INGENIOUS MECHANISMS

FOR DESIGNERS AND INVENTORS

VOLUME I

Mechanisms and Mechanical Movements Selected from Automatic Machines and Various Other Forms of Mechanical Apparatus as Outstanding Examples of Ingenious Design Embodying Ideas or Principles Applicable in Designing Machines or Devices Requiring Automatic Features or Mechanical Control

Edited by

FRANKLIN D. JONES

INDUSTRIAL PRESS INC.

200 MADISON AVENUE, NEW YORK 10016

Industrial Press Inc.
200 Madison Avenue
New York, New York 10016-4078

INGENIOUS MECHANISMS
FOR DESIGNERS AND INVENTORS—VOLUME I

37 39 41 43 45 44 42 40 38 36

PREFACE

WHEN the designer or inventor begins to originate or develop some form of automatic machine or other mechanical device, he is confronted by two important problems: The first one is purely mechanical and relates to the design of a mechanism that will *function* properly. The second problem is a commercial one and pertains to designing with reference to the cost of manufacture.

In order to solve the mechanical part of the problem, especially when an intricate motion or automatic control is required, a wide knowledge of the principles underlying those mechanical movements which have proved to be successful, is very helpful, even to the designer who has had extensive experience. The purpose of this treatise is to place before inventors and designers concise, illustrated descriptions of many of the most ingenious mechanical movements ever devised. These mechanisms have been selected not only because they are regarded as particularly ingenious, but also because they have stood the test of actual practice. Many of these mechanisms embody principles which can be applied to various classes of mechanisms, and a study of such mechanical movements is particularly important to the designer and student of designing practice owing to the increasing use of automatic machines in almost every branch of manufacture.

The second problem mentioned, that of cost, is directly related to the design itself which should be reduced to the simplest form consistent with successful operation. Many mechanical movements *are* ingenious because they are simple in design. Simplified designs usually are not only less costly but more durable. Almost any action or result can be obtained mechanically if there are no restrictions as to the number of parts used and as to manufacturing cost, but it is evident that a design should pass the commercial as well as the purely mechanical test. In this connection it is advisable for the designer to study carefully mechanical movements which actually have been applied to commercial machines. Practically all of the mechanisms shown in this treatise have been utilized on automatic machines of various classes.

CONTENTS

CHAPTER I

CAMS AND THEIR APPLICATIONS

A STUDY of the various mechanical movements and automatic regulating devices used on automatic and semi-automatic machines of different types, will show that mechanical movements based on the same general principles are often applied to machines which differ widely as to type and purpose. For instance, a mechanism for obtaining an intermittent motion may possibly be utilized in connection with almost any mechanical device requiring such motion, after certain changes have been made. Frequently these necessary changes will alter the form and perhaps the entire arrangement without changing the underlying principle governing the operation. This explains why designers of automatic machinery find that a knowledge of mechanical movements of all kinds is valuable, because an understanding of one design often suggests an entirely different application.

Many of the most ingenious mechanical movements and regulating devices ever devised will be found in this treatise. Some of these with more or less modification have been used so generally on different classes of machinery that they may be considered standard, whereas many others are more special and are not so generally known. All of these mechanisms, however, are believed to embody some mechanical principle that is likely to be useful to designers and inventors. These various mechanisms have been grouped in chapters according to the general types or classes to which they belong, partly to show different modifications of a given type and also to assist users of this book in finding a mechanical movement suitable for a particular application.

This first chapter deals with cams because they are widely used in the design of automatic machines of practically every type. In fact, by the use of some form of cam it is possible to obtain practically an endless variety of movements and irregular motions, many of which could not possibly be derived by other mechanical means. Even though some other type of mechanism might be substituted, the cam provides the simplest and cheapest method of obtaining most of the special unusual actions required in automatic machine design.

General Classes of Cams. — The name "cam" is applied to various forms of revolving, oscillating, or sliding machine members which have edges or grooves so shaped as to impart to a follower a motion which is usually variable and, in many cases, quite complex. Cams are generally used to obtain a motion which could not be derived from any other form of mechanism. Most cams revolve and the follower or driven member may have either a rectilinear or oscillating motion. The acting surface of the cam is in direct contact either with the follower or with a roller attached to the follower to reduce friction. The exact movement derived from any cam depends upon the shape of its operating groove or edge which may be designed according to the motion required.

Cams may be classified according to the relative movements of the cam and follower and also according to the motion of the follower itself. In one general class may be included those cams which move or revolve either in the same plane as the follower or a parallel plane, and in a second general class, those cams which cause the follower to move in a different plane which ordinarily is perpendicular to the plane of the motion of the cam. The follower of a cam belonging to either class may either move in a straight line or receive a swinging motion about a shaft or bearing. The follower may also have either a uniform motion or a uniformly accelerated motion. The working edge or groove of a uniform motion cam is so shaped that the follower moves at the same

velocity from the beginning to the end of the stroke. Such cams are only adapted to comparatively slow speeds, owing to the shock resulting from the sudden movement of the follower at the beginning of the stroke and the abrupt way in which the motion is stopped at the end of the stroke. If the cam is to rotate quite rapidly, the speed of the follower should be slow at first and be accelerated at a uniform rate until the maximum speed is attained, after which the motion of the follower should be uniformly decreased until motion ceases, or a reversal takes place; such cams are known as "uniformly accelerated motion cams."

Plate Cam. — Several different forms of cams are shown in Fig. 1. The form illustrated at *A* is commonly called a "plate cam," because the body of the cam is in the form of a narrow plate, the edge of which is shaped to give the required motion to the follower. This follower may be mounted in suitable guides and have a reciprocating motion (as indicated in the illustration) or it may be in the form of an arm or lever which oscillates as the cam revolves. When the follower is in a vertical position as shown, it may be held in contact with the cam either by the action of gravity alone or a spring may be used to increase the contact pressure, especially if there are rather abrupt changes in the profile of the cam and the speed is comparatively fast.

Positive Motion Cam. — The cam illustrated by diagram *B*, Fig. 1, is similar to the type just described, except that the roller of the follower engages a groove instead of merely resting against the periphery. Cams of this general form are known as "face cams" and their distinctive feature is that the follower is given a positive motion in both directions, instead of relying upon a spring or the action of gravity to return the follower. The follower, in this particular case, is in the form of a bellcrank lever and is given an oscillating motion. One of the defects of the face cam is that the outer edge of the cam groove tends to rotate the roller in one direction and the inner edge tends to rotate it in the opposite direction. A certain amount of clearance must be provided in the groove and, as the roll changes its contact from the inner edge to the outer edge, there is an instantaneous reversal of rotation which is resisted, due to the inertia of

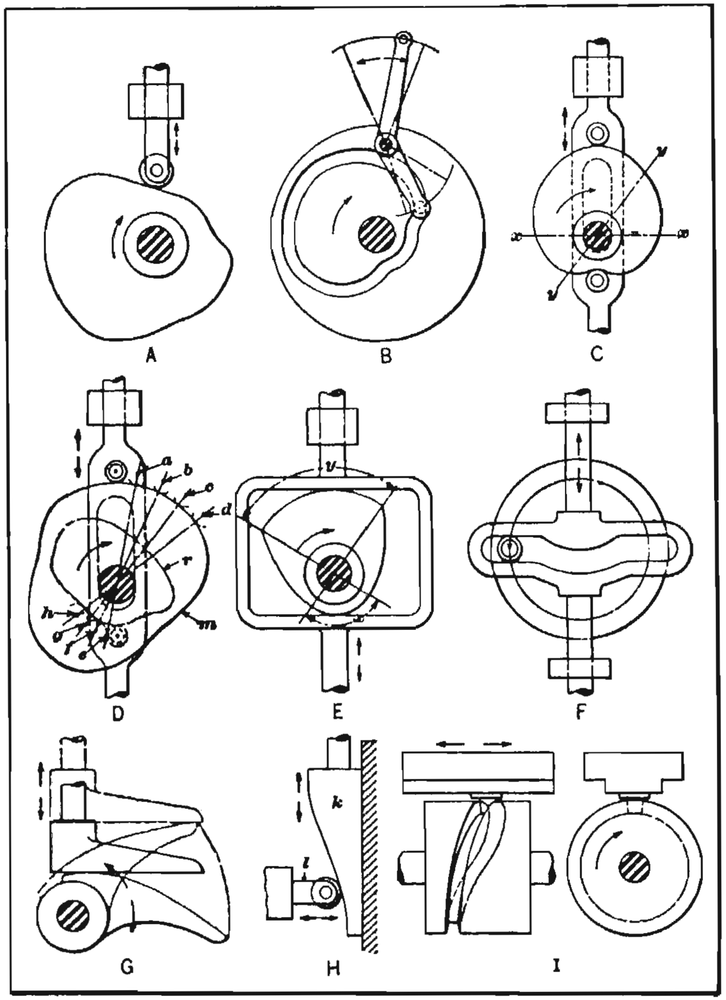


Fig. 1. Different Types of Cams

the rapidly revolving roll; the resulting friction tends to wear both the cam and the roll. This wearing action, however, may not be serious when the cam rotates at a slow speed. If

the speed is high, there is also more or less shock each time the follower is reversed, owing to the clearance between the roller and the cam groove.

Plate Cam Arranged for Positive Motion. — In order to avoid the defects referred to in connection with the face cam, the follower of a plate cam is sometimes equipped with two rollers which operate on opposite sides of the cam, as shown at *C*, Fig. 1. With such an arrangement, the curve of the cam for moving the follower in one direction must be complementary to the curve of the remaining half of the cam, since the distance between the rollers remains constant. In other words, this cam may be designed to give any motion throughout 180 degrees of its movement, but the curvature of the remaining half of the cam must be a uniform distance from that of the first, at all points diametrically opposite. Then the distances measured along any center line, as at *xx* or *yy*, are constant and equal the distance between the follower rollers. For this reason, the term *constant diameter cam* is sometimes applied to this class which is adapted for heavier work than the grooved face cam illustrated at *B*. The follower or driven member is slotted to receive the camshaft, and this slot acts as a guide and keeps the rollers in alignment with the center of the cam.

Return Cam for Follower. — When the curvature of one half of a cam is not complementary to the curvature of the other half, a special return cam is necessary, if the follower is equipped with two rollers in order to secure a positive drive. A main and return cam is illustrated at *D*, Fig. 1. The main cam may be laid out to give any required motion for a complete revolution of 360 degrees, and the return cam has a curvature which corresponds to the motion of the return roller on the follower. After the main cam is laid out to give whatever motion is required, points as at *a*, *b*, *c*, *d*, etc., are located on the path followed by the center of the roller, and, with these points as centers, the points *e*, *f*, *g*, and *h* are located diametrically opposite, and at a distance equal to the center-to-center distance between the rollers. These latter points lie in the path followed by the center of the return roller, and by striking arcs from them having a radius equal to the roller radius, the curvature or working surface of the return cam may be laid out. One method of arranging these two cams is to place the follower between them and attach the rollers on opposite sides of the followers. The camshaft, in some cases, carries a square block which is fitted to the elongated slot in the follower to serve as a guide and a bearing surface.

Yoke Type of Follower. — Another form of positive motion cam is shown at *E*, Fig. 1. In this case, the follower has a surface which is straight or tangential to the curvature of the cam. With a follower of this kind, there is a limitation to the motion which can be imparted to it, because, when the contact surface is flat or plane, it is evident that no part of the cam can be concave since a concave surface could not become tangent to the straight face of the follower, and even though the follower is curved or convex any concave part of the cam must have a radius which is at least as great as the radius of any part of the follower. The type of cam shown at *E*, like the one illustrated at *C*, can only be laid out for a motion representing 180 degrees of cam rotation; the curvature of the remaining half of the cam must be complementary to the first half or correspond to it. The follower of the cam shown at *E* has a dwell or period of rest at each end of its stroke, the parts x and y being concentric with the axis of the camshaft. This general type of cam has been used for operating light mechanisms and also to actuate the valves of engines in stern-wheel river steamers.

Inverse Cams. — On all of the cams previously referred to, the curved surface for controlling the motion has been on the driving member. With a cam of the inverse type, such as is shown at *F* (Fig. 1) the cam groove is in the follower and the roller which engages this groove is attached to the driving member. The motion of this cam can be laid out for only 180 degrees of movement. The inverse type of cam is used chiefly on light mechanisms, the particular cam illustrated at

F being designed to operate a reciprocating bar or slide. The curved part of the slot in the follower has the same radius as the path of the driving roller, and serves to arrest the motion of the slide momentarily. The well-known Scotch yoke or slotted cross-head is similar to an inverse cam having a straight slot that is perpendicular to the center line of the follower. (The motion obtained with the Scotch yoke and its practical application is referred to in Chapter IX.)

Wiper and Involute Cams. — The form of cam shown at *G*, Fig. 1, is simply a lever which has a curved surface and operates with an oscillating movement through an arc great enough to give the required lift to the follower. A cam of this kind is called a "lifting toe" or a "wiper" cam, and has been employed on river and harbor steamboats for operating the engine valves. Many involute cams are somewhat similar in form to the type illustrated at *G*, and they are so named because the cam curve is of involute form. Such cams are used on the ore crushers in stamp mills. Several cams are placed on one shaft and as they revolve the rods carrying the stamps are raised throughout part of the cam revolution. Disengagement of the cam and follower then causes the latter to drop.

Cams having Rectilinear Motion. — Some cams instead of rotating are simply given a rectilinear or straight-line motion. The principle upon which such cams operate is shown by diagram *H*, Fig. 1. The cam or block k is given a reciprocating motion in some form of guide, and one edge is shaped so as to impart the required motion to the follower l. An automatic screw machine of the multiple-spindle type is equipped with a cam of this general type for operating side-working tools, the tool-slide receiving its motion from the cam which, in turn, is actuated by the turret-slide. This type of cam is also applied to an automatic lathe for operating the radial arm or tool-holder.

Cams for Motion Perpendicular to Plane of Cam. — The cams previously referred to all impart motion to a follower which moves in a plane which either coincides with or is parallel to the plane of the motion of the cam. The second general class of cams previously referred to, which cause the follower to move in a plane usually perpendicular to the plane of the motion of the cam, is illustrated by the design shown at *I*, Fig. 1. This form is known as a "cylinder" or "barrel" cam. There are two general methods of making cams of this type. In one case, a continuous groove of the required shape is milled in the cam body, as shown in the illustration, and this groove is engaged by a roller attached to the follower. Another very common method of constructing cylinder cams, especially for use on automatic screw machines, is to attach plates to the body of the cam, which have edges shaped to impart the required motion to the follower. When a groove is formed in the cam body, it should have tapering sides and be engaged by a tapering roller, rather than by one of cylindrical shape, in order to reduce the friction and wear.

Automatic Variation of Cam Motion. — Ordinarily the motion derived from a cam is always the same, the cam being designed and constructed especially for a given movement. It is possible, however, to vary the motion, and this may be done by changing the relative positions of the driving and driven members by some auxiliary device. This variation may be in the extent or magnitude of the movement or a change in the kind of motion derived from the cam. The cam mechanism shown at *A* in Fig. 2 is so arranged that every other movement of each of the two followers is varied. The bellcrank levers *a* and *b*, which are the followers, have cam surfaces on the lower ends, and they are given a swinging motion by rolls *d* and *e* pivoted to arm *c* which revolves with the shaft *h* seen in the center of the arm.

The requirements are that each lever have first a uniform motion and then a variable motion; it is also necessary to have a change in the variable stroke until twelve strokes have been completed, when the cycle of variable motions is repeated. For instance, every other vibration of each lever is through a certain angle, and for twelve alternate vibrations the stroke is changed from a maximum to a minimum, and *vice versa*,

the angle of the uniform vibration being the mean or average movement for the variable strokes. The uniform vibration is obtained when roll *d* engages the cam surface on either lever *a* or *b*, and the variable movement is derived from roll *e* on the opposite end. This roll is mounted eccentrically on bushing *f* which is rotated in its seat by star-wheel *g*, one-twelfth revolution for each revolution of arm *c*; consequently, the roll is moved either toward or away from the axis of shaft *h*, thus varying the angle of vibration accordingly.

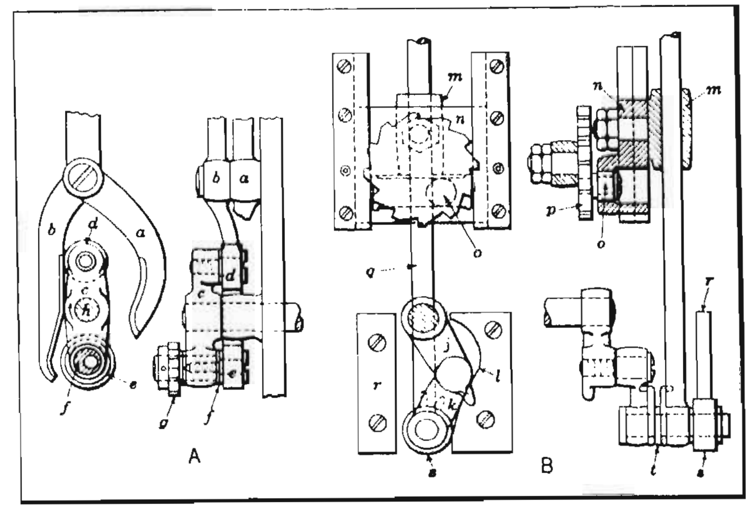


Fig. 2. Mechanisms for Varying Motion Normally Derived from Cams

Another mechanism which serves to vary the motion derived from a cam surface is shown at *B* in Fig. 2. This mechanism is used in conjunction with one previously described. A motion represented by the curvature *l* of a plate cam is reproduced by the upper end of the rod or lever *q*. One movement of the rod end is an exact duplicate of the cam curvature, and this movement represents the mean of a cycle of twelve movements, each of which is a reproduction of the curvature on an increasing or diminishing scale from maximum to minimum, or *vice versa*. The lever returns to the starting position with a rectilinear or straight-line motion.

The lever is given a reciprocating movement by crank *j* and connecting link *k*. The roll *s* at the lower end of the lever is kept in contact with cam surface *l* by spring *t*. The lever *q* is fulcrumed and slides in the oscillating bearing *m* which is supported by the slotted cross-head *n*. This cross-head is operated by roll *o* which is carried by a crankpin on a twelve-tooth ratchet wheel *p*. When the mechanism is in action, the crank *j* throws connecting link *k* out of line with lever *q* and the resulting tension on spring *t* causes roll *s* to follow the outline or curvature *l* of the cam until the upper end of the

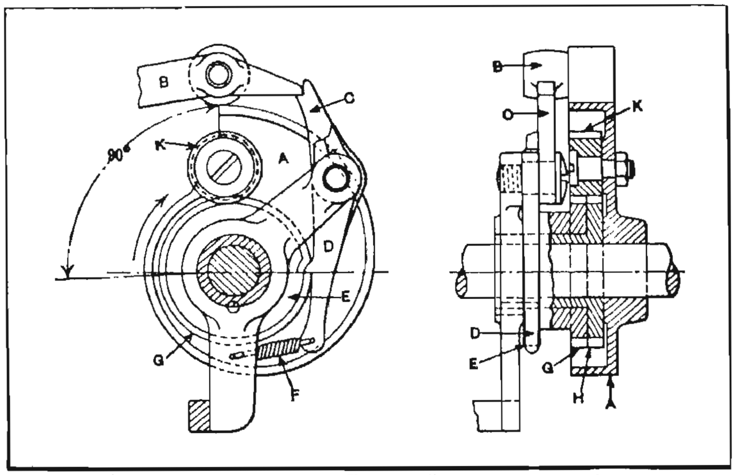


Fig. 3. Arrangement for Varying Dwell of Cam Follower

travel is reached; then the connecting link *k* is thrown out of line with lever *q* in the opposite direction, which causes spring *t* to force roll *s* against the straight return guide *r*. For each revolution of the crank, a pawl turns the ratchet wheel *p* one tooth, so that the slotted cross-head *n* and the bearing *m* are gradually raised and then lowered. As the result of this upward and downward movement of bearing *m*, which is the fulcrum for lever *q*, the motion is increased and then diminished the desired amount.

Varying Dwell of Cam Follower. — The mechanism illustrated in Fig. 3 is for varying the dwell of a cam follower or the length of time it remains stationary. The cam *A* lifts

lever *B* during three-fourths of a revolution, and during the dwell the follower *B* is held up by the latch *C*. This latch is controlled by pawl *D*, cam *E*, and spring *F*. The cam *E* has ratchet-shaped notches in its edge and is made integral or in one piece with a twenty-four-tooth gear *G*. The ratchet and gear are revolved upon the hub of a twenty-five-tooth stationary gear *H*, by the planetary pinion *K*, once for every twenty-four revolutions of cam *A*. With this particular mechanism, the lever *B* is given a dwell of 90 degrees for the first revolution; thereafter the dwell increases 360 degrees after each

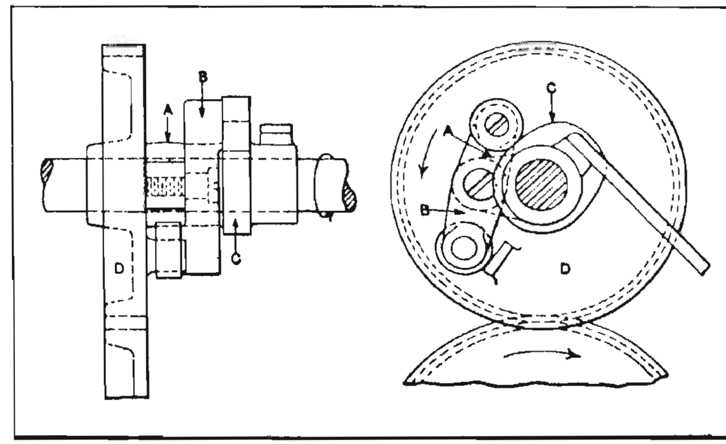


Fig. 4. Application of Cam for Varying Rotary Motion

rise of the follower, until the fourth period (which gives 1530 degrees dwell) when the dwell decreases until it is again 90 degrees; that is, during the fourth period the rise occurs while the cam makes three-fourths revolution, and then there is a dwell equivalent to 4¼ revolutions. Twenty-four revolutions are required to complete a cycle of movements. When milling the teeth in cam *E*, the index-head was arranged for twenty-four divisions, but teeth were cut only at the following divisions: 1-2-4-7-11-16-20-23. When the mechanism is in use, latch *C* is disengaged whenever pawl *D* enters a notch in cam *E*, thus allowing lever *B* to drop suddenly.

Variable Rotary Motion derived from Cam. — An unusual application of a cam is illustrated in Fig 4. In this case, a cam is used to impart a variable angular velocity to a gear which makes the same number of revolutions as its driving shaft. The driving shaft carries a casting A to which is fulcrumed the lever B which, in turn, has a roll on each end. One roll engages a cam C which is supported upon the shaft but does not revolve with it. The other roll bears upon a lug on the side of gear D which is also free upon the shaft, but is constrained to revolve with it either faster or slower, accord-

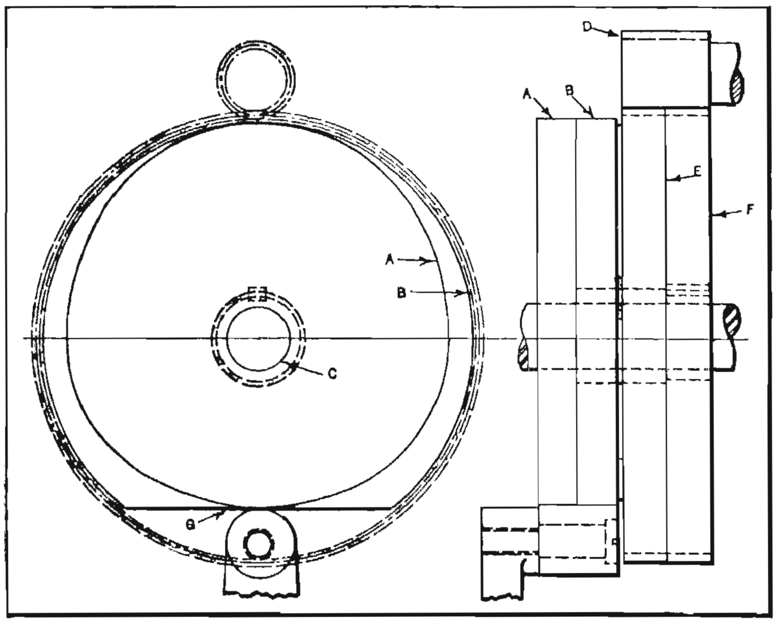


Fig. 5. Two-part Cam which Alternately Increases and Decreases Stroke of Follower

ing to the relative positions of lever B and cam C.

Cam which Alternately Increases and Decreases Motion of Follower. — An automatic paper-tube rolling machine has a driven member that must dwell during three-fourths of a revolution of shaft C (Fig. 5) and then be given a stroke that varies gradually in length for successive cam revolutions. This variation is obtained by using a cam having two sections A and B. These two sections are both driven by pinion D through gears E and F. Gear E is integral with cam B and

has 105 teeth, whereas gear F is keyed to the hub of cam A and has 104 teeth. Both gears have the same outside diameter, and the difference in tooth numbers provides a differential movement between the cams, so that one cam is continually changing its position relative to the other.

The dwell is obtained when the roll of the follower is in contact with the concentric part of cam B. When cam A is in the position shown, the maximum stroke occurs as the follower traverses across the flat edge G of cam B. The stroke of the follower is gradually reduced as A turns relative to B, thus filling the segment-shaped space at G, so that finally the cam is nearly concentric all around. The motion of the follower is somewhat irregular, but for this particular application, the irregularity is immaterial, as the essential requirement is to have the follower, after the 364th revolution of the pinion, at a distance from the center of shaft C equal to the dwelling position.

Automatic Variation of Cam Rise and Drop According to Pressure Changes. — The special design of cam illustrated in Fig. 6 normally has a 120-degree rise, a 60-degree dwell, a 90-degree drop, and a 90-degree dwell. In the operation of the machine to which this cam was applied, however, it was necessary to vary the motion derived from the cam in accordance with the pressure exerted upon a certain part of the machine; for instance, if the pressure exceeds a given limit during a dwell, the rise must take place in 90 degrees instead of 120 degrees; whereas, if the pressure decreases below the desired amount, the drop must be lengthened to 120 degrees. The mechanism for automatically varying the cam motion is comparatively simple, as the illustration indicates.

The main cam A carries two auxiliary cams B and C. These cams are driven by pins, which pass through them as shown by the sectional view, and they are free to slide upon these pins and the shaft, parallel to the axis of the shaft. Cam B carries a roller K and cam C, a roller L. Adjacent to these movable cams, there is a disk D having two sets of ratchet teeth and two side cams M and N. (The end view of this disk is shown at the lower part of the illustration.) A pawl *F* rests upon the block *G* until the increase or decrease of pressure interferes with the balance of the spring shown and causes pawl *F* to drop into engagement with a ratchet tooth. As soon as this engagement occurs, disk *D* stops rotating and cams *M* and *N* come into engagement with rollers *K* and *L* and force cams *B* and *C* over toward cam *A*, so that they engage the wide cam-roller on the follower, and give it the re-

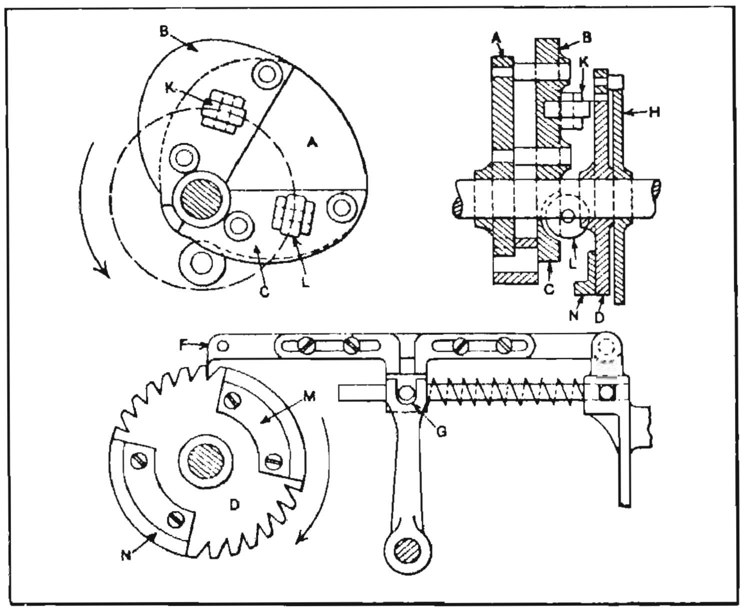


Fig. 6. Cam equipped with Mechanism for Varying Rise and Drop According to Predetermined Pressure on Another Part of the Machine

quired variation of movement. The cam *H* returns pawl *F* to the neutral position.

Sectional Interchangeable Cams for Varying Motion. — A flexible cam system was required that made it possible to vary the motion relative to the complete cycle of movements by substituting one interchangeable cam section for another, instead of using a large single cam for each variation. Two distinct methods of obtaining practically identical results were successfully evolved. One mechanism was a rotary type and

the other involved the use of rectilinear motion for the cam sections. Both mechanisms might properly be called "magazine" cams, because the cam sections are continually placed in action and then replaced by others in successive order.

The rotary design is illustrated in Fig. 7. The cam sections shown at *A* are semi-circular. The continuity of the cam surface is obtained by making each semi-circular section in the form of a half turn of a spiral with close-fitting joints, the complete cam appearing like a worm. The sections are fed longitudinally along the shaft and successively under the

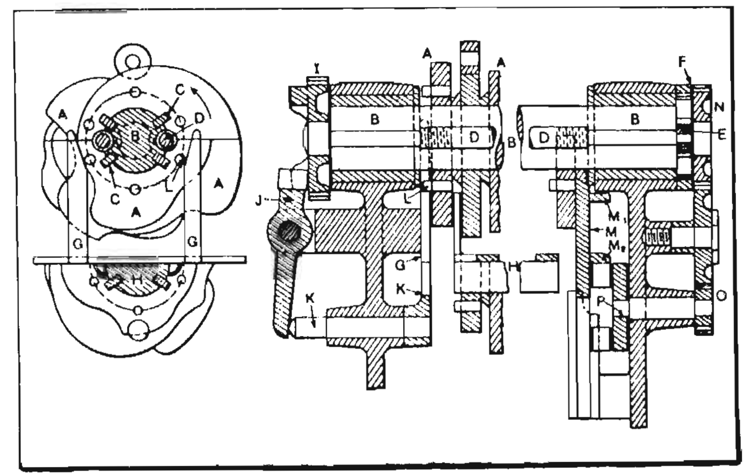


Fig. 7. Cam Mechanism Provided with Interchangeable Sections for Varying Motion of Follower

lever roller at a rate of advance equaling the lead of the spiral. Four feathers *C* are provided to guide and retain the cams. The two screws *D* producing the longitudinal movement are driven by pinions *E* meshing with an internal gear *F*, which is fastened to the bearing. As the feathers extend only to within the width of one cam from the left bearing, two sections drop from the shaft at every revolution, the dropping sections being guided by the guides *G*. The double cam upon the driving gear *I*, the lever *J*, and the carrier-slide *K* provide the means for hanging the semi-circular cams upon the magazine bar *H*. The slide *K* catches each piece by the pins L and, by pushing one, causes the further one to slide onto the lifting slide M which engages its grooved hub. The gears N and O, in the ratio of 1 to 2, and disk P operate a slide for returning the cams to their shaft. The rollers on P successively engage the steps M_1 and M_2, thus raising the slide which drops back automatically.

To facilitate engagement between the cam threads and the screws, the square threads of the latter are V-shaped at the entering ends, and, to insure locking the cams to the shaft quickly, the ends of the feathers recede into pockets and fly out by the action of springs. Any part of the system may be

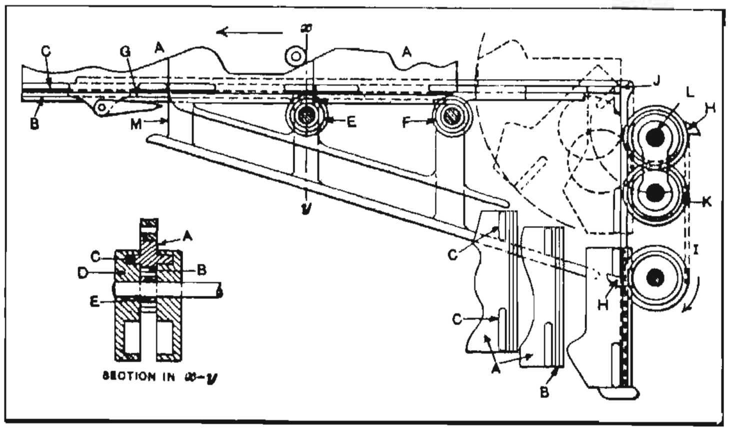


Fig. 8. Interchangeable Cam Sections which have a Rectilinear Motion

changed by placing the desired section in a holder and introducing it between the slide K and the magazine bar. The cam to be removed—the dropping cam—comes out upon an inclined runway of the holder.

The alternate design is the rectilinear cam system shown in Fig. 8. The mechanism consists of the cam sections A, provided with rack teeth at B. (See also detail sectional view.) Each section has four lugs C which act as guides in the ways D. A pinion E feeds the sections along beneath the lever roller, and the frictionally driven pinion F assembles them. When any section has passed beneath the roller, it is automatically hung upon the magazine chute. The for-

ward lugs C are made slightly longer than the rear ones, to span the gap G; but the rear lugs enter the gap just as the forward lugs clear the ways. The sections are taken from the lower part of the ways in the magazine by spring-controlled forks H upon the chain I which engage the lugs and lift the cams until the smaller lugs strike at the corner J. The linked gear K meanwhile engages the rack, and as it swings about the center L, it lifts the cam up against the ways; here the resistance offered to further motion of the links causes K to rotate about its own center and slide the cam into place.

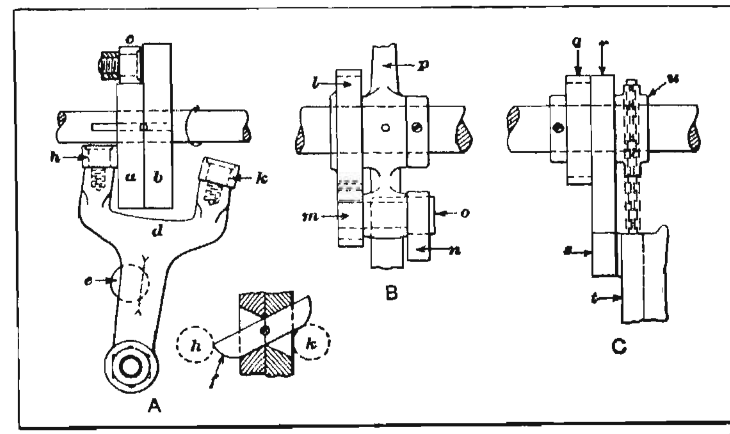


Fig. 9. (A) Double-shifting Cam; (B) Lever Vibrated from Shaft on which it is Fulcrumed; (C) Shaft Oscillated by Cam Located on it

Substitute sections are introduced at M, and the replaced sections are lifted from the ways.

Double Two-revolution Cam of Shifting Type. — The cam mechanism illustrated at A in Fig. 9 is so arranged that two revolutions of a double cam are necessary in order to give the required motion to a follower. One revolution is required for the rise or upward movement of the follower and a second revolution for the "dwell," during which the follower remains stationary. The cam sections a and b are fastened together and are free to slide upon their shafts a distance equal to the face width of one section. The two cam sections are driven by means of a spline. Roll c is attached to the follower or driven member and, in the illustration, is shown in contact with the spiral cam *a*, from which the upward movement is derived. The cam *b* is simply a circular disk mounted concentric with the shaft. The lever *d* for shifting the double cam is operated by a "load-and-fire" mechanism having a spring plunger at *e*. (The load-and-fire principle is explained in Chapter VI on "Reversing Mechanisms.")

When the mechanism is in operation, cam *a* lifts roll *c* to its highest position, when lever *d* shifts the double cam along the shaft, leaving roll *c* upon cam *b*, where it remains during a dwell of one revolution; the cam is then immediately shifted in the opposite direction, thus allowing roller *c* and the driven member to drop instantaneously upon cam section *a*. The movement of shifting lever *d* is derived from the double-ended lever *f* (see detailed view) which extends through a slot in the cams. This lever is pivoted at the center and is free to swing in one direction or the other, until it rests against the sides of the opening. With the double cam in the position shown in the illustration, end *f* engages roll *h* and forces it to the left until spring plunger *e* comes into action and suddenly throws the lever over the full distance. The opposite end of lever *f* swings far enough to clear roll *k* before this roll is thrown over.

Lever Vibrated from Shaft on which it is Fulcrumed.— A cam which is used for vibrating a lever twice for each revolution of a shaft on which it is fulcrumed is illustrated at *B* in Fig. 9. A gear *l* attached to the shaft drives a pinion *m* which is one-half the size of the gear. This pinion revolves cam *n*, and the shaft for the pinion and cam has a bearing in the end of lever *p*. The cam revolves in contact with a stationary roll *o* which causes the lever to vibrate about the shaft as a center twice for every revolution.

Shaft Oscillated by Cam located on it. — Fig 9 shows, at *C*, how a shaft can be given an oscillating or rocking movement by a cam which is mounted on the shaft. The cam *r* is attached to gear *q* which is driven from an outside source. As the cam revolves in contact with roll *s*, a reciprocating motion

is imparted to slide *t*. A chain attached to this slide passes over a sprocket *u* which is fast to the shaft. The other end of the chain is fastened to a tension spring beneath the slide, which serves to hold the roll *s* into engagement with the cam.

Double-track Cam. — A cam that provides the required motion and "dwells" for a slide on a special flat-wire form-

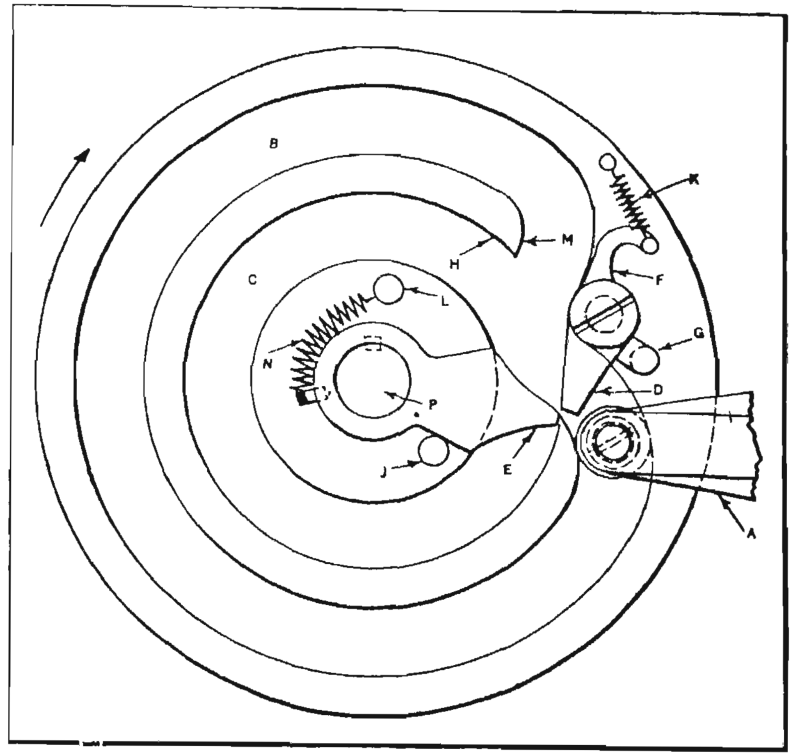


Fig. 10. Cam having Two Concentric Grooves which are Engaged Alternately by a Roller on the Driven Slide

ing machine (See Fig. 10) is so designed that the follower *A* has a dwell at each end of its stroke. The cam has two concentric grooves *B* and *C*, and as it rotates, the roller on follower *A* is transferred alternately from one groove to the other by means of the switching levers *D* and *E*. The roll in the illustration is about to come into contact with lever *D*, which will swing around until the lug *F* engages pin *G*;

then edge D will be flush with edge H and follower A will pass into groove C. The path thus formed to guide the roll into groove C is positive, as lever D is against stop G and lever E is against J.

As soon as the roll passes the end of lever D, the latter snaps back to the position shown, through the action of spring K. The follower now dwells at one end of its stroke, as groove C is concentric. When the cam has revolved far enough to bring the roll into contact with lever E, the latter swings around until it strikes stop-pin L, and then the edge E is flush with end M. A path has now been formed which leads the roll into the outer groove B, after which lever E snaps back to the position shown, through the action of coil spring N. The cam rotation then continues until the roll is again in the position illustrated, when the cycle just described is repeated.

The follower is rigidly connected to a slide (not shown) which operates a mandrel for forming the stock. The cam receives its motion from shaft P. The required movement could have been obtained by the use of an ordinary cam, thus reducing the speed of shaft P one-half, but because of difficulties due to conflicting machine speeds, it was considered advisable to employ the special cam described.

Spiral Cam for Reciprocating Motion. — A positive spiral cam drive for imparting a reciprocating motion to a slide is shown in Fig. 11. The cam C, which has a spiral groove, revolves continuously in the direction indicated by the arrow, and transmits motion to slide D through engaging rollers A and B which are connected by rocker arm E, and are arranged to engage the cam alternately. If roller A is in the inner position or at the inner end of the cam groove as shown, it will be traversed to the outer end of the groove while the cam makes 1½ revolutions; as this roller approaches the outer end of the groove, it engages a cam insert F (see also detail sectional view) placed in the groove; consequently, roller A rides up the inclined surface of this cam insert, which causes rocker E to force the other roller B down into engagement

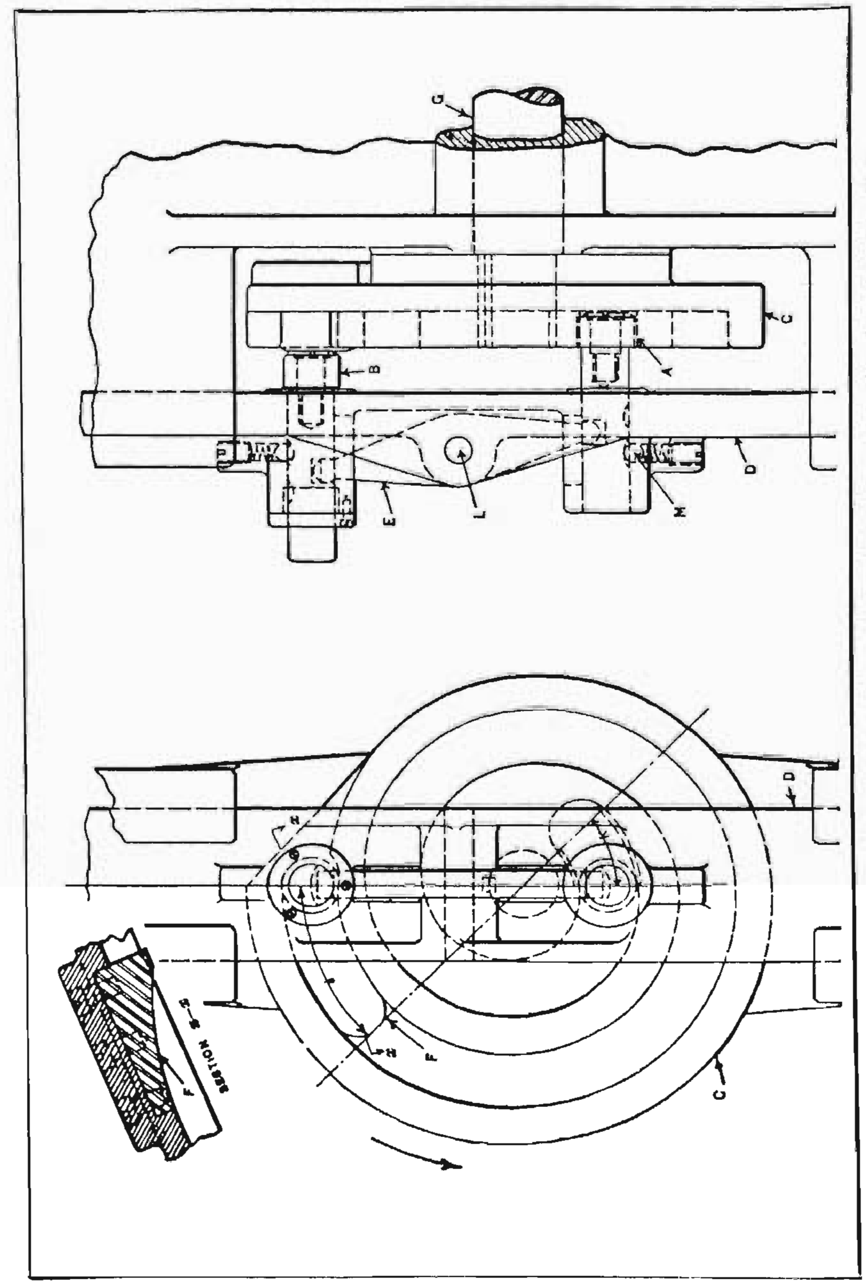


Fig. 11. Spiral Cam which is Engaged Alternately by Two Rollers on Rocker Arm of Driven Slide

with the inner part of the cam groove; then the return stroke of the slide begins as the cam continues to revolve, and when roller B has reached the outer position, thus completing one cycle, the action is reversed, roller A being again forced into engagement at the inner position of the cam groove. It will be seen then that three cam revolutions are required for the forward and return strokes of the slide, and the rollers successively traverse from the inner to the outer positions.

At the beginning and end of the spiral, the groove is milled concentric with driving shaft G (as indicated by the arrows

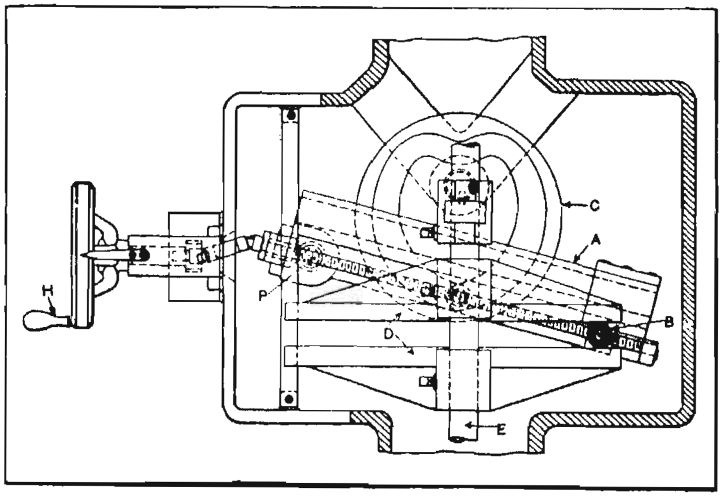


Fig. 12. Cam and Slotted Cross-head Combination with Adjustment for Varying Stroke

J and K) which provides a dwell equivalent to one-eighth revolution of the cam at each end of the stroke. The concentric sections J and K also permit the rolls to enter and leave the groove freely. The spiral groove advances uniformly so that a uniform motion is imparted to the driven slide. The rocker E, which swings on pin L, has rounded ends that engage grooves cut in the roller plungers. Pawl M which is backed by a spring, drops into either of two half round grooves in the plunger for locating it in the upper and lower positions. The other plunger has the same arrange-

ment. The cam insert F is of hardened tool steel and the rolls are beveled at the bottoms to correspond with the curve of the insert.

Cam-stroke Adjustment without Stopping Machine. — The mechanism shown in Fig. 12 is for traversing the table of a grinding machine along the bed. This machine, which is of a comparatively small size, is intended for internal and external grinding operations; thus it is necessary to provide means for readily changing the stroke of the table. With the mechanism illustrated, any variation in stroke can be obtained from zero to the maximum while the machine is operating. The motion for the table is derived from a heart-shaped cam C mounted on a vertical shaft which is driven through a speed-changing mechanism. This cam engages a roll attached to the lower side of an oscillating arm A having on its upper side another roll B which can be adjusted relative to the pivot P about which the arm oscillates. This upper roll operates between the parallel faces of yoke D, and the latter is attached to a rod E located beneath the table of the machine. On the under side of the table and extending throughout its entire length is a dovetailed slide-way in which is fitted a block that is attached to and moves with the reciprocating rod E. By means of a suitable lever, this block, which fits into the dovetailed slide-way, can be clamped in various positions for changing the location of the table. The action of the mechanism is as follows: When the cam C is rotating, arm A oscillates about pivot P and, through roller B, transmits a rectilinear motion to yoke D, rod E, and the table The length of this movement or stroke is governed by the position of roll B relative to pivot P, which may be varied by means of a screw that is connected through a universal joint with a shaft upon which handwheel H is mounted. When roll B is moved inward until it is directly over pivot P, no movement will be imparted to yoke D or the table.

Crank and Cam-lever Combination. — An interesting form of mechanism is illustrated in Fig. 13. This mechanism is used on moving picture cameras and also for feeding films through printing machines. It is commonly referred to as a "claw" mechanism or movement. The claw or hook *A* is double and engages evenly spaced perforations that are along each edge of the film. When this device is applied to a moving picture camera, the film is drawn, from a roll in the film box, down in front of the lens and then passes to a reel in the receiving box. The film remains stationary during each exposure and is drawn downward between successive exposures which are made at the rate of sixteen a second. The hook *A*, which engages the film and moves it along intermittently and with such rapidity, receives its motion from a crank and cam-lever combination. The two intermeshing gears *B* and *C*

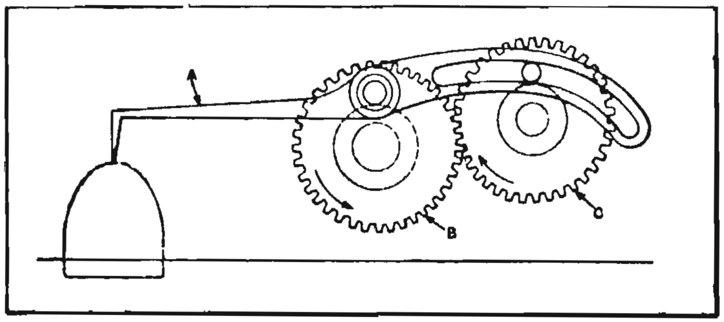


Fig. 13. Crank and Cam Combination for Operating Claw Mechanism of Moving Picture Camera

revolve in opposite directions. Gear *B* has a crankpin upon which the hook is pivoted. An extension of this hook has a curved cam slot that engages a pin on gear *C*. As the two gears revolve, the hook is given a movement corresponding approximately to the D-shaped path indicated by the dotted lines. While this mechanism is shown in a horizontal position in the illustration, it would normally be vertical with the hook uppermost, when in operation. Some of the other claw mechanisms in use differ from the one shown in regard to the arrangement of the operating crank and the cam or curved slot for modifying the crank motion. For instance, the cam, in some cases, is a separate part that is placed between the crank and the film hook, a pin on the hook lever engaging the

cam slot. Another type of claw mechanism derives both the downward motion for moving the film and the in and out movements of the film hook from separate cam surfaces.

Group of Cams engaged Successively. — The mechanism to be described was designed to engage with the driving shaft first one and then another of the cams in a group of five mounted upon the same shaft. It was necessary to have these cams operate their respective levers successively back and

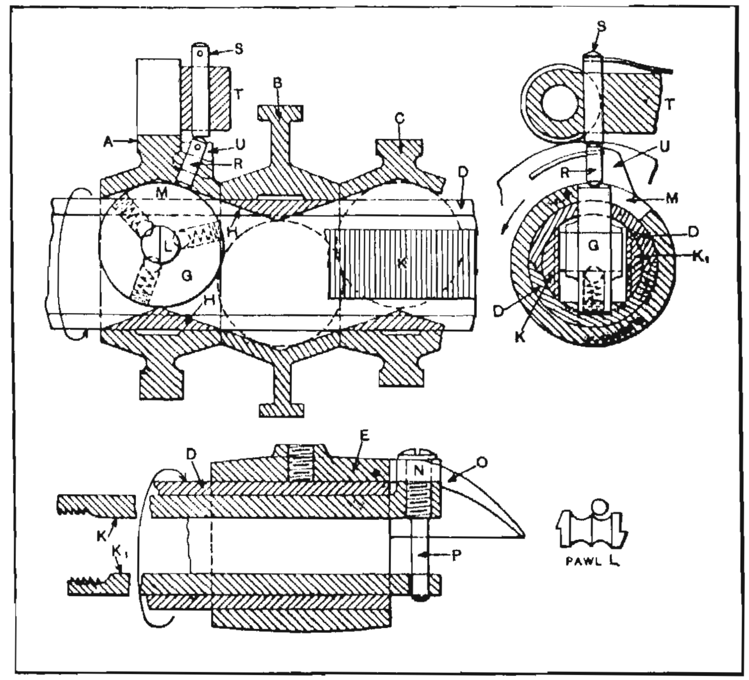


Fig. 14. Cams in a Group Engaged Successively

forth from one end of the group to the other, and while any one cam was in action the others must remain stationary with their lever rolls on a 90-degree dwell. Eight revolutions of the shaft were required to complete one cycle of movements. The device for controlling the action of these cams is shown in Fig. 14. The cams *A*, *B*, *C*, etc., are mounted upon a hollow shaft *D* carried in bearings *E*. The engagement of successive cams with the hollow shaft is effected by a roll-key *G* which is caused to move inside of the shaft from end to end. This motion of the roll-key is obtained from ratchets K and K_1. (See longitudinal section at lower part of illustration, which is taken at an agle of 90 degrees to upper section in order to show more clearly the construction.) As the roll-key is moved along, it follows the inclined surfaces H which bring it into engagement with the respective cam keyways, as at M. Within the roll-key there is a double-ended pawl L (see also detail view) which is held into engagement with either ratchet K or K_1 by balls and springs. The ratchets are cut oppositely and are given a reciprocating movement by cam O, roll N, and roll screw P which causes both ratchets to reciprocate together. A similar equipment on the opposite end of the ratchet makes the motion positive. When the roll-key has engaged the last cam in one direction, the return of the ratchet causes the pawl L to rise onto a higher surface, thereby throwing it into mesh with the other ratchet and effecting the reversal.

Obtaining Resultant Motion of Several Cams.—A driven member or follower is given a motion corresponding to the resultant motion of four other cam-operated followers by the mechanism to be described. These followers are in the form of levers, which are equally spaced and fulcrumed upon one bar. Four of the levers are operated independently by four positive-motion cams. The fifth lever, which is in the center of the group, receives the resultant motion of all the others; that is, the forces acting upon the other four levers are automatically resolved and their resultant in magnitude and direction is transmitted positively to the fifth lever. It is not necessary to show the cams or levers to illustrate the principle involved, but the ingenious apparatus by means of which the resultant motion is obtained is shown in horizontal section in Fig. 15. Each of the four levers is connected by a knuckle joint to one of the racks A, B, C, and D. These racks are free to slide up and down independently and are arranged in two pairs. One pair meshes with pinion E and the other pair with pinion F. As the arrangement of the

mechanism is symmetrical it will only be necessary to describe the action of one side. Any movements of the levers connecting with racks A and B will be transmitted to pinions E and G, which are mounted on one stud and rotate together. A stationary rack H and a sliding rack J engage pinion G. The sliding rack J carries a pinion K which, in turn, engages a stationary rack L and a sliding rack M. Pinion N is located on sliding bar P to which is attached the fifth lever previously referred to.

In order to illustrate the action of this mechanism, assume that rack A lifts one inch, rack B drops one-half inch, rack C is stationary, and rack D lifts one-quarter inch. The resultant is a three-quarter inch rise. In analyzing the motion, it should be remembered that a pinion moving along a stationary rack will cause a movable rack on the opposite side to travel with

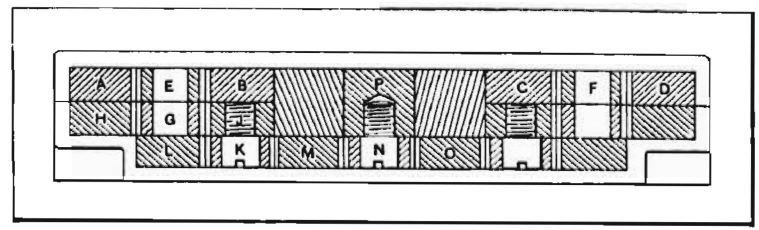


Fig. 15. Mechanism for Obtaining Resultant Motion of Several Cams

twice the pitch-line velocity of the pinion, which fact and its converse are here applied. The racks A and B acting upon pinion E will cause it to rise $\frac{1}{2} \times (1 - \frac{1}{2}) = \frac{1}{4}$ inch. This movement is doubled in the sliding rack J which, therefore, travels one-half inch, and it is again doubled in sliding rack M which as a movement of one inch. Rack M, in turn, moves pin N and the fifth lever slide P one-half inch. If the action of racks C and D is analyzed in a similar manner, it will be found that rack O has a movement of one-half inch, and rack N, one-quarter inch, which gives a total rise of the lever attached to slide P of three-fourths inch. To further illustrate the action, if all of the cam levers should drop one inch simultaneously, the result would be a drop of four inches for the middle lever attached to slide P.

CHAPTER II

INTERMITTENT MOTIONS FROM RATCHET GEARING

It is frequently necessary for machine parts to operate intermittently instead of continuously, and there are various forms of mechanisms for obtaining these intermittent motions. A tool-slide which is given a feeding movement at regular intervals is an example of a part requiring an intermittent movement. Automatic indexing mechanisms which serve to rotate some member, periodically, a definite part of a revolution, after the machine completes a cycle of operations, represent other applications of intermittent movements. The usual requirements of an intermittent motion, when automatic in its action, are that the motion be properly timed relative to the movement of parts operating continuously and that the member receiving the intermittent motion be traversed a predetermined amount each time it is moved. The movement may be uniform or it may vary periodically. When the machine part which is traversed intermittently must be located in a certain position with considerable accuracy, some auxiliary locating device may be utilized in conjunction with the mechanism from which the intermittent motion is obtained. For example, the spindle carriers of multiple-spindle automatic screw machines are so arranged that the carrier is first rotated to approximately the required position by an intermittent motion, and then it is accurately aligned with the cutting tools by some form of locating device.

Ratchet Gearing. — One of the simplest and most common methods of obtaining intermittent movements is by means of ratchet gearing. This type of gearing is arranged in various

ways, as indicated by the diagrams in Fig. 1. In its simplest form, it consists of a ratchet wheel *a* (see diagram *A*), a pawl *b*, and an arm or lever *c* to which the pawl is attached. The arm *c* swings about the center of the ratchet wheel, through a fractional part of a revolution, as indicated by the full and dotted lines which represent its extreme positions. When the movement is toward the left, the pawl engages the teeth

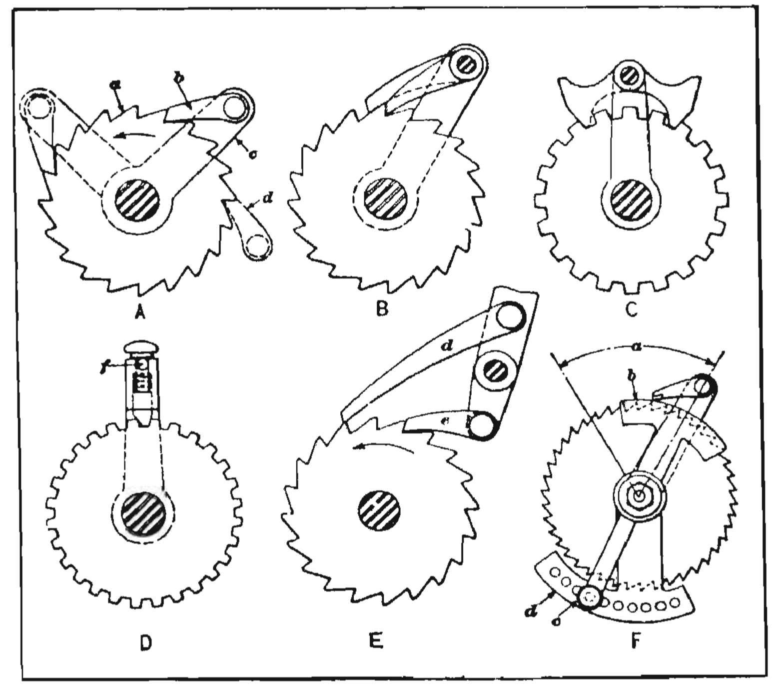


Fig. 1. Different Arrangements of Ratchet Gearing

of the ratchet wheel so that the latter turns with the arm. When the arm swings in the opposite direction, the pawl simply lifts and slides over the points of the teeth without transmitting motion to the ratchet wheel. If a load must be sustained by the ratchet gearing, a fixed pawl located at some point, as indicated at *d*, is used to prevent any backward rotation of the ratchet wheel.

With gearing of this general type, the faces of the ratchet teeth against which the end of the pawl bears should be so formed that the pawl will not tend to fly out of mesh when a load is applied. In order to prevent such disengagement, the teeth should be so inclined that a line at right angles to the face of the tooth in contact with the pawl will pass between the center of the ratchet wheel and the pivot of the pawl. If the face of this tooth should incline at such an angle that a line at right angles to it were above the pawl pivot, pressure against the end of the pawl would tend to force it upward out of engagement with the ratchet wheel.

Multiple Pawls for Ratchets. — When a single pawl is used as shown at *A*, Fig. 1, the arm which carries it must swing through an arc equal to at least one tooth of the ratchet wheel; hence the pitch of the teeth represents the minimum movement for the wheel. If two or more pawls are used, a relatively small motion of the arm will enable successive teeth to be engaged without decreasing the pitch of the ratchet wheel. The principle is illustrated by diagram *B* which shows two pawls in position instead of one. As will be seen, one pawl is longer than the other by an amount equal to one-half the pitch of the ratchet teeth. With this arrangement, the movement of the arm may equal only one-half the pitch, if desired, the effect being the same as though a single pawl were applied to a wheel having teeth reduced one-half in pitch. By using three pawls, each varying in length by an amount equal to one-third of the tooth pitch, a still finer feeding movement could be obtained without actually decreasing the pitch of the teeth and thus weakening them.

Reversal of Motion with Ratchet Gearing. — A simple method of obtaining a reversal of motion is illustrated by diagram *C*, Fig. 1. A double-ended pawl is used and, in order to reverse the motion of the ratchet wheel, this pawl is simply swung from one side of the arm to the other, as indicated by the full and dotted lines. Reversible ratchet wheels must have teeth with bearing faces for the pawl on each side.

Another method of obtaining a reversal of motion is shown at *D*. The pawl, in this case, is in the form of a small plunger which is backed up by a spiral spring. One side of the pawl

is beveled so that the pawl merely slides over the teeth on the backward movement of the arm. When a reversal of movement is required, the pawl is lifted and turned half way around, or until the small pin *f* drops into the cross-slot provided for it, thus reversing the position of the working face of the pawl.

Frictional Ratchet Mechanisms. — The types of ratchet gearing previously referred to all operate by a positive engagement of the pawl with the teeth of the ratchet wheel. Some ratchet mechanisms are constructed on a different principle in that motion is transmitted from the driving to the driven member by frictional contact. For instance, with one form, the driving member encircles the driven part which has cam surfaces that are engaged by rollers. When the outer driving member is revolved in one direction, the rollers move along the inclined cam surfaces until they are wedged tightly enough to lock the driven part and cause it to turn with the operating lever. When the driver is moved in the opposite direction, the backward motion of the rollers releases them. This general principle has been applied in various ways.

Double-action Ratchet Gearing. — It is sometimes desirable to impart a motion to the ratchet wheel during both the forward and backward motions of the ratchet arm or lever. This result may be obtained by using two pawls arranged as illustrated by diagram *E*, Fig. 1. These pawls are so located relative to the pivot of the arm that, while one pawl is advancing the ratchet wheel, the other is returning for engagement with the next successive tooth.

Variable Motion from Ratchet Gearing. — Ratchet gearing, especially when applied to machine tools for imparting feeding movements to tool-slides, must be so arranged that the feeding motion can be varied. A common method of obtaining such variations is by changing the swinging movement of the arm that carries the operating pawl. In many cases the link which operates the pawl arm receives its motion either from a crank or a vibrating lever, which is so arranged that the pivot for the rod can be adjusted relative to the center of rotation for changing the movement of the operating pawl and the rate of feed.

One method of adjusting the motion irrespective of the movement of the operating pawl is illustrated at *F* in Fig. 1. The pawl oscillates constantly through an arc *a*, and this angle represents the maximum movement for the ratchet wheel. When a reduction of motion is desired, the shield *b* is moved around so that the pawl is lifted out of engagement with the ratchet wheel and simply slides over it during part of the

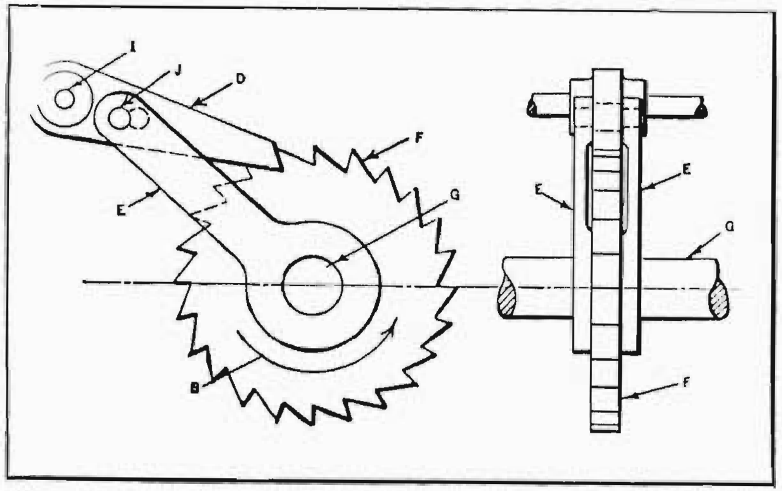


Fig. 2. Ratchet Mechanism to Prevent Reversal of Rotation and Arranged to Lift Pawl and Eliminate Noise when Ratchet Wheel is Rotating Clockwise

stroke. Thus, when the shield covers three of the teeth as shown in the illustration the motion of the ratchet wheel is reduced the same as though the swinging action of the pawl lever had been diminished an amount corresponding to three of the teeth. With the particular arrangement illustrated, the shield is held in any position by means of a small spring plunger *c* that engages holes in a stationary plate *d*.

Ratchets having Lifting Pawls to Prevent Noise. — Fig. 2 shows the construction of a ratchet mechanism that was designed for use on machines in which the noise of the pawl passing over the teeth of the ratchet is objectionable. Inci-

dentally, the continuous wear on the ratchet teeth and the end of the pawl is eliminated by the arrangement shown.

The ratchet wheel *F* revolves with the shaft *G*. The pawl *D* swings freely on the pivot *I*, which is held in the stationary part of the machine. The connecting links *E* are free on the shaft *G* and are held together at their upper ends by the rivet *J* which has a shoulder on both sides. This permits the links to be tightly fastened together and still be free to swing on

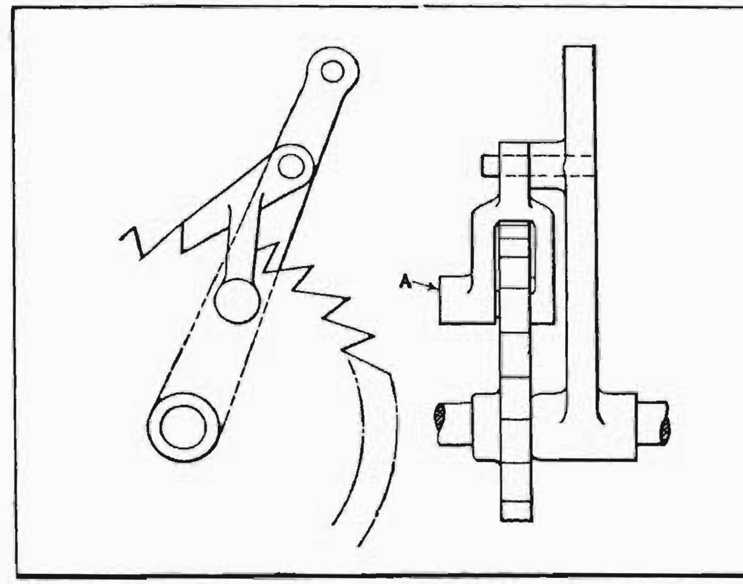


Fig. 3. Ratchet Mechanism with Silencing Device

the pawl *D*. The links *E* are sprung together, or toward each other at the lower end, so that they have a slight friction bearing on the sides of the ratchet wheel *F*. There is an elongated hole in pawl *D* through which rivet *J* passes.

The action of the mechanism is very simple but effective. When shaft *G* is turned clockwise, ratchet wheel *F* turns with the shaft. The friction on the sides of ratchet wheel *F* has a tendency to revolve links *E* with the wheel. The tendency to revolve, however, is prevented by rivet *J* which passes through pawl *D*. As rivet *J* shifts to the right-hand end of the slot in D this action results in lifting pawl D out of contact with wheel F and holding it out of contact as long as shaft G is turned in a clockwise direction. The height that pawl D is lifted above the ratchet wheel is controlled by the length of the slot through which rivet J passes. As soon as shaft G revolves in the opposite direction, as indicated by arrow B, links E tend to revolve with the ratchet, and this results in bringing pawl D downward into contact with the teeth of the ratchet wheel, as shown in the illustration.

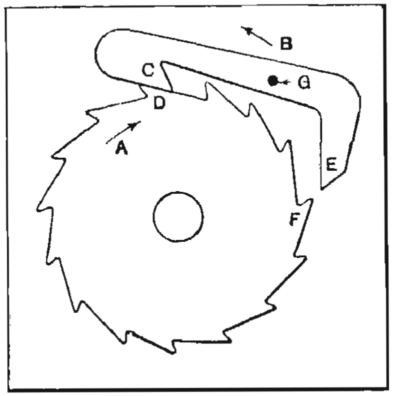


Fig. 4. Ratchet Mechanism having Double-ended Pawl

Another ratchet equipped with a silencing device is illustrated in Fig. 3. Boss A contains a spring plunger provided with a fiber tip. This plunger produces a slight friction on the ratchet sides and so causes the pawl to be lifted from the ratchet teeth on the idle stroke and kept from contact with the teeth until the working stroke. Many modifications of this principle are possible.

Silent Ratchet having Double-ended Pawl. — The ratchet mechanism shown in Fig. 4 has a double-ended pawl which operates silently. When the ratchet wheel turns in the direction indicated by arrow A, or when the pawl rotates in the direction indicated by arrow B, the end C of the pawl is raised by tooth D, thus bringing the end E into position to be engaged by tooth F. The engaging faces of the teeth are sloped so that the pawl will slide to the root and obtain a full contact. No spring is attached to the pawl.

When used as a feeding device, a frictional resistance, such as a friction washer placed on the fulcrum pin G, must be provided to eliminate rattle and insure the proper functioning of the pawl. When used simply to prevent the reversal of either member, no frictional resistance is necessary. In laying out a ratchet of this type it should be borne in mind that one of the pawls is just on the point of passing the tip of one of the teeth when the other pawl is midway between the tips of two teeth. It should also be noted that this type of ratchet, when used as a feeding mechanism, provides for feeding or indexing in multiples of one-half of a tooth space.

Silent Ratchet of Ball or Roller Type. — The design of ratchet shown in Fig. 5 should not be confused with the friction type. Power is transmitted by gripping the balls or

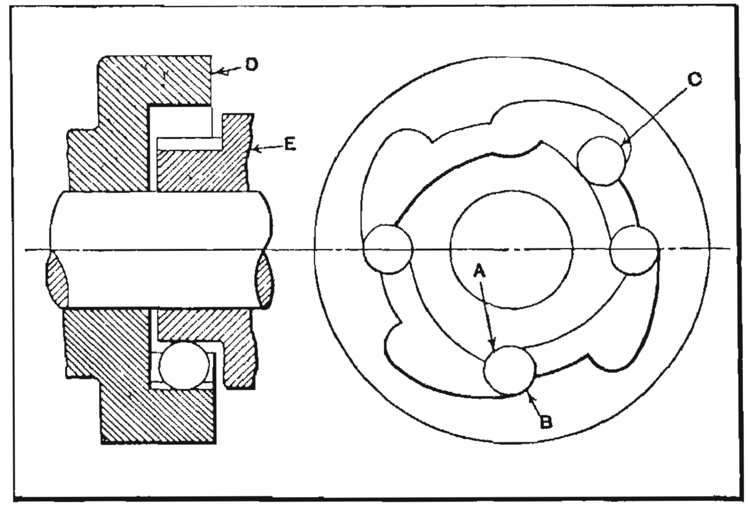


Fig. 5. Design that Transmits Power by Gripping Balls or Rollers between the Driving and Driven Members

rollers between surfaces A and B of the driving and driven members, not between cam surfaces. No springs are employed to bring the balls into place, gravity alone being relied on. Usually only three balls are in action; in the illustration it will be observed that ball C is not in engagement. Either member D or E may serve as the driver. When this mechanism is used in a drive where the movement need not be accurate, it is not necessary to machine the engaging surfaces, and iron castings serve well unless the strain is severe.

Ratchet Designed to "Dwell" Automatically. — When a feed-shaft or other driven member requires a "dwell" after every partial revolution, this may be obtained by a double ratchet mechanism arranged like the one shown in Fig. 6. This particular mechanism is designed to give a dwell equivalent to 3 teeth, or 3/16 revolution of the ratchet wheel, after every movement equal to 13 teeth, or 13/16 revolution.

Ratchet wheel *B* has the idle period or dwell, and ratchet wheel *A* carries a shield or guard *F* which prevents the pawl *E* of wheel *B* from operating during the dwell. Ratchet wheel *B* is keyed to shaft *D*, and the auxiliary ratchet wheel *A* is

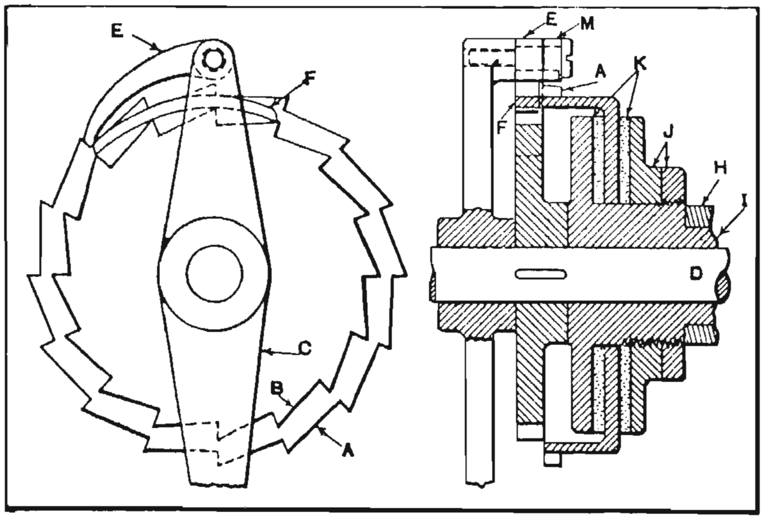


Fig. 6. Double Ratchet Having Shield which Prevents One Pawl from Engaging Wheel During Dwell

confined between two leather disks *K*, the pressure required being obtained from check-nuts *J*. Pawl *E* engages wheel *B*, as mentioned, and pawl *M* engages wheel A. These two pawls are pivoted to and operated by lever *C*, which gives them a movement that is slightly greater than three ratchet wheel teeth.

The function of the auxiliary ratchet *A* is merely to carry shield *F* around so as to prevent *E* from engaging wheel *B* during the idle period. The illustration represents the beginning of the dwell, which will continue until pawl *M* has moved *A* around so that shield *F* does not interfere with the action

of pawl *E*. Shaft *D* is a running fit in sleeve *I*, which is a force fit in part *H* of the machine.

Automatic Variation in Ratchet Feed Motion. — A special attachment on a wood-turning machine requires a comparatively heavy feed at the beginning, followed by a finer feed for finishing. This alternate retarding and accelerating feed motion is obtained automatically from a ratchet mechanism

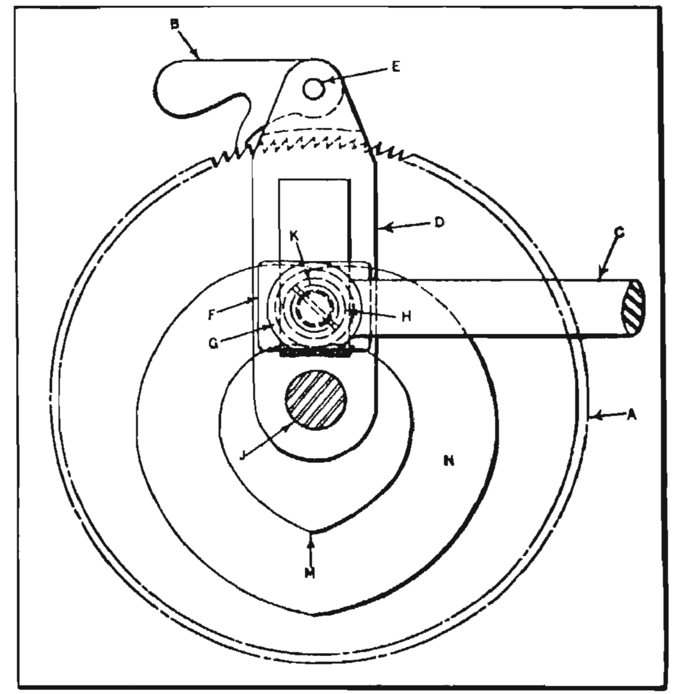


Fig. 7. Ratchet Feed Movement which is Increased and Decreased Alternately as Cam Varies Radial Position of Crankpin

(see Fig. 7) which is so designed that the radial position of the ratchet lever crankpin is continuously increased and decreased by a cam. The ratchet wheel *A* is secured to the feed-screw shaft and the cam groove *N* is cut in one side. Ratchet lever *D* is free to turn on shaft *J*, and it carries the feed pawl *B*. Lever *D* is given a swinging or oscillating movement by link *C* which connects with stud *K*. This stud is driven into the slide or cross-head F, and it has a projection on the other side carrying the cam roll G which engages cam groove N.

It is evident that as ratchet A is intermittently rotated, the cam will first increase the radial position of pin K until point M is passed, and then will return pin K to the minimum radial position shown by the illustration. This increase and decrease between the centers of shaft J and pin K will, of course, have a corresponding effect upon the arc through which lever D swings and the resulting movement imparted to ratchet wheel A and the feed-screw.

Automatic Reduction of Intermittent Movement. — The mechanism to be described is applied to a chucking grinder for automatically reducing the cross-feeding movement and depth of cut, as the diameter of the part being ground approaches the finished size. The head which carries the grinding wheels (three or four wheels are used on this machine) is given a reciprocating motion on the bed of the machine, and the work-spindle head is mounted on a bracket that can be set at an angle relative to the motion of the wheel-carrying slide for taper grinding. The shaft which transmits motion to the cross-feed mechanism shown in Fig. 8 derives its motion from a cam surface on a swinging member of the wheel-head reversing mechanism, which is of the bevel gear and clutch type controlled by a load-and-fire shifting device. The universally jointed telescopic shaft F_2 transmits motion to the cross-feed mechanism at whatever angle the swiveling bracket and work-spindle may be set. The cross-feed screw M_2 has mounted on it a handwheel K_2 and a spur gear N_2. This spur gear is connected with ratchet wheel H_2 by a tumbler gear arrangement controlled by lever J_2, which thus provides for reversing and disengaging the feeding movement. The ratchet wheel is operated by a pawl O_2, pivoted to lever G_2 which, in turn, receives its movement from rockshaft F_2. This movement is positive in the direction which operates the ratchet wheel H_2, and through it the cross feed. In the other direction, motion is derived from a spring R_2 until the point of plunger S_2 brings up against the adjustable stop T_2. As

the position of T_2 governs the extent of the movement of the swinging of lever G_2, a greater or less cross feed is effected at each stroke.

The position of stop T_2, and the amount of feed, is governed by two things. In the first place, the knurled nut U_2

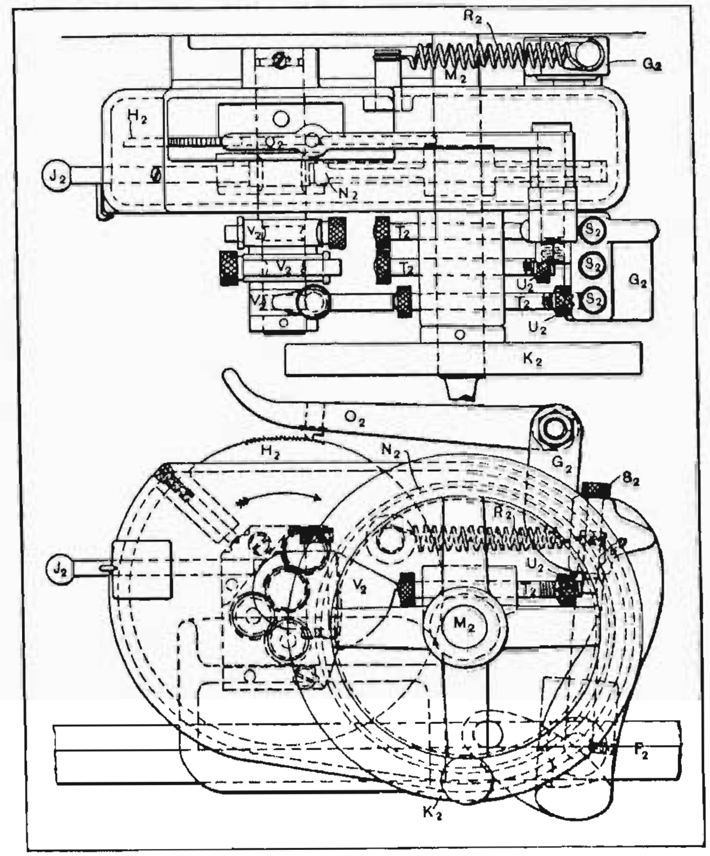


Fig. 8. Ratchet Feeding Mechanism Arranged to Automatically Diminish the Feeding Movement

furnishes a check to its backward movement, and thus regulates the rate of cross feed. Screwing this nut out increases the feed — screwing it back decreases it. In the second place, the feed is controlled by cam V_2, which is adjustably clamped on the shaft of ratchet wheel H_2, and revolves with it in the direction of the arrow. As the feeding progresses, the lower edge of V_2 comes into contact with the left-hand end of stop T_2, gradually limiting its movement from that permitted by the adjustment of U_2 until finally, in the position shown, the swinging of lever G_2 is stopped altogether, thus stopping the cross feed. The diminishing depth of cut thus provided for, as the desired finished diameter is approached, tends to improve the work in regard to accuracy and finish. It will be noted in the plan view that there are three stop cams V_2, three stops T_2, and three feed adjusting nuts U_2 and plungers S_2.

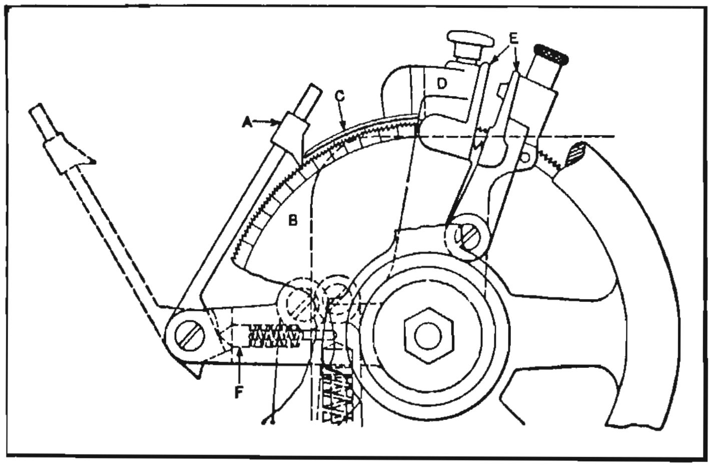


Fig. 9. Ratchet Gearing Arranged to Disengage Automatically after a Predetermined Movement

Any one of these three latter may be pressed down into working position, thus giving a separate cross-feed stop and rate of feed for each of three operations.

Automatic Disengagement of Ratchet Gearing at a Predetermined Point. — The action of ratchet gearing can be stopped automatically after the ratchet wheel has been turned a predetermined amount, by equipping the wheel with an adjustable shield which serves to disengage the pawl after the required motion has been completed. This form of disengaging device, as applied to the cross-feeding mechanism of

a cylindrical grinding machine, is shown in Fig. 9. This mechanism is used to automatically feed the grinding wheel in toward the work for taking successive cuts, and it is essential to have the mechanism so arranged that it can be set to stop the feeding movement when the diameter of the work has been reduced a predetermined amount. When the pawl A is in mesh with the ratchet wheel B, the grinding wheel is fed forward an amount depending upon the position of screws (not shown) which control the stroke of pawl A. The automatic feeding movement continues at each reversal of the machine table, until the shield C, which is attached to head D, intercepts the pawl and prevents it from engaging with the ratchet wheel, thus stopping the feeding movement. The arc through which the ratchet wheel is turned before the pawl is disengaged from it, or the extent of the inward feeding movement of the grinding wheel, depends upon the distance between the tooth of the pawl and the end of the disengaging shield. With the particular mechanism illustrated, a movement of one tooth represents a diameter reduction of 0.00025 inch, so that the amount that the wheel moves inward before the feeding motion is automatically disengaged can be changed by simply varying the distance between the shield and the pawl. To facilitate setting the shield, a thumb-latch E is provided. Each time this thumb-latch is pressed, the shield moves a distance equal to one tooth on the ratchet wheel. For instance, if the shield is at the point of disengagement and the latch is pressed sixteen times, the shield will move a distance equal to sixteen teeth. As each tooth represents 0.00025 inch, a feeding movement of 0.004 inch will be obtained before the pawl is automatically disengaged. This mechanism prevents grinding parts below the required size, and makes it unnecessary for the operator to be continually measuring the diameter of the work. It is located back of a handwheel (which is partly shown in the illustration) that is used for hand adjustment. The pawl is kept in contact with the ratchet wheel and is held in the disengaged position by a small spring-operated plunger F.

Non-stop Feed Ratchet Adjustment—1.—Ordinarily, the feeding movement obtained with a ratchet feeding mechanism is varied by changing the radial position of the operating crankpin, but this is not readily accomplished without stopping the machine. The variable ratchet feeding mechanism shown in Fig. 10 may be adjusted while operating. It consists of a fixed crankpin *A* mounted on a crank disk *B*, which, in

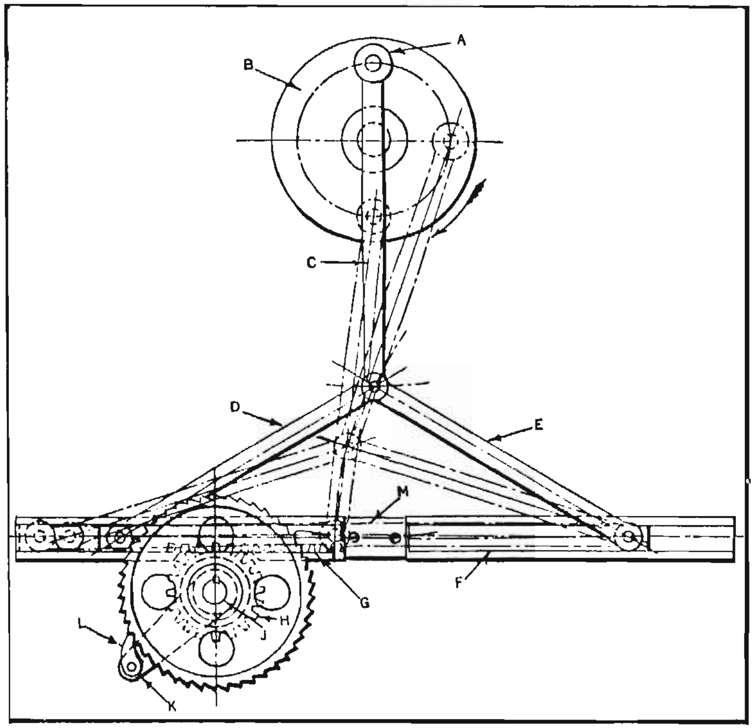


Fig. 10. Ratchet Feed Mechanism which may be Adjusted without Stopping the Machine or Driving Crank

turn, is mounted on a main drive shaft. One end of the pitman *C* is connected to the crankpin, and the other end is connected to links *D* and *E*. Fastened to link *E* is a sliding bar *F*, while link *D* is fastened to the rack *G*. Rack *G* meshes with pinion *H*, which is free on feed-shaft *J* but is connected to an arm *K* carrying pawl *L*. This pawl meshes with the ratchet wheel which is keyed to the feed-shaft.

Thus arranged, the driving motion of the main shaft is transmitted to the pawl arm with the sliding bar *F* abutting against the block *M;* and if the sliding bar is held against *M* during the downward stroke of the crankpin, all of this motion will be imparted to the pawl arm, swinging the pawl the maximum distance back around the ratchet wheel. On the upward movement of the crankpin, the pawl arm will be pulled upward and the pawl, engaging with the ratchet wheel, will turn the wheel through a maximum degree of rotation.

However, if the sliding bar is allowed to have some movement away from block *M* and the movement of the rack and pawl arm is impeded, part of the downward motion will be transmitted to the sliding bar and subtracted from that of the pawl arm, depending upon how far the sliding bar, on the one hand, and the rack and connected elements, on the other hand, are allowed to move. For example, if no limit is placed on the movement of the sliding bar, and no movement of the rack and connected parts is allowed during the downward stroke of the crankpin, all of the movement will be imparted to the sliding bar, and on the upward stroke of the crankpin, the sliding bar will merely come back to *M*, and there will have been no feeding motion imparted to the ratchet wheel and feed-shaft.

It will thus be seen that by regulating the proportion of movements of the sliding bar and the rack, any desired degree of rotation of the ratchet wheel and the feed-shaft may be effected. Suitable arrangements may be provided for setting the position of the sliding bar for each stroke of the machine. This can be easily done without stopping the machine. This mechanism is especially suitable for feeding structural steel, etc., through a punching machine, where the spacing of the holes is variable. The main drive shaft then represents the main shaft of the punching machine.

Non-stop Feed Ratchet Adjustment—2.— Many machines have feed rolls in one form or another, the two roll shafts being geared together, as for example, by gears *A* and *B* (see Fig. 11). The lower shaft of this particular design has a

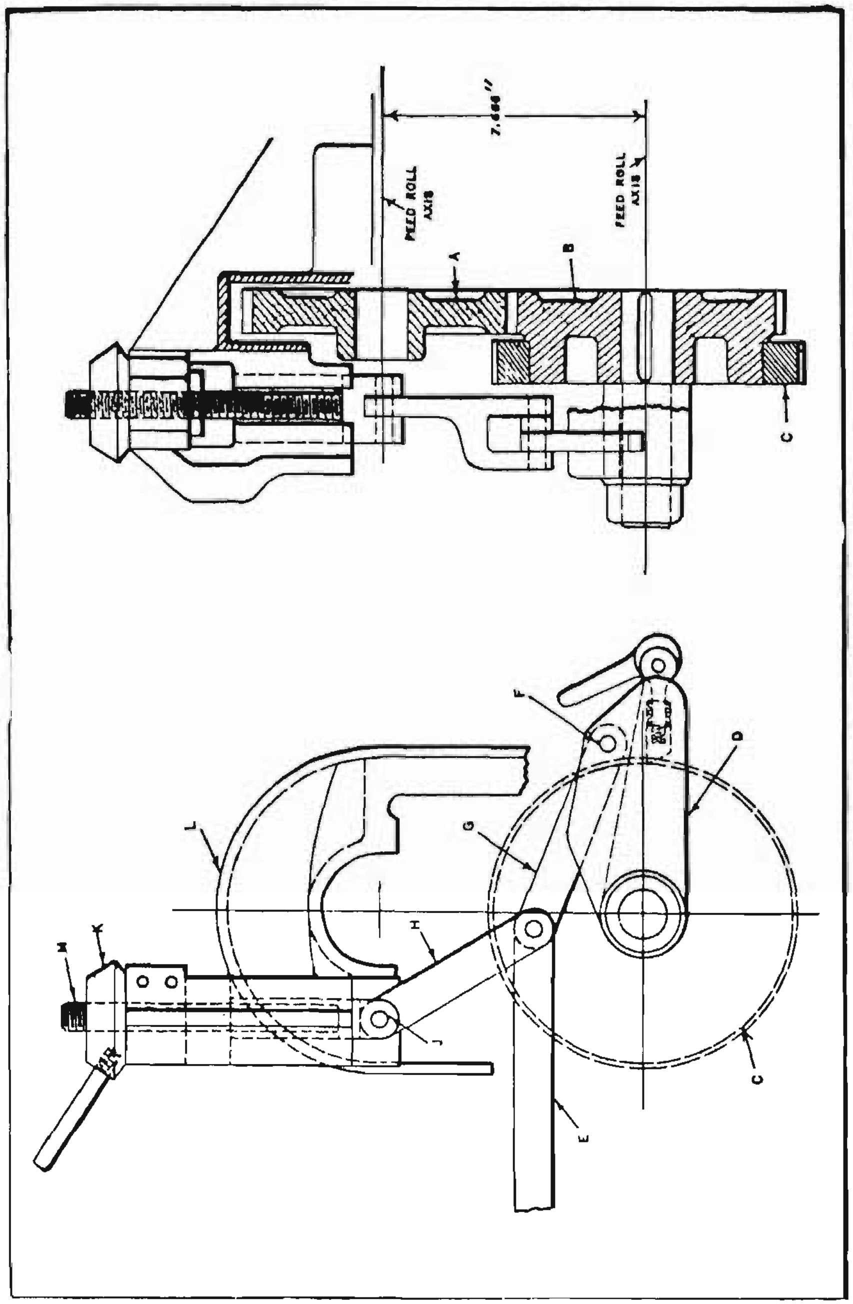


Fig. 11. Another Mechanism for Adjusting Ratchet Feeding Movement without Stopping the Machine

ratchet wheel *C* operated by a pawl carried by lever *D*, which, according to the former arrangement, was given a reciprocating movement by the direct action of a connecting-rod *E* pivoted to it, the motion imparted to *E* depending upon the radial position of a crankpin. When connecting-rod *E* is directly connected to lever *D*, changes in the feeding movement requires stopping the machine in order to increase or decrease the radial position of the crankpin (not shown) to which connecting-rod *E* is attached. Moreover, such trial adjustments may have to be repeated before the desired feeding movement is obtained. In operating these machines, which run at 200 revolutions per minute, it is important to feed the correct amount because the feeding is against a fixed stop; consequently, if the feeding movement is too long, the ratchet teeth and feed rolls wear out very rapidly, and too short a movement would cause scrap blanks.

To avoid wasting time in adjusting the feed mechanism and also secure more accurate adjustments, a feed mechanism was designed which permits varying the feed a very slight amount if desired and while the machine is running at full speed. This improved design is so arranged that the motion is transmitted from connecting-rod *E* to pawl lever *D* through link *G* and a link *H* which swings about a pivot *J* that may be adjusted vertically by turning nut *K*. As pivot *J* is moved upward less motion is transmitted to pawl lever *D*, whereas a downward adjustment of the pivot has, of course, the opposite effect, the feed being increased.

The casting *L*, which supports this adjusting mechanism, has a vertical slot in which the rectangular head of bolt *M* slides as nut *K* is turned. This nut is prevented from moving vertically by the retaining collar seen in the side view. Link *H* is pivoted at *J* to the forked end of bolt *M*. The parts should be so proportioned that link *H* will not swing back beyond the vertical line. Links *H* and *G* of this mechanism measure 5 11/16 inches from center to center of the holes, and connecting-rod *E* is about 4 feet long. The crankpin to which *E* connects is first adjusted roughly to give sufficient motion, and then after the machine is started pivot *J* is adjusted vertically until the feeding movement is exactly what is required. This mechanism works perfectly, and not only saves time, but has lengthened the life of both ratchets and feed rolls. Although some form of friction ratchet would make it possible to obtain a theoretically perfect adjustment, a positive action is desired for this mechanism. A triple form of pawl now is used. The distance between these pawls is equal to one-third of the pitch of the ratchet wheel teeth; hence, in effect, the ratchet wheel has three times the actual

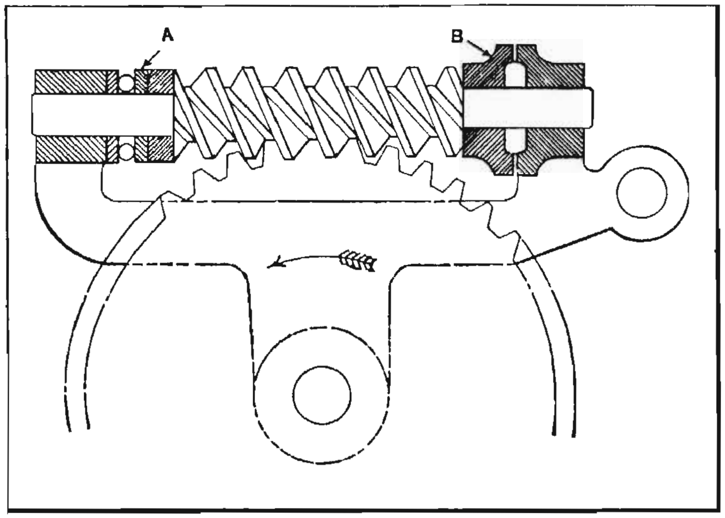


Fig. 12. Friction Ratchet with Worm Type of Pawl

number of teeth, so that comparatively fine adjustments are possible without eliminating the positive drive.

Friction Ratchet of Worm-pawl Type. — In Fig. 12 is shown an ingenious ratchet consisting of a worm-wheel and a worm mounted on a forked lever. When this design is enclosed in an oil-tight case and the parts are properly made, it functions with a high degree of accuracy. However, it is a comparatively expensive arrangement. Either the worm or the worm-wheel may be used for driving the mechanism. It will be seen that the worm-shaft is equipped with a ball

thrust bearing at *A* and is provided at *B* with a bearing that sets up an appreciable friction when subjected to a load. Bearing *B* and one race of bearing *A* are keyed to the shaft.

When the forked lever is operated in the direction indicated by the arrow, the thrust is against the plain bearing *B*, and the frictional resistance prevents the worm from revolving; hence the worm acts as a ratchet and turns the worm-wheel. On the contrary, when the forked lever is operated in the opposite direction, the worm revolves, because it overcomes the relatively slight friction of bearing *A*; consequently, the

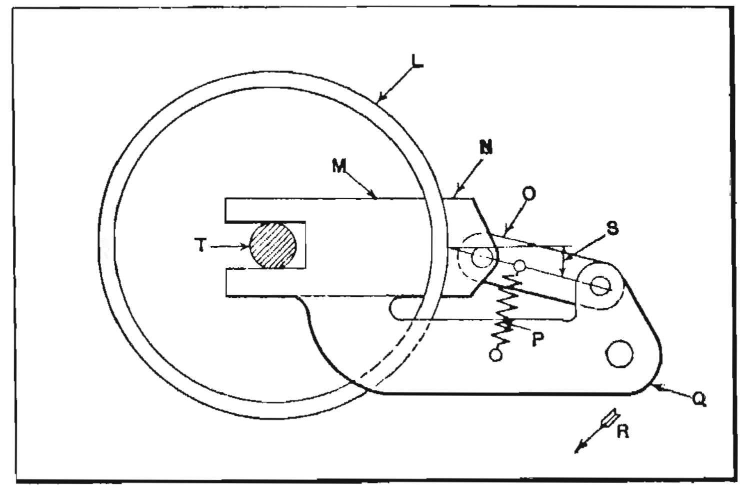


Fig. 13. Friction Ratchet of Toggle Type

worm-wheel remains stationary while the worm is swung backward.

By regulating the amount that the forked lever is moved in the counter-clockwise direction when the worm is the driving member, the movement of the worm-wheel can be varied. This device should not be regarded as a worm and worm-wheel mechanism in the ordinary sense, as no rubbing action takes place between the worm and the worm-wheel teeth when the two members are under a load. Added advantages of this construction are taking up the load without shock, and silent operation. An important point to be observed in designing a ratchet of this type is to make the helix angle of the worm such that the worm will just revolve when the load is on bearing *A*, and will remain stationary when the load is on bearing *B*.

Friction Ratchet of Toggle Type. — The friction ratchet shown in Fig. 13 is so arranged that the flange *L* of the friction wheel is gripped between the body *M* and shoe *N*. The friction surfaces are kept in contact by means of a light spring *P*. When the arm *Q* moves in the direction indicated by the arrow *R*, body *M* and shoe *N* merely slide over flange *L*, but when it moves in the opposite direction, the mechanism is

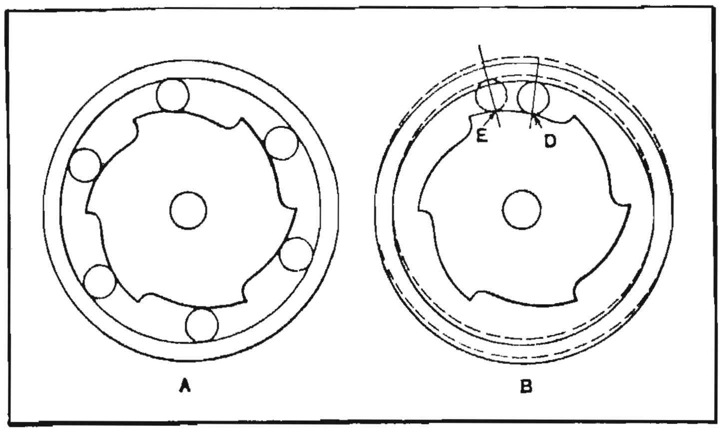


Fig. 14. Two Ratchet Designs that Function by Jamming Balls between the Driving and Driven Members

friction-locked, so that arm *Q*, body *M*, shoe *N*, and link *O* all move together as a solid piece with the wheel.

The advantage of this construction is that the sliding or friction surfaces have a comparatively large area and therefore great durability. This has the disadvantage, however, that, should a heavy film of oil accumulate on flange *L*, the ratchet will slip unless the angle *S* is made not larger than 7 degrees. A thin layer of oil absorbed on flange *L* will not cause the ratchet to fail. In designing this type of ratchet, care should be taken to see that body *M* is made very rigid. Provision should also be made for a radial movement of body *M*; this may be done by slotting the end to fit the central

shaft *T*. Flange *L* and body *M* may be made of cast iron, and shoe *N* of soft steel.

Friction Ratchets of Cam Type. — Two styles of friction ratchets are illustrated in Fig. 14. In the one at *A* the driving rollers are jammed between an internal circular surface of the outer member and cam surfaces on the inner member. With a ratchet of this type, springs are often utilized to force the rollers into contact with the jamming surfaces. One

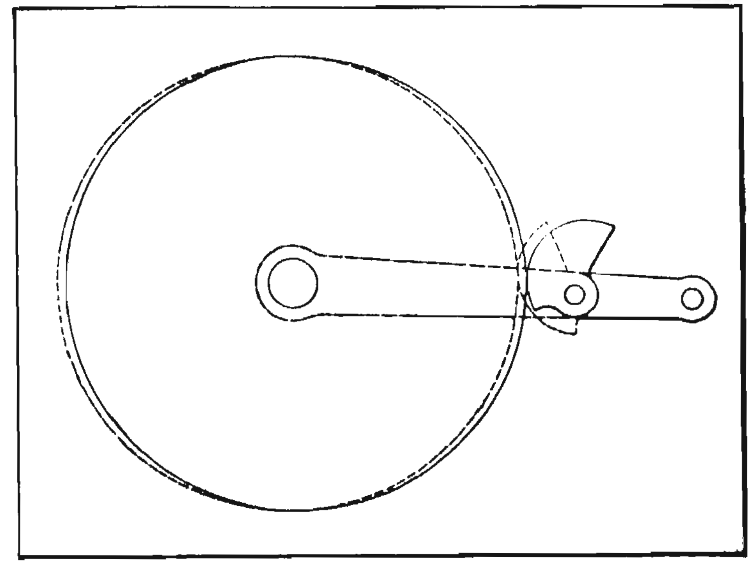


Fig. 15. Friction Ratchet Consisting of a Disk, Segment, and Lever

disadvantage of this design is that the rollers always bear at the same point on the inner member with the result that in time a depression is formed on each curved surface. When this occurs, the device can no longer be relied upon, as a jamming action is impossible because of the shoulders that are formed. The ratchet shown at *B* was designed to overcome this disadvantage. The outer member is placed slightly eccentric relative to the center about which the inner member revolves, as indicated on an enlarged scale by the dotted lines, and so the jamming of the rollers is distributed on each curve between points D and E. By this provision the forming of depressions is obviated.

A type of friction ratchet, which is fairly reliable in the ordinary form, when the disk is concentric, is shown in Fig. 15. However, if the disk is slightly eccentric relative to the fulcrum of the lever to which the engaging segment is attached, much better results will be secured, because the jamming is again distributed over a considerable area of the segment surface, and the forming of a depression on this member is avoided. This will be apparent by reference to the dotted positions of the ratchet disk and segment. Care must be taken to see that the amount of eccentricity is not too great, or the segment will not jam properly on one half of the disk periphery.

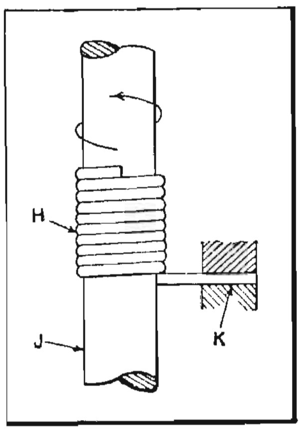


Fig. 16. Ratchet of Coil Spring Type

Friction Ratchets of Coil Spring Type.—The friction ratchet shown in Fig. 16 consists of a close-wound coil spring H which is assembled on the main spring shaft J, and has one end fixed at K. The inside diameter of the spring is slightly less than the diameter of the shaft J so that it normally grips the shaft. A torque in the direction indicated by the arrow tends to open the spring an imperceptible amount, beginning at the free end, until the friction is overcome. The shaft then turns freely. A torque in the opposite direction, however, increases the friction between the spring and shaft, owing to the "belt-like tension," and thus locks the shaft.

A simple method of employing a coil spring to provide a ratchet drive is illustrated in Fig. 17 (see upper illustration). The drum A is fastened to shaft B, and spring C is wound to fit snugly on drum A. One end of the spring is fastened to lever D. If lever D is rotated in the direction opposite that

indicated by the arrow, the spring will slide or slip on the drum. If rotated in the direction indicated by the arrow, the spring will tighten on the drum and drive shaft B.

This type of ratchet tends to produce a frictional drag upon the driven member during its idle stroke due to the gripping action of the spring. This drag can be made to serve a useful purpose in some cases, or it can be reduced to a negligible value by increasing the number of turns and increasing the inside diameter slightly so that the grip on the drum is loosened.

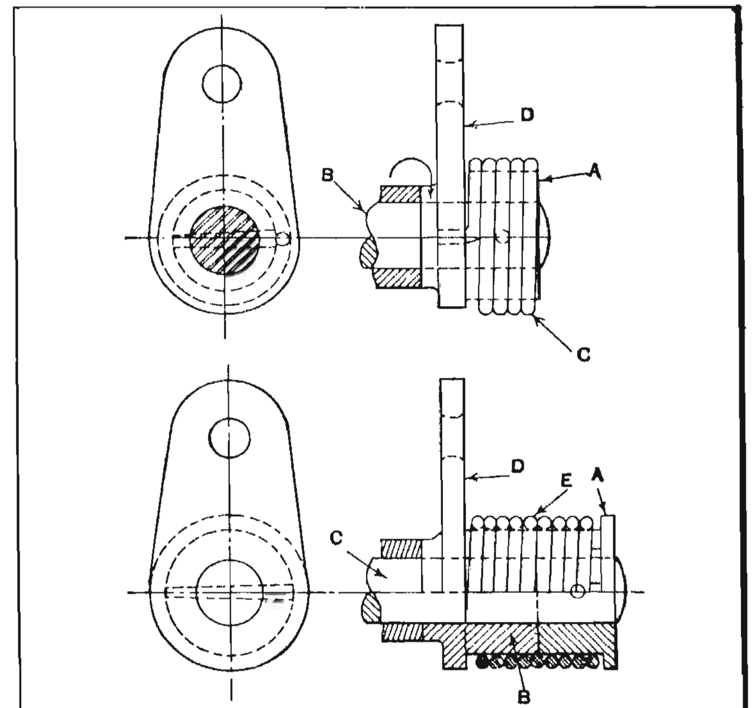


Fig. 17. Coil Spring Ratchet Drives

The spring ratchet shown by the lower illustration depends for its drive upon its gripping action on both the driver and the driven member and neither end of the spring is fastened. The drum B, machined on lever D, drives the drum A on shaft C, by the gripping action of spring E.

Intermittent and Continuous Friction Ratchet Feed Mechanism. — Mechanisms for obtaining both intermittent and continuous feeds that will permit of minute adjustments are sometimes required. A mechanism of this kind which has been successful is shown in Fig. 18. This mechanism was first applied to a horizontal drilling machine used to drill a hole 1 3/32 inches in diameter and 18 inches long in a chrome-nickel steel crankshaft. It was found, upon trial, that a feed of 0.007 inch per revolution was the ideal feed for this work. For some unknown reason, a feed of 0.008 inch made the drill chatter, while a feed of only 0.006 inch caused the work to become glazed so that the drill would not cut after being

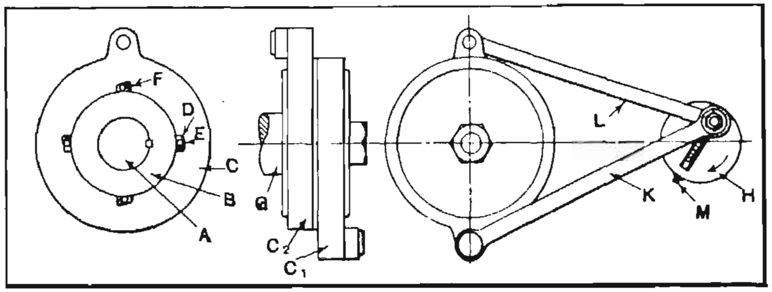


Fig. 18. Adjustable Friction Ratchet Feeding Mechanism which may be Arranged either for Intermittent or Continuous Motion

in use a short time. The drill used was of the oil-tube twist type, with flutes 20 inches long, and was made of high-speed steel.

A feed of the same type as was used for the crankshaft drilling operation was successfully applied to a machine for cutting fabrics. A feeding mechanism of the type described can also be used to advantage in connection with the roll feeds of punch presses. The mechanism used for the drilling operation is of the continuous-feed type, but when used with the fabric cutting machine or in connection with punch press feeding rolls, it is employed to give an intermittent feed.

Referring to the accompanying illustration, the feed or drive shaft A is keyed to a hardened and ground steel collar B around which is placed a hardened and ground ring C. In

ring C are suitable recesses D in which are placed hardened rolls E that are backed up by springs F. The action of this unit is as follows: When the outer ring C is rotated counter-clockwise, there is practically no resistance to its movement, but if it is rotated clockwise, the hardened rolls E are forced against collar B as they ride up the inclined surface of recess D. The rolls finally become wedged, and thus drive collar B which is keyed to shaft A.

In order to obtain an intermittent feed, it is merely necessary to use one of the friction units which is only capable

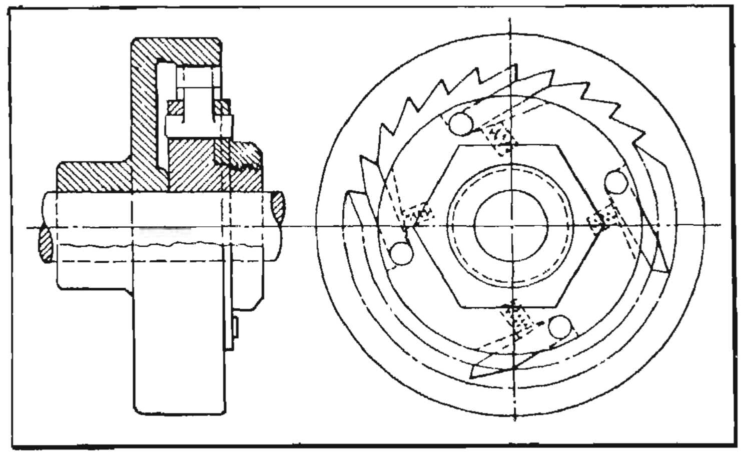


Fig. 19. Internal-tooth Ratchet especially Suitable for High-speed Drives

of driving shaft A in one direction. This unit may be connected to a rotating shaft on the machine by means of an adjustable crank, so arranged that its throw can be easily adjusted to obtain a fine or a coarse feed. For a continuous feed (of varying velocity due to the crank action) it is necessary to employ two units, as indicated in the two views at the right-hand side of the illustration. In this mechanism two outer rings C_1 and C_2, and one collar B are required. If both rings are arranged to drive when turning clockwise, then link L and ring C_2 will be driving while the pin of crank disk H is moving through the upper half of its circle, or from left to right, and link K and ring C_1 will be driving while the crankpin is moving through the lower half of its circle, or from right to left. If the two rings are arranged to drive while rotating counterclockwise, the action just described will be reversed, ring C_2 being the driver when the crankpin is moving from right to left, and C_1 driving when the pin is moving from left to right.

The crank disk *H* may be placed on the main drive shaft or any other shaft that may be conveniently located for the purpose. The screw at *M* provides a means of regulating the throw of the crank and may be equipped with a dial reading to hundredths or thousandths of an inch to facilitate the accurate adjustment of the rate of feed.

As there is no perceptible interval between the release of one ring and the engagement of the other, a practically continuous feed is obtained. If desired, it is permissible to make the recess for the friction rolls in collar *B* instead of in the rings. This construction has the advantage of giving a slightly better contact for the friction rolls owing to the fact that the contact surface of the outer ring is formed to a larger radius.

Internal Ratchet for High-speed Drive. — A ratchet having internal teeth is the best design to employ in a high-speed drive, because centrifugal force tends to assist the action of the pawl; this is particularly true when the pawl is mounted on the driving member. Fig. 19 shows an internal-tooth ratchet in which power is transmitted to the driven member through the medium of four pawls, each of which is assisted in its engagement with the ratchet teeth by a small coil spring.

Cam-operated Ratchet Pawl. — A ratchet feeding mechanism employed to revolve a heavy cylinder is shown in Fig. 20. This device is designed to start the rotation of the cylinder with a sudden movement, stop it quickly, and hold it stationary for a predetermined period, and then repeat the operation. The view at the left-hand side of the illustration shows the mechanism in the position it occupies when the cylinder is being held stationary. The view at the right-hand side shows the mechanism in the position occupied just before the indexing movement begins.

It will be noted that there are but four principal parts to the mechanism, namely, the ratchet wheel *A*, the pawl *B*, the crank *C*, and the cam *D*. Cam *D* rotates continually in the direction indicated by the arrow. When the roller *E* on crank *C* reaches the highest point on cam *D*, the ratchet *A* will have been completely indexed and be locked by the tooth *f* on the crank *C* from further rotation in the direction of the arrow, while the pawl *B* will prevent it from being rotated in the reverse direction.

Ratchet Mechanism for 90-degree Indexing Movement. — A 90-degree indexing mechanism which has no idle return

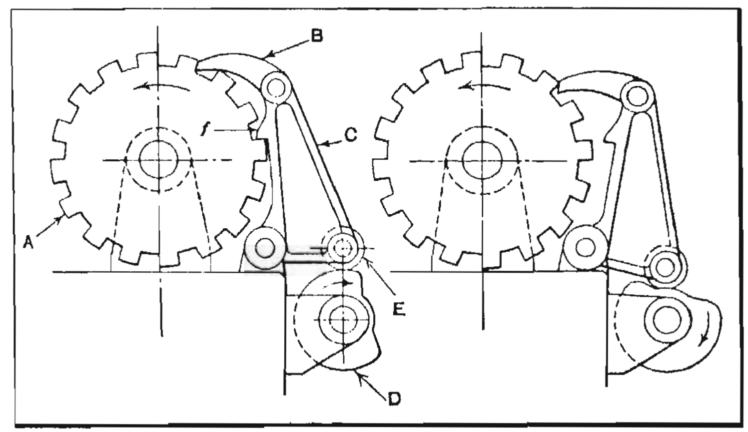


Fig. 20. Cam-operated Ratchet Feeding Mechanism

throw and which can be operated by a short stroke of the indexing member or lever is shown in Fig. 21. The screw or stud *A* is connected to the part of the machine that produces the indexing motion. The links *B* and *C* are connected to the arms *D* and *E* which carry the pawls *F* and *G*. Arms *D* and *E* are free to revolve on the indexing shaft *H* to which the index-wheel *J* is fastened.

To impart an indexing movement of 90 degrees to wheel *J*, pawls *F* and *G* are given a movement of 45 degrees in one direction and then returned to their normal position. A downward movement of stud *A* of the right distance, as indicated by dimension K, will draw levers D and E down so that they are rotated through an angle of 45 degrees. This downward movement brings pawl F backward, so that its point will coincide with the center line X-X, and pawl G will move forward until its point also coincides with center line X-X. Thus pawl G indexes wheel J through an angle of 45

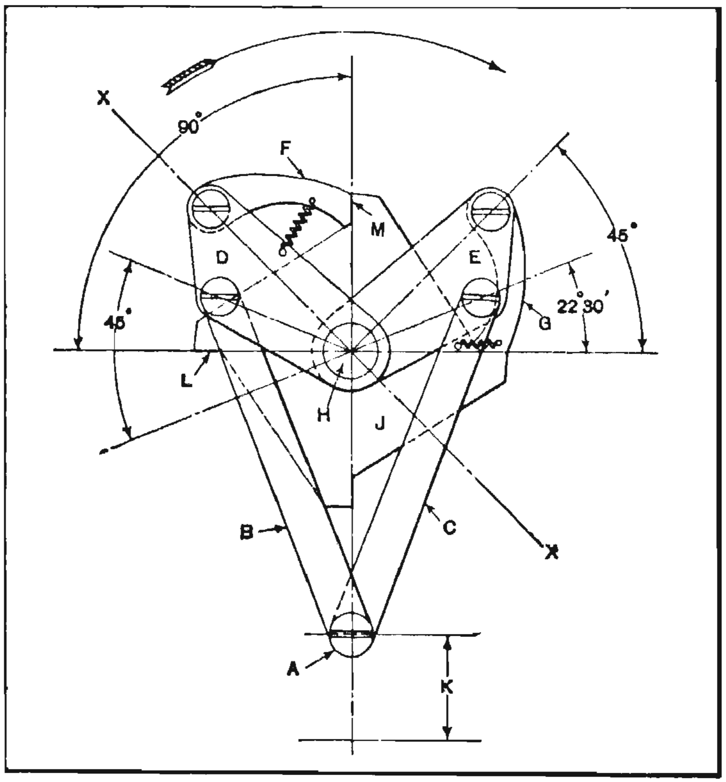


Fig. 21. Mechanism for Producing a 90-Degree Indexing Movement

degrees so that pawl F will catch on the tooth face L of the index-wheel. The return movement of stud A causes pawl F to move forward and to index wheel J the remaining 45 degrees, while the pawl G moves backward to its former position into contact with the face M of the succeeding tooth ready for the next indexing movement.

Index Mechanism for Ratchet Dial Feed. — A limited quantity of small parts, of a design that could be made most economically on a dial feed press, was required. As the comparatively small quantities to be produced did not warrant the purchasing of a new machine, the dial feed shown in

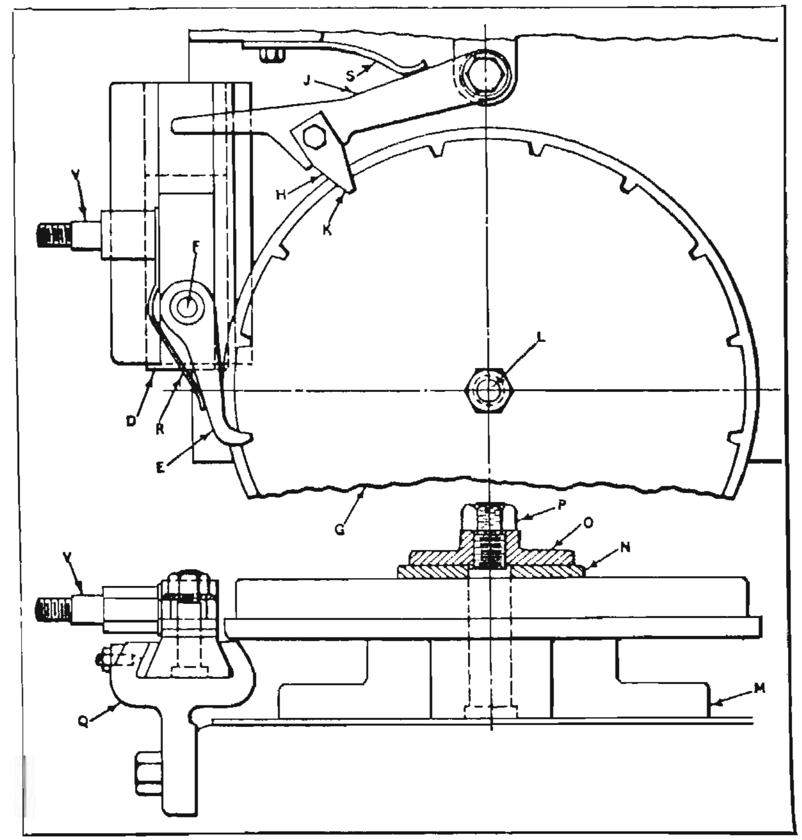


Fig. 22. Ratchet-operated Dial Feed for Press

Fig. 22 was designed and applied to an old press. On the last part of the up stroke of the press ram, a crank-operated rocker arm (not shown) having an elongated slot for receiving stud V, causes the ratchet slide D to be advanced. The pawl E, pivoted on the stud F in slide D, engages one of the notches in the dial-plate G and indexes this member to the next station. At the completion of the cycle, point *H* on lever *J* drops into the notch on plate *G*, locking this member during the down stroke of the press ram and the first part of the upward stroke. The dial-plate *G* is therefore locked while the dies are in operation.

As the slide *D* approaches the end of its rearward movement, it comes in contact with the end of lever *J* and lifts the locking point *H* from the notch in plate *G*. While the point *H* is still held away from plate *G* the point of pawl *E* drops into the indexing notch. When slide *D* has moved forward far enough to permit point *H* to come in contact with plate *G*, the indexing notch at *K* has moved from under point *H*, which then rides on the periphery of the plate until pawl *E* has completed the indexing movement, at which time it drops into the next notch.

The dial-plate *G* is shown blank, without the work stations cut in it, in order to eliminate unnecessary details. The dial-plate is made a good running fit on the stud *L*, which is driven into the die-bed *M*. The die-bed is provided with a flange which supports the dial-plate. A friction washer *N*, made from wood, is held in contact with the upper face of the dial-plate by the washer *O* and nut *P*. The bracket *Q* in which slide *D* operates is provided with an adjustable gib to compensate for wear on the dovetail faces of the bracket and slide.

The flat spring *R* serves to keep the pawl *E* in contact with the dial-plate. The locking point *H* is made from tool steel, and is rigidly secured to the arm *J*. The flat spring *S* serves to hold the locking point in contact with the plate. The stroke of slide *D* may be changed by adjusting the crank which operates the rocker arm.

Escapements for Clock Mechanisms. — An escapement may be considered as a form of ratchet mechanism having an oscillating double-ended pawl for controlling the motion of the ratchet wheel by engaging successive teeth. Escapements are designed to allow intermittent motion to occur at regular intervals of time. The escapement of a clock is illustrated

in Fig. 23. As applied to a pendulum clock the escapement serves two purposes, in that it governs the movement of the scape wheel for each swing of the pendulum and also gives the pendulum an impulse each time a tooth of the scape wheel is released. An escapement should be so arranged that the pendulum will receive an impulse for a short period at the lowest part of its swing and then be left free until the next impulse occurs. The time required for a pendulum to swing through small arcs is practically independent of the length of the arc. For instance, if a stationary pendulum receives an impulse, the time necessary for its outward and return movement will be approximately constant regardless of the impulse and arc of swing, within ordinary limits. Thus, if the impulse is of considerable magnitude, the pendulum starts with a relatively high velocity, but the distance that it travels counteracts the increase of speed so that the time remains practically constant for any impulse or arc of swing. A pendulum that is swinging freely will adapt the length of its swing to the impulse it receives, and any interference which might be caused by the locking or unlocking of the escapement will affect the regularity of movement less if it occurs at the center of the swing rather than at the ends. As the arc of swing increases, there is a very slight increase in the time required for the movement, and, therefore, it is desirable that the impulses given to a pendulum should always be equal.

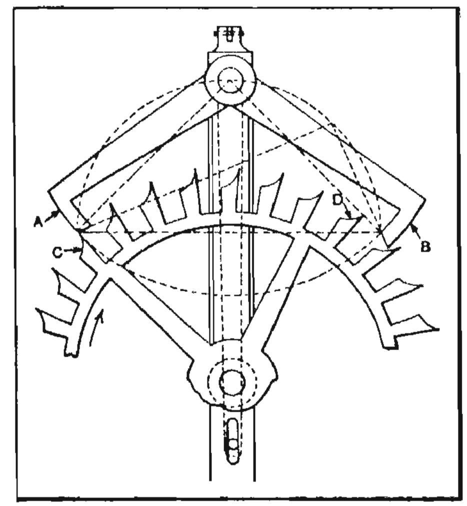


Fig. 23. Escapement for Controlling Action of Clockwork

One of the earlier forms of escapements was known as the "anchor" or "recoil" escapement. With this type, the pendulum was never free, but was controlled by the escapement throughout the swing. To avoid this effect, the Graham "dead-beat" escapement, illustrated in Fig. 23, was designed and has been extensively used. When the escapement is in action, the pallets A and B alternately engage the teeth of the scape wheel, which revolves intermittently in the direction indicated by the arrow. With the mechanism in the position illustrated, the point of tooth C is about to slide across the inclined "impulse face" or end of the pallet A, thus giving the pendulum an impulse as it swings to the left. When tooth D strikes the "dead face" of pallet B, the motion of the scape wheel will be arrested until the pendulum reverses its movement and swings far enough to the right to release tooth D; as the point of D slides past the inclined end of B, the pendulum receives another impulse, and this intermittent action continues indefinitely or until the force propelling the scape wheel around, which may be from a spring or weight, is no longer great enough to operate the mechanism. In designing an escapement of this type, the pallets are so located as to embrace about one-third of the circumference of the scape wheel. One of the features of the dead-beat escapement is the effect which friction has on its operation. During each swing of the pendulum, there is a rubbing action between the points of the scape wheel teeth and the surfaces of the pallets, so that the pendulum is retarded constantly by a slight amount of friction. This friction, however, instead of being a defect, is a decided advantage, because, if the driving force of the clock is increased so that the impulse on the pallets becomes greater, the velocity of the pendulum tends to increase, but this effect is counteracted by the frictional retardation caused by a greater pressure of the teeth of the scape wheel on the faces of the pallet. If the driving force be increased, the frictional retardation increases relatively in a greater proportion than the driving effect and, up to a certain point, the time of vibration of the pendulum diminishes. If the force or weight propelling the clock mechanism is continually in-

creased, a neutral point is finally reached, beyond which a greater force causes the time of vibration to increase instead of to diminish. In the design of clock mechanisms, it is desirable to have a driving power of such magnitude that it neither accelerates nor retards the motion of the pendulum. Many modifications of the escapement previously referred to have been devised to meet special requirements. The escapements of watches and of some clocks and portable time-keeping devices have a balance wheel instead of a pendulum to regulate the period of the intermittent action, but all of these escapements operate on the same general principle.

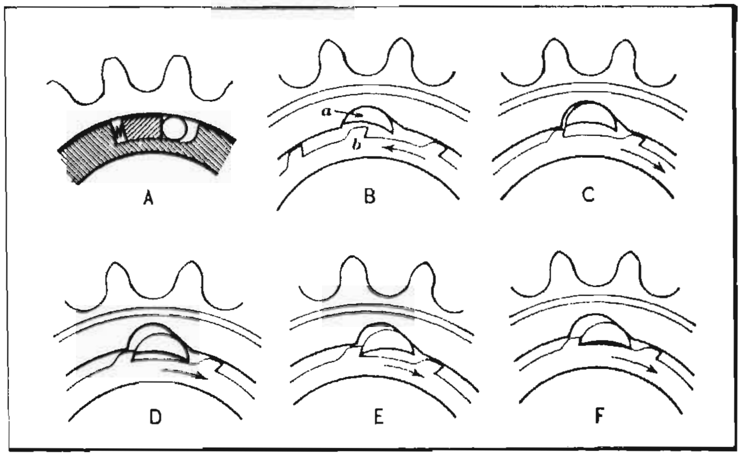


Fig. 24. Ratchet Mechanisms for Releasing Sprockets

Ratchet Mechanisms for Releasing Sprockets. — Some ingenious ratchet mechanisms have been applied to the sprocket wheels of bicycles to permit the pedals to remain stationary while coasting down a grade or hill. A design that has been extensively used is illustrated in principle by the detailed sectional view at A in Fig. 24. The sprocket wheel is not attached directly to the inner member which is shown in section, but motion is transmitted from one part to the other through frictional contact. The inner ring has a series of recesses equally spaced about the circumference. Each of these recesses contains a hardened steel roller or ball, and the bottoms of the recesses are inclined slightly. The rollers are lightly pushed up these inclined surfaces by blocks behind which are small spiral springs. Any relative motion of the inner and outer members of the sprocket causes these steel rollers to either roll up the inclined surfaces and lock the two parts together or to move in the other direction and release the driving and driven members, the action depending upon the direction of the relative movement. For instance, if the outer sprocket is revolved in a clockwise direction, all of the rollers are immediately wedged in their recesses. If the motion of the outer sprocket is suddenly arrested and the inner member continues to revolve, the rollers are immediately released.

An entirely different type of ratchet mechanism designed for use on the sprockets of bicycles is shown by diagrams B to F, inclusive, which illustrate its method of operation. The exterior sprocket is recessed on the inner side for the reception of a crescent-shaped piece a, which acts as the pawl. The depth of the recess and the shape of part a are such that the teeth on the inner ring b can pass freely when moving in the direction indicated by the arrow at B; with motion in this direction, part a simply is given a rocking movement in its recess to allow the successive teeth to pass. When the relative motion is in the opposite direction, as indicated by diagram C, the teeth on the inner member swing part a around in its seat, as shown by the successive diagrams, until it is finally wedged firmly between the two parts as shown at F. These so-called "free-wheel" mechanisms were subsequently replaced by an arrangement operating on the same general principle so far as the releasing mechanism was concerned, but so designed that a backward movement of the pedal also applied a brake.

Ratchet-controlled Press Shearing Mechanism. — In manufacturing a certain design of automobile radiator, two fins are cut from the stock by knife blades held in a bracket mounted at the right-hand side of the press. Seven strokes of the ram are required for piercing all the holes in a fin and so these knives function but once at every seven strokes. This

intermittent operation is produced by the mechanism illustrated diagrammatically in Fig. 25. On the right-hand side of the ram is mounted a 14-tooth ratchet A, which is brought into contact with a spring-actuated finger B on the housing of the machine at each upward stroke of the ram, so that the ratchet is indexed one-fourteenth revolution at each stroke.

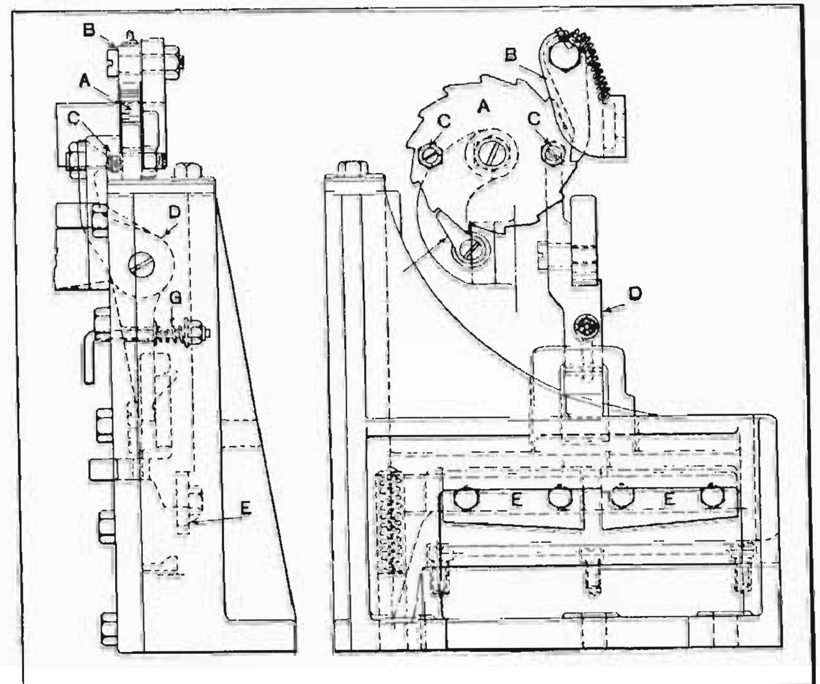


Fig. 25. Mechanism that Operates the Cutting-off Knife Blades in the Power Press at Every Seventh Stroke of the Ram

Diametrically opposite each other on ratchet A are two screws C, and at each seventh stroke of the ram, one of these contacts with the upper end of lever D, swiveling the lower end of the lever to the right so that it engages with the frame containing shear blades E. Then, at the next stroke of the ram, lever D forces the knife frame down to shear off the fins. The spring-actuated pawl F prevents the ratchet A from turning backward when contact is made with lever D, and spring G

insures that the latter will disengage from the knife frame when either of screws *C* has passed the upper end of the lever.

Ratchet Mechanism for Indexing Revolving Chuck. — The chuck shown in Fig. 26 is equipped with two special indexing jaws for gripping a three-way brass fitting and indexing it for machining each of the three open ends without stopping the turret lathe. The indexing is controlled by a lever on the side of the machine, which operates the automatic chuck and bar feed mechanism.

The jaw block *A* contains the indexing mechanism. The indexing jaw *B* is integral with the stem *C*. At *D* there is an eight-tooth ratchet (see also the plan view). Pawls *E* and *F* engage this ratchet, and rotate the jaw. The slides *G* and *H* carry the pawls, and are tongued and grooved into the jaw block *A*.

The forward movement of slide *H* engages the push-pawl *E* and rotates the ratchet one-eighth of a revolution, and the return movement engages the hook-pawl *F*, which completes the one-quarter turn required for each of the three positions. The springs *I* hold the pawls in the proper position.

The plate *J* is keyed to the stem *C*. The under side of plate *J* is drilled and reamed taper at three places to receive the lock-bolt *K* for retaining each indexed position. The lock-bolt is operated by a cam made integral with the slide *H*, as shown by the detail view of the slide (upper right-hand corner of illustration). Referring to the plan view of the jaw, it will be noted that there is considerable clearance between pawl *E* and ratchet *D*. This is so that the cam on the jaw *H* will pass under the roll of the side lever and rock it far enough to extract the lock-bolt from the index-plate before the pawl *E* starts rotating the ratchet.

Looking at the end view of the chuck, two lugs are attached to the chuck face at *L*. The lower ends of the yokes *M* are pivoted to lugs *L*. The upper ends of the yokes *M* are pivoted into grooves cut transversely across the slides *H* and *G* at *N*. A projection of the yoke connects through a link *P* with a plunger *Q* which slides in the spindle.

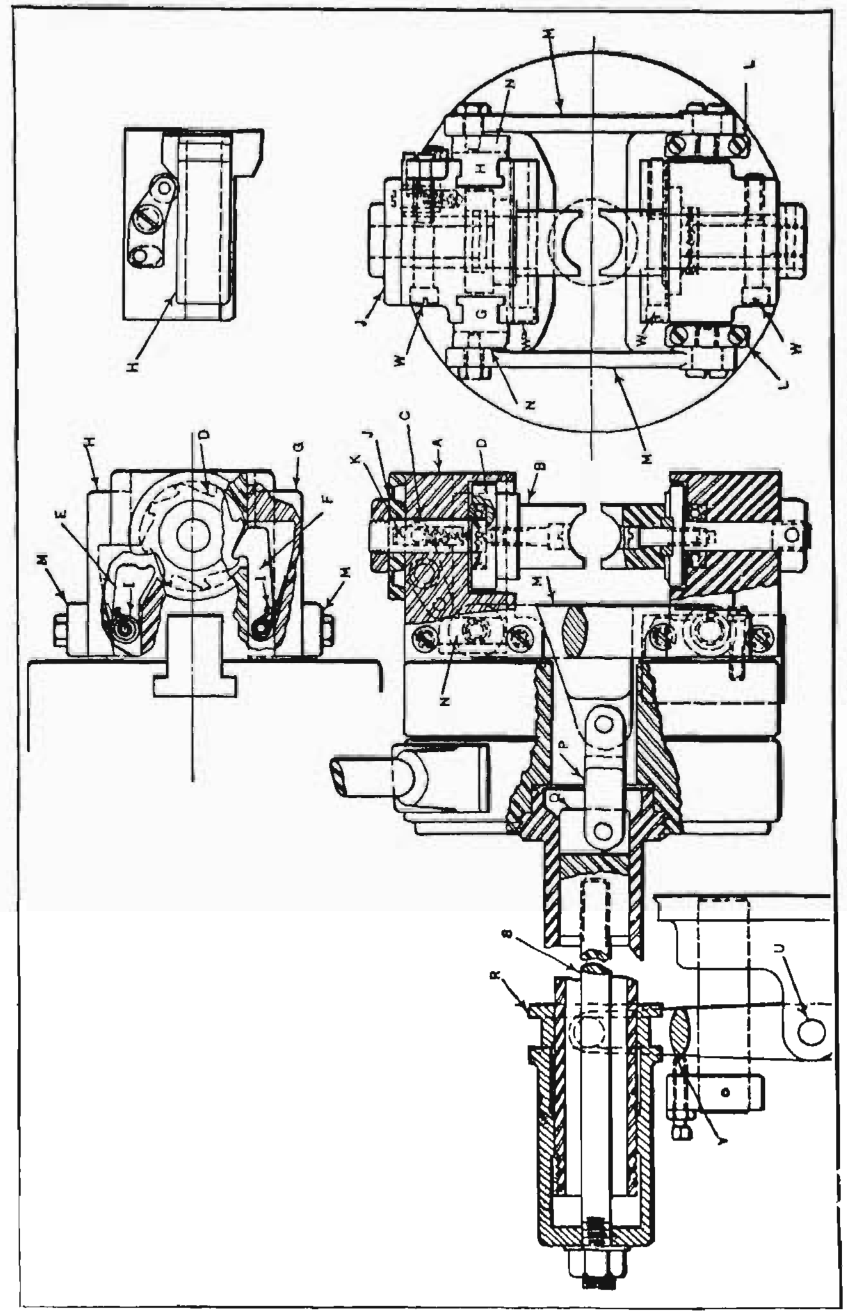


Fig. 26. Turret Lathe Chuck which may be Indexed while Revolving

The sleeve R slides over the rear end of the spindle and replaces the wedge used in the regular automatic chuck mechanism. Sleeve R also connects with the plunger Q through the rod S. A yoke has rolls that engage the groove in the sleeve R. This yoke pivots at U in a bracket on the machine, and its lower end is attached through a link to a rocker arm, swung by a hand-lever which operates an ordinary bar feed mechanism. Stop V limits the rearward movement of the sliding mechanism.

The indexing jaws are both supported on ball thrust bearings. The lower jaw is only a carrier. The jaw blocks are grooved to slip over the master chuck jaw and are bolted entirely through the master jaw, as shown at W in the end view.

These jaw blocks are attached to a two-jaw chuck, but they can be designed for fixtures or faceplates. The material used throughout in the jaws is steel; the yoke and sleeve are of cast iron. Jaws designed as described should not be considered for other than light and medium light work, as the gripping required for heavy work causes a thrust so great that the jaws cannot be revolved by the hand-lever, and to release the jaws slightly for indexing will cause misalignment of the machined surfaces.

In machining the brass parts for which this indexing chuck is used, each of the three diameters is bored to a different depth; hence, three boring heads must be used on the turret. The threads on each of the three diameters, however, are identical and, therefore, could be cut by using only one die-head, which is the method formerly employed. This method required a lot of extra turret indexing in order to swing the die-head back to the work. The present method is to use three die-heads, there being one after each boring position. This arrangement, in conjunction with the indexing mechanism, saves much time, as 665 pieces can be machined daily.

CHAPTER III

INTERMITTENT MOTIONS FROM GEARS AND CAMS

WHEN a shaft which rotates continuously is to transmit motion to another shaft only at predetermined intervals, intermittent gearing is sometimes used. Gearing of this type is made in many different designs, which may be modified to suit the conditions governing their operation, such as the necessity for locking the driven member while idle, the inertia of the driven part, or the speed of rotation. With some forms of intermittent gearing, the driven gear rotates through a fractional part of a revolution once for each revolution of the driver, whereas, with other designs, the driving gear transmits motion to the driven gear two or more times while making a single revolution. The number of times that the driven gear stops before it is turned completely around is varied in each case according to the requirements; the periods of rest may also be uniform or vary considerably.

Gears for Uniform Intermittent Motion. — The design of intermittent gearing illustrated by diagram A, Fig. 1, is so arranged that the driving gear, which has only one tooth, revolves fourteen times for each revolution of the driven gear. Each time the tooth of the driver engages one of the tooth spaces in the driven gear, the latter is turned through an arc a. The driven gear is locked against rotation when the driving tooth is not in mesh, because the circular part of the driver fits closely into the concave surfaces between the tooth spaces as they are successively turned to this position. The radius of the driver should be small enough to insure adequate locking surfaces between the tooth spaces, but not so small that sharp weak points will be formed at the edges of the tooth spaces. Counting mechanisms are often equipped with gearing of this general type. In order to vary the relative movements of the driving and driven gears, the meshing teeth may be arranged in various ways. For instance, if a second tooth were added to the driver on the opposite side as indicated by the dottted lines, the driven gear would receive motion for each half revolution of the driver. The diagram at *B* illustrates another modification. In this case, the driven gear has a smaller number of rest periods, and it is turned farther for each revolution of the driver, as the latter has three successive teeth.

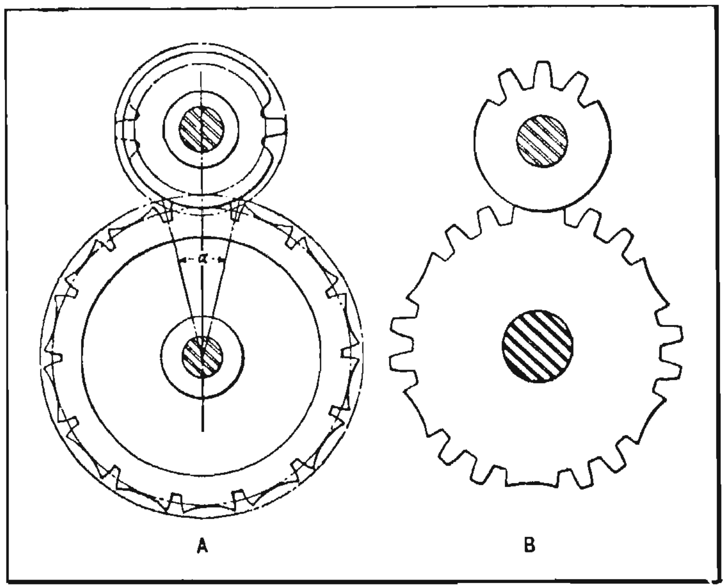


Fig. 1. Gears for Uniformly Intermittent Motion

Variable Intermittent Motion from Gearing. — With some forms of intermittent gearing, the driven gear does not move the same amount each time it is engaged by the driver, the motion being variable instead of uniform or equal. The diagram *A*, Fig. 2, shows an example of the variable motion intermittent type. The driving gear has four driving points around its circumference with numbers of teeth at each place varying from one to four. The tooth spaces on the driven gear are laid out to correspond so that the motion received

by the driven gear is either increased or decreased progressively depending upon the direction of rotation of the driver. Gearing of this general type may be arranged in many different ways and is designed to suit the particular mechanism of which it forms a part. After laying out gears of this kind, it is often advisable to make brass templets in order to ascertain by actual experiment if the gears are properly formed and give the required motion.

Intermittent Gearing for High Speeds. — The design of gearing illustrated by diagram *B*, Fig. 2, is considered pref-

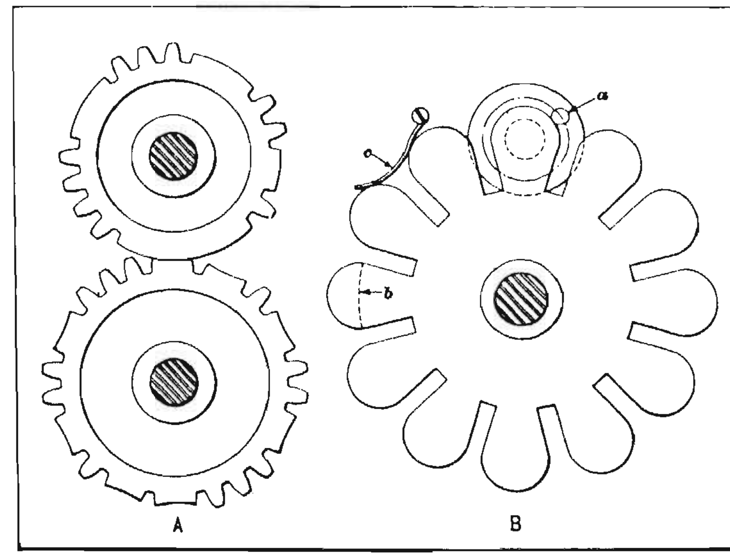


Fig. 2. (A) Gearing for Variable Intermittent Motion; (B) High-speed Intermittent Gearing

erable to the forms previously described, where the driving member revolves at a comparatively high speed. With this gearing, the driven member is stationary during one-half revolution of the driver. The latter has a stud *a* or roller which engages radial slots in the driven gear while passing through the inner half of its circle of travel. The flat spring illustrated at *c* is used to hold the driven wheel in position so that the driving roller will enter the next successive slot without interference. The projections or teeth on the driven gear may have semi-circular ends as shown, or all of the ends may be concentric as indicated by the dotted line at *b*. If the semi-circular ends are not provided, there should be some form of positive locking device to insure alignment between the radial slots and the driving pin or roller. The corners should also be rounded to facilitate engagement of the roller.

Another form of intermittent gearing designed to eliminate shocks when operating at relatively high speeds is illustrated in Fig. 3. The speed ratio between the driving and

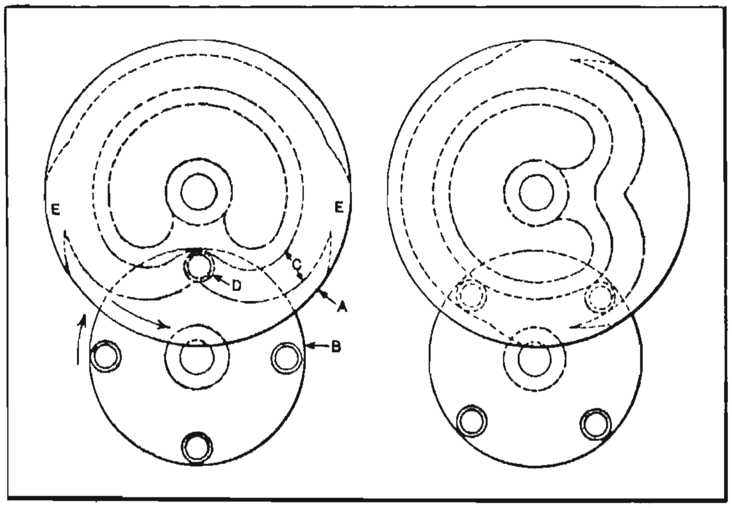


Fig. 3. Another Form of Gearing Designed to Eliminate Shocks at High Speeds

driven members is 4 to 1, each revolution of the driver turning the driven wheel one-quarter revolution, or 90 degrees. The driver *A* has a cam groove *C* which is so shaped that the motion of the driven wheel *B* is gradually accelerated and retarded at the beginning and end of its movement. This groove is engaged by rollers *D* on the driven wheel. The rollers enter and leave the cam groove through the open spaces at *E*, and when the driven wheel is stationary, two of the rollers are in engagement with this groove, thus effectually locking the driven member. The illustration at the left shows the driven wheel at the center of its movement, and the view

to the right shows the relative positions of the two parts after the movement is completed. As the roller at *D* is revolved 45 degrees from the position shown, the following roller enters the cam groove through the left-hand space *E*.

Rapid Intermittent Motion for Moving Picture Projector. — A very rapid intermittent motion is required on moving picture projectors. The film is not moved continuously, but each view or positive on it is drawn down to the projecting position while the shutter is closed, and the film remains stationary for a fractional part of a second while the picture is exposed on the screen; then, while the shutter is again closed, the next successive view is moved to the projecting position. It is apparent, therefore, that moving pictures are, in reality, a series of stationary pictures thrown upon the screen in such rapid succession that they are, in effect, blended together and any action or movement appears continuous. It is important to give the film a very rapid intermittent motion, because it is necessary to have the shutter closed when this movement occurs; and the length of time that the shutter is closed should be reduced to a minimum. This shutter is in the form of a wheel or disk, and it has a fan-shaped section which passes the projector lens while the film is being shifted. In order to avoid flicker on the screen, the shutter has two additional fan-shaped sections. With these three equally spaced sections, the light is not only shut off from the screen during each successive film movement, but twice between each movement at uniform intervals. By closing the shutter twice while the picture is on the screen, the flicker that would be visible and annoying if the shutter were only closed while moving the film is multiplied to such an extent that it becomes almost continuous and is practically eliminated as far as the observer is concerned, assuming that the projector is operated at the proper speed. The width and area of that section of the shutter which is passing the lens when the film is being moved is governed by the time required for the film movement. Theoretically, the area of each section or segment of the shutter should be equal. although, in practice, the two extra sections are made of somewhat smaller area than the main one, in order to increase the open space and the percentage of area left for the passage of light.

There is an important relation between this shutter wheel and the intermittent motion or gearing of the projector. This is due to the fact that the shutter must be closed while the film is being shifted. With the mechanism to be described, the film movement is very rapid so that the shutter blades may be proportionately reduced in area, thus leaving more open space for the light. The intermittent motion referred to is

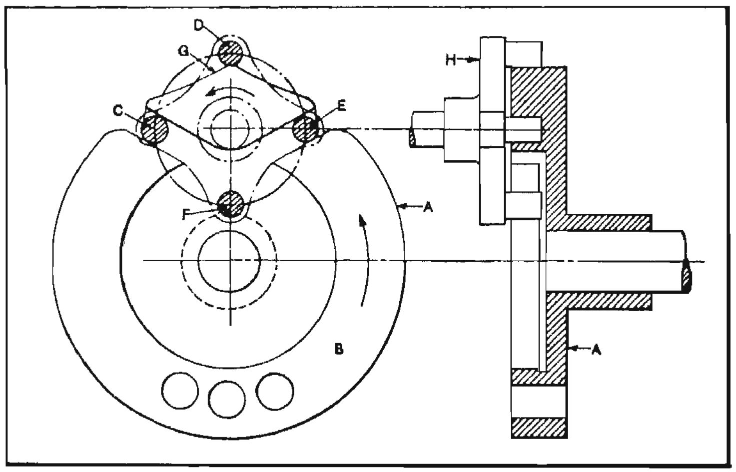


Fig. 4. Rapid-acting Intermittent Gearing of Moving Picture Projector

shown in Fig. 4. This mechanism is composed of a disk or wheel *A*, having an annular flange or ring *B*, which has two diagonal slots across it as shown; this wheel, which is the driver, imparts an intermittent motion to the follower *H*, which carries four equally spaced pins or rollers that engage the ring *B* on wheel *A*. Each time this wheel makes one revolution, the follower *H* is turned one-quarter revolution and in the same direction, as indicated by the arrows. The follower is stationary except when it is engaged by the slots or cam surfaces formed on one side of ring *B*. During this stationary period, the ring *B* simply passes between the four

pins on the follower, two of these pins being on the outside and two on the inside of the ring.

The quarter-turn movement is obtained in the following manner: When the projection or cam surface *G* on the revolving wheel *A* strikes one of the pins, the rotation of the follower begins, and the pins are so spaced that one on the outside moves through a diagonal slot in ring *B* while a pin on the inside moves outward through the other slot. For instance, if the pins *C* and *D* are on the outside and *E* and *F* on the inside, pin *D* will first be engaged by cam surface *G* and, as the follower revolves, pin *C* will pass in through one diagonal slot while pin *E* is moving to the outside of the ring through the other slot. At the completion of the quarter-turn movement, pins *C* and *F* will be on the inside and *D* and *E* on the outside. As wheel *A* continues to revolve, ring *B* simply passes between these closely fitting pins which lock the follower against movement until projection *G* again comes around and strikes the next successive pin on the follower.

The follower operates a toothed wheel or sprocket which connects with the film and moves it downward each time the shutter is closed. Above and below the intermittent gearing there are other sprockets which rotate continuously, and these are so timed that a loop of film is formed above the intermittent gearing that is just large enough to provide for one film movement, which is equivalent to the length of one view or positive. As the film is drawn down rapidly by the intermittent mechanism, a loop is formed below it which is taken up by the lower sprocket as the film is wound upon the lower receiving reel. The normal speed of wheel *A* is sixteen revolutions per second, and it has been operated at two or three times the normal speed. The time required for turning the follower one-quarter revolution is approximately one-sixth of the time for a complete revolution, or 1/96 second, when running at normal speed. With the Geneva motion, which has been applied to many projectors, approximately one-quarter of the time is required for the intermittent action; therefore, the shutter blades must be of larger area than when the film movement occurs in one-sixth of the time. The mechanism shown in Fig. 4 is claimed to be superior to the Geneva motion in that there is less tendency to subject the film to injurious stresses. The locking of the follower during the stationary period is also more secure, especially at the critical time when near the operating point. The three holes drilled in the ring *B* are to compensate for the slots on the opposite side and to balance the wheel *A*.

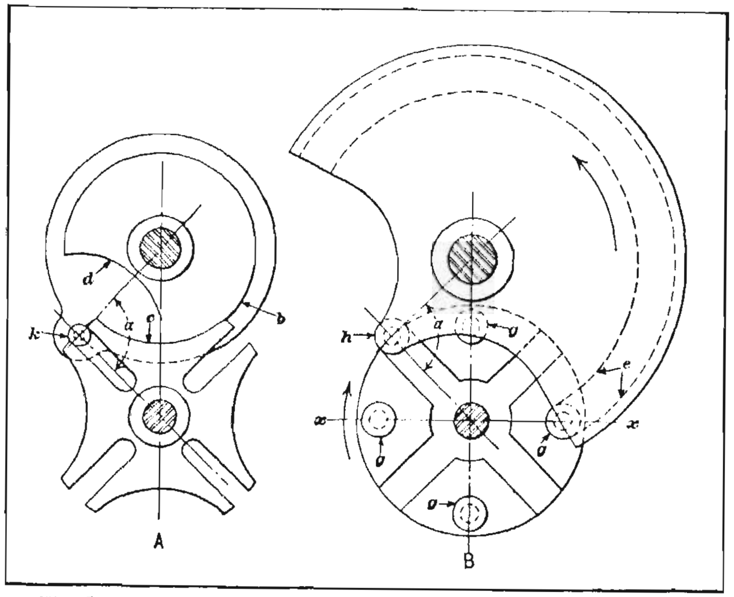


Fig. 5. Geneva Wheels which Vary in Regard to Method of Locking Driven Member during Idle Period

Geneva Wheel for Intermittent Motion. — The general type of intermittent gearing illustrated in Fig. 5 is commonly known as the "Geneva wheel," because of the similarity to the well-known Geneva stop used to prevent the over-winding of springs in watches, music boxes, etc. Geneva wheels are frequently used on machine tools for indexing or rotating some part of the machine through a fractional part of a revolution. The driven wheel shown at *A* in the illustration has four radial slots located 90 degrees apart, and the driver carries a roller *k* which engages one of these slots each time

t makes a revolution, thus turning the driven wheel one-quarter revolution. The concentric surface *b* engages the concave surface *c* between each pair of slots before the driving roller is disengaged from the driven wheel, which prevents the latter from rotating while the roller is moving around to engage the next successive slot. The circular boss *b* on the driver is cut away at *d* to provide a clearance space for the projecting arms of the driven wheel.

The Geneva wheel illustrated by diagram *B* differs from the one just described principally in regard to the method of locking the driven wheel during the idle period. The driven wheel has four rollers *g* located 90 degrees apart and midway between the radial grooves which are engaged by the roller of the driver. There is a large circular groove *e* on the driver having a radius equal to the center-to-center distance between two of the rollers *g*, as measured on the center line *xx*. This circular groove engages one of the rollers as soon as the driving roller *h* has passed out of one of the grooves or radial slots. Each time the driver makes one revolution, the two rollers on the center line *xx* are engaged by the locking groove. The illustration shows the driving roller about to enter a slot and the locking roller at the point of disengagement. When the driven wheel has been moved 90 degrees from the position shown, the roller which is now at the lowest position will have moved around to the left-hand side so that it enters the locking groove as the driving roller leaves the radial slot.

When designing gearing of this general type, it is advisable to so proportion the driving and driven members that the angle α will be approximately 90 degrees. The radial slots in the driven part will then be tangent to the circular path of the driving roller at the time the roller enters and leaves the slot. When the gearing is designed in this way, the driven wheel is started gradually from a state of rest and the motion is also gradually checked.

Geneva Wheel Designed for Slight Over-Travel. — An ingenious special Geneva wheel was developed to operate the conveyor of an automatic weighing machine. This conveyor, by means of brackets attached to it, pushes the packages to be

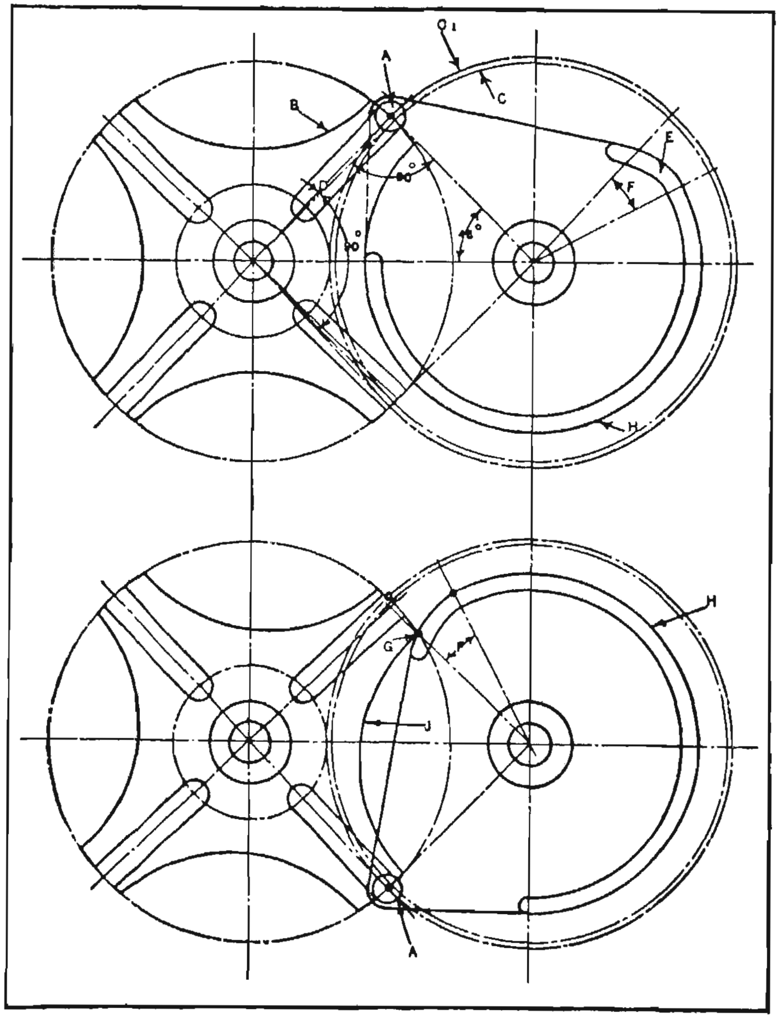


Fig. 6. Special Geneva Motion which Gives Slight Over-travel and Equal Return Movement to Provide Clearance

weighed on the scale platforms of the machine and then stops and remains stationary while the weighing takes place.

Originally, this conveyor was operated by an ordinary Geneva wheel, but the packages sometimes rubbed against

the conveyor brackets, which seriously affected the accuracy of the weighing. To prevent this, it was necessary to have the conveyor moved back about 1/8 inch after delivering the packages to the scale platforms, thus providing clearance between the conveyor brackets and the packages. This clearance has been obtained by a slight change in the Geneva wheel, the result being that the conveyor is first given a small amount of over-travel and is then withdrawn a distance equal to this over-travel, thus providing the clearance desired.

The over-travel has been obtained by enlarging the diameter of the path traversed by the driving roller A (see Fig. 6). Circle C represents the normal path described by the center of roller A, or the radial position of the roller when the movement of driven wheel B is 90 degrees during each roller engagement. By increasing the diameter of this path, as represented by circle C_1, driven wheel B and the conveyor are given the required amount of over-travel. The center lines of the slots in wheel B are approximately tangent to circle C_1, so that engagement takes place without shock and the mechanism operates smoothly. Angle D is equivalent to one-half of the angular over-travel imparted to wheel B .

When roller A reaches the position shown by the lower view, it has moved wheel B 90 degrees plus twice the angle D. Now as the roller leaves its slot, cam surface E, which extends through angle F, comes into contact with corner G of the slotted wheel and pushes the wheel back an amount equal to twice the angle D, thus withdrawing the conveyor brackets from the packages during the time required for weighing. The driven wheel is locked during this dwell by the engagement of concentric surface H with arc J on the driven wheel, the wheel being released as soon as roller A again moves around into engagement with a slot, or in the position indicated by the upper view.

This simple method of preventing frictional resistance between the conveyor brackets and packages during the weighing operation did not introduce any difficulties in manufacturing the intermittent motion described.

Intermittent Motion for Dial Feed.—An intermitttent motion which is incorporated in an automatic station-dial machine for buffing brass shells is so designed that shaft *A* (see Fig. 7) revolves intermittently and has eight dwelling periods per revolution, each dwell being equivalent to 3/5 revolution of driving shaft *B*. The disk *C* is fastened to shaft *A*, and in it there are eight equally spaced hardened steel pins *D* and an equal number of larger pins *E*. As the sectional view shows, pins *D* are located on a higher level than pins *E*. As shaft

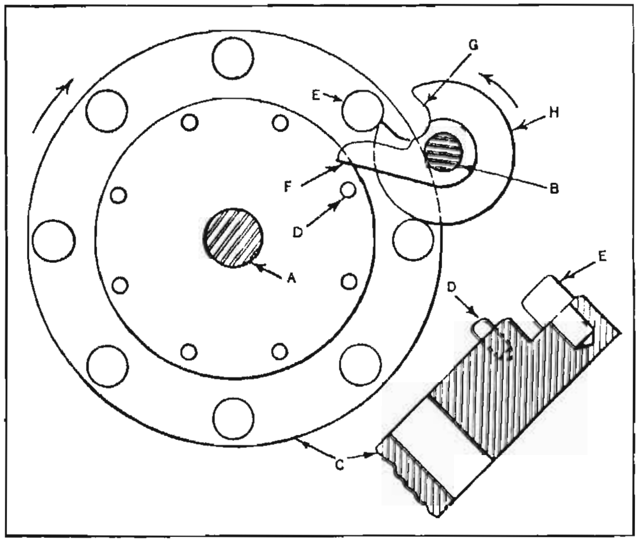


Fig. 7. Intermittent Motion for Dial Feed Mechanism

B revolves in the direction indicated by the arrow, a finger *F* first engages pin *D* and turns disk *C* until the larger pin *E* is engaged by notch *G*; when *F* leaves pin *D*, the positive drive between *E* and *G* continues until one-eighth revolution of disk *C* is completed. Then the concentric surface *H* is in contact with and tangent to two of the pins *E*, thus locking the driven disk in the dwelling position until the driver is again in position for an indexing movement.

One-twelfth Turn of Driven Shaft to One and One-quarter Turns of Driver.— The solution of an interesting problem

in design is indicated in Fig. 8. The requirements were that for every one and one-quarter revolution of a continuously rotating shaft *A*, a second shaft *L* in alignment with the driving shaft must rotate intermittently, with equal velocity and in the same direction as shaft *A*, one-twelfth revolution or through an angle of 30 degrees. An eccentric bushing *C* is keyed to the driving shaft *A*. A 96-tooth gear *D* is loosely mounted on eccentric bushing *C*, but is prevented from rotating by lever *E*; the pitch-line of gear *D*, however, is always tangent to the pitch-line of the 120-tooth gear *F* and to that of the planetary pinion *G*. This pinion is carried by a double

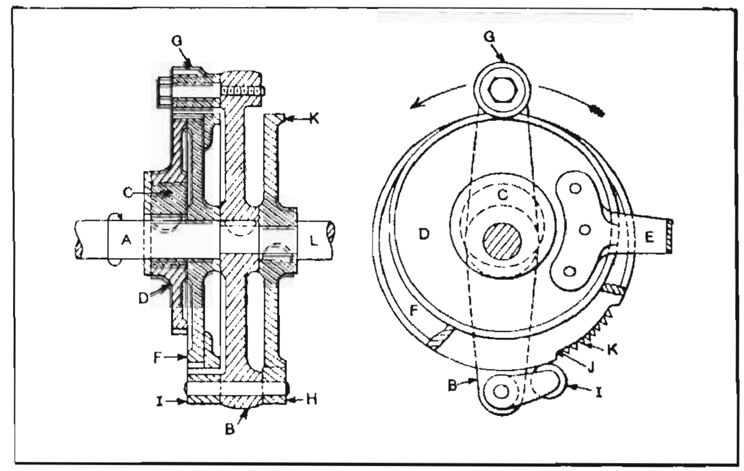


Fig. 8. Mechanism for Rotating Driven Shaft Intermittently and at Same Velocity as Driver

arm *B* which is also keyed to the driving shaft *A*. As arm *B* traverses the pinion around gear *D*, gear *F* is revolved on shaft *A* in the ratio of 120 to 96 or 1.25 to 1. The end of arm *B* opposite the pinion carries a link and roller *I* which runs on a flange of gear *F* until a depression in the periphery allows the roller to drop and permits pawl *H* to engage ratchet wheel *J* which is keyed to shaft *L*; each time the pawl engages the ratchet wheel, the latter is turned forward until roll *I* runs up on top of the flange again. As gear *F* advances one-fifth revolution for each revolution of the arm and pawl, and since 30 degrees equals one-twelfth revolution, the opening or depression for the roll must be shortened $1/5 \times 1/12$ of 30 degrees, or to $29\frac{1}{2}$ degrees.

Two-speed Intermittent Rotary Motion. — The fast and slow motion of the pattern cylinder of a certain type of loom is derived from the reversible intermittent gearing shown in Fig. 9. The large gear *A* is mounted on the pattern cylinder shaft, and receives its motion either through the segment gear and crank combination *B* or through a similar combina-

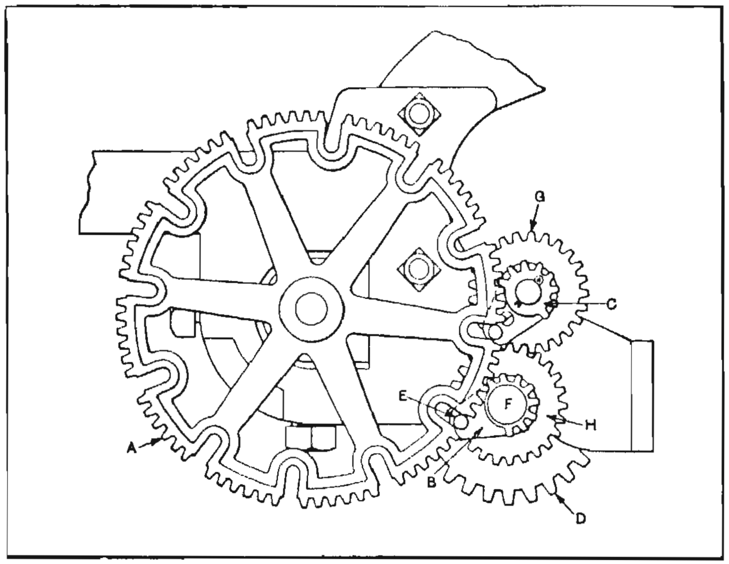


Fig. 9. Two-speed Reversing Intermittent Gearing

tion *C*, these two combinations being used to reverse the direction of rotation. Gear *D* is the driver for this train of mechanism. Whether the motion is transmitted from gear *D* to the pattern gear *A* through the crank and segment gear combination *B* or through combination *C* depends upon the position of a sliding key *F*. An intermittent fast and slow motion is obtained with either combination. When key *F* locks the crank and gear *B* to the shaft, the pattern wheel is rotated at a relatively slow speed when the segment pinion is acting as the driver, and at a faster speed when the crank-

pin *E* comes around into engagement with one of the radial slots in the pattern gear. When this direct drive is employed, the gears *G* and *H* revolve idly with the upper crank and gear combination *C*. When a reversal of motion is required, sliding key *F* is pushed in to engage gear *H*, which then drives gear *G* and the combination at *C*.

Adjustable Intermittent Motion. — The intermittent feed mechanism shown in Fig. 10 is so arranged that the intermittent action may be varied according to requirements by means of a simple form of "skipping" device. A pitman con-

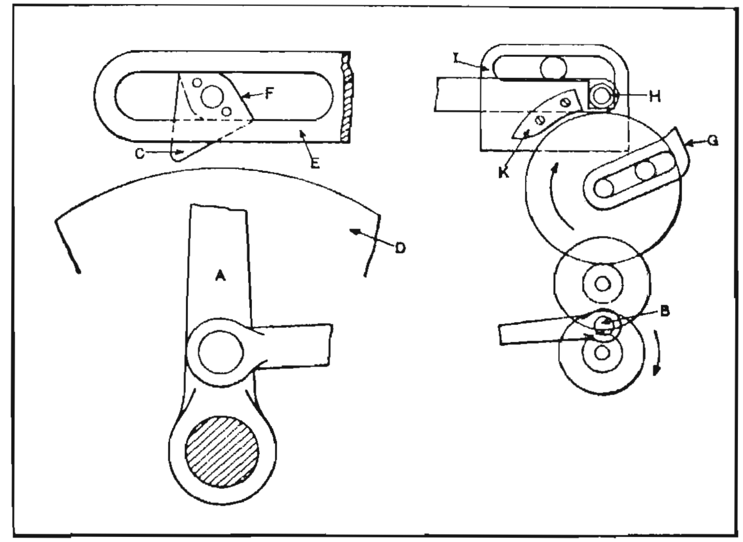


Fig. 10. Feed Mechanism with Skipping Device for Varying the Intermittent Motion

necting with crank *B* transmits an oscillating movement to lever *A*. This lever carries a stud on its free or upper end upon which is pivoted a fiber pawl *C*. This pawl engages the smooth periphery of disk *D* and turns the latter a fractional part of a revolution when lever *A* is moving to the left, unless the engagement of the pawl is prevented by the mechanism to be described. The pawl is formed of two pieces attached to opposite sides of a diamond-shaped block *F*. This block is within the slot and, being slightly thinner than the bar, causes the projecting sides C to frictionally engage the lower side of the bar. Any motion of lever A towards the right causes the pawl to turn to the position shown so that it clears the disk D for the return stroke. The reverse motion of lever A changes the position of block F so that the ends C grip the disk D, which is given the required feeding movement. The skipping of the feed is accomplished by a train of change-gears and a cam G. This cam serves to lift the pin H clear of its seat, so that the bar carrying pawl C is free to slide horizontally as lever A moves to the left;

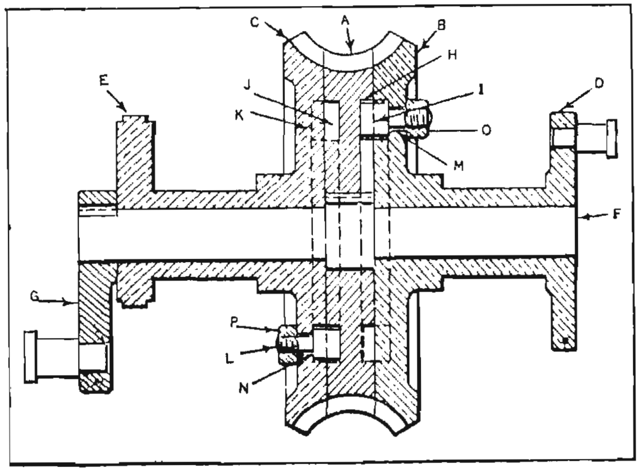


Fig. 11. Triple Worm-gear for Transmitting Three Intermittent Movements

the result is that the pawl is not turned by frictional resistance to the gripping position, and it simply makes an idle stroke. The cam G is pushed in when it strikes dog K and is suddenly thrown outward by a spring after passing the dog; this sudden release disengages pin H from its seat, into which it drops again upon the return of bar E. The number of feeding strokes before an idle stroke are governed by the ratio of the change-gears.

Triple Intermittent Worm-Gear. — The mechanism here described was part of a barbed wire fence-making machine. While the accompanying drawing of the device (Fig. 11) is

not made to any scale, the description will explain the principle of the design. This device transmits three distinct movements, all of which can have a dwell of different length, and this dwell can be varied in time to at least 180 degrees of the cycle.

The worm-gear is made up on three sections, A, B, and C. Section A is keyed to shaft F; section B revolves on shaft F and carries a disk crank D; section C also revolves on F and has an eccentric E. Shaft F has a crank G, and D, E, and G are the work levers. Worm section A has two concentric slots J and H on each side. In each of these slots there is fastened a stop (not shown). In sections B and C there are two concentric slots M and N cut long enough to meet the required adjustment of timing. Dog bolts L and I are held in the desired position by nuts O and P.

A section of the teeth long enough to make complete disengagement from the worm is cut from A, B, and C. We will assume that section C where the teeth are out is set central relative to the center of the worm; then section C would stay in this position until section A rotated enough to bring the stop in slot J into engagement with dog L. This action, of course, must be timed so that when engagement occurs, the worm teeth will register. By the varied settings of these two dog bolts, the timing of any section can be changed to meet the requirements of whatever machine is being designed.

Intermittent Rotary Motion Varied by Changing Cams. — Some novel features are embodied in the design of the intermittent motion mechanism shown in Fig. 12. Referring to the upper view, the shaft B revolves continuously, receiving its motion from one of the constantly rotating shafts of the machine on which it is employed. Another shaft C, imparts the intermittent movements obtained by the mechanism. With the cam pieces N and O, the shaft C makes ½ revolution, and then dwells while shaft B makes 7½ revolutions. Seven additional sets of cam pieces are provided, which can be used in place of those shown at N and O for obtaining different intermittent movements.

The driving shaft B runs in the bearings L and W. A spur gear V, pinned to the shaft B, meshes with the gear cut

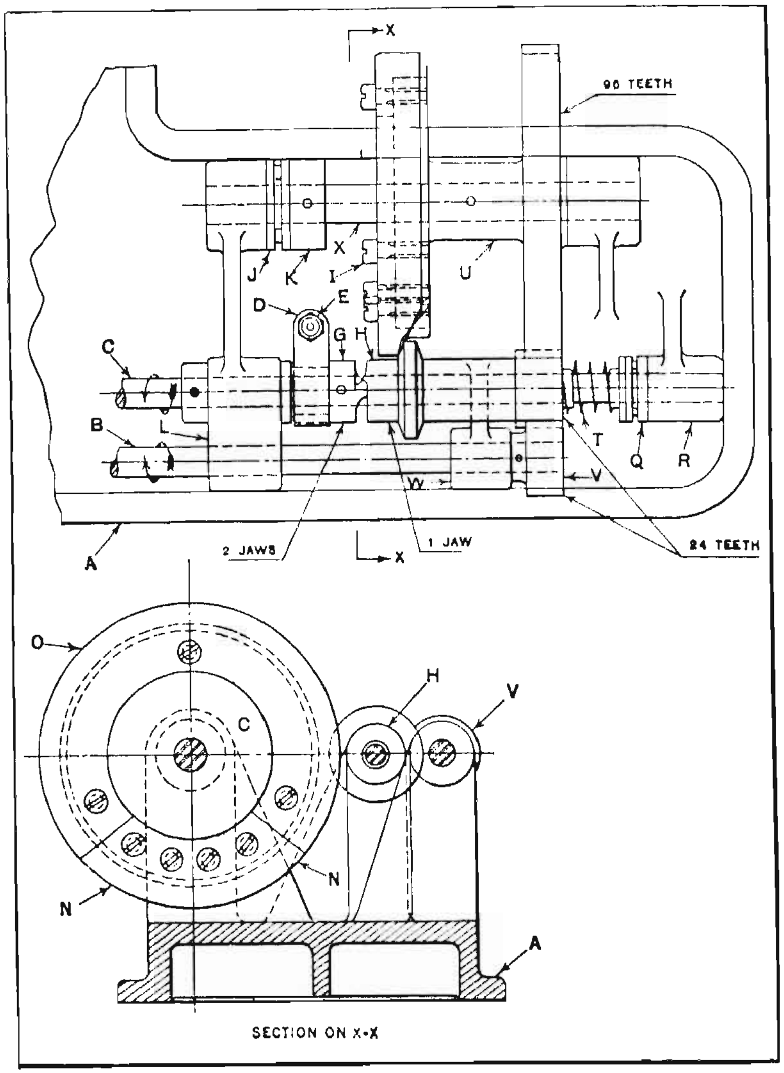


Fig. 12. Variable Intermittent Motion Obtained by Changing Cams.

integral with the sleeve H. The gear on sleeve H also meshes with a gear on the hub U. Hub U is fastened to shaft X by a pin, and has, on the opposite end from the large gear, a

turned disk on which the removable cam pieces N and O are mounted. These pieces are fastened to the disk by screws I. The collar K is pinned to shaft X, and the end thrust resulting from the cam action is taken by the ball thrust bearing J.

Sleeve H, which slides and also turns on shaft C, has a shoulder or collar which is beveled on both sides and which acts against the cam pieces N and O. Sleeve H has a single jaw machined on one end which engages one of the two jaws on member G. The member G, which is fastened to shaft C, is equipped with a leather-lined band brake D. The friction drag of the brake is controlled by adjusting nut E, which closes or spreads the band as required. The spring T acts against sleeve H, and forces the beveled shoulder against the cam, the resulting thrust being taken up by the ball thrust bearing Q.

As there are eight different intermittent motions to be transmitted, the eight sets of cams must all be of different lengths. When a shorter cam than the one shown at O is used, the two cam pieces N are slid back, so that they make contact with each end of the shorter cam, and are then fastened securely in place by the screws I. Additional tapped holes are provided in member U for fastening the new cam pieces in place. It will be noted that the pitch circles of the gears on hubs H and U and of the cam and the beveled shoulder are the same, so that nearly all sliding action between the cam surfaces is eliminated.

The operation of the mechanism may be explained as follows: Sleeve H is driven by shaft B, and makes the same number of revolutions per minute. Shaft X revolves one-fourth times as fast as shaft B and sleeve H. With the particular cam O shown in the illustration, shaft C will dwell or remain stationary while shaft B makes $3\frac{1}{2}$ revolutions, after which shaft C will make $\frac{1}{2}$ revolution, thus completing one cycle. Cam N simply slides sleeve H back, thus disengaging the clutch until the cam again moves around to the clutch-engaging position. The friction brake serves to stop the shaft C as soon as the clutch teeth are disengaged, and also takes up any lost motion that may occur.

Mechanism for Controlling Length of Rotating and of Idle Periods. — An intermittent motion was required to drive a shaft at the rate of one revolution per second and have provision for adjusting the length of the periods at which the shaft remained stationary. The first work for which the mechanism was employed required the shaft to make one revolution in one second and remain stationary for fifty-nine seconds. The second class of work required the shaft to make three revolutions in three seconds and then remain stationary for fifty-seven seconds, while a third operation required the shaft to make five revolutions in five seconds and remain stationary for fifty-five seconds.

The mechanism is shown in Fig. 13 as arranged to meet the first requirement. Upon the machine base *A* is mounted the bearing stand *R* which is bored out to a running fit for the small end of the worm *U*. Shaft *N*, in turn, is made a running fit inside the worm *U* and transmits the intermittent motion to the machine proper, which is not shown in the illustration. The clutch member *K* slides on shaft *N* and is prevented from turning by a key *L*. On the face of the large end of worm *U* is a clutch tooth which engages the tooth on member *K*. The worm meshes with the worm-gear *E*, imparting a continual and uniform motion to the cam-plate *F*. The cam-plate pushes the lever *C* outward and thus disengages the clutch at the required intervals. The spring P provides for the return of lever *C* and the engagement of clutch *K* at the end of the period in which the shaft *N* is required to remain stationary. The brake-drum *Y* is keyed to shaft *N*, and is equipped with a steel brake-band *M* having a leather-lined contact surface. The pressure of the brake-band on the drum is maintained by a spring *V*.

When the mechanism is in operation, the grooved pulley *S* is driven by a belt from the lineshaft at a speed of 60 revolutions per minute. This pulley rotates the worm *U*, causing the worm-gear *E* and the cam-plate *F* to make one revolution per minute. As the worm has a single thread, a point on the circumference of the cam-plate will revolve a distance *Z*, equal

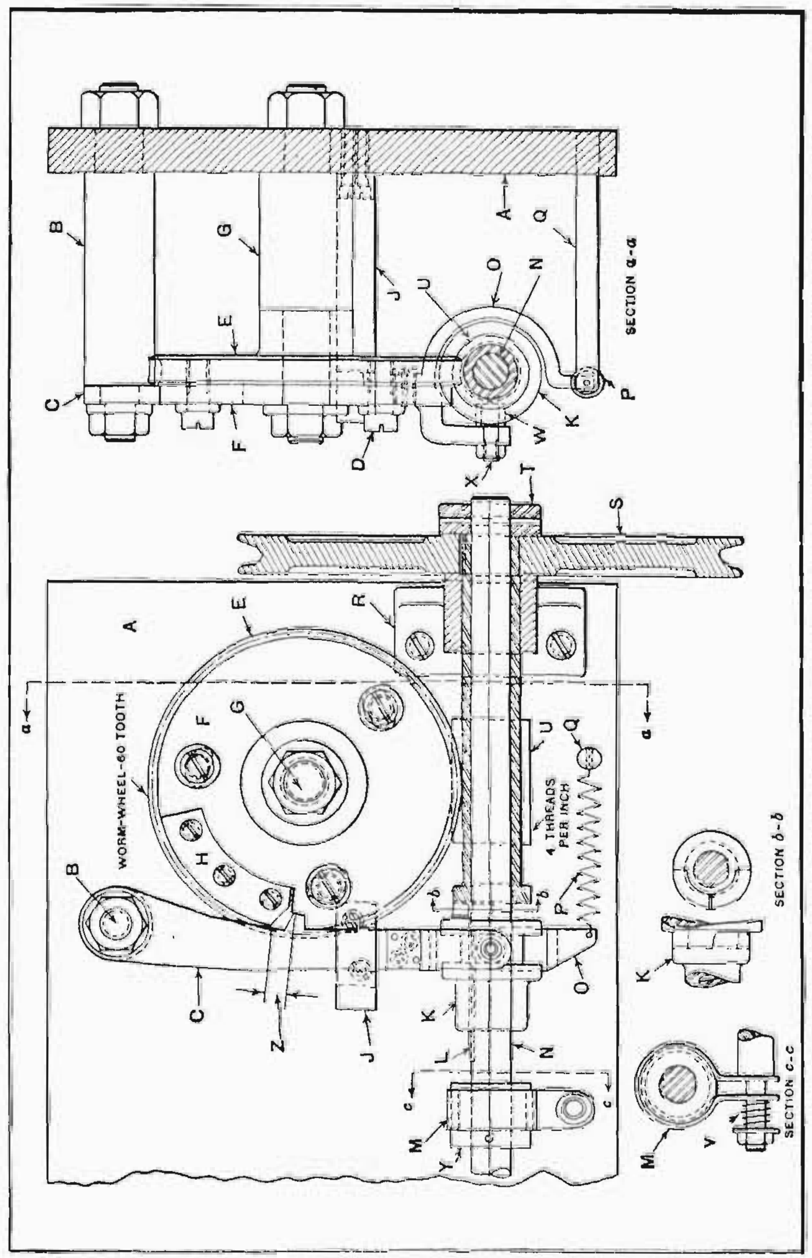


Fig. 13. Intermittent Mechanism which may be Arranged to Vary Length of Rotating and of Idle Periods

to the lead of the thread, or about ¼ inch, per revolution of the worm and of shaft *N*.

After the cam-plate has been properly adjusted, it will not require changing except to compensate for wear. At the instant the cam-plate has reached the end of the travel *Z*, the movement of lever *C* disengages the clutch *K*, so that the rotation of shaft *N* is stopped exactly at the end of one revolution. The friction brake consisting of drum *Y* and band *M* prevents the momentum of the clutch and shaft from causing a further movement of the shaft.

In order to obtain the necessary dwells required for the second and third operations, the distance *Z* must be increased. This necessitates the use of interchangeable cam-plates *F*. For the second job, the dimension *Z* is three times as great as for the first job, and for the third job five times as great. Any other desired length of dwell may be obtained by providing a suitable cam-plate.

Intermittent Motion which Automatically Increases and Decreases. — The mechanism described in the following is designed to turn a driven shaft through a small arc for every other revolution of the driving shaft and according to the following requirements: The feeding movement of the driven shaft is to increase by small amounts until a maximum feed is obtained; the feed then decreases to a minimum, again increases to a maximum, and at this point instantly begins at the minimum again. If a line is drawn representing these movements graphically, it will readily be seen that there are two periods of increasing feed and one period of decreasing feed for every cycle of movements. It was necessary to derive the feeding motion from a shaft running at twice the desired speed.

The principle upon which this mechanism operates is shown partly in diagrammatical form, in Fig. 14. The arm *C* carries a sliding block *D* which is connected to lever *L* by a pitman *K*. Block *D* is fed to or from the center of *B* by screw *E*, working in a divided nut. On the upper end of *C* are two intermittent motion star-wheels *F* and *G*, of six teeth each, with their

planes at right angles. Wheel *F* is pivoted on the side of *C*, and *G* is fastened onto *E*. Wheel *F* has three projecting pins F_1, F_2, and F_3, placed on alternate teeth and denoted in the illustration by black dots. Suppose star-wheel *F* is rotated one tooth (denoted by the straight lines). If that tooth is one of the three with the projecting pins on the side, wheel *G* and screw *E* will be rotated one-sixth revolution. Conversely, if the tooth on *F* has no projecting pin, *G* will not rotate. This

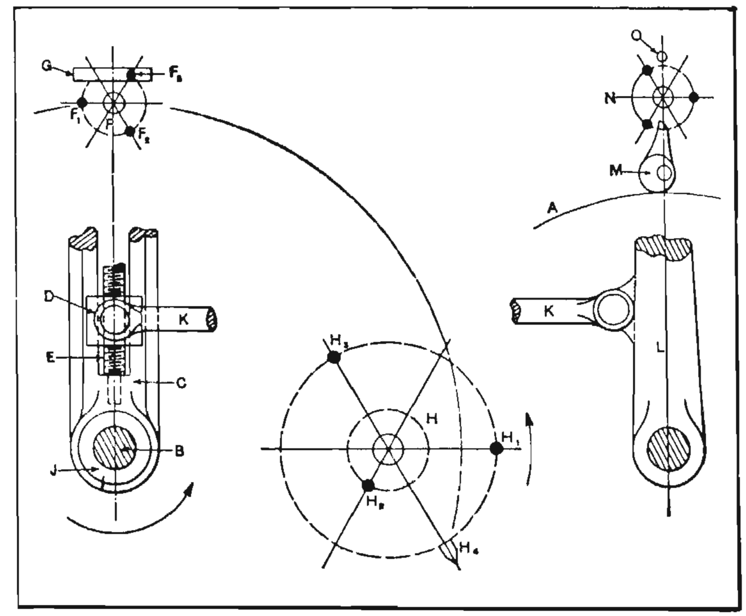


Fig. 14. Diagram of Mechanism for Automatically Increasing and Decreasing Intermittent Motion of Driven Shaft

method gives the alternating feed through *K* and *L*, as the arm *C* revolves about the center of *B*.

The center of *F* revolves about *B* in a path as shown by the broken line, which passes midway between the two circles at *H*. These circles represent the controlling star with six teeth or points (denoted by straight lines). A pin (not shown) upon the block *D* is caused to engage two teeth of *H*, on two successive revolutions before it is retracted. This is done only when the slide is at the upper end of its travel.

Star-wheel H merely operates star F. Three projecting points (denoted by black dots) engage point after point of F as it comes around. Suppose it is projections H_1 or H_3 that engage with F. Then, by reason of their being on the outer side of the center of F, star-wheel F must revolve in an opposite direction to that in which it would revolve if H_2 were the projection engaging F, because H_2 is on the inner side of the center path of F.

It is now apparent that block D moves up (or down) through the action of H upon F and F upon G; and the motion takes place only for every other revolution of B. When block D reaches the outer limit, the pin upon the block is released as mentioned, which, in turn, revolves H two teeth before it is retracted, thereby engaging the opposite side of F, reversing the direction of rotation of F and G, and returning D toward the center. Pin H_3 is now brought into action and D goes outward again to the extreme position. The controlling star is rotated as before, returning H_1 to position and bringing H_4 into action to open the split nut at screw E, which allows a spiral spring in J to return the block D and pitman K instantly to the center, thus completing one cycle. Through link K, lever L, and pawl M, ratchet A is rotated. A star N is carried by L, which is operated by the hinge tappet O and which is provided with three projecting pins to lift the eccentric pawl from the ratchet every alternate stroke of L.

Constant Intermittent Rotary Motion from Variable Rotary Motion. — The feeding movement of a planer tool, which occurs at the end of each return stroke, is derived from a shaft which revolves in first one direction and then the other, the number of revolutions depending upon the length of the stroke which is adjusted to suit the work. The simple mechanism to be described makes it possible to obtain the same rotary movement for operating the feed-screw of the tool-slide, regardless of the number of revolutions made by the shaft which drives the feeding mechanism. A crank at the end of the driving shaft turns part of a revolution and then remains stationary while the shaft continues to revolve.

One method of securing this fractional part of a turn and then stopping the motion of the feed disk is illustrated at A in Fig. 15. The link f connects the crankpin of the feed disk with a rack which, through suitable gearing, transmits motion to the feed-screw. The main pinion shaft of the gear train for driving the planer table has attached to its end the cup-shaped casting a, which forms one part of a friction clutch. The crank disk b has a hub c, which fits into the tapering seat in part a and forms the other member of the clutch. If this

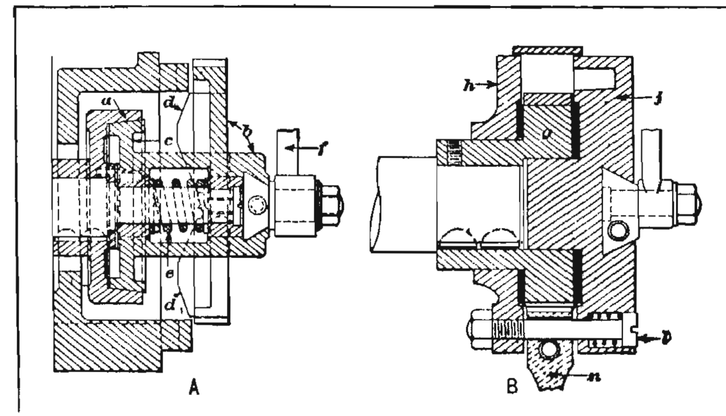


Fig. 15. Mechanisms for Deriving an Unvarying Rotary Movement from a Driving Shaft regardless of the Number of Revolutions Made by the Shaft

friction clutch is engaged when the planer is started, the crank disk b revolves until one of the tapered projections or cam surfaces d strikes a stationary taper lug, thus forcing part c out of engagement with a against the tension of spring e. The crank disk then remains stationary until part a and the driving shaft reverse their direction of rotation at the end of the stroke. This reversal of motion disengages the tapering surface d or d_1, as the case may be, and allows the friction clutch to reëngage; the crank disk is then turned in the opposite direction, until the other tapering projection strikes a second lug which again stops the motion of the feed disk. This intermittent action in first one direction and then the other is continued as long as the planer is in operation, and the feed disk oscillates through the same arc regardless of the length of stroke or the number of revolutions made by the driving shaft; consequently the feeding movement of the tool will not be varied by a change in the length of the stroke.

Another planer feed mechanism which operates on the same general principle as the one just described is illustrated at *B*, Fig. 15. In this case, the hub *g* is keyed to the shaft and the flange formed on this hub is between plates *h* and *j*. This flange does not come directly into contact with the plates, as there are leather washers on each side as indicated by the

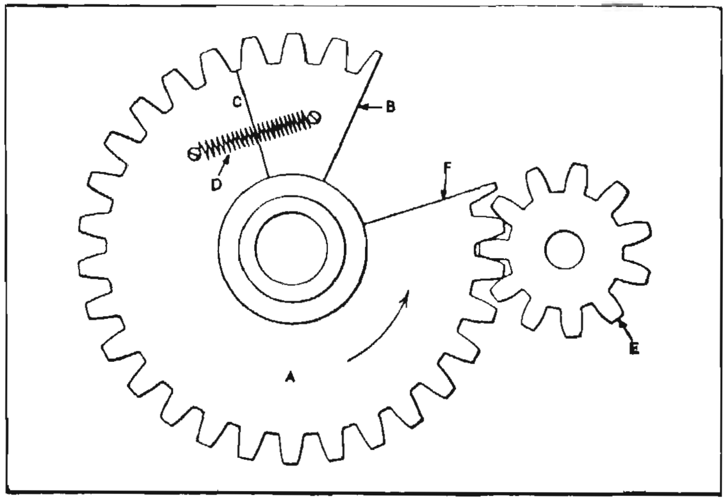


Fig. 16. Intermittent Motion Derived from a Swinging Gear Sector

heavy black lines. The plates *h* and *j* are held in contact with these washers by three bolts *l* having springs under the heads. The hub *g* is surrounded by a band which is split on the lower side and has lugs *n* into which is fitted a pawl of such shape that, when it strikes a fixed stop, the band is opened and released from the hub. This releasing of the band occurs after the crank disk has turned far enough to give the necessary feeding movement. The crank is held in position while the driving shaft continues to revolve, by the friction between plates *h* and *j* and the leather washers previously referred to. When reversal occurs at the end of the stroke, the hub *g* re-

volves in the opposite direction and the band again grips it until the pawl of lug *n* strikes the opposite stop.

Intermittent Gear with Swinging Sector. — The gearing illustrated in Fig. 16 has one period of rest for each revolution of the driver which has a sector *B* that is free to swing in the space provided for it, but is normally held in the position shown by a spiral spring *D*. The driver revolves at a uniform speed in the direction shown by the arrow and, when the sector *B* comes into engagement with the driven gear, the latter stops revolving while the sector is swinging across the open space or until side *B* strikes side *F*, when the driven gear is again set in motion. As soon as the sector is released by

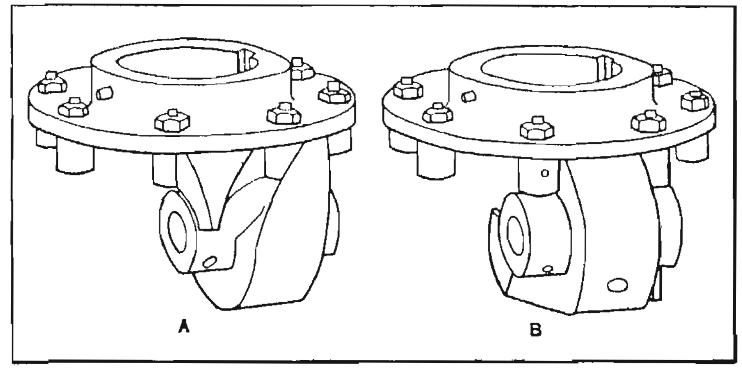


Fig. 17. Intermittent Gearing for Shafts at Right Angles

the driven gear, the spring draws it back to the position shown in the illustration, preparatory to again arresting the movement *E*. The resistance to motion offered by gear *E* should be great enough to overcome the tension of spring *D*, as otherwise the sector would not swing away from the position shown. In order to avoid shocks, this gearing would have to be revolved quite slowly; while the design is not to be recommended, the principle may be of some practical value.

Intermittent Gearing for Shafts at Right Angles. — When driving and driven shafts are at right angles to each other, intermittent gears which are similar to bevel gears in form, but constructed on the same general principle as the spur gearing illustrated in Fig. 1, may be employed. The smooth or blank space on the driving gear for arresting the motion of the driven member corresponds to the pitch cone and engages concave locking surfaces formed on the driven gear. Owing to the conical shape, such gearing is more difficult to construct than the spur-gear type.

A form of intermittent gearing for shafts at right angles to each other but not lying in the same plane is illustrated in Fig. 17. The driving member is in the form of a cylindrical cam and has a groove which engages, successively, the rollers on the driven wheel. Diagram *A* shows the cam in the driving or operating position, and at *B* the driven wheel is shown

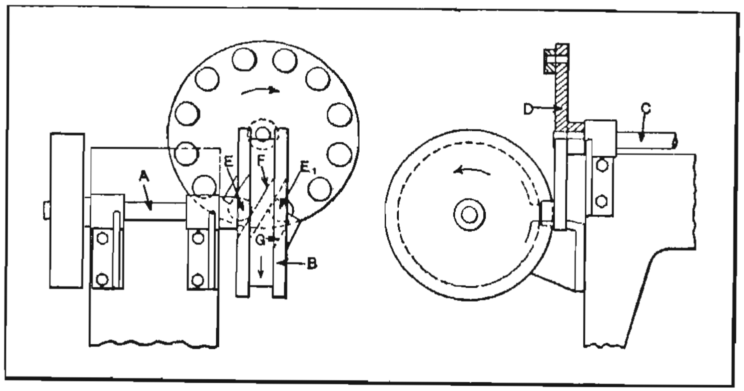


Fig. 18. A Modification of the Type of Gearing Shown in Fig. 17

locked against rotation during the period of rest. The locking action is obtained by parallel faces on the cam which fit closely between the rollers and are located in planes at right angles to the axis of rotation. This mechanism was designed for a high-speed automatic machine requiring an accurate indexing movement and a positive locking of the driven member during the stationary period. The gearing operated successfully at a speed of 350 revolutions per minute, and it was because of the speed that this design was used in preference to the Geneva-wheel type of gearing previously described. The curvature of the operating groove on the driving cam is such that the driven wheel is started slowly and, after the speed is accelerated, there is a gradual reduction of velocity. The driven wheel has no lost motion for any position and the mechanism operates without appreciable shock or vibration, and is practically noiseless.

Another form of intermittent drive for shafts located at right angles but not lying in the same plane is illustrated in Fig. 18. This mechanism operates on the same general principle as the one just described, but differs in regard to the form of the driving member or cam. This cam *B* is attached to the end of the driving shaft *A* and has an annular groove corresponding in width to the diameter of the rollers on the driven wheel *D* carried by shaft *C*. This annular groove is not continuous as there are inclined openings on both sides. When the cam revolves in the direction indicated by the arrow, the inclined surface *F* pushes roller *E* over to the left, thus causing disk *D* to turn; at the same time, roller E_1 enters the opening on the opposite side and is pushed over to the central position by cam surface *G*. This roller E_1 remains in the groove until the cam has made one revolution, thus locking the driven wheel against rotation. This locking roller then passes out at the opposite side and another roller is engaged by the groove. The ratio of this gearing, which was used to provide a feeding movement on an automatic machine, depends upon the number of rollers on the driven wheel.

Locking Plates for Intermittent Gears.—In the design of intermittent gearing, it is essential that the driven gear be provided with some form of locking device to prevent it from moving during a period of rest. If this provision were not made, the driven gear might not be in the correct position to mesh with the first tooth of the driver when the two gears come into contact. Various devices are used for this purpose. The simplest form of locking circle is formed by milling the blank space of the driving gear down to the pitch line as shown at *a* (Diagram A, Fig. 19). During each period of rest of the driven gear the locking circle on the driver rotates in one of the stops *b* on the driven gear and, of course, does not transmit any motion to that gear. In applications where the driven gear acts as an idler, driving a third gear and consequently containing a complete set of teeth, it is impractical to make use of a locking circle of the type just mentioned. In cases of this kind, a locking circle may be employed which is in a different plane from that of the gear teeth and which may be in the form either of a plate riveted to the gear or a flange cast integral with it. In these designs, the locking circle should not interfere with the action of the teeth.

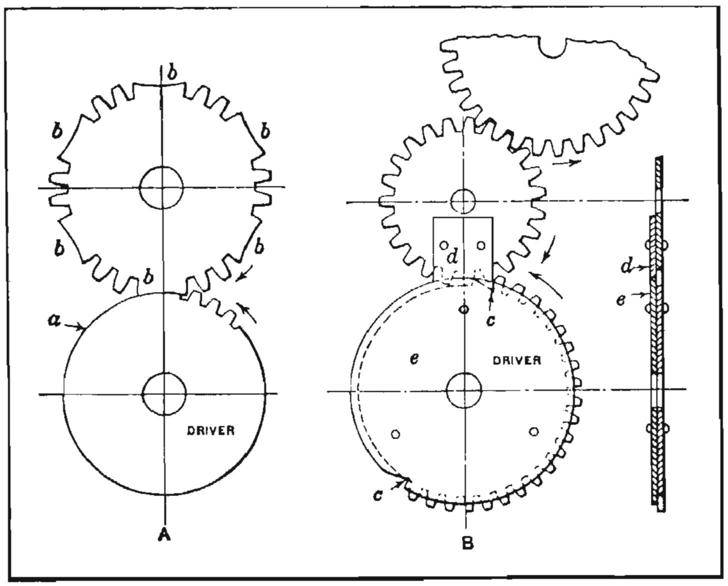


Fig. 19. Application of Locking Plates to Intermittent Gearing

The form of locking device consisting of plates riveted to the gears is used on light gearing, principally where the gears themselves are stampings. This construction is shown at *B*. When such plates are made use of, notches, as shown at *c*, must be provided in the locking circle in order to permit the corners of the plates which form the stops *d* to pass the plate *e* on the driving gear at the beginning and ending of each period of rest. The shape of this notch may be readily obtained by laying out the gears when in the respective positions of entering and leaving the periods of rest. Such notches are not

necessary on gears of the type shown at *A*, the spaces between the teeth being such as to take care of this condition.

The form of locking circle, or ring, which is cast on the gears is the type which is used on heavier construction. The ring is cast integral with the driving gear. The stops are cast integral with the driven gear, and should be designed to allow sufficient cutter clearance, at the ends of the gear teeth for cutting the gear teeth. Notches are also provided in this type of ring to permit the stops to enter and leave just as described for the riveted-plate type of locking device.

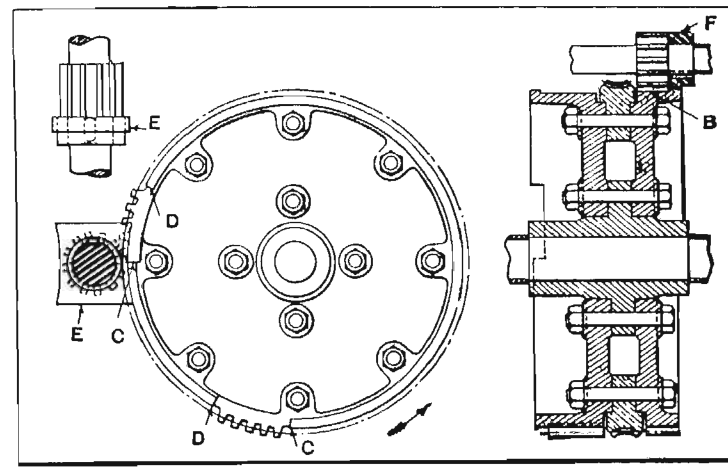


Fig. 20. Intermittent Gears with Locking Circles

Locking Circles Cast Integral with Intermittent Gears.— In Fig. 20 is shown a design for intermittent gears which is suitable for either light or heavy construction. An intermittent gear is bolted to each side of the worm-wheel and hub, one gear providing motion for indexing while the other operates a clutch mechanism. The portions of the rims which have no teeth are cast to a smaller diameter than the root diameter of the teeth, so as to allow sufficient clearance for the pinion teeth while the intermittent spaces on the gear rotate past the pinion. This clearance may be plainly seen at *B*. The locking circle of each gear is cast integral with the gear, but is broken between the points C and D where the pinion starts and stops rotating. It is not necessary for the diameter of the locking circle to be equal to the pitch diameter of the gear; it may be made to any convenient dimension, but it is well to make this diameter as large as possible, thereby reducing any possible binding on the locking plate. The cutting of the gear teeth presents no difficulties, but it is sometimes necessary to reduce the addendum of the teeth adjacent to points

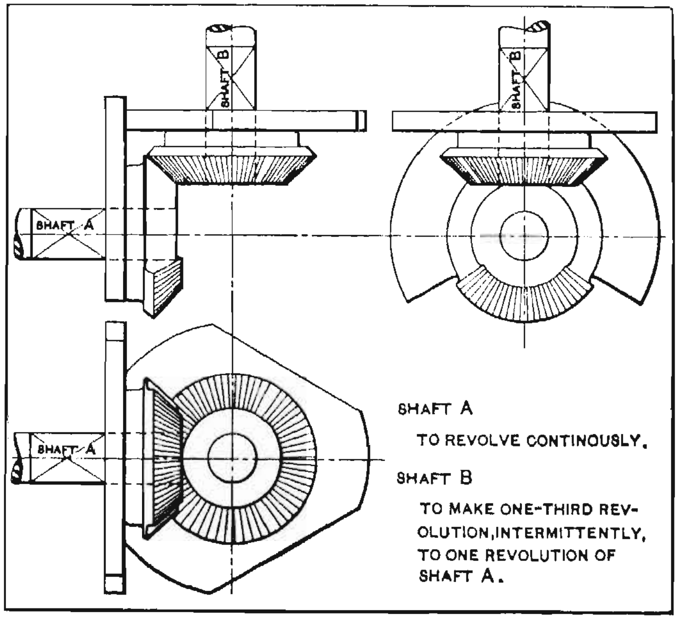


Fig. 21. Intermittent Bevel Gears Provided with Auxiliary Locking Device

C and D so that they will enter and leave the pinion teeth without interference. The locking plates E and F are secured to the pinion shafts at the sides of the pinions by means of feather keys. Gears of this nature run smoothly, and as their manufacture offers no difficulties to the machine shop they might well be used more extensively in the design of automatic machinery.

Auxiliary Locking Device for Intermittent Bevel Gears. — The intermittent bevel gearing illustrated in Fig. 21 is provided with auxiliary locking plates which regulate the motion of the driven gear and hold it stationary while disengaged from the driver. The driving gear is on shaft A and revolves continuously. It is only provided with enough teeth to rotate the driven gear and shaft B one-third revolution to one complete revolution of the driver. This mechanism is used to actuate feeding rolls requiring an intermittent motion. Formerly the gearing was used without the locking device to be described, but there were slight variations in the movements of the driven shaft so that the gears did not always mesh

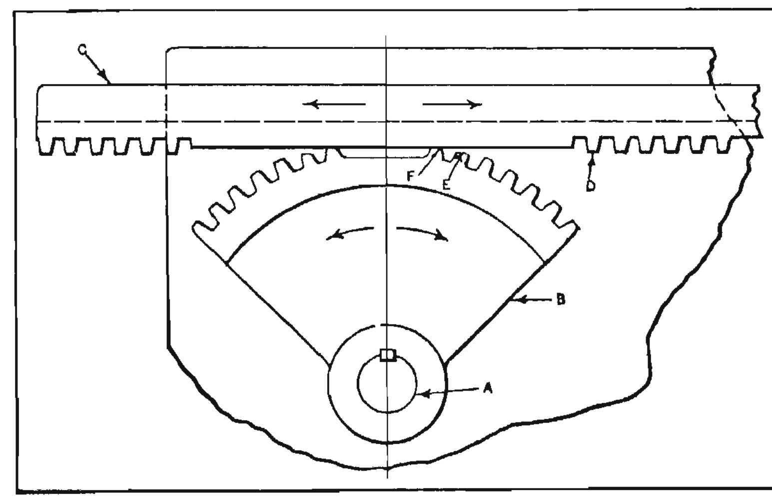


Fig. 22. Reciprocating Rack which Engages Segment Gear at Each End of Its Stroke

correctly, which caused them to break, and also interfered with the timing of the feeding movement. These defects were eliminated by applying locking plates to the shafts A and B, one plate being located just back of each gear. The plate on shaft B has three equally spaced flat sides or edges and the plate on shaft A is cut away to provide a clearance space for the protruding sections of the plate on shaft B when this shaft is in motion. As the plan view shows, the flat side of the plate on shaft B, during the idle period, is intercepted by the plate on A so that the driven shaft is not only locked but its motion is limited to one-third revolution for each complete turn of the driving shaft.

Intermittent Motion from Reciprocating Rack. — The intermittent motion to be described is part of a device for feeding brass shells to a dial press. In the operation of this feed mechanism, it was necessary to turn shaft *A* (Fig. 22) and segment gear *B* through part of a revolution at each end of the stroke of reciprocating rack *C*. The illustration shows the segment gear in the dwelling position. As the rack moves, say, to the left, the rack teeth beginning at *D* engage the segment gear and turn it. When the rack reverses, the segment gear is turned in the reverse direction until the rack teeth at the right leave it, and then dwell occurs until the teeth on the left-hand side engage the segment gear. Tooth *E* (and the corresponding tooth on the opposite side) is cut away to avoid interference as *D* comes into contact with *F*.

The "Beaver-tail" Stop Mechanism. — The "beaver-tail" stop mechanism is used in conjunction with a geared drive to prevent or minimize inertia shock or impact at some point in a repeated cycle where a clutch is thrown or tools are brought into contact with each other or with the work as in power press operation. The name "beaver-tail" is applied to this mechanism because of the shape of the cam which forms an important part of it. The driving pinion *A* (See diagram, Fig. 23) revolves continuously, and drives gear *B* through ordinary gear teeth except when the "beaver-tail" mechanism comes into action, at which time the motion of gear *B* is controlled by the two rollers *R* and R_1 and the "beaver-tail" cam located between the rollers. If driven gear *B* is to be stopped once during each revolution, only one cam is attached to it. If two stops per revolution are required, two cams located 180 degrees apart, are used. The teeth of the driven gear are cut away at each stopping position, and the large developed tooth or cam takes their place.

The rollers on the driving pinion are diametrically opposite each other, and their centers are on the pitch circle of the pinion. When the beginning of the blank space on gear

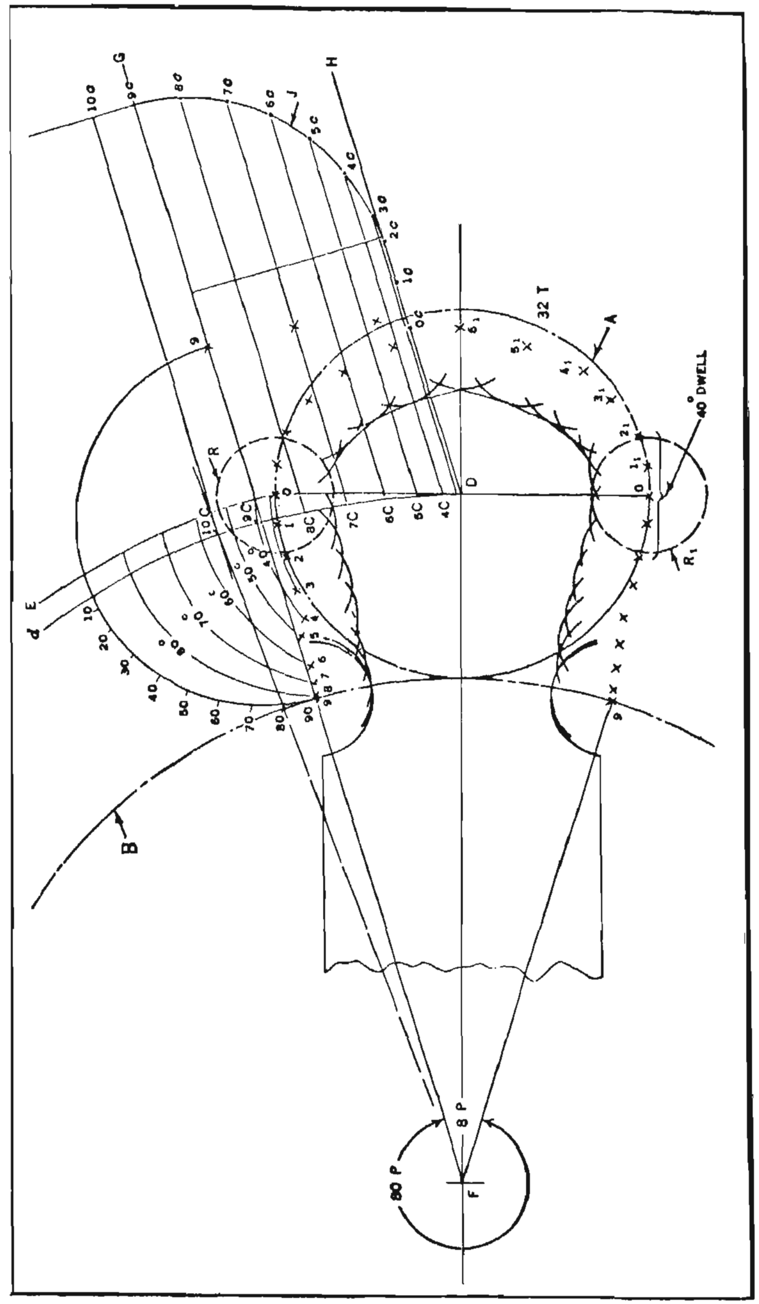


Fig. 23. Development of a "Beaver-tail" Cam for a 3 to 1 Gear Ratio and a Dwell Equal to About 40 Degrees of Pinion Rotation

B reaches the pinion and during the next quarter revolution of the pinion, one roller moves along the "beaver-tail" cam and brings the gear to rest with a harmonic motion. The center of the roller at the point of engagement coincides with the point of tangency of the two pitch circles, so that engagement takes place without shock. The driven gear is locked during the brief dwell which occurs while the rollers are revolving about a concentric part of the cam. This stationary or neutral position is shown by the diagram. After the dwell, the other roller, during a quarter revolution of the pinion, engages the cam and accelerates the gear until it has the same speed as the pinion, when the gear teeth mesh and the ordinary gear drive is resumed.

There may be one, two, or more stops per revolution, depending upon requirements and the number of stopping positions provided. Both stopping and starting are accomplished with harmonic deceleration and acceleration, so that there is no shock to the mechanism (except from possible backlash) due to the reversal of strains. While the deceleration is taking place, the driven member acts temporarily as the driver, returning energy to the drive as the revolving parts are brought to rest. During a brief dwell period of twenty or thirty degrees of driver rotation, while the cam is in the neutral position, a clutch on the driving shaft may be thrown in or out without inertia or load effect from the driven parts. As the load picks up, the driver has temporarily a greater mechanical advantage, due to the position of the roll on the cam, than it normally would have because of the ratio of the gears.

Application of "Beaver-tail" Stop to Power Presses. — The "beaver-tail" stop is an invention of Charles R. Gabriel. It was developed about twenty-five years ago for use in connection with the feed mechanism of gear-cutting machines built by the Brown & Sharpe Mfg. Co. During the last five or six years this stop mechanism has been utilized very successfully by the E. W. Bliss Co. in conjunction with the drives of power presses designed for special purposes.

For example, a small press used for curing celluloid is arranged with a double "beaver-tail" stop to permit disengaging and engaging the clutch at the bottom of the stroke while the press is under full load. This stop mechanism has also been used on larger machines having shafts up to 5- or 6-inch sizes. Machines equipped with this stop are used either for curing celluloid or for other products which must be held under pressure for a period of time. The rolling key clutch is used on the back-shaft or driving shaft, and since the clutch is disengaged and engaged as the rollers are passing across the dwell or neutral position on the cam, it is protected from an otherwise unduly severe strain. Presses are sometimes built for this service without using the "beaver-tail" cam, but if the load is heavy, the clutch life is likely to be short.

A rather different use of the motion is its application for removing the impact load from the tools as on a press used for the extrusion of collapsible tin tubes. The "beaver-tail" is so placed that it brings the slide and punch to rest just as contact is made with the slug or blank to be extruded. The extrusion then takes place with an easy accelerated motion, the whole action resulting in various manufacturing advantages and an enormously greater tool life.

With the clutch on the back-shaft, the driven gear must be fitted with two "beaver-tails," one at the point of contact of the tools, and the other opposite it, to favor the clutch in stopping the press. A variation of this practice is to place the clutch in the gear on the main shaft and use only one cam at the tool contact position. This cam need not have any allowance for dwell.

Size of Driven Gear Used with "Beaver-tail" Stop. — The pitch circle velocity and travel of the gear and pinion are the same except during the slowing down and picking up periods, when the pinion moves half its circumference and the gear moves a much shorter distance. This difference in travel reduces the number of teeth or pitches on the gear (including those teeth that are actually cut away) and hence also reduces the gear diameter.

As an example to illustrate the procedure, assume that a pinion makes four revolutions to one of the gear. In this case, if the pinion has 32 teeth, there would normally be 128 gear teeth ($4 \times 32 = 128$). If in the case selected the gear moves only the space of ten teeth during the slowing down and picking up periods, while the pinion moves a half revolution or sixteen teeth, then the difference ($16 - 10 = 6$) must be deducted, leaving 122 teeth or pitches as the gear circumference. The pitch diameter of the gear relative to the pinion is in the ratio of 122:32.

Profile of "Beaver-tail" Cam. — The construction or development of this motion is worked out in Fig. 23. The method is based upon keeping the gear stationary and revolving the pinion about it as in planetary gearing, determining the relative progress of the center of the pinion (along the arc *Dd*) and then plotting the corresponding positions of the rolls which thus outline the proper shape of the cam.

The pitch circles of the pinion and gear are indicated at *A* and *B*. The rollers *R* and R_1 are shown in the neutral or locking position 0-0 which is their position when the center of the pinion coincides with the center line of the cam at *D*. This is one of the limiting positions. The other is either of the points 9, when the two rolls and the center of the pinion are on a radial line from the center of the gear. Here one roll center coincides with the point of tangency between the pitch circles of the gear and pinion, and the driving action shifts from the rolls to the gear teeth (or vice versa).

The spacing of points 0-0 depends upon the pinion, which we have assumed in this case to have 32 teeth (the number must be even) and, say, a 4-inch pitch diameter. The pinion moves through just half a revolution from the point where one roll engages the cam to the point where the other roll leaves it on the other side, the movement being equivalent to 16 teeth. The gear obviously moves a shorter distance, which we have assumed in this case to be 8 pitches. (A greater distance reduces the dwell but eases the accelerating or decelerating action.) This is 8 pitches less than the pinion movement. Accordingly, the pitch circumference or number of teeth of the gear for a 3:1 ratio is $32 \times 3 = 96 - 8 = 88$. The gear diameter equals $(4 \times 88) \div 32 = 11$ inches.

The order of construction, after determining the pitch diameters of the gear and pinion is as follows: Lay off the center line *FD*, the pinion pitch circle *A* about the center *D*, and the gear pitch circle *B* tangential to it, about the center *F*. Locate the roll centers 0-0 in their neutral position on the intersections of the pinion pitch circle and the vertical center line through *D*.

Locate the starting and finishing positions of the roll centers 9-9 on the pitch circle *B* of the gear—in this case, 8 pitches apart or 4 pitches each side of the center line *FD*. Draw the radial line *FG* from the center of the gear through one of the points 9. The centers of the pinion and both rolls must lie on this line when the roll *R* is at 9. Draw the arc *Dd* and the arc 0*E* from the center *F* and the points *D* and 0.

Following the scheme of revolving the pinion about the gear to develop the shape of the cam, the center of the pinion for various positions of the roll *R* must move along the arc *Dd*. When the rolls are at the locking or stop position 0-0, the pinion center is at *D*. When the roll *R* is at the point 9, the pinion center is at 9*C*, the intersection of the arc *Dd* and the line *FG*. Here the gearing is in normal mesh and the gear and pinion are moving relatively at full speed. Since the pinion moves through a quarter revolution (90 degrees) in accomplishing this change of speed, we have taken nine positions for the roll, representing a change of 10 degrees each on the pitch circle of the pinion. The corresponding positions of the center of the pinion must be on the arc *Dd* between 9*C* and *D*, and should be spaced to give harmonic deceleration. To get a starting point for this harmonic change, lay off from 9 on the gear circle *B* a distance equal to 10 degrees on the pinion pitch circle. (Note the small arc at the division marked 80). A radial line from the gear center *F* through this point, intercepts the arc *Dd* at the point 10*C*. Then the distance 10*C*-9*C* represents the travel of the center of the pinion for 10 degrees of rotation (direct-geared, full speed).

To locate the pinion center positions below $9C$ for harmonic deceleration, the following approximate construction has been used: Draw the line DH from D parallel to FG. Swing the arc J from a center on FG, tangent to DH. From the point $9c$ at the intersection of the arc J and the line FG, lay off the distance $10c$-$9c$ about the arc J, continuing along the line DH. Transfer the points $8c$, $7c$, etc., on arc J, by means of lines parallel to FG, to the arc Dd. This gives the pinion center positions $8C$, $7C$, etc. Since the points $0c$, $1c$, and $2c$ all fall on the point D, there is a relative dwell in the motion during the roll positions 0, 1, and 2.

To locate the roll center positions, swing an arc equal to the pitch radius from the proper center and measure off the corresponding number of degrees from the arc $0E$ as a starting point. Thus the points 1 and 2 are on the radius from the center D, and are 10 and 20 degrees, respectively, from 0. The point 5, for example, is on a pinion radius swung from the center $5C$, and is measured off 50 degrees from the intersection of that radius with the arc $0E$. This gives the positions of the center of the roll R between the points 0 and 9. Corresponding positions of the roll R_1 may be obtained at the same time, as they are at the other extreme of the pinion diameter in each case. Thus the point 5_1 is at the intersection of the diameter line through the points 5 and $5C$ and the pinion circle arc swung from the center $5C$. The remaining positions may be obtained by repeating the construction on the other side of the center line FD, or since the cam is symmetrical, by transferring the points 0 to 9 and 0 to 6_1 across the center line FD on perpendicular lines to the corresponding points on the other side of it. The diameter of the rolls R and R_1 is determined to suit the case, and the profile of the cam is outlined, as shown, by drawing in a portion of the roll circumference, swung from each of the roll center positions.

Note that the dwell is about 40 degrees; that is, in laying out the uniform distance $10c$-$9c$ around the arc $9c$-D, the

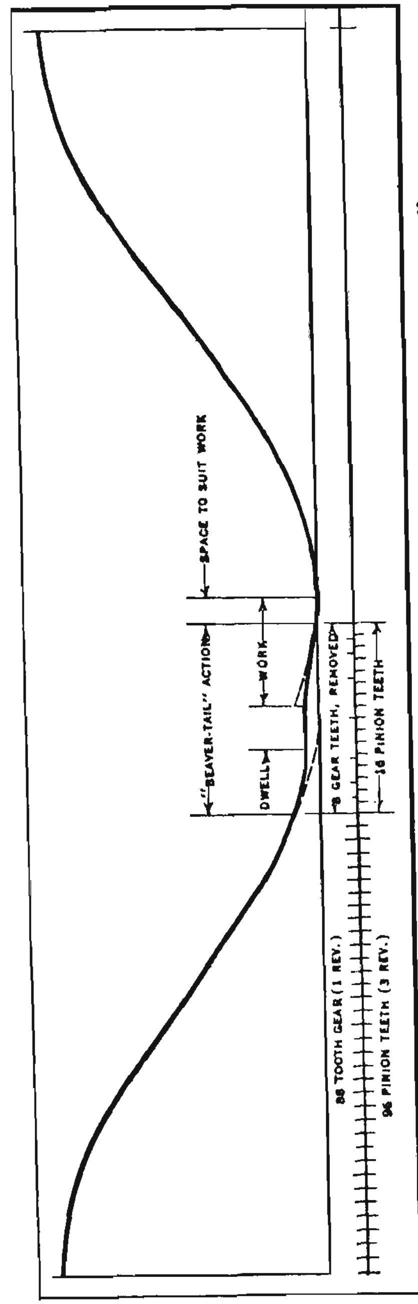


Fig. 24. Modified Crank Motion of an Extrusion Press Using the "Beaver-tail" Shown in Fig. 23

points $2c$, $1c$, and $0c$ all fall on the neutral center D, so that rotation of the pinion through the corresponding positions does not result in any motion relative to the cam. If it is desired to reduce this dwell, the number of pitch spaces between the points 9-9, assumed in this case as eight pitches, should be increased. For convenience in gear-cutting, this distance should always be a whole number of pitches.

The action resulting from the use of the "beaver-tail" cam in the drive of a crank press and the modification of the crank motion is shown by the curve in Fig. 24. This is based upon the design shown in Fig. 23. The motion of the pinion, which has uniform speed, is used as the time basis for the curve. The spacing of the crank is either fixed or adjustable to suit the work, which in this case may be assumed to be extrusion. Note that a close approximation to uniform motion during the working period results from the combination of harmonic acceleration of the "beaver-tail" and harmonic deceleration of the crank.

Automatic Indexing Mechanism. — The indexing or dividing of circular work requiring equally spaced grooves milled

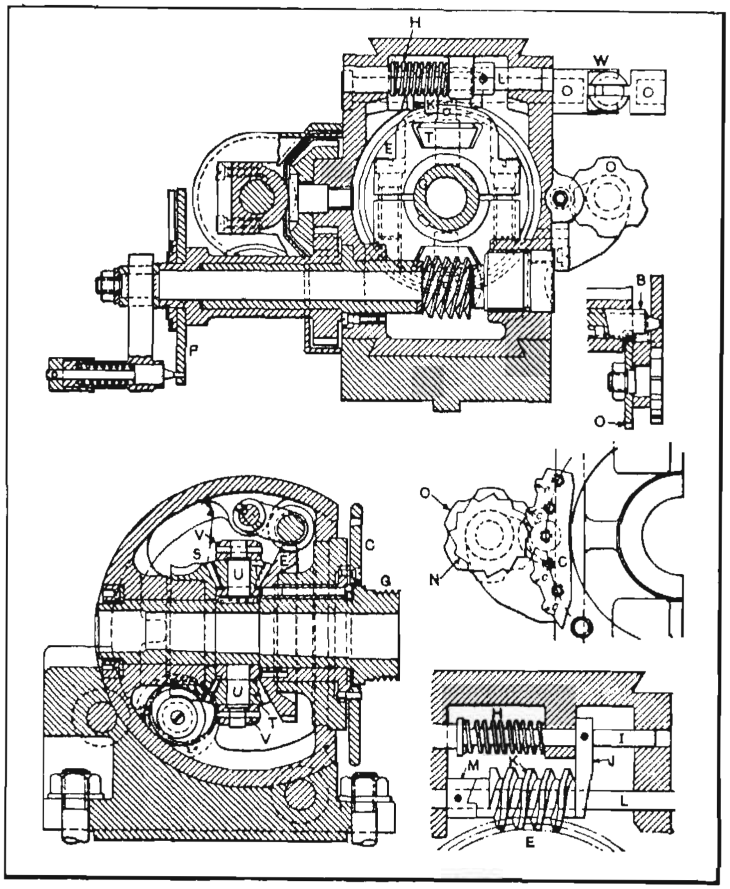


Fig. 25. Automatic Indexing or Dividing Mechanism

across the periphery may be controlled automatically by the dividing-head illustrated in Fig. 25. In addition to the transverse and longitudinal sections shown, there are three detail views which illustrate important features. The mechanism for controlling the indexing automatically derives its motion

from a spindle *L* driven through coupling *W* from a special pulley carried on a bracket attached to the bed of the milling machine. The clutch *M* (see detail view) on spindle *L* locks worm *K* to the spindle when the worm is pressed against the clutch *M* by a spring *H*, acting through rod *I* and finger *J*. This engagement with clutch *M* occurs when lock-bolt *B* is withdrawn from plate *C*, so that worm-wheel *E* is free to revolve. The movement of plate *C* at each indexing is controlled by a counter mechanism consisting of a dividing-plate *C* having teeth on the periphery which engage the teeth of disk *N*, thus rotating stop-plate *O* which controls the engagement of lock-bolt *B* and the extent of the indexing movement.

The table of the milling machine on which this mechanism is used should be arranged to return automatically. When one groove has been milled across the work, the table returns and, when near the end of the return stroke, lock-bolt *B* is withdrawn by a suitable mechanism (not shown in the illustration). When this bolt is disengaged from dividing-plate *C*, the worm-wheel *E* is free to revolve. The pressure of spring *H* forces rod *I*, finger *J*, and worm *K* to the left, the worm engaging clutch *M* on spindle *L* which is constantly revolving. As worm *K* and worm-wheel *E* revolve, rotary motion is transmitted to dividing-plate *C*, and also to spindle *G* through epicyclic gearing consisting of bevel pinions *T* mounted on pins *U* attached to part *V* which is keyed to spindle *G*. The indexing movement continues until bolt *B* enters one of the succeeding holes in plate *C*. The movement of worm-wheel *E* is then arrested and the worm, as it continues to revolve, disengages itself from clutch *M* and stops rotating. The dividing-plate *C* has a number of teeth *c*, the number corresponding to the number of its holes. Whenever this plate is set in motion, these teeth engage disk *N* and turn the counter *O*. A solid portion of this counter is thus placed in front of lock-bolt *B*, which prevents the bolt from reëngaging with plate *C* until a rotation equal to the required number of holes has been completed. One of the concave notches in counter *O* then releases bolt *B* which engages plate *C*. The number of teeth in plates C and N and the notches in counter O depend upon the number of divisions required. This dividing-head may be used the same as the hand-operated design. The hand-operated indexing movements, as well as the automatic movements, are transmitted to spindle G through the train of epicyclic gearing previously referred to.

Combined Indexing and Locking Mechanism. — The automatic indexing and locking mechanism illustrated in Fig. 26 was designed for a multiple-spindle automatic screw machine. The motions of this machine are all controlled by cams on a

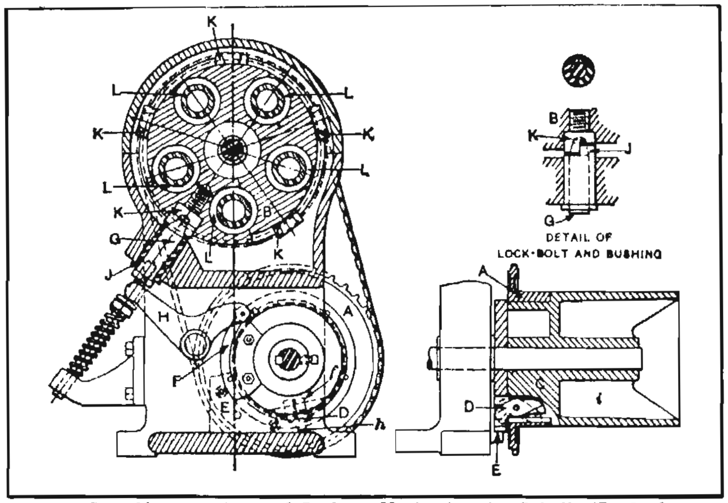
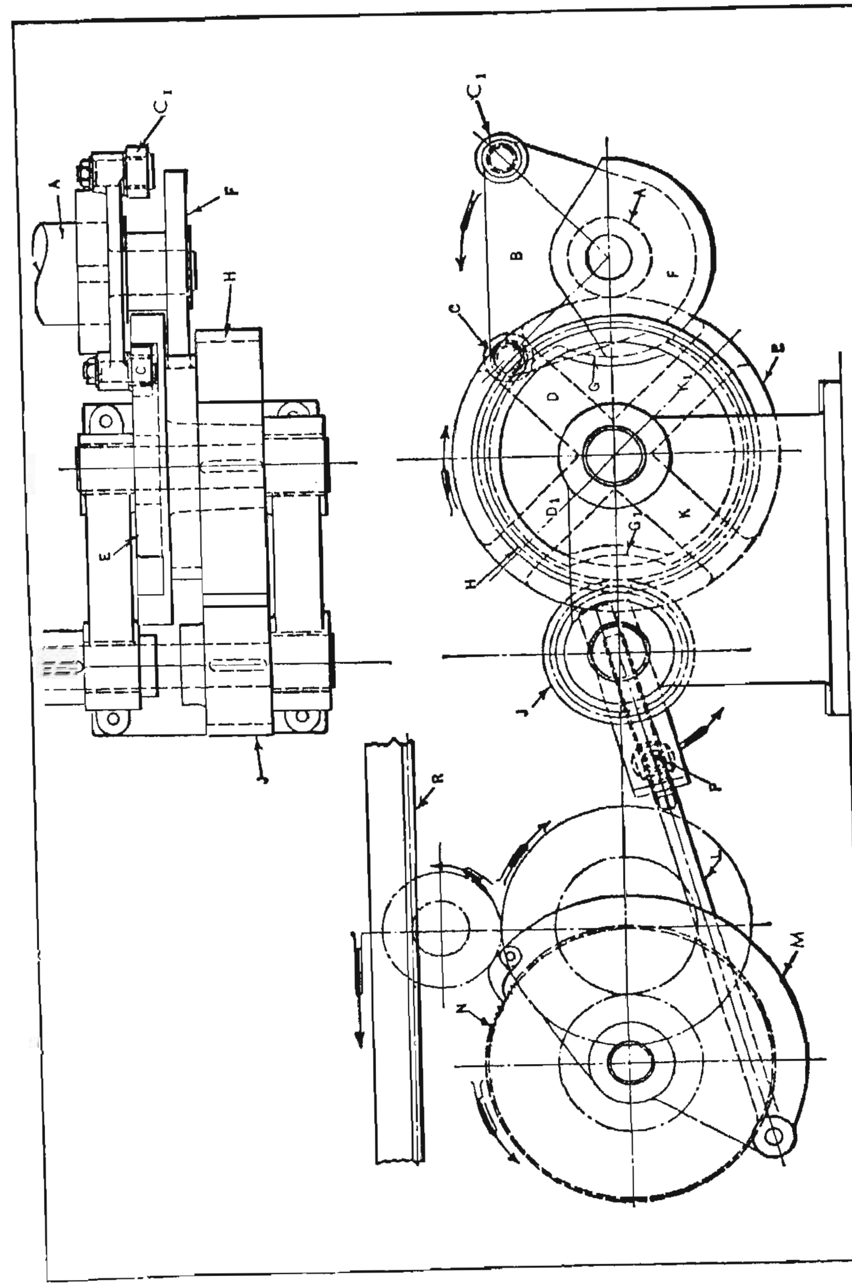


Fig. 26. Indexing and Locking Mechanism for Spindle Head of Multiple-spindle Automatic Screw Machine

camshaft which transmits motion for indexing by means of a chain and sprocket gearing. The sprocket wheel A on the camshaft is directly connected by a chain with a sprocket wheel fast to the spindle head B. Sprocket wheel A is normally loose on its seat on cam C but it is engaged with the cam for indexing by means of a dog D contained in a slot in the cam. One end of this dog is arranged to engage a recess inside the hub of sprocket wheel A and the outer or projecting end is in position to be acted upon by stationary cam E. Normally the dog D is out of engagement with the sprocket wheel, but for the indexing movement, cam E throws the dog into



engagement, thus revolving the sprocket and the spindle head to a new position. As there are five spindles in the head, in this case, the spindle head must be revolved one-fifth revolution at a time. After the indexing movement is completed, cam *E* disconnects the dog from the sprocket automatically.

As it is necessary on machines of this class to locate the spindle head very accurately each time it is indexed, some form of auxiliary locating and locking mechanism is employed. In this case, the locking bolt is at *G*, and is forced into its seat by the spiral spring shown. The action of the bolt is controlled by a lever *H* and cam *F*. This cam allows the bolt to drop in place as soon as the indexing motion is completed. The conical point of bolt *G* engages a seat of corresponding form in whichever plug *K* is in position. These plugs are spaced equidistantly about the periphery of the spindle head. The tapered seat for the end of bolt *G* is formed partly in the plug *K* and partly in the bushing *J* through which the plug passes, as indicated by the detailed view. With this arrangement, the location of the spindle head does not depend upon the closeness of the fit of the cylindrical part of the locking bolt in bushing *J*.

Intermittent Spacing Mechanism of Geneva Type. — The feeding or spacing mechanism to be described is used on a large perforating press. This mechanism serves to advance the plate automatically after each series of holes has been punched. It is designed to advance the plate as quickly as possible, so that the idle stroke of the punch will be reduced to a minimum. With this mechanism, the advancing or feeding movement occurs while the crank that operates the press slide makes one-fourth revolution. To accomplish this result, a modification of the Geneva movement is utilized.

The feeding mechanism is driven from shaft *A* (see Fig. 27) to which is attached plate *B* carrying rollers *C* and C_1. Roller *C* comes into engagement with groove *D* of the Geneva wheel when the main crank of the press is within 45 degrees of its upper position. After roller *C* has turned wheel *E* one-quarter of a revolution and as it is leaving groove *D*, roller

C_1 comes into engagement with groove D_1, so that the turning movement of wheel *E* is continued for another quarter revolution. It will be seen that while plate *B* makes one-half revolution, wheel *E* also turns the same amount; then as plate *B* turns another half revolution, wheel *E* remains stationary.

Attached to shaft *A* beyond plate *B* (see plan view) there is a circular cam or segment *F* which engages an arc *G* of corresponding radius, thus locking wheel *E* while rollers *C* and C_1 are out of engagement. During the second revolution of plate *B*, roller *C* engages groove *K* and roller C_1, in turn, engages K_1. During the following half turn of plate *B*, cam *F* is in engagement with arc G_1, so that Geneva wheel *E* is again locked. This wheel is integral with a 40-tooth spur gear *H* which drives a 20-tooth pinion *J*; consequently, for every half revolution of wheel *E* pinion *J* makes one complete turn.

On the shaft carrying pinion *J* there is a crank having an adjustable crankpin. This crank imparts an oscillating movement through link *L* to the segment-shaped part *M* to which is attached a pawl engaging ratchet wheel *N*. The intermittent motion thus imparted to the ratchet wheel is transmitted through a train of gearing to rack *R*, which is attached to the plate feeding table. By varying the radial position of crank *P*, and consequently the movement of the feed pawl, the advancement of the work can be varied as desired.

Since a quarter turn of wheel *E* causes one-half turn of pinion *J*, it will be seen that the engagement of roller *C* with wheel *E* results in the forward or feeding stroke of the ratchet wheel, whereas engagement of roller C_1 returns the feeding pawl to its starting position. This mechanism operates smoothly, because wheel *E* is gradually accelerated during one-eighth revolution and then gradually decelerated during one-eighth revolution as each roller successively comes into engagement with it. This is an important feature, since the primary object of this mechanism is to provide the required feeding movement in a minimum length of time.

Indexing Mechanism of Screw-slotting Machine. — The automatic machine to which the indexing mechanism shown in Fig. 28 is applied mills the screw-driver slots across the heads of screws. The screws to be slotted are placed in a slowly revolving hopper from which they are conveyed by a chute to the work-holder or turret *M*, which, in turn, locates each successive screw beneath the narrow cutter or saw, which mills the screw-driver slot. The work-holding and cutter-feeding movements are derived from a camshaft at the back of the machine which is connected with the main driving pulley through change-gearing so that the rate of operation may be varied to suit the size of the work.

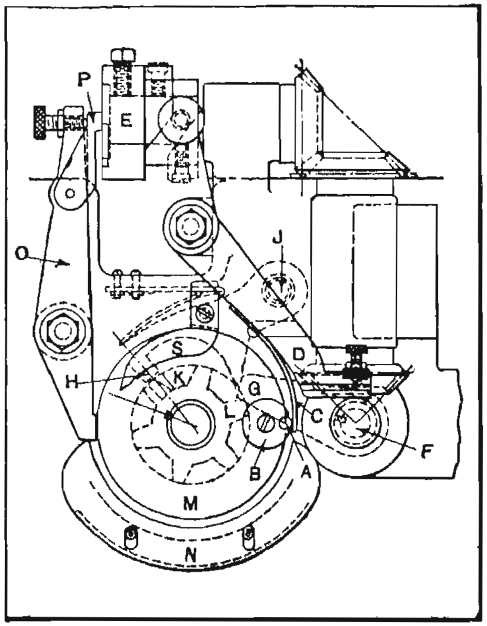


Fig. 28. Automatic Indexing Mechanism of a Screw-slotting Machine

Fig. 28 is a plan view showing the turret operating mechanism. After a screw is released from the chute, it falls into the position shown at *A* where it is held between a seat in bushing *B* and spring *C*, which is attached to escapement lever *D*. This escapement permits the blank to fall into position in the work-holding turret and also holds the screw blank in place in bushing *B*. The lever *D* receives its motion from cam *E* on a camshaft at the rear of the machine, which is driven through change-gears. At *F* is a vertical shaft extending down through the bed of the machine, which is driven through bevel gearing from the shaft on which cam *E* is mounted. This vertical shaft carries a revolving arm *G* that strikes locking lever *H* pivoted as *J*, and raises it from the slot in disk *K* in which it is seated. All of these parts are shown

by dotted lines, as they are located beneath the turret or work-holder of the machine. As arm *G* continues its movement, it strikes against one of the teeth of star-wheel *L* and revolves it one-sixth revolution. When this indexing movement has been completed, locking lever *H* again drops into a slot in *K*, thus locking the turret in position.

The turret carries six equally spaced bushings *B*, although only one is shown in the illustration. The slotting saw (not shown) is located on the side opposite bushing *B* and the screw blanks, after being placed in the work-holder at *A*, are indexed around to the saw by the intermittent action of the indexing mechanism, which movement occurs after each screw-head is slotted. As the screw blanks leave position *A* they are held loosely in place in the bushings by guard *N* which may be adjusted in or out to agree with the body diameter of the screw blank. As each screw arrives at the operating position beneath the slotting saw, it is held firmly against its seat in the bushings by the inner end of lever *O*. This lever receives its motion from the left-hand face of cam *E*. This cam does not bear directly against the end of the lever *O*, but acts through the intermediate lever *P* which is adjustable by means of the thumb-screw shown. It is thus possible to regulate the pressure with which *O* bears against the work, the adjustment being varied according to the size of the screw blank.

The slide carrying the slotting saw moves vertically and is fed downward by a cam as each successive blank is located beneath it. After the slot is milled, the saw is moved upward rapidly by a spring action, and lever *O* releases the slotted screw which drops through a chute and into a receptacle. If the screw blank does not release readily, the continued rotation of the turret brings it into contact with the curved edge of the ejector *S*. Incidentally, the bushings *B* are provided with a number of seats in their periphery so that, by simply turning these seats outward, the bushings are adapted for holding screws of a number of different sizes. The indexing mechanism is so arranged that any inaccuracy which may occur is in a direction lengthwise of the screw slots and not at right angles to the face of the saw, so that the centering of the slot in the head of the screw is not affected.

Semi-Automatic Indexing and Locking Mechanism. — The fixture illustrated in Fig. 29 was designed for a two-spindle

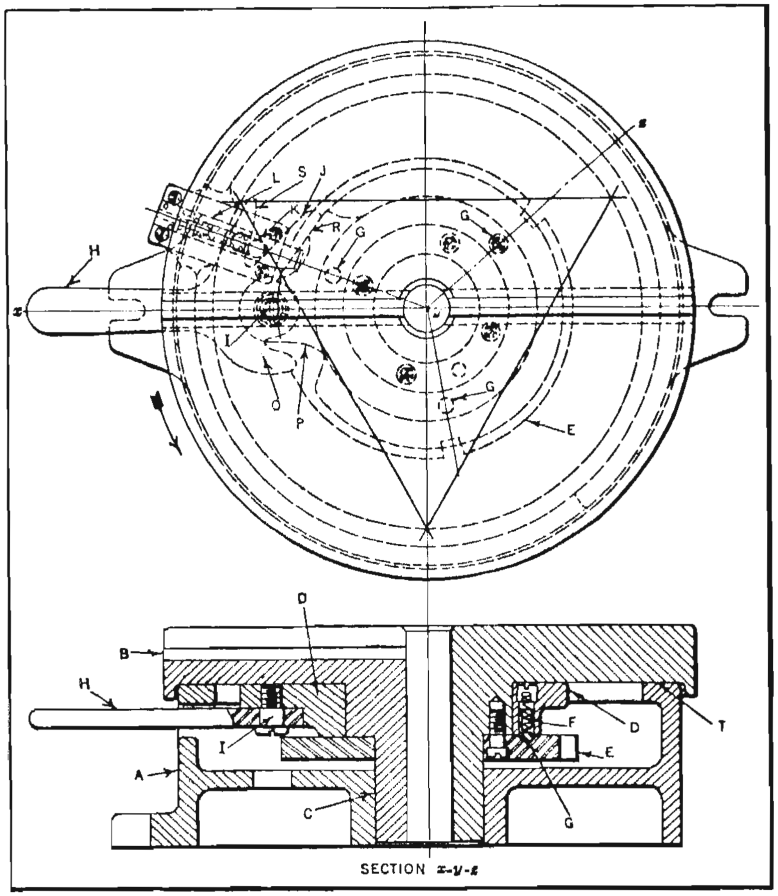


Fig. 29. Sectional and Plan Views of Semi-automatic Indexing Fixture

drilling machine. There is one drilling and one reaming position, and a third station directly in front of the operator for loading and unloading the work. The triangular center lines on the plan view represent these three positions. The

operator has to give the indexing lever only two strokes, one forward and one return, to unlock the fixture and index the work. After indexing, the lock-bolt falls into the next notch and the fixture is locked until released by another movement of the operating lever.

The illustration does not show the work or fixtures, which are mounted on top of the rotary table. The base of the fixture is fastened to the machine table, lugs being provided at each side, as shown by the plan view. The table B is free to revolve on the base A around the central bearing C.

On the table hub, and free to rotate, is ring D to which is pivoted the indexing lever H. The index-plate E is attached to the table by screws and dowel-pins, and it has three index notches cut in its periphery. These two members D and E act as one unit during the forward stroke of the lever, owing to the fact that plunger F engages drill point spots G in the index-plate. Lever H, which is pivoted at I, projects through a slot in the base A. As this lever is pulled toward the operator to turn the table in the direction of the arrow, finger J (see plan view) comes into contact with pin K fastened in lock-bolt L, and releases this bolt from the index-plate E. Bolt L has one taper side to permit easy engagement and eliminate backlash. It is supported by the base and is backed up by a spring.

After bolt L is withdrawn to clear index-plate E, finger O contacts with surface P, which is part of ring D; then lever H and ring D act as a unit, and further movement rotates the index-plate. After the lock-bolt is released by finger J acting against pin K, this bolt rides on the periphery of the index-plate until it falls into the next notch. The motion of lever H is then reversed or pushed backward, so that finger O leaves surface P and finger J comes into contact with surface R on ring D. As this backward movement occurs, plunger F which is backed up by a spring, is lifted from spot G and slides over the surface of plate E until it reaches the next indentation. At the same time, finger S engages pin K and locks the bolt. Provision should be made for lubricating the movable parts.

CHAPTER IV

TRIPPING OR STOP MECHANISMS

THE different devices used for controlling motion may either be manually or automatically operated, and be adjustable for varying the time of disengagement, or non-adjustable so that the tripping action occurs at the same point in the cycle of operations. Tripping and disengaging devices also vary in that some operate periodically or at regular intervals, whereas others act once and then must be re-set by hand preparatory to another disengagement. The application of disengaging mechanisms varies greatly. With some classes of machinery, an automatic trip of some form is used to stop the machine completely after it has performed a certain operation or cycle of movements. On many machine tools, trips are used to disconnect a feeding movement at a predetermined point, not only to prevent the tool from feeding too far, but to make it unnecessary for the operator to watch the machine constantly, in order to avoid spoiling work. (When a feeding motion must be disconnected at a certain point within close limits, it is common practice to use some form of positive stop for locating a slide or carriage after the feeding movement has been discontinued by a trip acting through suitable mechanism.) The function of some tripping devices is to safeguard the mechanism by stopping either the entire machine or a part of it, in case there is an unusual resistance to motion, which might subject the machine to injurious strains.

The three mechanical methods of arresting motion which are most commonly employed are by means of clutches, by shifting belts, and by the disengagement of gearing. When the tripping action is automatic, some design of clutch is generally used to disconnect the driven member from the

driver or source of power. The action of the clutch is controlled in various ways. Shifting belts are not ordinarily applied to machines as a part of the regular mechanism, but are very generally used to control the starting or stopping of an entire machine; clutches are also used extensively for this purpose. Gearing which is engaged or disengaged to start or stop a driven member is used in some cases. Feeding mechanisms of some types have a worm-wheel driven by a worm which is dropped out of mesh when the feeding action is discontinued. The method of controlling motion may depend upon the speed of the driving and driven members, and the necessity of eliminating shocks in starting, or upon some other factor, such as the inertia of the driven part or the frequency with which starting and stopping is required.

How Tripping Action is Controlled. — Automatic tripping devices may be adjustable and be set beforehand to act after a certain part has moved a given distance, or they may only act when a machine begins to operate under abnormal conditions. The adjustable form of trip, if for a part having a rectilinear motion, may consist simply of a stop which is placed in such a position that it will disengage a clutch after the part under the control of the trip has moved the required distance. If a rotary motion is involved, the same principle may be applied with whatever modification of the mechanism is necessary. When the trip is designed to act automatically only when the machine is operating under adverse conditions, the action may be governed by variations of pressure or resistance to motion, or the product on which the machine is working may cause the trip to act in case the operation is not as it should be. The following examples will illustrate a few of the applications and the possibilities of tripping mechanisms of different types.

As various tripping devices are used in conjunction with reversing mechanisms to change the direction of motion, instead of stopping it entirely, Chapter VI on "Reversing Mechanisms" shows additional applications of automatic tripping appliances. Additional examples will be found in Chapter

VII on "Overload Relief Mechanisms" and "Automatic Safeguards."

Trip which Disengages a Clutch. — One of the simplest forms of automatic tripping mechanisms is illustrated diagrammatically at *A*, Fig. 1. This general type is applied to some classes of machine tools for disengaging the feeding movements of a tool-slide at a predetermined point. The tool-slide, which may be the carriage of an engine lathe, is moved along the bed by a feed-screw *a* or a splined rod which is rotated through a clutch *b*. The shifting member of this clutch is operated by a lever *c* the lower end of which connects with rod *d*. This rod extends along the bed a distance equivalent

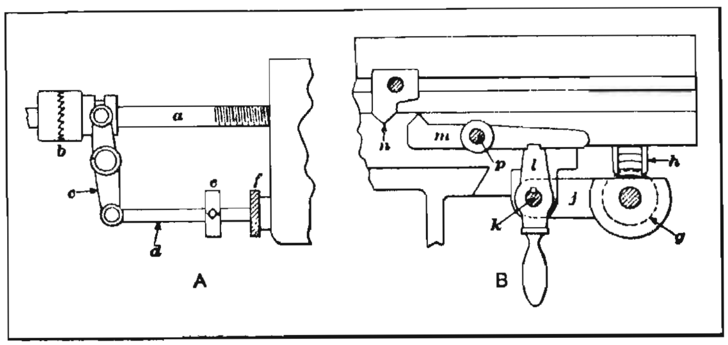


Fig. 1. Simple Forms of Automatic Tripping Mechanisms

to the carriage movement and carries an adjustable stop collar *e*, which is engaged by some projecting part *f* on the carriage; when this engagement occurs, the rod is shifted in a lengthwise direction, thus throwing the clutch out of mesh and stopping the feeding movement. Obviously, the point at which disengagement occurs depends upon the position of stop collar *e* which is set in accordance with the length of the part to be turned. There are other trip mechanisms of the clutch-shifting type which differ from the kind described in regard to the details of the mechanism for shifting the clutch.

Trip which Disengages Gearing. — Diagram *B*, Fig. 1, illustrates a form of automatic trip which serves to disengage worm gearing instead of a clutch. The worm *g* revolves

worm-wheel *h* and the table feed-screw. This worm is carried by an arm *j* pivoted at *k* and held in position by the engagement of lever *l* with a notch in lever *m*. When the adjustable trip dog *n*, attached to the work table, strikes lever *m* and swings it about pivot *p*, the worm *g* drops out of engagement with worm-wheel *h* and the feeding motion stops. The point of this engagement may be varied at will by simply changing the position of the trip dog *n*. Some of the trip mechanisms on vertical drilling machines operate on this same general principle.

Many different designs of automatic tripping mechanisms, especially of the type used on machine tools for controlling feeding movements, are of the same principle as those described, in that trip dogs are attached either directly to the driven member or to some auxiliary mechanism such as a revolving disk geared to the driven part, and these dogs stop the feeding movement either by disengaging a clutch or gearing. If the feeding movement is intermittent and is obtained through ratchet gearing, the pawl may be prevented from engaging the gear teeth of the ratchet wheel after the latter has turned a predetermined amount. An example of this type of tripping device is described in Chapter II (see Fig. 9).

Automatic Stop for Drilling Machine. — Fig. 2 shows a side elevation and plan of an automatic stop or trip for a vertical-spindle drilling machine, which operates by disengaging a friction clutch. The feeding movement is transmitted to the spindle from the friction gear *c* to the disk *d* and through worm gearing at *k* to a pinion meshing with rack *l* attached to the spindle sleeve. The position of friction gear *c* is controlled by hand lever *g* which, through link *e*, lever *f*, shaft *a*, and collar *b*, moves the friction gear in or out of engagement with disk *d*. Lever *g* is held in the engaged position by the latch or trigger *n*. An adjustable stop collar *h* is set by means of graduations to automatically disengage the feed after a hole has been drilled to whatever depth is required. This collar acts by simply striking the end of latch *n*, thus releasing lever *g* and the friction gear *c*. Any wear in the friction clutch is compensated for by adjusting set-screw *j* in the end of connecting link *e*.

Duplex Automatic Tripping Mechanism.—Another form of tripping device for a vertical-spindle drilling machine is illustrated at *B*, Fig. 2. This stop may be set to disengage the worm *e* from the worm-wheel on the pinion shaft, or it may be utilized to disengage miter gear *g* which drives the worm-shaft. The tripping dog is attached to a bracket or

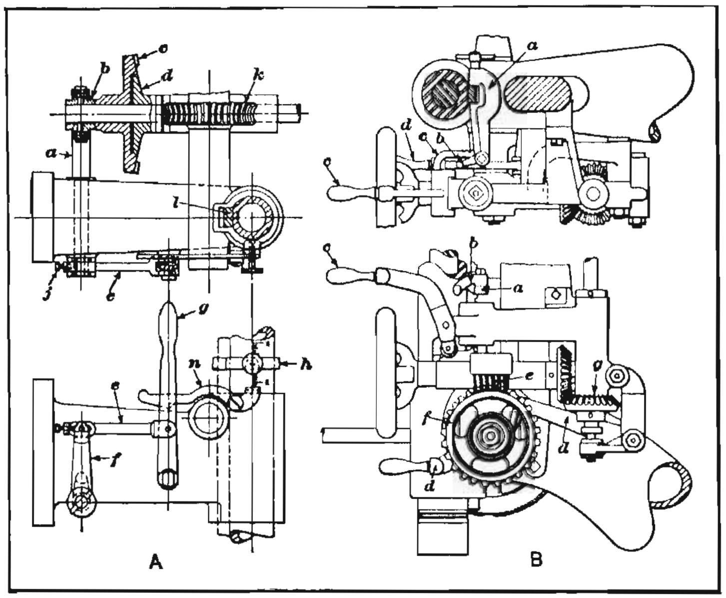


Fig. 2. Automatic Stop or Tripping Mechanisms of Vertical-spindle Drilling Machines

arm *a* clamped to the feed rack on the sleeve. The dog *b* may be swung so as to engage either levers *c* or *d*; as shown in the plan view, it is in the latter position. Lever *c* controls the engagement of worm *e* with wheel *f*, whereas *d* serves to disengage the bevel gear *g*. When the worm is out of mesh, the spindle may be moved vertically by the hand-feed lever, for facing or similar operations, after a hole has been drilled. Ordinarily, gear *g* is disengaged, but this does not leave the spindle free for rapid adjustment.

Adjustable Dial Type of Tripping Mechanism. — The automatic tripping mechanism shown in Fig. 3 was designed for drilling machines and may be adjusted to disengage the downward feeding motion of the drill at any depth up to 14 inches. The feeding movement is transmitted through the drill spindle from shaft *A*, through worm gearing, to shaft *B* which has a pinion engaging the rack on the spindle quill. The automatic disengagement of the feed is controlled by the engagement of pawl *H* with lever *N*. The distance that the spindle feeds downward before the feed is tripped is regulated by the gradu-

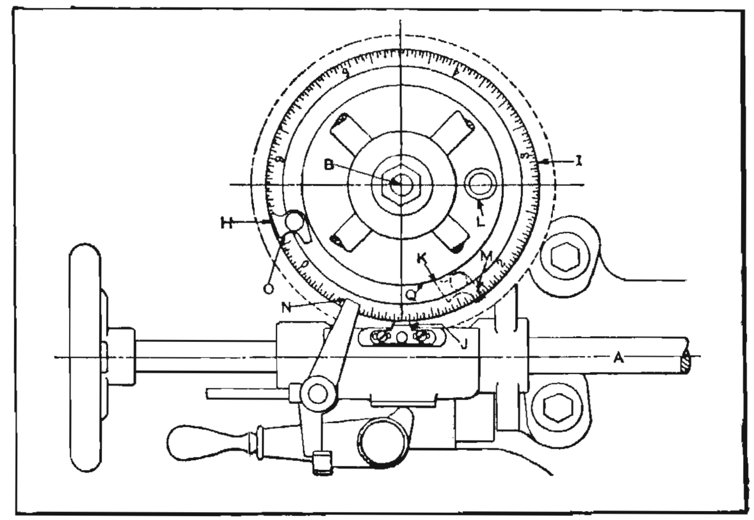


Fig. 3. Automatic Feed-tripping Mechanism having Graduated Adjusting Dial for Controlling Time of Disengagement

ated adjustable dial *I*. The graduations on this dial indicate 1/32 inch of the spindle travel, and one complete revolution represents 7 inches of spindle travel. The pawl *H* is so designed that it can be set to allow two revolutions of the dial before engaging lever *N*.

The operation of the mechanism is as follows: If the feed is to be tripped automatically in 7 inches or less, pawl *H* is set as indicated by the dotted lines at *K*; if it is desired to trip the feed at a distance greater than 7 inches, pawl *H* is turned to the position shown by the full lines. For example, if it should be required to automatically trip the feed at a depth of 3 inches, the knurled nut L would first be loosened and the graduated dial I turned until the figure 3 on it was opposite the mark on pointer J, after which nut L would be tightened. The pawl H would then be set in the position shown by the dotted lines, with the result that, when the drill had traveled 3 inches, the surface M would come into contact with the side N of the trip arm and disengage the feed. On the other hand, if it were required to drill to a depth of 9 inches before the feed was automatically tripped, the dial I

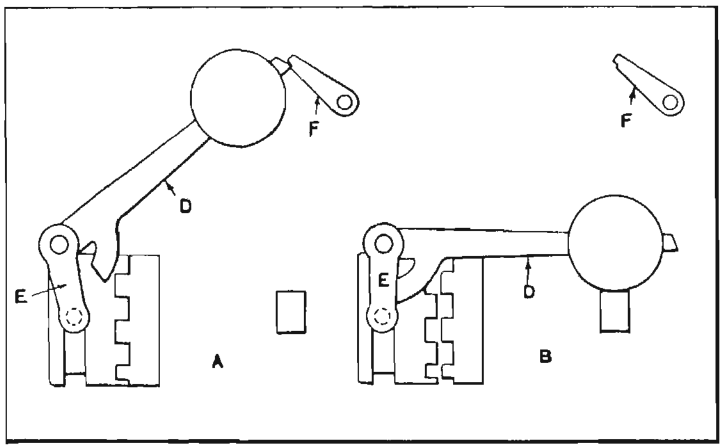


Fig. 4. Releasing Mechanism in which a Falling Weight Disengages a Toothed Clutch

would be set with figure 2 opposite the mark on pointer J, and pawl H would be turned to the position shown by the full lines. With the pawl in this position, the contact of surface O with lever N would not throw out the feed, as the pawl, being loose on its stud, would simply turn and pass the tripping arm without moving it. After the pawl had passed the arm, it would then be in the position shown by the dotted lines, that is, with the end in contact with a projecting sleeve, as at K, thus preventing further rotary movement, so that, when it again came around to the tripping lever, the feed would be disengaged. If the knurled nut L is loose, the feed cannot be automatically tripped at any point.

Clutch Release of Gravity Type. — The disengagement of a toothed clutch can be accurately accomplished at a predetermined point in the traverse of a sliding machine member by an arrangement similar to that shown at A and B in Fig. 4. This design enables disengagement to be accomplished so quickly that wear on the clutch teeth is reduced to a minimum. The mechanism consists of a weighted lever D connected at

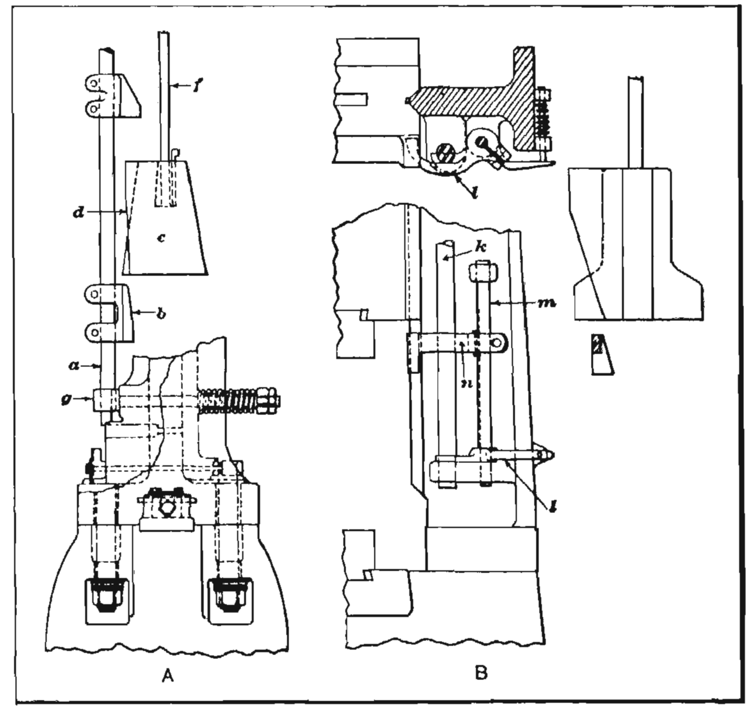


Fig. 5. Board Drop-hammer Tripping Mechanisms

one end to two links E which control the axial movement of the driven clutch member. When the clutch is engaged as shown at A, the opposite end of the weighted lever is supported by trip-lever F, but when lever F is tripped during the operation of the machine, the weighted lever falls, and thus forces the driven clutch member away from the driving member. The relative positions of the various details when the clutch is disengaged are illustrated at B. A similar mechanism may be used for releasing a worm from a worm-wheel.

Tripping Mechanism of Drop-hammer. — When a board type of drop-hammer has fallen and is rebounding, the friction rolls grip the board and elevate the hammer preparatory to the delivery of another blow. The eccentrically mounted gripping roll is moved inward against the board for elevating the hammer, when a "friction bar" is released by a tripping mechanism and allowed to fall. Most of these tripping devices operate on the same general principle as the design illustrated at A in Fig. 5. The friction bar a is attached at its upper end to a lever that controls the position of the eccentrically mounted friction roll; when the bar falls, the lifting board is gripped between this front roll and one at the rear that revolves in one position. Before the friction bar is released, the lower end rests upon a seat which prevents it from falling. When the hammer c descends, an incline surface d on it engages bracket b and pushes bar a off of its seat. The weight of this bar is sufficient to give the roll referred to the required gripping pressure on the board f, so that the hammer is lifted to the top of its stroke. As the hammer rises, it engages a lever and raises the friction bar which, in this particular case, is returned to its seat by a spring-operated guide g. In order to operate the hammer properly, it is necessary to release the friction bar at exactly the right time, which must be varied according to the thickness of the hammer dies.

The tripping mechanism must be so set that, as the hammer rebounds, its upward movement is continued by the action of the friction rolls. If the release of the friction bar occurs too soon, the rolls will grip the board either before the hammer strikes its blow or before it has had time to rebound. On the other hand, if the release occurs too late, the hammer will fall back after rebounding and the roll will have to pick up a "dead" or stationary load. The point of release depends upon the vertical position of bracket b.

A trip mechanism of the swinging latch type is shown at B, Fig. 5. In this case, the friction bar k is held in the upper

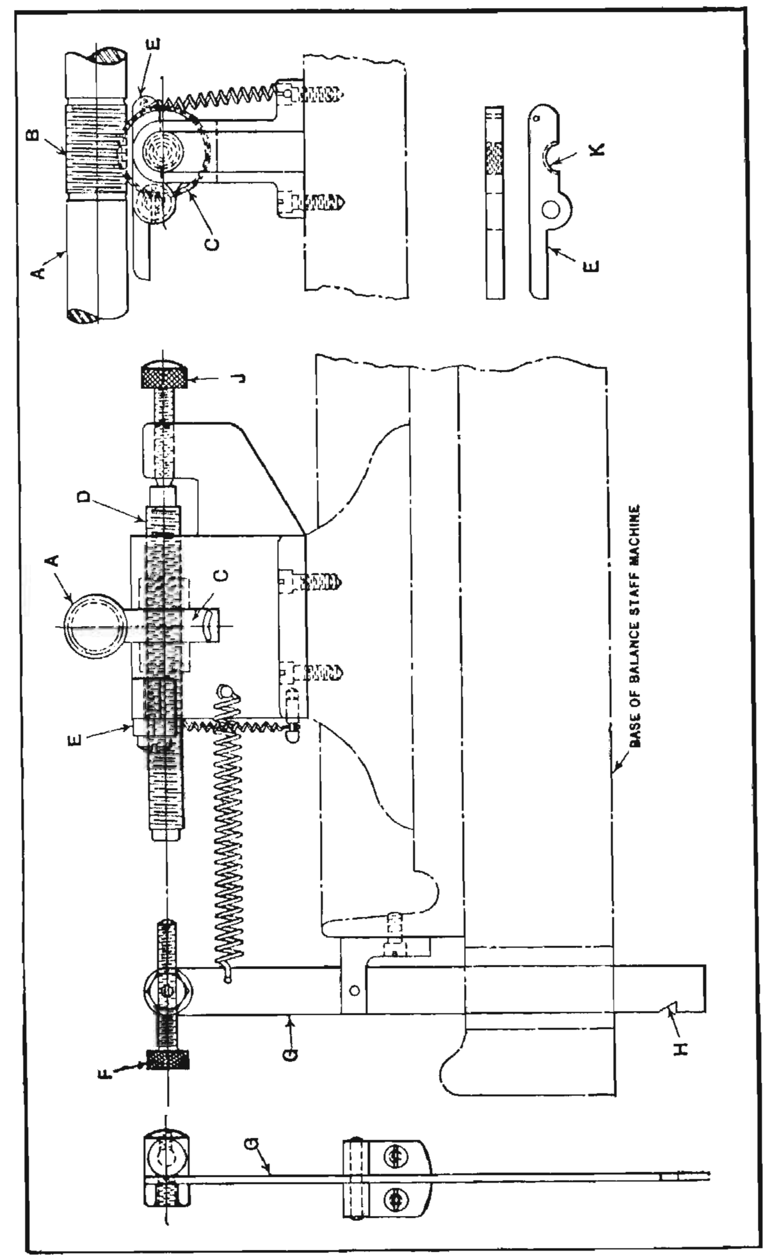


Fig. 6. Device for Automatically Stopping the Operation of a Balance-staff Machine when the End of the Stock is Reached

position by a catch l which engages a slot in the bar and is attached to the short vertical shaft m. This shaft also carries a lever n that extends out far enough to engage an inclined surface on the hammer. As the hammer descends, lever n, bar m, and catch l are turned, thus releasing the friction bar k and allowing the rolls to grip the board for elevating the hammer. The point at which release occurs may be varied by changing the vertical position of lever n.

Stop for an Automatic Machine. — An automatic stop designed to prevent the jamming of stock in the feeding chuck of a balance-staff machine when the end of the stock has been reached is shown in Fig. 6. Several automatic stops were tried out on these machines without success previous to the development of the stop shown. This stop is a modification of the type used on bench screw machines. Before proceeding with the description of the stop it may be well to mention that a balance-staff machine is used for turning the balance staffs for watches. The balance staff is the shaft on which the balance wheel of a watch is secured. At each end of this shaft is a fine pivot or tapering point. When assembled in the watch these pivots are supported by jewel bearings which are cupped out to receive the pivots.

The balance-staff machine is similar in design to a small bench screw machine. It consists of a spindle containing a chuck and a mechanism for feeding the stock forward, a camshaft with a set of cams for operating the chucking mechanism, for reversing and controlling the feed, and for operating the front and rear cross-slides. The front cross-slide carries the cutting-off milling cutter, and the rear cross-slide the milling cutters that form the pivots. It might be added that the balance staff is only rough-milled in this machine; it is afterward put into a pivot-turning machine and the pivots turned down to the required diameter and then polished. During the milling of the pivots and the cutting-off operation, the work is firmly held in a support. The spindle camshaft and the cutters in the cross-slides are all driven separately from the main driving shaft.

In applying the stop it was necessary to alter the regular camshaft A somewhat. A left-hand quadruple-thread worm having five threads per inch with a lead of 0.20 inch and a pitch of 0.050 inch was cut on the shaft at B. This worm is employed to drive the worm-gear C. When the machine is in operation the camshaft turns the worm-gear C which, in turn, revolves shaft D. A keyway is cut the entire length of the threaded part of shaft D. A key held in worm-gear C is made a sliding fit in this keyway. Now as shaft D has a left-hand thread which fits the threaded part K of piece E (see view in lower right-hand corner), shaft D will move toward the set-screw F when the machine is in operation. When lever G is pushed back by the end of shaft D coming in contact with screw F, the stopping device of the machine, which is controlled by a latch held in V-slot H, is released and the machine stopped. The time at which this releasing movement occurs may readily be varied by adjusting screw F.

The stock used in the manufacture of the balance staffs comes in three-foot lengths, and the stopping device is set to stop the machine automatically when about 2 inches of stock is left in the feeding chuck. This prevents the jamming of the stock between the cutters and enables the operator to easily take out the piece left in the chuck and to feed in a new piece of stock by lifting part E sufficiently to disengage the thread of shaft D. The latter member may be pushed back into contact with the stop-screw J, when lever G will spring back to its former position as shown in the illustration. The machine is then started by lifting up the starting latch so that it will be engaged by the V-slot H.

Stop Mechanism which Operates After Predetermined Number of Revolutions. — This mechanism was applied to a coil spring winding machine for turning a shaft a predetermined and exact number of revolutions and then stopping it instantly. To obtain the exact number of turns needed, the motion is transmitted through a multiple-disk drive arranged as shown in Fig. 7. The drive is from pulley A through friction disks B to shaft C, which rotates part D, having a hole at E in

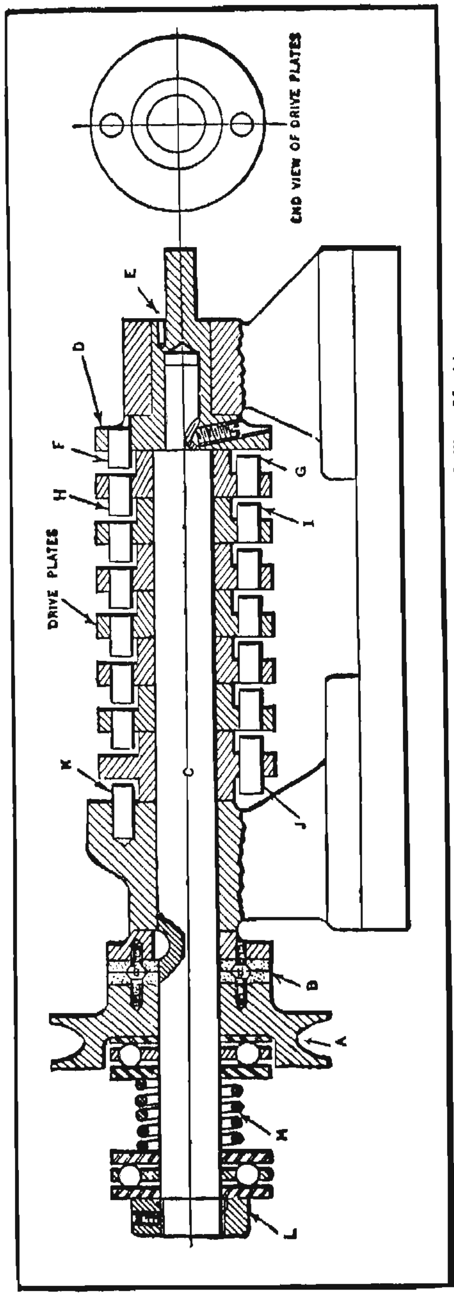


Fig. 7. Accurate Stop Mechanism Used on a Spring Coiling Machine

which one end of the wire to be coiled is inserted.

In the flange of part *D*, there is a pin *F* which strikes pin *G*, causing pin *H* to turn and strike pin *I*. In this manner, the pin in each successive disk strikes a pin on the disk following, after whatever part of a revolution is needed to bring the successive pins into contact. Finally, pin *J* in the last disk engages the fixed pin *K*, which locks the entire combination of disks and stops shaft *C* immediately as the friction disks *B* slip, permitting the pulley to continue revolving. The coil spring is then removed, and the drive, which must be equipped with a reverse countershaft, is reversed, the disks being unwound or turned backward until they are again locked in the reverse position. It will be noted that the frictional resistance between

disks *B* can be varied by means of nut *L*, which is used to regulate the pressure from spring *M* located between two thrust bearings.

Quick Stop Mechanism for Spring-winding Machine.— A quick stop mechanism which has been used successfully

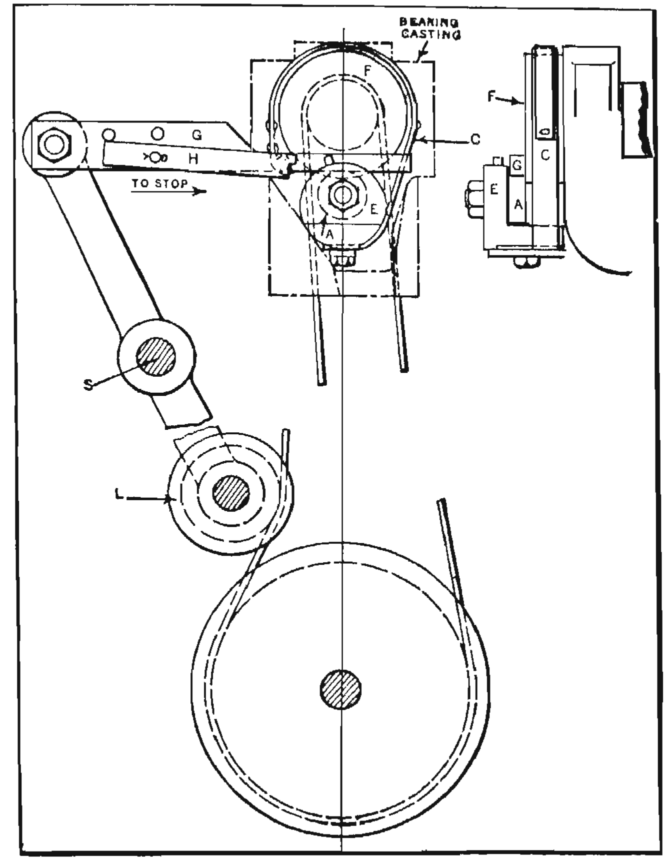


Fig. 8. Stop Mechanism which Slackens Driving Belt and Applies Brake

on a machine for winding springs has an operating shaft *S* (see Fig. 8) which withdraws idler pulley *L* and advances arm *G* on which pawl *H* is attached. This pawl pushes against a pin in eccentric *E* and turns it, thus tightening brake-band *C* on brake-drum *F*. It will be seen, therefore, that as soon as the driving belt is slackened by withdrawing pulley L, the brake is applied. Arm G is supported by an extension which rests on collar A.

When the brake-band is adjusted to the proper tension, the pawl will tighten it and then slip over the pin, allowing the brake to release and leaving the spindle free after stopping it; or if desired, the adjustment can be such that the pawl will not slide over the pin, in which case the spindle is stopped and held stationary. The spindle speed is 7,000 revolutions per minute and there is very little wear on the parts. The operating shaft S has a handle on one end, conveniently located for the operator, but this motion could, of course, be applied automatically if desired.

Device for Stopping a Machine Momentarily. — It is an easy matter to provide a means of stopping a machine with its own power by the use of a cam or some similar mechanical device, but it is a more difficult problem to design a device that will automatically stop a machine and start it again so that it is only momentarily brought to a full stop. With the usual type of device designed simply to shut off the power, there is no stored up energy that can be utilized to start the machine again without attention on the part of the operator. Fig. 9 shows a simple and automatic device for shutting off the power momentarily.

The shaft A is a part of the machine that rotates while the machine is in operation. When it is desired to stop the motive power, latch B is moved to the left, pivoting about pin P. The machine is started again by moving latch B back to the right-hand position. The problem is to provide a means of throwing latch B to the left or stopping position and then back again to the running position without attention on the part of the operator. The latch can be readily moved to the left by means of a cam, such as shown at C, but the instant that the cam has so acted the machine will stop, and the cam will be left in such a position that the lever cannot be moved back to the starting position, either through the action of a spring, such as shown at D, or by other means.

To overcome this difficulty, shaft A was provided with a flange member E secured by the pin F. The cam member C is free to turn on shaft A, but its movement relative to flange E is limited by two pins G and H, which are driven into flange E and allowed to project through the slots J and K in the cam member. On the cam member is a stud L to which is secured one end of a spring M, the other end being secured to some part of the machine.

When shaft A travels in the direction indicated by the arrow, the flange E will drive the cam member C in the same

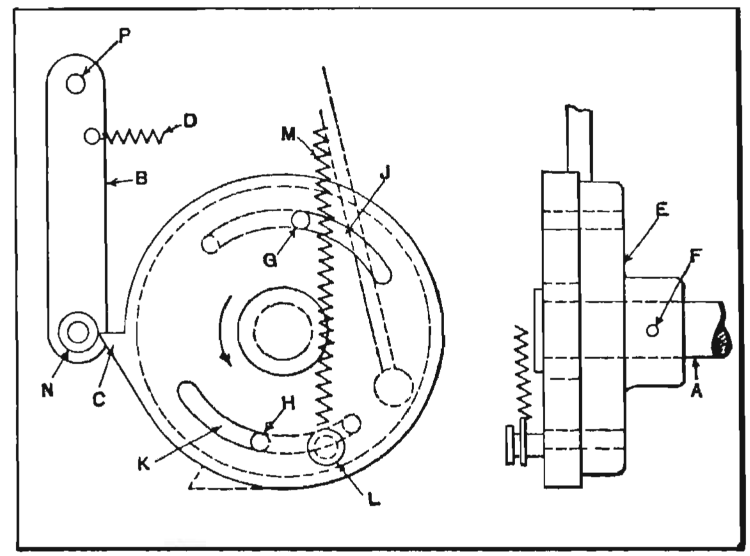


Fig. 9. Mechanism for Automatically Starting a Machine after Stopping it Momentarily

direction. As the pin L reaches its highest point and begins its descent, it will act against the pressure exerted by spring M until the instant that pin L passes a vertical line through the center of the shaft. At this point spring M will accelerate the movement of the cam member and not only push lever B to the left, thus bringing the machine to a stop, but will also carry the point of the cam past the roller N, so that lever B will be returned to its original position by the action of spring

D. The machine will thus be started automatically after having been stopped for an instant.

Mechanism which Stops Rapid-moving Slide at Top of Stroke. — A stencil-cutting machine used for cutting odd-shaped openings in stencils of various fibrous materials has a fixed knife *A* (see Fig. 10) and an upper knife *B* which is given a rapid reciprocating motion. The stop mechanism to be described is so arranged that when a foot-lever is depressed, the cutter-slide is disconnected from the driving member, which allows the cutter-slide to stop instantly and always at the top of its stroke, thus permitting the work to be moved either away from the knife or from one opening to another.

The machine is driven by a constant-speed motor through a belt connecting with a pulley on the rear end of shaft *C*. The foot-lever which controls the action of the cutter-slide is pivoted in the machine base and connects with the rear end of lever *D*. The drive to the cutter-slide is through two hinged fingers *E* and *F* (see end view which shows cover plate removed). These clutch fingers are pivoted to a cross-head which is given a rapid reciprocating movement by crank-pin *G*. The foot-lever normally is held in its upper position by a spring (not shown). When the foot-lever is depressed, clutch fingers *E* and *F* are forced out of engagement with angular notches in the cutter-slide, and as soon as the latter is released, it is pulled to the top of its stroke by spring *H*. (The spring *H* also takes up any lost motion that there might be in the reciprocating cross-head or connections.) The cutter-slide remains in this upper position until the clutch fingers on the reciprocating cross-head are released so that they are again free to engage the angular notches.

The mechanism for controlling the disengagement and the engagement of the clutch fingers operates as follows: Lever *D* is pivoted at *J* so that its forward end moves upward as the foot-lever is depressed. The round end of lever *D* connects with and transmits this upward movement to bar *K* and to cross-arm *L* (see end view). Cross-arm *L* is secured to bar *K* by a set-screw, which allows the cross-arm to be ad-

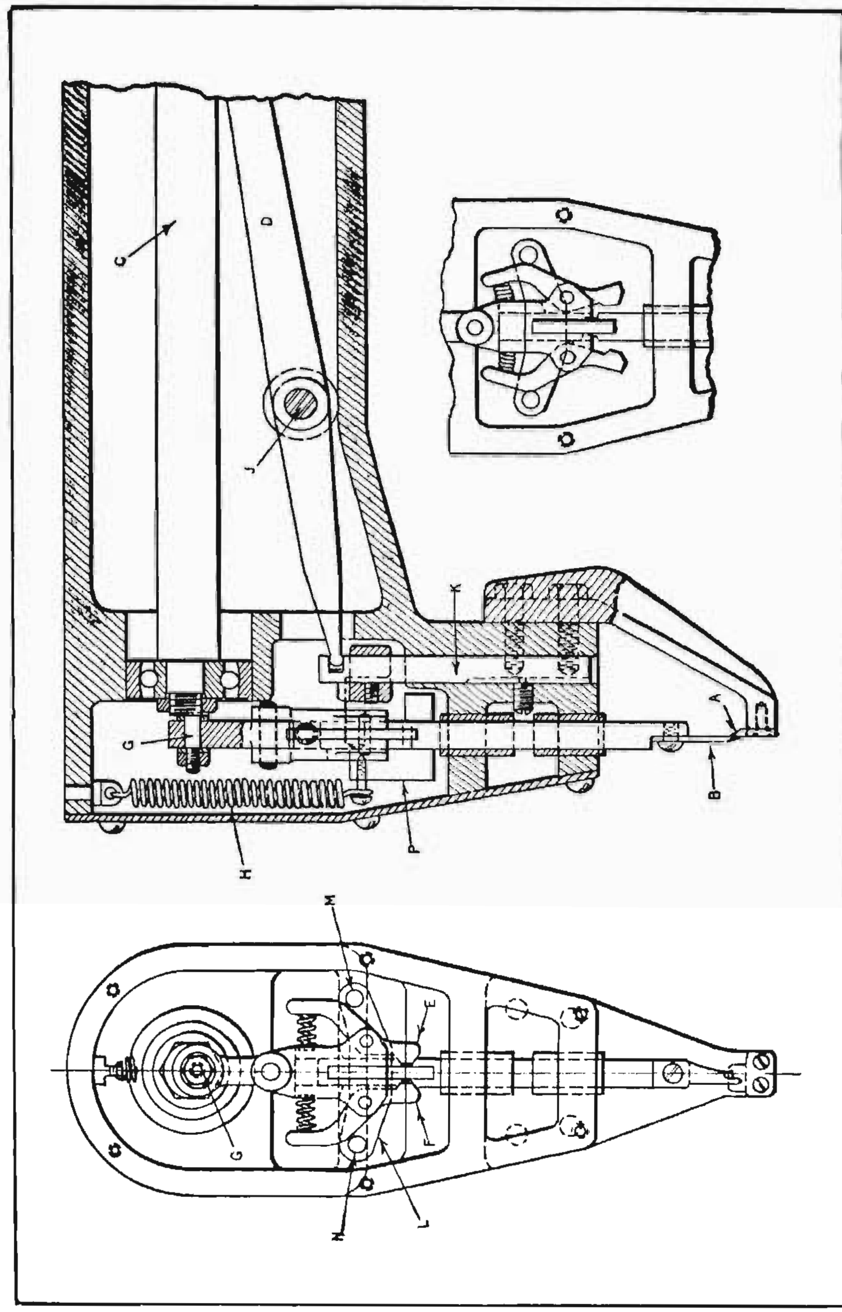


Fig. 10. Mechanism for Quickly Stopping Cutter-slide of Stencil-cutting Machine, at the Top of its Stroke

justed up or down to accommodate any variation in any other part of the machine. This cross-arm has pins *N* and *M*, which act against the upper ends of the clutch fingers, compressing the springs shown and releasing the fingers, as indicated by the detail view in the lower right-hand corner of the illustration.

When the foot-lever is released, it is pulled upward by its spring, which, through the mechanism described, moves cross-arm *L* downward, and engages a tongue on the cutter-bar, forcing it down and thus allowing the springs to close the clutch fingers upon the cutter-bar so that they can engage the angular notches. A little clearance is provided in the bottom of each notch to allow for any wear on the tooth or the notch. The upper end of the cutter-bar fits into a hole in the cross-head. The cutter-bar also has a tongue *P* which serves as a guide for the upper end, the cross-head being slotted to receive this tongue, which also acts as a stop when the clutch fingers are released.

With this mechanism, starting and stopping is almost instantaneous, with the main drive shaft of the machine running at high speed. The knives may be formed to various curved shapes to facilitate cutting any fancy scroll work or small round holes. This machine, incidentally, is used in cutting leather gaskets, design stencils, and various other articles not required in large enough quantities to warrant the cost of a punch and die. The machine may also be used to advantage in trimming and cutting odd-shaped patterns and uneven edges of leather goods after the stitching operation. The upper knife has a sharp point or pilot on the lower end for piercing the work when starting the cut.

Clutches that Automatically Disengage. — The clutches used on power presses are designed to automatically disengage after making one or more revolutions. The clutch connects the flywheel or driving gear of the press with the driven shaft, whenever it is tripped, by pressing down a foot-treadle. As long as this treadle is held down, the clutch remains in engagement and the press continues to run; if the treadle is

released, the clutch is disengaged when the ram or slide of the press is approximately at the top of its stroke. The downward movement of the treadle releases a pin, key, or some other form of locking device which quickly engages the driving member; when the treadle is released, the locking device encounters some form of trip or cam surface which withdraws it and stops the press. There are many designs of clutches of this general type.

Automatic Clutches of the Key Type. — Fig. 11 shows a clutch of the type having a key which is engaged or disengaged with the hub *A* of the flywheel. This flywheel revolves freely on the shaft until the dog *D* is pulled down by the action of the foot-treadle; then the key *C* is forced downward into engagement with the flywheel by a strong steel spring *E*. When

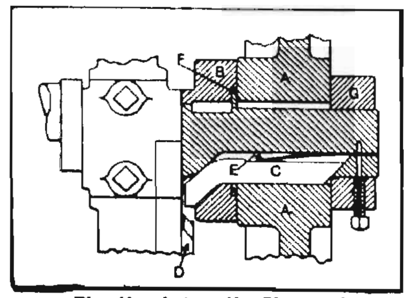


Fig. 11. Automatic Clutch of Shifting-key Type

the foot-treadle is released, the dog *D* is forced up, and when key *C* comes into contact with the dog, it is pushed back into the shaft, thus allowing the flywheel to again run freely. If the treadle is depressed and then released, the press will make one revolution before stopping, but if the treadle is held downward, the press will continue to run. This clutch is equipped with a safety device to prevent the ram or slide of the press from descending unexpectedly while setting dies or making adjustments. This safety device consists of a steel ring *F* having a keyway or slot in it for receiving the key *C*. When the press slide is at the top of its stroke and dog *D* is up, the key is entirely within the shaft and may be held in this position by turning ring *F*, thus preventing accidental engagement of the clutch. Ring *F* has an extension arm that enables it to be turned readily.

A clutch is shown in Fig. 12 that has a rocking key instead of one that moves radially. This key *A* extends across the shaft and, when the press is not in motion, the key rests in a semi-circular seat and occupies the position shown in the end view. When in this position, the lever *B* at one end of the key is in engagement with the latch *C*, which is connected with the foot-treadle. As soon as latch *C* is swung out of the way by depressing the treadle, lever *B* and the key tend to turn as they are acted upon by the compressed spring *E*. When the flywheel has turned far enough to bring one of the recesses *F* opposite the key, the latter, by making a quarter turn in its seat, engages the recess and locks the flywheel and

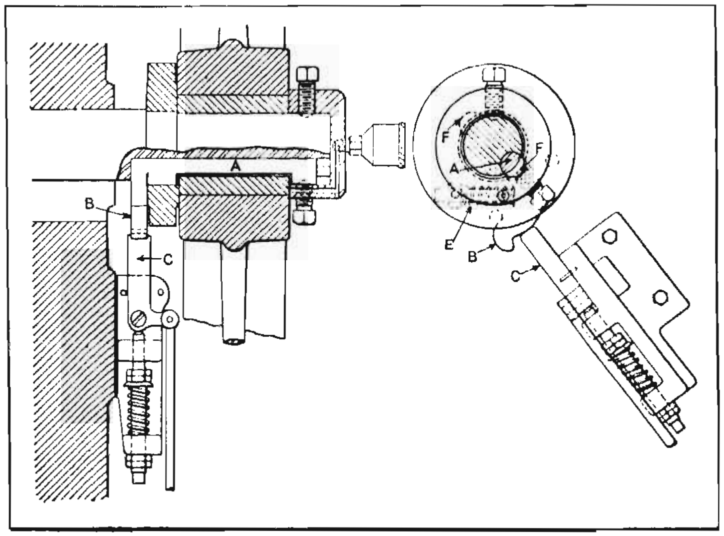


Fig. 12. Automatic Clutch of Turning-key Type

shaft together. If the treadle is immediately released, thus allowing latch *C* to swing back to the vertical position, it will engage lever *B* when it comes around and force this lever and the key back out of engagement with the flywheel.

Clutches Engaged by a Wedging Action. — Some designs of automatic clutches are engaged by a wedging action of some locking member between cam or eccentric surfaces, instead of employing pins or keys. An example of the cam type of clutch is shown at *A*. Fig. 13. A cam *a*, having a series of eccentric or cam surfaces, is keyed to the crankshaft, and surrounding this cam there is a slotted ring *c* con-

taining rollers *b*, which, in turn, are surrounded by a hardened tool steel ring *d*. These parts are inserted in a recess formed in the hub of the flywheel. On the slotted ring *c*, there is a lug *f* which is in engagement with the pivoted stop lever *e* when the press is not in operation. As soon as the stop lever is drawn downward by means of the foot pedal, the rollers are carried around by the action of the flywheel until they are wedged tightly between the cam surfaces and the outer ring *d*; the crankshaft is then driven with the flywheel and continues to revolve until lever *e* is released and, by striking

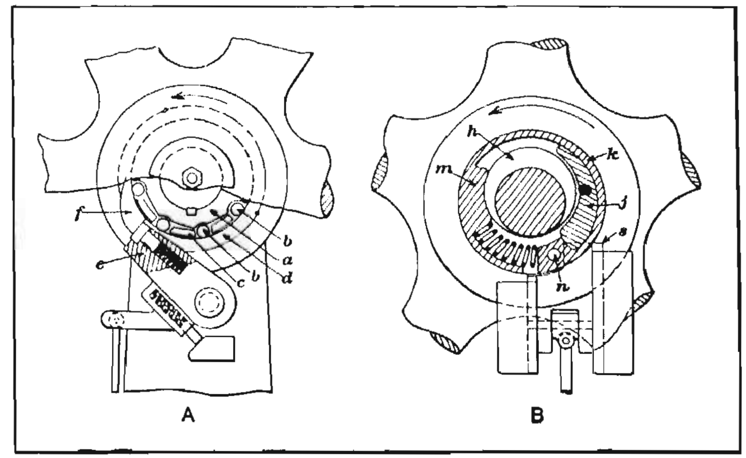


Fig. 13. Automatic Clutches of the Cam or Wedging Type

stop *f*, throws the rollers out of engagement. The slotted ring *c* has a spring attached to it (not shown) which turns the ring and rollers toward the high points of the cam when the ring is released by the lowering of lever *e*.

The design of clutch illustrated at *B* is equipped with an eccentric *h* which is solid with the crankshaft and a wedge-shaped member *j* which serves to lock the flywheel and crankshaft together. This wedge *j* is located between the eccentric and a ring *k* inserted in a recess in the hub of the flywheel. The ring is split and compressed somewhat so as to exert a pressure against the wall of the recess. When the stop *s* is in engagement with pin *n*, the flywheel simply revolves about the expansion ring *k*. When stop *s* is withdrawn, the ring *k* expands, and, as it begins to revolve with the flywheel, the wedge *j* is forced between the eccentric *h* and the inside of the ring; consequently, the flywheel, expansion ring, and the shaft are firmly locked together. When the foot pedal is released and stop *s* engages pin *n*, the ring contracts and remains stationary while the flywheel continues to revolve. The surface at *m* serves as a brake, so that the crank-shaft is stopped when the slide is approximately at the top of its stroke.

Clutch of One-revolution Coiled-spring Type. — A spring clutch designed to disengage automatically at the end of one

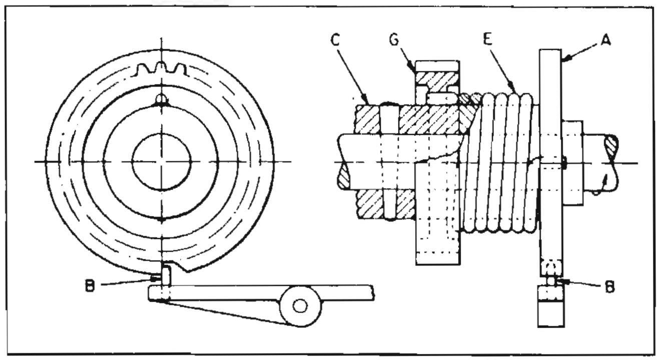


Fig. 14. Spring Type of Clutch which Automatically Disengages After Making One Revolution

revolution of the driving shaft is shown in Fig. 14. Wheel *A* is free to rotate on the driving shaft; gear *G* is free to rotate on bushing *C* which is fastened to the drive shaft; gear *G* and wheel *A* are attached to opposite ends of spring *E*; and spring *E* is a snug fit on bushing *C* but is not fastened to it.

With the drive shaft rotating, gear *G* will not move as long as wheel *A* is held from rotating by dog *B*; but when the dog is withdrawn momentarily, wheel *A* is dragged around by the spring which now grips bushing *C* firmly and hence rotates with it. Since gear *G* is attached to the spring, it too will rotate at the same speed as the bushing and drive shaft. When the rotation of wheel *A* is stopped by the re-engagement of dog *B*, the spring will open slightly, slip on the bushing, and no longer drive gear *G*.

By making wheel *A* large and heavy, like a flywheel, the clutch will take hold gradually and with a smooth action. When the clutch is disengaged, the driving spring will be opened slightly, due to the rotating action against the stationary wheel. When wheel *A* is released, it rotates forward a slight amount and causes the spring to grip the drum. As the driven members of the clutch start to rotate, their motion will be resisted by the inertia of the heavy wheel, which, in turn, prevents the clutch from engaging quickly.

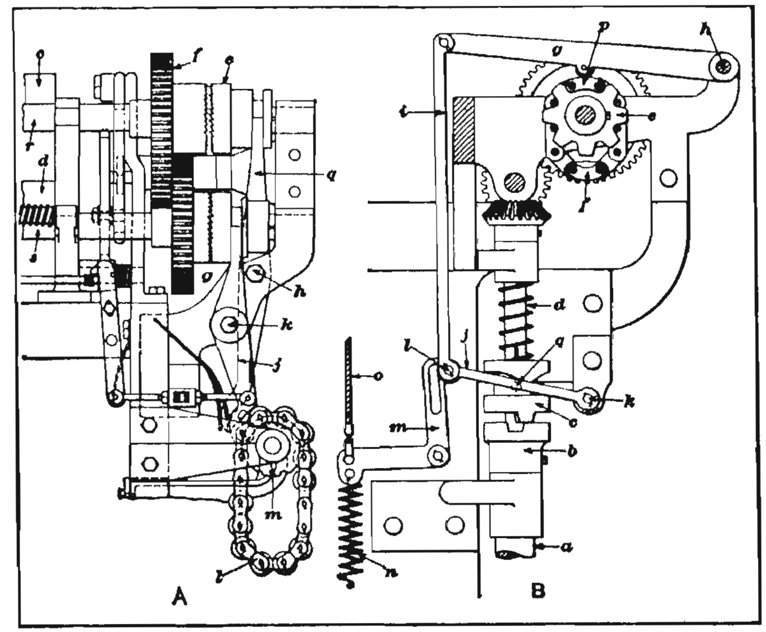


Fig. 15. Mechanisms Equipped with Endless Chains for Controlling Engagement and Disengagement of Clutches

Variable Clutch Control by "Pattern Chain." — The ingenious method of controlling clutches illustrated at *A* in Fig. 15 is applied to a textile machine known as a "twister." The variations in the yarn are obtained by controlling the action of two sets of delivery rolls. The lower rolls *r* and *s* of each set support the upper rolls *c* and *d*. Splined to the end of roll *r* is a shifting clutch member *e* which revolves the roll when engaged with the clutch teeth on the hub of gear *f*. A similar clutch and gear combination is located at *g* for driving the lower set of delivery rolls. The upper clutch is connected with lever *q* pivoted at *h*, and the lower clutch, with lever *j* pivoted at *k*. The action of these clutch levers is governed by a pattern chain *l* suspended on a drum *m*. As this drum revolves, the rollers of the pattern chain come into engagement with the lower ends of the clutch levers, thus shifting the clutches in and out of engagement. By changing the position of the rolls or risers of the pattern chain, the pattern of the yarn may be varied and different fancy effects be obtained. The chain drum is revolved by means of change gearing for varying the speed according to requirements. The clutch gears are rotated continuously, and the delivery rolls are only stopped when a knob or knot is being formed, both sets of rolls being rotated while the yarns are being twisted together between the knots.

Another application of an endless chain for controlling the engagement and disengagement of a clutch at predetermined intervals is illustrated at *B*, Fig. 15. This mechanism is applied to a loom. The vertical shaft *a* is driven through bevel gearing (not shown) at the lower end, from the driving shaft of the loom. The upper end of shaft *a* carries a clutch member *b*, which is engaged by the shifting clutch member *c* splined to shaft *d*. Shaft *d*, through the bevel and spur gearing shown, is connected with the chain drum or cylinder *e* carrying the clutch controlling chain *f*. Above this chain, there is a lever *g* pivoted at *h* and connected by link *i* with another lever *j* pivoted at *k*. The pin *l* connecting the link and lever engages a slot in bellcrank *m*, the movements of which are controlled by a spring *n* and a connector *o* which extends to another part of the machine. The vertical slot in lever *m* has a short horizontal section at the upper end.

The action of the mechanism is as follows: When the clutch members are engaged, the chain drum and chain revolve, and when one of the links *p* engages lever *g*, the lower lever *j* is raised, thus locating pin *q* in the upper part of the annular groove of the shifting clutch member. As soon as pin *l* at the end of lever *j* reaches the upper end of the vertical slot, the bellcrank lever *m* swings over under the action of spring *n*, thus engaging pin *l* with the horizontal part of the slot and locking the lever *j* in the upper position. As soon as the lever *j* is raised, a projection engages pin *q* and disconnects the clutch, thus stopping the rotation of shaft *d*. The link *p* on the pattern chain is no longer under the roller of lever *g*, but this lever is still held in the upper position, by the engagement of pin *l* with the horizontal slot in bellcrank lever *m*. The clutch remains disengaged until the connector *o* swings the vertical part of lever *m* to the right, thus allowing

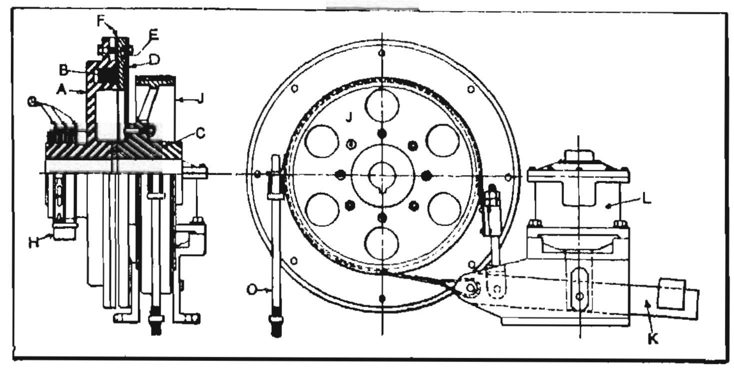


Fig. 16. Magnetic Clutch Equipped with Automatic Band Brake which Operates when Clutch Releases

the upper clutch member *c* to re-engage the lower part. The movements of the connector are controlled by another chain which operates on the same general principle as the one referred to.

Magnetic Clutch with Automatic Band Brake. — The magnetic clutch illustrated in Fig. 16 is equipped with an electrically operated brake which acts automatically when the clutch is released, in order to stop the driven part as quickly as possible. The driving shaft carries the field *A* which is provided with a magnetizing coil *B*. The hub *C* on the driven shaft has attached to it a flexible spring-steel disk or plate *D*. This plate carries the armature *E* which is prevented from coming directly into contact with the magnetizing coil by a ring of frictional material at *F*. This friction ring, which is made of woven asbestos and brass wire, provides a frictional surface for driving. The ends of the winding of the magnetizing coil are attached to the rings *G* which are in contact with a pair of brushes *H* connected with the electrical circuit. The automatic brake, which is of the band type, engages drum *J*, and the ends of the band are pivoted to lever *K* at two points as shown. The plunger of a solenoid enclosed in cylinder *L* is attached to lever *K*, and at the outer end of this lever there is a weight which serves to apply the brake when the clutch is disengaged.

In the operation of this clutch, the current is gradually admitted to the magnetizing coil by means of a rheostat. The magnetic attraction between this coil and the armature causes the friction ring *F* to be held firmly against the driving member, so that motion is transmitted between the driving and driven shafts. The solenoid is also energized so that lever *K* is pulled upward and the band brake about drum *J* released. This brake is held in circular form and out of contact with the drum by a spring and rod *O*. As soon as the circuit is broken, the clutch is released, and the solenoid allows the weighted lever *K* to fall, thus supplying the brake automatically to the driven part. This feature is of particular advantage when the driven side of the clutch is connected to some part which tends to revolve quite a long time after disengagement.

Multiple-disk Clutch Equipped with Brake. — It is sometimes necessary to start and stop machines or certain parts of machines smoothly, with great rapidity, and in synchronism with other moving parts. With light or slow moving apparatus, the problem is relatively simple, but the difficulties multiply as weight and speed are increased. The clutch mechanism shown in Fig. 17 has proved very efficient for the class of service mentioned. This design of clutch is used on a machine transmitting a load of 20 horsepower and operating about 3600 times per day, under unusually trying conditions

This machine picks up its load from dead rest, makes three revolutions and comes to rest again in three-fifths second, or at an average rate of 300 revolutions per minute, without the slightest shock or effort. When it is considered that the clutch drum is driven at only 340 revolutions per minute, and the engagement is only a fraction of a second, it will be seen that the slip is very slight indeed. The absence of shock may be attributed to the perfect cushioning of the pressure applied to the clutch and to the liberal friction area provided,

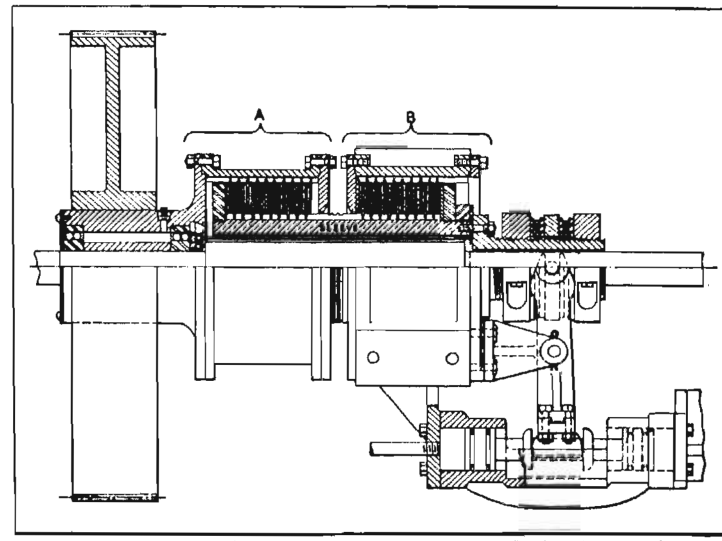


Fig. 17. Quick-acting Multiple-disk Clutch and Brake Combination

there being nearly a square inch for each pound of pull at the average radius of the disks.

The device consists essentially of two multiple-disk friction clutches of the dry type mounted tandem on a single sleeve which is fitted to slide, but not to turn, on a shaft that is directly coupled to the intermittent load. The driving clutch is shown at *A* and the brake clutch, at *B*. The two clutches are built up in the usual form for disk clutches, that is, with two alternate series of disks, one keyed to the driving member and the other to the driven member; one set is preferably faced with friction fabric. One series of disks in a set is provided with internal projections to engage longitudinal slots on the sleeve, while the other disks have external projections loosely fitting the internal slots of the driving and braking clutch drums. The projecting lugs on the disks are reinforced to provide greater bearing surface on the sides of the slots in which they travel. As both clutches are mounted on the same sleeve, and the outer part of the driving clutch is continuously driven, the sleeve becomes the driven member of the driving clutch and the driving member of the brake clutch. The driven member of the brake clutch is solidly bolted to the frame of the machine of which the clutch constitutes a part, so that, in reality, it is not driven, but acts as a brake to bring the sleeve to rest when this clutch is engaged.

Both the clutch drums are built in skeleton form to facilitate the egress of material wearing off the friction facings, and to permit of the easy application of castor oil to the facings. If this treatment is not neglected, a set of facings may last two years or more in constant service, but, if the facings are allowed to become entirely dry, they will be less durable. The sleeve is provided with a flange on each end so that, when it is moved endwise, the disks of one of the clutches will be clamped between one of the sleeve flanges and the head of one of the clutch drums, while the pressure on the disks of the other clutch will be released. Movement of the sleeve in the opposite direction will release the disks of the first clutch and clamp those of the second. In the illustration, the parts are shown in the position of rest, or with the driving clutch disengaged and the brake clutch set.

The controlling mechanism is operated pneumatically and may be made automatic by connecting with other moving parts to actuate the valves. The actual movement of the sleeve which engages and disengages the clutches is derived from two opposed pneumatic cylinders and the connections shown. It will be apparent that the cylinders must work alternately, that is, when one is under pressure the other must be open or free to exhaust. The distribution of air is controlled by two valves, together with a series of interconnecting pipes. With

the valves in the "up" position, compressed air is free to pass through the pipe to one of the cylinders and to the top of the other valve for forcing it down, cutting off the air supply of the cylinder it serves, and opening it to exhaust.

A small hole near the live-air inlet leads to the annular space below the valve proper, around the stem, and is open continuously, admitting air to hold the valve in the "up" position when so placed. As the only connections between the controlling valves and the cylinders are pipes, the control may be somewhat remote and placed in any convenient position. Experiments have been made to determine the practicability of operating the valves magnetically, and also of moving the clutch sleeve by means of magnets, but both have been found far less efficient and much slower than air, the slowness of the electrical operation being due to the time required for the magnets to "build up." The drift of the shaft after the operation of the stopping valve has been found to be very small and practically constant, the shaft stopping within a few degrees of the same position every time. Any wear of the friction disks or their facings is automatically compensated for by additional travel of the pneumatic pistons, so that mechanical adjustments are rarely required.

CHAPTER V

ELECTRICAL TRIPPING MECHANISMS

WHEN an automatic machine must be safeguarded in some way, electrical tripping mechanisms are sometimes preferable to purely mechanical devices. For some purposes a mechanical tripping mechanism would be so complicated as compared to an electrical device as to be inferior to the former, if not entirely impracticable. Another advantage of the electrical devices lies in the fact that they may in some cases, be used as a check on the accuracy of preceding operations and thus avoid finishing pieces of work that are defective. The application of electricity to automatic mechines may be regarded as a complication in itself, but this is far from being true if the tripping devices are properly applied.

The following examples are typical applications of electromagnetic tripping devices to automatic machines, and by studying these designs, one may readily understand how similar tripping mechanisms could be applied to other classes of machinery. In most of these examples, the tripping devices constitute part of attachments for standard machines that were converted into "automatics."

Methods of Closing Electromagnetic Circuit. — A metallic cartridge shell is shown in Fig. 1 (view to right) in place on a machine which pierces the primer hole, and a shell is illustrated at *H* on which the piercing operation has been performed. After the hole has been pierced, the primer is inserted in the primer cavity *J*. These operations are performed on a standard Waterbury-Farrel cartridge primer. The shells were formerly placed on dial pins by hand and indexed under the cross-head for piercing and inserting the primer; they were then removed from the dial pins automatically. An improvement was made in the method of operation by applying an

automatic feed mechanism to place the shells on the dial pins, but this did not dispense with the necessity of an operator for each machine, as there are three possible conditions that may result in the production of imperfect work: 1. The feed mechanism might fail to deliver the shell to the dial pin, or the supply of shells might become exhausted, while primers would continue to feed and thus be wasted. 2. The piercing punch might break and the machine would then continue to place primers in the cavities of shells which had not been pierced, and such shells would obviously be useless. 3. The primer

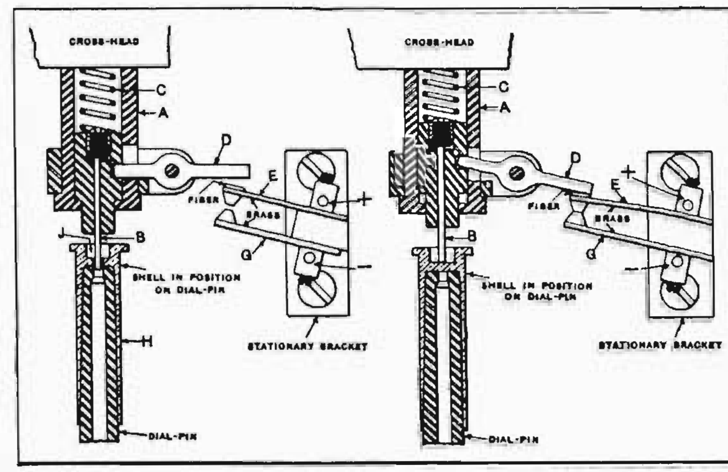


Fig. 1. Mechanism for Closing Circuit and Stopping Machine in Case Punch Fails to Pierce the Shell

feed might fail to work properly, or the supply of primers might become exhausted.

The application of a suitable electromagnetic tripping mechanism to this machine takes care of all of these contingencies. First, consider the possibility of the feed mechanism failing to deliver a shell to the dial pin. Referring to diagram *A*, Fig. 2, it will be seen that the shells are carried on pins on the dial and are indexed under the punch *a*. If a shell is in its place on the dial pin, it contracts the spring *b* when the ram descends, but should the mechanism fail to deliver a shell to the pin, the sleeve *g* passes down over the dial pin and pushes the upper contact *d* of the tripping mechanism down upon the lower contact *e*. This closes the electrical circuit and stops the cross-head on the upstroke. The contacts are fastened to the frame of the machine and the method by which the tripping mechanism operates will be described in detail later.

The way in which the piercing operation is safeguarded by the electromagnetic tripping mechanism is illustrated in Fig. 1. The punch-holder *A* is located at the index point immediately after the completion of the piercing operation. If a shell is pierced, the pin *B* descends through the hole in the shell, as shown at the left of the illustration; but if the piercing operation does not take place, the punch is held in the position indicated in the right-hand illustration, thus contracting the light spring *C* and throwing the lever *D* against the contact *E*. This closes the electrical circuit and causes the machine to be stopped so that shells cannot have primers inserted in them when the primer hole has not been properly pierced.

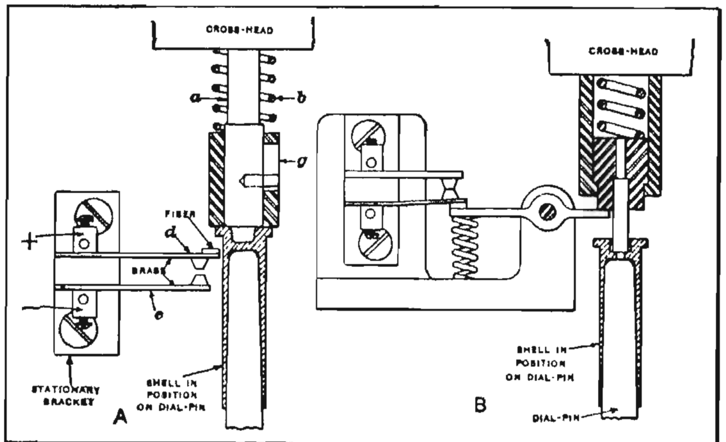


Fig. 2. Circuit-closing Device which Acts when Shell has not been Placed on Dial Pin; (B) Trip which Prevents Passing a Shell without a Primer

The failure of the machine to feed a primer into the primer cavity of the shell is guarded against by the mechanism illustrated at *B* in Fig. 2. The design of this tripping mechanism is practically the same as that used to control the piercing operation, and will be readily understood without further description.

Fig. 3 illustrates the mechanism used on a press for assembling the brass cups *A* and *B*, the cup *A* being inserted in the cup *B*. These cups are held in hoppers on each side of the machine from which they are taken by notched dials. The cups *A* are dropped into holes in the machine dial which passes over the dial carrying the cups *B*. The operation of the machine will not be described; it should be mentioned, however, that a plunger descends in such a manner that the cup *A* is forced into place in cup *B*. Several conditions may occur that will result in loss or damage. The feed mechanism could fail to deliver either one or both cups to their respective dials,

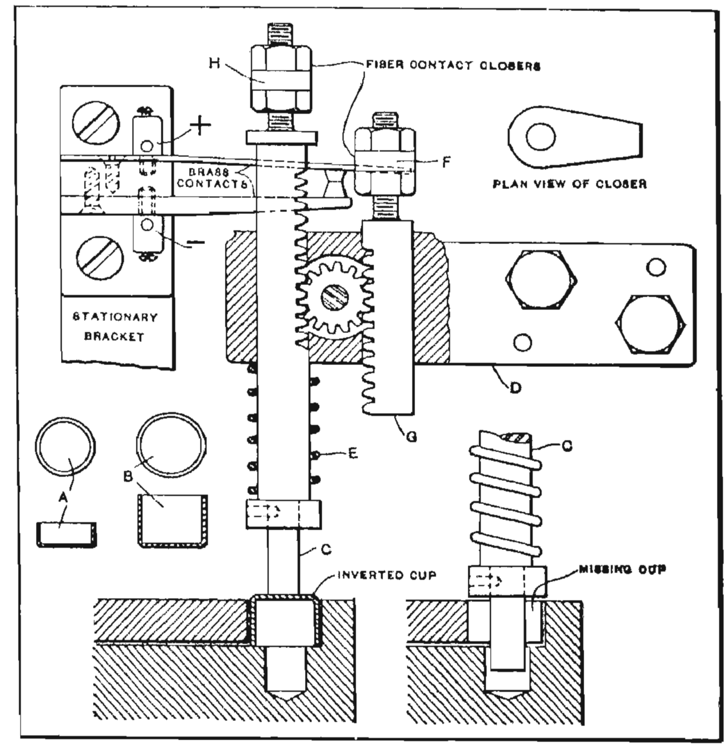


Fig. 3. Circuit-closing Device Used on Machine for Assembling Shells A and B

or it could deliver them to the dials in an inverted position. Either the absence or inversion of either or both cups is detected by an electromagnetic tripping device which automatically stops the machine until the error has been corrected. The punch C is located at an index point preceding the assembling punch, and is carried by a bracket which is fastened to the cross-head. In the case of an inverted cup, the punch C is held on the bottom of the cup and pulls the rod G down through the action of the pinion, which engages with rack

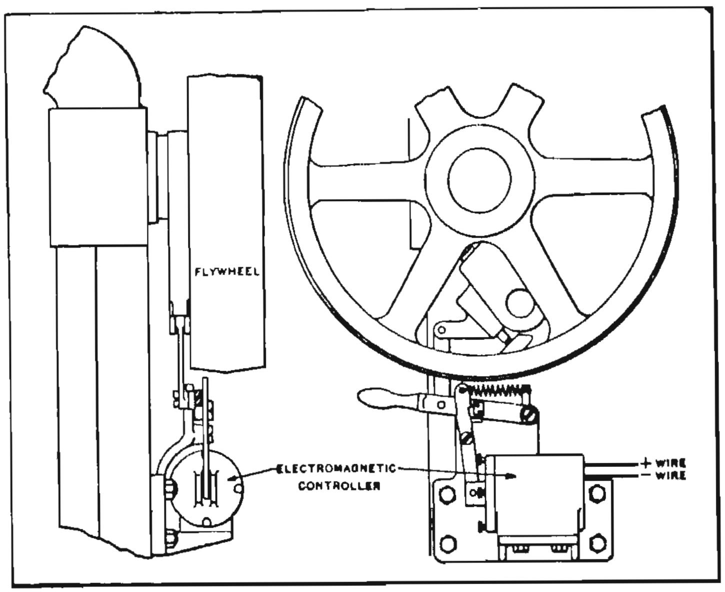


Fig. 4. Electromagnetic Controller Applied to Power Press for Operating Clutch

teeth cut in the rods C and G. The descent of the rod G causes the contact closer F to pull down the upper electrical contact until it closes the circuit and causes the machine to be stopped.

The detail view at the right shows the punch and die when the feed mechanism has failed to deliver a cup to the dial plate. In this case, the upper electrical contact is pulled down by the contact closer H and causes the machine to be stopped as previously described.

Electromagnetic Controller. — Fig 4 shows the electromagnetic tripping device used on the machines referred to in the foregoing, for stopping the machine. In this illustration, the tripping device is shown in place on a power press equipped with a Horton clutch. The arrangement of the tripping mechanism will be more readily understood by referring to Fig. 5, which shows an end and cross-sectional view. This tripping mechanism is self-contained and can be applied to any style of press or type of machine. The bracket A carries the magnet B, pole-piece C, and levers D and E. The brass pole G is wound with No. 14 double-covered wire and the con-

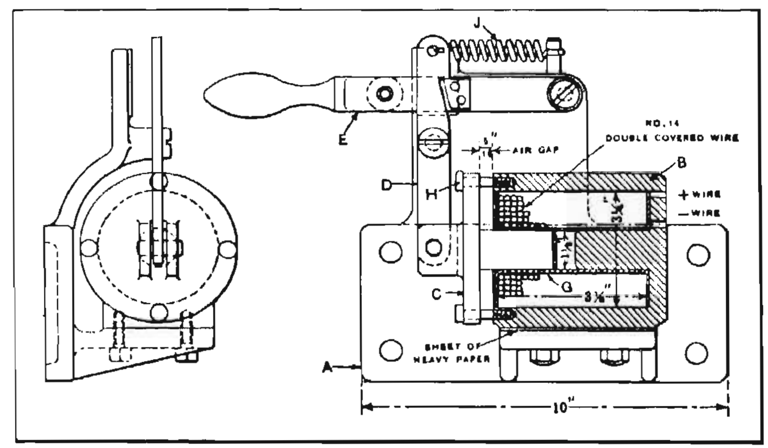


Fig. 5. End and Cross-sectional Views of Electromagnetic Controller

necting wires extend through the back of the spool. The brass pins H help to support the pole-piece C and provide adjustment for different widths of air gap, which should be as small as possible.

In order to start the press, the lever E is pulled down. This engages the flywheel clutch (see Fig. 4) and allows the spring J to pull the lever D over the hardened knife-edge, thus setting the pole-piece C at the proper working distance from the magnet. The inside dimensions of the device are given in Fig. 5. When the magnet is energized by two dry cells, it gives an initial pull of from twelve to fifteen pounds. As the dry cells are used on open-circuit—except for the fractional part of a second during which the contacts meet—they have a long life.

The initial pull provided by an electromagnet of this kind varies with the material used for the magnet and the pole-piece. Where cast iron is used, the pull of the magnet can be calculated by the formula:

$$NI = 3000\, Z \sqrt{P \div D},$$

in which

N = number of coils of wire on the spool (ampere-turns);
I = current in amperes;
P = pull in pounds;
Z = air gap in inches;
D = diameter of plunger in inches.

The electromagnet shown in Fig. 5 was designed to give a pull of 15 pounds, and it will be seen that $Z = 5/16$ inch and $D = 1.125$ inch. Then:

$$NI = 3000 \times 5/16 \sqrt{15 \div 1.125} = 3423.19 \text{ ampere-turns.}$$

Assuming that there are 375 turns of wire on a spool, the amount of current required will be found to be $\frac{3423.19}{375} = 9.14$ or, say, 10 amperes. Two good dry cells connected in series will average 15 amperes during their useful life and give a considerably higher current when new. As 10 amperes is sufficient to enable the electromagnet to do the work required of it, it will be seen that an ample factor of safety is provided. When designing devices of this kind, moving wires and moving contacts should be avoided and the mechanism should be made as simple as possible. The dry cells should be used on open circuit, the contacts carefully insulated from the machine, and covers provided for contacts and terminals.

Electric Stop for Drawing Presses.— The automatic electric stop to be described was designed for use on roll-feed double-action presses employed in the production of screw shells such as are used on incandescent electric lamp bulbs

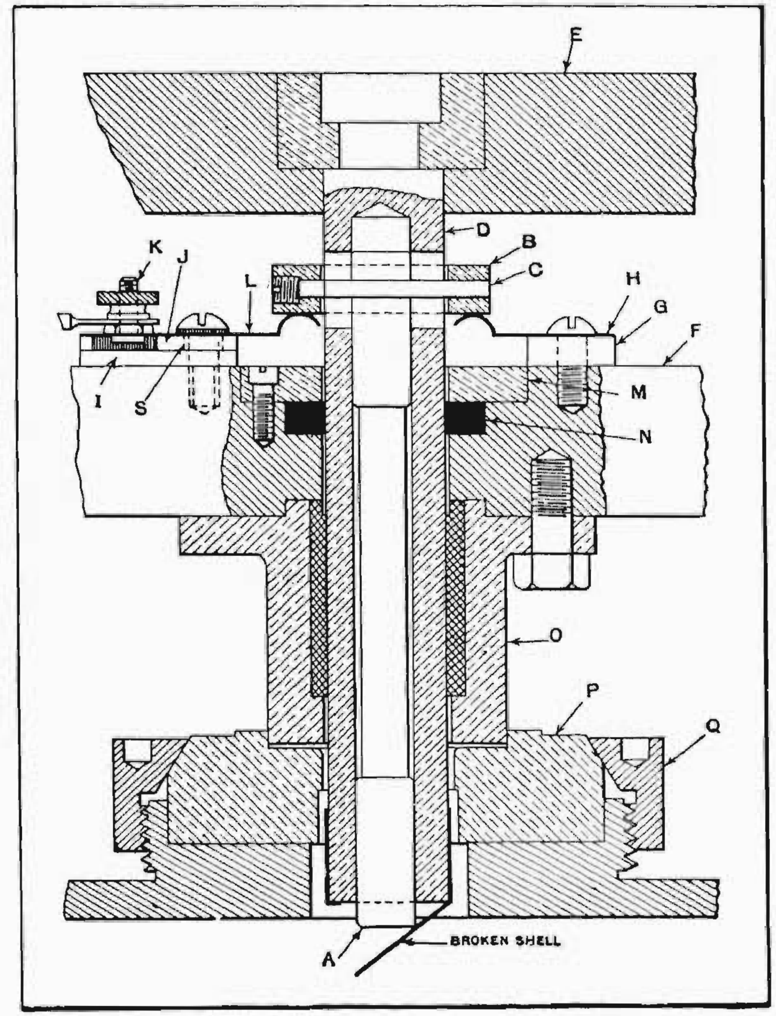


Fig. 6. Shell Blanking and Drawing Die Equipped with Electric Stop

and sockets. Each press was equipped with a gang or multiple die designed to blank and draw five shells at one stroke. In drawing the shells from 0.010-, 0.008- and 0.006-inch metal it occasionally happened that the bottom of a shell would be pushed through, as shown in Fig. 6. If the press was not stopped immediately when this occurred, the scrap shells would pile up on the die, frequently stopping the press. In such cases it was necessary to employ a long lever to release the dies. The dies were often broken or otherwise made unfit for future work as a result of this treatment.

The electric stop eliminated this trouble by automatically stopping the press whenever the bottom of a shell pushed through, thus making it possible for one operator to attend four presses, where it had previously been necessary to have an operator for each press. An electric bell was also placed on the presses so that the operator's attention would be called to a machine as soon as it stopped. The construction of the automatic stop is as follows: A collar *B*, Fig. 6, is attached to each of the gang punches, and held to the push-pin *A* by a threaded dowel-pin *C*. This collar is designed to make contact between the brass springs *H* and *L* when the bottom of the shell is pushed through. Spring *H* is fastened to a brass block *G*, which is grounded on the press through the cutting punch-holder *F*. Spring *L* is prevented from grounding on the cutting punch-holder by the fiber blocks *J* and *I*. In order to prevent the screw that holds blocks *J* and *I* in place from making a short circuit between the punch-holder *F* and the contact *L*, an insulating bushing *S* was made of fiber and the screw inserted through this bushing. At the outer end of spring *L* is placed a terminal post *K* to which is fastened a wire. This wire, in turn, is connected in series with contacts *L* on the other four punches of the same press as indicated in Fig. 9.

Each press is also equipped with a make-and-break device, such as shown in Fig. 7. This device is provided with a plunger-holder *A* fastened to the blanking slide by the bracket *C*. When the slide is down within 1/16 inch of the bottom of the stroke, the spring-actuated plunger *G* makes contact with the brass plug *D* which is held in the fiber insulating block *E*. The fiber block is fastened to the frame of the press by the bracket *F*. By completing the circuit in this way, the electrically operated latch shown in Fig. 8 can be tripped only when the press is within 1/16 inch of the bottom of the stroke.

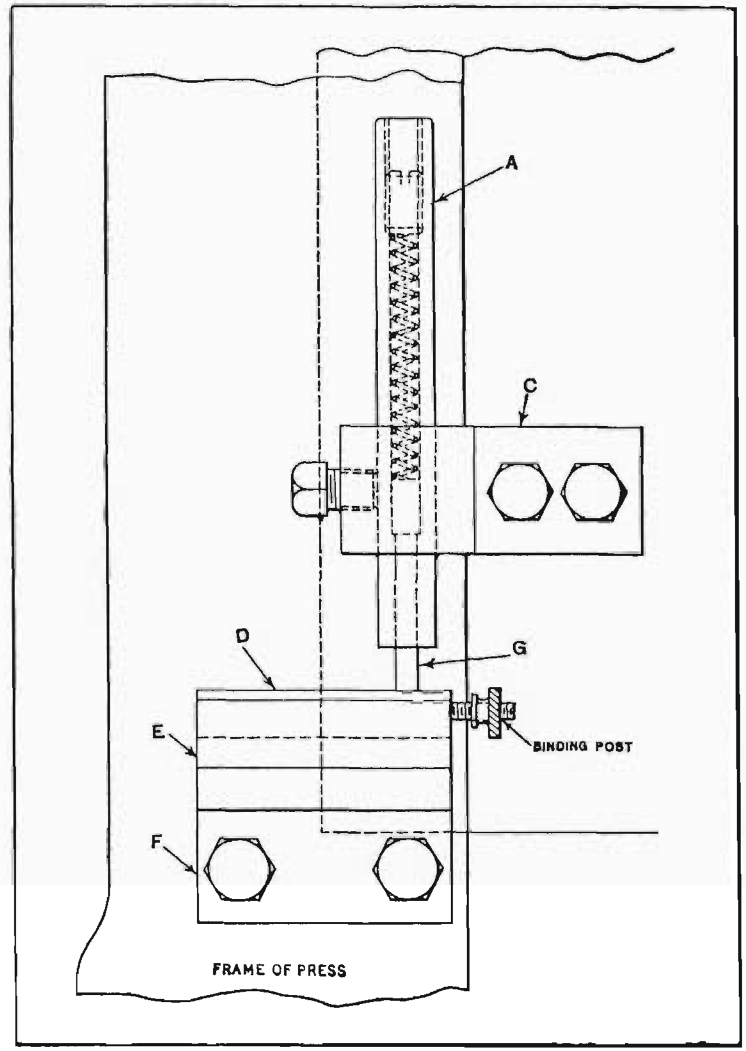


Fig. 7. Make-and-break Contact Device Used with Press Stop

The plunger-holder *A* is adjustable, however, so that this distance can be decreased or increased, as the nature of the work may require. The rod that connects the clutch key and treadle of the press was replaced by a tripping mechanism. The clutch trip lever *U* is connected to the lever *W* by rod *V*. The treadle is also connected to lever W by rod X. When the treadle is pressed down by the operator's foot in starting the press, lever W is caught and held under roll Y on the electric latch. The clutch trip lever U is thus drawn down so as to clear the clutch key. When the latch is released, the levers fly up into the position shown by the dotted lines. The clutch trip thus engages the key and stops the press. If it is desired to stop

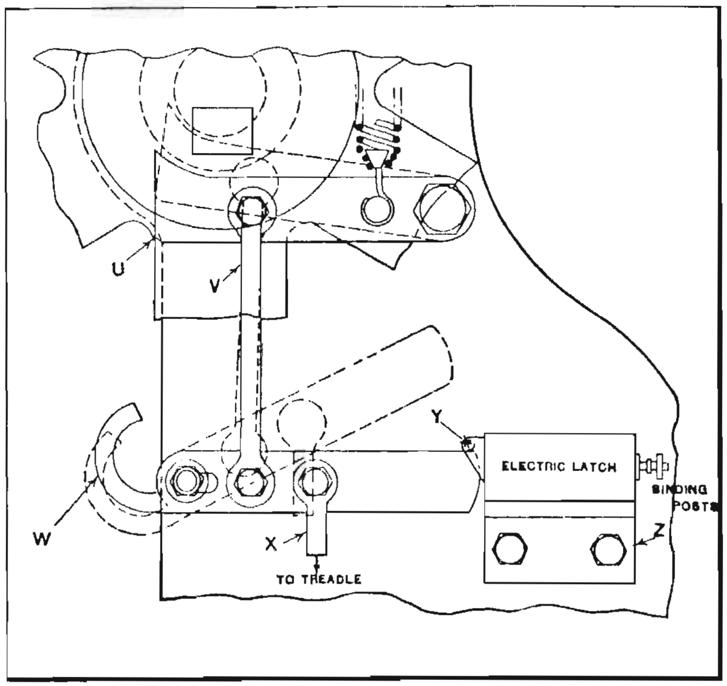


Fig. 8. Press Clutch Trip Operated by Electric Latch

the press without releasing the electric latch, all that is required is a slight pull on the hooked end of lever W which has an elongated hole in it that permits a lengthwise movement of about 3/8 inch.

The operation of the stop is as follows: Referring to Fig. 6, the metal is fed under punch O which cuts the blank and holds it in the drawing die P; drawing punch D then descends and draws the shell. If the bottom of the shell is not pushed

out, the press continues to operate. The make-and-break device shown in Fig. 7 makes a contact when the blanking slide is down within 1/16 inch of the bottom of its stroke and continues to maintain this contact until the slide is moved upward about 1/16 inch on the return stroke. The press is so timed that although the drawing slide has a longer stroke, both slides start upward together and maintain the same speed for about 1/4 inch of the upward stroke.

It is clear, then, that if the bottom of the drawn shell is not pushed out, the push-pin A, Fig. 6, will be held flush with the

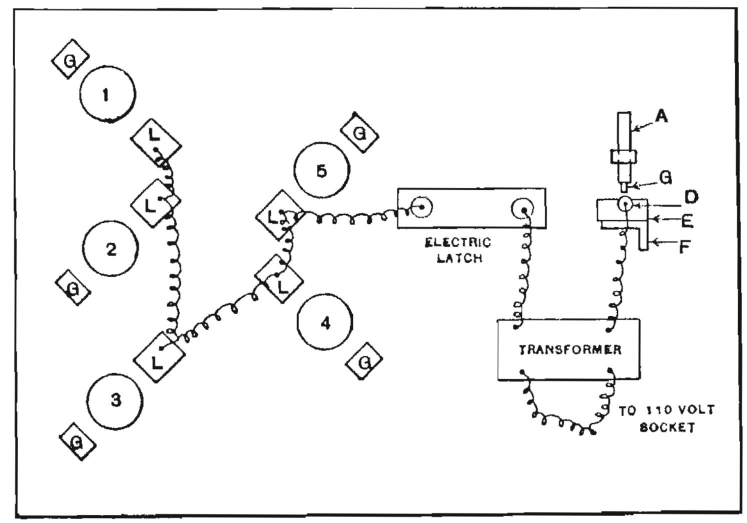


Fig. 9. Wiring Diagram of Electric Stop System on Gang Punch

bottom of the drawing punch. Pin A must project from the punch end about 1/8 inch in order for collar B to make contact with springs L and H. As the drawing slide returns, the shell is stripped from the drawing punch by the bottom of the die P, and before the push-pin can drop so that it projects 1/8 inch below the punch, the contact made by the make-and-break device is broken. It requires a movement of only 1/16 inch to break the contact so that it is impossible to trip the press automatically until contact is made again at the end of the next downward stroke.

If the bottom of the shell is pushed out on the drawing stroke, push-pin *A* is permitted to drop and the collar *B* to make contact with the springs *H* and *L*, which results in releasing the electric latch *Y* shown in Fig. 8. When the electric latch is released, the tripping levers on the press take the position shown by the dotted lines, causing the clutch trip to throw out the clutch and stop the press before another revolution is made. Some difficulty was experienced at first from the lubricant used on the metal being forced into the contacts and causing short circuits. A rubber washer *N*, Fig. 6, placed in the punch-block *F* prevented the lubricant from reaching the contact points and thus eliminated this trouble. When the end of a roll of metal is reached, the contacts act in the same way as when the bottom of the shell is pushed out, and stop the press.

CHAPTER VI

REVERSING MECHANISMS FOR ROTATING PARTS

A REVERSAL of motion is essential to the operation of many different forms of mechanism. Machine parts having a rectilinear or straight-line motion must, of necessity, reverse their movement, and many rotating parts also revolve first in one direction and then the other. The reversal in some cases is applied to a single shaft or slide and, in other instances, an entire train of mechanism is given a reversal of motion. The types of reversing mechanisms vary considerably, both as to principle of operation and as to form or design. Some are so arranged that the reversal of motion occurs at a fixed point in the cycle of movements, whereas, with other designs, the point of reversal may be changed by means of adjustable dogs or tappets which are attached to the movable part and control the action of the reversing mechanism. The adjustable type is required on some machine tools for varying the length of the stroke made by a cutting tool or machine table so that the stroke will conform to the length of the work. Reversing mechanisms also differ in that some are hand-controlled and others are operated automatically.

Intermediate Spur Gears for Reversing Motion. — A simple method of obtaining a reversal of motion by means of spur gears is shown at *A* and *B* in Fig. 1, where the reversing gears used on some designs of lathe headstocks are illustrated diagrammatically. The two intermediate gears *b* and *c* are mounted on a swiveling arm which can be adjusted for engaging either one of the intermediate gears with the spindle gear. When the gears are in the position shown at *A*, the drive is from *a* through *c* to *d*. When the arm carrying the intermediate gears is shifted as indicated at *B*, the motion is transmitted through both intermediate gears or from *a* through *b* and *c* to *d*, thus reversing the direction of rotation. This mechanism, as applied to a lathe, is used for reversing the rotation of the lead-screw when cutting left-hand threads.

Another method of obtaining a reversal of rotation by means of an intermediate gear is illustrated by diagram *C*. In this case, there are two sets of gearing between the driving and driven shafts. For the forward motion, the drive is from gear *e* to *f*. When the rotation of the driven shaft is to be reversed, gear *e* is shifted to the left and into mesh with the

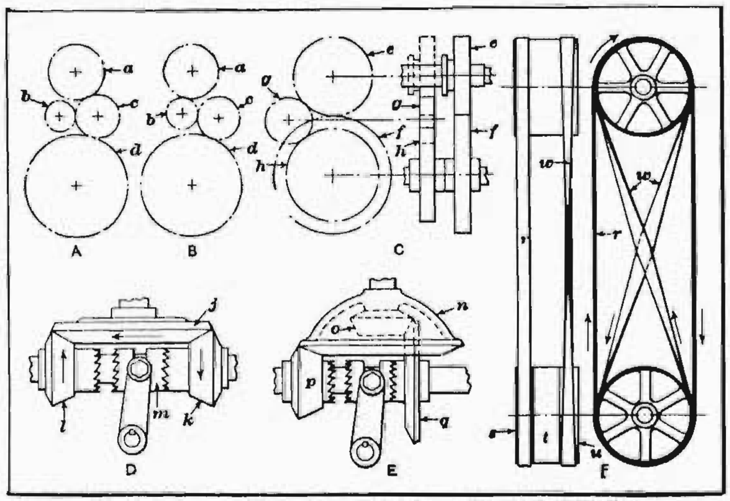


Fig. 1. Common Methods of Obtaining a Reversal of Motion

intermediate gear *g*, as shown by the dotted lines, so that motion is transmitted through *e*, *g*, and *h*. This general arrangement for obtaining a reversal of rotation is applied extensively to the transmission gearing of automobiles.

Bevel-gear Type of Reversing Mechanism. — A combination of three bevel gears, as illustrated by diagram *D*, Fig. 1, is applied to many different classes of mechanisms for obtaining a reversal of motion, especially when the reversing action is automatically controlled. With the usual arrangement, gear *j* is the driver and it is constantly in mesh with the bevel pinions *l* and *k*. These bevel pinions are loose upon the driven shaft and have a clutch *m* interposed between them. This

clutch is free to move endwise along the shaft, but it slides along a key or feather which compels it to revolve with the shaft. Each bevel pinion has teeth corresponding to clutch teeth, so that the engagement of the clutch with either pinion locks it to the shaft. Since these bevel pinions revolve in opposite directions, as indicated by the arrows, the rotation of the driven shaft is reversed as clutch *m* is shifted from one gear to the other. When the clutch is in the central or "neutral" position, it does not engage either gear, and no motion is transmitted to the driven shaft. Many of the reversing mechanisms which are equipped with this bevel gear combination differ in regard to the method of operating the clutch. For instance, clutch *m* might be shifted by the direct action of a slide or table having a rectilinear motion, or an auxiliary mechanism might be utilized to give the clutch a more rapid movement at the point of reversal. Some of these auxiliary features will be referred to later.

Two-speed Reversing Mechanism of Bevel-gear Type. — On some classes of machinery, it is desirable to have a relatively slow motion in one direction followed by a rapid return movement, in order to reduce the idle or non-productive period. One design of reversing mechanism of the bevel-gear type, by means of which a slow forward speed and a rapid return speed may be obtained, is illustrated at *E* in Fig. 1. In this case, there are two driving as well as two driven gears. The larger driver *n* is made cup-shaped so that a smaller driver *o* can be placed inside. When the clutch engages the smaller driven gear *p*, the fast speed is obtained, and, when the clutch engages gear *q*, the speed of the driven shaft is reduced an amount depending upon the ratio of the slow-speed gearing. Reversing mechanisms of this general type are not adapted for reversing the motion of heavy slides or work tables nor for fast-running machinery, because of the excessive shocks and stresses incident to a sudden reversal of movement in case of high velocities or heavy loads.

Reversal of Motion with Friction Disks. — When motion is transmitted between shafts located at right angles to each other by the type of frictional transmission shown in Fig. 11, Chapter XI, a reversal of rotation is easily obtained. As disk B is shifted inward along the face of disk A, the velocity ratio is gradually reduced, and when disk B passes the axis of disk A, the direction of rotation is reversed. This form of transmission has been applied to the feeding mechanisms of certain types of machine tools, and to other classes of machinery, especially where simplicity of design and ease of operation and control are essential factors. One method of arranging this form of drive, as applied to an automobile transmission, is to mount the driving member on a sliding shaft which enables the driving and driven disk to be readily disengaged, thus combining in one simple mechanism the clutching, speed-changing, and reversing functions.

Reversal from Open and Crossed Belts.—Shafts are often connected with open and crossed belts for permitting a reversal of rotation. The arrangement is illustrated by the diagram F in Fig. 1. There are three pulleys on the driven shaft. The central pulley t is keyed or attached to the shaft, whereas the outer pulleys s and u are loose and free to revolve upon the shaft. When the "open" belt r is shifted onto the tight pulley t, the driven shaft revolves in one direction and its rotation is reversed when the crossed belt w replaces the open belt on the tight pulley.

This form of drive is sometimes modified by having two pulleys on the driven shaft and a clutch interposed between the pulleys, so that either of them may be made the driven member. Thus, when the clutch is engaged with the pulley connecting with the open belt, the rotation is the reverse of that which is obtained when the clutch engages the pulley driven by the crossed belt. The countershafts for engine lathe and other machine tools which may require a reversal of movement are commonly arranged in this manner. Open and crossed belts are also applied to belt-driven planers for reversing the motion of the platen or work table. Many planer drives have pulleys which are so proportioned as to give a rapid return movement. A common arrangement is to place a central or tight pulley on the driven shaft which has two steps or diameters, the smaller one of which is for obtaining a fast return motion. Many modern planers are driven by motors which transmit power direct to the planer transmission.

Incidentally, belt drives of the type referred to are often used in place of gearing, for reversing heavy or fast running parts, because the belts slip somewhat if the load becomes excessive, due to the stopping and starting at the points of reversal, and this slipping action automatically protects the mechanism from injurious shocks or stresses.

Operation of Reversing Clutches.—When a reversal of motion depends upon the action of a clutch which is shifted from one gear to another revolving in an opposite direction, it is essential to operate the clutch rapidly and to secure a full engagement of the clutch teeth. Provision should also be made against disengagement of the clutch as the result of vibrations incident to the operation of the machine. There are two common methods of controlling the clutches used in connection with the bevel-gear type of reversing mechanism illustrated at D in Fig. 1. One form of control may be defined as the swinging-latch type and the other as the beveled-plunger type. The general principle of operation is the same in each case, and is as follows: When the work table, or whatever part is to be reversed, approaches the end of its stroke, a spring is compressed, and then a latch or trip allows this compressed spring to suddenly and rapidly throw the reversing clutch from one gear to the other. Reversing mechanisms of this general design are often called the "load-and-fire" type, because the spring is first loaded or compressed and then tripped to secure a rapid movement of the clutch and a reversal of motion at a predetermined point within close limits. The action of the compressed spring also insures a full engagement of the clutch teeth and prevents the clutch from stopping in the central or neutral position, which might occur if a spring were not used and the momentum of the part to be reversed were insufficient to carry the clutch across the space intervening between the two reversing gears.

Reversing Mechanisms of the Face Gear Type. — An interesting application of the face gear is shown in Fig. 2. The double spur pinion *B*, mounted on and keyed to shaft *C*, may be moved along its axis, thus engaging the "face gear" with step *D* as shown, or step *E* with the face gear at *F*. This is an economical and substantial means of reversing the direction of rotation.

A face gear may be cut on a Fellows gear shaper. The teeth are cut on a face at right angles to the axis of the gear. They are produced by a cutter provided with involute teeth

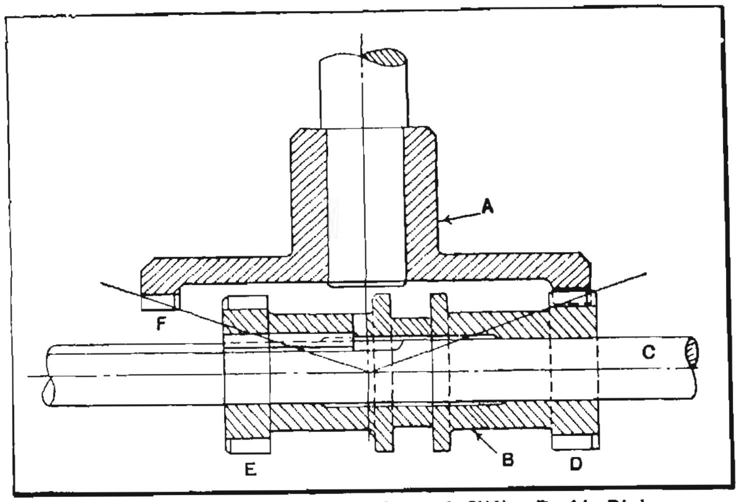


Fig. 2. Reversible Face Gear Drive with Sliding Double Pinion

corresponding in number and outline to the teeth in the pinion that is to mate with the face gear. The cutter and work are geared to rotate in the ratio of their respective teeth, and as the cutter reciprocates in contact with the work teeth are developed in the face gear by the molding-generating process.

In order to operate properly with the face gear, the axis of the mating pinion must be located at exactly the same distance from the face of the gear as the axis of the producing cutter. Proper tooth action is obtained, regardless of the axial position of the pinion, assuming, of course, that it is not moved out of engagement. This is an important point

since cone adjustment may be disregarded. These gears are adapted to ratios of 3 to 1 and higher. Drives of 1 to 1 ratio are unsatisfactory, since the active face width is too limited.

Latch Type of Reversing Clutch Control. — The reversing mechanism illustrated in Fig. 3 is a bevel-gear type equipped with the swinging latch form of clutch control. This mechanism is applied to a cylindrical grinding machine for reversing

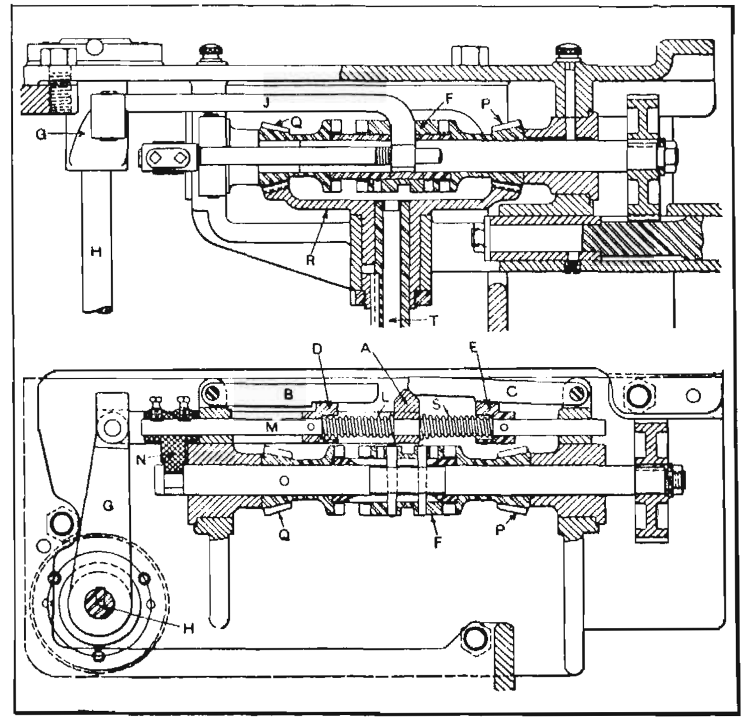


Fig. 3. Spring and Latch Type of Reversing Clutch Control

the motion of the work table, and is located at the rear of the machine. The rockshaft *H* extends through to the front of the machine and has attached to it a lever which is engaged by dogs on the work table, the distance between these dogs being varied according to the length of stroke required. At the rear end of rockshaft *H* there is a lever *G* which, by means of link *J*, transmits motion to the reversing mechanism. As the work table approaches the end of its stroke, lever G swings either to the right or left as the case may be. If the motion is to the left, tappet A, connected to link J, compresses spring L on the rod M and forces block D against a square shoulder on the lower side of latch B. Continued movement of tappet A to the left causes the beveled side of A to lift latch B, thus releasing block D, which, with rod M, is thrown rapidly to the left under the impulse of the compressed spring L.

After the movement of shaft M to the left, the shoulder on latch C drops in behind block E. The fork N on rod M also throws shaft O to the left and with it the reversing clutch F which is keyed to this shaft. The motion which prior to reversal was transmitted through bevel pinion P to the main gear R is now from pinion Q to R so that the movement of the work table is reversed. When the work table approaches the end of its stroke in the other direction, tappet A is moved to the right, thus compressing spring S. Then latch C is lifted by the beveled edge on A and the parts M, N, and O are quickly shifted to the right by spring S, thus again reversing the motion.

If the operator desires to stop the traversing movement at the end of the stroke, this may be done by the movement of a knob located in the center of the table-traversing handwheel at the front of the machine. This knob is connected with a plunger T which, by pressing the knob, may be held under pressure against the reversing clutch F. When this clutch is shifted at the end of the stroke either by springs L or S, plunger T drops into a groove in clutch F, thus holding it in central or neutral position. The knob previously referred to may be set at any part of the stroke to stop the traversing movement at the end of that stroke. The withdrawal of the knob again starts the traversing movement without requiring any further action on the part of the operator. The shaft connecting with bevel gear R extends to the front of the machine and, through suitable gearing, transmits a rectilinear motion to the work table of the grinding machine. This mechanism is an example of the "load-and-fire" type referred to.

Beveled Plunger Control for Reversing Clutch.— An example of the beveled plunger type of clutch control for a reversing mechanism is shown in Fig. 4. This design is also intended for a cylindrical grinding machine. The point of reversal is controlled by the tappets A which are adjusted along the work table to vary the length of the stroke. These tappets alternately engage lever B at the ends of the stroke and, by swinging this lever about its pivot, shift bar C which transmits motion to the reversing clutch. If the work table is moving toward the right, the tappet at the left engages

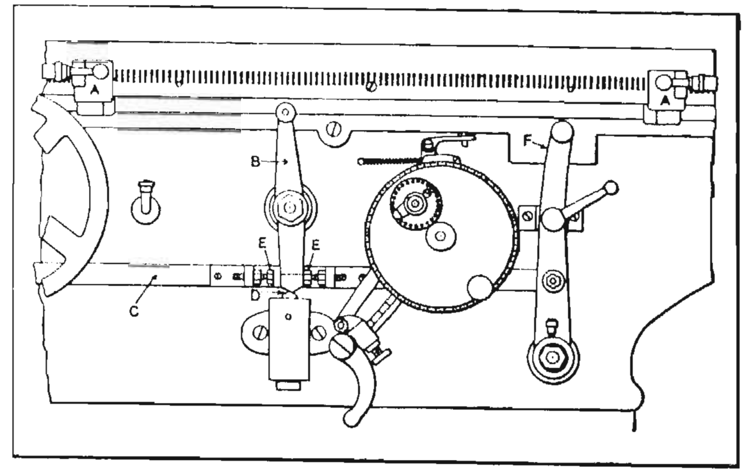


Fig. 4. Spring and Beveled Plunger Control for Reversing Clutch

lever B as the table approaches the end of its stroke. The movements of the lower end of reversing lever B towards the left forces the beveled plunger D downward, thus compressing a spring that is located beneath it. When the point of the V-shaped end of lever B has passed the point of plunger D, the latter is suddenly forced upward by the compressed spring and lever B, rod C, and the reversing clutch are shifted rapidly.

There is a certain amount of lost motion between the studs E on bar C and the reversing lever B. As the result of this lost motion, the clutch is not entirely disengaged until the V-shaped point of the reversing lever has passed the point of plunger D; the reversing clutch is withdrawn slowly from the bevel pinion which it engages until the sudden action of plunger D causes it to shift rapidly into engagement with the opposite bevel pinion. The clutch is held in engagement until the next reversal of motion by the upward pressure of the plunger against the beveled end of the reversing lever B. With the particular design illustrated, the point of reversal can also be controlled by hand lever F which is connected to rod C; by placing this lever in a central position, the clutch is shifted to neutral and the movement of the work table discontinued.

Controlling Point of Reversal by Special Mechanisms. — The points of reversal for a reciprocating slide are usually controlled by trip dogs mounted directly on the slide and adjusted to give the required length of travel or stroke. It is not always convenient, however, to control the reversal in this way. For instance, if the operating slide is at the rear of a machine where the trip dogs cannot be adjusted readily, some form of mechanism which operates in unison with the slide may be used to permit locating the trip dogs at the front of the machine. A simple method of controlling the points of reversal from the front of the machine is applied to a certain design of cylindrical grinding machine. The wheel slide travels along ways at the rear of the machine and the length of stroke is regulated in accordance with the length of the work by two trip dogs mounted on a wheel or circular rack at the front of the machine. The shaft carrying this wheel extends through the machine and is connected by gearing, so that it has an oscillating or turning movement in first one direction and then the other, which movements correspond to, and are in unison with those of the wheel carriage at the rear. Worm teeth are formed on the periphery of the trip-dog wheel and the dogs are held in position by worms which may be lifted out of engagement when the dogs are to be adjusted considerably. The dogs alternately strike a tappet or lever which controls the movements of the reversing clutch.

Another method of controlling the reversing points of a rear slide is by means of a shaft connected through gearing

with the reciprocating slide and having at the front end a pinion meshing with a sliding rack carrying the trip dogs. As the rear slide operates, it turns the pinion shaft in first one direction and then the other, which imparts a reciprocating motion to the rack. The trip dogs attached to the rack, by engaging a lever, cause a reversal of motion by means of a clutch-and-gear type of reversing mechanism.

An indirect or independent method of controlling the points of reversal on an automatic bevel gear cutting machine is

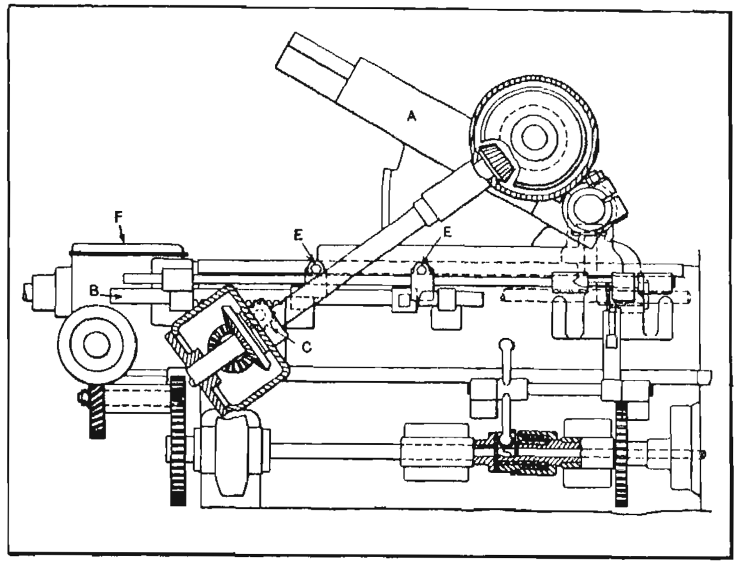


Fig. 5. Independent Method of Controlling Reversal of an Adjustable Slide on a Bevel Gear Cutting Machine

illustrated in Fig. 5. The cutter-slide A must be set at an angle corresponding to the inclination of the gear teeth to be cut, so that it would be difficult to have the trip dogs attached directly to this slide. To avoid such an arrangement, a sliding rack B is employed. This rack meshes with a pinion C which rotates in unison with the feeding of the cutter-slide, since this pinion and the slide derive their motion from the same shaft. As pinion C rotates in first one direction and then the other, it traverses the rack B, which, by means of the adjustable dogs

E, controls the action of the reversing mechanism enclosed at *F*. With this arrangement, the traversing movement of the rack can be made less than the travel of the cutter-slide, if this is desirable because of limited space. On the other hand, if the traversing movement of the slide is to be very short and it is essential to reverse it at a given point within close limits, the movements of the reverse controlling rack can be

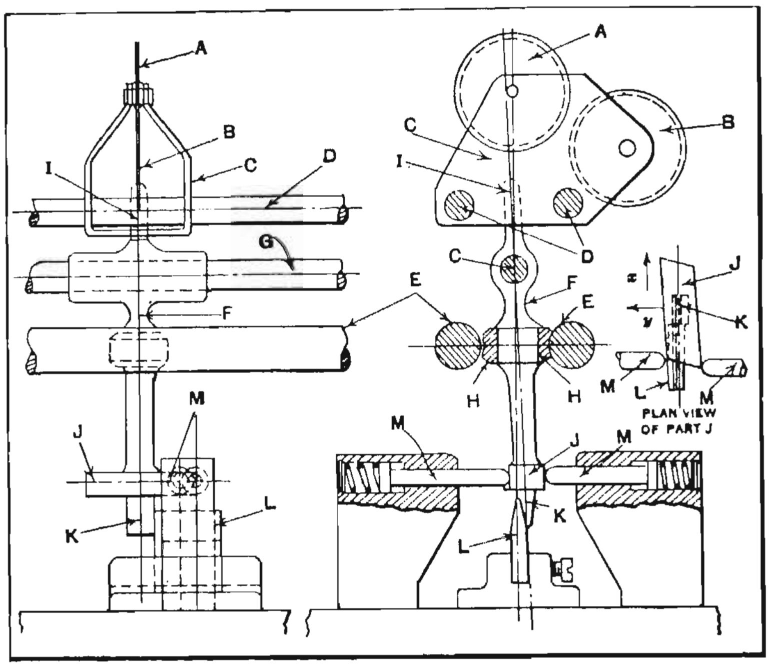


Fig. 6. Quick Reversing Mechanism of Wire-coiling Machine, which Prevents Turns of Wire from Piling Upon One Another at the Coil Ends

increased considerably as compared with the motion of the cutter-slide.

Quick-reversing Mechanism for Wire Coiling. — The rapid reversing mechanism to be described is embodied in a machine for winding coils having a number of layers one upon the other, as used in the electrical trade. As the device that guides the wire on the coil arrives opposite one end of the coil, instantaneous reversal of direction is necessary in order to

avoid the turns of wire piling upon one another. The results obtained with this reversing mechanism are highly successful.

The wire is fed from the supply drum over pulley *A* (see Fig. 6) and under pulley *B* and then on the coil spool. As the latter is not driven in a special manner, it is not shown. The two pulleys are kept very thin, so as to get close up to the flanges of the spool. These V-pulleys are mounted in the carriage *C*, which is capable of sliding along the guide rods *D*. Beneath these rods are located two lead-screws *E* of a pitch equal to that of the wire when wound on the spool. These are rotated in opposite directions.

Suspended between the lead-screws is the lever *F*, which slides along the guide rod *G*. Embodied in this lever are two half-nuts *H* which, as the lever swings from one side to the other, engage alternately with the lead-screws *E*. Thus the lever will travel to the right or left according to which screw it is in mesh with.

The top of the lever carries an extension *I* which engages in a slot in the base of the carriage *C*, thus giving it the necessary motion. It will be obvious that the peg *I* must be free to swing as the lever is moved, but must have no side play. Lower down on the lever is fixed a hardened steel rhombus-shaped part *J*. This is located at an angle, as seen more clearly in the plan view. Beneath this is a hardened steel wedge *K* which is kept in contact with another inverted wedge *L* by means of the spring plungers *M*.

It will be seen, then, that when one half-nut is in contact with a lead-screw the steel wedges either slide or are ready to slide against each other. The method of securing the reversal will be more easily seen by examining the plan view of part *J*. Here part *J*, fixed to the lever, has moved along in the direction of arrow x. By so doing and due to its angular location, it has compressed the right-hand plunger. At the same time it has reached the end of its traverse and the two wedges *K* and *L* are about to separate.

Thus, at this moment, part *J* will be forced by the plunger, in the direction of arrow y, and the half-nut will mesh with the opposite lead-screw. The lever *F* will then immediately reverse its direction, and the cycle indicated will be repeated.

For winding coils of varying length, it is merely necessary to increase or decrease the length of the lower wedge *L*. As this device is automatic, except for the placing and removing of bobbins, and the starting of the wire, one operator can take charge of several wire coiling machines of this type.

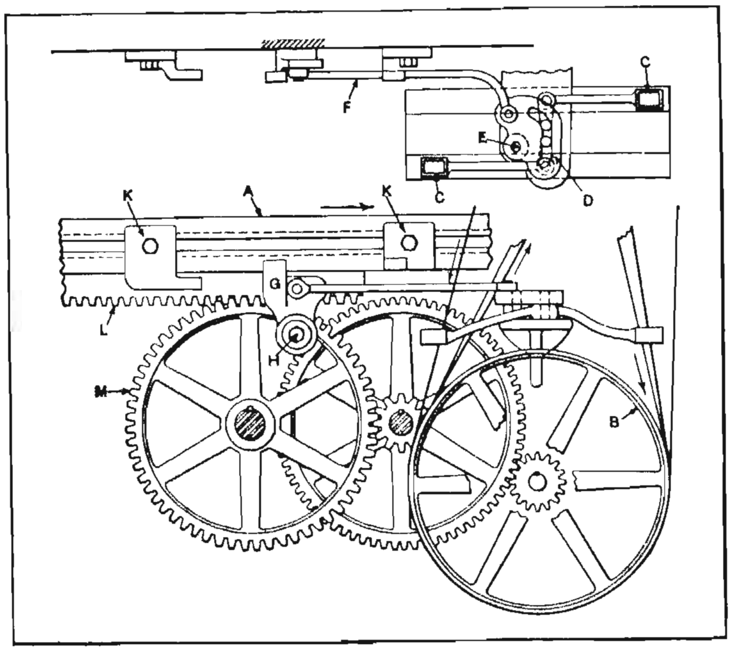


Fig. 7. Reversing Mechanism of a Belt-driven Planer

Mechanism for Shifting Open and Crossed Belts.— The open and crossed belts illustrated by diagram *F*, Fig. 1, are shifted automatically for obtaining a reversal of motion, when used to drive such machines, as planers, broaching machines, or other classes of mechanisms which are designed for continuous operation and equipped with this form of drive. A side elevation and plan of the automatic belt-shifting device used on a planer is illustrated in Fig. 7. The shaft on which the belt pulleys *B* are mounted transmits motion to the planer

table *A* through a train of gearing which gives a suitable speed reduction. In order to reverse the motion of the work table, this entire train of gearing is reversed by alternately shifting the open and crossed belts onto the central pulley, which is attached to the shaft. The length of the stroke is governed by the distance between the two dogs *K* which may be adjusted along a groove at the side of the table. The position of each belt is controlled by a guide *C* having an opening at the end through which the belt passes. These two guides or shifters, which are in the form of bellcranks, are pivoted and the inner ends carry small rollers that engage a groove in the cam-plate *D*. This cam-plate is pivoted at *E* and is connected by a link *F* with the arm or lever *G* which is pivoted at *H*.

When the planer is in operation, the table moves in a direction depending upon which belt is on the tight pulley. When this movement has continued far enough to bring one of the dogs *K* into contact with arm *G*, the latter is pushed over about its pivot, thus imparting a swinging movement to the cam-plate *D*. The groove in this cam-plate is so formed that the belt on the tight pulley is shifted to the loose pulley and the other belt is moved over to the driving position on the tight pulley. At the end of the return stroke, the other dog engages arm *G*, thus swinging the cam-plate in the opposite direction and again reversing the motion.

Reversal of Motion through Epicyclic Gearing.— A train of epicyclic or differential gearing may be designed to give a reversal of motion. This form of transmission has been applied to some automobiles of the smaller sizes. The principle governing the operation of one of the earlier designs is shown by the diagram, Fig. 8. Two sets of differential gears, indicated at *A* and *B*, are mounted inside of drums. These drums may be revolved independently for obtaining the slow forward speed and a reverse motion, or they may be locked together so as to revolve as a unit with the crank-shaft for obtaining the direct high-speed drive. The central gear *a* is the driver in each case, and is keyed to the crank-shaft. The slow forward speed is obtained with the combination illustrated at *A*. To obtain a reduction of speed, the internal gear *b* is held stationary by the application of a brake-band to its periphery; the pinions *c* carried by the driver member are then forced by the driving gear *a* to roll around inside of the internal gear, thus transmitting a slow rotary motion to the driven member attached to the pinions. In order to obtain a reversal of motion through the combination of gearing illustrated at *B*, the disk carrying the pinions is prevented from rotating by the gripping action of another brake-band, so that the pinions merely revolve on their studs and rotate the internal gear in a reverse direction. In this case, the in-

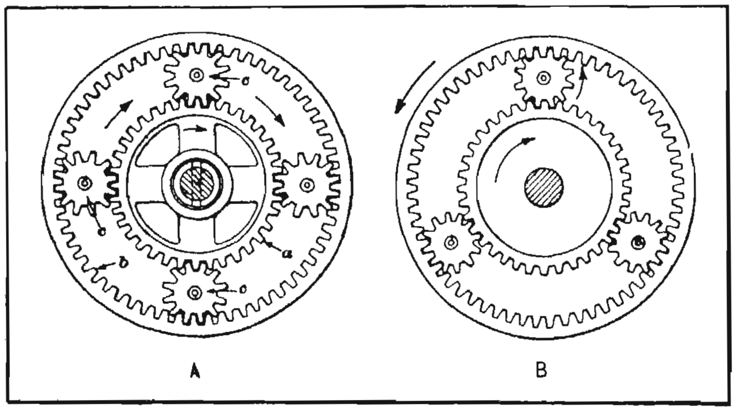


Fig. 8. Diagram Showing Arrangement of Epicyclic Gearing for Obtaining Forward and Reverse Motions

ternal gear is the driven member and transmits motion to the driving sprocket.

A reversal of motion may also be obtained with the train of epicyclic gearing shown in Fig. 9. In this case, there is no internal gear. Gear *A* is mounted on the sleeve of sprocket A_1, gear *D* is keyed to shaft *K*, and gear *F* is attached to the extended hub of drum *H*. The three gears, *B*, *C*, and *E* are locked together and revolve upon a pin carried by drum *G*. A duplicate set is also located on the opposite side of the drum, as the illustration shows. When this drum is held stationary by a brake-band, gear *A* and sprocket A_1 are driven at a slow

forward speed through gears *D*, *C*, and *B*, gears *D* and *A* revolving in the same direction. The direct high-speed drive is obtained when clutch *J* is engaged, the whole mechanism then revolving as a unit with shaft *K*. When drum *H* is held stationary by a brake-band, gear *D* causes gear *E* to revolve about the stationary gear *F* in a direction opposite to the rotation of *D;* consequently, gear *A* is forced to follow in the same direction in which drum *G* and the planetary gears *B*,

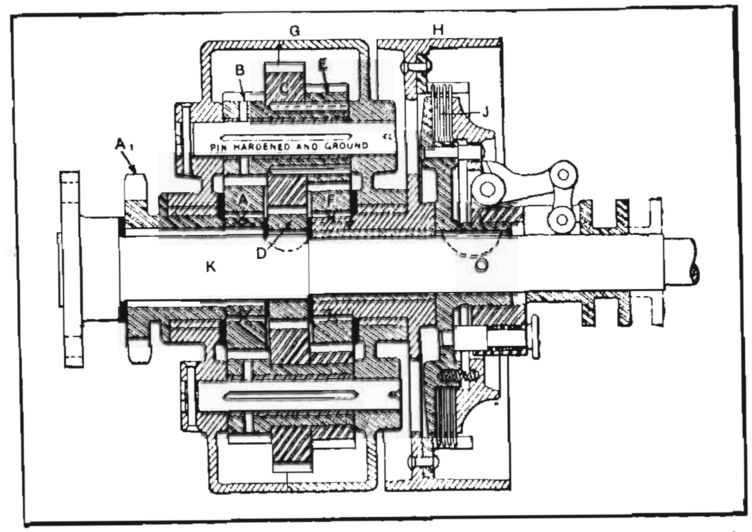


Fig. 9. Another Arrangement of Epicyclic Gearing which Gives Forward and Reverse Motions

C, and *E* are moving, thus reversing the motion of gear *A* and the sprocket.

Operation of Reversing-ratchet Pawl. — Fig. 10 shows a simple change in a ratchet movement which resulted in a more positive action. Originally, the arrangement was as follows: The ratchet wheel *A*, which is keyed to shaft *F*, served to impart motion to a sliding table (not shown). The motion of the shaft *F* was imparted to the table by means of gearing engaging with a rack on the under side of the table. The lever *B* was given an oscillating motion by a crank, to which it was connected by the connecting-rod *E*. The pawl

C, pivoting on the stud I, engaged the ratchet wheel A, the weight H serving to keep the pawl engaged. Originally pawl C was not slotted as shown, but carried pin K, the part D with its weight G not being used. As the table moved along, a dog on the edge struck pin K, and swung pawl C to the opposite

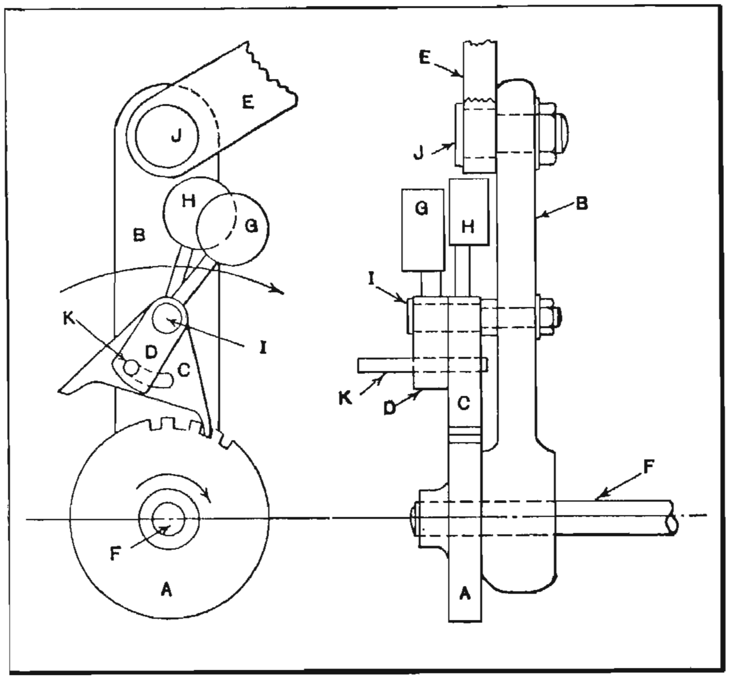


Fig. 10. Reversing Ratchet, the Pawl of which is Operated by a Swinging Weight

side, reversing the rotation of shaft F and the movement of the table.

In general, this arrangement operated satisfactorily, except on rare occasions, when the position of the dog on the table would be such as to swing pin K just far enough to balance pawl C, when the movement of the table would stop until reversed by the operator.

In order to overcome this condition, pawl C was slotted as shown, and the part D was added, pin K being a press fit in part D and sliding in the slot in pawl C. With this arrangement, pawl C, operating independently of part D, remains in contact with ratchet wheel A until the table has moved far enough to push pin K over so that weight G falls on the opposite side, the unbalanced effect causing pawl C to swing on stud I, and engage on the opposite side, producing a more positive effect than by swinging the pawl direct.

Automatic Ratchet Reversing Mechanism. — The simple design of ratchet reversing mechanism illustrated in Fig. 11 enables a ratchet wheel to be automatically reversed after making a predetermined number of revolutions, and the arrangement is such that the time of reversal or the number of revolutions made by the driven ratchet prior to reversal may be varied at will throughout a wide range. The double pawl A is carried by an oscillating arm (not shown), and this pawl engages the driven ratchet B. Mounted concentrically with B there is a smaller controlling ratchet C which is normally restrained from rotating by suitable frictional resistance. The larger diameter of ratchet B prevents pawl A from engaging the smaller ratchet C, except when the deep notch D is reached by the pawl which then drops down into engagement with C.

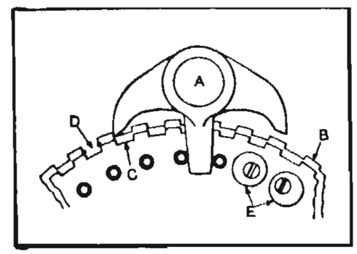


Fig. 11. Ratchet Mechanism which will Automatically Reverse after Making a Predetermined Number of Revolutions

The reversal of motion is effected by the engagement of the extension on pawl A with one of the trip dogs E. The number of revolutions made by ratchet B prior to reversal depends upon the number of deep notches D and the position of the trip dogs E. When this mechanism is in operation, ratchet B receives an intermittent motion from the oscillating pawl A and the controlling ratchet C remains stationary until one of the deep notches D is engaged by pawl A; then ratchets B and C rotate together an amount depending upon the motion of the pawl. Controlling ratchet C then remains stationary until another deep notch is engaged. The repeated movements of ratchet C each time the pawl drops into a deep notch, finally bring one of the trip dogs E into contact with the projection on the pawl; the latter is then swung around so that its opposite end engages ratchet B and, consequently, the direction of rotation is reversed. The time of reversal may be controlled by varying the distance between the trip dogs and by having one or more deep notches in the driven ratchet.

Combined Reversing and Feeding Movements.— Some reversing mechanisms are so designed that the longitudinal movement of a reversing rod is accompanied by a rotary

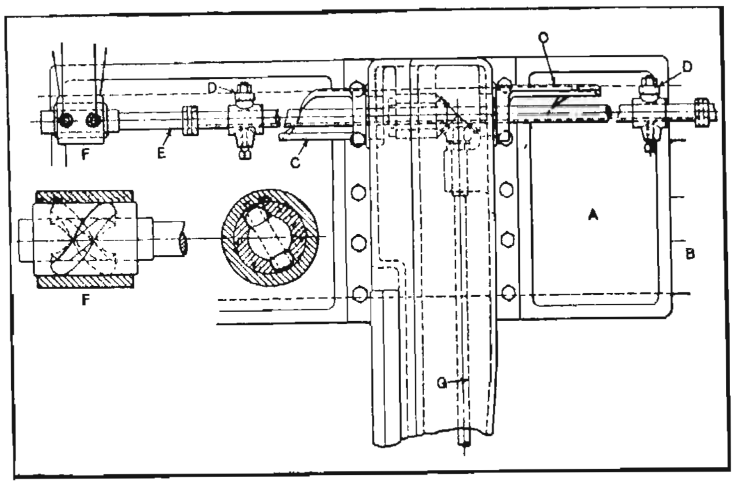


Fig. 12. Reverse Controlling Mechanism so Arranged that Motion of Reversing Rod is Accompanied by a Rotary Movement for Feeding Tool

motion for imparting a feeding movement at the time reversal occurs. A reversing device of this kind, as applied to a Richards' side-planing machine, is illustrated in Fig. 12. The saddle A is traversed along the bed B by means of a screw, the rotation of which is reversed by open and crossed belts that are alternately shifted from loose pulleys to a tight pulley attached to the screw. The two projecting arms C which are bolted to A strike dogs D mounted on rod E, which, by its longitudinal movement, actuates the belt-shifting mechanism. When rod E is shifted in a lengthwise direction, it is also given a rotary motion in the following manner: Within the

bearing F there is a bushing having cam grooves cut into it, as shown by the enlarged detailed view. These grooves receive rollers carried by a pin that passes through the rod E. With this arrangement, any endwise movement of rod E is accompanied by a rotary motion resulting from the engagement of the rollers with the helical grooves in the fixed bushings of bearings F. This rotary movement is transmitted through bevel gears to a rod G which imparts a downward feeding movement to the feed-screw of the tool-slide, through the medium of ratchet gearing.

Automatic Control of Spindle Reversal.— Fig. 13 represents a sectional view through the bed of an automatic screw machine, beneath the headstock, and illustrates the mechanism for automatically controlling the reversal of the spindle rotation. This machine is driven by a single belt pulley rotating at constant speed. The various movements of the machine, other than revolving the spindle, are derived from a shaft at the rear which rotates at a constant speed. On this shaft H is mounted a series of automatically-controlled clutches which are similar in action to those used on punch-presses. These clutches control the feeding of the stock, the opening and closing of the chuck, the revolving of the turret, the reversing of the main spindle and the changing of the speed from fast to slow, or *vice versa*. This back-shaft H is connected by change-gearing through a worm drive, with a slow moving camshaft A at the front on which are mounted the cams for the turret and cross-slide movements and a series of dog carriers B carrying tappets or dogs which control the action of the different clutches on the back-shaft. The ratio of the change-gears previously referred to determines the duration of the cycle of operations and, consequently, the length of time it takes to make a given piece of work.

The main spindle is reversed by a clutch located between two clutch members revolving in opposite directions. The carrier B has an annular T-slot in which adjustable dogs like the one shown at C are mounted. These dogs engage a tappet D on lever E, the rear end of which carries a screw F, the cylindrical point of which enters a cam groove in clutch G. This clutch is mounted loosely on shaft H which revolves continuously. A plan view of the cam is shown in detail above the end view. The cam groove is exactly the same on the other side as on the side shown, the clutch being arranged to engage each half revolution and then automatically disengage. The normal position of the pin F is in the recess at a. When it is lowered entirely out of the groove by the action of dog

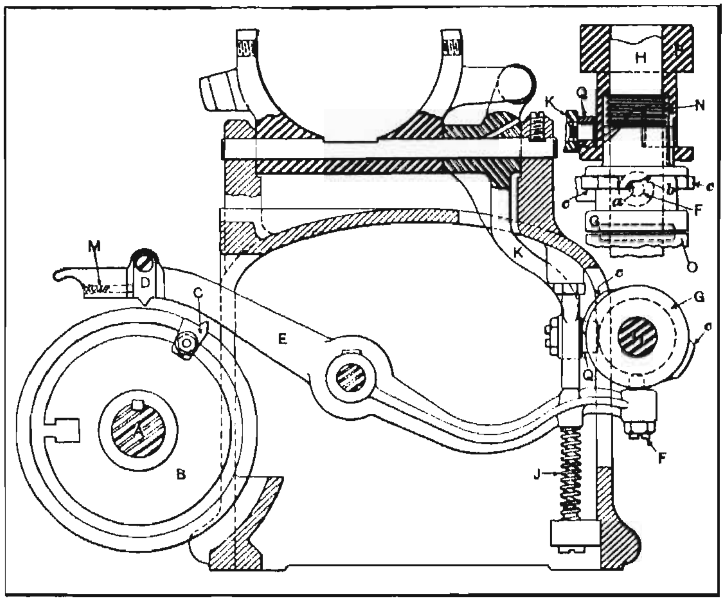


Fig. 13. Arrangement for Automatically Controlling Spindle Reversal

C on tappet D against the pressure of spring J, this releases clutch G, which is forced forward by a spring N coiled about the shaft, until it engages a mating member O, fastened to shaft H, and begins to revolve. Meanwhile dog C has passed tappet D, allowing pin F to drop into the cam groove again. The clutch G, as it revolves, brings inclined face b of the groove (or a similar incline on the opposite side) into contact with F, and the continued revolution of G, through the action of this inclination on the pin, forces the clutch teeth out of engagement, stopping G again with the pin in position

a as at the start. A cam P, also loose on the shaft H, is keyed to G. This cam engages a roll Q on the end of lever K, which operates a clutch fork, controlling the position of the main spindle clutch. When it is time to again reverse the spindle, another dog C is set in the proper position, and the clutch is tripped, revolving for a second time a half revolution and stopping, thus operating lever K and the spindle clutch to change the direction of the spindle rotation. This represents the normal procedure in cases where the time taken to make one piece is short enough so that the rotation of dog carrier B is reasonably rapid. For many pieces, however, this movement is so slow that dog C does not come out from under tappet D in time to allow pin F to drop into the cam groove before the clutch has made the required half revolution. In such cases, incline b would pass without disengaging the clutch and pin F could not enter until the next recess came around and the next incline b; hence the clutch would be stopped at the end of one revolution instead of a half revolution. This difficulty has been very simply overcome by the following means:

Tappet D is pivoted to lever E as shown, and is forced back against a shoulder to the position indicated, by a spring M located in a drilled hole and pressing against a plunger bearing on D. This spring is of such strength as compared with spring J that the first effect of dog C, when it strikes D, is to move the latter backward without raising lever E. When D has been pressed so far back that it strikes the shoulder at the left, further movement being impossible, E is raised, pin F is withdrawn from the cam slot in the clutch G, and the latter is allowed to engage fixed member O on the shaft H, and starts to revolve. A cam surface c is provided on G which, immediately after the clutch begins to rotate, strikes pin F and depresses it still further, thus raising tappet D clear above the point of dog C, and allowing it to swing back to its normal position against the shoulder at the right under the influence of spring M. Lever E is then ready to drop instantly, as D and C are entirely clear of each other. As soon as the end of cam projection c passes, F drops into the groove and the rotation of the cam is arrested after a half revolution, as required. When it is known that shaft H revolves at 120 revolutions per minute, so that the half revolution of G occupies but one-fourth second, it will be seen that the device has a difficult duty to perform, but operates in a very satisfactory manner.

Automatic Variation in Point of Reversal. — One of the many interesting mechanisms found on textile machinery is the one employed on fly frames for controlling the winding of the roving on the bobbin. The bobbins are driven at a decreasing rate of speed as the diameter increases and they not only revolve but are given a vertical reciprocating motion, in order to wind the roving onto them in successive helical layers. This winding of the roving onto the bobbin involves, in addition to decreasing the speed as the diameter increases, a decrease in the traversing speed of the bobbin and a gradual shortening of the bobbin travel as one layer of roving is wound upon another. The bobbin should move a distance equal to the diameter of the roving while it rotates relative to the "flyer" a distance equal to one revolution; therefore, as the bobbin speed gradually diminishes, it is also necessary to decrease the rate of traverse, so that each layer of the roving will be coiled closely. The change in the point of

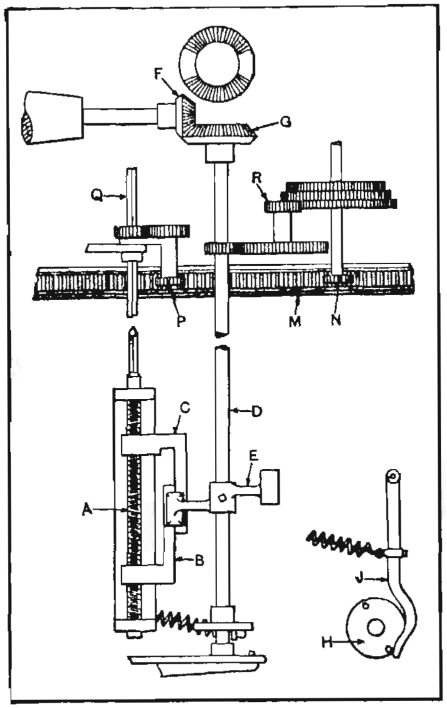


Fig. 14. Mechanism for Varying Point of Reversal and Speed of Rotation

reversal in order to shorten the stroke as the bobbin increases in diameter is required in order to form conical ends on the wound bobbin and a firm winding that will not unravel and cause trouble, such as would be the result of attempting to wind each layer the full length of the bobbin. These changes occur simultaneously, although they will be referred to separately in describing the "builder motion" illustrated diagrammatically in Fig. 14.

The plates B and C engage a screw A, which has a right-hand thread extending along one-half its length and a left-hand thread, along the remaining half. These plates and the screw are traversed vertically with the bobbin carriage. The vertical shaft D carries a dog E having two arms located 180 degrees apart. At each end of the stroke, shaft D makes a half turn which motion is utilized for reversing the motion, for shifting the cone belt slightly in order to decrease the speed, and also for shortening the stroke of the bobbin. As the plates B and C move vertically, one end of the tumbling dog E bears against them until it slides off at one end. Prior to the disengagement of the dog with one of the plates, gear F on the cone-pulley shaft revolves idly in a space on the rim of gear G where the teeth are omitted. There are two of these spaces located 180 degrees apart, as the illustration indicates. One of the projecting pins on the disk H at the lower end of shaft D is in engagement with a lever J which has attached to it a spring that holds the lever aganst a pin and tends to turn shaft D. When dog E slides off one end of a plate, shaft D is turned far enough by the action of the spring and lever J to bring gear G into mesh with pinion F; consequently, gear G is revolved one-half turn or until pinion F engages the space on the opposite side where there are no teeth. The partial rotation of shaft D shifts the reversing gears through a connection at the lower end and starts the bobbin carriage and plates B and C in the opposite direction. As the opposite end of the tumbling dog E swings around, it engages one of the plates and again causes a reversal of motion as it slides off of the opposite end.

The shifting of the belt on the cone-pulley at each reversal, for gradually decreasing the speed as the bobbin winding increases its diameter, is obtained by connecting rack *M* with shaft *D* through the pinion *N* and the train of gearing shown. This rack *M* has a fork attached to it that connects with the cone-belt, and it is traversed slightly each time dog *E* slides off a plate and allows shaft *D* to turn one-half revolution. The reduction in the length of the carriage traverse is obtained by revolving screw *A* at each reversal and thus shortening the distance between the plates *B* and *C*. This rotation of the screw is effected by pinion *P* which engages rack *M* and transmits motion through the other gears shown to the extension *Q* on the screw, which is made square and is free to slide through the gear hub as the carriage moves vertically. As the plates *B* and *C* are moved toward each other, the tumbling gear *E* has a shorter surface to traverse before it is disengaged. These two plates both move the same distance, so that the point of reversal decreases at each end and the bobbin is wound conical at both ends. The roving delivered by the front roll is either tightened or slackened by engaging pinion *R* with one of the three gears shown.

Oscillating Mechanism which Varies Point of Reversal.— The intermittent reversing mechanism shown in Fig. 15 is an important part of a power-driven valve-grinding machine designed for automatically grinding-in any desired number of valves at one time. The grinding movements imparted to the valve consist of rotating the valve approximately two-thirds of a revolution in one direction, then reversing the direction of rotation for one-third of a revolution and so on, the point of reversal being constantly changed, a feature which tends to prevent the cutting of grooves by the abrasive grit.

In addition to the reversing or oscillating movements, the valve is raised intermittently to allow the grinding or abrasive material to flow in between the seat and the valve. The various parts of the mechanism are designated by the same reference letters in the different illustrations.

The shaft *C* of the reversing mechanism imparts the oscillating movements to the valve-grinding spindles through bevel gears which are not shown as they do not affect this mechanism. A shaft driven from the reversing mechanism, is

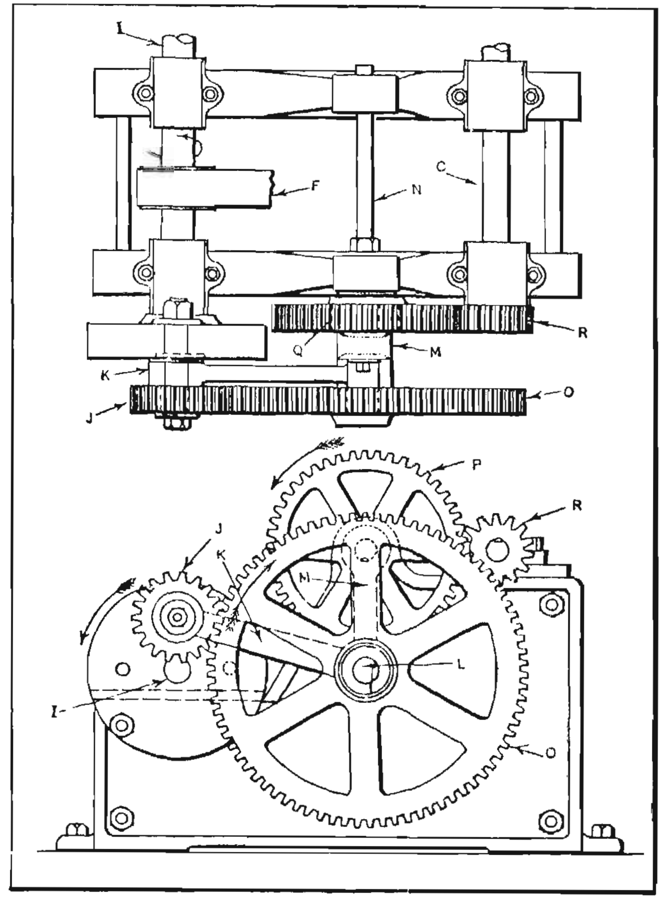


Fig. 15. Intermittent Reversing Mechanism

equipped with cams which transmit the intermittent valve-lifting movements to studs in contact with the valve stems.

Shaft *I* is connected with the lineshaft by a direct belt drive and is revolved continuously in the direction indicated by the arrow. Keyed to one end of shaft *I* is a disk having a small gear J secured to it in an off-center position with respect to the shaft. The connecting-rod or link K has a bearing on the solid stub shaft of gear J at one end, and a bearing on the shaft L at the other end. Shaft L is journalled on an oscillating arm or pendulum M, the upper end of which pivots on shaft N. The gear O keyed to shaft L is thus kept in

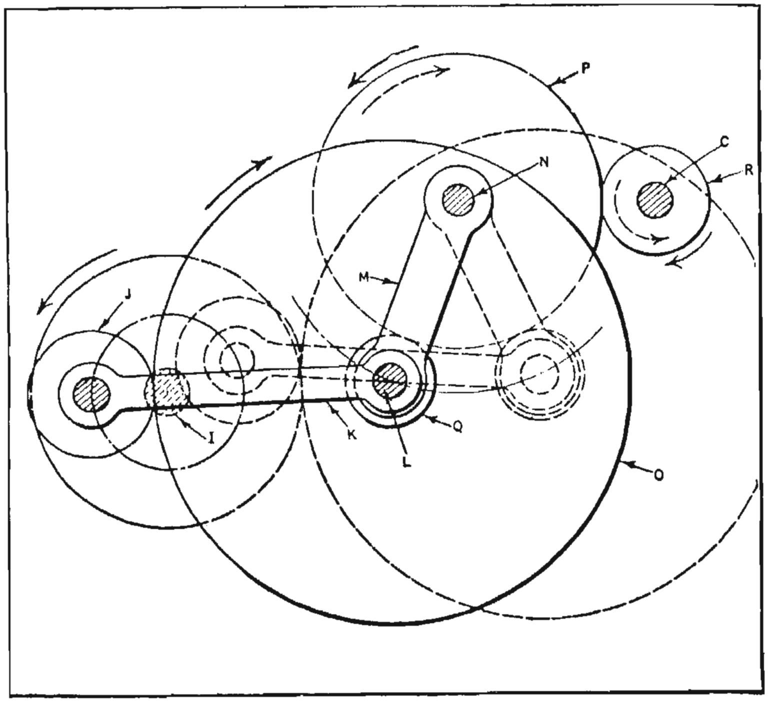


Fig. 16. Diagram of Mechanism Fig. 15, Showing Parts at Beginning of Forward Driving Movement in Full Lines and at Beginning of Reverse Movement in Dotted Lines

mesh with gear J at all times and has an oscillating movement similar to the pendulum of a clock, which results from the movement given shaft L by connecting-rod K.

In addition to the oscillating movement, gear O, with its shaft L, is rotated slowly by the driving gear or pinion J, one revolution of shaft I being equivalent to one revolution of gear J under the usual operating conditions. Keyed to

the rear end of shaft L immediately back of the oscillating arm M, is a pinion Q (see Fig. 16) which meshes with a gear P journalled on shaft N on which arm M is pivoted. The slow drive transmitted to gear O by gear J will also be transmitter to gear P. In addition to this drive, gear P is also revolved by the oscillating movement of gear Q keyed to shaft L, being revolved first in one direction and then in the other. The design of the gearing is so calculated that when the connecting-rod K moves forward, gear P will also be revolved forward; that is, when the connecting-rod K moves from the position shown by the full lines in Fig. 16, to its extreme position to the right, there will be added to the slow driving movement of pinion J, through gears O and Q, the driving action of the pendulum or swinging movement also transmitted by the teeth of gear Q. Thus, gear P is driven forward in the direction indicated by the full line arrow until the connecting-rod starts its backward movement at approximately the position shown by the dotted lines in Fig. 16.

On the back stroke of rod K, gear P will be revolved in the opposite direction, as the driving speed in the reverse direction, resulting from the oscillating movement, is much greater than the slow forward driving movement of the pinion J. Thus, the driven gear R on shaft C is revolved first in one direction and then the other, the gear ratios being so calculated that the valve-driving spindles will revolve approximately two-thirds of a revolution in the forward direction, then in the reverse direction for one-third of a revolution, but never reversing in the same position due to the constant flow forward drive of gear J. These oscillating gear movements can be adapted for purposes other than that described. The complete valve-grinding machine is patented and was designed for a large concern engaged in the manufacture of tractors.

Reversing Shaft Rotation After Eight Revolutions.—The shaft reversing mechanism to be described is used in the construction of a washing machine. The principal driving members consist of the commonly employed gear and double pinions, the latter being located diametrically opposite each other. The reversing parts are of simple construction and are designed to eliminate wear and friction as far as possible. They have been found to work faultlessly and are positive in their action. The same reference letters are used on the different views of Fig. 17 to denote the same parts.

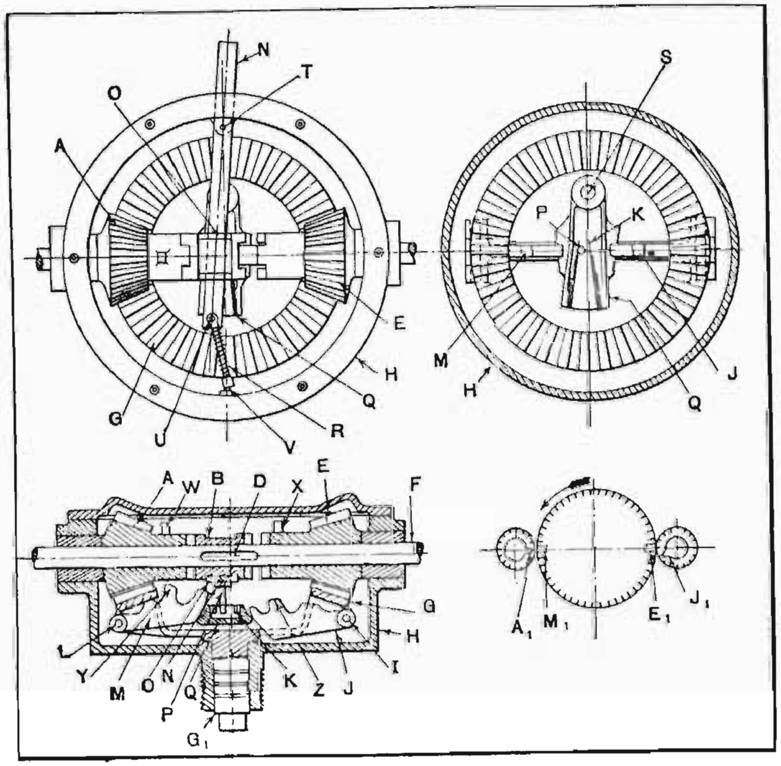


Fig. 17. Mechanism Used on Washing Machine for Reversing Barrel after Eight Revolutions in Each Direction

The cast-iron housing *H* carries the vertical shaft G_1, which connects the main gear *G* with the drum on the washing machine. The two pinions *A* and *E* run on the horizontal shaft *F* which is connected with the motor or power source. This shaft is provided with a key-slot and key *D* upon which the double-ended clutch jaw *B* slides. The center of this clutch jaw is recessed to receive a semicircular bushing *O*. This

bushing has a square section on its outer side which is in contact with the starting and stopping lever *N*. The bushing *O* is provided with flanges on both sides which keep lever *N* in its proper position.

Another lever *Q* swivels on a stud *S* driven into gear *G*. Movement is imparted from lever *N* to lever *Q* by a pin *P* which is driven into lever *N*. This pin extends down into a recess in lever *Q* and works against two shock-absorbing springs *K*, one spring being located on each side of the recess. Gear *G* carries, besides stud *S* and reversing lever *Q*, the two cams *J* and *M* located in slots which extend below the gear and toward the center. These cams are arranged diametrically opposite each other and are pivoted on their respective studs *I* and *L*.

On the upper side of the cams are located lugs *Y* and *Z* which at intervals come in contact with lugs *W* and *X*, the latter being cast on the extensions of pinions *A* and *E*. The purpose of the cams is to push lever *Q* over the center, one cam coming into action when the main gear is revolving to the right and the other when it is revolving to the left. Thus when cam *M* is in the position shown with lever *Q* over to the left of the center, lug *Y* comes in line with lug *W* on pinion *A* at a certain moment. When this happens lug *W* pushes the cam down and this, in turn, forces lever *Q* over the center to a similar position on the right, carrying with it the other movable parts—pin *P*, lever *N*, bushing *O*, and clutch jaw *B*. The latter member is now disengaged from pinion *A* and engaged with pinion *E*, thus imparting the reciprocating or reversing motion to gear *G*.

When lever *Q* moves over to the right it also pushes cam *J* up into a new position, ready to act similarly to the other cam. To obtain a good working action, there must be a certain ratio between the gear and the mating pinions, and this ratio must be based on the number of revolutions which the washing machine barrel is required to make before reversing. In this case eight revolutions are chosen for half a period, or double that (sixteen revolutions) for one whole period. To obtain this, the ratio must be 8 to 23. The number of teeth in this case is 16 for the pinions and 46 for the gear. Another thing to be taken into consideration is the location of the lugs on the pinions. The diagram in the lower right-hand corner of the illustration shows the relative positions of these lugs in regard to each other and also to the cams. Assume that the barrel of the washing machine has completed one period of sixteen revolutions and is ready for the next period, and that the gear will now revolve in the direction indicated by the arrow for the next eight revolutions. After completing the eight revolutions, the direction of rotation is reversed by means of lug J_1, which comes in contact with cam E_1. To obtain this action, the gear and pinions must be so assembled that lug J_1 lacks just two tooth spaces of making contact with cam E_1, when lug A_1 is in contact with cam M_1, as illustrated. When the required eight revolutions are completed, lug A_1 again comes in contact with cam M_1 and the motion is reversed again for the next eight revolutions.

A coil spring R, holds lever N in the required position on each side of the center. This spring is seated at point V on the housing, and is guided on the lever by means of a pin U. Lever N is mounted on a pin T located diametrically opposite point V on the housing. The lever is extended outside the casing so that the operator can get a good grip on it for starting and stopping the mechanism.

Reversal of Motion after Predetermined Number of Revolutions up to 100,000. — With the mechanism illustrated in Fig. 18, a driven shaft may be reversed after making any predetermined number of revolutions from 1 to 100,000 and the motion may be discontinued entirely after the shaft has made any given number of reversals up to 10,000. This mechanism was applied to a textile machine. The reversing shaft A is driven from the vertical shaft shown through either one of the miter gears B which revolve in opposite directions and are alternately engaged with the shaft by the sliding clutch C. The reversal of rotation after a predetermined number of revolutions is controlled by a system of ratchets

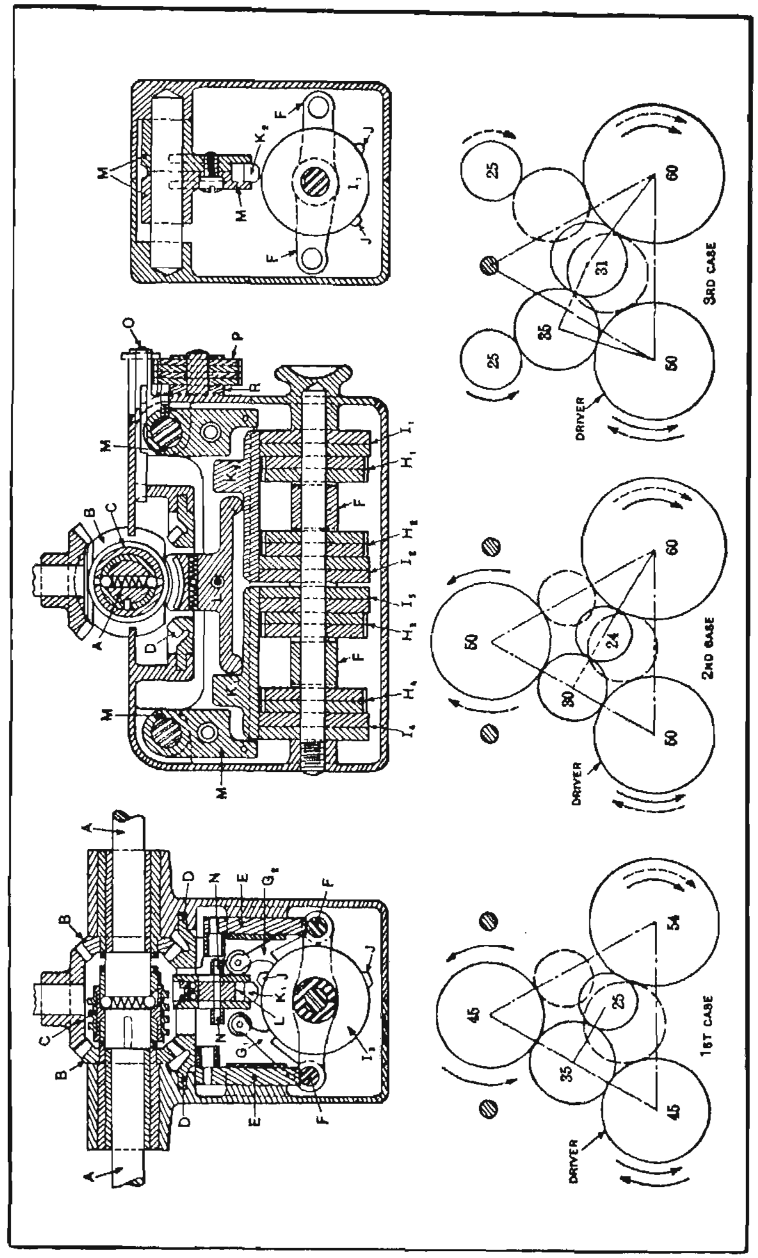


Fig. 18. Mechanism for Reversing Motion after any Predetermined Number of Revolutions between 1 and 100,000, and for Stopping Driven Shaft after any Number of Reversals up to 10,000

and pawls. Another ratchet-and-pawl mechanism is also utilized for stopping the rotation of A after a given number of reversals, by placing the shifting clutch C in the central or neutral position.

The lower miter gear D has a cam which engages the rollers on the upper ends of bars E. The lower ends of these bars oscillate the rockers F which carry two sets of pawls, G_1 and G_2. The set of four pawls G_1 at the left is in the operating or working position, as shown in the illustration. There are four pairs of ratchets H_1, H_2, H_3, and H_4. The teeth of each pair of ratchets are cut oppositely and the four pawls on one side of the ratchet shaft are for engaging the ratchets which control the number of revolutions made by shaft A in one direction, whereas the four pawls on the other side of the shaft are for operating the reverse motion ratchets. These ratchets operate progressively and transmit motion to disks I_1, I_2, I_3, and I_4. These disks have projections or cam surfaces J, which serve to shift the reversing clutch C after shaft A has made a predetermined number of revolutions, which number is regulated by adjusting the ratchets before the mechanism is put into operation. This system of cam disks and ratchets will be referred to as the "combination."

Each ratchet has 20 teeth with one deep cut or tooth. Each tooth of ratchet H_4 is equivalent to 8000 revolutions of shaft A; each tooth of ratchet H_3 is equivalent to 400 revolutions; each tooth of ratchet H_2, 20 revolutions; and each tooth of ratchet H_1 is equivalent to one revolution of shaft A. The mechanism is set for a given number of revolutions by turning each ratchet so that the deep tooth is away from the operating pawl a certain number of teeth, the number depending, in each case, upon the number of revolutions of A represented by each tooth. For instance, to set the combination for a reversal of motion after shaft A makes 49,763 revolutions, ratchet H_4 is so located that there are six teeth between the operating pawl and the deep tooth; these six teeth are equivalent to 48,000 revolutions of A. Ratchet H_3 is then set at four teeth, representing 1600 revolutions of A; ratchet H_2

is set at eight teeth, equivalent to 160 revolutions of A, and, finally, ratchet H_1 is set at three teeth, representing three revolutions of A. The mechanism is now set for a total of 48,000 + 1600 + 160 + 3 = 49,763 revolutions.

After the mechanism has been set in the manner described, its action is as follows: The pawl G_1 which is actuated once for every revolution of gear D, drops into the deep tooth or notch of ratchet H_1 after engaging three teeth on H_1, since, in this particular case, this ratchet was adjusted so that there were three teeth between the pawl and the deep tooth. As soon as this deep tooth is engaged by the pawl, the ratchet H_2 is turned a distance equivalent to one tooth; ratchet H_2 then remains stationary until H_1 has made a complete turn and its pawl again drops into the deep tooth, when H_2 is again moved one tooth. The pawls are so located that the first one must engage the deep notch before the next successive pawl can engage its ratchet at all, and the relation between the other pawls is the same. Ratchet H_2 continues to be moved a single tooth for each complete revolution of H_1 until it has moved eight teeth, in this particular instance. The pawl of H_2 then drops into a deep tooth and ratchet H_3 is moved one tooth. Ratchet H_3 now remains stationary until H_2, by the continued action of H_1, makes a complete revolution, when H_3 is moved another tooth. After H_3 has moved a distance equivalent to four teeth, its pawl, in turn, drops into the deep notch, and ratchet H_4 is turned one tooth. A complete revolution of H_3 turns H_4 another tooth and, when H_4 has moved six teeth, in this case, the shaft A will have made a total of 49,763 revolutions.

This result will be verified in order to more clearly show the action of the mechanism. As previously mentioned, each ratchet has 20 teeth. Each tooth of H_1 represents one revolution of shaft A and the movement of three teeth prior to engagement with the deep notch equals three revolutions. Since H_2 is set for a movement of eight teeth, H_1 will have to make eight complete turns, which will be equivalent to 160 additional turns of A. Now the four complete turns of H_2 necessary for moving H_3 four teeth require $4 \times 20 \times 20$ or 1600 additional turns of A, giving a total of 1763 revolutions. Finally, the movement of H_4 six teeth requires $6 \times 20 \times 20 \times 20 = 48{,}000$ additional turns of A, so that the total number of revolutions made by A prior to reversal equals 49,763, when the ratchet mechanism is set as previously described.

The progressive action of the ratchets gradually revolves the cam disks preparatory to shifting the reversing clutch. The cam J on disk I_4 first engages and lifts the floating lever K_1 at the left-hand end and the lever L one-half as much. When the other cam disks act upon K_1 and K_2, these levers, together with part L, are lifted the full amount and spring balls in L cause it to be thrown quickly into mesh with the clutch C. This clutch is threaded and two threads are also formed on the upper side of part L. As levers K_1, K_2, and part L are contained in a carriage M, all are constrained to move parallel to the axis of shaft A, because of the action of the screw threads on clutch C. This results in breaking the combination; that is, the floating levers are all removed from the cams, the four pawls G_1 are disengaged from their ratchets, and the idle set of pawls G_2 comes into action, thereby reversing the rotation of the controlling mechanism. As soon as the travel of carriage M is completed, which requires $1\frac{1}{2}$ revolution of the clutch, the latter is constrained to act along the threads on L while making one revolution, until feathers attached to the clutch over-ride the spring balls in shaft A; the clutch is then instantly thrown out of mesh with one bevel gear and into mesh with the other, thereby reversing the rotation of shaft A. Just as the clutch starts this rapid shifting movement, the cam on gear D engages one of the rollers N on part L and throws the levers all down and the threads on L out of mesh with the clutch.

The rotation of shaft A is stopped after a predetermined number of reversals by means of a separate mechanism which arrests the movement of carriage M midway of its travel. The worm threads on clutch C then act upon part L and withdraw the clutch until it is out of engagement on one side and cannot engage on the other. Carriage M is stopped by a pin O which drops into a groove in M after the four ratchets P, having ten teeth, are properly aligned as regards a deep notch in each ratchet. These ratchets are operated consecutively by a stepped four-fingered pawl on R through the medium of a pin connecting with M.

The three diagrams in the lower part of Fig. 18 illustrate the systems of gearing controlled by the mechanism described in the foregoing. The requirements, as illustrated by these three diagrams, are as follows: 1. That a reversing gear shall drive two others continuously in the same direction but in opposite directions relative to each other. 2. That a reversing gear shall drive one of the two gears continuously in the same direction and the other in the same direction as that of the reversing gear. 3. That a reversing gear shall drive one gear continuously in the same direction and shall drive two others alternately in the same direction as itself. The full arrows and the full circles on these diagrams belong together, and likewise the broken or dotted lines and arrows. The full lines connecting the centers indicate that those gears are linked, whereas the broken lines denote spring connections. The movement of the reversing driver and the friction of the links swing the idler gears.

CHAPTER VII

OVERLOAD RELIEF MECHANISMS AND AUTOMATIC SAFEGUARDS

SOME tripping or stop mechanisms are designed for machines requiring an automatic safeguard either against overloads and resulting excessive strains on the machine parts or as a means of protecting the machine against some other dangerous operating condition. A tripping mechanism of this general class may be so arranged that it will transmit power under normal conditions but disconnect the driving and driven parts if the resistance or strain becomes excessive, thus safeguarding the machine from injury. Automatic trip mechanisms are also utilized in some cases to prevent excessive speeds, or possibly for stopping the machine automatically if a part being operated upon is not in the proper working position. The following examples illustrate typical applications of different types of automatic safeguards.

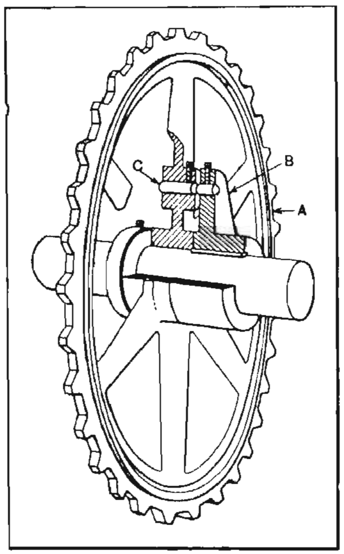


Fig. 1. Sprocket Driven Through Pin which Breaks in Case of Excessive Overload

Breakable Pins to Prevent Excessive Overload. — Some types of machines are so arranged that any unusual resistance to motion will automatically stop either the entire machine or whatever part is affected, in order to prevent damaging the mechanism or straining it excessively. A simple form of safety device consists of a pin which shears off or breaks in

case the overload becomes excessive. The sprocket A shown in Fig. 1 is provided with a pin of this kind. This pin C connects the driving hub B with the hub of the sprocket. The sprocket, instead of being keyed to the shaft, is loosely mounted on it, and the hub B is keyed to the shaft instead. The pin C is grooved or reduced in diameter an amount depending upon the maximum amount of power to be transmitted. If this pin is subjected to an unusual strain, it will break, thus leaving the wheel free and protecting the driven parts.

This same method of protection against overload has been applied in various ways, and, while it is simple, there are certain disadvantages. In order to avoid replacing a broken pin, the machine operator sometimes inserts a pin that is stronger than it should be to afford adequate protection against injurious strains. The ideal safety device is one which does not break in case of overload, but simply disengages and is so arranged that it can readily be re-engaged. In electrical work, this principle has been applied by substituting circuit-breakers for fuses which melt when the current becomes excessive.

Automatic Clutch Control to Prevent Overload. — The principle governing the operation of an automatic device for disengaging a clutch when the overload becomes excessive is illustrated by the diagram, Fig. 2. This mechanism was applied to a metal-cutting machine, the object being to automatically disengage the feed in case the resistance to the rotation of the tool becomes abnormally high. The mechanism is also arranged to reverse the feeding movement if, for any reason, the excessive resistance should continue after the feed has been disengaged. The spindle to which the cutting tool is attached is represented at A. This spindle is driven through worm-wheel M and worm L from the driving shaft B, which receives its motion from a countershaft through a belt operating on pulley K. The driving shaft B is free to move in a lengthwise direction within certain limits. The clutch C is keyed to this shaft so that it will rotate and move axially with the shaft. The gears *D* and *F* on each side of clutch *C* are free to revolve upon the shaft, but are prevented from moving in a lengthwise direction. The inner side of each gear is provided with clutch teeth corresponding to those on clutch *C*, which is used to lock either gear to shaft *B*. The shaft *I*, which transmits feeding movement to the cutting tool, is driven either through gears *D* and *E* or through gears *F*, *P*, and *H*. When clutch *C* engages gear *D*, the cutting tool is fed forward by shaft *I*, and a reversal of the feeding movement is obtained when clutch *C* is shifted into engagement with gear *F*.

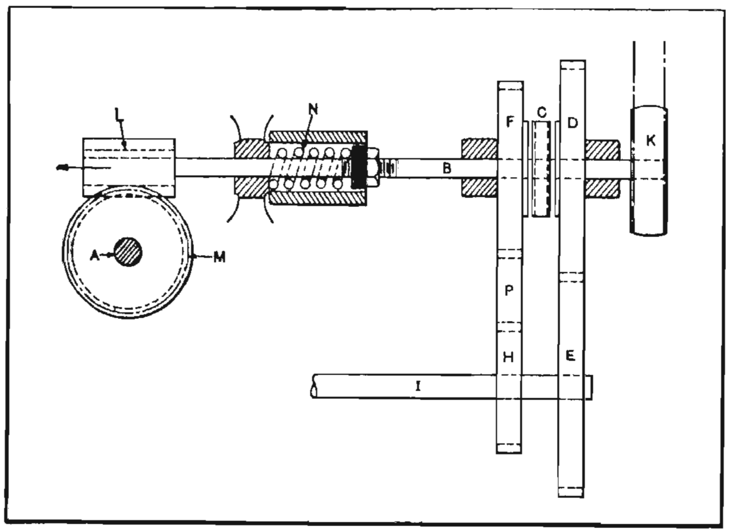


Fig. 2. Device for Automatically Stopping Feeding Motion when Resistance to Rotation Becomes Excessive

When clutch *C* engages with gear *D*, excessive resistance to the motion of the cutting tool will cause the clutch to be shifted to the neutral position, thus stopping the feeding movement. This automatic action is obtained as follows: The shaft *B* is normally held by spring *N* in such a position that clutch *C* engages gear *D*, so that the feeding movement is forward. The tension on this spring is regulated by the nut shown. In case the resistance to the rotation of the cutting tool and spindle *A* should become excessive, the pressure between the teeth of the worm *L* and the worm-wheel *M* causes the worm to move in the direction indicated by the arrow, the worm-wheel acting somewhat like a nut. This lengthwise movement of worm-wheel *L* and shaft *B*, against the tension of spring *N*, disengages clutch *C* from gear *D* and stops the feeding movement. If the resistance to rotation again becomes normal, clutch *C* is automatically returned into engagement with gear *D*. On the other hand, if the resistance to rotation increases, clutch *C* may be drawn over into engagement with gear *F*, thus reversing the feeding movement.

Overload Relief for Worm Drive. — Other mechanical devices for automatically disengaging the driven member whenever the resistance to motion increases excessively are shown at *A* and *B* in Fig. 3. These devices operate on the same general principle as the one previously described, but differ somewhat in regard to the arrangement. The mechanism illustrated by diagram *A* is designed to allow a worm-wheel to make on revolution and then stop; the movement, however, may be discontinued before the revolution is completed, if the resistance to rotation becomes excessive. The sleeve *a* is revolved constantly by a pulley on its outer end. The inner end of this sleeve has clutch teeth intended to engage corresponding teeth on the end of sleeve *b*. The latter is attached to the shaft and both are free to move slightly in an endwise direction. The body of sleeve *b* is threaded to form a worm which engages worm-wheel *c*. The spring *e* tends to shift sleeve *b* to the left and into engagement with clutch teeth on sleeve *a*. The stop at *d* is utilized in this particular case to disengage the driving clutch after the worm-wheel has made a revolution. If stop *d* is withdrawn, the spring *e* revolves the worm-wheel slightly and moves the worm and clutch *b* to the left and into engagement with the constantly revolving clutch *a*. The worm-wheel then begins to revolve and continues until the lug *g* strikes the stop *d* or until some unusual resistance too great to be overcome by the spring is encountered; then, as the worm-wheel remains stationary, it forms a nut for the worm which screws itself out of engagement with clutch a. The strength of spring e is proportioned with reference to the safe or maximum load to be transmitted. One of the advantages of this type of mechanism is that the motion is positively transmitted until an excessive load causes the driving clutch to be disengaged. Provision may readily be made for the adjustment of spring e so that the tension can be varied according to conditions.

Diagram *B*, Fig. 3, illustrates a modification of the same general type of mechanism. The shaft m is free to move

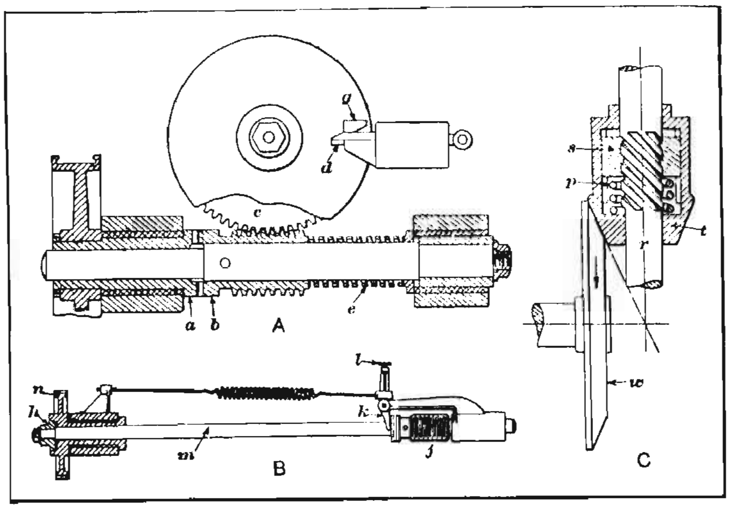


Fig. 3. (A and B) Devices for Automatically Disengaging the Driven Member whenever Resistance to Motion Increases Excessively; (C) Friction Gearing Designed to Vary Contact Pressure According to Load

slightly in an endwise direction and is keyed to the tapering disk h, which fits into a seat of corresponding taper in the hub of gear n, thus forming a friction clutch. Motion is applied to gear n and is transmitted by worm j to a worm-wheel (not shown), for any desired purpose. Shaft m turns freely in the hub of gear n, but is attached to worm j. The lever k, which has a spring fastened to it above the fulcrum or pivot, supplies the necessary amount of thrust to keep h in engagement with n under ordinary conditions. This thrust may be regulated by the thumb-screw l which changes the position of

the block to which the spring is fastened. If the resistance to the motion of the worm-wheel becomes excessive, the worm moves bodily along the teeth of the wheel, as though it were a nut, and, by moving shaft m and disk h to the left, disengages the friction clutch. The endwise thrust from lever k might be obtained by means of a weight instead of a spring.

Pressure of Friction Gearing Varied According to Load.— A novel design of friction gearing, in which the pressure between the two friction wheels is automatically regulated by the amount of power transmitted, is shown at *C* in Fig. 3. The wheel w which is the driver revolves in the direction shown by the arrow. The driven pinion t is free to either rotate or slide in a lengthwise direction upon shaft r within certain limits. This shaft has a screw of coarse pitch which passes through nut s. This nut slides in grooves in the friction pinion t so that the pinion and nut revolve together. A spiral spring p inserted between nut s and the pinion forces the latter against the driver w with a pressure depending upon the position of the nut. If wheel w is revolving in the direction shown by the arrow and the driven shaft meets with an unusual degree of resistance to rotation, as soon as shaft r lags behind or stops revolving, nut s moves downward, owing to the action of the screw, and increases the compression on spring p and also the pressure between pinion t and wheel w. When the resistance to rotation again becomes normal, the spring moves the nut slightly upward and reduces the endwise thrust. While this device may not be entirely practicable, it embodies an interesting principle.

Cam and Spring Type of Overload Release Mechanism.— The valve driving mechanism of a certain rotary valve type of gasoline engine is provided with an automatic release mechanism as a safeguard in case the rotors should seize, due to a failure of the oil or water supply. The crankshaft transmits motion to hollow shaft *A*, Fig. 4, through a pair of bevel gears, and this motion is transmitted through pin *B*, sleeve *C*, and pin *D* to shaft *E*, which turns the rotors through another pair of bevel gears. Hollow shaft *A* fits over an extension on shaft *E*, and is held against endwise movement by a thrust collar.

Pin *B* engages a slot in sleeve *C*, this slot being similar in form to the well-known bayonet lock. Pin *D* in shaft *E* engages a cam surface formed on the end of sleeve *C*. The sleeve is held firmly against pin *D* by spring *F*, the tension of which may be adjusted by changing the position of collar *G*. A ball thrust bearing *H* is located between the spring and sleeve *C*.

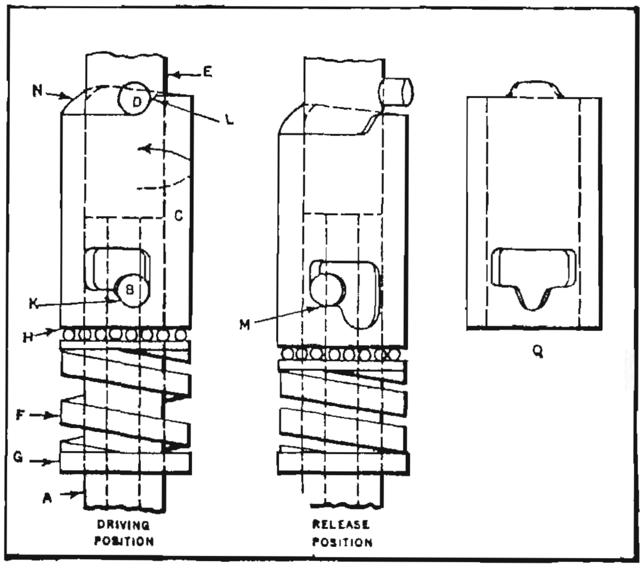


Fig. 4. Overload Release Mechanism

Normally, the entire assembly rotates in the direction indicated by the arrow, and it will be evident that sleeve *C* tends to slide downward, as pin *B* bears against an inclined surface *K* and pin *D* bears against an inclined surface *L*. Ordinarily, however, sleeve *C* is prevented from sliding downward by the upward pressure of spring *F*.

If the torque reaches the danger point, the spring thrust is overcome and sleeve *C* moves downward far enough to release pin *D*, which moves along a slight rise and finally disengages pin *B* from slope *K*, thus allowing the level surfacc *M* to en-

gage pin *B*, which now rests against the end of the slot. Pin *B* holds sleeve *C* in this released position, and rotation of *A* and *C* continues while shaft *E* remains stationary.

To re-engage the drive, the crankshaft is given a few backward turns. As shaft *A* revolves sleeve *C*, due to the upward pressure on pin *B* and the frictional resistance between *B* and *C*, surface *N* comes into contact with pin *D*. At this point, any further turning movement would force sleeve *C* downward. However, the resistance between pin *B* and surface *M* is slight in comparison with the force required to rotate surface *N* beneath pin *D*. Therefore, shaft *A* and pin *B* turn until the pin engages the lower part of this slot and is again in the driving position. The clicking sound made by the dropping of pin *B* shows that it is unnecessary to continue turning the crankshaft backward, although more turns than required will do no harm, because each succeeding revolution simply results in forcing the sleeve downward as surface *N* engages pin *D*, the sleeve snapping upward into place again when pin *D* passes surface *L*.

This mechanism automatically retimes the rotary valves as well as the ignition, because the various parts always occupy the same relative positions when the forward drive begins. The special sleeve shown at *Q* is intended to provide the automatic release, regardless of the direction of rotation. While this design *Q* is not actually being used, it doubtless would be satisfactory, as the principle of operation is the same. The cam surfaces are altered to provide release in either direction of rotation and to permit re-engagement by turning backward relative to the direction for driving.

Sensitive Tripping Clutch for Delicate Machinery. — A sensitive tripping clutch designed for use on delicate machines of various classes, and whenever quick disengagement is required, is so arranged that the tripping action can be controlled automatically by whatever method is most suitable for the particular application.

The driving shaft *D*, Fig. 5, transmits motion to the driven sleeve *A* through engaging clutch teeth which give a positive drive. When the clutch is to be tripped to stop the rotation of the driven sleeve, a tripping pawl engages a notch in trip-ring *H*, which instantly disengages the driving and driven members, as described in detail later.

One arrangement of the tripping pawl *T* is shown in Fig. 7; this view includes ring *H* of the clutch in order to illustrate clearly the general arrangement. When pawl *T* engages one of the notches in ring *H*, as shown, the clutch is tripped. Just how this tripping action occurs will be apparent by referring to the details of Fig. 5. The driving shaft is connected to

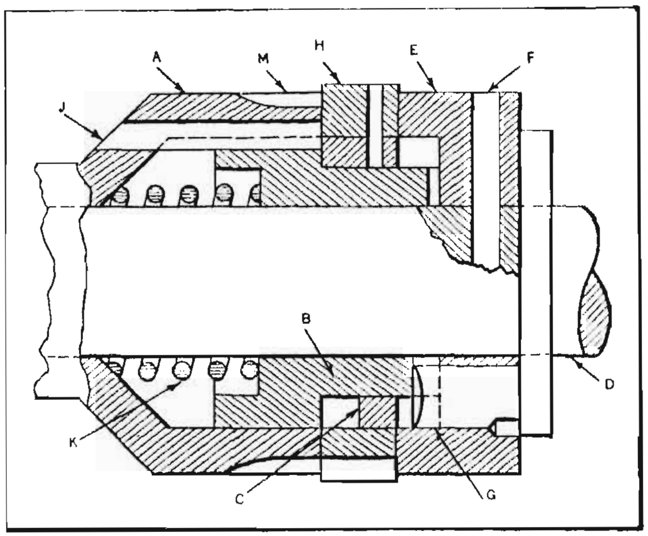


Fig. 5. Sectional View of Sensitive Tripping Clutch

sleeve *E* by pin *F*, and in sleeve *E* there are four driving pins *G*. These pins engage teeth on the small diameter of throw-out cam *B*. The cam (which is shown in detail at *B*, Fig. 6) is secured to the driven sleeve *A* by four pins *J*, Fig. 5, in such a manner as to allow cam *B* to slide axially. When in the running position, cam *B* is held in engagement with driving pins *G* by spring *K*.

Trip-ring *H* and ring *C* are connected by pins, so that they function as a single part, which can revolve but cannot move in any other direction. Clutch teeth on ring *C* are in alignment with corresponding teeth on the larger part of cam *B* (see detail of ring *C* in Fig. 6). When the tripping pawl, which normally is out of engagement with ring *H*, snaps into the tripping position, rings *H* and *C* stop revolving. Throw-out cam *B*, however, and the driven sleeve continue to revolve until the teeth engaging ring *C* slide backward far enough to disengage the teeth engaging driving pins *G*, thus stopping

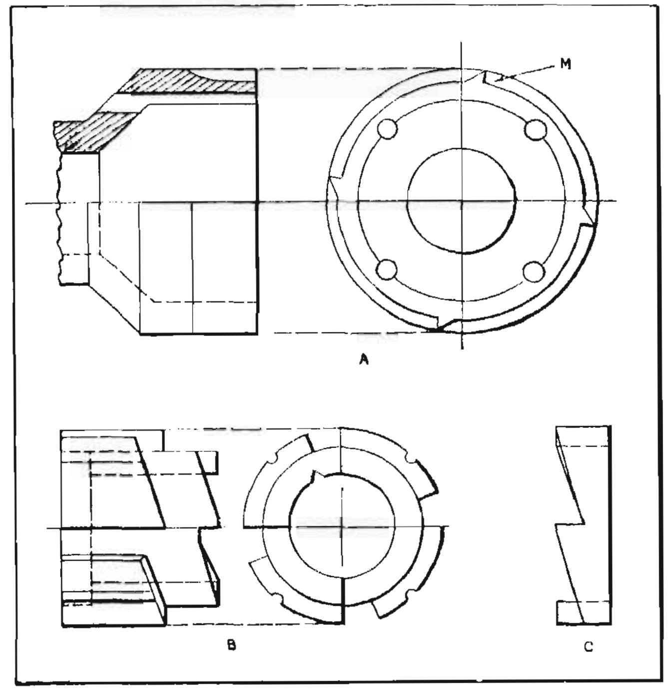


Fig. 6. Detail Views of Parts A, B and C, Fig. 5

the rotation of *B* and the driven sleeve. It will be seen, therefore, that the teeth on ring *C* act as cams to disengage the teeth on the smaller diameter of *B* which mesh with the driving pins.

To prevent continued rotation of the driven sleeve, as the result of friction, small teeth *M* are provided in the driven sleeve (see also detail view *A*, Fig. 6). These teeth *M* are also engaged by the tripping pawl, but they are so spaced in assembly that there is time enough after the stopping of ring H for the driving and driven teeth to disengage before a tooth M on the driven sleeve comes around into engagement with the tripping pawl. The end of the driven sleeve can be fitted with either a gear or sprocket, according to the driving requirements.

The method of tripping the clutch depends upon how the clutch is applied. The diagram Fig. 7 represents an arrange-

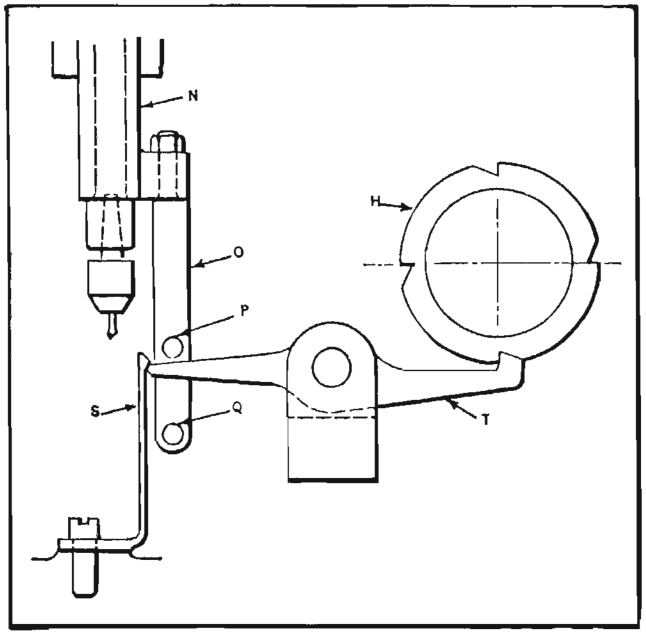


Fig. 7. One Method of Tripping Clutch Automatically

ment intended for a special drilling fixture. The rack sleeve N is free to move vertically, but it does not revolve. The attached bar O carries the trip-pins P and Q. These pins might be held in a T-slot to provide adjustment if necessary. It will be apparent that pawl T moves into engagement with ring H when pin P pushes the end of T down past the apex of the tripping spring S. The tripping pawl snaps quickly into engagement with ring H after its pointed end passes the apex of the tripping spring. The disengagement of the trip-

ping pawl and the engagement of the clutch occurs when pin Q strikes the pawl as it ascends. By varying the angle of the engaging points on pawl T and spring S, the sensitiveness of the tripping action may be varied as required. This general method of tripping can be applied when tapping to certain depths and for various similar purposes.

An entirely different tripping arrangement is illustrated by the diagram Fig. 8. This particular arrangement might be utilized in connection with fine wire drawing or whenever the tripping of the clutch is to occur automatically in case of wire or thread breakage. The diagram represents wire drawing, although it is evident that the same principle might be applied to other processes. The winding drum is indicated at

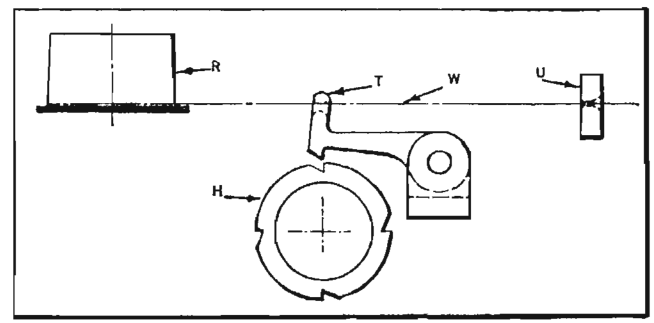


Fig. 8. Clutch-tripping Arrangement for Wire Drawing or Similar Process

R, the drawing die at U, and the wire by line W. The wire passes through a bell-mouthed hole in tripping pawl T. The pawl, in this instance, is supported by the wire under normal working conditions. If the wire should break accidentally, or if the end passes through the die, the pawl drops by its own weight and engages the trip-ring H. It will be evident from the foregoing examples that the tripping of the clutch might be controlled in many other ways.

Automatic Relief Mechanisms for Forging Machines.— Forging machines are equipped with a tripping or relief mechanism which prevents excessive straining or breakage of the parts controlling the motion of the movable die, in case the stock to be forged is not placed in the grooves of the dies, but is caught between the flat faces. These relieving mechanisms differ somewhat in design, but the object in each case is to temporarily and automatically release the movable die from the action of the driving mechanism, in case the operating parts are subjected to a strain or pressure that is abnormally high. The release may be obtained by inserting bolts or "breaker castings" in the mechanism, which will shear off or break if there is an excessive strain; another type of relief mechanism depends for its action upon a spring which is proportioned to resist compression for all ordinary strains but to compress sufficiently to release the pressure on the dies when that pressure increases beyond a safe maximum.

Spring and Toggle Relief Mechanism. — The plan view of a forging machine, shown in Fig. 9, illustrates one method of arranging a spring and toggle relief mechanism. When this machine is in operation, the stock is gripped between the stationary die *A* and the movable die *B*. The heading slide *C*, which carries a ram or plunger for performing the forging operation, is actuated by a crank on the crankshaft *D*. The gripping slide *E* to which die *B* is attached is moved inward for gripping the stock and outward for releasing it, by means of two cams *F* and *G*. These cams transmit motion to slide *H*, which is connected with slide *E* through a toggle and link mechanism. Cam *F*, acting upon roll *T*, moves the slide *E* for gripping the stock, whereas cam *G*, in engagement with roll *V*, withdraws the die after the forging operation is completed. The upper detail view to the right shows the relief mechanism in its normal position, and the lower view shows it after being tripped to relieve any abnormal pressure on the dies.

When the machine is operating normally, link *J*, which connects with link *K* of the main gripping toggle, oscillates link *K* about pivot *L* and, through link *M*, imparts a reciprocating motion to the gripping slide *E*. If a piece of stock or some other part is caught between the flat die faces, the gripping action continues until the strain exceeds a certain amount; then the backward thrust upon link *N* causes it to swing about

pivot *O* (see lower detailed view) carrying with it the other links of the "by-pass toggle" and compressing the spring *S* which is shown in the plan view at the left. As the result of this change in the position of the by-pass toggle, pressure on the gripping die is released. Meanwhile the heading tool at-

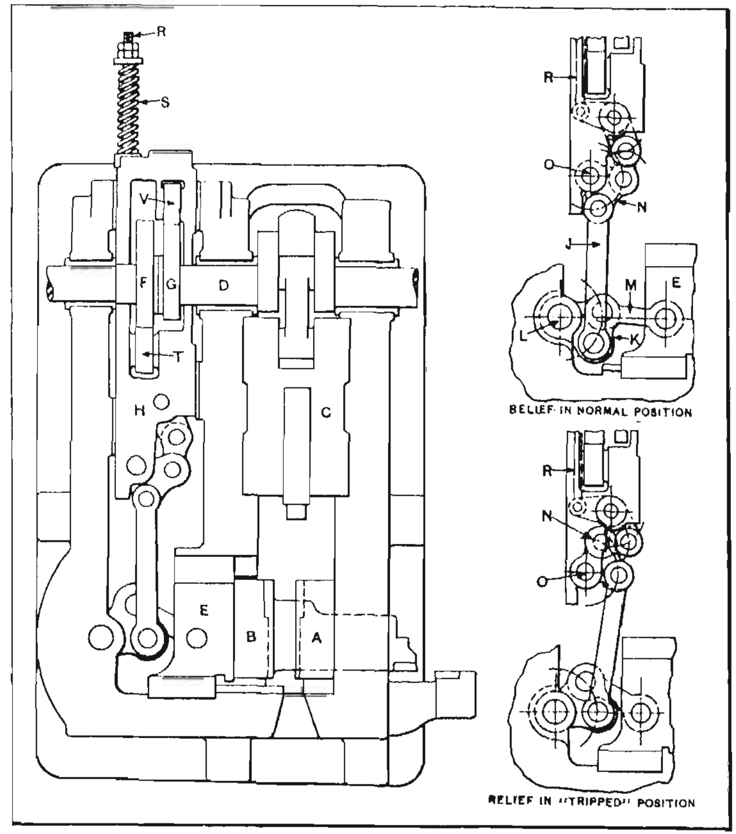


Fig. 9. Plan and Detail Views of Forging Machine Showing Automatic Relief or Tripping Mechanism

tached to slide *C* completes its full stroke and, upon the return stroke, the by-pass toggle is re-set automatically by spring *S* which expands and, through rod *R*, swings the toggle links back to their normal position shown in the upper detailed view. This automatic re-setting of the toggle makes it unnecessary to stop the machine, as is necessary with safety devices of the breaking-bolt type. There is no movement of the by-pass toggle, except when a "sticker"—to use the shop expression—is caught between the gripping dies. While this relief mechanism safeguards the working parts of the forging machine from excessive strains, it is capable of transmitting enormous pressures to the gripping dies.

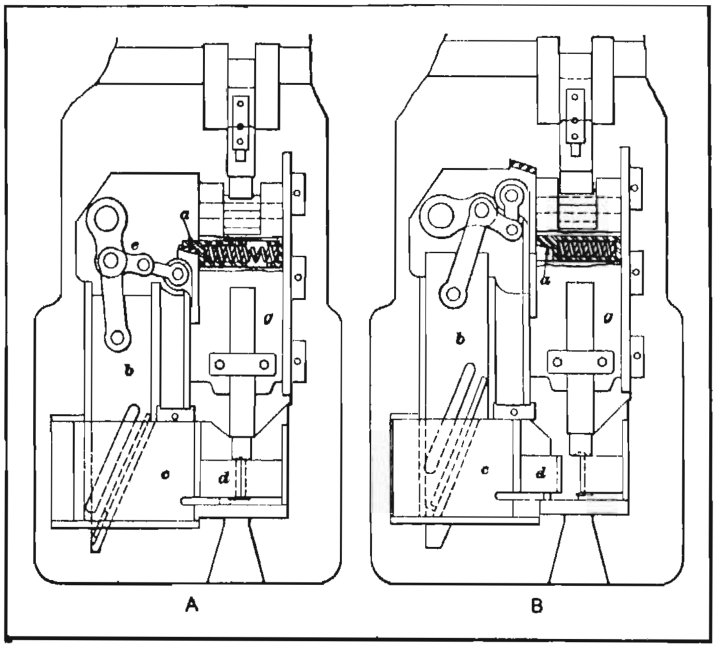


Fig. 10. Bevel Spring Plunger Type of Relief Mechanism on Bolt and Rivet Header

Beveled Spring-plunger Relief Mechanism. — The type of relief mechanism illustrated in Fig. 10 is applied to a wedge-grip bolt and rivet header. The movable die *d* is attached to slide *c* which is beveled to correspond with the tapering end of slide *b*. Slide *b* is given a reciprocating movement by the toggle mechanism at *e*, and, when slide *c* is pushed inward for closing the dies, the beveled end of slide *b* forms a solid metal backing, which securely locks the movable die during the heading operation. When forming the heads on bolts or rivets, it is neces-

sary to place the stock directly in the impression in the gripping dies, and not between their opposing faces, as these dies are intended to come together, so that the stock is firmly held in the impression between them while the rivet or bolt head is formed by the tool attached to slide *g*. The relief mechanism, which comes into action in case the stock is caught between the dies, consists of a spring plunger *a*, which has a beveled end and is held outward by the spring shown. The beveled end of this plunger bears against an angular projection on a slide for transmitting motion, through the toggle mechanism, to slide *b* and the movable die. If this die, however, is prevented from moving inward by a piece of stock that is not in the die impression, but caught between the faces, the increased pressure on plunger *a* forces it back against the tension of the spring and off of the beveled seat, as indicated by illustration *B*.

Overload Release with Positive Lock Acting During Short Period. — Often it is desirable, in feeding or other operations, to have a safety device that will function at any time except just before the end of the stroke, when the action must become positive as, for example, when a part is being locked or held solid while it is being operated on by a punch or other tool, after having been pushed into position with the automatic release ready to act in case of any obstruction. The device shown in Fig. 11 was designed to provide such an overload release and positive lock during the last 1/16 inch of the stroke. In this instance, the automatic release is used in connection with a slide that transmits a tensional or pulling force, but this type of release has also been applied with slight changes where a compressive or pushing force is utilized. Such a mechanism is applicable in conjunction with auxiliary rams or slides of punch presses, and it may also be applied to automatic machines or other mechanisms.

In the particular design illustrated, connection with the source of power is at *A*, and motion is transmitted through link *B* to slide *C* and through the safety device to the point of application at *D*. (In some cases, the operating stroke might be derived from an eccentric at the side of a press or

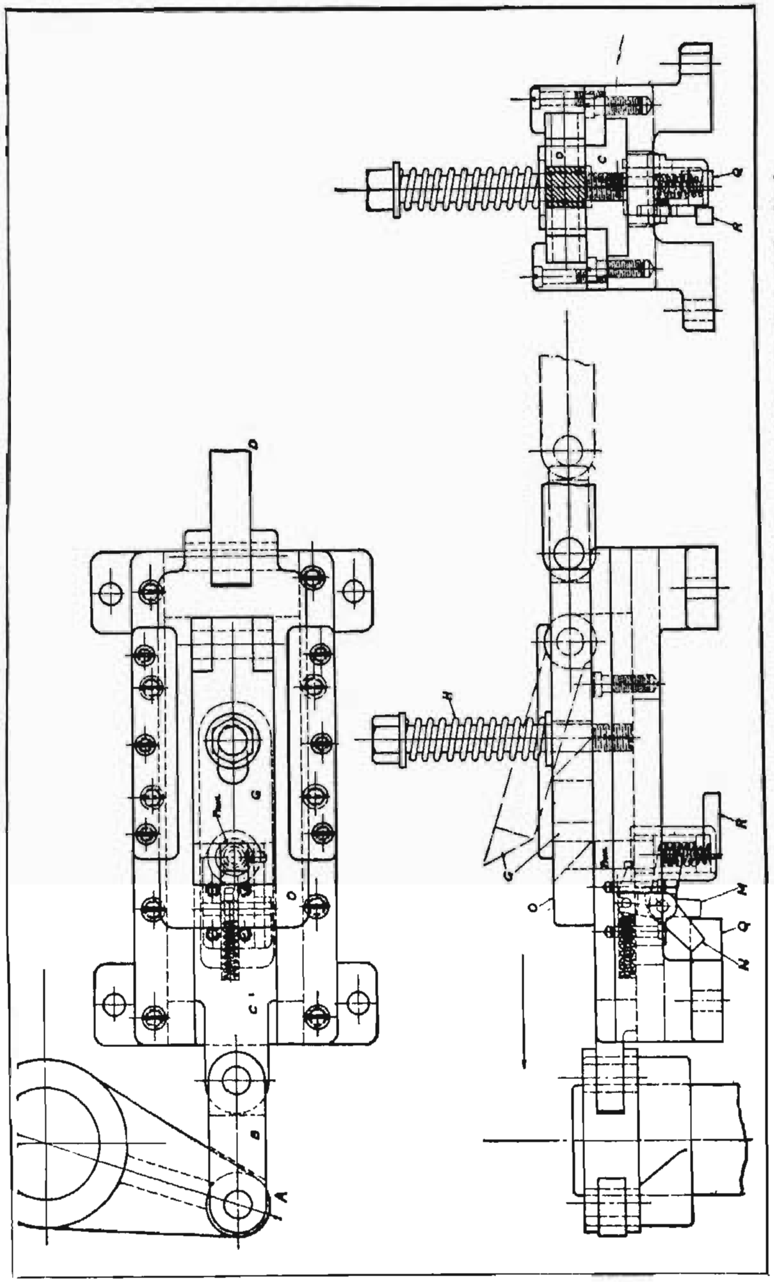


Fig. 11. Safety Overload Release which Changes to a Positive Drive Just Before the End of the Stroke

from a crankshaft.) Pivoted to slide *C* is a releasing plate or flap *G*, which is free to swing about its pivot or pin, as indicated by the dotted lines in the lower view. This releasing member *G* normally is held in the closed or driving position by spring *H*. As the end of plate *G* is beveled and engages an inclined surface on the driven part *O*, plate *G* tends to swing upward during the working stroke, but is prevented by spring *H* unless unusual resistance is encountered. The part *O* is in the form of a strap or loop which surrounds plate *G*, as indicated by the plan view, and *O* is positively connected to member *D*.

Unless some obstruction or abnormal resistance causes disengagement of the driving and driven part through the release mechanism described, movement of the slide continues in the direction of the arrow until lever *M* strikes projection *Q*, thus releasing the pawl, the location of which is indicated on the side and plan views. As the pawl is instantly forced upward by a spring beneath it, the drive is transferred from the beveled surfaces between parts *O* and *G* to vertical surfaces between parts *O*, *G*, and the pawl, so that the release mechanism is no longer effective and the stroke becomes positive.

During the return stroke, lever *N* strikes projection *R* and withdraws the pawl, so that it is in position for the next forward stroke. The illustration shows the relative positions of the parts just before the pawl is released or tripped. A spring is used to force the pawl upward when released, because this provides the instantaneous action desired.

If during the working stroke an obstruction had been encountered by the driven member, part *O* would have stopped as flap *G* disengaged from it and slide *C* would have traveled forward without injuring whatever tools or other members are connected at *D*. The action should, of course, be timed by properly locating the projections or stops for manipulating the pawl levers. The angle of the bevel between parts *O* and *G* depends upon the amount of force to be transmitted, and adjustment of spring *H* provides a more delicate means of regulating the releasing action.

Automatic Stop for Wire-winding Machine. — The diagram, Fig. 12, shows an automatic tripping device that is applied to a machine used for winding small wire onto spools. In this illustration, *A* represents the reel which contains the stock of wire, and *B* is the spool upon which the wire is wound. This spool is driven at a constant speed. If, for some reason, the wire should not uncoil easily from reel *A*, it might be broken or the mechanism damaged, assuming that the wire passed directly from the reel to the spool. In order to avoid trouble from any resistance to uncoiling which may occur, the wire, after leaving the reel, is guided by idler pulleys, so

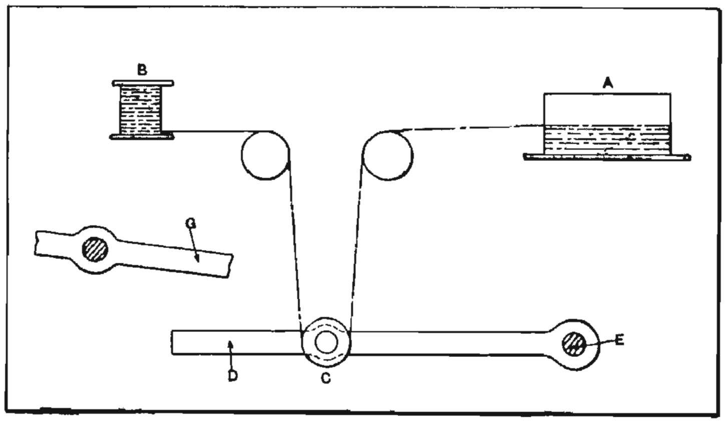


Fig. 12. Safety Tripping Device for Wire-winding Machine

as to form a loop; at the end of this loop, there is an idler pulley *C* mounted on a lever *D* which is free to swing about fulcrum *E*. When the uncoiling and winding is proceeding under normal conditions, the weight of lever *D* is sufficient to prevent the wire from lifting it; any abnormal resistance, however, such as might be caused by a kink on reel *A*, will result in swinging lever *D* upward into contact with trip *G*, which, by disengaging a clutch, stops the machine.

Automatic Stop Mechanisms for Textile Machines. — Some very ingenious tripping mechanisms or "stop motions" are applied to different classes of textile machines. The examples described illustrate the possibilities of the use of com-

paratively simple devices for automatically controlling the action of machines under conditions which might, at first, seem to be very complex and difficult.

The stop motion shown in Fig. 13 is applied to a machine used for twisting yarn. The yarn passes from the guide wire at *A* around the rolls *B* and *C*, through an eye in wire *D* and out through the guide at *E*. The wide *D* is attached to another wire *F*, which is normally held by the yarn in the position shown by the full lines. If the yarn or thread should break, the wires fall to the position shown by the dotted lines, thus bringing wire *F* into engagement with the lower roll *C*. Contact with this roll immediately moves the wires to the left until a tongue *G* enters between the rolls and raises *B* out of contact with *C*, which prevents it from revolving and stops the delivery of yarn.

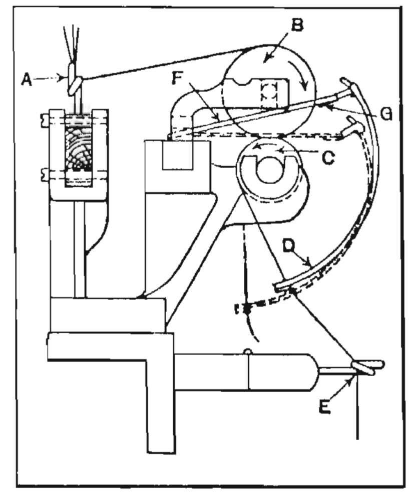


Fig. 13. Tripping Device for a Textile Machine

Another stop motion which acts when a thread is broken is shown in Fig. 14. This mechanism is applied to a machine used for winding thread on spools. It is designed to raise the spool out of contact with a flange which drives it by friction, if a thread breaks, thus arresting the motion of the spool without stopping the spindle on which the spool is mounted. The device is also arranged so that the wire which drops when a thread breaks is raised automatically to its normal position for re-threading. The thread *A* passes through the eye of a drop wire *B* and serves to hold this wire in its normal position. Attached to this wire there is a lever *C* pivoted at *D* and connected by link *E* with the catch *F*. The lever *G* is normally held in a horizontal position by catch *F*. If a thread breaks, however, the dropping of wire *B* releases catch *F* and lever *G* falls to the position shown in the illustration. This lever is connected by a rod *H* with a sleeve *J* pivoted at *K*. The downward movement of lever *G* swings the sleeve about its pivot and brings a pin under the flange *R* of the spool, thus raising it from the supporting disk *L*, as shown in the illustration; at the same time, the flange of the spool engages a rubber disk *m* which stops the rotation. Attached to the shaft of lever *G*, there is a small finger *O* which is given a partial turn when the catch lever falls. As the result of this movement, the finger engages lever *C* and swings it with the drop wire *B* back to the normal position ready for re-threading. As soon

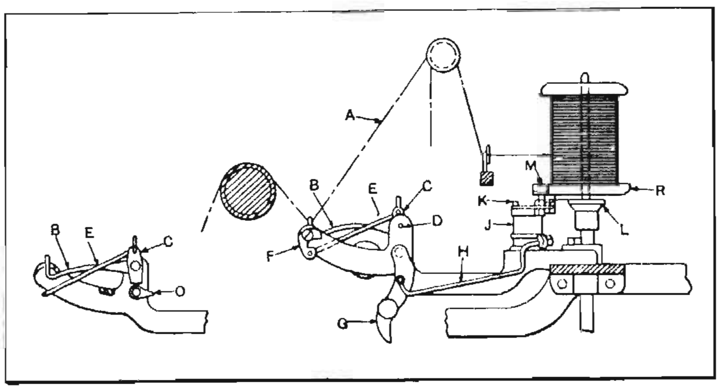


Fig. 14. Another Tripping Device or "Stop Motion" for a Textile Machine

as the catch lever has been re-engaged with the catch *F*, the spool drops into contact with its driving flange and again begins to wind the yarn.

Stop Mechanism That Operates When Work is not in Position. — A machine for placing covers on cans required a device to prevent a cover from being dropped when there was no can in place to receive it. The cans are pushed one at a time in the direction of the arrow (see plan view of the diagram, Fig. 15), and they slide along bridge *B* and stop over table *A*. This table is slotted to allow it to move upward until the recessed surface is raised above the surface of the bridge. As the table rises, the can is centered by a chamfered

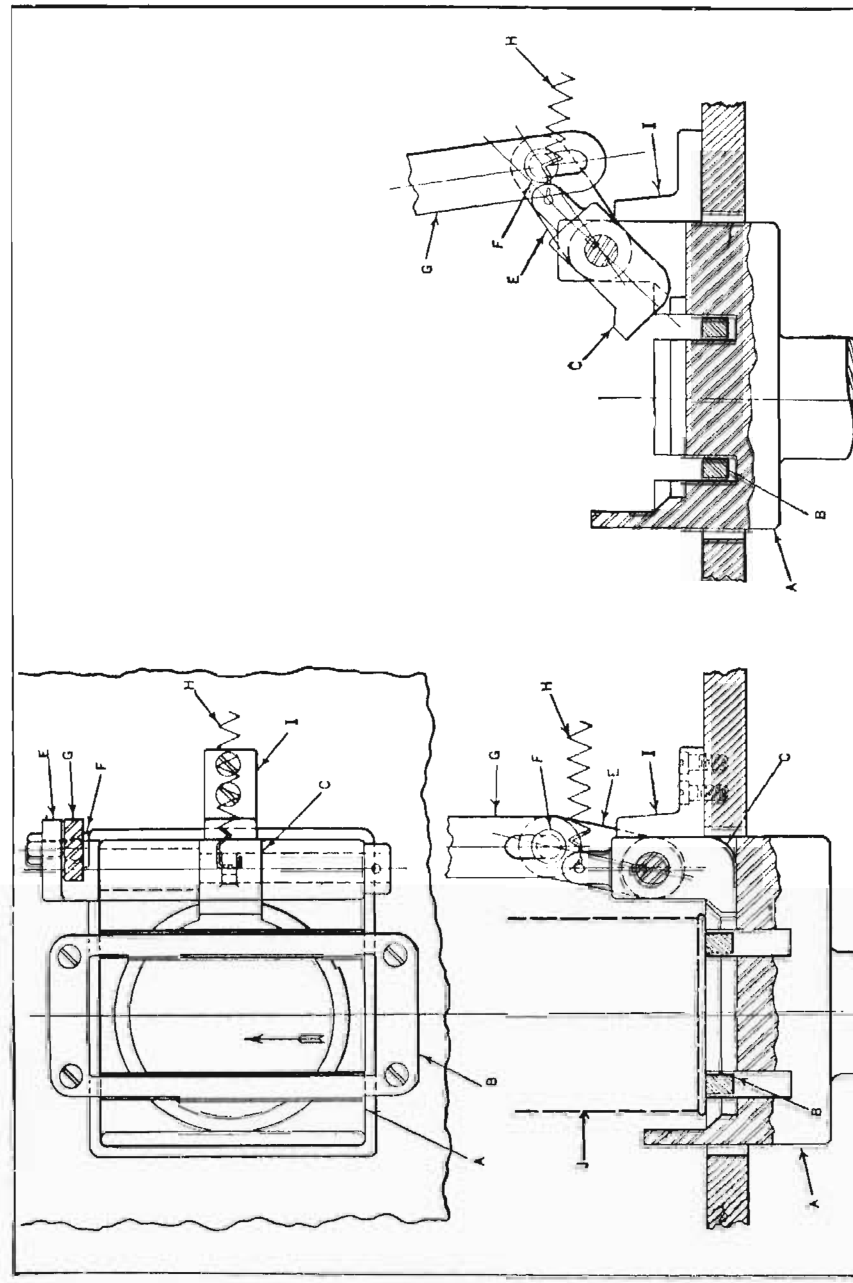


Fig. 15. A Machine which Places Covers on Cans is Equipped with this Mechanism which Stops Action of Cover-dropping Mechanism if Can is not in Position

edge, and raised to receive a cover. The cover dropping mechanism is operated by link *G* which, in turn, is operated by stud *F* on arm *E*.

This device is most easily explained by first considering its action when table *A* rises without a can in place. Referring to the sectional view at the left, a detector foot is shown at *C*. Spring *H* is attached to the upper end of the detector foot and tends to swing it to the right, but this turning is prevented by block *I*. The detector foot is keyed to a shaft which is carried by the table, and when the center of the shaft passes the top of the block *I*, the foot is pulled over by the spring, as shown in the right-hand view. This also allows arm *E* to swing over without causing an upward movement of the link *G*. It will be noticed that this link is slotted so that stud *F* cannot move it until after the detector foot has reached a point where it can swing. Thus it is seen that if there is no can on the table, the detector foot, arm, and link will swing without actuating the cover-dropping mechanism.

Now suppose that a can *J* is in place on the bridge. As the table rises, the can settles into the recess, and when the shaft center passes the top of block *I*, the toe of the detector foot will come into contact with the rim of the can. The can prevents any further swinging movement of the foot, and therefore stud *F* pushes link *G* upward, thereby causing a cover to drop. The center of the stud in arm *E* is placed slightly to the right of the center of the shaft; so that while there is some tendency to swing the foot, it is not enough to crush the can.

Safeguard for Delicate Mechanism of Adding Machine.— The delicate and intricate mechanism of an adding machine is safeguarded from injury resulting from careless or rough operation, by the ingenious mechanism shown in Fig. 16. This controlling device is so arranged that the force or power exerted upon a hand lever by the operator is not transmitted directly to the mechanism, but the operation of the machine is subject to spring action at a certain known rate and with a known driving force. The operator is only allowed to supply

the power for stretching certain springs and releasing their action, the arrangement being such that he cannot apply his strength directly to the mechanism.

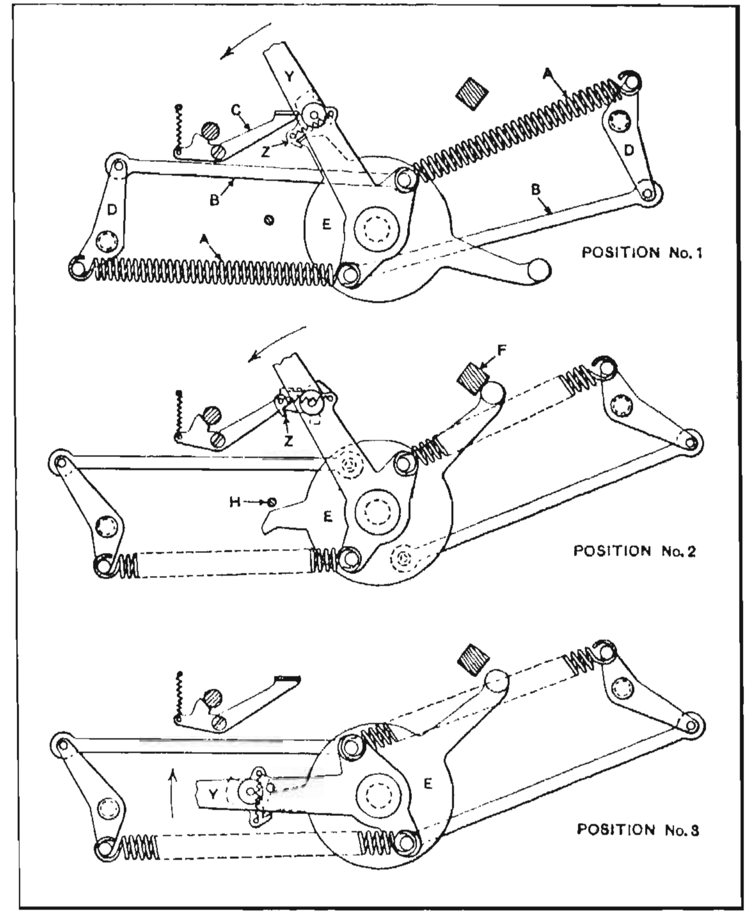


Fig. 16. Spring Controlling Device which Safeguards the Delicate Mechanism of an Adding Machine

The inner end of the operating lever is shown at *Y*. The upper view marked "position No. 1" shows the lever after it has been pulled forward for operating the machine. This lever revolves freely on its shaft and the operating parts are driven by the member *E*. As lever *Y* is pulled forward, catch

C releases pawl *Z*, thus allowing member *E* to fly forward as springs *A* contract and transmit motion to *E* through levers *D* and links *B*. This free movement of part *E* operates the forward stroke of the machine, and the rate of action is controlled by an oil by-pass governor (not shown). The movement of *E* is stopped by abutment *F* in position No. 2. The forward movement of operating lever *Y*, which is now free of the mechanism, is continued by the operator until the lever arrives at position No. 3. At this point, the end of pawl *Z* strikes pin *H*, throwing the pawl back into engagement with the projecting lug on part *E*. As the lever moves from position No. 2 to position No. 3, the springs *A* are extended, and acting through levers *D* and links *B* they return lever *Y* to the starting position, the lever carrying with it part *E* and the mechanism of the machine. This reverse movement is also under the control of the oil governor. Near the end of the return movement, latch *C* rises to permit pawl *Z* to pass, and the various parts return to position No. 1 ready for another stroke.

The provision of two springs *A* is simply for balancing the strain on the mechanism. The double action of these springs, which makes it possible for them to operate the mechanism on both the forward and backward strokes, is due to the fact that they are connected to movable members at each end. In one case, the connection is to levers *D* and in the other to operating lever *Y*. The contraction of these springs between positions No. 1 and No. 2 operates the forward motion, and their contraction from position No. 3 to No. 1 operates the backward motion.

Centrifugal Type of Speed-limiting Device. — The automatic speed-limiting device described in the following was designed for application to gas or gasoline engines. In case the speed becomes excessive, owing to the failure of the governor, this tripping mechanism, which is of the centrifugal type, operates by breaking the ignition circuit. It may be attached either to the secondary shaft or to the main shaft. The controlling element is a weight *A* (Fig. 17) which is attached to a rod connecting with a spring *B* on the opposite side of the hub. This weight is located within a casing *C* carried by a stud *D* screwed into the end of the shaft. Pivoted near the casing is a latch *E* which normally holds the weighted trip-lever *F* in the position shown. The ignition switch is located at *G* and, when the lever *F* is held up by latch *E*, the ignition circuit is closed. If the speed of the engine is increased to such an extent that the action of centrifugal force causes weight *A* to fly outward against the tension of spring *B*, the end of rod *H*, by striking catch *E*, releases lever *F* and allows it to fall, thus breaking the ignition circuit.

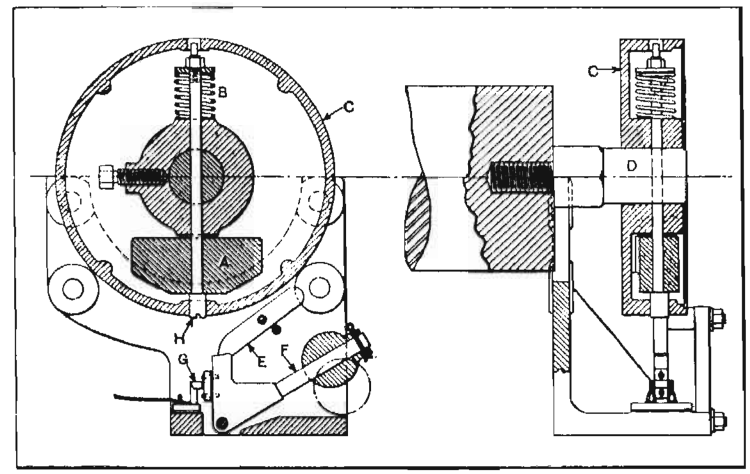


Fig. 17. Centrifugal Type of Speed-limiting Device Designed for Gas or Gasoline Engines

Over-speed Limiting Device for Electric Locomotives. — There exists, on certain railroads, the necessity of employing a reliable over-speed limiting device, which will prevent the application of further propelling power to an electric locomotive after a predetermined speed has been reached. The prevention of over-speeding protects the traction motors or other locomotive parts from being injured by speeds in excess of that for which they were designed. The device should be so designed that the engineer may again apply power as soon as the speed has decreased to some permissible value.

A device that meets these requirements has been developed for application to electric locomotives. It is, however, capable of alteration so that its field of usefulness can be extended to steam or other motive power, as the fundamental device is capable of various simple modifications which will perform different functions as desired. By mounting a Veeder counter on the contact box, a record can be obtained of the number of times an engineer has over-speeded his train. The system may also be interlocked with the air brake by the use of additional relays. One relay can be made to give the engineer a warning and at the same time cause a counter to register. Another relay can start a time element device which will, after the elapse of a definite time, cause the brake pipe to be vented, thus applying the brakes. In case the engineer reduces the speed of the train, when given warning that he is exceeding the prescribed speed, before the automatic feature has had time to work, nothing further will happen and the device will then return to its normal position.

The centrifugal member seen at the right in Fig. 18 is mounted on the center of an idle axle. One half of this member carries the centrifugal arm and adjusting spring, so that adjustments can be made on the test floor and left undisturbed when applying to the axle. The centrifugal arm has two cam surfaces A and B which are offset one from the other. These cams engage light arms A_1 and B_1 on the contact box, which is mounted close to the centrifugal member, as shown by the relative positions in the illustration.

Normally, a spring D holds the centrifugal arm in the "in" position. At some predetermined speed, the arm flies outward about the center C, compressing the spring, while cam A strikes arm A_1. This motion causes a toggle switch in the contact box E to break contact and thus interrupt the control current to the master controller. Thus further application of power to the locomotive is prevented until the speed is reduced. Upon a reduction of speed to the necessary value, the centrifugal arm moves inward about the center C and cam B strikes arm B_1, causing the toggle switch again to make the circuit

in the contact box. This permits the engineer again to apply power to the locomotive. As developed at present, the range of speed between trip out and reset is from 39 to 37 miles per hour in one case and 65 to 60 in another.

In designing the centrifugal member, it was necessary to reduce friction to a minimum and also to produce bearings that would require almost no attention for maintenance. This arm, therefore, was mounted in a double-row, deep-

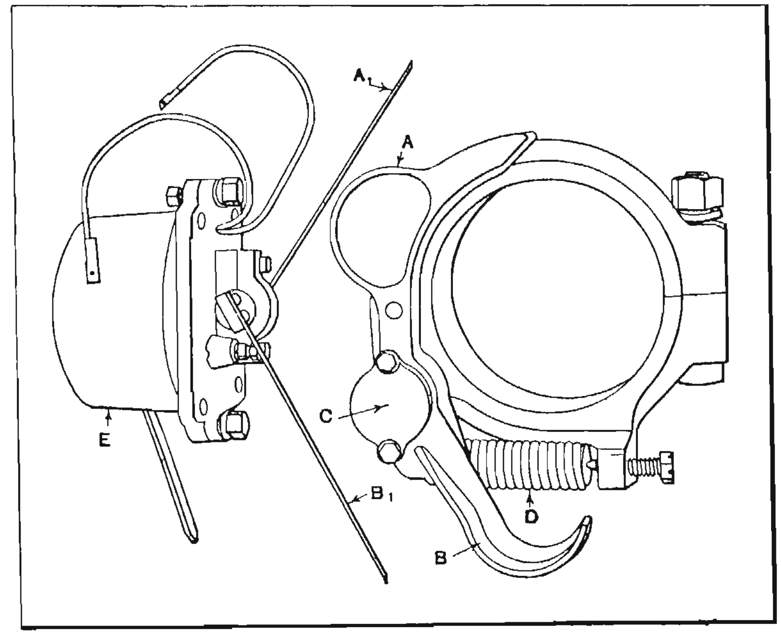


Fig. 18. Device for Electric Locomotives which Automatically Prevents Over-speeding

groove ball bearing, which insures minimum and constant friction over a long period of time. The spring D operates between hardened conical points or centers J and K, bearing in hardened pockets (see Fig. 19). It is restrained from buckling by hardened sliding guides F and G within the spring. The outside of guide G has an increased diameter at H to prevent the spring from buckling. This mounting resulted, on test, in an extremely sensitive centrifugal arm which, depending on the characteristic of the spring, could be made to fly out positively and return for a difference in speed of approximately 1 per cent.

The contact box is designed with special reference to withstanding the hard blows received from the cams. The striking arms are made of hardened steel springs, tapered toward the outer ends to reduce inertia and consequent rebound when the cam strikes. It was found necessary to provide a spring loaded friction drag in box *E* to prevent the striking arms

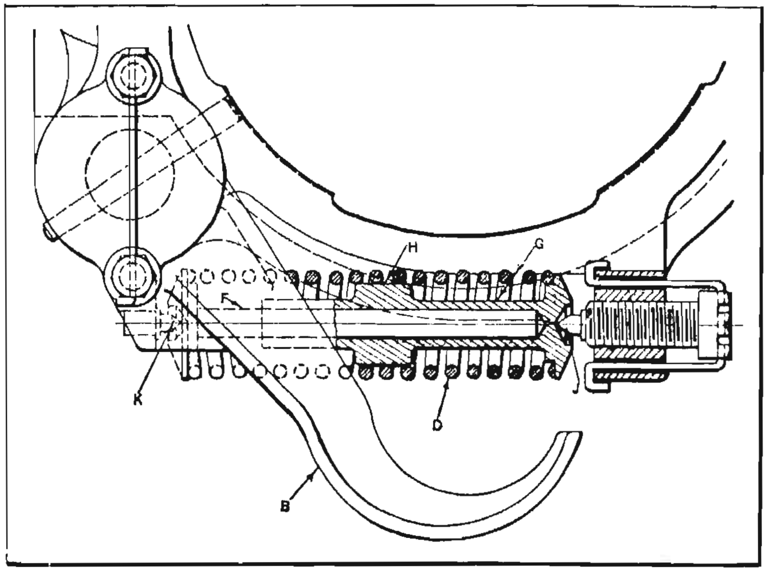


Fig. 19. Detail of Sensitive Spring Mounting for Centrifugal Arm of Over-speed Limiting Device

from rebounding and rubbing against the cams. The moving contact is a light weight element, which is connected to the striking arms only by means of two "over the center" springs, and is supported on knife-edge bearings. The contact does not move until after the striking arms have passed the center. It then snaps over quickly, regardless of the speed at which the striking arms are moving. The contacts are made of phosphor-bronze and in the form of flat springs which bear against copper-graphalloy buttons.

Speed-limiting Device for Steam Engines. — A speed-limiting device which is governed by the inertia of a weight and the tension of a spring is shown in Fig. 20. This automatic stop was designed for application to steam engines but devices operating on the same general principle could doubtless be applied to other classes of machinery. This mechanism is primarily a safety device and is intended to stop the engine and prevent damage such as might be caused by a bursting flywheel, in case the governor failed to operate. The lever *A* is pivoted at *B* to the engine cross-head and is normally pre-

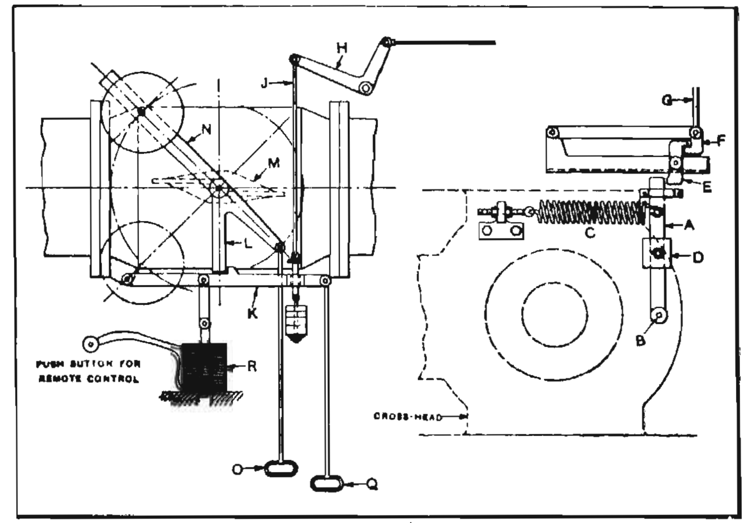


Fig. 20. Automatic Speed-limiting Mechanism for Steam Engines

vented from swinging about pivot *B* by the spring *C* attached near the upper end. The inertia of weight *D*, which may be adjusted along the lever *A*, tends to swing the lever to the right when the motion is suddenly reversed and the cross-head moves to the left. When the cross-head is at one end of its stroke, the upper end of lever *A* is quite close to the catch *E*, which engages latch *F*. Rod *G* attached to this latch connects through whatever additional rods or levers may be needed, with the tripping mechanism used in conjunction with a quick-closing valve which controls the flow of steam to the engine cylinder. This valve and its operating mechanism is shown in detail at the left of the illustration. Rod *G* is connected in any convenient way with bellcrank lever *H*, from which rod *J* carrying weights at its lower end is suspended. This rod passes through trip-lever *K*, which normally engages lever *L* connected with the quick-closing valve *M*. If, for any reason, the speed of the engine becomes excessive, the lever *A* and its attached weight resists the sudden reversal of motion at the end of its stroke sufficiently to overcome the tension of spring *C*, and lever *A* strikes catch *E*, thus releasing latch *F*; as rod *J* drops, the flange on it strikes trip *K* and allows the steam valve to be closed by the weighted lever *N*. This speed-limiting device may be adjusted by varying the tension of spring *C* and also by changing the position of weight *D*. The greater the spring tension and the nearer the weight is to the pivot *B*, the faster the speed will have to be to overcome the tension of the spring at the point of reversal. The handle *O* is for resetting the steam valve and handle *Q*, for tripping the valve by hand. If remote control is required, this may be obtained by the use of rods or cables directly connected to latch *K*, or by the use of a solenoid *R*, as indicated by the illustration. This automatic safety stop is recommended as being simple, positive in action, adjustable, inexpensive, and easily applied to almost any engine.

CHAPTER VIII

INTERLOCKING DEVICES

THE primary purpose of an interlocking device is to prevent the simultaneous engagement of conflicting mechanisms, so as to eliminate, as far as possible, the danger of straining or breaking any of the machine parts. There are many examples of interlocking devices in which the component parts are much more costly to produce than is necessary or justifiable, considering the function of the device. Notwithstanding that the purpose of an interlocking device is to protect expensive parts of the machine from destruction, there is seldom any need to provide other than the simplest mechanism to accomplish the desired end. Elaboration or complication in this connection is unnecessary, and should be guarded against.

A separate movement should not be required to lock or unlock such a device, but the ordinary manipulation of the parts controlled by the interlocking mechanism should, without any further effort insure the correct functioning of the safety device. Otherwise, the operative time will be increased so that there will be much less available time for productive work. Over-elaboration in design as well as a tendency to make the parts much stronger than is necessary should be avoided. Although the latter fault may have very little influence on the cost, a more careful consideration of the design and the stresses imposed might result in much greater compactness and neatness.

Two-lever Type of Interlocking Mechanism. — Fig. 1 shows a typical interlocking device, which, in various forms, is extensively used. The two levers *A* and *B* usually control two pairs of sliding gears by means of which one shaft is driven from another shaft at four different speeds. Since all the four pairs of gears are of different ratios, it is necessary to have some element introduced that will insure that one or the other of the two levers shall always be in the neutral position. Obviously, if some such provision were not made, there would always be a serious risk of one of the levers being accidentally engaged while the other was in engagement. This would, of course, be rather disastrous for the gears and other trans-

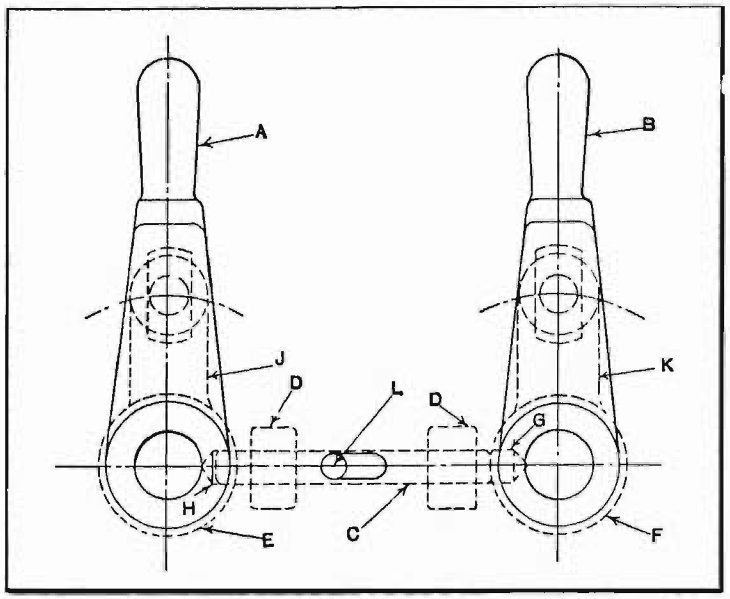


Fig. 1. Simple Arrangement of Interlocking Gear-change Levers

mission elements concerned, and it is to avoid such a possibility that the levers are designed to interlock so that one is always in the neutral position.

The locking element itself consists of a short piece of round steel *C* carried in bearings *D*, and engaging with holes *H* and *G*, in the bosses of the levers *J* and *K*. There is really no reason, except for the sake of appearance, why the sliding rod *C* should not be outside the gear-box housing and engage the bosses of the hand-levers *A* and *B*. A pin *L*, working in an elongated slot and fixed in the rod *C*, is the medium through

which the rod is moved from left to right, or vice versa, as required. This arrangement is non-automatic and locking and unlocking are accomplished in a separate movement. As previously mentioned, this is not a desirable feature.

In the illustration, the lever *B* is shown unlocked ready for engagement. When it is swung either to the left or to the right, the hole *G* in the boss of lever *K*, is moved out of alignment with the rod *C*. It is therefore impossible for rod *C* to be moved, and since its left-hand end is engaged with the hole

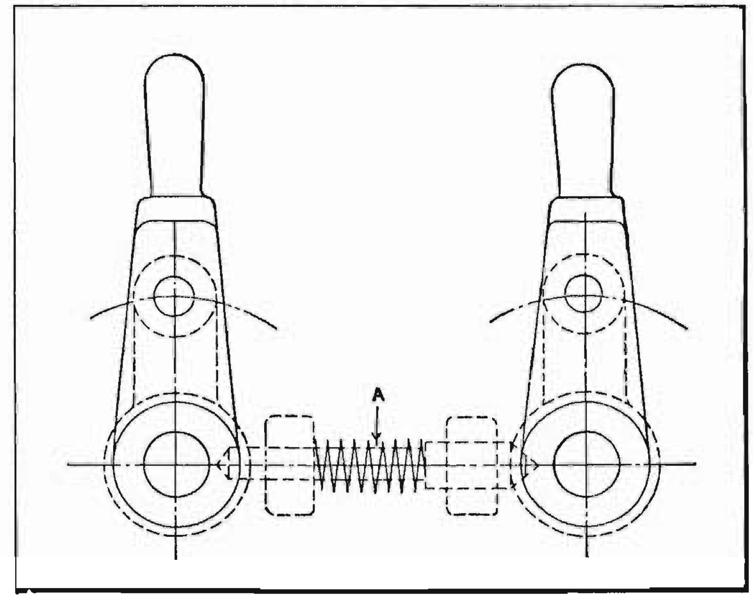


Fig. 2. Gear-change Levers with Automatic Interlocking Arrangement

in lever *J*, it is also impossible to move the lever *A*, and the gears that it controls. If it were desired to use one of the gears controlled by lever *A*, it would first be necessary to move lever *B* to the vertical, or neutral, position, then slide the rod *C* to the right by means of the pin *L*, after which lever *A* could be moved either to the right or left as required, and *B* would be definitely locked out of engagement. It will be seen that the interlocking device involves another movement by the operator, and, since this can be eliminated without adding to the cost of the mechanism, it should be done, if possible, as it insures easier and more convenient operation.

Automatic Interlock for Speed-changing Levers. — Fig. 2 shows an automatic interlocking mechanism in which the objectionable feature of the preceding design is overcome. Here a spring plunger A automatically performs the interlocking function. The plunger is supported in much the same way as in the previous example, but the spring tends to push it to the right. At the right-hand end it has a conical point which engages with a depression in the boss of the lever. Whenever the lever at the right is moved into the neutral position, the plunger is pushed into the hole in the boss by the action of spring A. This unlocks the lever at the left, and allows it to be immediately shifted as required. When the left-hand lever has been moved, however slightly, from the neutral position, the end of the plunger comes up against the boss and prevents the plunger from moving to the left.

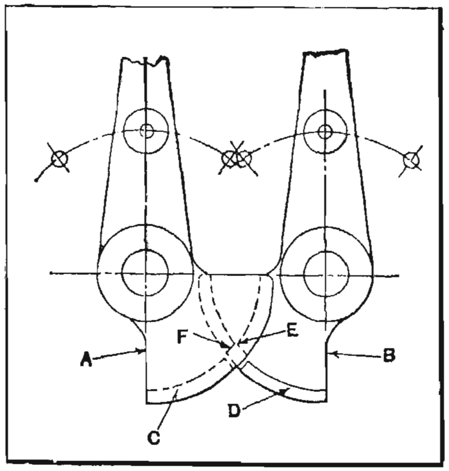


Fig. 3. Interlocking Levers that Can be Made Economically

Since the right-hand lever can be shifted only if the plunger is allowed to move to the left, this lever is positively locked in the neutral position whenever the left-hand lever is in any position other than neutral. The arrangement shown in Fig. 2 is therefore to be preferred to that shown in Fig. 1. It not only works automatically, without requiring any attention or effort on the part of the operator, but it has also the additional advantage that it is rather cheaper to produce.

Levers having Interlocking Segments. — Fig. 3 shows another arrangement for use under conditions similar to the two

previous ones. The two levers are each fitted with a fan-shaped projection A and B. On the front side of the latter and the rear side of the former are two ribs D and C, respectively, which interfere with each other. Grooves E and F are cut across the center of the ribs to allow either of the levers to be moved as required. When the two levers are in the position shown in the illustration, the gears are in the neutral position, and either lever may be shifted. Whichever lever is moved, and no matter whether it is moved to the right or to the left, one or the other of the ribs is moved across the

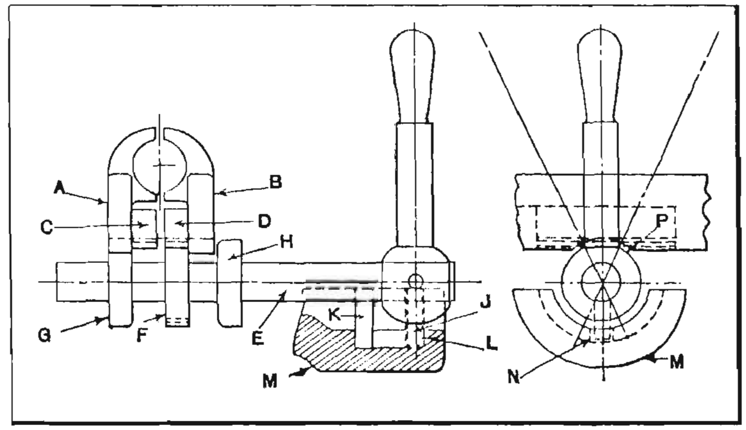


Fig. 4. Interlocking Single-lever Control for Two Sets of Gears

gap in the other rib, thus positively locking it in the out-of-gear position. The arrangement is quite an effective one and very cheap to produce, the only criticism that might be made against it being that it takes up more space than some of the other types. Good designers take a pride in concealing such devices, as a rule, but whether the practice is worthwhile when it involves greater expense is a matter for individual decision.

Single-lever Interlocking Device. — While the three previous examples are much used in machine tool practice, none of them is ideal because there are two levers which must be manipulated in order to operate the sliding gears. The design shown in Fig 4 is much superior in this respect, since only one hand-lever is required, and this lever is so arranged that it not only operates both sets of gears but also functions as an interlocking device. This combination permits quicker operation and, incidentally, gives greater compactness to the design.

The two sets of sliding gears are engaged by forks carried on bars *A* and *B* to which are secured racks *C* and *D*. The hand-lever shaft *E* has a pinion *F* and two plain disks *G* and *H* formed solid with it, and is so arranged that it can be shifted endwise as well as rotated on its own axis. The hand-lever is pinned or keyed to the shaft, and is fitted, in addition, with a pin *J* which projects into two segmental grooves *K* and *L* in a hollow semi-cylindrical casting *M*.

Connecting the two grooves and corresponding with the vertical position of the hand-lever, is a cross-groove *N*, approximately equal in width to the diameter of the pin *J*. With the hand-lever in the position shown, the pinion *F* is in mesh with rack *D*, and consequently the gears controlled by slide-bar *B* will be operated if there is any angular movement of the hand-lever. If the lever and its shaft are pushed bodily to the left, the rack *C* will be engaged by the pinion *F*, and the second set of gears will thus be brought under control. The function of the two disks *G* and *H* is to lock the slide-bar that controls one set of gears in a central position when the other set is in mesh. This is done by the disk being moved into suitably shaped hollows formed in each of the slide-bars, as shown at *P*.

In the illustration, the disk *G* is shown engaged with slide-bar *A*, so that the gears controlled by this bar are definitely locked out of engagement. If it is required to operate slide-bar *A*, the hand-lever must first be moved to the vertical position (all gears then being out of mesh), after which the lever is pushed to the left, thus engaging the pinion with rack *C*, and the disk *H* with slide-bar *B*. In this position, the gears controlled by slide-bar *B* are positively locked out of engagement.

The purpose of the grooves in casting *M* is to insure that the lever-shaft will not be moved axially, except when it is in the neutral position. This part of the device and the pin *J*

could, if necessary, be carried inside the gear case, and the pin *J* could be fitted to either of the disks *G* or *H*. On these lines the design would be somewhat neater, though it is possible that it would not be quite so convenient in operation, as the grooves would then be out of sight and the exact vertical position of the hand-lever would have to be judged, more or less, by the "feel."

Interlocking Device for Lathe Apron. — Lathe aprons are usually fitted with some arrangement for interlocking the conflicting gear mechanisms. In some cases the cross and

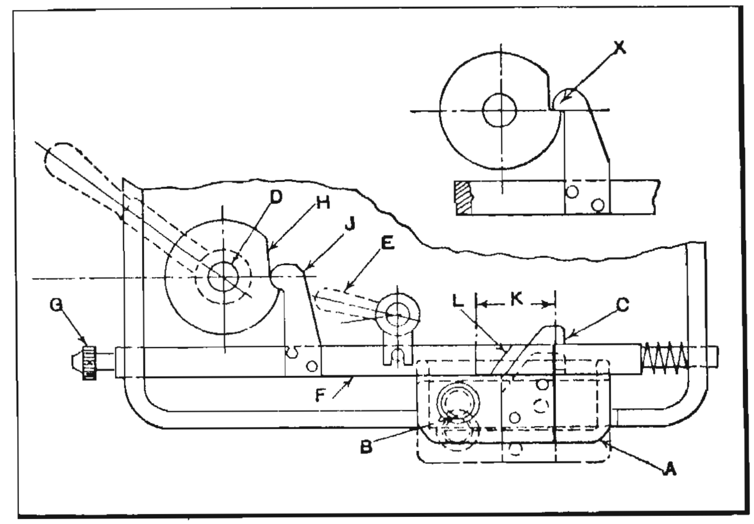


Fig. 5. Interlocking Device for Lathe Apron

longitudinal feeds are not interlocking but provision is made to prevent the engagement of the lead-screw nut when either of the other two feeds is in use. The design shown in Fig. 5 is of this type, the only interlocking action being between the lead-screw nut and the feed combinations.

The feeds are engaged by a drop-worm, carried in box *A* and engaged by handle *B*, the trigger *C* holding the drop-worm box in the engaged position. The square rod *F* has a slight endwise movement, and is cut away near the worm-box to provide a support for trigger *C*. To release the feed, the rod is moved to the right by means of handle E, while a knurled-head screw G, for fine adjustment, forms the abutment for a dog fixed to the bed of the lathe which trips the feed automatically at any predetermined point. At D is the camshaft for operating the lead-screw nut. A counter-clockwise rotation of this shaft from the position shown closes the nut on the screw. Fixed to the camshaft is a cam H which, when the lead-screw nut is engaged, pushes the rod F to the right through the medium of lever J which is firmly secured to the rod. This action releases the trigger C and allows the drop-worm to fall out of engagement with the worm-gear, thus disengaging both the cross and longitudinal feeds before the lead-screw can be used. There are, however, several details in this type of design that are open to criticism, as will be pointed out in the following.

The arrangement illustrated does not meet all the requirements of a satisfactory interlocking device. Suppose a machine were fitted with a device of this type, and the work required that the lead-screw be geared up, and running, the whole time, as would be the case on almost any job that involved both turning and threading.

Suppose now, that the tool had been set up for, say, facing a moderately deep shoulder and that the cut had nearly reached the body of the shaft. If at this time the handle on the camshaft that operates the lead-screw nut happened to be accidentally depressed, the cross-feed would be disengaged, but the lead-screw would come into action and there would be the possibility of a smash-up. There are also similar possibilities with the longitudinal feed and the lead-screw, though in this case the damage would probably not be quite so extensive.

The trouble is due to the fact that the feeds are not really interlocked, because this term means, that when one feed is engaged it is impossible to engage another; in any case it does not mean that the accidental engagement of one particular feed merely disengages the one that is in use, as in the type shown in Fig. 5. If this illustration is carefully examined, it will be seen that the portion of the rod F that is cut away at K to

form the trigger is much wider than it need be. If the width were reduced and the left-hand side were beveled as at L to match the back of the trigger piece, the whole principle of the device would be changed, and it would become an effective interlock, provided cam H and lever J were made of such a shape that the former could not be made to impart any movement to the latter. To meet these requirements, cam D and lever H should be designed as shown at X.

When the trigger is engaged with the rod, that is, when the worm is in mesh, the lead-screw nut cannot be closed, because lever X is held up by the cam, so, obviously, it is not possible to engage the nut when the worm and gear are in mesh. Further, when the nut is engaged, the beveled face L prevents the engagement of the worm, as it interferes with the back side of the trigger when the rod F is positively prevented from movement toward the left. This simple alteration has converted an indifferent and imperfectly operating device into one that is correct.

Lathe Apron Interlocking Mechanism for Three Feeding Movements. — Fig. 6 shows another type of lathe apron in which all three feeds are interlocked to prevent conflicting engagement. In this design there is no drop-worm, the actual starting and stopping of the feeds being effected by a hand friction knob A, which couples gear B to the worm-gear C. At D is the rack pinion on which is keyed the large gear E, while at G is the cross-feed screw pinion which permanently engages with an intermediate gear H. Approximately midway between the intermediate gear H and the rack gear E is an eccentric shaft J carrying a gear F and a pinion Y. The eccentric shaft is controlled by a small lever K, which swings into three positions—the one shown, one to the right, and another to the left. In the position shown, the feed gears are entirely disconnected; if the lever is swung over to the right, pinion Y engages gear E, and gear F engages friction gear B. If swung the opposite direction—to the left—the eccentric shaft gear F connects the intermediate gear H to gear B.

In the first case the longitudinal feed gears are engaged, while in the latter case the cross-feed gears are connected. Lever K is used only for engaging the required feed gears; that is to say, it does not actually start or stop the feed, this being done by the hand friction knob A. Up to this stage the mechanism is perfectly interlocked, because when lever K is swung over, either to the right or to the left, only the particular feed represented by the given location of the lever can be engaged, the other being totally inoperative—that is, of course, as an automatic feed.

Since the lever cannot be in two positions at once, only one feed can be engaged at a time, the change from longitudinal to cross, or vice versa, necessitating, first, the disconnection of the feed by the hand-knob; second, the correct placing of the lever for the feed required; and third, the actual starting up of the feed by the hand-knob. So far as the cross and longitudinal feeds are concerned, then, we may assume that fairly complete interlocking is secured. However, there still remains the screw-cutting mechanism.

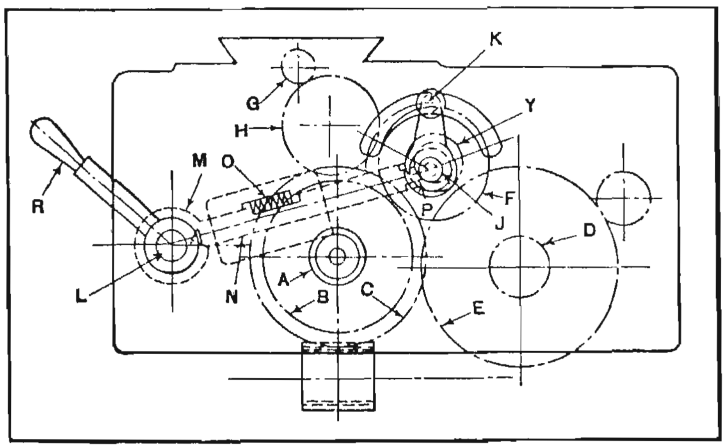


Fig. 6. Lathe Apron with All Feeds Interlocking

The lead-screw nut is operated by a camshaft L, and on this shaft is a collar M in which is cut a V-shaped groove. On the inside of the apron casting is a rectangular groove running from camshaft L to the eccentric shaft J, and fitting into the groove is a plunger N which is pressed always in the direction of the camshaft by a spring O. The lower or left-hand end of the plunger is pointed to fit into the V-shaped groove in collar M so that, in effect, the plunger acts on the camshaft as a snap plunger, thus reducing the risk of its being moved by vibration. At the opposite end, the plunger is narrowed down so as to enter a cross-groove P, cut in the boss of lever K. When the lever is in the position shown (the feed-gears being out of mesh), the camshaft handle R can be depressed and the nut connected with the lead-screw. The plunger N, meanwhile, is pushed to the right into the groove P in lever K, thus effectually locking the latter in the central position.

Suppose, now, that the lever K had been moved to either the cross or longitudinal feed position. The groove P in the boss on the lever would not then be in alignment with the plunger, so the latter could not move toward the right. This being the case, the nut-operating handle R could not be depressed, as it would be locked by the V-point of the plunger. Thus, only when the lever K is in the central or out-of-gear position can the lead-screw nut be engaged, and on the other hand, only when the lead-screw nut is disengaged can the feed-lever be moved, all three feeds being by this simple means, interconnected so that no two can possibly be in simultaneous engagement.

Simple Interlock for Lead-screw and Longitudinal Feed.— An example of effective design combined with low production costs is shown in Fig. 7, the interesting feature of this particular arrangement being that a perfect form of interlock is provided without the introduction of a single extra part. The device is fitted to a lathe apron and is of the type that applies only to the lengthwise feed and screw-cutting movement, the cross-feed not being interlocked in any way. The eccentric shaft A operates a sliding clutch which engages the longitudinal feed when handle B is correctly manipulated. When the handle is pulled outward, the clutch is engaged and the feed started. At C is the cam-plate which controls the lead-screw nut; in the position shown the nut is disengaged.

On the right-hand side of the cam-plate is cast a projection *E,* while on the boss of handle *B* is a lug *D,* the lower side of which is roughly located at the same height as the upper end of the cam-plate projection *E* when the lead-screw nut is disengaged. If the handle *B* is pulled outward to engage the feed, lug *D* moves over the end of projection *E* and prevents the cam-plate from being rotated to engage the nut. On the other hand, should the nut be first engaged, projection *E* moves upward into the position shown by the dotted lines at *F* and prevents the engagement of the feed by interfering with

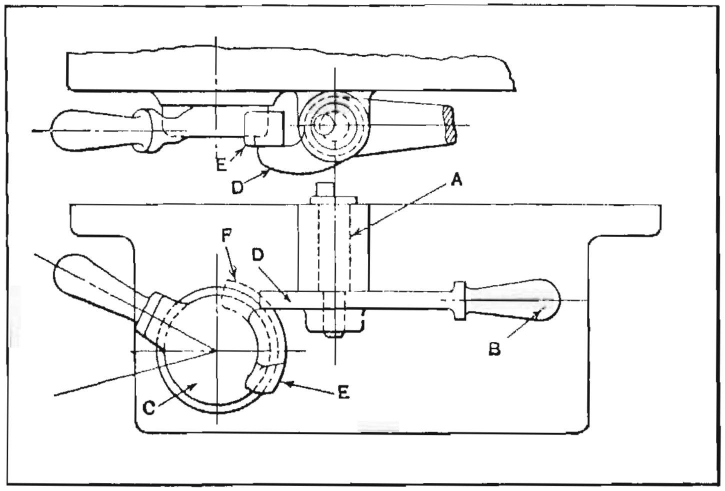


Fig. 7. Simple Interlocking Device for Lead-screw and Lengthwise Feeding Movement

lug *D* on the hand-lever. So far as it goes, the design is perfectly effective and a good example of simplicity and cheapness, though, of course, it is rather lacking in completeness in that it does not also interlock the cross-feeding movement of the tool slide.

Interlocking Device for a Back-shaft Type of Lathe. — Fig. 8 shows still another example of a feed interlocking device as applied to the saddle of a back-shaft type of lathe. Ordinarily, in the old style of back-shaft lathe, the three feeds, cross, longitudinal, and screw-cutting, are almost always entirely disconnected, so there is nothing to prevent all three being simultaneously engaged. To overcome the risk from this possibility, the arrangement illustrated in Fig. 8 was patented some years ago by an English firm. At the back of the saddle are two gears *A* and *B,* the former mounted on the cross-feed

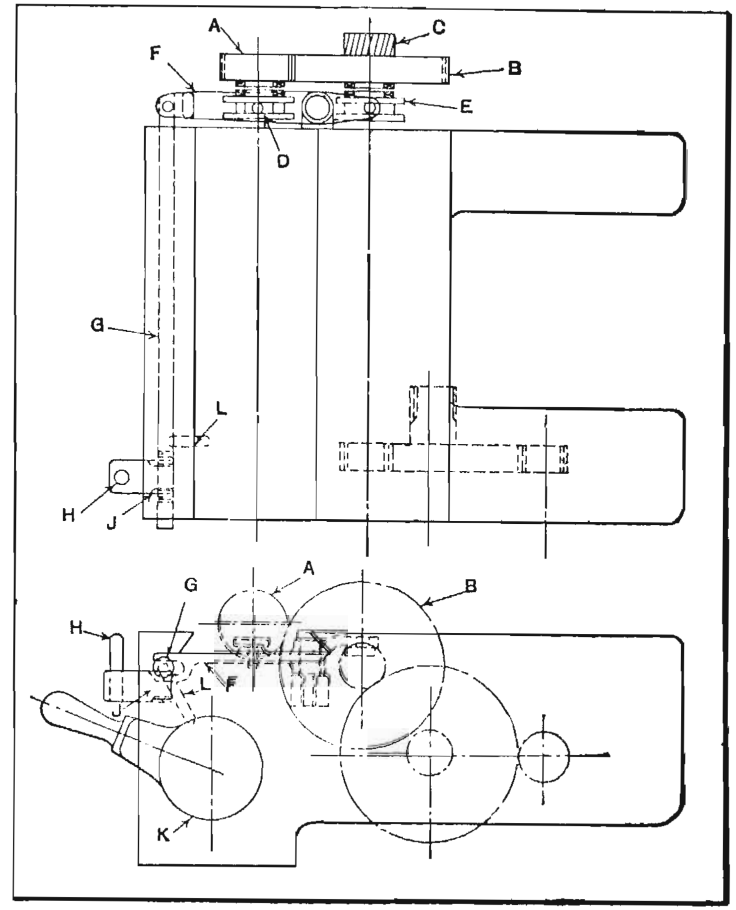


Fig. 8. Back-shaft Type of Lathe Saddle with All Feeds Interlocking

screw and the latter on the feed-shaft for the longitudinal feed. Gear *B* is driven from a worm *C* which is keyed to the back-shaft and free to slide on it, and both gears *A* and *B* run loosely on their shafts and are provided with clutch teeth on their front faces, which are engaged by clutches *D* and *E,* which are feathered to the cross-feed screw and the feed-shaft, respectively. Both clutches are operated by a single lever *F* through a small handle *H* and rack and pinion *G* and *J*.

When lever *H* is pushed toward the rear, clutch *D* is engaged with gear *A*, the other clutch *E* being meanwhile moved further away from its mating gear *B*. If the handle *H* is pulled toward the front, clutch *E* is moved into engagement, thus starting up the longitudinal feed. When the handle is vertical or in the mid-position, both clutches are out of engagement, and the cross and longitudinal feeds are inoperative. Obviously, then, it is impossible for the two feeds to be engaged at the same time.

Referring now to the screw-cutting motion, the nut is engaged through the usual type of cam-plate *K* by depressing the handle, so that the two half-nuts are engaged with the lead-screw. Immediately above the cam-plate is a short length of round rod *L*, the lower end of which is pointed to fit into a conical depression in the periphery of the cam-plate. The upper end just touches the round rack *G* when rod *L* is in its lowest position. Corresponding with the neutral position of the two feed clutches is a hole in rack *G* opposite rod *L*. When handle *H* is in the mid-position, the cam-plate handle can be depressed, rod *L* meanwhile being slightly raised until its upper end enters the hole in rack *G*, thus preventing the latter (and with it the feed clutches) from being moved. When the cam-plate handle is in the upper position so that the half-nuts are disengaged from the lead-screw, either of the feed clutches can be engaged as required, but no matter which is engaged, the hole in rack *G* is moved out of line with rod *L*, so it is impossible for the rod to be raised, and consequently for the lead-screw nuts to be operated. It will be seen, therefore, that only one of the feeds can be in operation at any given time, so that the engagement of any of the three feeds definitely locks the remaining two out of engagement.

Interlocking Device for a Geared Drive. — Fig. 9 shows a more complicated system of interlocking, as utilized in the design of a six-speed, all-geared drive machine tool. In the

example to be considered the transmission consists of nine gears and three shafts. The drive is from a pair of tight and loose pulleys. The speed of the pulleys is about 500 revolutions per minute, and it is this comparatively high speed that makes it necessary that all the speed-changing levers be interlocking, so as to minimize the risk of damage by preventing the engagement of the gears while under load. The correct relative location of the shafts is shown in the end view at the left. The plan view at the right merely shows the three sets of gears as if they were all in the same plane. At *A* is the

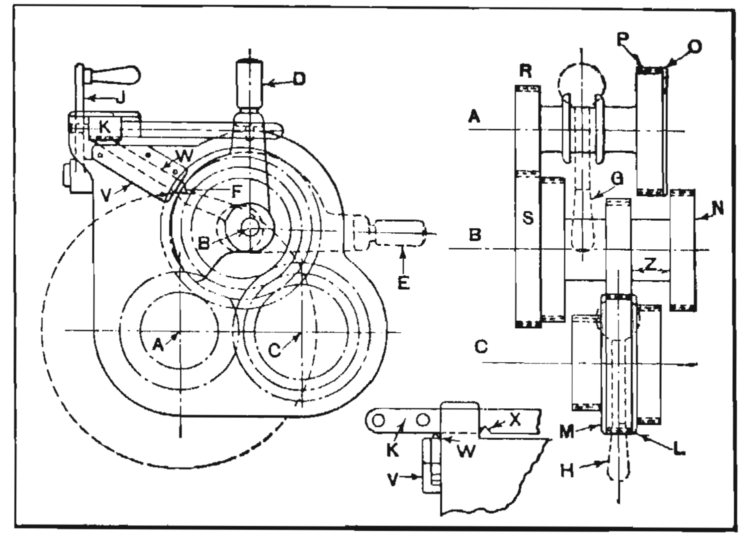


Fig. 9. Gear-box Drive with All Controls Interlocked

pulley shaft which carries twin gears having an axial movement that permits them to be engaged with either of two gears on shaft *B*. The latter is an eccentric shaft carrying four gears, all of which are fixed together, but run loose (as a whole) on the shaft. Shaft *C* carries three gears which may be engaged at will with the three mating gears on shaft *B*.

The gears on shaft *B* give three speeds, which are doubled by the twin gears on shaft *A*, thus giving six speeds in all. It might be mentioned, at the outset, that this design of gear-box is not presented as an ideal one, but it is an excellent example of interlocking, and it is only with this point that we are immediately concerned. The eccentric shaft *B* is really the foundation of the whole system. This shaft is controlled by a lever *D*, fitted with a spring plunger which holds it in the vertical position. Slightly more than a 90-degree angular movement is required to bring the lever into the position indicated at *E*, which causes the gears on the eccentric shaft to be thrown entirely out of mesh both with the gears on shaft *A* and with those on shaft *C*.

Referring now to shaft *A*, the larger of the two gears carried by this shaft has a washer *O* secured to it, which is of a diameter equal to the external diameter of the gear itself. This washer must lie either on the right or on the left of gear *N*; that is, it cannot be located anywhere in the width of face of gear *N*, because it would interfere with the teeth. The washer *O* thus insures that either the gears *R* and *S* or *P* and *N* are in mesh. Were there no washer, it would be possible for the twin gears *R* and *P* to be in a mid-position, with two pairs of different ratio gears in mesh at the same time. Of course, this trouble could be obviated by increasing the distance between the gears, but such a course would affect the interlocking properities of the design as originally conceived. The washers *M* and *L* on the largest of the three gears on shaft *C* serve a similar purpose to washer *O*. These washers prevent the half and half engagement of the gears, and necessitate that the eccentric shaft be thrown into the out-of-mesh position before the sliding gears can be shifted. It is really the object of the washers to insure that the eccentric-shaft gears are disconnected before a change of speed is made.

The next step is the connecting of the eccentric shaft to the belt-shifting gear, and this is provided for quite simply. The belt forks are mounted in a fairly substantial belt bar *K*, which is moved by the lever *J*. On the end of the gear-box is bolted a short slide *V* provided with a plunger *W* which has pointed ends. The lower end enters a V-shaped groove *F* in the eccentric-shaft lever, and the plunger is held down against the lever by gravity. In the belt bar *K*, is another V-

shaped groove which receives the opposite end of the plunger.

When the belt is on the fast pulley, the belt bar is in its extreme right-hand position, and the V-groove *X* is out of line with the plunger, the plunger end coming snugly up against the body of the bar. In this position, the eccentric-shaft lever is locked by the plunger and is engaged with groove *F* in the quadrant of the lever. Since none of the sliding gears can be moved (because of the washers previously referred to) until the lever has been swung to position *E*, it follows that no change of speed can be made until the belt is first shipped on the pulley and the power thus shut off. Further, when the belt bar is moved to the left until groove *X* is opposite the upper end of the plunger—the belt then, of course, being on the loose pulley—the act of swinging down the eccentric-shaft lever raises the plunger into groove *X*, thus locking the belt on the loose pulley until the eccentric shaft is again returned to its "in-gear" position.

With the lever in position *E*, the two sets of sliding gears may be manipulated as occasion requires, and when the desired change has been made, the gears are again engaged by returning the lever to its original position, after which the belt is shipped to the fast pulley. If the sliding gears are not correctly positioned sidewise, the washers *O*, *M*, or *L*, by interfering with their respective gears, will not allow the eccentric shaft to go fully back into its normal "in-gear" position, so that, here again, the operator has little chance to blunder .

To alter the speed, the sequence of movements is as follows: (1) Move the driving belt on the loose pulley; (2) swing down the eccentric-shaft lever; (3) move the two sets of sliding gears to the required positions by means of levers *G* and *H*; (4) swing back the eccentric-shaft lever; and (5) ship the belt on the fast pulley. The interlocking arrangements insure that this sequence is always followed, and though it may sound a little complicated and involved, it is much more quickly done than described, and the arrangement certainly removes practically all risk of injury to the gear.

Regarding the introduction of an eccentric shaft, there may be some difference of opinion as to the efficacy of this arrangement. It is open to the objection that it tends to slow down the operation of the machine, because of the necessity for allowing the machine to come to rest before the eccentric shaft can be safely swung back into the "in-gear" position. When ordinary sliding gears are used, the power is first shut off, the necessary speed changes made, and the belt reshipped, before the machine has fairly come to rest, thus saving a certain amount of time. On the other hand, there is one advantage to be derived from the use of the eccentric shaft, and that is the fact that double helical gears can be used for the driving gears, if desired. If this is done, the drive will be much smoother and less noisy — two features of importance when the speeds are so high.

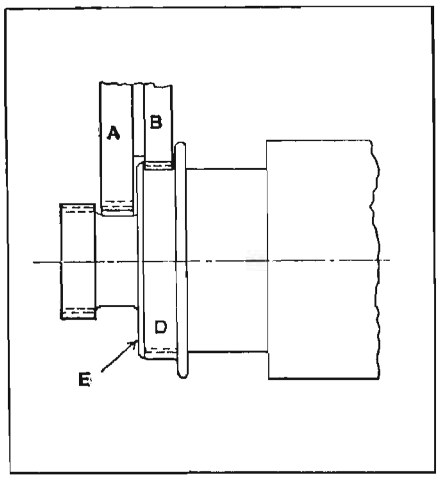


Fig. 10. Interlocking Device for Double Back-gears

Another advantage of the eccentric shaft is that it reduces the over-all length of the gear-box, because it does away with the necessity of having a space of two face-widths between the gears, as at *Z* With ordinary sliding gears, this distance would have to be slightly greater than twice the face width of the gear so as to allow one gear to slide out of engagement before the other entered. By swinging the gears out of mesh, as is done by the eccentric shaft, the distance *Z* between the fixed gears is reduced to slightly over half, thus giving greater compactness.

Interlocking Arrangement for Back-gears. — Another application of the washer or shroud idea is illustrated in Fig. 10, which shows a simple device by means of which the double

back-gears of a lathe or other machine are prevented from being engaged so as to cause damage. As ordinarily made, there is nothing to prevent double back-gears from being engaged with both pairs of teeth in mesh, in which case serious damage would probably result. If, however, the gears are arranged as shown, with a washer, or shroud, equal to the external diameter secured to the larger cone gear *D*, there is not the slightest possibility of the gears being otherwise than correctly engaged.

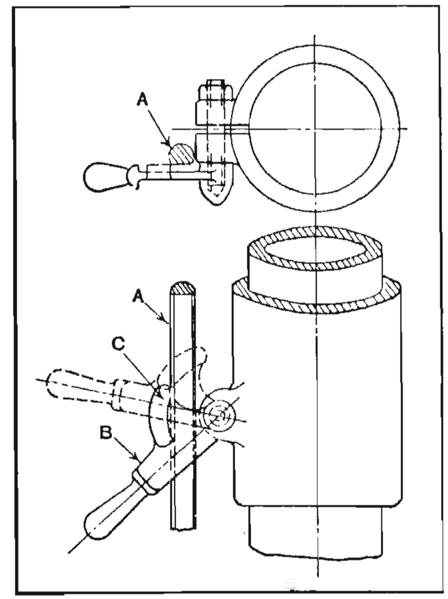


Fig. 11. Interlocking Device for Power Elevating Mechanism

In some cases the two sliding gears *A* and *B* on the quill are not provided with washers and are pushed across from one position to the other by hand, so that should they happen to stick in some intermediate position, both pairs of gears would be engaged at once if the eccentric gear-meshing shaft were manipulated.

With the arrangement shown, the gears must be correctly located sidewise; if they are not, the washer *E* will prevent their engagement by coming in contact with the top of the teeth of sliding gear *B*. Incidentally, it might be mentioned that double helical gears in this connection would do away with the necessity for any interfering device whatever, as the V-shaped teeth would prevent any axial movement unless the gears were first thrown out of mesh.

Interlocking Device for Power-driven Machine Tool Elevating Mechanism. — Power elevating mechanisms, which are often used in machine tool design, should be equipped with some form of interlocking device which would insure that the parts to be elevated are unclamped before the power is applied. The arrangement illustrated in Fig. 11 is designed to prevent such accidents. The shaft *A*, which by partial rotation controls the elevating gears, has a wide flat on one side, while the clamping lever *B* has a fan-shaped projection *C* which, when the lever is in the upper position, fits snugly against the flat of the shaft and thus prevents the shaft from being rotated. When the lever is pulled into the lower position (as shown by the full lines), the parts to be elevated are unclamped, the tail of the lever moves clear of shaft *A*, and the elevating gears may then be engaged as required, in the certainty that everything is in order and that there is no risk of breakage.

In some cases a different method from the one just described is followed. Somewhere in the elevating mechanism—usually in the connection between the elevating screw and its driving gear—a slipping device is fitted, this device being so adjusted that it is quite capable of performing its normal duty; but should there be any accidental over-running, or should the operator have forgotten to unclamp, or any other abnormal circumstances arise which unduly increase the load, there would be a slippage and not a breakage, thus saving the machine from serious damage and breakdown.

CHAPTER IX

DRIVING MECHANISMS FOR RECIPROCATING PARTS

Machines of many different types are equipped with some form of mechanism for changing a rotary motion to a rectilinear or straight-line motion, or *vice versa*. The design of such a mechanism may depend upon the kind of motion required, the amount of power to be transmitted, or other considerations. In this chapter, various mechanisms, especially of the more unusual designs, are described.

Relative Motions of Crankpin and Cross-head. — In some cases, especially in connection with steam engine work, it is important to note the relative motions of the crankpin and cross-head, or whatever part has a straight-line movement. The crankpin has a practically uniform velocity, but the sliding member, which in the case of a steam engine consists of the cross-head and piston, has a variable velocity. Each time the cross-head reaches the end of its stroke, it starts from a state of rest and the velocity increases during approximately one-half of its stroke and then decreases until the cross-head again comes to a state of rest at the opposite end of the stroke. The relative positions of the crankpin and cross-head also vary at every point of the stroke. The position of the crank when the cross-head has traversed one-half its stroke is indicated by the diagram Fig. 1. If the crank were rotating in the direction indicated by the arrow, it would turn through some arc *a* less than 90 degrees, to bring the cross-head to its mid-position, and through a greater arc *b* for the remaining half of the stroke of the cross-head. It will thus be seen that the relative motion between the cross-head and crank during the first half of the stroke is different from that of the second half. This variation in movement is further illustrated by locating the distances that the cross-head moves for equal movements of the crank; for example, if the crank is moved through an arc *c*, from the dead-center position, the cross-head will move a distance *y*, but if the crank is placed on the opposite dead center and then moved through an arc *d*, which is equal to *c*, the cross-head will move a distance *z*, which is less than *y*. This is due to the fact that one-half of the crankpin circle curves toward the cross-head, whereas the other half curves away from it. This variation of motion has an important effect on the design of steam-engine valve-gears, and it is objectionable in some types of mechanisms. The length of the connecting-rod from the center of the cross-head wrist-

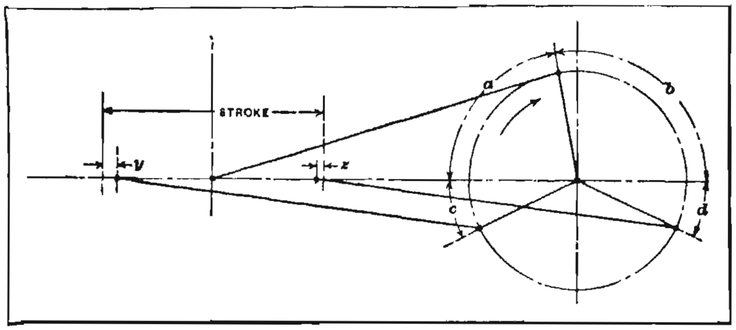


Fig. 1. Diagrams Showing Relative Motions of Crankpin and Cross-head

pin to the center of the crankpin is usually equal to from 4½ to 6½ times the crank radius, on steam engines.

Crank and Slotted Cross-head or "Scotch Yoke." — The irregularity in the motion of a cross-head relative to the crank with the ordinary form of crank mechanism depends upon the length of the connecting-rod. The greater the length of the connecting-rod, the less the irregularity of motion. If it were practicable to use a connecting-rod of very great length, the horizontal movement of the cross-head would be practically the same as the movement of the crankpin measured horizontally. If the connecting-rod were of infinite length, theoretically the movement of the cross-head and crankpin in a horizontal direction would be alike.

A simple form of mechanism for eliminating the irregularity of motion common to all ordinary crank drives, is known as a "crank and slotted cross-head" or the *Scotch yoke*. The cross-head *a* (see Fig. 2) has a slot which is at right angles to the center-line *xx* representing the direction of rectilinear movement. The crankpin carries a block, which is a sliding fit in this slot, and is free to revolve about the pin. As the crank revolves, the distance which the crankpin moves, as measured

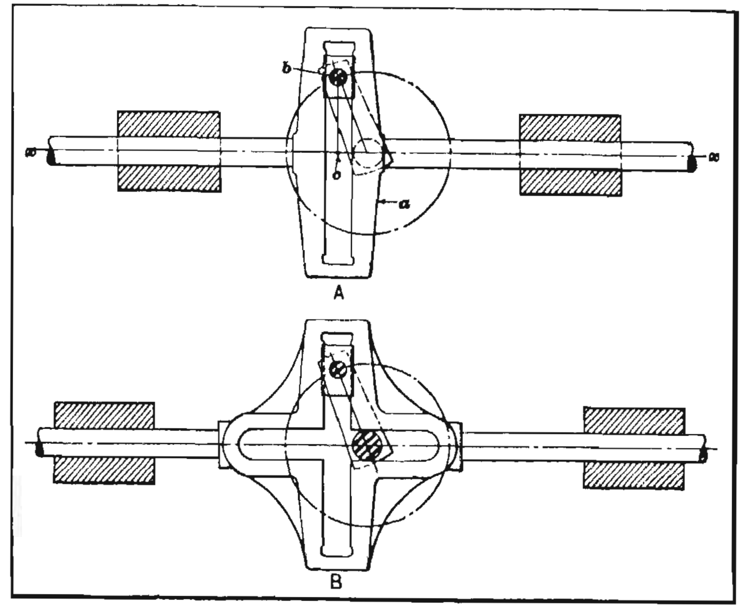


Fig. 2. Slotted Cross-head or Scotch Yoke

in a horizontal direction, will be the same as the movement of the cross-head. This mechanism is sometimes called a *harmonic motion*, because if the crank rotates uniformly, the cross-head will be given a harmonic motion. When a point, as at *b*, moves with uniform velocity along a circular path, point *c* will have a harmonic motion along the center-line *xx*; hence, harmonic motion may be defined as the movement of a point along the diameter of a circle, which is projected from a point moving with uniform velocity along the circumference.

The crank and slotted cross-head has been applied to some types of steam pumps. One of the rods extending from the slotted cross-head carries the steam piston and the other, the water piston. The crank is a driven member, and its radius regulates the length of the stroke. By mounting a flywheel on the crankshaft, steam may be cut off before the end of the stroke and used expansively, because of the energy stored in the flywheel. The crank and slotted cross-head is a very compact form of mechanism, although the sliding motion of the block in the slotted member causes more friction and wear than the ordinary crank and connecting-rod. The latter is also simpler in construction and is, therefore, used almost exclusively as an engine connection, as well as for many other classes of machinery.

The diagram *B*, Fig. 2, shows a modification of the crank and slotted cross-head or Scotch yoke. This mechanism gives the same motion as the one illustrated at *A*, but the cross-head has two slots at right angles to each other, so that it can be placed on a continuous shaft. The vertical slot is for the sliding crank block, whereas the horizontal slot forms a clearance space for the shaft. With this design, the crank could be placed at any intermediate point on the shaft without using a center crank. It is not as compact, however, as form *A*, and the vertical slot is not continuous, which is an objectionable feature.

Scotch Yoke Modified to Give 60-degree Dwell at Each End of Stroke.—With the design of Scotch yoke shown in Fig. 3, the driven cross-head or slide has a dwell at each end of the stroke equivalent to about 60 degrees of crank rotation. This mechanism is part of a double flanging press used in the manufacture of certain cans. The flanging is done in three operations at the rate of eighty cans per minute, and as a slight amount of time is required for the cans to drop from one working position to another, the dwell obtained with this mechanism allows for this. The drive is from pinion *A* to gear *B*. The eccentric crank *C* has a bearing in gear *B*, and carries at one end (see plan view) a pinion *D* which meshes

with a stationary gear *E*, the ratio of these two gears being 2 to 1.

As gear *B* revolves, the planetary pinion *D* rotates around fixed gear *E*; consequently, crankpin *C* turns about the axis *F* (see Fig. 4) of its bearing in gear *B*, while this axis fol-

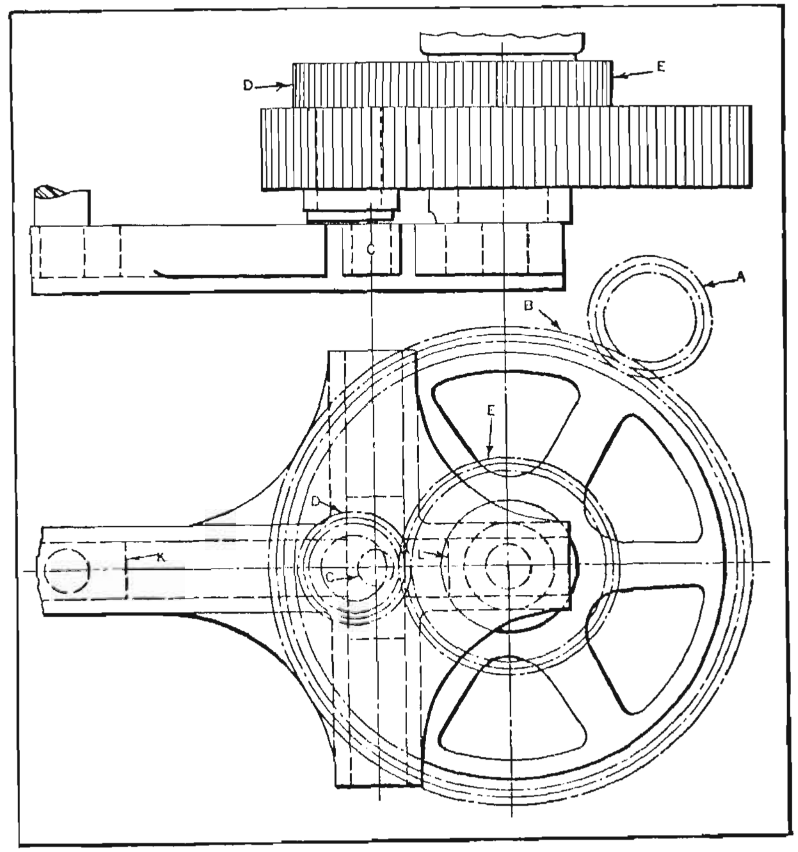


Fig. 3. Scotch Yoke Having an Eccentric Planetary Crankpin which Provides Dwell at Each End of Stroke

lows a circular path *G*. These combined rotary motions of *C* about *F* and of *F* around path *G* cause the axis of eccentric pin *C* to describe an oblong path, as indicated by line *H*. The straight sides of this oblong path represent the dwelling periods at each end of the stroke. This approximately straight-line movement of the center of pin C during nearly 60 degrees of crank rotation on each side is due to the fact that the center of crankpin C moves inward toward center J, so as to offset, during this period, the circular movement. The driven slide or cross-head is supported on pivoted guides K and L, Fig. 3. This mechanism doubtless can be applied to various classes of machinery requiring dwell at the ends of the stroke.

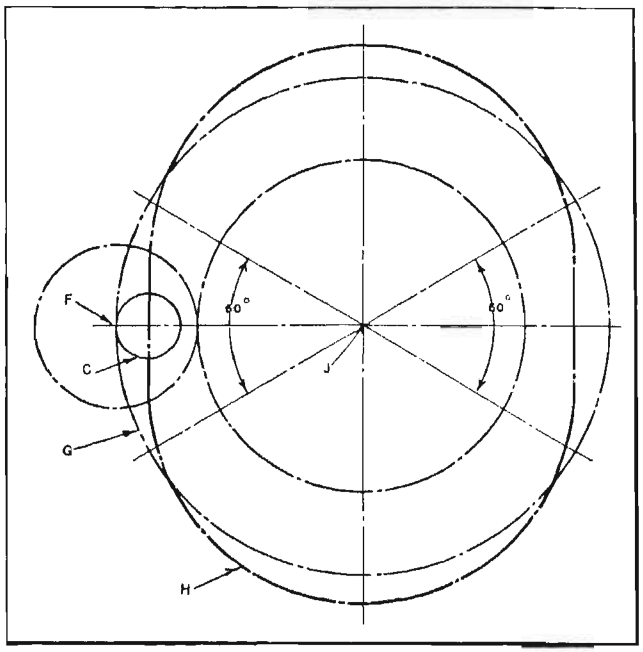


Fig. 4. Diagram Showing How Eccentric Rotation of Crankpin Causes Dwell at Ends of Stroke

Crank Designed to Traverse Slide at Uniform Rate.—An interesting and unique mechanism for controlling the traverse motion of the carriage on a grinding machine is shown in Fig. 5. This mechanism is designed to feed the carriage by means of a crank motion. The accelerated motion given to the crank at the ends of the stroke is the principal feature of the device. This acceleration gives the carriage a uniform rate of travel the full length of the stroke which

prevents under-grinding at each end of the spindles. The illustration shows a plan view and two positions of the mechanism.

In Position 1, the parts are shown as they slide the carriage toward the end of the first traverse, while in Position 2 the return traverse has started. The driving rod, which is not shown, is connected to the connecting arm K, by an adjusting

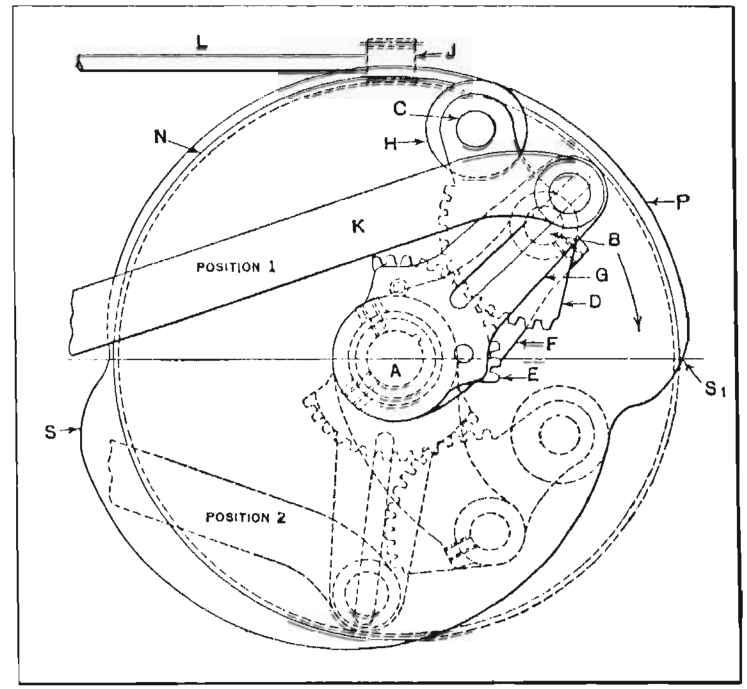


Fig. 5. Mechanism for Obtaining Uniform Rate of Traverse of Grinder Carriage, with Crank Motion

screw and nut. Worm-shaft L, which receives its motion from the main drive of the machine by means of bevel gears, carries worm J, which drives the upright shaft A through the worm-wheel N in the direction shown by the arrow. Driving arm F, is keyed and fastened by set-screws to A, as shown, and carries, at its outer end, a stud which is fastened by set-screws in hole B. Segment D has a forked or clevised end, in which the roll H is carried, revolving freely on stud C. The segment and roll are capable of pivoting on the stud at *B*. Segment *E* is loose on shaft *A*, and carries crank-arm *G* fastened to it by means of a screw and dowel-pin. The segment is prevented from lifting off by a head on the upper end of the shaft.

The grinding starts with roll *H* at point *S*, one full traverse being from *S* to S_1, where the return motion starts, ending automatically at *S*, after the necessary number of traverses has been completed. As the arm *F* swings around on *A*, segment *D*, which meshes with segment *E*, forces roll *H* against cam *P*.

When *H* reaches point S_1, and one traverse has been completed, the heel of cam *P* throws *H* toward the center, forcing the segment *E*, which is free to turn clockwise on *A*, to suddenly revolve about five teeth. This action throws arms *G* and *K* forward quickly, and incidentally the carriage as well, preventing a dwell at the end of the spindle. The resulting relation of the parts is shown in Position 2. Arm *G* has a T-slot, in which the connection *K* may be adjusted, in or out, to obtain the desired throw, that is, the proper length of the traverse.

The feed of the carriage being constant, the same amount of time is consumed on either long or short traverses. Since the speed of *A* is always the same, as the length of the spindles increase, the number of traverses required is also increased, by means of a knock-off device. It is obvious that the depth of the cut on short spindles is greater than that on long spindles, and that fewer traverses are required to grind short spindles to the desired size, and vice versa.

Crank Mechanism for Doubling the Stroke. — A crank and link mechanism is shown in Fig. 6 which makes it possible to obtain a rectilinear motion approximately equal to twice the throw of the driving crank. This mechanism is shown applied to an air pump for use on automobiles, either for the inflation of tires or in connection with engine starting apparatus requiring compressed air. The crank proper is of the center type with a bearing on each side. The connecting-rod

is attached to the yoke *A* which is mounted on the main crank-pin. The opposite end of this yoke is pivoted to link *B* which is suspended from a pin attached to the compressor casing. As the crankshaft rotates, this link oscillates and so controls the position of yoke *A* that the stroke of the piston is approximately doubled. The view to the left shows the piston at the lower end of its stroke. As the crank turns in a counter-clockwise direction, link *B* swings to the right so that the right-hand end of yoke *A* is forced downward and the left-hand

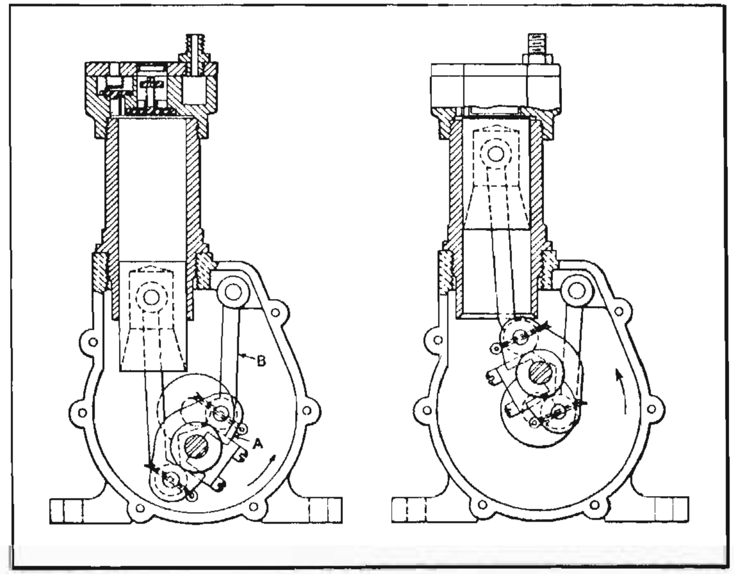


Fig. 6. Crank Mechanism for Doubling the Stroke

end upward, as indicated by the right-hand illustration which shows the piston at the top of its stroke. The advantage of this crank mechanism is that it enables a comparatively large capacity to be obtained from a small compact pump.

Pinion and Rack Mechanism for Doubling Stroke. — Another method of doubling the stroke when a crank of relatively small size is necessary, owing to a limited space, or desirable, in order to obtain a compact design, is by means of a fixed and a movable rack having a crank-driven pinion interposed between them. The pinion is pivoted to the end of the crank connecting-rod so that it is free to roll along the stationary rack when the crank revolves. As the result of this rolling movement of the pinion, the movable rack is given a rectilinear motion equal to twice the stroke of the crank, or twice the diameter of the path described by the crankpin. This mechanism has been used for driving the beds of cylinder presses.

A modification of the plain gear-driven crank is shown in Fig. 7 which illustrates the bed motion of a two-revolution pony press. The driving and driven gears *A* and *B* are of the elliptical form in order to compensate for the motion

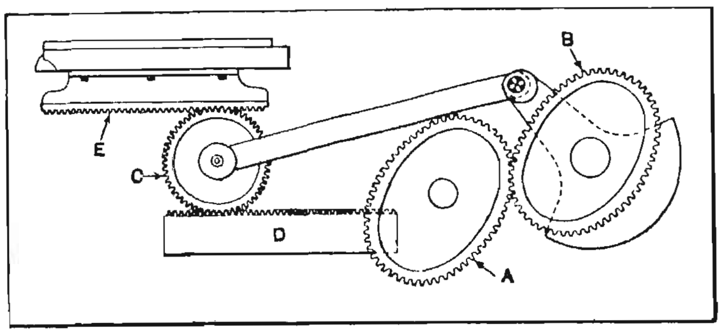


Fig. 7. Crank-driven Pinion Engaging Stationary and Movable Rack for Doubling Stroke

derived from a crank rotating at uniform velocity. The driven gear *B* revolves the crank which, in turn, transmits motion to pinion *C* by means of the connecting-rod shown. This pinion is rolled in first one direction and then the other along the stationary rack *D*, and imparts a rectilinear motion to rack *E* and the press bed. The press bed moves a distance equal to twice the distance that the axis of gear *C* moves, or four times the radius of the driving crank. The elliptical gears are so proportioned and located relative to the crank as to give a more uniform motion to the press bed than could be obtained with a crank rotating at uniform velocity. With an ordinary crank, whatever part is given a rectilinear motion

starts from a state of rest, and the velocity gradually increases toward the center of the stroke and then decreases until it again becomes zero at the opposite end of the stroke. With the elliptical gearing shown, as the pinion *C* approaches either end of its stroke and the crank advances toward the "dead-center" position, the long side or radius of the driving gear comes into engagement with the driven gear and increases its velocity, and also the velocity of the crank. As the return stroke begins, the velocity of the driven gear and crank gradually decreases, because the radius of the working side of the driving gear gradually diminishes; the result is that, when the crank is at right angles to the line along which the axis of pinion *C* moves and is in a position to impart the maximum velocity to pinion *C*, the speed of the crank is slowest, because it is then driven by the shortest radius of the driving gear. As the crank moves away from this central position at right angles to the center-line of motion, the speed is gradually accelerated again so that pinion *C* does not slow down as it would with a crank rotating at uniform speed. The reversal of the heavy press bed is assisted by means of "air springs" or cushions, the same as on cylinder presses in general. This mechanism is intended for small presses.

Compact Long-stroke Mechanism. — The mechanism shown in Fig. 8 was designed to give a long stroke within a small space. A harmonic reciprocating motion is obtained. The device consists of two gears *A* and *B* fitted on eccentrically positioned shafts *C* and *D*. Two concentric surfaces turned on each gear act as eccentrics with respect to the shafts. Two yokes *E* and *F* with two stays secured by bolts *G*, keep the two gears in mesh as shown, the gears being meshed in such a manner that the shafts are nearest each other at one end of the stroke. The drive shaft *C* is mounted in bearings on the machine frame, and the other shaft *D* is attached to the reciprocating part of the machine with its gear free to turn on the shaft or with the shaft.

If desired, the machine can be equipped with ball or roller bearings throughout, and it can be made fairly light by using hollow shafts and gears. The full lines show the mechanism in the position it occupies at the upper end of the stroke, the dotted lines at *H* show the mechanism in the position it occupies when the drive shaft has rotated through an angle of 90 degrees, while the dotted lines at *J* show the mechanism at the end of the down stroke, after the drive shaft has rotated through an angle of 180 degrees. If the driven gear is located 180 degrees from the position shown, rotation of the driving

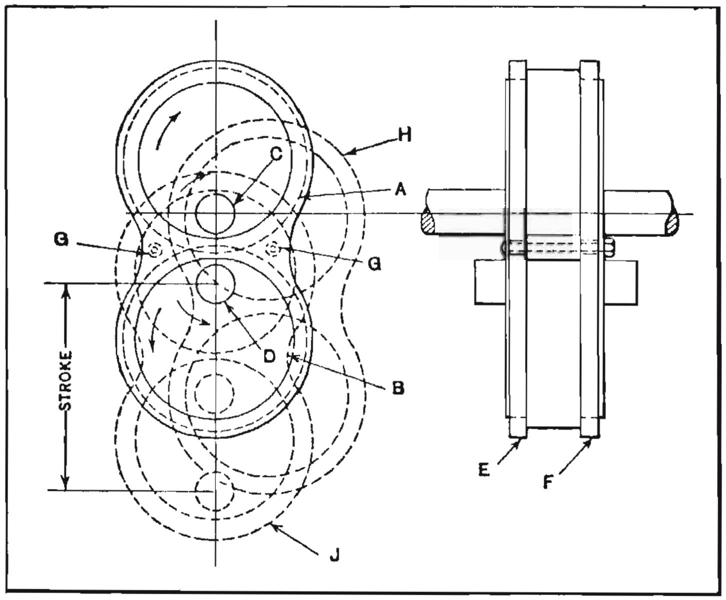


Fig. 8. Long-stroke Mechanism so Designed that Length of Stroke Depends upon Relative Positions of Eccentric Gears

gear will not result in reciprocating motion; therefore, a stroke from zero to the maximum can be obtained by varying the relative positions of the driven and driving gears. For any relative position except that which gives maximum stroke length the stroke will not be a simple harmonic motion.

Long-stroke Mechanism for Windmill Pump. — An ingenious mechanism used in the head of a windmill in order to obtain a long stroke for the pump rod and still allow the

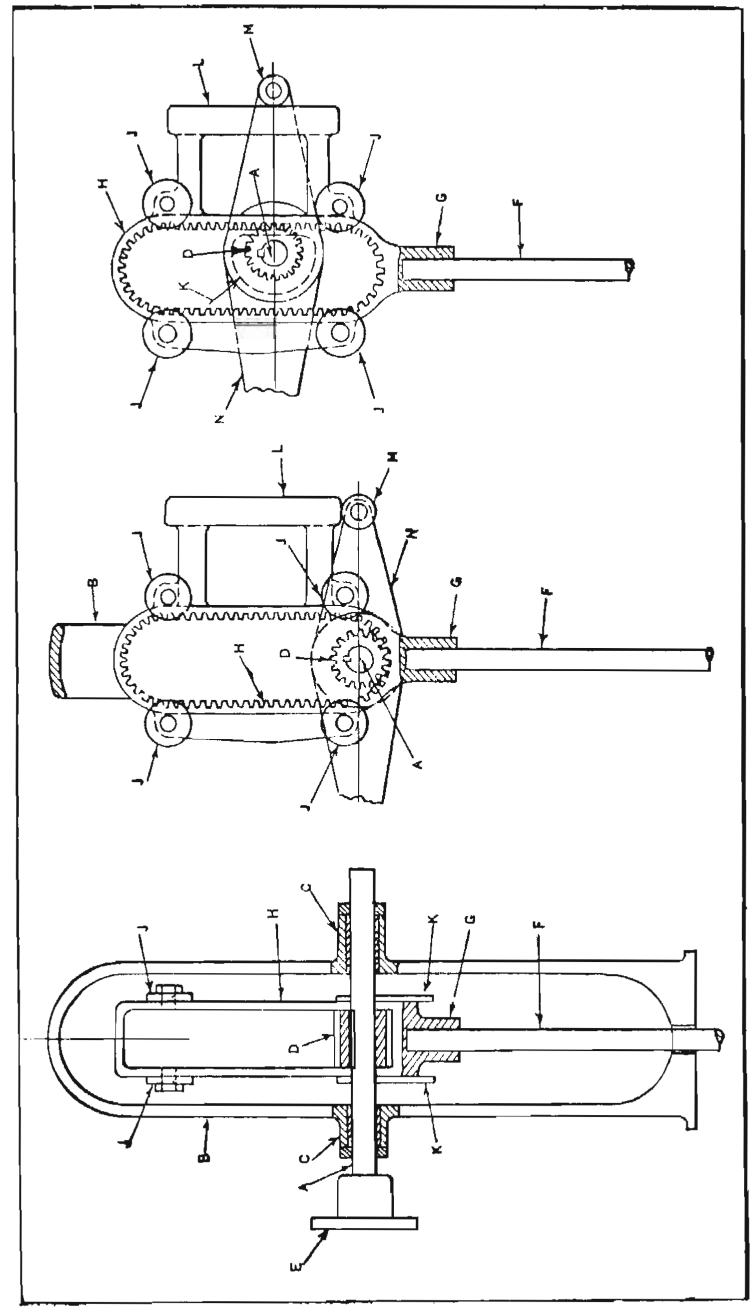


Fig. 9. Windmill Pump Drive for Obtaining Long Stroke and Efficient Windmill Speed

wind wheel to run at an efficient speed is shown in Fig. 9. A vertical section through the gearing is shown at the left, a front view in the center, and a front view at approximately midstroke at the right. In these different views, each part referred to is indicated by the same reference letter.

The main shaft *A* is supported in the main bearings *C* (left-hand view) which are held in housings in the head casting *B* of the mill head. The actual wind wheel is connected to the shaft at *E*, a coupling only being shown. On the main shaft is keyed a pinion *D* which meshes with an internal rack *H*. This rack consists of two vertical sections which the pinion *D* engages alternately, and connecting semicircular sections at top and bottom. The rack is provided with four steel rollers *J*, which engage with the cam *K* at the bottom and top of the pump stroke. These cams throw over the rack at the end of each stroke and thus reverse the direction of travel of the pump rod *F*, which is shown connected to the bottom of the rack at *G*. An extension of the rack at *L* consists of a steel guide plate which is in contact with a flanged roller *M*, and this keeps the pinion *D* in mesh with the rack *H*. This is clearly illustrated in the right-hand view, which shows the pump at approximately midstroke. The roller *M* is shown in the central view, passing across the bottom portion of the guide plate; it will make contact next with the inside face of the guide plate, and thus keep the pinion *D* in mesh with the opposite side of the rack *H*. The swing of the rack is only about 1¾ inches; consequently there is very little angular motion of the pump connecting-rod.

With the gearing shown there is one stroke of the double-acting pump to two and a half revolutions of the wheel. On account of the slow motion of the gear, the wear of the working parts is slight and the mechanism as a unit is efficient.

Reciprocating Motion from Double Rack and Shifting Spur Gear. — If a gear rotating continuously in one direction is between parallel racks, so that it can be engaged with first one rack and then the other, these racks will be moved in opposite directions. For instance, if the top side of the gear moves

one rack to the right, the lower side will move the other rack toward the left. Some flat-bed printing presses are equipped with this double-rack and shifting-gear mechanism for driving the bed in first one direction and then the other. With mechanisms of this class, one rack is first traversed past the gear; when the gear and rack are entirely disengaged, the gear is shifted axially far enough to align it with the other rack. While this shifting movement takes place, the motion of the bed is arrested, and it is reversed by some auxiliary mechanism which moves it far enough to bring the other rack into engagement with the driving gear. Press bed motions of this general type differ principally in regard to the method of moving the press bed at the ends of the stroke, at the time when the driving gear and rack are disengaged.

Napier Motion for Press Beds. — When a gear or pinion is in mesh with a single rack and rotates in one position, obviously both the gear and rack must reverse their direction of motion at the end of each stroke. The gear, however, may rotate continuously in one direction if it is arranged to engage the upper and lower sides of a rack designed especially to permit such engagement. A mechanism of this type, known as the Napier motion and also as "mangle gearing," has been used for imparting a rectilinear motion to the tables of flat-bed printing presses. The principle of the Napier motion will be apparent by referring to Fig. 10. The rack *A* is attached to a frame *B* which is secured to the table of the printing press. The rack teeth are of such a form that the gear *C* may mesh with the rack on either the upper or lower sides. The shaft *D*, upon which the gear *C* is mounted, is rotated through a universal coupling, which permits it to swing in a vertical plane so that the gear may pass from the upper side of the rack to the lower side, and *vice versa*. The gear shaft is made to move in a vertical plane by a stationary slotted guide *E* having a vertical slot that is engaged by a sliding block mounted on the shaft. Spherical-shaped rollers *F* are mounted at each end of the rack, and the gear has a socket or spherical depression formed in it for engaging the rollers, each time the gear moves around the end of the rack when passing from one side to the other. Opposite each end of the rack, there are guide plates *G* having curved surfaces which are concentric with the rollers at the ends of the rack. The gear *C* also carries a roller *H* which engages these curved guides as the gear moves upward or downward at the points of reversal.

The action of the mechanism is as follows: If the gear is on the upper side of the rack, as shown in the illustration, and it is revolving to the left or counter-clockwise, the rack will be driven to the right with a velocity equal to the motion at

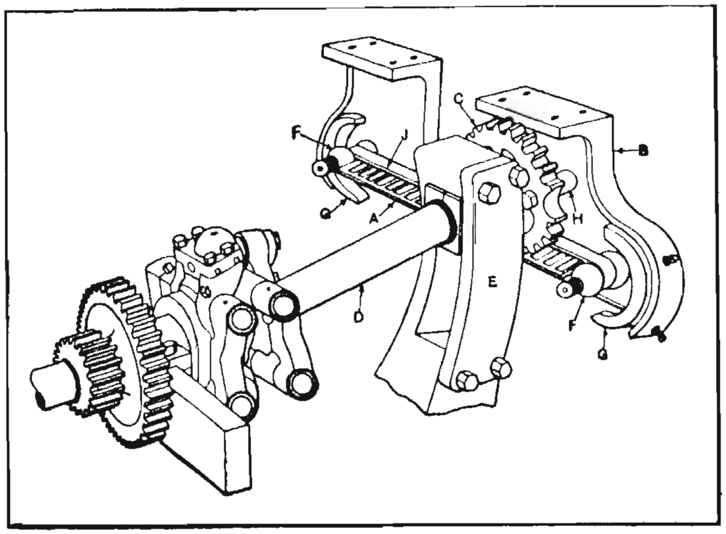


Fig. 10. The Napier Motion for Flat-bed Printing Press

the pitch circle of the gear. As soon as the gear engages the roller *F* on the end of the rack, it begins to move downward in a vertical plane, because its motion is constrained by guide *E*. When the gear is in mid-position so that its axis coincides with the center-line of the rack, it will have made a quarter turn, thus moving the center of roller *F* farther to the right, a distance equal to the radius of the pitch circle. Farther movement of the gear downward causes the rack to reverse and move toward the left; the gear then operates on the under side of the rack until the roller at the right-hand end of

the rack is engaged, when the upward movement of the gear takes place and there is another reversal of motion.

The total length of the stroke is equal to the distance between the centers of the rollers on the rack, plus the pitch diameter of the gear. The length of the rack must equal the pitch circumference of the gear or some multiple of it, so that the rollers at the end will engage the socket or depression in the gear at the points of reversal. If a gear is used having two roller spaces located 180 degrees apart, the length of the rack or the center to center distance between the rollers may be some multiple of half the pitch circumference. The teeth on each side of the rack incline from the horizontal at the same angle as the gear axis when in its upper and lower positions, to obtain a full contact of the gear teeth. The gear also has a plain cylindrical shoulder on the inner side, which rolls upon a plane surface *J* at the base of the rack, to give a smoother action than would be obtained from a gear supported entirely by tooth contact. This arrangement of gearing imparts a uniform motion to the press table, excepting any variable movement resulting from a universal joint, and gives a gradual reversal of motion at the ends of the stroke. The Napier motion may be designed for any length of stroke, although the stroke remains constant, as there is no way of making an adjustment.

Modified Napier Motion for Saw-filing Machine. — An interesting mechanical movement for obtaining the motion required in filing the teeth of handsaws is shown in Fig. 11. As one file is being drawn across the saw, the other file, which has been raised to clear the saw, is returning. When the file in the raised position reaches the end of the return stroke, it is lowered to the saw, and during the filing stroke, the first file mentioned is being returned in the raised position, the files operating alternately.

The mechanical motion for operating the two files is derived from the combination of a special internal gear driven by a pinion having a projecting shaft which engages a slotted cam guide that keeps the pinion and internal gear in the proper

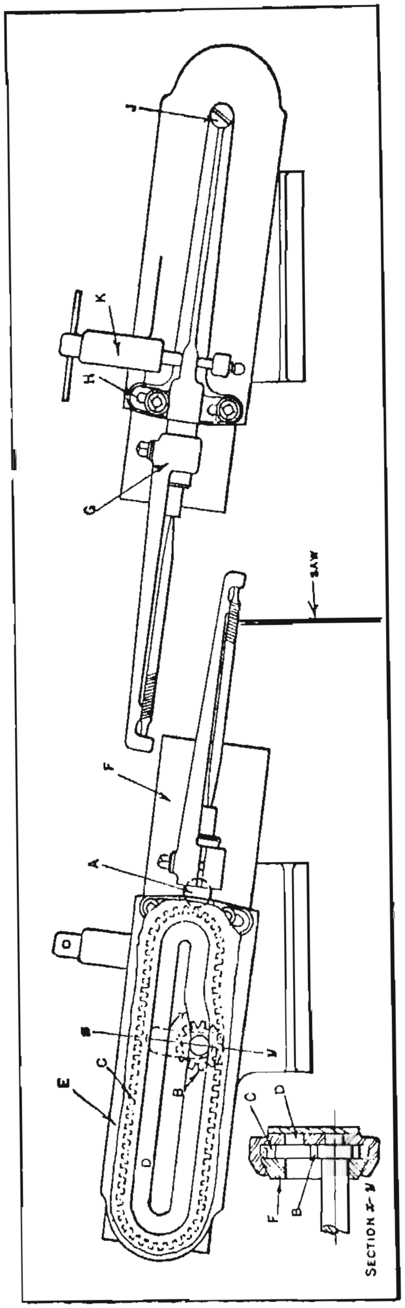


Fig. 11. Mechanism of a Duplex Handsaw-filing Machine, which Raises Each File During the Return Stroke. Left-hand View Shows File Near the End of its Cut and About to Lift Away From the Saw

relative positions. The illustration shows a front view of the two filing heads and a section x-y through the left-hand head. The outside of one filing head is shown at the right of the illustration, and the inside mechanism is shown by the left-hand view of the opposite head.

Each slide is pivoted on a shaft A, and pinion B revolves in mesh with the internal gear C. The pinion is held in contact with the internal gear teeth by an extension of the pinion shaft beyond the face of the pinion gear. The end of this shaft revolves in the slotted guide D which controls the motion. The file-holding head E operates on the slide F during the filing and return strokes. On the filing stroke, the internal gear teeth are being driven from the bottom of the pinion gear, whereas on the

return stroke the internal gear is being driven from the top of the pinion gear. The file-holding arm G is under the pressure of a coil spring in K. The file-holding arm pivots from J, there being slots at H to allow for the necessary swinging movement.

The left-hand view shows the file nearly at the end of its cut and about to leave contact with the saw and travel a short distance to clear the end of the file-holder; then as it is being raised to its highest point, the file of the right-hand head is lowered to start its cut. A plan view would show the files working at opposite angles, to file the proper bevel into the

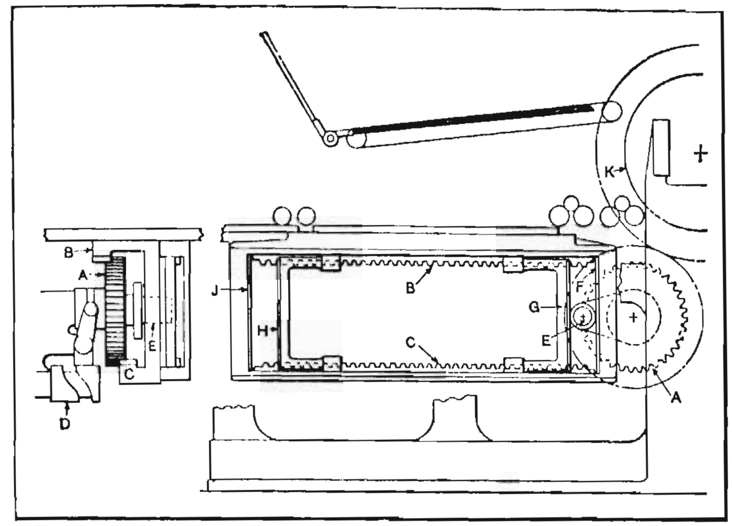


Fig. 12. Double-rack Shifting-gear and Crank Combination for Traversing Bed of a Printing Press

handsaw teeth. An interesting feature is a 5 to 1 gear ratio, but due to a loss of one-half revolution at each end of the stroke, the ratio is 4 to 1 for a complete filing cycle.

Crank Type of Reversal for Press Bed Motion. — An ingenious mechanism of the double-rack and shifting-gear type is shown diagrammatically in Fig. 12. This design is applied to some flat-bed or cylinder presses. In the operation of presses of this general type, the sheets to be printed are carried around by a revolving cylinder K so that contact is made with a flat form on the press bed which moves horizontally beneath the cylinder. This cylinder makes one revolution during the printing stroke and a second revolution while the press bed is being returned. In order to avoid contact between the cylinder and the bed or form during the return stroke, the cylinder is raised slightly by a suitable mechanism. The rotation of the cylinder is continuous in one direction and it is imperative that the cylinder and press bed move exactly in unison. The circumferential velocity of the cylinder should equal the linear velocity of the bed, because any relative motion would cause slurring on the printed sheet and it would be impossible to obtain sharp clean-cut impressions. As the cylinder revolves at a uniform speed, obviously the mechanism for driving the bed must be designed to give a uniform motion while the impression is being made. In order to properly time the motion of the cylinder and bed, the cylinder is connected by gearing and suitable shafts with gear *A*, which transmits motion to the bed; therefore, the press bed motion must be designed to reverse the movement of the bed without reversing the motion of gear *A*, since this gear rotates in unison with the cylinder or continuously in one direction.

This driving gear *A* is mounted between parallel racks *B* and *C*, both of which are attached to and travel with the bed. The distance between the pitch lines of these racks corresponds to the pitch diameter of the driving gear *A*. The racks are not directly in line, but are offset as shown by the end view, so that, when the gear is in mesh with one rack, it will clear the other one. The lateral movement of gear *A* for aligning it alternately with racks *B* and *C* is derived from cam *D*, which transmits motion by means of a lever and yoke engaging the gear hub.

When the press is in operation, the bed is moved in one direction by the engagement of gear *A* with rack *B* and in the opposite direction by meshing gear *A* with rack *C*. If gear *A* is revolving in a clockwise direction while in mesh with rack *C*, the latter and the press bed (the motion of which is constrained by guides) will move toward the left. When the press is in motion, this movement toward the left continues until the rack is entirely out of mesh with gear *A*; just before the disengagement of gear *A* and rack *C*, the crankpin *E*, which is provided with rollers, comes around and enters between the parallel faces of a fixed reversing shoe *F* and a swinging or movable reversing shoe *G*. The fixed shoe is rigidly attached to the press bed and rack frame, whereas the movable shoe is pivoted and free to swivel. This swinging reversing shoe has a pin on its lower side (not shown) which engages a slot or cam that controls its swinging movements. As soon as rack *C* has moved far enough to the left for shoe *G* to clear the crankpin, the cam swings the shoe inward so that crankpin *E* is confined temporarily between the faces of shoes *G* and *F*, which form a vertical guide or slot. As the crankpin passes its lowest position and begins to move upward, the roller on it bears against the face of *G* and "picks up" the load as gear *A* moves out of mesh with rack *C*.

When crankpin *E* arrives at the position shown in the illustration, the motion of the press bed is reversed, because a roller on the crankpin then engages the face of shoe *F* thus moving the driven member toward the right. The motion continues to be derived from the crank independently of the disengaged gear and rack, until the crankpin has passed the top quarter or highest position; then gear *A* enters the upper rack *B* and the motion is transmitted entirely through the gear and rack until the crank again comes into action at the opposite end of the stroke. At this end, the crankpin is again confined between a swinging shoe *H* and a fixed shoe *J*. After rack *B* has moved out of engagement with gear *A*, crankpin *E*, which is now in its highest position, comes into contact with shoe *H* and continues the movement toward the right while making a quarter turn, and then reverses the motion as it swings downward against the face of shoe *J*. While crankpin *E* is controlling the motion and gear *A* is entirely out of mesh, this gear is shifted by cam *D* out of line with the rack *B* which it just left, and into line with rack *C*.

An ingenious feature of this mechanism lies in the provision of two rollers for crankpin E and locating the fixed and swinging shoes in different vertical planes. With this arrangement, each roller is free to revolve in opposite directions as the crankpin moves along the vertical faces of the shoes. The momentum of the bed is gradually checked at the points of reversal, by air cushions or "air springs." A plunger enters a cylinder at each end of the stroke and air is compressed to arrest the movement, and, by expanding, this air assists in accelerating the heavy bed when its motion is reversed. Provision is made for regulating the air cushion or pressure according to the speed of the press. The air cushion is a feature common to flat-bed or cylinder presses in general.

Reversal of Motion by Reciprocating Pinions. — The mechanism illustrated in Fig. 13 is similar, in some respects, to the press bed motion just described, in that the parallel-rack and shifting-gear construction is employed. The method of operating the press bed at the ends of the stroke, however, is entirely different from that shown in Fig. 12, as reciprocating pinions are used to pick up the load and reverse the motion. The uniform motion of the bed is derived from pinion A which is constantly in mesh with gear D carried on the main driving shaft. Pinion A is located between parallel racks B and C which are attached to the press bed. These racks are offset, as in the design shown in Fig. 12, so that the pinion will clear one rack while in engagement with the other one. The shifting of the pinion is controlled by cam E which transmits motion to the pinion by means of lever F. The pinions for reversing the motion of the bed are located at G and H. The shafts upon which these pinions are mounted are connected to a heavy yoke J which has a vertical slot or groove in which a swiveling block attached to the crank K operates. This crank is rotated by the main driving shaft, and transmits to yoke J and pinions G and H a rectilinear motion equal to the throw of the crank. This is a harmonic motion, as yoke J and the sliding crank-block operate on the same principle as the well-known Scotch yoke. The outer ends of yoke J are supported by horizontal guides, and the pinions G and H

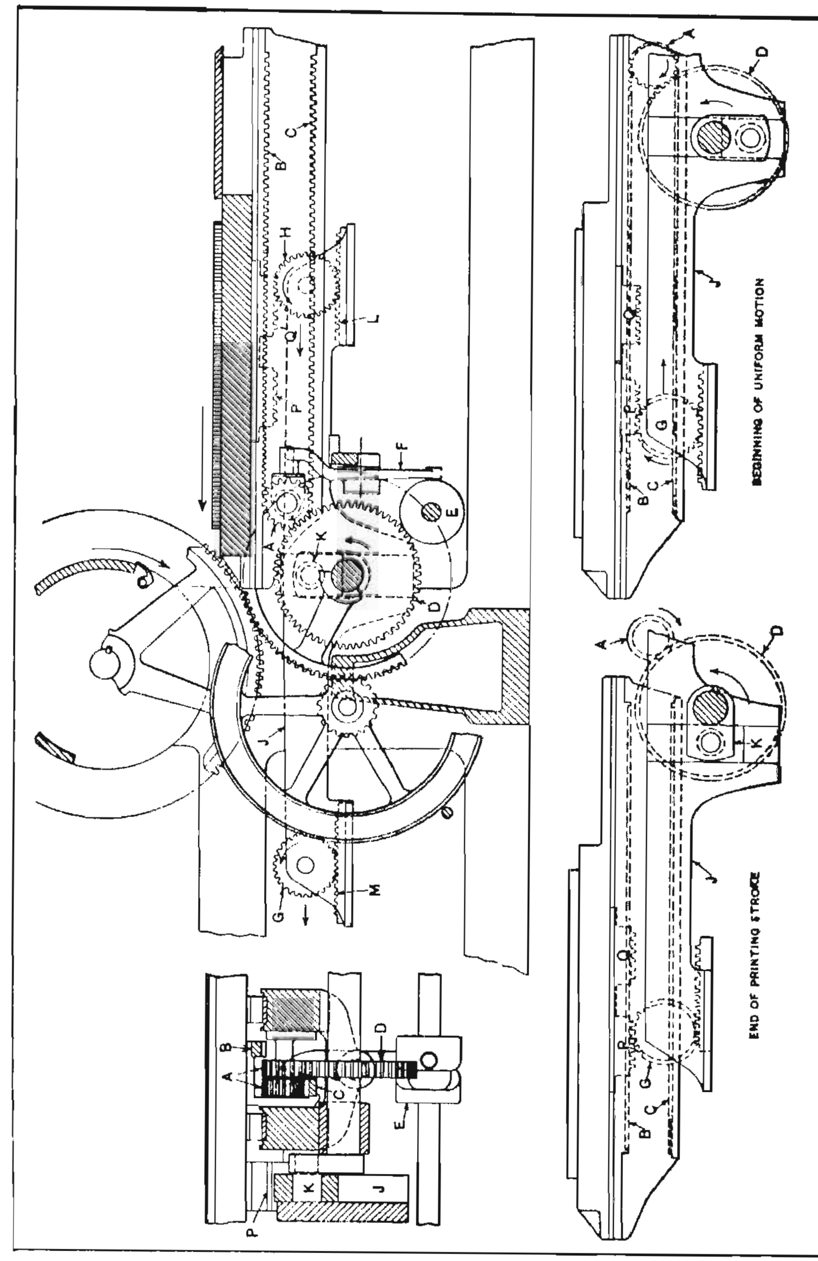


Fig. 13. Double-rack and Shifting-gear Mechanism for Press Bed having Reciprocating Pinions for Controlling Motion at Ends of Stroke

are constantly in mesh with short racks M and L along which the pinions roll as the crank moves them to and fro.

The action of the mechanism will be apparent by considering the various movements which occur during a forward and return stroke. The side view of the assembled mechanism shows the press bed in the position where the driving pinion A has just come into engagement with the lower rack C. As this pinion rotates in a clockwise direction, the bed will be driven to the left with a uniform motion. (The relative positions of pinion A and racks B and C are clearly shown by the end view.) When the bed has moved so far to the left that pinion A is about to roll out of mesh at the right-hand end of rack C, pinion G, which, meanwhile, has been moving along its rack M, comes into engagement with another short rack P (see also end view) attached to the bed. To insure the proper engagement of pinion G with rack P, the action of crank K relative to the motion of the bed is so timed that pinion G is rolling to the left when rack P which is also moving to the left comes into engagement with it. As pinion A leaves rack C, pinion G, which is then in mesh with P, continues the movement of the bed toward the left until crank K is in the position shown by the diagram in the lower left-hand corner of the illustration, which represents the end of the printing stroke. Further rotation of crank K in the direction indicated by the arrow causes a reversal of the rolling motion of pinion G and starts the press bed toward the right, motion being transmitted from G to rack P. While this reversal of movement occurs, pinion A is being shifted by cam E into alignment with the upper rack B.

When crank K has moved a quarter revolution from the position it occupies at the extreme end of the stroke, pinion A comes into mesh with the upper rack B and the short rack P leaves pinion G. The view at the lower right-hand corner of Fig. 13 shows pinion A about to enter rack B and pinion G leaving rack P. As the rectilinear motion of yoke J is harmonic, the movement of the bed is uniformly retarded as it approaches the point of reversal and is then accelerated until pinion A engages its rack, when the motion is uniform. When pinion A enters at the end of either rack, the velocity of the movement derived from crank K and the reciprocating pinion corresponds to the velocity obtained from the driving pinion A, so that there is no abrupt change of motion as the load is being transferred from the reversing pinion to the driving pinion A. As the press bed approaches the opposite end of its stroke, pinion H comes into engagement with rack Q and continues the movement for a short distance each side of the point of reversal or while pinion A is out of mesh with either rack and is being shifted, the action being the same as previously described.

Variable Reciprocating Motion. — The fly frames used in the manufacture of cotton goods are equipped with a mechanism for traversing the rovings or slightly twisted slivers of cotton as they pass between the rolls of the fly frame, which is used to make the rovings more slender and give them a twist. The reason for traversing the roving as it passes between a steel and a leather-covered roll is to prevent wearing the leather covering at one place. On some machines, this reciprocating or traversing motion is obtained from a crank or a cam. This simple arrangement distributes the wear but, if the length of traverse is uniform, the tendency is for the leather covering to wear the most at the points of reversal. In order to distribute the wear more evenly, the mechanism shown in Fig. 14 was designed. With this arrangement, the length of traverse gradually increases until it reaches a maximum and then decreases until the shortest length of traverse is obtained; the gradual increasing and decreasing of the stroke are then repeated.

The diagram A illustrates graphically the action obtained with a crank motion, and diagram B illustrates the variable stroke derived from the mechanism to be described. The guide-bar C, which extends the full length of the rolls, has small holes opposite each roll section through which the rovings pass, and it is this guide-bar which receives the reciprocating motion. The automatic variation of the traversing movement is derived from two eccentrics *D* and *E*, which revolve at different rates of speed. These eccentrics are formed on the hubs of gears *F* and *G*, which are adjacent to each other, and are both driven by one worm *H* as shown by the end view. The motion of the eccentrics is transmitted to guide-bar *C* through rods *J* and *K* and the bracket *L*. One of the gears meshing with worm *H* has one more tooth than the other, which causes the gears to rotate at a varying speed. The result is that the eccentrics formed on the two gear hubs are

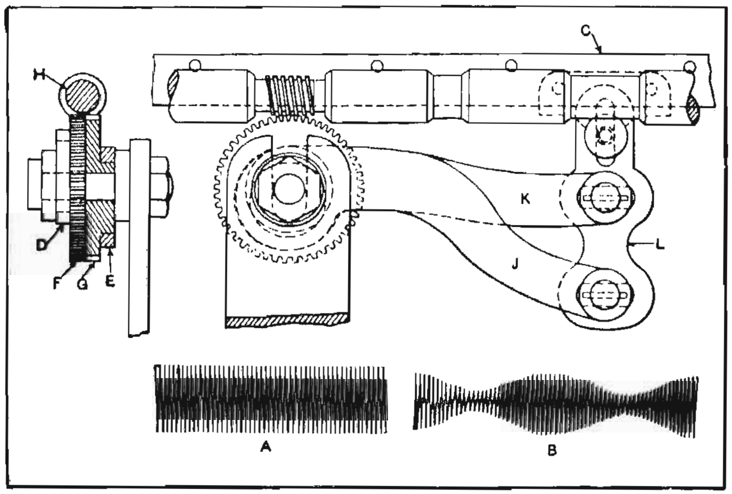


Fig. 14. Double Gear and Shifting Eccentric Combination for Automatically Varying Traversing Movements

continually changing their position relative to each other, which automatically varies the length of traverse for guide-bar *C*. For instance, at one period during the cycle of movements, both eccentrics will move rods *J* and *K* in the same direction, and, at another period, one eccentric rod will be moving backward while the other is moving forward, thus reducing the stroke of the guide-bar. The connections between the eccentric rods and the bracket are adjustable; an adjustment is also provided where the bracket is attached to the guide-bar, so that the maximum traversing movement may be varied to suit the requirements.

Another Design of Variable Reciprocating Motion. — A mechanism used on cotton fabric machinery for varying the traverse of rolls is designed to give a variable reciprocating motion to slide *A* (see Fig. 15), which has a movement varying from zero up to the maximum of 1 inch with a gradual reduction back to zero. The slide is traversed

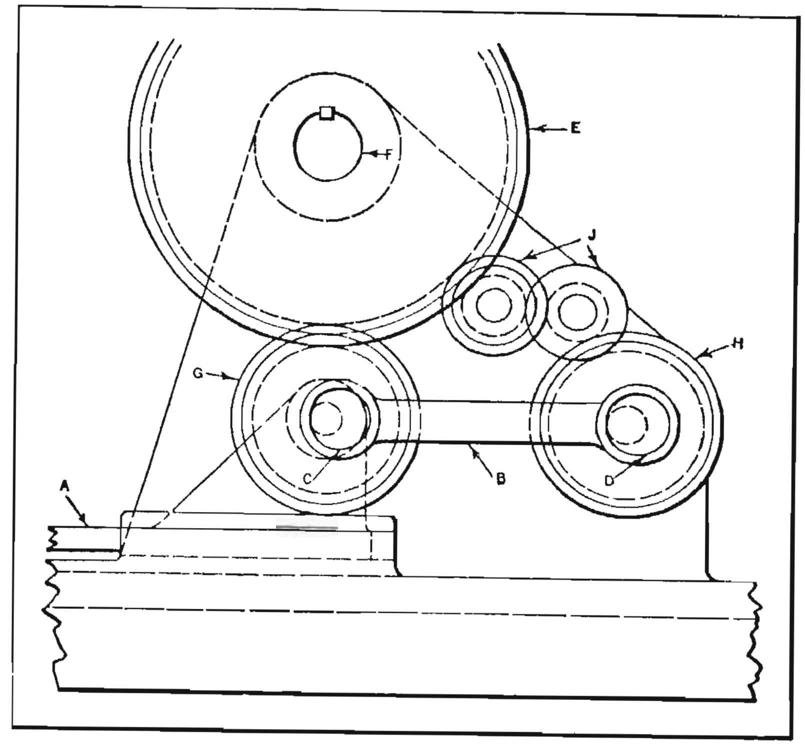


Fig. 15. Mechanism for Alternately Increasing and Decreasing Movement of Slide

through link *B*, and the variation in stroke is obtained by eccentrics *C* and *D*, the relative positions of which are changed, thus lengthening or shortening the effective length of link *B*.

The driving gear *E* is keyed to shaft *F* which receives its motion from the machine proper. Gear *E* meshes with gear *G* directly, and with gear *H* through two idlers *J*. Gears *G* and *H* turn freely on studs. The stud for gear *G* is fastened into the driven slide, and the stud for gear H into a bracket. The eccentrics are integral with their respective gears, and there is a variation in the eccentricity due to a difference of one tooth in the gears G and H. When the movement of one eccentric neutralizes that of the other, the driven slide remains stationary, and this movement gradually increases from zero up to the maximum as the relative positions of the two eccentrics change. This traversing movement causes a change in the center-to-center distance between gears F and G, but since involute teeth are used, this variation does not interfere with the tooth action.

Rapid Reciprocating Motion from Epicyclic Gearing. — What is known as a "wabble" gear is used on mowing machines for imparting a rapid reciprocating motion to the cutter bar. The arrangement of this gearing and the other parts of the mechanism is shown in Fig. 16. The internal gear C is so mounted that it cannot rotate but is free to oscillate on a universal gimbal joint D. The gear B which meshes with one side of C is mounted on the main shaft which connects with the driving wheels. The frame J is rigidly connected to gear C and is pivoted in the revolving part H. By this means, gear C is given an oscillating or wabbling movement, so that the entire gear describes or follows a circular path. This circular motion causes the teeth of gear C to mesh with those of gear B all around the circumference for each rotation of the part H. This part H turns on a fixed shaft E and acts somewhat as a flywheel to maintain steadiness of action besides constraining gear C to follow a circular path.

In this case, gear C has forty-eight teeth and gear B, forty-six teeth; therefore, if gear B were free to turn on its shaft, it would be displaced two teeth for each rotation of part H or each time gear C completed a circular movement. Consequently, twenty-three revolutions of part H and a like number of oscillations of frame J would be required to turn B one revolution. Tracing the motion in the opposite direction, it will be noted that one rotation of gear B, which acts as the driver when the mechanism is in operation, will cause twenty-

three oscillations or wabbling movements of gear C and a like number of rotations for part H. The frame J is connected to the cutter bar by the ball joint at K, so that one turn of the driving wheels which are mounted on shaft A will traverse the cutter bar twenty-three times. This combination of gearing makes it possible to use a gear B having only two teeth less than the number in gear C, which would be practically impossible with gears having teeth parallel to the axis of the shaft. With the usual forms of epicyclic gearing, in which a high velocity ratio is obtained, the efficiency of transmis-

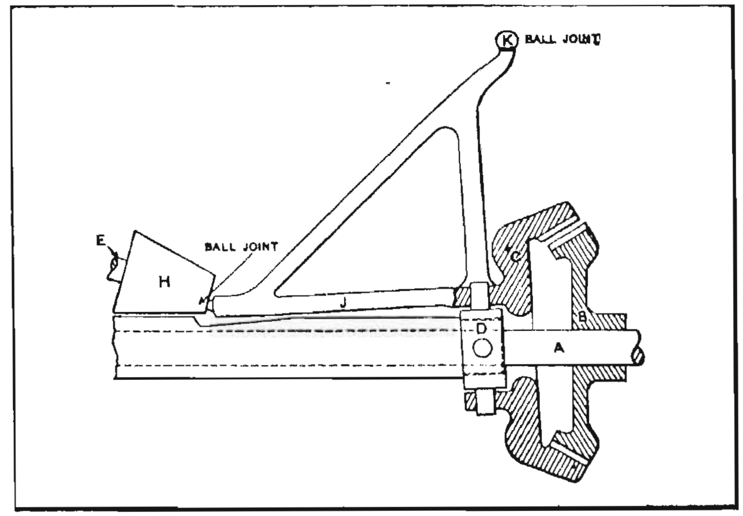


Fig. 16. Epicyclic or "Wabble" Gearing for Producing a Rapid Reciprocating Motion

sion is low on account of the excessive tooth friction, but, in this case, the efficiency is said to be nearly as high as that obtained with a train of spur gears having the same velocity ratio.

Epicyclic Gear and Crank Combination. — The mechanism illustrated in Fig. 17 is applied to an electric coal-puncher. One of the difficulties encountered in designing coal-punchers, excepting the solenoid type, has been in changing the rotation of the motor into a reciprocating motion for the drill. If the blow is directly dependent upon the motor, the latter causes trouble, owing to the vibrations and strains incident to the blows of the pick, and if springs are utilized they are liable to break. Types having separate motors and flexible shaft connections have also been tried in order to avoid some of these difficulties, but complications were introduced which at least partially offset the benefits derived.

The coal-puncher of which the mechanism shown in Fig. 17 forms a part uses both compressed air and electricity.

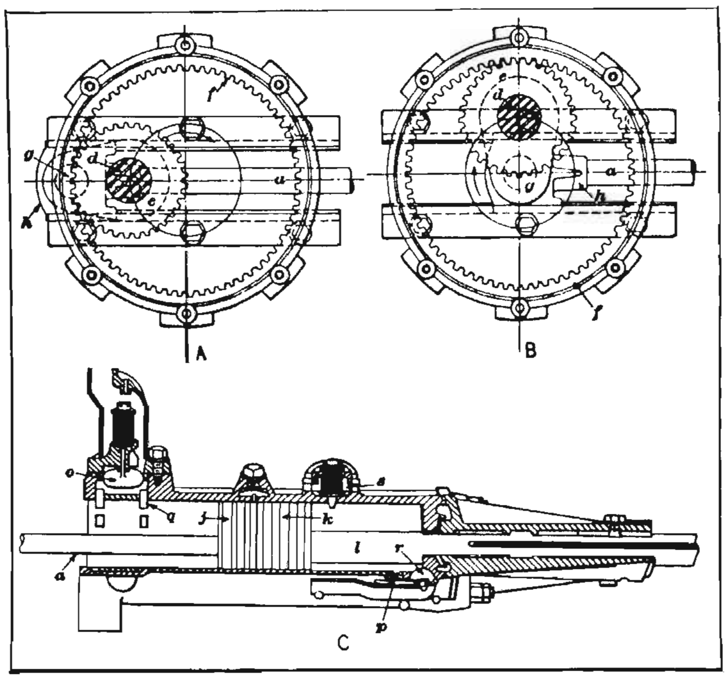


Fig. 17. Epicyclic Gear and Crank Combination from which Reciprocating Motion is Derived

Power for operating the coal-puncher is obtained from a motor and the compressed air gives the blow. There is no direct connection between the motor and striking pick, so that the vibrations are cushioned. The illustration shows the mechanical means by which the rotation of the motor armature is changed to a reciprocating motion for driving the air-compressing piston. A small pinion attached to the armature

shaft engages a large driving gear (not shown) which has a solid web carrying the stud *d* upon which the crank pinion *e* is mounted. This crank pinion has 33 teeth and meshes with internal gear *f* which is rigidly fastened to the frame of the machine and is concentric with the main driving gear which surrounds it. The pitch diameter of the crank pinion *e* is just one-half that of the internal gear *f* which has 66 teeth. The crankpin *g* is attached to the pinion *e* and engages cross-head *h* which is mounted in guides and receives a rectilinear motion as pinion *e* revolves around the internal gear. Attached to the cross-head, there is a piston-rod *a* which enters the air-compressing cylinder and has a piston secured to its forward end.

When the main driving gear is revolved by the motor, the crank pinion stud *d* describes a circular path, as indicated by the arrows, thus causing pinion *e* to revolve about the stud and around the internal gear. When the pin *d* has moved one-quarter of a revolution, it will be in the position shown by the illustration to the right, and pin *g* attached to cross-head *h* will be in the center of the internal gear. At the completion of one-half a revolution, pin *g* will have moved in a straight line a distance equal to the pitch diameter of the internal gear, and will be at the right-hand end of its stroke. Similarly, at three-quarters of a revolution, the pin will again be in mid-position, and at the completion of a full revolution, it will be at the starting point, as shown by the view to the left. In this way, the crank pinion, as it revolves around the internal gear, transmits to pin *g* and the attached cross-head *h* a rectilinear forward and backward movement. The cross-head is mounted in guides, but pin *g* would follow a straight line even though guides were not used.

The way in which the air is compressed and utilized to impell the pick-carrying piston forward, all in one cylinder, will be described. A sectional view of the air cylinder is shown at *C* in Fig. 17. The air cylinder contains two pistons *j* and *k*. The rear piston *j* is attached to rod *a* connecting with the cross-head. The front piston *k* has no connection with *j*, but it is attached to the drill or pick socket by the rod *l*. The first stroke of the pick is purely mechanical. The rear piston *j* moves forward, pushing the front piston *k*. During this stroke, air is drawn into the cylinder behind the piston *j*, through the main inlet valve *o*. On the return stroke, this air is compressed and at the same time the front piston *k* is drawn back by the partial vacuum created by the piston *j*, air being admitted in front of *k* through a port *p*. When the return stroke is completed, the rear piston has passed the by-pass opening *q* in the cylinder, which opening is between the two pistons at the time. This allows the compressed air to force the front piston forward, exactly as in any compressed air drill. In this way, the first real stroke of the machine is made; that is, the mechanical stroke previously mentioned is made only once or when starting from rest. On the forward stroke of the piston *k*, the air in front escapes through the port *p*, but after the piston has passed and, therefore, closed this port, a sufficient amount of air remains to cushion the blow and prevent damage to the front cylinder head. This cushion of air may leak somewhat, and to prevent an insufficient supply remaining, which would have the effect of creating a partial vacuum in this space and holding the piston on the return stroke, a small inlet valve *r* is placed in the forward part of the cylinder. This allows air to flow in under these conditions before the open port is passed. When the front piston *k* has made its forward stroke, the rear piston follows, mechanically driven as before, and would compress the air which has just made the stroke of the front piston, were it not for the so-called vacuum valve *s* which allows all air between the pistons above a certain pressure to escape to the atmosphere. This action prevents the two piston faces from coming together.

A mechanism operating on the same general principle as the one shown in Fig. 17 has been applied to printing presses of the flat-bed type, for imparting a rectilinear motion to the bed. This mechanism has the advantage of giving a long, gradually increasing and decreasing motion with a short crank and without the use of a connecting-rod or a slotted cross-

head; therefore, it can be applied to some classes of mechanisms when there would not be sufficient room for a connecting-rod or in preference to the slotted yoke, because of mechanical objections to the latter. In designing this mechanism, the center of pin *g* should exactly coincide with the pitch circle of the internal gear; then, if the internal gear has twice as many teeth as the revolving gear, the center of *g* will move in a straight line, even though its motion is not constrained by means of guides.

Converting Fast Rotary Speed to Slow Reciprocating Motion. — A novel method of converting a fast rotary speed into a very slow straight-line motion consists in using two

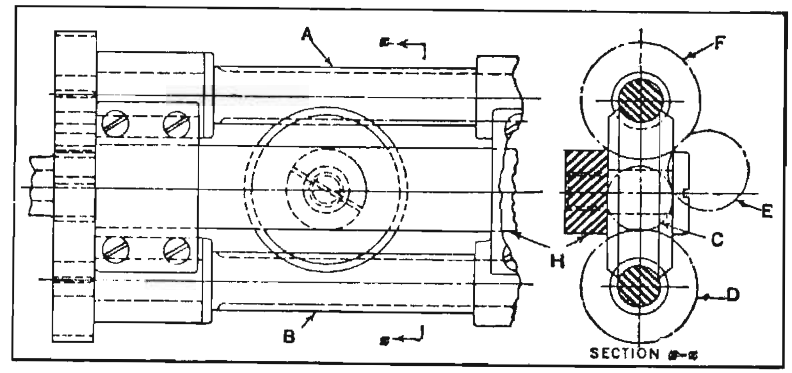
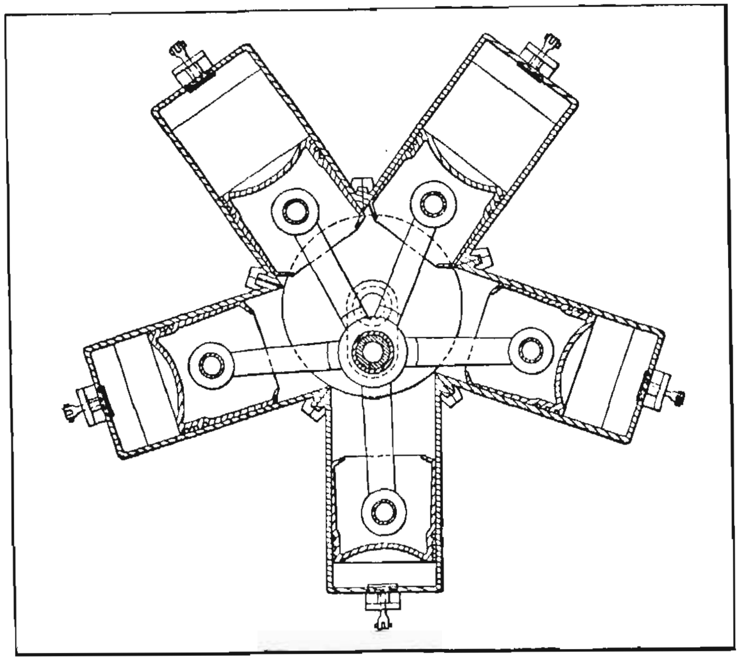


Fig. 18. Mechanism for Reducing a High Rotary Speed to a Slow Straight-line Motion

worms *A* and *B* (see Fig. 18) which differ very slightly in pitch and which mesh with opposite sides of a worm-wheel mounted on the part that is to receive the slow straight-line motion. Worm *B* is driven directly by pinion *C* and gear *D*, and worm *A* is revolved in the opposite direction through the provision of an idler gear *E* between pinion *C* and gear *F*. The worm-wheel turns freely on a shoulder stud which is held in slide *H*.

Worm *B* has six threads per inch, and worm *A*, 5 31/32 threads per inch, all threads being right-hand. Now if the pitch of both worms were exactly the same, no motion would be transmitted to slide *H*, but as there is a difference in pitch equal to 1/1146 inch, the center of the worm-wheel and slide will move one-half this amount or about 0.0004 inch per revolution of the worms, or 0.0002 inch per revolution of the driving pinion *C*, as the latter has 15 teeth, whereas gears *D* and *F* each have 30 teeth. When the slide *H* reaches the end of its stroke, engagement of a dog with a suitable trip



19. **Stationary Crank and Revolving Cylinders**

operates a clutch and the traversing movement of the slide is reversed.

Cylinders which Revolve about a Stationary Crank. — A crank which is connected to a piston or other reciprocating part ordinarily revolves, but a piston may be given a rectilinear motion relative to a cylinder by holding the crank in a fixed position and revolving the cylinder, connecting-rod, and piston about the crank. An example illustrating this method of utilizing the crank is shown by the diagram, Fig. 19, which illustrates the general arrangement of an early

type of airplane motor. With this form of motor, as the cylinders revolve about the stationary crank, the pistons move in and out relative to the cylinders, the same as though the latter were stationary and the crank revolved. The cylinders form the flywheel and drive the propeller and, as they revolve rapidly, the temperature is reduced sufficiently by air cooling and without any auxiliary cooling device.

Combined Reciprocating and Rotary Movements. — The piston of the pump shown in Fig. 20 has, in addition to a rectilinear movement, a rotary motion. This pump was de-

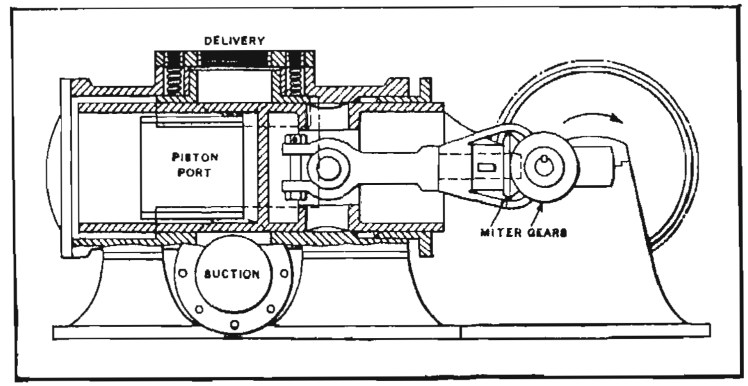


Fig. 20. Piston Having Combined Rectilinear and Rotary Movements

signed for pumping water or other liquids containing foreign materials, such as weeds, pieces of rope, paper, etc., which might enter the pump cylinder. Instead of using suction or discharge valves which would become clogged and cause trouble, the opening and closing of the ports is controlled by the rotary movement of the piston, and any foreign materials of the kinds mentioned are sheared off by the edges of the ports. The rectilinear motion of the piston is obtained from a crank. A miter gear keyed to the end of the crankpin meshes with a mating gear keyed to the end of the connecting-rod, so that, as the piston is moved in and out, it is also given a rotary motion. The piston is of the trunk type with an opening at both ends and a partition in the center. The head end at the left of the partition contains a port which alternately registers with the suction and delivery ports. When the piston is in the position shown, both ports are closed, but, as soon as the pump rotates in the direction indicated by the arrow, the suction port begins to open. When the crank has moved 90 degrees, the piston port will be exactly over the suction

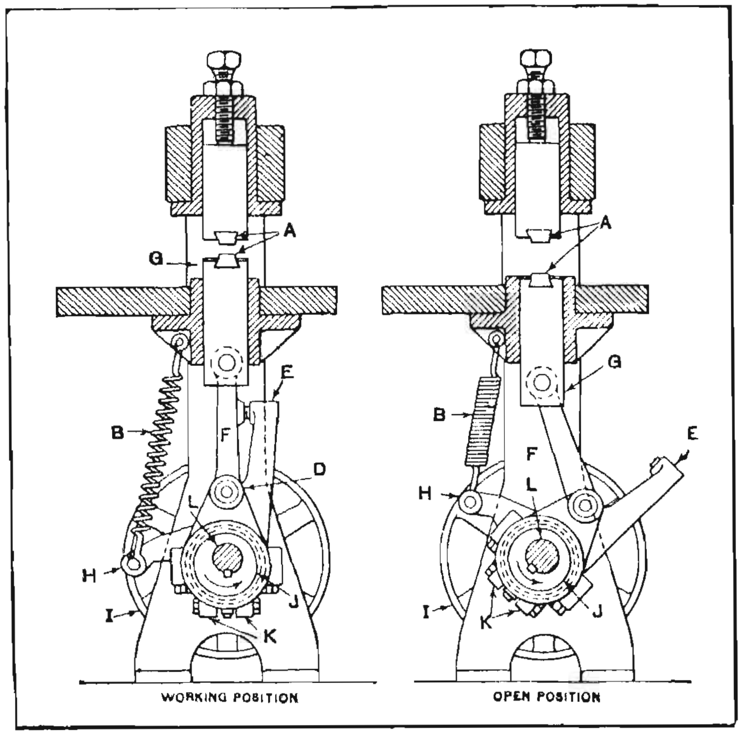


Fig. 21. Mechanism for Shifting Reciprocating Part from Working Position Automatically

port and, when the opposite dead center is reached, both ports will again be closed. When the crank is on the bottom quarter or at the center of the return stroke, the piston port will be opposite the delivery port.

Shifting Reciprocating Part from Working Position. — The machine shown in Fig. 21 is used in a certain branch of the leather business to press a leather product between a pair

of dies *A* by a series of reciprocating motions given to the lower die, which is afterwards withdrawn to the "open position" shown at the right, to allow the removal and insertion of the work. The mechanism to be described serves to automatically locate the lower die in the open position when the driving belt is shifted to the loose pulley, and into the "working" position as eccentric *J*, which imparts motion to ram *G*, is rotated by shifting the belt to the tight pulley. The shaft to which this eccentric is keyed turns in the direction shown by the arrow. The upper half of the eccentric strap is pivoted to the connecting-rod at *D* and carries an arm *H* to which is attached the long spring *B*. If there were nothing to prevent it, the spring would evidently tend to pull the joint *D* over, as shown by the right-hand view. In this position, with the belt on the loose pulley, the machine is ready to receive the work. The lugs *K* are attached to a leather band friction, bearing on an extension of the eccentric surface, and shown in dotted lines behind the eccentric strap. A finger screwed to the lower half of the strap and projecting between the lugs serves to keep the brake in position. If the machine is started by throwing the belt onto the tight pulley, the brake grips the eccentric with sufficient force to overcome the slight tension of spring *B*, and joint *D* is moved back to the central working position, where buffer *E* has reached its seat on the connecting-rod. As the shaft continues to turn, the brake slips on its seat and the eccentric gives the desired movement to the ram. When the operation is completed, the belt is shifted to the loose pulley, and spring *B* turns the shaft, eccentric, and strap backward until the machine is again in the open position with the ram lowered to allow a change of work.

Rectilinear Motion from Revolving Pawls. — The mechanism for driving a conveyor is shown in Fig. 22. This conveyor consists of a pair of endless chains between which the conveyor buckets are carried. These buckets are hung on pivots, so that they are kept in an upright position by gravity. The chains are equipped with wheels which run on tracks. The chain and buckets are propelled along the tracks as indicated by the arrow, by a system of rotating pawls which receive their motion from a large gear *D*. Each pawl, in turn, engages one of a series of pins on the chain and, after having pushed the conveyor ahead, the pawl is raised by cam *C* and the next pawl repeats the operation. When a pawl, as at *A*, is passing through the lowest arc of its travel, the conveyor is propelled forward. The pawl shown at *B* has passed the lowest point, and it gradually lags behind the conveyor, so that the end of the pawl is readily lifted out of engagement

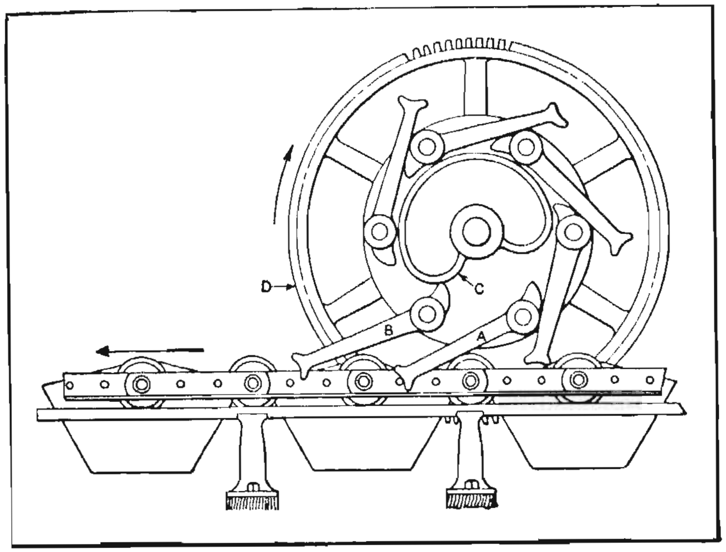


Fig. 22. Arrangement for Obtaining Rectilinear Motion from Revolving Pawls

without interference. As will be seen, the inner end of pawl *B* is in contact with the cam surface which controls its position.

Compact Reciprocating Mechanism of Air Compressor.—A sectional view of the air compressor of an oil burner is shown in Fig. 23. In this equipment, four pistons *A* are reciprocated by a revolving thrust plate *B*, which is located at an angle of 12 degrees. As this plate is revolved by worm-gearing *C*, the pistons will, of course, be forced upward due to the angular location of plate *B*, and the return of each piston is insured by the action of a spring *D* which keeps the

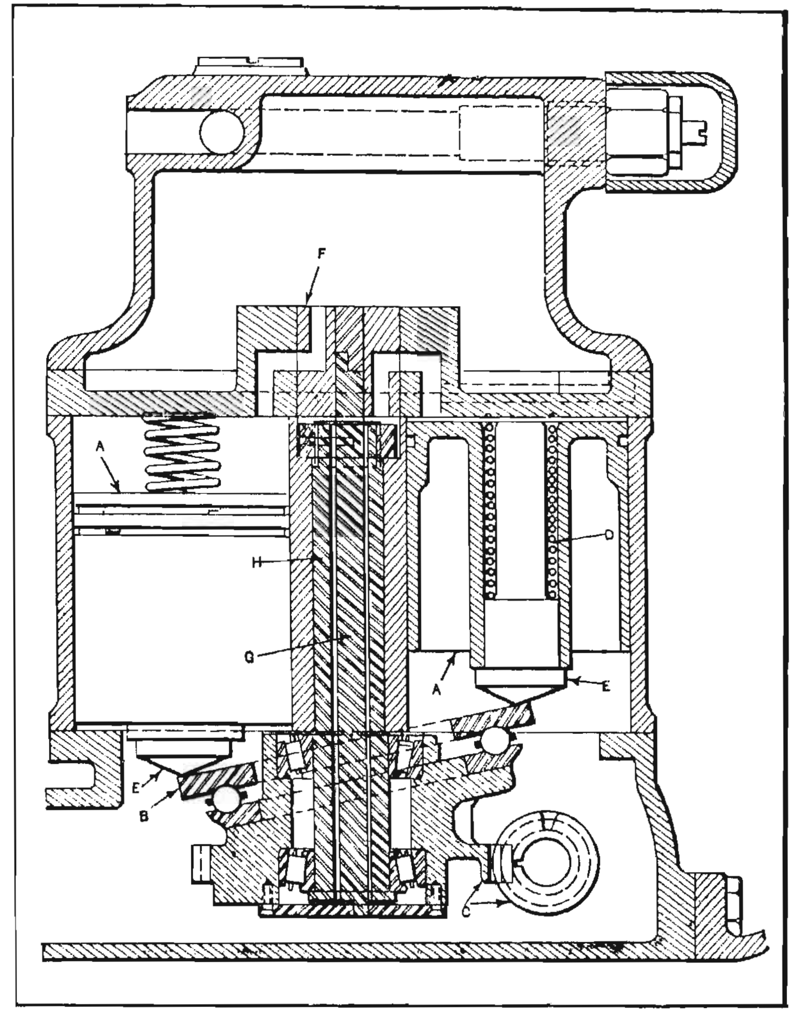


Fig. 23. Reciprocating Mechanism of Air Compressor for an Oil Burner

hardened steel plug *E* in contact with plate *B*. Every revolution of the worm-wheel causes each of the four pistons to make a forward and return stroke. This worm-wheel revolves on Timken roller bearings, and the thrust plate is supported on a ball bearing. The rotary valve *F*, which opens and closes the air ports, is driven from the worm-wheel by a shaft *G*.

In the illustration, the right-hand piston has delivered its quota of air and is ready for the downward stroke, the valve being timed to allow air to be drawn through shaft *H* as the piston descends. The valve, as shown, has connected the compressed air reservoir with the left-hand cylinder, although actually the port does not open until the piston has completed nearly three-fifths of its stroke, thus preventing, as far as possible, a pulsating action of the air due to the small capacity or space above valve *F*. The four cylinders, each of which is 3 inches in diameter, are so located that their center lines form a 3¼-inch square, as seen on a plan view of the compressing mechanism.

In this oil burner, the oil is atomized by the use of compressed air generated by this mechanism, and a surplus of low velocity air for complete combustion of the fuel is furnished by a fan, not shown. The worm-gearing is continually submerged in oil to half its depth, and the splash from the worm lubricates the Timken bearings and the pistons. The oil spray, carried along by the air being compressed, is sufficient to lubricate the rotary valve. A relief valve in the cover above valve *F* allows any surplus air not needed for atomizing the oil to be returned to the intake side so that it can be compressed again.

Drop-hammer Lifting Mechanism. — The drop-hammers used for making drop-forgings are so designed that the hammer head is raised by rolls which run in opposite directions and bear against opposite sides of a board attached to the hammer head. Front and side elevations of a drop-hammer lifting mechanism are shown in Fig. 24. The board *A* passes between the rolls *B* and *C*. One roll rotates in a fixed position and the other one is alternately pressed against the board and then withdrawn from it, when the hammer is in operation. The pressure of the movable roll is applied for raising the hammer head and released for allowing it to drop upon the work. The roll that is withdrawn is usually the front one which has an eccentric bearing so that a slight rotary movement will cause the roll to release the board. As the hammer

drops and approaches the bottom of its stroke, it engages some form of trip or latch which holds the eccentric roll in the outward position so that the roll moves in against the board; the hammer is then immediately elevated preparatory to striking another blow. As the hammer approaches the top of its stroke, the eccentric roll is again automatically withdrawn, thus stopping any further upward movement. The hammer will then fall and repeat the cycle of movements and will continue to run automatically, provided the board clamps at *D* are not allowed to grip the board. The position of these clamps is controlled by a foot-treadle. When this treadle is released,

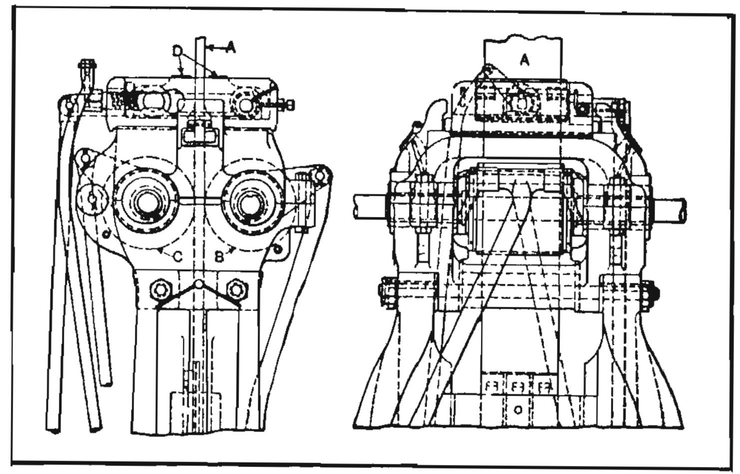


Fig. 24. Board Drop-hammer Lifting Mechanism

the clamps grip the board as it reaches the top of its stroke and starts to move downward, so that the hammering action discontinues until the foot-treadle is again depressed. This mechanism for transmitting the rotary motion of the rolls to board *A*, which has a rectilinear movement, is similar in principle to the rack and pinion, except that motion is transmitted entirely by frictional contact instead of by means of teeth which give a positive drive.

Toggle Joint. — A link mechanism commonly known as a *toggle joint* is applied to machines of different types, such as drawing and embossing presses, stone crushers, etc., for increasing pressure. The principle of the toggle joint is shown by the diagrams A and B, Fig. 25. There are two links, b and c, which are connected at the center. Link b is free to swivel about a fixed pin or bearing at d, and link c is connected to a sliding member e. Rod f joins links b and c at the central connection. When force is applied to rod f in a direction at right angles to center-line xx, along which the driven member e moves, this force is greatly multiplied at e, because a movement at the joint g produces a relatively slight movement at

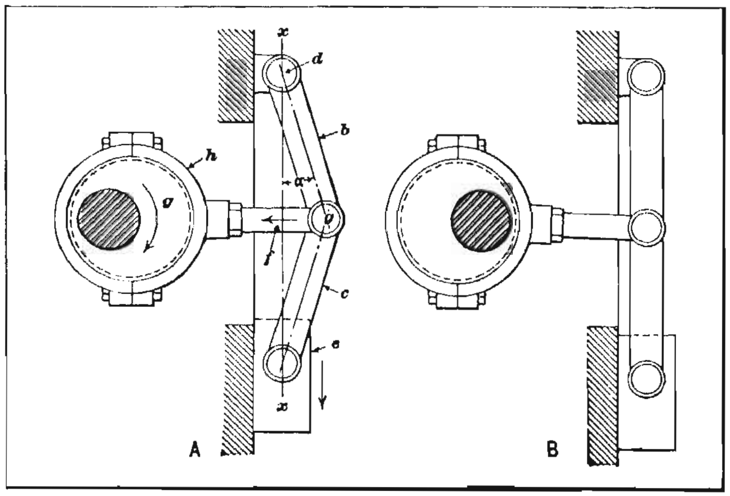


Fig. 25. Diagram Illustrating Action of Toggle Joint

e. As the angle a becomes less, motion at e decreases and the force increases until the links are in line, as at B. If R = the resistance at e; P = the applied power or force; and a = the angle between each link and a line xx passing through the axes of the pins, then:

$$2\,R \sin a = P \cos a.$$

Single- and Double-stroke Toggle Mechanism. — A toggle mechanism is often utilized for changing a rotary to a rectilinear motion, especially when a powerful squeezing action is required. An arrangement of this kind is used on some cold-

heading machines, such as are employed for forming heads on bolts, rivets, etc. The diagram A, Fig. 26, illustrates a crank-driven toggle mechanism which gives a forward and return stroke for each revolution of the crank. When the links of the toggle are straightened, as indicated by the heavy lines, the punch which forms the head on the work is at the end of its stroke, and it is then withdrawn as the crank makes another half revolution. This form of drive as applied to a cold-header is known as the "two-cycle type," because two revolutions of the crankshaft are necessary to complete a rivet or bolt requiring two blows of the punch.

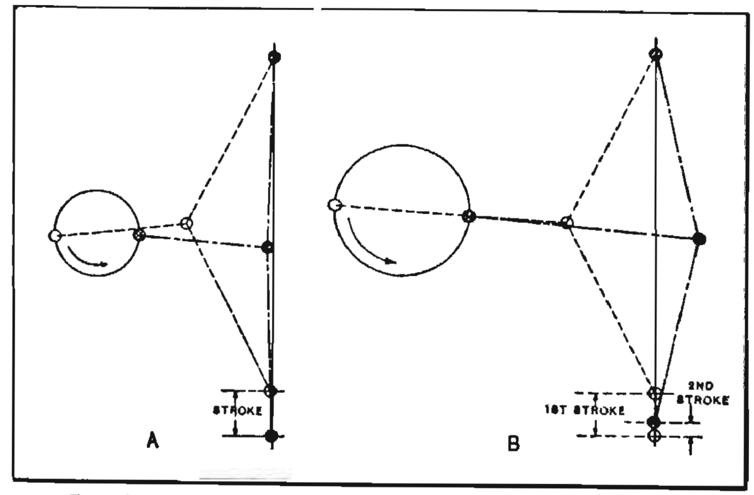


Fig. 26. Action of Single- and Double-stroke Toggle Mechanisms

Many classes of work cannot be done satisfactorily with a single stroke, owing to the amount of metal that must be upset in order to form the head of a bolt or rivet. A design of toggle mechanism which is extensively used on double-stroke machines is illustrated by the diagram B, Fig. 26. With this arrangement, two blows are obtained for each revolution of the crank connecting with the toggle. The location of the crank is such that the links of the toggle are straightened before the crank has made one-half revolution; consequently, when the half revolution is completed, the links of the toggle are carried beyond the center-line, as indicated by the diagram, which causes the ram and die to be withdrawn preparatory to making a second stroke. As the crank continues to revolve and the toggle is again straightened, a second working stroke is made and then the ram and die are withdrawn; this cycle of operations is repeated for each revolution of the crank. The two strokes which are obtained for each revolution of the crank may be of unequal length, as shown by the diagram, or of equal length, depending upon the position of the crank relative to the line of the straightened toggle. A cold-header having this form of drive is known as a "one-cycle" machine, since it will impart two blows to the work for each revolution of the crankshaft.

Toggle Mechanism of Drawing Press. — One way of arranging the toggle mechanism of a drawing press is illustrated in Fig. 27. When a press of this kind is in operation, the sheet of metal to be drawn is pressed firmly down upon the die face by a blank-holder, while the drawing punch forces the metal into or through the die. The blank-holder prevents the sheet stock from buckling, and it should remain in the downward position while the drawing punch is at work. Toggle mechanisms are employed on large drawing presses to operate the blank-holder. The toggle mechanism illustrated is operated from crank *H* on the main crankshaft. This crank connects with link *A*, the lower end of which is attached to yoke *B*. The upper end of yoke *B* is guided by link *E*, which is pivoted to the frame of the press, and the lower end is guided by another link *C* pivoted at *D*. These two links *C* and *D* compel the yoke to move in practically a vertical straight line when it is traversed by the action of crank *H*. Attached to the yoke are two other links connecting with bellcranks *F* and *G* which, in turn, are pivoted to the side of the press frame. The outer arms of these bellcranks are connected by long links or rods with cranks on the ends of two rockshafts *J* and *K*, at the front and rear of the press, respectively. From these rockshafts, motion is transmitted to the blank-holder by means of arms *L* and links *M*. The dotted lines on one

side indicate the action of the rockshaft and its connecting link when in the extreme upper position. The bellcrank levers *F* and *G*, together with the links connecting them with the rock-shafts *J* and *K*, form a toggle mechanism which is straightened out at the same time that the driving crank *H* is passing its center and the arms *L* and links *M* are in line. This central or straight-line position for the toggles occurs while the blank is being held for the drawing operations; the

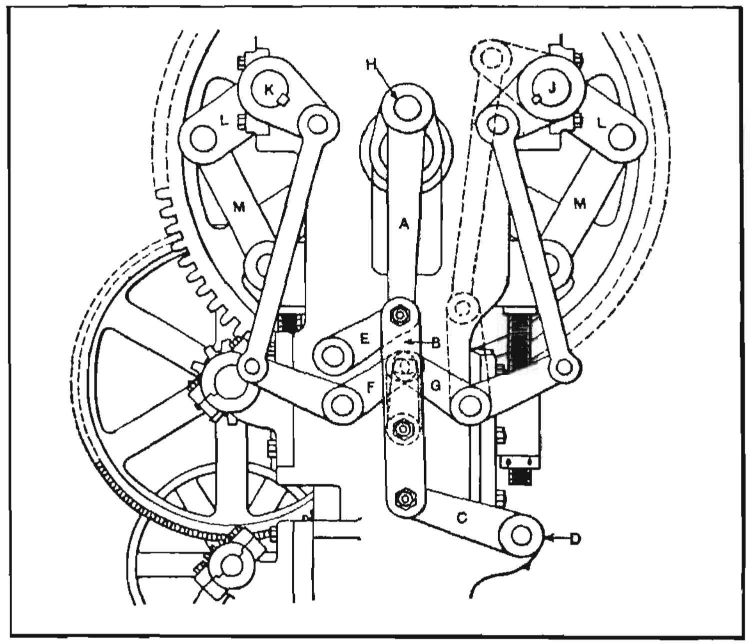


Fig. 27. Application of Toggle Mechanism to a Drawing Press

blank-holder dwells or remains down long enough to enable the drawing punch to complete its work before the sheet metal stock is released by the blank-holder. The slide to which the drawing punch is attached receives its motion from the main crankshaft.

Adjustable Double-motion Eccentrics. — The diagram Fig. 28 represents a disk at *A* encircled by a strap in which the disk is free to run. If this disk is mounted on a concentric shaft perpendicular to it, no motion will be imparted to the strap. However, if the shaft passes through the geometrical center of the disk, but the disk is located at some angle a (see diagram B), then as the shaft makes one complete revolution, the disk and its strap will oscillate harmonically through angle $2a$ in a direction parallel to the shaft.

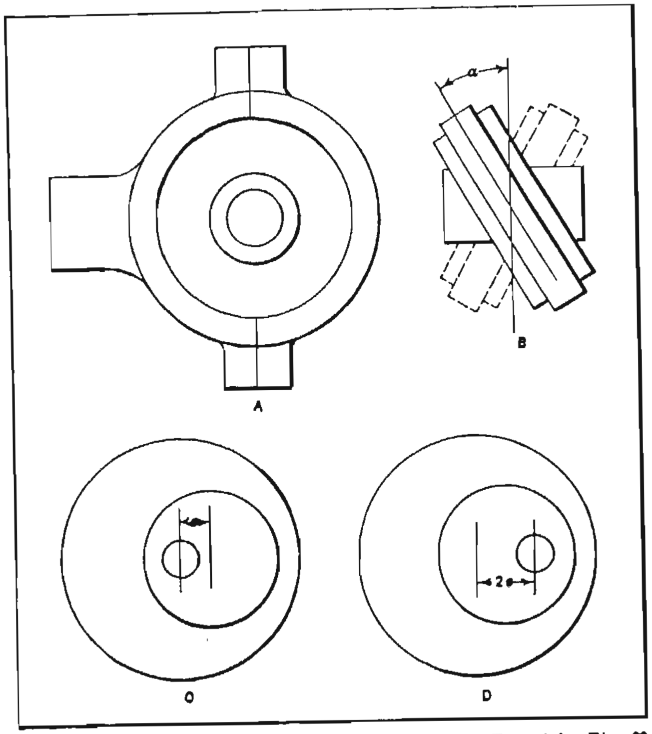


Fig. 28. Diagrams Showing Principles Combined in Eccentric, Fig. 29

Referring now to diagram C, two eccentrics are shown in outline, each having an eccentricity equal to e. The outer eccentric is adjustable relative to the inner one, and it is assumed that they can be fastened in any angular relation to each other. When clamped in the position shown, one eccentric offsets the other, so that the resultant throw is zero. When the positions are changed, as shown by diagram D, the eccentricities are added, and intermediate positions will, of

course, vary the throw from zero to a maximum distance $2e$.

Fig. 29 illustrates how the principles shown separately in Fig. 28 are combined. The inner eccentric E is made spherical, so that the outer eccentric F may be set at an angle to the shaft, as indicated by diagram B, Fig. 28. These inner and outer eccentrics, Fig. 29, have the same eccentricity, so that any throw from zero up to the maximum may be secured by adjustment, as represented by diagrams C and D, Fig. 28. Finally, the mechanism as a whole may be arranged for a circumferential adjustment about the shaft. This mechanism or some modification embodying the same principle may be

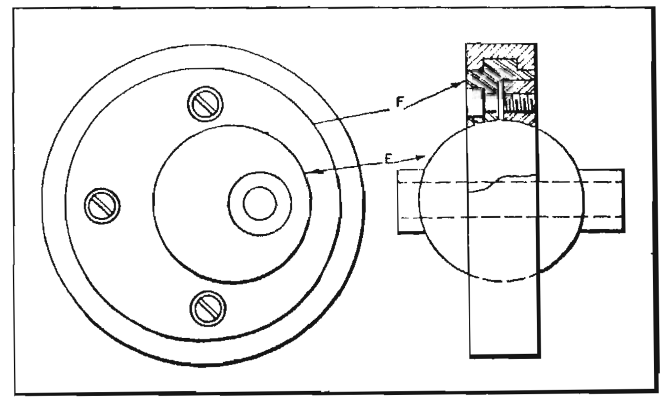


Fig. 29. Eccentric Adjustable as to Throw and Angle Relative to Shaft

used to obtain a harmonic motion in perpendicular planes with adjustment as to amplitude and phase.

Reversing Screw. — When a relatively slow but powerful reciprocating movement is required, a reversing screw may be employed. Many broaching machines of the horizontal type, which operate by pulling long broaches through holes in castings and forgings, are equipped with the reversing screw type of drive. As the broaching is done by a series of cutting teeth which gradually increase in size in order to produce a hole of the required shape progressively, considerable power is required for pulling the broach through the work, especially when cutting hard tough metal. Therefore, the draw-head to which the broach is attached is given a rectilinear movement by means of a screw which does not revolve but is moved in a lengthwise direction by a nut. The screw passes through this nut which is held against endwise movement, and, with one design, is rotated from the driving shaft through suitable gearing. This gearing is so proportioned that a comparatively slow motion is imparted to the nut and screw for the cutting stroke and a faster movement for the return or idle stroke. The nut which engages the screw is alternately connected with these two combinations of gearing by means of a clutch that is shifted by adjustable tappets or dogs that control the length of the stroke. Some of the smaller broaching machines intended for lighter work have belt pulleys that revolve about the screw in opposite directions, and are alternately engaged with a central clutch which transmits motion to the draw nut on the screw.

Screws for Power Transmission. — When screws are used for power transmission, multiple-threaded screws are preferred ordinarily, as they are much more efficient than single-threaded screws, the efficiency being decidedly affected by the helix angle of the thread. Single-threaded screws are sometimes preferable, owing to their mechanical advantage as compared with the multiple-threaded form. Thus a heavier load can be moved by a single-threaded screw than by the multiple-threaded type, assuming, for example, that the screw diameters and the force applied are the same in each case. The single-threaded screw, in this instance, moves the load only one-half as far as a double-threaded screw, so that it is capable of overcoming greater resistance for a given applied force, although the mechanical efficiency is lower than that of the multiple-threaded screw.

Force Required to Turn Screw. — In determining the force that must be applied at the end of a given lever arm in order to turn a screw (or nut surrounding it), there are two conditions to be considered: (1) When rotation is such that the load *resists* the movement of the screw, as in raising a load with a screw jack; (2) when rotation is such that the load

assists the movement of the screw, as in lowering a load, assume that:

F = force applied at end of lever arm;
L = load moved by screw;
R = length of lever arm;
l = lead of screw thread;
r = mean or pitch radius of screw = outside radius minus one-half of thread depth;
μ = coefficient of friction.

When load resists screw movement:

$$F = L \times \frac{l + 2r \times 3.1416\mu}{2r \times 3.1416 - \mu l} \times \frac{r}{R}$$

When load assists screw movement:

$$F = L \times \frac{2r \times 3.1416\mu - l}{2r \times 3.1416 + \mu l} \times \frac{r}{R}$$

If lead l is large in proportion to the diameter so that the helix angle is large, F will have a negative value, which indicates that the screw will turn due to the load alone, unless prevented by a force F which is great enough to prevent rotation of a non-locking screw.

Coefficients of Friction. — According to experiments by Professor Kingsbury made with square-threaded screws, a coefficient of 0.10 is about right for pressure less than 3000 pounds per square inch and velocities above 50 feet per minute, assuming that fair lubrication is maintained. If the pressures vary from 3000 to 10,000 pounds per square inch, a coefficient of 0.15 is recommended for low velocities. The coefficient of friction varies according to lubrication and the materials used for the screw and nut. For pressures of 3000 pounds per square inch, using heavy machinery oil as a lubricant, the coefficients were as follows:

Mild steel screw and cast-iron nut, 0.132; mild steel nut, 0.147; cast-brass nut, 0.127.

For pressures of 10,000 pounds per square inch, using a mild steel screw, the coefficients were, for a cast-iron nut, 0.136; for a mild steel nut, 0.141; for a cast-brass nut, 0.136.

For dry screws, the coefficient may be 0.3 to 0.4 or higher.

Coefficient of Friction for Angular Thread Forms. — Frictional resistance is proportional to the normal pressure, and for a thread of angular form the increase in the coefficient of friction is equivalent practically to $\mu \sec \beta$, in which β equals one-half the included thread angle; hence, for a U. S. standard thread, a coefficient of 1.155μ may be used.

Effect of Helix Angle on Efficiency. — The efficiency between a screw and nut increases quite rapidly for helix angles up to 10 or 15 degrees (measured from a plane perpendicular to the screw axis). The efficiency remains nearly constant for angles between about 25 and 65 degrees, and the angle of maximum efficiency is between 40 and 50 degrees.

In determining the efficiency of a screw and a nut, the helix angle of the thread and the coefficient of friction are the important factors. If E equals the efficiency, A equals the helix angle, measured from a plane perpendicular to the screw axis, and μ equals the coefficient of friction between the screw thread and nut, then the efficiency may be determined by the following formula, which does not take into account any additional frictional losses, such as may occur between a thrust collar and its bearing surfaces:

$$E = \frac{\tan A\ (1 - \mu \tan A)}{\tan A + \mu}$$

This formula would be suitable for a screw having ball-bearing thrust collars. Where collar friction should be taken into account, a fair approximation may be obtained by changing the denominator of the foregoing formula to $\tan A + 2\mu$. Otherwise the formula remains the same.

The square form of thread has been used chiefly for power transmission, because it has a somewhat higher efficiency than threads with sloping sides. However, when the inclination is comparatively small, as in the case of an Acme thread, the preceding formula, which was deduced for a square thread, may be applied without serious errors. The Acme thread

has practical advantages in regard to cutting and also in compensating for wear between the screw and the nut.

It is evident that the screw of a jack, or of any lifting or hoisting appliance, will not be self-locking if the efficiency exceeds 50 per cent, as higher values would mean that the screw would turn in its nut under the action of the load. In actual designing practice, it will be understood that the maximum efficiency, even for a screw intended solely for power transmission, may not be practicable, as, for example, when it is necessary to employ a helix angle smaller than that representing maximum efficiency, in order to permit moving a given load by the application of a smaller turning moment.

CHAPTER X

QUICK-RETURN MOTIONS FOR TOOL SLIDES

MANY machines, especially of the type used for cutting metals, are equipped with a driving mechanism which gives a rapid return movement after a working or cutting stroke, in order to reduce the idle period. For instance, shapers and slotters are so arranged that the tool, after making the cutting stroke, is returned at a greater velocity, thus increasing the efficiency and productive capacity of the machine. The method of obtaining this rapid return varies with different types of machine tools. In some cases, motion for the return movement is obtained by using two belts which alternately come into the driving position and rotate the driven member at two rates of speed. This method is employed with belt-driven planers, the belt for the return movement of the table connecting with pulleys having a higher speed ratio. The rapid return movement for some other types of machines is obtained by transmitting motion through a different combination of gearing which is automatically engaged at the end of the working stroke. The term "quick-return motion," however, as applied to machine tools, generally relates to a driving mechanism so designed that the increased rate of speed for the return movement is obtained through the same combination of parts which actuate the driven member during the forward or working stroke, and the quick-return feature is due to the arrangement of the mechanism itself.

Crank and Oscillating Link. — A simple form of quick-return mechanism which has been applied extensively to shapers is shown diagrammatically in Fig. 1. The pinion A drives gear C at a uniform speed, and this gear carries a swiveling block B which engages slotted link L. The lower end of this

link is pivoted at D and the upper end connects by means of a link with the ram of the shaper. As the crankpin or swiveling block B revolves with gear C, it slides up and down in the slot of link L and causes the latter to oscillate about the fixed pivot D at its lower end. The ram of the shaper is mounted in guides or ways so that it is given a rectilinear movement.

A quick-return movement is obtained with this form of drive owing to the fact that the crankpin B moves through an arc a during the cutting stroke, whereas, for the return stroke, it moves through a much shorter arc b. As gear C

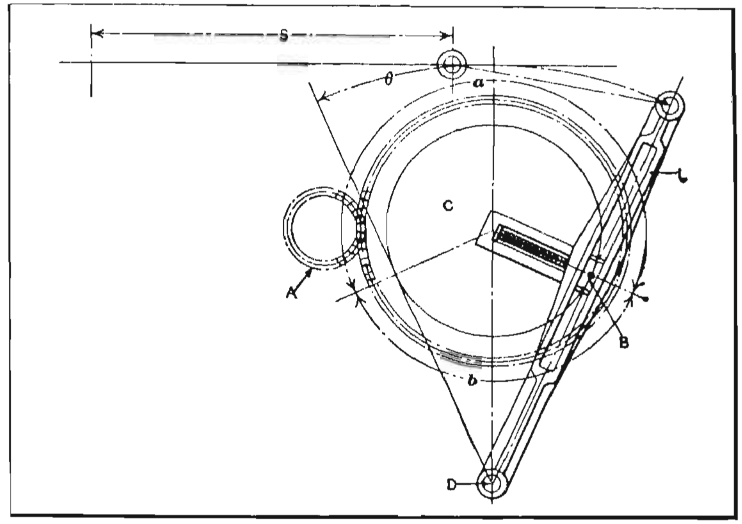


Fig. 1. Quick-return Motion from a Revolving Crank and Oscillating Slotted Lever

rotates at a uniform speed, obviously the time required for the return stroke, as compared with the cutting stroke, is in the same proportion as the lengths of the arcs a and b. The radial position of block B may be varied in order to change the length S of the stroke. This mechanism imparts a variable speed to the ram, the speed increasing toward the center of the stroke and then diminishing. The angle made by the crankpin for the forward stroke equals 180 degrees + the angle θ through which slotted link L moves; for the return stroke, the crankpin moves through an angle equal to 180 degrees — the angular movement θ of the slotted link. The sine of one-half angle θ equals the radius of the crank divided by the distance from pivot D to the center of the gear C.

Whitworth Quick-return Motion. — A type of quick-return motion that has been widely used in slotter construction is illustrated in Fig. 2. This mechanism, which is known as the "Whitworth quick-return," is similar in principle to the crank and oscillating link combination previously referred to, although the construction is entirely different. The pinion A drives gear C at a uniform velocity, and this gear carries a block B which engages a slot or groove in part D, which is

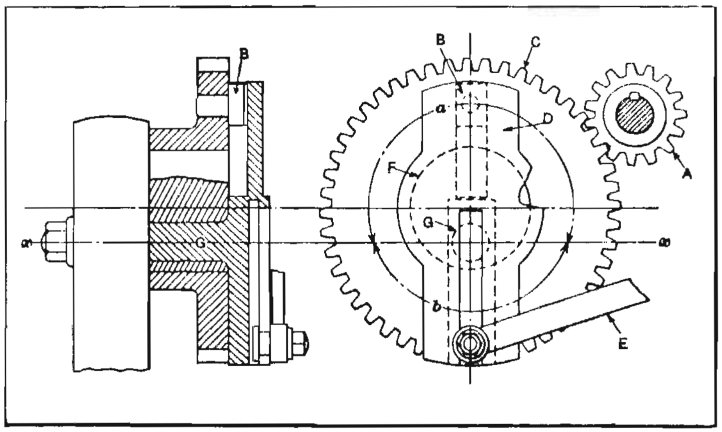


Fig. 2. Whitworth Quick-return Motion

connected by a link E with the tool-slide of the machine. The line xx represents the center-line of motion for the tool-slide. The gear C revolves upon a large bearing F which is a part of the machine frame. The slotted member D has a bearing G, within F, and the center about which D rotates is offset with relation to the center of driving gear C; consequently, the crankpin or block B moves through an arc a during the cutting stroke and through a shorter arc b for the return stroke, so that the latter requires less time in proportion to the respective lengths of arcs a and b. The stroke is varied by changing the radial position of the pin which connects with link E.

So far as the principle of operation is concerned, the chief difference between the Whitworth motion and the crank and slotted link is that, in the former case, the bearing for the slotted or driven member is inside of the crankpin circle, whereas, with the crank and slotted link combination, the pivot is outside of the crankpin circle. As the result of this difference in arrangement, part D in Fig. 2 has a continuous rotary motion, whereas the slotted link L, in Fig. 1, swings through a definite angle. With the Whitworth quick-return, the ratio of the time required for the forward and return strokes is not varied by changing the length of the stroke. With the crank and oscillating link, a change of stroke does affect this ratio, the latter increasing as the length of the stroke is increased.

Modification of Whitworth Motion. — A quick-return mechanism that is a modification of the Whitworth motion combined with the slotted link and rotating crank is illustrated by the sectional view, Fig. 3. This form of drive has been applied to a shaper in order to secure in addition to a quick return a cutting speed that is practically constant throughout the working stroke. The driving gear F transmits its rotary movement through a swiveling block A to a ring E which turns about an eccentric C. On the opposite side of this ring there is a second swiveling block B, which drives the crank-disk G, on which is mounted the main crankpin block H, engaging the vibrating arm or link L that, in turn, is connected with the ram. The eccentric C is offset with relation to the center of the driving gear F, and it remains permanently in a fixed position; therefore, the circular path of the eccentric ring blocks A and B is not concentric with the path described by the main crankpin H. In other words, the circle which these blocks describe as they are driven around by gear F has a constantly varying radius from the center of the gear, which compensates for the irregularity of speed obtained by a plain slotted link, and gives a practically constant movement during the working stroke.

Quick Return from Elliptical Gearing. — Elliptical gearing has been used to obtain a quick-return motion, although such gearing is difficult to cut without special attachments, and comparatively few mechanisms requiring a quick-return motion have this type of drive. The driving and driven gears are of the same proportions and size as shown in Fig. 4, and each gear revolves about one of its foci as a fixed center. The distance between the shaft centers is made equal to the length

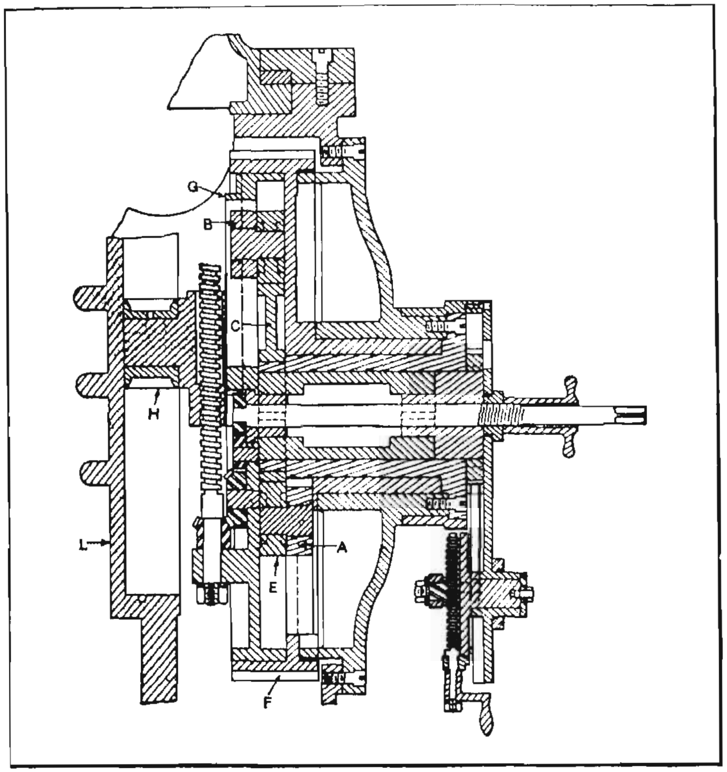


Fig. 3. Crank-operated Quick-return Motion Designed to Give a Uniform Forward Speed

of the common major axis. The angular velocity ratio varies according to the respective radii of the driving and driven gears at the point of contact. If A is the driving shaft and it rotates at a uniform speed, the angular velocity of shaft B will increase during the first half revolution from the position shown in the illustration, and then decrease during the remain-

ing half revolution. When the gears are in the position shown, the angular velocity of the driven shaft B is minimum, because that side of the driver having the shortest radius is in contact with it; as the driver revolves, the radius at the point of contact gradually increases, and, consequently, the angular velocity increases until tooth C is in mesh, when the angular velocity is maximum. When point C representing the longest radius of the driving gear has passed the point of contact, the angular velocity gradually diminishes until it is again at a minimum.

The actual number of revolutions made by each shaft in a given time is, of course, the same, and the driving and

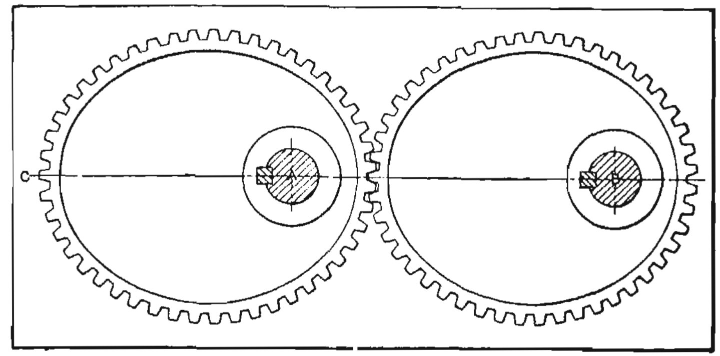


Fig. 4. Elliptical Gearing Arranged to Return Driven Part Quickly

driven gears both require the same time to complete the half revolution between the two positions representing the minimum and maximum angular velocities. The variable motion of the driven gear, however, may be utilized to give a quick-return movement to a driven tool-slide or other part.

This type of quick-return motion has been applied to shapers in order to return the tool quickly after the cutting stroke. The driven gear is connected to the ram by a link. The bolt or crankpin on the gear which connects with the link may be adjusted along a groove for varying the distance from the center of the driven shaft and the length of the stroke. Elliptical gearing has also been used for operating the slide valve of a steam stamp, such as is used for crushing rock. In this case, the variable motion obtained from the gearing is utilized to so control the motion of the valve as to admit steam above the piston throughout almost the entire downward stroke, whereas, on the upward stroke, just enough steam is used to return the stamp shaft, in order to reduce steam consumption.

Eccentric Pinion and Elliptical Gearing for Quick-return Motion. — The eccentric and elliptical gear combination, in conjunction with gears mounted concentrically, as shown in Fig. 5, has been utilized to secure a quick-return motion. The pinions A and B are keyed to the driving shaft. The smaller pinion A is concentric with the shaft and meshes with a half spur gear F. The larger pinion B is eccentrically mounted on the shaft and is in line with a half elliptical gear H, the two gear segments on the driven shaft being offset as shown by the end and plan views.

In the operation of this gearing, the semi-circular gear F is driven by the small pinion A and the elliptical gear by the eccentric pinion B. The elliptical gear makes one-half revolution to each complete revolution of its eccentric driving pinion. If the driven shaft is revolving in a counter-clockwise direction, the eccentric pinion will be the driver from C to D. At the latter point, the elliptical gear segment leaves the eccentric pinion and the smaller pinion A comes into mesh with the half spur gear and continues to be the driver through the remaining half revolution of the driven shaft, or until the elliptical gear again comes around into mesh with the eccentric pinion. Owing to the difference in the diameters of the half spur gear and its pinion A, the latter must make two revolutions before the eccentric pinion can again engage the teeth of the elliptical gear.

At the point C where the eccentric pinion again becomes the driver, the radius of pinions A and B is equal, and the transfer of the load from A to B does not cause an abrupt change of speed for the driven member. As the eccentric pinion, however, begins to swing the elliptical gear around, the speed of the driven shaft is increased until the maximum

radius of the eccentric pinion is opposite the minor axis of the elliptical gear. The speed is then at maximum and, as the movement continues, the speed gradually decreases until the load is transferred to the concentric pinion A which imparts a uniform velocity to the driven member.

With the eccentric-elliptical combination of gearing just described, one revolution of the driven shaft is obtained for every three revolutions of the pinion driving shaft, two revolutions of the concentric pinion A being required for a half

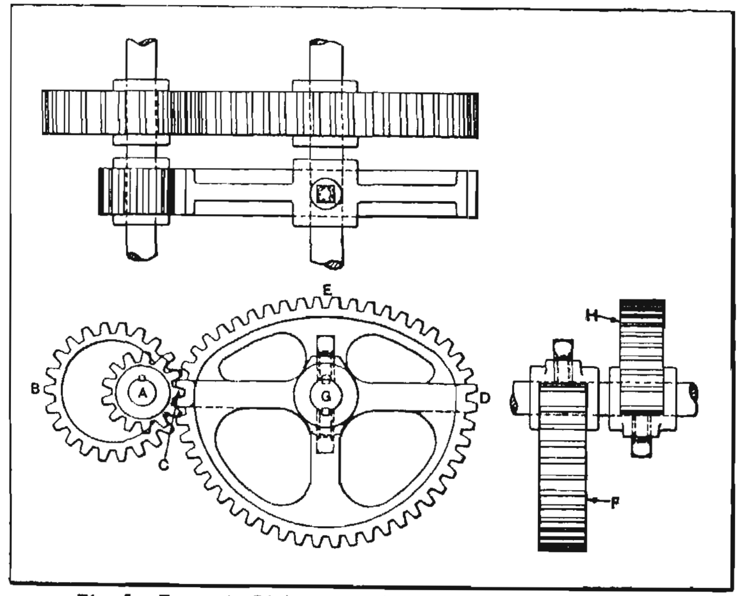


Fig. 5. Eccentric Pinion and Elliptical Gear for Accelerating Return Movement

revolution, and one revolution of the eccentric pinion B for the remaining half revolution. If this mechanism is applied to a slotter or other machine requiring a similar movement, the cutting stroke will occur while pinion A is the driver, because a relatively slow and uniform speed is imparted to the driven shaft. As the eccentric pinion starts the drive, the speed of the driven shaft is gradually accelerated and, after reaching the maximum, is reduced to the cutting speed, so that the tool-slide is rapidly returned to the starting position ready for the next cutting stroke. The ratio of the quick return to the cutting speed should not be too great, because a jerky motion and excessive vibrations in the machine will result. It has been found, by experiment, that a ratio of 2 to 1 is about the highest that will give satisfactory operation.

When laying out gearing of this kind, there are a few fundamental points which must be observed in all cases: 1. The long radius AB of the eccentric pinion from the shaft center to the pitch line should equal one-half the distance between the centers of the driving and driven shafts. 2. The short radius AC of the eccentric pinion should equal one-half the diameter of the concentric pinion. 3. The major axis CD of the elliptical gear should equal twice the distance between the shaft centers, minus twice the short radius AC of the eccentric pinion. 4. The minor axis of the elliptical gear, or twice the distance EG, should equal the distance between the centers of the shafts. 5. The elliptical gear, assuming that it were complete, should have twice the number of teeth that there are in its eccentric driving pinion, and the number of teeth that there are in its eccentric driving pinion, and the number of teeth in both the elliptical gear and eccentric pinion should be even. 6. The shaft hole for the elliptical gear should always be located at the intersection of the major and minor axes, or in the center of the gear. This type of gearing is employed when it is especially desirable to secure a uniform motion during the entire cutting stroke.

Quick-return Movement which Operates Independently.— On one design of automatic screw machine, the quick-return and advance movements of the turret-slide are controlled independently of the turret-slide feed cam by means of a crank. The turret A (Fig. 6) is carried by a slide that moves horizontally along the machine bed. The movements of the turret-slide are derived from two different sources. When the turret tools are at work, the slide is operated by a lead cam through lever B, which has teeth at its upper end meshing with rack C. While the turret is being indexed, it is withdrawn rapidly and then quickly advanced to the working position

again, by the action of crank E which is revolved once for each indexing movement. The rack C transmits motion to the turret-slide through connecting-rod F, which is pivoted to crank E on the turret-slide. This crank is on the "dead center," as shown in the illustration, while the tools are cutting; when the turret is to be indexed to bring the next successive tool in position, it is first withdrawn far enough for the tool to clear the work, and then the shaft carrying crank E is turned one revolution, through suitable gearing, by the

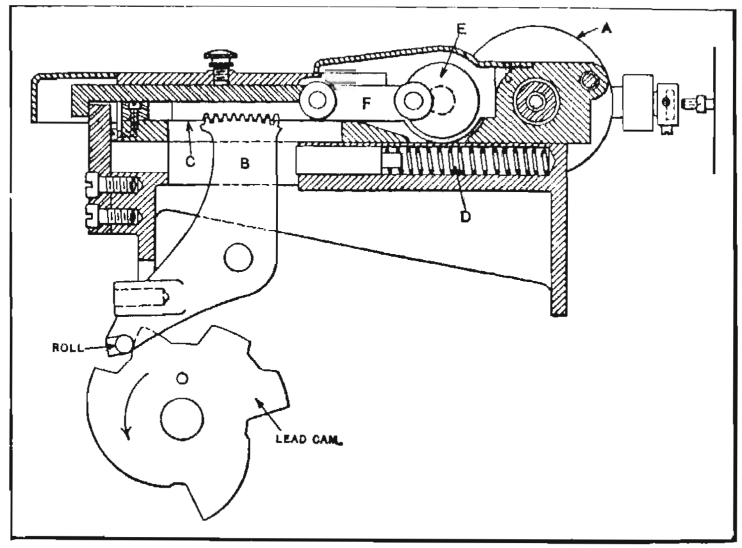


Fig. 6. Independent Quick-return Movement for Screw Machine Turret-slide

engagement of a clutch the action of which is controlled by a trip dog. When the crank revolves, it allows spring D to draw back the turret-slide without rack C, while making one-half turn, and then advance it during the remaining half turn, the rate of movement being increased by the motion derived from the cam, which is laid out to suit the work. This quick-acting crank operates while the roll on the lower end of lever B is passing from the highest point of the cam lobe to the point for starting the next cut.

CHAPTER XI

SPEED-CHANGING MECHANISMS

MECHANISMS for changing the speeds of rotating parts may be divided into two general classes. Those in one class provide convenient means of varying speeds to suit operating conditions. Usually a range of several speeds may be obtained from mechanisms of this class and the change from one speed to another may only require the shifting of a lever or some other form of controlling device.

Speed-changing mechanisms in the second class are designed either for reducing or for increasing the speed a fixed amount, the mechanism being designed for one speed ratio only. For example, the speed between driving and driven shafts may be reduced in the ratio of, say, 10 to 1, or to secure whatever speed change is required. Ordinarily, mechanisms in this class are for reducing the speed of a driving shaft, but in some cases an increase of speed is required. A speed-changing mechanism of this general class may be incorporated in the design of a machine for changing the relative speeds of certain parts, or such a mechanism may be located between a driving motor and the machine to reduce the relatively high motor speed down to the normal working speed of the machine.

General Methods of Speed Regulation. — When speed variations are essential to the operation of machines such, for example, as are used for some kinds of manufacturing work, the changes are usually obtained by hand-controlled speed-changing devices. If such variations are seldom required, it may be necessary to stop the machine and make an adjustment, or replace one or more gears with others of different diameters. When changes of speed are frequently needed, the machine is generally equipped with some mechanical device enabling one or more variations to be obtained rapidly, by sim-

ply moving a wheel, lever, or rod which controls the combination or velocity ratio of the mechanism through which the motion is transmitted. If the machine is of the automatic type, the speed may be regulated according to varying conditions, by the mechanism of the machine itself, which is constructed or adjusted beforehand to give the proper changes. The exact arrangement of the details depends, in any case, upon conditions such as the speed variation required, the importance of rapid changes, the relation of the speed-controlling mechanism to other parts of the machine, etc.

Mechanical devices for varying the speed are of special importance on machine tools. In fact, most machine tools are so constructed that the speed of the cutting tool or of the part being operated upon can be varied, the range or extent of the variation depending upon the type of machine. These changes are desirable in order to cut different kinds of metal at the most efficient speed; for example, soft brass may be turned, drilled, or planed at a much higher speed than cast iron or steel, and, by using the fastest speed that is practicable, obviously the rate of production is increased. Another important reason for speed variation is to secure the proper surface speed for revolving parts, regardless of the diameter, and the correct cutting speeds for rotating tools of different sizes. In the case of lathes or other turning machines, the speed of the work is increased as the diameter decreases, in order to maintain a cutting or surface speed which is considered suitable for the kind of metal being machined. Similarly, drilling or boring machines are so designed that the speed of the drill or boring bar can be varied in accordance with the diameter of the hole being drilled or bored. The design of this part of any machine tool involves determining the minimum and maximum speeds that would ordinarily be required, the total number of variations, the amount of increment by which each step or change varies, and the design of the mechanical device for securing speed changes and transmitting them to the work-spindle or tool. These speed-changing devices usually consist of different combinations of gearing, although belt-driven pulleys and friction gearing are often utilized.

Types of Mechanical Speed-changing Mechanisms. — When a variation of speed is obtained by changing the velocity ratio of two or more parts forming a train of mechanism, one of the following methods is generally employed: (1) By means of conical pulleys connected by a belt or cone-pulleys having "steps" of different diameters upon which a connecting belt may be shifted; (2) by the use of cone-pulleys in conjunction with one or more sets of gears; (3) by means of toothed gears exclusively, with an arrangement that enables the motion to be transmitted through different ratios or combinations of gearing; (4) by employing a friction transmission consisting of driving and driven disks, pulleys, or wheels, so arranged that one member (or an intermediate connecting device) can be shifted relative to the axis of the other for varying the speed. These different types or classes of speed-changing mechanisms are constructed in various ways.

Combination of Cone-pulley and Gearing. — One method of changing speeds by using a cone-pulley in conjunction with gearing is illustrated by diagram *A*, Fig. 1. This particular arrangement is commonly employed on engine lathes and is known as "back-gearing." When the pulley is driving the spindle direct, it is usually locked to the spindle by means of a bolt which connects it with the "face gear" *d*. For the direct drive, the back-gears are disengaged and the main spindle and cone-pulley revolve together. By disengaging the cone-pulley from gear *d* so that it rotates freely about the spindle and engaging the back-gears, motion is transmitted from the "cone gear" *a* to gear *b*, and from *c* to *d*; in this way, the range of speeds obtained by the direct drive is doubled. With a four-step cone-pulley, there would be four direct speeds and four slower speeds with the back-gears engaged, the drive being so proportioned that a gradual increase of speeds from the minimum to the maximum, or *vice versa*, may be obtained. The sleeve which carries the two back-gears revolves about a shaft having eccentric bearings at the ends, so that, by turning this shaft, the back-gears are engaged or disengaged.

Many modern engine lathes have double back-gears, one arrangement being shown at *B*. There are two cone gears *a* and *b* and two mating gears *c* and *d* on the rear shaft, so that a double range of geared speeds may be obtained, in addition to variations secured with the direct drive; thus, with a three-step cone-pulley, there would be a total of nine speeds. The gears *c* and *d* are shifted along the rear shaft for changing their position relative to the cone gears. A modification of

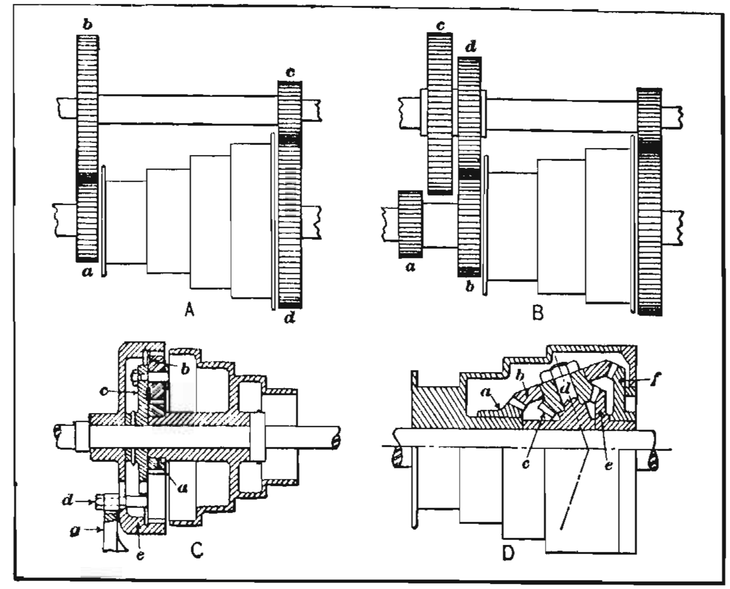


Fig. 1. Gearing and Cone-pulley Combinations for Varying Speed

the double back-geared drive is so arranged that the two gears on the rear shaft are connected by a friction clutch controlled by a conveniently located lever. Another design of lathe headstock gearing is commonly known as "triple gearing," although this term is not always applied to the same form of drive by machine-tool builders. Ordinarily, however, a lathe is said to be triple-geared when there are two gear shafts. The cone-pulley speeds are doubled by driving through one combination of gears, and a third range of speeds is obtained by transmitting the motion through the other combination, the pinion of the second shaft being engaged directly with a large internal gear on the faceplate. Triple gearing is used on large lathes and the direct drive to the face-plate provides a very powerful turning movement, such as is required for taking heavy cuts on castings or forgings of large diameter.

Cone-pulley and Epicyclic Gearing. — The use of a cone-pulley and planetary or epicyclic gearing is shown at *C*, Fig. 1. The cone-pulley has a pinion *a*, which meshes with pinion *b*, mounted on a stud carried by plate *c*. Pinion *b* also meshes with an internal gear forming part of casting *e*. This casting and the cone-pulley are both loose upon the shaft, but plate *c* is keyed to it. When lock-pin *d* engages a notch in plate *c*, the gears are locked together and the shaft is driven directly by the cone, the entire mechanism revolving as a unit. When lock-pin *d* is engaged with a stationary arm *g*, the internal gear is prevented from rotating and motion is transmitted to the spindle of the machine from the cone-pulley, as pinion *a* causes pinion *b* to revolve about the stationary internal gear and carry with it plate *c*, which transmits a slower speed to the spindle than is obtained with the direct drive. This design, which has been applied to some upright drilling machines, is sometimes known as a "differential back-gear."

Another cone-pulley containing epicyclic gearing is shown by the diagram *D*, Fig. 1. Bevel gears are employed in this case, instead of spur gears, and the combination is known as "Humpage's gear." This gearing was designed originally to replace the back-gearing of a lathe, but it has been applied to various classes of machinery. When used in conjunction with a cone-pulley, the arrangement is as follows: The cone-pulley is loosely mounted on its shaft and carries a pinion *a* which meshes with gear *b*. This gear is locked to pinion *c*, thus forming a double gear that is free to turn about arm *d*, the hub of which is also loosely mounted on the spindle or shaft. Gear *b* meshes with gear *f*, whereas pinion *c* meshes with gear *e*. Diametrically opposite arm *d*, there is another arm which carries gears corresponding to *b* and *c*. This additional gearing is included because of its balancing effect and

need not be considered in studying the action of the gearing. The gear *e* is keyed to the spindle, and, except when a direct drive is employed, gear *f* is stationary. With the fulcrum gear *f* stationary and gear *a* revolving, gear *e* and the spindle are rotated at a much slower speed, as the arm *d* and the intermediate connecting gears roll around gear *f*. The direction in which gear *e* rotates for a given movement of gear *a* depends upon the ratio of the gearing, and the direction may be reversed by changing the relative sizes of the gears. When the ratio $\frac{f \times c}{b \times e}$ is less than 1, gears *a* and *e* will revolve in the same direction, whereas, if this ratio is greater than 1, they will revolve in opposite directions. This is compact gearing and the velocity ratio may be varied considerably by a slight change in the relative sizes of the gears.

The velocity ratio when $\frac{f \times c}{b \times e}$ is less than 1 may be determined by the following formula, in which the letters represent the numbers of teeth in the gears marked with corresponding reference letters in the illustration:

$$\text{Ratio} = \frac{\frac{f}{a} + 1}{1 - \frac{f \times c}{b \times e}}$$

If gear *a* has 12 teeth, *b*, 40 teeth, *c*, 16 teeth, *e*, 34 teeth, and *f*, 46 teeth, then,

$$\text{Ratio} = \frac{46/12 + 1}{1 - \frac{46 \times 16}{40 \times 34}} = \frac{4\frac{5}{6}}{\frac{39}{85}} = 10.53$$

Therefore, gear *a* will revolve 10.53 times while gear *e* is making one revolution. If the expression $\frac{f \times c}{b \times e}$ is greater than 1, the formula may be changed as follows:

$$\text{Ratio} = \frac{\frac{f}{a} + 1}{\frac{f \times c}{b \times e} - 1}$$

Geared Speed-changing Mechanisms. — When toothed gearing is used exclusively in a speed-changing mechanism, the most common arrangements may be defined as the (1) sliding-gear type; (2) the clutch-controlled type; (3) the gear-cone and sliding-key type; (4) the gear-cone and expanding-clutch type; (5) the gear-cone and tumbler-gear type; and (6) the multiple crown-gear and shifting-pinion type. Diagram *A*, Fig. 2, illustrates the principle of the sliding-gear design. One of the parallel shafts carries two fixed gears, *a* and *c;* the gears *b* and *d* on the other shaft are free to slide axially so that motion may be transmitted either through gears *a* and *b* or *c* and *d*. The first combination gives a faster speed than the latter, because driving gear *a* is larger than gear *c*. For obtaining a greater range of speeds, two or more sets of sliding gears are used in many cases.

Clutch Method of Control. — Diagram *B*, Fig. 2, illustrates the use of a clutch for controlling speed changes. This clutch is located between the two driven gears and it can be engaged with either of these gears by a lengthwise movement effected usually by a lever. While this clutch is free to slide axially, it is prevented from revolving about the shaft by a spline or key. The driven gears, however, turn freely about the shaft unless engaged by the clutch. A positive clutch is shown in the diagram, or one having teeth which engage corresponding notches in the hubs of the gears; many of the clutches for speed-changing mechanisms, however, are of the friction type.

In the diagrams *A* and *B*, single-belt pulleys are shown upon the driving shafts. This is a common method of rotating the initial driving shaft of speed-changing mechanisms of the all-geared type, the shaft rotating at a constant speed and all of the changes being obtained by the shifting of gears or clutches. On many machines, however, the single constant-speed belt pulley is replaced either by a motor of the constant-speed type or one of the variable-speed type.

Intermeshing Gear Cones and Sliding Key. — The use of intermeshing gear cones and a sliding key for changing speeds is represented by diagram *C*, Fig. 2. Two cones of gears are

mounted upon parallel shafts so that they intermesh, one shaft being the driver and the other, the driven member. All of the gears on shaft *a* are attached to it, whereas those on shaft *b* are free to revolve around the shaft, except when engaged by the key *c*, which can be shifted from one gear to another by moving rod *d*. If the key were in the position shown by the diagram, the drive would be through gears *g* and *e;* if *a* were the driving shaft, the speed of shaft *b* could be increased by

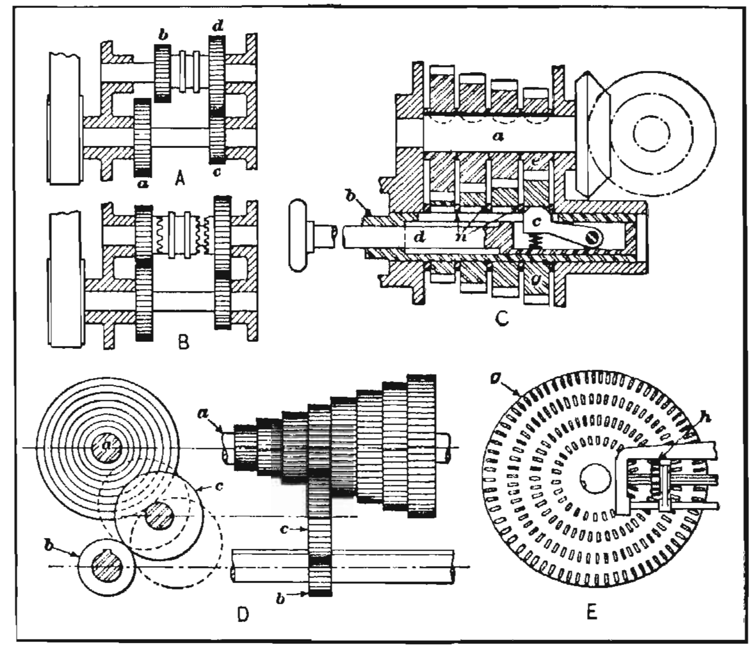


Fig. 2. Diagrams Illustrating Different Types of All-geared Speed-changing Mechanisms

engaging the key with gears to the left. Obviously, the number of speed changes corresponds to the number of gears in the cone.

The driving end of the key projects through a slot in the shaft and the edges are beveled to an angle of about 45 degrees, so that, as the key is moved in a lengthwise direction, it will be depressed by the action of the beveled edge against a steel washer or guard *n* placed between each pair of gears.

With this arrangement, the key is completely disengaged from one gear before meshing with the next one, which is essential with a drive of this kind. The key is forced upward into engagement with the keyways of the different gears, by means of a spring beneath it. A modification of the mechanism just described is so arranged that, instead of locking the gears in the upper cone by means of a sliding key, each gear is fitted with a ring which may be expanded by means of a wedge, the action of which is controlled by suitable means. The gear-cone and sliding-key mechanism is applied to many different types of machine tools, although this form of mechanism is usually installed either for transmitting feeding motion or in connection with spindle drives which require a relatively small amount of power.

Gear-cone and Tumbler-gear Mechanism. — The arrangement of a gear-cone and tumbler-gear mechanism is represented by diagram *D*, Fig. 2. There is a cone of gears on shaft *a* and a pinion *b* which is free to slide on a splined shaft and is connected with cone gears of different diameters, by means of the tumbler gear *c*. The tumbler gear is carried by an arm which can be shifted parallel to the axis of the gear cone for aligning the tumbler gear with any one of the cone gears; this arm can also be moved at right angles to the axis of the gear cone for bringing the tumbler gear into mesh with the various sizes of gears composing the cone (as shown by the dotted circles), and provision is made for locking the arm in its different positions. Cone-and-tumbler gearing is not always arranged as shown by diagram *D;* for instance, the tumbler gear, instead of engaging with a pinion mounted upon a splined shaft, may mesh with a long pinion, or the tumbler gear may be carried by a frame which is adjusted to bring the tumbler gear into mesh with the different cone gears. Another modification consists of a cone of gears which are adjusted axially for alignment with the tumbler gear which is only moved in a radial direction.

Multiple Crown-gear and Shifting Pinion. — The multiple crown-gear type of speed-changing mechanism is represented by diagram *E*, Fig. 2. The crown gear *g* has several concentric rows of teeth, and the speed is varied by shifting the pinion *h* so that it engages a row of larger or smaller diameter. This mechanism has been applied to drilling machines for varying the feeding movements of the drill.

The design and application of the various kinds of speed-changing mechanisms previously described, and the exact arrangement of the gears or other parts are governed very largely by the type of machine and the general nature of the work which it does. Mechanisms of the same general type are often constructed along different lines.

Gear Ratios and Speed Variations. — Proportioning a train of gears to obtain a given velocity ratio, or possibly a given series of speeds, is frequently encountered in the design of geared transmissions. When the problem is simply that of obtaining a given velocity ratio, and when the latter is so large that more than one pair of gears should be used, a uniform reduction between the different pairs is conducive to the highest efficiency. Whenever this arrangement is practicable, the ratio of each pair in a train may be determined by extracting the root of the total ratio. If there are two pairs of gears, extract the square root; for three pairs, extract the cube root, etc. For example, if the total ratio between the first driving and the last driven gear is to be 125 to 1 and three pairs of gears are to be used, the ratio of each pair should preferably equal 5 to 1, since $\sqrt[3]{125} = 5$.

Speeds in Geometrical Progression. — In designing gear combinations for varying spindle speeds or feeding movements, it is general practice, among machine tool builders particularly, to vary the speeds in geometrical progression, successive speeds being obtained by multiplying each preceding term by a ratio or constant multiplier. Thus, if the slowest speed is 50 revolutions per minute and the ratio is 1.3, the succeeding speeds will equal

$$50 \times 1.3 = 65$$
$$65 \times 1.3 = 84.5$$
$$84.5 \times 1.3 = 109.8$$

When the fastest speed f and the slowest speed s in a series are known and also the total number of speeds n, the ratio may be determined by the well-known formula:

$$\text{Ratio} = \sqrt[n-1]{\frac{f}{s}}$$

Since logarithms would ordinarily be used for the extraction of this root, the ratio may be obtained as follows:

Rule. Subtract the logarithm of the slowest speed from the logarithm of the fastest speed and divide the difference by the total number of speeds minus 1. The result will equal the logarithm of the ratio.

Ratios for Machine Tool Drives. — In actual practice, the exact progression obtained may be modified slightly to permit using gears of a certain diametral pitch. For machine tool transmissions, the ratio of a geometrical progression should, as a general rule, be between 1.3 and 1.5, as otherwise there will be either too small or too great a difference between successive speeds. There would be no practical advantage in a series of speeds varying by small increments equivalent to a ratio of say, 1.1, whereas, if the ratio were 1.7 or possibly 2, the changes from one speed to the next would be excessive. Feeding mechanisms may be designed for ratios of 1.2 or less, depending on the type of machine.

Speeds of machine tool drives and especially feed changes are sometimes varied according to "chromatic scale progression," with a ratio of either 1.4142 or 1.189 in case a lower ratio is required. The first ratio is the square root of 2, and the second the fourth root of 2. The object of using these particular ratios is to obtain a series of speeds or feeds containing the even ratios, 2, 4, 8, 16, etc.

Speed Calculations for Lathe Headstock. — As an example of gear designing, for obtaining speed changes, assume that the problem is to design a lathe headstock with a stepped cone pulley. The headstock is to have a five-step cone with back-gears, giving ten speeds in all, and the speeds are to range from 300 down to 10 revolutions per minute.

The first thing to determine is the geometrical progression

of ten numbers from 10 to 300. As the maximum and minimum speeds and the number of speed changes are known, the ratio of the progression may be determined by using the formula previously given. Inserting the values in this formula we have,

$$R = \sqrt[10-1]{\frac{300}{10}} = 1.46$$

Therefore to have speeds from 10 to 300 revolutions per minute, ten in number, we must use a ratio of 1.46, which on multiplying gives the results shown in the first column of Table 1. Five of these speeds are obtained from the five-step cone. The function of the back-gear is to double the number of speeds, making ten in all. We have decided, from observation or experience, that the least practicable diameter for the

Table I

Speeds Varying in Geometrical Progression	Speeds Actually Obtained
10......... = 10.0	9.7
10 × 1.46 = 14.6	14.5
14.6 × 1.46 = 21.3	21.0
21.3 × 1.46 = 31.0	30.0
31 × 1.46 = 45.3	45.0
45.3 × 1.46 = 66.0	65.0
66 × 1.46 = 96.4	97.0
96.4 × 1.46 = 140.5	140.0
140.5 × 1.46 = 205.0	202.0
205 × 1.46 = 299.3	300.0

smallest cone step is 5 inches. This gives a starting point from which all the other cone steps may be calculated, as their ratios must be to one another as the ratios of their speeds.

Cone Pulley Diameters For Crossed Belts. — The highest speed is obtained, of course, when the belt is driving from the largest step of the countershaft cone to the smallest step of the headstock cone and the back-gear is out of mesh. Countershaft and headstock cones are usually made alike; consequently the two middle steps will be the same diameter, which makes the countershaft speed the same as the middle speed of the cone, without the back-gears, or 140.5 revolutions per minute.

The size of the largest step may be obtained by the formula:

$$\text{Diam. largest step} = \frac{\text{Max. spindle speed} \times \text{Diam. smallest step}}{\text{Speed of countershaft}}$$

$$\text{Diameter of largest step} = \frac{300 \times 5}{140.5} = 10.7 \text{ inches}$$

The middle step should next be determined. It is the sum of the large and the small diameters divided by 2:

$$5 + 10.7 \div 2 = 7.85 \text{ inches}$$

The second largest step will be the mean between the middle and the largest step:

$$7.85 + 10.7 \div 2 = 9.27 \text{ inches}$$

The second smallest step in like manner will be:

$$5 + 7.85 \div 2 = 6.43 \text{ inches}$$

Calculating the Back-gears. — The back-gears obviously must be of such ratio as to reduce the first five speeds to a second or slower five. The ratio between the slowest of the first five and the slowest of the second five is as 10 to 66 revolutions per minute or 6.6 to 1. As the back-gear consists of two pairs of gears, this ratio is the product of their respective ratios, and we must extract the square root to find what these respective ratios may be. The square root of 6.6 is 2.56. Therefore the ratio of each pair of gears should be 2.56 to 1. We may now proceed to find the number of teeth in the gears. It is customary to make the gears next to the faceplate of heavier pitch, as they are the slowest running and consequently have the greatest tooth pressure. Suppose we find by calculation, experience, or comparison with other makers that the faceplate gears should be 4 diametral pitch and the back-gear shaft admits of using a 15-tooth pinion. Then:

$$15 \times 2.56 = 39 \text{ teeth, approximately}$$
$$15 \div 4 = 3\tfrac{3}{4} \text{ inches pitch diameter}$$
$$39 \div 4 = 9\tfrac{3}{4} \text{ inches pitch diameter}$$

One-half of the sum of the pitch diameters, or $6\frac{3}{4}$ inches, equals the center-to-center distance. Now we want to make the other pair of gears 5 diametral pitch, and 5 diametral pitch gears will not fit into $6\frac{3}{4}$-inch centers. The solution of the problem is to make the faceplate gears with 16 and 40 teeth, respectively, which will make the centers an even 7 inches.

$$16 \div 4 = 4 \text{ inches pitch diameter}$$
$$40 \div 4 = 10 \text{ inches pitch diameter}$$

Therefore the center distance equals $\frac{10 + 4}{2} = 7$ inches.

This alters the ratio slightly, as the ratio of 40 to 16 is 2.5. The difference must be made up in the other pair of gears as follows:

$$6.6 \div 2.5 = 2.64 \text{ ratio for 5 diametral pitch gears}$$

Twice the center distance or 14 inches is the combined pitch diameters of the gears, and with 5 diametral pitch the combined number of teeth is 70. The desired ratio is 2.64. Add 1 to this ratio and divide it into the total number of teeth. Thus:

$$70 \div 3.64 = 19, \text{ approximately}$$

Therefore we have 19 teeth in the pinion and 70 — 19 = 51 teeth in the gear. The spindle speeds will now be checked as

Table II

Countershaft Speed, R.P.M.	Countercone Diam. Inches	Headstock Cone Diameter, Inches	Back-gears	Spindle Speeds, R.P.M.
140 ×	10.7 ÷	5.00	Out	= 300
140 ×	9.27 ÷	6.43	Out	= 202
140 ×	7.85 +	7.85	Out	= 140
140 ×	6.43 +	9.27	Out	= 97
140 ×	5.00 ÷	10.7	Out	= 65
140 ×	10.7 ÷	5.00	× (19 × 16 + 51 × 40)	= 45
140 ×	9.27 +	6.43	× (19 × 16 + 51 × 40)	= 30
140 ×	7.85 ÷	7.85	× (19 × 16 ÷ 51 × 40)	= 21
140 ×	6.43 ÷	9.27	× (19 × 16 ÷ 51 × 40)	= 14.5
140 ×	5.00 ÷	10.7	× (19 × 16 + 51 × 40)	= 9.7

shown in Table 2 to see how close we have come to the desired result. We will make the countershaft speed 140 revolutions per minute even. The results shown in Table 2 are also compared with the ideal speeds in Table 1. It is possible to get even closer results by a little juggling, but such accurate speeds are seldom necessary.

Planetary or Epicyclic Gear Trains. — If one of the gears in a train is fixed or stationary, and another gear (or gears) revolves about the stationary gear in addition to rotating relative to its own axis, the mechanism is known as an *epicyclic train of gearing,* because points on the revolving gears describe

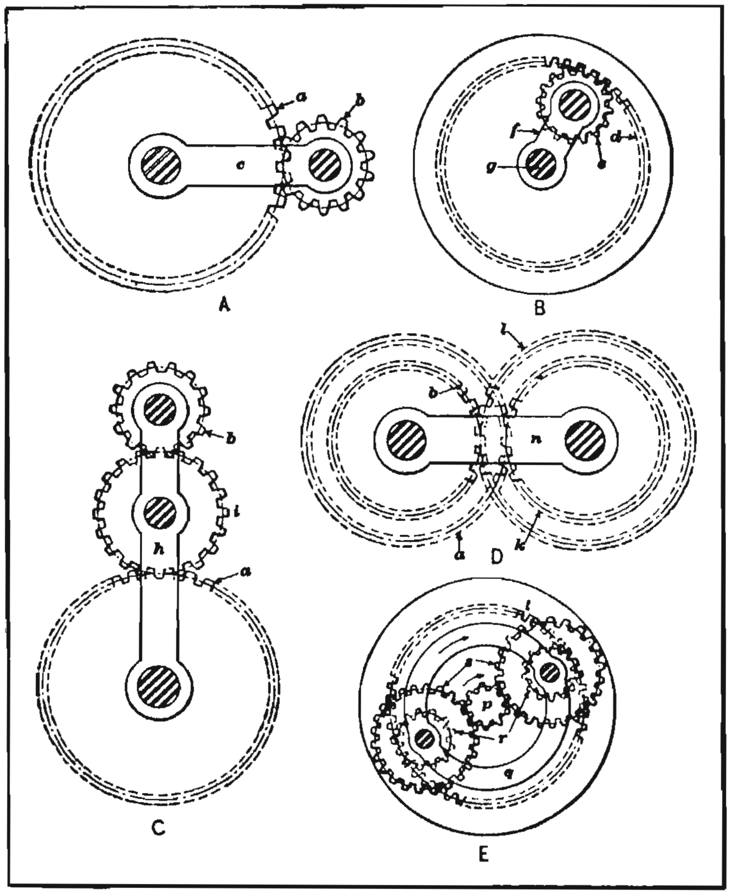


Fig. 3. Epicyclic or Planetary Gearing

epicycloidal curves. The two gears a and b (see diagram A, Fig. 3) are held in mesh by a link c. If this link remains stationary and gear a makes one revolution, the number of revolutions made by gear b will equal the number of teeth in a divided by the number of teeth in b, or the pitch diameter of a

divided by the pitch diameter of b. If a and b represent either the pitch diameters of the gears or numbers of teeth, the revolutions of b to one turn of a equal $\frac{a}{b}$. If gear a is held stationary and link c is given one turn about the axis of a, then the revolutions of gear b, relative to arm c, will also equal $\frac{a}{b}$, the same as when gear a was revolved once with the arm held stationary. Since a rotation of arm c will cause a rotation of gear b in the same direction about its axis, the total number of revolutions of gear b, relative to a fixed plane, for one turn of c, will equal 1 (the turn of c) plus the revolutions of b relative to c or $1 + \frac{a}{b}$. For example, if gear a has 60 teeth and gear b, 20 teeth, one turn of arm c would cause b to rotate $\frac{60}{20}$, or 3 times about its own axis; gear b, however, also makes one turn about the axis of gear a, so that the total number of revolutions relative to a fixed plane equals $1 + \frac{60}{20} = 4$ revolutions.

In order to illustrate the distinction between the rotation of b around its own axis and its rotation relative to a fixed plane, assume that b is in mesh with a fixed gear a and also with an outer internal gear that is free to revolve. If the speed of the internal gear is required, it will be necessary, in calculating this speed, to consider not only the rotation of b about its own axis, but also its motion around a, because the effect of this latter motion on the internal gear, for each turn of link c, is equivalent to an additional revolution of b.

Diagram B, Fig. 3, represents an internal gear d in mesh with gear e on arm f. If arm f is held stationary, the revolutions of e for one turn of d equal $\frac{d}{e}$, d and e representing the numbers of teeth or pitch diameters of the respective gears. If the internal gear is held stationary and arm f is turned about axis g, the rotation of e about its axis will be clockwise when f is turned counter-clockwise, and *vice versa;* hence, the revolutions of gear e, relative to a fixed plane, for one turn of f about g, will equal the difference between 1 (representing the turn of f) and the revolutions equal to $\frac{d}{e}$.

Method of Analyzing Epicyclic Gear Trains. — A simple method of analyzing epicyclic gearing is to consider the actions separately. For instance, with the gearing shown at A, Fig. 3, the results obtained when link c is fixed and the gear a (which normally would be fixed) is revolved are noted; if gear a is revolved in a clockwise direction, then, in order to reproduce the action of the gearing, the entire mechanism, locked together as a unit, is assumed to be given one turn counter-clockwise. The results are then tabulated, using plus and minus signs to indicate directions of rotation. Assume that gear a has 60 teeth and gear b, 20 teeth, and that + signs represent counter-clockwise movements and — signs clockwise movements. If link c is held stationary and gear a is turned clockwise (—) one revolution, gear b will revolve counter-clockwise (+) $^{60}/_{20}$ revolution. Next consider all of the gears locked together so that the entire combination is revolved one turn in a counter-clockwise (+) direction, thus returning gear a to its original position. The practical effect of these separate motions is the same as though link c were revolved once about the axis of a fixed gear a which is the way in which the gearing operates normally. By tabulating these results as follows, the motion of each part of the mechanism may readily be determined:

	Gear a	Link c	Gear b
Link Stationary............	-1 turn	0 turn	$+\frac{60}{20}$ turn
Gears Locked..............	$+1$ turn	$+1$ turn	$+1$ turn
Number of Turns...........	0	$+1$	$+4$

The algebraic sums in line headed "Number of Turns" indicate that, when gear a is held stationary and link c is given one turn about the axis of a, gear b will make 4 revolutions relative to a fixed plane in a counter-clockwise or + direction, when link c is turned in the same direction.

The application of this method to the arrangement of gearing shown at B, Fig. 3, will now be considered. Assume that gear d has 60 teeth and gear e, 20 teeth. Then, if gear d is turned clockwise with link f stationary, and the entire mechanism with the gears locked is turned counter-clockwise, an analysis of the separate motions previously referred to will give the following results:

	Gear d	Link f	Gear e
Link Stationary...........	-1 turn	0 turn	$-\frac{60}{20}$ turn
Gears Locked.............	$+1$ turn	$+1$ turn	$+1$ turn
Number of Turns..........	0	$+1$	-2

Effect of Idler in Epicyclic Gear Train. — If an idler gear i is placed between gears a and b (diagram C, Fig. 3), the latter will rotate about its axis in a direction opposite to that of the link (the same as with the arrangement shown at B), and the revolutions of gear b, relative to a fixed plane, for one turn of link h about the axis of a, will equal the difference between 1 (representing the turn of h) and the revolutions equal to $\frac{a}{b}$. Assume that gear a has 60 teeth, idler gear i, 30 teeth, and gear b, 20 teeth. Then the turns of b, relative to a fixed member for one turn of h about the axis of a, are shown by the following analysis:

	Gear a	Idler i	Link h	Gear b
Link Stationary.........	-1 turn	$+\frac{60}{30}$ turn	0 turn	$-\frac{60}{20}$ turn
Gears Locked...........	$+1$ turn	$+1$ turn	$+1$ turn	$+1$ turn
Number of Turns.......	0	$+3$	$+1$	-2

The direction of rotation of b, relative to a fixed member, may or may not be in the same direction as that of link h, depending upon the velocity ratio between gears a and b. If gears a and b are of the same size, one turn of link h will cause b to revolve once about its own axis, but, as this rotation is in a direction opposite to that of h, one motion neutralizes the other, so that b has a simple motion of circular translation relative to a fixed member. If gear b were twice as large as a, it would then revolve, for each complete turn of link h, one-half revolution about its own axis, in a direction opposite to the motion of h; this half turn subtracted from the complete turn of link h gives a half turn in the same direction as h, relative to a fixed member.

Compound Train of Epicyclic Gearing. — Diagram *D*, Fig. 3, illustrates a compound train of epicyclic gearing. This arrangement modified to suit different conditions is commonly employed. Gear *a* represents the fixed member and meshes with gear *k*, which is attached to the same shaft as gear *l*. Gear *l* meshes with gear *b* the axis of which coincides with that of fixed gear *a*. Assume that gear *a* has 36 teeth, gear *k*, 34 teeth, gear *l*, 35 teeth, and gear *b*, 35 teeth. Then one turn of link *n* about the axis of gear *a* would give the following results:

	Gear *a*	Link *n*	Gears *k* and *l*	Gear *b*
Link Stationary...	−1 turn	0 turn	$+\frac{36}{34}$ turn	$-(\frac{36}{34} \times \frac{35}{35})$ turn
Gears Locked.....	+1 turn	+1 turn	+1 turn	+1 turn
Number of Turns.	0	+1	$+2\frac{1}{17}$	$-\frac{1}{17}$

From this analysis, it will be seen that, for each counterclockwise turn of link *n*, the rotation of gear *b* equals $1 - \frac{a}{k} \times \frac{l}{b}$ in which the letters correspond either to the pitch diameters or numbers of teeth in the respective gears shown at *D* in Fig. 3. If the value of $\frac{a}{k} \times \frac{l}{b}$ is less than 1, gear *b* will revolve in the same direction as link *n*, whereas, if this value is greater than 1, gear *b* will revolve in the opposite direction.

Compound epicyclic gearing may be used for obtaining a very great reduction in velocity between the link *n* and the last gear *b* in the train. As an extreme example, suppose gear *a* has 99 teeth, gear *k*, 100 teeth, gear *l*, 101 teeth, and gear *b*, 100 teeth. The speed of gear *b* will equal $1 - \frac{99 \times 101}{100 \times 100} = \frac{1}{10{,}000}$ revolution; hence link *n* would have to make 10,000 revolutions for each revolution of gear *b*. The arrangement of epicyclic gearing shown at *D* is known as a *reverted train*.

Diagram *E* shows another arrangement of reverted train. An internal gear *t* forms part of the mechanism, and either this gear, frame *q*, or pinion *p* may be the stationary member, depending upon the application of the mechanism. In this case, instead of a single set of gears between *p* and *t*, there is

a double set located diametrically opposite and connected by a suitable frame *q*. This arrangement is similar to the mechanism of a certain type of geared hoist. The central pinion *p* is the driving member, internal gear *t* is stationary, and the frame *q* is the driven member and imparts motion to the hoisting sheave.

Sun and Planet Motion. — A mechanism of the general type illustrated by diagram *A*, Fig. 3, was employed by Watt for transmitting motion from the connecting-rod to the engine shaft, because the crank motion had been patented previously. This mechanism is known as a "sun and planet" motion, the fixed gear *a* representing the sun and the revolving gear *b*, the planet. In applying this mechanism to an engine, one gear was keyed on the shaft and the other was fixed to the connecting-rod. The connecting link between the gears was loose on both shafts. A forward and return stroke of the piston caused the connecting-rod gear to pass once around the shaft gear, but without revolving on its own axis, as it was attached to the connecting-rod. With this arrangement, if both gears are of the same diameter, the shaft gear will make two revolutions for one turn of the connecting link between the gears or one revolution for each stroke.

Differential Gearing for Large Speed Reduction. — When it is necessary to obtain a speed reduction of large magnitude, some arrangement of differential gearing may be preferable. Belting may not be suitable, either because it does not give a positive drive from the driving to the driven shaft, or because there is not sufficient room for the pulleys. Lack of room may also prevent the use of chains and sprockets. Spur gear trains often require too many gears, thus introducing high costs and an undue amount of power loss through friction. Differential gearing as a means of obtaining a satisfactory speed reduction mechanism of compact form is utilized on many classes of machines. With the form shown in Fig. 4, the speed reduction is made from the shaft *A* to the shaft *B*. As may be seen, the gears *C* and C_1 are fixed on shaft *A*. Gear *E* is merely an idler meshing with gears *C* and *D*. Gear

D is fixed on the hub of bevel gear F, which is bushed and free to revolve on the shaft B. Gear C_1 meshes directly with gear D_1. Gear D_1 is mounted on the hub of the bevel gear F_1 which is bushed and free to revolve on shaft B, the same as gear F. The crank H is keyed to the shaft B, and on its two arms are mounted the bevel idler gears G and G_1 which are bushed and free to revolve on the arms.

Now as C and C_1 are keyed to the shaft A, it is evident that

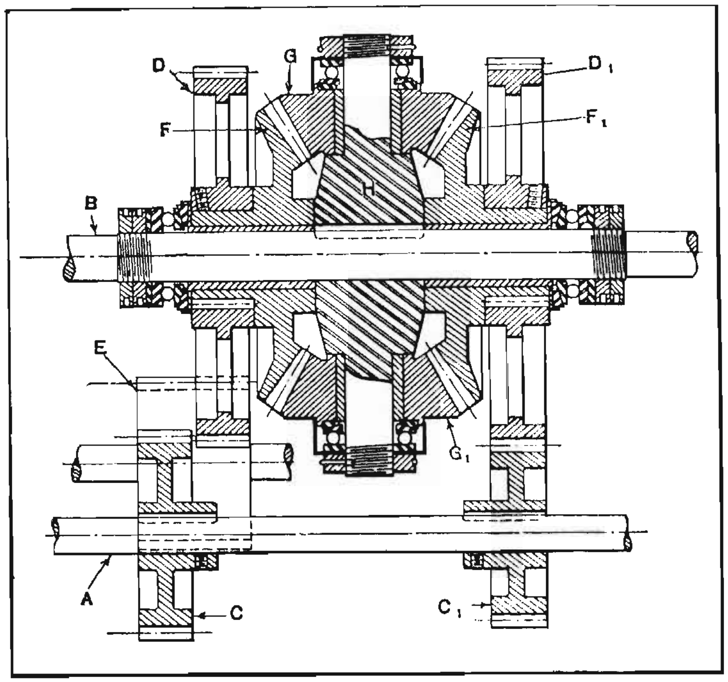


Fig. 4. Speed Reduction Mechanism Employing Differential Gearing

they must always turn or rotate in the same direction. It should be noted further that C_1 drives direct to D_1 and that C drives D through the idler gear E. Therefore D and D_1 must revolve in opposite directions. Their speeds are also different, as gear C has more teeth than gear C_1. Gears F and F_1 must, of course, revolve with gears D and D_1 which are mounted on their respective hubs. The bevel gears G and G_1

meshing with both F and F_1 must therefore revolve on their own axis. If F and F_1 should revolve at the same speed but in opposite directions, gears G and G_1 would be stationary with respect to the axis of the shaft B, that is, they would revolve on their own axis, but they would not revolve about shaft B. Now if F and F_1 should revolve at different speeds, G and G_1 must revolve about the axis of the shaft B, and thus revolve shaft B through crank H.

The number of revolutions per minute of shaft B, which is driven by crank H will be one-half the algebraic sum of the number of revolutions per minute of the speeds of the bevel gears F and F_1. In order to make this clear, let it be assumed that a point on the pitch circle of gear F_1 is traveling at a rate of $x + y$ feet per minute, and that gear F is stationary. It is obvious that a point the same distance from the center of shaft B on the axis of G and G_1 will travel at a speed of $\frac{1}{2}$ $(x + y)$ feet per minute. Assume that a corresponding point on gear F is traveling x feet per minute in the opposite direction or $-x$ feet per minute, and consider gear F_1 to be stationary. The corresponding point on the axis of gears G and G_1 will now travel at a rate of $\frac{1}{2}$ $(-x)$ feet per minute. Next assume that both gears F_1 and F are traveling at their respective speeds of plus $x + y$ and minus x feet per minute. Now adding the speeds of F_1 and F we obtained $x + y - x = y$ feet per minute. Then a speed of one-half y feet per minute equals the speed of the point under consideration on the axis of gears G and G_1.

As the pitch diameters of gears F and F_1 are equal and cannot be otherwise, and as these gears mesh with gears G and G_1, it is evident that the number of revolutions per minute can be readily employed to designate the speed. Therefore, the speed of shaft B can be expressed as one-half the algebraic sum of the revolutions per minute of gears F and F_1. The selection of the bevel gears is merely a matter of choosing such sizes as can be used in the available space, and still be of sufficient size to give the necessary tooth strength for the material used and the load imposed. The diameters of gears

G and G_1 should, of course, be made as large as permissible so that they will not revolve on the crank H at a higher speed than necessary. In determining the sizes of the spur gears to be used, it is only necessary (not considering the available space) to select such sizes as will give the desired difference in speed between F and F_1, together with a suitable surface speed.

The following example will serve to make clear the procedure followed in laying out or designing a speed reduction device of the type shown in the illustration. We have shaft A running at a speed of $-$ 625 revolutions per minute (anti-clockwise), and it is desired to drive shaft B from shaft A so that shaft B will revolve in a clockwise direction at a speed of approximately 3 revolutions per minute. The algebraic sum of the speeds of gears F and F_1, must, therefore, be about $+$ 6 revolutions per minute. Taking 300 revolutions per minute as an approximate speed for F_1, we may proceed to determine the pitch and size of the gears to be used.

Let it be assumed that we select an 8-pitch gear having a pitch diameter of 7½ inches for D_1 and an 8-pitch gear having a pitch diameter of 3½ inches for gear C_1. These gears will give bevel gear F_1 a speed of $+$ 291.67 revolutions per minute. The speed of gear F should therefore equal approximately $-$ 291.67 $+$ 6 or $-$ 285.67 revolutions per minute. Now if gear D has a pitch diameter of 7⅜ inches and gear C has a pitch diameter of 3⅜ inches, gear F will have a speed of approximately $-$ 286.02 revolutions per minute. The difference between the number of revolutions of gears F and F_1 will then equal 5.65 revolutions per minute, and the speed of shaft B will be one-half as great or $+$ 2.82 revolutions per minute.

It is, of course, difficult to obtain an exact speed for shaft B, but by using gears of as fine a pitch as possible without an undue sacrifice of strength, we can obtain very nearly the exact speed desired. The center distance between the two shafts can in some cases be varied, thus increasing the range in speed reduction that may be obtained.

The possibilities of a speed reduction device of this kind is readily apparent to the designer. The speed of the shaft B with respect to shaft A is governed entirely by the speed ratio existing between the gears D and D_1. The speed of shaft A may be as high as the successful operation of the gears will allow, and still by the right combination of gears it may be possible to obtain a very low speed for shaft B. At the same time the whole gear assembly is very compact, and may be installed where other forms of speed reduction mechanisms cannot be used. In order to obtain the best results, all the running parts should, of course, be well oiled and this may be readily accomplished by running the whole assembly in an oil bath or in a case partly filled with oil. Operated under these conditions, the friction loss is very low.

Differential Mechanism for Reduction of 840 to 1. — In machine design large reductions in speed are often obtained by the use of a differential mechanism, especially when the reducing unit must occupy a comparatively small amount of space. A differential mechanism in which a reduction of 840 to 1 is secured by using only two spur gears, two worms, two worm-wheels, two bevel gears, two bevel pinions and a yoke, is shown in Fig. 5. The reduction ratio of this gearing may be altered to suit conditions by simply varying the number of teeth in the two spur gears. Dimension X is approximately 7⅛ inches on this particular lay-out.

Spur gear A is the driving member of the differential unit. It is mounted on the same shaft as worm C, and through these two parts drives worm-wheel E. Gear A also meshes with gear B, and thus drives a shaft running parallel to that on which it is mounted. On the second shaft is a worm D by means of which power is transmitted to worm-wheel F. Gears A and B revolve in opposite directions, and as the thread of both worms is right-hand, worm-wheels E and F also rotate in opposite directions. Pinned to the adjacent sides of the two worm-wheels are bevel gears G, which, together with their respective worm-wheels, are free to turn on the driven shaft. These bevel gears mesh with two idler pinions on studs at

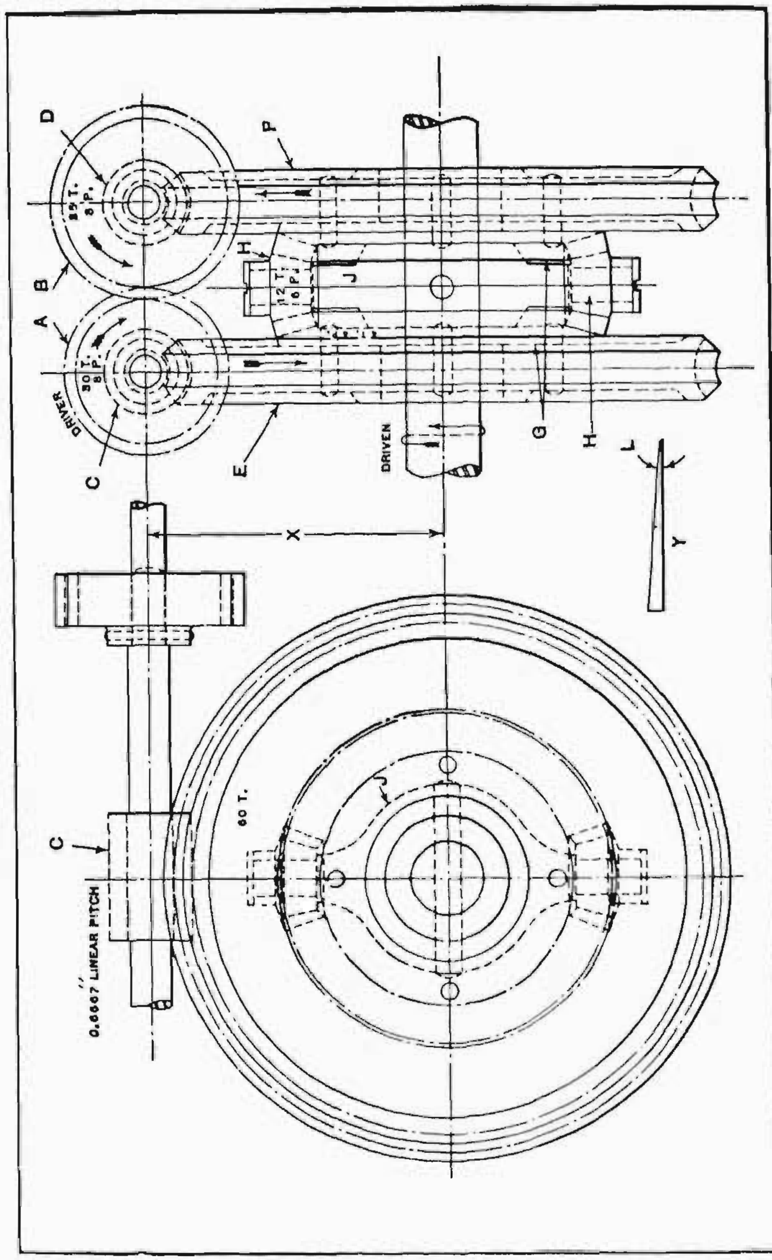


Fig. 5. Geared Mechanism for Obtaining a Speed Reduction of 840 to 1 Within a Small Space

opposite ends of driving yoke *J*, which is pinned to the driven shaft. The idler pinions are held on the studs by means of screws.

The large reduction obtained by this unit is due to giving spur gear *B* five more teeth than gear *A*. As a consequence, gear *B* makes only 30/35 revolution per revolution of gear *A*. The ratio of each set of worm-gearing is 60 to 1, and so for each revolution of gear *A*, worm-wheel *E* makes 1/60 revolution. The 30/35 revolution imparted to gear *B* at the same time causes worm-wheel *F* to turn 1/70 revolution in the opposite direction to that in which worm-wheel *E* is revolved. Because of the two worm-wheels revolving in opposite directions, the result on yoke *J* is the same as if worm-wheel *F* were held stationary and worm-wheel *E* were moved the difference between 1/60 and 1/70 revolution. This would mean a forward movement of worm-wheel *E* of 1/420 revolution.

As the center of driving yoke *J* is located half way between bevel gears *G*, the actual movement of the yoke per revolution of gear *A* will be only one-half the difference between the forward and backward movements of the two worm-wheels, or 1/840 revolution. This can be easily proved by constructing a triangle as shown at *Y*, in which the line adjacent to angle *L* represents the pitch diameter of pinions *H*, and the length of the opposite side equals the distance that a point on the pitch circle of the bevel gear on worm-wheel *E* would move during 1/420 revolution. Then if the adjacent side is bisected and a perpendicular is erected at that point, the length of the perpendicular will be one-half the length of the opposite side of the triangle; this perpendicular line will equal the distance a point on the center line of the yoke, located from the center of the driven shaft a distance equal to the pitch radius of bevel gears *G*, will move during the movement of the bevel gear on worm-wheel *E*. This yoke movement equals 1/840 revolution.

Compound Differential Gears for Varying Speeds.—The differential speed-changing mechanism shown in Fig. 6 has spur gears and pinions but no internal gear. This is a compound or reverted train and is intended for an automatic screw machine of the heavier class in order to provide a slow and powerful movement to the spindle for heavy thread-cutting operations, or for any other heavy work which requires a powerful drive. The gearing is contained within the spindle driving pulleys on the back shaft of the spindle head. There are three pulleys and the slow speed is obtained by shifting the belt to the center pulley *A*, and engaging the sliding clutch *B* with gear *C*; as this clutch slides upon a square shaft and cannot revolve, the gear *C* is held stationary. There are two sets of planetary pinions *D* and *E* located diametrically oppo-

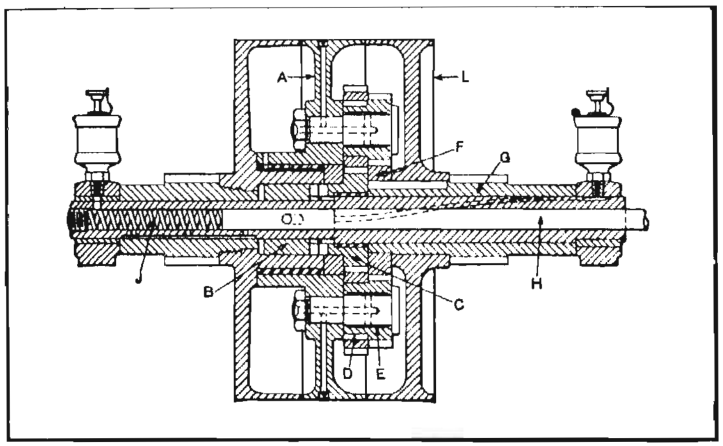


Fig. 6. Compound or Reverted Train of Epicyclic Gearing for Reducing Speed

site. The pinions on each stud are locked together but they are free to revolve about the stud. Pinions *D* rotate around the fixed gear *C*, while pinions *E* revolve the driven gear *F* at a slow speed, but with considerable power. The gear *F* is keyed to the extension of pinion *G* which meshes directly with the front spindle gear of the machine. When this slow speed is not required, the clutch *B* is disengaged, so that the entire train of differential gears is free upon the loose center pulley *A*. Two spring plungers (not shown) attached to pulley *A* engage the rim of pulley *L* and cause both pulleys to revolve together when the slow-speed attachment is not engaged, so

that the planetary pinions will not revolve upon their studs at this time. The clutch *B* is shifted by a cam-operated rod *H* acting in conjunction with a spring *J*.

With this arrangement of gearing, the differental action and reduction of speed is the result of the difference in the diameters of pinions *D* and *E* and their mating gears. When the slow-speed attachment is operating, the larger pinions *D* roll around the stationary gear *C* and force gear *F* to follow slowly in the same direction. This action will be more apparent if that part of the larger pinion *D* which is in engagement with stationary gear *C*, at any time, is considered as a lever pivoted at the point where the teeth mesh with the stationary gear. As the pinion *D* revolves and the imaginary lever swings around its fulcrum, the teeth of the smaller pinion *E* in contact with gear *F* force the latter to move in the same direction in which the rolling pinions *D* and *E* and pulley *A* are moving.

Slow Starting Motion for Textile Machine. — The slow starting motion attachment shown in Fig. 7 is applied to textile machines that are used for winding the warp threads onto a loom beam, as a precaution against breakage of the threads, caused by sudden starting. Shaft *A*, which is a short auxiliary shaft on which the pulleys and other parts are carried, is mounted in a bearing which is bolted to the frame of the machine. Bushing *B* is fastened by a set-screw to *A* and forms the bearing for the loose pulley *C*. The slow-motion pulley *D*, which has a twelve-tooth gear cast on its inside hub, turns freely on the shaft *A*. The casting *E*, which is also fastened by a set-screw to *A* carries the steel pinion *F*, the shank of which revolves in a bearing at the end of the casting. This pinion has twelve teeth and on the opposite end of the shaft a gear of nineteen teeth is assembled, which meshes with the twelve-tooth gear on the slow-motion pulley *D*. The forty-tooth clutch gear *G* meshes with *F* and is loose on shaft *A*. This gear has a five-tooth clutch, as shown in the right-hand view at *g*. The driving pulley *H* is also loose on *A* and has a brake pulley *I* and a driving gear H_2 cast integral with it, the gear H_2 driving the main gear of the machine. On the inside of pulley H are three short studs J, each of which carries a small pawl K.

The belt-shifting mechanism is not shown in the illustration, but it is operated by a foot-treadle, which is fastened to the treadle shaft. On the end of the treadle shaft, just inside the frame, is a segment arm, which meshes with the teeth on one end of a double gear. The teeth on the opposite end of this gear operate a sliding rack which projects from the frame, just above the pulleys. The belt guide is bolted to this rack, and as the rack is run outward and the belt shifted to pulley C, the brake, which is also attached to the treadle shaft, is

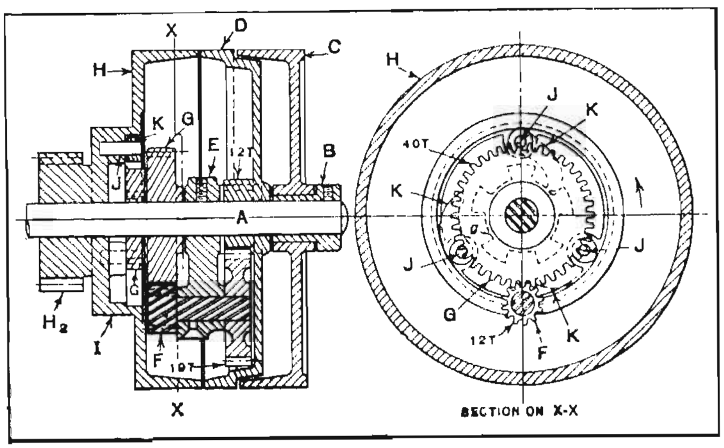


Fig. 7. Slow Starting Motion Mechanism of Textile Machine

brought into contact with I and the machine is quickly stopped. This prevents the ends of yarn that are being wound on the beam from becoming slack or entangled, which would be a troublesome and awkward condition.

When the belt is shifted from C to D, the number of revolutions per minute of gear G is reduced to about one-fifth that of pulley D as is shown in the ratio of the teeth $\dfrac{12 \times 12}{19 \times 40}$, or approximately 1/5. The relation of the three pawls K to the five teeth on the clutch g is such that one pawl is always down; that is, one will always drop into the position occupied by the

upper pawl, as shown in the right-hand view of the illustration. The pawl being thus engaged, the engaging tooth of the clutch, which is revolving counter-clockwise and at reduced speed, pushes against the pawl, driving pulley H, and incidentally gear H_2, slowly in the direction shown by the arrow. When the belt is shifted onto pulley H, which is also a loose pulley, gear H_2 will assume the full speed, pawls K will simply drag over the teeth in the clutch, and pulley D will become idle, as a result of the disengagement of the pawls and clutch.

Rotary Speed Varied Each Half Revolution. — An electric switch testing machine required that shaft A (see Fig. 8)

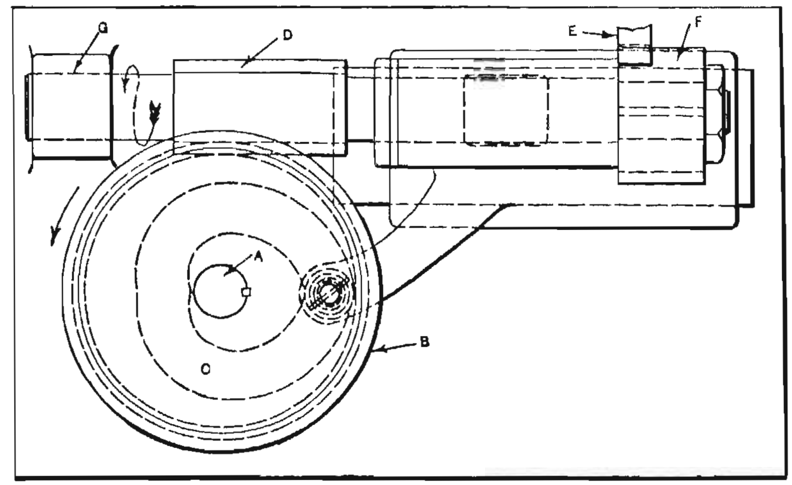


Fig. 8. Mechanism for Changing Speed of Driven Shaft Every Half Revolution

make one-half revolution in three seconds at a uniform speed, and the following half revolution in four seconds, also at a uniform speed. This result is obtained by means of a cam C (on the driving worm-wheel B), which imparts a uniform reciprocating motion to the driving worm D, causing the worm-wheel alternately to be advanced and retarded as the worm moves first with and then against the worm-wheel rotation.

The mechanism is driven by gear E, which meshes with pinion F. This pinion is attached to the worm-shaft and is wide enough to provide for the lengthwise movement of the worm. The worm-wheel *B* and cam *C* are integral. The cam roller is carried by an arm which is part of the slide that forms a bearing for one end of the worm-shaft. The other end is supported in bearing *G* through which it is free to slide when the worm is moved axially.

Worm-wheel *B* has 56 teeth and worm *D* has 4 threads per inch. The speed of worm *D* is 8 revolutions per second. When shaft *A* is being turned one-half revolution in three seconds, the worm moves with or in the direction of rotation of the worm-wheel *B,* so that *B* is turned somewhat faster than it would be if worm *D* were not moved axially. The increased movement causes shaft *A* and worm-wheel *B* to turn one-half revolution in one second less than when worm *D* moves backward against the rotation of worm-wheel *B.*

Since worm *D* makes 8 revolutions per second, there are 24 revolutions in 3 seconds, and at the same time the worm advances one inch due to the action of the cam. Now the pitch of the worm thread and circular pitch of the worm-wheel is $\frac{1}{4}$ inch; hence, 1 inch axial movement of the worm is equivalent to 4 teeth of the wheel, so that the total movement equals $24 + 4 = 28$ teeth, or one-half revolution, as the wheel has 56 teeth. When worm *D* moves backward against the rotation of the worm-wheel, it makes 32 revolutions in four seconds, but the 1 inch axial movement, in effect, subtracts motion equivalent to 4 teeth, or $32 - 4 = 28$, or one-half turn in four seconds, as compared with three seconds for the opposite direction.

Two-gear Clock Mechanism of 12 to 1 Ratio. — In the design of mechanisms in general, eliminating useless parts and obtaining the desired result by using the most simple and direct means is often the most essential part of the problem, especially when even a single unnecessary part would greatly increase the manufacturing cost. The interesting feature of the mechanism to be described is that it contains only two gears which give the same speed ratios between the driving and driven members as are ordinarily obtained with a train

of four gears. One application of the mechanism is to clock-work or time-pieces. The hour hand of an ordinary clock is driven through a compound gear train, called "dial gears," which serves to turn the hour hand one-twelfth revolution while the minute hand makes a complete revolution. The two-gear mechanism which is shown in Fig. 9, enables the

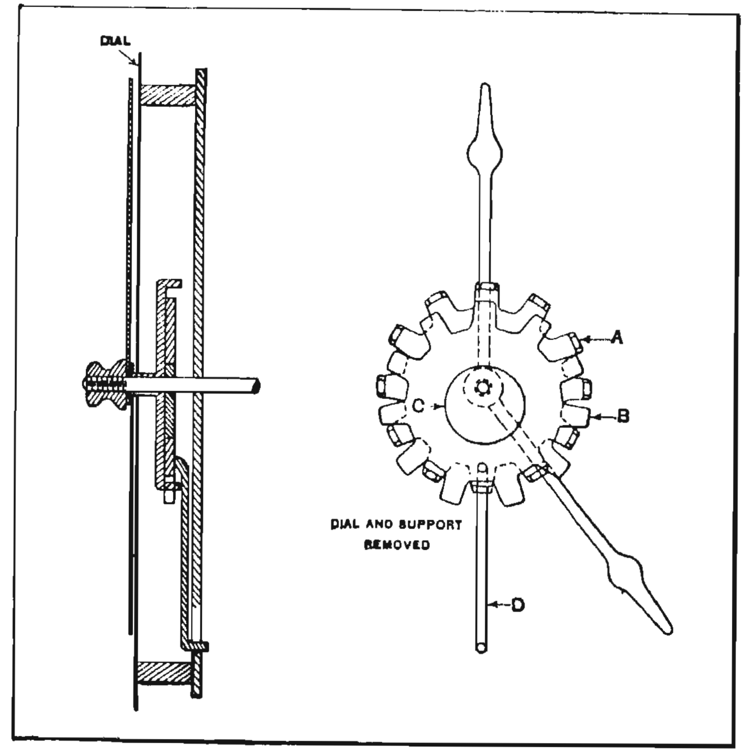


Fig. 9. Two-gear Clock Mechanism of 12 to 1 Ratio

same 12 to 1 ratio between the minute and hour hands to be obtained.

The hour hand is attached to a gear *A* having 12 teeth, the ends of which are bent so that they lie parallel to the axis, similar to a crown gear. Within gear *A* there is another gear *B* which has 11 straight or radial teeth. This inner gear is mounted on an eccentric *C* attached to the shaft for revolving the minute hand. As this shaft turns, the eccentric causes the axis of gear B to revolve, but the gear does not turn about its own axis, as there is a rod D which prevents such movement. The lower end of this rod is bent at right angles and engages a slot, thus allowing the planetary movement of gear B, but preventing it from revolving about its own axis.

Each revolution of the minute-hand shaft and eccentric, causes the teeth of gear B to withdraw from whatever tooth spaces of gear A they happen to be in mesh with and engage the next successive tooth spaces; consequently, gear A is advanced an amount equal to one tooth or one-twelfth revolution for each complete turn of the eccentric and minute-hand shaft. In this way, the desired speed ratio between the minute and hour hand of a clock is obtained. Of course, this speed ratio might be varied to suit different requirements. This mechanism is not intended only for clockwork, but can be applied wherever a simple reducing device is required in mechanisms of the type used in connection with counting and recording instruments.

The relative positions of the driving and driven parts during any fractional part of a revolution depend upon the length of the rod D attached to gear B. If this rod were of infinite length, a uniform rotation of the eccentric shaft would cause the driven part also to revolve at a uniform rate; but with rods of ordinary length, the motion of the driven part (which, in the case of a clock, is the hour hand) is accelerated and then retarded as it moves from one figure to the next on the dial. If rod D were dispensed with and gear B prevented from rotating by a pin attached to some point within the gear teeth, and engaging a fixed slot as in the case of rod D, the driven part would remain nearly stationary during a large part of the driver's revolution, and the movement from one division point to the next would be quite rapid. This rapid motion would be desirable for counting mechanisms in order to have the indicating hand opposite the numbered divisions, except when rapidly moving from one position to the next successive number on the dial.

Gearless Variable-speed Transmission.— There are many machines and mechanical units that varying circumstances make it desirable to be able to drive at a barely perceptible speed, an intermediate speed, or a high speed. A patented variable-speed unit of this type is so arranged that while its operation is entirely mechanical, any speed from zero to maximum is obtainable without the use of a single gear. The changes in speed are made without shocks or undue stresses.

Fig. 10 shows an assembly drawing of the unit. The drive is delivered to the transmission by shaft A on which tight and loose pulleys are mounted. The shaft, of course, may also be driven direct by motor or other means. Two cranks on shaft A impart motion through connecting-rods C to oscillate a shaft D on each side of the transmission. At the forward end of each of these shafts is a crank E in which there are two blocks F which are adjustable along grooves in the crank. It will be seen that the blocks in each crank are connected by means of the link G to another crank H which is bushed on the shaft B. The position of blocks F on cranks E is adjustable by turning handwheel J to raise or lower screws K, which are each connected by means of a link to the respective blocks F. The pairs of blocks on the opposite sides of the transmission are adjusted in unison by sprocket L, mounted on the same nut as handwheel J, which drives sprocket M through a chain. Instead of these sprockets and chain, screws K are sometimes connected by means of a shaft and helical gears. Then, too, instead of handwheel J being mounted directly on the nut of one of screws K, the handwheel may be placed in a convenient position for the operator, and connected to the nut by a long shaft and bevel gears.

It will be evident that at each revolution of shaft A, cranks E are rocked once forward and backward, with the result that cranks H oscillate similarly. The angular movement of cranks H becomes less as blocks F are moved from the extreme end of cranks E toward the axis of the cranks. When the center of the blocks coincides with the axis of the cranks, no movement is imparted to links G and cranks H. Integral

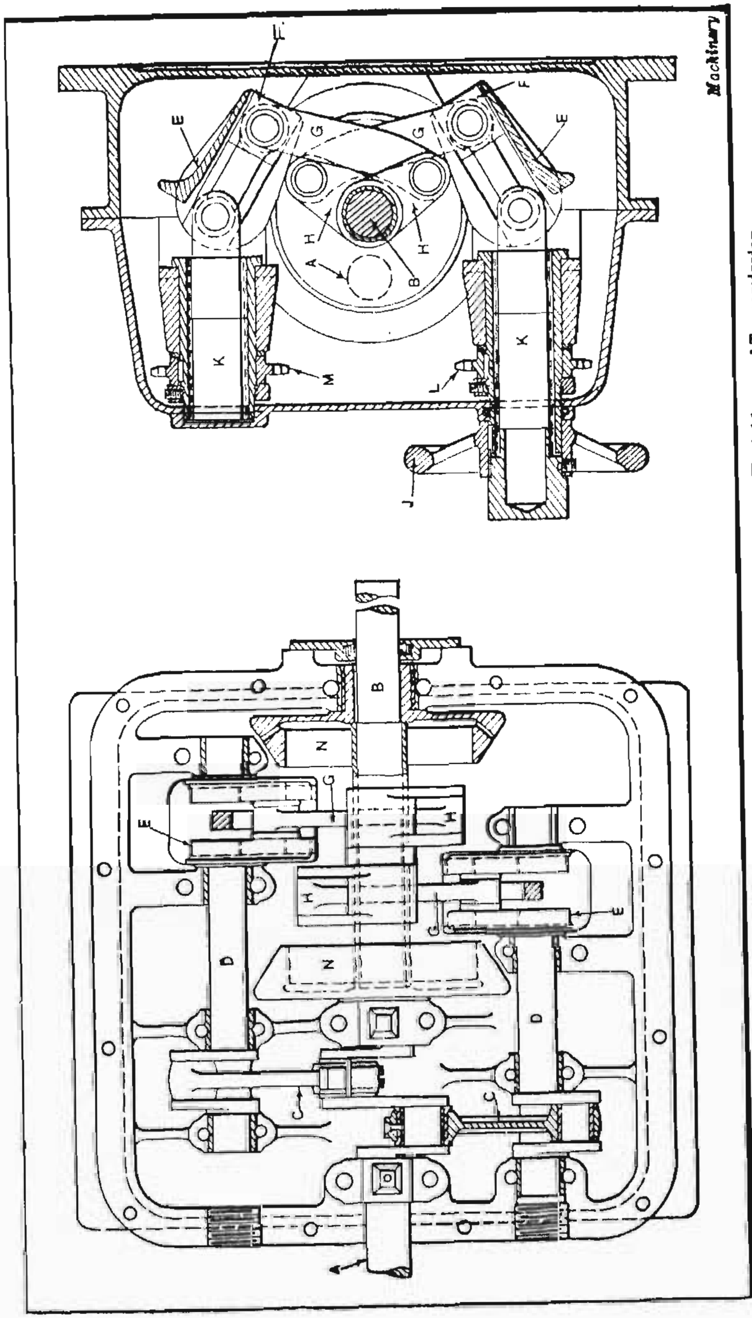


Fig. 10. Assembly Drawing Showing the Construction of the Variable-speed Transmission

with each crank H are two eccentrics which impart radial movements to two impeller parts. These serve as a quick-acting clutch to transmit the drive to drums N when they are expanded. Drums N are both keyed to shaft B to drive it when they are rotated, both drums being turned in the same direction. The two drums are engaged simultaneously so as to obtain a double drive. The impeller devices consist of an ingenious design in which rollers are employed with the double eccentrics to give the quick locking and unlocking action.

From the foregoing description it will be obvious that with a machine equipped with this transmission in the main drive, it is possible to obtain any feed or speed between zero and maximum of the tool or work. With a speed of, say, 1150 revolutions per minute of shaft A, the maximum speed of shaft B would be about 800 revolutions per minute. The housing is oil-tight, dust-proof, and filled with lubricant which is supplied to all parts by the splash obtained from the movement of the cranks, etc.

Frictional Speed-changing Devices.— Friction gearing of various forms is applied to some classes of machinery as a means of obtaining speed changes. The frictional type is simple in design and has the further advantage of providing very gradual speed changes. If a definite relation, however, must be maintained between the driving and driven members, the frictional transmission is not suitable, but, in some cases, the fact that it is not positive and tends to slip when subjected to excessive loads is a desirable feature, as it serves to protect the driven mechanism against excessive stresses.

Fig. 11 shows a type of frictional speed-changing mechanism which has been quite generally used, the details of construction being modified somewhat, owing to variations in the amount of power to be transmitted and other factors affecting the design. The particular arrangement referred to is applied to a running-balance indicating machine. The motor which drives the machine revolves the leather-faced driving disk A which is in contact with a steel wheel B. The vertical shaft passing through the driven wheel transmits motion to a horizontal shaft (not shown) at the top of the machine, which, in turn, revolves whatever part is to be tested for running balance. Variations in the speed of the work are obtained by changing the position of wheel *B* relative to the axis of the driving disk *A*. The adjustments of wheel *B* are controlled by a hand lever provided with a notched quadrant for holding it in a given position. This hand lever is connected with the slide of wheel *B* by link *C*. A reversal of

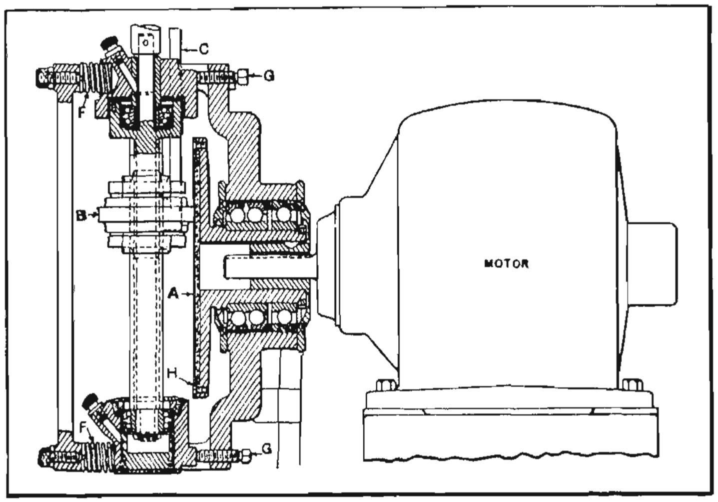


Fig. 11. Speed-changing Mechanism of Friction Disk and Wheel Type

motion is obtained by simply shifting wheel *B* to the opposite side of the axis of the driving disk. The wheel is held against the leather-faced disk with sufficient pressure by means of springs *F* which are provided with screws for varying the compression. If the leather disk becomes flattened out or thin from wear, the wheel *B* may be adjusted inward by means of stop-screws *G*. The leather disk is held in place by a retaining ring *H*. The adjustments for changing the speed should only be made when the driving disk is running.

Friction Disk and Epicyclic Gear Combination for High Velocity Ratio.—A very high velocity ratio or great reduc-

tions of speed, as well as extremely small variations of speed, may be obtained by the mechanism to be described. This mechanism (see Fig. 12) is a combination of friction disks and a train of epicyclic or differential gearing. The two disks *D* and *E* are free to revolve upon the vertical shaft *C*, and the hubs of these disks form the bevel gears *F* and *G*. Between these two bevel gears are the additional gears *T* and *J* mounted on pin *H*, which is attached to shaft *C*. The disks *D* and *E* are in frictional contact with wheels *N* and *O*, and their position is regulated by screw *K*, which is rotated through disks *L* and *R*. If wheel *N* is revolved and disks *D* and *E* are equidistant from the axes of wheels *N* and *O* (as shown in the illustration), both disks will revolve at the same speed, but in opposite directions. As gears *F* and *G* also rotate at the same speed, the intermediate gears *T* and *J* merely revolve idly upon pin *H*, which remains in one position. Any change in the position of disks *D* and *E* relative to the wheels *N* and *O* will result in reducing the speed of one disk and increasing the speed of the other one; consequently, gears *T* and *J* begin to advance around whichever gear *F* or *G* has the slower motion, so that pin *H* and shaft *C* revolve in the same direction as the more rapidly revolving gear. If disks *D* and *E* are only moved a small amount from the central position, the differential action in the gearing and the motion of shaft *C* will be at a very slow

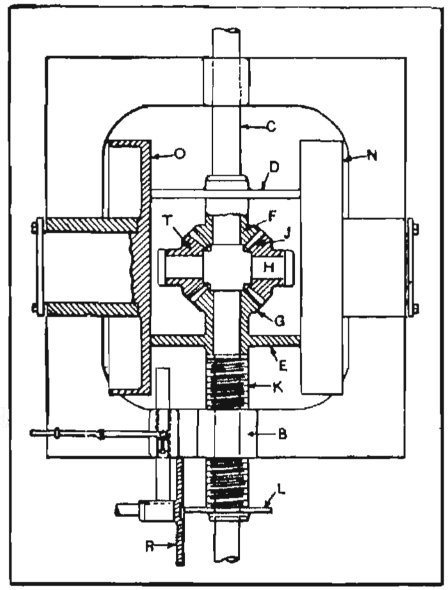


Fig. 12. Combination of Friction Disks and Epicyclic Gear Train for Obtaining Great Reduction of Speed

rate. The direction of rotation may be changed by moving the disks upward or downward relative to the central position shown by the illustration.

Friction Speed-changing Mechanism of Disk, Ball and Roller Type. — A speed-changing mechanism of the friction type which was designed for use in naval nautical instruments,

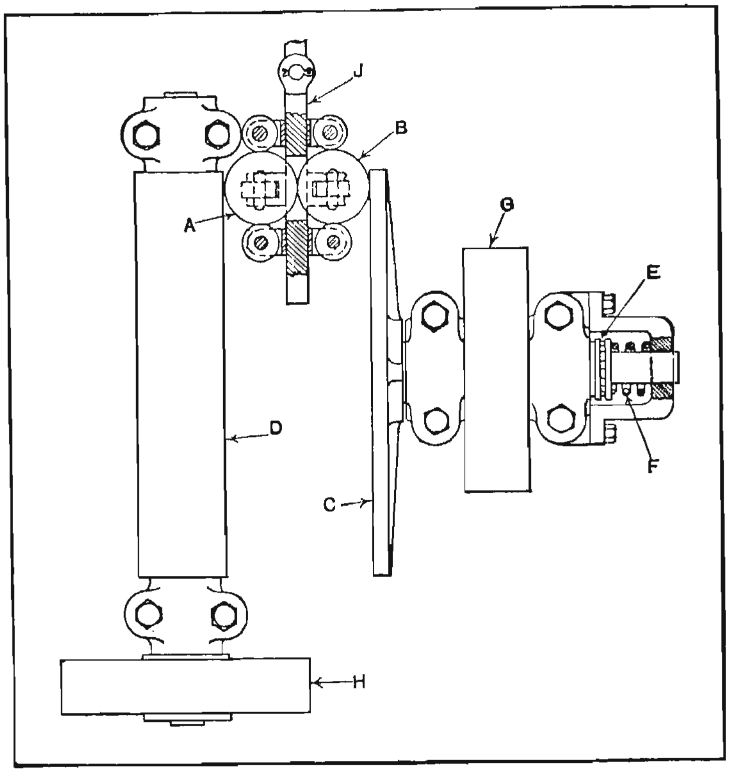


Fig. 13. Speed-changing Mechanism of Friction-disk and Ball Type

is shown in Fig. 13. In this application of the device the power transmitted is very small, say about 0.01 horsepower. It has also been applied to mechanical fuel stokers, the power transmitted in this service being from one to two horsepower.

For transmitting up to two horsepower, excellent results may be obtained. This transmission is particularly useful when frequent changing of speed is required under conditions

that would preclude the use of gearing for the purpose. A remarkable feature of this device is that the force required to effect a change from full speed in one direction to full speed in the reverse direction is so small as to be negligible, and this operation is unaccompanied by any jar or shock.

As will be noted from the illustration, the device consists essentially of two balls *A* and *B* revolving in contact with each other between a driving disk *C* and a driven roller *D*. The shaft of the driving disk is mounted in a ball thrust bearing *E* back of which there is a spring *F*. The disk may be driven by a gear or pulley *G* or it may be connected directly to an electric motor or gasoline engine. The power is transmitted from the driven roller by pulley *H*. The two balls employed to transmit the power from disk *C* to roller *D* are mounted in a carrier *J* in which they are loosely held by small rolls mounted in brackets. The carrier is so mounted as to be capable of movement across the face of the disk.

The balls *A* and *B* are capable of two movements; that is, they may rotate on their own centers and also roll across the disk and roller between the two latter members. The disk, the balls, and the roller are all made of hardened steel. When the balls are near the edge of the disk the roller runs at its highest speed. Upon moving the ball carrier nearer the center of the disk, the roller gradually loses speed, until at the center it will cease to rotate. As the carrier continues to move, passing the center, the roller rotates in the opposite direction.

The pressure of the spring *F* is not excessive, and while the tractive force between the balls would appear at first to be small, there is actually no slippage in practice. This is in accordance with the law of friction which states that the amount of friction is independent of the amount of surface in contact but depends entirely upon the pressure. An illustration of this law may be observed in the case of locomotive driving wheels, where the tractive force of, say, six points of contact with the rails, is sufficient to draw a heavy train.

Friction Roller Between Cones. — Many speed-changing mechanisms of the friction type have opposing cones which are connected by some intermediate member that may be adjusted to vary the speed. The use of an ordinary belt has already been referred to. Fig. 14 shows an arrangement for regulating the speed of a driven shaft, by changing the position of a wheel *A* placed between the driving cone *B* and the driven cone *C*. These two cones are made of cast iron and the bearing surface of the intermediate wheel is formed of leather disks held in place between two flanges or collars. This particular mechanism is used for varying the feeding movement of a cold-metal saw. The handle *D* connecting with a screw is used for controlling the position of the intermediate wheel and the rate of speed. A dial at *E* shows the rate of feed per minute, this dial being connected through shaft *F* and a gear at the lower end with a rack on the adjustable member, so that any change in the position of the wheel is indicated by the dial. The lower friction cone is held in contact with the wheel by means of a spring *G*, the tension of which may be regulated by lever *H*. This lever is provided with graduations so that the same tension as well as the rate of feed per minute may be duplicated.

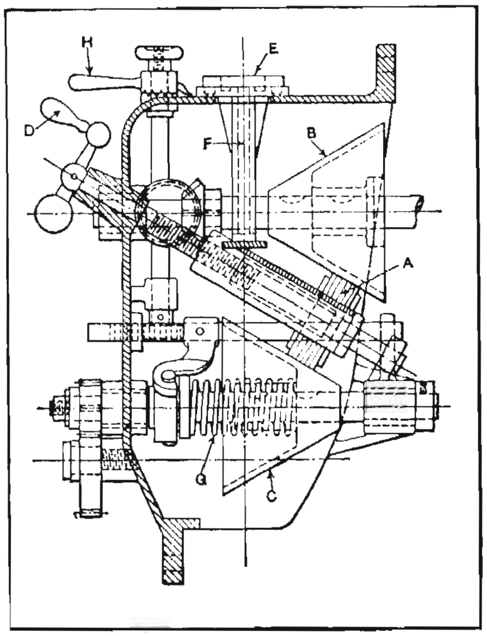


Fig. 14. Friction Cones and Intermediate Wheel for Varying Speeds

Band or Ring Between Cones. —Another method of transmitting motion from a driving to a driven cone is shown in Fig. 15, which illustrates the Evans friction cones. The two

cone-pulleys are not directly in contact with each other, but bear against a band or ring of leather which serves to transmit the motion. The speed of the driven cone is varied by simply shifting this leather ring so that it bears against a larger or smaller part of the cones. If cone *A* is the driver, the speed of cone *B* would be gradually increased if belt *C* were shifted toward the right, since the practical effect of this shifting movement is to increase the diameter of the driving pulley. This mechanism is used ordinarily as a variable-speed counter-shaft. There are two general methods of

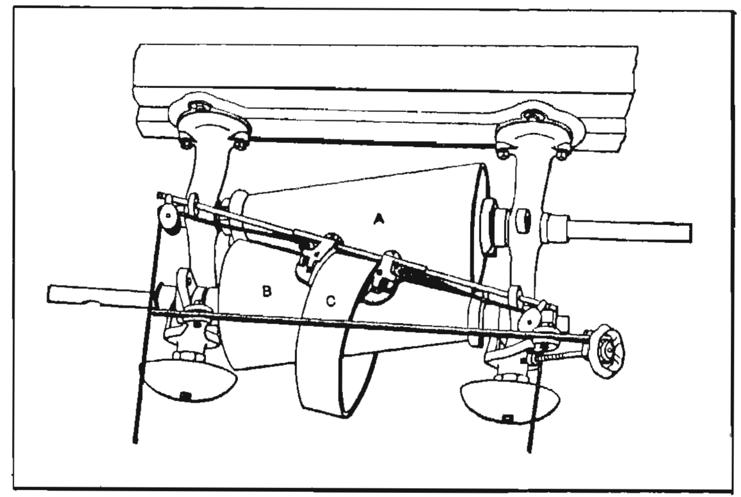


Fig. 15. Friction Cones Which Transmit Motion Through Adjustable Leather Ring or Belt.

starting or stopping the driven members. Some friction cones are so arranged that the leather ring is shifted to a parallel part of the cones for disengaging the drive, and others are so designed that one cone is raised and lowered by the shifting lever, thus starting and stopping at the same speed.

Spherical Rollers Between Disks. — The necessity for accurately adjusting or controlling the speeds of driven shafts has resulted in the development of a variety of variable-speed mechanisms. Among these is the friction-driven type of variable-speed mechanism. The design shown in Fig. 16 has a

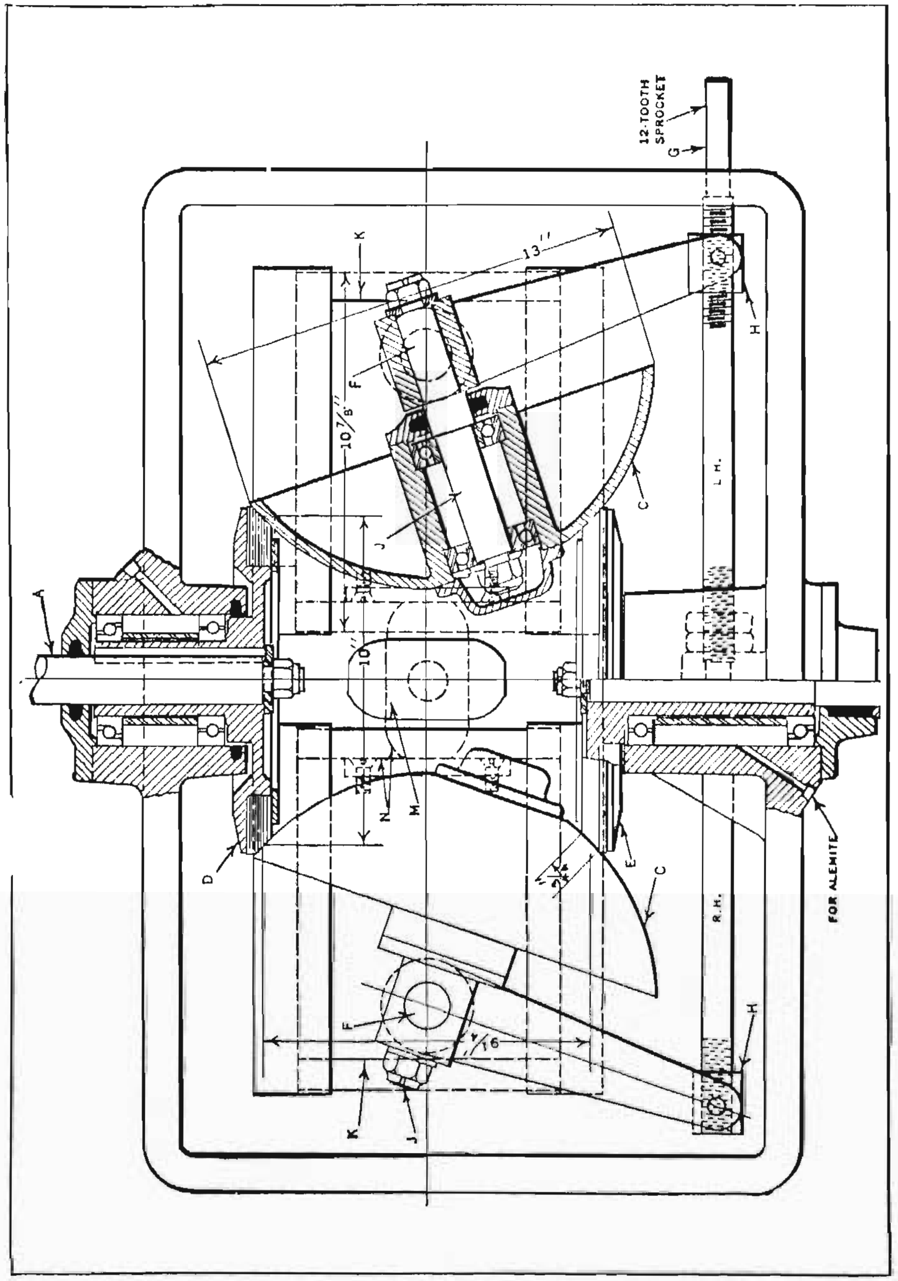


Fig. 16. Variable-speed Friction Drive Consisting of Spherical Rollers Between Driving and Driven Disks

maximum driving ratio of 4 to 1, and provides an accurate means of adjusting the speed of the driven disk E and its shaft within the required range of from 2 to 0.5 revolutions per minute when the driving shaft A revolves at a constant speed of 1 revolution per minute. Like most friction drives, this design is employed where the load is light.

The drive consists primarily of two spherical members C which engage the friction disk D of the driving shaft and the disk E secured to the driven shaft. The variation in speed is obtained by rotating the spherical members C about the points F. When the spherical members are in the positions shown, the driven shaft will be rotated at approximately its slowest speed, as the driving disk D makes contact with the spherical members at their outer edges, while the driven disk E makes contact near the centers of the members C, where the surface speed is comparatively low.

The operator adjusts the speed by rotating the shaft G, a sprocket and chain arrangement (not shown) being employed for this purpose. The right- and left-hand threads on shaft G, together with the nuts H attached to the arms of the bearing shafts J, provide means for accurately positioning the spherical members C to give the desired speed.

The members C are mounted on slides K. These slides are connected by springs which serve to keep the friction members in driving contact with the disks D and E. The friction drive can be disengaged at the will of the operator by means of a lever outside the housing, which controls the cam M. When the cam is turned, the slides K are forced apart, thus disengaging the friction driving members and causing the driven shaft to stop. The dotted lines at N show the position of the cam and slides when the drive is stopped.

Concave Friction Disks and Inclined Wheel. — The frictional variable-speed transmission shown in Fig. 17 is an example of the type having annular concave frictional surfaces engaged by an intermediate wheel the inclination of which is varied for changing the speed. The principle upon which the device operates is illustrated by the diagram at the left. The two disks having annular concave surfaces are rotated from some source of power and run loose on shaft A which is driven at a variable speed. The intermediate wheel D is pivoted at O to arm B, so that it can be inclined as indicated by the dotted lines. The drive to shaft A is transmitted through arm B. When wheel D is parallel to shaft A, as shown in the illustration, and the two disks G and C are revolving in opposite directions at the same speed, wheel D will simple revolve about pivot O, and arm B and shaft A will remain stationary. If wheel D is inclined, however, as

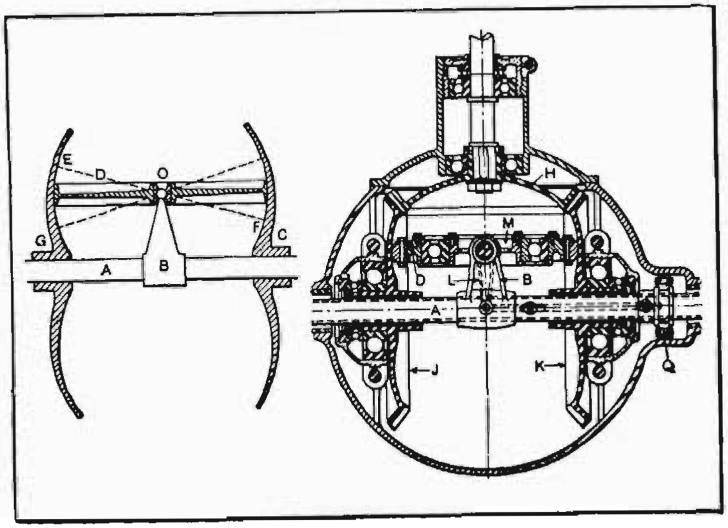


Fig. 17. Variable-speed Transmission Having Annular Concave Surfaces and Inclinable Friction Wheel

indicated by the dotted line EF, the contact surface at E will be revolving at a higher circumferential speed than the surface F on disk C; consequently, pivot O, arm B, and shaft A will be given a rotary motion, the rate of which depends upon the angularity of wheel D. The greater the angularity, the greater will be the difference in the diameter of the contact surfaces of disks G and C and the higher the speed of shaft A. By inclining wheel D in the opposite direction, the rotation of shaft A can be reversed.

A variable-speed mechanism designed on this principle is shown at the right of the diagram in Fig. 17. A bevel gear H mounted on the end of the driving shaft revolves the two bevel gears J and K mounted on shaft A, which is the driven member. These bevel gears J and K have annular concave surfaces which engage the cork surface of wheel D. This wheel revolves on an annular ball bearing, the inner race of which is attached to ring M pivoted on a stud carried by arm B. The angular position of wheel D is controlled by a lever L integral with the pivoted ring M. This lever is connected with ring Q which is engaged by a forked lever similar to the form used for shifting clutches.

An objection to variable-speed mechanisms based on this principle is that the variation of speed does not change the torque, so that, even though there is considerable speed reduction, the torque will not be proportionally greater, because the limiting factor for the torque is the frictional adherence between the driving and driven contact surfaces, and this frictional resistance is independent of the speed at which the shaft A is running; consequently, while variable-speed devices in general are of such construction that the torque increases when the speed decreases, in the present case the speed is variable, while the torque remains constant. As the main feature of variable-speed devices is often not the variation of speed as much as the increased torque obtained by a decrease in speed, the objection referred to is one of great importance.

Materials for Friction Gearing.—In the selection of material for the driving member of friction gears, good frictional qualities combined with a reasonable degree of durability are essential. In order to determine the relative merits of different kinds of friction materials, tests were conducted by Professor Goss of the University of Illinois. The driving and driven wheels used for these tests were each 16 inches in diameter and were mounted on parallel shafts and run with the peripheries or edges in contact. The driving wheels were made of the fibrous materials referred to later and were 1¾ inches wide; the driven wheels were ½ inch wide and made of cast iron, aluminum and type metal. It was found that leather fiber is exceptionally strong and has a high coefficient of friction when in contact with cast iron or an aluminum alloy — materials commonly used for the driven member. Straw fiber has a somewhat lower frictional value, and is not so durable as leather fiber, but nevertheless is satisfactory and has the advantage of being obtained readily. Tarred fiber is exceptionally strong, but its frictional coefficient is comparatively low. Sulphite fiber has the highest coefficient of any of the materials listed, but it is the weakest. Leather is inferior to plain straw fiber as to both frictional qualities and strength. A summary of other results of these tests follows.

Power Transmitted by Friction Gears. — The power transmitting capacity of friction wheels of given size and running at a given speed depends upon the pressure of contact and the coefficient of friction. Since the life of the fibrous or leather driving wheels depends upon the contact pressure (the diameter decreasing due to a yielding of the material as the pressure becomes excessive) the allowable working pressure is determined with reference to durability. Allowable pressures per inch of face width should be approximately as given in

Allowable Working Pressures and Coefficients of Friction

Material	Allowable Working Pressure per Inch	Coefficient of Friction when Driver is in Contact with	
		Cast Iron	Aluminum
Leather fiber	240	0.31	0.30
Straw fiber	150	0.26	0.27
Tarred fiber	240	0.15	0.18
Sulphite fiber	140	0.33	0.32
Leather	150	0.14	0.22

the accompanying table, which applies to the 16-inch wheels previously mentioned. This table also includes working values for the coefficients of friction. The frictional coefficient for the wheels tested approaches its maximum value when the slip between the driving and driven wheel amounts to 2 per cent; the coefficient diminishes when the slip exceeds 3 per

cent. As a general rule, the percentage of slip, according to the results of the tests previously referred to, should not be less than 2 per cent nor more than about 4 per cent.

The number of horsepower transmitted by friction gearing may be determined by the following general formula in which H = the number of horsepower; D = diameter of driving wheel in inches; N = revolutions per minute; P = allowable working pressure in pounds per inch of face width (see table); W = face width in inches; f = coefficient of friction (see table).

$$H = \frac{\pi DPWNf}{33{,}000 \times 12}$$

For a given coefficient and contact pressure, this formula may be simplified by determining the value of the expression $\frac{\pi Pf}{33{,}000 \times 12}$ and inserting this value X in the formula, $H = DNWX$. The fibrous wheel should always be the driver to avoid wearing a flat spot on it in case the driven wheel stalls, and as rigid a support as possible is essential.

Driving Disk Engaging Side of Driven Wheel. — When the driving disk engages the side of the driven wheel (in order to provide for speed changes and possibly a reversal of rotation) pure rolling action is not obtained because the driver makes contact with the driven disk at various diameters; consequently the velocity of the driven disk at one side of the driver differs from the velocity at the other side where contact is at a smaller or larger radius, depending upon the side. In order to avoid an excessive amount of slippage between the driving and driven disks, the ordinary running positions of the driver should be such that the minimum distance from the center of the driver face to the center of the driven wheel, will not be less than twelve times the width of the driver face. For example, if the driver face is ½ inch, the minimum distance to the center of the driven disk should preferably be 6 inches. When the driver is closer than this minimum distance, the coefficient of friction is reduced and also the power-transmitting capacity.

Obtaining Contact Pressure. — The method of applying contact pressure is adapted to conditions, but, in general, the lever-operated eccentric box or thrust box is commonly used; it is a simple method for giving hand or power control. In some cases, more elaborate devices are used. The pressure may be positively applied and it may be made to vary automatically as the load increases or decreases. As friction is essential to the operation of this type of gearing, care should be taken to prevent any great reduction of the driving power by the accumulation of grease or other foreign matter on the friction surfaces. Rigid support for the friction wheels and the maintenance of a good contact between the working surfaces are also of importance. Friction gearing is not a suitable form of transmission where it is essential to maintain a prescribed relation between driving and driven parts of a mechanism throughout an entire cycle of operations. In some cases, however, a transmission which is not positive is preferable in that it constitutes a safety device and prevents the transmission of shocks or an excessive amount of power to parts of a mechanism which might thereby be injured. Friction gearing is also very simple in design, and operates smoothly and quietly.

Multiple-disk Type of Speed-changing Mechanism. — The variable-speed mechanism shown in Fig. 18 is an ingenious design used on certain cylindrical grinding machines for changing the rotary speed of the part being ground and also the rate of the table traverse. Three levers grouped around a dial at the front of the machine are used for controlling the mechanism. The position of lever *A* governs the rotary speed of the work, and another lever in front of the circular dial (not shown in the illustration) serves to change the rate of the table traversing movement. These changes of work speed and table traverse are entirely independent. The long lever *R* is used for starting and stopping the rotation of the work and the traversing movement of the table simultaneously. The mechanism is driven from a driving shaft which runs at a constant speed and connects with coupling *B*. The sprocket

C is connected to the reversing mechanism and drives the table traverse. Another sprocket (not shown) is connected by a pair of silent chains and a splined shaft, with a driving member for the headstock.

The mechanism operates as follows: The shaft *F* carrying coupling *B* drives shafts *G* and *H* at a constant speed through spur gearing. The shafts *G* and *H* carry a series of hardened steel disks mounted on square portions of the shafts. These

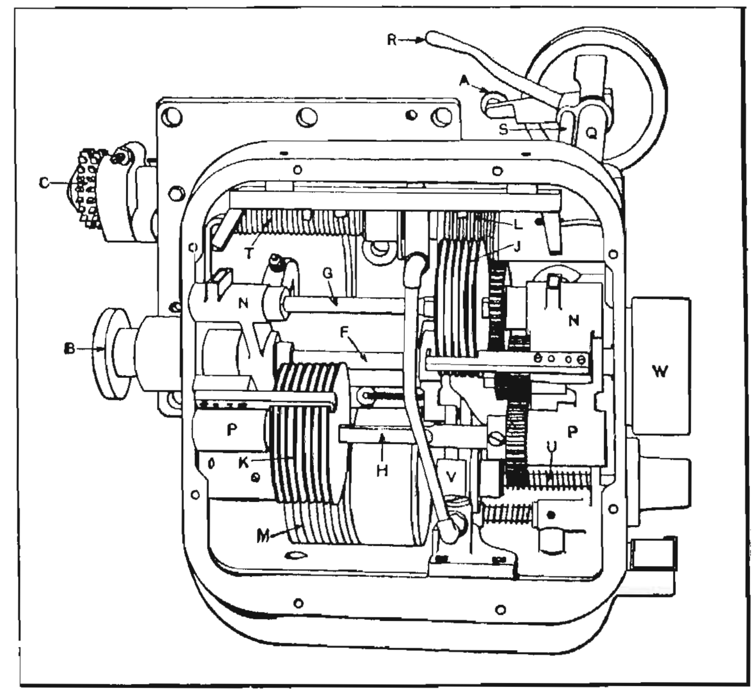


Fig. 18. Multiple Friction Disk Type of Speed-changing Mechanism

disks *J* and *K* are ground slightly convex and each group of disks intermeshes with another group or series of hardened steel disks *L* and *M*. Each of these driven disks has a rim at the periphery so that the point of contact with the driving disk is always at the outer edge. The shafts *G* and *H* are mounted in swinging brackets *N* and *P*, both of which pivot on shaft *F*, thus allowing the position of disks *J* and *K* to be varied relative to the disks L and M. If the convex disks J are swung towards the recessed disks L, the surfaces of disks J, which actually do the driving, decrease in radius and, consequently, the speed of disks L and their shaft also decreases. The lever A controls the position of bracket P and the speed of the headstock, whereas the lever at the front of the dial (not shown) controls bracket N and the feeding movements of the table. Motion is transmitted to these brackets through bevel pinions meshing with segment gears on the brackets.

With this mechanism, slight variations in speed may be obtained while the machine is in motion. When lever R is shifted for stopping the machine, a cam at the end of shaft S operates a lever which relieves the pressure applied to disks L and M by the springs shown at T and U. This lever also applies brakes which quickly stop the table and headstock. When the lever is raised for starting the mechanism, the disks L and M grip the intermeshing disks J and K, and the driven members are started without shock, the action being very similar to the well-known multiple-disk friction clutch. A plunger pump at V pumps oil from the bottom of the case to a distributor at the top which lubricates the entire mechanism.

Governors for Speed Regulation. — When the regulation of speed is automatically controlled, some form of governing mechanism of the centrifugal type is commonly employed. Many of the governors used on steam engines depend for their action upon the effect of centrifugal force on a rotating element. In the case of a "fly-ball" governor, weights or balls attached to pivoted levers are revolved by the engine and if the speed increases above normal, the balls or weighted levers move outward from the axis of rotation, owing to the increase in centrifugal force. This change in the position of the revolving balls may be transmitted through suitable connecting levers and rods to a valve which partly closes, thus reducing the steam supply. When a governor of this type is applied to a Corliss engine, the release of the steam valves and the point of cut-off is controlled directly by the governor. Most governors of the fly-ball type have one or more springs which

tend to resist the outward movement of the revolving balls.

The inertia or centrifugal-inertia governor, which is used so extensively, is attached to the fly-wheel and regulates the speed by varying the position of the eccentric or crankpin that operates the valve. The general principle upon which this type of governor operates is illustrated by the design shown in Fig. 19. This particular governor has an inertia bar A with enlarged ends to increase the weight at the ends. This bar is pivoted at B where there is a roller bearing to reduce the frictional resistance. The eccentric C is attached to the inertia bar and it has an elongated hole or opening to permit movements relative to the crankshaft. Directly opposite the eccentric is a third weight D, which balances the effect of gravity on the eccentric. A heavy coil spring E is attached to the inertia bar. A rod F is pivoted to the bar on the opposite side of bearing B and is connected to a loose-fitting piston in the oil dashpot G.

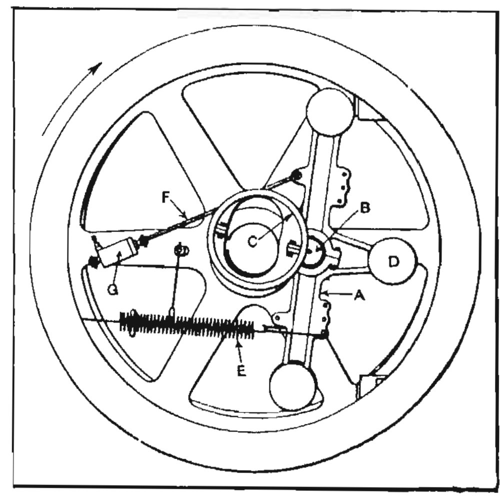


Fig. 19. Centrifugal-Inertia Type of Engine Governor

The flywheel revolves in the direction shown by the arrow and speed variations cause a slight movement of the inertia bar about its bearing in one direction or another, thus changing the position of the eccentric, which changes the point of cut-off. If the speed increases, the inertia bar lags behind momentarily and the steam is cut off earlier during the stroke because the eccentric swings inward and shortens the travel of the valve. If a sudden increase of load should cause the engine to run slower, lever A, as a result of its inertia, would tend to continue running at the faster speed, which would swing the lever forward about bearing B in the direction of rotation, thus increasing the valve travel and admitting more steam to the cylinder by delaying the point of cut-off. The spring end of the inertia bar is the heavier and the speed of rotation depends entirely upon the equilibrium between the centrifugal force acting upon the inertia bar and the tension of the spring, while the actual movement of the governor parts is effected by the inertia of the weighted end of the bar. The sensitiveness of the governor may be varied by adjusting a by-pass valve upon cylinder G. Other governors of this general type vary in regard to the form of the weighted lever and the arrangement of springs or other details. The inertia type is preferable to the purely centrifugal design for engines subjected to sudden and decided load changes.

CHAPTER XII

DIFFERENTIAL MOTIONS

When a motion is the resultant of or difference between two original motions, it is often referred to as a differential motion. The differential screw is a simple example of a motion of this kind. This is a compound screw from which a movement is derived that is equal to the difference between the movements obtained from each screw. The diagram A, Fig. 1, illustrates the principle. A shaft has two screw threads on it at e and f, respectively, which wind in the same direction but differ in pitch. Screw f passes through a fixed nut and screw e through a nut that is free to move. The motion of the movable nut for each revolution of the screw equals the difference between the pitches of the threads at e and f.

This combination makes it possible to obtain a very slight motion without using a screw having an exceptionally fine pitch and a weak thread. Another form of differential screw is shown at B, which illustrates a stop that enables fine adjustments to be obtained readily. The screw bushing g is threaded externally through some stationary part and is also threaded internally to receive screw h which is free to move axially but cannot turn. Both screws in this case are right-hand, but they vary as to pitch. If bushing g has a pitch of $\frac{1}{32}$ inch or 0.03125 inch and screw h a pitch of $\frac{1}{36}$ inch or 0.02777 inch, one complete turn of g will advance screws h only 0.00348 inch ($0.03125 - 0.02777 = 0.00348$), because, as bushing g advances $\frac{1}{32}$ inch, it moves screw h back a distance equal to the difference between the pitches of the two threads. By turning the bushing only a fractional part of a turn very small adjustments may be obtained.

Differential Motion of Chinese Windlass. — The Chinese windlass shown by the diagram C, Fig. 1, is another simple example of a differential motion. The hoisting rope is arranged to unwind from one part of a drum or pulley onto another part differing somewhat in diameter. The distance that the load or hook moves for one revolution of the compound hoisting drum is equal to half the difference between the circumferences of the two drum sections.

The well-known differential chain hoist illustrated at *D* operates on the same general principle as the Chinese wind-

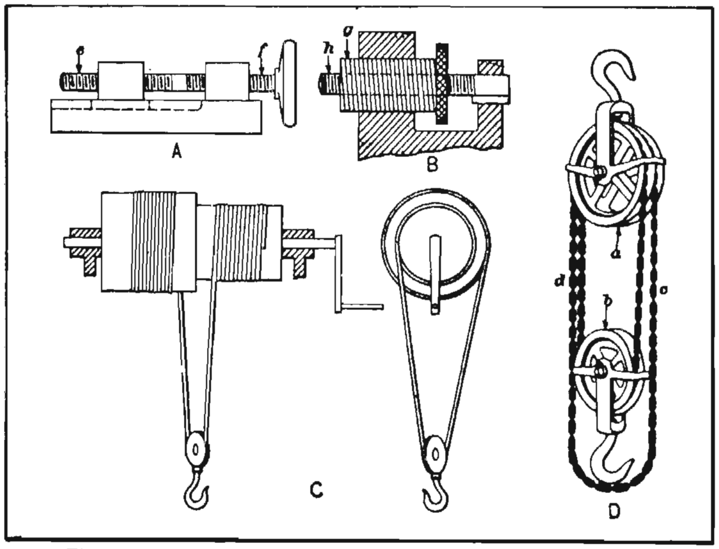


Fig. 1. (A and B) Differential Screws; (C) Chinese Windlass; (D) Differential Hoist

lass. The double sheave *a* has two chain grooves differing slightly in diameter, and an endless chain passes over these grooves and around a single pulley *b*. This pulley *b* and the hook attached to it is raised or lowered, because, for a given movement, a greater length of chain passes over the larger part of sheave *a* than over the smaller part. If the upper sheave is revolved by pulling down on the side *d* of the chain that leads to the groove of smaller diameter, the loop of chain passing around pulley *b* will be lengthened, thus lowering the pulley; the opposite result will be obtained by pulling down on chain *c* which leads up to the larger diameter of the sheave.

Differential Motions from Gearing. — Most differential motions are derived from combinations of bevel or spur gearing. The epicyclic bevel gear train illustrated by diagram *A*, Fig. 2, is applied to many mechanisms of the differential type, and its action under different conditions should be thoroughly understood. The shaft *a* has mounted on it two bevel gears *b* and *c* and an arm *d*. The arm is attached to the shaft and carries a pinion *e* which meshes with each gear and is free to revolve upon the arm. There are several conditions that can exist with a gear train of this kind.

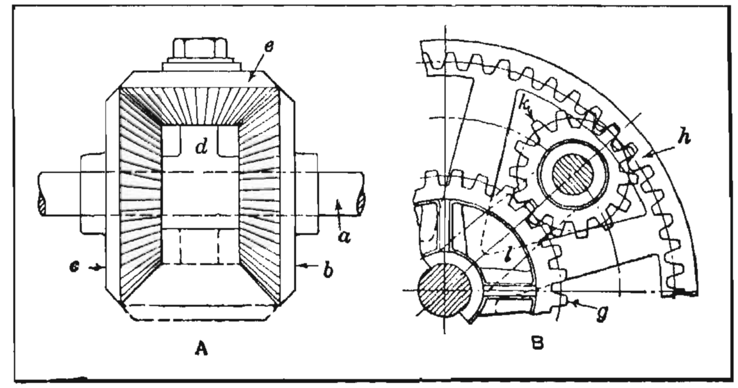


Fig. 2. Epicyclic Trains of Bevel and Spur Gearing

First, assume that gear *b* is stationary and *c* loose on the shaft. If the shaft and arm *d* is revolved, motion will be transmitted from arm *d* to gear *c*, through pinion *e*, and gear *c* will make two turns for every one of arm *d* and in the same direction as the arm. If gear *b* should rotate instead of being stationary, this motion, combined with that of the arm, would modify the motion of gear *c* and it would also make a difference whether gear *b* turned in the same direction as the arm or in an opposite direction.

Second, suppose the preceding conditions are reversed and one of the bevel gears *b* or *c* is revolved while the other gear remains stationary, and that arm *d* carrying the bevel pinion constitutes the driven element. With only one gear revolving, the arm will turn in a direction corresponding to that of the gear and at half its speed. If both gears rotate in the same direction at different speeds, the arm will follow in that direction and with a speed intermediate between the two. If the gears are driven in opposite directions at different speeds, the arm will follow the more rapidly moving gear, and if the speeds are equal, pinion *e* will revolve upon the arm, but the latter will remain stationary.

Third, assume that arm *d* remains stationary and gears *b* and *c* are loose on the shaft. If gear *b* is the driver ,the pinion *e* will simply transmit motion to gear *c* in the opposite direction, the three gears in this case forming a simple train with pinion *e* acting as the idler. The force tending to rotate arm *d* will be twice the force transmitted from gear *b* to gear *c*. A practical application of this last principle is found in the Webber differential dynamometer. The arm of this dynamometer which supports the scale pan and weights corresponds to arm *d* and is pivoted on a shaft carrying two bevel gears. On the arm and meshing with these two bevel gears are bevel pinions and the amount of power transmitted through this train of gearing is measured by the weights in the scale pan. The combination of gearing illustrated by diagram *A* usually has two or more pinions meshing with the bevel gears. In many cases, there are two pinions located diametrically opposite, as indicated by the full and dotted lines. The addition of other pinions, however, does not affect the action of the gearing.

Differential Spur Gearing. — The diagram *B*, Fig. 2, shows an arrangement of spur gearing which gives a differential motion. This combination consists of ordinary spur gear *g*, an internal gear *h*, and a pinion *k*. This pinion is free to turn on a stud that is attached to arm *l*. In the application of this gearing, there are three possible conditions. In the first place, the internal gear *h* may be stationary, and the gears *g* and *k* may revolve. Second, the arm *l* may be stationary, in which case either the internal gear *h* or gear *g* may be the driver. Third, gear *g* may be stationary and the motion be transmitted

in either direction between gear *h* and arm *l*. Fig. 3 shows a practical application of this gear combination. In this design, there are two intermediate pinions (corresponding to *k* in diagram *B*, Fig. 2) which are mounted on an arm and located diametrically opposite. This arm is keyed to the end of a shaft. The large internal gear is stationary and forms part of a casing enclosing the gears. The central gear is keyed to another shaft which is in line with the shaft carrying the pinion arm. This arrangement is simply used to obtain a

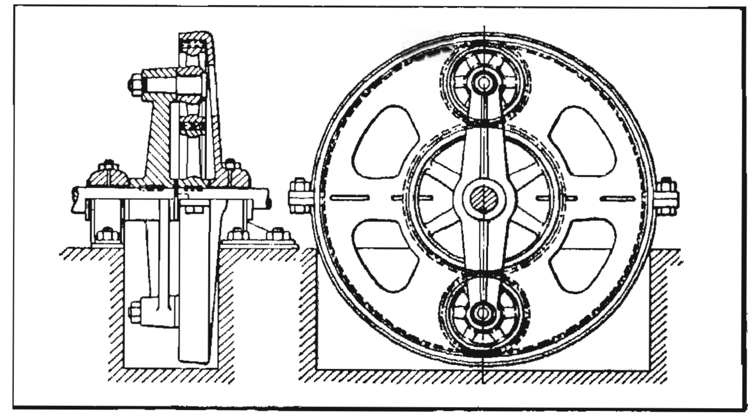


Fig. 3. Epicyclic Gearing for Obtaining Speed Reduction by Differential Motion

reduction of speed. The design is compact, although differential or epicyclic gearing, in general, is inefficient as a transmitter of power. Such gear combinations, however, have certain mechanical advantages, and they are often utilized by designers for a variety of purposes as indicated by the different mechanisms to be described.

Differential Motion between Screw and Nut Rotating at Different Speeds. — Variations of movement are sometimes obtained by the differential motion between a revolving screw and a nut which is rotating about the screw at a different speed. One application of this principle is illustrated by the variable-speed mechanism of a milling machine shown in Fig. 4. This mechanism is designed to increase the efficiency of a machine by accelerating the speed of the table when the cutters are not at work. The machine table moves rapidly up to the cutting point, then the speed is reduced while milling and, after the operation is completed, the table is quickly returned to the loading position so that the idle or non-cutting period is reduced.

This mechanism is located beneath the machine table *C*, which is traversed by a screw *D*, that passes through the plain bearings *E*, *F*, and *G*, mounted upon the base of the machine. The pinion *H* is confined longitudinally between bearings *E* and *F*, and it is splined to screw *D*, so that the latter must

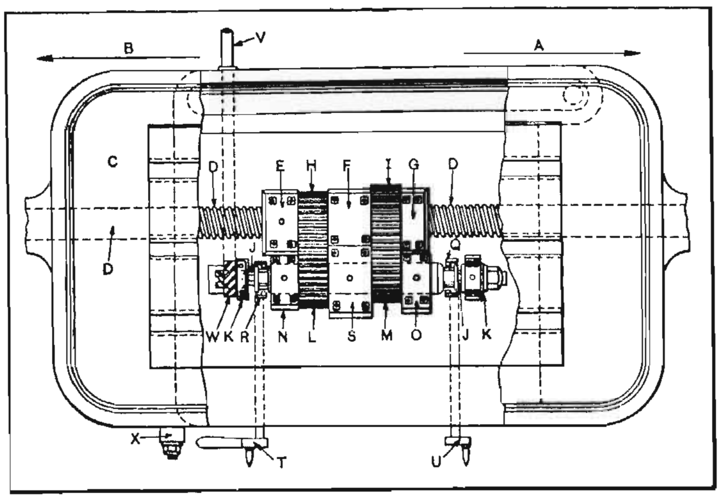


Fig. 4. Variable Feeding Mechanism Which is Partly Controlled by the Differential Movement Between a Revolving Screw and Nut

turn with the pinion but is free to slide in a lengthwise direction. The hole through gear *I* is threaded to fit screw *D* so that it is practically a nut and gear combined. The auxiliary shaft *J* supported in bearings *K* carries two pinions, *L* and *M*, which are loosely mounted upon the shaft. This shaft *J* is rotated continuously in one direction through spiral gears *W* from the driving shaft *V*. Within the housings *N* and *O* are clutch sleeves which encircle the shaft *J*. The sleeves are splined to the shaft, but are free to slide upon it, and they may be locked with teeth formed on pinions *L* and *M*. These

clutches are controlled by levers *T* and *U* at the front of the machine which are connected by the shafts shown, with the clutch shifting devices at *R* and *Q*. The action of the clutches is controlled automatically by adjustable stops located on the front of the machine table.

The clutch connecting with gear *L* is first engaged by hand lever *T*. The table then moves forward rapidly (in the direction indicated by arrow *A*) as gear *H* revolves screw *D* and causes it to turn through the gear nut *I* which is held stationary at this time. Just before the milling cutter begins to act upon the work, lever *U* strikes a stop, thus engaging the clutch with gear *M*. The gear nut *I* is then revolved in the same direction as gear *H* but at a slower speed, so that the forward movement of screw *D* is reduced, because of the differential action between the screw and nut. Both sets of gears continue to operate while the cut is being taken; when the milling operation is completed, another stop engages lever *T*, thus stopping the rotation of gears *L* and *H*. As the gear nut *I* continues to revolve about the screw, the movement of the machine table is reversed, since screw *D* is not rotating. The motion continues in the direction indicated by arrow *B* until a third stop to the right of lever *U* trips the latter, thereby stopping gear *I* and the table movement. The table is now in position for removing the finished parts and replacing them with others that require milling.

Differential Feeding Mechanism for Revolving Spindle. — The spindle of a horizontal boring, drilling, tapping, and milling machine is given a lengthwise feeding movement by the differential action between the revolving spindle and a revolving nut which engages a helical groove in the spindle. The spindle is driven by a large gear *A* (see Fig. 5) which connects with the back gearing of the machine. The hub of this gear has two keys which engage the splined spindle. The sleeve *B* on which gear *A* is mounted has gear teeth cut in one end which mesh with three planetary pinions *D* that engage one side of the double internal gear *E*. The other side of this internal gear meshes with pinions *N*. These pinions, in turn, mesh with gear teeth formed on the rotary nut L which engages directly with a spiral or helical groove cut in the spindle. A flange on this nut rotates between large ball thrust bearings, as shown, in order to take the end thrust in either direction.

When nut L rotates at the same speed as the spindle, the latter does not move in a lengthwise direction, but, by revolving nut L either faster or slower than the spindle, a feeding movement in one direction or the other is obtained. The rotation of nut L is regulated by the gearing at G. When the feeding movement is stopped, gear F, which carries the

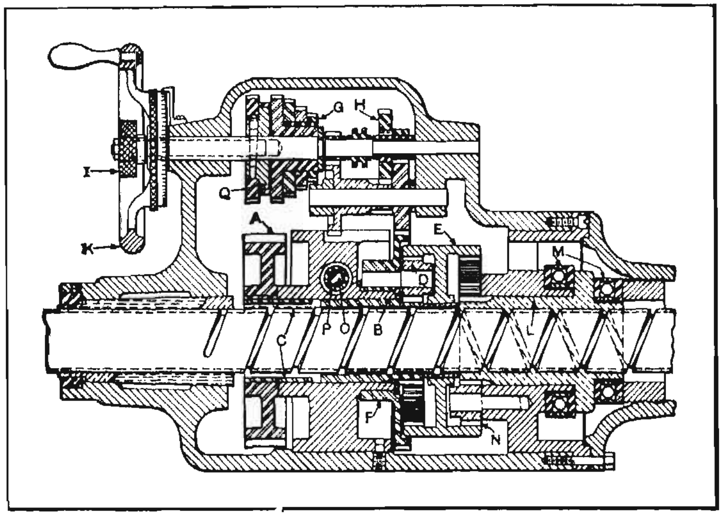


Fig. 5. Mechanism of Differential Type for Feeding Spindle in Lengthwise Direction

planetary pinions D, does not revolve and nut L rotates with the spindle, which, therefore, remains in the same longitudinal position. When gear F which is connected indirectly with the feed change-gears G is revolved by these gears, the nut L is revolved independently of the spindle and at a different rate of speed.

Application of Floating Lever Principle.— What are known as "floating" or "differential" levers are utilized in some forms of mechanisms to control, by the application of a small amount of power or force, a much greater force such

as would be required for moving or shifting heavy parts. Floating levers are commonly applied to mechanisms controlling the action of parts that require adjustment or changes of position at intervals varying according to the function of the apparatus subject to control. The initial movement or force may be derived from a hand-operated lever or wheel, and the purpose of the floating lever is to so control the source of power that whatever part is to be shifted or adjusted will

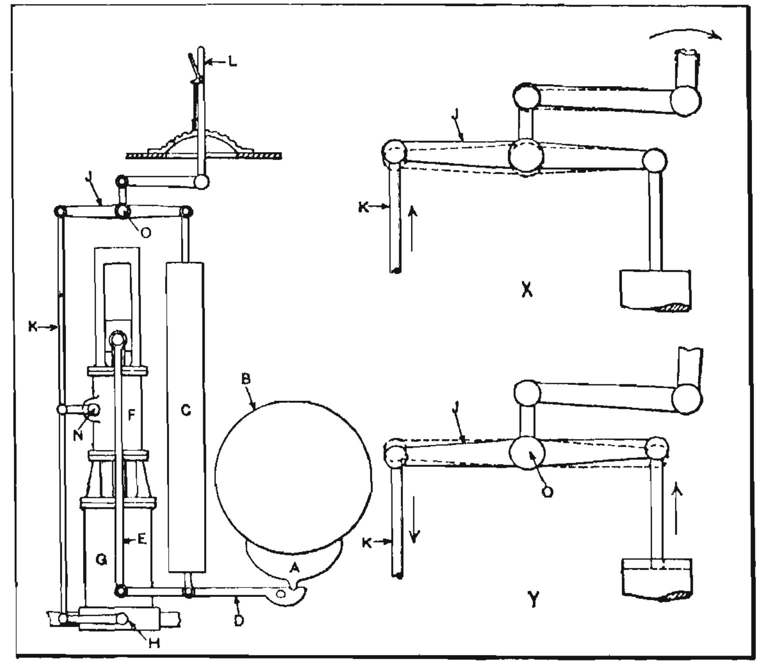


Fig. 6. Diagrams Illustrating Application and Action of Floating Lever

follow the hand-controlled movements practically the same as though there were a direct mechanical connection. A floating lever is so called because it is not attached to fixed pivots and does not have a stationary fulcrum, but is free to move bodily, or to "float" within certain limits and in accordance with the relative forces acting upon the different connections.

Fig. 6 illustrates one application of the floating lever. The diagram at the left represents an auxiliary braking apparatus for a large hoist. The brake shoe *A* is applied to the brake drum *B* whenever the dead weight *C* rests upon the lever *D*. This lever is connected by rod *E* with a cross-head attached to the upper end of a piston rod extending through the oil cylinder *F* and into the steam cylinder *G*. When steam is admitted beneath the piston in cylinder *G* by opening a valve at *H*, the weight is raised and the brake released, and, if for any reason the steam pressure should be suddenly reduced, weight *C* would fall and the brake be applied automatically. The movements of the piston in cylinder *G* and, consequently, of weight *C* are controlled by hand lever *L* through floating lever *J*, in such a manner that the weight rises and falls, as the lever is shifted, practically the same as though the force for moving the weight were derived directly from the lever by means of a rigid mechanical connection. The action of the mechanism is as follows: If the weight is down and the brake applied, and lever *L* is moved from its central position to the right, the left-hand end of lever *J* will be raised (as shown on an exaggerated scale by diagram *X*), thus lifting rod *K* and opening valve *H*; this valve has no lap, so that any movement of the lever admits steam to the cylinder. As soon as the piston begins to rise, the right-hand end of lever *J* also rises (see diagram *Y*) and turning about pivot *O* immediately begins to close the steam valve. If the lever *L* is moved through a small arc, the valve is closed quickly and the weight only rises a short distance; on the contrary, if the lever is thrown over to the extreme position, the piston and weight must move upward a proportionately greater distance before the valve is closed. If the lever, after being thrown to the right, is moved towards the left, valve *H* opens the exhaust port and the weight descends; as soon as it begins to move downward, the left-hand end of the floating lever is raised, which tends to close the exhaust port and prevent further downward motion.

An apparatus of this kind responds so quickly to adjustment that the weight follows the motion of the hand lever almost instantaeously and the end of the floating lever connected to rod *K* has very little actual movement. The oil cylinder *F* is used to stabilize the action of the weight and prevent overtravel which would occur if there were only the cushioning effect of steam. The by-pass valve *N* controls the flow of oil from one end of the cylinder to the other as the piston moves up or down, so that the motion of the weight ceases as soon as the steam and oil valves are closed.

Controlling Mechanism of Steering Gear. — The practical effect of the floating lever previously described for controlling the movements of power-driven apparatus may be obtained by other mechanical devices, examples of which are found on steamships for controlling the action of the steering engines. Engines used for this purpose are commonly equipped with a control valve which distributes steam to the engine valves. The latter are generally of the hollow piston type and are arranged to receive steam either at the ends or in the center, the exhaust varying accordingly. The admission of steam either to the ends or in the center is governed by the position of the control valve. For instance, if the control valve is moved in one direction, steam may be admitted to the ends of the engine valves and be exhausted in the center. If the control valve were moved in the opposite direction, this order would be reversed and also the direction in which the engine rotates; therefore, each engine valve requires but one eccentric, the control valve acting as a reversing gear. The mechanism which operates this control valve is so designed that, when the engine is set in motion to move the rudder either to port or starboard, this same motion is utilized to shift the control valve in such a way that the movement of the rudder coincides with the motion of the steering wheel. While the floating lever has been used in connection with this controlling mechanism, the common form of control depends upon the action (which is often differential) either of gearing or of a screw and nut.

With the arrangement illustrated at *A*, Fig. 7, the control valve of a steering engine is governed by the action of a screw that is operated by the steering wheel, and a nut that is revolved by the engine. The shaft *a* is connected with the steering wheel and transmits rotary motion to screw *b* which is splined to, and free to slide through, gear *c*. The rod *d* serves to operate the control valve of the steering engine. Any rotary motion of shaft *a* moves screw *b* in a lengthwise direction in or out of the nut on worm-wheel *e*, unless this nut is revolving at the same speed as the screw. The action of the mechanism is as follows: If worm-wheel *e*, which meshes with a worm on the steering engine crankshaft, is stationary, the rotation of shaft *a* will turn screw *b* in or out of the nut and shift the control valve, thus starting the engine in one direction or the other, depending upon which way the control

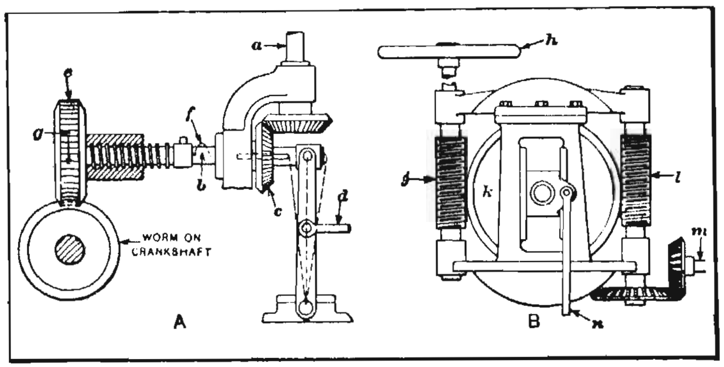


Fig. 7. (A) Controlling Device for Steering Gear; (B) Mechanism Used as Substitute for a Floating Lever

valve was moved. As soon as the engine starts, worm-wheel *e* and the nut begin to revolve, which tends to move the screw and control lever in the opposite direction. Suppose screw *b* were revolved in the direction shown by the arrow *f*, thus moving the screw and control lever to the right; then, as the engine starts, worm-wheel *e* and the nut revolve as shown by the arrow *g*. Now as soon as the rotation of shaft *a* and screw *b* is stopped or is reduced until the speed of rotation is less than that of worm-wheel *e*, the screw is drawn back into the nut and the control valve is closed. If the steering wheel and screw *b* were turned slightly and then stopped entirely, the rudder would only be moved a corresponding amount.

because the control valve would soon be shifted, by the action of worm-wheel *e*, to the closed position. Steering engines, in general, are equipped with some form of stopping device which automatically limits the movement of the rudder.

Rolling Worm-wheel Type of Controlling Mechanism. — The ingenious substitute for the floating lever illustrated at *B* in Fig. 7 depends for its action upon a worm-wheel which is interposed between two worms. The handwheel *h* controls the rotation of worm *j*, which meshes with the worm-wheel *k*. The worm *l* on the opposite side of the worm-wheel is rotated by whatever apparatus is to be controlled. The shaft of the worm-wheel is journaled in boxes which are free to slide up and down the vertical slides in the framework shown. Any vertical displacement of the worm-wheel is transmitted to rod *n* which operates the valve, clutch, or other mechanical device used for starting, stopping, and reversing the driving machinery. Assume that the mechanism is at rest with the worm-wheel midway between its upper and lower positions in the vertical slides of the housing. When the handwheel *h* is revolved in a direction corresponding to the motion desired, worm *j* revolves, and worm *l* is stationary, since the mechanism is not yet in motion; therefore, the rotation of the handwheel has the effect of rolling the worm-wheel *k* between the two worms either up or down, depending upon the direction in which the handwheel is rotated. Any vertical displacement of the worm-wheel will, through the medium of controlling rod *n*, start the power-driven machinery. This motion is immediately transmitted to shaft *m* and worm *l* which acts to move worm-wheel *k* in the opposite direction vertically, provided worm *j* is stationary or is revolving slower than *l*. The result is that the power-driven member is moved or adjusted proportionately to the rotation of the handwheel *h*. The handwheel, for instance, might be turned to a position corresponding to a certain required adjustment, which would then be made automatically.

Control Mechanism having Differential Bevel Gearing. — The steering gear controlling mechanism illustrated in Fig. 8

operates on the same general principle as the design previously described, although the construction is quite different. The control valve, in this case, operates with a rotary motion, instead of moving in a lengthwise direction. Shaft *A* is revolved by the steering wheel and transmits rotary motion to shaft *B* through the gearing shown. The differential action for regulating the position of the control valve is obtained by means of three gears *C*, *D*, and *E*. Gear *C* is keyed to shaft *B*, and gear *E* on the extended hub of worm-wheel *F* is free to revolve about shaft *B*. Gear *D* interposed between gears

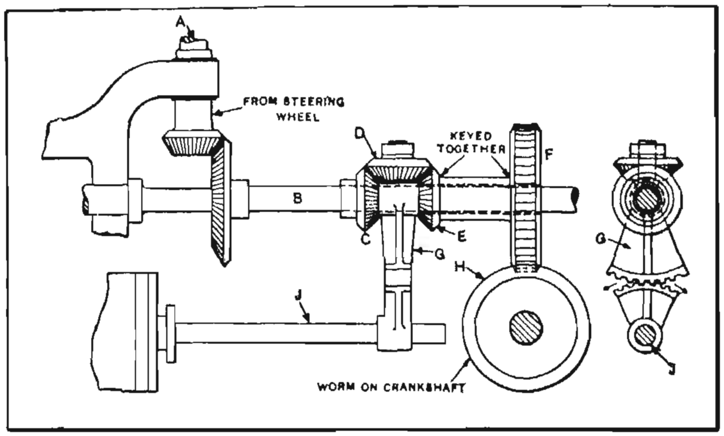


Fig. 8. Steering Gear Control Mechanism Having Differential Bevel Gearing

C and *E* is mounted upon a segment gear *G* which engages another segment gear on the control valve spindle *J*. If shaft *B* is revolved while gear *E* and the worm-wheel are stationary, gear *D* rolls around between the gears and, through the segment gear, turns the control valve, thus starting the steering engine and with it the worm *H* on the crankshaft which drives worm-wheel *F* and gear *E*. As soon as the rotation of shaft *B* is stopped, gear *E* which has been revolving in the opposite direction to that of *C* rolls gear *D* back to the top position, thus closing the control valve and stopping the engine. If gears *C* and *E* are revolved at the same speed, gear *D* simply rotates between them and the control valve remains open. If

the speed of gear *E* exceeds that of *C*, the valve begins to close, and if *C* revolves faster than *E*, the valve is opened wider and the engine continues to operate. This general principle has been applied to various classes of mechanisms.

Differential Governors for Water Turbines. — Many of the automatic governing devices used for controlling the speed of water turbines have a differential action. A simple form of governor is illustrated in principle by the diagram *A*, Fig. 9. An auxiliary water motor drives the bevel gear *a* by belt *d*, and bevel gear *c* is driven by belt *e* from a shaft operated by

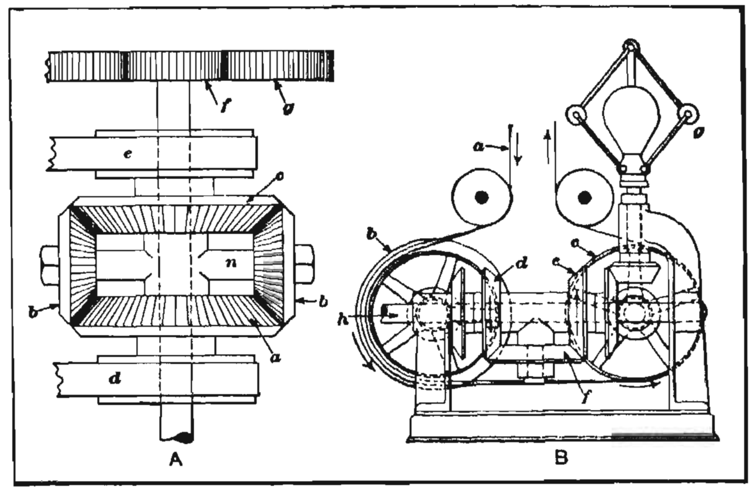


Fig. 9. Differential Governing Devices for Water Turbines

the turbine to be governed. Both gears *a* and *c* are loose on their shaft, but the arm *n* which carries the bevel pinions *b* is fast to the shaft. On one end of the shaft there is a pinion *f* which meshes with a rack *g* that operates the turbine gate, and thus controls the flow of water to the turbine. As the auxiliary motor has no work to do except to drive part of the governing mechanism, it runs at practically a constant speed; the variations due to the rise or fall of the water level are so small a percentage of the total head of water that the speed of this motor is little affected. It will be assumed then that the speed of gear *a* is practically uniform. The speed of gear c, however, changes with an increase or decrease of the load upon the turbine, and, as gear c runs faster or slower than gear a, the arm n follows it around one way or the other and thus opens or closes the turbine gate.

The governor shown at B also has a differential action, but it is controlled by centrifugal force acting on a fly-ball governing device. The governor is operated by a belt a connected with the turbine. This belt passes around idler pulleys and over the wide-faced pulleys b and c. These pulleys, through bevel gearing, drive the differential gearing composed of gears d, e, and f. Gears d and e are loose from their shafts and pinion f is pivoted on an arm that is keyed to the shaft. Gear e is connected by the gearing shown with a centrifugal governing device at g. The belt pulley b is conical and the diameter at the center is the same as that at pulley c. When the turbine is operating at normal speed, the belt is at the center of the conical pulley b and, consequently, gears d and e revolve at the same rate of speed in opposite directions. The result is that the arm carrying pinion f remains stationary. If the turbine begins to run too fast, the balls at g move outward under the action of centrifugal force, and belt a is shifted by a mechanism not shown to a smaller part of the conical pulley b. The resulting increase in the speed of gear d causes the arm carrying pinion f and the shaft h to which it is attached to revolve in the same direction as gear d. As a result of this movement, the turbine gate is lowered by means of gearing not shown, and the speed of the turbine wheel is reduced. If the turbine should begin to run more slowly than the normal speed, the shifting of belt a by governor g would cause gear d also to revolve slower, thus turning shaft h in the opposite direction and raising the gate.

Another modification of the differential governor is shown by the diagram, Fig. 10. This type of governor is equipped with two sets of epicyclic gearing. The gears A and B are free to turn on the shaft, but may be retarded by brake bands at E and F. The inner gears C and D are driven by belts connected in some way with the turbine. One of these belts

is open and the other crossed, so that the gears revolve in opposite directions. The brake bands are so arranged that, when one tightens, the other loosens its grip on the brake drum. Both of these bands are operated by a shaft G and the tightening of the bands is effected by a double ratchet mechanism (not shown) having two pawls. One pawl rotates shaft G in one direction and the other in the opposite direction. When the speed increases or decreases, one pawl or the other is operated by a fly-ball governor driven from the turbine. As the result of this motion of the pawl, one band is tightened and the other released, so that one of the gears

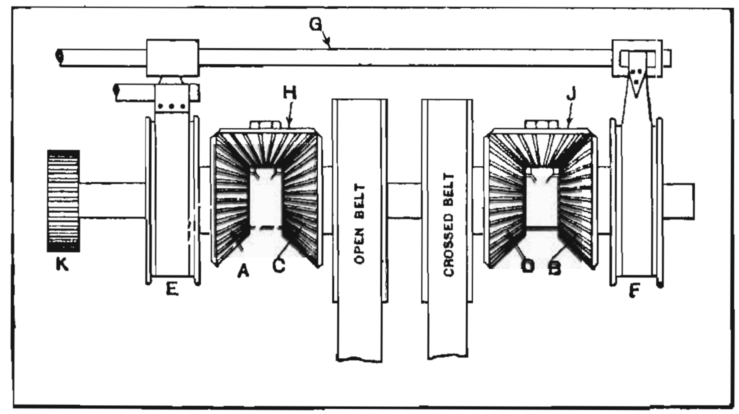


Fig. 10. Differential Governing Mechanism Controlled by Ratchet-operated Brakes

A or B is held with a greater or less degree of friction or is prevented from turning altogether, while the other one runs free. If gear A is held by the brake, the arm carrying pinion H will begin to turn in the same direction in which gear C turns, whereas, if gear B remains stationary, the arm carrying pinion J will follow gear D; consequently, the pinion K on the end of the shaft will by means of a rack raise or lower the turbine gate. This governor depends for its sensitiveness upon the fly-ball governing device, and for its power upon the transmitting capacity of the open and cross-belts.

Differential Gearing of Automobiles. — One of the important applications of differential gearing is found on automobiles. The object of transmitting motion from the engine to the rear axle through differential gearing is to give an equal tractive force to each of the two wheels and, at the same time, permit either of them to run ahead or lag behind the

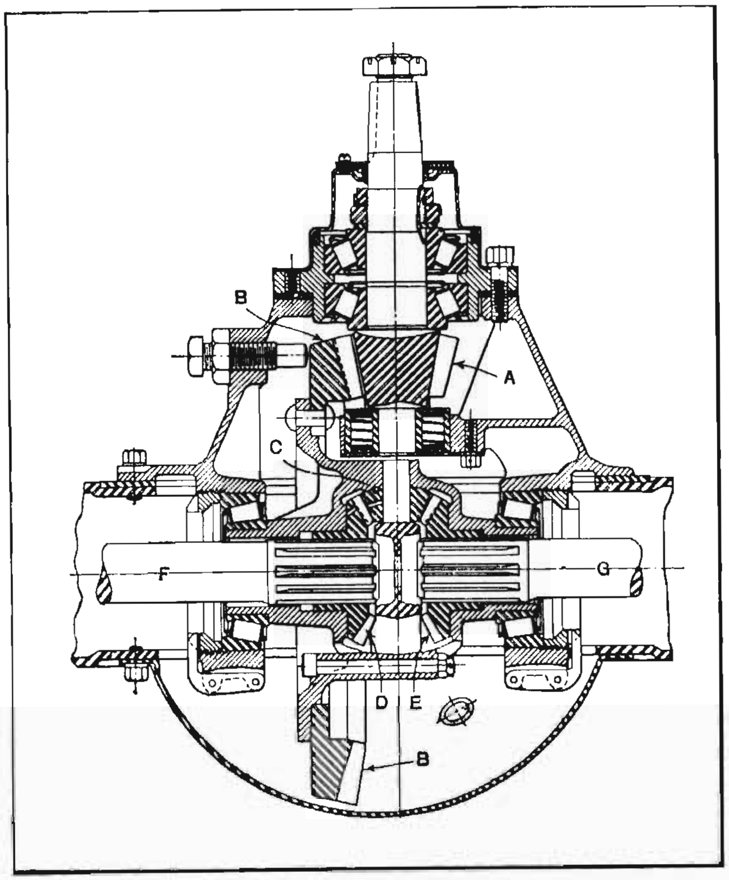


Fig. 11. Differential Gearing of an Automobile

other as may be required in rounding curves or riding over obstructions. The axle is not formed of one solid piece, but motion is transmitted to the right- and left-hand wheels by means of separate sections, the inner ends of which are attached to different members of the differential mechanism.

The principle of this mechanical movement will be understood by referring to Fig. 11. The propeller shaft extends from the transmission case where speed changes are obtained, and revolves the bevel pinion *A* which drives the large bevel gear *B*. Gear *B* is attached to a casing which contains the differential gearing. As this casing revolves it carries around with it a "spider" upon which is mounted either three or four equally-spaced pinions *C*. These pinions are free to turn about the bearings formed on the arms of the spider, and they are located between and mesh with the side gears *D* and *E*. These side gears are mounted upon the splined ends of the right- and left-hand axles *F* and *G* and the side gears rotate with these axle sections.

Under ordinary conditions, the rotation of gear *B* causes gears *D* and *E* to both revolve at the same rate of speed, since the connecting pinions *C* are moved around with the casing, but do not revolve. To illustrate the action, assume that the wheels are jacked up and are simply revolving in one position; then, if one wheel is held from turning so that, say, gear *E* is stationary, the rotation of bevel gear *B* will roll pinions *C* around on gear *E* with the result that gear *D* will revolve twice as fast as when gear *E* is revolving with it and at the same speed. On the other hand, if the opposite wheel and gear *D* were held stationary, the gear *E* would run at twice its normal speed; moreover, if the speed of either of the gears is reduced, the other side is speeded up a corresponding amount. The differential gearing is ordinarily incorporated in the rear axle, except when power is transmitted to the wheel by means of side chains, in which case the differential is in the countershaft. Gears *A* and *B* usually are either the "spiral bevel" or the "hypoid" type.

Speed Regulation through Differential Gearing. — When the speed of a driven part is governed by drives from two different sources, differential gearing may be used to combine these drives and allow any variations in speed that may be required. An application of this kind is found on the fly frames used in cotton spinning for drawing out or attenuating the untwisted fiber or roving, by passing it between different pairs of rolls which move, successively, at increased speeds. After the fiber is attenuated, it is wound on bobbins and at the same time given a slight twist. The diagram, Fig. 12, represents a mechanism for controlling the speed of the bobbins, one of which is indicated at *B*. This bobbin receives its motion through a train of gearing connecting with the main shaft of the machine and also through another combina-

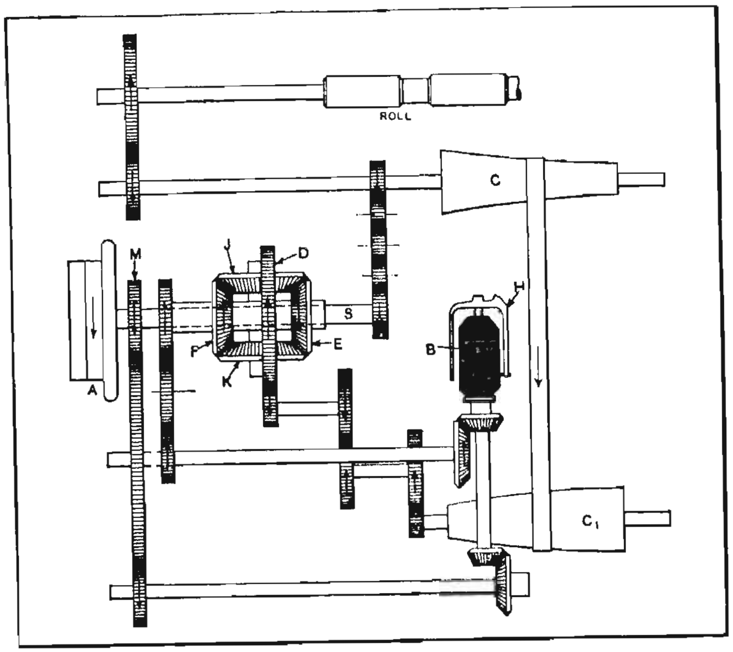


Fig. 12. Diagram of Mechanism Having Differential Gearing Through Which Speed Changes are Transmitted

tion of gearing which is driven by a pair of cone-pulleys for decreasing the speed of the bobbin as the roving is wound upon it and the diameter increases.

The main shaft is driven by pulley *A* and motion is transmitted through shaft *S* and the gearing shown to the cone *C* and the rolls, one of which is indicated at the upper part of the diagram. The cone *C* and the rolls move at a constant speed, and the roving is delivered by the rolls at a uniform

rate of speed. On the shaft *S* there is a bevel gear *E*, which is one of the gears of an epicyclic train that is commonly known as the "differential motion." The large gear *D* corresponds to the arm of the gear train, since it carries the two intermediate bevel pinions *J* and *K*. This gear *D* is driven from the lower cone C_1 which is connected by belt with the upper cone. Bevel gear *F* which meshes with the pinions carried by gear *D* is loose on shaft *S* and is connected through gearing with the bobbin *B*. With this arrangement, the speed of the bobbin depends first upon the speed of bevel gear *E*, which is constant, and also upon the speed of gear *D*, which may be varied by shifting the position of the belt on the cones. Any variations in the relative speeds of gears *D* and *E* will produce twice the variations in the speed of the bobbin.

The roving is wound on the bobbin in successive helical layers by means of the flyer *H* driven at a constant speed by gear *M* on shaft *S*. The roving passes from the rolls to the flyer, and entering the top of its hollow spindle, is threaded down through one arm of the flyer and then wound on the bobbin. The flyer and bobbin revolve in the same direction, but the bobbin has a higher velocity and, for that reason, draws the roving from the flyer and winds it in successive layers as the bobbin travels up and down, so as to cover its entire surface. As each successive layer is added, the bobbin increases in diameter, and its speed relative to that of the flyer must be decreased in order to prevent breaking the roving. This change of speed is transmitted to the bobbin through the differential gearing referred to by shifting the belt on the cone-pulleys.

Differential Gear and Cam Combination. — The differential gear and cam combination described in the following is used on fly-frames in conjunction with the same general class of mechanism illustrated by the diagram, Fig. 12. This mechanism differs from the differential ordinarily used in that it has no epicyclic train of gearing. As previously explained, a differential motion is employed in connection with a shifting belt and cone-pulleys for changing the speed of the bobbins. The differential action is obtained, in this case, by means of a crown gear A (Fig. 13) which is attached to the main driving shaft B; the crown gear C secured to sleeve E, which carries the bobbin driving gear F, and the double crown gear D, which is mounted on a spherical seat and engages gears A and C at points diametrically opposite. This double crown gear operates in an oblique position, so that a small part of the gear meshes with gear A on one side and a small part on the other side meshes with gear C. The spherical bearing allows the intermediate crown gear D to swivel in any direction, and it is held in position by a cam surface on the edge of sleeve G. The gear C has the same number of teeth

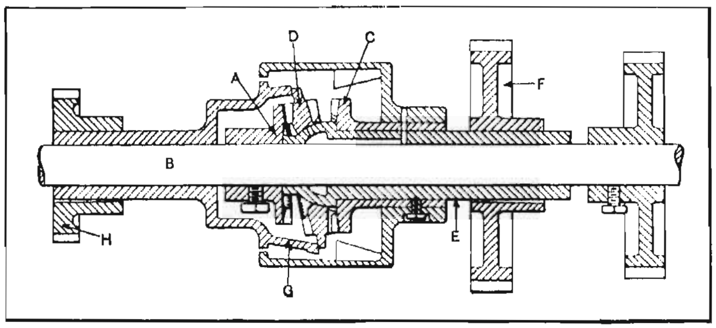


Fig. 13. Differential Gear and Cam Combination

as the intermediate gear D, but gear A has a somewhat smaller number of teeth.

The differential action is obtained by the relative motions between gear A and cam G. This cam is driven from the lower belt-cone of the machine which is connected with gear H. If cam G were revolving at the same speed as gear A, the same teeth on gears A and D would remain in contact and the entire gear combination would act practically the same as a clutch. As soon as the speed of the cam differs from that of gear A, the position of intermediate gear D is changed so that different teeth are successively engaged. As the result of this differential action, the speed transmitted to gear C is either increased or decreased. The extent of the differential

motion depends upon the difference between the speeds of gear A and cam G. As this difference diminishes, the speed of gears D and C increases; inversely, as the speed of cam G is reduced, the speed of gear C is also reduced, since the motion from gear A is lost as the result of differential action. The advantages claimed for this mechanism are quiet operation and reduction of friction.

Differential Hoisting Mechanism. — An ingenious method of utilizing differential action to vary the speed of a hoisting mechanism is illustrated by the diagram, Fig. 14, which represents the crane to which this mechanism is applied. There are two chains attached to the crane hook. One of these chains

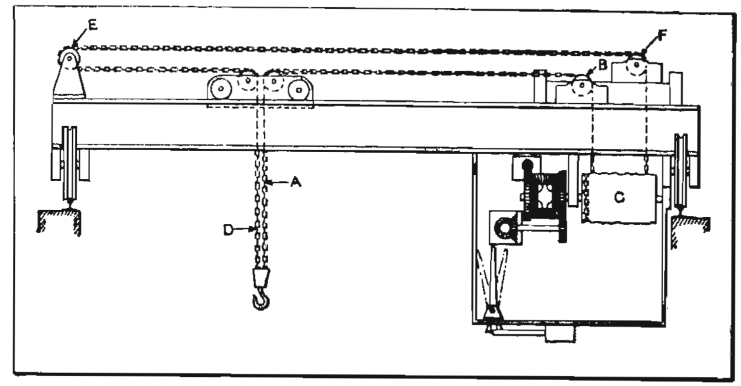


Fig. 14. Crane Equipped with Differential Hoisting Mechanism Shown Diagrammatically in Fig. 15

A passes over a pulley on the trolley and over pulley B to the winding drum C. The other chain D passes upward over its trolley pulley to the left, and over pulley E to pulley F, and then down to a drum located back of drum C. These chains may be wound upon their respective drums either in opposite directions or in the same direction, and at varying rates of speed. If both drums are rotated in opposite directions at the same speed the effect will be to raise or lower the hoisting hook, whereas, if the drums rotate in the same direction and at equal speed, the chain will be taken in by one and given off by the other, thus causing the hook and its load to be carried horizontally without raising or lowering it. Any difference in the speed of the two drums when moving either in the same or opposite directions will evidently cause the hook to move both vertically and horizontally at the same time.

The mechanism for operating the two hoisting drums is illustrated diagrammatically in Fig. 15. There are two electric motors J and K. Motor J drives the worm-wheels L in opposite directions and also the attached bevel gears. The other motor K drives the spur gears M and the upper bevel gears. The intermediate pinions N between the bevel gears revolve on arms Q which are keyed to the shafts of their re-

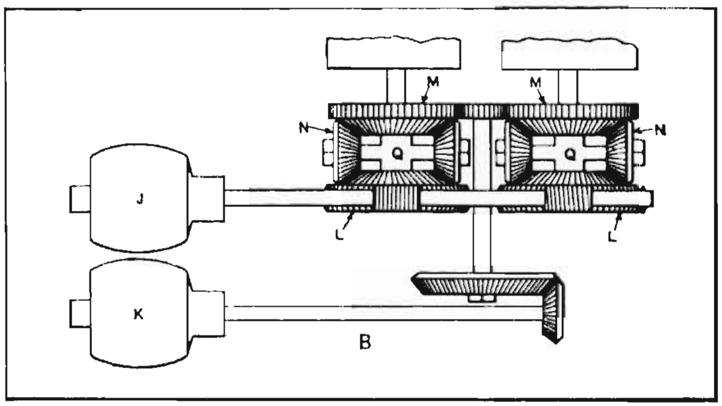


Fig. 15. Differential Hoisting Mechanism

spective drums. The bevel gears with which the pinions mesh are loose on their shafts. With this arrangement, if motor K is stationary, motor J will drive the drums in opposite directions and raise or lower the hook as previously explained. On the other hand, with motor J stationary, motor K will operate the drums in the same direction and move the crane hook horizontally. As these motors may be reversed or operated together at varying speeds, any desired combination of movements and speeds for the hook and its load may be obtained.

Variable and Reversing Rotation for Feed-Rolls. — The requirement of the mechanism here described, which is used

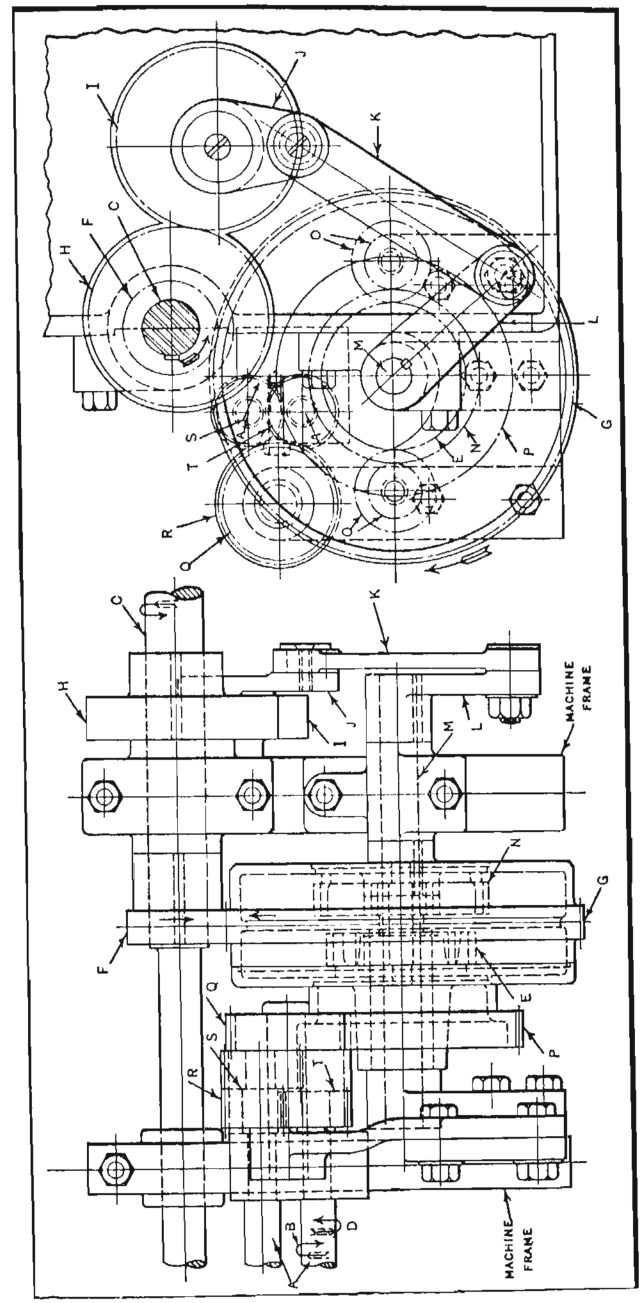


Fig. 16. Special Design of Differential Mechanism for Transmitting to Feed-rolls a Variable and Reversing Rotation

on a cotton combing machine, is to rotate feed-rolls *A* (see Fig. 16) with their respective top rolls (not shown) 1.4 revolutions in the direction of arrow *B* while driving shaft *C* makes 0.6 revolution, and, during the remaining 0.4 revolution of shaft *C*, to rotate feed-rolls *A* 0.7 revolution in the direction of arrow *D*.

The object is to feed approximately 4 inches of cotton

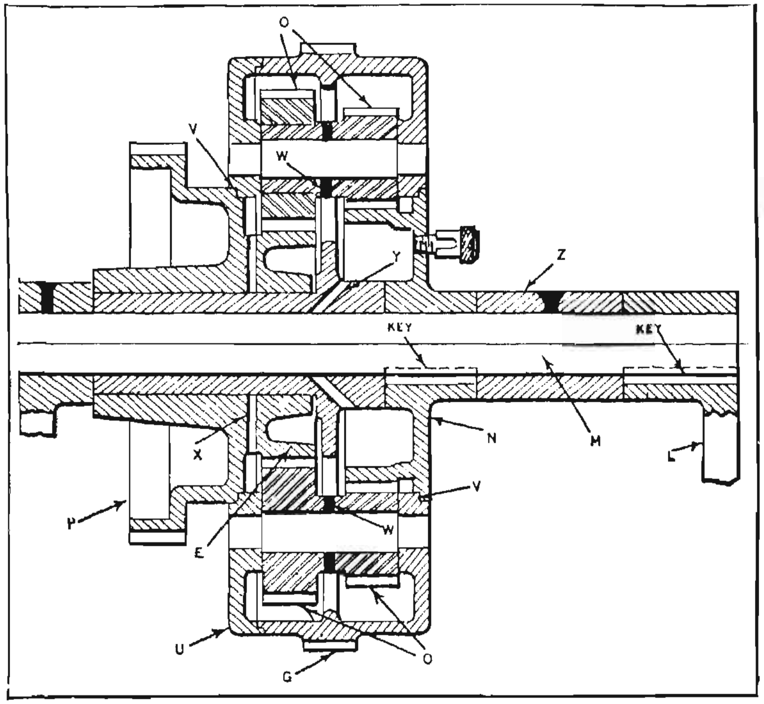


Fig. 17. Cross-section of Differential Gearing

forward, as indicated by arrow *B*, and then reverse and feed approximately 2 inches of material backward, as indicated by arrow *D*, and repeat for each revolution of driving shaft *C*, thus performing a doubling process in conjunction with other elements of the machine.

The required result was obtained by combining a constant motion and a variable oscillating motion, the two motions co-acting on a common gear *E* (see also Fig. 17) through an

epicyclic or differential gearing combination. Referring to Fig. 16, it will be seen that the constant motion is effected by driving pinion *F* and housing gear *G;* also that the variable oscillating motion is produced through eccentric gears *H* and *I*, crank-arm *J*, connecting link *K*, rocker arm *L*, rocker shaft *M*, and rocker shaft gear *N*. Through the planetary gears *O* these two motions are permitted to combine and drive feed-rolls *A* forward and backward as mentioned, through gears *E*, *P*, *Q*, *R*, *S*, and *T*.

The timing of the forward and backward rotation of rolls *A* is controlled by eccentric gears *H* and *I*, which transmit a quick-return motion to rocker arm *L* for the forward rotation of rolls *A*, as indicated by arrow *B*, and next a slower motion in the opposite direction to rocker arm *L* for the backward rotation of the rolls *A*, as indicated by arrow *D*. Eccentric gear *H* is keyed to driving shaft *C*, while *I* is keyed to the hub of crank arm *J*, which revolves on a suitable stud fastened to the frame of the machine.

Housing gear *G*, which runs loose on rocker shaft *M*, and all other gears and bearings inside of it, are splash-lubricated through oil-holes *X*, *Y*, and *W*, Fig. 17. By pouring one-half pint of oil into the housing, three weeks' supply of lubricant is provided, which is well distributed to all bearings by planetary gears *O* revolving through it. Oil-tight joints are provided for gears *N* and *P*, as shown at *V*. Gears *E* and *P* are integral and revolve on the extended hub of housing gear *G*. Rocker arm *L* and rocker shaft gear *N* are keyed to rocker shaft *M*. This mechanism replaced a cam and clutch arrangement which was too noisy and did not wear well.

Differential Speed Indicator. — A sensitive speed-indicating device which shows variations of speed between two rotating parts is shown, partly in section, in Fig. 18. This indicator operates on the differential principle. It is equipped with two cylindrical rollers; one roller is shown at *A* and the other is located in a similar position on the opposite side of the vertical center line. The axes of the roller shafts are in the same vertical plane, and on the ends of these shafts are mounted belt pulleys C. These pulleys are connected with the shafts the relative speeds of which are to be compared. Each roller A is in contact with a spherical steel ball B three inches in diameter. The ball is held in position by a small stop D at the rear and by a small roller E at the front. This roller is mounted on an arm fixed to a spindle which is free to rotate and to the outer end of which is attached the pointer F. When both the supporting rollers A are driven at the same speed and in the same direction, the spherical ball will rotate about a transverse horizontal axis and will carry the wheel E vertically up or down, as the case may be. The direction of movement will be indicated by the pointer F.

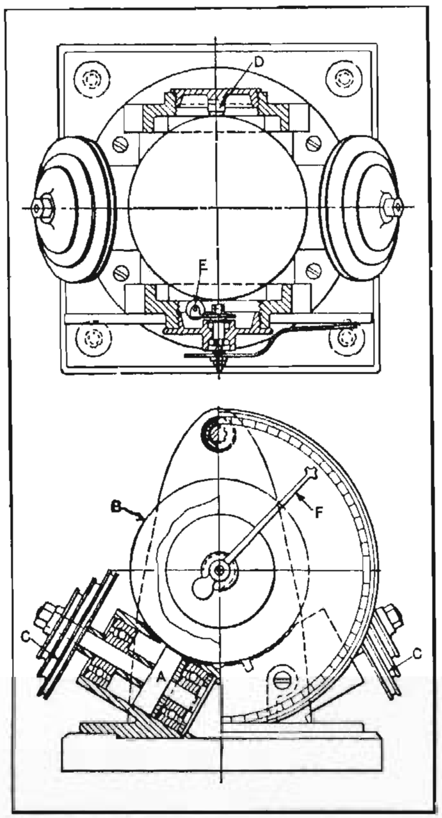


Fig. 18. Differential Speed Indicator

If either of the supporting rollers runs faster than the other, the ball will rotate about some inclined axis and wheel E will naturally turn so that its axis is parallel to that about which the sphere rotates. This instrument is said to be very sensitive as an indicator of speed variations. For instance, it is claimed that a difference in the speed of the rollers due to a variation of 0.001 inch in the diameter of driving pulleys having a nominal diameter of $2\frac{1}{2}$ inches can be detected.

CHAPTER XIII

STRAIGHT-LINE MOTIONS

A COMBINATION of links arranged to impart a rectilinear motion to a rod or other part independently of guides or ways is known either as a *straight-line* motion or a *parallel* motion, the former term being more appropriate. Mechanisms of this type were used on steam engines and pumps of early designs to guide the piston-rods, because machine tools had not been developed for planing accurate guides. One application

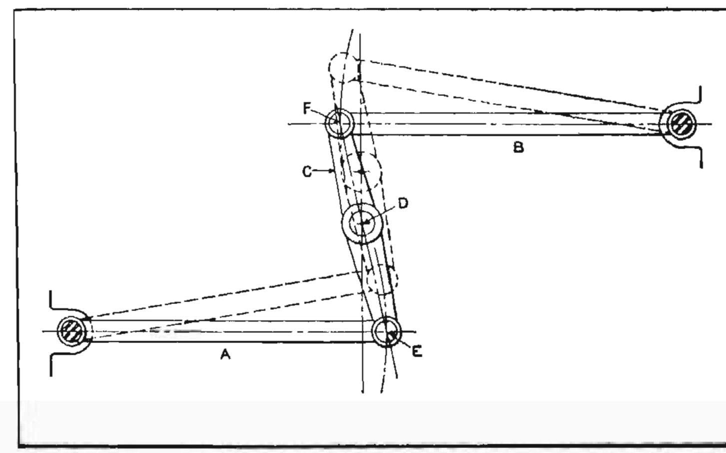


Fig. 1. The Watt Straight-line Motion

of straight-line motions at the present time is on steam engine indicators for imparting a rectilinear movement to the pencil or tracing point. The principle of the well-known parallel motion, invented by James Watt in 1784, is illustrated by the diagram, Fig. 1. Links A and B are free to oscillate about fixed pins at their outer ends, and are connected by link C. A point D may be located on the center-line of link C, which follows approximately a straight line when links *A* and *B* are given an oscillating movement, because, when *A* moves from its central position, the center of pin *E* moves to the left along its circular path while the center of pin *F* moves to the right. As the motion of point *D* is affected by both links *A* and *B*, it moves very nearly in a straight line, provided *D* is correctly located and the angular motion of the links does not exceed about 20 degrees. Very few straight-line mechanisms produce a motion which is absolutely straight, and the general practice is to so design them that the guided part will be on the line when at the center and extreme ends of the stroke.

Scott Russell Straight-line Motion. — The mechanism illustrated in Fig. 2 will give an exact straight-line motion, but

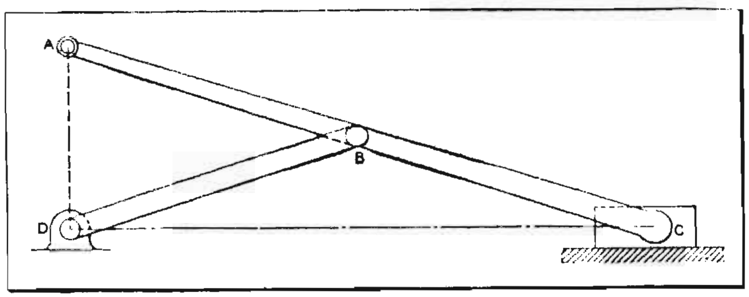


Fig. 2. The Scott Russell Straight-line Motion

it is necessary to have an accurate plane surface upon which block *C* can slide. In addition to this sliding block, there are two links *AC* and *DB*. The link *DB* is one-half the length of *AC* and the shorter link is connected at a point *B* midway between *A* and *C*. The shorter link oscillates about a stationary pivot at *D* as end *A* is moved up or down along the straight line *AD*. Since *AB*, *DB*, and *BC* are equal, a circle with *B* as the center will intersect points *A*, *D*, *C* for any angle *DCA*; consequently, the line *AD*, traced by point *A* is perpendicular to *DC*, since *ADC* is always a right angle.

Instead of having guides or a plane surface for the sliding block *C*, the mechanism is sometimes modified by attaching

the block end of link *AC* to another link which is free to oscillate about a fixed pivot so located that the link will be perpendicular to the line *CD*, when in its mid-position. The longer this link and the greater the radius of the arc described by the connecting point at *C*, the more nearly will *C* move in a straight line; hence, the longer this link, the less point *A* deviates from a straight line. This modification of the Scott Russell straight-line motion is sometimes called the *grasshopper motion.*

Straight-line Motions for Engine Indicators. — Some form of straight-line motion is necessary on a steam engine indi-

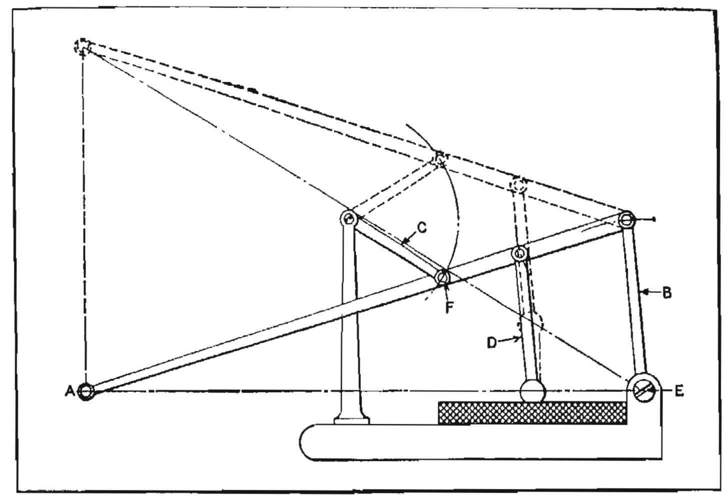


Fig. 3. Straight-line Motion of Thompson Indicator

cator in order that the motion of the indicator piston will produce a parallel movement of the tracer point or pencil, which draws a diagram on the paper or indicator card. The cylinder of the indicator is open at the bottom and is connected by suitable pipes with each end of the steam engine cylinder, so that the under side of the indicator piston is subjected to the varying pressure acting upon the engine piston. The upward movement of the indicator piston resulting from the steam pressure is resisted by a spiral spring of known resilience, and a rod extending above the piston connects with some form of link work designed to give a straight-line motion to the tracer point. When the engine is running and the indicator is in communication with the steam cylinder, variations of pressure will be recorded by the vertical movement of the pencil or tracer which is brought into contact with paper wound about a cylindrical drum that is rotated by the reciprocating motion of the engine cross-head.

The straight-line or parallel motion of one indicator is shown in Fig. 3. The arm A which carries the pencil at its outer end is pivoted to link B which, in turn, is pivoted to the top of the indicator. As arm A moves upward, the outer end is guided along a straight line by link C, which oscillates about

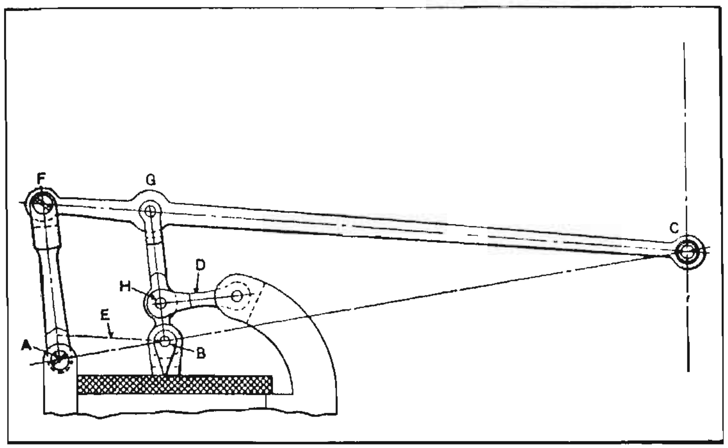


Fig. 4. The Crosby Straight-line or Parallel Motion

a fixed pivot and is connected to arm A at F. This mechanism is so proportioned that a line from A to E intersects the point at which link D is attached to the piston.

The straight-line motion of another steam engine indicator is shown in Fig. 4. This mechanism, like the one previously described, is so arranged that the fulcrum A of the entire mechanism, the connection B, and the pencil point C are always in a straight line. The fundamental principle of this mechanism is that of the pantograph. If link D were removed and replaced by another link at E, both parallel and equal in length to FG, this would result in a well-known form of pantograph mechanism. The length of link D to replace E may be determined as follows: The procedure is to first ascertain, by trial, a convenient location for the point at which link D is to connect with link BG. The path followed by point H as end C is moved along a straight line is plotted on a large scale for all positions on the linkage within the required range of movement. This path will be approximately the arc of some

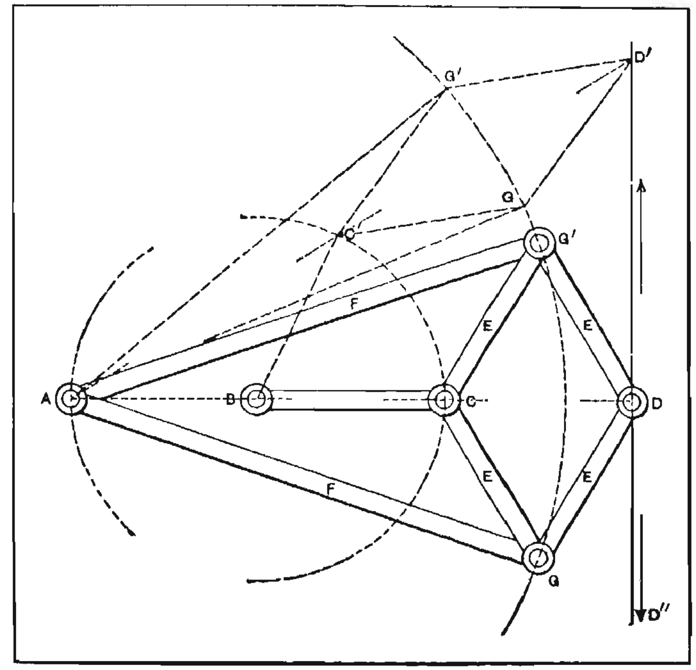


Fig. 5. Arrangement of Peaucellier Linkage for Straight-line Motion

circle, and the fixed pivot for link D is located at the center of this circle. If a link at E were actually used in place of link D, a straight-line motion at C could be obtained, providing the pivot B had a straight-line motion. Any form of guide intended to insure a straight movement at B would be objectionable, since it is desirable to reduce the friction of mechanisms of this type to a minimum. It is also essential to have the parts as light as possible in order to minimize the inertia and the effect of momentum, which is especially troublesome when taking cards from engines operating at high speed.

With the parallel motion of another indicator, a pin on the pencil arm corresponding to the one shown at F in Fig. 4 engages a curved slot in a stationary plate which is secured to the indicator in a vertical position. This curved slot takes the place of a link, and its curvature is such as to compensate for the tendency of the pencil to move in an arc.

Peaucellier Straight-line Motion. — The link mechanism shown in Fig. 5 will give an exact straight-line motion. This

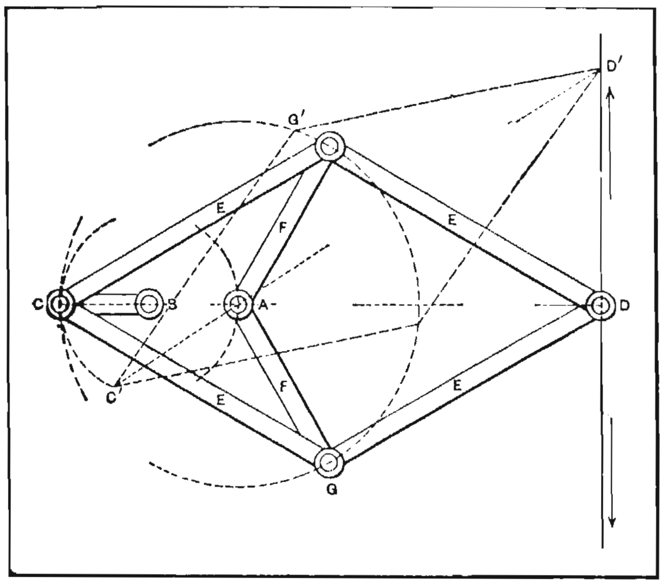


Fig. 6. Modification of Peaucellier Straight-line Mechanism

mechanism was invented by Peaucellier, a French army officer. It is composed of seven links moving about two fixed centers of motion, A and B. The four equal links E form a rhombus; the links F are equal, and the center B is midway between A and C. If the point D be moved in the direction of the arrows, it will be constrained to move in the straight path $D'D''$, which is perpendicular to the line of centers $ABCD$. This may be tested experimentally. The path of the point C is the circum-

ference $AC'C$; and the path of GG' is the arc described with the radius F. If the center-line of the links E and F be assumed in any position such as $AC'D'$, it will be found that the rhombus the sides of which represent the length of the links E takes the position shown in the drawing.

In Fig. 5, the centers A and B are external to the links E. A variation of the linkage is shown in Fig. 6, in which the centers A and B are within the rhombus. The links F are equal, and center B is midway between A and C, as in Fig. 5. The corresponding links and points in the figures are labeled with the same letters; it may be shown experimentally that the point D is compelled to move in a straight line perpendicular to the line of centers $CBAD$.

CHAPTER XIV

MISCELLANEOUS MECHANICAL MOVEMENTS

THE mechanisms described in this chapter are of such a miscellaneous character that they cannot be placed in any of the general groups or classifications covered by preceding chapters. They are included in this treatise, however, to add to the variety of the mechanisms described.

Mechanism to Insure Full-stroke Movement of Operating Lever. — Mechanisms are sometimes so arranged that hand-operated movements are, to some extent, controlled mechanically, to prevent motion in the wrong direction or incomplete action. The full-stroke ratchet mechanism shown in Fig. 1 is used on an adding typewriter to prevent the operator from starting handle *A* and not completing the required movement. For instance, if handle *A* is in the upper position, as shown at the left, any downward movement must be continued until the handle has made a complete stroke before it can be reversed for returning it to the original or upper position. Similarly, if the lever is at the lower end of its stroke, as shown by the view to the right, any upward movement must be completed before the direction of motion can be reversed. This positive control of the action of handle *A* is obtained in a very simple manner. As the handle is moved downward or upward, pawl *B* is carried with it. This pawl is pivoted to part *D* and normally held in a vertical position by a spring. When handle *A* is at the upper end of its stroke, as shown at the left, and a downward movement is started, pawl *B* engages sector *C* and its upper end swings to the right; as the downward movement of handle *A* continues, pawl *B* engages successive notches in sector *C*, and locks into one of these notches if an attempt is made to return handle *A* before the downward stroke is completed. When handle *A* has been pushed all the

way down (as shown to the right), pawl *B* drops into the enlarged notch *E* of sector *C* where there is enough room to permit the pawl to swing around to the vertical position; consequently, as soon as handle *A* is moved upward, the top of pawl *B* swings to the left and again engages successive notches in sector *C*, thus preventing any return of handle *A* to the lower position until the pawl has cleared the upper end of the sector and again swings to a vertical position.

Lock to Prevent Reversal of Rotation. — Some shafts must be free to rotate in one direction but be locked instantly against

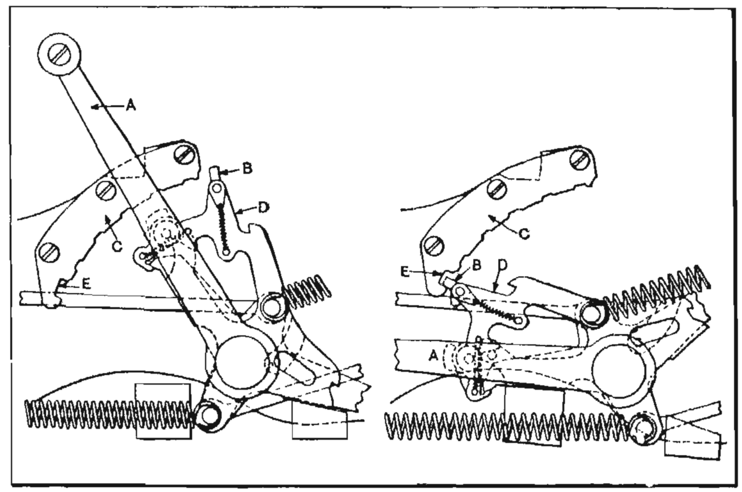


Fig. 1. Full-stroke Mechanism to Prevent Starting the Operating Lever and not Completing its Movement

a reversal of rotation. A ratchet mechanism may be objectionable because of its noise and backlash. Under these conditions, the arrangement shown in Fig. 2 was found to be satisfactory. Shaft *A* is free to rotate clockwise, but a reversal is not allowed, although when the shaft is not running clockwise, there is always a tendency toward reversal because of torque exerted on the shaft. Keyed to the shaft is a ring *B* into which are cut three wedge-shaped recesses containing rollers *C*. Ring *B* rotates within bracket *D*, which may be bolted to the wall or some immovable structure. Ring *B* and bracket *D* are kept in alignment by retaining plates *E*, which are bolted to the ring. The tool-steel rollers *C* are kept in contact with ring *B* and bracket *D* by means of light springs *F* which are riveted to their keepers.

When shaft *A* and ring *B* rotate clockwise, the rollers tend to move relatively in the opposite direction, thus compressing springs *F*. This movement releases any wedging action between the roller and members *B* and *D*, although the rollers always remain in contact with these members. Any back-

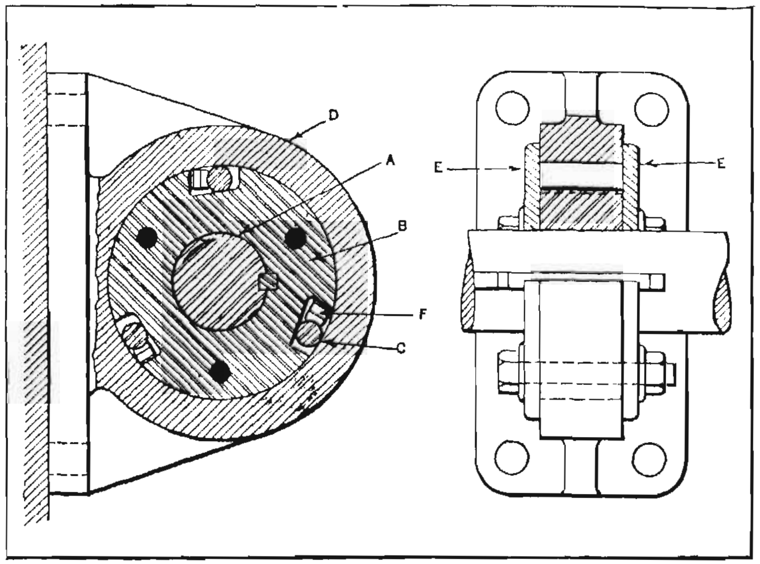


Fig. 2. Reversal of Shaft Rotation is Prevented by Wedging Action of Rollers

ward or counter-clockwise rotation is stopped instantly, because the rollers become wedged between parts *B* and *D*, thus locking them together. It is evident that the greater the torque counter-clockwise, the greater will be the locking effect within the limits of the strength of materials used. This simple contrivance proved to be very effective for preventing a reversal of shaft rotation.

Device to Rotate Shafts Synchronously in Opposite Directions. — Two shafts which are in alignment are rotated synchronously and in opposite directions by the simple ar-

rangement shown by the diagram Fig. 3. The two shafts *S* and S_1 have cranks of equal throw. On the outer ends of these cranks are universal joints. Balls *K* are shown, but

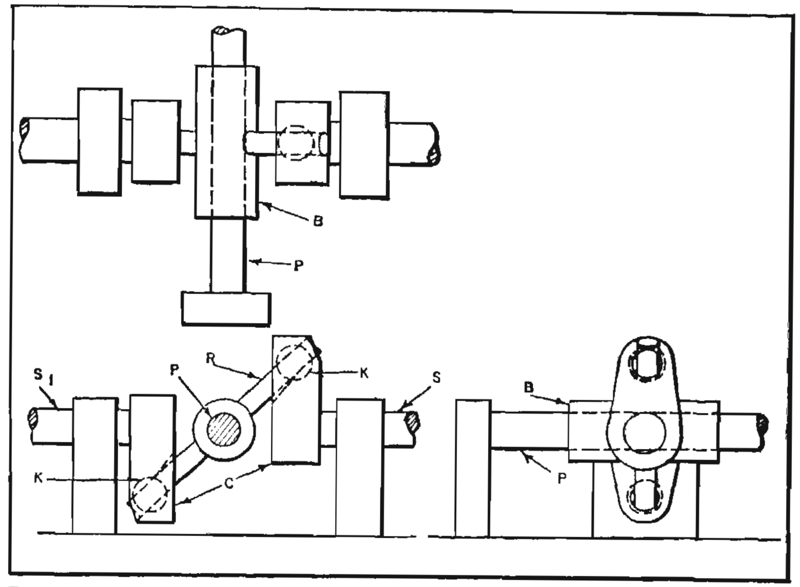


Fig. 3. Diagram Showing Device for Rotating Shafts in Opposite Directions, the Shafts Being in Alignment

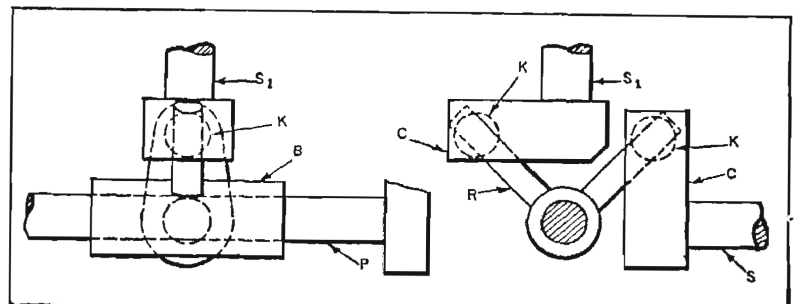


Fig. 4. Transmission Shown in Fig. 3 Applied to Shafts Located at Right Angles

any equivalent joint is satisfactory. The arms *R* of rocking beam *B* are free to slide through holes in balls *K*. This rocking beam is free to move axially and rock upon fixed shaft *P*, which is located at right angles to shafts *S* and S_1 and in their planes. This fixed shaft is also midway between the planes of rotation of the ball centers.

This is a flexible arrangement, as it may be applied to shafts located at any angle. Fig. 4 illustrates shafts at right angles. The connecting member is the same in principle. The arms of the rocking beam are at right angles, and the axis of the fixed shaft in this case is at the vertex of the angle made by the driving and driven shafts. If the driving and driven shafts are not in alignment, the rocker arms must be offset the same amount as the shafts. One driver using one rocking beam and having arms suitably arranged can operate a number of shafts parallel to the driving shaft or making an angle to it, provided the driving and driven shafts have a common plane; or in case the driven members are angularly disposed, provided the planes of the driven members have a common vertex through which the axis of the driver passes.

This mechanism is adapted for complete enclosure and oil-immersed operation. This form of transmission was applied where gears were not desired, although if gears had been used, five would have been necessary.

Disengagement of Worm-gearing for Rapid Adjustment. — A hand-operated winding drum used for adjusting the height of airplane models in a "wind tunnel" is so arranged that the drum may be rotated either through worm-gearing for precise adjustments, or directly, by disengaging the worm-gear, when rapid adjustments are desired. The airplane model is supported by wires (not shown) which pass up over pulleys and then to the winding drum *B* (Fig. 5), one wire extending forward and the other backward relative to drum *B*. The rotation of this drum winds or unwinds both wires simultaneously. The knurled handwheel *C* is used for fine adjustments, motion being transmitted through a single V-thread worm and a straight-faced worm-wheel *D* located at one end of the drum. If a rapid adjustment is required, the worm-wheel is disengaged from the worm merely by pulling the worm-wheel and drum axially on bolt *F*; the drum is then

turned directly by hand, the flange *G* being knurled to provide a better grip.

When the outward pull is released, spring *H* immediately forces the worm-wheel back into mesh, thus relocking the setting. The handwheel *C* is graduated on top so that the amount of adjustment can be determined. Although this device is fitted to laboratory apparatus, the idea of sliding a worm-wheel axially out of mesh with the worm might be utilized for other purposes; however, it is evident that only straight-

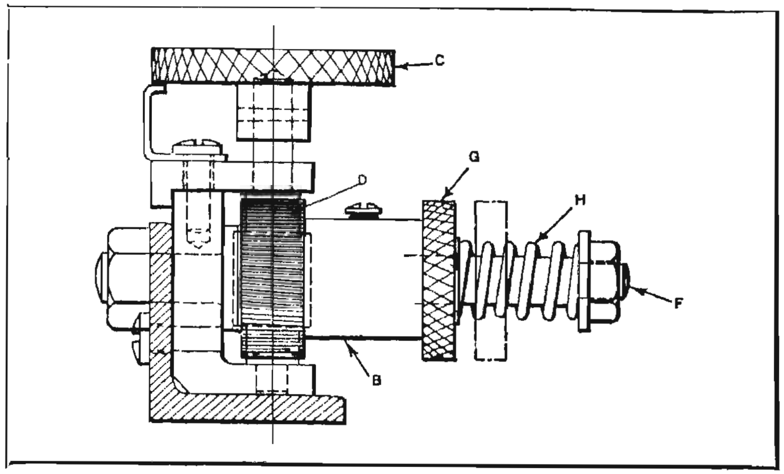


Fig. 5. Worm-gearing Arranged for Quick Disengagement When Rapid Adjustment is Required

faced worm-wheels could be used. For some applications, it might be preferable to replace the knurled portion of the winding drum with a spur gear arranged to slide into engagement with driving gears when the worm-wheel was disengaged. Another variation for possible application to small lifting blocks would include a friction brake for rapid lowering by gravity.

Centrifugal Chuck-closing Mechanism. — The automatic chuck-closing mechanism illustrated in Fig. 6 is operated by the action of centrifugal force upon balls or spherical weights which move outward when chuck rotation begins, thus automatically closing the collet chuck upon the work. This device is used on a plain screw machine.

The aluminum body *A*, which contains the chuck-closing mechanism, is mounted at the rear of spindle *C*. Sixteen equally spaced steel balls *B* are located in slots formed around the edge of ball-holder *D*. This ball-holder is free to slide forward or backward for opening and closing the chuck, and it is centered on three supports *E* that form part of body *A*.

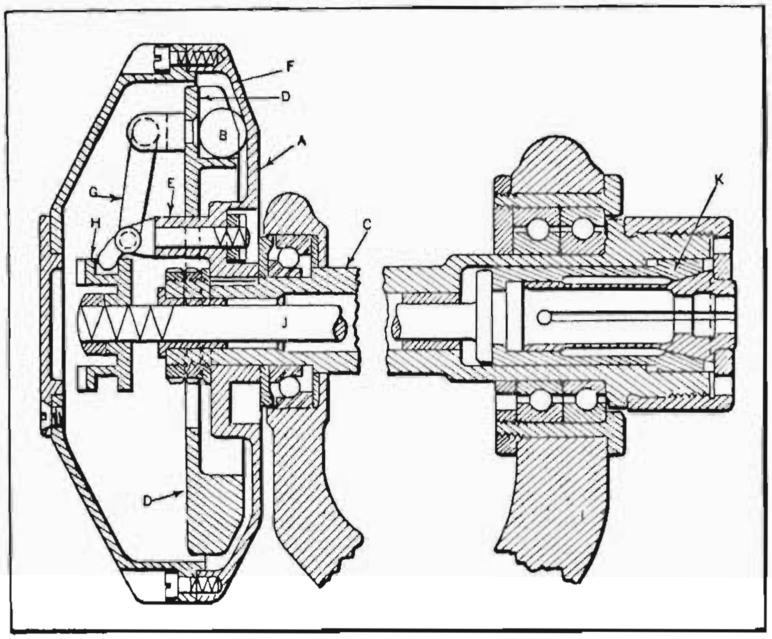


Fig. 6. Mechanism Which Automatically Closes Chuck When Spindle Rotates

The front or chuck end of the spindle is a standard type, and has a collet chuck, as shown. When the spindle and chuck-closing mechanism begin to revolve, balls *B* move outward, due to centrifugal force, and as they engage the inclined surface *F*, ball-holder *D* is pushed backward with a force which increases as the rotary speed increases. This backward movement of *D* causes levers *G*, acting through collar *H*, to push rod *J* and chuck sleeve *K* forward, thus closing the collet

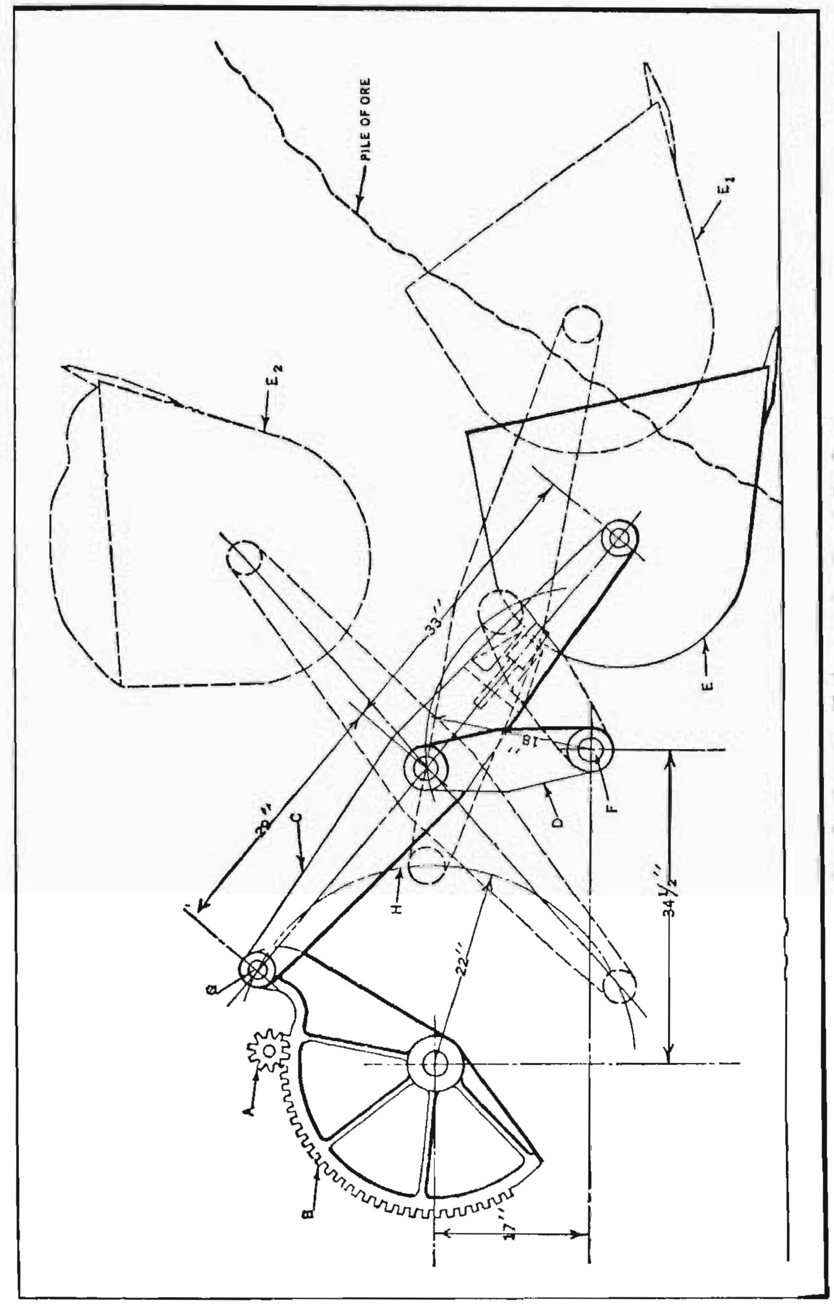


Fig. 7. Scooping Mechanism for Shovel Truck

chuck about the work. When the machine is stopped, the balls return to their inner positions and the spring collet chuck opens. This machine has a friction clutch on the spindle with a foot-treadle which controls starting and stopping.

Mechanical Scooping Motion. — A mechanically reproduced scooping motion incorporated in a truck designed principally for handling lead ore is illustrated by the diagram Fig. 7, which merely shows those parts that are essential to the motion required. Pinion *A*, which is driven by a motor, meshes with the gear sector *B*. Arm *C* is pivoted to one end of gear *B*; the other end is pivoted to bucket *E* and the middle part of arm *C* is pivoted to rocker arm *D*, which swings about a fixed pivot *F*.

This combination, when proportioned according to the dimensions given, provides the required scooping action. The truck is driven forward to locate the bucket close to a pile of ore. Then as pinion *A* turns gear *B*, pin *G* moves downward along arc *H*, and the bucket *E* is forced into the pile, as at E_1, at the same time being forced upward with an efficient scooping action similar to that obtained with the large steam shovels. The dotted lines at E_2 indicate the elevated position of the bucket when loaded.

This mechanism has proved to be a great time- and labor-saver, as the truck loads itself and at the same time lifts the ore high enough so that it can readily be charged into the furnace. The truck is also very compact, so that it can be run into a freight car for unloading. The complete mechanism permits sluing the loaded bucket 90 degrees each way from the central position, so that the truck can be run up a narrow aisle for charging furnaces with the ore.

Power Press Stock Gage. — This stock gage is made part of the press equipment instead of part of the die, as is customary. The mechanical gage finger that has been developed embodies a very simple tripping device. In the diagram, Fig. 8, the device is shown in its operating position on the back of a straight-side blanking press. A hook-block *A* slides in gibs in the cross-arm *L*, and carries on its lower end the

stop-finger *M* secured by the set-screw *N*. The threaded stem *Q* of the slide extends through the stop-lug *H* and carries stop-nuts above and below this lug. The ram of the press *K* carries the trunnion block *J* in which swings the hook *B* on the

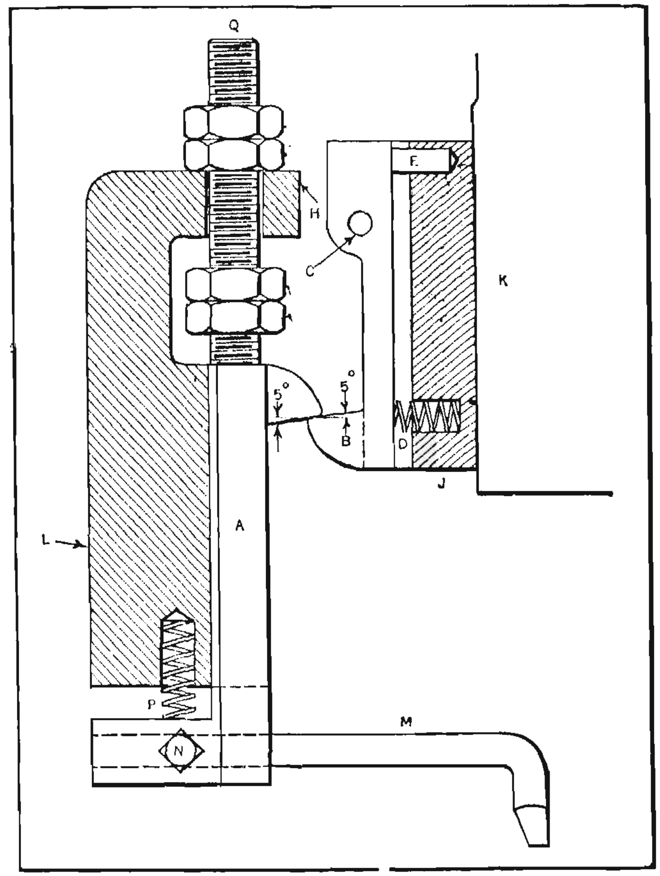


Fig. 8. Stock Gage Which Moves with the Punch After Part Has Been Blanked but Holds Stock During Blanking Operation

cross-pin *C*. Hook *B* is pressed outward by the spring *D* until the upper end strikes the stop-pin *E*. The radius of the tips of the hooks is approximately 1/16 inch.

In operation, the action is as follows: With the sliding hook down, the stop-nuts are against the top of lug *H*, and the tip of the finger *M* rests on the die. The ram *K* descends carrying *J* and *B* with it. The large radii of *A* and *B* engage pressing *B* back against the spring *D* until *B* passes *A*. On the up stroke, the sloping surfaces of *A* and *B* engage and *B* lifts *A* until the stop-nuts strike the under side of the lug *H*, which prevents further movement of *A*. The reaction between the sloping surfaces of *B* and *A* forces *B* and *A* apart, so that they unhook, allowing *A* to drop through the force of

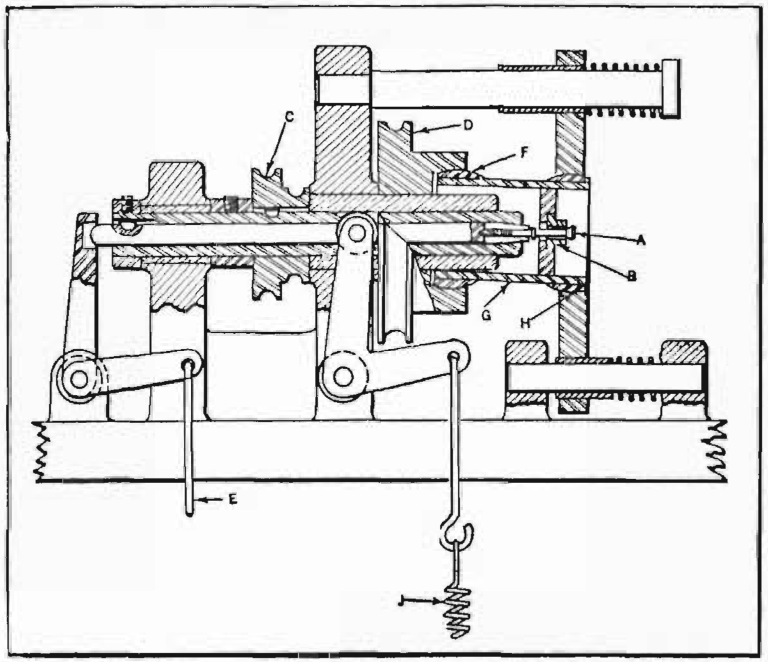


Fig. 9. Machine for Burnishing Ball Valve Seats

gravity and the action of spring *P*. On the next stroke of the press, the action is repeated. It will be noted that the finger *M* moves with the punch after the stock has been blanked, but holds the stock stationary during the blanking. This one feature largely reduced the number of spoiled blanks caused by movement of the stock during blanking.

Eccentric Motion for Burnishing Ball Valve Seats. — This mechanism for burnishing a ball valve seat *B* (see Fig. 9)

causes the valve *A* to rotate about its own axis, and at the same time this axis has a planetary conical motion, the apex of the cone being at the center of the spherical surfaces of the valve and seat; hence, local irregularities are eliminated and the density of the metal increased.

The valve *A* is held by a spindle collet which is closed by a foot-pedal attached at *E*. The valve seat *B* fits into a hexagon socket which prevents it from turning. The spindle pulley *C* turns 1500 revolutions per minute, and pulley *D* revolves 100 revolutions per minute. This pulley *D* has a ball socket *F* bored off center, which gives the axis of sleeve *G* a conical

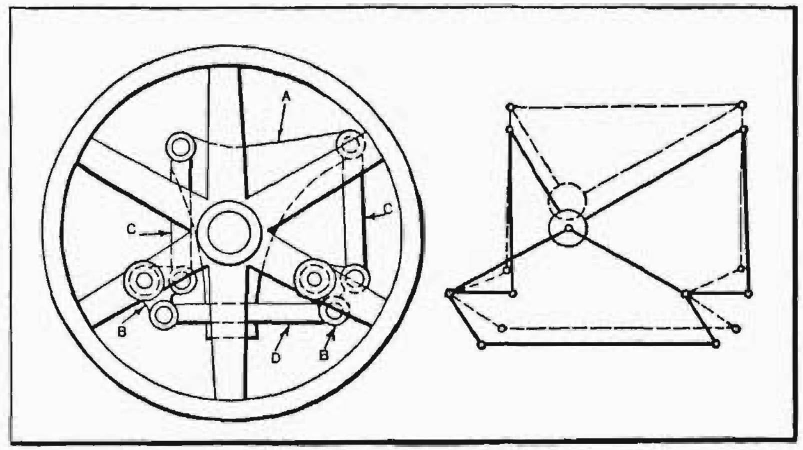


Fig. 10. Shaft Coupling Designed for Lateral or Angular Displacement

motion, the apex of the cone being at point *A* which is also the center of the ball socket *H*. Sleeve *G* does not revolve. A foot-pedal attached to spring *J* is used to hold the valve seat *B* against valve *A* with sufficient pressure to burnish the valve seat properly. Both valves and seats are of brass, and oil is used in burnishing.

Shaft Coupling Rigid in Torsion Only. — The linkage shown in Fig. 10 was devised in order to secure a flexible coupling that would permit a comparatively large and constantly changing amount of misalignment. With this arrangement, misalignment does not cause the coupling to heat or become noisy. Mounted on the driven member so they can rotate are two bellcranks *B*, which are joined by a link *D* as shown, so that they rock in unison. These bellcranks are also connected with the driving spider *A* by links *C*, which should be provided with ball brasses if the angular misalignment is excessive. A line diagram of the linkage is shown at the right in the illustration. The broken lines indicate the position taken by the links and the bellcranks when vertical displacement takes place.

Angular Transmission for Shafts.—If a designer is seeking a means for transmitting power between two shaft ends

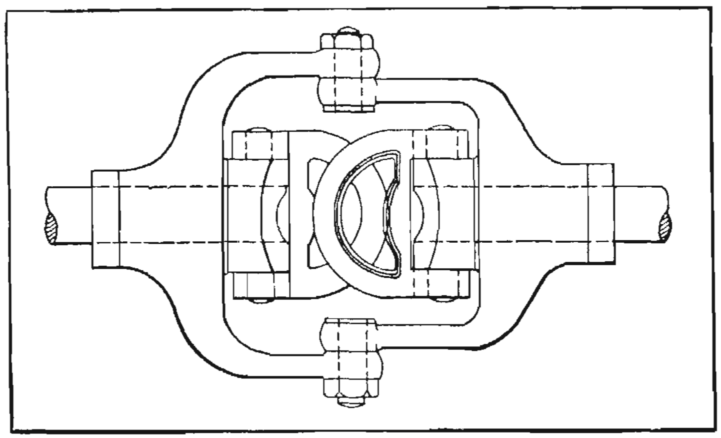


Fig. 11. Transmission with Swiveled Ends

at right angles where the four requirements are positive motion, flexibility, compactness, and quietness of action, what form of mechanical motion will he adopt? A survey of the small group of mechanisms available for angular drives will reveal a rather meager list from which to choose. There are angular transmissions that are positive and compact, but are neither flexible nor quiet. There are transmissions that are flexible and quiet, but neither positive nor compact. It may reasonably be assumed, then, that a description of any new mechanical movement that meets these requirements will be of interest to designers and power transmission engineers.

The use of the Bartlett angular transmission to be described is not confined to right-angle transmissions, but is applicable to any shaft angle from 0 to about 120 degrees, although at the latter angle the contact between the rubbing surfaces would be considerably reduced. If the shaft ends are swiveled, as in Fig. 11, a complete angular sweep of 180 degrees, or 90

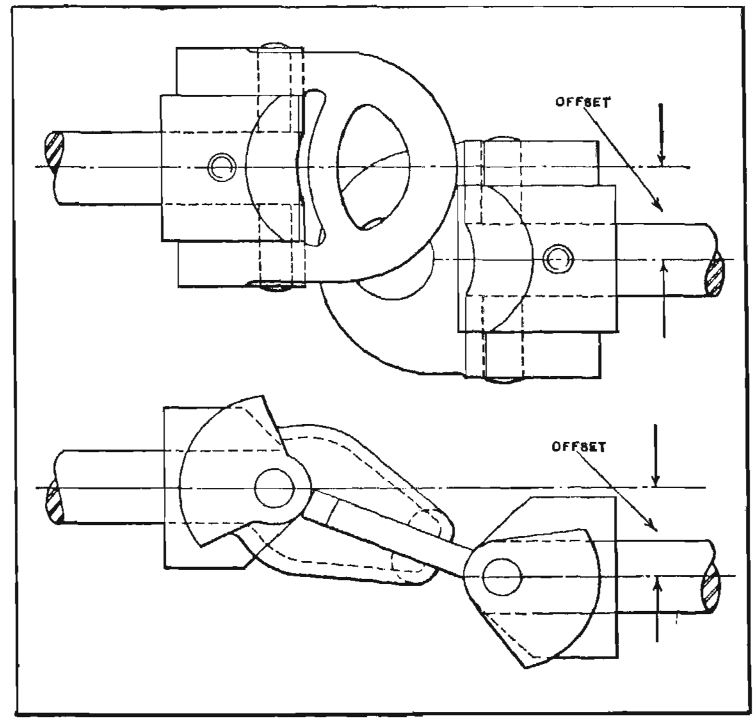


Fig. 12. Diagrams Showing Drive Employed on Offset Shafts

degrees in either direction, is possible; and by adding a ring and a second swivel-pin, a universal joint of a hitherto unattainable magnitude of angular sweep may be produced. Such a universal joint has the unusual property of maintaining a uniform angular velocity ratio of 1:1 between the driving and the driven shafts. It is well known that the common type of universal joint produces an increasing departure from a uniform ratio as the shaft angle increases; for example, with a shaft angle of 30 degrees, Hooke's joint causes a total variation of about 8¼ degrees between the two shafts twice in each revolution, and the actual variation from uniform angular velocity is 28.87 per cent.

Where the shafts are parallel, but considerably offset, this drive can perform the same function as the well-known Oldham coupling, the angular velocity ratio still being uniform. Two positions of the drive when used in this way are shown

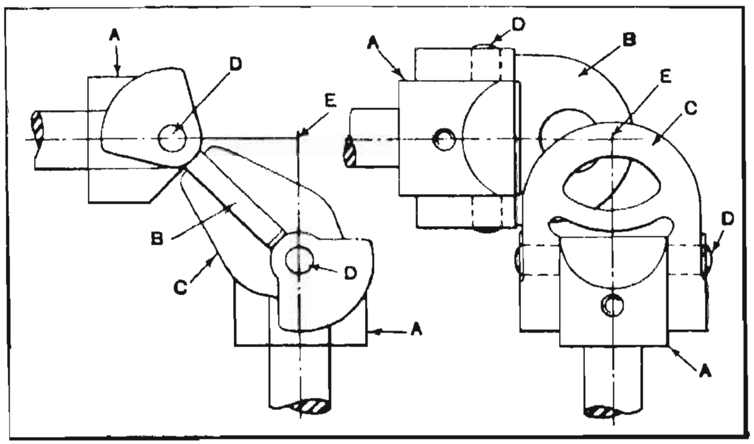


Fig. 13. Transmission Used for Right-angle Drive

in Fig. 12, where the positions in the two cases differ by a quarter of a revolution of the shaft.

When used as a right-angle drive for lineshafting, the drive is connected direct to the shaft ends, which are preferably mounted in self-aligning bearings secured to a special hanger. This hanger provides a rigid connection between the bearings and allows for adjustment in all directions. It also provides a convenient surface on which an oil-case of simple design may be attached.

From Fig. 13 it is possible to obtain a good idea of the construction and action of the parts when the transmission is used for a right-angle drive. The two hubs *A* are keyed to the shaft ends. Each hub carries two hardened steel pins *D* over which the driving members *B* and *C* are free to turn

through an angle of somewhat more than 45 degrees on each side of the shaft axis. The member *C* is of cast iron, semi-steel, or bronze, and is slotted as shown to provide a sliding fit for the member *B,* which is of steel with the working surfaces hardened and ground. The openings in the semicircular parts are for the purpose of reducing the weight, and the extra metal on the opposite side of the pins acts as a counterbalance, but is not needed except for high speeds.

For a perfectly uniform angular velocity ratio, the following conditions must exist:

1. The center line of each pin must intersect the axis of the shaft.
2. The center lines of the pins must lie in the mid-plane between the sliding surfaces of the tongue and slot.
3. The axes of the two shafts must intersect in a point *E.*
4. The center lines of the two pins must be equally distant from the intersection *E* of the shaft axes.

If these four conditions hold, it must follow that for all positions of the sliding members, the center lines of the two pins lie in the same plane, which is the mid-plane of the tongue and slot. Also this mid-plane is always inclined at equal angles with the axes of the two shafts. Hence any angular motion of one shaft must be accompanied by an equal angular motion of the other shaft.

While deviation from the four conditions mentioned will affect the uniformity of the velocity ratio to some extent, it is not necessary to hold to the same degree of precision in mounting these units as would be required for gears.

This drive is actually a flexible shaft coupling applied to shafts whose axes are set at angles varying from 0 to 90 degrees. When the shafts are not in perfect alignment, the bearings are relieved of undue stress, and the action is smooth and quiet. Shaft misalignment is not possible with gears which mesh properly, nor, in general, with most other types of angular drives.

Water-operated Automatic Switching Mechanism. — In making steel wire, a billet of the proper size is heated and reduced in cross-section in a series of mill stands. The steel emerges from the last or finishing stand in the form of a small rod which is several thousand feet long and moves at a speed of 25 miles per hour or more. This rod is guided through pipes to a horizontal reel, the speed of which is automatically regulated according to the speed of the finishing stand, and this forms the rod into a coil. The coiled rod is removed from the reel by a suitable mechanism and then reduced to a wire by cold-drawing through a series of dies.

In order to obtain high efficiency, the rods follow one another very closely; in fact, there is only a space of a few feet

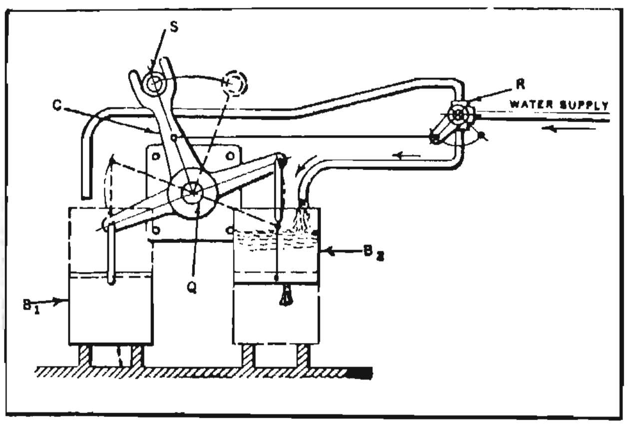


Fig. 14. Mechanism of Automatic Switching Device

between the rear end of one rod and the forward end of the next rod. Since it would be impossible to clear the reel in these short time intervals, it is necessary to employ two reels and coil the rods alternately on them. This requires a switching device which directs the rods to the two reels in alternation.

Before describing the switching device, it might be well to summarize the conditions: (1) The rods are white-hot; (2) they move at a high rate of speed; (3) the time available for the switching device to act is very short, and the action must be instantaneous; (4) the rods are of different lengths, so that switching must occur at irregular intervals; (5) the

device must possess a maximum degree of reliability, because its failure would cause considerable danger and a decrease in production. The device to be described meets all of these conditions in a very simple manner.

Pipe S (see Fig. 14) can swing around a fulcrum in such a manner that while one end always registers with a pipe leading to the finishing stand, the other end will register with either a pipe leading to the right-hand reel or with another pipe leading to the left-hand reel. Bellcrank C can swing around stationary pin Q, moving pipe S into either of the two necessary positions. A three-way valve R is operated by the bellcrank. Parts B_1 and B_2 are buckets, each having a hole in the bottom. Water flows into each bucket alternately, in a volume exceeding that of the outflow at the bottom, so that the water level in the bucket gradually rises.

In the position shown in full lines, bucket B_2 is receiving water and becoming slowly heavier, while bucket B_1 is emptying itself and becoming lighter; hence the bellcrank tends to swing into the dotted position, thereby reversing the three-way valve, which diverts the stream of incoming water to bucket B_1 and allows bucket B_2 to become empty.

Now the only obstacle to such a motion of the bellcrank is furnished by the rod which, running from pipe S to whichever reel pipe it is in line with, bridges the gap between the two pipes and acts, so to speak, as a splice. But at the instant that the tail end of the rod leaves pipe S, the motion of the bellcrank takes place. This action is, of course, reversed when the tail end of the next rod leaves pipe S. The amount of water is so regulated that the scraping action on the side of the rod is slight and therefore not injurious. This device has provisions for manual control in case of emergency.

Mechanical Lapping Mechanism for Plane Surfaces.—The mechanical movements required in accurately lapping the parallel surfaces of small brass rings are obtained from the mechanism shown in Fig. 15. This mechanical lapping process enables one man with one machine easily to lap 2500 rings a day to within 0.0002 inch of parallelism.

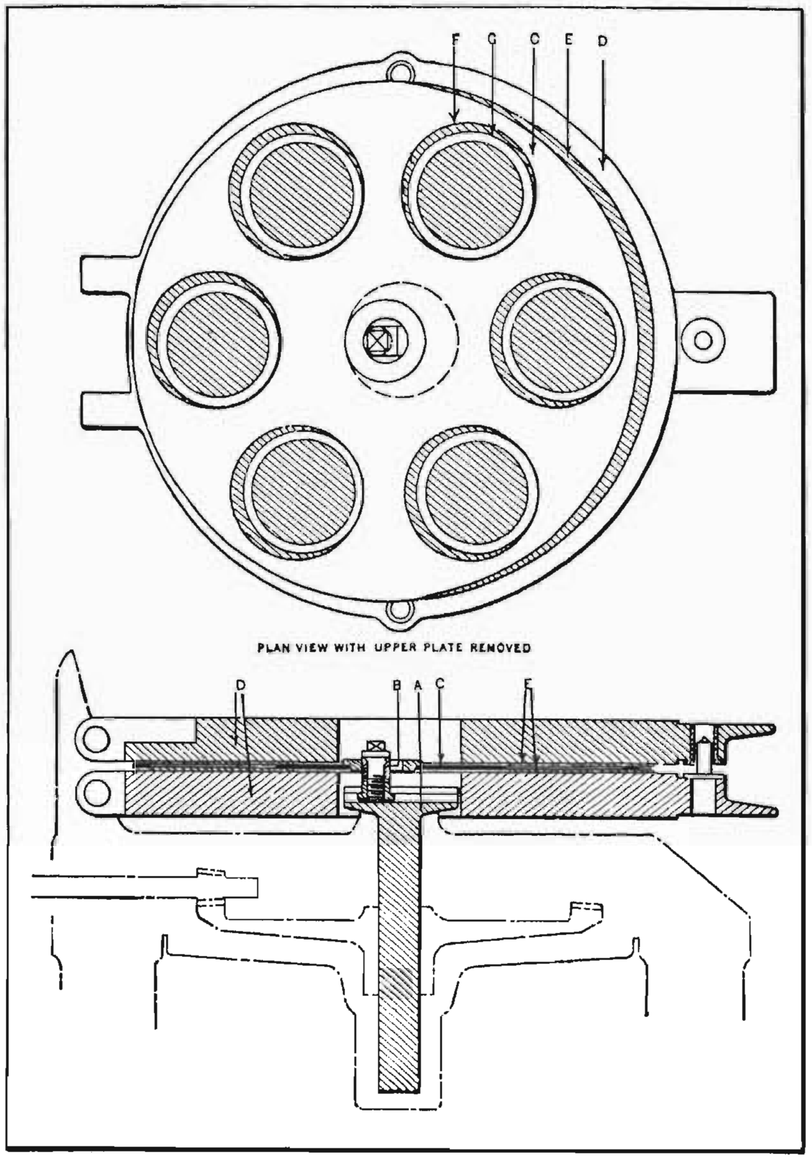


Fig. 15. Mechanical Lapping Mechanism for Finishing Small Brass Rings

Referring to the illustration, a vertical shaft *A* is rotated at 150 revolutions per minute. An adjustable eccentrically located roller *B* on the upper end of the shaft engages a hole in a driving disk *C*, the hole being about half again as large as the roller. Thus the disk is oscillated in a horizontal plane 150 times a minute. As the roller is smaller than the hole in the disk, and therefore assumes a position eccentric to the disk, and as the point of eccentricity is continuously progressing around the circle, due to the inertia of the unequally divided mass of the disk, the disk itself assumes a slow motion of rotation.

The driving disk oscillates and rotates between two plates *D*, which are nominally stationary, but which have a certain amount of freedom in all directions. These plates are covered, on adjacent faces, with disks of abrasive cloth *E*. The driving disk has openings *F* cut in it to receive the rings *G*, these openings being on a circle which is of the same diameter as the mean diameter of the abrasive disks.

When the roller is adjusted to the proper eccentricity for the size of the rings, the oscillation of the driving disk will pass the rings over the outside and inside edges of the abrasive disks an equal amount. The openings in the driving disk are a little larger than the rings, so that the latter are free to rotate around their own centers. They do this because the path around the plates, of any points on the outside edges of the rings, is greater than that of any points on their inside edges; therefore the lapping action is equalized as the rings twist around their own centers.

Thus, by an ingenious adaptation, a simple crank translates to a number of work-pieces, at one time, an oscillatory motion, as well as motions of rotation around both their individual and common centers, and a complication of mechanical movements is obtained, without precise construction or skilled attention, which produces precision results—a good example of simplification in machine design.

Pantograph Mechanisms for Reproducing Motion on a Different Scale. — A pantograph is a combination of links which are so connected and proportioned as to length that any motion of one point in a plane parallel to that of the link mechanism will cause another point to follow a similar path either on an enlarged or a reduced scale. Such a mechanism may be used as a reducing motion for operating a steam engine indicator, or to control the movements of a metal cutting tool. For instance, most engraving machines have a pantograph mechanism interposed between the tool and a tracing point which is guided along lines or grooves of a model or pattern. As the tracing point moves, the tool follows a similar path

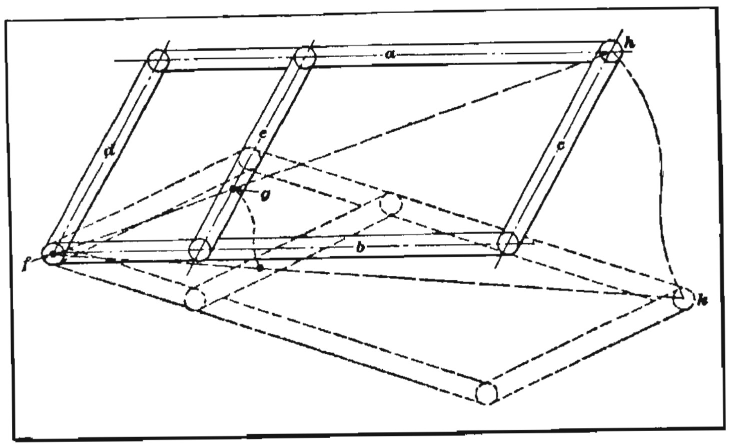


Fig. 16. Pantograph for Reproducing Motion on a Reduced or Enlarged Scale

but to a reduced scale, and cuts the required pattern or design on the work.

A simple form of pantograph is shown by the diagram, Fig. 16. There are four links, *a*, *b*, *c*, and *d*. Links *a* and *b* are equal in length, as are links *c* and *d*, thus forming a parallelogram. A fifth connecting link *e* is parallel to links *c* and *d*. This mechanism is free to swivel about a fixed center *f*. Any movement of *h* about *f* will cause a point *g* (which coincides with a straight line passing through *f* and *h*) to describe a path similar to that followed by *h*, but on a reduced scale. For instance, if *h* were moved to *k* following the path indicated by the dotted line, point *g* would also trace a similar path.

Another form of pantograph mechanism is shown at *A* in Fig. 17. This pantograph, which is sometimes called "lazy tongs," is used to some extent for obtaining the reduction of motion between an engine cross-head and the indicator drum when taking indicator cards. The pantograph is pivoted at *b* by a stud which may be secured to a block of wood or angle iron attached to a post or in any convenient place. The end

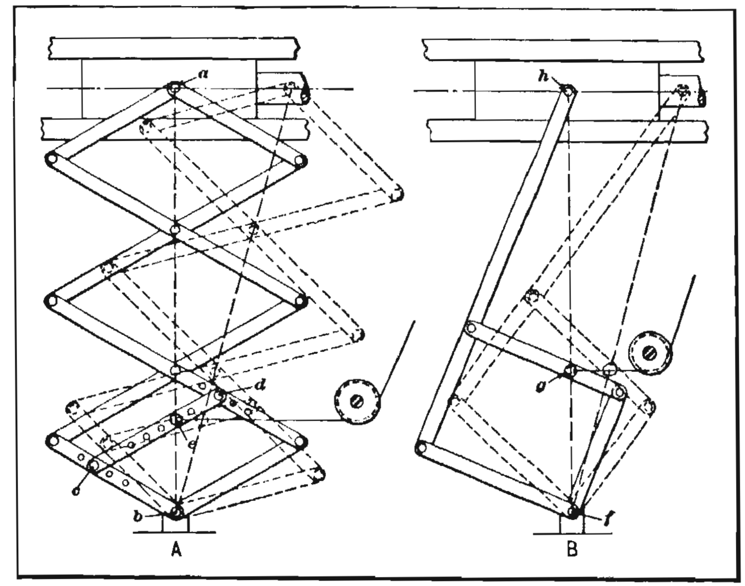


Fig. 17. Pantograph Mechanisms Applied to Engine Cross-head to Reduce Motion When Taking Indicator Cards

a has a pin which is connected to the cross-head of the engine. The cord which transmits motion to the indicator drum is attached to the cord-pin *e* on a cross-bar. This cross-bar may be placed in different positions relative to the pivot *b*, by changing screws at *c* and *d;* the cord-pin *e*, however, must always be in line with the fixed pivot *b* and pin *a*. The position of the cross-bar in relation to pivot *b* determines the length of the travel of cord-pin *e* and, consequently, the rotary movement of the indicator drum and the length of the diagram which the pencil traces upon the indicator card. The objection to this reducing mechanism is the liability of lost motion resulting from wear in the numerous joints.

The pantograph reducing mechanism shown at B in Fig. 17, has four links joined together in the form of a parallelogram, and one of the links is extended and pivoted to the engine cross-head. The swiveling movement of the pantograph is about the fixed pivot f, and the cord which operates the indicator drum is attached at g. As the illustration indicates, this point of attachment g coincides with a line passing through the pivots f and h, the same as for the pantograph shown at A. If F = the length of the engine stroke and L, the length required for the indicator diagram,

$$F : L = fh : fg, \quad \text{or} \quad \frac{F}{L} = \frac{fh}{fg}.$$

Action of an Adding Mechanism.— The adding mechanism to be described is applied to a machine which is a typewriter and adding machine combined. With this machine debit and credit accounts may be written down indiscriminately; each set of items added, and the total amount printed beneath each vertical column. The writing is done on the typewriter in the regular way, and the figures are set up and printed with the adding mechanism at the same time that the reading matter is written. Two adding mechanisms or "accumulators" are required, one being for the debit and the other for the credit column. This machine may also be used in various other ways. For instance, a list of items may be printed in a series of vertical columns and these columns added to obtain the total amount in both horizontal and vertical directions; finally, these totals, both horizontal and vertical, may be added together to obtain the grand total. Discounts may also be reckoned, amounts may be subtracted from each other, and many other operations performed in connection with commercial work.

The adding keyboard is composed of nine vertical rows of nine keys each. The lower key of each row is numbered one, the next two, and so on up to nine. Of the nine vertical rows, the first on the right is for units, the next, for tens, etc., or, since the reckoning is usually in dollars and cents, the first row is for cents, the next, for dimes, and the succeeding rows, for dollars. Figs. 18 and 19 show diagrammatical sections through the machine along the line of any one of the vertical rows of adding keys, which are shown at G. Other important parts of the mechanism are the rack A, the type sector F, by means of which the numbers are printed on paper carried by roller K, and the accumulator wheels B, by which the addition is performed. These parts, as well as the other moving mechanism shown, are duplicated for each one of the nine rows of keys, there being nine racks, nine type bars, nine sets of accumulator wheels, etc., in all.

The adding mechanism is operated by the movement of rack A. This movement takes place under the influence of spring O whenever stop N is swung back as shown in Fig. 19 by the operation of the handle of the machine. The length of the movement which spring O thus gives to rack A is determined by keys G. If the figure $476.34 is set up on the keyboard, for instance, key "4" will be depressed in the cents column, and, when the movement of the rack takes place, the rack teeth beneath the accumulator wheel will have moved four spaces.

This is more clearly seen in Fig. 18 where each of the keys G is shown to be mounted on a stem which carries a stop H at the lower end. When the keys are depressed, these stops come into line with corresponding steps formed at the left-hand end of the rack A. These steps are so proportioned that when key 1 is depressed, for instance, the rack is allowed to move one tooth before striking its abutment. When key 2 is depressed, it moves two teeth and so on, as shown by the numbered arrows at the lower part of the illustration. When no key is depressed, indicating zero, then a stop is interposed which prevents any movement of the rack. When key 9 is depressed, the rack takes the full movement of nine teeth allowed by the striking of the projections on the under side of the rack against supporting bar J.

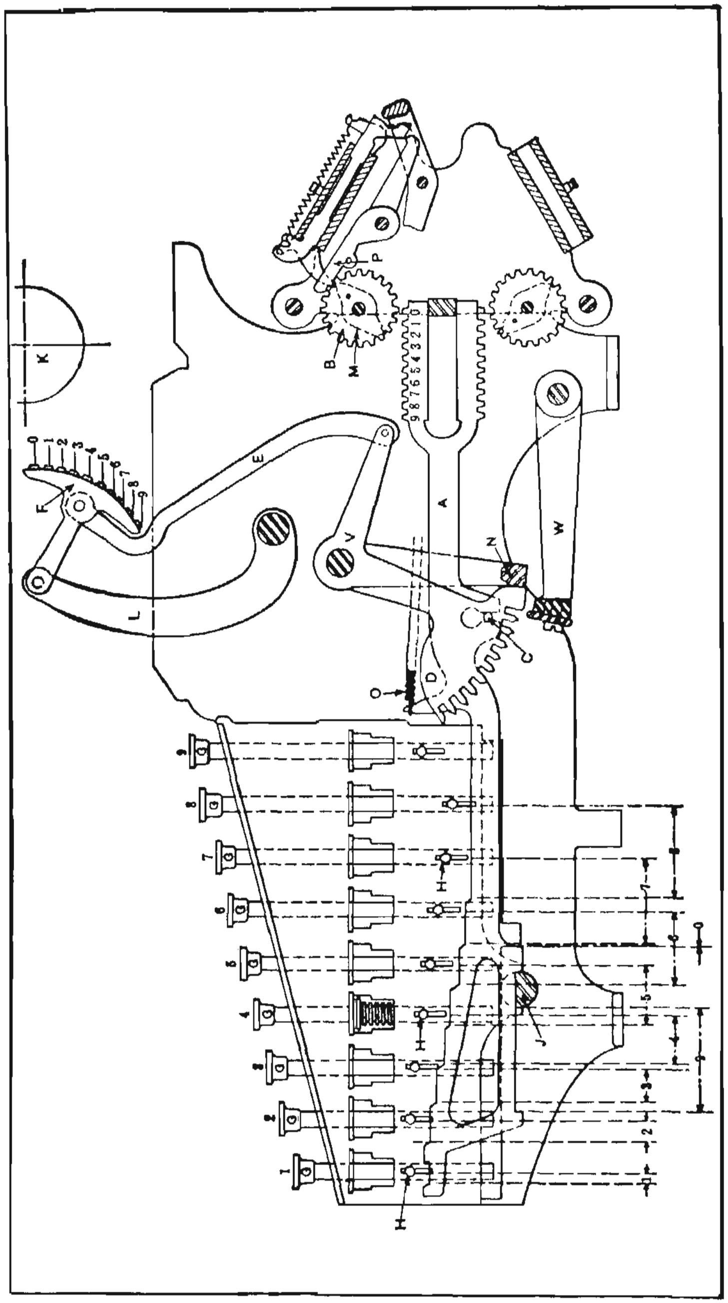


Fig. 18. Diagrammatical Cross-section of Adding Mechanism Showing Relation of Keys, Racks, Accumulators, and Printing Device

Each rack A has cut in it a slot engaging pin C in sector D. Each sector is, in turn, connected by link E with the type bar F having numbers from 0 to 9. Whenever a key (key 4, for instance) is depressed as shown in Fig. 19, and the rack is allowed to move four teeth backward under the influence of spring O, the type bar F is thereby set at the corresponding figure. The throwing forward of lever L to which the type bar is pivoted then prints this figure "4" on paper wrapped about roll K. It is important to remember that rack A and type bar F are positively connected under all conditions. It should, perhaps, be mentioned that the teeth in sector D simply provide for more accurate alignment of the type in printing than would otherwise be possible. Just before the printing stroke takes place, arm W swings up, carrying a plate which enters the corresponding tooth space in each one of the nine sectors D, aligning all the figures on type bars F and giving a good, evenly printed number on paper.

The Accumulator Mechanism. — The accumulator mechanism, by means of which the adding is done on the machine previously referred to, will now be described. There are ordinarily nine accumulator wheels for each of the nine racks. This particular machine, however, has two sets of nine wheels each, one set being above rack A (see Fig. 19), and the other below it. The upper one is the debit accumulator for addition in the debit column and the other is the credit accumulator for the credit column. Only the upper or debit accumulator will now be considered. This set of nine accumulator wheels, of which only one is shown at B, may be swung into and out of engagement with the teeth of racks A, at will. These accumulator wheels have 20 teeth each; they could have ten, except for the fact that it would make them inconveniently small. Each wheel is provided with a two-tooth ratchet M positively pinned to it. This ratchet spans ten of the wheel teeth between its points. Pawl P is adapted to engage the teeth of ratchet M, and is connected with the mechanism by means of which the tens are carried from one column to another (that is, from one accumulator wheel to another) as

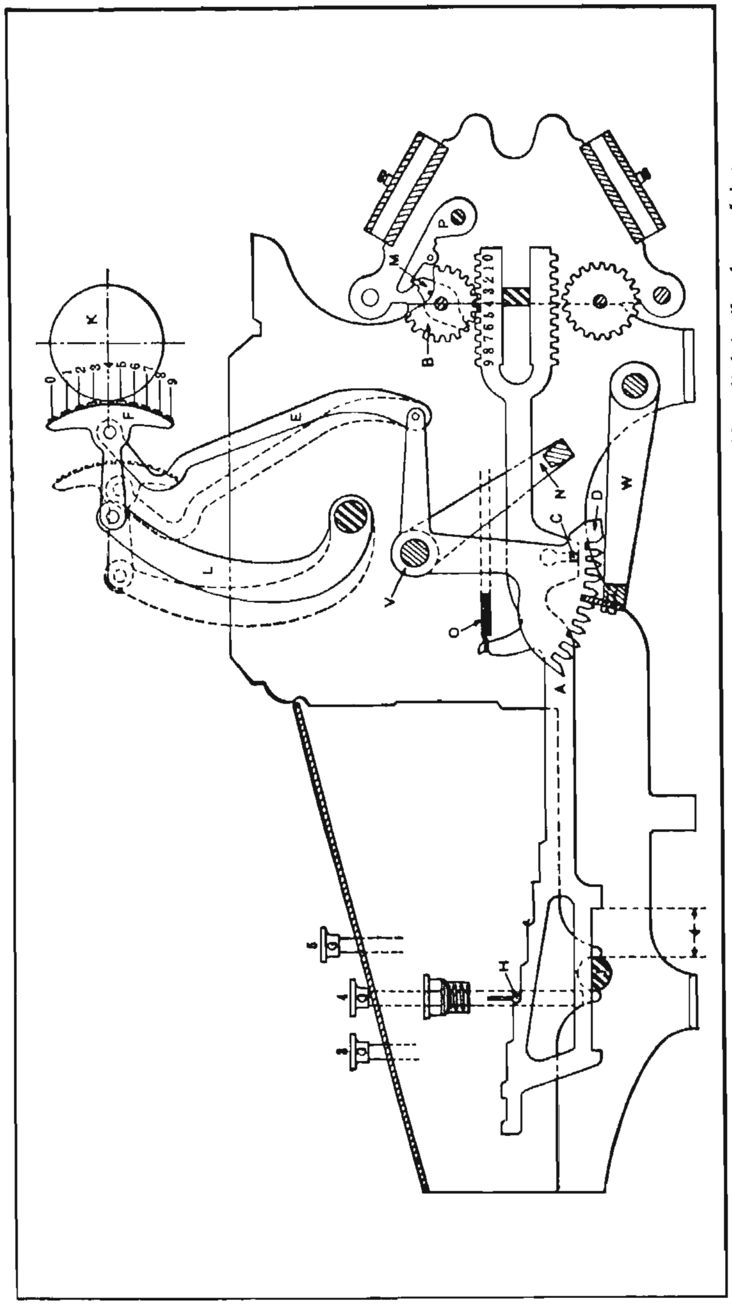


Fig. 19. Adding Mechanism Arranged for Printing the Number 4 and Adding it into the Accumulator

will be described in connection with diagrams Figs. 20 and 21.

Order of Operations for Adding. — Figs. 20 and 21 show, in diagrammatical form, the method of procedure followed in the simple problem of adding 4 to 9, and obtaining the sum 13. At *A*, Fig. 20, the machine is shown "clear," that is, with the accumulator wheels at zero, which means that one tooth

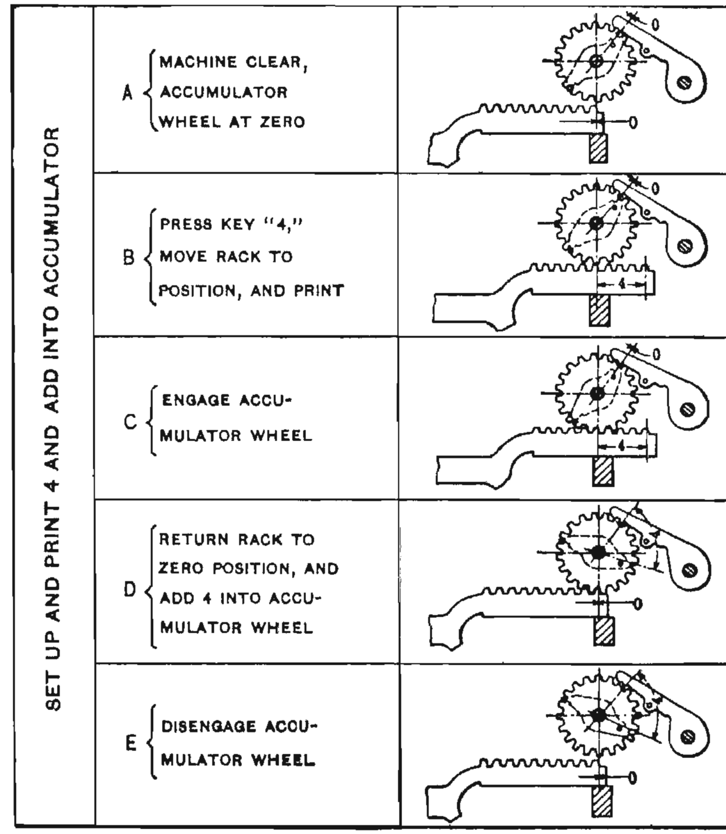


Fig. 20. Diagrams Illustrating Action of Adding Mechanism

of the two-tooth ratchet is up against the hook of the pawl. Key 4, corresponding to the number to be added, in this case, into the accumulator wheel, is now depressed and the operating handle of the machine is pulled over. The first thing that takes place is that the rack is allowed to move four teeth

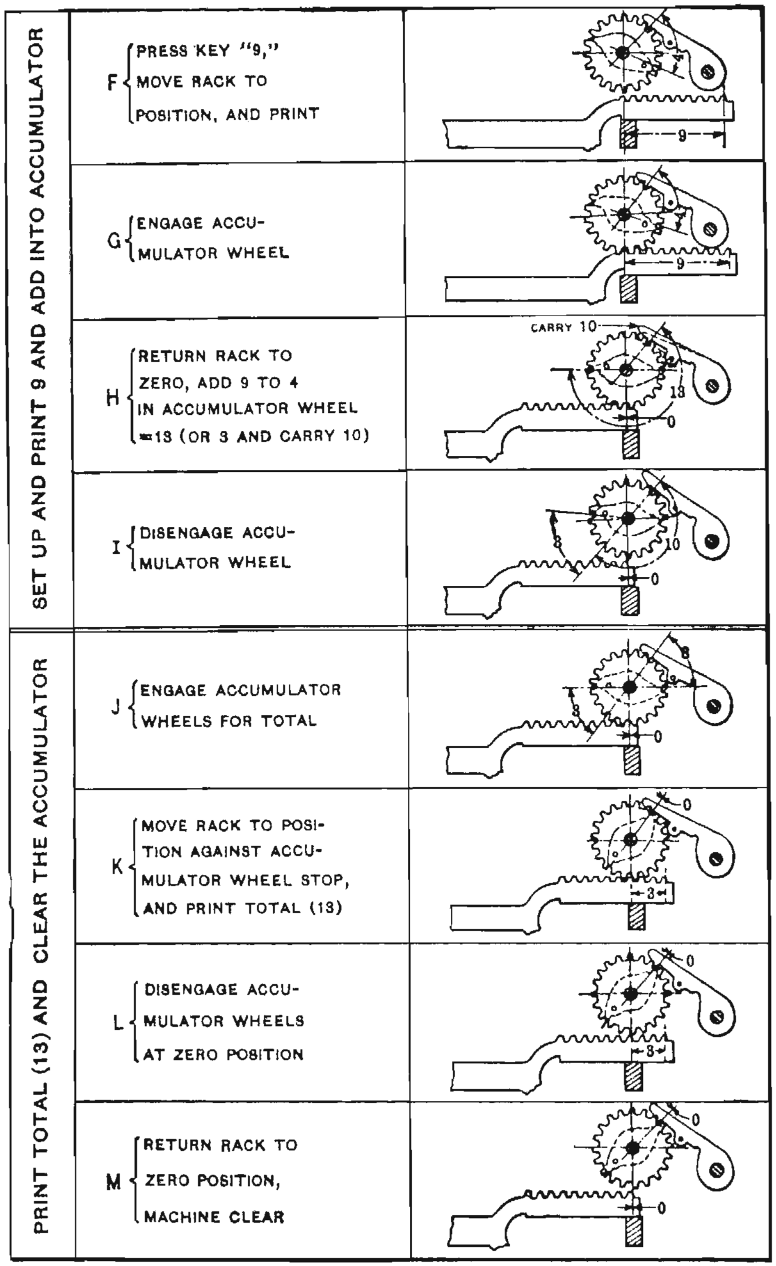


Fig. 21. Continuation of Diagrams Illustrating Operation of Adding Mechanism

to the right, as shown at *B* (see also Fig. 19). In this position, the number "4" is printed. Next (as shown at *C* in Fig. 20) the mechanism automatically throws the accumulator wheel down into engagement with the rack. Then as the operator allows the handle to return, the rack moves back to the zero position again as shown at *D*, carrying the accumulator wheel with it a space of four teeth from its zero position. The mechanism then disengages the accumulator wheel, leaving the machine ready for the next operation with the 4 added into the accumulator, as shown at *E*.

To add 9 to the 4, key 9 is depressed and the operator pulls the handle. This results in a movement of nine teeth of the rack as shown at *F* in Fig. 21. The figure 9 is then printed. The accumulator wheel is next engaged, as at *G*. Then the rack is returned to the zero position as at *H*, and the accumulator wheel is disengaged as at *I*. This evidently moves the accumulator wheel $9 + 4 = 13$ teeth as shown at *H*. In doing this, one of the teeth of the two-tooth ratchet lifts the pawl as it passes under it. This raising of the pawl operates a spring-loaded mechanism, which shifts the next accumulator wheel (that for the tens column) one tooth, when the wheels are returned from engagement in operation *I*. This operation corresponds to that of "carrying" when adding with pencil and paper, except that it is done automatically. This carrying mechanism will not be described in detail as the parts are small and rather complicated, although the action is simple. The mechanism may be understood more clearly by considering the actions of the wheels when every one of them in the accumulator, from cents up to the millions of dollars, is set at 9 — that is, when they are set up for 9,999,999.99. Now suppose that one cent is added, so that the first wheel is moved beyond 9 — that is, to 0. The tooth of the ratchet *M* will then pass under the first pawl, raising it. When the accumulator wheels return from engagement, this raising of the first pawl releases a spring-loaded mechanism which moves the next wheel from 9 to 0. This, in turn, moves the next wheel from 9 to 0 and so on until each one of the row has been advanced one tooth, setting the whole row at 0,000,000.00. This operation is done so rapidly that one cannot distinguish between the successive operations, but each one is dependent upon the preceding one. The operations required for finding a total are shown at *J, K, L,* and *M,* Fig. 21. The first thing the operator does is to depress the "debit total" key at the left of the keyboard, the sum having been added into the upper or debit accumulator. He then pulls the operating handle, and the accumulator wheels are engaged with the racks as shown at *J*. The next operation is the release of the racks so that the springs move them toward the right. There are, in this case, no keys depressed in the keyboard, so that the racks would move the full distance of nine teeth, were it not for the fact that they have to carry the accumulator wheels with them, and the ratchets on these wheels come in contact with the pawls, thus arresting their movement and stopping the movement of the racks.

The previous operation of adding 9 to the 4 in the wheel set the "units wheel" three teeth beyond the point of the ratchet, and the "tens wheel," one tooth beyond the point of the ratchet. It is evident, then, that in operation *K* the units rack will be allowed to move three teeth and the tens rack one tooth. This will evidently set up the unit type bar at "3" and the tens type bar at "1." On the return of the handle, the printing mechanism is operated, transferring the total "13" to the paper. The accumulator wheel will then be released, and the rack will be allowed to return to the zero position as shown at *M*. This leaves all the accumulator wheels back in the zero position, with the teeth of the ratchets back against the pawls, leaving the machine "clear" and ready for the next operation.

It might have been desired to print a sub-total instead of a total; that is, a total for the addition as far as it had proceeded, but not to clear the machine, thus permitting more figures to be set up and printed and added into the same sum. Sub-totals can be printed at any point in the adding up of a line of figures, as required, by a simple change in the operation shown at *J, K, L,* and *M* in Fig. 21. This consists simply in allowing the wheels to remain in engagement at *L,* so that the racks, when they return in operation *M,* will bring the wheels to the same position as they had in *J,* thus leaving the totals still set up in the accumulator. Since there are two independent accumulators, it is evident that a number can be added into either one or both of them; or a total or sub-total can be taken from one of them and added into the other — all depending upon the manipulation of the keys and the time of throwing the accumulator wheels into and out of action.

This adding machine has what are known as "controlling keys." These are named "non-add," "debit add," "debit sub-total," "debit total," "credit add," "credit sub-total," "credit total," "repeat," and "error." The pressing down of the non-adding key permits the printing of a number without adding. In other words, this keeps the accumulators permanently out of engagement with the racks. The debit and credit add keys permit a number to be printed and added into the corresponding accumulator, even though the carriage is not set in the proper position for that accumulator. The use of these keys, therefore, gives a flexibility to the machine which is necessary for special operations such as horizontal adding. The debit and credit sub-total keys take and print a total from either the debit or credit accumulators without clearing the accumulators. The debit and credit total keys, on the other hand, take the total from either the debit or credit accumulators, as the case may be, and clear the accumulator after the total is printed. The pressing down of the repeat key holds in the downward position whichever of the number keys have been depressed, allowing the same number to be repeatedly printed and added as many times as the operating handle is pulled. This is useful in multiplying by repeated additions and for other similar uses. The pressing of the error key will release every other key on the keyboard, both of the number keys and of the operating keys as well.

The keyboard is provided with an interlocking mechanism connected with the controlling keys of the machine and with the operating lever. This mechanism, among other things, prevents the keys from being pressed down or changed after the operating lever movement is started. The keyboard also has a connection with an error key, the pressing of which releases all the keys that may be depressed at the time. Means are also provided for automatically releasing and returning the keys after each operation.

Accumulator Controlling Mechanism. — The engagement of the accumulators with the racks, and their release, in the

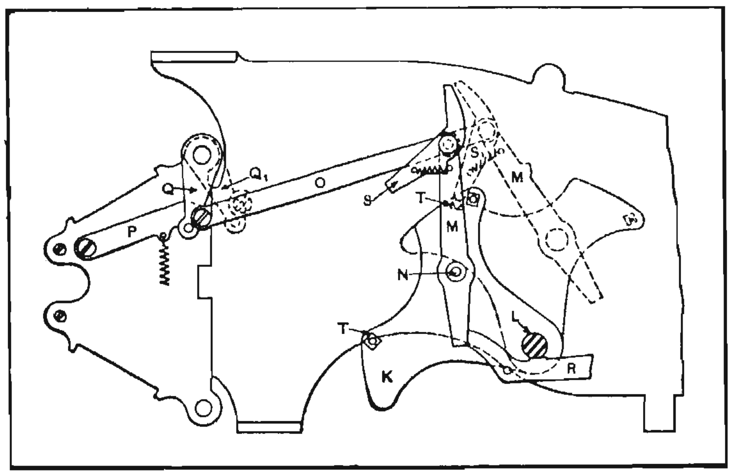


Fig. 22. Flying Lever Connection Between Operating Shaft and Accumulators of Adding Mechanism

operation of the adding mechanism previously described, is effected as follows: The sector K (see Fig. 22) is directly connected with the operating shaft L controlled by the operating handle. It is provided with connections with both accumulators, although this illustration only shows the connections with the debit accumulator. Flying lever M is connected with the debit accumulator by means of links O_1 and bellcrank Q. Member P is simply a spring detent to locate Q for either the engaged or disengaged position of the accumulator wheels.

As sector K starts on its stroke toward the dotted position, flying lever M is carried with it, owing to the resistance which

the end of the latter meets with against abutment R. When K has gone far enough so that the end of the lever has dropped off R, the lever M becomes free. The movement has been sufficient, however, to move accumulator lever Q to position Q_1 which throws the wheels into engagement. If it had been desired to throw the wheels into engagement at the end of the stroke instead of at the beginning, detent R would have been withdrawn from the position shown, leaving flying lever M free. Near the end of the stroke of K, however, the end of the pawl S would have struck stud T, making M and K solid, for all practical purposes, and moving Q to the position Q_1

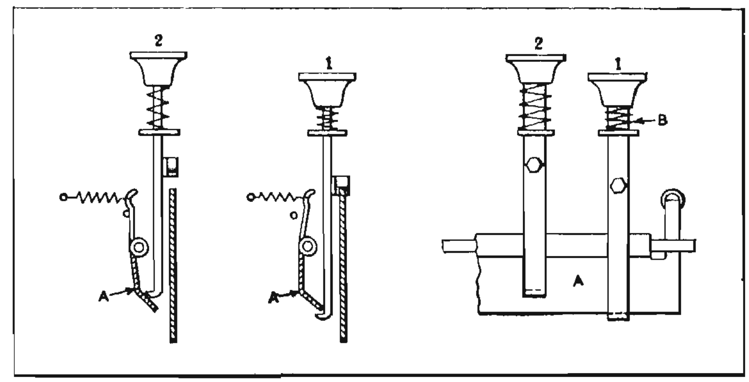


Fig. 23. Simple Arrangement for Holding in the Downward Position Only One Key at a Time in a Row of Adding Machine Keys

at the end of the stroke. If it had been desired to keep the accumulator wheels out of engagement altogether, R would have been lowered out of the position shown, and S would have been moved to a position clear of stud T. Then flying lever M would have been entirely free of K, and no movement of Q would have taken place. The provisions for throwing the accumulator out of engagement at either the commencement or end of the return stroke are similar to those just described.

Adding Machine Key Control. — The keyboard of an adding machine is said to be "flexible" when so arranged that, if a key has been depressed, it will stay down, but the pressing down of another key in the same vertical column will release the first key. With this arrangement, if an attempt were made to depress two keys successively, the releasing of one by the downward action of the other would eliminate a possible error. As a further advantage, if the wrong key were pressed, the depression of the right one restores the wrong one to its normal position. The simple, but ingenious, device for controlling the action of the keys on one of the commercial adding machines is illustrated in Fig. 23. If key No. 1 is depressed, the lower hooked end of the stem on which it is mounted springs past the end of a long pivoted strip *A* that extends throughout the entire length of the vertical row of keys. The result is that the key is held in the downward position by this hooked end until some other key is depressed. For instance, if the operator presses down on key No. 2, this will swing the strip *A* about its pivot to allow the hooked end of the stem to pass, and this movement of strip *A* releases the hooked end of key No. 1 which immediately is forced upward to its normal position by a spring *B*. In the same manner, any key which may be pressed down will throw back the strip and release any other key which may at the time be depressed.

CHAPTER XV

HYDRAULIC TRANSMISSIONS FOR MACHINE TOOLS

Within recent years many standard machine tools have been designed with hydraulic feed mechanisms built in, as a part of the machine. Among the first tools to be so equipped were broaching machines. The production capacity, flexibility of control, and low maintenance cost of these hydraulically operated tools attracted the attention of many machine tool builders and users. Later, grinding machines and drilling machines were successfully equipped with hydraulic feeds. Following this, several lathe and chucking machine manufacturers and builders of milling machines began the development of hydraulically equipped machines.

In practically all new applications of hydraulic transmission to machine tools, no accumulators are employed. Thus, the new system of feeding or driving consists essentially of an oil-pump and a cylinder having a piston driven by oil circulated by the pump and controlled by piping and valve equipment, to give the piston any movement required for feeding or driving. Where hydraulic rotary drives have been applied, a motor similar in construction to the rotary oil circulating or driving pump takes the place of the cylinder and piston arrangement.

Although the basic principle of hydraulic operation of feeds and drives appears simple, the actual development of a practical system involves considerable engineering. The requirements of one machine may be met by comparatively simple equipment, whereas the hydraulic operation of another machine may require a system of piping, specially designed control valves, two or more driving pumps and hydraulically driven devices of various kinds.

Advantages of Hydraulic Operation. — The following is a brief resumé of the principal advantages claimed for hydraulic, as compared with mechanical, operation of machine tool feeds and drives.

1. Straight line or rotary transmission of power at any desired point of application.
2. Higher cutting speeds.
3. Longer life of cutting tools.
4. Greater flexibility of speed control.
5. Quick reversal of feed, with practically no shock.
6. Simple and efficient control, both hand and automatic, of all rapid traversing, feeding, and reversing movements.
7. Quiet operation.
8. Low power consumption, which varies automatically to meet resistance offered to cutting tool or driven member.
9. Safety insured by relief valves, which can be set to stop the feeding movement at any predetermined pressure.
10. Ability to "stall" against obstruction, thus protecting parts against breakage and providing an ideal method of cutting shoulders to exact positions and facing to length, by using positive stops.
11. "Slip," which permits movement to slow up when tool is overloaded without "windup" of mechanical feed gear.
12. Fewer moving parts.
13. Provision for checking and comparing action or condition of cutting tools by pressure gage, which indicates cutting force.
14. Adaptability for operating auxiliary devices, such as work-holding clamps, clutches, diamond dressing tool, indexing pins, etc.
15. Comparatively simple centralized control over one or any number of hydraulically operated units arranged or located according to requirements.
16. Greater flexibility of design which permits feed to be adjusted to most efficient rate after machine is assembled.

17. Reliability and low up-keep cost due to comparatively few wearing parts.

Two Fundamental Circuits Employed. — The designs of practically all hydraulic feeds and drives recently applied to machine tools have been based on one of the two fundamental hydraulic circuits shown diagrammatically in Figs. 1 and 2, or a combination of these two circuits. In both circuits, the pistons *A*, which impart the required feeding movements, are driven by a liquid delivered from their respective pumps *B*. The pressure is low if the piston encounters no resistance, and the distance it is moved corresponds to the discharge rate of the pump, whether the resistance be high or low.

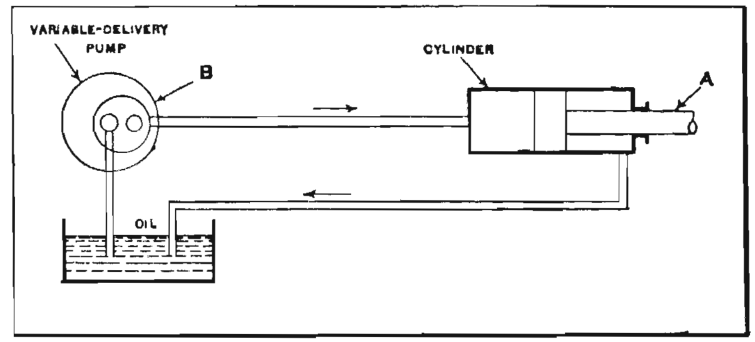


Fig. 1. Hydraulic Feed with Closed Circuit

In the case of the circuit shown in Fig. 1, the rate of feed is changed by varying the volume of liquid delivered by the pump *B*. With the circuit shown in Fig. 2, the feed is controlled by opening or closing the choke valve *C*, causing more or less of the liquid from the constant-volume pump *B* to be by-passed by valve *D*, but admitting liquid to cylinder *E* at the rate necessary to give the required feed.

Pressures Used for Feeds and Drives. — Both high- and low-pressure systems are used, depending on the type of machine and its requirements. For instance, the reciprocating tables of light weight internal grinders may be driven by constant-displacement low-pressure gear pumps, as comparatively little pressure is required to move the tables of such machines. On the other hand, a certain car-wheel boring machine is equipped with four variable-delivery pumps, each having a capacity of 3060 cubic inches per minute at a pressure of 1000 pounds per square inch. Two of these pumps are used for feeding the two boring heads, which have 9-inch feeding cylinders. This equipment gives an available feeding force of 60,000 pounds on each carriage. The other two pumps are used for operating the mechanism for chucking the car wheels.

Variable-delivery Pumps Arranged with Closed Hydraulic Circuits. — For simplicity, Fig. 1 shows the pump pushing

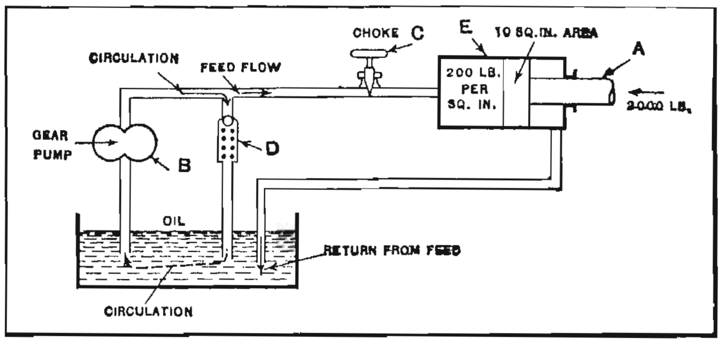


Fig. 2. Hydraulic Feed with By-passed Circuit

the piston outward, a valve which must be used to reverse the flow in the discharge and return pipes on the in stroke being omitted in the diagram. The complete arrangement gives a definite rate of feed proportional to the metered discharge of the pump less leakage from the closed pressure side of the circuit through the pump pistons and the feed piston. The rate at which the piston moves will never exceed the rate corresponding to the fixed discharge of the pump, and greater or less resistance to the movement of the piston will only raise or lower the pressure in the cylinder and connecting pipe to the pump without materially changing the speed at which the piston moves.

Perhaps the most outstanding advantage claimed for this system of hydraulic transmission of power is the sensitive

and easy control of the speed of the driven mechanism. If we pass the entire output of the pumping unit to the driven unit, the speed of the latter may be regulated by varying either the speed or displacement of the pumping unit. This constitutes a very efficient means of control, and one that is limited only by the mechanism involved.

The by-passed circuit illustrated diagrammatically in Fig. 2 shows the gear pump B pushing the piston A at a rate corresponding to only a fraction of the displacement of the gear pump. The excess displacement escapes through a relief valve D into the oil-pot, and is again taken up by the gear-pump suction pipe and continuously circulated.

Types of Pumps Used in Feeding Machine Tools. — At present there are three types of pumps in general use for feeding machine tools:

1. An accurately made gear pump capable of delivering a constant volume of oil at a constant pressure. These pumps are usually arranged to deliver oil at pressures up to 250 pounds per square inch. A relief valve is used in connection with this type of pump for maintaining an even pressure.

2. A multiple-piston pump with variable stroke. This type of pump is built to deliver a variable amount of oil at pressures up to 1000 pounds per square inch.

3. A pump which combines the first and second types and is arranged to deliver a large volume of oil at about 250 pounds pressure from a gear pump, and a smaller volume at a higher pressure from a variable-delivery piston pump. Both pumps are built into one housing and interlocked as to control.

There is nothing unusual about the gear pumps employed, which are simply required to deliver the necessary volume of oil at a constant pressure. The variable-stroke pumps or units, such as the Oilgear automatic variable-delivery pump are necessarily more complicated. This pump was designed for use in equipping the smaller sizes of milling, boring, drilling, and similar machines with hydraulic feeds. The following specifications for this pump may be of interest to designers:

Forward and reverse feeds and forward and reverse rapid traverse are provided. Either hand or automatic control may be employed. Pipe connections with the gear pump are provided for operating fixtures or other auxiliary equipment. Different positions of the control valve give full speed (rapid approach), feed forward, neutral, feed reverse and full speed reverse (rapid return). When used with a 3⅞-inch cylinder, this pump has a feeding range of from 1.66 to 23 inches per minute and a reverse feed of from 3.32 to 46 inches per minute. The rapid traverse speed in either direction is 93 inches per minute. The maximum working pressure is 1000 pounds per square inch and the power consumption at maximum capacity is 2 horsepower. The drive shaft speed is 860 revolutions per minute or lower. The pump is about 19½ inches high.

Rotary Drives for Long Strokes and Rotary Tables. — When very long table strokes are required, as in the case of some types of grinding machines, or when the table has a rotary movement, as in some milling machines, it is desirable to use a rotary motor in place of the feed cylinder. The working parts of a motor of this type are identical with those of the corresponding variable-stroke pump. The displacement of one such motor is 4.6 cubic inches per revolution; and the maximum torque, 690 inch-pounds at 1000 pounds per square inch. The maximum speed is 860 revolutions per minute, and the output at this speed, 9.4 horsepower.

Operating Multiple Feeds. — To have complete individual speed control of two or more hydraulic cylinders or motors, each cylinder must be driven by its own pump and the entire flow from the pump must go through that cylinder. However, drilling machines of the multiple-spindle type having several feed cylinders operated simultaneously by oil supplied by a single gear pump have proved practical. If sufficient oil is pumped at all times, so that under the worst condition of usage there is still a slight amount being by-passed through the relief valve, the rate of movement of the piston of any one cylinder can be changed without any material change in

the rate of travel of the pistons in the other cylinders.

If the volume of oil delivered is not sufficient to maintain the pressure adjusted by the by-pass valve in the case of two or more motors or cylinders operated in parallel, the motor or cylinder encountering the least resistance may take the entire flow until its stroke or work is finished or stopped by closing a valve. The remaining units will operate successively according to the order of their resistance values. The total time required for all the cylinders to perform this work under this condition will be the same as though they operated simultaneously, assuming, of course, that the volume of oil delivered by the driving pump is the same. By employing a variable-delivery pump for changing the volume of oil flow in a system of this kind, the operator can obtain any desired feed for each cylinder as it comes into operation.

Cylinders Operated in Series. — If the speed control required on two cylinders is simultaneous and proportional, the two cylinders may be placed in series in a closed circuit with a single pump. Such a circuit with the three pumps in series is shown diagrammatically in Fig. 3. This circuit is applicable to a drilling machine equipped with three drilling heads.

The movement of such heads may be coordinated mechanically by racks and pinions or by linkage, but the hydraulic method is more flexible, less liable to damage through breakage, and in some cases, cheaper. The three cylinders must be graduated to the speed requirements of the heads. If one head is to move twice as fast as another, its cylinder must have one-half the volume. Also, each cylinder must be so designed that the volume displaced in its piston-rod end is equal to the volume displaced in the head end of the next succeeding cylinder. This is evident from the diagram, as the oil supplied to the head end of each cylinder after the first one, comes from the rod end of the preceding cylinder.

In order to keep the movement of such a set of pistons properly coordinated, the pistons must run against their cylinder heads at the termination of each cycle, so that they always start the next cycle in the same relation. The relief valves

A, *B*, and *C*, permit oil to pass around any piston that has stalled against its cylinder head during the back stroke, thus bringing all the pistons successively back against their cylinder heads.

Use of Multiple Transmitter. — The series circuit of Fig. 3 divides the total working pressure into as many parts as there are cylinders. This reduces the maximum working pressure available in each cylinder, and tends toward large cylinder diameters. For this and other reasons, it is sometimes better to use a multiple transmitter, consisting of one double-acting

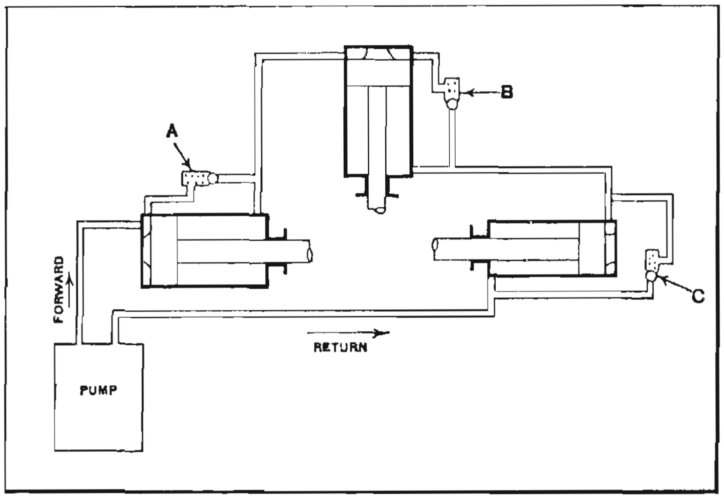


Fig. 3. Diagram of Cylinders Operated by Series Circuit

cylinder reciprocated by the pump and operating several cylinders whose piston-rods are attached to a single cross-head. Each of these secondary cylinders acts as a pump or impeller for its own individual driven or feeding cylinder. This system establishes several separate closed hydraulic circuits, each of which operates its feeding cylinder at definite speeds, the pressure in each separate circuit depending on the resistance against the piston-rods of the respective feeding cylinders and on the corresponding piston areas.

The strokes of all the impelling cylinders are the same, but

their diameters and the diameters of the feeding cylinders may be varied to give any desired feeding forces and strokes to the respective feeding cylinders, provided the totals are within the power capacity of the pump. All speed variations and distances traveled by the pistons of the respective feeding cylinders are proportional to the speeds and distances traveled by the piston of the main cylinder connected to the pump.

In this case also, it is necessary to provide means similar to those shown in Fig. 3 for bringing each of the feeding cylinders against this cylinder head at the end of every cycle

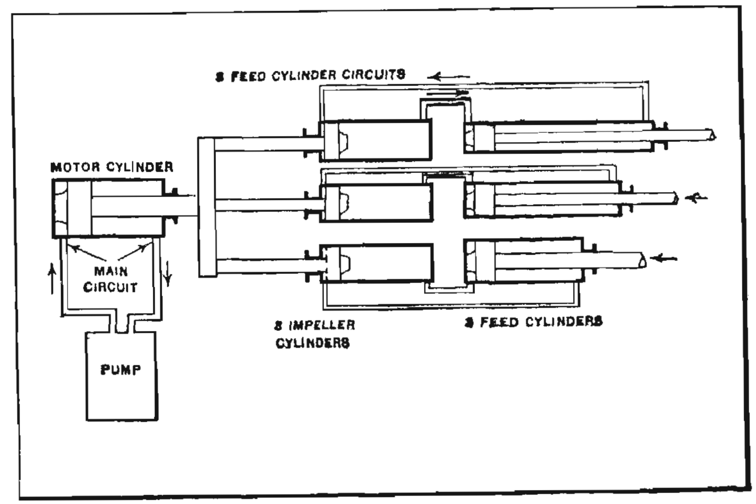


Fig. 4. Diagram of Multiple-impeller System

to keep the pistons in coordination. A system of this kind is indicated in Fig. 4, only the principal circuits being shown. In practice, the circuits of both Figs. 3 and 4 require low-pressure make-up lines from the pump to each circuit, and other details, which are omitted for the sake of clearness.

Slip and its Effect. — A tool-holder fed by an oil-pressure piston and a volumetric pump cannot be used to chase a thread, because its rate of feed is not absolutely constant. There is always a certain amount of leakage (across the bridges and through the plunger fits) in the pump, and this leakage is greater as the cut becomes heavier and the oil pressure rises. If the feed is low, in inches per minute, this leakage may be a considerable percentage of the pump delivery. This is the "slip," and in the early designs of hydraulically fed tools, was generally assumed to be a defect. Geared feeds do not slip, and the tool must cut the given thickness of chip, whether the material is hard or soft, the cut deep or shallow. If the tool manages to back off slightly due to "windup" in the rods and gearing, the lost travel must be made up, and the average thickness of chip throughout the cut must be equal to the geared feed rate.

If the tool is fed by oil from a volumetric pump, it will never move faster than the nominal feed rate, but higher cutting pressures will cause the feed to slow up. The travel lost by this increased slippage is never made up. Hence the tool is not damaged by being forced to maintain the given feed rate, as may be the case when overload causes winding up of the mechanical feed gear.

The rate of leakage in a hydraulic feeding system depends upon the pressure, and the pressure is directly caused by the resistance encountered by the tool. Consequently the flow of oil actually delivered by the pump into the feeding cylinder is less as the resistance increases. In actual practice, this reduction of feed with increasing pressure may be a very significant fraction of the theoretical feed rate, especially with heavy cuts at slow feeds.

For instance, a standard 3⅞-inch diameter feed cylinder has a piston area of 11.8 square inches and can deliver a net feeding force of 11,800 pounds to a cutting tool. When working at this maximum pressure of 1000 pounds per square inch, the slip of the entire apparatus would quite likely amount to 15 cubic inches of oil per minute. In ordinary cuts, such a feeding cylinder usually operates at pressures of 250 or 300 pounds per square inch, and the slip is, say, 5 cubic inches per minute.

Therefore, if the feed were adjusted to give 4 inches per minute under 250 pounds pressure, the additional slip of 10

cubic inches of oil as the pressure rises to nearly 1000 pounds would reduce the rate of feed by nearly ⅞ inch per minute, leaving a net feed of about 3⅛ inches per minute during the excessively heavy cut. This amounts to a 20 per cent reduction in the 4-inch per minute rate of feed. If the feed rate were set at 16 inches per minute, the reduction would be 5 per cent, as the amount of slip is practically constant for given pressures. These rates are based on a type of pump having relatively large leakage through a distributing valve. Other types would show about one-half as much slippage.

Speed-changing Hydraulic Transmission.— The hydraulic transmission illustrated by Figs. 5 and 6 is so designed that the speed of the driven pulley may be varied from zero up to the full speed of the driving pulley, so that this mechanism may be utilized as a clutch or for changing speeds. This transmission is intended for general application.

The driving pulley *A* on shaft *B* (Fig. 5) revolves gear *C* and two idler gears *D* and *E* (see Fig. 6). These idler gears are housed in case *F* to which the driven pulley *G* is attached. The gears referred to act as a pump. and circulate oil through ports *H*, *J*, *K*, and *L* (as indicated by the arrows), provided the ports in the cylindrical or plug valves *M* and *N* are open.

If valves *M* and *N* are fully open, the gears will rotate freely, because the oil can circulate through the passageways without resistance; consequently, the driven member and its pulley will remain stationary. If, however, the valves *M* and *N* are closed gradually, there will be a corresponding increase in resistance to the rotation of the gearing, and as a result, the driven member will rotate at a rate of speed depending upon the amount of resistance. When valves *M* and *N* are completely closed, all rotation of the gears is prevented, and the driving and driven members rotate at the same speed. The transmission then acts like a clutch in engagement, whereas when valves *M* and *N* are fully open and the driven member is stationary, the action is similar to a clutch that has been disengaged. Thus it will be seen that the gears revolve as a unit only when the valves are fully closed, and they rotate

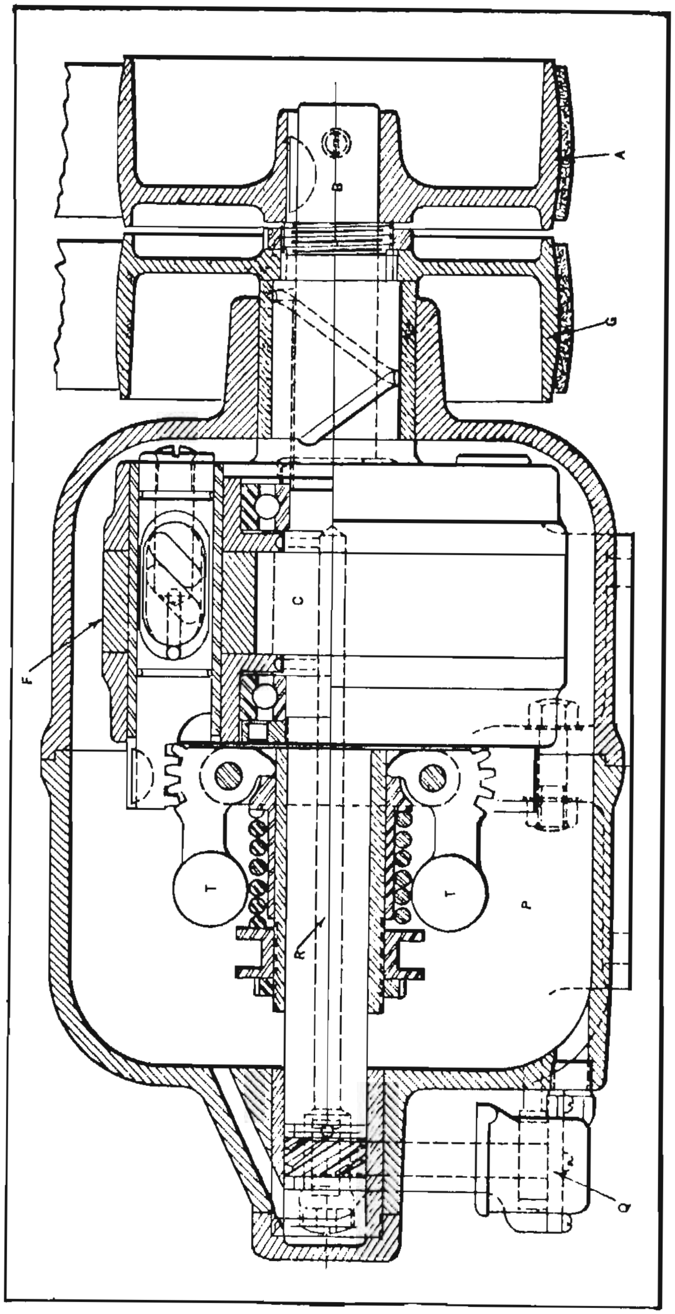


Fig. 5. Hydraulically Controlled Transmission Which May be Used Either as a Clutch or for Varying Speeds

about their axes when the valves are partially or entirely open for the purpose either of varying the speed or discontinuing the drive entirely.

The main supply of oil is in the main casing at P (Fig. 5)

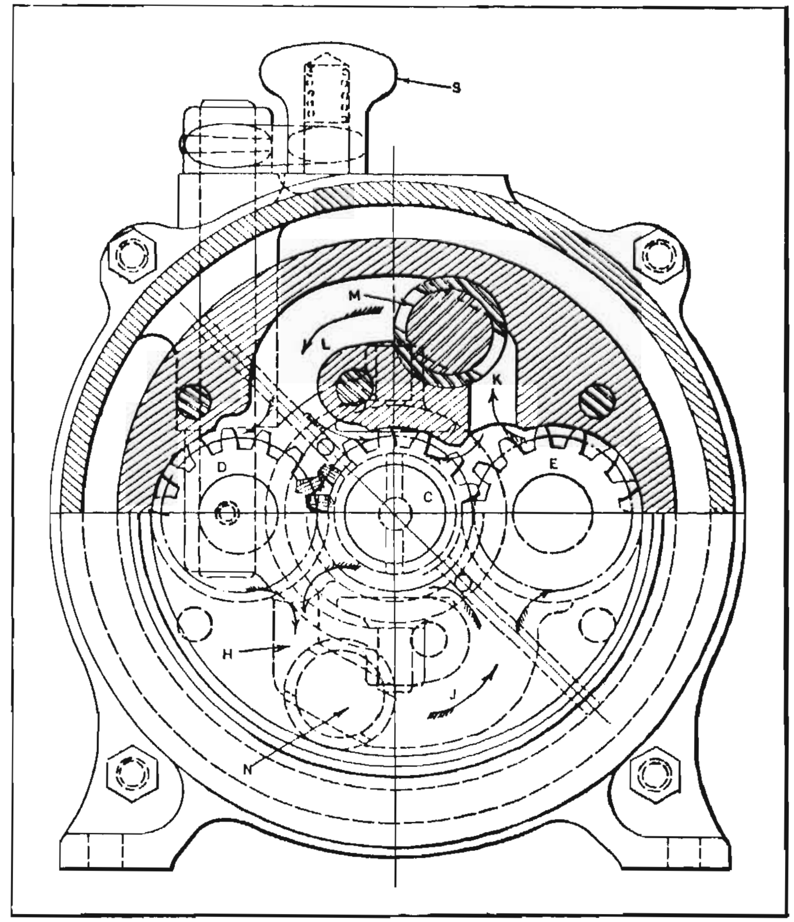


Fig. 6. Cross-section, Showing Gears and Control Valves of Hydraulic Transmission

and a small pump at Q, driven through spiral gearing from the main shaft, forces the oil through a central opening R in this shaft. Lever S (Fig. 6) serves to control the positions of valves M and N and the speed variations. Any variation in the speed for which the mechanism is set, caused by changes in load, is regulated by the centrifugal governor *T* (Fig. 5).

In order to relieve the oil pressure at the points where the teeth of gears *C*, *D*, and *E* intermesh, small radial holes are drilled through these teeth and connect with diagonal holes leading to the spaces between the teeth, thus relieving the oil pressure and lessening friction. This transmission is also designed to provide reversal by special arrangement of gearing connection with the driven member. The hydraulic feature of the transmission, however, is the same as described.

CHAPTER XVI

AUTOMATIC FEEDING MECHANISMS

MACHINES which operate on large numbers of duplicate parts which are separate or in the form of individual pieces are often equipped with a mechanism for automatically transferring the parts from a magazine or other retaining device, to the tools that perform the necessary operations. The magazine used in conjunction with mechanisms of this kind is arranged for holding enough parts to supply the machine for a certain period, and it is equipped with a mechanical device for removing the parts separately from the magazine and placing them in the correct position wherever the operations are to be performed. The magazine may be in the form of a hopper, or the supply of parts to be operated upon by the machine may be held in some other way. The transfer of the parts from the hopper or main source of supply to the operating tools may be through a chute or passageway leading directly to the tools, or it may be necessary to convey the parts to the tools by an auxiliary transferring mechanism which acts in unison with the magazine feeding attachment. These automatic feeding mechanisms are usually designed especially for handling a certain product, although some types are capable of application to a limited range of work. The feeding mechanisms described in the following include designs which differ considerably, and illustrate, in a general way, the possibilities of automatic devices of this kind.

Attachments having Inclined Chutes.— One of the important applications of magazine feeding attachments is in connection with the automatic screw machine. Most of the parts made on these machines are produced directly from bars of stock, but secondary operations on separate pieces are sometimes necessary, and then an automatic or semi-automatic attachment may be employed to transfer the parts successively to the machine chuck where the tools can operate upon them. Many of these attachments have magazines which are in the form of an inclined chute that holds the parts in the

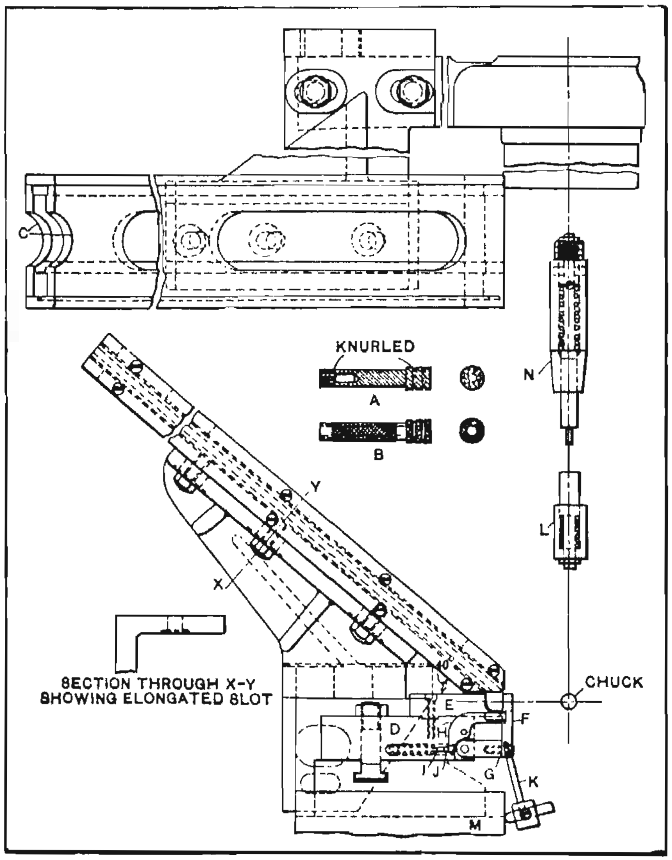


Fig. 1. Automatic Screw Machine Magazine Attachment

correct position and from which they are removed, one at a time, by a transferring device. An example of this type of magazine attachment is shown in Fig. 1. This attachment was designed for feeding the handles of safety razors. The preliminary screw machine operations involve turning, form-

ing, knurling, drilling, tapping, and cutting off the handle, thus producing a piece of the form shown at *A* in the illustration. These partly finished handles are then placed in the chute or slide of the feeding attachment, from which they are transferred to the chuck, so that a hole can be drilled clear through the handle as indicated at *B*, and one end of the hole be slightly enlarged. The upper and lower plates *C* of the chute have grooves milled in them to correspond to the enlarged parts of the handle. As each successive handle reaches the lower end of the chute and drops into the small pocket shown, a spring plunger *L* attached to the turret advances and pushes the work out into the chuck of the machine.

As the ends of the handles have shoulders, the pocket at the bottom is automatically enlarged to permit the passage of this shoulder. The work-carrier consists principally of two blocks *D* and *E* and a finger *F*. Block *D* is held in the cross-slide and block *E* is attached to the top of block *D*. The forward end of block *E* is cut out to fit the work, which is held in place by finger *F*. This finger is fastened to lever *G*, pivoted on block *D*, and normally held in position by a pawl *H* engaged by plunger *I* and pin *J*. When a piece of work drops into the pocket in block *E* and the front cross-slide has advanced far enough to bring the work in line with the hole in the chuck, the enlarged part of the plunger *L* trips the finger *F* after the work has been partly inserted in the chuck. This action is caused by the contact of plunger *L* with a beveled edge on pawl *H* which disengages the V-shaped end of the pawl from a groove in lever *G* and, at the same time, pushes back spring plunger *I*, thus allowing finger *F* to drop away from block *E*. The pawl *H* serves as a locater for the work and, when disconnected from lever *G*, it swings down and the work is pushed into the chuck by plunger *L* which is held in the advancing turret. After a piece has been inserted in the chuck, the cross-slide, as it moves outward, brings trip *K* against casting *M* which, through the combined action of lever *G*, pawl *H*, and spring plunger *I*, closes the work-carrier. The piece in the chuck is forced in against a spring plunger

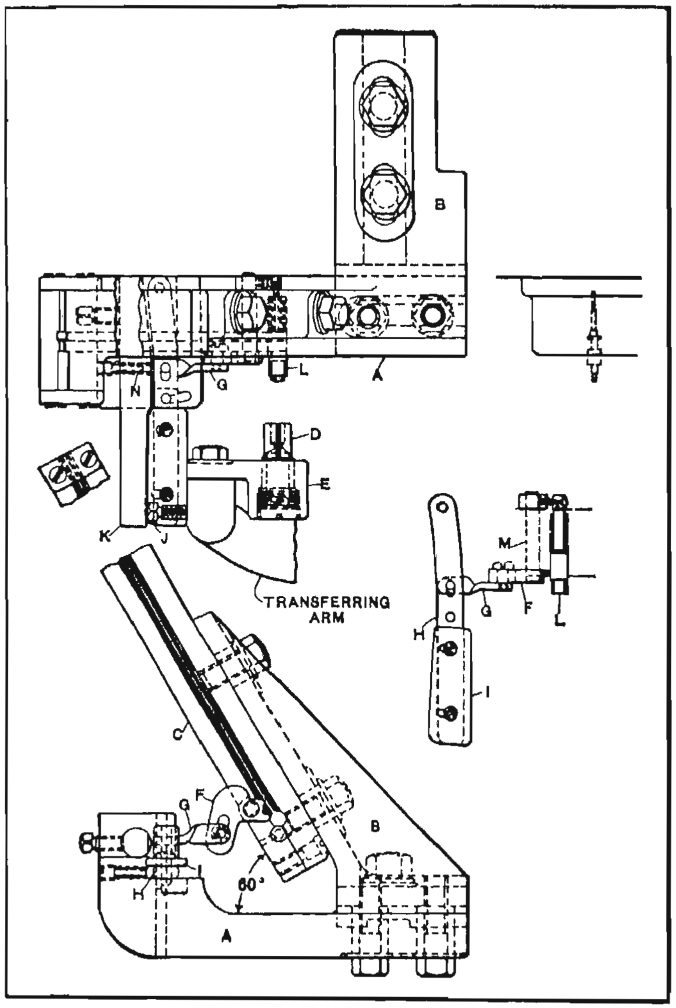


Fig. 2. Magazine Attachment for Pinion Staffs

held by feed finger *N*. This spring plunger ejects the work when the machining operation has been finished and the chuck is opened.

Feeding Attachment for Pinion Staffs. — The magazine feeding attachment shown in Fig. 2 was designed for handling pinion staffs of the form illustrated by the dotted lines in the

upper right-hand corner of the illustration. The chute *C* is supported by a bracket *B* which is attached to a boss provided on the automatic screw machine for holding special attachments. The bracket *A* is attached to *B* and carries the mechanism for feeding the pinion staffs successively to the place where they can be removed by the transferring arm. The two main parts of the chute are grooved to fit the pinion staffs, so that the latter are held in the correct position. The operation of this attachment is as follows: The chute is filled with pinion staffs and the lower one is held back temporarily by trip *F*. This trip is connected to link *G*, which carries a pin that engages a slot cut in lever *H* (see detailed view). Lever *H* has fastened to its upper side a trip-lever plate *I* the inclination of which may be varied. When the transferring arm swings upward, it is stopped in the correct position by set-screw *J*, which engages stop *K*, the arm itself bearing against plate *I* and forcing it back, together with lever *H*. This action, through connecting link *G*, operates trip *F* and allows one piece to drop into the pocket formed at the end of this trip. The transferring arm carrying a split bushing *D* then advances and pushing back the nest *L* passes over the end of a pinion staff and grips it. The transferring arm then recedes and swings down to the chuck in which the pinion staff is placed. When the transferring arm descends, the spring *N* returns trip-lever plate *I* and lever *H* to their former position. Trip-lever *F* also swings back in order to catch another piece, the pinion staff in the trip being deposited in the nest *L* ready for transferring to the split bushing *D* the next time the transferring arm ascends.

Magazine Attachment for Narrow Bushings. — The narrow bushings shown at *A*, Fig. 3, are blanked out and drawn in a die to the shape shown; they are then turned, faced, and threaded (as indicated at *B*) in an automatic screw machine. Two separate operations are required, but the magazine attachment shown in this illustration is used for both. The bushings are placed in the inclined slide or chute, and the lower one is retained temporarily by a finger *i*, which is held upward by spring k, the exact position of the finger depending upon the adjustments of set-screw j which engages a projecting end. The transferring arm, which removes the work from the lower end of the chute and conveys it to the chuck, has a swinging or circular movement, as indicated by the dotted line. The work is gripped as the holder (shown in detail at C) advances, and then, as the transfer arm starts to swing downward toward the chuck, the finger i is depressed,

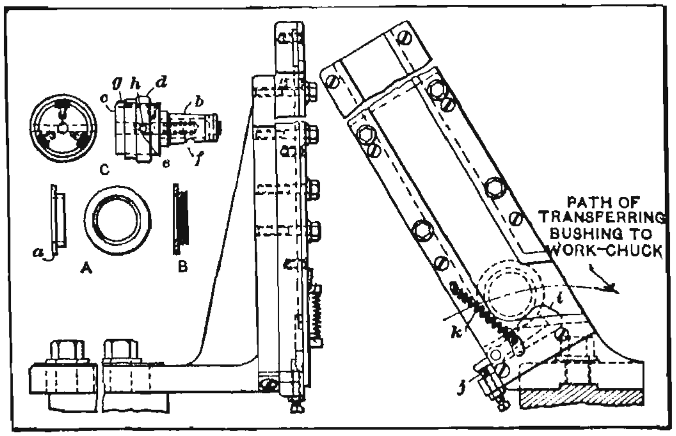


Fig. 3. Magazine Attachment for Handling Parts Shown at A and B

thus allowing the bushing to slide out of the chute. The work-holder has a taper shank b which fits into the main body c. On this body is held a ring d through which a pin is driven. The pin h in this ring d fits into an elongated hole in body c and enters spring plunger e. A slot in body c receives a flat spring g, which is provided to grip the work securely. This spring also compensates for slight variations of diameter.

The degree of inclination for chutes of magazine attachments varies from 20 to 60 degrees and depends upon the size and shape of the work. The chute should incline at a greater angle for small work than for large work. The chutes of attachments used for handling flat pieces, such, for example, as might be cut out in a blanking die, are usually held in a vertical chute instead of one that is inclined.

Hopper Feeding Mechanism for Screw Blanks. — The automatic feeding mechanism to be described is used on a thread rolling machine of the type having straight dies between which the blanks are rolled to form the threads. The faces of the dies are in a vertical position and one die is given a reciprocating motion in a direction at right angles to the axis of the screw blank. The automatic feeding mechanism

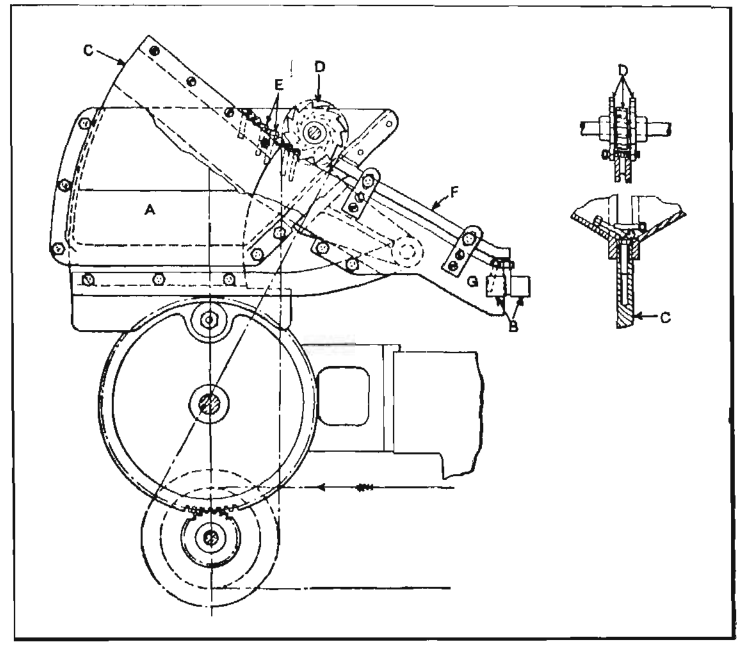


Fig. 4. Hopper-feeding Mechanism for Screw Blanks

shown in Fig. 4 is arranged to transfer the screw blanks from the hopper A to the dies at B in such a way that each successive blank is in a vertical position when caught between the dies. The hopper A, which is at the top of the machine, is equipped with a plate or center-board C which passes through a slot in the bottom of the hopper and is given a reciprocating motion by a gear-driven cam. This center-board has a vertical slot extending along the upper edge (see detail sectional view) which is a little wider than the diameter of the screw blank bodies. As the center-board moves up through the mass of screw blanks, one or more of these blanks are liable to drop into the slot and hang suspended by their heads. If a blank does not happen to be caught for any one stroke of the center-board, the mass of blanks is disturbed and it is likely that one or more blanks will fall into the slot on the next successive stroke of the center-board.

As some blanks are picked up while in a crosswise or other incorrect position, an auxiliary device is employed to dislodge such blanks. This device consists of three revolving wheels at *D* which have teeth like ratchet wheels. The arrangement of these wheels is shown by the detailed view. The center wheel, which is the smallest, revolves above the heads of the blanks which are moving down the slot of the center-board in the proper position, as indicated at *E*. The two outer wheels, which are larger than the central one, revolve close to the outer edges of the center-board. If a blank is not in the correct position, it will be caught by these wheels and be thrown back into the hopper, but all blanks that hang in the slot pass between the outer wheels and beneath the central one without being disturbed. After the blanks leave the center-board, they pass down the inclined chute *G*, which is provided with a guide *F* that holds them in position. As each successive blank reaches the lower end of the chute, it swings around to a vertical position and is caught between the dies which roll screw threads on the ends.

Feeding Shells with Closed Ends Foremost.— The possibilities of mechanical motion and control are almost boundless, if there is no limit to the number of parts that may be incorporated in a mechanism, but as complication means higher manufacturing cost, and usually greater liability of derangement, the skillful designer tries to accomplish the desired results by the simplest means possible; it is this simplifying process that often requires a high degree of mechanical ingenuity. The feed-chute shown in Fig. 5 illustrates how a very simple device may sometimes be employed to accomplish what might appear at first to be difficult. This is an

attachment used in conjunction with an automatic feeding mechanism for drawing shells in a punch-press. These shells are fed from a hopper, and it is essential to have them enter the die with the closed ends down. If a shell descends from the hopper with the open end foremost, it is automatically turned around by the simple device shown. The view to the left illustrates the movements of a shell which comes down in the proper position or with the closed end foremost. In this case, the bottom of the shell simply strikes pin *B* and, after rebounding, drops down through tube *C*. If the open

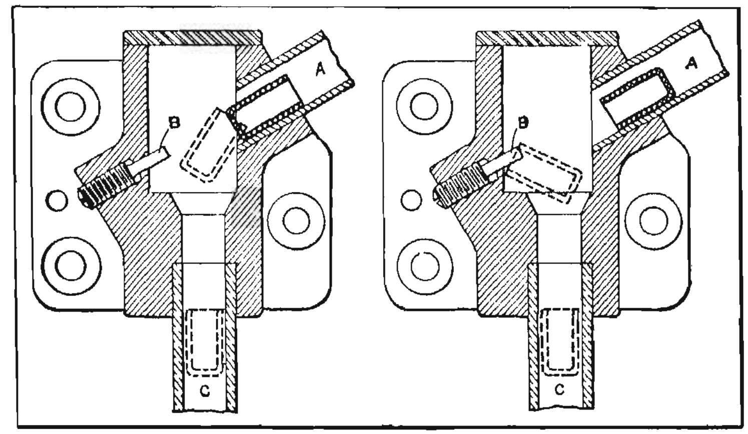


Fig. 5. Simple Attachment of an Automatic Feeding Mechanism for Turning Shells Which Enter Open End Foremost

end of a shell is foremost, as illustrated at the right, it catches on pin *B* and is turned around as the illustration indicates. If a shell enters the die with the closed end upward, the drawing punch will probably be broken.

Feeding Bullets with Pointed Ends Foremost.— An attachment for feeding lead bullets or slugs to press tools with the pointed ends foremost, regardless of the position in which the bullets are received from the magazine or hopper, is illustrated in Fig. 6. This attachment is applied to a press having a 4½-inch stroke. The bullets enter the tube *A* which connects with a hopper located above the press. An "agitator tube" moves up and down through the mass of bullets in the hopper and the bullets which enter the agitator tube drop into tube *A*. As each bullet reaches the lower end of this tube, it is transferred by slide *C* (operated by cam *D* attached to the cross-

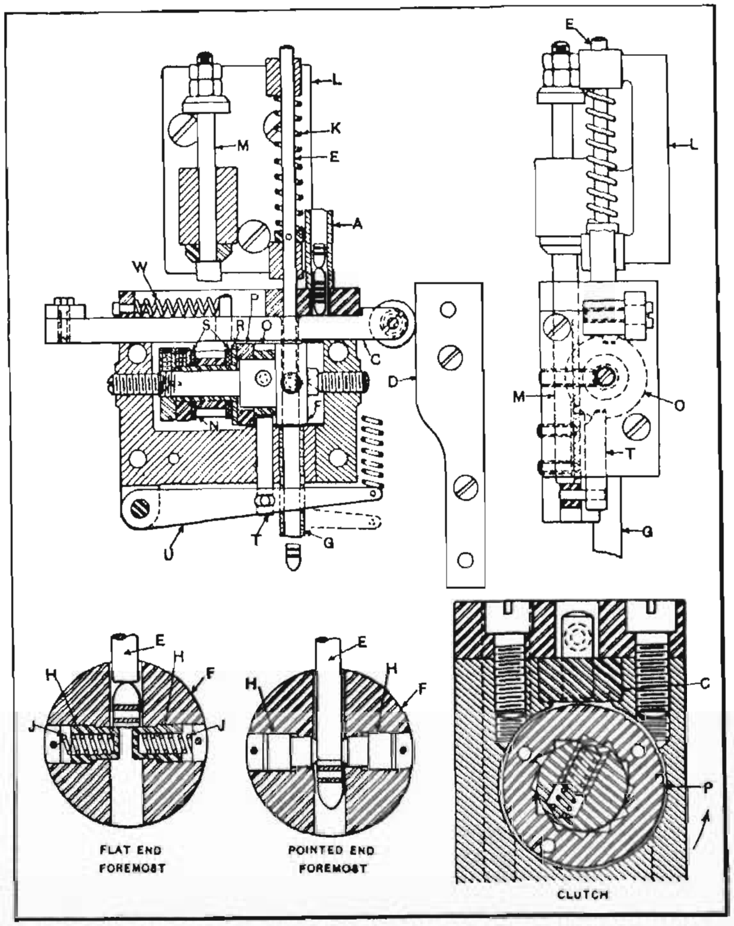


Fig. 6. Attachment for Hopper Feeding Mechanism Which Delivers All Bullets to a Dial Feed Plate with Pointed Ends Foremost

head) to a position under the rod *E*. The rod-holder *L* is also carried by the cross-head. Whenever a bullet enters tube *A* with the rounded or pointed end downward, it is simply pushed through a hole in dial *F* and into feed-pipe *G* leading

to the dial feed-plate of the press. This feed-plate, in turn, conveys the bullets to the press tools where such operations as swaging or sizing are performed.

The arrangement of dial *F* is shown by the detailed sectional views at the lower part of the illustration. Whenever a bullet enters the dial with the pointed end foremost, the plungers *H* are pushed back against the tension of springs *J* and the bullet drops into the tube beneath. If the blunt or flat end is foremost, the plungers are not forced back, and as rod *E* is prevented from descending further, it simply moves upward against the tension of spring *K* as the cross-head continues its downward motion. A mechanism is provided for turning dial *F* one-half revolution so that every bullet that is not pushed through the dial will be turned around with the pointed end foremost before it drops into the feed-tube *G*. This rotary motion of the dial is derived from a rack *M* attached to bracket *L*, and a pinion *N* with which the rack meshes. The location of the dial is governed by an index plate *O* and a plunger *T* which enters one of the notches in the index plate; the latter is attached to dial *F*. A clutch *P* (see also detailed sectional view) is fastened to sleeve *R*. Fiber friction washers *S* are used to prevent breakage in case anything unusual should happen.

When the cross-head descends, the rack *M* revolves the clutch in the direction shown by the arrow. When within one-quarter inch of the lower end of the stroke (this position is shown in the illustration), the rack *M* strikes lever *U* and disengages the index plunger *T*. The rack descends far enough to give it time on the return stroke to move dial *F* sufficiently to prevent the returning index plunger from re-entering the hole it just occupied. On the return stroke, the lost motion of the rack in its bracket provides time for the withdrawal of rod *E* before dial *F* is revolved. This lost motion can be adjusted so that the highest point of the upward stroke is reached just as dial *F* has turned 180 degrees, thus bringing the other index slot in line with plunger *T*. If the rack should move too high, the friction washers *S* will allow for this excess movement by slipping. This half revolution of dial *F* turns a bullet that is not pushed through it end for end, so that it drops down in the pipe *G* with the pointed end foremost. The slide *C* is returned for receiving another bullet from tube *A* by the action of spring *W* which holds the slide roller firmly against the cam-plate *D*.

Feeding Shells Successively and in Any Position.— A feeding mechanism designed to feed shells or cartridge cases one at a time and in any position is shown in Fig. 7. Owing to the weight of the heads of cartridge cases, they may readily be arranged upon a table heads downward, and the particular mechanism to be described is arranged for changing the shells from a vertical to a horizontal position before dropping them into a trough by means of which they are conveyed to the operating tools. The table *A* upon which the shells are placed is slightly inclined so that the shells readily slide towards a horizontal disk *B* which is rotated constantly by a belt and pulley. As the disk revolves, the shells are carried towards the funnel-shaped mouth of a guideway *C* where there is a wheel *D* having teeth of irregular form. This wheel is revolved in the same direction as disk *B* so that it continually pushes back some of the shells and prevents jamming. The shells which move too near the center of disk *B* to enter the mouth of the guide-way are carried around until they meet the edge of an inclined fence *E*, which is just above the disk near the center, but is arched near the periphery so that shells can pass under it. This fence causes the shells to move out towards the circumference of disk *B*, so that they may enter the guide-way as they again come around.

Just beyond the wheel *D* there is a feed-wheel *F* which has teeth of regular form that fit between the cartridge cases. This wheel is rotated in the direction shown by the arrow, so as to feed the shells forward at a definite rate along the guide-way *C*. This guide-way, excepting at the mouth, is only slightly wider than the shell diameter, so that all the shells in it form a continuous and orderly row. The guide-way may be curved gradually in any direction, so that the shells which

enter it with their axes vertical may be turned to any desired position as they pass along. As previously mentioned, the guide-way, in this case, changes from a vertical to a horizontal position. At the end of the guide-way there is a pair of stops that act alternately to allow one shell to issue at a time from the guide-way. The first stop consists of a pair of fingers *G* which rise up through the floor of the guide, and the

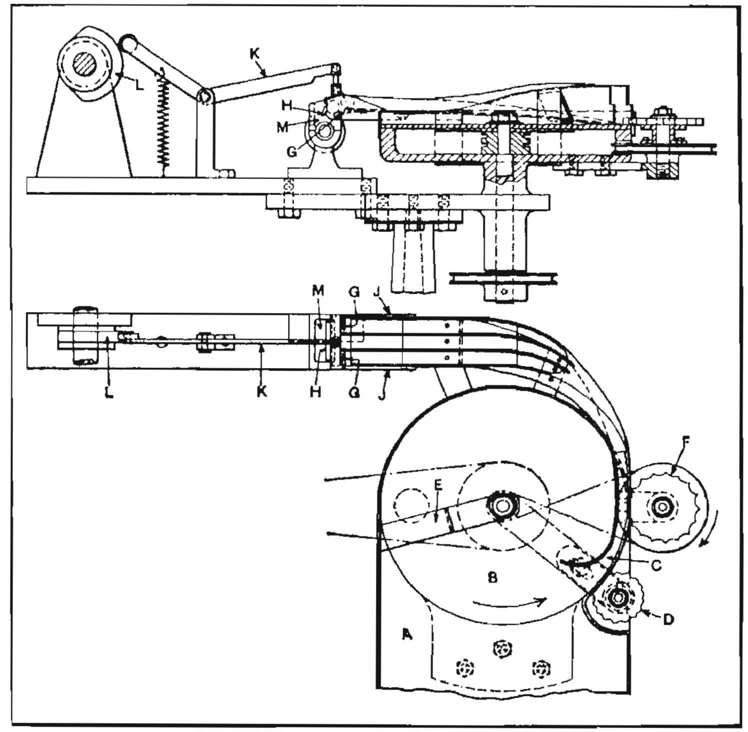


Fig. 7. Mechanism for Automatically Feeding Shells One at a Time

second stop is in the form of a gate *H* which moves down in front of the foremost shell of the row. These two stops are carried on a pivoted frame *J* so arranged that, as the gate rises to allow the foremost shell to pass from the mouth of the tube, the fingers *G* rise in front of the second shell to hold back the whole row. The frame *J* is connected with a lever *K* which is intermittently rocked by the cam *L*. The successive shells drop into the trough M as they are discharged from the guide-way.

Feeding Shells Successively and Gaging the Diameters. — The mechanism described in the following is part of a cartridge-making machine, and its function is to feed cartridge cases or shells from a tube, one at a time, and provide means of detecting shells having heads that are over the standard diameter. The shells are placed heads downward onto a fixed

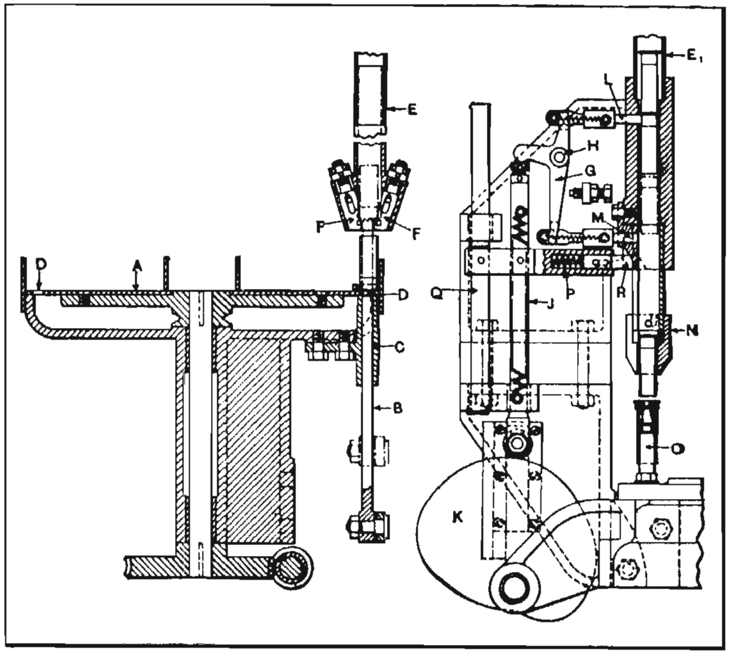


Fig. 8. Mechanism for Feeding Shells Successively and Gaging the Diameters

table from which they are pushed by hand onto a revolving disk A, Fig. 8. This feed disk operates on the same general principle as the one illustrated in Fig. 7. As each successive shell passes from the guide-way of the revolving disk, it is placed directly over a push-rod B. This push-rod is pivoted to the end of a lever which is oscillated by a cam, thus causing the push-rod to move vertically through a guide C and through one of the slots D formed in the periphery of the feed disk A.

Each time the push-rod B moves upward, it pushes a shell into the end of tube E. This tube has two gravity fingers F and, as the shell rises, its rim lifts these fingers and separates them far enough to allow the rim to pass; the fingers then drop back behind the rim and prevent the shell from falling when the push-rod recedes. When this push-rod makes the next successive stroke, the shell lifted by it pushes the first shell up into tube E which is bent over to form an arch and terminates at E_1.

When the vertical section of tube E is filled and the shells passed over the top of the arch, they fall open end first down into the vertical section E_1. Just below the end of tube E_1, there is a device for releasing the shells one at a time. This consists of a three-armed lever G, which is pivoted at H and is given an oscillating or rocking movement by vertical rod J having a roller in contact with cam K, against which the rod is held by a spring. As lever G oscillates, it withdraws, alternately, two fingers L and M which project into the passageway for the shells. These fingers are withdrawn against the tension of suitable springs and the upper one catches the cartridge shells by the rim, whereas the other one extends beneath the open end. When the upper finger is withdrawn, a shell drops against the lower finger and, when the latter is withdrawn, this shell is released and, at the same time, the upper finger moves in and prevents the next successive shell from dropping out until it is released by the backward motion of finger L. As each successive shell drops, it passes through a gage N and then falls over one of the vertical pins O, which are equally spaced around the periphery of the machine table. This table is revolved intermittently in order to locate the shells beneath a series of tools carried by a tool-holder having a vertical reciprocating motion.

Attached to the rod J, there is a bar P the movements of which are steadied by a bar Q mounted in suitable guides. The bar P carries a spring plunger R having a beveled end which engages a beveled surface as shown; consequently, as rod J and bar P are lifted by cam K, plunger R is pushed back far enough to clear the rim of the descending cartridge. When rod *J* descends, however, plunger *R* moves inward and bears downward on the head of the cartridge beneath it, thus pushing it through the gage *N* and onto one of the series of pins *O*. If the rim of a cartridge should be so large that it would not readily pass through the gage, the resistance overcomes the tension of the spring that holds *J* into contact with the cam, and the cartridge remains in the gage until the next stroke of the machine. As the table moves around, the attendant will notice that there is a pin without a shell upon it and, therefore, he will remove the next successive shell, because, ordinarily, the shells are not so large as to resist being forced through the gage by a second stroke of the push-down bar *P*. If an exceptionally large head will not pass through the gage, the machine must be stopped and the shell removed by hand.

Feeding Mechanism for Taper Rolls.— The device here described was designed for taking taper rolls, of the kind used in roller bearings, from a hopper, selecting these rolls, and feeding them small end first into a centerless grinding machine. The hopper used is of the type generally applied to thread-rolling machines. The center board is arranged with a V-groove in place of the usual slot, and the rolls are picked up and allowed to slide down into the selecting mechanism lengthwise.

The mechanism is shown depositing a roll *R* (see Fig. 9) into the fixture ready for the feed-bar *K* to come back and pick it up. The feed-bar is actuated by a cam on the opposite end of the machine (not shown in the illustration). The body *A* of the fixture is fastened to the hopper by a bracket. Sliding on body *A* are two plates *B* and *C*, retained by gibs. These plates are moved inward by pawls *D* and outward by compression springs *E*. Pawls *D* are oscillated about pivot pins *F* by the action of the cam surfaces *G* on slides *H* which are dovetailed into the body.

Recessed into slides *H* is a dog *J*. Another dog *L* is attached to feed-bar *K*, and is held in adjustment by clamp *W*. Pivoted in the body is a bellcrank *M*, the forked end of which

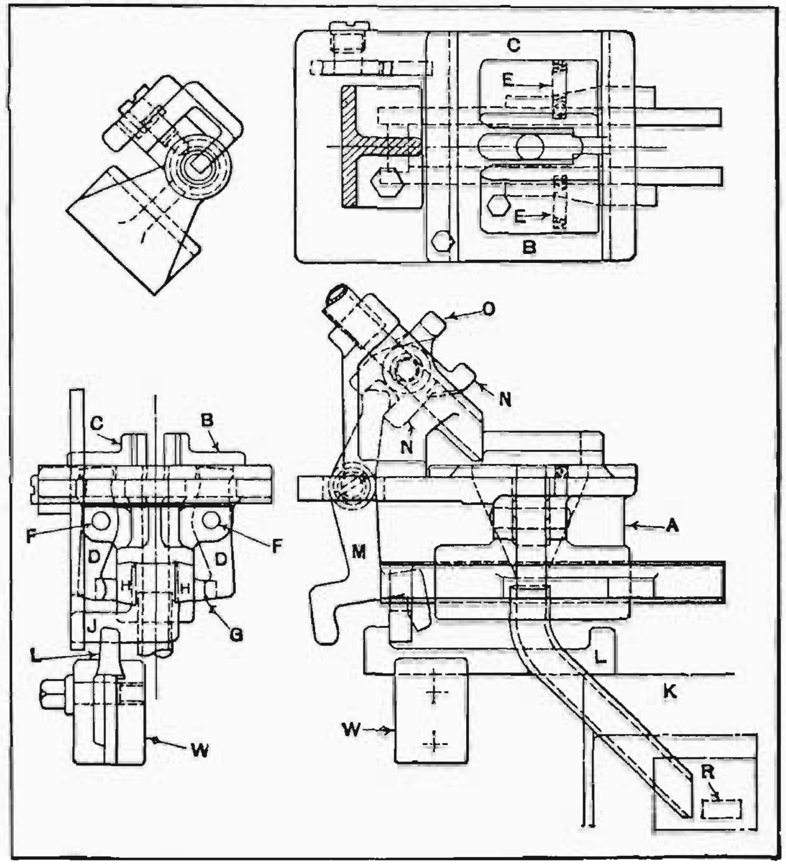


Fig. 9. Device for Feeding Taper Rolls Into Centerless Grinding Machine

straddles dog *J*, while the ball end meshes with a fork on the escapement pawl *N*. The function of the escapement pawls *N* and *O* is to cut off the feed of the rolls, as they come down from the hopper, and allow them to slide down one at a time on plates *B* and *C*.

In operation, the fixture works as follows: Feed-bar *K* is shown in its maximum "in position," and a roll *R* has been deposited ready to be picked up by the feed-bar on its return. Plates *B* and *C* are in their open positions, and escapement pawl *N* is shown holding back the rolls in the feed-tube. As the feed-bar returns, one of the projections on dog *L* engages dog *J* and carries slides *H* outward, forcing plates *B* and *C* inward through the action of pawls *D* which ride on the cam surfaces on slides *H*.

As dog *J* continues its outward movement it engages a prong on bellcrank *M*, causing it to pivot in the body and oscillate the escapement pawl enough to allow one roll to slide out on plates *B* and *C*. The remaining rolls in the line are retained in the feed-chute by pawl *O*, which bears on the top of the following roll. As feed-bar *K* completes its stroke, the roll at *R* is fed down by a finger (not shown), and the feed carries it between the grinding wheels on its return stroke. While the feed-bar is on its return stroke, the outer projection on dog *L* engages dog *J*, carrying the slides *H* inward.

Pawls *D* ride down the cam surfaces on the slides, and the compression springs *E* force plates *B* and *C* outward. During this outward movement of the plates the small end of the roll tilts downward, and as the plates continue to move apart, the small end drops through, leaving the roll suspended by its large end between the plates. The plates continue outward, allowing the roll to drop small end first through the feed-chute into the position *R*. Meantime bellcrank *M* is engaged by the dog *J* and oscillates pawl *N* downward. Pawl *O* is carried up and the rolls slide forward against pawl *N*, thus completing one cycle.

Revolving Magazine on Feeding Attachment.— The automatic feeding attachment shown in Fig. 10 has a revolving carrier of magazine *B* for holding the blanks to be operated on. This attachment is used for feeding the blanks from which the barrels for watch springs are made. The shape of these barrels, which are about ¾ inch in diameter, is indicated at *M*. The magazine wheel *B* is recessed, as shown by the side view, to form a pocket for the blanks, and it is provided with slots around the edge in which the blanks fit, as indicated at *N*. The blanks are inserted in the attachment or magazine wheel through slot *C* which connects with pocket *D*. The wheel *B* is rotated by a belt which transmits motion

from a pulley on the front camshaft to a pulley located on shaft *S*. As these two pulleys are of the same diameter, the magazine wheel rotates at the same speed as the front camshaft. The blanks, as they are carried around by the wheel, drop into slide *H* and from there into a pocket in a bushing held by a carrier. The block *I* of this carrier (see enlarged detail view) is counterbored to receive a bushing *O* which contains plunger *P*, and the bushing is cut out to receive the spring fingers *E*. These fingers are attached to plugs *F* which are held in drilled holes in block *I*. The bushing *O* is free to

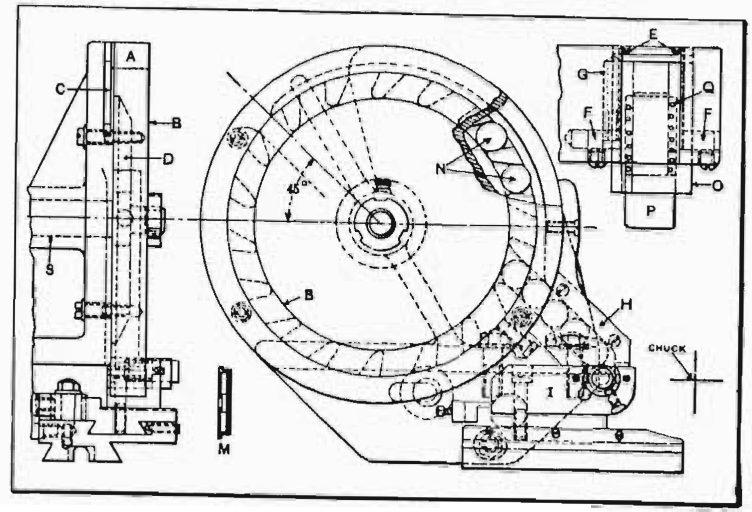


Fig. 10. Magazine Attachment of Revolving Type

slide in block *I* and is held back by spring *G*, which bears against a pin driven into the bushing. As a blank rolls down the slide *H*, it is deposited in bushing *O*. The cross-slide upon which the attachment is mounted then advances to locate the blank in line with the hole in the chuck. When in this position, the turret advances and a stop on it pushes plunger *P* forward, thus forcing the blank from the fingers *E* and depositing it in the chuck. The spring *Q* which returns plunger *P* is made much heavier than the spring *G* used for holding back the bushing *O*. The object of this arrange-

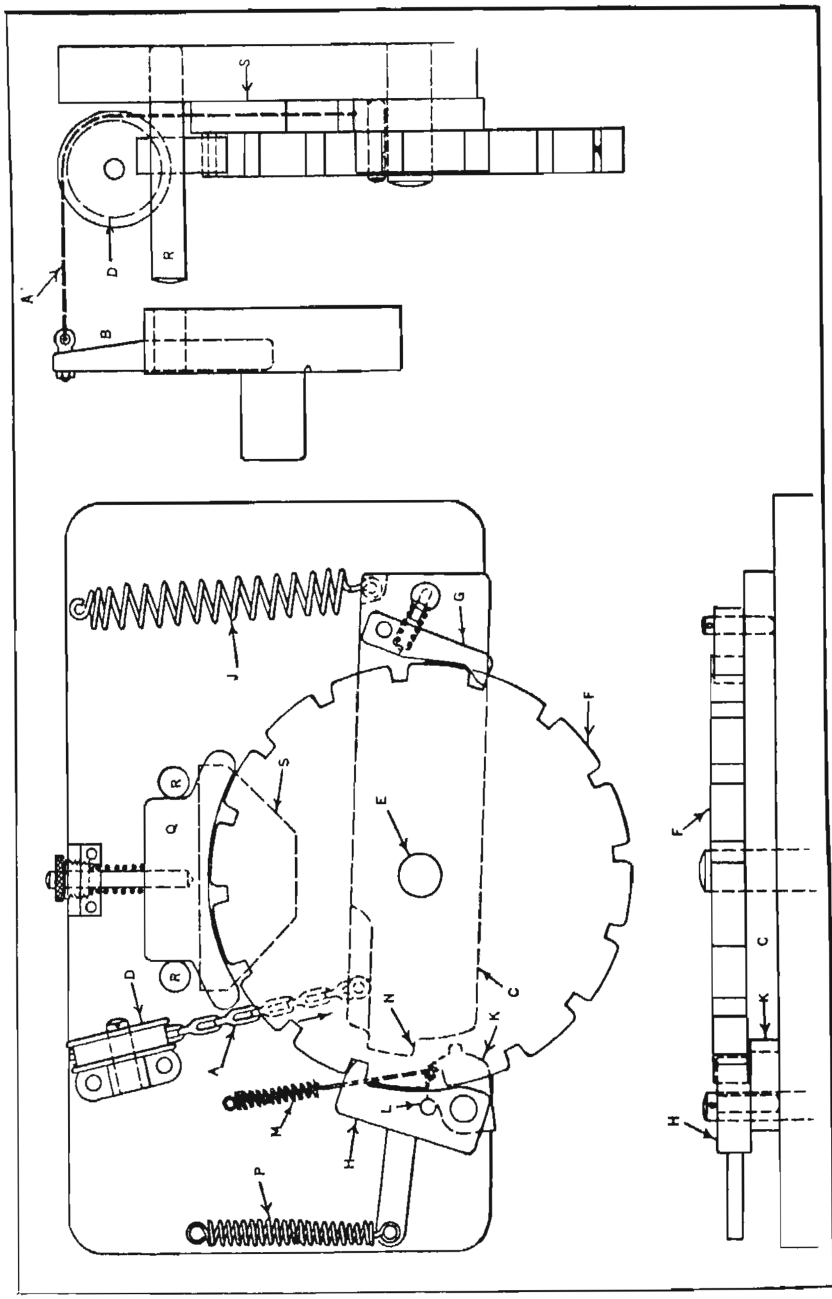


Fig. 11. Automatic Dial Feed for Press Work, Which is Self-contained and Readily Placed on or Removed from the Press

ment is to insure that the bushing will be pushed out close to the face of the chuck before the plunger forces the blank out of the spring fingers.

Self-contained Automatic Dial Feed. — In this design, which is for power presses, the necessary indexing and locking movements are obtained by very simple means through a self-contained mechanism which can be mounted on a press or removed from it quickly, without drilling holes in the press or attaching connecting-rods, levers, or other operating parts, to the crankshaft. A simple chain connection with the punch-holder provides the motions required for unlocking the dial and indexing it to the next station or working position.

The upper end of this chain *A* (see Fig. 11, which is partly diagrammatic) is attached to extension *B* on the punch-holder, and the lower end connects with indexing lever *C*. As the end view shows, the chain passes around a guide pulley *D*. The swinging movements of lever *C* about its pivot *E* are utilized in conjunction with spring controls, as described later, to unlock, index, and again lock dial *F*.

Pawl *G* is used for indexing dial *F*, and pawl *H* for locking the dial so that each successive die is accurately located relative to its punch and the dial is held securely during the working stroke. How the indexing and locking movements are derived will now be explained by describing the action of the different parts during, first, a downward and then an upward stroke of the ram.

The dial is shown in its normal or locked position. As the ram moves downward, the horizontal part of the chain moves in the direction indicated by the arrow (see also upper view, Fig. 12) and spring *J* turns lever *C* around its pivot *E*. The locking-pawl release-lever *K* is normally held against stop-pin *L* in locking pawl *H*, by a light spring *M*, as shown in Fig. 11, but when the projection *N* on lever *C* engages lever *K*, as shown by the upper view Fig. 12, lever *K* turns about its pin, allowing *C* to pass. Pawl *H*, however, is not disturbed, the dial remaining locked.

This turning movement of lever *C* also withdraws indexing

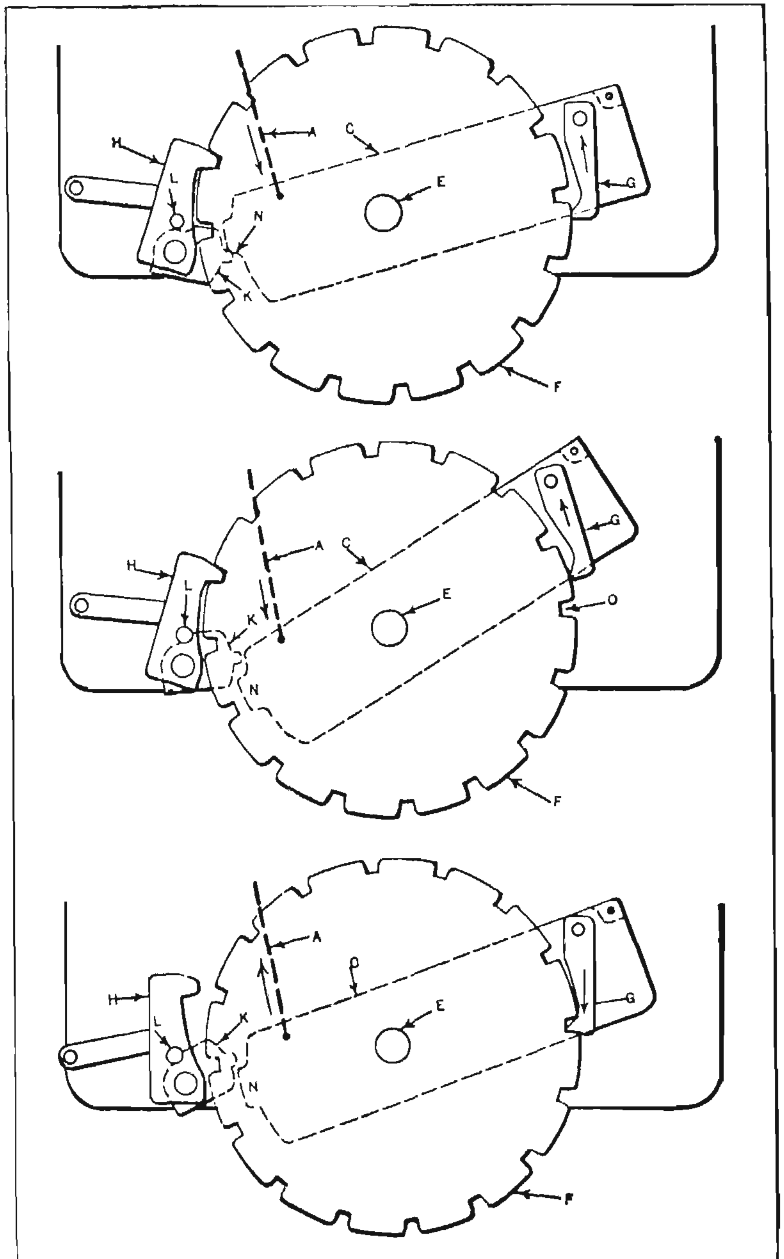


Fig. 12. Three Views Showing the Action of the Locking and Indexing Mechanism of the Automatic Dial Feed

pawl G preparatory to the next indexing movement. The central view in Fig. 12 shows the relative positions of the parts when the ram is near the bottom of its stroke. The projection N has passed lever K, thus allowing lever K to swing back to its position against stop-pin L. Meanwhile, ratchet G has withdrawn nearly a space and a half around dial F.

As the upward stroke of the ram begins, the movements are, of course, reversed, as indicated by the arrows in the lower view, Fig. 12. The chain connecting with the punch is now pulling lever C in the opposite direction. While pawl G is moving from the position shown in the central view around into engagement with slot O, the dial is unlocked by the engagement of lever C with K, which, in turn, acting against pin L, swings pawl H back to the position shown by the lower view. The continued movement of lever C, acting through pawl G, indexes the dial, and just before the ram reaches the top of its stroke, lever K clears projection N, thereby allowing the larger and more powerful spring P, Fig. 11, to swing pawl H into the locking position against the tension of the lighter spring M. This completes the cycle of movements.

It will be noted that the important motions required for unlocking and indexing are derived from the positive action or pull of the chain. This dial feed is used on a press that runs at 90 revolutions per minute. It is advisable to have a hard wood brake Q to assist pawl H in preventing the dial from over-running at the end of the indexing movement. Two guide pins R assist in aligning the punch and in keeping all parts together when the attachment is removed from the press. A hardened steel plate S takes the thrust of the punching, forming, or drawing operation. Spring P is ½ inch in diameter, 4 inches long, and made of 0.060-inch steel wire; spring M is 5/16 inch in diameter, 2½ inches long, and made of 0.035-inch wire; and spring J for the indexing lever is 1 inch in diameter, 4 inches long, and made of 0.080-inch wire.

The particular dial feed illustrated was designed for use on a standard press having a 2-inch stroke, and it is used in assembling small locks requiring a number of operations, such as bending lugs, upsetting pins, etc., the work being indexed successively under the different punches (not shown) attached to the punch-holder. Locating gages or pockets are attached to plate *F* and the completed parts are ejected in front by air pressure. Plate *F* also serves as a bolster plate in order to provide ample die space.

The ease and rapidity with which this dial feed can be placed in position or removed from the press is an important feature of the design, as it can be applied or removed as quickly as an ordinary die having leader or guide pins. Owing to the simplicity of the design of this mechanism, it costs little to construct, so that it is practicable to have a number made for different operations or parts, the self-contained feeding mechanisms being interchanged on the press, the same as dies.

CHAPTER XVII

DESIGN OF AUTOMATIC FEEDING MECHANISMS

WHEN an automatic machine designer has solved a great many problems during the course of his experience, he reaches a point finally where he does not need to do so much experimenting before originating a plan or design for a certain operation. Two experienced men, however, working on the same problem will seldom decide upon the same method of handling, yet both solutions may be equally good.

Two jobs may be similar but they are not often exactly alike. A machine may have been designed and built for a certain operation, and the designer may be called upon to design another for a piece of almost the same shape. A small difference may make it necessary to use an entirely different method of handling. When experimenting with a model, if it is found that it works properly without a slip or failure under every test, it may be considered satisfactory, but when it fails to function *just once,* the trouble must either be overcome or a different scheme tried. It is best to develop an idea along lines that previously have been found successful, whenever this is possible, but for much of this work there is no precedent, and a new idea must be worked out to suit the case. Take nothing for granted, until it has been proved.

A customer may say, "I want tc dump all these three sizes of pieces into a hopper and have them come out through three chutes separated according to their size. Can you do it?" Assuming that the designer is confronted with a case of this kind (which is by no means uncommon) it may be best to answer the question by asking another, for example: "Why don't you separate them first as they come from the manufacturing machines, and keep them separate?" There may be conditions which prevent this, but usually much can be done to simplify a design by adopting improved methods for previous operations. Many people think that one can do anything with an automatic machine, and while there is some truth in this, the cost may be prohibitive. One complicated machine can sometimes be designed for several operations on a difficult piece of work, but it is always expensive and likely to get out of order. It is nearly always better to use two or three simple machines than to make one complicated one, for in the latter case, when anything goes wrong with the machine the entire production stops until the trouble is overcome. The careful designer considers these points before he starts actual work.

A designer is frequently called upon to handle and arrange pieces that are dumped into a hopper and many times a much simpler arrangement could be used by starting at the beginning of the problem instead of in the middle. Any piece of work that is made in a machine is removed from it in a uniform manner, and if, after completion, these pieces are to be packed in a carton, package, or box, the feeding into the packing machine can be greatly simplified by devising a simple arrangement to apply to the machine that does the manufacturing. Such a device can often be made to stack the pieces into a removable carrier or magazine which can then be attached directly to the packing machine. The nature of the work sometimes prohibits the use of anything of this sort, but it is well to keep it in mind as a possibility.

In the selection of pieces from a mass, there are several principles frequently used: Gravity, vibration, oscillation, rotation, and centrifugal force. Some kinds of pieces will fall by gravity, perhaps assisted by vibration of inclined planes or hoppers. Others need to be oscillated in order to disturb the mass and change the arrangement continually. The principle of rotation is often applied in many ways. Centrifugal force can be applied successfully to separating devices, but its application in automatic machine design is seldom appreciated by the average designer. Gravity and vibration are

probably most used, yet both of these methods must be applied properly to secure positive results. If we should place a single piece of work of a given shape in a fixed position on an inclined plane, we know that it will slide downward if the angle is great enough, but if other pieces are in contact with it, we cannot tell what may happen. Several examples will be shown to indicate the advantages of prearrangement of pieces before handling.

Feeding Shallow Boxes Top Side Up. — At *A* in Fig. 1 is shown a cup-shaped wooden disk which is to be passed through a machine and the depression filled with a composition. Our problem calls for feeding the pieces in such a way that they will always come through the machine cup side up. If they are arranged in a stack as at *B*, the matter is quite simple, as they will drop easily into the carrier *C* ready for any other operation.

If we are required to handle the pieces from a hopper, the first step is to flatten them out so that they will slide edgewise down a chute, as indicated at *D*. To do this, they may be placed on a vibrating inclined plate *E* so that they pass through the opening at *F* into the chute. An oscillating brush at *G* can be used to disturb the pieces and break up any fixed arrangement. When the pieces have passed through into the chute *D*, the cup sides will not face in the same direction—some will be one way and some another. Referring to the enlarged view at *H*, the end of the chute is seen at *K* with one piece *L* entering the selector, and another following it at *M*. The selector revolves intermittently in the direction shown by the arrow at *N*. There are six slots equidistantly placed in the edge of the selector, and a guard plate covers its face, as shown at *O* in the sectional view. As the selector revolves, it takes one piece at a time from the chute and carries it around to the point *P*. If the piece has entered in the position shown at *Q*, it drops down through the chute *R*, but if it has entered in the reverse position as at *S*, it is carried around to the next station *T* and drops through the other chute *U*.

The mechanism governing this is simple. The revolving selector V has at each one of the six slots a spring plunger W, at the end of which there is a hardened roll X which travels around the circular plate Y. This plate is continuous except at the point P, where it is interrupted. If the cups face in the direction Q, the light spring behind plunger W

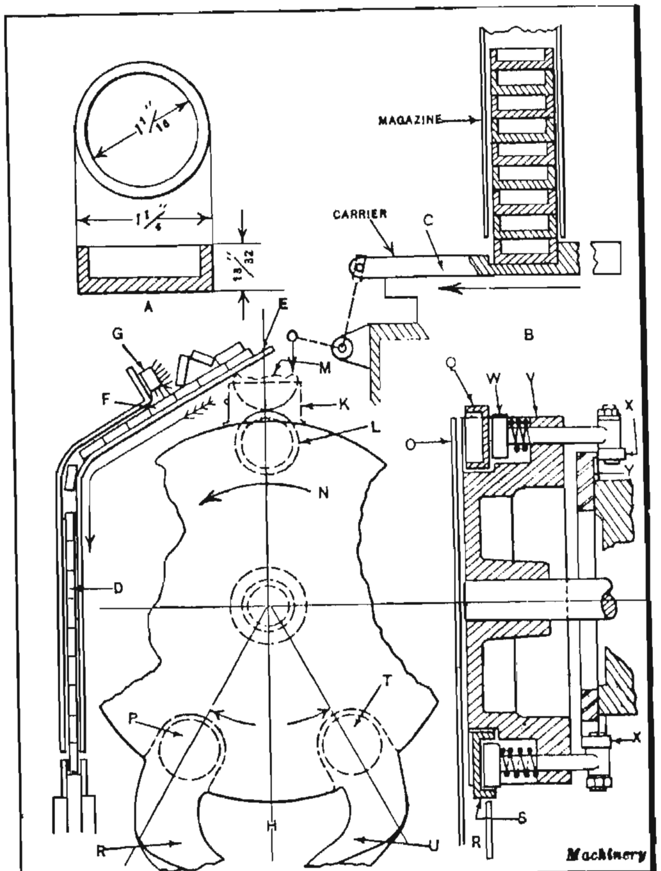


Fig. 1. Feeding Device for Cup-shaped Wooden Disk Shown at A

simply throws the piece out into the chute at R, but if the cup lies as at S, the plunger enters the cup and prevents it from dropping until the selector has passed that point. Before reaching T, roller X rides up again on plate Y, thus withdrawing the plunger and allowing the piece to drop into

chute U. It is only necessary to twist the chutes R and U in opposite directions, making a quarter turn, to have both pieces come out facing the same way at the bottom of the chutes, at which point they can be easily handled as required. The spring behind the plunger must be very light in order to prevent friction when the pieces slide out.

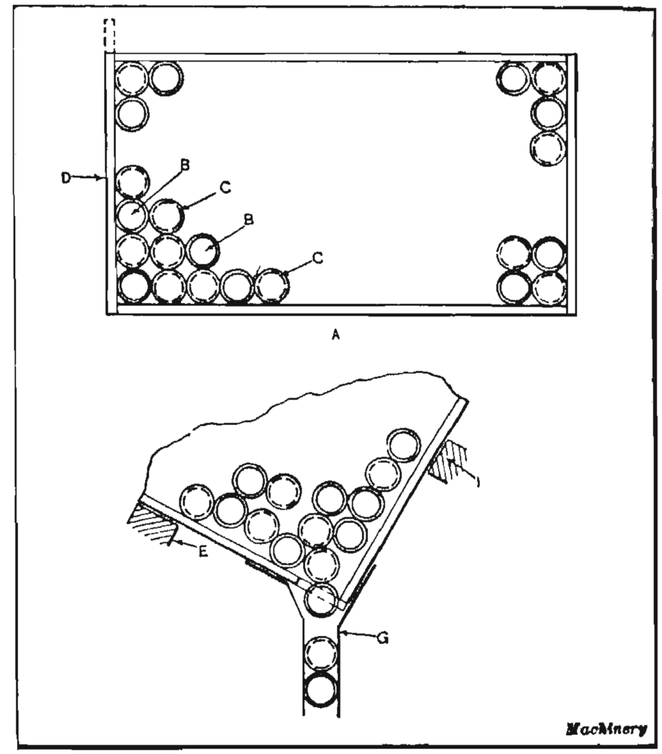


Fig. 2. Inclined Tray Method of Feeding Pieces Into a Chute

Considerable expense can be saved by using trays in which to arrange the pieces before feeding them into the machine, instead of using a hopper. Fig. 2 shows at A a form of tray that can be used, the pieces being rapidly spread out by hand so that only one layer is in the tray. Some of the pieces will be right side up, as shown at B, while others will be upside down, as at C. If the tray is made with one end D so that it can be pushed back far enough to allow one piece at a time to come through, this opening will serve as a gate, and the tray can be set up on an incline and against supports at *E* and *F*, so that the pieces will drop down one at a time into the chute *G*. From this they can go into the selector as in the preceding case.

Feeding Mechanism for Plain Flat Disks.— Let us take another example of a disk *A* (Fig. 3). This is just like the other, except that it has no depression and could be handled from the hopper or from a stack by means of a reciprocating slide, as shown at *C* in Fig. 1, or by a rotating selector without the plunger. If flat pieces are being handled which are to be packed in bottles or boxes, the preceding method would be much too slow. Assuming that the tablet shown at *A* is to be deposited in a bottle, thirty pieces by count, it would be out of the question to use any such method as that mentioned. The pieces are to be dumped into a large hopper, several thousand at a time, as the requirements for production are such that not more than one second can be allowed for counting and putting them into the bottle. It would be difficult to make a counting device to operate as rapidly as this and with certainty, but the pieces could be weighed, or they could be arranged thirty in a row approximately 11¼ inches long, as shown at *B*. If they could be arranged this way, it would still be difficult to keep the chute full and deliver the pieces as rapidly as required.

If the pieces are dumped into a hopper, the previously demonstrated methods can be employed for making the parts come through an opening flat side down on an incline great enough so that they will slide down it to a gateway shown at *C*. This gateway is open at *D* over a rubber-faced disk *E* which revolves slowly. A guard strip *F* runs across the revolving table at an angle, and there is also a guard at *G*, the lower end of which blends into the chute *H*. From point *K* to *L* there is just room enough for thirty pieces to lie. In operation, as the pieces drop through on the disk at *D*, the circular movement combined with gravity causes the pieces

to roll rapidly down the incline and into the chute *H* until they have filled it completely. They will then back up against the guard *G* until relieved by the opening of valve *M*. This is timed so that, as it opens, another valve at *N* closes. As there are four or five pieces *O* still in contact with the rubber

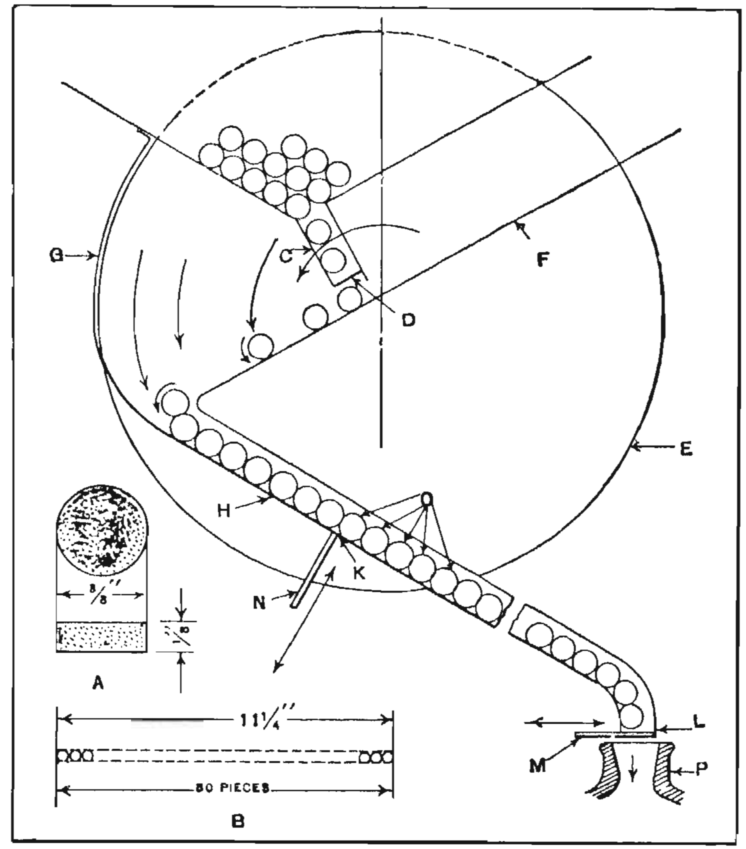


Fig. 3. Principle Applied in Counting a Large Number of Pieces

face of the disk, these will receive a rapid movement from it, pushing out those ahead so all the thirty pieces will shoot into the mouth of the bottle *P*.

Care must be taken in designing a mechanism of this sort to work out the important points by means of a model, as otherwise it will not function satisfactorily. A slight change in the angle of the guard plates, chutes, and the speed of the rotating disk make a great difference in the operation, but if the principle is understood, it is comparatively easy to make up a simple model which can be used for demonstrating purposes and to obtain the right relation of the various guards and chutes.

Filling Small Boxes with Tablets.— Fig. 4 shows at *A* a pasteboard box containing twenty-four disk-shaped tablets.

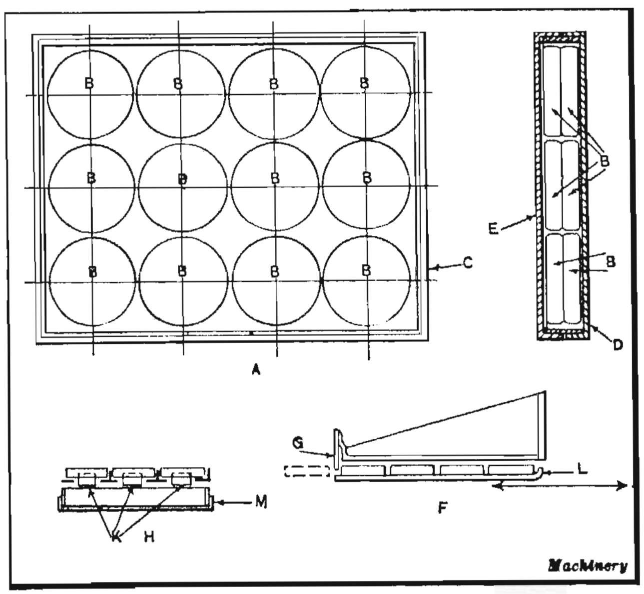


Fig. 4. Method of Packing Pieces Into Boxes

They are arranged in the box in layers, as shown at *B*. The problem is to take the pieces from a hopper, arrange them in the box as shown, and put on the cover *E*. The production required is not less than twenty boxes per minute, and therefore not over one second must be consumed in putting in each layer. Arranging the pieces and putting them in the box is not particularly difficult, but we must also feed the boxes and covers into position and put on a cover. It is advisable to use two sets of chutes for the two layers in the box: this can

be done easily enough, although care must be taken to provide a sufficient supply so that both chutes will be kept full and still have a reserve. By using two chutes, one layer is put into the box which then moves over to the other chute and receives the second layer. If the chutes are arranged properly, the production time will be only that required for putting in one layer.

The arrangement for feeding is simple, and needs but a brief description. The tablets lie in a nesting device, a part of which is shown at *F*. They are held back by a rubber cowl *G*, and lie between guides shown in the end view *H*. These guides are open at the bottom and contain long steel fingers *K*, having a hook on the end at *L*. At the proper time these fingers move forward over the box *M*, pulling with them the tablets, which pass under the cowl *G*. The fingers *K* then move backward again, and as they do so, the cowl *G* restrains the pieces from following, and they drop into the box one after the other in their proper arrangement. It is advisable to make the fingers lie as close to the top of the box as possible, in order to minimize the amount of drop.

Let us now consider the handling of the box and cover. As both of these are rectangular and regular in shape, it is not difficult to arrange a feeding device for them, but the putting on of the cover is more difficult. If we take up one thing at a time, and decide to load the boxes from a stack, as shown at *A* in Fig. 5, we must realize that even with fifty boxes to a stack, a new magazine full would be required about every two minutes and a half. Therefore, it may be well to arrange for an indexing holder *B* containing six magazines, which can be quickly removed and replaced when empty. A device can be made which will index the table one station at every fifty strokes, either (1) by using a ratchet and pawl and a dog, which will drop into an index-plate at the proper time; (2) by putting exactly fifty holders on the conveyor belt and using a suitable dog on the side to index the table; or (3) by means of a cam. The first or third method is to be preferred.

The conveyor belt *C* removes the boxes one at a time by an intermittent movement, and carries them along to the first loading station D, at which point the first layer of tablets is put in by the method previously described. The conveyor is arranged with cross-pieces E spaced only a trifle further apart than the width of the box. As it passes under the magazine

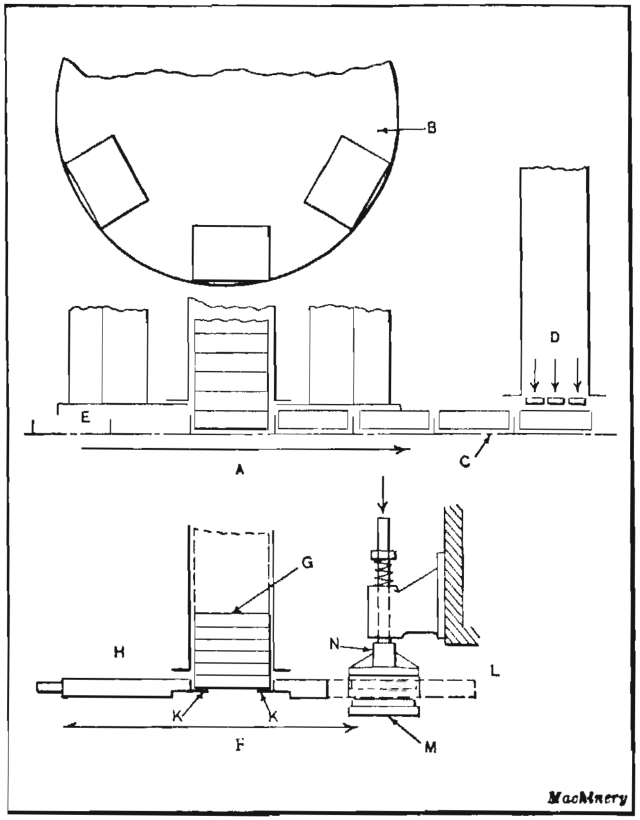


Fig. 5. Feeding the Boxes Into the Machine and Putting on the Covers

it stops, allows one box to drop, and continues to do this regularly. The next step is to put on the cover.

Placing Covers on Boxes.—By arranging the covers in a stack of similar form to that used for the boxes, a reciprocating slide can be used to remove them from the bottom of the stack, as shown by the diagram at F. Fig. 5. The cover G

drops into the slide H and lies on top of two flexible rubber strips K which prevent it from falling through. At the proper time, the slide reciprocates and takes the position shown by the dotted lines at L, the cover then being directly over a box full of tablets shown at M. A pusher N lies directly over the

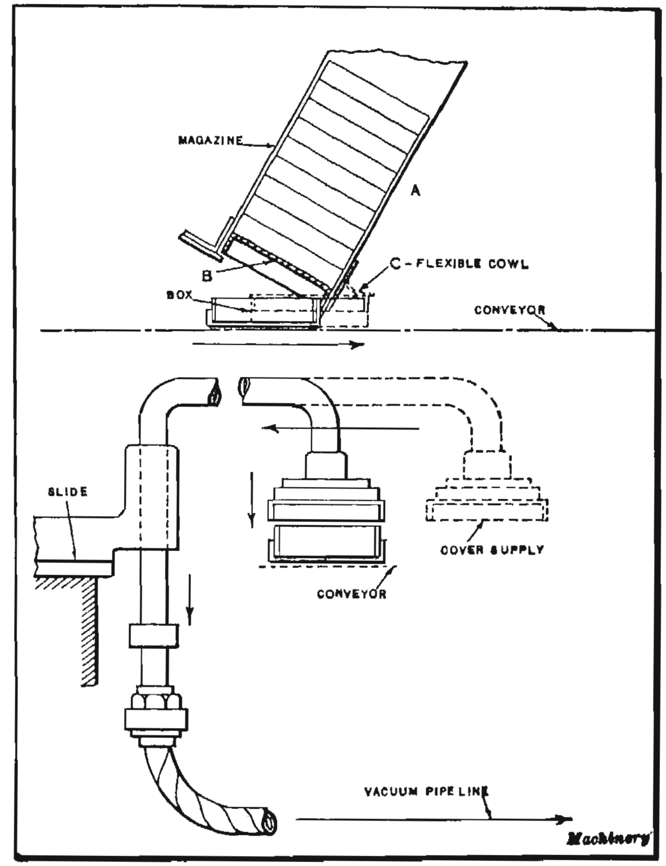


Fig. 6. Other Methods of Putting Covers on Boxes

cover, and is used to press the latter down on the box. The control of this mechanism is obviously very simple and does not need detailed description.

There are other ways of putting on a cover, one of these being indicated in Fig. 6 at A. Here the chute is set at such an angle that one side of the cover hangs low enough to be caught by the edge of the box *B* as it passes under it. A light rubber or cloth cowl at *C* drags over the cover as the conveyor pulls it through. This device might be satisfactory if it did not depend on gravity for the operation, but unless the covers are fairly heavy there is a possibility of a miss now and then.

The lower view shows still another method often used for picking up pieces of this sort. Here the covers fall down a curved chute and come to rest as indicated. A perforated suction plate picks up one cover by vacuum, moves it over to the box and presses it down by a vertical movement of the ram. This device is rather more complicated than the others, but could be made very satisfactory. The pneumatic method of handling can often be used for conditions of this kind, but care must be taken with any materials that lie close together and that are somewhat porous, as there is always a possibility of the suction going through more than one piece and picking up two, when only one is wanted.

It will be noted that in working out this problem it was considered in unit form, and attention has not yet been given to the power application. Often a great deal of unnecessary trouble is caused by the designer attempting to decide upon the method of driving and timing up a unit during the progress of the design. The units can be designed and located from the preliminary freehand sketch, and after their positions have been settled, the main driving shaft cams, levers, etc., can be placed most advantageously.

CHAPTER XVIII

HOPPER DESIGN FOR AUTOMATIC MACHINERY

WHEN materials are to be handled in bulk, they are often dumped into a hopper, from which they are removed through an opening in the bottom. It is difficult to classify the various kinds of hoppers and divide them rigidly into groups, but by considering the kinds of materials handled, we can make an arbitrary distinction sufficient for our purpose. The following list covers the types most commonly required:

1. Granular materials, which flow readily.
2. Liquids of all kinds, heavy or light.
3. Plastic materials, such as cement, asphaltum, and other mixtures, candy in certain forms, cream cheese, flour dough, etc., some of which are sticky or spongy.
4. Disk-shaped or oval pieces of uniform size and shape.
5. Cylindrical or conical pieces of uniform size or shape.
6. Irregular shapes of various kinds.

The first thing to consider in the design of a hopper for materials of any kind which are to be handled in bulk, is the general shape of the material; and the next, the manner in which the pieces or materials can be taken from the hopper—that is, their arrangement as they issue from it. Problems connected with these two points are some of the most important in automatic machine design, for upon their correct solution the functioning of the machine is largely dependent. A hopper may be cylindrical, conical, rectangular or any other shape best suited to the material. The different conditions encountered can best be understood by citing a few examples.

Hoppers for Different Substances.— Granular substances, unless they are very irregular (as might be the case if the material is formed in crystals) will fall down almost any kind of an incline and through almost any form of opening. Liquids, of course, will do the same, but for pasty or gummy substances a gravity feed is usually impractical, and the ma-

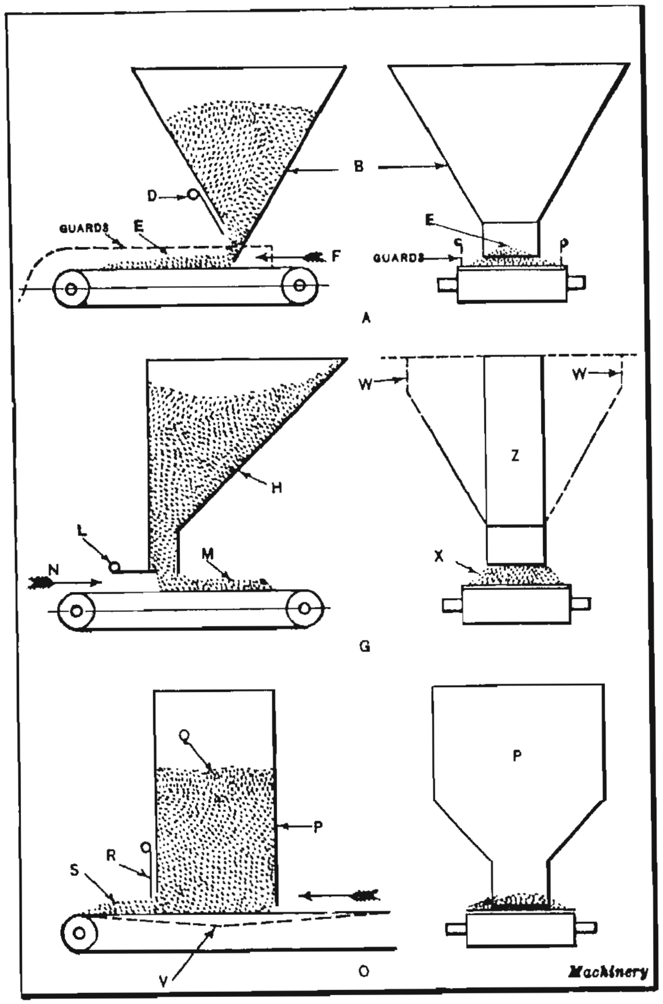


Fig. 1. Examples of Hoppers Used for Granular Materials

terial must be forced through the hopper by mechanical means. Disk-shaped or oval pieces are often fed by gravity and removed by inclined chutes. In such cases it is frequently neces-

sary to use an agitator or some other form of mechanical feed to keep the flow constant. Cylindrical or conical pieces are very likely to clog in a hopper, and must be stirred up continually or agitated to prevent lodging. In some cases it is not even possible to use a regular form of hopper, and the pieces must be spread out on a large inclined plane. In analyzing a given condition, think first of the arrangement of pieces as applied to the method of handling after the work has passed through the hopper.

Examples of Hopper Design. — Fig. 1 shows several examples of different forms of hoppers, to illustrate the manner in which the material passes through the hopper and is discharged from it. In example *A*, the hopper *B* is of pyramidal form. The material has no tendency to wedge or crowd, but flows readily in a continuous stream to the bottom of the hopper at which point an adjustable gate or valve *D* controls the amount passing through. If handled on a belt moving in the direction *F*, the material will naturally spread out as shown at *E*, so that it can be carried and delivered in the quantities required. Side guards can be used (as shown by dotted lines) if desired. In example *G*, one side of the hopper *H* is straight and the other is angular. Often the space available makes it more convenient to have a hopper of this form. In the other direction, the sides may be straight, as at *Z*, or tapered, as at *W*. A valve at *L* permits adjustment according to requirements, and the material is deposited on the belt *M*. Care must be taken in proportioning the opening of the hopper so that too wide a mass is not distributed on the belt, as shown at *X*, for in a case of this kind there would doubtless be considerable loss as the belt moves along.

In example *O* a somewhat objectionable form is indicated at *P*. With certain materials there might be no great objection to this form, but when handling anything of a heavy nature, the weight of the rectangular mass at *Q* might be sufficient to deflect the belt, as shown at *V*, thus causing variable amounts to pass by the valve *R*. Assuming that the belt travels in the direction of the arrow, the material would be drawn out at S and carried away as needed. With a wide-mouthed hopper like this, a great deal of pressure is continually being exerted by the column of material above the belt, and this pressure varies according to the quantity of material in the hopper. Conveyors for this sort of material are often made concave and lie on three rollers, in which case no guards are required, as the concave form keeps the material from falling off.

In Fig. 2, the example shown at K is a pyramidal hopper L containing a granular material M which is to be removed,

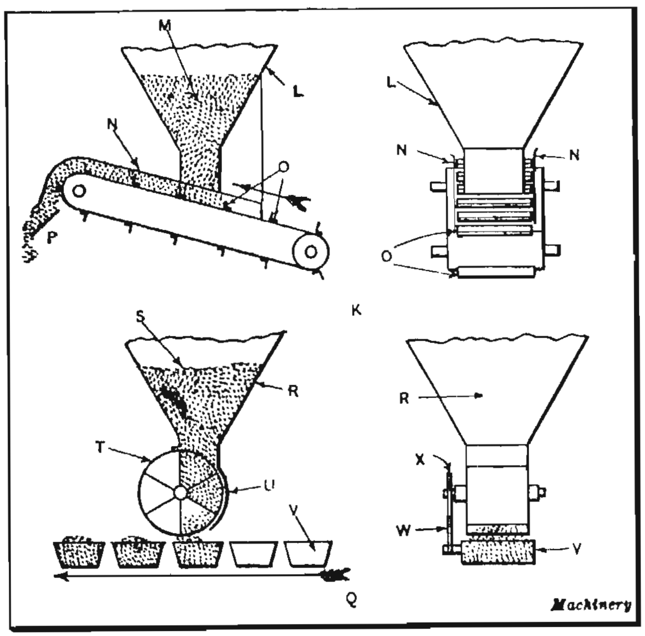


Fig. 2. Guarded Hoppers for Granular Substances

carried upward, and discharged at point P, intermittently. A plain belt conveyor is used, and on this are fastened at equal spaces, the clips O which hold the material and prevent its rolling down the incline. The guards on each side at N lie close to the cross strips and have leather or rubber ends which prevent loss of material. This type of hopper is useful when granular materials are to be fed in approximately uniform

quantities and carried from one level to another. The example illustrated was used to remove bulk candy and transfer it to a higher level, where it was spread out into distributing chutes.

Valve Arrangements in Connection with Hoppers. — The example shown at Q, Fig. 2, is quite different in its general form, although the material is held in a hopper R of the ordinary form. The lower part of the hopper is narrowed so that the material passes down through it in the form of a rectangular column. A rotating valve T having six openings of equal size is located as shown, and one portion of the hopper U extends down around the valve. When in the position shown, the material fills the compartment shown uppermost and the guard U prevents the previously filled compartment from emptying until it is nearly upside down. The buckets V pass along under the valve, and a given quantity of the material is emptied into each bucket as it goes by. The movement of the buckets may be regular or intermittent, and the rotation of the valve can be easily controlled by means of projecting arms on the buckets, as shown in the side view at W. These arms can be made adjustable and placed so that they will strike a spider X and carry it around a given distance as they move along. In designing devices of this kind, it is a good plan to make the length of the arms adjustable and the positions of the striking portions on the buckets also adjustable, in order to compensate for slight variations in movement.

Another form of valve arrangement which is independent of the movement of the buckets, but is timed to suit their progress, is shown in Fig. 3. The usual form of hopper A carries the material B in the pyramidal portion, but the lower part is rectangular and quite long, being so proportioned that its contents will fill one of the buckets N. The valves C and D lie in guides at the top and bottom of the rectangular portion of the hopper. In the upper valve, there is an opening E, and in the lower an opening F, the two openings being so placed that when one is in line with the hopper the other is closed. The arm *G*, pivoted at *K*, connects the valves by sliding joints at *H*. Lever *L* connects with rod *M*, which pivots on block *O* in slotted disk *P*. A reciprocating movement is transmitted to lever *L*, as the disk revolves.

The kind of material that is being handled determines whether a small amount of leakage is permissible or not, but in normal cases the slide valves can be made of cast iron or brass working in guides of steel. Careful fitting is not usually

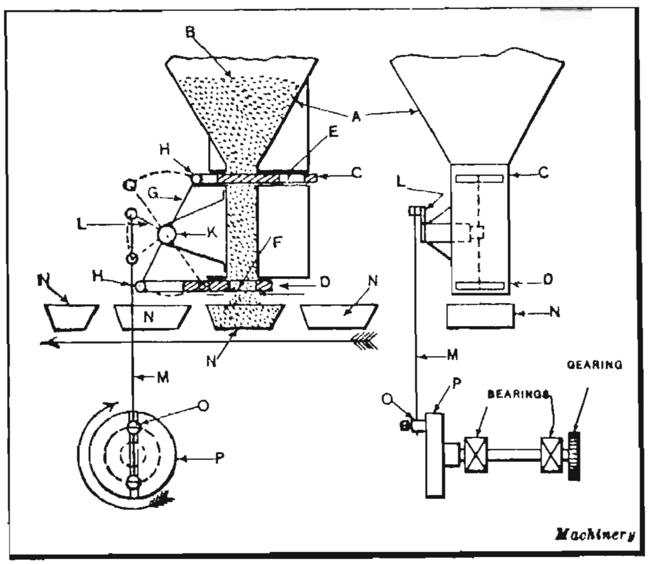


Fig. 3. Arrangement of Hopper Valves for Handling Granular Substances

required on this kind of work, but the guides should always be long enough to permit a free sliding movement and good alignment.

Continuous Rotary Hopper Valves. — In the example shown in Fig. 4, a continuous rotary feeding device is shown. The hopper *O* and the valve *P* are similar to that shown at *Q* in Fig. 2, but the operation is by means of a ratchet gear *R* mounted as shown. The arm *T* has a pawl on it at *S* so arranged that it engages with ratchet *R*. The disk *X* is slotted at *W*, and the block *V* is adjustable in the slot to control the

movement of lever *T* through the connecting-rod *U*, as in a preceding example. The disk is driven by a chain from the sprocket *Y*, and the buckets *Q* are controlled in their movement by suitable gears or sprockets and chain running from the same shaft. When two separate units are to be timed accurately, a chain drive or gearing should always be used.

Fig. 5 shows a form of rotating hopper valve which may occasionally be found useful when quantities are to be measured and deposited accurately in buckets or other receptacles. The hopper *A* contains the material *B* which flows downward until it enters the section *H* of the rotary valve. There are

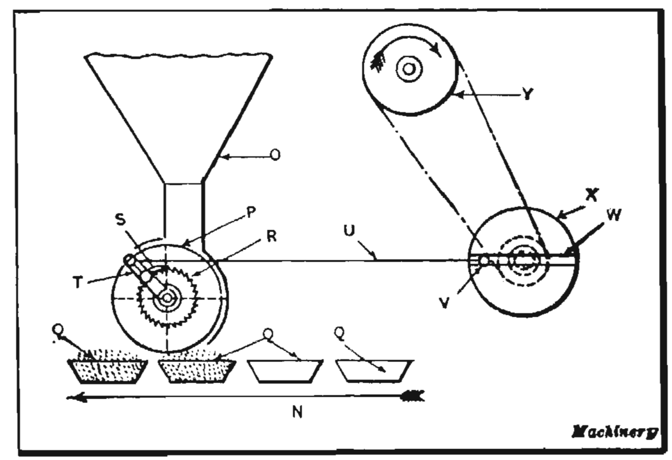


Fig. 4. Continuous Rotary Feeding Devices

six chambers *H* arranged radially about a center on which the valve rotates. The entire valve mechanism is supported by brackets *C* and *D* arranged according to the general design of the mechanism. The valve plate *F* fits closely against the under side of the hopper, and in each compartment there is a hole leading directly into the throat of the hopper. The plate *G* also has a series of corresponding holes, but there is only one opening in plate *K* from which the chute *L* leads. As the buckets *M* pass under the spout *L* the indexing mechanism operates at the proper point and discharges the contents of one of the compartments into the buckets, as indicated. If small quantities only are delivered and if the conveyor movement is proportioned properly, the materials can be dumped without stopping the conveyor, but for larger quantities it may be necessary to use intermittent gears or some other convenient method to stop the movement for a moment when the valve L is open to insure delivering the required amount.

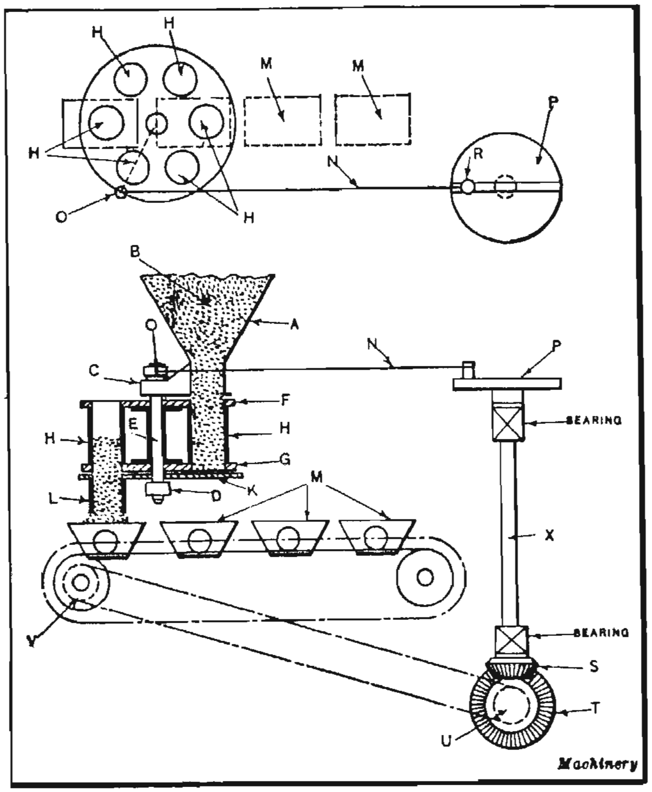


Fig. 5. Rotating Hopper Valve Mechanism

The operating mechanism of this device is very similar to one previously described, the lever O being attached to a pawl working on a ratchet gear, the movement of which is controlled by the rod N which pivots in block R. The disk P has a slot in which the block may be adjusted to govern the movement of the pawl. By driving shaft X from the bevel gear S by means of the ring gear T, the "take off" for the conveyor comes from the same shaft through sprockets U and V connected by a chain. Naturally, the location of the various driving units is dependent largely upon the general design of the machine and the other portions of the mechanism which must be driven. It is, of course, essential that all units which must function in a certain relation to each other must be controlled from a single shaft. When accurate timing is not necessary, intermediate gears and lever movements can sometimes be used to simplify the design.

Hoppers for Liquids.—So far we have considered only the hoppers used for materials that flow readily by their own weight, and as liquids of thin consistency are led through the hoppers in much the same manner, we will consider the problems arising in their handling. Although granular substances and liquids will both flow readily, there is considerable difference in the valve arrangements. In granular substances, a small amount of leakage around the valve is not serious, but when liquids are handled, more attention must be paid to this feature. Several points of importance should be noted:

1. The nature of the liquid, its value and its consistency are matters that affect the design of the valve mechanism. Waste and cleanliness do not go together; if you were to see any machine handling liquids which were dripping all over it, you would not think the machine thoroughly efficient. It is not a good plan to allow surplus material to run down or overflow into a tank or tray arranged to carry it off, unless for one reason or another the liquid must be flooded over the receptacle. Conditions of this kind are sometimes found in the making of molded products in order to be sure that the molds are thoroughly filled. A thin liquid will flow readily through a valve which can be shut off by suitable mechanism without difficulty, but a heavier fluid may drip all over the machine and the receptacles unless a suitable valve is used.

2. The distribution of the product into containers affects the design of the valve. Measuring valves are most commonly used for controlling the flow of liquid. When the valve

opening is very small, the pressure of the outside air may be sufficient to prevent the liquid from flowing unless a "vent" is provided. It is usually necessary to provide some means also for controlling the drip from the spout to prevent waste.

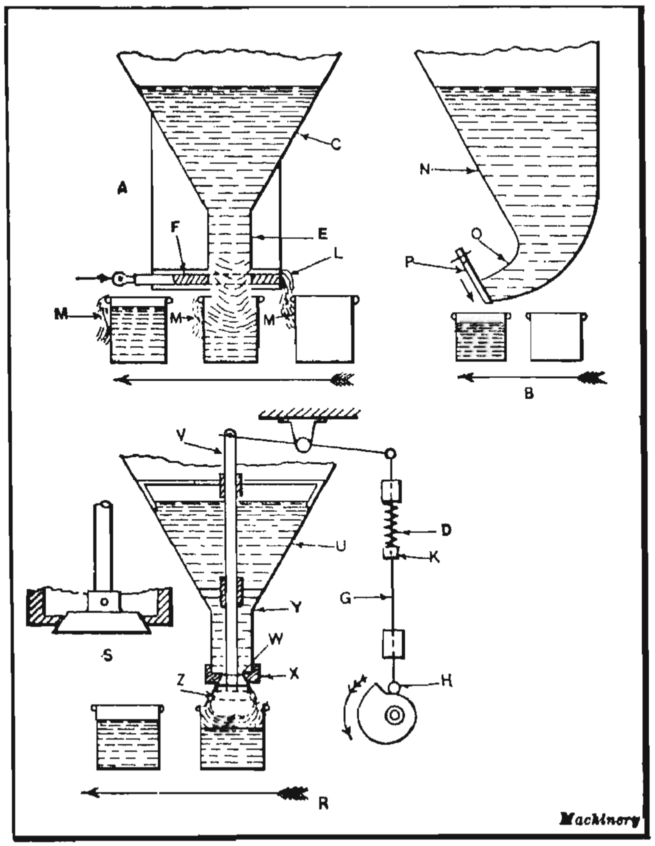


Fig. 6. Containers and Valves Designed for Thin Liquids

Valves Used for Thin Liquids. — Fig. 6 shows several forms of valves used for thin liquids. Some of these are good, while others are objectionable. At *A* the hopper *C* is of angular form. A point of importance is the variable pressures caused by small or large quantities of liquid in the hopper. The method of supply makes considerable difference in the

design of the hopper, and it is quite common to find a cylindrical form used with a conical bottom leading to another cylindrical portion as shown at *E*. If the liquid is poured into the hopper from time to time, the pressure on the valve is variable, but if it flows into the hopper through a supply pipe, a float valve can be easily applied which will shut off and open the pipe as the liquid falls or rises in the container. By this means the pressure can be kept nearly uniform, and better results can be obtained. The type of valve shown in this example is unsuitable for use in handling liquid. The valve *F* slides backward and forward through guides of similar form to those used in handling granular substances, but it is almost impossible to make a mechanism of this kind tight, and the liquids will drip out at *L* and run down over the sides of the containers below. If the fluid is sticky, it may cling to the outside, as shown at *M*. If not, it will run down the sides and get all over the machine. In cases where the ends of the valve are covered, material is likely to collect there and cause trouble in cleaning.

The form shown at *B* is considerably better, as the hopper *N* terminates in a spout *O*, and the fluid is held back by a hinged shut-off valve *P* which fits tightly against the face of the opening. As this valve is operated it cuts off the supply in such a way that there is only a small amount of drip. This can be caught without difficulty by the containers, and although a few drops may remain with the heavier variety of liquids, those that are thin will not drip to any great extent. This form of shut-off valve is very frequently used and is recommended for its simplicity.

A form that is adapted to a variety of uses is shown at *R* in the same illustration. The hopper *U* contains a liquid which flows in the lower part and against the valve *W* which prevents it from passing through until the valve is opened. The valve seat *X* is fastened to the lower part of the hopper, and the angular surface of the valve keeps it tight. The rod *V* is guided as shown, and is connected by a sliding joint to an operating rod. This is pivoted in a bracket which may be attached in some convenient way, depending upon the design of the machine. The connecting-rod *G* has a roller *H* at the lower end which contacts with the cam shown. The roller is kept in contact with the cam by the spring *D* which thrusts against the collar *K* on the shaft. As the cam revolves in the direction shown by the arrow, the valve is opened and the liquid flows down over the conical portion *Z* and into the container. The spring *D* closes the valve quickly when the roller drops off the cam, but the spring must be fairly stiff in order to make a tight joint. The enlarged detail at *S* shows the construction of the valve seat and the valve. The conical surface should be made as narrow as possible, as it is easier to keep it properly fitted if this is the case.

When any sort of liquid is to be handled through a valve, it is very important that the construction permit quick and easy removal of the units for the purpose of cleaning. This point is of great importance when handling any sort of food product or any liquid that has a corrosive action on the metal. The selection of metals used in valve construction is also an important factor in the design. Stainless steel can often be used in valve work.

Experiments will usually show the effect of a fluid on a certain kind of material, but it is important to have all conditions that will obtain in using the machine fulfilled when making the test. Rubber is sometimes used for valves, and in such cases replacements should be made as easy as possible, because frequent renewals are necessary. A designer who does not consider such points as these fails in the essentials of good design. When considerable labor is necessary in replacing a certain part subject to wear, or when cleaning the machine after the day's work, this labor is just so much lost time that might have been avoided by greater forethought in designing.

CHAPTER XIX

MAGAZINE FEEDING ATTACHMENTS FOR MACHINE TOOLS

THE magazine attachment in any automatic machine must, first of all, deliver the work, even if it should be of irregular form, in a uniform manner. Some pieces are much more difficult to handle than others; for example, the bevel gear forging *A*, Fig. 1, is larger and therefore much heavier at one end than at the other. Also, as it is a forged piece, it may have fins, seams, or other small projections which will interfere with smooth movement. Obviously, a straight-guide magazine cannot be used for this piece, assuming that it is to be machined on centers or in a chuck. In any case it must be delivered from the magazine with the same side always in a given position. When irregular pieces are to be handled in a magazine, the greatest care must be used in planning the arrangement, in order to prevent wedging, interlockng, or cramping of any kind. Such conditions may develop from many causes, such as too much or too little clearance in the magazine guides, or a lack of forethought in arranging the pieces. Also, if forgings or castings are made from several dies or patterns, they will likely vary somewhat and cause trouble in feeding.

The first step in designing a magazine for an irregular piece is to look carefully at its general shape, the size of the fillets, and the location of seams, fins, or raised numbers which may prevent a smooth rolling movement. In considering the work shown at *A*, we know that it will undoubtedly have fins longitudinally where the dies part; that fillets will be in evidence at the junction of the head with the stem; and that it must be delivered from the magazine in about the position shown. It is often necessary to find the approximate center of gravity of a piece in order to determine whether it will be likely to overbalance if handled in a certain way or in a particular position. Usually this can be determined by laying the work on a straight edge, such as a ruler or scale, or even by balancing it on the finger, and then marking the approximate center of gravity with a piece of chalk.

A sample piece is not always available in the preliminary stages of design, but an approximation can be obtained graphically from a drawing of the piece or from a wooden model

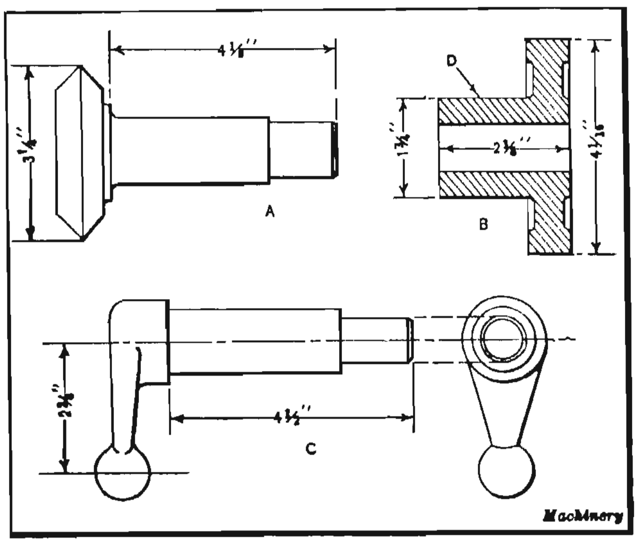


Fig. 1. Examples of Irregular Work for Which Feeding Magazines are to be Designed

which can usually be made in a short time. When gravity is used entirely for feeding the pieces, the possibilities of friction must not be overlooked, as much trouble is caused by neglecting this important factor, particularly when the pieces are arranged to slide down an inclined plane or when they rub against each other in the magazine. All surfaces which come into contact with the work or on which it slides, should be made narrow in order to reduce friction to a minimum.

In determining the shape of a magazine for a given piece,

several sketches should first be made of different schemes, and then a careful examination will usually show which idea is most practical. The use of paper, pasteboard, tin, and wooden models will often lead to an excellent development of an idea, the operation of which, on paper, might be problematical on account of unforeseen conditions. By this means it is possible to determine, without great expense, whether a certain mechanical contrivance will function properly.

One of the most important points in machine design is the elimination of the element of chance in the operation of some unit. A design may look all right on paper and may be mechanically correct, yet when it is built, some unforeseen condition may prevent it from functioning properly without more or less expensive changes in the construction. If the actual operation of a unit could be seen before it is built, many troublesome factors would be eliminated. The designer will do well to experiment with simple inexpensive models whenever there is an element of doubt regarding the working of a mechanism.

Designing a Magazine for a Bevel Gear.— Having considered the arrangement of pieces so that they will come to the carrier in the desired position, we have only to decide upon the best design of magazine. Gravity feeds are unsatisfactory for some kinds of work, and it is necessary to resort to mechanical operation. However, this is seldom required in magazines for feeding bar work, forgings, or castings of the sort being dealt with. There are several ways in which forging *A*, Fig. 1, can be fed. One form of magazine is shown in Fig. 2, the plan view clearly showing the general arrangement of the guide plates. In the right-hand elevation the magazine is shown emptying a piece into carrier *A* in which it is clamped by spring *B* and carried over to the centers at *C* on which it is to be turned. The clamp is withdrawn, as shown at *D*, after the chucking. In this magazine the upper pieces are held back by the shutter mechanism *E*. An adjustable stop for the carrier is provided at *F*.

The fillet at the point where the stem of the forging joins the head might cause trouble unless clearance is provided. If there were only a small fillet, there would be no trouble, but a large one might result in wedging and stop the progress of the pieces through the magazine. If designed with sufficient clearance to take care of all possible variations, there would doubtless be no trouble with this design. The general

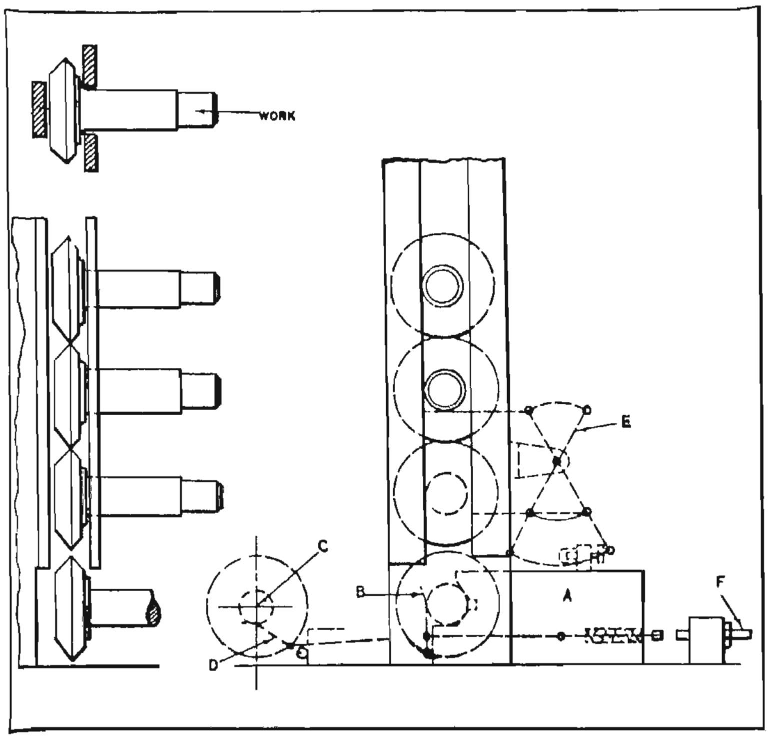


Fig. 2. Example of Design for Bevel Gear Magazine

construction of a magazine of this sort needs no particular comment, but it is advisable to make the back plate and one side plate adjustable so that the magazine can be set to suit varying conditions.

Magazine Used for Two Operations. — Example *B*, Fig. 1, is a small bronze casting which is to be fed from a magazine and faced, turned, and bored while held by stem *D*. In another operation, the work is to be held on a plug and gripped on the large finished diameter while the other side is finished and the chucking stem cut off. Assume that a magazine adjustable to suit both operations is highly desirable, as the work is to be done in a special chucking machine and the device used in connection with the turret. The magazine

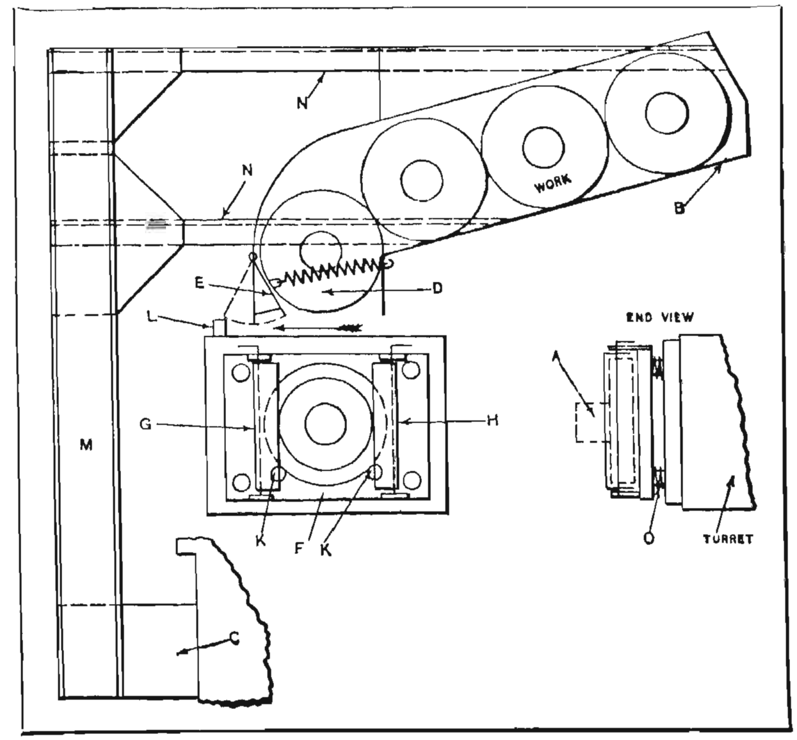


Fig. 3. Turret Lathe Magazine for Feeding Casting B, Fig. 1

should be set in such a position that at one complete revolution of the turret the chucking device would receive work from the magazine. The turret would then move forward and allow the piece to be gripped by the chuck jaws.

Fig. 3 shows the general arrangement of a suitable magazine for this piece, and its position with respect to the turret. It is loaded from the front of the machine by the operator, who can obviously attend to several machines without difficulty. Sufficient details of construction are shown for the designer to understand, but most important of all are the principles involved, as these can frequently be applied to other work of a similar kind. Magazine B should be supported in some suitable manner, depending upon the type of machine and the position in which the magazine is held. In this case bracket C is fastened to a pad at the back of the machine and on it is mounted a structural steel channel M, to the upper part of which angle-irons N are fastened. These extend forward and are riveted to the sheet-metal guides of the magazine. It is usually much better to arrange the supporting members at the back of the machine rather than in front for several reasons, one of these being that there is less likelihood of interference with the working parts of the machine, and another, that, when so situated, it is not in the way of the operator.

The magazine chute is constructed of sheet steel, bent to shape, and is slightly inclined to allow the pieces to roll down easily to point D, at which position the first piece is supported by the hinged valve E. The turret chucking device F is fastened to the face of the turret, and has metal guides G and H at each side. The two locating pins K should be made adjustable to take care of variations in the size of the pieces handled. On the upper side of the turret, a trip L is located in such a position that when the turret is indexed to the location shown, the trip strikes valve E and opens it, as indicated by the dotted lines, thus allowing one piece to fall down into nest F. After the piece has dropped, the turret moves forward and carries the work to the chuck, which should be air-operated and have jaws of suitable form to grip the stem ready to machine the part. After the work has been done, the jaws should open automatically, and the piece drop down into the bed of the machine, leaving the chuck ready to receive another piece. Coil springs O allow for variations in placing the work in the chuck.

In designing a magazine and carriers of this sort, the designer must follow through the entire process carefully, always keeping in mind that when one piece is removed from the magazine another immediately takes its place, and so provision must be made to receive it and not permit it to follow the first piece into the carrier. When springs are applied for

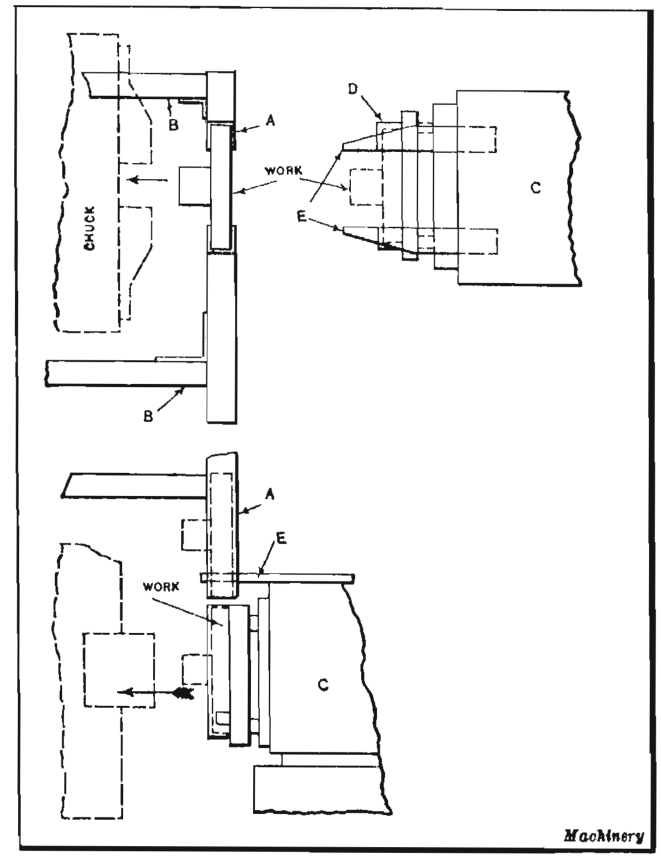


Fig. 4. Turret Lathe Magazine Placed Directly Over the Spindle

closing magazine valves, a little experimenting is sometimes necessary to obtain the proper form of spring and one that has the required strength. Also locations of pins on which the work rests may not always be correct, and then the piece will not be properly centered for chucking. Therefore, it is generally better to make such points adjustable so that suitable settings can be made without difficulty.

Magazine Located Above Spindle.— Another magazine for placing work in the chuck is indicated by the diagram Fig. 4. This magazine is for parts of the general form shown at *B*, Fig. 1. Here the chute *A* is arranged above the spindle, and is supported by brackets *B*. The design of the chute is much the same as in the previous example, but the work comes down and rests against a double valve instead of a single one. Turret *C* comes forward at the proper time with a carrier *D* having projecting fingers *E* above it which open the valve and allow the work to drop into the nest on the turret. A continuation of the turret movement carries the work forward to the chuck jaws. The dotted diagram in the lower part of the illustration shows the position of the work in the carrier after dropping and before chucking. In almost every form of magazine, the carrier or chucking device design is controlled by the shape of the work and the form of the chute. When mounted on the turret and used to push work into a chuck, such a device must be flexibly mounted if the nature of the work requires that it should be pushed back firmly into the chuck jaws. When this is not necessary, the movement can be controlled by the cam that operates the turret.

Magazine Placed at Rear of Machine.— A magazine for application to the rear of a machine is shown diagrammatically in Fig. 5. The mounting is similar to those used for bar or other work held on centers. The work lies in the chute with one piece over another, and the lowest one either resting on carrier *A* or in the chucking position *B*. With this arrangement the necessity for a spring valve is obviated, although the carrier must have suitable provision for keeping the pieces vertical, and also spring releasing members *C* which will allow the work to pass into the chuck when pushed by a device on the turret. The carrier is equipped with adjustable work supports *D*. The design of any magazine should nearly always be of open construction for several reasons: It is easy to

see when the magazine needs refilling; it is not likely to become clogged with dirt; and it is cheaper to make than a closed form. There are cases when the closed form is preferable, but seldom on this kind of work. Sheet metal can be bent without difficulty into various forms and readily riveted or brazed together, which makes it a desirable material from which to construct

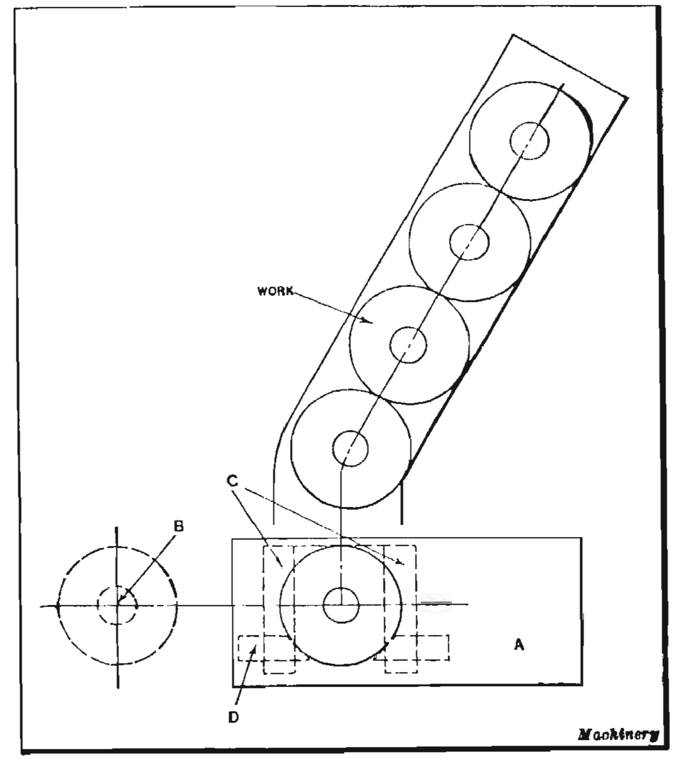


Fig. 5. Magazine Designed for Same Part as that in Fig. 4 but for Application to Rear of Machine

chutes. The inside surfaces should be kept smooth by using countersunk-head rivets, where necessary, or better still, by designing the magazine in such a way that rivets do not pass through walls where moving parts are carried.

Supports for the magazine should nearly always be made adjustable to provide for variations in castings or forgings.

More than one adjustment is often necessary for the size of the work and its relation to the carrier. When a gravity feed is used, it is not always necessary to support the magazine rigidly, but when mechanical feeding devices are applied, the supports must be rigid enough to carry out the operation without undue vibration.

Friction Between Rolling Parts. — Friction surfaces should be made as small as possible consistent with good design, always bearing in mind that line or point contacts with the work are to be preferred. Pieces that are to roll into

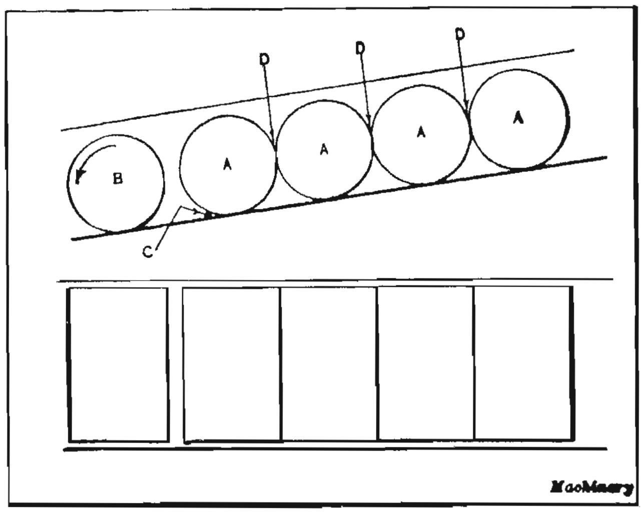


Fig. 6. Effect of Friction on the Rolling of Parts in Magazine

position should be fed in a chute having a sufficient incline so that they will roll freely. Care must always be taken with circular pieces having long surfaces that come into contact with each other, as shown in Fig. 6. Pieces *A*, considered one at a time, would roll down a slight incline unless they were badly distorted. However, when in a group, the first one starts to roll and gets away from the others, as shown at *B*, but a grain of sand or small chip in front of one of the others, as at *C*, may prevent it from starting because of the resistance at this point and the friction of the contact at points *D*. Of

course, if the angle of the incline is great enough no trouble will be found, but while a 5-degree angle is sufficient to carry a single piece it is advisable to make the incline steep enough to prevent any possibility of sticking.

The importance of a smooth-working positive valve action for releasing pieces from a magazine should be emphasized,

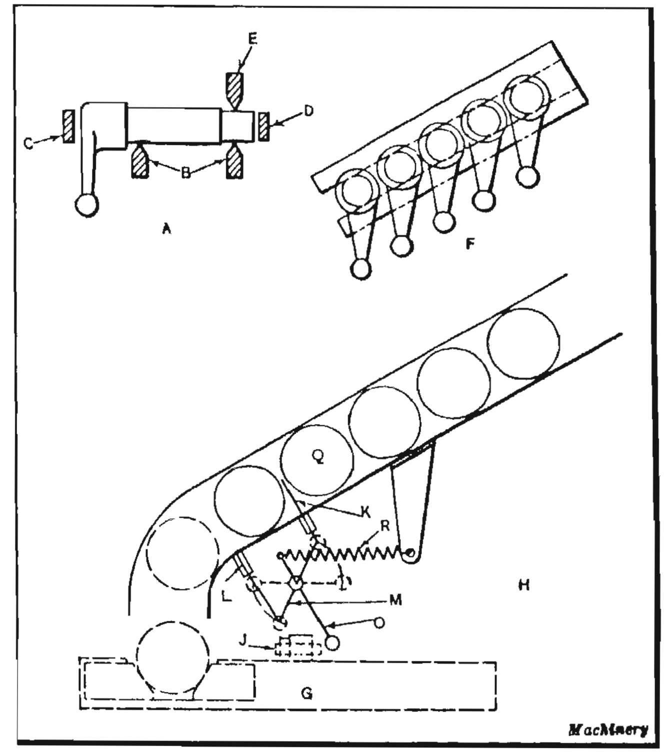


Fig. 7. Design for Handling the Topheavy Forging C, Fig. 1

for on the correct admission of the pieces to the carrier depends the success of the entire mechanism. Cases are found where springs, rubber bumpers, or sheet rubber "cowls" are necessary at the mouth of the magazine.

Feeding Forgings of Irregular Shape. — Piece *C*, Fig. 1 is to be machined on centers that have previously been drilled. Assume that a faceplate driver is used with an air-operated device for driving the work by gripping the ball-end, a floating movement being provided to allow for variations in the distance between the shaft center and the ball. There are several interesting problems connected with designing an attachment for this piece. First, the faceplate must be stopped at a specified position in relation to the work, in order that the arm may enter the holding device properly.

The magazine for this work must be different from any of the previous examples, as the pieces are so heavy at the ball end that they cannot be balanced on the cylindrical portion. Therefore a special arrangement of guide rails and guards is necessary, as shown at *A*, Fig. 7, in which the pieces rest on knife-edges *B* with a guard *C* and *D* at each end spaced far enough apart to allow clearance. Another guard *E* prevents the piece from tipping out of position. These pieces cannot be held in a vertical chute because the overhanging ends would strike each other and cause incorrect positioning. However, they can be held in guides at an angle of 45 degrees, as shown at *F*, and while they will not roll they will slide easily down this incline without wedging or cramping.

There is a necessity for a stop arrangement at the lower end of this device similar to a valve, but this can be of trigger form to allow one piece to come through and stop the others, as shown in the enlarged diagram at *H*. There are two sliding cut-off blades *K* and *L*, both of which are controlled by a movement of lever *M* which is attached to a pivoted shaft. To this shaft another lever *O* is connected in such a way that when carrier *G* comes into position for a load, it strikes the end of lever *O* and pulls it over, thus withdrawing blade *L* and letting one piece of work fall into the carrier. At the same time blade *K* is moved up to prevent the next piece from following. When the carrier returns again, spring *R* pulls lever *O* forward returning blade *L* into place and withdrawing blade *K*. There is an adjustable trip *J* on the carrier. Care must always be taken to design shut-off mechanisms with sufficient adjustments so there will be no difficulty in timing up the movement.

CHAPTER XX

DESIGN OF MAGAZINE CARRIERS AND SLIDES

In transferring parts from a magazine chute or hopper to the cutting position in an automatic machine, a carrier or some other mechanical means is required, unless the work can be fed directly by gravity. The important factors in designing a carrier are shape of work, form of magazine, method of transferring work from carrier to holding units, and method of holding the work. If an automatic machine is to be designed for continuous work on the same piece day after day, it is not necessary to provide adjustments, except for taking up lost motion and wear. However, it is well to remember that a standard product may be redesigned and that even minor changes in size or shape may make it necessary to adapt the machine to new conditions. It is nearly always possible, without much extra expense, to provide for such contingencies in designing, and it is therefore advisable to make locating points, guides, stroke of carrier, etc., adjustable within reasonable limits.

Points Involved in Designing a Carrier.— Let us consider the conditions encountered in designing a carrier for delivering bars from a magazine to a chucking device in which the work is held on centers and gripped on the outside by floating jaws. The easiest way to determine the points of importance in developing any design is to consider the things that must be done. In this case, the work must fall from the magazine into the carrier, which must move over toward the supporting centers, and at the end of the travel, the work must be pushed out of the carrier and gripped by the chuck; the carrier must then move back into position for receiving another piece from the magazine. It is evident that the work must be properly located in the carrier and perhaps held in a fixed position while the carrier moves forward to the chucking position. At this point any device used for holding the work must be positively released, leaving the work free to be pushed forward as required. Provision must also be made for returning the carrier without interfering with the piece that is left on the centers.

It will be assumed that the magazine delivers the work to the carrier in a position parallel to the axis of the centers on which it is to be placed. The problem, then, is to move the

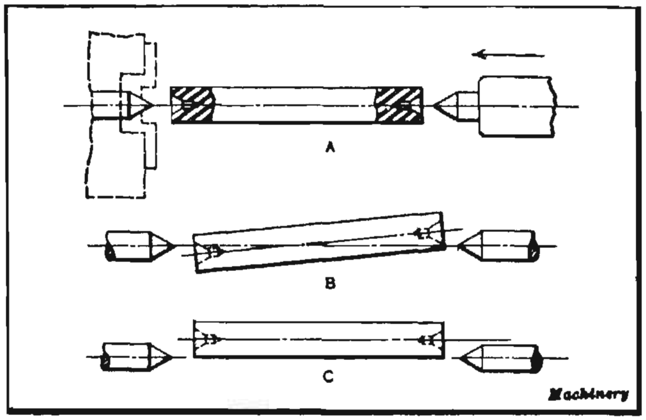


Fig. 1. Diagrams Showing Proper and Improper Methods of Delivering Bar Work to Centers

work in the carrier from the magazine toward the front of the machine, until it reaches position *A*, Fig. 1, which represents a plan view. At this point it is pushed forward out of the carrier by a movement of the tailstock center, until it is located on the two centers and gripped by the floating jaws of the chuck.

The carrier must not interfere with the longitudinal movements of the work nor cause it to assume an incorrect position in relation to the centers or the chuck. At *B* is shown, exaggerated, a condition that might obtain if the work were located improperly in the carrier. In such a position, the

centers would not enter the countersunk ends of the work and trouble will result. Of course, the centers will align work when it is not badly out of alignment, but they can only justify a certain amount of error. Then, again, the work may be parallel with the centers, but above, below, or at one side of them, as shown at *C*. Such a condition should exist only when the carrier supports are poorly designed, allowing chips or dirt to collect on the bearing seats, or when the adjustments for the carrier movement are loose or incorrectly set.

Carrier Supports. — Several examples of carrier supports for bar work are shown diagrammatically in Fig. 2, that at

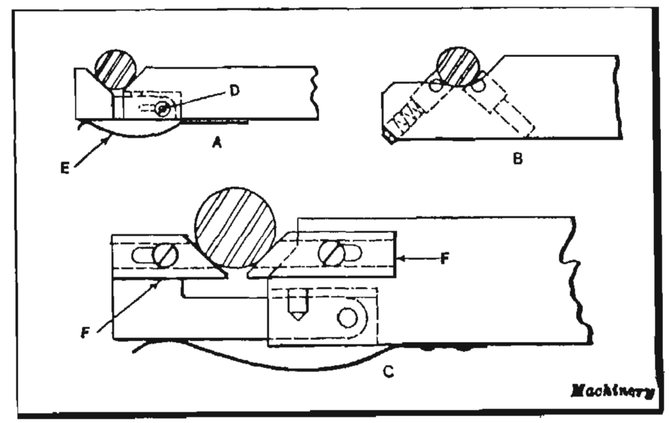


Fig. 2. Examples of Different Methods of Supporting Cylindrical Work in Carriers

A being a plain V-form with one portion hinged at *D* to permit its withdrawal from the work after the latter has been placed on the centers. This design can easily be made adjustable by slotting the hole provided in the swinging piece for the hinge-pin, thus permitting a change in the relation of the two sides of the vee to suit different diameters of stock. A spring *E* is needed to hold the hinged portion in position, and this spring must be stiff enough so that it will not be depressed by the weight of the work. It must be so located as not to interfere with other parts of the mechanism.

In example *B*, the work is supported on two pins, one of which must either be of a spring-plunger type or held in a portion of the carrier that is hinged as in the preceding example. There are some advantages obtained in using a construction like this. It is simple and not likely to get out of order nor change its position in an operation. Another advantage is that the pins do not collect chips or dirt and can easily be replaced if worn or broken. Also, it is easy to provide an adjustment for various kinds of work.

In the final example *C*, the carrier is provided with two strips *F*, both of which may be made adjustable, or one fixed and the other adjustable, according to conditions. This question depends somewhat on the arrangement used to regulate the stroke of the carrier. If the stroke is not adjustable, both jaws of the vee must be adjustable, as otherwise they could not be arranged to receive and center the work properly. In this type of carrier also, one of the jaws must be hinged to allow easy withdrawal after chucking.

Examples of Carrier-slide Design. — Fig. 3 shows the arrangement of the magazine, carrier-slide, and carrier on an automatic machine handling bar stock. The work falls from the magazine to carriers *A* which are mounted in slides *B*. These slides, in turn, rest in saddles at *C*, which are adjustable lengthwise on a dovetail way extending longitudinally at the rear of the machine. The carriers and slides are moved back and forth across the machine by the bar *D* which passes through a boss at the rear of each slide. This construction permits longitudinal adjustment of the slides without disturbing the feeding mechanism.

Another construction is illustrated in Fig. 4, in which carrier *A* is a steel bar which is adjustable on slide *B*. The latter part is an iron casting, ribbed for stiffness, and dovetailed to fit saddle *C*. A taper gib *D* provides adjustment for wear. Carrier-slides should always have a long bearing in the saddles, extending out if necessary, as at *Y*, with an extension on the saddle serving as a support. The shaft used to move the carrier-slides backward and forward passes through the

hole in boss *E*, and is a sliding fit in the hole so that the carriers may be readily adjusted for long or short work without disturbing any other mechanism.

Saddle *C* can be adjusted along dovetail *F*, after loosening gib *G*, which should again be tightened when the desired positions are reached. Adjustment of the carriers on the slides is possible by means of the slotted holes *H*. Portion *J* of the

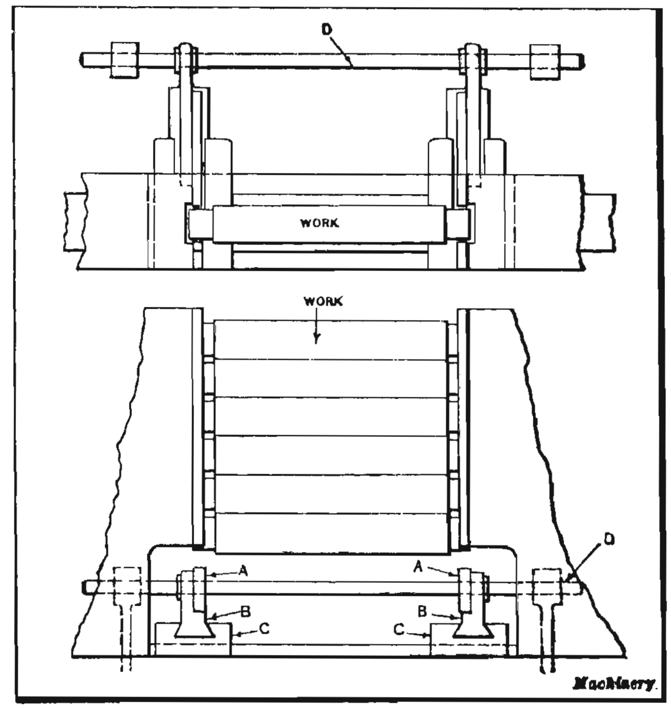


Fig. 3. Arrangement of the Magazine, Carrier-slide, and Carrier on an Automatic Machine

vee into which the work falls is hinged and supported by a flat spring as before, a slotted hole again permitting adjustments for diameter. Stud *S* limits the upward movement of jaw *J*. The dotted lines at *K* illustrate the operation of the swivel jaw when withdrawing from work placed on the machine centers.

In detail *X*, the magazine supports are dovetailed at *L* to

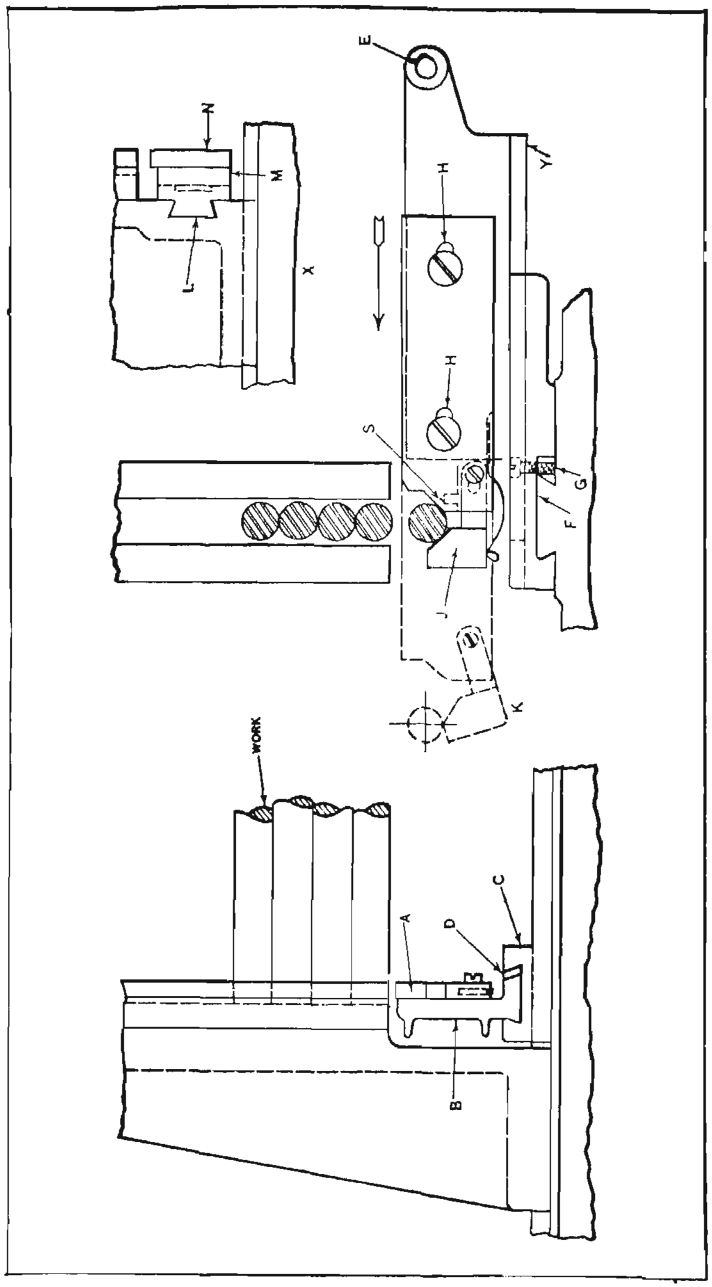


Fig. 4. Carrier-slides, Carrier, and Magazine for Bar Work

receive the carrier-slide *M* to which the carrier *N* may be fastened in any convenient way, so long as it is adjustable to some extent. There are both advantages and disadvantages in this construction, one advantage being that there are fewer pieces in the mechanism and adjustments are therefore simplified. The magazine supports must extend farther to the rear in order to give a long bearing to the slide. One disadvantage is that as the relation between the carriers and the magazine guides must always be the same, no adjustment is possible. A change in relation can only be obtained by substituting an offset carrier or adding special filler blocks between the slide and carrier to bring the latter into the required position. In designing this type of slide, the front face of slide *M* should be about in line with or a trifle back of the adjacent end of the work. In general, it will be found that the ideal construction is one in which simplicity is combined with a compact design requiring a minimum amount of space.

Control of Carrier-slides. — The movements of carrier-slides are usually controlled through levers actuated by a cam. The position of the camshaft, timing of the slide movements, and general method of operation are dependent to a large extent on the design of the machine. It is customary to use a single camshaft for controlling the movements in order to avoid the lost motion likely to occur if several shafts were used with intermediate gearing. With all movements taken from one shaft, it is much easier to regulate the timing and there is less possibility of a certain movement starting too late or too early on account of lost motion.

One method of operating a carrier-slide is illustrated in Fig. 5. The work *A* is shown just as it has dropped from the magazine into carrier *B*. Shaft *D* extends through both carrier-slides and enters slots in operating arms *C*, which control the movements. Both arms *C* may be fastened to pivot shaft *E* by means of keys, pins, or set-screws. Lever *F* has a roller *G* at one end, which engages with a cam *H*, so proportioned as to give the required movements at the proper time relative to other movements of the machine. Spring *K* keeps the roll in contact with the cam. The axis of the cam-shaft is in line with the work-holding centers.

Safety Devices for Slides. — With automatic machinery there is a possibility that the work may not lie properly in the slide, and this will cause considerable damage unless some

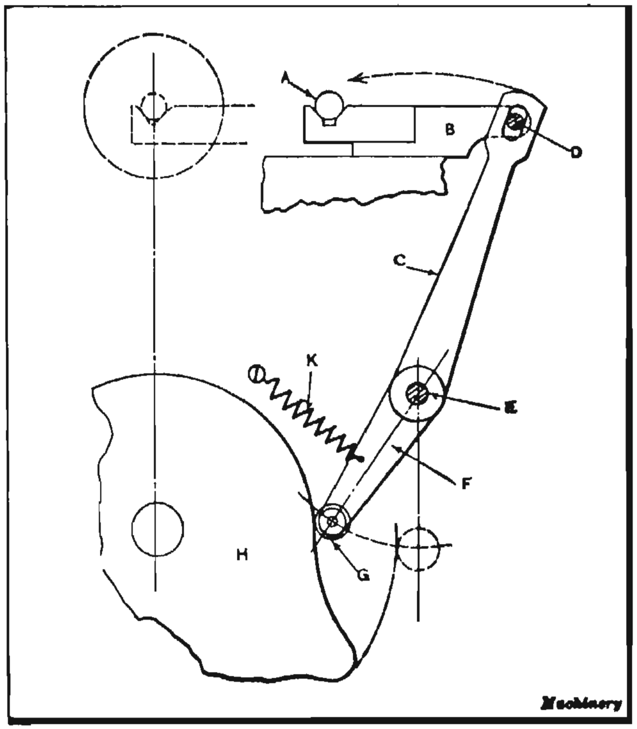


Fig. 5. Diagrammatic View Showing One Method of Imparting the Forward and Return Movements to the Carrier

provision is made to guard against such a contingency. It is well to anticipate such possibilities in designing by using the imagination and trying to conceive what the result would be if the piece did not fall properly into the carrier. One of the simplest safety devices that can be used for carrier-slides is a friction clutch, so placed that it will not drive the mechanism if an excessive strain occurs.

In Figs. 6 and 7 are shown examples of clutches suitable for such an application. In Fig. 6 the rocker-shaft is designated by *A*, and on it are mounted levers *B* which control the carrier-slide, only one lever being shown. A friction collar *C*, with a leather disk *D* secured to its face, is keyed or pinned

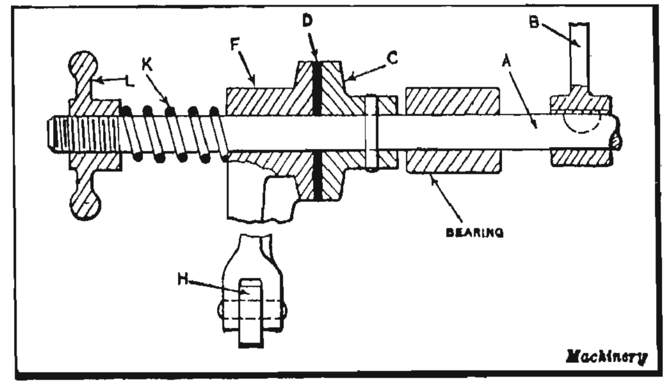


Fig. 6. Friction Clutch which may be Applied as a Safety Device to Carrier Mechanisms

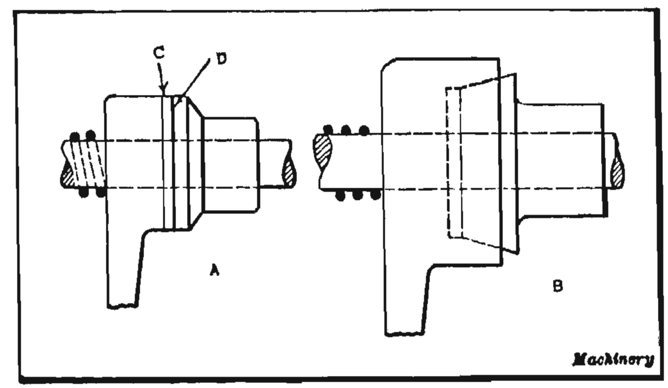


Fig. 7. Two Friction Clutch Designs which may be Used with Carriers

to the shaft. Lever *F* may be a steel or malleable-iron casting or a drop-forging, and is a running fit on the shaft. Roller *H* is held in contact with the cam in a manner similar to that shown in Fig. 5.

The hand-knob *L* is used to compress coil spring *K* and produce the desired friction in the device by forcing the face of the hub on lever *F* against the leather face of collar *C*. The friction should be sufficient to permit the transmission of power to shaft *A* under normal conditions, but the clutch faces should slip when undue resistance is met. As the power required for this particular movement is small, an excessive pressure is not needed. It is an easy matter to regulate the friction by releasing or compressing the spring.

For mechanisms in which only a slight pressure is required, a steel disk *C* working against a cast-iron surface *D*, as shown at *A* in Fig. 7, is often sufficient, the steel disk being pinned to the lever. Another clutch that could be used for the same purpose is shown at *B*. This is of the conical type, possessing considerable pulling power which, of course, is dependent on the angle, diameter, width, etc., of the faces. This type is more expensive than the leather-disk form shown in Fig. 6, and it is not as well suited to the purpose.

Carriers for Irregular Pieces. — In handling symmetrical bar work, two parallel carriers of more or less fixed design are almost invariably used, but such forms are not often adapted to the handling of irregular work. Some of the important points that apply in designing a carrier for this kind of work are as follows:

1. The shape of the piece to be held; it is essential to use a carrier that will hold the work invariably in the position required for its proper transfer to the chuck. If the work is fed to the carrier by gravity, and if it is heavier at one end than the other, provision must be made to prevent it from tipping or tilting to one side when entering the carrier.

2. The release of the work as it is transferred; if there is more resistance on one portion than on another, the piece may be forced out of position so that it will not locate properly in the chuck. The pressure of any springs used for holding the work is often a troublesome matter. Experiments are frequently required before the most suitable form can be determined.

3. The return of the carrier to the magazine for another piece; the carrier should not "drag" unduly on the piece that has just been chucked, and it should close as soon as possible after leaving the piece so as to prevent the accumulation of chips or dirt. An air blast can be directed on the locating surfaces just before a new piece of work falls from the magazine, but this is necessary only when a great many chips are formed. Chips and dirt can also be eliminated by means of suitable guards so placed that they will not interfere with the work.

For certain kinds of irregular work, the carriers previously mentioned can be used, it being possible to make adjustments to suit conditions. When work of larger diameter is to be handled, however, another form is sometimes required, in which only one slide is used. Fig. 8 shows an example of this kind, which can be used when the magazine is arranged at the back of the machine. The work is contained in a vertical magazine *A* as indicated, and fed by gravity into carrier *B*. This piece of work has a chucking stem *C*, as illustrated clearly in the enlarged view *Y*, which is grasped in the jaws of a special chuck. As the work lies in the carrier, it is in front of the chuck, and so it must be pushed to the left to bring it into a position where the chuck jaws can seize it. This is accomplished by means of a pusher-rod on the turret of the machine and by a carrier design that readily releases the work.

Attached to the front side of parts *B* and *D* are three leaf springs *E*, mounted as clearly shown in the enlarged detail. The springs are made of sheet metal, bent over on each side to form a lug *F* adjacent to the walls of the carrier and assembled with a pin to form a hinge. Another spring *G* normally holds each spring *E* upright. On the opposite side of the carrier from the chuck, are three guard plates *H* which, in combination with springs *E* and the vee, form a more or less flexible "nest" in which the work is held upright while it is carried forward into the chucking device. When the carrier reaches this position, the turret pusher-rod advances and pushes the work past springs *E* into the chuck.

The carrier must make the forward motion without interference and must withdraw after the work has been chucked. In addition, it must hold the work approximately upright while moving from the magazine to the chucking position. A carrier having a wide range of adjustment is not always required for this kind of work, although it is advisable to provide a limited amount of adjustment to take care of normal

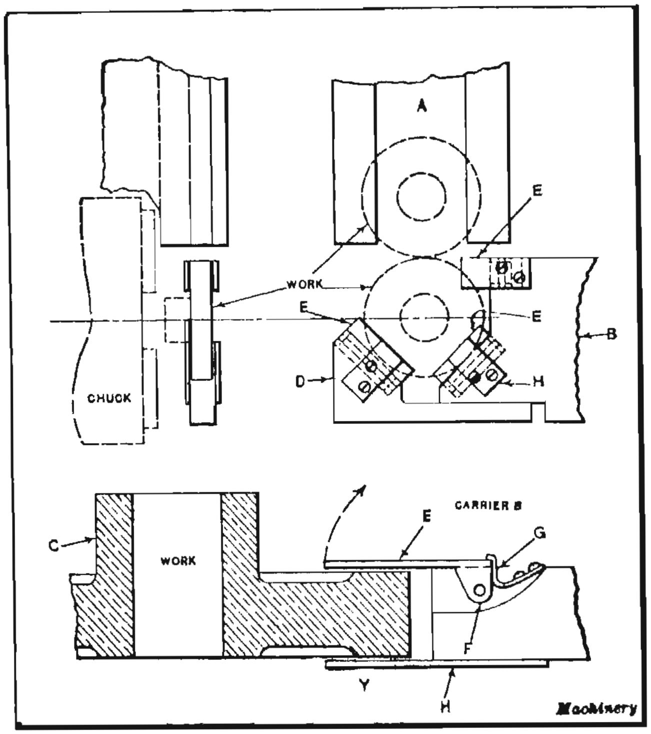


Fig. 8. Carrier in which the Work is Held by Means of Springs that Open Up in Placing the Work in the Chuck

variations in the size of work and to provide for minor changes in the design. This can be easily done by making member *D* separate from the carrier so that it can be moved backward or forward with relation to the other side of the vee.

Occasionally the same device may be used for several pieces, while in other cases it must be made to suit the particular piece. The method of holding the work in the carrier and

the type of chuck employed affects the design to some extent. A similar spring valve arrangement can be applied to various forms of carriers, whether the magazine is at the rear of the machine, near the chuck, or above the turret. The requirements are that the carrier shall locate the work properly for chucking, hold it securely while moving into position, and release it readily when placed in the chuck.

Every form of carrier has its own peculiarities, and the designer must be continually on guard to avoid overlooking some small matter that would vitally affect the functioning of the mechanism. For example, when protruding chuck jaws are used for holding the work, if the chuck is stopped in a certain position when the piece is inserted, the position of the jaws may affect both the carrier and the pusher by means of which the work is inserted in the chuck. Also, when the work is picked up by a moving chuck, it may be necessary to so design either the carrier or the pusher that a certain portion can revolve while the work is being inserted in the chuck. Conditions like this tend to complicate the design and make it more difficult to avoid "hitches" in the operation.

It is not possible to give much information of value applying to conditions of this kind; the solution of such a problem is dependent largely upon the skill of the designer. Experiments with wooden or other models are almost always necessary in order to develop a scheme that will produce satisfactory results. The preliminary idea can often be developed on paper in proper proportion in order to make sure that it can be applied to the machine, but in perfecting the design, it is highly important to test it by using a working model.

INDEX

INGENIOUS MECHANISMS

FOR DESIGNERS AND INVENTORS

VOLUME II

Mechanisms and Mechanical Movements Selected from Automatic Machines and Various Other Forms of Mechanical Apparatus as Outstanding Examples of Ingenious Design Embodying Ideas or Principles Applicable in Designing Machines or Devices Requiring Automatic Features or Mechanical Control

Edited by

FRANKLIN D. JONES

INDUSTRIAL PRESS INC.

200 MADISON AVENUE, NEW YORK 10016

Industrial Press Inc.
989 Avenue of the Americas, New York, NY 10018
Tel: 212-889-6330 Toll-Free: 1-888-528-7852 Fax: 212-545-8327
www.industrialpress.com Email: info@industrialpress.com

INGENIOUS MECHANISMS
FOR DESIGNERS AND INVENTORS—VOLUME II

38 40 37

CONTENTS

SECOND VOLUME OF INGENIOUS MECHANISMS

THIS additional volume of INGENIOUS MECHANISMS FOR DESIGNERS AND INVENTORS has been published as a companion book to Volume I in order to present illustrated descriptions of a large variety of mechanisms and mechanical movements not at hand when Volume I was produced. The continual demand for Volume I from engineers and machine designers, both here and abroad, is not only a tribute to the value of this treatise, but an indication of the need for information on outstanding mechanical movements. The publication of this second volume makes it possible to present many additional mechanisms of great practical value to designers of automatic machines or other devices, as well as to students of the general subject of mechanism.

Many of the main sections or chapters in Volume II have titles similar to those found in the first volume to assist the user of both books in locating all the information on a given subject, but the mechanisms illustrated and described in the two volumes are entirely different in design. While the second volume is a continuation of the first one, each book is an independent treatise; taken together, they constitute an unusually complete work of reference on the very important subject of mechanism. The numerous mechanical movements featured in Volume II, like those in Volume I, have been applied successfully to automatic machines and many other forms of mechanical apparatus. While it is not feasible in any work of this kind to include mechanisms that are directly applicable to every type of machine and operating condition, it is believed that the numerous designs found in Volumes I and II embody mechanical principles which may be utilized in the solution of practically any mechanism designing problem likely to be encountered.

CHAPTER I

CAM APPLICATIONS AND SPECIAL CAM DESIGNS

Cams in their various forms doubtless are more useful to designers of automatic and semi-automatic machines than any other type of mechanical device. Mechanical movements which would be difficult or impracticable to obtain by other means may, in numerous instances, be derived readily either from a single cam or from two or more cams used in combination. The cams which follow illustrate a variety of interesting applications taken from different classes of mechanical equipment. Other applications of cams and cam-operated mechanisms will be found in Chapter I, Volume I, of INGENIOUS MECHANISMS FOR DESIGNERS AND INVENTORS.

Indexing Cam for Varying Stroke of Follower.—For a given number of strokes of a slide, almost any variation in the length of each successive stroke may be produced by means of an indexing cam mechanism like that shown in Fig. 1. The construction of this cam is economical and the design is unusually simple, when the movements involved are considered. The cam member consists of a core *A* in which are secured eight cam inserts *B*. Each insert is tapered at a different angle and has a throw corresponding with the required movement of the follower roll *C*.

The core is keyed to a shaft turning in bearings on the slide *D*, which is reciprocated through a rack and gear by a member of the machine in which the cam is used. To one

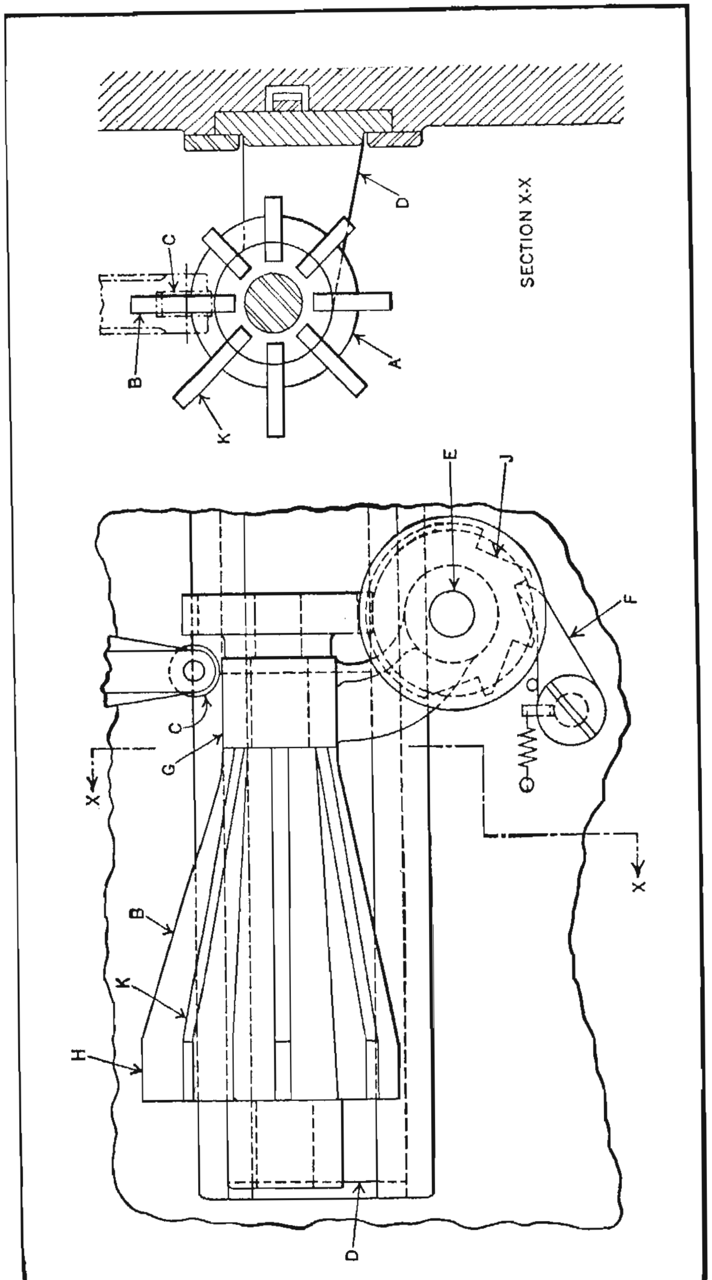


Fig. 1. Sliding Cam which is Indexed after Each Stroke to Present a Different Cam Edge to the Follower, thus Varying the Follower Stroke

end of the core shaft is keyed a helical gear, meshing with a similar gear on the vertical shaft *E*. This shaft, running in two bearings cast integral with the slide, carries a ratchet wheel *J*, which is operated by the pawl *F*, pivoted to the machine base. There are as many teeth in the ratchet wheel as there are inserts.

The various movements are obtained in the following manner: From the position indicated, the slide moves toward the right, causing the follower roll to ride along the bearing *G* and on the insert *B* to point *H*. The slide now returns, during which time the follower roll is also returned by means of a coil spring (not shown). Toward the end of the return stroke, as the roll dwells on bearing *G*, a tooth in ratchet J engages the pawl *F*. Upon the continued movement of the slide, the pawl forces the ratchet wheel around one tooth, causing the core to rotate until insert *K* is in line with the follower roll. Thus, on the return stroke, the roll rides on insert *K*, which imparts a shorter movement to the follower than the preceding one. In this way, each succeeding movement of the follower is varied until the core has been indexed one revolution. At this time, the roll will again be in line with the insert *B* and the cycle of movements will be repeated.

Although not shown, a friction brake should be applied to either the core or the ratchet-wheel shaft to prevent overrun of the cam due to the momentum imparted by the pawl. Other combinations than that shown here may be obtained by using different inserts to vary the throw or a different number of inserts to increase or decrease the number of follower movements per cycle. In the latter case, the number of teeth in the ratchet wheel must be changed to correspond with the number of inserts.

Cam-Plate with Four Adjustable Lobes.—In developing an automatic machine, it was necessary to provide means for transmitting an oscillating motion to an arm or lever from a rotating shaft. The arm was attached to a slide

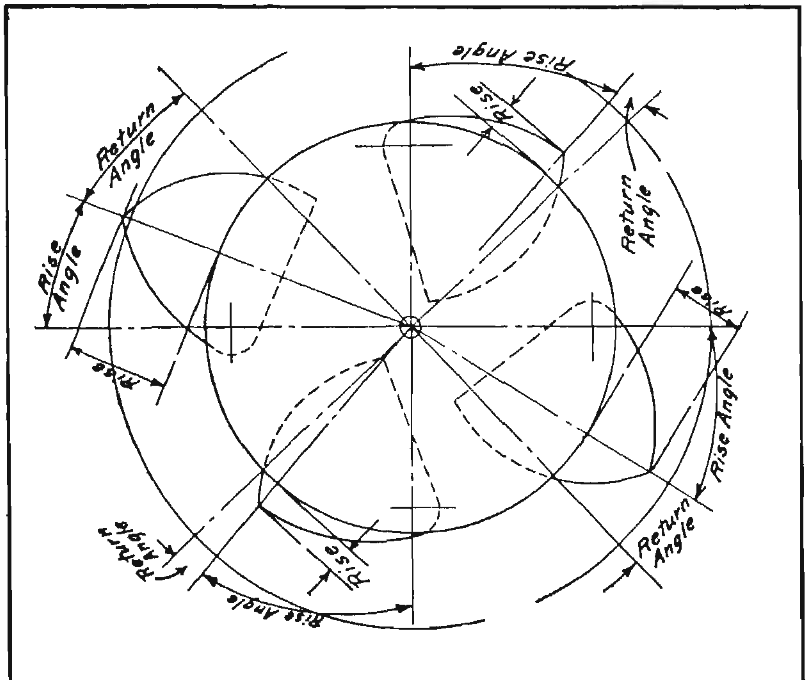


Fig. 3. Diagram Showing Cam Lobes Set in Different Positions to Indicate the Range of Adjustment

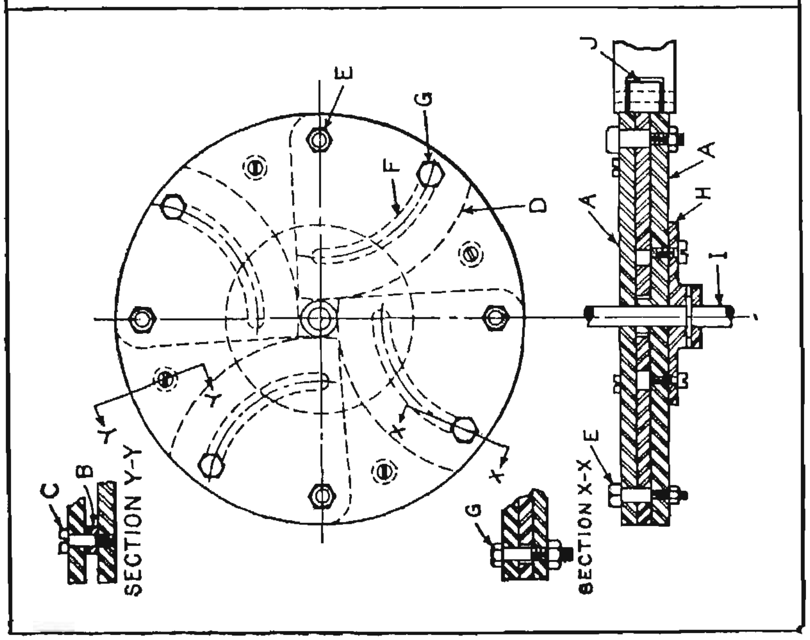


Fig. 2. Cam-plate Provided with Four Adjustable Lobes for Varying the Throw

which was returned to the zero position by means of a coil spring after having reached its maximum position.

The variation in the sizes of the product made necessary an occasional change in the length of travel or movement of the slide. To obtain the desired adjustability with the least number of actuating parts, the adjustable cam-plate shown in Fig. 2 was designed. The two side plates A are spaced a given distance apart by spacers B and are clamped together by screws C. Four cam-plates D, spaced 90 degrees apart, are held on pivot studs E. Circular slots F are milled in these plates through which clamp bolts G are inserted.

These plates are a sliding fit between the plates A, so that when bolts G are tightened, cam-plates D are held securely in place. To the outer side of one end plate A is fastened the flanged bearing H, which is held by a cross-pin to the driving shaft I. The lever arm roller J was made wide enough to allow it to ride on both the central cam lobe and on the periphery of side plates A. The various distances to which the cam lobes can be projected and the angles of rise and fall are shown in Fig. 3. This particular cam has four adjustable lobes, but a larger or smaller number of lobes can be used.

Cam for Guiding a Follower Along a Square Path.— A mechanism for guiding a pointer along a square path is illustrated in Fig. 4. The rotating shaft A, through the action of cam B and dovetailed slides C and D, causes the pointer E to follow the square contour indicated by the dot-and-dash outline. The horizontal slide D is mounted in the stationary member F, which also serves as a bearing for the shaft A; and the vertical slide C is mounted in the slide D. Elongated holes are provided in both slides so that the slides will clear the shaft in operation.

Alternate vertical and horizontal movements of slide C are obtained through the action of the positive cam B, the lay-out of which is shown in Fig. 5. Here it will be seen that sections PQ and RS are concentric with the shaft A, and sections RP and SQ are drawn by scribing arcs having centers at S and R, respectively, to form the rises of the cam.

Referring again to Fig. 4, pointer E, attached to slide C,

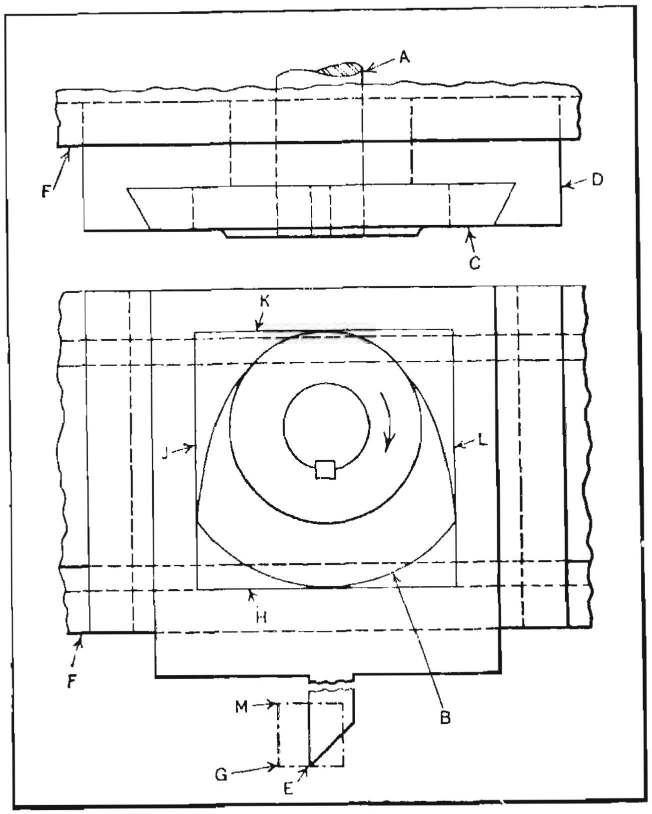


Fig. 4. Positive Cam Movement for Guiding Pointer E along Square Path Indicated by Dot-and-dash Lines

is in its lowest position and has just completed one half of the horizontal movement imparted by the cam B. As the cam continues to rotate in the direction of the arrow, slide C will move toward the left, carrying pointer E to position G. During the entire horizontal movement the concentric por-

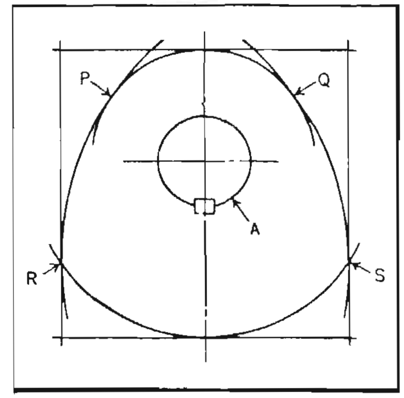


Fig. 5. Lay-out of Cam Used on Mechanism Shown in Fig. 4

tions of the cam are in contact with the surfaces K and H, preventing slide C from moving vertically. However, when the pointer E reaches position G these concentric surfaces contact with surfaces J and L and prevent horizontal movement of slide C. In the meantime, the cam rises come in contact with the surfaces K and H and raise the slide C until the pointer reaches position M. These alternate vertical and horizontal movements guide the pointer E in its required path.

Combination Cam and Parallel Motion for Guiding Spindle in Square Path.—The mechanism shown in Fig. 6 was designed to guide the center of the spindle A along a square pathway indicated by lines M and N. It is used in conjunction with a woodworking machine for gouging out an endless grooved recess of square contour into which a decorative insert is fitted. The movement involves two separate motions—a cam motion and a parallel motion. The former is the actuating member which imparts the movement to the follower, while the latter serves merely to maintain the direction of motion of the follower. By the use of interchangeable cams and follower plates, as explained later, the follower can be made to follow paths of various dimensions.

The mechanism is mounted on the machine frame B, and consists chiefly of cam C, follower D which carries the cutter-spindle A, and the parallel motion links E, F, and G. The follower is connected to the stationary bracket H through these links. With this arrangement, the angular position of the follower will remain unchanged, regardless of its location relative to the cam. The cam is of the triangular type, imparting movement in the four directions required.

From the position shown the cam is rotated in a clock-

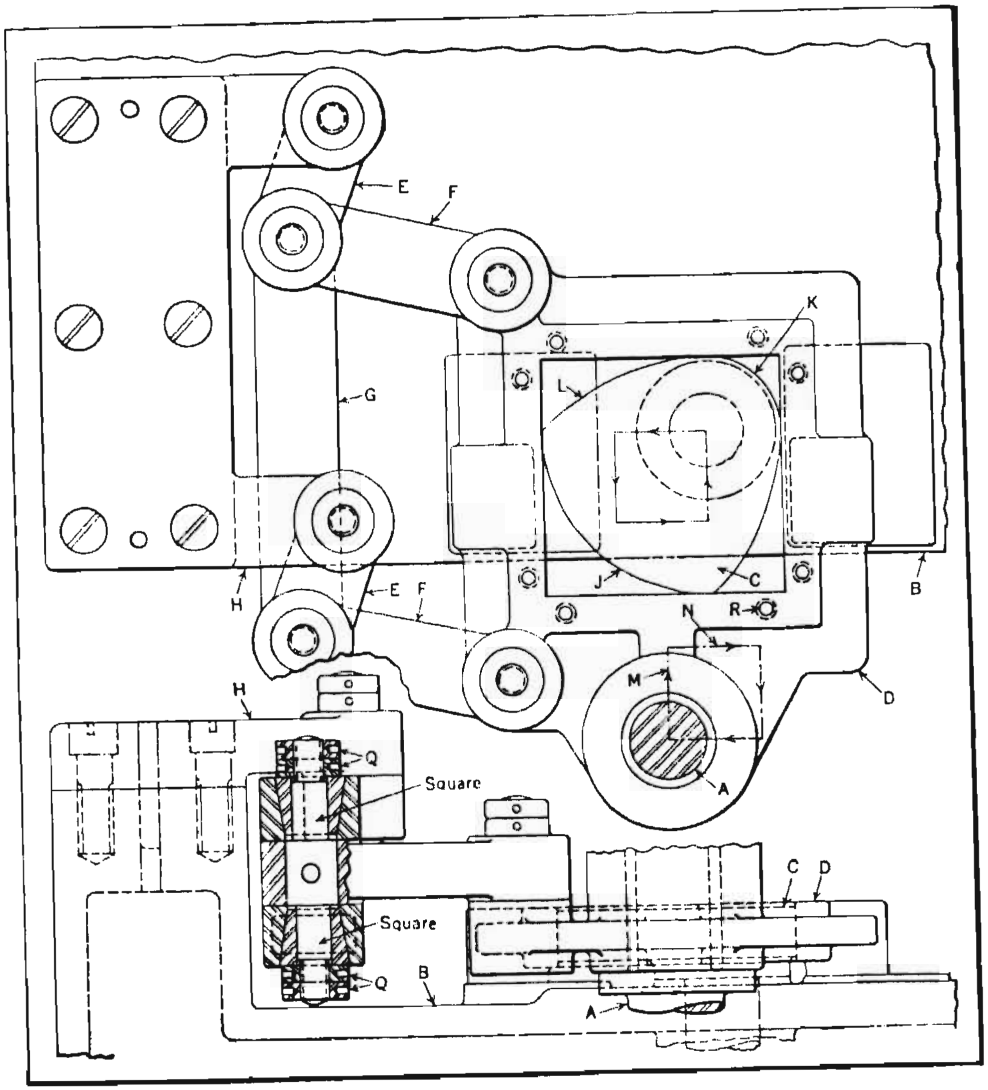


Fig. 6. Mechanism for Guiding Tool along a Path of Square Outline

wise direction. Since edges J and K are concentric with the camshaft, continued rotation of the cam will not impart a horizontal movement to the follower; but the upper edge L of the cam will raise the follower so that the center of the spindle will move along a path coinciding with the line M. When the curved edge J becomes tangent to the top cam surface of the follower, the center of the spindle will coincide with the top end of line M and the vertical movement of the follower will cease. Edge L will now force the follower toward the right so that the spindle center will follow a path coinciding with line N.

The action of the cam and follower is the same for each side of the square over which the spindle center passes. This cam is of the positive type, since the distance between the two points at which the edges of the cam intersect a line passing through the center of the camshaft is the same, regardless of the angularity of the line.

If the spindle is required to follow a square path of smaller dimensions, the cam surfaces of the follower are lined by means of four flanged plates, and a cam giving the required throw is substituted for the one shown. The camplate can be quickly attached by means of screws which pass through the plate flanges into tapped holes R.

Owing to the movement of the spindle in a plane normal to its axis, the upper end of the spindle is provided with two universal joints and a sliding sleeve. This provides a flexible connection with the upper driving shaft of the machine. Since, however, this arrangement is a common one, it is not shown. The follower is supported in its overhanging position from the bracket H by two pads integral with the follower, which rest on finished pads cast on the machine frame. In order to compensate for wear in the lever and link connections, the connection pins were designed as shown in the cross-section. With this arrangement, any wear can be taken up by tightening the check-nuts Q.

Changing Cam Speed by Transforming Uniform Circular Motion into Periodic Variable Motion.—Fig. 7 shows a mechanism by means of which a uniform circular motion is transformed into a periodic variable circular motion. The driven shaft A and the driving shaft B rotate in bearings located on the same axis. Disks C and D are securely

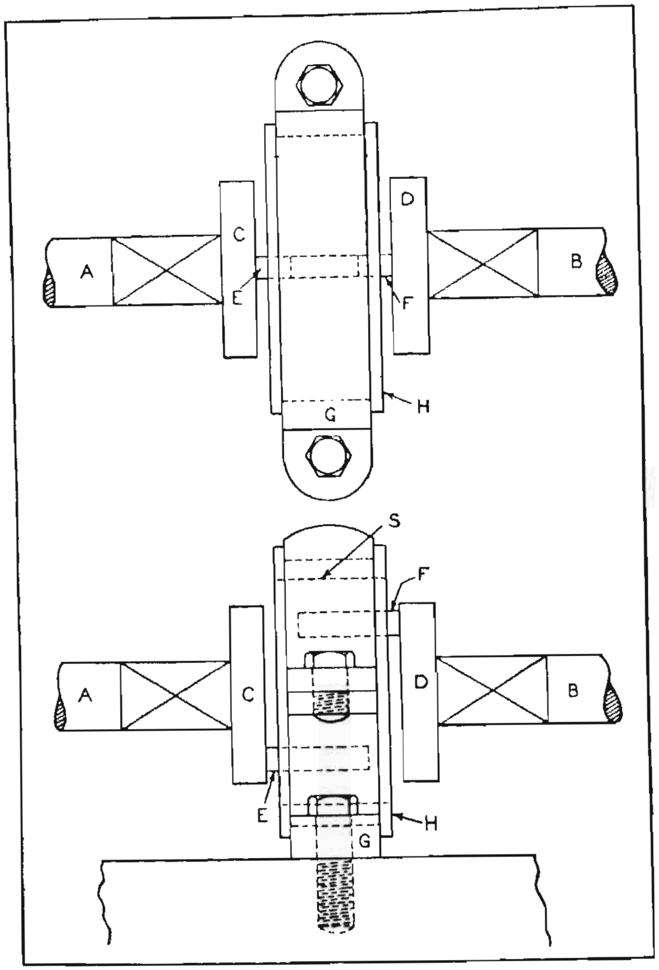


Fig. 7. Mechanism for Accelerating Speed of Driven Shaft During a Portion of Each Revolution

mounted on shafts A and B. The bearing G carries the flanged disk H, which is slotted at S to receive the pins E and F in the disks C and D. The feet of bearing G are slotted so that the position of the bearing may be changed in relation to shafts A and B. The motion of shaft B is transmitted to shaft A by the pins E and F, which act in the slot S in disk H.

In the position shown in the illustration, bearing G is so located that the axis of disk H coincides with that of shafts A and B, in which case the motion of shaft B is transmitted uniformly to shaft A. If the bearing G is moved to one side, the axis of disk H is thrown out of alignment with those of shafts A and B. As disk H then revolves in the same plane but on a different axis from shafts A and B, the pins E and F will alternately approach and recede from the center of disk H, thus imparting a periodically fast and slow motion to disk C. The amount of variation in the motion given shaft A is controlled by the amount of movement given bearing G.

An interesting application of this mechanism was made on a machine on which shaft A carried a cam. The speed with which the operating point of the cam passed under the follower was varied by shifting the bearing G.

Right- and Left-Hand Threaded Cam for Converting Rotary into Oscillating Motion.—A simple mechanism for converting rotary into oscillating motion consists of a cylinder having a right- and left-hand thread and a half-nut made as shown in Fig. 8. This mechanism was incorporated in a specially constructed printing press for the purpose of imparting a reciprocating motion to the rollers which assists in distributing the ink. A similar arrangement can be used in numerous other applications, when the speed of rotation is not too high and the load is not too great.

In the application referred to, three rollers were used for distributing the ink. The two outside rollers were operated by a double rocker arm actuated by the crank-arm *A*, which is fitted with a half-nut *B*. The right- and left-hand threaded cylinder *C* at one end of a rotating shaft serves to oscillate or move the end of arm *A* forward and back. The center ink-distributing roller is moved by a single rocker arm driven by another threaded cylinder similar to the one shown at *C*. The rocker arms are pivoted

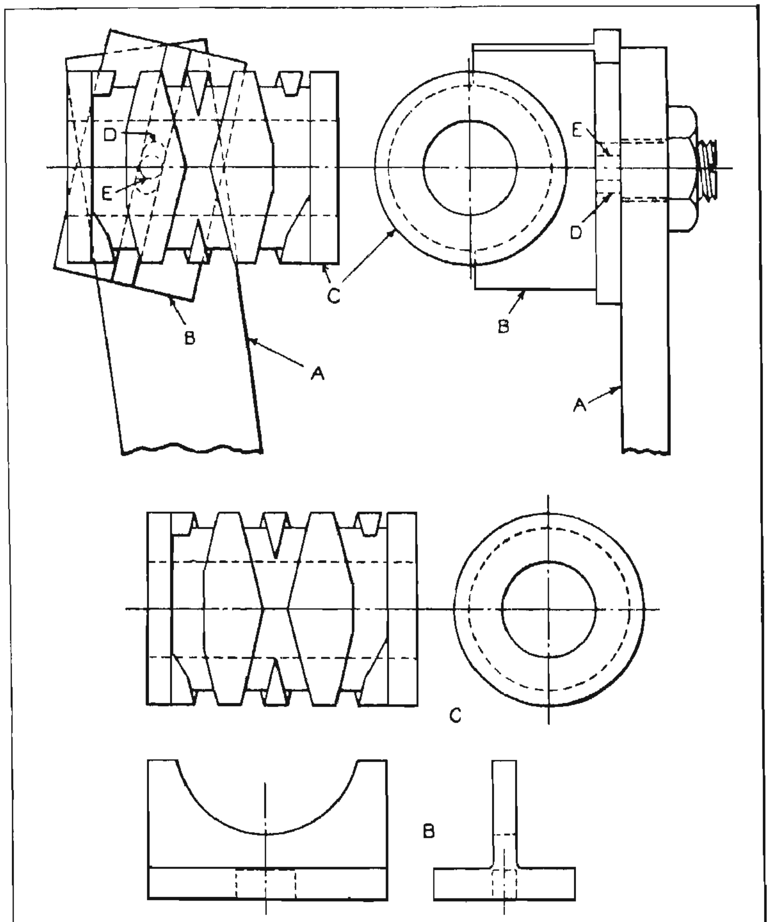


Fig. 8. Cylinder Cam C with Right- and Left-hand Threads Designed to Reverse Direction of Travel of Half-nut B at Each End of Stroke, thus Imparting an Oscillating Motion to Lever A

and carry ball-bearing pins that work against the flanges of spools on the ink-distributing rollers. Thus, as the rocker arms move back and forth, they transmit the required motions to the ink-distributing rollers.

The half-nut *B* is made from a T-shape, the thickness of the stem being equal to the width of the thread groove. The stem is formed to a concave shape to fit the contour of the root diameter of the thread, while its over-all length is made somewhat greater than the outside diameter of the thread. Its minimum length must be such as to more than span the gap made by the crossing of the right- and left-hand threads. At the center of the T-shaped bar is an elongated hole *D*, which slides over a pin *E* attached to the crank-arm. Thus, pin *E* causes the crank to rock back and forth with the longitudinal travel of the nut. An elongated hole is necessary for pin *E*, since the arm swings in an arc while the nut travels in a straight line.

When the half-nut approaches the end of its travel in one direction, its axis is on an angle with the center line of the shaft. This angle is equal to the pitch angle of the screw. In order to reverse the travel, the axis of the half-nut must pivot about pin *E* until it is in the proper angular position for the reverse traverse motion imparted by the thread of the opposite hand lead.

The last thread on the cylinder *C* is cut back a sufficient distance to allow the half-nut to pivot, and the "following" edge where the thread runs out at the end is filed back sufficiently to allow the nut to clear this surface and the end flange. The nut is also beveled at the edge where it enters the thread. The threaded cylinder *C* and the half-nut *B* are shown separately in the views to the left.

This mechanism operates smoothly, having a short dwell at each end of the stroke while the nut reverses and picks up the opposite thread. In the printing press application, the two outside rollers are operated by a double rocker arm which causes them to move an equal amount in opposite

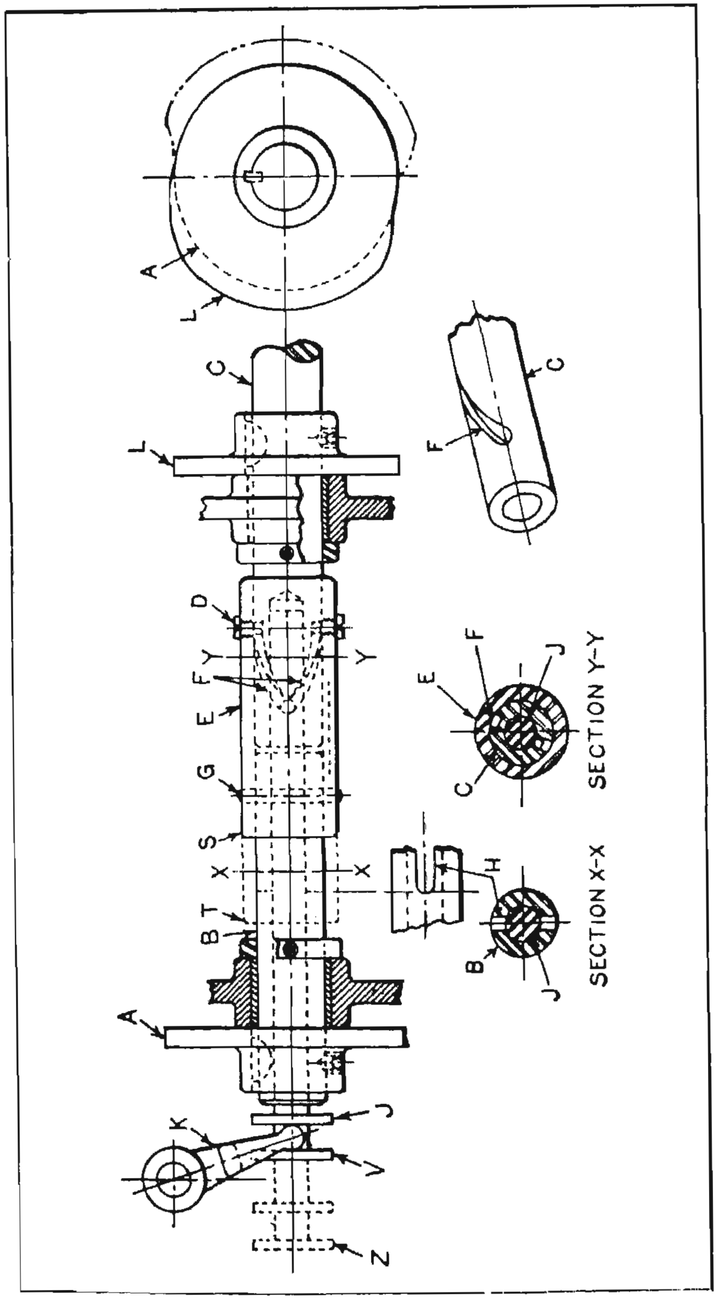


Fig. 9. Mechanism for Changing Angular Positions of Feed-cams A and L to Vary Rate of Tool-feeding Movements as Required for Different Machining Operations

directions. It is desirable to introduce as much variety as possible into the motion of the three rollers in order to smooth out the ink more effectively. For this reason, the leverage for the crank-arm of the center roller is made somewhat different from that for the outside rollers. In this case, the length of the thread on the cam for actuating the crank-arm of the center roller is longer than that of the cam for the outside rollers. With this arrangement, the center roller continuously varies its position in relation to the outer rollers.

Mechanism for Making Quick Change in Angular Positions of Feed-Cams.—The staggered production requirements and the available tool equipment for rough-turning several parts of similar design necessitated changing the angular relationship of the two principal feed-cams on one shaft for each tool set-up. The arrangement provided to permit the positions of the cams to be changed quickly to suit the machining requirements of the different parts is shown in Fig. 9.

Cam *A* on shaft *B* is driven by shaft *C* through keys *D* in sleeve *E*. Keys *D* operate in spiral slots *F*. Pin *G* fits in sleeve *E* and extends through slots *H* of shaft *B*. Pin *G* also extends through the shifter shaft *J* in shaft *B*. Axial movement of shifter shaft *J*, by means of lever *K*, from position *V* to *Z* causes cam *A* to advance clockwise in relation to cam *L*. This movement of sleeve *E* from position *S* to *T* causes keys *D* to operate in slots *F* of shaft *C*. Shaft *J* is piloted in shaft *C* to maintain the alignment of shafts *B* and *C*. The follower on cam *A* is released when changing cam positions. Shifter lever *K* is provided with conventional means (not shown) for locking in any of the required positions.

Cam and Eccentric Combinations.—The vertical ram *B*, Fig. 10, is given the required motion by combining a cam and a crank or eccentric motion. In this mechanism, gear *C* supports cam *D* and gear *E* supports the eccentric *A*. The sliding member F which carries the fulcrum of the lever G is driven by means of a connecting-rod H. The movements of both the cam and the crank serve to give the lever G a long stroke. A small slideway at the outer end of lever G provides the connection with the ram B.

The sliding member F may be omitted in some cases and the end of lever G connected directly to an eccentric on a

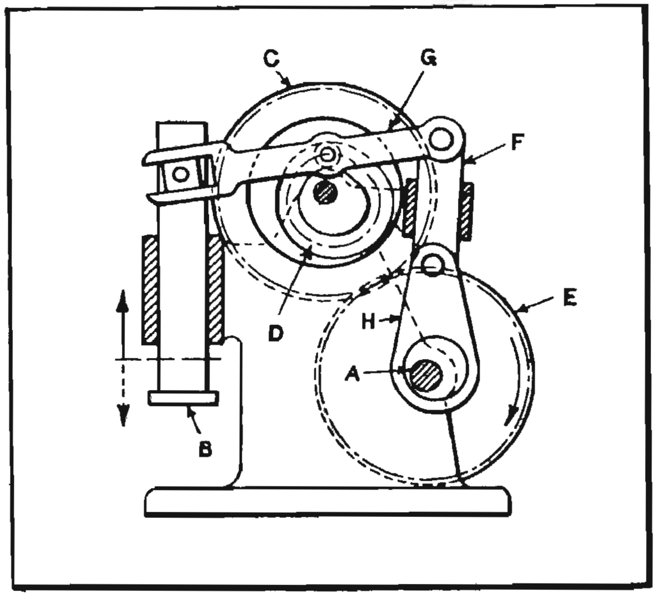


Fig. 10. Combination of Gears, Crank, Cam and Eccentric for Operating Ram

crankpin, as shown in the upper view, Fig. 11. For special cases, the cam profile may also take the form of an eccentric disk. In that event, the form of the mechanism can be changed, so that the cam is replaced by an eccentric which actuates a rod. The crank-arm can also be replaced by an eccentric A, as shown in the lower view. This gives a simple mechanism having two eccentric rods coupled together by means of a link G.

Compound Cam Drive to Reduce Cam Rise.—The possibilities of mechanisms consisting of links, gears, and cams for imparting oscillating movements to bellcranks are indicated by the illustrations Fig. 12. These mechanisms are incorporated in a shoe-sewing machine. The first design, shown in the view to the left, consists of a simple cam drive. The motion of the swinging member A is trans-

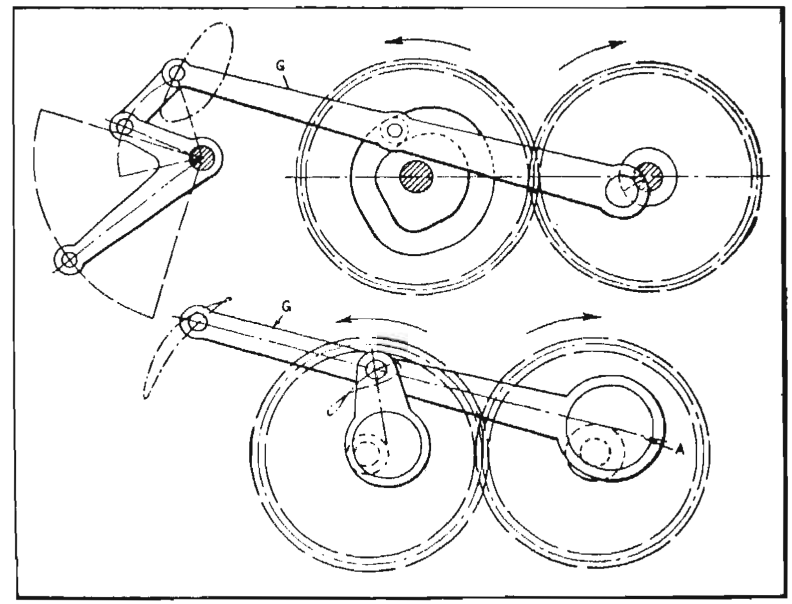


Fig. 11. Levers Operated by Cam and Eccentric Combinations

mitted by means of a rod B to the bellcrank C, which is required to oscillate or swing back and forth through an angle of approximately 80 degrees about the center of shaft D.

To obtain this movement, the driving cam E must have a rise of about 1 21/32 inches. The quick rise in the cam groove required to meet this condition, however, prevented the mechanism from being satisfactory for this particular

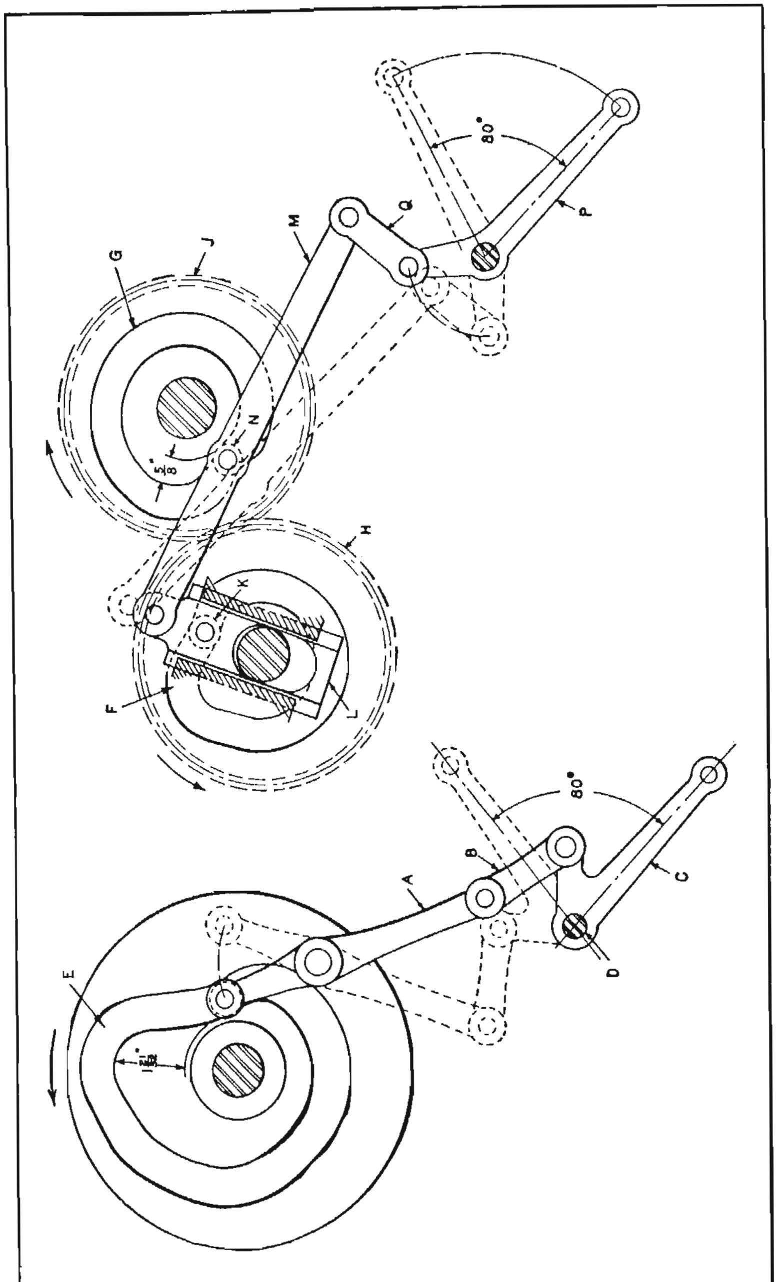


Fig. 12. Combination Link and Cam Mechanisms Used to Impart Oscillating Movement to Bellcrank

application. An enlargement of the base diameter of the cam disk or a change in the distance between the fulcrum of the lever and the cam axis to overcome this difficulty was impractical.

For this reason, the improved mechanism shown in the view to the right was designed. In the latter mechanism, an additional motion is imparted to the swinging lever by the use of another cam. The two cams *F* and *G* are driven in opposite directions by spur gears *H* and *J*, which are of equal size. These gears revolve on the same axis as the cams that they drive. Cam *F* actuates roller *K* attached to the slide *L*, which is mounted between guides on the cam-plate. The lever *M* is attached to the slide *L* and receives an additional motion from the cam disk *G* through the roller *N*.

The motion of bellcrank lever *P* is derived from lever *M* through rod *Q*. As shown in the illustration, the stroke of lever *M* is greatly increased by the two cams *F* and *G* and the slide *L*. The rise of each cam in the new mechanism is reduced to about 5/8 inch. The mechanism described works satisfactorily at speeds ranging from 400 to 500 revolutions of the driving gears per minute.

Long-Stroke Cam of Small Diameter with Rapid Return.— Cylindrical cams of the usual type for imparting a relatively long and powerful stroke to the follower must necessarily be large. Frequently this is undesirable, especially in a machine of light construction. In designing a certain machine for inserting the packing in stuffing-boxes, a rather long stroke of a slide was required to press the packing into place. The return or idle stroke was to be rapid. Because of the light construction of the machine and the elevated position of the cam, it was desirable to have the cam of small diameter, as well as light.

To meet these requirements, the cam mechanism illustrated in Fig. 13 was developed. It consists of the cylin-

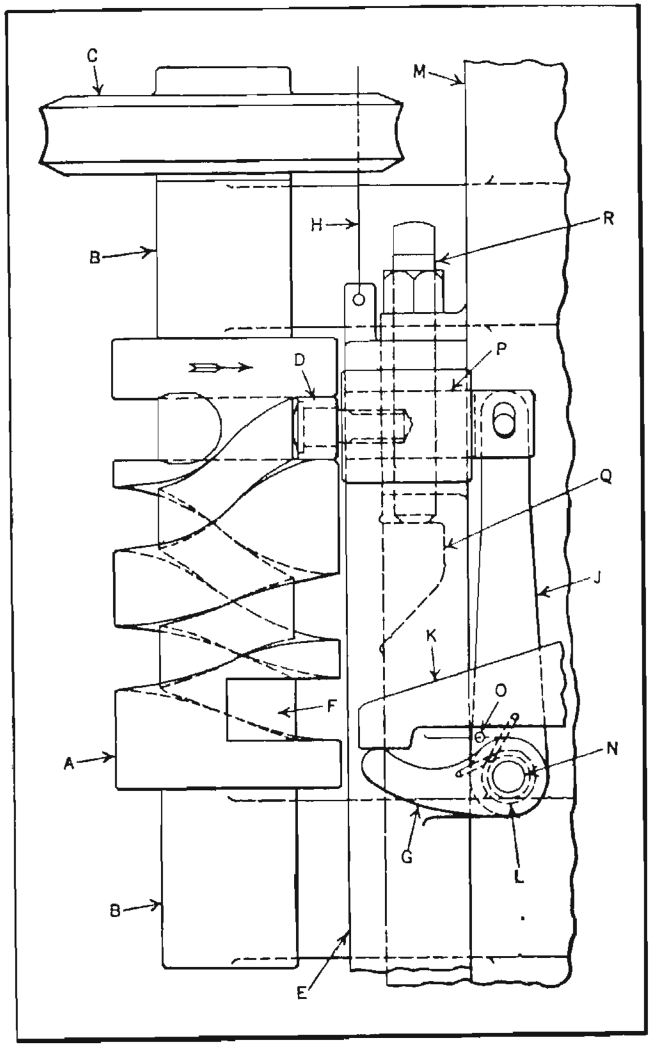


Fig. 13. Rotary Cam in which Roll is Disengaged at the Bottom of the Stroke to Allow the Slide to be Returned Rapidly by a Counterweight. Re-engagement Takes Place at the Top of the Stroke

drical cam *A*, secured to a shaft running in the stationary bearings *B*. The cam is rotated by means of a worm (not shown) and worm-gear *C*, and is engaged by the follower roll *D* on the slide *E*. This slide is mounted on the machine column *M* and carries the levers *G* and *J* on the shaft *N*.

Although both levers are free to rotate on the shaft, the lower lever *G* is normally held against the pin *O* in lever *J* by means of the coil spring *L*. The upper end of lever *J* is connected to a plunger *P*. This plunger slides in a bearing cast integral with the slide and carries the follower roll. A counterweight on cable *H* returns the slide to its upper position, the upward movement being limited by stop *Q* on the slide and adjusting screw *R* on the machine column. Stop *K* is fastened to the machine column and serves to operate levers *G* and *J* for engaging the roll with the cam groove at the top of the stroke as explained later.

In the position shown, the slide is about to begin its downward stroke. As the cam is rotated in the direction of the arrow, the slide moves downward until the roll has reached the part of the groove at *F*. Here the bottom of the groove is sloped gradually toward the outside of the cam; thus when the cam continues to rotate, the roll is forced out of the groove and the slide is returned to the upper position by the counterweight. Just before the slide reaches the upper position, the lever *G* comes into contact with the stop *K* and swings lever *J* with plunger *P* toward the left carrying the end of the roll stud against the cam. At the top of the stroke the roll is forced into the groove through the action of coil spring *L*. As the cam continues to rotate, the slide is once more carried downward.

Axial Movement from Mating Cam Sections Rotating at Different Speeds.— Certain copper tubes used in connection with steam-heating apparatus are covered with strips of copper, the strip being wound around the tube and soldered. The strip and the solder must be removed from the ends of the tubes to provide a bare length of 1 inch

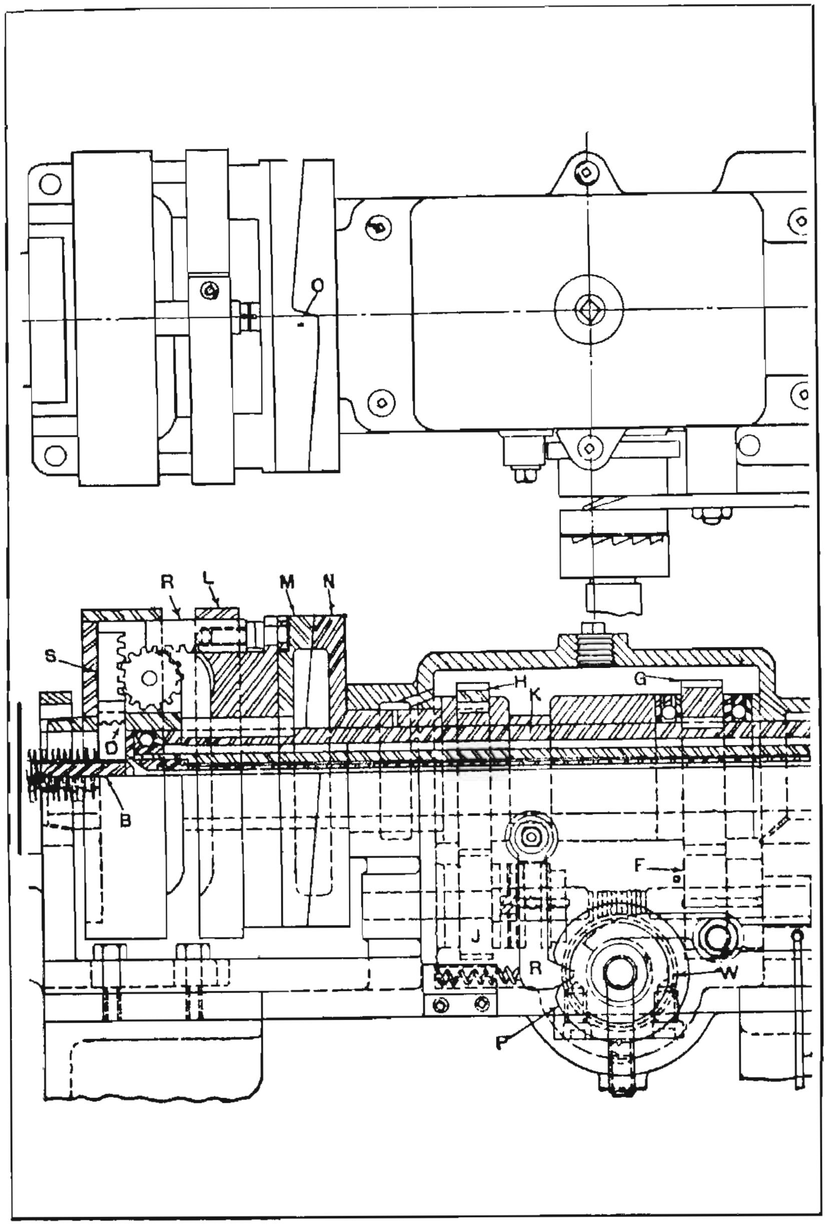


Fig. 14. Sectional and Plan View Showing Tool-feeding Mechanism Equipped with Mating Cam Sections which Rotate at Different Rates to Provide the Motion Required

for connection to a tank or header. This "stripping" of the tube ends is done by using a machine having three cutters, which are held radially and feed inward as the cutter-head rotates about the tube. The machine used for this work is shown by the sectional and plan views.

The end of a wound tube (represented by the zigzag lines) is pushed over a stationary pilot *B*, Fig. 14, which fits snugly inside the tube. An air-operated clamp is next tightened and the tube is ready for the stripping operation. The head of the machine, which contains the three cutters (one of which is shown at *D*), revolves continually at the rate of 600 revolutions per minute, and when a clutch is tripped by a foot-pedal, the three tools feed inward a distance of 3/4 inch at the rate of about 0.018 inch per revolution. The mechanism for obtaining and controlling this feeding movement is the interesting feature of the machine.

The drive from the motor to the cutter-head is through gears *F* and *G*. Gear *G* is attached to the main spindle *K*, which connects with the cutter-head. A head *L*, which is rotated by the cutter-head proper, is free to slide for a limited distance along spindle *K*. Attached to sliding head *L* there is a cam *M* which fits a mating cam *N*. Cam *N* is free to revolve on spindle *K*, and it has attached to it a gear *H* which meshes with the gear *J*.

Before the tool feeding movement begins, cam *M* drives the mating section *N* through the step or shoulder *O* (see plan view), and gears *H* and *J* revolve idly. When the tools are to be fed inward, cam *N* is rotated 40 1/2 revolutions to 40 revolutions of cam *M*. The result is that cam *N* exerts a wedging effect against *M*, causing the latter, with head *L*, to slide along the spindle. When this sliding movement occurs, racks *R*, attached to sliding head *L*, transmit this movement through pinions to racks *S*, attached to the cutter-holders. The method of obtaining this differential movement between cam sections *M* and *N* will now be described.

In order to start the tool-feeding movement, a clutch trip lever is raised by depressing a foot-pedal. This releases a clutch dog or plunger connecting plate P through a clutch with the shaft of worm-wheel W, which is rotated continually from the driving shaft. As soon as plate P begins to revolve, the dog or clutch lever R is forced out of the notch in plate P, thus connecting, through a clutch, the driving shaft with gear J; consequently, cam section N is now driven from shaft E through gears J and H, and since it rotates 40 1/2 revolutions to 40 revolutions of cam M, the wedging action and traversing movement previously referred to occurs. This difference in the speeds of cams M and N is due, of course, to the ratios of gears F and G as compared with gears J and H. Gear F has 25 teeth and G 40 teeth; hence, for each turn of gear G, F makes 40/25 turn. Therefore, 40 turns of G require $\frac{40}{25} \times 40 = 64$ turns of shaft E and gear F. For each turn of gear J, H makes 31/49 turn, as J has 31 teeth and H 49 teeth; hence, if J makes 64 turns then H will make $\frac{31}{49} \times 64 = 40\ 1/2$ turns.

While the driving shaft is turning sixty-four times in order to complete one cycle in the movement of the feeding mechanism, plate P is turned 64/65 revolution, as the worm-wheel W has sixty-five teeth. At the end of the cycle, clutch lever R is again opposite the notch in plate P and gear J is disconnected from the driving shaft, thus stopping the feeding movement automatically. Shoulder O on cam N is also around to the point where section M can slide back into engagement, which it is forced to do by means of springs concealed in the cutter-head. The difference in the speeds of the two cam sections is so slight that this re-engagement occurs easily and without objectionable shock.

Duplex Cam Action for Turning Cam Contour and Maintaining Proper Cutting Angle of Tool.—A special movement, embodied in a camshaft lathe, controls the turning tool by two sets of cams, so that the cutting angle in relation to the cam outline will always be the same. A partial cross-section of the lathe showing the carriage slide can be seen in Fig. 15. Both the cam at the top and the one

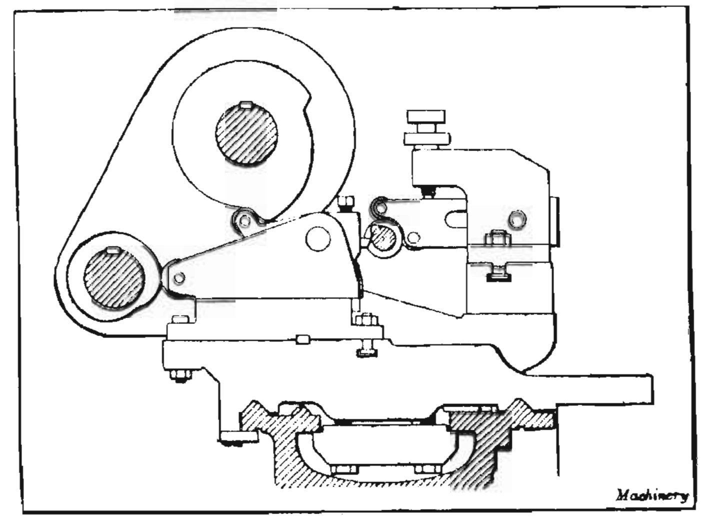


Fig. 15. Cross-section of Lathe, Showing Cams that Maintain Constant Cutting Angle of Tool

at the left revolve at the same number of revolutions per minute as the camshaft to be machined. The cam at the left is used as a master while the top cam controls the swinging motion of the tool about the horizontal axis in such a way that the cutting angle remains constant.

With the combined movements of both cams, the desired result is obtained. The master cam at the left is ground accurately in the lathe by using a grinding wheel in place of the cam roller and of the same size, while a suitable member engages a revolving cam of correct form on a shaft between the lathe centers.

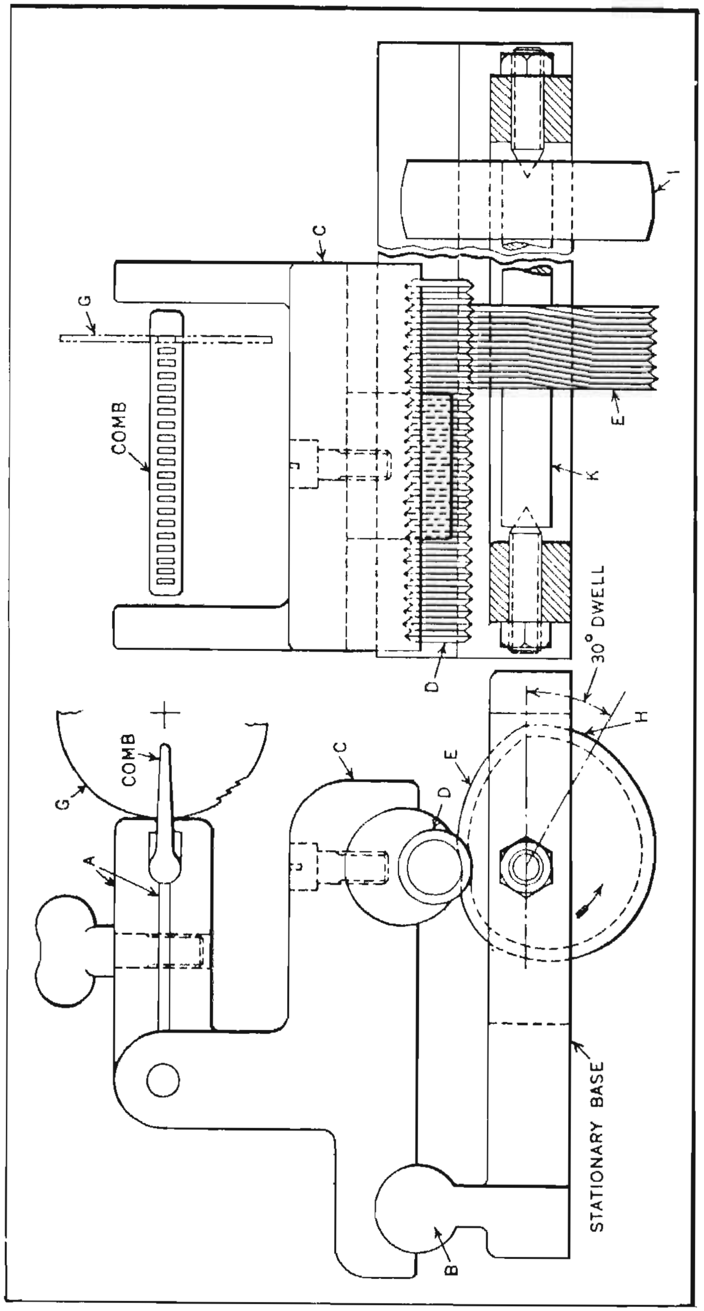


Fig. 16. Mechanism in which a Cam Imparts an Oscillating Movement Followed by a Horizontal Indexing Movement

Double-Acting Cam which Oscillates Follower and also Indexes it Horizontally.—In the mechanism shown in Fig. 16, a cam is used to impart an oscillating movement, as well as a horizontal indexing movement, to the table of a machine for sawing teeth in combs. A comb is clamped rigidly between the straps *A* on the table *C*. This table oscillates about the bearing *B* and receives its motion from the cam *E* acting against the follower *D*. The cam is driven by a belt passing over the pulley *I*. The circular saw *G* revolves in stationary bearings, the comb being fed to it by the oscillating action of the table.

The principal feature of this cam motion is the manner in which the horizontal movement is imparted to the comb for cutting the successive teeth. On the outside of the cam is cut a continuous V-groove which engages corresponding grooves in the follower *D*. The several turns of the groove on the cam follow a parallel plane perpendicular to the center line of the shaft *K* until they approach the dwelling portion *H*, where they are deflected to one side a distance equal to the pitch of the groove. This pitch is also equal to that of the slots being cut in the comb. The grooves in the follower, however, are not continuous but are a series of separate grooves.

In the position shown, the table is at its lowest point and the saw has just completed cutting a tooth in the comb. As the cam continues to revolve in the direction of the arrow an upward movement is imparted to the table. When the dwelling surface of the cam has come in contact with the follower, the comb is clear of the saw, and the horizontal or indexing movement of the table begins, continuing until the follower has passed over the angular portion of the cam groove. Further movement of the cam carries the table downward, causing the comb to be fed against the saw for cutting the next tooth. This completes the cycle of operations. The follower *D* does not revolve in actual operation, but can be adjusted to present new wearing surfaces. The design of this mechanism permits the use of interchangeable cams and followers for slotting combs having teeth of different pitches.

Straight-Line Movement Applied to a Cam Follower.— A practical application of a straight-line movement obtained by means of a link and a lever is shown in Fig. 17.

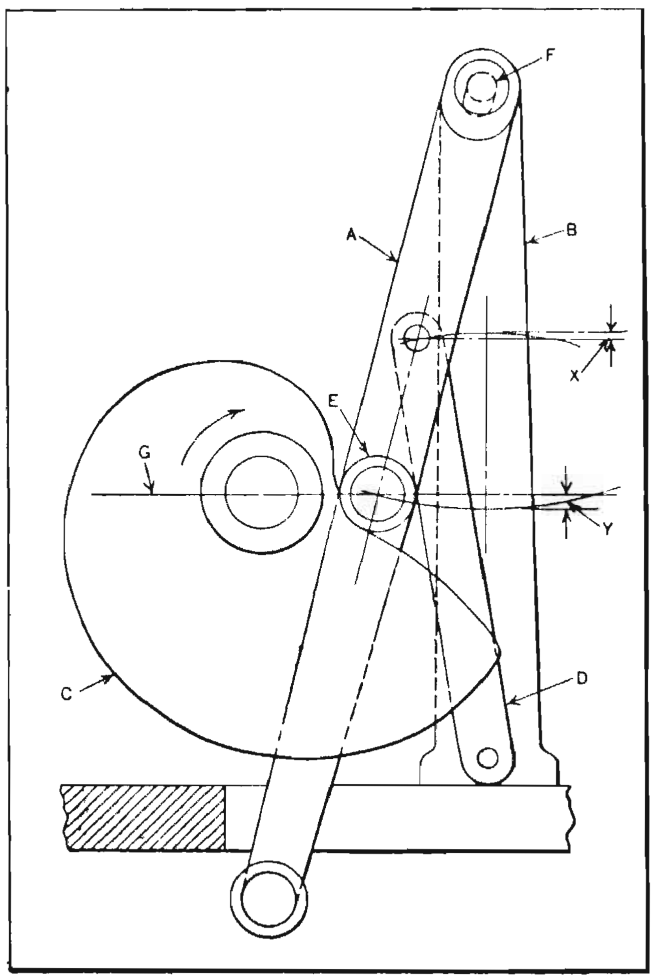


Fig. 17. Application of a Simple Straight-line Motion to the Follower Roll E

This movement is applied to the follower roll of a cam on an automatic machine intended for sawing slots in latch needles.

The roll *E* moves 3 inches forward and backward, and the return movement is effected during one-twelfth of a revolution of cam *C*. Originally, the roll lever *A* was pivoted to the stationary bracket *B*, and was not equipped with the auxiliary link *D*. Consequently, the center of the roll *E* followed a curved path, and on the return of the lever *A*, the roll had a tendency to leave the cam surface, especially when the machine was operated at high speed. This action caused the roll to strike the low point of the cam with an appreciable impact during each cycle.

To overcome this condition, the pivot hole in the upper end of the lever *A* was elongated and the link *D* added to force the roll *E* to travel in a straight instead of a curved path. The center distance between the pivot *F* and the roll *E* is equal to that of the holes in the ends of the link. The lower end of the link is pivoted to the bracket, while its upper end is pivoted to the lever. This pivot is located in such a position that the distance *X* equals one-half of the distance *Y*.

As the cam rotates from the position shown, the upper end of lever *A* is gradually lifted and lowered through the action of the link, so that the center of the roll *E* follows very closely the center line *G*. Thus, when the steep incline of the cam is reached, the roll is returned along the same straight line and remains in contact with the incline instead of leaving the cam surface, as when the roll followed a curved path.

Varying the Cam Dwell with Two Adjustable Follower Rolls.— An increase in the variety of products manufactured in one plant made it necessary to alter some of the wire-forming machines so that the dwelling periods of their slides could be varied. To do this, instead of employing one follower roll for each slide, two adjustable rolls were used, as shown in Fig. 18. The two dwelling periods of the slides can thus be varied to suit requirements. The cam, indicated at *A*, is secured to the driving shaft and engages both rolls *B* and *C*. The rolls are mounted on flanged bushings and secured to slide *D* by studs. They can be adjusted to any position along the curved T-slot *E*.

The amount of dwell and the timing of the rise and fall of the slide depend upon the distance between the two rolls

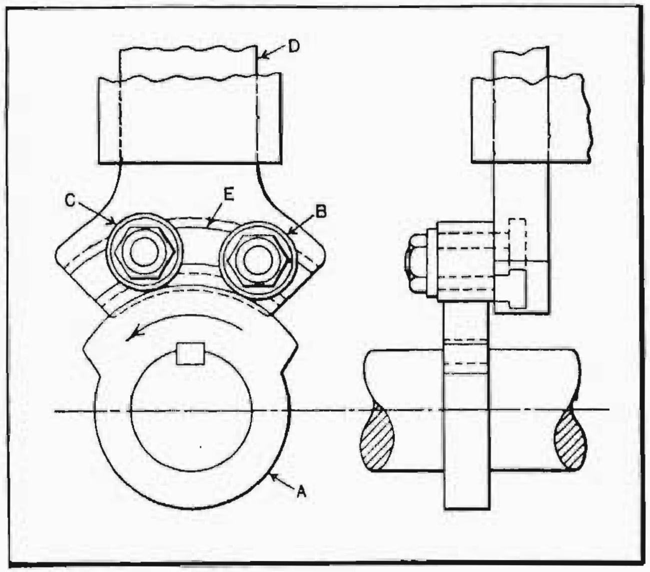


Fig. 18. The Dwelling Period and Timing of This Cam can be Varied by Simply Changing the Positions of the Two Follower Rolls

and their location along the T-slot. For instance, if the slide were required to dwell longer in its upper position, the distance between the rolls would be increased. On the other hand, if the dwelling time in the upper position was to be decreased, the rolls would be brought closer together. The time at which the rise and fall of the slide occurs may be varied by adjusting the rolls along the T-slot without changing their center distance.

Double-Faced Cam for Rapid Rise without Excessive Side Thrust.— The cam for operating the slide of a certain machine required a rapid rise without excessive side thrust. To meet this requirement, a double-faced cam was used (see Fig. 19). Each face or edge of this cam *C* has a rise equal to one-half the total rise required. The cam has a sliding fit on shaft *A*, and it is revolved by the driving gear *G* which meshes with gear teeth extending around the center of the cam.

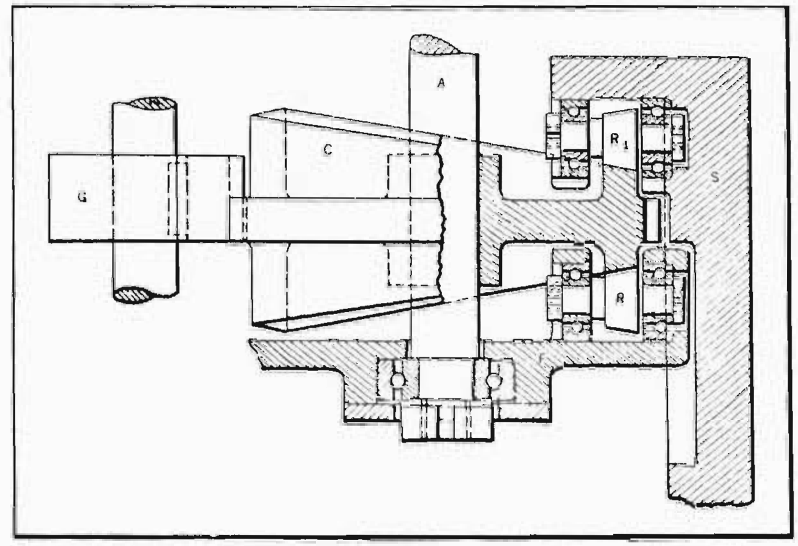


Fig. 19. Double-faced Cam which Moves Driven Slide a Distance Equal to Sum of Leads of Both Faces

As the cam rotates it rises, owing to the fact that it rests on a roller *R*, which is supported by the machine frame *F* and remains stationary except for rotation about its own axis. Bearing against the top face of the cam is another roller R_1, which is supported by slide *S*; this slide is the one that is operated by the cam. It will be evident that when the cam makes one revolution, slide *S* moves a distance equal to the sum of the leads of both cam faces, but roller R_1 and the slide take the thrust of only one cam face.

Reciprocating Motion to Square Bar from Cam Made of Helical Gear Segments.—A novel and what proved to be a very practical application of helical gear segments and pinions is shown by Fig. 20. Shaft *A* has an intermittent rocking movement, which is alternately clockwise and counter-clockwise. The range of these movements is through an angle of about 5 degrees. This rocking lever is required to impart an endwise movement to the square bar or shaft *B*. For this purpose, a segment of a single helical

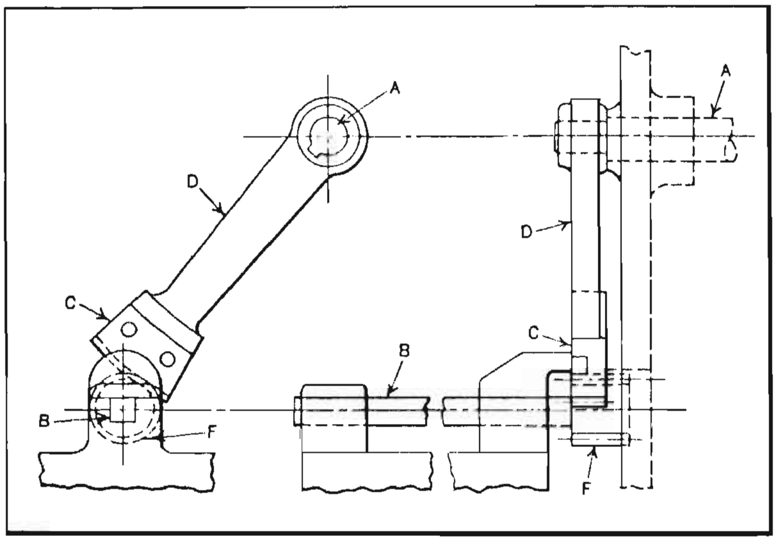


Fig. 20. Helical Gear Segment and Pinion Used as Cams to Produce Longitudinal Reciprocation from Rocking Movement

gear *C* is attached to lever *D*, and a helical pinion *F* of equal angle, but of opposite hand, is fitted to the shaft *B*. Shaft *B*, being square, cannot rotate, and is therefore forced to move endwise.

The helical segments and the helical pinions used in this construction were much less expensive than cams. A complete ring gear furnishes enough segments for several machines. By making the number of teeth in the pinion a

multiple of 4 and cutting four keyways in the shaft hole, it is possible to bring new teeth of the pinion into the working position when wear takes place by changing the position of the gear on the shaft. When the square shaft *B* can be made to serve equally well in any position, only one keyway is necessary, as the shaft and gear can be keyed together as a solid unit and relocated in one of four positions to bring unworn teeth into contact with the segment *C*. The segment *C* is supported on each side, a roller

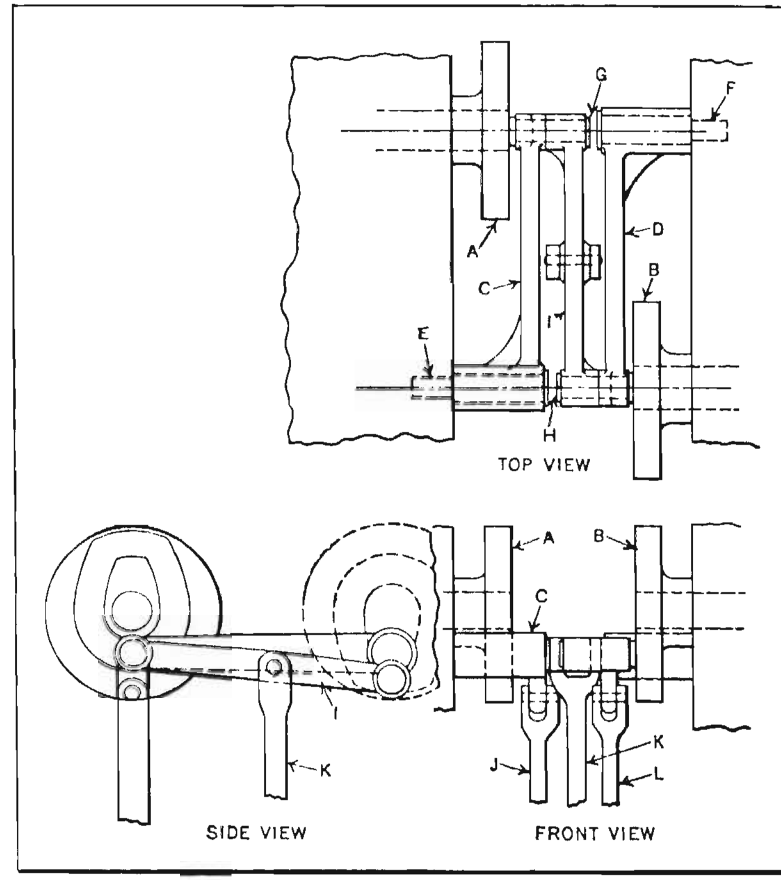


Fig. 21. Mechanism for Transmitting Motion to Three Levers of a Wire-bending Machine

support being used when necessary to reduce the friction load. Gears with teeth having a helix angle of 45 degrees or more give satisfactory performance in this kind of service.

Double Cam Drive for Three Reciprocating Rods.— The lever-motion mechanism shown in Fig. 21 is used on a wire-forming machine to obtain the motions described in the following: Two rods *J* and *L* are given a reciprocating motion, the timing relationship of which must be adjustable. Each of these rods must pass through a complete cycle of motions for each revolution of the drive shaft, although they never operate simultaneously. A third rod *K* is given a similar reciprocating motion of lesser magnitude. The latter rod, however, must pass through two complete cycles for each revolution of the drive shaft, each cycle being performed simultaneously with the cycle of the other two rods. Any change in the timing relationship between the first two rods must be automatically transmitted to the third rod.

The cams *A* and *B* operate at the same speed, and impart the required oscillating movements to the levers *C* and *D*. Cams *A* and *B*, although similar in outline, are set with their lobes approximately 180 degrees apart, and each cam can be adjusted slightly in its timing relationship with the other cam. Lever *C* fulcrums on stud *E*, while lever *D* fulcrums on stud *F*. Studs *E* and *F* are so located that levers *C* and *D* are in a horizontal position when their oscillating ends are held at their lowest points by the cams.

Lever *I* is supported on studs *G* and *H*, carried on the oscillating ends of levers *C* and *D*, respectively. Rods *J*, *K*, and *L* are attached to levers *C*, *I*, and *D*, respectively, and serve to transmit the motion to the required points. As lever *C* is oscillated by cam *A*, lever *I* is given a similar motion, being pivoted on stud *H*, which is held in a fixed position by cam *B*.

After lever *C* has passed through its cycle and come to rest, lever *I* is given a similar movement at the opposite

end by cam *B* through lever *D*. As the movement of lever *I* is produced entirely by levers *C* and *D*, it must always operate in exact synchronism with these levers, regardless of the adjustment of cams *A* and *B*. The front view shows the levers at rest. The side view shows rod *K* moved to its lowest point by the action of lever *I*.

Obtaining Instantaneous Movement of Cam-Operated Lever.— One of the best known means of imparting a very quick movement in one direction to a reciprocating part of an automatic machine is by a cam and spring mechanism, such as shown in Fig. 22. The member to be actuated (not shown) is attached to the upper end of link *A*. The other end of this link is connected to the rocker lever *B*, pivoted on stud *C*. Lever *B* acts in conjunction with cam *J* through roller *E* and spring *F*.

The left-hand diagram shows the mechanism just at the end of a dwell period of the lever *B*. Further rotation of the cam in the direction indicated by the arrow will result in roller *E* dropping into the recess of the cam and thus producing a quick downward movement of lever *B* and link *A*. It is clear that no matter how heavily spring *F* is loaded, there is a relatively slow accelerating movement of lever *B* while point *G* of the cam moves from the position shown to point *K* on roller *E*, or along the arc *GK*. Only when the cam has made an angular movement equivalent to angle *GDH* does lever *B* completely lose the restraint imposed upon it by the cam and roller and allow spring *F* to pull lever *B* downward with a quick motion. The point *H* is found at the intersection of the cam outline with an arc *KH* swung about stud *C* as a center and tangent to roller *E*.

The angle *GDH*, through which the cam rotates during the delayed action, depends primarily on the length of the radius of the roller *E* and to a much smaller extent upon the lengths *CK* and *GD*. This angle represents, in terms of angular velocity of the cam, the delay in the time of the

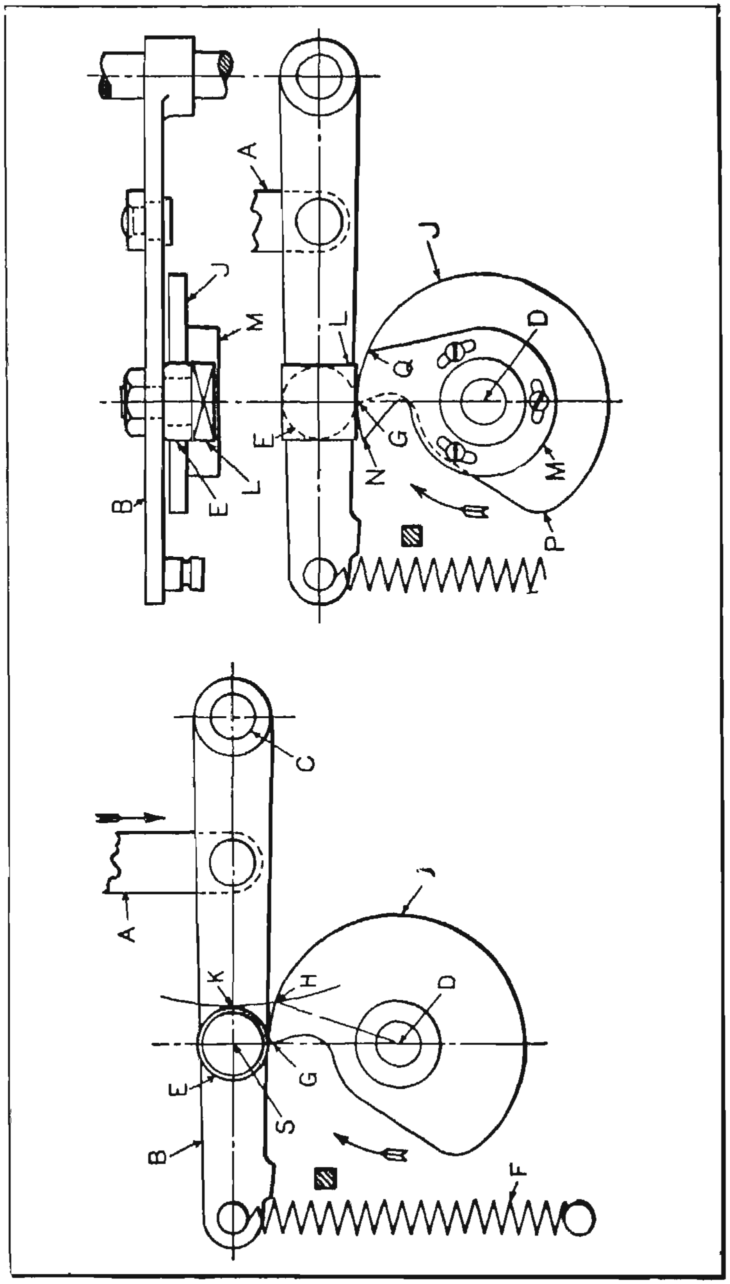


Fig. 22. Cam and Spring Mechanism for Obtaining Quick Downward Movement of Link A and Slow Return

snappy spring action, the delay being greater the larger the roller size, the shorter the roller arm *CS*, and the slower the rotation of the cam.

In most cases, this delay is not objectionable. It may even be welcome in some cases, as it results in much less shock to the mechanism. However, there are occasions when this delay must be eliminated, as for instance, when a hot fluid which sets very quickly must be pumped into a mold. Under such conditions, a very sudden action on the fluid-forcing pistons is desired. The right-hand diagram shows how this action can be effected by the addition of a few parts.

At the side of cam *J* is mounted an auxiliary shoe *M*, which is rotated about shaft *D*. This shoe engages the square block *L*, which is held rigidly to the lever *B*. Lobe *NQ* of the shoe *M* remains in contact with block *L* for some time after cam *J* has lost contact with the roller *E*. During the time cam *J* and shoe *M* are rotating from the position shown to the point of release, lever *B* remains nearly stationary, as the lobe *NQ* of shoe *M* slides underneath the flat face of block *L*. Further movement of the cam and shoe in the same direction results in an instantaneous drop of lever *B*.

Following the sudden drop of lever *B*, shoe *M* and block *L* are inoperative. After the desired dwell, the follower is restored to its initial position by the lobe *P* of the cam, which acts upon the roller *E* alone. Just before the end of the cycle of shaft *D*, both the roller and the block engage the cam and the shoe simultaneously. A little care in the design of the details insures smooth operation.

The angular margin between points *N* and *G* can be materially reduced without danger of the roller interfering with the cam during the sudden drop. If desired, this angular margin can be increased, provided lobe *NQ* of the shoe is made of sufficient size. This is a very desirable feature, as the exact moment of the drop can be adjusted within fairly wide limits, independent of the exact moment of the withdrawal movement.

Cam Mechanism that Returns Lever to its Starting Position when Machine is Stopped.—In a certain type of machine, an oscillating lever is required to return to its starting position, or very near it, regardless of the part of the cycle in which the machine is stopped. This lever, which is indicated at *A* in Fig. 23, controls the movement of an independent feeding device on the machine.

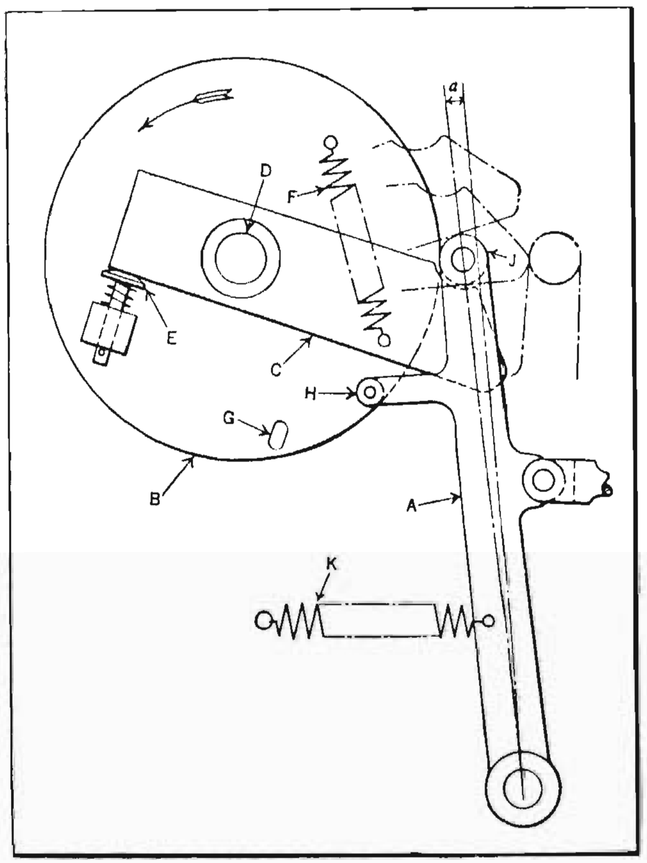


Fig. 23. Cam-actuated Lever that Always Returns to its Starting Position, Regardless of the Part of the Cycle in which the Machine is Stopped

The mechanism consists chiefly of a disk *B* which rotates continuously in the direction of the arrow and a cam *C* which is pivoted on the disk shaft *D* and held normally against the spring bumper *E* by spring *F*.

As the disk rotates from the position shown, cam *C* and lever *A* become locked and remain stationary until the lug *G*, secured to the disk, comes in contact with the roll *H* on the lever. Further movement of the disk then forces the lever toward the right, so that the upper roll *J* on the lever will move out of the hooked part of the cam. The remaining part of the stroke of lever *A* is imparted by the cam through the action of spring *F*. As the end of the cam passes roll *J*, the lever is immediately returned to its starting position by the spring *K*.

With this arrangement, it is obvious that the movement of the lever is obtained through a trigger action between cam *C* and roll *J* and regardless of the position in which disk *B* may stop, the lever will always return to its starting position when roll *J* is released.

If, however, the disk is stopped during the angular movement *a* of lever *A*, that is, before the roll *J* is released, the lever also will stop and will not return to its starting position. In the present application, however, the lever does not begin to function until it has moved through this angle, and hence is sufficiently near its starting position to fulfill the conditions required.

Single Cam Action Performs Four Different Functions.—An excellent example of a multiple cam action in which four movements are obtained essentially by one simple edge-cam is shown in Fig. 24. It is applied to a device used for capping bottles, and although two cams are used here, they are identical and impart the same movements simultaneously. The cam arrangement is such that by swinging the forked lever *G* toward the right, a split collar or "table" grips the neck of the bottle, the table being automatically locked in this position while continued movement

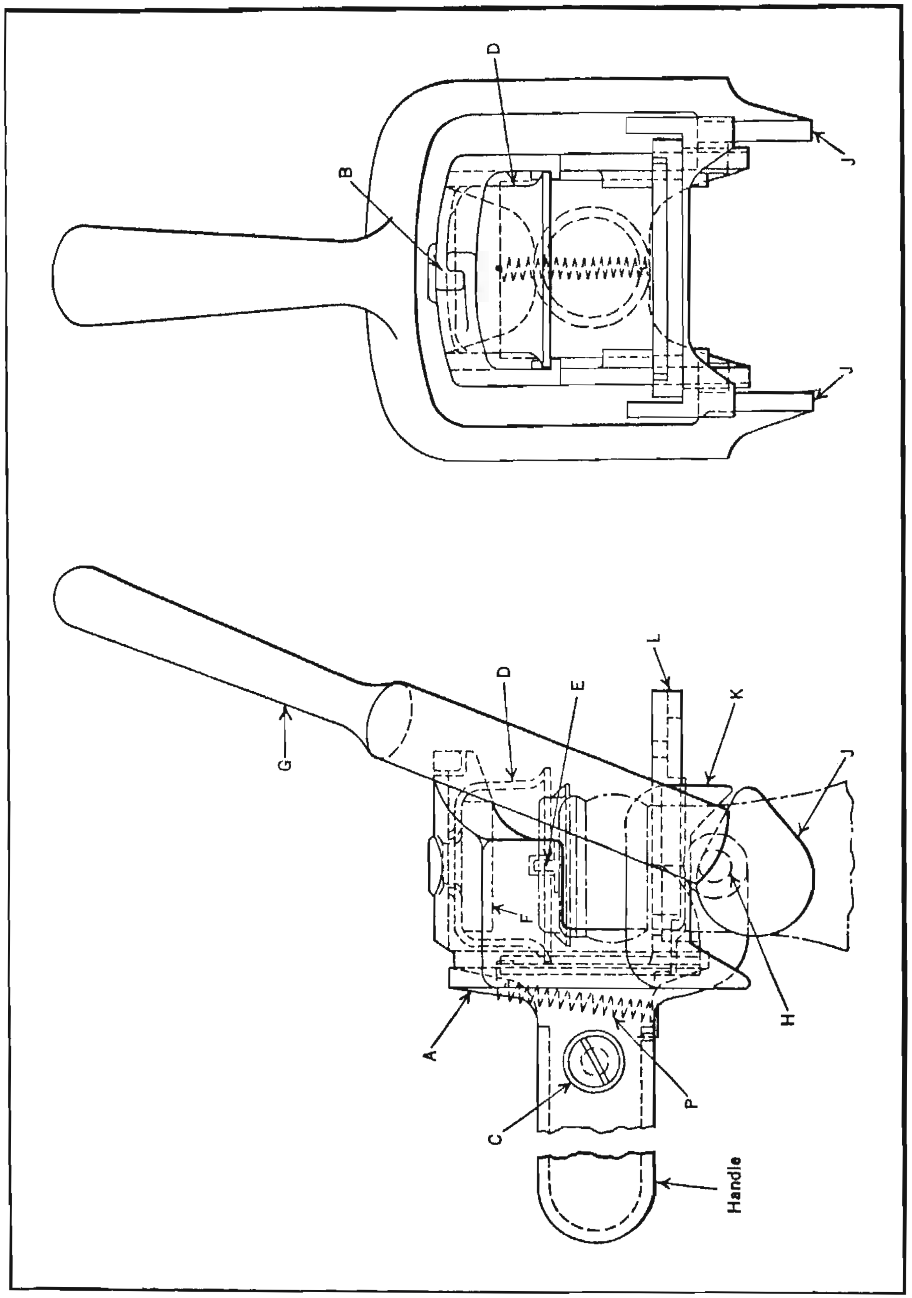


Fig. 24. Arrangement in which Only One Cam-lever is Required for the Operation of a Bottle-capping Device

of the lever causes the cap to be forced in place. To remove the device from the bottle after the capping operation, the lever is merely returned to its original position.

One of the outstanding features in the design of this device is that only one screw is required in its assembly. No machining is done on any of the parts, as sufficient clearance has been allowed to permit the use of unfinished castings. This arrangement resulted in an inexpensive product which in no way affects its utility. The body *A* is cast in two parts, which are held together by the interlocking hooks at *B* and the screw *C*. The cup-shaped capping hood *D* is held in place between the two halves of the body and prevented from rotating by the two lugs *E*. Inside the capping hood is a rubber pad *F* which is forced into place and held by a stem projecting through a hole in both the hood and the body.

The forked capping lever *G* has two pins *H* cast integral with it. These pins serve as a pivot for the lever and engage holes in the lower part of the body. On each side of the forked lever is a cam *J*. The most important part of the device is the split collar or table which consists of two parts—the lifting cam-plate *K* and the guide plate *L*, the latter having a sliding fit in the body. Both parts of this table are interlocked, as shown in Fig. 25.

The cams *J* on the capping lever impart four different movements. When lever *G* is in its farthest position toward the left, the table halves are separated in order to permit the open end of the bottle to pass through. Separation of the table halves, as indicated in both sectional views, is accomplished as the point of the cam engages the projection *M* on the cam-plate. Referring to the extreme left-hand view, it will be noted that the table halves are together, in position to grip the neck of the bottle. This is done with the portion *R* of the cam as it engages the projection *N* on the cam-plate when lever *G* is swung toward the right. Continuation of this lever movement (see ex-

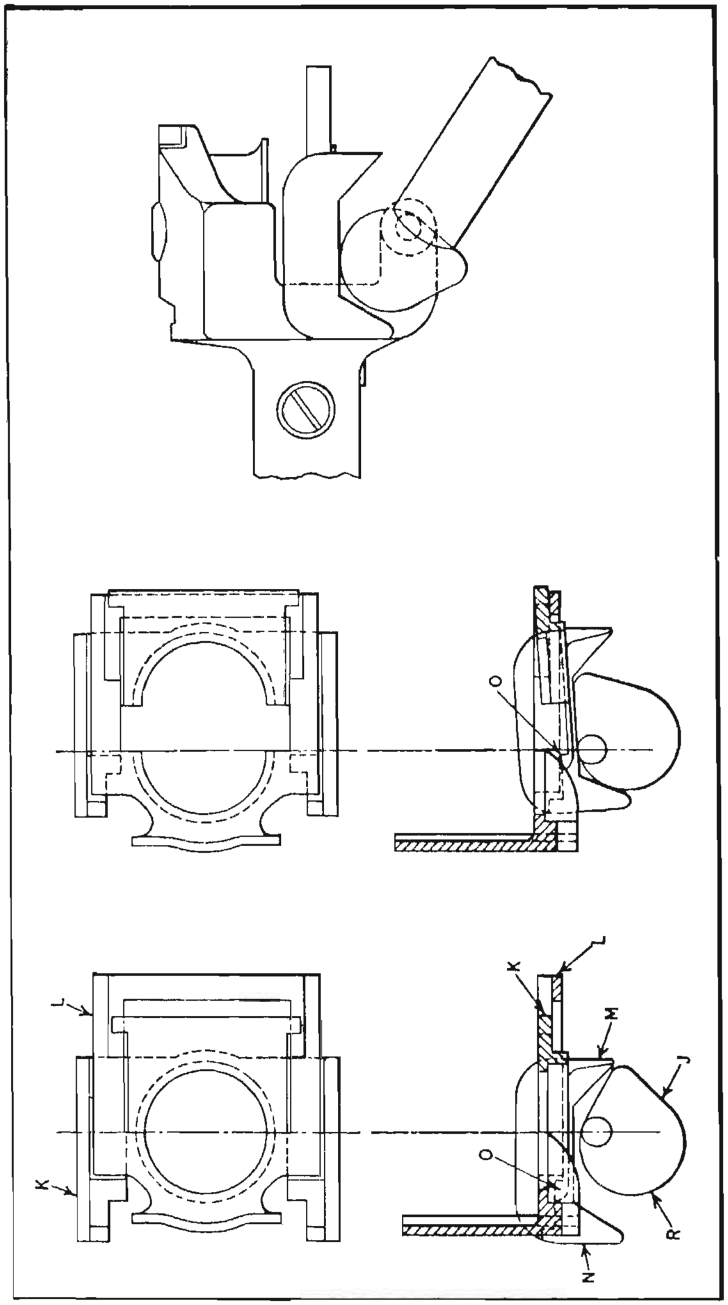


Fig. 25. Views Showing the Closing, Locking, Capping, and Opening Actions Obtained by the Twin Cams

treme right-hand view) causes the cam to rotate into a position where it has forced the cap down over the bottle top, thus completing the operation.

During the capping operation, there is a side thrust on the cam-plate *K* which tends to separate the halves of the table. This would, of course, permit the bottle neck to pass through the table. To prevent this, latches *O* are provided on the cam-plate. These latches hook over the end of the guide plate and lock the two halves after they have been closed around the neck of the bottle.

When the lever *G* is swung toward the left to open the table halves, the cams on this lever tilt the cam-plate *K* enough to disengage the latch and permit it to pass under the guide plate *L*. This is shown clearly in the central view. A spring *P*, Fig. 24, keeps the cam in contact with the cam-plate. One end of this spring is fastened to the body and the other end to the guide plate.

Switching Arrangement for Cylindrical Cam with Intersecting Grooves.—Cylindrical cams having intersecting roll grooves are sometimes used when a cam of small diameter is desired, or when two revolutions of the camshaft are required to one cycle of the follower. These cams have also found application in sewing machines, gas engines, etc. In the ordinary cam of this type, the break in the grooves at their intersection necessitates the use of a follower of special design, because a roll would become wedged at this point. The roll is usually replaced by an oblong shoe, the sides of which curve inward at the ends so that the shoe will be a sliding fit in any part of the groove.

This arrangement is not always satisfactory when a smooth action of the follower is required, owing to the increased clearance around the shoe at the intersection of the groove. Moreover, at this point, the pressure of the sides of the shoe against the corners of the groove causes a great deal of wear on both members. These objections

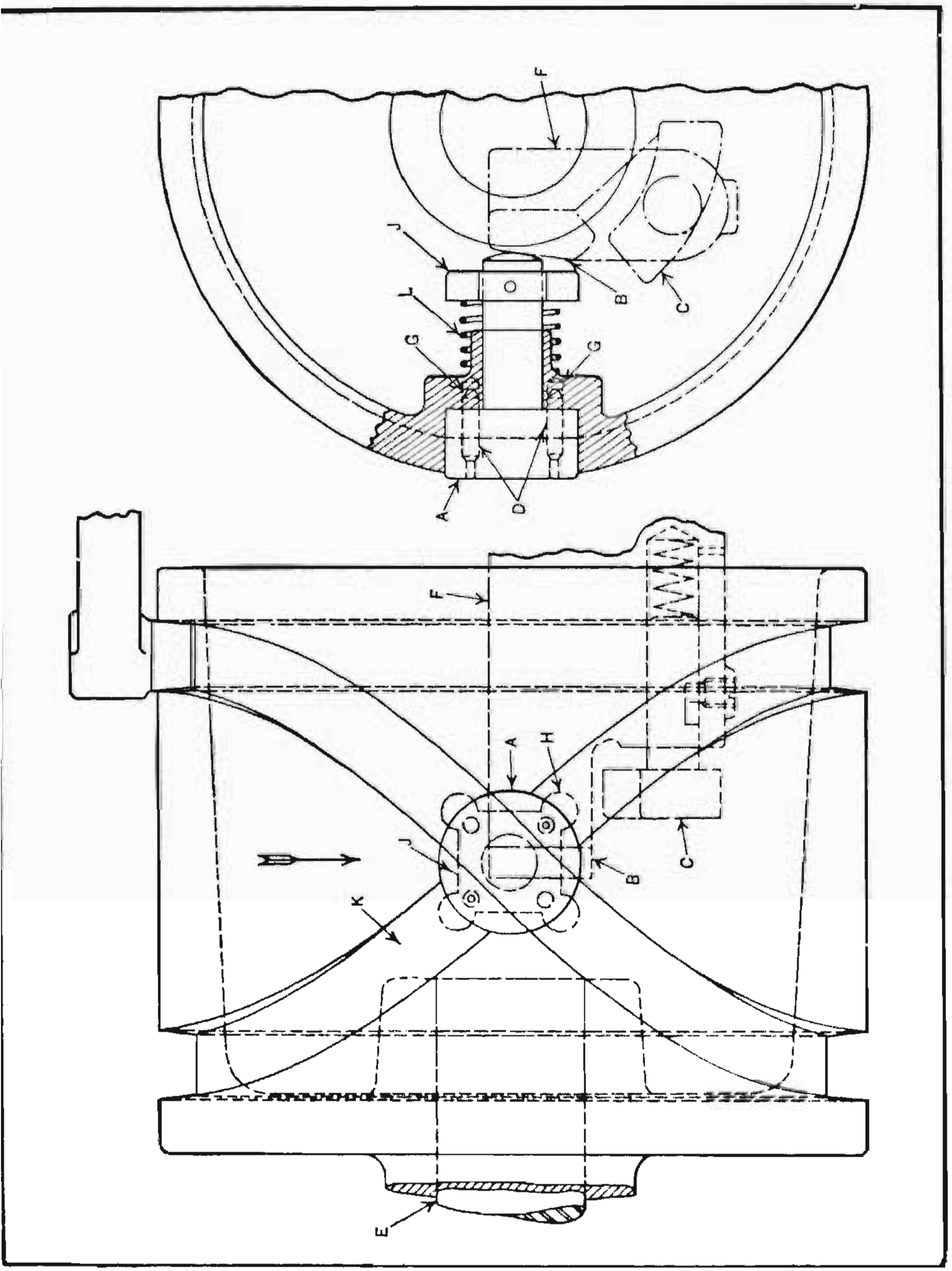


Fig. 26. Cam Having Two Grooves and an Automatic Switch at their Intersection

are overcome, however, by the rather ingenious switching arrangement on the cam illustrated (Fig. 26). It is entirely automatic and provides a continuous groove for the roll, regardless of which groove the roll is in.

The arrangement consists of the grooved plunger *A*, member *J* secured to the plunger shank, and the stationary cams *B* and *C*. These cams are mounted on the arm *F* extending within the cored portion of the cylindrical cam and serve to rotate the plunger 90 degrees for every revolution of shaft *E*. Cam *C* has a shank which is a sliding fit in a hole bored in the arm *F*. The shank is backed up by a coil spring to compensate for the interference of cam *C* when engaging with member *J*. Pins *D* lock the plunger in position after each indexing movement.

In the position shown, the lower end of plunger *A* has engaged cam *B*. Further rotation of the cylindrical cam in the direction of the arrow will cause cam *B* to force the plunger outward until pins *D* have been withdrawn from holes *G*. The plunger is now free to rotate. As the cylindrical cam continues its rotation, the end of cam *C* comes in contact with lobe *H* on member *J* and rotates the plunger 90 degrees. In this position, the pins *D* are directly over another set of holes like those at *G*, and the plunger is seated through the action of the coil spring *L* and locked in position by the pins as they enter these holes.

The groove in the plunger is now aligned with cam groove *K* in which the roll is guided as the cylindrical cam continues to rotate. For each succeeding revolution of this cam, the indexing action of plunger *A* is repeated, so that the plunger groove is always in line with the proper cam groove. In designing a cam of this type, it should be remembered that the cam grooves must cross at an angle of exactly 90 degrees; otherwise inaccurate alignment of the cam and plunger grooves will result. Hardened bushings in the cylindrical cam may also be provided for the indexing pins to reduce wear at these points.

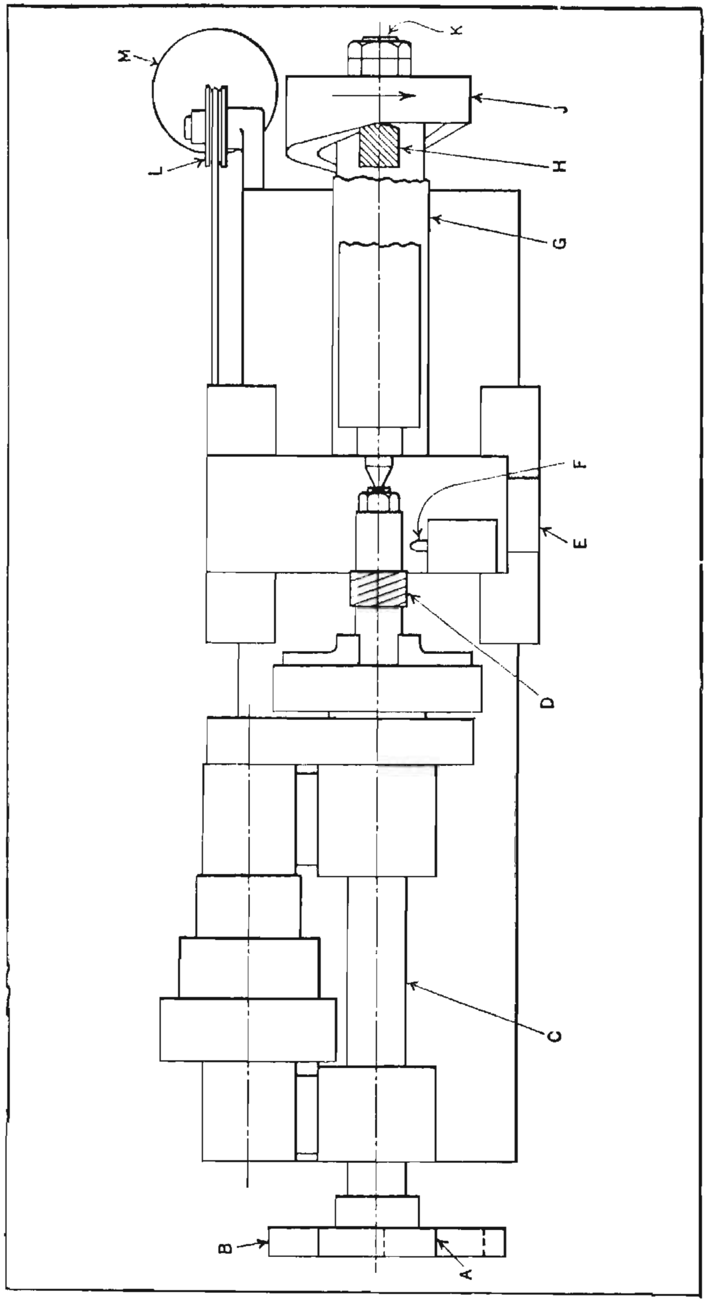


Fig. 27. Redesigned Automatic Threading Lathe in which a Simple Cam and Change-gears Serve to Index the Work when Chasing Quadruple Threads by Rotating the Work 1 3/4 Turns to One Cam Revolution

Cam-Operated Threading Tool and Automatic Indexing for Multiple Thread Cutting.—To meet the demand for an economical method of chasing quadruple threads on the short sleeves, an automatic threading lathe was redesigned. The usual lead-screw was replaced by a cam, and the cross-feed was arranged to feed once in every fourth pass of the tool. By employing a cam and proportioning the change-gears correctly, it was possible to index the work automatically from thread to thread with each longitudinal cycle of the tool.

A diagrammatic plan view of the lathe is shown in Fig. 27. The work, indicated at *D*, is mounted on an arbor and supported in the lathe in the usual manner. The gear *A* on the spindle and gear *B* on the camshaft *K* are connected by means of an idler. On the right-hand end of the camshaft is keyed the cam *J* which imparts an intermittent reciprocating movement to the carriage *E* through the bronze follower *H* attached to the slide *G*. In order to maintain contact between the follower and cam, a weight *M* was provided, which is connected to the carriage by a cable passing over pulley *L*.

The ratio of gears *A* and *B* is such that, for every revolution of the cam, the work rotates 1 3/4 revolutions. In other words, the work assumes a new angular position at the beginning of each cut. This change in position is equivalent to 90 degrees. Thus the work is indexed smoothly from thread to thread without employing a complicated indexing mechanism.

Calculating Gear Ratio and Developing Cam.—The method of calculating the gear ratio and developing the cam for the multiple-threading operation will now be described. In Fig. 28, the line *ON* was drawn equal to the cam circumference, and on this line the cam was developed. Line *ON* was divided into seven equal parts by vertical lines numbered as indicated. Since the thread to be cut was quadruple, each one of these equal parts was assumed to repre-

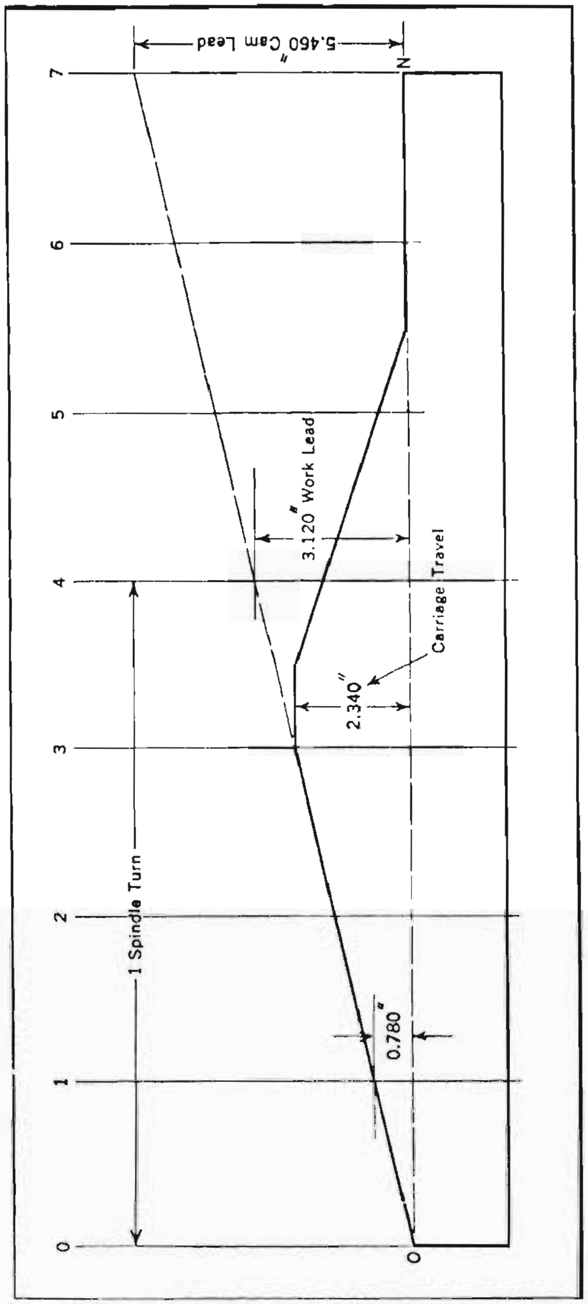


Fig. 28. Development of Cam for Cutting Multiple Thread on Sleeve

sent one-fourth of a spindle turn. Thus the ratio of the spindle turn to the cam turn is 7 to 4; that is, when the cam completes one turn, the spindle makes 7/4 or 1 3/4 turns. Gears corresponding to this ratio have 40 and 70 teeth. The larger gear or the one having 70 teeth is mounted on the camshaft.

Now it is obvious that the part of the cam that forms the thread must be an accurate helix, the development of which is a straight line. To develop this line, a point was located on vertical No. 4, 3.120 inches (the thread lead) above line *ON*. Through this point a straight line was drawn from point *O* to vertical No. 7. It was found that stopping the thread-forming portion of the cam on vertical No. 3 provided ample carriage travel for cutting the thread.

The exact dimensions for the cam rise were found by first dividing the thread lead (3.120) by 4 to obtain the rise for one-quarter revolution of the spindle, or 0.780 inch. This rise was then multiplied by the number of spaces from *O* corresponding to three-quarters of a revolution of the spindle, and the total cam rise, 2.340 inches, was obtained. The lead (5.460 inches) for the working portion of the cam was found by multiplying the entire number of spaces by the rise during one-fourth revolution of the spindle.

The dwell at the top of the cam allowed time for backing the tool out of the thread before the carriage started on its return movement. The longer dwell at the bottom allowed time for moving the tool forward and for the functioning of the cross-feed. With this arrangement, the work rotates continually in the same direction. To enable multiple threads of different leads to be cut on this machine, the sizes of suitable cams and gears can be computed by the method described.

This lathe can also be used for chasing internal threads in short bores. In this case, the action of the cross-feed is reversed, so that the cutting tool will be moved toward the center of the bore at the end of each cut.

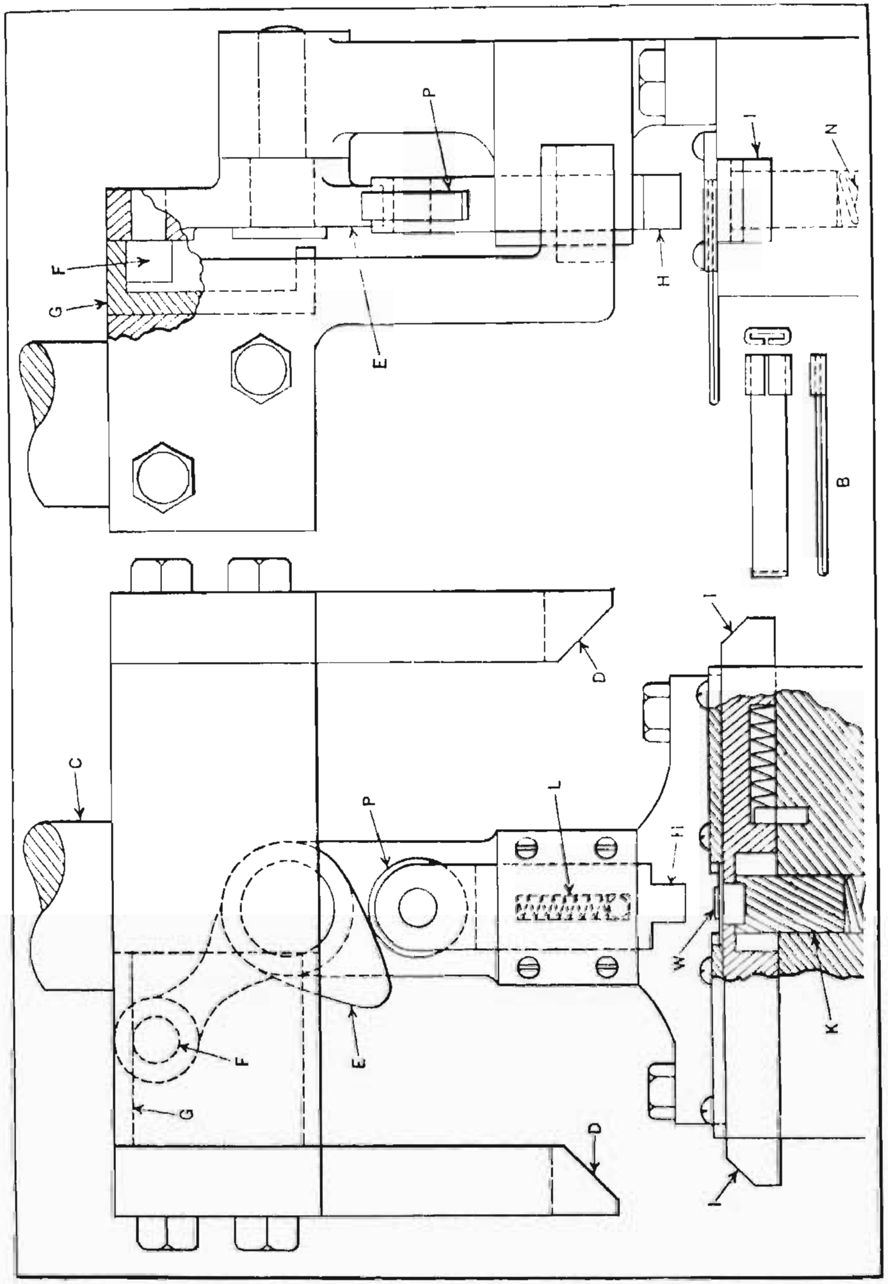


Fig. 29. Folding Die with Pivoted Cam for Rapidly Advancing and Withdrawing Plunger H in Advance of Cams D

Double-Acting Pivoted Cam Mechanism for Folding Die.— The forming plunger H and slides I of the die shown in Fig. 29 are so actuated by an ingenious mechanism that the two tabs of a flat blank are folded and tightly clenched over the central portion, as shown at B, in one stroke of the punch-holder C. After placing the piece in the position shown at W, the press is tripped.

The upper surface of part G, coming in contact with stud F, causes cam E to act on plunger H. Plunger H, acting on the work, forces it between the ends of slides I, causing the tabs of the work to be bent upward. As the lobe of the cam E passes the roller P, the spring L in plunger H reacts on cam E, causing it to swing quickly to the right until it is restrained from further movement by stud F coming in contact with the lower flange of part G, as shown in Fig. 30. This has the effect of causing plunger H to rise rapidly and thus avoid interference with the inward movement of the slides I. As the ram reaches its bottom position, the slides I are operated by the cams D, causing the tabs in the work to be folded over.

Fig. 30 shows the die with the ram at the bottom position. As the ram ascends, the slides I return to the positions shown in Fig. 29, while the cam E, being returned to its original position, again acts on plunger H, causing it to press tightly on the folded tabs of the work. On this stroke of plunger H, there are three thicknesses of metal under it, whereas on the first stroke there were only two thicknesses. This causes plunger K to recede a distance equal to one thickness of the stock. Thus the pressure on the work will always be equal to that exerted by the spring N, and can never be great enough to crush the work.

Operating Two Slides in Opposite Directions with One Single-Groove Cam.—In redesigning a tapping machine to be used for tapping opposite sides of a part simultaneously, two tapping heads were required to travel in opposite directions. This movement, as first suggested, was to be obtained by means of individual cams. A simpler method was devised for transmitting movement to both heads from a single cylindrical cam being used, as indicated by the diagram, Fig. 31.

The diagram is so clear that it hardly requires a descrip-

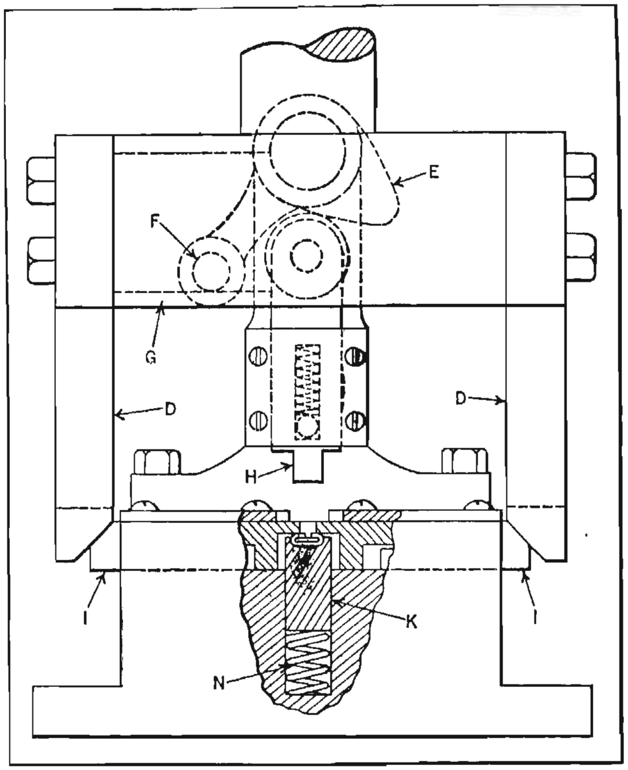


Fig. 30. Die Shown in Fig. 29 in Final Tab-clinching Position

tion. There are numerous cases to which this idea may be applied, with a great reduction in construction and upkeep cost. As indicated, one cam groove serves both heads (not shown). In stationary guides *A* and *B*, on opposite sides of the cam, are slides *C* and *D*. These slides carry the rolls *E* and *F*, both of which engage the same cam groove. The

slides are connected to their respective heads by the tie-rods *G* and *H* through which the required movement is transmitted.

Sliding Triangular Cam for Reducing Cam Size and Stroke.—In many automatic machines, sliding cams are employed for transmitting a straight-line movement to the tool or the work-holder, followed by a dwell to permit load-

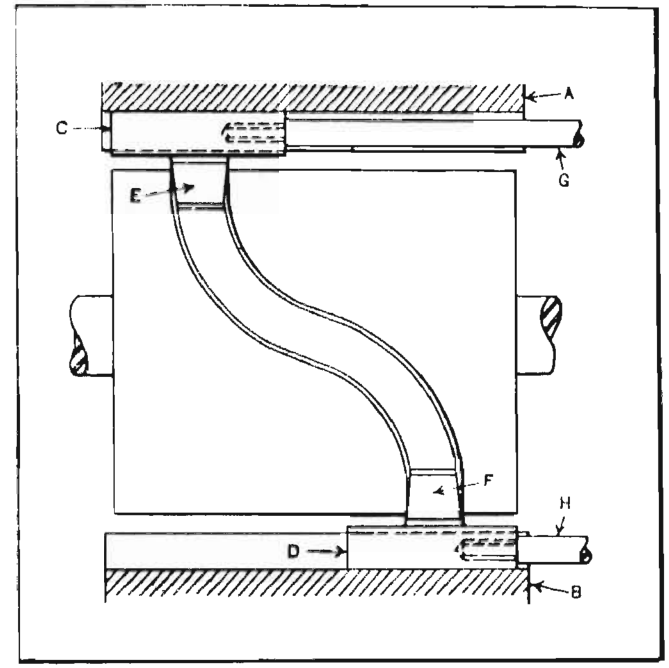


Fig. 31. Cylindrical Cam with One Groove which Serves to Move Two Slides in Opposite Directions

ing and unloading of the work. Frequently this dwell is unusually long, and the movement of the cam follower considerable; hence, a cam of the usual design would not only be large in proportion to other parts of the machine, but also would require a comparatively long stroke. These objections are overcome with the cam shown in Fig. 32. This cam is positive and compact.

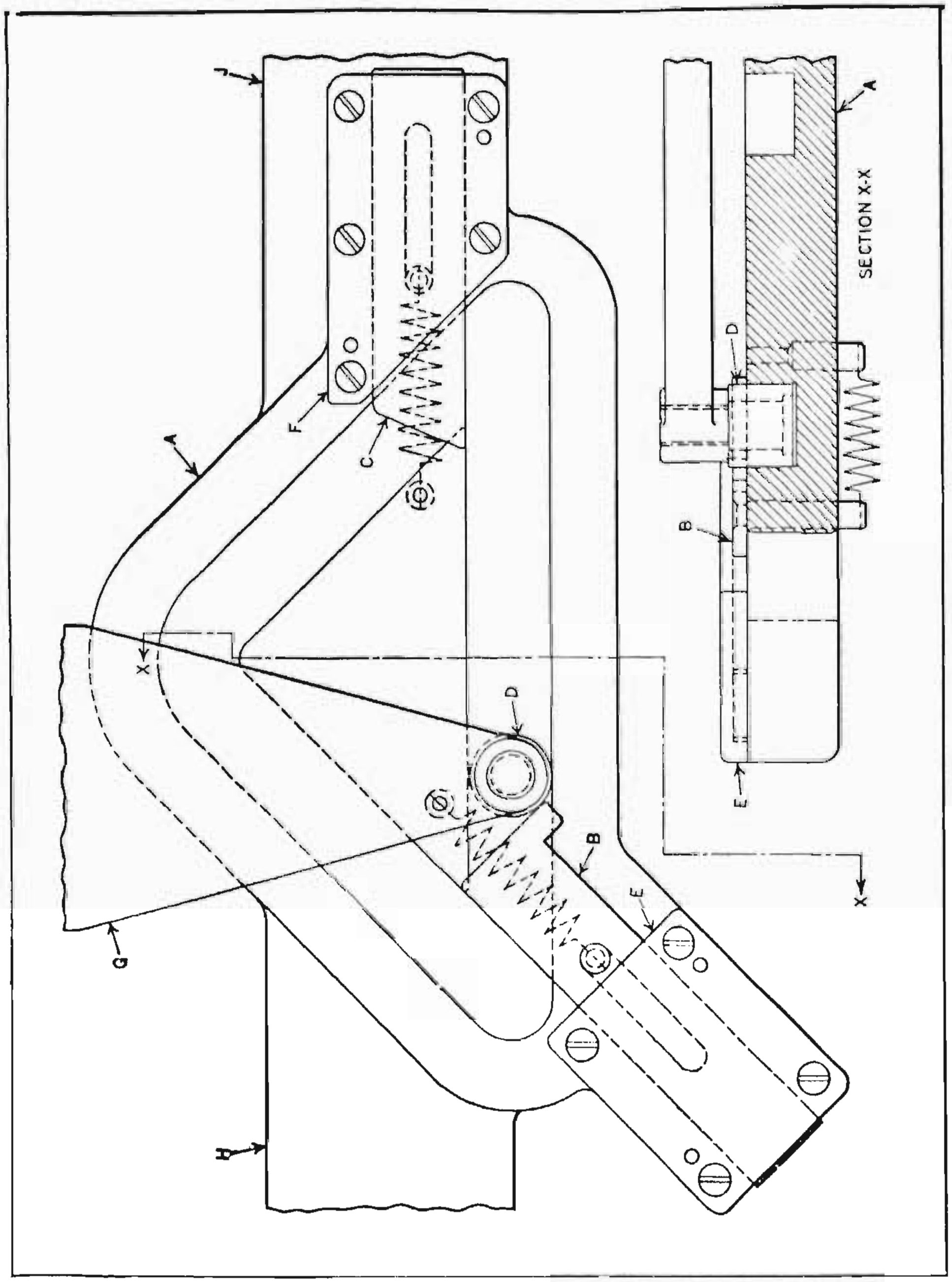


Fig. 32. Triangular Design of Cam for Obtaining Long Stroke and Dwell from Relatively Small Cam

The cam A is supported at the ends H and J by suitable bearings, and is given a reciprocating horizontal movement by some member of the machine. It has a continuous roll groove following a triangular path, and is equipped with locking plates B and C for retaining the roll D in the proper section of the groove. These locking plates are a sliding fit in the caps E and F, respectively, and are normally held in the position shown by coil springs.

In the position indicated, the cam has nearly completed its stroke toward the right. Further movement of the cam will cause the roll to depress the locking plate B; and at the end of the stroke, when the roll has reached the end of the horizontal section of the groove, the plate will once more return to the position shown.

As the movement of the cam is reversed, the roll is forced upward along the edge of the plate and finally into the groove, imparting a vertical upward movement to the follower arm G. This movement of the follower arm continues during the first half of the cam stroke. During the remainder of the stroke, however, the follower arm is returned to its starting point, after having passed the locking plate C, which is similar to plate B.

At this point, the movement of the cam is reversed and the roll simply rides in the horizontal section of the groove for the entire return stroke of the cam. During the latter stroke no vertical movement is imparted to the roll, and hence the follower arm G dwells at this time. This completes the cam cycle. The distance that the cam follower moves, as well as the timing, may be varied by changing the angle of the angular groove sections.

Double-Action Cam that Rotates Follower and Moves it Axially.—The interesting mechanism Figs. 33 and 34 is used in a four-slide spring-winding machine. Springs 1/2 inch in diameter and 1 1/4 inches long are made in this machine. At one station, the spring is coiled and cut off. It is then carried, by means of a transfer arm, to another

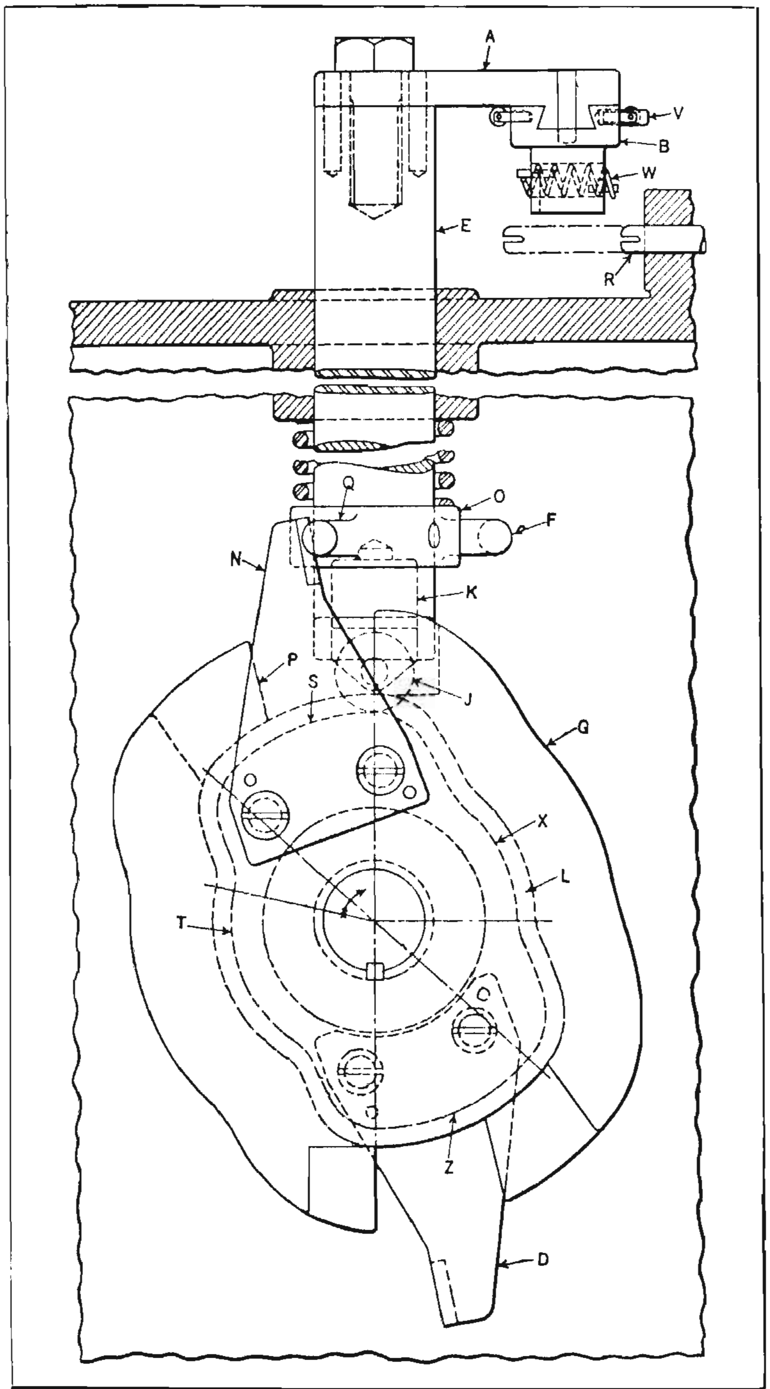


Fig. 33. Cam Mechanism for Operating Transfer Arm of Spring-coiling Machine

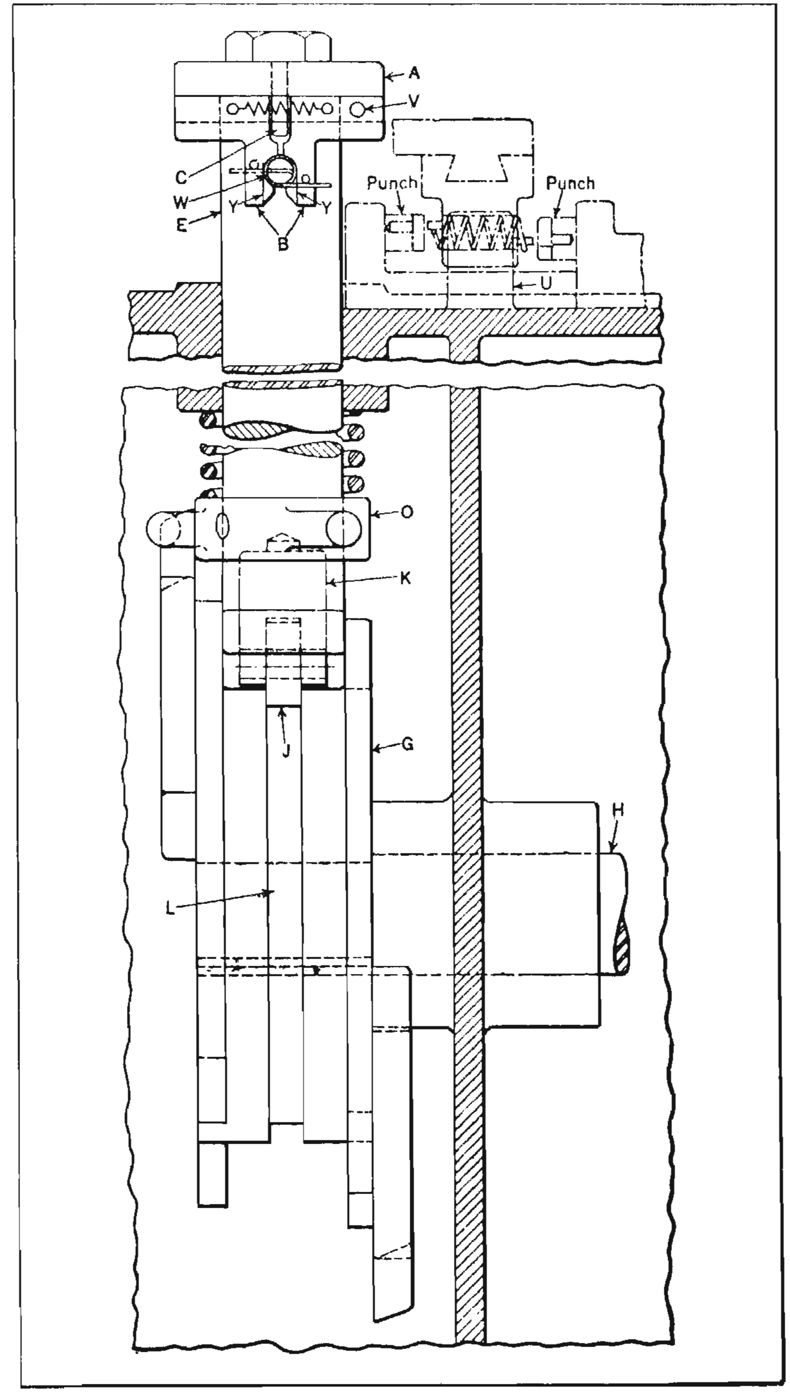


Fig. 34. Side View of Cam Mechanism Shown in Fig. 33

station, where the ends are bent parallel with its axis, after which the completed spring is ejected from the machine. The transfer arm moves through three different planes during each cycle, yet all its actions are controlled by a single cam. The reason for forming the spring ends at a separate station is that another spring is being wound while the preceding one is being formed. With this arrangement, the production is practically double that obtained when the forming was done on the coiling mandrel. As a matter of fact, it would have been extremely difficult to perform all the operations at one station.

The transfer arm *A* has two jaws *B* mounted on its overhanging end. Between the jaws is gripped the coil spring *W*, on which the coiling and cutting operations have been performed. The jaws are centralized by the pin *C* against which they are held normally by the coil spring shown. The arm is bolted and doweled to the vertical plunger *E*, which is a free fit in a long bearing cast in the machine frame. The plunger is given a combined vertical and rotary movement by means of the cam *G* mounted on the drive shaft *H*.

The connection between the cam and the plunger is made by the roll *J* pivoted in the plug *K*. The plug, in turn, is a free fit in a hole bored in the lower end of the plunger. Thus the plunger can be rotated to any position, yet the roll will remain in the same plane, being constrained by the continuous cam groove *L*. As indicated, jaws *B* have grasped spring *W* and elevated it vertically to the position shown, through the action of cam *G*. Incidentally, the coiling arbor *R* has automatically receded to permit the spring to pass upward. The lower end of plunger *E* is square and is a sliding fit between the flanges of the cam. Thus, when the square end is confined between the flanges, the plunger cannot rotate. However, at certain points in the flanges, gaps are provided to permit rotation of the plunger for swinging the transfer arm 90 degrees to the forming station indicated in dot-and-dash outline at the upper part of Fig. 34.

The finger *N* is fastened to the cam for the purpose of rotating the plunger at this time. This finger engages a lug on the collar *O*, pinned to the plunger, and starts the rotation of the plunger. The rotary movement is then picked up and continued as the end *P* of the flange comes in contact with the squared end of the plunger. This action is more clearly illustrated in Fig. 35. Here the cam is rotating in a clockwise direction and the finger *N* is about to swing the plunger in the direction indicated by the arrow.

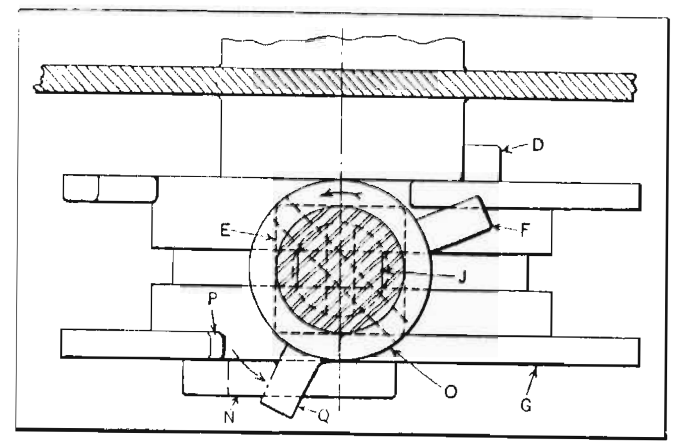


Fig. 35. Plan View of Collar O, Fig. 33, Showing Finger N About to Rotate Plunger E

As the cam rotates the finger *N* engages lug *Q* and rotates the plunger until the flange end *P* comes in contact with the squared end of the plunger and continues the rotation of the latter until it has completed its 90-degree movement. At this position, the squared end of the plunger enters between the flanges, thus preventing further rotation of the plunger, and in addition, the finger and lug absorb the entire starting torque. The rotary movement of the plunger occurs while the roll is passing over the concentric portion *S* of the cam; hence, the height of the arm remains constant at this time. However, as the cam continues to rotate, the roll descends to the low concentric part of the cam lobe at *T* and dwells, causing the arm also to descend and dwell. In descending, jaws *B* enter between two stationary guides *U* which prevent the jaws from opening during the forming operation. The operation at this station consists of bending the projecting ends of the spring outward so that they will be parallel with the axis of the spring. This is done by the automatically controlled punches which advance, with their slides, and bend the ends over the corners *Y* of the jaws. The position of the spring ends relative to the jaws is maintained by the two pins indicated.

When the ends have been formed, the cam raises the arm vertically to its former height. At this time, the roll engages the cam surface at *Z* and, as on the opposite side of the cam. the plunger ana arm are brought back to their original position. In this case, however, finger *D*, engaging lug *F*, starts the rotation of the plunger, after which the corresponding flange end completes the 90-degree movement. When the arm is swung back, a latch (not shown) engages the pin *V* and opens the right-hand jaw, allowing the completed spring to drop into a chute. The cam then allows the arm to dwell until the succeeding spring has been coiled and cut off. Next, the roll passes to the cam surface *X*, causing the plunger and arm to descend until the jaws snap over and grip the spring. This completes the cycle.

The heavy coil spring on the plunger insures constant engagement of the roll with the cam. Although this mechanism was designed primarily for a two-station machine, the same principle can be used for three or more stations by merely modifying the cam throws and adding the required fingers and lugs. In order to facilitate the machining of the cam, the cam is made in two sections and fastened together with screws, the parting line coinciding with the side of the roll groove. For the purpose of simplification, this sectional construction is not shown.

CHAPTER II

INTERMITTENT MOTIONS FROM GEARS AND CAMS

The term "intermittent motion" is applied to mechanisms for obtaining a "dwell" or possibly a series of dwells or moving and stationary periods of equal or unequal lengths. Many different designs of intermittent motions are in use because they are required on so many different types of automatic and semi-automatic machines. The intermittent motions illustrated and described in this and the two following chapters, supplement the two chapters on this general subject found in Volume I of INGENIOUS MECHANISMS FOR DESIGNERS AND INVENTORS.

Advancing Reciprocating Motion with Dwell at Each Point of Reversal.—In coating certain parts of household appliances with enamel by means of a combination dipping and baking machine, the parts are hooked on an endless conveyor chain and passed through a bath of enamel and then through an adjacent heating oven for drying the coated surfaces quickly. In order to facilitate the spreading of the enamel while the parts are passing through the bath, the chain is given an advancing reciprocating movement. The chain advances to deliver the parts to the oven.

This movement of the chain is obtained by the mechanism shown in Fig. 1. The mechanism is mounted on the base *A* of the machine. It consists essentially of a combination of planetary gearing and a double intermittent gear arrangement. The intermittent gearing provides the reciprocating movement, while the planetary gearing is necessary to transmit this movement to the chain sprocket.

Beginning with the intermittent gearing, ring gear *B*

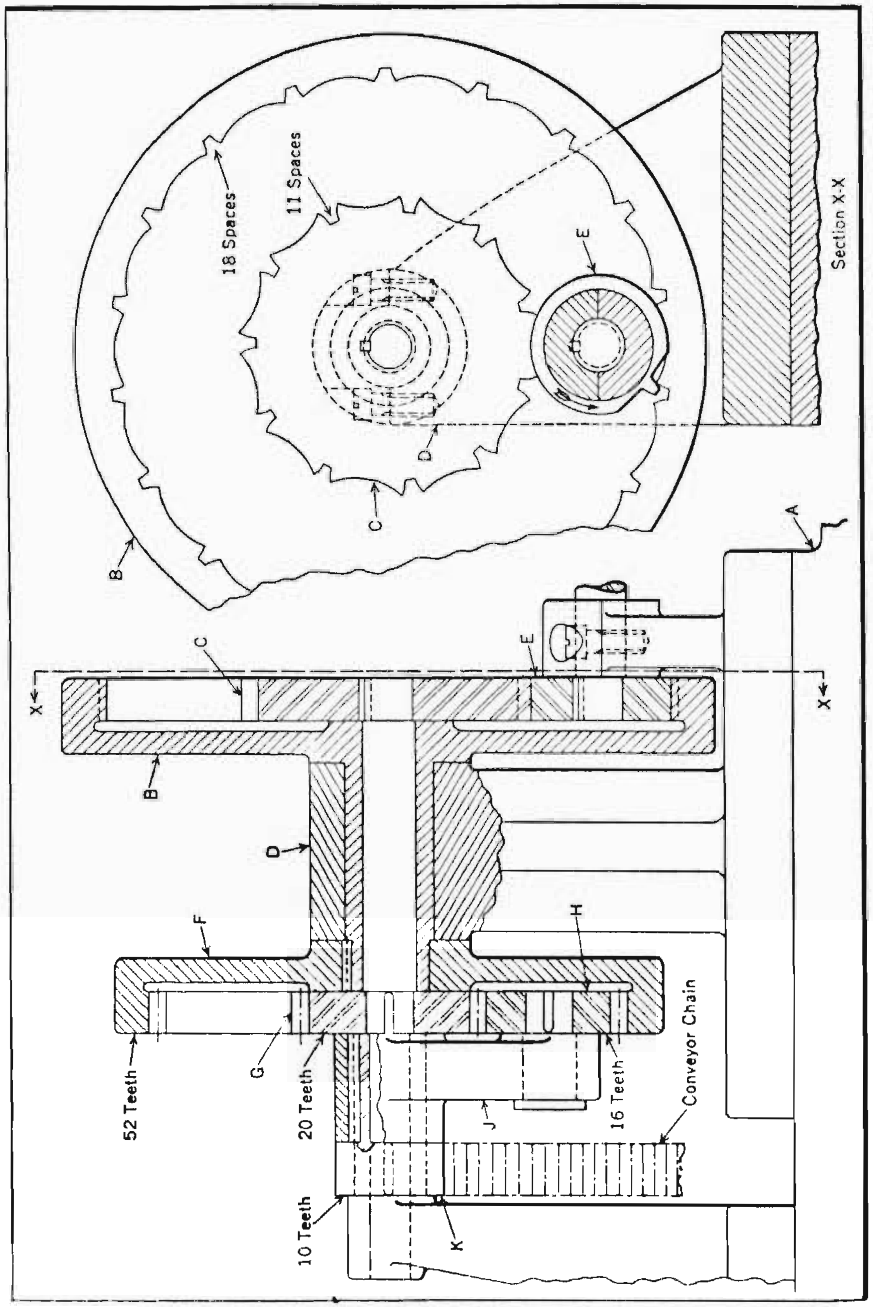


Fig. 1. Mechanism for Imparting Advancing Reciprocating Motion to Conveyor Chain of Enamel Dipping and Baking Machine

and center gear *C* are supported in the stationary bearing *D* and mesh with the driving pinion *E*, which rotates in a stationary bearing. At the left-hand end of the sleeve that forms the journal for gear *B* is keyed an ordinary internal ring gear *F*, and on the shaft to which gear *C* is secured is keyed the pinion *G*. Gear *H* is free to turn with the stud in arm *J* and meshes with internal gear *F* and pinion *G*. The arm *J* is keyed to an extension sleeve integral with the conveyor chain sprocket *K*, the sleeve being free to rotate on the center shaft.

When driving gear *E* rotates in the direction indicated by the arrow, the single tooth will engage the adjacent tooth space in gear *B* and rotate the latter 1/18 revolution. During this movement, gear *C* will be locked in a stationary position by gear *E*. Hence, the partial rotation of gear *B* will rotate gear *F* and cause gear *H* to roll around the stationary pinion *G* and swing arm *J*, with the sprocket *K*, in the same direction.

As the gear *E* continues to rotate, its cylindrical portion locks gear *B* and the single tooth engages a tooth space in the center gear *C*, rotating the latter 1/11 revolution, after which the cylindrical portion of gear *E* locks it in a stationary position. Rotating gear *C* in this way causes gear *G* to rotate and roll gear *H* on the now stationary gear *F*. In this manner, gear *H* carries arm *J* and sprocket *K* around the center shaft in a direction opposite to that of the driving gear *E*. This completes one cycle of movements.

The required angular movements of the sprocket are as follows: 14 1/2 degrees, or approximately 0.04 revolution, in a clockwise direction, as observed from the right-hand end of the mechanism. The sprocket then dwells and reverses its movement, rotating 9 degrees, or 0.025 revolution. The angular advance of the sprocket for each cycle is 0.04 — 0.025 = 0.015 revolution, or about 5 1/2 degrees. In calculating the ratios and the number of teeth and tooth

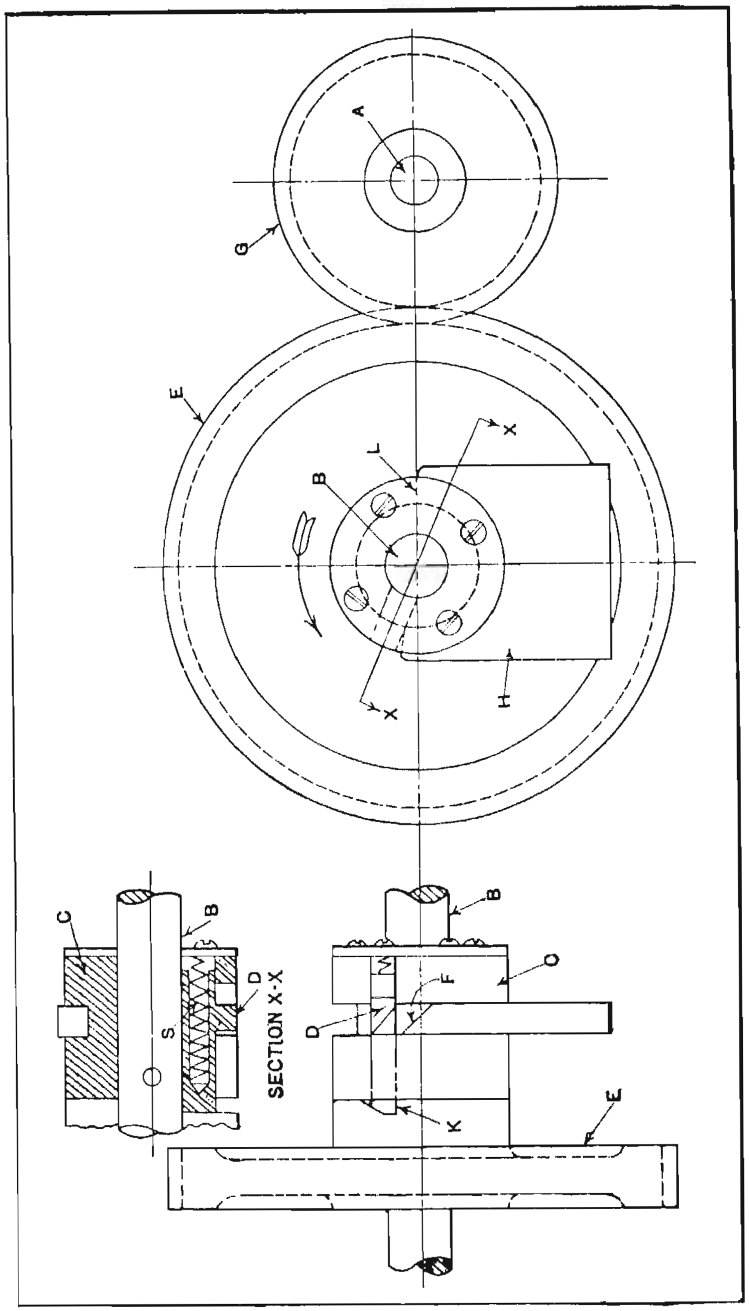


Fig. 2. Intermittent Gear Drive Actuated by Cam-operated Clutch Dog

spaces in the gears, two separate conditions are involved: First, the sprocket movement when gear E rotates gear B while gears C and G are locked; and second, the sprocket movement in the opposite direction when gear E rotates gear C while gears B and F are locked.

One Revolution of Shaft is Followed by Dwell Equivalent to One Revolution.—The shaft A of the drive shown diagrammatically in Fig. 2 is required to make a revolution and then stop or dwell for a period equivalent to one revolution. Shaft A is driven by shaft B, which rotates continuously. The gear G is keyed to shaft A and meshes with the gear E, which is a running fit on shaft B. The sleeve C is pinned to the driving shaft B. When the drive is in operation, the clutch dog D, which is a sliding fit in a slot in sleeve C, drives gear E one-half revolution; then as the angular face on the dog comes in contact with the angular or cam face F of the stationary piece H, the dog is withdrawn from contact with gear E at point K, allowing gear E to remain stationary while shaft B makes one-half revolution.

After shaft B has made one-half revolution, the dog D passes out of contact with the piece H at point L, allowing the dog to re-engage gear E through the action of spring S. Gear E then makes one-half revolution, following which the cycle of movements described is repeated. Thus gear E rotates one-half revolution and then remains stationary while shaft B rotates one-half revolution.

As gears E and G have a driving ratio of 2 to 1, gear G is given the required intermittent motion. A wide range in the timing of the intermittent motion may be obtained by varying the ratio of the gears and the length of the actuating or cam surface of the piece H.

Positive High Speed Intermittent Rotary Motion.—The mechanism shown in Fig. 3 provides the intermittent rotary motion required for operating the conveyor of a wire stitcher. The member A receives its intermittent rotary motion from the continuously rotating shaft B through the positive indexing action of an eccentric strap C operated by the eccentric L keyed to shaft B. A sprocket or gear—not shown—attached to the hub or face of member A transmits the intermittent motion to the conveyor.

At each revolution of shaft B the member A is indexed 1/11 revolution by pin D which engages one of the eleven evenly spaced slots S. During the idle or return movement of pin D, the member A is locked in position by pin E. The ratio of the idle time between the indexing movements, to the indexing time, is 73 to 107 in the design illustrated.

At the beginning of the indexing movement, member A

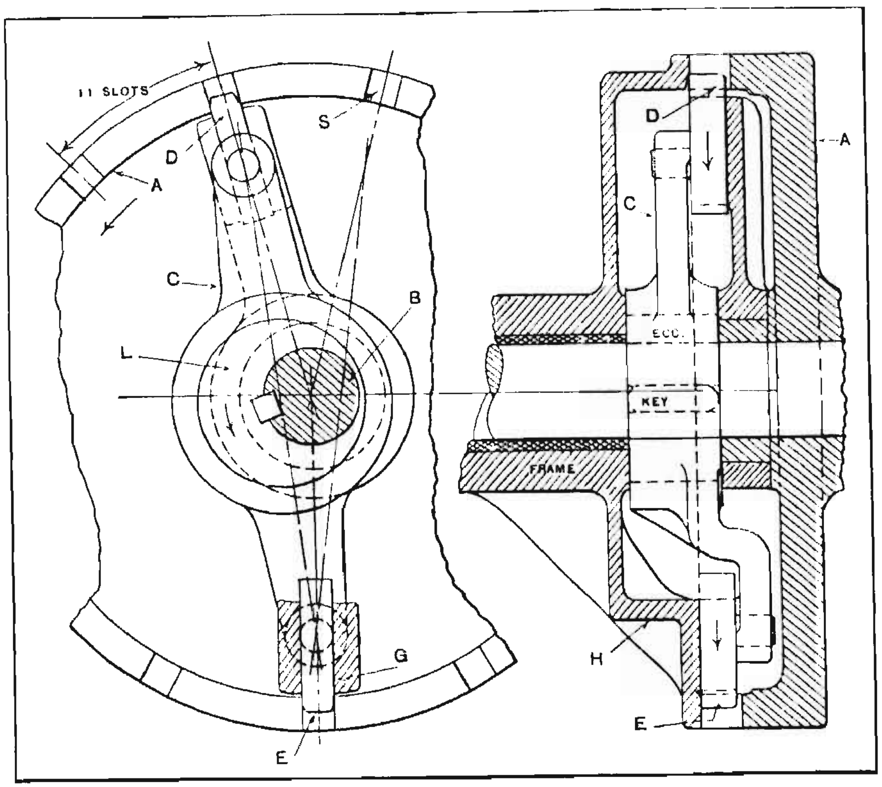


Fig. 3. A Constantly Rotating Shaft B and Eccentric Imparts a Positive Intermittent Motion to Member A which is Locked During the Idle Period

moves slowly, but the speed increases rapidly to the maximum and then slows down as the end of the movement is reached. As the mechanism stops the load slowly and without shock, it can be operated at high speeds, as compared with the usual ratchet wheel and pawl mechanism, which is difficult to balance and has a tendency to "overthrow" under appreciable loads.

The member A is always engaged by one or both of the actuating teeth D and E. These teeth are pivoted to the extreme opposite ends of the eccentric strap C. Overthrow is prevented by the locking tooth E, which is a free sliding fit in the groove G machined in the frame H. This slot restricts the movement of the pivoted tooth E so that it is forced to travel in a vertical direction.

The view at the left shows the parts of the mechanism in the positions they occupy at the completion of the indexing movement. It will be noticed that the tooth E has entered one of the slots in member A before tooth D has become disengaged from another slot of the member.

The peculiar motion imparted to the eccentric strap C by the eccentric L, due to the vertical path which its lower end is forced to follow, causes the top end of the strap to move in an elliptical path, carrying with it the actuating tooth D. Tooth D is always held in a radial position by a slot in the guiding arm, which is a free fit on the hub of member A. The elliptical motion and the radial guide force the actuating tooth D to engage and disengage successively the slots in the edge of member A, thus converting constant rotary motion into a positive intermittent motion.

The mechanism is equally efficient when operated in either direction. In adapting it for other purposes, the following characteristics should be considered: The slot spacing in member A controls the amount of intermittent motion and also the idle time. There must be an odd number of slots if the best action is to be obtained, but as a gear or sprocket drive of the proper ratio can be used to suit individual requirements, this characteristic is not a vital fault.

If too few slots are used, the eccentric throw will be excessive and the action of the mechanism will not be so smooth as with a greater number of slots. It is well to bear in mind that fewer slots decrease and more slots increase the idle time.

The interior of the mechanism shown is filled with grease, but the flanged parts of the frame could be fitted with a cover and a packing ring could be provided on the hub of member *A* so that the mechanism could be filled with oil. With this form of lubrication, the carrying power would be increased to handle greater loads at higher speeds.

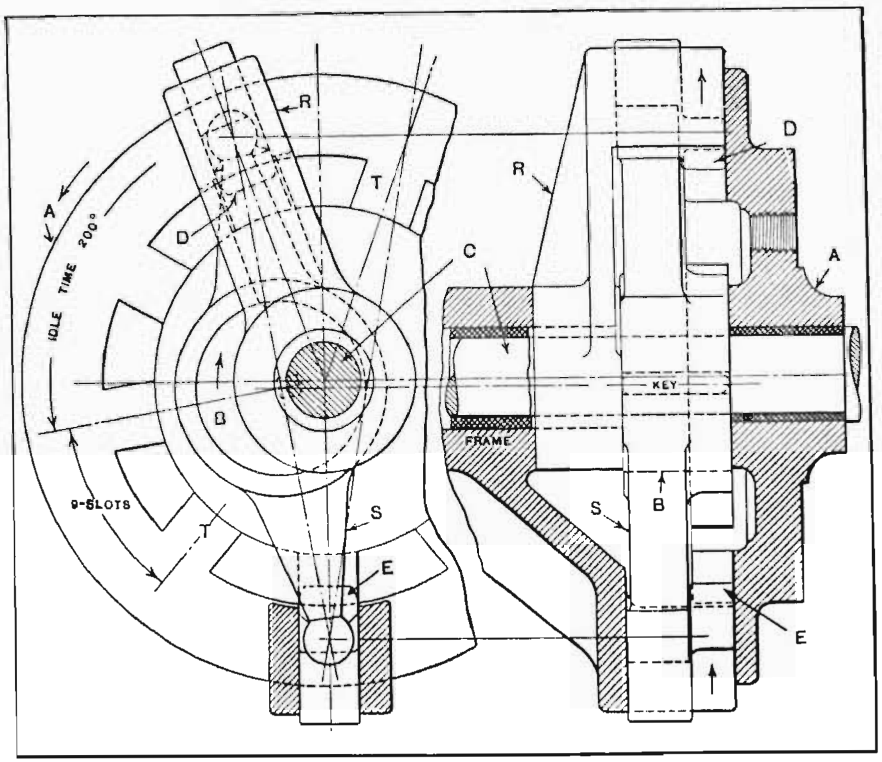


Fig. 4. Intermittent Mechanism which Provides a Longer Idle Period than the Design Shown in Fig. 3

Another Design of Positive High-Speed Intermittent Motion.— The intermittent rotary motion just described is suited for use where the idle time is short, as compared with the feeding time. The total idle time between the rotary or feeding movements in the case of the mechanism about to be described is equivalent to more than 180 degrees per revolution, while in the case of the previously described mechanism it was less than 180 degrees.

The principal difference in the designs is found in the location of the actuating teeth *D* and *E* (see Fig. 4) in relation to the member *A* to which the intermittent or indexing motion is imparted. The actuating teeth operate on the outside of the slotted ring of member *A*, so that the idle time occurs while the eccentric throw travels above the center line during the return stroke of strap *S*. The forward or feeding movement, therefore, occurs while the eccentric throw travels below the horizontal center line. With this arrangement, the longer throw of the eccentric is utilized for the idle or return movement of arm *S*, while the shorter throw is employed for the feeding movement. It will be noted that the shaft and its eccentric driver *B* rotate in a direction opposite to that of the driven member *A*, whereas in the design previously described, these members rotate in the same direction.

When the mechanism is in operation, the central shaft to which the eccentric driver *B* is keyed rotates at a constant speed. The lower end of the strap *S* is restricted to a vertical motion by the tooth *E*. Tooth *E* is pivotally mounted on strap *S*, having a wide bearing on the strap, and slides freely in a groove in the rigid frame. The tooth *D* at the opposite end of strap *S* has a similar pivoted connection to the strap and slides freely in a groove cut in the radial rocker guide *R*, which is a free turning fit on the central shaft bushing. The guide rocker *R* serves to maintain the tooth *D* in a radial position with respect to the central shaft. When the mechanism is in operation, the eccentric B causes the teeth D and E alternately to engage and disengage the equally spaced slots T cut in the annular ring and produces the intermittent motion of member A.

The vertical motion of the bottom end of the strap S, combined with the rotary motion at the center imparted by the eccentric B, gives the top end of the strap carrying tooth D a peculiar elliptical motion. This motion is such that tooth D alternately engages and disengages successive slots T, thereby imparting the required intermittent motion to the driven member A. The length and shape of the teeth are such that either or both teeth are always in engagement with slots in member A, thus giving a positive control over the motion. Tooth E locks the part A in position while tooth D is on its return or idle stroke. Tooth D engages a slot preparatory to the forward movement before the locking tooth E is disengaged. A hub at the side of member A is provided so that a sprocket or gear can be attached to it, through which the motion is transmitted to other parts.

This mechanism has several desirable characteristics, such as its slow starting and stopping action, absence of over-throw, positive locking between movements, compactness, and ability to operate in either direction. It can also be operated at relatively high speeds. The idle time is determined by the number of slots, and is always equivalent to more than 180 degrees per revolution. With nine slots, as in the design illustrated, the idle time is 200 degrees per revolution. With fewer slots, the idle time would be greater, and with a greater number, the idle time would be less. For the best action, there should be an odd number of indexing slots.

Planetary Intermittent Gearing.— Intermittent gearing of the planetary type may be used to advantage in cases where the driving and driven shafts must be in line with each other, and where a large number of dwells per revolution of the driven shaft is required. A drive of this type is shown in Fig. 5.

The ring gear A is stationary, and the single-tooth gear B is driven by means of the shaft C through the gears D and E. Gear D is keyed to shaft C, while gear E is integral with gear B. Both gears E and B are free to turn on the shaft J, mounted in the arms F and G. A hub on the lower end of arm F turns freely in the stationary bearing H, the intermittent movement being taken from this hub. Gears

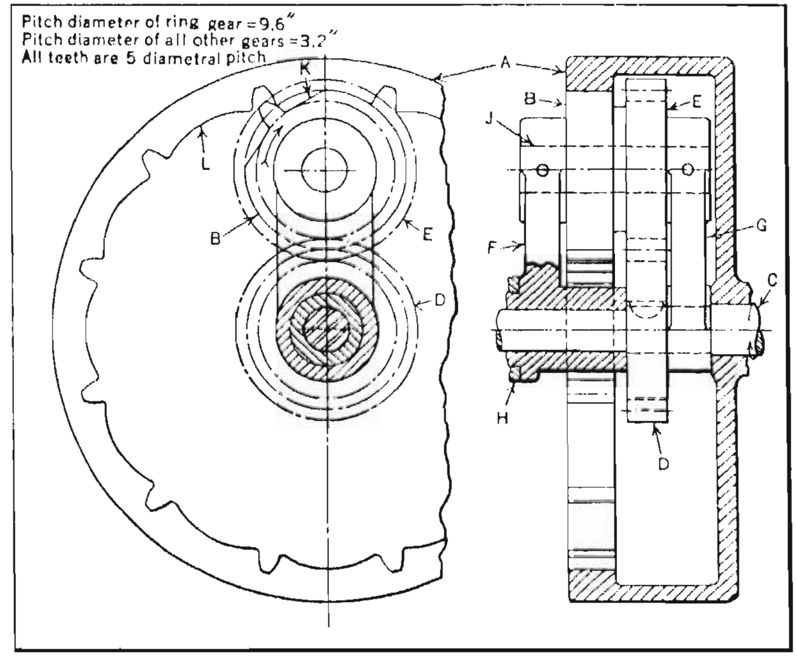


Fig. 5. Intermittent Gearing of Planetary Type

D, E, and B, in this case, have the same pitch and pitch diameters; hence, according to the principles of epicyclic gearing, one-third of a revolution of shaft C is required to index the arm F one division.

In the position shown, the single tooth in gear B is about to engage a tooth space in the ring gear. As soon as this engagement occurs, arms F and G will start to rotate on shaft C in a counter-clockwise direction. Rotation of the

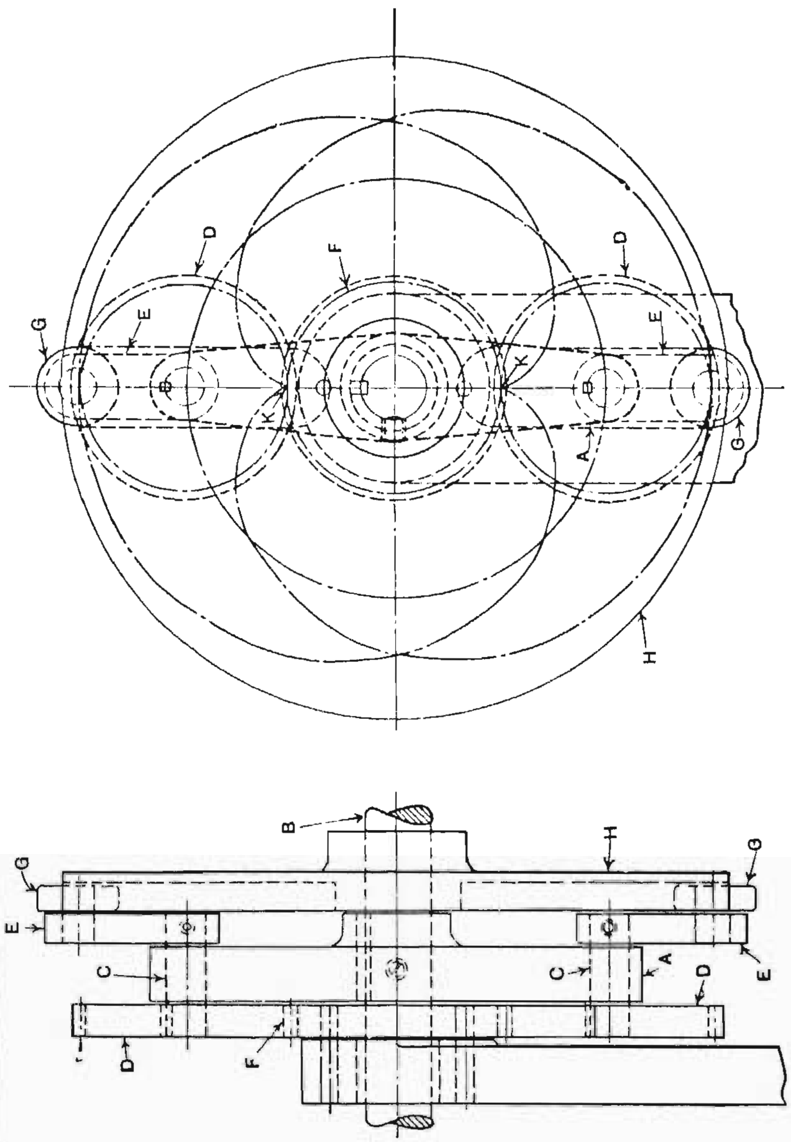


Fig. 6. Planetary Type of Crank Motion for Gradually Varying Speed of Driven Member from Zero to Maximum Velocity and Back to Zero

arms continues until the single tooth has left the tooth space, at which time the concentric portion of gear *B* engages the corresponding cylindrical surface *L* in the ring gear, locking gear *B* and causing arms *F* and *G* to dwell. The arms continue to dwell until the tooth in gear *B* has engaged the next tooth space in the ring gear, which causes the arms to move toward their next dwelling position. In designing the single-tooth gear *B*, sufficient clearance should be provided, as indicated, at *K*; otherwise, interference with the ring gear will result.

Rotary Motion which Varies from Zero to Maximum and Vice Versa.—The purpose of the mechanism illustrated in Fig. 6 is to produce an intermittent rotary motion which will start and stop a driven member without shock and yet keep it under positive control throughout the cycle. This motion is obtained by the practical application of the mathematical curve known as the epicycloid, which is the curve traced by a point on a circle as the latter revolves on the outside of another fixed circle.

The arm *A* is keyed to the driving shaft *B* which revolves continuously at a constant rate. At each end of arm *A* is a revolving shaft *C*, which has a revolving gear *D* keyed to one end and a short arm or crank *E* keyed to the other. The two revolving gears *D* mesh with a fixed gear *F* which is concentric with the driving shaft. All three gears are of equal diameter. At the end of each crank *E* is a roller *G*, the center of which lies on the pitch circle of the corresponding gear *D*. The circular plate *H* revolves freely on the driving shaft *B* as it is driven by rollers *G*, each of which engages a radial slot in the plate. The drive is taken from plate *H* in any desired manner.

It will be seen from the illustration that the centers of the rollers *G* trace the epicycloids shown by the broken curves in the end view. As the arm *A* rotates, the angular velocity of the centers of the rollers and of the driven plate gradually increases from the zero point at *K* until, in the position shown, the angular velocity is greater than that of the driving shaft; from this point and during the following half revolution of the driving arm, the angular velocity of the roller center and of the driven plate is gradually reduced again to zero at point K. Since the gears are of equal

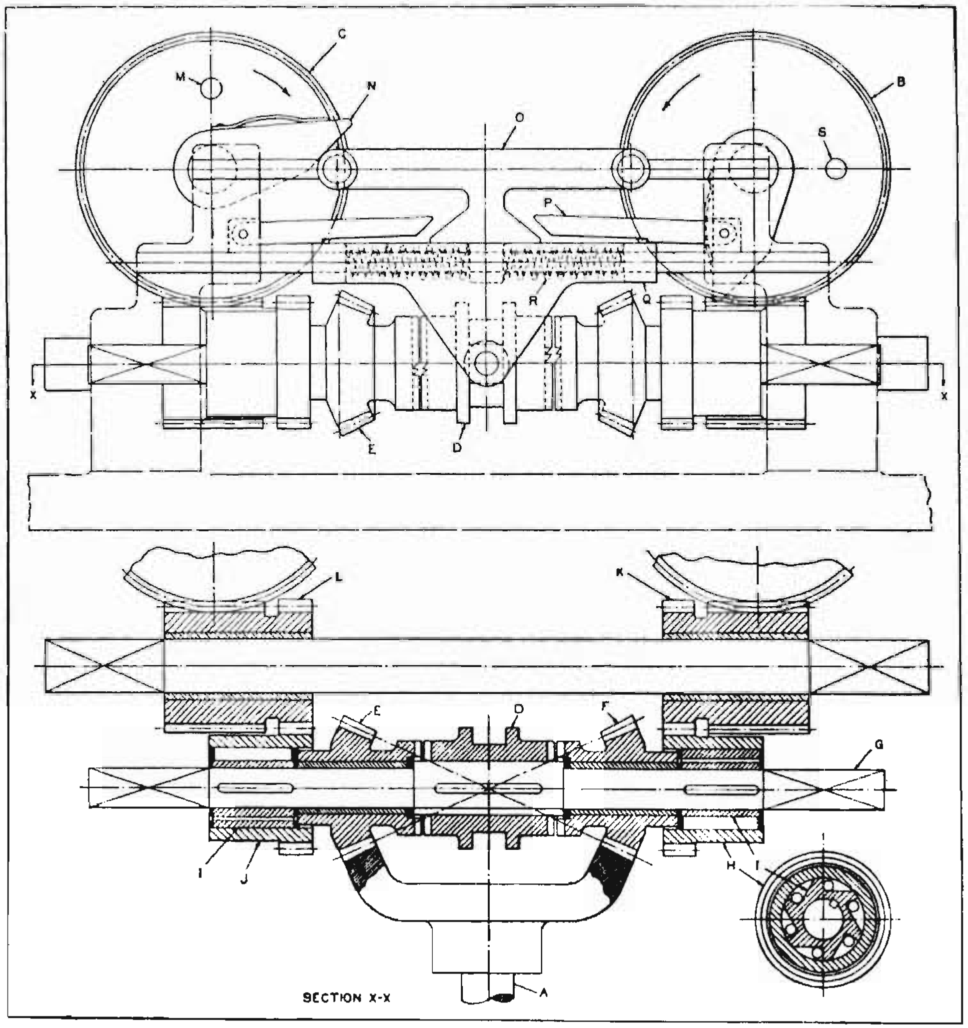


Fig. 7. Mechanism for Driving Worm-wheels C and B Intermittently and Alternately from Shaft A

diameter, the roller centers and the driven plate make a complete revolution in the same time as the driving shaft.

The angular velocity of the driving shaft is constant. If θ equals the angular position of the driving arm from the zero point K, and ω equals the angular velocity of the driven plate H, then $\omega = \dfrac{6a\,(1 - \cos\theta)}{5 - 4\cos\theta}$. The maximum angular velocity of the driven plate is 1.3 times that of the driving arm.

Mechanism for Driving Two Shafts Intermittently and Alternately.—The mechanism shown diagrammatically in Fig. 7 was designed for use on a special machine. In operation, the constantly rotating shaft A, through a gear train, drives worm-wheel B one revolution in the direction indicated by the arrow, after which the gear-shifting mechanism functions automatically, causing worm-wheel B to dwell and the driving motion to be transmitted to worm-wheel C through another gear train. After worm-wheel C has been driven one revolution, the gear-shifting mechanism again functions, causing worm-wheel C to dwell while the driving action is again transmitted to worm-wheel B, thus completing the cycle, which is continuous as long as the driving shaft A rotates.

The clutch member D, which is slidably keyed to shaft G, is shown in the neutral position, but when the mechanism is in operation, this clutch is in engagement with either pinion E or F, causing shaft G to rotate in one direction or the other, depending upon which pinion is engaged. The driving of shaft G in either direction from the crown gear on shaft A is made possible by the "free-wheeling" type friction clutches, consisting of two members H and I, and the friction rollers arranged as shown in the section view in the lower right-hand corner of the illustration. Two members I of the proper hand are keyed to the shaft and the two friction members H and J are slipped over them. Thus, when the clutch member D is in mesh with pinion F, the wedging action of the friction rollers serves to lock members H and I together as one piece, while member J runs freely over its mating member I. When the clutch engages pinion E, the direction of rotation of shaft G is reversed, the drive being through friction clutch J, while member H runs free. Thus, the direction of rotation of shaft G is controlled by the movement of clutch member D.

Spur gear teeth on members H and J mesh with the spur gears K and L, which have worms cut on their hubs that mesh with the worm-wheels B and C, respectively. The manner in which the clutch is automatically controlled to give the worm-wheels B and C their respective intermittent movements is shown by the upper view.

Assume that the mechanism is in operation and that clutch D is in engagement with gear E, so that worm-wheel C is being driven in the direction indicated by the arrow. When pin M on the worm-wheel comes in contact with the flat spring on the swinging arm N, which is a free turning fit on the worm-wheel shaft, it causes the swinging arm to come in contact with the roller on shifter lever O. The latch P would be down at this time instead of in the position shown. In the down position, a step on the latch engages a collar Q on the shifter slide, preventing the slide from moving to the right. Thus, continued rotation of the arm N serves to compress the spring R until a cam surface on the shifter lever lifts latch P, releasing the spring, which forces clutch member D to the right into mesh with gear F, engaging the drive to worm-wheel B, and allowing worm-wheel C to dwell. When this takes place, the shifter lever O, being released from the pressure exerted by spring R, also moves to the right and the arm N is rotated past the roller on lever O by the flat spring, previously compressed by pin M.

When worm-wheel B has rotated through the required angle, the pin S comes into contact with the flat spring on

a swinging arm similar to the arm N previously described. The movement of the clutch member D into engagement with gear E is accomplished automatically, the same as the movement in the opposite direction. This cycle of operations is repeated automatically.

Adjustment of Intermittently Driven Sprockets while Drive is Operating.—Motion picture projectors are designed to give the film a rapid intermittent movement, stopping it sixteen times every second. These dwelling periods in the movement of the film are so timed that the light is projected through the film only when it is stationary. Moving pictures, therefore, are in reality a series of sixteen stationary pictures projected on the screen every second at normal operating speed, and owing to the "persistence of vision" this rapid succession of still pictures causes the successive views to blend into one another and produce the effect of continuous motion.

The edges of the film have accurately spaced perforations which are engaged by teeth on a double sprocket E (see Fig. 8). The drive to this sprocket is through the internal driving gear A, the driven gear B, driving disk C of the intermittent motion, and driven member D on the sprocket shaft. This driven member turns 1/4 revolution for every complete turn of driver C, and this quarter turn of D occurs during one-fifth of the revolution of C. During the remaining four-fifths of a turn of C, member D is locked in the stationary position.

This mechanism has an original feature which makes it possible to shift the film sprocket E from, say, its lowest position, which is the one illustrated, to a higher position, without interfering with the intermittent drive and while this drive is in operation. It was discovered accidentally that four points in a plane may be so located relative to one another that two of these points, if moved along straight lines perpendicular to each other, will cause a third point to describe an arc about the fourth; thus, if points a, b,

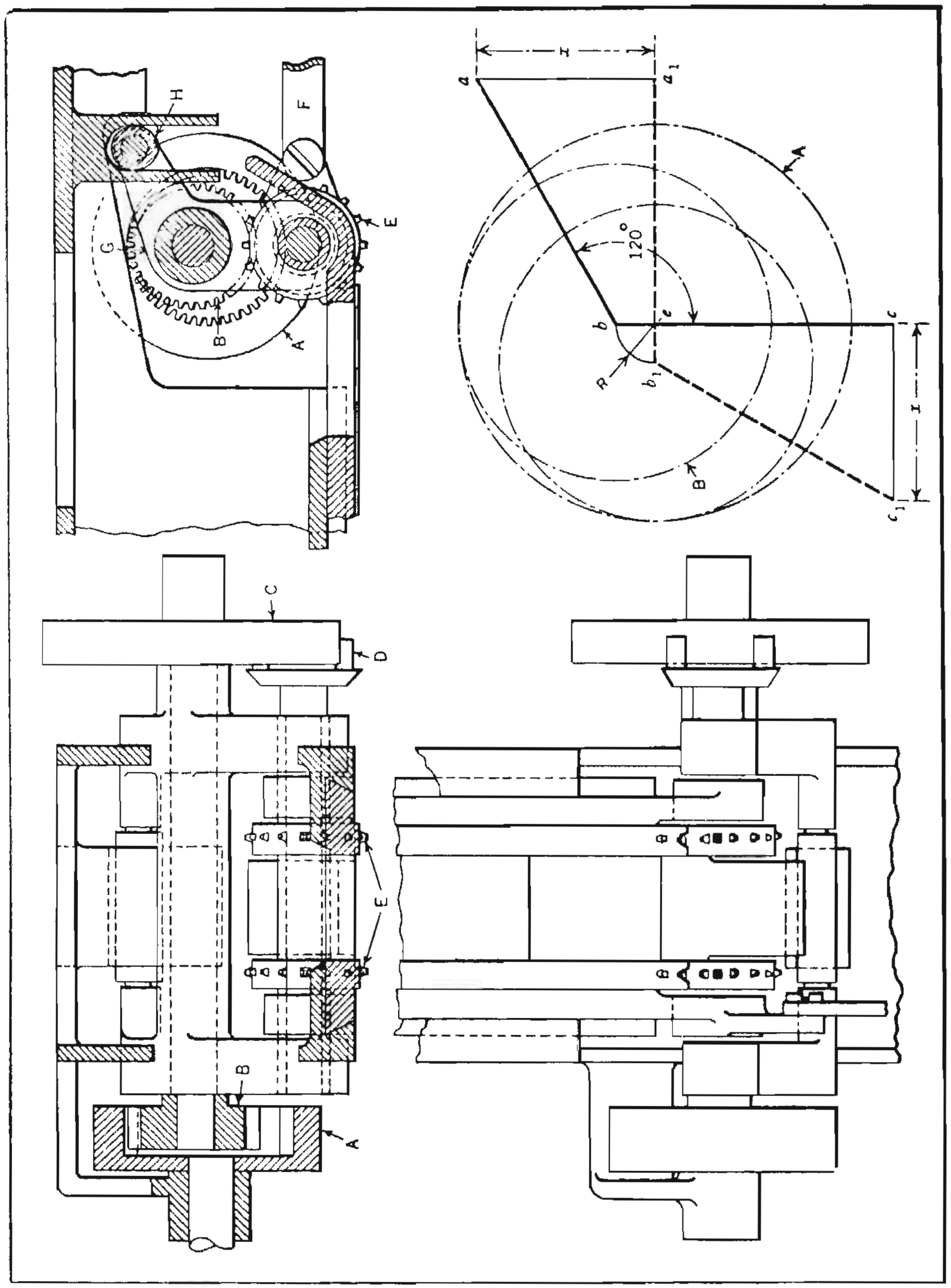


Fig. 8. Mechanism of a Moving Picture Projector which Makes it Possible to Adjust Film Sprocket Along a Straight Line without Interfering with its Intermittent Drive

and c on the diagram are located properly, the movement of point a to a_1 and of c to c_1 will cause point b to describe an arc of radius R about point e.

Before describing the essential requirements in this design, its practical application will be explained. This application is illustrated by the diagram just referred to, in conjunction with the sectional view just above it, which represents the actual mechanism. The line abc of the diagram represents the center line of the arm G. When the axis of the sprocket E is shifted along a straight line, as indicated on the diagram at cc_1 the axis of roller H moves along a perpendicular straight line aa_1. In conjunction with these two straight-line movements, the axis of driven gear B (represented at b on the diagram) describes an arc bb_1 of 90 degrees. As this arc is concentric with the axis of driving gear A, driven gear B continues to mesh properly with A during the straight-line movement of sprocket E, which is the requirement.

From what has preceded it will be evident that, in designing this mechanism, the problem is to so proportion the angular arm abc that when c and a move along straight lines at right angles, point b will follow a circular arc having a radius R equal to one-half the difference between the pitch diameters of gears A and B. To obtain this circular movement of point b, the design must be according to the following requirements:

Points a and c must be equidistant from point b.

The angle between arms ab and bc must be 120 degrees. If x equals the length of the straight-line movement, and y equals the dimension ab or bc, then,

$$y = \frac{x}{\cos 30^\circ\ (2\cos 30^\circ - 1)} = 1.57x$$

$$\text{Radius } R = \frac{x}{2\cos 30^\circ - 1} \times \left(\frac{1}{\cos 30^\circ} - 1\right) = 0.21x$$

The movement of sprocket E is effected by a hand-lever

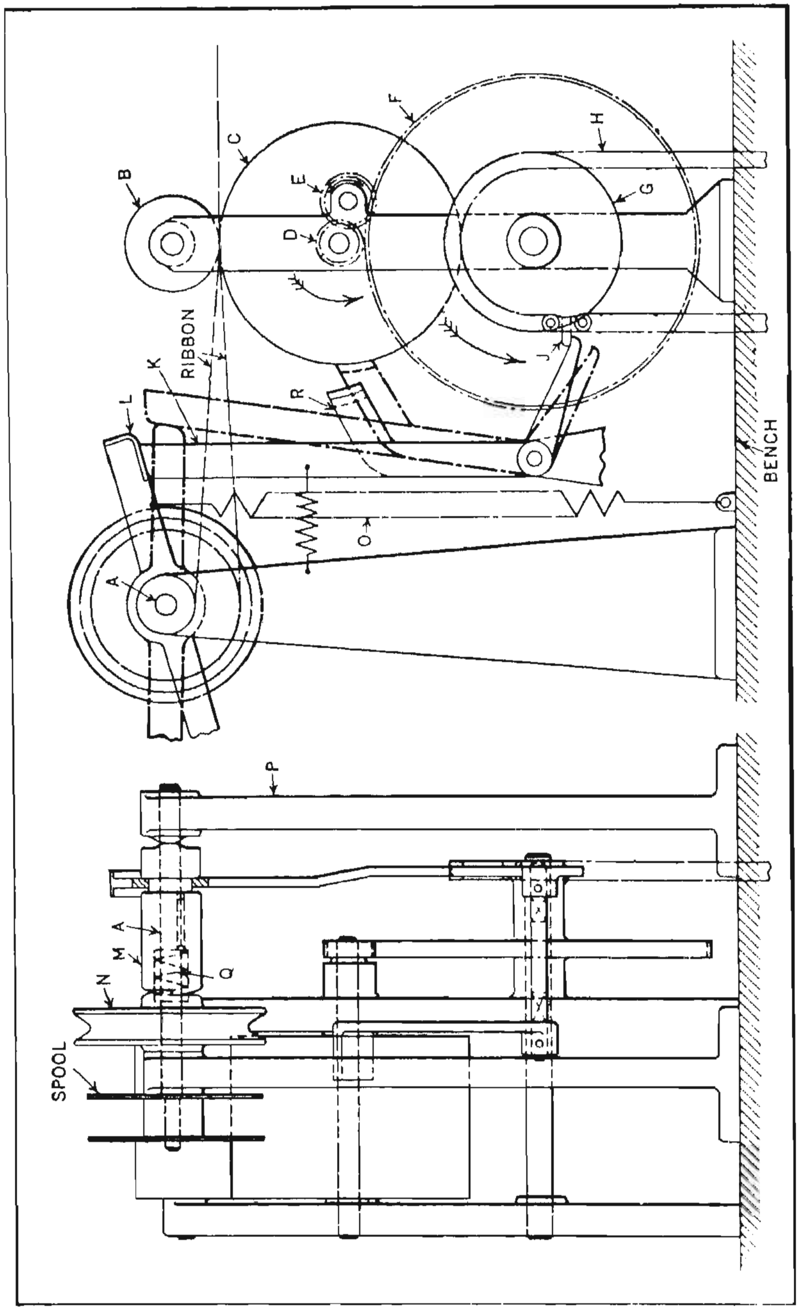


Fig. 9. Mechanism for Winding a Predetermined Length of Typewriter Ribbon on a Spool

connected with link F. This adjustment in the position of the sprocket is only used when an improperly made splice in the film requires what is known as "framing."

Intermittent Rotation for Measuring Typewriter Ribbon as it is Wound on Spool.—In winding typewriter ribbon on spools, a device like the one shown in Fig. 9 is used for stopping the rotation of the spool when a predetermined length of ribbon is wound up on it, at which time the ribbon is cut off. The spool is slipped on the end of the power-driven shaft A, and the ribbon is drawn from between the two rolls B and C. A coil spring (not shown), acting upon the upper roll, serves to keep a constant pressure of the rolls on the ribbon, so that as the ribbon is wound on the spool, both rolls are rotated.

Roll C, through the medium of the gears D, E, and F, rotates the chain sprocket G, over which the chain H is hung. Protruding from one of the links in this chain is the pin J which, through the levers K and L, disengages the clutch M from the driving pulley N and thus discontinues the rotation of the spool. The length of the ribbon wound on this spool depends upon the circumference of the roll C, the ratio of the gears, the number of teeth in the sprocket, and the number of links in the chain.

For every cycle of this chain, the pin depresses the lower end of the lever K, and in doing so, forces the upper end of the lever toward the right, allowing the hand-lever L to swing in a clockwise direction under the pull of the spring O. This hand-lever is secured to the clutch member M, at the right-hand end of which is a cam-shaped projection engaging a similar projection on the stationary hub of the bracket P. As the end of lever L moves downward, the clutch member is oscillated on the shaft A and the cam projections are disengaged, allowing the coil spring Q to force the clutch member to the right and disengage its teeth from those of the driving pulley, thus discontinuing the rotation of the shaft and the spool.

To prevent further rotation of the rolls (due to inertia) after the clutch has been disengaged, the brake arm *R* is provided. This arm is attached to the pivot shaft of the lever *K*, so that just as soon as this lever has been tripped, or immediately after the clutch has been disengaged, the end of the hand-lever *L* swings downward and wedges against the edge of lever *K*, forcing the brake-shoe against the roll *C*.

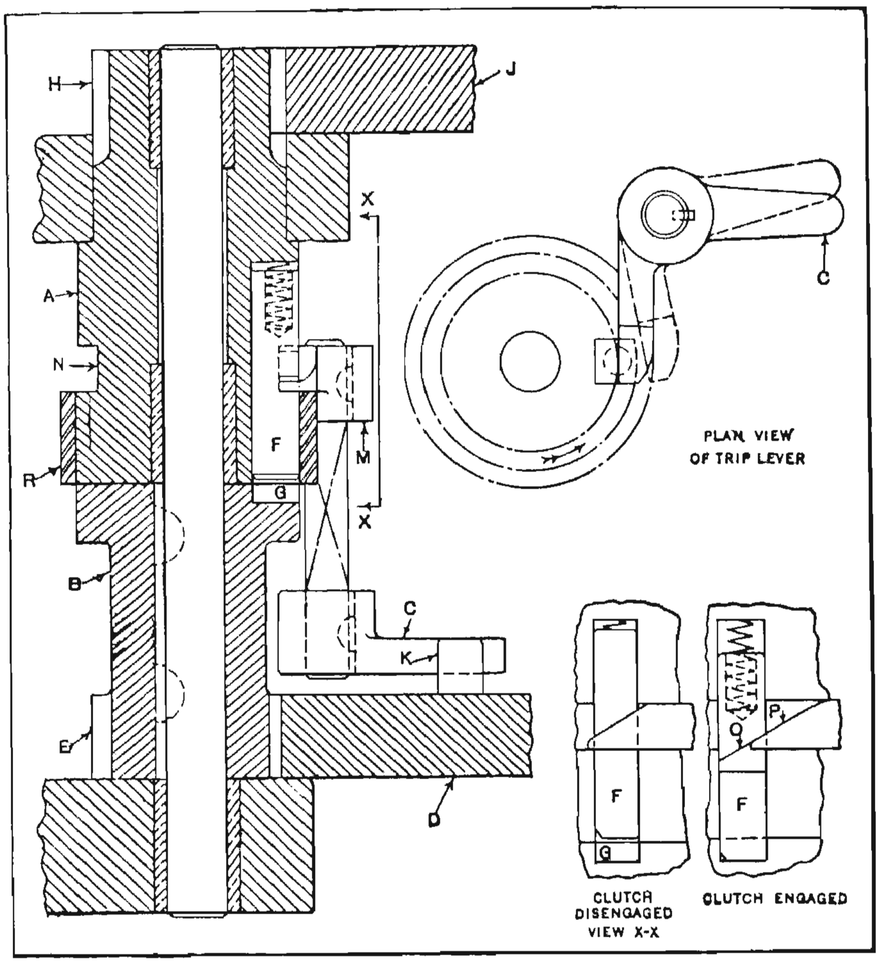


Fig. 10. This Clutch is Operated Intermittently through the Action of a Trip-lever

In the design shown, roll *C* has a circumference of 12 inches, and the ratio of the gears and sprockets is such that when the roll rotates once, the linear movement of the chain is equal to the pitch of the chain; hence, the number of links in the chain corresponds to the number of feet of ribbon upon the spool. Although designed primarily for winding typewriter ribbon, this device could doubtless be used successfully for other applications.

Intermittent Movement from Continuous Rotary Motion.—A mechanism for transforming continuous rotary motion into intermittent rotary motion is shown in Fig. 10. A movement such as this is often applied to indexing plates or tables of multi-stage drilling or chucking machines. The mechanism consists chiefly of two clutch members *A* and *B*, which are automatically disengaged at uniform intervals by means of a key actuated by the trip-lever *C*.

The driving gear *D* is rotated uniformly, receiving its motion from some other member of the machine. This gear meshes with the pinion *E* on the lower clutch member *B*. In the upper clutch member *A* is a sliding key *F*, which is backed up by a coil spring. This key is forced by the spring into the slot *G* when the lower clutch member is rotated into a position where the key and slot are in alignment. When this engagement occurs, both members of the clutch are locked together. The ring *R*, which is shrunk on the lower part of the clutch member *A*, serves to retain the key in its slot.

Pinion *H* is integral with the upper part of the clutch member *A* and meshes with the gear *J*. This gear, in turn, serves to drive an indexing plate (not shown). The clutch is engaged through the action of the pin *K* in the driving gear *D*; and at a certain point in the rotation of this gear, the pin trips the lever *C* so that the lever and the dog *M* assume the position indicated by the dotted lines in the detail plan view at the right. The shaft on which the lever and dog are keyed rotates in a stationary bearing secured to the machine. As soon as the point of the dog is lifted out of the wedge-shaped slot in the key, the latter is free to drop down into the slot G, provided the two members of the clutch are located in the proper position radially. When the key enters the slot G, the gear J rotates. In the meantime, the pin K has passed the lever C, and the dog M is returned to its original position in the annular groove N by a spring (not shown).

When the clutch has rotated nearly a complete revolution, a bevel face O on the key F (see detail in lower right-hand corner) comes into contact with a bevel P on the dog; and as the clutch continues to rotate, the key is forced upward and out of the slot G, as shown in the detail view.

This mechanism will operate satisfactorily at speeds up to 50 revolutions per minute, but at higher speeds, the key is not given sufficient time to drop into the slot G. If accurate indexing is required, the usual plunger arrangement for the indexing plate is used in conjunction with the mechanism described. A mechanism of this type lends itself very well to jobs where it is necessary to vary the indexing ratios. To obtain the various ratios, the gear H may be made demountable with respect to clutch member A. In this way, gears H and J can be changed to suit the required indexing ratio.

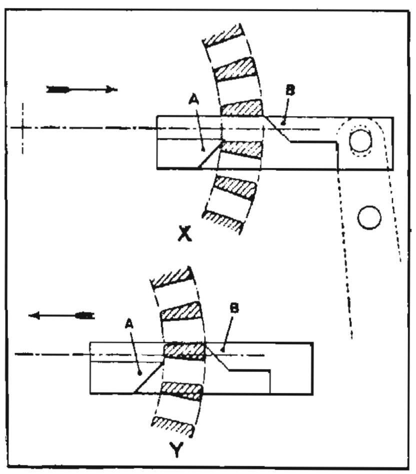


Fig. 11. Simple Indexing Mechanism which is Operated Rapidly by One Lever

Escapement Type of Indexing Mechanism.— A simple indexing mechanism consisting of a rotating ring having a number of radially milled slots and sliding indexing

fingers or cams is shown in Fig. 11. The device is operated by means of a hand-lever, which is indicated by dotted lines. At X, the finger B has just left a slot, and the inclined face

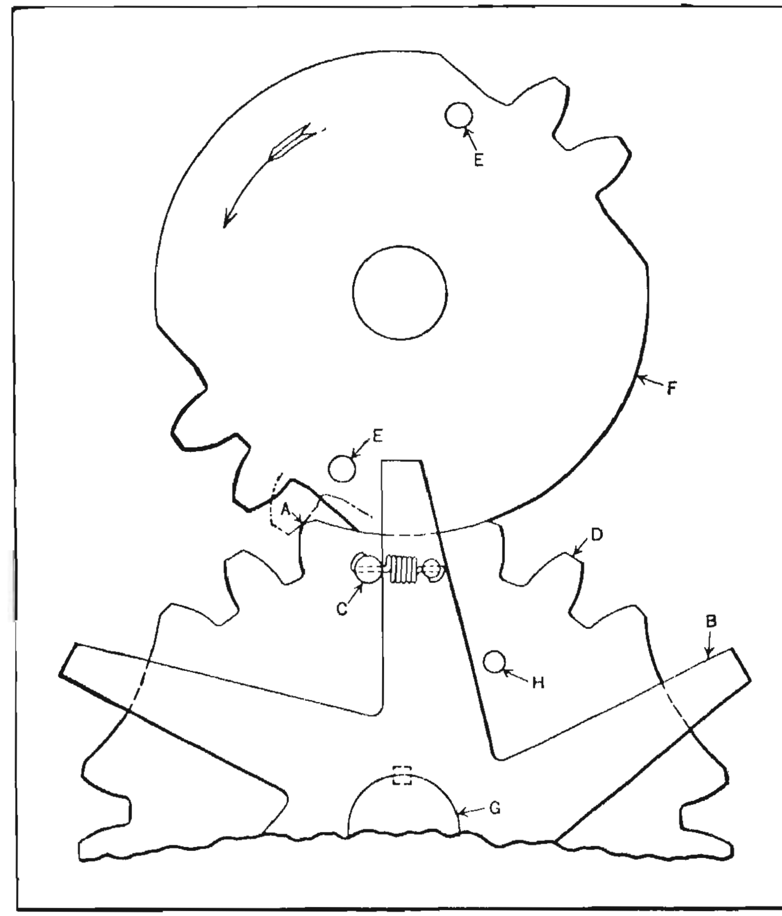


Fig. 12. High-speed Intermittent Gearing with Arrangement for Reducing Tooth Impact

of the finger A is engaging a corner of the same slot. As the finger continues toward the right and enters the slot, the ring is moved through a little more than half a division. The movement of the finger is now reversed and the finger B engages the corner of the next slot, as seen at Y. As the movement continues toward the left, the finger B enters the slot and pushes the ring around to its correct indexing position.

Shock Absorber for High-Speed Intermittent Gearing.—One of the greatest objections to the intermittent type of gearing when used for transmitting high-speed movements is the impact of the mating gear teeth at the beginning of each intermittent movement. This action is due, of course, to the inertia of the driven gear and the offset position at which tooth contact takes place, as indicated at A in the illustration.

The greater part of the wear, tooth breakage, and noisy operation resulting from the tooth impact is prevented by means of the arrangement shown in Fig. 12. Here a steel spider B having as many arms as there are dwelling positions in the driven gear D is mounted on the shaft G. This spider, although free to rotate on the gear-shaft, is held normally by a coil spring against pin C in gear D. The movement of the spider on the shaft is limited by pin H.

Just before the tooth contact at A occurs, one of the pins E in gear F forces the top arm of the spider toward the right, causing the spring to exert a pull on pin C and start gear D gradually. Thus the inertia of gear D is overcome before the contact at A occurs; hence the force of the impact at the point of engagement of the teeth is greatly reduced.

Auxiliary Friction-Driven Gear to Reduce Starting Shock of Intermittent Gearing.—In the operation of intermittent gear trains the impact of the teeth at the beginning of each movement may not be serious at lower speeds, but for higher speeds, the operation of the mechanism is likely to be noisy and the leading teeth are either soon battered out of shape or broken. To overcome this condition in an intermittent gear train operating an automatic hopper, a second set of gears was incorporated, as shown in

Fig. 13. These gears A and B serve to start the driven shaft G rotating with very little shock just before the leading teeth in the intermittent gears come into contact. Another advantage is that the starting torque is borne by a number of teeth in gears A and B instead of by two teeth only, as in the usual type of intermittent gear train, thus reducing tooth wear.

The intermittent gears, which are keyed to their respective shafts, are indicated at D and C. The second set of gears is also mounted on these shafts. Gear A, however, is free to turn on its shaft, while gear B is keyed to the lower shaft (not shown). Both the gears A and B have teeth all around their circumference, the tooth pitch and pitch diameters being the same as in the corresponding gears D and C. It will be noted that gear A is confined between friction washers, which tend to transmit a turning movement to the driven shaft. The pressure of the washers against the gear can be varied by adjusting the lock-nuts, which changes the tension of the coil spring. With this arrangement, the pitch-line speed of gear A and of gear D (when in motion) are the same.

In operation, the driving gears C and B rotate in the direction of the arrow. With the gears in the position shown, it is obvious that unless special provision is made, the entire force of impact in starting the indexing movement will be at point H. In the present design, however, part of the force is divided between several teeth in gears A and B. As soon as point F has passed point E, gear D begins to rotate, through the action of the friction drive, before the leading teeth in gears D and C come into contact. This rotation is started with practically no shock, and continues until the teeth of both intermittent gears are properly meshed.

Some experimenting may be required before the check-nuts are adjusted so that the tension on the coil spring is sufficient to balance the normal load imposed on the mech-

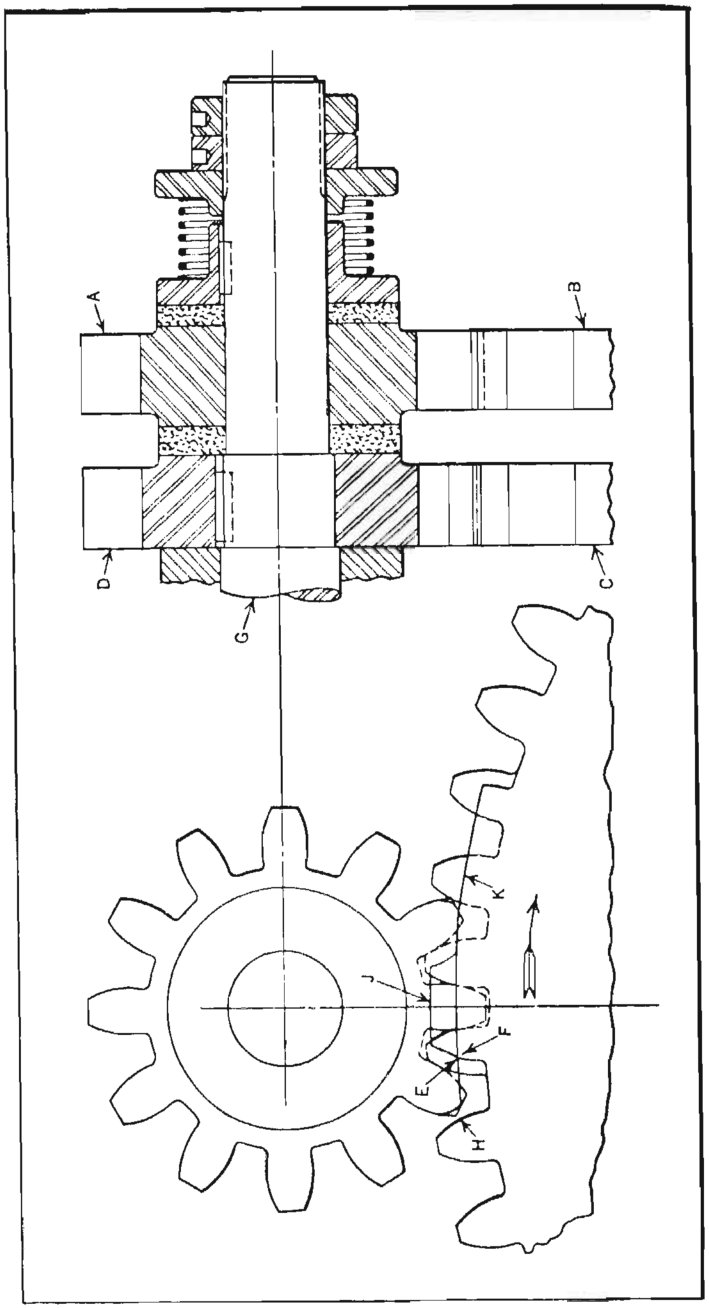


Fig. 13. Intermittent Gear Train in which Impact Shocks are Reduced by a Friction Gear Drive that Starts Each Indexing Movement

anism. The clearance J in gear D also deserves some mention. By removing the metal at this point, a longer dwelling surface K is obtained, thus reducing the time, at the beginning and end of each dwell, in which the gears are in their unlocked positions. It should be understood that this mechanism is suitable for light loads only. If the load is too great, the wear on the dwelling surfaces of the intermittent gears will be excessive due to the torque produced by the friction drive during each dwell. Rapid wear, however, can be prevented by the use of hardened inserts in the dwelling surfaces.

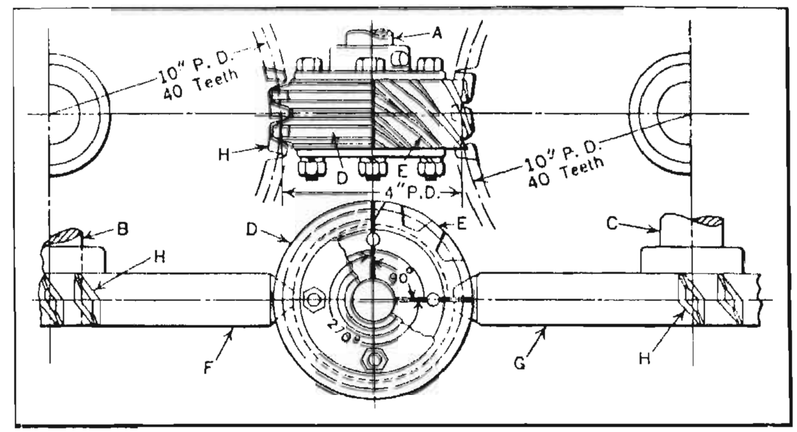


Fig. 14. Gear Drive with Special Gears Designed to Have Shaft A Drive Shafts B and C Intermittently

Gear Drive for Imparting Intermittent Motion Alternately to Parallel Shafts.—In designing a transformer tap changer, provision had to be made for alternately moving the arms of two tap adjusters with a dwell between each movement. Also the arms were required to be locked between movements. It was desirable to have the driving shaft at right angles to the shafts that operated the tap adjuster arms. The speed reduction was required to be approximately 1 to 12.

These conditions were fulfilled by the mechanism shown

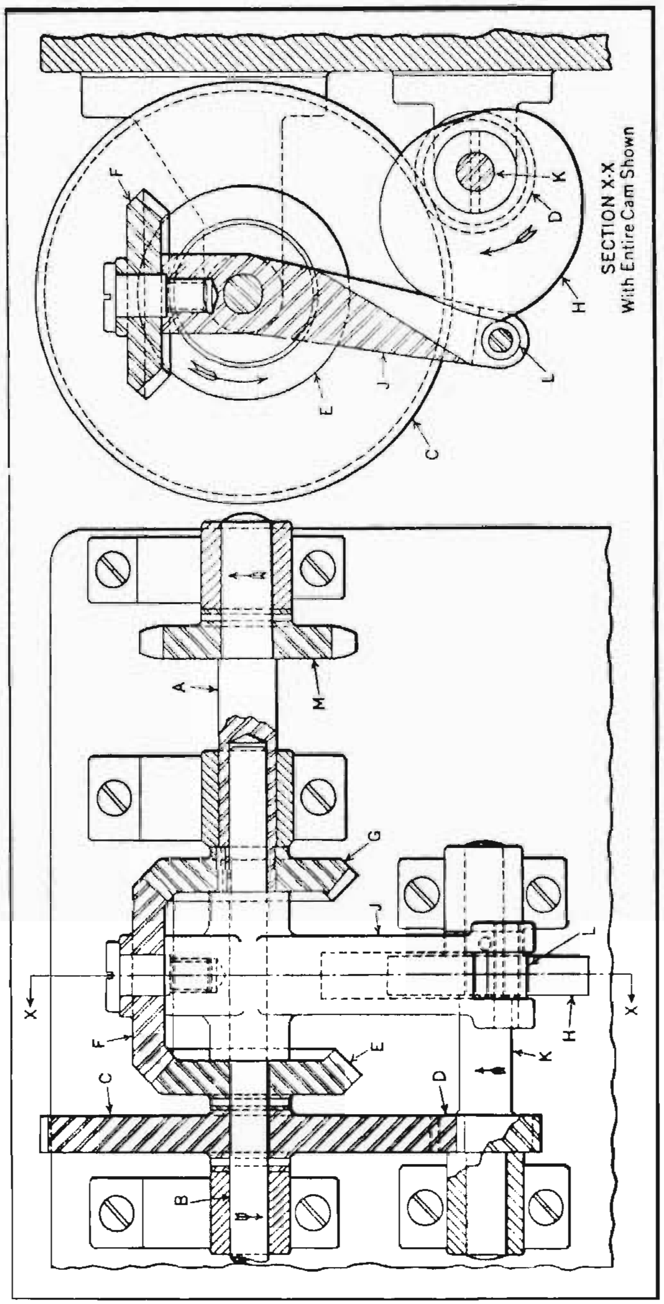


Fig. 15. Sprocket Drive for Conveyor with Mechanism that Causes Conveyor to Dwell at Operation Stations without Starting or Stopping Shock

in Fig. 14. The gear drive consists of a combination worm on the driving shaft *A* which meshes with two gears on the shafts *B* and *C* connected to the tap adjusters. The worm is built of two parts *D* and *E*. Part *D* has tooth spaces that appear simply like annular grooves. This part comprises a segment of 270 degrees. The other part *E* has a helix angle of 53.1 degrees. These two parts have grooves in their sides into which annular ribs on the side plates fit when the four members are bolted together as shown.

The two gears *F* and *G* are alternately in mesh with both parts *D* and *E* of the worm. This is accomplished by making gears *F* and *G* with teeth, as indicated at *H*, which will mesh with the teeth in both section *D* and section *E* of the driving worm. When the teeth in *F* and *G* are in mesh with the teeth in section *D*, no motion is transmitted from the driving to the driven shaft. One rotation of segment *E* past *F* or *G* serves to rotate either one of these gears through an angle of 27 degrees. The gears *F* and *G* each have 40 teeth. The section *E* is a 90-degree segment of a 12-thread worm. With this arrangement, gear *F* and then gear *G* will be turned through an angle of 27 degrees. There is a stop or dwell between each movement corresponding to three-fourths revolution of the driving shaft. Both pinions are locked between their respective rotational movements.

Combination Cam and Differential Gear Movement for Chain Conveyor.—Sprocket chain conveyors are used extensively for conveying containers through filling machines, and frequently the drive is arranged so that the chain dwells at regular intervals to permit the filling of the containers. One rather interesting drive for obtaining this intermittent conveyor movement is shown in Fig. 15. Its design embodies a cam which transmits a rocking movement to a differential planet gear for controlling the rotation of the driving sprocket of the conveyor chain. This mechanism has its application in a machine for filling glass vials with liquid. To prevent the spilling of the liquid from the vials as they pass along on the conveyor, provision is made to eliminate shock in stopping and starting the chain.

The intermittent movement is transmitted to the sprocket shaft *A* from the constantly rotating drive shaft *B* through the spur gears *C* and *D*, miter gears *E*, *F*, and *G*, and also through the cam *H*. Gears *C* and *E* are pinned to the drive shaft, the end of which turns freely in the end of the driven shaft *A*. On this shaft is keyed gear *G* which meshes with gear *F*. Gear *F* is mounted on the arm *J*, which is free to turn on shaft *B*. The outer end of arm *J* carries a follower roll *L* which engages the cam *H*, the latter being pinned to the pinion shaft *K*. In order to synchronize the conveyor movement with that of the rest of the machine, each intermittent cycle of the conveyor chain must occur during one-quarter revolution of the drive shaft *B*.

There are four vial stations to each length of conveyor chain equivalent to the pitch circumference of the sprocket; hence, in order to cause the chain to dwell as each station passes the filling valve, the sprocket *M* must dwell after each quarter revolution. It was found by experiment that a vial could be filled in the same time that it takes shaft *B* to rotate one-eighth revolution. Thus, having determined the angular movement of this shaft during the dwell period, it remains to proportion the gears and cam to impart the required rocking motion to arm *J* for obtaining this dwell; that is, to cause gear *F* to roll on gear *G* without rotating the latter and the sprocket.

Assuming that arm *J* is stationary, one-eighth revolution of gear *E* in the direction of the arrow would rotate gear *G* the same amount in the opposite direction. Now suppose that during this one-eighth revolution of gear *E*, arm *J* were rotated one-sixteenth revolution in the same direction. Then gear *F* would merely roll on gear *G* and the latter would remain stationary. Since we know the movement of arm *J* required to cause gear *G* and sprocket *M* to dwell during one-eighth revolution of the drive shaft, the contour of the cam can be developed. The throw of the cam will, of course, correspond to the angular movement of the arm. One complete cycle of the cam is required for each one-quarter revolution of the drive shaft. Therefore, the ratio of gears *C* and *D* must be 4 to 1.

Thus, while the drive shaft *B* rotates one-eighth revolution from the position shown, the cam will rotate one-half revolution and gear *F* will roll on gear *G*, causing the latter and the sprocket to dwell. During the next one-eighth revolution of shaft *B*, however, the cam will complete its revolution, swinging the arm in the opposite direction and causing gear *F* to rotate gear *G* one-quarter revolution, or twice the amount it would rotate if arm *J* were stationary. In this way, it will be seen that shafts *B* and *A* have the same angular movement for each station movement, although shaft *A* rotates at a higher velocity, owing to lost motion resulting from its dwell.

By observing the contour of the cam, it will be noted that it is developed to impart a constant rise for the first half revolution. This constant rise is important if a steady dwell is to be obtained. For the remaining half of the cam, the contour is such that the beginning of the upward movement of the arm is accelerated and then retarded at the top. This accelerating and retarding of the arm, when transmitted through the gears, results in a corresponding movement being imparted to the conveyor chain, the shock to the chain being so slight that spilling of the liquid in the vials does not occur. The working torque transmitted through the gears is sufficient to maintain engagement of the follower roll on the cam.

Parallel Slides with Latch and Cams for Operating One Slide Intermittently.—In connection with a certain extrusion process, it was found necessary to withdraw two sliding members of a stripping mechanism up to a predetermined point, after which one slide had to remain stationary while the other completed its full travel. On the return stroke, the stationary slide had to be "picked up" and carried along with the other slide. Fig. 16 shows how this is accomplished by the use of a swinging latch, the principle of which might well be applied to other devices where one of a pair of slides must have a temporary dwell. This latch operates between two flat profile cams with oppositely disposed notches shaped to receive the rollers on the latch.

The upper cam A is secured to the moving platen or slide

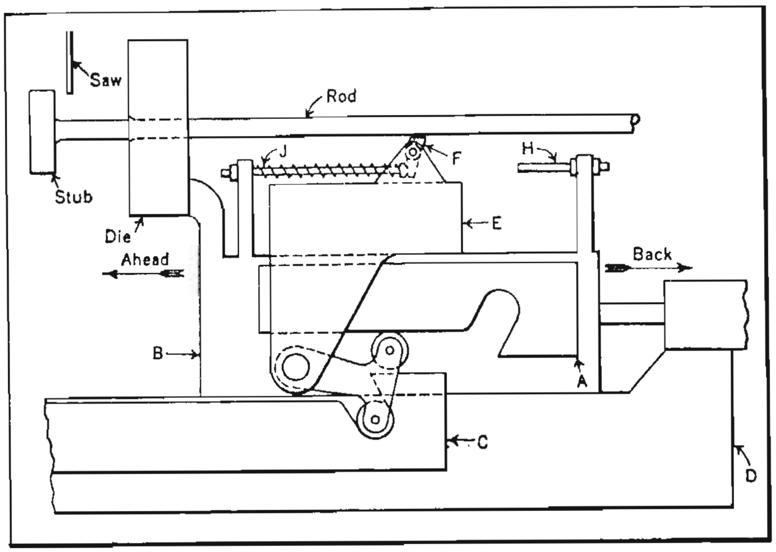


Fig. 16. Two Slides, One of which is Operated Intermittently by a Latch Actuated by Opposing Cams

B, while the lower cam C is fixed to the bedplate D. The carriage E has a limited range of sliding movement on the slide B equal to the longitudinal distance between the two cam notches when the platen is in its "back" position. At both ends of the carriage travel, adjustable trip-rods J and H engage and disengage pawls F, respectively, these pawls gripping the extruded rod.

As soon as the "stub" has been severed from the rod by the saw, the slide B is started ahead. At this time the rod is held stationary by the pawls F on the carriage, which is

now anchored to the bedplate; as the die is attached to the moving slide, the effect is to strip the die from the rod. When the two notches in the cams come opposite each other, the latch swings out of the lower notch and into the upper one, so that the carriage is picked up and carried along with slide B. Just before the latch swings upward, at which time the rod is clear of the die, the rear trip-rod H releases the pawls, leaving the rod free to be removed.

Intermittent Movement of Reciprocating Slide.—Many ingenious slide movements are to be found in the various types of wire-forming machines. One intermittent movement, applicable to these machines, is shown in Figs. 17 and 18. In this design, the two adjacent slides A and B are actuated by the connecting-rod C. Slide A is connected directly to this rod and is given a continuous reciprocating movement. Slide B operates intermittently. For each cycle of the mechanism, slide B moves with slide A for one working and one return stroke, dwelling for three succeeding working and return strokes.

Both slides operate in the stationary guideway D. On slide A is mounted a locking device consisting of housing E, locking plunger F (Fig. 18) which engages bushing G, and cam H with its indexing pins J. This device is actuated by the spring pawl K, which slides in a boss on the guideway.

In the position shown, the slides are locked together by the plunger F; consequently, both slides are moving together. They have just completed their working stroke and are about to return in the direction indicated by the arrow (Fig. 17). On the return stroke, pawl K engages one of the pins J and rotates the cam 90 degrees, causing the projection L (Fig. 18) to slide upward along the deep notch in the housing E and drop into one of the three shallow notches M. This results in the plunger being withdrawn from bushing G in the lower slide just before the return stroke is completed. Hence, some means must be provided

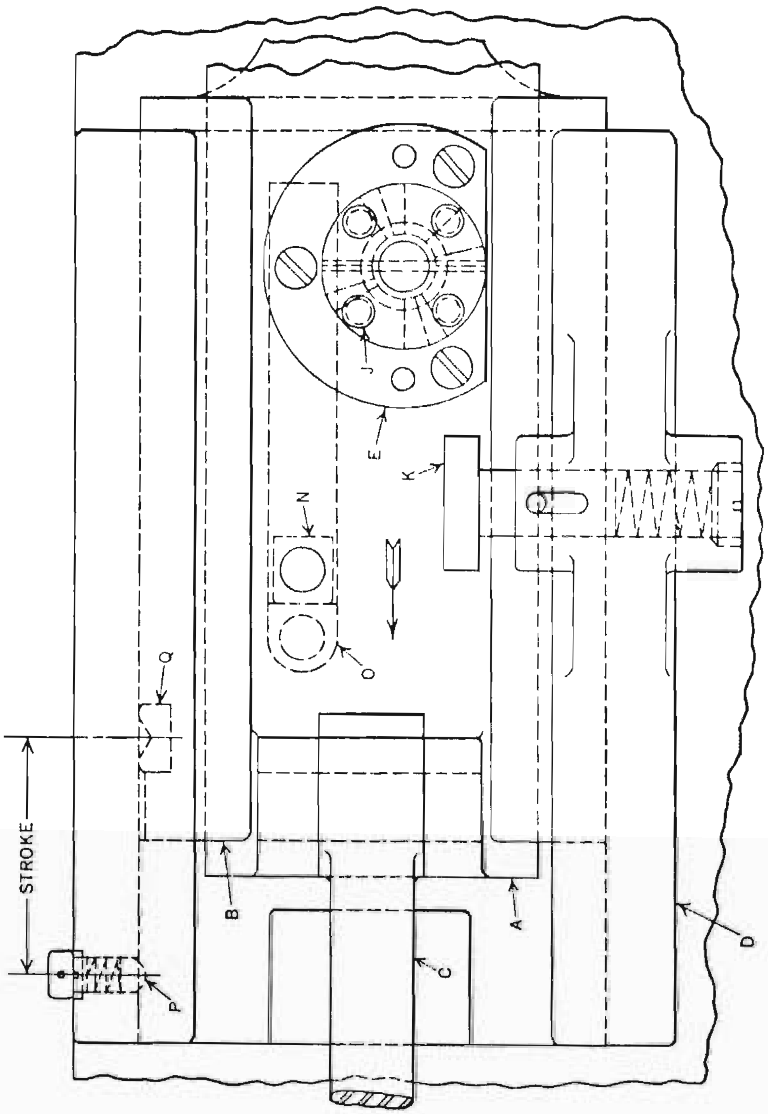


Fig. 17. Plan View of Double-slide Movement Shown in Fig. 18

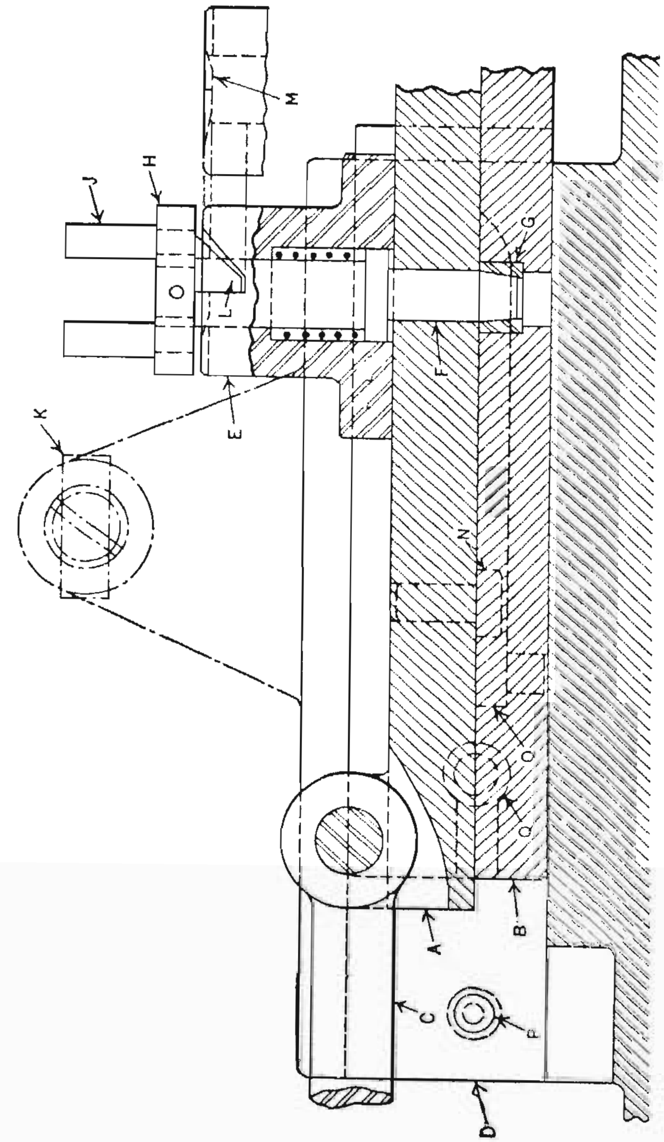


Fig. 18. Double-slide Movement with an Arrangement that Causes One Slide to Dwell During a Predetermined Number of Strokes of the Other Slide

for completing the return movement of slide *B*. Stops *N* and *O* serve this purpose. As these stops are in contact with each other at this time, the upper stop resumes pushing the lower slide to the end of its return stroke. At this point, spring button *P* engages the depression in the pad *Q* and prevents slide *B* from moving toward the right (during its dwell), due to frictional contact with slide *A*.

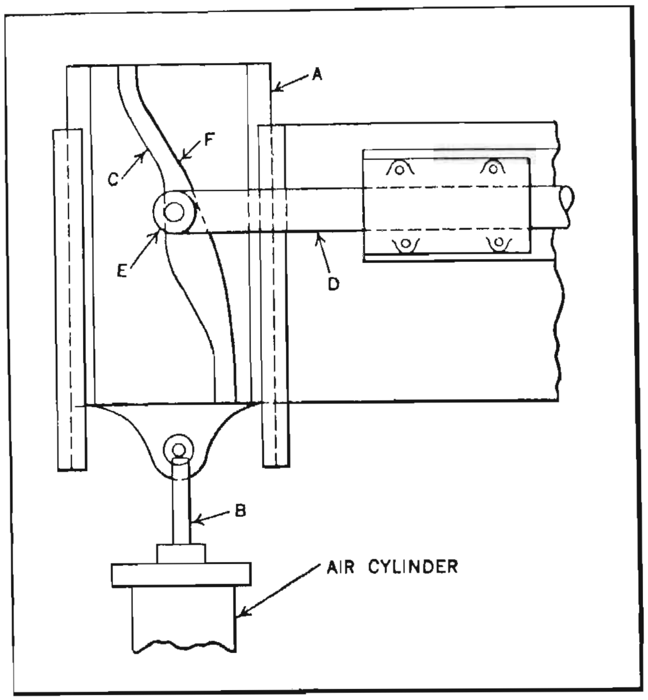


Fig. 19. Mechanism for Converting a Constant Reciprocating Movement into an Intermittent Movement

For each of the two succeeding working and return strokes of slide *A*, cam *H* is indexed 90 degrees, as previously described; but as the projection *L* enters a shallow notch for all three indexing movements, slide *B* remains stationary for three working and three return strokes of slide *A*. On the last return stroke, however, cam *H* is

again indexed. This time the projection enters the deep notch, allowing plunger *F* to drop down and enter bushing *G*. At the end of this return stroke the cycle is completed, and on the succeeding working and return strokes both slides travel together. The object of the shallow notches in housing *E* is to prevent the cam from reversing its movement after being indexed due to the back drag on the pins *J* when they leave the pawl.

Intermittent Movements from a Constant Reciprocating Movement.—A feeding mechanism operated by an air cylinder was required to convert the constant reciprocating movement of the air piston into an intermittent movement on the outward stroke. This movement was to be at right angles to that of the piston. On the return stroke, the motion was to be continuous and at a constant speed. The mechanism for obtaining these movements is shown diagrammatically in Fig. 19. The reciprocating piston *B* is attached to the slide *A*. Slide *A* has a cam groove with the side *C* formed with a dwell to impart the required intermittent movement to the feeding plunger *D* on the outward movement of the piston.

On the return movement, the side *F* of the cam groove returns the plunger *D* to the starting position without the intermittent motion required on the outward movement. There is sufficient friction in the mechanism to keep the roller *E* of the plunger *D* in contact with the sides *C* and *F* on the outward and inward movements, respectively. Automatically operated air valves control the dwell at the end of each stroke. The speed of the feeding and return movements of plunger *D* is governed by the rate at which air is admitted to the cylinders by the air valves.

Adjustable Clock-Controlled Intermittent Mechanism.—The mechanism shown in Fig. 20 is used in a bottle-cap counting machine. It is the function of this mechanism to swing a pivoted chute alternately from one position to another, allowing the chute to remain in each position long

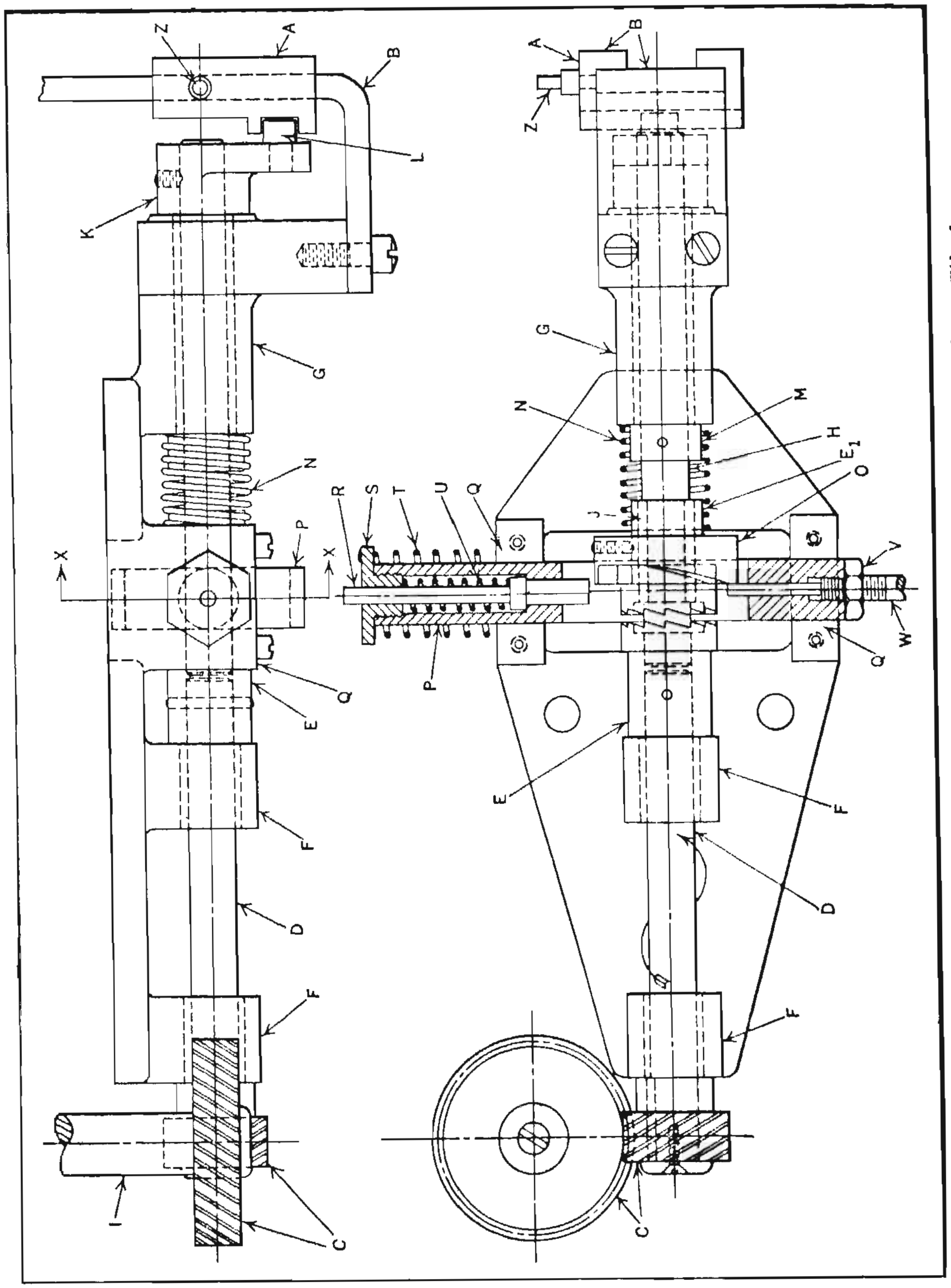


Fig. 20. Clock-controlled Mechanism for Imparting Intermittent Reciprocating Motion to Slide A

enough to permit a packing case to be filled with bottle-cap crowns which are delivered by the chute. Although the movements obtained with the mechanism here illustrated could be duplicated by other mechanical arrangements, none of the available types met the particular requirements of the counting machine. The mechanism here described has proved successful.

The required movements are transmitted to the chute by the slide *A*. The pin *Z* in slide *A* engages a slot on the under side of the pivoted chute. When slide *A* is in the position shown in the upper view, Fig. 20, the chute discharges into one of the packing cases. As soon as the packing case is filled, a clock, having its actuating lever connected to rod *W* of the yoke *P*, releases the latter member, which causes the clutch to engage the driving and driven shafts and then disengage them after the driven shaft has carried the crank-arm *K* around one-half revolution.

The pin *L* of the crank-arm engages a slot in slide *A* and carries the slide to the opposite position, where it remains while the packing case under the chute is being filled. The clock then acts again, and the chute is automatically transferred to the other filling position. This cycle of operations is repeated continuously, the mechanism being driven by the constantly rotating shaft *I* through the helical gears *C* and shaft *D*. With this arrangement, the transfer movement of the chute is accomplished very quickly and smoothly. The timing of the movements and the duration of the rest periods are controlled by the clock, which can be adjusted to meet any operating requirements.

Construction and Operation of the Clock-Controlled Intermittent Mechanism.—Referring now to the construction of the mechanism, the toothed clutch member *E* is fastened to the continuously rotating shaft *D*, mounted in the bearings *F*. Another part of the mechanism is supported in the bearing *G*, and consists of shaft *H*, on which the toothed clutch member E_1 is a sliding fit. Rotation of E_1 on shaft H is prevented, however, by the two small feather keys J, which are fixed in the shaft and are a close sliding fit in the keyways in E_1.

The crank-arm K, previously referred to, is fastened to

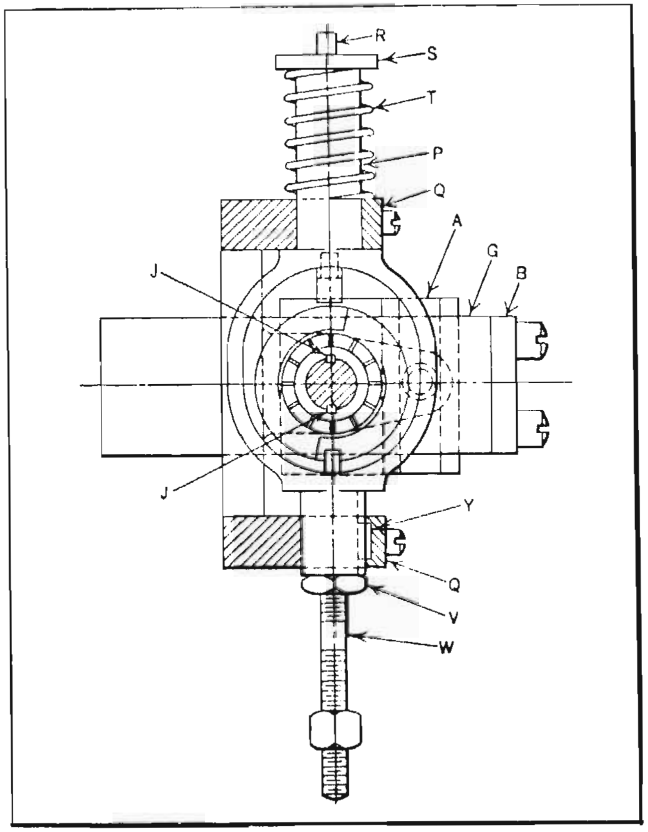


Fig. 21. Section X-X, Fig. 20

the outer end of shaft H. The collar M prevents any lateral movement of the shaft and at the same time serves as a guide for the spring N. When the mechanism is released, the spring N forces the clutch element E_1 into engagement with element E, thus providing for the positive rotation of the crank K through one-half revolution.

The cam O, which causes the slide A to pause at the end of each stroke, or one-half revolution of shaft H, is fastened to the clutch element E_1. The contour of cam O, when rolled out in a flat position, is shown in the views to the right, Fig. 22. It will be noted that there are two gradual

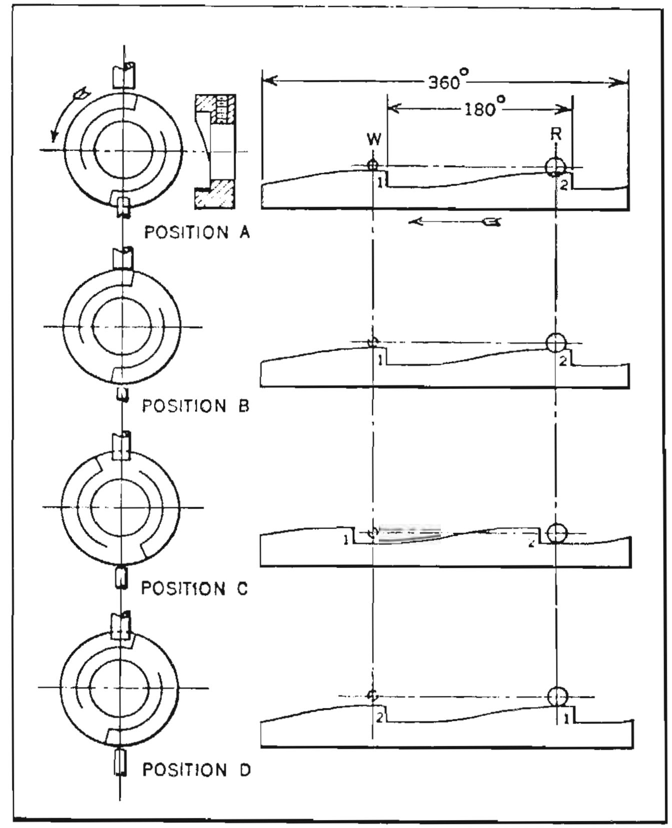


Fig. 22. Form and Operation of Cam O, Fig. 20

rises in the cam surface, after which a sudden drop follows. These drops are just 180 degrees apart.

To disengage the clutch elements, it is only necessary to move the clutch element E_1 a certain distance, depending upon the depth of the clutch teeth plus a reasonable amount of clearance between the teeth. In this case, the depth of the teeth was 1/16 inch. The rise of the cam contour was 7/32 inch, giving a clearance between the teeth of 5/32 inch. This large clearance is necessary, as will be made clear from the following description of the cam followers. The cam, which is machined in the form of a ring, is fastened to the clutch element E_1 by a set-screw, so that these two members are free to move laterally along the shaft H.

The cam follower arrangement is somewhat more elaborate than the average type of follower, and is the most interesting and unique feature of the mechanism. Referring to Figs. 20 and 21, the yoke P is guided in the bearings Q. These bearings are split for the purpose of facilitating the assembling of the mechanism. To prevent the yoke from rotating in its bearing, a small key Y is provided and a keyway is cut in the lower stem of the yoke. This restricts the yoke to a vertical movement.

The lower stem of the yoke is provided with the adjustable pin follower W, which may be locked in place by nut V. The adjustment of this pin is very important, as it determines the amount of movement necessary in the lever mechanism (not shown here), which is attached to the pin follower W. This lever mechanism is connected to the clock that times the movements. The clock-operated mechanism will remove the lower pin and permit the clutch elements to make contact under the action of the spring N. The upper stem of the yoke is counterbored and provided with the plunger pin follower R and the light spring U, which is held in place by the nut S. The spring T keeps the yoke in the upper position with its shoulder against the lower side of the bearing Q.

The function of the two pin followers will be more easily understood by referring to the four diagrams in Fig. 22, which show the main positions of the pin followers with relation to the cam contour. At A is shown the normal position of the pin followers at the starting position. The lower pin is in contact with the cam contour, while the upper pin is free. As soon as yoke P, Fig. 20, is pulled downward, the lower pin is removed and the upper pin strikes the outer edge of the cam ring, as shown in position B, Fig. 22.

It will be noted that the upper pin is larger in diameter than the lower one. The purpose of this feature will become obvious on further consideration of the mechanism. When the upper pin strikes the outer edge of the cam ring, the spring U, Fig. 20, is compressed. However, this condition only exists momentarily, inasmuch as the spring N forces the clutch elements into contact as soon as the lower pin is removed from the cam contour, resulting in the rotation of the cam and all its attached parts.

As soon as rotation begins, the upper pin is freed and drops down under the action of the spring U, so that it makes contact with the cam contour as illustrated in position C, Fig. 22. When the cam has rotated 180 degrees, the clutch elements are separated and the rotation ceases. This last step is illustrated by position D, where the point marked 1 has been replaced by the point marked 2 under the upper pin follower R. In the meantime, the crank has traversed from one end of its stroke to the other and stopped. The lower pin is still out of contact with the cam contour, the upper pin having performed the action of separating the clutch elements. As soon as the yoke is released, it is raised by the spring T, Fig. 20, and at the same moment, the upper pin is removed from the cam contour and replaced by the lower pin. This explains the necessity for having the upper pin slightly larger in diameter than the lower one.

The upper pin causes the cam contour to move, or be set back slightly from the edge of the lower pin follower. This permits the lower pin to rise freely into position opposite the cam contour. The cam and the clutch member E_1 move toward the clutch member E as the upper pin leaves the contour of the cam, but this movement is stopped by the lower pin.

Two very important details should be noted. First, the distance between the ends of the upper and lower pins must be such as to bring the lower pin opposite the cam contour before the upper pin is entirely removed from contact with the cam. If this condition does not exist, the spring *N* will force the clutch elements into contact before the lower pin is in place to hold it back when the upper pin is removed. The second important detail is to note that the rise of the cam contour is determined by the size of the upper pin. The upper pin must obviously be able to fall in with the lower part of the cam contour before it can perform its function.

If the yoke is released before the cam has rotated through 180 degrees, the lower pin itself will perform the function of separating the clutch elements and leave the upper pin inactive as far as contact with the cam is concerned. In this particular case, there was no absolutely definite time release for the yoke, so that a positive operating arrangement had to be provided which would allow a rotary motion of only 180 degrees at each releasing movement of the yoke *P*, regardless of how long the yoke was held in the lower position. This feature accounts for the use of two cam followers instead of one.

It might be of interest to mention here that this mechanism is operated at a speed of about 100 revolutions per minute with no difficulty. However, it might be necessary to provide small depressions in the surface of the cam contour at the points where the followers rest if the speed is much above 200 revolutions per minute. This will prevent overrunning of the cam due to the inertia developed in the rotating parts.

Intermittent Reciprocating Motion Derived from Cam Operated by a Chain.— At times it is necessary to obtain a positive reciprocating motion, followed by a period of dwell, from a moving chain. Such a motion can be imparted by each link of the chain with the device shown in Fig. 23; or by omitting certain cam-rolls, the device can be made to operate only as sections of the chain pass it.

Referring to the illustration, the chain *J* is constructed of flat steel links which are joined together by two lengths of tubing, one within the other, in a way to permit a free turning action at the link joints. Each link is equipped with a spindle *G*, the top end carrying the work-holder (not shown) while at the lower end is mounted a set of three rolls. The smallest roll is a slip fit on the hub of one of the larger ones and acts against the flat cam *A* when the chain is in motion. The two larger rolls come in contact with the bar *C*, thus providing the necessary support for the chain while under the action of the cam. The cam is fastened by screws and dowels to the swinging arm *M*, which is pivoted in the bracket *B* by the pin *N*, held in the bracket by the set-screw *O*. The bracket is secured to the machine table by screws, the supporting bar *C* being mounted on its upper part. There are also two other plain brackets (not shown) to support the extreme ends of this bar. Connected to a projection on arm *M* is the link *P* which carries the reciprocating motion to the required part of the machine.

In operation, the roll *D*, as shown in the side view, is about to force the point *K* of the cam away from the chain, and as the cam is pivoted on pin *N*, the end *L* will move toward the chain between the two rolls *F* and *E*. Upon further movement of the chain, edge *X* will come in contact with the roll *E*. At this time, the center of roll *D* has passed the point *K*, so that as roll *E* forces edge *X* away, point *K* swings toward the chain and between rolls *D* and *F*. The projection *Y* on the cam prevents the point *K* from swinging further than is shown toward the center of the chain. The cam is in action only during a movement of the chain approximately equal to the diameter of the cam-rolls, and as the projection to which link *P* is connected is integral

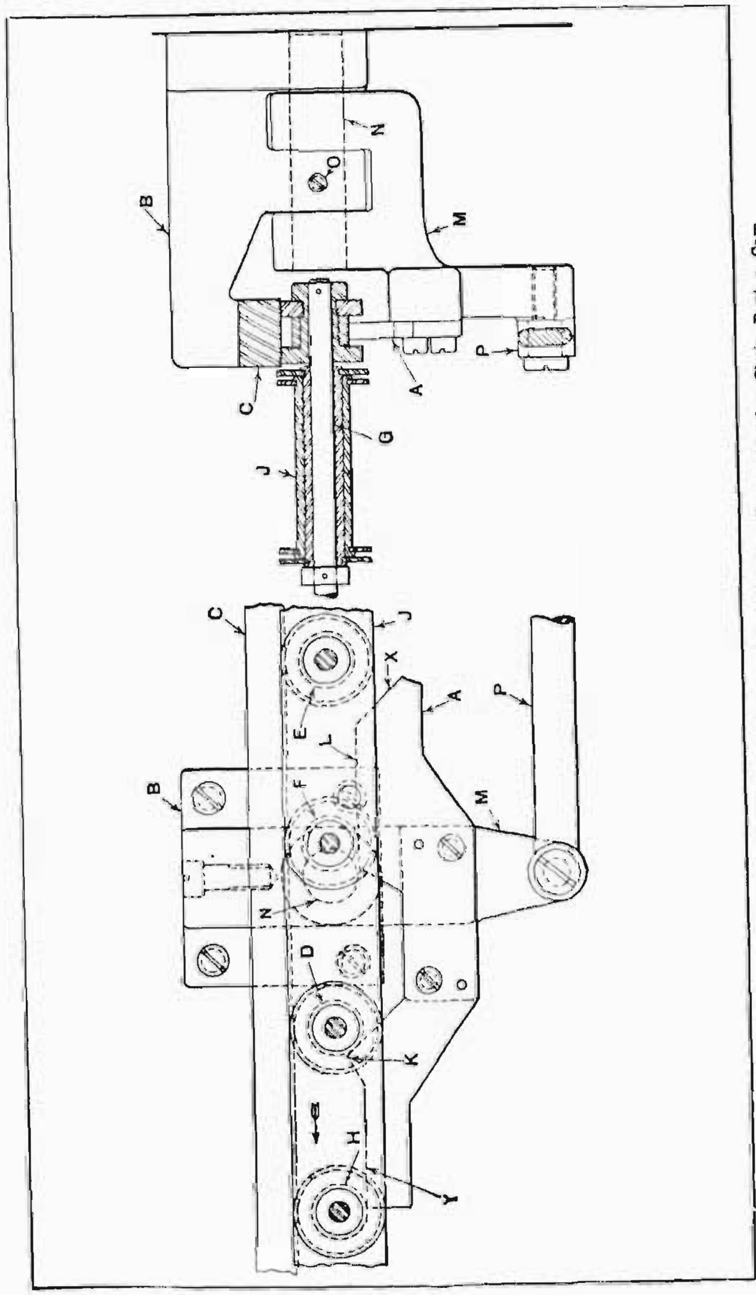


Fig. 23. Intermittent Reciprocating Motion, Produced by the Movement of a Chain Past a Cam

with the cam, the motion of the latter, as described, will produce the required reciprocating movement of the link P.

Intermittent Motion for High Rotary Speeds.—Various forms of intermittent motions have been designed for driving machine parts that must alternately turn through part of a revolution and then dwell or remain stationary between each fractional turning movement. Some mechanisms of this class, however, are not adapted to high speeds owing to excessive shocks each time the driven member is started. The design here illustrated (see Fig. 24), which is similar in principle to those used on motion picture machines, although much larger, operates quietly and smoothly at high speeds.

This particular mechanism is used on a milk-bottle cap-making machine. The driver, which is 24 inches in diameter, makes four revolutions to one of the driven member, which has four equally spaced arms each equipped with a roller, as the illustration shows. The speed of the driver is 960 revolutions per minute, and it requires a minimum movement of 90 degrees to operate the driven member smoothly and quietly at this speed. The action may be extended over a larger angle, thus permitting higher speeds and shortening the idle time.

General Design of High-Speed Intermittent Motion.—The rollers on the driven member engage a large annular track (see end view), and after each quarter turn, the driven shaft is securely locked during the idle period. As the driver turns in the direction of the arrow, it rotates the driven shaft intermittently in the same direction. The surface at G (Fig. 24) of the outer track acts against roller B until roller A enters the groove H. When roller A has fully entered groove H, roller C begins to enter groove J. When point P on the driver passes roller B, roller A is about half way through groove H, and as roller B begins to engage surface K, roller A emerges from track H.

When roller A has reached the position marked A_1,

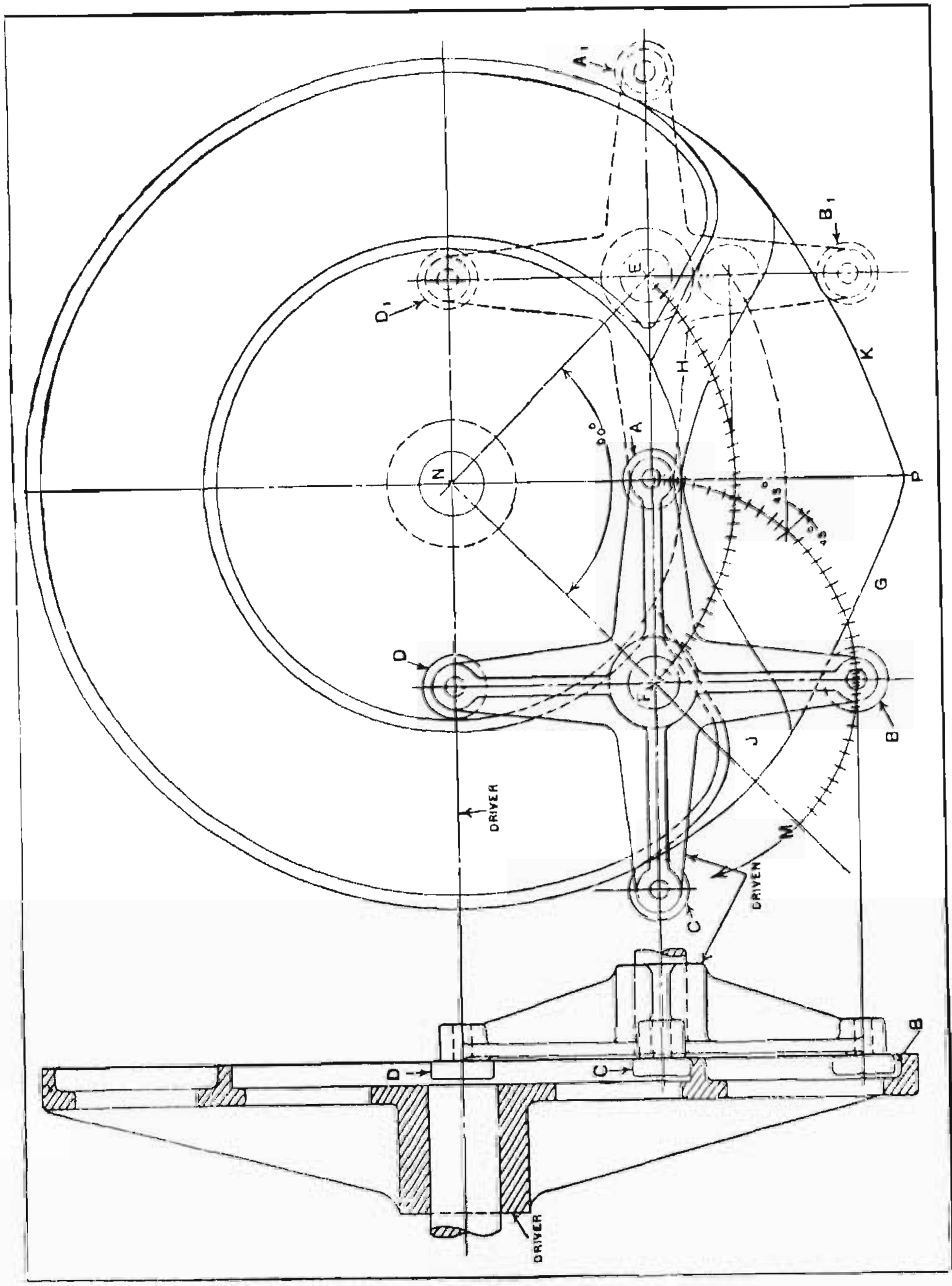


Fig. 34. High-speed Intermittent Motion with Dwell of 270 Degrees or Less

roller D has swung around so that it is again in contact with the inner track as at D_1, roller B is at B_1, and the driven member has turned one-fourth of a revolution. The entrance to groove H now passes roller C, which follows roller D_1 around the inner track. At no time is the driven member free to turn in either direction, except as it is revolved by the tracks or cam grooves, and two or more rollers are always in engagement with the driver.

Laying Out the Cam Curves.—In order to avoid shocks, especially at high speeds, it is necessary to gradually accelerate and then gradually retard the movement of the driven member. The method of developing or laying out the cam curves to obtain this result will now be explained. With N as a center, draw an arc FE through the axis of the driven shaft and divide this arc into thirty-six equal spaces. Next, with F as a center, draw an arc through the axes of rollers A and B and extend this arc 45 degrees to point M. Beginning at the center of roller A, lay off a division of 1 degree, then a division of 2 degrees, followed by one of 3 degrees, 4 degrees, and so on, up to and including 9 degrees. The total number of degrees thus laid off equals 45, since the nine divisions progressively increase from 1 degree up to 9 degrees by increments of 1 degree.

This procedure is now reversed; that is, the divisions begin at the 45-degree point and progressively decrease from 9 degrees down to 1 degree. Beginning at the center of roller B, the order is again reversed, the divisions beginning with 1 degree and increasing up to 9 degrees, ending at M.

Each division from A to M is now bisected; consequently, between the centers of rollers A and B there are now thirty-six divisions, the same as between the centers F and E. Assume that the divisions from F to E are numbered from 1 to 36, and that the divisions from A to B are also numbered from 1 to 36.

From these divisions we shall now proceed to locate various points on the center lines of the cam grooves *J* and *H*. With *N* as a center, draw an arc through division number 1 on arc *AB*, extending it to the right and left of the vertical center line *NP* a short distance. Draw another arc through division number 2 and continue up to division number 36 next to the center of roller *B*. The arcs through these various division points need not be continuous, but they should be located to the right and left of the vertical line *NP* far enough, as near as can be judged, to intersect the center line of the cam grooves.

Now set the compass to the radius of the driven member, or from the center of shaft *F* to the center of one of the rollers. With division number 1 (adjacent to *F*) as a center, draw an arc intersecting arc number 1 struck from center *N* and to the *right* of the vertical line *NP*. Continue until arcs of the radius of the driven member have been struck from each of the thirty-six divisions on *FE*, thus intersecting all of the thirty-six arcs (to the right of *NP*) struck from center *N*.

The thirty-six centers thus located lie on the center line of the cam groove *H*, and various points along the sides of this groove are located by setting a bow pencil or bow pen to the radius of the driven rollers and drawing a series of arcs. The sides of groove *H*, which are tangent to these arcs, can then be drawn.

To lay out the cam groove *J*, again use the compass set to the radius of the driven member, and in striking arcs to intersect those extended to the left of *NP* from *N*, work from *E* to *F*; for example, with division number 36 as a center (adjacent to *E*) intersect arc number 1 struck from center *N* and to the *left* of the line *NP*; continue until finally division number 1 (adjacent to *F*) is used in striking an arc intersecting arc number 36 struck from *N*. In this way, the series of roller centers for groove *J* can be located.

In laying out the curve *GPK* of the outer track of the driver, proceed as follows: With *N* as a center, and from

each of the eighteen divisions on the arc from the center of *B* to *M*, strike eighteen arcs to the right of center *P* and eighteen to the left, all adjacent to the surfaces *K* and *G*; then with the compass set to the radius of the driven member, intersect the thirty-six arcs just struck by another series of thirty-six arcs from the divisions on arc *FE*.

If we assume that the thirty-six arcs adjacent to *G* and *K* are numbered from 1 to 36 from left to right, then division number 1 on *FE* will be used to intersect arc number 1, and so on, until the thirty-sixth division, adjacent to *E*, is used to intersect the thirty-sixth arc at the extreme right of surface *K*. Arcs equal to the roller radius are now struck from these thirty-six centers to locate points along the profile *GPK*.

Although the ratio of this particular mechanism is 4 to 1, other ratios are possible by adding more arms and rollers to the driven member and extending the cam action over a longer arc on the driver. This mechanism has one objectionable feature—the driven shaft must end at the driver and cannot be supported on each side, as will be evident by examining the end view. However, the continuous positive relation between the driving and driven members, the good distribution of wearing surfaces, and the adaptability to high speeds with smooth action compensate for the objectionable feature mentioned.

CHAPTER III

INTERMITTENT MOTIONS FROM RATCHET GEARING

Intermittent motions so designed that a ratchet mechanism constitutes an important element will be found in this chapter. These motions of the ratchet type have been segregated to facilitate finding an intermittent motion likely to meet the requirements of any given design, as, for example, when the application of some form of ratchet gearing appears to offer the best solution.

Ratchet Mechanism for Uniform Intermittent Movement and Heavy Duty.—The ratchet and pawl mechanism illustrated in Fig. 1 was designed for moving, intermittently and accurately, a heavy load at medium speed. The ratchet wheel is positively locked during the idle period, and a positive stop prevents over-travel and insures uniform intermittent movements.

The ratchet wheel *W* is free to turn on the driving shaft, which is shown in section. Behind the ratchet wheel and attached to the driving shaft there is an eccentric *E* connecting with the short arm of bellcrank *B*, which is pivoted at *P*. The operating pawl *O* is pivoted to the long arm of the bellcrank. Pawl *R*, which locks the ratchet wheel during the idle period, is pivoted at *U* and is shown in the locking position. Both pawls *O* and *R* are normally held in engagement with the ratchet wheel by coil spring *C*, which is attached to each pawl. At the upper end of the long arm of the bellcrank there is a steel plate *S* which engages a flat spring *F*, thus lifting the locking pawl *R* to which spring *F* is attached.

This ratchet mechanism will operate with the constant-speed driving shaft turning in either direction in relation

to the ratchet wheel. The eccentric *E* attached to the shaft starts and stops the load gradually like a crank; the full and dotted lines show the extreme positions of bellcrank *B* and pawl *O*. Before pawl *O* comes into engagement with

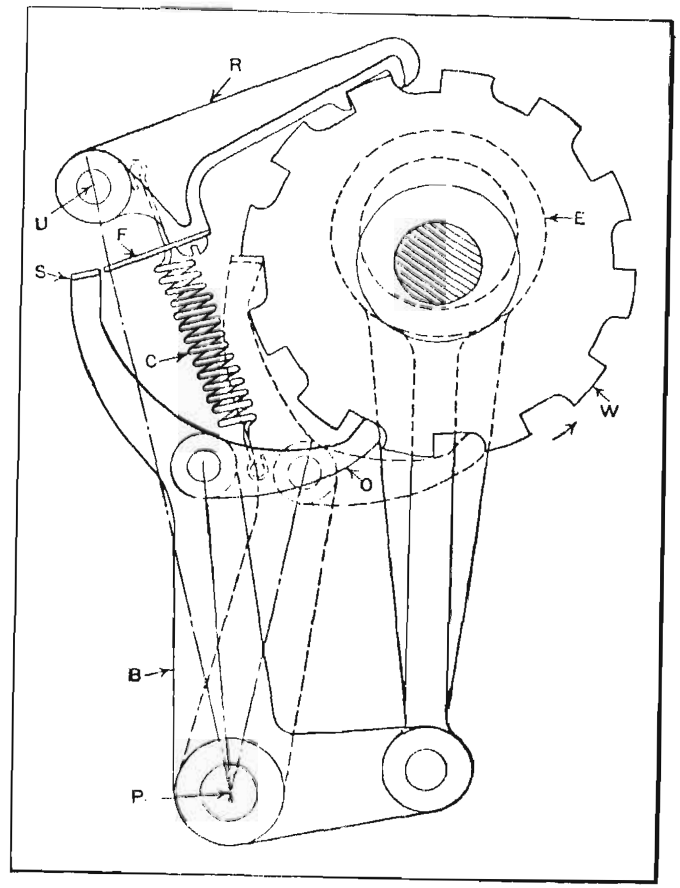


Fig. 1. Ratchet Mechanism Designed to Move a Heavy Load Intermittently and Accurately

a tooth on wheel *W*, plate *S*, by engagement with spring *F*, lifts pawl *R*, thus unlocking the ratchet wheel.

A short movement of plate *S* causes it to pass the center line between pivots *P* and *U*; consequently, it is soon disengaged from spring F, but not until pawl O has moved wheel W about half a tooth, so that when plate S passes spring F, the hook end of pawl R falls on top of the next approaching tooth. Before pawl O reaches the end of its forward movement, plate S enters a space ahead of the radial face of an approaching tooth, so that this tooth face comes into contact with plate S at the end of the stroke, and any over-travel is thus prevented. During the backward movement of plate S to the starting position, it strikes

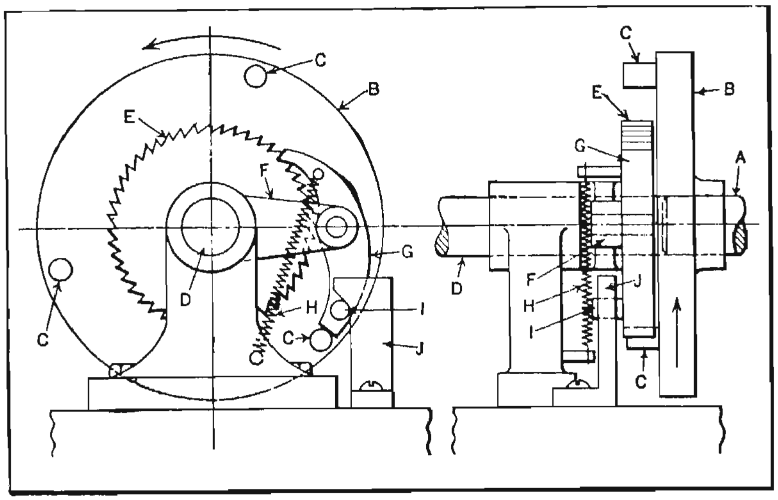


Fig. 2. Intermittent Motion Drive Mechanism Used on Wire-forming Machine

spring F and bends it upward slightly, which insures seating the locking pawl firmly. This ratchet mechanism has a ratio of 12 to 1, there being twelve turns of the driving shaft to one complete turn of wheel W. The connection between the ratchet wheel and the driven member which it operates is through gearing not shown.

Intermittent Rotary Motion from Constantly Rotating Shaft.—The mechanism shown in Fig. 2 is designed to transmit an intermittent rotary motion to the shaft D from the constantly rotating shaft A. This intermittent movement operates the feeding device on a wire-forming machine which requires three partial revolutions of the driven member for each rotation of the driving member. Drive shaft A with the attached disk B rotates continuously in the direction indicated by the arrow. The disk B has three equally spaced pins C on one side. The ratchet wheel E is keyed to shaft D. Lever F carries the pawl G and is free on shaft D. Spring H serves to keep pawl G in contact with the ratchet wheel E and also tends to rotate lever F in a direction opposite to that in which member B is driven.

Pawl G is so shaped that when the actuating end is in contact with ratchet wheel E, the opposite end lies in the path of the pins C. As one of the pins C makes contact with pawl G, the pawl is carried with it, causing ratchet wheel E to rotate. When the pin I on pawl G comes in contact with the cam J, pawl G is disengaged from ratchet wheel E, which then stops moving.

Continued movement of disk B causes the end of pawl G to be further depressed by the action of cam J until the pawl slips under pin C. The two views of the mechanism show pawl G about to be disengaged from ratchet wheel E. As soon as the pawl has discontinued positive contact with pin C, the action of spring H causes pawl G and lever F to rotate in a direction opposite to that of the driving member B, until the upper end of lever F strikes the end of cam J which limits its movement and controls the angular movement of ratchet wheel E. The driving movement is repeated as each pin C comes into contact with the pawl.

High-Speed Ratchet-Feeding Mechanism with Positive Lock.—The positive intermittent indexing or ratchet-feeding mechanism shown in Fig. 3 is designed for operating a paper feed-roll R used in connection with a printing unit of a machine. One end of the roll R and the ratchet wheel Z, with its actuating and locking pawls, are mounted

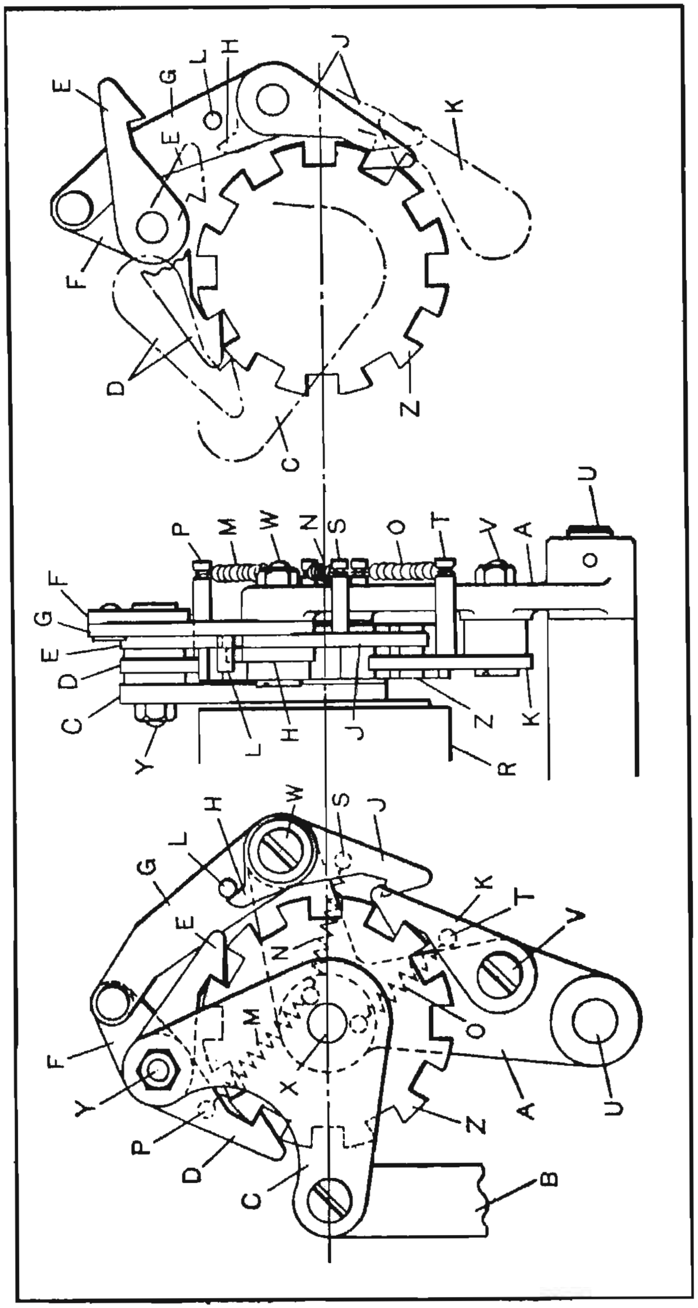


Fig. 3. Positive Ratchet-feeding Mechanism Designed to Operate at High Speed

on arm *A*. Because of the rocking motion of arm *A* about shaft *U*, it is necessary to have roller *R* positively locked against rotation during the return movement of the indexing pawls *D* and *E*. For this reason, the use of a friction feeding device was impracticable.

The indexing mechanism is not affected by the rocking motion, as the ratchet-operating link *B* has a movement corresponding to or parallel with that of arm *A* when it is not being used to actuate the indexing pawls. The indexing pawl *E* and lever *F* are pinned together, and locking pawl *J* is pinned to pawl latch *H*, but this is not shown in the illustration.

To index the roller *R*, link *B* is given a reciprocal motion which rotates the bellcrank *C* clockwise. The indexing pawls *D* and *E* are connected to bellcrank *C* by stud *Y*. Pawl *D*, being held in position by spring *M*, engages ratchet wheel *Z* which is connected to shaft *X*. Pawl *E* is held in position by the connections to levers *F* and *G*.

The locking pawl *J* is spring-connected to lever *G*, and the latching pawl *K* is held in engagement with the ratchet wheel *Z* by the spring *O*. The pawls *J* and *K* are attached to rocker arm *A* by studs *W* and *V*. As the bellcrank *C* rotates, the indexing pawl *D* rotates ratchet wheel *Z*. The indexing pawl *E* is disengaged by the action of levers *F* and *G*. As lever *G* moves outward, pin *L* allows locking pawl *J* to strike the top of the ratchet wheel *Z* through the action of spring *N*.

As the ratchet tooth passes, the spring *N* finally pulls locking pawl *J* into engagement with the tooth. Latching pawl *K* is disengaged by the tooth of ratchet wheel *Z* and is snapped into engagement at the end of the stroke by spring *O*. The positions of the pawls *D*, *E*, and *J* at the end of the forward or indexing stroke are shown by the full lines in the view at the right. The dot-and-dash lines in this view show the positions of the latches and pawl *K* when the bellcrank has almost completed its return stroke.

The dot-and-dash lines of pawl latch H and locking pawl J show the positions of these pawls when J rides against the top of ratchet wheel Z.

On the return stroke, the ratchet wheel Z is held stationary by pawl K which has returned to its original position. Pawl D is disengaged by the tooth of the ratchet wheel, and pawl E is returned by the action of levers F and G. Pawl J is not disengaged until the last quarter of the stroke of bellcrank C, when it is disengaged by pin L which forces the pawl latch over against the pull of spring N. Referring to the dot-and-dash outlines of the pawls D, E, and J in the view at the right, pawl J is shown in the position it occupies just after disengagement with the ratchet wheel Z; pawl E is shown about to engage the ratchet wheel; and pawl D is shown riding against the top of the ratchet wheel.

It will be noted in the central view and the view at the left that all spring connections are made by bringing studs P, S, and T out from the pawls. This was done to simplify the assembling and disassembling of the unit. The delayed action of locking pawl J is the main feature of this unit. On the return stroke of bellcrank C, lever F, when rotated about one-third of its total movement, causes lever G to move only a small fraction of its total movement. After the second third of the movement of lever F, lever G will have moved one-half of its stroke. Thus pawl J is not disengaged until after three-quarters of the return stroke of bellcrank C is completed.

Ratchet Motion which Varies Movement of Driven Shaft Twice per Revolution.—A ratchet mechanism that automatically increases and decreases the movement of the driven shaft twice in each revolution is shown in Fig. 4. This motion is applied to a wire-forming machine to produce a constantly varying rate of feed. The gear B and the ratchet wheel C are mounted on shaft A and revolve with it. The oscillating lever D is free on shaft A and

carries the pinion E, which meshes with gear B. Lever D also carries the pawl J, which engages ratchet C.

Lever F is attached to the side of pinion E, and is connected to the link G at its lower end. The upper end of link G is carried on stud K, which also carries one end of the rod H. Stud K fits into a hole in the block M, which slides in a dovetail groove on the upper end of lever D.

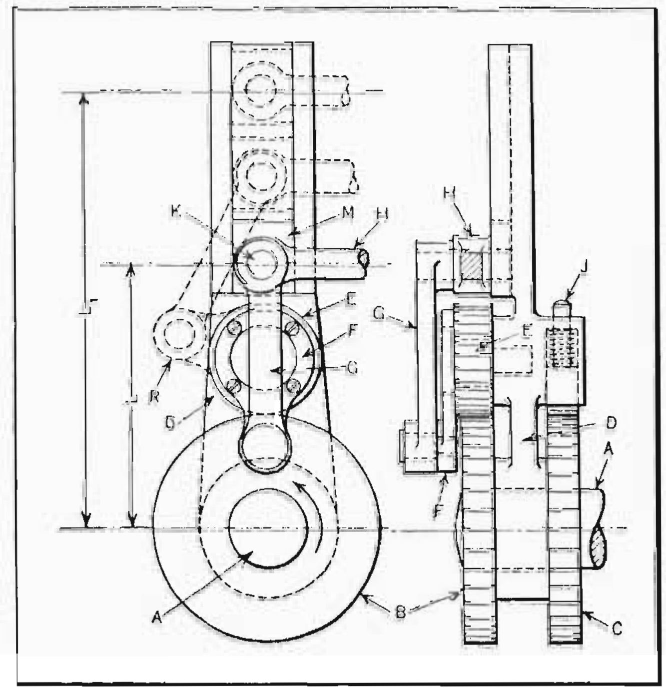


Fig. 4. Ratchet Mechanism that Automatically Increases and Decreases Effective Length of Ratchet Arm Twice During Each Revolution

In operation, rod H is given a reciprocating motion by a cam. The movement of rod H produces an oscillating motion of lever D on shaft A. On the forward stroke, pawl J engages with ratchet C, causing the entire assembly to make a partial revolution in the direction indicated by the arrow on gear B. On the return stroke, pawl J rides over the teeth of ratchet C, while shaft A, with gear B,

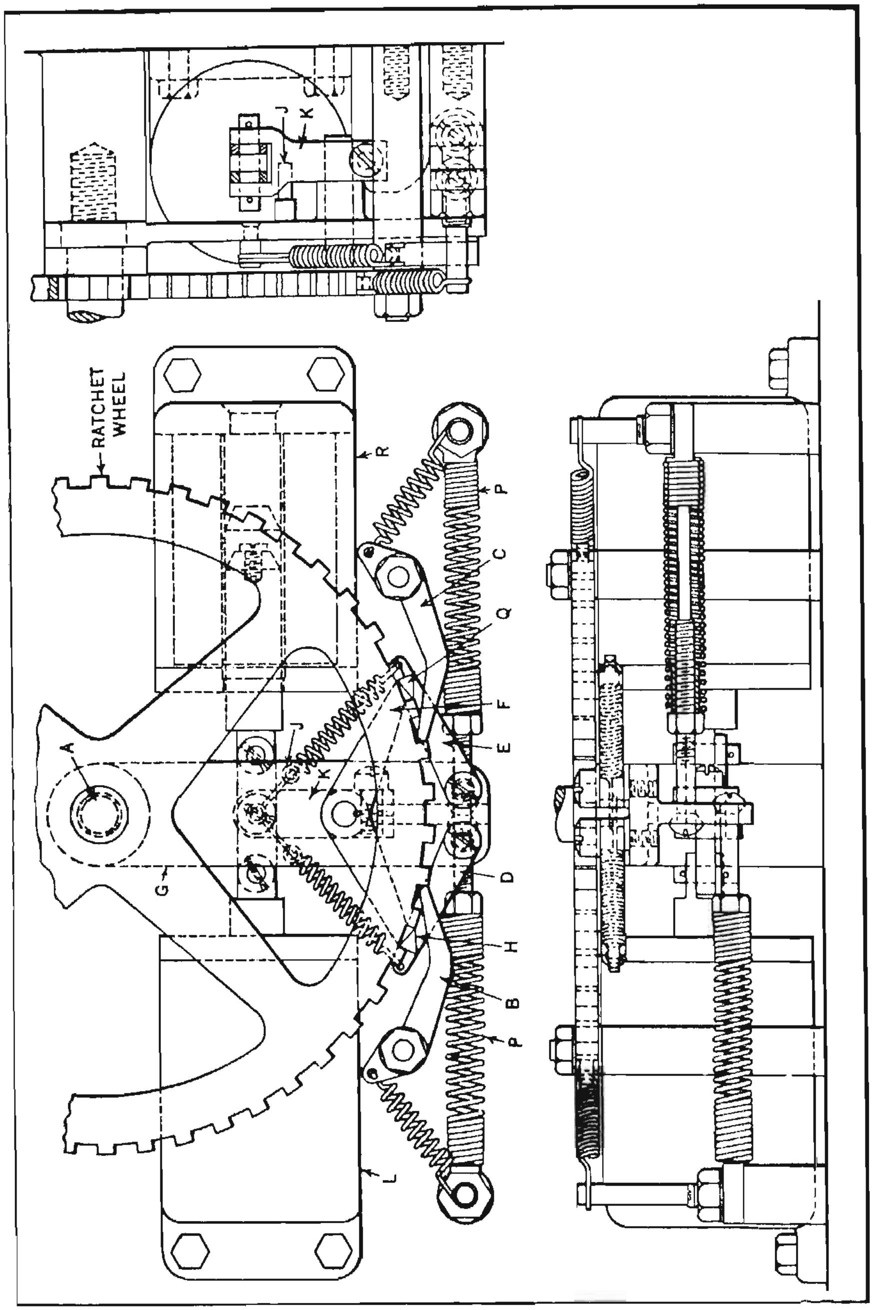


Fig. 5. Electrically Operated Ratchet Mechanism that can be Indexed Rapidly in Either Direction

remains stationary. This causes pinion E, which meshes with gear B, to make a partial revolution.

The movement of pinion E carries lever F to the position indicated by the dotted lines R, causing the link G, connected to rod H by stud K, to move upward, increasing the center distance between stud K and shaft A. In this manner, the length of the lever arm is varied from L as a minimum to L_1 as a maximum. The number of cycles performed per revolution of shaft A is determined by the ratio between pinion E and gear B, which, in this case is 2 to 1.

Solenoid-Operated Reversible Ratchet Mechanism Adapted to Remote Control.—The mechanism shown in Fig. 5 has interesting possibilities as a means for controlling machines or equipment from a distance. By sending current through one of the two electromagnets or solenoids L and R one of the two centering springs P will be stretched so that it will index the ratchet wheel one tooth in one direction when the current to the solenoid is cut off. Sending electric current through the other solenoid indexes the ratchet wheel one tooth in the opposite direction. The solenoids act very quickly and the spring centers the mechanism without shock. Thus the ratchet wheel can be indexed rapidly in either direction at the will of the operator. No switching arrangement for sending the current through either of the solenoids is shown, but copper contacts set in the periphery of a revolving fiber or Bakelite disk can be arranged to furnish the intermittent electrical impulses necessary to energize the magnets so that each impulse will move the ratchet wheel one tooth in the desired direction.

The reversible ratchet mechanism shown could be used to actuate an elevator position indicator, for example. By using some of the parts and eliminating others, a self-locking device that will index in one direction only could be obtained. Such a device would be suitable for an automatic feed for a notching press. Numerous other applications are possible for this device when used as a reversible ratchet with solenoid control or when controlled by mechanical means. In some cases, it has been used as a single-direction ratchet with either magnetic or mechanical control.

The electromagnets are used to set the mechanism and to stretch or extend one of the adjustable tension springs *P*. These springs are adjusted to overcome the friction of the mechanism and of the machine part actuated by the mechanism. When these springs contract, they gradually exert less force on the parts actuated, so that there is less shock to the mechanism when the stopping pawl drops into its proper notch in the ratchet wheel.

A solenoid should not be used to move the ratchet wheel, because the pull of a solenoid increases very rapidly as the length of the air gap in the solenoid is decreased. For instance, with an air gap 1 inch long, we might obtain a pull of, say, 5 pounds, but when the gap of the same solenoid has been decreased to 1/32 inch, the pull may be as high as 2000 pounds. The hammer blows delivered by the application of such force would flatten the end of the stopping pawl and the sides of the ratchet wheel. It would also cause a rebound of the parts, which would not give sufficient time for the stopping pawl to become properly seated, and the shock of the sudden stopping action might upset the parts actuated. Hence, an adjustable initial tension spring of sufficient strength to overcome the working friction of the mechanism and the parts actuated by it is preferable.

Operation of the Reversible Ratchet.—In considering the operation of the mechanism, let us first assume that an electric current is sent through the solenoid *R*. This causes arm *K* to be pulled to the right until it strikes the right-hand pin *J* on the arm *G*. This, in turn, causes the claw *F* on arm *K* to turn and move the pawl *E* so that the triangular projection *Q* is released from the slot in the ratchet wheel. Next, arm *G* moves to the right about the pivoting

point *A*, pulling the triangular boss *H* of pawl *D* up the side of the tooth of the ratchet wheel. This action lifts the stationary locking pawl *B* from its notch, as shown in Fig. 6, and drops boss *H* into the notch formerly occupied by pawl *B*. At this point of the operation only pawl *C* is engaged. This pawl prevents the ratchet wheel from being moved in a counter-clockwise direction by the friction developed by the moving parts.

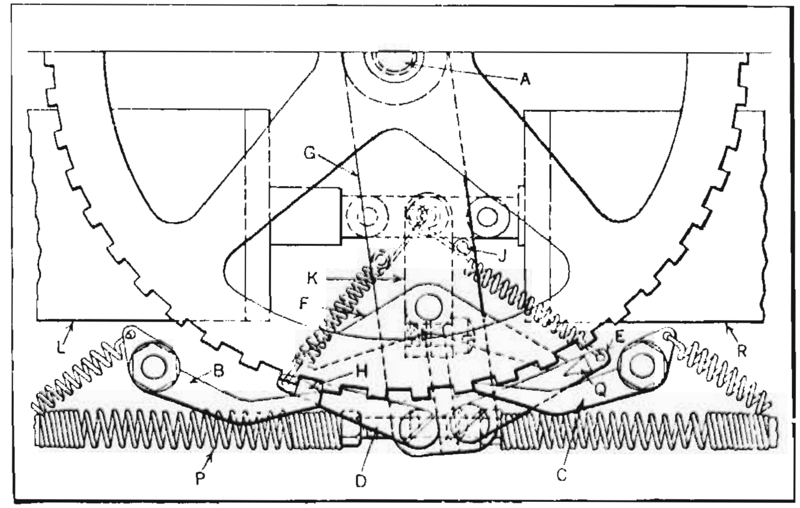


Fig. 6. Mechanism Shown in Fig. 5, with Working Parts in Positions Occupied on Application of Electric Current

The tension spring *P* has now been extended, and when the current through the solenoid is broken, this spring returns arm *G* to the central position. The triangular projection *H* on pawl *D* pushes the ratchet wheel one tooth in a clockwise direction. Pawl *B* rides down the side of the triangular projection *H* and stops the movement of the ratchet wheel. Thus the ratchet wheel is rotated an amount equivalent to one tooth space and positively stopped each time the electrical circuit is completed and opened. The direction of rotation is controlled at the will of the operator, rotation in the opposite direction being obtained

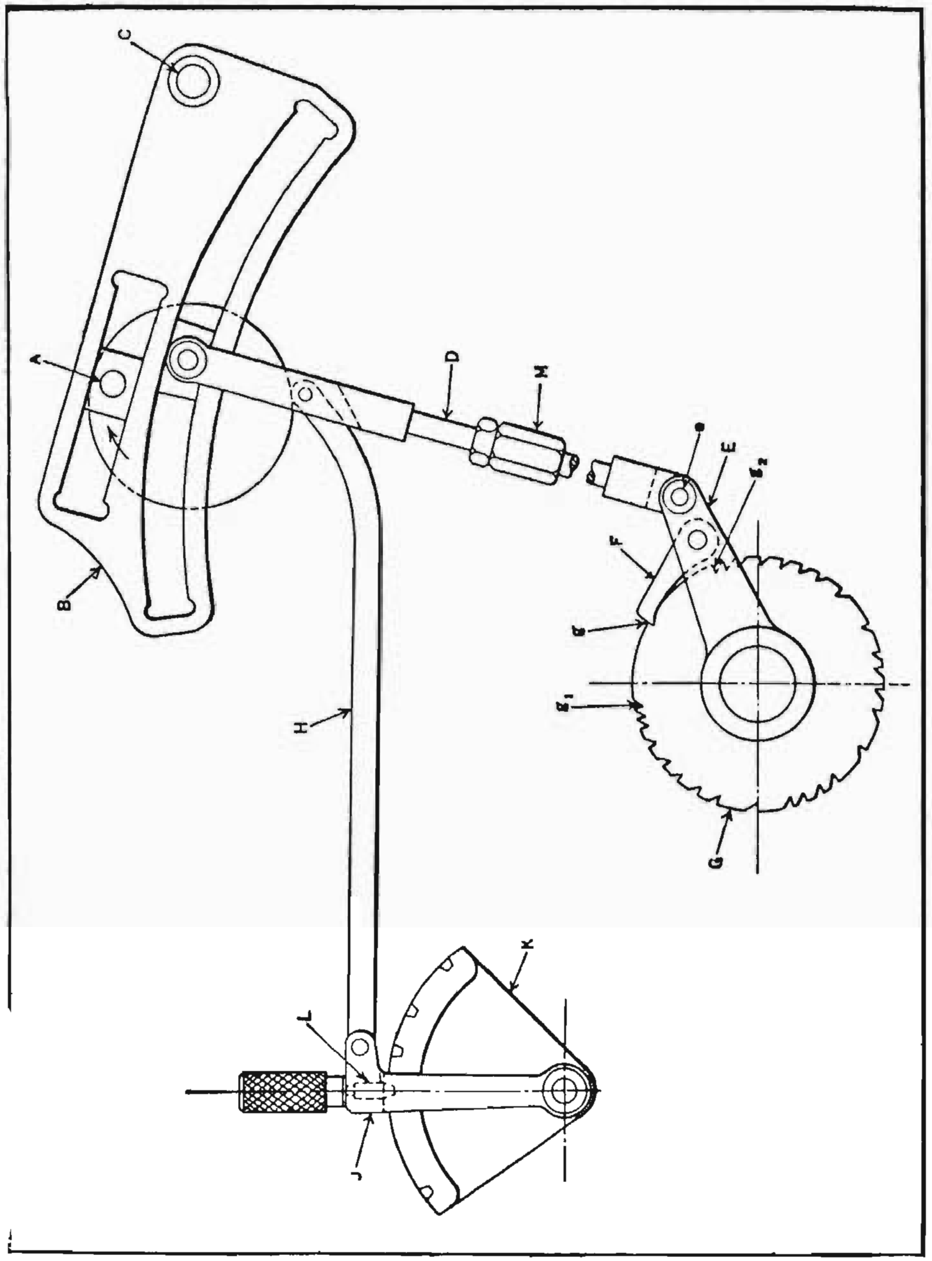


Fig. 7. Design of Ratchet Feed Mechanism with Link Motion Adjustment

by making and breaking the circuit through solenoid L instead of solenoid R.

The springs for pawls B, C, D, and E are designed to give just sufficient tension to operate their respective pawls satisfactorily, while the initial tension centering springs P are adjusted to give just sufficient tension to move the work, the mechanism, and the plunger cores of the solenoids.

Fig. 5 shows the mechanism in a neutral position—that is, with no electric current applied to the magnets. The arm K is shown in a central position for clearness in illustrating the details, although this arm would probably never remain in this position when the mechanism was in use. It may be noted here that there is no need for the spring to be so adjusted that the arm K will be exactly centered when no current is passing through either of the solenoids.

Ratchet Feed with Link-Motion Adjustment.—The ratchet and pawl feed shown in Fig. 7 was designed to fulfill the following conditions: (1) The rate of feed must be changeable without stopping the machine; (2) the pawl must always terminate the feeding stroke in the same angular position; (3) after making four complete revolutions, the feeding movement must cease for an interval and the pawl must always engage the same notch on the final movement.

The feeding movement is derived from the crankpin A, which imparts a non-varying angular reciprocating movement to plate B about the fulcrum stud C. By having the crankpin A rotate in the direction shown by the arrow, a quick-return motion is obtained for the pawl. The link D transmits the movement to pawl lever E, and pawl F transmits the feed to ratchet wheel G.

The swinging link H is connected to the link D and the anchor lever J, which can be swiveled to various positions along the segment K by withdrawing the spring plunger L. The operator can easily adjust this member on the return stroke of the pawl F when the parts are not under a load.

Adjustment of lever J causes the upper end of link D to slide along the lower slot in plate B. Thus by locating the upper end of lever D either nearer or farther from the fulcrum C, a shorter or longer movement of the pawl F may be obtained, as desired.

The lower slot in plate B is formed to a radius of the same length as link D. Thus, when plate B is in the highest position, as shown in the illustration, the center e of the arc-shaped slot will always be in the same place. It will be seen that when the motion is arrested in the position shown, the upper end of link D can be traversed the whole length of the lower slot without imparting any movement to the pawl F. The turnbuckle M provides the adjustment required for locating the point e accurately.

The ratchet wheel G is given a rather unusual form in order to meet the third requirement. The number of indexing movements per cycle ranges from 20 to 36, and as four revolutions of the ratchet wheel are completed per cycle, we have $20 \div 4 = 5$ notches and $36 \div 4 = 9$ notches The number of notches required for the different numbers of indexing movements within this range are obtained in the same manner. The essential feature is that the first notch for all feeding movements shall be located at g.

With this ratchet feed, it is obvious that the coarsest feed will require a minimum angular movement of pawl F, equivalent to $360 \div 5 = 72$ degrees, and that the finest feed will require an angular movement of $360 \div 9 = 40$ degrees. The notches nearest notch g, that is, g_1 and g_2, are located 40 degrees each side of point g, or 80 degrees apart.

Now, if the anchor lever J is moved during the running period to increase the feed to the maximum amount, the first one or two movements following the change may be erratic and the pawl may fall short of the required movement; but even if it happens to engage notch g_1, which rightly belongs to the 40-degree feed or the 36-movement

indexing cycle, a swing of 72 degrees plus, say, 3 degrees for clearance, will be insufficient to engage the tooth at g_2, which, as previously stated, is 80 degrees from g_1. Thus no movement of ratchet G will occur during the next feeding stroke, and there will be no indexing movement until pawl F advances to and engages the correct notch g. Ratchet G will then be rotated until notch g reaches the correct finishing point.

From the preceding description it will be obvious that the coarsest feed must be slightly less than twice the finest feed in order to insure proper functioning. Thus, if we let S equal the smallest number of divisions and G the greatest, then G must equal $2S - 1$. In the case described, $S = 5$; therefore, $G = 9$. If $S = 8$, $G = 15$.

Theoretically, we could then use all the numbers from 9 to 14, inclusive. In practice, however, some of these divisions would interfere, but this trouble could be avoided by having two ratchets mounted side by side with the divisions split up between them and the zero notches on both ratchets in alignment. Individual pawls, would, of course, be necessary. If a slight irregularity in feed is not objectionable, one ratchet could be used, employing the maximum number of equally spaced notches consistent with strength, but with the spaces from g_1 to g and g to g_2 left blank. Each of these spaces would equal $360 \div 15 = 24$ degrees.

Intermittent Rotation of Ratchet Wheel During Forward and Reverse Movements of Pawl Lever.—The mechanism shown in Fig. 8 was designed to provide an automatic infeed for the reciprocating table of a grinding machine. The mechanism is so designed that the table is fed inward at the moment of reversal at each end of the stroke. Ratchet wheel G turns counter-clockwise when left-hand pawl E moves downward, and G also turns in the same direction when right-hand pawl E moves downward. Pawls E are attached to and operated by lever D. The cross-slide or unit B, which is actuated by the feeding mechanism, is

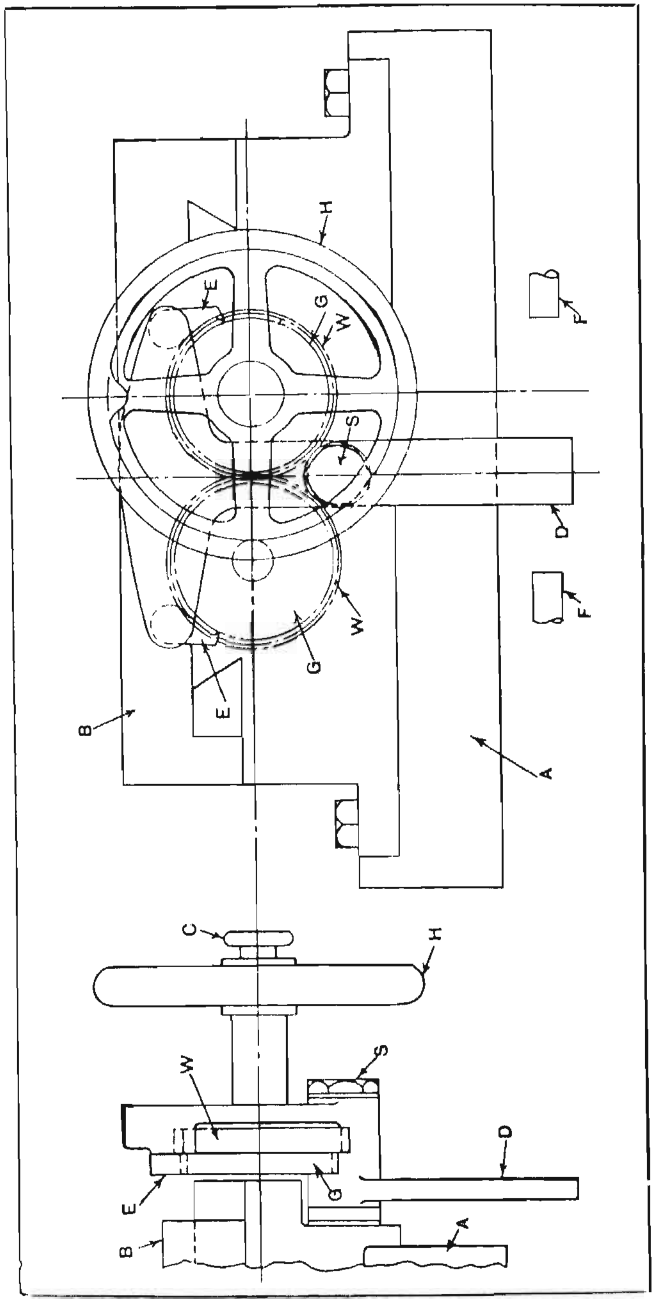


Fig. 8. Reciprocating Grinder Table Equipped with Automatic In-feed Mechanism

mounted on the reciprocating table *A* of the grinding machine. The member *D* is mounted on a fulcrum stud *S*. When the table *A* is about to reverse at either end of its travel, the lower end of the lever strikes one of the adjustable stops *F*. This action causes member *D* to pivot about stud *S*, thus transmitting a rotary motion to one of the ratchet wheels *G* through the pawl *E*. The ratchet wheels are secured to meshing gears *W*, one of which is mounted on the cross-slide feed-screw. Thus the pivoting or angular movement of member *D* causes the feed-screw to be rotated a certain amount in the same direction when the table reverses at each end of the stroke.

The amount of angular movement of member *D* is determined by the position of the stops *F* with respect to the position of the table at the moment of reversal. After member *D* has been actuated by coming in contact with one of the stops *F*, it is inclined in the proper direction for operation by the stop *F* at the opposite end of the table.

The handwheel *H* can be connected with the cross-slide feed-screw by setting the knob *C* to engage an internal clutch. When the handwheel is thus connected, it provides a means for feeding the cross-slide in or out. With the automatic feeding mechanism applied as shown in the illustration, however, the pawls *E* must be thrown up out of contact with the ratchet feeding wheels in order to permit the slide to be fed inward by the handwheel.

Intermittent One-Revolution Drive.—The mechanism to be described is designed to revolve a disk intermittently, so that it will make one revolution at the conclusion of the feeding movement of a certain machine. The ratchet wheel *A*, Fig. 9, which is keyed to shaft *B*, revolves continuously at a constant speed. Shaft *B* rotates freely within the boss of disk *C*, but when pawl *D*, attached to disk *C*, is allowed to engage ratchet *A*, the parts *A* and *C* revolve as one unit. Gear *E*, which is keyed to shaft *B*, then transmits motion to a slide (not shown).

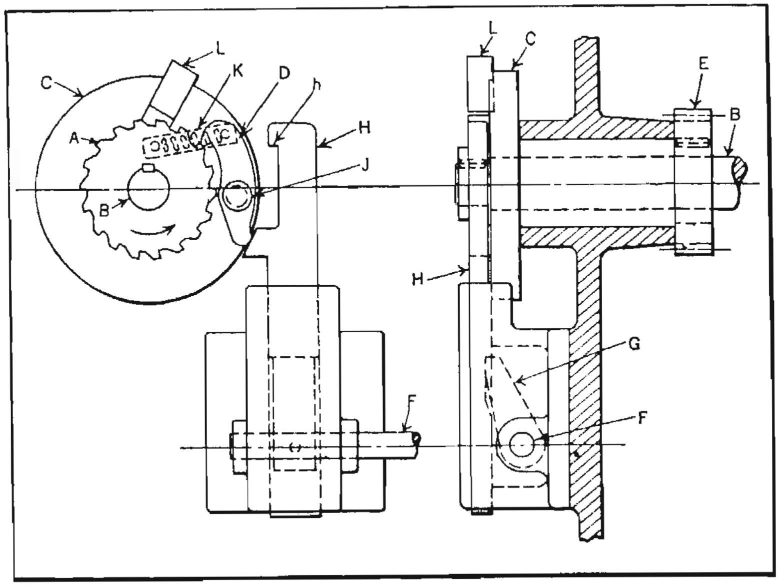


Fig. 9. Mechanism by which Rotating Shaft B Imparts One Revolution to Gear E when Shaft F Releases Latch G

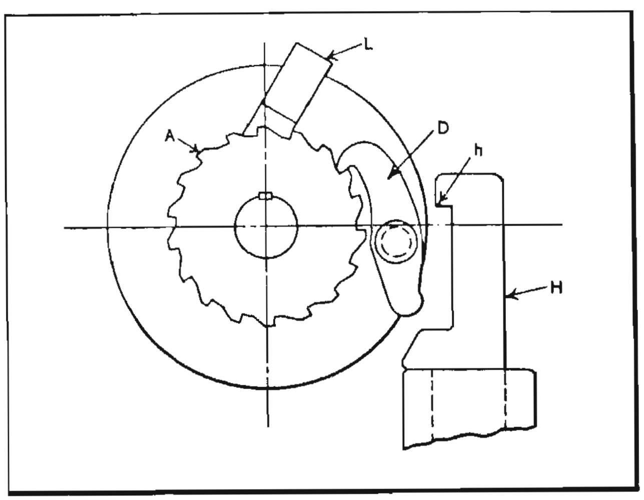


Fig. 10. Position of Bar H after Being Released by Latch G, Fig. 9

At the completion of the feeding movement, shaft *F* is rocked in a clockwise direction until latch *G*, keyed to shaft *F*, moves clear of the drop-bar *H*. This allows bar *H* to drop and clear the end of pawl *D*, which then pivots on stud *J*, under the action of the tension spring *K*, and engages the revolving ratchet wheel in the manner illustrated in Fig. 10.

In order to limit the rotation of disk *C* to one revolution, the projecting lug *L*, affixed to and rotating with disk *C*, engages the upper hook *h* on bar *H*, as shown in Fig. 11. The lug *L* then lifts bar *H* high enough to allow latch *G*, Fig. 9, to re-enter the notch in bar *H*, and hold the latter member in the upper position. This repositioning of bar *H* takes place before one revolution is completed. The continued movement of disk *C* causes the end of pawl *D* to come in contact with the lower projection on bar *H*, thus forcing the pawl out of engagement with the revolving ratchet *A* and arresting the movement of disk *C* after it has completed one revolution. When bar *H* is down, the upper hook *h* must clear the end of pawl *D*, and the lower hook on bar *H* must clear projection *L*, in order to have the mechanism function correctly.

With proper attention to lubrication, bar *H* will function satisfactorily, but dirt or gummy oil will cause it to become stuck in the upper position, as only the weight of the bar itself is relied upon to cause the return movement. It was therefore decided to change this detail. Although a spring could have been used to exert a downward pull on bar *H*, a lever *M*, as shown in Fig. 12, was substituted for bar *H*. This construction was less expensive than the first one, and the rotating movement about fulcrum stud *N* produced less frictional resistance than the sliding movement of bar *H*.

The only other difficulty experienced in the design illustrated in Fig. 9 was with the time interval required to free latch *G* after the drop-bar had been allowed to fall. The rate of the rotation of ratchet wheel *A* is comparatively

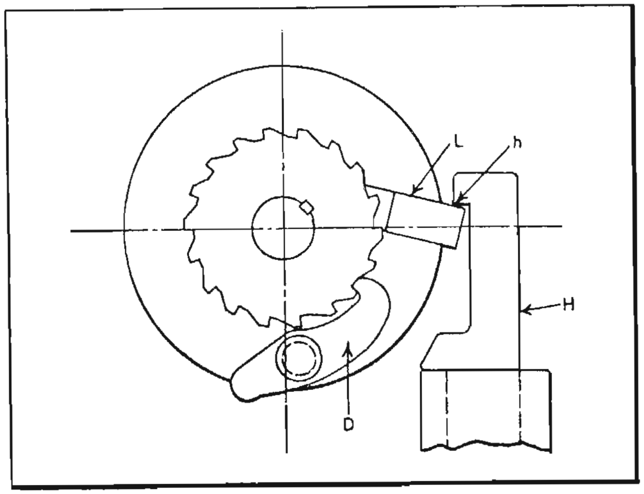


Fig. 11. Lug L Engaging Hook h by which it Lifts Bar H

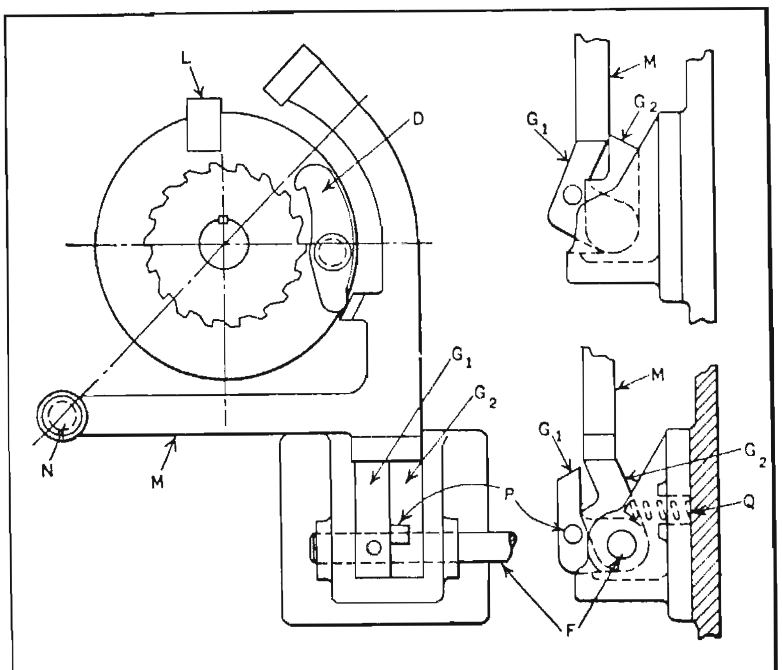


Fig. 12. Improved Development of Drive Shown in Fig. 9

high, and with a very slow feed, the trip motion shaft F did not operate quickly enough. Thus, latch G would not be ready to retain bar H when the latter member was lifted by the lug L. This meant that the disk would be given a second revolution before pawl D was disengaged. To avoid this difficulty, latch G was replaced by two latches, as shown at G_1 and G_2, Fig. 12. Latch G_1 is keyed to shaft F, but latch G_2 is free to revolve.

When shaft F is rocked clockwise, latch G_1 pushes latch G_2 from under lever M by means of the pin P. Lever M then occupies the position shown by the view in the upper right-hand corner. Thus lever M is raised slightly, but when it is released by latch G_2, latch G_1 occupies a position directly under lever M, thus preventing it from dropping the full distance. When shaft F is released and latch G_1 returned to the original position, the pressure of spring Q tends to return latch G_2, but cannot do so because the latch strikes the side of lever M. Lever M is then free to drop between the two latches and start the one-revolution drive. This arrangement prevents the one-revolution drive from functioning until shaft F has been tripped and returned to its original position.

Dwell of Sprocket-Chain Conveyor at Regular Intervals for Loading.—Endless sprocket-chain conveyors are used extensively for carrying lacquered or paint-sprayed parts through drying ovens. The work-holders, as a rule, are mounted at each link joint. In one application of this type, the chain travels a distance equal to the length of seven links and then dwells long enough to allow these links to be loaded by means of an automatically operated feeding device. This alternate movement and dwell of the chain is repeated continuously, so that the chain is always fully loaded as it passes through the oven.

The arrangement for obtaining this intermittent movement is shown in Fig. 13. It consists of driving sprocket A, ratchet wheel B, pawl C, and driving sleeve D. In each link joint of the chain is mounted a slender shaft, threaded at its upper end for a work-holder. The lower end of these shafts is provided with a roll, which, at a certain point in the chain travel, comes in contact with a rapidly moving endless leather belt (not shown). This causes the shaft, work-

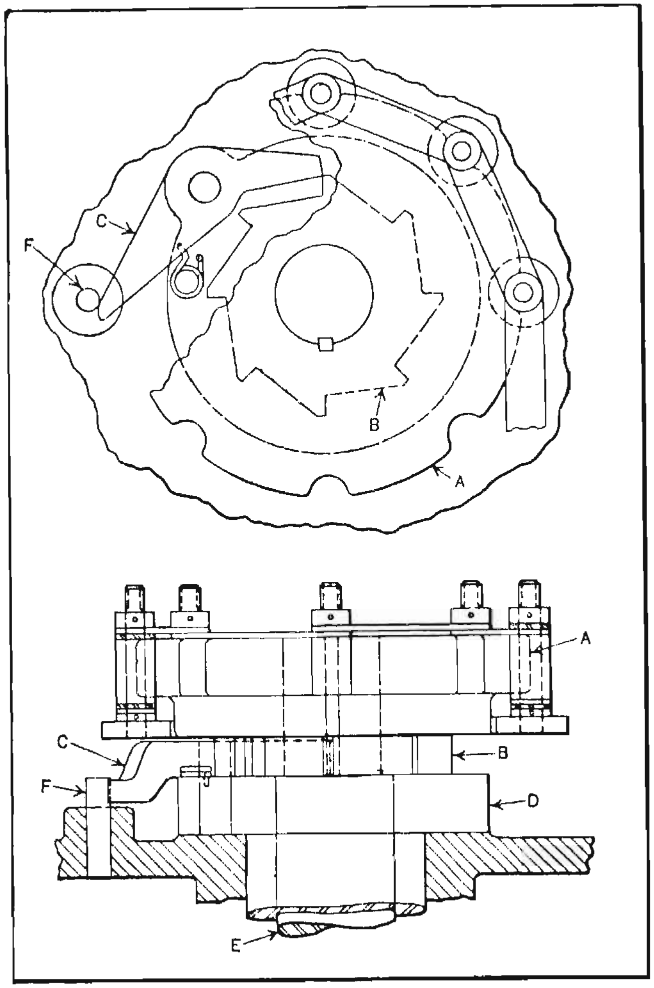


Fig. 13. Ratchet Mechanism that Disengages a Sprocket Wheel from its Driving Member at Regular Intervals to Obtain an Intermittent Movement of the Chain

holder, and work to rotate, so that the paint or lacquer being applied will be distributed evenly.

Both the sprocket and ratchet wheel are keyed to shaft *E*, which turns freely in the sleeve *D*. The sleeve is rotated continuously in a clockwise direction by another member of the machine, and at its upper end is pivoted the ratchet pawl. As indicated, the pawl has been forced out of engagement with the ratchet wheel by pin *F*. This causes the sprocket and chain to dwell long enough for seven work-holders to be loaded. As sleeve *D* continues to rotate, the outer end of the pawl passes by pin *F*, permitting the other end of the pawl to engage the next tooth and carry the sprocket wheel around to the position shown. Here, the pawl is again forced out of engagement with the ratchet, causing the sprocket wheel to dwell a sufficient time for seven more work-holders to be loaded. The movements described are repeated for each succeeding revolution of sleeve *D*.

Duplex Type of Intermittent Drive.— The mechanism shown in Fig. 14 has the component parts so arranged that it can be used for two distinct purposes. In the application to be described first, the continuously rotating shaft *A* transmits an intermittent rotating movement to the shaft *B*. In disk *C*, fastened to shaft *A*, is a pin *D* which contacts with the surface of the internal cam on the combination cam and lever *E*.

Lever *E* is pivoted on shaft *B* and is provided with a lug to which the spring pawl *F* is fastened. Two spring pawls are shown. but one or more may be provided, depending upon the movements imparted to lever *E*, as will be explained later.

The enlarged view, Fig. 15, illustrates the method of producing the oscillations in lever *E*. The movement of pin *D* on the internal cam surface of lever *E* forces the lever to one side until a point is reached, as shown by the dotted lines, where pin *D* passes over the ridge or shoulder in the

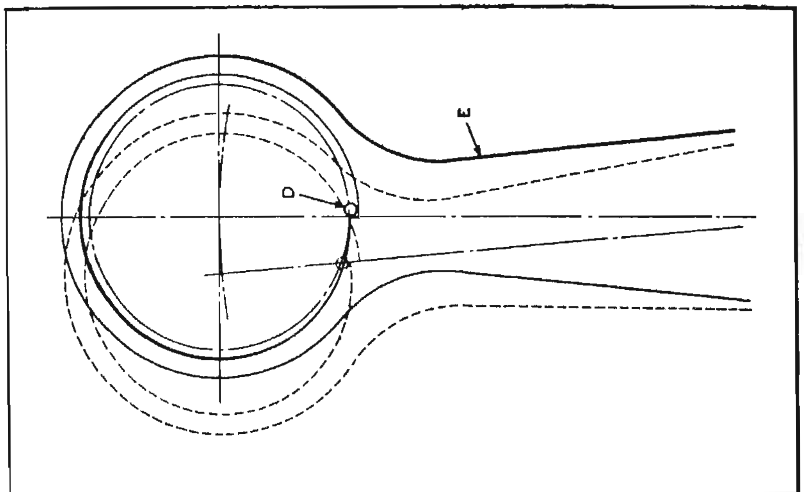


Fig. 15. Enlarged View of Members D and E, Fig. 14

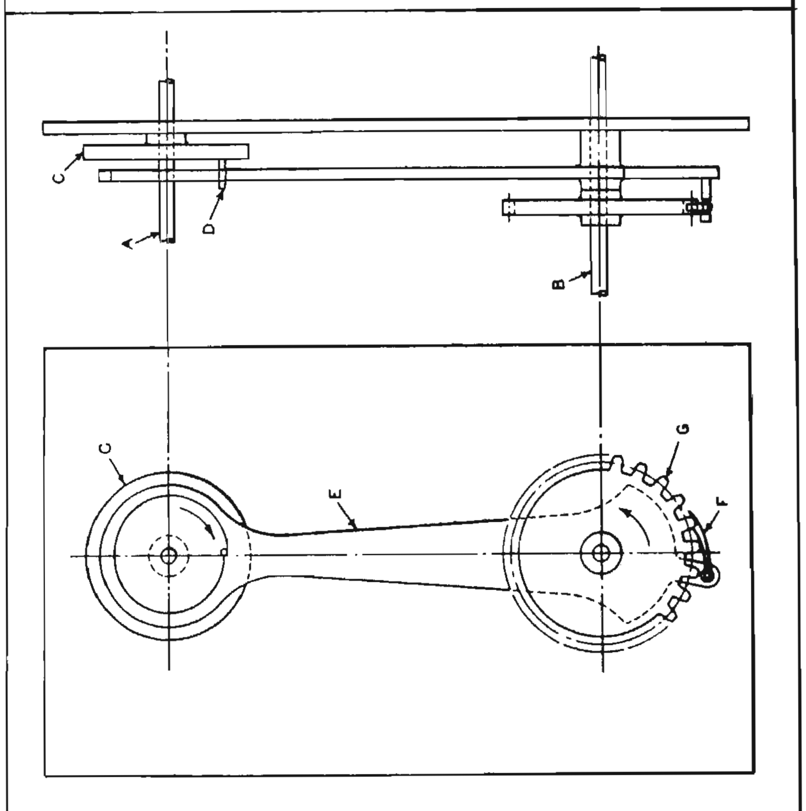


Fig. 14. Mechanism for Transmitting Intermittent Motion from A to B or from B to A

cam surface. The location of pin *D* inside the cam produces a positive motion in the lever and eliminates the use of any form of spring.

If the stroke of the lever movement is sufficient to produce a movement in the pawl equivalent to one-half the pitch of the teeth in the gear or ratchet wheel *G*, then two spring pawls are required, as illustrated, the inner pawl being shorter by one-half the pitch of the teeth. If the lever stroke is equivalent to one-third of the tooth pitch, three spring pawls are required, each being shorter than the next outer one by one-third the tooth pitch. In this manner, the number of intermittent movements of gear *G* can be varied to conform with requirements.

In the second application, shaft *B* is driven continuously in a counter-clockwise direction. Through a friction clutch arrangement (not shown) shaft *B* drives shaft *A* in a clockwise direction. Lever *E* stops the rotation of shaft *A* at predetermined intervals, so that shaft *A* rotates intermittently.

Referring to Fig. 15, assume that pin *D* has reached the position shown by the dotted lines and that shaft *A* is being driven by shaft *B* through the friction clutch drive. As shaft *A* with its disk *C* revolves, pin *D* acts on the cam surface of lever *E*, causing it to pivot about shaft *B* in a direction opposite to that of the rotation of shaft *B* until pin *D* reaches its lowest position, as shown by the full lines in both Figs. 14 and 15. At this point the pin comes in contact with the shoulder on lever *E* and rotation of shaft *A* is stopped, due to the engagement of pawl *F* with the teeth on gear *G*.

It should be mentioned here that the drive between shafts *B* and *A* is designed to rotate shaft *A* at a faster speed than the driving shaft *B*. Slippage of the friction clutch in the drive from *B* to *A* occurs, of course, at this point and continues until shaft *B*, rotating in a counter-clockwise direction and carrying the spring pawls *F* around with it, causes

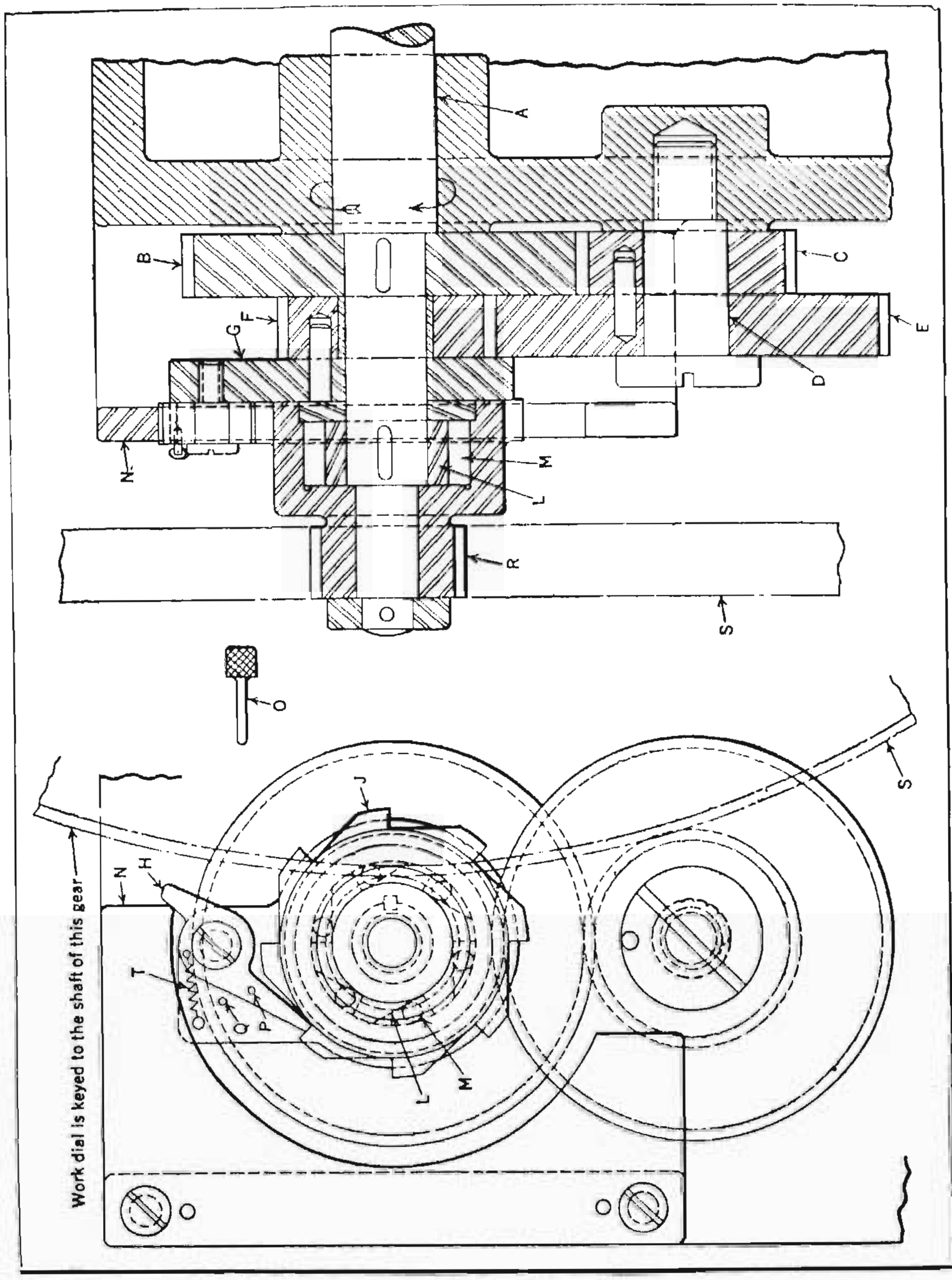


Fig. 16. Variable-velocity Rotary-motion Mechanism for Work-carrying Dial of Polishing Machine

lever E to pivot or swing around to the position shown by the dotted lines in Fig. 15. Pin D is released when contact with the shoulder on lever E is made at this point, and the cycle is repeated as described.

Combination Roller Clutch and Ratchet for Imparting Variable Rotary Movement.—The over-running or "free wheeling" feature of roller friction clutches is used to advantage for imparting a variable rotary motion to the work-holding dial of a polishing machine. In connection with this movement, a ratchet mechanism is employed. Similar pieces of two different diameters are polished on this machine. The work stations are so spaced that the large pieces will be close together on the dial, in order to reduce to a minimum the non-productive time of the wheel in passing from one piece of work to the other. However, the same dial and the same station spacing are employed for polishing the small-diameter work; hence, without the special variable-motion mechanism illustrated (see Fig. 16), an appreciable loss in production time would result, owing to the gaps between the work over which the wheel must pass.

With the mechanism shown, each station is advanced rapidly toward the polishing wheel until the polishing action commences. At this time, the rotary movement of the dial is immediately decreased to the desired speed. This speed of the dial continues until the part passes out of contact with the wheel, and the speed is then immediately increased, so that the succeeding dial station is moved rapidly to the wheel. Provision is made for imparting a constant rotary movement to the dial when polishing the larger work.

The mechanism for imparting the variable dial movement is relatively simple. The dial is driven by the shaft A, which rotates at a constant velocity. On this shaft is keyed the gear B, meshing with gear C on stationary stud D. Gears C and E are pinned together, the latter meshing with gear F, which is free to turn on shaft A. Pinned to gear F and also free to turn on shaft A is the plate G, to which is pivoted the pawl H. This pawl intermittently engages the ratchet wheel J, its engagement being controlled by the stationary cam N. Wheel J turns freely on shaft A, and its left-hand end forms the pinion R for rotating the dial, the pinion meshing with the bull gear S secured to the back of the dial. The inside of the ratchet wheel is bored out to receive the roller clutch arrangement, which consists of the core L and rolls M. The core is keyed to shaft A.

The ratio of the gears B, C, E, and F is such that plate G rotates four times as fast as shaft A. Hence, owing to the over-running feature of the roller clutch, if pawl H is in engagement with the ratchet wheel, the dial will be rotated at high speed through gear R. However, provision is made for automatically disengaging the pawl just as the polishing wheel comes in contact with the work. This is accomplished by means of the cam N.

For example, the pawl is shown engaged, the dial having been rotated at high speed to advance the work to the wheel. With the ratchet wheel in the position indicated, the wheel is just about to come in contact with the work. As the plate G continues its rotary movement, the cam N swings the pawl out of engagement with the ratchet wheel, allowing the roller clutch to pick up the motion and continue the rotation of the dial at one-fourth the preceding angular velocity.

This new angular velocity is constant and continues until the polishing wheel has passed over and commences to leave the work. At this point, the pawl leaves the lower part of cam N and coil spring T forces it into engagement with the ratchet wheel, so that the high speed of the plate is once more transmitted to the ratchet, thus rotating the dial at a corresponding velocity to advance the next station toward the polishing wheel. In this way, the two driving members G and L alternately transmit the required speeds to the

pinion gear R. Cam N is designed to hold the pawl out of engagement with the ratchet wheel for one-half revolution of gear R. Hence, since plate G rotates four times as fast as core L, the angular movement of gear R for one-half turn will be four times that for the remaining half turn of the cycle.

When the larger work is being polished, a constant angular velocity of the dial is required. To obtain this condition, the holes P and Q in the pawl and plate, respectively, are aligned, and the pin O is inserted through them. Thus, the pawl is held out of engagement with the ratchet wheel to allow the roller to transmit the constant rotary movement of shaft A directly to the gear. There is sufficient friction in the various moving parts of the mechanism to prevent over-run of the roller clutch each time the pawl is disengaged. In applications where the inertia of the rotating part is likely to cause over-run, a simple brake can be mounted on one of the driving members.

This mechanism is designed to operate with the dial in a vertical plane, as the engagement of the rollers in the clutch is dependent upon gravity. However, the mechanism can be readily adapted to any other position of the clutch by inserting coil springs behind the rolls. Creeping movement of the clutch rolls in this unit does not accumulate and change the dial station location, since the creep for each cycle of the gear R is compensated for by the positive action of the ratchet.

Differential Ratchet for Imparting Slight Axial Movement to Feed-Screw.—In designing machines, it is sometimes necessary to make provision for transmitting an intermittent movement to a feed-screw from a reciprocating slide. For a comparatively large movement of the screw, an ordinary ratchet arrangement is suitable, but for an extremely small movement, such as that required for the feed-screw illustrated (see Fig. 17), special means must be provided.

The screw G provides the transverse feed for the wheel of a surface grinding machine used in grinding wood planer knives. It has the very short axial movement of 0.00045 inch for each cycle of the reciprocating slide A. To obtain the desired feed, two ratchet wheels C and D, operating on the differential principle are used. One wheel has 23 teeth and the other 24. Axial movement of both wheels is prevented by the bracket B fastened to the machine. This

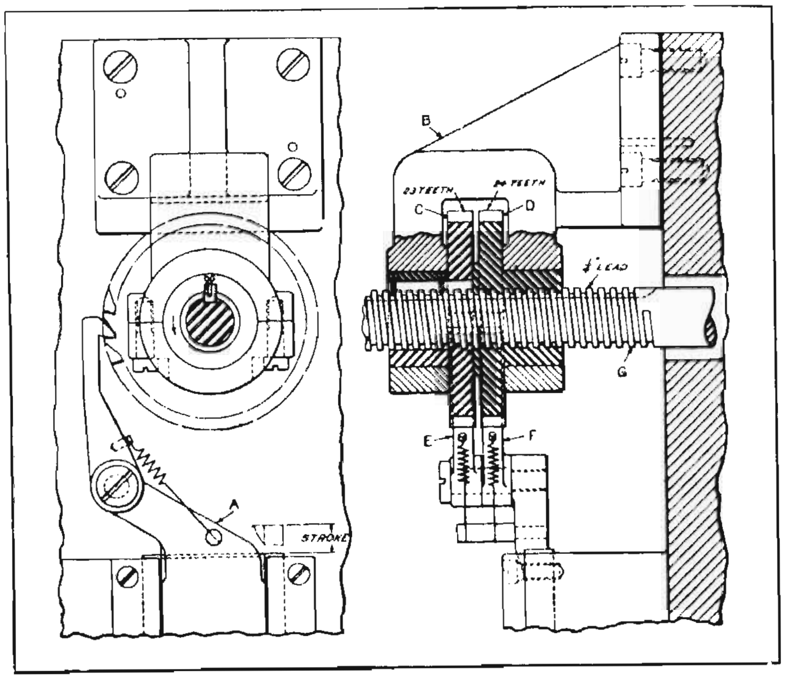


Fig. 17. Ratchet Mechanism for Imparting Slow Movement to Feed-screw

bracket also forms the bearings for the wheels. It will be noted that wheel C is provided with a feather key which engages the spline in the screw and that the screw is free to slide axially in this wheel. The bore in wheel D, however, is threaded to engage the screw. The angular movement is imparted to the ratchet wheels by their respective pawls E and F, pivoted to the slide A.

In explaining the action of this mechanism, let us assume, for simplicity, that both ratchet wheels have the same number of teeth. Thus, for every cycle of slide A, both the wheels, as well as the screw, would rotate together; consequently, there would be no axial movement of the screw. Now returning to the actual case, wheel C has one less tooth than wheel D; therefore, during one cycle of the slide, pawl F will rotate wheel D 1/24 revolution. Owing to the difference in the number of teeth, however, wheel C will be rotated by pawl E 1/23 revolution; and the axial movement of the screw will be equivalent to the difference between these two movements multiplied by the lead of the screw, or:

$$\left(\frac{1}{23}-\frac{1}{24}\right)\times\frac{1}{4}=0.00045 \text{ inch}$$

Perhaps it should be mentioned here that, in so far as the preceding description is concerned, both pawls could have been incorporated into one wide pawl encompassing both wheels. However, the requirements of the machine were such that a faster feeding movement of the screw was required for certain jobs. To accomplish this, the pawl E is swung to the left to clear wheel C and held in this position by a latch. A plunger is then released which locks wheel C to prevent its rotation. All movements for disengaging and locking the wheel are obtained by shifting one lever. However, the latch, plunger and operating lever are not shown. With the pawl E disengaged, one cycle of the slide will cause only ratchet wheel D to rotate, its angular movement being 1/24 revolution. The corresponding movement imparted to the screw is approximately 0.010 inch.

Adjustable Pawl Shield to Vary Ratchet Wheel Movement.— Alterations made in a certain product necessitated shortening a ratchet movement on the production machine. The new ratchet arrangement is shown in Fig. 18. Oscillating lever A is actuated by a cam (not shown) and carries the pawl E, which through the ratchet wheel C, transmits the required intermittent movement to shaft F. Adjustment of the angular movement of this shaft is obtained by means of the shield D. Bearing B was turned down on one end to serve as a support for the shield, which is held in a stationary position by a set-screw. Near the end of the back stroke of the lever, the shield lifts the pawl away from the ratchet wheel. Thus, part of the subsequent forward stroke of the lever is completed before the shield permits the pawl to engage the ratchet wheel, so that the angular

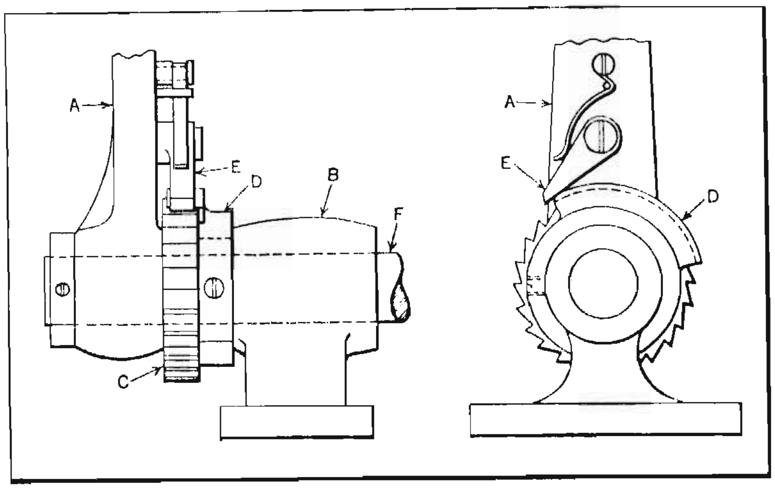


Fig. 18. Application of a Shield to a Ratchet for Reducing the Angular Movement Transmitted

movement of the shaft is shortened. By varying the position of the shield, the angular movement of the shaft will also be varied.

Combined Eccentric and Friction Ratchet for Automatically Varying a Feeding Motion.—On a special polishing machine, the work is passed under a set of oscillating brushes charged with abrasive. In the original design, the work-table was fed at a uniform rate by means of a toothed ratchet, which transmitted its movement through a shaft to the work-table. However, under certain light con-

ditions, the surface of the polished work showed a series of marks corresponding to the movement of the work-table. Though it was considered impossible to eliminate the marks entirely, it was thought advisable to break up their symmetry, so as to render them less noticeable. This was accomplished by the use of a variable ratchet movement, the design of which is shown in Fig. 19.

The shaft A, which operates the work-table, carries the eccentric B, which is keyed to it. The eccentric B is en-

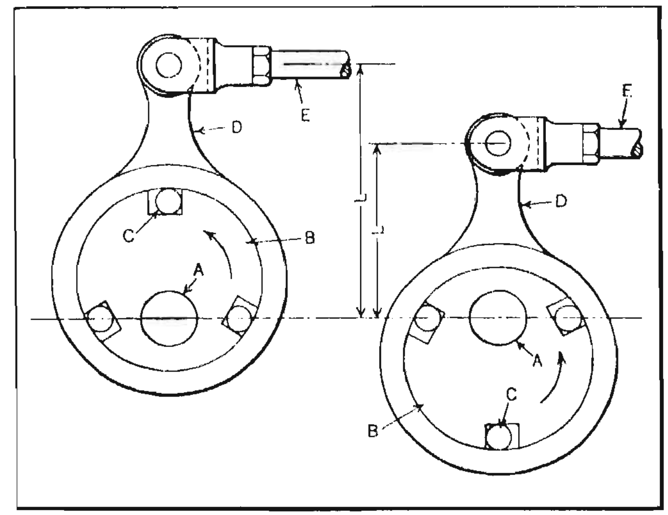


Fig. 19. Mechanism in which Eccentric Mounting of Ratchet Member B Varies Effective Length L of Ratchet Arm from Maximum to Minimum Once in Each Revolution of the Driven Shaft A

circled by the strap D, which is given an oscillating motion by the rod E. Eccentric B is grooved to receive the rollers C, forming a conventional type of roller clutch or ratchet which operates through the wedging action of the rollers C between the eccentric B and the strap D.

As the eccentric B revolves with shaft A, the effective length L of the lever arm changes constantly, the range of variation being controlled by the throw of the eccentric B. The view to the left shows the mechanism with L at its

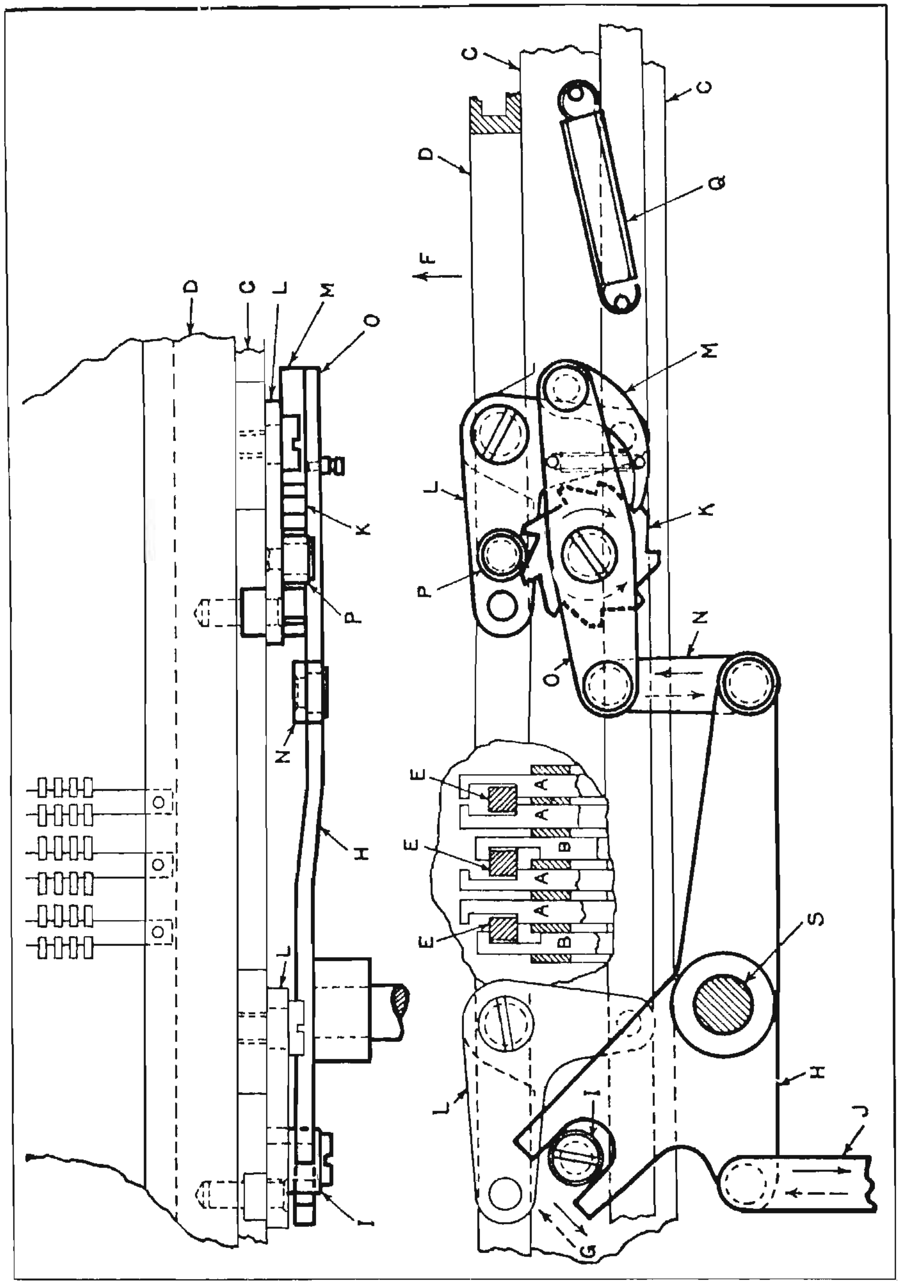


Fig. 20. The Parts Shown by Heavy Lines are Applied to a Type-casting Machine to Produce an Oscillating Movement of Frame D while the Frame C Moves through the Downward Cycle of an Oscillating Movement

maximum, while the view to the right shows L at its minimum. As the reciprocating movement of rod E is constant, the degree of movement imparted to shaft A is controlled by the length L of the lever arm. One cycle of variations is produced at each rotation of shaft A.

Mechanism for Oscillating a Part Mounted on a Moving Member.—Sometimes a designer finds it necessary to provide means for imparting a short, quick oscillating movement to a machine part on a member that is also in motion. The mechanism shown in Fig. 20 was designed to meet such a requirement. In this mechanism, the part or unit D is required to have a quick movement upward from the frame or part C in the direction indicated by arrow F, and then return to the starting position. This movement takes place while frame C is moving through the downward cycle of its oscillating movement, the direction of which is indicated by the full arrow at G.

The mechanism shown is a stop-pin bed used in a type-casting machine for the purpose of selecting groups of type. The bed consists of a multitude of flat stop-pins, slidable in a honeycomb holder. Some of these pins A are shown in the restored or cleared position, while others, such as those marked B, are shown in their depressed positions. These stops are arranged in rows, and in plan view present a two-dimensional field in which code patterns can be depressed.

Frame D is built above bed framework C with a number of bars E attached to it, and forms a sort of rigid grate which, when raised at right angles to the bed in the direction indicated by arrow F, restores all the depressed stops B to the clear positions A. When lowered into contact with the frame C as shown, the restoring unit, consisting of the members D and E, permits a new pattern to be depressed in the stop-pin bed to suit the new cycle.

Now, it is necessary to perform this clearing operation by raising and immediately dropping the unit consisting of members D and E, while the bed section is making a short, quick stroke in the direction G. This is effected by means of a single outside connection—a lever H acting with the bed frame through a roller and slot at I. The lever is propelled by a cam (not shown) through link J.

Under one of the several interconnected bellcranks L provided to secure true parallelism in the motion of members D and E is mounted a ratchet K. This ratchet is

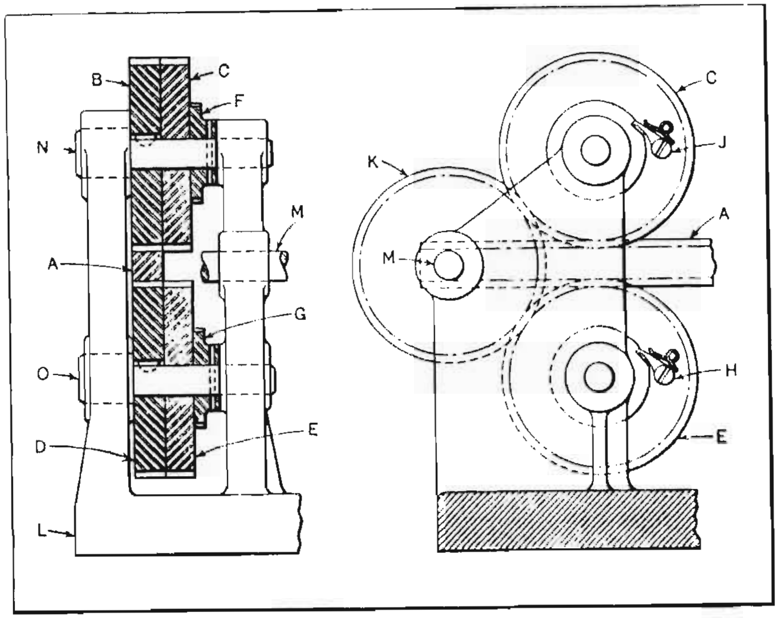


Fig. 21. Train of Gears Operated by a Rack for Imparting an Intermittent Motion in One Direction to Shaft M

mounted idly on a stud in the bed frame C and is made to act in combination with the roller P on the bellcrank L. A pawl M is arranged to receive its motion from the lever H through the connecting link N and rocking plate O, mounted idly on the same stud as the ratchet K.

It can be readily seen that during the stroke indicated by the dotted arrows there is skipping of the pawl M over the ratchet teeth, while during the stroke in the direction indicated by the solid arrows, the ratchet wheel is given positive indexing which forces the bellcrank L and the restoring unit upward. As soon as the crest of a ratchet tooth passes the roller, the frame unit is restored to its lower position on the bed frame by virtue of its weight, assisted by the tension in the spring Q.

The lay-out of the ratchet gear is such that, when the roller is in any of the ratchet tooth gashes, the roller clears the adjacent teeth by a slight amount. The stroke of the pawl is sufficiently in excess of the tooth spacing to cover this clearance and insure a full tooth spacing. The number of teeth in the ratchet is, of course, of no consequence, and such requirements as desirable size of the roller, easy camming of the roller by the back slope of teeth, available space for the whole device, etc., are factors governing the design of the ratchet.

Rectilinear Movement Converted to Intermittent Rotary Movement.—By means of the mechanism shown in Fig. 21, the reciprocating rack A imparts an intermittent rotary movement in one direction to the shaft M through the gears B, D, C, E, and K, and the ratchet wheels F and G. During each stroke of the rack, shaft M rotates one-half of a revolution and then dwells. The length of this dwell, as well as the velocity of shaft M, is controlled by automatic valves on an air cylinder (not shown) which actuate the rack.

Teeth cut in opposite sides of the rack engage gears B and D, keyed to shafts N and O, respectively. Gears C and E are free to turn on their shafts and mesh with gear K keyed to shaft M. Pawls, pivoted to gears C and E, engage ratchet wheels F and G, fixed to their shafts.

When rack A moves toward the left, gears B and D rotate in opposite directions, and pawl J simply rides over the ratchet teeth without imparting motion to gear C. Pawl H, however, engages ratchet wheel G and causes gear E and ratchet wheel G to rotate together. Now, as gear E is in mesh with gear *K*, gear *K* will rotate in a clockwise direction. When the rack reaches the end of its stroke, the automatic valves close for a predetermined time, thus holding the rack stationary and causing shaft *M* to dwell.

When the valves open, air is admitted to the opposite side of the piston and the rack moves toward the right. In doing so, pawl *H* rides over the teeth of ratchet *G*, and pawl *J* engages the teeth in ratchet *F*, causing gear *C* and ratchet wheel *F* to rotate together; and as gear *C* is in mesh with gear *K*, gear *K* will rotate in a clockwise direction as before.

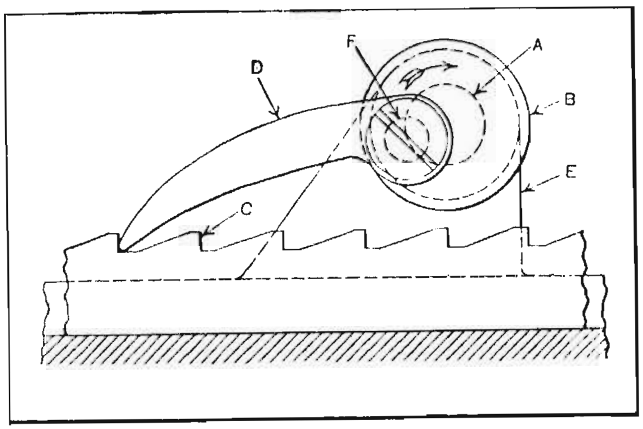


Fig. 22. Crank-driven Pawl that Imparts a Slow Movement at Both Ends of its Stroke to Prevent Jerking and Over-run

The movement of shaft *M* continues until the rack has reached the end of its stroke, at which time the automatic valves close once more to obtain the required dwell.

The angular movement of shaft *M* after each dwell depends upon the stroke of the rack and the ratio of the gears. In this case, all the gears have the same number of teeth; consequently, during one stroke, the travel of the rack must equal one-half the pitch circumference of gear *K*, or slightly more, to allow the pawls to engage properly.

Ratchet Pawl Having a Slow Movement at Both Ends of Its Stroke.—A feed-slide in a special tube cutting-off machine is given an intermittent movement by means of a pawl engaging evenly spaced teeth on the slide. To eliminate jerking at the beginning and over-run at the end of each slide movement, the pawl was actuated by means of a crank, as shown in Fig. 22. Here the teeth on the slide are indicated at *C*, one of which is engaged with the pawl *D*. The pawl turns freely on pin *F* in the crank disk *B*. The disk is integral with the end of the continuously rotating shaft *A*, which turns in the stationary bearing *E*.

As the shaft *A* rotates in the direction of the arrow, the pawl is carried toward the right until it engages the next tooth. At this time the pin *F* is diametrically opposite the position in which it is now shown. Continued rotation of shaft *A* causes the slide to start very gently toward the left, owing to the curvature of the path through which the pin travels. The velocity of the slide, however, increases as the pin approaches the bottom of disk *B*, and decreases as it approaches the position shown, the movement at the end of the stroke being barely perceptible. Consequently, the momentum of the slide is greater at the middle of the stroke and decreases at the end of the stroke.

Automatic Indexing Head with Self-Locking Mechanism.—Automatic indexing heads are used on many special machines having work-tables of the reciprocating type. One design of head particularly adapted for these machines, especially where an unusually large number of divisions is to be indexed, is shown in Fig. 23. The work is secured by some suitable means to the left-hand end (not shown) of shaft *A*. At the end of each indexing movement, the shaft is locked to prevent rotary movement of the work during the machining operation.

The indexing head housing *B* is fastened to the reciprocating machine table. Extending from one side of this head is the shaft *C*, to which are keyed the bevel gear *D* and the forked lever *E*. The forked end of this lever engages a stationary pin *F* secured to the machine, while gear *D*

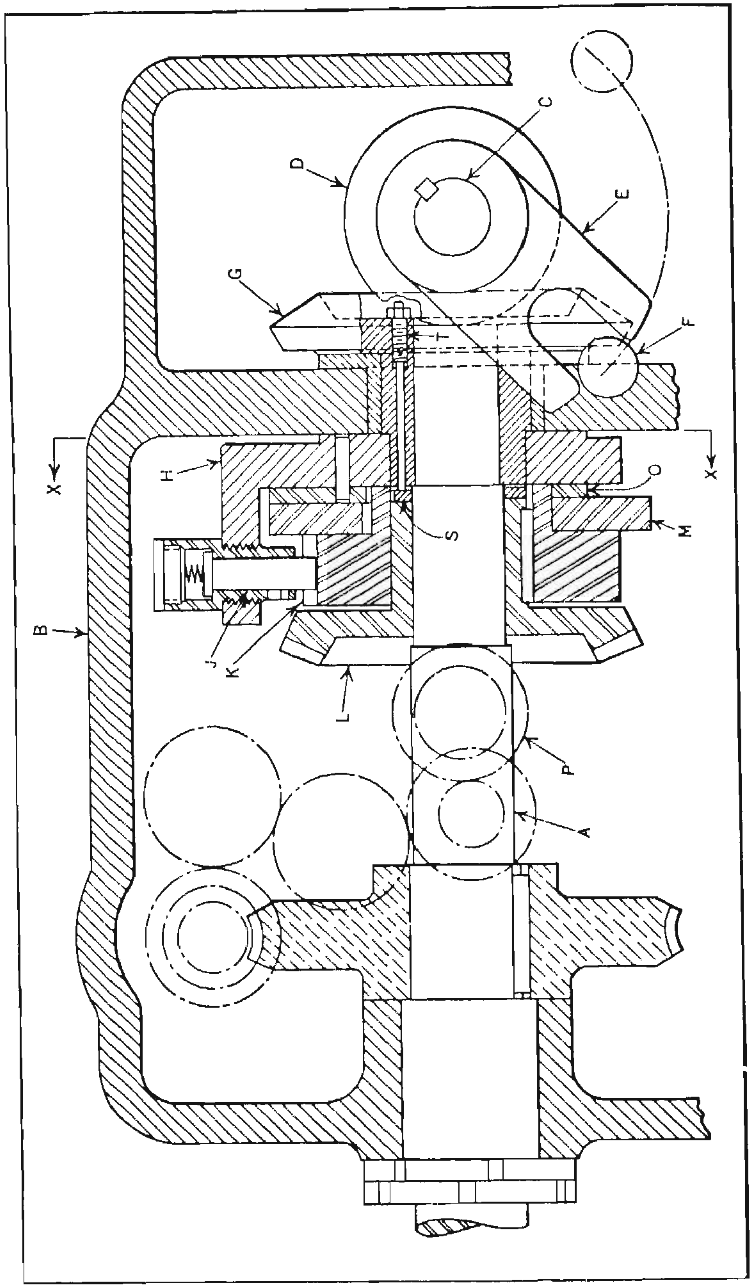


Fig. 23. Self-locking Indexing Head for Obtaining a Large Number of Divisions

meshes with gear G, keyed to the arm H. Arm H carries a spring-actuated pawl J that engages a ratchet wheel K, keyed to the bevel gear L.

A locking ring M is keyed to the ratchet wheel and, as indicated in Fig. 24, engages the plunger N in the housing B. Cam O, which is a thin plate pinned to the arm H, dis-

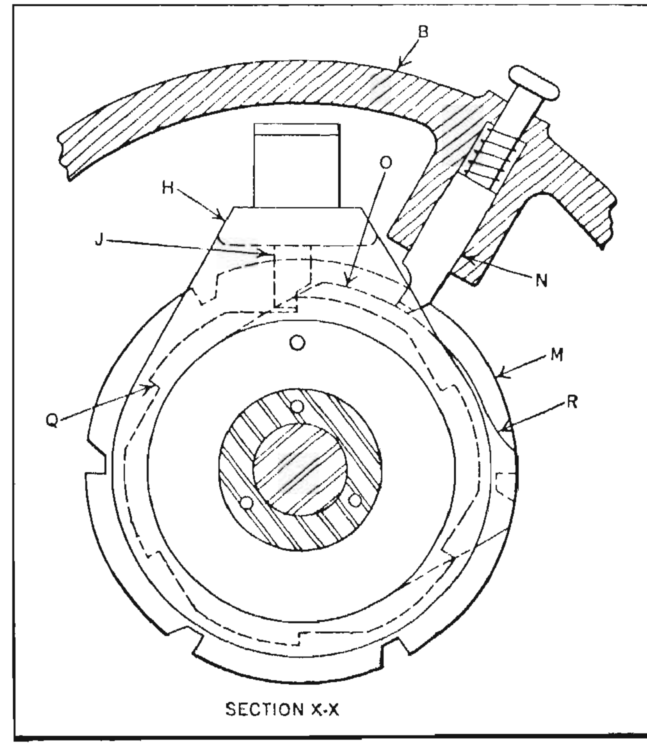


Fig. 24. Cross-section of Indexing Mechanism, Fig. 23, Showing Action of Locking Cam

engages the plunger N from the locking ring just before each indexing movement. Plunger N is wide enough to engage both the locking ring and the cam. To reduce the indexing movement so that a large number of divisions could be indexed, a combination spur and worm gear train was introduced, as indicated by the dot-and-dash lines in

Fig. 23. This gear train is operated by the bevel gears *L* and *P*.

Shaft *A* is indexed as the head moves toward the right just before the cutter engages the work, and is idle during the return stroke, while the machining is being done. Incidentally, if the cutter thrust is against the head, these movements should be reversed by mounting lever *E* and gear *D* on the other side of gear *G*. In the position indicated, the head has just completed the end of its indexing stroke toward the right. Now, as the head returns toward the left, the lever *E*, meshing with pin *F*, will be swung in a counter-clockwise direction, causing gear *G* and arm *H* to rotate about 63 degrees.

By referring to Fig. 24, it will be seen that during this movement of arm *H*, pawl *J* will be carried to the left and will engage ratchet tooth *Q*. However, just before pawl *J* engages this tooth, plunger *N* is disengaged from ring *M* by the lobe *R* on cam *O*.

On the return or indexing stroke of the table, lever *E* is swung in the opposite direction (clockwise), causing pawl *J* to rotate ratchet wheel *K*, with gear *L*, one-sixth revolution. This movement of gear *L* is transmitted through the spur and worm gear train, causing the shaft *A* to turn one division.

A washer *S*, Fig. 23, is provided to eliminate any backlash in the bevel gears and preserve the accuracy of the head. The backlash is taken up by tightening the screw *T*, thus bringing the gears into closer mesh. The number of divisions obtained with this type of head can be varied by changing the number of teeth in the ratchet wheel, by changing the gears or by varying the throw of lever *E*. This lever, with pin *F*, may be replaced by a rack and pinion.

Intermittent Motion for Feeding Wire to a Cutting-Off Machine.—Short pieces of twisted wire, approximately one inch long, are used in a certain product. Measuring and cutting off these short lengths by hand was found to be a slow and unsatisfactory process; hence the machine shown in Figs. 25 and 26 was designed to do this work automatically. It consists essentially of a mechanism for feeding the wire intermittently to two rotating shear blades.

All the working parts of the machine are mounted on a steel baseplate. Bracket *A* provides a support for the fixed shear blade *B* and also contains a double bearing for the shaft *C* (Fig. 25) on which a rotary shear blade head is mounted. It will be noted that bracket *A* is threaded to receive the two bronze bearing bushings *D* for shaft *C*. These bushings provide the necessary adjustment for setting the blades of the rotating shear head close to the stationary blade *B*. After this adjustment is made, the bearing bushings are locked in position by tightening the screws *E*.

Shaft *C* is driven by a 1/4-horsepower motor through reduction gearing, and drives the shaft *F* by means of helical gears. Shaft *F* runs free in the grooved roll *G* and in the ratchet wheel *H*. Keyed to shaft *F* is the feed-crank *K*, and connected to this crank is the link *L* (Fig. 26). This connection is made by the screw *M*, which also serves as a pivot on which link *L* oscillates. Secured to the head of screw *M* and to the link *L* is a spring, the purpose of which will be explained later. This spring, which has been omitted to avoid confusion, is of the "mouse-trap" type and is wound around the head of the screw *M*.

At the lower end of link *L* is secured the roller *N* and the feed pawl *O*. The pawl engages the ratchet wheel *H*, while roller *N* rides upon the periphery of cam *P*. Cam *P* is a running fit on the hub of feed-crank *K* (Fig. 25), and is prevented from turning by screw *Q* (Fig. 26). Roll *G* is keyed to the ratchet wheel *H*.

Operation of Wire-Feeding Mechanism.—The manner in which the feeding movement is imparted to the ratchet wheel and roll *G* is as follows: As shaft *F* revolves, the feed-crank *K* and its connecting members *M*, *L*, *N*, and *O*

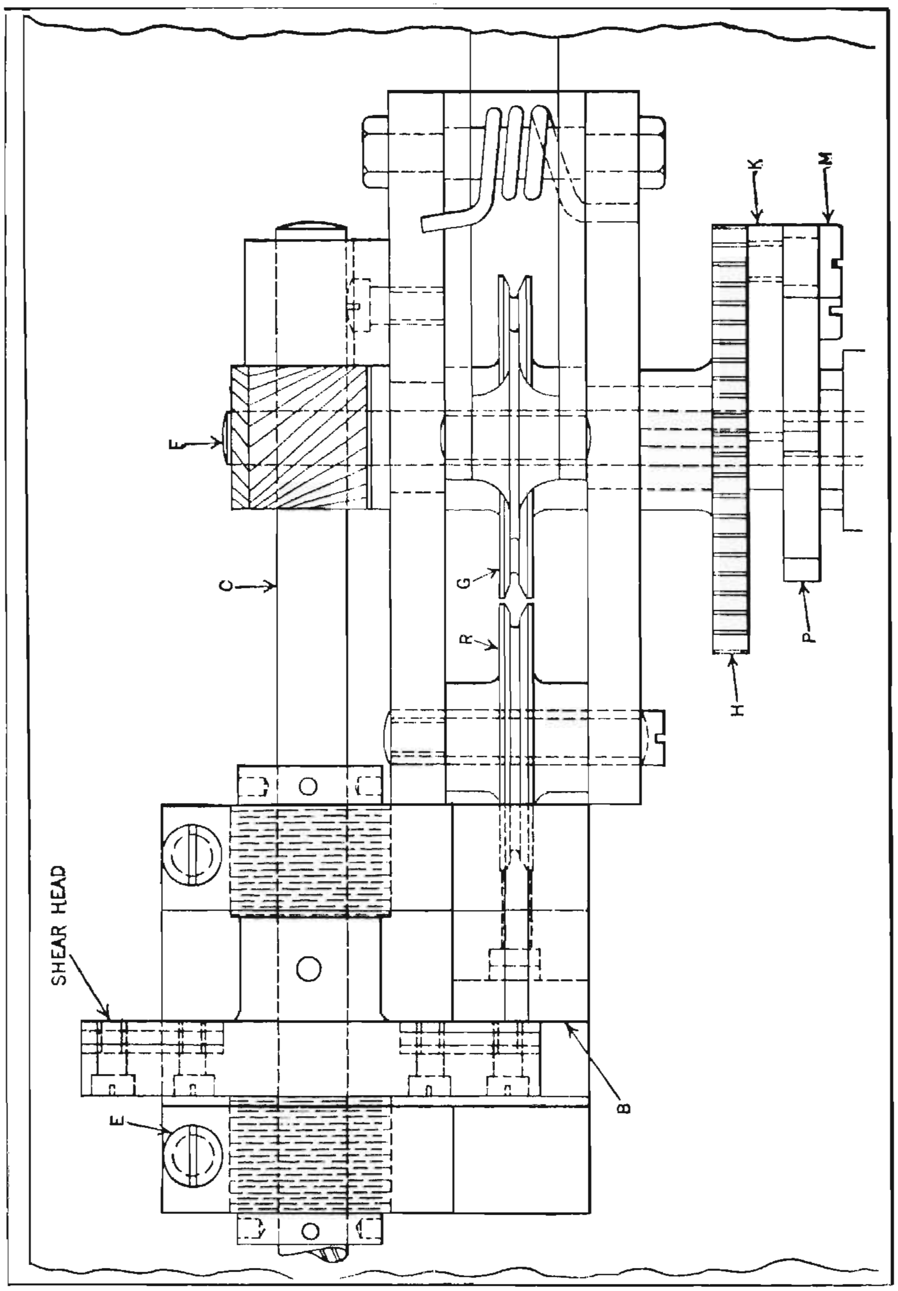


Fig. 25. Plan View of Intermittent Wire-feeding Mechanism Shown in Fig. 26

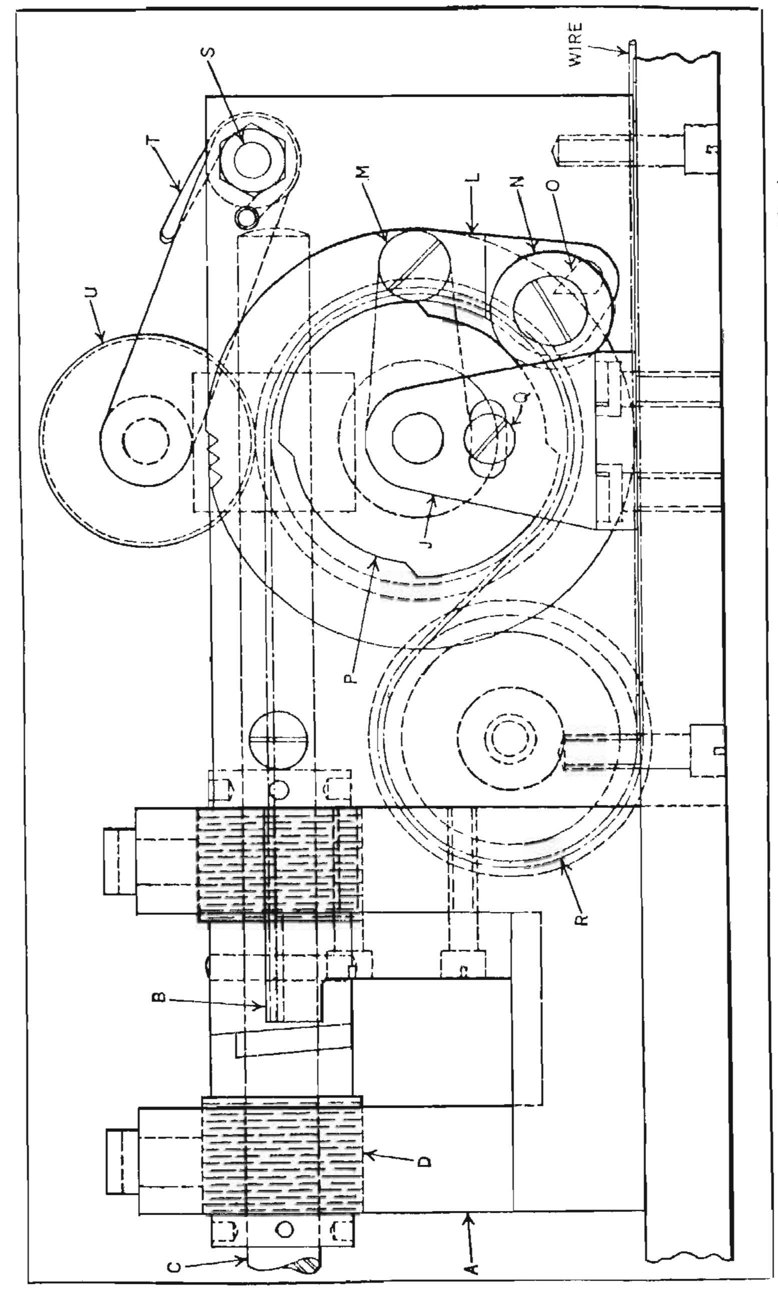


Fig. 26. Intermittent Feed Mechanism on Machine which Cuts off Wire at the Rate of 120 Pieces per Minute

are carried around cam *P*. Roller *N* is forced to maintain contact with the cam by means of the "mouse-trap" spring previously mentioned. The cam has two low places corresponding to the feeding intervals.

As roller *N* drops into these low places, the link *L* is pulled toward the center of the cam, carrying pawl *O* into engagement with the ratchet wheel *H* and thus rotating the ratchet wheel and roll *G*. This movement continues until roller *N* engages the high part on the cam and is forced outward, carrying link *L* outward also, and disengaging pawl *O* from the ratchet wheel.

In operation, the end of the wire is carried by hand under and over idler roll *R*, under and over feed-roll *G*, through a short piece of tubing (not shown) to keep it from buckling, and then over the edge of fixed shear blade *B*, where it is cut to length by the blades in the rotating head after the machine is started. Idler roll *U*, mounted on two arms pivoted on the stud *S*, serves to exert a pressure on the wire against roll *G* through the medium of spring *T*. This provides the necessary traction to pull the wire from the reel, which, although not shown, is located on the steel baseplate at the right.

Ninety-nine teeth were cut on the ratchet wheel, so that all the teeth would come into action. In this way, the wear is distributed over all the teeth. These teeth have a face angle of 15 degrees to permit the pawl to disengage readily under load. Any feeding movement from 1/99 of the circumference of the roll *G* to approximately one-half this amount can be obtained by substituting suitable plate cams.

As cam *P* has two low places on its periphery, it is obvious that two feeding movements take place for every revolution of shaft *F*. Through reduction gearing, shaft *C* operates at 60 revolutions per minute, cutting two wires at each revolution, or 120 wires per minute. Thus for a period of eight hours, the production is approximately 57,000 wires.

Indexing Mechanism with Interchangeable Turrets for Either Three or Four Stations.—A reciprocating slide on an automatic machine designed for drilling and counterboring small fiber parts is provided with two turrets which are interchangeable. One turret has three equally spaced tool stations, while the other has four stations. The three-station dial is replaced by the four-station dial when the work requires the use of four different tools. By the elimination of the extra indexing movement through the use of the three-station turret whenever possible, an appreciable saving is realized.

Referring to Fig. 27, the tool-slide is shown at *A*. Upon the slide is mounted the permanent indexing dial *B* which is free to turn on the stud *C* fixed in the slide. The four-station turret *D* is shown mounted on the dial. The pin *E*, which is a drive fit in the dial and a slip fit in the turret, prevents the turret from rotating on the dial. The turret is indexed by means of the blade *F*, which is integral with the spring-actuated plunger *G*, the blade engaging the pins *H*, *J*, *K*, and *L* in the dial.

It will be noted that the four pins *H*, *J*, *K*, and *L* are in equally spaced holes in the dial, which correspond with the four turret stations. When this turret is replaced by the three-station turret (not shown), only three pins are used in the dial. In that case, two pins are located at *M* and *N*, pin *H* remaining in the position shown. The indexing of the turret is accomplished during the idle part of the stroke indicated.

In the position shown, the slide, moving toward the right, is approaching the working part of its stroke. On continuing this movement, the tool opposite the work performs its operation, after which the slide starts on its return stroke. As pin *H* leaves the heel of blade *F* on its movement to the left, the dial is prevented from reversing its movement by the pawl *P* pivoted to the slide. This pawl engages teeth cut in the periphery of the dial. Incidentally,

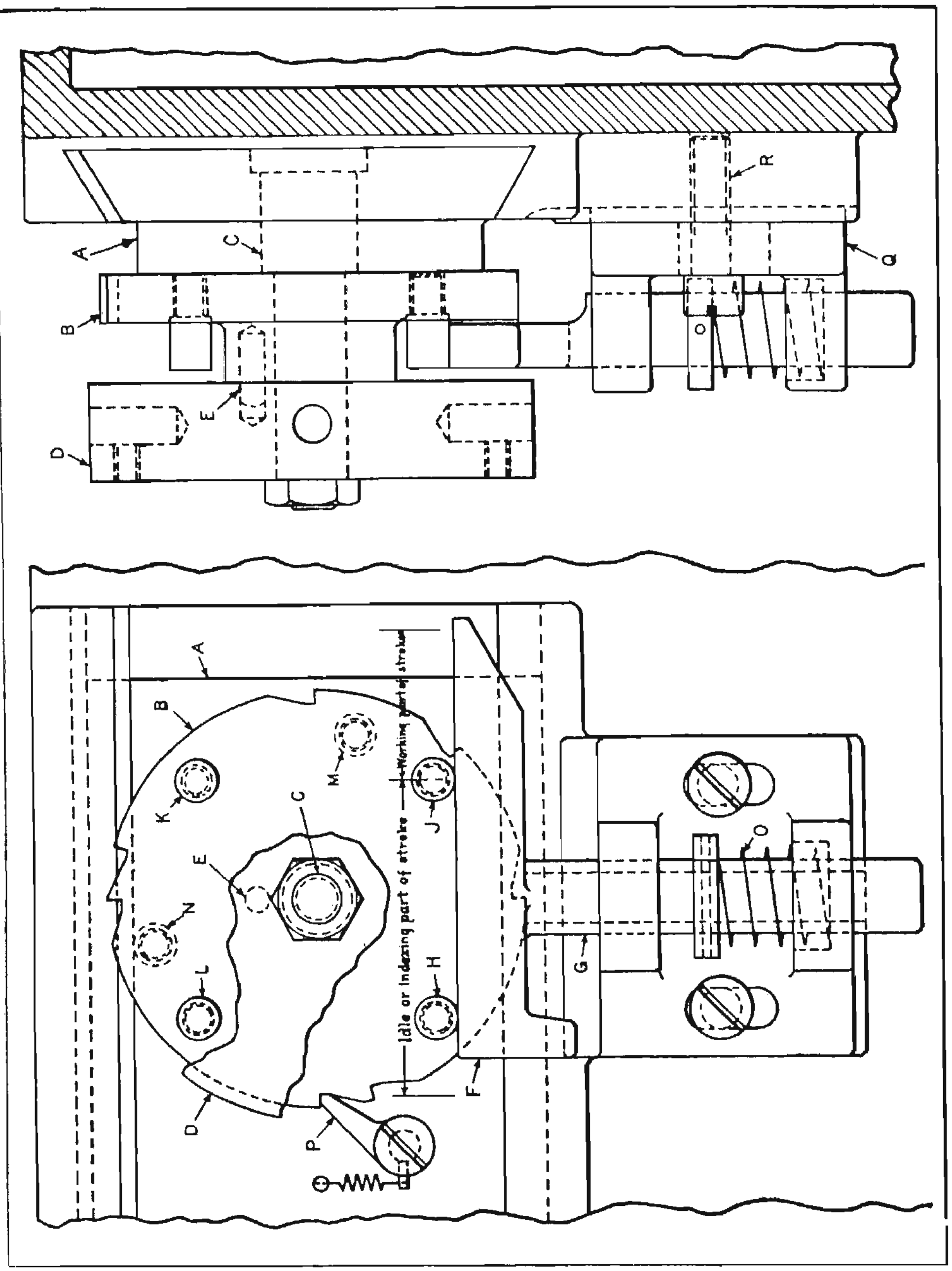


Fig. 87. Mechanism for Indexing Turret to Either Three or Four Stations

without this ratchet arrangement, the reversal of the dial would prevent the engagement of the blade with the succeeding pin, thus preventing the indexing of the dial. After pin J has left blade F, the latter is forced forward by spring O, a distance equal to about three-quarters the diameter of the pin.

Now, when the slide reverses its movement, pin J comes in contact with the left-hand end of blade F, so that further movement of the slide will cause the blade to rotate the turret. Continued rotation of the turret results in pin K coming in contact with the blade; and as the slide continues, the blade is pushed outward, so that pins J and K become located in contact with the long edge of blade F. The tool in the second turret station is now in position for performing its operation.

No means other than blade F are provided for locking the turret. The tool pins, pressing against the blade F as shown, serve to prevent the turret from rotating. This arrangement has been found entirely satisfactory for the class of work handled on the machine on which it is used. When the three-station turret is employed, it is necessary to move the bracket Q inward, so that the heel of blade F on plunger G will pass the center line of the right-hand pin when the slide is moved to its extreme left-hand position. To make this adjustment, it is only necessary to loosen the screws R which lock the bracket in place.

CHAPTER IV

INTERMITTENT MOTIONS OF THE GENEVA TYPE

The Geneva type of intermittent motion is based upon the principle of the Geneva stop which has been applied to watches, etc., to prevent winding the main spring too tightly. This stop mechanism, as the name implies, is intended to prevent rotation after a certain number of revolutions. This is not the case, however, when the principle of the Geneva stop is applied to intermittent gearing. Geneva wheels or mechanisms are used to transmit an intermittent motion to some driven member at regular intervals which may be repeated indefinitely, as, for example, when some part of a machine tool requires indexing or rotating through some fractional part of a revolution at certain intervals while the machine is in operation.

Geneva Motion Designed to Reduce Rate of Acceleration and Deceleration of Driven Member.—The Geneva stop mechanism is used frequently because of its simple design and serviceability. In the form generally used, the driving roll follows a circular path. With this arrangement, the disk begins its movement from a stationary position and comes to a stop without shock, but the acceleration and deceleration in the velocity of the disk occur at a rapid rate, producing a relatively high angular velocity in the rotating disk. In order to eliminate these disadvantages, a German inventor developed a modified form of Geneva stop mechanisms in which the driving roll that transmits intermittent motion to the cross or slotted disk is operated by a mechanism consisting of four articulated members, as shown at *A*, Fig. 1.

In this mechanism, the driving member *D* rotates on

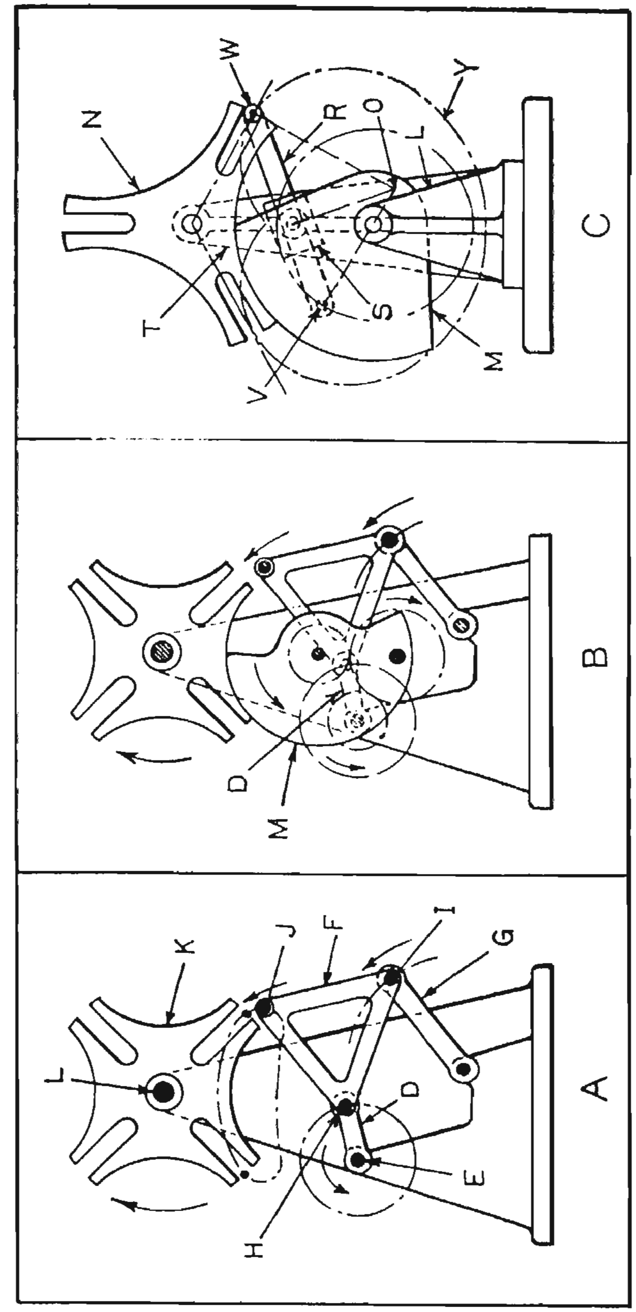


Fig. 1. Geneva Mechanisms Designed to Reduce Rate of Acceleration and Deceleration of Driven Member

axis E, and by means of rod F, gives member G a swinging motion. As the center of the connecting stud H describes a circle and stud I moves through only a part of a circle, all other points on the rod or member F describe curves of a distinct form. A stud at J supports the driving roll for the Geneva stop mechanism. When crank D completes a full rotation, roll J enters a slot in disk K and drives the disk to the next stopping position, after which it leaves the slot.

The difference between this mechanism and the older well-known arrangement is that the height of the curve followed by the roll on stud J is not so great; thus the angular velocity of the disk K, which depends on the distance of point L from the top of the curve, is considerably reduced. To prevent any unintentional movement of disk K, a blocking disk is necessary. For this purpose, a disk M, as shown in view B, is supplied. This disk is driven by intermediate gears from crank D. The addition of this blocking system, however, considerably complicates the mechanism. Another disadvantage of this drive is the bulky unsymmetrical design.

Another similar drive which functions through a turning block linkage is shown at C. The small fixed bracket L forms the bearing for the shaft of the driving crank M. This crank-arm also serves as the blocking disk for holding the driven disk N stationary during the dwelling periods. The crank-arm M is connected at V to the rod R, which slides in a block S, pivoted on the stand T. On the opposite end of rod R is mounted a roller W. As shown in the illustration, roller W describes a heart-shaped curve Y. The upper or spear-shaped portion of the curve Y is used for imparting the driving movement to the three-armed cross or driven disk N.

As roller W enters the slots in disk N, tangents to the path followed by the roller at this point must pass through the center of the disk and the center of the slots, which are

radially located. The normal of curve Y is found by extending a line from V through the center of the shaft on which M is mounted, so that it intersects a line perpendicular to rod R, drawn from the center of the fixed stud on which block S is pivoted. A line from the point of intersection O to the center of roller W forms the desired normal to curve Y.

Combined Geneva and Intermittent Gear Movement.— In order to modify the operating characteristics of the

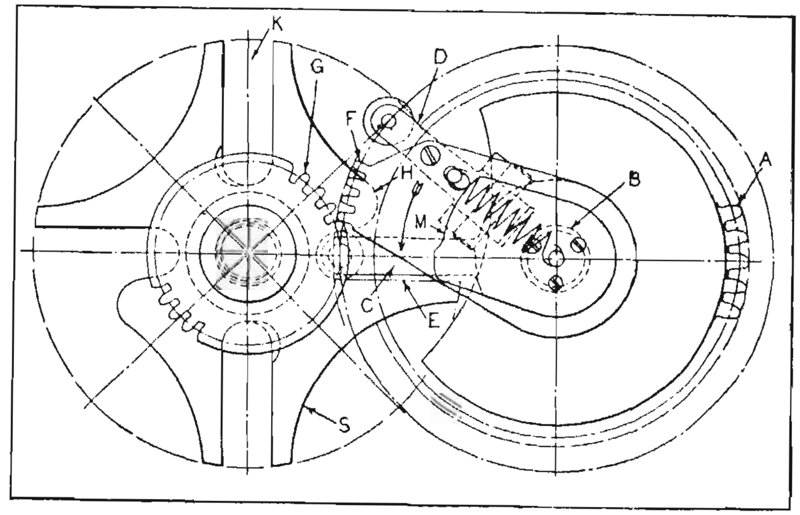


Fig. 2. Intermittent Drive Mechanism Designed to Accelerate and Decelerate Motion at Start and Finish of Driving Movements

well-known Geneva gear movement to adapt it for a particular purpose, intermittent gearing was incorporated in the design, as illustrated in Figs. 2 and 3. After laying out the design on the drafting board, a model was made which operated satisfactorily. The mechanism consists of a modified double driving arm Geneva wheel with intermittent gear segments. The gear segments are so placed that they transmit a practically uniform speed movement to the driven member from the instant the driving arm ends its accelerating movement until the second driving arm begins its decelerating movement in stopping the driven member. One advantage of the mechanism, in its application to an automatic machine, is that the driver requires a movement of only about 130 degrees to rotate the driven member 180 degrees. This leaves 230 degrees of the driver cycle

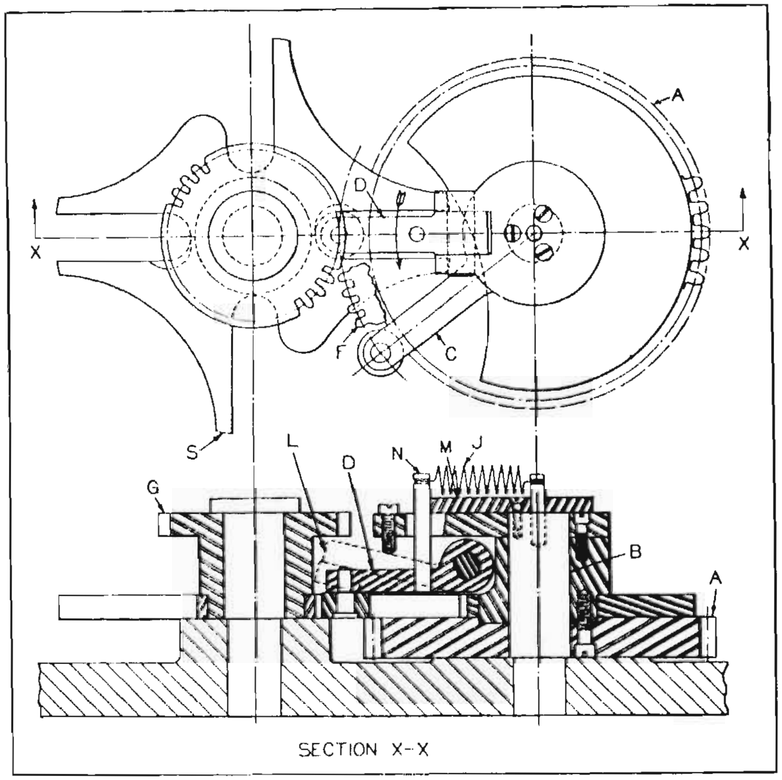


Fig. 3. Mechanism Shown in Fig. 2 with Various Members in Different Operating Positions

free to perform other useful work or operations while the driven member dwells.

Referring to the illustrations, *A* is the driving gear, which operates at a uniform speed. The whole driving unit is mounted on the stationary stud *B* and rotates in the direction shown by the arrow. The first driving arm *C* is

integral with the driving member, while the second arm *D* is pivoted to it and suitably spaced from the first arm.

When arm *C* engages slot *E* in the driven spider *S*, it will start rotation of the latter member and accelerate its speed until arm *C* reaches the center line between the two members. At this point the intermittent gear segment *F* meshes with its mating segment *G* on the driven member. As the pitch line of the intermittent gears corresponds with the center line of the path in which the arm rollers rotate, the gears continue the motion of the driven member at approximately the same speed as was attained by the roller arm *C* at the instant it passed the line between the centers of the driving and driven members. The slot or arm on the far side of the spider *S* is shortened and so shaped at *H* that the roller cannot interfere with the uniform motion imparted by the gears as the roller recedes from the slot.

The ratio of the intermittent gears is such that the driven gear *G* will rotate 90 degrees while the gears are engaged, the remainder of the 180-degree movement being derived from the two driving movements of 45 degrees each, imparted by the accelerating arm *C* and the decelerating arm *D*. The latter arm, because of its pivoting feature (see Fig. 3) and the tension of the spring *J* is held out of engagement with its slot *K*, as indicated by the dotted lines *L*, until just before it reaches the center line, when the action of the lobe of the stationary cam *M* on the pin *N* forces the arm down into engagement with its slot. This engagement occurs at the instant when the intermittent gears pass out of engagement. The Geneva gear action of the arm *D*, in its further rotation, decelerates the driven member to a stop 180 degrees from the point where the accelerating arm *C* started its rotation.

Inverse Geneva Wheel Motion.— The term "inverse" is applied to an unsual form of the well-known Geneva mechanism for producing intermittent circular motion, because the driving and driven members rotate in the same direction, whereas with the usual form of Geneva motion, the rotations are reversed. The arrangement is such that the driving crank axis and the crank circle are entirely within the radius of the plate or driven member, and this produces a vastly different effect in the timing, acceleration, and the velocity of the plate. These effects are things to be considered in applying the mechanism to a machine design. In some designs, the effects produced may not be altogether desirable, while in others they may have distinct advantages and introduce an improvement.

The inverse Geneva stop or wheel motion was developed to fill the requirements of a particular type of drive for feeding strip stock into power press dies. Since its inception a variety of successful applications in automatic machinery have been made.

Typical forms of the inverse Geneva wheel are shown in Figs. 4 and 5, the former showing a three-station and the latter an eight-station plate. The essential parts are few and simple, consisting of a constant-velocity driving crank C and a variable-velocity driven member D, called the plate. The plate rotates in equal intermittent movements from station to station, stopping for a short interval of time at each station. As the rotation of the plate is caused by the motion of the crank-pin roller E in passing through radial grooves in the plate surface, the number of stations is dependent upon the number of grooves.

The smallest number of radial grooves with which a Geneva mechanism will function is three. The greatest number is infinite, being limited only by the diameter of the plate and the width of the grooves, both of which may theoretically be made to any proportions. In actual practice, however, the number of grooves required is not very great.

Working and Idling Angles of Driver Rotation.—By comparing Figs. 4 and 5 it will be seen that as the number of grooves increases, the working angle a of the driving

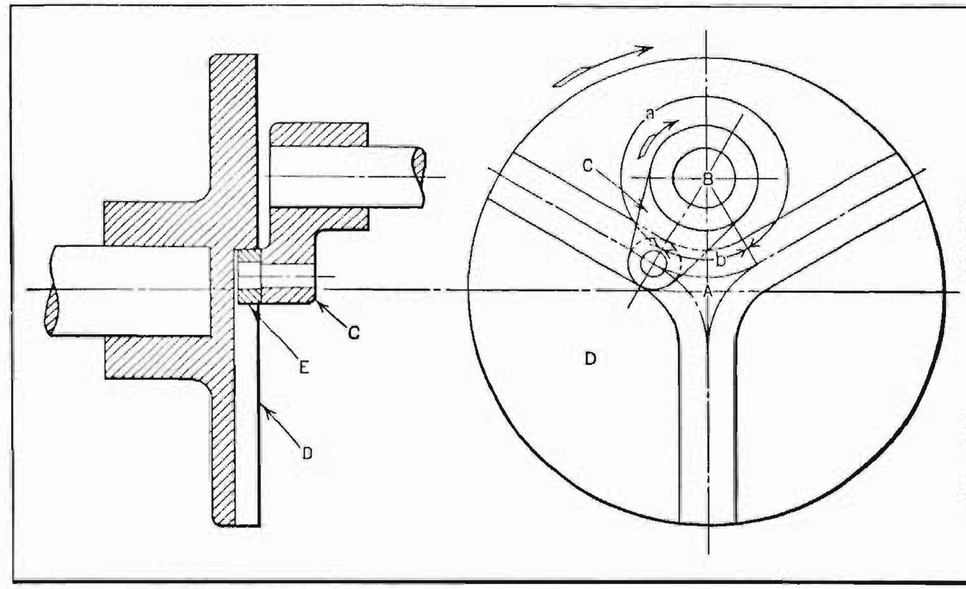


Fig. 4. Three-station Inverse Geneva Wheel Mechanism

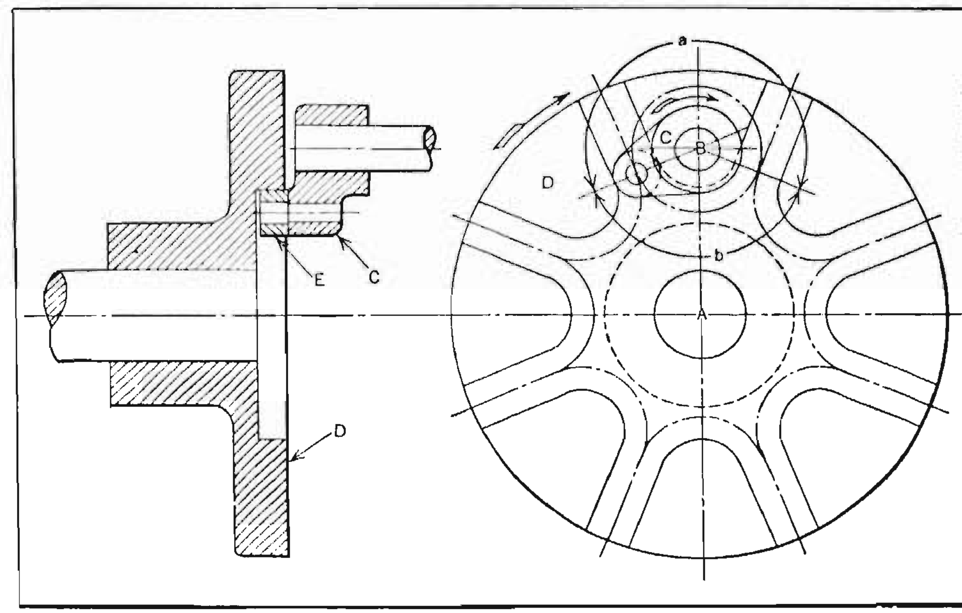


Fig. 5. Inverse Geneva Wheel Mechanism with Eight Stations

crank decreases and the idling angle b increases. These angles are determined as follows: Referring to Fig. 6, the angle s between two adjacent radiants on the Geneva plate is equal to 360 degrees divided by the number of radiants N. As the roller enters and leaves the grooves when the crank center line is at right angles to the radiants, two equal triangles are formed by the lines AEB and $AE'B$, from which it is seen that angle b equals $180 - s$. Then angle a equals $360 - (180 - s) = 180 + s$.

Fig. 6 also shows some of the practical points to be considered in the design of an inverse Geneva wheel or stop. In both Figs. 4 and 5 the inner ends of adjacent grooves are joined by a circular arc which is concentric with the crank circle. This arc is of little or no use, and to facilitate machining, it is preferable to connect the grooves with straight lines, as at h, Fig. 6. The corners should be broken by a small radius to permit the roller to enter the grooves more easily.

Locking the Driven Member.—In any sort of intermittent motion device it is desirable, and usually necessary, that some means be provided for locking the driven member in position while it is at rest. The locking feature employed in this mechanism is shown in Fig. 6. In this illustration, a circular arc lobe, machined concentric with the driving crank axis, is shown as an integral part of the crank at d.

During the idling period of the driving crank, this lobe is in contact with one of the locking segments e, which are made to project from the face of the plate and are machined to the radius of the lobe just described. This feature prevents any accidental rotation of the plate while it is in one of the rest positions. The angles f and g subtended by the arcs on e and d, respectively, are equal, and their magnitude is a matter that should be given careful consideration.

The angles should be made as large as possible in order to keep the plate locked during the entire time that it is

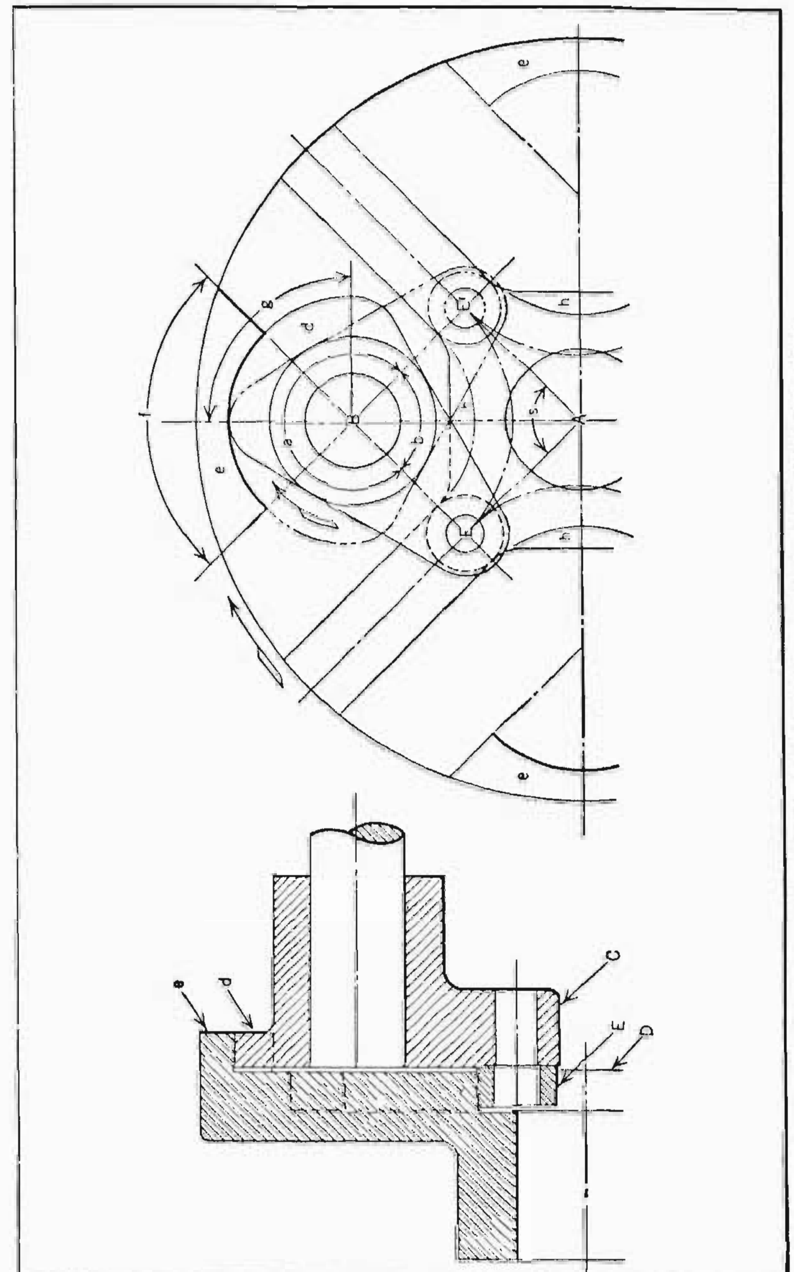


Fig. 6. Locking Arrangement Employed on Inverse Geneva Wheel Mechanism

not in motion, but making them too large will result in interference of the parts. The best results are obtained by making angles f and g each equal to angle b. This permits the locking action to begin the instant the crank roller leaves one groove and to end the instant it enters the next groove. This, it will be seen, is the case in Fig. 6. The heavy full outlines show the crank in the act of entering a groove, while the left half of the locking segment is just being cleared by the lobe on the crank so that rotation may begin in the plate. The light dot-and-dash outlines show the crank in the act of leaving the groove, with the lobe engaging the left half of the locking segment.

A locking segment e is placed midway between each two grooves, as shown. Their centers represent the relative positions of the crank during the idling interval between working periods. The radius of the locking segment is more or less arbitrary, but it will be limited by the radius of the plate. The locking segments do not add greatly to the cost of manufacturing the plate, because their shape is quite simple. In fact, the structural lines of the entire plate are made up of simple geometrical figures and are easily machined without the use of templets or masters.

Geneva Type of Work-Reversing and Transfer Mechanism.—Many types of automatic machines must be provided with means for turning over or reversing the position of the work at some point during its progress through the machine. This is accomplished very effectively in one case by the mechanism shown in Fig. 7. The work at A is turned over and transferred to position B. To accomplish this, the work A is fed in the direction indicated by arrow C to the position D in the reversing mechanism E. A plate F in this mechanism holds the work by means of pressure applied by two springs, only one of which can be seen in the illustration. Attached to the reversing mechanism is a gear H which is mounted on a shaft J.

The Geneva mechanism shown below gear H turns the

entire unit through an angle of 180 degrees in the direction indicated by arrow K each time it functions. This movement transfers the work from position D to L. When the next piece is pushed into position D, it comes in contact with pad N on plunger M, pushing it to the right. Thus the pad P on the other end of the plunger pushes the work into the position indicated at B. This action is repeated

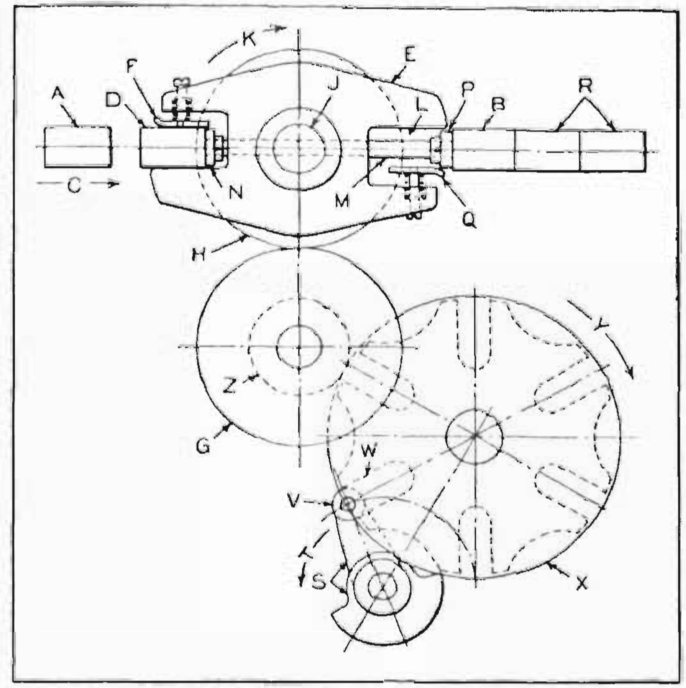


Fig. 7. Mechanism for Reversing Work D and Transferring it to Position B

at each cycle of the machine causing the work, which has been reversed, to be pushed along, as shown at B and R. The indexing is accomplished by means of a Geneva movement, in which the combination lever and locking segment S revolves in the direction indicated by arrow T through one complete revolution for each 180-degree indexing movement of member E.

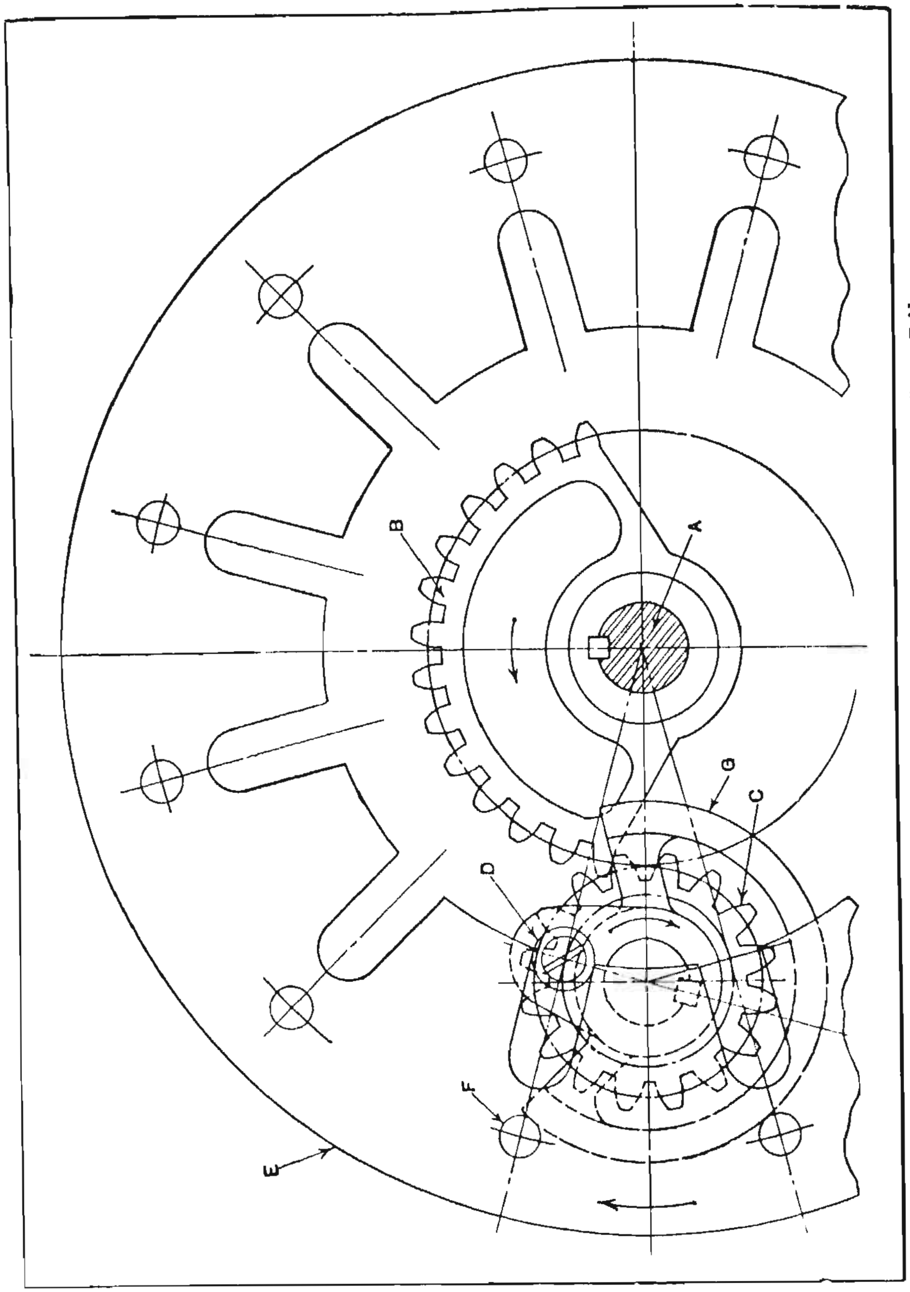


Fig. 8. Mechanism for Producing Intermittent Rotary Motion for a Heavy Table

The roll *V*, at each revolution, engages one of the slots *W* in the plate, causing the large spur gear *X* to revolve in the direction indicated by arrow *Y* through one-sixth of a revolution. The speed ratio between gear *Z* and gear *X* is three to one. Thus gear *G* is revolved anti-clockwise one-half revolution, causing gear *H* to revolve one-half revolution in the direction indicated by arrow *K*. This completes one cycle in the operation of the automatic machine.

Segment Gear and Geneva Wheel for Intermittent Rotary Motion.— The mechanism shown in Fig. 8 was designed to give a large heavy table or turret an intermittent rotary motion. The drive shaft *A* carries a gear segment *B* which contains just enough teeth to cause gear *C* to make one revolution. On the same shaft with gear *C* is a crank carrying a roller *D* which engages slots in plate *E*. Plate *E* is so mounted on the table or turret that it is free to revolve.

Plate *E* carries pins *F* which are engaged by the rim or periphery of the circular segment *G* which is a part of the crank. This action locks the plate *E* in position while the roller is out of engagement with the slots. By varying the number of teeth in the pinion, the size of the segment, and the number of slots, the length of the dwell period can be increased or decreased. During the dwell period, while segment *B* is out of contact with the gear *C*, the roller *D* remains in the position shown. Very little shock occurs when the segment comes into contact with the gear. This mechanism is adapted for moving heavy loads, as the power is applied near the periphery of the table.

Locking Driven Wheel of Geneva Movement.—The modified Geneva movement shown at the left in Fig. 9, illustrated and described in Volume I, "Ingenious Mechanisms for Designers and Inventors" (page 74), provides positive locking of the driven member between the indexing movements. The locking is accomplished by having one or two of the rollers *R* engage the annular groove *G*. The roller P, carried by the driving disk B, is shown about to leave its slot, having completed the indexing of shaft S. The roller R, shown entering the groove G, serves to lock the disk and shaft S until the next indexing movement. This mechanism has the disadvantage of being rather large. Also, roller R is so located that the driver B must be of such a large diameter D that it would interfere with the shaft S if it were extended through the driven wheel. This construction necessitates placing the driven wheel at the end of the shaft, thus preventing the use of an outboard bearing.

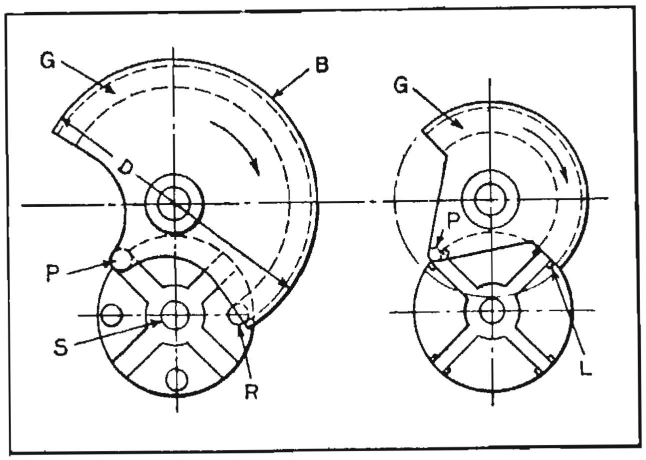


Fig. 9. Examples of Modified Geneva Movements

These objections have been overcome by the arrangement shown by the diagram at the right in Fig. 9, in which the locking of the driven disk is accomplished by lugs L which extend beyond the radial grooves of the driven member, so that they are engaged by the groove G in the driving member. Rollers can be substituted for the lugs L, but they are more expensive. It will be noted that the same diameter of driven wheel requires a driver of much smaller diameter than the mechanism shown at the left. It will be noted also that the lugs L are located in the most effective positions for locking the driven member, whereas the rollers R are so positioned that they lose about 30 per cent of their effectiveness. In other words, a given clearance between the locking members will permit more play or looseness in the case of the mechanism shown at the left. The loads on the locking pin required to resist a given torque will be about 50 per cent greater than on the lugs L.

Another advantage of the improved mechanism is that it is easier to make, especially in shops not accustomed to handling precision work, because the groove G extends for exactly half the driver area, or through an angle of 180 degrees, while the groove G of the other mechanism is somewhat over 180 degrees and must be carefully calculated and laid out. The rollers R must not only be accurately spaced between the grooves of the star wheel of the driven member, but they must also be accurately located at the correct distance from shaft S. On the other hand, the lugs L can easily be centered on the slots in the star wheel, and their location from the center is also easily accomplished. Still another advantage is that by cutting away a little material on the star wheel and modifying the arm of the driver that carries the roller P, either the driver or the driven disk can be assembled or dismantled without disturbing its mating part by sliding one part past the other.

Application of Geneva Wheel to Turret Indexing.—A well-known method of indexing the turrets of automatic machines is by the use of the principle of the "Geneva" motion. This has the advantage of giving a slow starting movement which gradually accelerates and then slows down before reaching the stopping point, thus securing rapid indexing and at the same time avoiding shock. An example illustrating the application of a Geneva wheel to turret indexing is shown in Fig. 10. In this case, the pin A engages the slots in the disk B to index the turret. The cylindrical portion of the pin carrier C engages concave portions of the disk B to locate the turret approximately; the automatic spring-operated latch D accurately locates the turret by engagement with notches in the large dividing wheel E. The turret is afterwards locked by a sliding steadyrest. This indexing mechanism gradually accelerates the heavy turret at the time of indexing and

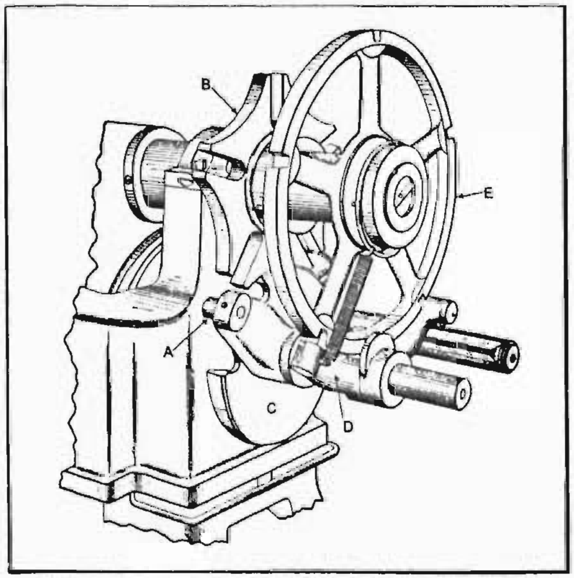


Fig. 10. Turret Indexing Mechanism of the Geneva Type

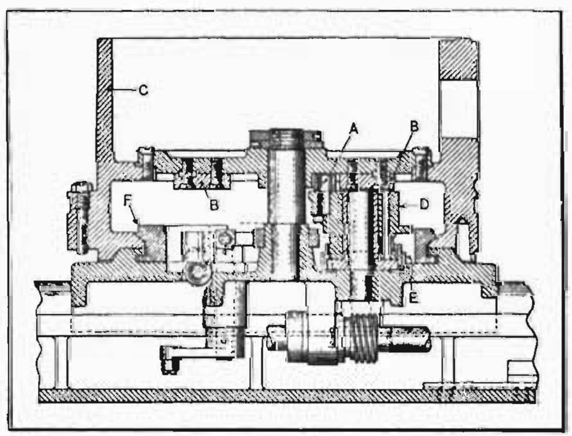


Fig. 11. Another Turret Indexing Mechanism of the Geneva Type

then gradually checks its momentum. Fig. 11 shows another application of the Geneva drive, in which a roller A engages slots formed between blocks B for indexing the turret C. The roller is carried on a sleeve D which is intermittently turned by gear E.

Graphical Analysis of the Geneva Mechanism.—In designing a Geneva mechanism for intermittently indexing a shaft or some other machine member through part of a revolution, it is frequently difficult for the designer unfamiliar with the mechanism to study its action. The following analysis is presented with a view to making the study of this mechanism easier. In the analysis it will be shown that a pair of imaginary arms connected by an imaginary link can be substituted for the Geneva mechanism and be kinematically identical with it. This is true for every point of the working range of the motion. The imaginary arms and connecting link will be of varying lengths at the different points of action, but at every point will be subject to the very simple laws covering the action of link work.

Fig. 12 shows in outline a typical form of the Geneva transmission at the beginning of the cycle. For simplicity, four slots are shown in the driven wheel N, although this analysis is equally applicable to any number of slots. The driving arm is shown at M, the center of the driving arm at A, the center of the driven wheel at B, and the center of the roller at E. The same mechanism is illustrated in Fig. 13 at an intermediate point in the cycle; here the letter O represents the imaginary driving arm, P the imaginary driven arm, and Q the imaginary connecting link. The imaginary arms and link are laid out as follows: Draw a line connecting E, the center of the roller, and B, the center of the driven wheel. Passing through E, draw the normal ED, and through A draw a normal to ED, intersecting at D. The imaginary driving arm is now length AD, the imaginary driven arm EB, and the imaginary con-

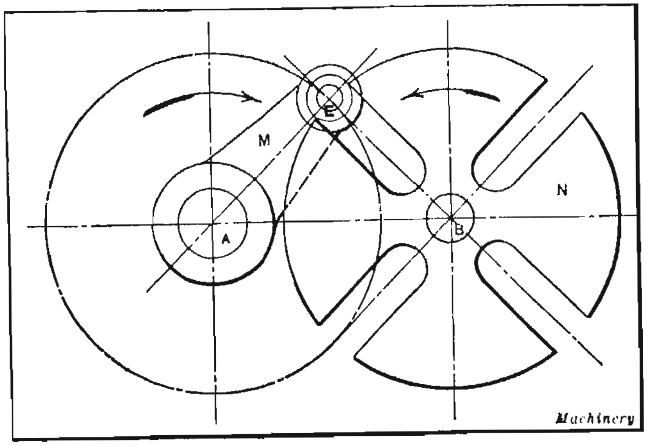


Fig. 12. Outline of a Typical Form of the Geneva Mechanism at the Beginning of a Cycle

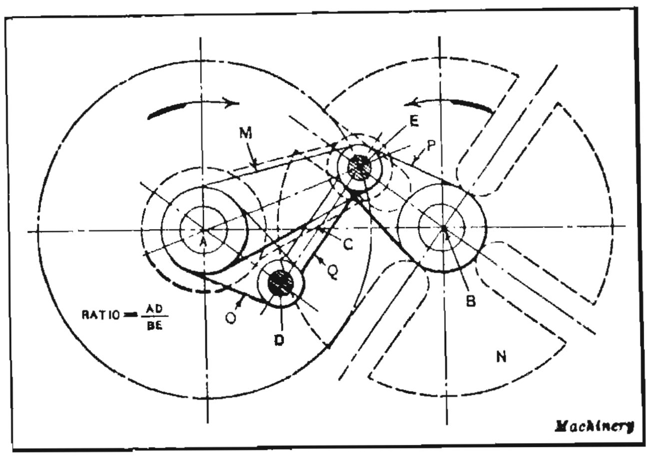


Fig. 13. Geneva Mechanism, with Imaginary Arms and Link that are Kinematically Identical with the Geneva Motion

necting link, ED. This imaginary linkage system kinematically replaces the Geneva mechanism for the point of the cycle at which E is in this illustration.

Fig. 14 shows the mechanism laid out with center E at a point still further advanced in the cycle. It will be noted from this illustration that the imaginary driving arm O has lengthened and that the arm P and link Q have short-

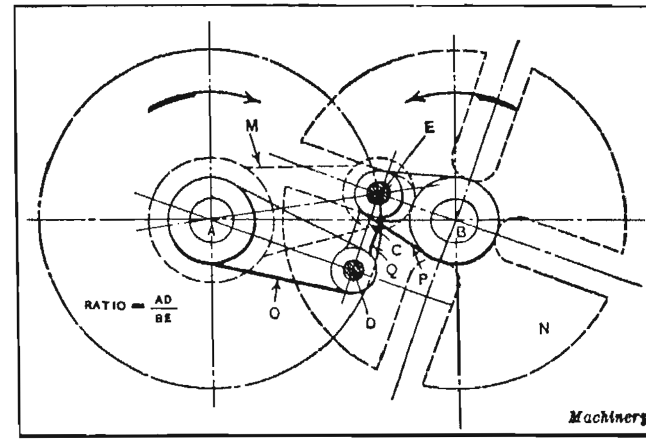


Fig. 14. Illustration Showing Geneva Mechanism Further Advanced along its Cycle

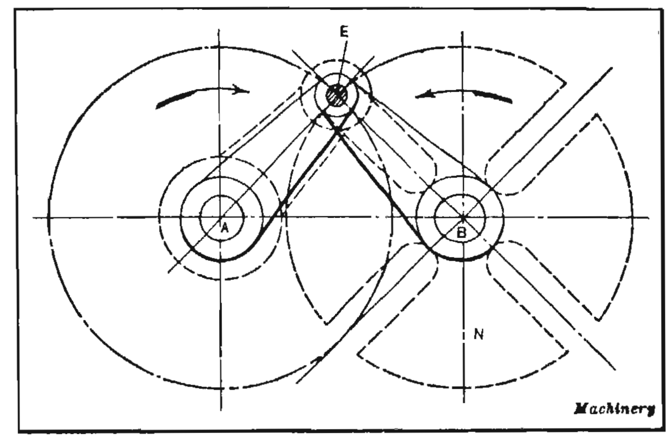


Fig. 15. Initial Position of Geneva Transmission, at which Point the Velocity Ratio is Zero

ened. In Fig. 16, the mechanism is shown at the middle of the cycle, where the imaginary driving arm attains its maximum length and coincides in length and position with the actual driving arm M. The imaginary driven link has shortened to the minimum, coinciding in position and length with line BE of the driven wheel, and the length of the connecting link is now zero. At this point the velocity ratio is at the maximum.

At the initial position of the Geneva transmission, which is illustrated in Fig. 15, the length of the imaginary driving arm is zero and the length of the driven link equal to BE, the maximum acting radius of the driven wheel. The length of the imaginary connecting link is also zero. In this case, the velocity ratio is evidently zero.

Determining the Velocity Ratios at Intermediate Points.— It is now in order to show the method of finding

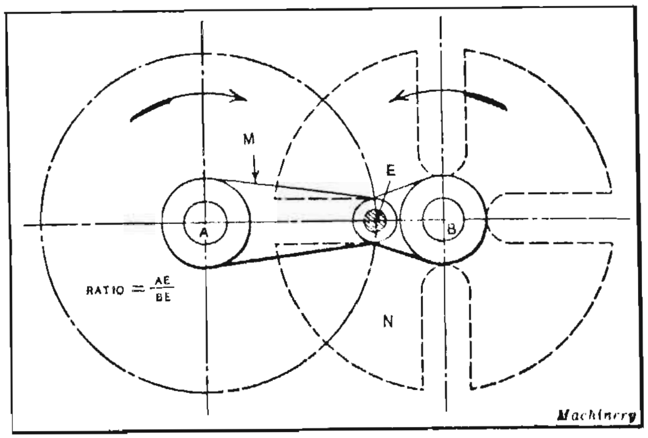


Fig. 16. Mechanism at Middle of Cycle, where Velocity Ratio is at the Maximum

the velocity ratio at intermediate points. In Fig. 13, the ratio is, by the law of leverages, $\frac{AD}{BE}$, as the connecting link is normal to both levers, and in Fig. 14 the velocity ratio is also $\frac{AD}{BE}$. (Compare with a pair of pulleys of radius AD and BE, connected by a belt DE). By laying out a number of lever systems, as in Figs. 13 and 14, the velocity curve of a Geneva mechanism can be determined for as many points as desired.

There is, however, a more direct method of determining the velocity ratio. In Figs. 13 and 14, the triangles ADC and BEC are similar by construction; therefore $AD:BE::AC:BC$. Hence $\frac{AD}{BE}$ equals $\frac{AC}{BC}$, and the velocity ratio is equal to $\frac{AC}{BC}$. Lines AC and BC are the segments into which the imaginary connecting link Q divides the line of centers AB.

Laying out the Velocity Curve.— To lay out the velocity curve, first divide the circumference EE_1, Fig. 17, into any number of parts, preferably equal. Then from B draw lines BX, BY, etc., through these points spaced out on the circumference. From these lines BX, BY, etc., draw normals 40-40, 35-35, etc., intersecting the points on the circumference and the line of centers AB. Now, assuming that A-15 along the line of centers measures 2.4 inches, and B-15, 6 inches, the velocity ratio at point 15 on the circumference equals $2.4 \div 6$ or 0.4. Compare this result with the velocity curve in Fig. 18.

Now prepare for laying out the velocity curve by erecting ordinates on the base line 0-90, Fig. 18, at equal distances apart. Lay off on each ordinate a distance corresponding to the quotient obtained by dividing $\frac{A\text{-}5}{B\text{-}5}$, $\frac{A\text{-}10}{B\text{-}10}$, etc., and connect these points. The resulting curve will be tangent to the base line at 0 and 90 and tangent to a line parallel to the base at the vertex, as shown. The velocity begins at zero, gradually increases to a maximum at the vertex of the curve, and then gradually diminishes till at the end of the cycle it again becomes zero.

Using the Velocity Curve.— If it is desired to find the point at which a velocity of 100 per cent occurs, bisect the line of centers AB, as in Fig. 19. On these segments draw the semicircles ADC and CEB. Through point E where one

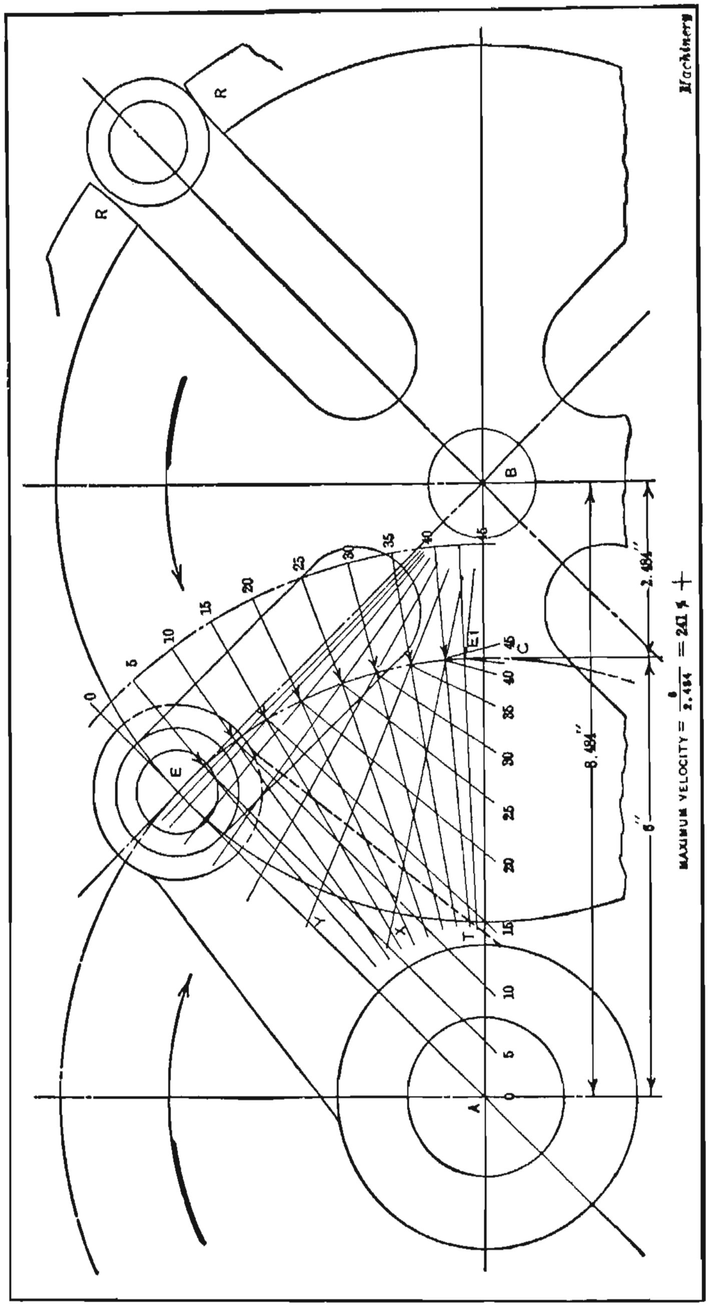


Fig. 17. Procedure Followed in Graphically Determining the Velocity of the Driven Member at Various Points along the Path of the Roller Attached to the Driver

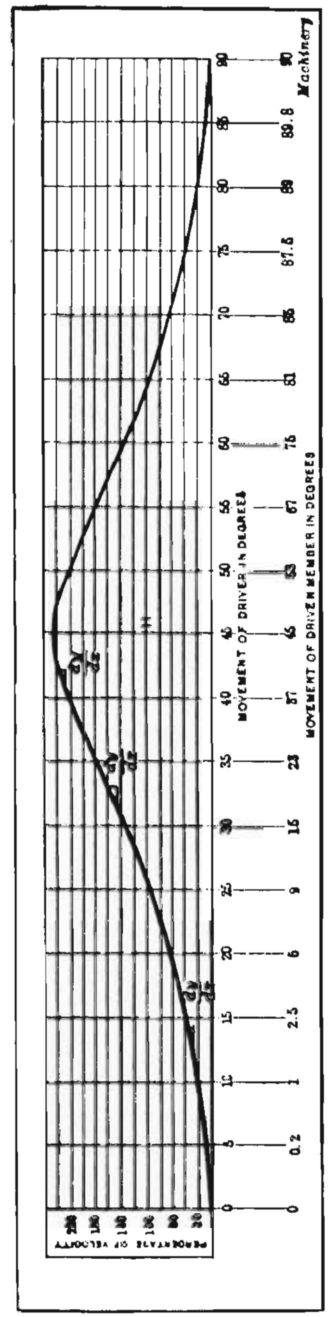


Fig. 18. Curve which Shows the Velocity Ratio Between the Driving and Driven Members of a Geneva Mechanism at Different Points along the Path of the Roller Attached to the Driver

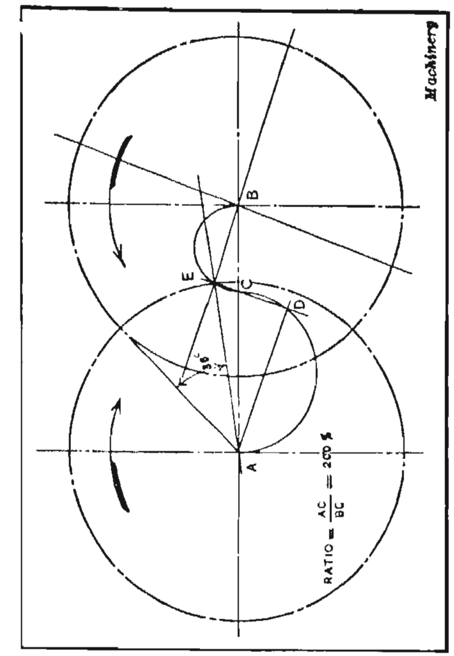


Fig. 20. Construction Used to Determine at which Point a Velocity Ratio of 200 Per Cent is Obtained Between the Driving and Driven Members

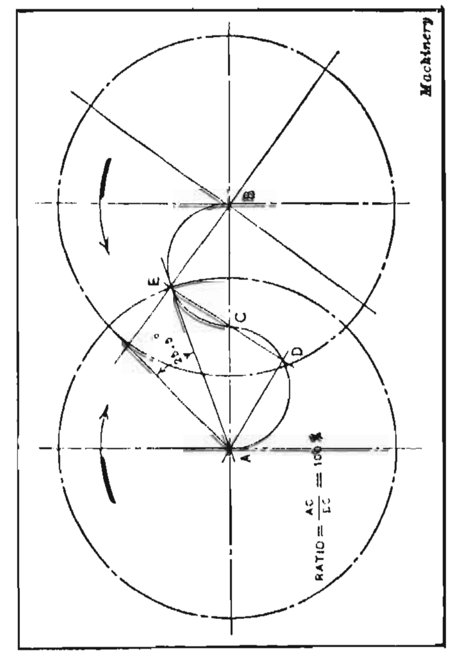


Fig. 19. Method of Determining at which Point a Velocity Ratio of 100 Per Cent is Obtained in the Movement of the Driving and Driven Member

semicircle intersects the path of the center of the roller, draw the line *ED*, intersecting the line of centers *AB* through *C*. The angle *CEB*, being drawn on a diameter, will be a right angle. From *A* drop a normal *AD*, intersecting *DE*. The velocity ratio is $AC \div BC$, which equals 1.

The construction for the 200 per cent ratio is shown in Fig. 20; in this case, the line of centers is trisected. Horizontal lines may be drawn intersecting the velocity curve in such a manner as to afford instant means for determining the velocity at any desired point. A ready method is to determine by the means suggested in the preceding paragraph, the height of ordinate *H*-45, Fig. 18, for a 100 per cent increase. Then divide *H*-45 into five equal parts, each of which will be equal to 20 per cent. Draw horizontal lines through these points of division and where these lines intersect the velocity curve, the velocity percentage will be known. The whole range of the velocity curve can be treated in this manner. It will be obvious that the velocity, after the driving member has moved 10 degrees, is found at the intersection of the ordinate 10 and the horizontal line marked 20 per cent.

To find the angular position of the driven wheel, prolong the lines *BX*, *BY*, etc., in Fig. 17, until they intersect arc *ET*. Measure angles *ABX*, *ABY*, etc., and lay off below the velocity curve, as shown in Fig. 18. The angular position of the driven wheel corresponding to any position of the driver can then be read off directly.

The only practical point in the design of the Geneva transmission that will be referred to here is to call attention to the desirability of enlarging the diameter of the driven wheel, as shown at points *R*, Fig. 17. This permits of operating a locking mechanism while the driven wheel is constrained by the driver, and makes the Geneva stop of very general application in automatic and semi-automatic machinery.

CHAPTER V

TRIPPING OR STOP MECHANISMS

Mechanisms of this general class may be used to stop a machine automatically either at the conclusion of a series of operations, possibly for stock renewal, or after a predetermined number of revolutions. Another function of a stop mechanism is to prevent the transmission of power to the machine whenever an abnormal operating condition would result in damage to the machine. These and other applications will be described.

Mechanism for Stopping Machine Automatically when Reel is Filled with Wire.—The mechanism shown in Fig. 1 is part of a machine for insulating electric wire. The purpose of this mechanism is to automatically disengage the machine clutch and thus stop the machine when the reel upon which the finished wire is being wound has been filled. This leaves the operator free to attend to other duties while the wire is being wound on the reel.

As the wire reel *A* gradually becomes filled, the roll *B*, resting on the layers of wire, is forced outward, causing the arm *C*, through a sliding clutch mechanism, to disengage the power actuating the reel. The roll *B* is held snugly against the wire by means of a weight (not shown) connected to the arm *C* by the cable *Y*.

The driving shaft *D* for the reel is supported in bearing *E* bolted to the machine base *G*. On this shaft is shrunk the clutch member *H* which engages the clutch teeth on sleeve *J*, to which gear *K* is keyed. Gear *K* meshes with gear *L* keyed to the shaft *M* on which the reel is secured. Sleeve *N* is a free fit in bearing *F*, and at its right-hand end has a turned collar. One end of coil spring *O* is placed

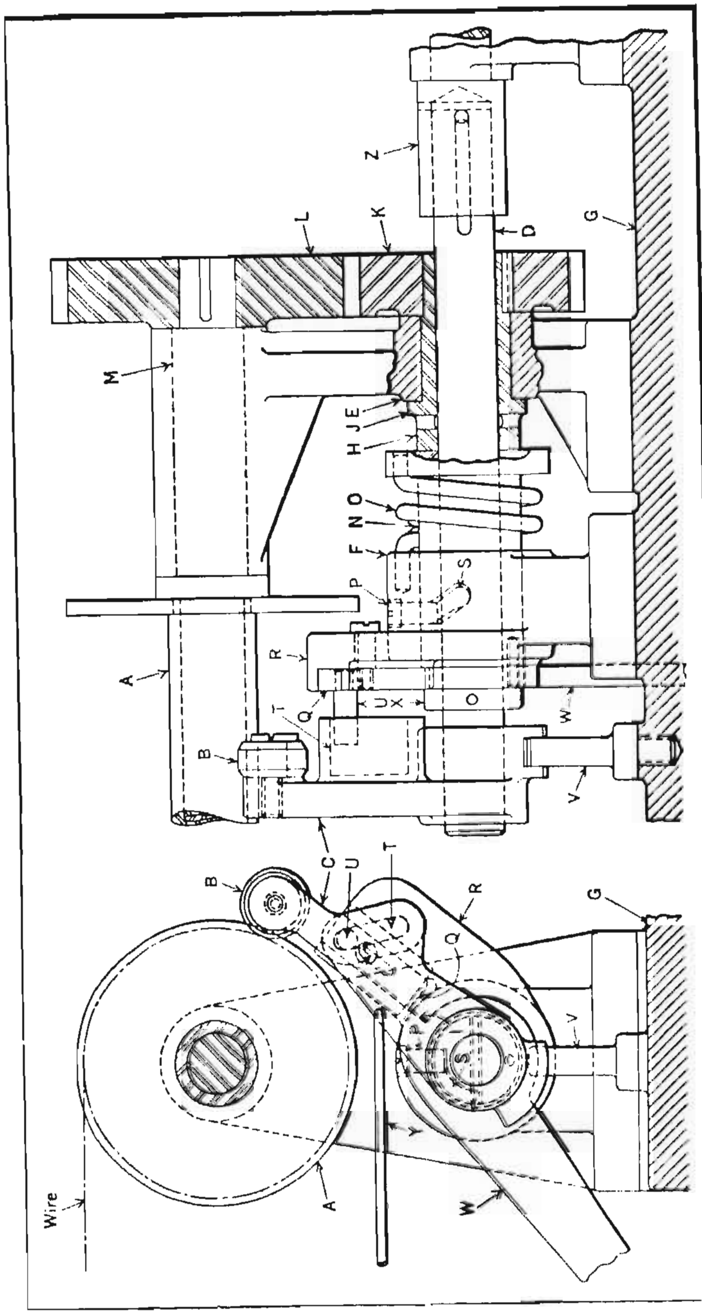


Fig. 1. Reel A of this Mechanism is Stopped Automatically when Filled with Wire which Forces Roller B Outward, so that Latch Q Releases Sleeve N, Permitting Torsion Spring O to Revolve Sleeve N. This Causes the Pin P in the Helical Slot S to Slide the Clutch Member H out of Mesh with the Clutch Sleeve J, thus Disconnecting the Drive from Shaft D to Reel A through the Gears K and L

in a hole drilled in the end of bearing *F*. This spring, when released as explained later, serves to rotate sleeve *N* in bearing *F*, so that the screw pin *P*, engaging a cam slot *S* in the sleeve, causes the clutch to move axially, disengaging the clutch members *H* and *J*.

The latch for releasing the spring is shown at *Q*. This latch slides radially in the guide *R* cast integral with bearing *F*. The movement of this latch is controlled by the pin *U* which is riveted to the latch and engages the deep cam slot *T* in a projection on arm *C*. Arm *C* turns freely on drive shaft *D* and is prevented from moving axially by the stationary pin *V* which engages a segmental groove in its hub.

When in its lowest position, the latch engages the projection *I* on the hand-lever *W*, which is a free fit on shaft *D* but is kept from moving axially relative to sleeve *N* and shaft *D* by collar *X* pinned to this shaft. Incidentally, lever *W* is pinned to the end of sleeve *N* and must therefore rotate with it. Sleeve *Z* provides for an axial movement of shaft *D*; the shaft slides in the sleeve, but is prevented from rotating by a key engaging a spline in the shaft. This sleeve is part of a shaft which is connected directly to the driving motor shaft.

The wire passing on to a nearly full reel is indicated in dot-and-dash lines in the end view. At this time, the last layer of wire on the reel has forced the roll *B* and arm *C* very nearly to their farthest right-hand position. During this movement of the arm, the cam pin *U* has been forced outward radially until latch *Q* is just about to leave projection *I* on the hand-lever *W*. As soon as another layer of wire is wound on the reel, the lever *C* will swing to the right a corresponding amount and cam slot *T* will raise pin *U*, so that latch *Q* will be entirely disengaged from projection *I*.

It should be mentioned that a torque has been developed by the coil spring *O* which, up to this point, has held the projection *I* tightly against the latch. As soon as the latch releases the lever, the energy stored in the spring causes the lever to swing in a clockwise direction, rotating sleeve *N* until pin *P* comes into contact with the opposite end of cam slot *S*. The action of the pin in this slot will cause the sleeve, together with lever *W*, shaft *D*, and clutch member *H*, to move axially toward the left and thus disengage clutch member *H* from the clutch teeth in sleeve *J*. By disengaging these clutch members, the power is thereby disconnected from the reel, causing the latter to stop, so that it can be replaced by an empty one.

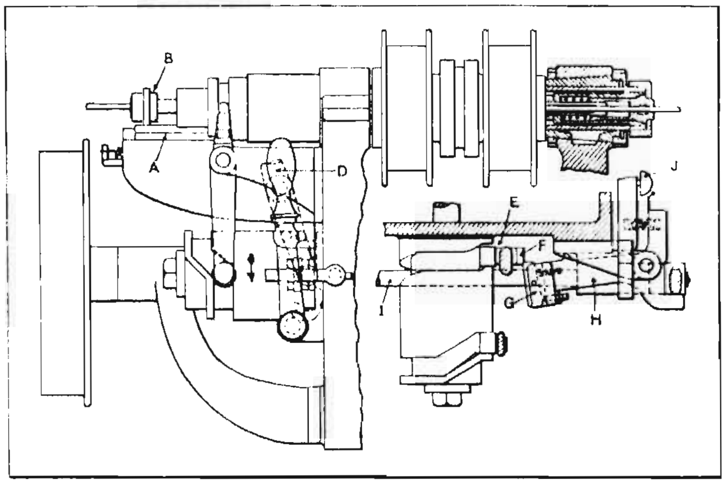


Fig. 2. Mechanism for Automatically Stopping the Machine when New Bar of Stock is Required

When the empty reel is in place, the hand-lever *W* is merely swung downward to start the reel rotating. This causes roll *B* to rest on the core of the reel and the cam slot *T* to allow pin *U* to move downward, so that latch *Q* once more engages the projection *I* and locks the lever in position. This downward movement of the hand-lever, of course, rotates the sleeve *N* so that the reverse action of the pin in the cam slot *S* occurs, moving the sleeve *N* toward the right and engaging the clutch members *H* and *J*, thus rotating the reel.

One outstanding feature of this arrangement is that

there is no excessive end thrust on the bearing, a condition which is typical of mechanisms of this type.

Stopping Machine for Stock Renewal.—A device for stopping the machine when the bar of stock has all been used is shown in Fig. 2; this result is accomplished by a mechanism controlled by the disengagement of the feeding device with the stock. The mechanism is so designed as to stop the machine with the jaws of the chuck open, so that a new rod of stock may be inserted; it is also devised so that the machine is not stopped nor the chuck opened until the length of stock projected by the forward movement of the feeding mechanism is acted on and severed from the remaining stock. This is accomplished by so constructing the stop mechanism that it is thrown into operative position when the feeding devices are disengaged from the stock, but does not operate to stop the machine until the feeding devices are again advanced.

In this construction, the slide *A*, connecting with the feeding tube by the grooved collar *B*, is drawn back by a spring when the stock passes beyond the feeding fingers, there being no friction to hold it; this operates lever *D* (shown dotted), the movement being made possible by the widened space in the cam groove at *E*. This movement allows the projection *F* to pass the latch *G* so that, on the next revolution of the cam, the lever *H*, carrying the latch, is rocked together with shaft *I*, which throws the driving mechanism out of operation and also sounds the gong *J* to notify the operator that a new piece of stock is needed.

Cork Cap-Disk Feeding Mechanism which Operates Only When Caps are in the Receiving Position.— The device shown in Fig. 3 is used in conjunction with a cap-feeding mechanism for inserting cork disks in the caps. As the caps are fed down the line the device places a cork disk in each cap, after which the caps continue on their way to other stations. The outstanding feature of the device is that it will not feed a cork disk *R* from the mag-

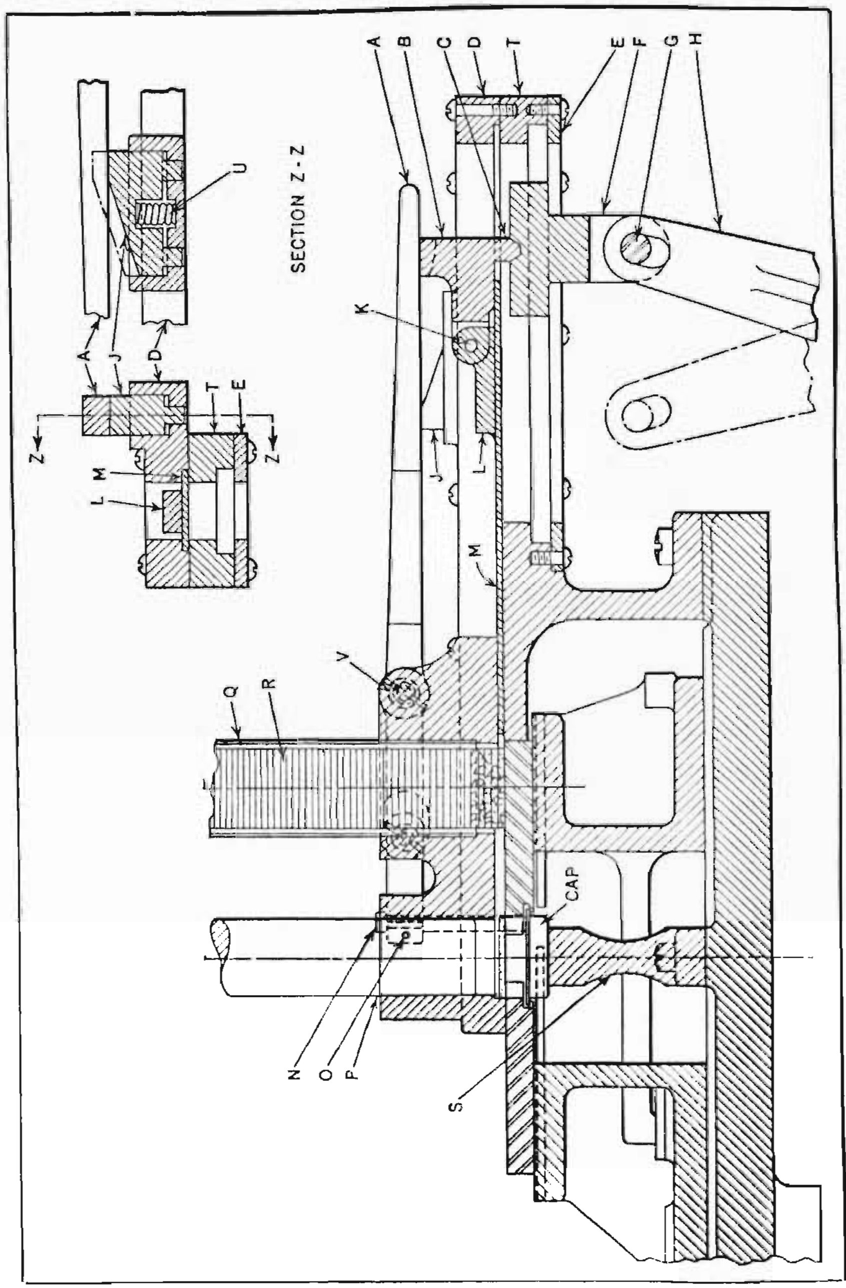


Fig. 3 Mechanism for Inserting Cork Disks in Metal Caps: an Automatic Stop Operates if a Cap is not in the Receiving Position

azine *Q* unless there is a cap ready to receive it. This is quite important, because it frequently happens that the flow of caps is interrupted. If the device continued to feed the cork disks, they would be wasted and in all probability, the mechanism would jam.

This cork disk feeding mechanism is synchronized with the cap-feeding mechanism, so that there is no chance for a misstep in production. The cork disks *R* are stacked in the vertical magazine or tube *Q*, which is kept in continuous agitation so that the disks assume a horizontal position, the upper ones falling down when those at the bottom are removed. A feeding finger *M*, slightly larger in width than the diameter of the cork disk, passes back and forth under the stack of disks, pushing them, one by one, under the plunger *P*. The plunger then forces the cork disk into the cavity provided for it in the cap. An anvil *S* is provided under the caps to take the pressure of the plunger.

The finger *M* slides in a groove provided for it between *D* and *T*. A hinged latch, consisting of parts *K* and *L* and the latch part *B*, is fastened to the feeding finger permanently. Below the feeding finger is the driving slide *F*, which functions in a groove between *T* and *E*. The driving slide reciprocates continuously under the action of the rocker arm *H* through the pin connection *G*. A groove cut in the top of the driving slide corresponds in shape to the projection *C* on latch *B*. When projection *C* rests in the groove on the top of the driving slide, the feeding finger *M* is reciprocated under the cork disk stack.

The mechanism is synchronized, so that a cork disk is in place ready to be forced down by the plunger just at the time when the cap begins to move from its place directly ahead of the cork disk feeding mechanism. A feeler bar *N* is placed at this point ahead of the cork disk feeding mechanism and, by its vertical motion, controls the feeding of the cork disks. The feeler bar is guided in a slot cut in the cap guide bar, and the opposite end rests in a slot in the

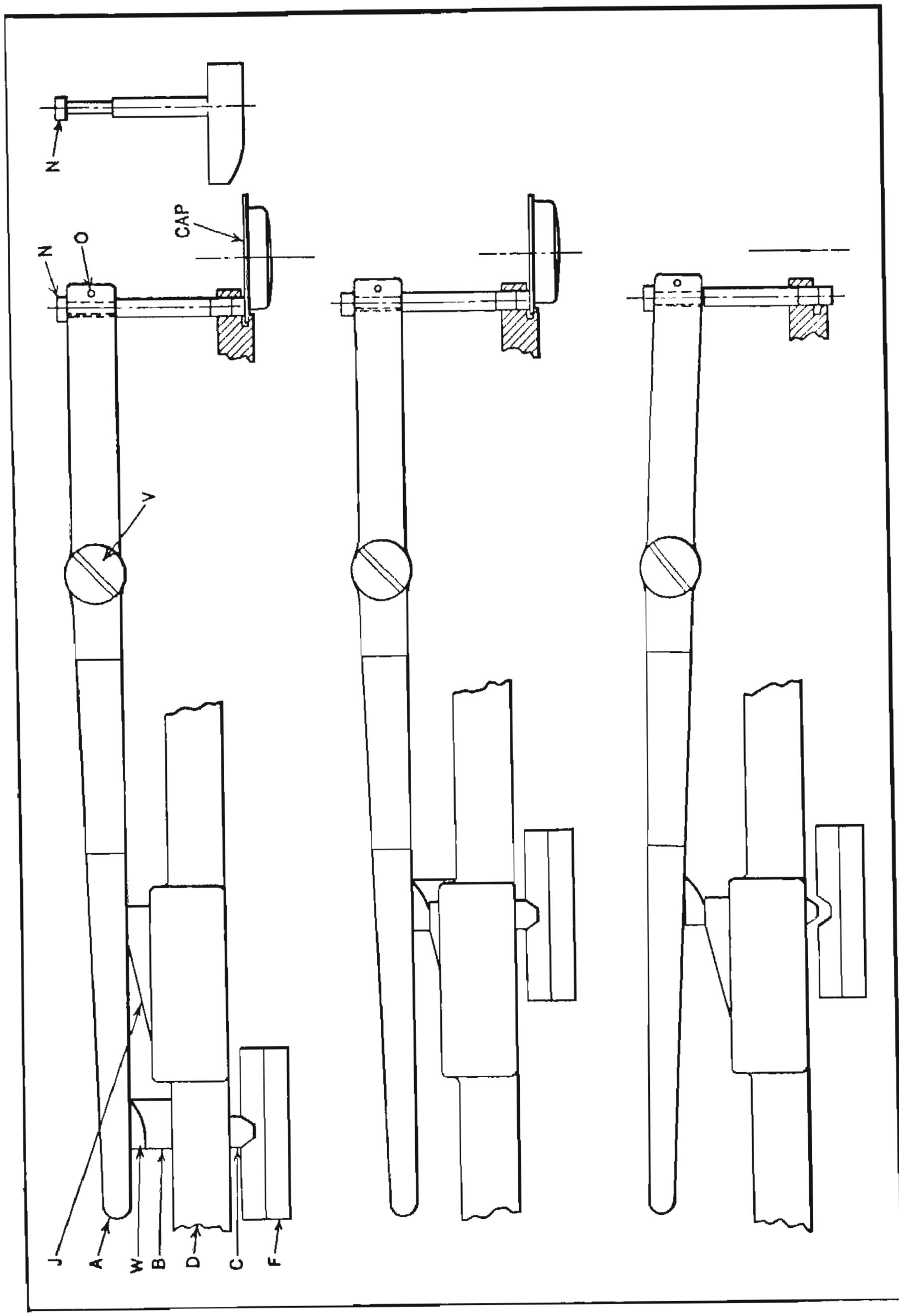


Fig. 4. Diagrams Illustrating Operation of Device that Stops Feed of Disks when Feed of Caps to Assembling Position is Interrupted

end of the lever *A*, where it is retained in place by a pin *O*. Lever *A*, in turn, is pivoted on a screw *V*, and is balanced about its pivot point so that the weight will be slightly greater at the end where the feeler bar is located. This insures the proper contact between the caps and the feeler bar.

The opposite end of lever *A* rests on a horizontal projection on latch *B* and on the tapered button *J*. The tapered button has a smooth vertical motion under the action of the light spring *U* in the well provided for it in *D*.

The operation of this device will be clearer by referring to the three views in Fig. 4, which show lever *A* in three different positions. The upper view shows the lever in the position assumed when a cap is under the feeler bar. The projection *C* on the latch is located in the slot on the driving slide *F*. At the same time, the tapered button *J* is in contact with the lever. The central view shows the position of the parts when the feeding finger has pushed a cork disk into place under plunger *P*. Lever *A* is in the same position as in the upper view, which indicates that another cap is under the feeler bar. The projection *W* on latch *B* has depressed the tapered button, so that the projection *C* engages the driving slide *F*. When there is no cap under the feeler bar, the parts assume the positions shown in the lower view. In this case, lever *A* has been raised from the tapered button so that as the projection *W* rides up the tapered surface it lifts projection *C* out of the slot in the driving slide and thereby stops the movement of the feeding finger *M*, Fig. 3.

Device that Prevents Engagement of Clutch Until Slide is in Operating Position.—The rotating spinning tool of a machine for spinning an inaccessible joint in kitchenware had to be of the expanding and contracting type to allow access to the work. The machine clutch was required to be disengaged and the rotary movement of the tool positively stopped while the tool entered the work, as

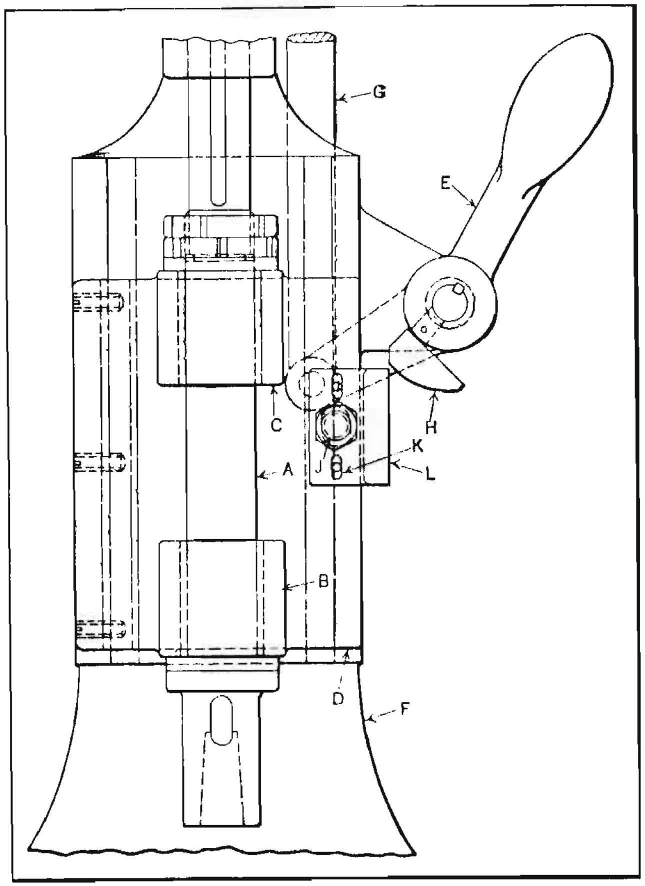


Fig. 5. Safety Device that Prevents Expanding Spinning Tool from Being Operated while it is Entering or Leaving Work

otherwise, the centrifugal force would cause the tool to expand and damage the work. To prevent the operator from accidentally leaving the clutch engaged at this time, the simple locking arrangement shown in Fig. 5 was devised.

The tool-spindle *A* rotates in bearings *B* and *C*, which are cast integral with a vertical tool-slide *D*. This slide is fed downward by a foot-pedal, which actuates a rack and pinion. The foot-pedal and rack and pinion are not shown in

the illustration. The machine clutch is operated by lever *E*, pivoted to the bracket cast on the machine frame *F*. This lever is connected to the clutch mechanism by a link *G*. The interlocking arrangement consists merely of the dog *H* on the hub of lever *E* and the stop *L*, which is secured to the vertical slide by a nut on the stud *J*. This stop is of angular shape and has a slight vertical adjustment to accommodate similar work of different sizes. The adjustment is provided by the elongated holes for the aligning pins *K* and stud *J*.

The slide is shown in its working or lowest position, and the stop is down far enough to allow dog *H* to pass when the lever is swung in a clockwise direction to engage the machine clutch. In swinging the lever for this purpose, however, the dog is moved toward the left, thus blocking the return of the stop, with the slide, while the clutch is engaged. After the work has been spun, the operator must shift lever *E* back again to disengage the clutch before returning the slide to its upper position. Thus, when the slide is at any other point than the lowest point indicated, the dog will come in contact with the stop and prevent the lever from being swung clockwise to engage the clutch.

Roller Clutch with Tripping Device.—In designing special machinery, it is often necessary to provide a tripping clutch similar to that used on a power press. Such clutches may be required to have the added safety feature of not repeating should the operator fail to take his foot off the starting treadle. Special wire or pipe bending machines, special cutting-off machines, and single-action machines for such operations as gluing, notching, stamping, and scoring, are typical machines on which clutches of this kind are used. To meet the requirements of such machines, the clutch shown in Fig. 6 was designed. Although not new in principle, the design has been developed to a point where the device is light in weight, compact, and effective in action. Different applications may, of course, necessitate

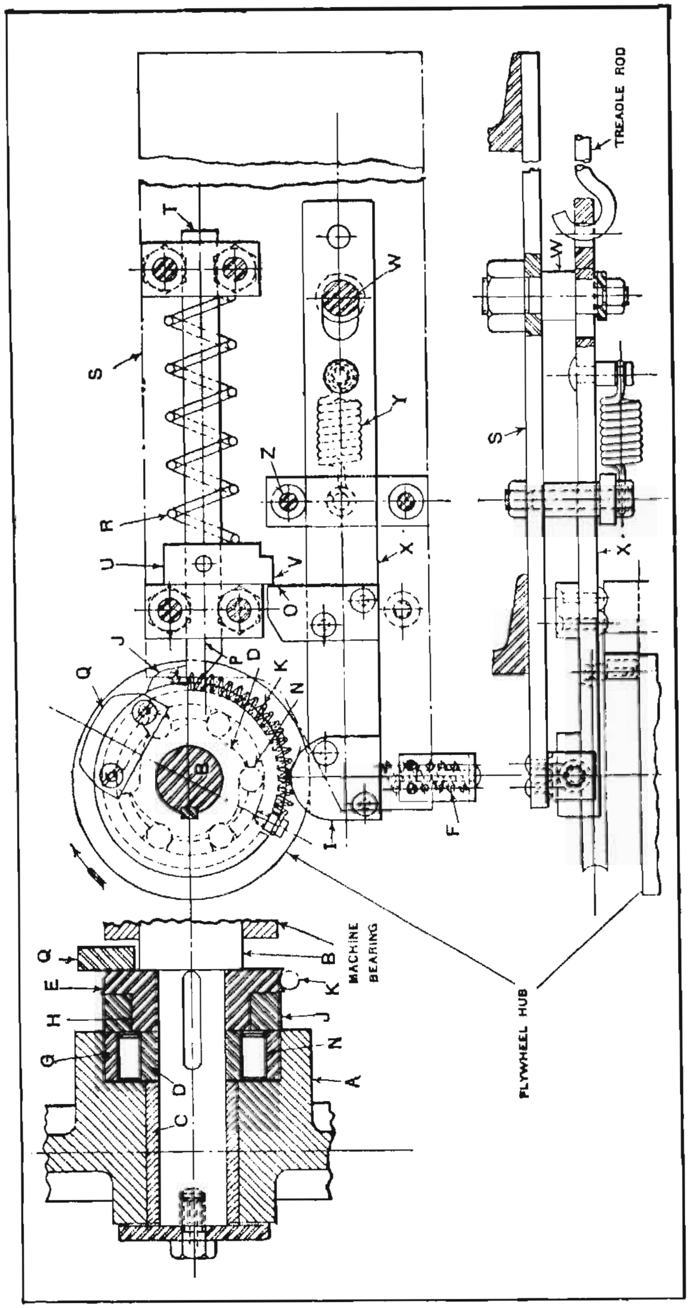


Fig. 6. Roller Clutch with Tripping Device Designed to Prevent Repeating

changes in the mounting, treadle action, and driving means.

When the machine is in operation, the flywheel *A* revolves continuously. Shaft *B* remains stationary until the operator depresses the foot-treadle. When the treadle is pressed down, shaft *B* makes one complete revolution and stops, regardless of whether the operator removes his foot from the treadle or keeps the treadle depressed. In order to cause the shaft *B* to make another complete revolution, the treadle must be allowed to return to its normal position and be depressed again.

Briefly, the action of the tripping device is as follows: When the foot-treadle is depressed, the treadle-rod pulls plate *X* to the right against the tension of spring *Y*. This movement brings latch *O* into contact with collar *U* at *V*, causing rod *T* to move to the right against the tension of spring *R*. The downward movement of the foot-treadle is continued until the latch *P* at the end of rod *T* is disengaged from the nose *J* of the friction roller cage, also shown in Fig. 7. The friction roller cage, being thus released, is revolved clockwise on the hub of collar *E* through the action of spring *K* fastened to pins *L* and *M*, Fig. 7. Referring to Fig. 6, it will be noted that collar *E* is keyed to shaft *B*.

Now as the roller cage revolves, it carries the rollers *N* with it, forcing the rolls to climb up the cam surfaces of the cam member *D*, which is keyed to shaft *B*. The rollers finally reach a point where they act as wedges between cam *D* and the hardened steel ring *G*, which is pressed into the hub of the flywheel *A*. Flywheel *A* then drives shaft *B* forward in the direction indicated by the arrow, the roller cage and collar *E* revolving with the shaft.

The shaft *B* is revolved but a fractional part of a revolution before the cam *Q*, secured to collar *E*, comes in contact with the cam *I* at the end of plate *X*, causing plate *X* to pivot about pin *W* against the tension of spring *F*. This action serves to disengage collar *U* from contact with latch *O* at point *V*, allowing spring *R* to force rod *T* back into the position shown, with the latch P ready to engage nose J of the roller cage when it has made a complete revolution.

As the rolls are prevented from moving forward by the roller cage when nose J is stopped by latch P, their wedging action between the ring G and the cam D is released and

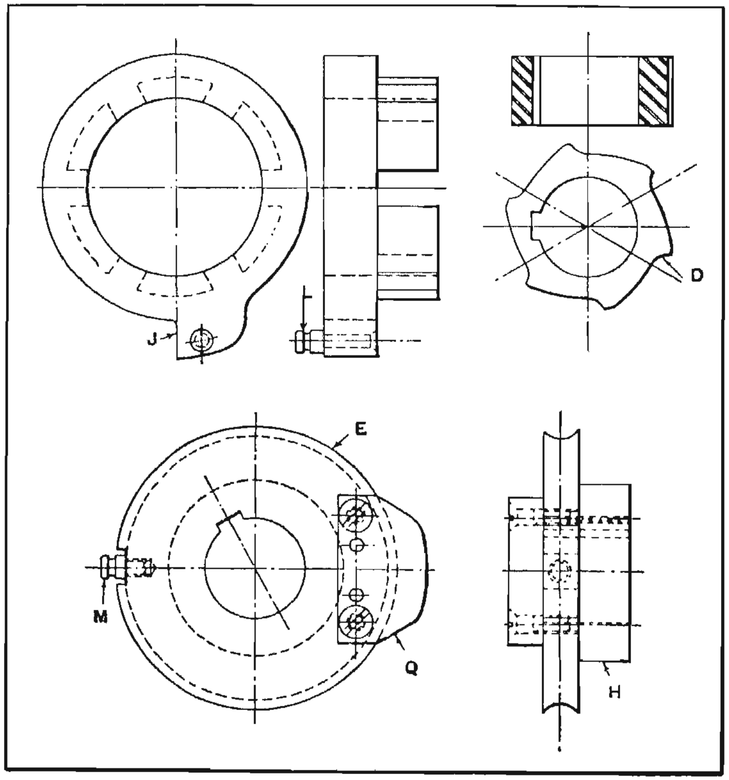


Fig. 7. Roller Cage J, Collar E, and Cam D of Clutch Shown in Fig. 6

shaft B stops, while the flywheel continues to revolve. All parts of the device are now in the positions shown in Fig. 6, ready for the tripping operation to be repeated. While the treadle-rod and the latches are shown in a horizontal position in the illustration, they are usually located in a vertical position, plate S being mounted on the machine frame, portions of which are shown in cross-section in the

lower view. However, the construction is such that the latch can be located at any desired angle relative to the roller cage. To insure efficient operation of the safety device, care should be taken to see that the rod T slides freely in its bearings, and that spring R has a very snappy action. All wearing parts should, of course, be hardened.

Tripping Device for Bead-Chain Cutting-Off Machine.— Bead chain made of brass is used in large quantities for electric-light pull-sockets. This chain is wound on spools

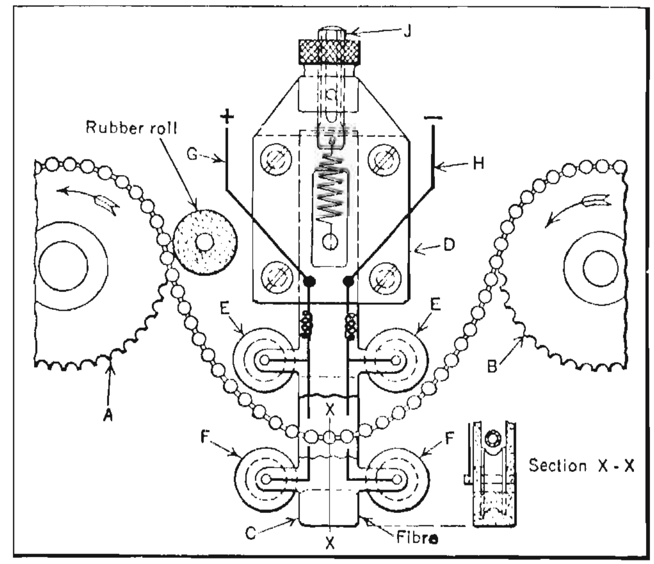


Fig. 8. Electrical Tripping Device that Stops Machine when Chain Breaks or when the Slack Varies

in bead-chain forming machines and then delivered to other machines where it is cut off to the required length for assembly in the sockets. In the cutting-off machines, the chain is passed over two sprockets A and B, Fig. 8. Creeping of the chain on the sprockets is one of the major troubles experienced with these machines; and if the machine continues running after creeping occurs, mutilation of the chain in another part of the machine results.

It was found that this difficulty could be overcome by maintaining a certain amount of slack between the sprockets; and to obtain this condition, the tripping arrangement shown was incorporated in the machine. The tripping mechanism is so arranged that if the slack becomes appreciably greater or less than that indicated, the chain closes the electric circuit of a solenoid. This causes the core of the solenoid to release a clutch which stops the machine. The operator then gives the chain the required amount of slack. One of the advantages of this type of tripping device is that the chain is not required to lift or support any weighted latch member in order to close the circuit; the chain itself closes the circuit. In addition to this, if the chain breaks, the circuit is also closed, causing the solenoid to stop the machine.

The tripping arrangement consists chiefly of the fiber slide *C*, which is guided in the stationary block *D*. On the slide are mounted two sets of rolls *E* and *F*. At the end of the small pins on which the rolls turn, copper wires are soldered. These wires are connected to the two main wires *G* and *H* leading to the solenoid (not shown). It will be noted that slide *C* has a floating action, its weight being supported by the spring attached to screw *J*. This prevents excessive pressure of the upper rolls on the chain. Screw *J* can be adjusted so that the chain is normally half way between the upper and lower sets of rolls. The slide is made from fiber in order to insulate it from the machine, and the machine is separated from its foundation by layers of insulation to prevent grounding of the current.

The action of the device is as follows: If the chain creeps forward on sprocket *B*, the slack will increase until the chain rests on rolls *F*. This closes the circuit formed by wires *G* and *H* and operates the solenoid, which, in turn, releases the clutch and stops the machine. If the creeping of the chain is such that the slack is reduced, a similar action of the solenoid occurs, the chain being drawn against rolls *E*, in this case, and thus closing the circuit.

Quick-Tripping Mechanism for Clamping Device.— In designing a special clamping device, it was necessary to incorporate a quick-tripping mechanism which would provide for a rather slow releasing and an almost instantaneous clamping action. The mechanism designed for this purpose consists of a common plate cam *A* (Fig. 9), and an

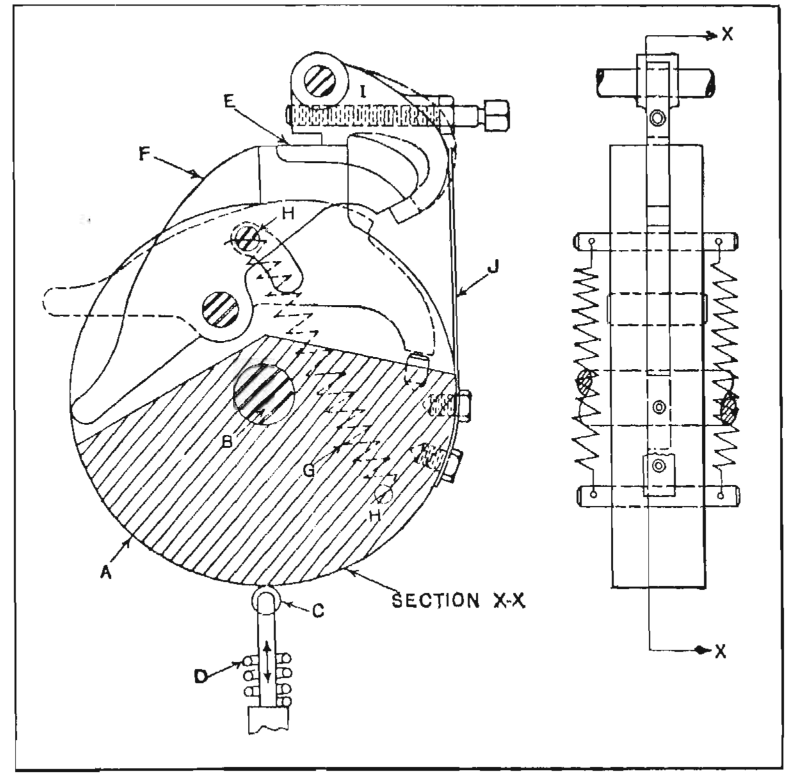


Fig. 9. Quick-tripping Mechanism for Clamping Device

involute cam *F* pivoted in a slot cut through the middle of the plate cam. These two cams actuate the cam-roll C which performs the required tripping and releasing operations.

The shaft *B*, to which the plate cam *A* is keyed, has a reversing motion. It revolves approximately 180 degrees

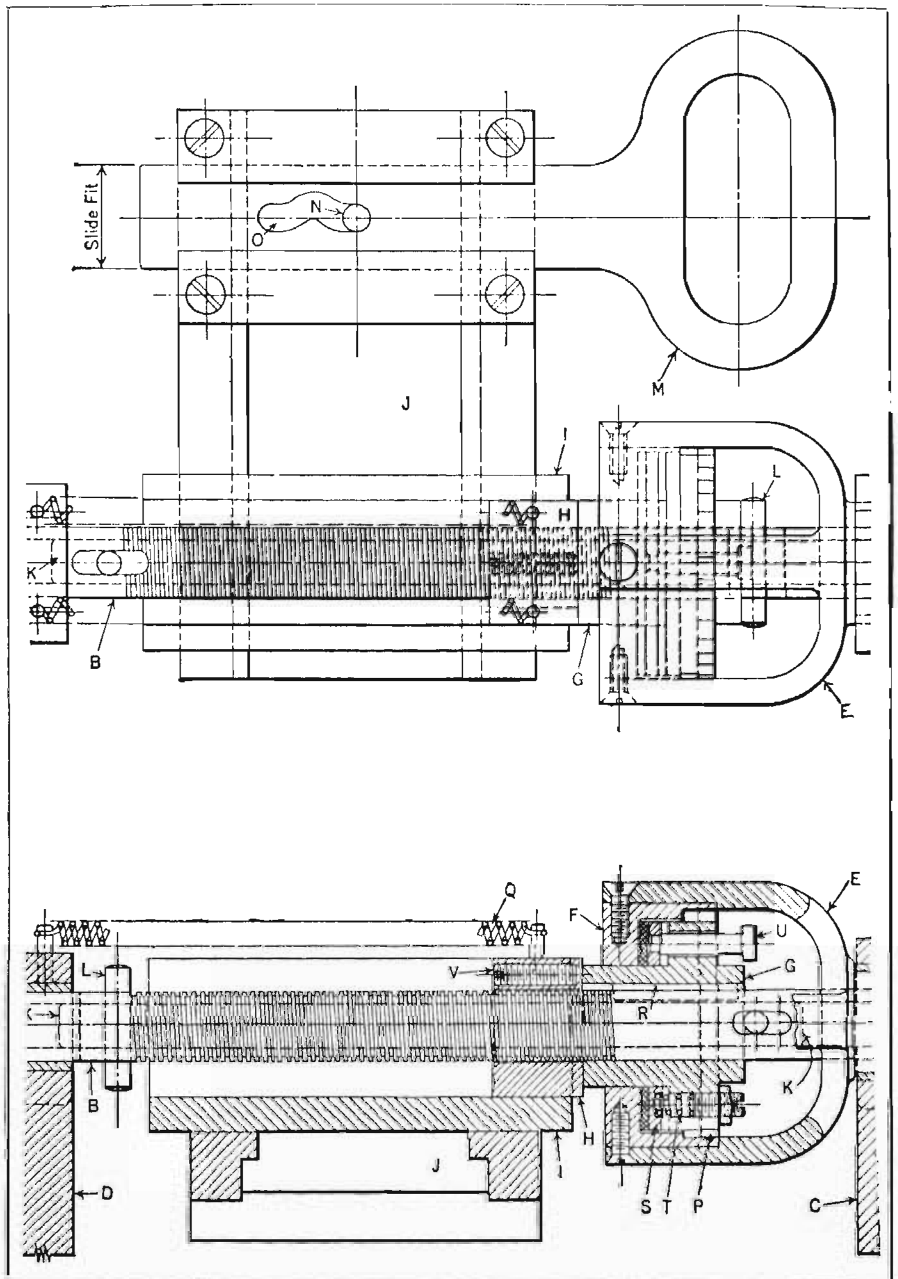


Fig. 10. Mechanism with a Lead-screw and Nut Arranged to Disconnect the Driving Clutch after a Given Number of Revolutions

in a counter-clockwise direction, reverses, and then returns to the starting position, this oscillating movement being continuous. As the plate cam *A* revolves in a counter-clockwise direction, the roll *C* is gradually depressed by the cam *F*, causing the clamping device to be released slowly. When the cam *F* reaches the point where the roll *C* is in contact with the lobe *E* of the cam *A*, the clamp is fully released, and continued rotation of the cam brings the end of the adjusting screw of latch *I* into contact with roll *C*, causing the latch to release cam *F* and allowing the springs *G* to draw the cam into the position shown by the dotted lines.

At this point the direction of rotation of shaft *B* is reversed by a mechanism on the machine, which is not shown in the illustration. As the cam assembly turns in a clockwise direction, the roll *C* reaches the end of the dwell surface of lobe *E* and is returned into contact with the concentric portion of cam *A* with a very sudden action through the tension exerted by spring *D*. This sudden movement of roll *C* engages the clamping device almost instantly. As the cam continues to rotate, the roll *C* comes into contact with the protruding end of cam *F*. As the tension of spring *D* is sufficient to overcome the tension of springs *G*, the cam *F* is returned to the position shown by the full lines, where it is held by latch *I*, which is under the tension of the flat spring *J*. The mechanism is thus set in position for a repetition of the cycle just described, upon the reversal of the shaft *B*.

Mechanism for Stopping a Machine After a Given Number of Revolutions.—The mechanism shown in Fig. 10 is designed for use on either a hand- or a power-driven machine. The object of the device is to control the number of pieces fed into an assembly from a magazine, by automatically stopping the machine at the end of the count. The device is applicable to any kind of machine in which a shaft or the complete machine is required to be stopped after a given number of revolutions. The machine or shaft remains idle until the work is removed and the handle M is moved to the starting position.

A pulley (not shown) drives the main tubular shaft B through clutch members E, F, and G. Shaft B, in turn, drives the machine. The pulley is keyed to the hub of clutch spider E. The circular barrel F, having thirty internal teeth, is positively secured within the four arms of spider E and revolves with the pulley. The connecting and disconnecting circular clutch member G between B and F has thirty external teeth designed to mesh with the teeth in F at P. Member G is a sliding fit on shaft B, but is prevented from rotating on it by key R.

The concentric spring-pad ring S, mounted on three sliding pins U, is backed up by three compression springs T. On the face of this pad is riveted a piece of brake lining or fabric which provides a frictional engagement between F and G. This friction clutch is adjusted to allow two revolutions before the speeds of the two revolving members become synchronized and the teeth become positively engaged at P.

Within the shaft B is a sliding rod K having crosswise holes near each end, through which two pins L are driven. These pins project through slots in the sides of the shaft. Shaft B has either a buttress or a square thread with a lead and a horizontal length that is sufficient to provide for the largest number of revolutions required. The half-nut H is a sliding fit in the channel I, mounted on the slide J. The threaded hole in nut H is elongated on one side an amount equal to a little more than double the depth of the thread. The thread is also cut away until only about one-half of the threaded circumference is left for engagement.

The pin N in the cross-slide actuates the transfer channel I, either engaging the half-nut H with the lead-screw or disengaging it, by its action on the cam slot O in the sliding portion of handle M. Two coil tension springs Q are attached to the half-nut, their opposite ends being positively fixed to the left-hand bearing on the machine. The half-nut is shown just making contact with the front of clutch member G. As shaft B continues to revolve, the nut advances until the clutch teeth at P are disengaged. The member G then comes in contact with pin L as both G and K are moved forward, disconnecting the clutch pad S from frictional contact with F. The machine is thus stopped, allowing the clutch barrel to run idle.

When it is desired to start the machine, the handle M is pulled forward, causing the nut to become momentarily disengaged from the lead-screw, so that it returns instantly to its left-hand position, and is in mesh again with the lead-screw thread. On its return movement, nut H strikes the pin L at the left and causes the friction pad S to engage F. This, in turn, causes G to rotate in synchronism with E, so that the teeth at P are engaged by means of rod K and pins L. This starts the machine, and the nut begins to travel on the lead-screw toward the clutch, where it repeats the stopping operation. Cam slot O is designed to lock the nut channel in its proper position while the nut is traveling on the lead-screw.

When the nut is disengaged from the lead-screw by handle M, it is returned by springs Q and caused to strike pin L at the left, thus being held momentarily in contact with the pin. The nut cannot start back until it has caused the clutch to engage and thus commence the count. The springs T under the pad S act as buffers against the sudden return impact of the nut on pin L, and allow the clutch to engage more smoothly. Screws at the back of the compression springs T provide means for adjusting the spring pad to give the proper synchronizing friction and buffer action. Screw V in the half-nut is adjusted to give the exact number of revolutions required. Any desired number of revolutions within the range of the lead-screw can be obtained by using a split washer of the right thickness on the contacting face of nut H or by screwing pins of the required length into the face of the nut.

For a large number of revolutions which would require a lead-screw of excessive length, if arranged as illustrated, the nut can be operated on an independent screw in a channel at one side of the clutch, using speed reducing gears between it and shaft B. In this case, the nut is made wide enough to surround shaft B and long enough to lead properly in its guiding channel. However, with a lead-screw 12 inches long, having 18 threads per inch cut on shaft B, over 200 shaft revolutions can be obtained before the clutch is disengaged. This is sufficient for most counting and machine stopping operations.

Stopping Spring Fatigue Testing Machine at Time of Breakage.— An old punch press is used as a fatigue testing machine for shock absorber springs. The equipment operates twenty-four hours a day and a small counter indicates the number of times the spring is compressed. In the past, when a spring broke during the night, the machine continued to operate and a wrong number of compressions was recorded.

The problem was solved by installing a photo-electric relay in such a position that the light beam passes beneath the bottom of the plunger when the plunger is in its lowest position. With this arrangement, the spring intercepts the light beam and prevents it from falling upon the photo-tube under ordinary circumstances. When the spring breaks, however, it collapses and the beam passes over it to the photo-tube, which actuates a relay and stops the operation of the machine.

Power Press Stop Mechanism which Disengages Clutch when Magazine Feed Jams.—When parts become jammed as they are fed from a magazine to the dies of a press, it is likely to prove disastrous, not only to the die members, but to the press members as well. The most practical method of preventing damage in such a case is to stop the

press instantaneously at the top of its stroke by some automatic means. This is done by the mechanism shown in Figs. 11 and 12, which makes it possible for one operator to tend three presses running at about 75 revolutions per minute.

This arrangement, as applied to a battery of power presses equipped with automatic feed mechanisms, has proved highly satisfactory. In addition to eliminating damage to press and die members, this mechanism also provides a valuable safety feature in that the movement of two levers is required to start the press; hence, both of the operator's hands must be on these levers at this time and out of the danger zone.

Only the magazine feed-slide and tripping mechanism are shown. They are mounted on the baseplate A, which is secured to the press at a point adjacent to the magazine. The feed-slide, which is indicated at B, reciprocates in guides on the block C, this block being fastened to the baseplate. The reciprocating movement of the feed-slide is obtained through link D, rocker arm E, lever F, and cross-head G. Rocker arm E pivots about the stationary sleeve J, and lever F is pivoted to arm E at K. The lower end of lever F enters a slot in the side of sleeve J and is held against the side of plunger L by the spring M (Fig. 12). With this arrangement, arm E and lever F oscillate normally as an integral unit. Thus, link D, which is attached at its upper end to the press ram, oscillates arm E and lever F, imparting an oscillating movement to the cross-head G, which operates in the vertical guides H on the feed-slide.

The oscillating movement of the cross-head causes the feed-slide to reciprocate and push the work from the magazine into the die. However, should jamming of the feed-slide occur, the lower end of lever F would immediately swing away from the side of plunger L and cause the tripping levers N, Z, and P to operate, as will be explained

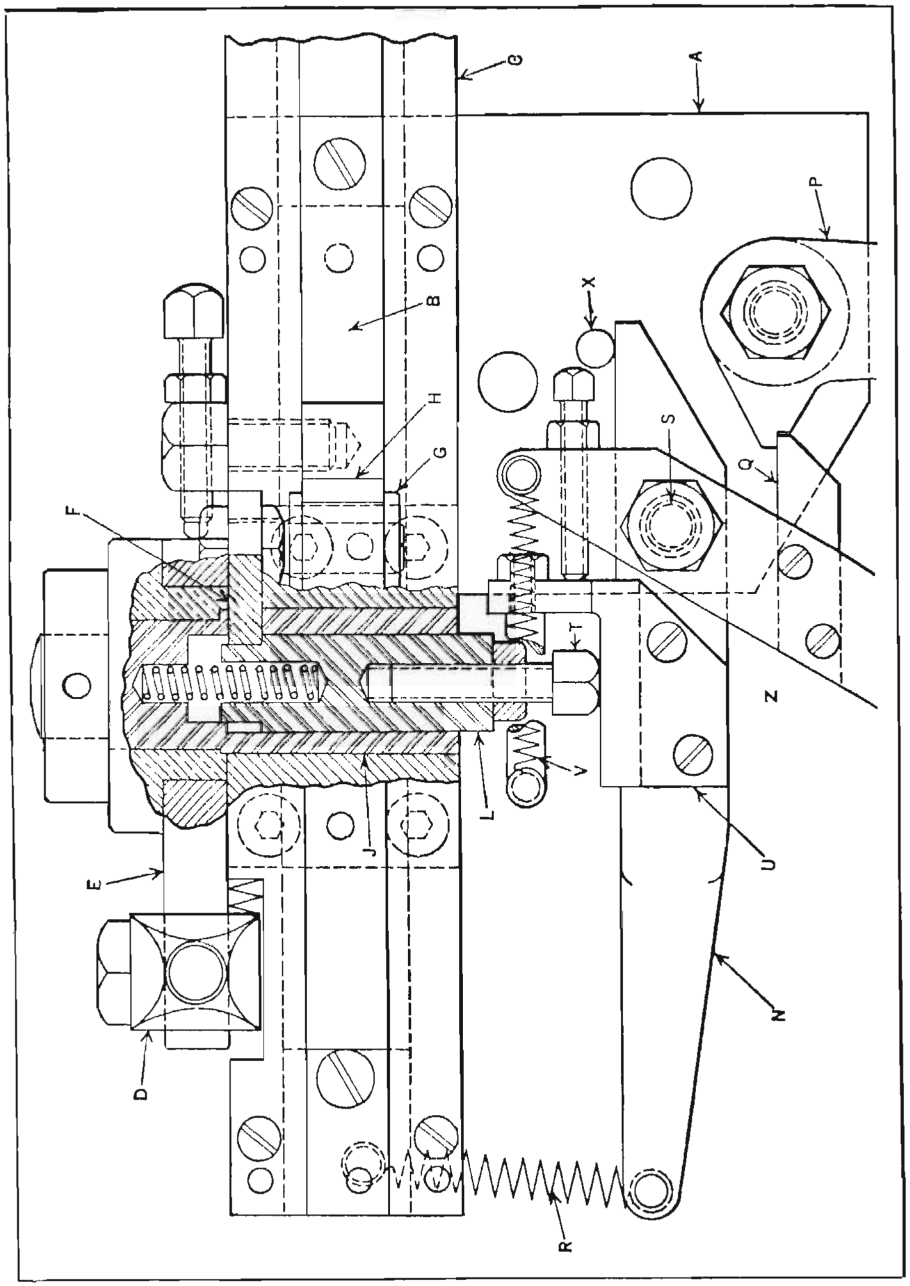


Fig. 11. Plan View of Power Press Stop Mechanism (shown in Fig. 12) which Disengages Clutch when Magazine Feed Jams

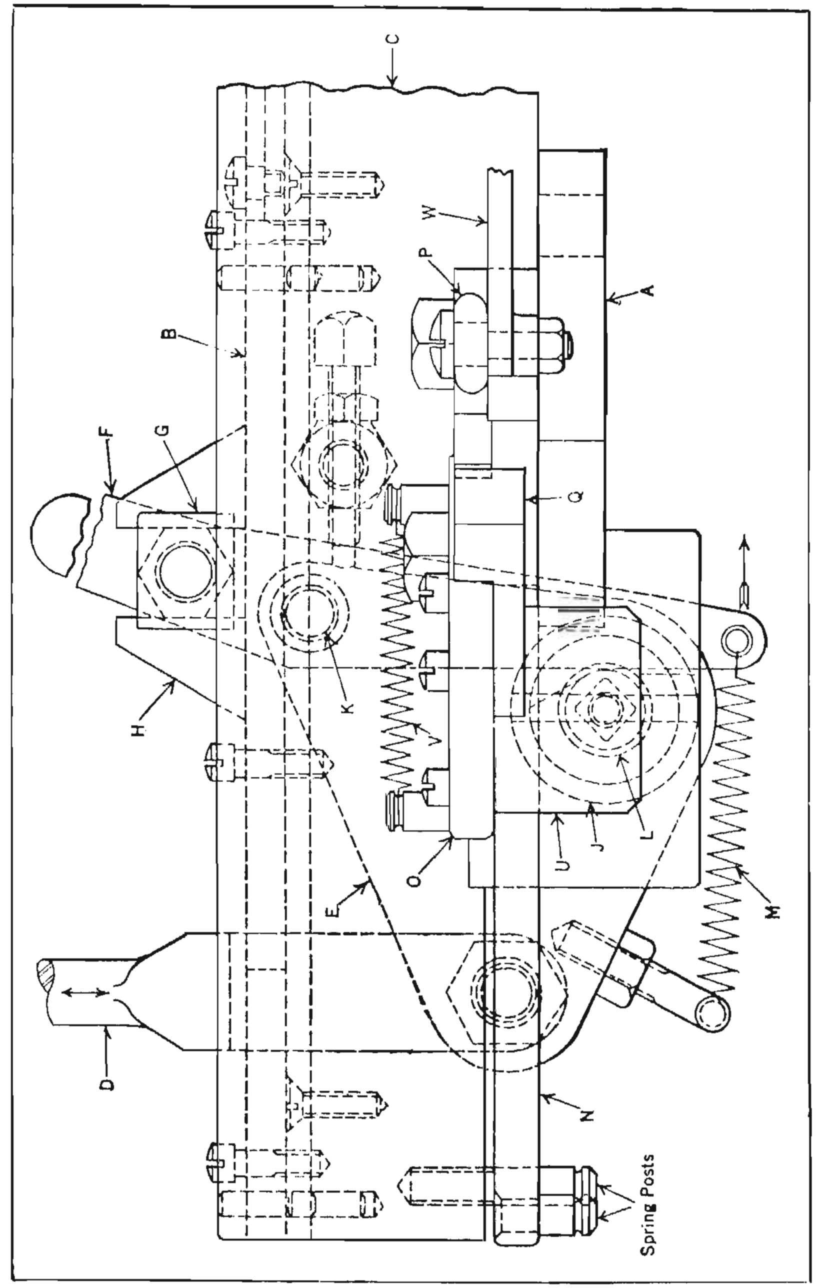


Fig. 12. Tripping Mechanism which Automatically Disengages the Press Clutch when the Magazine Feed Jams

later. Levers N and Z are pivoted at S. Bracket U, secured to lever N, is held against the adjusting screw T in plunger L (Fig. 11) by means of the coil spring R, thus holding plunger L in engagement with lever F. Latch Q, on lever Z, is normally held in engagement with lever P by the spring V. Link W (Fig. 12) is connected to the clutch mechanism, and when moved toward the right disengages the clutch and stops the press ram at the top of its stroke.

The Tripping Action when a Jam Occurs.— Assuming that the feed-slide B has jammed, the movement of the slide will be shortened or discontinued altogether, so that the lower end of lever F will leave plunger L, allowing the plunger to be forced further into the sleeve J by the lever N. As levers N and Z swing together (by resetting and when jamming occurs only), the latch Q leaves the notched end of lever P, allowing this lever and link W to move toward the right under the action of a spring (not shown) and stop the press at the top of the stroke. The cause of the jamming can then be removed.

To restart the press, the operator grasps levers Z and P. Lever Z is swung toward the right, carrying lever N back into contact with the stop-pin X, and thus allowing the shoulder at the end of plunger L to once more engage the lower end of lever F. After this, lever Z is swung slightly to the left to clear the notched end of lever P. Lever P is then swung to the left until its notched end engages the latch Q, thus throwing in the press clutch.

As the work is fed to the dies on the upward stroke of the ram, jamming of the feed-slide usually occurs at this time; hence, the press is stopped at the completion of this stroke, thus preventing damage to the press tools. This arrangement also insures the safety of the operator, as both hands are occupied with levers Z and P when starting the press.

Mechanical Device Stops Press if Punch Breaks.— Electromagnetic devices for stopping a punch press by re-

leasing the clutch when the machine or dies fail to function properly are sometimes used (see Ingenious Mechanisms, Vol. I, page 148). Serious damage to the dies or press is often prevented by such devices. Electromagnetically controlled devices have numerous advantages over the all-me-

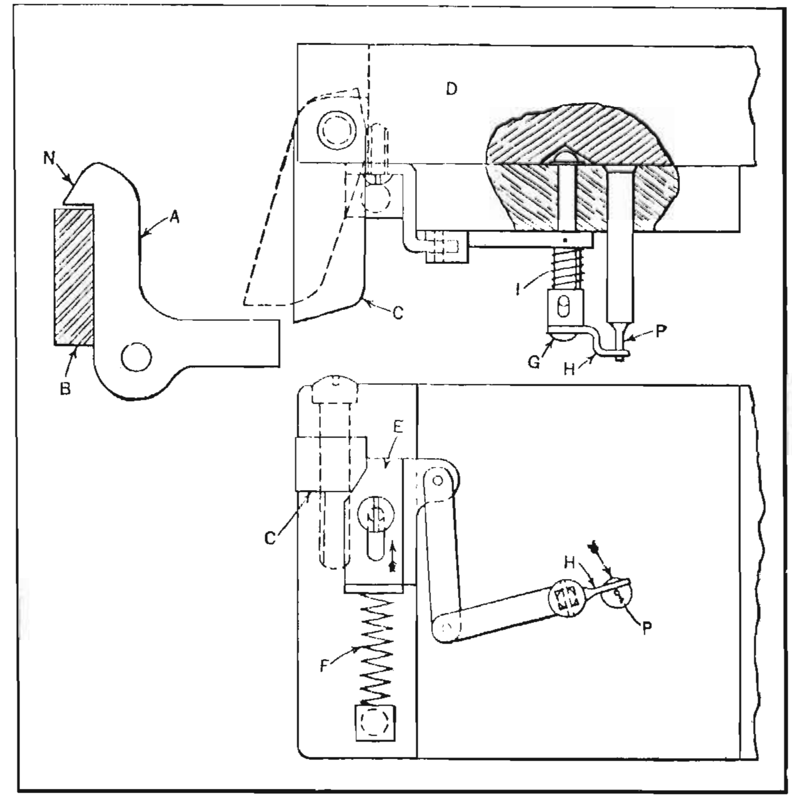


Fig. 13. Mechanical Device for Stopping Press if Piercing Punch is Broken

chanical type, but there are times when such equipment is not desirable, especially when it is possible to install a non-magnetic device that is cheaper and practically as simple as the circuit-closing mechanism required for operating the electromagnet. Such a device is shown in Fig. 13. It is designed to stop the press and prevent the work from being spoiled in the event that the small piercing punch P becomes broken.

Referring to the illustration, latch A is attached to the frame of the press in such a position that the hook N will snap over the hand trip-lever B or an auxiliary member that operates parallel with the trip-lever when the latter is depressed sufficiently to engage the press clutch. If punch P is broken, the dog C, which swings freely in an extension of the punch-holder D, is pushed outward into the position shown by the dotted lines. On the next down movement, dog C trips latch A and releases the clutch.

The mechanism by which dog C is pushed outward is clearly shown in the illustration. The pressure exerted by the spring F is just sufficient to overcome the friction of the moving parts and the weight of dog C. Thus the pressure of the finger H on the small punch is not great enough to deflect the punch. The direction in which the spring pressure acts is indicated by the arrows.

The cam E is used to operate dog C in preference to a direct connection with the levers, because any jamming effect on the dog will not be transmitted through the levers to the small punch. Finger H is backed up by a spring I, and moves up and down the punch, the knob G striking a depression in the stripper plate. The movements are so timed that finger H does not touch the strip stock. It is evident that the basic principles here described can be employed in a great variety of lever arrangements that may be designed to suit different conditions.

Automatic Brake Mechanisms.—To safely hoist, hold, and lower a load, hoisting machinery is usually equipped with so-called safety, automatic, or retaining brakes. These brakes permit a load to be lifted freely by the motor, and lock the brake by the gravity action of the load as soon as the lifting torque of the motor ceases to act in the hoisting direction. The load is retained by the brake in any position, and only when the motor runs in the lowering direction is the acting power of the brake diminished, allowing the load to descend. The speed at which the load drops is regulated and determined by the lowering speed of the motor, while the brake, in the meantime, absorbs by friction the greater part of the potential energy of the dropping load, and generates heat in the brake.

Fig. 14 represents what is known as the *Weston* brake, which is the typical form of a very large class of automatic brakes used on hand and electric cranes to control the load. A pinion A mounted loosely on the shaft has formed on one hub a spiral surface normal to the shaft, and on the opposite end a faced surface to present to the friction disks e. A collar D, fast on the shaft, has a spiral surface which engages that of the pinion hub, and is backed up by a split washer or other device to resist end-pressure along the shaft. A flange B, loose on the shaft, has a faced surface similar to that on pinion A, and carries a ratchet to engage with a pawl C. A series of friction disks e is placed between the faced surfaces on A and B in such a manner that the disks in contact with A and B are keyed by sliding feathers to B and A, respectively, as shown at X and Y.

This gives each disk a motion opposite to that of its neighboring surfaces, and each two surfaces in contact having opposite directions of rotation form one friction surface of the brake. Thus the brake shown has five friction surfaces and four washers or disks. These disks are made of various materials; alternate disks of steel and brass, or steel and fiber are frequently used, and also polished saw steel for all the disks. The shaft revolves in the direction of the arrow on the right to hoist, and with the arrow on the left to lower; ratchet teeth are formed to permit the rotation of the flange B when hoisting, and prevent it when lowering; pawl C is counterweighted to throw it into engagement with the ratchet; flange B is backed against a shoulder on the shaft, so that all the end-thrust is taken by the shaft between this shoulder and the split collar *E*, and the brake is self-contained.

Action of the Weston Type of Automatic Brake.—The action of this brake is as follows: Suppose a load acts on the pinion *A* (Fig. 14) tending to revolve it in the direction of the left-hand arrow, and the shaft begins to turn in the direction of the right-hand arrow. *D* being fast on the shaft will revolve opposite to *A*, which will cause the spirals to slip or bind slightly and thrust *A* toward *B*, thus clamping the disks *e* between *A* and *B*, the end-thrust of *D* and *B* being taken by the shoulders on the shaft. In this manner

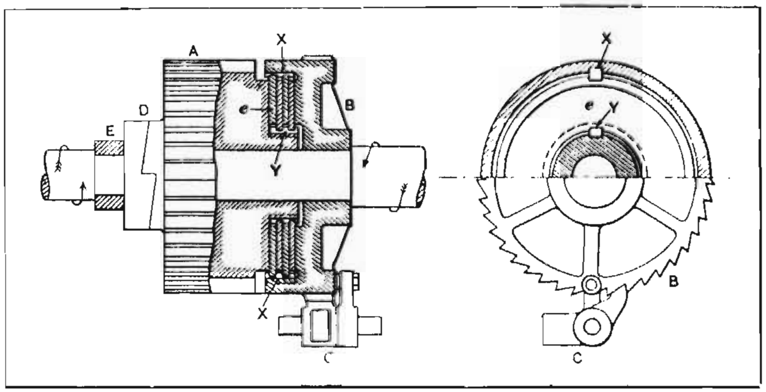


Fig. 14. Weston Type of Automatic Brake for Use with Hoisting Machinery

the whole mechanism consisting of *D*, *A*, *e*, and *B* is locked solidly together, and is made fast upon the shaft; thus the pinion *A* is driven and the load raised.

To lower, the shaft is turned in the direction of the left-hand arrow, carrying *D* with it, and since *A* (at the beginning) is clamped tightly to *B* through the disks *e*, and *B* is prevented from rotating by the pawl *C*, *D* is given motion relative to *A* in the direction of releasing the spirals, and hence the thrust upon *A*. As soon as this thrust is relieved, *A* turns freely in the direction of the left-hand arrow under the influence of the load, and, overhauling the shaft with its collar *D*, brings the spirals again into contact, reestablish-

ing the locked condition and holding the load suspended. A further motion of the shaft results in a repetition of this cycle, and the act of lowering the load consists of an in-

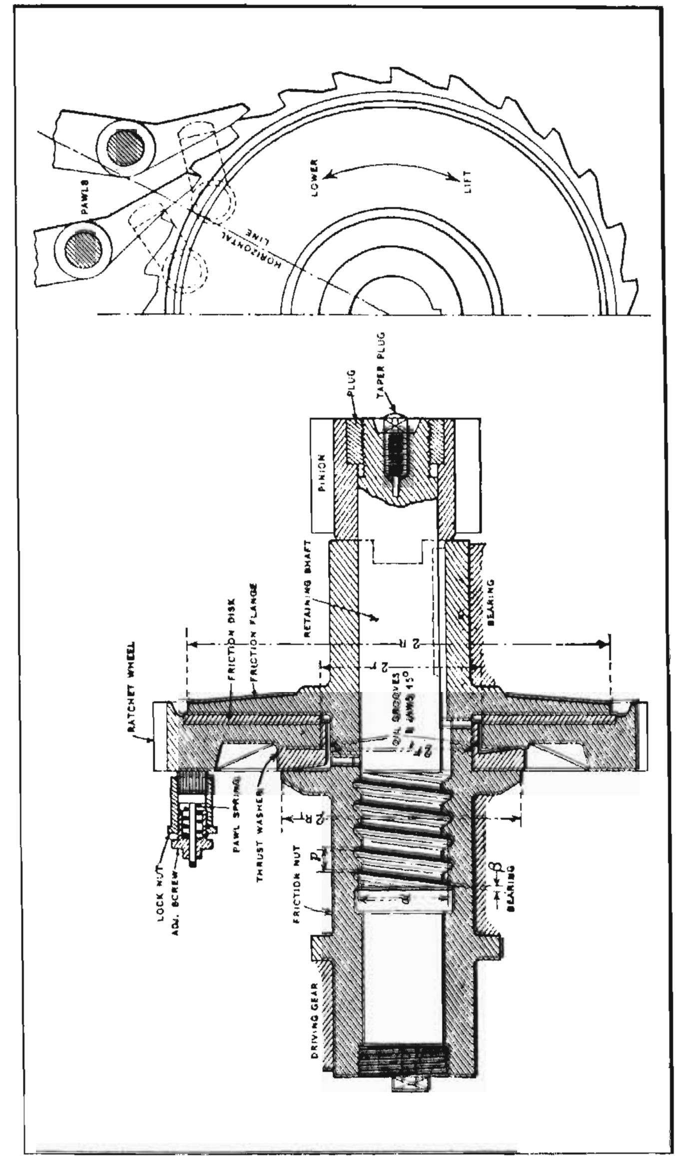


Fig. 15. Another Design of Weston Load Brake for Cranes and Other Hoisting Machinery

finite number of such repetitions in a unit of time, the motion of the load resulting from each cycle being infinitesimal, thus making the motion of the descending load uniform.

Special Type of Weston Brake.— Another type of Weston brake, embodying exactly the same principle as that shown in Fig. 14, is shown in Fig. 15. The ratchet is free to revolve when hoisting, but is held by two silent pawls from turning in the lowering direction. The friction nut is geared to the motor and the retaining shaft with gear pinion leads to the hoisting drum. The retaining shaft and friction nut are threaded either right- or left-hand, according to the hoisting direction. The friction flange is keyed to the retaining shaft and mates with the friction nut by means of three jaws which have about 15 degrees angular play. The friction flange drives the pinion direct through tongued and grooved projections between the pinion and flange. Any tendency of the load to revolve the retaining shaft when the motor is at rest causes the friction flange with friction disk to be pressed against the ratchet wheel and the thrust washer of the nut, due to the action of the threads. The friction of this washer against the ratchet wheel, which, as already explained, does not turn in the lowering direction, is sufficient to hold the load. Upon starting the motor to lower, it turns the friction nut and relieves a certain amount of pressure on the washers, until the pressure is overcome so far as to permit the load to revolve the friction flange in unison with the speed of friction nut, or motor. In hoisting, the jaws of the friction nut and flange engage, thus relieving the brake of all friction.

CHAPTER VI

OVERLOAD RELIEF MECHANISMS AND AUTOMATIC SAFEGUARDS

Certain types of machines or other forms of mechanical apparatus are likely to be subjected to excessive overloads resulting possibly in breakage of one or more parts unless provision is made to prevent, automatically, any dangerous overloading. These overloads are due to some abnormal operating condition and the function of the relief or release mechanism is to automatically disconnect the machine or driven member from the source of power, thus safeguarding it against excessive strains and serious damage. These overload relief mechanisms may be classed as a form of tripping or stop mechanism designed especially to safeguard a machine or its parts against excessive strains and breakage.

Automatic Overload Release for Worm-Gear Drive.— A machine for cutting coal in mining is subjected to such strains, jerks, and shocks that some overload protection is essential. A cast-iron safety washer which has been applied on mining machinery for many years has certain disadvantages which have been overcome by the improved overload release to be described. The safety washer is used in conjunction with a worm-gear drive, as shown by the left-hand sketch, Fig. 1. This washer is placed over the worm and is held by a nut and a short section of pipe. The idea is to make this washer strong enough to hold the worm in place under normal loads. If the load is excessive, however, the thrust of the worm will break the washer, thus releasing the worm from its driving key.

An overload release designed to eliminate certain disadvantages of the safety washer is shown by the sketch at the right of the illustration. The worm is held in the running position by the coil spring *A*, the tension of which may be adjusted by nut *B* on the worm-shaft. A jaw

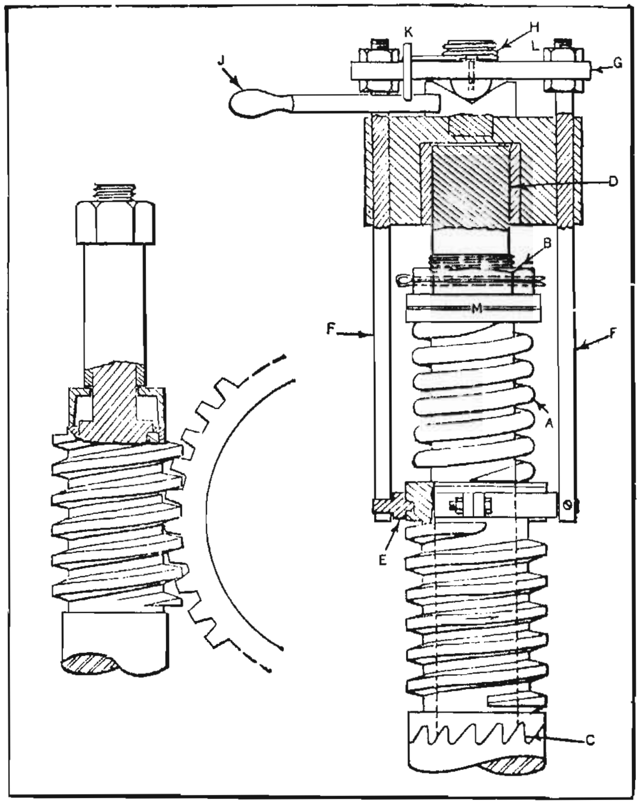


Fig. 1. (Left) Cast-iron Safety Washer Release; (Right) Improved Type of Overload Release

clutch *C* is used instead of a driving key, and the extended worm-shaft has an outer bearing at *D*.

The action of the mechanism is as follows: During normal load, the parts are in the relative positions shown. If there is an overload, the worm thrust compresses spring A so that clutch *C* is disengaged. This axial movement of the worm is transmitted through ring *E* and rods *F* to plate G,

causing the small coil spring at *H* to swing clutch handle *J* from position *K* to *L*, which locks the worm in the out or disengaged position. To reset, lever *J* is simply returned to its former place, which permits the driving and driven parts of clutch *C* to come into engagement.

It is important to have the ball thrust bearing *M* between spring *A* and nut *B*, because when the clutch is in the disengaged position, the worm-shaft and nut must necessarily continue to turn, while the worm is idle. The teeth of clutch *C* should be rounded at the edges to prevent damage at the moment of disengagement under load.

This mechanism is quick and positive, and can be applied to various other drives, especially when a machine is likely to encounter some obstruction due to careless adjustment or operation. It can be utilized to provide overload protection when a machine is running in one direction but not in the other. For example, many machines are geared for a higher speed during the return stroke, and overload protection is desirable for this reversal or backward movement. Various other applications will be apparent to designers. It is obvious that every machine subject to overload should have its safety device, for the same reason that motors need fuses and circuit-breakers.

Worm-Gear Equipped with Friction Drive that Prevents Overload.—A friction release is incorporated in the worm-gear shown in Fig. 2. This gear was designed for use in a wrapping machine in which failure of any part to function would merely result in slippage of the drive gear. The same principle, however, has many applications in special and automatic machinery.

Instead of making the gear from one piece, it is constructed from three pieces, namely, a hub, a ring on which the teeth are cut, and a friction disk. These are assembled, as shown at the left, by six bolts. Originally helical springs were placed between the disk and the bolt heads, as shown at *A*, but in this particular application, it was found that

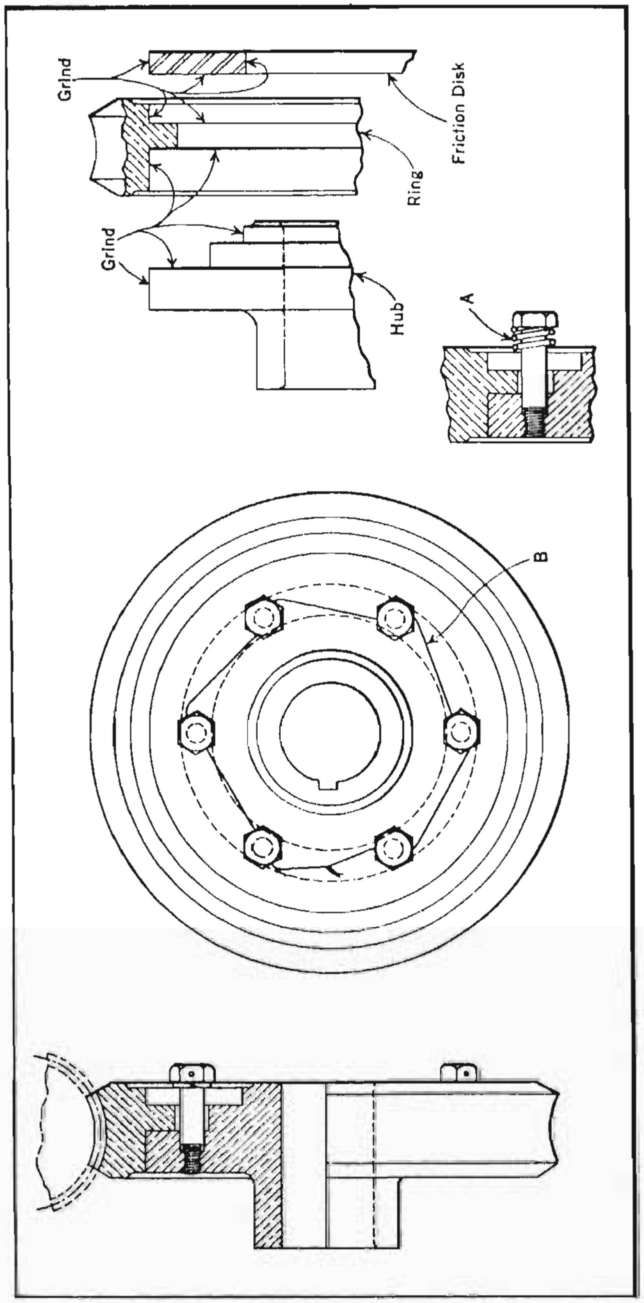


Fig. 2. Worm-gear with Friction Clutch that Prevents Overloading the Driving or Driven Units

spring lock-washers were satisfactory if the studs were not screwed up too tight. The use of helical springs, however, is recommended when slippage must occur at any accurately specified stress. After the proper adjustment has been made, the bolts are restrained from turning by the wire *B* which passes through holes in the heads of the bolts. To insure concentricity, it is best to grind the surfaces as indicated at the right, allowing just enough clearance to offer a free-running fit. These units are then assembled, after which the teeth are cut just the same as in any regular gear.

Before adopting this design, the gear was tested by means of a prony-brake mechanism, comprising a pinion drive, a brake-shoe, and an arm that worked in conjunction with an ordinary weight scale. The precision with which the drive could be made to release was quite surprising. A prony-brake mechanism is recommended for adjusting units for a given load that must be maintained closely. The hub and gear ring of the worm-gear are made of bronze, and the friction disk is made of steel.

Another Overload Release of Friction Type for Gears or for Other Rotating Members.– The release clutch shown in Fig. 3 is of the friction type and has proved very satisfactory in protecting parts of machine drives against overloading. The device can be built directly into a spur or worm gear and requires no additional space; hence it can be easily incorporated in a drive where no provision was originally made for such a device. The clutch is of simple design and very economical to build, since standard gears requiring only a little extra machining can be used. The friction disks used are standard Ford parts, costing less than five cents each.

Gear *A* is provided with six equally spaced holes *B* containing the pins *C*. These pins engage notches in the friction disks *E* and act as drivers. Other disks *F* are provided with lugs *H* which engage corresponding notches drilled in hub *J*. Disks *E* and *F* are free to slide on each other. They are held tightly against the web of gear *A* by hub *J* and collar *L*, the required pressure being transmitted to the hub and collar by the springs *M* on the shoulder-screws *N*. With this arrangement, gear *A*, pins *C*, and disks *E* comprise the driving member of the clutch,

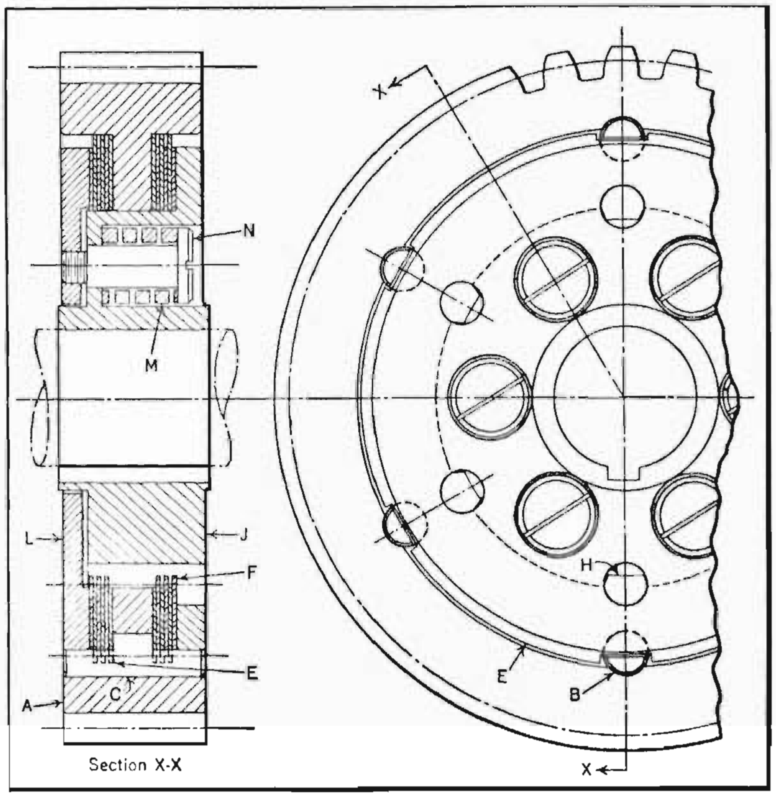


Fig. 3. Gear Equipped with Friction Drive for Stopping the Driven shaft when Excessive Loads are Applied

and disks *F*, hub *J*, springs *M*, screws *N*, and collar *L*, the driven member.

Any excessive torque applied to the driven shaft will cause the friction disks to slide on each other, thus stopping the rotation of the driven shaft until the excessive torque is removed. The magnitude of the torque transmitted is directly proportional to the total axial spring pressure applied on the clutch disks and their coefficient of sliding friction, and can be controlled by proper spring adjustment.

Overload Friction Release for a Large Gear Drive.— Each machine operating the large valves for filling and emptying the locks of a certain ship canal has embodied within the main spur gear an overload friction release.

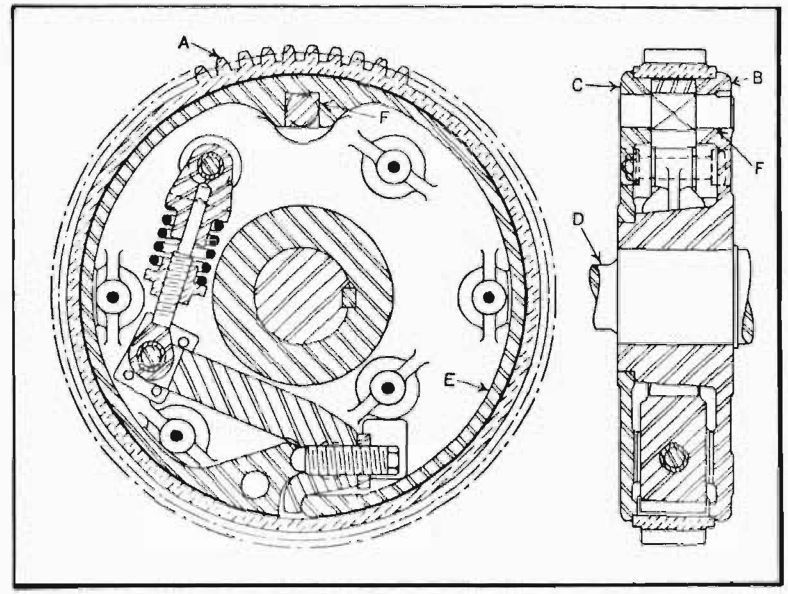


Fig. 4. Overload Friction Release with Adjustment for Controlling Point of Release

The object of this device is to prevent damage to the valve-lifting mechanism in case the valve gate should be suddenly stopped by some obstruction.

The main spur gear unit consists of a manganese bronze rim *A* (Fig. 4), about 37 inches in diameter, with teeth cut on its outer face. The gear is free to rotate in a groove formed by the cast-steel casing *B* and the cover *C*, which are bolted rigidly together and keyed to the gear-shaft *D*. In the casing is an internal brake-band *E* of cast steel,

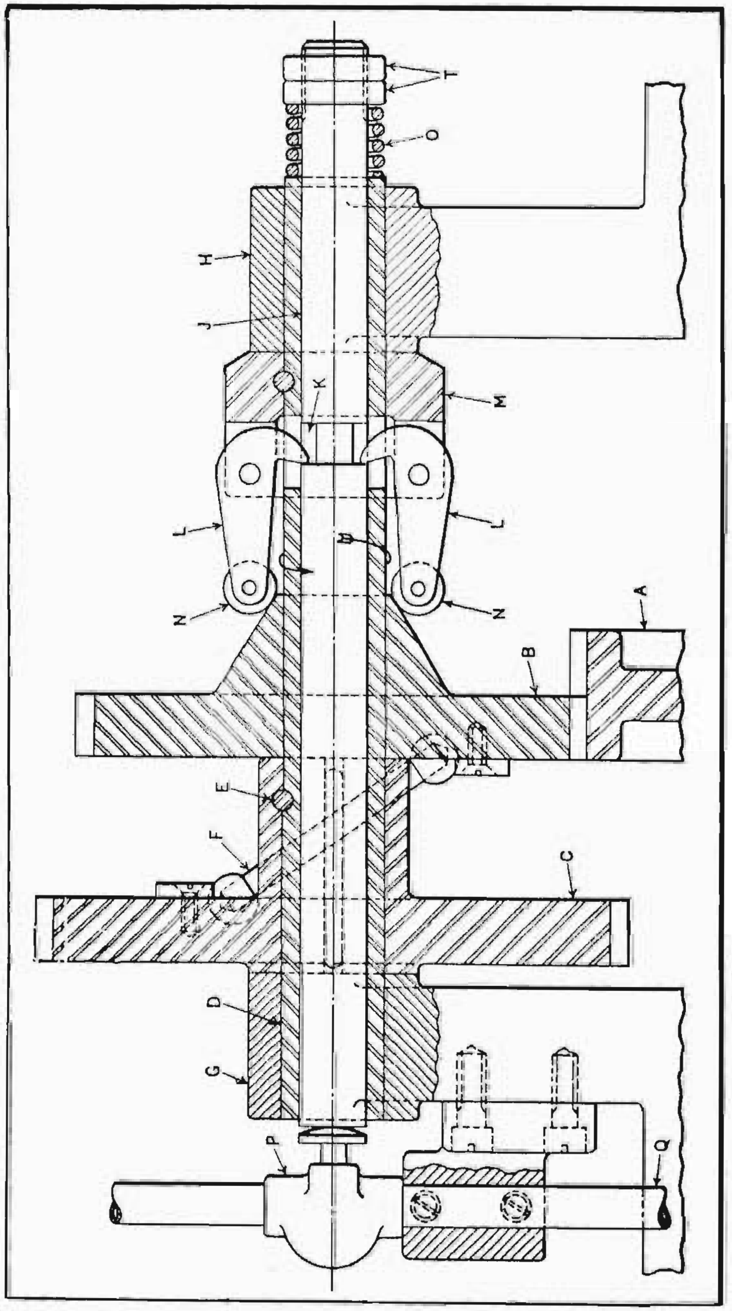


Fig. 5. Mechanism for Pneumatically Disengaging Clutch at Remote Point when Machine is Overloaded

which has an asbestos lining secured to it with copper rivets. The band is pivoted to the casing by the pin *F*.

A spring-actuated lever is provided to expand the brake-band and press it against the rim. Thus the torque is transmitted from the gear teeth through the brake-band to the casing and then to the shaft *D* to which the valve-operating drum is keyed. The spring mounting is adjustable, so that the proper load can be applied to the lever and the load regulated to compensate for wear on the band and rim. With this arrangement, the gear unit acts as a whole. However, should a log or other foreign material obstruct the valve, slippage would occur between the bronze rim and the brake-band and thus prevent damage to the machine or valve.

Pneumatic Overload Relief Mechanism for Automatically Disengaging Clutch at Remote Point.—A conveyor system is employed in a certain plant for delivering gravel over a relatively long distance to a washing and screening machine. Too large a quantity of material fed into the machine is likely to cause damage; to prevent this, an overload relief mechanism is provided on the machine for stopping the conveyor, the power for which is applied at some distance from the point of delivery of the gravel. With this arrangement, the relief mechanism opens a valve in a compressed air line when the machine is overloaded and delivers air to a cylinder, the piston of which disengages the conveyor clutch.

This mechanism is shown in Fig. 5; the air cylinder and conveyor clutch are omitted, as their design is generally known. The driving gear *A*, which rotates at a constant speed, transmits the required rotary movement to the machine through gears *B* and *C* and another gear (not shown). Gear *C* is secured by pin *E* to the hollow shaft *D*, supported in the stationary bearings *G* and *H*. Gear *B* is a running and sliding fit on this shaft, but when the machine is not overloaded, is caused to rotate with gear *C* by the bar *F*.

Balls secured to the ends of this bar engage corresponding ball sockets in gears B and C.

Plunger J which operates the valve P in the compressed air line Q when the machine is overloaded is a sliding fit inside of the hollow shaft. Plunger J contains a groove K which is engaged by the two fingers L. These fingers are pivoted in the collar M, which is pinned to the hollow shaft. At the left-hand ends of fingers L are rollers N, which rest on the tapered hub of gear B. The fingers L, gear B, and plunger J are held normally in the position shown by the coil spring O. The collar-nuts T provide the necessary adjustment for setting the tension of spring O to hold gear B in the position indicated when the machine is not overloaded.

In operation, gear A rotates shaft D in the direction indicated by the arrow. When the machine is running under a normal load, gear B maintains the axial position shown. However, if the load becomes excessive, the pressure against the ball ends of bar F will increase so that the bar will push gear B toward the right. As this movement occurs, the tapered hub of gear B opens fingers L, which causes plunger J to be forced toward the left and the button in valve P to be depressed.

In this way, air is admitted to the line Q leading to the clutch-operating cylinder, which causes the piston to disengage the clutch and stop the conveyor. When the excessive load on the machine is relieved, gear B once more returns to the normal position shown, causing plunger J to move toward the right and close the air valve. Springs provided on the air cylinder then return the piston and thus re-engage the conveyor clutch.

It should be mentioned that a small hole is drilled in the air cylinder head at the pressure end to permit the air to escape when the piston is actuated by the springs. This allows the air to leak out of the cylinder fast enough to permit the springs to return the piston when the air valve P

is closed, but not so fast that full line-pressure will not operate the piston. This leak hole has another important advantage in that it prevents the operation of the clutch cylinder through leakage which might occur in the valve P.

Overload Relief for Oscillating Lever.—A release mechanism that was designed for a feed slide subject to jam-

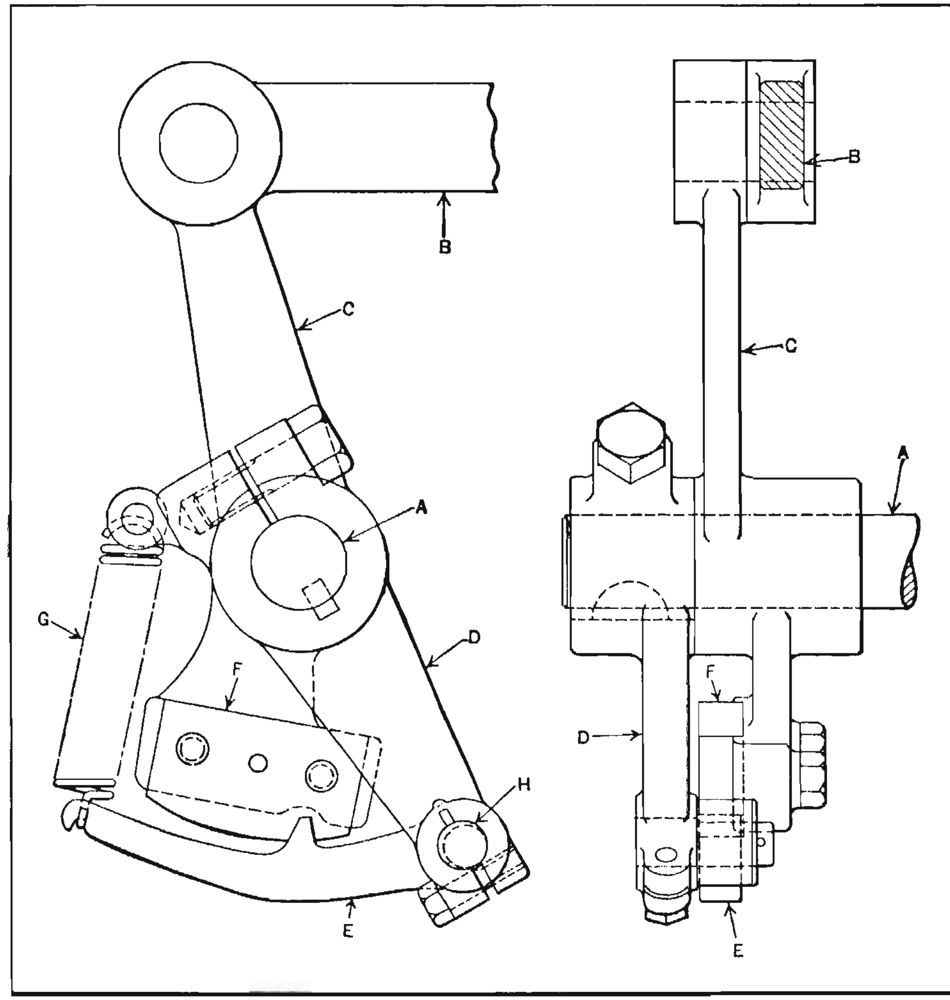


Fig. 6. Arrangement for Automatically Disengaging a Driving Lever from its Shaft when the Load Becomes Excessive

ming but that can also be applied to various types of movements is shown in Fig. 6. Oscillating shaft *A* transmits a reciprocating movement to link *B* connected to the feed slide (not shown) through the lever *C*. Lever *C* is a slip fit on the shaft, but is prevented from turning by a locking arrangement consisting of lever *D*, locking bar *E*, locking plate *F* secured to a projection on lever *C*, and spring *G*. At the outer end of lever *D*, which is keyed to the shaft, is pivoted the bar *E*. A tooth in this bar engages a notch in plate *F* and is held in this position by the spring *G*.

Normally, the entire mechanism is locked together and rocks back and forth with the shaft. However, if link *B* becomes overloaded, lever *C* will stop oscillating and shaft *A* will merely turn in the hub bore of this lever. Lever *D*, being keyed to the shaft, will continue to oscillate and cause the tooth on bar *E* to ride out of the notch and slide along the now stationary plate *F*. The tooth will continue to slide back and forth along this plate and in and out of the notch until the overload on link *B* is removed. When this is done, the tooth will engage the notch and the entire mechanism will once more function as a unit. An eccentric stud *H* is provided so that the angular position of lever *C* can be adjusted to vary the position of link *B* at the beginning and end of its stroke.

Overload Slip Arrangement for Feed-Screw.—The overload slip mechanism, Fig. 7, is so designed as to allow for the application of varying loads. The slide *A* of this mechanism is operated by means of a threaded sleeve or feed-screw *B* in the threaded hole *C*. The rod *D* passes through sleeve *B* and has a handwheel pinned to one end. The hub *E* of the handwheel has a cam-shaped end which is in contact with a similar cam face on the end of sleeve *B*. The opposite end of rod *D* is threaded and fitted with lock-nuts *F*. When the handwheel is turned until the screw *N* comes in contact with button *O*, any additional movement of the handwheel will cause the cam face on hub *E* to ride up on

the cam surface on sleeve *B*, compressing the spring *P*. When the cam load reaches the high point, spring *P* causes rod *D* to return to its original position. Varying pressures from zero to maximum can be obtained either by adjusting nuts *F* to vary the loading of spring *P* or by increasing or decreasing the angle on the cam faces of the handwheel hub and sleeve *B*. Both the spring pressure and the angle of the cam faces can, of course, be adjusted when this seems desirable.

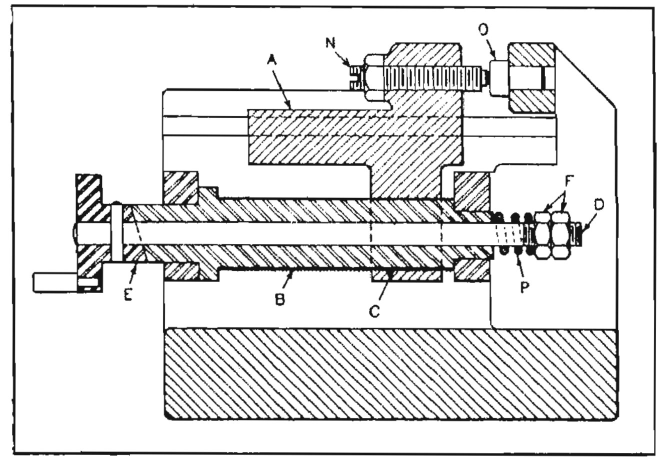


Fig. 7. Feed-screw Operated by Handwheel that Ceases to Turn the Feed-screw when the Slide Meets Obstruction or is Overloaded

Ratchet Feed with Automatic Overload Safety Stop.—A ratchet feed mechanism provided with a safety attachment that protects the mechanism from breakage in case the feed becomes jammed, and that also serves to stop the machine when this occurs is shown in Fig. 8. The attachment is so designed that the feed can be reengaged as soon as the obstruction has been removed. Previous to the installation of this attachment, a shear pin was used to protect the feeding mechanism from breakage. The shear pin arrangement merely protected the feed mechanism and did not prevent the loss in production that resulted from operating the machine while the feed was jammed; in addition, it required the suspension of production while the shear pin was being replaced.

Referring to the illustration, the feed-shaft *A* is given an intermittent rotary movement by means of the ratchet wheel *B* and the pawl *C* which is carried on lever *D*. Lever

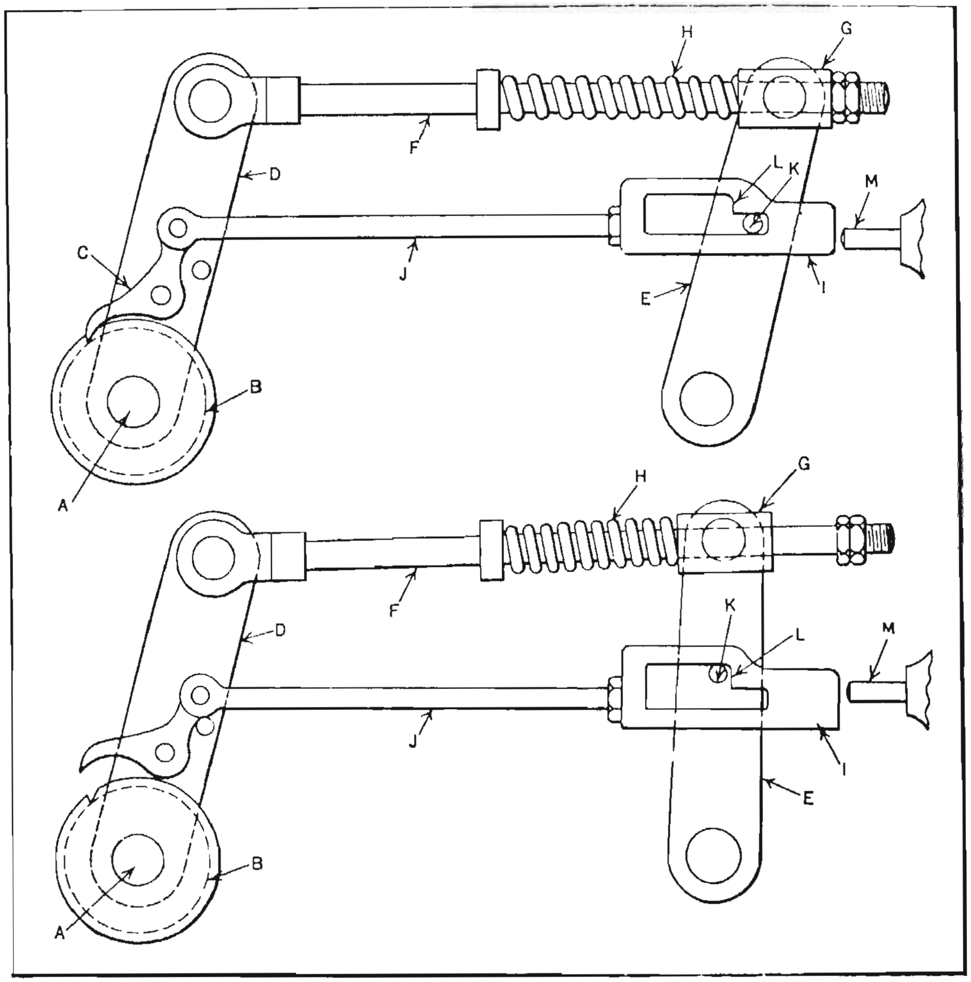


Fig. 8. Ratchet Feed with Attachment for Protecting Mechanism and Stopping Machine if Feed Jams

D receives its movement from lever *E* through the connecting-rod *F*. Lever *E* carries the swinging yoke *G* through which rod *F* passes. Rod *F* carries the spring *H*, which is compressed when the load on shaft *A* exceeds a predetermined limit. Pawl *C* is connected to the plate *I* by the connecting-rod *J*. Plate *I*, which has an irregular-shaped hole, rests on pin *K* carried on lever *E*. Under normal conditions, connecting-rod *J* carries no load, merely riding between levers *D* and *E*.

The upper view of the illustration shows the mechanism in its normal operating position. The oscillating movement of lever *E* is transmitted to shaft *A* by pawl *C*, which is held in engagement with the ratchet wheel *B* by a spring (not shown). Should the movement of shaft *A* be prevented, the continued movement of lever *E* would simply result in compressing spring *H*, thus preventing the breaking of parts. The forward movement of lever *E* carries pin *K* into the larger portion of the irregular hole, as indicated in the lower view. On the return stroke of lever *E*, the pin *K* engages the shoulder at *L* on plate *I*, thus causing pawl *C* to remain out of engagement with ratchet wheel *B*, and preventing further movement of shaft *A*, although the machine may not be stopped immediately. In this position, plate *I* extends beyond lever *E* sufficiently to push over the rod *M* far enough to open the electric switch that controls the driving motor, and thus cause the machine to come to a stop.

When plate *I* is lifted and disengaged from pin *K*, the machine is again ready to start, but if the resistance of shaft *A* is greater than the tension of spring *H*, the machine will again be stopped on the first stroke of lever *E*. In the actual construction, two pairs of levers *D* and *E* are used, the mechanism being located between them. In order to show the mechanism more clearly, however, the outer levers have been omitted in the illustration.

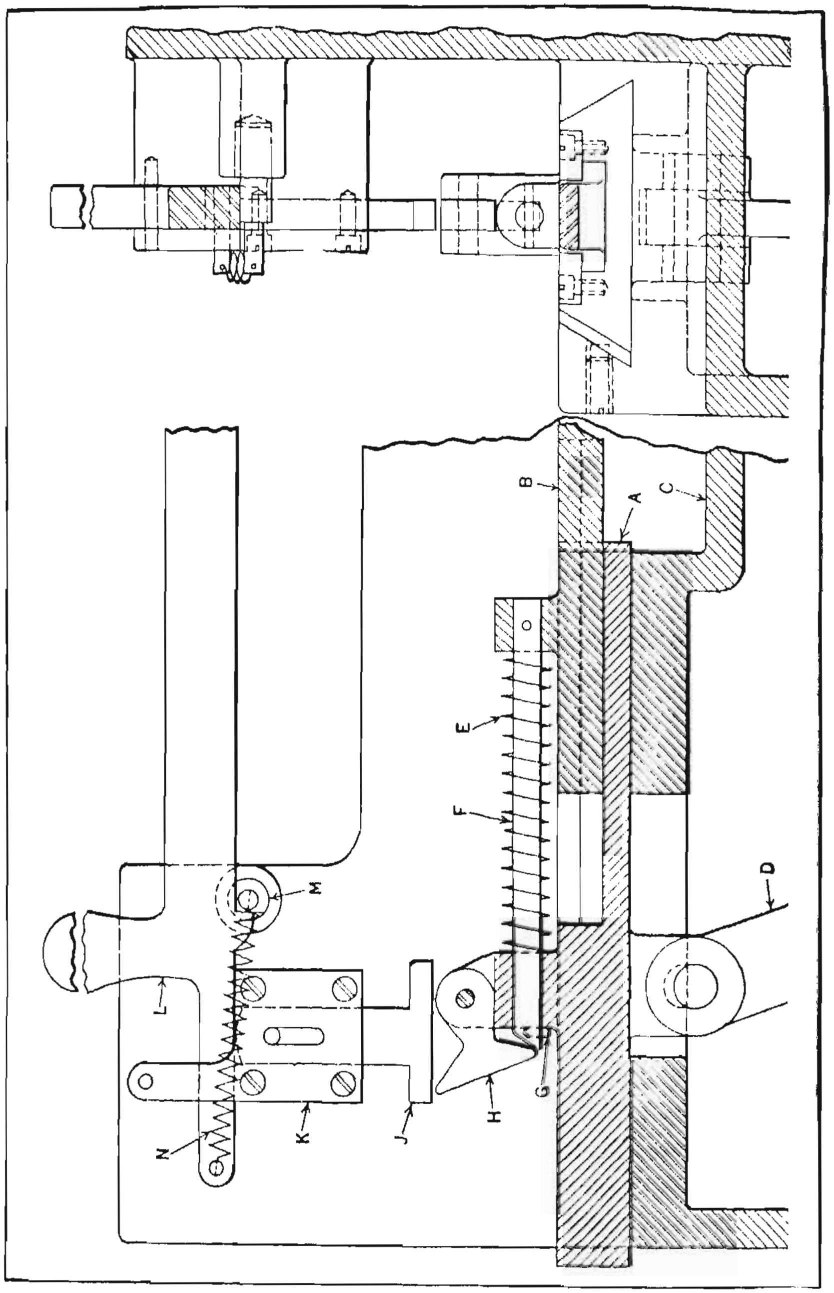


Fig. 9. Mechanism for Stopping Slide Automatically when it Becomes Overloaded

Mechanism for Instantly Disengaging Clutch at Point of Overload.— Many mechanisms designed to disconnect the power drive to the machine when it becomes overloaded function only at one point in the operating cycle. With the mechanism shown in Fig. 9, however, the machine clutch through which power is transmitted to the slide is disengaged instantly, at the exact point in the slide movement at which the overload occurs.

This arrangement is incorporated in a machine for assembling metal caps on electric fuse plugs, a number of the plugs being capped simultaneously. The capping tool slide actually consists of two slides *A* and *B*, slide *B* being superimposed upon slide *A*. Slide *B* carries the capping tools and is normally held in one position relative to the main slide *A* by means of a stiff coil spring *E*. If the tool-carrying slide *B* meets with an obstruction, it telescopes into the main slide, actuating a latch *H*, through rod *F*, which causes a spring-operated hand-lever *L* to shift and disengage the machine clutch. Although not shown here, a band brake operated by the same hand-lever prevents over-run of the machine members after the clutch is disengaged. Slide *A* is reciprocated in a dovetail guide in the machine frame *C* by the oscillating lever *D*. This lever is actuated by another member of the machine (not shown).

Rod *F*, together with stop-pin *G*, limits the telescoping movement of the slides, in addition to tripping the pawl *H* when the slide meets an obstruction. Pawl *H* is pivoted at the top of the main slide lug, and when swung upward engages latch *J* sliding in the guide *K* on the machine frame. The upper end of this latch, when the latter is raised by pawl *H*, serves to disengage the clutch lever *L* from the stationary pin *M* in the machine frame.

The coil spring *N*, secured to pin *M*, then forces the lever *L* toward the right, disengaging the machine clutch and applying the band brake. All these movements take place at practically the same instant that the overload occurs, so

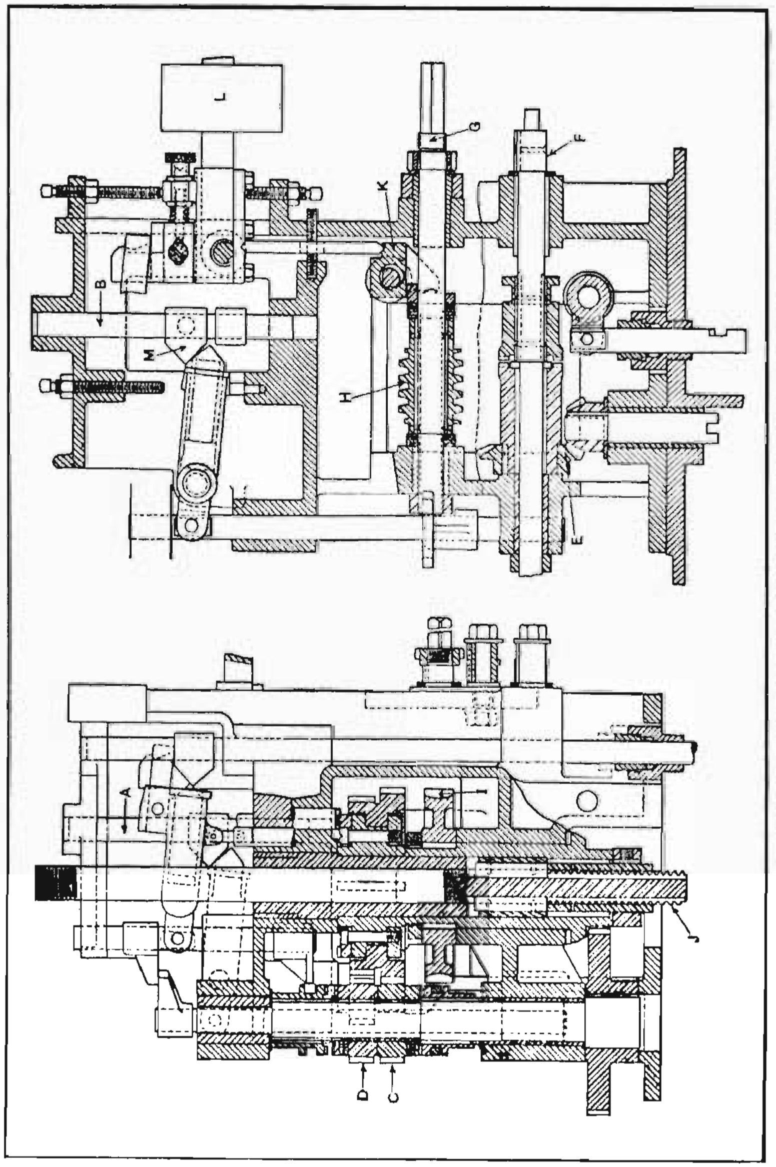


Fig. 10. Mechanism which Automatically Disengages the Feed if Pressure on Tool Becomes Excessive

that no further movement of slide *B* results after meeting the obstruction. Slide *A* is then moved to the left by hand and the obstruction removed, after which the machine is started by shifting lever *L* back to its original position.

Disengagement of Feeding Mechanism when Pressure on Tool is Excessive.—The mechanism Fig. 10 is part of an "automatic" of the vertical type. The rods *A* and *B* carry dogs which engage stops on the frame of the machine and, under normal conditions, trip the advance and return feed movements respectively. The rapid-traverse motion, which retracts the tools quickly and can also be utilized to bring them forward rapidly to the point of cutting, is operated through gears *C* and *D* controlled by clutches. The advance feed for cutting is through bevel gears *E* and change-gears connecting shafts *F* and *G*, the worm *H* on the shaft *G* driving worm-wheel *I*. These two trains of mechanism give the desired advancing and retracting movements through connection with screw *J*. A feature of this feed is in providing means for automatically tripping it whenever the pressure on the cutting tool becomes excessive. This is accomplished by providing a thrust bearing for the worm *H* which, through bellcrank *K*, is held in place by an adjustable weight *L*. When the pressure is sufficient to raise the weight, the mechanism operates to trip the latch *M* and engage the return motion the same as if the regular tripping point had been reached.

Spring-Plunger Release Mechanism for Preventing Damage to Reciprocating Parts.—Mechanisms for automatically preventing damage to reciprocating parts are sometimes required when there is the possibility of such an occurrence resulting from the jamming or overloading of the machine. A mechanism of this kind is shown in Fig. 11. The reciprocating movements of the parts to which this mechanism is applied are obtained by means of the lever *A*, which swings back and forth as indicated by arrows *E*.

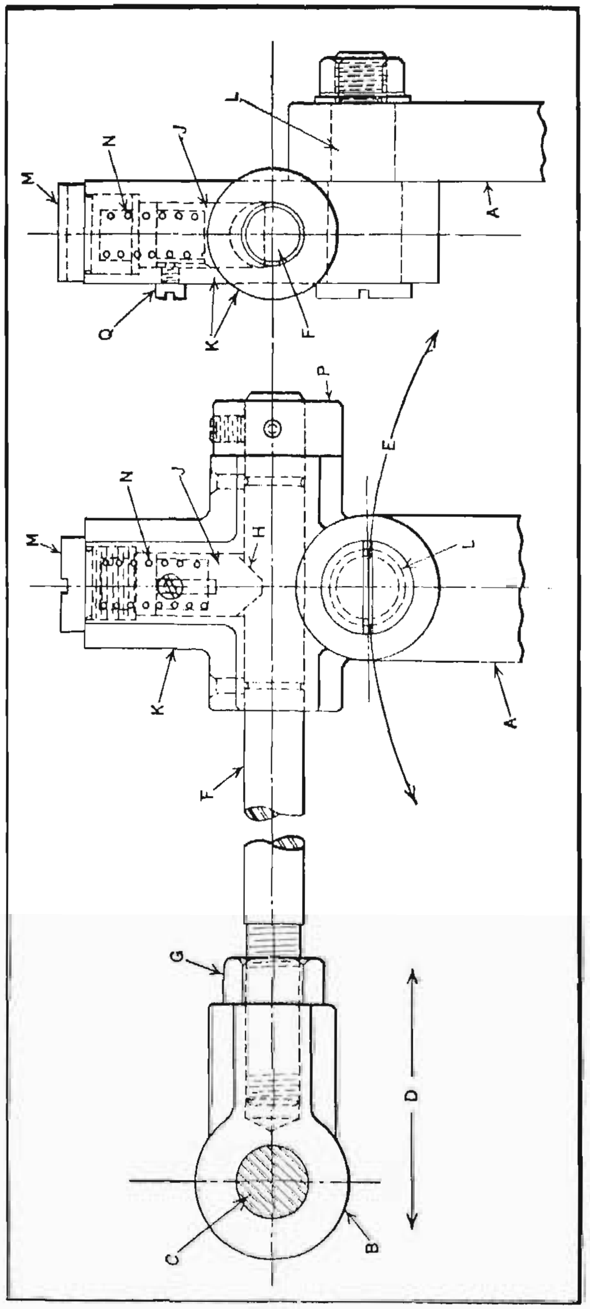


Fig. 11. Release Mechanism for Reciprocating Lever Designed to Prevent Breakage Due to Overloading or Jamming of Machine Parts

Lever *A* is mounted on a shaft (not shown) which is operated by a cam-and-roll mechanism. At *B* is shown a connecting link, through the center of which passes a short shaft *C* by means of which the rocking action of the lever *A* is transmitted to a vertical reciprocating slide. This slide travels back and forth in the directions indicated by arrows *D*, being propelled through the medium of the connecting-rod *F*, which is adjustable in the link *B*. Rod *F* operates in a vertical position instead of in the horizontal position shown in the illustration. This rod is securely locked in place by nut *G*, the adjustment being obtained by making a complete revolution of rod *F*, so that the crosswise notch at *H* will be in the proper position to receive the locking plunger *J*.

Ordinarily, the movement of lever *A* is imparted directly to shaft *C*. The safety unit is provided to prevent breakage in case the machine becomes jammed or overloaded. This safety device consists of the cast-iron housing *K*, which is free to pivot on stud *L* in lever *A*. Screw *M* holds spring *N* in place, so that the plunger *J* is forced into the notch *H* in rod *F*. Under normal operating conditions, the unit acts as a non-yielding driving block. In the case of an overload on shaft *C*, spring *N* yields sufficiently to permit plunger *J* to snap out of notch *H*, with the result that lever *A* and block *K* will reciprocate without imparting any motion to rod *F*, thereby preventing the driven parts from being broken or damaged.

When the obstruction is removed, plunger *J* automatically springs back into notch *H* and the normal operation of the machine is resumed. The collar at *P* is provided to insure a positive driving movement for rod *F* on the down stroke. Thus, the driving movement imparted to rod *F* is interrupted only on the upward stroke. There is a pin-shaped end on screw *Q* that enters a keyway in plunger *J* and thereby prevents the latter member from turning in block *K*.

CHAPTER VII

REVERSING MECHANISMS OF SPECIAL DESIGN

Reversing mechanisms may be designed to act at a fixed point in the cycle of movements or to vary the point or time of reversal. A reversal of motion in some cases may also be accompanied by a change of velocity. This chapter deals with reversing mechanisms of the different types mentioned and includes only special designs not found in Chapter VI of Volume I (pages 161 to 197).

Compact Reverse Mechanism of Rapid-Acting Parallel Worm Type.—In a certain type of can-seaming machine, the work is controlled by a mechanism having a continuous reciprocating movement. This mechanism provides a traverse movement of constant velocity. The reversals are positive and practically instantaneous. A compact design was essential in this instance, because the mechanism was used in making an alteration to a machine where the small space available made it impossible to use a long-throw cam. A nut and feed-screw provided with the usual dog-operated reversing mechanism was considered, but was rejected owing to the lost motion attending each reversal.

The mechanism is mounted on the machine frame *A*, Fig. 1, and consists essentially of the two worms *B* and *C* and the follower-roll *D*. Both the worms have right-hand threads and are rotated at a constant velocity in opposite directions by means of gears *E* and *F*, mounted on their respective worm-shafts. These gears, in turn, are rotated by gear *G* on the shaft *H*, which is driven by another member of the machine. Follower-roll *D* is free to turn on its bearings in cross-slide *K*, which moves laterally in the

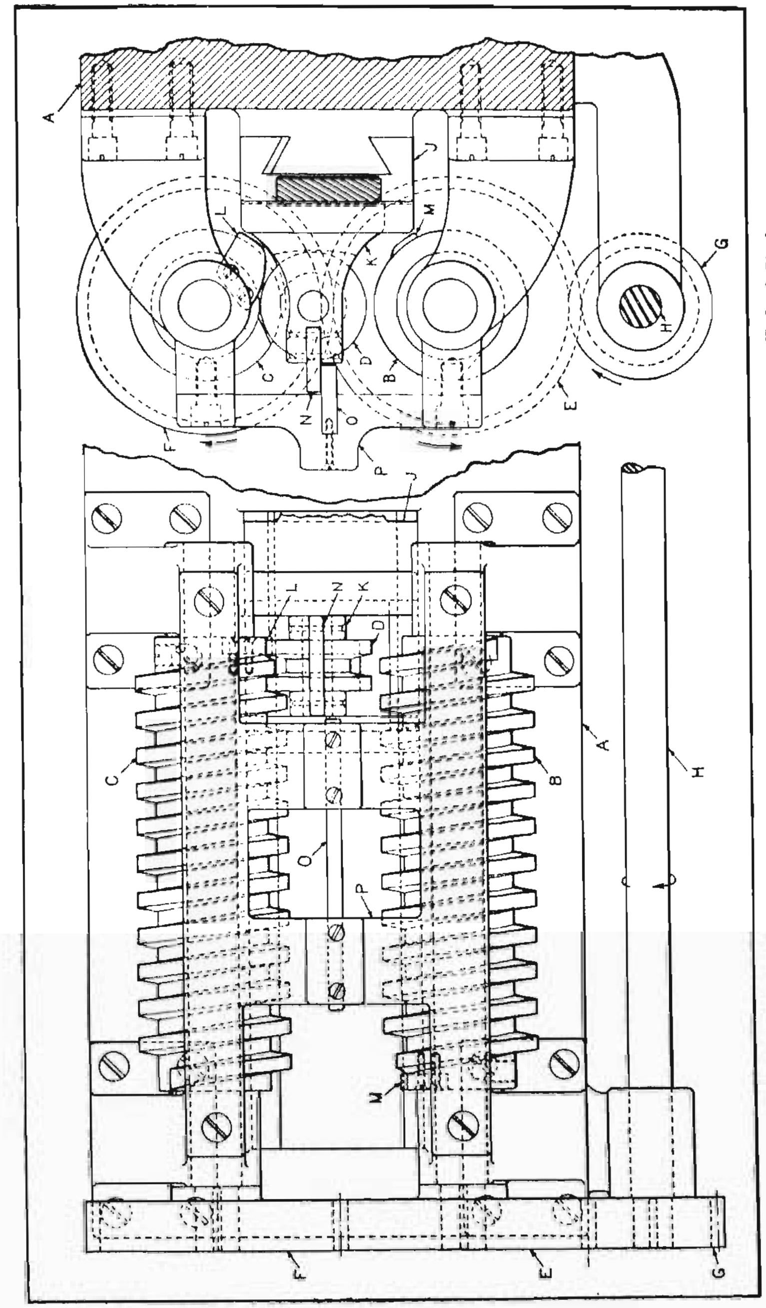


Fig. 1. Mechanism for Obtaining Rectilinear Movement with Rapid Reversal at Each End of Stroke

slide J. Slide J is mounted on a dovetail guide on the machine and transmits the required movement to the work. This slide is given a reversal of its movement at each end of its stroke through the action of cam lugs L and M, secured by screws to opposite ends of worms C and B, respectively. The roll is held in engagement with each worm by the insert N which rides along one side of stationary bar O, depending upon which worm is engaged with the roll. Bar O is held in the stationary position by the top plate P, secured to the bearings of the worm-shafts.

When the follower-roll has reached the position indicated, slide J is at the end of its right-hand stroke. It will be noted that insert N on the roll cross-slide has just passed the end of bar O. Now as the worm continues to rotate, the cam lug L comes in contact with one flange of the roll and forces it over into engagement with the beginning of the thread on worm B, holding or locking it in this angular position until the worm thread has carried the left-hand end of the insert N past the right-hand end of bar O. Worm B then reverses the movement of the follower with slide J, carrying it toward the left until, at the end of the stroke, cam M comes into contact with the other follower-roll flange, which forces the latter over into engagement with worm C.

The roll is held in this position until its movement toward the right carries insert N past the end of bar O, the bar preventing disengagement of the roll and worm during the remainder of the stroke. Thus worm C returns the roll and slide J to the position shown, where the reversal of the slide J is repeated. The reversals of slide J are effected rapidly and with absolutely no shock. The pressure between insert N and bar O is insignificant, owing to the relatively small angle of the worm thread. However, in order to insure a long life, as well as to increase the efficiency of the unit, both of these members are hardened and ground on their wearing surfaces.

Mangle Gear Mechanism for Changing Direction of Rotation.—The mangle gearing mechanism shown in Fig. 2 is designed to drive, from a continuously rotating shaft P, a shaft S a portion of a turn backward and forward. The pinion shaft P is driven through universal joints which permit it to move back and forth in the slot in guide B. To

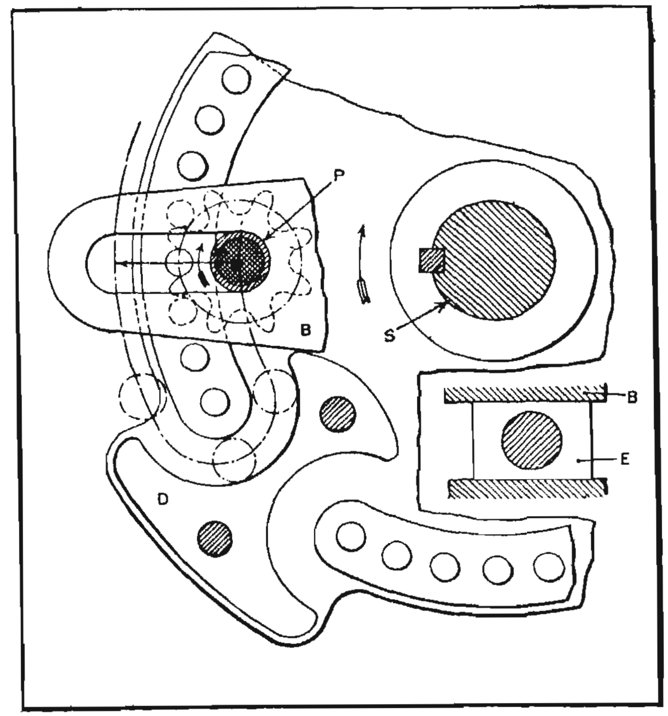


Fig. 2. Pin Type of Mangle Gearing for Reversing Rotation of Driven Shaft S

the shaft S is keyed a center plate to which is attached concentrically the mangle gear proper, consisting of a ring of cast iron or steel fitted with a number of pins which act as gear teeth and which mesh with the teeth of a gear of the sprocket type.

To the center plate is attached a reversing dog or guide D into which the end of the sprocket shaft passes, restricting the movement of the sprocket, and causing it to move from one side of the pin ring to the other as the sprocket and ring rotate together. After the passage of the sprocket from one side of the ring to the other, the ring turns in the opposite direction; that is, its motion is reversed, but its velocity remains unchanged, except, of course, during the passage of the sprocket to the other side of the ring while it is in contact with guide *D*. In passing through the guide, the end of the sprocket shaft travels from one end of the

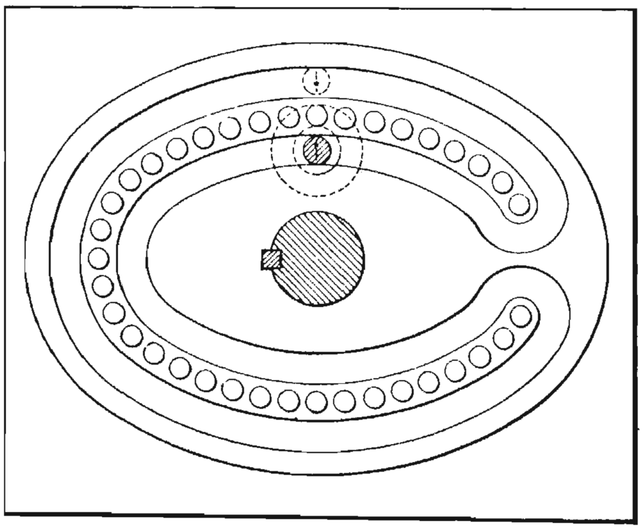


Fig. 3. Reversing Mechanism Similar to that Shown in Fig. 2, but Designed to Produce Variable Velocity

slot in the fixed bracket *B* to the other. The ends of the slot prevent the sprocket from being forced out of mesh with the pins. When the center plate has rotated through its full complement of a turn, the sprocket is again transferred to the side of the ring with which it was previously in mesh through the action of the opposite guide. Instead of rotating directly in the bracket *B* as a bearing, the shaft may rotate in a sliding block *E*, such as shown in the small cross-sectional view.

In Fig. 3 is shown a variation of the type of mechanism

just described. This design is arranged to vary the velocity of the driven shaft. The end of the pinion shaft may be fitted with a ball journal bearing.

Mangle Gearing for Reversing Rotation of Shaft After One Complete Turn.—The pin type of mangle gearing mechanism shown in Fig. 4 is designed to reverse the driven shaft after it has made a complete turn. With this

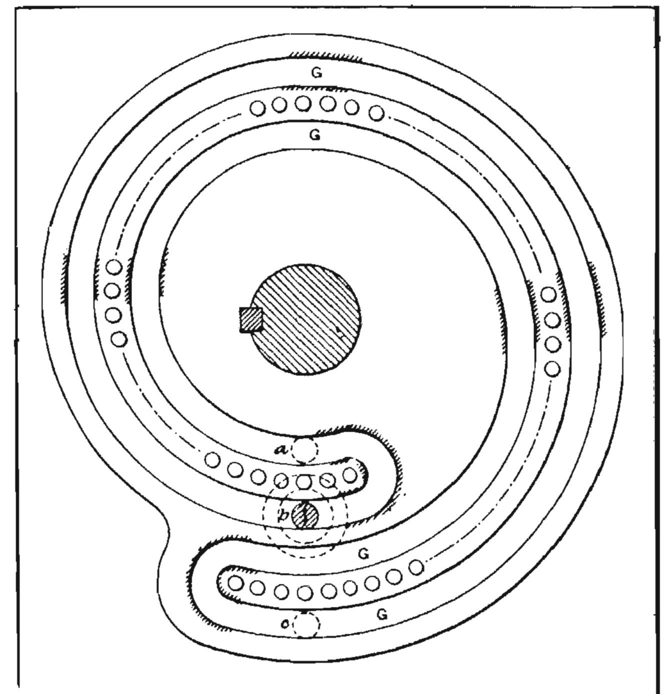


Fig. 4. Pin Type of Mangle Gearing Arranged to Reverse Driven Shaft after One Complete Turn

mechanism, the driven shaft has a somewhat variable motion. The smaller the lead of the spiral in relation to the distances of the pins from the shaft center, the less will be the velocity variation. Fig. 5 shows one arrangement for driving the pinion shaft *b*, Fig. 4, which provides for the required oscillation of the pinion or sprocket shaft.

It is obvious that an infinite number of velocity combinations are possible by varying the shape of the mangle gear shown in Fig. 4. The continuous groove G serves to keep the sprocket in mesh with the pins. The end of the pinion shaft b may either rotate directly in contact with the groove or it may bear against the groove side through a ball journal bearing attached to the shaft.

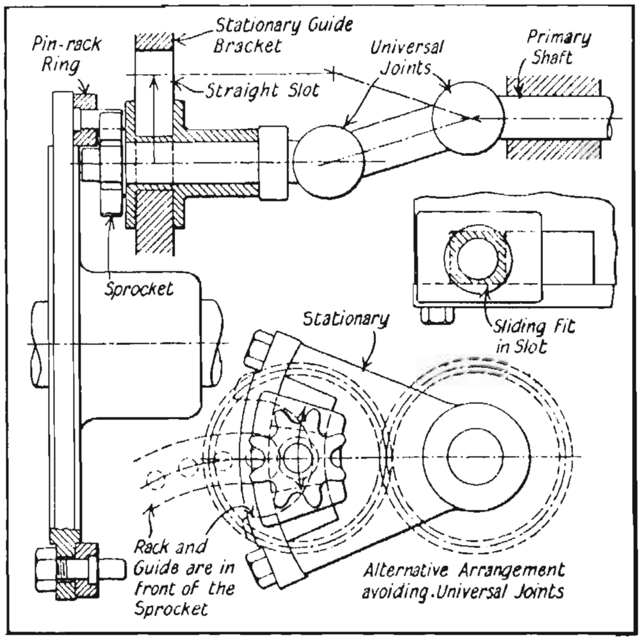


Fig. 5. Types of Drives Arranged to Permit Oscillation of Driving Shafts such as the One Shown at b, Fig. 4

Perhaps the most common method of driving the pinion of a mechanism of this kind is by bevel gears, the bevel gear on the pinion shaft serving as a universal joint. This method has the objection, however, that owing to the oscillation of the shaft, the pinion occupies different angular positions, not only during the oscillation but when it is driving. To allow for the change in the angle, the teeth of the pins must be barrel-shaped. Generally they are permitted to assume this shape through wear. The driving

method shown in Fig. 5 is preferable to the bevel gear drive.

Shaft-Reversing Mechanism Giving Higher Velocity in One Direction.—When a shaft-reversing mechanism is required in which the velocity in one direction must or can be greater than in the other, the driver and driven elements may be ordinary gears or gear segments, such as shown in Fig. 6. With this type of mangle gearing, the velocity of

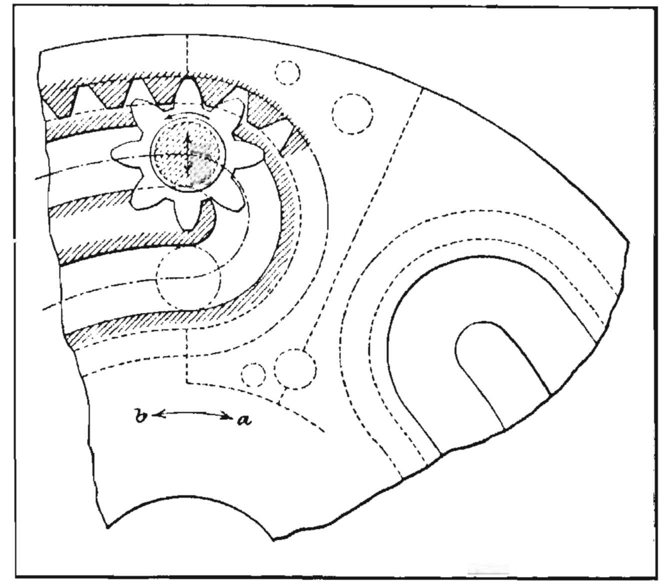


Fig. 6. Shaft-reversing Mechanism that Gives Higher Velocity in One Direction

the driven shaft is greater in the direction indicated by arrow a than in the direction b. The velocity is uniform, however, in each direction. The pinion shaft in this case is guided wholly by a groove in the center plate, into which the end of the pinion shaft projects.

In Fig. 7 is shown a double-edge rack segment-form mangle gear for obtaining reversal of the driven shaft. The round end c of the gear ring can be half of a pinion, having a boss or hub by which it is located in a drilled hole in the plate.

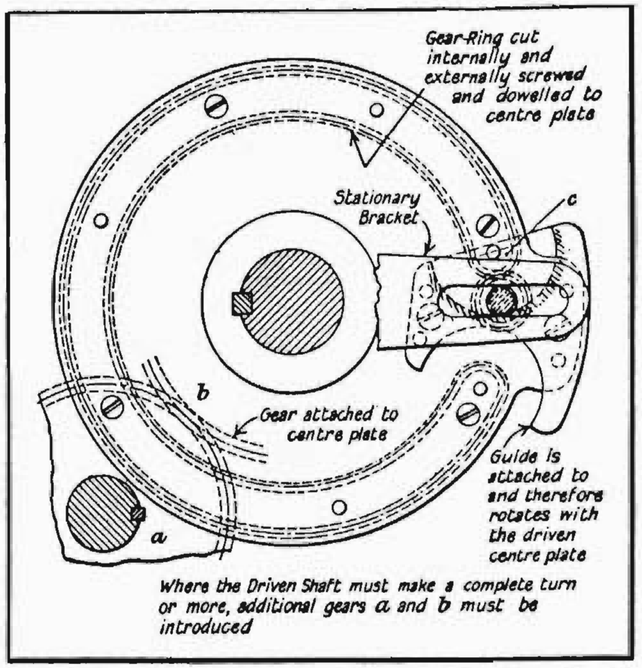


Fig. 7. Mangle Gear Shaft-reversing Mechanism with Additional Gears for Obtaining a Complete Turn or More than One Turn

Shockless Reversing Mechanism which Varies Point of Reversal.—Some mixing machines of the agitator type, employed for mixing liquid or plastic materials, require a reversing movement of the agitators; at the same time, however, the point at which reversal occurs must advance uniformly. These combined movements may be obtained by means of the mechanism shown in Fig. 8.

Here the drive shaft D, rotating at a uniform speed, imparts the required movement to the shaft G through the action of a combination planetary and eliiptical gear train. All three shafts D, L, and G rotate in stationary bearings, and owing to the ever changing radii of the elliptical gears at the tooth contact, an alternating accelerated and retarded movement is imparted to shaft L. This movement, in turn, is transmitted by spur gears J and K to the ring

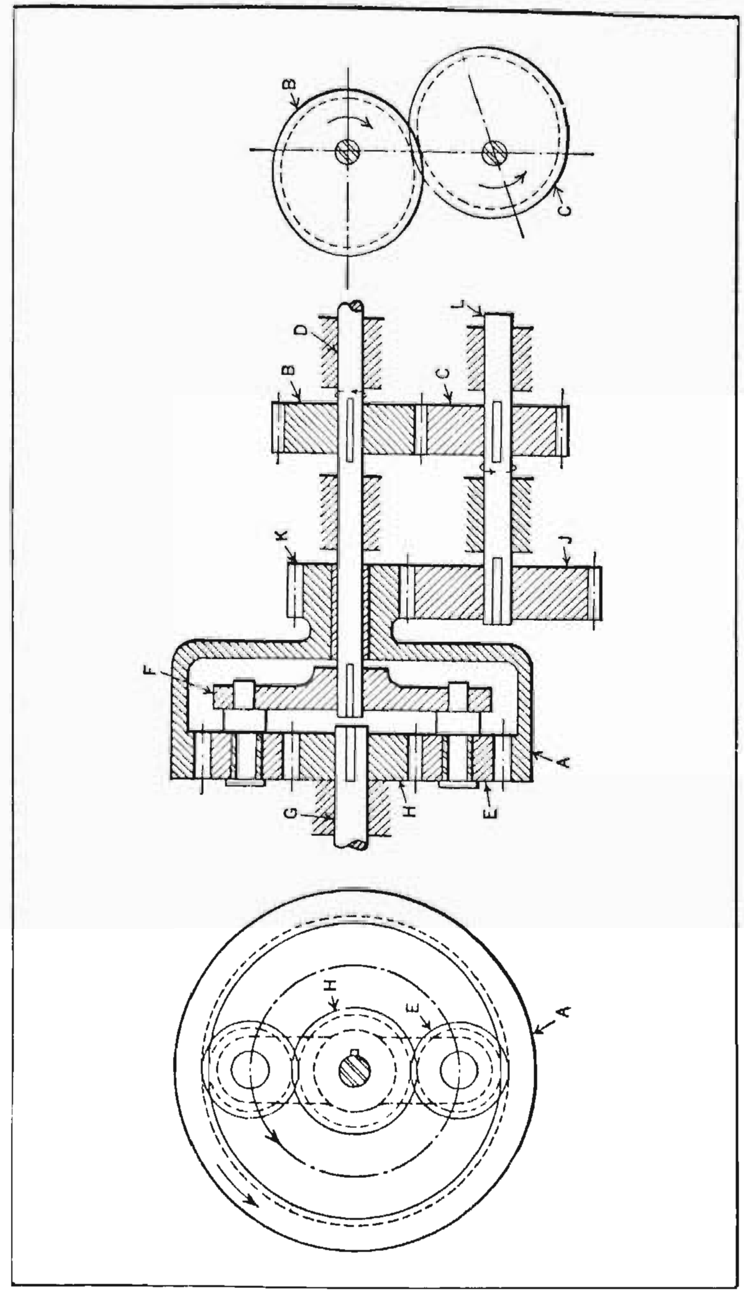


Fig. 8. Planetary Gear Train which is Given a Variable Movement by Elliptical Gears to Reverse the Movement of Shaft G

gear *A*. Now assume that the elliptical gears *B* and *C* have rotated into the positions indicated in the end view at the right, and that the ratio of spur gears *J* and *K* and also the ratio (momentarily) of the elliptical gears is such that the velocity of the centers of pinions *E* is one-half that of the pitch line velocity of the ring gear. Then, according to the principle of epicyclic gear trains, the pinions will simply roll about and not rotate the spur gear *H*.

Now if the shaft *D* rotates in the direction of the arrow, the ratio of the elliptical gears at the tooth contacts will gradually change so that the movement of shaft *L* and ring gear *A* will be retarded. Therefore, as the pitch line velocity of the ring gear decreases, the velocity of pinions *E* relative to the spider *F* will also decrease, and the lag of these pinions will cause the gear *H* and shaft *G* to rotate in the same direction as shaft *D*. This movement of shaft *G* will be accelerated until the elliptical gear *B*, whose engaging radius is gradually diminishing, has rotated through an angle of 90 degrees. At this point the ratio of the elliptical gears is at its minimum and as they continue to rotate, the ratio increases. This has the effect, through the movement transmitted to the pinions, of retarding the angular movement of gear *H* and shaft *G* until the elliptical gear *B* has passed through another 90-degree angle. At this time, the elliptical gears are once more (momentarily) in a position where the pinions roll about but do not rotate gear *H*; and, on further rotation of the elliptical gears, the velocity of the pinions will be gradually increased with respect to spider *F*, thus reversing the angular movement of gear *H* and shaft *G*. This movement of shaft *G* will be accelerated during a 90-degree movement of the elliptical gear *B* and then retarded through the next 90 degrees, at the end of which time the point of reversal has again been reached and the mechanism has passed through a complete cycle.

The movement transmitted to the pinions during the first half revolution of the elliptical gears is slower than

the movement transmitted during the second half; and since the velocity of these pinions governs the amount of angular movement of shaft *G*, then the angular movement of this shaft, in a counter-clockwise direction, is less than that in a clockwise direction. Therefore, the point of reversal of the shaft *G* will vary or advance about the shaft center an amount equal to the difference in these two angular movements. By varying the ratio of the spur gears *J* and *K*, the advance of the reversal points may be increased or diminished to suit the requirements; or in case no variation of the reversal point is required, the same procedure may be followed. This type of mechanism, owing to its retarding and accelerating movements, is particularly desirable where reversal must take place without shock.

Oscillating Motion Converted to Variable Reversing Motion.— In a special electrical switch testing machine, an oscillating motion of one shaft is converted to a reversing motion in another shaft, the latter alternating at each reversal between the two speeds of 60 and 30 revolutions per minute.

The shaft *X* (Fig. 9), on which the segment gear *A* is keyed, is the oscillating member. The shaft *T*, to which the irregular motion is transferred, turns in the machine bearings (not shown) and serves as a pivot for the arm *B*. Gears *O* and *P*, located under this arm, are keyed on shafts *U* and *Y* and are connected by the three gears *S*, *V*, and *G*. The concentric grooves *E* and *D*, milled in the segment gear, are joined at both ends to form one continuous groove and serve as a guide for the cam-roll *C* in the end of arm *B*. Dogs *R* and *N*, which engage projection *Q* on the arm, are fastened securely to the segment gear. Latches *J* and *M* swing on shoulder-screws, and normally bear against pins *I* and *K*, due to the tension of the coil springs.

In the position shown in the illustration, the segment gear *A* is oscillating in the direction of the arrow, and the dog *R*, against lug *Q*, is about to swing the arm *B* around shaft *T*. A further upward movement of dog *R* will throw gear *O* out of engagement with the segment gear. However, just before the teeth of gear *O* have become disengaged, a partial engagement of the teeth in gear *P* and segment *A* takes place. While gears *O* and *P* are being shifted, roller *C* swings up to the beginning of the groove

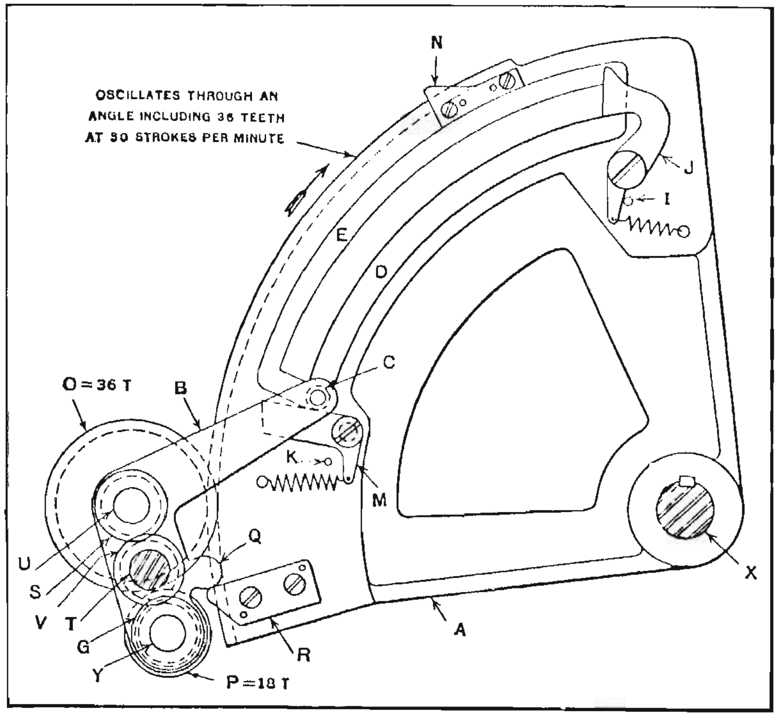


Fig. 9. Mechanism for Converting Oscillating Motion into Reversing Motion

E. When the roll reaches this position, the oscillating segment *A* has come to the end of its upward stroke and is about to return. The latch *M* closes the end grooves and prevents the roll from dropping back to groove *D* when the segment reverses.

The roll now follows groove *E* and serves to hold gear *P* in mesh with segment *A* until dog *N* comes in contact with the lug *Q*. This disengages gear *P*, after which gear

O is engaged with segment *A* again. In the meantime, roll *C* has forced latch *J* to one side and is swung down to the end of groove *D*, being prevented from coming out of this groove by the return of latch *J*. The roll, running in groove *D*, serves to hold gear *O* in mesh during the return stroke of the segment. This completes one cycle of the movements.

Because of the difference in the number of teeth between gears *O* and *P*, as noted in the illustration, and the arrangement of the gear train, the uniform oscillation of segment *A* will result in one clockwise revolution of shaft *T* for every up stroke of the segment, while the down stroke will result in two counter-clockwise revolutions of the shaft. With some slight modifications in the design, shaft *T* may be made to revolve at varying speeds other than described and in the same direction instead of reversing. This may be done by varying the number of teeth in the gears and adding an idler between any two of the gears *S*, *V*, or *G*.

Mechanism for Reversing Tap Spindles in Drill Head.— When more than one tap is used in a drill head, the problem of reversing the taps is often simplified by having one tapping spindle drive on the "in feed" and another spindle drive on the "return feed." The arrangement of the gearing for such a drive is shown diagrammatically in Fig. 10. In this case, the large drill head carries a number of drilling spindles (not shown in the illustration), in addition to the four tapping spindles, *A*, *B*, *C*, and *D*.

The drill head slides up and down on column *E*, being kept in alignment by an external projection that slides in a vertical track. The drive is obtained from a vertical shaft *F* at the end of which is keyed the pinion *G*. This pinion is in mesh with the gear *H* which drives the drilling and tapping spindles. The drive for the drill spindles is very simple and is not shown in the illustration. The drive for the tapping spindles begins with the clutch shaft *I*, which is driven from the gear *H* through the pinion *J*. The shaft *I* carries a sawtooth double clutch *K* which can be engaged with either the upper member L or the lower member M.

When the downward feed of the head begins, an arrangement of levers similar to the belt-operating mechanism on a planer causes the clutch K to engage the upper member L. The drive to the tapping spindles is then through the idler

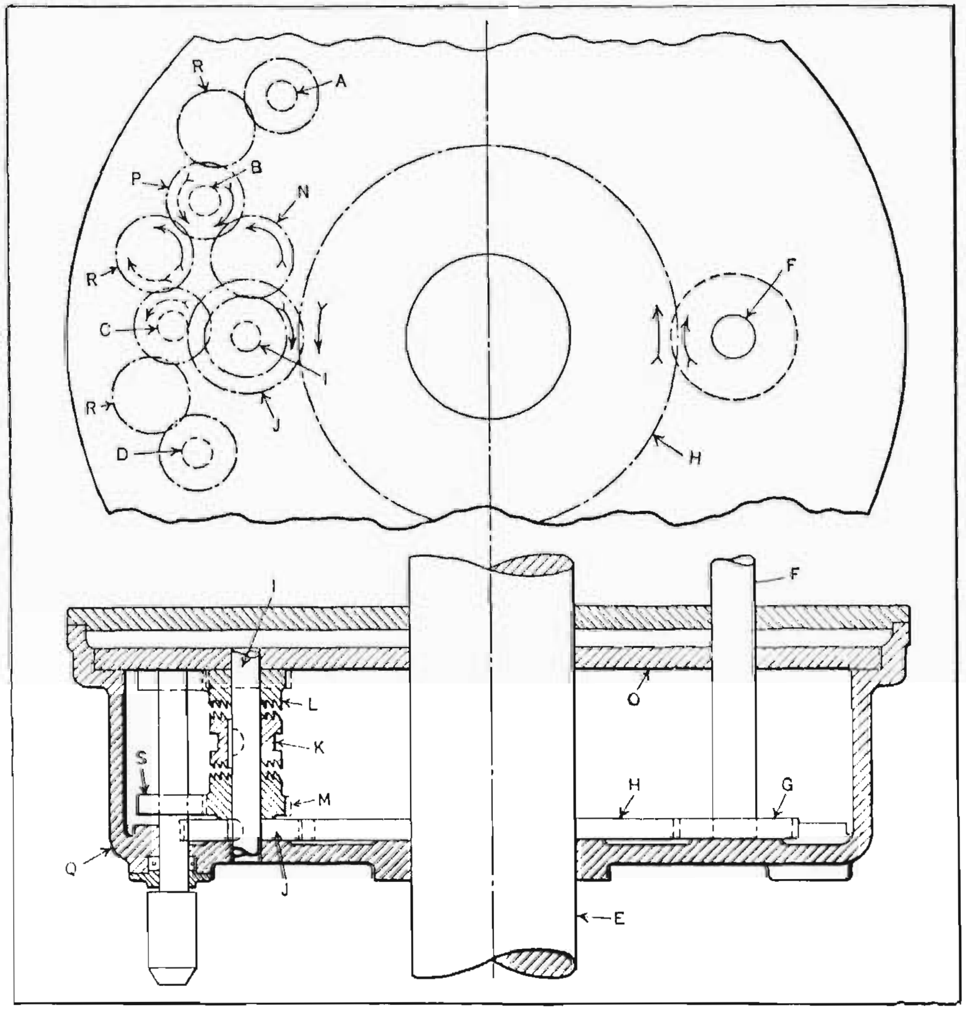


Fig. 10. Reversing Mechanism for Taps Used in Multiple-spindle Drill Head

gear N, which is mounted on the top plate O, and thence to the gear P on the top of the tapping spindle B. The tapping spindle is then revolved in the direction required for tapping. The other three spindles A, C, and D are driven in the same direction through the idler gears R which are mounted on the bottom of the gear-case Q.

As soon as the head begins to travel upward, the clutch K comes into engagement with the lower clutch member M and drives directly through gear S, which is fastened to the bottom of the tapping spindle C, revolving it, together with the other spindles, in the opposite direction. The full-line arrow-heads show the direction in which the meshing gears revolve when tapping, while the dotted arrows indicate the direction in which the gears revolve when the spindles are reversed on the "out feed."

Rotary Reversing Mechanism for Varying Angular Movement and Dwell of Driven Shaft.— Wire-forming machines of the four-slide type usually have various ingenious mechanical movements incorporated in their design that are applicable to machines used for other purposes. For example, in one four-slide, wire-forming machine, there is a reversing movement for a feed-slide shaft that has unusual features. This mechanism is designed to give the driven shaft a short dwell at each point of reversal. Besides, provision is made for varying the angular movement of the driven shaft without altering the dwell. The movements are transmitted from another shaft which oscillates continuously at a constant angular velocity.

On the oscillating driving shaft (not shown) is an arm to which is connected the link A, Fig. 11. This link, in turn, is pivoted to the sector or segment B. Sector B is free to oscillate on the stationary stud C and is provided with a sliding gear sector F which meshes with the driven gear G. Sector B is also provided with adjustable split stops D and E. These stops are used for regulating the angular movement and the dwell of the driven shaft H, to which gear G

is keyed. The stops are clamped in place by bolts on the dovetailed periphery of the sector *B*.

An important part of the mechanism is a friction stop or brake on shaft *H*, which is necessary to prevent over-run of this shaft at the point of reversal. The brake arrangement, however, being of simple and well known design, is not illustrated.

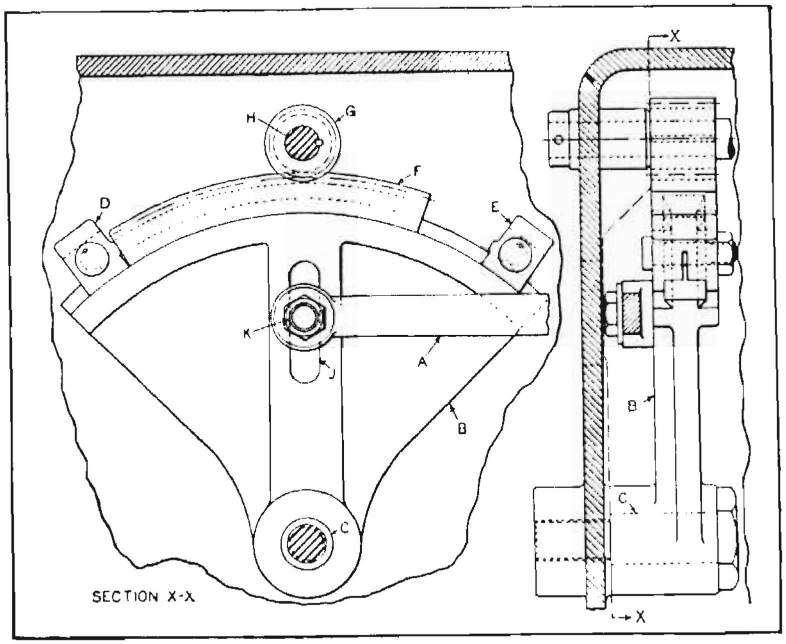


Fig. 11. Mechanism for Imparting Rotary Oscillating Movement to Shaft H, which Permits Varying Angular Movement and the Length of Dwell at Each Reversal

When the machine is in operation, the sector *B* is oscillated by link *A* at a constant angular velocity. In the position shown, the sector has moved toward the right to its central point, rotating gear *G* in a counter-clockwise direction. This motion continues until the sector has reached its farthest position at the right. The sector then reverses its movement and the rack *F* and gear *G* remain stationary until the end of the rack comes into contact with the stop *E*. At this time, continued movement of the sector toward the

left will carry the rack segment toward the left, rotating gear *G* in the opposite direction. The rotary movement of this gear continues until the sector comes to the end of its movement toward the left. Now as the sector once more moves toward the right, gear *G* dwells until stop *D* comes in contact with the sliding gear segment. Thus, gear *G* and shaft *H* are given a rotary reciprocating motion with a short dwell at each point of reversal. Owing to the different kinds of jobs adapted to this machine, a variation in the angular movement of shaft *H* is frequently required, the dwelling period remaining constant. This is obtained by the combined adjustment of the link and the stops. The extent, however, to which the angular movement can be increased is limited by the length of the sliding gear segment.

Suppose, for example, a greater angular movement of the shaft were required. In this case, the stud *K* would be adjusted to a lower point and stops *D* and *E* would be moved farther apart to avoid reducing the time periods of the dwells. In making these adjustments, a few trials are usually necessary in order to obtain the proper positions of stud *K* and the stops. Obviously, the same arrangement can be used to reduce or increase the dwell within certain limits, the angular movement of the driven gear remaining the same.

CHAPTER VIII

DRIVES OF THE CRANK TYPE FOR RECIPROCATING DRIVEN MEMBERS

The special designs of crank mechanisms described in this chapter are for transmitting motion to slides or other parts having a reciprocating action. These drives may be arranged to produce some special movement, such, for example, as arresting the motion of the slide momentarily during some part of the stroke or providing a quick return movement to reduce the idle period; or the design may be special in that provision is made for adjusting either the length or position of the stroke while the machine is operating.

Crank Motion that Causes Slide to Dwell at Center of Stroke.— The crank mechanism Fig. 1 is incorporated in a certain carton wrapping machine for changing the position of the carton as it passes through the machine. This mechanism imparts a reciprocating movement to the work-slide *A*, with a dwell at the center of its stroke in each direction. The slide is reciprocated in the stationary guide *D* through link *B* by the crank *E*. This crank is a free fit on shaft *F*, but rotates with *F* whenever spring-actuated plunger *H* engages one of the notches cut in the flange on the crank. A pin *J* in the plunger projects through and below arm *G*. When this pin engages the cam-block *K*, which is secured to the machine frame, the pin and the plunger are moved radially outward. This causes the plunger to disengage the notch and allows the crank and slide to dwell while the shaft continues to rotate.

The shaft and arm *G* are rotated in the direction indicated by the arrow. In the position shown, pin *J* has engaged cam-block *K* and has withdrawn the plunger from

the notch *L*. At this time, the slide is at the center of its stroke; and since the plunger is all that locks the crank *E* to the shaft, the shaft will turn in the bore of the crank hub and allow the crank and slide to dwell. A spring-actuated V-plunger is provided at *N* to hold the slide securely in the "dwell" position.

As the shaft and arm continue to rotate, pin *J* leaves cam-block *K* and allows the end of the plunger to ride on

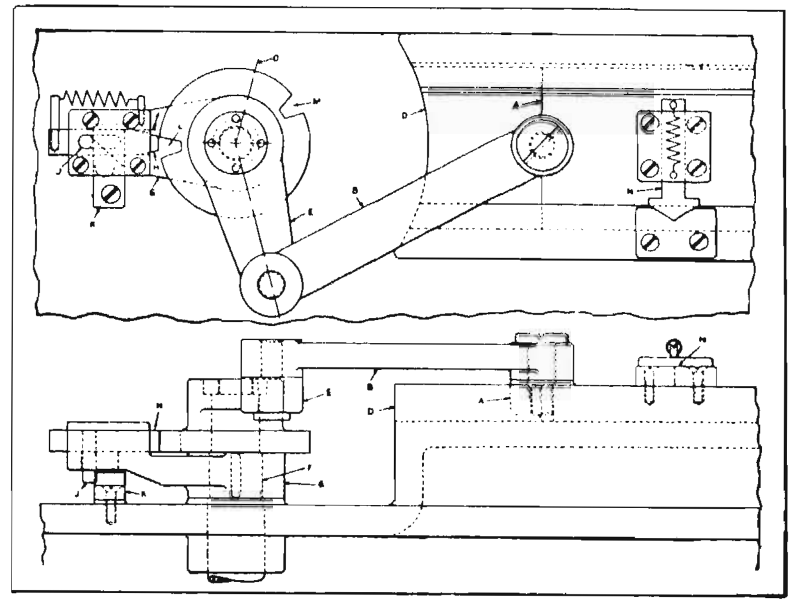


Fig. 1. Reciprocating Slide Mechanism with Dwell at Center of Stroke

the periphery of the crank flange until it drops into notch *M*. When this occurs, the crank is once more locked to the shaft so that continued rotation of the shaft will cause the slide to move toward the left and return to the position shown. At this time, the center line of the crank will coincide with center line *O* and cam-block *K* will have forced pin *J* outward, thus disengaging the plunger from its notch and allowing the crank and slide to dwell.

The withdrawn plunger then rides along the periphery of the crank flange until it drops into notch L and locks the arm to the crank again. The rotating arm now rotates the crank, causing the slide to move this time toward the right and then back to the position shown. At this point, the cam-block once more disengages plunger H. This completes one cycle of movements, which is repeated for each revolution of shaft F.

It will be noted that the angular movement of the crank is different for each half of the slide cycle, owing to the angular position of the connecting-rod B. This results in a variation of the time interval for each succeeding dwell and stroke. Fortunately, however, this variation is permissible. In other applications, where the dwell and stroke must have the same time interval, the well-known Scotch yoke crank movement could be used instead of the crank and connecting-rod shown. In this case, the notches would be located in the flange diametrically opposite each other.

Planetary Type of Crank Motion for Obtaining Dwell.— In attempting to bend a stranded copper cable into a U-shape by means of a kind of wing die, it was found, while experimenting with a punch press, that a distinct stop or dwell was required at a certain point in the bending stroke to permit the copper to set. If the dwell was omitted, a springing back of the metal occurred, resulting in variations in the form of the bent section. This dwell had to take place before the end of the stroke, because the latter part of the stroke was utilized to eject the formed piece.

A special machine was designed to actuate the slide from which the bending die receives its motion. This operating slide also requires a dwell at the top of the stroke to allow time for inserting unbent parts into the die, so that the machine can be operated continuously instead of using a single-stroke clutch and tripping device. The planetary type of crank motion used causes a crankpin to follow, during the dwelling periods, an arc having a radius equal approximately to the length of the connecting-rod, so that the

crank end swings without transmitting motion. Fig. 2 shows the general arrangement.

The housing A contains shaft C, which is driven through worm-gearing and carries a crank disk D. An internal gear H having 120 teeth is bolted to housing A and meshes with a 24-tooth planetary pinion J attached to the eccentric crankpin E; consequently, when crank disk D revolves,

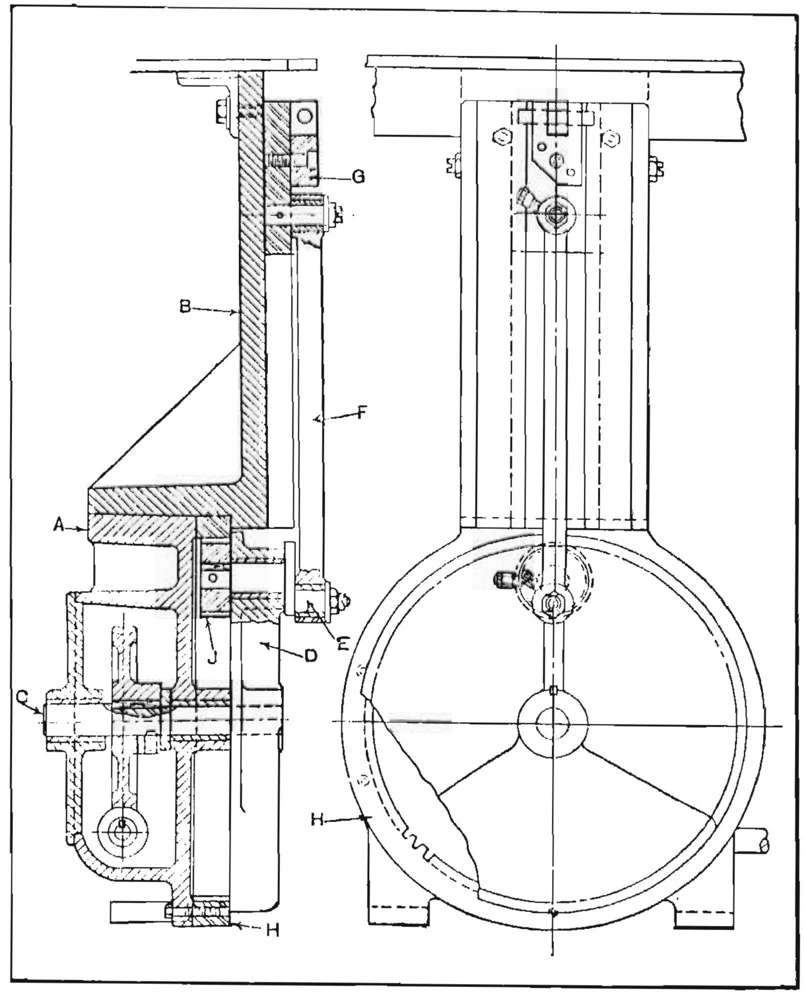


Fig. 2. Planetary Type of Crank Motion for Obtaining Dwell of Bending Die

pinion J and the crankpin revolve around their own axis and also around shaft C. These combined rotary movements modify the motion imparted to slide G and cause the axis of the pin E, to which connecting-rod F is attached,

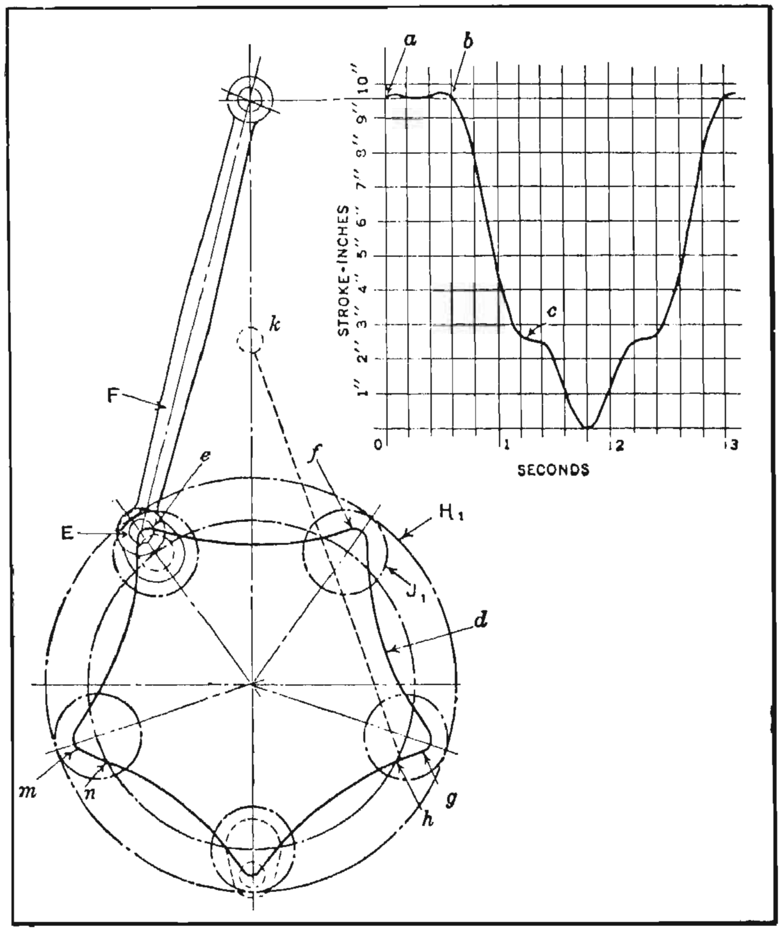


Fig. 3. Path Followed by Eccentric Crankpin, and Chart Showing Dwelling Periods

to follow the path indicated by the heavy line d, Fig. 3.

The curve in the upper right-hand corner of Fig. 3 illustrates how the action of the driven slide is changed during one complete cycle. The cycle begins at a point representing the top of the stroke of the slide. The vertical dimensions on the chart represent the stroke, in inches, and the horizontal dimensions, the time in seconds. One revolution is represented as 3 seconds, because the machine is designed to run about 20 revolutions per minute.

The relative positions of the internal gear H and the pinion J (Fig. 2) are indicated in Fig. 3 by dot-and-dash pitch circles H_1 and J_1. The radius of the eccentric crank E is considerably less than the pitch radius of the pinion, which causes the axis of the eccentric crankpin to describe a five-lobed curve d. The dwell of the driven slide at the top of the stroke occurs between points a and b on the chart and during about 6/10 of a second. This dwell is due to the fact that the length of the connecting-rod equals the radius of an arc which approximates that part of the crankpin path from e to f. As the lower end of the crankpin swings from e to f it transmits only a slight movement, and there would be none at all if this portion of curve d were a perfect arc with a radius equal to the connecting-rod center-to-center length. It is the dwell at this point that is utilized for removing the work and inserting unbent blanks.

The pause during the down stroke to allow the metal to set after bending occurs between points g and h where the curve d is practically tangent to the arc of the connecting-rod. This pause or dwell is represented on the chart at c, and at this time, the upper end of the connecting-rod is at k. An unnecessary dwell is made during the return stroke between points m and n, which correspond to g and h, but this slight delay in the upward movement does not affect the practical working of the mechanism. The entire device is located under a table about 2 feet square, which indicates that it is quite compact.

Oscillating Crank-and-Toggle Mechanism for Rapid Reciprocation of Slide.—In a metal ribbon crimping machine, four complete cycles of a slide are obtained from an oscillating arm as the latter passes through one cycle. This arrangement, which is shown in Fig. 4, has the advantage of simplicity of design and an unusually smooth action. The slide *A* that controls the crimping tools is mounted in guides *B*, cast integral with the machine frame *C*. Arm *D* is the driving member and is keyed to the shaft *E*, which oscillates at a constant angular velocity. This arm transmits

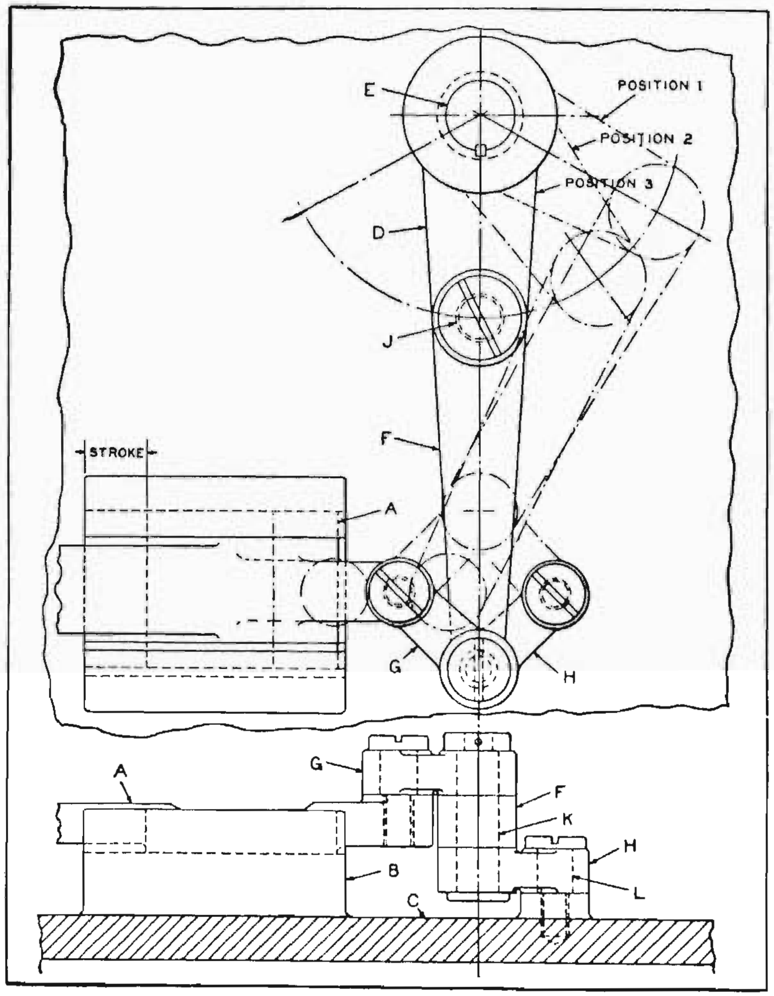


Fig. 4. Reciprocating Slide of a Metal Ribbon Crimping Machine, Operated by Oscillating Arm and Link Mechanism

the movement to the slide through a toggle arrangement consisting of links *F*, *G*, and *H*. Link *F* is pivoted at its upper end to arm *D* by pin *J*, and at its lower end to links *G* and *H* by pin *K*. The outer end of link *G* is pivoted to the slide, and the outer end of link *H* is pivoted to the shoulder screw *L* in the machine frame.

Three positions of the arm and links are shown. At Position 1, the toggle links *G*, *F*, and *H* are at their highest points; hence, slide *A* has been drawn to its farthest point at the right. As the arm swings downward to Position 2, these links assume a horizontal position, causing the slide to move to its farthest position at the left. The arm then continues its movement until it arrives at Position 3, where link *F* has forced the toggle links down to their lowest position, causing the slide to be carried back to the position indicated. Thus, during this one-quarter cycle of the arm, slide *A* has passed through a complete cycle. Consequently, as a repetition of these slide movements occurs during each quarter cycle of the arm, the slide will complete four cycles for each cycle of the arm *D*. An added advantage of this toggle arrangement is the unusually high working pressure that is delivered at the end of the stroke toward the left at the point where the pressure is needed most.

Auxiliary Crank that Assists Crankpin Past its Dead Center.—One method of overcoming the dead center condition in transmitting rotary movement to a shaft by means of a crank is shown in Fig. 5. Two of the outstanding advantages of this drive are its positive action and its low cost. The driven crank is actuated by a similar crank keyed to the driving shaft. By incorporating an auxiliary or "dummy" crank, the driven crankpin not only is helped past its dead center positions, but the angular velocity of the driving and driven shafts is held constant. In addition to this, the torque transmitted to the driven shaft is uniform at its various angular positions.

The shaft-to-pin center distance is the same for all three cranks. The driving crank is indicated at A, the driven crank at B, and the "dummy" crank at C. It is important to note that the connecting-rod is of solid construction and connects all three crankpins. With this arrangement, the position of all three cranks is the same at any part of the machine cycle.

In the full outline, the cranks and connecting-rod are approaching the dead center position. When they reach

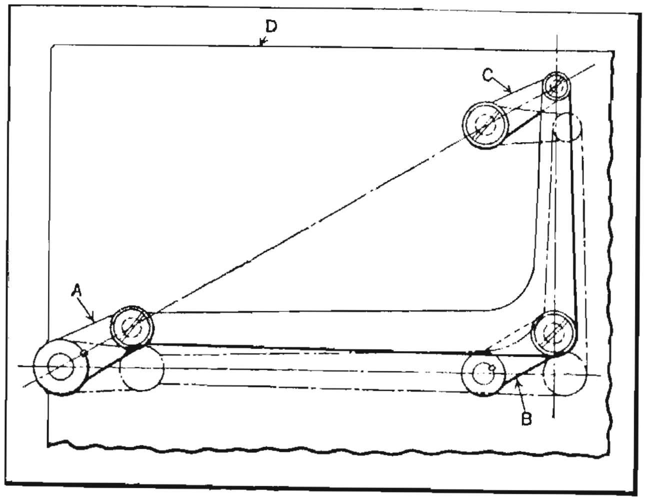


Fig. 5. Arrangement for Preventing Crankpin from being Stopped on Dead Center

this position, they will coincide with the dot-and-dash outline. Here it is obvious that the crankpin in the "dummy" crank has passed its dead center and can continue its movement unrestricted. Now, owing to the rotary action of crank A, crank C will swing downward, and as a result, crank B will be forced past its dead center. The same action occurs in reverse order when crank C is on its dead center relative to crank A. That is, crank B, having passed its dead center, will serve to force crank C past its dead center position. Incidentally, the location of crank C can be varied to suit existing conditions, although it should not

be located too close to a straight line passing through the driving and driven shafts.

Auxiliary Crank for Quick Return Movement.— The cam-operated turret-feed mechanism of an automatic screw machine is shown in Fig. 6. The advance feed is obtained by the cam A operating through the segment lever B to feed the turret-slide C. The return motion is accelerated by the revolution of the crank D which brings the turret

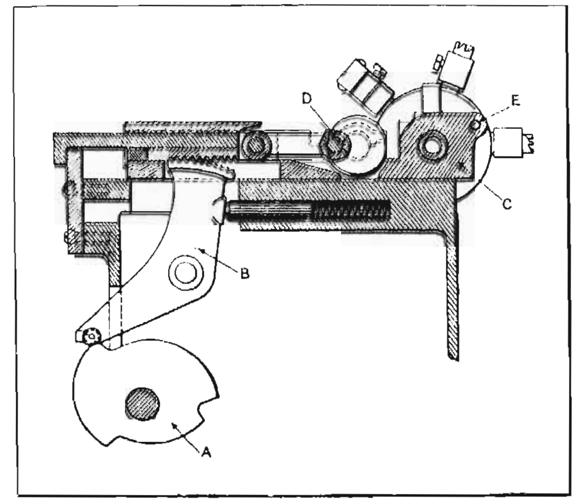


Fig. 6. Turret Slide is Withdrawn Quickly when Crank D Rotates

back quickly, a distance equal to the throw of the crank.

In the operation of a machine for high-speed work, it becomes important, both in securing the desired speed and in avoiding objectionable shocks, to move and reverse the lightest parts. For this reason, machines having turrets of the "revolver" or "barrel" type, in which each spindle can be fed independently, are especially adapted to high-speed work. In such machines, each tool carrier is connected successively with a reciprocating feed slide, and only the feed slide with one of the tool carriers connected with it requires to be reciprocated for the feed and return movements. In order to "speed up" this type of machine still further, the use of an auxiliary slide has been resorted to. This auxiliary slide alone is moved during that part of the quick-return movement required to retract each tool, and even this slide is disconnected for the remainder of the return movement, thus avoiding the shock which would result from rapid movement of the slide.

Fig. 7 shows an application of such an auxiliary slide with its disconnecting means. The turret *A* carries a series of tool spindles which are successively indexed to come into operative positions and be engaged by the block *B*. A main

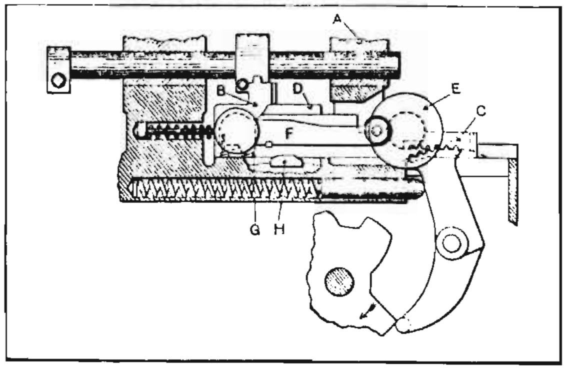


Fig. 7. Another Application of a Crank Motion for Obtaining a Quick Return

slide *C*, on which is an auxiliary slide *D*, is mounted on the bed of the machine. A crank *E* which is also on the main slide is connected to the auxiliary slide by the two-part connecting-rod *F*, one end of which is connected with the main slide and the other with the auxiliary slide. A latch at *G* holds these two parts together except during the quick-return motion which is obtained by revolving the crank disk *E*; then the latch is disengaged by passing over the cam *H*, which thus breaks the connection with the auxiliary or supplemental slide for the remainder of the crank throw and gives the quick-return movement and the quick-advance movement up to the point of cutting.

Rapid Return Movement Obtained by Roller Clutch and Crank Arrangement.—In designing machinery, it is frequently possible to make use of a roller friction clutch for reducing the time consumed during the idle part of the production cycle. This application is exemplified by the simple crank motion of the Scotch yoke type shown in Fig. 8. It is employed for actuating a slow-moving slide in a machine for forming plastic materials.

The slide indicated at *A* is reciprocated vertically. The crank is composed of the core *B*, integral with drive shaft *C*; the member *D*, which is bored to provide a running fit for the core; and the rollers *E*. The roller *F* on the stud that is secured in the projection on member *D* serves as the crankpin and engages a slot extending across the slide. As the crank rotates in the direction of the arrow, the slide is given its upward or working stroke, the movement being comparatively slow. During this stroke, the weight of the slide causes the rollers *E* to grip both the core and the member *D* tightly, so that both members rotate positively together. When the roll *F* has passed the center line *G*, the weight of the slide, which has caused the rolls to wedge tightly on the upward stroke, releases the gripping pressure of the rolls between the core and member *D* and allows the latter to rotate one-half revolution, returning the slide to its lowest position at a relatively higher velocity.

The downward stroke is the idle one, and its velocity in this particular case is unimportant in so far as the timing of the slide movements is concerned. This condition made it possible to use this crank. At the bottom of the stroke, the rolls *E* once more pick up the motion and move the slide upward at the slow speed required for the operation. It is estimated that with this design, an approximate gain of 30 per cent in production time is obtained over the time that would be required if a crank of the solid type were used.

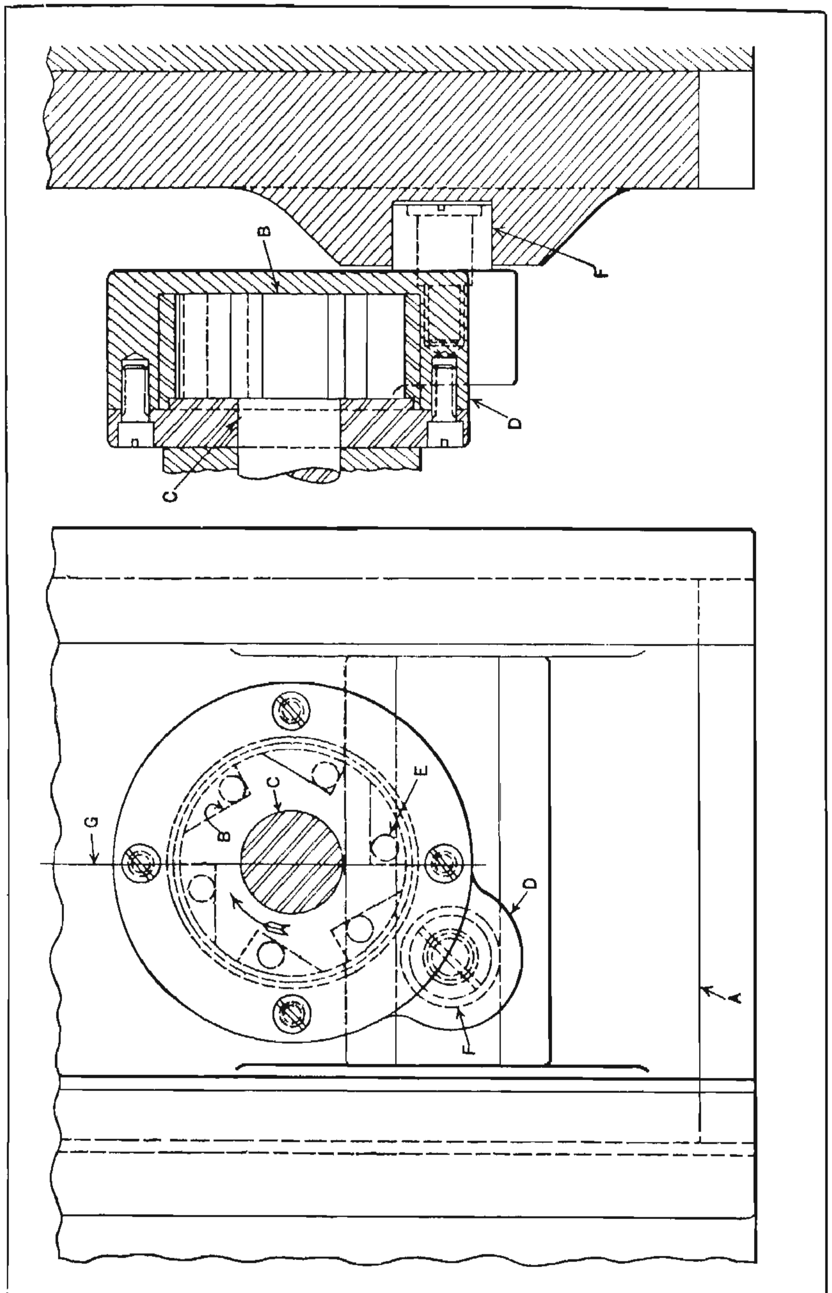


Fig. 8. Crank-operated Slide with Quick-return Movement

Quick-Return Crank Motion with Adjustment for Varying Velocity of Stroke.—Slotting machines, as a rule, are provided with some means of varying the cutting speed to suit the different materials to be machined. However, in reducing the cutting speed, the production is also reduced a corresponding amount, because the velocity of the entire cycle of the machine is slowed up. This objection was overcome in the case of one slotting machine by using a crank

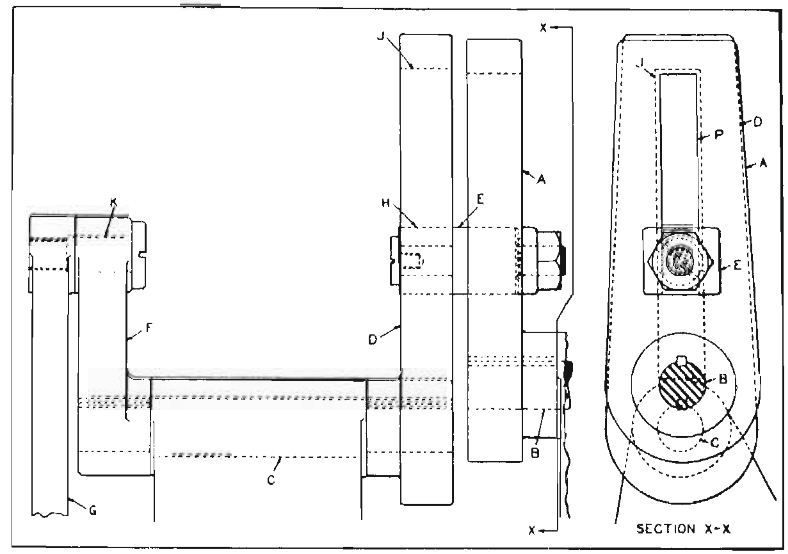


Fig. 9. Quick-return Crank Motion Mechanism with Means for Varying Velocity of Stroke

motion for actuating the slotting ram, the principle of this motion being shown in Figs. 9 and 10. With this arrangement, the velocity of the working stroke can be varied within certain limits without changing the time taken for the ram to pass through its cycle. Therefore, varying the cutting speed of the tool in this way does not change the rate of production, because, as explained later, the loss in velocity during the working stroke is compensated for by increasing the velocity of the return stroke.

The crank mechanism consists chiefly of the arm A,

Fig. 9, keyed to driving shaft *B;* the jack-shaft *C*, keyed to the arm *D* in which slides a cross-head *E* fastened to and adjustable along arm *A;* and crank *F*, also keyed to jack-shaft *C* and connected to the slotting ram by the connecting-rod *G*. It will be noted that jack-shaft *C* is offset from the driving shaft *B*. Consequently, as arm *A* rotates arm

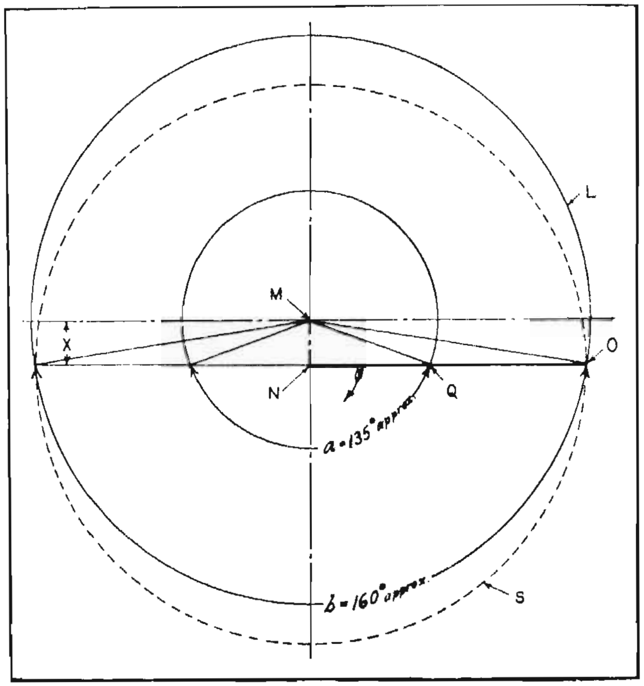


Fig. 10. Diagram Indicating Operating Principle of Mechanism Shown in Fig. 9

D and crank *F*, the pivot block *H* on cross-head *E* slides back and forth in the slot *J*. Arm *D* and crank *F*, therefore, will be given an irregular rotary movement; that is, the crankpin *K* will travel faster in its circular path *S*, Fig. 10, when below the horizontal center line of shaft *C* than when above this center line. This action will be more clearly understood by referring to Fig. 10. Here let circle *L* represent the path of cross-head *E*, point *M* indicating

the center of driving shaft *B*. Let *N* indicate the center of shaft *C*, and let the heavy line represent the arm *D* with the cross-head at *O*.

Now, if the arm *D* is horizontal, as indicated by the heavy line, the cross-head, with arm *A* and shaft *B*, will rotate in the direction of the arrow only 160 degrees, in order to rotate arm *D* one-half revolution. Thus, during this movement, which corresponds with the return stroke of the ram, arm *D* and crank *F* rotate faster than arm *A*. In completing their revolution, however, the cross-head and arm *A* rotate 200 degrees to turn arm *D* and the crank the remaining half revolution. Hence, during the latter movement, which corresponds with the working stroke of the ram, crank *F* rotates more slowly than arm *A*. Thus, a slow working stroke and a rapid return stroke are obtained.

If it is required to reduce the velocity of the working stroke, the cross-head is adjusted inward in slot *P*, Fig. 9, to a new position, say, to *Q*, Fig. 10. In this case a 135-degree movement of arm *A* is required to rotate crank *F* through its return stroke, and a 225-degree movement to rotate the crank through its working stroke. Thus, the velocity of crank *F* is increased during the return stroke and reduced during its working stroke. Crank *F* and arm *A*, however, complete their cycle in the same time, so that the reduction in velocity of the working stroke does not affect the production rate of the machine. Incidentally, a greater range in the velocity variation of the crank can be obtained by increasing the offset *X* between shafts *B* and *C*. This change will, of course, affect the length of the slots in the arms.

Adjusting Operating Position of Reciprocating Slide without Stopping Machine.—Occasionally it is necessary to provide means for adjusting the operating position or point of reversal of a slide having a fixed length of stroke without stopping the motion of the slide. A parallel to this requirement is found in a vertical shaping machine, in

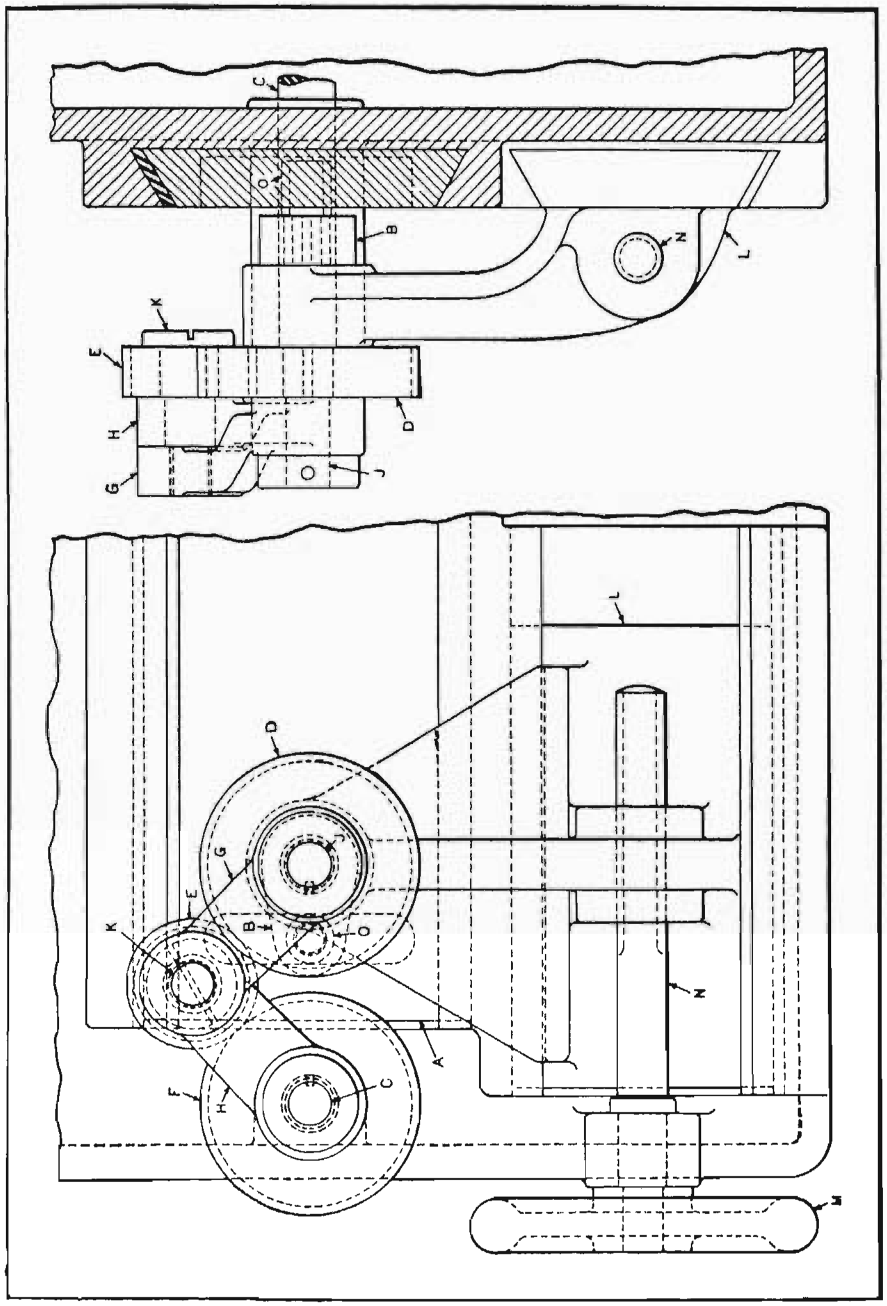


Fig. 11. Reciprocating Slide with Mechanism for Adjusting Operating Position

which the reversal point of the tool-slide is varied manually. A mechanism for obtaining this variation is shown in Fig. 11, the tool-slide being indicated at *A*. This slide is driven by the crank *B* keyed to shaft *J*. Shaft *J* is driven by shaft *C* through gears *D*, *E*, and *F*. Roll *O*, mounted on a stud in crank *B*, engages a groove in the tool-slide and operates on the principle of the Scotch yoke.

The center distances between gears *D* and *E* and between gears *E* and *F* are maintained by the links *G* and *H*, respectively. These links are a free fit on the gear-shafts *J* and *C*. Gear *E* and link *H* are also a free fit on screw *K*. Shaft *J* turns freely in a bracket cast integral with the adjusting slide *L*, and this slide is actuated by the handwheel *M* on the feed-screw *N*. Screw *N* engages a nut cast on slide *L*. Slide *A* is shown in its extreme left-hand position. Assume that both the left-hand point of reversal and the right-hand point of reversal are required to occur farther toward the right. To effect this change, the operator merely turns handwheel *M* the required amount or until slide *L* has carried shaft *J* a corresponding distance toward the right. In doing this, the links tend to straighten out, yet the gears remain in mesh and continue the rotation of the crank. The range of variation for changing the point of reversal is controlled by the diameters of the gears. If a larger idler gear *E* is used, the slide will have a greater range of adjustment.

Adjusting Crank Throw of Wire-Forming Machine while Machine is Running.—In the operation of a wire-forming machine, difficulty was experienced in holding the parts to a uniform shape, due to variations in the hardness of the low grade of wire used. These variations in hardness necessitated frequent adjustment of the forming dies to prevent excessive variations in the depths of the formed portions. As stopping of the machine for this purpose seriously affected production, it was decided to provide means for making the necessary adjustments while the ma-

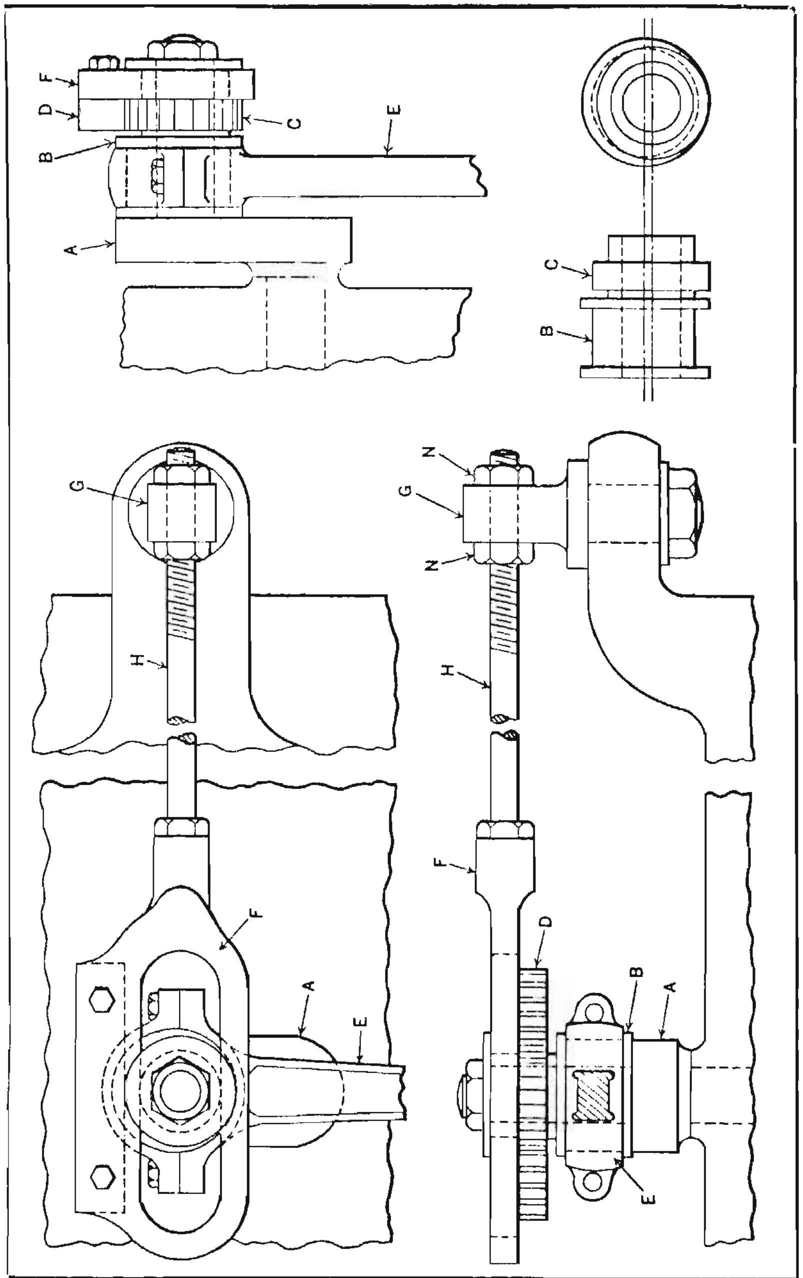


Fig. 12. Mechanism that Provides Means for Shortening or Lengthening Stroke of Connecting-rod E while Machine is in Operation

chine was in operation. This was satisfactorily accomplished by applying the stroke-changing mechanism shown in Fig. 12. With this mechanism, the length of stroke of connecting-rod *E*, which operates one of the dies, can be changed while the machine is in operation, simply by adjusting the nuts *N* on rod *H*.

The length of stroke of rod *E* is varied by means of the eccentric bushing on the pin of crank *A*. Crank *A* is of the conventional open-end type, except that the crankpin is longer than would ordinarily be required. The bushing *B* on the crankpin is turned eccentric with the bore to fit the bearing in the connecting-rod *E*. The hub of bushing *B*, which is machined concentric with the bore, carries the gear *C*, as shown in the view in the upper right-hand corner of the illustration. The yoke *F* is carried on the hub of bushing *B* outside of gear *C*. This yoke carries rack *D* which meshes with gear *C*. Rod *H* is fastened to yoke *F* and is threaded on its outer end where it passes through stud *G*. Stud *G* is located in a fixed position, but is free to turn or swing. All three assembly views show the crank *A* in its upper position. The bushing *B* is shown adjusted for the maximum length of stroke.

As the crank *A* rotates in either direction, the crankpin carrying bushing *B* moves in the slot in yoke *F*. This produces a rotating movement of bushing *B* on the crankpin as a result of the action of rack *D* and gear *C*. The number of teeth in gear *C* is such that a half turn of crank *A* produces a half turn of gear *C* and bushing *B*. Thus the throw of eccentric bushing *B* is reversed in relation to the crankshaft as the crank *A* reverses its position. This causes the throw of eccentric *B* to be added to the throw of crank *A*, thus increasing the stroke of connecting-rod *E*. This condition exists only in the opposite positions of the crank *A*, as the bushing *B* is constantly changing its position throughout the cycle. As the nuts *N* on rod *H* are changed, the rela-

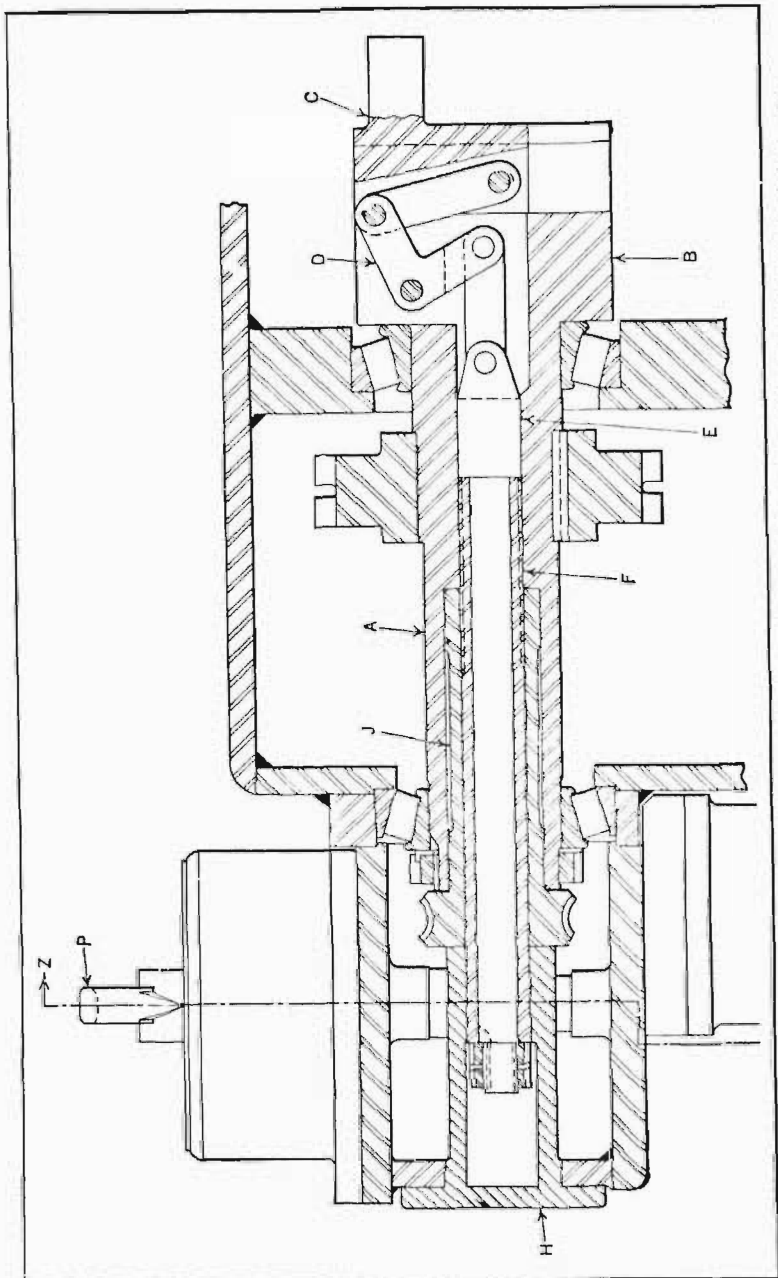


Fig. 13. Motor-driven Mechanism Combined with an Electric Switch for Changing the Radial Position of a Crankpin to Vary the Crank Throw

tive position of the bushing *B* is changed, causing a change in the throw of crank *A*.

Electric Control that Varies Throw of Crank While Machine is Running.—The mechanism illustrated in Figs. 13 and 14 provides a rapid adjustment of the throw of a crank while the machine is in operation. The crankshaft indicated at *A* serves to impart a reciprocating motion to another member of the machine, and any throw of the crank between zero and the maximum is instantly available.

The crankshaft is mounted on tapered roller bearings. Crankhead *B* is integral with the shaft and carries a sliding block of which the crankpin *C* is an integral part. This block is connected by means of link *D* with the draw-bar *E*, which is free to slide axially in shaft *A*. Sleeve *F* is threaded at its right-hand end and has rack teeth on it that mesh with gear *G* (Fig. 14).

Sleeve *F* is keyed to stationary cap *H* to prevent it from turning, but is free to slide axially in this cap. The worm-wheel nut *J* is a running fit in shaft *A*, and is threaded to fit sleeve *F*. A reversing motor rotates the worm-wheel nut through worm *K*, and thus moves the sleeve *F*, with draw-bar *E*, axially, so that, by means of link *D*, the radial position of the crankpin is changed.

Mechanical and Electrical Mechanism for Regulating the Crank Throw.—The apparatus for controlling the radial movement of the crankpin is shown in Fig. 14. It is contained in a separate housing, and consists of a special electric switch designed to control the reversing motor. This switch has a disk *L*, which is connected by a bushing to gear *G* and is provided with two semicircular contact segments *M*. The segments are insulated from disk *L*. Member *N* is connected to a handle and its pointer *P*.

The links *Q*, hinged to member *N*, are each equipped with a contact blade *R*. The links, with their blades, are held against the contact segments by a coil spring, as indicated. This spring also serves to hold the pointer against the

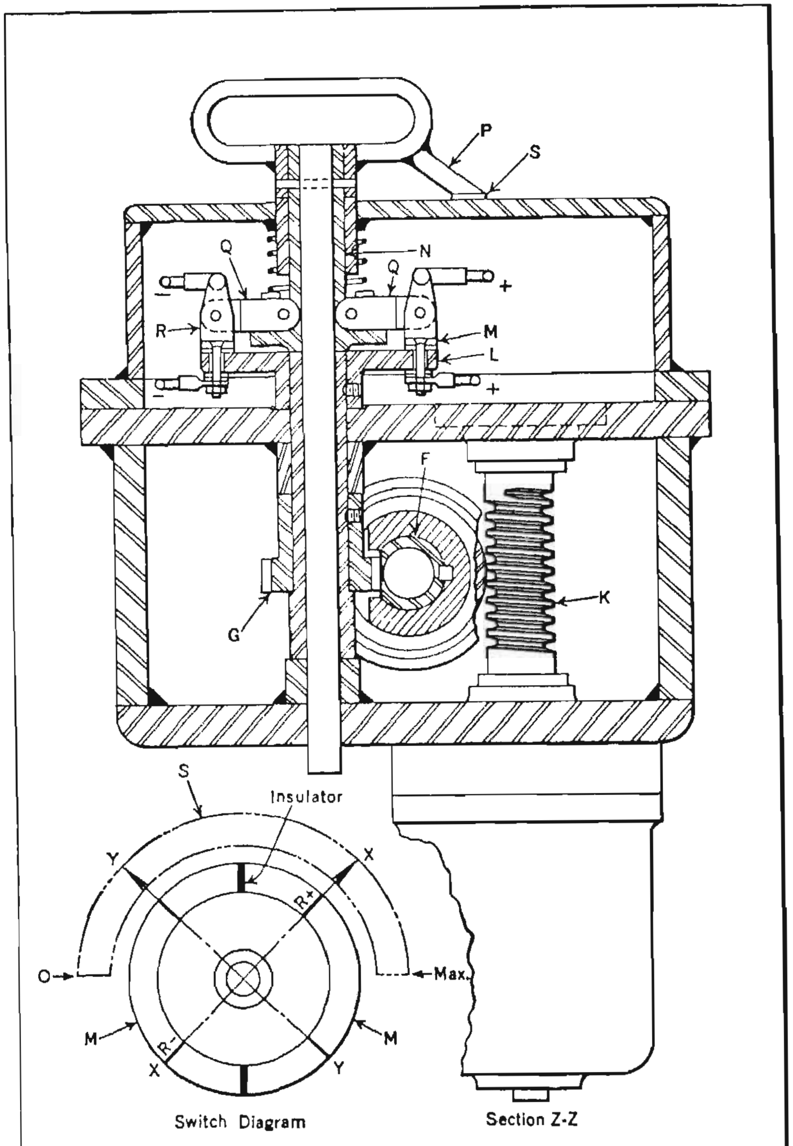


Fig. 14. Section of Mechanism in Fig. 13, Showing the Operation of the Electric Control Switch

graduated dial S and prevents its shifting after an adjustment of the crankpin has been made. To further prevent shifting, the finger is formed like a knife-edge and rests in radial grooves in the dial, which also serve as the graduations.

The operation of the switch will be understood from the switch diagram. Point $R+$ represents the positive contact blade and $R-$ the negative blade. The contacting segments M are connected with the motor and are separated from each other by insulators. When the $R+$ and $R-$ blades are on the insulators, the motor is idle, as the circuit is open, and the throw of the crank is indicated by the position of the finger on the dial S. If the finger is moved toward the right, so that the blades $R+$ and $R-$ coincide with line X-X, the blade $R+$ is in contact with the right-hand segment and $R-$ with the left-hand segment, and the electric circuit is closed. Consequently, the motor will start and shift the crankpin, as already explained. In the meantime, through the axial movement of sleeve F and the resulting rotation of gear G, the disk L turns clockwise until the contact blades engage the insulators. At this point the circuit is broken and the motor stops, leaving the crankpin in a radial position corresponding to the position of the pointer on the dial S.

If the pointer is moved toward the left, say on line Y-Y, the motor will run in the opposite direction and move the crankpin back toward its former position. In this way, to either shorten or lengthen the throw of the crank, the operator merely swings the pointer handle so that the pointer engages the graduation on the dial corresponding to the required throw. This adjustment, besides being rapid, is made with a minimum amount of effort, as the motor does the actual work of shifting the crankpin. In designing the switch, great care should be taken to thoroughly insulate the electrical contacts.

CHAPTER IX

RECIPROCATING MOTIONS DERIVED FROM CAMS, GEARS, LEVERS AND SPECIAL MECHANISMS

In designing the driving mechanisms for some parts having a reciprocating motion, cams, gears or levers are substituted for a transmission of the rotating crank type. Examples of these different forms of reciprocating drives will be described. As with the crank type of drive, the object of using cams or combinations of levers may be to vary the stroke in some way or the object may be to obtain a mechanical movement essential to meet a particular operating requirement.

Double Lever Mechanism to Provide Strokes of Unequal Length Synchronized During Part of Stroke.— The mechanism shown in Fig. 1 fulfills an unusual requirement in a simple manner. Two slides of a wire-forming machine were required to operate with different lengths of travel, the slide having the longer travel being arranged to operate in synchronism with the other during a portion of its stroke. Adjustability, both as to the length of travel and the period of synchronization, was also required on the slide with the longer travel.

Referring to the illustration, bearing *H* carries the shaft *A*, which is given an oscillating motion by a cam-operated lever (not shown). The motion of shaft *A* is transmitted to lever *B*, which is keyed to it. Rod *E* transmits the motion of lever *B* to one slide, and rod *D* transmits the motion of lever *C* to the other slide. Lever *C* oscillates on stud *K*, carried on lever *B*, and has gear teeth cut on the end.

The gear teeth on lever *C* mesh with teeth cut in disk *L*, which is carried free on the hub of lever *B*. Disk *L* carries

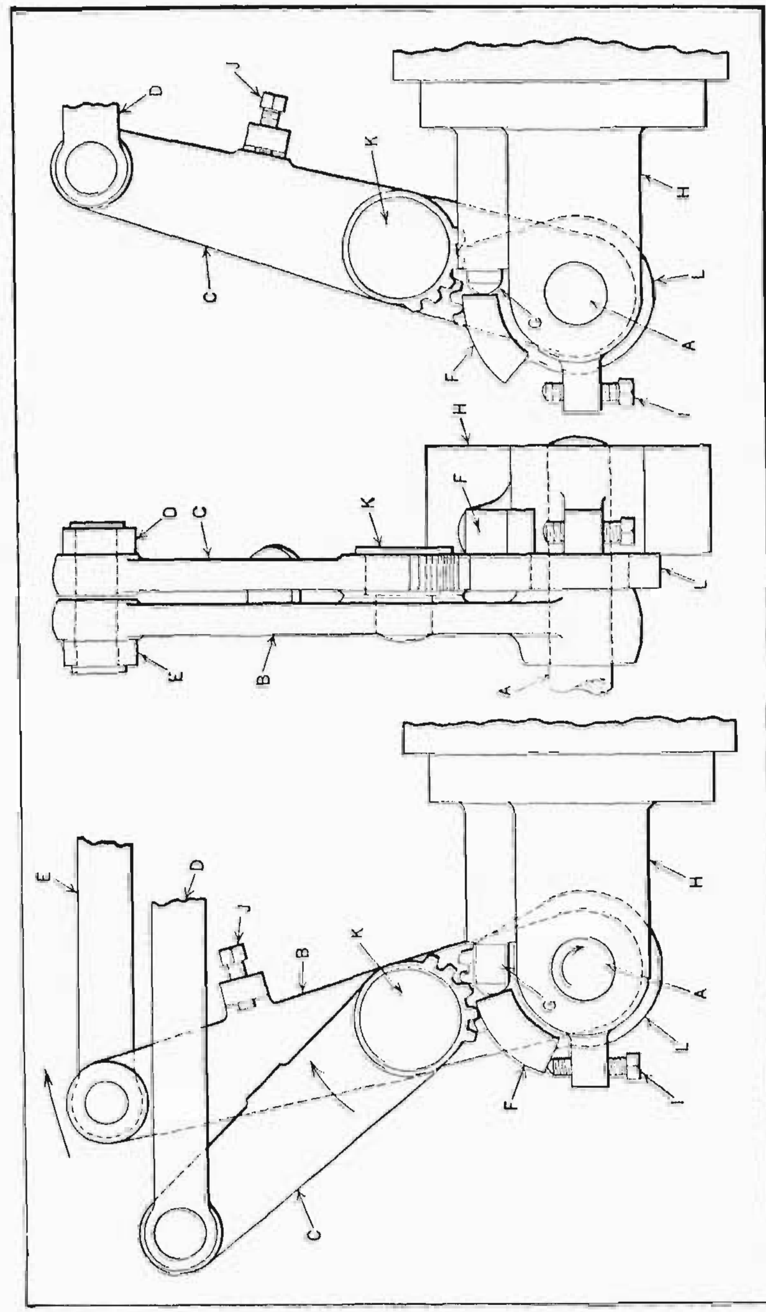


Fig. 1. Mechanism for Operating Wire-forming Machine Slides having Unequal Strokes and Synchronized Movements During Part of Stroke only

a pad F, which is in constant contact with plunger G in bearing H. Plunger G is backed up by a stiff spring. The action of plunger G against pad F tends to hold the pad down against the screw I.

As the shaft A rotates in the direction indicated by the arrow in the view to the left, lever B is carried in the same direction, but disk L is restrained from movement by the pressure of plunger G against pad F. This causes the gears to operate, so that the lever C is swung on stud K in the direction of the arrow. Rod D is thus given the combined movement of lever B and lever C, which continues until lever C makes contact with the stop-screw J on lever B. At this point, lever C is restrained from further rotation on stud K, and continued movement of lever B causes disk L to be carried around with it, due to the locking action that takes place between the gears and screw J.

As soon as disk L turns with lever B, levers B and C revolve around a common axis—the center of shaft A—and they move in synchronism from that point on. The view to the right shows the levers B and C in their extreme forward position, while an end view of the mechanism in the same position is shown by the central illustration. On the return stroke, synchronism is maintained until pad F again makes contact with screw I, at which time the movement of lever C is increased by the action of the gears. Screw J controls the period of synchronization, while screw I controls the travel of lever C.

Slide which Always Dwells During Initial Movement of Parallel Slide.—The mechanism shown in Fig. 2 provides a dwell or delay in the movement of slide B while slide A enters upon the first portion of its cycle. On the return stroke, slide B dwells in the same manner while slide A begins its movement back to its original position.

This requires a delay arrangement that will operate at each end of the cycle, so that the first slide will remain stationary in each position for a given length of time. The

dwells could, of course, be obtained by means of cams. However, the mechanism shown is simple and more compact than a cam arrangement. In this mechanism, disks N and D, with their contacting pins, are arranged similarly to the tumblers employed on a combination lock. Referring to the illustration, the slide A moves a given distance at the start before slide B moves in the same direction. At

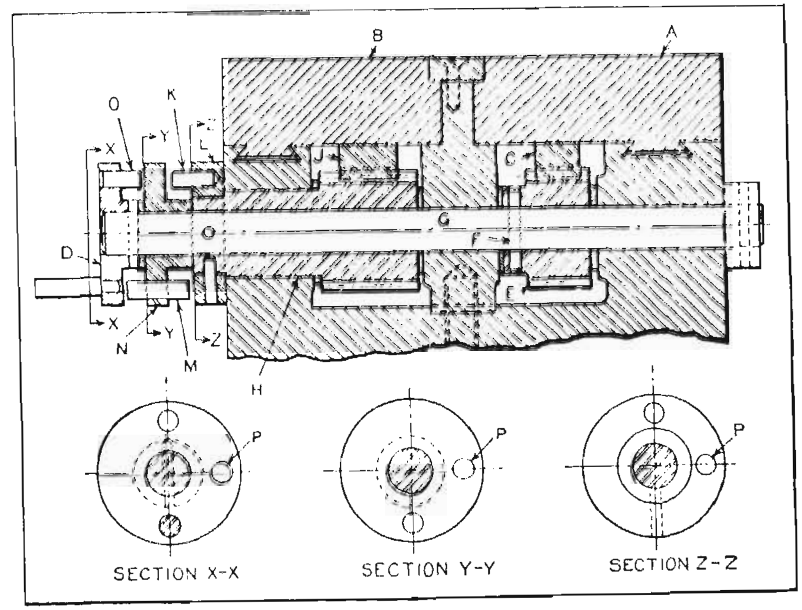


Fig. 2. Mechanism that Enables Slides A and B to be Moved in Either Direction by Turning Handwheel D, Slide B Always Dwelling for a Certain Period During the Initial Movement of Slide A

the end of the stroke, slide A moves in the reverse direction the same distance as at the start before slide B begins its return movement.

These motions are obtained in the following manner: Slide A, through rack C, is connected directly to the disk or handwheel D by gear E, pin F, and shaft G. Shaft G, however, is allowed to rotate freely in the combination gear and bushing H. Gear-bushing H is connected to handwheel D through pin K in collar L which comes into contact with pin M in the free-running collar N. The opposite end of pin M comes into contact with pin O in handwheel D. Thus pin O makes one revolution minus an amount equal to the thickness of the pin before it comes in contact with pin M. The opposite end of pin M can then make one revolution less the thickness of the pin before coming in contact with pin K which moves gear-bushing H, causing slide B to move. On the return stroke, the reverse action takes place.

This means that two revolutions of disk D, less the thickness of two pins, can be obtained before slide B follows the movement of slide A. Less than this amount of movement

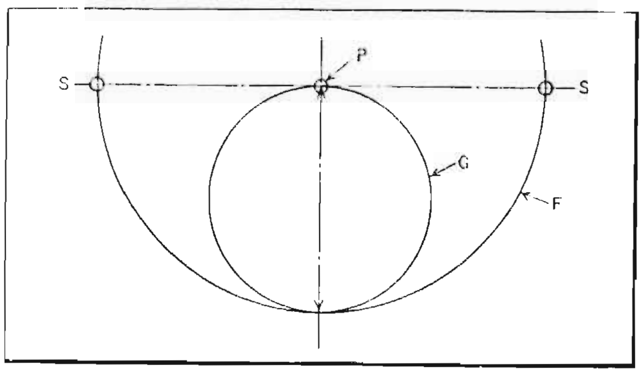


Fig. 3. Diagram Illustrating Application of Hypocycloid to Reciprocating Mechanism

can be obtained by placing two pins in each of the three disks at such angular positions as to give the required movements. Thus, in the case shown, the total movement of A in advance of B is 1 1 4 revolutions minus the thickness of three pins if pins P are inserted. This movement lends itself very readily to operations that require the withdrawal of a certain tool from the work before the entire carriage is withdrawn.

Mechanism for Converting Rotary into Reciprocating Motion by Application of Hypocycloid Principle.—The principle of the hypocycloid, as illustrated in Fig. 3, has been applied very effectively in the mechanism shown

in Figs. 4 to 6. This mechanism is designed to convert rotary motion into reciprocating motion. The hypocycloid SS, Fig. 3, is generated by the point P in the generating circle G as it rolls on the inside of the circle F. The hypocycloid thus generated by point P is a straight line when the

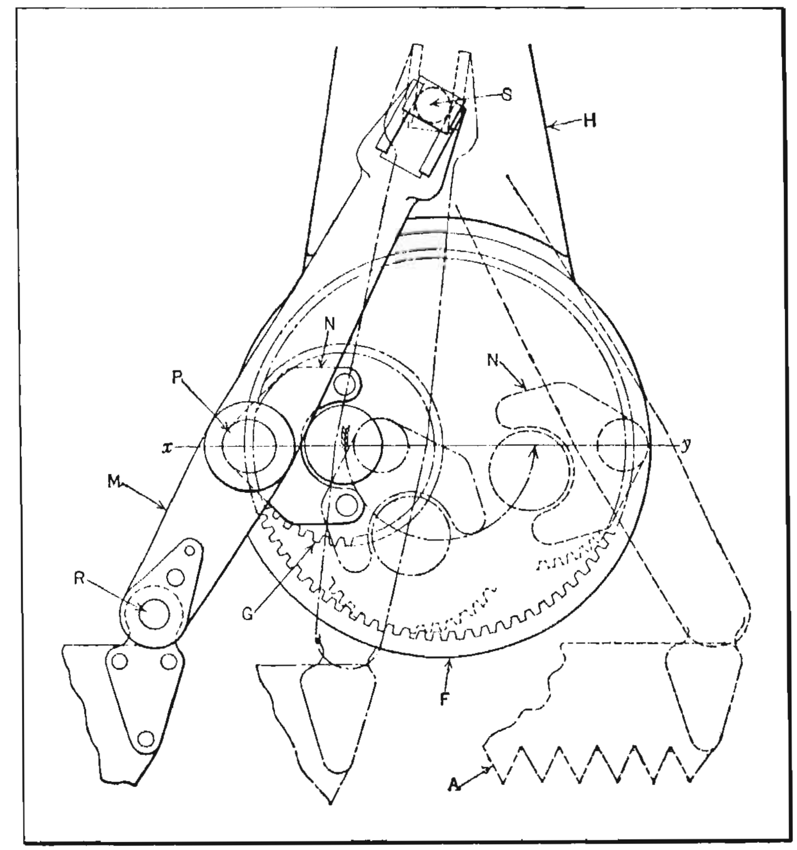


Fig. 4. Saw-reciprocating Mechanism Based on Hypocycloid Principle

diameter of the generating circle G equals the radius of the circle F. A mechanism designed on this principle will give a long stroke with a minimum number of small strong parts arranged in the most compact form.

In the principal application to be described, the circle F

becomes the pitch circle of a fixed internal gear, and the generating circle G becomes the pitch circle of a pinion that rolls around on the inside of the internal gear. The point P, which is at all times located at the exact intersection of

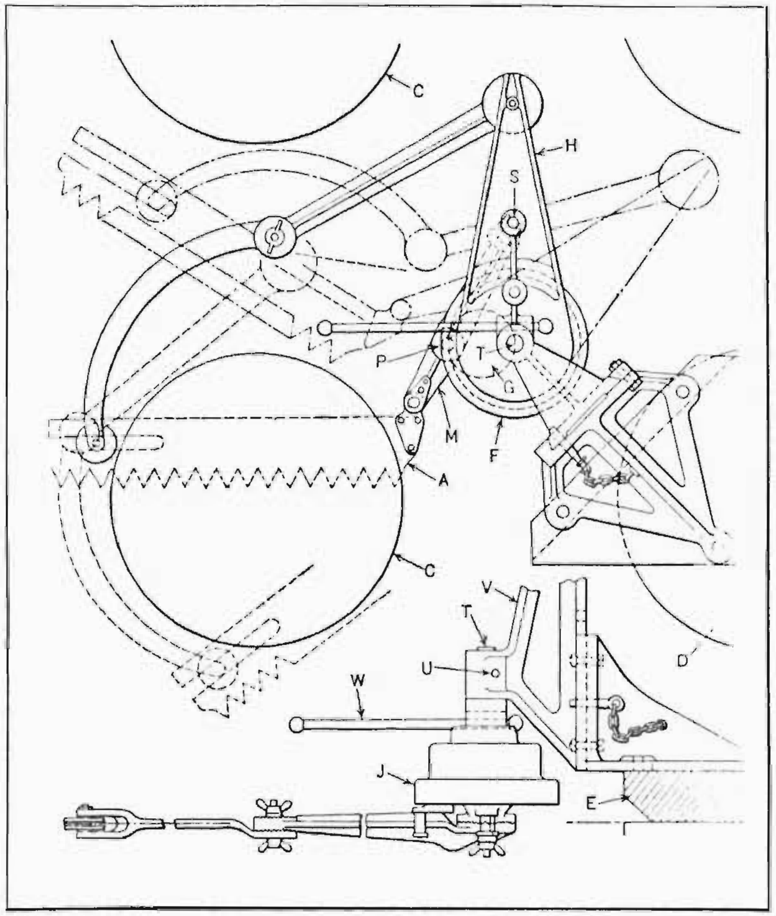


Fig. 5. Assembly of Saw Operated by Mechanism Shown in Fig. 4. Dotted Lines Show Starting and Finishing Positions of Saw in Cutting off Closely Spaced Piles

the pitch line G with the center line SS of the internal gear, becomes the stroke pin. As the pinion rolls, this pin moves back and forth on a straight line from S to S, which is the equivalent of the pitch diameter of the internal gear.

Hypocycloid Principle Applied to Saw-Reciprocating Mechanism.— The arrangement shown in Figs. 4 to 6 forms the operating mechanism of a saw for sawing off piling. The saw A, Fig. 4, has a stroke of 12 inches, and the internal gear F has a pitch diameter of 8 inches. The complete machine is made of Duralumin, and, without the motor, weighs 43 pounds.

The specific problem was to design and build a one-man portable machine for sawing off piles C (Fig. 5) at low tide. These piles were 18 inches in diameter and were spaced 30 inches apart, center to center. Plan and elevation views show the assembled machine attached to a 2- by 12- by 48-inch timber E, supported on pile D, which was hand-sawed. The saw and the guide arms are shown in three positions by dotted lines to indicate how the reciprocating members clear the adjacent piles. Fig. 4 shows the reciprocating mechanism to a somewhat larger scale. Sectional elevation and inverted plan views of the power-driven parts are shown in Fig. 6.

The direction of the stroke is determined and fixed when assembling stroke pin P and the pinion G in the internal fixed gear F. In Figs. 4 and 6, the line x-y is at right angles to the extension arm H of the frame J. The internal gear F is made as a separate piece only for convenience in cutting the teeth and to provide a bottom bearing for the driving pinion K, Fig. 6. The outside end L of pinion K was squared and connected to an air motor which runs at a speed of 800 revolutions per minute. This gives the saw about 72 strokes per minute.

The stroke pin P and its bracket N, riveted to the rolling pinion G, are shown in three positions in Fig. 4 to illustrate how the pin P follows line x-y, carrying with it the forked link M which has pin R pivoted on its short end. The long end of link M is forked around the squared fulcrum pin S, Fig. 6, which swivels in the hub Q of the extension arm H. The shoulder-stud T, Fig. 5, is supported in frame V and fixed in position by pin U. A vise handle W in frame J permits the operator to swing the whole assembly on stud T to adjust and feed the saw.

Referring to Fig. 6, the motor-driven gear X is a running

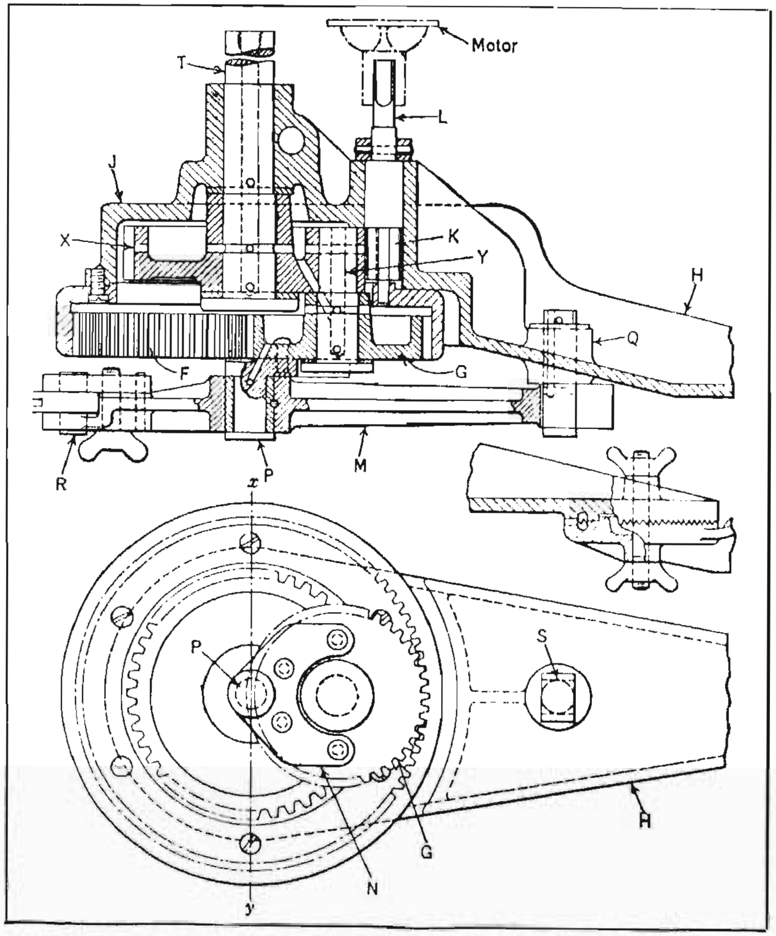


Fig. 6. Cross-section and Plan Views of Saw-reciprocating Mechanism

fit on stud T, the head of which serves as a support for the gear. The crankpin Y, fixed in gear X, has pinion G mounted on its head. Suitable thrust washers are provided for both gears X and G. The stroke-pin bracket N is riveted

to pinion G, with the center line of the pin P located on the pinion pitch line. The pin P turns freely in a bushing made in halves to facilitate assembling. The halves of this bearing are pinned securely to the forked link M. Holes and grooves for providing ample lubrication from one grease cup screwed into the top of stud T are shown.

Uniform Reciprocating Motion.—A uniform reciprocating motion often is required in machine design, and the

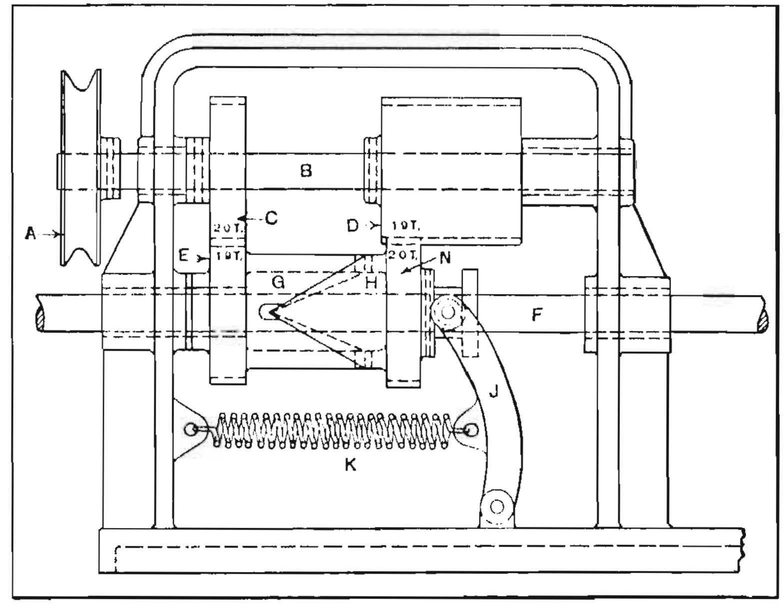


Fig. 7. Mechanism which Imparts an Even Reciprocating Motion to a Rotating Shaft

mechanism to be described produces such a movement. A belt drive to pulley A, Fig. 7, rotates shaft B, which drives gears C and D. Gear C meshes with and drives gear E. Cam G is integral with gear E and is opposed to the mating cam H, which is integral with gear N. Cam H and gear N are attached to shaft F, which rotates and also receives a reciprocating motion. Cam G and gear E are free to revolve around this shaft.

Gear C has twenty teeth, and gear E nineteen teeth, whereas gear *D*, which drives gear *N*, has nineteen teeth, and gear *N* has twenty teeth; consequently, gear *E* and cam *G* are driven somewhat faster than the mating cam *H* and gear *N*, so that there is a differential motion between the two. The result is that cam *G* forces cam *H* and shaft *F* to the right at a constant speed until the point of the driven cam passes the point of the driving cam, when the return stroke begins. It will be noted that spring *K*, acting

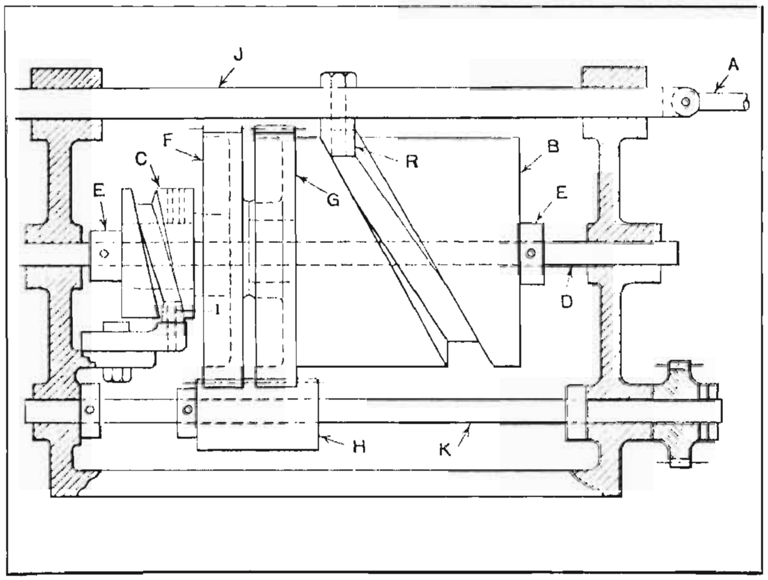


Fig. 8. Double Cam and Gear Combination for Producing Variable Stroke

through lever *J*, holds cam *H* in contact with cam *G* during the return movement. Gear *D* is made wide enough to permit gear *N* to continue in mesh during the entire stroke. This mechanism, with more or less modification to suit the purpose, could be applied to various classes of machinery.

Variable-Stroke Mechanism.—The purpose of the mechanism shown in Fig. 8 is to impart a variable-stroke motion to rod *A*. This is accomplished by two cams *B* and *C*, mounted on shaft *D*. These cams are a free running fit on shaft *D* and are held in place by two collars *E*. Cam *C* is

keyed to the hub of gear *F*, which has 100 teeth. Gear *G*, which has 111 teeth of modified form and is of the same diameter as gear *F*, is keyed to the hub of cam *B*, which transmits motion to rod *A* through roll *R* and bar *J*.

The gears *F* and *G* are both driven by the wide-faced pinion *H*, secured to shaft *K*. The cam roll *I* is mounted on a bracket secured to the machine frame, and remains in a fixed position. As the gears *F* and *G* revolve, the former gains eleven teeth on the latter at each revolution, thus shortening and lengthening the throw of slide bar *J* and rod *A*, the stroke being lengthened when the relative angular positions of the cams are such that they impart motion in the same direction, and shortened when the motion imparted by the cams is opposed.

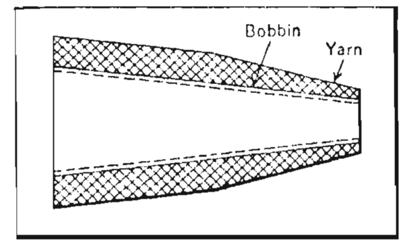


Fig. 9. In Winding this Bobbin, the Yarn is Guided by Means of the Mechanism Shown in Fig. 10

Varying a Reciprocating Movement at One Point of Reversal.—In a certain textile machine, the member that guides the yarn as it is wound on conical bobbins is given a reciprocating movement of uniform length until several layers of yarn have been wound. Then the length of this movement is gradually diminished so that when completely wound, the yarn on the small end of the bobbin forms a cone of greater taper than the bobbin itself, as shown in Fig. 9.

The mechanism for producing this movement is shown in Fig. 10, the member for guiding the yarn being indicated at *A*. This member slides on the stationary guide *C* and receives its motion from the reciprocating cross-head *G* through the bellcrank lever *M*, pivoted to the cross-head at *H*. The cross-head slides on stationary bars *E* and *F*, and is reciprocated by means of cam *K* on shaft *L*.

On the lower arm of lever *M* is a roll *m* which engages a channel cut in the bar O, pivoted at n. Another roll P at the free end of bar O engages the groove in the cam Q. This cam controls the angular position of bar O, and is rotated at the required speed by the worm and worm-gear S and r. It will be noted that the path of cam Q is con-

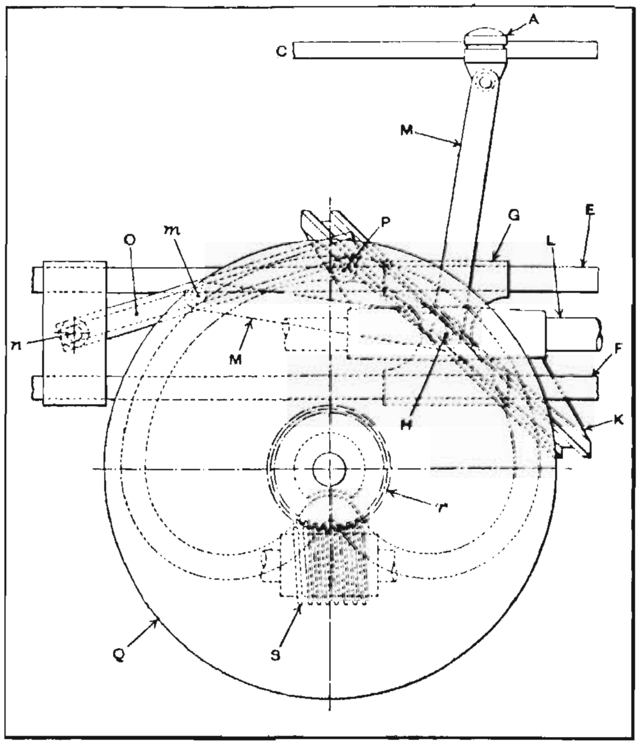


Fig. 10. Reciprocating Mechanism for Varying the Length of the Stroke of Member A which Guides the Yarn as it is Wound on the Bobbin Shown in Fig. 9

centric with its shaft for 180 degrees. Hence, while roll P is passing over this part of the cam, bar O will remain stationary and the length of the stroke of member A will remain constant. This is clearly shown in the diagram, Fig. 11, where the length of the stroke at this time is indicated at S_1. It will be seen that this stroke is equal to the movement of the cross-head G, Fig. 10, plus the movement

of the upper end of lever M resulting from its engagement with bar O. It is during the cam dwell that the first layers of yarn are wound along the length of the bobbin and parallel to its conical surface.

At the end of the dwell, however, roll P moves toward the center of the cam, swinging the bar O downward and thus changing the angular position of the lever. As a result, the stroke of member A is gradually diminished until

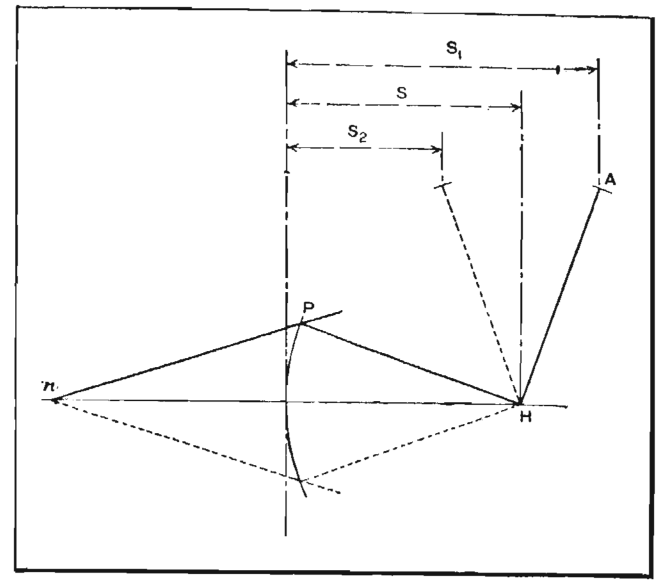


Fig. 11. Diagram Showing how the Oscillation of the Bellcrank Lever Shortens the Stroke in the Mechanism Illustrated in Fig. 10

the channel bar and the lower arm of the lever are in line. In this position (momentarily), the linear speed of member A and cross-head G are equal and their movement is indicated at S in the diagram, Fig. 11.

As roll P, Fig. 10, continues toward the center of the cam, the stroke of member A decreases still more until the channel bar and lever have assumed the position indicated by the dotted lines in Fig. 11. The stroke now is equal to S_2 and at this time, the bobbin is completely wound. Re-

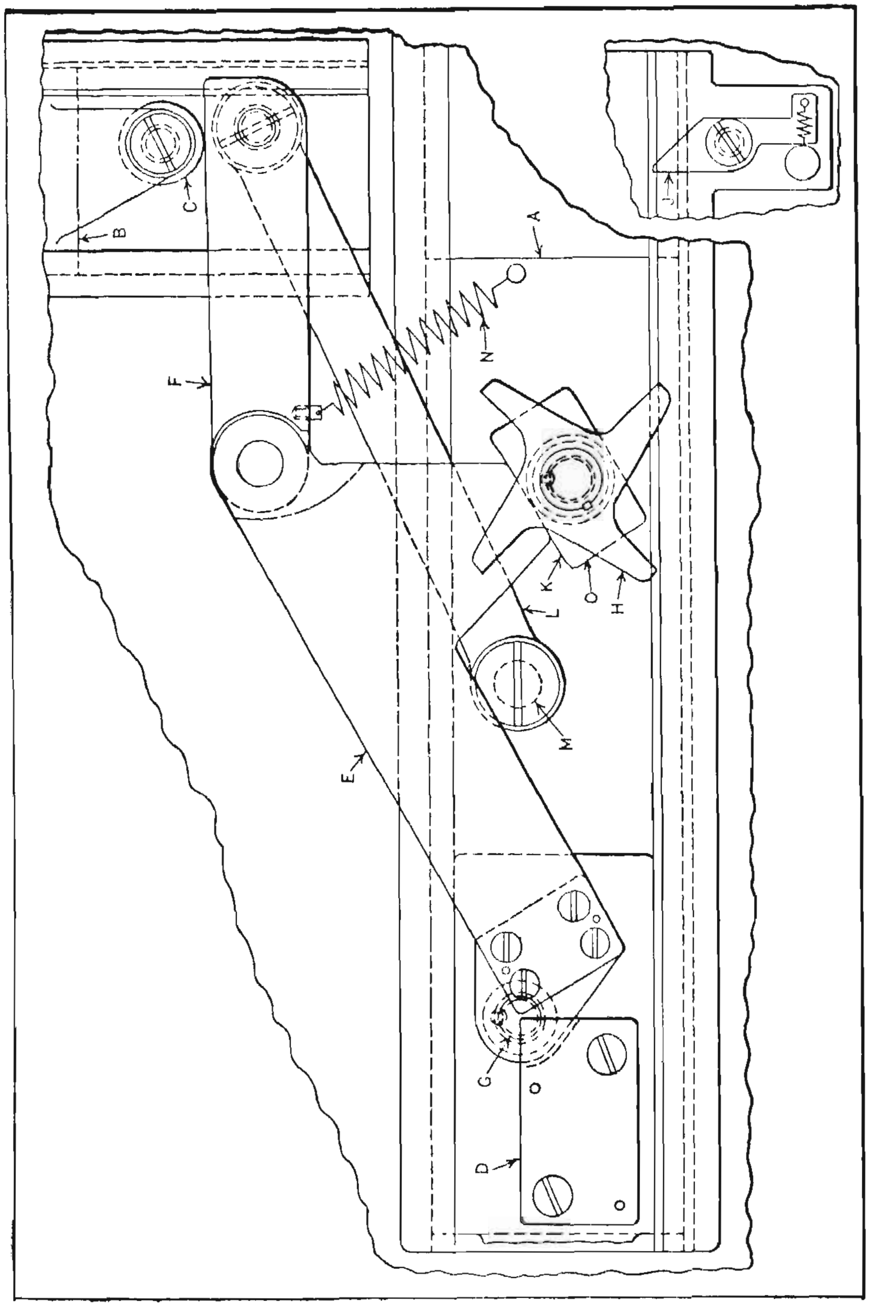


Fig. 12. Mechanism by Means of which the Horizontal Reciprocating Slide A Alternately Imparts a Long and a Short Stroke to Slide B

ferring to the diagram, it will be seen that the stroke is shortened only at one end. Consequently, as each successive stroke is shortened, the length of each successive layer of yarn is decreased a corresponding amount. Hence, the wound yarn at the small end of the bobbin forms a cone having a greater taper than the bobbin itself. This increased taper depends on the contour of cam Q and also on the rotary speed of the cam. This type of mechanism is used in many winding machines other than textile machines.

Alternately Imparting Long and Short Stroke to Slide.— The cam mechanism, Fig. 12, imparts a long and a short stroke alternately to a slide, a dwell occurring at both ends of each stroke. This slide serves to change the position of the carton of a carton-stapling machine, relative to the stapling tools.

Perhaps the most interesting feature of this arrangement is the fact that only one point of reversal is varied to obtain the two different strokes. The cam is made in three parts, consisting of block D, arm E, and bar F. It is mounted on the slide A, which is given a constant reciprocating movement by another member of the machine (not shown). The required movement is imparted to slide B by contact of the roll C with the cam.

Block D is secured to slide A and causes slide B to dwell at the left-hand end of the stroke. Arm E is pivoted at G to slide A, and its angular position is varied every other stroke by means of the star-wheel H and the pawl J. Star-wheel H, together with the block K, is keyed to a shaft that is free to turn in its bearing in slide A, while pawl J is pivoted to the machine frame. Contact between block K and arm E is maintained by spring N. Bar F maintains a horizontal position on both the long and short stroke of slide B, and it was to obtain this condition that the link L was incorporated. This link is pivoted to slide A at M and is connected to the bar F. With this arrangement, bar F remains in a horizontal position when arm E changes its angular position, thus maintaining the dwell at the right-hand end of the cam.

Reciprocating Slide with Cam Mechanism for Operating Tool or Drill Slide.— The reciprocating slide *A*, Fig. 13, is driven by an eccentric connected to rod *D*. This slide has a cam mechanism by means of which motion is applied to the slide *B*. The motion of slide *B* is employed for feeding metal-cutting tools to the work. It can be applied to

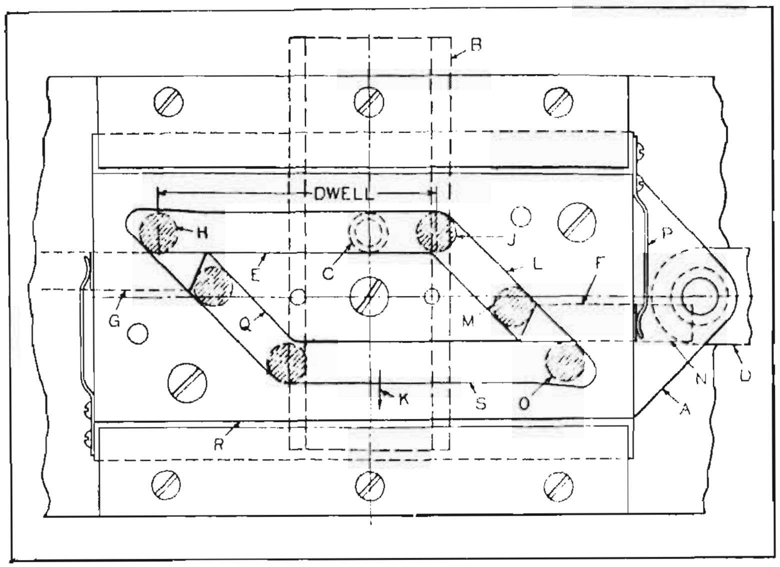


Fig. 13. Reciprocating Slide Mechanism for Operating Tool-slide

the spindle of a drill press, for example, to advance and withdraw the drill. By making suitable changes in the cam slot the drill can be given a rapid approach and reduced feeding movement, followed by a rapid return to the starting position and dwell. The movements required for this operation can be obtained by changing the cam-plate *Q* and guide plate *R*. These plates are secured in place by screws and dowels.

The cam-roll *C* is mounted on the driven slide *B*, which can move only in a direction at right angles to the move-

ment of slide *A*. The cam slot *E* in slide *A* has latches or slides *F* and *G* which project into the slot and prevent the cam-roll from reversing its direction of travel in the cam slot. When slide *A* is at its extreme right-hand position, the cam-roll will be located in the cam slot as indicated at *H*. The roll remains stationary while the slide *A* moves to the left until it reaches the position indicated at *J*, thus allowing the slide *B* to dwell. Any further movement of slide *A* to the left will cause roll *C* to move down the inclined portion *L* of the cam slot, moving slide *B* in the direction indicated by the arrow *K*. When the cam-roll reaches the point *M*, it forces the latch *F* back to the position indicated by the dotted lines at *N*. As soon as the roll reaches position *O*, latch *F*, under pressure from spring *P*, snaps back to the closed position. At this point, the eccentric that reciprocates slide *A* has reached its highest point and the slide commences to travel in the reverse direction. On the return stroke, the same cycle is repeated, causing the slide *B* to be returned to its original position when the cam-roll reaches the position indicated at *H*. Within reasonable limits, the dwell positions of the cam slot, as at *S*, can be changed to produce any sequence of movements or dwells required.

Combined Reciprocating and Elevating Movement.— The device illustrated in Fig. 14 is used for skimming dirt and oxides from the surface of molten lead in a galvanizing vessel. Pieces to be galvanized are dipped in this vessel, and in order that they shall have a smooth and bright surface, all foreign matter must be removed from the lead before the pieces are withdrawn. The skimming is done by means of the reciprocating blade *G*. The blade is in contact with the lead on the stroke from right to left. The return stroke, however, is made with the blade in an elevated position, as shown in the end view. With the blade in the latter position, the pieces to be coated can be readily placed in or withdrawn from the vessel.

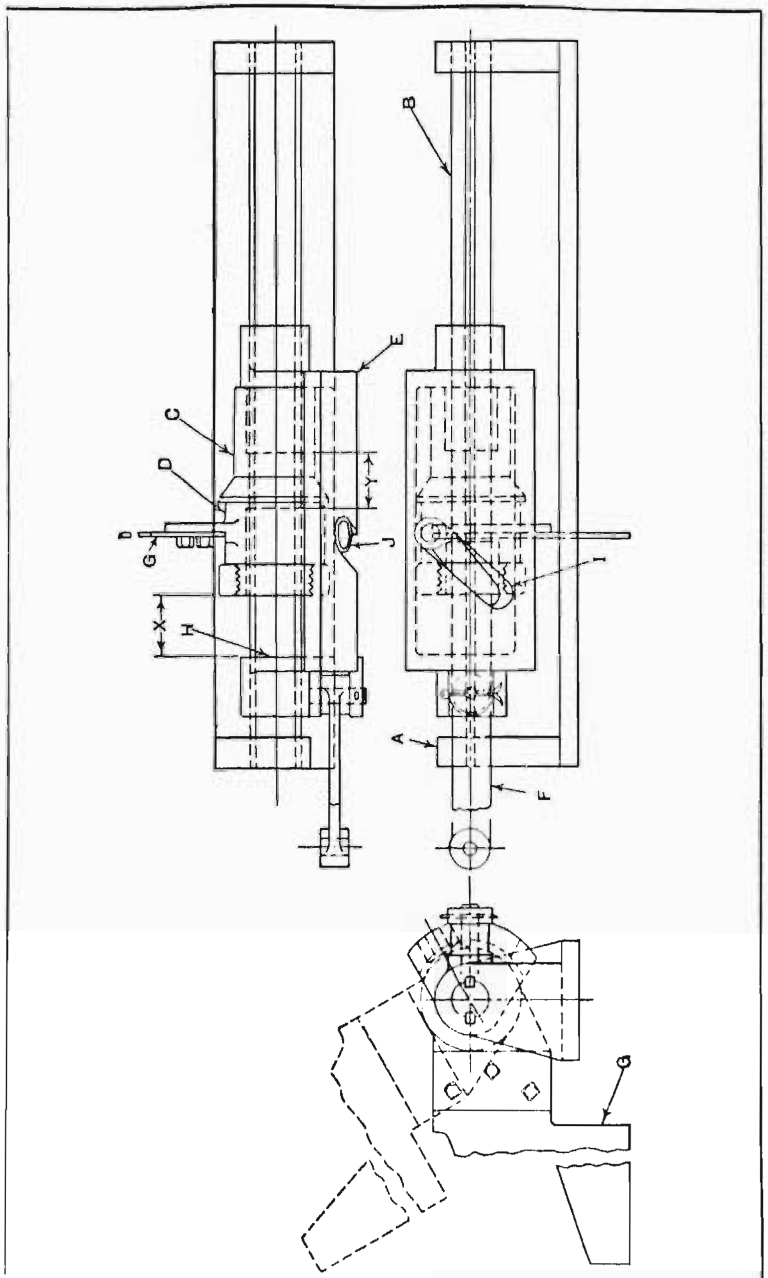


Fig. 14. Combined Reciprocating and Elevating Movement Applied to Skimmer on Galvanizing Vessel

On the bracket A, secured to the side of the vessel, is fastened a stationary shaft B. Sliding on this shaft is a sleeve C which, in turn, forms a bearing for the bushing D. On one side of this bushing is an extension to which the skimmer is fastened, while on the other side is mounted a cam roller J which engages an angular slot in the carriage E. The carriage slides on shaft B, and is given a reciprocating motion by a crank (not shown) through the connecting-rod F. Both members C and E are prevented from rotating by keys in shaft B.

Referring to the plan view of the illustration, it will be seen that the skimmer blade is at its farthest position to the left. The carriage E now moves toward the right, and after traveling a distance X, the surface H on the carriage boss comes in contact with the end of the sleeve C. While the distance X is traversed by the carriage, the sleeve C is stationary and the roll J is forced downward due to the angularity of the cam slot I. This movement of the roll will cause the blade G to rise above the molten lead.

The carriage E continues to move to the right with the blade in its elevated position until the end of the stroke is reached. On the return of the carriage, the gap shown at X will be on the other end of the sleeve C. As the width of this gap decreases, the cam roll will ride to the top of the slot I, causing blade G to enter slightly past the surface of the lead. The blade, held in this position, skims the surface of the lead as it continues its stroke to the position shown in the plan view.

There must be sufficient friction between the shaft and the sleeve so that the latter will remain stationary while the roll J raises the skimmer blade. This friction is obtained by counterboring both ends of the sleeve until the length of its bearing on the shaft is shortened to a distance Y. The location of this short bearing surface is such as to cramp the sleeve enough to obtain the desired friction. In

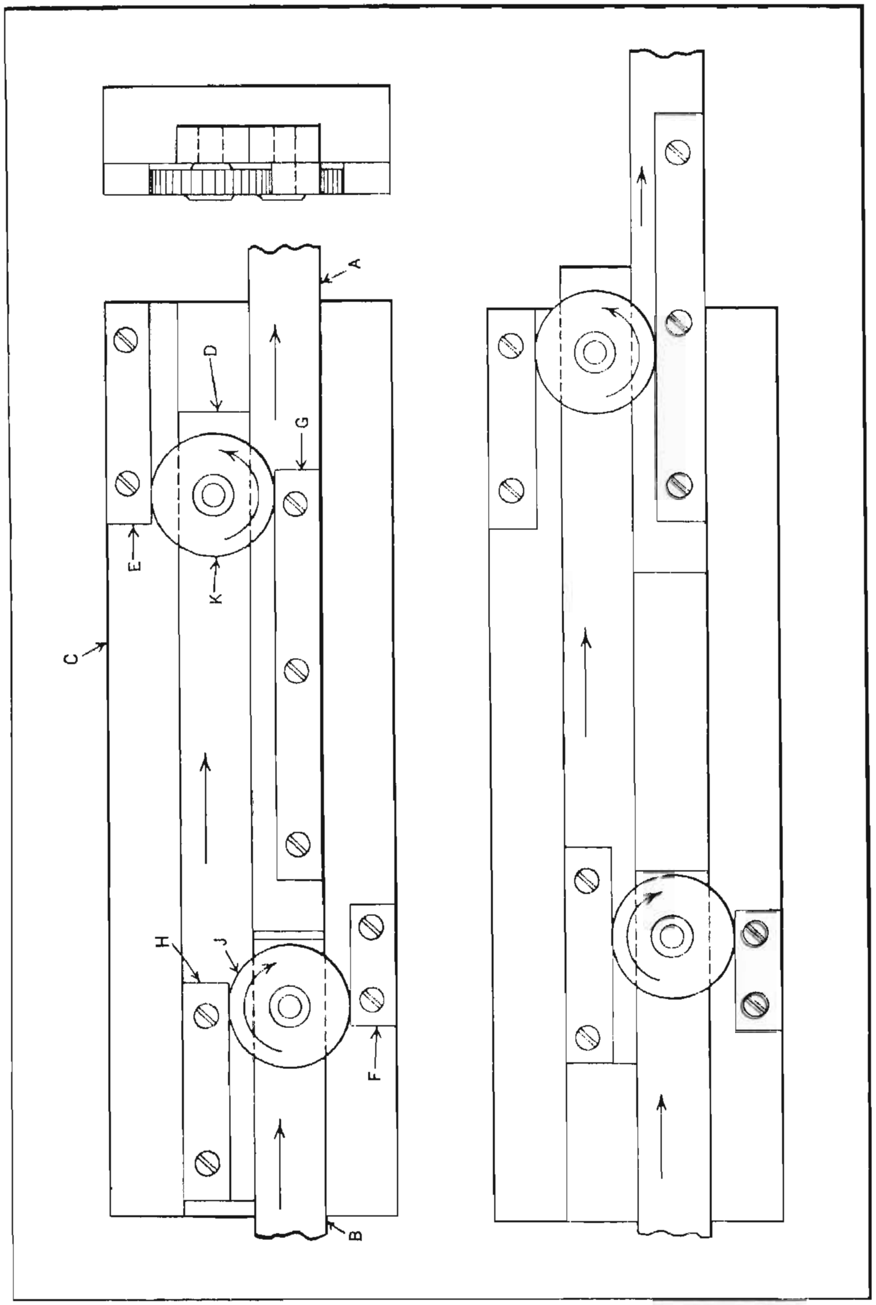


Fig. 15. Rack and Gear Mechanism for Increasing the Stroke Imparted by a Slide to Four Times its Original Travel

this simple way, a very effective and dependable frictional grip is obtained.

Quadrupling the Travel of a Slide.—In the mechanism illustrated in Fig. 15, the reciprocating slide *A* has a stroke four times as long as the slide *B* from which it receives its motion. This is effected through a series of racks and pinions, the pinions moving in a straight line and meshing with two opposite racks, one of which is fixed and the other free to slide. Obviously, the rack that is free to slide will move twice as far as the center of the pinion. This design is advantageous when a compact arrangement is required, and by using more than two gear combinations, the stroke imparted by slide *B* can be increased to any length.

The mechanism is mounted on the stationary block *C*, which is grooved to receive the three reciprocating slides *A*, *B*, and *D*. On each of the slides *A* and *D* is secured a rack, as indicated at *G* and *H*. Two stationary racks *E* and *F* are fastened to block *C*. On the ends of the slides *B* and *D* are the pinions *J* and *K*, each of which meshes with a fixed and a sliding rack. Full lines are used to represent the pitch lines of the gears and racks.

Now it will be seen that if slide *B* is advanced toward the right, say 1 inch, slide *D* will move 2 inches in the same direction through the action of pinion *J* meshing with the racks *F* and *H*. The same combination of gearing exists at the right-hand end of the block. Consequently, if slide *D* moves 2 inches, the stroke imparted to slide *A* will be 4 inches. At the end of this 4-inch stroke, slide *A* will be in the position shown in the lower view.

Intermittent Trigger Slide Having a Positive Working Stroke and a Swift Return.—A reciprocating slide having a trigger action is employed on an automatic nut-tapping machine for feeding the nut positively and at a relatively slow speed from the magazine to the tapping position. There, the slide dwells during the operation, after which the trigger action releases it, so that it returns swiftly to the magazine for another nut.

The mechanism is mounted on the machine frame *H*, Fig. 16. It consists of the dovetail slide *A*, on which is guided the auxiliary slide *B*, both slides being actuated by the oscillating segment gear *C*. Gear *C* oscillates at a constant velocity and receives its motion from another member of the machine (not shown). Secured in slide *B* is the round rod *D* which is connected at its left end to the work-carrier (not shown). Spring *E*, which imparts the rapid return movement to the work-carrier, is mounted on rod *D*, and the spring tension is released for the return stroke by the trigger *F*, pivoted on the stationary bracket *G*.

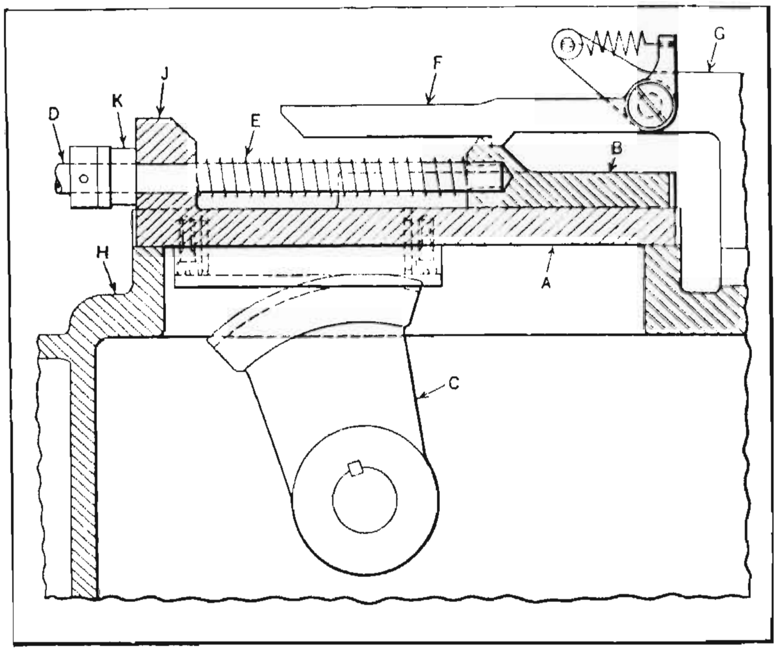


Fig. 16. Slide Having a Positive Working Stroke and a Rapid Return Movement

Gear segment *C*, in the position shown in the illustration, has carried both slides to their extreme left-hand position, and in doing so, has caused rod *D* to transfer a nut from the magazine to the tapping position. At this point, trigger *F* has engaged the projection on the auxiliary slide *B*. The gear segment now reverses its movement and carries slide *A* toward the right. During this movement, slide *B* remains stationary, since it is held by the trigger. Near the end of the stroke of slide *A* toward the right, the projection *J* engages the end of the trigger and lifts the latter away from the projection on slide *B*; consequently, slide *B* is released and under the action of spring *E* is carried toward the right until the rubber bumper *K* comes in contact with projection *J*. Thus, with the return of slide *B*, the work-carrier and rod *D* are returned swiftly to the magazine for another nut. This completes the cycle of the mechanism.

Slide which Dwells at One End of Its Stroke.—The screw shells on the plugs attached to electric extension cords are spun in place on the plugs in an automatic machine. The assembled shell and plug is delivered to the machine from the magazine by means of a feed-slide having a dwell at one end of its stroke. The slide is designed to have a positive action. The dwell occurs when the slide has carried the plug to its spinning position and continues until the shell has been spun and the finished plug ejected from the machine.

Referring to Fig. 17, the feed-slide *A* is mounted in guides on the machine frame *B*. The driving lever *C* is equipped with a roll *D* which engages a cam slot *E* in a projection on the slide. The illustration shows the empty slide in the position it occupies after being carried back toward the left to the magazine (not shown). As the lever reverses its motion, the slide is returned with a plug to the position indicated by the dot-and-dash outline, the lever rotating through angle *b*. This is the position of the slide while the plug is being spun. The dwell of the slide that permits this operation is obtained as the lever continues its movement along the curved portion of the cam slot, the latter being concentric with the lever shaft at this time.

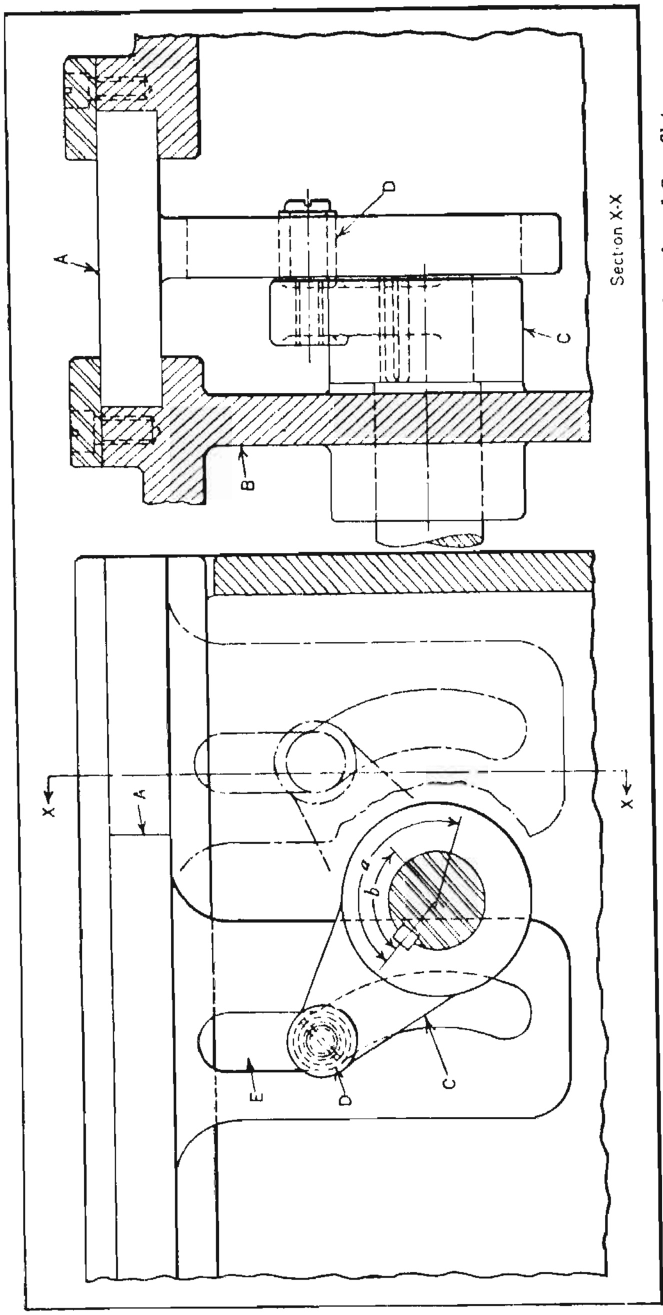


Fig. 17. Slide which Dwells at One End of its Stroke while Roller on Oscillating Driver Travels along Arc of Cam Slot

The dwell continues until the lever has moved through the angle a minus b and back to the position shown by the dot-and-dash outline. At this point the completed plug is ejected from the slide by a device not shown. The con-

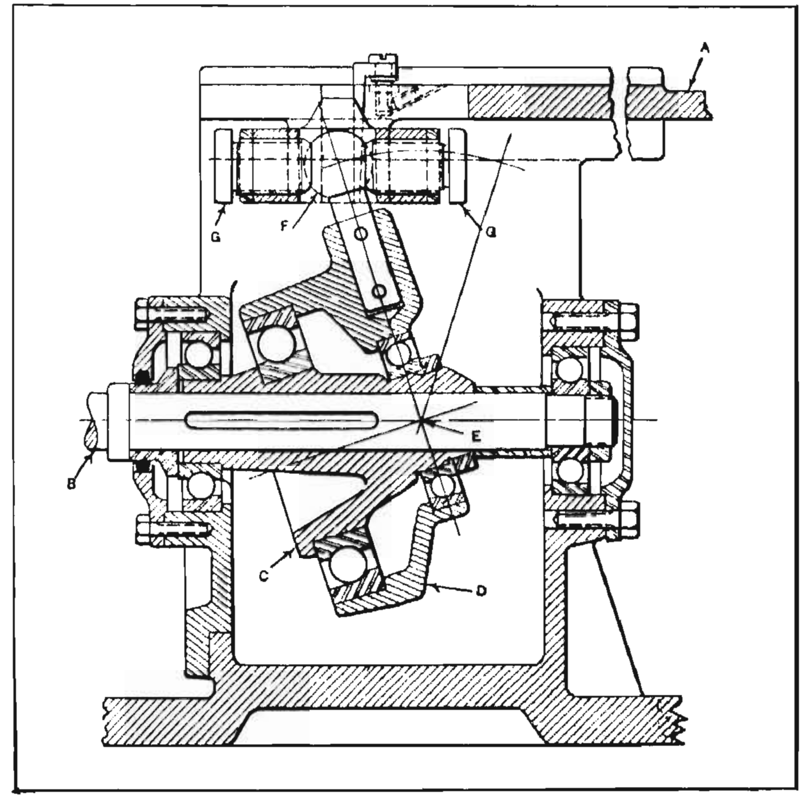


Fig. 18. Reciprocating Mechanism of a High-speed Machine for Operating Slide Parallel to Driving Shaft

tinued movement of the lever returns the slide to the magazine ready to pick up another plug.

Slide with Reciprocating Movement Parallel to Driving Shaft.—In a special high-speed machine used to shear and form fiber shields for electrical switches, it was required that slide A (see Fig. 18) have a reciprocating motion parallel to driving shaft *B*. It was also essential to operate slide *A* without lost motion due to wear, and the design here shown has proved satisfactory in this respect.

The reciprocating motion is obtained from an angular eccentric sleeve *C*, secured to driving shaft *B*. As this sleeve revolves, it imparts a swinging motion to part *D* about center *E*. This motion is transmitted to slide *A* through a rod which is fixed to *D* and has a ball-shaped end *F*. The spherical end engages concave seats in bronze screws *G*, held in a cross-head that is free to slide vertically far enough to provide for the rise and fall resulting from the circular movement of ball *F*. Screws *G* provide adjustment to eliminate play, and the vertical cross-head slide has adjustable gibs.

The two ball bearings that support the driving shaft and also the two between sleeve *C* and part *D* are of the combination radial and thrust type. The mechanism is enclosed in a bracket cast integral with the machine proper and forming a well so that the lower members are always in a bath of oil. This reciprocating mechanism operates smoothly and accurately, and requires little attention other than to add oil to the well at intervals of approximately two months.

Obtaining Two Reciprocating Motions from One Movement.— A change in a wire product necessitated changing the mechanism of a wire-forming machine so that the reciprocating motion originally used would be replaced by two similar movements of lesser magnitude in the same period of time. Fig. 19 shows how this was accomplished, using the same source of power. Originally the required reciprocating movement was furnished by rod *A*. In the new arrangement, this rod actuates rod *K*, causing it to move forward and back while rod *A* is moving in one direction.

Rod *A* is given a reciprocating motion from a distant source of power for transmitting the oscillating motion to

the lever *B*, which is fastened to gear *C*. Gear *C* and lever *B* are free on stud *D*, and oscillate in unison. Gear *C* transmits motion to gear *E*, which carries the lever *F*, both of which are free on stud *G*. Lever *F* carries the pin *H*,

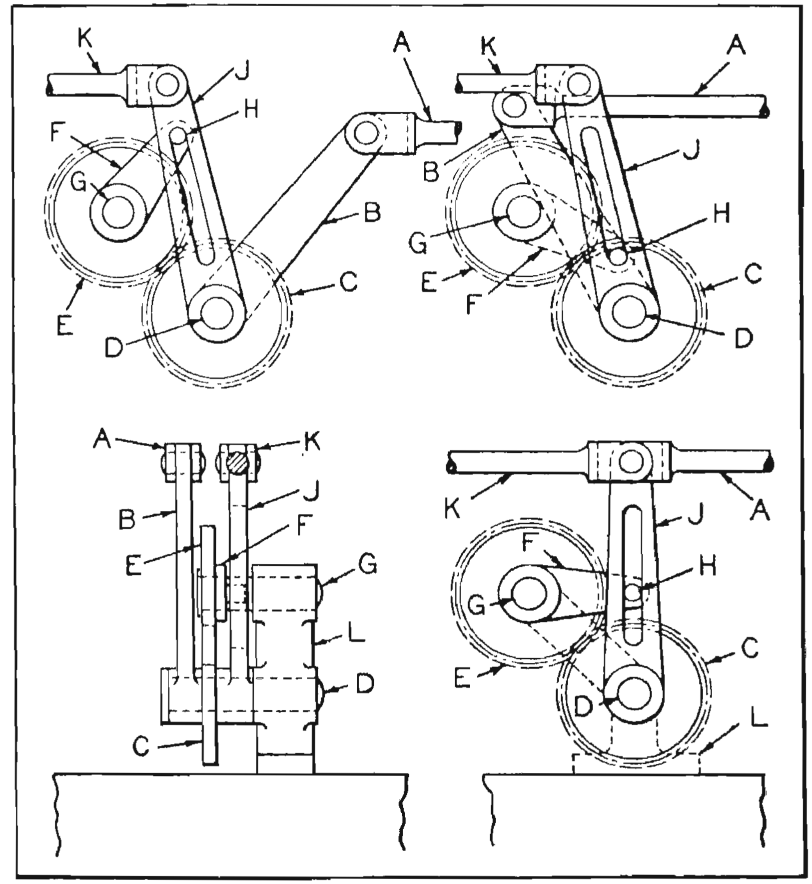


Fig. 19. Diagrams Illustrating Operation of Reciprocating Mechanism

which travels in a slot in lever *J*, transmitting motion to rod *K*. The assembly is supported by the bearing *L*.

In the two upper views, rod *A* is shown at its extreme right position and its extreme left position, representing half its cycle of operation. It will be noted that in both these views, lever J occupies the same position, having passed through one complete cycle and returned to its original position. Starting its movement from the position shown by the upper left-hand diagram, lever B is moved to the left by rod A, causing gear C to make a partial revolution. Gear E, meshing with gear C, is thus given a partial revolution in the opposite direction. As lever F is fastened to gear E and moves with it, pin H is moved downward in the slot in lever J, causing the latter to move to the right until pin H reaches the horizontal center line of stud G, at which time lever J is at its extreme right-hand position, as shown by the lower right-hand view.

Continued movement of rod A produces a further downward movement of pin H. As pin H passes the center line, it acts against lever J in the reverse direction, moving it to the left. As rod A reaches its extreme left position, lever J is also at its extreme left position, having completed its cycle, whereas rod A has completed but half its cycle. As rod A returns to its extreme right position, lever J again passes through its cycle. The magnitude of the movement of lever B may be determined by comparing its positions (see two upper views), while the movement of lever J will be understood by reference to the upper left-hand and the lower right-hand diagrams.

Slide with Dwell at Ends of Stroke and Quick Return.— A mechanism designed to give an intermittent movement to a reciprocating slide is shown in Fig. 20. For every revolution of the shaft I, the slide J rises at a comparatively slow speed until it reaches the position shown by the dotted outline; the slide then dwells at this point for a certain period of time, after which it returns to its original position. These movements are secured through the action of the lever arm D on the latch C and on the projecting lug K of the slide. The end of the arm, rotating at a uniform speed, engages the lug K and raises the slide until the latch C catches on the lower lug G.

The slide is held in this dwelling position until the arm trips the latch, when the slide drops down on stop-pin B to the position shown, thus completing the cycle. A further motion of the arm raises the slide. The member of the machine on which the end H of the slide acts (not shown) returns under the action of a coil spring, carrying the slide

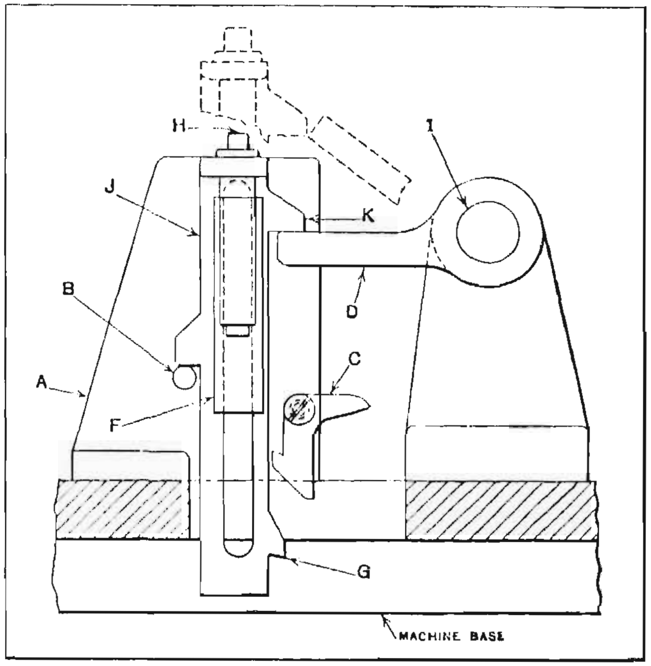


Fig. 20. Slide Mechanism Producing an Intermittent Motion and a Rapid Return

back also. It will be noted that the angle through which the arm must turn to raise the slide J the required height is governed by the over-all length of the arm D and the location of the shaft I. The slide is confined in its path by the T-shaped gib F on the bracket A.

Slide which Dwells During Every Other Cycle of Driving Slide.— On a certain bread-wrapping machine the mechanism, Fig. 21, controls the action of the bread-shifting slide.

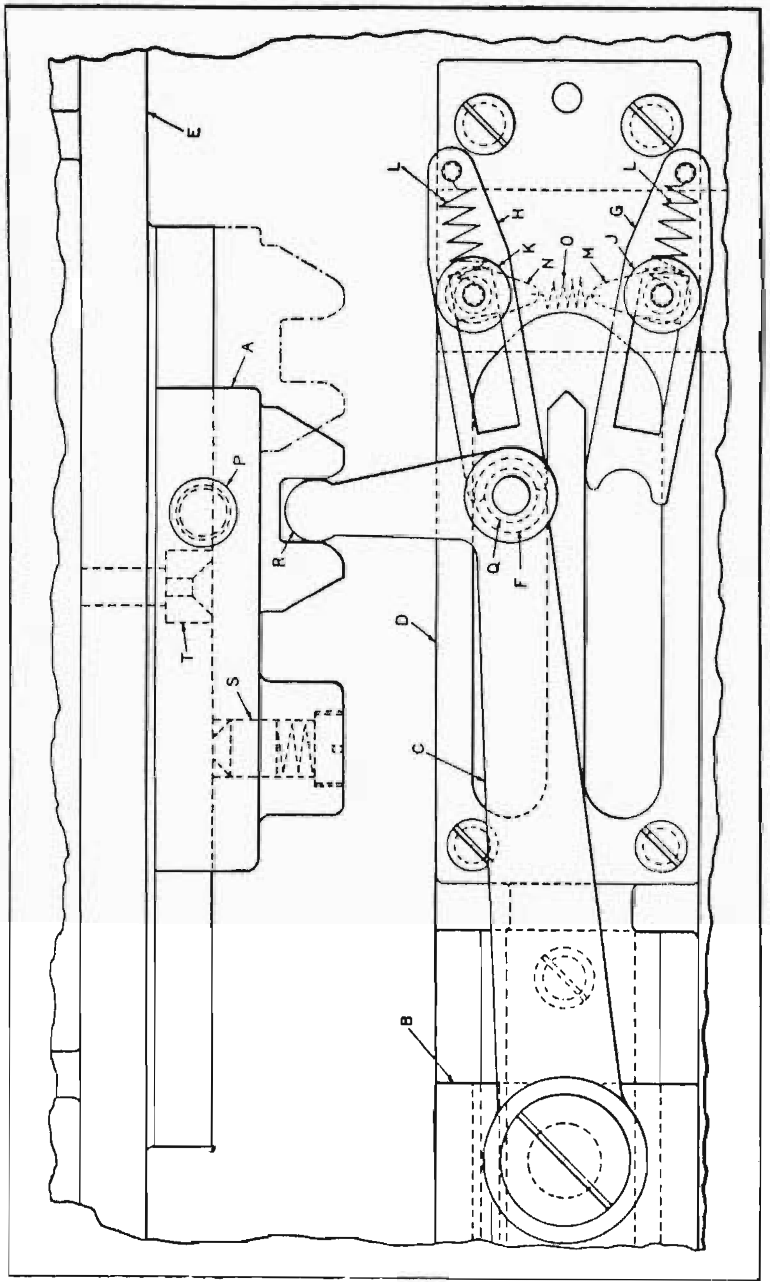


Fig. 21. Mechanism Driven by Constantly Reciprocating Slide B which Allows Slide A to Dwell During Every Other Cycle

This slide is operated by another sliding member and dwells during every other cycle of the driving member.

The bread-shifting slide is indicated at *A*. This slide transmits its controlling motion to the loaf-holding member (not shown) through stud *P*, and is actuated by the continuously reciprocating slide *B* through the connecting-rod *C*. Rod *C* is automatically disengaged from slide *A* after every other cycle of slide *B* by the switching arrangement mounted on the base *D*, which is secured to the machine frame *E*. Thus, base *D* serves also as a guide for the slide *B*. In the top of the base is machined a U-shaped groove with which the roll *F* on the connecting-rod engages.

Two spring-actuated switching arms *G* and *H* are pivoted to the base by the pins *J* and *K*. These pins are free to turn in the base and have a square shoulder near their upper end on which the arms slide. The arms are held normally in the position shown by the coil springs *L*. On the lower ends of the pins are secured fingers *M* and *N*, connected by the coil spring *O*. The tension of spring *O* serves to return the arms to their normal positions.

Slide *B*, together with connecting-rod *C*, moves slide *A* through part of its stroke toward the right. As these members continue their movement in this direction, the roll *F* enters the curved portion of the U-groove, withdrawing the projection *R* on the connecting-rod from its recess in slide *A*. This causes slide *A* to stop. In the meantime, however, the end of arm *H* engages the shoulder on the roll stud *Q* and is forced back toward the right. Thus, when the roll has reached its extreme right-hand position, the energy stored up in spring *L* forces the roll past the dead center. At this point, the slide *B* reverses its motion, and as it moves toward the left, the roll travels in the lower part of the groove. During this stroke of slide *B*, and also during its return stroke, the projection *R* on connecting-rod *C* remains disengaged from slide *A*. Hence, the latter dwells during this cycle. However, when slide *B* returns,

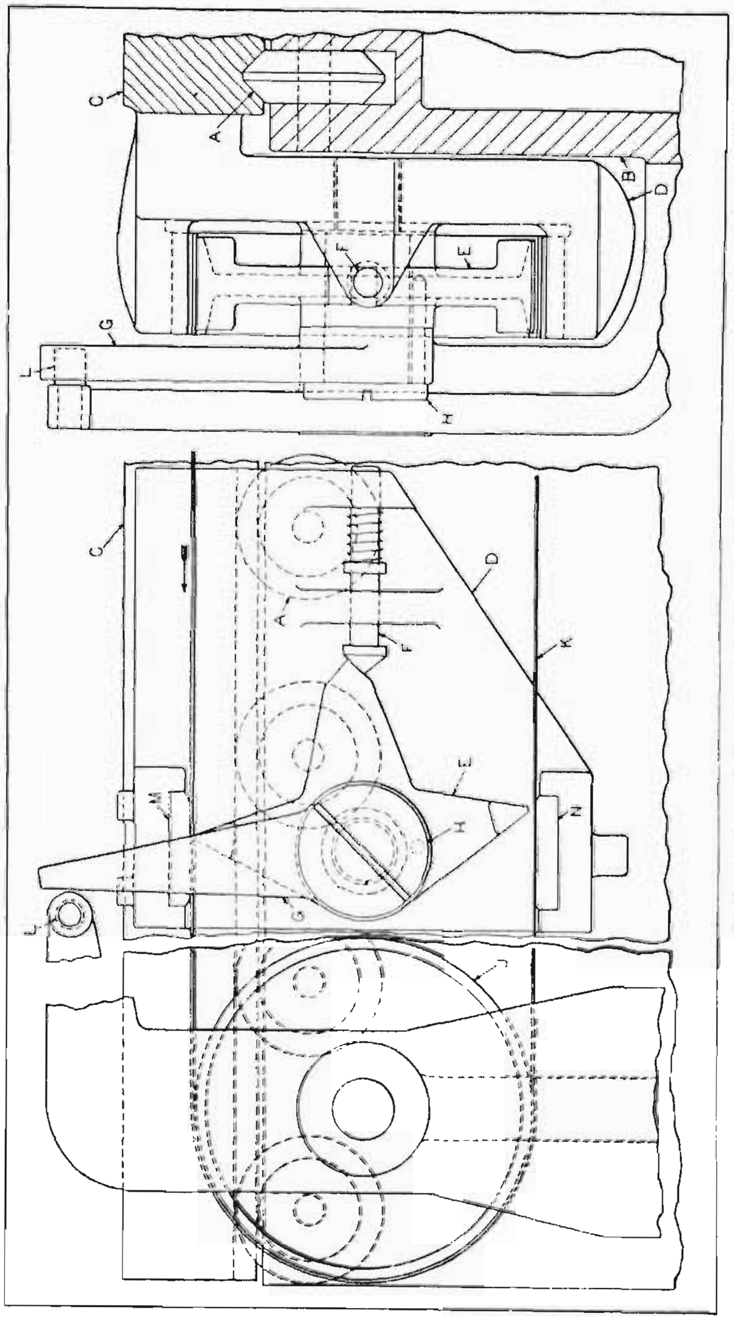


Fig. 22. Mechanism Used on Machine Table to Alternately Engage the Upper and Lower Sides of a Horizontal Belt, thus Obtaining a Reciprocating Motion

the roll stud *Q* engages arm *G*, so that when the roll reaches its extreme right-hand position in the groove, the arm forces the roll past the dead center and into the opposite section of the groove.

In entering this section of the groove, the projection on the connecting-rod again engages the recess in slide *A*, so that this slide is carried with slide *B* toward the left. It also returns with slide *B* to the position indicated by the dot-and-dash outline. At this point, the connecting-rod projection is again disengaged, as already described. Thus, slide *A* has a dwell equivalent to a complete cycle after every other cycle of the machine. In order to have slide *A* stop at exactly the same position every time it dwells, the spring-actuated plunger *S* was provided. The end of this plunger merely rides along the top of the guide for slide *A* until the slide reaches its dwelling position. When this occurs, the plunger drops into the depression in the bushing insert *T*, thus locking the slide securely during the idle stroke of slide *B*.

Reciprocating Motion Obtained by Alternately Engaging Upper and Lower Sides of a Steel Belt.—The turned shafts used in a certain type of machine tool are finished by polishing with abrasive cloth. This work is done on a machine in which the shaft is revolved between centers while the abrasive cloth, in a suitable holder, is held in contact with the shaft and moved back and forth by a reciprocating table. The long reciprocating movement of the table is obtained by alternately engaging and disengaging opposite sides of a horizontal steel belt. The mechanism secured to the table for automatically engaging and disengaging the belt is shown in Fig. 22.

The reciprocating table *C* is mounted on the beveled rollers *A*. An apron *D*, cast integral with the table, carries the reversing tumbler *E*, the spring-actuated plunger *F* and the dog lever *G*. Tumbler *E* and lever *G* are pinned together, but are free to turn on the stud *H*, secured in the

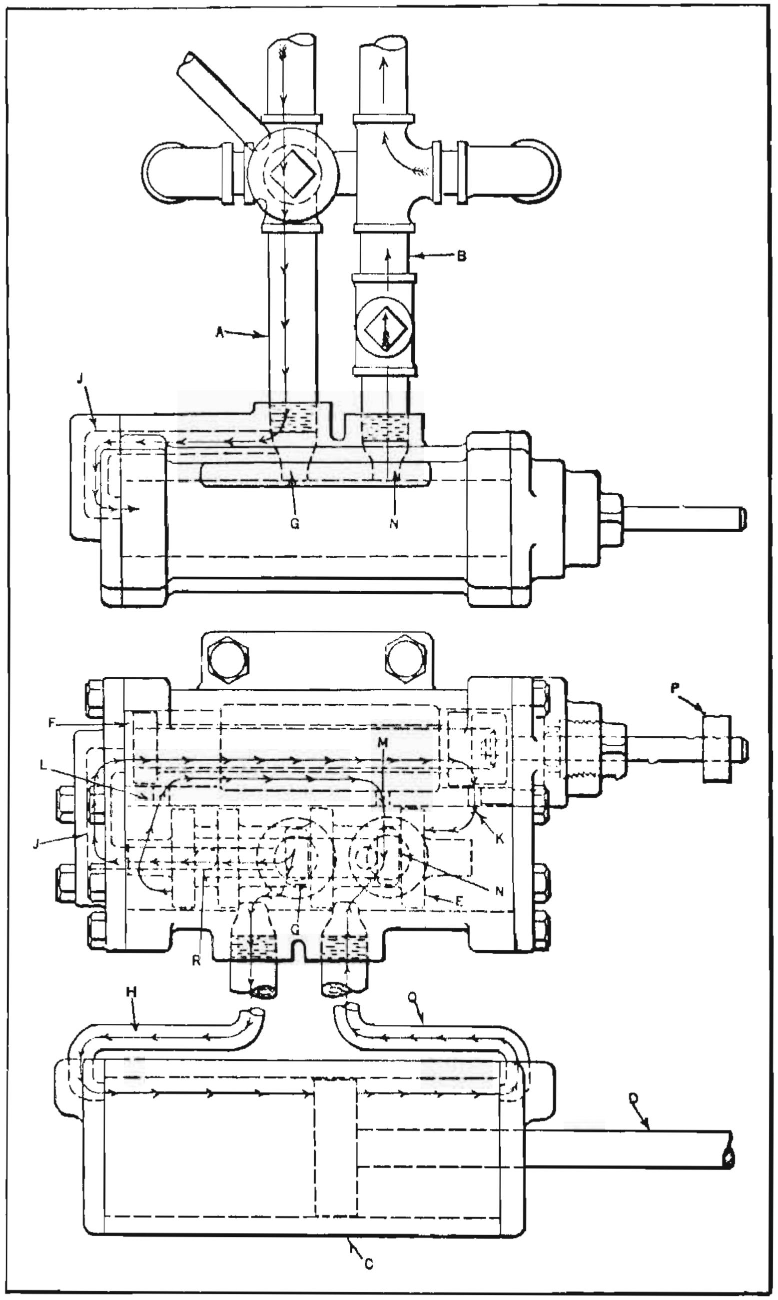


Fig. 23. Hydraulically Operated Reciprocating Mechanism

apron. The belt drums are mounted on shafts supported by brackets which are secured to the machine bed. Only the left-hand drum J is shown. This is the driving drum which rotates at a constant velocity. The other drum is the idler, and its bracket has a horizontal adjustment for taking up the slack in the steel belt K. On an extension on each of the brackets is a pin L, which engages the lever G at each end of the table stroke, causing the movement of the table to be reversed.

The spring on plunger F is made heavy enough to maintain engagement of tumbler E when the table is moving toward the right. As shown, the action of plunger F causes the top side of the belt to be gripped tightly between the top prong of the tumbler E and the block M secured to the apron. Consequently, the table must travel in the same direction as the top side of the belt, as indicated by the arrow. At the end of the movement in this direction, the lever G engages stationary pin L which swings the lever in a clockwise direction, thus disengaging the top prong of tumbler E from the belt.

Although the belt is then disengaged from the table, the resulting momentum causes the table to continue its motion in the same direction, so that the lower prong of the tumbler engages the lower side of the belt. This causes the belt to be gripped between the lower prong of tumbler E and the apron block N. As the lower side of the belt is moving toward the right, the movement of the table will be reversed. Wear resulting from use will eventually destroy the gripping action of the prongs and blocks, but this condition can be easily corrected by placing shims under the blocks M and N.

Hydraulic Reciprocating Mechanism for Machine Tools.—The hydraulic control valve (Fig. 23) is applied to hydraulically operated grinders. This valve controls the flow of oil to and from the hydraulic cylinder that contains the piston or plunger for operating the work-table.

A one-way pump supplies oil at a constant pressure through pipe *A*, and the return flow is through pipe *B*. The work-table is operated by piston-rod *D*, and the flow of oil to and from cylinder *C* is regulated by control valve *E* in conjunction with pilot valve *F*. The illustration has been made partly diagrammatic to show the arrangement more clearly.

When valve *E* is in the position shown, the oil from the pump enters through port *G*, which is connected with pipe *A*, and passes through port *H*, forcing the piston to the right. Oil from pipe *A* passes through port *J* and through the hollow pilot valve *F* and port *K*, thus exerting pressure against valve *E* which causes it to shift to the left-hand position shown. The arrows indicate the direction of flow. During this movement of valve *E* to the left, oil which previously entered the left-hand end of the chamber containing valve *E* is exhausted through ports *L*, *M*, and the main exhaust port *N*, as indicated by the arrows.

Now when a stop on the reciprocating part engages collar *P* on the pilot valve rod and moves the pilot valve to the right, port *K* is opened to the exhaust ports *M* and *N*, and oil under pressure flowing through port *J* passes through port *L* and shifts the control valve *E* to its right-hand position. The main inlet port *G* and port *Q* are now connected, so that the piston begins its movements to the left, and oil in the left-hand end of the cylinder is exhausted through ports *R* and then to the right through the interior of valve *E* and out through ports now opposite the main exhaust port *N*. The pilot valve *F* requires a movement of only 3/8 inch, and valve *E* is shifted quickly so that full port opening is obtained without delay and the flow of oil is not restricted.

CHAPTER X

SPEED-CHANGING MECHANISMS

Many different types of mechanisms for obtaining speed variations have been designed. These mechanisms, as applied to machines used in connection with manufacturing processes, permit the speed of cutters or other tools to be regulated to suit different materials or operating conditions, as in the case of machine tools. Numerous designs have also been developed for changing the speeds of moving vehicles. In all of these applications, the general object is to vary the speed of a driven member by mechanical means and independently of the driving engine or motor. Chapter XI of Volume I (pages 310 to 362) deals with different types of speed-changing mechanisms. The additional designs which follow are more or less special and embody interesting principles relating to the design of mechanisms of this general class.

Speed-Reducing Gearing for Operating Press Fixture.—An automatic fixture for a small punch press required a cam to operate it and it was necessary for the cam to make one revolution to seven revolutions of the punch press shaft. The compact mechanism for obtaining this speed reduction is sometimes known as "wobble gearing," owing to the eccentric motion imparted to one of the gears.

The punch press shaft *A* (see Fig. 1) has keyed to it an eccentric *B* (see also detailed view). This eccentric rotates within and transmits an eccentric motion to arm *C* to which is attached an internal or wobble gear *D*. Gear *D* meshes with and drives pinion *E* to which is attached cam *F*. This cam has two working edges for operating followers *G* and *H*, as these followers require different motions. At the lower end of arm C there is a stud J. One end of this stud is fixed to the press frame and the other end engages an elongated slot in arm C, thus preventing the latter from rotating about its axis, but permitting the axis to rotate around a circle equal in diameter to the throw of the eccentric.

The action of the mechanism is as follows: When the

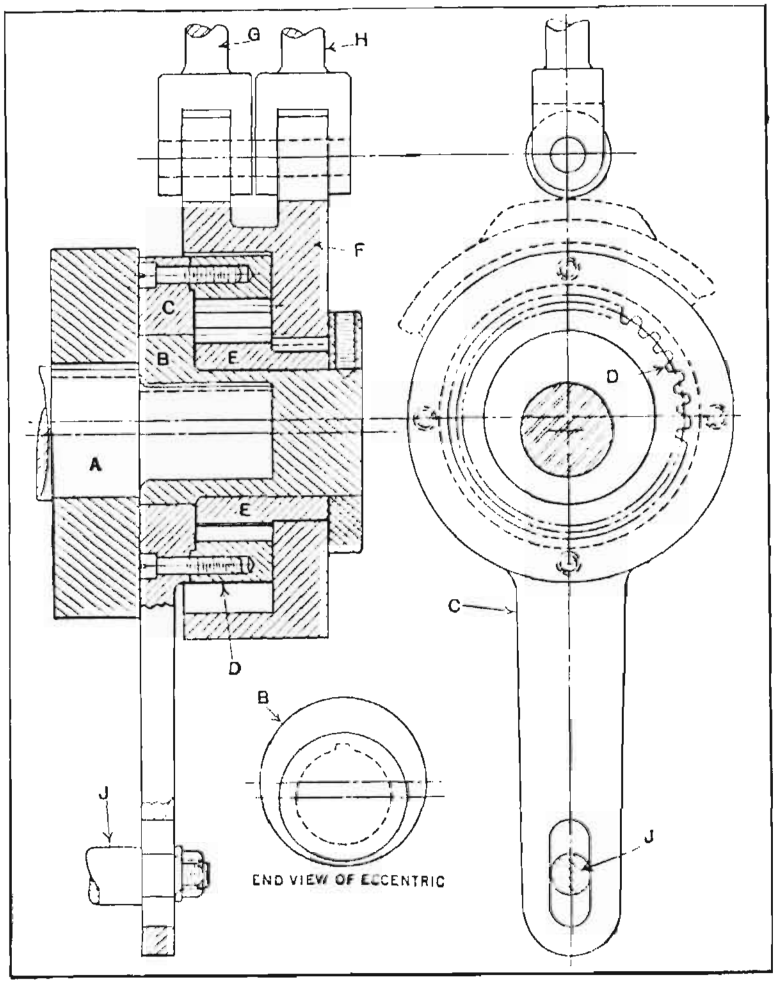


Fig. 1. Compact Speed-reducing Gearing for Operating Press Fixture

press shaft A rotates in a right-hand direction, the driven pinion E revolves in a left-hand direction. In one revolution of shaft A, the rotation of pinion E is equivalent to four teeth, this being the difference between the numbers of teeth in internal gear D and pinion E. Gear D has thirty-two teeth and pinion E twenty-eight teeth and $32 - 28 = 4$; therefore, pinion E will make one revolution for every seven revolutions of the punch press shaft, which is the reduction required. The gears are of 8 diametral pitch and the eccentric radius is 1/4 inch, giving 1/2 inch throw. The gear teeth are modified somewhat to provide clearance for the eccentric movement. All parts are made of machine steel. A 1/8-inch air hole (not shown) is drilled through part B opposite the end of shaft A to permit the air to escape when assembling A and B. The bearing surfaces also have suitable oil holes, which are not shown.

Nine-Speed Gear-Box with Single-Lever Control.—The single-lever control mechanism shown in Fig. 2 permits the operator to obtain instantly any one of nine different speeds. For instance, with a driving motor running at 960 revolutions per minute, the gear-box gives nine speeds ranging from 10 to 50 revolutions per minute. These speeds are obtainable, in the usual manner, with three pairs of sliding gears on two parallel shafts, by sliding each group of three gears. The gears and the gear-box of conventional design are not shown, but the yokes that control the sliding gears are indicated at J and K.

The interesting feature of this design is the arrangement for obtaining the nine changes of speed by means of the single lever A. A horizontal movement of lever A serves to rotate the segment gear F, which is in mesh with the rack teeth on the gear-shifting slide J, causing the slide to move endwise. A vertical or up and down movement of lever A imparts a similar sliding movement to the gear-shifting slide K, which has rack teeth meshing with the segment gear H. The range of movements imparted to the slides J and K by the horizontal and vertical movements of lever A permits any one of nine speeds to be selected.

When the handle A is moved horizontally, the housing B, in which the lever is pivoted at C, turns with the lever and the rod D. The teeth on gear E, secured to rod D, are a

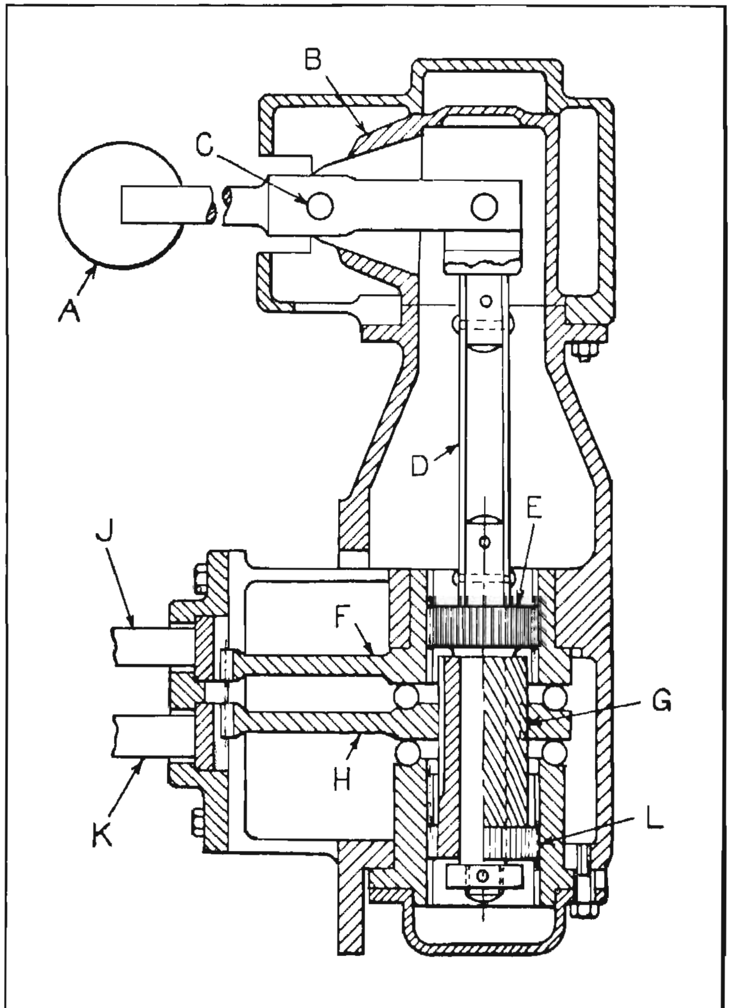


Fig. 2. Mechanism in which a Single Lever A Actuates Two Slides J and K that Control a Nine-speed Gear-box

sliding fit in internal teeth in the hub of the segment gear F. Thus a horizontal movement of lever A transmits a sliding movement to slide J through the rod D, gear E, and segment F. A vertical movement of lever A causes it to pivot on stud C, resulting in transmitting a vertical movement to rod D. The downward projecting end of gear E is a turning fit in the helical gear G; gear G has straight spur gear teeth at L meshing with teeth in the housing that prevent rotation of the helical gear. The vertical movement of lever A causes gear G to move vertically and impart a rotating motion to the segment H, which has internal helical teeth that are a sliding fit over the helical teeth on gear G.

Combination Differential and Speed Reducer.— The truck and tractor differential, Fig. 3, not only compensates for the variable speed of the rear wheels but has a second differential motion for obtaining two speeds at a ratio of 2 to 1; thus the pulling power of the car is increased at the low speed approximately 100 per cent. Both of these movements are enclosed in one casing.

In the sectional view of the rear axle assembly, the gears are shown enclosed in the housings A and B, which are mounted in ball bearings. A worm-gear C, clamped securely between these housings and driven from the engine by the worm, serves to drive the mechanism. The bevel gear E is locked by a pin to the housing A, while the gear F is free to turn in housing B and has a projecting hub on which the clutch G is held by a feather key.

By sliding this clutch to the right into engagement with the stationary member H, the gear F is prevented from rotating; and when the gear E is rotated, the pinions I, which are in constant mesh with bevel gears E and F, are advanced in the same direction as gear E. However, as these pinions are mounted on the four-armed spider J, their rolling action on the gears E and F will cause the spider to revolve at one-half the speed of the gear E.

The movement of the spider also carries the inner pin-

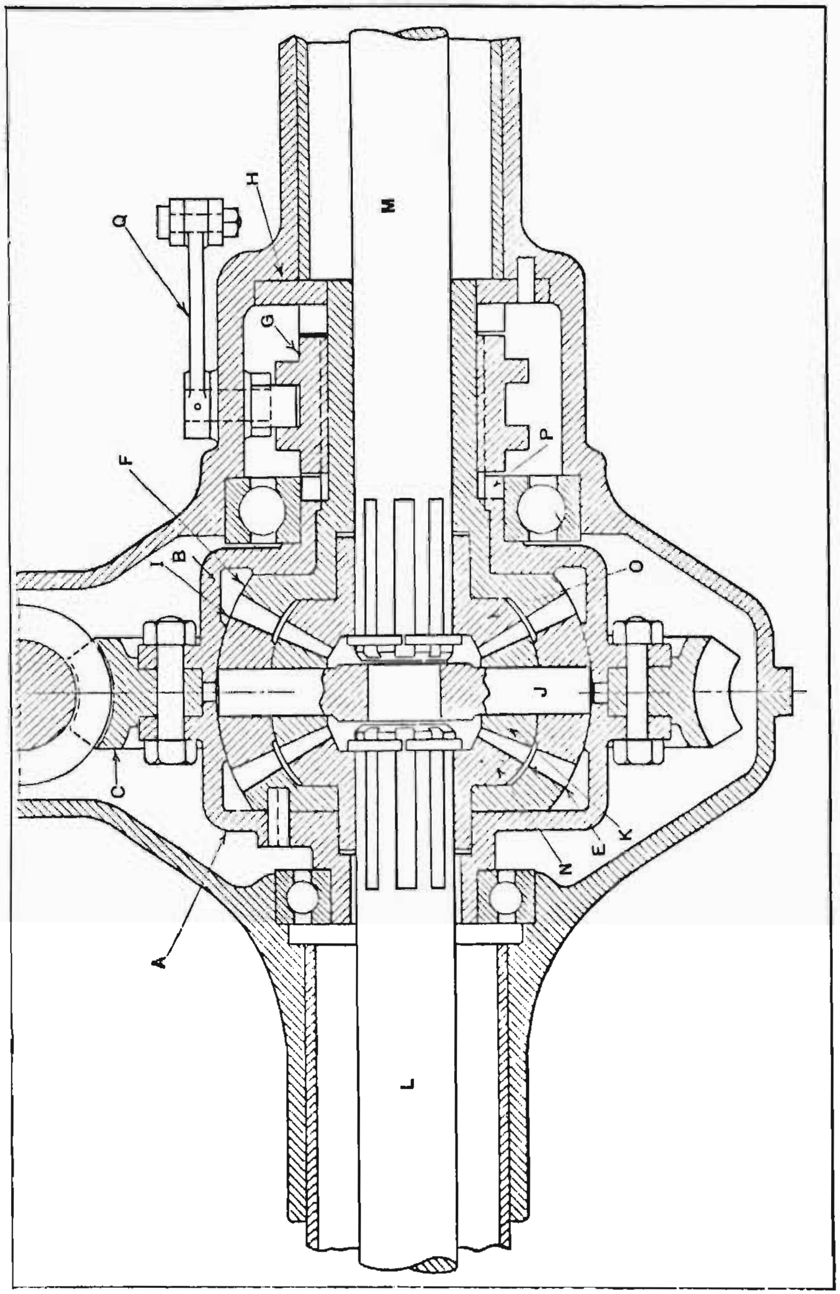


Fig. 3. Automobile Differential from which Two Speeds May be Obtained

ions K and the gears N and O around, driving the axles L and M. It is obvious that the gears N, K, and O form a differential similar to that found on most cars, permitting either axle to lag. When clutch G is moved to the left into engagement with the jaws P on the casing B, the entire assembly is keyed together and its action is the same as in an ordinary differential case. The spider J is free to float in the pinions K and I. The thrust of these pinions as they roll on the bevel gears is taken by the casings A and B, the spherical thrust seat being similar to that employed in a standard differential. The clutch G is operated by means of the lever Q, connected to a shifting lever near the driver's seat.

Compound Planetary Speed Reducer.—A compound planetary speed reducer, designed to give a large reduction in a unit of small size, is shown in Fig. 4. The gearing consists of a stationary gear F, doweled to housing D; a low-speed gear R, fastened to the low-speed shaft E; planetary pinions G and H; planetary gears L and M; and a high-speed pinion P, fastened to high-speed shaft Q.

Pinions G and gears M are carried on shafts J, while gears L and pinions H are carried on the sleeves K, the two assemblies being held on the planetary arm N. Pinions G and H, and gears F and R, have the same number of teeth; but gears L and M, although they are of the same pitch diameter, do not have the same number of teeth. In this case, L is a normal gear, while M has one tooth more than L, but is cut on the same pitch diameter.

The fact that gears M and L are both driven by the single pinion P makes possible the large speed ratio between the driver and driven shaft. It may be well to point out here that the ratio will be greatly decreased if pinion P is made with two sets of teeth in which one set is larger by a tooth than the other, and the gears L and M are made normal—that is, M is made to correspond with the lower half of the divided pinion P.

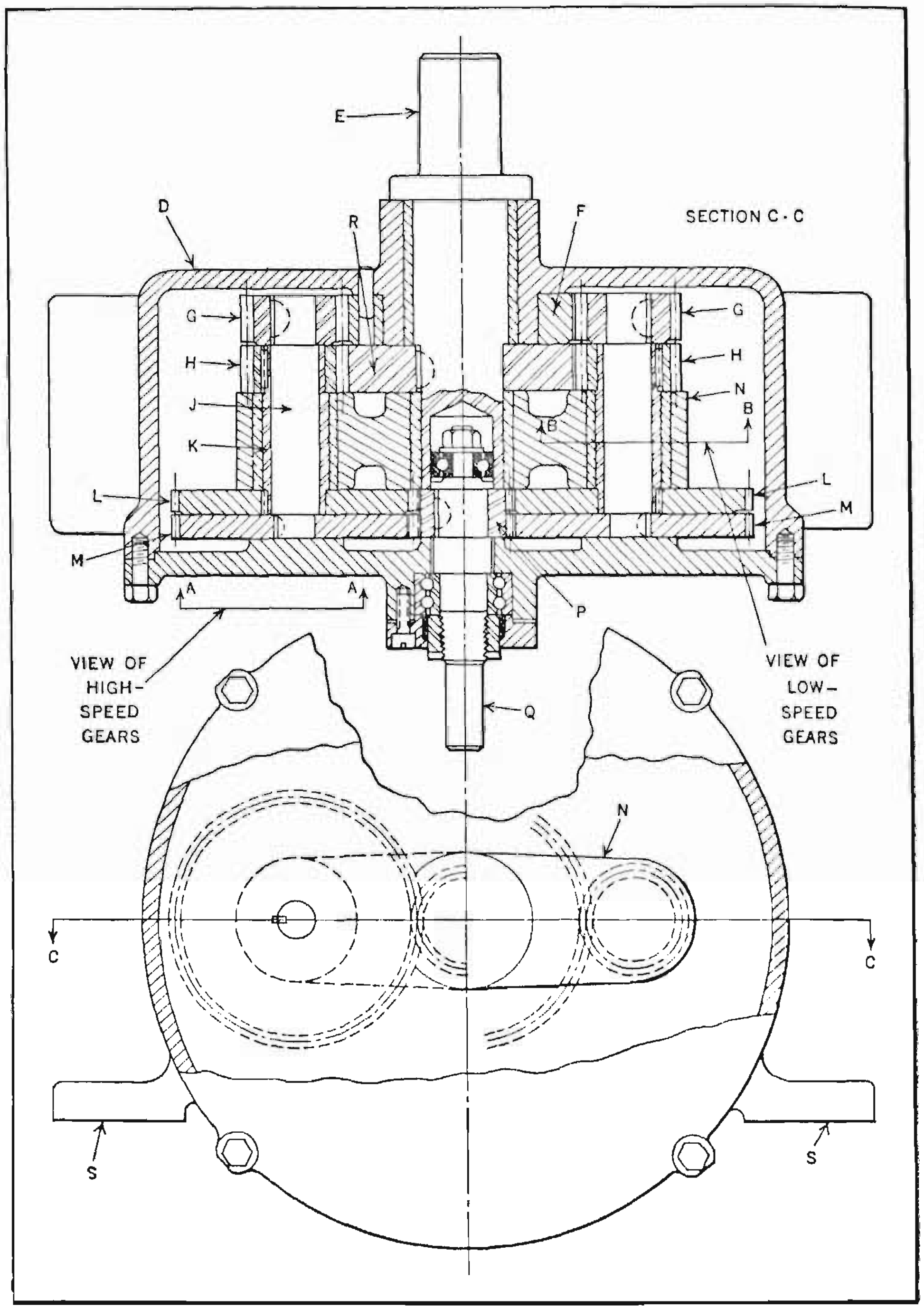


Fig. 4. High Ratio Speed Reduction Mechanism of Compact Design

Calculating the Speed Ratio.—The method of calculating the ratio of this kind of reducer is not very different from any other, although the fact that the movement goes through two sets of gears may make it confusing. An example will make the method clear.

Referring to the illustration, the number of teeth in the different gears of the mechanism are as follows: $F = 50$, $G = 20$, $H = 20$, $R = 50$, $L = 80$, $M = 81$, and $P = 32$.

First, assume that all the gears are locked tight and that the entire mechanism, case and all, is given one revolution. Thus, both shafts E and Q are given one revolution. Next, assume that arm N is held stationary, and that the case D is turned back one revolution. As this one revolution is made, we analyze the rotation of the various gears, noting first what happens to shaft Q and then to shaft E. The movement is added to or subtracted from the first revolution in each case, and the two results are set up as the ratio.

Equations can now be written from this information. Assuming that the first revolution was made in a clockwise direction, the second revolution is made in a counter-clockwise direction. It will be noted that when we turn the case back one revolution, while holding the arm still, the shafts revolve in a counter-clockwise direction; so we must subtract the calculated movement from the first revolution of both the driving and the driven shafts. Thus we have,

Movement of driver Q equals

$$1 - \frac{\overset{F}{50} \times \overset{M}{81}}{\underset{G}{20} \times \underset{P}{32}} = 1 - \frac{405}{64} = \frac{64 - 405}{64} = -\frac{341}{64}$$

and movement of driven shaft E equals

$$1 - \frac{\overset{F}{50} \times \overset{M}{81} \times \overset{P}{32} \times \overset{H}{20}}{\underset{G}{20} \times \underset{P}{32} \times \underset{L}{80} \times \underset{R}{50}} = 1 - \frac{81}{80} = \frac{80 - 81}{80} = -\frac{1}{80}$$

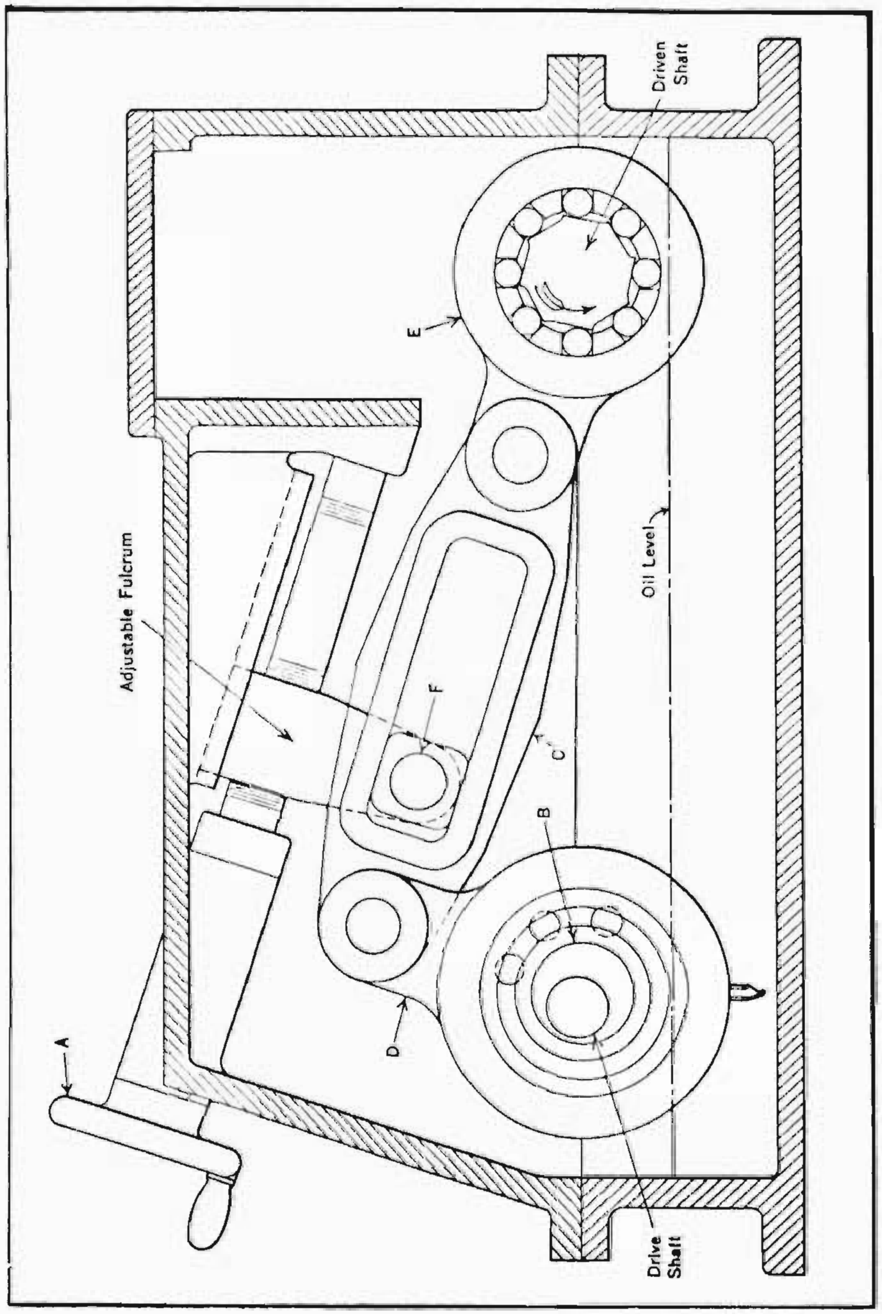


Fig. 5. Variable-speed Transmission in which the Driven Shaft is Rotated through the Successive Movements of a Series of Roller Clutches

Thus we have the ratio between the driving and driven shafts equals

$$\frac{341}{64} \div \frac{1}{80} = \frac{341 \times 80}{64} = \frac{426}{1}$$

It should be noticed that the equations for both Q and E are negative; hence their quotient is positive, which means that the driven shaft E runs in the same direction as the driver Q. If the positions of gears L and M are reversed, the same ratio is obtained, but with a negative sign, as will be found by working out the example as in the first case.

The effect obtained by making P in two pieces, with 31 teeth for the lower half to match the 81-tooth gear M, and 32 teeth in the upper half to match L with 80 teeth, will be to greatly reduce the ratio. Working out this example by the same method as was used for the first example shows the ratio to be 121.5 to 1.

The general construction of the speed reducer is shown quite clearly in the illustration. The low-speed shaft E runs in a bronze-bushed bearing, and the drive can be taken off to one side—that is, with a chain or gearing, if necessary. The high-speed shaft Q, in this case, was designed to be coupled directly to a motor. If it were necessary to drive shaft Q with a chain or gears, the double-row ball bearing would probably have to be split up into two bearings, one being arranged as in this design, and the other located close to the end of the shaft to take the radial load of the chain or gear. The method of mounting will naturally depend on conditions. In the application described, the case D was fastened to two channels running parallel with the shafts, legs S being provided for that purpose.

Gearless Variable-Speed Transmission.—In many drives, it is desirable to be able to shift from one speed to another without stopping the machine and also to be able to obtain any speed between the maximum and minimum. A design which meets these requirements is shown in Fig. 5. It is compact, and instead of using gears, chains, or belts for transmitting the rotary movement to the driven shaft, levers and roller clutches are employed.

In this particular model (built and patented by the Lenney Machine & Mfg. Co., Warren, Ohio), any speed between 15 and 150 revolutions per minute can be obtained with the motor running at 1750 revolutions per minute. The speeds are instantly changed by turning the handwheel indicated at *A*. On the drive shaft is mounted a series of eccentrics *B*. These eccentrics are connected to levers *C* by yokes *D*. Roller clutches on the driven shaft are connected to the levers by yokes *E*.

As the drive shaft rotates, the eccentrics impart an oscillating movement to the left-hand ends of the levers *C*; and as these levers are pivoted at *F*, their other ends will also oscillate and impart a rotary reciprocating movement to the roller clutches within the yokes *E*. Each reciprocating movement of the clutches will cause the driven shaft to rotate a fraction of a revolution; and as the eccentrics are spaced uniformly about the drive shaft, the impulses given the driven shaft will be successive and overlapping. In this way, a uniform rotary movement of the driven shaft is obtained.

The oscillating movement of the right-hand end of the links *C* determines the amount the driven shaft turns during each impulse, and this oscillating movement depends upon the position of the fulcrum *F* along the slot in the levers. For example, if the fulcrum is moved down toward the right by handwheel *A*, the reciprocating movement of the clutch will be shorter, and a longer time will be required to rotate the driven shaft. Obviously, the entire range of speeds is covered smoothly, enabling the mechanism to glide from one speed to another. Although not indicated, forward and reverse rotation of the driven shaft can be obtained by merely shifting a lever. This lever may also be shifted to neutral to stop the driven shaft.

Mechanism that Insures Changing Speeds According to Successive Gear Ratios.—An ingenious application of intermittent gears is incorporated in the mechanism shown in Fig. 6. This mechanism is designed for shifting change-gears axially in a metal-spinning machine. Provision is made for obtaining three different speeds and for shifting

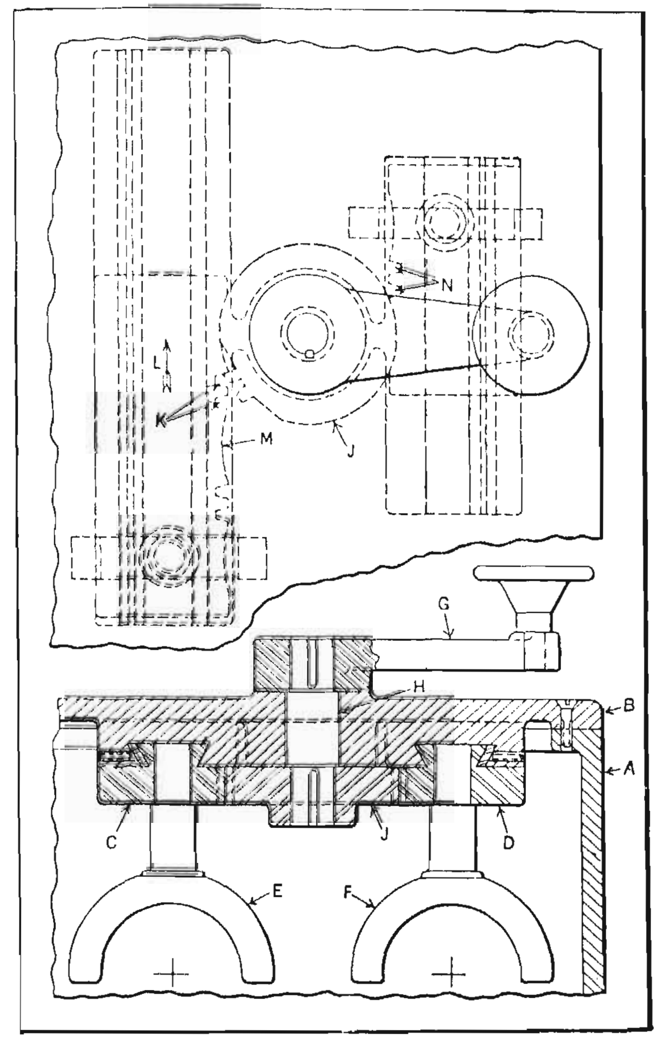


Fig. 6. Gear-shifting Mechanism that Insures Easy Changing of Gears in Order of Speed Ratios

the gears into the neutral position in the order of their ascending and descending ratios. Only one hand-lever is employed for manipulating the gears. The speeds are arranged in geometric progression and the provision for changing them in accordance with their ratios was made to avoid clashing of the gear teeth when changing speeds while the machine is in operation. With this arrangement, the pitch-line velocities of the gears to be engaged are so nearly the same that the teeth slide readily into mesh.

The gear-box is indicated at *A* and its cover at *B*. On the cover is mounted the entire gear-shifting mechanism, which consists essentially of the two slides *C* and *D* to which are attached the gear-shifting forks *E* and *F*; the hand-lever *G* keyed to shaft *H*; and the intermittent gear *J*, which is keyed to the shaft and engages the slides *C* and *D*.

With the hand-lever in the position shown, the change-gears (not shown) are in neutral. By rotating this lever in a clockwise direction, the two teeth in pinion *J* engage the tooth spaces *K* in slide *C*, causing the latter to move in the direction of arrow *L*. The movement of slide *C* continues until fork *E* slides the corresponding change-gear into mesh for imparting the lowest speed to the machine spindle. At this point, the cylindrical part of the pinion engages the corresponding depression *M* in the slide, locking slide *C* in a stationary position. To obtain the next higher spindle speed, the rotary movement of the lever is continued until the teeth in gear *J* engage the tooth spaces *N* in slide *D*. Up to this point, this slide has been locked in a stationary position by the cylindrical part of the pinion *J*.

As the lever continues to rotate, slide *D* is moved in a direction opposite that indicated by arrow *L*, causing the fork *F* to shift another gear into mesh and thus obtain the second speed. The next two speeds are obtained in like manner—that is, by continuing the rotary movement of lever *G* in a clockwise direction. Graduation marks on the

gear-box cover and an arrow attached to the lever hub indicate the positions of the lever for the various speeds, as well as the position when the gears are in neutral. In order to shift the gear to neutral when the lever is in the "high speed" position, the lever must be swung through an angle of approximately 450 degrees. However, owing to the successive arrangement of the gears, their action in shifting is so smooth that the lever can be shifted very rapidly between these two points.

Speed-Changing Transmission of Hydraulic Type.—Transmissions for cars driven by internal combustion engines must be designed to provide the required speed changes, a reverse or backward motion, and a neutral position to permit of stopping the car while the motor continues to run. The transmission to be described provides unlimited speeds from zero to the high or direct drive, and all speed changes, as well as the neutral position and the reverse, are controlled by a single foot-pedal, there being no gear-shifting lever.

This transmission has been tested under road conditions as explained later. Fig. 7 shows a cross-sectional view, and Fig. 8 an end view of the "fluid clutch" or hydraulic part of the mechanism.

The two opposed cylinders *A* are attached to the engine flywheel, and another pair of opposed cylinders *B* is attached to shaft *D*, which is offset, or eccentrically located, relative to the main center line *x-x*. The four pistons *C*, Fig. 8, located in the four cylinders, are in the form of a one-piece cross with arms of equal length, located 90 degrees apart; consequently, the two pairs of cylinders always rotate together and one pair is held at right angles to the other.

Rotation of the cylinders and pistons may or may not be accompanied by a reciprocating motion of the pistons in the cylinders. Such a motion will occur, due to the offset position of cylinders *B* and shaft *D*, unless the pistons are

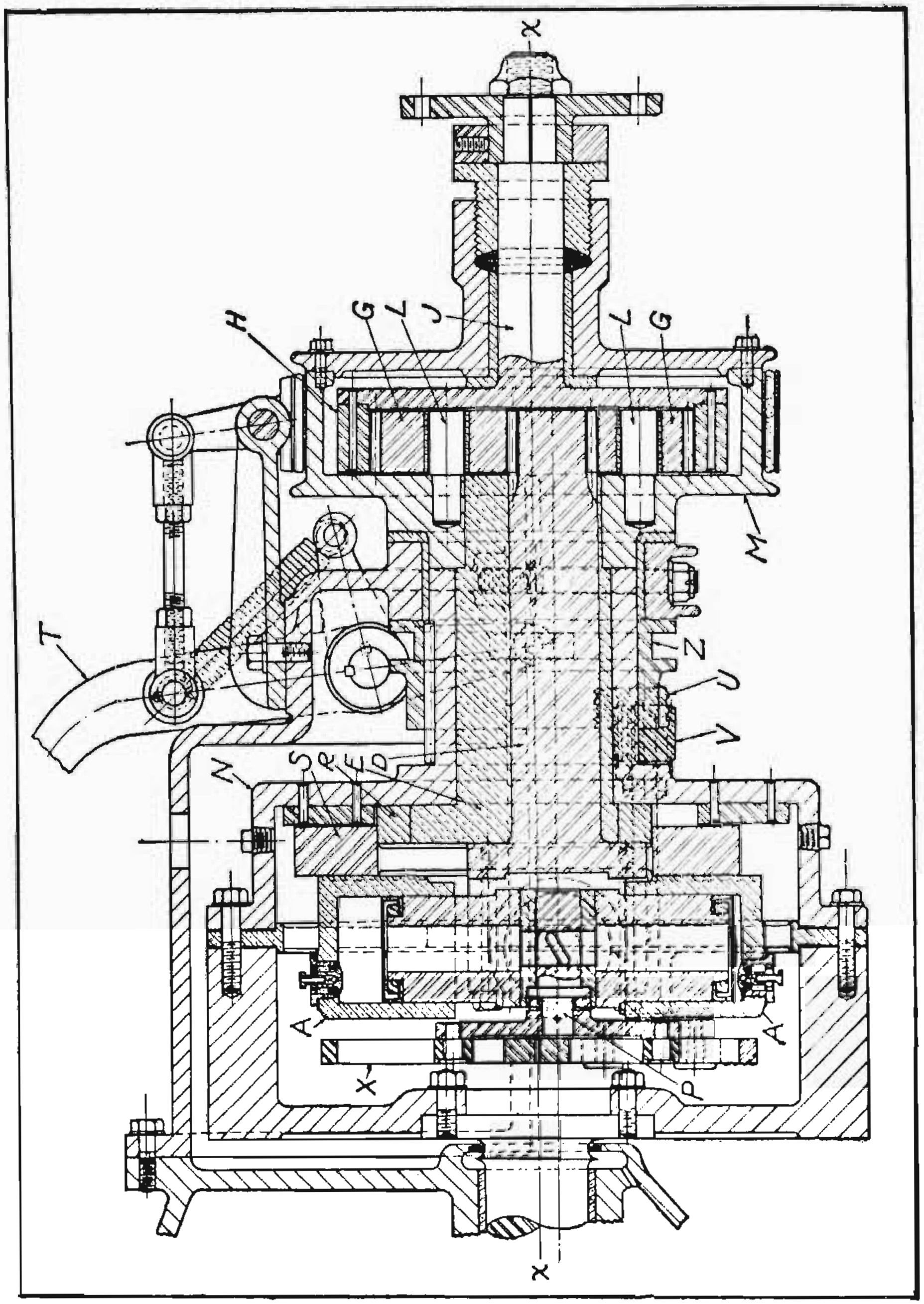


Fig. 7. Sectional View of Transmission, which is Controlled Entirely by a Foot-pedal

locked hydraulically so that movement is impossible. When the pistons are locked relative to the cylinders, the direct or high-speed drive is obtained.

How Direct Drive is Obtained.—The clearance spaces in all cylinders and the holes through the piston arms are always filled with lubricating oil. If the four-way plug valve *P*, Fig. 7, located at the intersection of the holes in the pistons, is closed so that oil is trapped in each cylinder, any movement of the pistons relative to the cylinders is prevented. (The control of this valve by foot-lever *T* will be explained later.)

When the pistons are thus locked, the various parts of the transmission from the motor shaft to the rear transmission shaft *J* rotate as a unit. These parts include, in addition to the cylinders, pistons, and eccentric shaft *D*, the eccentric bushing *E* (which is keyed to casing *M* and is free to turn with it in the main casing *N*), pinions *G* in casing *M* (which carries the pinion studs), internal gear *H*, and shaft *J* to which it is connected.

During this direct drive, the axis of eccentric shaft *D* rotates around the common axis x–x of the transmission, but shaft *D* does not turn about its own axis. The eccentric bushing *E*, in which shaft *D* is free to revolve, is forced to rotate about axis x–x by shaft *D*. While driving direct, pinions *G* cannot rotate about their own axes, because on one side they are in mesh with the driving pinion of *D*, which is locked against rotation about its axis, and on the other side they mesh with internal gear *H*, which offers resistance to rotation due to the fact that it is coupled indirectly to the wheels of the car. The result is that pinions *G* merely act as locking members or driving keys between the pinion of shaft *D* and internal gear *H*.

Reversing the Direction of Rotation.—The action of the mechanism during the intermediate speed changes and when in neutral will perhaps be easier to understand when the movement during the reverse drive is described. Valve

P, which was tightly closed for the direct drive, is wide open for reversal, thus allowing the oil to flow freely and permitting the pistons to reciprocate. The result is that eccentric shaft D now rotates about its own axis, but it also has a planetary movement about axis x–x, because when valve P is either partly or wide open, the eccentric bushing E, being keyed to casing M, is free to rotate when pinion gears G revolve inside internal gear H. When foot-pedal T is pushed down to the reverse position, it first opens valve P and then locks gear-case M and eccentric bushing E against rotation by gripping the gear-case with an external brake-band. The latter action is so timed that it does not occur until valve P is wide open and foot-pedal T controls the action of the brake, as well as that of the valve. With casing M and eccentric bushing E held stationary, motion from the motor is transmitted through cylinders A, the pistons, shaft D, pinions G (which now rotate about pins fixed in casing M), reversing internal gear H and shaft J.

Transmission in the Neutral Position.—If foot-pedal T, Fig. 7, is allowed to rise from the reverse to the neutral position, casing M and eccentric bushing E will remain released from the brake-band and valve P will be wide open. The revolving cylinders A then rotate cylinders B and eccentric shaft D, which merely turns about its own axis and axis x–x. The rotation of shaft D, however, is not transmitted to internal gear H and shaft J, because gear casing M and eccentric bushing E now are free to turn. The result is that pinions G merely rotate planetary fashion inside of gear H, because casing M and eccentric bushing E offer only slight frictional resistance to rotation, whereas internal gear H is coupled indirectly to the rear wheels of the car.

Action of Mechanism During Intermediate Speed Changes.—As foot-pedal T is allowed to rise from the neutral position toward the high-speed position, valve P is gradually closed. As it closes, there is a proportionate increase in the resistance to the flow of oil between the four cylinders; moreover, this resistance to the oil flow causes a corresponding increase in the resistance to the rotation of D about its axis until, finally, when valve P is completely closed, there is no such rotation, shaft D merely turning about axis x–x. However, when valve P is partially opened,

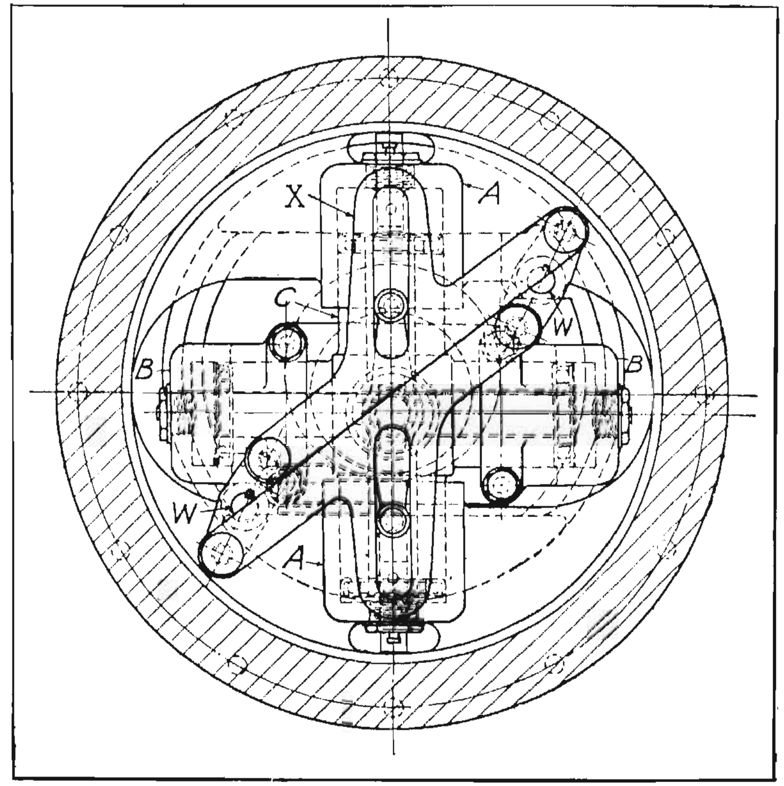


Fig. 8. End View, Showing Arrangement of the Cylinders and Pistons of the "Fluid Clutch"

shaft D has a planetary movement, there being rotation about its own axis and about axis x–x. The movement about axis x–x is, of course, accompanied by rotation of eccentric bushing E, which is forced to rotate by the studs in casing M when its pinions G revolve idly around H in the neutral position.

Now when shaft *D* begins to turn about axis *x–x* at an increased rate due to increasing the resistance of the oil, rotary motion will be transmitted to gear *H* at a rate depending upon the planetary movement of shaft *D* and its rotation about its own axis. The planetary movement increases and the rotation about the axis diminishes as valve *P* is closing. Finally, when the valve is entirely closed, thus preventing all rotation of shaft *D* about its own axis, pinions *G* act something like fixed keys that connect the pinion of shaft *D* with internal gear *H*, as previously mentioned.

The movement of foot-pedal *T* is transmitted to a sliding collar *Z*, which, in turn, slides a helical gear segment *U* that is continually in mesh with two helical pinions *V* keyed on short shafts *W* (Fig. 8). A quarter turn of these shafts can be obtained easily, as the gear segment slides in an axial direction while guided by a key to prevent rotation. This rotary motion of the pinion shafts is transmitted to the inside of the casing through oil-tight bearings. The elongated slots in the connecting link *X* allow space for the valve lever rolls to operate in when the pistons are reciprocating in the cylinders.

The car responds to the slightest touch of the foot-pedal without any jarring action or shocks. No trouble has been experienced from excessive heat generated by the compression of the fluid, as there is sufficient radiating surface; this has been proved by tests on the road. The main casing *N* of the transmission is made oil-tight and should be kept entirely filled with some good quality lubricating oil.

Safety valves are provided at the head of each cylinder, and they are set to blow at 1500 pounds per square inch. In conjunction with each safety valve, there is a sensitive one-way automatic check valve opening toward the inside of the cylinders. If a vacuum is created in the cylinders by the pistons due to insufficient oil as the result of leakage, the proper amount of oil will automatically be restored

through the check valves. This is an important provision, since the cylinders must always be completely filled to obtain a smooth, even starting torque.

It is necessary to have two counterbalancing plates *R* and *S*, Fig. 7. One is used to counterbalance the two offset cylinders *B*, and the other to counterbalance the pistons. The plate *R* is forced to move in opposition to the cylinders by eccentric bushing *E*, and the other plate *S* is driven by plate *R*, but it is keyed so as to slide in direct opposition to the pistons.

This transmission has been applied to a car and subjected to various driving conditions during 1000 miles of road tests. While there is doubtless considerable sliding friction in the design shown, no difficulties have been experienced from overheating, although continuous runs up to forty miles per hour have been maintained for three hours. Nevertheless, the mechanical efficiency can be increased by the use of ball or roller bearings, especially between the eccentric bushing and the main casing. Other changes may also be made subsequently in the construction of this transmission.

Automatic Speed Compensating Mechanism.—In recording sound on a sensitized motion picture film, it is necessary that the film have an absolutely uniform linear movement through a microscopic light beam which photographs on it electrical vibrations coming from a microphone. This film is pierced at the time it is manufactured with two rows of very accurately spaced sprocket holes. The pitch of this spacing changes from film shrinkage—when the film is aged by being stored, or otherwise—often as much as one-half per cent; that is, five feet per thousand feet of film. When new film is used for recording sound, sometimes there is no shrinkage at all. Most frequently, however, it will be found that the shrinkage amounts to about one-fourth of one per cent.

Since an absolutely uniform motion of the film is neces-

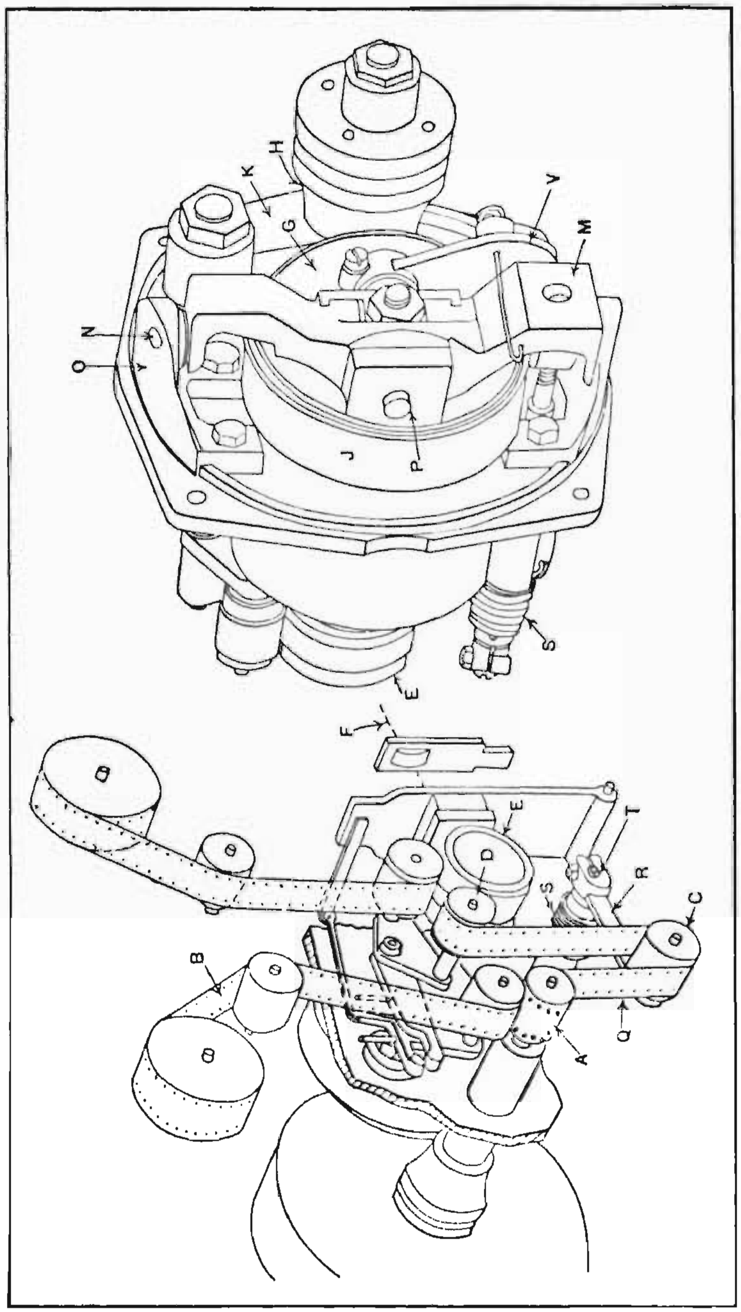


Fig. 9. Automatic and Precise Speed Compensating Mechanism for Maintaining Synchronism between Sound Record and Moving Picture Film Movement

sary for perfect sound recording, means must be provided to compensate for this shrinkage and thereby maintain synchronism between the sound record and sprocket holes, because the sound track record must synchronize with the pictures being taken by one or more cameras operated in synchronism with one or more sound recorders.

Referring to Fig. 9 (view at the left), the sprocket *A* causes the film *B* to travel in synchronism with a film passing through a corresponding camera, because the camera and sound recorder are actuated by synchronized electric motors. The film, after passing under the control roller *C* and over the roller *D*, snugly engages the drum *E* and is thereby caused to pass, with an absolutely uniform motion, through a microscopic oscillating light beam coming from a galvanometer, as indicated by the dotted line *F*.

Evidently, if the film has shrunk considerably, the periphery speed of the drum *E* must be correspondingly reduced in such a precise manner as to maintain the required uniformity in rate of film travel through the light beam. That is, the speed compensating mechanism about to be described must automatically select and maintain a speed ratio between the sprocket *A* and the drum *E* with a high degree of precision for the purpose of maintaining this uniform speed.

The perspective view at the right in Fig. 9 shows the mechanism as seen from the side opposite to that shown by the view at the left. Sectional views are shown in Fig. 10. By referring to these illustrations, it will be seen that control wheel *G* transmits motion from the driver *H* to the driven wheel *J* and drum *E*. The frictional contact between these wheels is maintained by a spring (not shown), which presses the driver *H* and its hinged bearing arm *K* toward the control wheel *G*.

The control wheel is journaled in a gimbal mounting formed by the members *L*, *M*, *N*, *O*, and *P*. The position of the control wheel is governed by the film loop *Q* through

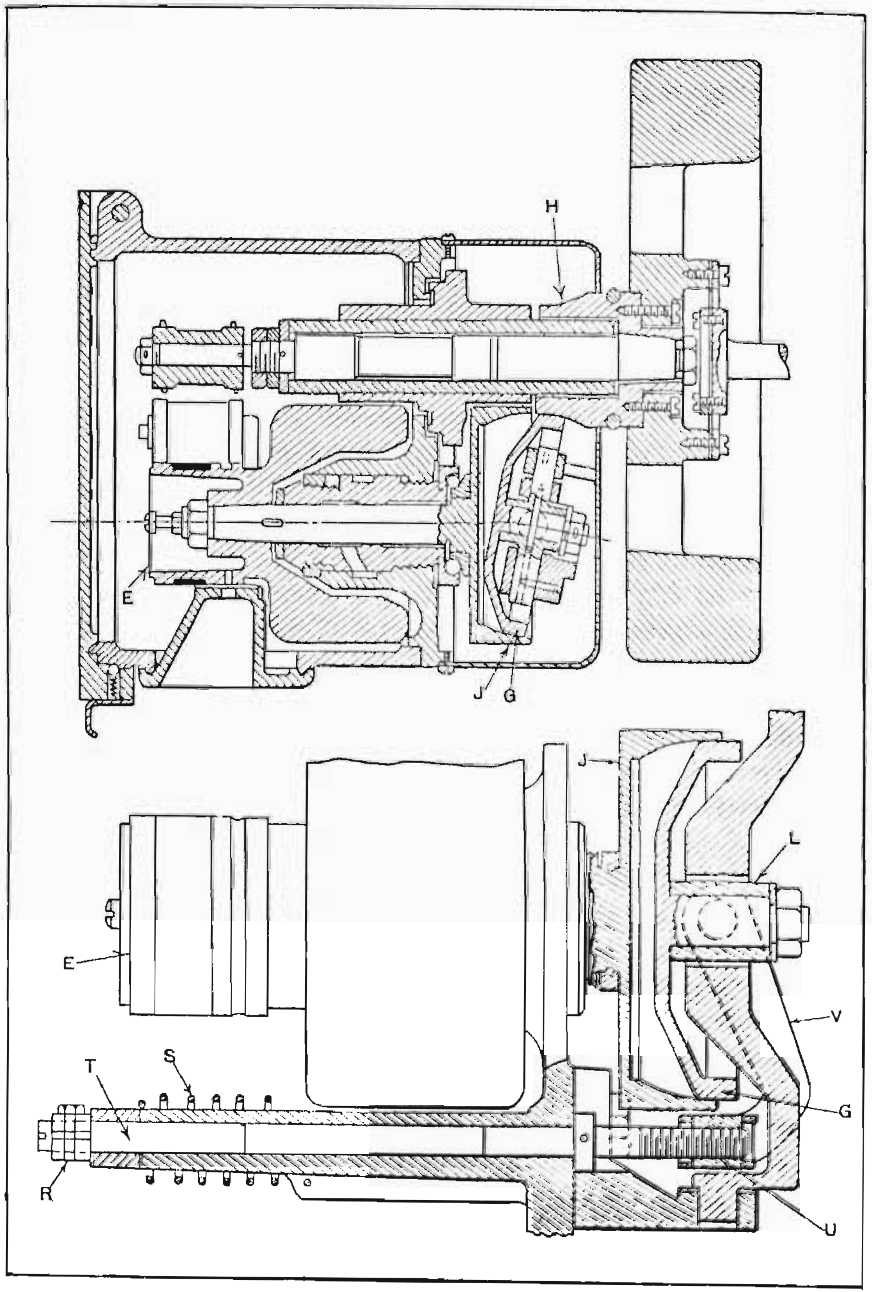


Fig. 10. Sectional Views, Showing how Minute Speed Variations are Obtained

the control roller C, lever arm R, spring S, control screw shaft T, nut U, and a swivel connecting with lever V. This lever is keyed to the trunnion P formed on the journal bearing L that supports the journal which is part of the control wheel G.

Owing to this construction, a very slight change (1/64 inch) in the position of the control roller C causes a corresponding change in the position of the control wheel G with respect to the driver H and driven member J. The friction contact surfaces between G and H are ground spherical, with great precision. The periphery of G is given a slight crowning to provide an actual contact friction driving surface of approximately 1/8 inch width. The slightest movement of the lever V tilts the control wheel G about its contact points with the wheels H and J, so that it automatically engages a new friction path that gives a corresponding change in speed ratio. Any slight wear or lost motion that may occur is taken up by means of springs.

In the apparatus, the arm R is 2 inches long, and a 1/64 inch movement of the control roller C gives approximately 1/800 turn of the screw shaft T, which has a double thread of 1/7 inch lead; consequently, the axial movement of the nut U is approximately 0.0002 inch. This slight movement of the nut automatically causes a new selection of speed ratio, thereby giving an extremely high degree of refinement in speed control. A very short time after starting under normal operating conditions, when running uniformly shrunk film, there is no appreciable change in position of the control roller C and its related parts.

Many sound recorders embodying the control mechanism described in the foregoing have been in successful operation for some time. The original model was developed in the Schenectady engineering laboratory of the General Electric Co. and operated daily over a period of two years, continually making perfect sound records without the least trouble.

CHAPTER XI

SPECIAL TRANSMISSIONS AND OVER-RUNNING CLUTCHES

The transmissions described in the preceding chapter are designed to permit changing the speed of the driven member relative to the speed of the driver. Even when speed changes are not required, some special type of transmission or connection between the driving and driven members may be necessary because of their respective positions or to provide for some other operating requirements. Interesting examples of these special transmissions will be found in this chapter.

One-Way Rotation with Reversing Driver.— The purpose of the mechanism shown in Fig. 1 is to obtain a one-way rotation for the driven shaft regardless of the direction of rotation of the driver. In other words, the driver *B* may at any time, at the will of the operator, reverse its rotation without changing the rotation of the driven sprocket *H*, and this result has been accomplished by a very simple mechanism consisting of few parts, as the illustrations show.

Gear *C* and sprockets *B* and *F* are mounted on and keyed to the sleeve, which is bushed and revolves freely about stud shaft *A*. Sprocket *B* is driven by a reversing motor. Spur gear *D* is driven by spur gear *C*. Gear *D*, sprocket *H* and sprocket *G* all revolve freely on stud *E*. Sprocket *F* drives sprocket *G*. Since sprockets *F* and *G* are connected by chain, they rotate, of course, in the same direction. Gear *D* always revolves in an opposite direction to that of sprocket *G*.

In the illustration the clutch teeth of sprocket *H* are

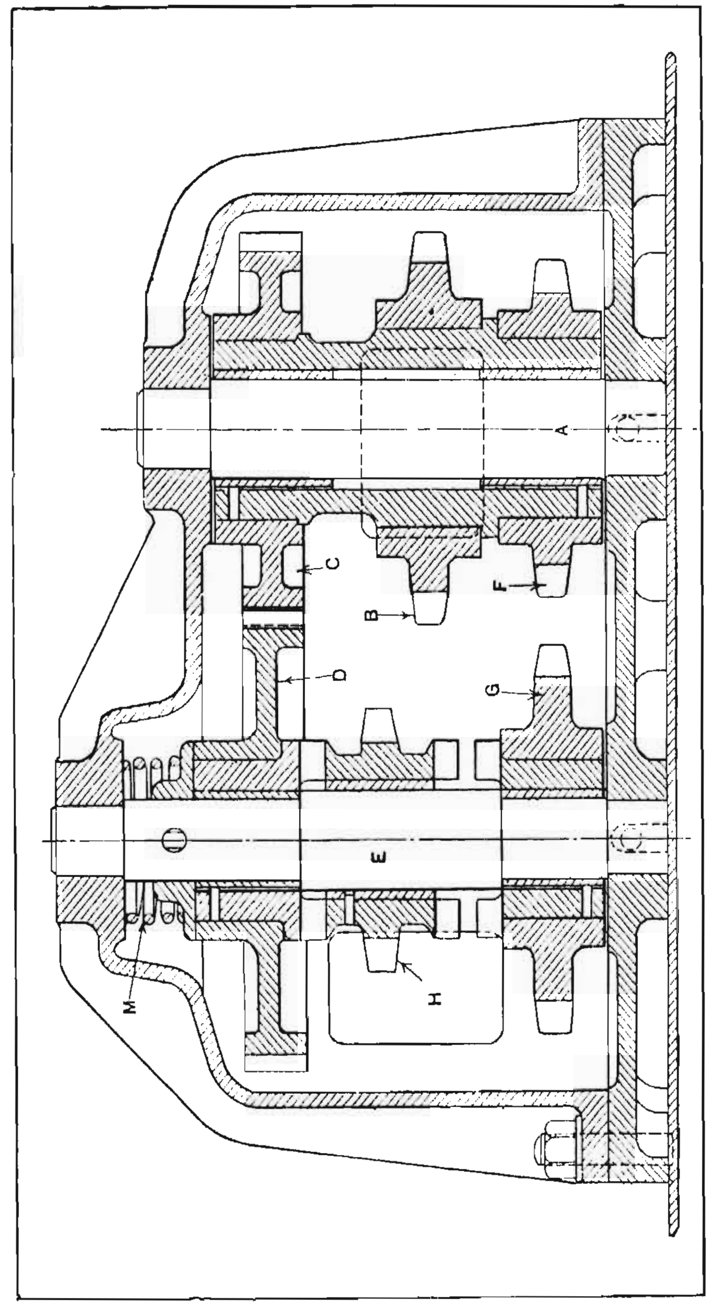


Fig. 1. One-way Transmission so Arranged that Reversal of Driver does not Change Rotation of Driven Sprocket

shown in mesh with the clutch teeth of gear *D*. Assume that gear *D* and sprocket *H* are revolving in a clockwise direction and that sprocket *H* is to continue rotating in that direction. Suppose now that the rotation of sprocket *B* is suddenly reversed to a clockwise direction, thus causing gear *D* to revolve counter-clockwise. This change of motion, owing to the shape of the clutch teeth, will throw sprocket *H* over into engagement with sprocket *G*, and the latter, which, of course, reversed its direction at the same

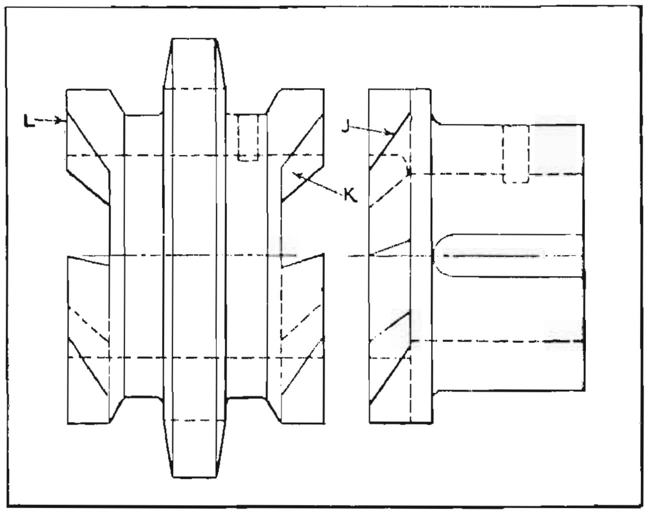


Fig. 2. Driven Sprocket of One-way Transmission and Form of Clutch Teeth Used

time as shaft *B* will now be rotating clockwise, which is the direction desired for sprocket *H*.

Why a reversal of motion causes sprocket *H* to shift from engagement with the gear over into engagement with the sprocket will be apparent by referring to the detail view Fig. 2, which shows this sprocket and the gear clutch. When this clutch (which is a duplicate of the sprocket clutch) is driving the sprocket, the under-cut surfaces of the clutch teeth are in engagement, but when the rotation of the driving clutch member is reversed, the tapering sur-

faces *J* act against the corresponding tapers *K* on the sprocket, thus exerting a wedging or cam action which thrusts the clutch over into engagement with the opposite side.

When this shifting of sprocket *H* occurs, it is evident that the tops or lands *L* of the clutch teeth might strike the tops of the teeth, say, on sprocket *G* instead of entering the spaces between the teeth. If this should occur, there would be a serious wedging action between gear *D* and sprocket *G*, because the width of the clutch part of sprocket *H* is somewhat greater than the clearance space between the clutches on gear *D* and sprocket *G*. If such wedging action should occur, however, it would be relieved instantly by the lateral movement of gear *D*, which is free to shift against the action of spring *M*. This relieving movement would be followed quickly by the return of *D* to its normal position as the clutch on sprocket *H* snaps into place. The machine on which this device is used consists of two conveyors driven by one reversing motor. One conveyor is reversed at the will of the operator, and the other must travel in one direction only. This mechanism has been fully covered by U. S. letters patent.

Gearless Transmission for Angular Drives.—An unusual form of transmission for shafts located at an angle is shown by the diagram, Fig. 3, which includes a side view and an end view. Motion is transmitted from the driving to the driven shaft through rods which are bent to conform to the angle between the shafts. These rods are located in holes equally spaced around a circle, and they are free to slide in and out as the shafts revolve. This type of drive is especially suitable where quiet operation at high speeds is essential, but it is only recommended for light duty.

The operation of this transmission will be apparent by following the action of one rod during a revolution. If we assume that driving shaft *A* is revolving as indicated by the arrow, then driven shaft *B* will rotate counter-clockwise. As shaft *A* turns one-half revolution, rod *C*, shown in the inner and most effective driving position, slides out of both shafts *A* and *B* during the first half revolution, and rod *C* will then be at the top; then during the remaining half, this rod *C* slides inward until it again reaches the innermost position shown in the illustration. In the meantime, the other rods have, of course, passed through the same cycle of movements, all rods successively sliding inward and outward.

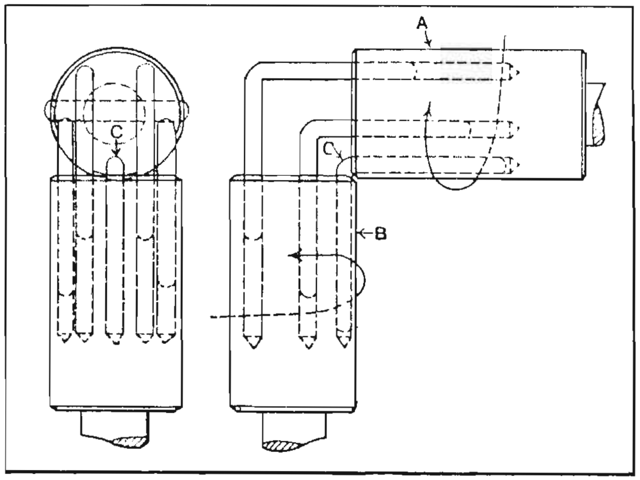


Fig. 3. Gearless Transmission Consisting of Shafts Connected by Rods which Slide in and out as the Shafts Revolve

Although this transmission is an old one, many mechanics are skeptical about its operation; however, it is not only practicable, but has proved satisfactory for various applications, when the drive is for shafts located permanently at a given angle. Although the illustration shows a right-angle transmission, this drive can be applied also to shafts located at any intermediate angle between 0 and 90 degrees.

One application that proved successful was on a special multiple-spindle drilling machine for drilling meter cases. This machine had between thirty and forty spindles

equipped with small drills which revolved at 1500 to 1800 revolutions per minute. This transmission was used to replace universal joints consisting of forked ends, each of which was pivoted by means of screws to a connecting block. These universal joints rapidly deteriorated, but the sliding rod transmission proved durable and quiet.

In making this transmission, it is essential to have the holes for a given rod located accurately in the same relative positions in each shaft; all holes must be equally spaced both in radial and circumferential directions. The holes in each shaft must also be parallel to each other, and each rod should be bent to the angle at which the shafts are to be located. If the holes drilled in the ends of the shafts have "blind" or closed ends there ought to be a small vent hole at the bottom of each rod hole for the escape of air compressed by the pumping action of the rods. These holes are also useful for oiling. To avoid "blind" holes, the shafts may have enlarged ends with holes extending clear through the enlarged part or shoulder. This transmission may be provided with a central rod, located in line with the axis of each shaft and provided with a circular groove at each end for a cross pin to permit rotation of the shaft about the rod, the central rod simply acting as a retaining device for shipping or handling purposes.

Changing Relative Positions of Two Revolving Shafts.— The relative positions of two revolving shafts may be varied while in operation by the mechanism shown in Fig. 4. The speed ratio between shafts *A* and *B* is constant, but the relative rotative positions of the two shafts can be varied by means of the worm *H* and the worm-gear *M*. This mechanism is used in a wire-forming machine on which the timing of one section of the machine must be changed relative to the timing of another section.

The driving shaft *B*, supported by bearings *D* and *E*, carries the gear *J*, which is keyed to it. Bracket *F* is free on shaft *B*, and carries the stud *N*, which supports gears *I* and K. The two latter gears are free to rotate as a unit. Gear L is keyed to shaft A, which is supported by bearing C. Each of the gears K and L have 18 teeth, while gears I and J have 24 and 12 teeth, respectively.

Power is transmitted from shaft B through gears J, I, K, and L to shaft A at a 2 to 1 ratio in the same direction. Worm H is keyed to shaft G, which is supported on the upper end of bearing E. Shaft G carries a handwheel (not

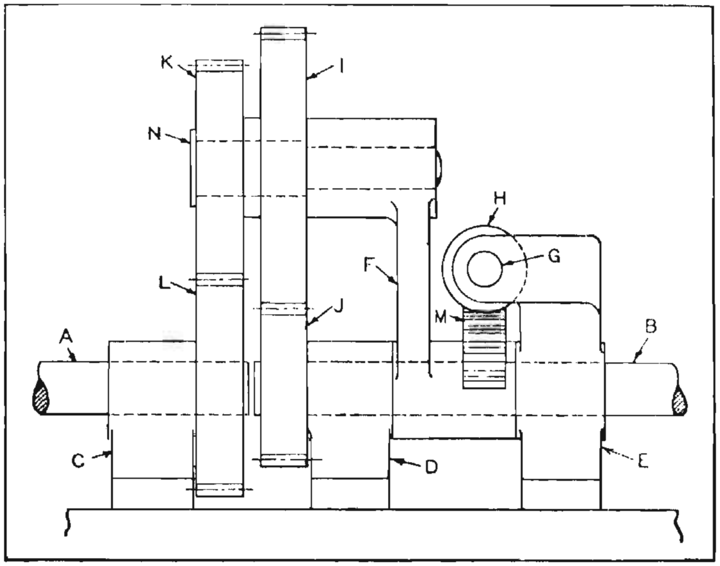


Fig. 4. Mechanism for Varying Relative Rotative Positions of Driving and Driven Shafts

shown) on one end, by means of which the worm is rotated. Worm H meshes with the worm-gear M on bracket F, causing the latter to rotate on shaft B when the handwheel on shaft G is turned. In Fig. 5, the dotted outlines of bracket F and gear I show the rotative movement around shaft B produced by the action of worm H and worm-gear M.

A better understanding of how the change of timing between shafts A and B is accomplished may be had by assuming these shafts to be stationary. Then, as bracket F is caused to rotate on shaft B as an axis, gear I, meshing

with gear J, will be rotated on stud N as an axis in the ratio of 2 to 1. As the axes of shafts A and B coincide, gear K rotates around shaft A.

If gear K were independent of gear I it would revolve on stud N as an axis in the ratio of 1 to 1 with shaft A, but as gears I and K are fixed together, and must revolve as a unit at a ratio of 2 to 1 with shaft B, gear L will be revolved in the reverse direction. The ratio of rotation between

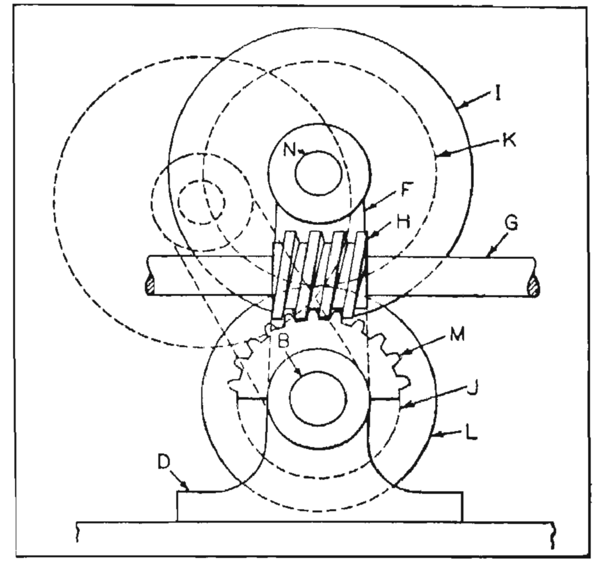


Fig. 5. End View of Timing Mechanism Illustrated in Fig. 4

gears K and L will still be 1 to 1, but each of the gears will revolve one-half turn in opposite directions. As gear L is keyed to shaft A, the latter must revolve with it. Under actual operating conditions, the effect is merely that of a gear train until the handwheel is turned, at which time shaft A is advanced or retarded as the bracket F is rotated forward or backward.

Changing Angular Velocity of Driven Member Twice During Each Revolution.—The velocity-changing mechanism to be described is incorporated in a hat-finishing machine. The finishing of the hat is done by a pad faced with sandpaper. This pad automatically travels from the top or center of the hat to the band while the hat is being revolved. An automatically controlled oscillating movement is also imparted to the finishing pad. The oval rotary motion mechanism referred to is required in order to keep the work in proper contact with the finishing pad.

Now, in order to finish the hat evenly all over, it is necessary to vary the angular velocity of the chuck on which the hat is mounted. If the spindle were revolved at a uniform speed, the front and back portions of the crown would be over-finished and the sides under-finished as a result of the difference in surface speed caused by the oval shape of the work. It is the purpose of the mechanism shown in Fig. 6 to increase the speed of rotation of the spindle as the front and back surfaces pass under the finishing pad.

As the oval head revolves, the quarter sections between the side and end portions must lift the working pad 3/4 inch in one-quarter revolution, thus increasing the pressure on the pad. As it passes from the end portion to the lower side, the pad drops 3/4 inch in one-quarter revolution, and this falling action gives less pressure on that portion. The oval head mechanism so acts that all portions of the section of the body with which the pad is in contact are raised to the same plane.

The same oval shape condition makes necessary the use of the differential angular velocity mechanism. The curvature of the end portions is sharper and the contact area of the pad on the hat is, therefore, less on these portions and the intensity of pressure greater. The side portions, being flatter, are subjected to less pressure by the pad. This condition would result in cutting the ends faster than the sides if the differential angular mechanism did not move the ends more quickly under the pad, the change in velocity being adjustable to suit the shape of hat.

The block that carries the hat is revolved by the spindle

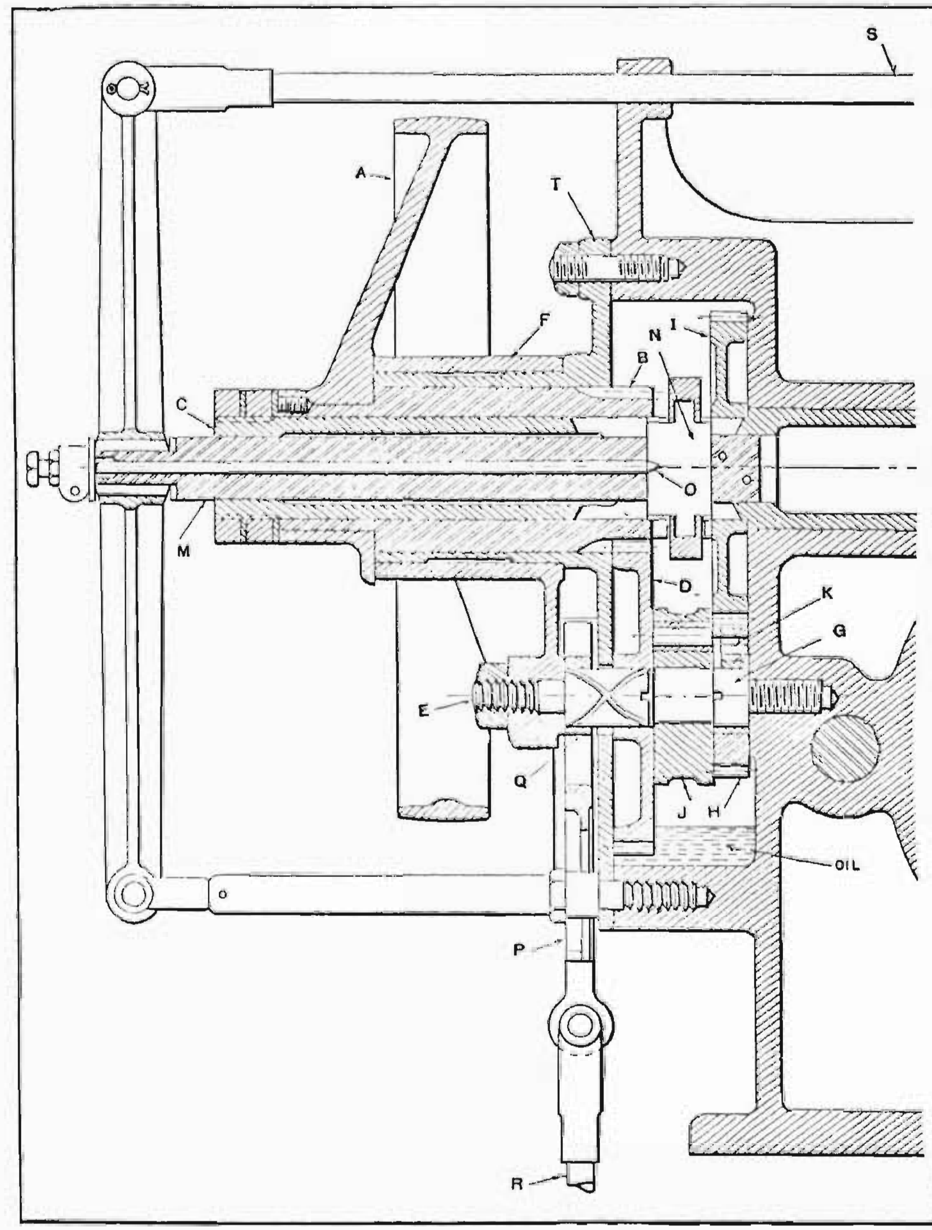


Fig. 6. Mechanism which Increases and Decreases Angular Velocity of a Shaft Twice During Each Revolution

C which extends through the machine frame at the right. By means of the control lever *S*, the spindle *C* can be connected directly with the driving pulley *A* so that it will run at the same speed, which might be required in finishing the band of a cylindrical or perfectly round hat. By moving the control rod *S* in the opposite direction, the variable velocity mechanism is brought into action. This mechanism causes spindle *C* to make one complete revolution to every four revolutions of the driving pulley *A*.

By means of the velocity-changing mechanism, the angular velocity of spindle *C* can be increased and decreased twice during one complete revolution. This change in velocity is synchronized with the oval motion mechanism, so that a uniform finishing effect is obtained. The amount of variation in the velocity is controlled by the cam-operated plate *P* which is actuated by the same cam that controls the oval motion mechanism.

Operation of Mechanism for Changing Angular Velocity.— The operation of the mechanism may be described by following the drive through from the pulley *A* to the spindle *C*, Fig. 6. Passing through a slot or opening in the push-rod *M* is a clutch finger *N* which is securely locked in the push-rod by the pointed rod *O*. Corresponding with the clutch finger opening in the push-rod are similar openings in the spindle *C* through which the clutch finger *N* freely passes.

When the push-rod *M* is moved backward by the control rod *S*, the clutch finger *N* engages the clutch face of the pinion *B*. The spindle *C*, through this contact, is driven at the same speed and in the same direction as the pinion *B* and its driving pulley *A*. When operating in this manner, the gear *D*, pinion *H*, and gear *I* all rotate on their free bearings and do not have any effect on spindle *C*.

When the push-rod *M* is pulled forward by rod *S*, the clutch finger *N* is disengaged from pinion *B* and engages

the clutch face of gear *I* which then drives the shaft *C*. The drive is now transmitted through the pinion *B* to gear *D*, through link *J* to pinion *H* and thence to gear *I* at a reduction in speed of four to one as compared with the direct drive from the pulley *A*.

The gear *D* is mounted on the stud *E* secured to the sleeve member *F*, which can be swung about the hub bearing on the cover plate *T*. When the stud *E* is in axial alignment with stud *G*, gears *H* and *D* have the same speed and this speed is constant. The slide *P* has an angular slot in it which fits over the slide shoe *Q* on stud *E* and serves to move stud *E* of gear *D* out of alignment with the stud *G* in accordance with the movement imparted to rod *R* by the cam that also controls the oval mechanism.

In Fig. 7, angle *x* and dimension *y* indicate the amount

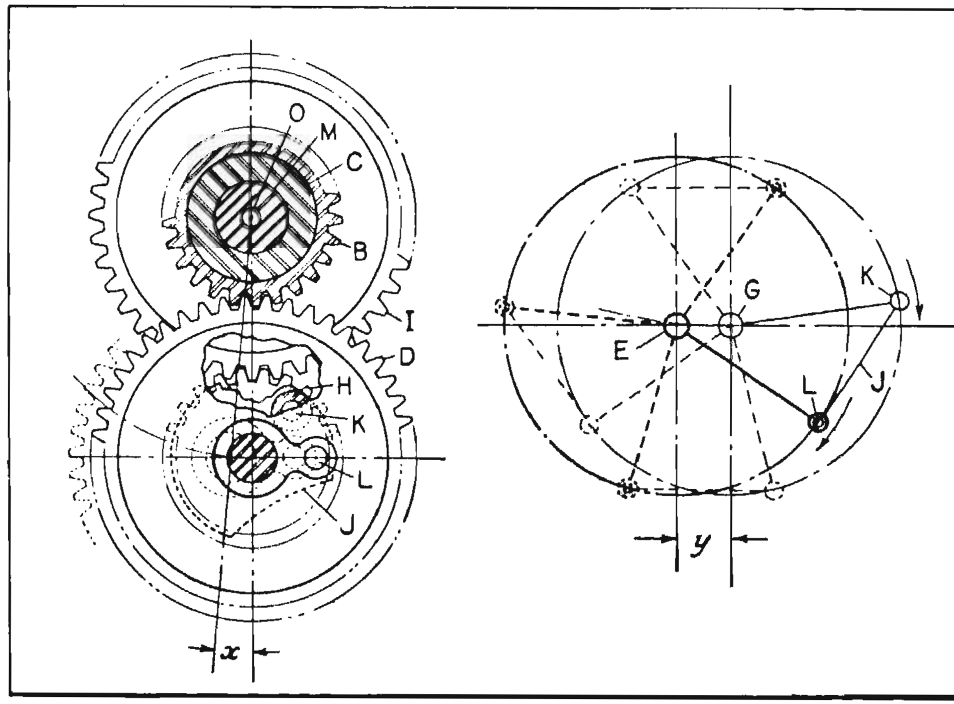


Fig. 7. Diagrams Used to Illustrate Operation of Velocity-changing Mechanism

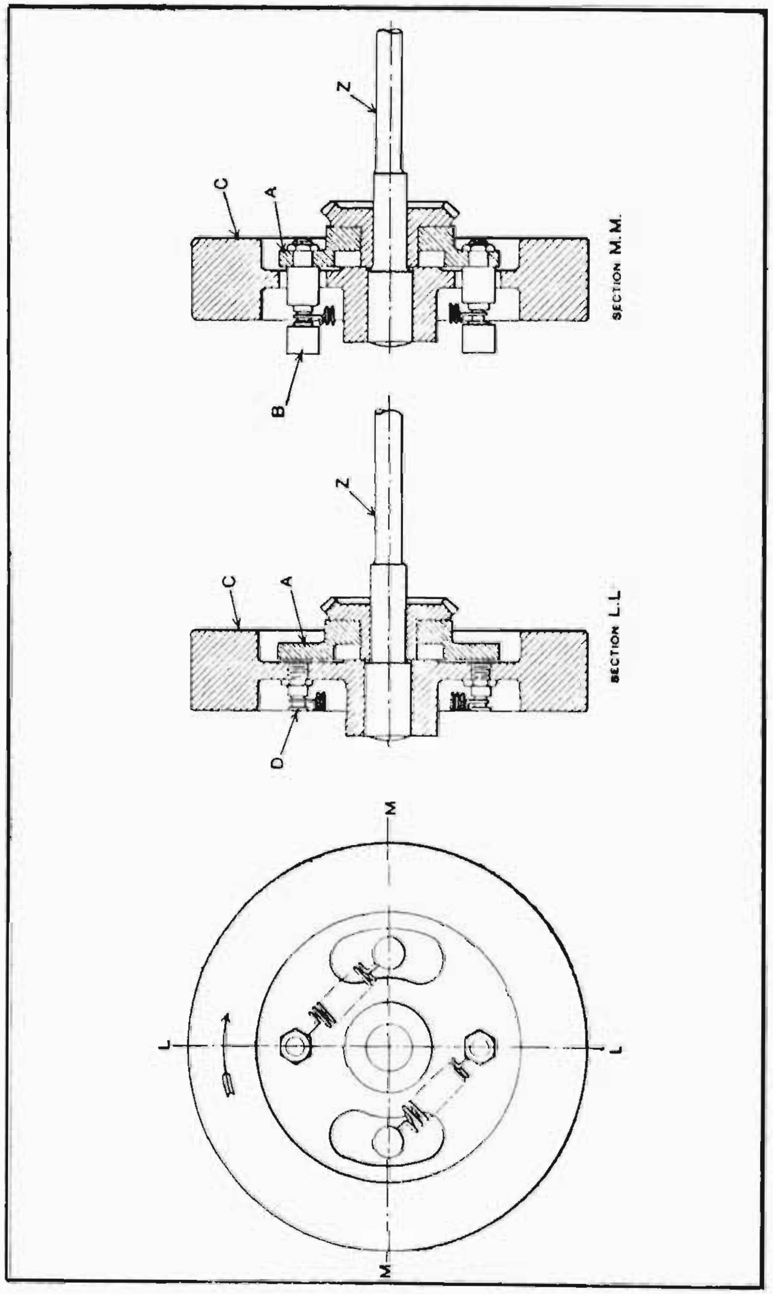


Fig. 8. Device for Converting Intermittent Motion to a Constant Drive

of "out of line" adjustment between the studs E and G. This adjustment is varied automatically from zero, when the finishing pad is at the top or center of the hat crown, to the maximum amount, when the finishing pad reaches the band of the hat. It is this changing of the axial relation of gear D and pinion H that controls the differential velocity. The diagram to the right will help make clear how the changing of the relative angular positions of the studs L and K and their connecting link J produces a variation in the velocity of the driven pinion H.

The change in angular velocity occurs once in each revolution of pinion H, but since pinion H makes two revolutions for each one of driven gear I, the change in the angular velocity of the latter gear and its spindle C occurs twice during each revolution. By changing the linkage connections between the cam and plate P, Fig. 6, it is possible to control the amount of variation in angular velocity. A segmental circular opening is provided in the lower portion of the cover plate T to allow sleeve F to swing on its hub bearing. The link J has no center bearing, and is held in position by the two drive pins K and L. The pin L projects from the inner face of gear D into a hole in link J. The pin K is made eccentric for convenience in assembling, but it would be possible to use a straight pin, as far as the operation of the mechanism is concerned.

Intermittent Motion Converted to a Constant Drive.— All moving projectors are equipped with some sort of intermittent device to cause the film strip to dwell at every picture. This is necessary, because running the film through the projector at a constant speed would result in a blur on the screen. With the advent of the "talkies," however, this motion had to be reconverted to a steady drive for the film strip while it passes through the sound-producing attachment, as otherwise the latter would not function correctly. This was done in one instance with the arrangement shown in Fig. 8.

The intermittent motion is transmitted to disk *A* through bevel gearing. This disk is equipped with two studs *B* which pass through elongated slots in the web of the flywheel *C* and are connected to two studs in the flywheel web by coil springs. The shaft *Z* is keyed to the flywheel. The intermittent motion transferred to disk *A* will be absorbed through the combination of the coil springs and the fly-

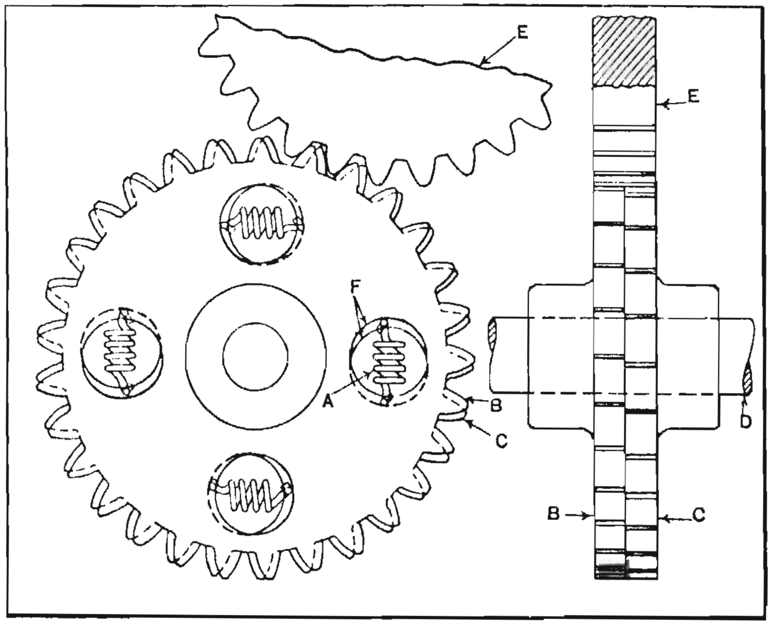


Fig. 9. Device for Eliminating Lost Motion in Gear Teeth

wheel, resulting in a constant speed for the shaft *Z*, which is the drive for the sound-producing attachment.

Double Gear for Eliminating Lost Motion in Gear Teeth.—In devices where gears are used, such as those operating graduated dials, it is often desirable to eliminate all lost motion resulting from wear occurring between the gear teeth. The gears shown in Fig. 9, which are only recommended for light transmissions, impart motion to a dial indicator that must register in both a clockwise and a counter-clockwise direction. It is obvious that any play in the teeth of the gears would produce inaccuracies in the dial readings.

The driving gear *E* meshes with both the gears *B* and *C*, the latter being fastened to the shaft *D*. In both gears *B* and *C* are drilled holes *F* for the springs *A*. One end of each spring is secured to gear *B* and the other end to gear *C*. The thickness of the teeth in both these gears is less than that of the driving gear *E*, so that normally, there would be considerable play between the meshing teeth. However, owing to the tension of the springs *A* the teeth in gear *C* are advanced ahead of those in gear *B* and serve to fill the tooth spaces in the driving gear. In this way, as wear occurs, it is obvious that all lost motion in the gear transmission is eliminated, and that no matter in which direction the gears are run, there will be no play between the teeth nor inaccuracy in the dial readings.

Over-Run Pawl Clutch to Permit Accelerating Driven-Shaft Speed.—The pawl clutch shown in Fig. 10 has been used very successfully in machines having camshafts or feed-screws which must be driven at accelerated or high speeds a part of the time. The low-speed shaft *S* drives the high-speed shaft *H* under normal operating conditions through the ratchet wheel *A* and pawls *G* of the collar *K*, which is keyed to the high-speed shaft. When the speed of shaft *H* is to be accelerated, a clutch mechanism, not shown, is engaged, and this drives the high-speed shaft at the accelerated speed through gearing connected with shaft *H* by gear *J*. The arrangement of the over-run clutch is such that it permits shaft *H* to be operated at the higher rate of speed without affecting the speed of shaft *S*.

Referring to the construction of the clutch, ratchet wheel *A* is keyed to shaft *S*. The pressure disk *B* is a sliding fit on shaft *S*, and is driven by ratchet *A* through pins *C*. The tension springs *D* tend to force the pressure disk *B* inward toward the ratchet wheel. Between disk *B* and the ratchet

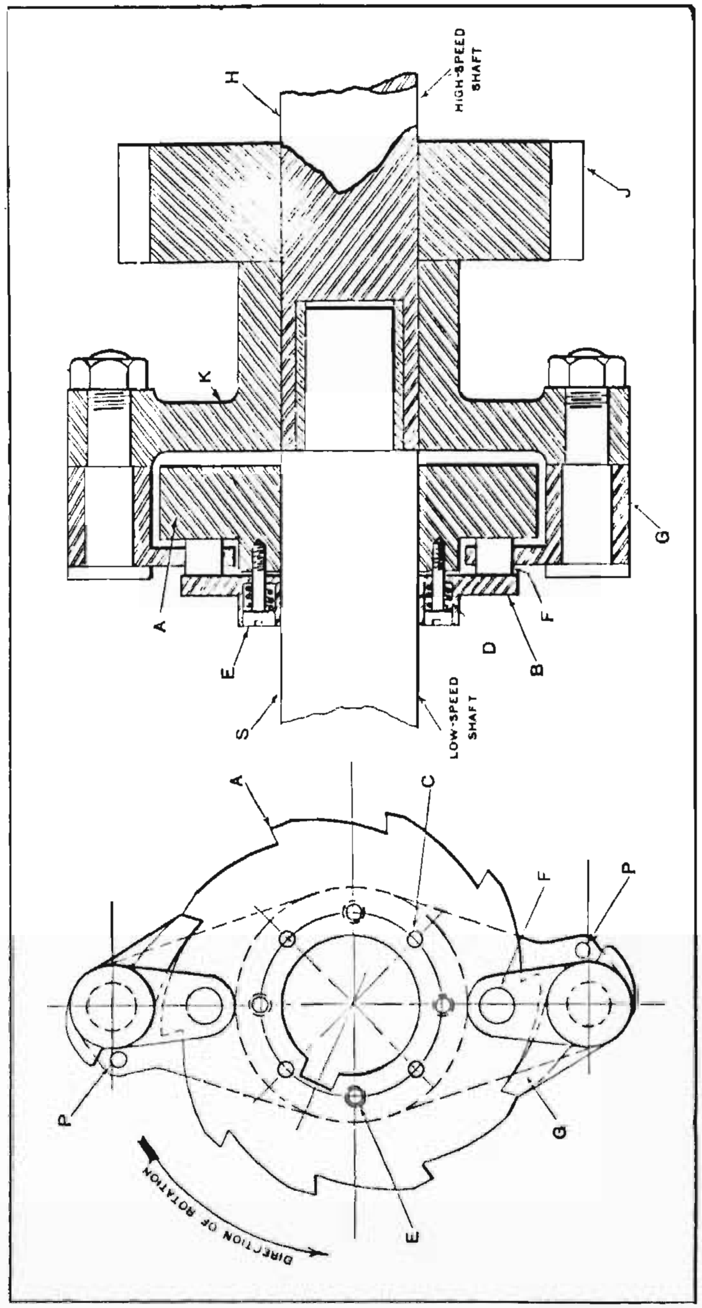


Fig. 10. Over-run Clutch which Permits Speed of Driven Shaft to be Accelerated without Affecting Speed of Driving Shaft

wheel are fiber drag-plugs *F*, which are set in holes in the ways of the pawls *G*. Behind the pawls are stop-pins *P* which keep the drag-plugs from swinging through too large an arc and thereby becoming entirely disengaged from the disk and the ratchet wheel. When the high-speed clutch is tripped and the gear *J* is driven at a higher speed than the ratchet wheel *A*, the pawls *G* are disengaged, and the plugs *F*, which then drag between the pressure disk and the ratchet wheel, cause the pawls to swing clear of the teeth in the ratchet member. The pawls remain in this disen-

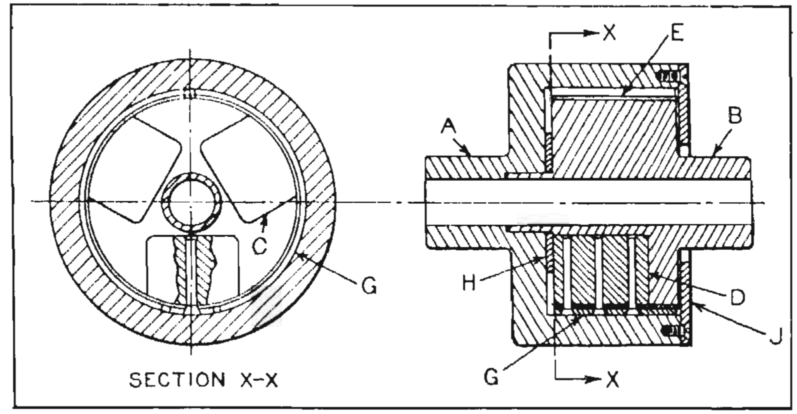


Fig. 11. Automatic Starting and Over-running Clutch

gaged position until the high-speed clutch is disengaged, at which time the drag on the plugs *F* is reversed, causing the pawls to be drawn down into engagement with the teeth and the ratchet wheel again.

Centrifugally Operated Starting and Over-Running Clutch.— In Fig. 11 is shown an automatic starting and over-running clutch of the centrifugal type. It consists of the driven housing *A*, in which rotates the driving clutch member *B*. Three cavities *C* are milled in member *B* to accommodate the three sliding weights *D*. To one of these weights is riveted the spring steel band *E*. Over the steel band and riveted to it is the brake lining material *G*. At *H* is a steel thrust washer, while at J is a retainer plate for keeping both halves of the clutch assembled.

The operation of the clutch is as follows: As the clutch is rotated by the driving member B, centrifugal force causes the weights D to move outward. This, in turn, expands band E, forcing the brake lining into contact with the inner surface of housing A. As long as the speed of driver B is maintained or increased, the gripping force is also maintained or increased, but the moment it is reduced, either from the slowing up of the driving force or because of resistance set up in the driven member, the sliding

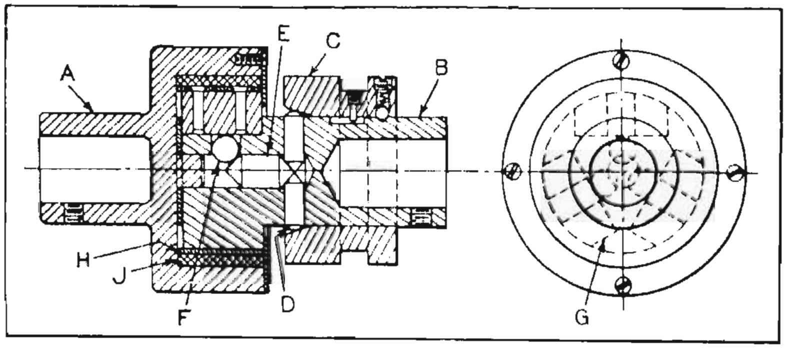


Fig. 12. Modified Design of Clutch Shown in Fig. 11

weights are forced inward by the spring band E, immediately disengaging the clutch and allowing half of the clutch to slip upon the other half. If the driving force is entirely cut off, the clutch disengages, allowing the driven half to over-run until it comes to rest.

Simplicity of construction and a large contact area are advantages of this clutch. It should be noted that the effective gripping power can be increased without increasing the diameter by merely lengthening the clutch. It should also be noted that, for a given velocity, the force exerted is proportional to the mass of the weights D, so that the larger the housing of the clutch, the greater will be the gripping force. This type of clutch can also be made in

the form of two halves of a brake-shoe, a centrifugal weight being used to force the two halves apart. However, this type of construction would involve a problem of dynamic balancing.

Fig. 12 shows the same type clutch employed in the reverse manner. Here the housing A serves as the driver, the driven member being the housing B, which is caused to rotate when the clutch collar C is forced over the pins D. These pins, of which there are three, force the connector pin E against the three steel balls F. Pins can be used in

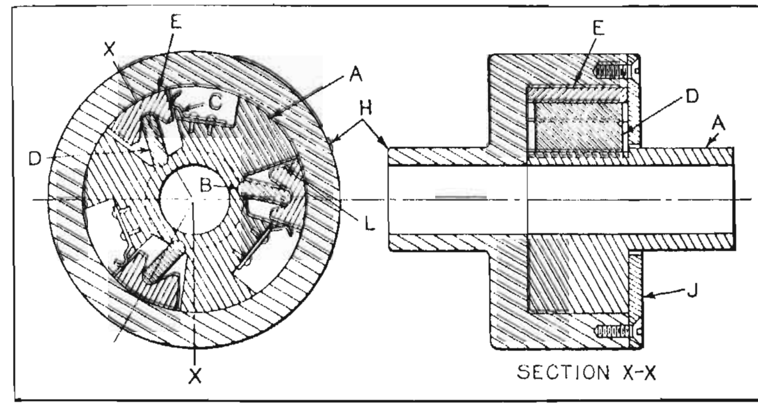


Fig. 13. Over-running Clutch with Toggle-actuated Shoes

place of the balls, if desired. The balls F force the three equally spaced sliding blocks G outward, causing the spring steel band H to expand, so that the brake lining material J will grip the outer housing, thus connecting the driving and driven members. The screw-pin is for the purpose of connecting the shaft collar C to the housing B, while the ball spring arrangement is provided to hold the shaft collar in either the open or the closed position.

Toggle Type of Over-Running Clutch.— An over-running clutch employing the toggle joint principle for obtaining the required locking action is shown in Fig. 13. It consists of the spider A having three milled slots spaced

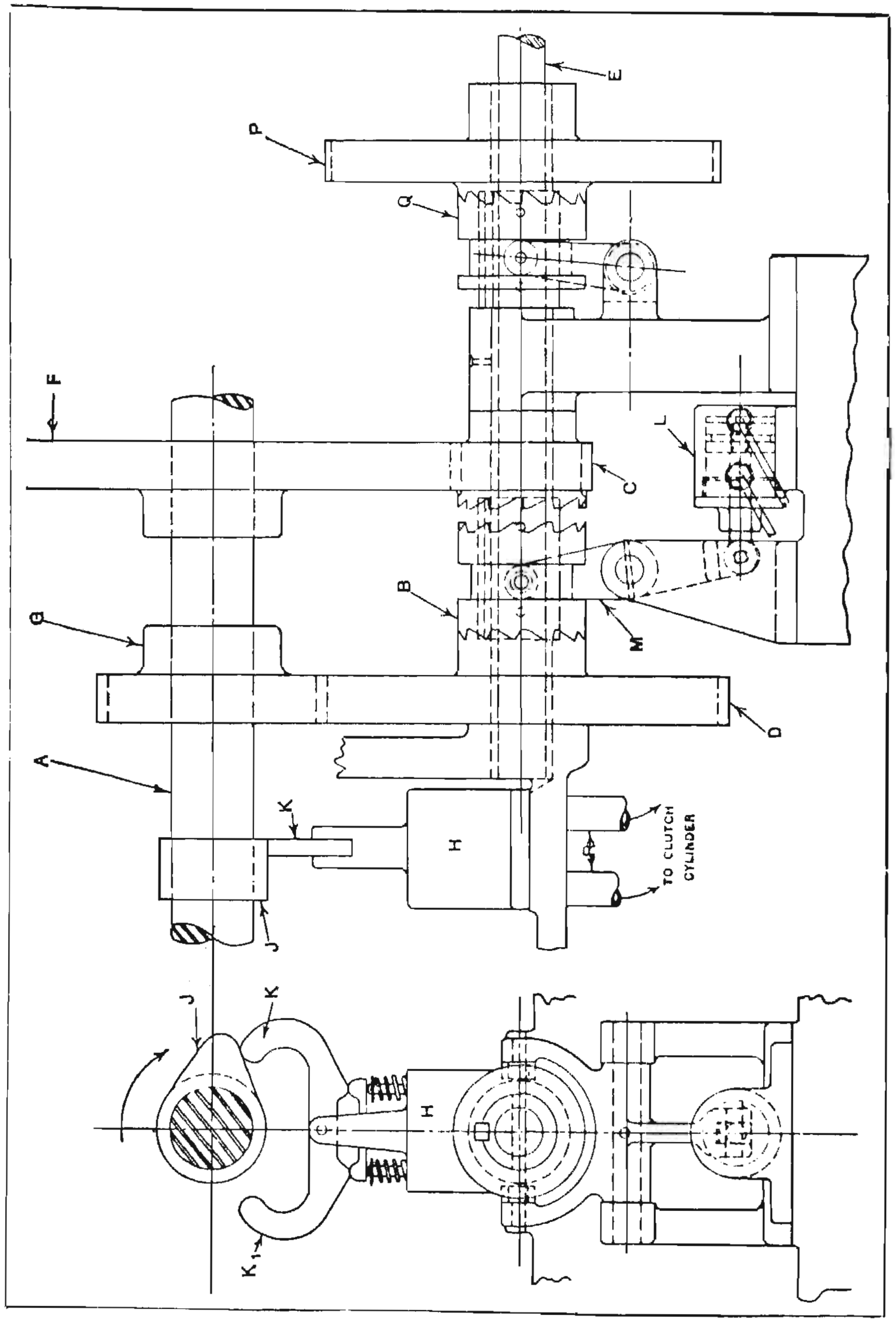


Fig. 14. Automatically Controlled Air-operated Clutch for Alternately Engaging High- and Low-ratio Gearing

120 degrees apart, the outer casing *H*, and the toggle levers *D*. At points *B*, which are slightly offset from the center lines of the slots, are milled semicircular recesses which act as seats for the toggle levers *D*. The shoes *E* are made slightly smaller than the width of the slots and have an outside diameter equal to the diameter of the spider *A*. Seats are milled in the rear sides of the shoes for the toggle levers *D*. The flat springs *C*, fastened in the slots in the spider, tend to keep the shoes *E* in contact with the inside of the outer casing *H*. A retainer plate *J*, held in place by screws, keeps the members of the clutch assembled.

The operation of the clutch is similar to that of any free-wheeling clutch. The toggle levers *D* are set at as slight an angle as possible, making due allowance for wear on the shoes *E*. After the shoes become worn, the faces *L* are machined or cut back. When the shoes have been cut back so that the toggles *D* become ineffective they are replaced.

Air-Operated Clutch for Two-Speed Drive.—The speed of a shaft that drives the feeding and indexing mechanism of a multiple-spindle drilling machine must be increased from 2.13 to 15 R.P.M. to permit indexing in 1 2/3 seconds. This speed change is controlled by a cam-operated, four-way air valve and an air-operated clutch which alternately engages the high- and low-ratio gearing.

Starting and stopping of the machine is controlled by hand-operated clutch *Q* (Fig. 14) which connects or disconnects gear *P* with shaft *E*. The main shaft *A* drives the feeding and indexing mechanism only. When the machine is drilling and the feeding mechanism is in operation, motion is transmitted from shaft *E* to *A* through the low-speed gearing *C* and *F*, as a result of the engagement of clutch *B* with gear *C*. When an indexing movement is required, clutch *B* is automatically shifted into engagement with gear *D*, thus driving shaft *A* through gears *D* and *G* and increasing the speed to 15 revolutions per minute, so that the indexing will be completed in the allotted time;

then the clutch is shifted back automatically to the feeding position.

Clutch *B* is shifted by means of compressed air acting against a piston which is within cylinder *L* and is connected to clutch yoke *M*. The admission and exhaust of the air to and from cylinder *L* is controlled by a cam *J* acting in conjunction with a four-way valve *H*. This valve connects with the main air line, and there are two 3/8-inch pipes *R* leading from it, provided with reducing bushings to fit the small tubes connecting with each end of the air cylinder *L*. When cam *J*, which is attached to shaft *A*, moves in the direction of the arrow, it comes into contact with end *K* of the air valve operating lever, thus admitting air to the left-hand end of the cylinder and exhausting it from the opposite end. The result is that clutch *B* is thrown into engagement with gear *D*, as the illustration shows. This high-speed drive of 15 revolutions per minute continues until cam *J* engages end K_1 of the lever, thus admitting air to the right-hand end of the cylinder and exhausting it from the left, which throws the clutch into engagement with gear *C*. An air pressure of 80 pounds per square inch is carried in the main line, and the total pressure exerted against the piston in cylinder *L* is about 98 pounds.

CHAPTER XII

SELF-CENTERING PIVOTED LEVERS AND SLIDING MEMBERS

It is sometimes necessary or desirable to provide certain machine elements with mountings that will permit them to be deflected from or forced out of their normal positions by other parts of the machine. Usually such elements must be so designed or equipped that they will return automatically to their normal positions when the deflecting forces are removed. The term "self-centering" is used in reference to the devices shown in Figs. 1 and 2, because they serve to return the elements to their central or normal positions when the deflecting force is removed. Probably the most familiar example of a self-centering device is the spring-actuated control rod employed to operate a dog-tooth clutch. With this type of clutch, the two members may not be in the correct angular relation to permit them to engage when the control lever is thrown over. Under these conditions, it is the function of the self-centering spring-actuated control rod to yield and permit the clutch lever to be thrown over without unduly straining the mechanism, causing the two parts of the clutch to engage as soon as they are in the proper positions.

Another application of self-centering devices is to the control levers of power-operated machines. For such applications, the lever is normally held in the neutral or central position by the self-centering device, and is returned to this position automatically as soon as it is released by the operator. Manual operation of the lever in one direction or the other places the machine either in forward or reverse motion, and its release causes the machine to stop.

Self-Centering Devices for Angular Movement.—The attachment of a weight, pendulum fashion, as shown at *A*, Fig. 1, is one of the simplest examples of a self-centering device having an angular movement. A device of this kind may be used for a lever having a range of movement of 20 degrees each side of the vertical center line. This type of self-centering device, however, may be objectionable on account of the inertia introduced by the weight. Another objection is that the weight offers little resistance to angular movements of small amount, as the self-centering force is zero at the central position. Still another objection is the tendency of the weight to oscillate after displacement.

The device shown at *B* is similar to the one at *A* except that spring tension is substituted for the weight. This type of self-centering device is often used and is fairly effective for some purposes. The self-centering effect, however, is zero at the mid-position of the lever, and effective centering forces are not developed until a considerable angular movement of the lever has taken place. The lever also has a tendency to vibrate after the decentralizing forces begin to act. In practice, short stiff springs are usually employed, which cause a heavy pressure to be exerted on the bearings or pivots of the lever.

The self-centering device shown at *C* typifies approved practice. Two clip levers pivoted about the boss of a lever are employed in this device. A pin carried by the lever is gripped between the two clip levers with a force depending upon the initial tension of the spring connecting the ends of the clip levers. Another pin of the same width as that on the lever is fixed to the stationary framework of the machine and interposed between the two clip levers. The result is that no movement of the lever in either direction can take place without forcing the clip levers apart against the resistance of the spring. The majority of self-centering problems involving angular movement can be solved by the application of the principles embodied in this device.

Self-Centering Device Applied to Electric Switch.—The self-centering device employed on a rotary snap-action electrical switch is shown at *D*, Fig. 1. The principle on which this centering device operates is similar to that of the device shown at *C*. In this case, a C-shaped spring is used to force the clip levers together, as the angular movement is only a few degrees. The centering forces are equal in both directions of angular movement in the devices shown at *C* and *D*.

It is sometimes desirable, however, that the decentering forces be more strongly resisted in one direction than in the other. A modification of the design shown at *C* to meet this requirement is illustrated at *E*. The unequal tension is obtained by employing two separate springs of unequal strength for the two clips. As the springs are anchored to the machine framework, it is a simple matter to provide them with means for adjusting the tension.

The designs shown at *C* and *E* have proved highly satisfactory in general practice, but when absolute precision in the self-centering action is necessary, the type shown at *F* is preferable. With the designs shown at *C* and *E*, any difference in the size of the two pins located between the clip levers will result in lost motion, whereas the design shown at *F* is free from this possibility.

In this design, one of the clip levers is dispensed with, and a spring is used to pull the main lever and the clip lever together. This provides a definite self-centering action without the possibility of the slightest lost motion, and has been successfully employed in the design of a shock absorber in a train of gearing connecting a gun to its elevation indicator. Under the shock of the recoil of the gun, the device yields, but immediately regains its proper position with respect to the indicating pointer. In this application, any lost motion would destroy the accuracy of the elevation indicator.

Roller and Spiral Cam-Operated Centering Device.—In the case of the self-centering device shown at *G*, Fig. 1,

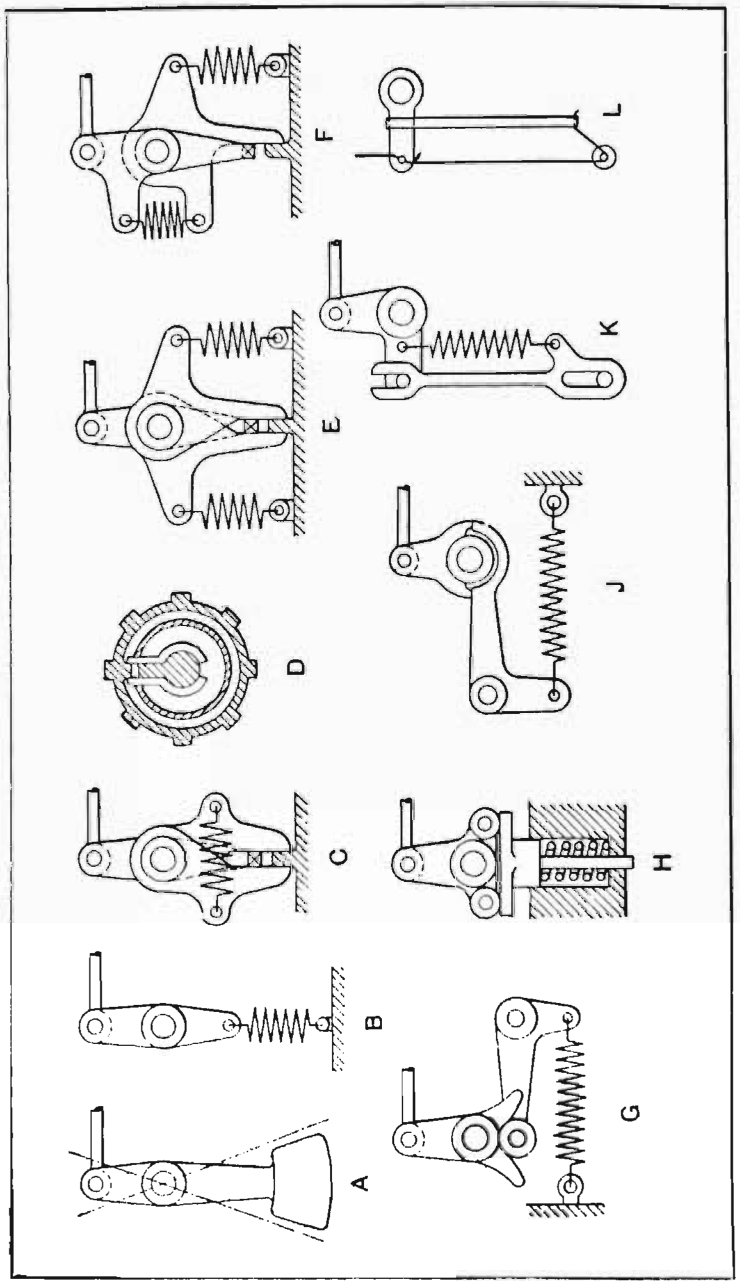


Fig. 1. Examples of Various Types of Self-centering Levers

the actuating forces are applied to the lever by a roller which is pressed into engagement with a spirally shaped cam through the action of a coil spring. The arrangement is similar to that of the "zeroizing" cam lever used in stop watches. By shaping the cam curves properly, any desired variation in the centering force can be obtained from the spring that presses the roller radially inward. A heavy pressure on the roller and the lever bearings would result if the pitch of the spiral cam were made small in order to secure a wide range of angular movement, and in such a case, it would be advisable to use ball bearings for the roller and for the lever bearings.

Another design, consisting of two arms extending in opposite directions and carrying rollers that are subjected to spring pressure from a radially sliding T-shaped piece, is shown at *H*. When the main lever is in the central position, the T-shaped piece exerts an equal pressure on both rollers. Angular movement of the lever in either direction results in one or the other of the rollers depressing the T-shaped piece against the action of the spring.

At *J* is shown a design similar to that at *H*, except that the rollers are omitted and a pivoted lever is used in place of the T-shaped piece. When there is a limited amount of space directly below the lever, this design may be used to advantage. The devices shown at *G*, *H*, and *J* possess one feature that is often desirable, namely, they permit the self-centering action to be rendered inoperative when desired by forcing the centering T-shaped piece or lever out of contact with the main lever.

At *K* is shown an interesting, effective and extremely simple self-centering device. There is no lost motion in this type, and it can be arranged to give equal or unequal resistance to movement in either direction. It requires but one actuating spring, which can be readily provided with a tension adjustment.

The device is simply a flat connecting-rod slotted at each end, one slot being engaged by a pin on the lever, and the other slot by a pin secured to the machine framework in a fixed position. The spring is fastened to the connecting-rod and to the lever. If the upper end of the spring is connected at a point midway between the fulcrum of the lever and the connecting-rod, the self-centering forces will be equal in both directions. If the upper end of the spring is moved nearer the lever fulcrum, the self-centering force opposing clockwise rotation of the lever will be reduced, while the force opposing anti-clockwise rotation becomes increased. The action of this design is illustrated by the model shown at *L*. Three pins, a cardboard lever, a piece of wire, and an elastic band suffice to provide this working model of the device.

Self-Centering Devices of the Sliding Type.—One of the earliest types of sliding self-centering devices is shown at *A*, Fig. 2. The centering forces are supplied by two springs acting in opposite directions on the sliding member. When flexible springs are used, the centering or positioning action is rather indefinite with this type of device, especially if the springs are flexible. The position of the sliding member depends upon the value of the force opposing the self-centering springs.

At *B* is shown what may be considered the usual or standard practice in the design of self-centering sliding members. In this case, the springs apply the centering pressure to washers placed on each side of a collar located on the sliding spindle. The inner sides of the washers engage the sides of a member that projects from the machine frame. Any vertical movement of the sliding member is resisted by the projecting member on which one of the two springs acts.

The springs may exert equal or unequal pressure on the collars, as desired. When an equal resistance in either direction is desired, the design is frequently modified, as shown at *C*, so that only one spring is required.

Methods of Eliminating Lost Motion.—Unless the distance between the faces of the washers employed in the sliding self-centering device is equal to the corresponding distance between the faces of the projecting members, lost motion will occur. Where the possibility of the slightest lost

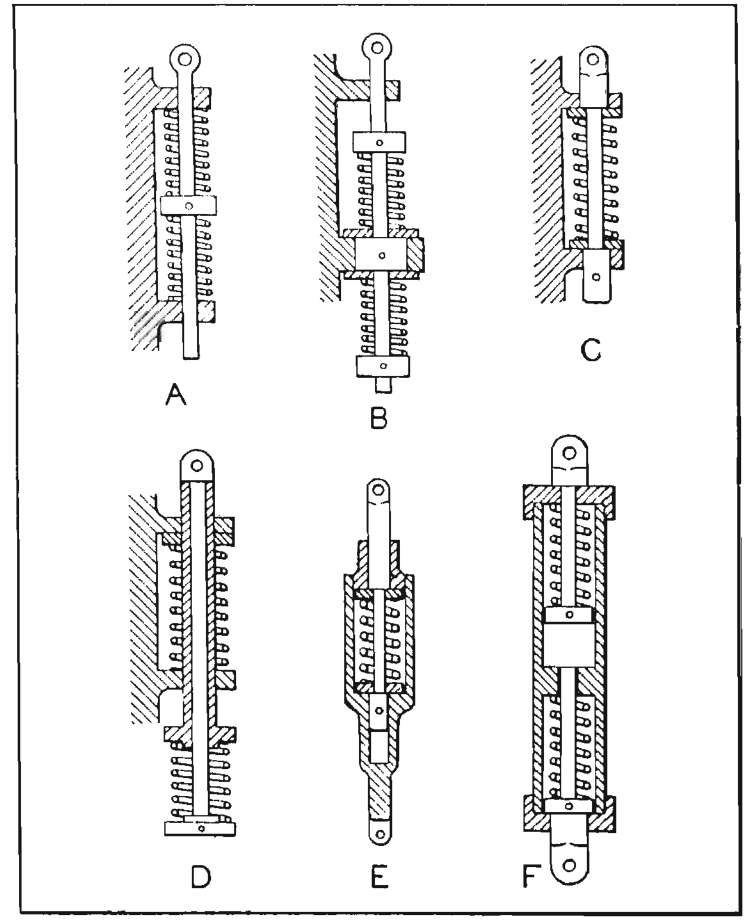


Fig. 2. Self-centering Devices of the Sliding Type

motion must be eliminated, it is better to use a device of the type shown at *D*, Fig. 2, in which the upper collar is pinned to the sliding member. In this design, the sliding member is mounted within a sleeve which is pressed upward against a projecting member on the machine frame by a spring. A depressing movement of the sliding member carries the sleeve with it. On the upward movement of this member, the sleeve cannot follow, and the sliding member rises out of the sleeve against the pressure of a second spring that acts between the two parts. With this arrangement, no

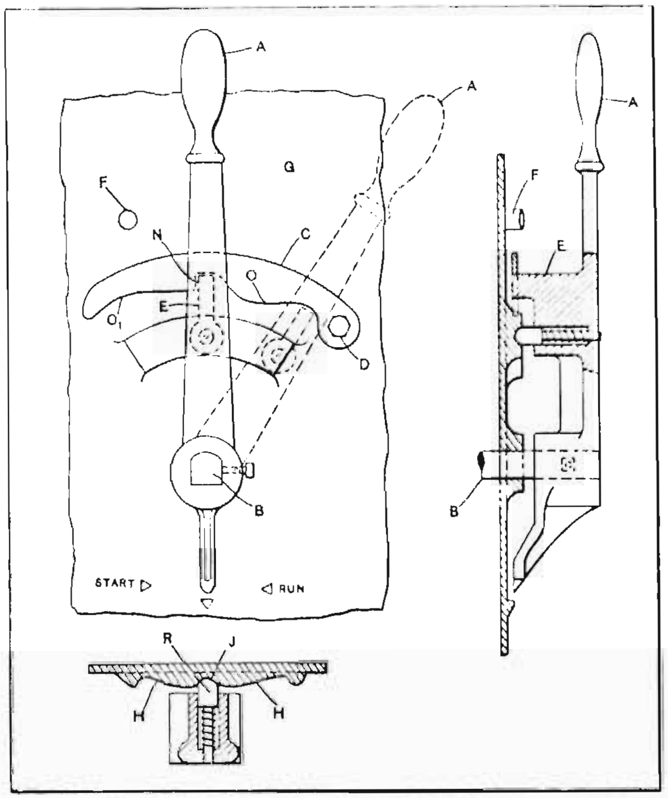


Fig. 3. Gherkin Latch Used on Electrical Equipment

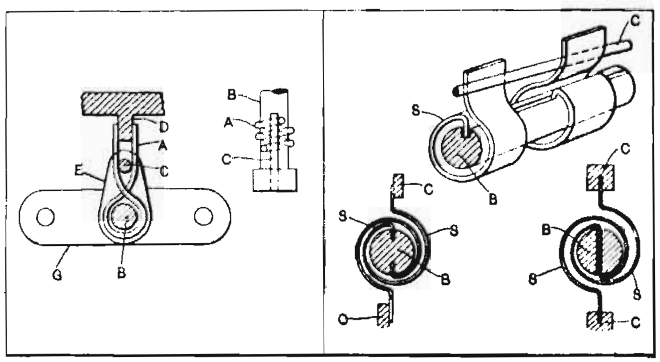
lost motion is possible and the centering action is precise.

The device shown at C is commonly employed in the design of spring-actuated connecting-rods used in dog-clutch control link work and for similar purposes. At E is shown a typical connecting-rod of this class.

At F is shown a design, similar to that illustrated at D, which is intended for use where the possibility of the slightest lost motion must be eliminated. In this design, both springs are of the same size and provide equal resistance to either compression or extension of the connecting-rod.

Gherkin's Latch.—Fig. 3 shows a mechanism which is known as a "Gherkin's latch" which is used in conjunction with certain types of centering devices. This latch consists of a handle A which is mounted on shaft B. The handle has a projecting boss E which engages with the latch C that is pivoted on the pin D. The pin D is carried by the case G and the movement of the latch is limited by a second pin F. This latch is so constructed that, when the handle is moved slowly to the left, the latch will prevent the handle moving beyond the notch N; but if the handle is moved over to the extreme right—as shown dotted in the illustration—and then thrown quickly in the opposite direction, the boss E on the handle will leave the top of the incline O on the latch with sufficient speed to enable it to jump across the notch N, in which case the handle will come to rest in the position marked O_1 which is the running position. The pin R riding on the surfaces H will lock the handle in the cavity J when the action of the centering spring becomes effective. In this way, the handle is securely held in the central or off position.

Applications of Gherkin's Latch.— This latch may be used in connection with either of the centering devices shown in Figs. 4 and 5. The centering spring has one end attached to the shaft B of Fig. 3 and the opposite end to the case G. Referring to the illustration of the centering spring, or so-called "cross-legged spring" shown in Fig. 4, it will be seen that a coiled spring A is employed to connect the shaft B with the case. The effectiveness of this spring is dependent on the following conditions: first, that the ends cross each other in a line which is perpendicular to the horizontal axis of the spring; second, that the ends of the spring extend far enough above the outside of the coils to enable a pin or bar C and a fixed boss D to be inserted between these two ends. The pin or bar C is carried by the crank E, which, in turn, is carried by the shaft B of the mechanism shown in Fig. 3. An arm G or some other means provides for transmitting the motion so that when the shaft B is moved in either direction the cross-legged spring will be put in tension by having one end turned through the action of the pin C. This spring tension provides for returning the arm G and the shaft to which it is

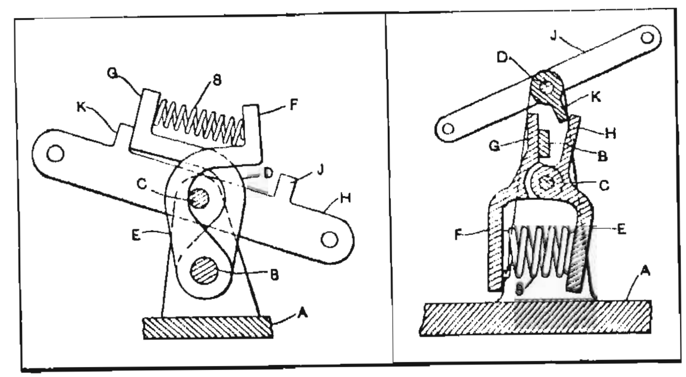


Figs. 4 and 5. Two Forms of Centering Devices that may be Used with a Gherkin Latch

secured to the central position when the action of the spring becomes effective. A very simple centering device is illustrated in Fig. 5, the only difference from the preceding type being that two separate springs are employed in this case. One end of each of these springs S is secured in a slot in shaft B, while the other end is held by the fixed pin C. When shaft B is turned in either direction, one of these springs is put under tension and this spring tension provides for returning the shaft to the central position when the action of the spring becomes effective. Two modifications of this design are shown in the lower corners of this illustration.

Miscellaneous Types of Centering Devices.— Another form of centering device is illustrated in Fig. 6. Referring

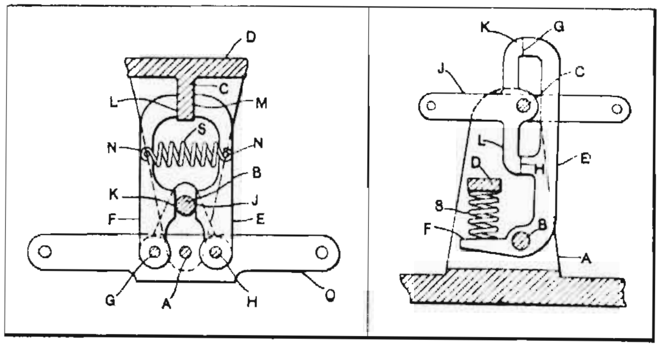
to this illustration, it will be seen that there is a bracket A which supports a pivot B and a pin C. Two arms D and E are mounted on the pivot B, on which they are free to swing. On the ends of the arms are projections F and G which are secured to opposite ends of the spring S. These projections F and G also engage with bosses J and K on the moving member H. This moving member is carried by the pin C, upon which it is free to swing. Assuming that the moving member H is rotated in a clockwise direction, the centering device will assume the position shown. The arm E is held against the pin C, while the boss G on the arm D engages the boss K on the movable member H. Further rota-



Figs. 6 and 7. Two Types of Double-arm Centering Devices

tion of the member H will compress the spring S and this will return the movable member to the central position when the action of the spring becomes effective. If the movable member H is rotated in a counter-clockwise direction, the action of the centering device will be exactly reversed, the arm D being held by the pin C while the boss F on the arm E engages the boss J on the movable member H, thus causing the compression of the spring S.

In the centering device shown in Fig. 7, the frame A is provided with a stop B and two pivots C and D. Two arms E and F are mounted on the pivot C, these arms being free to swing and provided with projections G and H at their upper ends. When the centering device is in its normal position, both of the projections G and H are in contact with the stop B, owing to the action of the compression spring S which forces the lower ends of the arms in opposite directions. It will be seen that the movable member J is mounted on a pivot D and provided with a projecting lug K on its lower side. This lug engages with the projections G and H, and when the member J is rotated in either direction, it swings one of the arms about the pivot C and compresses the spring S. When the action of this spring



Figs. 8 and 9. Two Types of Centering Devices in which Sliding Contact of the Control Arms is Employed

becomes effective, it returns the arm to the normal position; and the projection at the upper end of the arm acts on the lug K, causing the movable member J to be returned to the starting point.

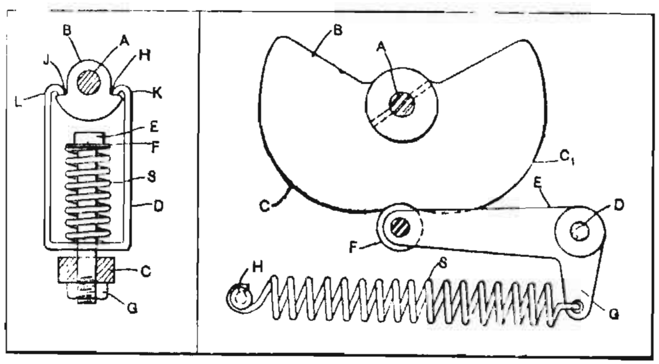
The centering device illustrated in Fig. 8 works on a somewhat different principle from that of the preceding types. The frame D carries a stop C and a pivot A. The movable member O is carried by this pivot A, and, in turn, carries two arms E and F. The arm E is carried by the pivot H and provided with two sliding surfaces J and M which work in contact with the pin B and the boss C, respectively. The arm F is supported on the pivot G and has

two sliding surfaces K and L which work in contact with the pin B and the boss C, respectively. The tension spring S is secured to the arms E and F at the points N. Assuming that the movable member O is to be rotated in a clockwise direction, the arm E will be pulled down by the supporting pivot H and in so doing, the surfaces M and J will slide on the boss C and pin B. For the same reason, arm F will be pushed up, the surface L sliding on the boss C and the surface K sliding on the pin B. This action increases the distance between the pins N to which the spring is secured, thus putting the spring S under tension; and when the action of this spring becomes effective, it will return the movable member O to the starting point.

In the device shown in Fig. 9, the frame A carries two pivots B and C, and a boss D to which one end of the spring S is secured. The arm E is pivoted at B and has a projection F which engages the opposite end of the spring S; this arm also has two sliding surfaces G and H, which work in contact with the projections K and L on the movable member J. This movable member is carried on pivot C. When the member J is rotated in either the clockwise or counter-clockwise direction, one of its projections causes the arm E to be rotated in a clockwise direction about the pivot B. When this rotation takes place, the projection F on the arm E compresses the spring S. When the movable member J is released, the action of the spring S becomes effective and returns it to the central position.

Device for Small Angular Movements.—Fig. 10 shows a form of centering device which is limited in its application, but has been found particularly effective in those cases where it can be used. It is only applicable for returning mechanisms which have a relatively small angular movement. Referring to the illustration, it will be seen that the shaft A has a hub B secured to it. An arm or other means of transmitting the motion may be secured to the shaft A. The hub B has two recessed surfaces H and J that receive the ends K and L of the yoke D, which is held in position by means of a bolt E. The opposite end of this bolt is carried by a lug C, and the bolt carries a spring S and washer F which supports the pressure of the upper end of the spring. When the shaft A is rotated in either direction, the hub B raises the yoke D, thus compressing the spring S, and when the action of this spring becomes effective, it pushes down the yoke and returns the shaft A to the starting point.

Cam, Bellcrank, and Spring Combination.—One of the simplest forms of centering devices that can be used consists of the combination of a cam, bellcrank, and spring, illustrated in Fig. 11. Referring to this illustration it will be seen that the shaft A has the two-lobed cam B secured



Figs. 10 and 11. Yoke and Cam Types of Centering Devices

to it. The bellcrank E is mounted on the pivot D; and this bellcrank has the cam roller F mounted at one end and the other end secured to the spring S. The opposite end of the spring is secured at the point H. When the shaft A is rotated in either direction, the lobe C or C_1 of the cam pushes down the end of the bellcrank; and this causes the crank to rotate about the pivot D. The result is that the opposite end G of the bellcrank swings out and places the spring S under tension. When the action of this spring becomes effective, it rotates the bellcrank in the opposite direction and returns the shaft A to the starting point through the action of the cam roller F on the cam.

Oil-Switch Control Centering Device.—A type of centering device which finds wide application in the control of oil switches is illustrated in Fig. 12. In this illustration, A represents a shaft to which a hub C is secured. This hub carries a projecting lug D which extends between the two arms E and F of the centering device. These arms are free to swing about the shaft A. The arm E is provided with an extension K to which one end of the spring S is secured, and a second extension G which limits the movement of the arm E in a clockwise direction through contact with the stop B. The arm F has an extension J, to which the opposite end of the spring S is secured, and a second extension

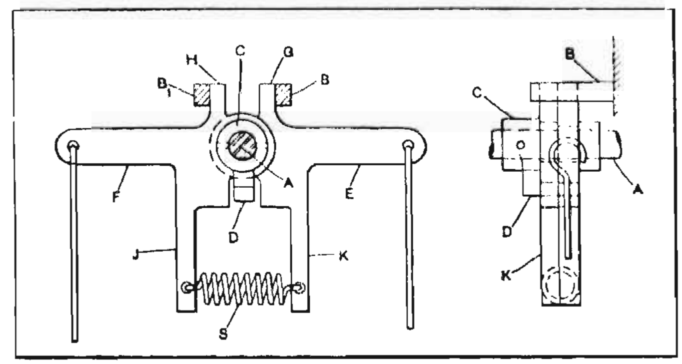
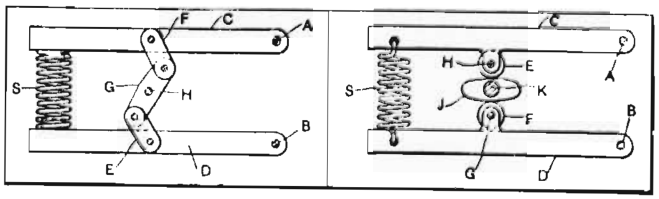


Fig. 12. Another Form of Double-arm Centering Device

H which limits the rotation of the arm F in a counter-clockwise direction through contact with the stop B_1. The tension of the spring S holds the arms E and F against their respective stops B and B_1. When shaft A is rotated, the hub C and its lug D are moved in either a clockwise or counter-clockwise direction as the case may be. This rotation causes either the arm E or F to be rotated against the tension of the spring S, and when this spring tension becomes effective it causes the shaft A to be returned to the starting point.

Lever-Returning Devices.— Figs. 13 and 14 show two simple devices for returning two levers to their normal positions. In the device shown in Fig. 13, the lever C is carried by the pivot A and the lever D is carried by the pivot B. An arm G is pivoted between these levers and connected to them by means of two links E and F. When arm G is rotated in a clockwise direction, it causes levers C and D to swing in about pivots A and B, thus placing the spring S under compression. When the force tending to bring the levers C and D together is released, the compression of the



Figs. 13 and 14. Centering Devices for Returning Two Arms Simultaneously

spring S will return the levers to their normal positions. The arrangement of the mechanism shown in Fig. 14 is quite similar to that of the preceding illustration, except that the levers C and D are provided with cam rolls E and F which are mounted on pivots H and G. When the cam J is rotated in either direction, by turning the shaft K, it causes the levers C and D to swing outward about the pivots A and B, thus placing the spring S under tension. When the torque tending to rotate the shaft K is released, the tension of the spring S returns the levers C and D to the starting point.

CHAPTER XIII

MULTIPLE-LEVER MECHANISMS WITH DWELLING OR IDLE PERIODS AND OTHER SPECIAL LEVER COMBINATIONS

Mechanical movements may in many cases be derived in the simplest manner by the use of properly proportioned levers or combinations of levers and connecting links. Several lever combinations which are arranged to provide a period of dwell during the cycle of movements will first be described and these designs will be followed by certain special lever applications or other devices allied in some way to mechanisms of the lever type.

Multiple-Lever Mechanisms Designed to Obtain Dwells in Lever Movements.— Levers, in combination with links, can be used as a means for obtaining a dwell or idle period during the cycle of movements imparted to the lever of a driven shaft. The cam and follower-roll mechanism is perhaps the only simple one in which complete elimination of motion is obtained during the dwell period, but it cannot always be applied conveniently; moreover, it is often difficult to obtain sufficient movement by means of cams.

When a close approximation to complete elimination of motion during the dwell period will meet requirements, levers and links provide a simple solution of the transmission problem, particularly when the driving and driven units are not located too close together. Fig. 1 illustrates a case of this kind, in which a hand-lever (not shown) mounted on the driving shaft operates two distinct mechanisms near the ends of its stroke, while the middle section of the stroke operates the driven shaft, which receives no noticeable movement when the other two mechanisms are being actuated.

Advantages of the Lever-Type Dwell Mechanism.—The lever type mechanism has many advantages. For example, with the lever and link mechanism, it is a simple matter to increase or decrease the movement imparted to the driven member and change the direction of movement.

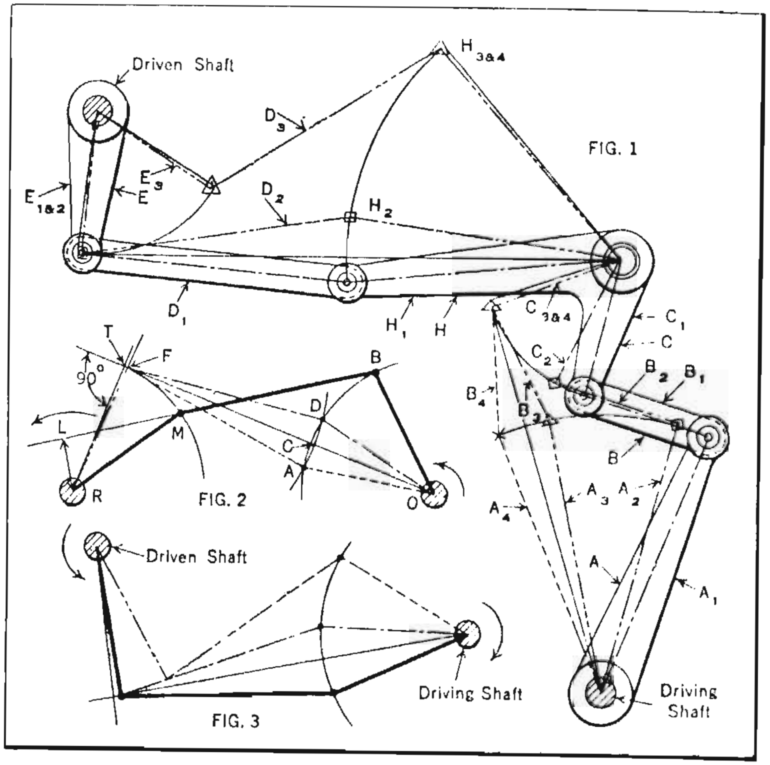


Fig. 1. Lever and Link Mechanism Having Dwell Period for Driven Shaft at Each End of Oscillating Movement of Driving Shaft. Fig. 2. Diagram Used to Illustrate Method of Laying out Lever and Link Mechanisms. Fig. 3. Arrangement for Obtaining Dwell at Beginning of Driving Lever Stroke with Shafts Turning in Opposite Directions

Such mechanisms can also be arranged easily to avoid obstacles. Only simple parts which can be made easily in any machine shop are required for the lever mechanisms. These mechanisms operate smoothly and quietly without requiring any attention other than an occasional oiling. All the levers shown in Fig. 1, together with some that are not shown in the illustration, were required for the transmission of motion from the front of a machine to a higher position at the rear. This is accomplished, with the additional feature of a dwell period of the driven shaft near each end of the stroke of the driving shaft.

Operation of a Lever-Type Dwell Mechanism.—In operation, the lever A (Fig. 1) on the driving shaft moves from position A_1 through A_2 and A_3 to A_4, serving to operate two mechanisms (not shown) from levers also mounted on the driving shaft. The movement from A_2 to A_3 is transmitted to the lever on the driven shaft, causing it to move from E_1 to E_3. Lever E on the driven shaft dwells while the lever on the driving shaft is moving from A_1 to A_2 and from A_3 to A_4. The first dwell of the lever on the driven shaft, as lever A moves toward A_4, occurs while the lever on the driving shaft moves from position A_1 to A_2, and while the longer or driving arm of the bellcrank H swings from H_1 through the neutral point to H_2, causing lever E to move only a very short distance, as indicated by the full line and the dot-and-dash line. The actual over-travel transmitted to lever E at this time amounts to an angular movement of only one minute. Even this small movement can be reduced by incorporating a similar arrangement at another point in the transmission system. As a matter of fact, the motion of lever E during the dwell period does not exceed 8 per cent of the total movement.

The movement of lever A from A_3 to A_4 results in transmitting practically no movement to link B and lever C. This movement simply causes positions C_3 and C_4 to become merged at one point, giving corresponding positions H_3 and H_4 to lever H. It is generally necessary to increase the length of the driving levers or decrease the length of the driven levers to compensate for the decreasing effects, on the driven levers, of the angular motions of the drivers as they approach the position of dwell. This can be done conveniently, but the designer must be careful not to reduce too greatly the moment arms of the driving forces in the links.

Incidentally, it may be mentioned that link and lever arrangements of this kind are self-locking against the reversal of the lever moments when the driving lever is at each end of its stroke. In the case illustrated, there was no necessity to utilize the toggle action of the levers for the production of heavy pressure or for locking purposes, but this feature might be useful in some cases.

Laying Out a Lever-Type Dwell Mechanism.—The general method of laying out a mechanism of this kind is illustrated in Fig. 2. This lay-out can be varied considerably without seriously affecting the results. First, the approximate or definite locations for the driving and driven shafts are laid out and the directions of the motions are determined, so that intermediate levers and links can be sketched in for a preliminary trial. Isolating one unit, as in Fig. 2, arcs representing the swing of the link-pins are drawn, and from the center of the driving shaft is drawn line OCT tangent to the arc of the driven link-pin.

It is not absolutely necessary to have line OCT in the tangential position, but this position ordinarily gives the best results. From the center of the driven shaft, line RT is drawn at right angles to line OCT. Now, CT represents the length of the connecting link. From C mark off at D and A arcs representing one-half the angle through which it is desired to eliminate the transmission of movement to M, and with length CT and centers A and D draw arcs which cut OCT at F. This point F is the final position of the driven link-pin. From OA lay off the point B, so that arc AB subtends the whole angle of rotation of the driving lever. Then point B is the starting position of the driving link-pin.

From point B, with a radius equal to CT, draw an arc cutting the arc of the driven link-pin at M. Now, RM indicates the initial position of the driven lever, while BM indicates that of the link. Thus, it will be seen that while the driving lever is given a continuous forward motion, the driven lever moves forward up to its final position and then has a slight additional forward motion, after which it returns to its final position. This last reciprocating movement between F and T is usually so slight as to be negligible. This slight movement also acts on the driven lever in such

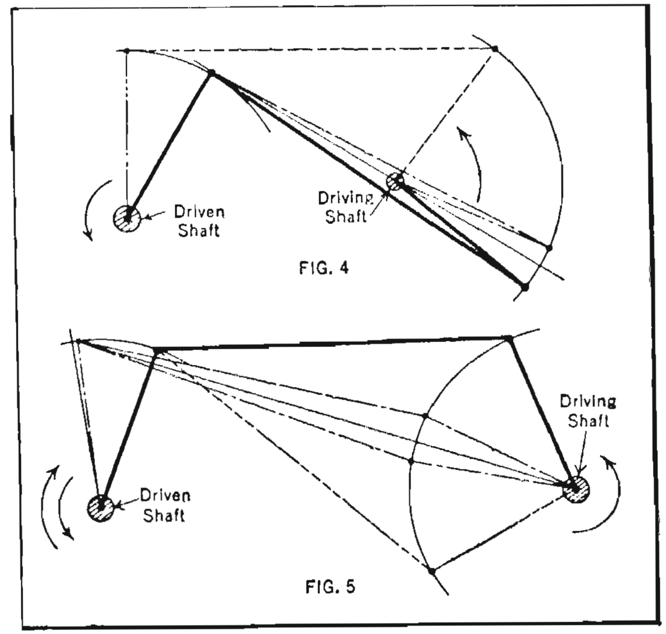


Fig. 4. Levers Arranged to Have Driven Shaft Dwell at Beginning of Driving Shaft Stroke with Both Shafts Turning Counter-clockwise. Fig. 5. Arrangement for Dwell at Mid-point of Driver Oscillation

a direction as to produce the minimum amount of angular movement.

In order to eliminate motion at the beginning of the stroke instead of at the end as just described, and at the same time retain the same direction of rotation for both levers, it will be necessary to carry the lever-pin to the right-hand side of the driving shaft. This necessitates using an overhanging transmitting lever and link, in order to allow the link to pass across the center of the shaft—see Fig. 4. If it is possible to reverse the direction of the driving lever, elimination of the motion at the beginning of the stroke can be achieved by the arrangement shown in Fig. 3.

Fig. 5 shows another arrangement in which the dwell occurs at the mid-point of the swing of the driving lever, so that the driven lever is given an oscillating motion with a dwell at the extreme left-hand position.

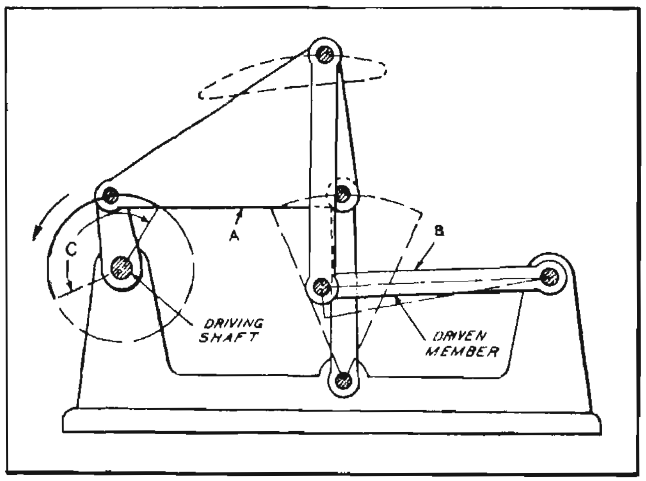


Fig. 6. Mechanism that Allows the Driven Member B to Remain in Position Shown while Crankpin of Driving Arm Passes through Arc C

Driven Lever which Dwells while Driving Crank Turns Part of a Revolution.— A link mechanism which allows the driven member to dwell for a relatively long interval is shown by the diagram Fig. 6. Any point in the flat triangular plate which constitutes the rod *A* can be used for a link connection. For the designing of such mechanisms, it is necessary to study the different forms of curves described by the various points on rod *A*. The forms of the curves traced by the different points on this rod vary distinctly according to their relative positions. To obtain long dwell periods, only the curves that have large circular sections on a part of their outline are used. Geometrical

methods have been evolved to determine such curves, but they are rather complicated. However, some very remarkable improvements in dwell movements have been made by applying such formulas.

There are curves that show a near relationship to circular forms and there are others in which the radii of cur-

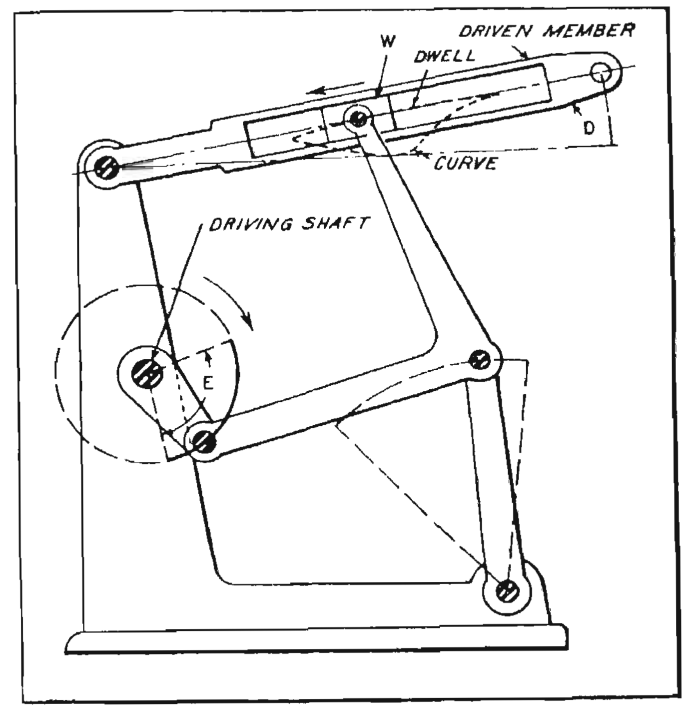


Fig. 7. The Driven Member D Remains in the Position Shown while the Crankpin of the Driving Arm Passes through Arc E

vature will increase to such an extent that the curves practically resemble a straight line. Mechanisms with dwells can be designed, in which the driven member connected with the base of the mechanism has a sliding way, such as shown at *W*, Fig. 7. A member with two vertical slide ways at 90 degrees or any other angle can be used in place of the swinging lever.

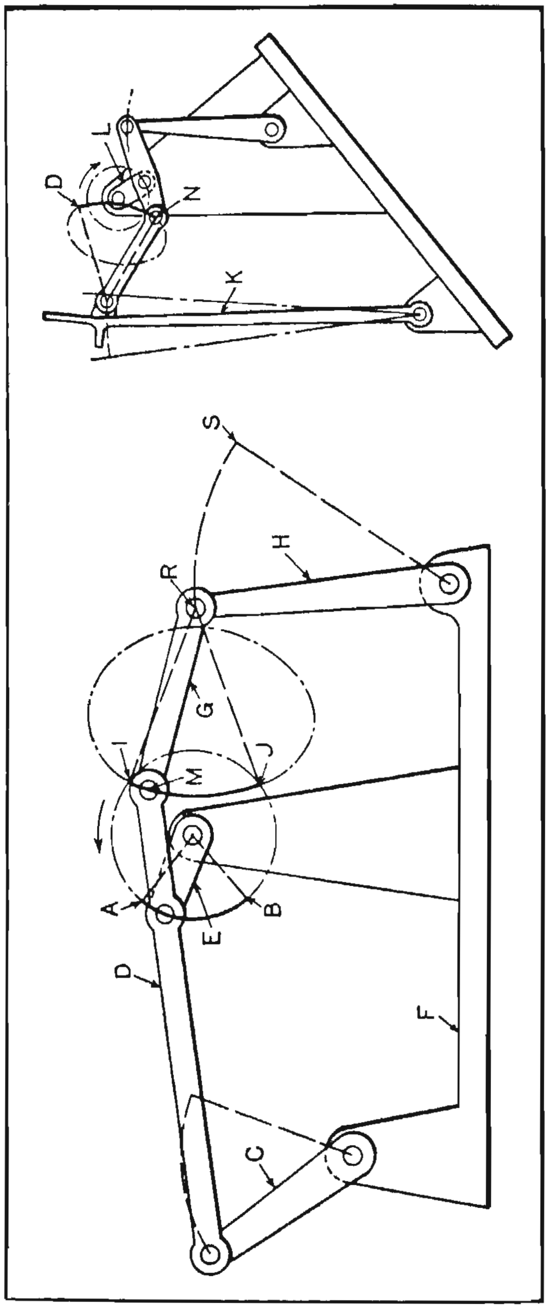


Fig. 8. (Left) Mechanism in which the Link H Dwells while the Crankpin of the Driving Arm E Travels from A to B. (Right) Application of Mechanism Shown at Left to a Textile Machine

Link Mechanism on Textile Machine for Obtaining Dwell in Lever Movement.— Crank-driven mechanisms which also provide lever movements having a dwell period are shown in Fig. 8. The view at the right shows a design applied to a textile machine which provides a dwell period for the lever *K* equal to nearly one-third the period required for a complete revolution of the crank-arm *L*. This mechanism is used on a weaving loom and has proved very successful. The pause or dwell obtained with this arrangement is of sufficient duration to permit the shuttle to pass from one side of the machine to the other.

Referring to the view at the left, which shows the principle of operation, it will be noted that the driving arm *E*, which revolves continuously in the direction indicated by the arrow, is connected to rod or link *D* at a point approximately one-fourth its length from the end connected to link *G* by the stud *M*. Link *G*, in turn, is connected with the upper end of the driven lever *H*, which oscillates through the arc *RS*, dwelling at *R* while the stud *M* travels from *J* to *I*. The lever or arm *C*, connected to the left-hand end of link *D*, also oscillates through an arc as indicated. The amount of dwell and the length of the arc through which the driven lever oscillates depends, of course, on the positioning and the lengths of the links and levers. The use of slide ways for varying the relative positions of the links may be advantageous in some of the many applications for which a mechanism of this kind is adapted.

High-Speed Oscillating Motion with Dwell at Each End.— It is possible, by means of a mechanism consisting of links and gears, but no guides or cams, to obtain an oscillating motion with a dwell at each end of the oscillation. A mechanism of this kind is shown in Fig. 9. This mechanism has two gears, the driving gear *A* and the driven gear *D*. Gear *D* is only half the diameter of gear *A*. It will be noted that the gear shafts are mounted in the fixed base *E* of the mechanism.

The lengths of crank-arms F and G, or the distance of the crankpins of the push-rods H and I from the centers of their respective gear shafts, as well as the lengths of the push-rods, can be of any suitable dimensions. The crank F works in a different phase from crank G. The push-rods H and I are connected by the joint J. If crank F makes one revolution, and crank G, by means of the spur gears,

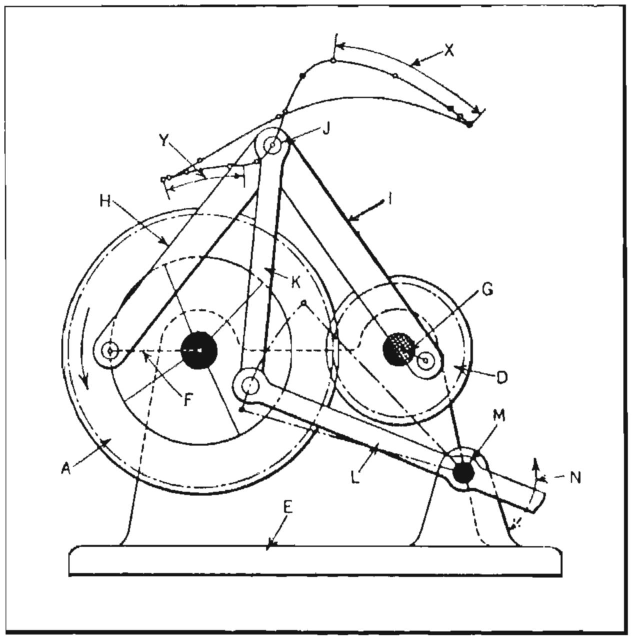


Fig. 9. Mechanism that Causes the Short End of Lever L to Oscillate through Arc N with Dwell at Each End of Arc

makes two revolutions in a reverse direction, joint J will describe a curve having the unusual form indicated.

The curvature is so selected from the possible forms that it has the same radius at the maxima and minima points. If this radius nearly corresponds to the length of member K, which is connected to joint J at one end and to swinging member L at the other, member L will be oscillated about a fulcrum bearing M in the base E and will pause or

dwell while the joint J describes the arcs X and Y. Member L is thus given an oscillating movement indicated by arc N, with a dwell at each end of the arc. The members H and I in the design shown are of identical size. A mechanism of this kind can be used for high speeds.

Straight-Line Motion for Oil Circuit-Breaker.—Straight-line motions are not used extensively, since they are only adapted to certain special conditions, and they have some inherent disadvantages. The particular straight-line motion illustrated by the diagrams Fig. 10 (which represent side and end elevations of the links) produces an approximate straight-line between the two points A and B. This mechanism is part of a heavy-duty oil circuit-breaker. Straight-line and toggle linkages have had considerable application to this type of electrical apparatus.

The pivots of what might be called the main lever are located at A, C, and E. The fixed pivots or hinge points are at D and F. The links CD and EF are free to rotate around their respective fixed pivots or bearings. The line EG indicates a link connecting with a crank or other form of driving member. A force indicated by the arrow at A acts along the path AB, and the triangular lever ACE moves from the "closed position" shown in full lines to the "open position" indicated at B, C_1, and E_1. This latter position represents approximately the lower limits of the main lever movement. The load from A to B varies in a manner characteristic of circuit-breaker operation, and the velocity in each direction also varies from zero to as high as 12 to 15 feet per second in order to operate the brushes fast enough to open or close in from 1/3 to 3/4 second. The load at A may vary from 150 to 3000 pounds during normal operation and may be higher under certain conditions. The line AB is vertical in the apparatus, and the line FP at right angles to AB.

In considering the advantages of this design, first note that the main lever ACE is a triangle with the load at one

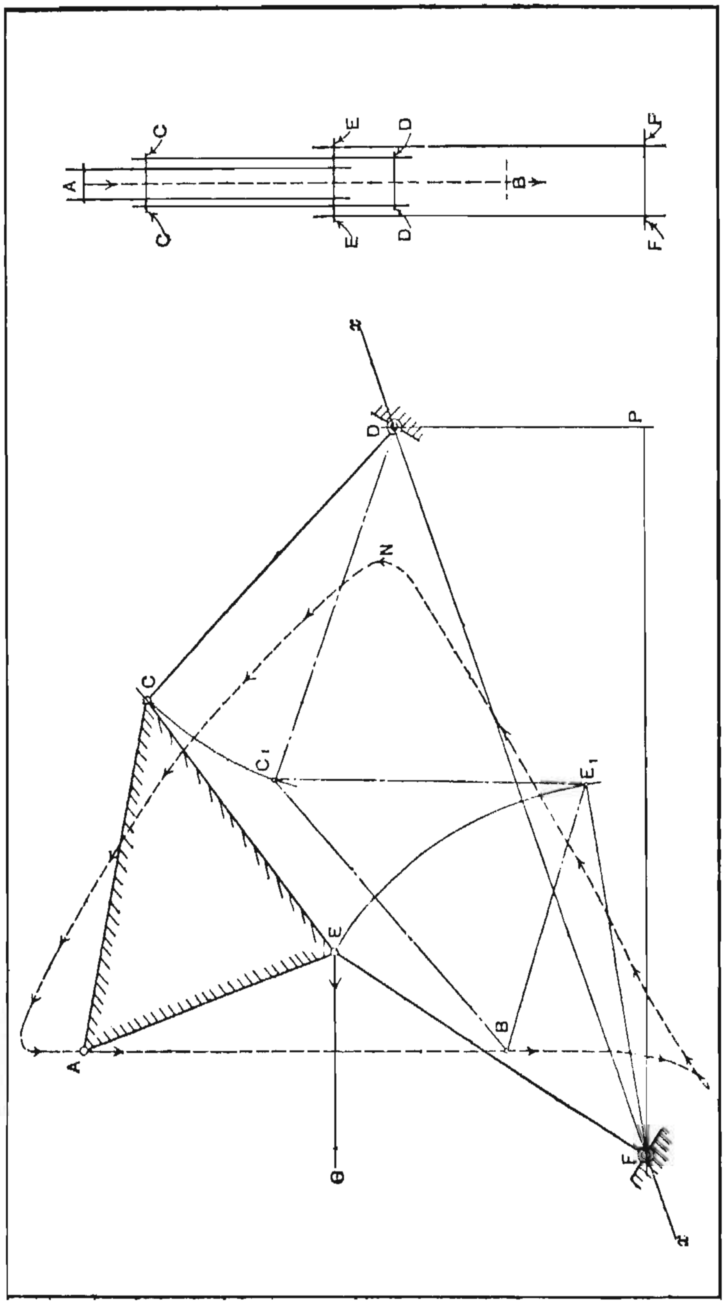


Fig. 10. Diagrams Representing Side and End Elevations of Straight-line Motion for Heavy-duty Oil Circuit-breaker

corner *A*. This form provides maximum strength and rigidity for a given amount of metal, and few straight-line motions, except the more complicated ones, have this feature. Second, the members *AE* and *EF* are struts well located with reference to the load at *A*. These struts are disposed to form the familiar toggle joint. Third, the rod from *A* to *B*, carrying the load, passes about midway between the points *E* and *F*, allowing proper clearances. Morever, the rod is in the center of the double link *EF*, there being a link on each side, as shown by the end elevation; this is also the case with link *CD*. As the result of this construction the pins are in double shear and with practically no bending due to overhang. The line from *A* to *B* deviates only slightly from a true one and is accurate enough for the purpose mentioned. This motion has the advantage that the pivot points and links can be varied at one place and compensated for somewhere else to an extent not possible with a number of other types. All pivot points, however, must be in the proper relation to obtain the most accurate line, although this does not necessarily require the particular arrangement shown.

The real geometrical reference axis of this linkage is indicated by the line x–x. Note that the dotted line and small arrows starting from *A* extend through *B* downward curving to the left and then, after making a small loop, extend upward through *N* and back to the starting point *A*. This line indicates the path which point *A* would follow if the motion were continued beyond point *B* and through a complete cycle.

The four-sided linkage *CDFE* is similar in principle to the skeleton for most of the straight-line and parallel motions from Watts down to the Roberts type. The radius arms *CD* and *EF* are of equal length, but would not need to be, if the main lever *ACE* were changed to an isosceles form of suitable length and the center *D* were swung clear over the top until directly over pivot *F*; then the Roberts compensating motion would result. It will be noted that the path of point A crosses the axis x–x twice. If all the links in the linkage $CDFE$ were of different lengths and if CE were longer than DF, and CD shorter than EF, then a point near the middle of the oscillating link CE would cross the axis six times. This will be recognized as the more general case of the irregular four-sided linkage. When crossed linkages are used, as in Watts and some other motions, the line is crossed only twice. From a practical standpoint this motion has a few decided and inherent advantages.

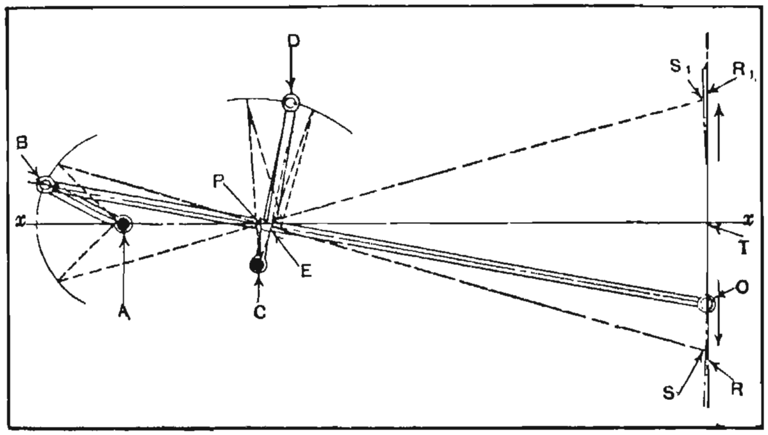


Fig. 11. Diagram of Straight-line Mechanism Used on Granite Gang Saw

Straight-Line Mechanism for Gang Saw.—The mechanical movement shown in the accompanying diagram (Fig. 11) is used in connection with a gang saw for sawing granite, to obtain an approximate straight-line motion with a combination of links. The bearings or pivots A and C are stationary. Link AB is free to turn about bearing A, and CD is free to turn about bearing C. The rigid bar or link OB has an extension ED at right angles to it, which is pivoted at D to the lever CD. As the end O is moved in the direction of the arrows, the pivot B swings about an arc having a radius AB, and the pivot D swings about an arc CD. The resultant movement of the point O is very nearly a straight line.

When this mechanism is applied to a granite gang saw, a slight rise at the ends of the stroke S and S_1 is required, so that links of special length are used. These lengths, in inches, are as follows: $AB = 8\ 1/2$; $OB = 66\ 1/2$; $EB = 22\ 13/16$; $AP = 13\ 1/2$; $CD = 16$; $ED = 12$; $RT = 12$; $R_1T = 12$. The rise at the end of the stroke is 1/4 inch, approximately.

Four mechanisms of this type are used on the granite gang saw. Each mechanism is so located that the center line x–x is vertical, the straight-line movement being horizontal. The machine is equipped with a rectangular "sash" in which there are numerous steel blades. Each corner of this sash is attached to one of these straight-line movements, and the sash is moved back and forth by a crank and connecting-rod. Steel-shot under the blades works against the stone and does the cutting at the rate of from 3 to 6 inches per hour. The object of the slight rise at the ends of the stroke is to allow the grains of shot to fall under the blades as the shot drops down from above. By shortening the dimensions AB and EB equally, the end O can be made to travel in an exact straight line for a certain distance.

Stroke Adjustment for Oscillating Lever.—The arrangement of levers shown in Fig. 12 provides a simple means of transmitting a variable oscillating movement to the lever D from the lever C. The oscillating or up and down angular movement of lever C about center H remains constant, while the oscillating or angular movement of lever D about center J can be varied to suit requirements. Adjustment of the angular movement of lever D is obtained by shifting the position of block A, which is in contact with the two levers. The levers are held in contact with block A by springs (not shown in the illustration). Lever C transmits the smallest angular movement to lever D when block A is in the position shown to the left. The largest angular movement of lever D is obtained with the block in the position shown by the dotted lines at K.

The track surfaces of the levers in contact with block A are set parallel with each other when the levers are at either the top or the bottom of their strokes, according to whether the motion is required to be more nearly constant in speed at the upper or the lower position. In Fig. 12 the levers are shown in their lowest positions. Contact block A can be adjusted by the machine operator to any position along the levers. The adjusting rod B is hinged to the control lever, so that it can swing in a vertical plane as lever C oscillates.

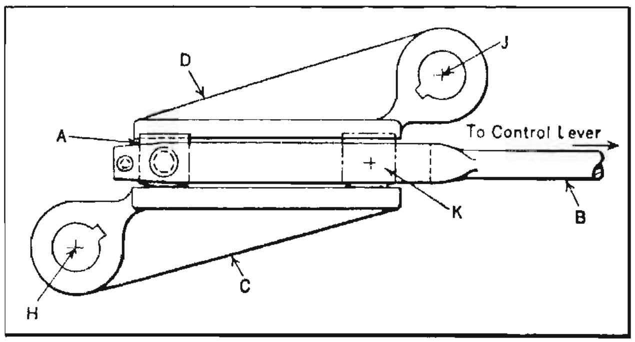


Fig. 12. Arrangement by which a Variable Oscillating Movement is Transmitted by Lever C to Lever D

If A is a single-piece block, it must either have a roller or else have a flat face contact with the track of the upper lever and a lower surface formed like the portion of a cylinder of relatively large diameter. The block and the tracks of the levers must be hardened. Grease lubrication is best where a flood of oil cannot be applied.

When the loads are heavy, sliding friction is likely to develop flats on block A and rough spots on the lever tracks. In such cases, the design shown in Fig. 13 is preferable. With this design, the sliding friction is reduced to a minimum by substituting rollers F and G for the plain bearing surfaces of block A, Fig. 12. On the upper side of the

trunnioned separating block E, Fig. 13, is a single roller F, while on the lower side are two similar hardened rolls G. With this arrangement, only a slight sliding motion is possible between the rollers and the lever tracks, the movement of the rollers being limited by the end flanges of the block. The close fit between the side plates and the rollers, and the squaring effect of the end flanges of block E, tend to keep the rollers parallel.

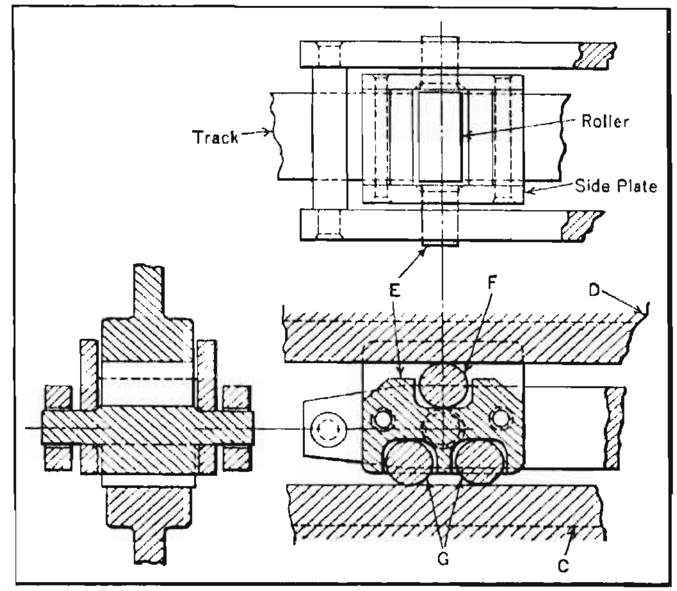


Fig. 13. Roller Bearing Block Used in Place of Solid Contact Block A, Fig. 12, when Lever D is Heavily Loaded

Cam and Rack Mechanism for Increasing the Movement of an Oscillating Lever.—When compactness is essential in a mechanism for producing a long movement with a short lever, the design shown in Fig. 14 may be used to advantage. The lever indicated at G is pivoted to a slide D confined in the guides of the machine. Cast on the lower end of the lever is a gear segment which meshes with a stationary rack H. Movement is imparted to the slide by the continually rotating cam A, which is mounted on a shaft in the stationary bracket C and engages roll F on the lever stud. Coil spring E, fastened to the slide and the machine frame, serves to hold the roll in engagement with the cam.

As the cam rotates in a clockwise direction from the position indicated, the slide D will move toward the right. The lever will, of course, travel with the slide, and owing to the engagement of the gear segment and the rack, will swing in a clockwise direction. The lever will continue to swing

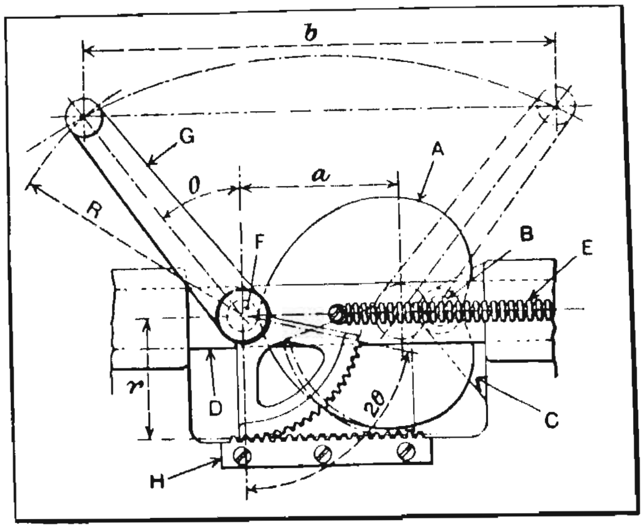


Fig. 14. Combination Cam-and-rack Movement for Increasing the Throw of a Lever

in this direction until the cam has carried the slide to its extreme position at the right. The position of the lever at this point is indicated by the dot-and-dash lines.

In determining the movement b of the upper end of the lever, it is only necessary to add the throw a of the cam to the normal angular movement of the lever, all measurements being taken horizontally. The movement of the lever can be varied by changing the throw of the cam, the pitch radius r of the segment gear, or the length R of the lever itself.

Push-Button Mechanism for Alternately Changing Position of Lever.— Fig. 15 shows a mechanism designed to change the position of lever C from the right to the left of the vertical center-line or vice versa each time the push-button D is operated. The cam-shoe F (left-hand view) which is an extension of push-button D, is in position for changing lever C to the position shown by the middle view, by coming in contact with the inclined face of pawl G when the button is pushed upward. The upward motion of cam-shoe F will cause pawl G to move in the direction of arrow O. When the cam-shoe reaches the end of the incline, the fulcrum of the connecting links J will rest against the flat face of the cam-shoe. Pawl G will then be lifted, causing lever C to swing and change from the right-hand to the left-hand position.

During this motion, the pressure of the shoe applied to the connecting links J automatically moves the link or pawl H toward the left. This obviously prevents pawl G from moving too far to the right and from becoming disengaged from the cam-shoe. The right-hand view shows the push-button D released and returned to its normal position by spring K. When the connecting links J are released from contact with the cam-shoe, the spring L causes pawls H and G to move toward each other until links J come in contact with pins M and N. Lever C is now ready to change from left to right when the push-button is again operated. The stem of the push-button D is square and is a sliding fit in the square hole in guide E.

Multiplying Action of Lever for Obtaining Quick-Acting Brake Movement.— Fig. 16 shows the construction of a mechanism designed to provide more than the customary amount of clearance for a brake-shoe without sacrificing any of the braking effect. This is accomplished by a system of levers that provide for a quick take-up of the clearance space, after which the brake movement is effected in the usual manner.

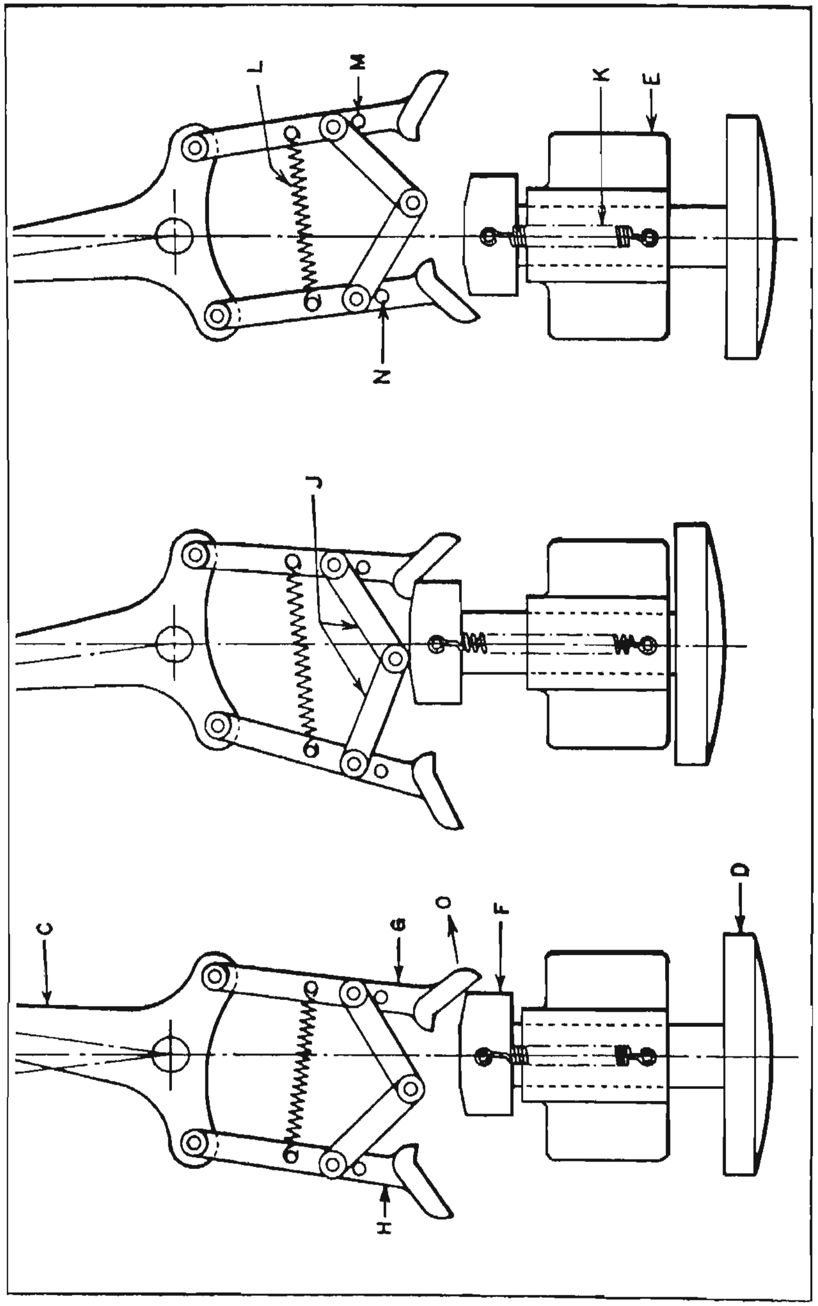


Fig. 15. (Left-hand View) Push-button Mechanism with Lever C in Right-hand Position. (Middle View) Mechanism with Push-button Pushed in to Transfer Lever to Left-hand Position. (Right-hand View) Push-button Released with Lever Remaining in Left-hand Position

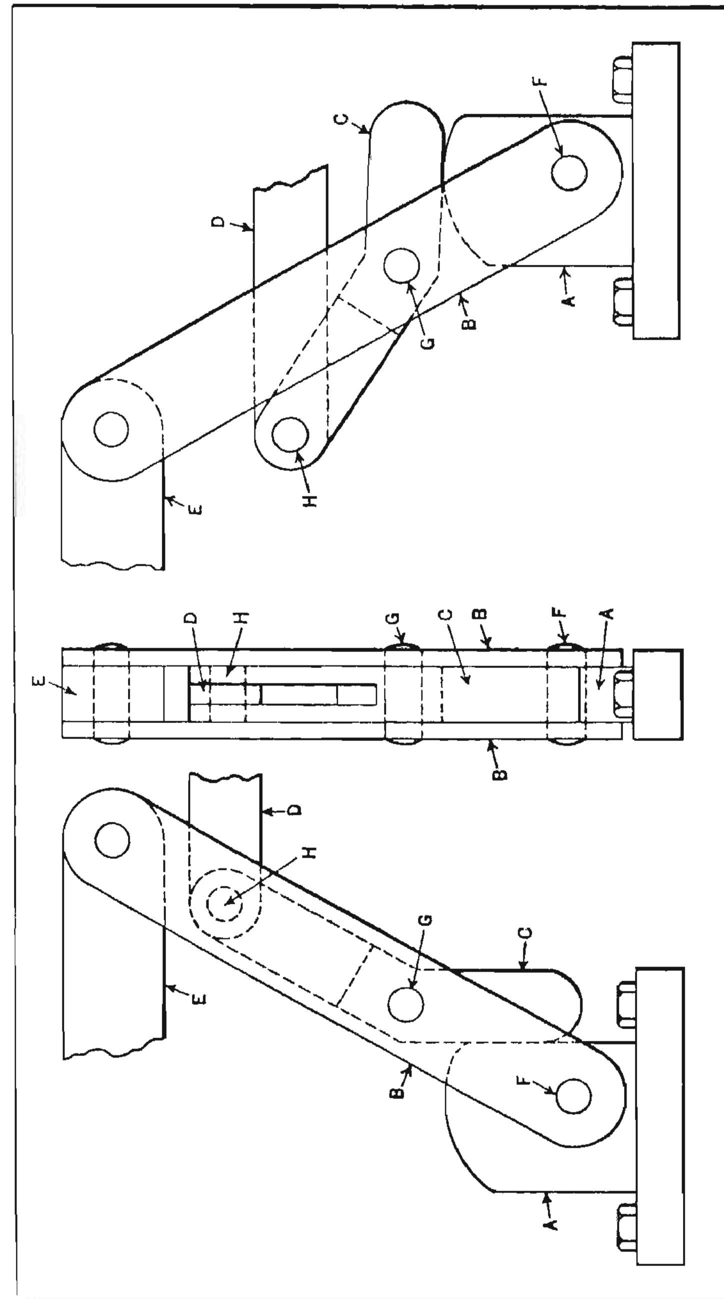


Fig. 16. Quick-acting Cam-and-lever Mechanism for Operating Brake-shoe

Part A is fastened to the stationary part of the machine and carries the pin F on which the double levers B pivot. The upper surface of part A is machined to conform with the arc of a circle of which pin F is the center. It will be noted that the pin F is located off center in part A and that the upper edge of part A terminates in a small arc-shaped surface on the right-hand end. Levers B carry between

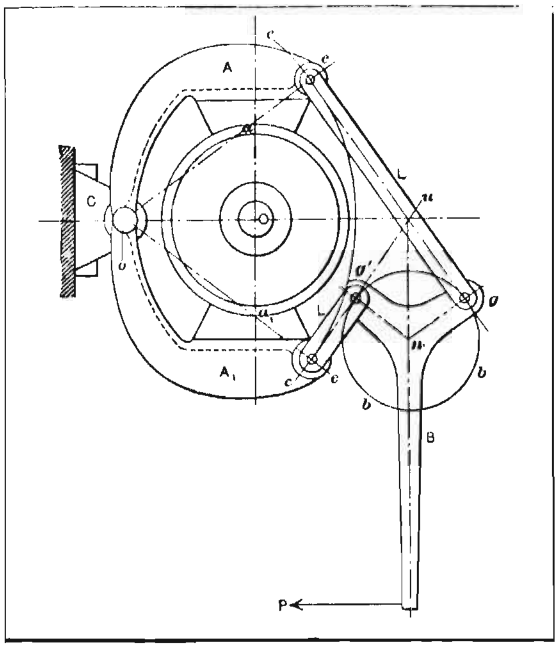


Fig. 17. Arrangement of Links and Levers for Operating the Clam-shell Type of Block Brake

them lever C which fulcrums on the pin G and carries at its upper end the bar D attached to it by the pin H. The opposite end of bar D is attached to the brake-shoe.

The side view at the left and the end view at the center show the arrangement with the brake-shoe in the released position. As the bar E is moved to the left, lever B fulcrums on pin F, and the upper right-hand corner of part A acts on lever C, which is caused to fulcrum on pin G. As the movement of the lower end of lever C is multiplied at the upper end, bar D is drawn forward quite rapidly in advance of levers B, thus quickly reducing the clearance space between the brake-shoe and the drum. As soon as the lower end of lever C has passed over the corner of part A, lever C ceases to act independently, and moves in unison with levers B.

Lever Mechanism for Block Brake.—The type of block brake known as the "clam shell" brake, Fig. 17, is often used in place of the band brake, over which it possesses the advantage of even wear on the blocks, and positive release, although not possessing as great a gripping power. The cast arms A and A_1 are pivoted at o to the frame of the machine, and carry blocks formed to grip the brake wheel. Links L connect these arms to the bellcrank B, having the floating center n. To lay out this brake to the best advantage, draw lines from o through the center points of contact a and a_1, on the rim of the wheel; also with o as a center, draw arc cc, intersecting these lines at points e. At these points, draw tangents to arc cc, intersecting at u, and draw un, bisecting angle gug'. Select a point n on un for the center of circle b, drawn tangent to eg, so that the required leverage will be obtained for the brake system.

When the brake is new, the exact nature of block contact is doubtful, and must be considered as only a line across the face of the blocks, but the wear on the blocks causes such a condition of pressures per unit area that the rate of wear is the same at all points of contact between the block and the drum. The point o may be placed below the wheel, making the axis aa_1 horizontal, the arms A and A_1 falling apart by gravity, when released. When the arms are not heavy enough to do this without one of them bearing against the wheel, while the other is free, light springs may be attached to points e to keep them apart, when released. The arms are sometimes extended so that points e may be connected by a spring which sets the brake, the release being made by toggles separating the arms when applied. The wheels of these brakes may be made V-shaped, the same as for band brakes. The blocks are often made to embrace a larger portion of the wheel than shown—sometimes nearly 180 degrees.

In Fig. 18 are shown two types of this brake, the fixed points being indicated by a dot within a circle, and the floating points by a plain dot. At *A* is shown a form of brake that is useful when there is no convenient way of pivoting the arms to the frame at *u*. The bellcranks *ace* and lever *mn* are pivoted as shown, but the point *u* is fixed in space only by its geometrical relations to points *a*. Since

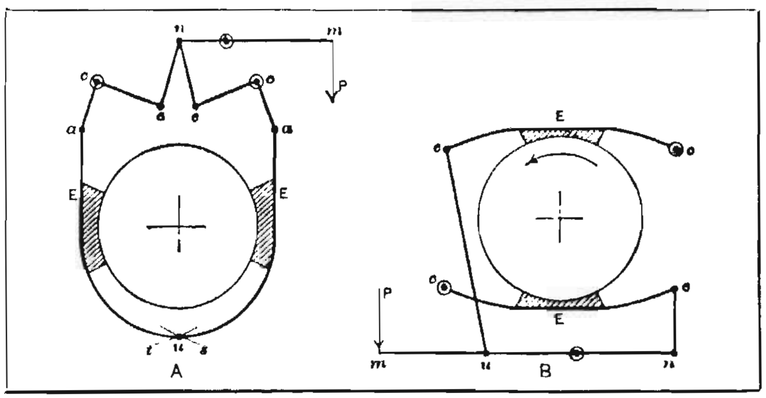


Fig. 18. Other Methods of Operating Block Brakes

the arcs *s* and *t*, struck from the points *a*, cross at *u*, it is evident that the point *u* becomes fixed in its relation to points *c* where the system is connected to the frame, and thus *u* is the fulcrum of the arms *E*, although not the point which receives the thrust of the brake arms, this being taken at points *c*. At *B* is shown a good type of brake in which both arms act as tension members in transferring the braking force to the fixed points *c*.

Safety Locking Device for Clutch Lever.— Safe operation of an extractor used for drying wiping cloths requires that the cover of the machine be tightly closed before the starting clutch is engaged. One device which complies with

this requirement is shown in Fig. 19. The stationary casing *A* and the cover *B* enclose the rotating container (not shown) for the cloths. The cover is hinged at the left-hand side of the casing and can be secured in its closed position

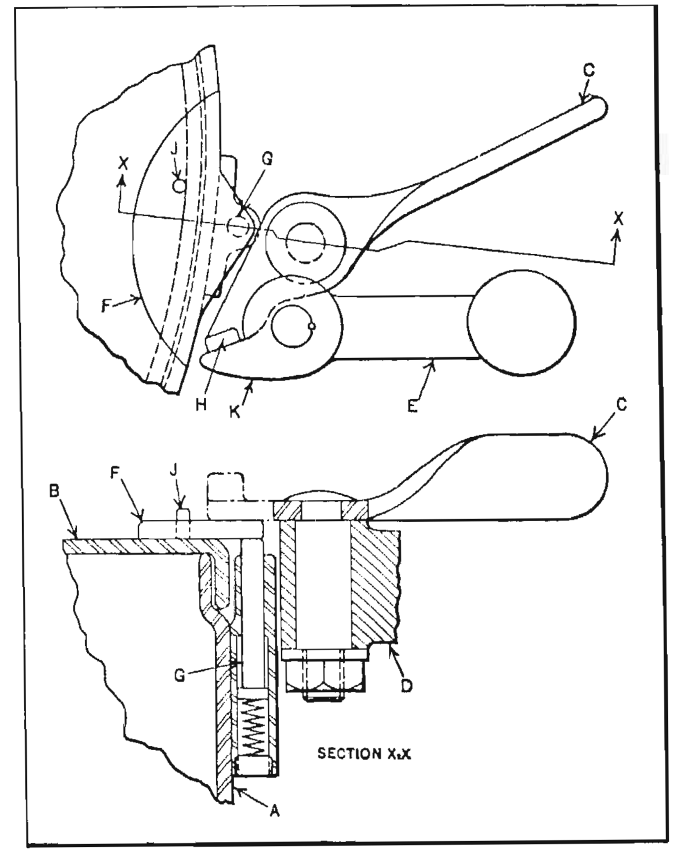


Fig. 19. Safety Locking Device which Prevents Accidental Shifting of the Clutch Lever

tion by the clamping lever *C*, pivoted on the stationary bracket *D*.

In the position shown, the cover can be swung upward on its hinge to allow loading or unloading of the container. While the cover is up, clutch lever *E* is prevented from being returned accidentally to its engaged position by the projection H on the lever C. Lever C, in turn, is prevented from returning to its clamping position by the spring-actuated plunger G, mounted on the casing. With this arrangement, both levers are automatically locked when the cover is up and are automatically released when the cover is in its closed position. As the cover descends, plate F, which is welded to the cover, depresses plunger G and allows lever C to be swung into its clamped position. At this time, finger K will clear the projection H, permitting lever E to be swung in position to engage the clutch. Plate F is provided with a stop-pin J to limit the clamping movement of lever C.

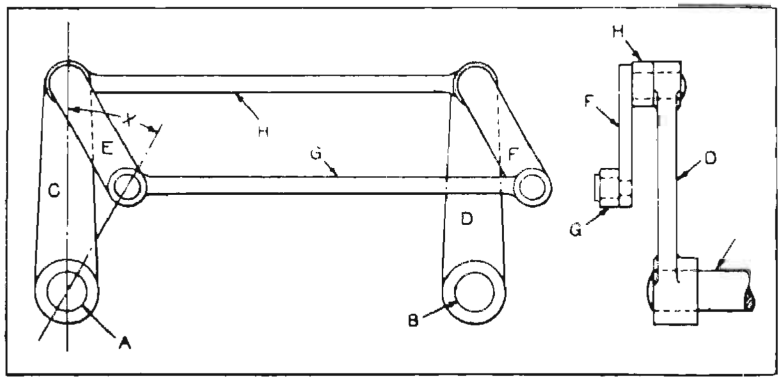


Fig. 20. Crank Motion with Auxiliary Rod that Eliminates Dead Center Effect

Crank Motion with Dead Center Eliminated.— When a rotary motion is transmitted from one shaft to another by means of cranks and a connecting-rod, the dead center positions may be avoided by the arrangement shown in Fig. 20. This mechanism is for a wire-forming machine and transmits power between two shafts located some distance apart. The purpose of the auxiliary rod G is to carry the driven shaft past the dead center positions. The question may be raised as to why a crank motion is used when a chain drive would produce the same effect. The reason is that on the machine in question, a reciprocating part of the machine passes into the space between the two shafts while the connecting-rods are "running over," or passing through the upper half of their cycle of rotation, withdrawing as the rods approach the center position. Obviously, this arrangement would be impossible with a chain or gear drive, which remains in the same position at all times.

Referring to the illustration, the shafts A and B carry the crank-arms C and D, respectively, which are connected by the rod H that runs free on its crankpins. In the actual installation, rods G and H were longer than shown. The length of these rods, however, does not affect the operation of the drive. The upper crankpins, which are keyed to the crank-arms C and D, carry the auxiliary arms E and F, which are set at an angle with arms C and D. Connecting-rod G, which is exactly the same length as rod H, connects arms E and F. Although this arrangement may be classed as being without a dead center, it really has two dead center positions, but there is a time element between the two which renders them both ineffective in arresting the driving motion. When one crankpin reaches dead center, the other is still approaching and is effective in forcing the first past the dead center.

It is essential that each pair of similar parts be of exactly the same length; otherwise, there will be a binding action. Although the length of arms E and F should be kept as short as possible for the sake of compactness, they should not be less than one-half the length of arms C and D. The movement will operate without any dead center effect over a wide range of positions for arms E and F, although the smoothest movement seems to be attained when the angle X is not less than 20 degrees.

CHAPTER XIV

FEEDING MECHANISMS AND AUXILIARY DEVICES

The expression "feeding mechanism" may indicate mechanical means of presenting parts successively for some manufacturing operation or this term may be applied to a mechanism for imparting a feeding movement to a metal-cutting or other tool. This chapter deals with feeding and allied mechanisms of various types and designed for miscellaneous application. It supplements the four chapters in Volume I which deal with this general subject (pages 447 to 519).

Hopper Feeding Mechanism Used in Soldering Fuse Plugs.— In manufacturing electrical fuse plugs, such as those used for house circuits, a thin fuse strip is soldered to a split rivet that has previously been assembled into the plug. The soldering is done on a special indexing dial machine in which the solder slugs are automatically dropped on the rivet on the inside of the plug. In another position of the dial, the plug dwells under a concentrated gas flame long enough to melt the solder slug. At the next station the operator places one end of a fuse strip in the molten solder. The dial is then indexed to another position where the plug is automatically ejected from the machine.

A detail view of the mechanism for automatically feeding the solder slugs is shown in Fig. 1. The dial indexes intermittently and is shown with a plug in position to receive two slugs which are held up in the end of the tube *B* by the stop *G* riveted to the arm *F*. This arm swings on the pin *I* and is normally held in position by the coil spring *C*.

The hopper *A* is fastened in the boss *D* of the machine by

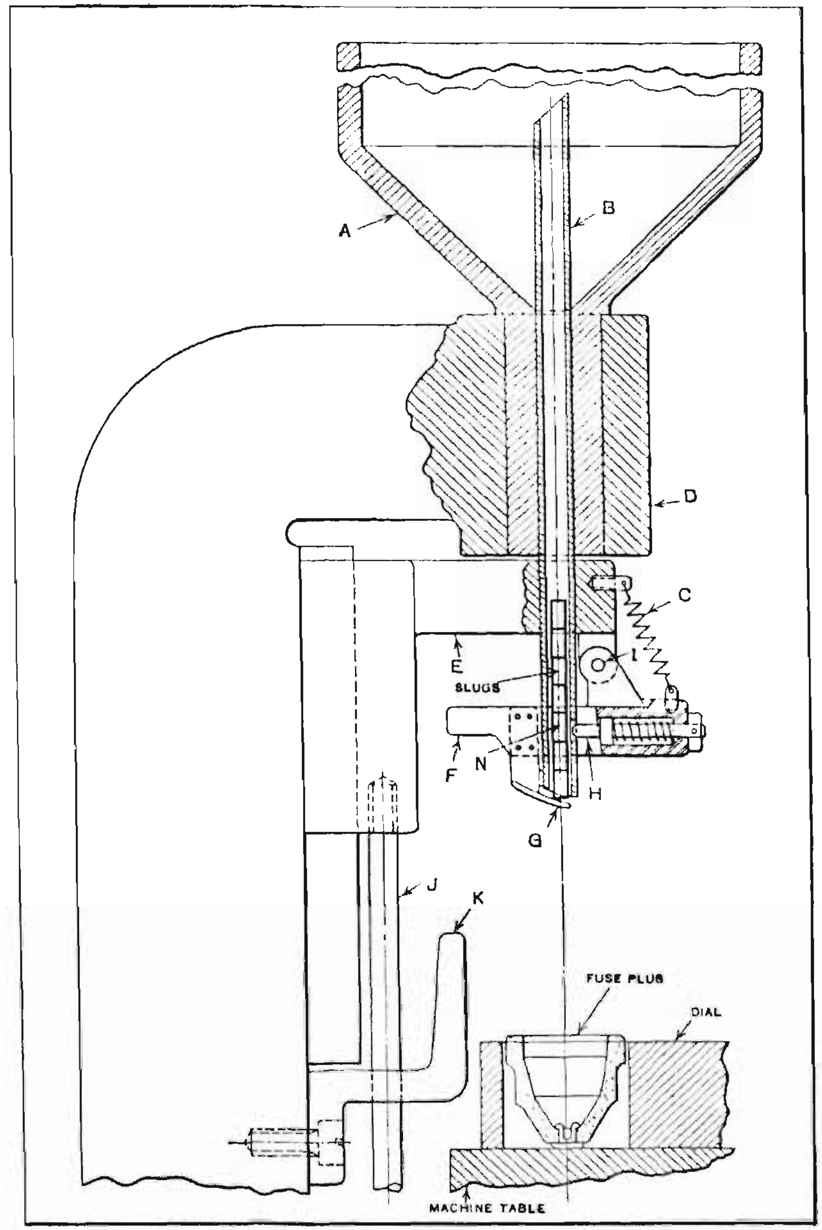


Fig. 1. Hopper which Delivers Two Solder Slugs to Each Fuse Plug in Soldering Machine Dial

a set-screw. Slide E carries arm F and tube B, and receives a vertical reciprocating movement through rod J, from a cam located on the machine. The slide completes one cycle during the indexing of each station. The top of tube B is cut at an angle to produce greater agitation of the slugs in the hopper. The slugs, collecting in the tube as the latter passes through the supply in the hopper, drop down on stop G.

At the end of the down stroke of slide E, the stop K engages projection F, swinging the latter up to the left. This motion withdraws stop G from the end of the tube and allows the two bottom slugs to drop out and into the fuse plug. Just before this happens, however, the spring-actuated plunger H forces the slug N against the side of the tube, holding back the flow of slugs until the return stroke of slide E disengages members K and F, permitting stop G and plunger H to return to the position shown in the illustration.

Agitating Device for a Pin Hopper.—Difficulty was experienced by a plant manufacturing electrical switches in maintaining a sufficient flow of switch pins from hopper to power press. The pins are made from brass rod and are about 3/16 inch in diameter by 3/8 inch long. The hopper is of simple design, consisting chiefly of a stationary conical shell in which the pins are placed. A length of tubing having an inside diameter slightly larger than the diameter of the pins is a slip fit in a vertical hole bored in the lower end of the hopper; this tubing is given a vertical movement, which is transmitted from the press crankshaft through a rack and pinion.

As the end of the tube passes through the pins, some of them drop into the tube and down to a connecting chute, which carries them to the press. It was found, however, that the pins had a tendency to collect around the tube horizontally instead of vertically, so that the number entering the tubing during each stroke was insufficient to supply the

press. After some experimenting, this difficulty was overcome by the use of an agitating device, as shown in Fig. 2.

Bracket A is stationary, the upper end (not shown) supporting the conical shell with its tube while the lower part carries the agitating device. The tube is secured to the sliding bar B, both parts being raised to the upper position by means of the reciprocating arm C pivoted to the cross-head D. A double-end latch E is also pivoted to the cross-head, and engages teeth cut in bar B, as well as projections on the plate F, which is secured to bracket A.

In the position shown, the cross-head has already started its upward stroke, carrying with it bar B and the tube. Further movement of the cross-head causes the upper end of the latch to be forced to the left by the lower projection on plate F. At the same time, the lower end of the latch is forced out of engagement with the lower tooth in bar B and allows the bar to drop back by gravity, aided by the action of coil spring G. The lower end of the latch is held away from the bar momentarily only; thus, as the upper end passes the projection on plate F, its lower end at once enters the next tooth space in the bar B, which then continues its upward movement until the latch engages the next projection on plate F. At this time, and for each remaining projection, the action described is repeated.

At the end of the upward stroke, bar B is held suspended by the latch, but upon starting its downward stroke, the latch again comes into contact with the upper projection on plate F and is released from the bar, allowing the latter to drop to its lowest position. Here the slide is picked up once more by the latch after the cross-head returns to the bottom of its stroke. Incidentally, a bumper (not shown) is provided to take the shock upon the return of bar B. A series of reciprocating movements is imparted to the tube as it passes upward through the pins, agitating them sufficiently to cause a greater number to enter the tube. It should be mentioned that the success of a device of this

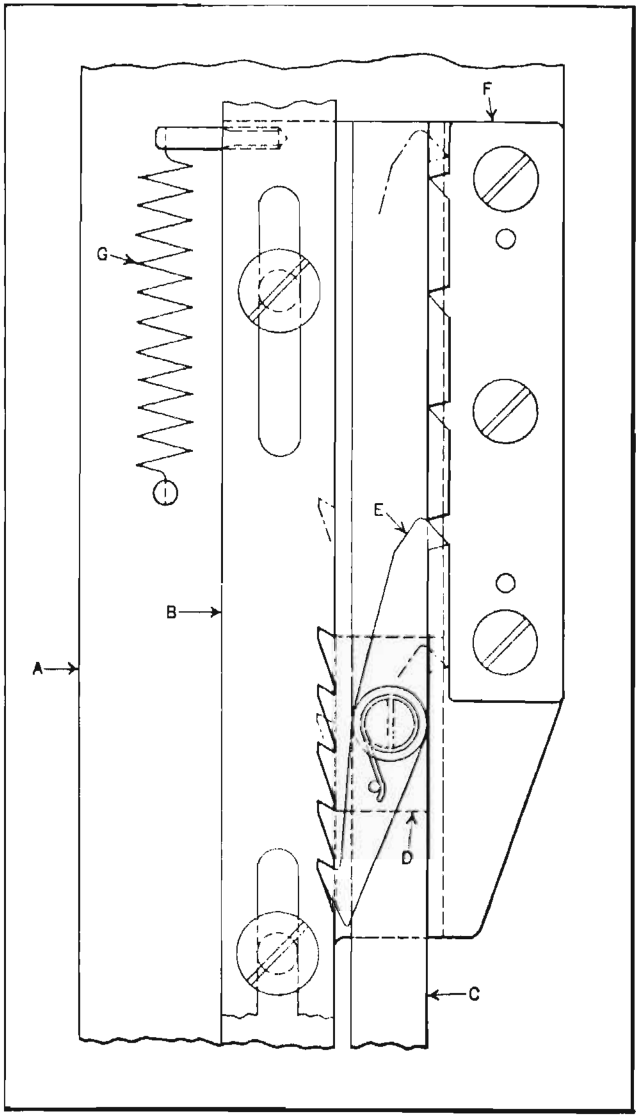


Fig. 2. Device which Increases the Flow of Work from a Hopper by Imparting Short Reciprocating Movements to the Feeding Tube as it Passes through the Parts

type depends upon the speed of the cross-head, because if the speed is too slow, the lower end of the latch will fail to catch the tooth in the sliding bar, thus rendering the device inoperative.

Mechanism for Feeding Granular Material Uniformly.— One of the methods used in a stamping mill for feeding

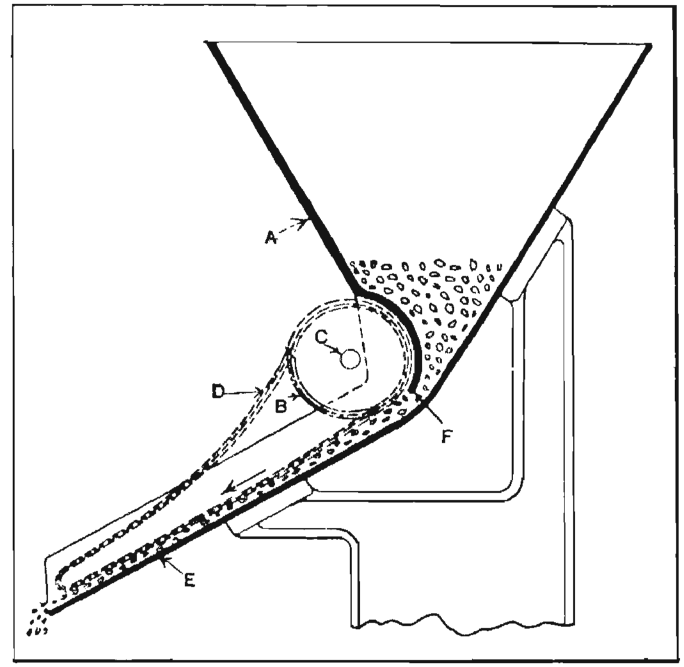


Fig. 3. Maintaining a Constant Flow of Ore from a Hopper by Means of Endless Chains

crushed ore at a uniform rate to a grading machine is shown in Fig. 3. The ore, which consists of pieces about the size of an egg, is dumped into the hopper A and passes through the opening F. As the ore must be delivered from the chute E at a uniform rate, some means must be provided for regulating its flow. This is accomplished in the following manner: A number of endless chains D passing over the sprockets B rest upon the ore as it flows down the chute.

If there is no movement of the chains, the friction resulting from the weight of the latter will prevent the ore from sliding down the chute. If motion is imparted to the chain so that it will travel at a constant speed in the direction of the arrow, the ore will be carried downward at a uniform rate and at approximately the same speed as that of the

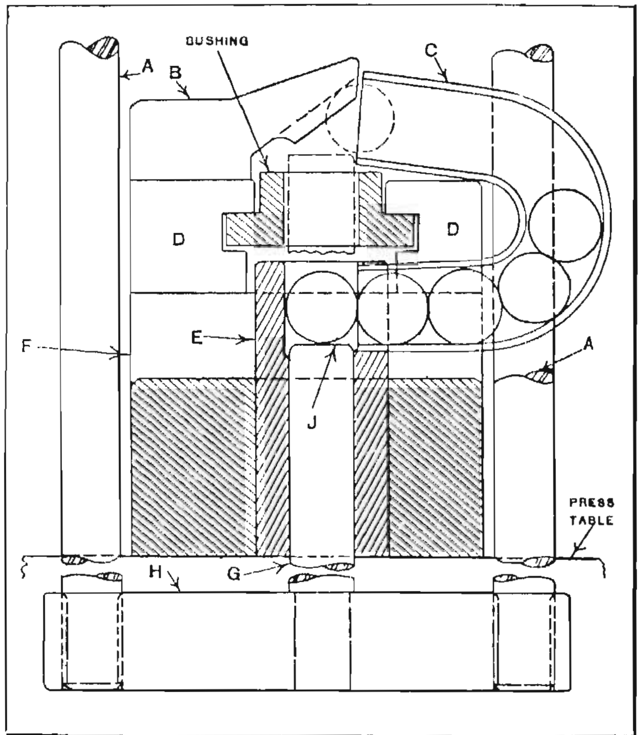


Fig. 4. Simple Press Fixture for Automatically Burnishing Bushings

chains. The movement of the chains is obtained by revolving the sprockets *B* on the shaft *C*, the latter being driven from the driving shaft of the grading machine. The nature of this ore is such that it will flow very freely, and little trouble is experienced from jamming at the mouth of the chute.

Automatic Ball-Feeding Attachment for Ball Burnishing.— Steel balls are frequently employed for burnishing holes when a very fine finish and an accurate job are required. With the automatic ball-feeding device shown in Fig. 4 the work can be burnished in a power press. The bushings to be burnished are fed into a chute, the end of which is shown at *D*, and carried down by gravity to a position directly over the reciprocating plunger *G*. When the plunger is at its lowest point, one of the balls in the return tube *C* rolls on the end *J*, directly under the hole to be burnished. As the plunger ascends, it pushes the ball up through the work. Continuation of this upward movement carries the ball against the angular surface of the block *B*, and into the return tube *C*, as indicated by the dotted outline of the plunger and ball when at their highest position.

The reciprocating motion of the plunger is derived from the ram of the press through the connecting posts *A* and the plate *H* in which the plunger is a drive fit. A clearance hole in the bolster plate is necessary to allow a through passage of the plate *H*. The bushing *E* is a drive fit in the fixture *F* and a slip fit for the plunger. One side of this bushing has an opening for the lower end of the return tube. This tube may be fastened by straps to the fixture, or it may be soldered to the bushing *E* and to the chute *D*.

Although not shown, the usual provision must be made for tripping the press clutch in case the work becomes jammed in the chute or fails to line up properly with the plunger *G*. One advantage of this fixture is that the wear incident to burnishing is distributed equally among a number of balls.

Chain Feed Mechanism with Periodically Accelerated Motion.— Fig. 5 shows a mechanism used for feeding tree trunks *T* to a sawing machine with a periodically accelerated motion that might also be applied to other machines. This motion is obtained by an interesting arrangement for simultaneously taking up slack in one side of the chain while giving out slack in the other side. The mechanism is driven by the spur gear *A* which drives gear *B* by means of an intermediate gear *C*, thus driving chain *D* at a uniform speed. The chain is carried over four pulleys in the base of the mechanism. The pulleys *E* and *F* are attached to a swinging arm *G*. This arm is oscillated by a crank and rod mechanism *H* and *J*, also connected to the driving gear *A*.

The gear *K*, driven by chain *D*, imparts the periodically varying feed motion to the rollers *M*, in contact with the tree trunk, by a second chain drive indicated by the light dot-and-dash lines at *L*. If pulleys *E* and *F* remained stationary, the feeding motion would be uniform. However, as the pulleys are on the periodically swinging lever *G*, the motion of chain *D* is changed in such a manner that it will remain stationary during one short period and will be accelerated or retarded in the other. The swinging lever *G* is slotted so that rod *J* can be adjusted to give the feeding motion required. A cam-actuated motion could be substituted for the crank motion obtained by crank *H* and rod *J*, the cam being given the profile necessary to obtain the desired motion.

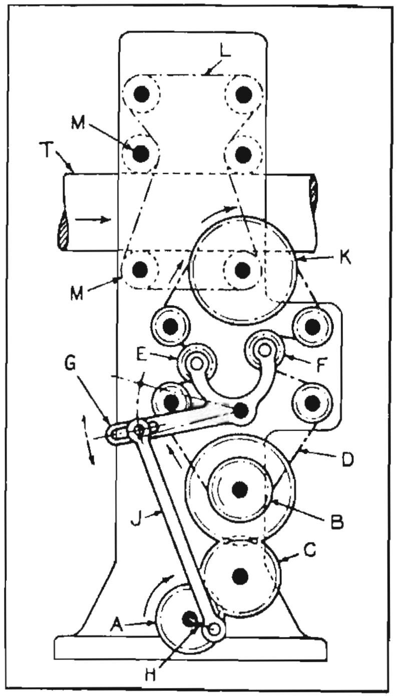


Fig. 5. Mechanism for Producing Periodically Accelerated Motion

Planetary or Differential Type of Feeding Mechanism for Internal Grinder.—The eccentric and differential type of feed controlling mechanism which is shown by a perspec-

tive view, Fig. 6, to illustrate the arrangement clearly, has been applied to internal grinders of planetary design, used for grinding holes in parts of such bulk that rotation is impracticable. The radius of the path of the grinding wheel, which has a planetary motion, is changed while the wheel is at work by an adjusting movement that is transmitted through differential gearing. The grinding wheel spindle 1 is located eccentrically in a cylindrical member 2, which is rotated to vary the radial position of the wheel. Center line *A* represents the axis of the main body 6 of the grinder head; center line *B* is the axis of cylindrical part 2; and *C* represents the axis of the grinding wheel spindle. The distance from *A* to *B* equals the distance from *B* to *C*, so that by turning part 2, axis *C* can be made to coincide with axis *A*, thus permitting the wheel to be located anywhere from a central position to its maximum position radially.

When the grinding wheel has been adjusted for a given cut, it has, in addition to rotation about its axis, a planetary movement about axis *A* of the grinding head. This planetary motion is obtained from the driving shaft which rotates head 6 through gears 9 and 10 at one end, and 5 and 8 at the other. The driving gears 8 and 9 are the same size, and the driven gears 5 and 10 are also equal in size; consequently, these two sets of gearing normally rotate at the same speed, but when a feeding movement of the wheel is required, gear 8, through an adjustment of the differential gearing located between gears 8 and 9, is caused either to lag behind or advance, thus shifting eccentric 2 through worm-gearing 3 and 4 and a screw gear which meshes with teeth on the inside of gear 5.

The action of the differential gearing will be explained in connection with Fig. 7 which shows a cross-sectional view. When the driving shaft *G* revolves, pinions *P* (14 in Fig. 6) which are mounted on studs fixed in a stationary housing of the differential, also revolve. This rotation of the pinions is transmitted to the internal gear *Q* (12 in

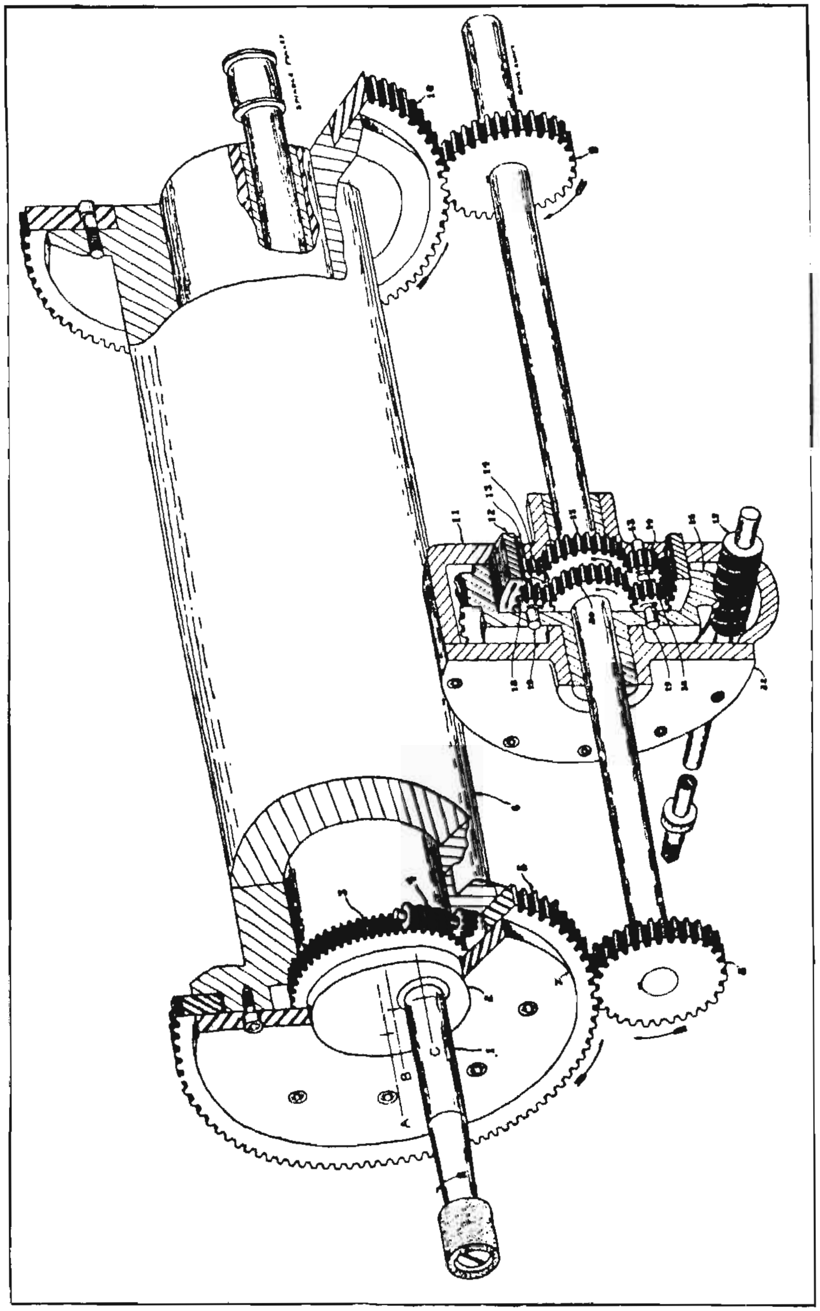


Fig. 6. Perspective View of Eccentric Feed Mechanism which is Operated by the Adjustment of Differential Gearing

Fig. 6) which is free to turn within worm-gear *R* (16 in Fig. 6). As internal gear *Q* revolves, it drives pinions *S* (18 in Fig. 6) which are mounted on pins fixed in worm-gear *R*. Pinions *S* rotate the left-hand section G_1 of the driving shaft at the same speed as the right-hand section, except when a feeding movement occurs.

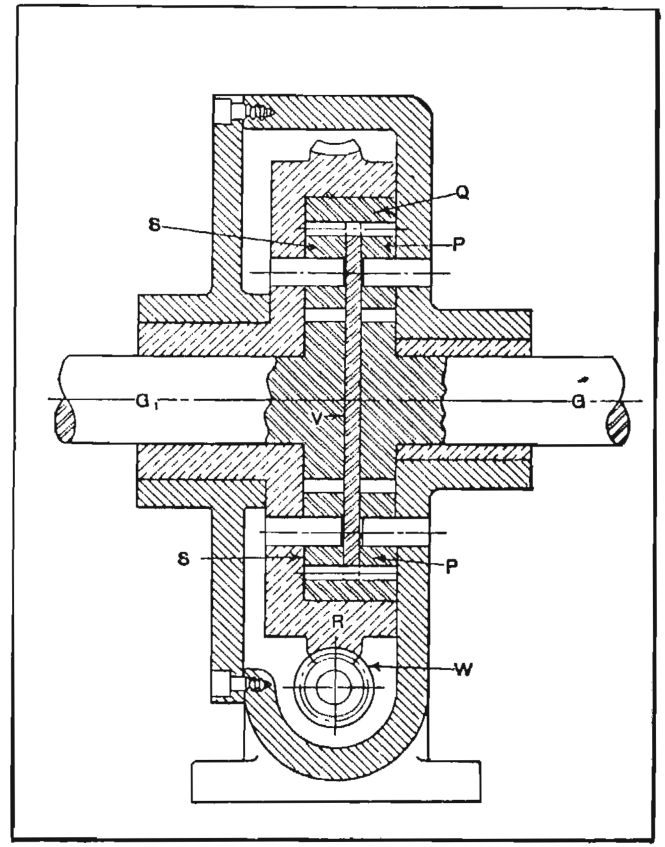


Fig. 7. Sectional View of Differential Gearing of Eccentric Feed Mechanism

For adjusting the grinding wheel in or out, worm *W* is turned by hand, thus turning worm-wheel *R*, which changes the position of pinions *S* relative to pinions *P*. If pinions *S* are advanced or moved in the direction of the rotation of

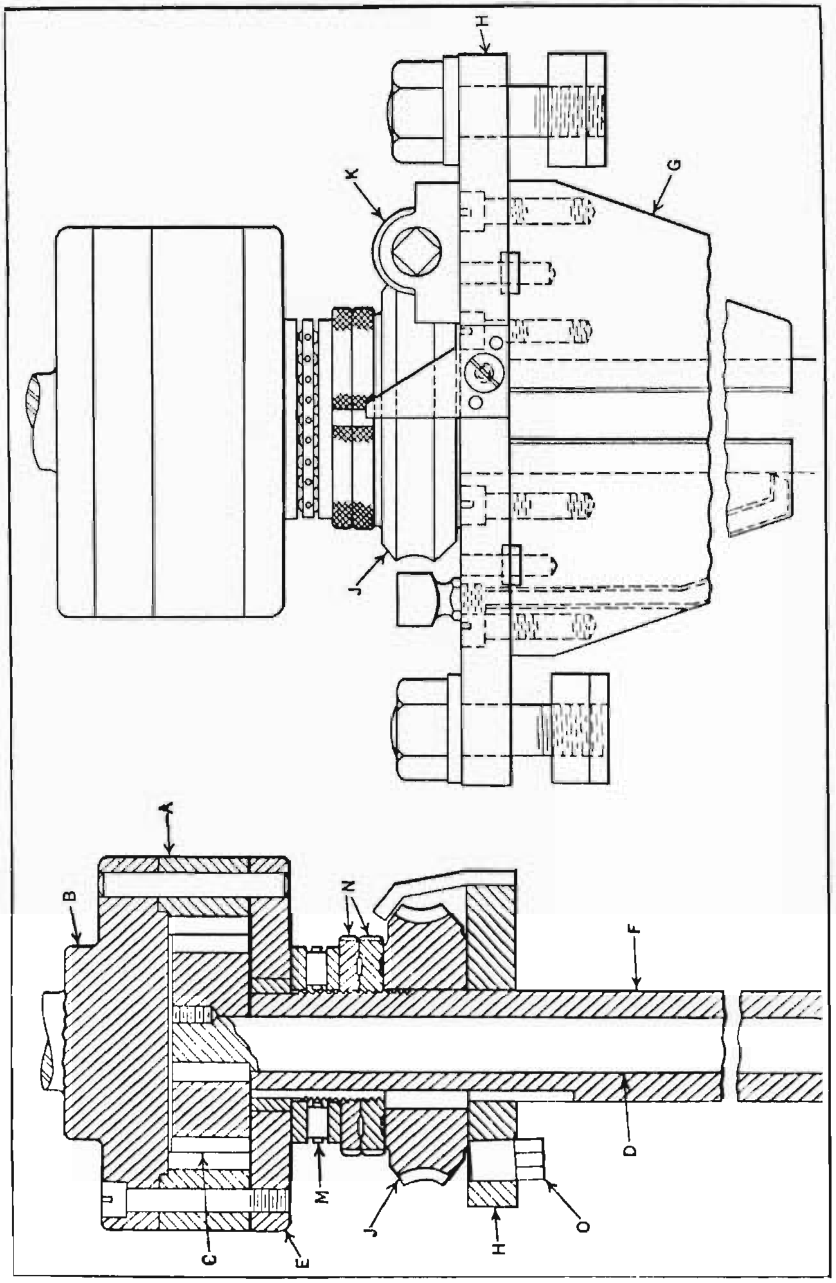


Fig. 8. Recessing Tool with Planetary Feed for Milling an Irregular Recess at a Depth of 8 1/2 Inches

internal gear Q, then during this period of adjustment, shaft G_1 will turn somewhat slower than G; consequently, there will be a movement of gear 5, Fig. 6, relative to the main head, thus causing worm-gearing 3 and 4 to rotate and changing the position of the grinding wheel. On the other hand, if the rotation of worm-wheel R is such as to move pinions S in the direction of the rotation of internal gear Q, the speed of shaft G_1 will be accelerated relative to G, thus adjusting the grinding wheel in the opposite direction. A spacer plate or disk V (Fig. 7) is located within the internal gear Q and between the two sets of gearing. The internal teeth of gear 5 (Fig. 6) were cut on a lathe, the indexing being done by disengaging the feed-screw

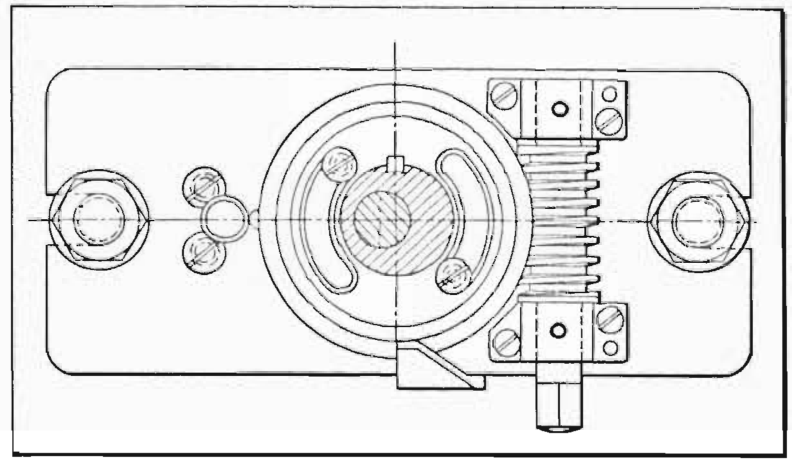


Fig. 9. Plan View of the Recessing Tool, Showing How the Cutter Shaft is Fed toward the Work by an Eccentric Sleeve

gears, and the teeth of the screw gear which engages the internal teeth of gear 5 were cut on a horizontal universal boring mill. The cutter used was a duplicate of the internal gear and formed the teeth by a generating action.

Planetary Adjustment for Feeding Deep-Hole Recessing Milling Cutter.— An ingenious application of planetary gearing to tool design is shown in Figs. 8 and 9. This tool is used in a radial drilling machine for milling, at a depth of 8 1/2 inches, an irregular recess in the ports of a steel forging, as indicated at *A* in Fig. 10. Internal gear *A* (Fig. 8) is fastened to the shank *B*, and meshing with this

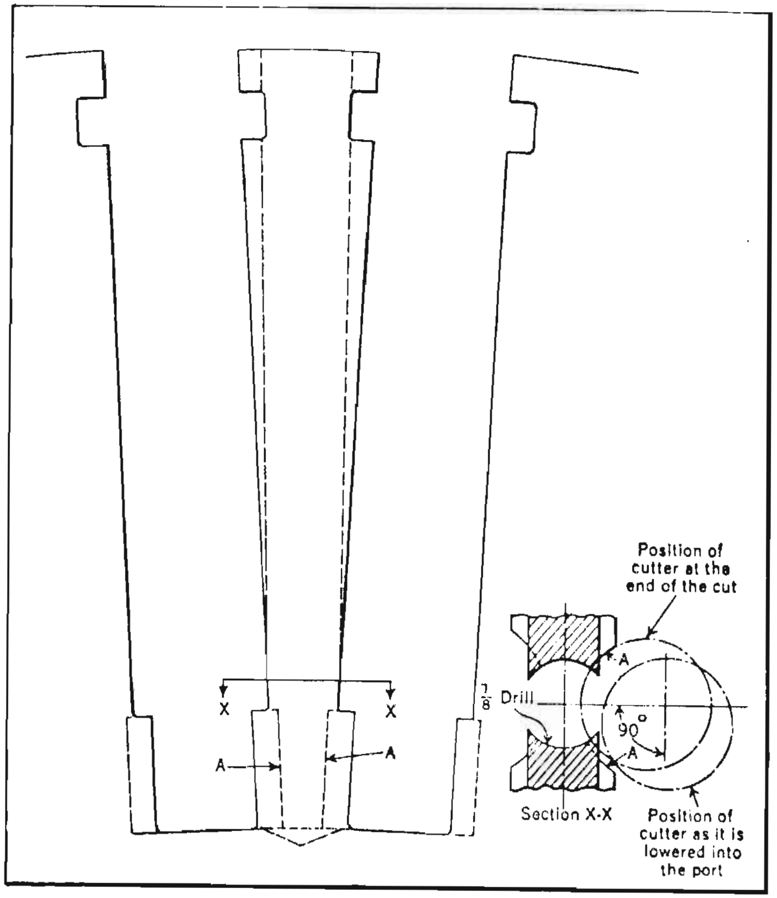


Fig. 10. Diagram of Work, Showing the Irregular Recess Cut by the Tool in Fig. 8

gear is the pinion *C*, secured to the end of the cutter shaft *D*. Endwise movement of both pinion and shaft is prevented by the retaining plate *E*.

The eccentric sleeve *F* provides a bearing for the cutter

shaft and is supported along its length by the two jaws *G*. These jaws are secured to the plate *H*, upon which is mounted the worm-wheel *J*, which is keyed to the eccentric sleeve. Meshing with this worm-wheel is the worm *K*, which serves to rotate the eccentric sleeve for feeding the cutter (attached to lower end of shaft *D*) into the work. The thrust of the cutter is taken by the roller bearing *M* through the check-nuts *N* on sleeve *F*.

To recess the port *A*, Fig. 10, the cutter is lowered to the bottom of the port, and the plate *H*, located by pin *O*, is fastened by T-bolts to the top of the forging. The jaws *G* are made slightly less in width than the port, so that they serve to centralize, as well as to support the eccentric sleeve. As the machine spindle rotates, the internal gear revolves about the pinion, rotating the latter with the shaft and cutter.

To start the cut, the eccentric is rotated by hand through the worm and worm-gear, a handwheel (not shown) being provided for turning the worm. As the eccentric sleeve rotates, the cutter is swung into the side of the port. The greatest depth of cut is reached when the worm-wheel has rotated 90 degrees, as indicated in Fig. 10 by the dot-and-dash circles representing the cutter. When the cut is completed, the worm-wheel is reversed to withdraw the cutter from the recess. The tool is then removed and set up in the next port, where the other recess is cut.

Friction-Grip Wire-Feeding Device.— The device shown in Fig. 11 can be used for feeding wire on any wire-forming or other machine requiring an accurate feed. The outer shell *S* is mounted on a slide or other reciprocating part of the machine which has a movement equal to the desired feed. Cage *C* is carried inside shell *S*, bearing with an easy sliding fit at both ends. This cage has two holes diametrically opposite each other into which balls *B* are placed. The holes are slightly larger than the balls, so that the balls move freely within them. The holes are not full size clear through to the central hole in part *C*, but end in a conical seat, thus allowing the balls to project through to the wire *W*, but preventing them from falling through when there is no wire there. Spring *P* pushes the cage so that the balls are carried into the taper portion of *S* and against the wire.

In action, the device is moved in the direction of the arrow shown on the diagram. As the balls are in contact with the wire and the shell, any resistance to the movement of the wire causes them to roll into the taper and grip the wire more tightly. This movement is very slight and does

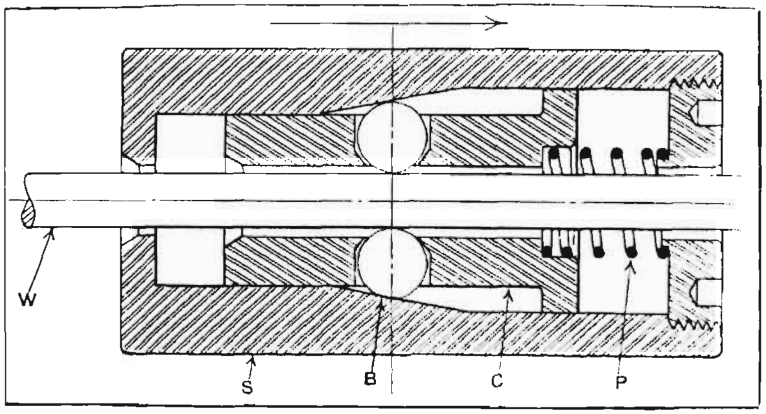


Fig. 11. Simple Design of Wire-feeding Device of Friction-grip Type

not affect the accuracy of the device for ordinary purposes. As the shell starts on its return stroke, the balls roll out of the taper, thereby releasing their grip on the wire. Theoretically, three balls should be used in this device, but it has been found that two balls are entirely practical. It will be noticed that several sizes of wire can be fed and that variations in the wire do not affect the accuracy of the feed.

Adjustable-Speed Wire-Feeding Unit for Wire-Cutting Machine.— Various lengths of wire for carrying electric current are used in the manufacture of a certain product. These wires are cut to length and the ends are stripped of insulation on a well-known make of wire-cutting machine.

The wire comes to the factory wound on heavy spools, a full spool weighing approximately 175 pounds. The inertia of such heavy spools caused an uneven feeding movement that resulted in a variation in the lengths of the wires. To overcome this trouble, an operator was employed to turn the spool, so that a small amount of wire would be kept slack at the feeding end of the machine. It became evident, however, that such an arrangement would be too expensive, as the job appeared to be one that would last for several years. The attachment illustrated in Figs. 12 and 13 was built to eliminate the necessity for hand-feeding.

The spool of wire is indicated at the left, Fig. 12. There are three large sheaves *C*, *D*, and *E*, and one idler *F*. Two of the main sheaves *E* and *D* are drivers, being geared together and driven by a sprocket and chain which is connected to a cross-shaft *H* at the end of the machine *A*, Fig. 13. In the driven sprocket is a free-wheeling clutch. The shaft to which this clutch is keyed is squared for a crank-handle. This permits the train of sheaves to be turned forward independently of the driving mechanism.

The original driving shaft of the machine was replaced by a longer one. To this longer shaft is secured a large friction disk *G*. Adjoining this disk and attached by suitable mountings to the end of the machine is the cross-shaft *H*. To the end of this shaft is secured the driving sprocket-wheel. Also keyed to this cross-shaft is a small friction wheel *I* which contacts with and is driven by the large friction disk. Set into a groove in the cross-shaft is a small screw *J* (see enlarged view). This screw is threaded through a half-nut which is secured to the side of the driven wheel. The screw is retained in the shaft by means of a plate which is fastened to the end of the shaft. By means of this screw, the driven wheel is adjusted across the face of the driving disk to obtain any desired ratio of speed. To facilitate setting the driven wheel, the cross-shaft is scored and numbered every half inch, the numbers corresponding

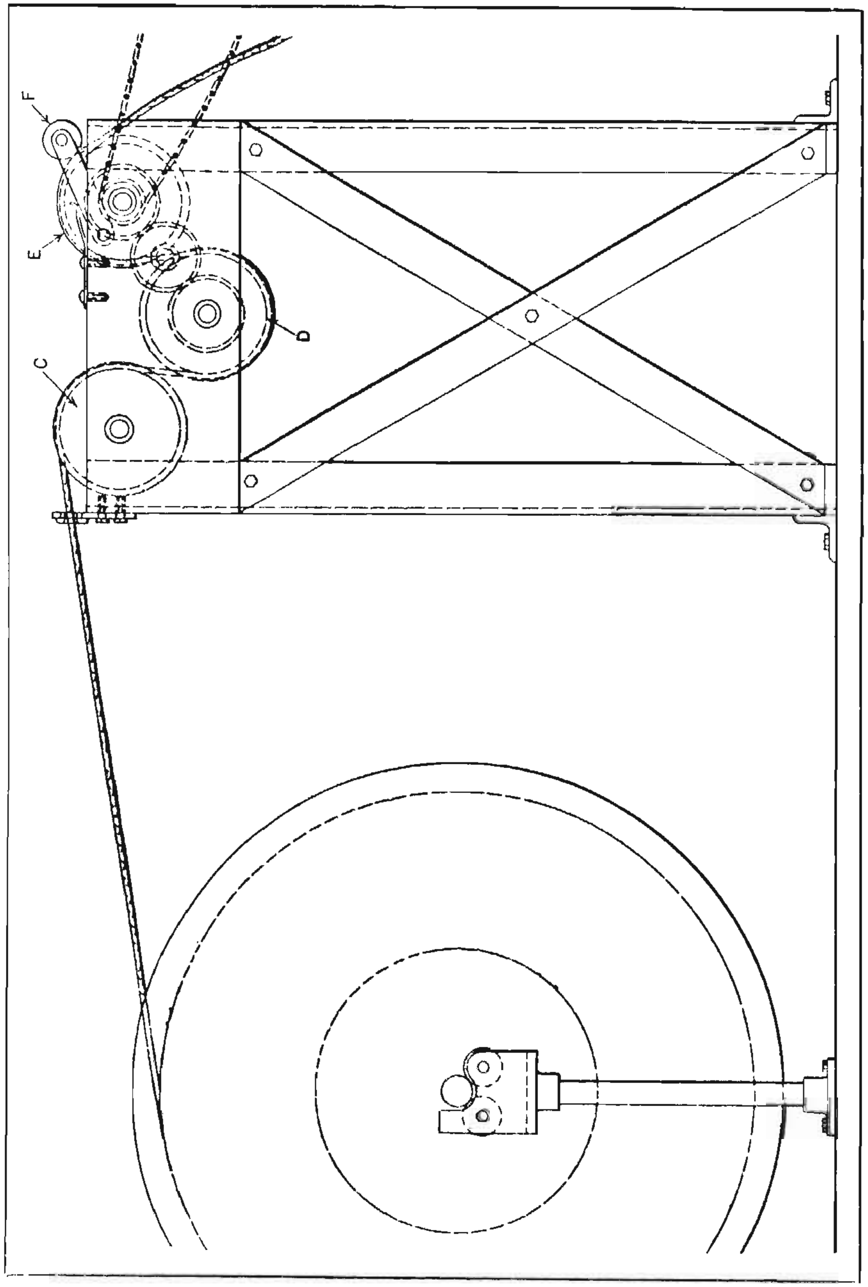


Fig. 12. Adjustable-speed Friction-driven Feeding Unit for Wire-cutting Machine—See Continuation, Fig. 13

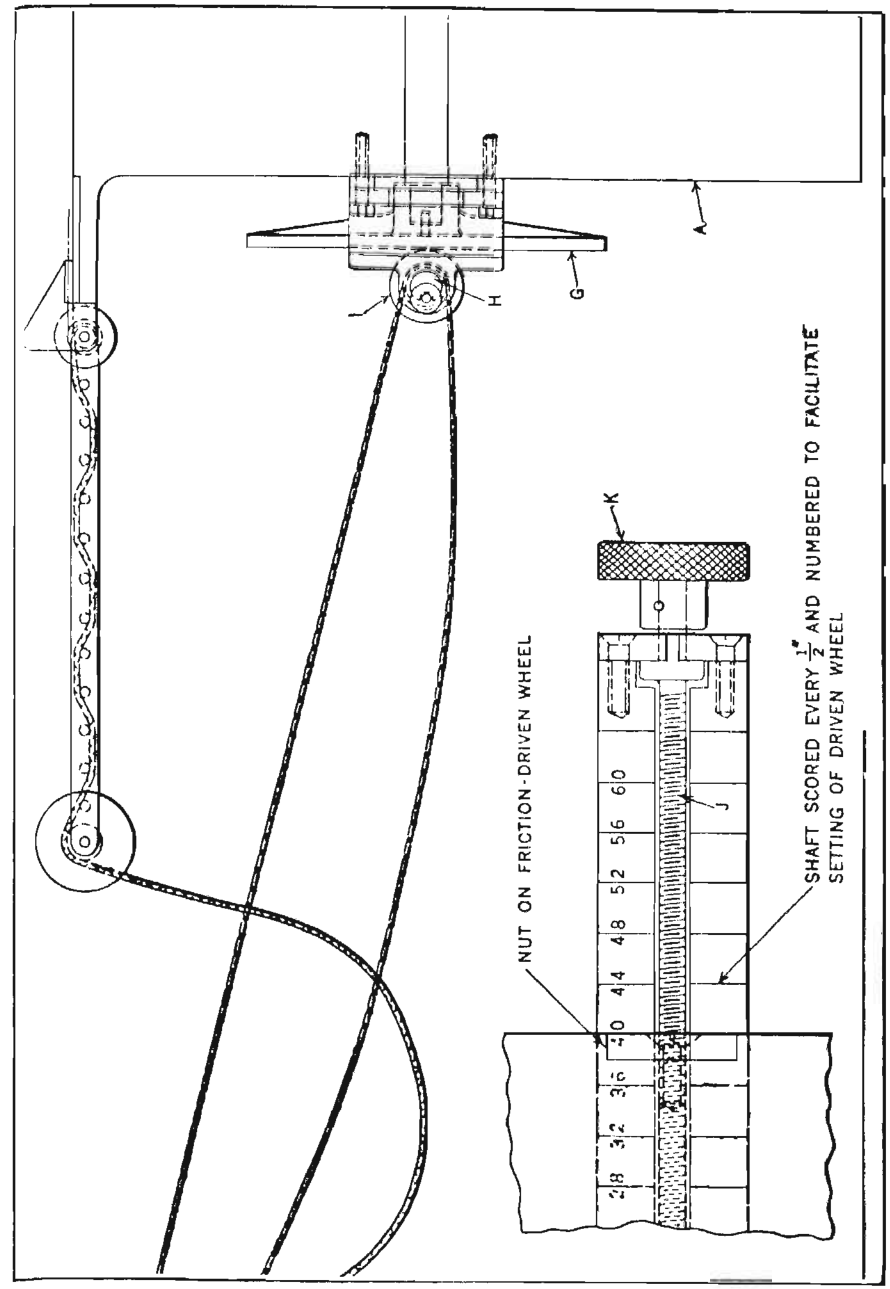


Fig. 13. Wire-feed Driving Mechanism which Transmits Motion through a Sprocket and Chain Transmission to the Feeding Sheaves, Fig. 12

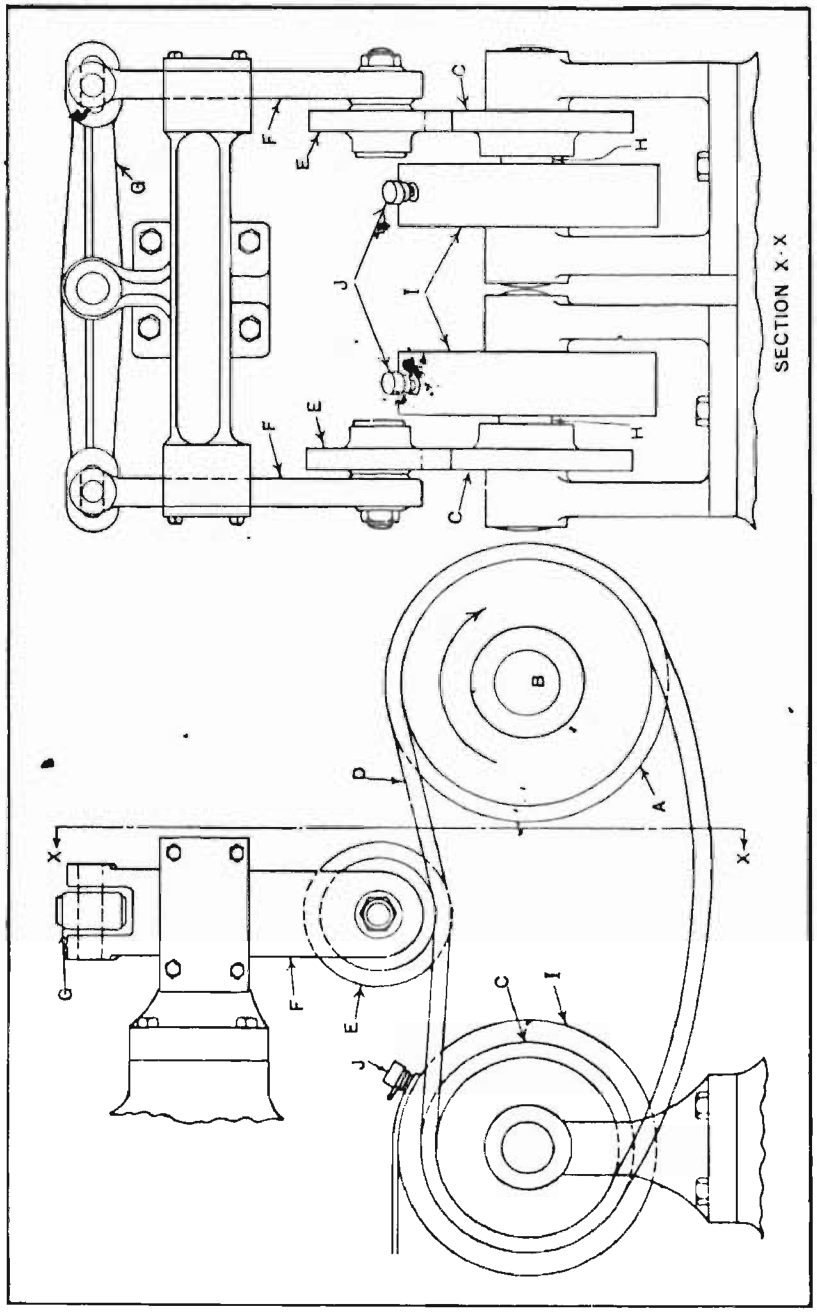


Fig. 14. Device in which the Slack in Two Sprocket Chains Equalizes the Tension in Two Wires Being Stretched

to the length of wire that will be fed by the rolls for that particular setting.

In operation, after the wire has been laced through the feed-rolls and into the machine, the operator places the crank-handle on the squared end of the shaft previously referred to, and turns the feed-rolls forward, feeding a small amount of slack wire ahead of the wire-cutting machine. The wire-cutting machine is then started, which also serves to start the feed-rolls. If the length of wire is one that has been cut previously, the setting, no doubt, will be fairly accurate and the same amount of slack wire will be maintained. However, should the length of wire to be cut be an odd size, an approximate setting is made. An occasional glance from the operator, while pursuing other duties, determines whether the feed-rolls are losing or gaining on the wire-cutting machine. In either case, a turn of knob K at the end of the screw in the cross-shaft readjusts the feed. This can be done while the machine is running.

Automatic Wire-Tension Equalizer.—On a special machine for producing a wire product, numerous strands of wire are woven or interlaced around two lengthwise strands. After the required number of interlacings have been made, the two lengthwise strands are pulled tightly and the whole locked together. It is essential, however, that both lengthwise strands have the same degree of tension during this locking action. Hand methods of tensioning had been used until the attachment shown in Fig. 14 was developed. This attachment automatically maintains an equal tension on the two wires.

The two sprockets A fastened on the driving shaft B drive the two sprockets C by means of the chains D, which have considerable slack. This slack, because of the direction of the drive, will be normally at the bottom. A uniform tension is maintained in the wires by means of the two idler sprockets E carried on slides F, which, in turn, are connected by the equalizing lever G. Sprockets C and

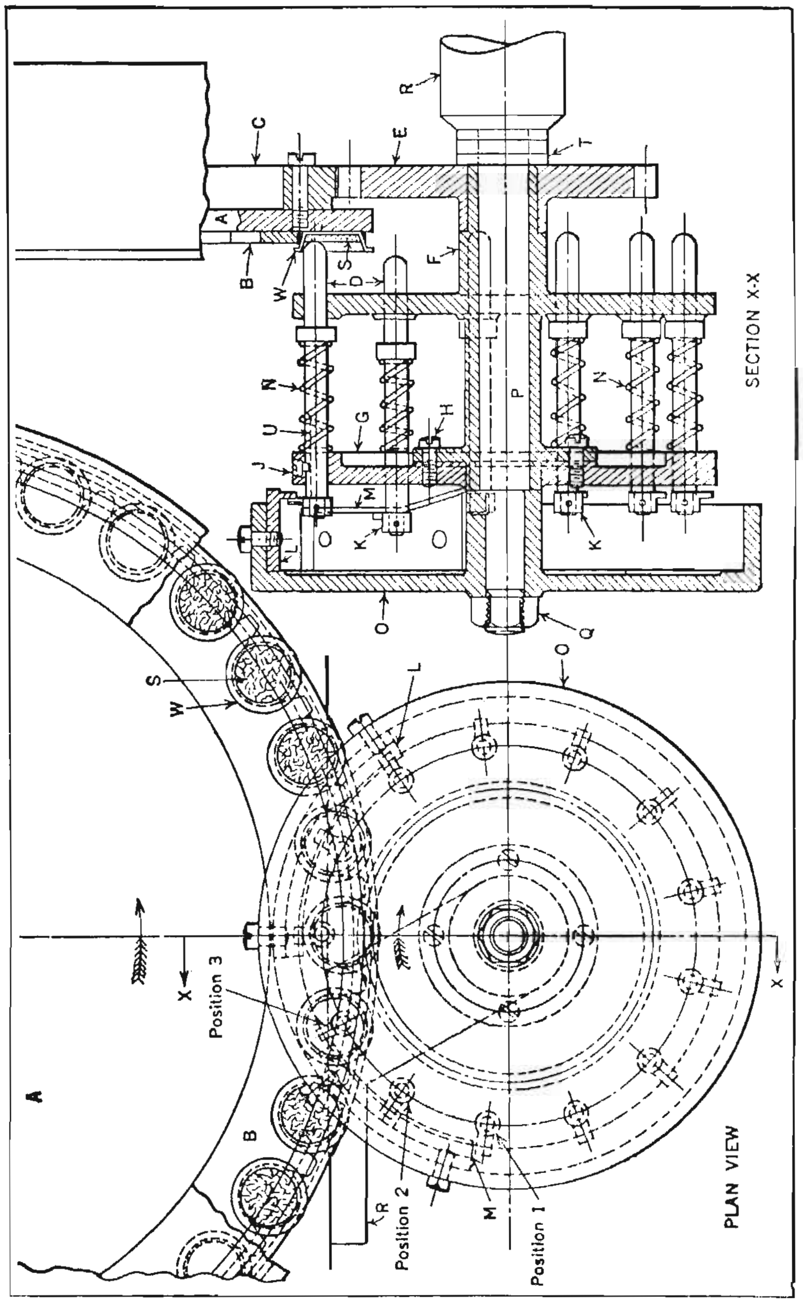


Fig. 15. Mechanism that Automatically Removes Caps W from Transport Wheel A when the Disk-feeder Fails to Supply the Caps with Cork Disks S

drums *I* are fastened on the short jack-shafts *H*. The lengthwise strands of wire are fastened to studs *J* on drums *I*.

When the wires are ready for tensioning, shaft *B* is given a slow rotary motion in the direction of the arrow. This motion is transmitted to the sprockets *A* and *C* and the drums *I* through the chains *D*. The idler sprockets *E* operate on the tight or load-carrying side of the chains *D*. Therefore, any increase in the tension of the wires will produce a corresponding increase in the tension of the chains *D*, and also in the pressure against the idler sprockets *E*.

As long as the tension of the two chains *D* remains equal, the lever *G* will be inactive. However, as soon as this tension becomes unequal, the sprocket *E* on the chain having the greater tension will be forced upward, causing the lever *G* to force the other sprocket *E* downward until the tension again becomes the same in both chains. Except at the very beginning of the tensioning operation, this attachment scarcely seems to operate. The slightest difference in the tension of the two wires is transmitted to the chains and idler sprockets, causing an almost imperceptible equalizing movement of the lever *G*.

Mechanism for Removing Incompletely Assembled Caps from Conveyor Wheel.— Cork disks *S*, Figs. 15 and 16, are assembled in caps *W* by automatic machinery having a conveyor or transport wheel *A*. Occasionally, the mechanism for feeding the cork disks into position for assembling becomes jammed, with the result that caps *W* pass to the ring *B* of the transport wheel *A* unprepared for the operations that are to follow.

The automatic mechanism shown in Fig. 15 for removing the caps that fail to receive cork disks was developed after some experimental work and attached to the transport wheel. This self-contained mechanism prevents the incompletely assembled caps from continuing along the line and thus causing unnecessary expense for useless work. The

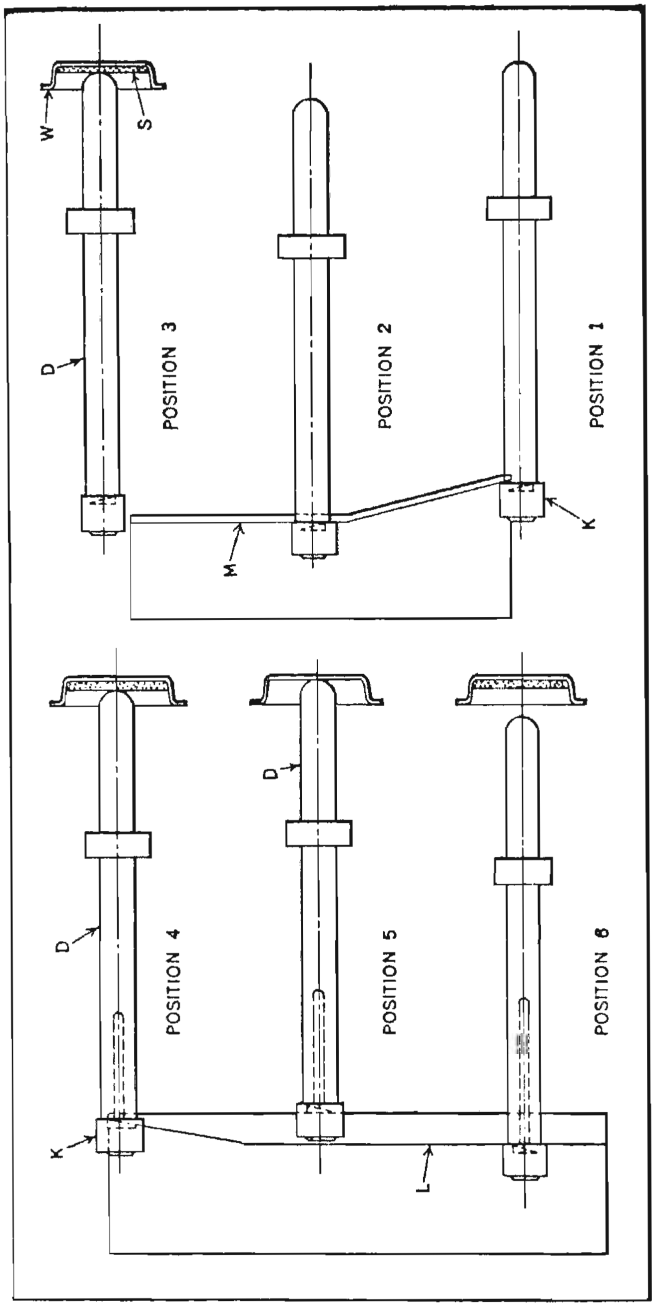


Fig. 16. Diagrams Illustrating Principle Employed in Mechanism for Removing Caps W from Machine when the Feeding Device Fails to Supply Them with Cork Disks S

assorting device operates satisfactorily and performs the desired sorting operation at a point that could not be reached readily by the operator's hand. The principle is simple and should be easily adaptable to other work of a similar nature.

Ring *B* is fastened to disk *A*, the caps *W* being carried in machined recesses provided in ring *B*. The ring-gear *C* is attached to the bottom of disk *A*, and through its connection with the gear *E*, serves to actuate the automatic assorter. Gear *E* is fastened to the flange member *F*, which is part of a spool consisting of two flange members *F* and *G*, held together by screws *H*. The spool rests on the thrust bearing *T* and revolves on the spindle *P*, which is part of the stationary support *R*.

Holes are drilled and reamed in the flanges *F* and *G* to permit a sliding fit for the plungers *D*. In order to prevent the plungers from twisting in their bearings, a slot *U* is provided in each one in which a set-screw *J* acts as a key. A light spring *N* keeps the plungers down. Each plunger is equipped with a lifter *K*, pinned in place at its upper end. The combined cover and cam-holder O is held in position on spindle *P* by the nut *Q*. Two cams *L* and *M* are fastened to the cam-holder *O*.

Operation of the Automatic Cap Assorter.— In operation, the movements of plungers *D* are controlled by cams *L* and *M*. The position of the first cam *M* is shown in the view to the left. In Position 1, Fig. 16, the lifter *K* is shown making its initial contact with the cam surface. Position 2 shows the plunger resting so as to clear the cap entirely. In Position 3 the plunger is shown released and dropped into the cap under the action of spring *N*, Fig. 15. Positions 1, 2, and 3, Fig. 16, correspond to the points marked 1, 2, and 3 in Fig. 15.

The function performed by cam *L* is illustrated in the views to the left, Fig. 16. In Position 4, plunger *D* is shown resting on top of a cork disk inserted in the cap. The lifter

K, in this case, is raised high enough to permit it to make contact with the cam surface. In Position 5, plunger D is shown resting on the bottom of a cap, with no cork disk in place. In this case, the lifter K is positioned too low to make contact with the cam surface, so that it passes under cam L. Since the plunger is not raised out of the cap in the latter case, it causes the cap to be removed from its position in the transport wheel by the plunger D. In Position 4, however, the cap moves on, because the plunger is raised sufficiently to allow the cap to retain its position in the transport wheel.

This device can be operated successfully at relatively high rates of production. In increasing the operating speed, it becomes necessary to place cams L and M farther apart, in order to give plunger D time to drop into the cap before the lifter K makes contact with cam L. The space between the plungers on the automatic assorter should be equal to the distance between the caps which are located on the transport wheel.

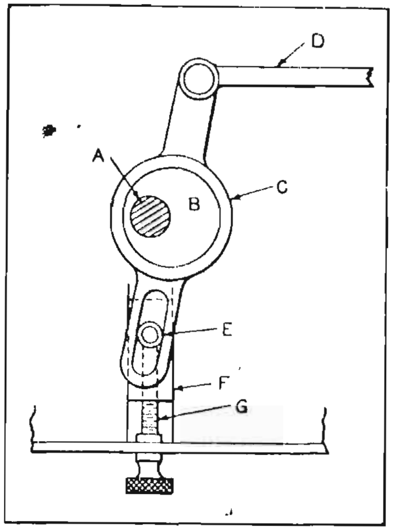


Fig. 17. Adjustable-stroke Mechanism

Adjusting Stroke without Stopping Machine.—The feeder slide on a wire machine is operated by an eccentric mechanism (Fig. 17), which has an arrangement for varying the length of stroke while in operation. The shaft A rotates the eccentric B, which carries strap C. The upper end of the strap carries an arm, the motion of which is transmitted to the slide through the rod D. The lower end of strap C has a slotted arm, which is positioned by the roller E. This roller is carried on block F which slides in a groove in the bed of the machine and is adjustable by means of the screw G.

The illustration shows the eccentric at its extreme right-hand position. The arms on strap C gradually assume an angular position after the eccentric has passed the top center. The effect is that of a lever with E acting as the fulcrum, so that rod D has a movement greater than would be obtained by direct connection with the eccentric strap C. If the roller E is moved toward or away from the shaft A, the effective length of the slotted arm is increased or decreased; and as the length of the upper arm remains constant, its movement is accordingly increased or decreased.

Adjustable Stroke-Feeding Mechanism for Sewing Machines.—The feeding mechanisms of sewing machines used for commercial production work must be designed to handle a great variety of fabrics. This requires a wide range of adjustment in the length of stitch. The parts of the mechanism must be so proportioned that they will be durable and require a minimum of power for their operation. While the method of adjusting the length of stitch should be simple and positive, it need not be of a character suitable for adjustment by the operators.

Referring to the feed-dog shown at B, Fig. 18, it is necessary that the path of travel of this part while above the throat plate N in the working part of its cycle of motion be approximately a straight line. It is also desirable that the working path of the feed-dog be capable of being tilted in either direction from a line parallel with the top of the throat plate. The mechanism shown accomplishes these several objects in the manner to be described.

The feed-dog B is attached at or near the front end of the feed-bar A. The rear end of this feed-bar is supported by rocker arm C by means of shaft D, about which it is free to pivot. In a similar manner, the front end of the feed-bar is carried by rocker arm E, which is free to pivot about the pin connection F. The rear rocker arm C is pivoted at its

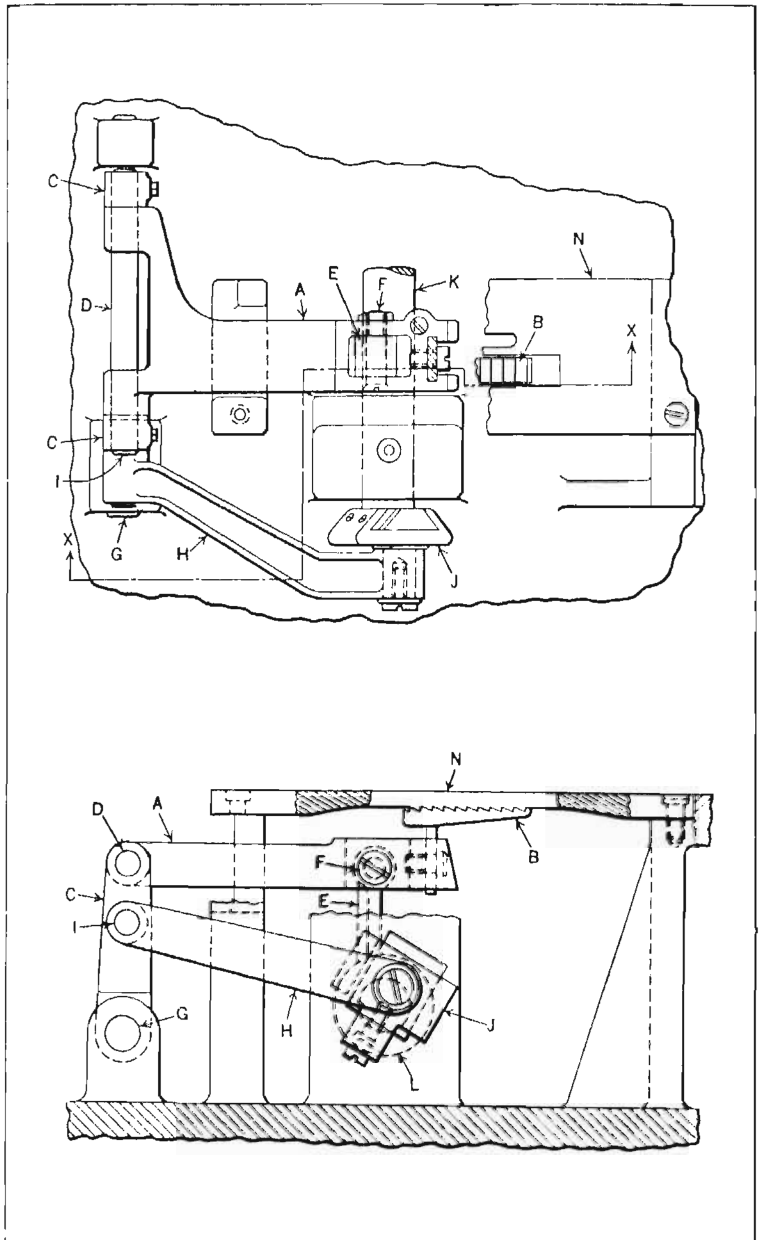


Fig. 18. Adjustable-stroke Feeding Mechanism for Sewing Machine

lower end to the frame of the machine at G and is driven by link H. Link H is pivoted to arm C at bearing I. The other end of link H is pivoted to the adjustable feed-crank J, carried on the end of main shaft K. The lower end of rocker arm E is in the form of an eccentric strap L, which engages an eccentric, driven by the main shaft K. This eccentric is termed the "feed-lift eccentric." Both the feed and lift motions are positive, and their combined action on the feed-dog results in a path of motion relative to the

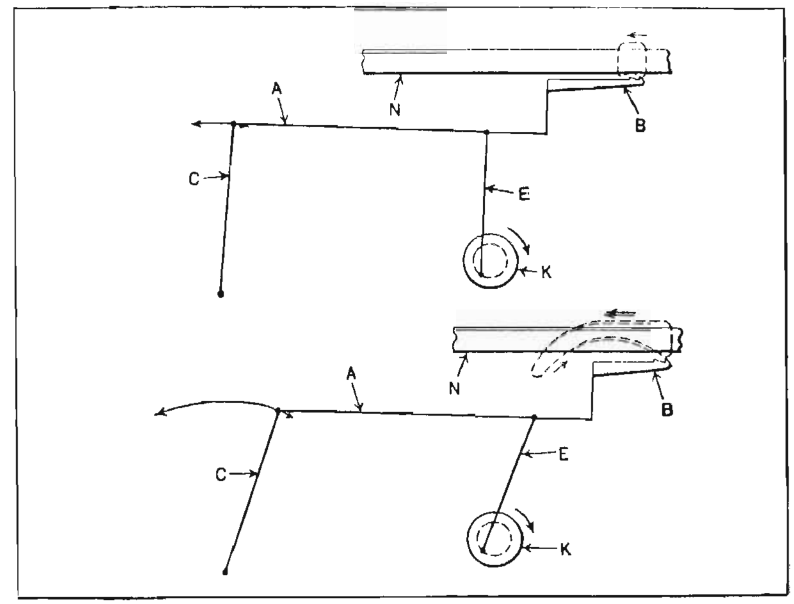


Fig. 19. Diagrams Showing Adjustments of Sewing Machine Feeding Mechanism for Both Short and Long Feeding Strokes

throat plate N such as is illustrated by the dotted lines of the two diagrams Fig. 19.

This arrangement causes the rear end of feed-bar A to rise and fall twice with each revolution of the shaft K. The amount of this rise and fall depends upon the length of the rocker arm and its angular displacement each side of the vertical position. In a like manner, the front end of the feed-bar will rise and fall, due to the relation between it and the front rocker arm. The front end of the feed-bar is also caused to rise and fall by the rotation of the lift-eccentric on the main shaft. It is evident, therefore, that the rise and fall of the front end of the feed-bar will be the result of these two actions. The rise and fall of the feed-dog will be similar to that of the front end of the feed-bar, but not exactly the same, depending upon its size and location relative to the front end of the feed-bar.

The upper diagram Fig. 19 shows the adjustment for a relatively short stitch, and the lower diagram, the adjustment for a relatively long stitch. These views indicate the relationship between the rocker arms, feed-bar, feed-dog, throat plate, and lift-eccentric. The dotted lines in these illustrations show roughly the path of the toe of the feed-dog. The path of the heel would be similar to that of the toe, but not exactly the same. By a suitable proportioning of the parts and adjustment of the angular relationship between the feed-crank and the lift-eccentric, the feed-dog may be caused to emerge through the throat plate parallel to the latter member and to travel a very nearly straight line parallel with the top of the throat plate.

By altering the angular relationship between the feed-crank and the lift-eccentric, the feed-dog may be caused to emerge from the throat plate toe first; that is, the feed-dog may be tilted backward at a slight angle. By changing this angular relationship in the opposite direction, the heel of the feed-dog may be caused to rise first. These various relations of feed-dog to throat plate are desirable because of the feeding requirements of different kinds of fabrics and the kind of seam required. In the design described, excessive wear and violent velocity changes have been avoided. This enables the mechanism to be operated at high speeds with relatively small wear and with a comparatively small consumption of power.

Mechanism for Operating Magazine Feed-Slide.—Electrical knife switches are automatically assembled on their

slate bases in a machine equipped with a magazine that feeds the bases to a dial by means of a pusher-slide. An interesting feature of this magazine is that, although the stroke of the feed-slide is only 4 3/4 inches, this slide serves to transfer the base over a distance of 7 1/2 inches. Referring to the illustration (Fig. 20), the magazine is shown at *A* fastened to the machine frame *B*. The feed-slide is indicated at *D*. It rests on the top of the machine and operates between two guides, one of which is shown at *F*. The slide is reciprocated by the oscillating lever *G* through the

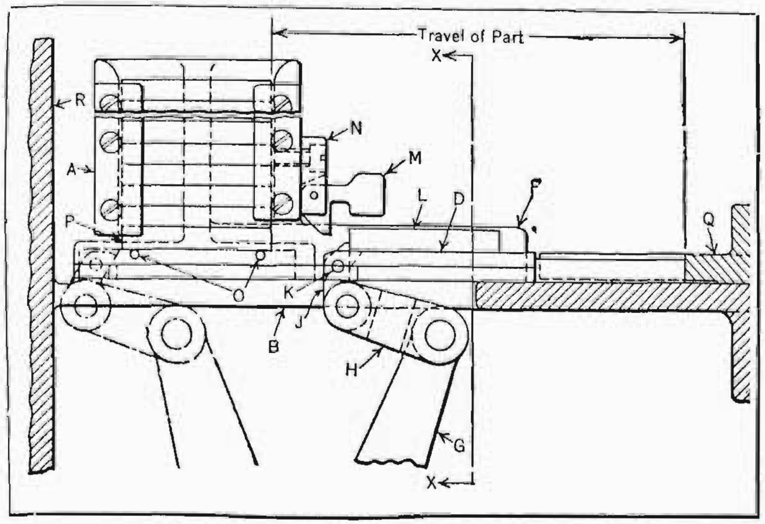


Fig. 20. Mechanism that Feeds Knife Switch Bases from Magazine A to Slots in the Dial Q

link *H* and the stop *J*, stop *J* being attached to the slide by the pivot pin *K*.

The lever *G* is shown in its farthest position to the right, where it has carried the base *L* out of the magazine. As the lever and slide move toward their left-hand position, base *L* is carried against the counterweighted pawl *M*, pivoted to the block *N* which is secured to the magazine. Continued movement of the lever toward the left causes the pawl to push the base off the slide, so that when the latter has reached its extreme left-hand position, the base drops on top of the machine frame *B* between the guides.

Since the bottom base in the stack rests on the pins *O*, the stop *J* must clear the bottom base as the slide moves to

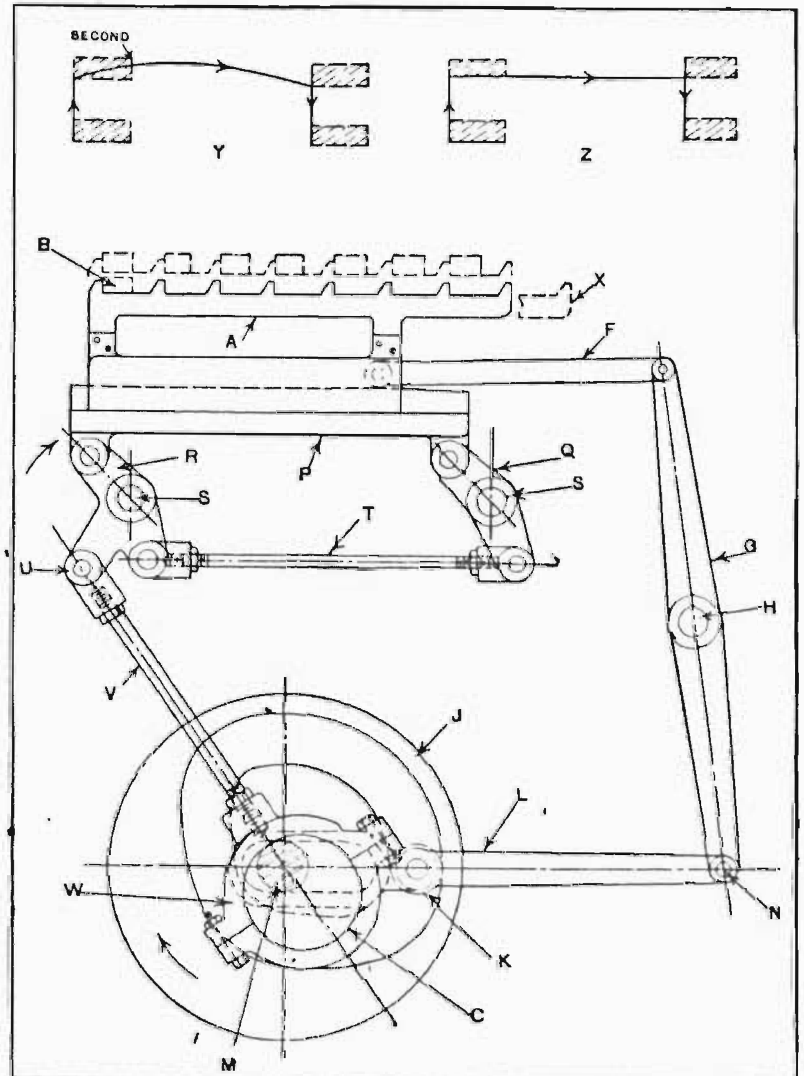


Fig. 21. Mechanism for Transferring Packages from Station to Station

its left-hand position. Obviously, as the lever swings toward the left, the stop will swing in a clockwise direction, so that its protruding point recedes below the top of the slide. The stop remains in this position until the lever and slide are at the left-hand end of their stroke, as indicated by the dot-and-dash outline. When the lever moves toward the right, the dog automatically swings back to its former position and pushes base *P* out of the magazine. In the meantime, base *L*, resting on the machine, is pushed by the front end of the slide into a slot in the dial *Q*. An added advantage of this mechanism is that it can be used effectively where space at the left of the magazine is limited, as it is in this case by the machine wall *R*. Ordinarily, a long slide would have been used which would have required considerable clearance at this point.

Transferring Parts from Station to Station.—In a machine for wrapping packages, the conveying mechanism shown in Fig. 21 is employed for transferring the packages to each successive station. In doing this, the transfer arms *A* must pick up the packages, carry them toward the right to the next stations, lower them into position, and then, after dropping enough to clear the bottom of the packages, return to their starting position. A package partly wrapped is deposited automatically on the carrier at *B* when the mechanism is moving up from the position shown.

The diagram *Y* indicates the path through which the packages theoretically are moved during one cycle, although they rest on bars in the upper position while the carrier drops below them when returning, the diagram representing the transfer arm travel. An eccentric in combination with a cam is used to obtain this movement, although two cams could be used that would cause the package to follow the path indicated at *Z*; or two eccentrics might be used if the motion imparted would be suitable.

The two carriers *A* support both ends of the packages, while the slide beneath supports the carriers and is connected by link F to the lever G pivoted on stud H. The slide is operated through this lever arrangement by the cam J which engages the roll K attached to the yoke L. Shaft M, which drives the entire mechanism, passes through the yoke, while a pin at N pivots the yoke to the lever. Supporting the slide that carries the transfer arms A is a bracket P to which the slide is dovetailed. This bracket is mounted on two levers shown at Q and R, which are free to pivot on the studs S. Connecting link T ties these levers together, and increases the strength of the assembly. Forming part of the lever R is an arm U to which is pivoted a connecting-rod V fastened to the eccentric strap W. Both the eccentric C and the cam J are pinned to the drive shaft M.

In operation, as the shaft M revolves, the movement of the eccentric causes levers R and Q to oscillate in the direction of the arrow, thus raising bracket P. In the meantime, a dwell on the cam prevents the slide from moving to the right or left. When the carrier reaches the position shown by the dotted outline, the cam operates lever G, so that the slide is moved to the right; the dwell on the cam then holds the lever stationary until the eccentric swings the lever R back so that the carrier will be in the position indicated at X. The cam then operates lever G to bring the slide back to its starting position. As the eccentric travels continuously through an arc, at no time will the slide be held stationary, the path of the carriers being curved; however, for practical purposes, this departure from a straight line movement may be disregarded. If it is desired to control this mechanism so that there is no up or down travel while the slide is traversing, a cam may be substituted for the eccentric. The dwells and rises on the cams may then be varied so that the path of travel will be as shown at Z.

Elevating Pile of Sheets to Keep Top Sheet in Alignment with Feed Rolls.—The top sheet in a pile is automatically kept in approximate alignment with a pair of feeding rolls

by the mechanism shown in Fig. 22. It is used for feeding sheets to a paper-tube rolling machine and can be readily adapted to the feeding of metal sheets as well.

The rectangular sheets A are stacked on the vertical slide B mounted on the machine C. This slide is given a vertical

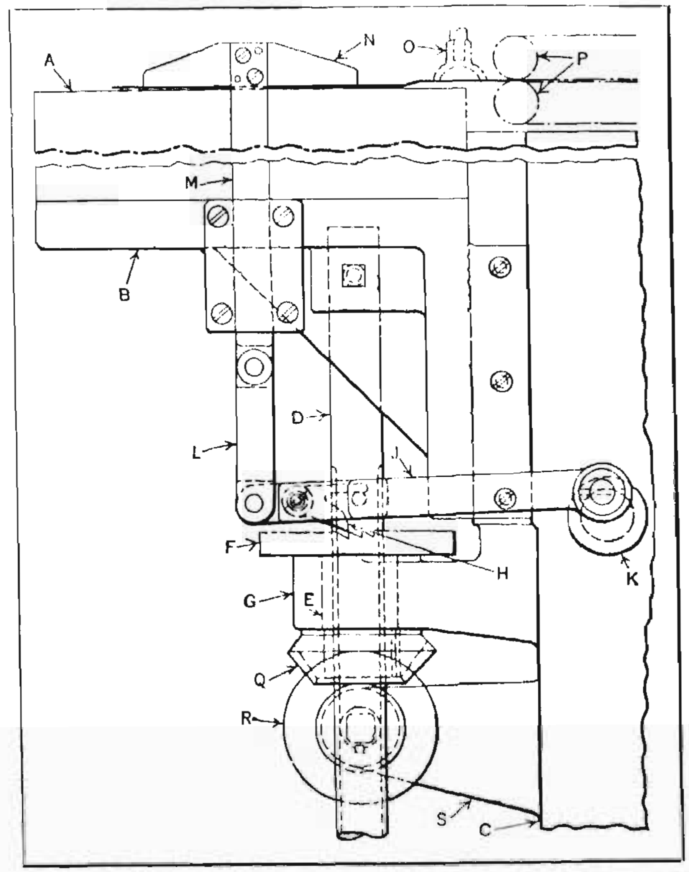
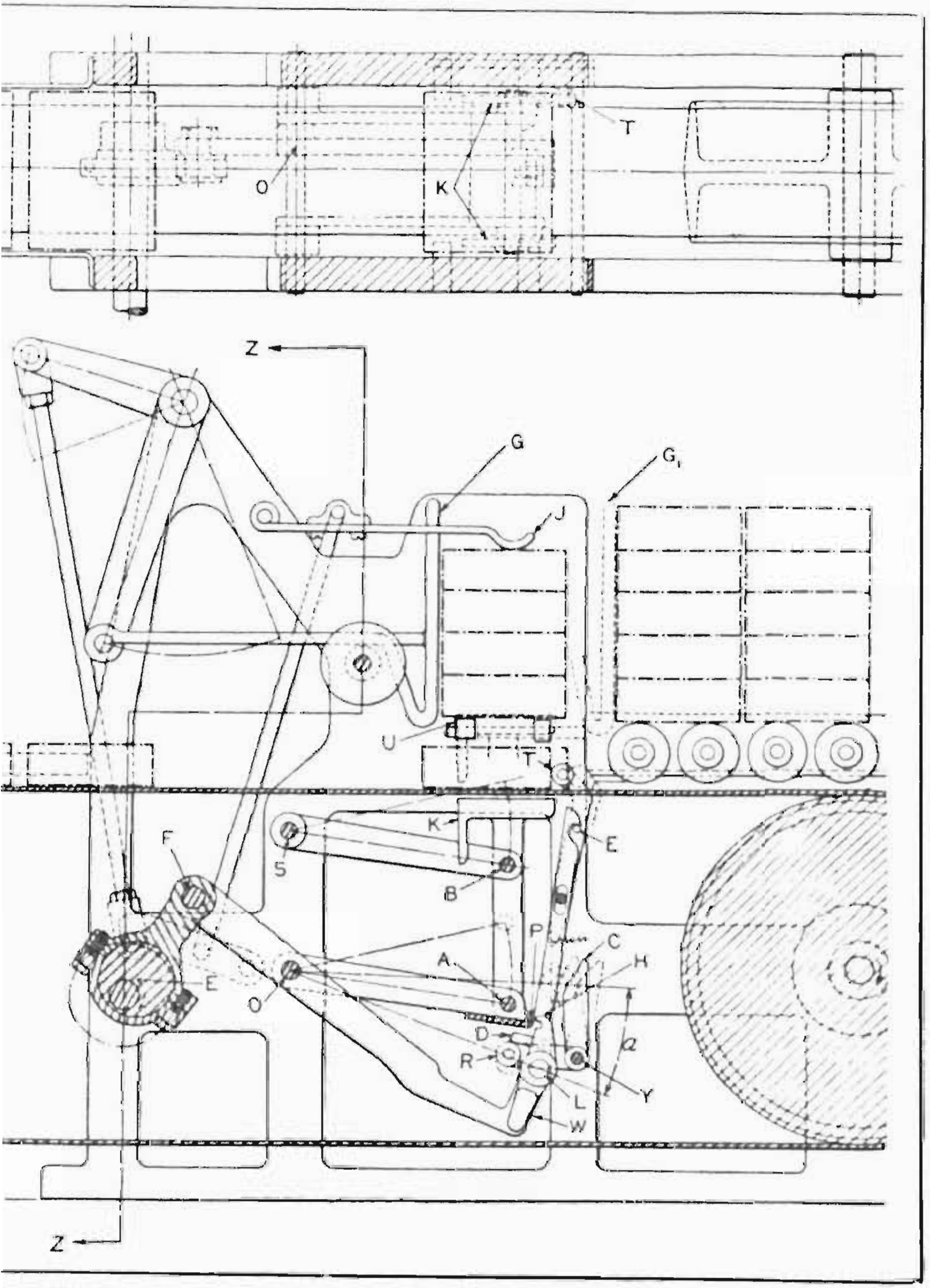


Fig. 22. Mechanism that Keeps Top Sheet of Stack in Magazine in Line with Feed-rolls

feeding movement by means of the screw D secured to the slide. The screw engages a nut E which is an integral part of the ratchet wheel F. Bearing G, cast on the machine frame, serves as a support for the nut and ratchet wheel.



g. 23. Stacking Mechanism at End of Conveyor Belt which Raises Pieces from Belt, Arranges them in Stacks, and Transfers the Stacks to a Roller Carrier

The latter is rotated to feed the screw and slide *B* upward by means of the pawl *H* pivoted to the oscillating bar *J*. This bar receives its movement through the constantly rotating crank *K*, and at its left-hand end is connected to link *L*. Link *L*, in turn, is connected to the bar *M* which slides in a guide on the slide *B*. At the top of bar *M* is a cross-piece *N*, which rests on the top of the sheets.

As indicated, slide *B* is loaded with sheets and the suction cups *O* have raised the end of the top sheet preparatory to carrying it forward and between the moving belts on the rolls *P*. These belts then transfer the sheet to the rolling mechanism. This top sheet is under the cross-piece *N*. The pawl *H* remains out of engagement with the ratchet wheel and there is no upward feeding movement of slide *B* until *N*, together with bar *M*, link *L*, and arm *J*, drops down far enough to cause pawl *H* to engage the teeth in the ratchet wheel, rotating the latter and feeding screw *D*.

Now, as cross-piece *N* travels upward with the top sheet, arm *J* will once more lift pawl *H* out of engagement with the ratchet wheel and thus stop the feeding action of the screw and slide *B* when the top of the pile of sheets has been elevated to the required level.

To permit reloading of the magazine, bevel gears *Q* and *R* are provided. Gear *Q* is keyed to the nut, while gear *R* is keyed to a shaft which turns freely in bracket *S* cast on the machine. The shaft for gear *R* is square at its outer end to accommodate a hand-crank used for moving the slide manually to its loading position. At this time, pawl *H* is swung up out of engagement with the ratchet wheel.

Mechanism for Stacking Articles at the Delivery End of a Conveying Belt.— The basic mechanical motion used in the mechanism illustrated in Fig. 23 has various applications and should be of interest to machine designers. In this case, it is applied to the problem of stacking articles which are being carried along a conveyor belt. Its advantage for this work is that it will handle articles in varying quantities or singly, as the case may be. In this instance, the articles, shown by dot-and-dash lines, are stacked five high and are discharged on a roller conveyor as shown.

The operation of the mechanism is based on the "firing" of a trigger T which is moved to the position shown by the dotted lines by a single article, which will then allow the conveyor belt to slip or pass beneath until the elevator K rises to the upper position, indicated by the dotted lines. This places the article at the bottom of the preceding articles which make up the stack. The article is prevented from dropping back by four latches U which are hinged in the side walls. Each article, of course, supports the one above it until the pile is complete, when the entire stack is moved to the right.

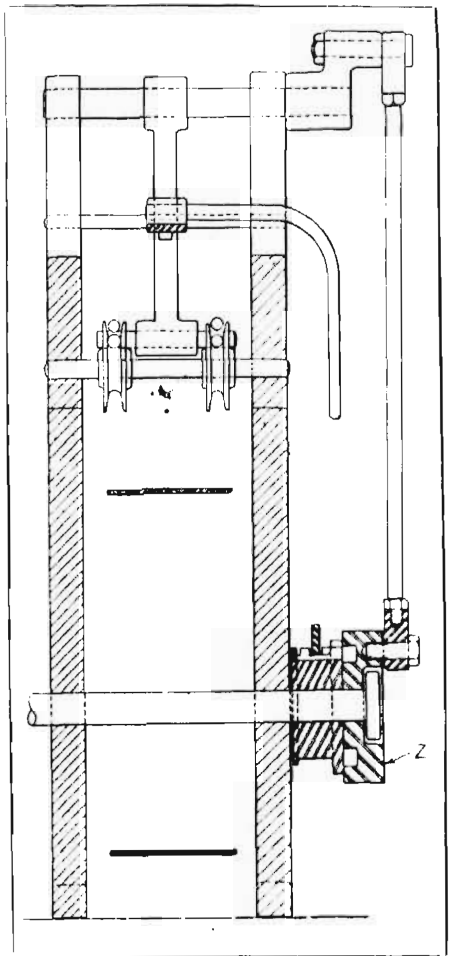


Fig. 24. Sectional View of Stacking Mechanism Shown in Fig. 23

The action of the lower part of the mechanism is as follows: An oscillating movement is imparted to the rocker arm OL by the driver shaft which rotates continuously. This shaft carries an eccentric E and, in turn, is connected

to arm OF. This arrangement causes arm OL to oscillate through the angle a. The secondary arm HLW, which is carried on arm OL, will pick up plate P under certain conditions, to be described, and swing arms OA and SB to their upper positions.

It is, of course, understood that these arms carry the elevators K and that the "pick up" action is accomplished as a result of the "firing" of the trigger T which allows the catch arm to drop off the small pin E and the lower end R to assume the position shown by the dotted lines. The cam arm YC and arm YR are tied together and turn or swing as one piece on the pin Y fixed in the frame. Part HLW also carries a small pin D which strikes roller R, and when in the lower position, locks into plate P on the upward part of the oscillating movement of arm OL. At the same time, cam C is moved to the position indicated by the dotted lines to the left. This action relatches the trigger T on pin E if no article is in position to be raised to the stack.

The discharge action may be arranged to take place on the return or downward part of the stroke. When the fifth article has raised arm J, the latter arm, which is connected to a bolt clutch finger of standard design, causes the part indicated at Z (Fig. 24) to rotate one complete revolution. Through suitable connections, which are clearly indicated, this action moves pusher G to the position indicated by the dotted lines at G_1 and returns it to its normal position. This movement transfers the stack of five pieces to the roller carrier and completes the cycle, after which the operations are repeated automatically.

Variable Rotary Movement for Operating Shell Hopper.— Brass shells are fed to a thread-rolling machine by means of a rotary hopper attached to the machine. After extensive experiments, it was found that a variable rotary movement of the hopper drum increased its efficiency; that is, more shells per minute could be fed by the drum when the pulsating movement was used. The mechanism shown

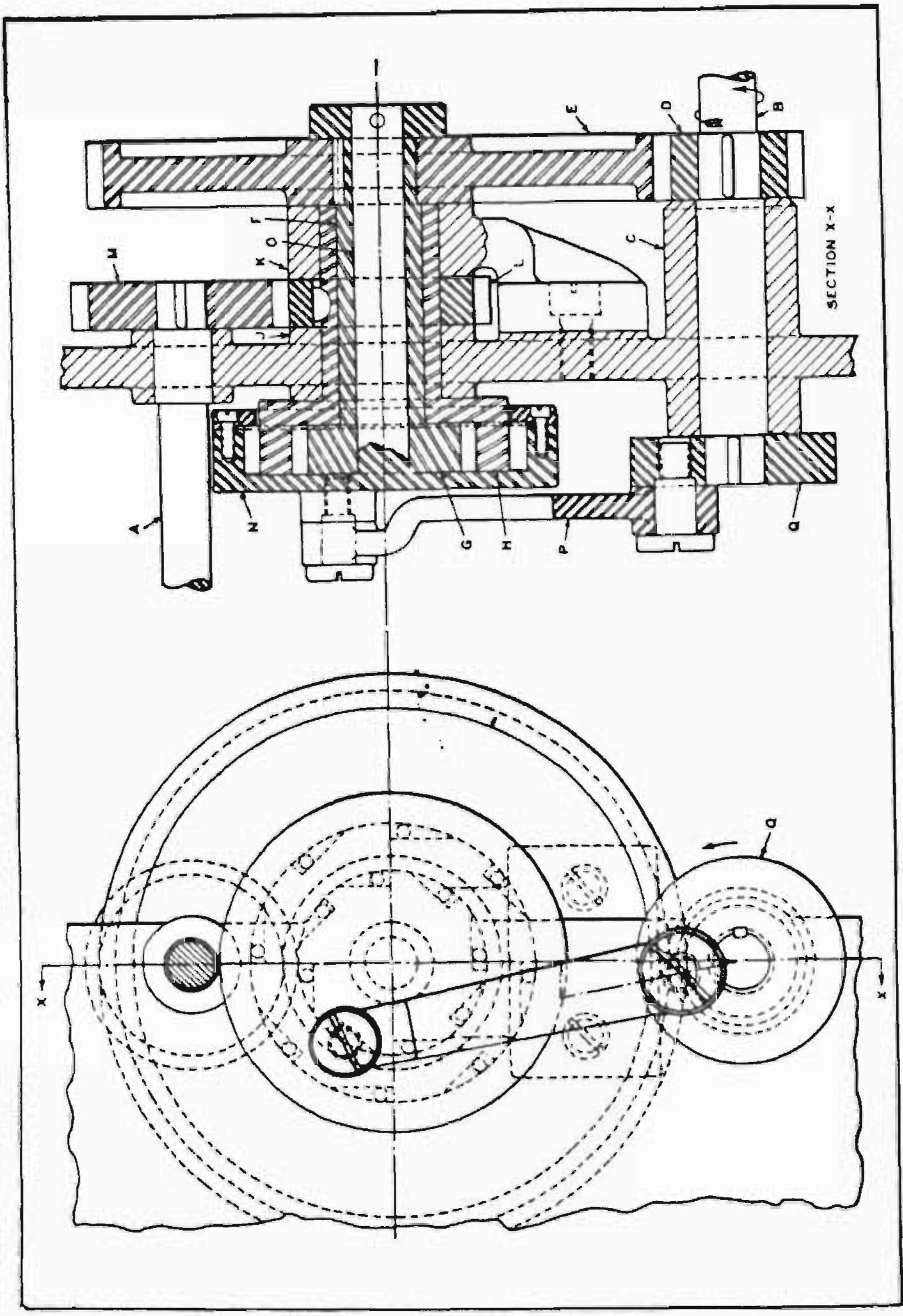


Fig. 25. Mechanism by which the Shaft B, Rotating at Constant Speed in One Direction, Imparts a Variable Rotary Movement to Shaft A

in Fig. 25 was designed to give the required pulsating movement.

With this arrangement, the drum is rotated a partial revolution at a slow velocity through a train of gears and a double roller clutch. The remaining part of the revolution is imparted by a crank which causes the roller clutch to over-run so that the drum rotates at a relatively rapid velocity.

The drive-shaft *B* is supported in the bearing *C*, which is an integral part of the machine. On this shaft is keyed the pinion *D*, which meshes with the gear *E*, keyed to sleeve *F*. The left-hand end *G* of this sleeve forms the core of a roller clutch; the outer ring for this core is indicated at *H*. The sleeve on ring *H* is supported in the bearings *J* and *K*, and its bore provides a bearing for the core sleeve *F*. Keyed to the ring sleeve is the gear *L*, which meshes with gear *M*, keyed to the drum shaft *A*.

It will be noted that ring *H* forms the core for the second or outer roller clutch, the ring for this clutch being indicated at *N*. The long shaft *O*, integral with this ring, is a free fit in the bore of core sleeve *F*, and a collar at its right-hand end serves to lock in position all the members supported in bearings *J* and *K*. On an offset boss on the side of ring *N* is pivoted the connecting-rod *P*, the lower end of which is connected to the crank disk *Q*, keyed to shaft *B*.

As shaft *B* rotates one-half revolution in the direction of the arrow, core *G* turns in a clockwise direction (see end view), rotating ring *H* and gear *L* with it. As a result, gear *M* and drum shaft *A* turn at a constant velocity in a counter-clockwise direction. In the meantime, crank *Q*, through rod *P*, rotates ring *N* in a counter-clockwise direction; but, as the clutch rolls between members *H* and *N* are free at this time, this movement does not affect the movement of shaft *A*. However, as soon as shaft *B* completes one-half revolution, the crank reverses the rotation of ring *N*. Now as this ring rotates at a much higher velocity than core G, the rolls between members G and H will be released, so that ring H will over-run and rotate gears L and M and drum shaft A at a high velocity. This high velocity of shaft A continues until shaft B has completed its second half revolution, after which the movement of ring N is again reversed, thus permitting the rolls to wedge between members G and H. This will cause member H to rotate the drum shaft at the slow velocity. There is practically no over-run of the drum when its velocity changes from high to low, owing to the frictional contact of the drum with the shells in the hopper. These slow and fast movements of shaft A are repeated alternately, imparting the required pulsating movement to the hopper drum.

CHAPTER XV

FEEDING AND EJECTING MECHANISMS FOR POWER PRESSES

Power presses and dies are utilized for such a large variety of manufacturing operations that many different types of feeding, ejecting, and other mechanisms have been designed. In fact, a large volume could be filled with mechanisms of this class alone; hence this chapter is not intended as a treatise on such mechanisms but it does contain illustrated descriptions of a number of feeding and ejecting mechanisms which incorporate in their design certain mechanical principles likely to be of value to users of a book on the general subject of mechanism.

Inverting Shells After they Leave the Hopper.— Some hoppers used for feeding shells to power presses are designed so that the closed end of the shell will enter the feed-tube first. To permit this type of hopper to be used for work in which the shells are required to enter the press dial with the closed ends at the top, some means must be provided for inverting the shells after they leave the hopper and before they enter the dials.

This may be done by employing the device shown in Fig. 1. Here it will be seen that the shells leave the hopper tube and drop into recesses in the disk A. These recesses are equally spaced and the disk is indexed one space for every cycle of the press. The indexing occurs during the upward stroke of the ram. Motion is transmitted to the disk for this purpose by means of the link B and the lever C. At one end of lever C is mounted a pawl which engages the ratchet wheel D. The ratchet turns freely on the shaft E and transmits the required rotary motion to the disk A by means of friction washers (not shown).

In the position shown, a shell has just entered the top depression in the disk, with its closed end at the bottom, while at the lower part of the disk another shell has dropped into the press dial with its closed end at the top. One-half revolution of the disk *A* is required to invert each shell.

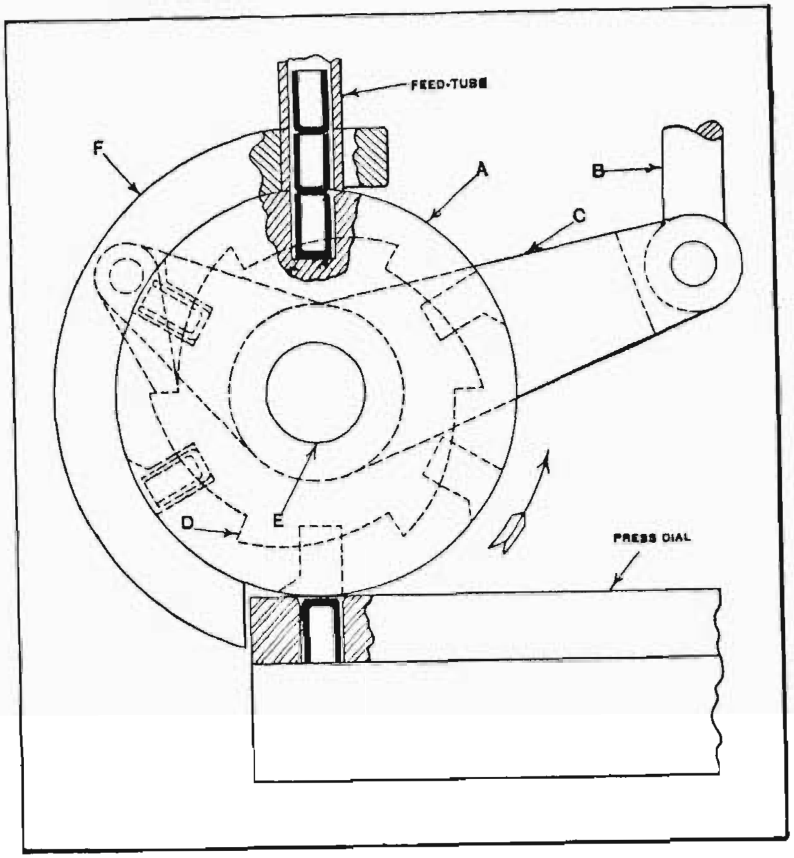


Fig. 1. Simple Device for Inverting Shells before they Enter the Dial Press

The stationary guard *F* provides for retaining the shells in the disk. It will also be noted that one corner of each of the impressions in the disks is beveled. This is done so that as this corner passes the hopper feed-tube it will not jam the shell in the end of the tube, but will force it upward into the tube. In case the stroke of the press is such as to cause the disk to be indexed more than one division, the link *B* can be equipped with a coil spring acting against the connecting member of the press ram, and a stop can be provided for lever *C*, so that the latter will oscillate only the required amount.

It may also be added here that the friction drive for the disk *A* provides a means for stopping the disk automatically in case of jamming when defective shells are fed through, in which case the guard *F* should be made removable, so that the shell can be extracted. After the shell is extracted, the disk must be rotated by hand until it assumes the correct position relative to the ratchet wheel. Corresponding lines scribed on both of these members may be employed for this purpose. It is evident that this arrangement may also be used for feeding shells into the dial with their closed ends at the bottom, provided, of course, that they leave the hopper tube with their closed ends at the top.

Hopper Attachment for Feeding all Shells with Their Closed Ends Up.— Regardless of whether shells are fed from a hopper with the closed end at the top or at the bottom, the attachment shown in Fig. 2 will deliver them to the press dial with the closed end at the top. This device greatly simplifies the design of the hopper, inasmuch as no attention need be paid to the position in which the shell leaves the hopper opening. By modifying the design of the attachment shown, the shells may also be delivered to the dial with the closed end at the bottom. Hence, by constructing two demountable attachments, shells may be made to enter the dial with the closed end at the top or at the bottom, only one hopper being employed in both cases.

The shells are fed from the hopper into the tube *A*. From the tube, they drop into openings in the annular ring *B*, which is given an intermittent rotary motion by means of the ratchet wheel *C* and ratchet lever *D*. This lever is given an oscillating movement through the link *E* which is carried

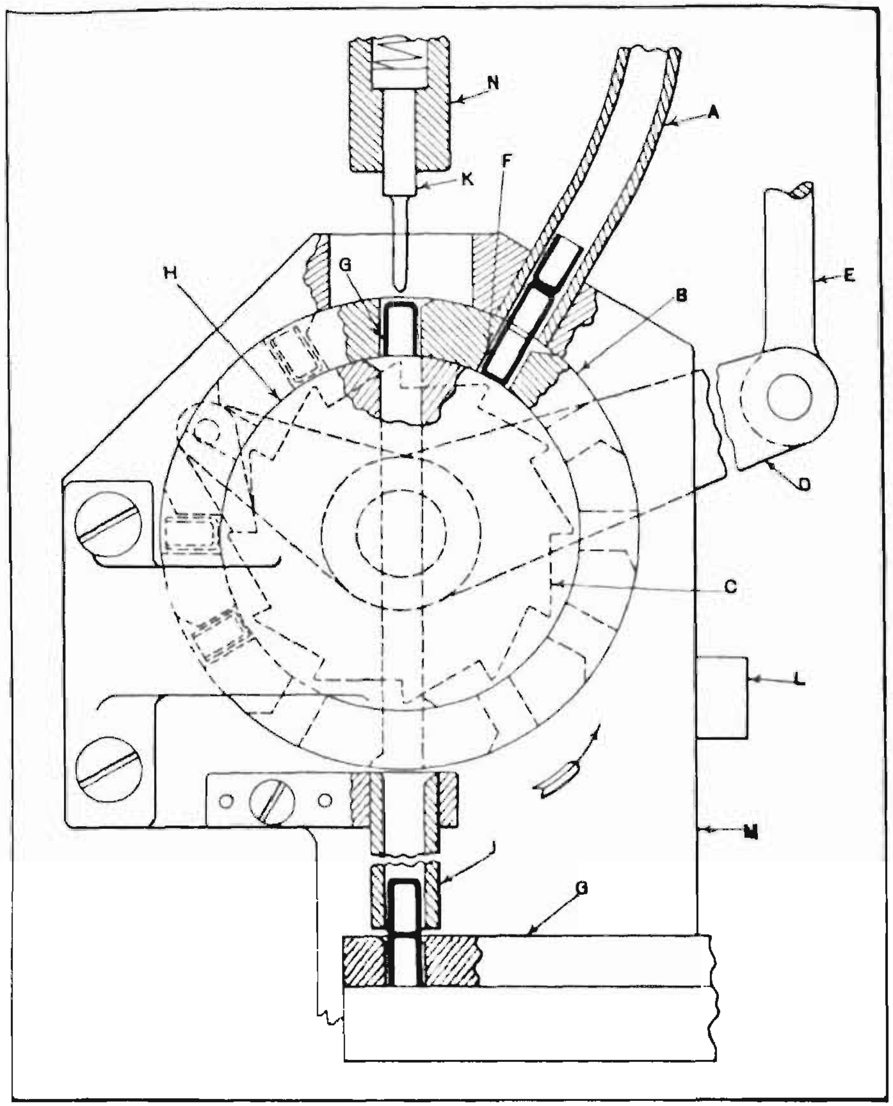


Fig. 2. Device for Feeding Shells into a Dial with their Closed Ends up Regardless of their Position as they Leave the Hopper

in the press ram; and for every cycle of the ram, the ring *B* is rotated one division. If all the shells were to drop into the ring in the position indicated at *F*, they would drop out of the lower end of this ring in the proper position to enter the press dial *G*. But when a hopper of the design mentioned is employed, the shells can take the position shown as they pass down into the tube *A*. Hence, provision must be made for delivering all the shells to the press dial with the closed end at the top. The shell *G* must be inverted before it reaches the dial. Instead of the shell being carried around with the ring, however, it will be forced down through a hole in the stationary core *H* by means of the plunger *K* secured to the press ram; and from there it will pass once more through the ring and into the tube *J*.

When a shell in the position indicated at *F* is indexed to the top of the ring, it will not pass down through the core *H*, as the plunger *K* will simply enter it without making contact. The shells are prevented from dropping by gravity at the top of the ring by a spring (not shown) which bears against the side of each shell as it passes this point. The tube *J* is made long enough to contain six shells, or half the number that can be held in the ring *B*. This length of tube was necessary, as, with a shorter tube, it is possible for the shells to pile up and be carried around once more past the tube *A*.

The ring is operated through a friction disk from the ratchet wheel *C*, and the position of the holes in the ring, relative to the hole in the core *H* and the tube *J*, is governed by the downward movement of the lever *D*, which is limited by the stop *L*, secured to the base *M* of the device. The base, in turn, is securely fastened to the press bed. Lever *D* strikes stop *L* before the downward stroke of the press has been completed, link *E* sliding, against the pressure of a coil spring, in a projection on the press ram. In case a damaged shell is indexed into position under plunger *K*, the plunger, instead of moving downward, will be held suspended, as it telescopes within the holder N, and the shell will be prevented from passing down into core H (see also simple device for feeding shells open end up, Volume 1, page 455).

Feeding Split Rivets to Power Press.—The automatic feeding mechanism, Fig. 3, is used for feeding split brass

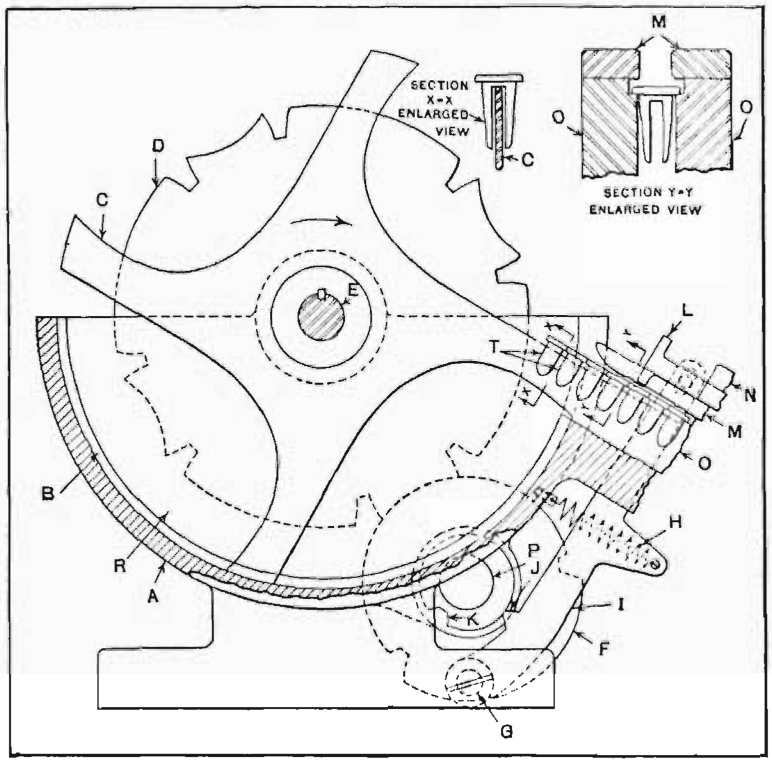


Fig. 3. Rotary Hopper for Feeding Split Rivets to a Dial Press

rivets to a dial on a power press, where they are assembled into porcelain fuse plugs. The hopper A, fastened to a bracket on the press in a position above the dial, is equipped with the four-bladed member C. For each revolution, member C is given an intermittent motion including four equally spaced dwells, through the action of the gears D and F

keyed to shafts E and P, respectively. Shaft P is the driving member for the hopper, and receives its motion through a chain drive connected to the press crankshaft.

The rivets are placed in the reservoir R, and as member C revolves in the direction of the arrow, some of them will straddle the blades, as shown in the cross-sectional view x–x. As the member C continues to revolve, the rivets slide along the curved edge of the blade until they arrive at the position shown at T. Each blade dwells long enough at this point to permit the rivets to slide off and into the chute O. To facilitate the delivery of the rivets to the chute, the latter was constructed to support the rivets under their heads (see section y–y), the strips M acting as a guard to prevent them from piling up on each other on the incline.

The top of the chute is kept clear of incorrectly delivered rivets by the sliding finger L. This finger receives its motion from the lever I, pivoted at G, and is actuated by the revolving cam K which engages the lug J. The spring H carries the finger toward the hopper, while the outward movement is positive. The plate N serves as a gib for the sliding finger L.

Feeding Round Pins to a Dial Press.—A hopper for feeding round pins to a dial press is shown in Fig. 4. With this design, the open end K of the revolving conical coil of tubing E is continually passing through the mass of pins in the hopper reservoir L, some of which enter the tubing at K and slide, both by gravity and by the pushing action of the pins entering, down the incline to the center of the coil where they pass into a stationary tube F leading to the press.

The bracket C, fastened by screws to another bracket A on the press, has two bosses which serve as bearings for the turned shank M on the revolving member G. The latter, driven by the flanged pulley B, which is connected by a belt to a pulley on the press crankshaft, serves as a holder for the coil E. The lower end of this coil is straight and passes

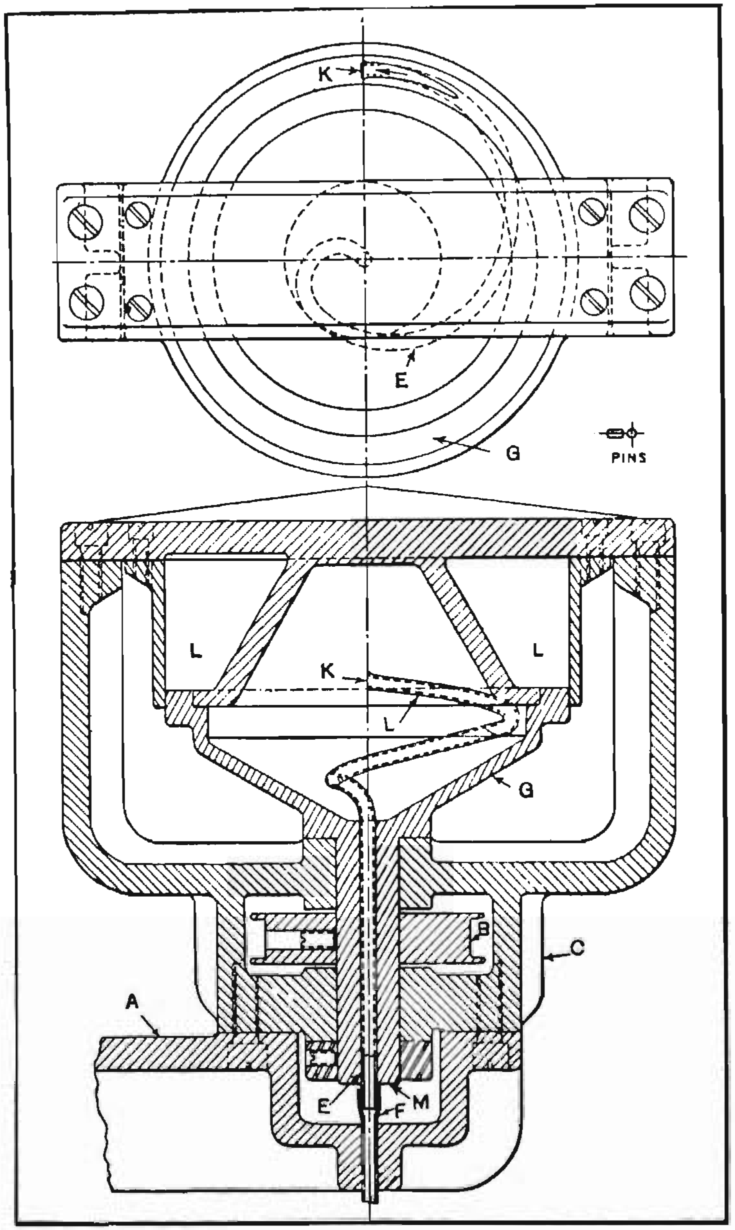


Fig. 4. Hopper with Revolving Coil of Tubing through which Pins are Fed to Dial

down through the shank M into the swaged end of the stationary tube F, while the upper end of the coil passes at an angle through the face of the outer flange on the member G, as illustrated at N. The hole for the tubing in member G at N is first cut through with an end-mill, and after the tube has been properly located, it is babbitted in place. The babbitt is next doweled to member G to complete the joint. With this type of hopper, the length of the pins to be fed governs the diameter of the coil. Such a hopper could not be used for very long pins, as the diameter of the coil would have to be so large as to be impractical.

Ejecting Shells that Enter Hopper Chute Wrong Side Up.— Among the many problems encountered in designing hoppers for feeding shells to dial presses is the delivering of each shell to the dial right side up. This is accomplished in one plant by means of the device shown in Fig. 5. The device is attached to the table of the press at the dial end of the chute, and is equipped with a vertical plunger for keeping in the chute those shells that have their open sides up. Another plunger ejects from the chute the shells that have their closed sides up.

A screw shell A is shown at the end of the chute B. This section of the chute is adjacent to the dial (not shown) and is secured in the bracket C; the bracket, in turn, is fastened to the side of the press table. Slide D, equipped with a spring-actuated ejecting plunger E, is mounted in bracket C, and is given a reciprocating movement by means of the bell-lever F. This lever is oscillated by a link on the arm G, which is bolted to the press ram.

A vertical sleeve H, also secured in this arm, supports the spring-actuated plunger J. This plunger serves to prevent those shells that have their open sides up from being ejected from the chute. For example, the shell A is shown with its open side up; now, as the press ram descends, plunger J enters and bottoms in the shell. As the ram continues to descend, the plunger remains stationary; in the meantime, the slide D has advanced toward the left, forcing plunger E against the side of the shell. Continued downward movement of the ram will merely result in both plungers J and E collapsing into their holding members. Thus, plunger J acts as a lock, preventing plunger E from forcing the shell out of the chute.

Just before slide D has completed its movement toward

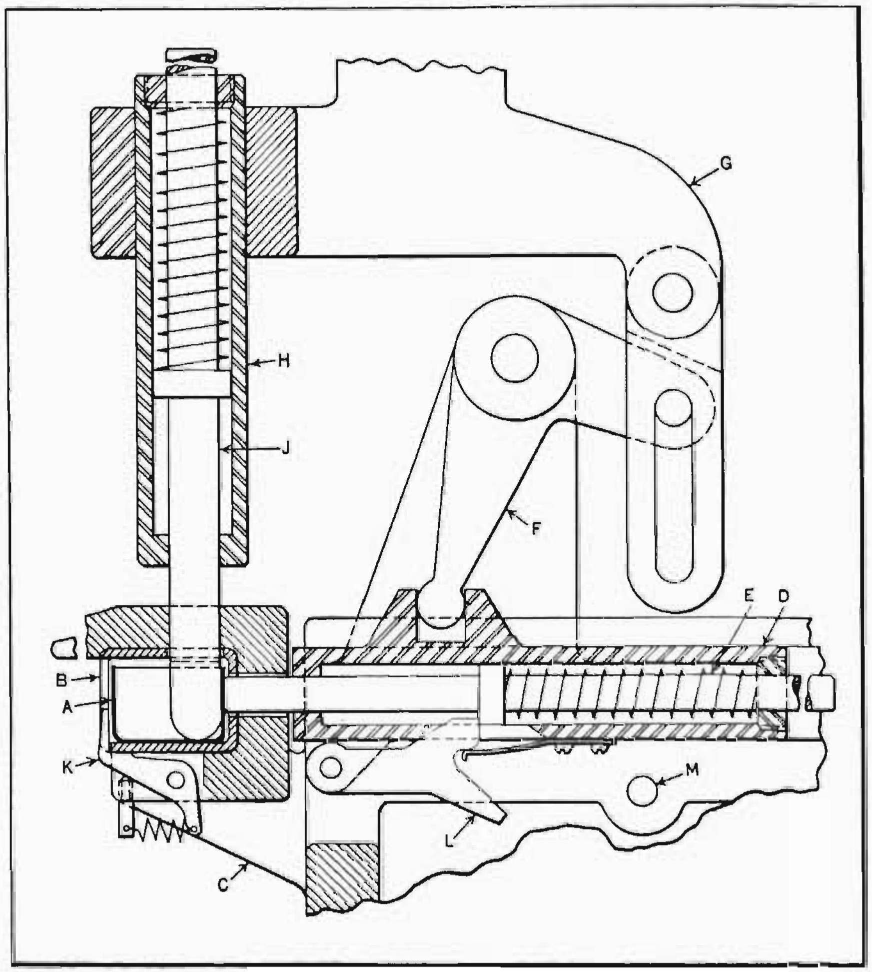


Fig. 5. Device for Ejecting Shells from Hopper Chute which Enter Chute Closed Side Up

the left, the latch L snaps into place behind the shoulder on plunger E, locking this plunger to the slide. Consequently, when the slide is returned toward the right, the pressure of plunger E on the shell will be released before the vertical plunger J leaves the shell. If this provision for locking plunger E were not made, the shell would be forced from the side of the chute. However, just before slide D has reached its extreme position at the right, latch L is disengaged from the plunger by coming in contact with the sta-

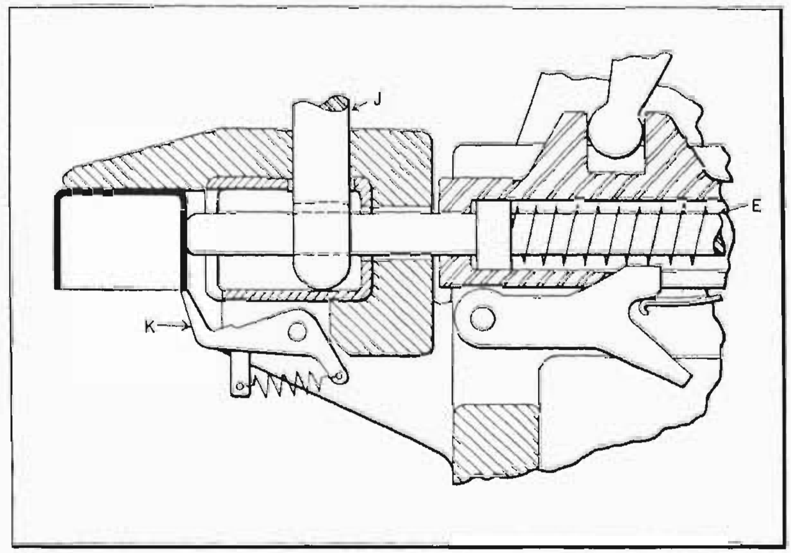


Fig. 6. Action of Device Shown in Fig. 5 in Ejecting a Shell from the Chute

tionary pin M. This allows plunger E to return to its ejecting position in the slide. At this time, however, both plungers are out of the chute; hence the shell is free to enter the dial.

When a shell enters the chute with its closed side up, it comes into position under plunger J, which, in this case, rests on top of the shell. Then, when plunger E moves toward the left, the shell will be pushed out of the chute, as shown in Fig. 6. During the ejecting process, the vertical

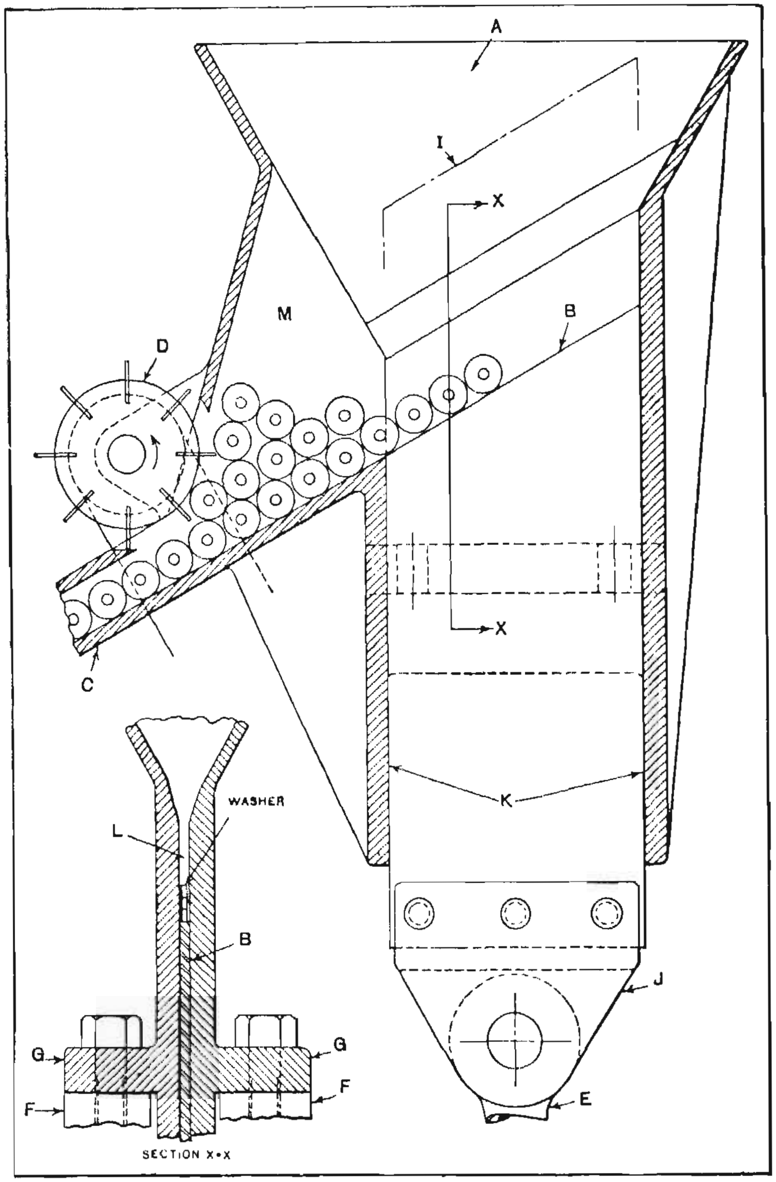


Fig. 7. Hopper of Simple Design for Feeding Washers to a Dial Press

plunger merely drops off the edge of the shell and to the bottom of the chute. This plunger is forked to allow it to straddle plunger *E*. It will also be noted that a guide rail *K* is provided to retain the shells in the chute while the ejecting device is inoperative. This rail is spring-actuated to allow it to return to its normal position. The ejected shells drop on a belt conveyor and are returned to the hopper.

Hopper for Feeding Washers to a Dial Press.—The hopper shown by the diagram Fig. 7 was designed for feeding special brass washers to a dial press, where they are assembled to electrical toggle switch levers. As slide *B* passes up and down through the mass of washers at *A*, some of the washers drop into spaces *L* or *M*, which are slightly wider than the washers, and roll down the incline at the top of the slide and thence into the chute *C*. The slide is shown in its lowest position, the highest position being indicated by the light dot-and-dash lines at *I*.

The reciprocating movement of slide *B* is obtained from a crank which is connected to the bracket *J* on the slide by the link *E*. The washers are prevented from jamming at the entrance to the chute by the wheel *D*. This wheel, revolving in the direction of the arrow, is driven by a sprocket and chain from the hopper crankshaft, and is equipped with eight flat spring projections which agitate the washers sufficiently to insure a uniform flow down the chute. The sides of the hopper are carried down at *K* to resist the side thrust of the slide set up by the hopper crank. Any dirt which may enter the hopper will pass out through the chute. The lugs *G*, cast integral with the hopper, provide a means for fastening the hopper to the bracket *F* on the press.

Automatic Ejector of Lift Type for Press Dial.—Fig. 8 shows a device for ejecting an assembled shell from the dial of an inclined press. The device is attached to the press ram by bracket *J*, which holds the post *B* with the pins *E* on which fingers *A* are pivoted. The operation is as follows:

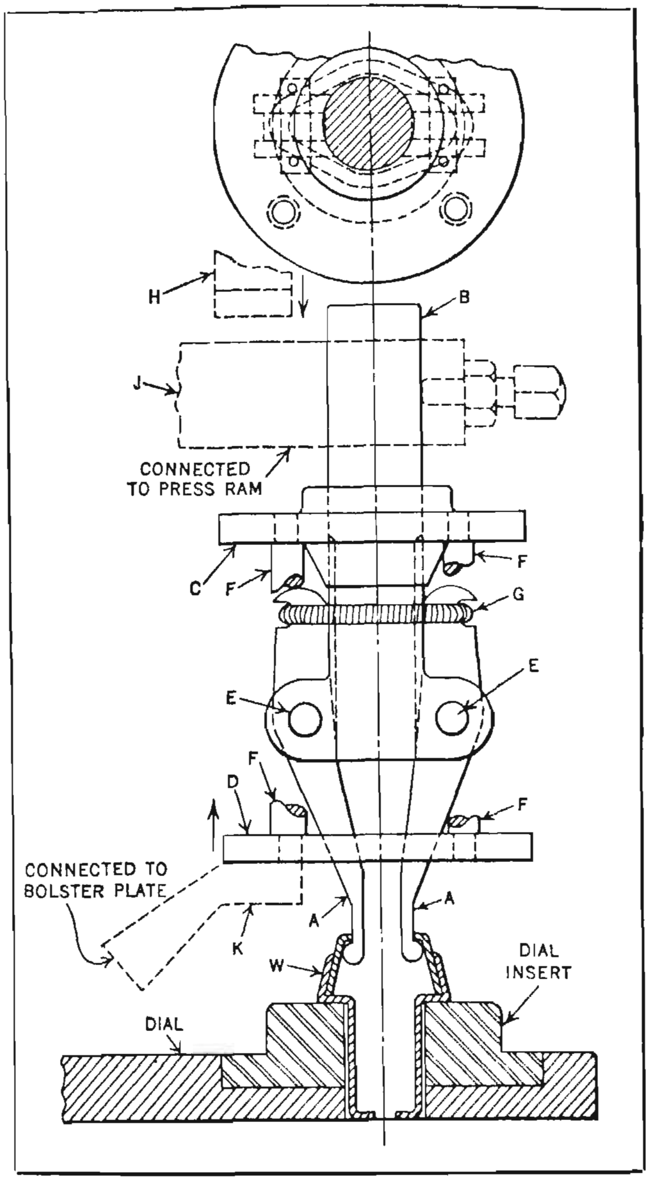


Fig. 8. Device for Ejecting Assembled Part from Dial of Inclined Press

When the ram starts to descend, the fingers *A* are closed at the lower tips, so that they will enter the hole in the work *W*, which consists of two assembled shells. Before the ram reaches the end of the down stroke, the stop *K* pushes the sliding plates *D* and *C* upward. These plates are connected by shoulder pins *F*. The illustration shows the beveled part on plate *C* leaving the fingers *A* so that the spring *G* is permitted to open the lower tips inside the work.

When the ram starts to ascend, the shell is gripped and lifted up out of the dial. Before the ram reaches the top of the upward stroke, the stop *H*, connected with the press frame, pushes plate *C* down. The beveled part of plate *C* then comes in contact with the upper end of fingers *A*, causing the tips to close and allow the work to drop down a chute and slide into a container, thus completing the cycle. The press is operated at a speed of about 75 revolutions per minute.

Device for Ejecting Fuse Plugs from Dial Press by Lifting Fingers.—A device used for ejecting porcelain fuse plugs from the dial of a press is shown in Fig. 9. In this press, the plugs are assembled with the metal caps that retain the isinglass covers. To eject the plugs, two spring fingers *A*, pivoted on the swinging arm *B*, are provided. The arm is mounted on a cam bushing *C*, the bore of which is a slide fit over the post *D* to allow vertical and rotary movement of the bushing. The post is stationary, and is secured to the press by means of the cast-iron bracket *E*. In the side of post *D* is secured the pin *F*, which engages the cam slot in the bushing and imparts the required oscillating motion to the arm *B* during the vertical movement of the bushing.

A flange at the top of this bushing engages the bracket *G*, which is secured to the press ram, so that as the ram reciprocates, the arm *B* will be given a combined vertical and oscillating movement.

As shown, the fingers *A* have gripped the plug preparatory to ejecting it from the dial. In this position, the ram is at its lowest point. As it ascends, the plug is lifted clear of the dial, and further upward motion of the ram causes the pin F to slide in the angular part of the cam groove and swing both bushing and arm enough to carry the plug over the edge of the dial. At this point the top ends of the two fingers A come in contact with the stationary stop H,

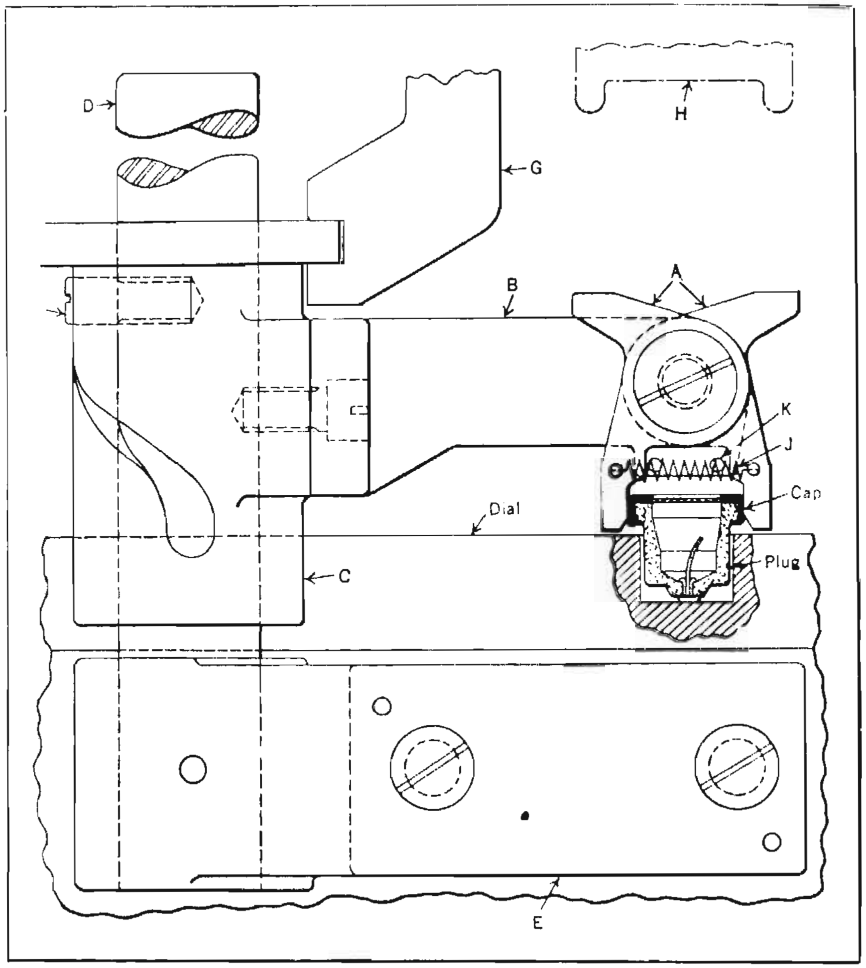


Fig. 9. Ejecting Device for a Dial Press which Lifts the Work from the Dial and Deposits it in a Container or Chute

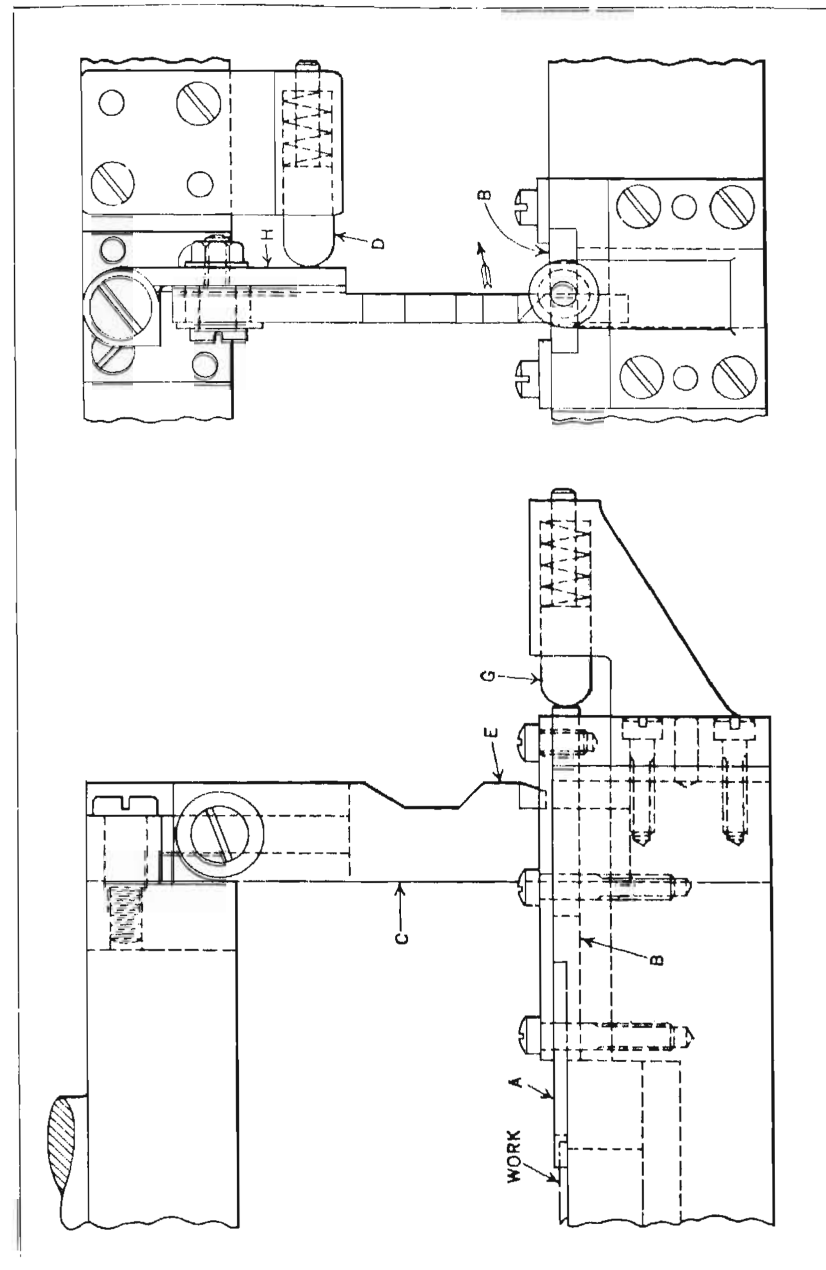


Fig. 10. Die Equipped with a Cam-arm that Imparts a Forward and Return Stroke to a Slide while the Ram Ascends. The Slide is Stationary while the Ram Descends

which opens the fingers, allowing the plug to drop into a chute at the side of the machine. From the chute, the plug slides into a container. A coil spring J is provided to give the required gripping pressure to the fingers. Also, in order to locate the fingers properly over the plug, centralizing pins K are provided. Although the stop H is shown in the illustration directly over the plug, this is not its actual position; it is placed at one side of the dial so that it will release the fingers only when they are above the chute.

Magazine Die with Stop which Shifts while Ejecting Work.— Several automatic magazine dies used for piercing

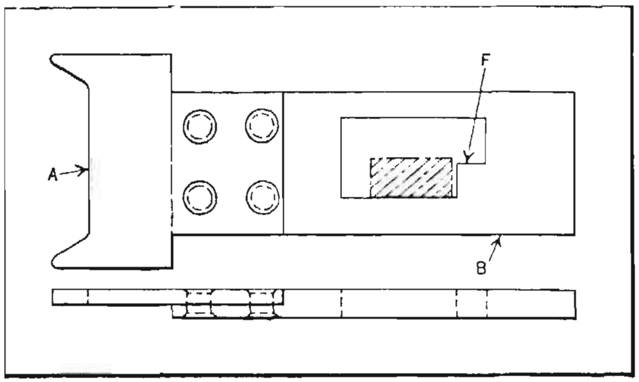


Fig. 11. Stop Operated by the Mechanism Illustrated in Fig. 10

and trimming operations on previously blanked parts are equipped with stop mechanisms like that shown in Fig. 10. During the upward stroke of the press ram, this mechanism serves to move the work locating stop A toward the right to permit the finished part to be ejected and a new blank to be put in place. Before the ram has reached the top of its stroke, the stop returns to its former position against the new blank. During the entire downward stroke of the ram, the stop is stationary.

As indicated in Fig. 11, stop A is riveted to the slide B, which is mounted in the die bolster. Cam-arm C (Fig. 10) is pivoted to the punch-holder and can be swung in the

direction of the arrow. At the top of the stroke, the lower end of this arm assumes a position corresponding with the sectional area in Fig. 11, and is normally held there by the spring-actuated plunger D. As the press ram descends, the angular edge at the bottom of the projection E on the arm engages edge F on the slide, causing the arm to swing in the direction of the arrow. Upon the continued descent of the ram, the projection E passes the corner at F, allowing the arm to swing back to its normal position. During the entire downward stroke, slide B is held against the die by the spring-actuated plunger G, and therefore remains stationary.

As the ram begins to ascend, the angular edge at the top of projection E engages the under side of the corner at F, causing the slide to be moved toward the right. At this point, the finished part is ejected from the die and a new blank is slid into place by means of the magazine feed slide (not shown). As the ram continues its upward movement, the projection E leaves the slide and the slide is returned by plunger G to a position against the new blank. The blank is thus located centrally over the die in readiness for the next downward stroke of the punch. The action of the stop slide can be timed accurately by adjusting the arm to its correct position along the hinge H.

Oscillating Arm for Dislodging Pieces that Obstruct Hopper Feed Exit.— In using hopper feeds of the flat, revolving disk type shown in Fig. 12, there may be trouble due to jamming of the work on the aligning strip A at the point where the pieces leave the hopper. The purpose of strip A is to line up the work so that it will enter the chute opening in a predetermined position. The clogging of the hopper exit results in loss of production and unnecessary wear on the punch and die members. As an operator often runs three or four presses of this kind, jamming or clogging of the work in the manner referred to may not be noticed immediately.

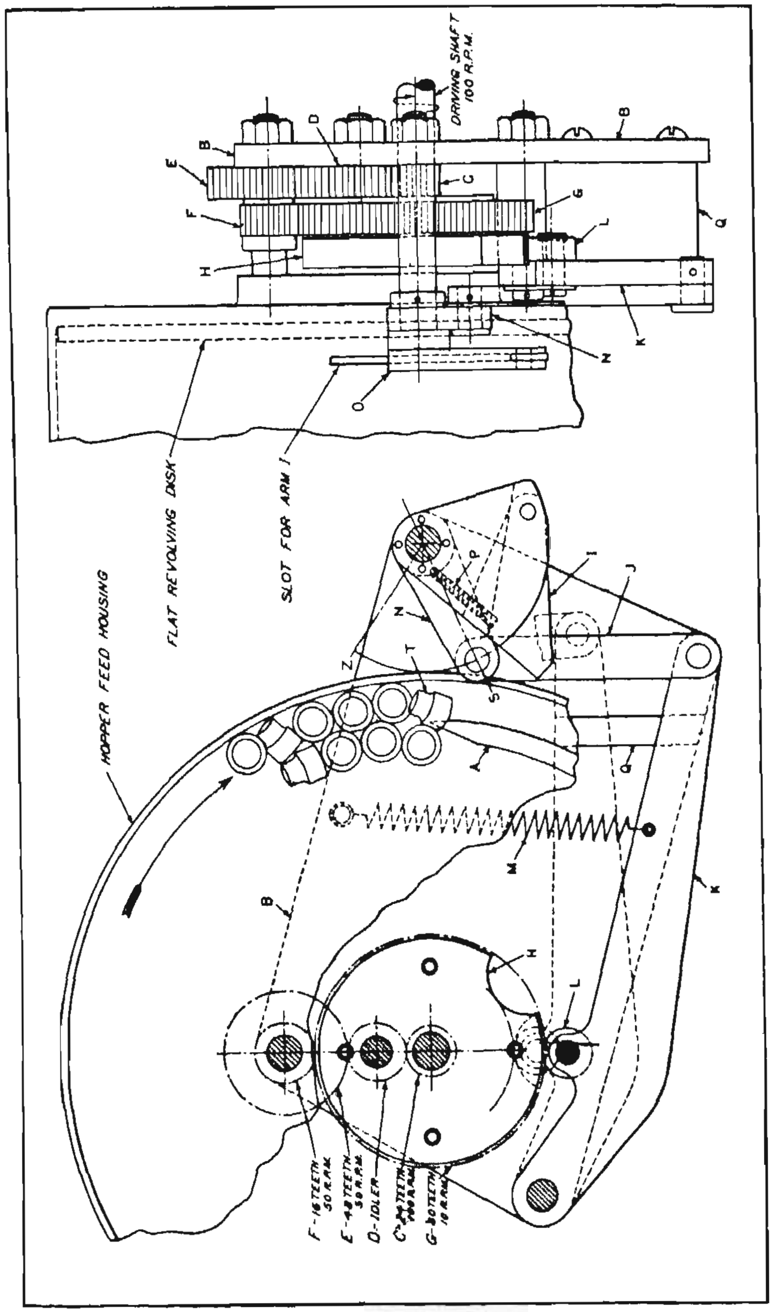


Fig. 12. Revolving Disk Type Hopper Feed with Mechanism for Dislodging Clogged Pieces

The entire device is mounted on the baseplate *B*. This plate is secured to the under side of the hopper feed housing. The gear *C* is an integral part of the driving shaft, which revolves at a speed of 100 revolutions per minute. The idler *D* transmits motion to the cluster gears *E* and *F*. Gear *F* meshes with gear *G*, which is free to revolve around the driving shaft and is riveted to cam *H*. Cam *H* revolves at 10 revolutions per minute.

The piece of work *T* which has obstructed the exit of the hopper is swept away or dislodged in the following manner: The circular cut-out on the cam *H* causes a sweeping motion of arm *I* from *S* to *Z* when the cam-roll *L* drops into the cut-out. The cam-roll is attached to lever *K*. Link *J* connects lever *K* to lever *N*. Lever *N* is riveted to the holder *O* of the arm *I*. Cam-roll *L* is kept in contact with the periphery of the cam by spring *M*. The small spring *P* provides flexibility for the arm *I* on the return movement. The shoe *Q* serves as a guide and steadyrest for lever *K*. The mechanism described can, of course, be applied to work of various shapes by making suitable alterations.

Transfer Mechanism for Stacking Parts on Rods as They Leave the Die.— In making parts such as shown in the lower right-hand corner of Fig. 13, it was necessary to stack them on rods with the irregular-shaped holes in correct alignment as they left the combination piercing and cut-off die. Stacking the parts in this manner facilitates subsequent operations. The transfer mechanism shown in Figs. 13, 14, and 15 provides an efficient means for stacking the parts. It is mounted at the right-hand end of the die and is operated by a cam attached to the punch-holder, the parts being stacked on rod *W*.

Referring to Figs. 13 and 14, the punch-holder *A* carries the piercing punch *B* and the cut-off punch *C*. On block *D*, which is secured to the die-bed, are mounted the slides *E* and *F*. Slide *E* carries the auxiliary cross-slide *G* which supports the shaft *H* at its left-hand end. The right-hand end of this shaft is supported by a double over-hanging bearing which is part of slide *E*. The left-hand end of shaft *H* is enlarged and recessed to slip over the end of the work shown at *J*.

Slide *F* is given a reciprocating movement by means of the cam *K* on the punch-holder through lever *L* and roller *M*. Bracket *X* supports lever *L* at its upper end. Slide *E*

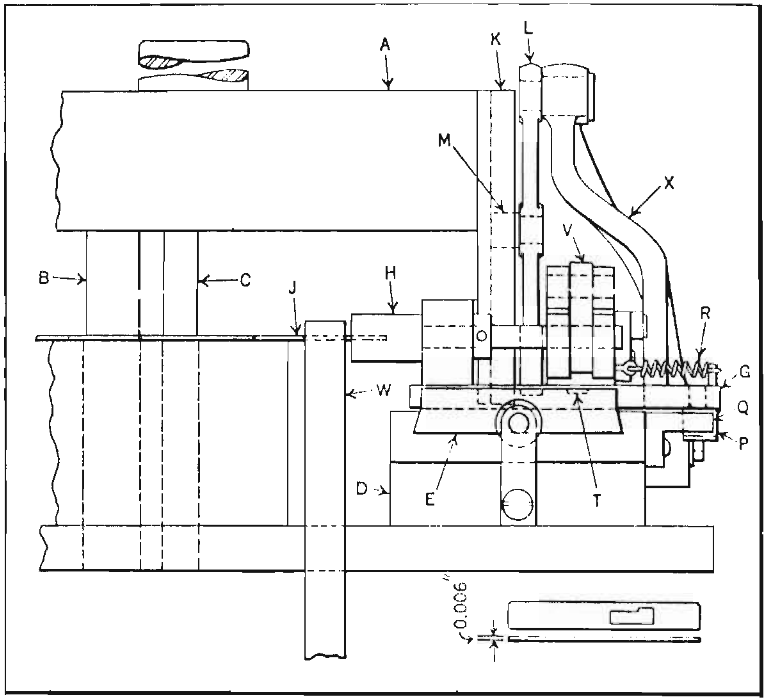


Fig. 13. Front Elevation of Mechanism for Transferring Parts Direct from the Die to the Stacking Rod

is backed up by the spring *N* on the flanged stud *O*. The flange of this stud serves as a stop for slide *E*. Slide *G* carries roller *P*, which is held in contact with cam *Q* by spring *R*. Slide *E* carries the spring-actuated latch *S* which engages slot *T* in slide *G*, and is connected to slide *F* by the slotted link *U*.

As indicated in all three views, the ram is in its lowest

position. The strip has been fed toward the right, its end entering the recess in shaft *H*. The part is then pierced and cut off by punch *C*. Slides *E* and *F* (see Fig. 14) are in their extreme right-hand positions. Latch *S* is held up out of engagement with slot *T* by link *U*, the screw at the rear end of which is in contact with the rear of slide *F*. Slide *G* is at its extreme left-hand position (see Fig. 13)

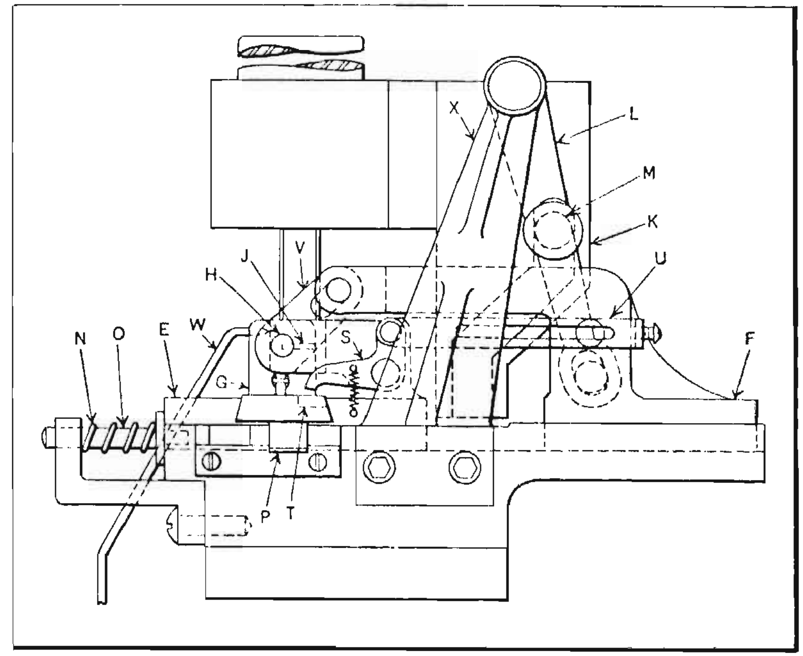


Fig. 14. Side Elevation of Transfer Mechanism, Showing the Action of the Slides Operating the Transfer Member

with the roller *P* in contact with the low part of cam *Q*. It will be noted in Figs. 14 and 15 that the front edge of work *J* is in alignment with the axis of the shaft *H*, so that when this shaft rotates, the work will rotate around its own front edge.

As the ram ascends, the cam *K*, acting on lever *L*, moves slide *F* toward the left until its inner end is in contact with the inner end of slide *E*, which, up to this point, has remained stationary. Prior to this, the motion of slide *F* has been transmitted to link *V*, which causes shaft *H*, to which the link is keyed, to revolve 90 degrees. At this point the work is standing on edge. Latch *S* has been permitted to swing downward; but as it is not in alignment with groove *T*, it merely rests on top of slide *G*.

Further upward movement of the ram causes slide *F* to push slide *E* toward the left, so that the work is placed over rod *W*. As the ram continues to move to the top of its

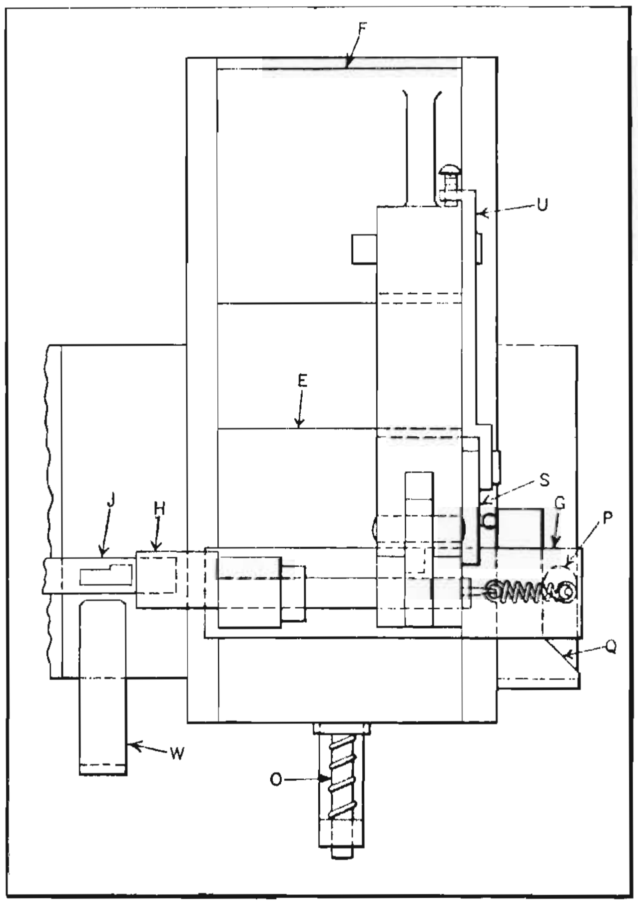


Fig. 15. Plan View of Transfer Mechanism, Showing the Position of the Part after Entering the Transfer Member

stroke slide *G* is drawn to the right by the action of roller *P* on cam *Q*. The work cannot follow this movement, being restrained by the rod *W* in the work-slot; hence shaft *H* is entirely removed from the work, allowing the latter to drop to the bottom of the rod. At this point in the cycle of operations, slide *G* is in its original position, at which time latch *S* is free to drop into the groove *T*, thus locking slide *G* to slide *E*.

As the ram descends, slides *E* and *F* move toward the right until the flange of stud *O* comes in contact with block *D*, which discontinues the movement of slide *E*. The movement of slide *F*, however, continues, and through link *V* revolves shaft *H* to its original position. As slide *F* approaches the end of its return movement, it engages the adjustable screw on link *U*, disengaging latch *S* from slide *G* and causing this slide to be drawn to the left (see front elevation, Fig. 13) by spring *R*. After slide *G* moves toward the left, the enlarged end of shaft *H* is in position to receive the end of a new part, thus commencing another cycle of movements. Rod *W* is about 4 inches high and will accommodate 500 pieces. It can easily be removed from the die when it has been completely filled, and replaced by another rod, after which the cycle of operations is repeated.

CHAPTER XVI

MISCELLANEOUS MECHANISMS OR MECHANICAL MOVEMENTS

Whenever mechanisms have a similar function or a common operating characteristic, they have been grouped together in chapters both in this volume and in Volume I to assist the user in finding whatever general type of mechanism may be wanted. In this chapter will be found mechanisms of such a miscellaneous character that they cannot be placed in any general classification.

Spherical-Elliptical Movement of Sewing Machine Double-Lock Stitch Mechanism.—Many of the sewing machines used in manufacturing clothing, bags, awnings, etc., use what is commonly known as a double lock stitch. These machines have two needles operating at right angles to each other. The lower needle operates beneath the throat plate that supports the goods being stitched. This needle is commonly termed a "looper," as it does not pierce the goods, but passes into and out of the loop of thread made by the other needle in its vertical motions. The loop of thread is formed at the desired location below the throat plate during the upward motion of the needle by permitting the thread to become slack at the proper time, thus causing it to buckle and form the loop. A mechanism used to impart the required motion to the lower needle, or looper, is shown in Fig. 1.

The looper *A* is required to pass very close to the needle in taking up the loop of thread. It must hold this thread loop and position itself on the opposite side of the needle's path by the time the needle has descended below the throat plate in forming the next stitch. From this it is apparent that the looper must have a back-and-forth motion at right

angles to the needle and at right angles to the direction in which the goods travels. In addition, the looper must have a back-and-forth motion at right angles to the needle in the line of travel of the goods being stitched. This latter motion is commonly called the "avoider" motion, since its object is to avoid or prevent interference with the needle. The path followed by any point on the looper *A* consists of a closed curve that resembles an ellipse bent to fit the surface of a sphere.

The main shaft *B* carries the flat strap eccentric *C* and the ball-joint eccentric *D*, which drive the rocker arm *E*, mounted in the spherical bearing *F*. Arm *E* carries the looper *A*. Eccentric *C* imparts motion to arm *E* by means of pin *G*, which passes through the center of bearing *F* with which it is in sliding engagement. This gives the looper *A* an oscillating movement. Eccentric *D*, by means of its connection with the ball-ended bellcrank *H* and link *J*, causes arm *E* to oscillate about the center of pin *G*, so that *A* moves back and forth between the positions shown by the full lines and by the dotted lines at *M*.

It will be noted that eccentrics *C* and *D* impart motions to *A* which are approximately at right angles to each other, in producing the spherical-elliptical movement. Eccentric *C* gives the needle the "avoider" motion, while eccentric *D* imparts the motion that causes the looper to pass in and out of the thread loop formed by the needle that pierces the goods.

Considering the action of eccentric *C* in producing the "avoider" motion, it will be noted that only one component of the circular motion of this eccentric imparts motion to the looper, the other component resulting in pin *G* sliding through the ball joint in *F*. With reference to eccentric *D*, it will also be noted that only one component of this circular motion is imparted to looper *A* through its connection thereto by means of bellcrank *H*, link *J*, ball-pin *K*, and rocker arm *E*, whereas the other component of this motion causes

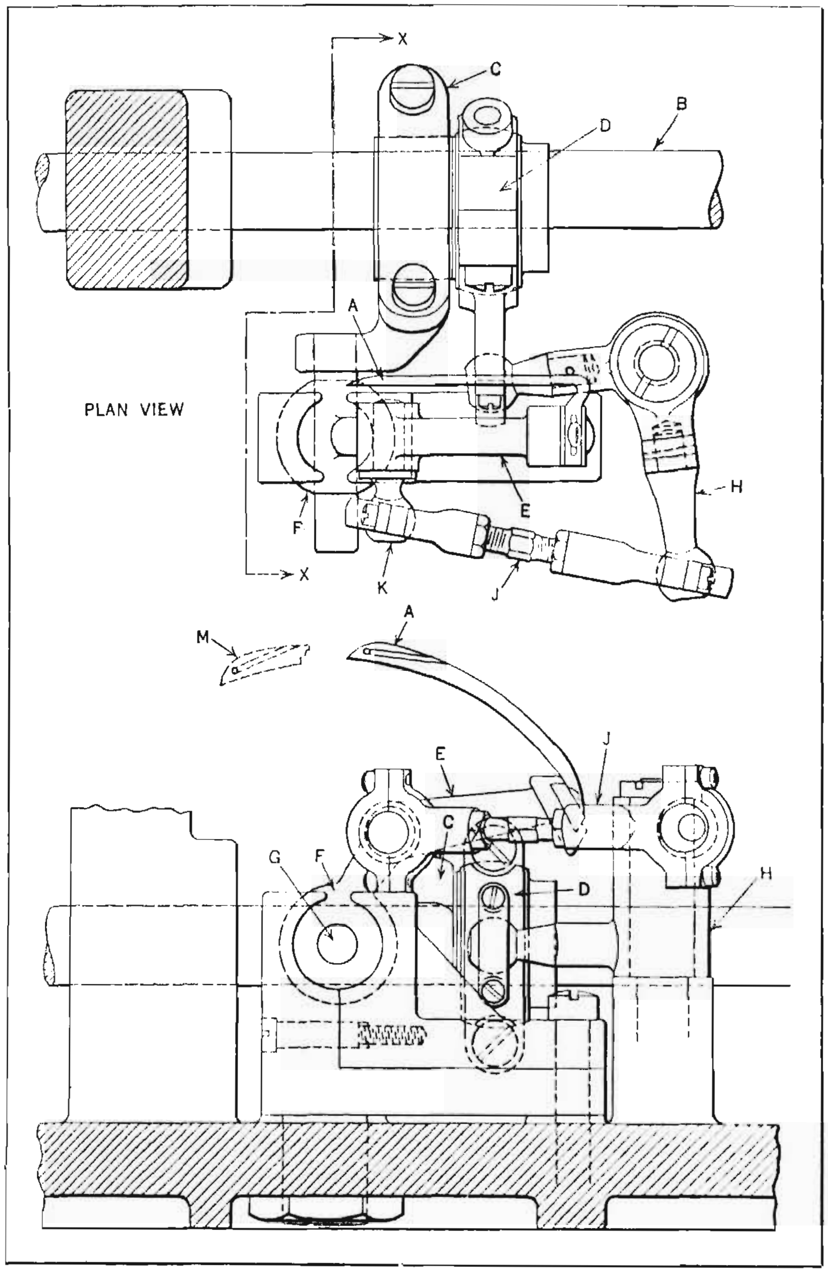


Fig. 1. Mechanism in which Ball Joints, Links, and Levers Operated by Two Eccentrics are Used to Obtain Spherical-elliptical Movement of Member A

rotation about its connection with the bellcrank *H*. Thus the motion imparted to the looper *A* may be considered as being the resultant of two circular motions at right angles to each other. The relative values of these two motions have been changed by means of levers, in order to give to each the desired amplitude.

The motion described could be obtained from a single eccentric, except for the mechanical difficulties encountered in obtaining suitable proportions for the components required for such motion. The motion produced by the mechanism illustrated is the resultant of two simple harmonic motions

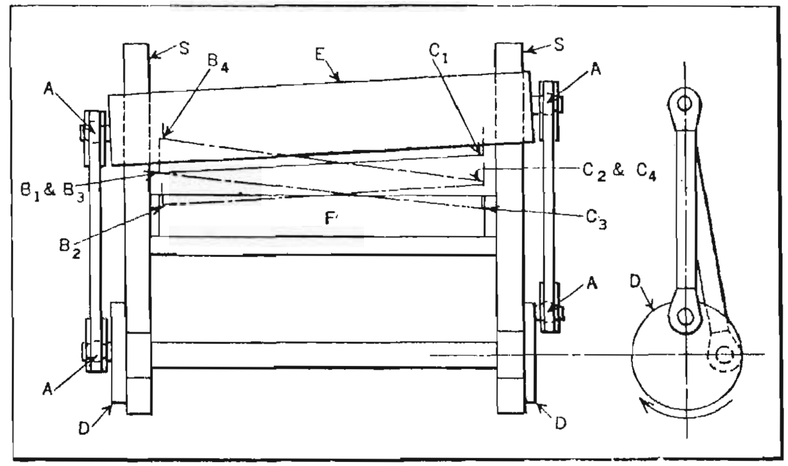


Fig. 2. Mechanism for Producing Shearing Motion

at right angles to each other, each being modified by the length and angularity of the connecting links. This form of looper drive is adapted to high speeds, since it is relatively simple, has few parts, and the few rapidly moving parts required can be made quite light. There are no violent changes in velocity, and the energy changes and friction losses are thus kept at a minimum.

Shearing Motion which Varies Angular Position of Blade.—The mechanism shown in Fig. 2 was applied to a shearing machine to obtain an action approximating that of hand-operated shears. The mechanism is arranged to make one complete cycle of movements automatically when the driving clutch is engaged. The sides S of the shearing machine serve as guides for the cross-bar E which supports the movable blade above the stationary blade F. As the crank disks D revolve, the ends of the movable blade are carried to different positions, as shown at points B_1, B_2, B_3, and B_4 on the left, and by C_1, C_2, C_3, and C_4 on the right. When the left-hand end of the blade is at B_1, the right-hand end is at C_1, and when the left-hand end is at B_2, the right-hand end is at C_2, and so on. This wabble or shearing action is obtained by locating one of the crankpins 90 degrees ahead of the other.

The successful operation of the mechanism is made possible by the spherical pin and connecting-rod bearings A. These bearings were easily produced from soft steel balls purchased from stock and bored out to a press fit on the bearing pins. The boxes at the ends of the connecting-rods were formed by pouring babbitt over the balls while the two parts were mounted on a surface plate in their proper relative positions. A few strokes with a soft-faced hammer served to loosen up the bearings sufficiently to permit them to operate satisfactorily.

Grease lubrication is provided by a hole drilled through the bearing after the babbitt was poured. This hole crossed another hole leading from the outer end of the pin. Side motion of the holder E is prevented by using curved guides having sufficient clearance to prevent binding. Machines equipped with this shearing mechanism are used in connection with the production of cotton batting and similar fibrous parts.

Transmitting Motion to Indexing Plunger by Steel Balls.— A rather unusual application of steel balls for transmitting motion to an indexing plunger on a dial press is shown in Fig. 3. The plunger J slides in the fixed bearing I and receives its motion from the lever A through the steel

balls H. The lever oscillates about the fixed stud B, and its lower end engages a slot in the member C. The latter is a sliding fit in the stationary bearings D and E, and as the lever A oscillates, member C forces the balls up the tube, causing the plunger to move upward into the dial. The plunger is returned to the position shown by the coil spring K which also keeps the balls in contact with member C. It is obvious that with this simple device the tube containing the balls may be bent to almost any shape desired, permitting it to clear any member of the machine.

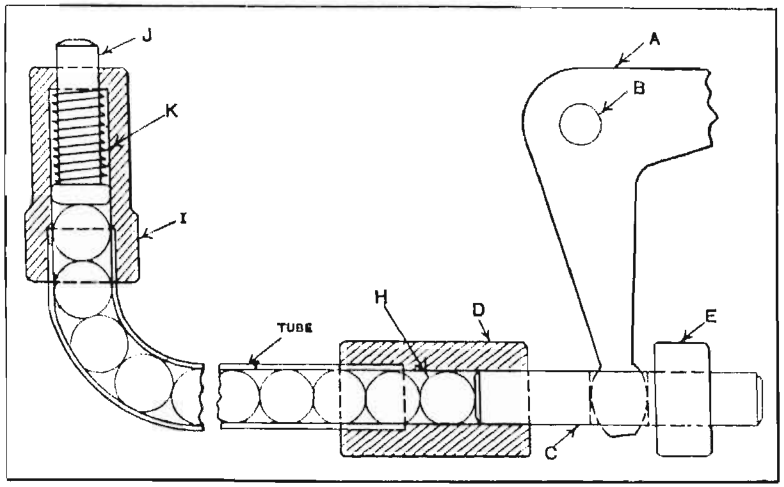


Fig. 3. Transmitting Motion to a Plunger with Steel Balls Confined in a Tube

Counter Used on Type-Setting Machine.— In setting type on a linotype machine (for lay-out pages), each line in the different sections is counted so that the type set will agree with the lay-out calculations. With the instrument shown in Fig. 4, however, no mental effort is required, as the actual count is registered automatically on the dial at X, which is returned to its zero position by one simple motion of the operator's hand.

The lower part of the counter is bolted to a stationary bracket on the frame of the machine. An adjustable finger

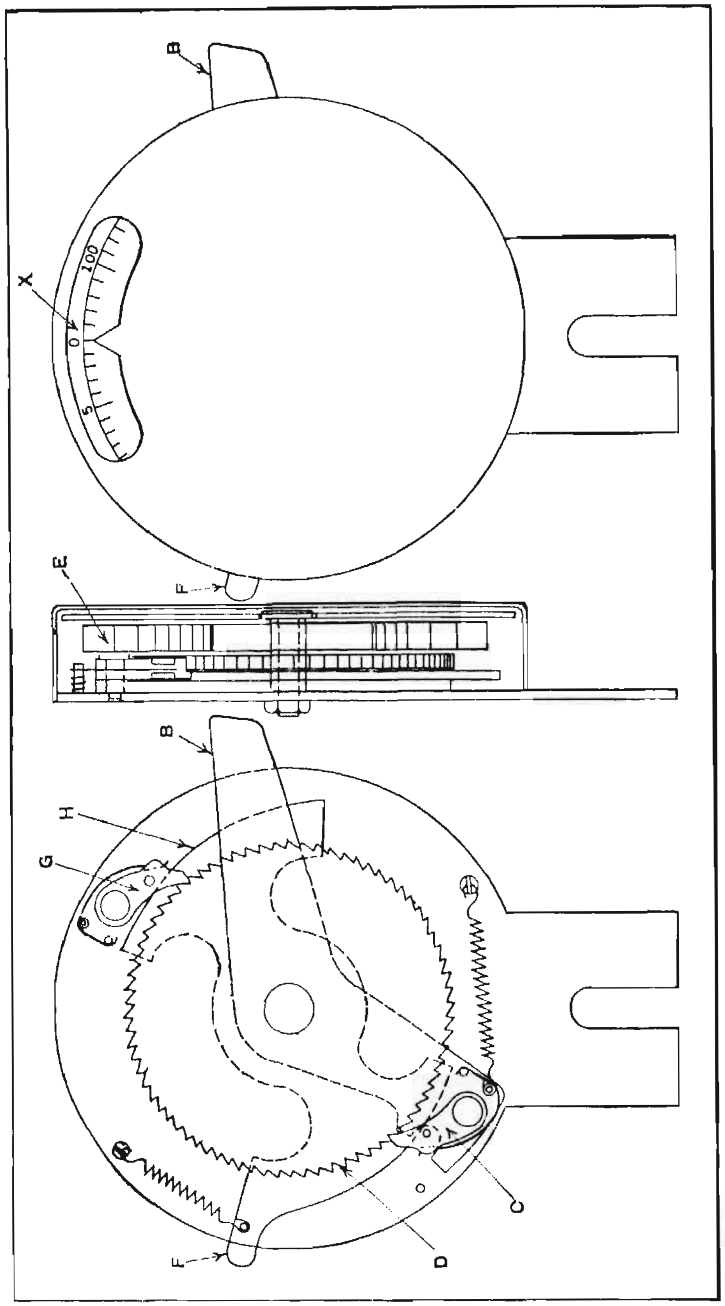


Fig. 4. Counter with Rapid-return Mechanism, Used on Type-setting Machine

attached to a reciprocating member on the machine strikes the end of the lever *B*, causing the pawl *C* to advance the ratchet wheel *D* one tooth, or one graduation on the dial. The dial continues to register until the required number of lines to be cast is indicated. The operator then depresses lever *F*, causing the dial to return once more to the zero position ready for the next group of lines. When the lever *F* is depressed, the cam-plate *H* forces the pins in pawls *C* and *G* outward, disengaging the pawls from the ratchet wheel and allowing the clock spring *E* attached to both cam-plate and dial to return the dial to the zero position. Stop-pins (not shown) are provided to limit the return movement of cam-plate *H* and lever *B*, as well as that of the dial on its return to the zero position. The counter is located on the machine so that the dial is at right angles to the line of vision, assuring easy and accurate reading.

Mechanism for Advancing and Lifting Parts to Clear Lugs of Conveyor Chain.—In a production line where a chain conveyor is used, it is sometimes necessary to lift the article conveyed clear of the lugs on the chain while the chain is in continuous motion.

The mechanism shown in Fig. 5 was developed to meet a requirement of this kind. The article conveyed in this case is required to be rapidly advanced ahead of the chain travel at the last station and brought into position where it can be elevated by an auxiliary mechanism before it is overtaken by the chain lugs.

The action of this entire unit may be summarized as follows: As the chain conveyor carrying the work travels along, arm *E* swings up and in back of the work, rapidly advancing it throughout its forward motion. After the work has been lifted clear, the arm swings in back of the next article.

In the side view the articles conveyed are indicated at *A* and *B*. There is a pair of chain conveyors at *C*, which travel in the direction indicated by the arrow *D*. The arm

E is in contact with the work and has advanced it ahead of the position it would normally have reached as a result of the conveyor movement. The mechanism is shown at the completion of the advance movement. This mechanism has a grooved cam *F* which revolves with shaft *G*. Over the block *H* is assembled a yoke *J*, which carries a roll *K* that rotates in the cam groove, causing yoke *J* to oscillate horizontally.

This action, through the medium of rod *L*, causes lever *M* to rock shaft *N* back and forth. A similar movement is imparted to a pair of levers *P* which, in turn, transmit a

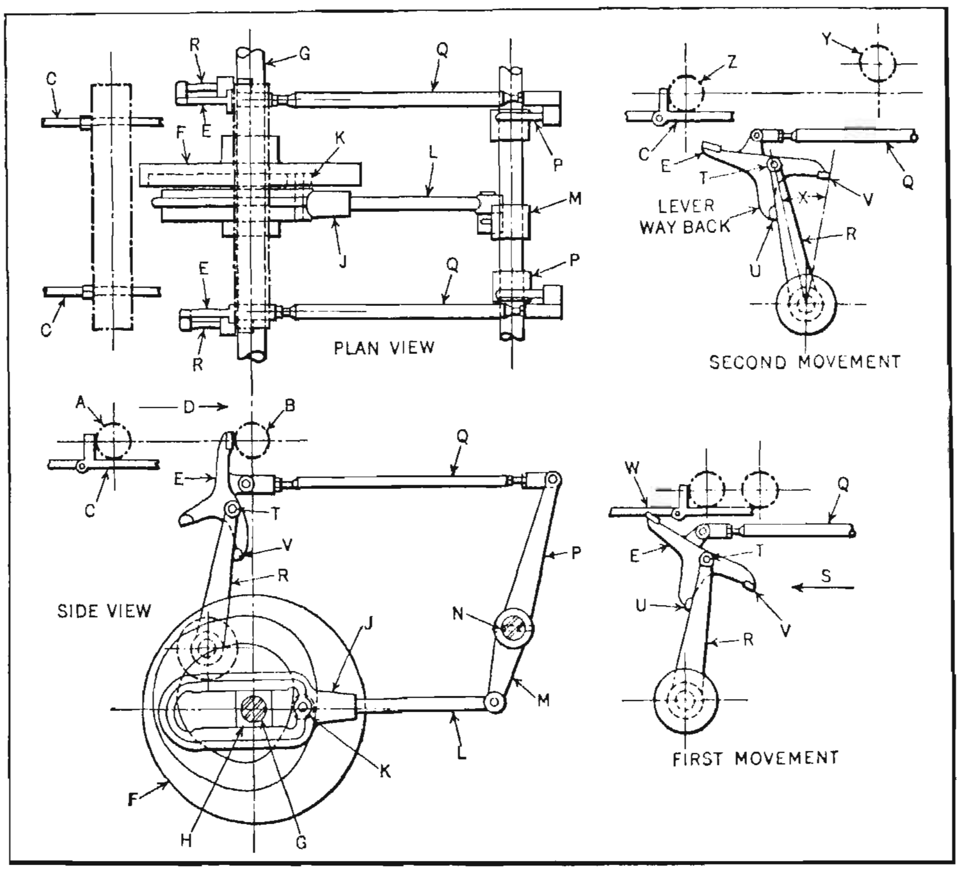


Fig. 5. Mechanism for Raising Work from Chain Conveyor and Advancing it to Operating Position

longitudinal reciprocating movement to a pair of rods *Q*. Rods Q transmit a rocking and transverse movement to the unit consisting of arms *E* and levers *R*, which results in advancing the work *A* and *B* and returning the unit to its starting point.

The side view shows the work *B* fully advanced, while the lower view at the right shows the position of the various members during the first portion of the return movement of arm *E*, as rod *Q* advances in the direction indicated by arrow *S*. This movement causes arm *E* to pivot about

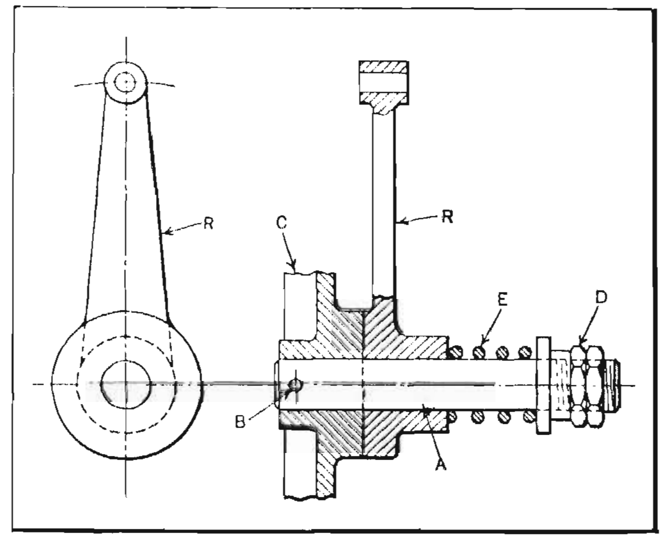


Fig. 6. Details of Friction Drag Applied to Lever R, Fig. 5

stud *T* as a center until lug *U* comes in contact with the side of the lever. It will be noted that, during the advance movement, the lug *V* was in contact with the other side of lever *R*. When the lug *U* is in the position shown in the view in the lower right-hand corner, the top of arm *E* is below the level of the work, as indicated at *W*, although lever *R* has not yet been moved. Continued movement of rod *Q* in the direction indicated by arrow *S* causes lever *R* to be pushed to the left until the entire mechanism assumes the position indicated in the upper view, where lug U is still in contact with the lever and the arc of travel is indicated by X.

Another article has now been brought forward, as shown at Z, and the position of arm E is such that rod Q, in pulling arm E to the right, will cause it to pivot about stud T until it swings behind the work Z, when lug V will be in contact with lever R.

Continued movement of rod Q will cause the article at Z to be rapidly advanced ahead of the chain travel, so that another mechanism (not shown) can pick up the work and raise it clear of the chain lugs into the position indicated at Y.

To insure that arm E will pivot about stud T during the first movement of the advance action and the first movement of the return action without imparting any movement to lever R, the latter lever is constructed as illustrated in Fig. 6. Lever R is shown as pivoting on the short shaft A, which is pinned at B to the side frame C of the machine. Two lock-nuts D permit the tension exerted by the spring E against the lever to be so adjusted that its movement is retarded until there is a definite pull in either direction that is sufficient to assure the correct functioning of the mechanism.

Air-Chuck Valve that Reduces Air Consumption Forty Per Cent.— In shops using a large number of air chucks for machining purposes, the air consumption may be reduced as much as 40 per cent by utilizing the type of pneumatic apparatus to be described. The full-line pressure is commonly used for both opening and closing the chuck. While the full-line pressure is required for closing the chuck, only a fraction of this pressure is needed to release the jaws.

By means of the valve shown in Fig. 7 the same air is employed for opening the jaws as is used for closing them. The principle can be more clearly explained by referring

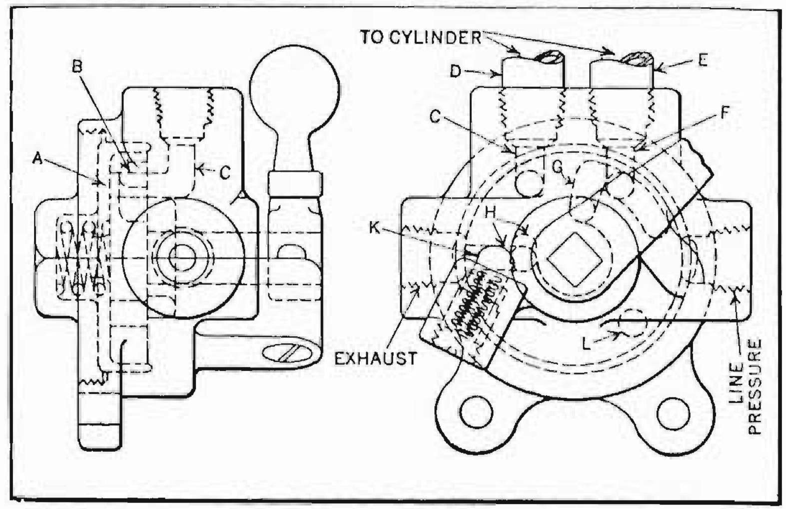


Fig. 7. Pneumatic Valve by Means of which the Same Air is Used for Opening and Closing a Chuck

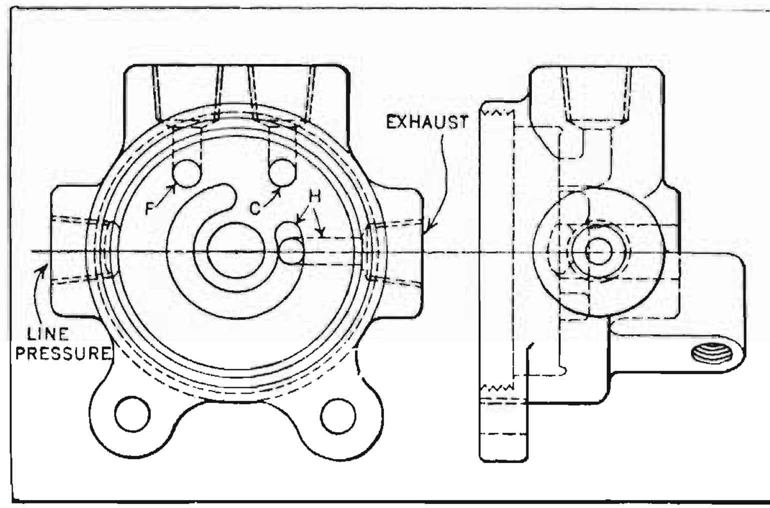


Fig. 8. Detail View of Body of Air Valve, Showing Positions of the Various Ports

to a diagram of the air cylinder (Fig. 10). This cylinder is designed so that when the chuck is closed, the space on both sides of the piston will be about the same, as indicated.

To release the jaws, the piston must be moved toward the left. This is done by exhausting air from side *A* into side *B* until the pressure in both sides is equal. The air in side *A* is then exhausted into the atmosphere, leaving side *B* with about one-half of the line pressure—about 50 pounds—to expand and push the piston toward the left, in this way releasing the jaws. It is very seldom that a pressure of more than 45 pounds per square inch is required to release the chuck jaws.

The valve (Fig. 7) that controls the air in the manner described is of the rotary self-seating type, which requires no packing. The position of the various ports in the valve body and disk are shown clearly in the detail views, Figs. 8 and 9, the reference letters corresponding in all views.

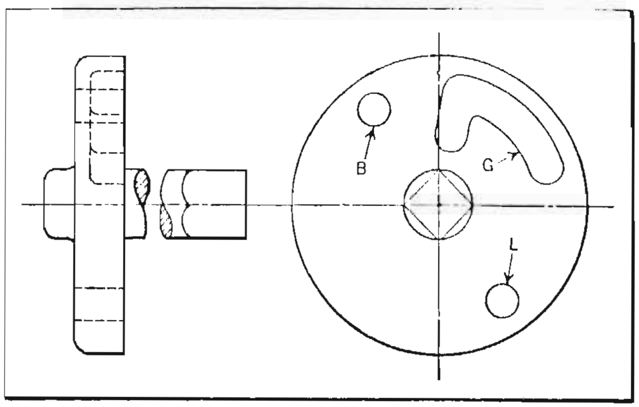


Fig. 9. Detail View of the Valve Disk

With the valve lever in the position indicated in Fig. 7, the air entering at line pressure passes over the disk at *A*, up through the hole *B* in the disk, out through port *C* in the body, and into the pipe *D* which leads into the left-hand side of the cylinder. The air entering the cylinder forces the piston toward the right and closes the chuck. During this movement, the air is exhausted from the right-hand side of the cylinder through pipe *E*, port *F* in the body, port *G* in the disk, port *H* in the body, and out through the exhaust opening to the atmosphere.

To release the jaws, the lever is swung toward the left until it comes into contact with the spring stop *K*. As the lever commences its movement toward the left, ports *G* and *H* are disconnected, thus closing the exhaust from the right-hand side of the cylinder. At this time, ports *C* and *B* are also disconnected, closing the inlet and confining the air at line pressure to the left-hand side of the cylinder. Continued movement of the lever causes port *G* to connect ports *C* and *F*, so that the air is by-passed from the left-hand side of the cylinder to the right-hand side until the pressure on both sides of the piston is equal. Further movement of the lever disconnects port *G* from port *F*, thus closing the latter and confining the air to the right of the cylinder.

As the lever comes in contact with the spring stop *K*, port *G* connects ports *C* and *H*, and the air on the left side of the cylinder is exhausted into the atmosphere. Thus, air at a pressure of approximately one-half the line pressure is left on the right of the cylinder. This air expands, pushing the piston toward the left and opening the chuck jaws. If for some reason this pressure is insufficient to release the jaws, the lever is forced farther toward the left, depressing the spring stop *K*. This additional movement of the lever causes port *L* to connect with port *F*, so that air at line pressure enters at the right of the piston and overcomes the resistance. When released, the lever will immediately spring back to the unchucking position, shutting off the line pressure.

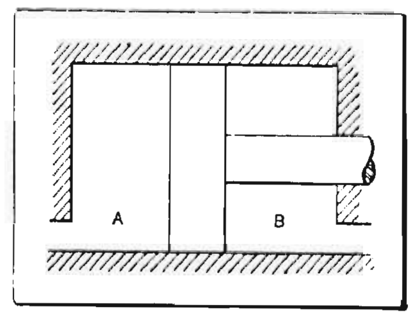


Fig. 10. Diagram of Air Cylinder with Piston in "Closed Chuck" Position

Mechanism for Winding Spherical Cores for Golf Balls.— Cores for golf balls are produced by winding a soft rubber band on a spherical rubber center. The thin rubber

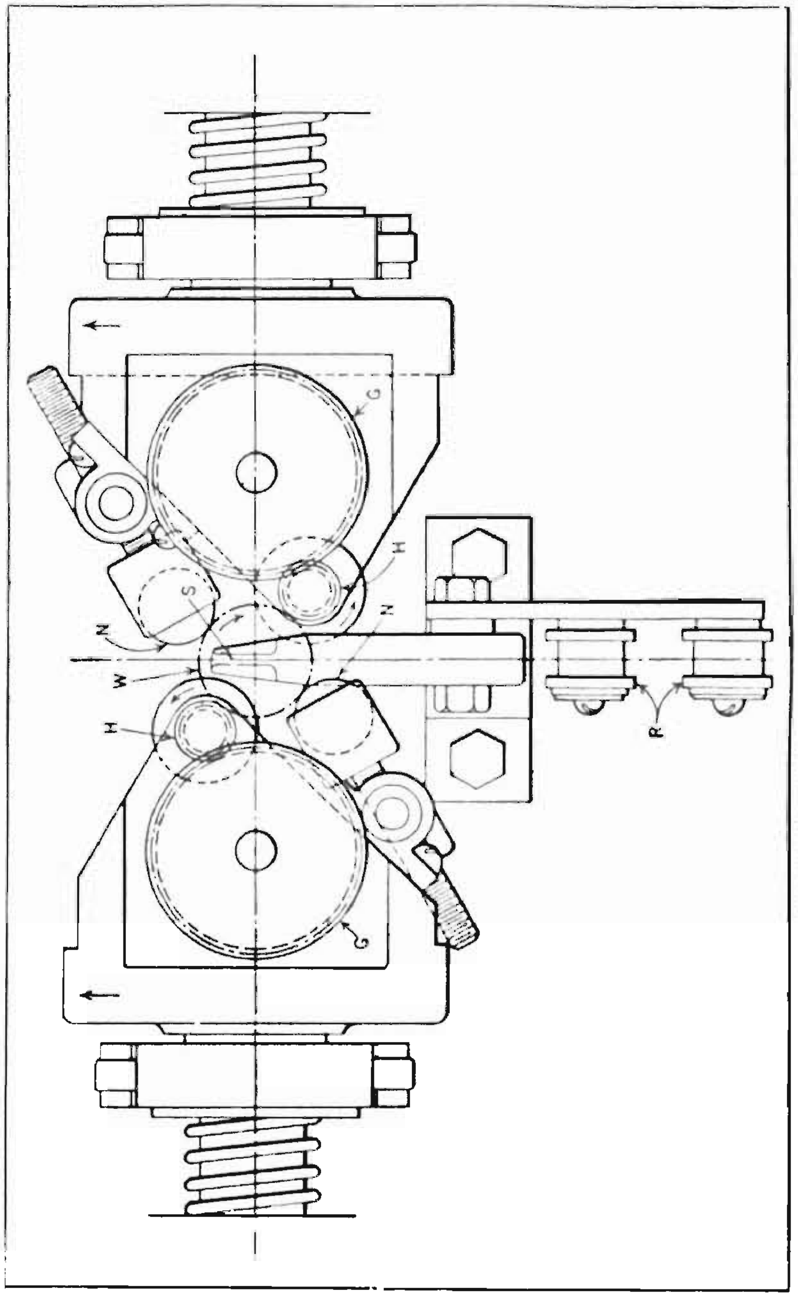


Fig. 11. Plan View of Winding Mechanism Shown in Fig. 12

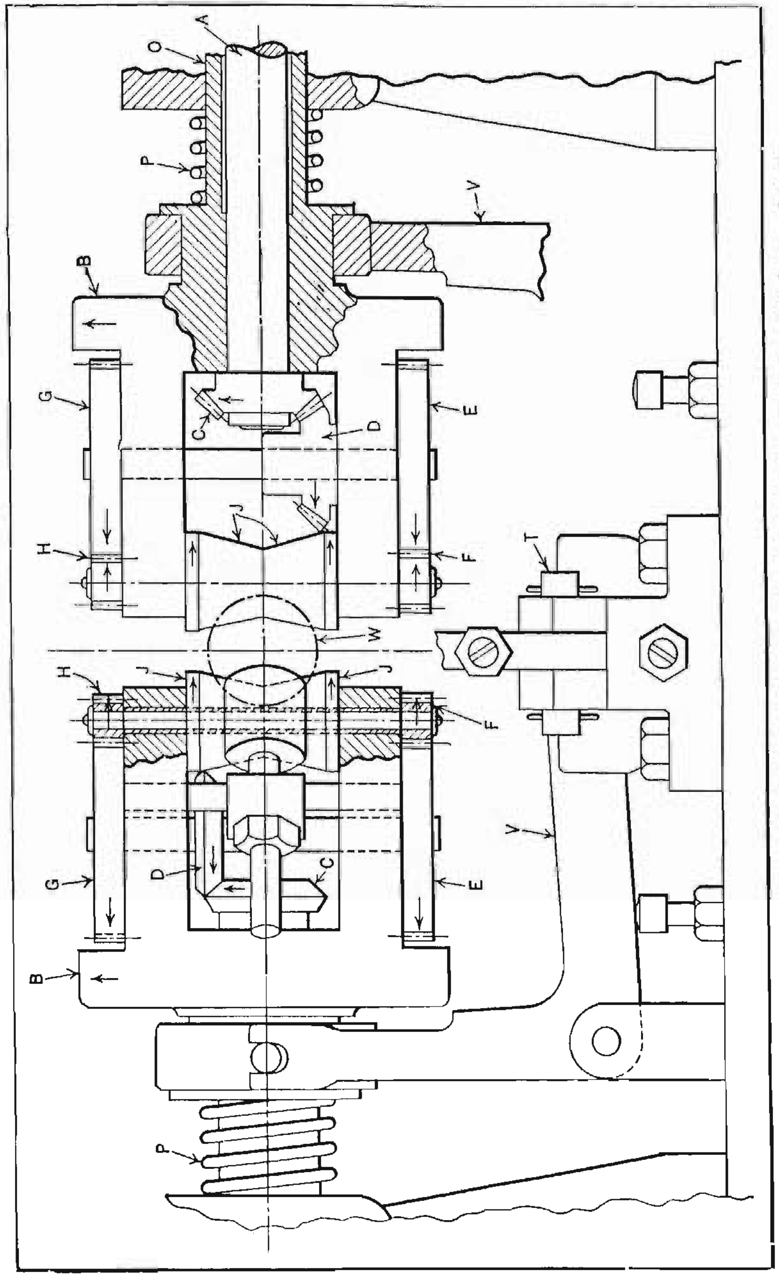


Fig. 12. Mechanism for Winding Thin Rubber Band Evenly over the Surface of a Small Spherical Center of Rubber

band is approximately 1/16 inch wide and must be wound evenly about the rubber center under uniform tension. The completed core must have a true spherical form. These requirements necessitate a somewhat complicated winding or rotational movement, in order to have the crossing points of the rubber band evenly distributed over the spherical surface of the core, which is constantly increasing in size during the winding operation.

The principal elements of the mechanism that provides the necessary winding movement are shown in Figs. 11 and 12. The work is started by placing the rubber center between the four truncated cones or rollers *J* (Fig. 12) mounted in the right- and left-hand winding heads *B*. The rollers are kept in contact with the core by the compressive action of the helical springs *P*, which allow the heads to recede equally as the diameter of the core increases. The bellcranks *V* equalize and centralize the outward movements of the winding heads so that the core is always kept in the central position. The bellcranks are connected with a treadle that permits the operator to withdraw the heads, so that the finished core *W* can be removed. The two ball casters indicated at *N* (Fig. 11) serve to guide or retain the core in its proper position between the rolls.

The two winding heads *B* (Fig. 12) have hollow shafts *O*, which are geared to a countershaft (not shown) at the rear of the mechanism. The countershaft is belt-driven and is provided with a handwheel at one end to allow the mechanism to be operated slowly by hand for making adjustments. The shafts *A*, which run inside shafts *O*, are also geared to the countershaft and run at a somewhat higher speed than shafts *O*. Shafts *A* provide for the secondary revolving movements imparted to the core by the four rollers through the bevel gears *C* and *D*, and the intermittent gears *E*, *F*, *G*, and *H*. The shafts of gears *H* drive the two upper rollers *J*, and the shafts of gears *F* drive the lower rollers *J*,

mounted on the two opposed heads *B*. Thus different movements are imparted to the upper and lower rolls.

The four rollers have their conical surfaces spotted to give a better frictional grip on the core. The winding operation consists of feeding the rubber band by hand over the first and under the second tensioning rolls *R* (Fig. 11) and on through the guide slot *S* down on the soft rubber core which revolves rapidly in two directions simultaneously. The feed guide with the tensioning rolls attached swings on a fulcrum pin *T* (Fig. 12). A helical spring (not shown) attached to the guide provides the proper winding tension. The winding movements are obtained from the constant rotation of the heads *B* in the direction indicated by the arrows, combined with the motions imparted to the core by the four rollers *J* driven by the intermittent gears *E*, *F*, *G*, and *H*. These movements are so timed that the rubber band is wound on the rubber core evenly.

When the core reaches the finished size, the machine is stopped automatically through an electrical contact made by the spindle of one of the receding heads. While the approximate winding movements described appear quite simple, the exact path followed by any particular point on the spherical surface of the core is somewhat complicated, as will be apparent when the effect of the difference in speed of the shafts *A* and *O* is considered in conjunction with the rotational movements obtained by the four intermittently rotated rollers *J* and the constant rotational movement of the heads *B*.

Loading and Discharge Door Control for Enameling Oven.— An oven for baking the enamel on automobile wheels is so arranged that the wheels roll down gravity runways through the oven and are automatically discharged into runways which lead them either to the next operation or to the loading dock for shipment. The baking time for a

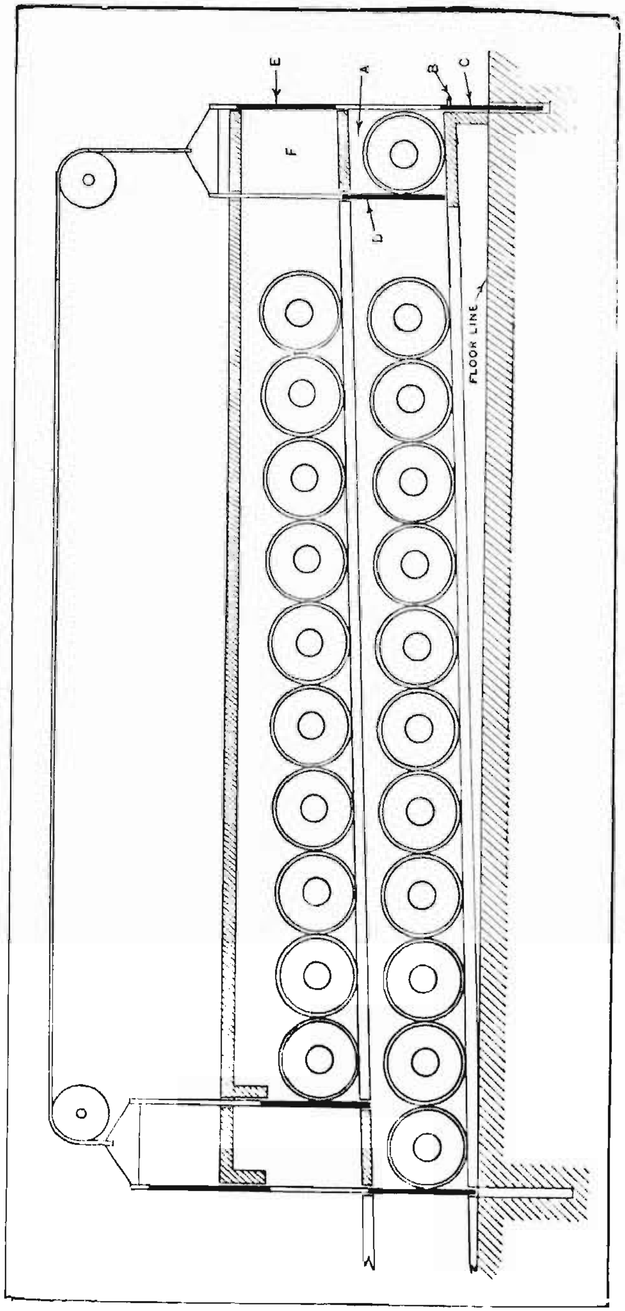


Fig. 13. Vertical Longitudinal Section of Enameling Oven which Automatically Discharges a Row of Baked Wheels when an Unbaked Row Enters the Oven at the Loading End

given length of oven depends entirely on the rate at which the wheels are put in.

In operation, the wheels are placed twelve abreast in compartment *A* (see vertical longitudinal section, Fig. 13). The operator, using handle *B*, then raises the door, which is made of three panels—*C*, *D*, and *E*—all mounted rigidly on a common frame and suspended from above. As the door is raised, panel *C* rises, closing compartment *A*, and panel *D* rises, letting the twelve wheels roll into the oven. Simultaneously panel *E* rises, opening compartment *F*, which has been closed at the back by panel *D*. Compartment *F* is now ready to be loaded with twelve wheels, after which the door is lowered again to reload compartment *A*.

The door at the discharge end is similarly constructed, and provides an automatic escapement for the wheels. The doors are connected by two cables which pass over sheaves above the oven and thus counterbalance each other. The two sheaves at each end are keyed to the shafts in order to keep the doors on an even keel and make them work freely in their guides. The two doors being connected, the discharge door is automatically lowered when the charging door is raised and vice versa, thus releasing a row of wheels every time a row is put in and keeping the oven full all the time. The man at the loading end sprays the wheels and places them in the runways, closing the door when he has inserted twelve wheels. Then, without further labor or attention on the part of the operator, they are baked the proper length of time and delivered to the next operation. With a given number of wheels per hour to be baked, any desired length of baking time may be obtained by making the oven of the proper length. Another point to be noted is that the doors are closed practically all the time, which cuts down the loss of heat to a minimum and also adds greatly to the comfort of the operator.

Short-Stroke Mechanism for Operating Fixture Lock-Pin.— A semi-automatic facing fixture used on a single-spindle drill press carries six castings that are equally spaced around the circular table of the fixture. As each casting is indexed around to the machining position, where it is faced by a profie cutter, the fixture table is locked in the proper position by a 3/8-inch tapered pin which enters a tapered hole. The mechanism to be described is for operating this stop-pin. Fig. 14 shows sectional and plan views of the stop-pin mechanism. The slide A, which is lo-

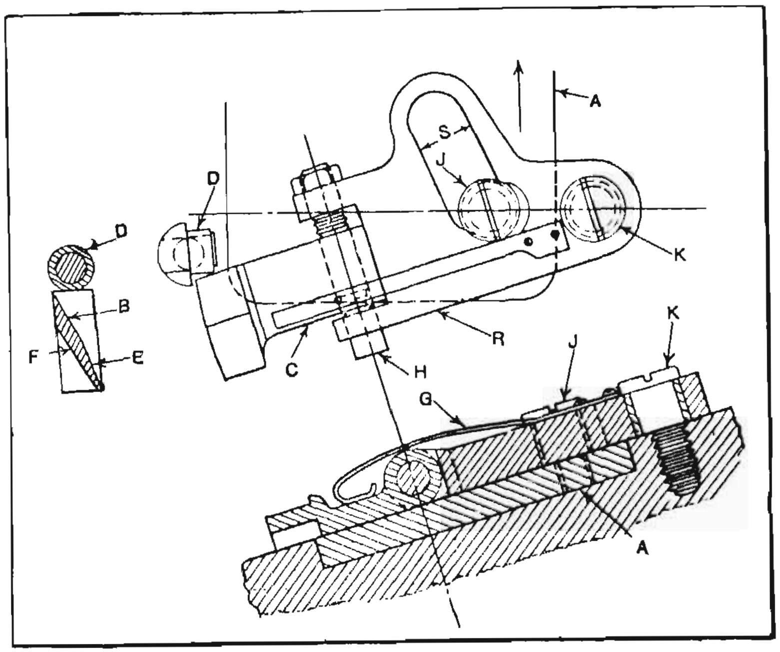


Fig. 14. Mechanism for Operating the Pin that Locks Table of Fixture During Machining Operation

cated on one side of the fixture, has a stroke of 1 5/8 inches. One and one-half inches of this stroke are utilized for indexing the fixture 60 degrees and the additional 1/8-inch movement is all that is necessary for withdrawing the stop-pin.

Attached to slide A is a pin J, which engages a slot S in lever R. The lever R is free to swing about the fixed pin

K, and carries an extension C, which is free to swing about pin H. The outer end of lever C has inclined surfaces B, E, and F, which come into contact with a roller D on the stop-pin when lever C-R is turned about pin K.

The plan view shows the mechanism in the position it occupies just before the stop-pin is withdrawn. When slide A moves in the direction of the arrow, pin J, acting against lever R, brings inclined surface B beneath roller D; 1/8-inch movement of slide A is sufficient to lift the stop-pin out of engagement. Before roller D has passed the end of surface E, the hole from which it was withdrawn has been moved from beneath the pin by the indexing movement of the fixture derived from slide A through a pawl (not shown), which engages one of six indexing pins. Soon after the end of surface E has passed roller D, the swinging movement of the hinged lever C-R discontinues, as slot S has moved around to a position parallel to the movement of slide A; consequently, slide A and pin J continue their movement without affecting the position of the slotted lever, and this additional movement of A is utilized to index the fixture 60 degrees.

During the return movement, slide A ejects the machined casting. When pin J engages the curved end of slot S, the lever C-R is forced back to the position shown, and the inclined surface F engages the opposite side of roller D, forcing it into the next stop-pin hole as lever C swings upward about the pin H against the action of spring G. As soon as the end of lever C has passed roller D, the lever is forced downward by spring G to the position shown in the sectional views. When in this position, lever C rests against slide A, as shown by the lower sectional view. The fixture is now in position for machining the next casting, after which the cycle of operations just described is repeated. The pin H, about which lever C swings, is made adjustable to compensate for wear or for variations caused by regrinding the chasers and the edges of the facing tool.

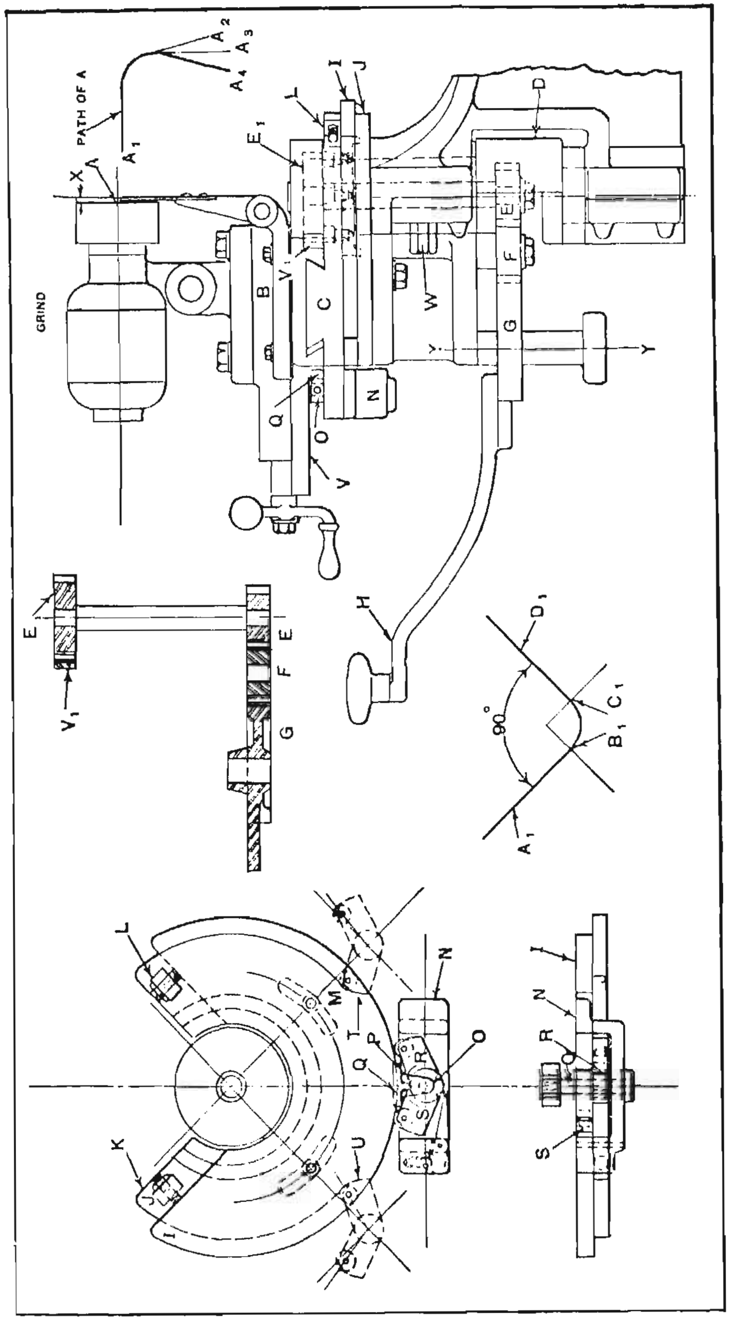


Fig. 15. Mechanism of Contour Grinding Machine Designed for Cutter Grinding

Mechanism of Contour Grinding Machine.—The mechanism shown in Fig. 15 controls the traversing movement of a grinding wheel in such a manner that hardened cutters can be quickly and accurately ground to contours such as are indicated by the three outlines (see upper right-hand corner). By simply moving the handle H from left to right, the grinding surface of the wheel at A will follow a path, such as is indicated by the line $A_1B_1C_1D_1$ (lower center diagram). In this case, the wheel grinds the straight surface from A_1 to B_1, and without pausing forms the radius from B_1 to C_1, continuing to the point D_1, the side C_1D_1 being ground at right angles to side A_1B_1 and the corner formed in one continuous pass.

By reversing the movement of handle H, the wheel is caused to travel back along the same path to the starting point A_1. Provision is made for setting the machine to grind the sides A_1B_1 and C_1D_1 to any included angle from 70 to 110 degrees instead of 90 degrees as shown, and with a radius of curvature at the corner of from 0 to 1 1/2 inches.

During the movement of the grinding wheel from A_1 to B_1 the handle H revolves about the center line Y-Y, transmitting the required longitudinal movement to carriage V which slides on C. The gear train that transmits motion from gear G to the rack V_1 secured to carriage V is shown in section. It will be noted that handle H is secured to gear G, which is actually a segment gear.

When the grinding surface of the wheel reaches point B_1, the revolving movement of handle H about axis Y–Y is automatically stopped and continued pressure on the handle causes the carriage V and slide C to revolve about the vertical axis of shaft D until the grinding surface of the wheel reaches the point C_1, when further movement about the axis of shaft D is stopped, and lever H again revolves about axis Y–Y, causing the wheel to travel from C_1 to D_1.

The grinding wheel is directly connected to the motor, which is mounted on the cross-slide B on carriage V. The distance X from the center line of shaft D to the grinding face of the wheel determines the radius of curvature of the corner. Carriage V is fitted to the slide C, which is secured to the top surface of shaft D. Shaft D thus supports the slide C, carriage V, cross-slide B, and the grinding wheel and its driving motor. These parts all swing together about the axis of shaft D during the corner-forming portion of the traversing movement.

Underneath slide C and fastened to the supporting frame or member are two plates I and J, also shown in the plan view (see left-hand illustration). These plates have stops K and L and cut-out portions on their peripheries which terminate in the cam surfaces at T and U. The spacing between T and L on plate I is definitely fixed, as is also the spacing or relationship of K and U on plate J. These plates are adjustable around the center of shaft D, and may be clamped in any desired position. The angle between T and U determines the angle of rotation of shaft D and slide C in forming the corner and, consequently, the included angle between the sides of the piece ground.

Midway of the length of slide C and in line with the center of the shaft D is a vertical shaft O, supported by bearings on slides C and N. On the upper end of shaft O is a two-tooth segment of a gear which meshes with a single rack tooth Q, attached to the mid point of the front face of carriage V. Also fixed to shaft O are two arms S and R, provided with rollers which make contact with the peripheries of plates J and I. The arms are offset vertically, so that S is in line with plate I, and R in line with plate J. The plan view shows the relative positions of the parts at the middle point of the corner-forming portion of the traverse movement. At this stage of the traverse movement of the grinding wheel, the shaft D is free to rotate in its bearings, and a movement of handle H to the right causes slide C and all parts attached to it to swing to the right about the vertical axis of shaft D. With the tooth Q in mesh with the two teeth of segment P, as shown, and the roller in arm S in contact with the periphery of plate I, movement of carriage V along slide C is positively prevented.

Rotation of slide C will continue until arm S drops into the cut away portion T, at which point slide C makes contact with stop L and further rotation about the axis of shaft D is positively prevented. When the roller of arm S drops into T and the parts are locked by stop L against further rotation about the axis of shaft D, the tooth Q is released from contact with the locking teeth of segment P by the rotation of shaft O. This leaves the carriage V free to slide on member C, so that resumption of the turning movement of handle H about axis Y–Y will complete the grinding movement along line C_1D_1.

When the movement of handle H is reversed, carriage V will slide along C until tooth Q enters the space between the two teeth of the segment P on shaft O. The continued motion of V and, consequently, the rotation of shaft O, causes the arms S and R to rotate until the roller in S is free of the opening T and the roller in arm R is in contact with the periphery of plate J.

When R makes contact with J, further motion of V on C is prevented, but rotation of C about D is permitted until R enters U and C makes contact with stop K, thus completing the corner-forming movement from C_1 to B_1. The carriage V can now slide on C and the revolving movement of handle H to the left about axis Y–Y completes the movement of the wheel from B_1 to A_1.

When the rotating movement required in forming the corner to a radius is taking place, all parts of the mechanism are locked against relative movement with each other, and the motion of handle H acting on gear E through gears G and F causes the complete assembly to revolve about the axis of shaft D. The combination of movements is caused by the continuous movement of H. There is no hesitation in the movement at the corners when rotation starts or

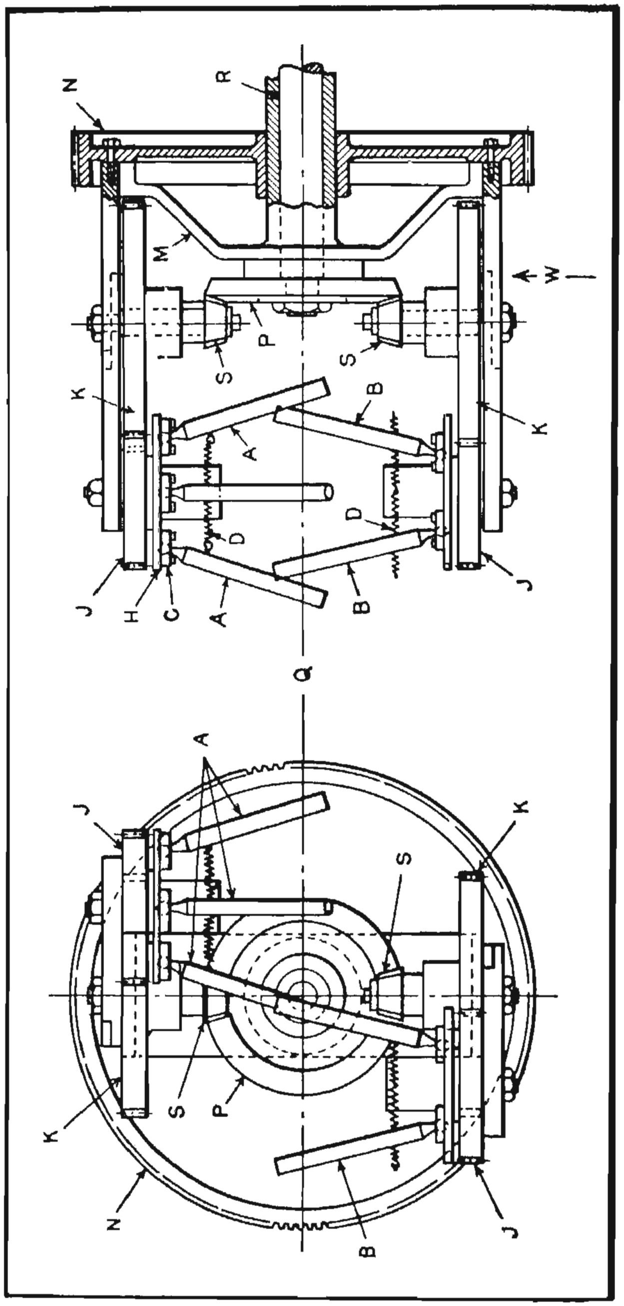


Fig. 16. Mechanism for Rotating Groups of Rods Arranged to Smooth Wrapper over Tapered Ends of Cigars

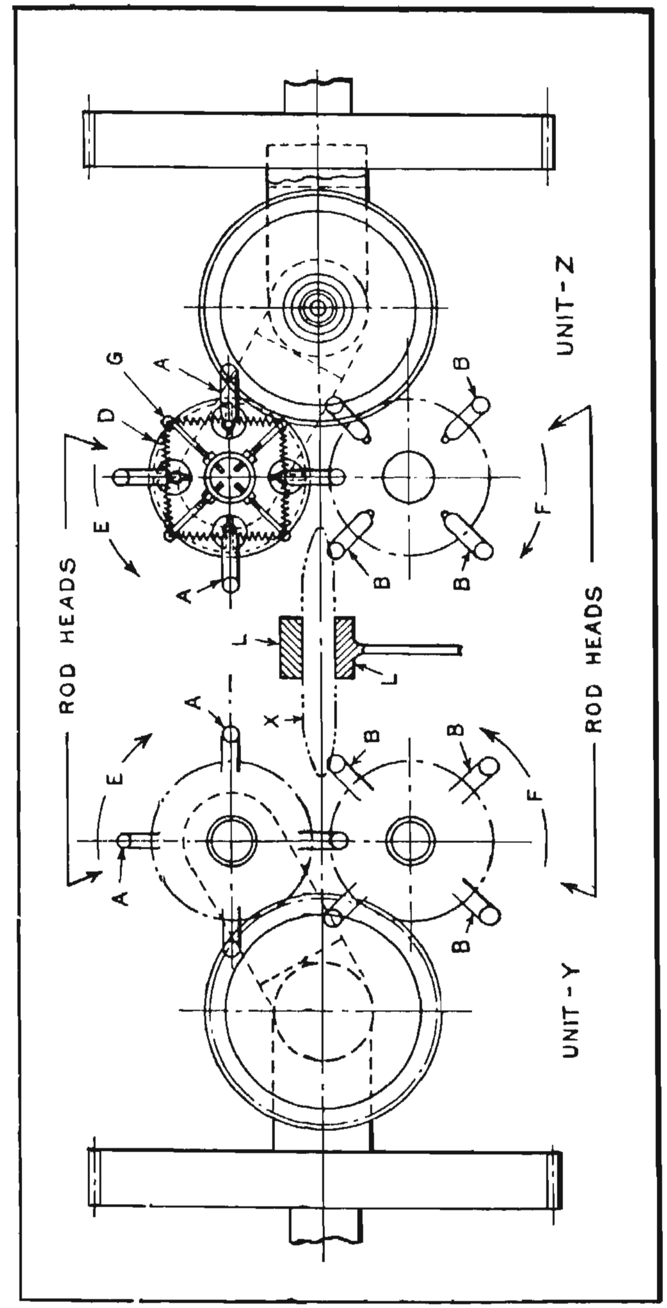


Fig. 17. Diagram Illustrating Operation of Wrapper-smoothing Mechanism Consisting of Two Units Like the One Shown in Fig. 16

stops, and no lost motion in the locking and unlocking action of the mechanism.

Mechanism for Smoothing Foil Wrapper Over Ends of Tapered Package.—Rods having a compound rotary motion are employed to smooth the creases of foil-wrapped cigars or other objects having tapering ends. Four round rods are arranged in a group at *A*, Fig. 16. Four rods are also arranged in a group at *B*. Each rod has a ball-shaped end mounted in a two-piece socket *C* in which the rod is free to pivot.

A group of springs *D*, of which eight are used in each unit, tie the rods flexibly together. In the center of Fig. 17 is indicated a tubular part *X* having an irregular shape with tapering ends, such as a foil-wrapped cigar. The cigar has been encased in its wrapper cylindrically and is to have the foil smoothed out and creased taperingly at the ends so that it folds smoothly over the irregularities and depressions of the rough-shaped ends.

To accomplish this, two entire mechanisms or units such as the one illustrated in Fig. 16 are employed, as shown in Fig. 17. The left-hand unit is shown at *Y*, and the right-hand unit at *Z*. The upper rods *A* are revolved in the direction indicated by the arrow *E* while the lower rods *B* are revolved in the direction indicated by the arrow *F*, the direction of rotation being determined by the bevel pinion drive. As each rod comes in contact with the wrapping on the work *X*, it smooths and creases it. The springs yield sufficiently to allow the rods to ride smoothly over the wrapping.

Each rod head, of which four are shown in Fig. 17, has eight springs *D* attached to four retaining screws *G*, two springs being attached to opposite sides of each of the rods *A* and *B*. The rod sockets are mounted on plates *H*, Fig. 16. Plates *H* are attached to gears *J*; thus the revolving of each large gear *K* causes the smoothing rods *A* and *B* to revolve continuously and stroke the work lengthwise while the foil-

wrapped part is held stationary in gripping fingers *L*.

In addition, the entire rod-actuating arrangement is mounted on a bracket *M*, which, in turn, is attached to a large gear *N*. Gear *N* is caused to revolve, thereby carrying the entire arrangement about the axis *Q*. Thus the rods not only work lengthwise along the wrapping but radially as well. These combined movements serve to draw the wrapping tight. The bevel gear *P* is revolved by shaft *R* which causes bevel pinions *S* to drive gears *K* that operate the bar-holding members. The rod-actuating units are comparatively small, and form one section of an entirely automatic machine.

Mechanism for Preventing Creep of Wire-Mesh Conveyor Belt.—Wire-mesh belts are used in a certain plant for feeding lacquered parts through drying ovens. Because of unequal stretching of the wire, however, the belt had a tendency to creep to one end of the driving roll and against the machine frame. This often resulted in damage to the belt. To overcome this difficulty, an idler roll was incorporated, as indicated at *G* in the diagram Fig. 18.

This roll is pivoted at *C*. End *D* is automatically swung either to the right or left, according to the direction of belt creep. For example, if the belt *E* creeps toward the rear of the driving roll *F*, idler *G* immediately swings toward the right, causing the belt to return to its normal path. If, on the other hand, the belt creeps toward the front, the idler will swing toward the left and return the belt as before to its normal path.

The automatic movement of this idler is produced by means of the ratchet mechanism shown in Fig. 19. This mechanism is indicated at *A* in Fig. 18, and operates a screw represented by the dot-and-dash line *B*. This screw engages a nut on the swinging bearing *D*. Base *A*, Fig. 19, is stationary, and on it is mounted ratchet wheel *B* and the double pawl *C*. The ratchet wheel is pinned to shaft *D*

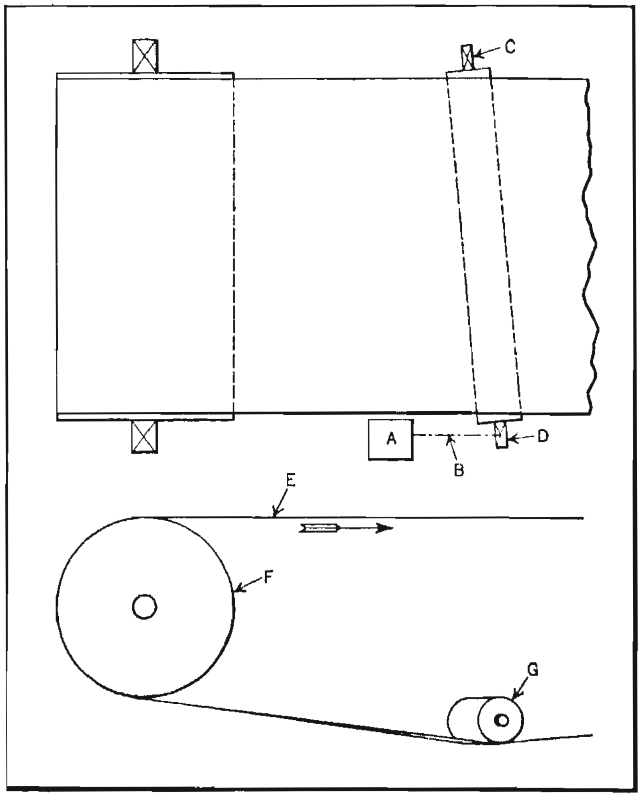


Fig. 18. Diagram Showing Swinging Idler Roll which is Actuated by Mechanism Shown in Fig. 19

and has closely spaced teeth cut on each side which are engaged by the pawl. Pawl *C* is pivoted on stud *F* in the vertical slide *G* and has an extension paddle *H* which is kept in contact with the edge of the belt by the counterweight *J*.

Slide *G* and pawl *C* are given a continuous vertical reciprocating movement by means of the pulley *K* through the crank *L* and the connecting-rod *M* which is pivoted at its upper end to the stud *F*. Pulley *K* receives its motion from another member of the conveyor (not shown). The screw engaging the nut on the swinging roll bearing is

shown at *N* and is rotated by the ratchet wheel through the miter gears *E*.

With the pawl in the position shown, the belt is running in its central position on the driving roll; hence neither end of the pawl engages the ratchet wheel. However, if the belt were to creep, say, toward the left, the edge of the belt would force paddle *H* also toward the left, causing the pawl to swing on stud *F*. As a result, the hooked end of the pawl

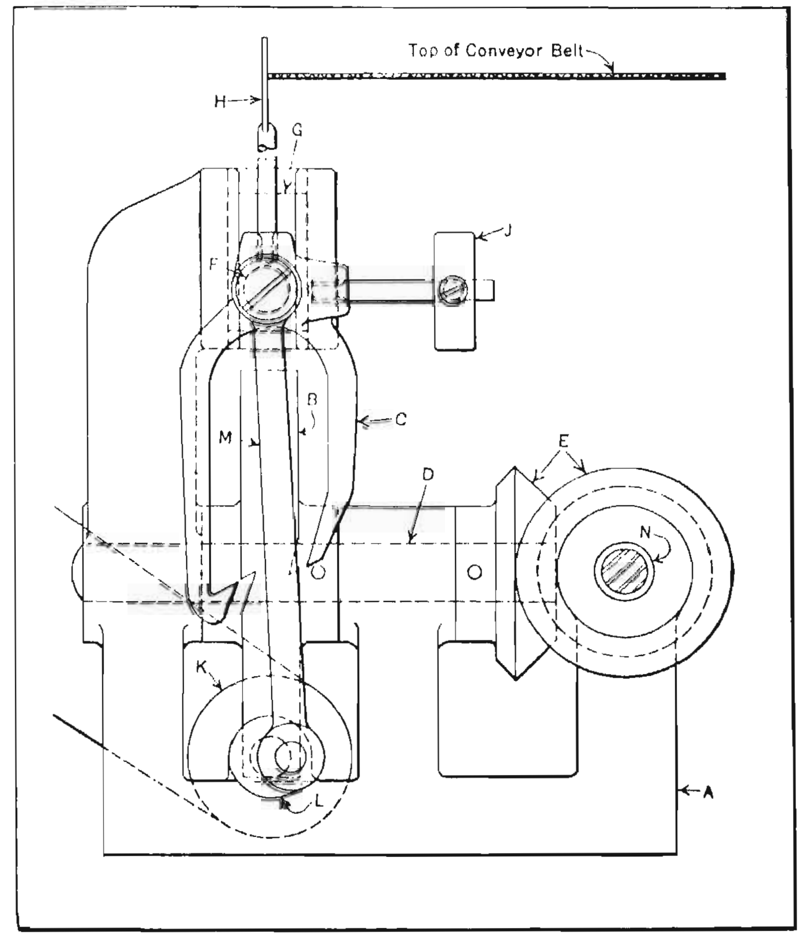


Fig. 19. Ratchet Mechanism for Automatically Swinging the Idler Roll into Position to Prevent Creep in a Wire-mesh Conveyor Belt

would engage the teeth in the ratchet wheel, rotating the ratchet wheel, miter gears, and screw *N*. As the screw rotates in the nut on the bearing roll, the roll is swung in the correct direction to guide the belt, through the angular position of the idler roll, back to the center of the driving roll. If the belt should creep toward the right, the movement of the paddle would cause the straight end of the pawl to engage the right-hand side of the ratchet wheel. As the teeth on this side of the wheel are cut opposite to those on the other side, the screw would rotate in the opposite direction and cause the idler roll, in turn, to swing in the opposite direction.

CHAPTER XVII

ENGINE VALVE DIAGRAMS AND THEIR APPLICATION IN STUDYING VALVE ACTION

The "valve-gear" or mechanism for operating the valve (or valves) which controls the admission and exhaust of steam in an engine cylinder is a comparatively simple mechanism especially if the engine has a single slide-valve; however, even the simplest form of valve-gear provides an interesting example of a mechanism requiring correct timing or relative action between certain main parts. This timing pertains to the action of the valve relative to the piston and steam ports, and it is secured first by proper design of the valve and its mechanism and finally by correct adjustment of the assembled parts. In connection with the design, so-called valve diagrams are used to determine in advance the action of a valve of given design. These valve diagrams and their application will be explained because of their relationship to the general subject of mechanism.

Diagrams which Show Valve Action.—When designing a slide-valve for a steam engine and the mechanism which operates the valve, it is desirable to be able to determine readily the position of the valve relative to the steam ports, for any given position of the crank or piston. Valve diagrams are commonly used for this purpose. These diagrams not only show graphically the relative positions of the valve and crank, but make it possible to design a valve with reference to a predetermined form of indicator card. Valve diagrams also indicate the effects of changes in the design of the valve on the steam distribution. In connection with steam engine work, certain problems or quantities relating to the point of cut-off, lead, etc., are assumed, and the remaining ones are required and may be determined by means of the valve diagrams. For instance, a designer might be given the point of cut-off, point of release, the lead, and the maximum port openings, the problem being to determine the valve travel, the outside and inside lap, and the angle of advance. By means of a suitable diagram, the valve travel, lap, etc., corresponding to these specified quantities may be readily determined. There are several different forms of valve diagrams, the *Zeuner* and the *Bilgram* diagrams being the ones most commonly used. The

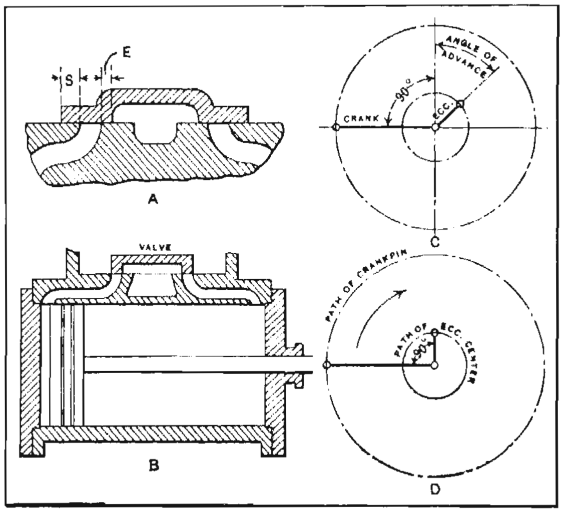


Fig. 1. Diagram Showing Slide Valves with and without Lap and Positions of Eccentric Relative to Crank

methods of laying out these diagrams and using them, in connection with steam engine work, will be described after considering some fundamental features of slide-valve design, so that the practical application of the diagrams may be more readily understood.

Position of Valve Relative to Ports.—A plain "D" slide-valve is represented at *A* in Fig. 1 on its seat and in mid-position. The *steam lap* or *outside lap S* is the amount by which the valve extends over the port on the admission side of the valve, when in mid-position on its seat. Similarly,

E is the *exhaust lap*, or *inside lap*, and is the amount by which the valve extends over the port on the exhaust side, when the valve is in mid-position on its seat. The necessity for having lap on a valve is shown by considering the lapless valve at *B*. Any movement of the valve to the right will admit steam behind the piston, and the other side of the piston will be open to the exhaust. The admission of steam on one side and the opening of the exhaust on the other will continue until the valve returns to its mid-position, which will occur when the piston is at the other end of its stroke. Such a valve arrangement as this would permit of no expansion of the steam in the cylinder; in other words, cut-off occurs at full stroke. This is an uneconomical type of valve.

For the lapless valve, the relative positions of the crank and eccentric are shown at *D*. It is necessary for the eccentric to be 90 degrees from the crank, in order that the valve shall be in mid-position when the piston is at the end of its stroke. Now, if there were any steam lap, it would be necessary to move the valve on its seat a distance to the right at least equal to the steam lap, in order that the port be open as soon as the piston starts on its stroke. By increasing the angle between the crank and the eccentric (for a direct motion), the valve can be moved along on its seat this amount. This angle is known as the *angle of advance*, and is indicated at *C*.

The Zeuner Valve Diagram.— The Zeuner valve diagram is illustrated in Fig. 2. As will be seen, there are two circles that are concentric and two smaller circles within the inner concentric circle. The larger circle need not be drawn to any particular scale, as it merely serves to represent the path of the crankpin. The smaller circle is either drawn to scale or full size, and has a diameter corresponding to the valve travel or twice the eccentricity. This is known as the *valve circle*. The two circles within the valve circle have a diameter equal to the radius of the valve circle, and are known as the *Zeuner circles.* The angle between the vertical center-line and the center-line upon which the two Zeuner circles are drawn is equal to the *angle of advance.* The cylinder is supposed to be on the left-hand side of the diagram, and the crank is turning in the direction indicated by the arrow.

For a head-end diagram or one for the head end of the valve and cylinder, the upper Zeuner circle is used for the admission or outward stroke, and the lower circle for the

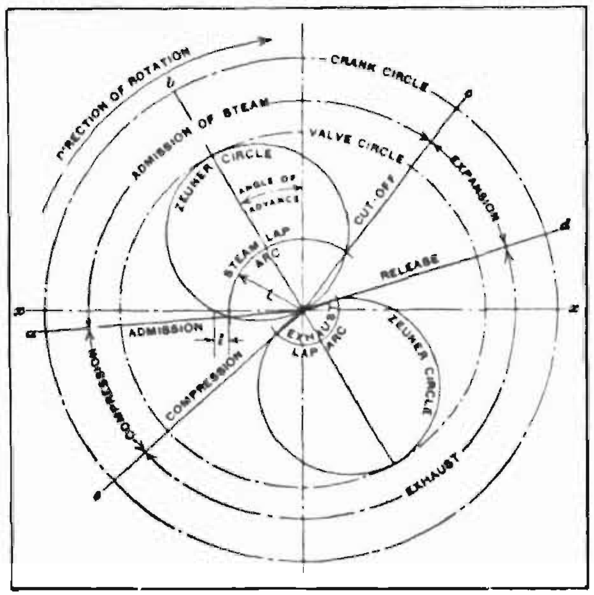


Fig. 2. Zeuner Valve Diagram

exhaust or return stroke. Fig. 2 is an example of a head-end diagram. With a center at the center of the valve circle, an arc with a radius equal to the steam lap is drawn, cutting the upper Zeuner circle. An arc with a radius equal to the exhaust lap is drawn on the lower circle in the same manner. These are called the *steam-lap* and the *exhaust-lap arcs.* The relation between the center-line of the crank for any given position, and these lap arcs and Zeuner

circles, indicates the position of the valve. When the crank center-line crosses the steam-lap arc and the Zeuner circle, the port opening equals the distance between the steam-lap arc and the Zeuner circle measured along the crank center-line. For instance, when the crank center-line is at a, the valve is off center a distance L equal to the steam lap, and the port is about to be opened. When the crank has moved

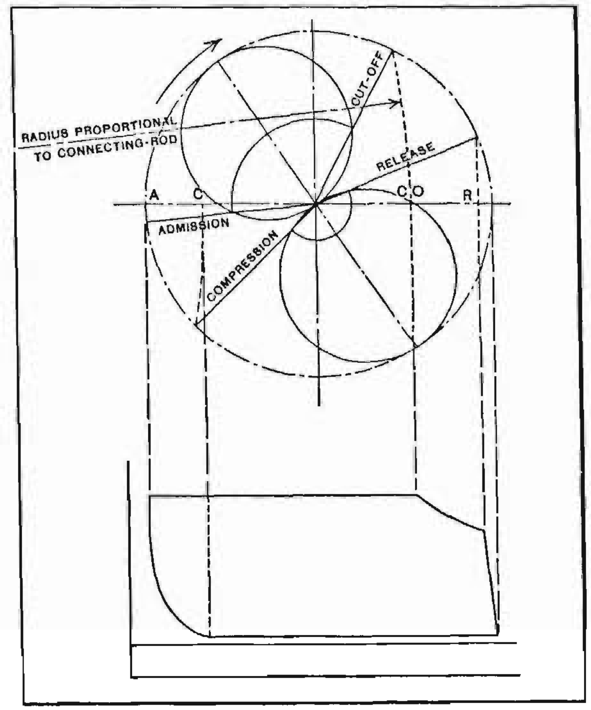


Fig. 3. Relation Between Zeuner Diagram and Indicator Card. Piston Positions are Marked A, for Admission; CO, for Cut-off; R, for Release; and C, for Compression

up to the horizontal center-line xx, representing the dead-center position, the port is opened an amount l equal to the lead, since l is the distance between the steam-lap arc and the Zeuner circle measured along the crank center-line. When the crank is at b, the maximum port opening occurs; at c, the steam is cut off; and at d, the exhaust port opens as shown by the intersection of the crank center-line with the lower Zeuner circle at the exhaust-lap arc. The exhaust port closes at *e* and compression begins. This cycle is then repeated.

From the foregoing, it will be seen that, as the crank center-line is rotated, it will intersect the valve circle, the two Zeuner circles, and the lap arcs. As the distance from the center of the valve circle to any point of the Zeuner circle, when measured along the crank center-line, shows the amount by which the valve has moved from its mid-position, the distance between the lap arc and the Zeuner circle shows the amount by which the port is uncovered.

Relation of Diagram to Indicator Card.—Since the valve diagram shows the relative positions of the crank and the valve for the various events of the steam engine cycle, it is possible to draw the valve diagram from the indicator card, or vice versa. Fig. 3 shows the relation between the two diagrams. The positions of the crank for admission, cut-off, release, and compression are indicated by the radial lines. To find the relative positions of the piston for these events, an arc is swung from the intersection of the crank center-line with the valve circle, to the horizontal center-line. The radius of this arc should be in the same proportion to the radius of the valve circle that the length of the connecting-rod of the engine is to the crank; that is, if the connecting-rod is five times as long as the crank, then the radius of the arc should be five times the radius of the valve circle. The center of the arc should be on the horizontal center-line extended to the left. The point which has been determined by this arc shows the relative distance which the piston has moved from the end of its stroke, for that particular position of the crank. These points can be projected down upon a diagram below to give the four principal points of the indicator card, the vertical distances on this card being determined by the relative steam pressures which exist at admission and at exhaust. The compression curve

and the expansion line can be drawn as equilateral hyperbolas. Starting with the indicator diagram, the four piston positions on the valve circle diameter, where admission, cut-off, release, and compression occur, are found by projecting upward from these points on an indicator card to the center-line of the valve circle. Then, by swinging arcs from these points to the valve circle, the crank positions are determined, and the valve diagram is laid out accordingly.

Application of the Zeuner Diagram.—A construction which is often necessary in working the Zeuner diagram is shown at *A* in Fig. 4. A small circle is drawn around the intersection of the valve circle and the horizontal center-line. The radius of the circle is equal to the lead. A tangent *ab* to this circle, perpendicular to the center-line *oc* of the Zeuner circle, will cut the valve circle at the crank positions of admission and cut-off *oa* and *ob*. If the point of cut-off and the lead are known, as is generally the case when a steam engine is designed, this lead circle can be drawn, the cut-off point determined, and a line *ab* from the cut-off point be drawn tangent to the lead circle. A perpendicular to this line *oc*, through the center of the valve circle, gives the center-line of the Zeuner circles and determines the angle of advance. The lap arc is at once determined, as it will always cut the Zeuner circle at the intersection of the crank positions *d* and *e*.

The diagram for the head end of the cylinder has been referred to in the foregoing. The crank-end diagram is similarly constructed and used. As shown at *B*, Fig. 4, the Zeuner circle for determining the admission of steam and the cut-off is on the *lower* side of the horizontal center-line, and the circle for determining the release and compression is on the *upper* side; in other words, the positions of the two circles are reversed from the head-end positions. Lap arcs are drawn as before, but the lead circle is at the right end rather than the left. The positions of the piston, cor-

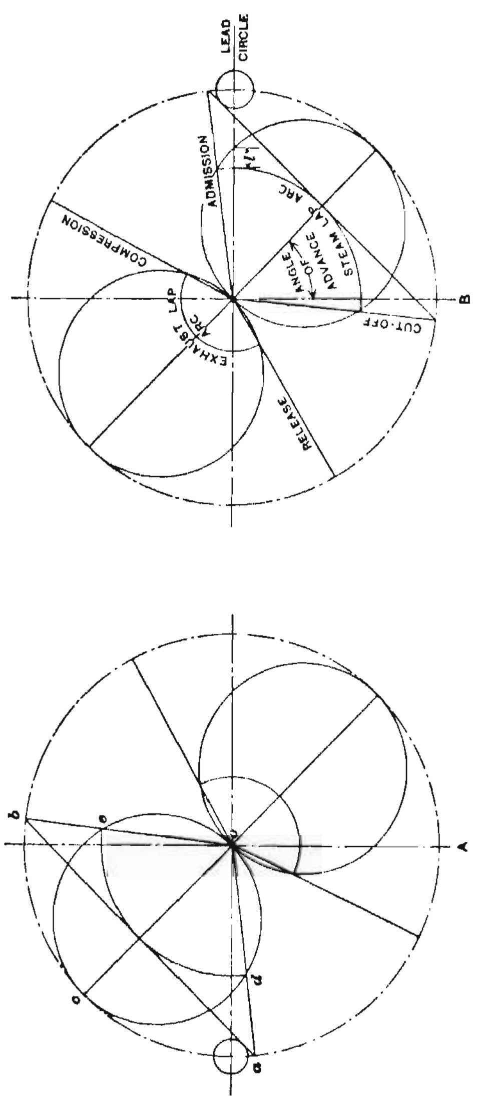


Fig. 4. (A) Construction of Zeuner Diagram when Lead is Known. (B) Diagram for Crank End of Cylinder

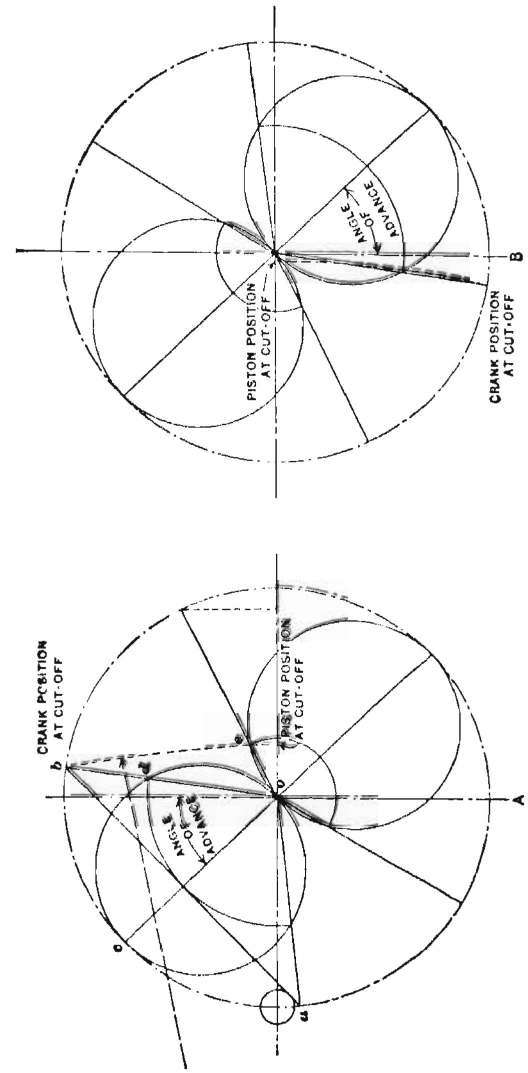


Fig. 5. Zeuner Diagrams Laid out from Given Data

responding to the crank positions at the four important points, are found by means of the arc proportional to the connecting-rod length, as before.

Construction of the Zeuner Diagram.—In order to illustrate the practical application of the Zeuner diagram, assume that a valve is to have a travel of 3 inches, a lead of 1/8 inch, and that the point of cut-off, under normal conditions, is to be at five-eighths stroke on the head end of the cylinder. The valve is to have an equal amount of outside lap on the two ends; hence, the cut-off will vary for the head and crank ends, owing to the angularity of the connecting-rod. As the result of this angularity, the amount of piston travel for the head- and crank-end strokes differs considerably for corresponding crank positions; consequently, a valve which has the same outside lap on both ends will give a shorter cut-off on the crank end than on the head end, and, in order to equalize the cut-off, it will be necessary to have a smaller lap on the crank end of the valve. If the amount of lap were varied, however, the lead would be unequal, and, as this is considered objectionable, it is common practice to make the lap equal and allow the variation in cut-off. Assume that the connecting-rod in this case is five times as long as the crank.

When constructing the Zeuner diagram, first draw a valve circle having a diameter equal to the valve travel, or 3 inches. Determine the piston position for five-eighths stroke, which represents the point of cut-off. (See Diagram *A*, Fig. 5.) Now, with a radius of 7 1/2 inches (or five times the radius of the valve circle) and with a center on the horizontal center-line of the valve circle extended, draw an arc upward to the valve circle, marking the position of the crank at the point of cut-off. Then, with a radius of 1/8 inch, draw the lead circle at the left of the diagram, as shown in the illustration. Draw a tangent *ab* to this circle which will terminate in the cut-off position. A perpendicular *oc* to this tangent and passing through the center of the valve circle will be the center-line of the two Zeuner circles. The angle between this perpendicular and the vertical is the angle of advance, and represents the angle between the eccentric and the crank. The steam-lap arc is drawn through the intersection *d* of the Zeuner circle with the crank position at cut-off, and will be found tangent to the line which was drawn tangent to the lead circle. The exhaust-lap arc is also drawn through the intersection *e* of the lower Zeuner circle with the crank position at release. The two laps may now be measured and the port openings for any position of the crank be determined. The maximum steam-port opening is measured along the center-line of the upper Zeuner circle between the valve circle and the lap arc.

To determine what will be the point of cut-off for the crank end of the cylinder, the process is as follows: Draw the valve circle as before, with a 3-inch diameter. Draw the lap arcs, reversing the positions as shown at *B*, Fig. 5, the steam-lap arc being in the lower right-hand quadrant and the exhaust-lap arc in the upper left-hand quadrant. The two Zeuner circles are next drawn, the center-line having the same angle of advance as in the diagram *A*. Now find the crank positions for the various events of the cycle by means of radial lines representing the different positions of the crank. Then, with a radius of 7 1/2 inches and with a center on the horizontal center-line extended to the left, as before, find the corresponding positions of the piston by swinging the arcs upward to the horizontal center-line.

Effect of Changing Eccentricity.—By means of the Zeuner diagram, it is possible to see clearly the effect of changing the eccentricity and the angle of advance in a slide-valve engine. If the eccentricity is increased without making other changes, the lead will be increased, cut-off occurs later, release sooner, and compression later. If the eccentricity is decreased, the lead is decreased, cut-off occurs earlier, the release later, and the compression earlier. Increasing the angle of advance (as the diagram, Fig. 2, shows) will cause the cut-off and all the other events to occur earlier, whereas decreasing it will have the opposite effect. These two methods of changing the operation of the valve are made use of in all types of shaft governors. A combination of the two, where the eccentricity is changed at the same time as the angle of advance, makes it possible to change the cut-off without changing the lead.

In the foregoing the piston has been assumed to be on the left of the diagram; if it were assumed to be on the right side, the Zeuner circles should be on the opposite sides of the vertical center-line and the other parts of the diagram similarly changed.

Summary of Principles of Zeuner Diagram.— When using the Zeuner diagram, it is well to have the following principles well in mind:

1. Any radial line on the valve circle represents a position of the crank.

2. The intercept on this radial line between the center and the Zeuner circle represents the movement of the valve from its mid-position.

3. The intercept on this radial line between the lap arc and the Zeuner circle represents the amount the port is open for that position of the crank.

4. The relative piston position can be determined, if the proportion of connecting-rod to crank is known.

5. The radial line representing the crank position for cut-off must pass through the intersection of the lap arc and the Zeuner circle. This also applies to release, compression, and admission.

6. A perpendicular to the crank position at the intersection with the lap arc and Zeuner circle will intersect the valve circle at a point coinciding with the center-line for the Zeuner circles; moreover, a perpendicular to any crank position at the intersection of the Zeuner circle will intersect the valve circle at the intersection of the center-line for the Zeuner circles.

Bilgram Diagram.— The methods of laying out a Bilgram diagram for both the head and crank ends of a cylinder are shown at *A* and *B*, Fig. 6. The diameter ss_1 of the outer circle represents the stroke of the engine; this circle may be drawn to any convenient scale. A smaller circle is drawn from the center *a*, having a diameter equal to the travel of the valve. This valve-travel circle may also be drawn either to a reduced scale or to full size, if more convenient. Assuming that the inside and the outside lap and the lead are known, proceed as follows: Draw a line *ll* parallel to *ss*, and a distance above it equal to the required lead or amount of port opening at the beginning of the piston stroke. Next draw another circle from center *b* having a radius equal to the outside lap of the valve. As the diagram shows, this circle should have its center *b* on the valve-travel circle and should be tangent to line *ll*. About center *b*, draw a smaller circle having a radius equal to the inside lap of the valve. Now, to obtain the location of the center-line of the crank at the point of cut-off, draw a radial line *ac* tangent to the outside of the lap circle. A vertical line *cd* intersecting the stroke line *ss* shows approximately how far the piston travels before the steam is cut off. This vertical line will not locate the exact point of cut-off, owing to the angularity of the connecting-rod and the resulting variation between the movements of the crankpin and the piston. To obtain the exact position of the piston at the point of cut-off, line *cd* should be an arc having a radius proportional to the length of the connecting-rod and drawn from a center located at some point on an extension of center-line *ss*.

A line *ae* through center *b* indicates the angular position of the eccentric corresponding to a given amount of outside lap and lead, whereas, angle eas_1 equals the angle of advance. Line *af* tangent to the inside-lap circle (on the lower side for diagram *A*) locates the crank position when

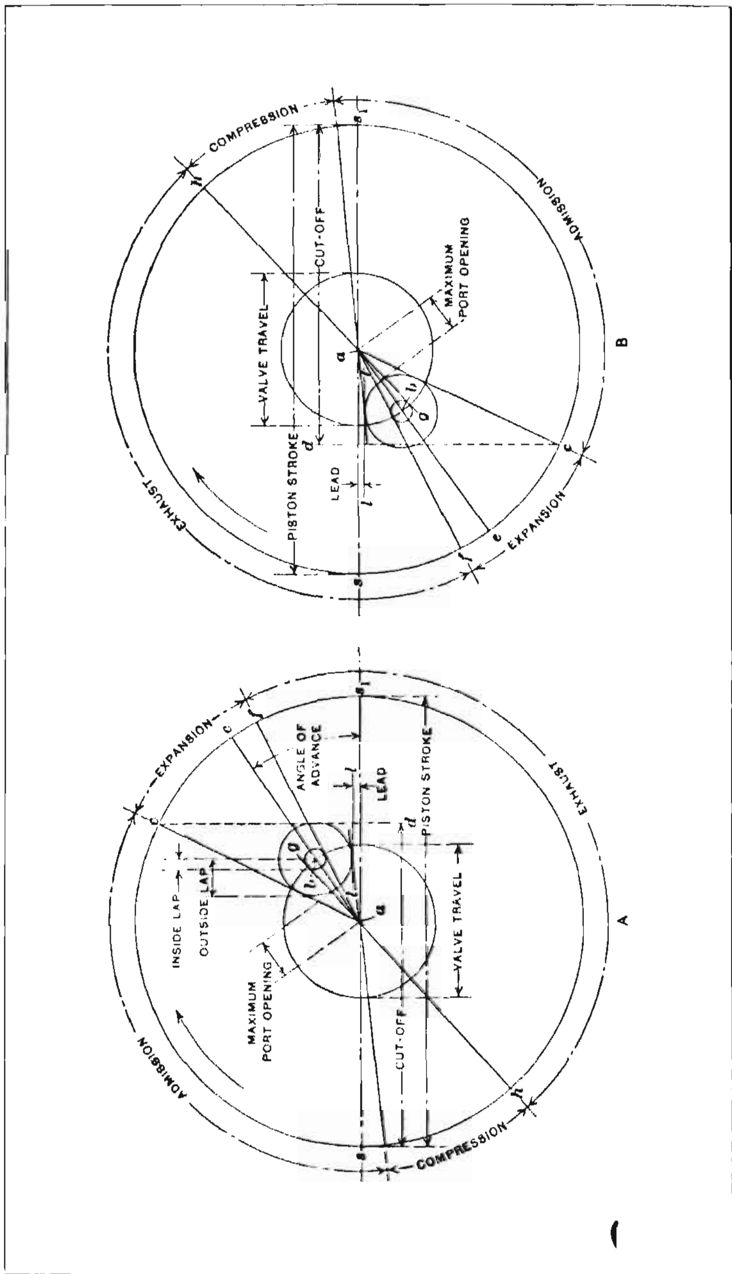


Fig. 6. Bilgram Valve Diagrams for Head and Crank Ends of Cylinder

the steam is released by the opening of the exhaust port, whereas line gh, tangent to the opposite side of the inside-lap circle and passing through center a, shows the position of the crank for the point of compression or the closing of the exhaust port. The maximum steam-port opening is represented by the distance from center a to the outside-lap circle, the measurement being taken along the center-line ae.

By studying these diagrams A and B, the effect of changes in the design of the valve may readily be determined. For instance, if the outside lap is increased (thus increasing the diameter of the lap circle), the point of cut-off will occur earlier in the stroke, giving a greater range of expansion; this change, however, will also cause an earlier compression, even if the inside lap is not increased, because the effect of enlarging the outside-lap circle is to change the position of the inside-lap circle and the angular position of line gh, so that the exhaust port is closed earlier. Increasing the inside lap also increases compression and delays the point of release or exhaust. By means of this diagram, if the point of cut-off, lead, maximum port opening, and point of compression were given, the necessary inside and outside lap and valve travel could be determined.

INDEX

INGENIOUS MECHANISMS

FOR DESIGNERS AND INVENTORS

VOLUME III

Mechanisms and Mechanical Movements Selected from Automatic Machines and Various Other Forms of Mechanical Apparatus as Outstanding Examples of Ingenious Design Embodying Ideas or Principles Applicable in Designing Machines or Devices Requiring Automatic Features or Mechanical Control

Edited by
HOLBROOK L. HORTON

INDUSTRIAL PRESS INC. NEW YORK, N. Y.

Industrial Press Inc.
989 Avenue of the Americas, New York, NY 10018
Tel: 212-889-6330 Toll-Free: 1-888-528-7852 Fax: 212-545-8327
www.industrialpress.com Email: info@industrialpress.com

INGENIOUS MECHANISMS
FOR DESIGNERS AND INVENTORS—VOLUME III

26 27 28 29 30

THIRD VOLUME OF INGENIOUS MECHANISMS

IN this third volume of INGENIOUS MECHANISMS FOR DESIGNERS AND INVENTORS a large number of mechanisms and mechanical movements not previously described in Volumes I and II have been brought together for convenient study and reference. The steady demand for Volumes I and II indicates the continuing need on the part of machine designers, engineers and students for detailed information about unusual, yet practical mechanical movements.

As in the previous volumes, the mechanisms described are the work of numerous contributors to MACHINERY and represent successful applications of a wide variety of types and designs. Of particular interest to many will be the chapter on Hoppers and Hopper Selector Mechanisms for Automatic Machines which appeared as two articles in MACHINERY by J. R. Paquin.

While it is not feasible in any work of this kind to include mechanisms that are directly applicable to every type of machine and operating condition, it is believed that the numerous designs found in Volumes I, II and III embody mechanical principles which may be utilized in the solution of practically any mechanism designing problem likely to be encountered. Although this volume is an independent treatise, the same general classification and chapter headings used in Volumes I and II have been retained, as far as possible, in Volume III to facilitate the use of all three volumes as a correlated reference library on the subject of mechanism.

CONTENTS

CHAPTER 1

Cam Applications and Special Cam Designs

In the design of mechanisms to obtain irregular movements of various kinds, cams are frequently employed. Those which are described or illustrated in connection with the mechanisms covered by this chapter are notable for some ingenious arrangement or design. Other applications of cams and cam-operated mechanisms will be found in Chapter 1, Volume I, and Chapter 1, Volume II, of "Ingenious Mechanisms for Designers and Inventors."

Cam Designed to Provide Longer Stroke without Enlarging Operating Space.—The cam shown at *B* in Figs. 1, 2 and 3 serves to impart a reciprocating motion to the machine slide *G*. The slide is required to have a longer stroke than could be produced by a cam of conventional design and of a size which could be assembled in the recess *W*. This unusual and interesting design of the cam was therefore necessary to provide the long stroke required without increasing the diameter of the cam.

Plan views of the mechanism are shown in Figs. 1 and 3, Fig. 2 being an end view. The complete assembly is shown by Figs. 2 and 3, whereas Fig. 1 shows the mechanism with the slide *G* removed. This view, however, shows the cam follower rollers *C*, *D*, *E*, and *F* in the respective positions they occupy when assembled on the under side of slide *G*.

Referring to Fig. 1, the shaft *A*, to which cam *B* is keyed, rotates in the direction indicated by the arrow. Cam *B* is provided with a series of projecting surfaces or cam tracks indicated by reference letters *H* to *N*, inclusive. The four roller followers *C*, *D*, *E*, and *F*, assembled on the under side of slide *G*, are equally spaced on the center line.

Rollers *C* and *D*, Fig. 1, act as cam followers on opposite sides of the track *H*. Rotation of cam *B* in the direction indicated causes these rollers to move toward the periphery of cam *B* at a rate of speed governed by the shape of track *H*. As roller *C* reaches the periphery of cam *B*, roller *D* reaches the position previously occupied by roller *C*, Fig. 1,

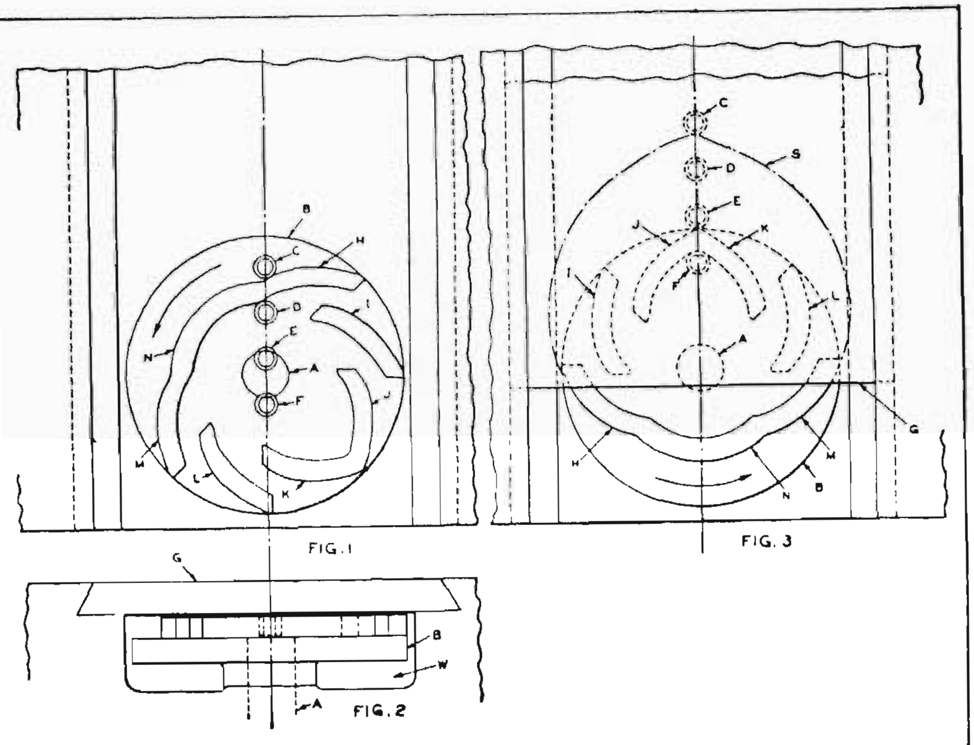


Fig. 1. View of Cam for Operating Slide Shown with Slide Removed but with Slide Follower Rolls in Positions They Occupy with Slide at Lowest Point of Travel. Fig. 2. End View of Cam B and Slide G. Fig. 3. Plan View of Assembled Cam B and Slide G with Latter Member at Highest Point of Travel.

since the center distance between rollers *C* and *D* is equal to the rise of cam track *H*.

When roller *C* reaches the periphery of cam *B* it ceases to function as a cam follower, but the motion of the slide *G* is continued by rollers *D* and *E*, acting as followers on track *I*, which at this point is in the position previously occupied by track *H*, as shown in Fig. 1. Track *I* then functions as a cam between rollers *D* and *E* until track *J* reaches the same position, when rollers *E* and *F* act as followers to continue the movement of slide *G*.

In Fig. 3, the rollers *E* and *F* are shown at the highest point to which they are carried by the track *J*, slide *G* having reached the extreme end of its upward travel. As cam *B* continues rotation, tracks *K*, *L*, and *M* pass in succession between rollers *F*, *E*, *D*, and *C*, returning the slide *G* to its original position. As track *N* is concentric with the center of rotation of cam *B*, the rollers *E* and *F* and slide *G* remain stationary until track *H* is again positioned, as shown in Fig. 1, to complete the cycle.

In the design illustrated, the upward stroke or travel of slide *G* at a uniform speed is produced by cam *B* during 135 degrees of its rotation. The reverse or downward stroke of slide *G* at uniform speed is also produced by cam *B* while rotating through an angle of 135 degrees. Following this complete reciprocating movement, the slide *G* remains stationary at its lowest point of travel while the cam *B* rotates 90 degrees to finish one complete revolution. With this type of cam, an irregular motion of the slide can be produced by varying the shape of the cam tracks, provided that each track is so designed that its leading end will continue the motion produced by the cam which preceded it. In order to compare the size of cam *B* with a cam of conventional design which would be necessary to produce the same length of stroke or movement of slide *G*, an outline of the track for such a cam is indicated by the dot and dash lines at *S*, Fig. 3.

Cam Mechanism with Variable Quick-Drop Adjustment.—The cam mechanism illustrated in Fig. 4 was designed to raise follower-roll *A* at the end of lever *B* at a uniform rate until the highest point of its travel is reached, and then to permit it to drop quickly a predetermined adjustable distance before resuming its downward

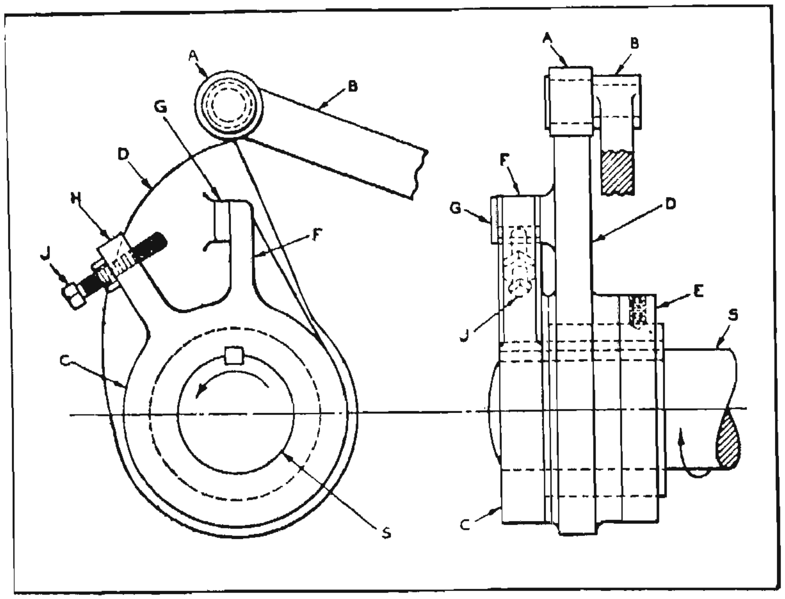


Fig. 4. Cam Mechanism Designed to Impart Upward Movement to Follower Roll A at Uniform Rate, Then Permit Quick Drop Followed by a Slow Descent to Lowest Point.

movement at a slower rate. This adjustable quick-drop cam mechanism is used on a wire-fabricating machine to transmit the particular motion required on one of the machine parts through lever *B*.

Driving shaft *S* revolves in a horizontal position in the direction indicated by the arrow, and carries with it flange *C*, to which it is keyed. Cam *D* is a free running fit on the hub of flange *C*, on which it is retained by collar *E*. With the members of the mechanism in the position shown, shaft *S* transmits motion to cam *D* through contact of arm *F* with

the projection *G* on cam *D*. The arm *F* forms an integral part of flange *C*.

The profile of cam *D* is designed to transmit a slow uniform upward vertical movement to roll *A*, followed by a rapid drop. Arm *H*, which is a part of flange *C*, carries stop-screw *J*. Roll *A* is kept in contact with cam *D* by a spring (not shown), which is attached to lever *B*.

As shown in the illustration, roll *A* is nearly at the top of its vertical movement. As soon as the peak of cam *D* passes under the center of roll *A*, the downward pressure of the spring attached to lever *B* reacts on the angular face of cam *D*, causing its rotation to be rapidly accelerated in the direction in which it is turning until projection *G* comes in contact with stop-screw *J*. Since this movement takes place rapidly, as controlled by the tension of the spring, there is a rapid drop of roll *A*, which is limited by the contact of projection *G* with adjusting stop-screw *J*.

Continued rotation of shaft *S* permits cam *D* to rotate at a slow rate of speed, the drop of roll *A* at this point being at the same rate as though cam *D* were keyed directly to shaft *S*. As the heavier side of cam *D* reaches a position opposite that shown in Fig. 4, it remains at rest until arm *F* again comes in contact with projection *G*.

Multiple Cam and Lever Mechanism.—The multiple cam and lever mechanism shown in Fig. 5 was designed to impart a movement to rods *D* and *E*, in the direction indicated by the arrows, through rotating shaft *B*. Interfering members and the considerable distance between shaft *B* and the rods made a simple drum cam and lever mechanism impractical for this particular application.

Rods *D* and *E* are prevented from rotating by slots in their ends, which slide along keys in the frame of the machine. Collars *G* and *H*, which are pinned to the rods, are machined on one side to form face cams. These cam surfaces slide on the cam surfaces of the two lever type collars *J* and *K*. The collars are joined by connecting-rod *L*. The roller *P* which is mounted on one end of this connecting-rod, is free to rotate on the pin Q and rolls along the pad at the upper end of lever R.

With cam A, roller C, lever Z, adjustable connecting-rod F, and lever R in the positions shown, the face cams have advanced rods D and E to their extreme forward positions. As cam A continues to rotate with shaft B in the direction

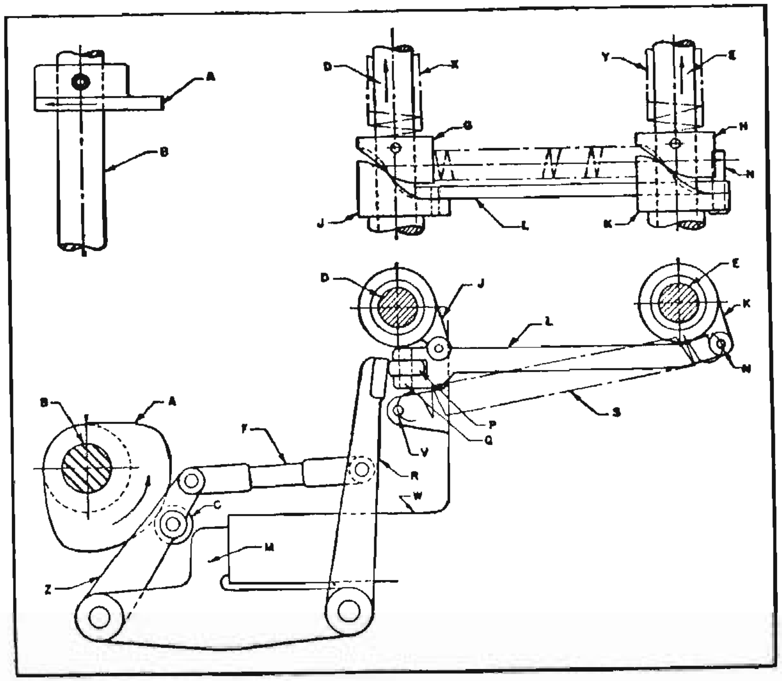


Fig. 5. Cam A through Connecting Linkage Imparts Motion to Rods D and E at Right Angles to Its Plane of Rotation.

indicated by the arrow, and roller C leaves the lobe of the cam, spring S returns the parts of the mechanism to their original positions. This spring is hooked over pin V in the frame of the machine and stud N in collar K. Springs X and Y, which are compressed during the advance stroke of rods D and E, are then free to return the rods and collars G and H to their starting positions.

Compound Cam Mechanism.—A mechanism employed on a French net-making machine that embodies an interesting application of a compound cam movement is shown in Fig. 6. The object of the mechanism is to give to the needle point n, Fig. 6, a closed path of the form shown in Fig. 7, and, additionally, an endwise movement normal to the plane of the paper. To effect this, there are two large gears, A_1 and A_2, meshing together, each of which carries a cam. In

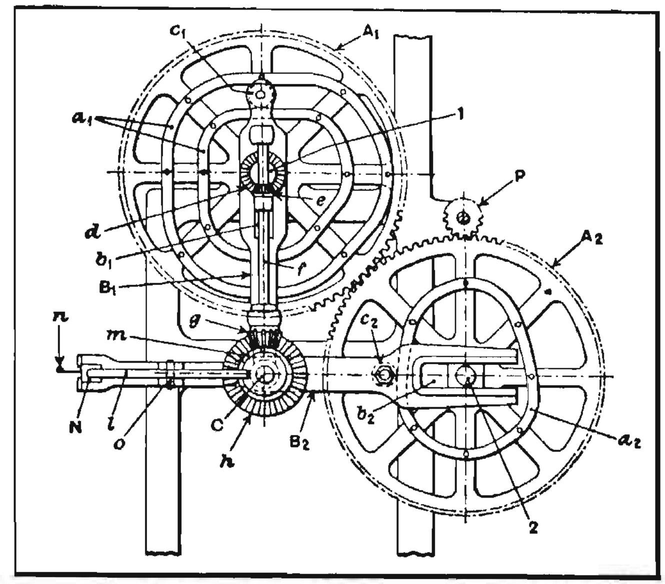


Fig. 6. Compound Cam Mechanism Employed on a Net-making Machine.

the case of gear A_1, the cam is formed as a closed track by attaching to the wheel the inner and outer members a_1, and in the case of gear A_2 by providing a rim a_2. A lever B_1 is slotted at b_1 so as to embrace the shaft 1, and is provided with a slipper block (not shown) for this purpose. At its upper end this lever carries a roller c_1, fitting the cam track previously described. A second lever B_2 is slotted or forked at b_2 to embrace the shaft 2, being provided with a slipper block as shown. A roller c_2 is employed to engage the cam

a_2, being kept in contact with it by means of a strong spring, not shown.

At C levers B_1 and B_2 are connected by a pin, so as to have a free turning movement. As shown, lever B_2 is continued to the left of C and carries a needle bar N.

So far, then, it will be seen that if gear A_1 were held stationary and A_2 were turned (not practically possible, of course), then the cam a_2 would give to the needle a substantially horizontal to and fro motion. Similarly, if we imagine gear A_2 held and A_1 turned, then the needle would receive a substantially vertical reciprocatory motion. Therefore, when both wheels are rotating, being driven by pinion P, a combination of these movements is effected, giving rise to an enclosed locus, as shown at the upper right in Fig. 7.

It is instructive to consider the method of finding the cam profiles for a given locus. When the needle point n is at position 1 of the locus, the levers are at right angles, as shown by heavy lines in Fig. 7. First, it is necessary to mark off on the locus, positions which will be occupied at successively equal time intervals, this being done from a knowledge of the required rapidity of movement between each such position. Such a series of points is shown as 1 to 12, at the upper right of Fig. 7. Next, circles are described on centers 1 and 2, Fig. 7, and are divided into twelve parts, as shown; the commencing position being noted, also the direction of numbering, thereon.

Consider a point such as 8. It can be seen that the center lines of the levers B_1 and B_2 must always pass through centers 1 and 2, respectively, and their lengths being fixed, it is a simple matter to determine the lever positions, and therefore the roller centers c_1 and c_2. Now, for this disposition of the linkage, the cam wheels will have turned through an angle such that division lines 8_1 and 8_2 will be at the positions first occupied by division lines 1_1 and 1_2, respectively, as shown in the diagram. The offsets x and y must also be noted. Now transfer the positions of points c_1, c_2 around to

divisions 8_1, 8_2, respectively, and thereby locate points P_1 and P_2, which will be points on the required cams. Repeating this process for the other positions of the levers will enable the whole of the cam profiles to be determined.

A further motion is incorporated in this mechanism, and this serves to move the needle point sidewise, transversely to the plane of the gears in Fig. 6. This is accomplished as

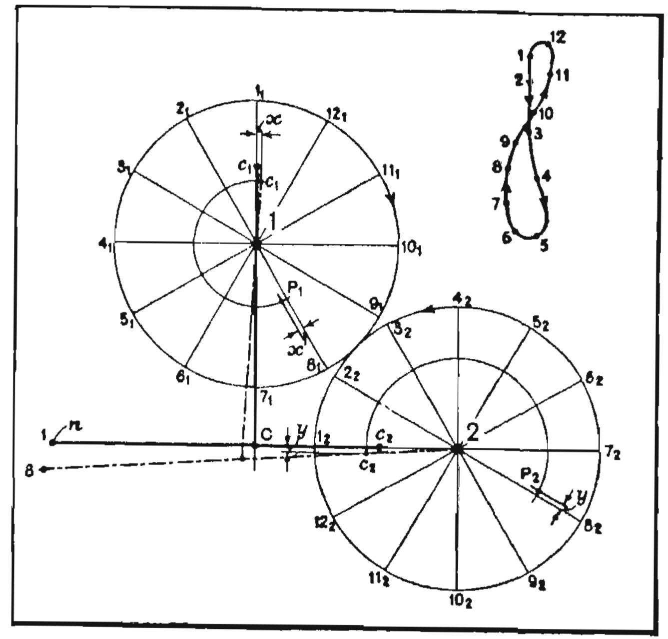


Fig. 7. At Upper Right Is Shown Path of Needle Point Controlled by Mechanism Shown in Fig. 6. The Central Diagram Shows Method of Determining the Cam Profiles.

follows, and during the time the points are moving from 2 to 3 in the locus diagram.

Attached to shaft 1 so as to rotate with it is a bevel gear d, and meshing with this is a bevel pinion e, keyed to shaft f which is carried in bearings on lever B_1. At the lower end of this lever is another pinion g, and at the junction C of the levers is a bevel gear h driven from pinion g, gear h carrying a face cam m. A boss is formed on lever B_2, and a transverse pin o in the boss provides an axle for a lever l, one end of which is engaged by cam m, the other end coming in contact with the end of the needle-point bar at N.

Thus, as the mechanism turns, the bevel gear train causes rotation of the face cam m, and accordingly produces the endwise movement desired.

The whole mechanism provides an unusual example of the compounding of movements, and is suggestive of other applications where complex paths have to be traced out by a moving element.

High-Lift Cam with Low Pressure Angle.— Cams acting on straight-line followers are generally limited to moderate pressure angles because of the side thrust developed against the follower bars. In order to produce a high lift without excessive side pressure, the cam shown in Fig. 8 was designed. It was applied in a wire fabricating machine.

The drive-shaft A, rotating in the direction indicated by the arrow, has keyed to it a cam body B that carries a cam-shaped bar C. This bar is shaped at each end to produce one-half of the total cam lift, and is a slide fit in the gibs at each side of it. A slot in the bar provides clearance for the driveshaft, and the bar is normally held in the position shown by the spring D.

The follower bar G, mounted on the machine member E, carries a block F in which is contained a roller H. Another roller J rotates on a stud which is held in a fixed position in its support I. The rollers H and J are offset relative to each other, so that each one contacts only one end of the cam bar C, which is relieved accordingly at both ends. This arrangement prevents the occurrence of two movements of the follower bar G in one complete revolution of the cam.

In the position shown, the follower bar is about to be raised by the upper end of the cam bar which contacts roller H, and the lower end of the cam bar is about to engage the fixed roller J, so that the cam bar will be raised at the same time that its upper end is raising the follower bar. As a

result, the movement of the cam bar, added to the normal rise produced by the upper end, moves the roller H to the position indicated by phantom lines at X. To produce this lift, with a conventional cam having the total rise on one lobe, a form such as that shown at Y would be required.

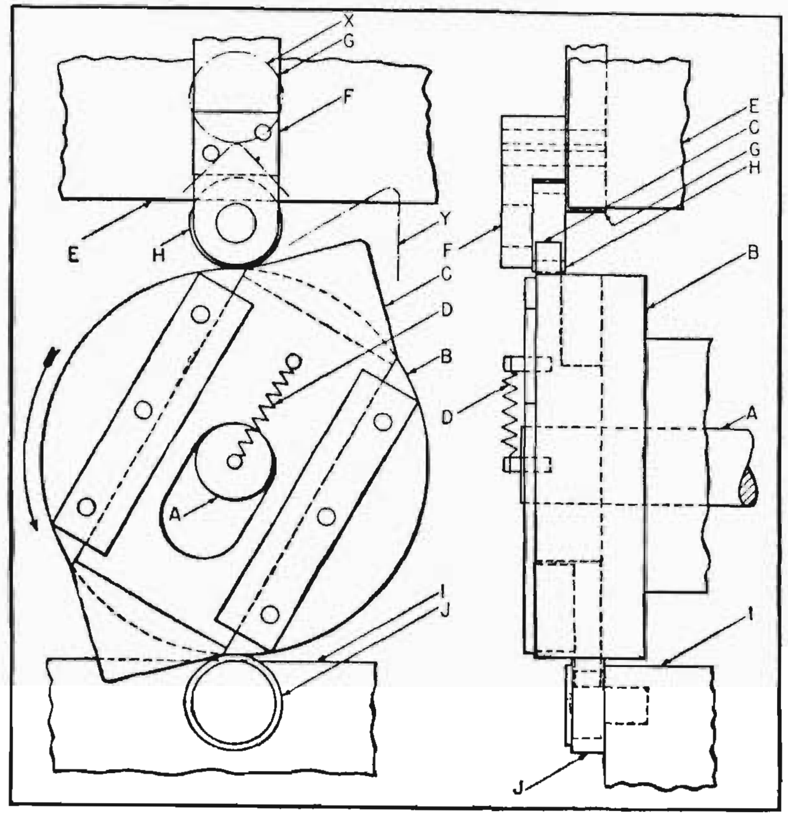


Fig. 8. A Sliding Cam Bar Having Each End Formed to Produce One-Half the Total Rise Provides a High Lift for the Follower Without Excessive Side Thrusts.

Cam Designed to Operate on Alternate Revolutions.— The design of a cam that transmits an irregular oscillating motion to a shaft on each alternate revolution of the camshaft is shown in the accompanying illustrations. This is accomplished by guiding the follower roll C along two dif-

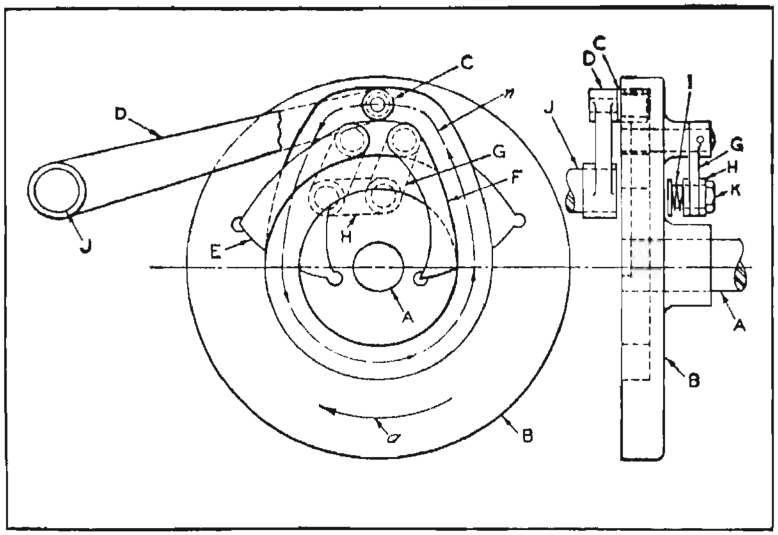


Fig. 9. Cam Mechanism Designed to Raise and Lower Lever D on Each Alternate Revolution of Shaft A.

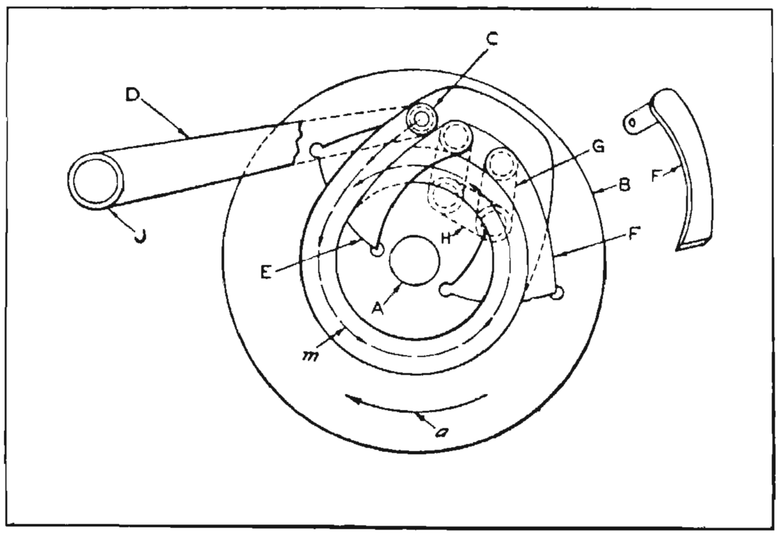


Fig. 10. Cam Mechanism with Levers E and F Set to Guide Roller C into Concentric Groove.

ferent tracks, the concentric track or path indicated by dotted lines at m, Fig. 10, which imparts no movement to lever D, and the cam path n, Fig. 9, which actuates the lever D, producing a rise and fall of this lever with a short rest period at the peak.

Referring to Fig. 9, which shows two views of the mechanism, shaft A carries the cam B, which rotates in the direction indicated by the arrow a. Shaft J receives its motion from cam B through the follower roll C carried on lever D. Cam B is machined to receive the levers E and F, which are recessed into it to less than half the depth of the cam groove.

A better idea of the shape of lever F can be obtained from the view in the upper right-hand corner of Fig. 10. The inner edges of levers E and F are shaped to form a part of the outer edge of the circular follower groove. The outer edges of these levers are shaped to form a part of the inner edges of the follower groove that produces the rise and fall of the roll C. Two levers G on the back of cam B are mounted on the shafts that carry levers E and F and that extend through the body of cam B. Two links H, one on each side of levers G, connect the lower ends of the levers by means of two screws K. Screws K carry the springs I, which act on links H to produce a light friction between levers G and links H.

As shown in Fig. 9, cam B has been rotated in the direction indicated by arrow a, causing roll C to follow the rising side of the cam groove, as indicated by the dotted arrow path n, to the point where the roll is at the peak of the rise. It will be noted that lever E is nested in the recess in the outer wall of the cam groove, while lever F is nested in the inner wall on the opposite side.

Continued rotation of cam B guides roll C to the falling side of the cam track, as shown in Fig. 10. However, this side of the cam track is obstructed by the lever E, as seen in Fig. 9. When roll C comes in contact with lever E, the latter is caused to swing to the opposite side of the groove, thus opening the falling side of the cam groove, as in Fig. 10.

Since lever E is connected to lever F by levers G and links H, lever F is caused to swing an equal amount in the same direction, thus obstructing the rising side of the cam groove.

Further rotation of cam B, Fig. 10, causes roll C to come in contact with lever F, but as lever F is locked against outward movement, roll C is guided in a circular path, as indicated by the dotted arrow path m. Continued rotation of cam B causes roll C to return lever E to its original position, so that roll C may again be guided up the rising side of the cam groove, as in Fig. 9.

The friction caused by the pressure of the springs I serves to hold the levers E and F in position until moved by the roll C. In this manner, shaft J is given an oscillating motion during one revolution of cam B while roll C is following the irregular groove, but is allowed to remain at rest during the next rotation of cam B while roll C is following the concentric path.

Cam Actuated Toggle and Lever Mechanism for Operating Pressure Pad.—The pressure pad A of the mechanism shown in Fig. 11 is moved up and down through a distance B along arc Z by the rotating cam N. The movement of the stud H of the pad along the path indicated by arc Z gives the pad a horizontal movement. This horizontal movement is so slight, however, that it can be disregarded in the case of the mechanism shown.

The action of the toggle mechanism operated by the cam N, roller P, lever M, and rod L enables the pad A to exert considerable downward pressure. The illustration shows the lever M and the pad A at the top of their strokes. When the rotating cam moves the lever M to its lowest position, shown by the dotted lines at Q, rod L will have pivoted lever J about rod F in the direction indicated by arrow K until the center line coincides with the line V.

This action results in bringing link R downward until it is almost in line with lever J, and causes lever S to pivot about the pin D fixed in the housing E until it is in a

vertical position on line W. The link T of the toggle arrangement is thus forced into a vertical position, causing pad A to move downward in the direction indicated by arrow C along the path of arc Z for a distance B. The function of lever G is to restrict the downward movement of the stud H to the path indicated by arc Z.

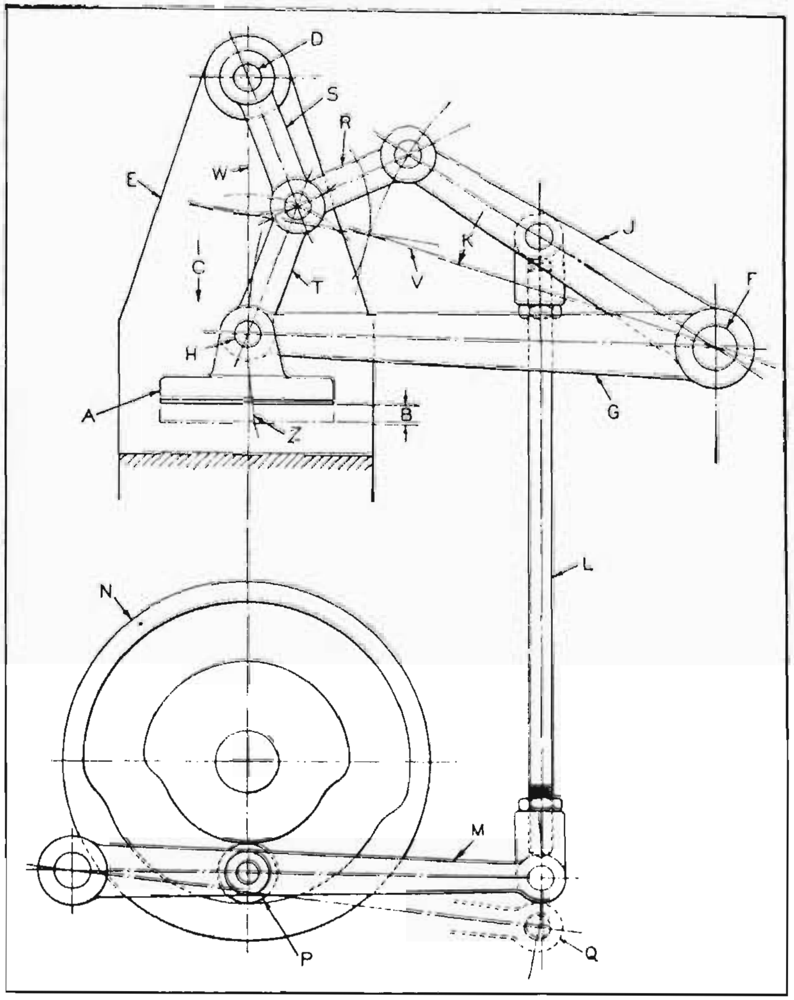


Fig. 11. Cam-actuated Mechanism for Imparting Stroke of Length B to Pressure Pad A.

Cam Mechanism Designed to Control Shaft Speed.—A shaft on a wire-forming machine must occasionally be rotated by a hand-crank. Owing to the nature of the product, it is necessary at one point in the cycle that the shaft be rotated at a predetermined speed. A speed that is either too fast or too slow results in defective work, although a fairly wide range is permissible between the maximum and minimum speeds. Fig. 12 shows how a cam is used to maintain the speed within the required range.

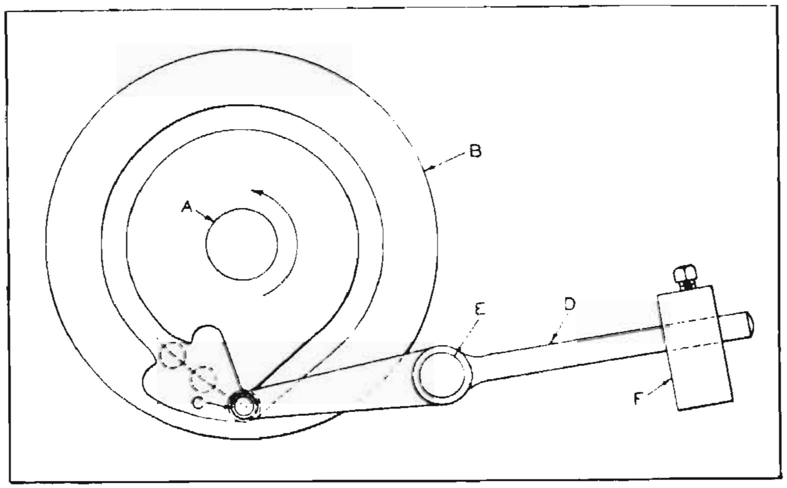


Fig. 12. Cam Arrangement for Controlled Speed of Shaft Driven by Hand-crank.

Shaft A, which is rotated by a hand-crank in the direction indicated by the arrow, carries the cam B. The cam roller C is carried on the end of the lever D which fulcrums on the stud E on a stationary part of the machine. The adjustable weight F holds roller C in contact with the inner cam surfaces.

In operation, if the speed of rotation of shaft A is within the required range, the roller C will follow approximately the path shown by the dotted circles. If the speed of rotation is too high, roller C will have insufficient time to follow its normal path and will lock in the pocket of the outer cam surface. If the speed of rotation is too low, roller C will roll down the sharp incline on the inner cam surface and become locked in the inner pocket. In either case, the cam roller must be returned to its normal path before further rotation is possible. The position of weight F is determined by experiment.

Cam and Planetary Gear Mechanism for Indexing Work-Table and Feeding Drill.—The diagram in Fig. 13 shows a mechanism for automatically feeding and withdrawing a drill and then holding the drill in the "out" position while the work-holding fixture is indexed to the next operating position. After the indexing movement has been completed, the succeeding drilling cycle begins automatically and the cycle of movements is repeated.

The drilling fixture is mounted on the indexing plate L, and is indexed by planetary gearing, which also drives a cam that controls the feeding and withdrawing movements of the drill. The sun gear E, which is driven continuously by the worm K and worm-wheel J, furnishes the drive for both the indexing and drilling movements. An interlocking device is provided which locks the indexing plate L while the drill is being fed, and then locks the drill feeding cam while the table is being indexed. Thus the indexing and drilling movements follow each other automatically and continuously.

Immediately below the indexing plate or table is the indexing ring, which is provided with slots P. These slots may be located at any desired position in the ring to give the angular indexing movements desired. Plate L and the indexing ring are connected to the spider H which carries the planetary gears F, only one of which is shown. Rod A is moved in and out by cam C and is connected to the drill feeding mechanism which it operates. The interlock or stop-bar N is supported by spring M and is moved vertically by cam D. Cams C, D and internal gear G are fastened together.

When the mechanism is in operation, with the members in the position shown in the diagram, cams *C* and *D* revolve while the interlock holds the drilling plate or fixture stationary. Cam *C* provides a short dwell at the end of the withdrawing stroke for the drill, at which time cam *D* pulls the interlock *N* downward, thus releasing the fixture plate for indexing, while locking the feed-cam *C*. When the next slot

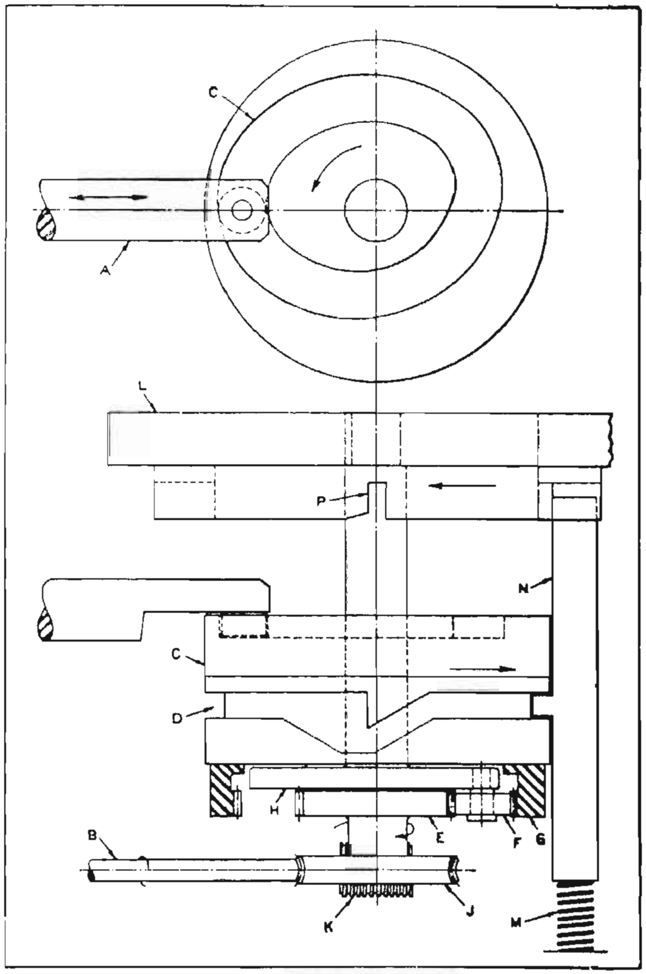


Fig. 13. Mechanism for Indexing Work-table and Feeding Drill.

P in the indexing ring reaches the position above the interlock *N*, the latter member is pushed up by the spring *M*, thus locking the indexing plate *L* and unlocking cams *D* and *C*.

In some cases, it may be desirable to provide means for preventing the interlock *N* from being forced back into the same slot *P*. In other cases, a light drag or brake on the plate *L* may be needed to insure having the cam *D* move around far enough to permit the interlock *N* to move upward freely into the indexing slot. The latter condition was found to exist in the case of the particular mechanism illustrated.

CHAPTER 2

Intermittent Motions from Gears and Cams

When the designer is called upon to provide for the intermittent motion of some part or parts of a machine, he may need to obtain only a stopping of movement for one or more periods in the operational cycle, such as would be required for loading or unloading a work-piece. On the other hand, the designer may also need to provide for the advance of the work or tool for a given distance at specific intervals with periods of rest between. Here, regular intermittent motion of an indexing type is called for.

Either type of intermittent motion may be obtained in various ways. The use of cams which are designed to produce periods of "dwell" is quite common. The employment of gears which are modified to produce intermittent motion is not as general. The mechanisms described in this chapter employ these two methods separately and in combination to produce one or more rest periods in the motion cycle. They supplement those presented in Volumes I and II of "Ingenious Mechanisms."

Intermittent Gear Mechanism.—A change required in a wire fabricating machine presented an interesting problem which was solved in an ingenious manner. On this machine, a uniformly rotating driving shaft transmitted its motion, in direct ratio, to the driven shaft through a pair of spur gears. Owing to a change in the manufactured product, it was required that the driven shaft be given alternately a half revolution and a period of rest corresponding to a full revolution of the driving shaft, and that the speed of the driven shaft and the center distance between the driving

and the driven shafts remain unchanged. Figs. 1 and 2 show how these conditions were met without resorting to complicated declutching mechanisms.

Referring to Fig. 1, which shows three views of the assembly, gear *A* is carried on the driving shaft *B*. The driven shaft *H* carries the two gears *C* and *D*, the faces of which are half as wide as the face of gear *A*, so that both can mesh with gear *A* from the same center of rotation. Gear *D* is free on shaft *H* and meshes with gear *A*. Gear *C* is pinned to shaft *H* and has the teeth removed at two diametrically opposite points, the number of teeth removed being sufficient to prevent the rotation of gear *A* from being transmitted to *C* at these points.

Gear *E* rotates freely on stud *F* and has a definite number of teeth removed from the upper half of its face, so that during a portion of its rotation there is no contact with gear *C*. Gear *E* is idle during the greater portion of the cycle, its main function being to assist in the control of the timing of the rotation of driven shaft *H*. The diameter of gear *E* and the number of teeth removed from it are governed, of course, by the motion required. In this case, gears *A*, *C* and *D* each have 24 teeth.

The number of teeth removed from gear *C* must be sufficient to prevent contact with gear *A* at the required points. The removal of the teeth, however, does not affect the operation of the rotative cycle, the only requirement being that the number of teeth remaining in each section be sufficient to maintain contact between gears *A*, *C*, and *E* during part of the cycle. The number of teeth in the full gear *E* equals the number of teeth in the driving gear *A* which will pass any given point during a complete cycle. As the complete cycle includes one rest period corresponding to a full revolution, and a rotating period corresponding to a half revolution of the driving gear *A*, the ratio between gears *A* and *E* will be 1 to 1.5; in other words, gear *E* in the lower or full-toothed side contains $24 \times 1.5 = 36$ teeth. The number of

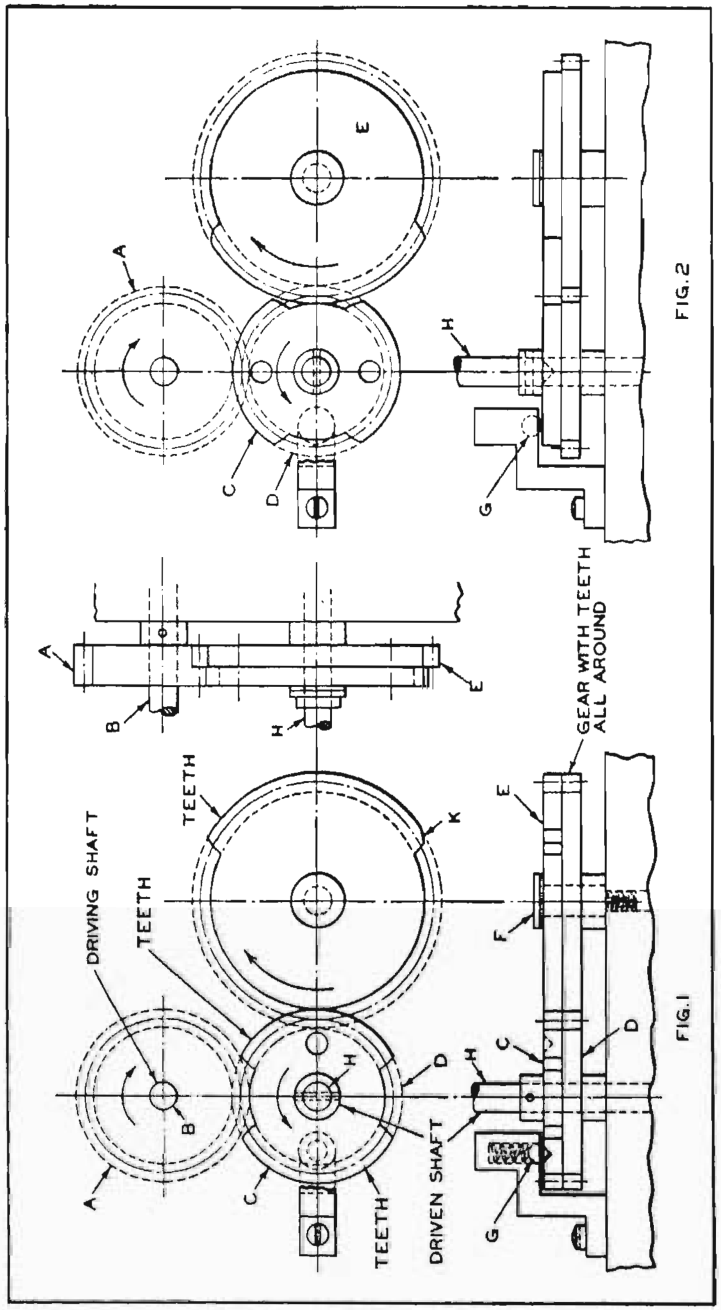


Fig. 1. Mechanism by Means of which Shaft B Transmits a Half Revolution to Shaft H, Followed by a Rest Period Equal to One Revolution of Shaft B. Fig. 2. Same Mechanism with Shaft B Driving Shaft H while Gear E Remains Idle.

teeth removed from the upper half of gear *E* corresponds to one rest period, or 24 teeth.

Fig. 1 shows the assembly at the middle of the rest period. Gear *A*, rotating in the direction indicated by the arrow, transmits its motion in the reverse direction to gear *D*, rotating idly on shaft *H*. As there is no connection between gears *A* and *C*, the latter remains stationary. The rotation of gear *D* is transmitted to gear *E*, which, at this point of the cycle, forms no connection with gear *C*. The leading tooth in the upper half of gear *E* must be cut away somewhat at *K*, for free entry when contact is made with gear *C*.

When the first tooth of gear *E* engages gear *C*, the latter begins its rotation, transmitting it to the shaft *H*. It will be noted that, at this point, although gear *E* is responsible for the rotation of gear *C*, it is not acting as a driver; the teeth of gear *E* merely act as keys to lock gears *C* and *D* together so that they rotate in unison. As the teeth of gear *C* mesh with those of *A*, the drive is direct from *A* to *C*, gears *D* and *E* operating idly.

Fig. 2 shows the mechanism with half of the rotative portion of the cycle completed. At this point, gear *E* serves no useful function, its work being completed as soon as gear *C* comes under the driving action of gear *A*. Although the toothed portion of the upper half of gear *E* is passing through the toothless portion of gear *C*, this is merely incidental and would not occur if the driven shaft *H* were to receive a full revolution between rest periods. As the last tooth preceding the toothless portion of gear *E* passes out of contact with its mating tooth in gear *C*, the latter ceases rotation and the rest period begins, as at this time the toothless section of gear *C* is again in position to clear the teeth of gear *A*, as shown in Fig. 1. The spring ball-stop *G* was added, to prevent accidental rotation of shaft *H*.

Intermittent Spur Gear Drive Mechanism.—The intermittent drive shown in Fig. 3 is incorporated in the design of a machine used in manufacturing a wire product. The two shafts M and N are required to rotate in opposite directions. Shaft N must make a complete revolution while shaft M makes one-half of a revolution, shaft N remaining stationary while the other shaft completes its revolution.

Gear A on the driving shaft M meshes with the gear B on the driven shaft N, the latter revolving at twice the speed of the former. Gear B has three teeth removed at one point

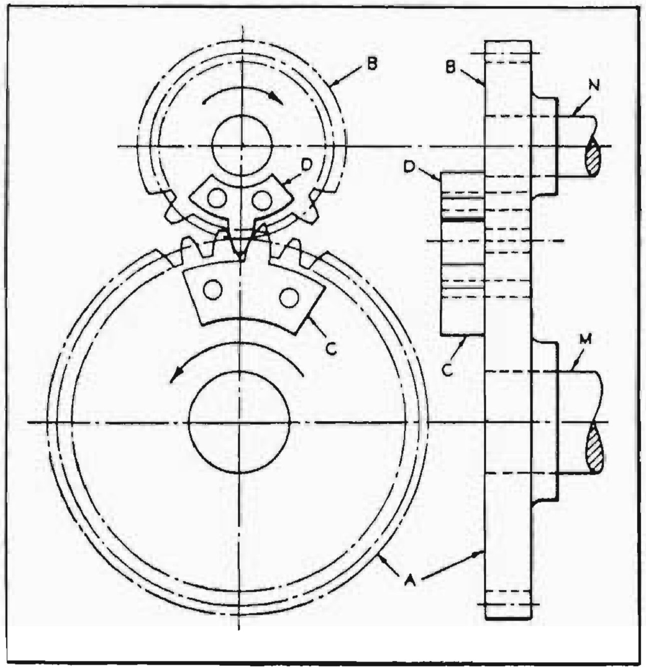


Fig. 3. Mechanism that Permits Shaft N to Remain Idle during One-half Revolution of Driving Shaft M.

where it receives no motion from gear A. The single teeth C and D are fastened to the outer sides of gears A and B, respectively, so that they make contact with each other and transmit motion the same as the regular teeth.

When teeth C and D are in contact, the rotation of gear A in the direction indicated by the arrow produces rotation of gear B in the opposite direction. Gear B receives its mo-

tion from gear A through teeth C and D until gear B is rotated sufficiently to bring the standard teeth into mesh. The single teeth are made slightly longer than the standard teeth to insure contact without danger of clashing. When the standard teeth have made contact, rotation of gear B continues until the space from which the teeth have been removed permits the teeth of gear A to pass, at which time the rotation of gear B is discontinued. At this time, tooth C is 180 degrees from the contact point with tooth D, so that gear B remains stationary until gear A has completed its revolution and teeth C and D are in contact again.

Adjustable Intermittent Rotary Movement.—A conveyor belt used on an assembling table required a dwell or rest period at specified intervals in order to permit loading. As the table was used for a variety of assembling operations,

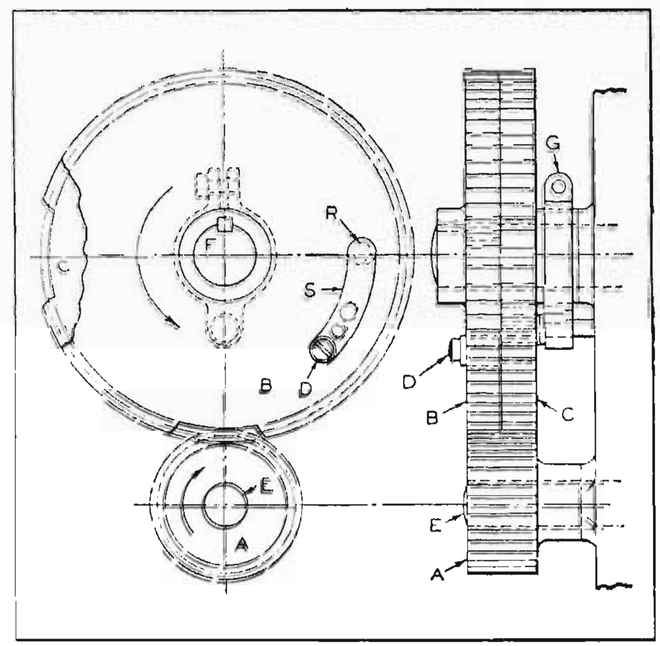


Fig. 4. Mechanism for Obtaining Adjustable Intermittent Rotary Motion.

some means for adjusting the rest period was necessary. The diagram in Fig. 4 shows the adjustable intermittent gear mechanism designed to meet these requirements.

The shaft *E* carries the driving pinion *A*, which rotates in the direction indicated by the arrow. Shaft *F*, which carries the conveyor belt, also supports the mutilated gear *B* which is keyed to it, as well as the mutilated gear *C*, which is free to rotate on shaft *F*. Both gears *B* and *C* mesh with the pinion *A*. Gear *B* has a slot *S* in it, while gear *C* carries the stop *D*, which is located in one of the tapped holes provided for it and is free to move in slot *S*.

The tapped holes for stop *D* in gear *C* are so located that when stop *D* is in any one of them, and in contact with either end of slot *S*, the teeth in gears *B* and *C* will be in exact alignment. The brake *G*, acting on the hub of gear *C*, provides sufficient resistance to prevent accidental rotation of gear *C* due to friction.

In the diagram, pinion *A* is shown in mesh with gear *C*, which it is driving in the direction indicated by the arrow. Gear *B*, however, remains stationary, as the mutilated portion is in line with pinion *A*. This arrangement provides the rest period required for the conveyor belt. Rotation of gear *C* causes stop *D* to be moved to the opposite end of slot *S*, as shown by the dotted circles at *R*. When it reaches this position, the movement of gear *C* is again transmitted to gear *B* through stop *D*, both gears then being rotated in unison and transmitting motion to the belt.

As the mutilated portion of gear *C* reaches pinion *A*, its movement ceases while gear *B* continues to rotate. Gear *C* remains stationary until stop *D* is again located in its original position in the slot in gear *B*, when both gears again rotate in unison. During this period the conveyor belt has continued its movement, the only purpose of the gap in gear *C* being to permit the two gears to change their relative positions, so that the stop *D* will again be in position for the next rest period. The relative positions of the mutilated sections of gears *B* and *C* are immaterial, provided that no part of the two gaps overlap.

When a shorter rest period is required, an additional stop *D* is placed in one of the other tapped holes, thus reducing the effective length of the slot in gear *B*. As the amount of movement of the conveyor belt between stations is controlled by the number of teeth in gear *B*, any change in the period of rest does not affect the length of travel between stations.

Modified Helical-Gear Indexing Mechanism.—Helical gears modified as shown in Figs. 5, 6 and 7 can be used directly for indexing. With the arrangement shown in Fig. 5, shaft *A* will be indexed one revolution for every four revolutions of the driving gear *B*. In other words, shaft *A* will be

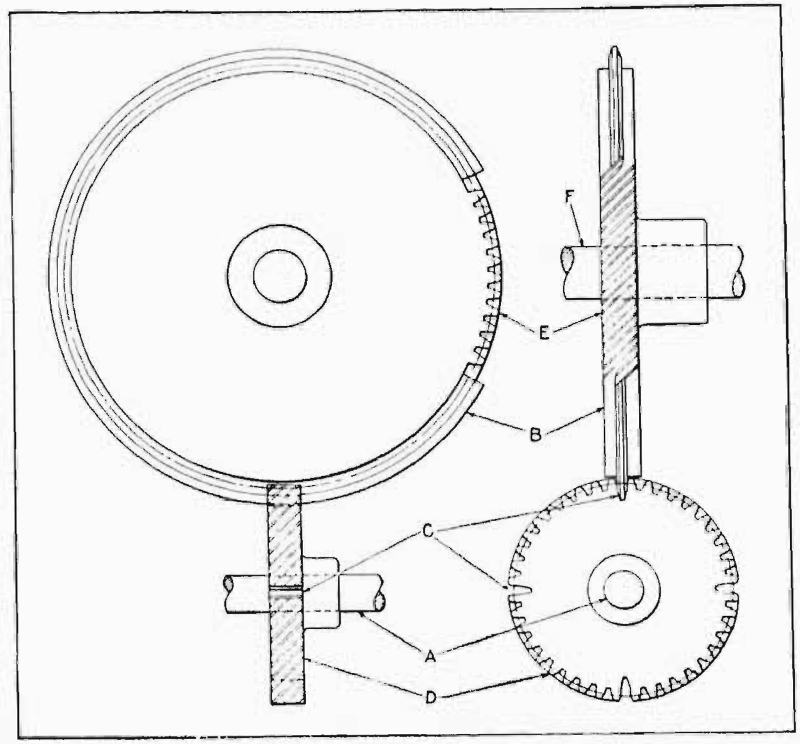


Fig. 5. Modified Helical Gears Index Shaft A One-fourth of a Revolution During One Revolution of Driving Shaft F.

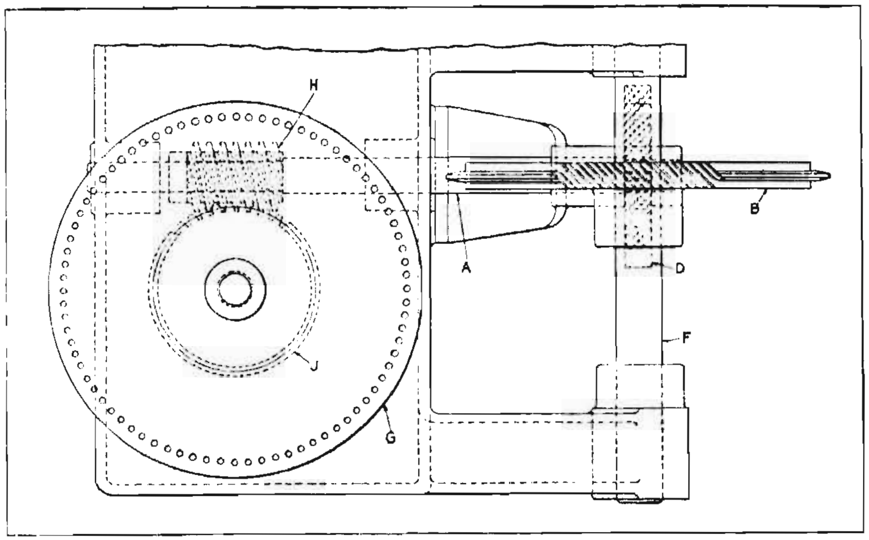


Fig. 6. The Number of Indexings of Plate G per Revolution is Increased by Interposing Worm Gearing Between It and Modified Helical Gears B and D.

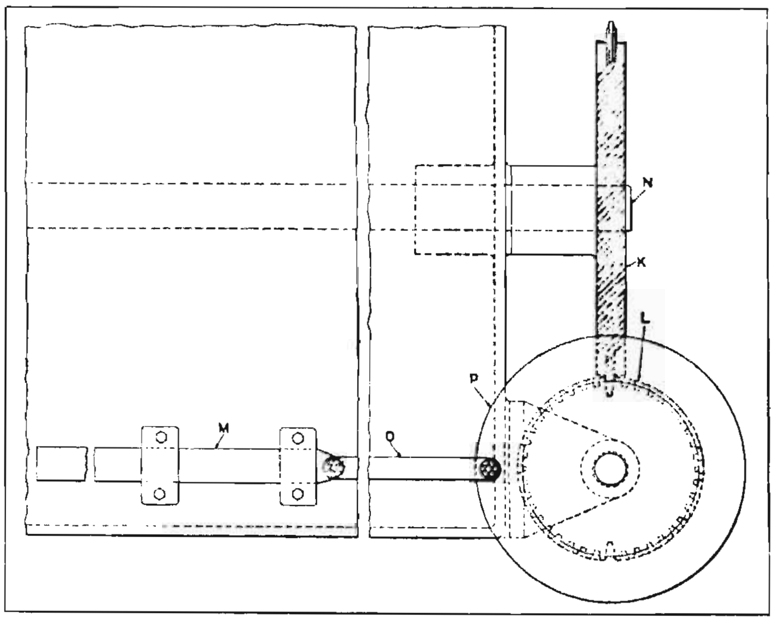


Fig. 7. Modified Helical Driving Gears N and L Arranged to Produce a Reciprocating Movement of Slide M with a Dwell at Each End of Its Travel.

indexed through an angle of 90 degrees during one revolution of the driving gear. The frequency of the indexing can be increased, with a corresponding decrease in the amount of rotation of shaft *A* per indexing, by increasing the number of slots *C* in gear *D* and reducing the number of helical teeth *E* in gear *B*. Similarly, shaft *A* can be indexed through a greater arc by decreasing the number of slots *C*. The number of helical teeth *E* in driving gear *B*, must equal the number of helical teeth in gear *D*, between any two adjacent slots *C*.

The modified helical gears *B* and *D* shown in Fig. 5 can be used as illustrated in Fig. 6 to increase the number of indexing stations without changing the number of revolutions per minute of the driving shaft *F*. This is accomplished by interposing worm-gearing between shaft *A* and index-plate *G*. The pitch of worm *H* may be so selected that worm-wheel *J* will index plate *G* the required number of divisions. In the case illustrated, a 1 1/2-inch pitch, double-thread worm was used to produce the required indexing.

In the mechanism shown in Fig. 7, modified helical gears *K* and *L* are used to reciprocate a slide *M*. This slide is moved to the right in one revolution of driving shaft *N*, and to the left (to the position shown) in the next revolution of the shaft. The motion of the slide in either direction is accomplished during one-half of a revolution of shaft *N*, the slide dwelling at the end of its travel during the other half revolution of the shaft. Link *O* connects the slide to plate *P*, which is keyed to the shaft on which gear *L* is mounted.

Cam-Actuated Intermittent Worm-Drive Mechanism.—A worm drive of rather unique design developed for use on a wire-forming machine is shown in Fig. 8. The mechanism comprising this drive converts a continuous rotary motion into an intermittent rotary motion at a reduced rotative speed. The object of employing the worm and worm-wheel is to give a compact high-ratio speed reduction in combination with the intermittent motion.

Referring to Fig. 8, shaft *A*, mounted in bearing *C*, carries the single-thread worm *B* and rotates in the direction indicated by the arrow. Shaft *A* receives its motion from the driving shaft *E* through the splined sleeve *D*, which per-

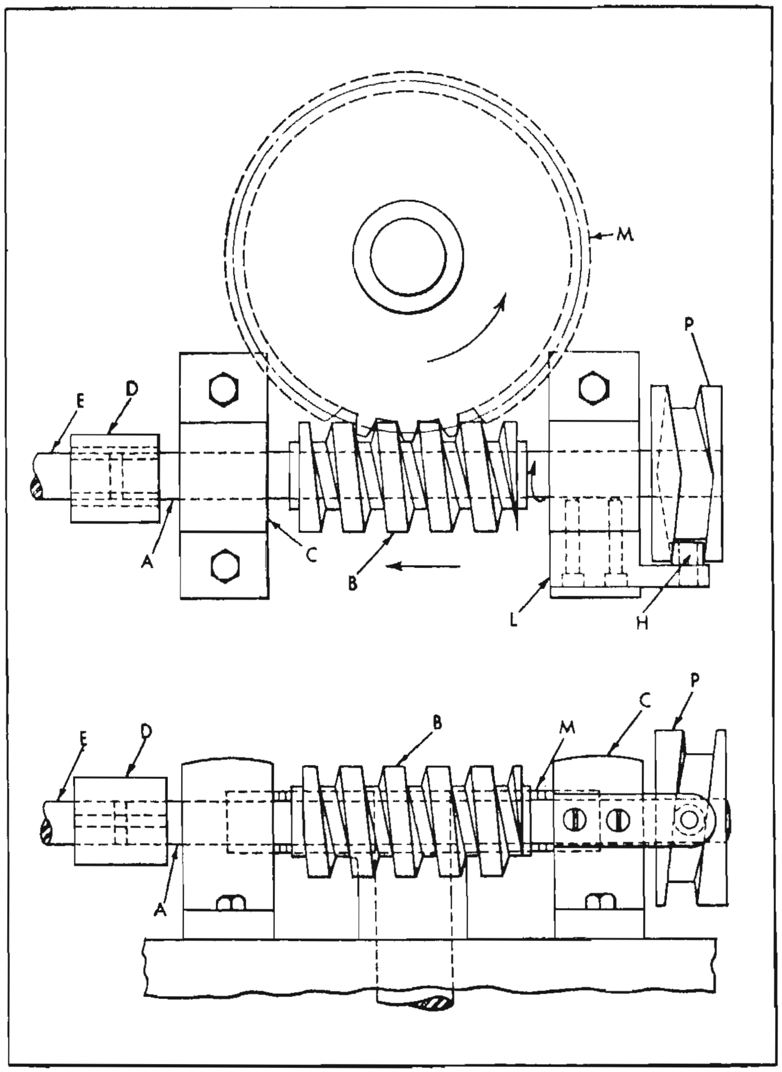


Fig. 8. Worm and Worm-wheel Drive with Cam that Provides Intermittent Rotary Movement.

mits axial movement of shaft *A*. Worm *B* meshes with the worm-gear *M*, to which it transmits motion in the direction indicated. Shaft *A* carries cam *P*, which rotates with it. Bracket *L*, attached to bearing *C*, carries roller *H*, which operates in the groove of cam *P*. It is obvious that, owing to the fixed position of roller *H*, the rotation of cam *P* will cause shaft *A* to be moved axially.

The groove in cam *P* is shaped to produce a uniform axial motion in one direction during one half revolution, and in the reverse direction during the other half revolution. The lead of cam *P* is equal to the lead of worm *B*.

If worm *B* were fixed against axial movement, one revolution of worm *B* would produce a movement of gear *M* equivalent to the lead of the worm. However, in addition to rotative motion, worm *B* is also given an axial motion by cam *P* acting against roller *H*, as mentioned; thus the rotation of gear *M* is effected by both the rotative and axial movements of worm *B*. As the lead of worm *B* and cam *P* are equal, the motion of gear *M* equals that which would be produced by an axially fixed worm of double the lead of worm *B*.

As shaft *A* continues to rotate, roller *H* is passed by the high point of cam *P*, which reverses the axial movement of shaft *A*. When this occurs, there is no movement of gear *M*, as the axial movement of shaft *A* produced by cam *P* is equal to the lead of worm *B*, but is in the reverse direction. In this manner, the axial movement of shaft *A* neutralizes the lead of worm *B*, the worm merely turning or threading itself back to its original position without imparting any motion to gear *M*. The effect is to produce a series of partial revolutions of gear *M* with equal rest periods between the movements.

Intermittent Mechanism with Hourglass or Cylindrical Cam.—The mechanism shown at the left of Fig. 9 consists of an hourglass cam with a helical cam surface that extends part way around the hourglass surface and then forms one side of an annular groove for the remainder of the circumference. This cam engages a toothed wheel as shown at the left of Fig. 9. As a tooth of the wheel is engaged by the helical cam surface, the wheel is revolved until the tooth reaches the annular groove. The wheel then ceases to rotate

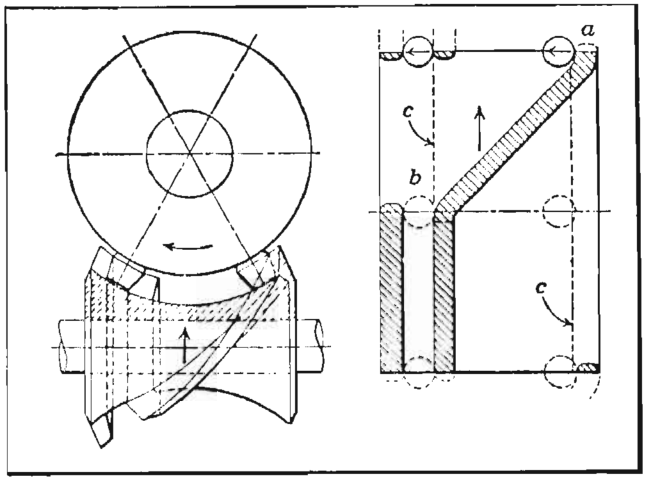


Fig. 9. Intermittent Worm-gear Drive in View at Left, with Development of Circumference of Similar Parallel Worm in View at Right.

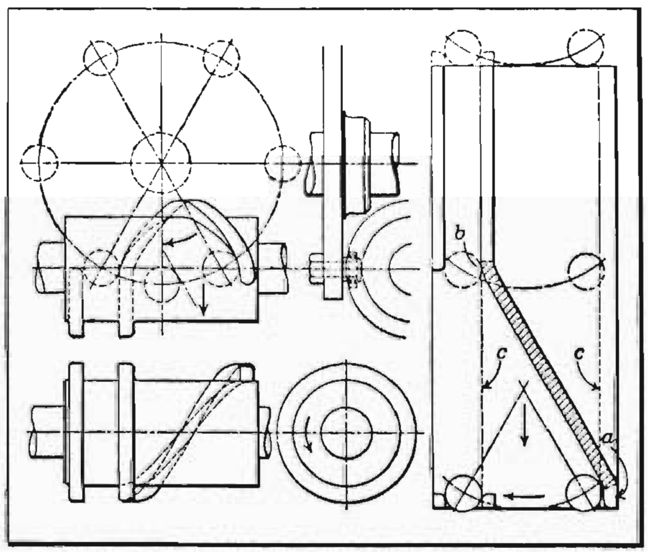


Fig. 10. Alternative to Intermittent Worm-gear Drive Design Shown in Fig. 9.

until the next tooth is engaged by the cam helical surface. The teeth on the wheel are spaced so that at the moment when one tooth has passed out of the annular groove the helical cam surface comes into contact with the next tooth, thus providing intermittent rotation of the wheel.

The helical cam surface is modified somewhat at the beginning and end to insure a gradual start and stop of the wheel. The wheel shown has six teeth so that the cam makes six turns for each turn of the wheel.

The view to the right of Fig. 9 shows a development of the pitch surface of a cylindrical cam with a similar helical cam surface and annular groove.

In Fig. 10, at the left, is shown a cylindrical cam with a cam surface and annular groove engaging a roller toothed disk. The view at the right of Fig. 10 shows a development of the pitch surface of this cam.

Obtaining an Intermittent Motion from a Uniformly Reciprocating Slide.—In tooling a wire-forming machine for a certain job, it became necessary to have a uniformly reciprocating vertical slide intermittently actuate a horizontal slide. The horizontal slide was to finish one-half of its cycle in each complete cycle of the vertical slide. This was accomplished by means of the mechanism shown in Fig. 11.

Slide *A* reciprocates uniformly in a vertical plane, carrying lever *B*, which is free to swing about stud *K*. At the lower end of the lever is a pin *C*, which enters a recess milled in fixed member *D*.

Horizontal slide *E*, which consists of a right- and a left-hand section, is free to reciprocate in ways provided in member *D*. The two sections of this slide are connected by plate *F*, which serves as a cam to produce the required motion. Rocker cam *G* can be swung about stud *L*, within member *D*, against light frictional resistance created between the under side of the rocker cam and member *D* by spring *H*.

As shown, vertical slide *A* is in its uppermost position and horizontal slide *E* is in its extreme right-hand position. At this point, the portion of pin C within the recess in member D is in contact with rocker cam G on one side and the cam surface of plate F on the other. When the vertical slide begins its downward stroke, the pin will be lowered in the groove formed between the rocker cam and member D until it has passed the lower end of the cam. At this point, the vertical slide completes its downward movement, and lever B will swing to a central position, as indicated by the broken outline Y.

On the upward stroke of the vertical slide, the pin will rise along the left-hand side of the rocker cam until it reaches the position indicated by broken outline Z. Here

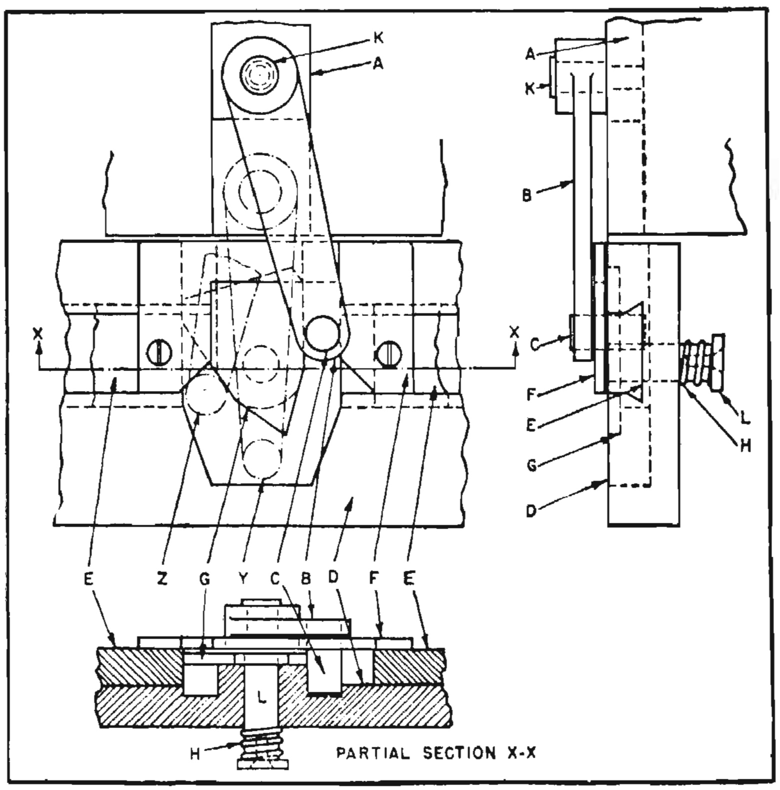


Fig. 11. Intermittent Motion of Horizontal Slide E is Obtained from a Uniformly Reciprocating Vertical Slide A by Means of Rocker Cam G and Cam Plate F.

the pin will contact the angular cam surface of plate F, causing the plate and the vertical slide to move to the left.

Simultaneously, the rocker cam will be pivoted so that its lower pointed end will lie on the left-hand side of the vertical slide center line. This completes half the cycle of the horizontal slide. On the next cycle of the vertical slide, the reverse action takes place, completing the cycle of the horizontal slide as it is returned to the position shown.

Thus the horizontal slide E is intermittently traversed first to the right and then to the left continuously by the reciprocating vertical slide A.

Mechanism for Converting Uniform into Intermittent Reciprocating Motion.—A machine for fabricating a wire product is required to move the work at various stages of the operating cycle by means of reciprocating push-rods. Because of a change in the product, it became necessary to reduce the length of the reciprocating movement and provide a period of rest without any major alteration in the actuating mechanism, which is required to operate other units of the machine. The changed mechanism by means of which the desired motion was accomplished is shown in Fig. 12.

Referring to the two upper views, a channel was machined in the original reciprocating push-rod B to carry the auxiliary rod C, which was made to slide within the channel. The block A serves as a guide for the assembly and, by means of the irregular slots D in its outer walls, aids in converting the uniform motion of rod B into the intermittent motion required for rod C. The irregular slot D is machined in both outer walls of part A so that the slots are in alignment and pin G will slide freely therein. A slot E likewise is machined in both walls of rod B so that pin G will slide freely. Rod C is also provided with a slot for pin G, as shown at F. This completes the assembly, except for the two plates P which serve as retainers for pin G. These plates, however, are not shown in the two

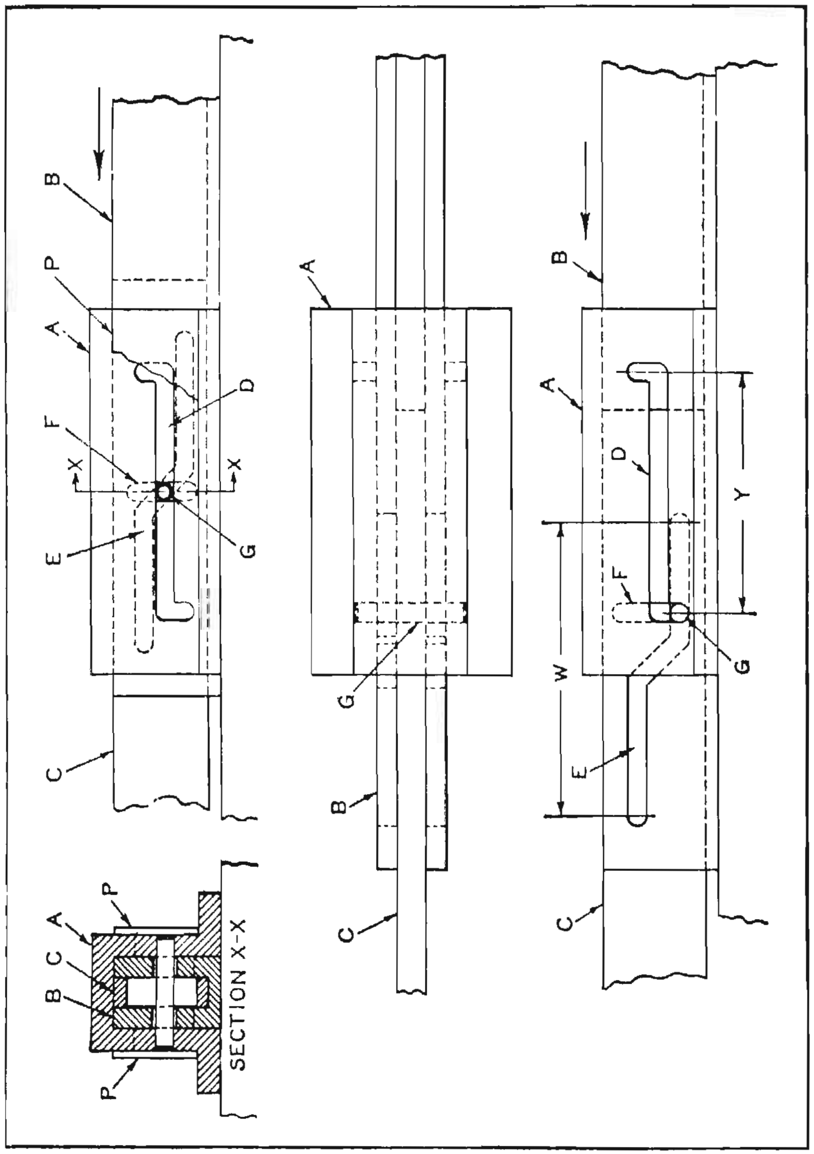


Fig. 12. Mechanism to Convert Uniform Reciprocating Motion of Bar B into Intermittent Reciprocating Motion of Bar C.

lower views, which illustrate the operation of the mechanism.

Referring to the top view, assume that bar *B* is moving in the direction indicated by the arrow and that pin *G*, at this point, lies in the angular section of slots *E* in the bar *B*, and in the center of the slot *F* in bar *C*. Any horizontal motion given to pin *G*, which passes through the vertical slot in bar *C*, must produce a corresponding movement of bar *C*. In the position shown, pin *G* is free to move horizontally in the slots *D* in part *A*, but is restricted from any vertical movement. As bar *B* is the actuating member and as pin *G* is locked in the angular section of slots *E* in bar *B* by the restricting influence of slots *D*, bar *C* is carried along with bar *B*, which transmits its motion to bar *C* through the slot *F*.

The rest or dwell portions of the cycle are accomplished in the following manner: Continued movement of bar *B* in the direction indicated by the arrow moves pin *G* to the ends of slots *D*, thereby preventing further horizontal movement of pin *G*. When this occurs, the angular portion of slots *E* in bar *B* forces pin *G* into the pockets at the ends of slots *D*, as shown in the bottom view. In this position, bar *C* is incapable of further horizontal movement, because it is locked to part *A* by the pin *G*. As pin *G* is now in the horizontal portions of slots *E*, bar *B* continues its motion in the direction of the arrow without transmitting any motion to pin *G* or bar *C*.

Reversal of bar *B* takes place before the ends of slots *E* strike pin *G*. The horizontal movement of bar *C* is thus controlled by the length of slots *D*, or the distance indicated by *Y* in the bottom view. The maximum movement of bar *B* equals the distance *Y* plus twice the distance indicated by *W*. The mechanism operates in a similar manner on the reverse stroke of the reciprocating driving bar *B*, pin *G* moving upward, however, when it reaches the ends of slots *D*, instead of downward.

Indexing Mechanisms for Small Film Projector.—The film indexing mechanism shown in Fig. 13 was designed for a small motion picture projector, the object being to obtain the desired indexing motion with the minimum number of moving parts. The indexing movement imparted to the shoe H by the mechanism carries the film from the position indicated at M to that indicated at N.

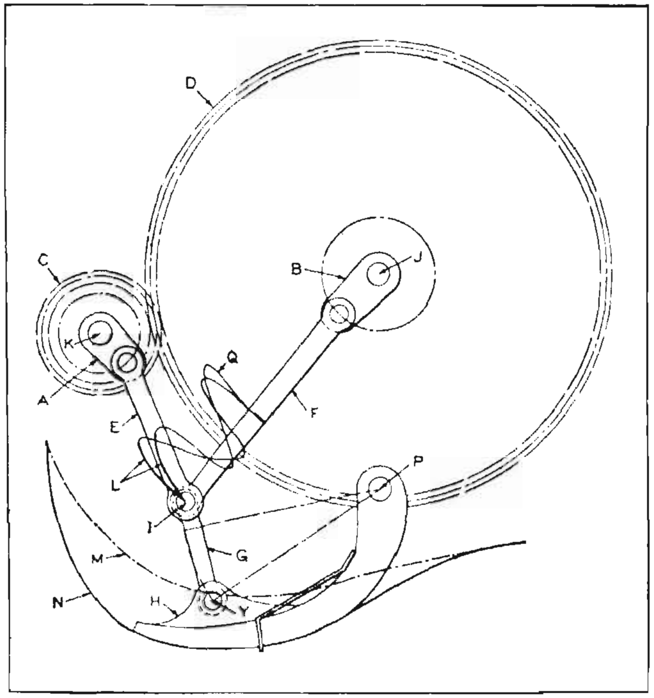


Fig. 13. Double-crank Link and Gear Mechanism for Indexing Motion Picture Film.

The cranks A and B are mounted on shafts carrying the two spur gears C and D, which are always in mesh. Gear D has four times as many teeth as gear C, so that crank A makes four revolutions to one revolution of crank B. The ends of cranks A and B are connected to rods E and F, the free ends of which are united by a third rod G.

This rod actuates the shoe or lever H which swings about the pivot pin P when indexing or moving the film from M to N.

During one revolution of crank B, which corresponds to four revolutions of crank A, a very complicated curve Q, having eight single strokes or four double strokes, is traced by the point I. The range of the curve is limited by four circles, the radii of which are equal, respectively, to the lengths of the following members: $F + B$ and $F - B$ drawn around point J as a center and $E + A$ and $E - A$ drawn around point K as a center. For the movement of the film, only the last two single strokes, indicated by the heavy lines at L, are utilized, whereas the other three double strokes are not used. During the time in which the

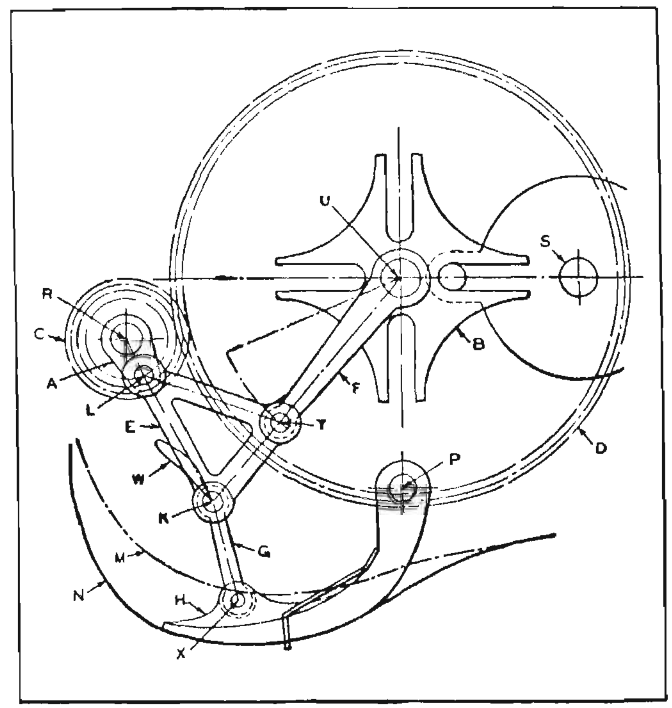


Fig. 14. Geneva Stop Motion Link and Gear Mechanism for Indexing Motion Picture Film.

point *I* traces the lines representing these three double strokes, gear *D* makes three-fourths of a revolution without transmitting any motion to the film.

Fig. 14 shows a similar driving mechanism that has been developed, which gives an equivalent indexing movement, but avoids the idle time period. This mechanism rests at first for three-fourths of a revolution of the driving shaft *S*. For this purpose, a Geneva stop motion with four stops and one roller is used. During one-fourth of the rotation of the driven member *D*, the point *K* of the mechanism is required to make a double stroke. Therefore, between the driving member *B* of the Geneva stop motion and crank *A* of the mechanism there must be a ratio of 4 to 1. Also, the index dial of the Geneva stop drive should be provided with a large gear *D*, whereas the crank *R* should have a small gear *C* which is one-fourth the size of gear *D*.

During the indexing motion, the crank *A* makes one full turn or rotation. This rotational movement is then changed by a four-bar-link motion involving the members ***A***, ***E***, ***F***, and a fixed member, causing them to give a swinging motion to the point *K*. Point *K* follows the path indicated by lines *W* if the connecting lines from the point *K* through the free links *L* and *T* pass through the respective fixed axles *R* and *U* of the mechanism.

For the sake of simplicity, the fixed axle *U* of the four-bar-link was selected as the fixed axle of the index dial. With the position of the curve point thus fixed, the mechanism can be laid out or drawn. If the lengths of crank *A* and the swinging lever *F* are so proportioned that crank *A* can make a full rotation, the point *X* will follow a path similar to that of point *Y*, Fig. 13.

Intermittent Rotary-Motion Gear and Cam Mechanism.— Shaft *A* of the mechanism shown in Fig. 15, rotating in a clockwise direction at a constant speed, is required to transmit intermittent rotating motion in a counter-clockwise direction to the driven gear *J*. Shaft *A* is keyed to a

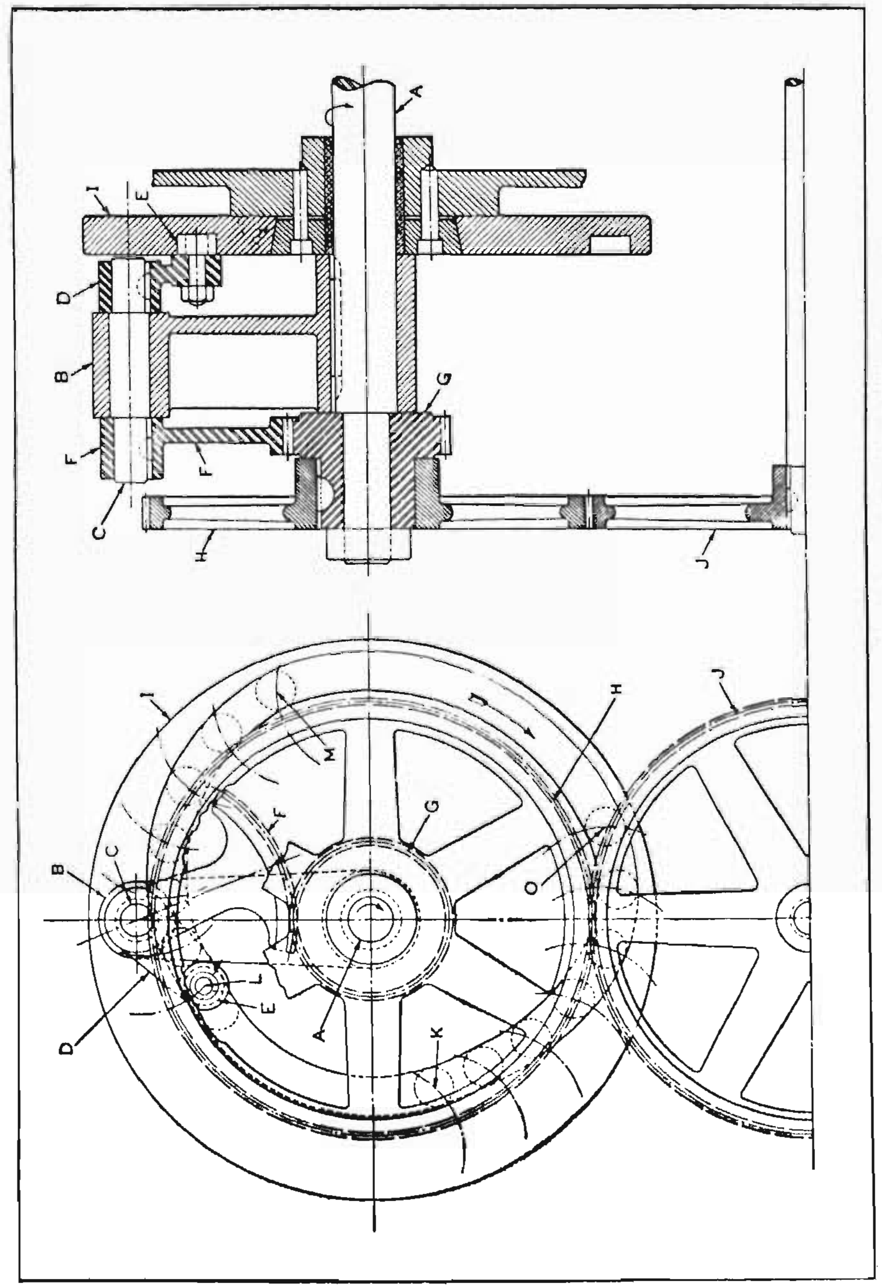


Fig. 15. Rotation of Shaft A at Constant Speed Imparts Intermittent Motion to Gear J.

driving arm B, which carries a short shaft C. Keyed to one end of shaft C is a crank-arm D with a cam-follower roller E which, traveling in a groove in stationary base cam I, transmits an oscillating rotary motion to shaft C.

To the front end of shaft C is keyed a segment gear F, which is in constant mesh with gear G. Gear G is a running fit on shaft A, and has a hub to which gear H is keyed. Gear H, in turn, is in mesh with gear J, keyed to the shaft that is to be given the intermittent rotating motion. Since gear J has the same number of teeth as gear H, it will have the same intermittent motion, but rotation will be in the opposite direction, or counter-clockwise.

Referring to the view to the left in Fig. 15, it will be clear that so long as the cam-follower roller E is traveling in the concentric portion of the cam groove from K to L, there will be no rotary motion of shaft C in arm B. During this period, arm B and segment gear F simply transmit rotary motion to gear H in a clockwise direction at the same speed as that of shaft A. Thus, so long as there is no rotary motion of shaft C in its bearing in arm B, the latter member, together with segment gear F and the gears G and H, are effectively locked together and rotate as a single member.

When shaft A and arm B, rotating in a clockwise direction, cause the follower-cam roller E to travel from L to M, the rise in the groove of cam I causes segment gear F and its shaft C to rotate in a clockwise direction in arm B. Since segment gear F is in mesh with gear G, a rotating motion is transmitted to gears G and H in a counter-clockwise direction, which is opposite to the clockwise rotation imparted to them by arm B alone.

Now, since the counter-clockwise movement imparted to gears G and H by segment gear F is equivalent to the clockwise movement imparted by arm B alone in making the quarter revolution required to carry roller E from L to M, gears G and H will remain idle during this period.

To obtain this intermittent or idle period, the rise of the cam groove from L to M must be just sufficient to cause segment gear F to transmit counter-clockwise motion at the same rate as clockwise motion transmitted by arm B

Continued clockwise movement of arm B during the next quarter revolution carries roller E from M to O. As this portion of the groove in plate I is concentric with shaft A, there will be no rotation of the segment gear in arm B, and shaft A, arm B, segment gear F, and gears G and H will revolve together for one quarter revolution.

During the next quarter revolution of shaft A, in which roller E travels from O to K, the fall in the cam groove will cause segment gear F to rotate in a counter-clockwise direction, and, consequently, transmit motion to gears G and H in a clockwise direction, which is in the same direction in which arm B is driving these gears. Therefore, in this case, the driving motion imparted to gears G and H by segment gear F is added to that imparted by arm B alone, thus doubling the rotating speed of gears G and H during this quarter revolution of shaft A. Gears G and H are, therefore, rotated through an angle of 180 degrees, while shaft A rotates through an angle of 90 degrees.

Summarizing the operation of the intermittent action of the mechanism, rotation of driving shaft A at constant speed has the following results: There is no movement of gears G, H, and J while arm B carries roller J through one-fourth revolution from L to M; the driven gears rotate at the same speed as shaft A and in the same clockwise direction while E travels from M to O; the driven gear J rotates at twice the speed of shaft A in a counter-clockwise direction while E travels from O to K; the driven gears rotate at the same speed as the driving shaft A while roller E is carried from K to L. The angular position of shaft A and arm B at which the dwell period of gear J begins can be varied by adjusting cam I and clamping it in place when the proper adjustment has been made.

Simple Gear and Star-Wheel Indexing Device.—The indexing device shown in Fig. 16 constitutes the main mechanism of an automatic stamping device. It has proved almost as efficient as the well-known Geneva stop motion, and is much easier to produce. The main drive-shaft *A* carries the disk *B*, which has a number of teeth cut on its periphery. This disk acts in the same way as the check disk of the Geneva motion.

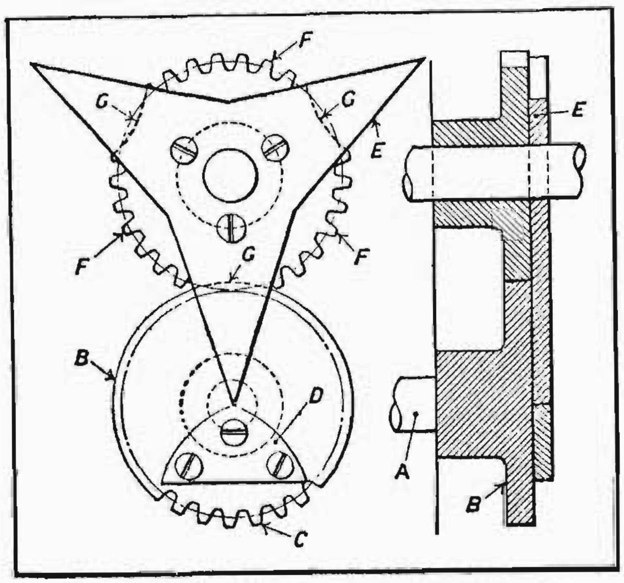


Fig. 16. Indexing Mechanism Developed for Automatic Stamping Device.

Fastened to the face of the disk *B* immediately above the toothed sector *C* is an operating cam or lug *D*. The driven member *E*, which is shaped like a three-pointed star, has three gear sectors *F* secured between the points of the star. As the main drive-shaft *A* is revolved, one of the sides of the operating piece *D* pushes against the side of the star-wheel and causes it to revolve. The toothed sectors on disk *B* and on star-wheel *E* will then engage each other, and in the particular case shown in the illustration, the star-wheel will be rotated through 120 degrees until its motion is arrested by means of the check portions *G* coming into contact with the untoothed periphery of the disk *B*, similar to that in the Geneva stop mechanism

High-Speed Intermittent Gearing.—One difficulty experienced with all types of intermittent gearing, including Geneva movements, is their inability to function properly at high speed. To permit ready engagement and disengagement at relatively high speed, it is necessary to have a certain degree of freedom of motion between the mating parts. This, in turn, causes clashing, incorrect timing, excessive operating noise and wear, and jamming of the mechanism in a short time.

To overcome these difficulties, designers resort to various expedients, such as spring- or inertia-operated prestarting elements which are intended to lessen the initial shock of engagement. These devices, when properly designed and applied, enable high speeds to be employed, but do not extend the range to the high speed sometimes required.

Fig. 17 shows an intermittent gear mechanism designed to meet the requirements of nearly noiseless operation at extremely high speed, and positive locking during the rest period. The mechanism consists of driving shaft *A* which carries a cylinder *B*, a driven shaft *C*, and an indexing shaft *D*. The driven and indexing shafts *C* and *D* are at right angles to the drive-shaft *A*. The cylinder *B* actually consists of two gears, a spiral gear *E* and a circular rack *F*. From each of these two gears certain numbers of teeth are cut away, so that there will be no engagement between the spiral teeth *E* and the spiral gear *G* on shaft *D* when the rack teeth *F* are engaged with the spur gear *H* on shaft *C*. The two shafts *D* and *C* are interlocked by gears *J* and *K*.

When the gearing is in operation, shaft D is locked in position during the time that circular rack F is engaged with spur gear H. The spiral teeth E engage spiral gear G at the moment when circular rack F becomes disengaged from gear H. While spiral teeth E and gear G are engaged, the shaft D is rotated a predetermined portion of a revolution, as determined by the part of the circumference occupied by teeth E. This movement is transmitted to shaft C through gears K and J.

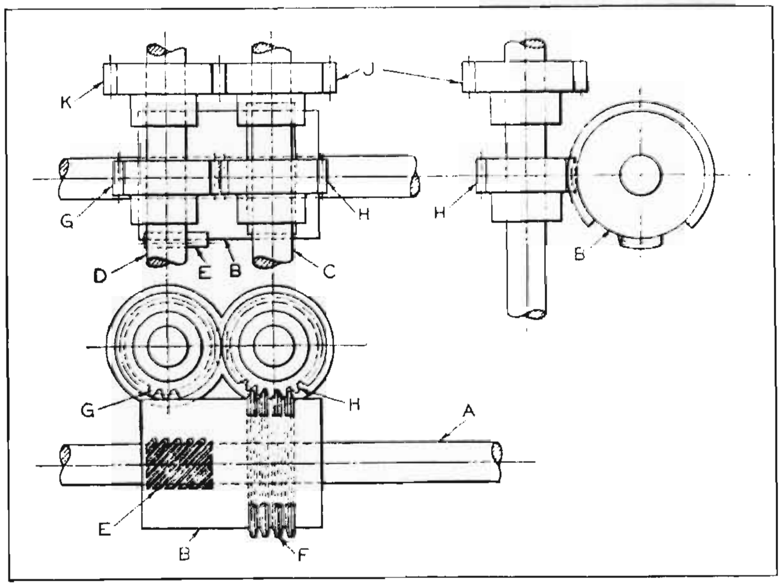


Fig. 17. Intermittent Gearing Designed to Operate at High Speed.

As the teeth E become disengaged from gear G, the rack teeth F enter gear H and lock shaft C against further rotation. The accuracy of the mechanism is not affected by the amount of backlash existing between gears E, G, K, and J, because the effective locking action is between F and H. To facilitate engagement, the entering ends of teeth F are pointed, the same as teeth E.

The mechanism described is positive, accurate and quiet in operation at extremely high speeds. These desirable features are obtained by having all contacts between the driving and the locking members made by sliding surfaces.

Irregular, Intermittent, Rotary-Motion Mechanism.—The design of an irregular, intermittent, rotary-motion mechanism which is used on a machine for producing a twisted wire product is shown in the accompanying illustration. With this mechanism, the driven shaft C is given a complete revolution during a half revolution of the driving shaft A; and during the second half revolution of the driving shaft, the driven shaft remains stationary, except for a slight change of position.

Referring to Fig. 18, left-hand view, driving shaft A carries gear B, which transmits its motion through gear D to shaft C. Gear D has a full complement of teeth, while gear B, which is of a pitch diameter equal to twice the pitch diameter of gear D, has only a sufficient number of teeth to produce a full revolution of gear D. Gear B carries an internal cam E, the groove of which receives roller F mounted on gear D.

In this view, gear D is shown beginning its rest period. The last tooth in the toothed section of gear B is just leaving its mating tooth in gear D, and roller F has entered the groove in cam E, the leading end of which is shaped to suit the path in which roller F travels while gear D is still being driven by gear B. From this point, the groove in cam E is formed to a true radius with the center of shaft A, which permits cam E to continue its motion without imparting any motion to gear D. It will be noted that, during the rest period of gear D, roller F is off the center line between shafts A and C, which locks gear D in position, preventing accidental rotation.

As the rise in cam E reaches roller F, the effect is to give a slight rotative motion to gear D in its original direction. Gear D then remains stationary in its changed position for a short time, returning to its original rest

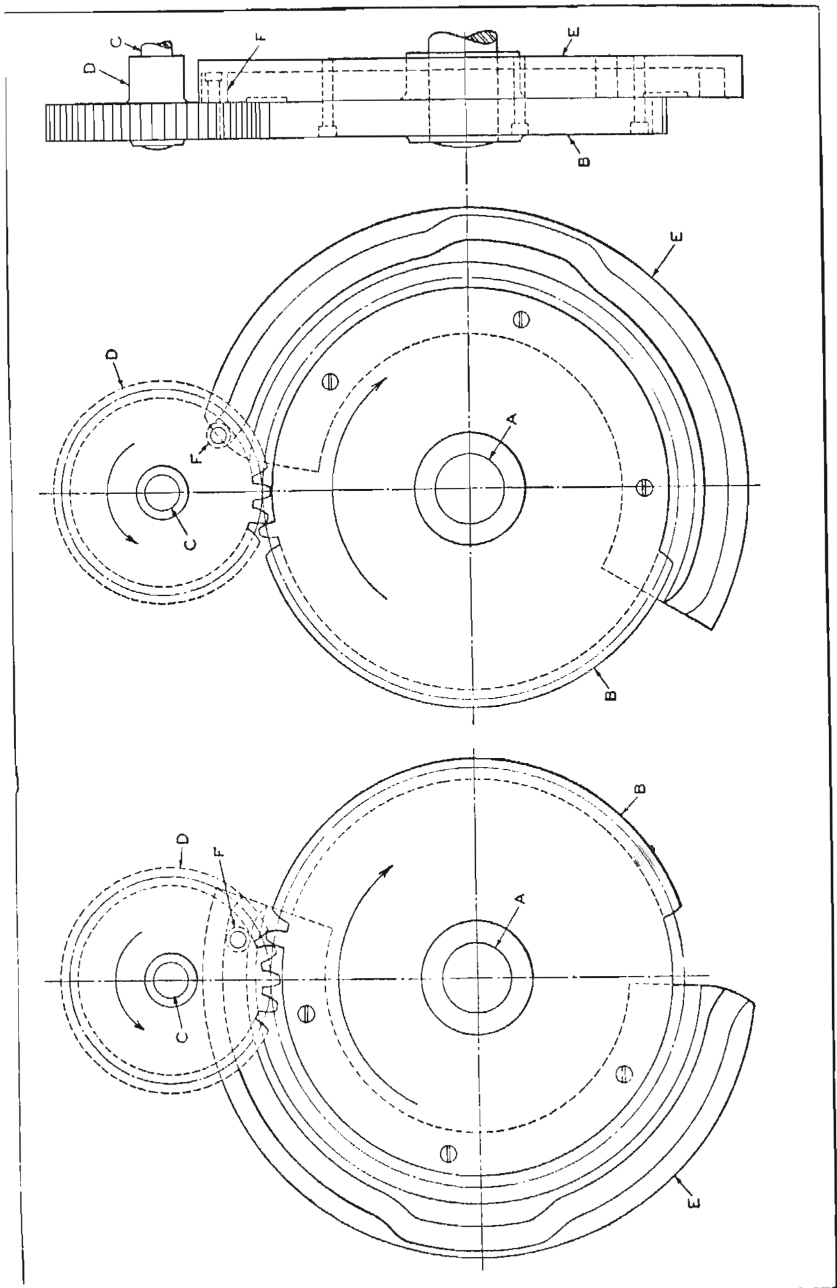


Fig. 18. (Left) Irregular, Intermittent, Rotary-motion Mechanism with Driven Gear D about to Begin Dwell; (Center) Driven Gear D about to Begin Rotation; (Right) End View of Mechanism.

position as the fall of cam E passes roller F. Gear D now remains stationary for the remainder of the rest period.

In Fig. 18, center view, gear D is shown just about to start its rotation, the first tooth of the toothed section in gear B engaging its mating tooth in gear D. Before these teeth become engaged, the rise at the end of cam E produces a gradual partial rotation of gear D corresponding to that which would be given it by one tooth. In this manner, gear D is already in motion when the teeth engage, thus eliminating shock on the first tooth. Gear D then makes a complete revolution before roller F again enters the groove in cam E. An end view of the assembled mechanism is shown in Fig. 18, at the right.

Intermittent Rotating Mechanism Designed for Smooth Operation.—The twisting spindle of a machine for fabricating a twisted wire product was required to finish its cycle in approximately half the time needed for the complete cycle of operations performed by the machine, and then to rest while the remaining portion of the cycle took place. Owing to space limitations, a mutilated gear was selected as the simplest means for producing the required movement. As the driven spindle was required to be positively locked during its rest period, a locking arrangement was attached to the gears; but, on trial, it was found that although the driven spindle rotated at comparatively low speed, the momentum, due to the weight of the rotating parts, was sufficient to produce a severe hammering effect at the end of the rotating period. The design that finally proved satisfactory is shown by the left and center diagrams of Fig. 19.

The plan view at the left shows the mechanism shortly before the termination of the rotating period of the driven spindle D. Driving shaft A carries the mutilated gear B, which rotates in the direction indicated by the arrow. Spindle shaft D, rotating in the opposite direction, carries the full gear C which meshes with gear B. Gear B carries

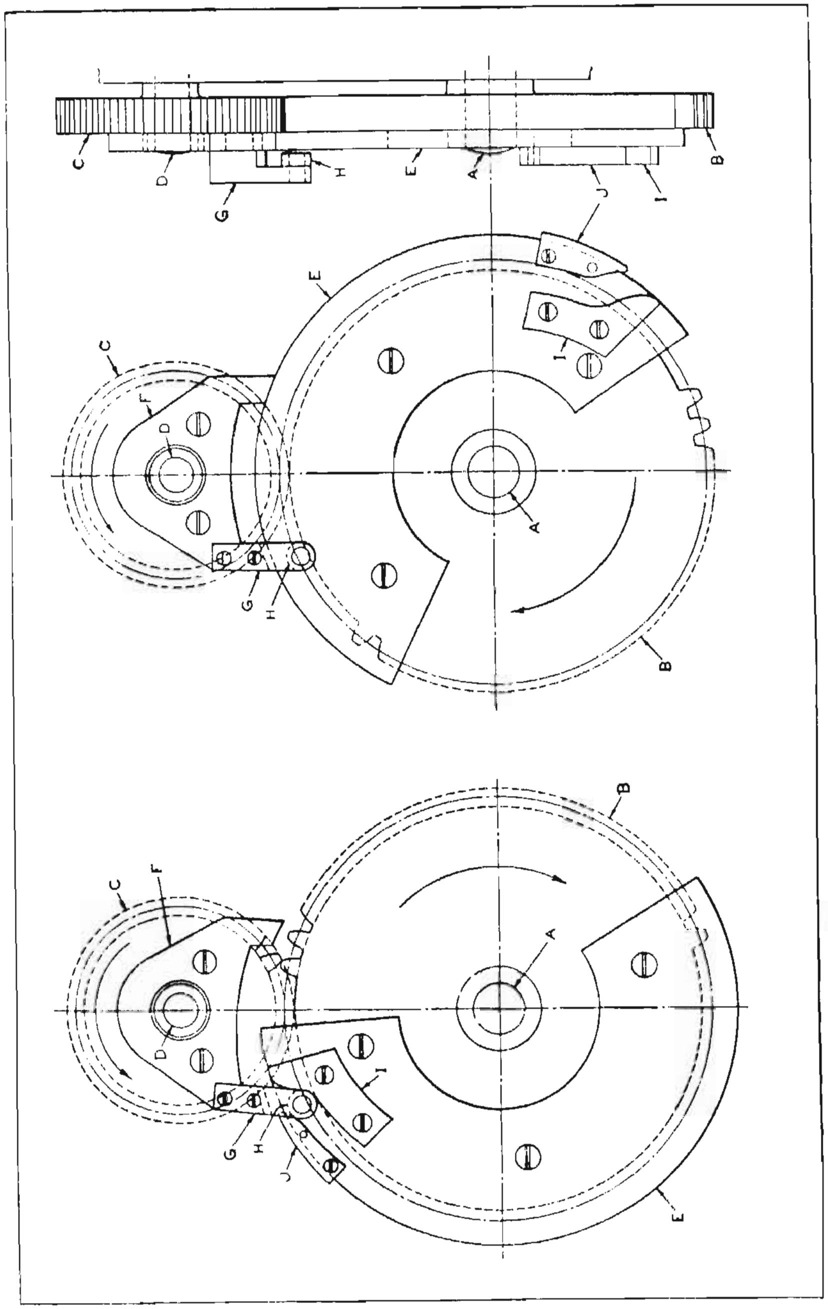


Fig. 19. (Left) Intermittent Rotating Mechanism with Driven Gear C about to Begin Dwell; (Center) Mechanism Shown with Driven Gear C Locked against Rotation; (Right) End View of Mechanism.

plate *E*, which is somewhat larger in diameter than gear *B* and is so located that it covers approximately the section of gear *B* from which the teeth have been removed.

Gear *C* carries, on plate *F*, two extensions which are shaped to an arc, the radius of which is a few thousandths of an inch greater than the radius of plate *E*. The number of teeth in gear *B* is sufficient to produce a complete revolution of gear *C*. As gear *C* completes its rotation, the projecting ends of plate *F* come in contact with plate *E* on gear *B*, locking gear *C* against accidental rotation during the rest period. This arrangement constituted the original mechanism.

The objection to this design was that the extension on the left side of plate *F* struck plate *E* a heavy blow as gear *C* and the parts carried on shaft *D* were brought to rest. In order to eliminate the shock incident to the sudden stopping of gear *C*, a tooth was removed from the trailing end of gear *B*. Guide plates *I* and *J* were attached to plate *E* and a roller *H*, carried on arm *G*, was attached to plate *F* in position to follow the cam path between plates *I* and *J*.

The last tooth of gear *B* is shown in the view at the left of Fig. 19 just about to terminate its contact with its mating tooth in gear *C*. Roller *H* has traveled downward into the groove formed between plates *I* and *J*. Up to this point, roller *H* travels in the path and at the speed determined by the rotation of gear *C*, the upper portion of the groove between plates *I* and *J* being shaped to conform to this path.

As the last tooth of the tooth section of gear *B* passes out of contact with its mating tooth in gear *C*, the latter gear no longer receives driving motion from gear *B*. The cam groove formed by plates *I* and *J* is so shaped that continued rotation of gear *B* draws roller *H* toward the center of gear *B*, causing the rotation of gear *C* to be continued in its original direction, but at a much slower rate. As roller *H* reaches the end of the groove, the leading end of plate *E* comes in contact with the right-hand foot of plate *F*, locking gear *C* against accidental rotation. The reduced speed of gear *C* toward the end of its period of rotation serves to eliminate the objectionable hammering effect.

As shown in the center diagram of Fig. 19, the leading end of plate *F* is riding on the periphery of plate *E*, thus locking gear *C* and spindle *D* against rotation.

A side view of the assembled mechanism is shown at the right of Fig. 19.

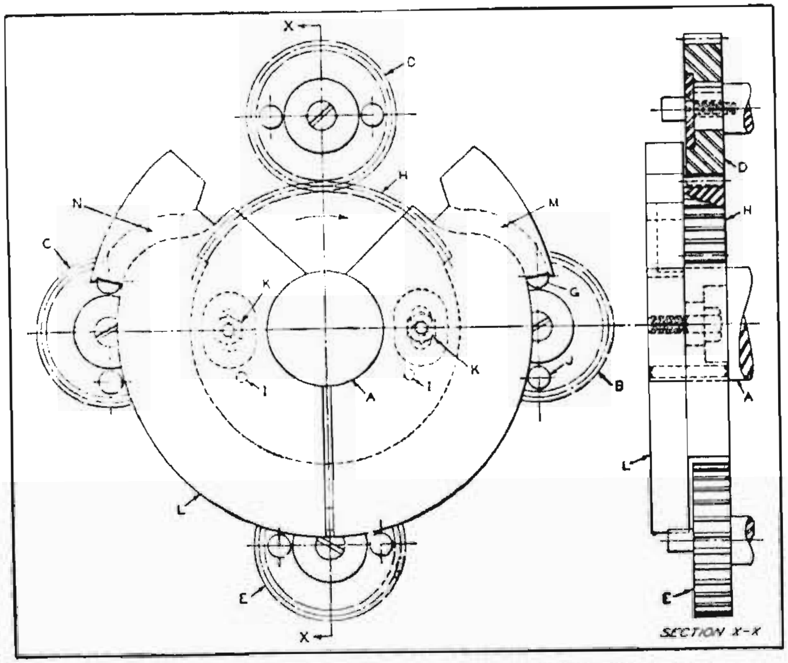


Fig. 20. Mechanism by Means of which Shaft A Intermittently Drives and Locks the Four Gears B, E, C, and D.

Mechanism for Intermittently Rotating and Locking Four Equally Spaced Shafts.—The shaft *A* of the mechanism shown in Fig. 20 transmits 1/3 horsepower at a speed of 180 revolutions per minute, and can be run at

this speed in either a clockwise or a counter-clockwise direction. In turning in a clockwise direction, as indicated by the arrow, shaft *A* rotates the gears *B*, *E*, *C*, and *D* intermittently and successively in a counter-clockwise direction. The drive to these gears is obtained through the combined or successive action of cam groove *M*, segment gear *H*, and cam groove *N*. The segment gear is keyed to shaft *A* and the cam members are adjustably fastened to the segment gear by studs *K*, dowel-pins *I* being used to maintain the permanent settings of the cams after they have been adjusted to the required position.

Starting with the various members of the mechanism in the position shown, one-half turn of shaft *A* will rotate gear *B* one revolution. Gear *B* is then locked in a fixed position until shaft *A* makes one complete revolution. As the four gears *B*, *E*, *C*, and *D* are spaced 90 degrees apart, the driving movement imparted successively to each of these gears begins when the preceding gear has made only one-half of a revolution. For example, when gear *B* has been given one-half turn, the gear *E* begins to turn. Similarly, when *E* has made one-half revolution, gear *C* begins to revolve.

A feature of the mechanism is its silent operation, which is obtained by employing the cam grooves *M* and *N* for respectively starting and stopping the driving movements imparted to the four gears, the segment gear *H* providing the intermediate driving movement.

In operation, the cam groove *M* engages the pin *G* and thus drives the gear *B*, accelerating its speed until the speeds of the two members are the same. This engagement of the members is accomplished without shock. The segment gear and the pinion *B* are thus engaged without shock when their pitch line velocities are the same. The segment gear continues to drive gear *B* until pin *J* enters the second cam groove *N*, the gear remaining in contact with the teeth of the gear segment until the cam groove has fully assumed its driving function. The cam groove *N* is designed to impart a decelerating movement to gear *B* until it has stopped. The concentric edge or rim *L* of the cam member now being in contact with the two pins *G* and *J* serves to lock gear *B* in a fixed position and hold it thus until cam groove *M* again engages the pin *G*. Each of the gears *E*, *C*, and *D* is successively rotated one revolution and then locked in a fixed position, the same as gear *B*.

This mechanism can, of course, be changed so that the intermittent drive will be imparted to one, two or three of the gears as desired. Also, it can be so modified that the gear or gears will be driven only by the cam groove. The contour of the cam groove *M*, for example, can be changed so that it will impart either accelerating or decelerating movements to the gear, and can also be made to include a dwell. The speed of shaft *A* can also be increased, and its rotation can be reversed.

Irregular Intermittent Motion Using Friction Drive.—The mechanism shown in Fig. 21 is designed to transmit an irregular intermittent rotating motion to shaft *J* from the driving shaft *A*, which rotates continuously at a constant speed. It is used on a machine that fabricates a wire screening material having a mesh of alternately increasing and decreasing size. The variations in the size of the mesh are controlled by the mechanism that varies the number of revolutions or fraction of a revolution that is made by the driven shaft *J* between the periods of dwell.

Referring to the illustration, the driving shaft *A* carries the disk *B*, which is keyed to it, and rotates at a uniform speed in the direction indicated by the arrow. Gear *I* is supported freely on shaft *A*, and carries a series of pins *G* which control the various steps in the operating cycle. Gear *I* also carries two studs *E*, which serve to connect it to the two-piece band *D* which encircles the disk *B*.

Band *D* is provided with a lining of frictional material *C*, and is held in place by two clamping bolts which can be adjusted to regulate the frictional driving force transmitted by disk *B*.

Gear *L* meshes with gear *I* and serves to rotate the shaft *J* which operates the feed mechanism of the machine. The latch *H*, which is supported freely on a stationary part of the machine, successively makes contact with the pins *G* on gear *I* as they arrive at the top position. Disk *B* carries the arm *F*, which actuates latch *H* through contact with pin *K*.

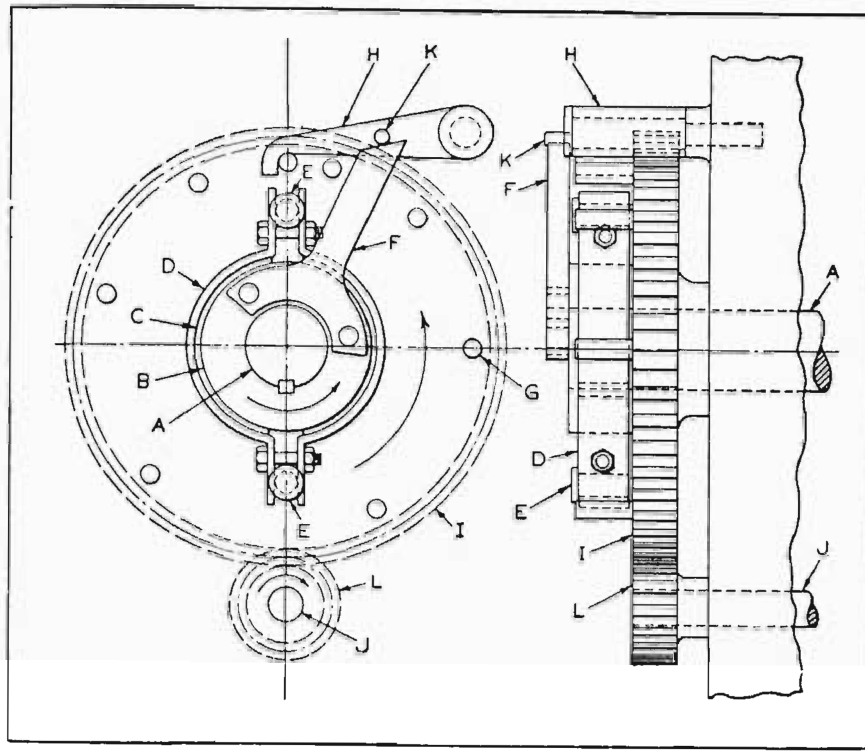


Fig. 21. Mechanism Designed to Transmit Intermittent Rotary Motion to Shaft J from Shaft A with Varying Length of Rotating Periods between Eight Successive Dwell Periods.

In operation, shaft *A*, disk *B*, and arm *F* rotate uniformly in the direction indicated by the arrow. The frictional driving force exerted on band D by disk B tends to rotate gear I through the connecting studs E. When the various members of the mechanism are in the position shown in the illustration, with the hook end of latch H in contact with one of the pins G, the gear I is restrained from rotating, and therefore no rotating motion is transmitted to shaft J, the disk B rotating against the frictional resistance of the band D.

Referring to the illustration, which shows arm F in contact with pin K on latch H, it will be evident that continued rotation of arm F will cause latch H to be lifted out of contact with pin G, permitting the friction drive from disk B to band D to transmit motion to gear I and shaft J through the medium of gear L.

When arm F has passed under pin K, latch H drops ahead of the next pin G, again stopping the rotation of gear I. Gear I remains stationary until arm F has completed a revolution, when it again lifts latch H through pin K. As the pins G are unequally spaced, shaft J will be rotated through a varying number of revolutions or through different fractions of a complete revolution between each of the dwell periods, as determined by the location of these pins.

Quick-Acting Intermittent Feeding Mechanism.—The feeding mechanism shown in Figs. 22 and 23 was designed for winding paper intermittently on an automatic machine. In this mechanism, advantage is taken of the toggle joint locking principle for quickly applying and securely holding a split nut in contact with the lead-screw.

The design of the machine necessitated placing the camshaft A, Fig 22, at a considerable distance from the lead-screw B. The two halves C and D of the split nut are shown in Fig. 22 disengaged from the lead-screw, so that the mechanism to which they are attached will not be moved transversely by the lead-screw. In Fig. 23, the

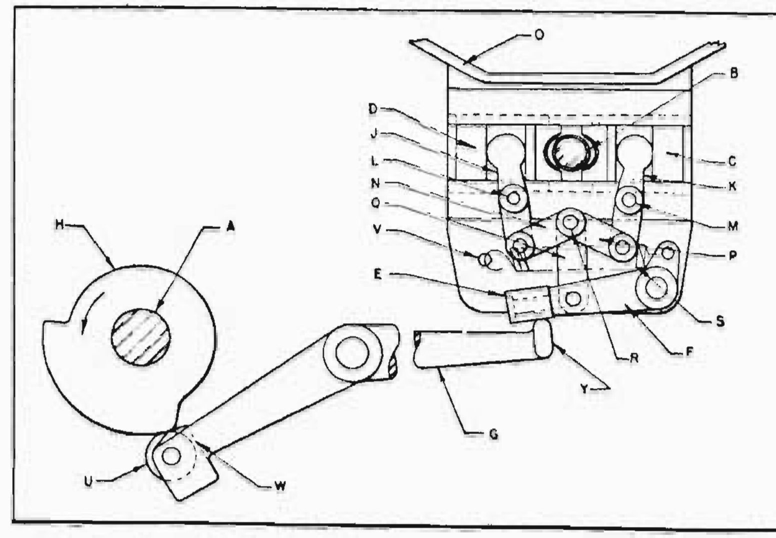


Fig. 22. Cam H, Bellcrank G, Rocker Arm F, and a Toggle Joint Provide Intermittent Feed by Periodically Engaging and Disengaging Lead-screw B and Split Nut C and D.

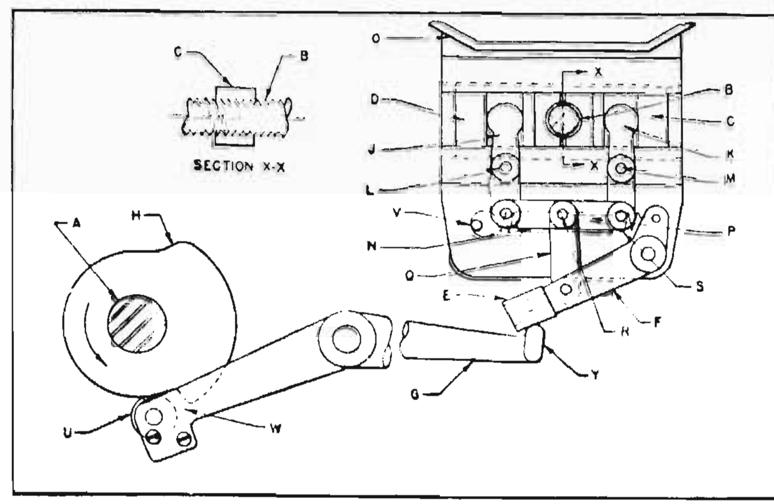


Fig. 23. Closed or Feeding Position of Mechanism Shown in Fig. 22. In this View, the Split Nut is in Engagement with the Lead-screw to Feed the Mechanism.

split nut is engaged with the lead-screw, and the mechanism carried on bracket O has been fed about 2 inches. Intermittent feed is accomplished by bringing together and separating the two halves of the nut at predetermined intervals in the operating cycle of the machine.

The upper ends of arms J and K engage slots in the sides of the split-nut halves. These arms pivot about studs L and M, and their lower ends are joined to a stud R by links N and P. Link Q connects this toggle joint to rocker arm F.

Roller E, at the lower end of the rocker arm, engages extension pad Y on the longer arm of bellcrank G. The bellcrank is pivoted by roller U or block W coming in contact with cam H, which rotates with shaft A in the direction indicated by the arrow.

The function of block W is to prolong the period during which the lever holds the split nut in its open position after the roller leaves the rise on the cam. The shape of the block provides a fast drop-off at the end of the "open" travel, as indicated in Fig. 23, and permits the split nut to snap quickly into engagement with the lead-screw. Spring S, which is attached to the upper end of rocker arm F and is hooked over pin V, exerts sufficient tension to hold the roller and block in contact with the cam, and also locks the toggle joint in either the open or closed position.

Rack and Gear Assembly for Intermittent Rotary Motion.—In the design of a wire-forming machine, a shaft was required with an intermittent rotary motion that exceeded the radial travel obtainable with ordinary ratchet and pawl mechanisms. The rack and gear assembly shown in Fig. 24 provided the desired motion efficiently. The reciprocating rack A meshes with gear B, to which it transmits an alternating rotative movement. Gear B is in mesh with gear C, which is smaller in diameter than gear B and does not mesh with rack A. Gear C is keyed on shaft D,

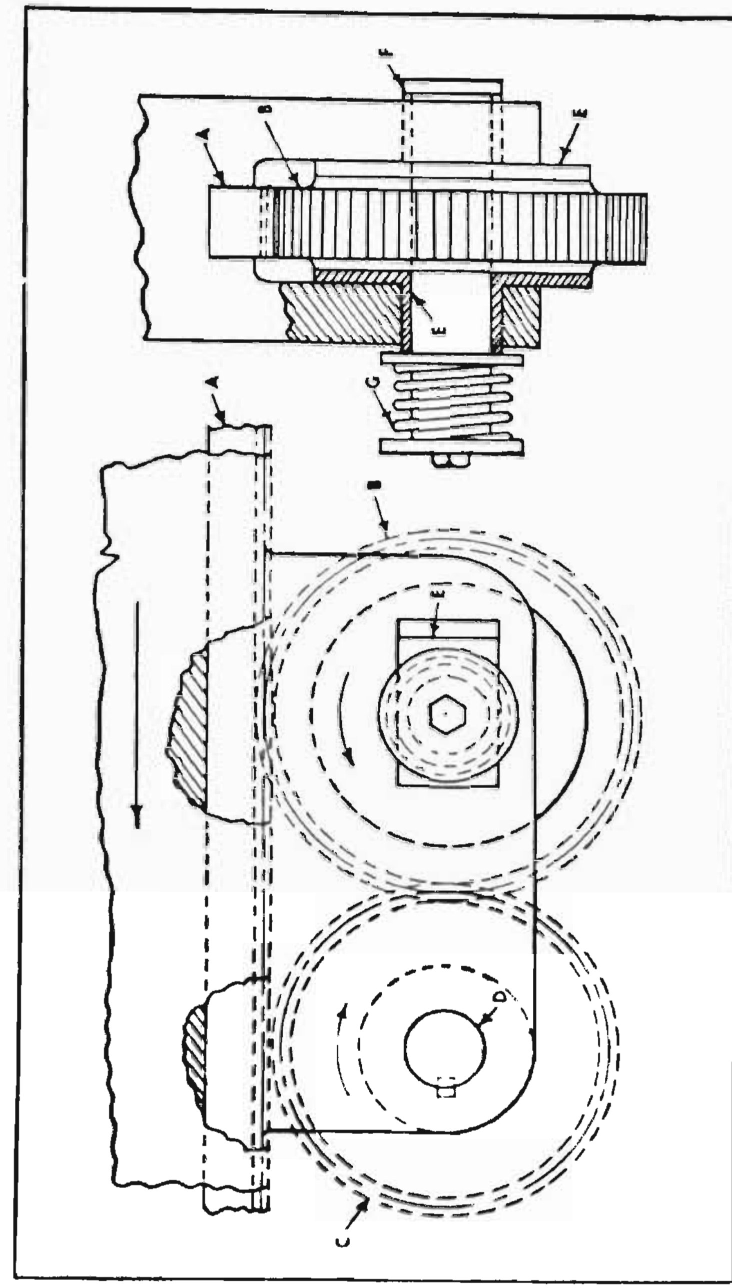


Fig. 24. Rack and Gear Assembly that Provides Intermittent Rotary Motion in One Direction.

which is given an intermittent, rotary motion in one direction.

Driving gear *B* is splined to shaft *F*, which is supported by two flanged bearings *E*. Bearings *E* are rectangular in section where they pass through the supporting member. The rectangular sections of bearings *E* are mounted in rectangular slots, which are somewhat longer than these sections to permit a horizontal sliding movement of the bearing. The hubs of gear *B* are of large diameter and are in contact with the flanges of bearings *E*. Shaft *F* is flanged on one end and is provided with a spring *G* on the opposite end, the pressure of which draws bearings *E* together so as to apply frictional resistance to the rotative movement of gear *B*.

As illustrated, rack *A* is moving in the direction indicated by the arrow, causing gear *B* to rotate in the same direction. Gear *B*, meshing with gear *C*, causes it and shaft *D* to rotate in the reverse direction. When the movement of rack *A* is reversed, the tendency for gear *B* to rotate in the reverse direction also is resisted by the friction applied through spring *G*. Inasmuch as there is no resistance to the horizontal movement of bearings *E*, the latter will slide in the rectangular slots, thus disengaging gear *B* from gear *C*. When bearings *E* come in contact with the ends of the slots, further sliding movement is prevented, and continued movement of rack *A* causes gear *B* to rotate; however, as gears *B* and *C* are out of mesh, no rotary motion is transmitted to gear *C*. When the movement of rack *A* is once more reversed (being then in the direction indicated by the arrow), bearings *E* immediately slide gear *B* into mesh with gear *C* and shaft *D* is rotated.

An idler between gears *B* and *C* will permit rotation of shaft *D* in the opposite direction to that illustrated.

Intermittent Motion for Changing Timing Interval for Air-Valve Functioning.—Fig. 25 shows how a shaft *D*, which originally served to actuate an air valve once for

each revolution of the chain-driven sprocket *A*, was equipped with an intermittent indexing mechanism designed to actuate the valve once every sixth revolution of the driven sprocket. Since the slow constant speed at which sprocket *A* rotated could not be changed, it was necessary to provide some means of driving shaft *D* from sprocket *A* at a reduced speed in the ratio of 6 to 1 to accomplish this change in the valve operating cycle.

Fortunately, the complete operation of opening and closing the air valve (not shown) could be accomplished in one-eighth of a revolution of shaft *D*. This made it possible to employ the cam-actuated intermittent mechanism shown, which indexes shaft *D* one-sixth of a revolution at each

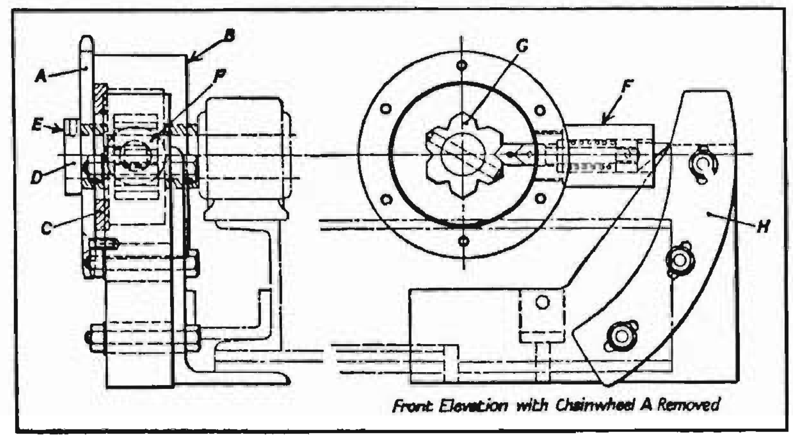


Fig. 25. Intermittent Indexing Mechanism Designed to Actuate a Valve Once Every Sixth Revolution of the Driven Sprocket.

revolution of sprocket *A*. With this mechanism, every sixth indexing movement of shaft *D* through one-sixth of a revolution served to open and close the air valve as required.

The necessary modifications in the drive included the securing of sprocket *A* to the idling drum *B*, which is a free running fit on shaft *D*, and the provision of a collar *E* for retaining the drum and sprocket assembly on shaft

D. The sprocket wheel is located on drum *B* by the machined ring *C*, and is held in place by countersunk-head screws. A steel housing *F*, attached to drum *B*, contains a spring-loaded pawl-ended follower. The follower, as shown by dotted lines in the view to the right, is made in two pieces, the pawl end being a square section bar with a V-shaped end. The rear end of this pawl is turned down to form a shank which is a sliding clearance fit inside the loading spring. The end of this shank is threaded to fit the threaded hole in the follower shown in contact with the cam-plate *H*.

Secured to shaft *D* is a hardened tool-steel hexagonal member *G*, which has a V-slot centrally located in each of its six flat faces. Normally, the loading spring holds the pawl out of contact with member *G*, but when the follower end comes in contact with cam *H*, the pawl end is forced into one of the slots in member *G*, which causes shaft *D* to rotate with sprocket *A*. The length of the indexing movement is determined by cam *H*, which keeps the pawl in engagement with member *G* through one-sixth of a revolution and then permits the pawl to withdraw, so that member *G* remains idle for the remaining five-sixths of a revolution.

Operation of the air valve is, of course, accomplished during one of the six indexing movements of member *G*, the other five indexing movements performing no valve-operating function. Cam *H* is provided with radial slots for the clamping screws to permit adjustment of the cam so as to insure correct timing of the engagement of the pawl and member *G*.

CHAPTER 3

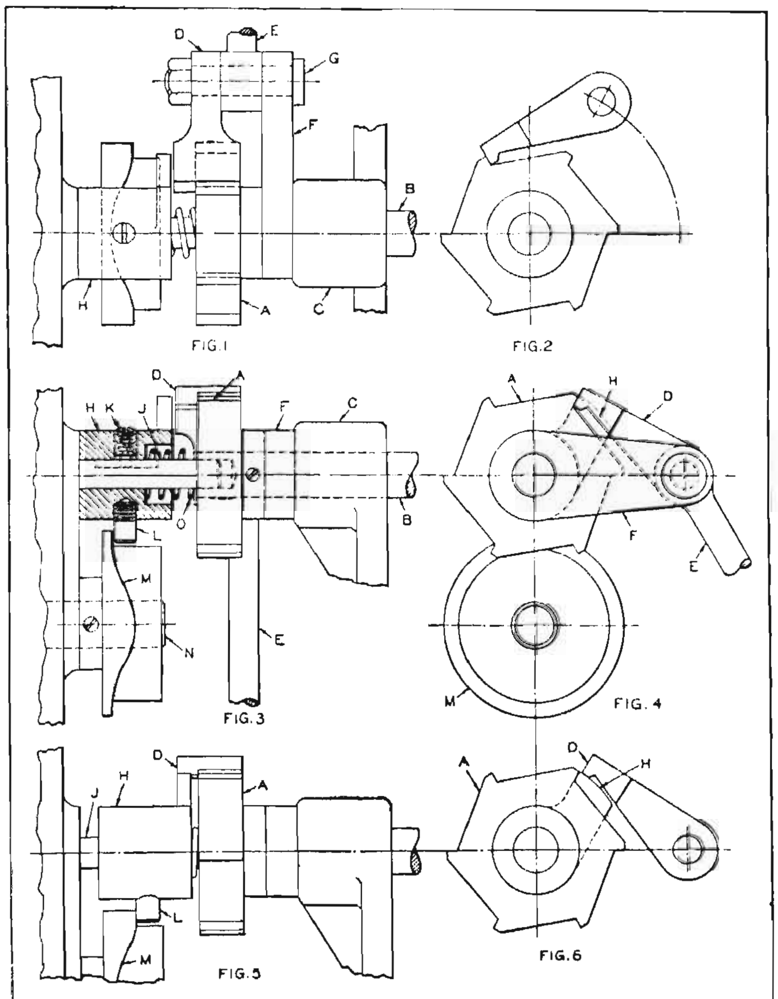
Intermittent Motions from Ratchet and Geneva Mechanisms

Two methods of producing intermittent motion in which the periods of rest are evenly spaced and of equal length are by means of ratchet gearing and by using some modification of the Geneva motion. In its basic form this motion is obtained by means of a Geneva wheel, acting as a driven member, which has four radial slots located 90 degrees apart that successively engage a roller or pin on the driving member. The Geneva wheel thus turns with the driving member through one-quarter of a revolution and is idle for the remainder of the revolution of the driving member.

A number of ingenious mechanisms in which a ratchet arrangement or a Geneva motion play a prominent part are described in this chapter. For other mechanisms of a similar type, the reader is referred to Volumes I and II of "Ingenious Mechanisms."

Ratchet Mechanism with Device for Controlling Engagement of Pawl.—A rather novel method of controlling the action of a pawl on a ratchet is incorporated in the ratchet mechanism shown in Figs. 1 to 6. In the particular application for which this mechanism was developed, the ratchet is required to operate at a slow and uniform rate, and at periodic intervals to skip one or more movements.

The ratchet wheel *A*, Fig. 1, is mounted on a shaft *B*, which, in turn, rests in a bearing *C*. The pawl *D* forms a part of the operating unit, which consists of the actuating rod *E*, connected to the upper mechanism, and the bar *F*, which serves to keep the ratchet and pawl in the same relative positions throughout their movements. A stud *G* holds the three members of the operating unit together. The pawl control unit consists of a displacing collar H, Fig. 3, which has a sliding fit on the pin J. The key K slides in a keyway in the pin J and serves to keep the member H in a fixed position relative to ratchet A. A cam follower L makes contact with the surface of the cam



Figs. 1 to 6. Diagrams Showing Ratchet Mechanism with Cam Arrangement for Disengaging Pawl.

M, which is fastened to a shaft N. Spring O tends to keep the displacing collar H away from ratchet A and at all times under the action of cam M. To properly align the ratchet mechanism and displacing collar, the pin J is extended into a hole in the end of shaft B.

In Fig. 4, the pawl is shown in position ready to rotate the ratchet through a distance equivalent to one tooth space. Fig. 2 shows how the pawl travels through an arc determined by the the length of bar F. In Figs. 1 and 2, the position of member H is as shown in Fig. 3, where it will be noted that the cam follower L is at the lowest point of the cam surface and the displacing collar is away from the ratchet. Fig. 5 shows the cam rotated to the position where follower L has caused member H to be moved forward toward the ratchet. The result is shown in Fig. 6, where it will be seen that the pawl has been raised, to prevent it from coming in contact with the next tooth, thus interrupting the ratchet movement.

The cam M can be arranged to provide any form of interrupted ratchet motion desired. It can be arranged to rotate continuously or intermittently, depending upon the nature of the application. The cam action is so timed that member H is moved forward into position to prevent the pawl from engaging the ratchet wheel at the moment the pawl is in the position shown in Fig. 2.

Ratchet Movement with Idle Period.—A ratchet movement operated by an oscillating lever in the conventional manner, except that the pawl is rendered inactive at a predetermined period, is shown in Fig. 7. This movement is used to operate a conveyor belt on a wire-forming machine, the purpose of the idle period being to increase the loading time at a certain point in the cycle.

Lever B is free to oscillate on shaft A. Ratchet wheel C is keyed to shaft A and carries on its hub a similar but narrower ratchet wheel D. The latter wheel is free to turn on the hub of wheel C. Pawl F, which transmits the motion of lever *B* to ratchet wheel *C*, is sufficiently wide on the working end to engage both ratchet wheels *C* and *D*. Ratchet wheel *D* carries the single-toothed dog *E*.

In operation, ratchet wheels *C* and *D* are given a partial revolution through the engagement of pawl *F* with their teeth, as the lever *B* swings to the left. On the return stroke, lever *B* swings to the right and pawl *F* rides over the teeth of the ratchet wheels, which remain stationary.

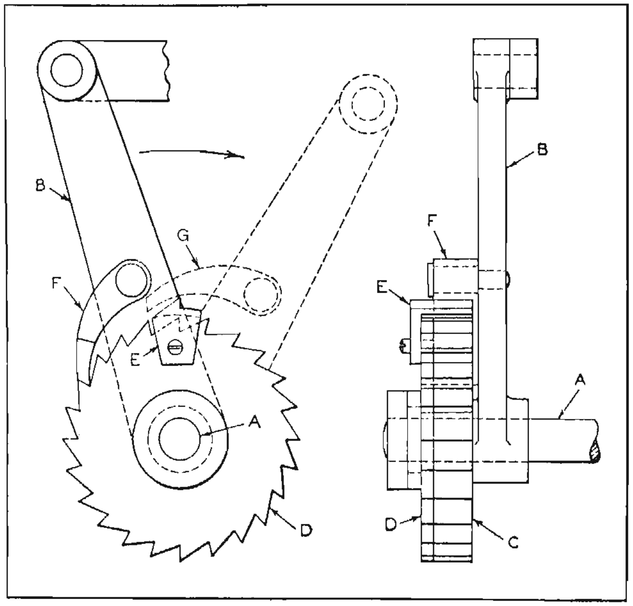


Fig. 7. Ratchet Mechanism with Intermittent Idle Period.

The illustration shows lever *B* at the end of its forward stroke and about to swing to the right in the direction indicated by the arrow.

Toward the end of the return stroke of lever *B*, pawl *F* is lifted out of contact with the ratchet teeth by dog *E*, as indicated by the dotted lines at *G*. As lever *B* reaches the end of its return stroke, pawl *F* drops behind the tooth of dog *E*, but is still held out of contact with the ratchet.

Then when lever *B* swings to the left, pawl *F* engages the tooth of dog *E*, giving the ratchet wheel *D* a partial revolution. As there is no connection between ratchet wheels *C* and *D*, wheel *C* remains stationary during this part of the cycle. On the succeeding forward strokes of lever *B*, pawl *F* again actuates the ratchet wheels *C* and *D* until dog *E* is once more brought under pawl *F*.

The movement, as shown, is designed to provide one idle period in six motions of lever *B*. Other cycles may be provided for by using more than one dog, but it is necessary that the angular movement of lever *B* be evenly divisible into the 360 degrees of a circle; otherwise it will not be possible to maintain a definite timing cycle. The rest period can be eliminated at will by merely setting ratchet wheel *D* so that pawl *F* will pass over dog *E* on the return stroke of lever *B*, thus allowing pawl *F* to engage the ratchet teeth at all times.

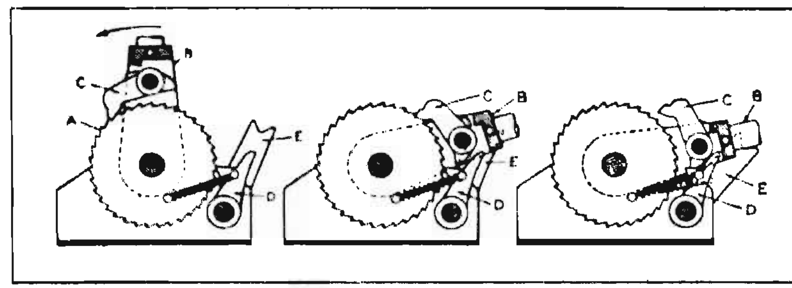


Fig. 8. Lever-operated Ratchet Mechanism with Special Pawl Arrangement.

Ratchet Mechanism with Special Pawl Arrangement.—A ratchet mechanism with rather an unusual pawl arrangement is shown with the operating lever in three different positions in Fig. 8. This ratchet mechanism is used to rotate a rope drum employed to provide the tension required for joining conveyor bands or belts. It can also be utilized as a conveying device for heavy machines and for other purposes.

The ratchet wheel A is directly connected with the rope pulley or drum. The operating lever B supports a ratchet pawl C which produces only a forward motion of the ratchet wheel. Any reverse motion is prevented by another pawl D, fixed in the base of the mechanism and pressed against the ratchet teeth by a spring. Operating lever B causes rotation of the wheel and the rope drum.

When the winding work is completed, the hand-lever is brought into the position shown in the middle diagram, where it is stopped by a pin which comes in contact with lever E. Pawl D remains in mesh with the ratchet teeth, so that any unintentional motion of the wheel is prevented should pawl C be accidentally disengaged. If it is desired to withdraw the rope, the hand-lever is placed in such a position that the fixed pin lies in front of lever E and causes it to tilt, as shown in the right-hand diagram. In this position, the cam-like surfaces of the pawls disengage the pawls from contact with the ratchet teeth, permitting the rope drum to rotate freely.

Ratchet Movement with Remote Control.—A reversing ratchet movement in which the operating pawl is tripped from a distant point is shown in Fig. 9. This movement is used to control the work-table of a metal polishing machine. Referring to the illustration, M is the work-table carrying the rack B which meshes with gear A. Gear A is free to turn on shaft J, which is supported on bearings (not shown). Shaft J carries the levers F and C at opposite ends, both levers being keyed to the shaft. Rod D transmits an oscillating motion to lever C.

Lever F carries pawl G and bar H, both of which are keyed to shaft P which passes through lever F. Shaft P is a free turning fit in lever F. Shaft K passes through shaft J carrying levers E and L at opposite ends. Lever E carries a plunger N, backed by a spring, which makes contact with bar H, thus engaging pawl G with gear A.

Referring to the view at the left, rod D is assumed to

be moving in the direction indicated by the arrow, the motion being transmitted through levers C and F, pawl G, and gear A to rack B, so that table M moves in the direction shown by the arrow. As lever E rests against the pin in lever F, motion is transmitted to lever L through shaft K. On the return stroke of rod D, gear A remains stationary, the pawl G riding back over the teeth.

As the movement of table M continues, pin O eventually strikes lever L, giving shaft K a partial revolution within shaft J, so that lever E is brought against the upper pin on lever F. This causes plunger N to act on the opposite end of bar H, swinging pawl G so that its lower end engages

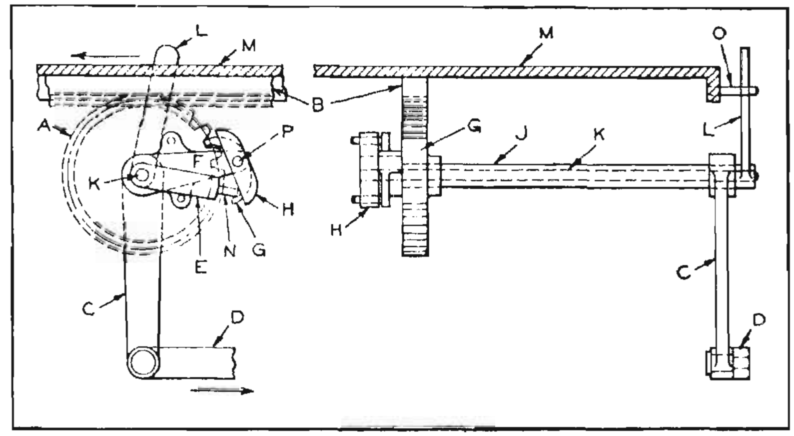


Fig. 9. Reversing Ratchet Movement which is Tripped by Contact of Pin O on Table M with Lever L.

gear A. In this manner, gear A is given a partial revolution on the forward stroke of rod D instead of on the pulling stroke as shown, so that the motion of table M is in the reverse direction. This continues until a pin at the opposite end of table M strikes lever L, again tripping pawl G and repeating the cycle.

Noiseless Ratchet Mechanism for Preventing Reversal of Shaft.—The ratchet mechanism shown in Fig. 10 was designed for noiseless or silent operation, and was originally intended for use on the head-shaft drive of belt conveyors, where a peripheral speed of 100 or more feet per minute is attained. However, it could easily be adapted to other uses where it is desirable to prevent a shaft from turning backward.

The mechanism is very simple, centrifugal force being utilized to keep pawls *A* from contact with the ratchet teeth *B* while the rotating member *C* is in motion. Member *C* is keyed to the shaft *K*. The instant *C* stops rotating, one of the three pawls *A* engages teeth *B*, preventing shaft *K* from rotating backward. When forward motion of the shaft is resumed, the pawl is instantly thrown out of contact with the ratchet teeth, the outward motion being restricted by stop-pins *S*.

The rotating member is composed of two indentical plates

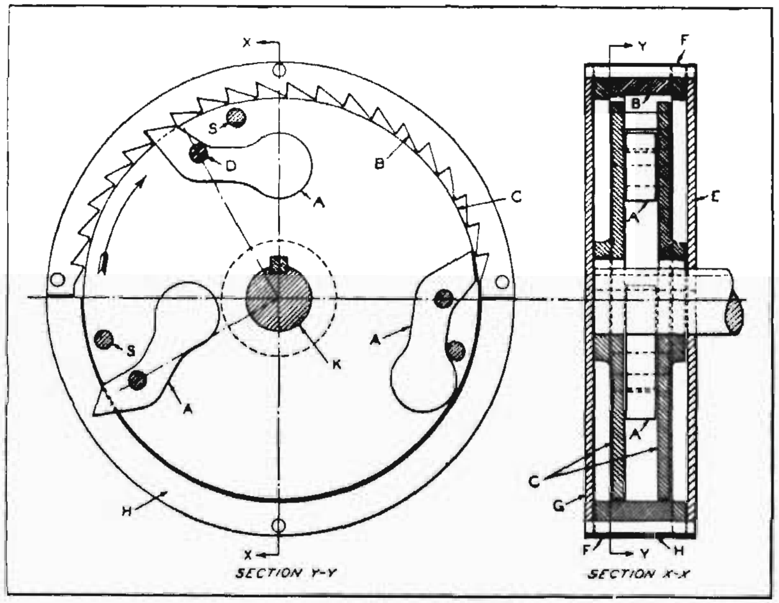


Fig. 10. Silent-operating Ratchet Designed to Prevent Shaft from Rotating in Reverse Direction.

C, as shown in the cross-sectional view to the right. The three pawls *A* are suspended between plates *C* in such a way that they are free to swing on their pivot pins *D* while the member *C* rotates in a forward direction. The ratchet teeth *B* are cut in a 180-degree segment mounted in the upper half of a dust-tight housing, the segment being held fixed between plates *E*, *F*, and *G*. The inner plate *E* is attached to the frame of the mechanism. The part *H* occupies the same relative position between the plates in the lower half of the housing as that occupied by the ratchet-tooth segment in the upper half.

Ratchet Mechanism that Converts Reciprocating Movement to Continuous Rotary Motion.—In designing a certain mechanism, the problem arose of providing a rotary drive for a shaft when the only available motion was reciprocation in a plane at right angles to the axis of the shaft. It was required that the rotation be continuous in one direction, but it did not need to be absolutely uniform. The problem was solved by the ratchet mechanism shown in Fig. 11.

In the illustration, the shaft to be rotated is shown at *A*. It is supported in suitable bearings (not shown). Ratchet *B* is keyed to the shaft *A*. On each side of the ratchet and turning freely on the shaft are pawl arms *C* and *D*. These arms are held in place by collars *E* which are pinned to shaft *A*. At the outer end of arms *C* and *D* are pins *F* and *G*, about which pawls *H* and *J* are free to swivel. These pawls are held in contact with the teeth of ratchet *B* by springs *K*. The latter are attached to the hubs of the pawl arms and bear against spring pins *L* mounted on pawls *H* and *J*.

The reciprocating member *M* is connected by links *N* and *P* to the outer ends of the pawl arm pins *F* and *G*. One link is above ratchet *B*, and the other link below the ratchet.

As reciprocating member *M* moves toward the right, the ratchet is rotated counter-clockwise by the pawl *J* engaging a ratchet tooth. During this movement, pawl *H* rides over the ratchet teeth. When member *M* moves to the left, pawl *H* engages a ratchet tooth and continues the rotation of both the ratchet and shaft *A* in a counter-clockwise direction. During this movement, pawl *J* slips over the ratchet teeth.

The pawls are beveled at their outer ends on the side adjacent to each other, as shown by pawl *H* in the plan view, so that the two pawls can pass each other without interference when they are at the extreme right-hand end

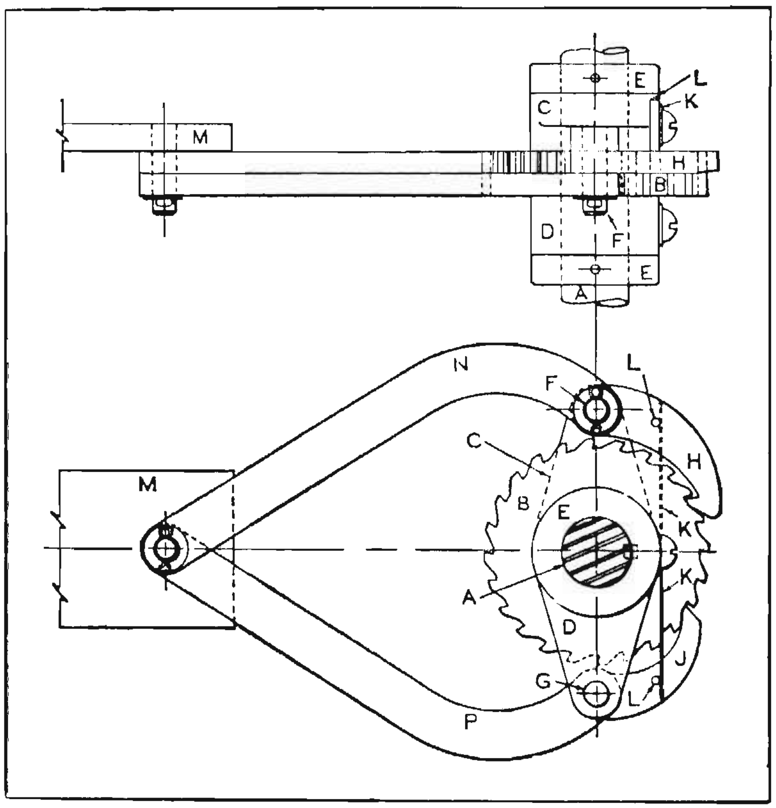


Fig. 11. Ratchet Mechanism which Provides a Continuous Rotary Movement that is Derived from a Reciprocating Motion.

of their travel. Likewise, links *N* and *P* are curved, so that they will readily clear ratchet *B* when member *M* is at the extreme right-hand end of its movement. A fly-wheel (not shown) promotes uniformity of motion of the driven shaft.

Lever-Operated Self-Locking Indexing Mechanism.– One continuous motion of lever *C* of the mechanism shown in Fig. 12 serves to index shaft *A* through angle *T* and lock it in position. This mechanism is employed on a hand miller in making two saw cuts in a steel arm. There are various sizes of arms and the angle between the cuts varies with the size of the arm. Provision is made for adjusting the indexing plates *S* and *R* to suit any required indexing angle *T*.

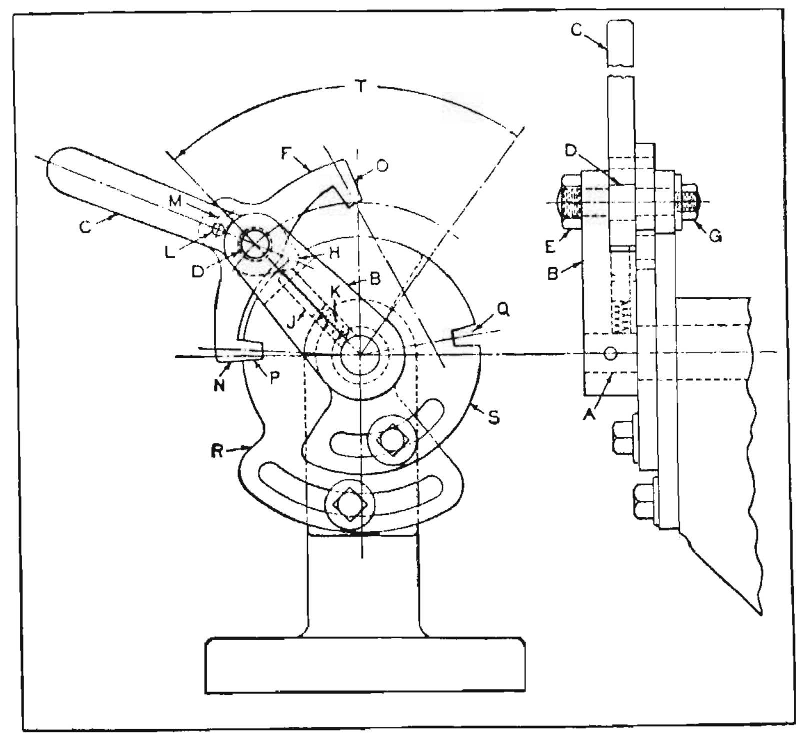


Fig. 12. Two-position Indexing Mechanism for Milling Machine.

Shaft A is part of the work-holding cradle of the fixture, which is to be indexed and locked in either of two positions. Arm B is pinned to shaft A. The stud D is held in arm B by nut E. The operating lever C and pawl F are free to swing on stud D, being retained on the stud by a washer and nut G.

The end H of lever C terminates in a point formed to a small radius. This pointed end engages a pointed plunger J which slides in a hole in the arm B and is forced against the pointed end of lever C by helical spring K. The action of the spring plunger against the operating lever C causes the lever to turn away from the plunger when the lever is swung to either side.

A pin L driven into a hole in lever C slides in a slot M in the pawl F. The length of the slot is such that the lever can be swung until its point snaps over the plunger before the pin reaches the end of the slot and turns the pawl F. The pawl projections N and O engage the slots P and Q, respectively, in the index-plates R and S.

With the members of the mechanism in the position shown, the pressure of the plunger J against point H of lever C causes the lever to turn counter-clockwise. The pin L is thus brought against the end of slot M in pawl F, turning the pawl and forcing projection N into slot P in the index-plate.

To be indexed to the other position, the lever C is turned clockwise until its point H snaps over the point of the plunger J and the pin L has moved to the other end of slot M in pawl F. The pressure of the spring plunger J on lever C now holds pin L against the end of slot M. The projection N of the pawl will be withdrawn from slot P and the pawl will be rotated clockwise until its projection O comes in contact with the edge of the index-plate S. Continued rotation of lever C causes arm B and shaft A, with its cradle, to be rotated through angle T, pawl projection O sliding on edge of index-plate S.

When projection O reaches slot Q, the spring plunger forces the pawl projection into the slot, thus completing the indexing operation. The reverse indexing is effected in the same manner by moving the lever counter-clockwise.

Double-Action Reversing Ratchet Movement.—A machine used for polishing a wire product has a traveling table which is given an intermittent motion by means of a rack and pinion actuated by an oscillating lever through a ratchet and pawl. In the original design, the pawl actuated the ratchet during one-half of the oscillating cycle of the lever, the pawl riding over the ratchet teeth on the return stroke in the conventional manner.

In the improved design, shown in Fig. 13, two pawls G and H are employed to rotate the gear A in a clockwise direction on both the forward and reverse strokes of the oscillating lever E. Referring to the illustration, rod T is given a reciprocating motion by means of a crank, thus transmitting an oscillating motion to lever E, which is free on shaft C. Gears A and R are keyed to shaft C, gear R meshing with the rack P, which is carried on the work-table S.

Referring to the view at the left, lever E transmits motion to lever F through the link I. Lever F and gear B are free to rotate on shaft D and gear B meshes with gear A on shaft C. Levers E and F carry the pawls G and H, respectively. Pawls G and H are slotted to receive pins on the ends of rods U and V, which slide in dovetailed grooves in levers K and L. Rod U is drawn upward by a spring, while rod V is drawn downward by a similar spring. Levers K and L are connected by the link J, which carries pin W at its center. Any horizontal movement of pin W causes levers K and L to move in unison. Stops M and N in work-table S serve to trip the pawls at both ends of the work-table travel.

Referring to the view at the left, the rod T, moving in the direction indicated by the arrow, transmits motion

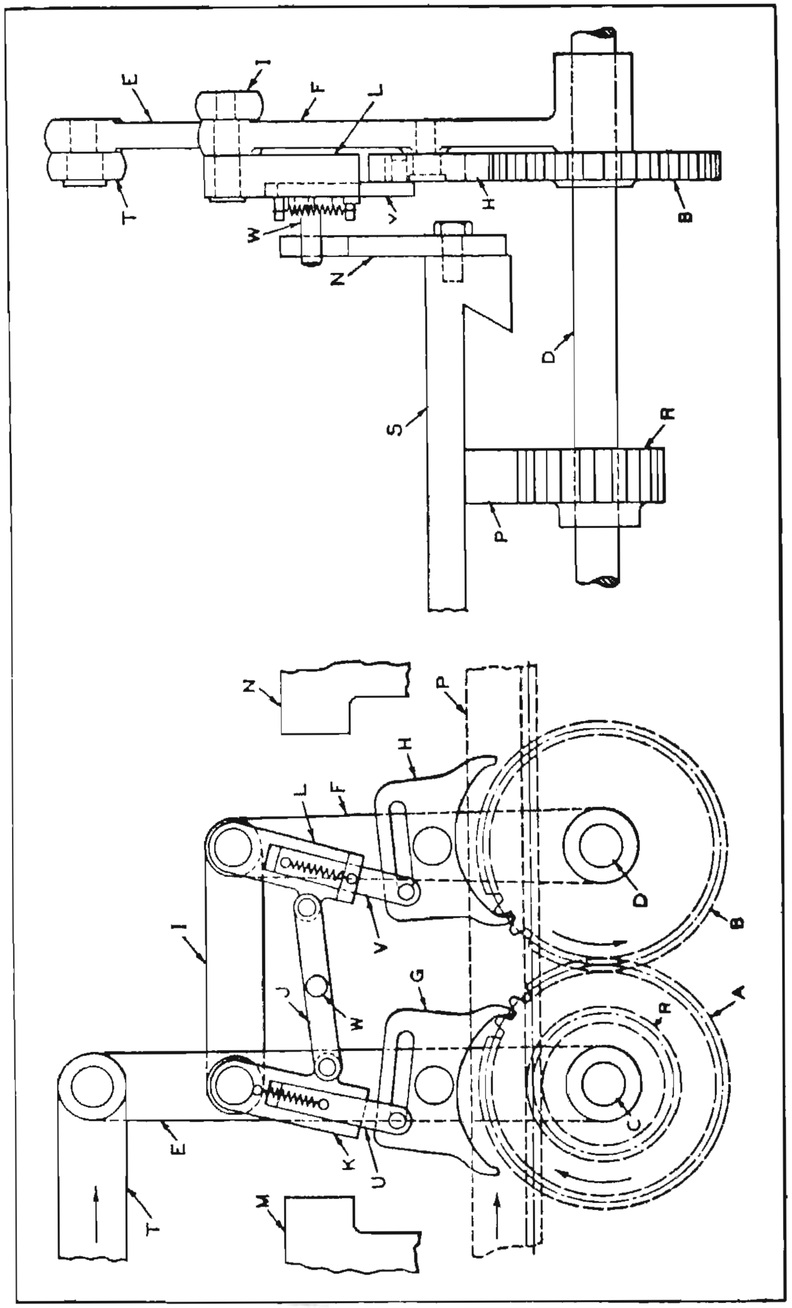


Fig. 13. Reversing Ratchet Mechanism Designed to Rotate Ratchet Wheel on Both Forward and Return Movements of Pawl Lever.

through lever E and pawl G to gear A, giving the latter a partial rotation in the direction indicated by the arrow. As both gears A and R are keyed to shaft C, the motion of gear A is transmitted through gear R to the rack P in the direction indicated. Gear A transmits its motion to gear B in the reverse direction, gear B serving no useful purpose at this time, merely turning under the pawl H.

As rod T reaches the end of its travel and reverses its direction, levers E and F also reverse their direction of travel. At this point, the reverse motion of lever F is transmitted to gear B through pawl H, the motion being continued through gears A and R to the rack P. In this way, both forward and return strokes of rod T are utilized to transmit motion to rack P in the same direction.

As the stops M and N are attached to work-table S, they move with the rack P. In the position shown in the view to the left, each movement of rack P brings the stop M closer to pin W. As stop M strikes pin W, the levers K and L are made to swing to the right, which, in turn, causes the outer ends of pawls G and H to engage the gears A and B. As this causes gears A and B to rotate in the reverse direction, the movement of rack P is reversed. This motion continues until stop N strikes pin W, when the direction of movement is again reversed.

Geneva Wheel Designed for Precise Intermittent Indexing Movements.—A modified type of Geneva wheel that was developed to obtain precise intermittent indexing movements is shown in Fig. 14. This special mechanism, shown diagrammatically at J, was designed to eliminate any backlash or over-running action of the driven member F at the end or beginning of each indexing movement, when the pin A of the driving member E enters or leaves the slot in one of the arms of the driven member F. The relative positions of the driving and driven members at the start and end of the indexing movement are shown in the diagrams J and L, respectively.

With the mechanism designed as shown, the driven shaft *H*, keyed to wheel *F*, is indexed one-fourth revolution for each revolution of the driving shaft *G*, which is keyed to wheel *E*. Each indexing movement starts smoothly as the long driving pin *A* enters one of the slots in wheel *F* and pin *C* passes out of contact with the flange on the rim of wheel *F*.

The driving pin *A*, as shown in section *X—X*, is long enough to make contact with the sides of the slots in wheel *F* for the full depth or thickness of the slotted

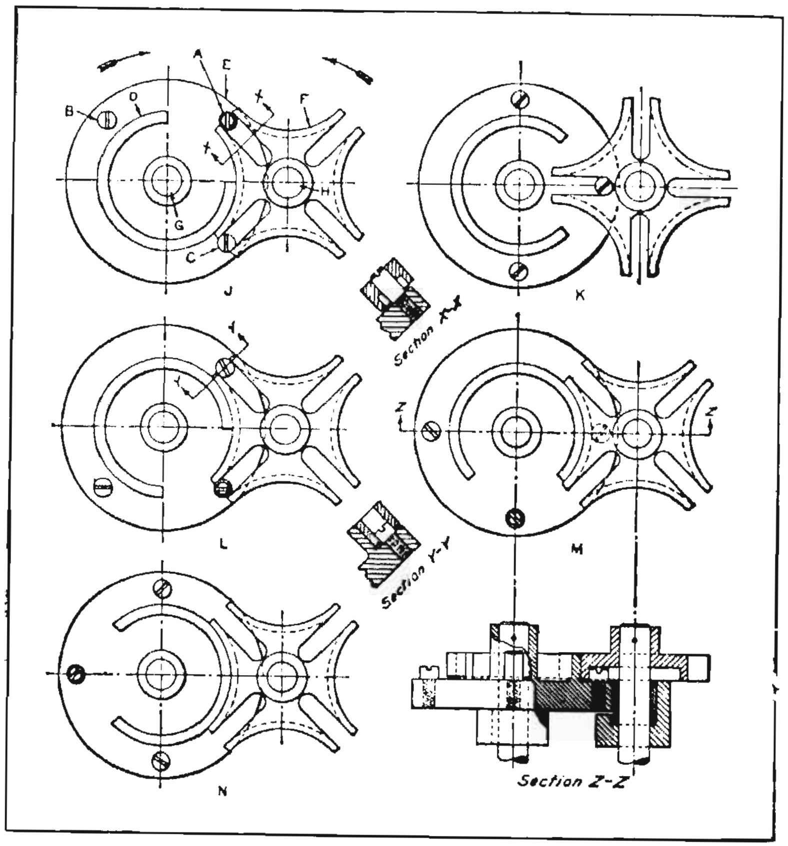


Fig. 14. Diagrams Illustrating the Operation of a Geneva Wheel Designed for Precise Intermittent Indexing Movements.

part of the wheel. The two shorter pins *B* and *C*, one of which is shown in section *Y—Y*, are made to clear the bottom of the recess machined in the under side of wheel *F*, as shown in section *Z—Z*, but are not long enough to contact the straight sides of the slots contacted by pin *A*.

The pins *B* and *C* are so positioned, however, that they make close running contact with the flange formed by the recess on the under side of wheel *F* (see the broken section *Z—Z*). The flange *D*, machined integral with wheel *E* to a close running fit with each of the four segments of wheel *F*, has a 90-degree section cut away opposite the driving pin *A* to provide clearance space for the projecting arms of the driven wheel during the indexing movement.

Referring to the diagram at *J*, pin *A* is just entering a slot in wheel *F*, while pin *C* is passing out of contact with the rim on the under side of wheel *F*. The pin *C*, being a close running fit on the inside of the flange on wheel *F*, and flange *D*, being a close running fit on the outside of the flange on wheel *F*, serve to hold wheel *F* stationary until the indexing actually begins, and also prevent it from further movement the instant the indexing is terminated. During part of the revolution of driving wheel *E*, the flange *D* alone serves to hold the driven wheel stationary in the dwell position, as indicated in diagram *N*.

The accurate positioning of pin *C* also prevents any movement of wheel *F* before pin *A* engages a slot. Thus the indexing movement of wheel *F* is started with a smooth, accelerating motion and stopped as smoothly with a decelerating motion without any over-run or backlash.

Diagram *K* shows the mechanism with the driven wheel *F* rotated through one-half of the first indexing movement, at which point it has reached its maximum speed of rotation.

Diagram *L* shows the driven wheel at the end of the indexing movement, with the short pin *B* of the driving wheel making contact with the flange on the under side of the driven wheel, so that it holds the outer side of the flange in contact with flange *D* of driving wheel *E*. Thus pin *B* prevents any rotational movement of the driven wheel as driving pin *A* leaves the slot in the driven wheel at the end of the indexing movement.

The diagram at *M* shows the driven wheel in the dwell position with pin *B* still in contact with the flange on the under side of the driven wheel. The broken section *Z—Z* shows the short pin *B* clearing the recess in the driven wheel, which is held stationary in the dwell position. Section *Z—Z* is broken, part of the view to the left being shown in full lines, in order to indicate the difference in the heights of the driving pin *A* and the two pins *B* and *C*.

The diagram at *N* shows the driven wheel still in the dwell position, where it remains stationary until rotation of the driving wheel *E* brings pin *C* and driving pin *A* into the positions indicated in diagram *J*. The indexing movement and dwell period described are then repeated, driven wheel *F* being indexed one-fourth turn for each complete revolution of driving wheel *E*.

Geneva Motion Mechanism of Unique Design.—In designing a machine for the automatic stamping of consecutive numbers on the corners of envelopes, it became necessary to develop some interesting mechanisms, among which was the Geneva dial motion shown in Figs. 15 and 16. This unique mechanism performs its intermittent indexing movements at a uniform rotational speed instead of at the accelerating and decelerating speed of the harmonic motion characteristic of the Geneva mechanism of conventional design ordinarily employed for such purposes.

Referring to Fig. 15, the mechanism is driven by a pinion at *A* which meshes with gear *B*. Indexing of turret *C*, as required to bring the correct numbers into their respective stamping positions, is controlled by a separate mechanism (not shown) which actuates trip-rod *D*. When rod

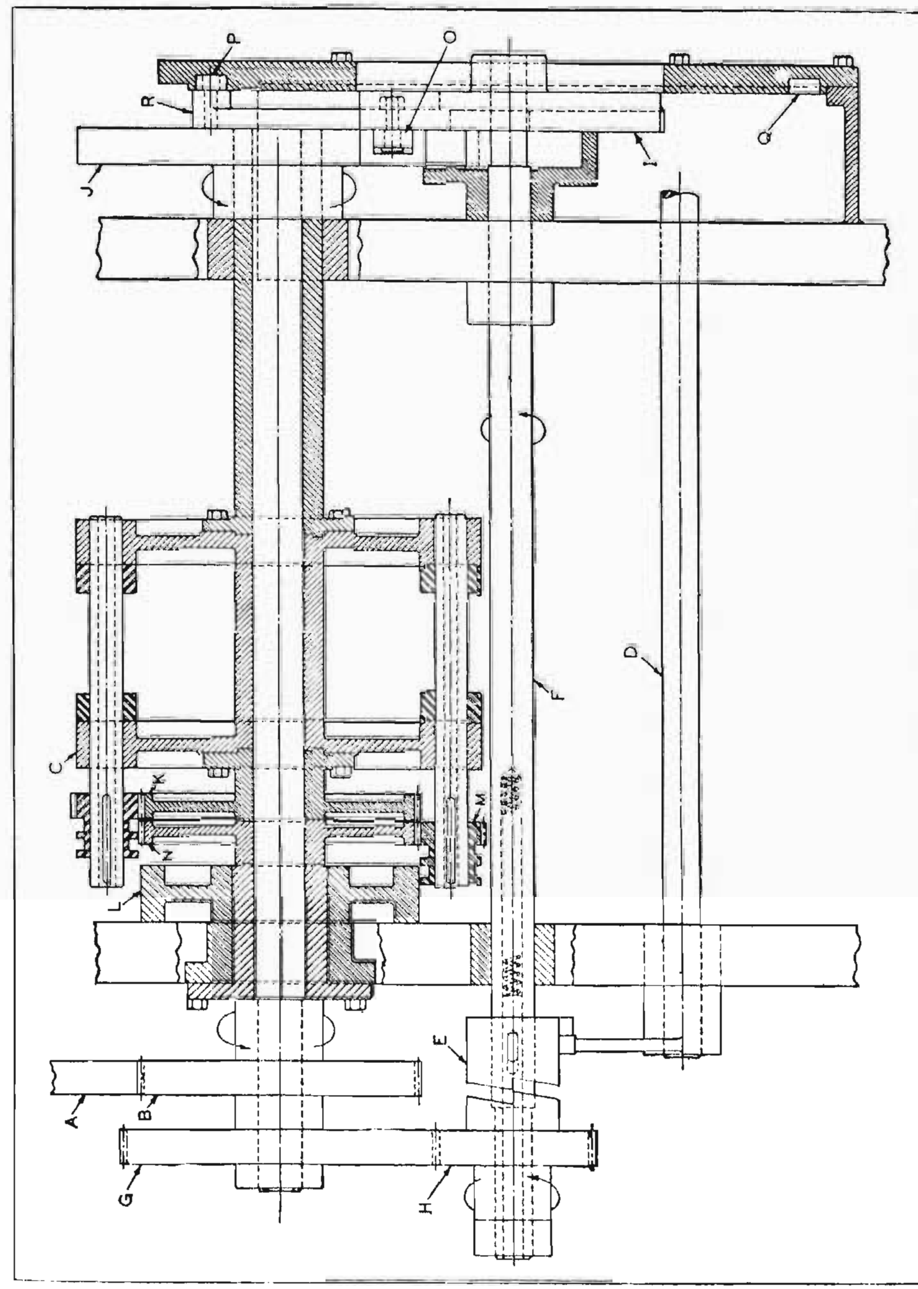


Fig. 15. Geneva Motion Mechanism which Indexes at Uniform Rotational Speed. Driving Member I Indexes Geneva Dial J

D is moved by the control mechanism to engage clutch *E*, shaft *F* makes one revolution, being driven by pinion *A* through intermediate gears *B*, *G*, and *H*.

The Geneva motion driving member *I* (also shown in Fig. 16) keyed to shaft *F* indexes the Geneva dial *J* one station, or one-sixth of a revolution. Dial *J*, being secured to a sleeve on which turret *C* is mounted, transmits the same indexing movement to the turret and gear

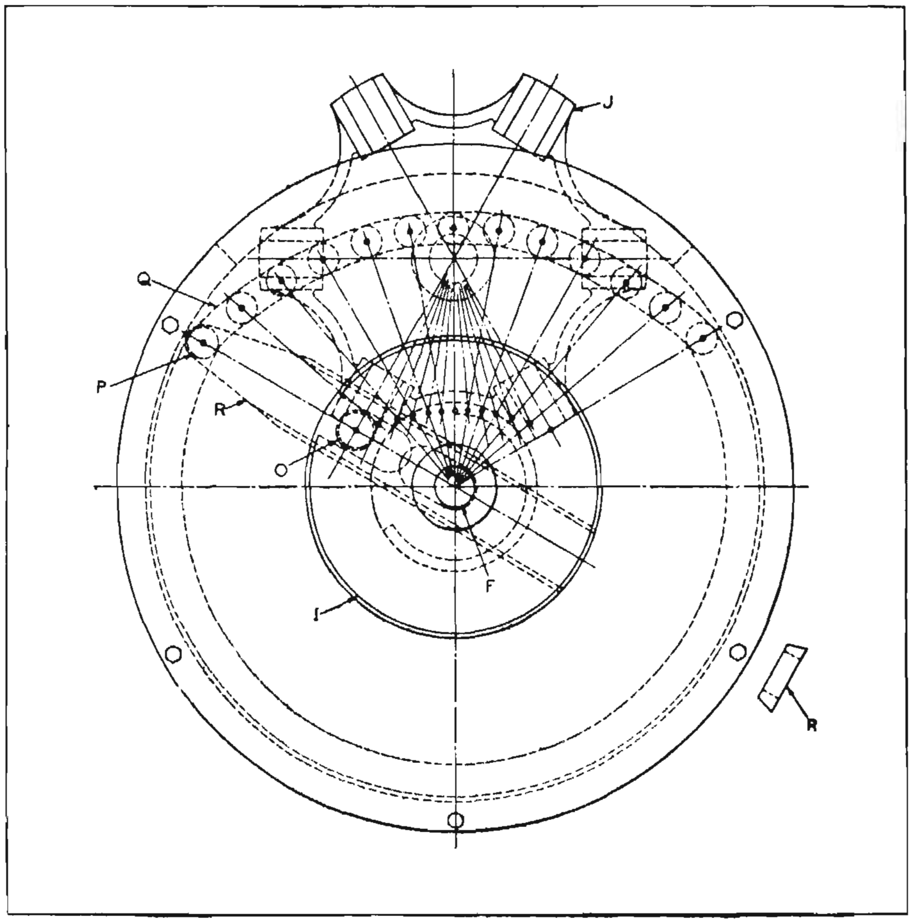


Fig. 16. End View of Geneva Motion Mechanism Shown in Fig. 15. Geneva Dial J has Six Stations.

K. During the indexing movement, cam *L* shifts pinion *M* out of mesh with gear *N* and into mesh with gear *K*. In order to permit this gear shifting to be accomplished, it is necessary that gears *N* and *K* have the same constant rotational speed, which would not be the case if the Geneva motion mechanism were of conventional design. Thus it was necessary to modify the design of the Geneva drive motion, as indicated in the end view of the mechanism, Fig. 16.

Referring to Fig. 15, pin *O* (which in the case of a Geneva motion mechanism of the regular type would be in a fixed position) is secured in a slide *R* mounted in a dovetail groove in driver *I*. At the outer end of slide *R* is mounted a follower-roll *P*, which runs in a cam groove *Q* in a stationary plate. This cam groove is so designed that when pin *O* enters the driving groove or slot in the Geneva dial *J*, slide *R* is forced inward, causing pin *O* to accelerate the rotational speed of dial *J* sufficiently at the entering and leaving positions of the indexing movement to result in a constant or uniform rate of rotation for dial *J* during the entire indexing operation. This uniform speed of rotation, being the same as that of gear *K*, permits pinion *M* to be readily shifted from gear *N* to gear *K* or from *K* to *N* during the indexing movement of dial *J* through one-sixth of a revolution.

Geneva-Wheel Mechanism for Obtaining Intermittent, Reversible Rotation.—Intermittent rotation, with means for reversing the direction of rotation, when desired, is provided by the mechanism shown in the accompanying illustration. Referring to Fig. 17, the vertical shaft *K* is driven at the same speed as the drive-shaft *A* through the medium of a pair of bevel gears *B* having a ratio of 1 to 1. The shaft *A* also drives the disk *C*, in which a pin *D* is mounted.

Pin *D* fits into the radial slots of the Geneva wheel *E*, which are spaced 72 degrees apart. Thus, for each revo-

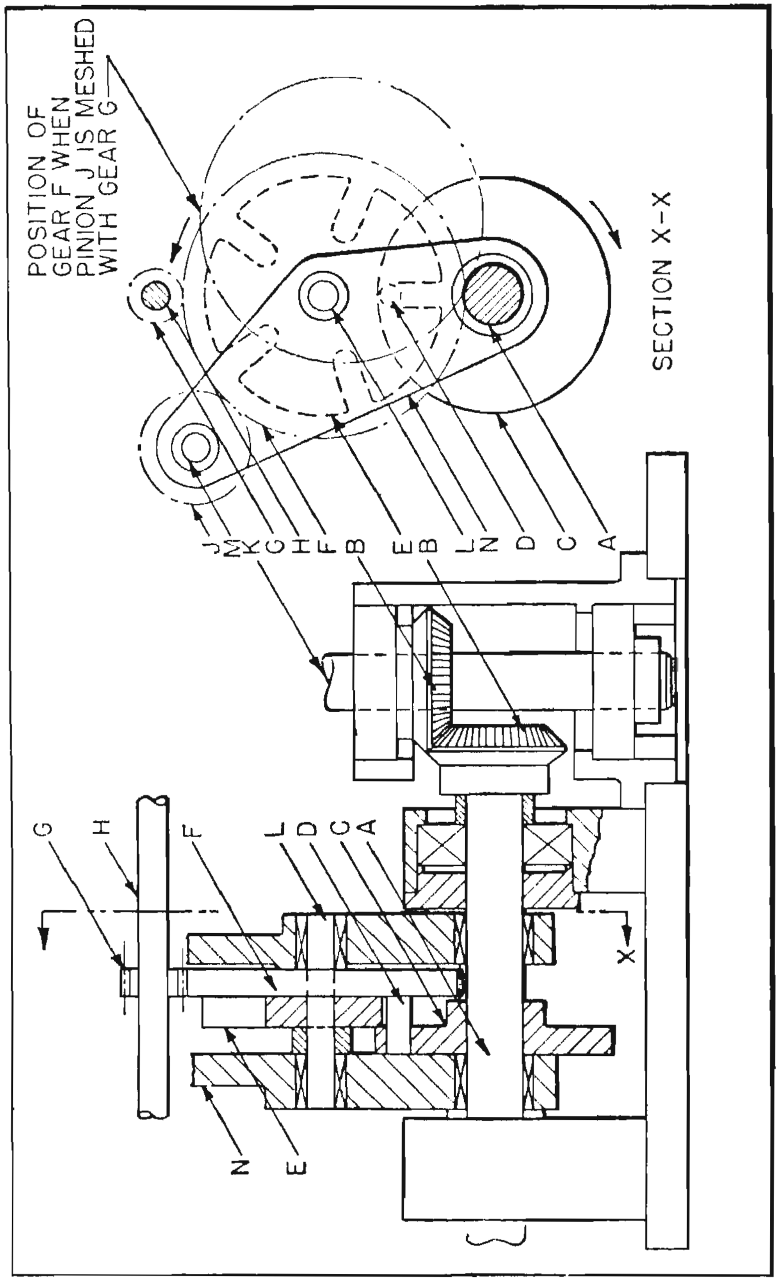


Fig. 17. Intermittent and Reversible Rotation of Shaft H is Obtained from Driving Shaft A by Means of Geneva Wheel E and Gearing.

lution of shaft A, wheel E is indexed through one-fifth of a revolution by pin D. Gear F, which is fixed to wheel E and rotates on the same shaft L, has sixty teeth of 24 pitch. Pinion G, having twelve teeth of 24 pitch, which meshes with gear F, is therefore driven intermittently through one revolution for each revolution of driving shaft A.

The direction of rotation of shaft H, on which pinion G is mounted, can be reversed by bringing gear J into mesh with pinion G. This angular movement disengages gear F from pinion G, as both shafts L and M are mounted in bracket N, which pivots about driving shaft A.

CHAPTER 4

Overload, Tripping and Stop Mechanisms

Mechanisms which automatically operate to stop an operation when overload occurs, to trip and start a new sequence or operation when a certain position or part of a cycle is reached, or to bring an operation to a halt at the end of a given cycle or when a given amount of motion has occurred, are described in this chapter. Other mechanisms performing similar functions are described in Volumes I and II of "Ingenious Mechanisms."

Tripping Mechanism Operated by Revolving Shaft.—The mechanism shown in Fig. 1 is designed to operate the tripping lever *A* from the rotating shaft *B*. The object is to move lever *A* from the position shown by the full

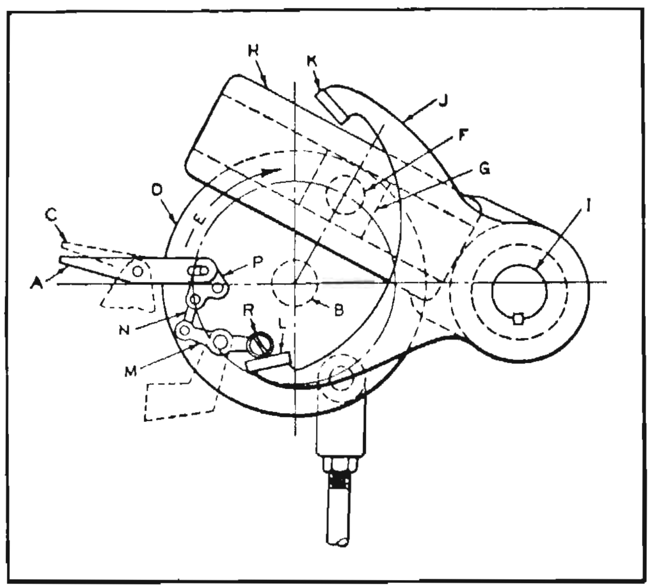


Fig. 1. Tripping Mechanism Operated by Rotating Shaft B which Shifts Lever from Position A to Position C and Back Again with Dwell Between Movements.

lines to that indicated by dotted lines at *C*, and back again, with an idle or rest period between movements. This is effected by means of disk *D*, which is keyed to shaft *B*, and linkage arrangement shown. Shaft *B* and disk *D* rotate in direction indicated by arrow *E*. A pin *F* in disk *D* fits a block *G* which is a sliding fit in a slot in lever *H*.

Lever *H*, being keyed to the shaft *I*, oscillates the latter shaft continuously as shaft *B* revolves. This oscillating movement is also transmitted to the fork-shaped member *J*, keyed or pinned to shaft *I*. Member *J* has two pads *K* and *L* which are alternately brought into contact with the roll *R* at one end of the lever *M*. The illustration shows the mechanism with lever *H* at approximately its highest point, and pad *L* in contact with roll *R*. With lever *H* in this position, the outer end of the trip-lever *A* is depressed to the position shown through the action of connecting link *N* and crank member *P*.

As shaft *B* continues to revolve, pad *L* moves away from roll *R*, allowing trip *A* to retain its position until pad *K* makes contact with the upper side of roll *R*, depressing it so that the outer end of trip *A* is raised to the position indicated by the dotted lines at *C*. Continued rotation of shaft *B* raises pad *K* from contact with roller *R*, so that the trip retains the position indicated by the dotted lines until pad *L* is again brought into contact with roll *R*.

Tripping Mechanism Operated by Revolving Shaft.—A modification of the design just described eliminates the large member *J* shown in Fig. 1.

As shown in Fig. 2, the lever *H* is provided with an integral extension piece *C* which acts upon roller *R*, causing it to transmit motion to the tripping lever *A*. Pad *L* is attached to lever *H*, as shown, and is located so as to come in contact with roller *R* when lever *H* is brought to its lowest position. The lever *H* has an oscillating movement obtained by means of shaft *B*, disk *D*, etc., as in the mechanism just described.

Two-Way Stop for Angular Movement of Shaft.—The mechanism shown in Fig. 3 comprises an arrangement for stopping the rotation of shaft *A* in two extreme angular positions by means of a single lever arm *B* attached to the shaft. The shaft *C* is connected to the lever arm by a pin, and is a slip fit through the trunnion *D*. The trunnion revolves around pin *G* and is held securely to the baseplate by the two bearing brackets *E*.

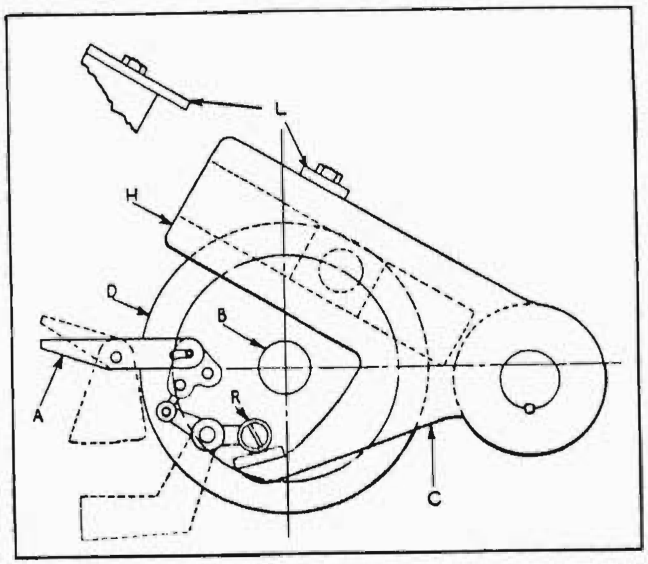


Fig. 2. Tripping Mechanism Operated by Rotating Shaft B which Shifts Lever A with Dwell between Movements.

The rubber shock absorber *F* on the end of shaft *C* stops against the trunnion block when shaft *A* is in either of the extreme positions *X* or *Y*. The position of the shock absorber is adjusted by means of the nut and lock-nut shown. By moving the trunnion block and its supporting brackets to either side of the center line, the extreme positions at which the angular movement of shaft *A* is stopped can be changed to suit a wide range of operating requirements.

Mechanism for Tripping a Rotating Lever.—The mechanism shown diagrammatically in Figs. 4 and 5 is designed to operate a trip-lever within a revolving housing at certain predetermined intervals, the exact time of each operation being controlled by a cam driven at the correct speed. The revolving housing *A*, Fig. 4, is mounted on sleeve *B* held on shaft *C*. The housing or head *A*, with its group of parts, revolves on sleeve *B* in the direction indicated by arrow *D*. The short lever *E* is free to pivot, and is pinned to shaft *F*, shown in cross-section. Lever *E* carries a roll *G* which comes in contact with a long radial dwell

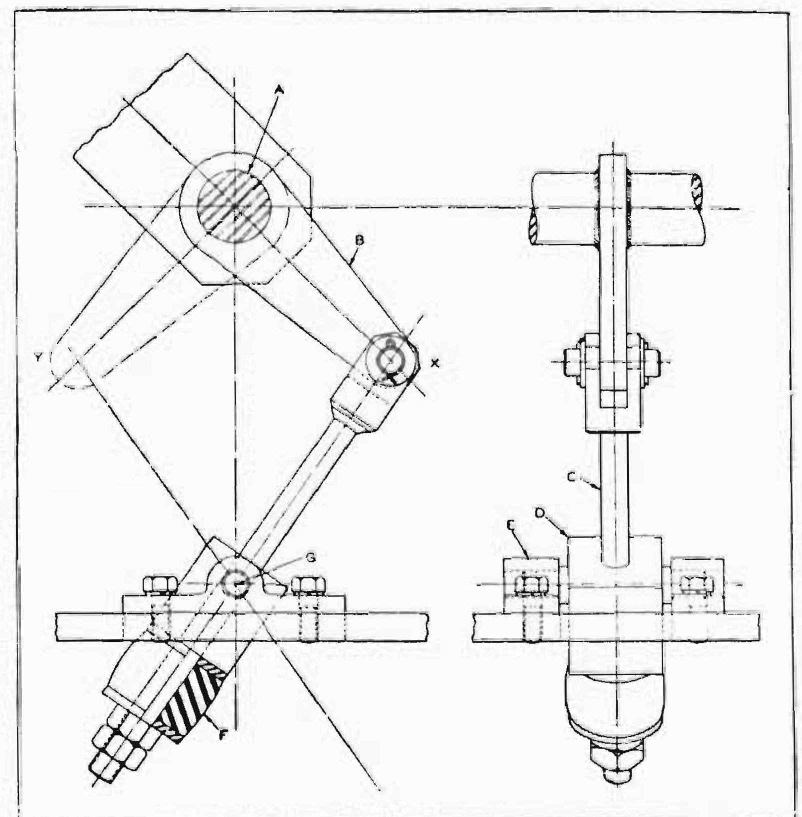


Fig. 3. Mechanism for Stopping Rotation of Shaft A in Two Extreme Angular Positions.

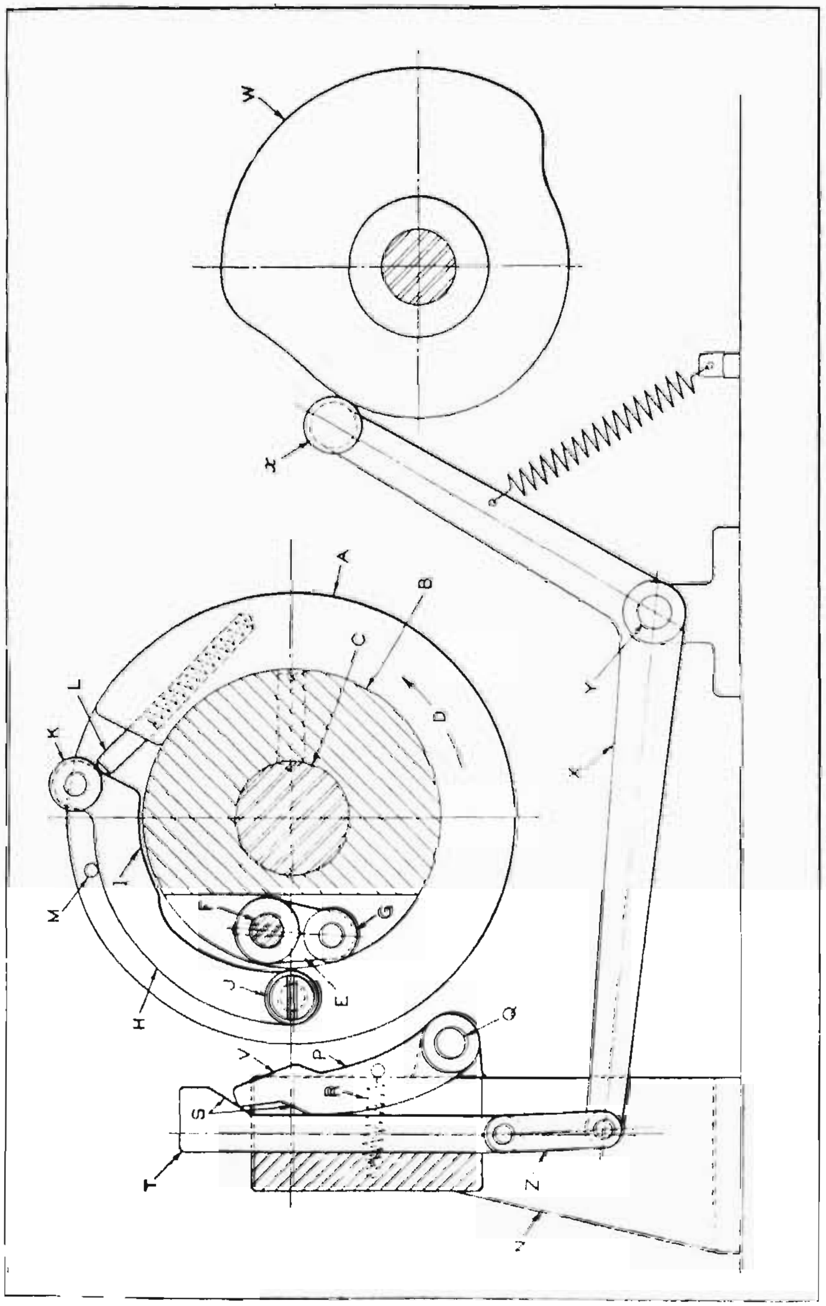


Fig. 4. Mechanism for Operating a Trip-lever on Shaft F within the Revolving Head A.

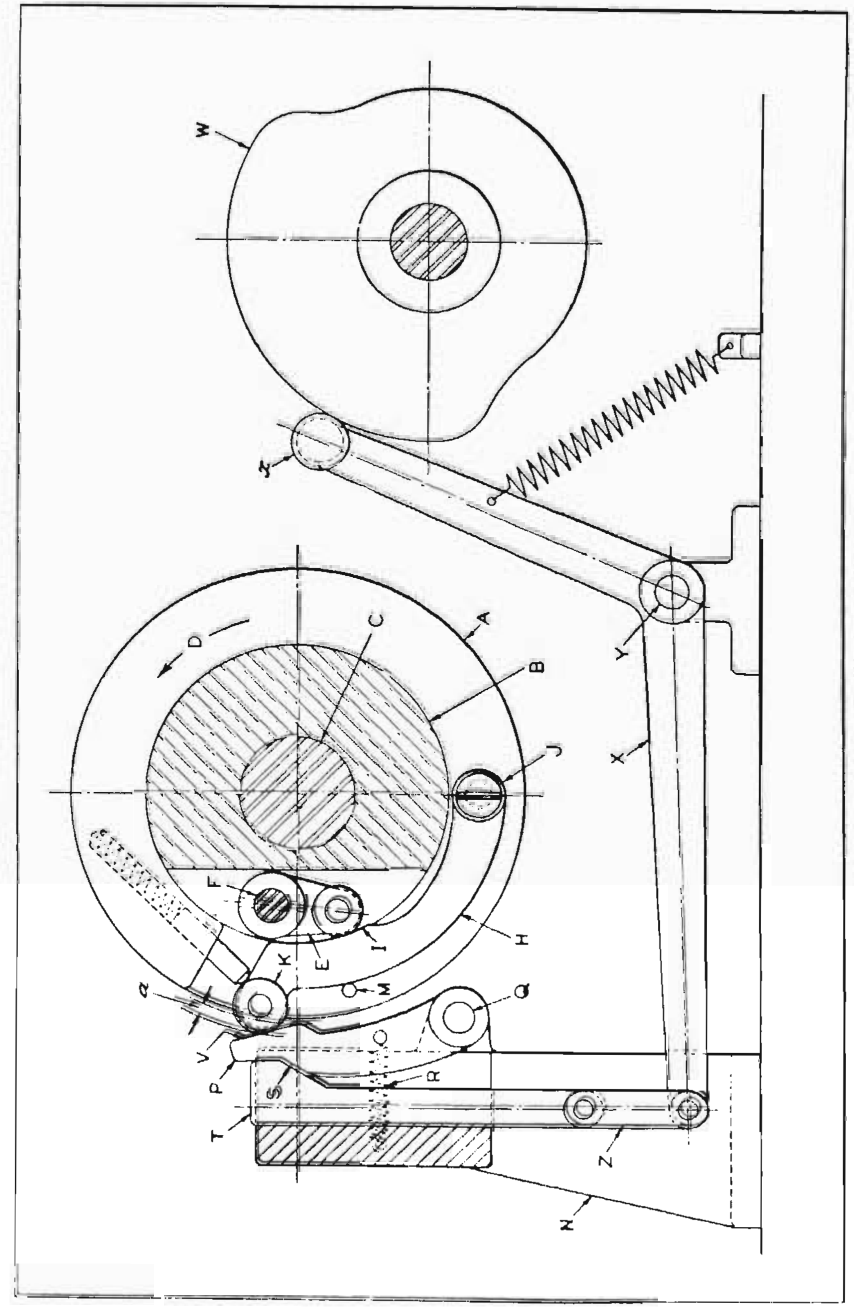


Fig. 5. Mechanism Shown in Fig. 4 with Lever P Brought into the Operating Position by Cam W.

I on lever *H*, the latter lever being attached to the housing by fulcrum stud *J*.

The long lever *H*, with a roll *K* at the end, is held in the outward position, away from the center of the revolving housing by means of spring pin *L*, the outward movement being limited by a stop-pin at *M*. When roller *K* moves in and out a distance *a*, Fig. 5, pivoting about stud *J* as a center, the short lever *E* rocks shaft *F*. This rocking action, in turn, operates a trip mechanism (not shown) which is located within the machine itself. A spring within the machine serves to hold the short lever in its outer radial position.

The mechanism for operating the long lever *H* at predetermined intervals as it revolves past a non-revolving cam surface *V* on lever *P* is actuated by cam *W*. Referring to Figs. 4 and 5, bracket *N* is attached to the side of the machine and carries lever *P*, which is pivoted at *Q*. Spring *R* serves to hold lever *P* back against the beveled or cam surface *S* of a sliding member *T*. Member *T* is

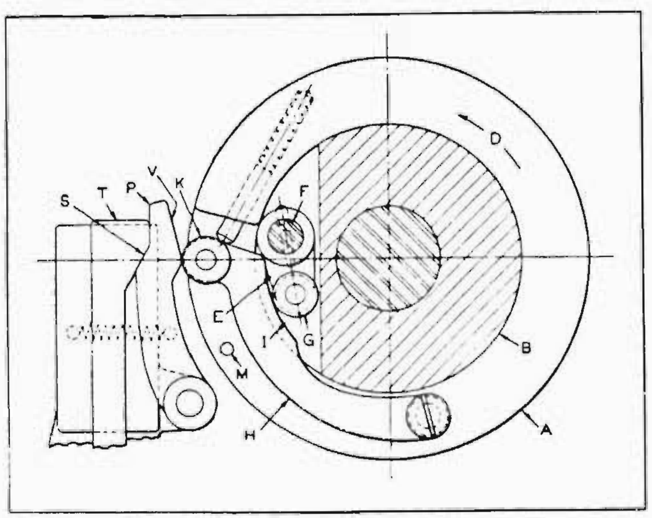


Fig. 6. Lever H is Forced Inward by Cam Surface V on Lever that Causes Cam I to Rotate Shaft F in Counter-clockwise Direction through Contact with Roller G.

free to travel up or down in a guide-way of the bracket. Cam surface *S* of the slide operates to push lever *P* to the right, so that roll *K* on the revolving lever *H* strikes the cam surface *V* at the correct time for operating the trip mechanism.

Actual timing of the trip is accomplished by cam *W* on a shaft which revolves in time or synchronism with the other operative units of the machine. Lever *X*, with a roller at *x*, pivots on a stud at *Y*. Lever *X* actuates slide *T* through connecting link *Z*; thus, when cam *W* and head *A* revolve in proper synchronism, the surface *S* on slide *T* is moved into engagement with lever *P* which, in turn, places surface *V* in position to push lever *H* over, as shown in Fig. 6, causing the short lever *E* to operate the tripping mechanism within the machine.

Overload Release Clutch Mechanism.—In the operation of an automatic machine that produces a formed wire product, it was impossible to prevent occasional jamming when changes in the wire size resulted in imperfect forming. To prevent serious damage to the machine, it was necessary for the operator to cut off the power immediately when the machine became jammed. As the operator was unable to maintain the close watch of the machine required to prevent damage, it was decided to attach an overload release clutch, as shown in Fig. 7.

The normal operating positions of the overload clutch members are shown in the left diagram of Fig. 7. Drive-shaft *A* carries a collar *B*, which is keyed to it. Collar *B* is grooved on the periphery to receive a pad on the long arm of bellcrank lever *D* which swivels on stud *E* carried on gear C. Stud *G* on gear *C* carries lever *F*, the lower end of which is provided with a V-shaped cam surface. The lower side of this V-shaped cam is in contact with the rounded end of lever *D*. Spring *H*, attached to the short arm of lever *D* and the lower end of lever *F*, provides sufficient tension to hold the pad on lever *D* in

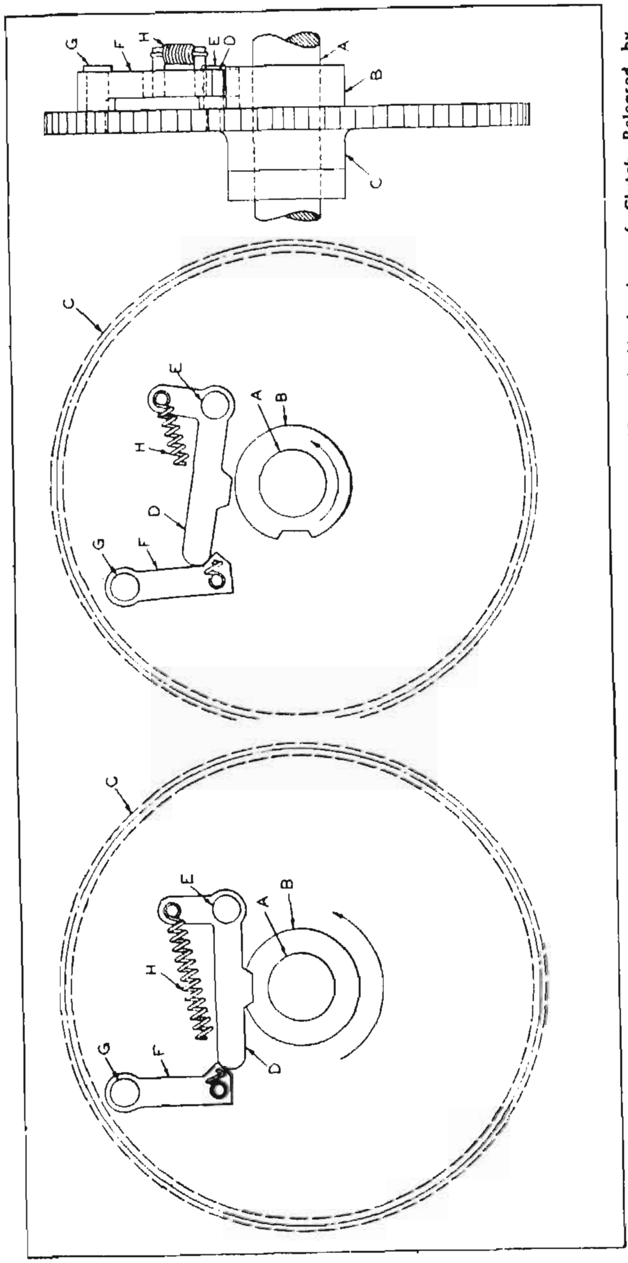


Fig. 7. (Left) Overload Release Clutch Mechanism with Driving Members Engaged; (Center) Mechanism of Clutch Released by Overload; (Right) End View of Clutch.

the groove of collar *B* during normal operation. The rotation of collar *B* is thus transmitted to gear *C* through lever *D* and stud *E*, the entire assembly rotating as a unit in the direction of the arrow, as shown in the left diagram of Fig. 7.

In the event that the clutch is overloaded, due to jamming of the work, the resistance to rotation of gear *C* overcomes the tension of spring *H*, causing the pad on lever *D* to be forced out of the groove in collar *B*, and thus disconnecting gear *C* from the source of power transmitted by shaft *A*. The movement of the end of lever *D* on the cam surface of lever *F* swings the latter member to the left; and as the rounded end of lever *D* passes over the high point of the cam surface on lever *F*, the upper edge of the cam surface on lever *F* acts on the lower edge of the end of lever *D*, lifting it completely out of contact with collar *B*, as shown in the center diagram of Fig. 7, and thus preventing lever *D* from returning to its normal driving position until it has been re-engaged by the operator.

An Overload Relief Device for Machine Protection.—An ingenious device designed to protect a driven machine mechanism from breakage in the event of accidental jamming of the work during normal operation of the machine is shown in Fig. 8. One feature of this device is that all forces are self-contained, and when it is tripped or released, no axial force is exerted on bearings or moving parts.

The mechanism consists primarily of a lever *A* which transmits a rocking movement to shaft *B* through suitable linkage. By itself, lever *A* is free to swivel about shaft *B*. It is held in one position on the shaft by a bearing at the right and arm *C* and collar *F* at the left. With plunger *D* seated in a conical socket in bushing *E*, however, motion is transmitted from lever *A* through arm *C* to shaft *B*, arm *C* being keyed to the shaft. Plunger *D* is held in the seated position by spring *H*.

The conical fit between plunger *D* and the socket of bushing *E*, as well as the load provided by spring *H*, should be designed for transmitting only the desired amount of torque. Then, when this predetermined torque is exceeded, the tapered end of plunger *D* will ride up out of its socket and lever *A* will become disengaged.

To keep the parts disengaged is the function of plunger *G*. At the top point in the disengaging movement of plunger *D*, the deeper of two flat spots on the plunger comes opposite the end of plunger *G*. Plunger *G* then snaps to the right and holds plunger *D* in the disengaged position. This

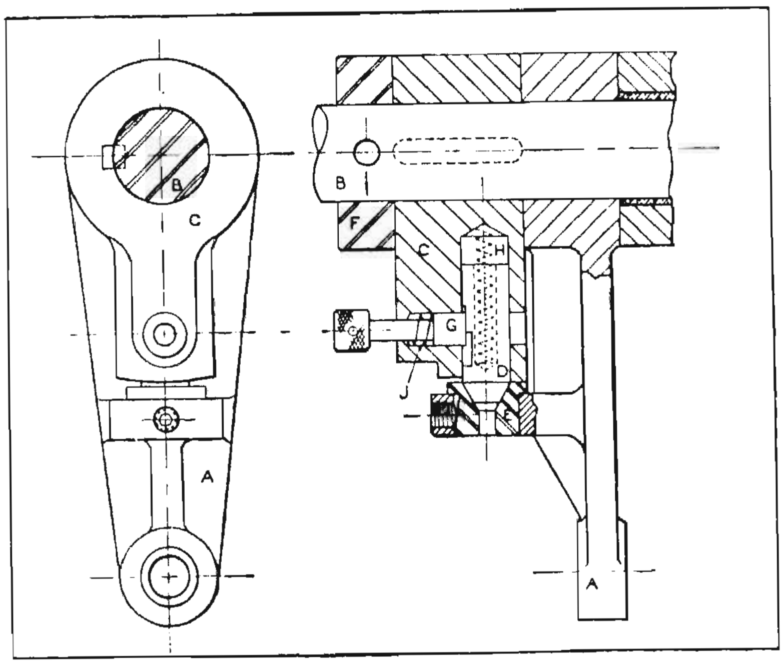


Fig. 8. Ingenious Device Designed for Protecting a Mechanism against Breakage if the Work Becomes Jammed.

permits the lever *A* to rock freely on shaft *B* until the device is manually reset, and power is, of course, no longer transmitted to shaft *B* through the mechanism.

To reset the device, the machine must first be cleared

of the obstruction, after which plunger *G* can be withdrawn to the left and plunger *D* reseated in the socket of bushing *E*. Plunger *G* is then free to return to the right, where it engages the shallower flat spot on plunger *D*. The purpose of this shallower flat spot is to prevent rotation of the plunger and to retain plunger *D* in position during disassembly of the mechanism.

This device is also applicable to rotary driving members, as well as to rocking members. In the case of rotating applications, the centrifugal force acting on plunger *D* would have to be considered as well as the load provided by spring *H*. The plungers *D* and *G* and bushing *E* should be hardened in order to insure satisfactory life.

Drive Unit with Overload Slip Mechanism.—The slip device shown in Fig. 9 was designed to prevent breakage of operative members, and is an important feature of the feeding mechanisms of certain machines. It is of the jump-pawl type, the pawl *A* being designed to operate with the radial plate *Q*. Pawl *A* is free to pivot upon stud *B*, although it is normally held in the position shown by the spring *C*.

The bellcrank lever *D* oscillates through the arc *E*, being actuated by the shaft *F* through pawl *A*, plate *Q*, and plate carrier *R* which is keyed to shaft *F*. Linkage levers *G* and *H* are moved up and down by means of connecting-rod *J*. Lever *G* is attached to a non-movable block at *K*, while lever *H* is attached at *L* to a sliding member *M*. The sliding member moves backward and forward in the direction indicated by the arrow *N*, once on the up stroke of the connecting-rod *J* and once on the down stroke, thus, in effect, producing a two-cycle movement of the member *M* for each reciprocation of rod *J*.

If slide *M* becomes jammed or is prevented from moving in a normal manner, pawl *A* will ride out of the notch in plate *Q* and travel over the radial surface of this member, thus disconnecting the drive.

Overload Feed-Trip Mechanism for Lathes.—The simple feed-trip mechanism shown in Fig. 10 was fitted to a heavy shell-turning lathe on which the feed had previously been of the entirely positive type, with no safety device to take care of overloading or jamming. The saddle of this lathe is fed in one direction only—that is, toward the headstock. Had there been a reverse for the feed-shaft, it would not have been possible to use the mechanism described.

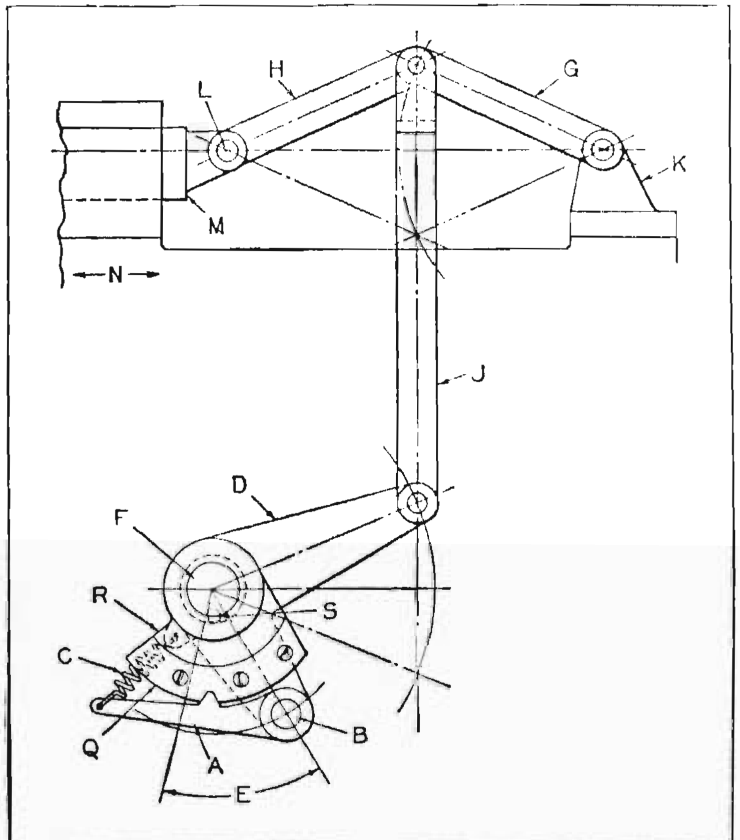


Fig. 9. Split-drive Mechanism by Means of which Oscillating Bellcrank D Transmits Reciprocating Movement to Slide M with Provision for Disengaging the Drive from Shaft F in Case Slide M Becomes Jammed.

A slipping type coupling in the feed transmission had been decided upon, but this proved too difficult to apply, particularly on the existing machines. It was suggested that a slight alteration to the trigger mechanism of the drop-worm would allow it to automatically force itself out of mesh with the worm-gear in the event of overloading or jamming. Such an arrangement would have an advantage over a slipping clutch because, immediately the worm tripped, all stress would be removed from the feed transmission, whereas, with the usual type of slipping

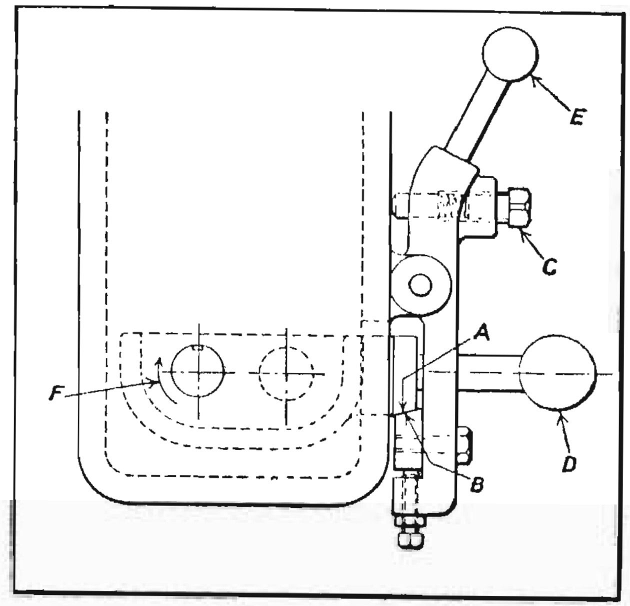


Fig. 10. Lathe Feed Overload Tripping Mechanism for Heavy-duty Turning Operations.

clutch, the whole of the feed transmission would be subjected to a heavy load, in the event of a jam, until noticed by the operator.

The simplicity of the mechanism shown can be judged from the fact that, in less than an hour after it was suggested, it had been tried out and found satisfactory. No new parts were made, the only work done consisting of a slight modification of the trigger elements of the drop-worm. Instead of being made practically square, to hold the drop-worm positively in mesh with its gear, the trigger faces *A* and *B* were made at an angle of 15 degrees, so that a downward thrust on *B*, resulting from an overload, would disengage the feed. Set-screw *C* compresses a spring, and can be adjusted to vary the pressure with which the trigger is held in the engaged position. The worm is lifted by knob *D*, and knob *E* operates the trigger to trip the feed. The feed-shaft runs in the direction indicated by arrow *F*.

Roll-Driving Mechanism that Stops Automatically in Case Material Breaks.—The mechanism shown in Fig. 11 was designed for unwinding thin tissue from large rolls. The problem was to provide suitable means for feeding the delicate, thin material, which is easily stretched and torn by improper handling. The arrangement shown provides for stopping the feed if the material breaks, and will also slow up the speed of the roll if the feeding rate becomes greater than the rate at which the material is consumed. Thus, the roll of thin material is kept under control at all times, so that the tissue is subjected to a minimum amount of tension.

The two fork-shaped side frames *A* are bolted to a base-plate *B*. They support two feeding rolls *C*, mounted on shafts *D*, to which they are fastened by taper pins *E*. Rolls *C* are provided with molded rubber covers *F* to increase the driving friction. The roll of tissue *G* is kept in position by shaft *H*, which is fitted in bushings *I*. The bushings are arranged to slide freely in the vertical slots in the side frames *A*.

Mounted on each of the feeding roll shafts *D* is a spur gear *J*, fastened in place by a taper pin *K*. Bearing blocks *L* are so constructed that the feeding roll assembly can be removed by unscrewing bolts *M*. Driving gear *N* of the

feeding control mechanism is fastened to stub shaft *O* by a taper pin. The driving gear meshes with the feeding roll gears *J*. Clutch member *P* is fastened to stub shaft *O*, in which there is a bore that supports the end of driving shaft *Q*. The driving shaft is thus supported between the stub shaft and a bearing in the side frame *A*. Clutch member *R* slides on shaft *Q*, guided by key *S* under the

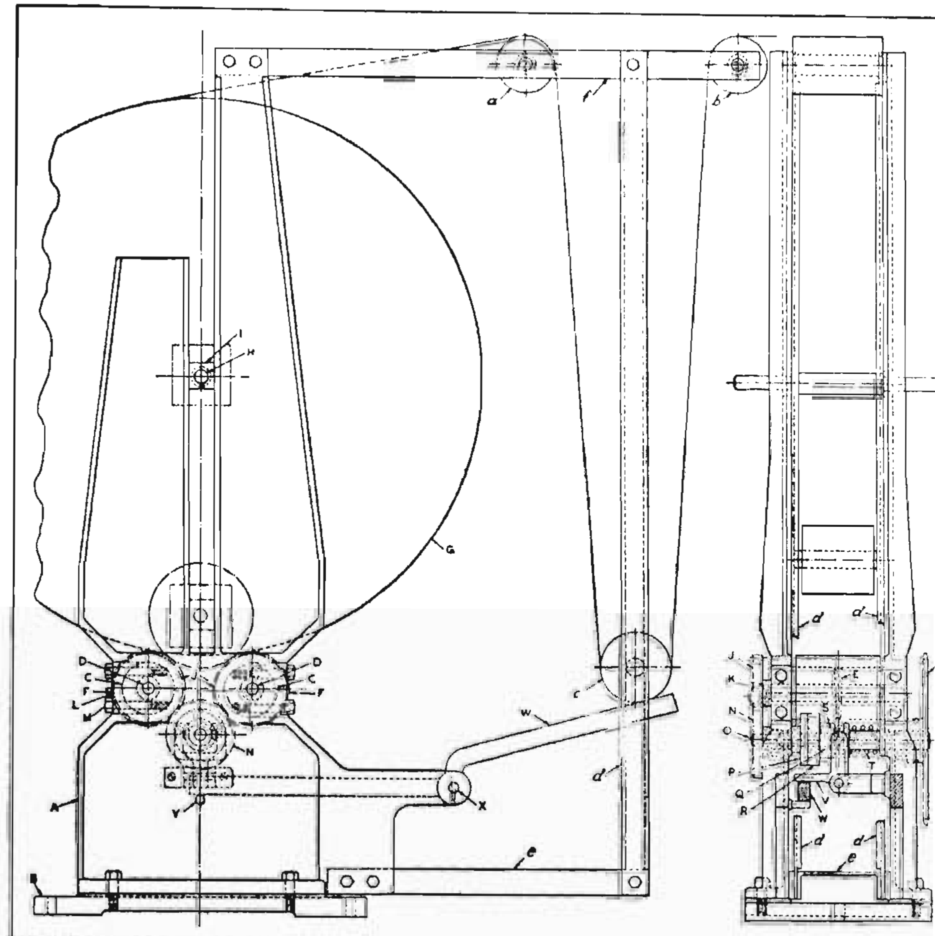


Fig. 11. Power-driven Mechanism for Unwinding Thin Tissue from Roll G which Stops if Material Breaks or if the Feeding Rate Exceeds the Rate at which the Material is Used.

action of spring T. Driving sprocket U is fastened to shaft Q.

Yoke V of the clutch-actuating mechanism is equipped with pins which fit into a groove in clutch member R. The yoke is supported by a bracket fastened to side frame A. Tripping bar W is pivoted on shaft X, which is supported in bearings provided in side frames A. The tripping bar is balanced, so that it rests on pin Y in the side frame.

The thin tissue material is guided over a series of three rolls, the positions of the two rolls a and b being fixed. The third roll c is free to move up and down in the side members or guides d. Guides d are fastened to horizontal members e and f which are, in turn, fastened to side frames A.

The operation of the unwinding mechanism is as follows: Sprocket wheel U actuates driving gear N through clutch members P and R, thus rotating feeding rolls C. The feeding rolls, in turn, rotate the roll of tissue, the entire weight of the material being supported by the feeding rolls. The tissue passes over guide roll a, under movable guide roll c, and up over the fixed guide roll b. In actual operation, movable guide roll c is raised from tripping bar W.

If the tissue is broken or ruptured, movable guide roll c comes to rest on tripping bar W, causing it to raise the opposite end from pin Y and make contact with yoke V. This action causes the yoke to draw clutch member R away from member P, thus stopping the feeding rolls. The same action takes place when the rate at which the material is being fed becomes greater than the rate at which the material is consumed. In such cases, movable guide roll c will fall on tripping bar W, causing the feeding rolls to stop and thus stop the rotation of the tissue roll G.

Clutch Equipment for Quick-Acting Brake.—The quick-acting brake in the double-ended jaw clutch shown in Fig. 12 was designed to permit the sudden stopping of a certain shaft E used for winding metal strip. The driving shaft A

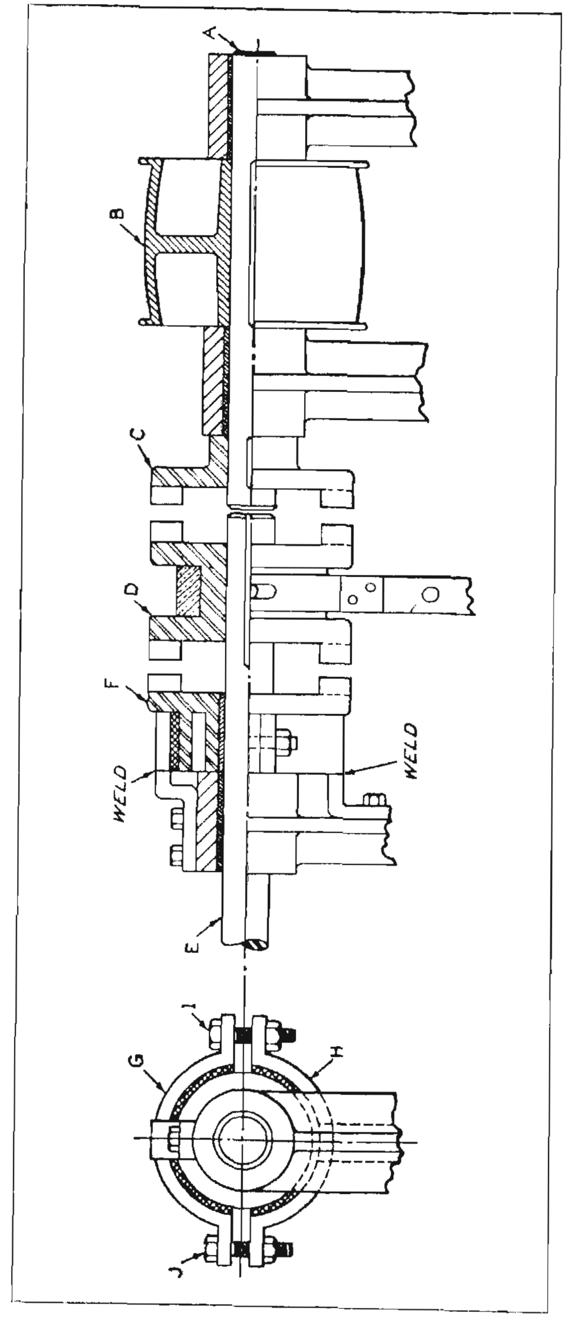


Fig. 12. Clutch Arrangement for Disengaging Shaft E from the Driving Shaft A and then Applying Brake to Shaft E.

is rotated continuously by a belt on the pulley B to which it is keyed. The double-ended clutch jaw D is free to move endwise along shaft E and slide along a key which compels it to revolve the shaft when engaged with the driving member C.

When jaw D is disengaged from member C, it engages the member F, which also acts as a brake-drum. Brake-shoes G and H, being tight at all times, will stop shaft E quickly upon the engagement of clutch D with member F. The brake-shoes are kept tight by means of bolts J and I. They are fastened to the bearing stand, which prevents them from turning. The brake linings are of standard width and can easily be replaced when worn out.

Safety Relief Mechanisms for Light Drives on Special Machines.—A relief mechanism of some type incorporated in the main drive provides adequate means of preventing breakdowns of many special light machines. It is sometimes difficult, however, to design or choose a simple effective device for this purpose that will entail a minimum of changes in existing mechanisms or machines. Figs. 13 to 18 show various pieces of mechanism that are capable of adaptation to almost any kind of special machine.

Practically all high-speed special-purpose machines have a number of connecting-rods, link arms, etc., one or two of which can be selected for modification. For example, the usual solid type connecting-rod can be replaced, as shown in Fig. 13, by one made of two parts A and B. Part B is tubular, and is threaded at one end with, say, twenty-six threads per inch to permit a fine adjustment of the operating arm by means of a knurled castellated nut G.

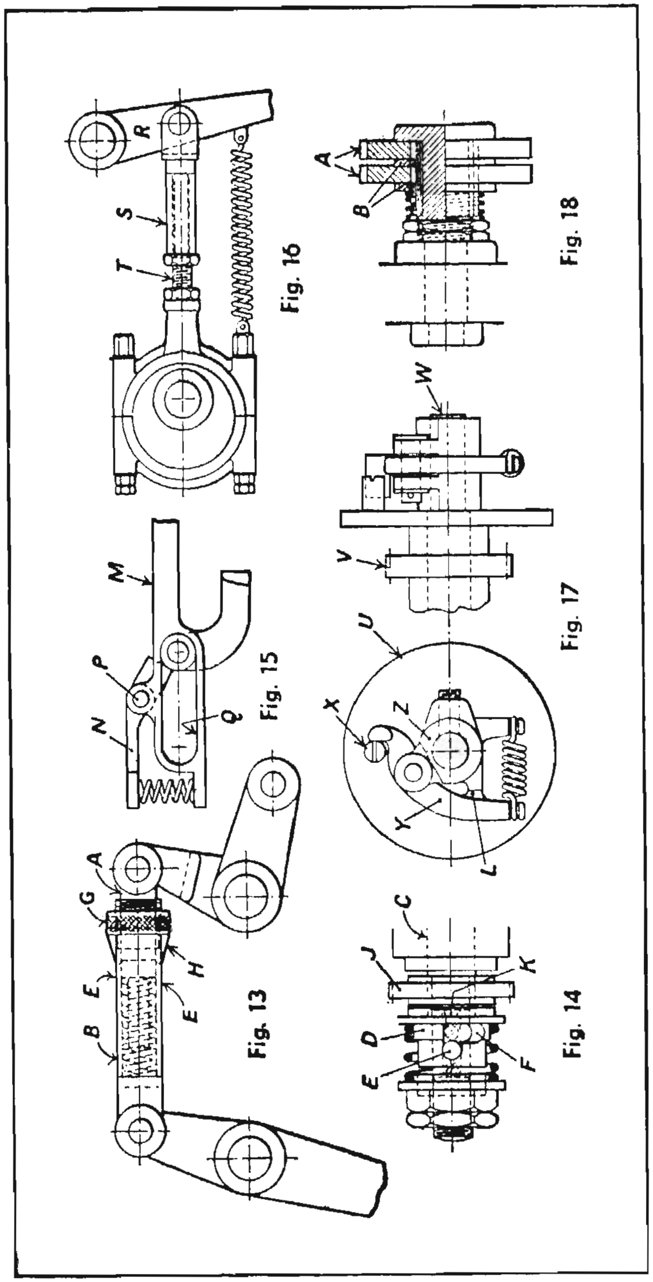
The other member A is a sliding fit in part B. A compression spring, located in part B, exerts pressure on member A. Two slots E are cut 180 degrees apart in B, and when this member is adjusted to the correct operating center distance by nut G, a pronged cotter-pin H is placed across one of the slots in the castellated head of nut G.

If the operating movement of the mechanism is obstructed, the arm simply collapses against the pressure of the spring until the mechanism has been freed. The amount by which the connecting-rod can be telescoped is limited by the length of slots E and the length of the prongs on cotter-pin H. The compression spring must, of course, exert sufficient pressure to operate the drive without yielding when the machine is working under normal conditions.

When the drive to a main camshaft or to a separate mechanism requires a safety relief, the one shown in Fig. 14 can often be used to advantage. On the main driving shaft C is a flanged sleeve D which is driven by key E, fastened to the main driving shaft. In the flanged sleeve there is an L-shaped slot F in which the projecting ends of the key are a sliding fit. A compression spring tends to force the bottom or end of the longitudinal leg of the L-slot into contact with the pin or key E. Thus the main driving gear or sprocket J is driven by means of pin E and its contact with slot F.

The flange also carries a clutch tooth K which is wedge-shaped, and at the instant the drive becomes excessive, the flange is forced against the spring and causes pin E to make contact with the other leg of slot F. The drive will then continue with the sleeve in an inoperative position until the obstruction is removed.

If a relief or safety device for a pusher member, slide, or folders is required, the simple attachment shown in Fig. 15 can be satisfactorily employed. The main cam arm should be replaced by an arm which has a slot Q at the operating end M. A small trigger-like arm N is pivoted about pin P. At the end of the trigger arm is a flared portion into which a recess is drilled to receive a compression spring. The compression spring tends to force the right-hand end of arm N slightly over the boss that carries the pusher, and under normal conditions has sufficient strength to transmit cam motion. If, however, the pusher strikes an obstruction,



Figs. 13-18. Six Types of Safety Relief Mechanisms for Light Drives.

the pressure causes the right-hand end of arm *N* to move upward and allow the arm to ride in the slot until the obstruction has been removed.

In a positive cam-driven mechanism, there is always danger of bending an arm and ruining the mechanism. To avoid this, the drive can be changed or replaced by that shown in Fig. 16. In this case, the driven arm *R* may have a connecting-rod *S*, and the cam arm may have an adjustable rod *T*. To introduce a safety device, one of the ends of rod *T* is threaded and screwed into the cam arm while the other end is left plain and inserted in a long bore in piece *S*. The two parts *S* and *T* are made of steel and are hardened. The bore of part *S* should be 1/32 inch larger than the diameter of the plain portion of rod *T*.

The cam arm and the driven arm are pulled together by means of a tension spring having sufficient strength to transmit the necessary motion. If, however, the mechanism becomes jammed, the spring is stretched and the plain portion of rod *T* reciprocates in the bore of piece *S* until the obstruction is cleared and arm *R* is again pulled into its operative position.

The drive for a conveyor or stock-feeding mechanism of a box-making machine, for example, can be safeguarded against breakage by the addition of a slip drive like that shown in Fig. 17. A flange *U* is fastened to the main driving gear or sprocket *V*, the whole assembly being allowed to rotate freely upon the drive-shaft *W*.

Flange *U* carries one or more pins *X*. When the positive drive is in operation, one of the pins *X* engages the drive lever *Y*, which is pivoted on a pin carried by the driving boss *Z*, fastened to the main drive-shaft by a set-screw. The driving lever *Y* is forced into contact with pin *X* by a tension spring, and stops *L* are filed to permit proper functioning. If the drive is obstructed, the spring yields and the main drive-spindle is allowed to revolve freely without rotating the driving gear or sprocket.

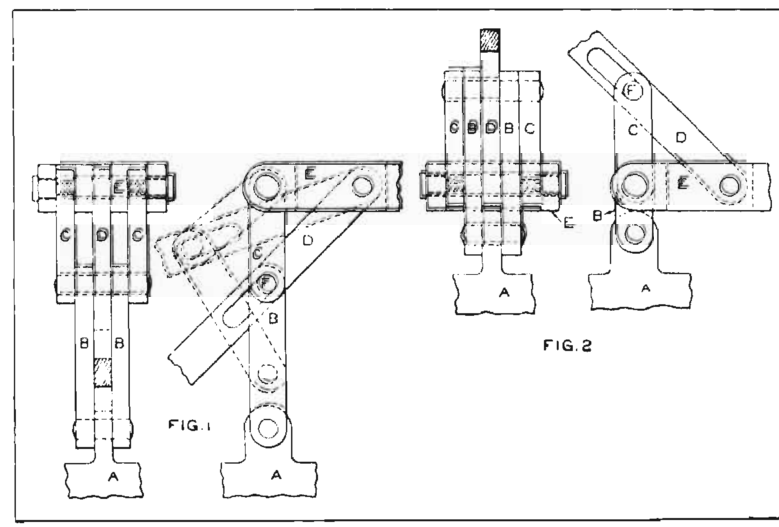
If the drive is of a lighter nature, such as that for a sheet-feeding device or a dating mechanism, a safety arrangement like that shown in Fig. 18 can be employed. The driving gears *A* can be made a running fit on the drive-shaft or stud. Each of the side faces of the driving gears comes in contact with the friction disks *B*, which are preferably keyed to the shaft in such a manner as to prevent rotation and yet allow a sliding motion on the shaft. These gears are forced into contact with the side faces by means of a compression spring adjusted to exert the required amount of pressure for driving the mechanism. Lock-nuts are provided, as shown, for maintaining the required setting.

CHAPTER 5

Locking, Clamping and Locating Devices

Means of positively locking a mechanism, clamping a work-piece or part, and locating work in the proper position for some operation to be performed on it, or locating a carriage or table in the correct loading position, are described in this chapter. In some cases, the locking or clamping operation is performed automatically, while in others hand operation is required. Similar devices are described in Volumes I and II of "Ingenious Mechanisms."

Double Locking Lever Motion.—A lever-and-link motion used to operate a slide that must be locked against movement at both ends of its vertical travel is shown diagrammatically in Figs. 1 and 2. The linkage automatically pro-



Figs. 1 and 2. Lever-and-link Motion which Locks Slide A at Each End of the Vertical Stroke.

vides the desired locking feature in both positions. The links *B* are connected to the slide *A* at their lower ends and are pivoted at their upper ends to the bracket *E*, attached to a stationary part of the machine. The hand-lever *D*, pivoted in the bracket *E*, passes between the links *B*, and is slotted to permit the pin *F* to pass through.

In Fig. 1, slide *A* is shown at its lower position. Links *B* and *C* have their centers in the same straight line, thus producing a dead center condition which locks slide *A* against movement in either direction. It will be noted that lever *D* is at an angle of approximately 45 degrees with links *B* and *C*. As the outer end of lever *D* is raised, the angularity of the slot in lever *D* causes pin *F* to be moved outward, so that links *B* and *C* are moved from their locked positions. Continued movement of lever *D* causes pin *F* to slide in its slot, and links *B* and *C* to swivel on their connecting pin, as shown by the dotted lines, Fig. 1, so that slide A is drawn upward.

Further movement of lever *D* causes links *C* to rotate 180 degrees on their upper pin, and links *B* to enter the spaces between links *C*. At the extreme upper position of slide *A*, as shown in Fig. 2, links *B* and *C*, again being in alignment, cause slide *A* to be locked against further movement. Thus the arrangement of the links is such that movement of slide *A*, when at its upper and lower positions, can be accomplished only through movement of lever *D*.

Lever Mechanism for Operating a Locking Pin and Clamping Bolts in One Movement.—The mechanism illustrated in Fig. 3 is employed on the swivel type fixture shown in Fig. 4 to withdraw a locking pin and release two clamping bolts in one movement of the hand-lever. The reverse operation of engaging the locking pin and tightening the clamping bolts is similarly accomplished by a simple return movement of the lever.

Referring to Figs. 3 and 4, it will be noted that hand-

lever *A* is located near one corner of the fixture in a convenient operating position and at some distance from locking pin *K* and the two clamping bolts *B* and *C* which it actuates. The clamping bolts act on both ends of the

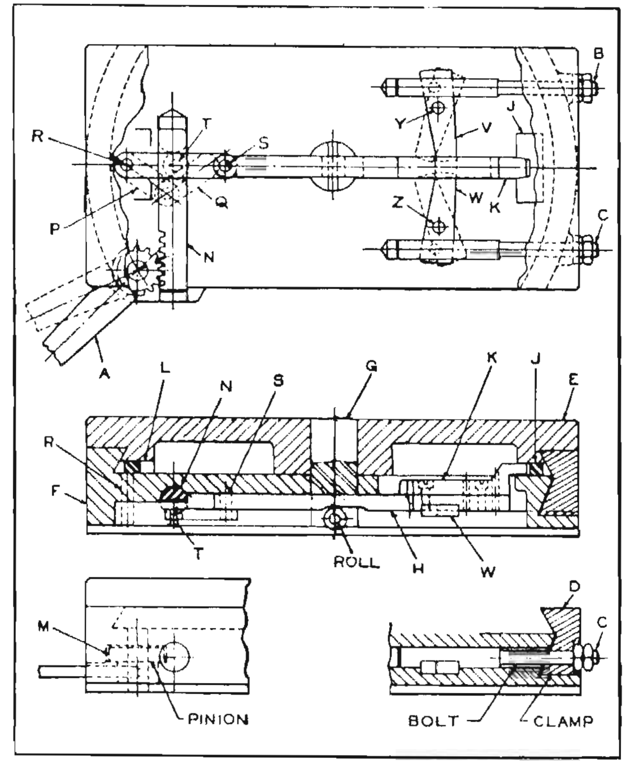


Fig. 3. Mechanism for Operating a Locking Pin and Two Clamping Bolts in One Movement of a Hand Lever.

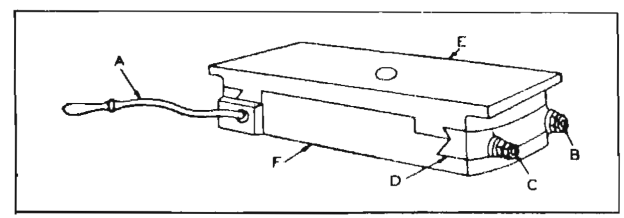


Fig. 4. Swivel-type Fixture Equipped with Locking Mechanism Shown in Fig. 3

dovetail segment *D*, as shown in detail at the lower right, lever *A* being the clamping medium.

The purpose of the entire unit is to bind upper portion *E* of the swivel base to the lower portion *F*. As shown in Fig. 3, stud *G*, which passes through the entire unit, is slotted and has a roll at its lower end. Bar *H* passes through the slot in stud *G* and rests on the roll. Stud *G* serves as a pivot on which section *E* can be swiveled. Swivel section *E* is located on the base by locking pin *K* which fits the slot in plate *J* attached to member *E*, while another plate at *L* serves to locate the swivel section when it is reversed or revolved through an angle of 180 degrees.

Pinion *M*, attached to operating lever *A*, meshes with a rack bar *N*. There are two toggle levers *P* and *Q*, lever *P* being pivoted at one end on pin *R*, fixed in base *F*. The other end of this lever is connected to one end of lever *Q*, the outer end of which is connected to bar *H* by pin *S*. Pin *T* acts as a fulcrum for the toggle joint.

At this fulcrum point, the ends of the levers are yoked into a cross-slot in the rack bar, so that, as the rack is moved into the position shown in Fig. 3, the toggles force bar *H* into the clamping position. When lever *A* is in the position shown by the dotted lines, the toggles are likewise in the positions shown by dotted lines, which results in withdrawing the clamping bar from plate *J* and unlocking dovetail clamp *D*, at which time two levers (shown at *V* and *W*) assume the positions shown by the dotted lines.

Levers *V* and *W* engage a slot in the lower side of bar *H* and pivot on pins *Y* and *Z*. These levers also have yoke connections with the two clamping bolts *B* and *C*, so that a solid metal-to-metal contact is provided. In effect, this forms a quick-operating, toggle-action mechanism by means of which the simple action of moving lever *A* is all that is necessary to unlock and unclamp upper member *E*.

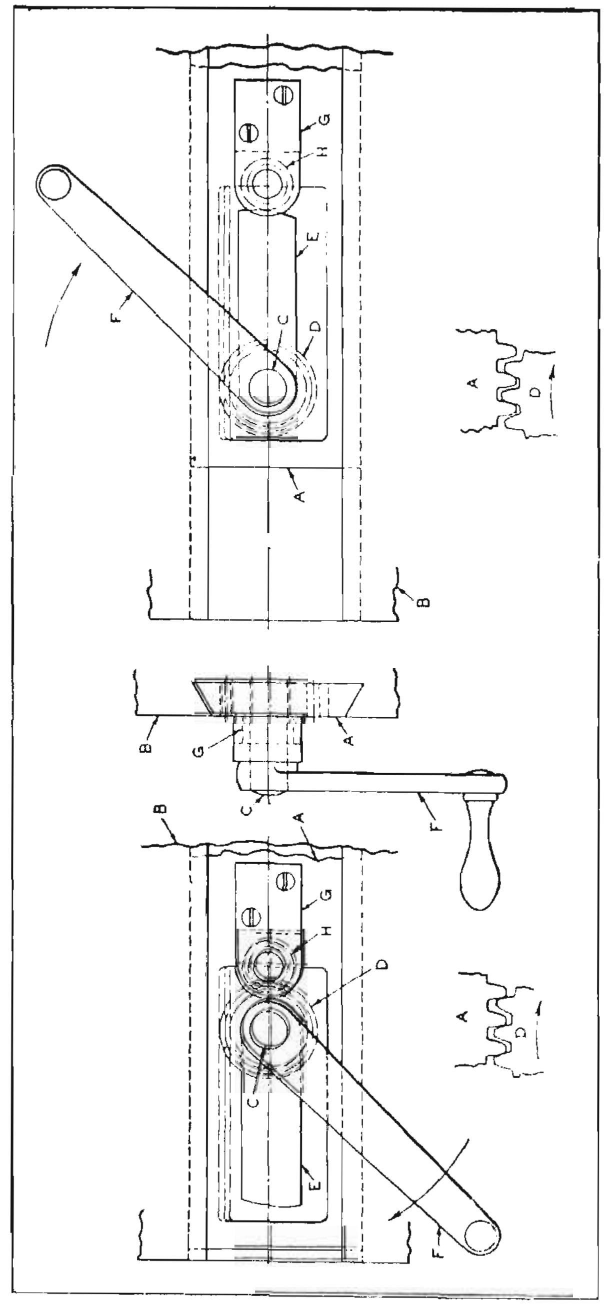


Fig. 5. (Left) Rack-and-pinion Mechanism with Rack Member A Withdrawn from Clamping Position; (Center) End View of Mechanism; (Right) Member A Locked in Clamping Position by Bar E.

Positive Lock for a Rack and Pinion Motion.—The mechanism shown in Fig. 5 is used on a fixture for operating the work-holding clamp. The shape of the work made it necessary to have the clamping foot enter a recess. For this purpose, the clamping foot was required to have a long stroke and be provided with a positive locking device. Side views of the mechanism employed are shown at the left and right of Fig. 5, while an end view is shown in the center of Fig. 5. Slide *A*, which carries the clamping foot (not shown), is dovetailed to a sliding fit in the stationary part *B* of the fixture.

Slide *A* is cut out to form a rectangular hole at one end. The upper edge of the hole has machined rack teeth which mesh with the teeth of pinion *D*; the pinion clears the lower edge of the hole, as shown. Pinion *D* is fixed on shaft *C*, which rotates in part *B*. Shaft *C* is so located that pinion *D* meshes loosely with the rack in slide *A*, so that there is a small amount of backlash. The reason for this backlash will be explained later.

Shaft *C* also carries bar *E*, which is shaped on its outer end to form a cam. Crank-handle *F*, which is fixed to shaft *C*, is used to operate slide *A* through pinion *D*. Slide *A* carries block *G* which supports roller *H* in alignment with cam bar *E*; the cam bar is revolved about shaft *C* with crank *F*.

The work is loaded into the fixture with handle *F* in the position shown in side view at the left of Fig. 5, slide *A* then being at its extreme left-hand position. When handle *F* is turned in a clockwise direction, pinion *D*, acting on the rack in slide *A*, causes the slide to move to the right. As slide *A* resists the effort of pinion *D* to move it, the backlash between pinion *D* and the rack in slide A is taken up in one direction, as shown enlarged at the left of Fig. 5.

In the side view at the right of Fig. 5, slide *A* is shown in its extreme right-hand position, at which point the work has been clamped firmly in position. It will be noted that, in this position, cam bar *E* has been rotated in a clockwise direction to make contact with roller *H*. Up to this point, the movement of slide *A* has been accomplished by the action of pinion *D* on the rack. Slide *A* now receives its motion from cam bar *E* through roller *H* and block *G*. As the rise of the cam surface on the end of bar *E* is very gradual, the effect is to produce a more powerful clamping action than would be obtained through pinion *D*.

In the position shown in the side view at the right of Fig. 5, cam bar *E* has pushed slide *A* slightly ahead of the position it would occupy if driven by pinion *D* at this point, so that the backlash between the gear teeth is taken up in the opposite direction from that shown in the side view at the left of Fig. 5. This condition is shown by the enlarged view at the right of Fig. 5. In this position, the back pressure on slide *A* cannot cause a reverse rotation of pinion *D*, as the teeth in the rack in slide *A* cannot make contact with the teeth of pinion *D* on the side that would cause reverse rotation. Slide *A* is locked firmly against the work by the wedging action of cam bar *E* between roller *H* and shaft *C*. This design provides a long quick travel of the locking slide, with positive locking, in a compact space.

Vise Operated by a Rack and Pinion with Locking Motion.—A rack and pinion that locks when it encounters resistance to motion, but that is free to operate when the resistance is removed or when its motion is reversed, can be used in a variety of shop mechanisms. For example, it can be applied on a vise, as shown in the accompanying illustrations, when a rapid change of opening is desirable and positive locking in any position is necessary.

As shown in the plan and front elevation views of the vise, Fig. 6, part *A* is grooved to carry rack *B*, to which the movable jaw *C* is attached. Pinion *D* fits in a recess in block *E*, and is keyed to shaft *F*, as is handle *G*. Block *E* is free to slide in a recess in the base; this recess and the contacting edge of the block are machined at an

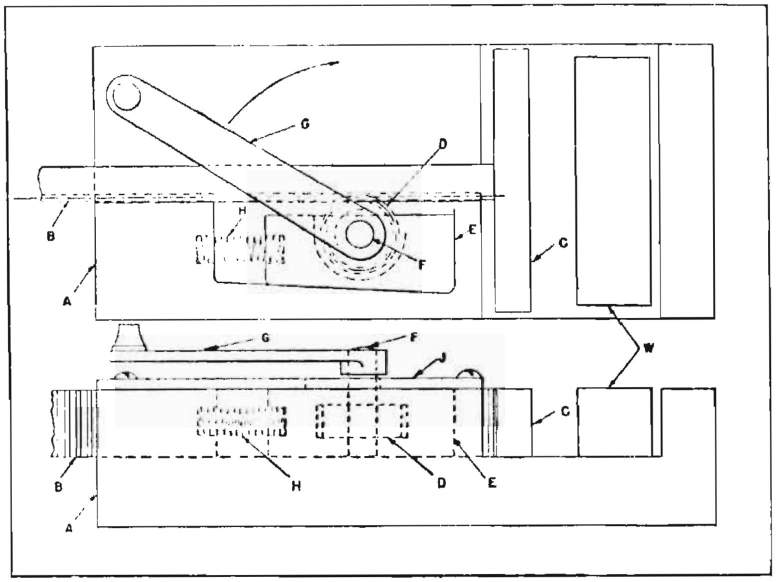


Fig. 6. Vise Equipped with a Rack and Pinion that Is Designed to Permit Rapid Clamping and Positive Locking in Any Position.

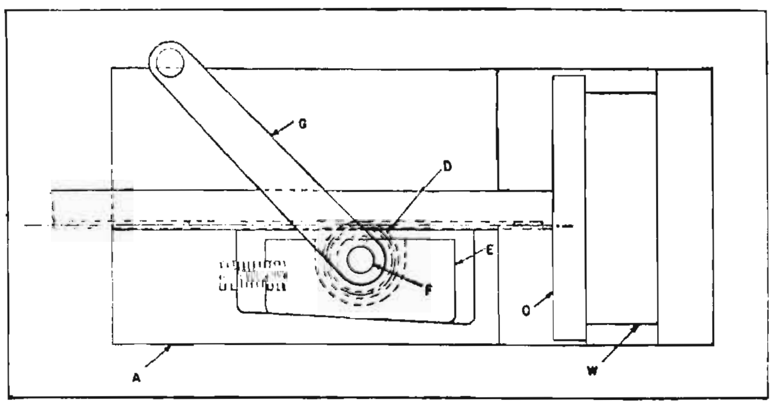


Fig. 7. When Jaw C Comes in Contact with the Work, Rotation of Lever G Wedges Block E Between the Rack and the Wall of the Recess, Thus Locking the Work in Place.

angle to the pitch line of the rack. Spring *H*, which is in compression, exerts pressure against the block and holds it against the right-hand wall of the recess with a force sufficient to overcome the frictional resistance of the rack to movement. Plate *J*, shown in the front elevation view, serves as a retainer for the assembly.

In operation, as handle *G* is turned in the direction of the arrow, the rotation of the pinion causes the rack to move to the right. This movement continues until jaw *C* comes in contact with the work-piece *W*, when further movement is prevented. Continued turning of the handle causes block *E* to be moved to the left, as shown in Fig. 7, by the force exerted by the pinion. Owing to the wedging action of the block, the rack is firmly locked against any reverse movement. However, when handle *G* is turned in a counter-clockwise direction, the block returns to its original position and the rack is freed for releasing the work.

A Non-Reversible Rack and Pinion Motion.—A rack and pinion motion is ordinarily reversible, so that rotation of the pinion will cause straight-line movement of the rack, and power applied to the rack will produce rotation of the pinion. In the assembly illustrated, movement of the rack can be obtained only by rotation of the pinion, and the mechanism is locked to prevent movement of the rack against the pinion.

Referring to Fig. 8, the bearing *A* supports shaft *B*, which carries the hand-crank *C*, pinned to the shaft, and pinion *D*, which is free to rotate. Collar *E* is also pinned to the shaft and serves to retain pinion *D*. Bearing *A* is counterbored to receive the closely wound spring *F*. The two ends of the spring are turned in, as shown in Fig. 9, which illustrates the counterbored end of bearing *A*. The clearances are exaggerated in the illustration for the sake of clarity.

Spring *F* is made to a diameter which requires that it be pressed into the counterbore of bearing *A*, so that a certain amount of frictional resistance to movement will be applied. Part *G* is a disk, keyed to shaft *B*, from which a segment-shaped section has been removed. The

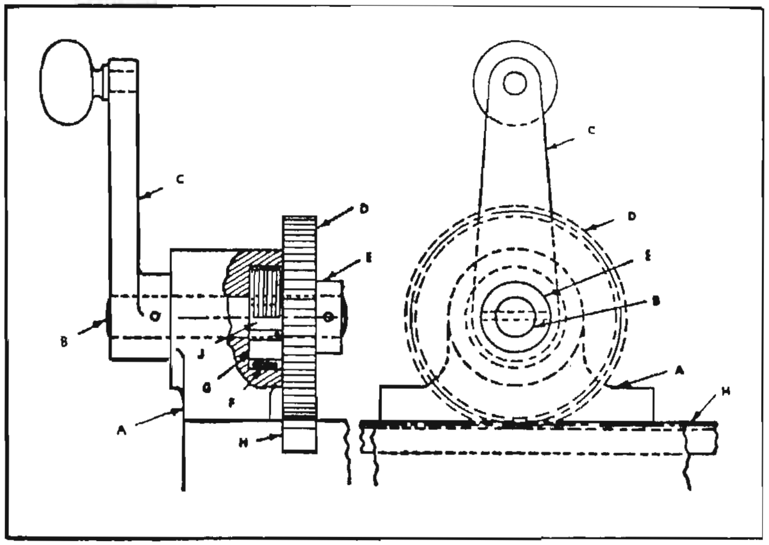


Fig. 8. A Rack-and-pinion Motion Assembly in which Movement of the Rack can be Obtained Only by Rotating the Pinion.

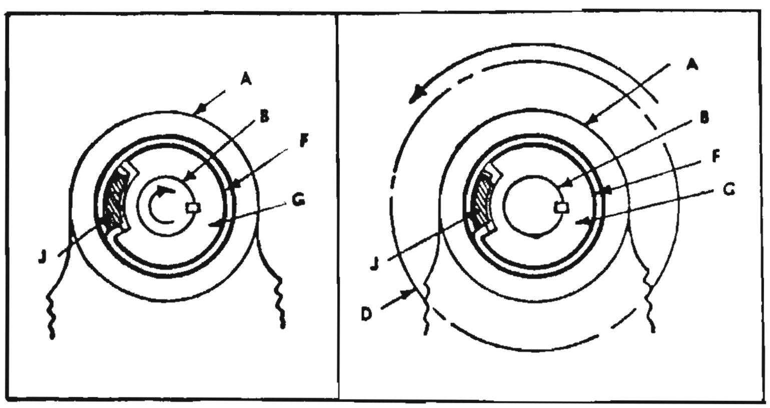


Fig. 9. (Left) View of Counterbored End of Bearing A Indicating Rotation of the Pinion in the Direction Shown by Arrow, which Transmits Motion to the Rack; (Right) View of Counterbored End of Bearing A Illustrating Operation of Locking Action which Prevents Movement of the Rack.

shape of disk *G* is shown clearly in Fig. 9. Pinion *D* has on its inner side a segment-shaped projection *J*, which enters the counterbore between the ends of spring *F* and the segment-shaped section which has been removed from disk *G*.

The two diagrams in Fig. 9 illustrate the operating principle. In the left-hand diagram, shaft *B* is being rotated by the hand-crank in the direction indicated by the arrow. As disk *G* is keyed to shaft *B*, it rotates with the shaft, one side of the cut-away section contacting one end of spring *F*. Since spring *F* is a tight fit in the counterbore of bearing *A*, the end of the spring at first offers resistance to further turning of disk *G*, but continued turning effort applied to the crank *C* and transmitted through disk *G* to the end of the spring, causes the latter to be wound tighter, thus slightly reducing the diameter of the spring. When this occurs, spring *F* is free to turn with disk *G*.

As projection *J* on pinion *D* is in contact with the end of spring *F*, and as pinion *D* is free to rotate on shaft *B*, pinion *D* is likewise carried around with disk *G*, the motion thereby being transmitted to rack *H*. When crank *C* is turned in the opposite direction, a similar effect is produced. Motion may, therefore, be transmitted to rack *H* by turning crank *C* in either direction.

The locking action operates as follows: If force is applied to rack *H* to rotate pinion *D* in the direction indicated by the arrow in the right-hand diagram of Fig. 9, the turning effort of the pinion will be applied to the end of spring *F* through projection *J* in the reverse direction to that shown in the left-hand diagram of Fig. 9. This causes spring *F* to unwind, thus increasing the diameter of the spring, so that the pressure against the walls of the counterbore in bearing *A* is increased. This pressure becomes greater as the force applied to the rack is increased, preventing movement of *H* in either direction.

In both diagrams of Fig. 9, the space between the end of spring F and the cut-away portion of disk G is greatly exaggerated. It is only necessary to provide sufficient clearance to permit a slight movement of the ends of the spring. The required clearance can be determined by trial.

Rack-and-Pinion Mechanism with Self-Locking Non-Reversing Feature.—In the operation of a wire-forming machine, it is frequently necessary to change the adjustment of a guide manually. The adjustment must be made quickly and must be maintained without any additional manual locking operation. The rack-and-pinion mechanism shown in Fig. 10 was designed to provide the required

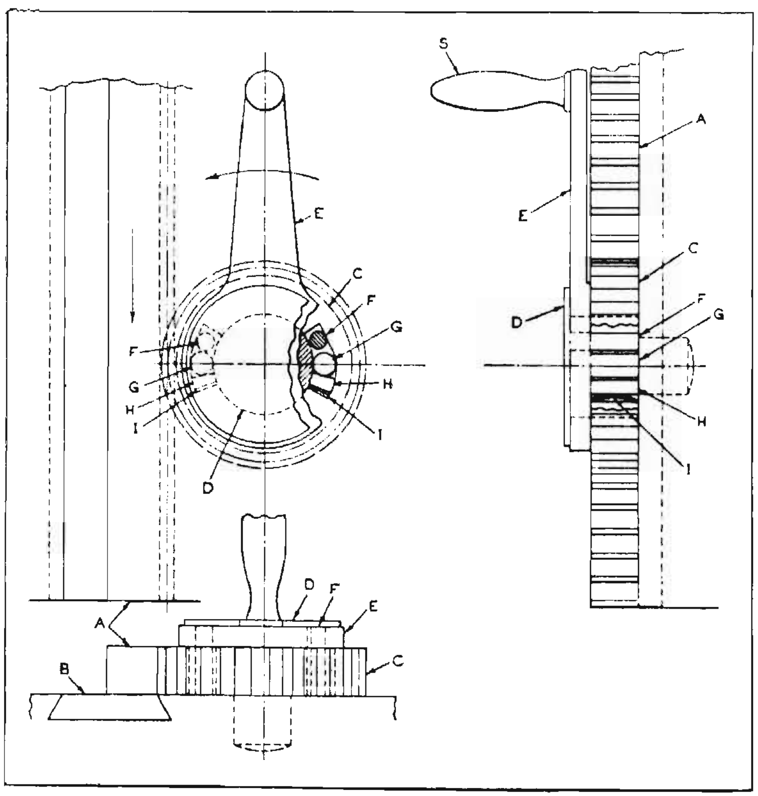


Fig. 10. (Left) End and Plan Views of Rack-and-pinion Mechanism with Self-locking Feature; (Right) Side View of Mechanism.

speed and ease of adjustment with the necessary locking feature. With this arrangement, the wire guide can be adjusted in either direction by turning handle S in a clockwise or a counter-clockwise direction. The mechanism is automatically locked in position.

Referring to the plan and end views of the arrangement shown at the left of Fig. 10, rack A is attached to slide B, which carries the wire guide that is to be adjusted. Stud D, locked on a stationary part of the machine, carries pinion C which meshes with rack A. Pinion C has grooves in the bore at two points, in which rollers G, blocks H, and springs I are assembled. The grooves in pinion C are cut deeper toward one end, so that rollers G will be free at the deeper end but will wedge tightly toward the center at the shallow end, forming the conventional type of free-wheeling clutch which permits free movement in one direction and locks or transmits motion in the other.

Blocks H are backed up by springs I, which tend to force rollers G toward the shallow end of the grooves, thus insuring a positive wedging action. It will be noted that the wedging action in the two grooves takes place when the driving member is rotated in opposite directions, so that gear C is normally locked against rotation in either direction.

Lever E, which is free to swing on stud D above gear C, carries two pins F which project into the grooves in gear C behind rollers G. A slight clearance is provided between pins F and rollers G, so that lever E will have a small amount of free movement. When lever E is moved in the direction indicated by the arrow, pin F on the left comes in contact with roller G, moving it toward the deeper section of the groove in gear C and thus eliminating the wedging action on that side. As the wedging action on the opposite side takes place in the opposite direction only, gear C is now free to rotate with lever E, causing rack A and slide B to be moved in the direction shown by the arrow. When lever *E* is rotated in the opposite direction to that indicated in Fig. 10, the pin *F* on the right-hand side comes in contact with roller *G*, eliminating wedging action on that side and freeing gear *C* to rotate with lever *E*.

When lever *E* is released, the wedging action of both rollers *G* again becomes effective in locking pinion *C* against rotation in either direction. As very little movement of rollers *G* is required for effective locking, rack *A* is locked against movement in either direction with almost imperceptible backlash, except when lever *E* is moved.

Locking and Releasing Mechanism for Traveling Carriage.—The stitching mechanism of a machine used for stitching a wire through a fabric must be firmly locked in position during the stitching operation, but must be immediately freed for resetting when the operation is completed. These requirements were effectively met in a simple manner by the mechanism shown in the accompanying illustrations. In Fig. 11 are rudimentary views of the machine with the stitching mechanism omitted to show the driving mechanism, which automatically locks and releases the stitching mechanism in its various positions.

The stitching mechanism is supported on carriage *A*, which rolls freely on two rails *B*, and is operated by gear *H*, which is also mounted on carriage *A*. Gear *H* is rotated by square shaft *G*, supported by bearings *C*. The motive power is furnished by belt *F*, operating on pulley D, which is keyed to shaft *G*. Loose pulley *E* is provided for disconnecting the power. Plate *J*, attached to bearing *C* at the right-hand end of the machine, is used for operating the disconnecting mechanism, to be described later.

In Fig. 11, belt *F* is shown on the loose pulley *E*. With the belt in this position, no power is applied to shaft *G*, and carriage *A* is free to be moved longitudinally to any position in its range of travel. Square shaft *G* passes through a square hole in gear *H*, to which it transmits

motion. The hole in gear *H* is somewhat larger than shaft *G*. During idle period of the stitching head, shaft *G* is located symmetrically within the square hole in gear *H*, as shown at upper left, Fig. 12, clearance space permitting free longitudinal movement of carriage *A*, Fig. 11.

When belt *F* is shifted to the tight pulley *D*, causing shaft *G* to rotate, the latter carries gear *H* with it, the

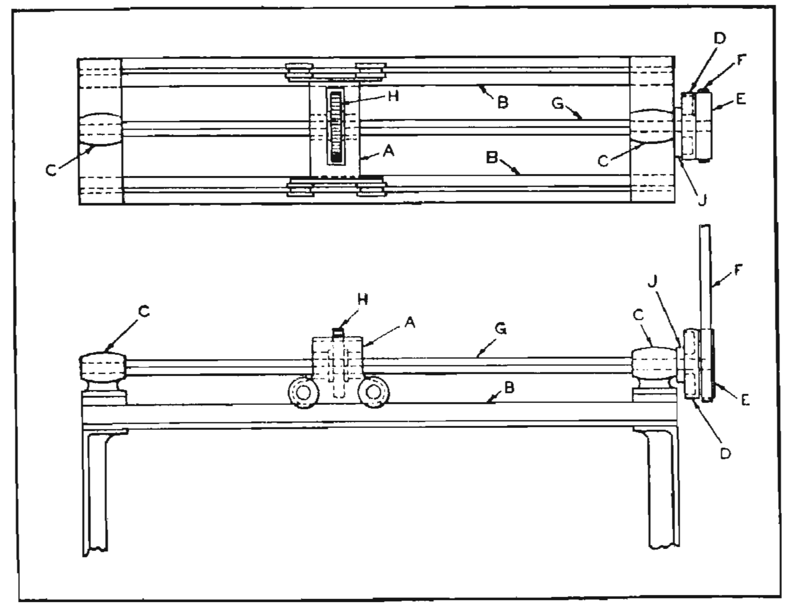


Fig. 11. (Top) Plan View of Wire-stitching Machine Frame and Carriage; (Bottom) Front View of the Machine.

corners of shaft *G* acting as driving members in the manner shown at the upper right of Fig. 12. The frictional resistance to rotation of the stitching head produces a sufficiently powerful wedging action of shaft *G* in the square hole in gear *H* to lock carriage *A* firmly against longitudinal movement. This condition exists as long as shaft *G* continues to rotate.

The mechanism that disengages carriage *A* from shaft *G* to permit longitudinal movement is shown in Fig. 12.

This illustration shows driving pulley *D* as viewed from the side next to the machine. The internal hub of pulley *D* carries lever *I*, which is clamped to it with sufficient tension to support its own weight, so that the lever and the pulley tend to rotate as a unit. However, the tension is light enough to be easily overcome by the driving power exerted by belt *F* on pulley *D*.

The split bore in lever *I* is lined with leather to prevent scoring the hub of pulley *D*. Plate *J*, the position of which is indicated by dotted lines in this view, is fastened rigidly to bearing *C*, shown in the lower or front view in Fig. 11. The outer end of lever *I* carries pin *K* which passes through a slot in plate *J*.

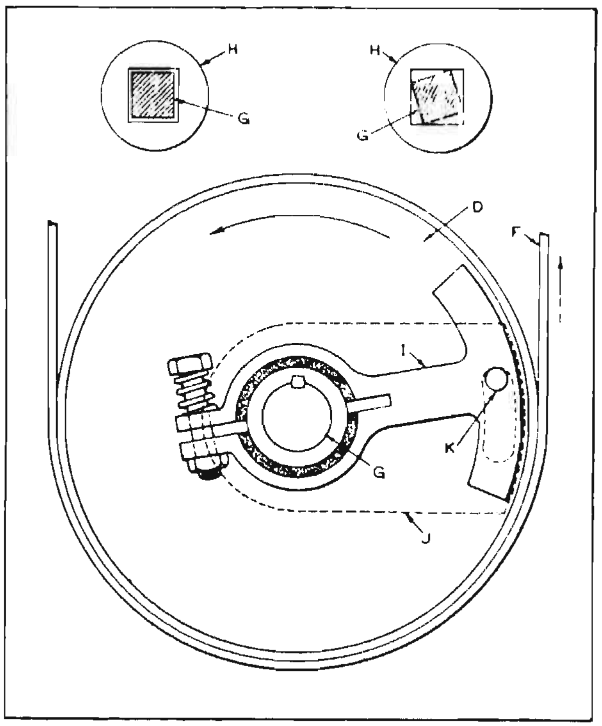


Fig. 12. (Top) Diagrams Showing Shaft G in Unlocked and Locked Positions in Gear H; (Bottom) Mechanism for Unlocking Shaft G.

In operation, belt *F* causes pulley *D* to rotate in the direction indicated by the arrow. The friction of lever *I* on the hub of pulley *D* tends to rotate the lever; but as pin *K* comes in contact with the upper end of the slot in plate *J*, further rotation of lever *I* is prevented, as shown in Fig. 12, this position of lever *I* being maintained throughout the stitching operation, or as long as pulley *D* rotates.

When belt *F* is shifted to loose pulley *E*, Fig. 11, the unbalanced weight of lever *I* causes pulley *D* to reverse its direction of rotation until pin *K* comes in contact with the lower end of the slot in plate *J*, at which time shaft *G* has again returned to its unlocked position in gear *H*, as shown at the upper left of Fig. 12. In this manner, carriage *A* is automatically freed for repositioning at any point of its cycle.

Mechanism for Operating a Floating Jaw Vise.—Fig. 13 shows a vise used on a routing machine for holding a wood block *A* while a cavity is being routed out, as indicated by the dot-and-dash lines at *B*. The jaws *C* and *D* of this vise are operated by a mechanism that permits them to float to suit the work, locking firmly when the work is finally gripped. The work *A* is located on two pins *E* which enter two previously drilled holes. The jaws *C* and *D* prevent the wood from bulging under the pressure of the cut where it closely approaches the sides of the piece.

As the routing must be accurately positioned relative to the holes and as the width of the work varies considerably, it is necessary that the jaws *C* and *D* have a floating action. That is, the jaws must grip the work in the position in which it is located by the pins *E* before they are clamped to the fixture bed *F*. The jaws *C* and *D* are slidably mounted on base *F* to provide the required floating action. The base *F* is slotted to receive keys machined on the under side of the jaws. Retaining plates *G* and *H* are screwed to the under side of the keys to hold the slidable jaws in place. The blocks *I* and *J*, located in slots in the jaws, swivel on pins at their upper end, and are held at the back of the slots by means of springs *K*. Blocks *I* and *J* carry at their lower ends the swivel-pins *L* and *M*, which are threaded to fit the right- and left-hand threads on the rod *P*. Rod *P* is equipped with a handwheel *W* which is turned clockwise for clamping the work. The blocks *I* and *J* are T-shaped, being wider at the bottom, as shown in the end view to the right. The lower ends of blocks *I* and *J* are therefore wider than the slot in base *F*.

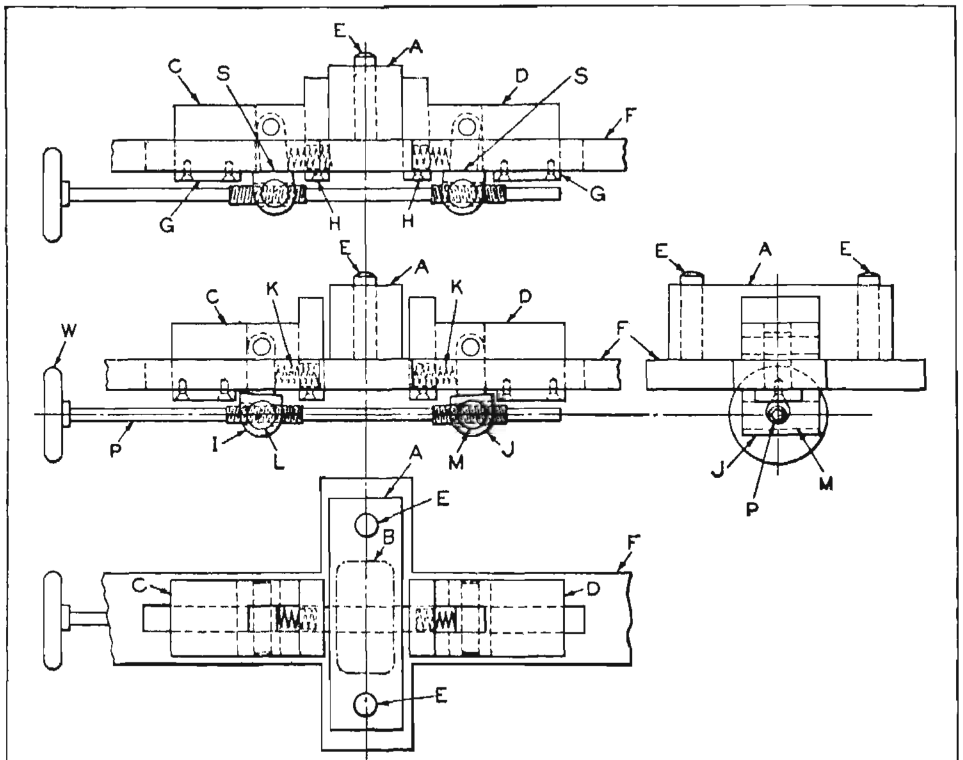


Fig 13. Mechanism that Causes Floating Jaws C and D to Grip Work A and then Grip Base F through the Clamping Action of Blocks I and J.

The two lower views in Fig. 13 show the vise in the open position, with work *A* located on pins *E*. In this position, the jaws and their blocks *I* and *J* are bound together by rod *P* and slide together as one unit on base *F*. When handwheel *W* is turned clockwise, the action of the right- and left-hand threads on rod *P* will throw swivel-blocks *I* and *J* toward each other. As pins *L* and *M* are prevented from swiveling by the resistance of springs *K*, jaws *C* and *D* are also thrown together.

When either jaw comes in contact with the work, its movement ceases, and the action of the screw is transmitted to the other jaw until they are both in contact with the work. Further turning of the handwheel causes blocks *I* and *J* to swivel on their upper pins, compressing the springs, so that at this point, work *A* is held by spring pressure. Continued turning of handwheel *W* causes blocks *I* and *J* to swivel still further until the enlarged or T-sections at their lower ends come in contact with the under side of base *F*, as indicated at *S* in the upper view, thus locking jaws *C* and *D* firmly in position.

Mechanism for Clamping and Releasing a Spring-Actuated Tailstock Center.—A mechanism designed for releasing a spring-actuated center and for clamping the center in the work-holding position is shown in Fig. 14. The fixture on which this mechanism is used is designed for holding the thin wooden part *G*, on which a routing operation is performed. The pointed ends of the work are supported on centers, the rear center being located in the block *H*. Although there is slight variation in the length of the wooden pieces, they must be held firmly in position, but without sufficient pressure to distort them.

Referring to the view to the left in Fig. 14, the rack *A* meshes with the gear *B* which turns freely on the flanged bushing *D*, mounted on stud *C*. The bushing *D* is slidably keyed in the bearing member *J* which supports the assembly. Stud *C* is threaded into the collar *F* which is fastened to bearing J. Handle E is pinned to stud C and serves to advance it into the threaded collar F. The work G is supported by the block H at the end of rack A. The opposite end of the work is similarly supported by a stationary block. A spring L furnishes the required pressure to support the work.

To place a piece of work between centers, the handle E is turned to the left into contact with the pin K in gear B, so that the gear turns with the handle. Then movement of gear B moves rack A to the right. Next, the

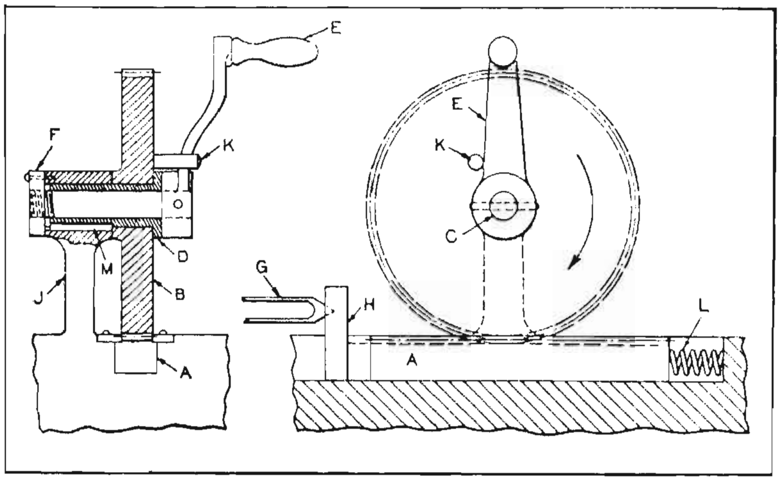


Fig. 14. Mechanism for Clamping and Releasing Tailstock Center.

work is located between the centers and handle E turned to the right, as indicated by the arrow. This permits the pressure of spring L, acting on rack A, to support the work with the maximum permissible pressure. Although this pressure will support the work, it is not sufficient to resist the pressure of the cut and some method of positively locking rack A is necessary.

Continued movement of handle E in the direction indicated by the arrow causes stud C to be screwed deeper into collar F, so that the hub of handle E, acting on the

flange of bushing D, locks gear B against the end of bearing J. As gear B now is locked, rack A is prevented from moving. Thus the support H holds the work G in position under the pressure exerted by the cutting tool. As bushing D is restrained from rotating by the key M, the rotary movement of handle E is not transmitted to bushing D, and the endwise pressure applied to the work when it is placed between centers consists only of that exerted on rack A by spring L.

Mechanism for Adjusting Arc-Shaped Levers around Rotating Cylinders.—The mechanism shown in Fig. 15 is designed to enable the arc-shaped members B of levers A to be closed around the rotating work X in such a manner as to smooth out a covering material placed on the cylindrical surface.

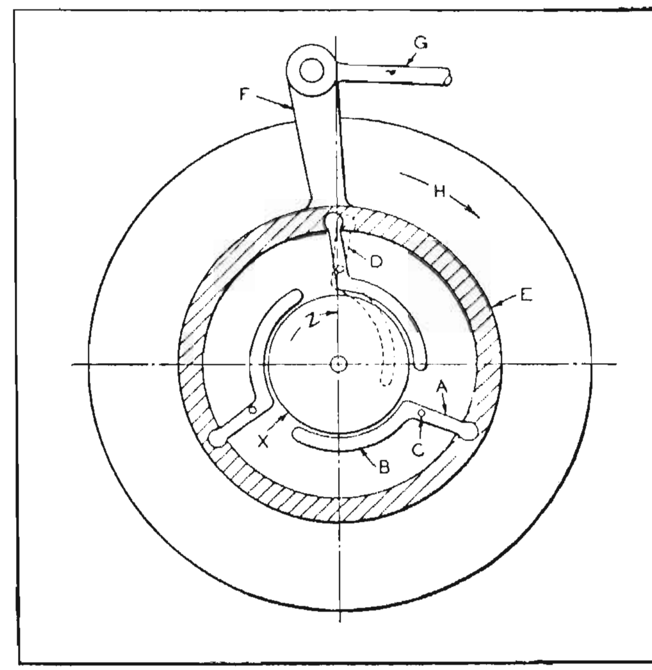


Fig. 15. Mechanism for Closing Arc-shaped Members B around Rotating Cylinder X.

The three smoothing levers A have their extensions B pivoted about pins C as centers, and thereby operate on the work to smooth out the covering material. The levers are shown at their outer positions, the dotted lines at D indicating the contracted position which would be assumed if work X were not in position. The levers are brought together under spring tension in the following manner: Each of the three levers A has its outer end located in a socket in a revolvable flange E having an extension arm F. Through the medium of the connecting-rod G, the flange is rotated in the direction indicated by arrow H to close the jaws. At the pulling end of the connecting-rod there is a tension spring, not shown. Thus when the smoothing fingers come in contact with the work, which revolves in the direction indicated by arrow Z, pressure is applied under spring tension to the levers B.

Milling Machine Spindle Brake and Circuit-Breaker Mechanism.—The purpose of the mechanism shown in Fig. 16 is to provide a means for locking the spindle of a milling machine positively when changing the milling cutter, in order to insure safer operation. When the machine is running, the milling spindle A is rotated by means of disk B. Disk B is provided with six holes C for the locking pin D which serves to prevent rotation of the spindle when it is raised to engage one of the holes in the disk. Pin D is actuated by means of lever E and the operating lever F which has a handle G.

A spring H presses pin D against disk B, so that it automatically engages one of the indexing holes, as shown in the middle view of Fig. 16. The braking lever I has the same center as lever F, but is independently fixed on a hollow shaft which terminates in a bevel gear K, shown in the lower view of Fig. 16. This gear meshes with another bevel gear Z on shaft M. The latter gear is connected with a crank disk N. Between the rotating pin U and another fixed link is attached a brake-band which

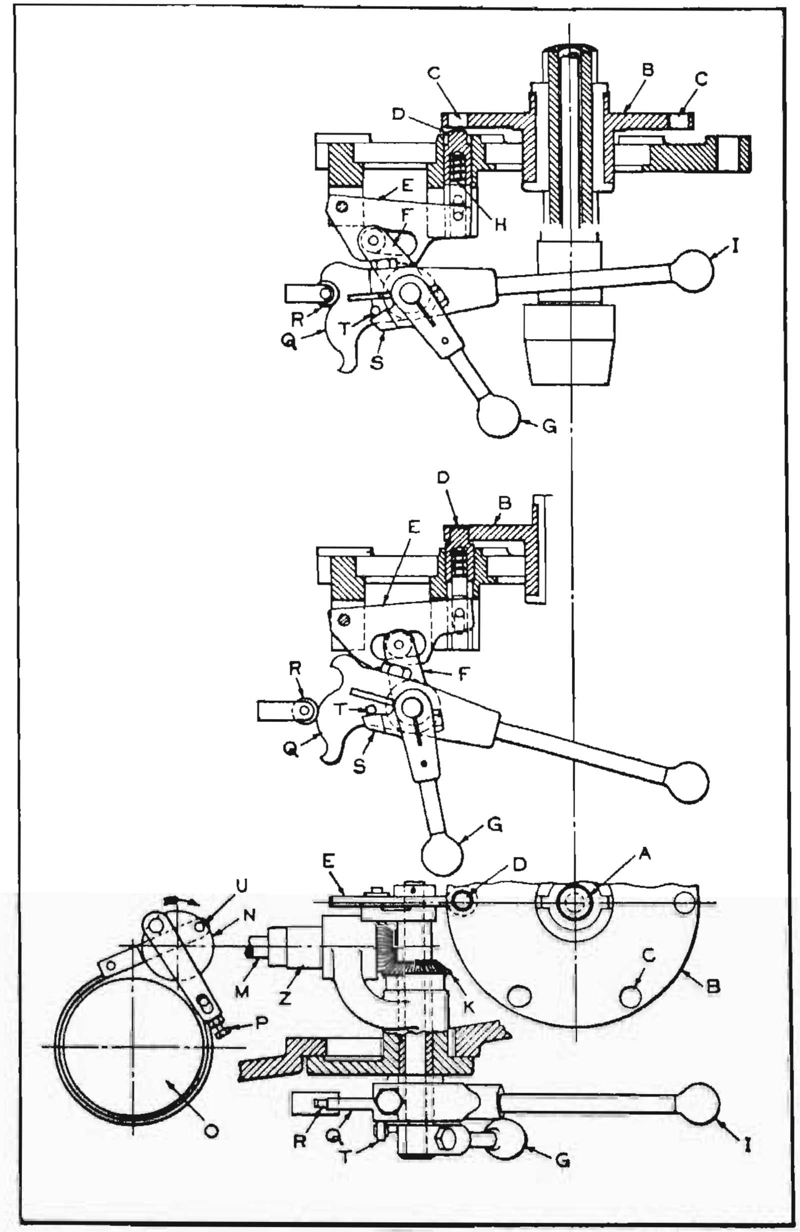


Fig. 16. Milling Machine Spindle Brake. (Top) Operating and Braking Levers G and I Disengaged to Allow Spindle to Rotate; (Center) Levers Positioned to Lock Spindle against Rotation; (Bottom) Top View of Mechanism and Side View of Brake.

acts upon the brake-disk O. The brake-band can be adjusted by means of a screw P. On the free end of the braking lever is a formed cam Q which actuates the limit switch of the electric motor by means of a roller R. A pin T on lever I and a projection on lever F provide the necessary interconnection.

When the machine is running, both levers are disengaged, as illustrated in upper view of Fig. 16. When brake-lever I is moved downward, roller R is pushed backward by the action of cam Q so that the current supply is interrupted. When lever I is moved downward further, the brake-band is applied, the index lever F remaining in its first position. However, with lever I in this position, it is possible to actuate lever F. Lever F can now be brought into such a position that the nose S comes into contact with pin T. The various members of the mechanism now occupy the relative positions shown in the middle view of Fig. 16, the spindle A being locked to prevent rotation by pin D entering hole C of the disk B.

If the brake-lever band is released by operating lever I, lever F is also returned to its first position by means of pin T. A further lifting movement brings lever I into an almost horizontal position, switching on the current. A false or unintentional movement of the levers is made impossible by having the length of lever F so short that it is first necessary to move lever I.

Automatic Work-Locating Mechanism for Milling Machine.—A large number of pieces similar to the one shown at the left of Fig. 17 are end-milled as indicated at M after the groove N is milled. The operation is performed on a vertical hand miller by setting the cutter to the proper depth and sliding the work into a clamping fixture, the central groove permitting the end-mill to enter, as indicated at the right of Fig. 17. The work-table is then moved to the right and left for cutting, and returned to the central position for removing the work.

As the central groove is but slightly wider than the diameter of the end-mill, it is necessary to stop the table in the position for loading and unloading with a fair degree of accuracy. Fig. 18 shows the construction of an attachment that locks the table positively in the loading position and permits the required travel for cutting.

Referring to Fig. 18, top view, the bar C is fastened to the machine table B, and carries the sliding block D, which is drawn to the left by the spring S. Bar C is notched in the center, as shown more clearly in the middle view of Fig. 18. Block G is fastened to the knee of the

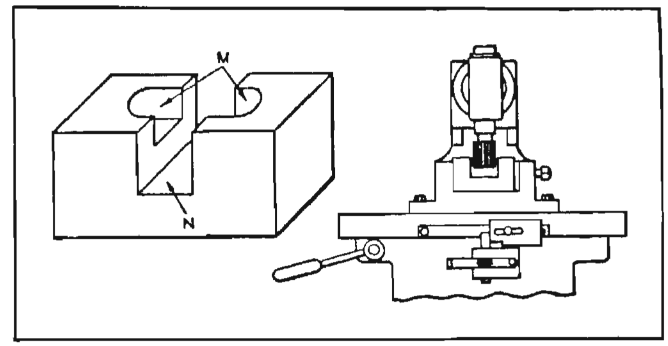


Fig. 17. Automatic Work-locating Mechanism. (Left) Work is End-milled as Indicated at M; (Right) Milling Machine with Fixture and Locating Attachment.

machine and carries the sliding key E, which is actuated by the lever F. The block D is reduced in width at the left-hand end. The travel of the table B is limited by the usual stops, not shown.

Referring to Fig. 18, middle view, which is the starting point of the cycle, the table B is locked by the key E being located in the notch in bar C. After the work fixture has been loaded, the lever F is depressed, withdrawing key E from the notch in bar C and permitting block D to be drawn to the left by the spring. When the lever F is released, key E comes to rest on the step on the end of block

D, as shown in Fig. 18, top view. The table *B* is then moved to the left, the block *D* being restrained from movement by key *E*.

When the travel to the left has been completed, and the table *B* is moved to the right, block *D* is permitted

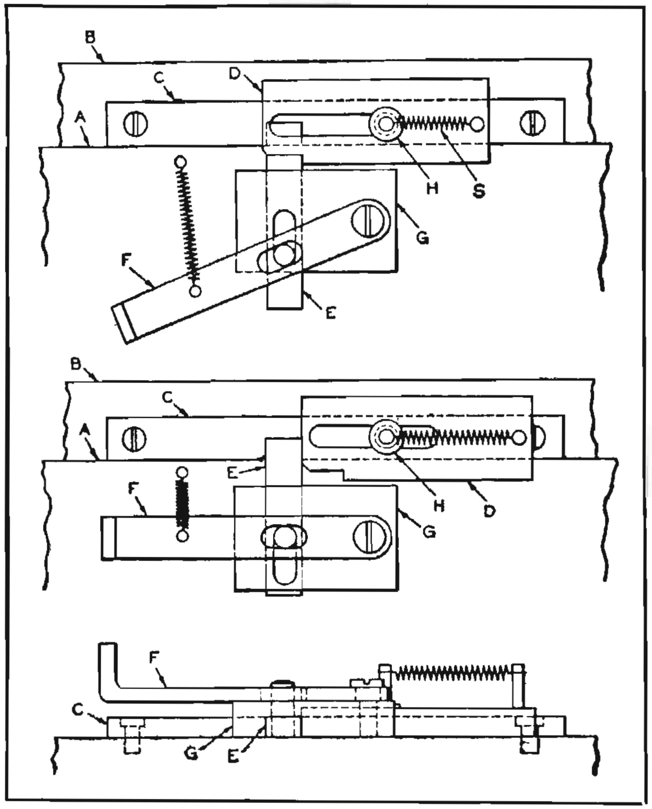


Fig. 18. (Top) Work Locating Attachment with Block D Drawn to Left by Spring S; (Center) Locating Attachment at Starting Point of Cycle; (Bottom) Bottom View of Attachment.

to slide toward its original position until the end of the slot comes in contact with the stud *H*. Continued movement of table *B* causes block *D* to be withdrawn from the end of key *E*, which then slides onto the solid portion of the bar *C*, the key *E* having been carried over and to

the right of the notch by the block *D*. When table *B* reaches the end of its right-hand travel, it is returned to its central position, the key *E* resting against the end of block *D* and sliding on bar *C* until it reaches the notch in bar *C*. Key *E* then enters the notch in bar *C*, again locking the table in the loading position.

Work-Locating Mechanism for Milling Machine.—The mechanism shown in Fig. 19 is designed to accomplish the same purpose as that just described. It is very accurate, can be operated rapidly, and has a wider range of adjustment. Machine tables *B* that do not have bolt slots in the front face can be fitted with a slotted plate *C* to which stops *D* are clamped by bolts *F*. Plate *C* is held in place on the table by two machine screws. It will be noted that a shallow double-angle or beveled slot *J* is machined in the center of plate *C*. This slot is engaged by the point of latch *G* through pressure exerted by spring *I*. This arrangement stops or locates the table in the correct position for loading and unloading the work without inter-

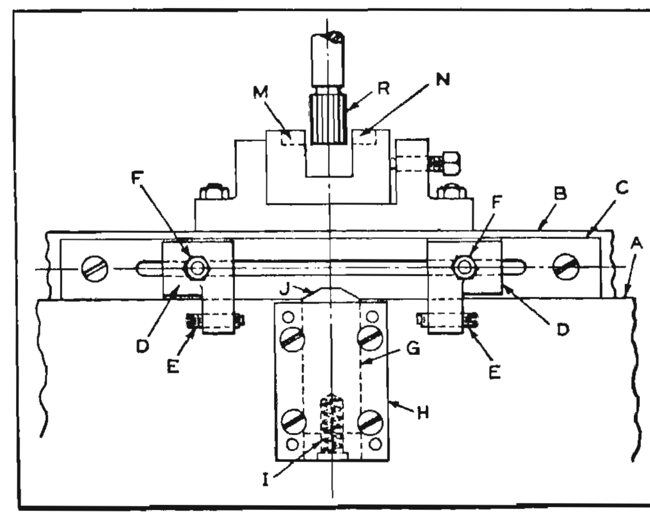


Fig. 19. Milling Machine Equipped with Mechanism for Locating Table in Correct Position for Loading.

ference of the cutter R with the sides of the slot in which the cavities are to be milled.

After the work is clamped in the fixture, cavities M and N are machined by the end milling cutter R. The pressure exerted by spring I is light enough to enable the machine or hand feed to disengage latch J. The machine feed can be used to within 1/32 inch of the final stops, after which the table is fed by hand until the points of set-screws E come in contact with latch casing H on the base A.

CHAPTER 6

Reversing Mechanisms of Special Design

Described in this chapter are various arrangements for obtaining reversal of motion. These include mechanisms for reversing a chain driven table; for providing a dwell at each reversal; for reversing a carriage traversing screw; for automatic stroke reversing in a variable speed traversing motion mechanism; for reversing a cable winding machine; for reversing a cross-head feed screw at any predetermined point; for varying the point of reversal; and a positive type of clut . that releases readily under heavy load. Other reversing mechanisms are described in Chapter 6 of Volume I and Chapter 7 of Volume II of "Ingenious Mechanisms."

Reversing Mechanism for Chain-Driven Table.—The accompanying diagrams illustrate the operation of a table reversing mechanism designed for use on a wire fabricating machine in which the table is driven by a roller chain. As shown in Figs. 1 and 3, the table is reversed or reciprocated by alternately disengaging and engaging toothed dogs M and N with the upper and lower portions of the driving chain at opposite sides of the sprockets. The mechanism, while not applicable to very heavy drives, possesses the advantages of simplicity and the ability to function when long table traversing movements are necessary. A front elevation of the machine and an end view on section XX are shown diagrammatically in Fig. 1. The table A is reciprocated by the chain B, its length of travel being controlled by the positions of the adjustable dogs C and D.

Referring to Figs. 2, 3, and 4, the vertical slide *E* is carried between two blocks *F* and *G*, which are attached to a supporting member *H*, Fig. 1. Blocks *F* and *G* are grooved to carry the horizontal slide built up of the plates *I* and *J*, which are spaced to accommodate springs *U* and *V*, as well as the levers *Q* and *R*. Plates *I* and *J* are held together by rivets. The plates *K* and *L* retain the horizontal slide in the blocks *F* and *G*.

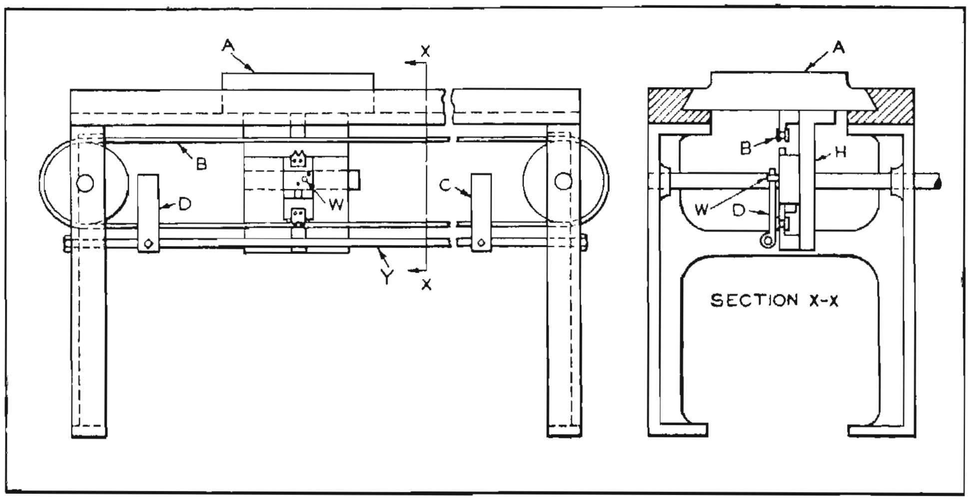
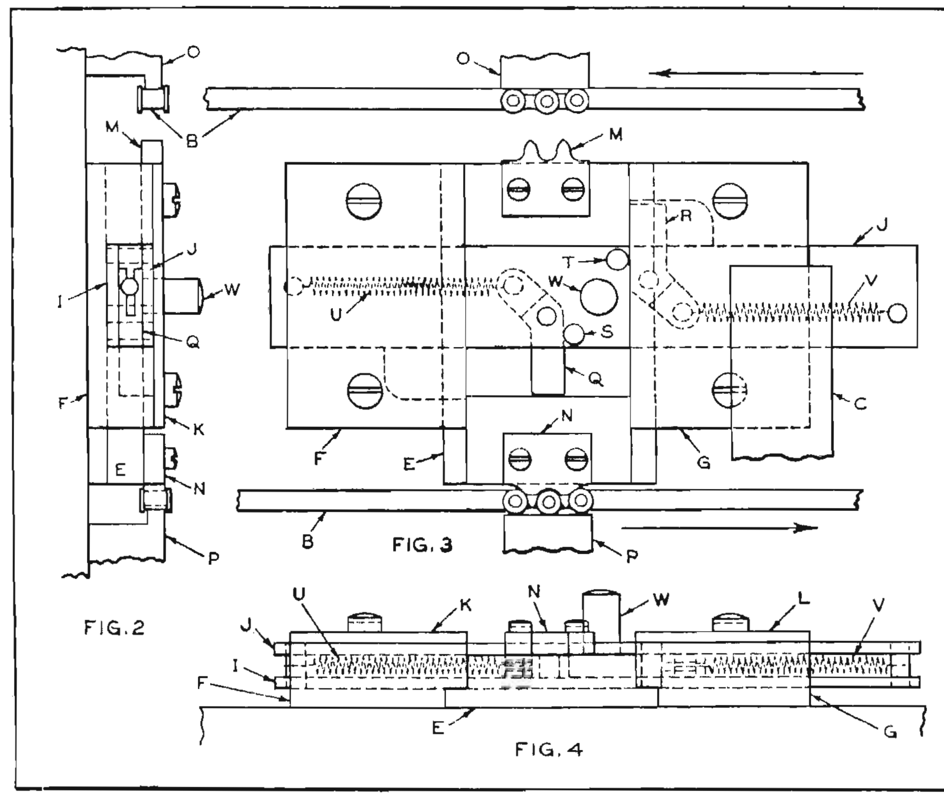


Fig. 1. Table Is Reciprocated by Alternately Engaging Toothed Dogs with Upper and Lower Portions of Driving Chain.

Vertical slide *E* is grooved to permit the horizontal slide to pass freely through it, and carries at its upper and lower ends the toothed dogs *M* and *N*, which are shaped like the teeth of a standard roller-chain sprocket. These toothed sections are alternately inserted between the rollers of the chain, and serve to transmit the movement of the chain to the table *A*. The chain *B* is supported by the guide blocks *O* and *P*, which are attached to the supporting member *H*, Fig. 1, and therefore travel with the table *A*. The levers *Q* and *R* are supported freely on pins between plates *I* and *J* of the horizontal slide, and are

normally held in contact with the pins *S* and *T* by the springs *U* and *V*. The pin *W*, carried by the horizontal slide, serves to operate the mechanism when it comes in contact with the adjustable dogs *C* and *D*, Fig. 1. It will be noted, by referring to Fig. 3, that lever *Q* projects out of the horizontal slide into the widened groove in the vertical slide *E*, while lever *R* projects into the recess formed in block *G*.

Fig. 3 shows the mechanism in the normal driving position, the teeth of dog *N* being engaged between the rollers of chain *B*, which is supported at that point by the guide *P*. Dog *N* cannot be disengaged from chain *B*, as the



Figs. 2, 3, and 4. Three Views of Table Mechanism in Normal Driving Position with Teeth of Dog N Engaging Lower Portion of Roller Chain.

position of lever Q prevents slide E from moving in its groove. As that side of chain B into which the teeth of the dog N are engaged is moving in the direction indicated by the arrow, table A is caused to travel with it to the right until pin W makes contact with the fixed dog C. This restrains the horizontal slide from further movement with the assembly.

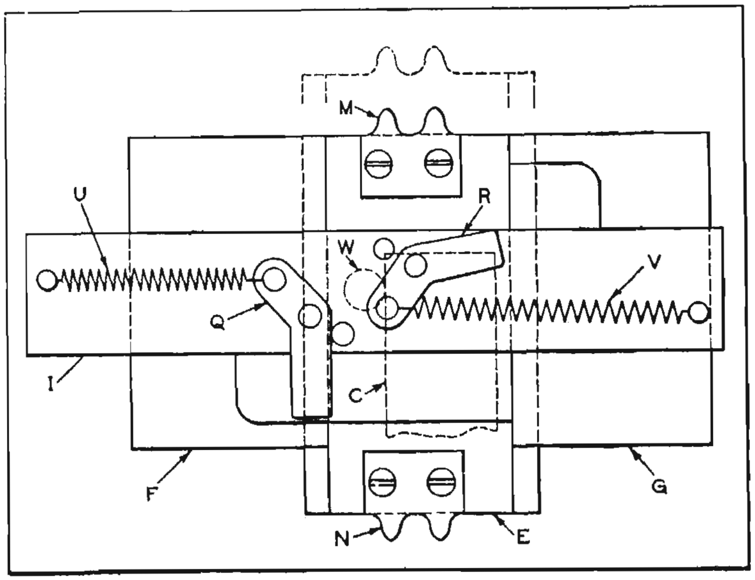


Fig. 5. Table Reversing Mechanism that Serves to Raise Slides E to Position Shown by Dotted Lines.

Continued movement of table A causes lever R to be turned on its pivot pin and enter the groove in slide E, as shown in Fig. 5. In this view, plates K, L, and J have been removed and the positions of pin W and dog C are shown by dotted lines. Lever R is shown held against the upper edge of the groove in slide E by the increased tension of spring V produced by the swinging movement of lever R. However, no movement of slide E takes place, as it is still locked in place by lever Q.

Further movement of the table causes lever Q to pass beyond the lower edge of the groove in slide E, at which time the tension of spring V, acting against the upper edge of the groove in slide E through the lever R, causes slide E to be raised quickly to the position shown by the dotted lines. The teeth of dog M entering between the rollers of the upper length of chain B carry the table in the opposite direction, or to the left.

Table A continues its movement to the left until pin W comes in contact with dog D, locked in position on the bar Y, Fig. 1. At this time, lever Q acts on slide E, again bringing about the engagement of dog N with the lower part of chain B. This causes table A to move to the right again.

Reversing Mechanism that Provides for Dwell at Each Reversal.—The mechanism shown in Fig. 6 is used to transmit motion from shaft A to shaft I in such a manner that when the direction of motion is reversed, there will be a definite, specified lag or dwell in the transmission of the motion to shaft I, regardless of the point in the cycle at which the reversal takes place. This mechanism is used on a machine for fabricating a wire product.

Referring to the illustration, shaft A, which carries pinion B keyed to it, rotates in the direction indicated by the arrow, transmitting motion in the reverse direction to gear C, which rotates freely on the hub of the machine member. Gear C carries the pin D, which comes in contact with the extended arm of the sector E, the latter member likewise rotating freely on the hub of the machine member. Lever G, which also rotates freely on the same hub, carries the pin F, one end of which enters the slot in sector E. The opposite end of pin F is extended so as to make contact with the lever H, which is keyed to the driven shaft I.

In operation, gear C, rotating in the direction indicated by the arrow, transmits motion to sector E through pin

D. Sector *E*, in turn, transmits motion to levers *G* and *H* through pin *F*. Lever *H* transmits motion to shaft *I*. On the reversal of the motion, and in practically one complete revolution of gear *C*, the pin *D* will come in contact with the opposite side of the extended arm of the sector *E*, causing the latter to rotate with gear *C* in a counter-clockwise direction.

When the end of the slot in sector *E* comes in contact with pin *F*, the lever *G* is likewise caused to rotate with gear *C*. On the completion of the second revolution of gear *C*, pin *F* is in contact with the opposite side of lever *H*, the position of the parts at this point being as shown by dotted lines in the illustration. At this point, the shaft *I* begins to rotate in unison with gear *C*.

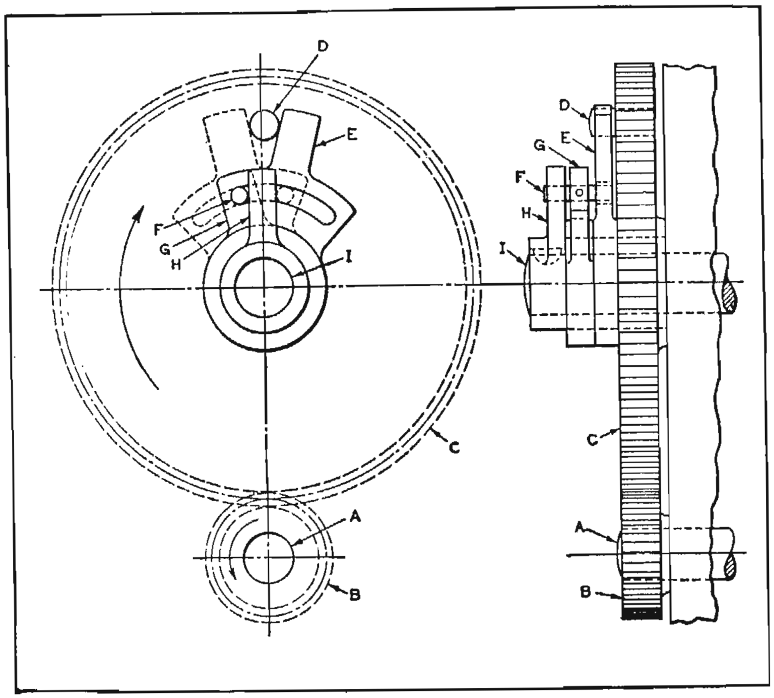


Fig. 6. Gear Drive Mechanism that Provides Dwell for Driven Shaft I at Each Reversal of Driving Shaft A.

It will be noted that in order to change the position of pin *F* from one side of lever *H* to the other, two complete revolutions of gear *C* are required, shaft *I* remaining stationary during the cycle. As the ends of the slot in sector *E* control the exact time at which shaft *I* begins to rotate, slight changes in timing can be accomplished to suit different applications by varying the length of this slot.

Reversing Rotating Motion Mechanism for Carriage Traversing Screw.—A simple reversing-motion mechanism constructed to operate a carriage traversing screw on a wire fabricating machine is shown diagrammatically in Figs. 7 and 8. This mechanism is so designed that there is no lag at the end of the traverse motion. A single-thread traversing screw is used, and there are no springs, sliding keys, or gears. Only moderate speeds are possible, however, due to the momentum developed at high speeds.

On the drive-shaft *A*, Fig. 7, are mounted the mutilated gears *B* and *C*, rotating in the direction indicated by the arrow. The double pillow block *F* carries the driven shaft *H* and the idler shaft *G*. Shaft *H* supports the gear *E*, which meshes with the gear *B* over one-half of its face and over the other half of its face with gear *D*. Gear *D*, on shaft *G*, has one-half of its face in mesh with gear *E* and the other half of its face in mesh with gear *C*.

Gears *B* and *C* each have a few more than half their teeth removed, and are keyed to shaft *A* with their toothed portions in diametrically opposite positions. The length of the gap between the two portions of gears *B* and *C* is governed by the center-to-center distance between shafts *G* and *H*, and must be such that the leading tooth of one gear comes in contact with its mating gear just as the trailing tooth of the other gear terminates its contact.

Referring to Fig. 7, gear *B*, rotating in the direction indicated by the arrow, meshes with gear *E*, causing it and the driven shaft *H* to rotate in the opposite direction,

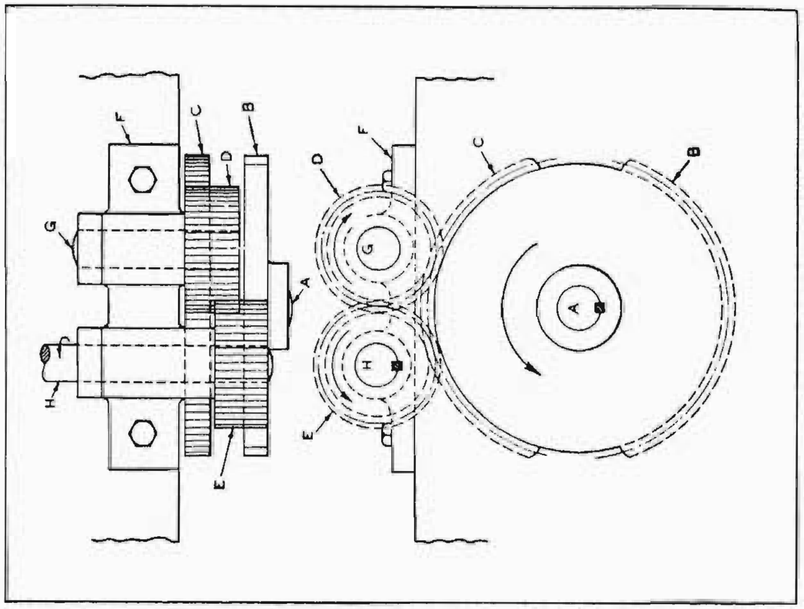


Fig. 8. Mechanism Shown in Fig. 7 with Driving Shaft A Rotating Driven Shaft H in a Counter-clockwise Direction.

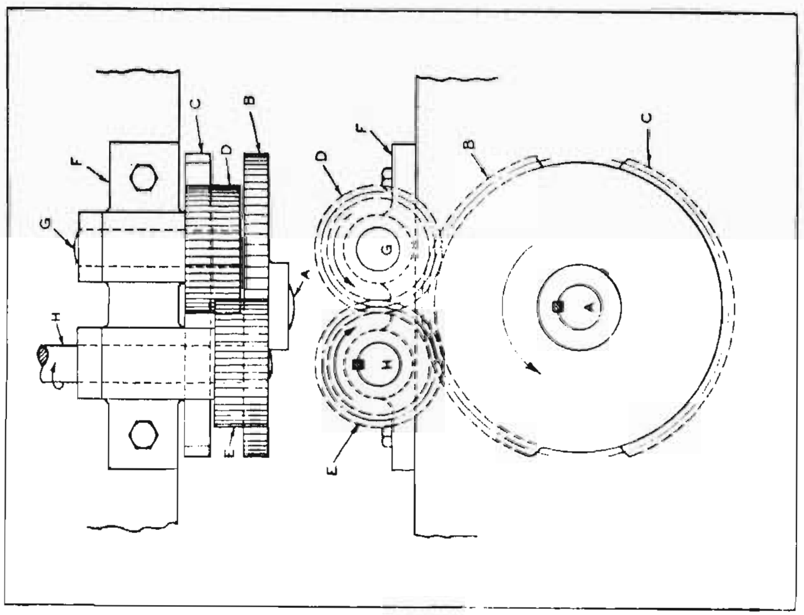


Fig. 7. Mechanism for Reversing Carriage Traversing Screw Shown with Driving Shaft A Rotating Driven Shaft H in Clockwise Direction.

as indicated by the arrow. At this point, the toothless portion of gear C is passing under gear D, and there is no connection between the two, but as gear D meshes with gear E, which, at this point, is being driven by gear B, gear D is caused to rotate in the direction indicated by the arrow. In rotating in this direction, however, no useful work is performed, the gear simply acting as an idler. Thus, the motion of shaft A is transmitted in the opposite direction to shaft H through gear E, while gear D and shaft G idle, or rotate without transmitting any motion.

Referring to Fig. 8, shaft A is shown in the position it occupies after one-half revolution, which brings the teeth of gear C into mesh with gear D. Gear E, meshing with gear D, is rotated in the direction shown by the arrow and, being keyed to shaft H, serves to drive the latter member. At this point, the toothless portion of gear B is passing under gear E, there being no connection between these two members. Thus the rotation of shaft A is transmitted in the same direction to shaft H through gears D and E. The end teeth of gears B and C are modified in shape as required to provide the clearance necessary for practically instantaneous reversal of the direction of rotation.

Automatic Stroke-Reversing and Variable-Speed Traversing Motion Mechanism.—The mechanism shown in Fig. 9 was designed in connection with the development of a machine for grading food products. It transmits a continuous reciprocating motion to the rod A at the rate of thirty strokes a minute, although provision is made for obtaining any desired traversing speed from one to sixty strokes a minute. The maximum length of the reciprocating stroke is 5 inches. The mechanism runs smoothly and is practically noiseless in operation.

The drum B is driven at a speed of 300 revolutions per minute by a motor which transmits power to the mech-

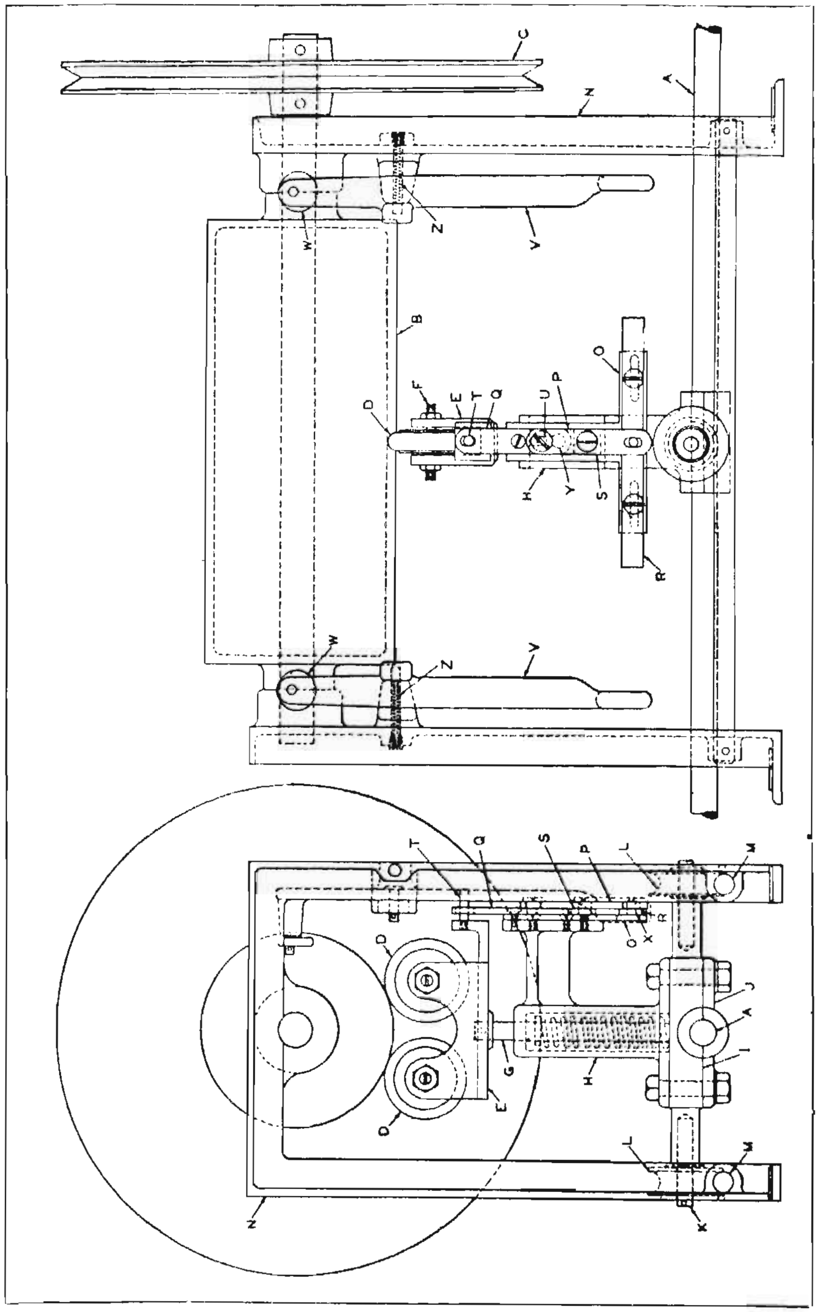


Fig. 9. Mechanism for Automatically Traversing Rod A Back and Forth at from 1 to 60 Strokes a Minute.

anism by means of a belt through the grooved pulley *C*. The drum *B* transmits motion to the rod *A* through contact with the friction-driven wheels *D* mounted in bracket *E*. The wheels *D* are equipped with hardened steel centers and revolve freely on conical screw points *F* which are adjusted and locked in the bracket *E*. The wheels *D* are equipped with semi-hard rubber or rawhide rings for friction driving, which operate smoothly.

Spindle *G*, fitted in bracket *E*, turns freely in the guide casting *H*. The assembly, consisting of the guide casting *H* and castings *I* and *J*, is supported by the grooved wheels *L*, mounted on pins *K* in the casting *I*. The guide wheels roll on the rods *M*, secured to the under frames *N*. The reciprocating rod *A* is clamped between the two castings *I* and *J*.

Bearing plate *O* is fastened to member *H* with countersunk machine screws. Assembled on plate *O* are levers *P* and *Q*. These levers have a compound action, and are actuated by the push-lever *R* which is held in place by screw pins acting in slots in plate *O*. The spacer *S* provides working clearance for levers *R*, *P*, and *Q*. Lever *Q* is pivoted on bearing plate *O*, and at its upper end has an elongated hole which engages the pin *T* in bracket *E*. The lower end of lever *Q* is slotted, as shown at *U*, to engage a pin on lever *P*. The lower end of lever *P* also has an elongated hole which engages a pin *X* on lever *R*.

When drum *B* is rotated, the traction wheels *D*, with their horizontal center lines parallel with each other and with the horizontal center line of drum *B*, revolve at speeds governed by the size and speed of the drum. In the position shown in the illustration, no traversing movement by the traction mechanism takes place, but any turning movement of bracket *E*, with spindle *G* as its axis, immediately causes the bracket *E* and its assembled members to travel to the right or left, as the case may be, and at a speed governed by the angle at which the wheels *D* are set in relation to the horizontal center line of drum *B*.

The travel of the traverse assembly, either to the right or left, brings lever *R* in contact with one of the levers *V*. Continued movement of lever *R* causes the eccentric rollers *W* to move in toward the revolving drum. Contact between the drum and rollers *W* causes the latter to revolve so that the high side of the rollers pushes back the reverse lever *V* with a rapid movement. Lever *V* thus moves lever *R* in the opposite direction. This action transmits, through levers *P* and *Q* and pin *T*, a turning movement to member *E*, which changes the angle at which wheels *D* make contact with drum *B*, thus reversing the direction of travel of rod *A*. This series of reverse movements is repeated when lever *R* makes contact with the lever *V* at the opposite end of its travel.

The enlarged end *Y* of the slot *U* locks the traction wheels *D* when the pin on lever *P* moves into this part of the slot. Pressure applied at any point above point *Y* when the pin on lever *P* is in the enlarged end *Y* of the slot will not unlock levers *P* and *Q*. The slightest pressure below point *Y*, however, particularly when applied to lever *R*, immediately unlocks the mechanism and places the wheels *D* in the reverse position. The springs *Z* serve to eliminate vibration of levers *V* and to keep them in their proper operating positions.

Reversing Mechanism for Cable-Winding Machine.— An automatic reversing mechanism for a cable-winding machine which is designed to allow reversal of the winding guide at any desired point from zero to maximum, so that reels of various widths can be wound on the same machine, is shown in Fig. 10. The winding of various pitches is accomplished by using pick-off gears. An interesting feature of this mechanism is the use of two free-wheeling flywheels to store up energy for completing the

automatic clutch-engaging movements at each reversal of the lead-screw *P* which drives the winding guide.

The mechanism is driven by the gears *A* and *B*, which rotate in opposite directions. These gears mate with gears *C* and *D*, which are pinned to the shafts *E* and *F*, formed with clutch slots *G*. For simplicity, only one slot is shown, but the clutches are of the usual multi-tooth type.

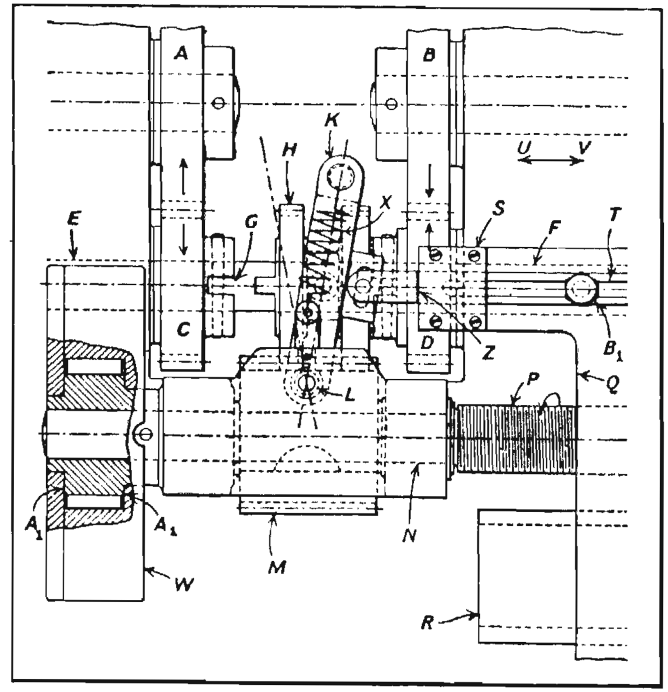


Fig. 10. Reversing Mechanism for Cable-winding Machine.

A combination gear and spool *H* runs freely on shafts *E* and *F*. The spool is shifted from side to side by the pins in the fork *K*, which is pivoted at *L*. It should be noted that the spool *H* remains constantly in mesh with the gear *M*, which is keyed to the shaft *N*. This shaft is formed integral with the lead-screw *P* which actuates the winding-guide casting *Q*. The casting *Q* is guided by the way R. The guide head carries the member S in which slides the slotted rod T. This rod is connected to the fork K as indicated. On the end of shaft N a free-wheeling flywheel W is pinned, which locks when the lead-screw rotates in the direction indicated, but unlocks in the opposite direction and runs free. On the other end of shaft N a second free-wheeling flywheel (not shown) is pinned, which operates in the opposite way to W. Spring X keeps the clutch in mesh with gear C or D until acted on by fork K.

In operation, spool H, being in mesh with gear D, causes lead-screw P to operate in the direction shown. This causes guide Q to move in the direction U until it comes in contact with the shoulder Z on rod T. The guide then pushes fork K, causing the spool to move out of the clutch slot until it is entirely disengaged. At this point lead-screw P would normally stop revolving. However, the energy stored in the free-wheeling flywheel W continues to turn the lead-screw until fork K passes through and slightly beyond the dead center position, whereupon spring X causes the spool to snap over against the face of gear C and into the slot. At this instant, the lead-screw reverses and flywheel W commences to free-wheel, while the opposite flywheel is driven to store up energy for the next reversal. It was found necessary after a trial run to make the surfaces A_1 in the free-wheeling flywheel a friction fit, so that after a short interval the energy would be dissipated, and it would then tend to drop down to a speed below that of the shaft. Thus, when again actuated, it could immediately pick up in the proper direction.

Upon the reversal of the lead-screw, guide Q travels in the direction V until member S strikes the stop B_1, when reversal again occurs.

Mechanism for Reversing Cross-Head Feed-Screw at Any Predetermined Point.—The purpose of the mechanism shown in Fig. 11 is to reverse the direction of rota-

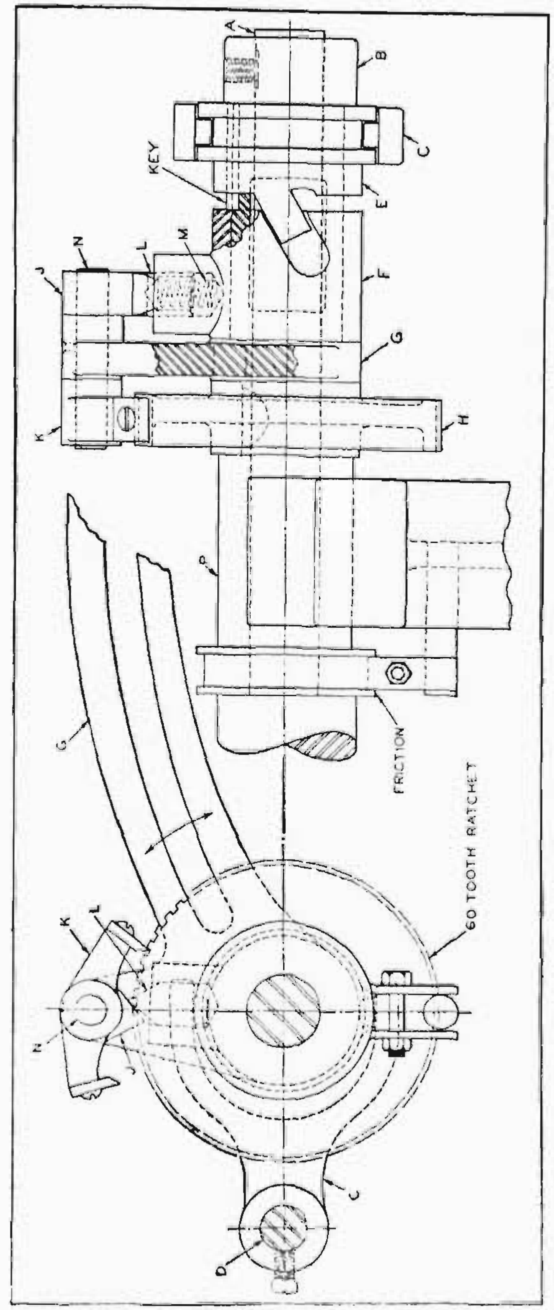


Fig. 11. Feed-screw Reversing Mechanism Using Double-pointed Pawl which can be Positioned to Revolve Ratchet Wheel Forward or Backward.

tion of a cross-head feed-screw when the cross-head has reached any predetermined point in its traverse. Adjustment of the cross-head speed is accomplished by a slotted pawl-arm. Reversal of the feed is obtained by a double-pointed pawl which can be positioned to revolve the ratchet wheel forward or backward. The mechanism has been applied to a machine for evenly distributing wire or cord on a reel or spool 50 inches in diameter, and also to a machine for a 30-inch spool.

The feeding and reversing mechanism consists of a single-threaded screw *A*, Fig. 11, supported in frame bearings *P*; pawl-arm *G*, which is free to rotate on screw *A* between ratchet *H* and collar *B*; the pawl-reversing cam *F*, placed over the hub of pawl-arm *G*; grooved sleeve *E* having a finger that engages a slot in the pawl-reversing cam *F*; fork *C*, secured to guide rod *D* for sliding sleeve *E* to the right or left when a cross-head (not shown) moves guide rod *D* through contact with a collar (not shown).

The collars are set to reverse the direction of rotation in accordance with the length of traversing movement desired. The sliding movement of sleeve *E*, Fig. 11, tips or rotates the reversing cam *F* which, in turn, depresses plunger *L* until it has passed the point of lug *J*. Spring *M* then forces plunger *L* outward against the side of lug *J* secured to shaft *N*. Shaft *N*, mounted in the short arm of pawl-arm *G*, is thus rotated, causing the pawl *K* to engage the ratchet on the other side of the arm, so that the lead-screw is revolved in the opposite direction. Pawl-arm *G* is oscillated by an eccentric on the drive-shaft. Should lead-screw *A* tend to back up, a simple friction brake can be applied.

Reciprocating Devices that Vary Their Points of Reversal.—Gradual variations of one or both reversal points of a reciprocating slide or shaft can be obtained automatically by using the star-wheel mechanism shown in Fig. 12 in combination with a proper intermediate drive,

four types of which are illustrated in Figs. 13 and 14. Eight different movements can be obtained with the various combinations; and any combination can be instantly converted to obtain a constant point of reversal by simply swinging the pawl *F* to its upper position so that it does not engage the star-wheel.

These movements are particularly adapted for mixing, valve-grinding, and textile machinery, where the stroke

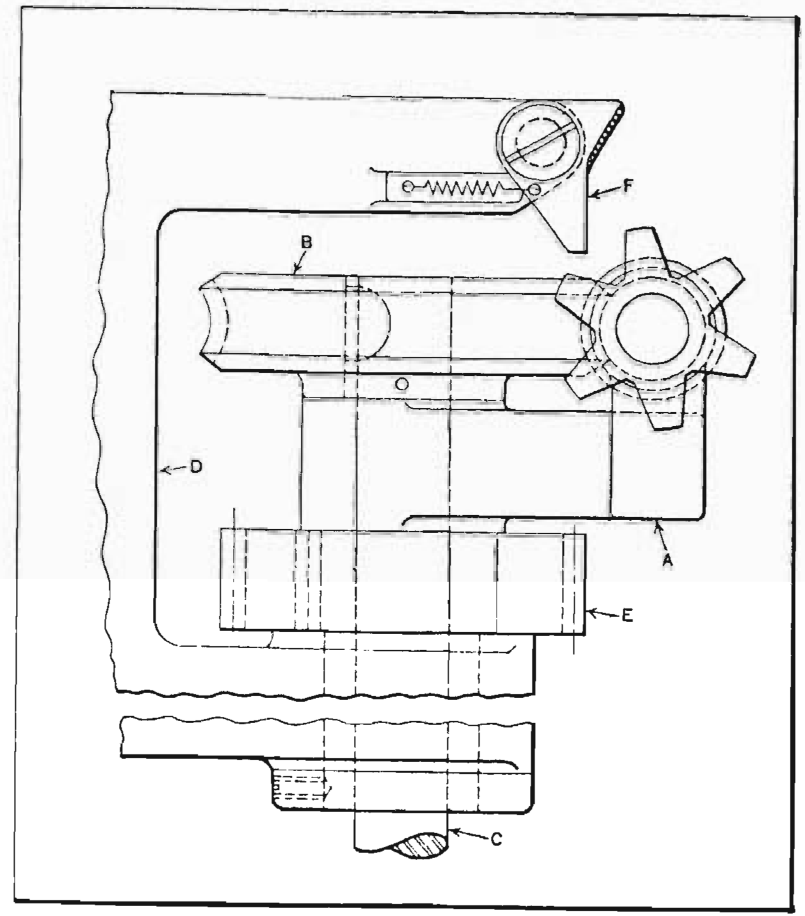


Fig. 12. Device Used with Any One of the Drives in Fig. 13 or Fig. 14 for Gradually Varying the Reversal Points of a Shaft or Slide.

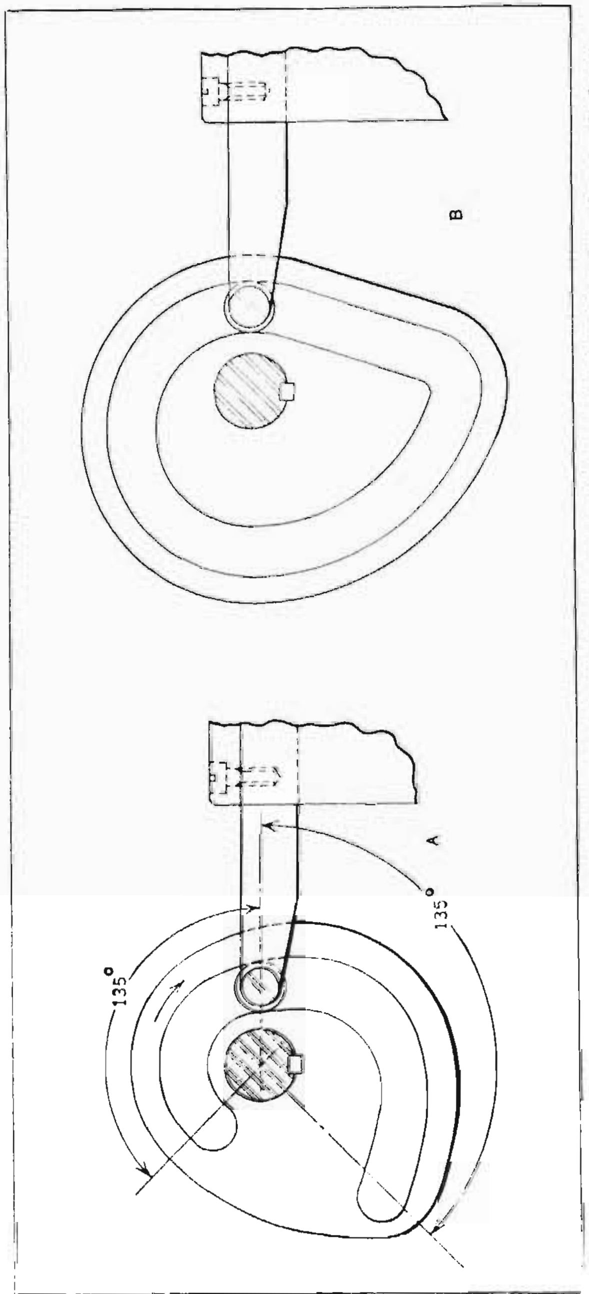


Fig. 13. When Used with Device Shown in Fig. 12, Intermediate Drive A Provides Uniform Increase of Stroke from Zero to Maximum. Intermediate Drive B Varies Time at which Reversal of Slide Occurs.

of a slide or a reciprocating shaft movement must be gradually increased or diminished, or where the positions of both points of reversal of the slide or the shaft must change gradually without changing the distance traveled between these two points.

The star-wheel mechanism consists principally of the arm *A*, Fig. 12, on which is mounted a shaft carrying the star-wheel, together with a worm meshing with the worm-wheel *B*. The worm-wheel is keyed to the shaft *C*, which has for its bearing a long sleeve, integral with the hub of arm *A*. This sleeve is free to turn in a stationary bearing on the bracket *D*, and has a spur gear *E* keyed to it. The driving member for this mechanism may be either a reciprocating or a continuously rotating gear, depending upon the movement to be transmitted through the intermediate drive; this gear (not shown) meshes with gear *E*.

Suppose that it is required to transmit movement to the slide at *A* in Fig. 13, so that the length of the stroke increases uniformly from zero to the maximum travel, the position of the point of its reversal at the left remaining constant. The cam shown here would be mounted on shaft *C* (Fig. 12), and the driving member for the star-wheel mechanism would be a reciprocating gear or rack. Now, as shaft *C* is locked to the arm by the worm and worm-gear, an oscillating movement is imparted to it by the driving member. However, during each cycle, the oscillating shaft will be advanced or retarded (according to whether the worm has a right- or a left-hand thread) by the action of pawl *F* on the star-wheel.

Assume now that the angular movement of the shaft *C* is 135 degrees, and that the cam is about to rotate in the direction of the arrow (Fig. 13). The cam-roll will follow the concentric portion of the cam groove and the slide will have no movement. During the next oscillating cycle of the cam, the position of both reversing points with respect to the cam will be slightly advanced in a

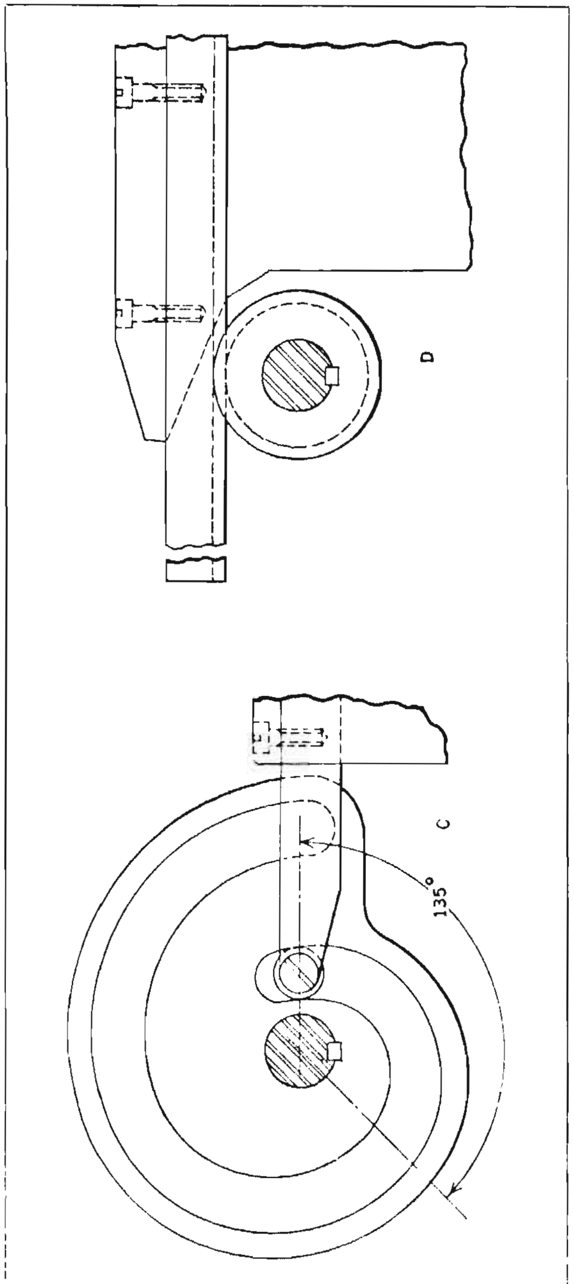


Fig. 14. When Used with Device Shown in Fig. 12 Intermediate Drive C Changes Position of Slide Gradually at Both Points of Reversal. Intermediate Slide D Produces Practically the Same Movement.

clockwise direction, due to the action of the star-wheel mechanism. This action will cause the roll to pass a short distance into the irregular portion of the cam groove, thus moving the slide a very short distance.

After each oscillation of the cam, the roll enters the irregular groove farther, uniformly increasing the stroke of the slide until it has reached its maximum travel. To prevent the roll from coming into contact with the end of the cam groove and damaging the mechanism, automatic dogs are usually provided on the machine for discontinuing the operation of the star-wheel mechanism; or in some cases, provision is made for automatically returning the cam to its starting position.

It should be noted, however, that owing to the fact that the roll passes over part of the concentric portion of the groove before the slide has reached its maximum travel, the slide will have a dwell at the beginning of each stroke. This dwell will decrease as the stroke increases, until finally, when the slide has reached its maximum travel, the dwell will equal zero.

To gradually vary the time at which the reversal of a slide occurs relative to the movements of the rest of the machine, the position of the reversal points remaining constant, the intermediate drive shown at *B* is used. In this case, the gear in the star-wheel mechanism is rotated continuously by another gear. Thus the angular movement of shaft *C* (Fig. 12) will be uniformly advanced or retarded (depending on whether the worm thread is right- or left-hand) relative to the movements of the rest of the machine members. Consequently, the time at which each reversal of the slide occurs will also be advanced or retarded relative to the movements of the other machine members.

When it is required to change the position of a slide gradually at both points of reversal, the drive shown at *C*, Fig. 14, may be employed. The cam, as before, is keyed to the shaft of the star-wheel mechanism, which, in this case, is driven by a reciprocating gear or rack meshing with gear E, Fig. 12. Assume, as in case A, that the angular movement of the shaft is 135 degrees. The rise of the cam groove is uniform. Hence, no matter what part of the groove the roll follows, the stroke of the slide will remain constant.

However, owing to the action of the star-wheel mechanism, the cam lags or advances after each cycle (depending upon whether the worm thread is left- or right-hand). Therefore, although the stroke of the slide remains constant, the positions of both reversal points of the slide change uniformly until the roll approaches the end of the cam groove. To prevent the roll from striking the end of the groove and damaging the mechanism, the same provision is usually made as mentioned for case A.

Another type of drive for producing practically the same movement as that just described is shown at D, Fig. 14. In this case, a pinion and sliding rack are used instead of the cam and roll. This is perhaps a simpler and more economical design, and it has the advantage of varying the position of the reversal points of the slide over a longer distance than would be practical with the cam type of drive.

To vary both points of reversal of a reciprocating shaft uniformly, no intermediate drive is required, because the shaft on the star-wheel mechanism—when driven by a reciprocating gear or rack—has this movement. All other reciprocating shaft movements corresponding to those of the slides shown in Figs. 13 and 14 can be obtained by combining the star-wheel mechanism with the proper intermediate drive and mounting a rack on the slide. The rack, meshing with a gear on the part to be reciprocated, will transmit the required movement.

Positive Type Reversing Clutch.—To design a positive type clutch that will release instantly under a heavy load without causing too much strain on the clutch fork or operating levers, usually necessitates considerable experimenting in order to determine the correct angle for the driving side of the teeth. When the proper angle is employed, the teeth will retain a sufficient grip to perform their work without releasing under the pressure of the load, yet the operator will be able to release the clutch with little effort.

The working conditions change, of course, with the method of operation, especially when the clutch is employed to control the movements of a slide weighing 2000 pounds. For example, the operating conditions are somewhat different when the slide is advanced by a series of short quick movements from when the slide is advanced smoothly and continuously at a uniform rate. Whether the movement is controlled by hand or by power feed, and whether or not oil or grease is employed to lubricate the ways on which the slide travels, are also factors to be considered. If the slide is fitted with a gib which must be kept fairly tight to prevent side play, as in the case of a wheel-slide on a surface grinder, this requirement must also be considered. Under these operating conditions, it is necessary that all the teeth make good contact so that the load will be distributed evenly, and thus permit the clutch to release instantly when reversing the direction of the slide movement either automatically or by hand.

The reversing clutch shown in Fig. 15 was first made with teeth cut straight without any releasing angle. With this type of teeth, it was found impossible to reverse the clutch when the slide was in motion and carrying a full load. It was, therefore, necessary to grind the cut teeth to an angle on the driving side. This was done by setting up an indexing head to hold the clutch on the table of a surface grinder and traversing the table back and forth by hand.

After several trials, an angle of 14 degrees, as indicated by view J, was decided upon as the most satisfactory for

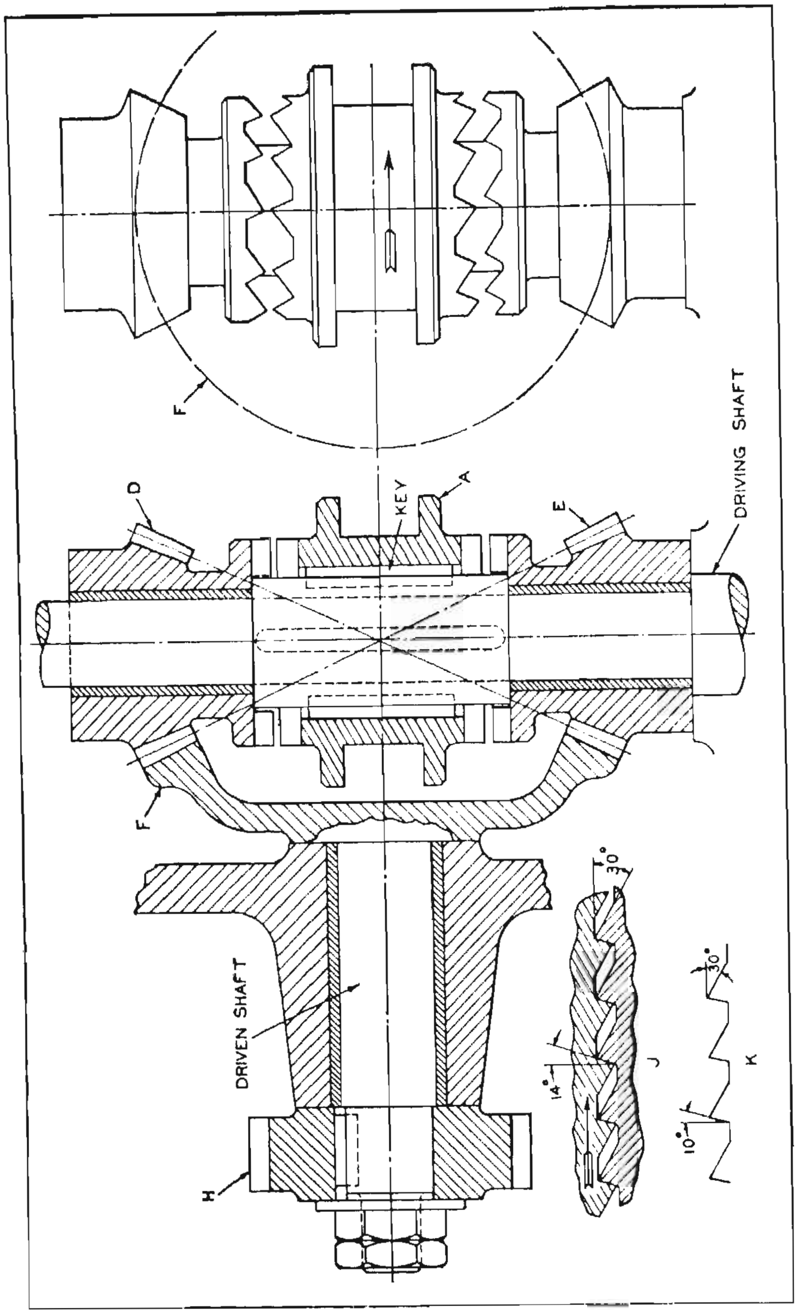


Fig. 15. Reversing Mechanism with Positive Clutch that Releases Instantly Under a Heavy Load.

the contact faces of the clutch teeth for moving a wheel-slide weighing 2000 pounds. For a slide weighing 1000 pounds with exactly the same operating conditions, an angle of 10 degrees, as at *K*, was found to be most satisfactory.

Various oils and greases were tried as lubricants for the wheel-slide, with the result that grease under pressure was found to give the slide the best working conditions. Oil, even in the heavy grades, would squeeze out on the sides of the ways due to the weight of the slide, which would then stick or freeze so badly that it was almost impossible to start the slide after it had remained idle for a short time. The grease, however, was found to stick to the ways, giving them a film that made it possible to move the slide easily.

It was found advantageous to have an odd number of teeth in the clutch face to facilitate milling the teeth. A clutch with an odd number of teeth will permit an over-running milling cutter to enter a tooth space on the opposite side of the clutch, instead of cutting into a tooth.

The clutch member *A*, which does the driving, has right-hand teeth cut in it at one end and left-hand teeth at the other end. The pinion bevel gears *E* and *D* have clutch teeth to match those cut on the clutch member *A*. The large bevel gear *F* meshes with the teeth in pinions *E* and *D*. A spur gear *H* meshes with a rack that moves the wheel-slide in or out.

CHAPTER 7

Reciprocating Motions Derived from Cams, Gears and Levers

Among the reciprocating mechanisms described in this chapter are: linkage arrangements for imparting reciprocating motion in a straight line and for operating a slide and plunger simultaneously; a chain and sprocket mechanism for providing a long stroke at uniform speed with a rest period at the end of each stroke; an ingenious arrangement for providing a reduction of speed at one end of a reciprocating movement without using a cam or changing the speed of the driving member; a simple means of providing reciprocating motion suitable for instrument use; an arrangement which causes a slide to be reciprocated from three different positions; a rack and intermittent gear mechanism which provides a uniform speed on both forward and reverse strokes; a mechanism designed to give various ratios of strokes to revolutions when converting rotary into reciprocating motion and for varying the phase of reciprocation continuously with respect to the rotary motion; a double-contact cam which provides reciprocating motion with positively locked rest periods; three types of sliding block mechanisms for converting rotary into reciprocating motion; a combination driving crank and Geneva dial mechanism for obtaining a positive, accurately timed reciprocating motion with a dwell at the end of each stroke and a means for converting reciprocating motion in one plane to similar motion in a plane at right angles to it.

Other reciprocating mechanisms based upon the action of cams, gears and levers are described in Chapter 9 of Volume I and Chapter 9 of Volume II of "Ingenious Mechanisms."

Straight-Line Reciprocating Mechanism for Disk Saw.— A mechanism consisting primarily of a link suspension frame for the rotating spindle of a saw for wood and similar materials is shown in Fig. 1. The interesting feature of this arrangement, which restricts the hand feeding movement to a horizontal motion along the straight line *X—X*, is the large amount of space it allows between the saw *S* and the supporting links *A*. There are two supporting links *A* and two links *B*. These four links are

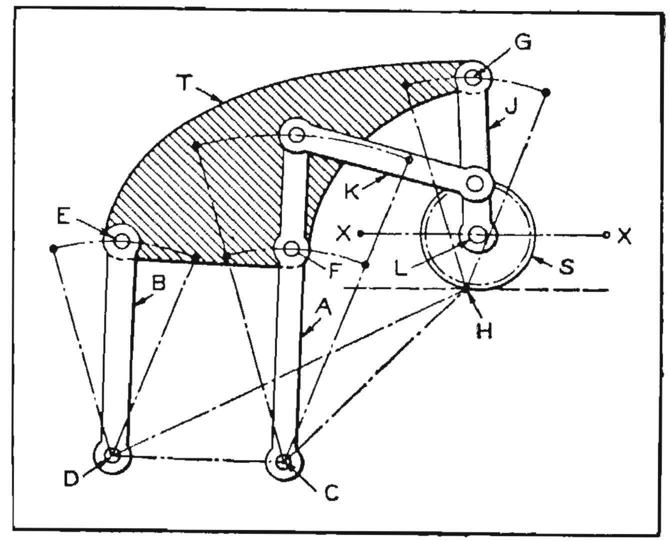


Fig. 1. Mechanism for Guiding Saw S Along Straight Line X—X.

mounted on bearing studs on the base at *C* and *D*. This arrangement permits links *A* and *B* to swing through the arcs indicated by the dot-and-dash lines.

The connecting bar *T* is triangular in shape. Triangles *E*, *F*, *G*, and *D*, *C*, *H*, are equal and are located in corresponding parallel positions. Link *J* is connected to *T* at *G* and to a rod *K* which is attached to the extension of link *A*. Thus the end *L* of lever *J* which supports the rotating saw *S* has a nearly straight-line motion. When member J is moved by hand, the saw S is given a straight-line motion, as required for cutting material. A mechanism similar to the one described is used on electric saws.

Straight-Line Reciprocating Motion.—The device shown diagrammatically in Fig. 2 was developed to obtain a movement of the point B in a straight line from D to E when point A of the fork H is moved from F to G. This, however, is only one of several possible motions that can be obtained by applying the same general principles to the construction of a motion-transmitting device of similar design.

Referring to the construction of the device, fork H is free to oscillate about pivot J on slide K. This slide is free to move along the fixed rod L. Rods M and N, fixed in fork H at angle x, are free to slide through members

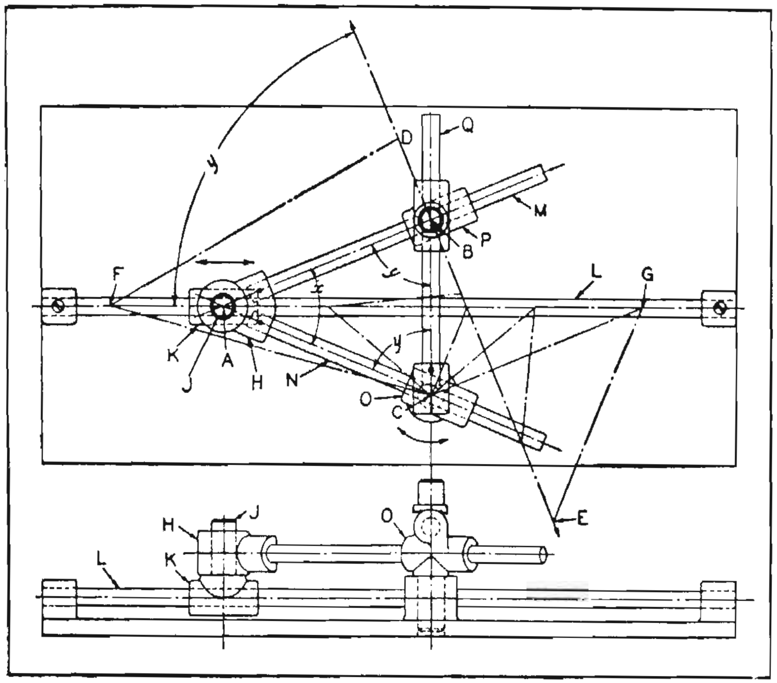


Fig. 2. Mechanism Designed to Move Point B in Straight Line from D to E when A is Moved from F to G.

P and O, respectively. Member O is pivoted at a fixed point in relation to rod L.

Rod Q is fixed in member O at the angle y of the isosceles triangle ABC. The rod Q is free to slide through member P at the fixed angle y in relation to rod M. As A is moved along rod L triangle ABC changes its altitude, but always remains isosceles. The triangle oscillates about A in relation to slide K. As it oscillates about A the point B travels in a straight line DE at the angle y in relation to rod L.

The movement of A is uniform with that of B in the proportion given in the equation:

$$\frac{\text{Movement of } A}{\text{Movement of } B} = \frac{\text{Leg } AB}{\text{Base } BC}$$

When point A of slide K has reached point G on rod L, the point B will be at point E. Various intermediate positions of rods M and N are indicated by the light dot-dash lines.

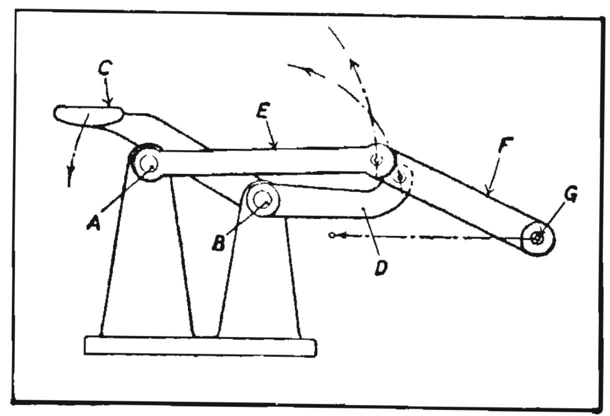


Fig. 3. Simple Foot-operated Mechanism for Obtaining Straight-line Motion.

Simple Straight-Line Reciprocating Motion Mechanism.—Fig. 3 shows a mechanism designed to withdraw a slide horizontally in a straight line to permit a part to be dropped through the opening into a chute of a conveying system

For the long stroke required, this device is quite efficient and can be adapted to other mechanisms. It is foot-operated and can be constructed at small cost. The paths through which the various members travel when the pedal is depressed are indicated in the illustration.

Pedal *C* is simply a continuation of lever *D*, which is pivoted so that it can be rotated freely about the center of stud *B*. Lever *E* pivots about the center of stud *A*. The other ends of arms *D* and *E* are connected by pivot-pins to the operating lever *F*. It will be noted that the inner of these two connecting points does not lie on the same center line as the end connections of lever *F*. At *G*, on the lower end of arm *F*, is a stud for connecting it to the slide arrangement. In use, the pedal is depressed, causing *G* to move in a relatively straight line toward the left, the motion being controlled by the two links *D* and *E*, which move in the paths indicated by the dot-and-dash lines. Two of these mechanisms are employed, one being mounted at each end of the slide.

Linkage Mechanisms for Operating Slide and Plunger Simultaneously.—The linkage arrangements shown in Figs. 4 and 5 are designed for advancing a slide and a plunger simultaneously during the first part of their movements, and then stopping the slide while the plunger continues to advance. The object of the devices illustrated is to advance the lower slide *A* for a distance *B* and then continue to advance plunger *F*. Slide *A* carries a receptacle *C* containing a sheet of paper *D* from a cut-off position to an operating position within the machine.

Resting on top of the slide is a cylindrical-shaped piece *E* which is carried along simultaneously by the plunger *F*, attached to another slide unit *G*. Plunger *F* also pushes piece *E* through the paper-holding receptacle *C*, and in doing so, causes the paper to be wrapped about the piece. The chute *H*, Fig. 4, leads from a hopper which furnishes a continuous supply of the parts to be wrapped.

A portion of the machine frame is shown at *J*. A bracket *K*, mounted on the frame, carries the dual slide mechanism. A lever *L* is employed to operate the slides. This lever is actuated by a cam *M* and a co-acting lever *N*, mounted on the rocker shaft *P*. Up to this point the mechanisms illustrated in Figs. 4 and 5 operate in a similar manner.

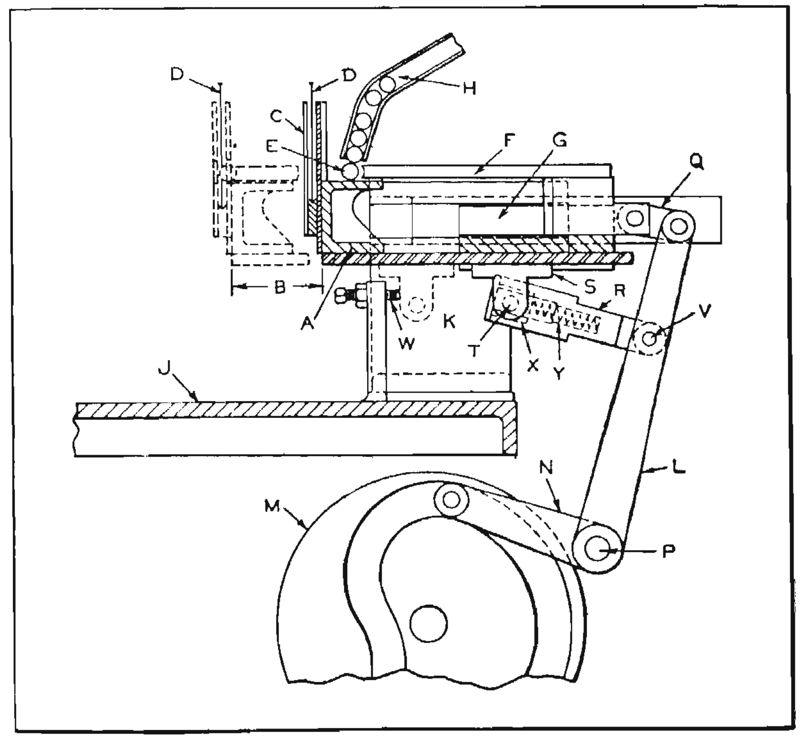


Fig. 4. Link Mechanism Designed to Operate Slide A and Plunger F Simultaneously.

Referring to Fig. 4, the upper end of lever *L* is attached to a link *Q* which moves the slide or plunger *F*. Attached to the lower sliding unit of which *A* forms a part, is a spring cushion link *R*. This link is connected to a block *S*, the pivot points being at *T* and *V*. When slide *A* is advanced a distance B, block S comes in contact with the stop-screw W, which prevents the lower slide from advancing beyond this position. At this point the spring cushioning mechanism comes into effect and permits the lever L to continue advancing. The continued movement of the upper slide and plunger F carries the cylindrical piece into the machine.

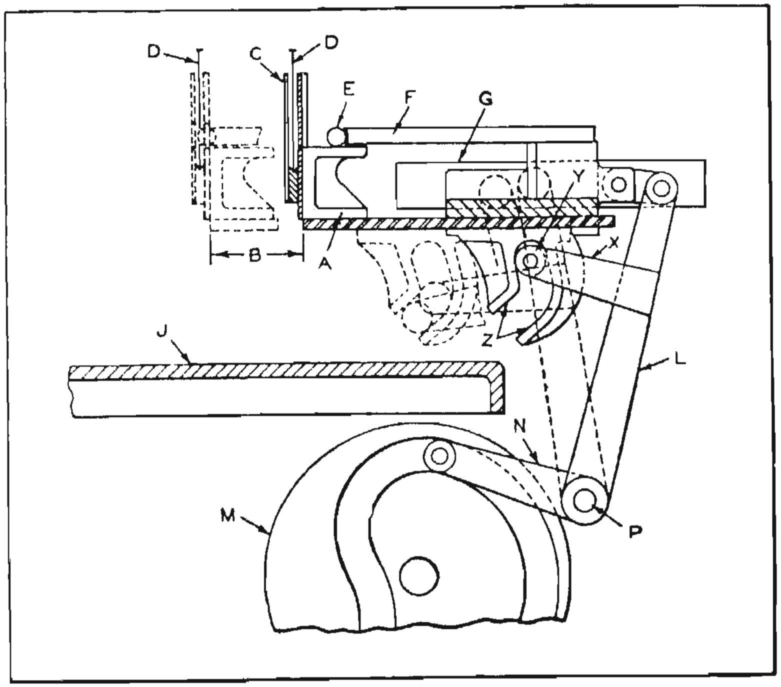


Fig. 5. Mechanism Shown in Fig. 4 Redesigned to Give Smoother Action.

The spring cushion consists of a movable block X in which the stud T pivots. This block slides in the connecting link R, while the spring at Y holds the sliding block in the position shown. As lever L advances, block X slides along the connecting link R, thereby compressing spring Y. Interrupted advance linkages requiring the compressing of a spring of sufficient stiffness to advance a slide

mechanism introduce into the machine what may be termed a harsh movement and one that may pound and wear out the mechanism quickly, with an attendant loss of power. To eliminate this action, the lower slide connection was redesigned as shown in Fig. 5.

The lever L, Fig. 5, is fitted with an extension X which supports a roll Y. Fastened to the under side of the slide bracket is an open cam Z. The contour of this cam is so designed that the roll will follow the angular portion of the cam indicated by the arrows at Z without advancing the lower slide after it has been advanced the required amount, although the upper slide and work pusher or plunger F continue to advance. It is obvious that the smoother action of the open cam Z will permit the moving parts to travel at a faster rate than is the case with the construction shown in Fig. 4.

Long-Stroke Reciprocating Motion for Guiding Wire on Reel.—Fig. 6 shows diagrammatically a reciprocating motion mechanism used for guiding wire on a broad reel. The mechanism is designed to give a long stroke, with a uniform rate of traverse speed and a period of rest at each end of the stroke.

Rod E, which is supported in bearing F, carries two pins P at its outer end for guiding the wire. Sprocket A drives sprocket B through the block chain C. The special link D, inserted in the chain in place of one of the standard links, carries the rod E on the pin G. The length of link D is such that pin G travels along the center line between the two sprockets. Link D, in traveling from sprocket to sprocket, transmits a uniform reciprocating motion to rod E.

At each end of the stroke, pin G remains stationary while the link D rotates half way around the sprocket, providing a period of rest which is determined by the diameter of the sprocket. On extremely long strokes, it is necessary to support the chain to avoid excessive sag.

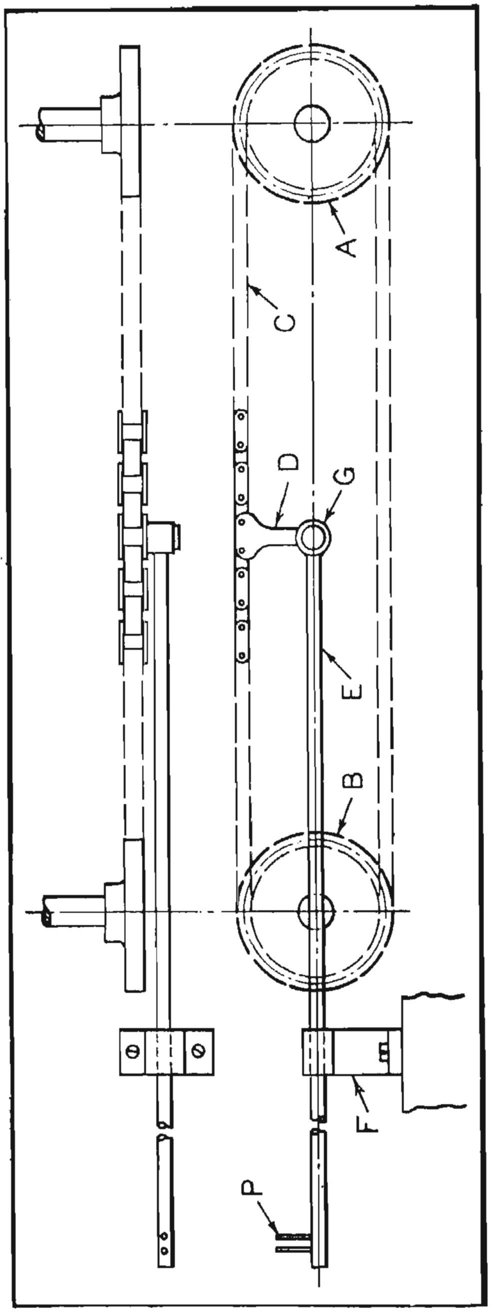


Fig. 6. Diagram Showing Arrangement of Chain and Sprockets for Obtaining Reciprocating Motion for Guiding Wire on a Long Reel.

Reciprocating Motion Used on a Wire-Forming Machine.—Fig. 7 shows the design of a reciprocating motion used on a wire-forming machine, which employs a unique method of providing for a reduction of speed at one end of the movement. This is accomplished without the use of a cam and without changing the speed of the driving member. The lower view shows a plan of the arrangement, and the two upper views show side elevations. The part *A* is a sliding fit in a dovetail slot in the stationary part *B*, and carries the lever *C*, which is connected at its upper end to the rod *F*, from which it receives its motion. The lever *D* is keyed to the same shaft as lever *C*, and carries at its lower end the roller *E*, which rides on the plate *G*.

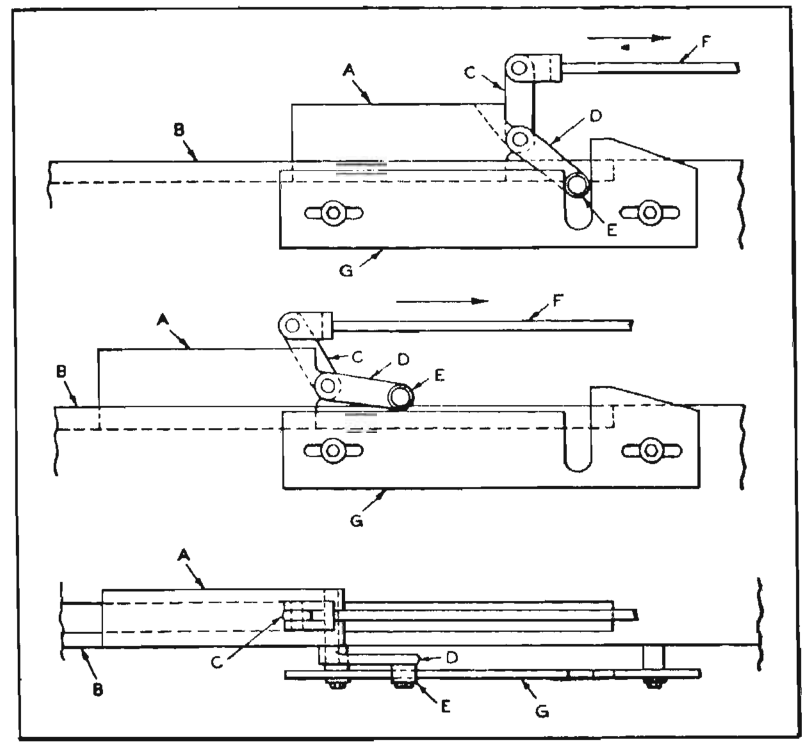


Fig. 7. Reciprocating Mechanism which Provides for Reduction of Speed at One End of Movement.

The part *A* is recessed, as indicated, to receive lever *C*, the recess being of a depth that will permit the roller *E* to ride on the upper edge of the plate *G* without any lost motion.

In the central view, rod *F* is shown as having completed its movement to the left and as moving to the right. As levers *C* and *D* are keyed to the same shaft and lever *D* is prevented from rotating downward by the roller *E* resting on plate *G*, there is no movement of either *C* or *D* relative to part *A* until roller *E* reaches the slot in plate *G*. The motion of *A* up to this point is directly transmitted to it by rod *F*. As roller *E* reaches the slot in plate *G*, it rides downward, as shown in the upper view. At this point, levers *C* and *D*, being fastened to the same shaft, operate as one lever, with roller *E* acting as the fulcrum. As levers *C* and *D* are of the same length, the speed of movement of part *A* from this point is reduced to about 50 per cent that of rod *F*. The length of travel at this point is likewise reduced. Plate *G* is slotted to provide adjustment.

Simple Reciprocating Mechanism for Chart Recording Pen.—A very simple means for producing reciprocating motion was discovered while developing a special instrument. Aside from the simplicity of the device, which is shown in Fig. 8, there is the desirable feature of lightness, with a minimum amount of power loss due to friction.

The mechanism consists of a cam *A*, fabricated from spring wire. The two legs *B* of the spring cam cross and pass through a hole in bar *C*, while screw *D* serves to hold the spring cam in place in the instrument case.

Cams *E*, acting on spring cam *A*, cause the position of the point where legs *B* cross to change as cam *A* is depressed and released. In changing the position of the crossing point, bar *C* is made to move back and forth. The upper view of the illustration shows the device at the starting position, with spring cam *A* completely relaxed,

while the lower view shows the relative position of the members of the mechanism when the maximum depression of spring cam *A* has been accomplished to displace bar *C* a distance equal to *F*.

Bar *C* is intended to support a pen for marking a continuous chart, and it is for this reason that lightness and easy operation are desired. The two cams *E* are not of

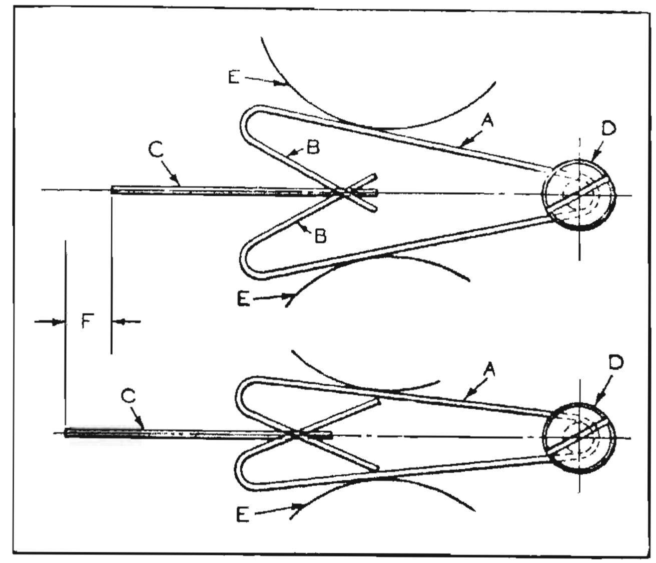


Fig. 8. Simple Method of Producing a Reciprocating Motion for a Chart Recording Pen.

the full rotating type. The motion of the cams about their pivots or centers is dependent upon the action of a bellows that is subject to pressure pulsations.

Pivoted Holder for Cam Follower Rolls Provides Unusual Reciprocating Motion.—The design of an interesting mechanism used on a wire-forming machine is shown in Fig. 9. This mechanism is used to operate a slide *G* which carries a forming tool, the slide being caused to reciprocate from three different positions. The lower cam *F* imparts the reciprocating movements to the slide, while the upper cam E intermittently locates the slide G in three different positions.

Referring to the illustration, the shafts A and B carry the meshing gears C and D which rotate in the directions indicated by the arrows. Shaft A is the driving member,

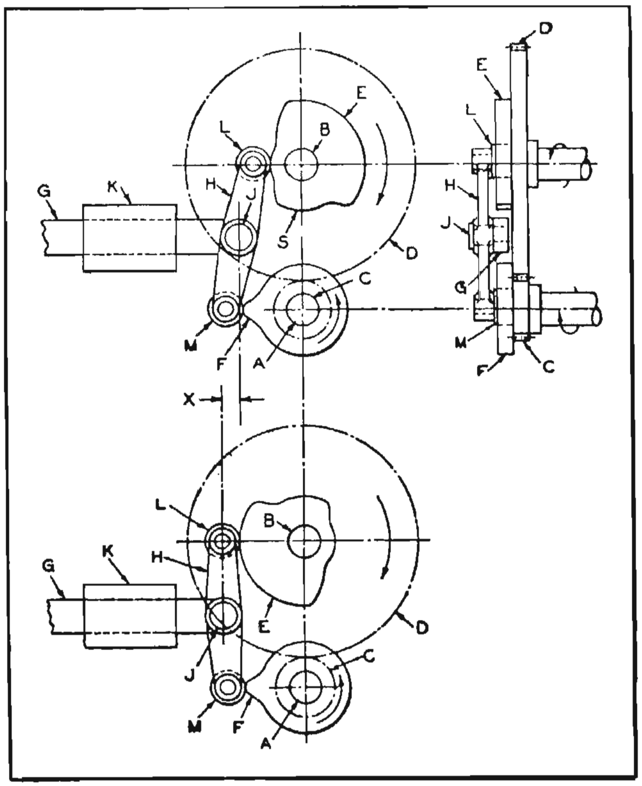


Fig. 9. Mechanism for Automatically Reciprocating a Slide and Locking It in Three Different Positions.

shaft B running idle in its bearings. Cams E and F are attached to gears D and C and operate with them. The speed ratio between gears C and D is 4 to 1. The slide G is supported in bearing K. Slide G carries the stud J, upon which lever H fulcrums at its center. Lever H carries the rollers L and M, which engage the cams E and F,

respectively, being held in contact by a spring (not shown) which draws slide G to the right.

Cam E is designed with three points of rest, one of which is for a period of 180 degrees, while the other two are for periods of 90 degrees each. The single-lobed cam F is designed to give a quick rise and fall. In the upper view to the left, the roller L is shown on the low point of cam E, while the roller M is positioned on the high point of cam F. As the lobe of cam F passes under the roller M, the lever H fulcrums on the center of the roller L, transmitting to the slide G a movement equal to one-half the rise of cam F.

As gear D rotates in the direction indicated by the arrow, the intermittent step S of cam E is brought under roller L, causing lever H to fulcrum on the center of roller M, thus moving the slide G to the left an amount equal to one-half the rise of step S on cam E, slide G now being in its second or intermediate position. The lobe of cam F again actuates the lower end of lever H, giving the slide G its second reciprocating movement from its new position.

Continued rotation of gears C and D brings the high point of cam E under roller L, as shown in the lower view. As compared with the upper view, the position of the stud J has moved to the left a distance equal to X. As the high point of cam E includes 180 degrees, the lobe of cam F acts twice on the roller M during this period. Thus the slide G is given one oscillating motion at each of two positions and two motions at the third position. The roller L then returns to the low point on cam E, completing the cycle. The slide bearing K, it will be noted, is not shown in the side view of the assembly.

Rack and Intermittent Gear Mechanism for Obtaining Uniform Reciprocating Motion.—The reciprocating motion produced by the experimental mechanism shown in Fig. 10 differs from that of the crank-driven movement so commonly used, in that member C moves at a uniform speed on both the forward and reverse strokes. No sliding gears or clutches, such as are usually employed in gear-driven reciprocating motions, are used in this mechanism, the motion being obtained by two modified gears and a rack.

Referring to the illustration, the intermittent driving gear *A* rotates constantly in the direction indicated by the arrow. The teeth of gear *A* mesh alternately with the teeth of rack *C* and gear *B*. In the position shown in the illustration, the leading tooth on gear *A* is about to make contact with the first tooth of gear *B*, causing the latter to rotate in the direction indicated. The motion of gear *B* is transmitted to the rack *C*, causing it to move to the right.

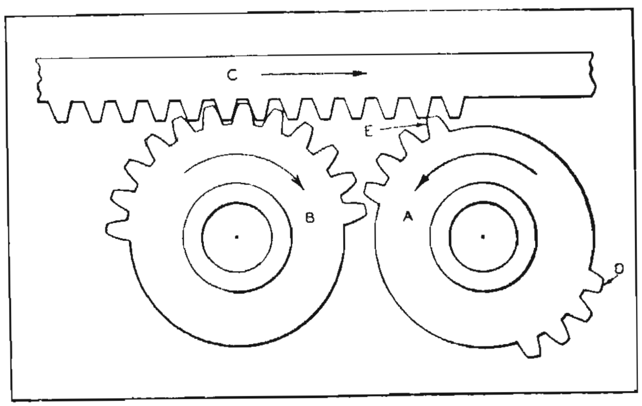


Fig. 10. Mechanism for Transmitting Uniform Reciprocating Motion to Rack C from Rotating Intermittent Gear A.

As the last tooth *E* in gear *A* completes its contact with the mating tooth in gear *B*, the tooth *D* in gear *A* is ready to engage the mating tooth in rack *C*. When this occurs, the direction of movement of rack *C* is reversed. At this point, the teeth on gear *B* are in the clearance space of gear *A*. Gear *B* is then driven by rack *C* until tooth *D* of gear *A* meshes with gear *B*. The alternate driving of rack *C* by gear *A* and gear *B* produces a uniform reciprocating motion of the rack.

It will be noted that the number of teeth in each tooth section of gear *A* is two less than one-quarter of the number of teeth that would be carried by a full gear of the same size and pitch. Although this is necessary to prevent clashing, the effect is the same as that produced by a quarter turn of a gear having the full number of teeth. It will also be noted that the two end teeth in each toothed section of gear *A* are modified to provide sufficient clearance to permit the reversal of rack *C*. On large gears, it will be necessary to modify the succeeding tooth as well. Also, it may be necessary to remove a small amount of material from the leading face of the leading tooth in each group of teeth in gear *A* to permit a slight lag in the reversal of rack *C*. The degree of modification of the teeth necessary for the proper functioning of the mechanism will vary with the size of the gears and their pitch.

The length of travel of rack *C* is determined by the diameters of gears *A* and *B*. The number of teeth in these gears must be divisible by 4. Although this mechanism can be used only where moderate speeds are employed, due to the necessity for suddenly arresting the momentum of rack *C* and gear *B*, its simple construction recommends its application wherever the speeds are within the permissible range.

Mechanism for Converting Rotary into Reciprocating Motion and a Single-Chain Differential Drive.—The mechanism shown in the upper diagram of Fig. 11 is used for converting rotary into reciprocating motion. It can be designed to give various ratios of strokes to revolutions and to cause the phase of the reciprocating motion to vary continuously with respect to the rotary motion. Such variation is desirable for lapping and polishing. For example, it can be built to have sleeve *T* make 0.565 reciprocation per revolution of sprocket *C*. The reciprocating motion is continuous and positive and takes place without shock.

The stationary housing K supports a bushing D, held in place by a screw E. The shaft A rotates in bushing D and is restrained from endwise motion by the hub of sprocket C and the base of the truncated cylinder F. At L is another truncated cylinder similar to the one at F.

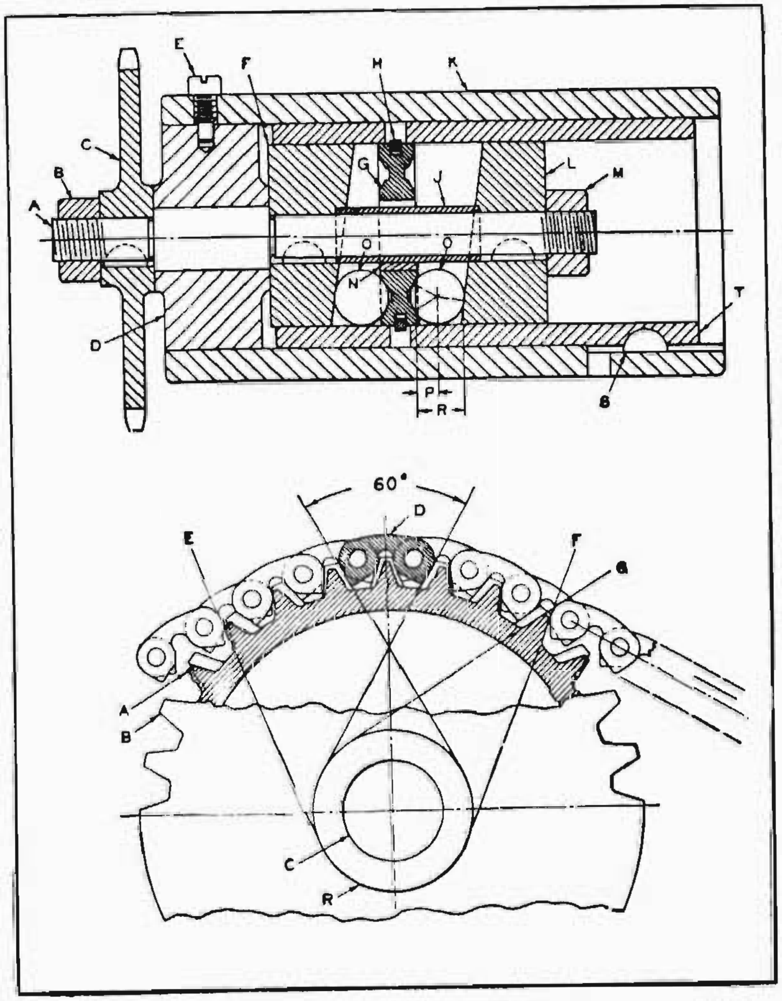


Fig. 11. (Top) Mechanism for Converting Rotary into Reciprocating Motion. (Bottom) Single-chain Differential Drive with Standard Sprocket A and Special Sprocket B.

These cylinders are keyed to shaft A with their truncated faces parallel and spaced by the sleeve J. The nut M serves to clamp the members F, J, and L together.

Sleeve T is a sliding fit in housing K, and is restrained from rotating by the key S. The internal collar or race G is held in sleeve T by the snap-ring H. In dismantling, the snap-ring may be reached through the holes in sleeve T. The two balls O are positioned by the ball cage N which straddles the internal collar G, so that the line joining the ball centers is parallel to the axis of shaft A. The cage N bears on the faces of the collar or race G and thus holds the balls in line.

Rotation of driving sprocket C rotates the truncated cylinders or face-cams F and L, thus causing the balls O to roll on the cam surfaces and on the internal collar G which they cause to reciprocate or shift from side to side. The reciprocating motion is transmitted from collar G to sleeve T through ring H.

If the faces of collar G were not grooved but constructed with two plain surfaces, the balls and their cage would make one revolution about the axis of shaft A to every two revolutions of shaft A. This gives two revolutions of A for each complete reciprocation of sleeve T. With the faces of the collar G grooved as illustrated, however, the ratio is somewhat different, as shown by the equation:

$$\frac{\text{Number of reciprocations of } T}{\text{Number of revolutions of } A} = 1 - \frac{P}{R}$$

where P and R are the horizontal distances shown in the upper diagram of Fig. 11.

It is not necessary to groove the faces of collar G in order to get a varying-phase relation between A and T. Any creep of the balls or the collar itself, however little, will cause a continuous though slight change in the phase relations. The ratio of revolutions of the driving sprocket C to the number of strokes of sleeve T can be further modified by mounting housing K so that it can be rotated about the axis of shaft A.

The chain mechanism shown in the lower diagram of Fig. 11 provides a convenient means for driving shaft A and housing K at slightly different speeds. Ordinarily, two concentric sprockets would be driven by two separate chains, each with its own take-up mechanism. The mechanism shown, however, drives the two concentric sprockets at slightly different speeds by means of a single chain which is wide enough to engage both sprockets. Referring to the lower diagram of Fig. 11, B is a standard sprocket, the teeth of which are shown in unshaded outline, and D a silent chain. The special sprocket A, the teeth of which are shown in shaded outline, has one tooth less than sprocket B, but has substantially the same outside diameter and root diameter as sprocket B. Its teeth are cut with the same cutter as is used for sprocket B, but the sprocket is indexed and recut to make the tooth thickness less than normal, so that the teeth A do not interfere with chain D and lift it off sprocket B.

The approximate tooth outline can be determined as follows: Draw the circle R tangent to the 60-degree faces of the standard gear tooth, and also draw a standard chain in contact with sprocket B, making the angle of wrap as small as possible without sacrificing proper functioning of the driving chain. Next, draw the line E tangent to the circle R and to the chain tooth which is about to engage the standard sprocket B. This establishes the driven face of the sprocket tooth on sprocket A. Next draw line F tangent to circle R and to the chain tooth that has started to leave the standard sprocket B to establish the opposite face of a tooth on sprocket A. Draw the line G tangent to the circle R and a whole number of tooth spaces from the line E, as measured on sprocket A. The space included between F and G will be the approximate tooth on sprocket A. It may be easier to place a chain on the sprockets and

cut the teeth of the second sprocket back by trial cuts until there is no interference than it is to lay out the tooth as described.

The operation of the mechanism is as follows: The chain, coming in wedge-like contact with a tooth of sprocket A along the line E, moves sprocket A a little faster than sprocket B. The narrow teeth on sprocket A permit it to move faster than sprocket B without interfering with the chain. One tooth at a time on sprocket A takes all the drive. If necessary, these teeth can be made shorter than standard teeth, as shown in the illustration.

The sprocket A shown in the lower diagram of Fig. 11 has 27 teeth, while the sprocket B has 28 teeth, so that the relative speeds of the two sprockets are inversely proportional to their number of teeth.

In using the single-chain differential drive on the mechanism shown in the upper diagram of Fig. 11, the standard tooth sprocket B was used to drive the housing K, which should be mounted so that it can rotate but is restrained from endwise movement, while the special 27-tooth sprocket A took the place of sprocket C. Thus when housing K and collar G have made 27 revolutions, the cams F and L will have made 28 revolutions. The cams, therefore, gain one revolution on collar G for each 27 revolutions of the housing. Since it requires approximately two revolutions of the cams with respect to collar G to produce one reciprocation, it will require 2×27 or 54 revolutions of housing K to produce one reciprocation when the collar G has plain faces. The sleeve J may be omitted and the nut M pulled up until the balls just grip collar G. The nut can then be locked with a cotter-pin or check-nut. A light spring could be used under the nut M, but this was not found necessary for the particular mechanism shown.

Care should be taken not to make the angle of the truncated face of the cylinders F and L too large. The limiting angle may be found by experimenting with a ball placed between two lubricated surfaces. The angle used for the mechanism shown was 8 1/2 degrees. This angle was found satisfactory for balls 3/8 inch in diameter.

Mechanism for Converting Rotary into Reciprocating Motion.—In Fig. 12 is shown a mechanism for converting the rotary motion of shaft *A* into a reciprocating motion of the sleeve which is composed of the two members *B* and *C*. This mechanism is similar to the one just described, the principal difference being in the construction of the reciprocating sleeve.

In the mechanism shown in the upper diagram of Fig. 11, the snap-ring *H* for transmitting the drive from collar *G* to the sleeve member was assembled by springing it into the groove in member *G*. Member *G* was then pushed into the sleeve until the ring reached the groove on the inside of the sleeve. The ring then expanded into the groove in the sleeve. In order to permit ring *H* to be compressed so that collar *G* could be removed, it was necessary to drill two holes through the reciprocating sleeve, as shown in the cross-section view in the upper diagram of Fig. 11. The diameter of these holes was somewhat larger than the width of the snap-ring and the holes were drilled diametrically opposite each other.

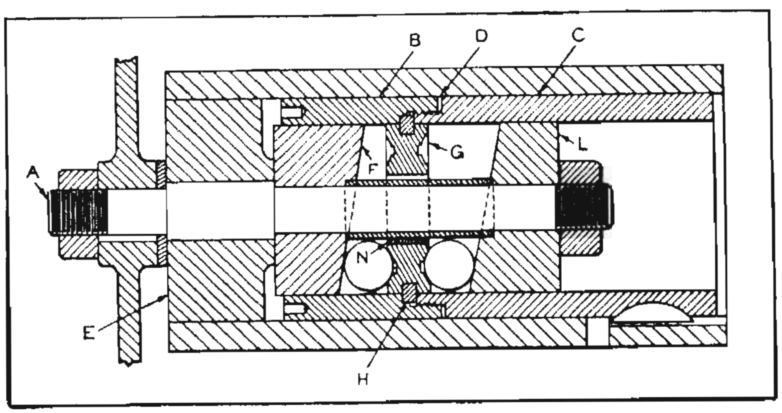


Fig. 12. Mechanism for Imparting Reciprocating Motion to Sleeve Members B and C.

With the design shown in the accompanying illustration, a heavier ring *H* is used and the necessity for drilling holes through the sleeve is eliminated. The ring *H* is fitted into the groove in collar *G*, which is then inserted in member *B* of the reciprocating sleeve so that ring *H* fits snugly into the counterbore. Threaded sleeve *C* is then screwed into sleeve *B*, a clearance of about 0.025 inch being left at *D* so that ring *H* can be clamped securely in place.

The operation of the mechanism is the same as the one just described and shown in Fig. 11. Shaft *A*, with the two truncated cylinders *F* and *L* keyed to it, revolves, causing the balls in contact with the truncated faces and the race of collar *G* to transmit a reciprocating movement to the sleeve members *B* and *C*. The balls are kept in alignment by a cage *N* which straddles the collar *G*.

Ball thrust bearings placed on both sides of the stationary bearing *E* might be a worth-while improvement.

Mechanism for Converting Rotary to Constant-Velocity Reciprocating Motion.—The mechanism shown diagrammatically in Fig. 13 was devised to impart a reciprocating motion to the sliding member *A* from the rotating shaft *B*. This reciprocating motion was required to operate the bellows of an artificial respiration machine in such a manner as to simulate the inspiration and expiration of natural breathing.

In natural breathing, the expanding movement of the ribs, accompanied by a downward pull of the diaphragm, produces a suction power for inhaling air which operates immediately at its full velocity and continues to do so to the end of the inhalation. It is apparent, therefore, that the accelerating and decelerating motion produced by a simple crank mechanism such as is usually employed to convert rotary to reciprocating motion cannot be used to drive the reciprocating bellows-operating member *A*.

The special mechanism devised for this purpose, like the natural mechanism of the chest, immediately starts the flow of air into the lungs at full velocity and continues the flow at a uniform rate to the end of the air injecting motion, after which it causes the air to be withdrawn from the lungs in a similar manner.

To duplicate the rate of natural breathing, the driving motor *C* was equipped with reduction gears to give shaft *B* a uniform speed of 16 R.P.M. The reciprocating motion is imparted to member *A* in the direction indicated by arrow *D* from the rotating shaft *B* through arm *E*, flat-faced wheel *F*, and a flexible steel band *G* in the manner to be described.

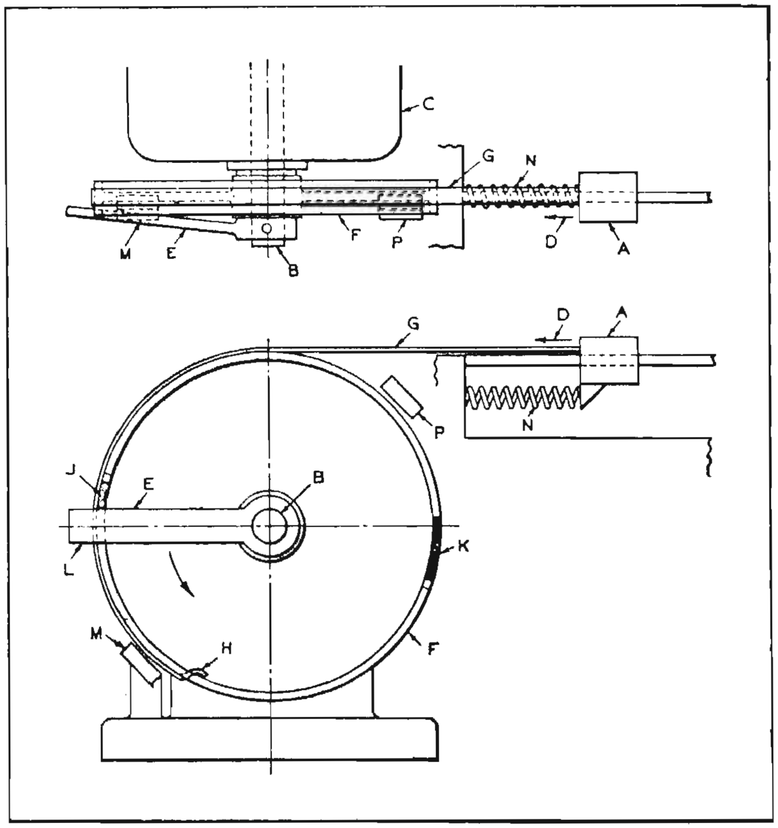


Fig. 13. Constant-velocity Reciprocating Motion Mechanism Designed to Operate the Bellows of an Artificial Respiration Machine.

Shaft *B*, with the arm *E* secured to it by a pin, rotates in a counter-clockwise direction, causing arm *E* to ride on the rim of wheel *F*, which is a free running fit on shaft *B*. One end of the flexible steel band *G* is fastened to wheel *F* at *H*, and the other end is secured to the bellows-actuating member *A*.

The arm *E* is made of spring material, and is so designed that it will automatically drop into slots or notches *J* and *K* in the rim of wheel *F* as it is rotated in a counter-clockwise direction. Fig. 13 shows arm *E* in slot *J*, in which position it acts as a driver for wheel *F*, causing the wheel to rotate counter-clockwise.

This rotation of wheel *F* serves to wind steel band *G* over the face of the wheel and thus pull the bellows operating member *A* to the left, in the direction indicated by arrow *D*. The movement of member *A* continues at a uniform speed until the outer projecting end *L* of arm *E* comes in contact with the cam-faced dog *M*, which lifts the arm out of the driving notch *J* in wheel *F*.

As soon as arm *E* is lifted from the driving slot, the spring *N*, which has been compressed by the movement of *A* to the left, instantly expands and forces member *A* to the right, thus completing the return stroke. This return stroke of member *A* causes wheel *F* to revolve clockwise back to its original starting position, where it is stopped by a device not shown.

The arm *E* then continues to ride or slide on the rim of the wheel until it drops into the slot *K*. The movement of member *A* to the left is then repeated in the manner described until another cam-faced dog *P* disengages arm *E* and allows the spring *N* to return the member *A* again to the starting point. Member *A* travels at a constant velocity during its stroke to the left because the rotating speed of wheel *F* is constant and therefore provides a constant speed for winding the flexible band *G* around the wheel face.

If a power return motion is required for member A in place of the spring-actuated return motion, gears can be employed to operate the mechanism in the reverse direction. Since the bellows are only a secondary part of the mechanism, they are not shown.

The notches J and K in wheel F are deep enough to seat arm E. One end of each notch is beveled or sloped to allow arm E to drop into the slot until it strikes the driving end of the notch which is at right angles to the rim. The dogs M and P are fixed, and are so positioned that they lift arm E out of the notches in wheel F at the proper time as it traverses around with the wheel rim, and then allow it to drop back into contact with the rim. It then slides along the edge and in contact with the rim in a counter-clockwise direction while the wheel is being reversed and being moved in a clockwise direction by spring N, until it drops into the next notch in the rim edge, which again immediately reverses the motion of wheel F.

Reciprocating Motion with Positively Locked Rest Periods.—A double-contact cam, such as is shown in Fig. 14, can be used to secure a reciprocating motion with a rest period for the sliding member. This mechanism is positive, self-locking, and has no backlash. The cam of this mechanism is keyed to the driving shaft at A. The contour of the cam is composed entirely of arcs of circles, and is constructed as described in the following: Let e equal the angular magnitude of the two rest periods of the cycle. Then 180 degrees minus angle e equals the angle of the two action periods of the operating cycle.

Next, triangle ABC is constructed with the angle at A equal to angle e as shown in Fig. 14. The lengths b and c are then selected for two sides of the triangle, so that $b + c - a$ equals the length of the reciprocating stroke S of the yoke. Then, with the vertex of the smallest angle of the triangle as a center, say at B, and with an arbitrary radius r, construct the arc DE; next, with C as a center,

construct the arc EF, and with A as a center construct the arc FG. With B as a center, construct arc GH and with C as a center, construct the arc HJ; finally, with A as a center, construct the closing arc JD.

The distance between any two parallel tangents to the cam is equal to the constant distance $2r + a + b - c$. Thus, the yoke follower maintains two contact points with the cam surface throughout the cycle, and operates without any backlash, except the small amount necessary for working clearances.

It will be noted that angles B and C are unequal, a fact that results in a different rate of acceleration or timing of motion on the forward and return strokes. Motion distribution or timing can be varied still further by changing radius r. Identical forward and return motions can be secured, if desired, by making angle B equal to angle C.

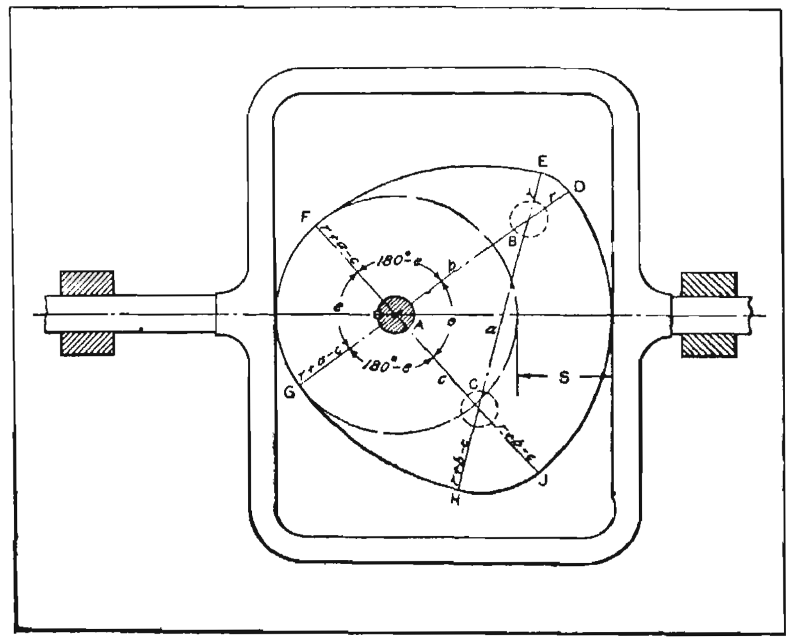


Fig. 14. Cam Mechanism Designed to Produce Reciprocating Motion with Positively Locked Rest Periods.

The shaft could have been keyed to the cam at points *B* or *C* as well as at *A*. In that case, however, a different rest period and a different travel of the follower would have been obtained with the same cam and follower, the rest periods corresponding in magnitude to the three angles *A*, *B*, and *C* of the triangle. The cam will operate with a rocker-arm follower as well as with a sliding type follower.

Sliding-Block Mechanisms for Converting Rotary into Reciprocating Motion.—Rotary motions are frequently converted into reciprocating movements by means of sliding-block mechanisms. Such mechanisms generally include a

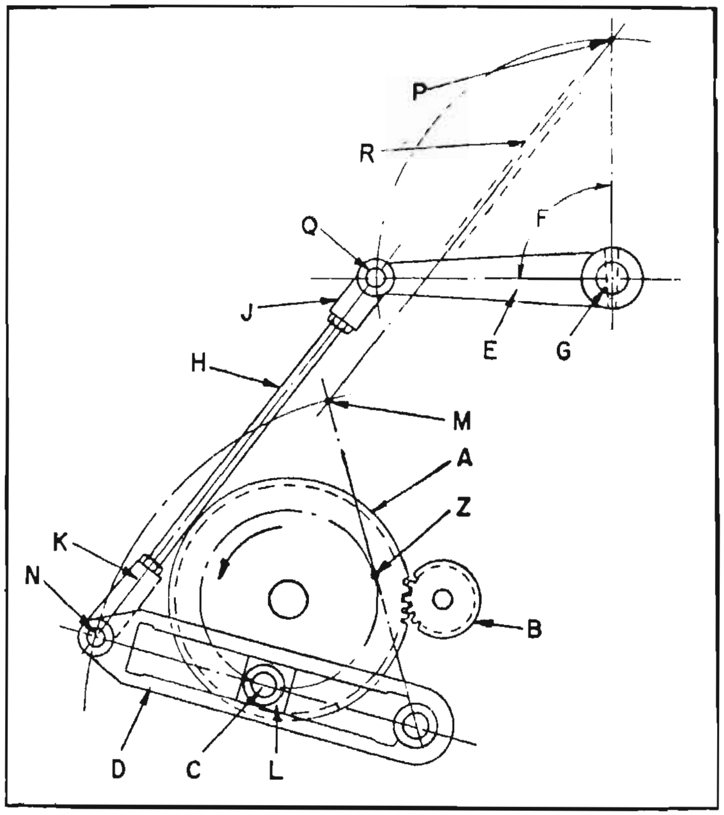


Fig. 15. A Long Eccentric Block Movement in which Lever E is Quickly Raised to a Vertical Position, then Slowly Lowered to Position Shown.

box-shaped cast-iron lever arm in which a bronze block slides. Three different applications of this type of mechanism are shown in Figs. 15, 16 and 17.

A long eccentric block movement is shown in Fig. 15. In this construction, the large gear *A* is driven by pinion *B*. Sliding block *L* is free to pivot about stud *C*, which projects from the face of the gear. When the gear is rotated in the direction indicated by the arrow, the sliding block will move lever *D* from the lower position shown to that indicated by the center line connecting points *M*

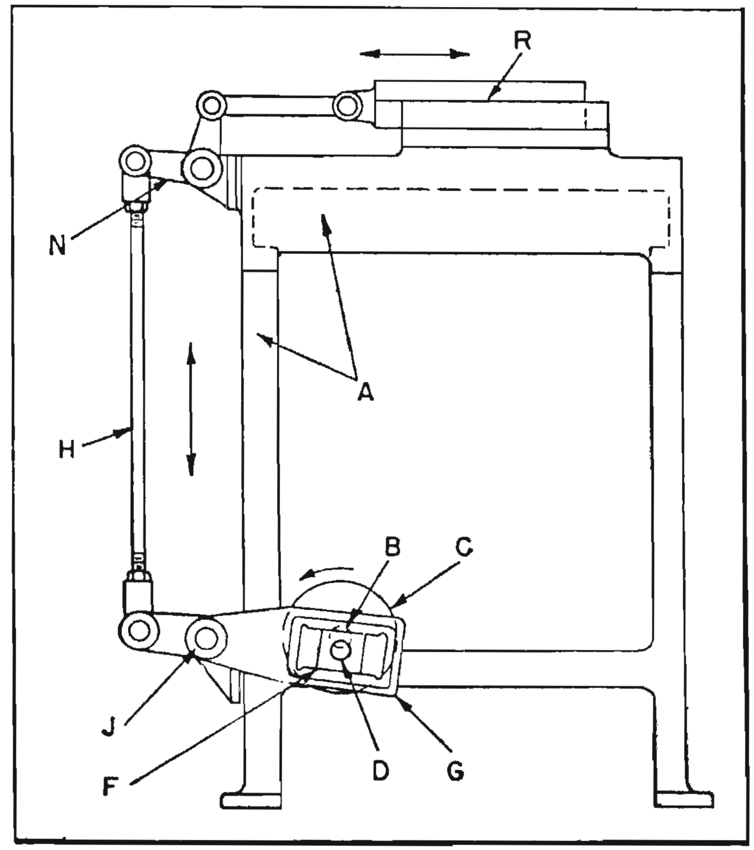


Fig. 16. Sliding Block F Mounted on Eccentric Stud D is Employed with Connecting Linkage to Reciprocate Member R on Table of Machine.

and *Z*. *M* represents the upper position of connecting pin *N*, and *Z* indicates the common center of the stud and block when the lever is in its uppermost position.

As block *L* slides back and forth in member *D*, the lever *E* and shaft *G* to which it is pinned are reciprocated through arc *F* to operate a mechanism not shown. Connecting-rod *H* and adjustable links *J* and *K* join levers *E* and *D*. Point *P* represents the upper position of pin *Q*, while the center line joining points *P* and *M*, and broken lines *R*, indicate the uppermost position of the connecting-rod.

One feature of this design is that only a little more than one-quarter of a revolution of gear *A* is required to lift lever *E* to its vertical position. To return the lever to the position shown requires almost three-quarters of a revolution. In other words, the slow motion of lever *E* and shaft *G* is obtained while work is being performed on the machine, and the quick motion returns them to their starting positions.

A contrasting application of the same mechanism is the so-called "short eccentric block movement," Fig. 16. In this case, sliding block *F* is free to pivot about an eccentric stud *D*, which projects from flange *C* on driven shaft *B*. The shaft is mounted between frames *A* of the machine. Block *F* slides within the slotted box-like lever *G*, pivoting the lever about shaft *J*. Connecting-rod *H* joins one end of lever *G* with bellcrank lever *N*, which, in turn, reciprocates member *R* in ways provided on the table of the machine.

An interesting variation of the sliding-block movement is the cam-operated mechanism shown in Fig. 17. In this case, the lever *M* containing the sliding block *P* is on the driven rather than the driving end of the mechanism. Roll *B*, which rotates about a stud extending from lever *C*, fits in the grooved face of cam *A*. Lever *C* is hinged on pin *D*, so that it oscillates through arc *H* as the cam is rotated.

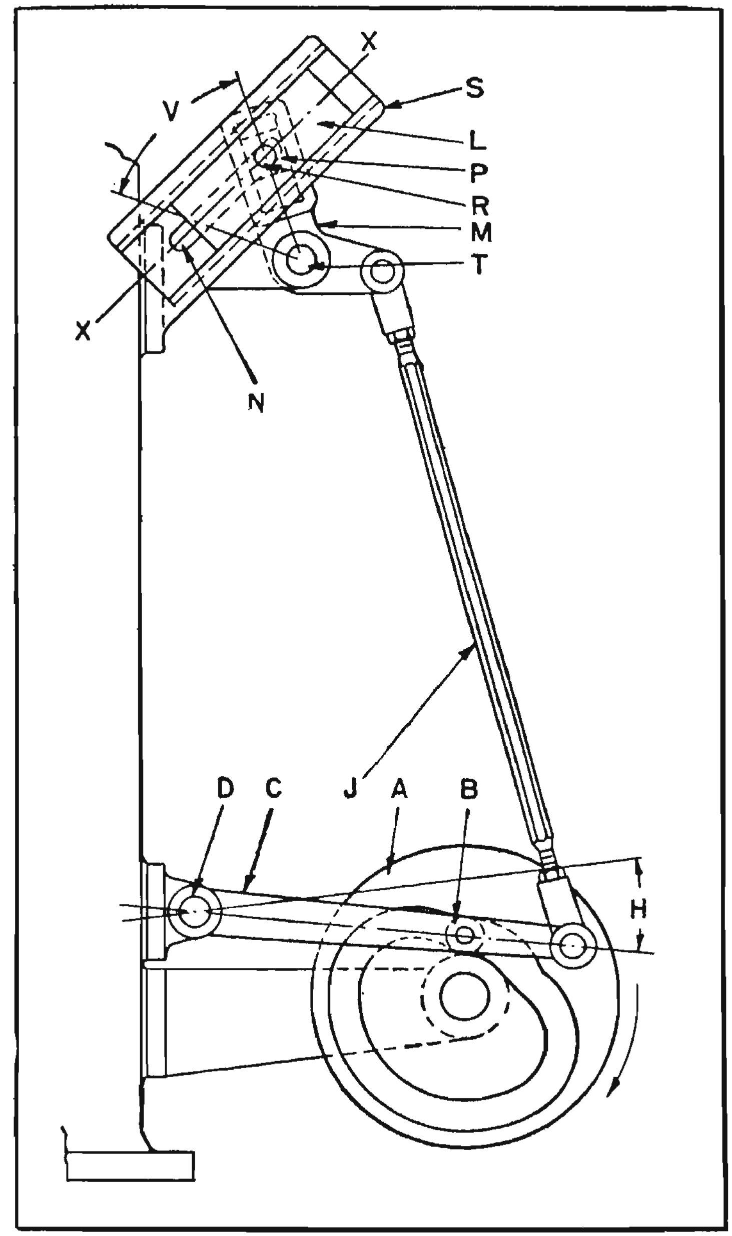


Fig. 17. Cam-operated Sliding Block Mechanism which is Employed to Reciprocate Slide L in Ways Provided on Bracket S.

This motion is transmitted to the bellcrank lever *M* by connecting-rod *J*. The bellcrank lever is free to pivot about pin *T*, and its longer arm is slotted to fit sliding block *P*. Stud *R*, which passes through the sliding block and is secured to slide *L*, is confined to movement along center line *XX* by an elongated slot *N* in bracket *S*. As the slotted arm of the bellcrank lever is pivoted through arc *V*, slide *L* is reciprocated along center line *XX* in ways on bracket *S*.

In making such sliding-block mechanisms, it is sometimes advantageous to have the slot in which the block travels open at one end. This type of forked construction facilitates assembly, and is generally satisfactory when the travel of the block is small. In cases of longer travel or high-speed operation, it is usually necessary to increase the strength of the forked member or resort to a box type lever.

Positive Reciprocating Mechanism.—Machines of certain types sometimes require feeding slides that are given a positive, accurately timed, reciprocating motion with a dwell at each end of the stroke. In the case of the mechanism shown in Fig. 18, the feeding slide *S* is required to be positively locked in the position indicated by the full lines while driving shaft *T*, rotating continuously in one direction, completes a certain portion of a revolution, following which the feeding slide *S* is moved to the position at the right indicated by dotted lines. The slide then remains locked in this position for a certain portion of a revolution of shaft *T*, after which it is returned to the position shown by the full lines. This cycle is repeated continuously as long as the driving shaft *T* continues to rotate.

While it would be possible to obtain a reciprocating and dwell motion of this kind by the use of simple cams, the usual cam and follower arrangement would not insure the necessary positive locking action, accurate positioning during the dwell period, and freedom from lost motion or backlash provided by the mechanism illustrated. As shown in

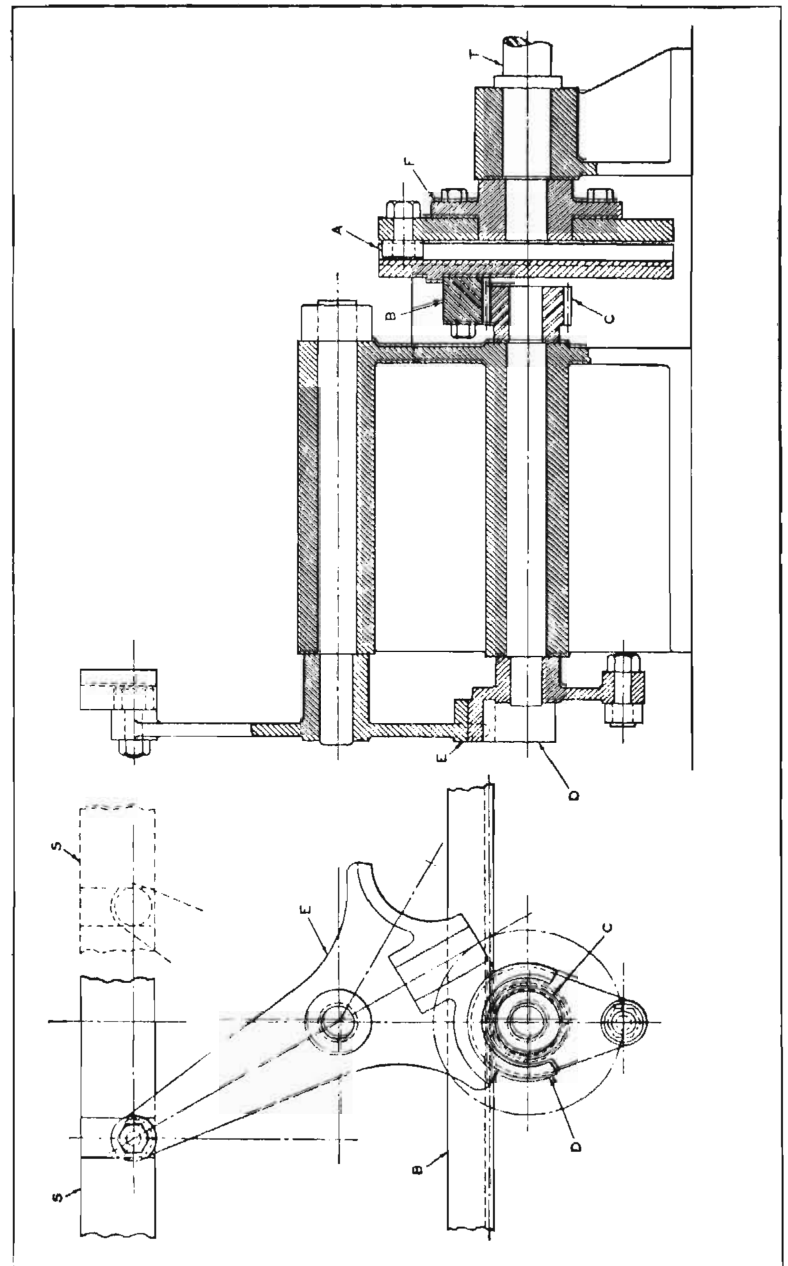


Fig. 18. Mechanism by Means of which Continuously Rotating Shaft T Causes Slide S to Reciprocate with a Dwell Period at Each End of Stroke.

the view to the right, the mechanism consists of two units—the driving crank unit shown at the right-hand end of the assembly or section view, and the Geneva dial mechanism indicated at the left-hand end.

Members D and E are laid out as a six-station Geneva motion, which allows D to revolve through an angle of 240 degrees while E remains stationary in either one of the two positions. This provides for the rotation of D through an angle of 120 degrees while transferring E from one dwell position to the other.

Reversal of member D after completing one revolution, as required to give slide S the reciprocating movements and dwells described, is accomplished by a crank with a roller A which revolves continuously with shaft T. Roller A engages a vertical slot in a member attached to rack B, as indicated in Fig. 19. Rack B, in turn, meshes with a pinion C on the shaft on which member D is mounted.

Thus continuous rotation of shaft T causes rack B to reciprocate, driving D first in one direction and then in the other. Rack B rotates driver D one full turn during

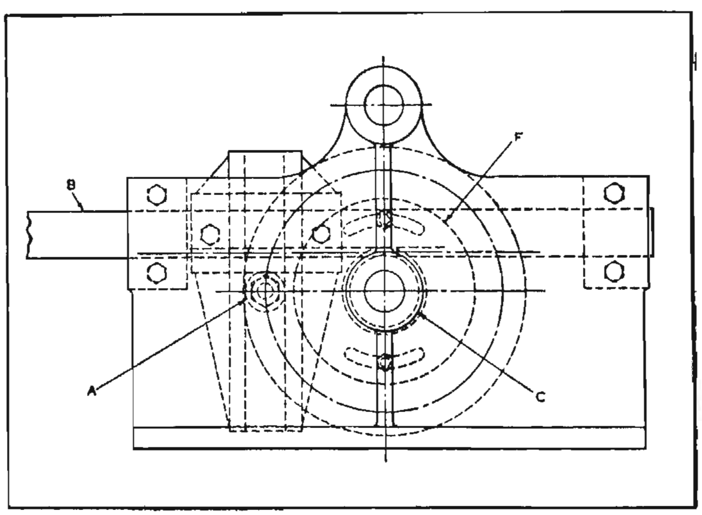


Fig. 19. End View of Mechanism Illustrated in Fig. 18.

one-half of a revolution of shaft T, and then reverses and drives it one full revolution in the opposite direction. Flange F, to which the disk carrying the driving roller A is attached, has elongated slots for the fastening studs in the flange to permit adjusting the angular position of roller A.

Swivel Joint Mechanism for Changing Direction of Movement.—Although the mechanism shown in Fig. 20, designed for changing the direction of movement transmitted by a lever, is not suitable for use where considerable rigidity is required, it nevertheless has much to commend it where the force applied is comparatively small.

The illustration shows one corner of the machine table on which the mechanism is used. The rod A has a reciprocating movement, rocking lever B about shaft C as a fulcrum. Shaft C is supported by two brackets M, the one at the front end of the shaft not being shown. This motion is transferred by the mechanism to a horizontal reciprocating slide D. There is a swivel link unit at E on the end of lever B. Lever H, being an integral part of the hub of lever B in the form of a separate arm, is also rocked back and forth by rod A.

Swivel joint F, of which S is a connecting-rod having a second swivel joint at G working in conjunction with the swivel link K, transmits motion in a horizontal plane to a rocker lever L. The horizontal lever L pivots on stud N, transmitting motion to link P, which, in turn, transmits the required reciprocating movement to slide D. Slide D is held on the base of the machine by means of guide plates Q. The opposite end of slide D serves to actuate transfer plates in the proper sequence of operations performed by the machine.

The principle involved in the swivel connections at E, F, and G is the same in all instances. The link E carries a pivot pin R, held in place by means of a washer and cotter-pin. This permits an up and down rocking movement of rod A. It also permits a rocking action of the joint forward or back, as well as sidewise, in the horizontal plane. The joint G at the end of the connecting link S allows a

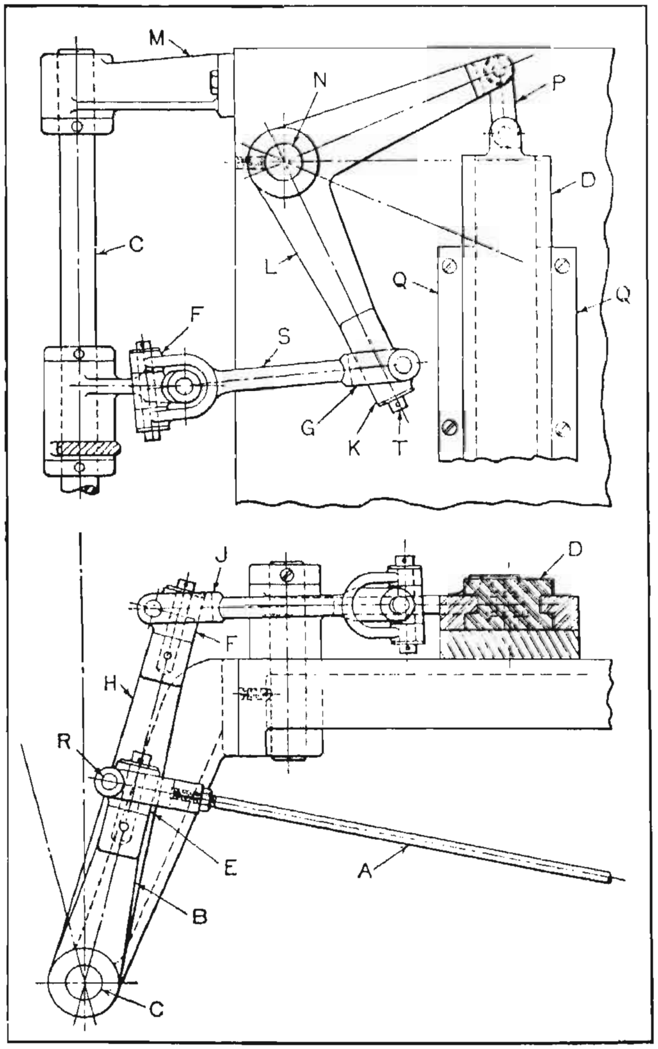


Fig. 20. Swivel Joint Mechanism by Means of which Reciprocating Rod A Operates Slide D.

swiveling motion in the horizontal plane, while pin T permits a swiveling motion in the vertical plane.

Therefore, regardless of the inclination of the connecting links or the variations in height due to the radial action of the several rocking levers, movement is transmitted from the rod A to the slide D. While the mechanism appears somewhat complex because of the many angles involved, it nevertheless often provides the simplest arrangement for obtaining the desired results.

CHAPTER 8

Crank Actuated Reciprocating Mechanisms

Among the special features of the reciprocating mechanisms described in this chapter are: operation of two slides alternately from one shaft; providing straight line reciprocating motion without support by ways or slides; synchronizing of horizontal and vertical motion; operation of two reciprocating slides from a single crankpin; provision of crank motion with dwell or rest period and obtaining reciprocating motions in two different positions in each cycle.

Other crank actuated reciprocating mechanisms are included in Chapter 9 of Volume I and are covered in Chapter 8 of Volume II of "Ingenious Mechanisms."

Mechanism for Operating Two Slides Alternately from One Shaft.—Figs. 1, 2 and 3 show the construction of a mechanism used on a packaging machine. Two slides *B* and *S* are operated alternately from the rotating shaft *D*, one of the requirements being that one slide start its movement when the other stops. Figs. 1 and 3 show the end and plan views.

The stationary part *A* of this mechanism is dovetailed to hold the two slides *B* and *S* on opposite sides, as shown in Fig. 1. Bearing *C* supports shaft *D*, which rotates in the direction indicated by the arrow. Lever *E* is free to turn on shaft *D*, and carries at its upper end the gear *K* which runs free on its stud. Gear *K* receives its motion from gear *L* which is keyed to shaft *D*. Connecting-rod *F* is carried on the stud on the upper end of lever *E* and the stud on slide *B*. Connecting-rod *J* is carried on a stud on gear *K* and a stud on slide *S*. Slides *B* and *S* are slotted, pin *G* passing through part *A* and acting as a stop for

both of these slides. Spring *H* serves to draw lever *E* to the right.

Taking Fig. 3 as a starting point in the cycle, slides *B* and *S* are held against pin *G* by spring *H* acting on lever *E*. The rotation of gear *L* in the direction indicated by the arrow causes gear *K* to rotate in the opposite direction; this, in turn, causes rod *J* to draw slide *S* to the left, slide *B* remaining stationary until slide *S* is again returned to its resting point against pin *G*. In Fig. 3, the dotted

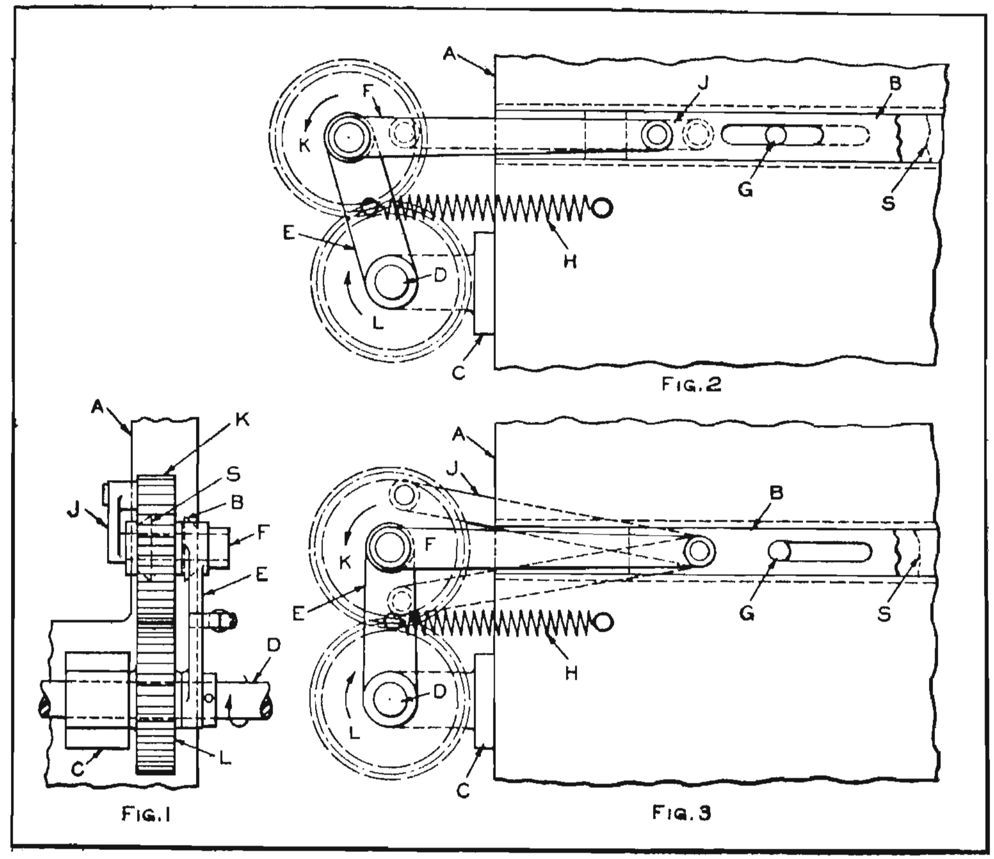


Fig. 1. End View of Mechanism, Showing Dovetail Slides B and S on Opposite Sides of Part A. Fig. 2. Plan View, with Slide B in Central Position and Slide S in Right-hand Position. Fig. 3. Slides B and S in Extreme Right-hand Positions.

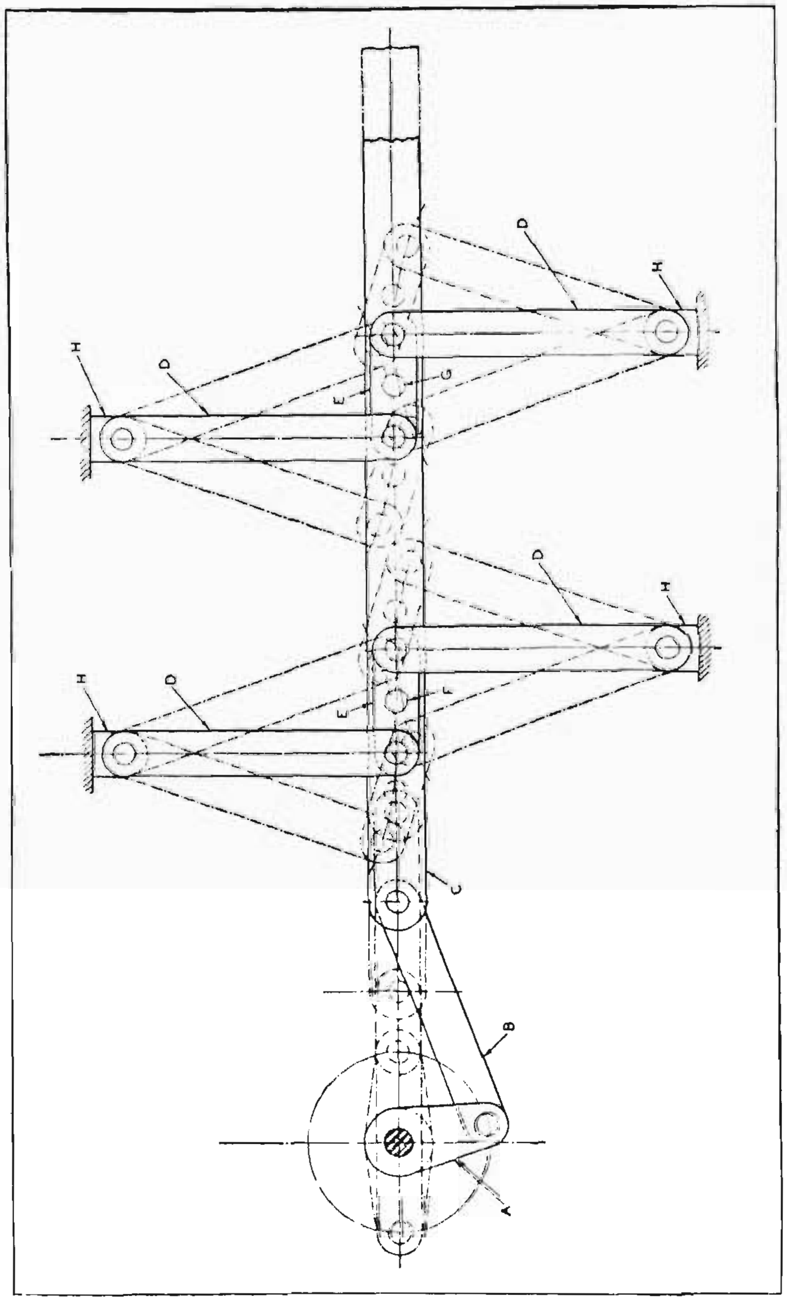


Fig. 4. Mechanism for Imparting Straight-line Motion to Ram C without Support by Ways or Slides.

outline of lever J indicates its position after slide S has completed its movement. As slide S is restrained from further movement to the right, continued rotation of gear K brings rod J to the center position, as in Fig. 2, thus causing lever E to swing to the left against the resistance of spring H. The movement of lever E draws slide B to the left, returning it to its resting point against pin G as rod J passes the center position, thus completing the cycle.

Straight-Line Reciprocating Motion for Link-Supported Ram.—A horizontal ram located in an atmosphere laden with abrasive material was required to have a straight-line reciprocating motion without being supported by ways or slides. The linkage devised to meet these requirements is shown in Fig. 4. Crank A and connecting-rod B impart the required reciprocating movement to ram C. In order to confine the motion to a straight path, four links D and two links E are utilized. These links are connected to ram C at the pivoting points F and G, the whole linkage mechanism being swiveled at the four stationary bearings H. All bearings are sealed to protect them from the abrasive material.

Links D and E are of such proportions that their centers F and G move in a straight line; and since the ram is connected at these points, it also moves in a straight path. Links E are approximately 38 per cent as long as links D, and the stroke of the ram does not exceed 65 per cent of the length of links D. This limitation on the length of the stroke with respect to the length of the links is necessary because points F and G only move in a straight line within a certain distance, beyond which they begin to move in a curved path.

Gear and Link Mechanism for Synchronized Horizontal and Vertical Motion.—The mechanism shown in Fig. 5 was designed to move work horizontally to a position where a vertical lift is applied. Referring to the illustration, the

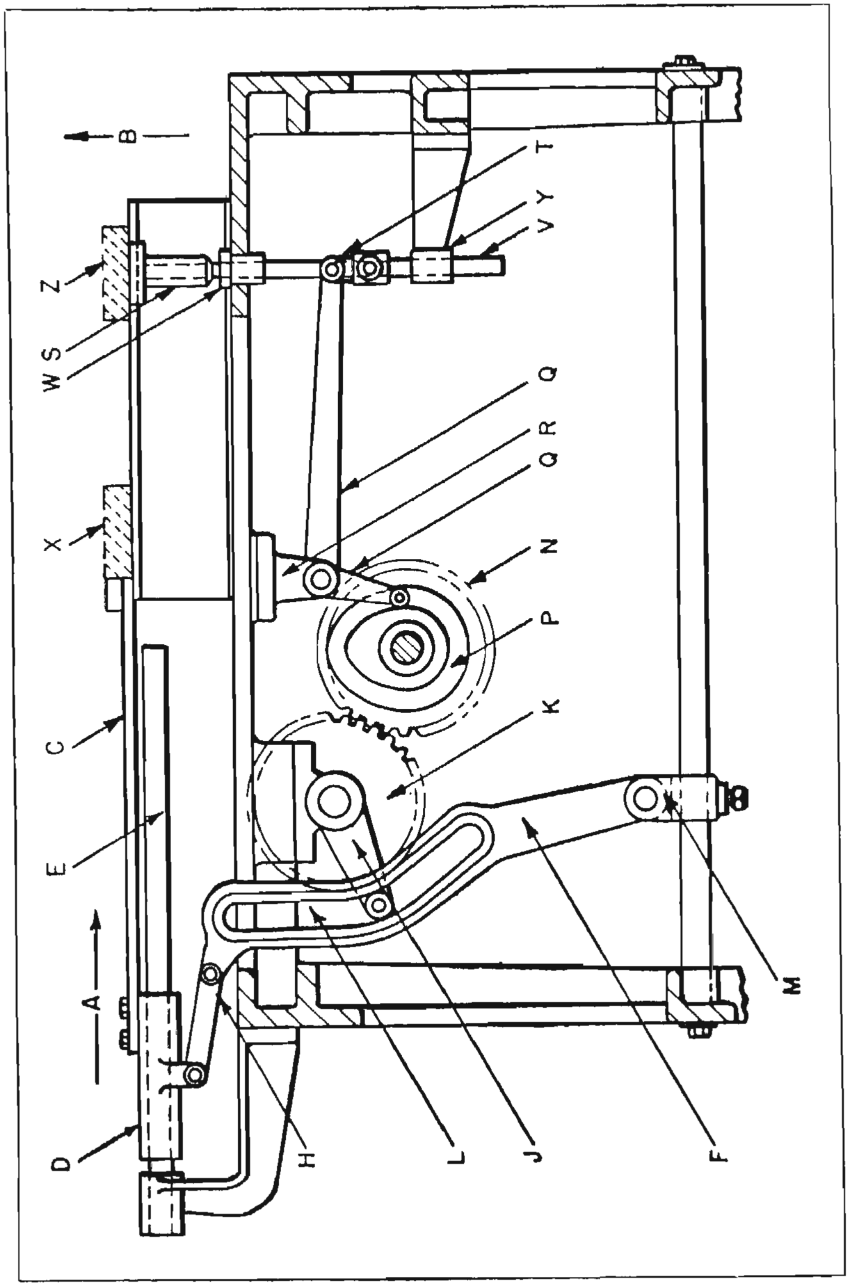


Fig. 5. Schematic Diagram for Gear and Link Mechanism for Synchronized Horizontal and Vertical Motion.

work X is to be moved in the direction indicated by the arrow A. Upon reaching position Z, the work is raised, as shown by the arrow B.

The horizontal pushing plunger C is attached to a slide D, which moves along two rods E when pulled by the lever F through the medium of the connecting link H. A crank J, rotating with gear K, operates in a slot L in the lever F, thus moving the lever so as to actuate the plunger slide to the right and left. Lever F is secured by means of a pivot to a stationary sleeve M, which is attached to a tie-rod in the base of the machine.

Gear K is driven by gear N in which there is a cam groove P. The purpose of the cam groove is to operate the bellcrank lever Q, which is attached by a pivot to the fixed bracket R. The movement of the bellcrank lever causes the lifting plate S to be carried up or down on the rod V, which is connected to the bellcrank lever by means of the linkage shown at T. Bearings at W and Y guide the lifting rod.

Timing of the horizontal and vertical movements relative to each other is accomplished by proportioning the slot L in lever F so that the forward movement of the work in the direction of arrow A is accelerated and then decelerated gradually as the crank J travels in the slot. A quick return stroke is provided by the angular section of slot L. The cam groove P in gear N is so positioned that lifting of the work at Z takes place during the return movement of lever F and plunger C.

Mechanism with Two Reciprocating Slides Driven from One Crankpin.—On a wire forming machine, a reciprocating motion was imparted to a sliding member by a crank disk to which it was connected by a pitman rod. Owing to a change in the design of the product, it became necessary to add another sliding member, which would carry the original sliding member. The added sliding member had to be given a reciprocating motion relative to the original

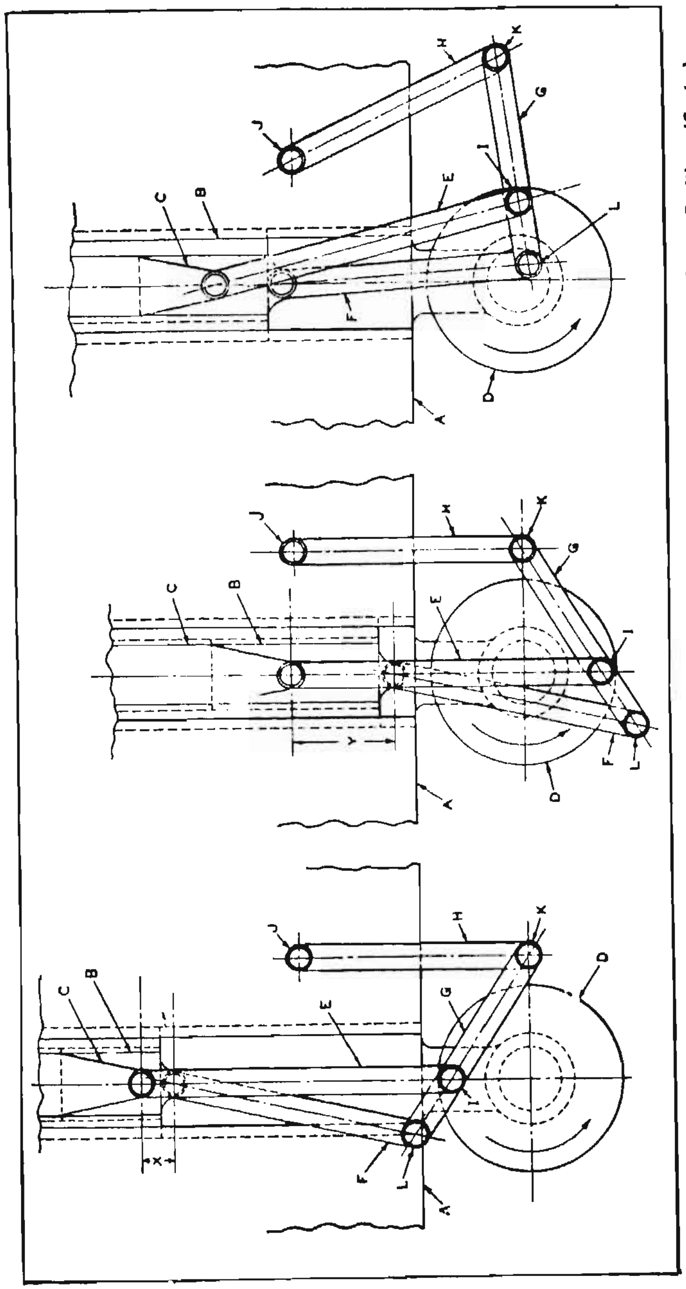


Fig. 6. (Left) Mechanism with Two Slides B and C Driven by One Crankpin I Shown at Top Dead Center Position. (Center) Mechanism with Crankpin I Shown at Bottom Dead Center Position. (Right) Mechanism Shown with Slides in Position Occupied After Crankpin I has made Three-fourths Revolution from Position Shown in Left View.

sliding member, both motions being transmitted from the same crankpin. This was accomplished in a comparatively simple manner, as shown in the accompanying illustrations.

Referring to the left-hand diagram in Fig. 6, the disk *D*, rotating in the direction indicated by the arrow, imparts a reciprocating motion to the slide *C* through the crankpin *I* and the pitman rod *E*. This constituted the original movement before any changes were made. The slide *B*, which was added, has a dovetailed base which is a sliding fit in a dovetail groove in member *A* of the machine, and carries the part *C* which is similarly mounted in slide *B*.

The lever *G*, pivoted on crankpin *I*, is connected at one end to the link *H* by the stud *K*, and at the other end to the pitman *F* by the stud *L*. Pitman *F* is connected to slide *B*, and pitman *E* to slide *C*, each transmitting the motion received from lever *G*, which is actuated by crankpin *I*. Link *H* is pivoted on stud *J*, which is stationary on member *A*.

In the left-hand diagram of Fig. 6, crankpin *I* is shown at the top dead center position. At this point of the cycle, slides *B* and *C* occupy the relative positions indicated by the dimension *X*, measured through the centers of the studs. In the center diagram of Fig. 6, crankpin *I* is shown at the bottom dead center. As lever *G* is operating in the third order, the greatest movement takes place at the work end, or at stud *L*. By virtue of this action, slide *B* is given a greater downward movement than slide *C*. The relative positions of the slides *B* and *C* are shown by the dimension *Y*, which is greater than dimension *X*, Fig. 6, left-hand diagram, indicating the difference in the movements of the two slides *B* and *C*.

In the right-hand diagram of Fig. 6, crankpin *I* is shown in the position it occupies after completing three-quarters of the cycle. This illustrates the action of link *H*, the purpose of which is to provide a floating fulcrum for lever

G without resorting to a slotted member or other form of track for stud K.

Crank Motion with Rest Period.—A crank motion with a dwell or rest period, designed for use on a wire forming machine, is shown in Fig. 7. This mechanism imparts a reciprocating movement to a part in the usual accelerating and decelerating cycle manner, except that a rest period is provided at one end of the stroke.

Referring to the upper left-hand view, the disk *A*, rotating in the direction indicated by the arrow, carries the bar *B* in a slot in which the bar slides freely. Bar *B* is retained by a cover plate (not shown), and is held in its extreme outer position by the spring *F*, carried on a rod that passes through the block *G*. The nut on the end of this rod restricts the movement of bar *B*, and is used for

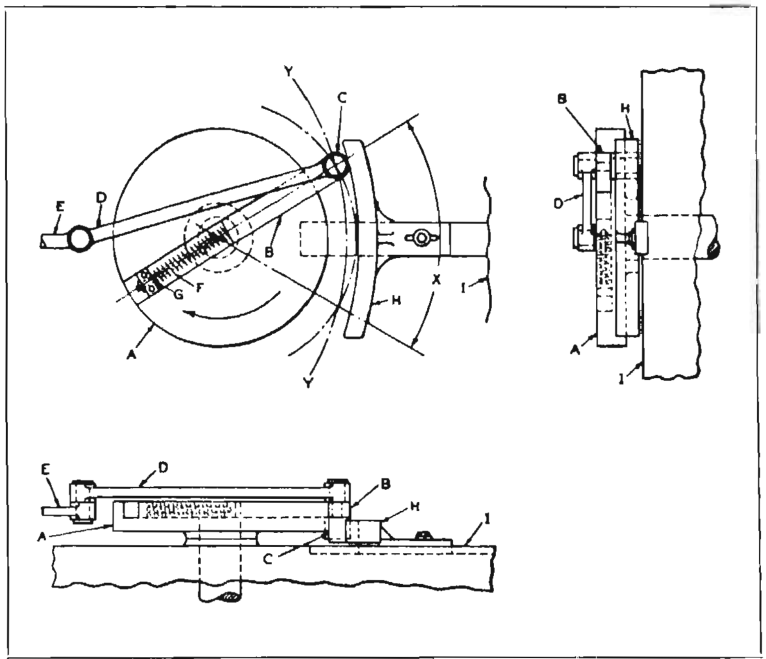


Fig. 7. Crank Drive for Reciprocating Slide that Provides Dwell at One End of Stroke.

making slight adjustments. Bar *B* is connected at its outer end with the rod *D*, which, in turn, is connected to the rod *E* to which the reciprocating motion is to be imparted by rotation of disk *A*.

The roller *C*, lower view of Fig. 7, carried on the under side of bar *B*, makes contact with the guide *H* during a portion of the cycle. Guide *H* is adjustably mounted on the stationary part *I* of the machine. The contact surface of guide *H* is formed to an arc of a circle having a radius equal to the center-to-center distance of the studs on rod *D* plus half the roller diameter.

Again referring to the upper left-hand view of Fig. 7, the rotation of disk *A* normally carries the center of roller *C* in the circular path indicated by the dot-and-dash circle concentric with disk *A*. However, as roller *C* contacts guide *H*, it cannot continue its normal path, but follows the path of the face of guide *H*. As this surface is formed in the arc of a circle that has its center on the center of the stud connecting rods *D* and *E*, rod *D* at this point rotates on this center, and there is no movement of rod *E* until roller *C* again leaves guide *H*. This causes a shortening of the stroke equal to the distance shown between the two arcs, and produces a rest period for rod *E* equal to the angle *X*.

Positive Crank Motion that Causes Slide to Dwell at Both Points of Reversal.—The use of a cam of ungainly size was avoided in the design of a wire-forming machine by employing a rather ingenious crank motion that produces a dwell at each end of its stroke. The mechanism employed is shown in Fig. 8. It imparts a relatively long stroke (6 inches) to a slide, and produces a dwell at each end of the stroke. The movement is positive and smooth. The compact nature and simplicity of the design may suggest its application to other types of machines requiring a similar movement.

The crank indicated at *A* is mounted on the drive shaft *B*, the latter being supported in the stationary bearing *C*.

A roll D engages the two grooves E and F in the driven slide G, which is mounted on the machine frame H. Positive action of the roll in passing from one groove to the other is assured by the switching arm J. This arm is secured to stud K, which is a free turning fit in the slide. It will be noted from Fig. 8 that the face of the roll is

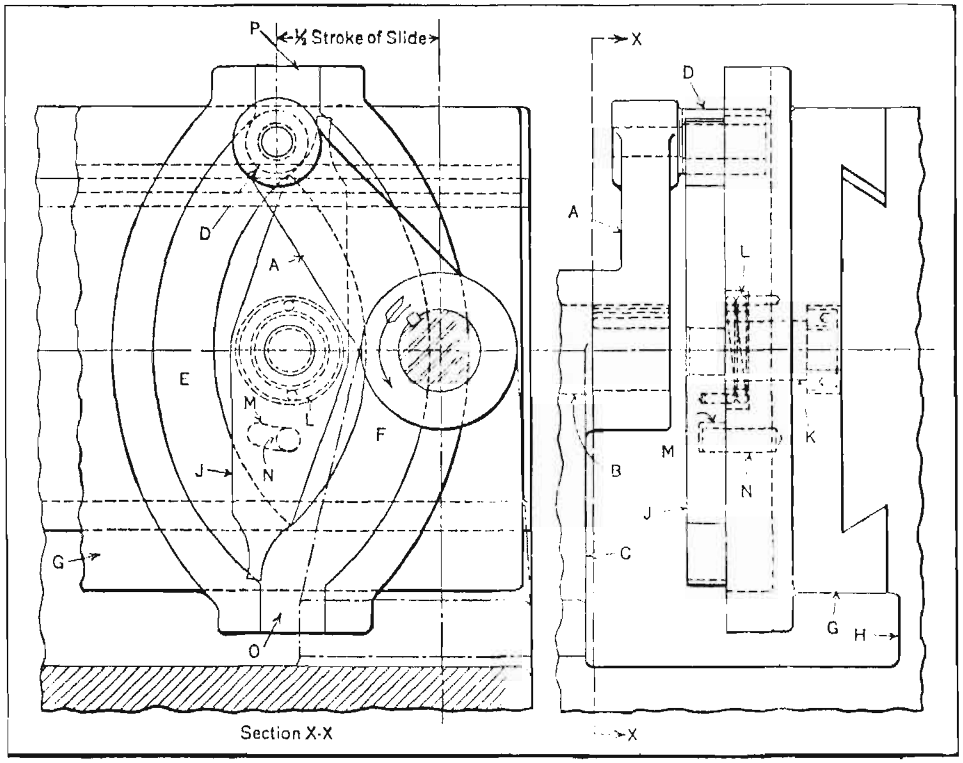


Fig. 8. Crank Motion that Reciprocates Slide, Allowing Dwell at Each Point of Reversal.

long enough to engage both the groove and the arm. Normally, the arm is held in the position shown by the torsion spring L, which forces the right-hand end of the elongated slot M in the arm against the pin N in the slide.

As the drive shaft rotates in the direction indicated by the arrow, the roll is guided by the arm into groove E.

Since the radius of this groove coincides with the radius of the path of roll D as crank A is rotated by the drive shaft, no motion of the slide will result and, consequently, the slide will have the required dwell at this end of the stroke. When the roll reaches the lower end of this groove, however, it pushes the arm J around in a counter-clockwise direction until the left-hand end of slot M engages pin N. In this way, the lower end of the arm forms a part of a continuous groove leading from groove E into the vertical groove O; and as the crank continues to rotate, the slide is carried toward the right to the end of its stroke. At the middle of this stroke, the roll is at its lowest position, at which time the lower end of the arm clears the roll. This allows the spring L to force the arm back to the normal position shown, so that a continuous groove leading from groove O to groove F results.

When the slide has reached the end of its stroke toward the right, the roll enters the concentric groove F, allowing the slide to dwell until the roll reaches the top of this groove. At this point, the roll engages the arm, forcing its upper end toward the left, so that it serves as a guide for transferring the roll to the vertical groove P. Continued rotation of the crank causes the slide to move toward the left to the position shown, the switching action of the arm being identical to that which took place at the bottom of the slide when the roll entered and left groove O.

It is possible that this mechanism would operate without the switching arm, but the action would not be certain. That is, the roll would be likely to jam at either point of reversal should the slide be accidentally displaced. Moreover, the impact of the roll, at each point of reversal against one corner of each of the grooves O and P would certainly damage the face of the roll or the groove corner. In addition to these disadvantages, a jerky action would be obtained at the points of reversal, which is absent when the switching arm is used.

Double Reciprocating Mechanism with Displaced Operating Positions.—The mechanism shown in Fig. 9 was developed to produce two distinct reciprocating movements of the feeding rod *S*. The limits of each of the reciprocations are identical, the difference being in the relative positions of the rod when the reciprocations take place. The action is best described by referring to the diagrams

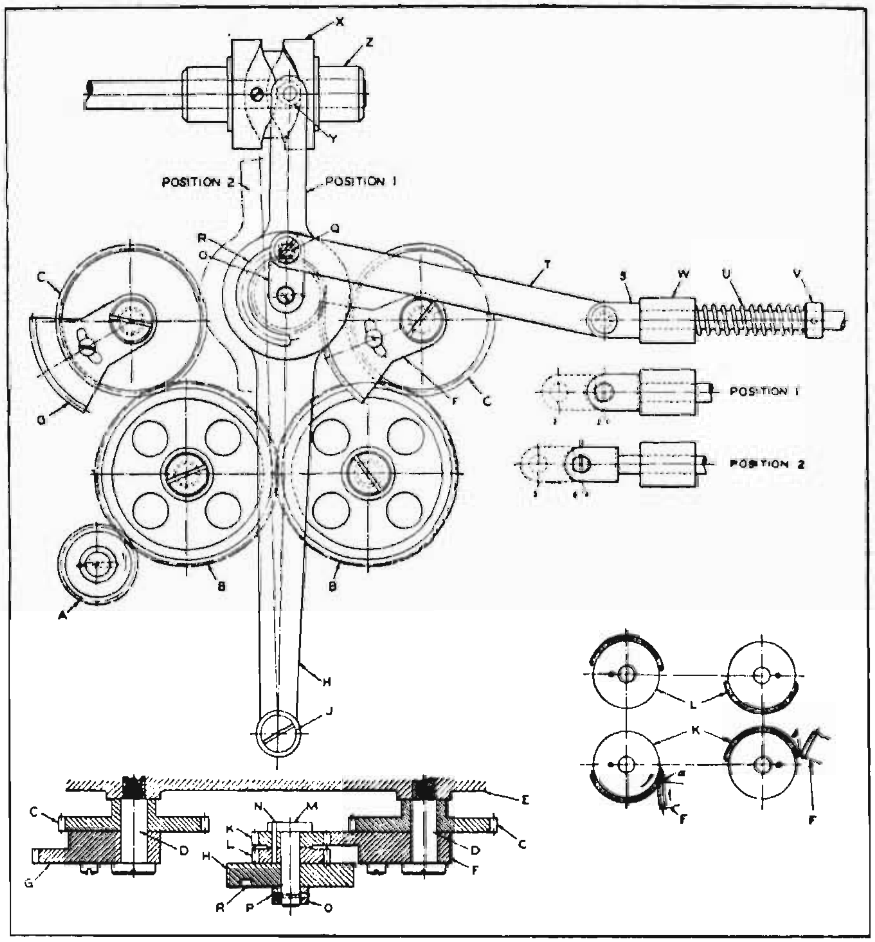


Fig. 9. Mechanism for Producing Reciprocating Movements of Rod S in Two Different Positions. At Lower Right are Shown Form and Positions of Gears K and L at Two Moments in Position 1.

which show the movements in the two reciprocating positions. The diagram below rod *S* designated Position 1 represents one reciprocating movement starting at a point marked 1, extending to 2, and finally returning to 3. When the mechanism is displaced, so that the upper end of rocker *H* occupies the position indicated by dotted lines in Position 2, the reciprocation starts at 4, extends to 5, and comes to rest at 6. This sequence of movements may be likened to the movements of a crank whose entire structure can be displaced or moved from Position 1 to Position 2, thereby giving the same reciprocating movement at two distinct points or positions separated by a definite distance. Provision is also made for time adjustment between the reciprocations.

The mechanism is actuated by gear *A*, which drives idlers *B*, the latter, in turn, driving the gears *C* on the studs *D*, in the main plate *E* of the machine. Gears *C* are provided with gear segments *F* and *G*. A slot in the gear segments permits adjusting their positions on gears *C* to produce a definite relation between the positions of the two segments in their continuous rotation with the gears.

The rocker arm *H* is pivoted on the stud *J* and is free to oscillate between gears *C*. Two mutilated gears *K* and *L* are fastened to the shaft *M* by pin *N*. In order to retain shaft *M* in place on the rocker arm *H*, the link or crank-arm *O* is fastened to the shaft *M* by pin *P*. Crank *O* is provided with the guide pin *Q* which fits into the concentric slot *R* in the rocker-arm head. Crank *O* is connected to the reciprocating rod *S* through the connecting-rod *T*.

The spring *U* furnishes the necessary tension between a collar *V*, fastened to rod *S*, and the bearing block *W*, which is a part of the main plate *E*. The cam *X* actuates rocker arm *H* between the gears *C*. A roller *Y* at the end of the rocker arm fits in the slot of the cam and produces a simple oscillating motion of the rocker arm. The bearing blocks Z support cam X.

The functioning of the mechanism is controlled by cam X. The cam is so designed that one revolution is necessary for a complete cycle of the reciprocating member S. One cycle consists of starting in Position 1 and returning finally to the same position. The cam can be given any irregular shape or form necessary to give any time cycle required of the reciprocating mechanism.

The mutilated gears K and L are alternately brought into contact with the gear segments F and G, which are displaced in such a manner as to make contact with their respective mutilated gears. Fig. 9 shows the form and relative positions of the mutilated gears at two moments in Position 1. First, the gear segment F is in contact with gear K at a ready to rotate the unit. The position of gear L should be noted at this time. The next view shows the gear segment F after it has completed its driving movement of gear K and is passing away from it at b. The changed position of gear L at this point should be noted. Gear L is now in such a position that when the rocker arm H is moved to Position 2, gear segment G can be engaged with it and made to produce a reverse rotation of gear L, returning link O to the upper part of the rocker-arm head and thus completing the reciprocation for Position 2.

It will be noted that points 1 and 3, as well as points 4 and 6, shown in the two position diagrams do not coincide. This is due to the fact that the slot R, subtending an angle of 180 degrees, combines with the angularity of rocker H to produce a slight difference in positions. However, this difference is necessary in the process to which the mechanism is applied. It is interesting to note that by slightly extending the slot at each end, the difference can be eliminated, and the points 1 and 3, as well as 4 and 6, can be made to coincide.

An interesting feature of the mechanism is that it provides a wide range for the timing of the interval between reciprocations. The timing is controlled by cam X, which is so interconnected with gears C that their positions are in a definite relation to the position of the cam X at all times. With the cam rotating intermittently, so that there is a sufficient pause in each position of rocker arm H to give the segment gear time to complete its function, there will be a definite time interval between reciprocations, dependent upon the relative positions of the two gear segments on their respective gears C. The interval can be varied by alternating the gear segments and bringing them closer together or spacing them farther apart.

CHAPTER 9

Variable Stroke Reciprocating Mechanisms

Means of adjusting the length, speed or timing of the reciprocating stroke are provided in mechanisms described in this chapter. Part of the stroke may be accelerated or retarded or the reciprocating motion may have a constantly varying length of stroke. Variation in the length of successive strokes may be obtained in one of the mechanisms described, while another provides a means of adjusting the length of stroke during operation. A mechanism for varying the stroke of a toggle-lever press is described. Another interesting arrangement is a double-lever mechanism for obtaining a variably controlled range of action from a single cam.

Adjustable Oscillating Mechanism for Reciprocating Movement.—The length of stroke of a reciprocating member on a wire-forming machine required occasional adjustment to suit changes in the product. The reciprocating movement on this machine was obtained by the cam-operated oscillating lever shown in Fig. 1 at *B*, rod *J* being used to impart the required movement to the reciprocating member. The cam *D* rotates in the direction indicated by the arrow, the roller *C* on lever *B* being in contact with the cam surface.

Lever *B* fulcrums on stud *G*, which is located on the upper end of the short lever *A*, the latter, in turn, fulcruming on stud *H*. The lower end of lever *B* has an extension which is acted upon by the spring *F*. Normally, this spring serves to keep levers *A* and *B* in alignment. The stop-screw *E* in a lug on a stationary part of the machine limits the movement of lever *A*.

In the view to the right, the roller *C* is shown in contact with the middle portion of the lobe on cam *D*, the rise from the lower lobe of the cam being accomplished while levers *A* and *B* operate on stud *H* as a unit. As the upper lobe of cam *D* reaches roller *C*, continued movement causes screw *E* to make contact with lever *A*, preventing further movement of the latter lever, so that lever *B* then fulcrums on stud *G*, as indicated in the view to the left. The change in the ratio of leverage increases the movement of rod *J*,

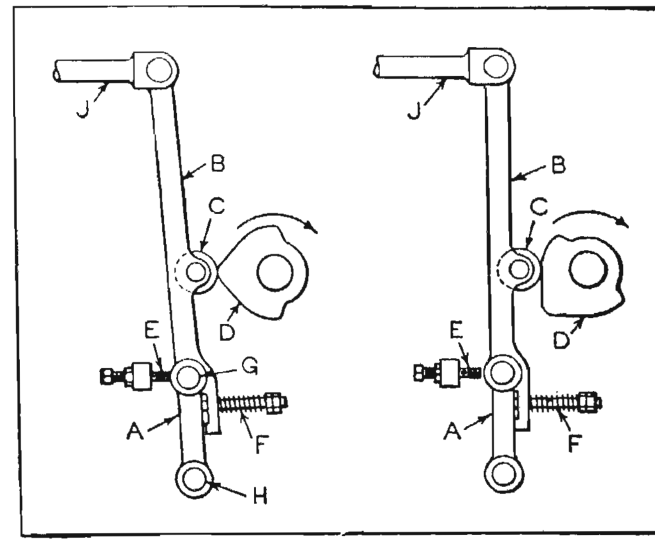


Fig. 1. Cam-operated Oscillating Lever with Stroke Adjustment, Used to Obtain Reciprocating Motion.

as controlled by the setting of stop-screw *E*. Roller *C* is held in contact with cam *D* by a spring which acts against the working end of rod *J*.

Mechanism for Varying Speed of Slide.—On a certain type of forming machine it was necessary to modify the action of the reciprocating slide carrying the metal-tape forming tools, so that a part of the forward and return stroke of each cycle of slide movement could be increased radial position, and is deep enough to insure free working of the roller as bellcrank lever F swings on its fulcrum G.

It will be seen that the rim of the cam-plate is built up in diameter for a short distance at the right-hand side of the slot O. The roller J is normally held in contact with the rim portion of the cam-plate at the left of the slot, being maintained in this position by the stepped portion P of the boss at the end of lever C. The straight side of this step is so arranged as to bear against the left-hand side of the bellcrank lever F. In this position, sufficient clearance must be left between the step and the side of the lever to permit the roller to rotate freely.

In operation, commencing the cycle of movements from the position shown in the left-hand view, shaft A is oscillated toward the right, carrying with it the lever C, and this, in turn, carries along at the same speed the bellcrank lever F and the connecting-rod attached to the slide. Throughout this stage the roller J runs along the circular rim of the cam-plate L in advance of the slot O. Thus the driven slide is actuated at the same slow speed as the driving shaft.

This slow speed will continue until the roller reaches the slot O in the stationary cam-plate. As soon as this point is reached, the roller passes down the slot, and continued movement of lever C causes the bellcrank lever F to be swung round on its fulcrum stud G, thereby imparting to the connecting-rod a movement at an increased speed—greater than that possessed by the rod during the first stage of its movements. The point of the start of this rapid rate of speed is shown in heavy broken lines superimposed upon the left-hand view. When the roller is in radial line with the slot O, the built-up side of the slot insures the correct entrance of the roller, which might not occur if the length of the slot sides were identical.

The position of the levers at the termination of the clockwise stroke of the shaft is shown in the right-hand view,

from which it will be noted that the roller has passed well down the slot O, and the bellcrank lever F is accordingly swung around its fulcrum stud. Thus the first half of the complete cycle of movements of the driving shaft A imparts a slow speed to the reciprocating slide, followed by a much quicker rate of travel throughout the later portion of its stroke.

When shaft A starts the second half of its cycle by moving backward toward the left, the driven slide will start off at a fast rate, followed by a slower speed as the roller J leaves the slot. Thus each cycle of motions affords to the slide two slow and two fast movements.

By providing a simple means of adjusting the position of the cam-plate radially on its hub bearing, thereby altering the position of slot O relative to the vertical center of the arrangement, the point of the beginning of rapid travel can be varied within appreciable limits. This provision can be secured by passing the retaining screws M through elongated slots in the bracket B, thus enabling the cam-plate to be shifted. This will entail the omission of the dowel-pins N, which can be used when the cam-plate does not have to be moved.

This mechanism gives extremely smooth running and shockless transmission. It is simple, both in construction and design, inexpensive, and has simple means for changing the timing and duration of different speed rates for the driven slide.

Variable Reciprocating Motion.—A reciprocating mechanism employed on a wire fabricating machine to convert a uniform motion into a variable motion is shown diagrammatically in Figs. 3 and 4. The reciprocating motion was required to meet conditions of operation resulting from a change in the design of a certain part of the machine. Originally, the reciprocating slide, traversing a stationary part of the machine, was given a uniform motion, coming to rest at a predetermined point.

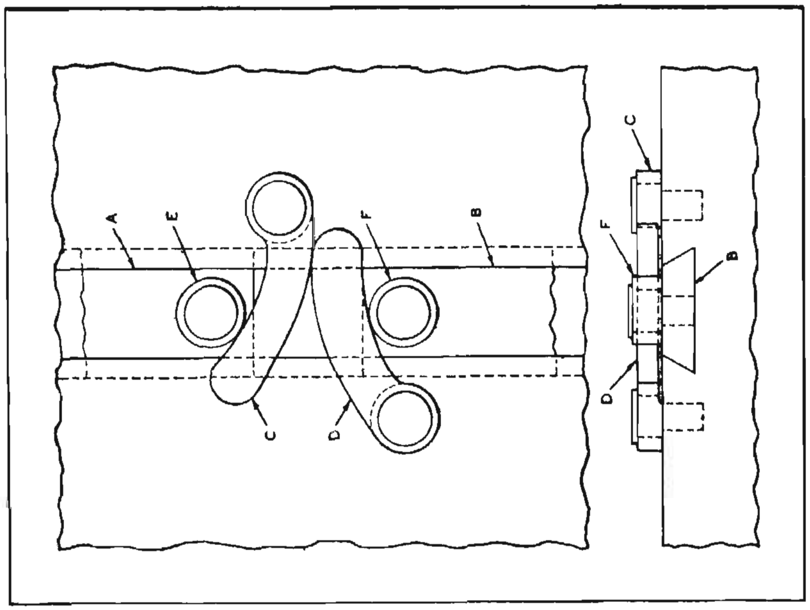


Fig. 4. Mechanism Shown in Fig. 3 with Slide Near Upper End of Stroke.

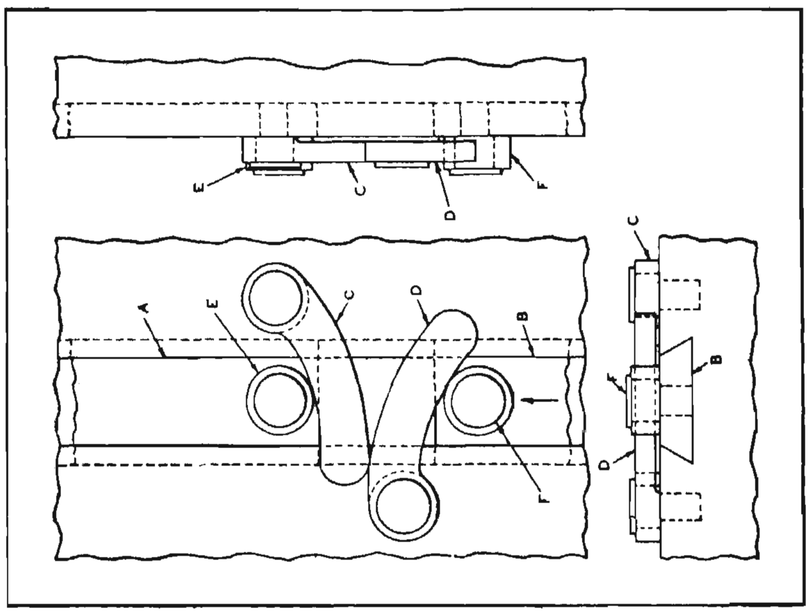


Fig. 3. Mechanism for Producing a Variable Reciprocating Motion.

Owing to the change in the part referred to, which reduced the clearance space available, it became necessary to retard the movement of the slide during a portion of its travel without changing the relative time at which the slide reached its ultimate position. This was accomplished by dividing the slide into two parts A and B, and introducing the levers C and D.

Referring to Fig. 3, slides A and B are dovetailed into a stationary part of the machine. Slide B, which is given a uniform reciprocating motion, carries the roller F which contacts with lever D, mounted on a fulcrum stud at the left. Slide A carries the roller E, which contacts with lever C mounted on the fulcrum stud to the right. Levers C and D are in contact at one point of their convex edges.

As slide B moves in the direction indicated by the arrow, its motion is transmitted through roller F to lever D, which is caused to pivot on its fulcrum stud. The motion of lever D is transmitted to slide A through lever C and roller E. It will be noted that D acts as a second-class lever on the end of lever C, which, in turn, acts as a second-class lever on roller E. Since it is a characteristic of the second-class lever to transmit the motion of the power member to the working point at a reduced speed and length of travel, it is obvious that slide A, at this point, is moving more slowly than slide B, thus providing the required clearance at the far end of the slide A.

As continued movement of slide B causes levers C and D to change their relative positions, the lengths of the lever arms are constantly changing, so that slide A is given an accelerated motion which, however, is less than the movement of slide B. As levers C and D change their relative positions, they eventually arrive in positions in which the contact points between rollers E and F and the levers lie on the same straight line. At this point, slide A is momentarily traveling at the same speed as slide B, since the motion is transmitted by direct contact, there being no lever action at this point. Further movement of slide B results in a continued acceleration of the speed of slide A, which is then moving faster than slide B. This is due to the fact, as shown in Fig. 4, that C and D are acting as third-class levers.

It will be noted that there is no change in the relative positions of slides A and B at the two extreme positions, indicating that the length of travel of slide A is equal to that of slide B. The ultimate effect of this arrangement is that of a continuous slide, with the added advantage of a retarded movement at the point where clearance space is required. Slides A and B are returned to their original positions by a spring (not shown).

Variable-Stroke Mechanism for Graduating Feed-Screw Dials.—The variable-stroke mechanism shown in Fig. 5 moves the head T up and down in synchronism with the indexing movement of the drum D, which stops the downward movement of the head as required for the production of the three different lengths of graduation lines on the work or dial A. The dials thus graduated are used on the cross-feed screws of turret lathes.

Referring to the illustration, the cutter-head T is actuated by the lever G and the connecting link J, which enters the sleeve H, in which it is securely held against the spring L by means of the spring K. The roll P at the lower end of sleeve H contacts with cam M, mounted on the main drive-shaft O. With this arrangement, the cutter-head T makes a full stroke at each revolution of the drive-shaft O and cam M.

The lengths of the graduation lines on dial A are controlled by the hardened steel circular drum or anvil D which comes in contact with the projection F on the sliding head T and thus interrupts the downward movement. While the cutter C is clear of the work, the anvil D is indexed by means of cam S on the main drive-shaft. Ten strokes of the cutter-head are required to complete the lines on one main division. Ten teeth, such as shown at R, are provided for indexing

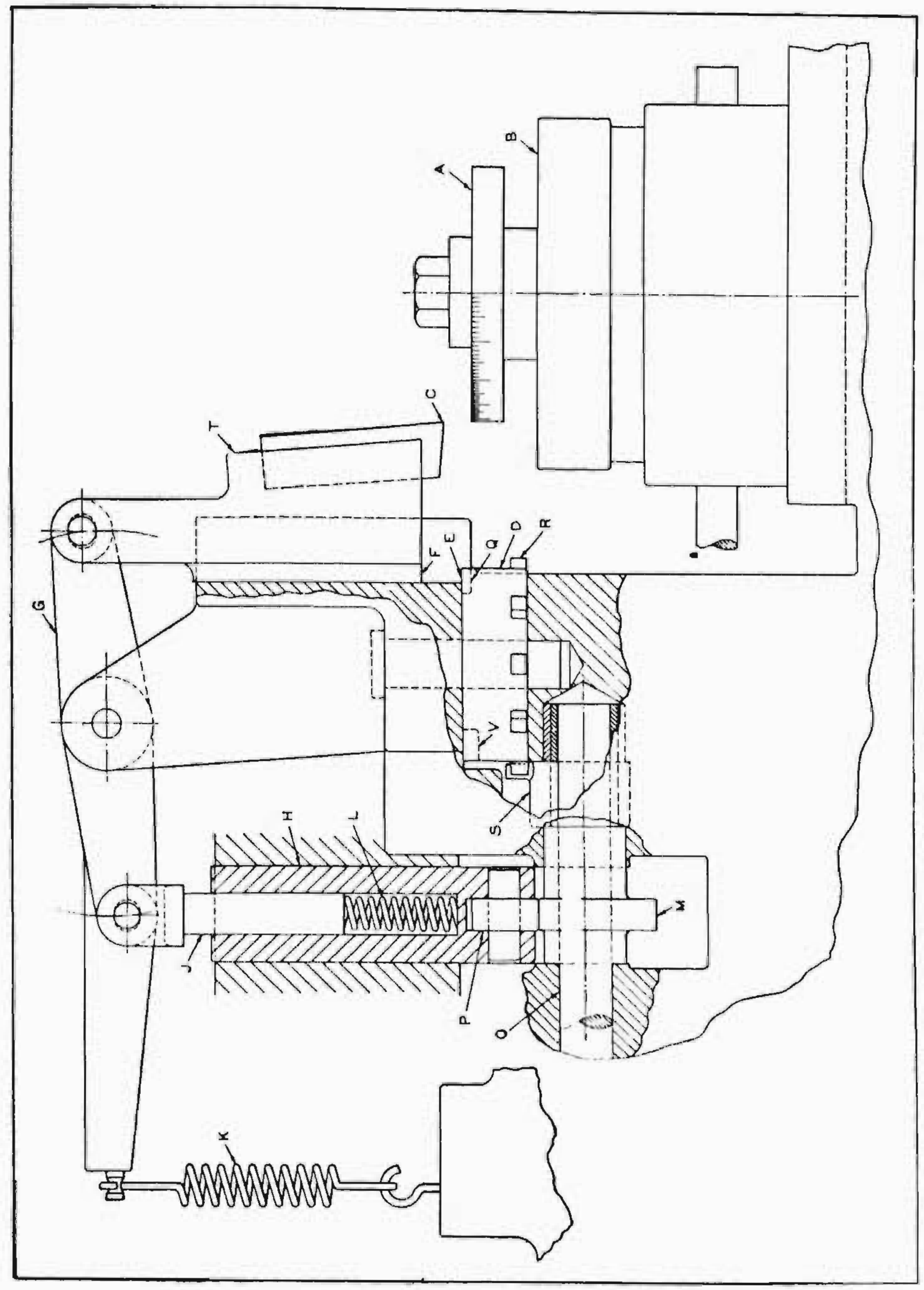


Fig. 5. Dial Graduating Mechanism which Automatically Varies Length of Stroke to Suit Different Lengths of Graduation Lines.

the anvil one complete revolution. Four of the strokes are interrupted at the short lines by the top face E of the anvil.

At the fifth indexed position, a slot Q in the face of the anvil receives the projection F, thus lengthening the stroke for the intermediate graduation line. A slot V, similar to the one at Q but diametrically opposite, allows the projection F to descend far enough to cut the longest graduation line.

The compression of spring L compensates for the various lengths of line. This spring also forces the cutter-head downward as sleeve H is lifted by cam M. Spring K serves to return the cutter-head to its uppermost position after the completion of the downward stroke. The faceplate B is indexed by a separate mechanism (not shown), which is properly synchronized with the variable-stroke mechanism.

Irregular Reciprocating Motion.—On a machine for manufacturing a wire product, it is necessary to guide the wire over a given path at an irregular rate of speed which decelerates toward the end of its travel. Although the required motion could readily be secured by the use of a cam, it was desired to obtain it from a crank used to operate another unit of the machine. The four views E, F, G, and H in Fig. 6 show how this was accomplished.

The crank A rotates in the direction indicated by the arrow, a reciprocating motion being transmitted to the slide-block C through the connecting-rod B. Gear teeth, cut in the large end of rod B, mesh with similar teeth on lever D, which is carried on block C. The wire being guided passes through a hole in the upper end of lever D.

In operation, the mechanism starts with lever D in the position indicated in view E. The angular movement of connecting-rod B causes lever D, by means of the gear teeth, to oscillate on its stud, so that the movement of the upper end of lever D is added to the movement of the block C as produced by the rotation of crank A, thus causing lever D to assume the position shown in view F. Continued rotation of crank A carries rod B to the horizontal position, causing lever D to return again to a vertical position, as indicated in view G, the movement of the upper end of lever D being deducted from the movement of block C.

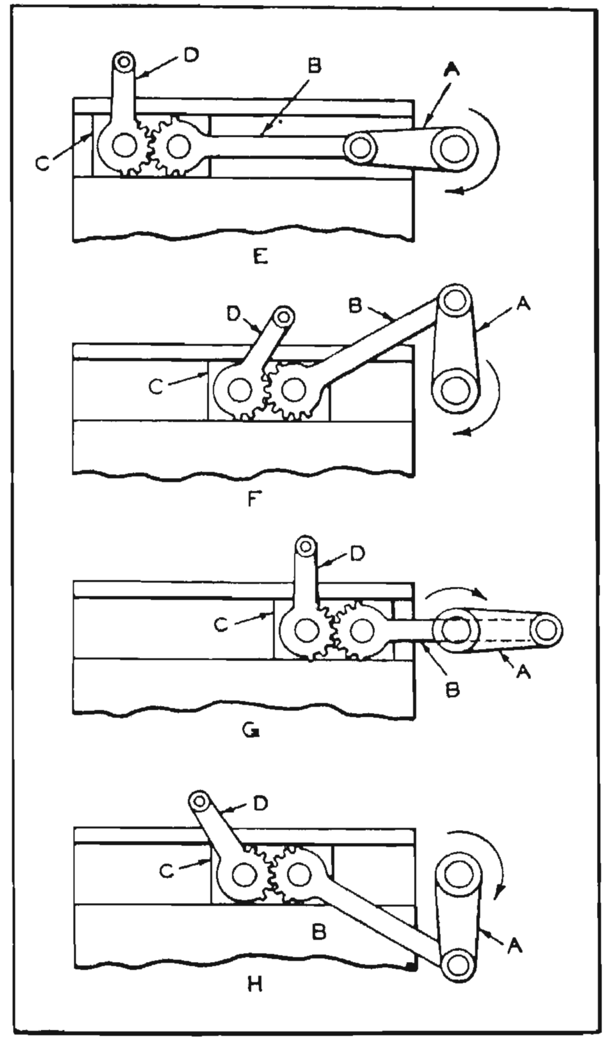


Fig. 6. Crank-operated Slide with Segment Gear Lever which is Given Irregular Reciprocating Motion Through Segment Gear on Connecting-rod.

A comparison of views E, F, and G indicates that during the first quarter revolution of crank A, the movement of the upper end of lever D is considerably greater than in the second quarter. In view H, which shows the mechanism at the end of the third quarter revolution, the condition is the same as in view F, but in the reverse direction.

Cam Drive with Variable-Stroke Mechanism.— In order to obtain a variable-lift motion from a simple cam mechanism, the length of the follower lever D was extended and equipped with an eccentric stop G, as shown in Fig. 7. With this arrangement, the stroke of lever D is lengthened or shortened as required by adjusting stop G. Baseplate H carries rotating cam B, which is in contact with follower roller C, mounted in the locking lever D. Lever D rocks or swings about pivot E and extends beyond follower roller C, terminating in a spherical end F, preferably made of fiber or laminated plastic material. End F is located opposite an adjustable eccentric G.

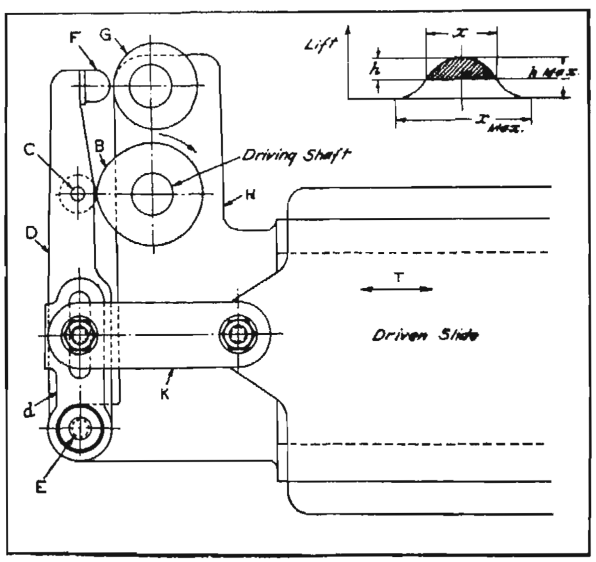


Fig. 7. Mechanism for Obtaining a Variable-lift Stroke by Adjusting Eccentric Stop G.

The extension d connected to lever D is provided with a long slot along which connecting-rod K can be adjusted and clamped. Rod K connects lever D with slide T, which represents the driven member. The position of slide T during the time in which the reciprocating motion is imparted to it by rod K can be changed by adjusting the driving pin in the slot in extension d. The length of the stroke can be changed by adjusting eccentric G.

This effect is shown diagrammatically in the upper right-hand corner of the illustration. The maximum and minimum lifts of the cam are represented by dimensions h, and the maximum and minimum angular movements during which the lift movement occurs are represented by dimensions x. By adjusting the stop, the values of these dimensions can be changed from maximum to minimum and, in extreme cases, they can be reduced to zero.

Variable-Stroke Mechanism for Toggle Lever Press.— Toggle lever presses and eccentric presses of the usual type have no provision for adjusting the stroke, and only by adjusting the length of the connecting-rod can the position of the ram at the end of the stroke be changed. With the new mechanism shown in Fig. 8, however, it is possible to make any desired adjustment of the stroke length while the press is running.

The ram I is actuated in the usual manner by a connecting-rod H attached to lever G. The length of lever H can be varied by means of a screw, so that the ram can be located at any desired position at the end of the stroke. Between the free end of lever G and the swinging lever D which has a fixed bearing in the press body, there is a short member F and a longer member C. To obtain a positive movement, point J on member C is compelled by member E to follow a circular path.

A rotating crank A drives member C by means of a connecting-rod B. The stroke adjustment is accomplished by means of the lever E. Lever E has the form of a quarter sector provided with holes that correspond with other holes in the press body. By changing the bolt K on which lever E swings to one of the other holes in the press body, the length L of the ram stroke will be changed. The position of the ram at the end of the stroke, however, remains the same.

The members D, C, and F being located in nearly the same vertical plane, as shown in the illustration, then exert the maximum pressure on the ram. In the actual design, sector E is replaced by a worm-wheel which permits regulating the position of bolt K on which lever E is pivoted. With this arrangement, the bearing or bolt K can be moved on a circle around the center of the worm-wheel which coincides with point J.

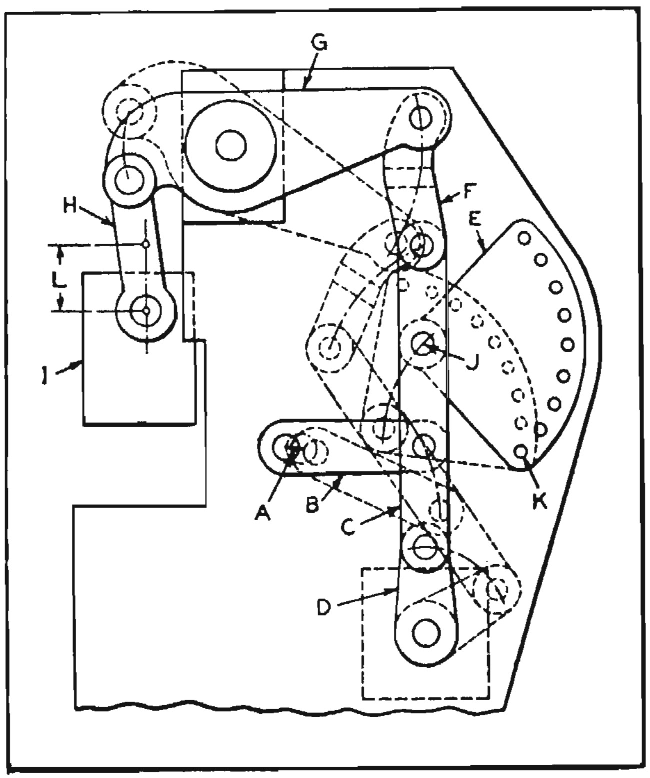


Fig. 8. Variable-stroke Mechanism for Toggle-lever Press.

The three views in Fig. 9 show the mechanism with the bolt K of member E located in three positions which give the maximum, medium, and minimum strokes. Precise adjustment of the stroke can be made before the press is started or while it is running. The crank A rotates through a complete circle. With clockwise rotation, the arc for the downward stroke is larger than for the return stroke; this results in the downward speed being somewhat reduced. Reducing the length of the stroke, in turn, reduces the time required for the downward movement.

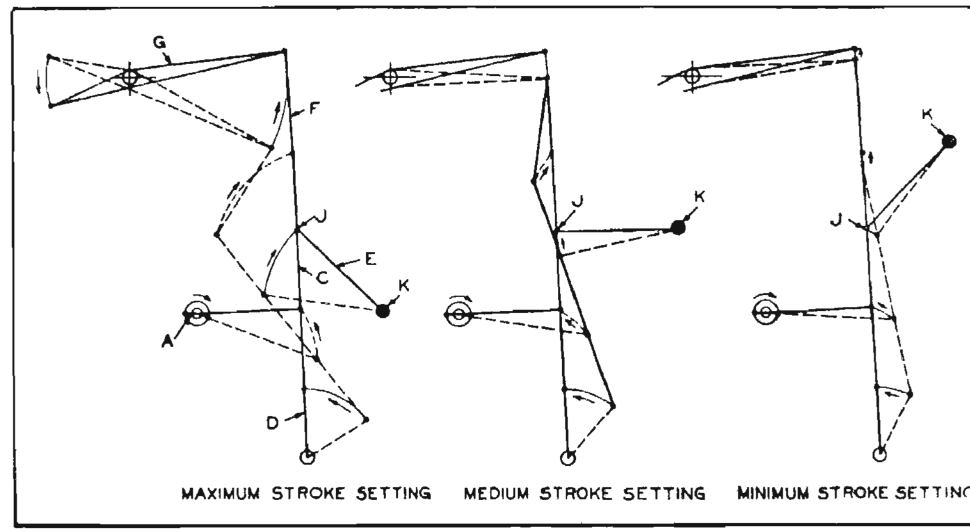


Fig. 9. Method of Adjusting Mechanism Shown in Fig. 8 for Different Length Strokes.

Variable-Stroke Mechanism with Adjustable Crankpin.—Fig. 10 shows an interesting design for a variable-stroke mechanism that was devised as part of the development of a feeding arrangement. The device consists of a crankpin wheel A which is provided with an integral shaft supported in a bearing in the plate B. A slot is accurately milled in the face of wheel A to accommodate the movable eccentric crankpin block C.

Block C is provided with rack teeth which mesh with pinion D. The pinion is an integral part of the shaft E which extends through a bore in the shaft of wheel A. A locking disk F is fastened to the shaft of wheel A by two set-

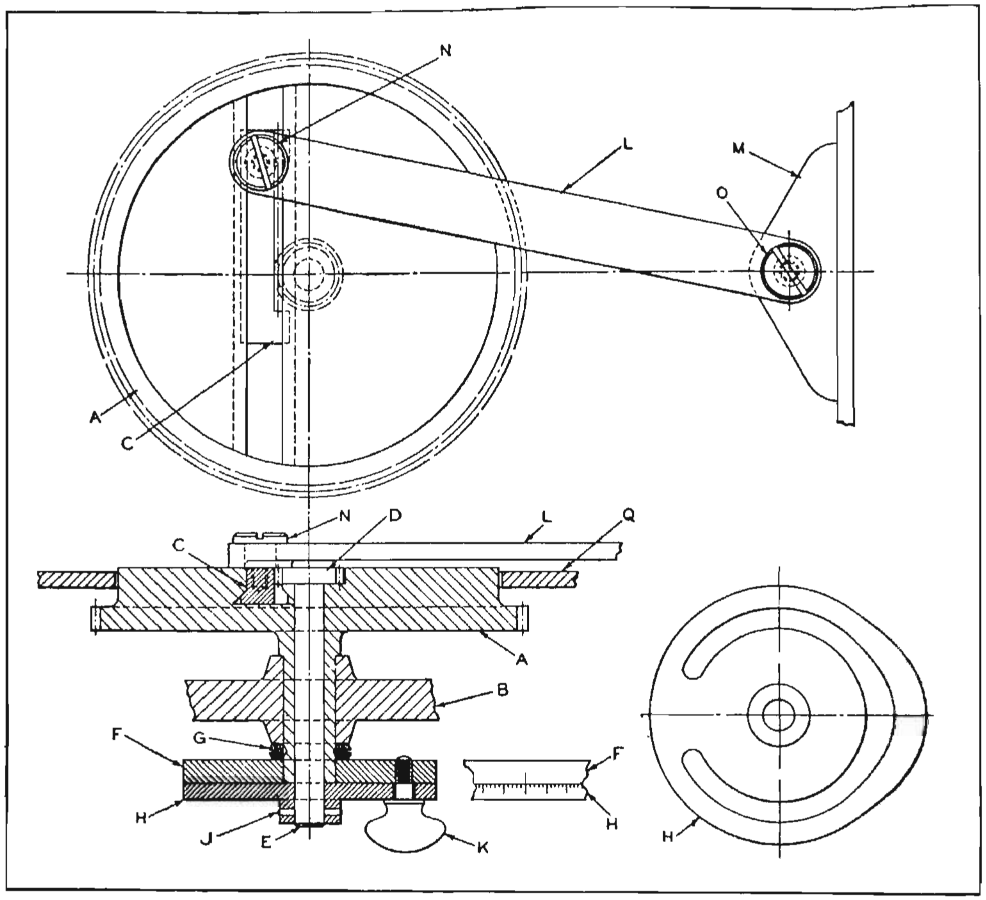


Fig. 10. Mechanism for Varying the Stroke of a Reciprocating Motion.

screws G. The adjusting disk H is fastened to shaft E by pin J. Disks F and H are locked together by screw K.

The eccentric movement is transmitted through link L to the pusher M. Studs N and O serve to connect the ends

of link L to their respective members. The wheel A is arranged in position below a table top Q.

The operation is quite simple. Power is transmitted through a system of gears, not shown, to the gear that is an integral part of wheel A. The eccentric location of stud N produces the required reciprocating motion in the pusher M through link L. To adjust the eccentricity, thumb-screw K is loosened and disk H is turned slightly in the direction required to produce the necessary eccentricity. The adjustment is transmitted through pinion D to the gear rack on the eccentric block C.

The pinion and rack are so constructed that they mesh with practically no free motion. Disks F and H are of a large diameter and are provided with graduations. The adjustment of the disks relative to each other results in a correspondingly small movement of pinion D and block C. This feature, coupled with the elimination of free motion in the linkage and gear teeth, produces a very accurate adjustment of pusher M.

Variable-Stroke Mechanism that Can be Adjusted While Operating.—The principle of mechanically shifting the crankpin to increase or decrease the length of the stroke, as employed in the variable-stroke mechanism just described, was applied to machine shop shapers sixty or more years ago. In the shaper applications, a screw was used in place of the gear and rack incorporated in the more recent design for changing the position of the crankpin.

A means of changing the stroke of a crank has been developed so that it can be employed to make adjustments while the mechanism is in operation. With this improved design, shown in Fig. 11, the stroke can be adjusted to any length within close limits from zero to the maximum length for which the mechanism is designed.

Referring to the illustration, driving gear B is keyed to driving shaft A. When gear B makes one revolution, it causes bevel gear J, which carries crankpin K, to make one revolution. Driving shaft *A* extends through the differential gear assembly to the bearing *O*, but is reduced in diameter to accommodate the small bevel gears *F*.

The bevel gear *C* is keyed to driving shaft *A*. Worm-gear *M* carries two bevel gears *D* and *E* on studs fastened within its rim. Worm-gear *M* and bevel gears *F* have a free bearing on shaft *A*. Worm *L* engages the worm-wheel, and, unless turned by means of its handwheel, remains stationary, in which case bevel gear *C* transmits its motion through bevel gears *D*, *E*, and *F* to gear *G*. Gear *G* is keyed to shaft *H*, to the upper end of which is keyed the small gear *I* that is in mesh with rack teeth cut in sliding bar *N* which carries crankpin *K*.

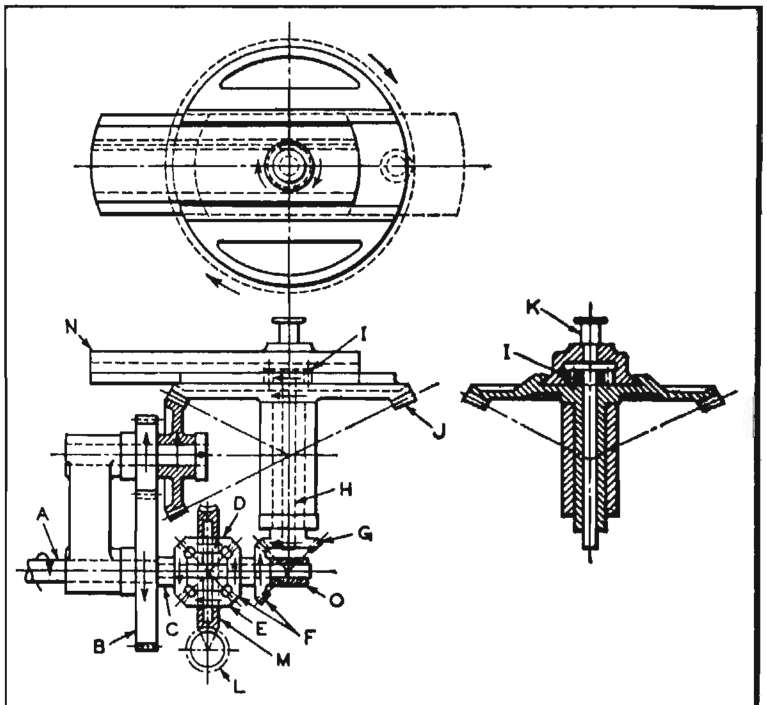


Fig. 11. Mechanism for Driving Crankpin K from Shaft A, which Enables Throw or Distance of Crankpin from Center of Its Shaft to Be Adjusted While Mechanism is in Operation.

When shaft *A* makes one revolution in the direction indicated by the arrow, gear *J* also makes one revolution. Simultaneously, the small gear *I* makes one revolution in the same direction as *J*, the result being the same as though gears *I* and *J* were fastened together. Any movement of the worm-wheel *M* by means of its handwheel, however, will cause gear *I* to revolve either slower or faster than gear *J*, depending on the direction in which the handwheel is turned. This increase or decrease in the speed of gear *I* will cause bar *N* to slide in or out, thus changing the radial position of crankpin *K* relative to the center of gear *J* and permitting the desired length of stroke to be obtained.

Exact setting for a given length of stroke is facilitated by a graduated dial on the handwheel. If the mechanism is required to operate at a fairly high speed, a specially designed double sliding bar would have to be substituted for slide *N* in order to keep the revolving unit properly balanced when adjusted to any stroke within its capacity.

Variable-Stroke Toggle-Lever Mechanism for Press.—The stroke of most presses operated by an eccentric cannot be changed, because the amount of eccentricity is fixed. However, only a slight adjustment of the position of the top bearing of a toggle-lever press is required to obtain a considerable range of adjustment in the length of the stroke. Crank *A* of the toggle-lever press mechanism shown in Fig. 12 transmits motion to toggle levers *C* and *D* by means of connecting-rod *B*.

The upper lever is located in a bearing *F* on a slide *G* which can be moved or adjusted in a horizontal direction at right angles to the direction in which the press ram moves. The adjustment of slide *G* is effected by a handwheel which operates an adjusting screw. Thus the rigidity of the press body is not impaired by the adjusting device. The positions of the press ram at the ends of its stroke can also be adjusted by means of the screw *E* and wedge *H* without changing the length of the stroke. The press-operating

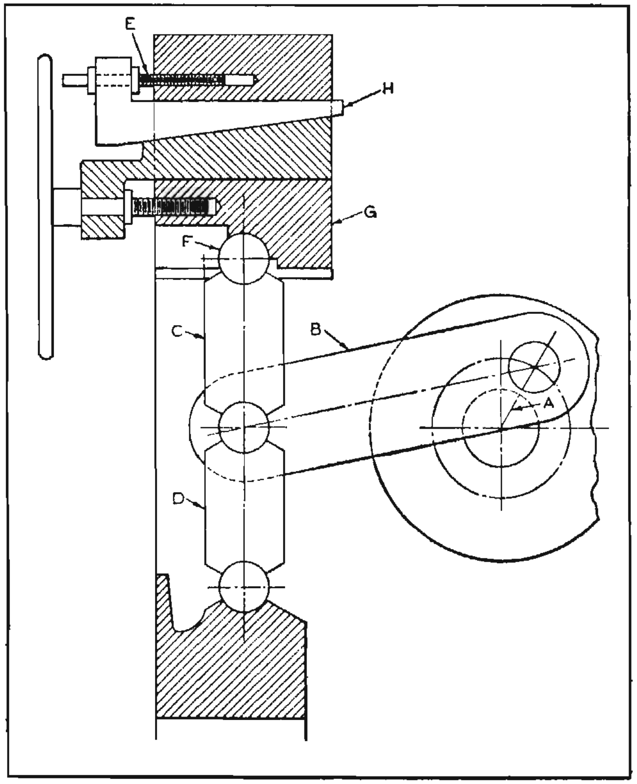


Fig. 12. Toggle-lever Mechanism Designed to Permit Adjustment of Position and Length of Press Ram Stroke.

mechanism described, which is the invention of W. Klocke, is covered by a British patent.

Variable-Stroke Cam-Actuated Mechanism.—The problem of providing a means for obtaining a variably controlled range of action from a single cam arises frequently in the design of various types of machines. The double lever mechanism, shown in Fig. 13, is simple, compact, and well adapted for solving problems of this kind. In its simplest form, it consists of a cam A and two levers B and C, pivoted on fixed centers and interlocked by a movable pivot stud D, which can be adjusted to vary the relative lengths of the active lever arms. One of the levers bears against the cam

face while the other imparts the required reciprocating movement to the rotating cylinder E.

Cylinder E is free to slide axially on the main shaft F to which it is keyed. A cylinder cam A is fastened to shaft F. The roller follower G which fits the cam groove is mounted on the bellcrank lever B which swivels about the fulcrum stud H. Roller I, which fits the groove in cylinder E, is simi-

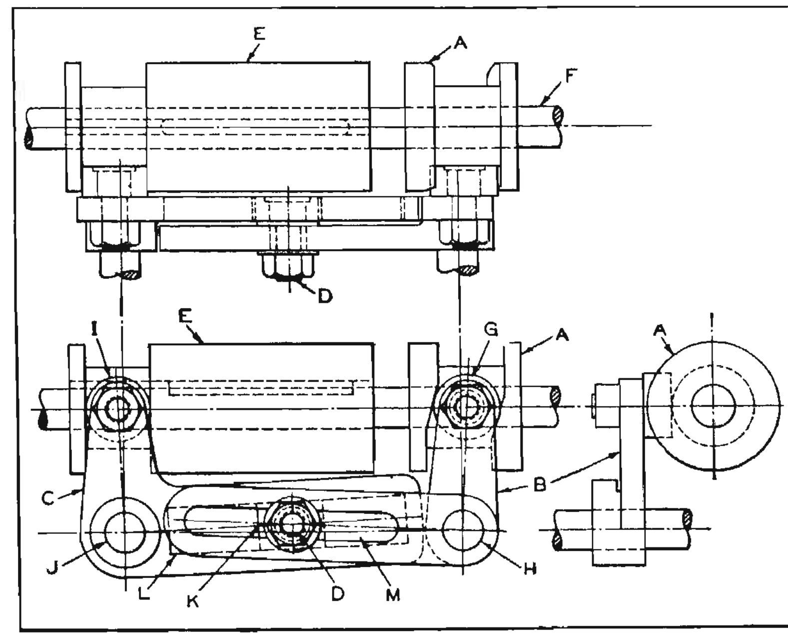


Fig. 13. Mechanism for Reciprocating Cylinder E on Shaft F.

larly mounted on bellcrank C that pivots on the fulcrum shaft J. The stud D carries a rectangular block K which slides in the wide slot L in bellcrank C. Stud D can be clamped in any desired position in slot M in bellcrank B.

When stud D is clamped midway between the two fulcrum points or studs J and H, the lever arms are of equal length, and consequently the axial movement imparted to the cylinder E is equal to the throw of the cam. To increase the movement of the cylinder, stud D is shifted toward fulcrum J. To decrease the movement of the cylinder, the stud is shifted toward fulcrum H. Not only can extremely fine variations of movement be obtained with this mechanism, but any desired range of movement can be obtained by properly proportioning the lengths of the slotted lever arms.

With this mechanism, complex movements are obtainable that might otherwise require elaborate mechanisms by making the fulcrum points adjustable, providing means for varying the position of the sliding block during the operating cycle, or substituting a cam of suitable curvature for the straight groove in cylinder E.

Mechanism for Producing Variable Reciprocating Movement.—A mechanism designed to produce a reciprocating movement of constantly varying length of stroke, constructed for use on a wire weaving machine, is shown in the accompanying illustrations. It was desired to produce an irregular pattern of no definite accuracy in the woven wire product, the main requirement being that the "repeat" patterns be identical. To produce the desired pattern, it was necessary that the length of travel of the reciprocating work-head vary throughout the repeat movement. By means of the mechanism illustrated, the work-head is given a gradual, but not uniform, increase and decrease in the length of stroke.

Referring to the left-hand diagram of Fig. 14, the drive-shaft A is carried in bearing I, attached to a stationary part H of the machine. Gear B is keyed to shaft A, and rotates in the direction indicated by the arrow. The hub on one side of bearing I is machined to support the lever J, which is free to oscillate on the hub. Lever J carries the gear C, which is free on its stud and meshes with gear B. Gear D rotates freely on the stud at the end of lever J, and meshes with gear C.

The connecting-rod F is attached to gear D by the stud L, and to the work-head G at the opposite end. The connecting-

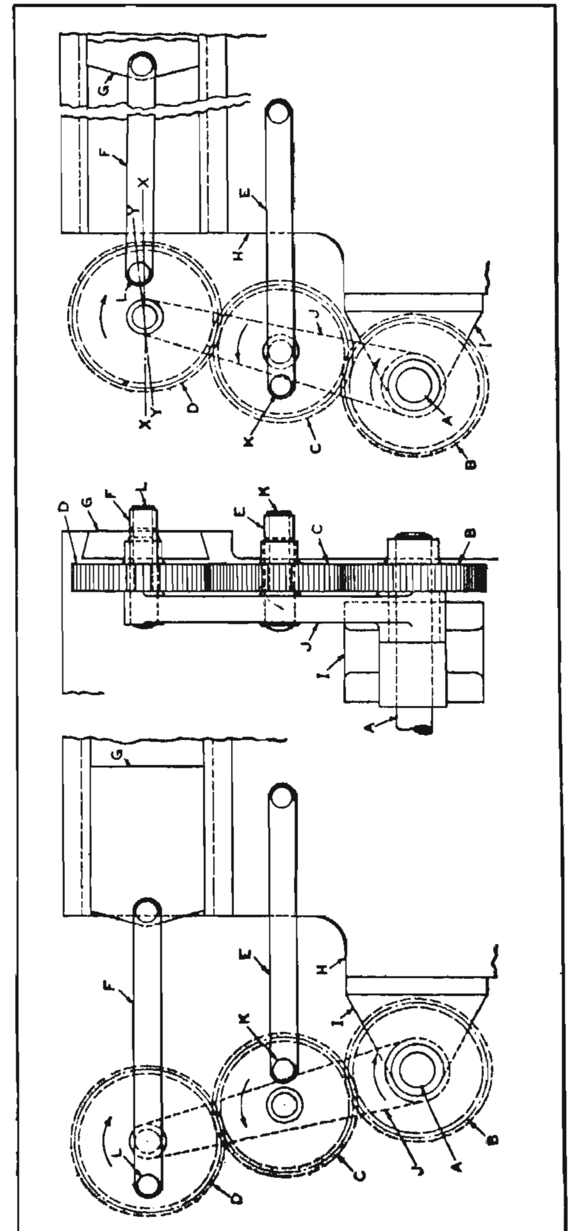


Fig. 14. (Left) Diagrammatic View of Mechanism for Producing Variable Reciprocating Movement of Slide G. (Center) End View of Mechanism. (Right) View Showing Relative Positions of Gears and Slide G after One-half Revolution of Driving Gear B.

rod E is attached to gear C by stud K at one end, and to the stationary part H at the other end. Gears B and C have the same number of teeth, while gear D has one more tooth. On the machine as constructed, gears B and C each have thirty-six teeth, while gear D has thirty-seven teeth, the effect being to maintain a constant definite relationship between gears B and C, and a constantly varying relationship between C and D throughout the cycle.

In the left-hand diagram of Fig. 14, the work-head G is shown at its extreme left-hand position. Gear B, rotating in the direction indicated by the arrow, causes gear C to rotate in the opposite direction. The rotation of gear C causes stud K to change its position so that lever J is moved or pivoted to the right, the extent of the oscillation being determined by the diameter of the circular path of stud K, as measured horizontally on the line of travel of the stud carrying gear C.

The rotation of gear C causes gear D to rotate on its stud in the direction indicated by the arrow, and as stud L rotates around the center of gear D, the radius of rotation is added to the horizontal movement of the upper end of lever J. As gears B and C have the same number of teeth, lever J passes through one cycle of oscillation to each revolution of gear B, but as gear D has one tooth more than gear C, it does not complete a full revolution to each revolution of gear B, and stud L, therefore, has not quite reached its extreme right-hand position, as is shown in right-hand diagram of Fig. 14.

This diagram shows gears B and C revolved one-half turn from the positions shown in the left-hand diagram of Fig. 14, but gear D is one-half tooth short of having completed a half turn. The line X–X drawn radially through the centers of gear D and stud L indicates the position which gear D occupied in the left-hand diagram of Fig. 14. The line Y–Y, similarly drawn, indicates the present position after the half-turn of gear B. Although the movement of stud L has been added to the movement of the upper end of lever J, the added movement is slightly, though negligibly, less than would be the case if gear D had completed a half-turn. Thus, with each rotation of gear B, there is one tooth space difference in the relative position of gear D, and a corresponding difference in the position of the work-head G at the ends of its stroke.

The left-hand diagram of Fig. 15 shows gear B after having completed four revolutions. Gear C likewise has completed four revolutions, lever J and rod E occupying the same positions as the left-hand diagram of Fig. 14. Gear D, however, is the equivalent of four tooth spaces short of having completed four revolutions, and the stud L has not reached its extreme left-hand position. The travel of the work-head G to the left, therefore, has been decreased at this point to a distance equal to the horizontal distance between the original and the present position of the stud L, represented by the dimension Z.

In the right-hand diagram of Fig. 15, lever J occupies the same position as in the right-hand diagram of Fig. 14, and the stroke of the slide or work-head G to the right has been shortened by the distance W. Thus, with each revolution of gear B, there is a decrease in the stroke of work-head G, until gear D has completed a half revolution on its stud, when the conditions are reversed and there is a gradual increase in stroke length until gear D has made a full revolution, which completes a cycle or "repeat" movement.

Mechanism for Regulating the Movement or Pressure Applied to a Rod.—The device shown in Fig. 16 is designed to provide adjustable control over the travel or pressure applied to rod A. The normal actuating movement is obtained by having the end B of plunger C rock to a position B_1, there being a spring return (not shown) for rod A which keeps it back in a position to make contact with the end B of the actuating plunger.

Plunger C is supported indirectly on a stud D that also carries a swinging block E, in which the plunger is free to

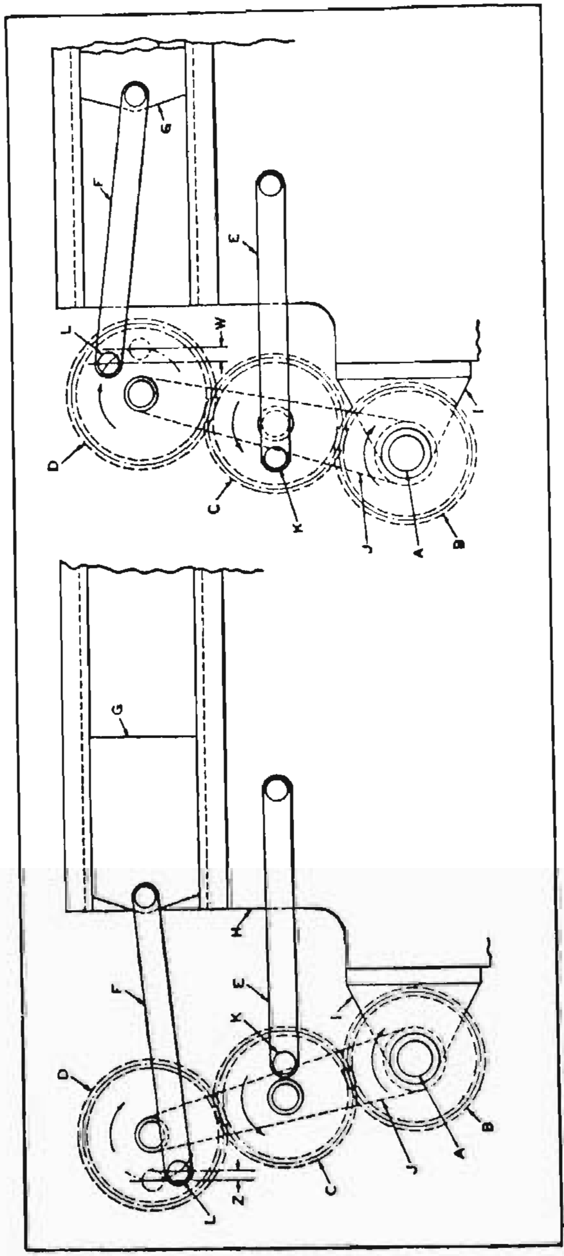


Fig. 15. (Left) Diagram Showing Relative Positions of Parts of Mechanism Shown in Fig. 14 after Gear B has Made Four Complete Revolutions. (Right) Mechanism with Lever J in the Same Position as in Right View of Fig. 14, but with Stroke of Slide G Shortened by Distance W.

slide up or down as required. The upward motion is obtained by raising the lever *F* through the medium of the link *G* operated from within the machine. Lever *F* pivots about the holding pin *H* and makes contact with a roll *J* attached to plunger *C*. There is a tension spring at *K* which holds the plunger *C* down against the lever *F* throughout its range of action.

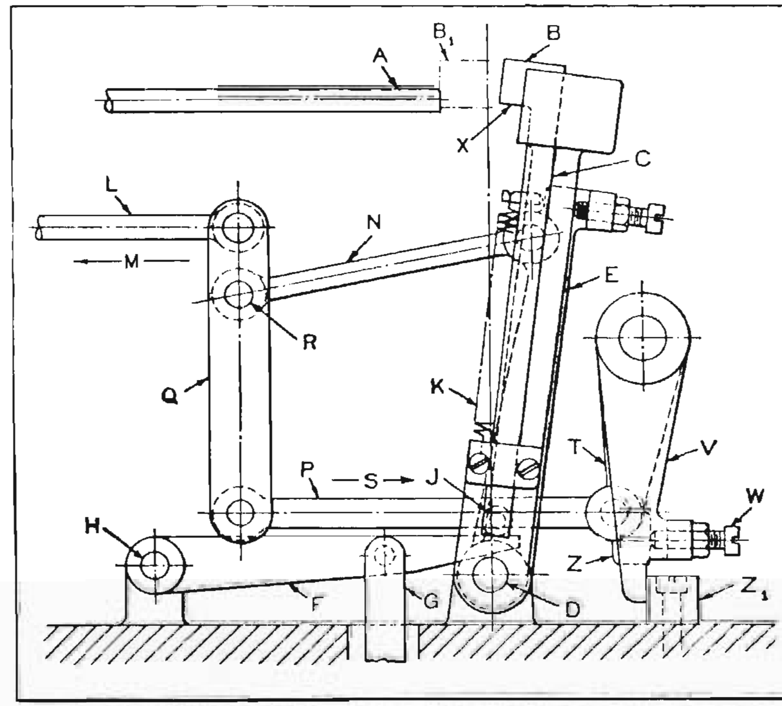


Fig. 16. Mechanism for Controlling Movement Imparted to Rod A by End B of Oscillating Plunger C.

When lever *F* raises plunger *C*, the entire swinging unit will rock back and forth without permitting the end *B* to strike rod *A*, as that portion *X* on the plunger will pass over the top of the rod. There is an ingenious arrangement for swinging the arm *E* back and forth, which is provided with means for adjustably controlling the travel from the right side of the mechanism, where the adjusting means is accessible to the operator of the machine.

Movement imparted to the connecting-rod *L* from within the machine in the direction of the arrow *M* pulls rod *N* in the same direction until the end *B* of the plunger meets resistance by coming against rod *A*, the fulcrum point for lever *Q* being about *R* as a movable center. This throws the thrust of link *P* in the direction indicated by arrow *S*, where it is carried to the lever *T*, which has an end *Z* that makes contact with an adjusting screw *W* in an adjacent lever *V*. Lever *V* makes contact with the block Z_1 attached to the machine bed. By adjusting screw *W*, the relationship between the two levers *T* and *V* is changed, thereby controlling within a limited range the action of *B* against rod *A*.

Automatic Variable-Lift Cam Mechanism.—A variable reciprocating movement is imparted to the slide of a wire-forming machine by the automatic variable-lift cam shown in Fig. 17. The requirements in designing this machine were that the slide be given four different degrees of movement during the cycle and that the timing of the movements coincide with each revolution of the driving shaft.

Referring to Fig. 17, shaft *A*, revolving in the direction indicated by the arrow, rotates the gear *B*, which is keyed to it. Gear *E* is free to rotate on shaft *A*, and carries the cam *F*, which is also free to rotate on the shaft. Gear *E* is revolved in a direction opposite to that of gear *B* through the idler gears *C* and *D*, which rotate freely on studs attached to a stationary part of the machine. The cam *J* is keyed to shaft *A*, and thus is caused to rotate with it.

Cam *J* consists of a heavy disk, which is grooved to carry the slide *H*, and a retaining plate, which is screwed to the disk. Slide *H* is shaped at its upper end to form the lobe of the operating cam. Roller *G*, which is attached to slide *H*, passes through a slot in the body of cam *J* and contacts the periphery of cam *F*. Slide *H* is slotted to permit shaft *A* to pass through. Roller *M* on slide *K* follows cam *J*.

When the mechanism is in the position shown, roller *G* is in contact with the high section of cam *F*. Gear *E* receives its rotary motion from gear *B*, reduced in the ratio of 1 to 4 by virtue of the relative pitch diameters of the gear train. Cam *F*, being attached to gear *E*, also rotates at the reduced rate of one-fourth revolution to one complete revolution of shaft *A*. As cam *F* is provided with four sections, each with a different radius, one of the four surfaces will be brought into contact with roller *G* at each revolution of shaft *A*. Thus roller *G*, being attached to slide *H*, causes slide *H* to move to one of four positions, depending on the relative position of cam *F*. The slide *K* is thereby moved a distance equal to the distance which the end of slide *H* projects beyond the periphery of the body of cam *J*. In operation, the outer end of slide *H* controls the movement of slide *K*, while the thrust of slide *K* reacts on cam *F*.

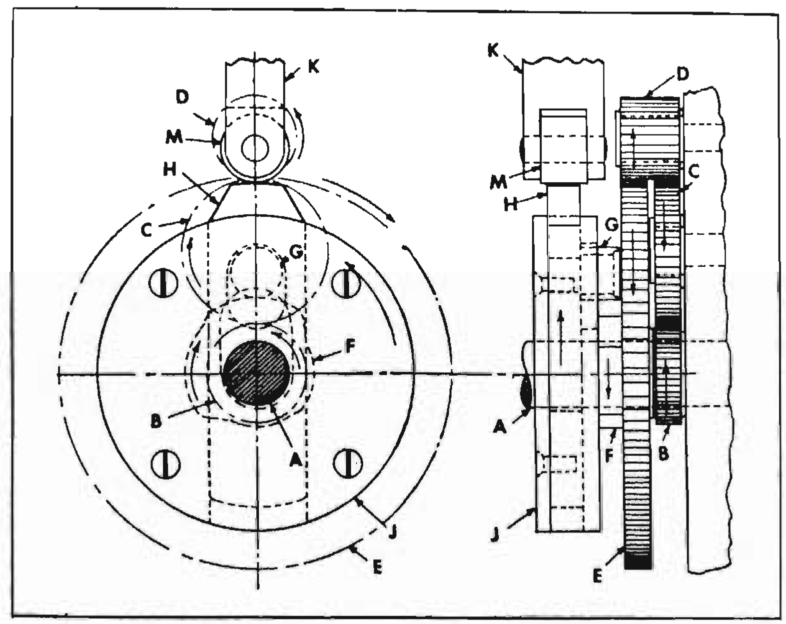


Fig. 17. Automatic Cam Mechanism which Gives a Variable-lift Movement to the Slide K.

Mechanism for Obtaining Variable Intermittent Oscillating or Reciprocating Movement.—The mechanism shown in Fig. 18 was designed to impart a rising and falling movement of magnitude, or height, *S* to roll *G* of lever *F* on a wire-forming machine at each revolution of the driving shaft *A*. (In the design shown, lever *F* is given an oscillat-

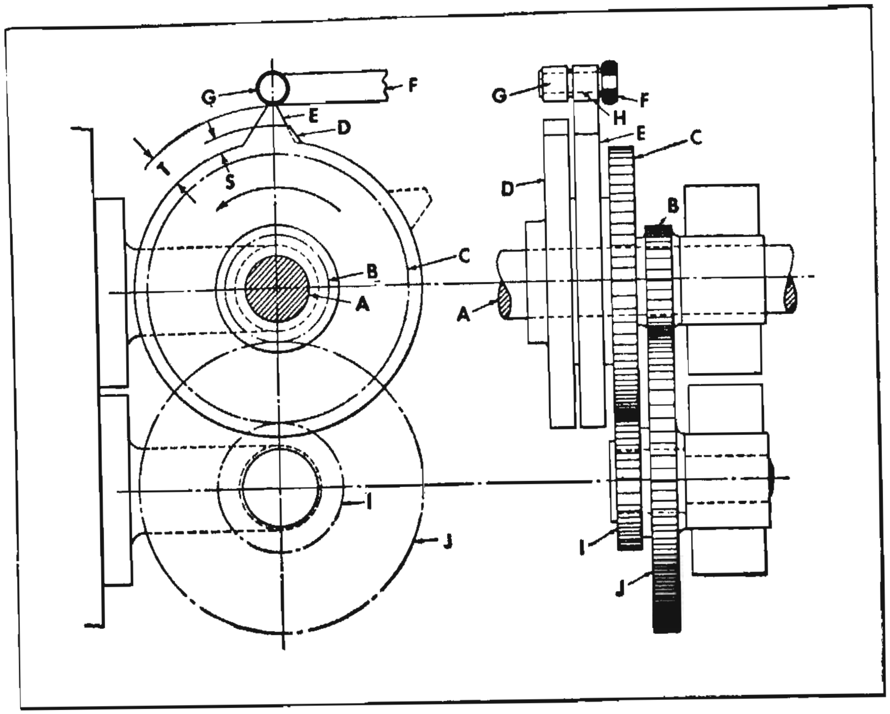


Fig. 18. Automatic Cam Mechanism which Imparts a Rise to Lever F of Magnitude T every Sixth Revolution of Shaft A and of Magnitude S at the Intermediate Revolutions.

ing movement, but if a slide were substituted for lever *F*, as in Fig. 17, a reciprocating movement would be imparted.) The mechanism is so designed that the magnitude, or height, of every sixth oscillation is approximately twice that of the five preceding oscillations, as indicated by the dimension *T*.

The required movement is accomplished in a restricted space by means of two cams *D* and *E*. The drive-shaft *A*,

rotating in the direction indicated by the arrow, carries the gear *B* and cam *D*, both of which are keyed to shaft *A*. Gear *B* meshes with gear *J*, which is keyed to gear *I*, both rotating freely on a stud. Gear *I* meshes with gear *C*, which is keyed to cam *E*. Both gear *C* and cam *E* rotate freely on shaft *A*. Lever *F* carries a stud on which the two rollers *G* and *H* rotate freely and independently of each other. Cams *D* and *E* are of the same size and outline, but it will be noted that cam *E* provides a lift *T* which is practically twice the lift *S* of cam *D*.

When the mechanism is in operation, cam *D* rotates with shaft *A*, transmitting motion to lever *F* through roller *G*. Gear *B* transmits rotative motion to cam *E* through gears *J*, *I*, and *C*, cam *E* transmitting motion to lever *F* through roller *H*. As the gear train consisting of gears *B*, *J*, *I*, and *C* is in the ratio of 6:1 with respect to the speed of shaft *A*, cam *E* acts on roller *H* once in six turns of shaft *A*.

Thus, lever *F* is given five oscillating movements by cam *D* followed by an oscillation of greater magnitude imparted by cam *E* on the sixth turn of shaft *A*. On the following fifth turn, or revolution, of shaft *A*, cam *E* arrives at the position shown by the dotted outline. On completion of the sixth rotation of shaft *A*, the position of cam *E* is that shown by the full outline, with the cam roller *G* raised to its highest position.

CHAPTER 10

Mechanisms Which Provide Oscillating Motion

In the mechanisms described, which produce an oscillating or back and forth rotary motion, there are a number of special features: one mechanism produces rotation that is adjustable for various radii of curvature and will also produce straight line motion or reverse the curvature from convex to concave; several others impart greater or less angular movement to the driven shaft than that of the driving shaft; another produces two oscillations of a shaft for each cycle of a slide; several mechanisms are described for transmitting oscillating motion from one plane to another at right angles to it. A means for providing interrupting control of oscillating movement is also described.

Centerless Oscillating Motion.—Circular motion, such as rotation or oscillation, is usually obtained by means of a guide or constraining member that utilizes as a pivot the center about which the circular motion takes place. This constraining member surrounds the pivot either wholly or partially and causes the moving object to travel in a circular manner about the center. In this case, the center must obviously be accessible.

There are instances, however, where the center does not lie within the workable confines of a given specimen in which true circular motion is desired, and it is necessary for a sliding constraining member to be used that has been specially formed to suit that particular curvature, or one very close to it. Even though such a constraining member does permit some variation in the curvature, the range of differences is comparatively small, because of difficulties in the practical application of the method.

A mechanism that produces circular motion without regard to the center of the curve, that can be adjusted for a wide range of curves, and that produces genuine rotation, though through limited arcs, is shown in Fig. 1. This mechanism is adjustable for various radii of curvature, from a curve of ordinary radius to one of infinity; it will produce straight-line motion or reverse the curvature from convex to concave.

Although the motion is confined to limited arcs, the device produces not merely revolution but rotation; that is, all points in the moving body travel in circular arcs about the same axis or the curve center C. The constraining elements are simple straight-line slides that perform their functions according to the law of geometry pertaining to the location of the vertex of an angle inscribed in a circle.

Referring to the illustration, A is the stationary base of the device, B the oscillating table, on which anything can be mounted for whatever purpose the circular motion may be desired, as for example, machining a curved surface. The particular design represented in the drawing has clarity for its purpose rather than refinement in construction or action, and is therefore somewhat diagrammatic.

For the sake of clarity, the illustration shows the mechanism in that adjustment that places the center of motion C within the confines of the illustration. The top and side views show only half of the entire mechanism, the other half being a duplicate. The following description applies only to the half shown.

From center C a circular arc of radius R may be passed through the four pivots D, E, F, and G. These four pivots will remain on this circular arc in whatever position the table B may be placed. Two of the pivots, D and F, are stationary on the base A, and the other two, E and G, are on the table B. Since the latter two follow the circular path, as will be shown, it follows that the entire table will have a circular rotational movement.

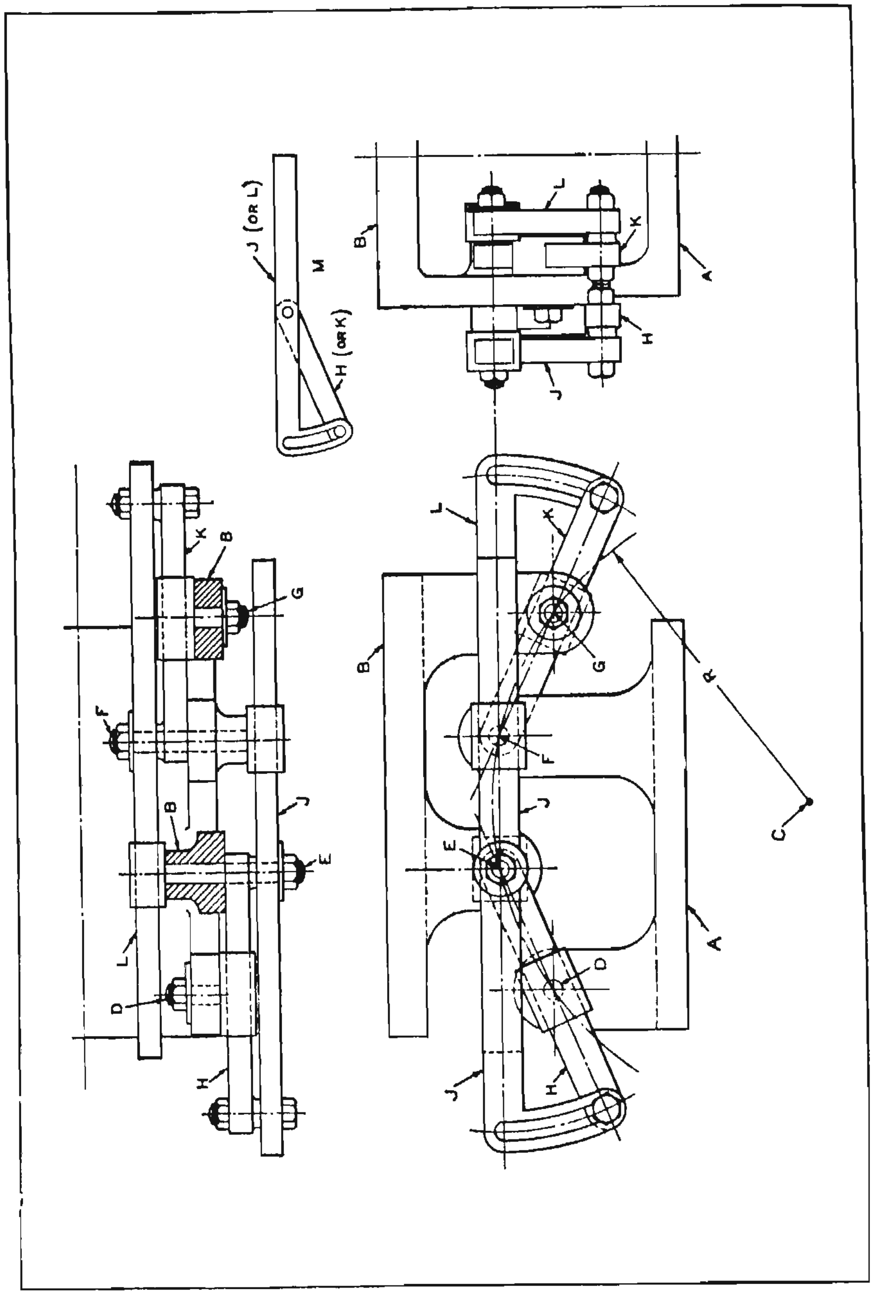


Fig. 1. Mechanism that Permits Table B to Oscillate about Center C without Requiring Any Restraining Member at the Center of Oscillation.

The table B is supported by and attached to the base A by means of two sliding bar sets consisting of bars H and J and K and L, one of which is shown separately at M. Each of these sets consists of two bars H or K and J or L, the shorter one, such as H, being pivoted to the middle of the longer one J. By means of the slotted arc, the shorter bar can be locked at any angle with the longer one within the range of the slot. One of these bar sets, H and J, is pivoted on the table at E, and the other, K and L, is pivoted on the base at F.

The bars of these bar sets slide in swivel guides, two of which are located on pivots E and F in line with but opposite to the pivots mentioned, and two more are located on pivots G and D. These guides are pivoted, respectively, in the table and base, as shown, and are free to turn, so as to assume the positions required by the bars that slide through them. The two bars H and J of the set that is pivoted at E slide in the guides on pivots D and F, respectively. The two bars K and L of the set that is pivoted at F slide in the guides on pivots G and E, respectively.

The action of the mechanism is as follows: The illustration shows the device in the mid-position of its motion. The points D, E, F, and G, are equally spaced. Therefore the angles in the two bar sets must be equal. As the table is moved either to the right or left, the vertex E of the angle in the bar set H and J follows along the circular path in accordance with the geometry of the inscribed angle and its subtended arc. Then the triangle EFG of the other bar set K and L is congruent to the triangle DEF of the first set, and since E and F already lie on the circle, it follows that G also lies on the circle by the same law of geometry. From the fact that the two points E and G of the table move in a circular path around the center C, it follows that the entire table rotates about this center.

The location of the center C and the corresponding radius of curvature depend on the angular adjustment of the bars in the slide bar sets, both of which must be alike—the smaller the acute angle the greater the radius. When the bars are in a straight line, the radius is infinite and the motion is in a straight line. If the angle is changed still further, it becomes negative and the curve of motion is inverted; that is, the curve becomes concave toward the upper side instead of convex. The same concave effect is obtained, however, by inverting the whole device.

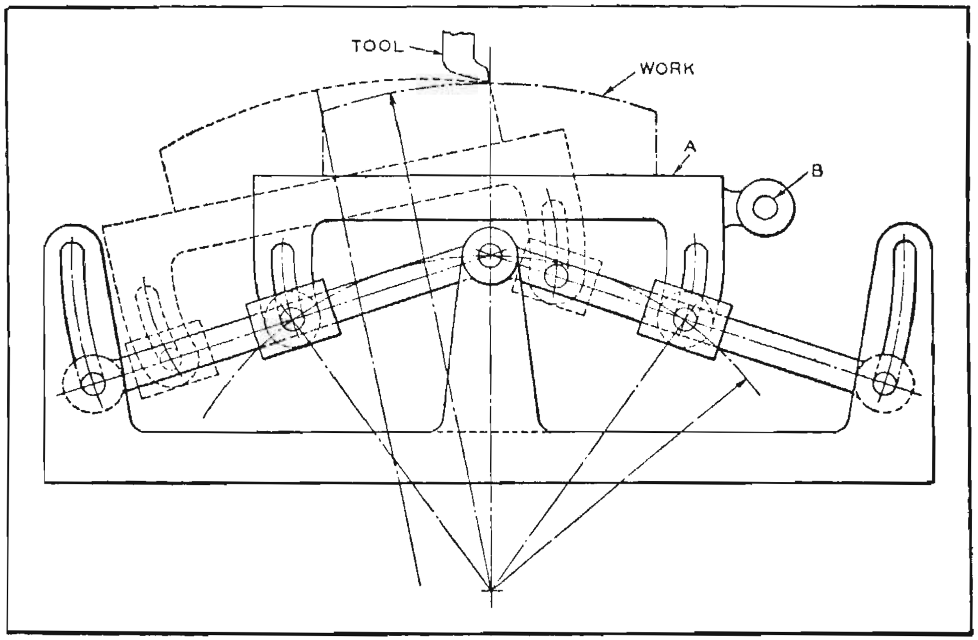


Fig. 2. Mechanism Designed for Use in Machining Convex Surfaces of Large Radii.

Mechanism for Use in Planing Convex Surfaces.—The mechanism shown diagrammatically in Fig. 2 is designed for use in planing convex surfaces that conform to parts or arcs of true circles having centers that may be located at any distance from the point of the tool. This device is similar to the one just described, although it is much simpler to construct. The diagram shows only one of the two sides

of the mechanism, the opposite side being similar to the one shown. The connecting-rod for oscillating the work-table A is attached at B.

As the motion of the work is not concentric with the convex surface being machined, a somewhat greater heel clearance is required for the tool than for ordinary turning, especially when work of the smallest diameter accommodated by the device is being machined. This clearance can be reduced as the work diameter is increased.

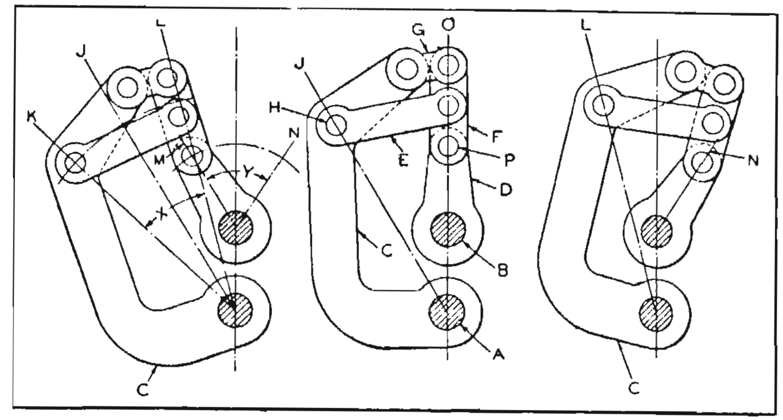


Fig. 3. Mechanism for Transmitting an Oscillating Movement from Shaft A to Shaft B, the Latter Shaft being Oscillated through a Larger Angle than Shaft A.

Crank and Link Mechanisms for Increasing Angular Movement of Shaft.—In Fig. 3 is shown a simple mechanism designed to transmit an oscillating movement from shaft A to shaft B. The mechanism is required to give the driven shaft a larger angular movement than that of the driving shaft. A mechanism of this type for multiplying the angular movement of a lever was developed by H. Lindars. It consists simply of the driving lever or crank C, the driven crank D, links E, F, and G, together with the five pins that connect the cranks and links. The cranks C and D are securely fastened to their respective shafts A and B and oscillate with them.

The view at the left shows the mechanism with the driving shaft *A* and its lever *C* rotated to their extreme left-hand or counter-clockwise positions, while the view at the right shows the mechanism with the driving shaft and its lever rotated to their extreme right-hand or clockwise positions. The center view shows the crank *D* and driven shaft *B* in the vertical position, with pin *P* on the vertical center line *O* and pin *H* of the driving crank *C* located on the radial line *J*, which is also indicated as *J* in the view to the left.

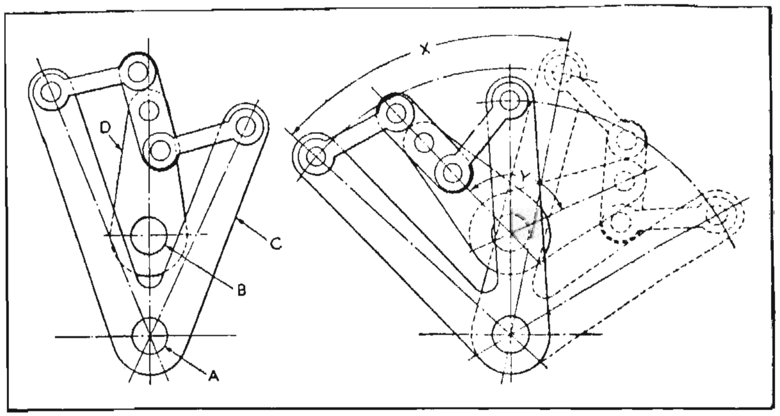


Fig. 4. Mechanism Designed to Transmit an Oscillating Motion from Shaft A to Shaft B with an Increase in the Angular Motion Imparted to Shaft B.

When shaft *A* rotates to the left so that pin *H* is in its extreme left-hand position on radial line *K*, crank *D* will also have been rotated to the left so that pin *P* is located on the radial line *M*. Similarly, when shaft *A* rotates to the right so that pin *H* is in its extreme right-hand position on line *L*, as indicated in the view to the right, crank *D* will have been rotated to the right so that pin *P* is located on radial line *N*. Thus, an oscillating movement of driving shaft *A* and its crank *C* through angle *X* serves to oscillate driven shaft *B* and its crank *D* through angle *Y*, which, as shown in the illustration, is larger than angle *X*.

It should be noted that, while crank *C* makes an angular

movement to the left of the central position shown in the center view equal to its angular movement to the right, the angular movement of crank *D* to the right from the central position is slightly greater than its angular movement to the left.

In the view to the left, Fig. 4, is shown another mechanism that is also designed to transmit an oscillating motion from one shaft to another with an increase in the angular movement imparted to the driven shaft. In this mechanism, which is patented in England under patent No. 465052, the oscillation of shaft *A*, and member *C* keyed to it, through angle *X*, as indicated in the view to the right, causes driven shaft *B* and its crank D to be oscillated through angle *Y*. In the view to the right, the mechanism is shown in full lines with its driving and driven shafts in the positions they occupy when they have reached the end of their rotating or oscillating movement in the counter-clockwise direc-

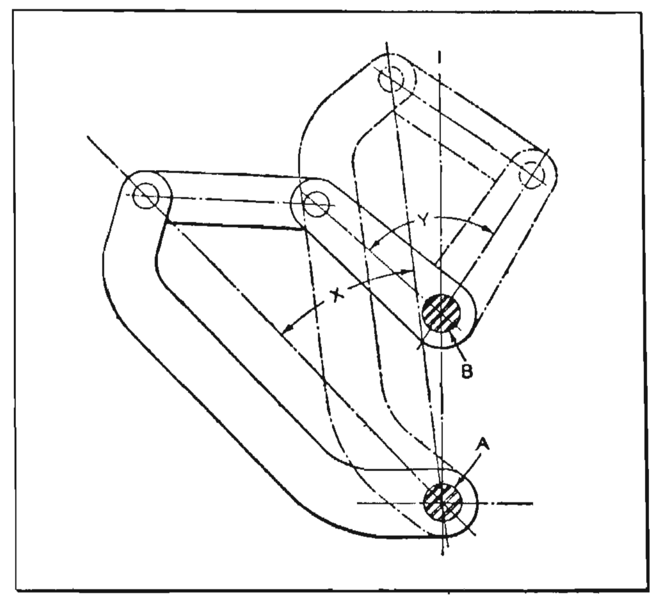


Fig. 5. Mechanism for Increasing Angular Movement of Shaft B as Compared with Shaft A.

tion. The dotted lines show the mechanism with its members in their positions at the extreme end of the clockwise movement.

Link Mechanism for Increasing Angular Movement of Shaft.—In Fig. 5 is shown an arrangement of links and levers for transmitting an oscillating movement from shaft *A* to shaft *B*. The mechanism is designed to oscillate shaft *B* through angle *Y*, which is greater than angle *X* through which driving shaft *A* oscillates. This mechanism is intended to accomplish the same purpose as the one just described.

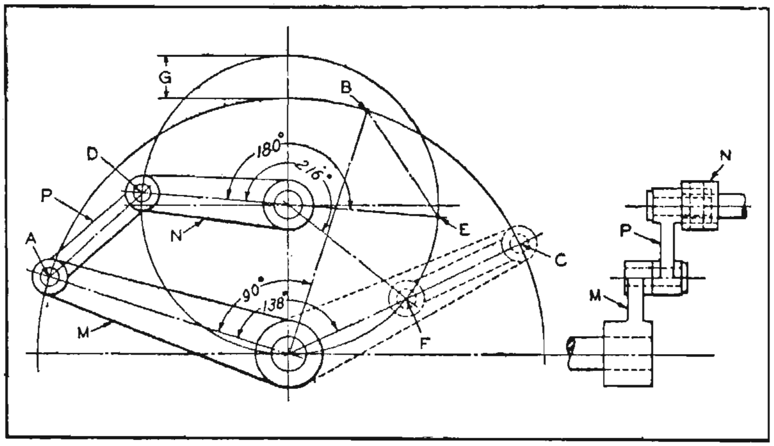


Fig. 6. Crank-and-Link Mechanism for Increasing Angular Movement of Driven Shaft.

Mechanism for Increasing Angular Movement of Shaft.—A crank-and-link type mechanism that produces a somewhat greater angular movement than either of the two designs just described is shown in Fig. 6. When the driving crank *M* travels from point *A* to *B*, through an angle of 90 degrees, the driven crank is rotated through an angle of 180 degrees. This represents a ratio of 1 to 2 in the increase in angular movement. When crank *M* oscillates 138 degrees, between points *A* and *C*, the driven crank will oscil-

late 216 degrees, between points *D* and *F*, at a ratio of 1 to 1.56. For this ratio *G* equals one-sixth the length of crank *M*, and the length of link *P* equals one-half the length of *M*.

In Fig. 7 is shown an arrangement for oscillating a crank that has too great an angular motion for practical operation by means of the connecting-rod of a revolving crank. In this case, crank *D* must have an oscillating movement of 144 degrees or more. This is accomplished by the introduction of an auxiliary crank *C*, which obtains its angular motion of 72 degrees from connecting-rod *E* driven by a revolving crank (not shown).

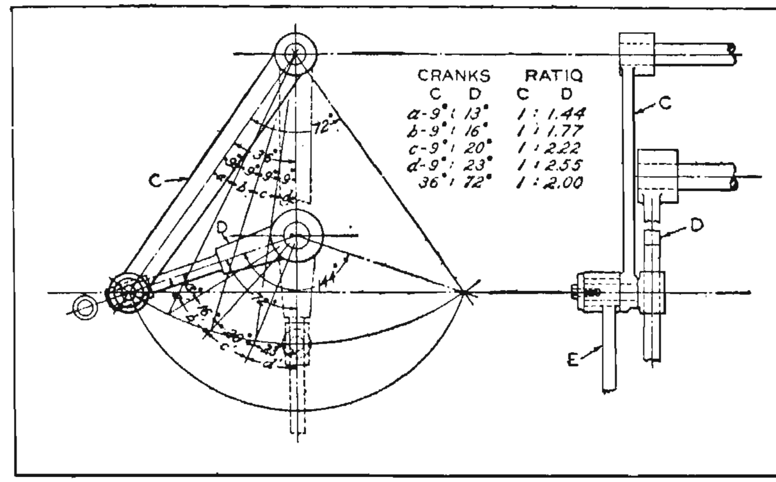


Fig. 7. Arrangement for Oscillating a Crank that has Too Great an Angular Motion for Practical Operation by Means of Connecting-rod of Revolving Crank.

The connecting-rod of this mechanism could, of course, be connected at any point on crank *C*; any position other than the one shown, however, would necessitate a change in the length of the stroke, although the 72-degree oscillating motion of crank *C* would remain unchanged. The motion of crank *D* is not uniform, but varies from a ratio of about 1 to 1.44 up to 1 to 2.55, a 1 to 2 ratio of oscillation being obtained for the half or complete angular motion. The complete angular motion of crank C is 72 degrees, and that of crank D, 144 degrees, giving a ratio of 1 to 2. The like angles a, b, c, and d, representing movements of crank C, produce unlike angular movements of crank D, as shown at a^1, b^1, c^1 and d^1, at ratios indicated in tabular form.

Mechanism for Reducing Oscillating Motion.—On a wire-forming machine, a part of the mechanism is operated by an oscillating shaft which receives its motion, through a system of links, from a crank. Owing to a change in the design of the product, the degree of oscillation of the shaft was required to be reduced. However, as other parts of the mechanism receive their motion from the same source, it was not permissible to change the throw of the crank, nor to change the positions of the connecting levers. Fig. 8 shows the design of a mechanism that successfully met the requirements referred to.

The shaft A, which is required to oscillate, is carried in the bearing B. Lever C is given an oscillating motion by

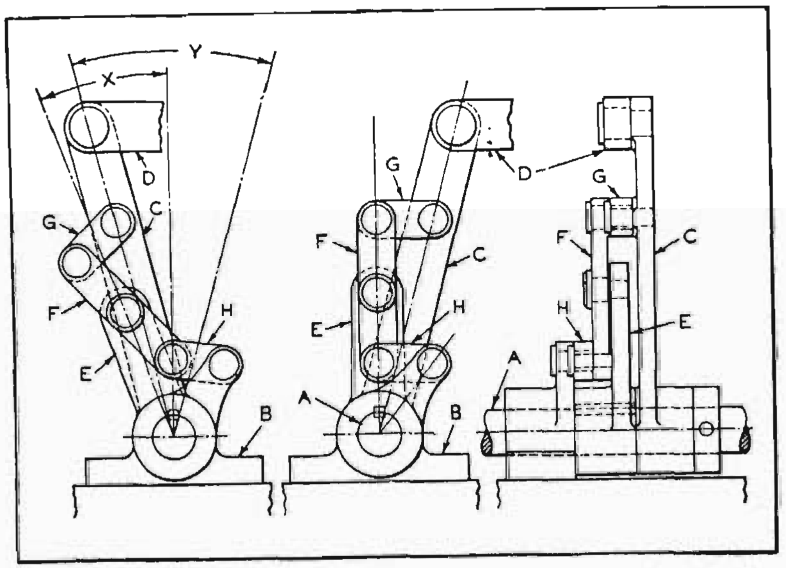


Fig. 8. Mechanism by Means of which Oscillation of Lever C through Angle Y Produces Reduced Oscillation of Lever E through Angle X.

the rod D. Lever C and rod D are of the same dimensions and occupy the same positions as before the change, except that originally, lever C was keyed to shaft A, whereas with the new arrangement it oscillates freely on shaft A. Lever E, which is keyed to shaft A, carries the lever F, which is pivoted at its center. Bearing B carries a fixed arm which is connected to the lower end of lever F by the link H. The upper end of lever F is connected to lever C by the link G. At the left and center of Fig. 8 are end views of the assembly, and at the right is a side view.

In the center diagram of Fig. 8, rod D is shown at its extreme right-hand position. By tracing through the linkage, it will be noted that any motion of rod D is transmitted through lever C to lever F through link G. Lever F transmits the motion further through lever E to shaft A.

In the left-hand diagram, rod D is shown at its extreme left-hand position. It will be noted that, as the fulcrum of lever F is at its lower end, it is acting as a second-class lever, and any motion given to the upper end of lever F, which is the power arm, will be transmitted to lever E in reduced proportion. In the left-hand diagram, the angle Y indicates the movement of lever C, and the angle X indicates the movement of lever E, the difference between the two indicating the reduction in angle of oscillation accomplished through the linkage.

Mechanism that Doubles the Oscillations Imparted to a Shaft by a Slide.—The fingers of a bag-closing machine are controlled by an oscillating shaft having a variable motion. The motion is imparted to the oscillating shaft from a feed-slide which reciprocates at a constant velocity. In Fig. 9 is shown the mechanism for transmitting the variable oscillating motion to the shaft D from slide A. During each cycle of slide A, shaft D passes through two oscillating cycles. For the first cycle, assuming that point R of link E is at P and that slide A is traveling downward, shaft D will oscillate rapidly through an angle of 190 degrees and return, at the same velocity, to its starting position. In the next cycle, the direction of rotation of shaft D is reversed, after which it oscillates slowly through an angle of 60 de-

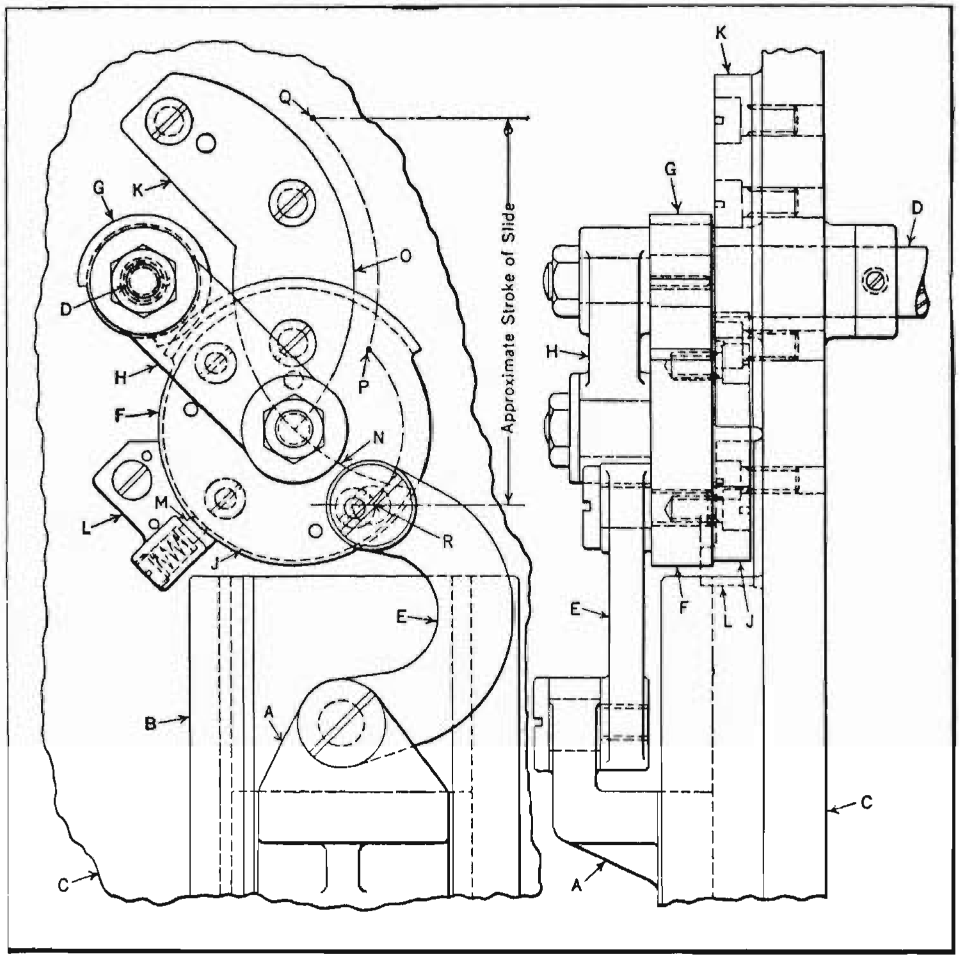


Fig. 9. Mechanism for Doubling and Varying Oscillations Imparted to Shaft D by the Reciprocating Slide A.

grees and then returns at the same velocity to its starting position. These oscillations are repeated alternately with no intermediate dwells.

The feed-slide A operates in the guides B, cast integral with the machine frame C. This slide transmits the required motion to shaft D through the link E, segment gears F and G, and the link H. Link E connects the slide with gear F and is pivoted to these members by shoulder-screws. Gear F is free to turn on a stud secured in link H, and meshes with the gear G, keyed to the driven shaft D. Link H is free to turn at the upper end on the driven shaft D.

On the under side of gear F is fastened a crescent-shaped plate J, which serves to guide this gear in its proper path along the contour of cam K, secured to the machine frame. Block L, containing the spring-actuated pin M, merely acts as a bumper to limit the downward movement of gear F and to absorb the shock of reversal at the low point.

In the position indicated, the slide is at the bottom of its stroke. As it moves upward, the upper screw in link E will be moved to point P. At the same time, gear F will be rotated through an angle of approximately 90 degrees; the lower point of cam K being in contact with the plate J holds link H stationary. This partial rotation of gear F causes gear G and shaft D to rotate 190 degrees in a clockwise direction. At the end of the 90-degree movement of gear F, the curved surface N on plate J coincides with the surface O on cam K, so that further rotation of gear F relative to link H is prevented. However, as surface O is concentric with the center of the upper end of link H and of gear G, continued upward movement of the slide to the end of its stroke will cause both gears and link H to swing upward as one member until the upper pivot screw in link E is at point Q. At the beginning of the latter movement, gear G and shaft D will have reversed their direction of rotation and their angular velocity will be reduced.

As the slide reverses its movement and returns, the slow angular movement of shaft D is also reversed. Shaft D then rotates slowly until the cylindrical part of gear F engages block L. At this point, link H is held stationary and gear F

rotates about the lower point of cam K. In doing so, gear F rotates gear G rapidly in a counter-clockwise direction, causing shaft D to reverse its movement and continue to rotate, but at a rapid rate, until the slide has reached the bottom of its stroke. Incidentally, a slight clearance should be provided between the adjacent surfaces of members K and J at this point in the slide cycle.

Obviously, the velocity of shaft D when the slide is approaching and leaving the bottom of its stroke is the same; and likewise, the velocity of the shaft when the slide is approaching and leaving the top of its stroke is also the same. Consequently, the alternate variation in the point of reversal of shaft D occurs at both ends of the stroke of the slide. By modifying the gear diameters and the stroke of the slide, various angular movements and velocities of shaft D can be obtained.

Lever Mechanism for Transmitting Oscillating Motion Simplified by Means of Universal Joints.—Transmitting a motion or movement developed at some point within a machine to some remote point for the operation of a secondary unit frequently introduces a perplexing problem. Often a combination of levers must be used for this purpose. In some cases, the problem can be simplified by equipping the connecting links or rods with universal joints, as here illustrated.

The mechanism shown, Fig. 10, is designed to transmit to lever X the oscillating motion imparted to lever J by cam C. Cam C, on driving shaft B, imparts a reciprocating motion to yoke E through roll D in the cam groove. Yoke E slides on block F located on shaft B.

It will be noted that both cam C and yoke E are located in a vertical plane and that lever J is in a horizontal plane, while lever X is in a vertical plane. Lever S, which is connected to reciprocating lever J by rod P, is also in a vertical plane. Thus, the problem of transmitting an oscillating motion to lever X becomes one of transmitting motion from

a horizontal to a vertical plane. Obviously, this problem is greatly simplified by using universal joints to connect rod P with levers J and S, as indicated by the dot-and-dash lines Z and Y.

Lever S is pinned to shaft T and transmits the required motion to lever X, which is also keyed to shaft T. Lever X, in turn, operates a slide mechanism which is connected to swivel-block W.

Referring to the construction of the mechanism, a portion of the machine frame is shown at A in both views. Bearing brackets on this frame support shaft B on which

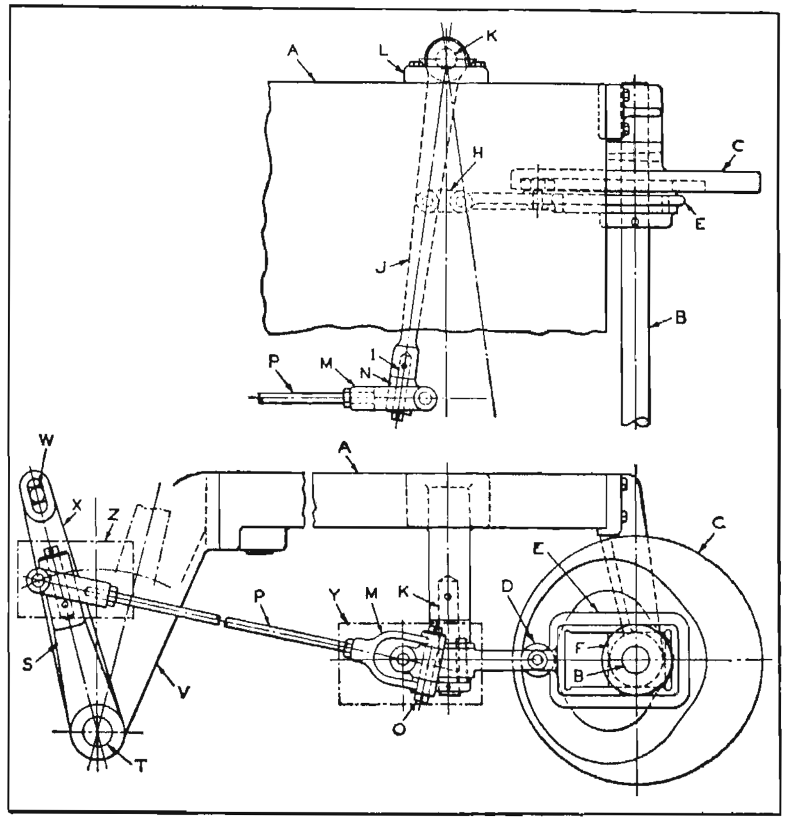


Fig. 10. Mechanism in which Universal Joints are Used to Simplify Transmission of Movement from Horizontal to Vertical Plane.

cam C is mounted. Yoke E is connected by pins to lever J through the medium of swivel-link H. Lever J is free to swivel about stud K as a center, stud K being carried in support L attached to frame A. In the end of lever J is a pin I on which block N is free to swivel. A pin O connects yoke M to block N. This construction provides a universal

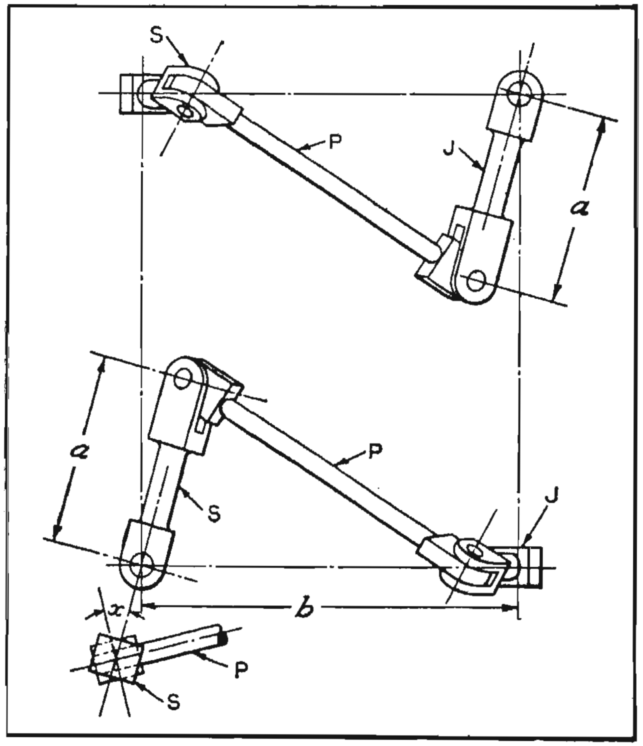


Fig. 11. Simple Mechanism for Converting Horizontal to Vertical Oscillation.

joint which permits yoke M to swivel in any direction. The two trunnion bearings of the two universal joints at the ends of rod P are located at right angles. Two brackets V support bearings for shaft T. With this arrangement, the tendency for the levers and connecting-rods to become cramped or jammed is eliminated.

Converting Horizontal to Vertical Oscillation by Simple Mechanism.—The linkage mechanism just described must have an appreciable amount of play in the pivot joints in order to operate successfully. If a few calculations are made, on the supposition that levers J and S are rigid bars, it will be found that the connecting rod P would be strained in torsion during the oscillating motion. In any practical mechanism, made as described and illustrated in Fig. 10, all distortion for which relief is not provided by the play allowed by the pivots will be taken up by the various parts of the mechanism.

However, the desired conversion from oscillation in the horizontal plane to oscillation in the vertical plane can be accomplished by a linkage mechanism of fewer parts which was devised by G. T. Bennett and described in *Engineering* in 1903. If applied to the mechanism in question, as shown in Fig. 11, it would require levers J and S to be of equal length, and the length of connecting-rod P to be the same as the distance between the axes of rotation of levers J and S. Then each of the universal joints shown would be replaced by a simple pivot joint. The angle between the pivots in rod P would be a right angle, while the sine of the angle x between the pivots in lever J would be equal to the ratio of the length of lever J to the length of rod P, or sine $x = a \div b$. Lever S would, of course, have the same angle x between its pivots as lever J.

The Bennett linkage is not an approximation; it is a mathematically correct method of converting an oscillating motion in one plane into an oscillating motion in another plane. There is no limit to the amount of oscillation; the cranks or levers may execute complete rotations or even continuous rotation if the links are designed to clear each other.

Space Linkage Mechanism for Transmitting Oscillating Crank Motion.—There are cases where obstructions or interferences may make it impossible to join the two links by a single connecting-rod, such as the mechanism just described. The interference can sometimes be avoided by the use of a connecting-rod made in two pieces which are simply

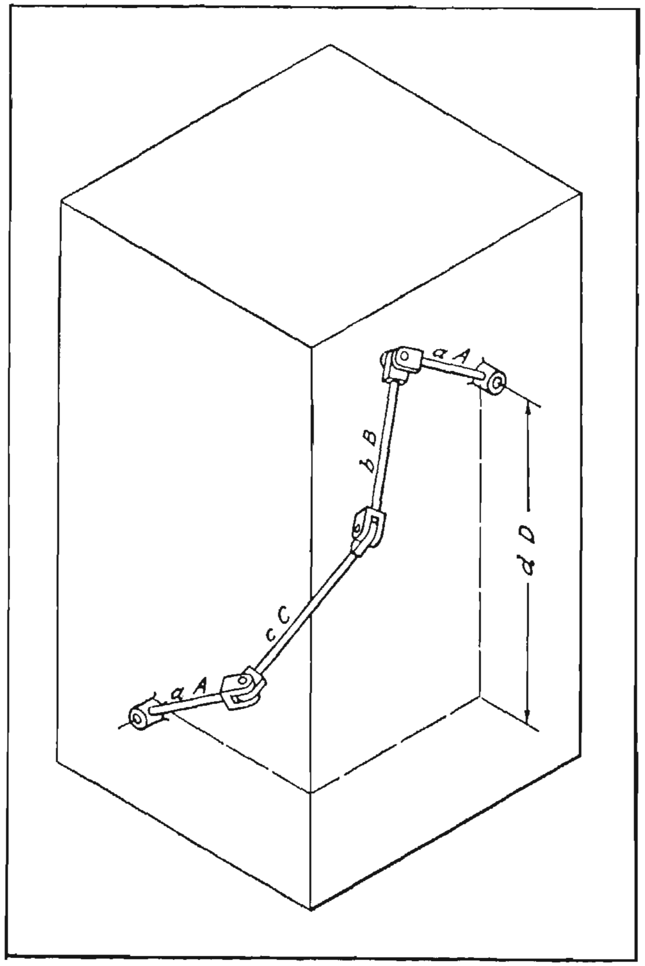


Fig. 12. Space Linkage Mechanism for Transmitting Oscillating Motion.

hinged together. The main frame, the two cranks, and the two parts of the connecting-rod, as shown in Fig. 12, constitute a hinged movable five-bar linkage in space.

A five-bar linkage of this kind must be designed to have certain mathematical relationships between its various members in order to operate. Let the length of a link be defined as the length of the common perpendicular between the hinges in the link. Let the twist of a link be defined as the angle between the hinges in the link. In the diagram and in the formulas below, the small letters refer to the lengths of the links, while the corresponding capital letters refer to the twists of the links.

The five-bar linkage will operate if the following conditions are observed: (1) The two cranks have equal lengths a; (2) the two cranks have equal twists A; (3) the sum of the lengths of the two parts of the connecting-rod is equal to the length of the main frame, $b + c = d$; (4) the sum of the twists of the two parts of the connecting-rod is equal to the twist of the main frame, $B + C = D$;

$$(5) \quad \frac{a}{\sin A} = \frac{b}{\sin B} = \frac{c}{\sin C}$$

To simplify the diagram, the cranks are shown to be oscillating in two perpendicular planes. The twist D of the main frame or link is, therefore, a right angle. Angle D need not, however, be restricted to a right angle; any other angle between the planes can be handled just as readily.

Oscillating Motion Transmitted from Vertical to Horizontal Plane.—The oscillating motion of a part in a vertical plane is transmitted to another part in a horizontal plane by means of the simple mechanism illustrated in Figs. 13 to 15. The magnitude of the oscillations is adjustable to suit various requirements.

A front elevation and plan view of the mechanism in the central or middle position of its cycle are illustrated in Fig. 13. Lever G oscillates in a horizontal plane about pivot rod H. Rod F, the position of which is adjustable in the slot of lever G, is fastened to the lever and operates with it. Part E is free to slide vertically on rod F and horizon-

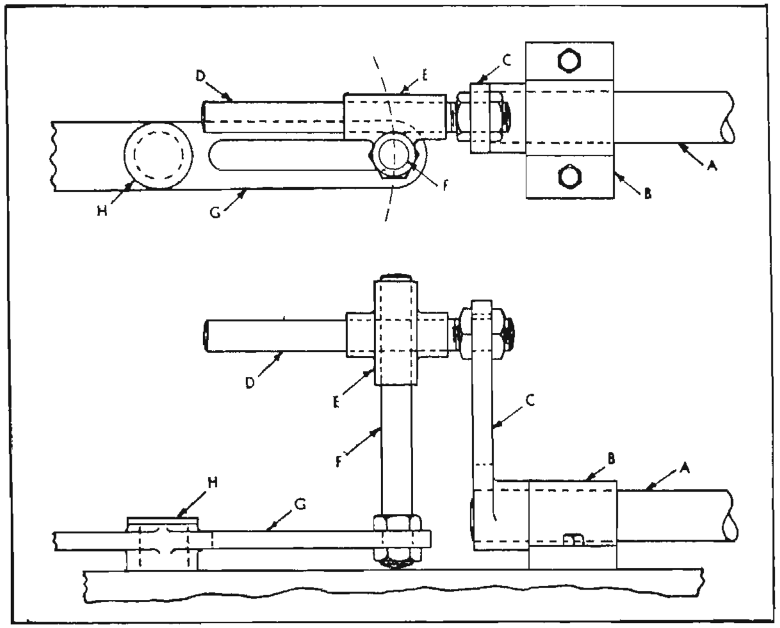


Fig. 13. The Oscillation of Lever G in a Horizontal Plane is Transmitted to Lever C which Oscillates in a Vertical Plane.

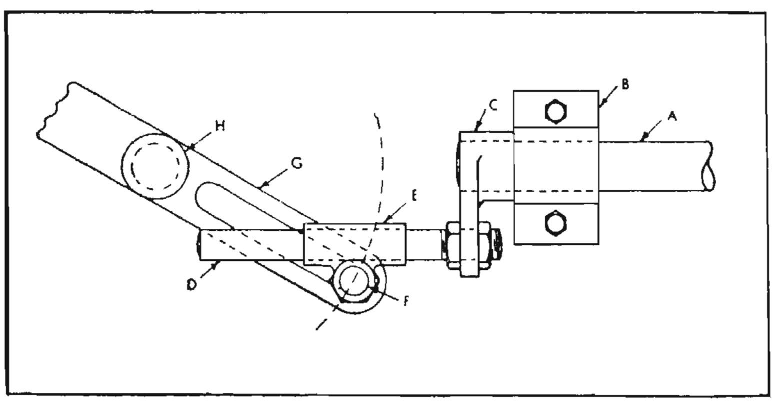


Fig. 14. Mechanism Illustrated in Fig. 13 Shown in Extreme Forward Position.

tally on rod *D*. Rod *D* is fastened to lever *C* and can be adjusted by the slot in the lever. Lever *C* is keyed to shaft *A*, which confines the movement of the lever to an oscillating motion in a vertical plane. Shaft *A* turns in bearing *B*. Thus, the oscillating motion of the lever *G* in a horizontal plane is transmitted to the lever *C*, which is made to oscillate in a vertical plane.

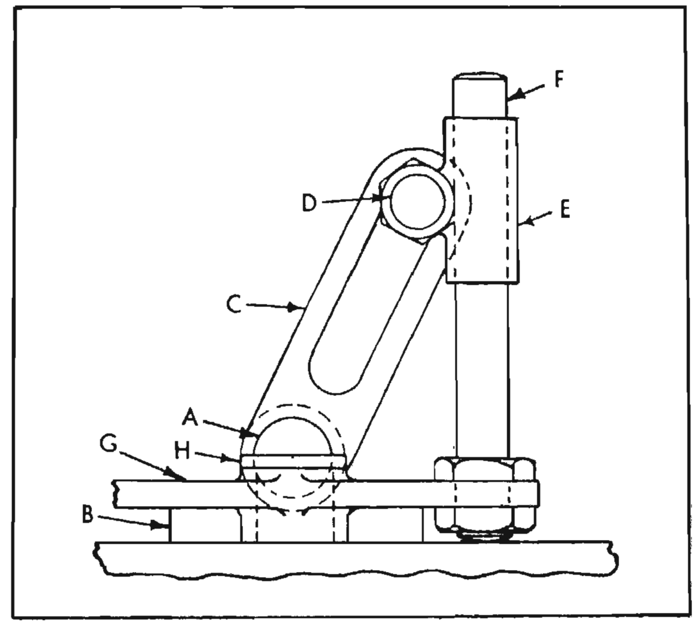


Fig. 15. Left-hand End View of Mechanism as Shown in Fig. 14.

The extreme forward position of the mechanism is illustrated by the plan view, in Fig. 14. Part *E* has moved out on rod *D*, away from lever *C*, and down on rod *F*, to permit movement of levers *C* and *G*, in their respective planes.

A left-hand end view of the mechanism when in the position indicated in Fig. 14 is shown in Fig. 15. Adjustment of the amount of oscillation is accomplished by changing the position of rods *D* and *F* in the slots of their respective levers.

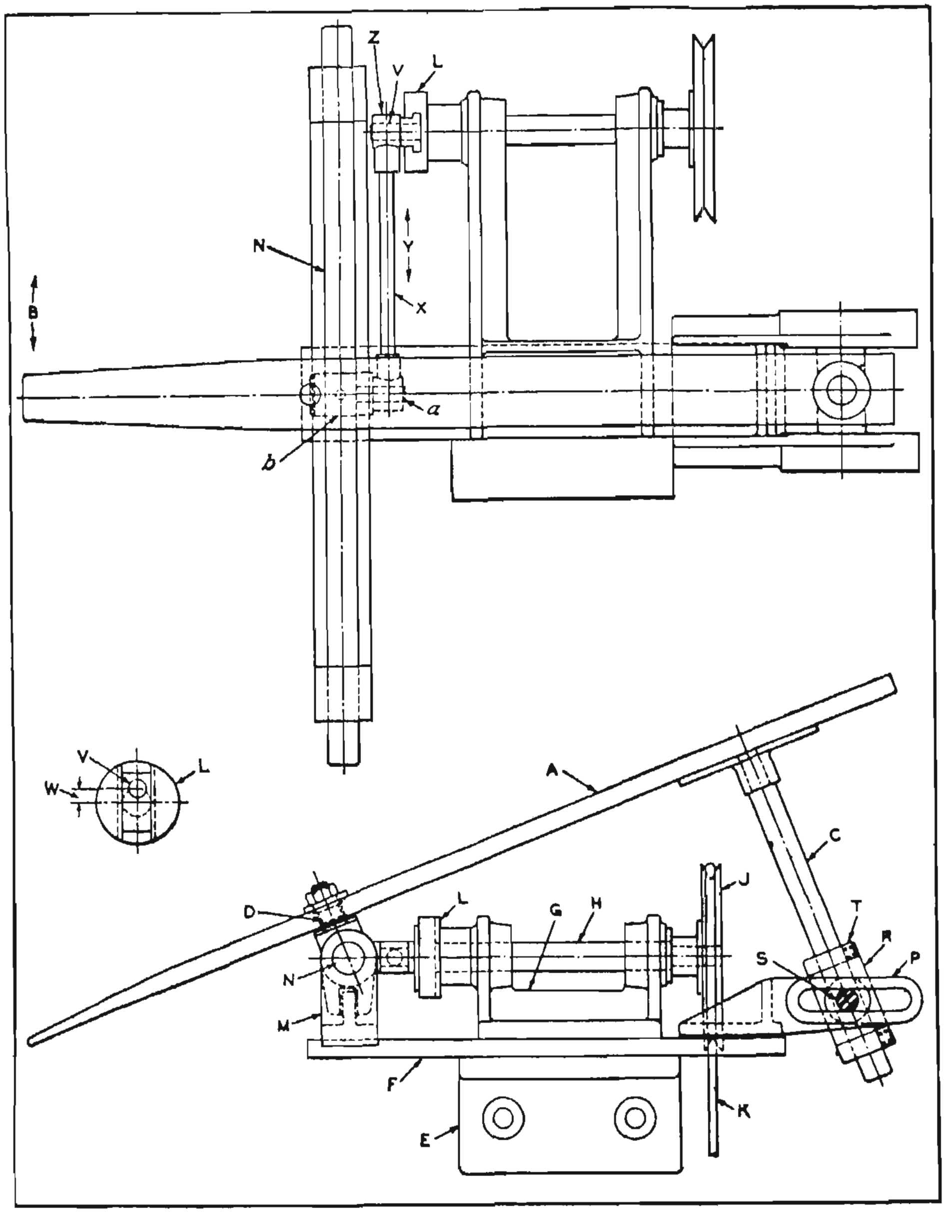


Fig. 16. Mechanism for Imparting Oscillating Motion to Arm A, which can be Adjusted to Various Angular Positions.

Mechanism for Imparting Oscillating Motion to Adjustable Arm.—The mechanism shown in Fig. 16 is designed to impart an oscillating or vibrating motion to the arm *A*, which can be adjusted to various angular positions. The mechanism is driven by a round belt *K* from a V-pulley within the machine on which the oscillating motion is employed. The complete mechanism, including arm *A*, is mounted on a baseplate *F*, which is secured to the machine by the bracket *E*.

Arm *A* pivots about the center of rod *C*, the oscillating motion indicated by the arrows at *B* being obtained from the reciprocating motion transmitted to the stud *D* of the cross-head *b*. The necessity for adjusting arm *A* to different angular positions is a factor that makes the design of this mechanism of particular interest. Provision for adjusting the length of the oscillating stroke also lends interest to the design.

The means for obtaining the oscillating motion will be described before considering the adjustment features. The shaft *H*, driven by belt *K*, is provided with a crankpin *V* for the bearing *Z* on one end of the connecting-rod *X*. The bearing at the opposite end of the connecting-rod is fitted on a pin *a* in a sliding block or cross-head *b*, of which stud *D* is a part. The reciprocating motion of stud *D* transmitted by the movement of the connecting-rod *X*, indicated by the arrows at *Y*, produces the required oscillation of arm *A*.

The bracket *M* supports the shaft *N* on which the cross-head *b* reciprocates. There is a slotted arm on the under side of the cross-head which is a sliding fit over the guide rail on the base of bracket *M*. The guide rail prevents the cross-head from rotating on shaft *N*. The length of the arc through which arm *A* oscillates can be changed to suit requirements by adjusting the sliding block in the flange *L* in which the crankpin *V* is mounted. Adjusting the block to increase the distance *W* of the crankpin *V* from the center of shaft *H*, as shown in the small insert diagram at the left of Fig. 16, increases the length of the arc through which arm A oscillates.

Stud D is fastened to arm A in such a manner that the arm can be tilted to any desired angle with the horizontal. The block R in which rod C pivots can be swiveled in bearing P and locked in the required angular position by a clamping bolt S. The collars T locate rod C in block R.

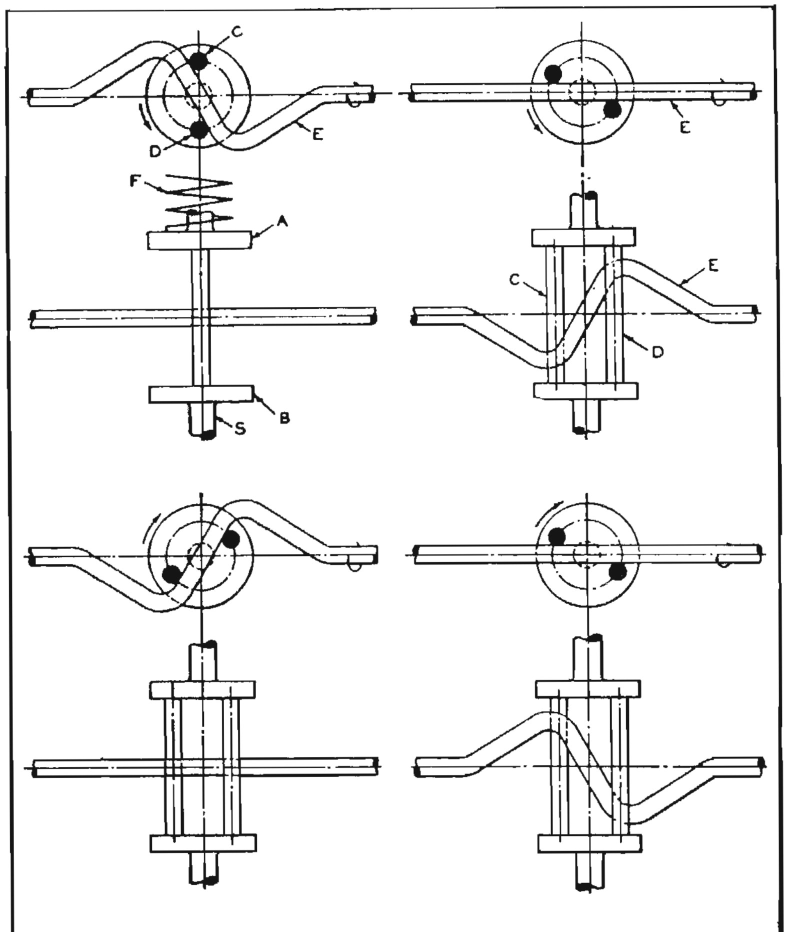


Fig. 17. Four Views of Simple Oscillating Mechanism, Showing How Rotating Shaft E Transmits Oscillating Motion to Shaft S.

Simple Mechanism for Producing an Oscillating Motion in a Shaft.—The mechanism shown in Fig. 17 was developed to produce an oscillating motion in the vertically suspended shaft S through the continuously rotating horizontal shaft E. It consists mainly of two disks A and B, both of which are a part of the vertical shaft assembly; the horizontal shaft E; spring F; and the pins C and D. The horizontal shaft E is bent to the shape indicated in the illustration, and is located between the pins C and D.

The four positions of the horizontal and vertical shafts shown in Fig. 17 indicate that the horizontal shaft serves as a cam, with th pins C and D acting as the cam followers. The shaft S is spring-loaded to keep the pins C and D in contact with the shaft E.

The mechanism can be operated at moderate speeds and, with a reasonable amount of tension in spring F, the oscillations will be imparted to the vertical shaft smoothly and without shock.

By changing the shape of the shaft E, sufficient variation in the oscillations can be obtained to meet different requirements. Some experimental work may be necessary when precise results are desired. However, a fairly accurate lay-out will usually be sufficient to give the desired action.

Double Toggle-Lever Mechanisms for Operating Presses.— The capacity of a straw-baling press designed as shown in Fig. 18 was considerably improved by redesigning the operating mechanism. The machine has an oscillating pressing piston C, actuated by a rod B from a rotating crank A, which is constructed as a gear wheel and rotated by a small gear D. The pressing action takes place when the three links 1, 2, and 3 form practically a straight line, a high pressure then being applied to the straw bale during the interval y_1–y_2 of the crank motion. This interval is so short that the baling action is not quite perfect.

The full exerted pressure loads the mechanism. The swinging piston is quickly pushed forward and is already on its return stroke when the knot is tied. Therefore the elasticity of the bale stretches the binding cord and hinders the binding mechanism. High friction in the links increases the wear, and the forward and return strokes are performed at an almost identical speed.

In the improved mechanism, shown at the left in Fig. 19, two members E and F are added, and the number of links is thereby increased from four to seven (as the connecting link between the three members B, E, and F must be calculated as two links). The new mechanism has two dead points; therefore, the toggle action shown at the left in Fig. 19 is extended to about three times hat of the older mechanism. The push exerted during the pressing action is taken off the gear and transmitted to the fixed link 5. The speed of the piston is gradually reduced until it is brought to a standstill. It remains in this position until the binding action is completed. The whole drive runs easily and without shock. The wear is consequently reduced to a minimum, and the capacity of the new machine is considerably increased.

Similar mechanisms can be employed in other industries, such as the one shown at the right in Fig. 19, which is used in a press designed for molding plastics. This mechanism

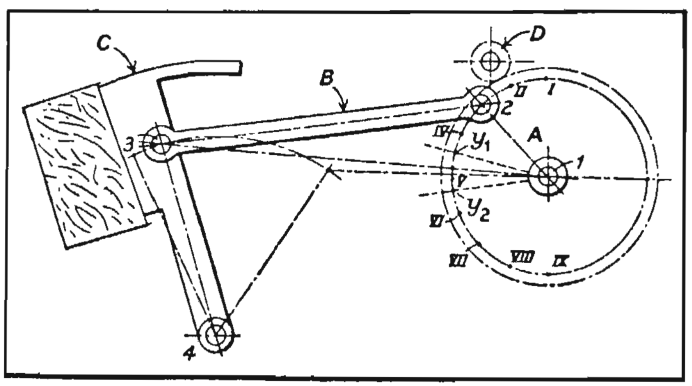


Fig. 18. Original Design of Mechanism for Straw-baling Press.

is the same as in Fig. 18, except that the swinging motion of the piston is replaced by the reciprocating sliding movement of a ram. Owing to its high capacity, better results are obtained with this press than with the older design having a simple crank mechanism.

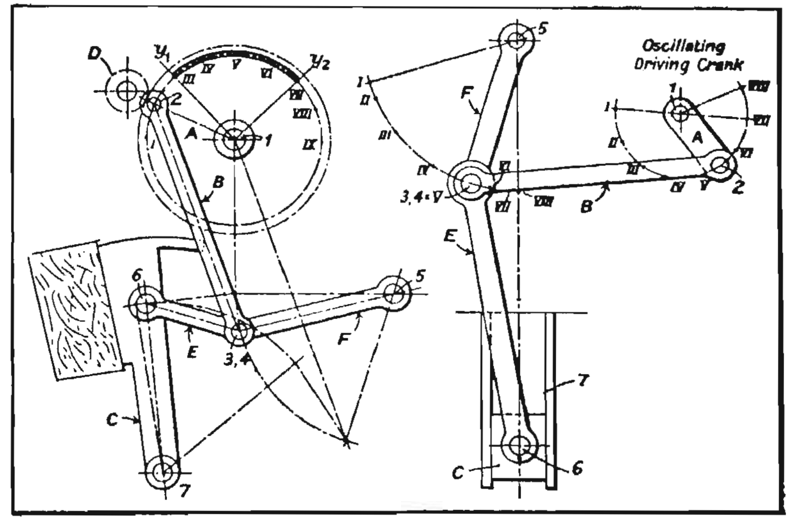


Fig. 19. (Left) Double Toggle-lever Mechanism for Straw-baling Press. (Right) Double Toggle Lever for Plastic Molding Press.

Mechanism for Applying Rotary or Oscillating Motion to a Driven Shaft.—The mechanism shown in Fig. 20 is designed to give a rotary or oscillating motion to the driven shaft E. The oscillating motion is made possible by the oscillating action of the segment gear M. When this motion is required, the driven shaft changes its direction of rotation every half-revolution and operates at one-eighth the full rotating speed of the motor driving shaft. When a continuous rotating motion is required for shaft E, sliding collar D of the clutch is engaged with clutch member C. Gear F then rides on the cylindrical part of the clutch member D, from which it is disengaged, and shaft E rotates continuously at the motor speed.

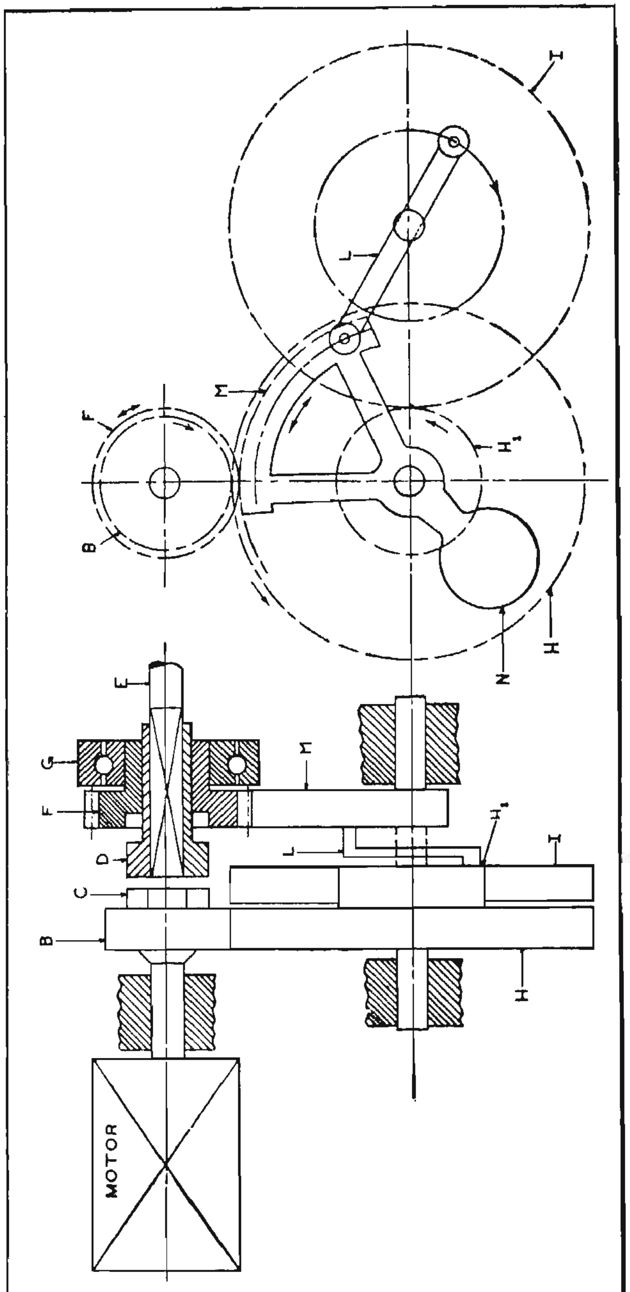


Fig. 20. Mechanism for Imparting Either Oscillating or Continuous Rotary Motion to Driven Shaft E.

The diagram at the left of Fig. 20 shows the mechanism in the neutral position. The gears B, H, and H_1 have a ratio of 1 to 8. The diagram at the right shows the gearing arrangement that makes possible the oscillating motion. Crank L is attached to gear I in an offset position, so that it provides a crank movement for oscillating segment gear M which meshes with gear F. The ratio of the pitch diameters is so selected that gear F is rotated 180 degrees in a forward and return movement; gear F is connected to the driven shaft E, giving it a similar motion when member D is moved to the right. This mechanism can be improved by introducing a second clutch to permit gear B to be disconnected from the motor shaft for the direct drive and also by attaching a balance weight N on the segment gear M, as shown in the right-hand diagram of Fig. 20.

Slight Oscillating Movement with Interrupting Control.—A mechanism for imparting a slight oscillating or vibrating movement to a shaft, which is equipped with an arrangement for interrupting the oscillating movement without

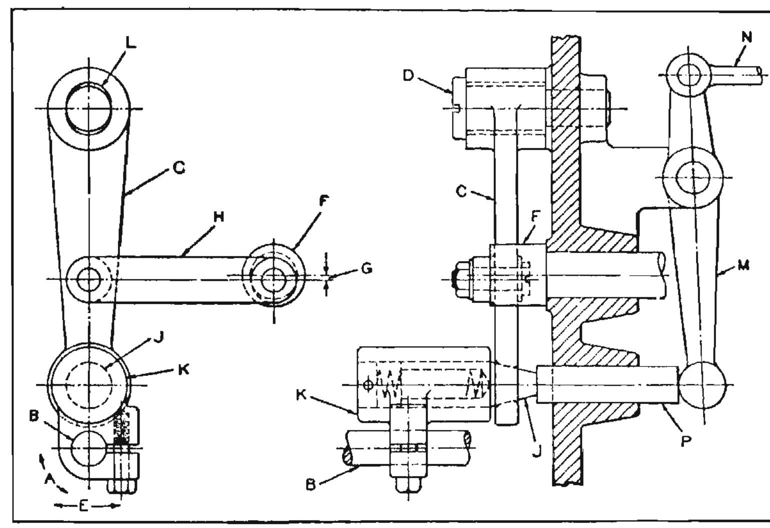


Fig. 21. Mechanism Designed to Impart Oscillating Movement to Shaft B with Provision for Interruption by Lever M.

stopping the driving shaft, is shown in Fig. 21. This device rocks shaft B back and forth, as indicated by arrow A, by having lever C, which is pivoted on stud D, rock about the center of the stud with a radial movement, as indicated at E. To operate lever C, an eccentric shaft is provided at F with an offset equal to G. As the shaft F rotates, the connecting link H is moved back and forth, imparting movement to lever C.

Lever C transmits a rocking movement to shaft B through engagement of the spring plunger J mounted in lever K, with a hole in lever C. The hole for the stud about which lever C oscillates is elongated at L to allow the two levers to pivot. Normally, the movement described is continuous, but at certain times it is necessary to stop the movement of lever K and to permit lever C to continue in operation.

This is accomplished by providing an arm at M which is actuated by a connecting-rod N so that a rod P pushes plunger J far enough to the left so that the clearance between the plunger and the tapered hole is great enough to permit lever C to oscillate without transmitting sideways motion to plunger J. This prevents the movement of lever K, which, in turn, stops the rocking movement of shaft B. Shaft B serves to operate other mechanisms within the machine. This plunger type of mechanism is suitable for use where only a slight rocking movement is necessary. The bevel plunger J is raised just a sufficient amount from the beveled hole to permit it to clear the hole at the extreme end of the oscillating stroke of lever C.

Feeding Mechanism Operated by Crank and Cam.— A very smooth oscillating movement is imparted to the fingers Y of the mechanism shown in Fig. 22 by means of a crank C and the cam B secured to the rotating shaft A. The oscillating movement is approximately as shown by the arrow E.

Arm D is free to slide up and down in a connecting block F, which is pivoted to the stud H. Stud H forms a part of

the lever C. As shaft A revolves, it carries block F to the left or right as indicated by the arrow E, and up and down through the medium of the connecting stud H. However, we are only concerned with the movement to the left and right, as the block merely slides up and down on the arm D, accomplishing no work while sliding vertically.

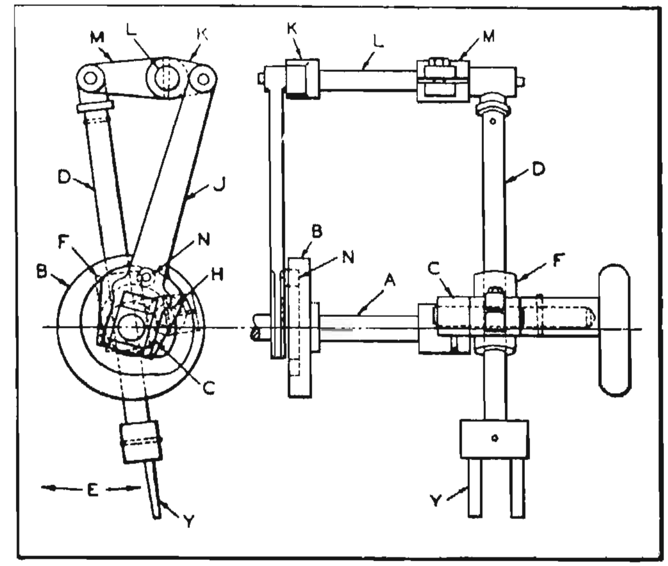


Fig. 22. Mechanism for Imparting Smooth Oscillating Movement to Fingers Y from Rotating Shaft A.

While the oscillating movement is taking place, the cam fork J moves up and down, being actuated by the roll N which is attached to it and travels in the groove of cam B. This causes lever K to rock the shaft L, thereby causing lever M to move arm D up and down. The combined up and down movement of arm D and left and right movement imparted by the eccentric throw of lever C results in a very smooth oscillating movement of the fingers Y.

Oscillating Mechanism for Milling Machine Shaping Attachment.—The oscillating mechanism shown in Fig. 23 actuates a slot- or groove-shaping attachment for a milling machine. The grooves G to be cut in the ends of bars W, shown at the upper right of Fig. 23, have their bottom surfaces machined to conform with an arc of a circle. The milling attachment was designed to handle this machining operation, because the bars were too long to be handled in a lathe.

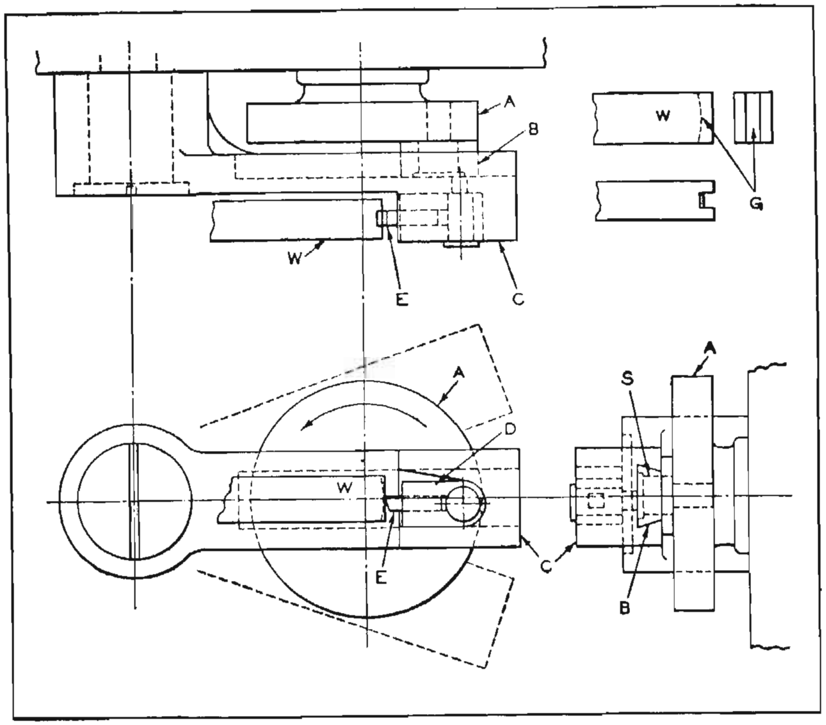


Fig. 23. Mechanism Applied to Milling Machine for Oscillating Shaper Tool Used to Cut Groove G in Work W, Shown at Upper Right.

Referring to the oscillating mechanism, the disk A is driven by the spindle of the milling machine, and carries a block B. Block B is free to swing on its stud S, and slides in a dovetail groove at the rear of lever C. Lever C is free to swing on its stud, which is fastened to the column of the milling machine. The rotation of disk A in the direction shown by the arrow imparts an oscillating motion to lever

C, as shown by dotted lines in the lower left view of Fig. 23, which indicate the two extremes of the oscillating motion.

The block D which carries the tool bit E is fitted into a recess in the front of the lever C, as shown in the lower left view of Fig. 23, and is free to swing on its stud. In this view, the tool bit E is shown in position for cutting, with the block D resting on the lower edge of the recess in lever C. The cutting action takes place on the upward swing of lever C. As the lever swings toward the bottom, the block D turns on its stud to prevent the cutting edge of the tool bit from dragging during the non-cutting half of the oscillating movement. Thus, block D operates in the same manner as the clapper block of a shaper. The work W is supported by clamping it on blocks on the milling machine table, the longitudinal feed being employed to advance the work to the shaping tool.

Oscillating Cam Made Adjustable.— On a recently designed automatic drilling machine, the work-holding fixture is operated by pneumatic cylinders. Compressed air is admitted to or discharged from the cylinders by means of a plunger valve operated by a cam supported on an oscillating shaft. For various types of work and different operations, it was necessary to provide some means of adjusting the timing for the application or release of the air pressure. The cam illustrated was designed for this purpose.

As shown in Fig. 24, plate B is clamped to and oscillates with shaft A. Cam-plate E is free to swing about the common center line of the plate and shaft, since it is loosely mounted on a hub of plate B. Fiber plates C (located between plates B and E) and plate D (on the outside face of plate E) are mounted on studs H. These studs are pressed into holes in plate B and pass through slots in cam-plate E.

Springs located between washers on the studs can be compressed by tightening the nuts. This causes the cam-plate to oscillate with plate B, the amount of oscillation being controlled by adjustable stops G and J.

Roller F, which is pinned to the plunger of the pneumatic valve, is shown in contact with the rise of cam E in Fig. 24. Movement of shaft A in the direction indicated by the arrow will cause the plunger to be depressed, thus permitting air to enter the cylinders.

Referring to Fig. 25, cam-plate E has moved to the right and the valve plunger is depressed. The cam-plate has come

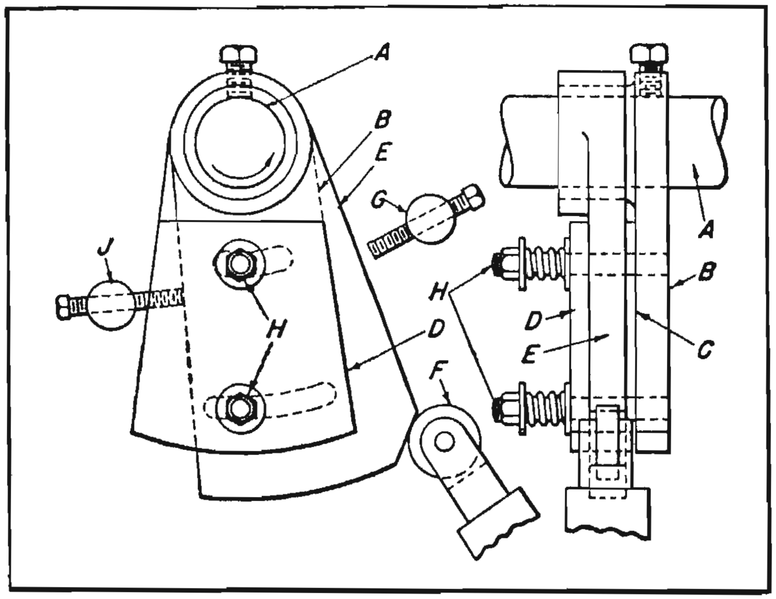


Fig. 24. Adjustable Oscillating Cam which Controls Admission or Discharge of Air to a Pneumatic Cylinder that Actuates Work-holding Fixtures. Amount of Oscillation of Cam-plate E can be Adjusted by Stop-screws G and J.

in contact with stop G so that further movement of the plate is prevented, regardless of the extent to which the shaft is oscillated. Then when the movement of the shaft is reversed, plate E returns to the position shown in Fig. 24, where further movement of the cam to the left is limited by stop J, and the valve is closed.

In this manner, the work-holding clamps are closed at the beginning of the first half of the cycle and released at the

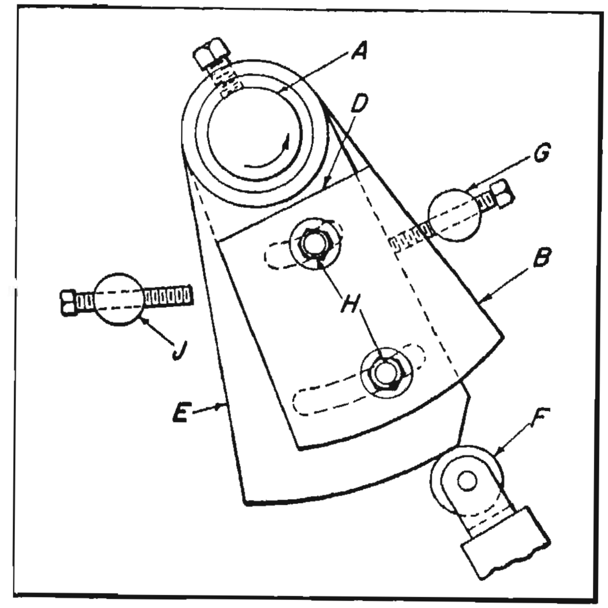


Fig. 25. Cam-plate E Shown in Fig. 24 has Here Moved to Right so that Valve Plunger Supporting Roller F is Depressed and Air is Admitted to Cylinder.

start of the second half of the cycle. Timing for the application or release of air pressure can be varied by adjusting the stop-screws. For example, if it is desired to open the valve later, the screw in stop J is backed off, thus causing a time lag before the valve is open. Similarly, the screw in stop G can be backed off to permit later closing.

CHAPTER 11

Mechanisms Providing Combined Rotary and Linear Motions

Described in this chapter are various mechanisms which impart a combined rotary or oscillating and linear or reciprocating motion to one or more elements. They provide, respectively, a combined rotary and traversing movement for a shaft; a reciprocating and rotary motion for a pneumatic drill; a reciprocating and rotary motion for a "flying needle" used in weaving wire cloth; a reciprocating shaft that rotates during part of its stroke; a ball and socket operation of a sleeve valve which reciprocates and oscillates; a crank which imparts an oscillating motion to one member and a reciprocating motion to another; a means of adjusting the radial position of a member while it is rotating; and a bearing designed for both rotary and reciprocating motions.

Mechanism for Producing Rotating and Traversing Movement.—A simple mechanism designed to impart a combination traversing and rotating movement to a machine member is shown in Fig. 1. Two gears C and D are keyed to the driving spindle K and retained by a flanged nut. The driving shaft L, which serves to rotate and traverse the machine member as required, has a screw thread and a keyway which is a sliding fit for the key P. The sleeve G is arranged to run freely in the bushings F, housed in the web E of the machine. End thrust imparted to this sleeve is taken by the large flange and the flanged nut N. Sleeve G is threaded on the inside to fit shaft L.

The gear B is keyed to the sleeve and the gear A runs freely on it, both gears being retained by the flanged nut M.

The part H is secured to gear A by three screws. Key P is held to part H, as shown, and slides in keyway in shaft L.

To make clear the operation of the mechanism, let it be assumed that gears A and C have forty-eight and twenty-four teeth, and gears B and D, forty-seven and twenty-five teeth, respectively. Let the member to be actuated, and consequently shaft L and gear A, make one revolution. Then gear C and shaft K will make $48 \div 24 = 2$ revolutions.

When gear D makes two revolutions, gear B makes

$$\frac{25}{47} \times 2 = \frac{50}{47} \text{ revolutions.}$$

Thus shaft L rotates relative to sleeve G, and a traverse feed movement is imparted to the member. If the screw thread on shaft L has 10 threads per inch, the distance through which shaft L is fed during one revolution equals

$$\left(\frac{50}{47} - 1\right) \times \frac{1}{10} = 0.0064 \text{ inch.}$$

Different feeds and speeds can be obtained by varying the ratios of A to C and B to D.

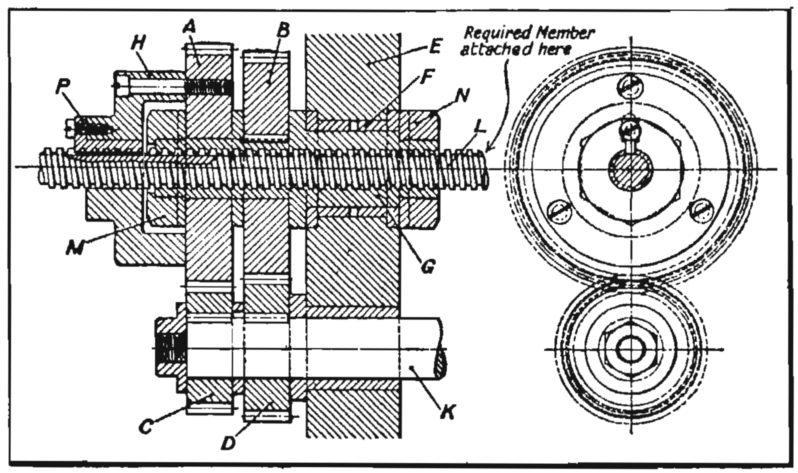


Fig. 1. Mechanism for Imparting a Rotating and Traversing Movement to Shaft L from the Driving Spindle K.

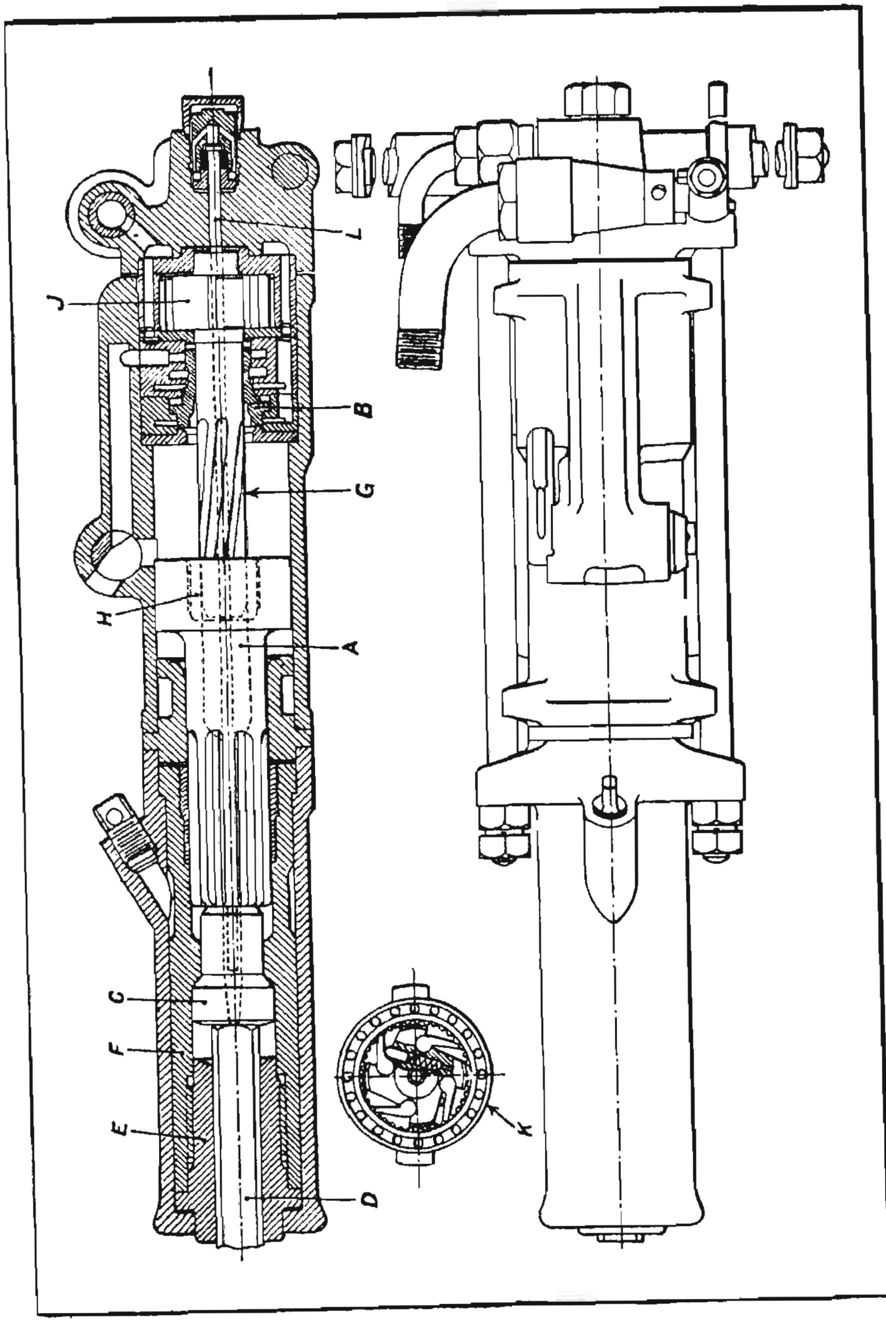


Fig. 2. Anvil-block Type Mechanism of Pneumatic Rock Drill.

Reciprocating and Rotary Motion for Pneumatic Drill.— Several different types of valve-gear mechanisms are employed for the operation of pneumatic rock drills. One of these types is shown in Fig. 2. This particular mechanism is used for a 3-inch anvil-block type hand hammer drill employed in the mining industry. The movements of the piston *A* are controlled by a spool valve *B*. The tool derives its name from the anvil *C*, which is interposed between the piston and the drill *D*. The particular drill shown has a shank of hexagonal cross-section. It is guided in the bushing *E*, which, in turn, is held in the tool-holder *F*. Anvil *C* is guided by tool-holder *F*.

The essential movements for rock drilling are an uninterrupted series of blows in rapid succession on the end of the drill, and a rotary motion of the drill, which must be turned a few degrees between successive blows. The blows, 1600 to 2200 per minute, are imparted by the piston *A* operating on the anvil block *C*, the velocity of the piston at the moment of impact being between 20 and 25 feet per second. The turning motion is imparted to the drill between successive blows by the helical-splined rifle bar *G*. This bar is engaged by a phosphor-bronze nut secured in the piston at *H*.

The rifle bar is provided with a ratchet mechanism at *J*, an end view of this mechanism being shown at *K* at the left of Fig. 2. The ratchet mechanism allows the rifle bar to turn in one direction only. On the return stroke of the piston, the ratchets prevent the rifle bar from turning and, in consequence, the piston turns a few degrees on the rifle bar in accordance with the helix angle of the splines. On the power stroke of the piston, however, the rifle bar is free to turn, and the piston moves forward without rotating, the inertia of the piston and the parts keyed to it causing the ratchet gear to slip. The piston is splined, and the splines are engaged by a nut in the tool-holder *F*. The turning movement of the piston on its return stroke, therefore, is transmitted to the tool-holder and to the drill.

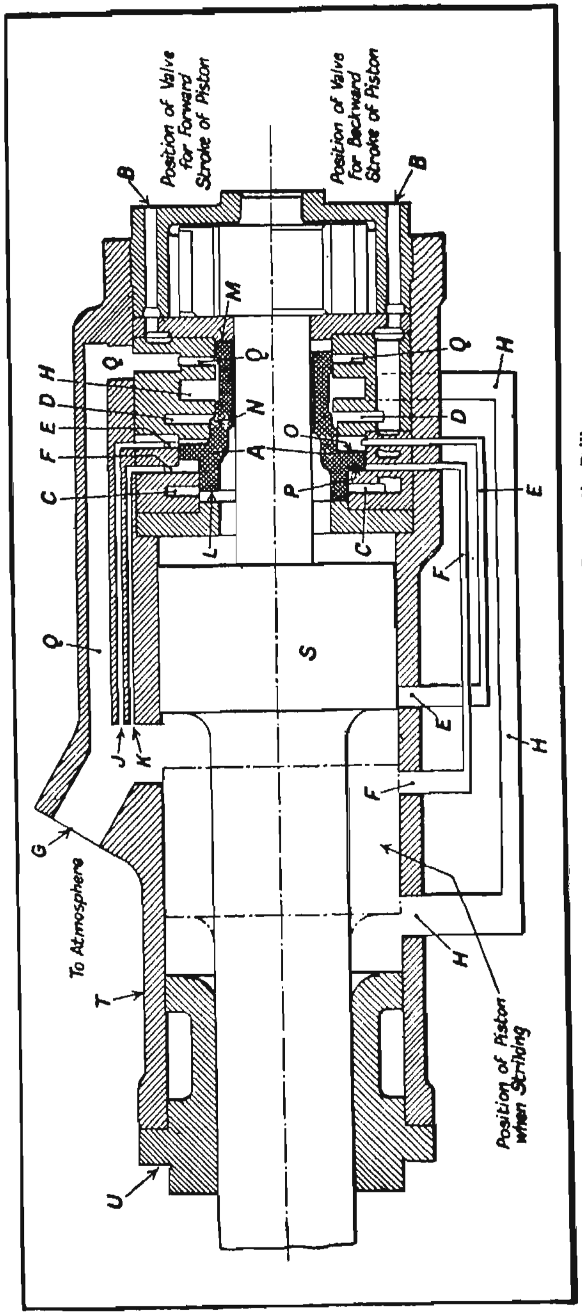


Fig. 3. Spool Valve Mechanism of Pneumatic Drill.

Water or air is almost always required at the drill point to facilitate the removal of the rock fragments or chips. In consequence, the drill is usually hollow, so that air or water can be fed through it to the point from the tube L by way of the hollow anvil block, piston, and rifle bar.

Fig. 3 is a diagrammatic sectional view of the pneumatic drill, illustrating the operation of the spool valve. The top half section shows the valve in position for admitting air to the rear of the cylinder, while the bottom half section shows the valve in position for admitting air to the opposite side of the piston.

The action of the valve is as follows: With the piston S in its rearmost position, air is admitted through the passages B and port C to the cylinder, as seen in the top half section, to drive the piston outward. The valve A is locked in this position by the air pressure acting on its front face L, the area of the front face being larger than that of the rear face M, and both faces being subjected to the same air pressure.

On the outward stroke, when the rear edge of piston S uncovers the port E in the cylinder, air at cylinder pressure passes up through the passage E to a port in the valve box, and acts on the face O of the valve. This moves valve A to the left, closing port C and cutting off the air supply which serves to drive the piston outward. At the same time, the valve opens communication between the live air port D and the port and passages H, permitting air to pass to the front of the cylinder and drive the piston back. Just before piston S delivers its blow, the rear edge of the piston uncovers the main exhaust port G, and the air behind the piston passes out to the atmosphere.

In the live air port D, there is a small face N on the valve, and the air pressure acts on the face, tending always to move the valve to the left. This serves to lock the valve in the position it takes up when air is being supplied to drive the piston back.

On the return stroke of piston *S*, when the rear edge has passed and closed exhaust port *G*, the remaining air in the rear of the cylinder is compressed until the pressure is sufficient to move valve *A* to the right, back to its initial position.

There is a front control port *F* in the cylinder, which is connected to the port *F* in the valve box. When the piston uncovers this port on its return stroke, air passes through the passage *F* and acts on the valve face *P*. This tends to move the valve to the right, but in ordinary running, the port *F* in the cylinder is inoperative, the compression pressure in the rear of the cylinder on the return stroke of the piston being the real cause of the movement of the valve. The purpose of port *F* in the cylinder is to prevent the piston from coming to rest with the main exhaust *G* slightly open to the front end of the cylinder, which would cause difficulty in restarting. On the return stroke of piston *S*, the main exhaust port *G* is uncovered before the piston is brought to rest, and the air in front of the cylinder exhausts to the atmosphere through this port.

As soon as the front end of piston *S*, in moving outward to deliver its blow to the drill, has closed main exhaust port *G*, there is a tendency for the air trapped in front of the piston to cushion the force of the blow delivered to the drill. This is, however, taken care of. Referring to the top half section of the illustration, it will be observed that when live air is being supplied to the rear end of the cylinder the front end of the cylinder has an opening to the atmosphere through passage *H*, the reduced diameter of valve *A*, and the port and passage *Q*. The air in front of the cylinder can therefore exhaust to the atmosphere, and cushioning of the piston prior to the blow is avoided.

It will be noted that the front air supply port *H* is partly covered when the piston is in position for striking the drill. This prevents damage to the piston in case the drill should be run without drilling actually taking place, or in the event that the operator does not hold the tool down to the work.

When the piston passes port *H*, air is trapped between the front face of the piston and the face of the division ring *U*, and the resulting compression prevents damage to the division ring.

Two small exhaust passages *J* and *K* are controlled by the faces *O* and *P* of valve *A*. These passages insure that no residual air pressure will remain in ports *E* and *F* of the

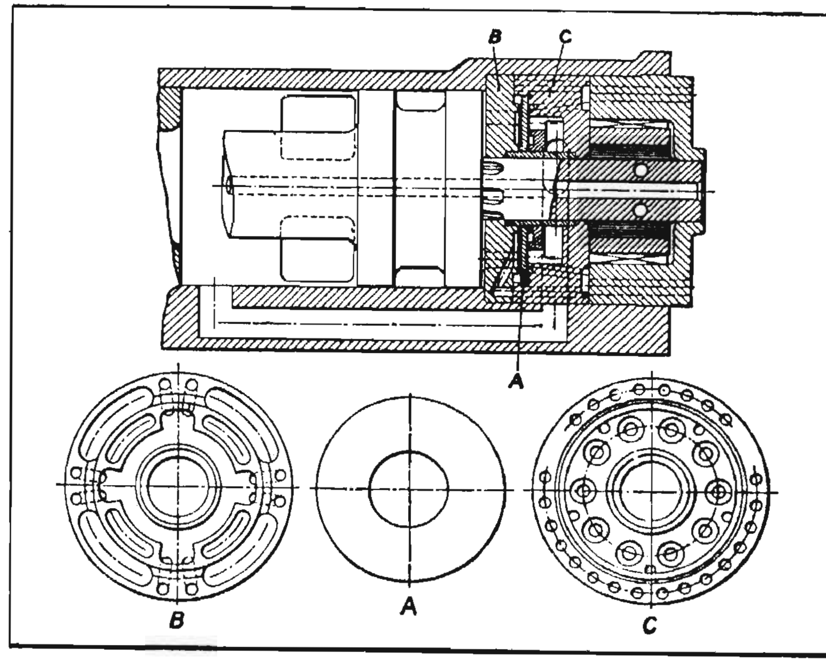


Fig. 4. Diagram Illustrating Operation of Disk Valve of Pneumatic Drill.

valve box after the valve has been moved to either end of its travel. The absence of residual pressure in these ports is essential for free movement of the valve.

Several other types of valve gear are employed. In the design shown in Fig. 4, the valve is of the disk type, the steel disk *A* operating between the valve seats *B* and *C*, and performing the same duties as the spool valve previously described.

Compact Mechanism Provides Combined Reciprocating and Rotating Motion.—A mechanism for producing a combined reciprocating and reversing rotating motion, which is of particular interest because of its simplicity and compactness, is shown in Fig. 5. This mechanism is used to operate a "flying needle" on a machine for producing a woven-wire product.

The shaft *A*, which is supported in bearings *B* and *C*, has a series of modified gear teeth on it which mesh with the teeth of gear *E*. The base of bearing *C* carries the roller *D* which engages a helical groove in shaft *A*. In operation,

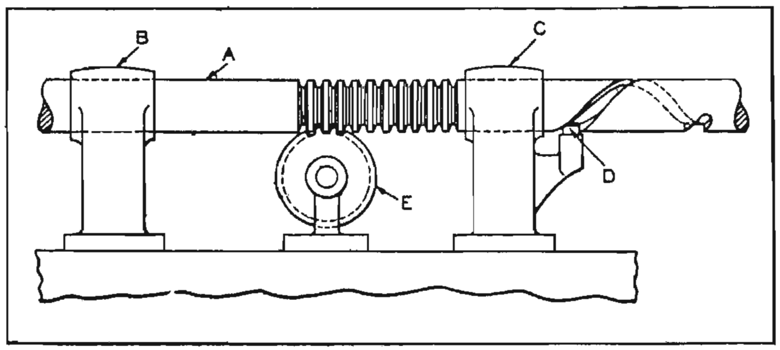


Fig. 5. Mechanism for Reciprocating and Rotating Shaft A.

gear *E* is given a reversing rotary motion by means of a sliding dog mechanism. As shaft *A* is reciprocated by gear *E*, it is also given a reversing rotating motion by the action of roller *D* in the helical groove. This arrangement provides a smooth uniform motion.

Reciprocating Shaft that Rotates During Part of Stroke.—A somewhat similar arrangement to that just described is shown in Fig. 6. Arm *A* receives a reciprocating motion from an independent source and transmits this motion to shaft *B*. When shaft *B* moves to the left, it is rotated through roller *C* engaging a helical groove in the shaft. This combined movement continues until the collar *D* comes in

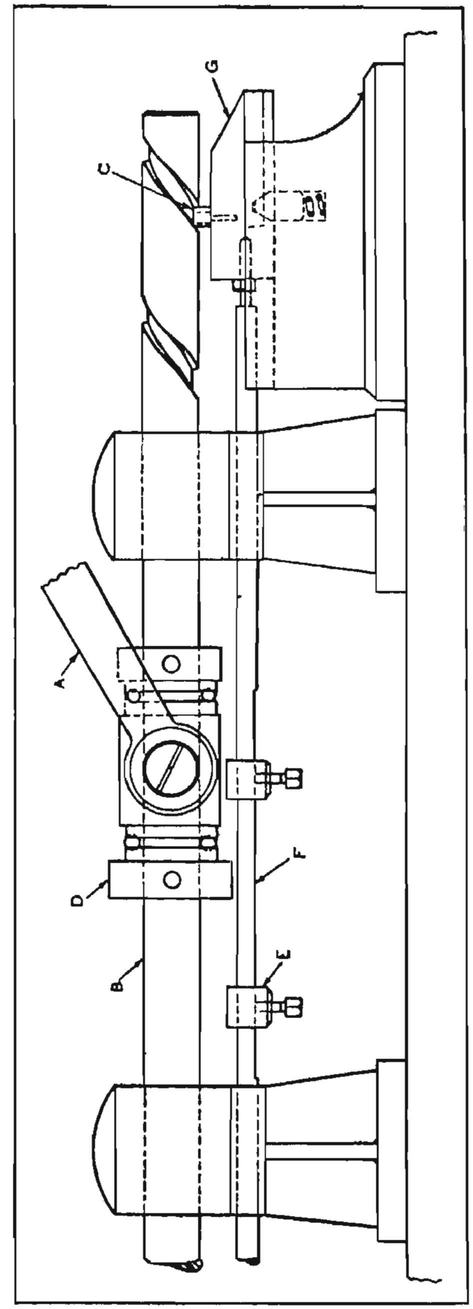


Fig. 6. Mechanism for Obtaining Combined Reciprocating and Rotating Motion.

contact with stop E. Upon further movement of shaft B to the left, rod F and slide G are drawn to the left at the same speed. It can be readily seen that when this simultaneous action is taking place, shaft B has no rotary motion. The position of stop E can be adjusted to vary the rotary motion of shaft B.

Ball-and-Socket Mechanism for Operating a Sleeve Valve.—In Fig. 7 is shown a ball-and-socket mechanism for imparting a combined oscillating and reciprocating motion to a sleeve valve. This mechanism was patented in England.

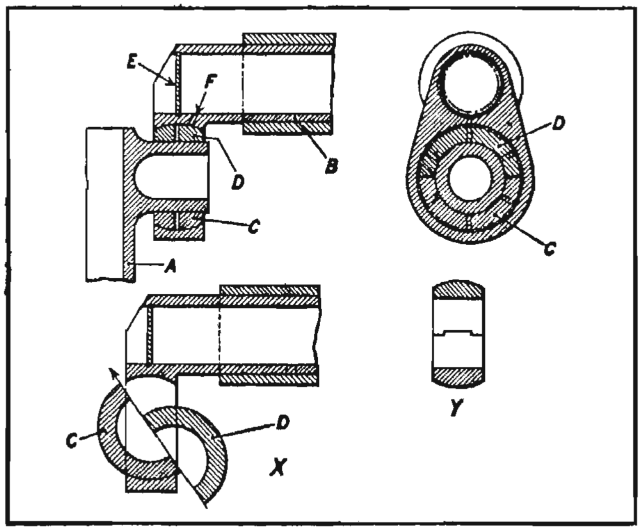


Fig. 7. Ball-and-Socket Mechanism for Imparting a Combined Reciprocating and Oscillating Motion to a Sleeve Valve.

Its most interesting feature is the novel method of assembling the simple but efficient ball-and-socket joint. The sleeve A is driven from the rotating crankshaft B, the socket being formed in the crank and the ball mounted so that it can slide on a pin projecting from the sleeve, as shown at the upper left of Fig. 7.

As shown, the annular ball member is made in halves C and D. To assemble the joint, one half of the ball member,

say C, is set in the socket with its axis at right-angles to that of the socket and its face inclined as illustrated in the view at X. The other half D is then slid into position in the direction of the arrow, and the two halves turned together through 90 degrees, so that they are co-axial with the socket. The pin projecting from the sleeve can then be inserted in the bore of the ball member, the two halves thus being maintained in the correct position. The two halves can, if required, be tongued and grooved, as indicated at Y.

In the arrangement illustrated, the hollow crankshaft is blanked off at E, and provision is made for feeding lubricating oil to the joint through the bore of the shaft and the hole F.

Foot-Operated Mechanism for Transmitting Rotary and Linear Movements.—The natural movement of the operator's leg as it swings a pedal forward is utilized in the mechanism shown in Fig. 8 to transmit a rotary motion to lever D, and, at the same time, impart an upward motion to shaft A, in the direction indicated by arrow B. The rotary motion occurs during the first part of the swinging movement of the foot-lever. Foot-lever E used for this purpose pivots on stud F. A stop-screw G locates the foot-lever in its starting position.

The operator, with his foot on the lever at X, kicks it back in the direction indicated by arrow H. At the beginning of the stroke, lever D is caused to move through the medium of connecting-rod J attached by adjustable rod ends K to studs L and M. This movement is transmitted by connecting link N to a unit within the machine on which it is employed. Lever D rocks on shaft P, which is supported in bearings in the end frames of the machine.

When pressure of the foot at X has moved the lever to the position indicated by the dotted lines at Q, lever D will have been rotated to the position indicated by the dotted lines at a, and surface R will be in contact with a roll at S on bellcrank T. The remainder of the foot-lever action, which carries lever E to position O, moves lever D to position b, and is utilized in actuating the toggle lever of which crank T and link V form a part. A vertical movement is imparted to shaft A through toggle connecting-link stud W traveling along path y.

Bellcrank T pivots on and is supported by a fixed stud Z. Link V is pivotally attached at I to a block pinned to shaft A. The effect of this toggle linkage and rotary lever action

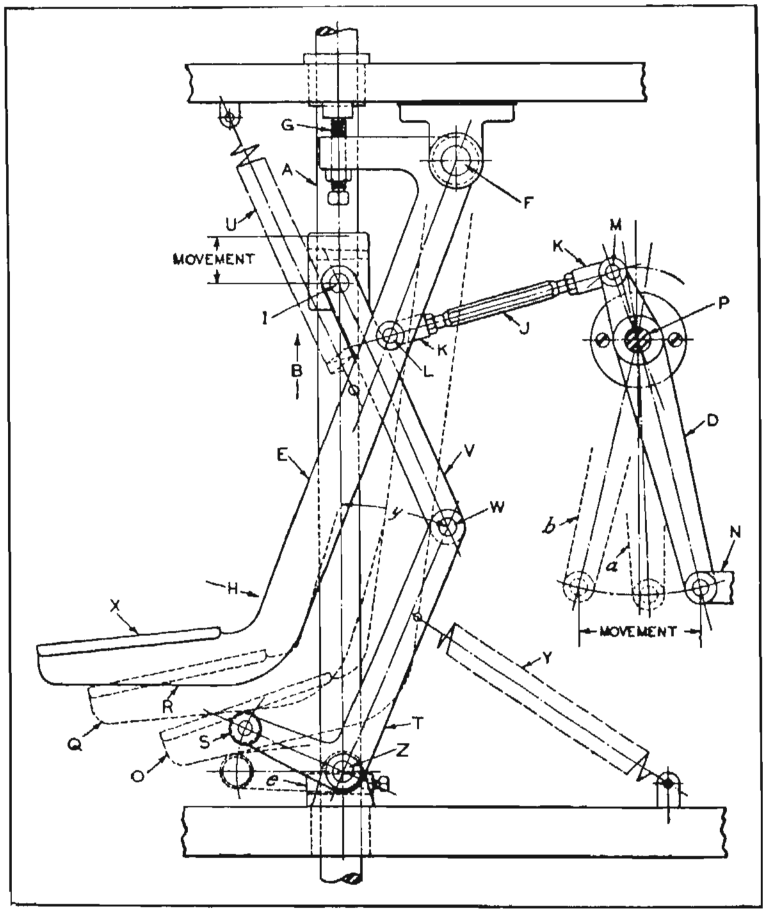


Fig. 8. Foot-operated Mechanism for Transmitting Rotary Motion to Lever D and Vertical Movement to Shaft A.

is to combine a crosswise movement of connecting link N with a delayed vertical movement of shaft A when the foot-lever is given a full swinging movement from the position shown by full lines to that shown at O. A spring U serves to return the foot-lever to its starting point. Another spring at Y returns bellcrank T and shaft A to their starting points which are determined by collar e.

External Control for Mechanism within Rotating Member.—Fig. 9 shows a device that provides exterior control of the movements and adjustments of some movable element on a rotating shaft. By such a device, for example, the radial adjustment of the tool on the arm of a boring-bar can be made while the bar is in motion, the adjustment being made through controls that do not revolve with the bar and that can be operated as conveniently as when the bar is stationary; or the blade of a revolving screw propeller can be adjusted for pitch without regard for the motion of the propeller shaft.

The illustration shows a more or less general arrangement of the device, many of its elements being subject to modification to adapt it to special needs and to the ideas of the designer. Also some construction refinements have been slighted for the sake of clarity, thus making the drawing somewhat diagrammatic.

The moving element to be adjusted is represented by the double crankshaft A, mounted in bearings that are carried on and rotate with the shaft B. The cranks are quartered, or set at 90 degrees to each other, and are driven by the connecting-rods C and D. Near the other end of the mechanism is another double-throw crankshaft E with quartered cranks having throws equal to those of crank A. It is to this crankshaft that the initial controlling movement is applied by hand-crank T.

Unlike shaft A, shaft E has no other motion than rotation about its own axis. Through the medium of the two connecting-rods F and G, this shaft imparts a reciprocating

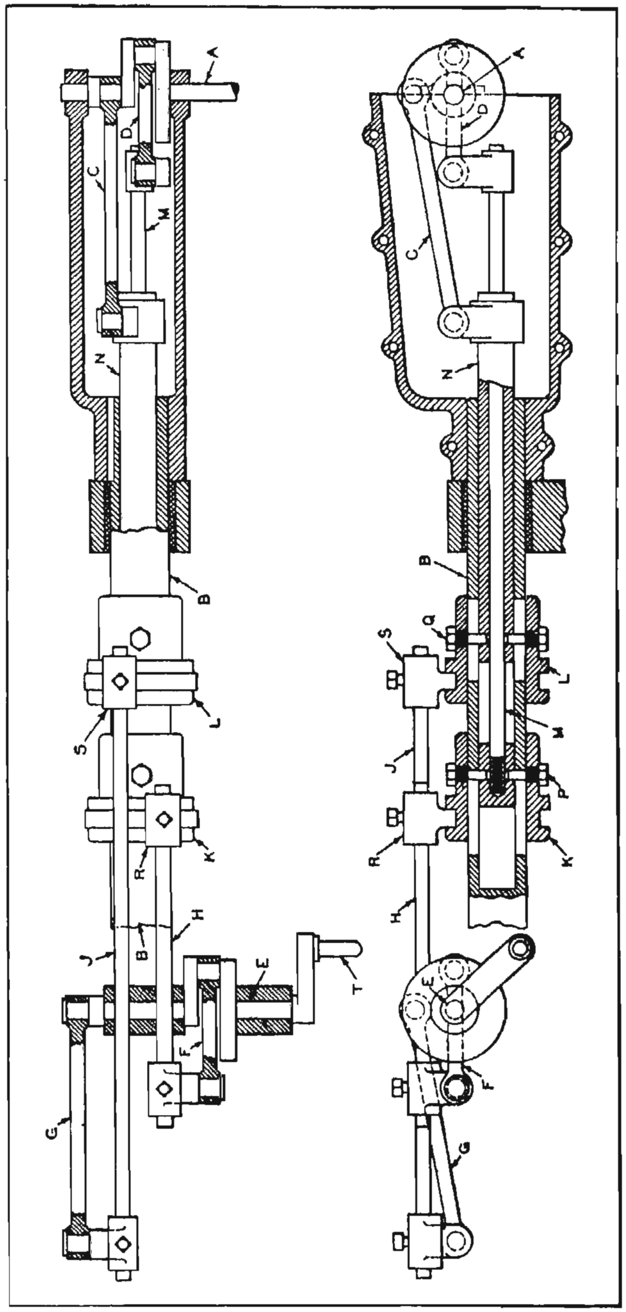


Fig. 9. Crank-operated Control for Mechanism within Rotating Member.

longitudinal motion to the rods H and J. This motion is transmitted to the sliding collars K and L, and thence to the central rod M and the sleeve N, which deliver the motion to the connecting-rods C and D. Any motion of the shaft E is thus duplicated in shaft A.

The two collars K and L rotate with shaft B, but are free to slide longitudinally on it. By means of pins P and Q, passing through slots in shaft B, the sliding motion of the collars is imparted to rod M and sleeve N. The reciprocating motions of rods H and J are transferred to the collars by means of the shoes R and S which ride in the grooves of K and L. It is thus seen that the motion of shaft E is transmitted to and translated on shaft A by longitudinal reciprocating motion at the points R and S, where the external stationary control parts meet the internal revolving control parts. This transmission of motion is thus independent of the rotative position of shafts B, and hence is entirely independent of its rotation.

Ball Reciprocating Bearings.–The term "ball reciprocating bearing" is used by the manufacturers of these bearings to designate a bearing capable of both rotation and axial reciprocation. Obviously, such a bearing cannot have definite ball tracks or grooves. Sometimes rotation only is required, and sometimes axial reciprocation alone is desired. For this reason, the outer raceway is the inside of a hardened and ground cylinder, while the inner race is either the outside of a cylinder or the surface of a hardened and ground shaft. The balls are placed between the two surfaces and are generally retained in the pockets of a metal cylinder serving as retainer.

It is well known that a ball pressed against a flat plate, especially against a surface the curvature of which is opposite to that of the ball, has less load-carrying ability than the same ball placed in a groove similar to that of the conventional Conrad type ball bearing. The load-carrying capacities in the two cases have a ratio of about 1 to 10.

In order, therefore, to secure the greatest load-carrying capacity in a ball reciprocating bearing for a given size, there are but few alternatives. One of these is to provide the greatest possible hardness and uniformity of parts, in view of the fact that all contacts are minute ellipses of very limited area, and the harder the parts the greater the unit pressure that can be applied. Another solution is found in accuracy of manufacture and proper design to distribute the load uniformly over the bearing. A third alternative is by finding a means of increasing the number of balls in the same size bearing so that a proportionate gain in capacity is obtained. Thus, if it is possible to install twice the number of balls, the capacity of the bearing is doubled.

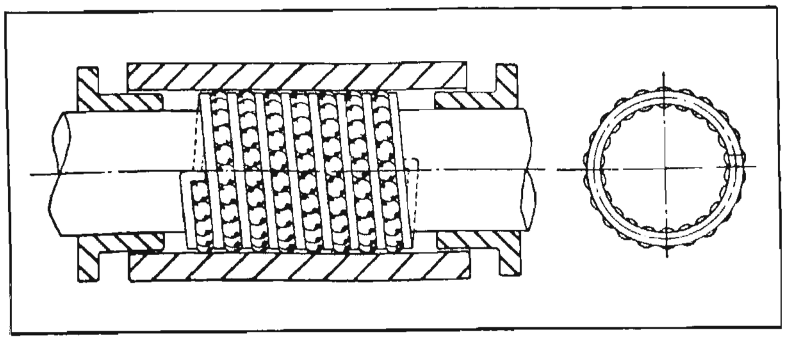


Fig. 10. Bantam Construction Ball Bearing Capable of both Rotation and Axial Reciprocation.

The Bantam Bearings Division of the Torrington Co. has been granted U. S. Patent No. 2,316,468 on means for increasing—in fact, doubling—the capacity of this type of bearing. The usual ball cage is a cylinder of bronze, steel, or aluminum drilled to form ball pockets of the greatest number per inch of cylinder length. The new Bantam construction consists of spirals of balls separated by coils of a specially formed helical spring, as shown in Fig. 10. The section of the spring wire may be triangular, round, square, or rectangular; but by making it in the form of an hourglass not only facilitates loading the balls, but retains them

between the races during shipping and installation. This construction is less expensive than the conventional type of retainer, both from the material and weight standpoint, and from a fabrication point of view.

In Fig. 10, it will be noted that the balls are restrained from end exit by the formed ends of the helix. When this bearing is used in rotation only, the helix of the retainer so positions the balls that each has its own special path as the shaft revolves.

In one test bearing, the pitch of the helix was 0.250 inch; each row contained twenty-two 3/16-inch diameter balls. Therefore, allowing no end play, each ball had a path (in rotation only) 0.0113 inch wide. Each path of the stationary race was thus loaded to maximum only once in about two revolutions, whereas in a grooved-race radial bearing with twenty-two balls in line, the maximum load on the stationary race would occur twenty-two times in two revolutions.

Ball reciprocating bearings are often used to operate in an axial direction only, and in such cases a large number of axial paths exist. The most common usage of this type of bearing is, however, in both reciprocation and rotation. When so used, the tracks of the numerous balls make intricate and overlapping designs, often covering every particle of the surfaces of both raceways. In this type of bearing, it is desirable to hold the diametral clearances close; a slight preload is permissible.

As a general guide for the capacity of this type of bearing, it may be said that the static capacity is dependent upon the number of balls, the diameter of the balls, and the radius of curvature of the inner race. Expressed as a formula,

$$C_s = KND^2F$$

in which C_s = static capacity, in pounds; N = number of balls; D = diameter of ball, in inches; F = curvature factor; K = constant = 200.

The constant K is derived from laboratory tests in which balls were loaded on hard flat steel plates up to the point where any greater load would cause "Brinelling" (indentations in the steel plate). The value of K is 200 for plates and balls of 60 Rockwell C hardness.

The value of the curvature factor F is given below. The value is found by determining R, the ratio of the inner race diameter to the ball diameter.

R	4	6	8	10	12	14	16
F	0.820	0.875	0.907	0.930	0.945	0.957	0.965

Values of F not given above may be interpolated.

To obtain the capacity C_r, in rotation only, the values of C_s may be modified by using $K = 100$ and applying speed factors. (These speed factors may be obtained from a manual published by the Bantam Bearings Division of the Torrington Co., South Bend, Ind.)

For both rotation and reciprocation, the capacity is little affected by the rate of reciprocation, as long as it is not excessive. Other factors that influence the capacity of this type of bearing are deflection of parts making up the assembly, accuracy of workmanship, over-travel of load, and lubrication.

Some of the devices in which this type of bearing is used are taper measuring instruments, engine governors, welding machine guides, printing press vibrator shafts, cloth-cutting machines, grinding-wheel dresser shafts, spool-winding machines, propeller-testing equipment, buffing machines, stem grinding and polishing spindles, wire-brushing machines, rotary-tool pipe cut-off spindles, airplane landing gear, inking rolls, sheet-polishing machines, gear grinders and lappers, paper-coating machines, remote-control gear-shift mechanisms, and hydraulic welding machines.

As the use of this type of bearing has occasionally been limited by its relatively low capacity, its field of usefulness should be greatly increased by this improved design which permits, in general, of doubling its load-carrying ability.

CHAPTER 12

Speed-Changing Mechanisms

Providing a fixed or adjustable speed of rotation of a rotating driven member that is different from the speed of rotation of the driving member can be accomplished in many different ways. Mechanisms described in this chapter illustrate the use of gears, ratchets, friction wheels, cams, pulleys and belts in combinations that are noteworthy for some ingenious feature or special function which they perform.

In one of the mechanisms discussed, the driven gear continues to rotate in the same direction at greatly reduced speed when the drive shaft is reversed. Also described are a cone pulley with epicyclic gearing for a high-ratio reduction drive; a friction drive designed for stepless speed variation; a cam-controlled variable-speed drive; a mechanism for controlling the speed of a driven shaft by changing the speeds of two driving motors; a cam-controlled worm wheel drive for controlling speed and direction of shaft rotation; a wabble-gear speed reducer; a device for automatically shifting a back gear in and out as the torsional resistance of the driven shaft varies; and a mechanism for changing cam speeds independently of camshaft speeds.

Other speed-changing mechanisms are described in Chapter 11 of Volume I and Chapter 10 of Volume II of Ingenious Mechanisms.

Mechanism for Producing Speed Change by Reversing Driving Shaft.—The mechanism shown in Fig. 1 is employed on a machine for fabricating a twisted wire product. This machine twists a group of wires together, the pitch or degree of twist being controlled by the rate at which the wire is fed into the twisting mechanism. The wire is twisted in one direction for a specified number of turns at a given pitch, and then twisted in the reverse direction for a number of turns at a greater pitch.

To obtain the required twist, the twisting mechanism is reversed while the feeding mechanism continues in the same direction but at an increased speed. The mechanism illustrated simply provides for changing the speed of gear *D*

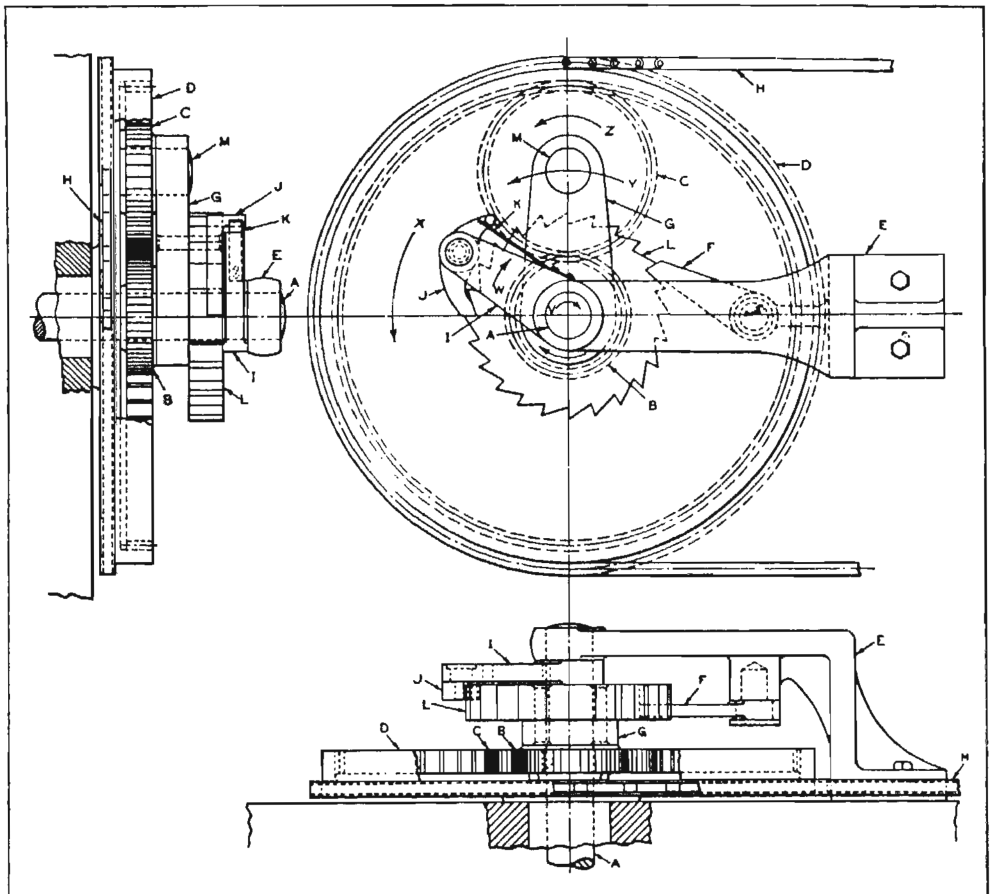


Fig. 1. Mechanism Designed to Drive Sprocket D from Shaft A at a Ratio of 1 to 4 when Shaft A Rotates in Direction Indicated by Arrow V, and to Continue to Drive Sprocket D in Same Direction but at a Ratio of 1 to 1 when Shaft A Is Reversed.

when the driving shaft *A* is reversed without changing the direction in which *D* rotates.

Referring to Fig. 1, the shaft *A* rotates in the direction indicated by the arrow *V*. Gear *B*, which is keyed to shaft *A* and rotates with it, meshes with gear *C*. Gear *C* meshes with the internal gear *D*, which is a free running fit on shaft *A*, causing it to rotate in the direction indicated by arrow *X*. The ratio between gears *B* and *D* of the mechanism, designed as illustrated, is 1 to 4, gear *C* acting as an idler while gear *B* drives gear *D*. Gear *D* is provided with sprocket teeth for the chain *H*, which transmits motion to the feeding mechanism at a distant point.

The lever *G* is free to turn on shaft *A*, and supports gear *C*, which rotates freely on stud *M*. Ratchet wheel *L* is also free to turn on shaft *A*, and is riveted to lever *G*. Lever *I* is keyed to shaft *A*, and carries the pawl *J*, which is held in contact with ratchet wheel *L* by the spring *K*. Bracket *E* supports the outer end of shaft *A*, which runs freely in its bearing. The pawl *F*, which engages ratchet wheel *L*, is also mounted on bracket *E*. As shaft *A* operates in a horizontal position, no spring is required to keep pawl *F* in contact with the teeth of ratchet *L*. All three views in Fig. 1 show the mechanism at the same point in the operating cycle.

In operation, shaft *A*, rotating in the direction indicated by arrow *V*, carries with it gear *B* and lever *I*. Gear *C*, meshing with gear *B*, rotates in the direction indicated by arrow *Z*, transmitting its motion to gear *D* in the direction indicated by arrow *X* at a reduced speed of rotation in the ratio of 4 to 1. Although lever *I* and pawl *J* rotate with shaft *A*, no motion is transmitted to ratchet wheel *L*, the teeth of which are designed for engagement with the pawl when rotating in the opposite direction. However, the rotation of gear *B*, acting on gear *C* at the point of contact, and the resistance of gear *D*, acting on gear *C* at the point of contact on the opposite side, tend to produce a turning movement of lever *G* in a clockwise direction.

As lever G and ratchet wheel L are riveted together, any motion or reaction affecting one also affects the other. Hence, the reaction on lever G serves to maintain engagement of pawl F with ratchet wheel L, which prevents clockwise rotation of the ratchet wheel. Thus the axes of shaft A and stud M are maintained in fixed positions; the assembly acts as a simple gear train, the power being transmitted from gear B to gear D through idler C in the ratio of their pitch diameters.

When shaft A, Fig. 1, which operates the twisting mechanism, is rotated counter-clockwise, or in the opposite direction to that indicated by arrow V, lever I also rotates in that direction, causing pawl J, which rotates with it, to engage the teeth of ratchet wheel L, so that the ratchet wheel will be rotated in the same direction, the pawl F at this point becoming inactive. As ratchet wheel L and lever G act as a unit, ratchet wheel L, stud M, and gear C rotate in the direction of arrow Y, with the axis of shaft A serving as a center. As the center of gear C is now rotating about the same axis and at the same angular velocity as gear B, there can be no motion of gear C about the center of stud M as an axis. Therefore, gear C can no longer operate as a gear, but merely serves as a connecting link for transmitting motion from shaft A directly to gear D. In this manner, the chain H is given a uni-directional movement at two speeds, as controlled by the direction of rotation of shaft A, which operates at a uniform speed, but is periodically reversed.

Two-Gear Speed-Reduction Mechanism Redesigned to Obtain Uniform Rotation.—Fig. 2 shows how the speed-reducing mechanism of the two-gear type illustrated and described in Volume I of "Ingenious Mechanisms for Designers and Inventors," pages 340-342, was redesigned to obtain uniform rotation. A standard internal gear G and pinion C were modified to operate without tooth interference in this redesigned mechanism.

On the driving shaft A is mounted an eccentric B. The

axis of driving gear C follows the motion of eccentric B, but is kept from rotating about its own axis by pin D, which works in the slot E. Linkage F is actuated by the eccentric B, which constantly maintains slot E in a perpendicular position through the action of the parallel links H, pivoted on studs J. Since the axis of gear C follows the motion of eccentric B and the gear does not rotate about its own axis,

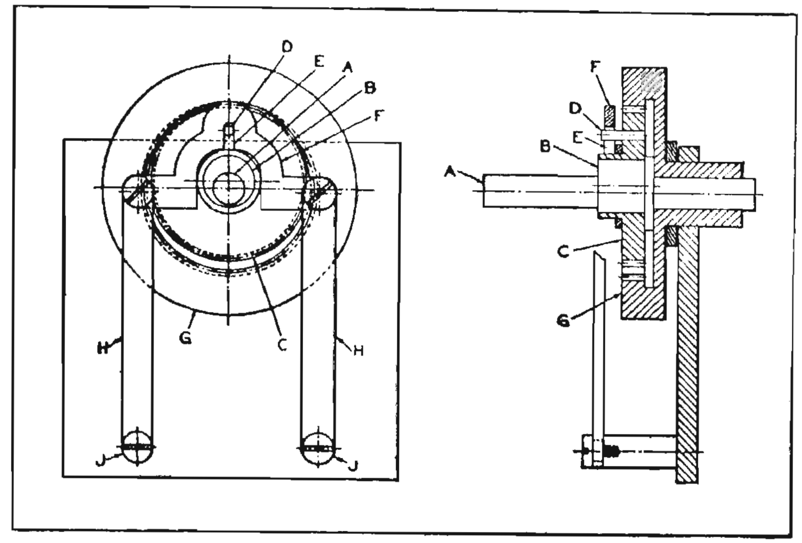


Fig. 2. Internal-external Gear Reduction Mechanism which Provides Uniform Speed.

the motion imparted to the driven gear G will be uniform and equal to

$$(\text{R.P.M. of } A) \times \left(\frac{N - n}{N}\right),$$ in which N equals the number of teeth in gear G and n equals the number of teeth in gear C.

Cone Pulley with High-Ratio Epicyclic Reduction Gearing.—The cone pulley with epicyclic gearing shown in the cross-section view in Fig. 3 was designed to provide a high-ratio reduction drive for the feed camshafts of several horizontal drilling machines. For this application, the cone pulley *P* was required to make 338 1/3 revolutions for each revolution of the driven shaft *S*. The drilling machines were being used on special automobile work at a time when it was impossible to obtain new or improved machines. The problem of designing the high-ratio reduction-gear drive was complicated by the necessity for keeping the size of the unit within certain dimensions, in order to permit it to be assembled in the space available. The design and construction of the reduction gearing unit was still further complicated by the fact that only a set of 16-pitch milling cutters was available for cutting the gears required.

Under these conditions, it was necessary to deviate from standard practice with respect to center distances between gears and the depth of the gear teeth. The special gears made to meet the unusual requirements, while theoretically incorrect, have given years of useful service. Exact center distances and details of the special gears are not given here, since it is unlikely that readers who may wish to build a similar high-ratio gear-reduction unit will be handicapped by the restrictions under which this unit was designed. With modern facilities for cutting any size and pitch of gear desired, it should be comparatively easy to build a similar gear-reduction unit of almost any desired ratio.

Referring to Fig. 3, the cone pulley *P* is of cast steel, finished all over and accurately balanced. Pulley *P* is fitted with an Arguto wood bearing and is mounted on the hub of the fixed 47-tooth sun gear *A*. The integral hub extending from the opposite side of sun gear *A* is pinned in a fixed position to the stationary housing *G*, which encloses two of the gears and is fastened to the drilling machine frame. The cone pulley is retained on the hub of stationary gear *A* by a fiber washer held in place by a dowel-pin and hexagonal head stud as shown.

The follower planet pinion *B*, having 20 teeth which mesh with the teeth of sun gear *A*, is pinned to the hub of the

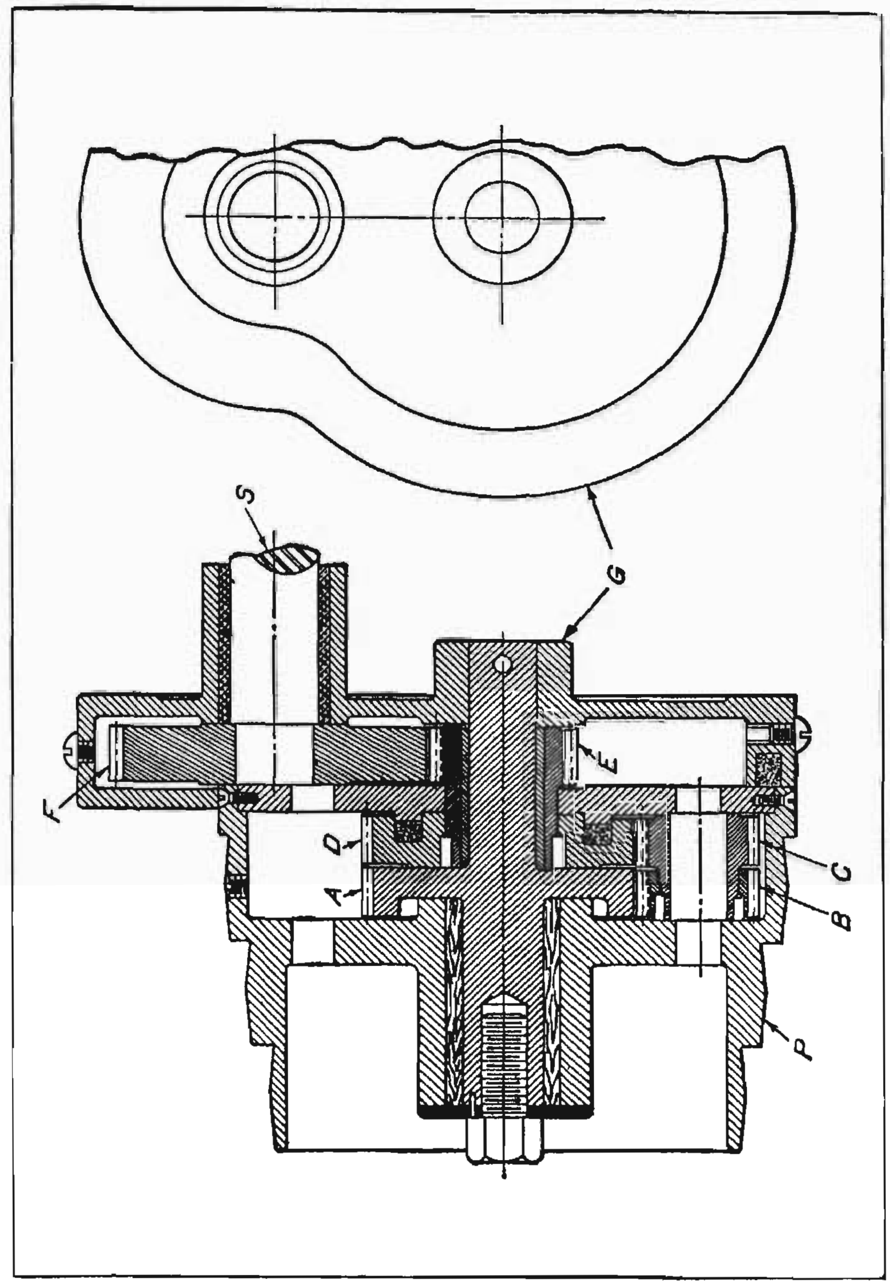


Fig. 3. Cross-section View of Cone Pulley with High-ratio Epicyclic Reduction Gearing and Partial End View of Gear Housing.

first driving planet pinion C, having 21 teeth. Pinions B and C revolve on a pack-hardened and ground tool-steel pin, one end of which is supported in the heavy web of the cone pulley P. The other end of this pin is supported by the steel cover plate secured to the large end of the cone pulley. Although not shown in Fig. 3, a second set of planet pinions, identical to those shown at B and C and mounted in the same manner, is located in a position diametrically opposite that of the ones shown.

The 49-tooth first-follower, revolving sun gear D meshes with planet pinion C and is keyed to the 24-tooth second-drive sun pinion E. The sun gear D and pinion E are fitted with a bronze bushing and run on the fixed hub of sun gear A. An annular groove in the face of sun gear D is fitted with a felt packing ring to retain oil in the chamber which encloses gears A and D and the two sets of pinions B and C.

The driving sun pinion E meshes with the 58-tooth final driven gear F keyed to the camshaft S, which runs in a bronze bushing in the stationary housing G. The chamber enclosing pinion E and gear F has a felt packing ring for retaining the gear lubricating oil. This chamber is filled with oil up to the bottom of the bushing in pinion E. The chamber which encloses gears A and D and pinions B and C is kept about one-third full of oil. The inner surfaces of both oil retaining chambers were coated with insoluble paint.

Three changes of speed are obtainable by shifting the driving belt on the three-step cone pulley. If necessary, the ratio between the speed of the cone pulley P and the driven shaft S can be reduced by using a smaller size driven gear F and a larger size driving pinion E. All gears are of machine steel and are pack casehardened. The bores and bearing surfaces of the gears are all accurately ground and fitted. The oil in the gear chambers should be changed frequently when the drives are new, flushing out the chambers with gasoline each time the oil is changed. After the gearing is well run in, oil need be added only as required to maintain the proper level.

It will be noted that in epicyclic gear trains gears that rotate on centers that also rotate are called planet gears or planet pinions, and gears that revolve about a fixed center are called sun gears. A sun gear may also remain in a fixed position, as is the case of sun gear A shown in Fig. 3. The planet pinions are carried by a rotating pulley, such as shown at P, Fig. 3, or a rotating arm, as indicated at P, Fig. 4.

Fig. 4 is a diagrammatic lay-out of the epicyclic gearing

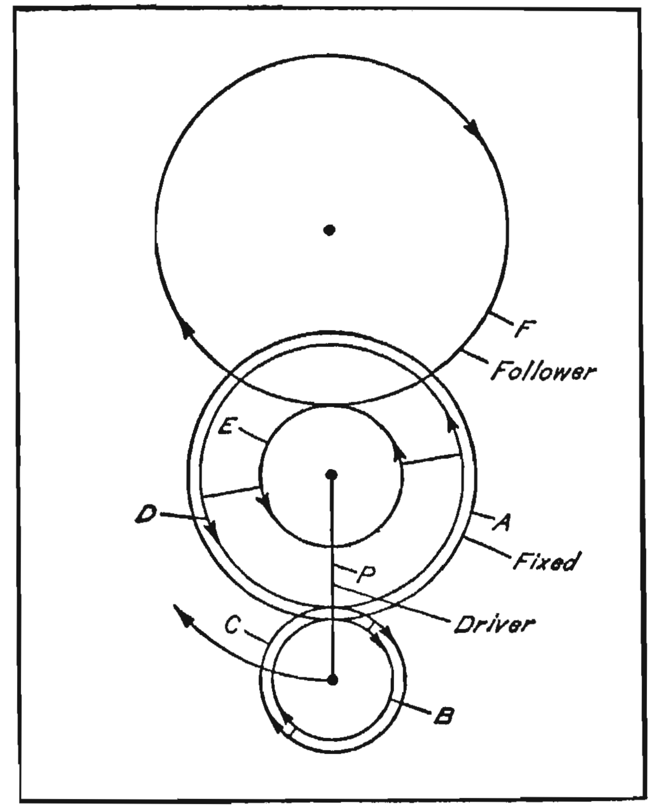


Fig. 4. Diagrammatic Layout of Epicyclic Reduction Gearing Used in Mechanism Shown in Fig. 3.

shown in Fig. 3, and is used to facilitate making the mathematical computations required in designing the epicyclic gear train. The same reference letters are used in both illustrations to indicate the same parts. This latter diagram has been made to a scale of one-half actual size, the circles in Fig. 4 indicating the pitch circles of the gears and pinions, and arm P representing the pulley P.

In the diagram, the arm P is shown rotating clockwise and carrying the pinions B and C with it. Since the teeth of pinion B are in mesh with the teeth of the stationary sun gear A and planet pinions B and C are keyed together, the latter two pinions revolve together in a clockwise direction as they rotate or are carried around the fixed sun gear A by pulley P. As the teeth of pinion C are in mesh with those of sun gear D, the latter gear will be driven in either a clockwise or counter-clockwise direction, as determined by the epicyclic gearing ratio formula given in the fourteenth edition of MACHINERY'S HANDBOOK, page 844, Fig. 17. Applying this formula, with the correct reference letter for each of the gears and pinions, we have the ratio

$R = 1 - \frac{A}{B} \times \frac{C}{D} =$ ratio of gear reduction or fraction of a turn imparted to sun gear D by one revolution of the pulley or driver arm P in which

$P =$ the pulley or driver arm;
$R =$ ratio of gear reduction;
$A =$ fixed sun gear with 47 teeth;
$B =$ planet pinion with 20 teeth;
$C =$ planet pinion with 21 teeth; and
$D =$ driven sun gear with 49 teeth.

Substituting numerical values (number of teeth in each gear) in the formula

$$R = 1 - \frac{A}{B} \times \frac{C}{D}$$

we have $R = 1 - \frac{47}{20} \times \frac{21}{49} = 1 - \frac{141}{140} = -\frac{1}{140}$

Therefore, one revolution of the pulley or driver arm P will turn the driven sun gear D 1/140 of a revolution. The direction of rotation will be counter-clockwise, or opposite that of the driver P, as indicated by the minus or negative sign which precedes the final result. Thus pulley P must make 140 revolutions for one complete revolution of sun gear D.

Now, since pinion E, which has 24 teeth, is keyed to sun gear D and drives gear F, which has 58 teeth, the number of revolutions of pulley P required to obtain one revolution of gear F, or the camshaft S, is obtained by the equation

$$140 \times \frac{58}{24} = 338\ 1/3$$

The direction of rotation of the camshaft is clockwise, or the same as that of the driving pulley P, as will be seen in Fig. 4.

It is interesting to note that a reduction ratio of 10,000 to 1 can be obtained with only the four gears A, B, C, and D if they are made with 101, 100, 99, and 100 teeth, respectively.

Quick-Change Two-Speed Belt Drive.—The two-speed belt drive shown in Fig. 5 was developed for use on a certain machine. The design provides two speeds without the usual requirement of shifting or moving belts. Two sets of pulleys, A and B, fastened to the driving shaft C and the driven shaft D, respectively, by means of pins, are employed.

Belts E and F are applied to their pulleys rather loosely. The rocker arms G and H, pivoted on the bearings J and K, are provided with two idler wheels L and M. Spindles N and O serve to carry the idler wheels and tie the two rocker arms together into a single unit. The bar P provides a means of connecting the rocker unit to a suitable actuating mechanism.

The operation of the drive is very simple. In the particular application illustrated, a very slow starting speed is required. After the driven pulley has reached the starting speed, it is immediately accelerated by shifting the drive to the next pulley. In other words, the rocker unit is placed in position 1 for starting, and is finally shifted to position 2. In thus operating the drive, the tension is first placed on belt E for slow speed and then shifted to belt F to obtain the high-speed drive.

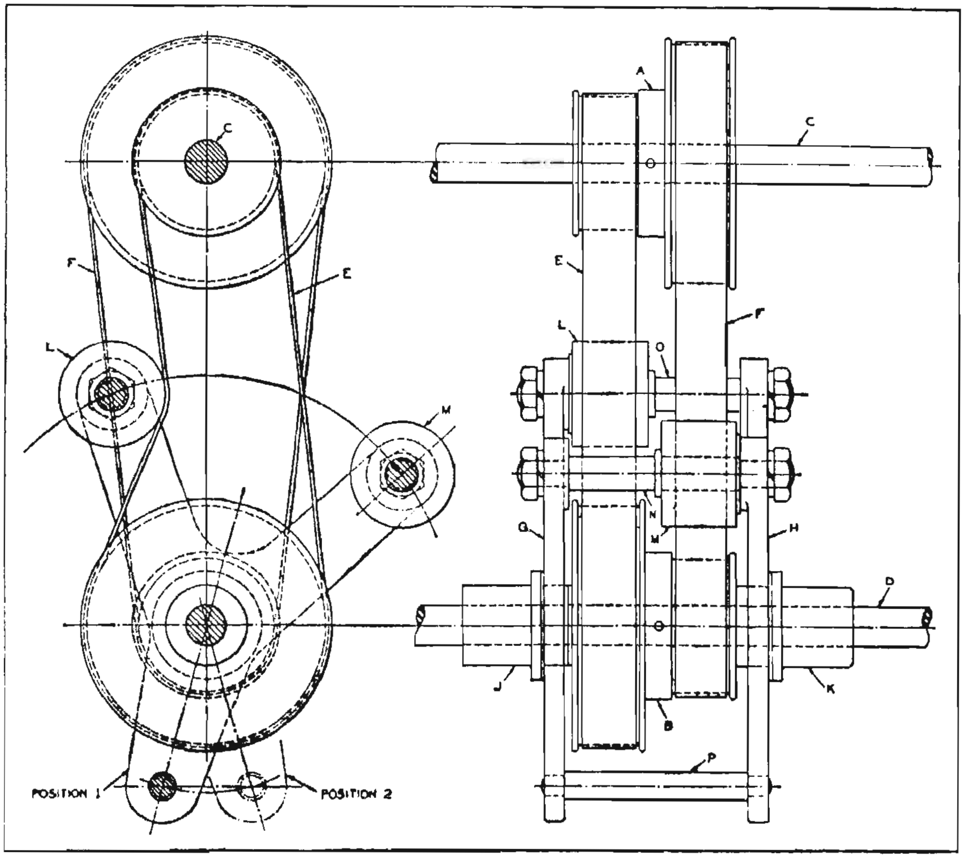


Fig. 5. Two-speed Quick-change Belt Drive.

In addition to providing a very easy means of varying speeds, this drive also permits placing the equipment at a remote place and controlling the speed through suitable linkage to the bar P.

Friction-Drive Mechanism Designed for Stepless Speed Variation.—Speed variation without steps can be obtained by sliding a belt along opposed tapering driving and driven drums; by a V-belt, steel ring, or chain driving between pairs of adjustable conical disks which permit varying the effective diameters; or by friction disks at right angles.

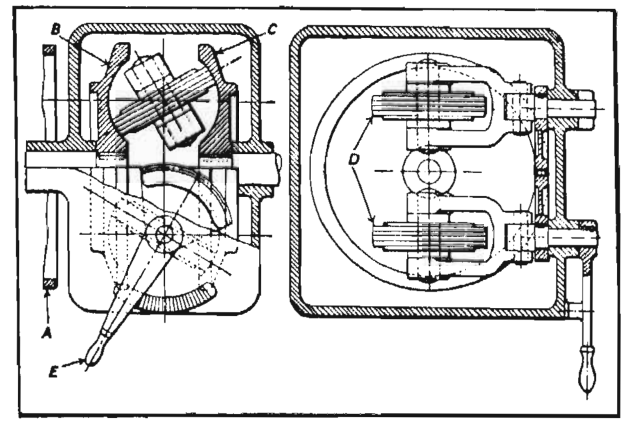


Fig. 6. Friction-drive Mechanism with Stepless Speed Changes Obtained by Moving Hand Lever.

The single-lever control friction drive mechanism shown in Fig. 6 is designed to give stepless speed variation between a minimum and maximum speed.

While heavy power transmission is not to be expected of friction drives, the arrangement shown possesses the advantage of a double drive. Driving pulley A is keyed to the duplex friction bowl B and transmits motion to member C by means of the two intermediate disks D. The disks are mounted in swiveling forks attached to gear segments operated by a single lever E. With the disks in the horizontal position, as shown in the right-hand diagram, the friction bowls run at equal speeds. A plus or minus movement of 30 degrees provides a stepless variable-speed range in the ratio of 7 to 1.

Variable-Speed Transmission.—The variable-speed drive shown diagrammatically in Fig. 7 was designed to replace the cone-and-belt type of drive employed on certain textile machines. It can, however, be applied to machines in other industries where a sensitive control is required which will respond instantly to a very slight variation in speed.

The main shaft *A* is driven at a constant speed. The shaft *B*, which provides the desired intermediate speeds, is cast integral with a sleeve *L*. Lever *V* is keyed to the quadrant shaft *U* and is located outside of the mechanism housing. This lever is connected to the "evener motion" which controls the drive, causing it to increase or decrease the speed of shaft *B* as required. The position of the quadrant *Q* indicated in the diagram shows that shaft *B* is being driven at nearly its maximum speed. When the small hardened-steel friction-driven roller *K* is almost at the center of the leather-faced disk *R*, gear *I* on shaft *J* and gear *H* will be almost at a standstill. Gear *G*, being integral with *H*, will also be practically at a standstill.

When gear *G* remains in a fixed position, shaft *B* is driven at its highest speed. Thus, if quadrant *Q* moves the shifting rod *P* and, consequently, the small friction roller *K*, toward the periphery of disk *R*, so that the speed of gear *G* is one-fifth greater than the main shaft *A*, shaft *B* will be at a standstill. From this it will be clear that a slight movement of quadrant *Q* will make a great difference in the speed of gear *G*. The spring *O* insures driving contact of roller *K* and disk *R*. The pressure exerted by spring *O* is just sufficient to prevent slippage and yet allow roller *K* to slide on shaft *M*, with its key sliding in keyway *W*. The main shaft does all the work and the small roller and disk serve as governors only. The gears to the left of wall *Z* are enclosed.

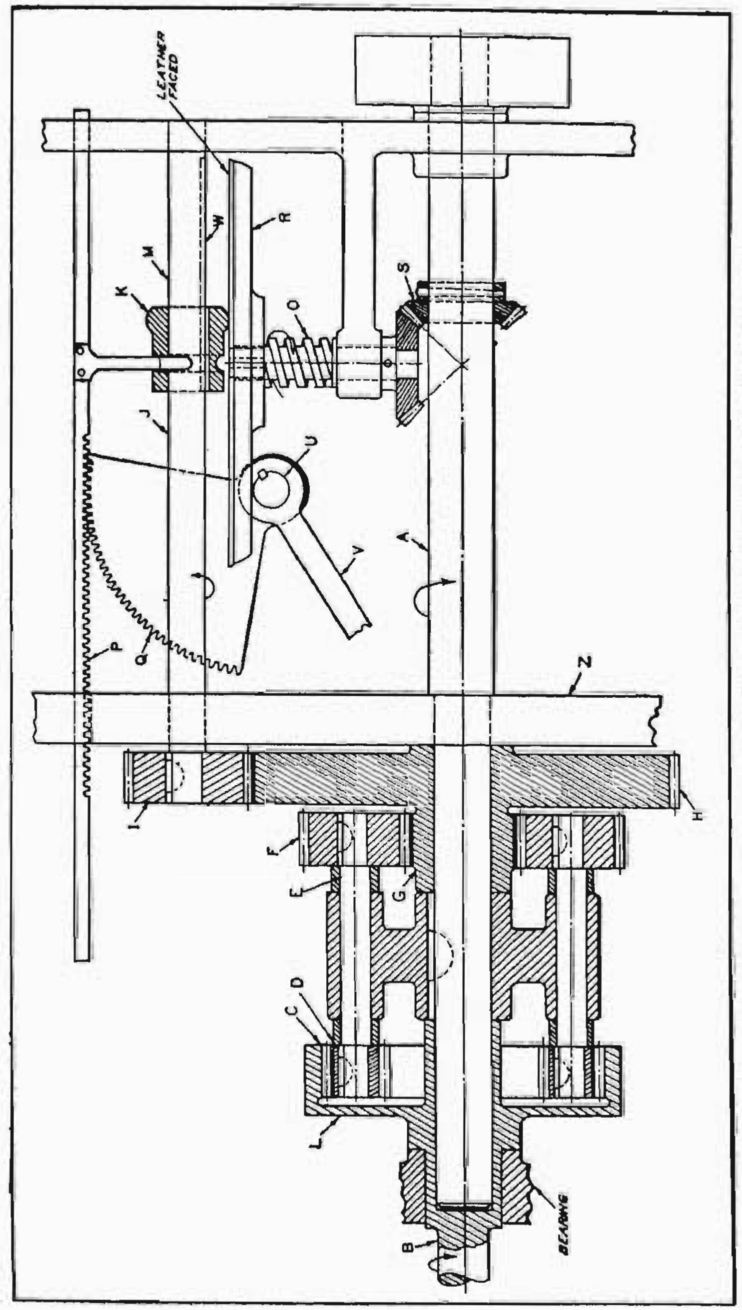


Fig. 7. Mechanism for Varying Speed of Driven Shaft Using Rack Controlled Friction Drive.

Cam-Controlled Variable-Speed Drive.— By properly proportioning the four gears and cam that constitute the important parts of the drive shown in Fig. 8, it can be caused to give any of a great variety of speed actions, the

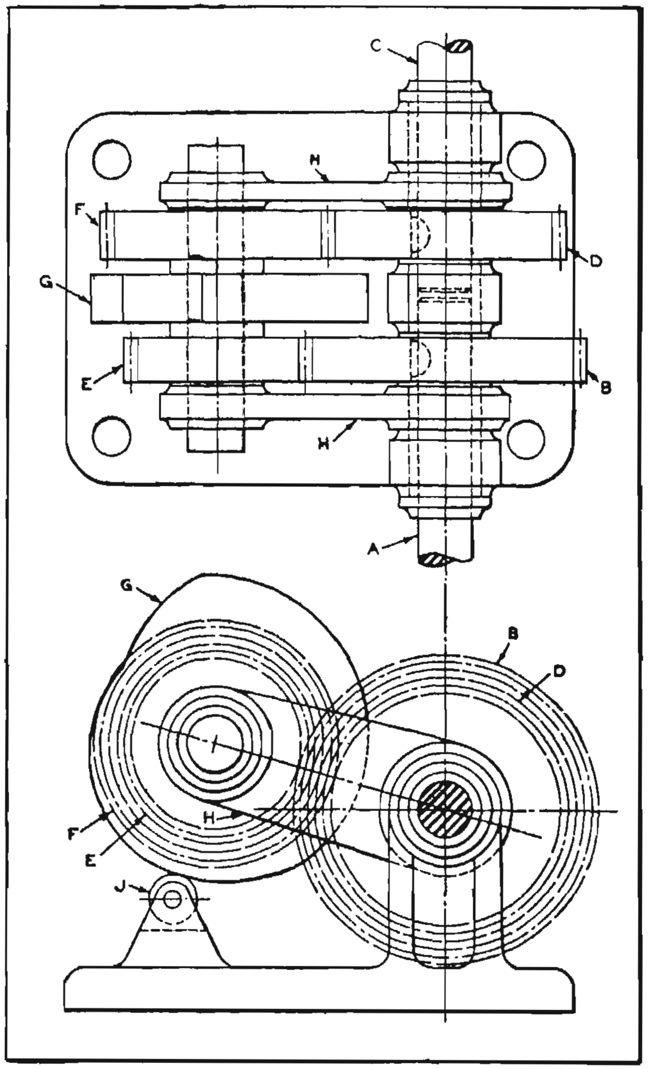


Fig. 8. Variable-speed Drive from Shaft A to Shaft C with Variation in Speed Obtained by Cam G.

simplest of which may be described as an intermittent or stop motion. When designed as a stop motion, it can theoretically perform this action in any one of a variety of ways. The ratio of the period of stop and the period of motion may be of any arbitrarily chosen value.

The number of stops per revolution may be either fractional or integral. To some degree, the stop intervals may have a non-uniform or non-periodic sequence. The action, instead of being the usual total arresting of motion, may also consist of merely a slowing down of the speed or it may be even more than a total stop—that is, it may be a momentary reversal of the motion.

Fig. 8 shows a simple but general arrangement of the device. The drive-shaft is shown at *A* and the driven shaft at *C*. Gear *B*, keyed to shaft *A*, drives gear *D*, keyed to shaft *C*, by means of back-gears *E* and *F*. At *G* is a cam that revolves as a unit with gears *E* and *F*. The frame *H* of the back-gear is a rigid unit, but pivots or is free to oscillate about the common axis of shafts *A* and *C*. The frame *H*, with its gears and cam, rests on the cam-roller *J*.

The gears *B*, *E*, *F*, and *D* constitute a train that may have any desired driving ratio within certain limits other than unity; that is, the ratio of *B* to *E* must not be the same as *D* to *F*. The action of the mechanism is as follows: The cam *G* rotates with the gears *E* and *F*, and as it rides on the roller *J*, its undulating contour gives a rocking motion to the frame *H*, which impresses a motion on the rest of the mechanism, in addition to that given by the driving shaft *A*. Thus if shaft *A* is held stationary, a movement of *H* will cause *C* and *D* to turn in one direction or the other, since the train value is not unity.

On the other hand, if cam *G* is lifted free of the roller *J*, and *H* is held stationary, the only motion that *A* can impart to *C* will be that transmitted by the gear train *B*, *E*, *F*, and *D*. Thus the motion of *C* due to that of *H* can be combined with that due to *A*, either positively or negatively, with the consequence that the motion of C results at one time from the sum of the two motions of A and H, and, at another time, from the difference of the two motions. By giving the proper value to the gear train B, E, F and D, and the proper contour to the cam G, any one of the actions mentioned can be obtained.

Mechanism for Starting, Stopping, Changing Speed, and Reversing Output Shaft.—A mechanism with two driving motors which permits starting, stopping, changing speed, and reversing the output shaft without stopping the motors is shown in Fig. 9. This control over the driven shaft is obtained by means of differential gearing without the use of friction clutches, gear shifts, or other well-known speed-changing and stopping devices, and is accomplished by simply changing the speeds of the two driving motors. This provides a stepless variation in driven shaft speed.

Each of the four bevel gears A, C, D, and F have the same number of teeth. Gear C is keyed to the motor shaft M, and gear A is keyed to the output shaft B. Both of these shafts extend to the center of the cross-piece G, which is cast integral with the spur gear H. The cross-piece provides a working bearing for gear H, as well as a support for the shafts of bevel gears D and F. Pinion E is keyed to the motor shaft N. The gears E and H are in the ratio of 2 to 1. The arrows on the faces of these gears indicate the direction of rotation.

Four tables of speeds, not shown, are used in operating the mechanism. These tables indicate the speed and direction of rotation of the gears E, C, H and A. The tables show that E, C, and H revolve in one direction continuously. The tables give the speed at which each motor must be operated to give the output shaft B any forward or reverse speed from 0 to 320 revolutions per minute in steps of 20 revolutions per minute, with the motor speeds ranging from 340 to 660 revolutions per minute. Of course these speeds may be increased.

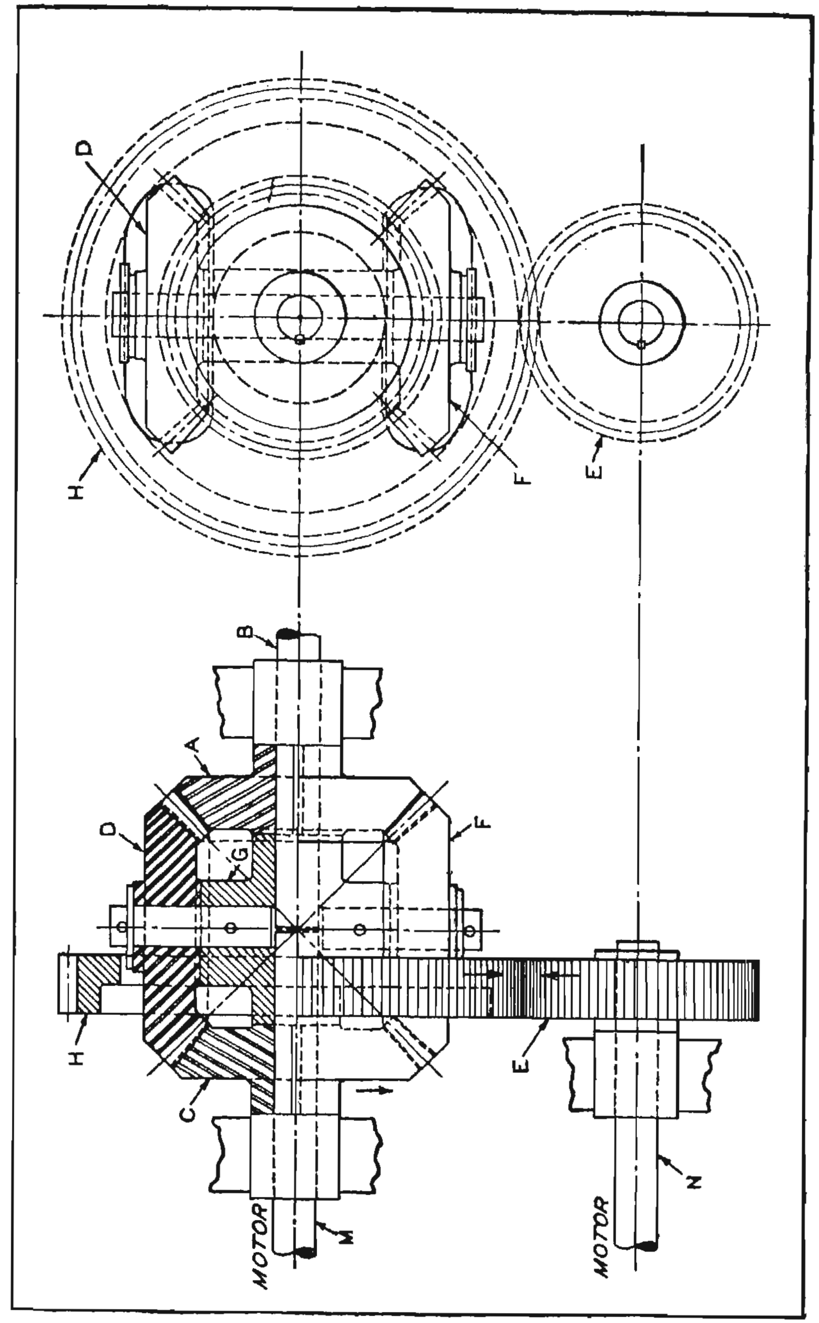


Fig. 9. Mechanism for Controlling Driven Shaft by Changing Speeds of Two Driving Motors.

With both motors operating at the same speed, bevel gear *A* and output shaft *B* remain stationary. In order to rotate the output shaft, the speed of one of the motors must be increased or decreased, depending on the direction in which the driven shaft is to be rotated.

To put the mechanism in operation, the motors are started simultaneously and their speed advanced in synchronism to 500 revolutions per minute, for example, without imparting motion to shaft *B*. Then, on decreasing the speed of the motor connected to shaft *N* 10 revolutions per minute, and increasing the speed of the motor connected to shaft *M* 10 revolutions per minute, for example, the output shaft *B* will be driven at a speed of 20 revolutions per minute in the same direction as shaft *N*.

Or, similarly, if the speed of the motor connected to shaft *M* only is increased, it will cause an equal ratio of speed increase in the output shaft *B*, but will make it revolve in the opposite direction to that of the motor shaft *M*, and if the speed of the motor shaft *N* only is increased, it will cause a like ratio of increase in the output shaft *B*, but in the opposite direction to that of the motor shaft *N*. While the tables show the speeds of the driven shaft obtained by increasing or decreasing the speed of the motors by steps of 20 revolutions per minute, the changes in speed are actually stepless and accomplished by a smooth acceleration or deceleration.

Mechanism for Varying Speed and Direction of Shaft Rotation.—A mechanism for controlling the speed or direction of rotation of the wheel or shaft of a machine is shown in Fig. 10. The worm *A* which drives the worm-wheel *B* is a sliding fit on the splined shaft *C* which rotates at a constant speed. The axial movement of worm *A* is controlled by the combined action of the yoke *D*, which is free to slide on the rectangular bar *E*, and the swinging cam-lever *F*, together with its actuating cam *G*.

It will be seen that the cam groove can be designed to give

the worm-wheel any desired movement, as to speed and direction of rotation, within certain limits. For instance, if the worm is moved axially in the direction its rotation would normally drive the worm-wheel, the speed of the worm-wheel will be increased. If the cam groove causes the worm to move axially in the opposite direction at a speed equivalent to the rotational speed of the worm-wheel at the pitch line, the worm-wheel will stop. By increasing the axial speed of the worm in the reverse direction, the worm-wheel will rotate in the reverse direction. Thus, with the combined movements of the worm and wheel, and the rack and gear action of this mechanism, it is possible to obtain variations in speed and reversal of the direction of rotation.

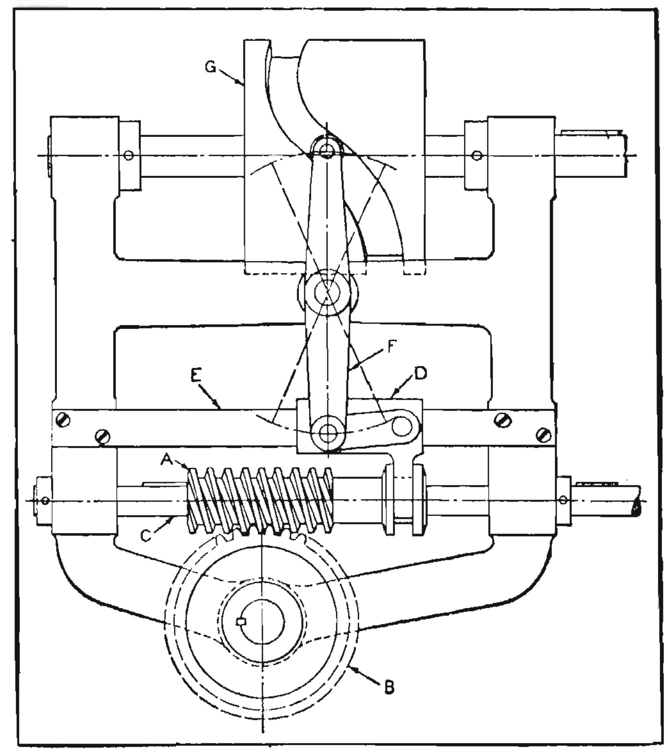


Fig. 10. Mechanism for Varying Speed and Direction of Shaft Rotation.

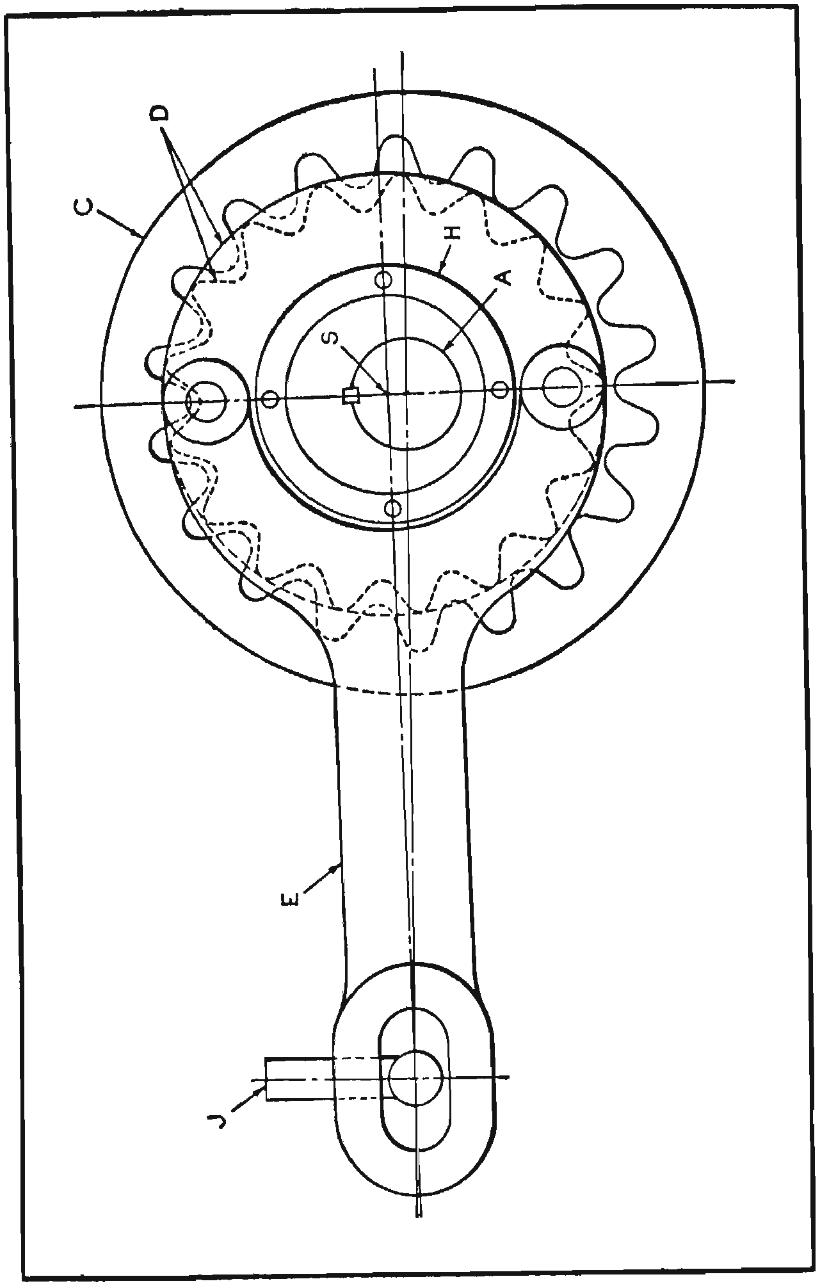


Fig. 11. Wabble Gear, Pinion, Strap and Locator of Mechanism Shown in Fig. 12.

Speed Reducer with Wabble-Gear Mechanism.—A wabble gear, cast in a mold made from a pattern originally designed for use in the production of gearing for boiler grates more than seventy years ago, performs a major function in the rugged, 18-to-1 ratio speed reducer shown in Figs. 11 and 12. Although this mechanism may appear

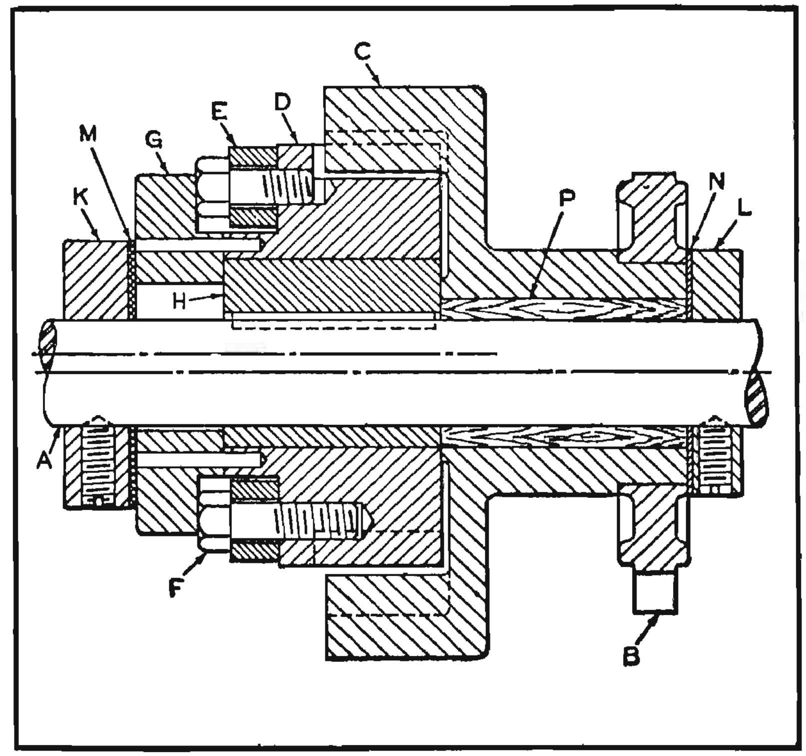


Fig. 12. Wabble-gear Speed-reduction Mechanism Used to Drive Tumbling Barrel.

somewhat crude, it has given excellent service for many years as a drive for tumbling barrels and similar equipment. It has the advantage of being comparatively simple and inexpensive to construct. The jack-shaft *A*, which runs at a speed of 180 R.P.M., drives the link-belt sprocket *B*,

Fig. 12, at a reduced speed of 10 R.P.M., and further speed reduction can be obtained by selecting sprocket *B* and its driven sprocket to give the desired speed, which in some cases may be as low as 2 R.P.M.

The tooth profiles of the wabble gear *C* and the cast wabble pinion *D* developed to operate with it are shown in Fig. 13.

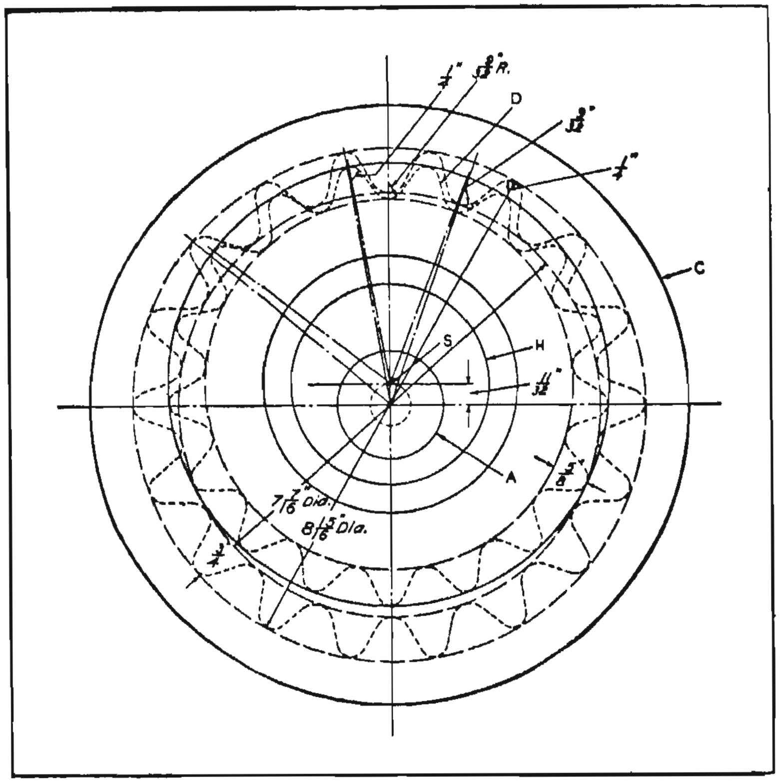


Fig. 13 Wabble-gear and Eccentrically Mounted Pinion of 18-to-1 Ratio Speed-reduction Mechanism

The strap *E* which is secured to pinion *D* by cap-screws *F*, as shown in Fig. 12, is also a casting. The retaining collar *G* serves to keep the eccentric *H* in place and to prevent the screws *F* from working loose. The eccentric *H* is keyed to the jack-shaft *A*. The wabble lever locator *J* is positioned

in the slot of wabble lever *E*. As indicated in Fig. 11, the end of locator *J* has 1/8 inch clearance or play on both sides of the slot in wabble lever *E* when it is in the center-line position.

Jack-shaft *A*, Fig. 12, is 1 15/16 inches in diameter, and the reduction mechanism is mounted as near to one end of the shaft as possible to facilitate assembly. Collars *K* and *L* are made light drive fits on shaft *A* and are tapped to give a tight fit for the retaining set-screws. Fiber washers *M* and *N* and the Arguto oilless bearing *P* give good service over a long period of time.

Referring to Figs. 11 and 12, it will be obvious that when shaft *A* is revolved, the eccentric *H* keyed to it will cause the center *S* of the wabble pinion *D* to follow a circular path about the center of shaft *A*. Since the wabble lever *E* prevents the wabble pinion gear *D* from revolving about its axis, the circular motion imparted to pinion *D* serves to revolve the wabble gear *C* through the equivalent of one tooth space for each revolution of eccentric *H*. As there are eighteen teeth in wabble gear *C*, shaft *A* will make eighteen revolutions to one revolution of the sprocket *B* attached to the wabble gear.

Automatic Back-Gear Shifter.—A device for automatically shifting a back-gear in and out as the torsional resistance of a shaft varies is shown in Fig. 14. This mechanism is a part of an automatic drill-press feeding device. Shaft *Q* is connected to the drill-press feed-spindle through an adjustable slip clutch (not shown), and motor-shaft *A* is the source of power. The automatic shifting of the back-gear permits the use of a small motor to deliver the high thrust necessary at the drill point for drilling large holes, and at the same time permits rapid traverse of the drill to and from the work. The back-gear is automatically disengaged for drilling small holes.

Hub *B* is keyed to input shaft *A*. Cantilever leaf springs *C*, attached to hub *B*, are under sufficient initial strain to transmit full torque through spherical-ended pins *D* without slippage. Collar *E*, gear *H*, and worm *N* are pinned to shaft *F*. Cluster gear *J*, which is supported in frame *M*, is in constant mesh with gear *H* and idler gear *G*. Worm-gear *P* is pinned to shaft *Q*. The thrust collar *K* floats on shaft *F* and supports the compression spring *L*.

At the start of the cycle, input shaft *A* drives shaft *F* at the same speed through the gripping action of pins *D* on collar *E*. During this time, gear *G*, which is a sliding fit on

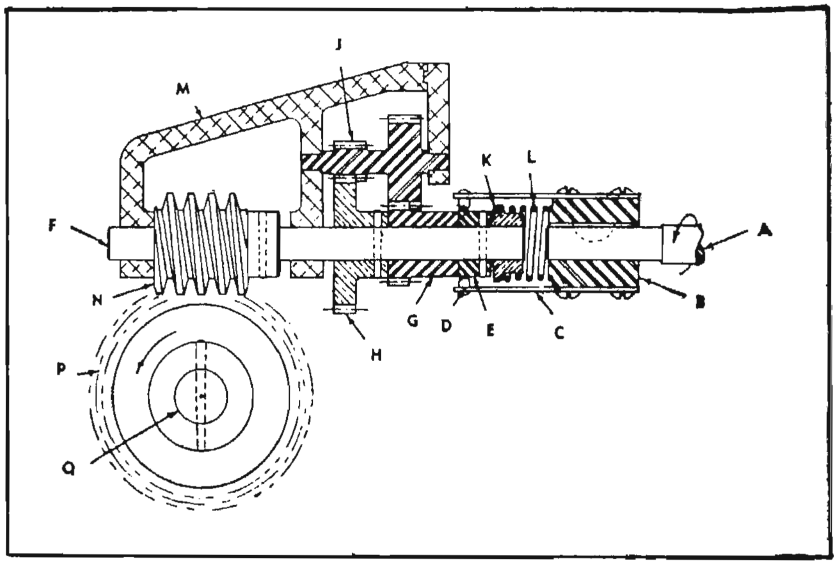


Fig. 14. Back-gear J Is Automatically Engaged or Disengaged as Torsional Resistance of Shaft Q Increases or Decreases.

shaft *F*, is idling at high speed. When the reaction of worm *N* to the torsional resistance of shaft *Q* builds up sufficiently to overcome the load exerted by spring *L*, shaft *F* will shift axially to the right. Thus pins *D* will transfer their torque from collar *E* to the hub of idler gear *G*, thereby transferring the drive through the cluster gear *J*, or back-gear, to shaft *F*. The back-gear is automatically disengaged upon the reversal of the input shaft *A*.

On this particular mechanism, the input shaft delivers 1/12 H.P. at 1500 R.P.M. Gears having 24 pitch furnish a 4 1/2 to 1 reduction through the back-gears. Pins *D*, collar *E*, and the hub of gear *G* are hardened. The initial load on spring *L* must be set lower than the minimum worm reaction to prevent pins *D* from dwelling at the parting line of collar *E* and gear *G*.

Mechanism for Changing Cam Speeds Independently of Camshaft Speeds.—A mechanism for reducing the rotary speed of a cam without altering the speed of the shaft on which the cam is mounted is shown in Fig. 15. This

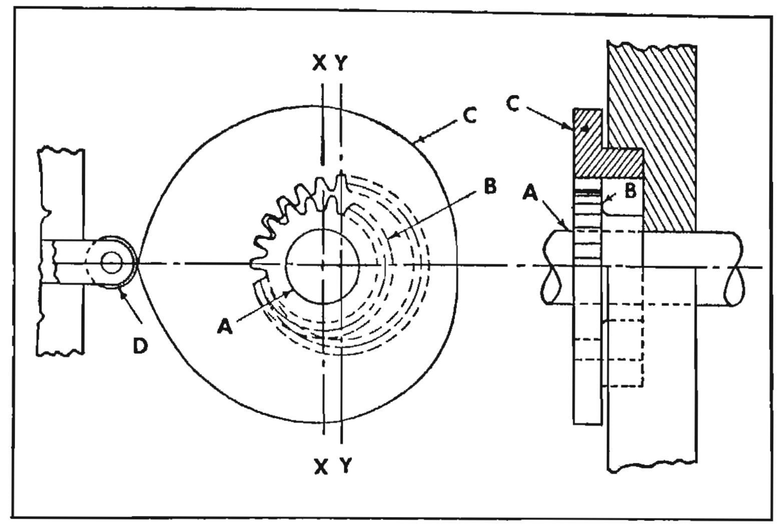


Fig. 15. Internal Spur Gear Teeth Cut in Cam Permits Reduction in Cam Speed without Changing Camshaft Speeds.

mechanism was applied to a machine for winding flat wire on spools. The wire-guiding mechanism was required to be operated by a uniform-motion cam, maintaining a definite rate of travel over a specific distance.

Because of a reduction in the width of the flat wire being spooled, it became necessary to decrease the rate of travel of the guiding mechanism by approximately one-third without altering the distance traversed or the speed of the cam-shaft, which operated other mechanisms.

To accomplish the required difference in rotary speed between the cam and its shaft, gear teeth were machined in the bore of the cam C to engage a spur gear B, which was assembled to the shaft A. The ratio of internal gear teeth to the spur gear teeth was made approximately 3 to 2. Thus the cam, which rotates about center line $Y—Y$, revolves at two-thirds the speed of the shaft, which rotates about the original center line $X—X$. In this way, the follower roll D, which actuates the wire-guiding mechanism, is driven at one-third the speed of shaft A.

CHAPTER 13

Speed Regulating Mechanisms

Machines which wind material such as paper, cloth or metal strip on spools or reels or which form or twist wire may require a synchronous rotation of two shafts with or without an occasional momentary acceleration or retardation of one shaft with respect to the other. In other machines the speed of the driven shaft must be maintained within close limits. The mechanisms described in this chapter have been designed to perform such special speed controlling functions. They include a hand-controlled arrangement for maintaining constant speed of a pull-roll unit; a friction-driving device which maintains web tension of rolled material within close limits; an automatic speed control for providing constant cutting speed on a lathe; a hand-operated mechanism for advancing a gear driven shaft while it is operating; a mechanism for insuring synchronous operation of two hand-operated shafts; a mechanism for obtaining a special cycle of speed relationships between two shafts; a mechanism for varying the pitch and twist in wire twisting machines; a mechanism which is used with a fluid power variable speed drive to maintain speed within 1/4 per cent of desired values.

Constant-Speed Pull-Roll for Winding Metal Strips.—Certain classes of metal strip material, after being put through various cleaning processes, are wound on steel spools mounted on a revolving shaft. The diameter of the spool hub, on which the metal is wound, is usually 6 inches. The spools were originally driven at a speed of 100 revolutions per minute. With this arrangement, the surface speed at the beginning of the winding operation was about 150 feet per minute. As the coil of metal became larger diametrically, the surface speed increased proportionately. The result was that by the time the full length of metal had been

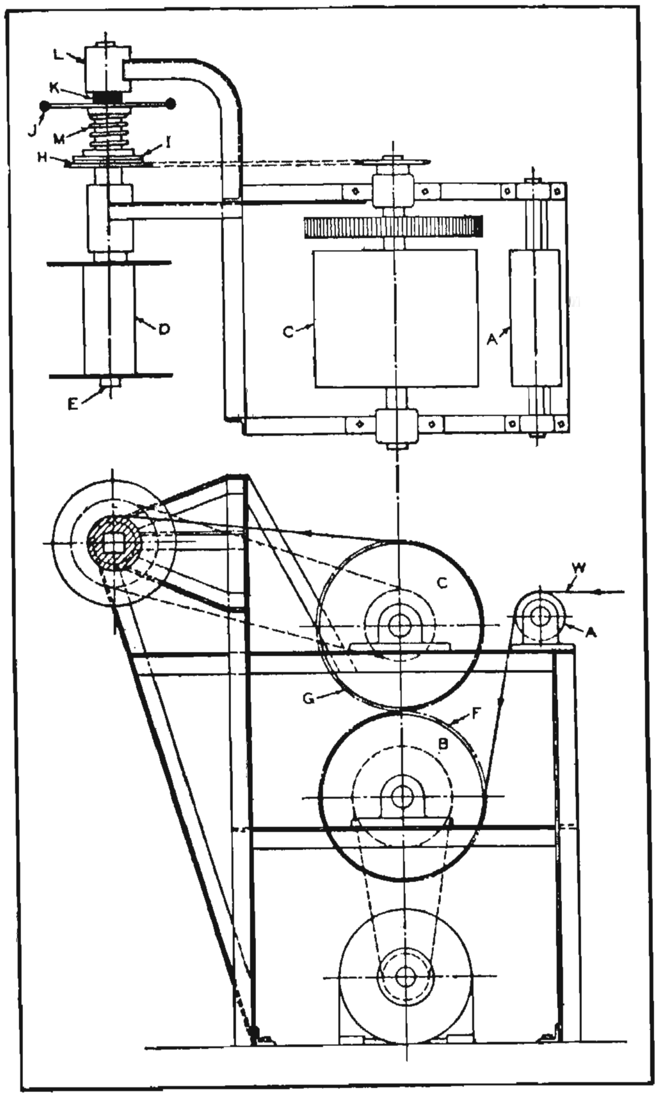


Fig. 1. Constant-speed Rolls for Pulling Metal Strip through Processing Machine and Winding it on Metal Spool.

wound around the spool, the surface speed had been increased about 100 per cent. This increase in speed resulted in scratched stock, which frequently had to be rejected.

As the processing operations required certain speeds for best results, it was necessary to maintain a constant winding speed. To meet these requirements and to overcome previous difficulties, the constant-speed pull-roll unit shown in Fig. 1 was built. This unit consists primarily of two large rolls *B* and *C*, driven at a constant speed and mounted one above the other on suitable bearings in an all-welded frame of standard structural steel. This unit also contains a square-ended revolving shaft *E*, mounted in bearings located on the outside of the main frame. The steel winding spool *D* slips over shaft *E*, the square end of which drives the spool.

As the metal strip *W* comes from the processing machine, it passes over the guide roll *A*. From roll *A* it passes under and part way around the pull-roll *B*, thence upward and half way around the pull-roll *C* to the winding spool *D* on the shaft *E*. From the illustration it will be noted that there is a small space between the pull-rolls. This space is necessary to allow the joints of the material to pass between the pull-rolls.

The unit is driven by a chain from the motor located below the pull-rolls, spur gears *F* and *G* transmitting power from one roll to the other. From pull-roll *C* a chain drives the winding spool *D* through friction disks *H* and *I*. More or less friction is obtained by turning handwheel *J* on the threaded bronze sleeve *K* which fits over the main shaft and is fastened to the rear bearing *L*.

As handwheel *J* is moved either clockwise or counterclockwise, the compression spring *M* increases or decreases the pressure on the friction disk *I*, so that the desired tension is maintained on the metal being wound. Thus the speed at which the metal is wound on the spool remains constant no matter what the diameter of the coil may be, the slipping that occurs between the friction disks compensating for the difference in speed. The rolls *B* and *C* do all the pulling of the metal through the processing machine and they also regulate the speed of the machine and govern the winding spool speed.

Friction Driving Mechanism for Rewinding Roll.—A friction driving mechanism that has been found practical as a means for driving a roll used in rewinding a paper or cloth web is shown in the accompanying illustration. This mechanism was designed specifically for rewinding cloth on a dyeing pad. Modern printing presses and paper-coating machines are, of course, equipped with an elaborately designed unit for automatically controlling the web tension and compensating for the constantly increasing diameter of the roll of rewind material. For a small proving press or laboratory model, however, the use of a simple and flexible friction driving unit is desirable.

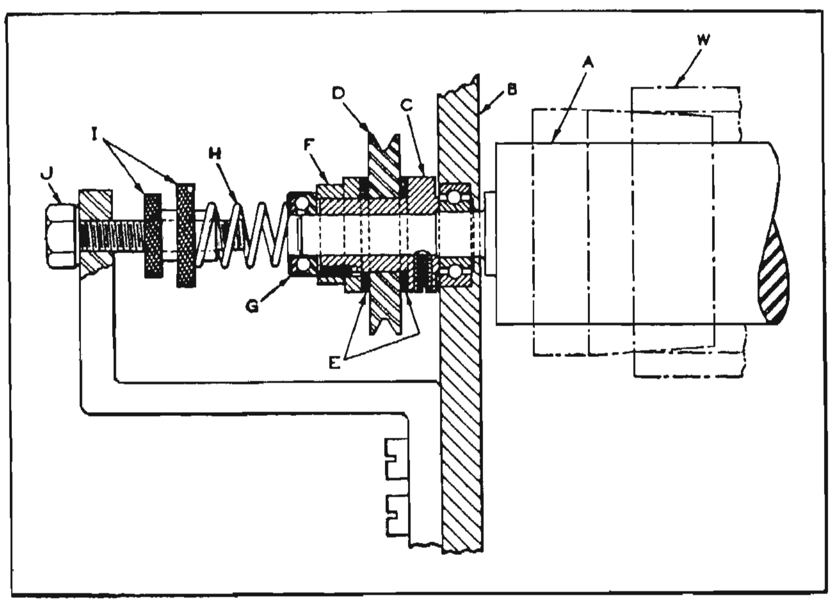


Fig. 2. Rewinding Roll Equipped with Friction Driving Mechanism.

In this case, the rewind unit is not required to pull the web through the operating rolls. It is only necessary for the unit to keep the material under tension within limits that will prevent wrinkling. Owing to the light amount of power needed, the friction adjustment must be very sensitive.

Referring to Fig. 2, the speed of the driven pulley *D* must be so adjusted that it is approximately 2 per cent higher than the minimum web speed when starting at the small diameter of the rewind core *W*. Pulley *D* is a running fit over bronze hub *C*, which is secured to the extension shank of the rewind roll *A* by set-screws or dowels. Two leather washers *E* on each side of the pulley provide the necessary friction for driving the roll when sufficient pressure of spring *H* is applied to collar *F*. All bearings are of the anti-friction type, including the roll-supporting bearing in the frame *B*.

A feature of this rewind mechanism is the spring pressure adjustment provided by stud *J* and nuts *I*, which can be easily changed while the machine is in operation. The thrust ball bearing *G* is preferred to an ordinary thrust washer because the face of the latter, revolving against the end of spring *H*, would reduce the sensitivity of the mechanism.

Automatic Variable-Speed Control Maintains Constant Cutting Speed when Facing Disks.—An automatic speed control applied to a Gisholt Simplimatic lathe in the machine shop of the Reeves Pulley Co., Columbus, Ind., makes it possible to maintain a constant cutting speed when facing disks. In other words, the number of revolutions per minute of the work decreases as the tool moves toward the circumference of the disk.

The machine is used for turning and facing disks in sizes up to 20 inches in diameter. The drive consists of a motorized mechanical variable-speed transmission with mechanical automatic speed control, arranged so as to provide a

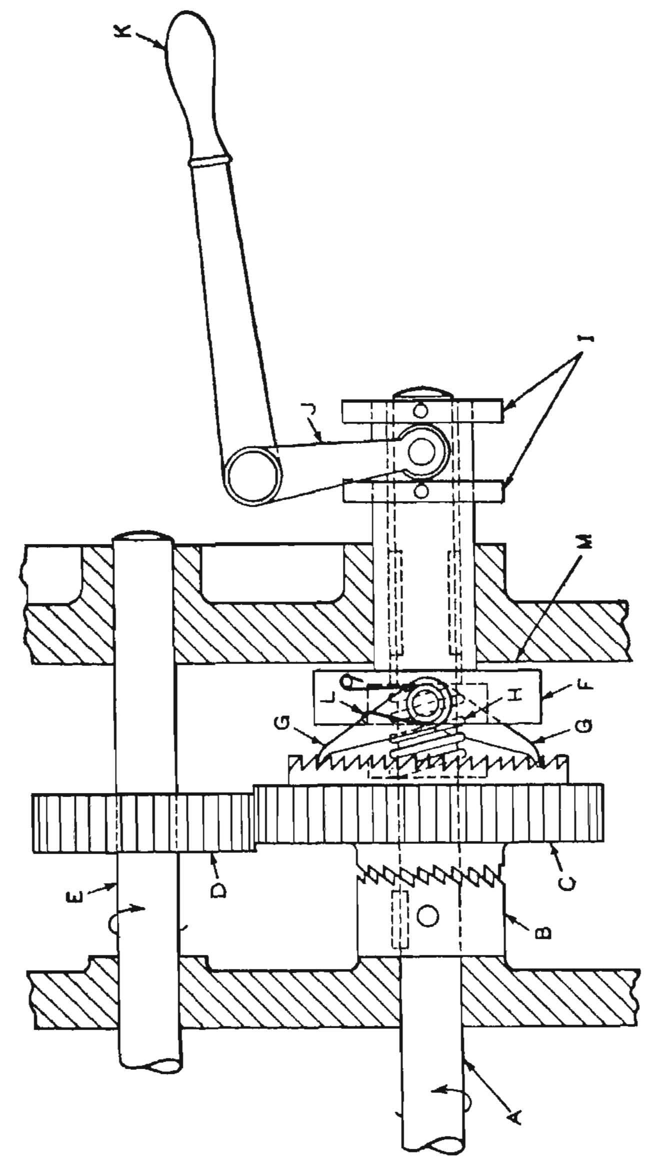


Fig. 3. Mechanism which Permits Driven Shaft E to be Advanced Ahead of Driving Shaft A by Means of Hand-lever K.

uniform cutting speed as the tool moves from the smallest diameter at the hub to the largest diameter at the circumference.

The speed control device is connected by a cable to the cross-feed of the lathe. As the cutting tool moves outward from the hub of the disk being faced, its movement is transmitted through the cable to the indicating lever of the mechanical variable-speed device, which, by means of a chain drive, transmits power to the speed control unit. The latter, in turn, is connected through a reducer and coupling to the drive-shaft of the lathe. Thus, as the cutting tool moves out, the motorized variable-speed transmission drive unit is automatically regulated to reduce the number of revolutions per minute of the disk, thereby maintaining a uniform cutting speed.

Lever-Operated Mechanism for Advancing Gear-Driven Shaft.— In operating a wire-forming machine, it is necessary to maintain a certain amount of slack in the wire as it passes through the machine. At times, the slack is taken up as a result of uncontrollable conditions and must be increased while the machine is in operation. Fig. 3 shows the construction of a mechanism by means of which the amount of slack is manually controlled.

The shaft *A*, rotating in the direction indicated by the arrow, normally drives shaft *E* through the gears *C* and *D*, the ratchet teeth cut on the side of the collar *B*, which is keyed to shaft *A*, being engaged with similar teeth cut on the hub of gear *C*. Spring *H* serves to maintain contact between the ratchet teeth, so that gear *C* is positively driven. Fig. 3, however, shows the ratchet teeth partly disengaged.

The sleeve *F* is splined to shaft *A* and rotates with it, but is capable of axial movement, being moved to the left, as shown, when pressure is applied to the left-hand collar *I* by the handle *K* through the fork-shaped lever *J*. The spring *H* normally holds sleeve *F* in contact with the frame at *M*. Two pawls *G* are carried on sleeve *F* and are held in contact with the ratchet teeth on the right-hand side of gear *C* by springs *L*.

As sleeve *F* is splined to shaft *A*, it normally rotates with it, the ratchet teeth in collar *B* and gear *C* being engaged and the pawls *G* merely resting on the ratchet teeth on gear *C*. In this position, the pawls *G* do not in any way aid in transmitting motion to gear *C*. Thus the entire assembly on shaft *A* rotates as a unit.

When it is necessary to increase the slack in the wire, the handle *K* is depressed, causing lever *J* to move sleeve *F* to the left, as shown, and the pawls *G* to spread or flatten out similar to toggle levers. As these pawls spread, they exert a turning effect on gear *C*, advancing it relative to collar *B* and causing the ratchet teeth on the hub of gear *C* to ride over those on collar *B*. When gear *C* has been thus advanced a distance equal to one tooth space, spring *H* causes it to slide to the left and re-engage the teeth on collar *B*.

Mechanism for Insuring Synchronous Motion.—On a machine for fabricating a wire product, two shafts, operated by means of hand-levers, actuate a tension mechanism. The shafts are required to operate practically in unison, a narrow range of latitude being allowed. Gearing the shafts together appeared to be the solution, but this method proved unsatisfactory, as, at times, one shaft was driven by the other. resulting in torsional stresses, which were objectionable. In order to insure equal power application and synchronous movement of the two shafts, the arrangement shown in Fig. 4 was devised.

The two gears *A* and *B* are carried on the shafts and mesh with the double rack *C*, which floats between them. Rack *C* carries a serrated slot at its lower end. A serrated pin *D*, fastened in a stationary part of the machine, is located in the slot of rack *C*. Pin *D* is given a minimum amount of clearance in the slot, the clearance being shown somewhat exaggerated. The plate *E*, which serves as a guide for rack *C*, is broken away to show the pin *D*.

In operation, if either of the handles is moved ahead of the other, the action of the gear on that shaft causes the rack *C* to swing on the teeth of the other gear as a fulcrum, so that the serrations in the slot of the rack engage the serrations on pin *D*, thus preventing further movement until the other handle is given a corresponding movement, thus disengaging the slot serrations from pin *D*.

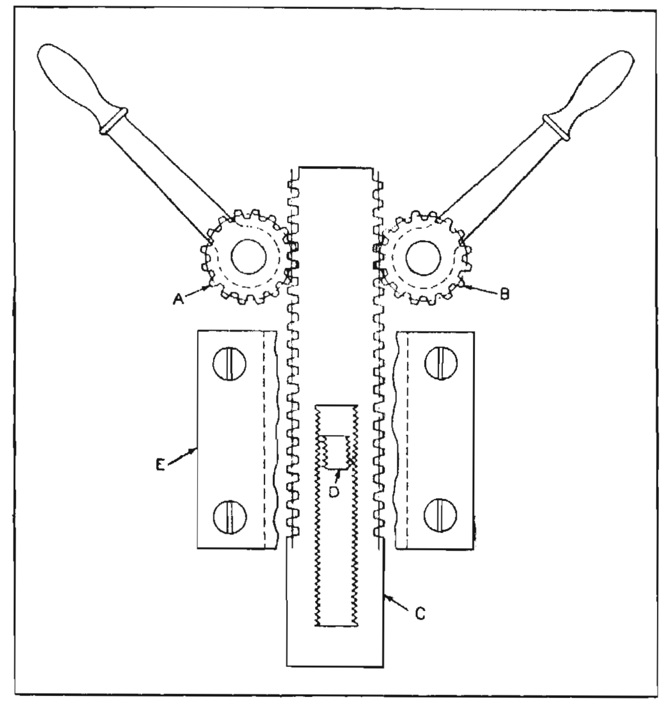


Fig. 4. Mechanism Designed to Insure Synchronous Movement of Two Levers under Equal Torque.

Mechanism for Obtaining Irregular Rotating Movement.—A machine for fabricating a wire product has two spindles that perform twisting operations in synchronism through approximately half of the operating cycle. During the remainder of the cycle, it is necessary for one of the spindles to accelerate its speed of rotation or to advance

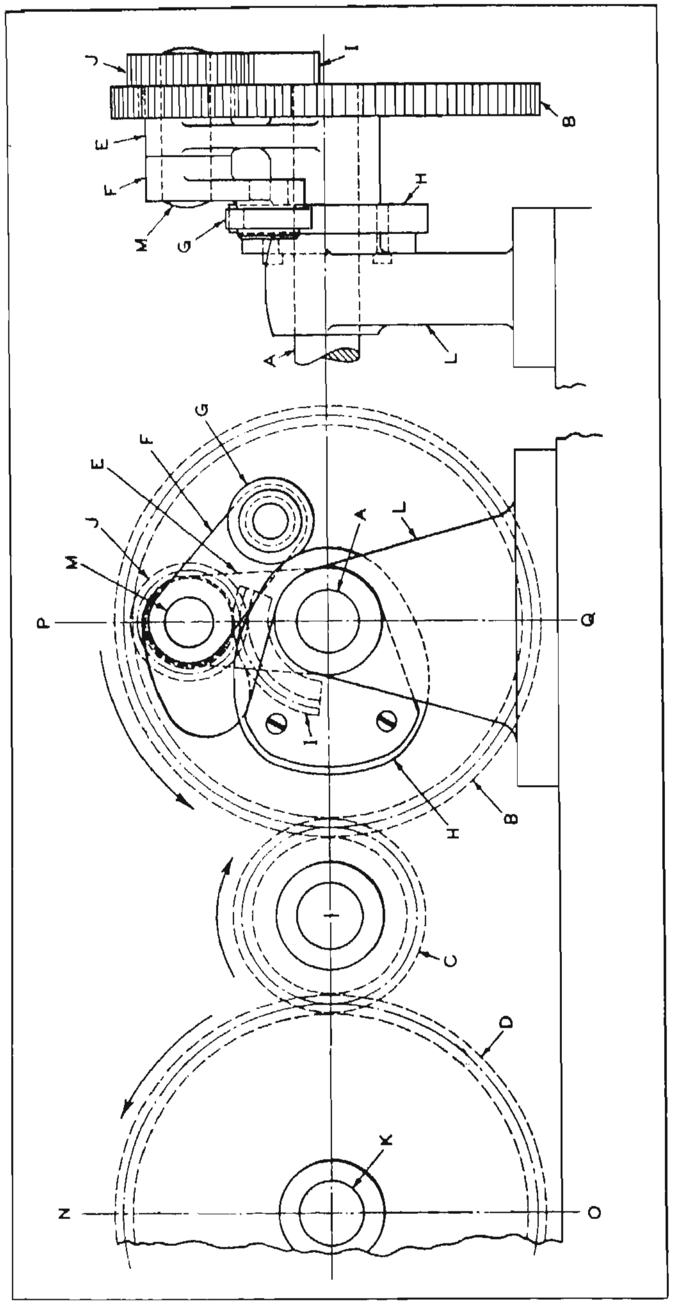


Fig. 5. Mechanism for Producing Irregular Rotation of Shaft A and Constant-speed Rotation of Shaft K from the Driving Gear C.

ahead of the other, returning to synchronous operation after a specified point in the cycle has been passed. The mechanism developed to obtain the necessary movement is shown in Figs. 5 and 6.

Referring to Fig. 5, gears *B* and *D* are driven by pinion *C*, rotating in the direction indicated by the arrows. Gear *D* is keyed to shaft *K*, while gear *B* is free on shaft *A*. Shaft *A* is supported in bearing *L*, which has a flange on one side on which cam *H* is mounted. Lever *E* is keyed to shaft *A*, and carries at its upper end the shaft *M* to which lever *F* and pinion *J* are keyed.

Lever *F* carries roller *G*, which rolls on cam *H*. Pinion *J* meshes with gear segment *I*, which is carried on, and rotates with, gear *B*. The boss on the upper end of lever *E* is extended to pass through the slot in gear *B*, as shown.

In operation, the rotation of pinion *C* is transmitted to gears *B* and *D*, which rotate in synchronism at a uniform rate. As gear *D* is keyed to shaft *K*, the latter rotates with it. Gear *B*, however, not being keyed to shaft *A*, does not transmit motion directly to shaft *A*. As shown in Fig. 5, the extended boss of lever *E* is in contact with one end of the slot in gear *B*; thus the motion of gear *B* is transmitted to shaft *A* through lever *E* in the direction indicated by the arrow, the effect being the same as though shaft *A* were driven directly by gear *B*.

Referring now to Fig. 6, in which bearing *L* is cut away to expose lever *F*, the rotation of gear *B* has caused roller *G* to rise to the high point of cam *H* and begin descending the other side of the cam. When roller *G* begins to ascend the rise of cam *H*, a rotative motion is imparted to pinion *J* through lever *F* and shaft *M*. This rotative motion of pinion *J*, which is in mesh with gear segment *I*, causes a slow rotative motion to be imparted to lever *E* in the same direction as the driving motion transmitted to gear *B*. As lever *E* is keyed to shaft *A*, any movement of *E* causes a change in the relative positions of gear *B* and shaft *A*.

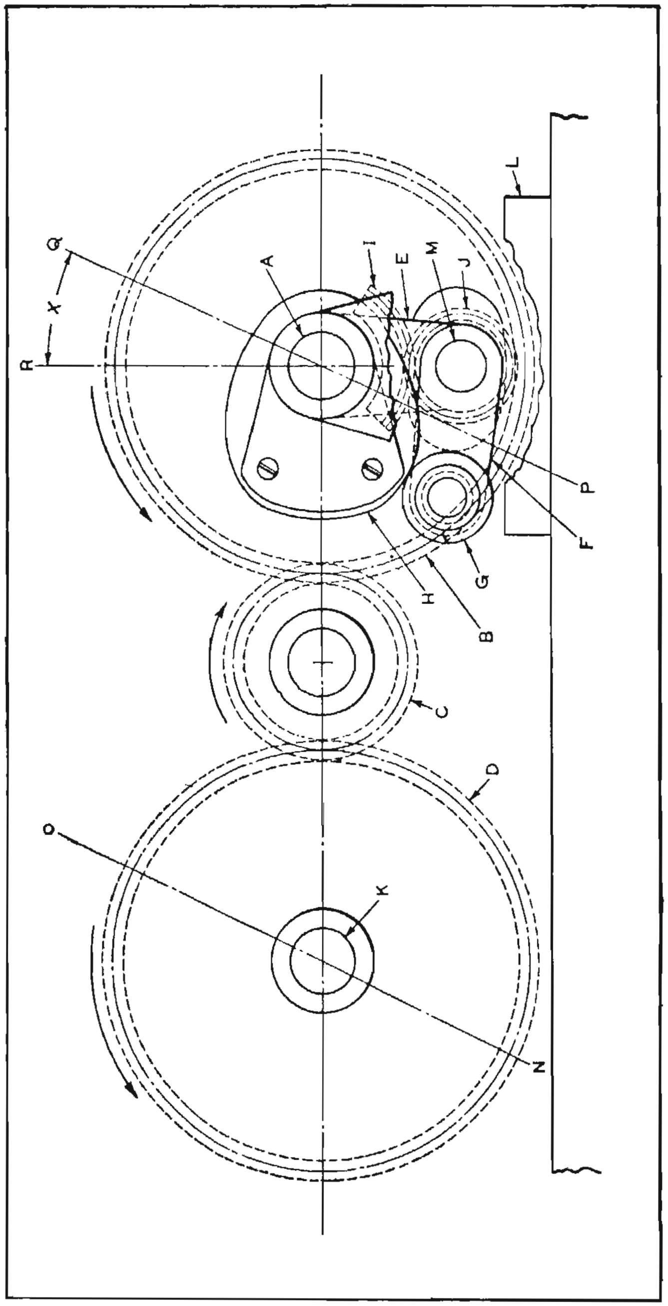


Fig. 6. Diagram Showing Mechanism in Fig. 5 after Shaft A has Rotated 180 Degrees.

When roller G reaches the high point of cam H, the entire assembly again rotates as a unit, their relative positions remaining unchanged until roller G begins to descend to the low point of cam H. In effect, shaft A is first given an accelerated movement, because the motion imparted by lever E is added to that imparted by gear B. The accelerated movement of shaft A is followed by a decelerated movement due to the reverse motion of gear J when lever E is returning to its original position.

The frictional resistance of the assembly, as used on the machine, combined with the resistance of the twisting operation, is usually sufficient to maintain contact of roller G with cam H; at high speeds, however, it may be necessary to attach a spring to lever F to maintain contact of the roller and cam.

In Fig. 5, the perpendicular radial center lines NO and PQ indicate the relative positions of gears B and D, and of lever E. In Fig. 6, center lines NO and PQ indicate that gears B and D have rotated approximately 150 degrees in synchronism. Center line R, through lever E indicates a rotation of 180 degrees of lever E, the angular advance of shaft A relative to gear B being indicated by angle X.

Wire-Twisting Mechanism Designed to Vary Pitch of Twist.— The purpose of the mechanism shown in Fig. 7 is to twist two lengths of wire A and B together, the pitch of the twist being varied to suit certain requirements. The two wires are fed at a uniform rate of speed through the twisting spindle C, the rotating speed of the spindle being automatically varied, so that a definite number of twists of uniform pitch are produced, followed by twists of constantly varying pitch. The wires twisted together in this manner are later cut to length. The two views in Fig. 7 show the interesting mechanism which was designed to produce the required variations in the speed of rotation of the twisting spindle.

As shown in Fig. 7, the bearing I supports the driving shaft *J* from which the mechanism receives its motion. Lever *K* is keyed to shaft *J* and carries the roller *L* at its outer end. Gear *E* is keyed to shaft *M*, which is carried on one end of the lever *N*. The roller *L* fits in a groove of gear

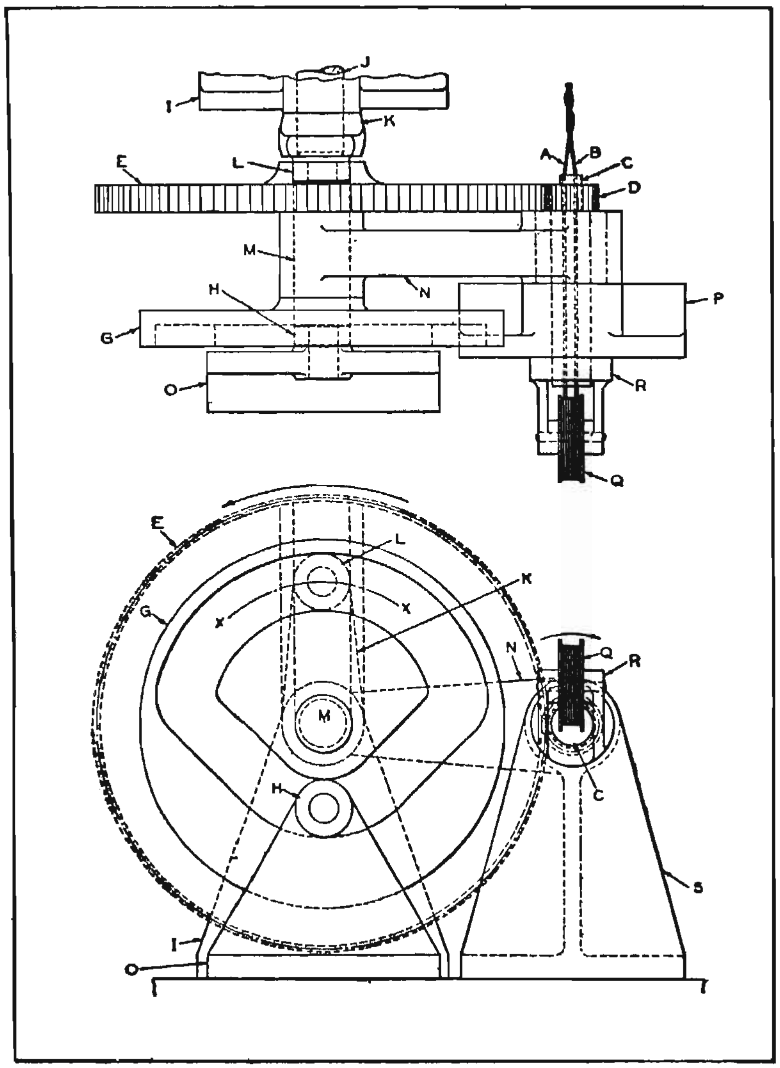


Fig. 7. Mechanism of Wire Twisting Machine Designed to Vary Speed of Gear E and Spindle C.

E and is used to transmit the rotary motion of shaft *J* to gear *E*. The internal cam *G* is keyed to the shaft *M*, and therefore rotates in unison with gear *E*.

The roller *H*, carried on stationary bracket *O*, travels in the groove in cam *G*. Lever *N* is supported freely on the hub of bearing *P* within which the spindle *C* rotates. Spindle *C* is provided with two holes through which the wires *A* and *B* pass, and receives its rotary motion from the gear *E* through the pinion *D*. The wire reel *Q* is carried on the hub *R*, which rotates with spindle *C*.

Referring to Fig. 7, the gear *E* is rotated in the direction indicated by the arrow, receiving its motion from shaft *J* through lever *K*. The spindle *C* is rotated in the direction indicated by the arrow, receiving its motion from gear *E* through pinion *D*, the ratio in this case being 8 to 1. The relative positions of shafts *J* and *M* are controlled by the position of the roller *H* in the groove in cam *G*.

In the position which is shown in the view at the top of Fig. 7, the axes of the shafts *J* and *M* coincide, and the arc of travel of roller *L*, indicated by the line *XX* in the lower view, is concentric with the axis of shaft *M*. Therefore, at this point, there is no movement of roller *L* in the groove on the back of gear *E* and the speed of rotation of gear *E* is uniform and at the same rate as that of shaft *J*. As the wires *A* and *B*, which are wound double on reel *Q*, are fed through spindle *C* at a uniform rate of speed, a twist of uniform pitch is formed as long as the roller *H* remains in the lower portion of the groove in cam *G*.

Referring to Fig. 8, continued rotation of gear *E* has brought the high point of cam *G* into operation on roller *H*, causing cam *G* and gear *E* to be raised by the lever *N*, swiveling on bearing *S*. In this position, the axes of shafts *J* and *M* no longer coincide, being separated by the distance *T*. Owing to the change in the position of gear *E*, it will be noted that the path of roller *L*, indicated by the line *YY*, is now closer to the pitch line of gear *E* than shown in Fig. 7.

The effect of this change is twofold. The peripheral speed at the pitch circle of gear E, when the gear is in the position shown in the lower view in Fig. 7, is greater than at the path followed by the roller L, in the ratio of the difference of their radii. In the view shown in Fig. 8, the path of roller L is closer to the pitch line of gear E, and although gear E

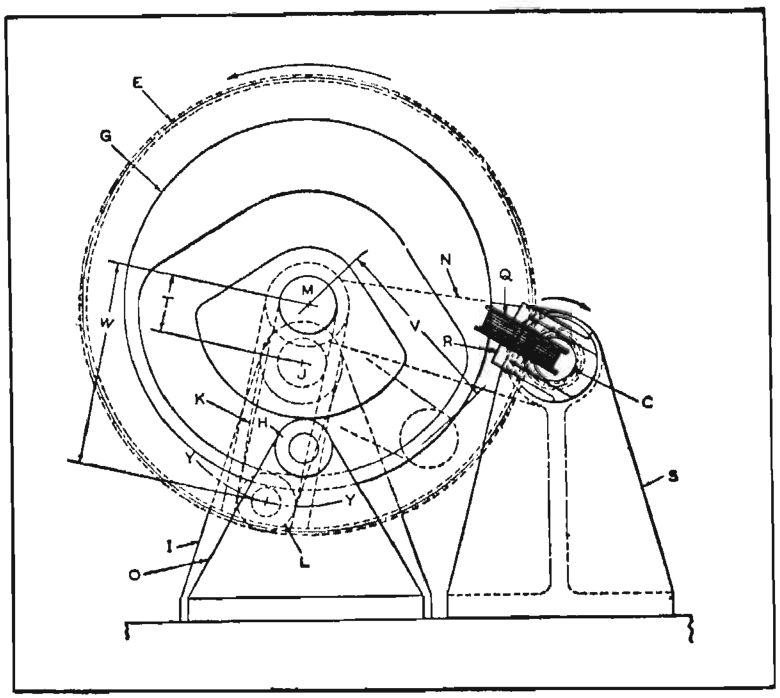


Fig. 8. In this Position Spindle C Rotates at a Lower Rate of Speed than in Position in Fig. 7, and Gear E is Given a Variable Speed Rotation.

and lever K are still rotating on the axes of their respective shafts, the change in the relative positions of the axes of shafts J and M produce the same effect at this point as would occur had the length of lever K been increased a corresponding amount.

As the speed of rotation of spindle C is governed by the peripheral speed of the pitch circle of gear E, spindle C is rotating at a lower rate of speed when in the position shown in Fig. 8 than when in the position shown in Fig. 7, the effect being to give a greater pitch twist to the wires A and B.

The second effect of separating the axes of shafts J and M is to produce a variable rotation of gear E. Regardless of the relative positions of the two axes, the effective length of the lever arm in transmitting motion from shaft J to shaft M is equivalent to the distance between the axes of these two shafts at any given point in the cycle, as indicated by the distance W. This distance will be greatest, and the rotation of gear E will be slowest, when the axes of shafts M and J and roller L are on the same straight line. As shown in Fig. 8, this point has not quite been reached.

When this point is passed, the length of the lever arm is gradually reduced, as indicated by the dimension V. During this period, spindle C is being rotated at a variable speed, thereby producing a twist of varying pitch in the wires. When the lower portion of cam G is again brought into action on roller H, the axes of shafts J and M again coincide, and spindle C rotates at uniform speed.

Precision Variable-Speed Mechanism Employed in Oilgear Drive.—The mechanism to be described is used in conjunction with fluid-power variable-speed drives designed to give precisely the driving speed desired, regardless of fluctuations in the speed of the drive to the pump end of the unit resulting from variations in the load, oil temperature, running fits, or power-line currents.

These precision variable-speed drives, developed by the Oilgear Co., Milwaukee, Wis., comprise a standard Oilgear fluid-power transmission with a Micro Servo-Motor stroke control cylinder which adjusts the pump stroke to give exactly the required hydraulic motor speed. Oil is admitted to the control cylinder by a pilot valve, actuated by a small differential unit which continuously compares the hydraulic motor speed with an accurate time-measuring unit or with the speed of any desired master unit.

One of the Oilgear units, when driven by an ordinary induction motor or lineshaft, will drive its load at any desired speed, either constant, adjustable, or continuously varying between zero and maximum.

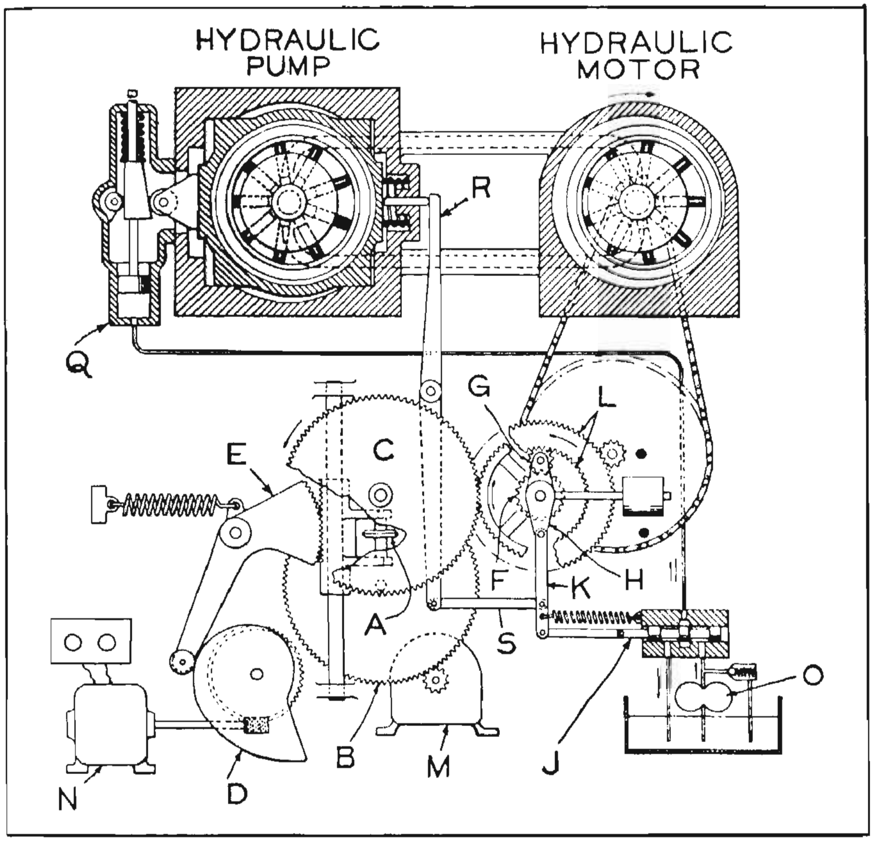


Fig. 9. Fluid Power Drive with Precision Speed Control Employing Synchronous or Selsyn Motor M as Master Speed Unit.

The mechanism, which maintains the speeds within 1/4 per cent of the desired values, is shown in Fig. 9 equipped with a small synchronous or Selsyn motor *M*, which serves as the accurate time-measuring unit. In cases where the frequency error in the power current supply available for

motor *M* is as much as 0.3 per cent and a closer control of speed is desired, the pendulum type precision control shown in Fig. 10 may be used. In this mechanism, pendulum *P* takes the place of motor *M* shown in Fig. 9.

The essential principle of the differential time precision control, as shown in Figs. 9 and 10, is that the hydraulic motor, driven by the flow of power from its variable-stroke pump, transmits the resulting speed to a differential comparator. This differential also receives a standard or master speed time control from the small synchronous motor *M* (Fig. 9) or the pendulum *P* (Fig. 10) which serves as time control units, or from some other unit such as a roll stand or float roll measuring control. In any case, the differential continuously compares the actual speed with the master speed and translates every discrepancy into exactly the increase or decrease of pump discharge necessary to correct the error.

It should be noted that while the use of a pendulum eliminates frequency error, it also limits the speed at which the comparator acts, and thus results in a slower response. When the master speed disk is driven by a motor, the action of the disk type control is so quick that fluctuations of speed due to ordinary load changes are caught and corrected within about one-tenth of a second. With moderate load fluctuations, such as usually exist in a continuous processing plant, the momentary governing errors are held within 1/4 of 1 per cent, plus or minus, while the integrated error over a period of time—say, ten seconds or more—is too small to be measurable by a stop-watch.

The differential time disk type control is shown in its usual form in Fig. 9, with a small synchronous or Selsyn motor *M* furnishing the master speed. If *M* is a synchronous motor, the control is by "time," and the speed of the hydraulic motor is given in revolutions per minute. If *M* is a Selsyn motor, it receives its current from a Selsyn generator driven by its master unit, and follows the speed of that master through its entire range from the standstill or stationary position to maximum speed.

The master speed is transmitted to the differential mechanism through the friction-disk ratio-changer, comprising a movable idler disk *A* that is spring-pressed between the driving or master disk *B* and the driven disk *C*. Disk *A* is mounted in a slide-block, and is adjustable by push-button control motor *N* or by a manual or float-roll control through cam *D* and segment *E*.

The master speed, as modified by the ratio-changer, is transmitted by gear teeth in the edge of disk *C* to a gear integral with the sun pinion *F*, which is the first leg of the differential comparator already mentioned. Planet gear *G* is mounted on a radial valve actuating crank *H* (the third

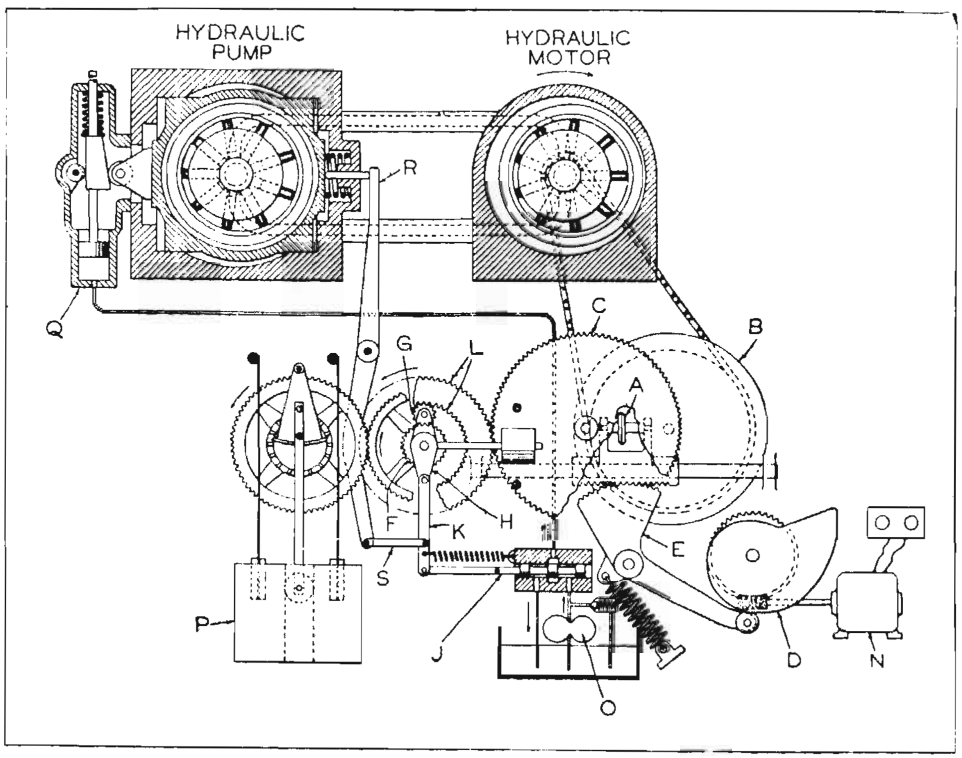


Fig. 10. Drive Similar to That Shown in Fig. 9 but with Different Type of Timing Unit.

leg of the differential) which, in turn, actuates pilot valve *J* through floating lever *K*. Planet gear *G* also meshes with an internal ring gear *L* which is exactly twice the diameter of sun pinion *F*.

Any movement of the pilot valve resulting from movement of the axis of the planet gear and crank *H* changes the pump stroke and hydraulic motor speed. To hold pilot valve *J* stationary, ring gear *L* (the second leg of the differential) must be driven by the hydraulic motor at exactly one-half the standard speed of gear *F*. Any discrepancy will result in a slight movement of valve *J*, permitting oil from gear pump *O* to enter the pump-stroke changing cylinder *Q*, and thus increase the pump discharge, or permitting oil to escape from *Q*, which will reduce the pump discharge. When the change in pump discharge has corrected the motor speed, so that it again balances the speed of the differential gears, the pilot valve will resume its neutral position.

An essential part of the mechanism is the follow-up lever *R* which responds to the movement of the pump slide-block, and by means of link *S*, closes the pilot valve as soon as the required correction has been made, thus preventing the so-called "hunting" action. As the disks *A*, *B*, and *C* are made of hardened steel, are accurately ground, and transmit no appreciable power, they show little or no wear after long service. This is true even when the speed is not changed and idler disk *A* runs in one location continuously.

A characteristic feature of the drive described is that it cannot be overhauled by a decelerating inertia load, so that winders and unwinders for paper machines, super-calenders, printing presses, textiles, strip and foil, etc., driven by the differential time controls will automatically start and stop heavy rolls of material without sag or added strain in the web. By means of the precise speed adjustment of the disk control, two or more rolls of paper or foil can be unwound from rolls of different diameter at identical surface speeds and at uniform tensions, and laid together.

CHAPTER 14

Feed Regulating, Shifting and Stopping Mechanisms

In all machines which perform operations on parts or on material, means must be provided for regulating, shifting and stopping the feed of either a tool or the work. Such mechanisms may be utilized to obtain coarse or fine feeding movements; to adjust the amount of automatic feed; to proceed through a complete cycle of advancement, cutting, withdrawal and replacement of workpieces; to properly position work for drilling or cutting; to stop the rotation of work in a particular angular position; to shift from one work-performing mechanism to another; and to stop operation of a machine when no more material or work is available for feeding. These and other operations are performed by the mechanisms described in this chapter.

Other mechanisms which perform similar functions are described in Chapter 16, Vol. I and Chapter 14, Vol. II of Ingenious Mechanisms.

Mechanism for Obtaining Coarse and Fine Feeding Movements.—It was in conjunction with the building of a small milling machine type of grinder for cutting slots 1/16 inch wide by 1/8 inch deep in some square, hardened pieces that the screw and worm-wheel mechanism shown in Fig. 1 was developed for the purpose of obtaining a quick approach to the grinding wheel and a fine feed for grinding the slot. A rubber cutting wheel was used for the slot-grinding operation.

The screw or worm *A* is a close sliding fit in the bore *B* and engages the worm-wheel *C* as shown. The worm-wheel is pinned to the stud *D* which has a bearing in the cast-iron bracket *E*. The stud *D* is made square at *F* to accommodate the lever *G*, and is threaded to receive the knurled nut *H*.

In operation, the handle *K* of lever *G* is moved in either direction. This causes the screw *A* to slide in or out with much the same action as a rack operated by a pinion. After using the lever to bring the work into the grinding position, the knurled nut *H* is tightened to hold the lever stationary. A fine feeding movement can then be obtained by turning the knurled nut *J*, worm-wheel *C* acting as a fixed nut for the screw *A*.

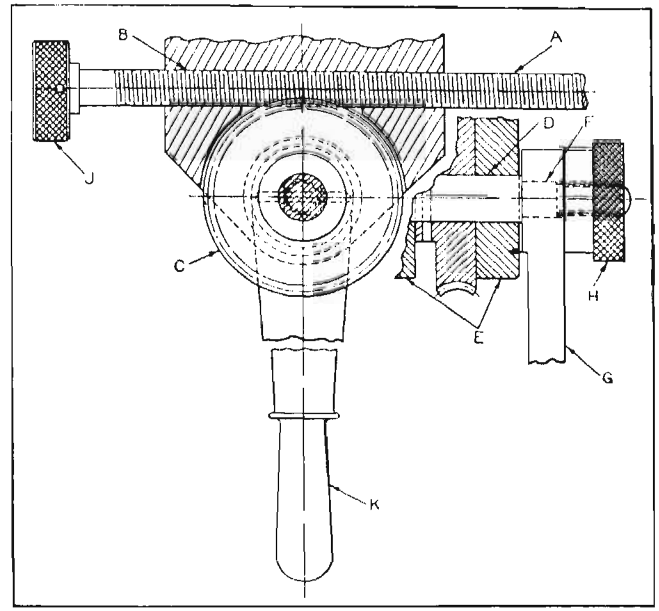


Fig. 1. Mechanism in which Worm Gear C Acts as a Pinion and Worm A Serves as a Rack.

Fine Feeding Mechanism Adjustable from 1 to 50 Microns per Revolution.—The fine feed of carriage *U*, Fig. 2, adjustable in increments of 1 micron from 1 to 50 microns (1 micron = one millionth of a meter or 0.000039 inch) per revolution of drive-shaft *C*, was obtained by means of the mechanism illustrated in Figs. 2 to 6. This design eliminates the reciprocating parts commonly encountered in feeding devices, which might cause vibration and chatter. The index-wheel Q, Fig. 3, can be turned in either direction while the machine is in operation, thus increasing or decreasing the feed within the range for which the mechanism is designed. The feeding range can be increased to any number in excess of 50 microns by increasing the diameter of the ratchet and cams to obtain the desired change in feed.

An advantage of this mechanism is that the fine feeds are

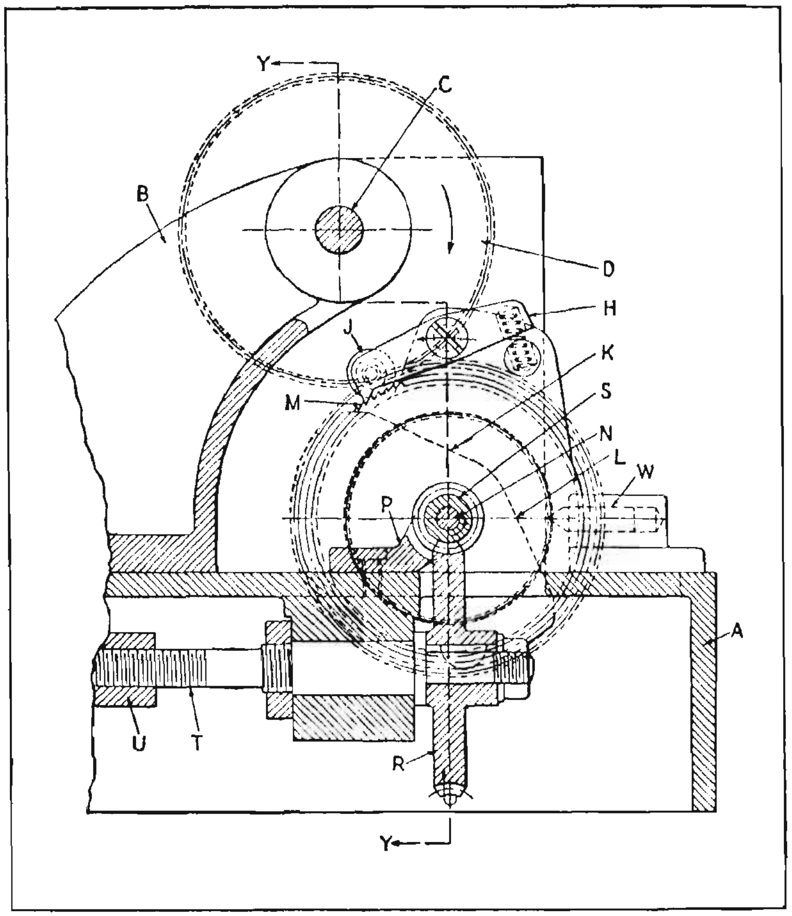


Fig. 2. End View of Feeding Mechanism Taken Through Section X-X of Fig. 3.

obtained with standard tolerances and parts. Commercial gears with some backlash are satisfactory. The ratchet wheel M has a relatively coarse pitch of 3/32 inch, with a correspondingly long life for the tooth of pawl H. The feed-screw T is 1/2 inch diameter with a 2 millimeter pitch thread which is relatively coarse for such a fine feed. The only point where extreme care must be exercised is between

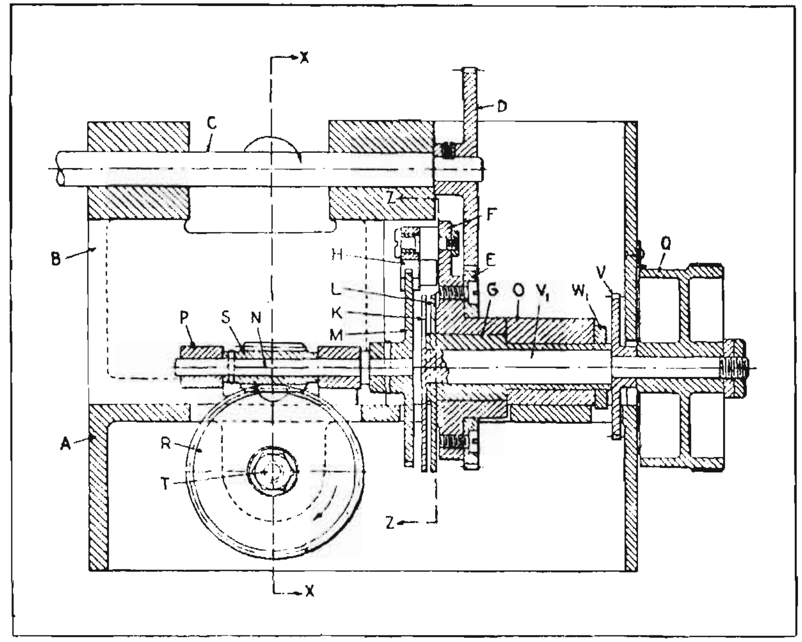


Fig. 3. Section Y-Y of Fig. 2 Showing Train of Mechanism between Drive Shaft C and Feed Screw T.

the feed-screw and the carriage nut, it being necessary to prevent any backlash.

Another advantage of this mechanism is that it is comparatively silent in operation. In conventional reciprocating type of feeding mechanisms, the pawl, on the return stroke, usually rides over the ratchet teeth and makes an annoying clicking noise. It also tends to wear both the pawl and ratchet teeth. This action is eliminated in the mechanism described.

The feeding mechanism consists essentially of a base A, on which is mounted an upright casting B that supports drive-shaft C, as shown in Figs. 2 and 3. Gear D, which is fastened to one end of this shaft, drives gear E. The latter gear, which is secured to pawl-carrier F, is mounted on

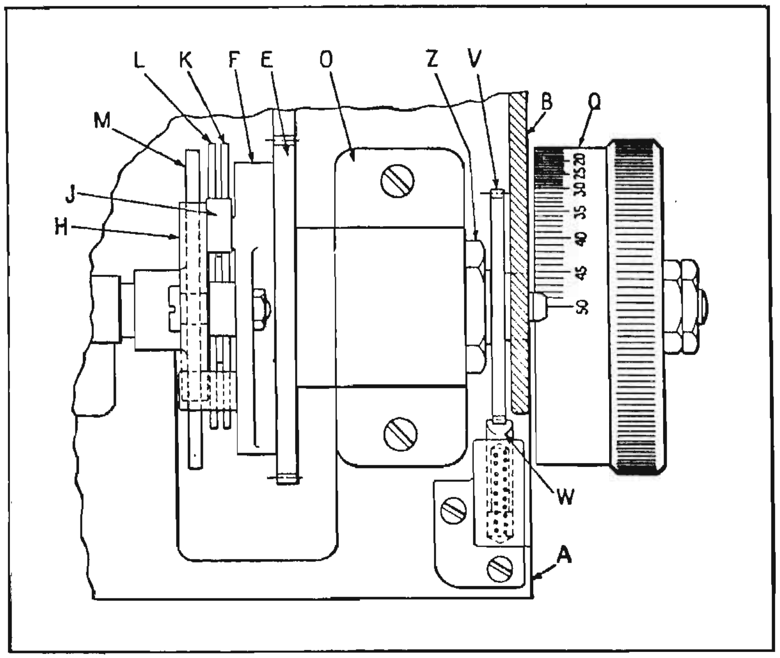


Fig. 4. Partial Plan View of Feeding Mechanism. Index Setting is Held by Plunger W Engaging Gear V.

sleeve bearing G. Pawl H carries a roller J on its driving end, and is held to carrier F by a stud on which the pawl pivots. As the pawl-carrier is rotated by driven gear E, the pawl is carried above the ratchet wheel M for a portion of each revolution when roller J rides on cams K and L. When roller J is carried past the cam surfaces, the pawl engages and drives the ratchet wheel through spring action.

Ratchet wheel M is pinned to shaft N, which is mounted on bearings P. Worm S, which engages worm-wheel R, is also pinned to this shaft. The worm-wheel is keyed to the feed-screw T. The 30 to 1 ratio of the worm to the worm-wheel is such that an advance of one tooth of the ratchet wheel rotates the feed-screw 1 micron. The feed-screw moves carriage U back and forth on ways which are not shown in the accompanying illustrations.

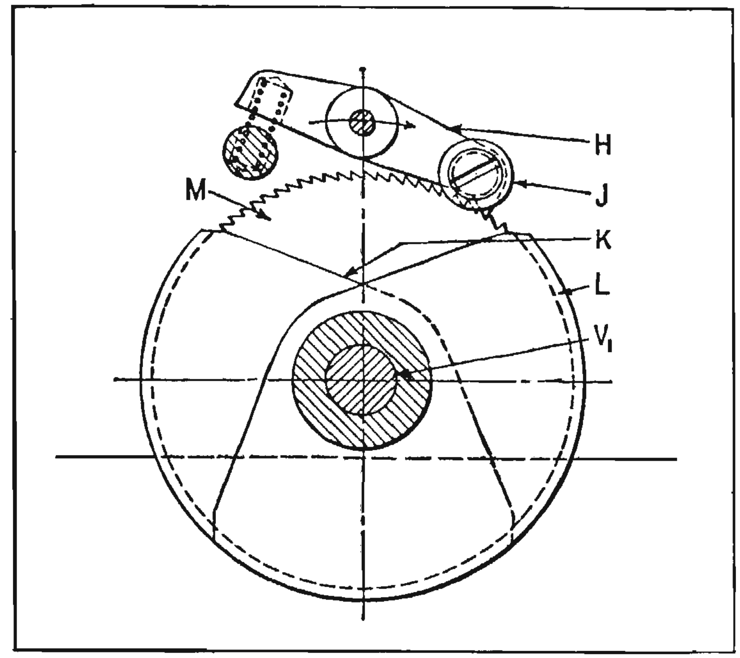


Fig. 5. Section Z-Z of Fig. 3 Showing Positions of Cams K and L for a 25-Micron Feed.

The relative position of cams K and L shown in Fig. 2 occurs when the index-wheel is set at 50. This setting and cam position cause the pawl to be lifted out of engagement with the ratchet wheel for half of the cycle—180 degrees—of the pawl-carrier. During the remaining half of the cycle, the pawl engages the teeth on the ratchet wheel, and through the worm and worm-wheel, imparts the maximum feed to the feed-screw of 50 microns per revolution of the drive-shaft. To reduce this rate of feed, the index-wheel is turned to any intermediate position between 0 and 50, which changes the position of cam *K*. The non-adjustable cam *L* is fixed to sleeve bearing *G*, which is held in bracket *O*.

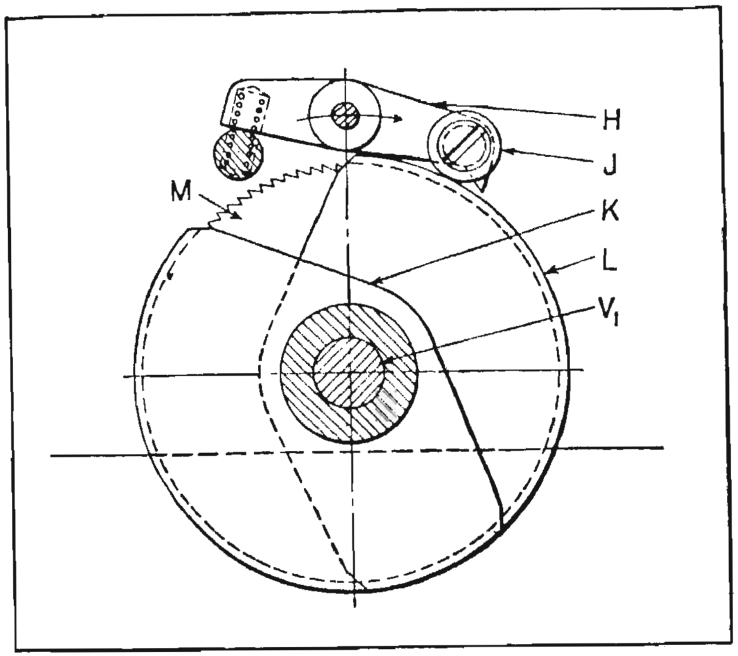


Fig. 6. Another View of Section Z-Z Showing Positions of Cams K and L for a 12-Micron Setting.

Index-wheel *Q* is mounted on shaft V_1, which turns in sleeve bearing *G*. Cam *K* is mounted on the other end of this shaft, and is turned an amount corresponding to the change in setting of the index-wheel. This varies the angular opening between the cams, permitting the pawl to engage more or fewer teeth on the ratchet wheel, depending upon the setting. When the index-wheel is set at 25, cam *K* is moved to the position shown in Fig. 5. Fig. 6 shows the relative position of the cams when the feed has been further reduced to 12 microns per revolution of the driving shaft.

A gear *V*, Fig. 3, having the same number of teeth as ratchet wheel *M*, is also mounted on shaft V_1. This gear is frictionally held to any setting on the index-wheel *Q* by means of a detent on spring plunger *W*, Fig. 4, which enters between the teeth of gear *V*. The number of teeth in this gear also corresponds to the number of graduations on index-wheel *Q*. The index-wheel is revolved, by gripping the knurled rim, to any number and held there by the detent.

The feeding cycle can be adjusted to begin at any predetermined point with respect to the drive-shaft *C*, Figs. 2 and 3, by changing the angular position of cam *L*. This is accomplished by loosening nut W_1, and revolving the sleeve bearing *G* and cam *L*, which is fixed to this bearing, to the desired position. The cam is then locked in this position by again tightening the nut.

Controlled In-Feed Mechanism for Screw Milling Machine.—Fig. 7 shows a dashpot designed to retard the rate of movement of the in-feed of a screw milling machine with a backed-off hob type cutter. The object of this device is to prevent the hob from jumping forward during the movement from one tooth to the next, so that smoother operation, with a uniform controlled in-feed, is obtained.

In view *X*, cutter-slide *A* houses a spindle carrying the hob *B*. Lever *C* operates the cutter-head and is extended to form a handle (not shown). In the arrangement illustrated, part *D* is integral with the cutter-head. Movement of lever *C* advances the hob toward the work until contact is made with stop-screw *E*, which is adjustable for the desired depth of cut.

Referring to view *Y*, control rod *F* is shown in contact with a piston contained in cylinder *G*. The oil or other suitable fluid behind the piston escapes through an adjustable needle valve H by way of suitable passages, at a rate determined by the setting of the needle valve. After completion of an operation, the oil returns through a valve to the other side of the piston under the influence of the return spring. Screws K act as filling plugs.

When slide D comes into contact with stop-screw E, it will meet the central rod F, and its rate of progress will be limited by the rate at which oil or other fluid can escape

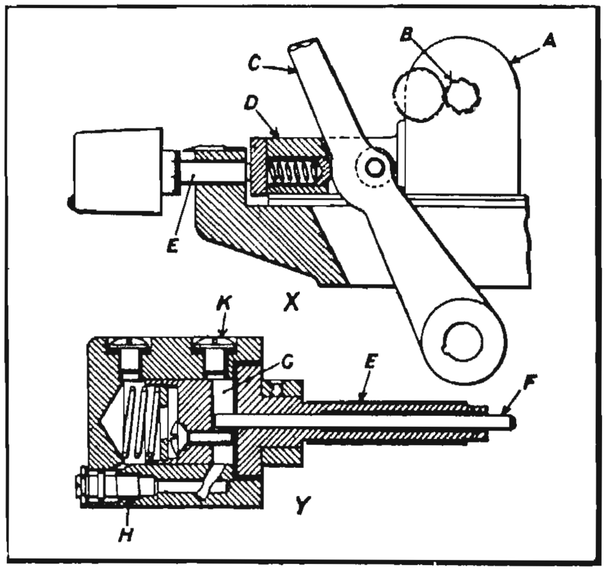


Fig. 7. Dashpot Mechanism Designed to Retard the Rate of Movement of Milling Machine In-Feed.

through the needle valve. The spring and plunger shown in the view at X provides a slight additional movement of lever C after contact with screw E, as required, to permit the operation of a trip motion.

Variable Feed Arrangement for Automatic Wheel-Dressing Device.—The actuating screw of an automatic wheel-dressing device employed on a grinding machine was designed to move the dressing diamond a given amount each

time it functioned, but there was no provision for varying the amount of this movement. Variations in the hardness and bonding of a grinding wheel, however, make it desirable that the amount of feed or movement of the truing diamond be adjusted to suit individual grinding wheels and the conditions under which they are used. Fig. 8 shows an

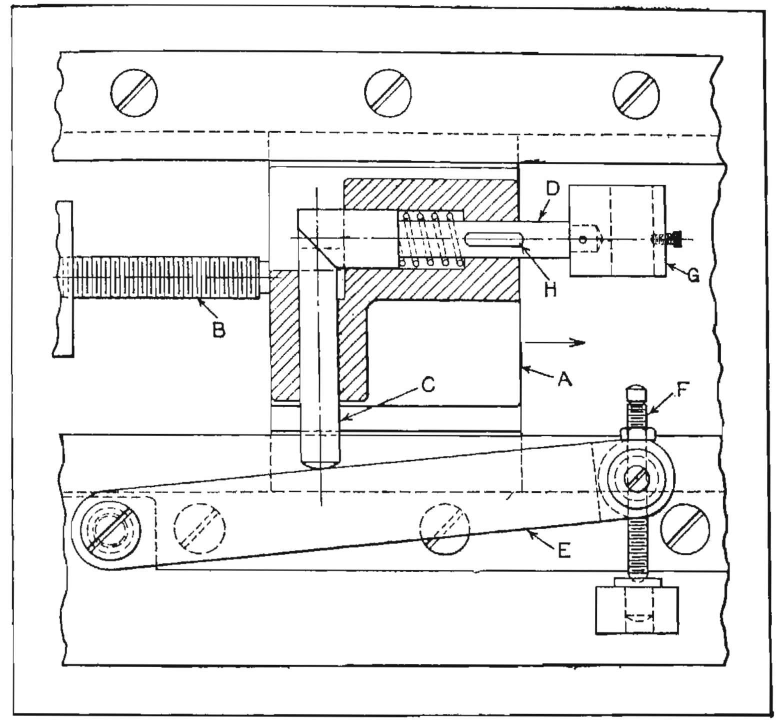


Fig. 8 Feed for Automatic Pressing Device Used for Truing Grinding Wheel.

arrangement designed to permit such adjustment. With this device, the feed can be varied any desired amount by adjusting the bar E.

Referring to Fig. 8, slide A is moved a fixed amount in the direction indicated by the arrow each time the actuating screw B functions. Mounted on slide A at right angles to each other are two plungers C and D. Plunger D is kept in contact with plunger C by means of the coil spring. Plunger C, in turn, is kept in contact with the adjustable bar E. The bar is also held in a fixed position by the pressure exerted by the spring.

The angle of the bar is adjusted by means of screw F which can be clamped in place by a locknut. The diamond-holder G is kept from rotating by means of the key H, which slides in a keyway cut in slide A. The amount of movement imparted to holder G by a given movement of screw B is determined by the angular setting of the sine bar E.

Automatic Feeding Mechanism for Drill Press.—The attachment shown in Fig. 9 has been applied to a drill press to provide automatic power feed and, in addition, to feed the drill into the work, disengage the feed when the drill has reached the required depth, return the spindle to the starting point, and then re-engage the feed. The work is removed and replaced by a new piece during the return movement of the spindle. This arrangement resulted in a considerable increase in production, with greatly reduced operator fatigue.

A front view of the attachment is shown at the lower right of Fig. 9, and a plan view at the upper right of Fig. 9. The spindle A carried in quill B is fed into the work by the rack D and pinion C. The rack and pinion are part of the original equipment; a handle attached to shaft S which carries pinion C has been removed. A worm-gear P is carried on shaft S in place of the handle. The bracket I pivots on stud J, which is carried on a bracket attached to the head of the press. This bracket is not shown, however, as it must be made to suit the individual application.

Bracket I supports the shaft M, which carries the worm K and sheave N. Worm K is keyed to shaft M, and is provided with a hub at one end which supports the compression spring L, shown partly compressed in Fig. 9. Shaft M is rotated in the direction indicated by the arrow by the

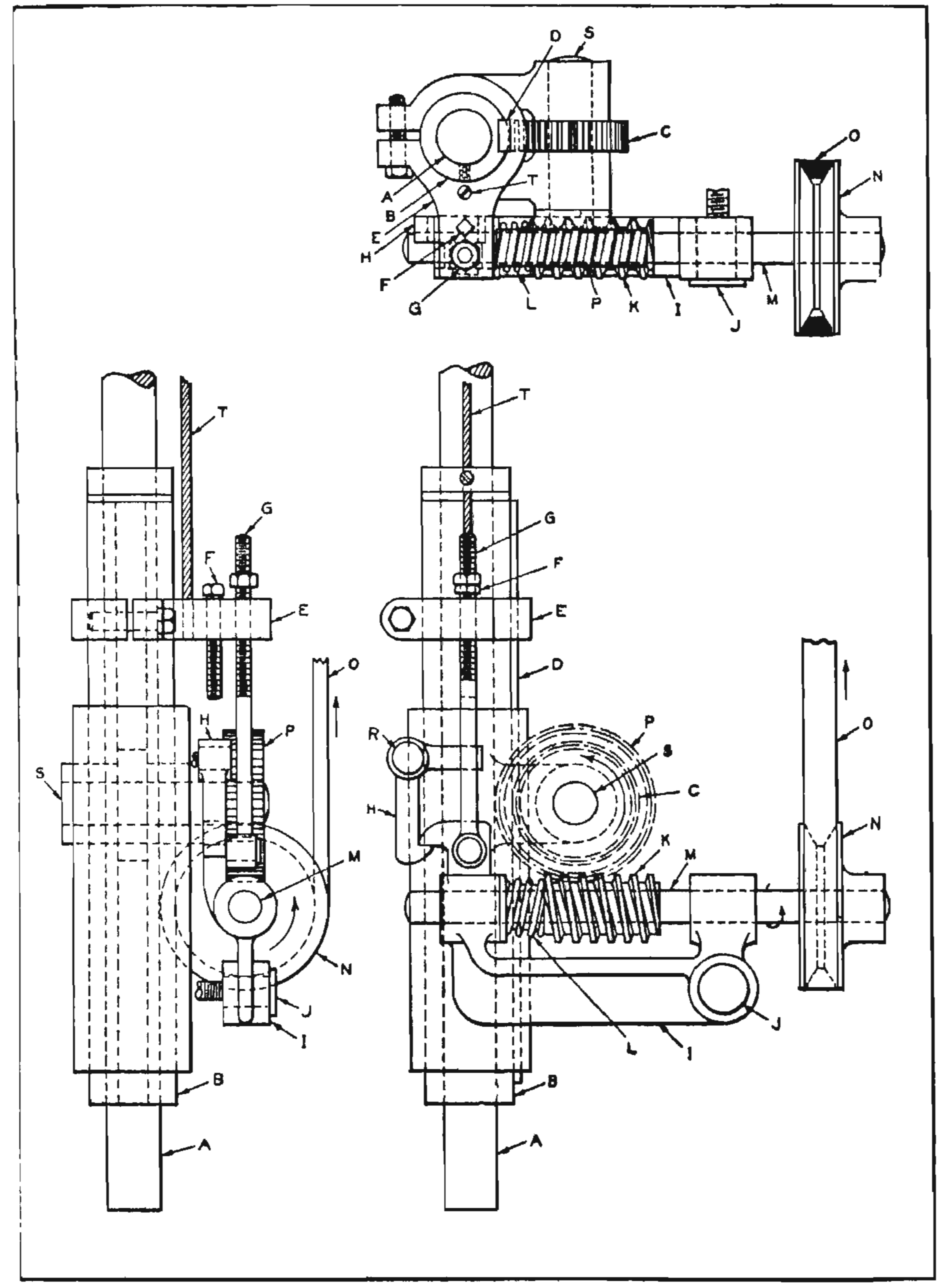


Fig. 9. Side, Front, and Top Views of Automatic Feeding and Reversing Mechanism Applied to Drill Press.

belt O driven from a sheave carried on the overhead horizontal shaft which furnishes the power for the driving spindle A.

The worm K meshes with the worm-gear P, rotating it in the direction indicated by the arrow. Worm-gear P transmits its motion through shaft S to pinion C which, meshing with rack D, produces a downward movement of quill B. Pawl H swivels on stud R, carried on the bracket attached to the head of the drill press. The hook-shaped end of pawl H engages the hooked end of bracket I, holding it in position to obtain proper meshing of worm K and gear P.

Bracket E is clamped to quill B and carries the screw F. Rod G passes through bracket E and is attached to the end of bracket I. Cable T, attached to bracket E, passes over an overhead pulley and carries a weight at the other end of sufficient size to raise the quill B and spindle A against the force exerted by gravity.

The attachment operates as follows: The rotation of shaft M drives the quill B downward through the action of worm K, worm-gear P, pinion C, and rack D. This movement continues until screw F, carried on bracket E and moving downward with quill B, comes in contact with the arm of pawl H and disengages it from bracket I. Bracket I, acted upon by the force of gravity and the pull exerted by belt O, is caused to swivel on stud J and disengage worm K from gear P. When this occurs, the downward movement of quill B discontinues and B is drawn upward by the action of the weight attached to the end of the cable T.

The upward movement of quill B continues until bracket E contacts with the nut on rod G, at which point bracket I is again raised to engage the pawl H, thereby re-engaging the worm K and gear P and again causing spindle A and the tool to be driven downward.

When worm K is disengaged from gear P, the spring L causes shaft M to be moved axially until worm K makes

contact with the bearing of shaft M in bracket I. The purpose of this movement is twofold in that it permits some of the shock resulting from re-engagement to be absorbed by spring L and also permits proper engagement of worm K and gear P in case they should strike on the tops of their teeth.

During the feeding operation of the cycle, worm K is held in contact with the bearing of the shaft in bracket I by the resistance caused by the friction of the quill B in the drill press head. The operation of the drill press is completely automatic, it merely being necessary for the operator to remove and replace the work.

Indexing Mechanism on Special Drilling Machine.— In tooling up for the production of a limited number of large Diesel engines, it was found advantageous to build two special semi-automatic drilling machines. These machines, operated by one man, perform all the angular drilling operations required in several parts of a similar design. The machine shown in Fig. 10 is designed to drill sixteen 3/32-inch holes in the piston shown at W. The holes are located in two rows around the periphery of the piston, and are equally spaced and drilled at an angle of 10 degrees with the axis of the piston.

The special machine shown in Fig. 10 consists of a drill press A, hooked up to an indexing device and mounted on a steel base. The piston is located on an arbor B and clamped securely by a tapered block C. The operator starts the drilling cycle by pushing a button. The arbor carrying the piston then moves through an intermittent rotating cycle until all sixteen holes are drilled. At the end of the cycle the machine is stopped automatically.

The indexing and the drill-feeding mechanisms are driven by a 1/4-horsepower motor D through a speed reducer E. Two properly synchronized cams G and M are mounted on the camshaft F. The grooved face-cam G actuates the ratchet H through pawl J, thus indexing the work as

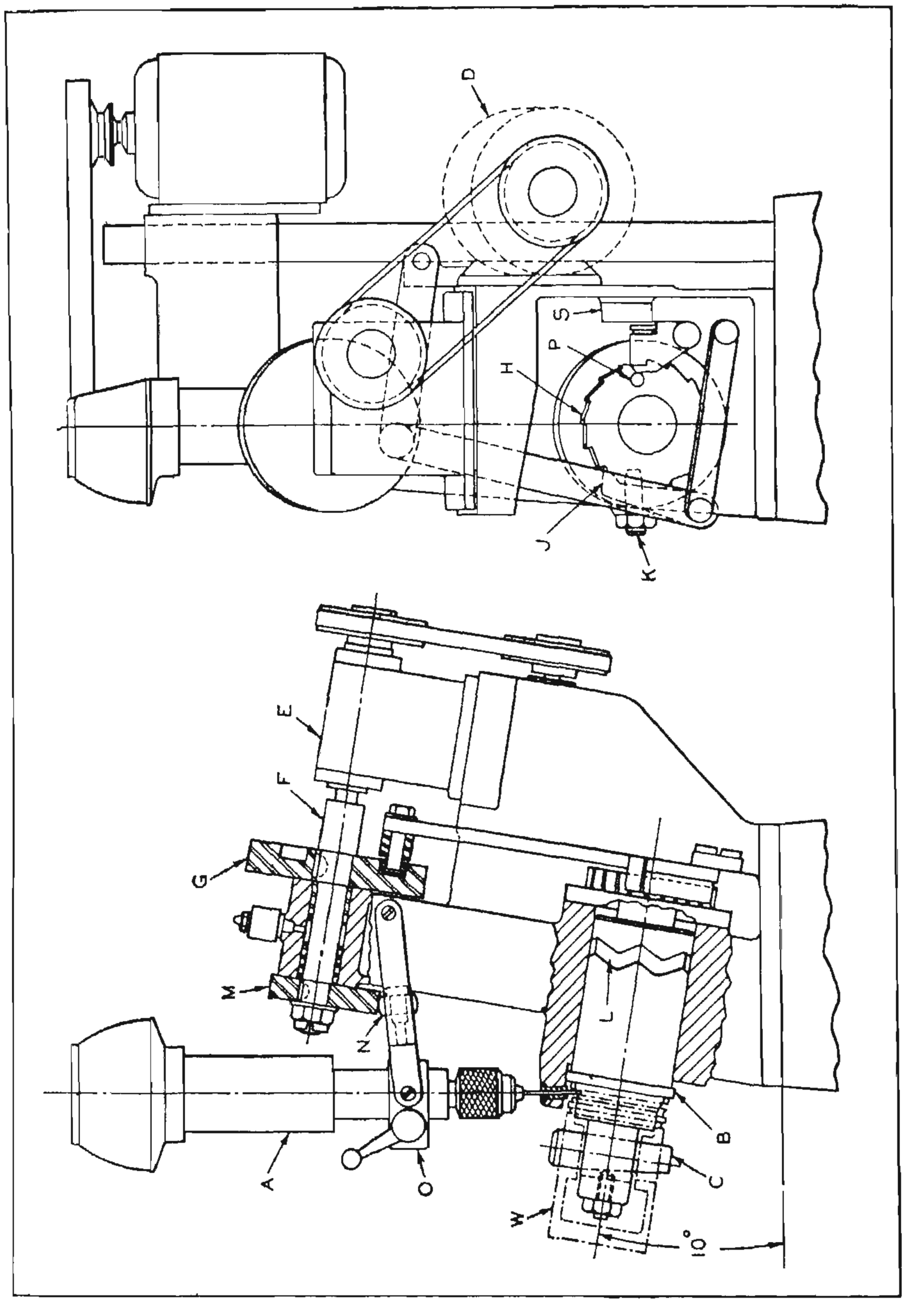


Fig. 10. Special Piston-drilling Machine with Automatic Indexing and Feeding Mechanism.

required for the angular spacing of the holes. A fixed plug *K*, engaging the cam groove *L* milled on the periphery of the work-arbor, causes the latter member to move in and out as required for drilling the two rows of equally spaced holes. The cam *M*, actuating the roller *N*, imparts the required feeding movement to the drill. The collar *O* can be adjusted to compensate for drill wear. A pin *P*, which is located on the ratchet wheel, comes in contact with a micro-switch *S*, thus stopping the machine automatically after the last hole in the piston has been drilled.

Spindle-Control and Collet-Operating Mechanism for Screw Machines.—The feeding of irregular-shaped automobile parts into the work-holding collets of a special multiple-spindle screw machine necessitated the provision of some means for automatically stopping the rotation of the spindles in exactly the same angular position at the loading station, opening the collet to receive the work, closing the collet on the work, and releasing the spindle-driving clutch. The mechanism developed to perform these functions consists primarily of a three-ball type spindle stopping and starting clutch and a collet opening and closing device, arranged as shown in Fig. 11. Details of the principal members are shown in Figs. 12, 13, and 14.

The spindles of the machine are all housed in a rotor *A*, Fig. 11, operated by a Geneva motion indexing mechanism. A center gear (not shown) which engages the pinions *E* serves to drive all the spindles. The collets are closed by the action of a stiff helical spring *S*.

This arrangement left little space for the spindle clutch, which was required to be of very compact design. Referring to the assembly view of the spindle, Fig. 11, the rocker arm *B* opens the collet by moving rod *C* to the right, causing spring *S* to be compressed. The spindle *D* is driven by the pinion *E*, which is bored out to house clutch ring *F*. The clutch ring is a drive fit in pinion *E* and is pinned to it, so that members *E* and *F* comprise an assembled unit. This

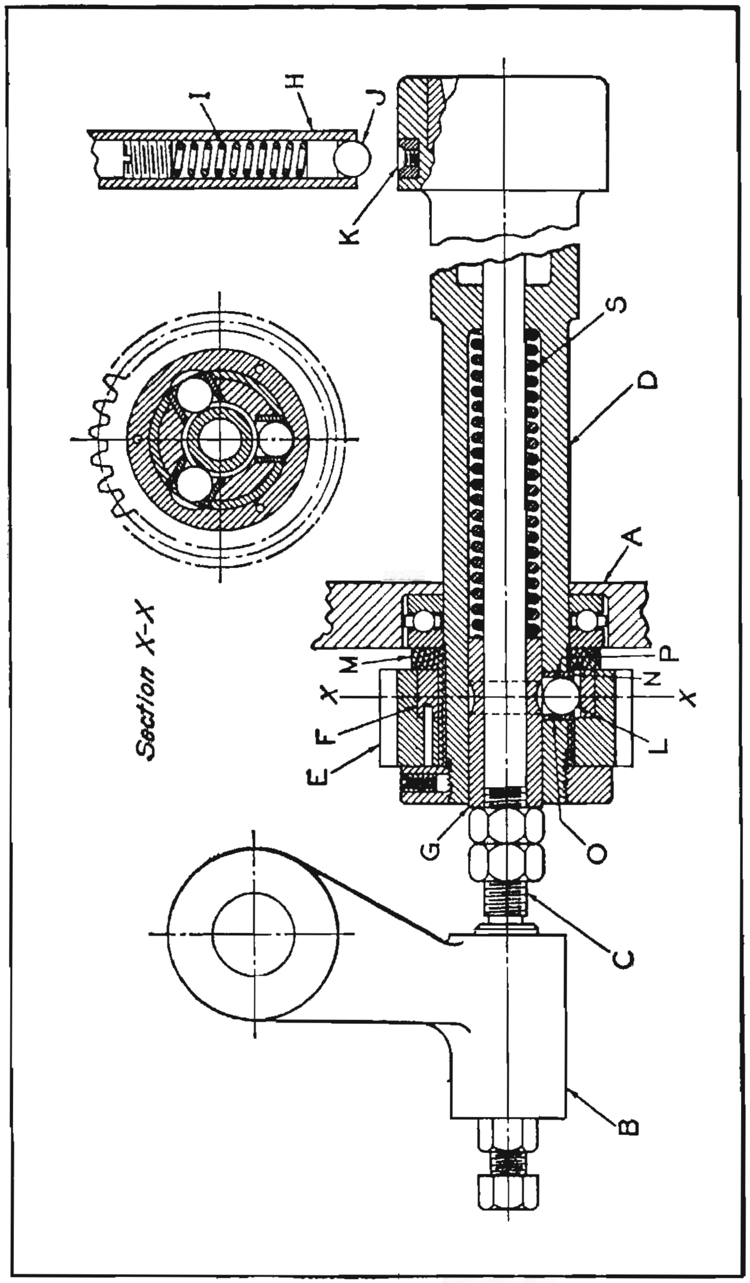


Fig. 11. Mechanism Designed to Automatically Start and Stop Rotation of Spindle D and Open and Close Its Work-holding Collet.

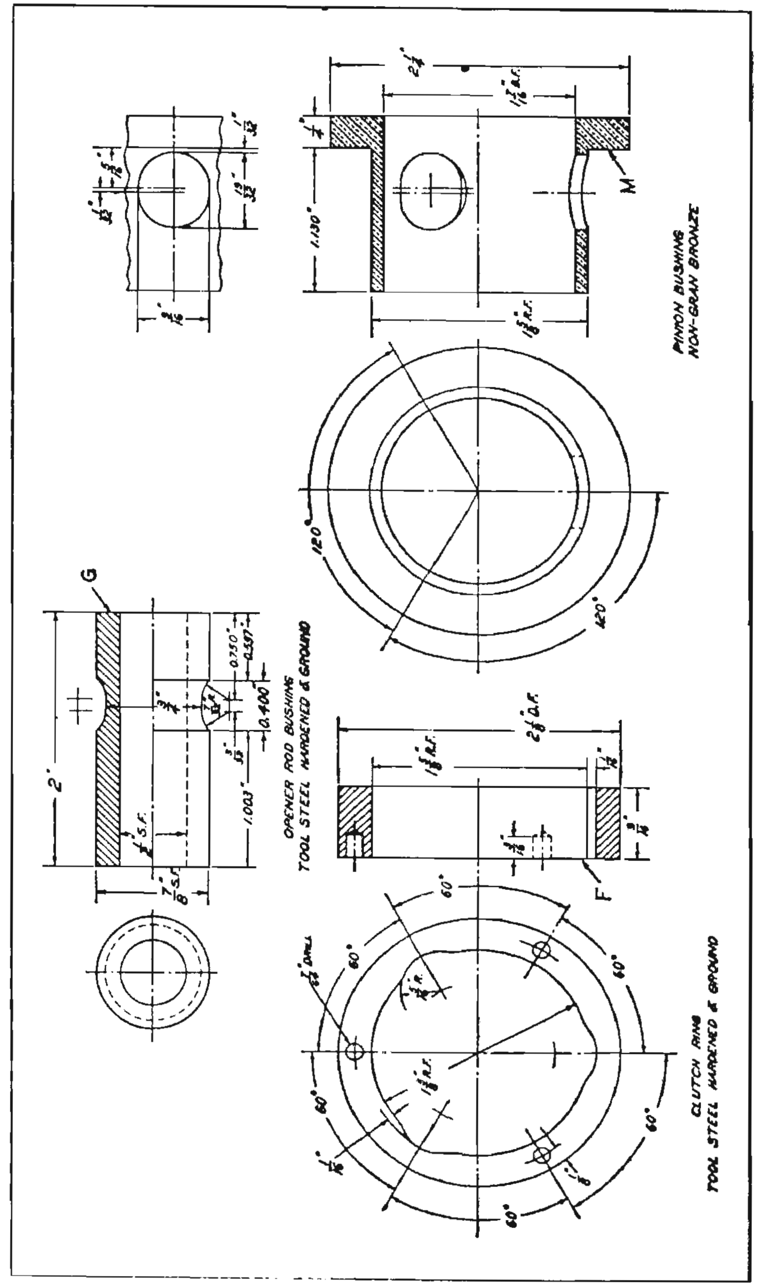


Fig. 12. Details of Collet-opening Rod Bushing G, Clutch Ring F and Pinion Bushing M of Mechanism Shown in Fig. 11.

unit runs continuously at a speed of 900 R.P.M. during the operation of the machine.

Three holes are drilled and bored through pinion bushing *M* to receive the bushings *N* which house the steel balls *L* of the clutch engaging and disengaging mechanism. These bushings are spaced 120 degrees apart, and are a drive fit

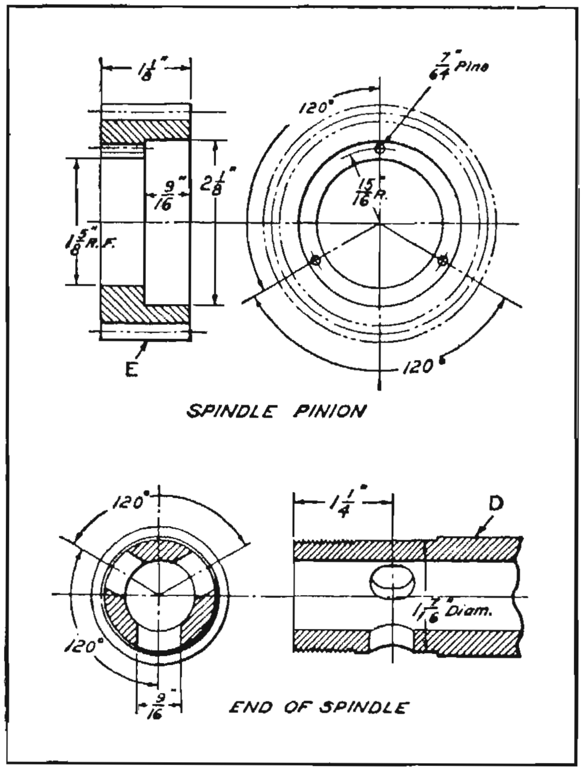


Fig. 13. Details of Spindle Pinion E and the Driven End of Spindle D in Fig. 11.

in the bored holes. Bushings *N* have a slot for holding spring *O* (see Fig. 14). A 5/64-inch hole *P* is also drilled in the bushing to form a retaining seat for the ball *L*.

The bushing *G*, Fig. 11, which is employed to guide the collet-opening rod *C*, is also used for operating the spindle

clutch. The tube *H* at the front end of the spindle rotor carries a spring *I* and ball *J*. The end of tube *H* is machined to prevent ball *J* from falling out of the tube. The hardened cup *K* in the spindle *D* is threaded to facilitate its removal when necessary. Tube *H* is moved downward by a cam device when the spindle clutch is released, so that the ball *J* is forced into the cup *K* to stop the spindle. Thus the

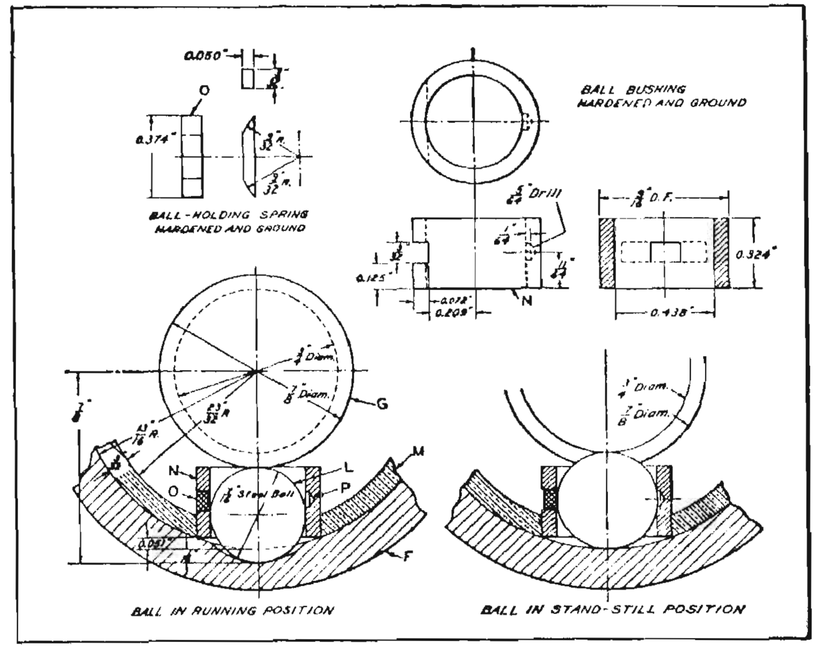


Fig. 14. Enlarged Scale Views of Details N and O in Fig. 11 and Broken Section Views Showing Spindle Clutch Balls L in Running and Idle Positions.

spindle is always stopped in the same position with respect to the rotor *A*, so that the work, in turn, can be fed into the collet in the correct position. This requirement presented a rather difficult problem, since the spindle runs at a speed of 900 R.P.M. A clutch made up for testing purposes was operated successfully at a speed of 1200 R.P.M. before the design was incorporated in the mechanism described.

In the assembly shown in Fig. 11, the 7/16-inch ball L at the bottom position is shown resting in the 5/16-inch radius cut-out in clutch ring F. The ball remains in this position only momentarily, however, since rotation of clutch ring F brings the cylindrical 1 5/8-inch diameter section of the latter member into contact with the three balls, forcing them inward toward the center of the spindle, where they are held by spring O and depression P.

The two enlarged views in Fig. 14 show exactly how the ball is located both in the running and in the stand-still positions. When arm B, Fig. 11, swings to the left, the draw-rod C closes the collet, and the three 7/16-inch balls are forced outward, so that they come in contact with the bottom of the 5/16-inch radius cut-outs in the clutch ring F and thus start the spindle rotating. When arm B swings to the right, the channel in the bushing G clears the three balls, causing the spindle to stop and the collet to open.

Device for Automatic Shifting between Two Cam-Operated Packing Mechanisms.—In designing a device for disposing of sheets from the delivery end of a paper finishing machine, it was desirable to provide a means of automatically and periodically shifting from one cam-operated packing mechanism to another. The shifting device here described permits five hundred sheets of paper to be placed on one tray while a second full tray adjacent to it is being replaced with an empty one.

Referring to Figs. 15 and 16, arms A and B, which are actuated by cams C and D, are secured to and alternately rock their respective pivot shafts A_1 and B_1. Each pivot shaft is connected to a separate packing mechanism (not shown).

One arm rocks its pivot shaft five hundred times, whereupon it stops at the top of its stroke and the other arm immediately begins its cycle of five hundred strokes. The working stroke occurs when the follower of the arm rolls into the hollow of its respective cam. The cams are secured

to shaft E, which rotates continuously at one revolution per cycle of the finishing machine.

A plate F, equal in diameter to the cams, is located between them and slides back and forth on a key in shaft E. When this plate is in contact with either cam, the follower is prevented from rolling into the hollow of that cam and is held at the top of its travel. Meanwhile, the follower on the other arm is free to roll into the hollow of its cam and

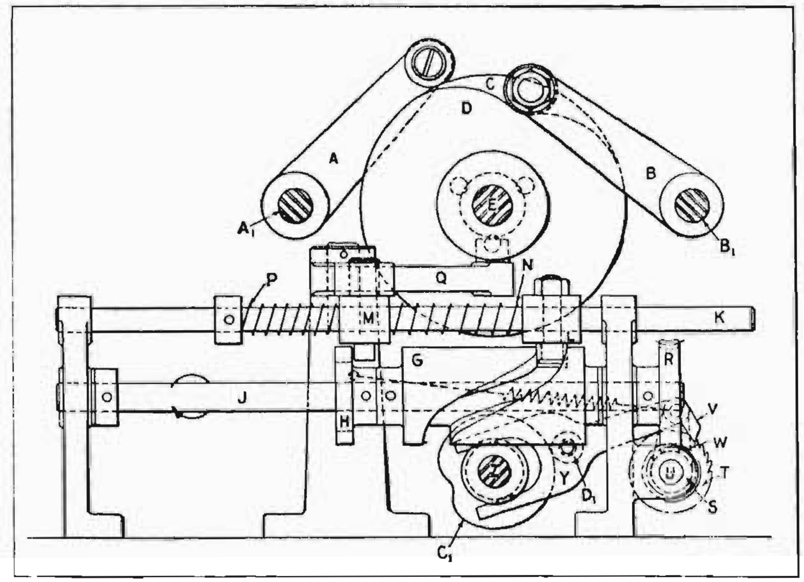


Fig. 15. Side View of Mechanism for Shifting Packing Action of Mechanisms on Shafts A_1 and B_1.

thus rock its pivot shaft. Upon completion of the cycle of five hundred strokes, plate F is shifted from one cam into contact with the other, thus shifting the packing cycle from arm A to arm B, or vice versa.

This shifting is accomplished by means of drum cam G and notched disk H, which are both pinned to shaft J. The follower carrier L for the drum cam is pinned to sliding shaft K, while block M and compression springs N and P

are free to move on this shaft. The follower carrier and block M are both free to slide on the parallel shaft K_1, which is secured to the frame of the mechanism.

Two notches, 180 degrees apart, in hardened disk H provide a sliding fit for the lower portion of block M, which is square in cross-section and also hardened. As the drum cam revolves in the direction indicated by the arrow, with the parts of the mechanism in the relative positions shown,

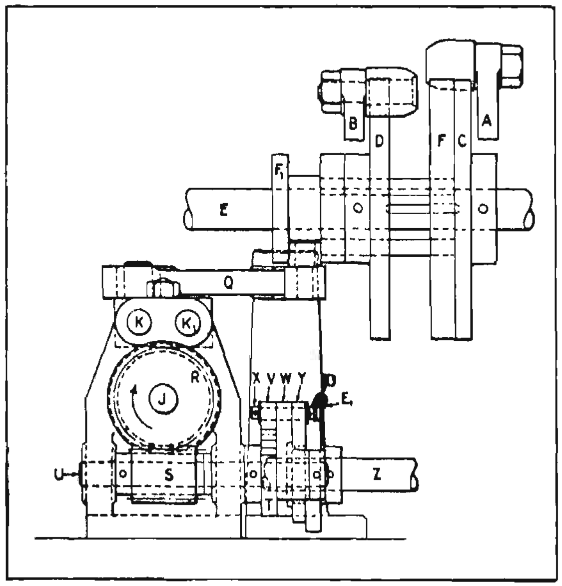


Fig. 14. End View of Mechanism Shown in Fig. 15.

the follower will be forced to move to the left. This rotation will prevent block M from moving to the left and entering the notch in the top of disk H. Thus, compression spring N will become loaded during a half revolution of the drum cam.

At this point, the second notch, shown at the bottom of disk H, becomes aligned with the block. Spring N then unloads, moving the block through the notch and thus rotating

bellcrank Q, one arm of which is pinned to the upper portion of the block. The other arm of this bellcrank is provided with a follower located between the flanges of spool F_1. The spool is connected to plate F by three rods, which run through cam D. Thus, the rotation of the bellcrank causes plate F to be shifted from one cam to the other.

Further rotation of the drum cam brings block M into contact with the opposite side of disk H, thus loading spring P. When the block reaches alignment with the notch shown at the top of the disk, it will pass through the disk, thus turning bellcrank Q in the opposite direction and again shifting plate F.

The purpose of the gearing at the right-hand end of shaft J is to turn this shaft one revolution for every thousand revolutions of shaft E. One revolution of shaft J, as previously explained, imparts two shifts to plate F, thus resulting in the desired packing of five hundred sheets in each tray.

Worm-wheel R, which is pinned to shaft J, meshes with worm S. This worm and ratchet T are pinned to shaft U. The ratchet is driven by pawl V through arm W, which turns freely on shaft U. In the outer end of arm W is securely fixed a pin X, which serves as a pivot for the pawl and cam-follower yoke Y.

Cam C_1 is secured to shaft Z, which rotates at the same speed as shaft E. Roller D_1, which is held in contact with this cam by spring E_1, is fastened to yoke Y. One tooth of ratchet T is thus indexed per revolution of shaft Z or E. The number of teeth on ratchet T and worm-wheel R is such that shaft J will make one complete revolution for every thousand revolutions of shaft Z or E.

Hand Control for Reciprocating Slide.—To facilitate setting up one type of wire crimping machine, the reciprocating slide that feeds the wire stock to the machine is equipped with a hand-lever which, when shifted, not only stops the movement of the slide at any point in its stroke, but

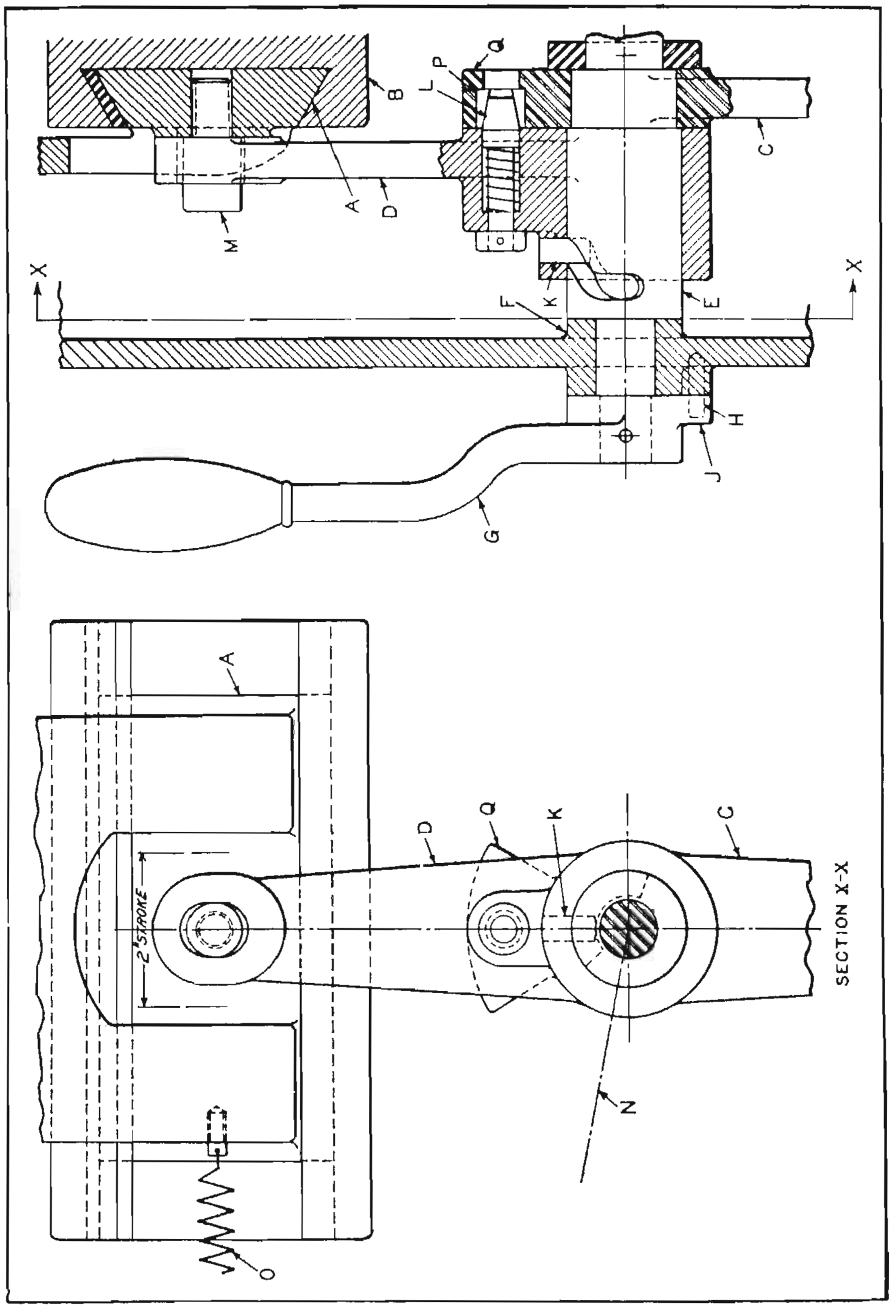


Fig. 17. Hand-lever Control Mechanism for Stopping Reciprocating Slide at any Point in its Stroke and Returning it to the Starting Position.

instantly returns the slide to its starting position. The arrangement for obtaining this result is shown in Fig. 17.

The feed-slide is indicated at *A*. It operates in the dovetail guide *B*, which is cast integral with the machine base. This slide receives its motion from the oscillating lever *C* through arm *D*, connected to the slide by stud *M*. Both arm *D* and lever *C* are fulcrumed on the shaft *E*. A bearing *F* on the machine casing supports one end of this shaft, and the other end is supported in another bearing (not shown) at the right. On the hub of hand-lever *G*, pinned to the end of the shaft, is cast a lug *J*, located between two pins *H* in the casing. With this arrangement, the angular movement of lever *G* is limited to about 80 degrees.

There is a cam slot in the body of shaft *E* which is engaged by a pin *K* in the hub of arm *D*. Also, in the hub of this arm is a spring-actuated plunger *L*, which locks the arm to lever *C*, causing these members to oscillate together.

However, if the operator wishes to stop the slide at, say, the center of its stroke toward the right and cause it to return to its starting position at the left, the hand-lever *G*, which at this time is in a vertical position, is swung toward the left until it coincides with the center line *N*. As this lever is shifted, shaft *E* turns with it, causing the cam slot to carry pin *K* with arm *D* toward the left (see right-hand view) until plunger *L* is entirely disengaged from lever *C*. At this point, coil spring *O* carries the slide back to its starting position at the left. Arm *D* remains connected to the slide, as pin *M* is long enough to compensate for the axial movement of the arm.

Now, if the operator wishes to start the slide again, the hand-lever is shifted back to its vertical position. This causes the cam groove to force arm *D* back into the position shown. Plunger *L* is held flush with the arm hub by the segment *Q* on lever *C* until the bushing and plunger are in alignment. The plunger then enters the bushing and locks the arm and lever, causing the slide to reciprocate.

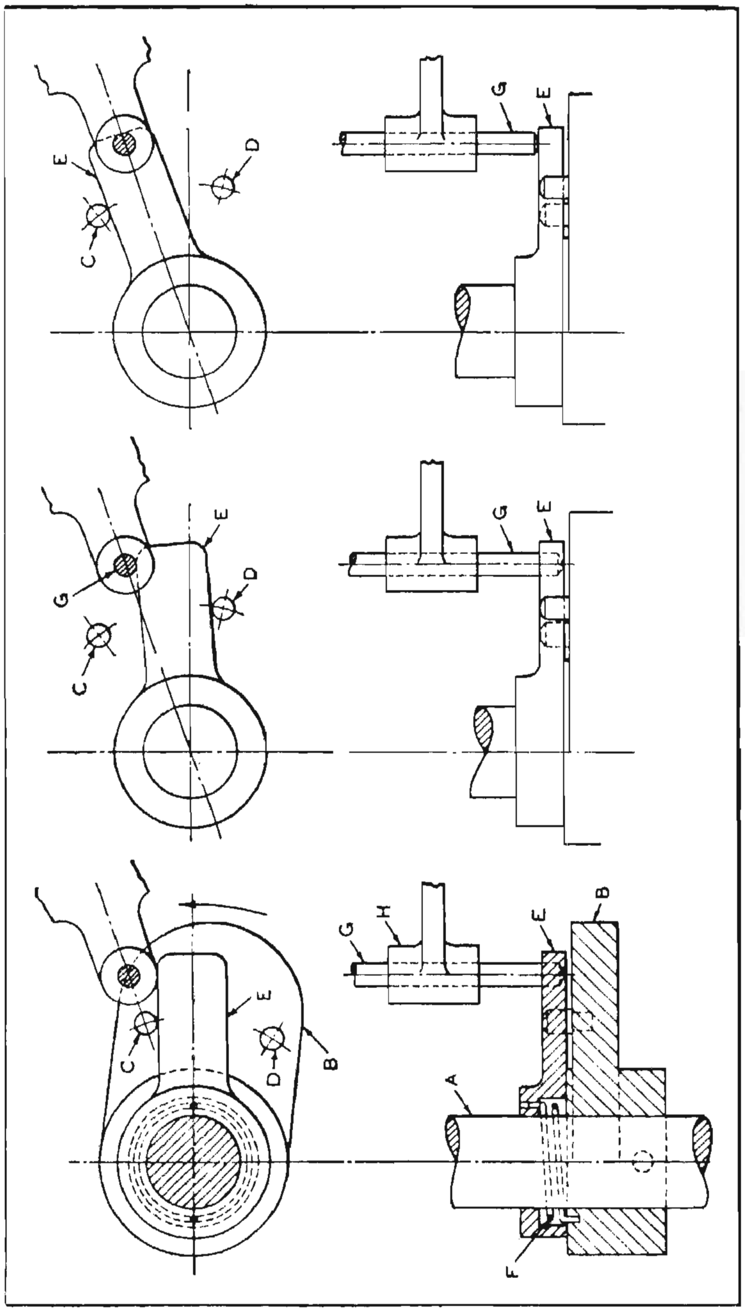


Fig. 18. (Left) Members B and E in Rotating Position With Rod G Lowered to Stop Rotation. (Center) Members B and E in Stationary Position. (Right) Release Rod G Raised to Permit Rotation of Members B and E.

It is important to note that the section of the cam groove engaging the pin *K* when the lever is in its vertical position, is at right angles to the axis of the shaft. This is essential, since at this time, the pin oscillates in the groove, and a helical path, however slight, would impart an undesirable axial movement to the arm for each cycle of lever *C*.

Stop with Safety Catch for Controlling Rotation of Shaft.—A machine that was developed for a special process is equipped with a device for feeding certain ingredients. This feeding device is controlled by a mechanism designed to proportion the amount or quantity of the ingredient fed, one complete revolution of the mechanism serving to feed the correct amount. If the supply of the ingredient has been exhausted or there is an insufficient quantity available in the bin, the mechanism will cease to function.

The feeding device (not shown) is attached to shaft *A*, Fig. 18, left-hand diagram. It is permitted to make one revolution only at a time. When the mechanism is once released for rotation, however, an arrangement is provided whereby the system is held open for operation at any time, regardless of whether or not it is prepared for rotation.

Segment *B* of the device is fastened to shaft *A*. The segment is provided with two pins *C* and *D*. The arm *E* is located between the pins and is attached to segment *B* by means of a small coil spring *F*. Spring *F* is arranged to force arm *E* against pin *C*. The release rod *G* slides vertically in the bearing *H*. The direction in which segment *B* rotates is indicated by the arrow.

The positions of the elements when the device is rotating are shown in Fig. 18, left-hand diagram. It will be noted that arm *E* rests against pin *C* under the action of spring *F*. Releasing rod *G* is in position to come in contact with arm *E* as rotation continues. When the device is stationary, the elements are positioned as shown in Fig. 18, center diagram, with arm *E* resting against pin *D*. In this case, arm *E* is forced away from pin *C* by the action of rod G, combined with the tendency of segment B to continue rotation.

Fig. 18, right-hand diagram, shows the positions of the various elements at the moment when release rod G is raised and the device is free to rotate. It will be noted that arm E has moved away from pin D under the action of spring F and is now resting against pin C. Any attempt to stop the rotation is now prevented by arm E, which is located under releasing rod G. Thus rotation of segment B and arm E will be certain to take place as soon as shaft A begins to rotate. The operator cannot stop the rotation or prevent its taking place unless he interferes with the action of arm E.

The operator can tell at a glance whether or not any of the ingredient has been fed into the machine by noting the position of the elements of the device. For instance, if they are set in the positions indicated in Fig. 18, center diagram, he knows immediately that rotation took place after he raised rod G because the device is now locked. If the device is as shown in Fig. 18, right-hand diagram, he is certain that rotation has not taken place and that no ingredient has been fed into the machine.

CHAPTER 15

Automatic Work Feeding and Transfer Mechanisms

This chapter deals with the automatic delivery of workpieces in the proper position for the operation to be performed on them. The mechanisms described include an automatic magazine-feed attachment for a centerless grinder; an automatic hopper feed for small cylindrical parts; an arrangement for inserting and heading tubular rivets; devices for filling containers and applying covers; an intermittent feeding mechanism designed to operate two slides from one cam; a rapid-motion, short-stroke wire-feeding mechanism; a novel intermittent feeding mechanism; and several transfer mechanisms.

Other automatic feeding mechanisms are described in Chapter 16, Vol. I and Chapter 14, Vol. II of Ingenious Mechanisms.

Automatic Magazine-Feed Attachment for Centerless Grinder.—The automatic magazine-feed attachment shown in Fig. 1 was designed for grinding the part or component shown in the detail view at W. This part, because of the shoulder, must be fed downward in the correct lateral position for the shoulders to pass into the clearance groove provided in both wheels. To accomplish this, a mechanical attachment is necessary. The illustration shows the attachment in position on the grinding machine, mounted on the side of the wheel-truing device. It can be easily adjusted to suit the position of the grinding and control wheels, both longitudinally and crosswise. The vertical position is permanent, and needs no further adjustment after the initial setting.

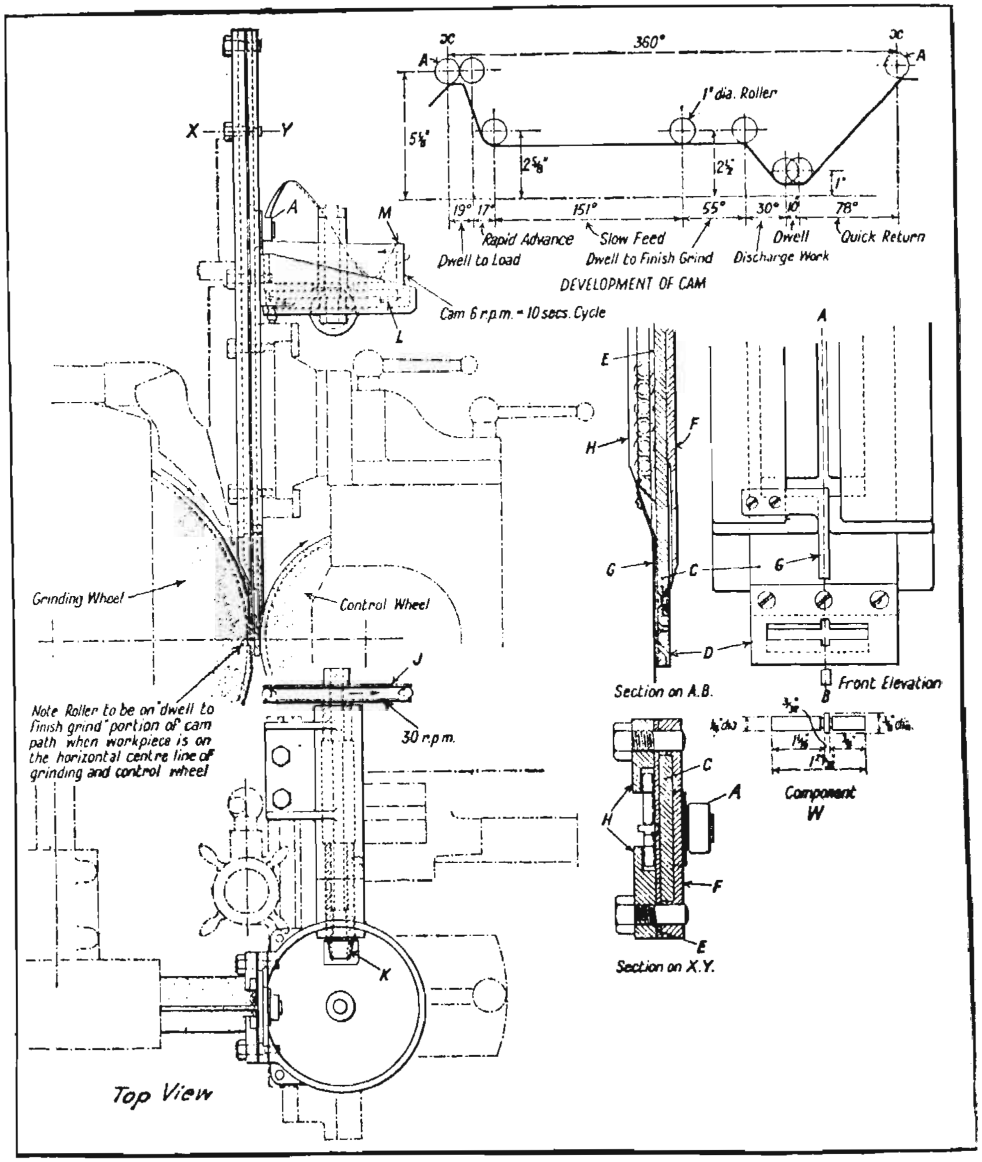


Fig. 1. Details of Automatic Mechanism Feed Designed for Centerless Grinder.

The section on the center line *A—B* shows an enlarged partial view of the sliding member *C*, the work-carrier *D*, which is made of hardened steel; the spacer plates *E;* the sliding member guide *F;* the spring finger *G*, which keeps the work-piece from falling forward when approaching the grinding wheel; and the strips *H*, which form the magazine into which the work-pieces are loaded.

The section *X—Y* shows an enlarged plan view of the magazine and its component parts. A piece to be ground is shown by dot-and-dash lines in the feeding position. The enlarged partial view of the front elevation (section *A—B*) shows the hardened-steel work-holder or carrier *D* which is attached to the sliding member *C* and the slot where the work-piece is retained and guided. Also shown is the position of the spring finger *G* in relation to the work-holder. The action of the whole mechanism is more readily understood if the cam development diagram at the top of the illustration is carefully inspected. This diagram shows clearly all positions of the sliding member *C*. It is necessary that the weight of the sliding member be sufficient to keep roller *A* in contact with the face of cam *M*.

Pulley *J* is driven by a belt running from the control-wheel shaft. The bevel pinion *K* is integral with the pulley shaft and engages bevel gear ring *L* attached to the under side of cam *M*, thus causing it to rotate in the direction and at the speed indicated.

The operator drops the work into the magazine at the open end; and when the sliding member *C* is in the top position, the slot in the work-holder *D* coincides with the curved angular path at the bottom end of the magazine strip *H*. The piece at the bottom of the magazine then rolls forward into holder *D*, which is carried downward on the rapid advance portion of the stroke actuated by cam *M* until it reaches the grinding position, where it is finished to size in the predetermined time period indicated on the cam lay-out.

Proceeding from the "dwell to finish-grind" portion on

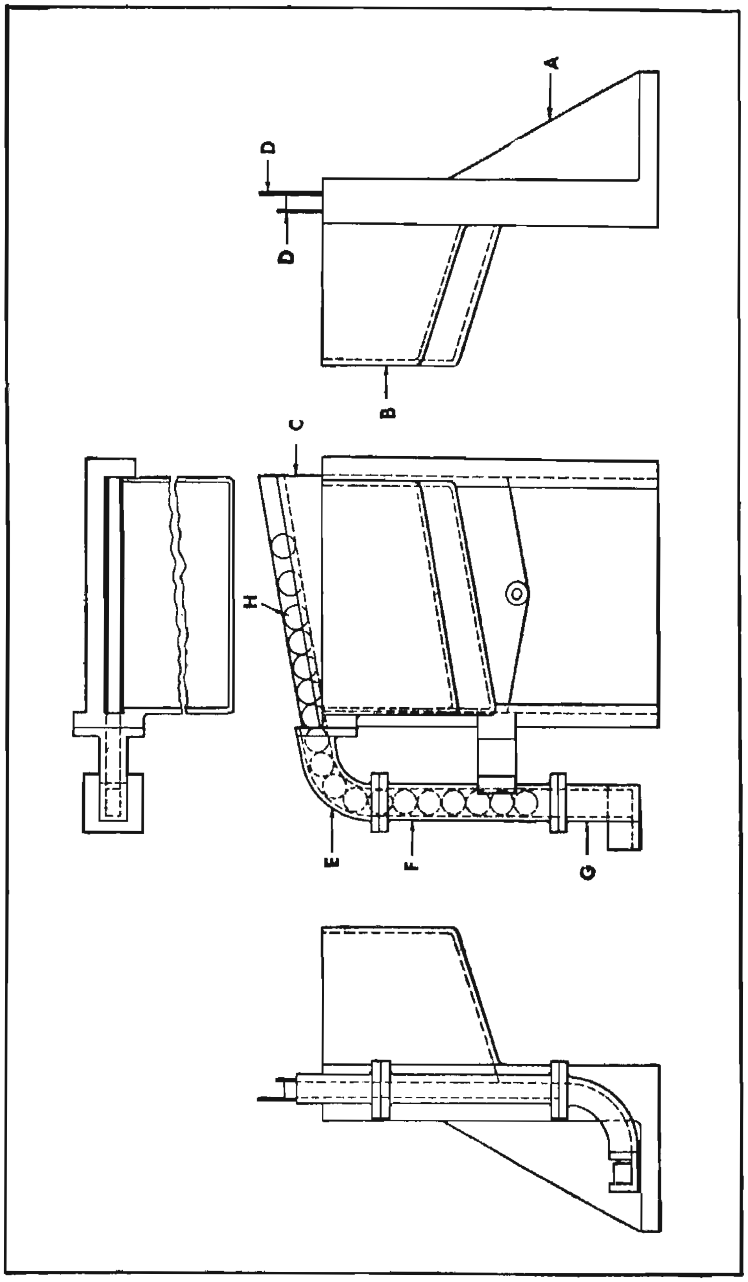


Fig. 2. Hopper Feed with Reciprocating Slide C Arranged for Handling Cylindrical Parts.

cam *M*, the sliding member *C* passes to the lowest position in its travel. During this period, the work is carried clear of the grinding and control wheels, and is free to drop out at the control wheel side of the work-holder into a receiving box. Sliding member *C* is now returned to the highest point of the cam path, which completes the time cycle per piece.

Automatic Hopper Feeds for Small Cylindrical Parts.— Rings or cylindrical parts can be automatically fed from a hopper by means of the mechanism shown in Fig. 2. The hopper shown in Fig. 2 is used to feed cylindrical pieces 1 1/4 inches in diameter by 1 inch high at the rate of forty per minute. This hopper consists essentially of a base *A*; a bowl *B*, in which the parts are placed; and a vertical reciprocating slide *C*. Slide *C* has two steel strips *D* attached to its sides. The strip nearest the base of the hopper is made higher than the one adjacent to the bowl. The slide has a vertical reciprocating movement of five strokes per minute. At the bottom of the stroke which aligns the slide with the bottom of the bowl, the work-pieces fall into the groove formed by strips *D* on slide *C*.

Slide *C* may be reciprocated vertically by means of a crank and slotted cross-head, or a Scotch yoke type of mechanism, which will permit the slide to dwell slightly at the lower and upper positions of its stroke, thus allowing the parts to fall into or roll out of the slide. When the slide reaches the upper end of the stroke, the pieces roll into the covered chute *E*, down the tube *F*, and into elbow *G*. The elbow changes the path of the pieces 90 degrees, and delivers each piece resting on a flat face. The weight of the parts following keeps the leading part against a stop. This part is fed by means of a pusher mechanism similar to the one shown in Fig. 3.

The hopper shown in Fig. 3, which is used for feeding brass rings, is similar to the one just described. However, the work-pieces are fed vertically instead of horizontally.

Details of the feeding mechanism are shown in the illustration. When slide *C* reaches the upper end of the stroke, the parts roll into the vertical covered chute *J*. As the slide moves down, roll *K* comes in contact with lever *L*. This action causes link *M* to move plunger *P* up, pushing one or more parts above the spring *S*. The spring supports the column of rings, which is advanced by each stroke of the slide.

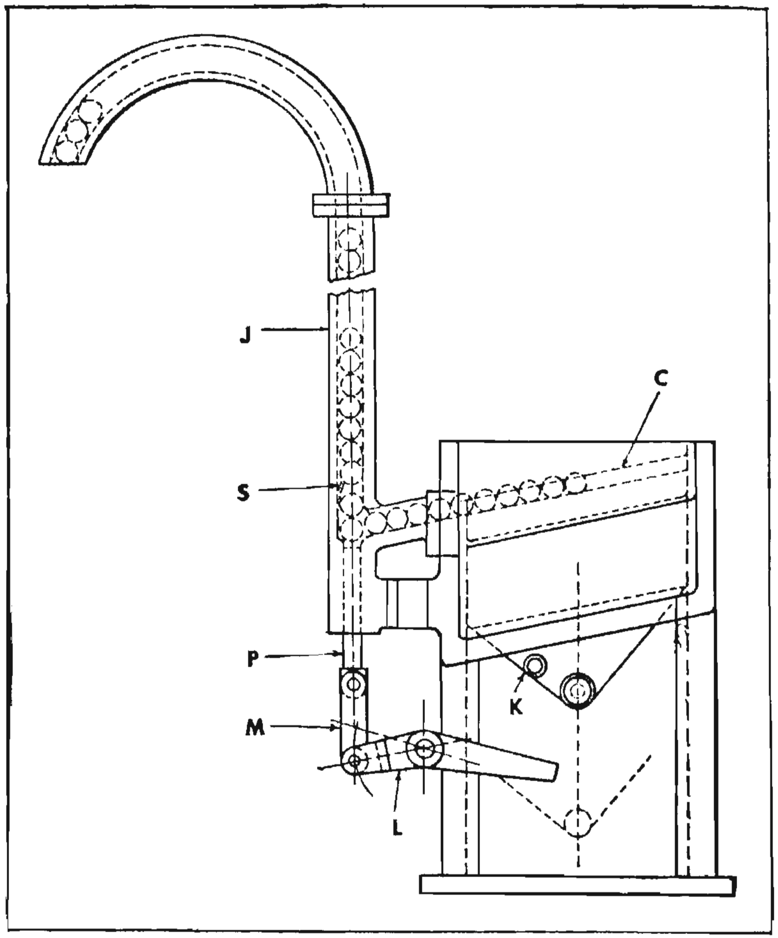


Fig. 3. Arrangement Designed for Feeding Rings Vertically from a Hopper.

Mechanism for Inserting and Heading Tubular Rivets.—The operating mechanism of the Chobert magazine riveting machine employed in a British bomber manufacturing plant is shown in Fig. 4. The sectional views at *X* show the successive stages in the riveting operation. The machine employs a pull-rod with a dolly or conical expanding tool *F* at one end, the diameter of which is slightly larger than that of the main bore through the tubular rivet. As the dolly is pulled through the rivet, it forms a head at the inner end and expands the shank so that it is a tight fit in the hole.

The machine has a hand-grip at *A* and a housing *B* for the crank-operated gear and cam mechanism whereby a sliding movement is imparted to sleeve *C*. Shroud *D* is attached to the end of this sleeve and moves with it to operate the jaws of chuck *E*.

The pull-rod or dolly *F* extends through the barrel of the machine and is held fast by jaws at *G*. Pressure on the jaws is applied by screw *H*, the arrangement permitting the rod to be removed when it is necessary to load the rivets, which are slipped over or threaded on the rod with the flanged ends toward the rear of the machine, as shown. A shouldered check-spring *K* is next threaded on the rod, the small end of which is inserted in the barrel, passed through the center of feed unit *L*, and gripped by jaws *G*. With chuck *E* closed, the first rivet is exposed with its flange seated against the chuck end.

The first rivet is inserted in one of the drilled holes in the gusset and tubular member, the dolly and end portion of the rivet projecting into the tube. Three turns of crank *M* are required to complete the riveting operation and reset the machine.

As mentioned, the cam mechanism in housing *B* imparts a forward movement to sleeve *C*, and with it, to shroud *D* and chuck *E*, the rod and rivets remaining stationary. Any movement of crank *M* when the rivet and chuck are pressed

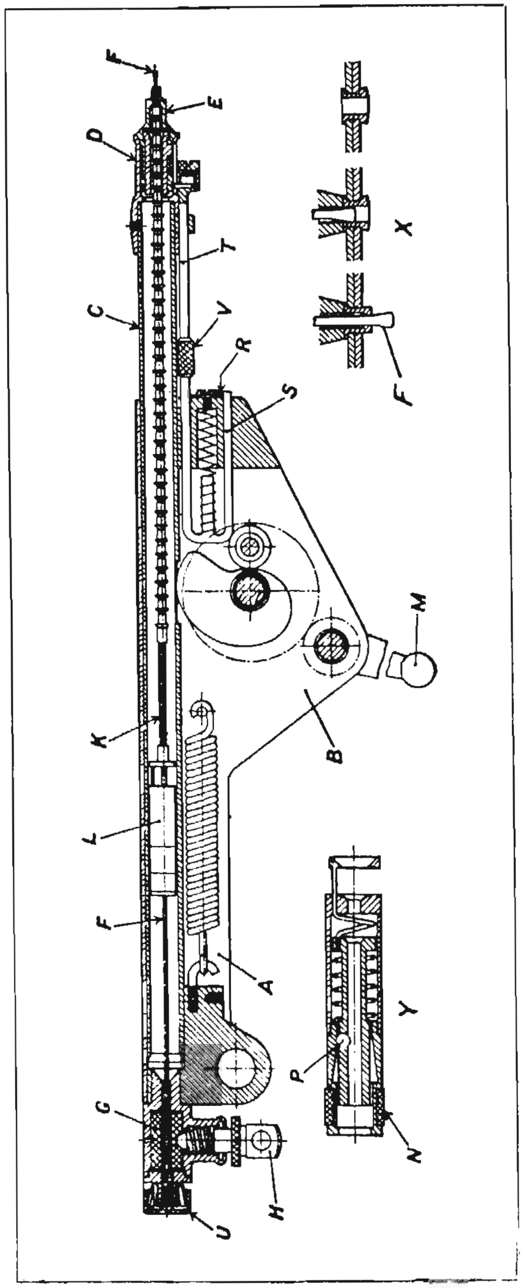


Fig. 4. Cross-Section of Chobert Riveting Machine Showing Essential Features of Operating Mechanism.

against the work will cause the body of the machine, and with it the rod, to retire. Thus, the dolly is withdrawn, completing the riveting operation as explained. This occurs during the first turn of the crank, after which the riveting machine is withdrawn slightly from the work.

During the second and third turns of the crank, the chuck jaws advance, permitting the passage of the next rivet, and close behind it. The machine is then ready for operation again. When operated by an experienced worker, the machine will close 1200 rivets per hour.

The feed unit of the machine is shown to a larger scale in the cross-section view at *Y*. A split ring *N* is a good sliding fit in sleeve *C*. Ball *P* rides to the large end of the taper bore shown when rod *F* is in position and is moved to the left during the riveting stroke. On the return or loading stroke, ball *P* is forced down the taper so that it grips rod *F*, the movement being assisted by the coil spring, and the entire unit is carried forward through a distance corresponding to the length of one rivet.

It is necessary to reload the rod when the magazine is empty, and to do this, jaws *G* are released by loosening screw *H*. Crank *M* is then given 2 3/4 turns to open the chuck jaws at *E*. With the jaws open, stop *R* is moved to engage a slot in the portion *S* of control rod *T*. Thus, the jaws are locked in the open position to permit the withdrawal of the empty rod and the insertion of another rod loaded with rivets. With the loaded rod in position, stop *R* is released, allowing the jaws to close. There should be a clearance of from 0.6 to 1 millimeter between the head of the exposed rivet and the end of the chuck jaws; if this is not the case, adjustment can be effected through the medium of cap-screw *U*, the threaded portion of which engages the end of rod *F*.

Mechanisms Designed for Filling Containers and Applying Covers.—The mechanisms here illustrated are employed in an automatic machine that places twelve pencil leads in a box and pushes the box cap in place. The box, or container, is molded by the injection process, the dimensions being indicated at the left in Fig. 5. The selecting mechanism, shown at the right in Fig. 5, takes the boxes from a hopper and presents them to the machine open end forward. The revolving disk carries hooks, so spaced as to allow a single box to be fed by each. If a box is in the wrong position, the hook enters its open end and retains it until the rubber-faced rotating roller A returns it to the

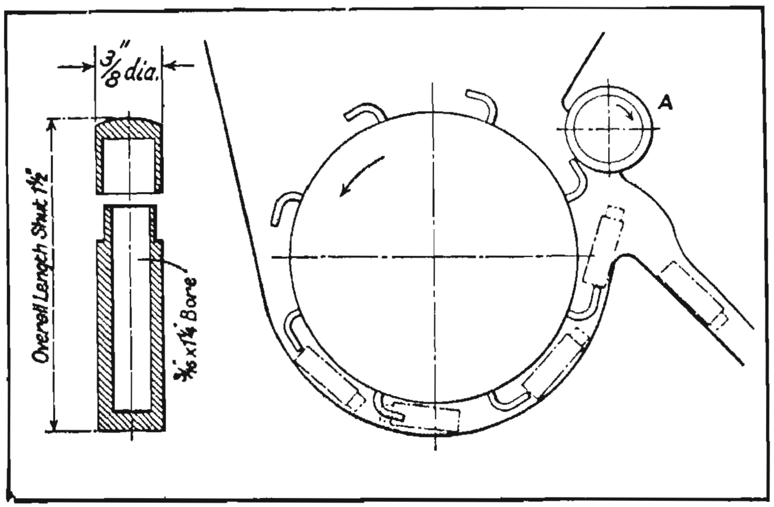


Fig. 5. (Left) Molded Box for Holding Pencil Leads (Right) Mechanism for Feeding Boxes Open End Forward.

hopper. Boxes fed correctly tip over into the delivery feed and slide toward the loading machine.

The continuous belt conveyor, shown in Fig. 6, has successive stations to which boxes, leads, and caps are brought in correct synchronism and quantities. Falling from the selector shown at the right in Fig. 5, the boxes drop into a single-column, vertical stack, the lower end of which is open at the sides. This opening faces one side of the belt conveyor, Fig. 6, which carries properly spaced finger clips

at regular intervals. An intermittent feed of the conveyor belt is obtained by means of a Geneva mechanism. The driver of the Geneva motion carries a face-cam that is set to impart a sharp blow to the lowest box in the stack during the stationary period of the belt, propelling the box into the spring clip on the belt. The belt then transports the box to the lead-filling station.

The lead-filling station, shown in Fig. 7, includes a counting and feeding mechanism. A circular shutter valve is

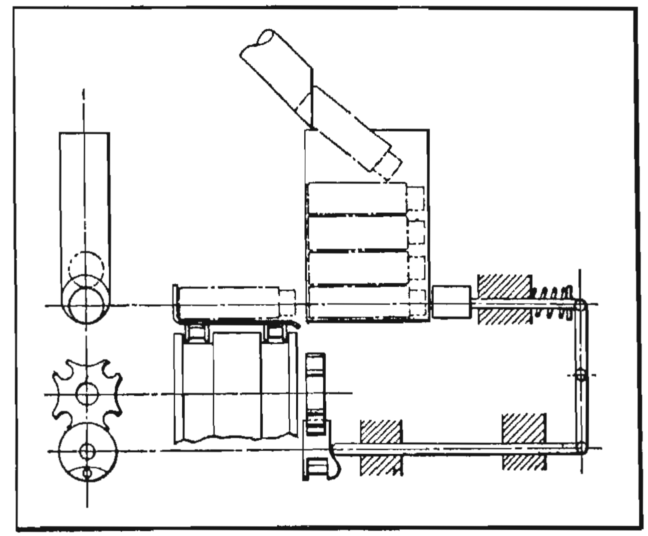


Fig. 6. Arrangement for Feeding Boxes Intermittently on Belt Conveyor.

employed to take twelve leads from the hopper. This valve contains only two rows between its shutters, which, in this case, are open-ended hollow cylinders provided with suitable ports. Only a sector of the hollow cylinders is required for the counting operation, but complete cylinders are employed for the sake of simplicity and rigidity of construction. These cylinders are oscillated by the lever arm, the lower shutter opening as the upper one closes. No overlap is shown in the illustration, but a certain amount of overlap is necessary, and this is provided by increasing the stroke of the operating lever.

From the valve, a dozen leads fall into the guide chamber, which is bored rather smaller in diameter than the boxes to be fed. The boxes are carried along the belt until they register accurately with the guide chamber. When in this position, a constant accelerating drum type cam moves a plunger forward and backward, feeding a supply of leads into the box without shock.

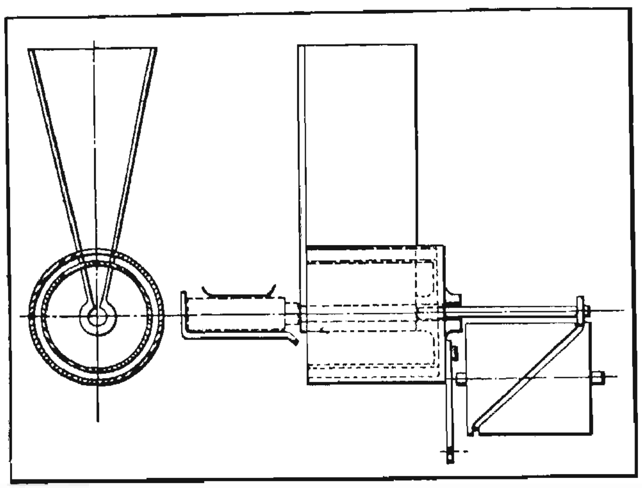


Fig. 7. Mechanism Employed to Select Correct Number of Pencil Leads and Feed them into Box.

The belt is next traversed to bring the box to the cap or cover station, where the cover is fed into position by a selector similar to the one used for the boxes. The "hand" of the selector must, however, be reversed, as the open end of the cover must be presented in a position opposite to that in which the boxes are fed to the conveyor belt. This station is shown in Fig. 9. The caps are fed down a tube, rolling out though an open incline into the single stack container which has open sides. The feeding mechanism is similar

to the one used for the boxes, although a more gradually accelerated thrust is applied by the face-cam. The conveyor chain belt is kept at the proper tension to insure accurate register at the different stations. The belt passes a deflector which ejects the filled boxes.

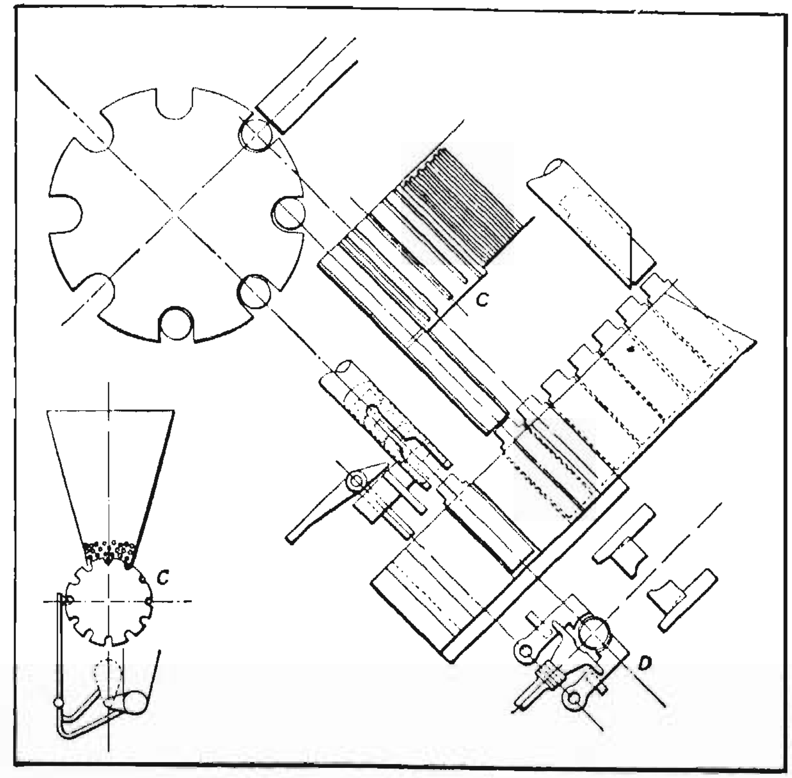


Fig. 8. Drum-type Mechanism Designed for Loading Pencil Leads into Boxes.

Before describing the alternative loading mechanism shown in Fig. 8, attention is called to a feature, without which much loss of time might be experienced in the high-speed automatic operation of the mechanisms described; namely, evidence that the feeding mechanism has functioned properly in placing the required number of leads in the container. The simplest way to detect empty boxes is by weight; loaded holders or boxes, on rolling down a flexible shelf, fall sooner than the empty ones. A dividing line, correctly positioned, is therefore used to separate the filled and unfilled boxes, thus providing a simple means for this inspection.

In the alternative mechanism, shown in Fig. 8, the belt is replaced by an inclined revolving drum. The selected boxes fall into a single-column chute. When a slot in the drum reaches a position opposite the chute, one box drops

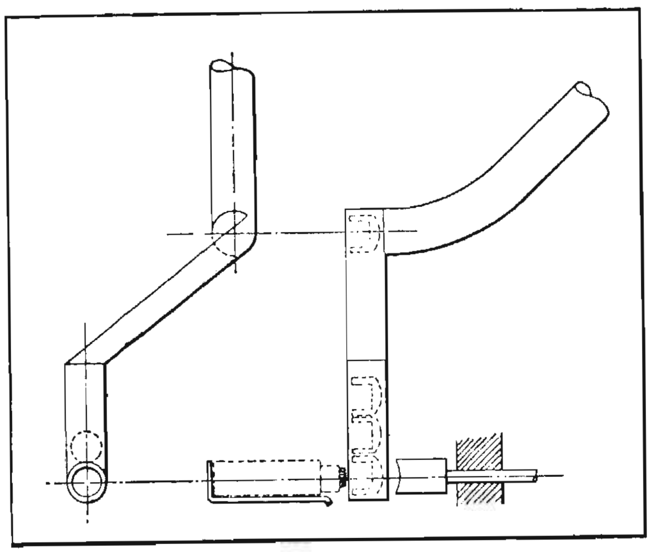


Fig. 9. Mechanism for Placing Covers on Filled Boxes.

into place and is carried around to the lead-filling station. The leads are also counted out by a revolving slotted drum *C*, two of the slots of which are extended beyond the others to operate the shutter movement that closes the inclined trough into which the leads fall from the drum.

At the capping station, selected caps slide down the supply tube, which has a spring arrangement designed to catch the lowest cap just before it reaches the exit. At the correct instant, the capping lever is rocked about its bearing, clos-

ing two jaws on the sides of the lowest cap while carrying it forward. A plan view of this mechanism is shown at *D* below the drum, cam-slides being provided to close the jaws during the forward movement. As the main drum revolves to the next position, the loaded boxes fall from their slots and again pass a mechanism that is arranged to detect unfilled boxes in the manner previously described.

Mechanism for Feeding Wooden Pegs into Magazine, Large Ends Foremost.—The purpose of the device shown in Fig. 10 is to feed round wooden pegs *A* into the magazine *B* with their large ends downward to facilitate assembling them in the product. The wooden pegs are 1 1/4 inches long and 1/4 inch in diameter for a length of 1 inch, and 3/16 inch in diameter for a length of 1/4 inch. These pegs are fed along a chute *C* into the selector cavity *D*, as shown in the plan view. A constant pressure is maintained against the line of pegs in chute *C*, so that when the first peg in the line is removed another moves forward to replace it.

The plate *E* is moved backward and forward, as indicated by the arrows at *F*, being actuated by means of a cam and return spring, not shown. Plate *E* carries a block *G* and has a rectangular slot or hole *H* cut through it which opens into the chute leading to the magazine *B*. Block *G* pushes the first peg in line to the opposite side of the cavity, and at the same time, temporarily prevents any other pegs from entering the cavity. The rectangular slot serves to open and close the opening in the floor of the cavity above the magazine chute. It will be noted that the slot in the slide is slightly wider than the 1/4-inch diameter of the wooden peg and also that it is shorter than the peg, being about 1 1/16 inches long.

In the right-hand wall of the selector cavity *D* is a notch *J*, as may be seen in the plan view. This notch is slightly wider than the 3/16-inch diameter of the peg but not as wide as the 1/4-inch diameter and is somewhat deeper than the 1/4-inch shoulder on the small end of the peg. Opposite this notch and on the same center line is a similar notch *K*, cut in the end of a movable block *L*, which is made to extend into and be withdrawn from the cavity of the selector by means of a suitable cam and return spring, not shown. The motion of this block is timed relative to that of the plate *E* in such a manner as to produce the operating cycle to be described.

Let it be assumed that all of the parts of the selector are in the position indicated by heavy lines in the illustration, and that the first peg has entered the cavity with its small end pointing backward along the chute *C*. The block *L* is next withdrawn and the slot *H* opens the floor of the cavity into the upper part of the chute leading to the magazine. Now the plate *E* moves forward, carrying the block *G* to the

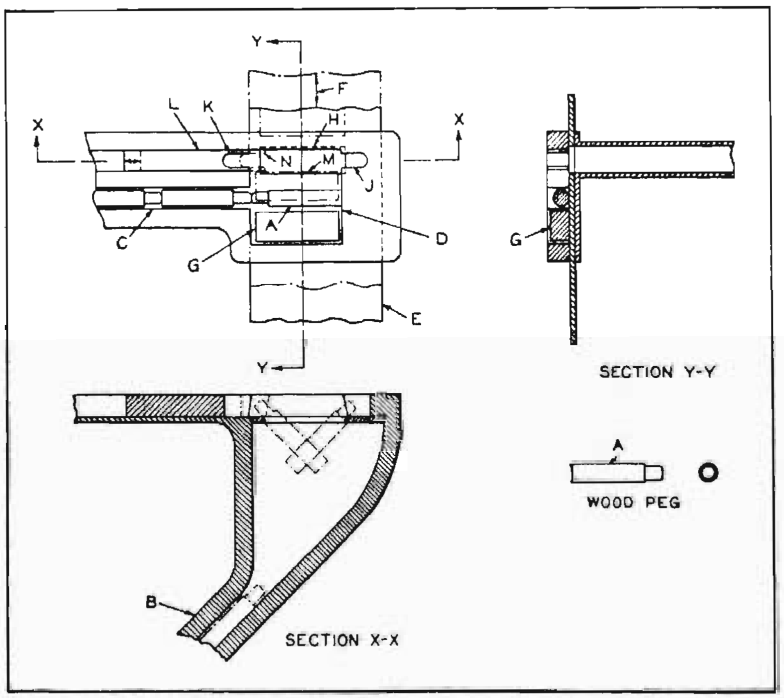


Fig. 10. Mechanism for Feeding Pegs A from Chute C to Mechanism B with Large Ends Foremost.

position indicated by the dot-and-dash lines at *M*. This pushes the first peg across and in line with the two notches *J* and *K* and, at the same time, cuts off the surplus flow of pegs into the cavity.

Next the block *L* moves to the right a distance of 3/16 inch, as indicated by the dot-and-dash lines at *N*, and the notch *K* engages the small end of the peg. The plate *E* then moves back to its original position, the peg being prevented from following the block by the confining notch. As soon as the slot *H* returns to its position above the magazine chute *B*, the peg, being unsupported at all points except under the small end, drops downward, the large end falling into the chute first so that the peg slides into the magazine with the 1/4-inch end to the left. The block *L* then returns to its original position.

The second peg, which enters the cavity small end first, will be considered next. The block *G* also pushes this peg across to the position between the notches, but when block *L* moves into the cavity this time, as the notch *K* is too small to engage the 1/4-inch diameter, the peg is pushed to the right, causing the small end to enter the notch *J*. When the slot *H* returns, it again leaves the peg unsupported at all points except under the small end, so that the large end falls downward and the peg slides along the chute into the magazine with its large end to the left.

This cycle of cam-actuated movements of slides *E* and *L* continuously feeds the pegs into the magazine *B* with their large ends downward. The magazine on which this selector is used has a capacity for handling about fifty units per minute. At this speed, the feeding device functions perfectly, and there appears to be no reason why it could not be operated at a much higher speed.

Intermittent Feeding Mechanism Designed to Operate Two Slides from One Cam.—The mechanism shown in Figs. 11 and 12 was designed to feed a continuous strip of corrugated flat wire stock *W* through a machine for further fabrication. The rapid, positive, intermittent feeding movement required is obtained through the operation of feed-slide *A* and work-gripping slide *B* by a single cam *C*. It is interesting to note that both slides *A* and *B* are operated simultaneously by cam *C* and that slide *B* is mounted in a dovetail groove in slide *A*.

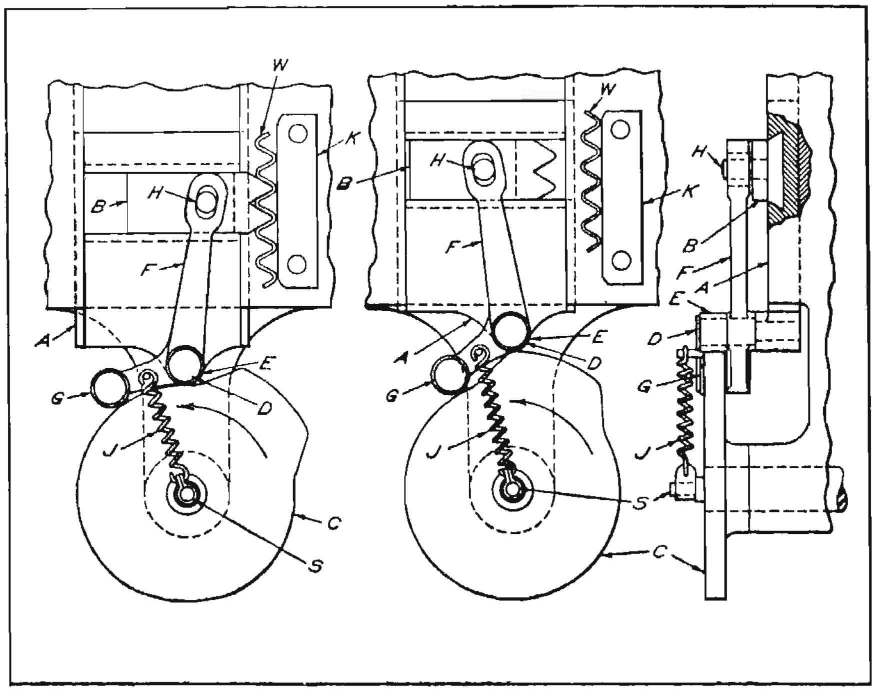


Fig. 11. (Left) Mechanism for Intermittent Feeding of Corrugated Flat Wire Stock W at End of Dwell Period in which Stock is Held Clamped against Block K by Slide B. (Center) Slide B Withdrawn from Work W and Slide A Moved One Corrugation to Rear through Action of Cam C. (Right) Side View of Feeding Mechanism.

Referring to Fig. 11, left-hand diagram, shaft *S*, carrying cam *C*, rotates in the direction indicated by the arrow. Stud *D* on slide *A* carries the cam roller *E* and the bellcrank lever *F*, which is free to oscillate. Lever *F* carries on its short arm the roller *G*, which is free to rotate on its stud. Pin *H* is fixed in slide *B*, and passes through the slot in the upper

end of lever *F*. Spring *J* serves to hold rollers *G* and *E* in contact with cam *C*. Work *W* is passed through the machine in contact with the guide strip indicated at *K*.

In Fig. 11, left-hand diagram, both rollers *G* and *E* are shown in contact with the low portion of cam *C*, which holds the slides in a fixed position during the rest period of the

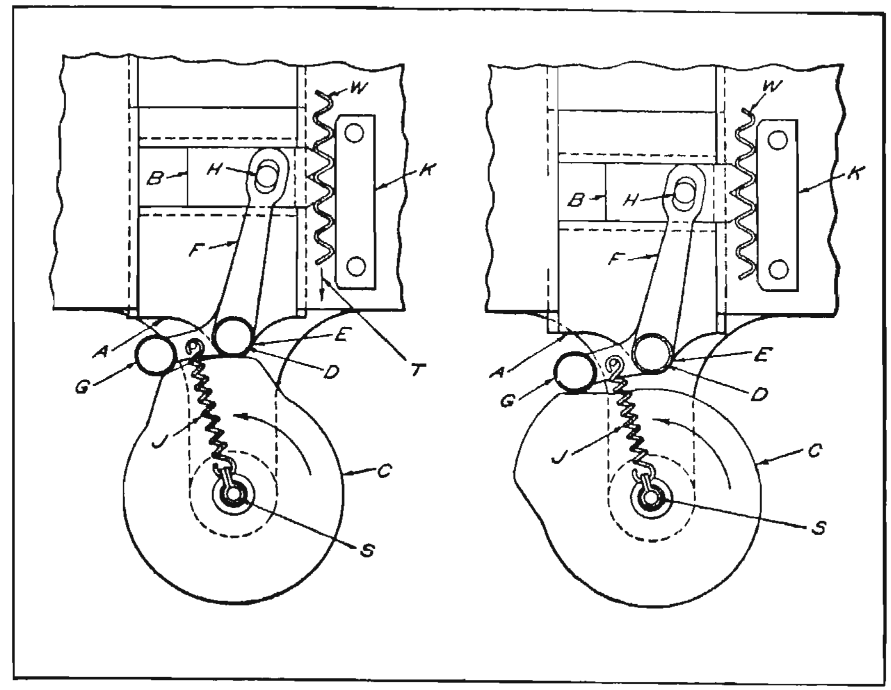


Fig. 12. (Left) Slide B Advanced to Engage Work W Preparatory to Feeding it Forward in Direction Indicated by Arrow T. (Right) Mechanism Engaged in Feeding Work W Forward with Cam C Approaching Dwell Position.

cycle. It is during this rest period that the fabricating operations are performed on the work. The formed end of slide *B* engages the corrugations in the work *W* as shown during this dwell period, holding it in contact with guide strip *K*.

Fig. 11, center diagram, shows the rise or lobe of cam *C* engaging roller *E* and causing slide *A* to move to the rear. Roller *G* has not yet been affected by the action of cam *C*, but the position of lever F has been changed so that, in addition to the rearward movement of slide A, the action of lever F on pin H has caused slide B to be withdrawn from contact with work W. Since the rearward movement of slide A is equal to the pitch or center-to-center distance of the corrugations of the work, no rearward movement is transmitted to work W. The slide B, however, has moved to the rear with slide A, and its forward end is in position to engage another set of corrugations in work W.

Continued rotation of cam C, as shown in Fig. 12, left-hand diagram, causes roller G to rise to the high portion of cam C. As roller E is in the same relative position as shown in Fig. 11, center diagram, there has been no movement of slide A. The change in the position of roller G, however, has caused lever F to pivot on stud D, actuating slide B and causing its forward end to again engage the corrugations in work W. Comparison of the left-hand diagram in Fig. 12 with that in Fig. 11 shows clearly the changes in the position of slide B relative to work W.

Fig. 12, right-hand diagram, shows the mechanism after cam C has passed from under roller E. The pressure of slide B on work W against guide strip K at this stage of the operating cycle prevents any movement of lever F and also prevents roller E from following the contour of cam C. In the position shown in Fig. 12, right-hand diagram, roller G still rests on the "drop" side of the cam lobe. The action of spring J serves to hold roller G in contact with cam C, causing slide A to be drawn forward. This action, in turn, causes slide B to feed work W forward. The forward motion of slide A continues until both rollers G and E come in contact with the low portion of the lobe on cam C, as shown in Fig. 11, left-hand diagram. Until the lobe of cam C again comes in contact with roller E, work W is held firmly in position. With this mechanism, work W is given a rapid, short, intermittent feed with a long rest period between the feeding movements.

Rapid-Motion, Short-Stroke Wire-Feeding Mechanism.—The mechanism shown in Fig. 13 is designed for feeding a strand of wire rapidly through a wire fabricating machine in a series of short strokes, with equal rest periods between strokes. Owing to the high speed of the machine, it was necessary that the mechanism provide a reciprocating move-

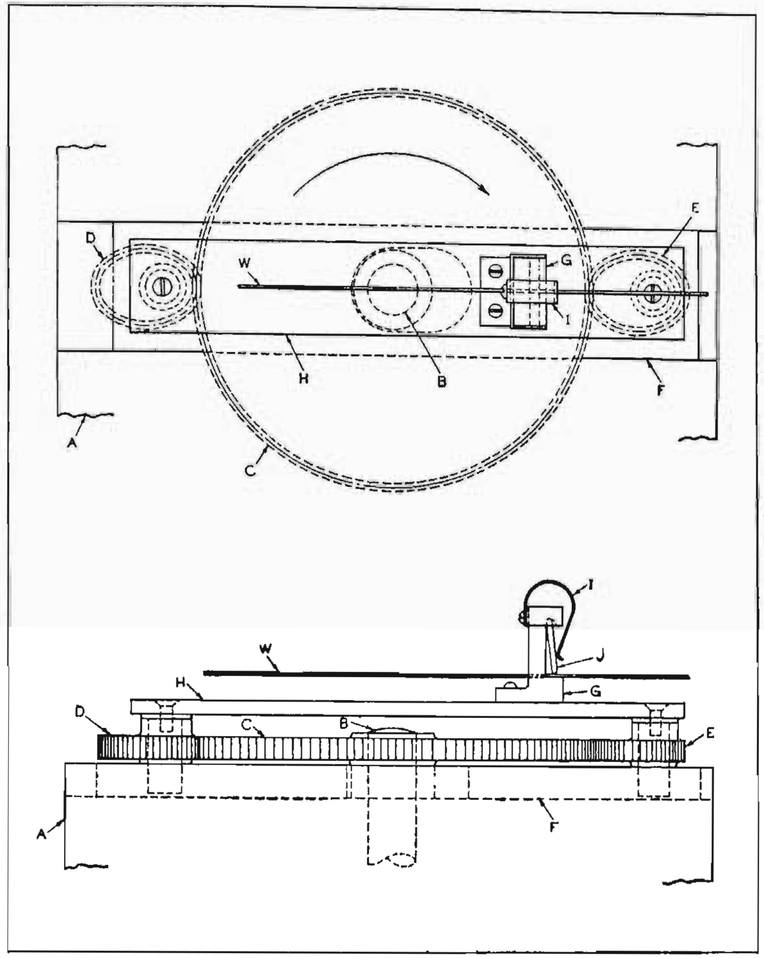


Fig. 13. (Top) Wire Feeding Mechanism Designed for Rapid Short Stroke Motion. (Bottom) End View of Mechanism in which Slide F is at End of Feeding Movement to Right.

ment with a minimum of vibration, and that a positive gripping arrangement be used for feeding the wire.

Referring to Fig. 13, upper diagram, the shaft *B* rotates in the stationary part of the machine *A*, in the direction indicated by the arrow, and carries the gear *C* that is keyed to it. The slide *F* is carried in the part *A*, and is slotted, as shown, to clear the hub of gear *C*. Slide *F* carries two studs on which the elliptical gears *D* and *E* revolve freely, receiving their motion from the gear *C*. In this view, the teeth on

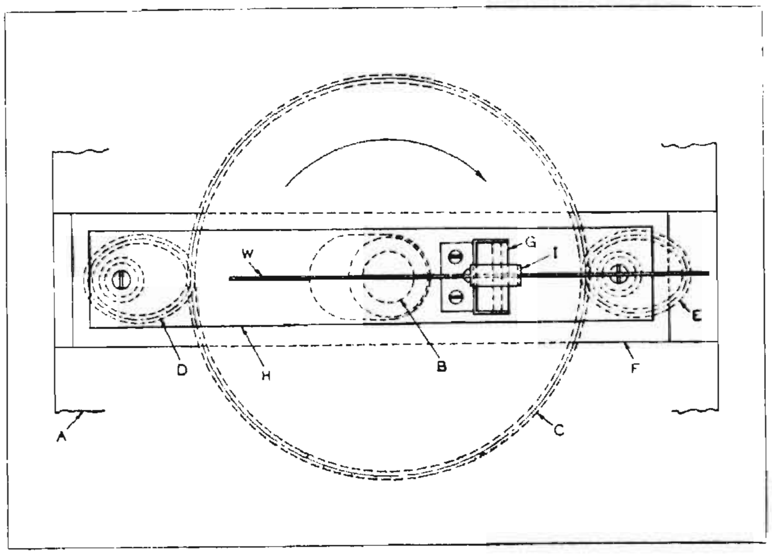


Fig. 14. Wire Feeding Mechanism with Slide F at End of Return Stroke.

the shorter side of the major axis of gear *D* are in mesh with the teeth of gear *C*, while the teeth on the longer side of the major axis of gear *E* mesh with the teeth on the opposite side of gear *C*. In this position, the center-to-center distance between the axes of gears *C* and *E* is greater than that between gears *C* and *D*. As the studs that carry the gears *D* and *E* are fixed in slide *F*, the latter is thrown off center, to the right, with relation to shaft *B*, the slide *F* being at its extreme right-hand position. The bar *H* carries the gripper

mechanism *G*, which feeds wire *W*, as will be explained.

In Fig. 14, gear *C* has made a partial revolution, resulting in a half revolution of gears *D* and *E* and reversing the condition shown in Fig. 13; at this point the slide *F* has been moved to its extreme left-hand position, the distance traveled by the slide being equal to the eccentricity of gears *D* and *E*.

The details of the gripper mechanism are shown in Fig. 13, lower diagram, which is an end view. The bar *H* is carried on the studs that carry gears *D* and *E*, and therefore moves with slide *F*. The wire *W* passes through the part *G*, which is mounted on bar *H*. The wire *W* rests on the ledge of part *G*, against which it is held by the wedging action of plate *J*, the flat spring *I* acting on plate *J* to insure a positive wedging action. As the bar *H* moves to the right, the plate *J* wedges between its seat in part *G* and the wire *W*, gripping the latter firmly, and feeding it into the machine. As the bar *H* moves to the left, the wedging action is destroyed and plate *J* slides over the wire *W*, without transmitting any motion.

Novel Intermittent Feeding Mechanism.—The mechanism shown in Fig. 15 is designed to intermittently index or feed a block chain or film *E* longitudinally from left to right. Each feeding movement advances the chain or film a distance *L*. The indexing is accomplished by a pin *D* which enters equally spaced holes or slots *H* in the chain, rising vertically into one of the slots and moving to the right until the chain has been moved a distance *L*. The pin then moves downward out of contact with the chain. With the simplest arrangement of this mechanism, using only one arm *B*, and without the block-rotating device *M* described later, the chain is fed a distance *L* for each revolution of shaft *S*, the feeding movement taking place during about one-fourth revolution of shaft *S*.

Pin *D* is carried in a square block *C*, which has a pin *G* that guides the movement of block *C* in a rectangular path determined by cam groove J. Pin G also enters a slot in the driving crank B attached to rotating shaft S. Block C is further guided in its rectangular path by rim K on the body A of the mechanism, which also has the cam groove J. Rim K prevents block C from turning about pin G, so that pin D will rise vertically into the slot in chain E for the feeding movement.

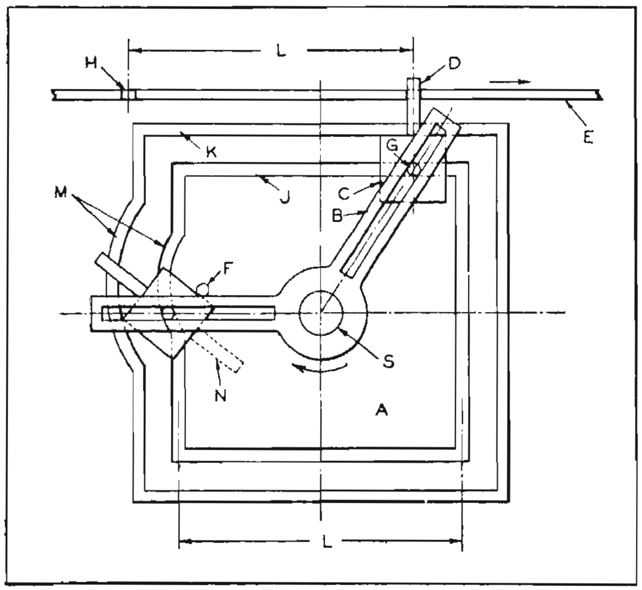


Fig. 15. Mechanism that Imparts Intermittent Movement to Chain E.

By modifying the forms of groove J and rim K as shown at M, and providing a stationary pin F in body A, it is possible to cause block C to rotate through an angle of 90 degrees. With this arrangement and only one block C and one driving crank B, pin D would be brought into the indexing position every fourth revolution of shaft S. By adding another pin N to block C, the chain would be indexed every second revolution of shaft S. Irregularity of the timing of the indexing movements or a variation in the number of indexing movements per revolution of shaft S can be obtained by employing more than one slotted crank B.

A variety of indexing movements can be obtained by varying the design. For example, block C can be fitted with one, two, three, or four pins D, and block-rotating arrangements M can be provided on the bottom and right-hand side of the mechanism.

Feeding Mechanism for Box-Nailing Machine.—The purpose of the device shown in Fig. 16 is to feed wooden boards, two at a time, in a vertical position into a box-nailing machine. The main feature of this feeding mechanism is its positive action in turning the boards from the flat position, in which they are stacked in the hoppers, to the upright position, in which they must be fed to the nailing machine.

The boards, which are 7 inches wide by 9 inches long and from 3/8 to 1/2 inch thick, are fed simultaneously in pairs at the rate of thirty-six pairs a minute. They are first stacked up by hand in the two opposite hoppers A, the two lowest boards being simultaneously carried into intermittently revolving cross-shaped members C by means of lateral feeding mechanisms or "kickers" B. These kickers are operated by the oscillating shaft D through levers E.

Members C, each of which is built up of four L-shaped plates so arranged as to form two passages at right angles to each other, revolve in troughs F, which have apertures G at the bottom for discharging the boards into the chutes H. End pivot shafts I of members C are journaled in an end plate J, and the opposite pivot shafts are journaled in a similar end plate at the opposite end of troughs F. Spur gears K are mounted on shafts I and mesh with the central spur gear L, the latter being attached to the four-notched wheel M, which freely rotates on pivot stud N. With this gearing, by turning wheel M, members C are made to rotate simultaneously.

Wheel M is operated as an ordinary ratchet wheel by pawl P, pivoted at the upper end of rocker Q, which, in

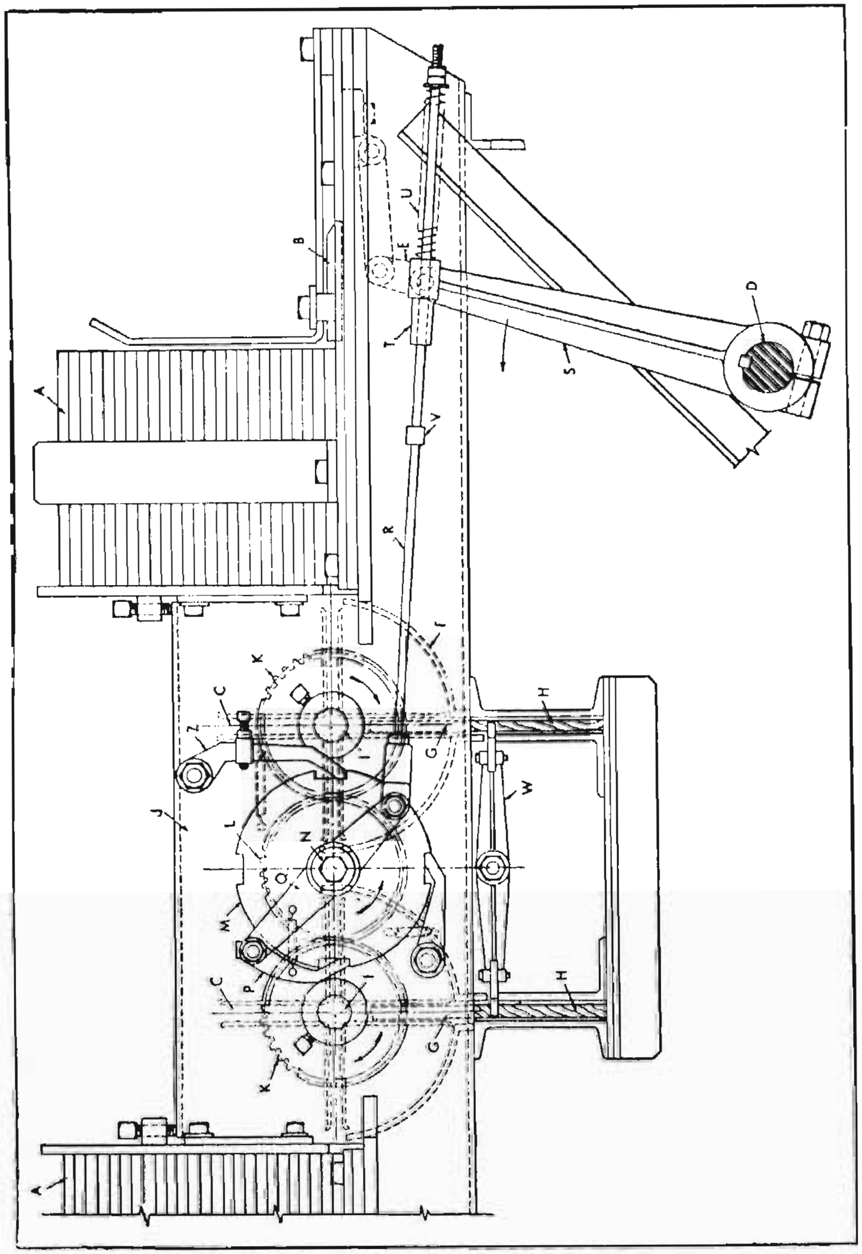


Fig. 16. Mechanism that Feeds Two Boards Simultaneously from Hoppers A to Chutes H.

turn, oscillates on pivot N under the action of the connecting-rod R attached to the lower end of rocker Q. The rod R is attached to the end of lever S, which is mounted also on the oscillating shaft D, by means of block T, through which the connecting-rod is free to slide. A precharged spring U constrains the motion of the connecting-rod in one direction, while a stop V, rigidly attached to the connecting-rod, constrains the motion in the other direction.

As levers E and S move together in the direction shown by the arrow, the lowest wooden boards in the hoppers are introduced into the momentarily stationary members C, while block T approaches stop V, finally reaching it and carrying it forward, together with the connecting-rod. This causes rocker Q to bring pawl P to the point where it will engage the upper notch of wheel M, and at the same time lifts stationary pawl Z, thus setting M free to rotate in the direction of the arrow.

As levers E and S return together to the starting position, kickers B will withdraw from under the stacks of boards, letting them drop under their own weight to the bottom of the hoppers. During this interval, lever S, pressing against spring U, will pull the connecting-rod to the right, causing wheel M to rotate until pawl Z drops into the next notch, thus preventing wheel M from advancing beyond one-quarter of a revolution.

As wheel M stops, lever S will continue to compress spring U until the end of the return stroke. Thus, there is a dwelling period between the completion of the quarter of a revolution of wheel M and the end of the return stroke of arm S, intended to give the wooden boards ample time to drop under their own weight to the bottom of chutes H, through which they are carried to the nailing-machine table by the cross-head W, which moves in synchronism with kickers B.

Mechanism Equipped with Suction Cup for Picking up and Feeding Thin Plates.—Three distinct phases in the sequence of movements imparted to suction cup W of the plate-feeding mechanism shown in Fig. 17 are produced and controlled by a single cam which transmits a sliding movement to the rack or actuating rod R. The three phases of the movement required to pick up plate A and carry it to the position shown at D, as indicated, consist of a vertical movement of suction cup W which raises plate A to the first position clear of the pile; a circular motion which carries the plate to the second or vertical position; and the final

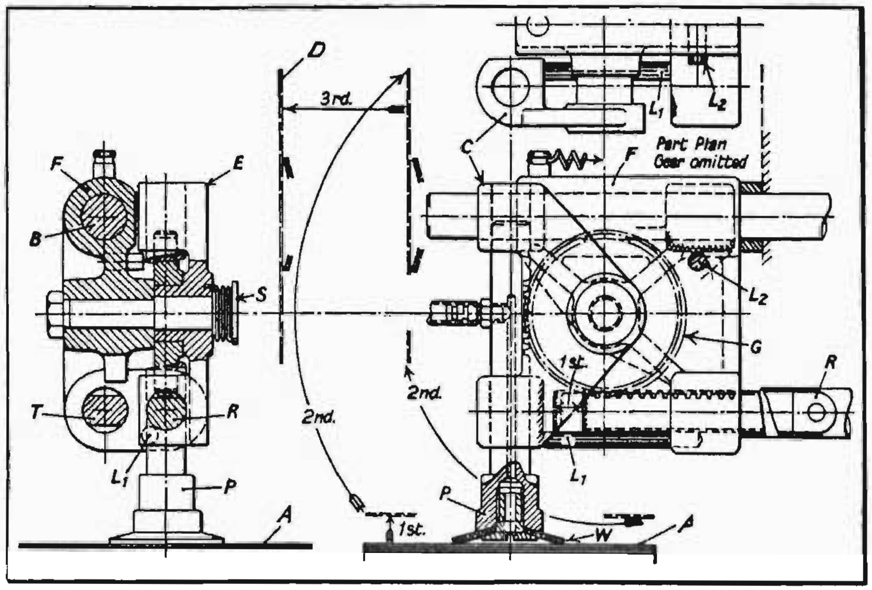


Fig. 17. Mechanism Designed to Pick Up Plate A and Feed It to the Position Indicated at D.

horizontal movement which carries the plate to the third position at D.

The light frame F of the mechanism slides on bar B and a flatted rod T, both of which are firmly fixed in the machine body. The triangular-ribbed frame F has its upper portion formed into a long boss. A central boss carries a horizontal shaft S on which a carrier C is so mounted that it can rotate freely through an angle of 90 degrees. The

maximum counter-clockwise position of the carrier, as shown by the full lines, is determined by a limit pin or stop L_1. A second stop-pin L_2 restricts the maximum clockwise rotation.

A gear G having a pitch diameter of 3 1/2 inches, freely mounted on the hub of carrier C, engages rack teeth on a hollow plunger P and the actuating rod R. Plunger P is provided with a suction cup W at the lower end. The required suction is maintained in cup W by connection with an exhaust pump and valve system.

A spiral spring, exerting a force of 2 to 3 pounds at the rack on rod R, tends to rotate carrier C in a counter-clockwise direction. Another tension spring, exerting a force of 4 to 5 pounds, tends to move slider frame F to the right. The weight of plunger P, which is approximately 1 pound, tends to push this member downward.

The relative strength of the springs and the weight equivalent of the plunger are important, as the movements of the several elements depend on the fact that these springs must collapse in a predetermined order. In operation, horizontal rack-rod R moves to the left to rotate gear G and raise plunger P, with its plate, until the shoulder on the plunger strikes the lower boss face of carrier C, actuating rod R being in a position that will not interfere with carrier C.

Continued movement of rod R to the left rotates gear G and carrier C through a quarter-turn against the action of a spiral spring until it is stopped by limit pin L_2. Any further movement of rod R now causes the slide assembly to move to the left against the pull of the spring. A reduction in the height of the pile of plates is automatically compensated for by a longer return movement of rod R by its actuating spring, which causes the plunger to fall to the pick-up position, regardless of the height of the pile of plates. Assuming that the plates are 6 1/2 inches square, the third or horizontal motion must be 2 3/4 inches to obtain adequate swing clearance.

Reversing Transfer Mechanism.—The mechanism shown in Fig. 18, is used on a wire fabricating machine for transferring a flat wire *W* from its original working position, shown in the upper view, to the position indicated by dotted lines at W_1 in the lower view. It will be noted that this transfer movement turns the flat wire upside down.

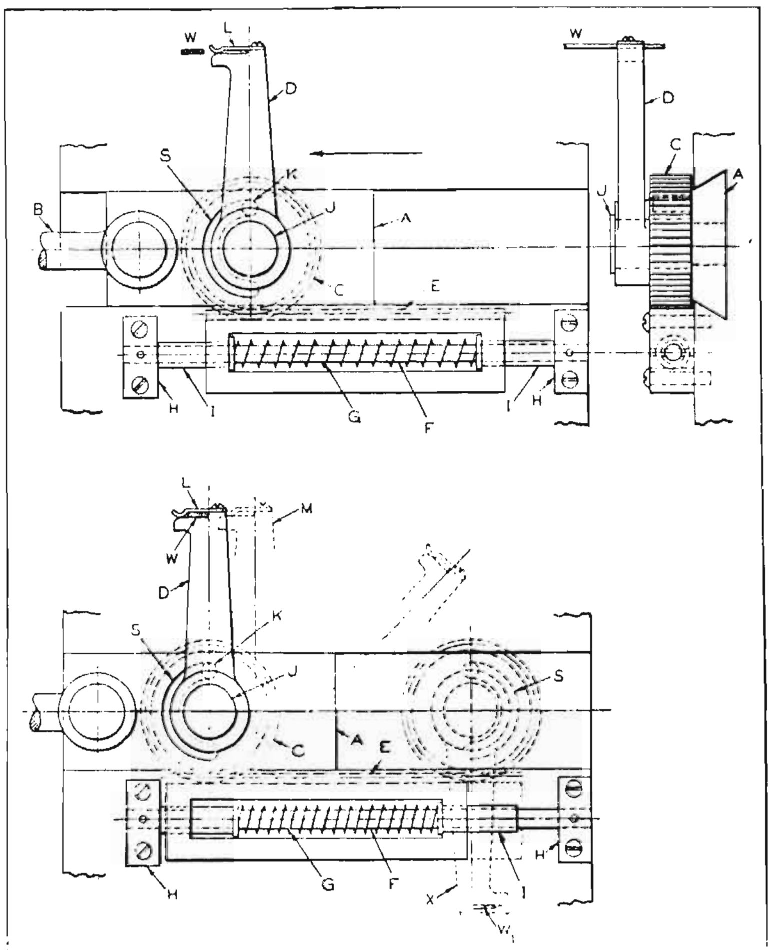


Fig. 18. (Top) Mechanism Designed to Pick Up Wire W and Transfer it to the Position Indicated by Dotted Lines at W_1 in Lower View. (Bottom) Diagram Illustrating Movement of Transfer Lever D.

Two synchronized assemblies or mechanisms like the one shown are used to support the wire at both ends.

Referring to Fig. 18, upper view, the reciprocating movement is transmitted to slide *A* by a cam-actuated connecting-rod *B*. Gear *C* and lever *D*, which are keyed together, are mounted on stud *J* which is fixed on slide *A*. Rotation of lever *D* is stopped when the ends of a semicircular slot *S* in gear *C* come in contact with a pin *K* in slide *A*. Gear *C* meshes with a rack on the upper edge of plate *E*, which carries the two flanged bushings *I* and spring *G*. The bushings *I* are free to slide in plate *E* on the shaft *F*, which is supported by the two blocks *H*, attached to a stationary part of the machine. The length of the bushings *I* is such that when in position in plate *E*, the length of the assembly is exactly the same as the distance between blocks *H*. Thus any movement of plate *E* in either direction is resisted by spring *G*, which serves to equalize the position of the plate between blocks *H*.

In Fig. 18, upper view, slide *A* and lever *D* are shown approaching the end of their travel to the left. It will be noted that the pin *K* is in contact with the end of slot *S* in gear *C*. The movement of slide *A*, up to this point, gives to gear *C* and lever *D*, to which it is attached, a rotative motion. Meanwhile, plate *E* is held stationary by spring *G*. In the position shown in Fig. 18, upper view, the upper end of lever *D* is advancing to receive the work *W*, which slides under the spring *L*. When the end of slot *S* comes in contact with pin *K*, the rotative movement of gear *C* is stopped and the continued movement of slide *A* is transmitted through gear *C* to slide *E*, which is then moved against the resistance offered by spring *G*. In this manner, a straight line movement of lever *D* is produced near the end of its travel to the left.

In Fig. 18, lower view, slide *A* is shown in the position it occupies at the end of its movement to the left, the work *W* being gripped under spring *L* on the end of lever *D*. On the return stroke of slide *A*, lever *D* returns in a straight-line movement to the position shown by the dotted line at *M*, which coincides with the position shown in Fig. 18, upper view. As plate *E* is returned to its central position, continued movement of slide *A* to the right causes gear *C* to be rotated on stud *J* until pin *K* is in contact with the opposite end of slot *S* in gear *C*, at which time lever *D* is in the position *X* indicated by dotted lines. A straight-line movement to the right is then transmitted to lever *D* until work *W* arrives at the point at which it is to be discharged. As the slot in gear *C* permits a rotative movement of 180 degrees, the work is given a full reversing or turning movement while being transferred from one position to the other.

Dial Transfer Mechanism for Chain Making Machine.—The dial transfer mechanism shown in Figs. 19 and 20 was developed for use in a chain making machine. The function of the mechanism consists of picking up a piece of work at *M*, Fig. 19, and transferring it to the position indicated at *N*. Since the limited space available made it impossible to employ an ordinary cam arrangement for transferring the work, it was necessary to develop the special mechanism here shown.

The problem of transferring the work from *M* to *N*, which was too long a distance to permit using a single cam movement, was solved by employing an indexing dial transfer movement, effected by a Geneva motion in combination with a comparatively short reciprocating motion obtained by a barrel cam.

Referring to Fig. 19, the transfer of a piece of work from *M* to *N* is accomplished by a combination of two indexing movements of the Geneva actuated dial *C* which carries the piece from *M* to *O* and a traversing movement of the dial *C* through a distance *T* by the action of the barrel cam *J*, Fig. 20. The latter movement carries the piece from *O* to *N*.

Assuming that dial *C* is indexed in a clockwise direction, the piece picked up at *M* will be indexed to position *P*, where it remains idle while the preceding piece at *O* is carried to

N. Then, on the second indexing movement of the dial, it is carried to position *O*, from which it is transferred by the movement of dial *C* through distance *T* to position *N* during the idle period of the dial.

The jaws *H* are spring-operated. The mechanism is driven by shaft *S* on which the barrel cam *J* is mounted. The entire

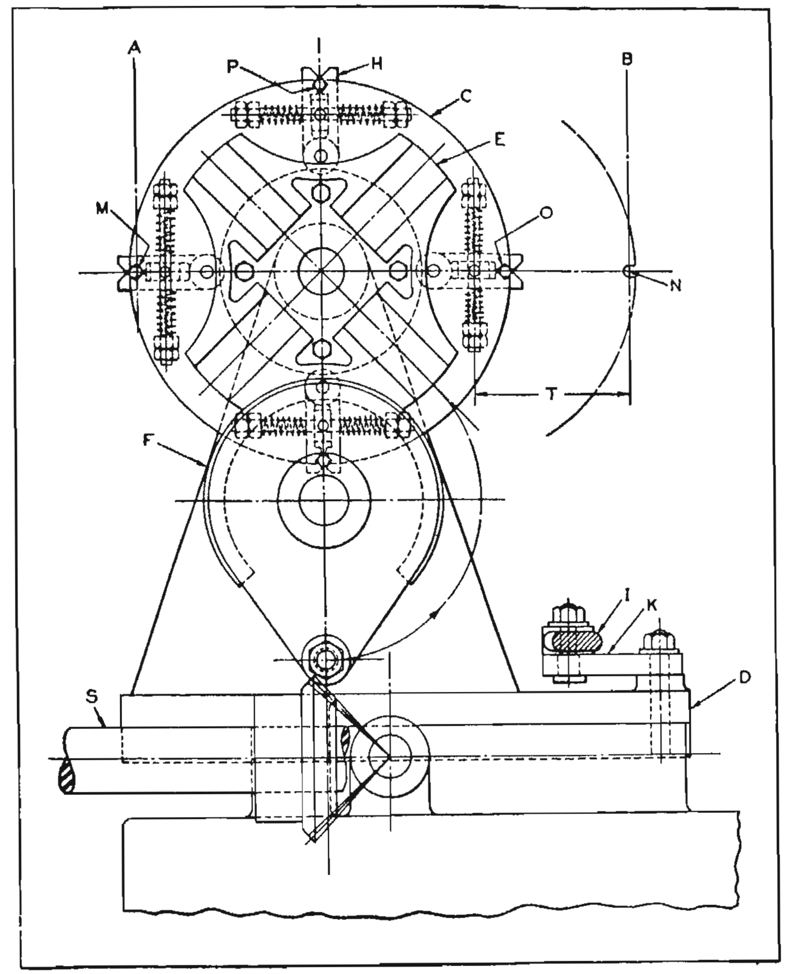


Fig. 19. End View of Dial Transfer Mechanism Used in Chain Making Machine.

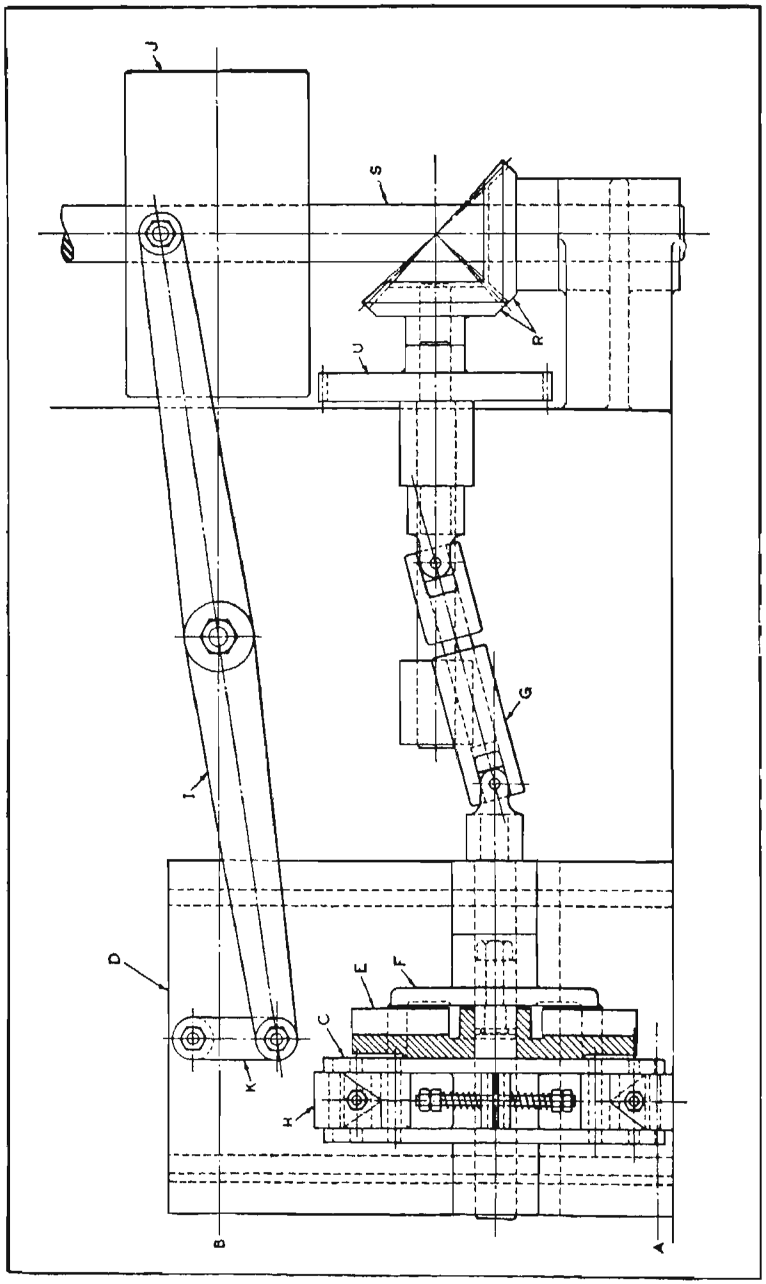


Fig 20. Plan View of Mechanism Shown in Fig. 19.

Geneva mechanism is mounted on a slide *D*, which is given the required intermittent reciprocating motion by barrel cam *J* through lever *I* and connecting link *K*.

As shown in Fig. 20, the driver *F* of the Geneva mechanism is driven from shaft *S* through miter gears *R*, spur gears *U*, and universal-joint shaft *G*, which permits the slide *D* to move on the base without interfering with the transmission of rotary motion to driver *F*.

Quarter-Turn Mechanism for Transferring Sheets from Press to Oven.—Tin sheets for making container parts are received from the conveyor belt of a printing or decorating press, swung through an angle of 90 degrees in a horizontal plane, and fed onto the conveyor of a drying oven by means of the mechanism illustrated. The work performed on the press necessitates that the tin plate be delivered in the position shown at A_1 in Fig. 21. However, the long drying oven is narrow and will only accommodate the sheets when they have been turned to the lengthwise position shown at A_2. As it would have been costly to rebuild the oven to handle the greater widths, the quarter-turn mechanism was designed and located between the conveyors of the press and the drying oven.

Sheet *A* is delivered from the printing press and received on conveyor belts *E* in the position indicated at A_1. The surface speed of conveyor belts *E*, which rotate on pulleys mounted on shafts *C* and *D*, is greater than that of the press conveyor belts. The front left corner of the tin sheet comes in contact with the conical-shaped, friction disks *F* and F_1, and a portion of the sheet slips between the faces of the disks. These disks are free to rotate on stud *R*, and are pressed together by spring *G*. The tin sheet is thus gripped between the disks, and its forward motion is temporarily halted.

The sheet is swung about the disks, which act as a pivot, by means of the rubber-covered pulley *H*, which is in contact with the lower surface of the sheet. This pulley, which

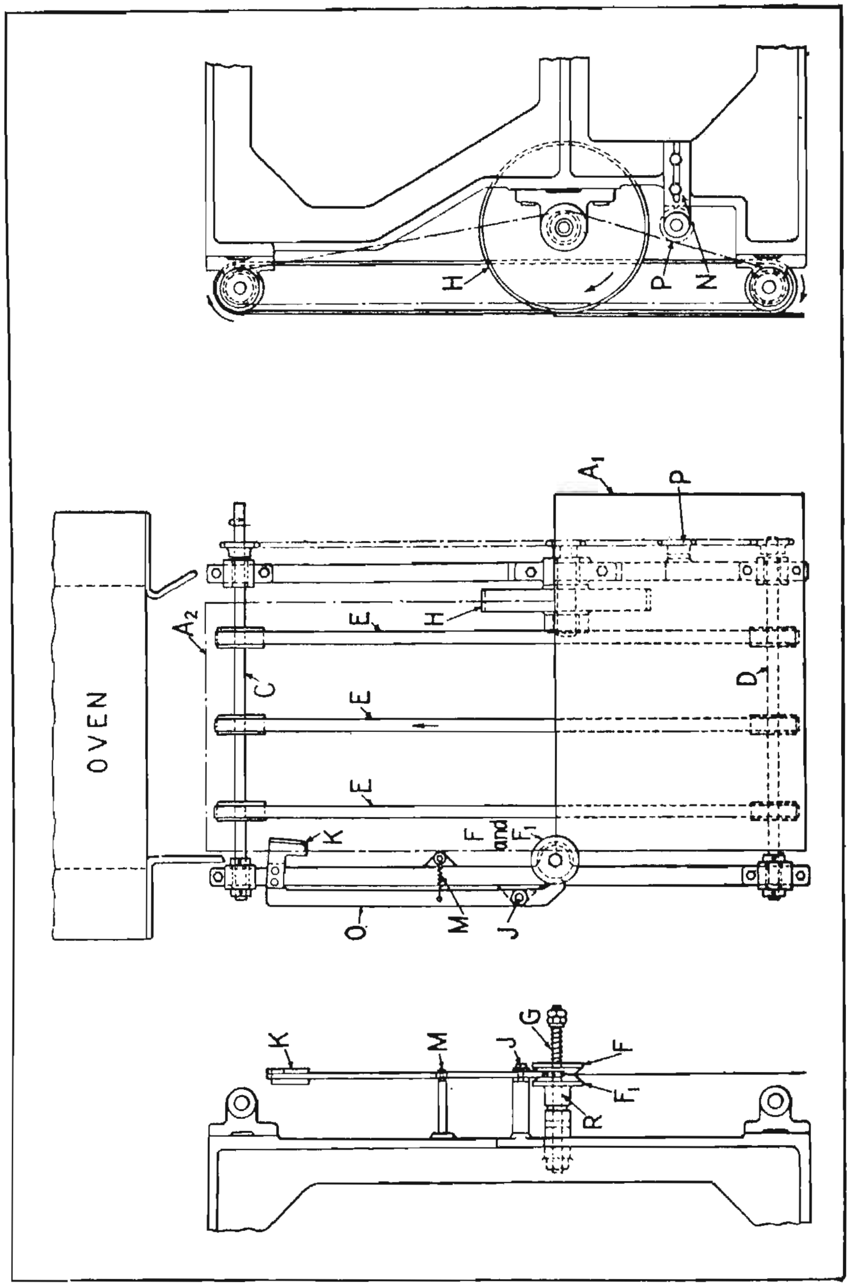


Fig. 21. Metal Sheets are Turned from Position A_1, as they Leave the Press, to Position A_2 for Feeding into Oven.

has a surface speed greater than that of conveyor belts *E*, turns the sheet to position A_2. As the sheet nears the desired position, its edge comes in contact with the face of the leather-covered bumper *K*. The force with which the sheet hits the bumper is sufficient to pivot lever *O* about stud *J* and cause the bevel-shaped end of the lever to separate friction disks *F* and F_1, and compress spring *G*. The sheet is thereby released and allowed to continue its forward motion on conveyor belts *E* and onto the conveyor of the oven.

Lever *O* is returned to its original position as soon as the sheet passes bumper *K* by means of spring *M*. The tension of this spring must be carefully controlled, so that it will not cause the bumper to apply too much braking action on the edge of the moving sheet. Shaft *C* is turned by a motor-driven chain. Shaft *D* and pulley *H* are rotated by means of sprockets and a single chain driven by shaft *C*. Idler sprocket *P* mounted on bracket *N* is used to adjust the tension of the chain.

Mechanism for Reversing Work and Transferring it to Another Spindle.—The purpose of the mechanism shown in Fig. 22 is to transfer the work *W* from spindle No. 1 of an automatic machine to spindle No. 2, and at the same time, reverse the position of the work, so that the chamfered end *S* is inserted in the chuck of the second spindle. The particular set-up shown was used in machining drawn shells such as shown at *A* to the form shown at *B*.

After the work has been chamfered, as shown at *S*, the housing *T* of the transfer arm is swung or rotated on its supporting bearing *X* to bring the jaws *D* and *E* in line with spindle No. 1. Rod *C* then pushes the work into place between jaws *D* and *E*. The housing *T* is next rotated or indexed on its bearing *X* to bring the work into line with the chuck on spindle No. 2. During this transfer movement, the shaft *K* is rotated 180 degrees, so that the chamfered end *S* is in the required position to enter the chuck on spindle No. 2 when the work is pushed from between the jaws *D* and *E*.

Referring to Fig. 22, spring F exerts sufficient pressure on jaws D and E to cause them to hold the work securely while it is being transferred from one position to the other. The housing T is indexed about bearing X by means of link G which moves forward or backward at the proper time.

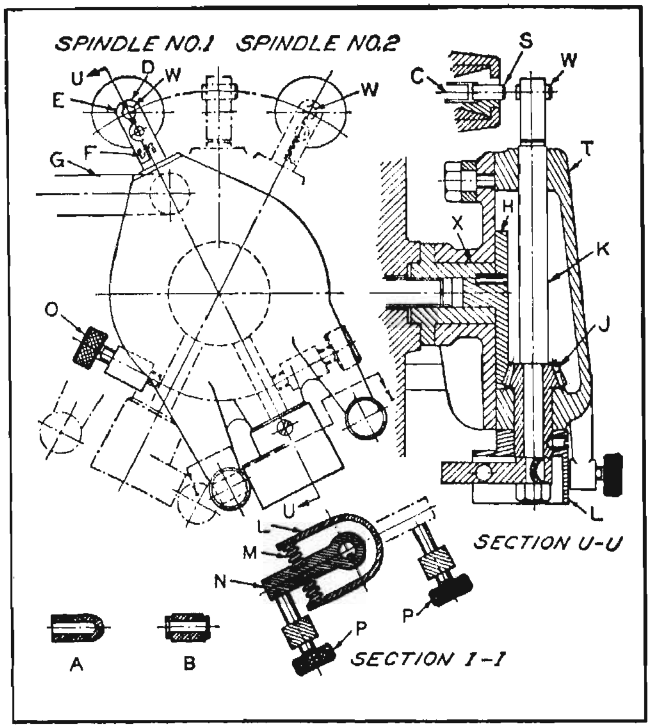


Fig. 22. Mechanism for Automatically Reversing Position of Work W and Transferring It from One Spindle to Another.

The rotation of shaft K for reversing the position of the work is accomplished by means of the stationary segment bevel gear H, which is in mesh with bevel pinion J. To pinion J is secured a member L which transmits the required turning movement to shaft K through springs M and the arm N secured to the end of shaft K. The rotating movement of shaft K is stopped at each spindle position by stops

P, which can be so adjusted that the work is revolved exactly 180 degrees while being transferred from one spindle position to the other. In the stopped position of arm N, the spring M which transmitted the motion to the arm is under sufficient pressure to keep the arm in contact with the stop. Stops O which limit the indexing movement of housing T are adjusted to bring the work into accurate alignment with the spindle.

Mechanism for Operating Two Slides Alternately.—In one automatic wrapping machine, two parallel slides, operating alternately, control the movement of each package through the machine. Both slides, indicated at A and B in Fig. 23, are given the required reciprocating movement by the cross-head C through a combination ratchet and cam mechanism.

The function of this mechanism is merely to lock the slides A and B alternately to the cross-head. The mechanism consists of locking slide D (see also Fig. 24) ; star cam G on shaft H engaging hardened pins L in the locking slide; ratchet wheel K keyed to the shaft and operated by the pawl J which is pivoted to the machine base; and locking plungers E and F, backed up by coil springs in the locking slide.

As shown in Fig. 23, slide B is stationary and slide A is locked to the cross-head C by plunger E. Locked thus, slide A has completed a working stroke and has nearly reached the end of its return stroke. At this time, pawl J engages ratchet wheel K, and, as the slide A completes its return stroke, rotates the ratchet wheel and cam G one-tenth of a revolution. This results in the cam forcing slide D toward the right, unlocking slide A and cross-head C, and locking the latter to slide B.

With slide A now stationary, slide B and cross-head C are locked and travel together through a working and return stroke. Near the end of the return stroke, pawl J once more engages the ratchet wheel and causes the locking slide D to move this time toward the left. This movement of slide D

unlocks slide B and cross-head C and locks the cross-head to slide A, which is then carried through a working and return stroke while slide B remains stationary.

To prevent over-run of the ratchet wheel, the ends of the cam lobes were slightly grooved to fit the hardened pins L. Both slides A and B operate at a relatively slow speed. It is

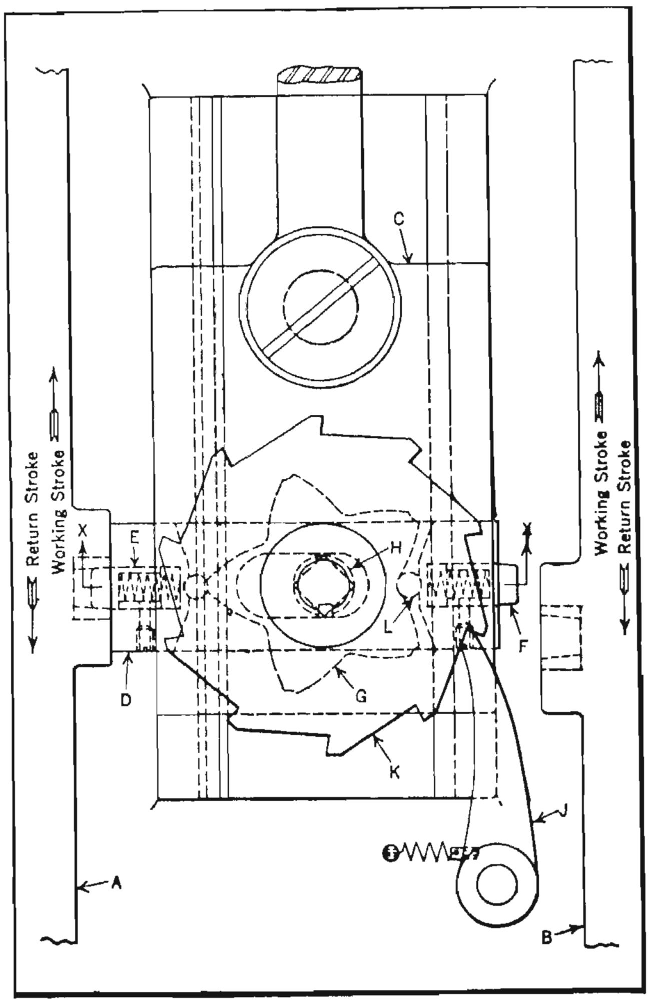


Fig. 23. Combination Ratchet and Cam Mechanism for Alternately Locking Two Parallel Slides to a Continuously Reciprocating Cross-head.

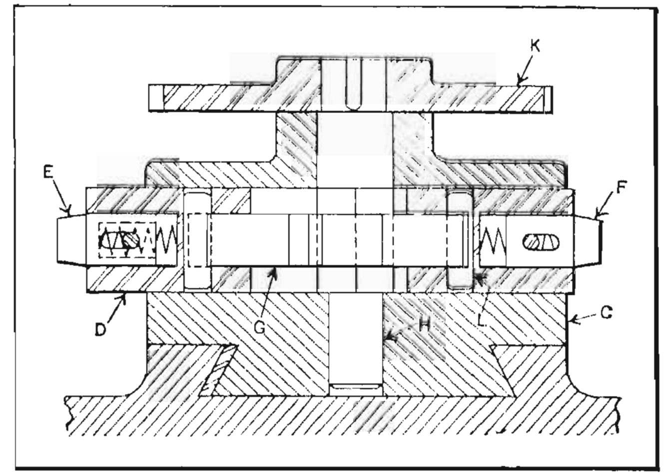


Fig. 24. Section X-X of Fig. 23 Showing the Action of the Locking Slide.

advisable, however, when this mechanism is applied to rapidly moving slides, to provide some sort of a friction stop or brake, so that the slides will stop in the same position at the end of each return stroke.

CHAPTER 16

Feeding and Ejecting Mechanisms for Power Presses

The use of a properly designed feeding and ejecting mechanism is an important factor in power press operation for maintaining a low percentage of spoiled work and a relatively high production rate. The mechanisms described in this chapter were designed for a wide variety of press functions, such as placing a drawn shell in a punching die; feeding strip material at various rates; carrying a printing type frame under the ram of a hydraulic press; operating an indexing type of dial feed; transferring work from one punch press to the proper feeding position in another; ejecting a formed part from a dovetail-shaped punch; retarding an automatic feed device; and stopping a press when stock fails to feed.

Other similar feeding and ejecting mechanisms will be found in Chapter 16, Vol. I and Chapter 15, Vol. II of Ingenious Mechanisms.

Automatic Feed for Placing Drawn Shell in Punching Die.—The die shown in Fig. 1 was designed for punching the bottom out of a drawn shell of the shape shown by the cross-section view at *S*. The arrangement provided for automatically feeding the shells *S* to the die resulted in a reduction of spoiled work and trouble from shearing of the punch due to improper location of the work in the die. At the same time, the use of the automatic feed served to increase the production of the die from 1300 to 2400 pieces per hour.

Formerly, the work was pushed down a chute into a locating nest with a stick. Often the work was not in the proper position when the punch came down, with the result that the punch was sheared or dulled. Part of the difficulty was

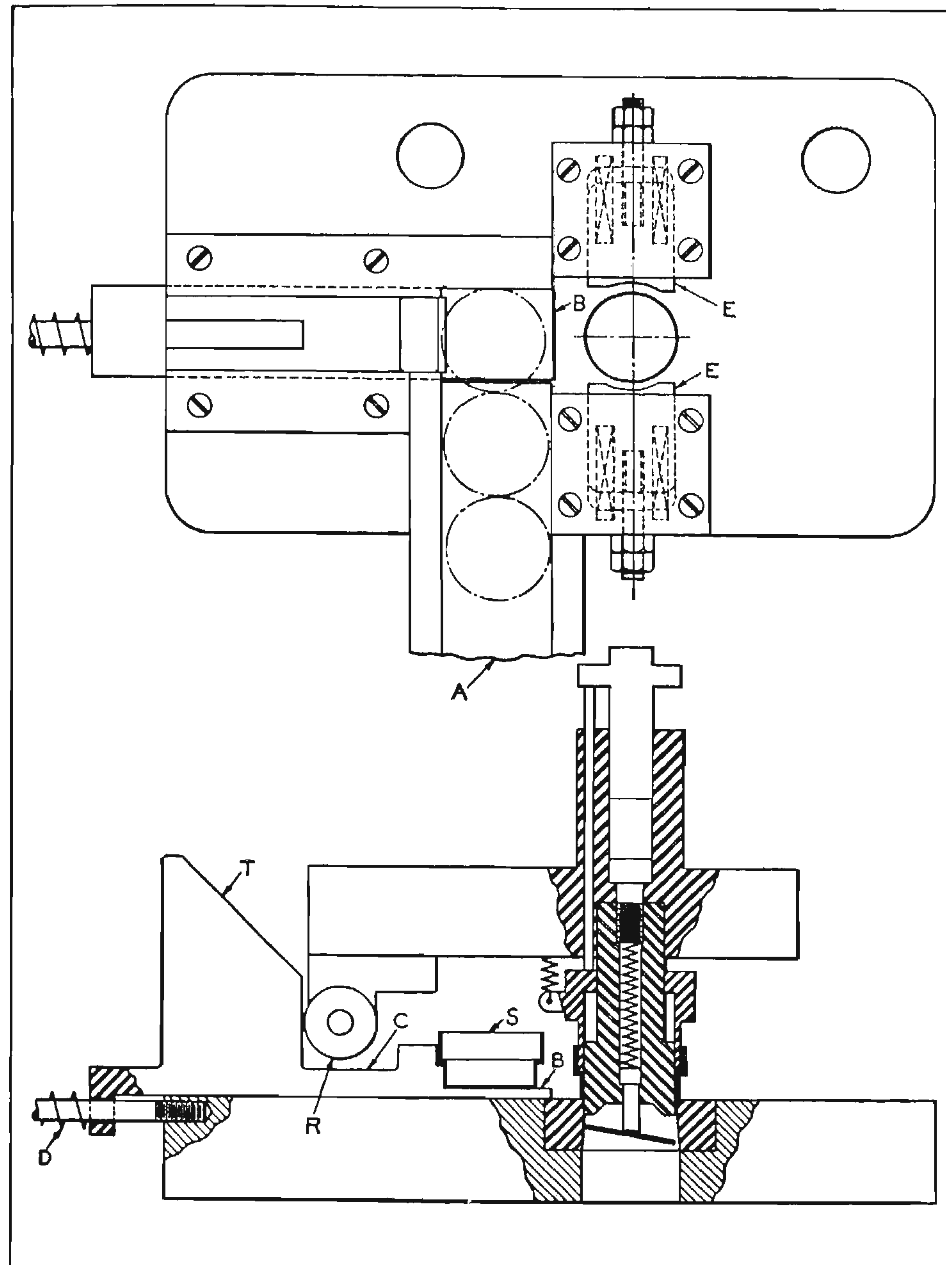


Fig. 1. Die Equipped with Automatic Feed which Locates Drawn Shells under Punch.

due to the fact that the work would not slide over the hole in the die, which was almost as large as the work, but would come to rest against the side of the hole in a tilted position.

With the automatic feed shown diagrammatically in the illustration, the work is fed into the inclined chute *A*, from which it slides down on a shelf on slide *C*, which pushes it between two spring jaws *E* located over the blanking hole in the die. The slide *C*, actuated by spring *D*, carries the shell *S* under the punch on the upward stroke of the press ram. On the downward stroke, the roller *R* acting on the cam *T* moves the slide *C* outward to the position shown in the illustration, and the punch centers the work, pushes it down onto the die, and punches the hole as shown in the lower view in Fig. 1. The round blank punched out of the bottom of the shell drops through the opening in the die. The work is stripped off the punch on the upward stroke of the ram, and ejected from the die by a blast of air.

The spring action used to push the slide forward prevents jamming in case the work is only half way in the slide when this member starts its forward movement. This happens occasionally when the operator fails to keep the chute filled with shells. The principle on which this lateral feeding mechanism operates can be applied, with certain modifications, to other work. For example, the slide shelf may be omitted when it is unnecessary to carry the work over the die opening.

Obviously, the automatic feed applied to this die makes it much safer to operate. Also, the die can be operated continuously, with less danger of damage to the die or work. The shells can be fed by hand into the chute or a hopper feed can be used. They can also be fed by chute directly from the drawing press.

Mechanism for Feeding Washer-Shaped Blanks to Die with Concave Sides Up.—Washer-shaped blanks of slightly concave form, as indicated at *a* in Fig. 2, are fed by gravity from a hopper through a chute *L* to the mechanism to be

described. The mechanism, in turn, automatically feeds the blanks, one at a time and concave side up, through chute *T* to the die of a punch press.

The part *a* is produced by piercing and blanking from mild strip steel. This operation leaves slight shearing burrs around the edge of the blank, as indicated, and gives it a concave or dished effect that varies from 0.010 to 0.040 inch. It is the function of the mechanism shown at *b* and *c* to insure feeding the blanks to the die through chute *T* with the concave side up toward the forming punch.

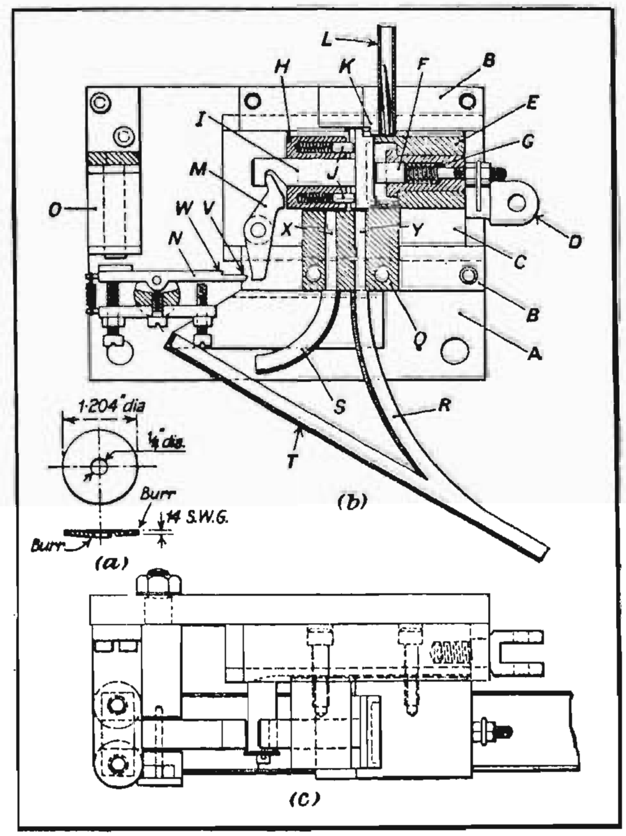


Fig. 2. (a) Views of Blank, Showing Distortion and Location of Burrs. (b) Sectional Side View of Selecting Device. (c) Plan View, with Chute L and Stop K Removed.

Referring to sectional side view *b*, baseplate *A* carries guides *B*, in which the sliding piece *C* is moved horizontally by the pin joint *D*. The pin joint is operated through a linkage connected to the press connecting-rod. A friction slipping device is incorporated to enable the sliding piece to be stopped at certain positions during the cycle. The contact body *E* is fixed to the sliding piece and is bored out, as shown, to receive an insulating bushing which carries the spring contact plunger *F*, this being adjusted to bring the end face level with the face of the contact body, where it is held by spring *G*.

Block *H* is also fixed to the sliding piece and carries the movable clamp *I*, which is forced toward the contact body by two spring pegs *J*. The upper side of the clamping head is milled to form a stop, so that when the slide is moved to the right, the clamp is arrested by a fixed stop *K* and the slide is brought to rest when block *H* comes up against the back of the clamping head.

In this position, a blank in chute *L* may fall into the gap between the contact body and the clamping head. A notch is milled in the clamp stem to engage the upper end of the lever *M* as shown, the lever being freely pivoted on a stud screwed into the sliding piece. The free end of the lever may engage either of the stops *V* or *W* when the sliding piece is moved to the left. The stop engaged depends on whether pivoted stop *N* is up or down, the position of *N* being controlled by an electromagnet *O* connected in series with the contact body *E* and a suitable source of direct current.

The two positions in which the clamp may be withdrawn and the blank released are indicated by the delivery slots *X* and *Y* in block *Q*. The arrangement of chutes *R* and *S* is such that a blank sliding down chute *R* will be turned 60 degrees in a counter-clockwise direction, and a blank sliding down chute *S* will be turned 120 degrees in a clockwise direction before passing down chute *T* to the press tool and its feeding mechanism.

Assuming that a blank is in chute *L* in the position shown in Fig. 2, and that slide *C* is moving to the right, the action of the device is as follows: Clamp *I* is arrested by stop *K*, and the contact body *E* moves on, allowing the blank to fall into the opening produced. The slide is then moved to the left, permitting pegs *J* to clamp the blank against the contact body, which results in the electromagnet *O* being energized. Stop *V* is moved out of the path of lever *M*, and the blank is carried along until lever *M* comes in contact with stop *W*, when the clamp is withdrawn and the blank is discharged down slot *X*. As soon as the blank leaves the contact face, the circuit is broken and the magnet de-energized, so that, on the return of the slide to the starting point, the stop *N* returns to its original position when cleared by lever *M*.

If the blank is located in chute *L* the opposite way round to that shown, then the central part of the disk will not touch the contact plunger and the magnet will not be energized; consequently, stop *V* will cause the blank to be discharged down slot *Y*. It will be apparent, therefore, that all blanks sliding down chute *T* will be hollow side up as required.

Adjustable Strip Feeding Device.—In feeding strip material, it is frequently desirable to vary the feeding stroke or adjust it accurately to meet changing conditions. The device to be described can be adapted for feeding strip material of any form or type and permits adjustment of the feeding interval or stroke to a minute degree by means of an adjusting screw. It is very efficient when moderate feeding rates are used.

The operation of the device is as follows: The plate cam *N*, shown in Fig. 3, is provided with a ridge around 180 degrees of the circumference which serves to raise the end of cam-lever *K* on which roll *M* is mounted. The oscillations thereby created by the cam in lever *K* are transmitted to the pressure lever *F* through the link *J*. Thus lever *F* serves to

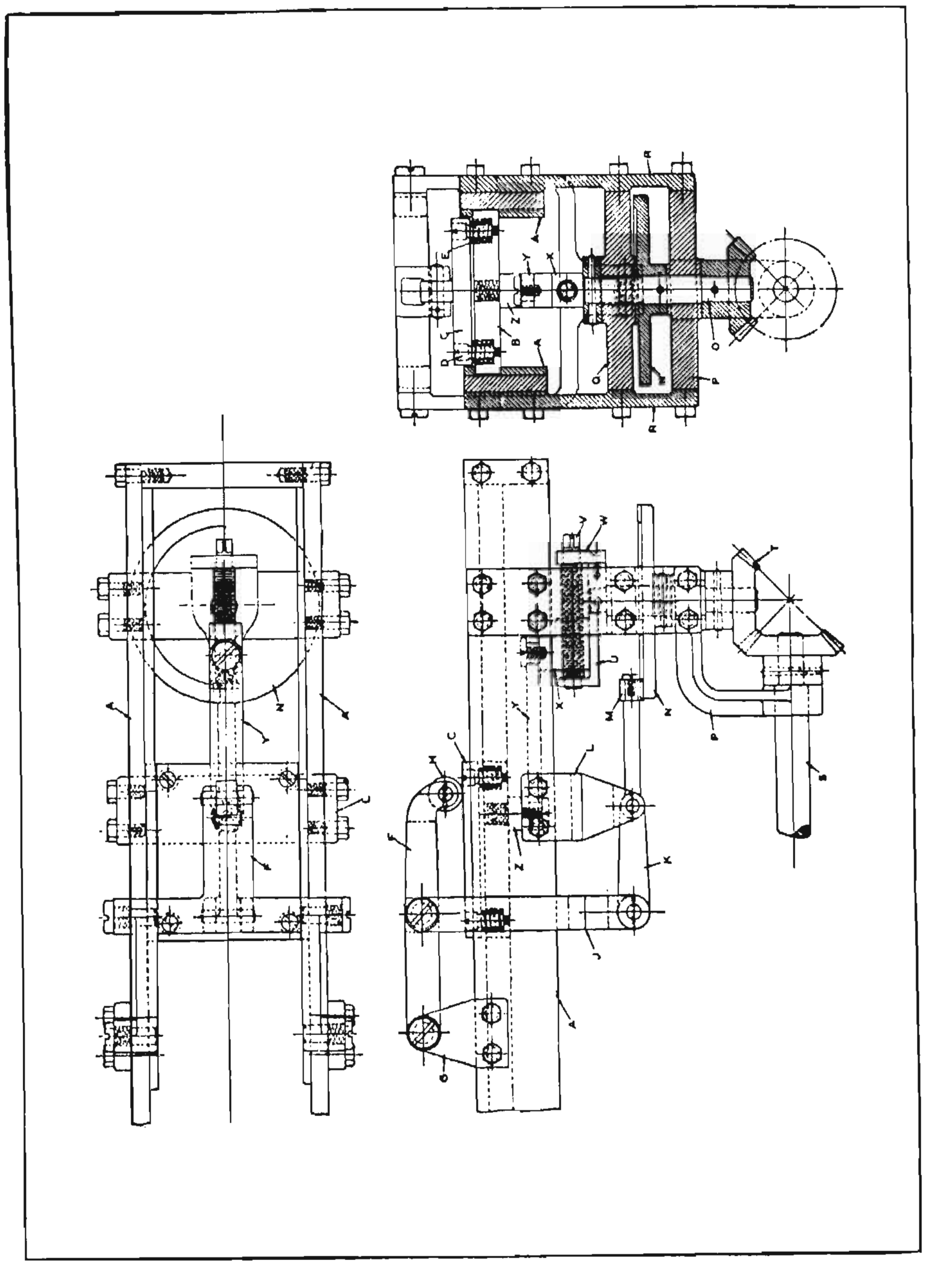


Fig. 3. Strip Stock Feeding Mechanism with Means for Adjusting Stroke.

depress and release the upper plate C which grips the stock at definite intervals, as determined by the position of cam N under the cam roller M.

The lower sliding plate B and the upper plate C, which form a unit through the connection of the screws D, are actuated by the eccentric block X through the connecting link Y. The continual rotation of block X causes the entire strip-gripping arrangement under roller H to reciprocate.

By combining the clamping motion created by cam N and the feeding movement imparted by the block X, the strip material is gripped in the jaws of the gripping assembly and given the required feeding movement. The smooth and regular operation of the device is assured by synchronizing the rotation of eccentric X with the rotation of cam N in such a manner that the gripping device will be released slightly before it has reached the end of its stroke. This will avoid the possibility of dragging the strip back slightly on the return stroke of the gripping assembly.

Strip material of any kind can be handled by the feeding device. Flat surfaces in the gripping assembly would serve well for materials that are slightly compressible. For hard materials, it may be necessary to roughen the surfaces in order to obtain a firmer hold. By altering the shape of the surfaces, it would be possible to handle materials of a form other than flat. For instance, wire can be easily controlled in a device of this type.

Referring to the construction of the device, the rails A are a part of the machine structure, and, in addition to serving as a support for the device, they have channels which guide the lower sliding plate B. The upper plate C is held in place by four fillister-head screws D, the heads serving as pilots for the upper plate. Four springs E around screws D tend to keep the upper and lower sliding plates apart. The whole assembly constitutes the strip-gripping arrangement.

The pressure lever F is supported in brackets G fastened to the rails A so that roller H rests on the surface of plate C. Link J serves to connect lever F with the cam-lever K. The cam-lever is supported in the bracket L, also fastened to the rails A, and is provided with the cam roller M which makes contact with cam N.

Cam N is fastened to the shaft O which is supported between the bearings P and Q. The bearings are fastened to the support bars R which, in turn, are attached to the rails A. The necessary power for actuating the device is obtained from the shaft S through the bevel gears T.

The stroke-adjusting arrangement consists of a screw-carrier U fastened to shaft O, the adjusting screw V, the end plate W fastened to the screw-carrier and serving to retain the adjusting screw in place, and the eccentric block X. Turning screw V causes eccentric block X to travel closer or farther away from the center of shaft O, thereby decreasing or increasing the eccentricity. The link Y serves as a connection between block X and the strip-gripping assembly through the extension block Z.

Compact Table-Feeding Mechanism.– The mechanism shown in Fig. 4 was designed to carry a chase or printing type frame P under the ram of a hydraulic press for making wax impressions from the type in the chase. The limited space available on the press prevented the use of the conventional crank movement, and made a compactly designed feeding mechanism necessary.

The table A on which the chase is loaded reciprocates between the loading position P of the chase at the left and the working position P_1 at the right. The travel of the table is guided by ways B. Table members C form a slot M which is at right angles to the center line X–X, representing the direction of rectilinear movement of the table. Block D, which is a sliding fit in this slot, is free to revolve about the stud S at one end of crank-lever F. Crankpin H is turned by crank G, which is revolved by means of the driving shaft

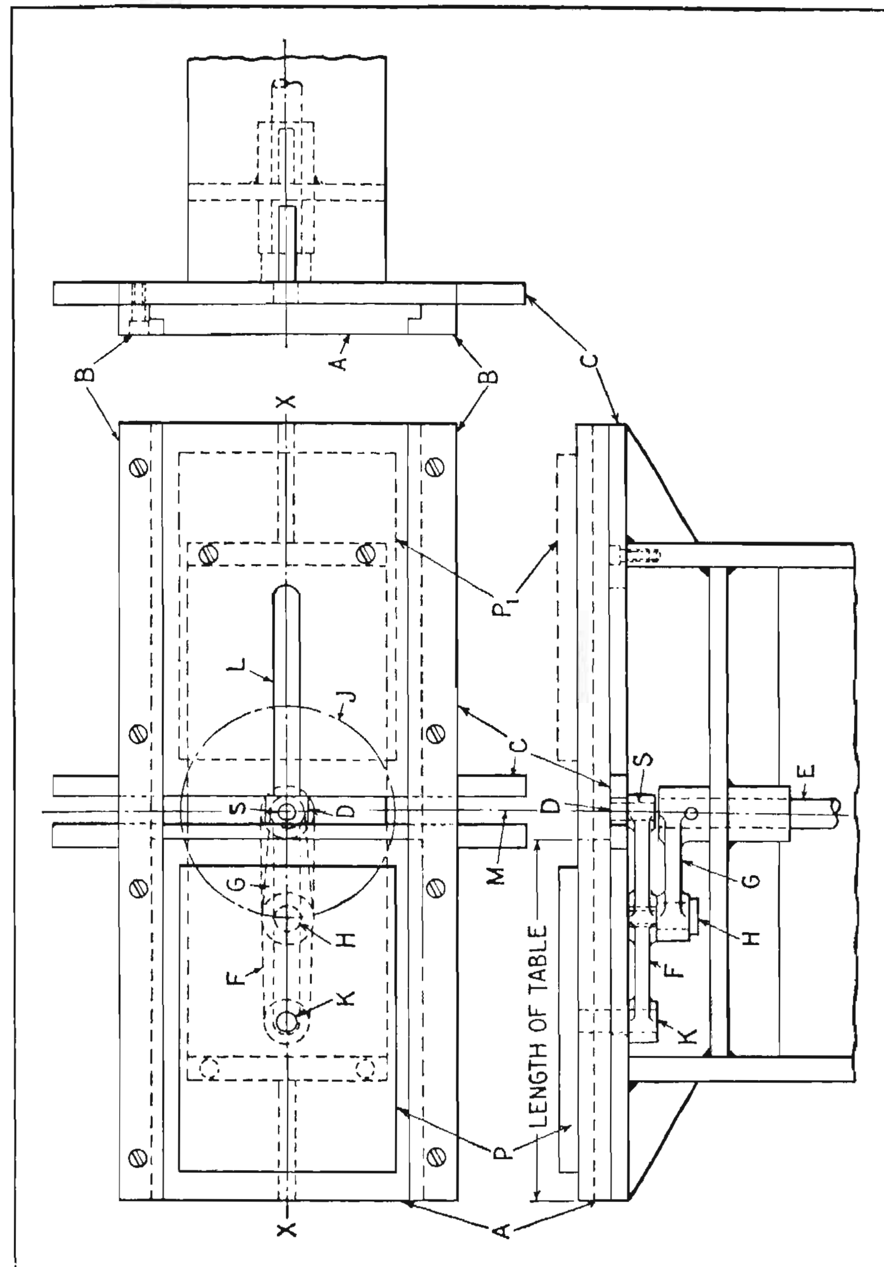


Fig. 4. Table A is Reciprocated along Center Line X-X by means of this Mechanism.

E to which it is pinned. The center line of the crankpin will follow a path shown by circle *J*.

The travel of one end of the crank-lever *F*, being fastened to block *D*, is confined to a reciprocating movement by the slot *M*. The other end of the crank-lever is fastened to table *A* by connecting-rod pin *K*. The travel of pin *K*, which carries the table with it, is confined by slot *L* formed by the table members *C* to a reciprocating movement. The common center line of the table and pin *K* follows the path indicated by center line *X–X*.

Mechanism for Operating Dial Feed and Radially Positioned Multiple Punches.—The mechanism shown in Figs. 5 and 7 was developed for operating an indexing type dial feed and radially positioned multiple punches used for the production indenting of thin-walled tubes, such as indicated at *B* in the enlarged view *A*, Fig. 6. The tube *B* serves as a means of assembling or joining the wooden rod *C* to the cylindrical rubber piece *D*.

The function of the dial feed mechanism is to pick up the assembled rod *C*, tube *B*, and rubber *D* at *E*, Fig. 7 at the left, and by successive intermittent indexing movements in the direction indicated by the arrow, bring these assembled members into the position indicated by the dot-dash lines at *F*. While the work dwells in this position, the eight radially located cam-operated indenting punches *G* are advanced to produce sixteen indentations, which serve to securely fasten tube *B* to rod *C* and rubber *D*.

After the indenting operation, the work is indexed around toward the rear of the dial feed, where it is unloaded on a conveyor or picked up by another feeding dial. Thus one assembly is indented at each dwell period between successive indexing movements of the dial.

Referring to Figs. 5 and 7, at the left, it will be seen that the feeding dial consists primarily of two disks *H*, each fitted with twelve radially positioned, equally spaced slides *J* having U-shaped slots at their ends which pick up and carry

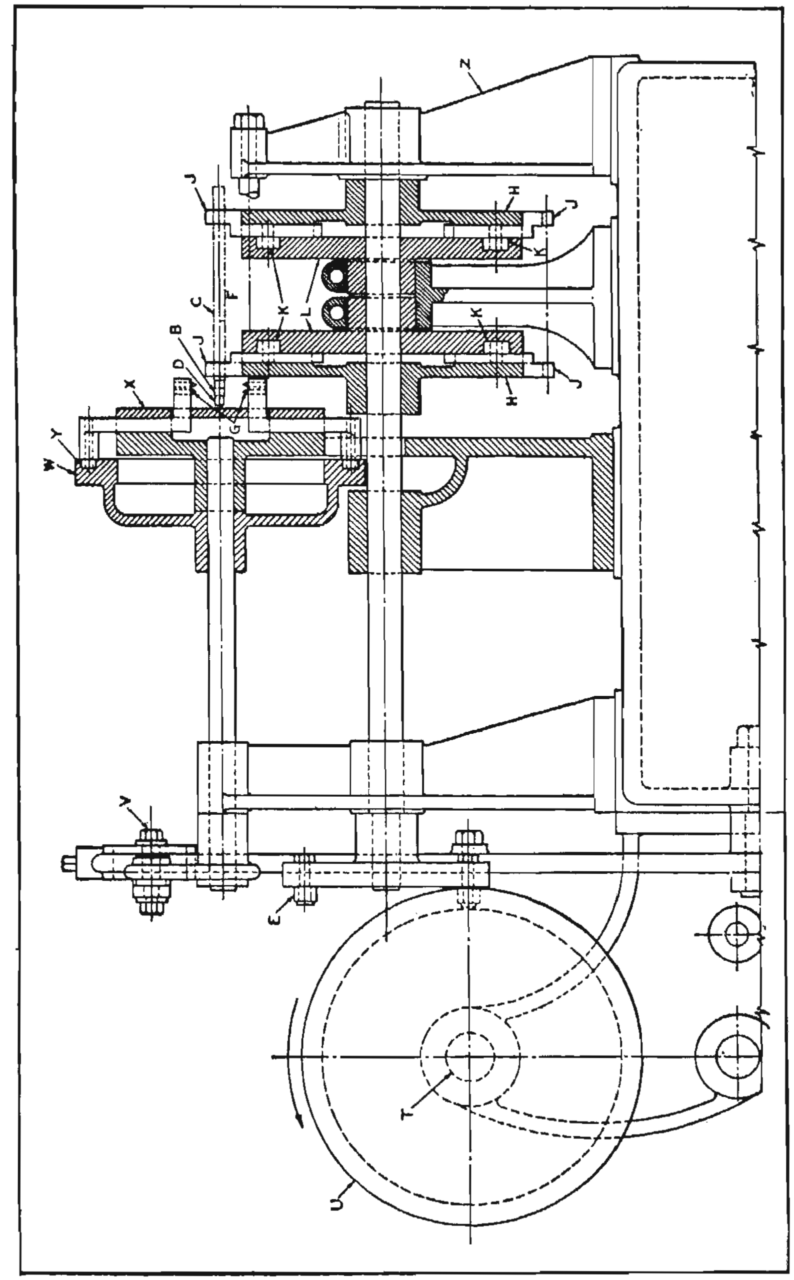


Fig. 5. Mechanism Developed for Synchronized Operation of Dial Feed and Punches for Indenting Tubes B on Assemblies such as Shown in Fig 6.

the work. Each slide J has a cam roller K which runs in a cam groove in the face of one of the two stationary cam-plates L.

The cam grooves in plates L are so laid out, as shown diagrammatically in Fig. 7, that roller K, instead of following the concentric path indicated by circle M, follows the path indicated by line N as the disk H is indexed from one position to another. This causes the slides J to carry the work along the path indicated by line P and the circles O.

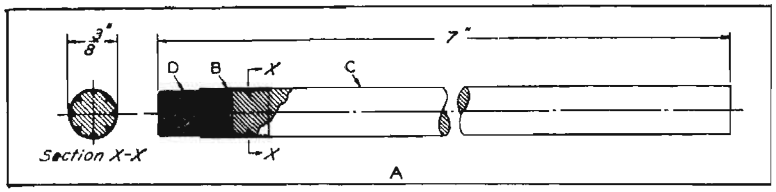


Fig. 6. Typical Assembly with Tube B Indented by Mechanism Shown in Fig. 5.

It will be noted that the work follows a path that leads away from the center of the dial or disk H as it leaves the loading position E, Fig. 7, at the left, until it reaches the position Q, after which it follows a straight horizontal path from Q to R, from which position it continues on a path that carries it back toward the center of the dial until it reaches the position indicated at S. It is necessary to have the work follow this irregular path in order to permit the tube B and rubber D, Fig. 5, to clear the indenting punches and holders as they are being indexed from Q into the indenting position at F and out again to the position R after being indented.

The intermittent indexing movements are transmitted to the feeding dial disks H from the driving shaft T, Fig. 5, through the cam U and dial driver E. The eight indenting punches are simultaneously moved inward radially to perform the indenting operation by means of the oscillating cam W operated from shaft T. The depth of the indentations can be controlled by adjusting the length of throw of

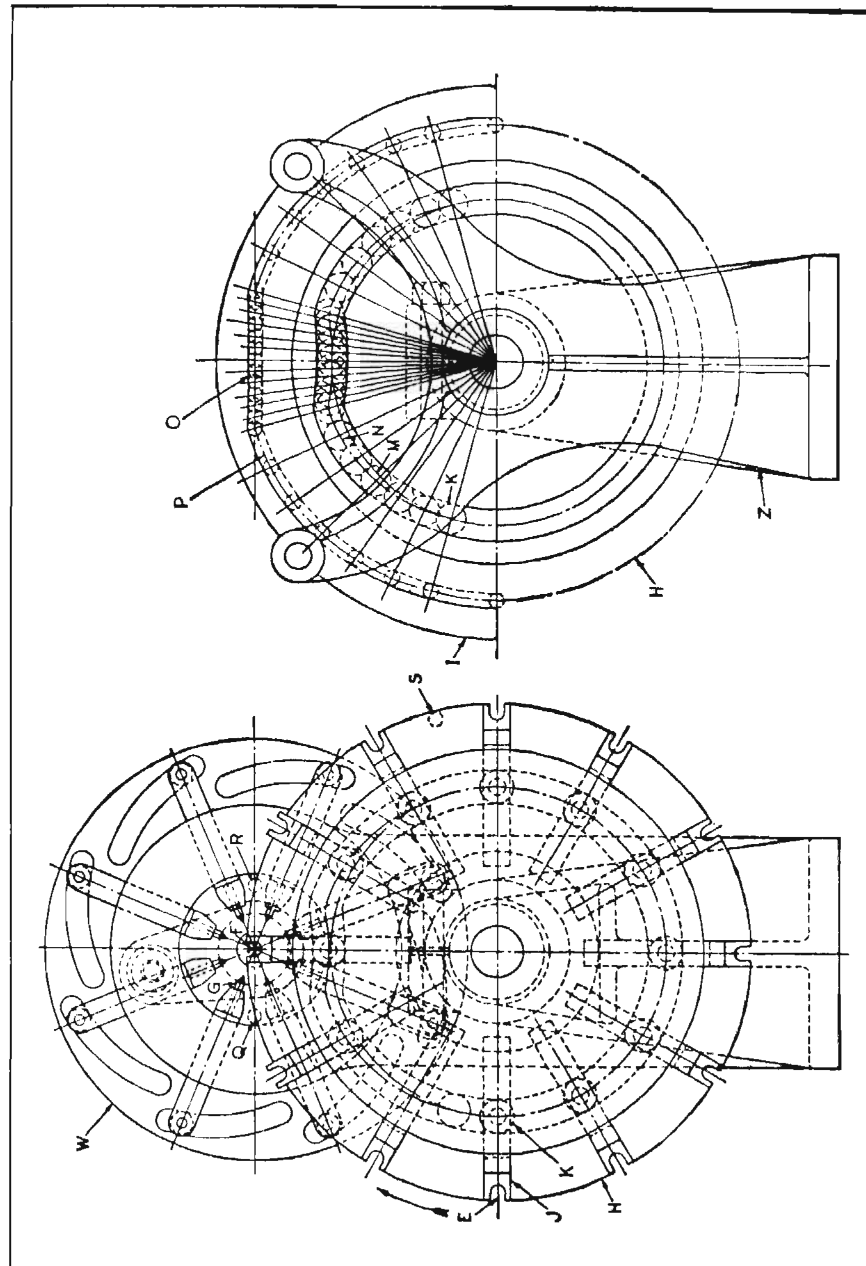


Fig. 7. (Left) End View of Indenting Punch and Dial Feed Shown in Fig. 5. (Right) Diagrammatic Layout of Cam L and Bracket Z of Fig. 5.

the cam oscillating mechanism. The oscillating movements of cam *W* are, of course, synchronized with the indexing movements of the work-carrying dial so that the indenting punches advance and withdraw while the work-feeding dial is stationary in one of its twelve dwell positions.

The holders of the indenting punches *G* are close sliding fits in the slots in the stationary head *X*, and have rollers *Y* which are running fits in their respective operating cam slots. The bracket *Z*, Fig. 7, at the right, supports a cover *I*, which keeps the work in place in the slots in slides *J* while it is being indexed from the loading position at *E*, Fig. 7, to the unloading position. The cams *L*, Fig. 5, are made with hubs mounted in a center pedestal equipped with a split bearing having clamping screws which provide means for individually adjusting the positions of the cams to obtain accurate alignment of dials *H*.

Automatic Transfer and Feeding Mechanism.—The automatic work transferring and feeding mechanism shown in Fig. 8 enables one person to operate a series of punch presses equipped to perform successive stamping operations. The stampings *A* are blanked and drawn in one press and are then blown by air pressure through a chute to the press shown in Fig. 8. The stampings are located in the die of the latter press by an automatic feeding mechanism. The feeding mechanism is shown on one press only, but other presses equipped with similar feeding devices can be added to the production line. This type of die with its lateral feeding mechanism can be adapted for handling different kinds and sizes of work. It operates at high speed and eliminates the need for a second operator in many instances.

The blanked and drawn stamping blown through the chute comes to rest against stop *E*, which is also shown in Fig. 9. The lateral feed-slide *F* now engages the stamping and pushes it forward into the spring-actuated receding jaws *G* which center it on the die *H*, Fig. 8. The slide is then withdrawn from the path of the descending punch.

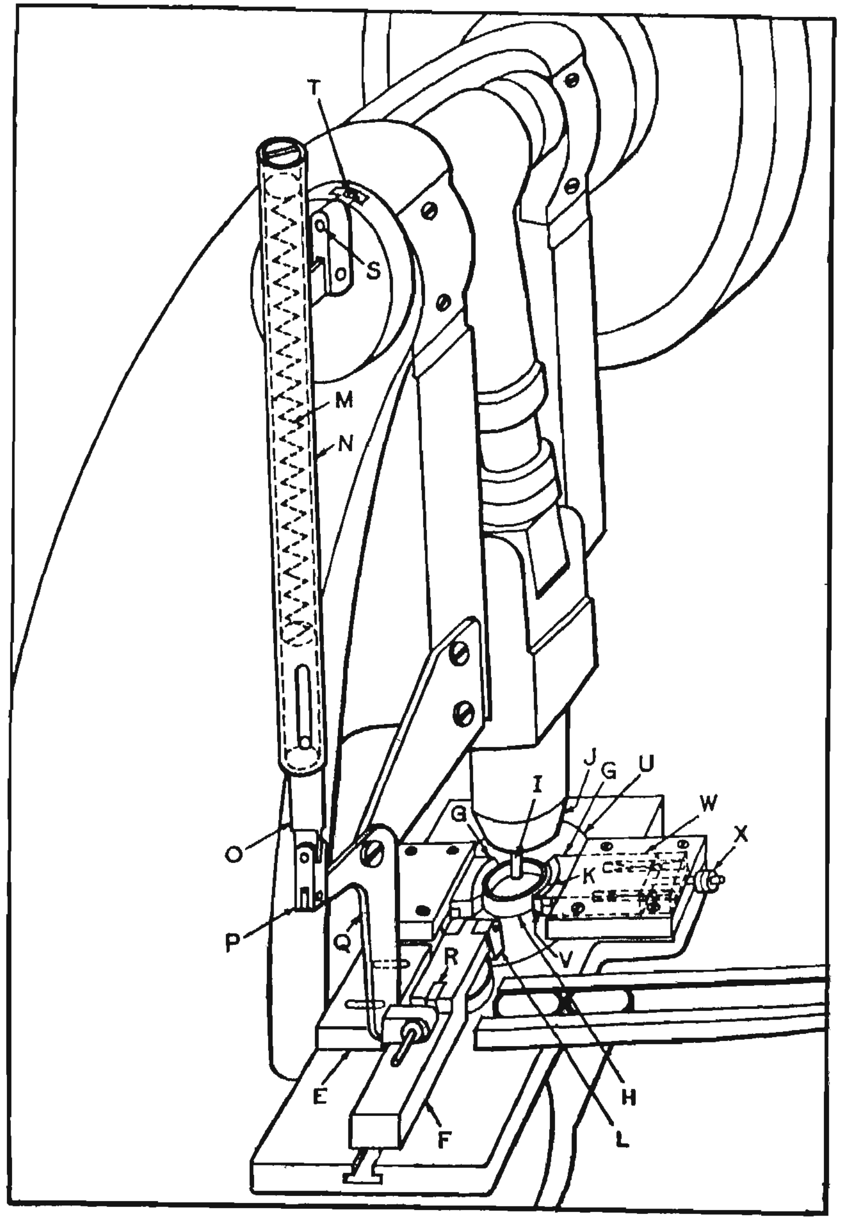


Fig. 8. Automatic Transfer and Feeding Mechanism which Delivers Shells from Another Press and Centers Them Successively on Die H.

On the down stroke of the press, a spring pin I projecting from the center of the punch comes in contact with the stamping and holds it against the die until the punch engages the work. The jaws are then pushed out of the way by contact of the tapered end J of the punch with the tapered surface K of the jaws.

On the up stroke of the ram, the stamping is pushed off the punch by pressure exerted by a spring-actuated stripper located within the punch. The stamping is held down by the spring pin I until it is gripped by the inward moving jaws. The finished stamping is pushed out of the jaws by a pawl attachment L on the front end of the feed-slide F.

The forward movement of slide F, which pushes the stamping from the jaws G, is spring-actuated, while the return movement is obtained by positive mechanical means. The yielding movement provided by the spring M on the forward stroke of the slide prevents jamming in the event that the slide is blocked in any manner, as, for instance, when a stamping is engaged by the slide before it has cleared the chute. The long spring M is contained in a two-part telescoping rod N which is connected to the slide through links O, P, Q, and R. The slide is actuated from the crankshaft of the press by crankpin S, which can be adjusted toward or away from the center of the crankshaft by screw T to vary the length of the stroke. Crankpin S is set 90 degrees ahead of the crankshaft, so that the slide moves forward during the last half of the up stroke of the press ram and during the first half of the down stroke. This setting gives the maximum time available for movement of the slide as required to permit it to clear the punch when handling large or deep work.

Changing the set-up for different operations is accomplished by simply removing the disk-shaped plate U which holds the die H and replacing it with another disk carrying the die for the new operation. The plate U, Fig. 8, is designated H in Fig. 9. Individual jaw adapters V which fit the

work are attached to slides W. The jaws are opened by means of adjusting nuts X. The position of stop E is adjusted to suit the size of the work. In certain cases, the positioning attachment on the front end of the slide is also changed. The most important other modifications apply to the perforating or blanking dies, and consist of providing

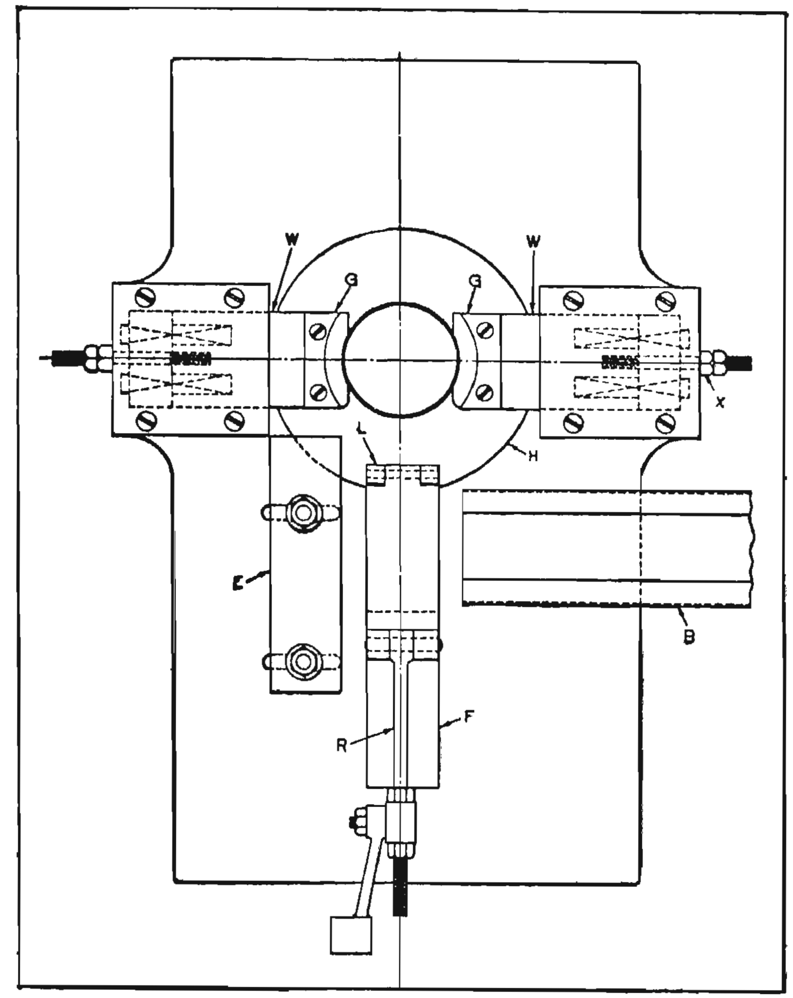


Fig. 9. Diagrammatic Plan View of Principal Members of Die, Feeding Slide and Work Locators of Press Shown in Fig. 8.

guide pins in plate U which are extended to the punch-holder in order to maintain accurate alignment.

When two or more presses are operated in a line, their treadles are connected by linkage, so that when one press is stopped, the others also stop. The presses are run at approximately the same speed, the last ones, however, running slightly faster. With this arrangement, no damage is done in the event that a press is operated without work in its die.

A die with this feed arrangement will handle any kind of stamping work which can be fed through a chute, whether round or square, shallow or deep, or of relatively large or small size. It is only necessary to push the work into the centering jaws approximately close to the required position and it will be automatically centered. This prevents the punch from cutting into a partially entered stamping, as in the case of gravity chutes which carry the blanks directly to the die. Jamming of the die or feed rarely occurs, as previously mentioned, due to the spring action incorporated in the design.

The lateral feeding mechanism which pushes the work into the centering jaws by means of the spring-actuated slide that moves at right angles to the direction in which the stampings are fed through the chute operates rapidly and is practically trouble-free. The operating principle of the mechanism illustrated has been described as it is applied to most types of work. Certain modifications can be made to suit the requirements of special cases.

Mechanism for Ejecting Formed Part from Dovetail-Shaped Punch.—In producing the piece shown in Fig. 10, upper right, on a punch press, it is necessary to strip the work from the forming punch. Although operations of this type are not unusual, this particular operation presented some difficulty because the work was 4 inches long, and the stroke of the press only 2 1/2 inches. As it appeared inadvisable to attempt to strip the work from the punch by a

direct leverage arrangement under such conditions, the stroke-multiplying mechanism shown was devised.

Referring to Fig. 10, lower left, the punch B is carried in the holder A; the die, being of conventional design, is not shown. Punch A carries the grooved bracket D in which the ejector E slides freely in dovetailed ways. The extension K on the forward end of the ejector passes directly under punch B in removing work C on the up stroke.

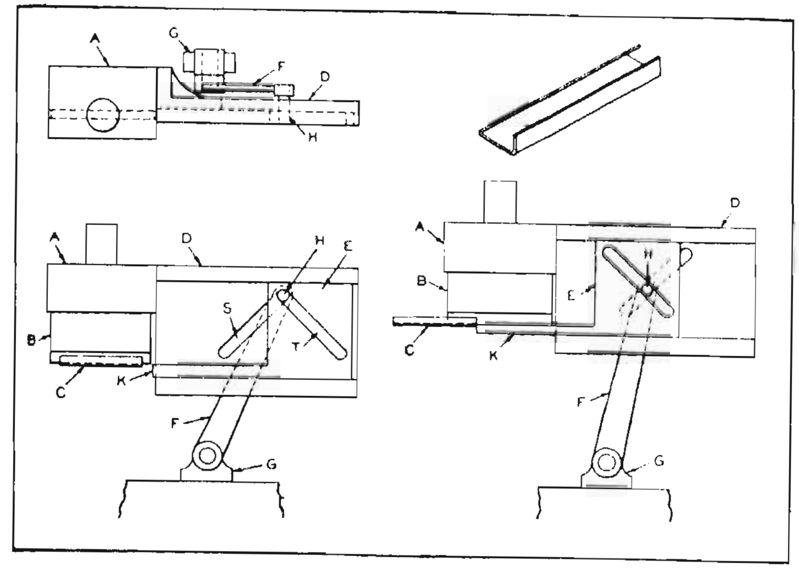


Fig. 10. Mechanism by which Vertical Stroke of Punch Produces Horizontal Movement of Stripper Equal to Twice the Length of Punch Stroke.

A slot S is machined in bracket D at an angle of 45 degrees with the horizontal. Slide E likewise has a 45-degree slot T, which is inclined in the reverse direction. Block G, fastened to the bolster plate of the press, provides a bearing for the oscillating lever F, which carries the pin H at its upper end. Pin H passes through the slots in both D and E at the point of their intersection. In Fig. 10, lower left, the ram of the press is shown at its lowest point, while slide E is held at its extreme rear position.

In Fig. 10, lower right, the ram is shown as it appears after having completed more than half of its upward stroke. As the ram ascends, pin *H* approaches the bottom of the slot in bracket *D*, causing lever *F* to swing forward. As pin *H* passes through the slot in slide *E*, the latter member is carried forward, but because of the angularity of the slot in slide *E*, its movement is twice as great as that of the pin *H*. Thus the vertical movement of the ram is transmitted through slide *E* in a horizontal direction and at an increased ratio sufficient to strip work *C* from punch *B*.

Mechanism for Retarding an Automatic Feeding Device.—A two-stage forming die with an automatic feed is used to form flanges on a relay armature in two different directions. A 1-inch feed on a 2-inch stroke punch press is used for this operation. Only the last 15/32 inch of travel of the 2-inch stroke does the forming. It was necessary, therefore, to retard the automatic feed caused by the 2-inch stroke of the press. This was accomplished by the mechanism here illustrated. This retarding mechanism keeps the feeding device stationary during the forming operation.

When the punch has completed 1 17/32 inches of its travel, bellcrank *H* halts on the head of stop-screw *K*, as shown at the left in Fig. 11. However, the punch continues its travel, and completes the forming operation. As shown at the right in the illustration, when the punch moves up and the bellcrank is lifted off the stop-screw, pawl *M* turns ratchet *N*, thereby feeding a new part into the first stage of the die, moving the partially formed part from the first to the second stage of the die, and pushing a completed part out of the die.

The mechanism was assembled by fastening eyebolt *A* to the punch-holder with screw *B*. Lock-nut *C* and spring-housing *D* were threaded onto the eyebolt. A threaded bushing *J* was placed over one arm of the bellcrank *H*. The end of this arm of the bellcrank was fitted with a spring support bushing *F*, held to the bellcrank by cap-screw *G*. A pawl *M*

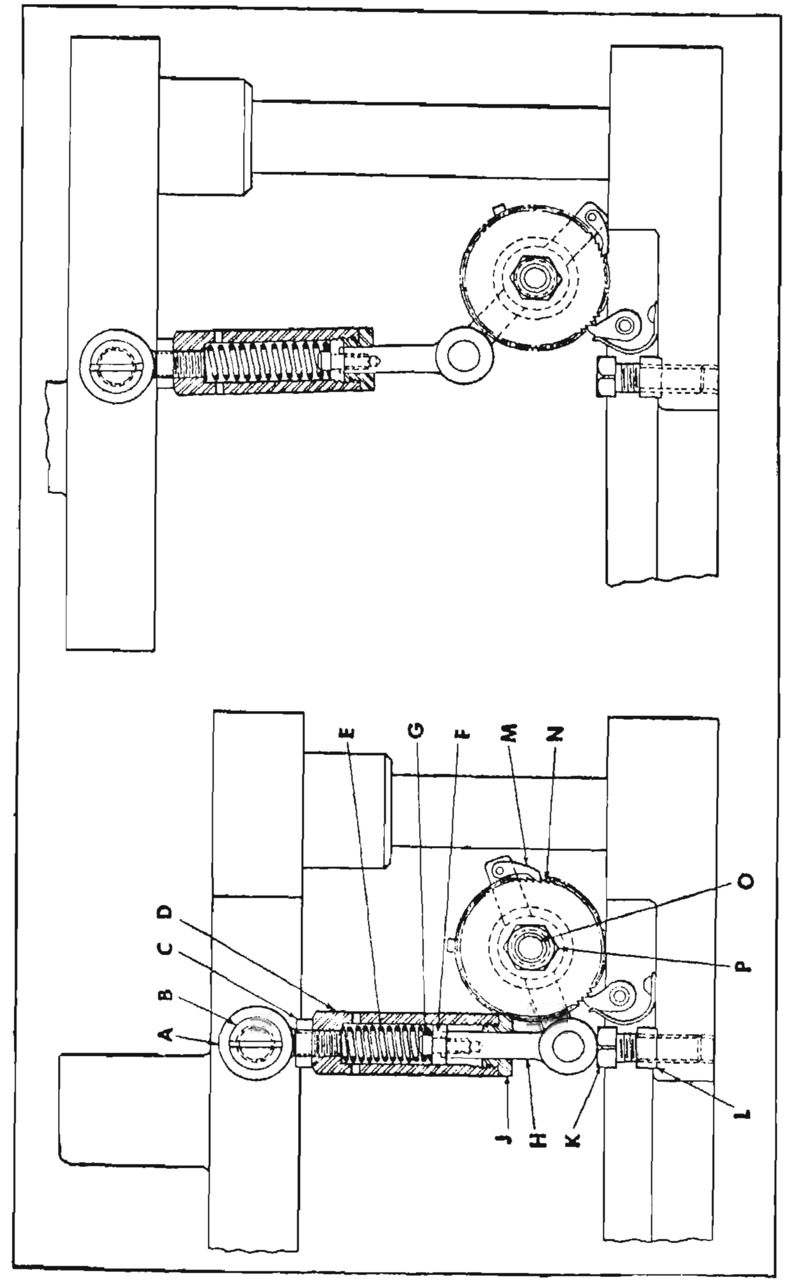


Fig. 11. (Left) Bellcrank H Is Halted by Head of Stop-Screw K as Punch Descends. (Right) As Punch Rises, Pawl M Turns Ratchet N, thus Operating Automatic Feed of the Punch Press.

was pinned to the other arm of the bellcrank. Spring E was placed in the housing D, and then the bellcrank assembly was fastened to this housing by the threaded bushing J.

The arm of the bellcrank that holds the pawl was fitted over drive-shaft O of the feeding device. This arm and ratchet N were retained on the drive-shaft by means of nut P. Drive-shaft O was attached to the die-holding surface of the die set. Stop-screw K was threaded into the base of the die set. Lock-nut L is used to secure the stop-screw in the desired position after adjustments have been made.

Mechanism for Automatically Stopping Press when Stock Fails to Feed.—A certain press operation involves the feeding of two steel strips from rolls into the forming station of the machine, where pieces are cut off from each strip and assembled in automatic dies. To avoid material waste caused by failure of either strip to feed, due to a break in the material or completion of the roll, it was necessary to develop the electrically operated, automatic stop mechanism here described. With this mechanism, the positive clutch of the press is automatically tripped, thus stopping the machine, when either of the strips fails to feed. Modifications of the mechanism can be made to accommodate other types of clutches.

As shown in Fig. 12, the two steel strips X and Z, being fed in the direction indicated by the arrows, are contacted by rolls A and B, respectively. These rolls, mounted on bellcranks F and E, are held against the strip stock by springs C and D. The feeding mechanism (which is not shown) draws the thin strips over idler roll M, thus increasing the tension of the strips. The bellcranks pivot about shaft G, which is supported in the U-shaped bar H that hangs from plate J. This plate is fastened to block K, which is mounted on spindle L of the machine through which the strips pass.

The upper arms of bellcranks E and F are fitted with screws V that make contact with either of the normally open switches R and S when the feeding of the strips is inter-

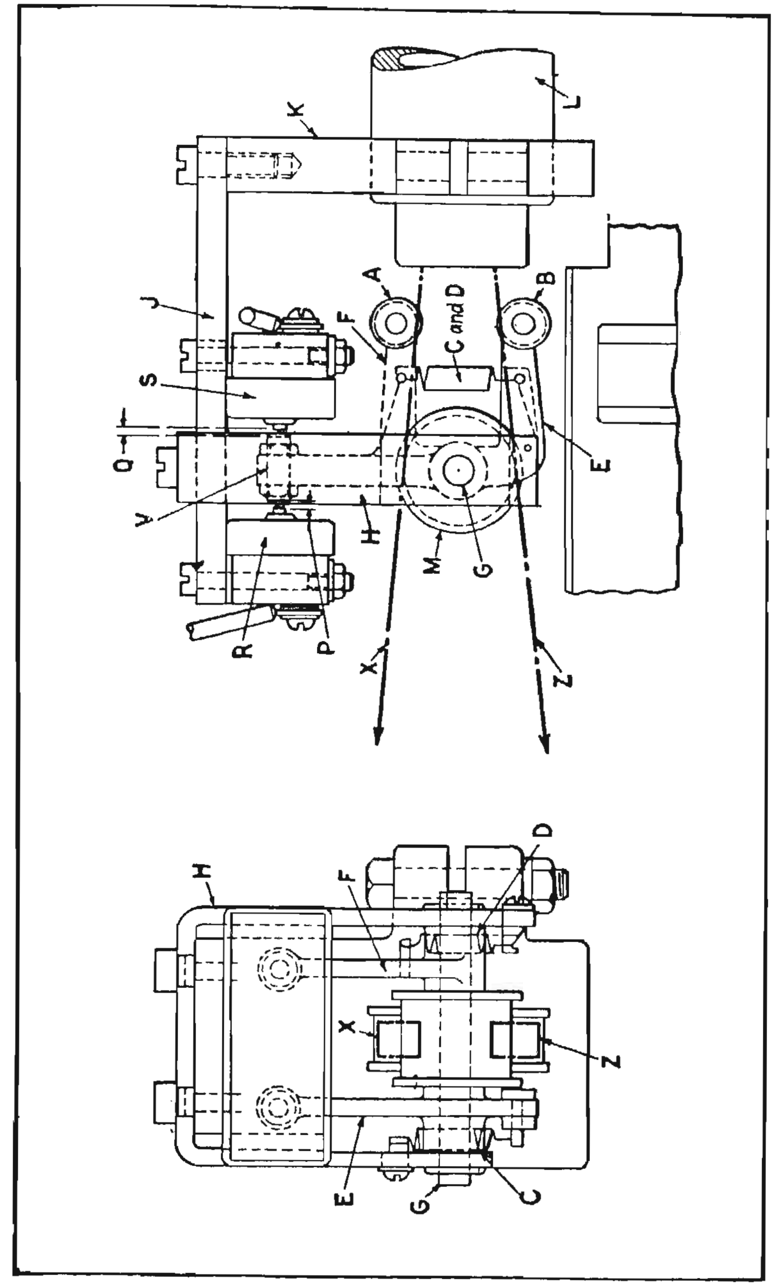


Fig. 12. Mechanism which, in Combination with that Shown in Fig. 13, Automatically Stops Press when Either Strip X or Z Fails to Feed.

rupted. This closing of the electrical circuit is accomplished through movement of bellcrank *E* or *F*, by spring *C* or *D*, depending upon which steel strip is broken, thus causing one of the screws *V* to close gap *P* or *Q*. The switches are electrically connected to solenoid *A*, Fig. 13.

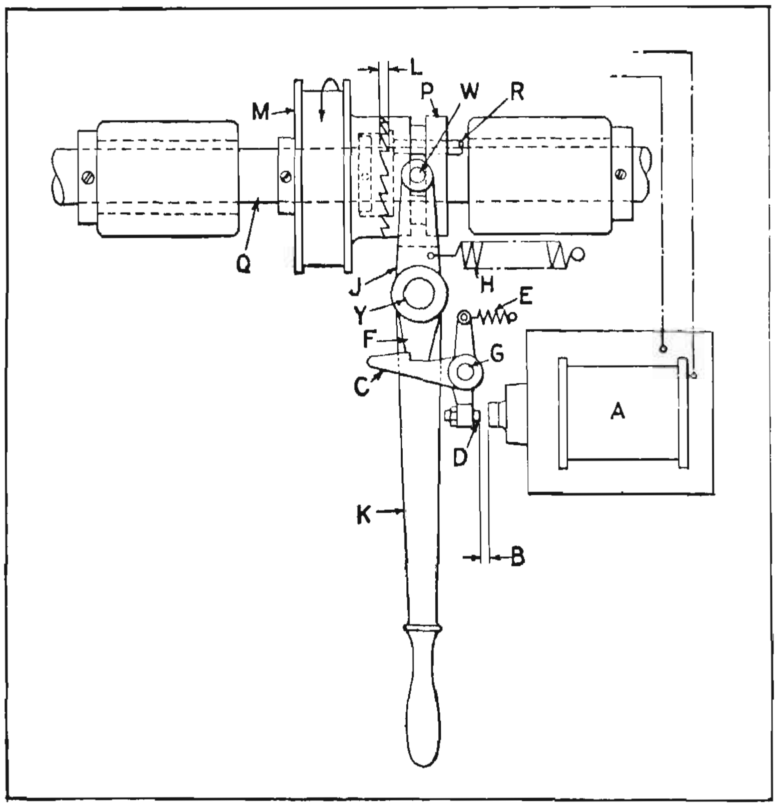


Fig. 13. When Stock Fails to Feed, Teeth on Clutch P are Automatically Disengaged from those on Hub of Pulley M, thus Stopping Press.

Referring to Fig. 13, when the electrical circuit to solenoid *A* has been completed, the contact point *D* will be moved toward the solenoid, closing gap *B*. This movement causes pawl *C* to pivot about stud *G*, thus releasing the end of pawl *F*. Spring *H*, which is connected to yoke end *J* of

pawl *F*, then pivots the pawl around shaft *Y*, thus pulling the clutch teeth out of engagement at *L* and causing the machine to stop.

Pulley *M* rotates in the direction indicated by the arrow, and is held in position on shaft *Q* by collars. Clutch *P* is keyed to the shaft, and when the teeth of the clutch are in engagement with those of the pulley, key *R* causes the shaft to revolve. When the teeth are disengaged, the pulley is free to rotate on the stationary shaft. Pins *W* in yoke *J* fit in the groove in the clutch. The clutch can be re-engaged by lever *K*.

CHAPTER 17

Hoppers and Hopper Selector Mechanisms for Automatic Machines

Tool engineers and machine designers are often faced with the problem of designing mechanisms to pick up parts from hoppers for delivery to the assembly machines. By "hopper feeding" is meant the indiscriminate dumping of a load of parts into a hopper of suitable size and shape, from which the parts are picked up, in the proper position, and deposited in a track for feeding to a machine by gravity. Ordinarily, the pick-up member is so shaped that the parts cannot enter the track if they are not in the right position, and therefore are dropped back into the hopper. Occasionally, the shape of the part and the speed requirements of the machine make it necessary to pick up parts that are not all in the same position. In that case, prior to going into the assembly machine, the parts are required to pass through an auxiliary mechanism, or separator, which arranges them all in the required position.

Many types of hoppers have been designed and built with varying degrees of success. One type of hopper may work successfully for a part of a certain shape, but may prove entirely unsuitable for pieces of a different contour. A great deal of thought must be given to the selection of a hopper for any particular job. Every new problem is unique in some respect, and will necessitate variations in the type of hopper selected.

Centerboard Design of Hopper.—Fig. 1 illustrates the centerboard hopper, a highly successful type when used to pick up parts within its limitations. The hopper body may be made of cast iron or cast aluminum, or it may be of welded steel construction. Side and end section views illustrate the general construction.

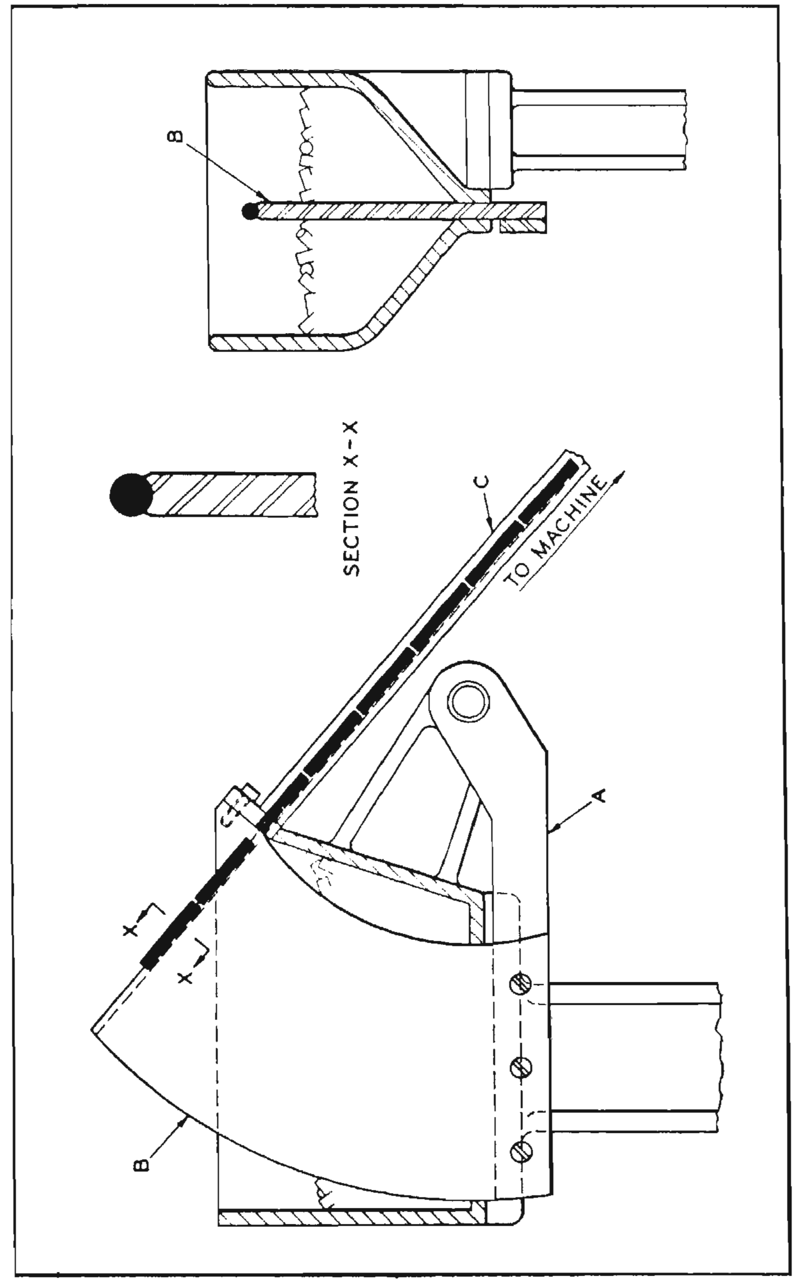


Fig. 1. Centerboard Hopper for Picking Up Rod-shaped Work or Parts of Angular Cross-section.

An arm *A* actuates a hardened centerboard blade *B*, which oscillates up and down through the mass of parts, picking up a few of them in the groove machined in its top edge. At the top of the stroke, this groove is in line with a track or tube *C*, and the parts slide down toward the machine. When the track is full of parts, those remaining on the centerboard fall back into the hopper. It is important, when the track is full, for the end of the last part to come flush with the end of the track. If it should project into the hopper, a jam would occur when the centerboard rose on the next stroke.

Another important design factor is to so lay out the centerboard arcs that the point of delivery, where the track joins the centerboard blade, is as high as possible. In that way, the greatest number of parts can be placed into the hopper at one time, and it will not be necessary to refill it as often. In at least one case the delivery point on several hoppers for the same machine was placed so low that the capacity of the hopper was too small for the speed at which the parts were being taken out. Careful thought to good hopper design will prevent these costly mistakes.

Section *X–X* shows the correct form for the top portion of the blade when used to pick up parts of round cross-section. It will be noted that the centerboard width is the same as the part diameter. However, it is machined so that it will bear on only one-quarter of the part diameter. This form has been found highly successful.

The centerboard blade should be chromium-plated. This serves two purposes. It allows the parts to slide more freely into the track, and the slippery surface thus provided prevents wedging of the parts between the blade and the bottom of the hopper on the downward stroke of the blade.

A cam is generally employed to actuate the blade, so that it will have a slow upward travel and a quick return. However, cranks have been used successfully for this purpose. They are run at speeds not greater than 40 R.P.M.

The centerboard type of hopper is recommended for all round parts having a length greater than twice the diameter. It can also be used for small disk-shaped parts by machining the groove at the top of blade to a depth of one-half the part diameter so that the parts are held as shown at *A* in Fig. 2.

An interesting application of centerboard hopper design is shown at *B* in Fig. 2. Here the blade *C* is machined at an angle, so that parts with angular projections can be picked up facing in only one direction. A stationary baffle plate *D* must be incorporated in this design to slide the parts over into the correct position at the top of the stroke. It can be seen that a part cannot be picked up in the wrong position, since it would slide off the edge of the blade, as shown at the right of Fig. 2.

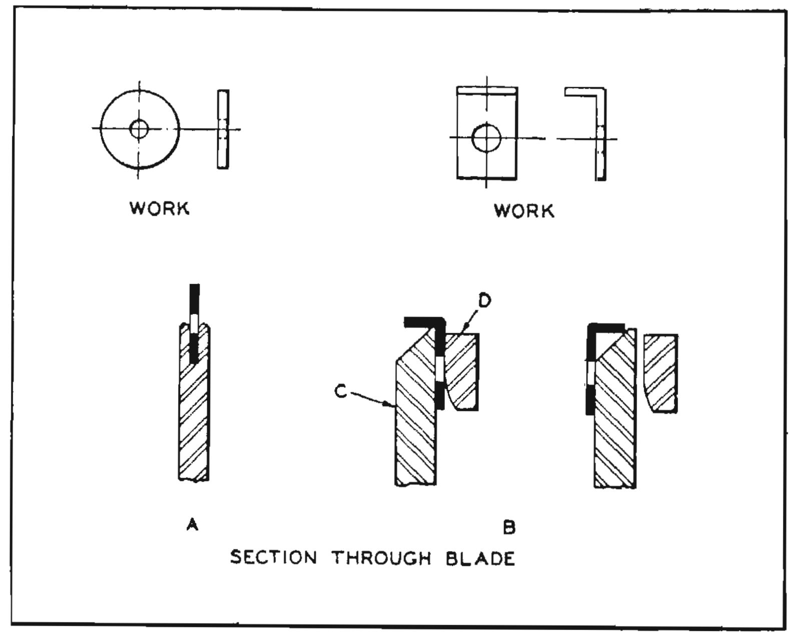


Fig. 2. Blade Designs Used for Picking Up Parts in Centerboard Hoppers.

Rotary Centerboard Hopper.– The rotary centerboard type of hopper can be applied to many automatic machines. By varying the blade cross-section, it can be easily adapted for parts of different shapes. This hopper has been found to be exceptionally satisfactory for channel-shaped parts of the type shown in Fig. 3.

The rotary blade *A* is given an intermittent motion, either by a Geneva wheel or by a pawl and ratchet arrangement.

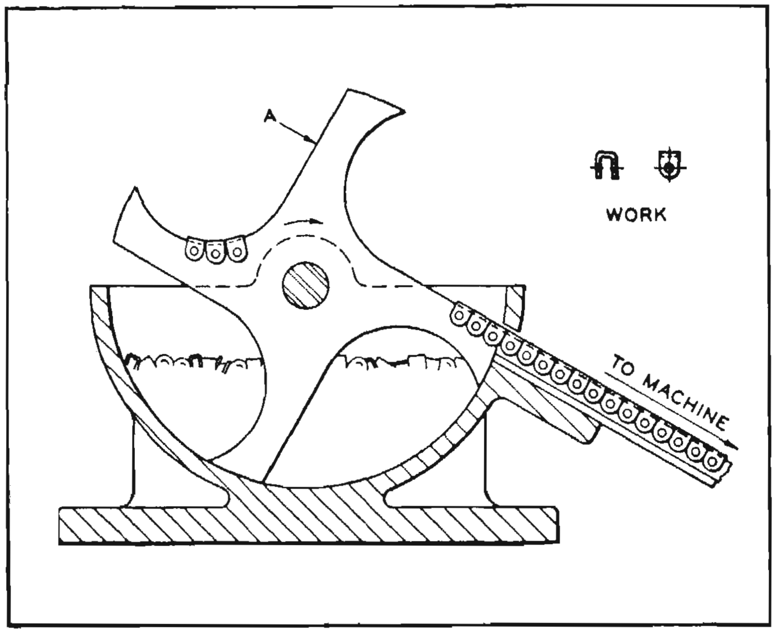


Fig. 3. Rotary Centerboard Hopper for Feeding Channel-shaped Parts to Automatic Machines.

The latter, because of its comparatively low cost, is generally used. As in the previous case, the blade should be hardened and ground, and then chromium-plated.

As with all hoppers for feeding parts, the wheel rotation should be as slow as possible, considering the feed requirements of the automatic machine which it is supplying. An ideal drive consists of a cam-operated ratchet, with a quick-

return pawl and a slow blade movement. A slight dwell lobe should be incorporated in the cam at the point where the blade comes opposite the track. This gives time for the parts to slide from the hopper blade to the track. The cam should be of the harmonic motion type that allows the wheel to start slowly, accelerate gradually, and finally come to a stop. In that way, the parts that have been picked up on other arms of the rotary blade will not be knocked off by abrupt stops.

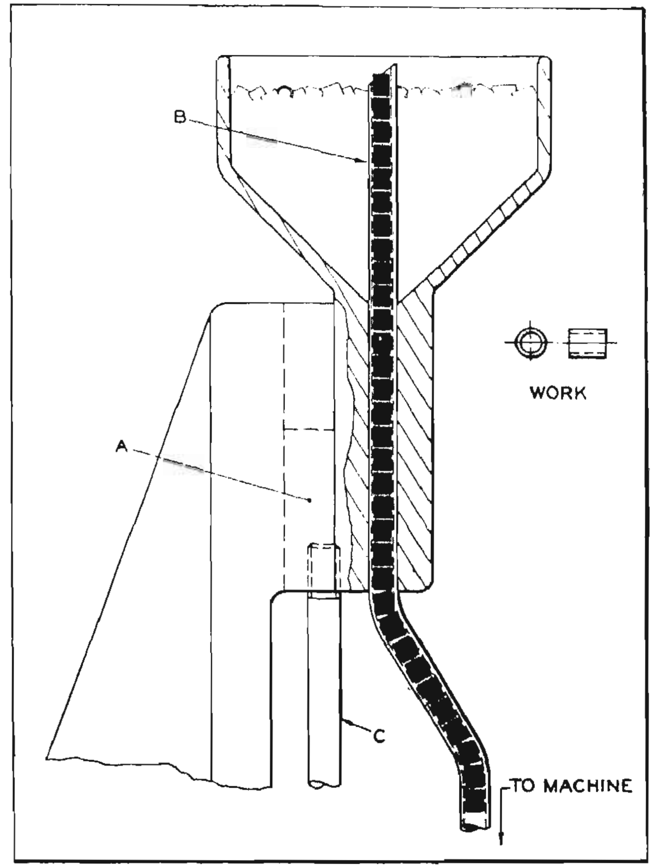


Fig. 4. Hopper Employing a Reciprocating Motion to Feed Short Parts through Stationary Tube to Machine.

Tube and Rotary Types of Hoppers.—Fig. 4 shows a good design of hopper for feeding short parts, such as illustrated at the right. The hopper body consists of a cast funnel which has a dovetailed slide *A* machined in its lower side. The feed-tube *B* is a sliding fit in a reamed hole in the hopper body as shown.

In operation, the hopper body is given an up and down movement by the actuating rod *C*. On the downward stroke, a number of parts enter the open end of the tube and slide down to the machine. The open end of the tube is machined at an angle so that, if a part falls crosswise of the opening, it will be knocked off by other parts on the upward stroke of the hopper. When the feed-tube is required to be bent, as shown, the inside diameter of the tube must be sufficiently greater than the diameter of the part so that the parts will not bind in turning the corner.

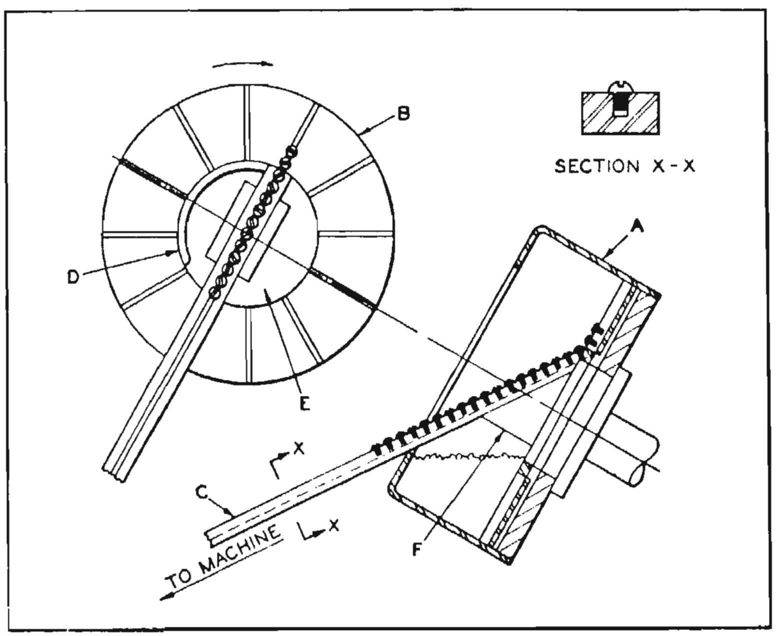


Fig. 5. Rotary Hopper for Handling Wide Variety of Work such as Screws.

The rotary type of hopper has proved practical on a wide variety of jobs. As illustrated in Fig. 5, it consists of a large round container *A*, mounted at an angle, with grooves machined in the baseplate *B*. When the baseplate rotates, some of the parts in the bottom of the hopper fall into the grooves in the baseplate and are carried up in line with the opening in track *C*. When the track is full, succeeding parts are carried past the track opening and fall back into the bottom of the hopper.

A baffle *D* is mounted on the stationary center *E* as shown. This prevents the parts that have been picked up from falling back into the hopper before they have passed the track opening. The track is mounted on a stationary bracket *F*. The set-up illustrated was designed for feeding short screws to an automatic screwdriver in such a way as to insure a continuous supply and correct positioning.

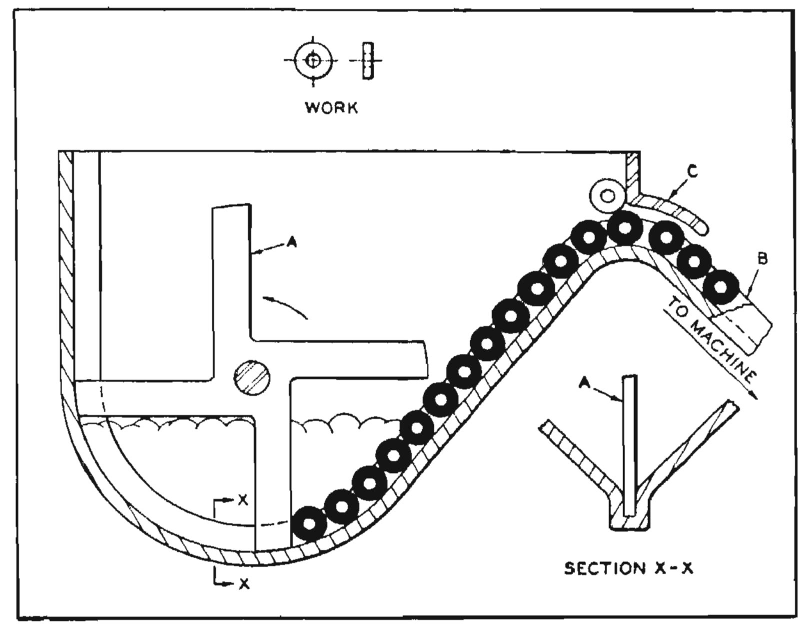


Fig. 6. Paddle-wheel Hopper for Feeding Flat Work, Varying in Shape from Square to Round.

Hopper for Feeding Flat Work of Both Round and Square Shapes.—The hopper shown in Fig. 6 is used for feeding flat work varying in shape from square to round. It is an inexpensive and very efficient design for such parts. Section *X–X* shows the shape of the groove at the bottom of the hopper. The parts that fall into this groove are pushed up the incline by the paddle wheel *A*. When the parts have been pushed up far enough, they enter the track *B*, down which they fall to the machine. When the track has become full, the succeeding parts ride up the baffle *C* as shown, and fall back into the hopper.

Barrel Hopper for Intricate Shapes.—The barrel type of hopper, while more expensive to build, is sometimes the only one that will successfully select parts of very intricate shape, including those that interlock when grouped together. It consists of a rotating hopper *A*, Fig. 7, which is shaped like a barrel and is open at both ends. The parts are dumped into hopper *A* through a loading hopper *B*.

On the inside of the rotating hopper are cast longitudinal fins, as shown in section *X–X*. Rotation of the hopper, which is mounted on bearings *F* and driven by the pulley *G*, causes the parts to flow in a steady stream down a blade *C*, which is set at a slight angle and connected at one end to an electric vibrator *D*. The other end of the blade is aligned with the track *E*, which leads to a machine.

The constant agitation of the parts resulting from this arrangement prevents interlocking, and some of the parts will fall on the blade in the right position. Vibration of the blade causes the correctly positioned parts to move toward the track and enter it. The loading hopper shown makes filling of the rotating hopper an easy matter.

Hopper for Feeding Rivets and Similar Shaped Parts.—The hopper shown in Fig. 8 is designed to select and feed rivets to a machine in the correct position for the riveting operation. However, it can also be used for many special parts of similar shape. The rotating portion *A* of the hop-

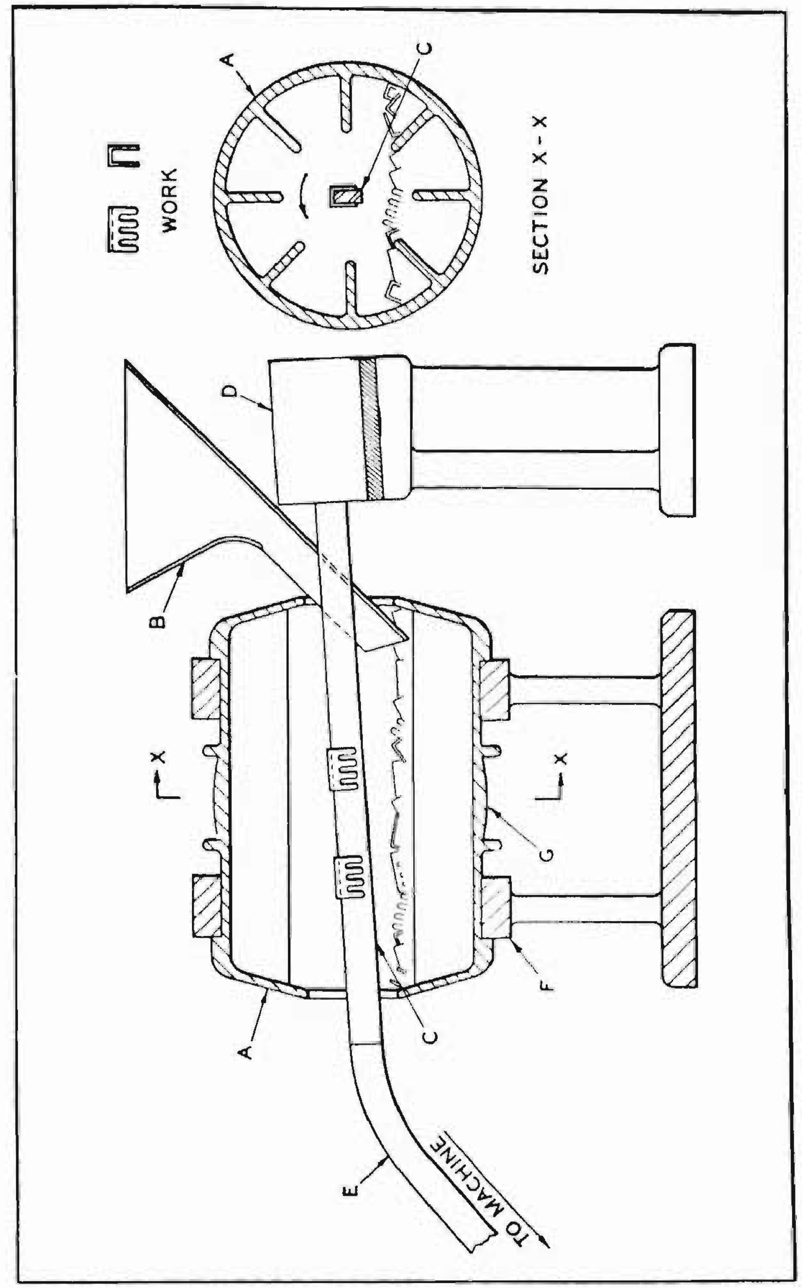


Fig. 7. Barrel Hoppers to Select and Feed Work of Intricate Shape, Including Parts that Interlock when Grouped Together.

per body contains a series of grooves as shown. Re-entrant groove *G* is machined in the stationary part *B* of the hopper body to accommodate the rivet heads. Rotation of the hopper by drive-shaft *C* allows a few rivets to enter the slots. When they are opposite the track *D*, an opening in the body allows the rivet heads to enter the track, and the rivets slide down toward the machine.

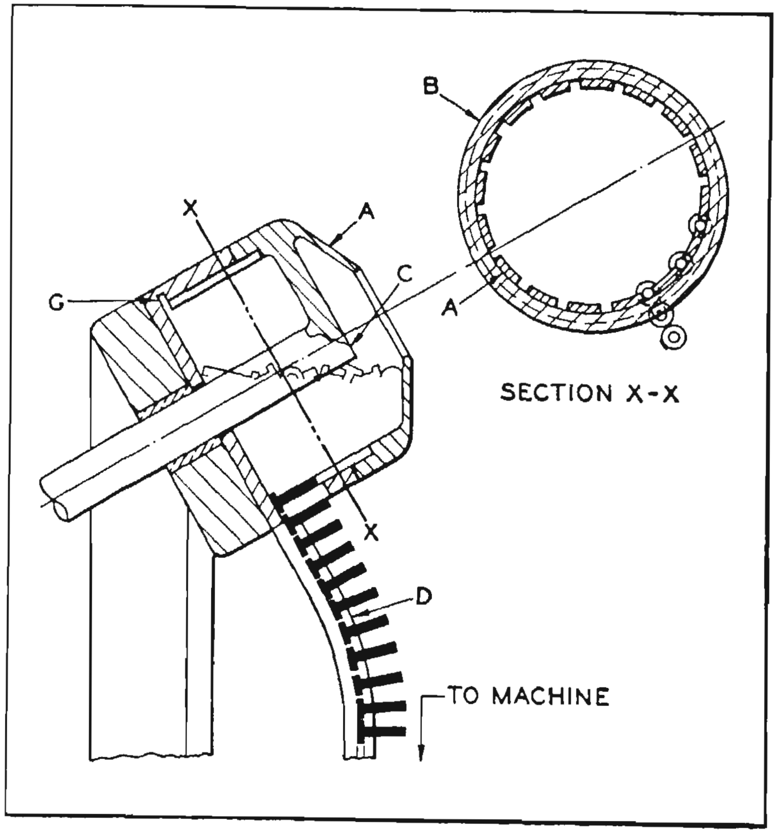


Fig. 8. Hopper for Feeding Rivets and Similarly Shaped Parts.

Tray Type Hopper for Comparatively Low Production.—The tray type hopper is a cheaply built hopper for feeding larger parts in comparatively low production. The operator

places the parts in the tray *A*, Fig. 9, and they are moved toward the track *B* by vibration. An agitator *C*, operated by a small crank, prevents jamming at the mouth of the track.

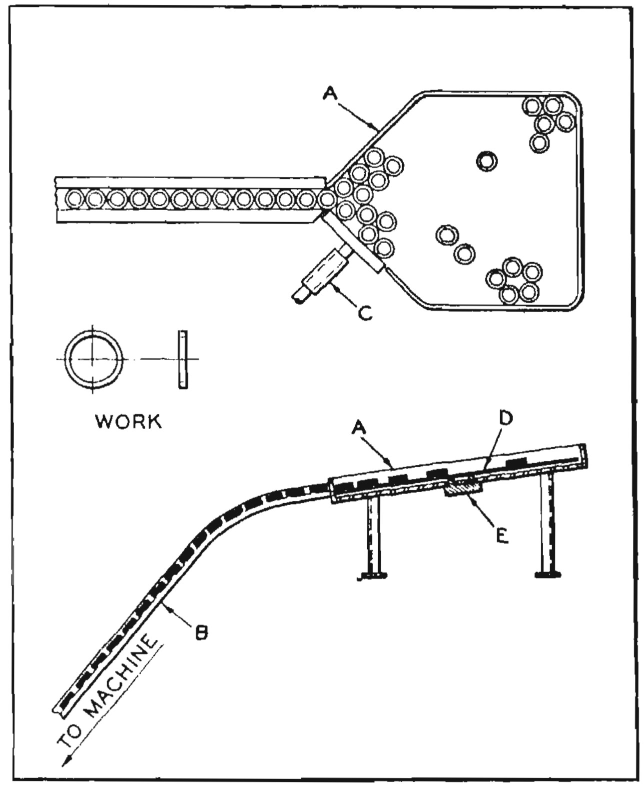


Fig. 9. Tray Hopper for Feeding Large Parts to Machines at Comparatively Low Production Rates.

The construction is simple. A vibrating plate *D*, set close to the bottom of the tray, is connected to the rods of a commercial electric vibrator *E*. The angle at which the tray must be set is determined by experiment, and is usually about 4 or 5 degrees. With the agitator operating, the parts then move readily to the mouth of the track.

Vibratory Hopper for Greater Production Requirements.—Where greater production requirements exist, the hopper illustrated in Fig. 10 will be found applicable to a wide variety of parts. It consists of a commercial vibratory feeder *A*, suitable guiding baffles *B*, a hopper C, and an agitator *D*.

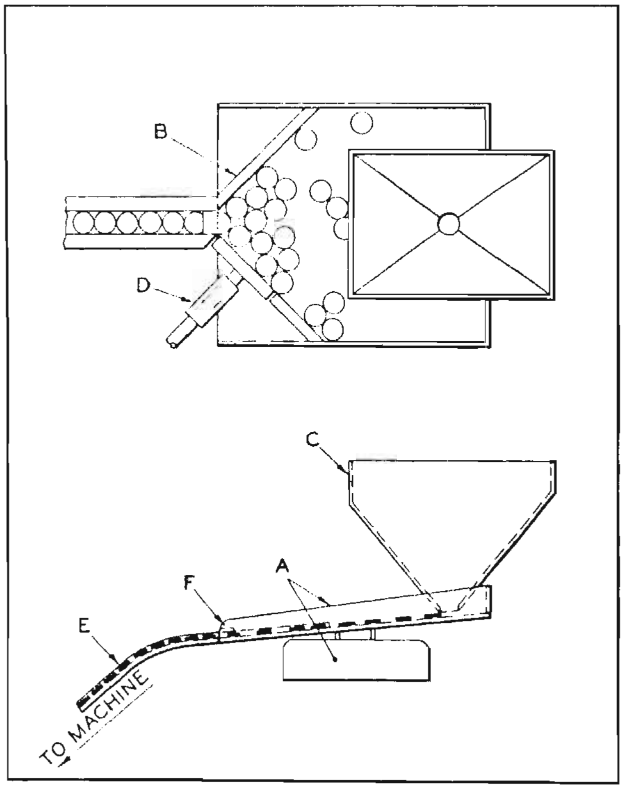


Fig. 10. Vibratory Type of Hopper for High Production Requirements.

The parts are dumped into the hopper, and the action of the vibratory feeder causes them to flow in a steady stream from its mouth. Owing to the slight incline of the pan, the parts flow toward the track *E*, being guided by the baffles.

As in the previous case, an agitator prevents jamming at the track mouth. A gate *F* is incorporated at that point, so that only one part can go through at a time. If a part is lying on top of another, the gate will prevent both from going through at once. The vibration will cause the lower part to enter, while the top part will fall off and enter the track in its turn.

Hopper of Magnetic Design.—An interesting hopper application is shown in Fig. 11, where Alnico magnets *A* are used to pick up the parts from the hopper *B*. The magnets

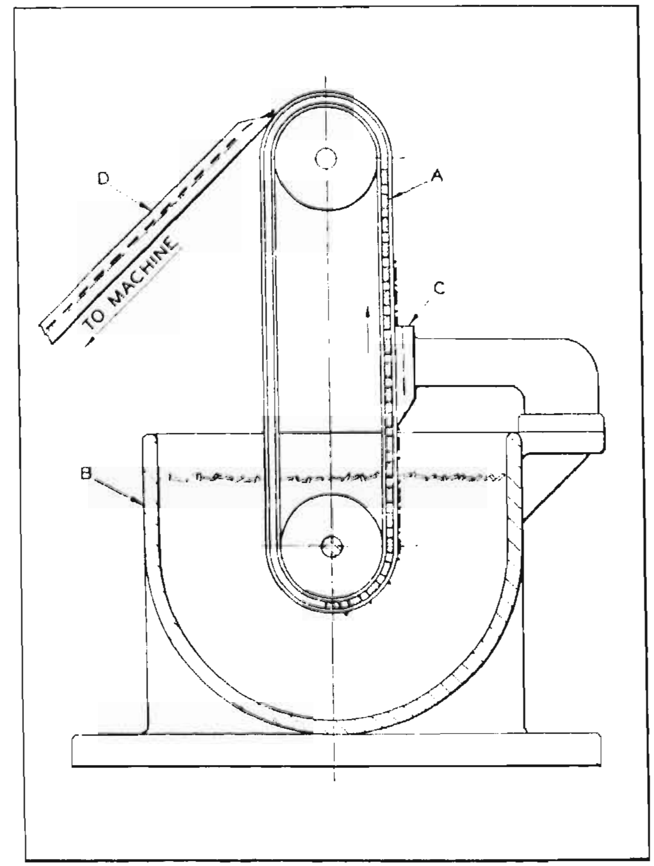


Fig. 11. Hopper which Uses Magnets to Pick Up Parts.

are incorporated in a conveyor belt which passes through the parts, some of which cling to the magnetic stations and are carried up. The stripper and side-guide arrangement *C* properly locates parts that were not picked up in quite the right position. In case one magnet picks up two parts, the side guide will strip the extra part, which falls back into the hopper.

At a point tangent to the top pulley, a track *D* strips the parts from the magnets. The track must be made of a non-magnetic material at that point. Further down, the parts pass through a demagnetizer if the magnetic properties imparted to them are objectionable.

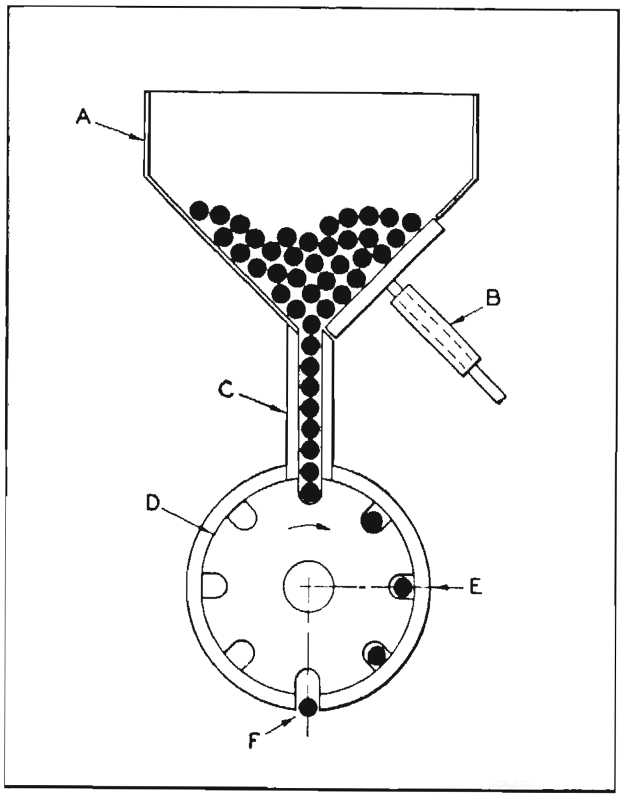


Fig. 12. Hopper for Feeding Long Rods with Agitator and Transfer Wheel which Indexes to Carry Rods to Work Station.

Simple Hopper for Feeding Long Rods.—Fig. 12 illustrates a simple hopper for feeding long rods to an automatic machine. The rods are loaded into the hopper *A*, and an agitator *B* insures a constant flow into the track *C*. At the lower end of the track, the rods enter grooves machined in a transfer wheel *D* and are carried to a work station *E* as the wheel indexes. At that point, the parts are automatically clamped and held securely while drill heads machine a hole in each end. At the ejection station *F*, the rods fall out into a box.

Spring-Actuated Pin Type Selector Mechanism.—In designing hoppers for feeding parts automatically into assembling machines, cases are often encountered where the work is required to enter the machine in a certain position for correct assembly with other components. To accomplish this, the hopper must be provided with a selector mechanism.

In the example shown in Fig. 13, one end of the workpiece is turned down to a slightly smaller diameter than the remainder of the piece, and the part is required to be fed into the machine with the small end first. A part of this shape would ordinarily be handled by a centerboard hopper such as described earlier in this chapter. However, the centerboard blade may lift the part up in either of two positions, with the smaller end toward the machine or with the larger end facing in that direction.

To insure that the piece will enter the machine in the correct position, a selector mechanism designed as shown in the illustration may be employed. This mechanism is located at some point between the hopper and the machine. Parts fed from the hopper fall into the selector wheel of the device, which is designed so as to allow the pieces that have been picked up correctly in the hopper to pass through into the machine, while those that are in the wrong position are turned, end for end, before entering the machine.

Referring to Fig. 13, it will be seen that the selector mechanism consists of a housing A, a stationary center plug B, and an intermittently rotating ring C, driven by a ratchet wheel J. A circular groove D is machined in the center plug, which is slightly wider than the small diameter of the workpiece. A hardened pin E is pressed into the center plug, and a spring-actuated pin F is located two stations away

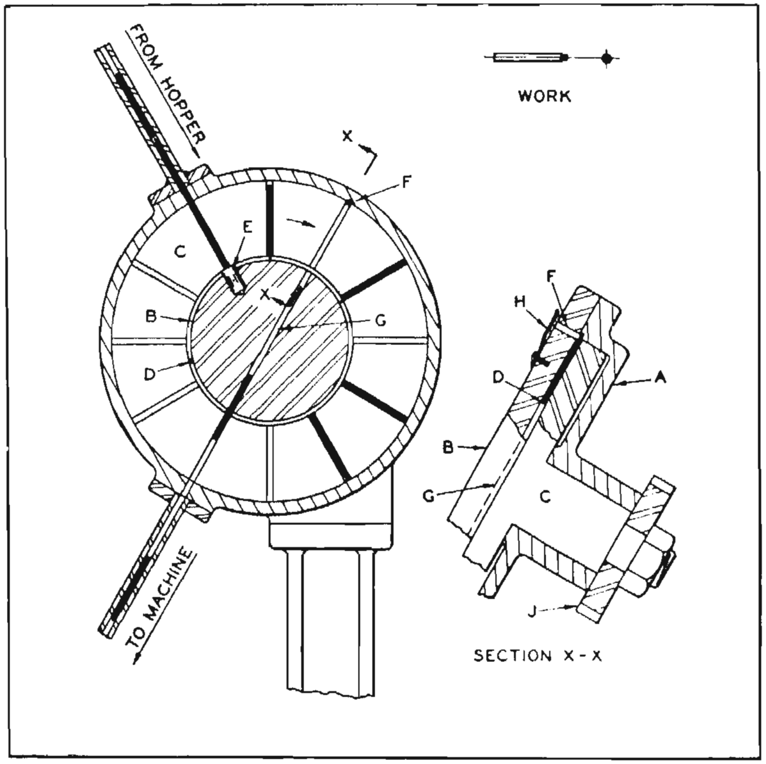


Fig. 13. Spring-actuated Pin (F) Prevents Work Fed from a Hopper in the Incorrect Position from Being Transferred to a Machine.

as shown. Pin F and the flat spring H may be seen more clearly in section X–X.

In operation, the parts slide down a tube from the hopper and enter a groove machined in the rotating ring C, coming to a stop against the end of the hardened pin E. When the

ring indexes, the parts are carried to the idle vertical station. As they leave the pin E, the small end of those pieces that are positioned correctly will enter groove D. Pin F at the next station will not touch the parts in the groove, due to their lowered position, and thus they will be free to enter a groove G machined in the center plug, through which they slide out of the selector and into a tube leading to the machine.

An incorrectly positioned part, having its small end toward the selector housing, cannot enter groove D. Hence, when it comes opposite groove G in the center plug, the spring-actuated pin F will bear on the end and hold it from sliding down the groove. As the wheel continues to index, the part is carried around, so that when it comes opposite the tube leading to the machine it will have been turned end for end, and thus enters the machine with the small end first, as required.

In laying out this type of selector wheel, the length of the tube between the selector and the machine must be great enough to hold at least six parts—preferably more. This would insure a sufficient supply of parts to the machine in case all the pieces for a certain period of time came from the hopper in the wrong position and had to be carried around by the wheel and turned before being fed to the machine.

This type of selecting device has been found successful for a wide variety of parts. Occasionally, a great deal of ingenuity is required to design the actual method of selecting, but once this is accomplished, hopper problems can be met that would otherwise be impossible to solve.

Circular Blade Type Selector.—To handle a part formed as illustrated in Fig. 14, the rotating ring in the selector would have a circular blade A pressed into it as shown. The parts that are in the correct position as they come from the hopper rest on this blade, and when they are opposite the groove in the center plug, slide down toward the assembly machine. Those parts that are incorrectly positioned are held by the blade, as shown at section *X–X*. In this case, as in the previous example, the part is carried around to the bottom of the ring, where it is in the proper position to enter the machine. There it slides over the inclined face of the retaining blade and into the feed-tube.

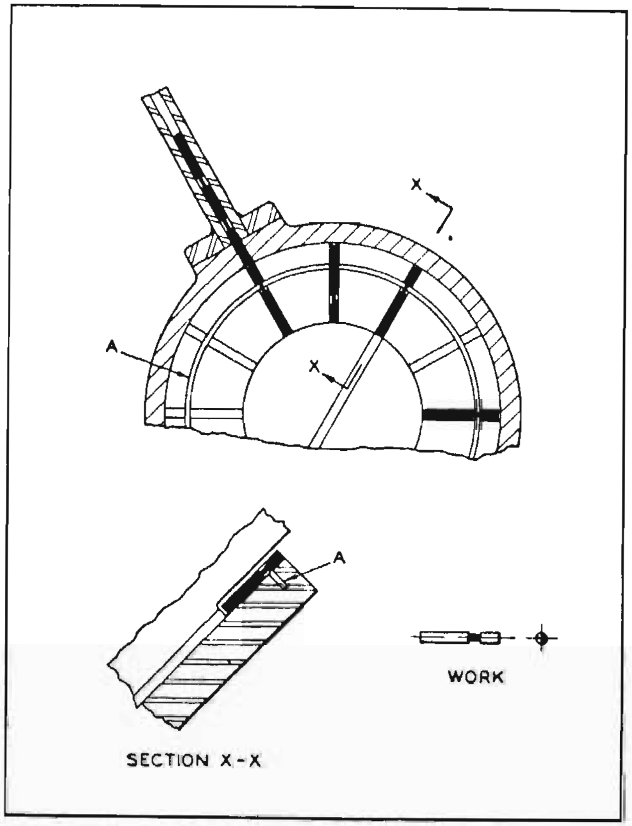


Fig. 14. Parts of Shape Shown are Uniformly Positioned by Selector that Holds Incorrectly Located Parts until They are Turned Over.

Magnetic Type Selector.— A magnetic selecting device can be used for certain parts, as shown in Fig. 15. In this case, a permanent magnet *A* is employed at the selecting

station for work having a pointed end, as seen in the illustration. The magnetic attraction is not great enough to hold the parts when the point is toward the magnet, and they slide through to the machine. However, when the flat end of the part is toward the magnet, it is held and carried around by the wheel so that it enters the tube leading to the machine in the desired position.

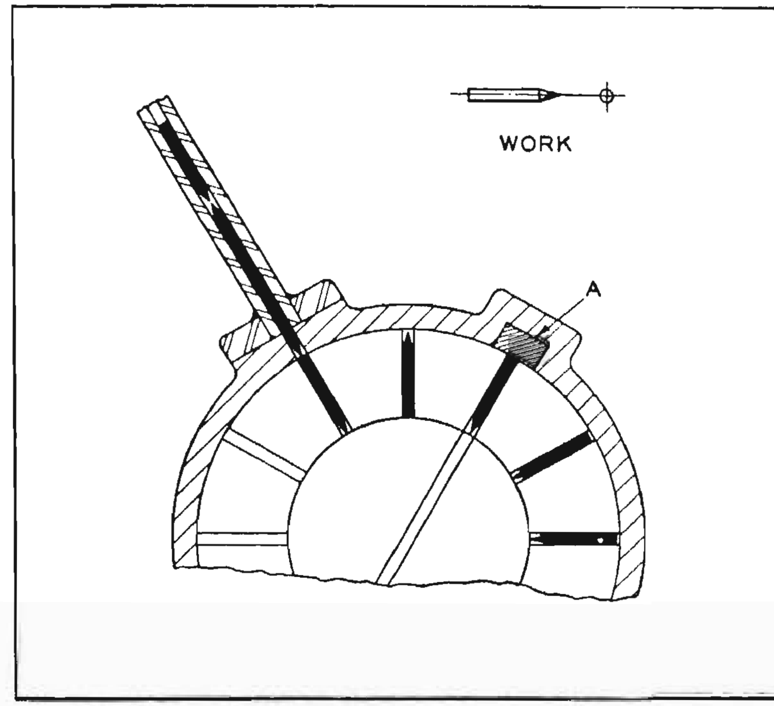


Fig. 15. Permanent Magnet (A) Prevents Parts that are Fed to it in the Wrong Position from Being Transferred to a Machine.

This is a cheap and efficient selecting device for certain types of parts. It is ideal for use in connection with pin-driving machines. In this case, the rotating ring must be made of some non-magnetic material, and so must the tube leading to the machine. Occasionally, it will be found necessary to provide a demagnetizing coil around the tube.

Selector for Flat Hooked Parts.—Fig. 16 illustrates a selector designed for flat, hook-shaped parts. A blade *A* in the cover engages the hook of incorrectly located parts and prevents them going through to the machine. When the parts are correctly positioned, the hook does not engage the

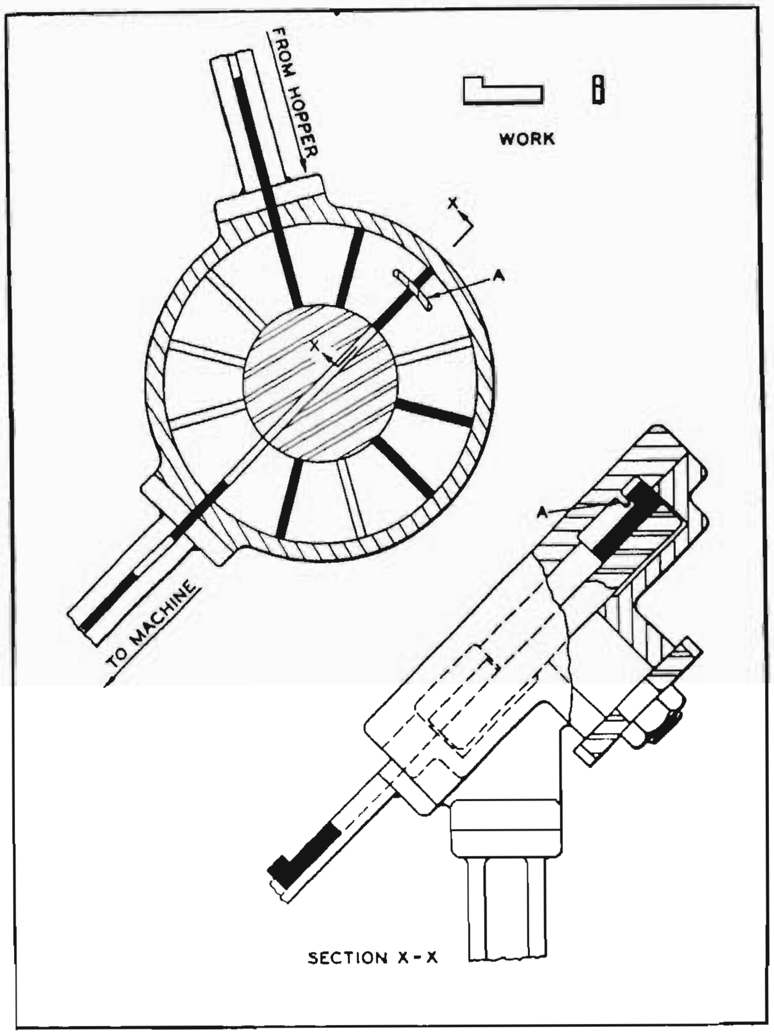


Fig. 16. Blade (A) Prevents Hook-Shaped Parts from Dropping Through to a Machine When They are Fed to the Selector in the Wrong Position.

blade, and the parts are free to go through. Section *X–X* shows the details of construction; it will be noted that the selector, in this case, is mounted at an angle rather than vertically. This is the most positive type of selector, and should be used whenever possible. Unfortunately, many parts do not lend themselves to such positive treatment, and spring pressure or magnetic properties must be resorted to.

Selector for Shallow Drawn Parts.—Shallow drawn parts, such as covers for various units, can easily be handled by means of the vertically mounted selector wheel shown in Fig. 17. With this arrangement, the covers enter slots in the rotating ring, and those with the open side up slide through the groove to the machine. Incorrectly placed covers are held from going through by the plugs *A*, and are carried around by the ring, falling into the feed-track right side up. In this case, the rotating ring must be fabricated sectionally in order to permit assembly of the plugs *A*.

Cup or Can Selector.—The simple device shown in Fig. 18 is used to position cans so that they will all be fed to a machine with their open ends up. In use, they slide down a tube from the hopper and hit a projecting pin *A*. Those that come down bottom first simply bounce from the pin and fall down the vertical tube in the same relative position. The cans that come down open end first catch on the projecting pin and flop over, so that they fall down the tube toward the machine in the desired position.

Selectors for Reversing Position of Parts.—It often happens that the easiest method of selection is to place the parts in the opposite position to that required by the sequence of operations in a particular machine. This was true in the case shown in Fig. 15, where the parts had to be fed to the machine point first, but were selected so that the opposite ends faced the machine.

In such cases, the track arrangement shown in Fig. 19 can be used to turn them end for end. This consists of a track with a break in it, arranged as shown. Two side plates

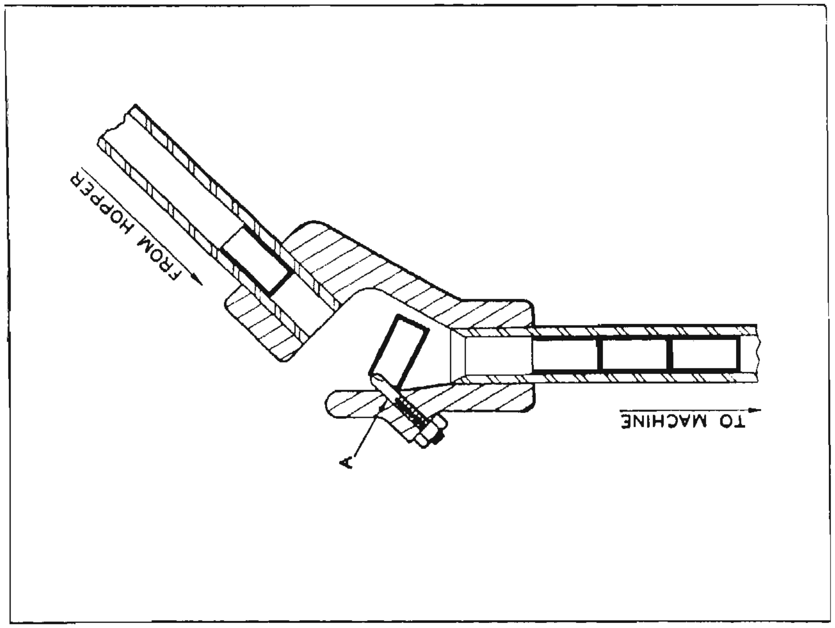


Fig. 18. Cups are Turned Over or Passed through this Selector so as to Enter a Track in a Uniform Position for Feeding to a Machine.

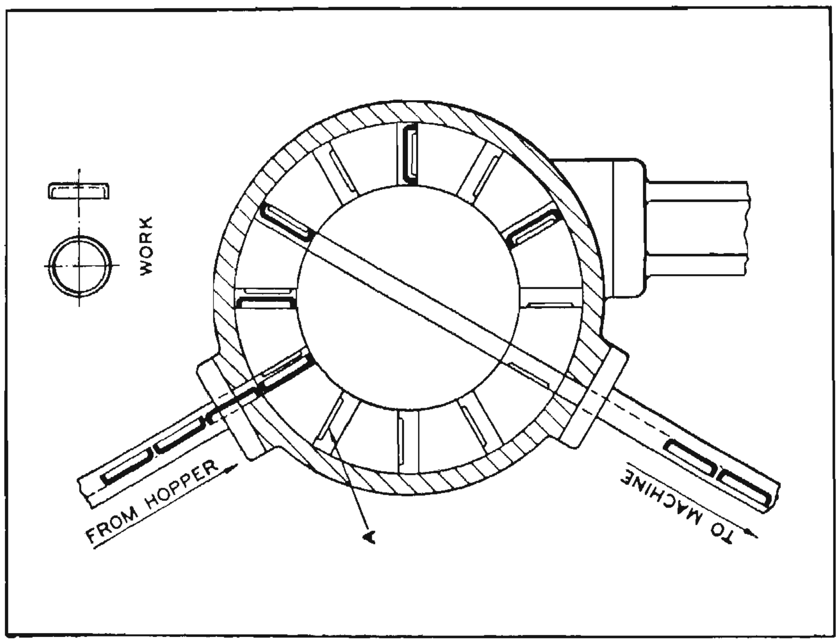


Fig. 17. Selector which Holds Incorrectly Positioned Shallow-drawn Parts by Plugs (A) and Prevents them from Being Fed to the Machine.

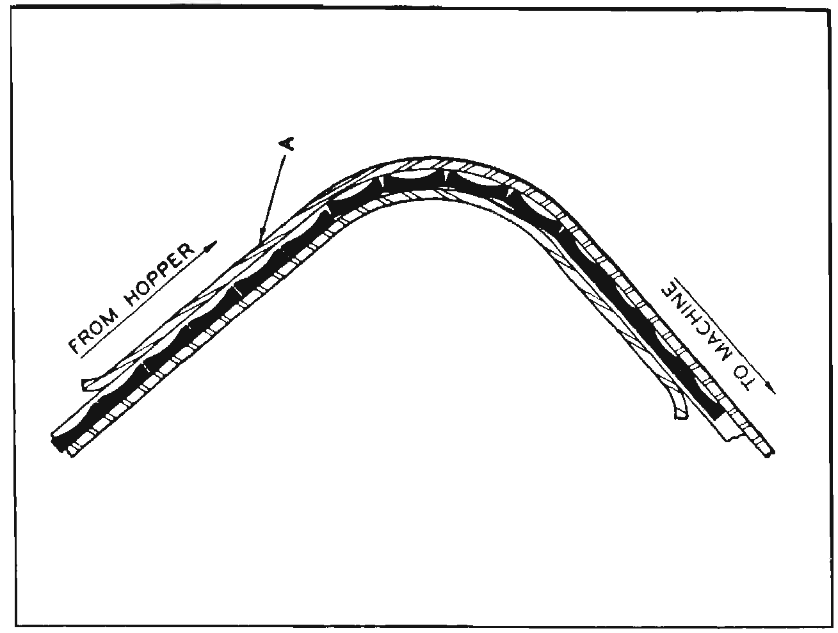


Fig. 20. Another Track Arrangement for Reversing Position of Parts Received from Hopper.

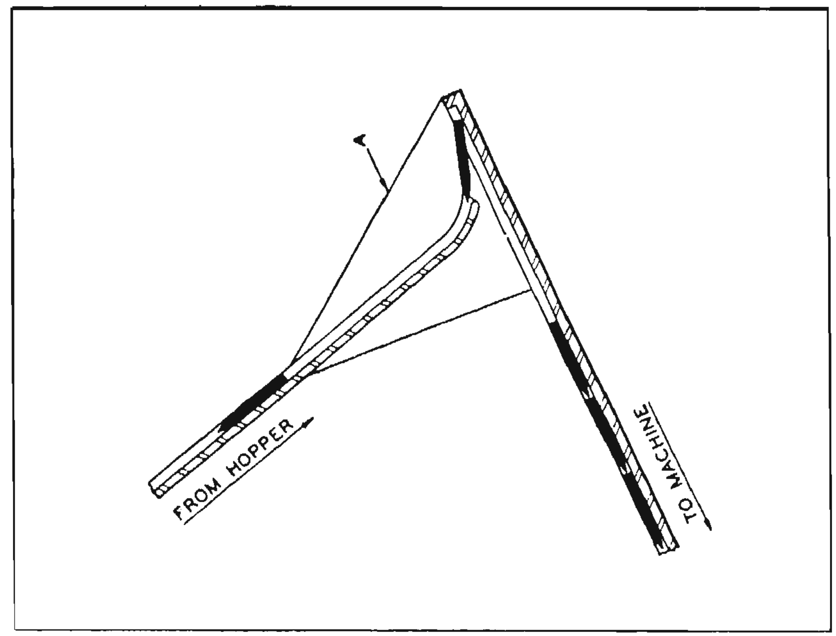


Fig. 19. Track Arrangement for Reversing Position of Parts Received from Hopper.

A prevent the parts from leaving the track. When they must be turned upside down, a simple bend in the track, as shown in Fig. 20, will accomplish this most effectively. It can be seen that in this case the cover *A* becomes the bottom of the track below the bend.

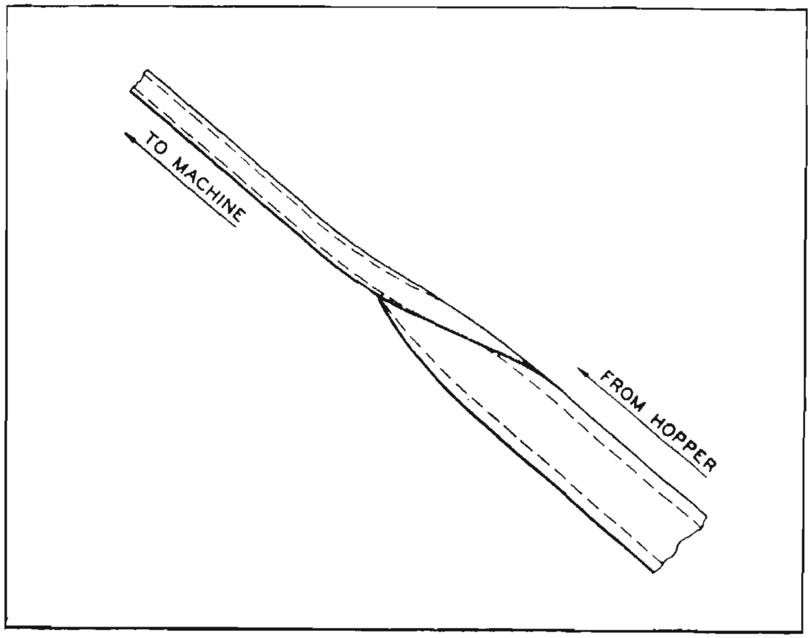


Fig. 21. Track with Quarter Twist for Turning Parts from Vertical to Horizontal Position.

Fig. 21 shows a quarter twist applied to a track. This is useful where the parts must be picked up in a vertical position by the hopper, but must be fed into the machine horizontally. Occasionally, a half twist is applied to the track to turn the parts upside down, where space requirements do not permit the use of a bend such as shown in Fig. 19.

CHAPTER 18

Miscellaneous Mechanisms

The mechanisms described in this chapter are those which were not readily classifiable in the general groups covered by the preceding chapters. They are included because of some interesting features or ingenious design.

Press Ram Mechanism that Gives Nearly Uniform Pressure During Latter Part of Stroke.—The mechanically operated presses generally employed for drawing and stamping work and for molding plastic materials produce their maximum pressures at or near the ends of their strokes. The mechanism shown diagrammatically in Fig. 1 has been designed to produce nearly a constant pressing force against the work for a considerable portion of the latter part of the ram stroke, thus giving somewhat the same characteristics as hydraulically operated presses.

The mechanism has two cranks *A* and *B* of the same length, mounted on shafts connected by two spur gears *C* and *D* of the same size. These gears, rotating in opposite directions, are so meshed that when crank *A* is in position 1, namely at the upper end of the stroke, the angular position of crank *B* is about 125 degrees in front of or ahead of crank *A*. The choice of this angular lead of crank *B* over crank *A* is very important in obtaining the required result, a lead angle of 125 degrees being the one best adapted for most requirements. Cranks *A* and *B* support, on their outer ends, the connecting-rods *E* and *F*, which are of the same length. The opposite ends of rods *E* and *F* are connected to the main connecting-rod *G* which transmits the reciprocating motion to ram *H*.

The connecting point J at which the three different members are joined follows a figure 8 path, as indicated at K. The larger part of the crank rotation—indicated by angle M—is utilized in moving the ram downward, so that the return stroke will be accomplished much more quickly.

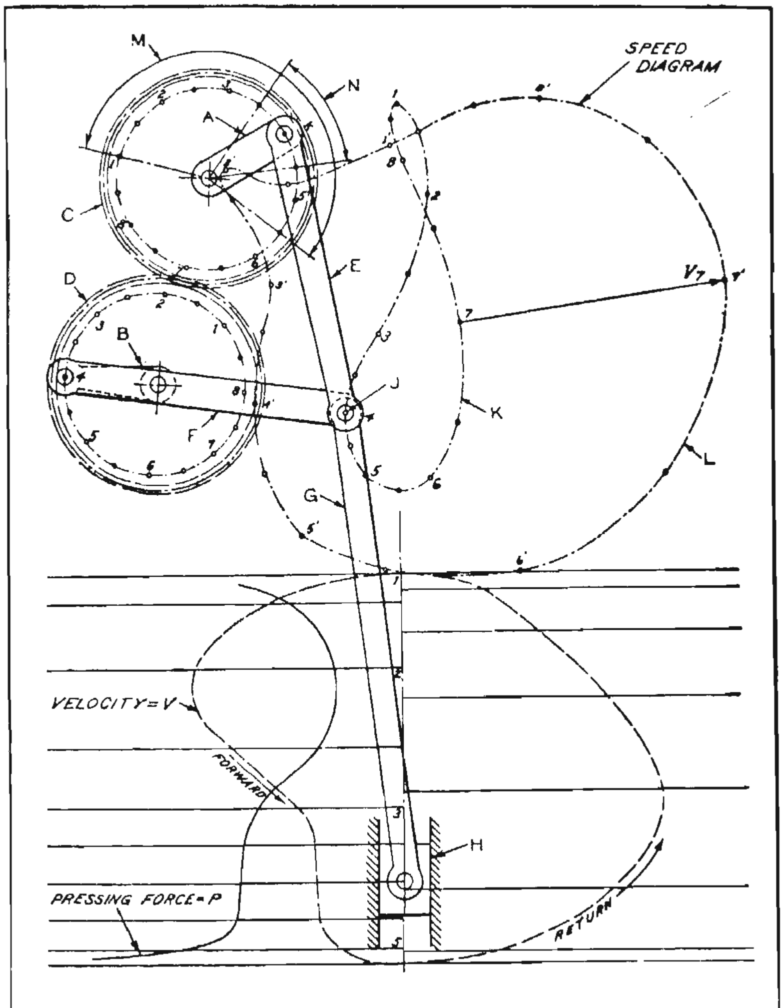


Fig. 1. Mechanism for Driving Press Ram, which Gives Nearly Uniform Velocity and Pressure During Latter Part of Down Stroke.

Besides the line K showing the path followed by connecting point J, there is also a speed diagram L which indicates the variations in its velocity. From this diagram, the speed of the ram was plotted for the down stroke, as represented by curve V. The velocity at any point on the downward stroke is indicated by the horizontal distance from the vertical reference line, which has been drawn through the center of ram H, to the given point on that part of the velocity curve which lies to the *left* of the vertical reference line. The velocity at any point on the return stroke is similarly measured, but to the given point on that part of the velocity curve which lies to the *right* of the vertical reference line as indicated in Fig. 1.

At the beginning of the stroke, the speed of the ram increases, the maximum speed being obtained at about one-third of the stroke. The speed then drops to a point where it is about one-third that of the maximum speed, and remains constant for a relatively large angular movement of the crank, as indicated by angle N. At the end of the stroke, the speed is reduced to zero.

The force diagram P, representing the force exerted by the ram, is the inverse form of the velocity diagram V. Also, in the interval during which the ram speed is practically constant, the force remains nearly constant. On the return stroke, the variation in speed is similar to that obtained with slotted crank drives.

Mechanical Equalizer for Hydraulic Press Ram.—A mechanical equalizer designed for mounting on the ram of a Greenerd hydraulic press is shown in Fig. 2. This device is being used to distribute the pressure exerted by the ram equally between two consolidating fixtures mounted on the press platen. The application of equalized pressure to the two fixtures takes care of any variations in height during the consolidation operation.

The pressure is transmitted to two fixtures (not shown) by the pins A and B, the pressure being equalized by the arm C which is pivotally mounted on pin D. The pins A and B are shown by solid lines in their normal operating positions, with springs E holding the pins in contact with the equalizer arm C.

Assuming that the fixture in contact with pin B offers greater resistance to the pressure exerted by the arm than

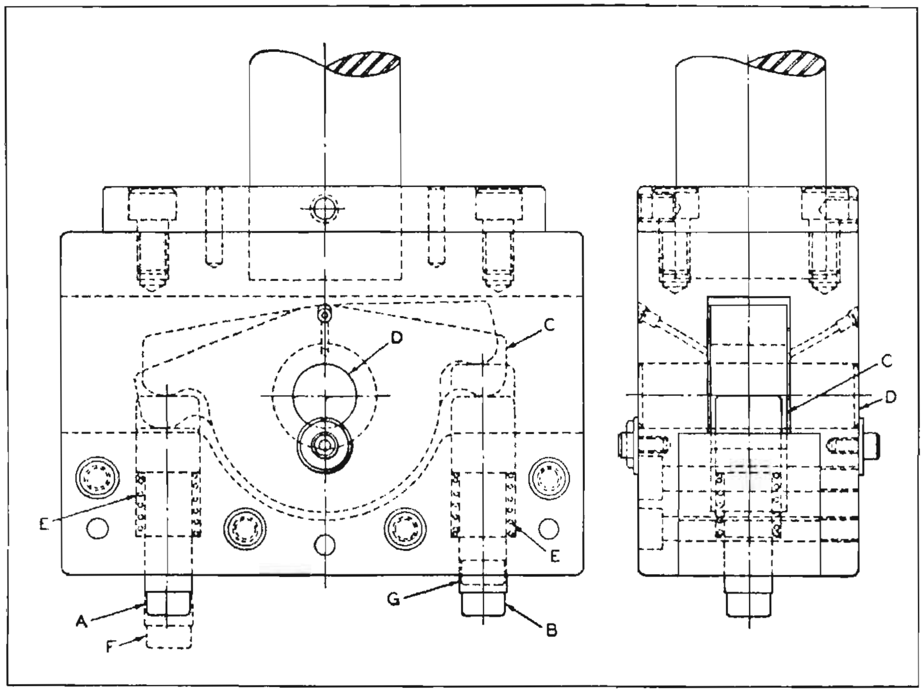


Fig. 2. Device Attached to Ram of Hydraulic Press to Obtain Equalized Application of Pressure by A and B.

does the fixture in contact with pin A, the pin A would advance while pin B receded, so that they would occupy relative positions such as indicated by the dotted lines at F and G.

Certain minor changes were made after the original design (illustrated) was developed, but the principle of operation remains the same.

Spring-Winding Mechanism Operated by Either Right- or Left-Hand Movement of Lever.—A switch for controlling automobile signalling lights contains a clock movement which is started by winding up a clock spring each time the control handle is turned. Turning the handle to the right lights the green signal, indicating a right-hand turn. Turn-

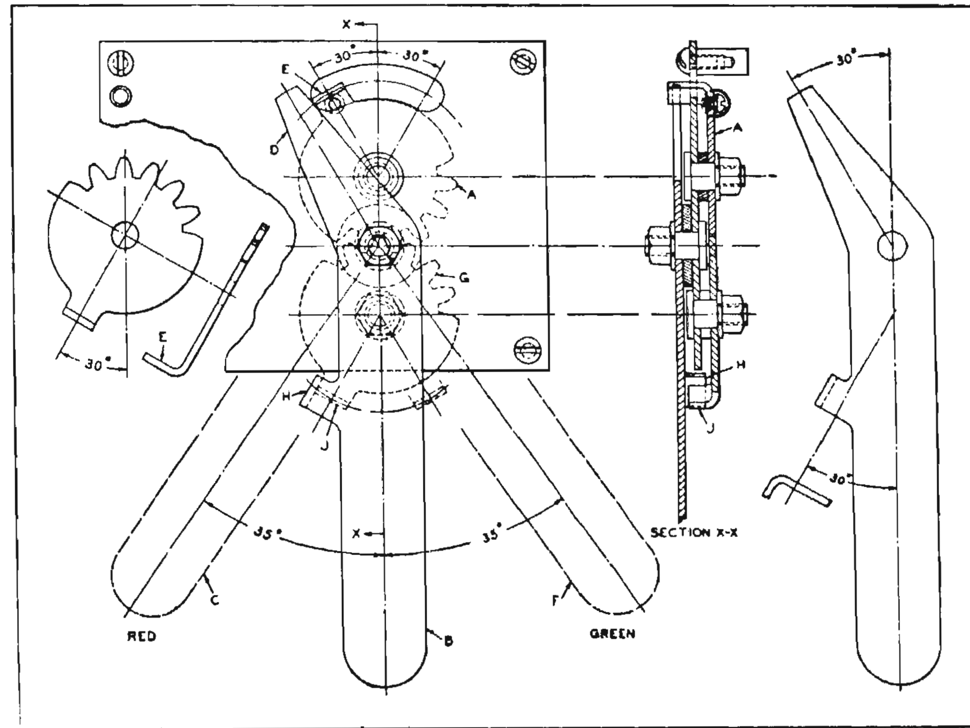


Fig. 3. Mechanism for Rotating Segment Gear in One Direction Regardless of the Direction in which the Operating Lever is Moved.

ing the handle to the left lights the red signal, indicating a left-hand turn. Turning the handle in either direction must always wind the clock spring in a clockwise direction. The problem of finding a movement which would do the winding under these conditions was solved by the mechanism shown in Fig. 3.

This mechanism serves to wind the spring regardless of which way the handle is turned. The leverage and the pressure are the same for movement in either direction. The distance the handle travels is also the same in each case.

Referring to Fig. 3, the spring to be wound is attached to segment gear *A*. Swinging the handle *B* to the position indicated by the dotted lines at *C* turns on the red light and rotates segment gear *A* in a clockwise direction through an angle of 60 degrees. This movement serves to wind the clock spring in a clockwise direction. Rotational movement is imparted to gear *A* by handle *B* through contact of the arm *D* with the projecting member *E*.

Swinging handle *B* to the right, into the position indicated by the dotted lines at *F*, serves to light the green light and also rotate the segment gear *A* through an angle of 60 degrees in a clockwise direction. In this case, however, rotational movement of gear *A* is transmitted from handle *B* through the segment gear *G* by contact of the projecting member *H* on handle *B* with the projecting member *J* on the segment gear *G*. The design of segment gear *A* is shown by the views to the left. Lever *B* is also shown in a separate view to the right.

Pivotal Joints for Special Purposes.—Pivoting joints of three different designs developed to meet special requirements are shown in Fig. 4. These joints are used to maintain proper alignment or free action of certain members. In the design shown in the upper view, the adjusting screw *A* pivots on the stud *B* which is attached to a swivel slide within the machine, a portion of which is indicated at *C*.

The knob at *D* has a curved surface at *E* which fits a mating surface in the lug *F*. As the knob is moved along the threaded portion of the screw, the end of the screw at *B* swings back and forth through a radial arc, the radial seat at *E* permitting the knob *D* to adapt itself to this swinging motion, which is in a horizontal plane at right angles to the center line of screw *B*.

In the central view of Fig. 4 a fork *A* used for shifting a clutch *B* on shaft *C* through the medium of the two pins *D* is shown. This shifting fork pivots on the screw *E*. A spring at *F* holds the shifting lever in contact with one side of the bearing, thus preventing it from vibrating.

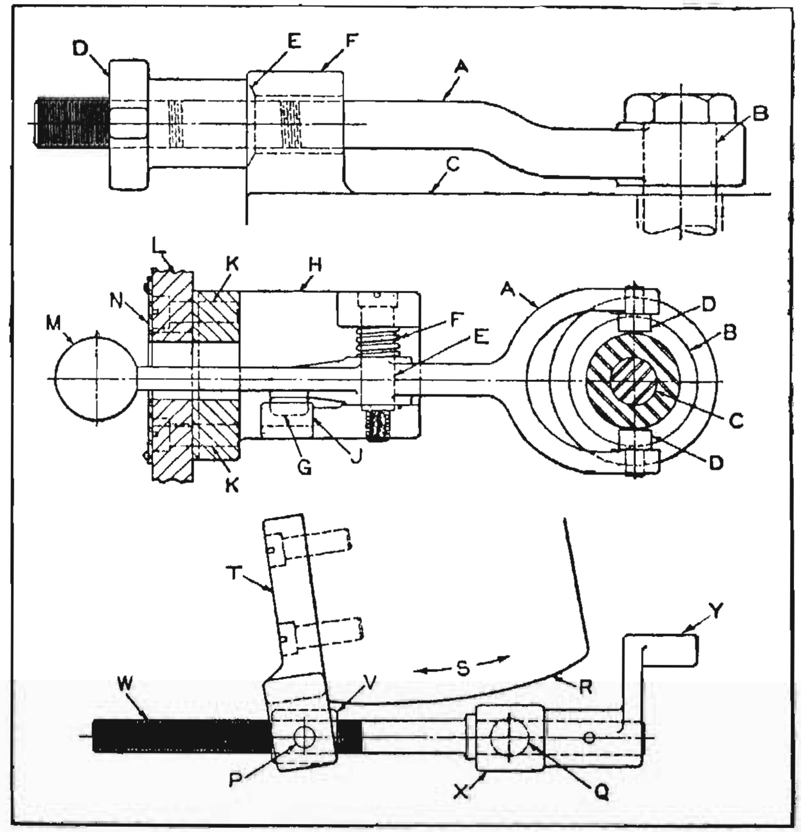


Fig. 4. Three Types of Pivotal Joints Designed for Special Purposes.

There is also a stop-lug at *G* which limits the sidewise movement of the lever by coming in contact with the side of lug *J*. This entire lever construction is attached to a bracket *H*, secured by screws at *K* to the front of the machine, a portion of which is shown at *L*. The shifting lever is operated by a round knob at M. A plate N is fastened over an elongated hole through which the unit is assembled.

Two pivot pins are incorporated in the arrangement shown by the lower view, which is employed for moving a swivel plate R. The pivot points are indicated at P and Q. Plate R is adjustable about a stud (not shown) with a circular oscillating movement indicated by the arrow S. On the side of plate R is mounted a block T. In a slot in the lower end of block T is carried a block V which is pivoted on two short pins P. Block V is tapped to receive screw W.

The pivot point at Q consists of a turned stem which is integral with the bearing X in which the screw W revolves when turned by the hand-lever Y, pinned to the shaft. The stem Q is free to revolve in a fixed base. Plate R, in traveling back and forth as indicated by arrow S, causes screw W to assume various angles, the swivel mountings permitting it to pivot at points P and Q.

Switch-Positioning Mechanism.—The lever arrangement shown in Fig. 5 is used on a spring winding machine which is reversed frequently by reversing the driving motor. The reversals are controlled by a double-throw drum type switch having sliding contacts. The lever mechanism serves to hold the switch in the neutral position when the mechanism is stopped to insure full disengagement of the contacts. With this arrangement, complete disengagement of the contacts is effected automatically after the operating lever has been moved to a given position.

The lever B is keyed to the shaft A, which operates the switch drum. The upper end of lever B is connected to the shifter rod G. Pin C at the lower end of rod B fits in the fork at the upper end of the lever D, which is free to oscillate on stud E. The lower end of spring F is fastened to a stationary part of the machine, while the upper end is fastened to the lower end of lever D.

The positions of the levers when the switch is in the neutral position are shown in the left-hand view of Fig. 5.

The tension of spring F, acting through lever D, serves to hold lever B upright. If the shifter rod is given slightly less than its full travel movement, the spring F will immediately return it to the neutral position.

In the right-hand view of Fig. 5, lever B is shown just before reaching its extreme left-hand position. At this

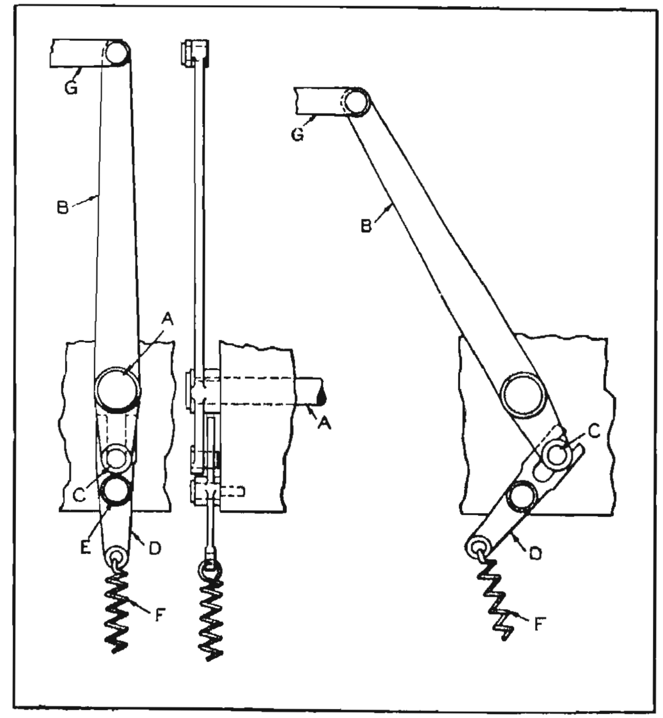


Fig. 5. (Left) Switch-operating Mechanism in Neutral Position. (Right) Switch Mechanism about to Make Contact.

point, the switch drum has not quite completed its partial revolution, and spring F, acting on pin C through lever D, still exerts sufficient pressure to return lever B to the neutral position if the movement of shifter G is not completed. However, as the movement of shifter G continues to the left, and the levers B and D more closely approach positions at right angles to each other, the pressure exerted by the fork at the end of lever *D* acts longitudinally on lever *B* and is no longer effective in returning it to the upright position. The switch contacts then remain in engagement until lever *B* is again returned to the position shown in the right-hand view of Fig. 5.

Follower Mechanism for Contour Milling of Grooves.— The usual method of machining a straight slot, the bottom of which changes from a parallel to a tapered surface at some point along its length is to make two passes with a milling cutter, one for each of the intersecting planes. With the proper type of contour follower, however, both surfaces can be machined in one pass, and production increased.

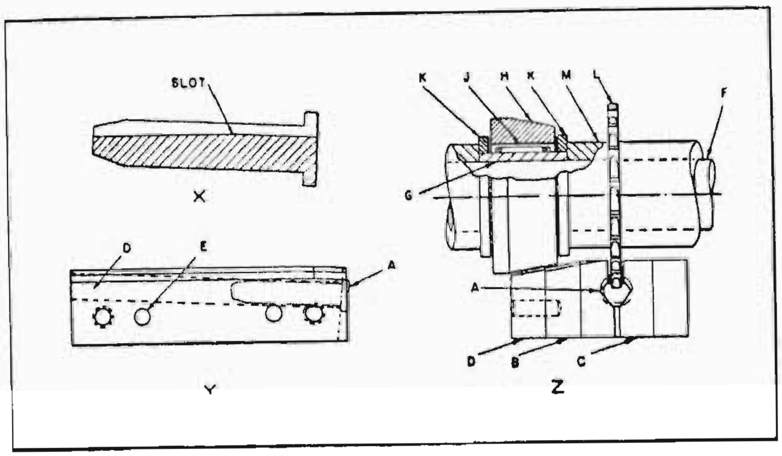


Fig. 6. Follower Mechanism Used in Milling a Contoured Slot in a Cylindrical Part as Shown at X.

The device shown at *Y* and *Z* in Fig. 6 was designed to form such a slot in a cylindrical part, as shown at *X*. The work *A* is located in V-blocks and clamped between a fixed jaw *B* and an adjustable jaw *C*, which are fitted to a 4-inch quick-acting vise. Templet *D*, which has the contour to be produced on the work machined on its surface, is fastened to the fixed jaw by dowel-pins *E*.

The follower roller *H* is attached to the milling machine arbor *F* in such a position that it rolls on the templet when the milling machine table is raised to maintain contact between the roller and templet during the feeding movement. Bushing *G* is made a slip fit over the arbor, and roller *H* is pressed on a needle bearing *J*, which revolves freely between two collars *K*. Both the templet and the roller are machined at an angle, as shown in view *Z*, so that the distance of the roller from the milling cutter *L* can be adjusted by adding or removing spacers *M*, thus varying the height of the cutter above the work and hence the depth of the slot.

In operation, the vise is mounted on the table of a hand milling machine. Attached to the end of the handle controlling the vertical movement of the cutter is a weight which maintains pressure on roller *H*, so that it is kept in contact with the templet. When the longitudinal feed of the machine is engaged, the roller follows the contour of the templet, causing the cutter to mill the slot to the same contour.

Lever and Spring Arrangement for Variably Increasing Tension on Slide.— In developing a wire-forming machine, it was necessary to provide means for gradually increasing the spring tension opposing the movement of a certain slide up to a given point in the cycle and then suddenly increasing the tension. The mechanism designed to accomplish the required variation in tension is shown in Fig. 7, the normal position of the mechanism being indicated in the upper view.

The slide *A*, fitted wi h roller *B* and cross-bar *C*, travels in a stationary part of the machine. Two studs *D* in the stationary part act as fulcrums for the levers *E*. Levers *E* carry the rods *F*, which pass through clearance holes in the cross-bar *C*. Springs *H* on rods *F* react downward against cross-bar *C* and upward against the rods *F*, thus holding levers *E* against the pins *G*, and the slide *A* against the pin *J*. In this position, springs *H* are only lightly compressed.

As slide *A* is raised, springs *H* are compressed by the movement of cross-bar *C*, the levers *E* being immovable, due to their contact with pins *G*. The tension on slide *A* is thus increased gradually until roller *B* comes in contact with the inner ends of levers *E*, causing the outer ends to move

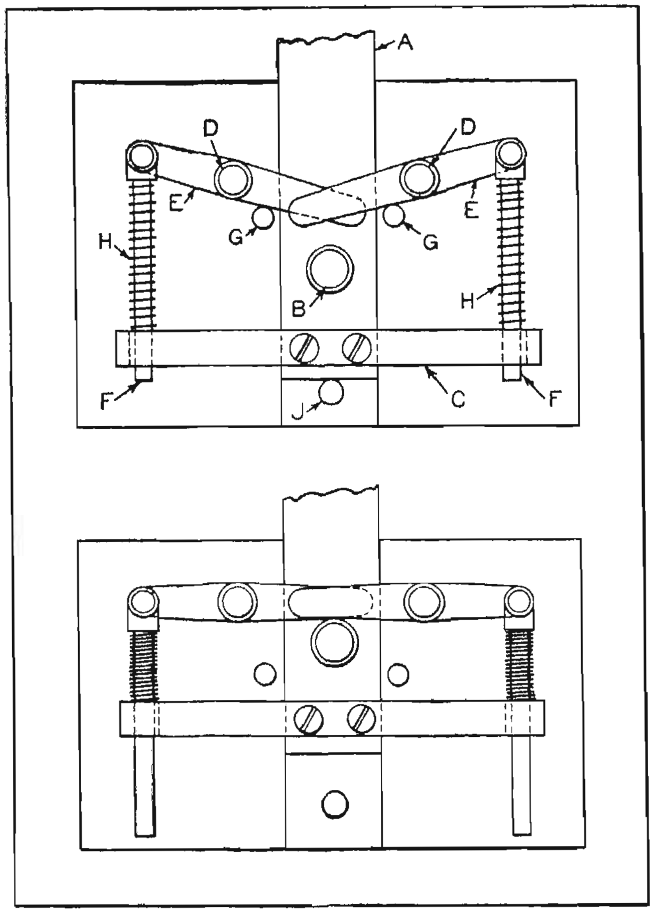


Fig. 7. Spring and Lever Arrangement for Increasing the Tension Opposing Upward Movement of Slide A.

in the reverse direction. This results in suddenly increasing the speed at which the springs are being compressed, which, in turn, causes a rapid increase in tension on the slide.

Rotating Mechanism for Creasing Flexible Material.—The mechanism shown in Fig. 8 is designed to actuate the jaws *A* and *B* for creasing a certain flexible material. The jaws and their actuating mechanism comprise only the creasing unit of a complete machine. In operation, jaw *A* simply pivots or swings on pin *C* from the position shown in the view at the left to that shown at the right, so that the distance indicated at *D* is reduced to that indicated at *E*. While this pivoting movement is taking place, jaw *B* is pivoted inward on pin *F* a similar amount, and, in addition, is rotated approximately one-half revolution. As a result of these two movements, jaw *B* is brought into position beside jaw *A*, as shown in the view to the right, in which the outlines of jaws *A* and *B* coincide.

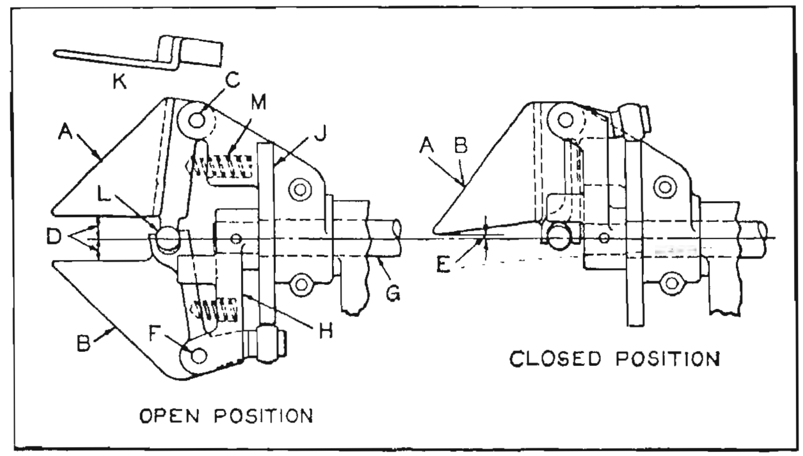


Fig. 8. Mechanism for Creasing Flexible Material.

The jaws are actuated as described by the shaft G, pinned to arm *H*. As shaft *G* rotates, the follower roll on an arm projecting from jaw *B* rides up on the lobe of the face cam *J*, causing the jaw to pivot or swing inward. A fork on jaw *B*, which fits over a ball *L* machined on jaw *A*, transmits the required pivoting motion to the latter jaw. Helical springs M maintain a constant opening pressure on the jaws. The jaws are offset, as indicated by view K, to permit them to be closed, as indicated in the view to the right.

Mechanism Designed to Crimp Ends of Heavy Paper Cylinder.—The lever and toggle action mechanism shown in Fig. 10 is designed to crimp the upper end of a heavy paper cylinder B, Fig. 9, for a length C preparatory to folding and flattening it over the end of an inside core or spool of twine A. After the paper has been folded over the twine, it is secured in place by cementing a circular label over the end.

To enable the end of the paper cylinder to be satisfactorily folded and flattened, the mechanism was designed to produce twelve crimps, equally spaced about its circumference, as shown in the plan view of Fig. 9. This required twelve individual toggle units, arranged in a circle around a central operating unit. Two of the toggle units are shown in the closed position in Fig. 10.

The spool X, Fig. 10, is lightly clamped in position, with the paper cylinder extending a distance Z beyond its upper end. The clamping means employed consists of a V-clamp A which slides on two tightening rods B, only one of which is shown. The tightening rods, with the spool located between them, pass through the frame of the mechanism and are fitted with clamping nuts. While the work is being clamped in place, the lever-shaped jaws C and D move upward, clearing the spool and paper.

The entire unit then moves down to the lower position, but with the jaws C and D held open. These jaws are next closed around the end of the paper cylinder, crimping it in twelve places, as required. After this has been accomplished, it is a simple matter to fold over the end of the paper and flatten it down into place to receive the label

Referring to the construction of the mechanism, housing E is a drum-shaped part to which is attached a series of brackets F. In the illustration, the drum is cut through the center to show two of the twelve units C and D. Each of

these units is pivoted on its own stud Q in the drum. The twelve angular extended ends G of levers D enter a groove in yoke H that is free to slide over the central shaft J. The central shaft has a pad K attached to the lower end which holds the spool down and acts as a control center.

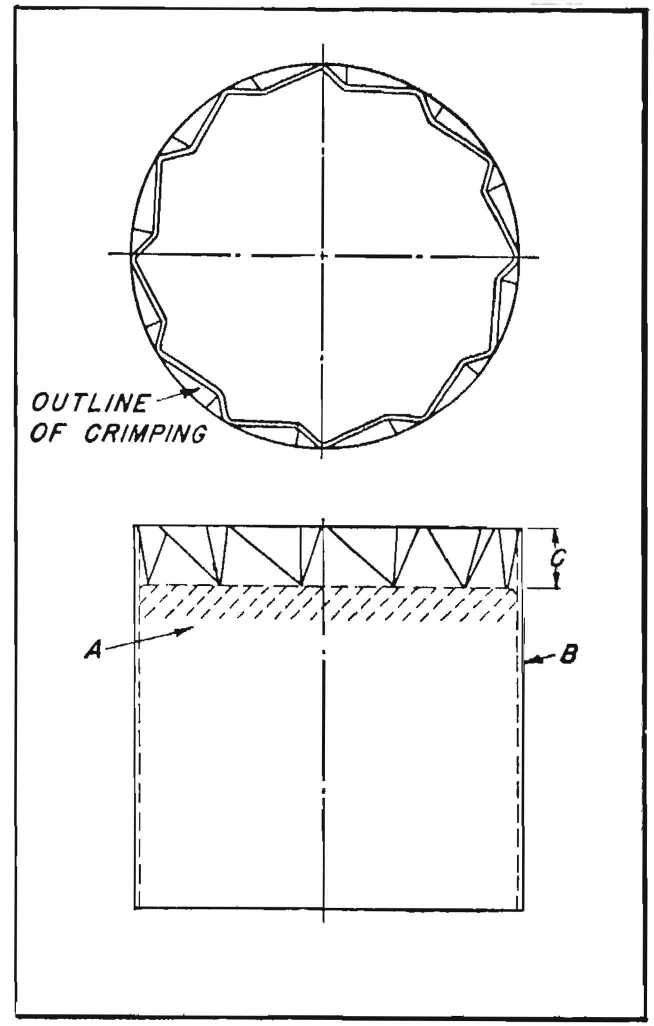


Fig. 9. Twelve Equally Spaced Crimps are Produced in Heavy Paper Cylinder by Mechanism Shown in Fig. 10.

The lever-shaped jaws C have straight extended ends L to which are attached, by means of connecting links M, a series of twelve right-angle levers N, all of which engage the groove in yoke P. Yoke P operates jaws C, bringing them into the open or closed position, while yoke H operates jaws D; thus the simultaneous action of the two series of

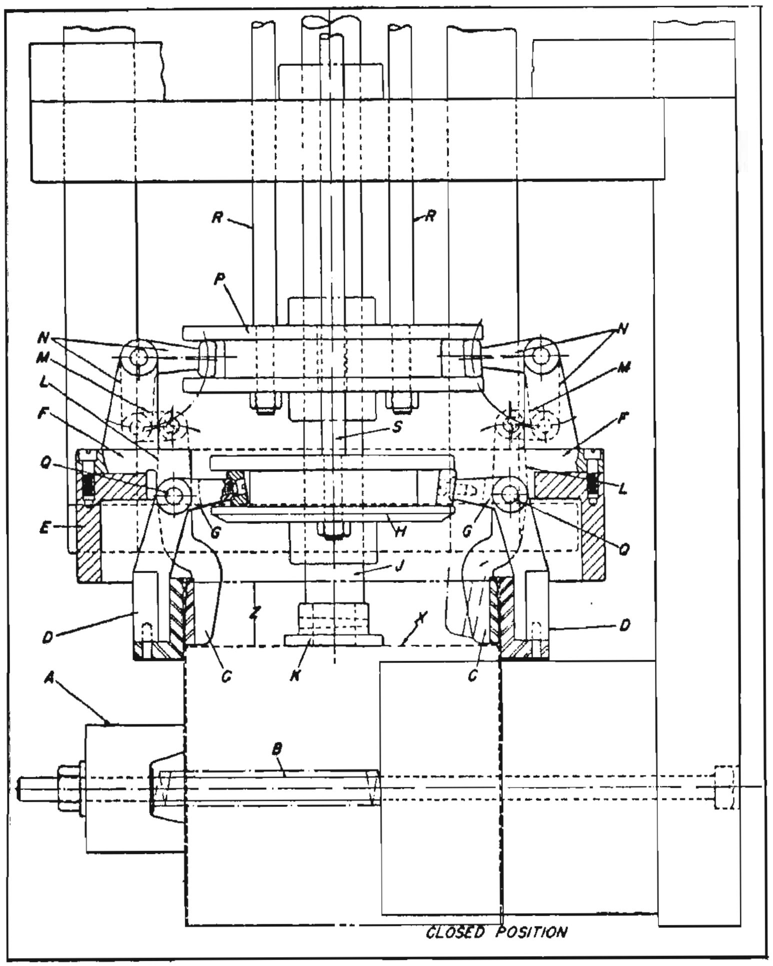


Fig. 10. Mechanism Designed to Crimp Ends of Heavy Paper Cylinders as Shown in Fig. 9.

jaws serves to crimp the circumference of the heavy paper. Two rods R attached to an operating mechanism within the machine serve to slide the yoke P up and down on the central shaft J. In a similar manner, two rods S operate the yoke H.

With the spool and paper cylinder clamped in place, the central shaft J moves down until pad K touches the spool. As this is done, the entire unit carried on drum E moves down into position. Jaws C are then closed by an upward movement of yoke P, after which jaws D are closed as yoke H slides upward along the shaft from the open to the closed position.

Selective Timing Mechanism for Actuating a Control Lever.—The timing mechanism shown in Figs. 11 and 12 has been used successfully on one of the textile machines manufactured by the James Hunter Machine Co., North Adams, Mass. As shown in the illustrations, it differs from the conventional type of timing devices. It covers a wide range of timing requirements and can be set for such operating intervals or periods as 1, 2, 3, 4, and 5 minutes, or 8, 16, 24, 32, and 40 minutes. An important feature of this mechanism is its extreme simplicity, the selection of the various intervals or time periods being instantly accomplished by merely turning a knob. This eliminates the necessity for locating or relocating various fingers or cam lobes about a disk, as in conventional timing devices.

The specific purpose of the mechanism illustrated is to impart one forward movement to lever V, Fig. 11, for a predetermined number of revolutions of shaft B. This forward movement of lever V can be used to release a clutch, make an electrical contact, or perform any duty necessary for starting other mechanisms or machines at the selected time intervals.

Crank A on the drive-shaft B transfers a reciprocating movement to the bellcrank C through the connecting-rod D. Bellcrank C is free to turn on stud E, and through its pawl

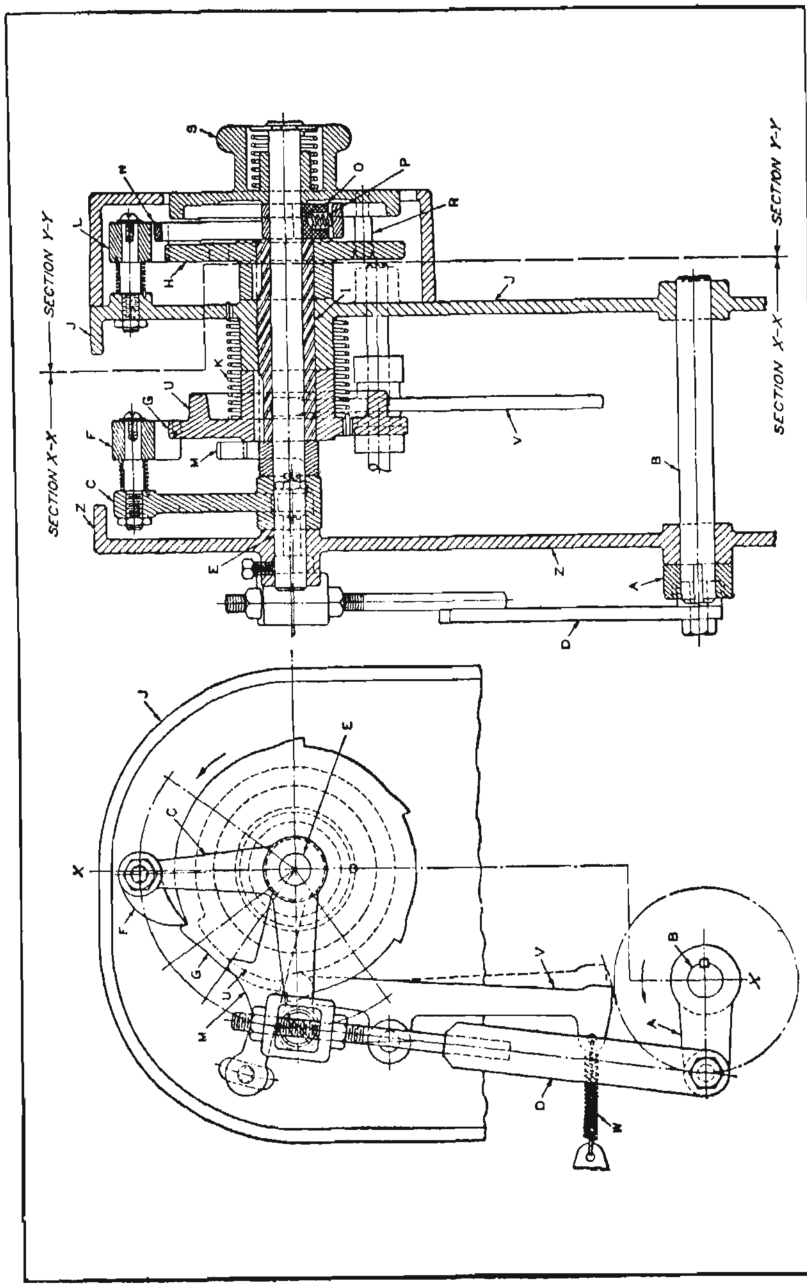


Fig. 11. By Setting Control Knob S to Any Number from 1 to 5, Lever V will be Moved Forward after Any Number of Revolutions of Shaft B from One to Five.

F, rotates the ratchet wheel *G*. The ratchet wheel *H* rotates in unison with ratchet wheel *G*, as both members are keyed to sleeve *I*, which is free to turn on stud *E* and extends through the stationary frame *J*. A helical torsion spring *K*, mounted between ratchet *G* and frame *J*, tends to rotate the two ratchets in an opposite direction to that imparted by bellcrank *C* and pawl *F*. Ratchets *G* and *H* are divided into six equal parts, five of which have teeth that are engaged by pawls *F* and *L*, respectively. Referring to Fig. 12, these teeth are marked No. 1_1, No. 2_1, No. 3_1, No. 4_1, and No. 5_1.

Referring again to Fig. 11, pawl *F* is released from ratchet *G* at the end of its stroke by cam *M*, but the ratchets are normally prevented from rotating under the action of spring *K* by pawl *L* mounted on frame *J*. Lever *N*, the function of which is to lift pawl *L* from ratchet *H*, is prevented from rotating too freely on stud *E* by the friction block *O* and spring *P*, and is carried forward under pawl *L* by the cam lobe *Q*, shown in Fig. 12, on ratchet *H*, and backward by pin *R* in control knob *S*. Pin *R* can be placed in any one of the holes in ratchet *H* numbered from 1 to 5. The pointer *T* on ratchet *H*, shown in Fig. 12, and numbers 1, 2, 3, 4, and 5 on the control knob *S* are used in making the required setting. The cam *U* is extended on the side of ratchet *G* and contacts with the lever *V*, held against it by the spring *W*, as shown in Fig. 11.

Shaft *B* rotates constantly when the mechanism is in operation, and through crank *A*, connecting-rod *D*, bellcrank *C*, and pawl *F* advances ratchets *G* and *H* one tooth for each revolution. Cam *U* is an integral part of ratchet *G* and acts upon lever *V* only when ratchets *G* and *H* are in the position shown. The position of ratchets *G* and *H* at the start of the timing cycle determines the number of revolutions of shaft *B* for each forward movement of lever *V*.

This is accomplished as follows: Assume that pin *R* is placed in hole No. 3, as indicated in Fig. 12. While cam *U* is acting upon lever *V*, the lobe *Q* on ratchet *H* carries lever

N under pawl L, lifting it clear of ratchet H. When pawl F is released from ratchet G by cam M, ratchets G and H turn backward through the action of spring K, until pin R comes in contact with lever N and carries it backward in time to allow pawl L to drop and engage tooth No. 3_1. Thus, during the third revolution of shaft B, the ratchets and cam U again come into position to move lever V forward.

Should pin R be placed in hole No. 4, lever N would be carried back from under pawl L by pin R one tooth later

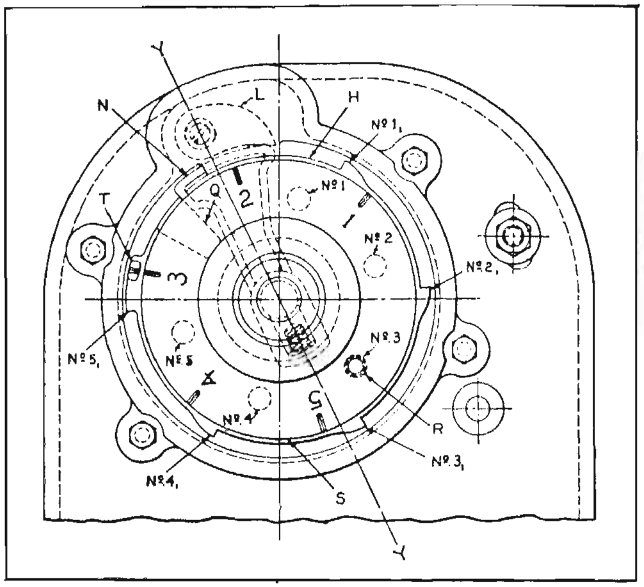


Fig. 12. Dial Control End of Mechanism Shown in Fig. 11.

when the ratchets were returned by spring K, and pawl L would engage tooth No. 4_1. Lever V would then be moved forward during the fourth revolution of shaft B. Should pawl L fail to engage any of the teeth, spring K would be prevented from being unwound by a safety stop, which is just in front of the end of pawl L on the vertical center line, Fig. 12. This stop prevents the pointer T from making a complete revolution.

Pin R is so arranged that it cannot be removed from in front of lever N. Therefore, if hole No. 2 is selected, lever N will be carried backward during the setting, and after imparting the forward movement to lever V, the return of the ratchets will be stopped on tooth No. 2_1. The forward movement of lever V will then take place during the second revolution of shaft B.

This mechanism can be designed for a different number of timing periods by dividing a cycle of the ratchets into one more division than the number of timing periods desired and proportioning the stroke of the connecting-rod accordingly.

Amplifying Mechanism for Precision Measuring Instrument.—A movement-amplifying mechanism developed to transmit movement from the contact or measuring point to the indicating pointer of precision measuring instruments has, as its most important part, a metal strip of rectangular cross-section which is twisted into a helix, as shown at A and B, Fig. 13. This twisted part, of unusual design, is employed in instruments for taking precision measurements of length, weight, pressure, electrical energy, etc., which require an amplifying unit that will operate with a minimum of frictional and energy loss and without back pressure.

The mechanism described and illustrated is protected by patents of Aktiebolaget C. E. Johansson, of Eskilstuna, Sweden. It has been employed in extensometers, electrocardiographs, micro-monometers, variometers, and surface finish testing instruments.

The metal strip A, Fig. 13, is twisted into the required helical form by fastening each end rigidly and winding from the center. The winding operation is continued until the metal has been formed sufficiently to retain the helical shape permanently. When the strip twisted in this manner is held at each end and stretched, the center of the strip will rotate about an axis which is the center of the cross-section of the strip. Actually, one end of the twisted strip is held in a fixed position, while the other end is attached to a lever or crank connected with the measuring point of the instrument, as shown in Fig. 14. The indicating pointer P is secured to the center of the twisted strip. With this arrangement, the indicating pointer will be moved over a graduated scale when the measuring point at the lower end of member A is moved.

In the case of the twisted solid strip A, Fig. 13, the metal in the center of the section is compressed in winding, and

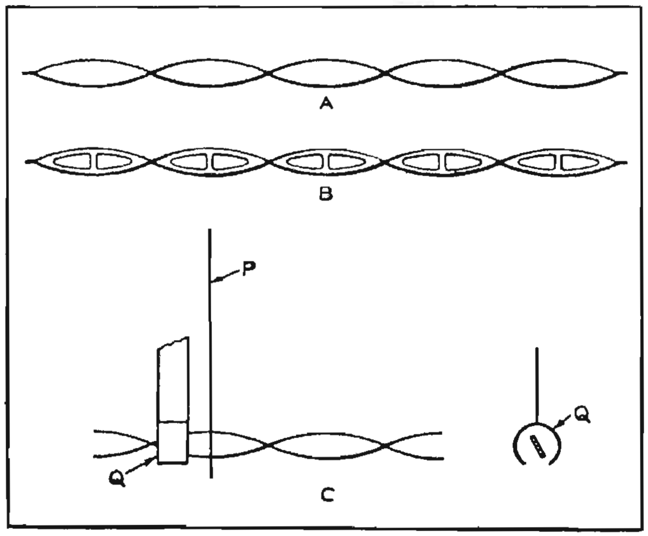


Fig. 13. Two Types of Metal Strip, A and B, Indicating Pointer P, and Split Tube Q Used in Amplifying Mechanism Shown in Fig. 14.

elongated when the strip is stretched. To correct this condition, a series of perforations may be cut out of the central portion of the strip, giving it the form at B. Such a strip requires less energy to operate and also gives a greater rotative movement with a given tension on the strip than the one shown at A. The relation between the cross-section of the strip, elongation, pitch of winding, and the stretching force required to produce rotation has been determined by

trying different combinations of cross-section dimensions, pitch of twist, and size and number of perforations.

Within a certain range, the rotation of a strip about its center is practically directly proportional to the elongation. On one type of experimental strip, this portion of the curve covers a range of about 60 degrees. The rotation of the strip within this range is approximately 18 degrees for an elongation of 0.00039 inch. Tests show that a force of one gram produces a rotation of 5 to 7 degrees.

Another strip which requires a much lower operating force and produces a much higher amplification gives such a high rotative or amplifying effect that it does not need to be perforated if used within a range of 145 degrees rotation. By "operating force" is meant the force required to hold the pointer in the starting or zero position. The latter strip is 0.0042 by 0.0002 inch in cross-section, 1.5748 inches long, and has a twist of 2160 degrees.

By varying the dimensions of the cross-section, length, and pitch of the twist in the strip, it is possible to produce many different amplification ratios. The strips mentioned are only examples, and do not show the full possibilities of their use in amplifying mechanisms. The twisted strips, when properly mounted in an instrument, are surprisingly strong. The elongating force or tension required to produce rotation of the strip about its axis can be reduced to a minute fraction of the amount normally required by balancing the normal or initial tension with a permanent magnet.

The "Mikrokator" amplifying and indicating mechanism shown in Fig. 14 is fitted with a strip B like the one shown at B, Fig. 13. Spindle A, Fig. 14, which carries the measuring point at its lower end, is forced downward against stop C by a coil spring. To provide a frictionless support for the spindle at the lower end, it is fastened to a metal diaphragm D. This diaphragm is cut out, as shown by the plan view E, so as to provide maximum flexibility and not interfere with the free movement of the spindle.

The upper end of the spindle is fastened directly to horizontal spring F and the horizontal member of spring "knee" G. One end of the twisted strip is fastened directly to the vertical member of spring knee G. The other end is fastened

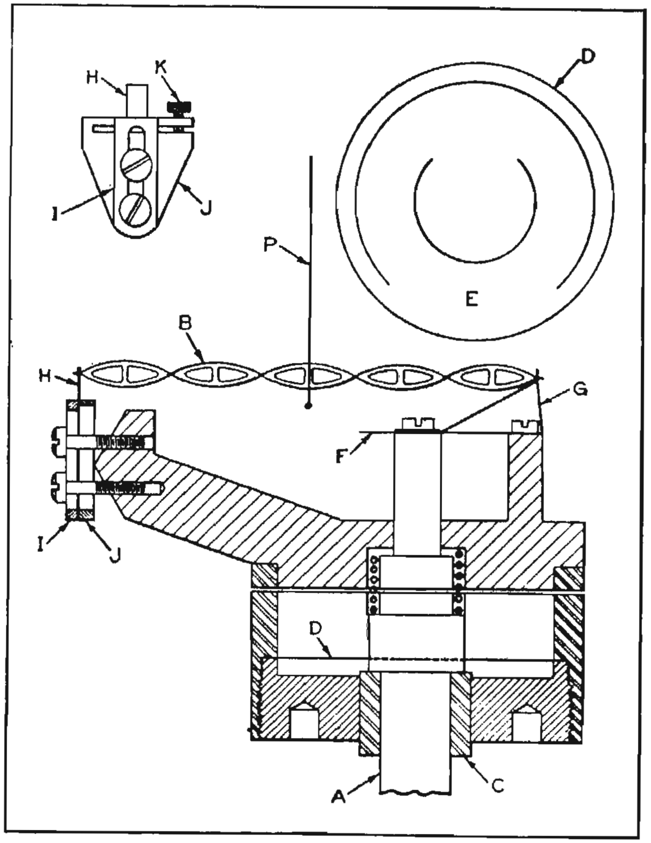


Fig. 14. Diagrams Showing Construction of Amplifying Mechanism of Precision Gage.

to the adjustable spring support H. An upward movement of the spindle will cause the vertical member of spring knee G to move to the right. This movement of the spring knee results in an elongation of the twisted strip, and causes pointer P, fastened to the center of twisted strip B, to rotate

across the scale of the instrument. Varying the height of the vertical member of the spring knee changes the ratio of amplification between the spindle and the pointer.

The adjustable spring support H is used to adjust the pointer position and movement to suit the scale. For economical production, it is more practical to produce identical scales than an individual scale for each instrument. To permit the use of identical scales, adjustable spring support H is provided for adjusting each mechanism until pointer movement corresponds exactly to scale graduations.

This adjustment is accomplished by an upward or downward movement of the plates I and J. These plates have elongated holes through which are passed the screws that clamp the plates to the frame. If the plates are moved up, the portion of spring support H that projects above the plates is reduced, and thus the spring support is stiffened; this causes greater elongation of the strip and greater movement of the pointer for a given movement of the spindle. If the plates are moved down, the portion of spring support H that projects above the plates is increased, and thus the spring support becomes more flexible, and, as it bends more easily, results in less elongation of the strip and a smaller movement of the pointer for a given movement of the spindle.

After the pointer has been adjusted so that it corresponds approximately to the scale graduations, the final adjustment is made by means of screw K. Adjusting plate J is slotted at the top, as shown, so that by turning screw K in a right-hand direction, the tongue at the top of the plate is moved upward, thus shortening the distance the spring projects above the supporting plate. By turning the screw in a left-hand direction, the tongue at the top of the plate is lowered, thus increasing the amount the spring projects.

This adjustment of the tongue has the same effect on the elongation of the strip and pointer movement as the adjustment of the plates I and J; but as the adjustment is made by means of a screw, minute adjustments of the pointer movement are possible. These movements can be adjusted until the actual spindle movement, as checked with gage-blocks, is made to correspond exactly with the indicated movement of the pointer on the scale. The adjusting plates can be changed to increase or decrease the initial tension.

As the only damping effect on the pointer movement is furnished by the resistance of the air, it is very important that the weight and the inertia be reduced to a minimum. The pointer, which is mounted in the center of the strip, is made of tapered glass tubing. The tubing at the large end is approximately 0.0024 inch in diameter, and, at the small or outer end, 0.0012 inch in diameter. As a pointer of such small diameter would be very difficult to see, it is provided with a small circular disk just below the tip. This glass tube is so flexible that it can be bent as easily as a hair without danger of breakage, and will return to its original shape after bending.

If the pointer is allowed to swing freely from the extreme right or plus 0.003 inch reading back to zero, it takes about three-fourths second for it to move this distance and come to an absolute stop. On some production measuring applications, where a damping interval of three-fourths second is too great, a quicker damping effect is obtained by having the strip rotate in a drop of oil. The oil is held in a short length of split tubing which encircles the strip close to the point at which the pointer is fastened to the strip, as shown at *Q*, view *C*, Fig. 13. This makes it possible to obtain almost instant damping of the pointer movement.

The highest amplification on a standard instrument of this type is 3000 to 1. On this instrument, a movement of 0.0001 inch of the measuring tip causes a movement of 0.300 inch of the pointer. On the corresponding instrument, with a scale graduated in metric units, a movement of 0.001 millimeter of the measuring tip causes the pointer to move 3 millimeters.

The highest amplification provided on a special instrument had a ratio of 27,600 to 1. In this case, a movement of 0.001 millimeter of the measuring tip produces a pointer movement of 27.6 millimeters. The scale has graduations for each 0.00002 millimeter. The width of these graduations is 0.55 millimeter. On a corresponding instrument graduated in the English system a movement of 0.0001 inch of the measuring tip produces a pointer movement of 2.76 inches. Scale graduations are 0.028 inch wide for each 0.000001 inch.

Mechanism for Obtaining Uniform Adjustment of Guide Rollers.— A number of strands of wire are fed into a machine for producing a woven wire product by passing the wires over grooved rollers. The rollers are spaced to meet certain specifications. At times, it is necessary to change the positions of the rollers in order to increase the "spread" of the wires. The adjustment must be accomplished while the machine is in operation, and the spacing between the wires must be uniform throughout the total spread. The two views in Fig. 15 show the design of a guide mechanism that fulfills these requirements. When applied to the machine, the mechanism is located in a vertical position instead of in the horizontal position shown in Fig. 15.

A stationary part *A* of the machine has a dovetail groove in it to receive a series of blocks *B*, several of which are shown. These blocks are free to slide in part *A*, with the exception of the block at the extreme right-hand end, which is pinned in position. Each of the blocks *B* has a roller *C* which supports a strand of wire, and all the blocks carry an externally threaded bushing *D*, the head flange of which fits into a recess in the adjacent block *B*. Shaft *E* passes through the assembly of blocks and bushings, and is splined, so that any rotative motion given it through the crank-handle *F* is transmitted to bushings *D* through a key in the bore of each bushing. The screws that fasten the keys in the bores of bushings *D* are shown in the head flanges of the bushings.

The lower view in Fig. 15 shows the mechanism with the rollers set in a position of minimum spread. It will be noted that each block *B* is in contact with the succeeding block. As shaft *E* is rotated by handle *F*, the housings *D* are rotated with it, causing each block *B* to be moved away from its adjacent block by an amount equal to the lead of the thread on bushing *D* multiplied by the number of rotations given shaft *E*.

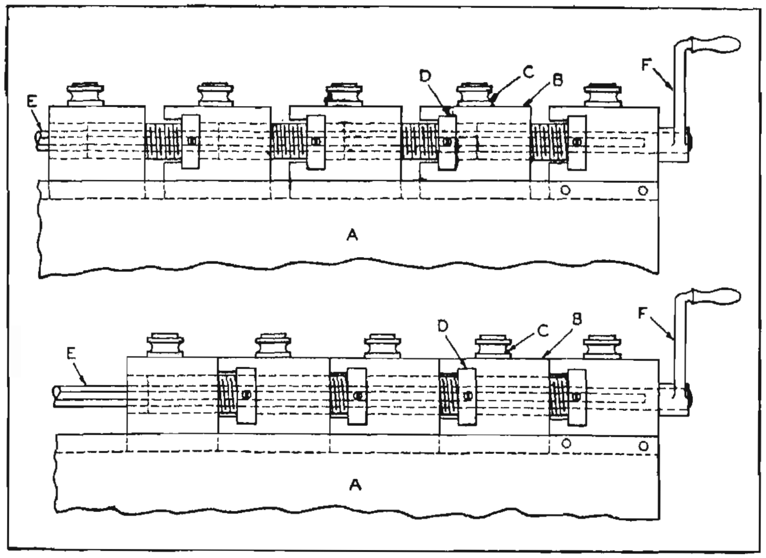


Fig. 15. Mechanism for Obtaining Uniform Adjustment of Spacing between Rollers C.

The upper view shows the mechanism as it appears after handle *F* has been given two turns. As each of the bushings *D* is rotated, it withdraws from the block into which it is threaded, causing that block and all those to the left of it to be moved the same amount. The increase in the spread between the two outside rollers represents the accumulative effect of the axial movement of bushings *D;* and as each of these bushings is rotated the same amount, the increase in the center distance between any two rollers is the same.

Link Mechanism for Operating Combination Furnace Door and Work Plate.—A door for a modern high-temperature furnace must be so designed that it can be opened quickly; it must be a tight fit in the closed position and take up a minimum amount of space. A door designed to meet these requirements is used on muffle type furnaces operating at temperatures up to 1000 degrees C. (1832 degrees F.). The door of the furnace is suspended on a multiple-

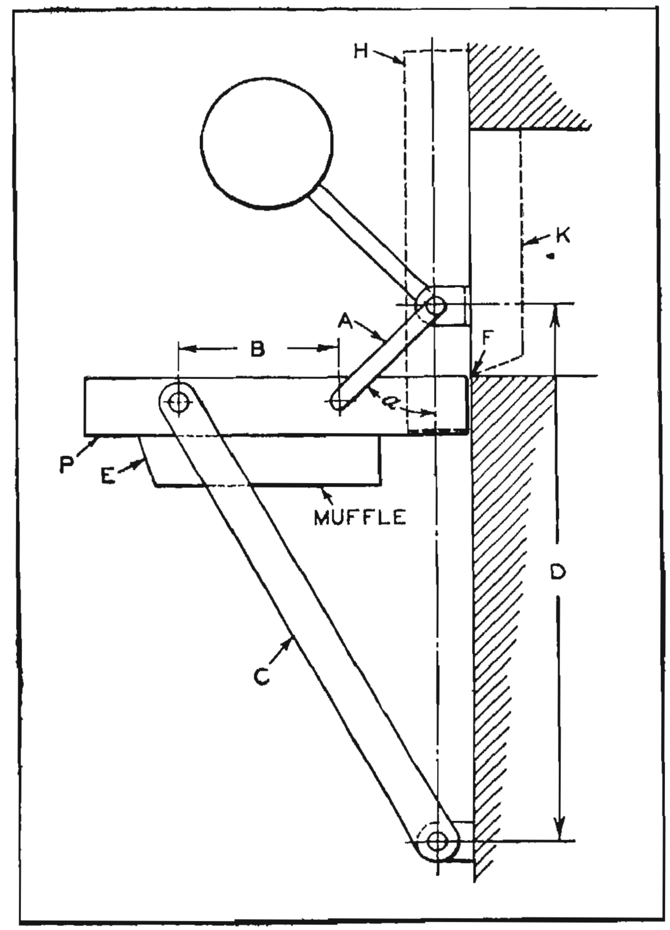


Fig. 16. Link Mechanism for Operating Furnace Door.

link arrangement, so that when it is opened it rotates through an angle of 90 degrees, which brings its upper side into a horizontal position for supporting the parts requiring heat-treatment.

The door plate shown at P in Fig. 16 is actuated by a link A and is guided by a longer link or arm C. When the door is closed by rotating link A in a clockwise direction, plate P will be in the vertical position indicated by dotted lines at H, with the muffle in the position shown by the dotted lines at K. After rotating arm A about 135 degrees counterclockwise, the door reaches the open horizontal position shown by the full lines. On the first part of the closing movement, the end of the door that is nearest the furnace rises without withdrawing from the furnace far enough to leave an opening through which small parts might fall.

The front edge of the muffle must be inclined as shown at E in order to permit it to clear the edge of the door opening at F. The door is counterbalanced by a weight attached to arm A, which serves also as a crank for operating the door. The length or dimension B between the pivoting points on plate P and the length of link C can be calculated if the length of link A and dimension D, as well as the angle a are given, using the formulas:

$$B = \frac{AD\ (1 + \cos a)}{A\ (1 + \sin a) + D}$$

and

$$C = A + D - B$$

Example—$A = 3$ inches; $D = 12$ inches; $a = 45$ degrees; $\sin a = 0.707$; and $\cos a = 0.707$.

Substituting the numerical values in the preceding formulas, we have,

$$B = \frac{3 \times 12\ (1 + 0.707)}{3\ (1 + 0.707) + 12} = 3.59 \text{ inches}$$

and

$$C = 3 + 12 - 3.59 = 11.41 \text{ inches}$$

The link mechanism described can also be used to advantage for other purposes, such as supporting tables where space is limited, as it permits the tables to be easily folded upward against the walls.

Stripper Mechanism for Wire-Forming.—In the operation of a wire-forming machine, one end of wire W—shown in the accompanying diagram, Fig. 17—is twisted around a pin, as indicated in the second view from the top, and then stripped off the pin in preparation for a subsequent operation. The twisted wire must be held in contact with die-plate K, as shown in the third view from the top, for a short time after pin F is withdrawn. Stripping fingers S must then be raised out of the way after the work has been stripped. The stripping operation is performed by frictionally operated levers in the manner shown by the diagrams.

The diagram at the top of Fig. 17 shows the levers immediately before the stripping operation is started. Shaft A, supported by bearing B, is given an intermittent oscillating motion by a cam (not shown). Shaft A is keyed to lever C, which carries pin F around which wire W is twisted. Lever E swivels on pin H, carried on lever C, and is shaped on the free end to form stripping fingers S. Lever D, which is made of bronze, is split (as shown) to permit it to be clamped around the finished hub extension of bearing B, the frictional resistance being adjusted by tension spring T. The hole in the outer end of lever D is slightly elongated and engages pin I, which actuates lever E.

The assembly is shown in its lower resting position in the top view of Fig. 17. Stripping fingers S of lever E are raised against the under side of lever C to allow space for the entrance of the initial forming die, which, at this point, has completed its work and withdrawn. As lever C is raised to withdraw pin F, lever D, being frictionally attached to bearing B, is not raised immediately; but, as pin H rises with lever C, lever E is caused to swivel on fulcrum pin I.

As lever C swings upward, the outer end of lever E is

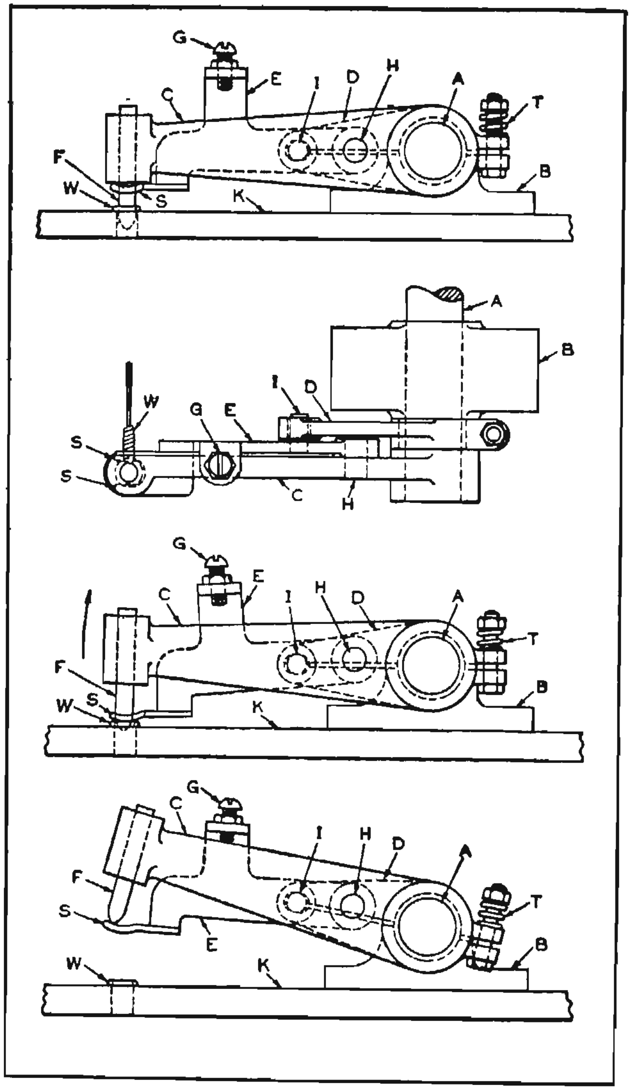


Fig. 17. (Top) Stripper Mechanism for Wire-forming Machine with Wire W Twisted around Mandrel F. (Center) Stripper Fingers about to Strip Wire W from Pin F. (Bottom) Mechanism with Pin F and Stripper Fingers S Raised to their Highest Points.

caused to swing downward, as shown in the third view from the top of Fig. 17. Since fingers S at the outer end of lever E press the work downward against die-plate K, the movement of lever E is restricted, and continued movement of lever C in the direction indicated by the arrow causes lever D to slip on the hub of bearing B. The frictional resistance of lever D on the hub of bearing B thus reacts as a downward pressure on the outer end of lever E, as indicated in this same view, and in the plan view shown in the second view from the top of Fig. 17.

As the movement of lever C continues, its position relative to lever E continues to change until screw G on lever E comes in contact with the upper edge of lever C. From this point, the entire assembly continues its movement as a unit, as shown in the bottom view of Fig. 17, until lever C reaches its extreme upper position, where it will permit the entrance of a forming die. As the motion of lever C is reversed, the frictional resistance of lever D on the hub of bearing B reacts on lever E in the reverse direction, so that the outer end of lever E is immediately raised until it comes in contact with the under side of lever C, when the entire assembly moves as a single unit until it reaches the position illustrated in the top diagram of Fig. 17, ready for the next stripping operation to be performed.

Differential Screw Design.—The use of a differential screw for very fine adjustment has well known advantages. However, these often seem to be outweighed by the difficulty of obtaining the required perfection in two co-axial threads differing minutely from each other in pitch. It is possible, however, to make up differential screws with effective pitches of only a few thousandths inch from screws of no extraordinary quality, and these can be used for very delicate adjustments of quite heavy members. Such a differential screw is shown in Fig. 18.

As shown in this illustration, a movable part of width M is to move a distance of plus or minus d relative to the fixed member, which has a width F. Let the pitch of the coarser threaded section of the screw which passes through the fixed member be designated P_t, and the pitch of the finer threaded section of the screw which passes through the movable member as P_m.

If the coarse-threaded part of the screw is at the extreme left in the fixed member, and the movable member is at the extreme right on the fine-pitch thread, then:

1. For each turn of the screw, the coarse-threaded part of the screw will move to the right a distance of P_t and the movable member will move to the left on the fine-pitch thread a distance of P_m.

2. The movable member will thus have a net movement to the right with relation to the fixed member that is equal to $P_t - P_m$.

3. Now if the movable member is to have a total net movement to the right of $2d$, then $\dfrac{2d}{P_t - P_m}$ turns of the screw will be required.

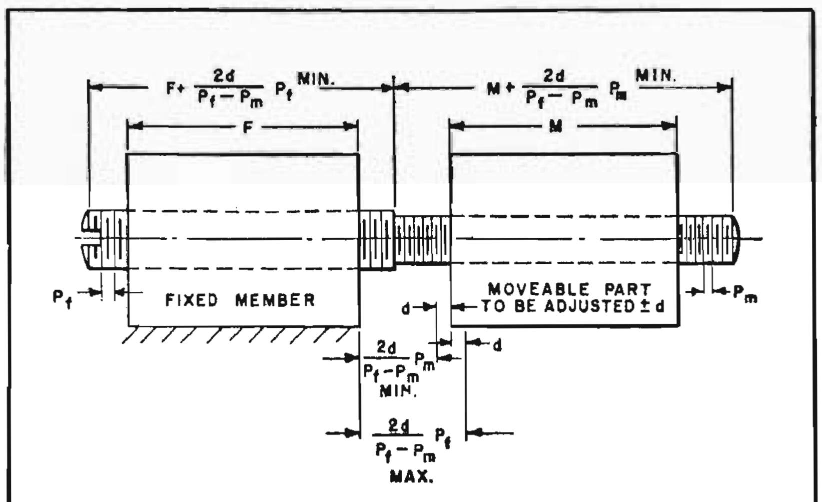


Fig. 18. Diagram of Differential Screw Showing Formulas for Calculating Minimum Lengths of Coarse- and Fine-threaded Sections.

4. For that number of turns of the screw, the coarse-pitch portion will travel through the fixed member a distance equal to $\dfrac{2d}{P_t - P_m} \times P_t$; hence, the minimum length of the coarse-pitch section must be $F + \dfrac{2d}{P_t - P_m} \times P_t$ as is indicated in Fig. 18.

5. Similarly, the movable member will travel on the fine-pitch section a distance equal to $\dfrac{2d}{P_t - P_m} \times P_m$, and the minimum length of that section will be $M + \dfrac{2d}{P_t - P_m} \times P_m$.

6. If both threaded sections are of minimum length, the maximum distance between the fixed and movable members will be equal to $\dfrac{2d}{P_t - P_m} \times P_t$, and the minimum distance will be equal to $\dfrac{2d}{P_t - P_m} \times P_m$.

Example—If the fixed and movable members are each 1 inch wide and a total range of adjustment of 1/8 inch in either direction is required, what will be the minimum length of the coarser thread, if it has 20 threads per inch, and of the finer thread, if it has 32 threads per inch? What will be the pitch of an equivalent thread that will provide the same fineness of adjustment? What will be the minimum and maximum distances between the fixed and movable members?

Solution—$F = 1$; $M = 1$; $d = \frac{1}{16} = 0.0625$;

$$P_f = \frac{1}{20} = 0.05;\ P_m = \frac{1}{32} = 0.03125$$

Minimum length of coarse thread =

$$1 + \frac{2 \times 0.0625}{0.05 - 0.03125} \times 0.05$$

$$= 1 + \frac{0.125 \times 0.05}{0.01875} = 1.333 \text{ inches}$$

Minimum length of finer thread =

$$1 + \frac{2 \times 0.0625}{0.05 - 0.03125} \times 0.03125$$

$$= 1 + \frac{0.125 \times 0.03125}{0.01875} = 1.208 \text{ inches}$$

Pitch of equivalent thread =
$0.05 - 0.03125 = 0.01875$ inch

Minimum distance between members = 0.208 inch

Maximum distance between members = 0.333 inch

Differential Screw Micrometer Mechanism.—A differential screw mechanism developed by the National Physical Laboratory of England to obtain the magnification of mi-

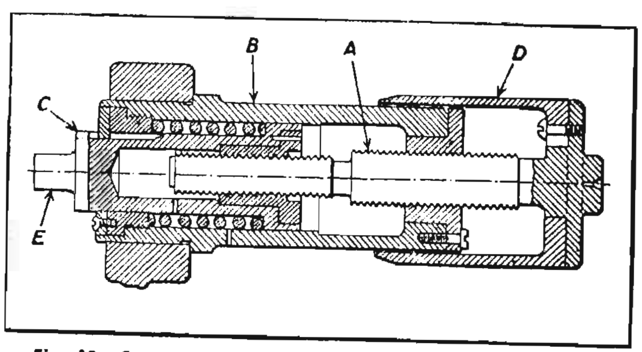


Fig. 19. Cross-sectional View of Differential Screw Mechanism of Micrometer.

crometer readings is shown in Fig. 19. In this design, two differential screws of relatively coarse pitch are employed to increase the accuracy of the micrometer reading. The larger screw A has 20 threads per inch, and the smaller one 25 threads per inch. Both threads are right-hand.

Screw A is engaged by a fixed nut at the right-hand end of barrel B, while the finer thread screw passes through a nut secured in sliding plunger C, the exposed end E of which forms one of the measuring anvils. A spring maintains contact between the micrometer screw and the two internal threads. The net movement of the plunger is $1/20 - 1/25 = 0.01$ inch for a complete rotation of the screw. The edge of thimble D, attached to the screw, is graduated in 100 divisions, each representing 0.0001 inch, the distance separating adjacent lines being about 0.04 inch.

The magnification with this arrangement is 400, compared with about 60 for an ordinary micrometer having 40 threads per inch and a thimble of about 1/2 inch diameter. A travel of 1 inch of the main screw moves the plunger only 0.2 inch. The total range of the instrument is, therefore, considerably reduced.

Pump with Flexible Rubber-Tube Action.—A pump designed to isolate the liquid or gas being pumped from the pump mechanism itself is a development of the Downingtown Mfg. Co., Downingtown, Pa. The operation of this pump, which is known as the Downingtown-Huber Squeegee type pump, is based upon the alternate squeezing and releasing of a rubber tube by a rocking compressor ring.

As shown in the diagrams, Fig. 20, the pump consists of six main parts. To the drive-shaft in the center is keyed an off-center rotor. This, in turn, can be keyed in any one of three positions to an adjustable eccentric. (In both diagrams the rotor is shown keyed in the central position.) The shaft, off-center rotor, and adjustable eccentric rotate as a unit in the compressor ring, and impart a rocking action to it.

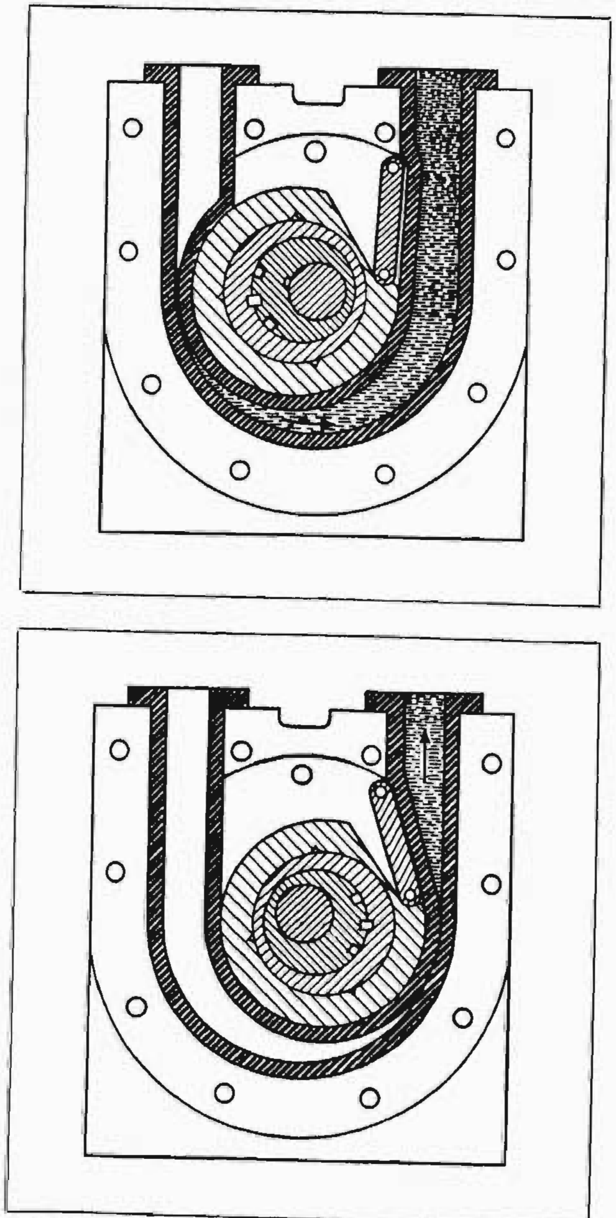


Fig. 20. Diagrams Showing Two Stages of Operation of Squeegee Type Pump in which Rotating Eccentric Member Forces Liquid in Rubber Tube from Inlet to Outlet.

This rocking action pushes the compressor ring out radially against the tube and continues progressively along the curved portion of the tube until it approaches the discharge end, when the tube is released and compression begins again at the intake end. Two stages of this action are shown in Fig. 20. Compression of the tube in this manner advances the liquid or gas being pumped toward the discharge side, while expansion of the tube back to its normal diameter produces a vacuum drawing more liquid or gas in from the intake side. A tube guide plate attached to the discharge side of the compressor ring prevents expansion of the tube beyond its normal diameter at this point where the tube is not surrounded by the pump housing or the compressor ring.

Thus, the liquid or gas being pumped is totally enclosed within the tube while passing through the pump. The tube itself can be made of pure gum rubber and various acid and oil resisting synthetic materials; in addition, it can be lined with synthetic materials for corrosion resistance or to prevent contamination. Solutions containing solids, whether abrasive or otherwise, cause little wear, as the inside of the tube is a smooth, continuous surface. The compressor ring merely rocks against the tube without any rubbing action, and hence causes practically no wear. A new tube is readily installed by simply fitting it inside the housing which forms its support and backing.

This pump is available in a fractional gallons per minute size which has a capacity range of from 0 to 6 gallons per hour, and develops a lift of 25 feet and a discharge pressure of 25 pounds per square inch. It weighs 3 pounds in a bronze housing, or 1 pound in a plastic housing. In the larger sizes, the capacities range from one-half gallon per minute up to 50 gallons per minute in the single-stage type, and up to 100 gallons per minute in the double-stage type.

Mechanism for Straightening Fine Wire.—Fig. 21 shows a comparatively simple machine designed for straightening coiled nickel-silver wire 1/64 inch in diameter. Straight pieces of this wire were required in lengths of 3, 6, and 12 inches. As it is difficult to straighten such short lengths, pieces 2 feet or more in length are straightened on the machine and then cut off to the required lengths. Wire in lengths as short as 2 feet, however, is generally drawn through the straightening machine twice. The end of the wire which leaves the machine last at the first pass is fed in first at the second pass. In some cases, more than two passes of the wire through the machine are necessary.

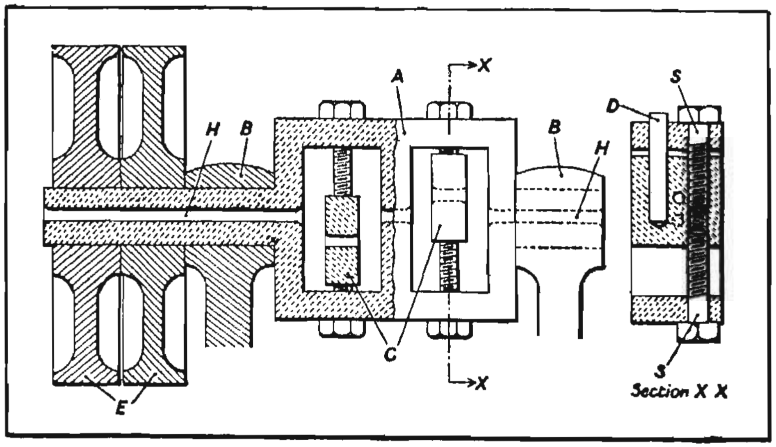


Fig. 21. Machine Designed for Straightening Fine Wire.

The machine is not complicated to make or to use. Similar designs are employed in the hosiery machine building trade for straightening the wires on which jacks are pivoted. Frame *A* of the machine is free to revolve in bearings *B*. It is shown driven by belt through the fast and loose pulleys *E*, although any other convenient form of drive can be provided. A suitable speed is about 100 R.P.M. A hole *H*, slightly larger in diameter than the wire to be straightened, is drilled through the center of the frame. Extreme accuracy in the diameter of this hole is not required, 1/8 inch being suitable for wires up to 3/32 inch in diameter.

Supported in slots in the frame *A* are the blocks *C*. These blocks are adjustable lengthwise in the slots by means of the screws *S*, while the dowels *D* prevent the blocks from turning. A hole of the same size as that drilled through the center of the frame is drilled in each of the blocks. These holes are so positioned that they can, by adjustment of the blocks, be aligned with the center hole in the frame, or they can be offset by an amount depending on the diameter of wire to be straightened.

In operation, the wire is fed into the center hole on one side of the machine until about 1/2 inch projects at the other side. This projecting piece is then gripped in a hand vise. The blocks are adjusted to positions that will cause the wire to be bent as it is pulled through the machine. The amount of this bending can best be found by trial, but it should be just sufficient to remove any kinks from the wire. The power is now applied, causing the frame to revolve, and the wire is pulled through the machine.

To prevent the wire from being scored, the ends of the holes in both the bending blocks and the frame should be provided with fairly large radii.

Mechanism for Taking Up Slack in Sprocket Chain.—A mechanism employing a weight-and-lever arrangement for taking up slack in a sprocket chain and for keeping the chain tight while it is in operation is shown in Fig. 22. On the non-movable base *A* is mounted a bracket *B* carrying a short shaft *C*, which supports a hinged lever *P*. Slidably mounted on the base is a unit *D* which carries the adjustable plate *E*. This plate supports the idler sprocket *F*, which is free to revolve on stud *G*. The function of the idler sprocket is to continuously exert sufficient pull on the sprocket chain *H* to keep it tight.

The cast-iron weight *K*, which exerts a pull in the direction indicated by arrow *L*, applies a pulling action to the connecting links *M* and *N* in the directions indicated by the arrows. This action transmits a compound movement to the hinged lever P, pulling back on rod Q and exerting sufficient pull on slide D to keep the chain H tight. It will be noted that the tendency of the weight to shorten the distance R as lever P pivots about the short shaft C causes the upper end of lever P to be forced to the left against the adjusting nut S on the thread at the outer end of rod Q.

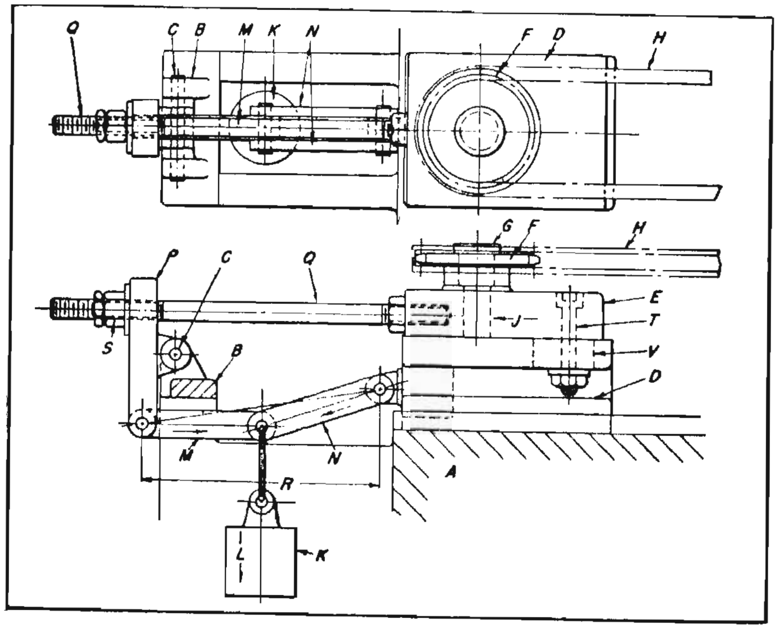


Fig. 22. Weight-actuated Mechanism Developed to Take up Slack in Sprocket Chain.

Plate E is adjustable on slide D and permits periodical adjustment to take up any major slack which may develop in the chain. Four screws of the type shown at T hold the plate in position, while the slide D is provided with an elongated slot V which permits adjustment of plate E. Adjusting nut S on rod Q permits setting the mechanism so that weight K will be in its most effective position for holding the chain under the tension required to take up the slack in the sprocket chain.

Mechanism for Operating Automatic Hoisting and Stacking Tongs.—The problem of providing more floor space to meet the requirements of companies engaged in armament work was solved in some cases by placing manufactured parts and partial assemblies into labeled barrels, boxes, or drums, and then piling up these packed containers to a height of several tiers. While this was found a satisfactory method of obtaining extra floor space, it required the use of a crane and good material-handling equipment. The quick-action automatic tongs, shown in Fig. 23, with operating mechanism designed as illustrated, were constructed especially for use in performing this work.

Referring to Fig. 23, center bar A and latches B are flame-cut from rough structural steel 3/4 inch thick. The center bar is purposely made heavy, so that it will descend by gravity when released. The four bars H are welded to the sides of the pivot bars, and the two lower cross-bars C are welded across the lower ends of bars H, as shown in the upper right-hand corner. The toothed gripping jaws D, four in number, are designed to obtain an equalizing grip on the container to be lifted. This feature permits the jaws to clamp and lift either round barrels, or containers or boxes having flat sides.

When the tongs are open, latches B are hooked over the two arc-shaped sections F that form part of center bar A. The tongs are then placed around the object to be lifted. A slight pull of the hand-chain G opens the latches and frees bars H and jaws D, so that they instantly approach each other and firmly grip any object to be handled, such as barrel K. The load is then hoisted by the crane hook and placed in the desired location.

When the crane hook is lowered, the center bar descends by gravity, as provided for by the vertical clearance slot S cut through its center. When it has descended far enough, latch hooks B are caused to snap together over the arc-shaped sections F through the pulling action of spring T

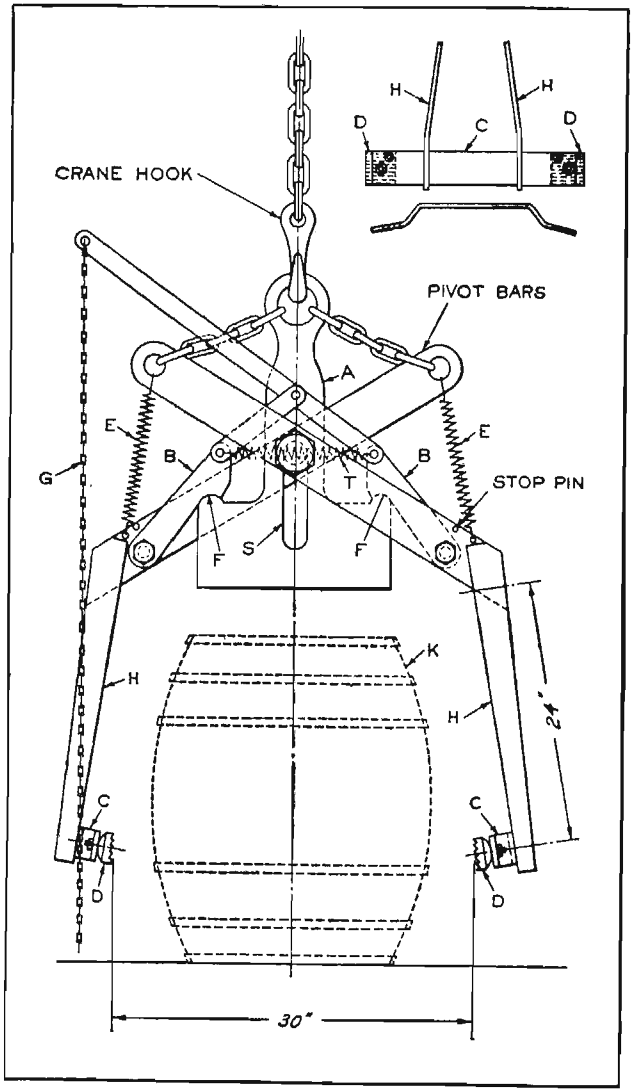


Fig. 23. Hoisting and Stacking Tongs with Mechanism for Gripping Barrel or Box when Control Chain G is Pulled and which Releases Load when Crane Hook is Lowered.

attached to their ends. When the crane hook is again raised, the latches cause the jaws to open so that the tongs are ready to be lowered for lifting the next container.

In removing packages from the tops of the stacked tiers, the operation of the device is simply reversed. The two light-tension springs *E* serve to prevent the jaws from collapsing or coming entirely together.

In building this equipment, no precision fits are necessary. The center block has a clearance of about 1/8 inch between the pivot bars. A separator bushing through the long vertical slot and around the pivot bolt has a similar clearance. These tongs grip securely anything that can be placed between them. Although the distance between the open jaws *D* is indicated as 30 inches in the illustration, this opening can be adjusted by providing several holes for the latch fulcrum bolts at different positions.

Friction Clutch for Grinding Machines Operated at High Speeds.—In Fig. 24 is shown a friction clutch that can be nicely balanced to permit operation at high speeds without perceptible vibration. The various parts of the clutch, shown in Fig. 25, are balanced separately and as a unit. The body *A*, Fig. 24, is made of semi-steel and finished all over. It has a bushing *K* and runs free when not engaged with the expanding ring *B*. The expanding ring is also made of semi-steel and is so shaped that it will balance itself. The raised portion at *J* shows the method of distributing the metal to obtain the required balance. At *C* is a hardened steel cone that slides along the shaft to operate the finger *D*. It is rounded at the front end *M* which acts as a cam when in contact with the point of the set-screw *I*. The set-screw *I* is screwed into a tapped hole in finger *D* and is locked in position by a check-nut. The round point of this screw is hardened to prevent wear.

The clutch finger *D* is a steel casting, shaped to assist in balancing the unit. It is pivoted about the screw *F*, which is threaded into and fastened to the expanding ring *B*. The

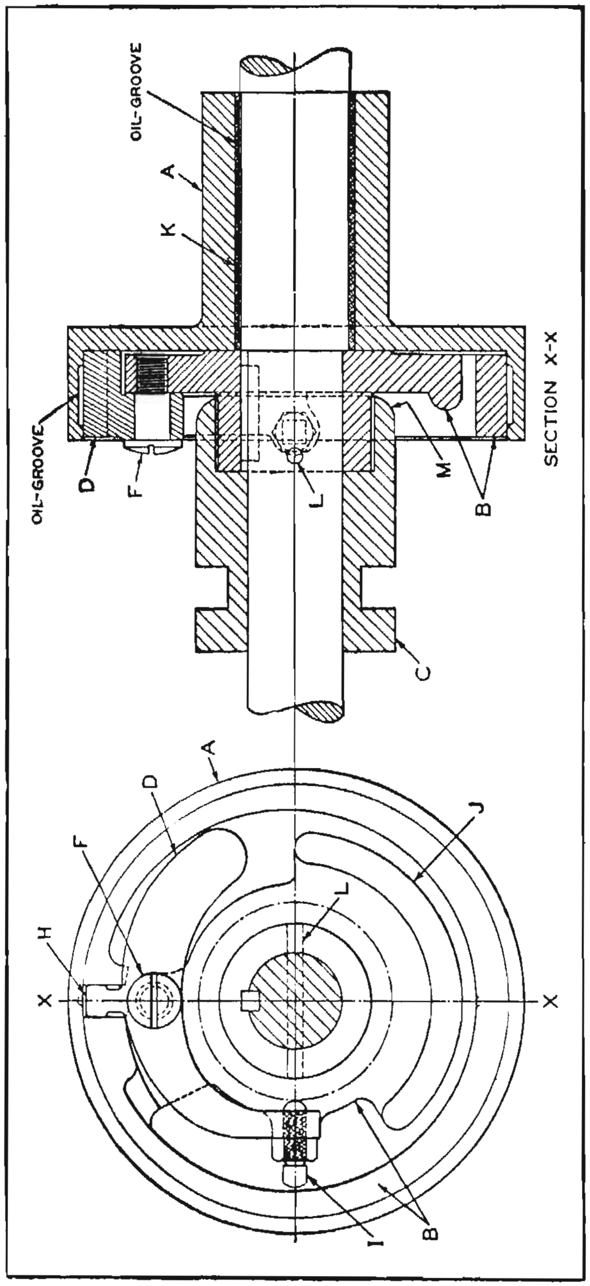


Fig. 24. Friction Clutch Designed for Use at High Speeds.

lever arm H is a part of finger D and serves to expand ring B. Ring B is keyed to the shaft and is held in position by a pin L.

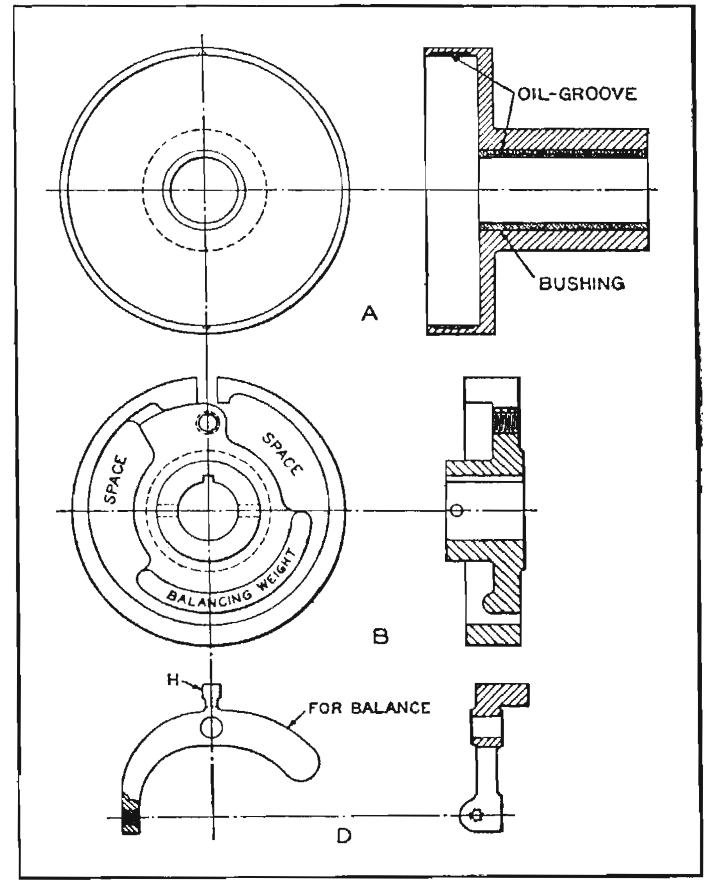


Fig. 25. Detail of Parts Comprising Clutch Shown in Fig. 24.

To engage the clutch, the operator simply slides the cone C toward the expanding finger D, bringing the cam face M into contact with the set-screw I and moving it outward, thus causing finger D to pivot about screw F. The lever arm H fitted in the slot in ring B causes the ring to expand

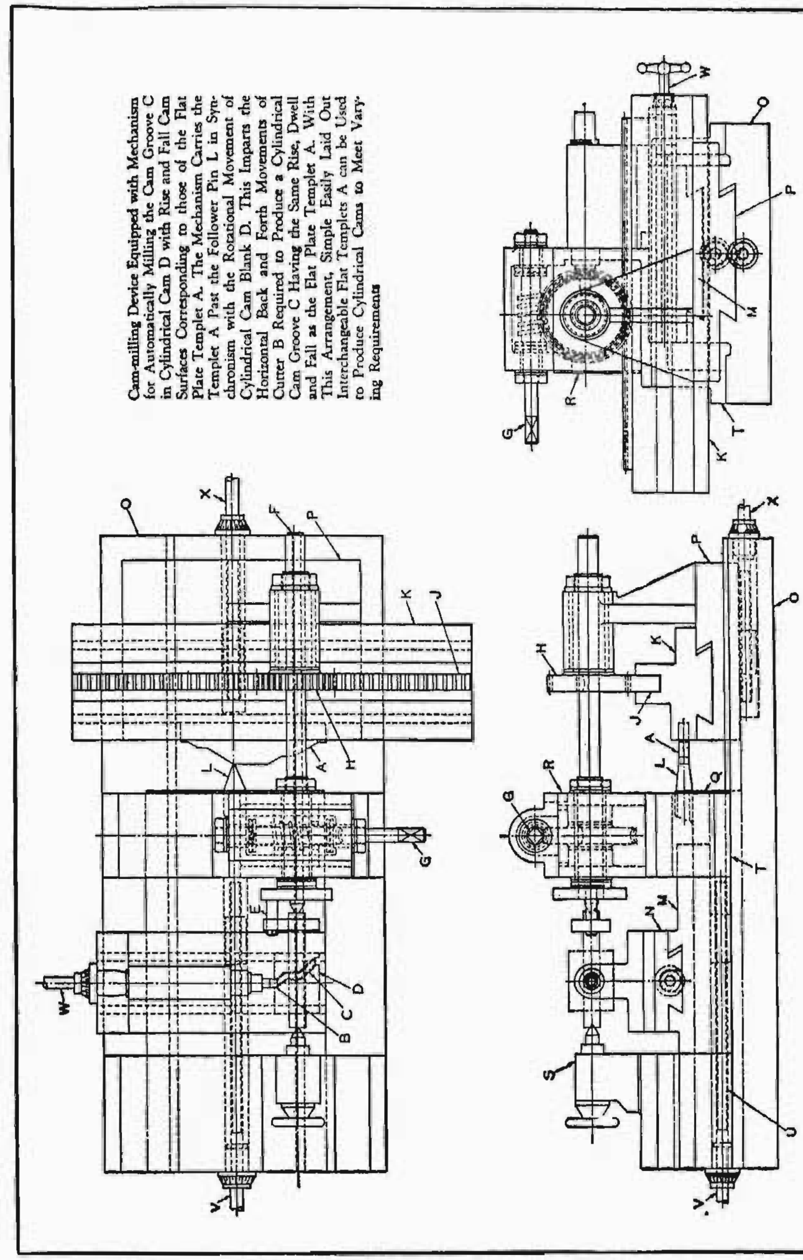
and grip the inner friction surface of the clutch body A. The expansion of ring B causes the entire unit to turn as one member. Cone C is counterbored to fit over the hub of member B, so as to enable the clutch cone to operate as near the center of the clutch ring as possible. All rotating parts of the clutch are lubricated with oil.

Mechanism for Controlling Cutter-Head Slide of Cam-Generating Device.—An automatic cam-generating device with a templet and follower-pin mechanism for controlling the cutter-head slide is shown in Fig. 26. One of the chief advantages of the mechanism here illustrated is the comparative ease with which the flat-plate master profile or contour templet A can be developed, laid out, and machined. Much work is saved as compared with developing the layout on a master cylindrical cam blank and machining the cam groove directly on the cylindrical blank.

Use of the flat-plate templet is especially advantageous in the case of slow-moving, uniform-motion cams in which the developed profile diagram of the various rise-and-fall surfaces consists of simple inclined straight lines. Development templets made from steel plate or flat ground stock are employed, as shown at A in the illustration, for the mechanism that controls the slide on which the cutter B is mounted for generating cam groove C in cylindrical blank D.

The cam blank D in which groove C is to be cut is mounted on an arbor between centers with a driving pin E arranged to rotate the work when the spindle F is rotated by turning worm-shaft G either with a hand-crank or a motor equipped with a reduction-gear drive. The worm on shaft G, meshing with the worm-wheel on shaft F, rotates pinion H, as well as the work. Pinion H, meshing with rack J on slide K, causes slide K to be traversed at right angles to the axis of shaft F simultaneously with the rotation of cam blank D.

As slide K moves transversely, the cam profile of templet A transmits longitudinal movement to slide Q through follower-pin L. The compound cutter-slides M and N, mounted



Cam-milling Device Equipped with Mechanism for Automatically Milling the Cam Groove C in Cylindrical Cam D with Rise and Fall Cam Surfaces Corresponding to those of the Flat Plate Templet A. The Mechanism Carries the Templet A Past the Follower Pin L in Synchronism with the Rotational Movement of Cylindrical Cam Blank D. This Imparts the Horizontal Back and Forth Movements of Cutter B Required to Produce a Cylindrical Cam Groove C Having the Same Rise, Dwell and Fall as the Flat Plate Templet A. With This Arrangement, Simple Easily Laid Out Interchangeable Flat Templets A can be Used to Produce Cylindrical Cams to Meet Varying Requirements

Fig. 26. Cam-milling Device Equipped with Mechanism for Automatically Milling Cam Groove C in Cylindrical Cam D Using Flat Templet A.

on slide Q, thus traverse the cutter B longitudinally to the left or right along the axis of the cam blank D in accordance with the profile or contour of the cam templet A. This simultaneous rotation of the cylindrical cam blank D and traversing movement of the rotating end-milling cutter B results in milling the cam groove C with the required rise-and-fall contours predetermined by the contour or profile of cam templet A. During the cam milling operation the follower-pin L is kept in contact with templet A by means of a cable and weight (not shown), which is attached to the end of the slide M.

Referring to the structural details of the device, base O is provided with machined ways throughout its entire length and carries the two longitudinal slides P and Q. Slide P is cast integral with the bracket bearing which supports work-spindle F, and it also carries the cross-slide K with rack J and templet A.

Slide Q, supporting follower-pin L, carries the compound slide M. Slide N is mounted on slide M and carries the spindle of cutter B. Headstock R and tailstock S are mounted on bridges T and U, respectively. Gear H, which is cast integral with a long sleeve bearing and is keyed to the work-spindle F, drives rack J secured to cross-slide K, thus imparting the traverse movement to templet A.

Compound slide M is provided with a long feed-screw V, which enables the cutter B to be set in any desired position along the surface of the work. Cross-slide N, carrying the spindle head, is provided with cross-feed screw W which enables cutter B to be adjusted radially to any desired distance from the center of the work. Screw W is also used for feeding the cutter into the work. The setting of slide P to accommodate templets of various sizes is accomplished by means of screw X.

Mechanism for Adjusting Throw or Radial Position of Block on Rotating Arm.— The mechanism here described was designed to permit a very fine, continuous adjustment

of block A, Fig. 27, left-hand diagram, along radial arm B while the arm is rotating about axis X–X. The arrangement of the mechanism, as adopted, is shown diagrammatically in the center diagram of Fig. 27, while an alternative arrangement (considered, but not adopted) is shown in the right-hand diagram of Fig. 27.

Screw C is used to move the block, and is, in turn, driven by shaft D through bevel gears. Shaft D carries one wheel of differential E, the other wheel of which is driven by a

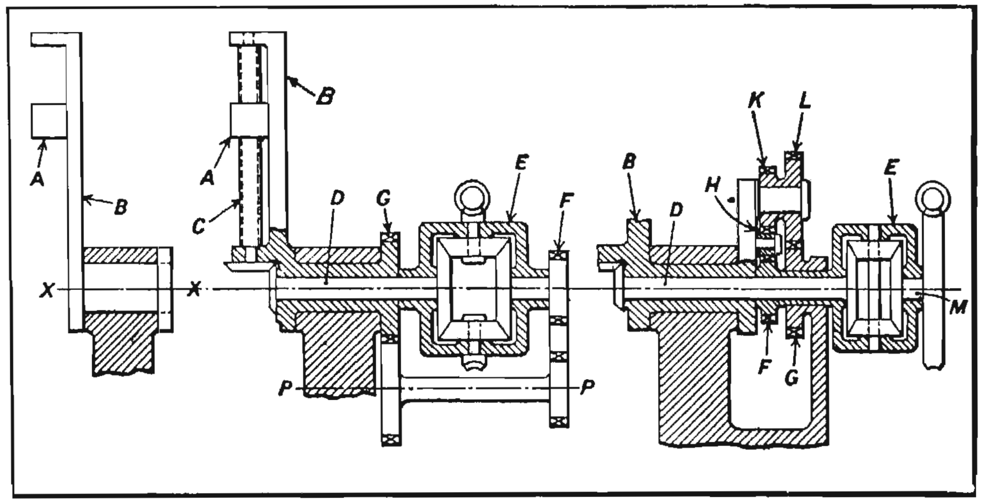


Fig. 27. (Left) Section of Mechanism that is Required to Have the Throw of Block A Adjusted while Arm B is Rotating. (Center) Sectional View of Mechanism Designed for Adjusting Throw of Block A. (Right) Sectional View of Alternative Throw-adjusting Arrangement.

train of gears from wheel G, which is fixed to arm B. The intermediate shaft rotates about a fixed axis P–P.

The ratio of the gear train beginning with driver G and ending with driven gear F is unity, and because of the reverse idler, shown at the right-hand end of the gear train, gears F and G rotate in opposite directions. Thus, when the housing of the differential is stationary, shaft D rotates in the same direction and at the same speed as arm B, and the position of block A on the arm is fixed. If, however, the differential housing is rotated, then its motion is added to, or subtracted from, that of shaft D, which, consequently, rotates relative to arm B and thus moves block A inward or outward. The block is, therefore, controlled by the worm and wheel drive to the differential housing. The drive to the mechanism and to arm B is through gear G.

In the alternative arrangement, shown in the right-hand diagram of Fig. 27, the ratio of the epicyclic train F to G must be 1 to 2, in order that the differential housing may be rotated in the same direction as, but at half the speed of, arm B. This requires the number of teeth in G, multiplied by the number of teeth in K, divided by the product of the number of teeth in L, and the number of teeth in F to be equivalent to the ratio 1 to 2, and also necessitates the use of intermediate gear H.

When shaft M is stationary, the rotation of the differential housing E causes shaft D to rotate in the same direction and at double the speed, so that it rotates in synchronism with arm B, and block A thus remains in a fixed position. When shaft M is rotated, the motion is transmitted to screw C and the block is moved. This arrangement, however, was not adopted because the drive to the arm was more difficult to arrange.

INDEX

INGENIOUS MECHANISMS

FOR DESIGNERS AND INVENTORS

VOLUME IV

Mechanisms and Mechanical Movements Selected from Automatic Machines and Various Other Forms of Mechanical Apparatus as Outstanding Examples of Ingenious Design Embodying Ideas or Principles Applicable in Designing Machines or Devices Requiring Automatic Features or Mechanical Control

Edited by

JOHN A. NEWELL

and

HOLBROOK L. HORTON

INDUSTRIAL PRESS INC.

200 MADISON AVENUE, NEW YORK 10016

Industrial Press Inc.
989 Avenue of the Americas, New York, NY 10018
Tel: 212-889-6330 Toll-Free: 1-888-528-7852 Fax: 212-545-8327
www.industrialpress.com Email: info@industrialpress.com

INGENIOUS MECHANISMS
FOR DESIGNERS AND INVENTORS—VOLUME IV

26 28 30 29 27

FOURTH VOLUME OF INGENIOUS MECHANISMS

A considerable period of time has elapsed since the publication of the third volume of Ingenious Mechanisms. During this period we have received many inquiries about the possible publication of a fourth volume in the series, indicating a continuing interest in this area.

This fourth volume follows the same pattern as its predecessors. The mechanisms described have been developed for application in a wide variety of fields. Rather than classify them by application, however, they have been grouped by type of mechanical movement. Thus, the reader is quickly guided to those mechanisms which may provide a possible solution to his problem.

Furthermore, the grouping is closely similar to, if not exactly the same as, that in the previous volumes so that the entire set may be used as an integrated reference library on the subject of mechanisms.

CONTENTS

CHAPTER 1

Cam Applications and Special Cam Designs

In the design of mechanisms to obtain irregular movements of various kinds, cams are frequently employed. Those which are described or illustrated in connection with the mechanisms covered by this chapter are notable for some ingenious arrangement or design. Other applications of cams and cam-operated mechanisms will be found in Chapter 1, Volume I; Chapter 1, Volume II; and Chapter I, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Cam Produces Motion on Alternate Revolutions

A conventional plate-cam was used on a machine producing a wire product to operate a forming press. The press had to be actuated once during each revolution of the driving shaft. A subsequent product change necessitated an alteration in the operating cycle of the cam — it was now required to operate the press twice during one revolution, then to remain at rest during the next revolution. Figures 1 and 2 show the design and operation of a cam which produced the desired movements with no alterations being required on the machine.

Cam body *A*, Fig. 1, is in the shape of a disc having an integral hub on its front face. The cam is keyed to shaft *B* and rotates in the direction indicated by the arrow. Two studs *C* pass through the disc and are free to rotate. Welded to them are curved bars *D* which act as cam lobes. Compression springs *E* apply sufficient frictional resistance to the studs to prevent movement by centrifugal force. Lever *F*, which operates the forming press, carries

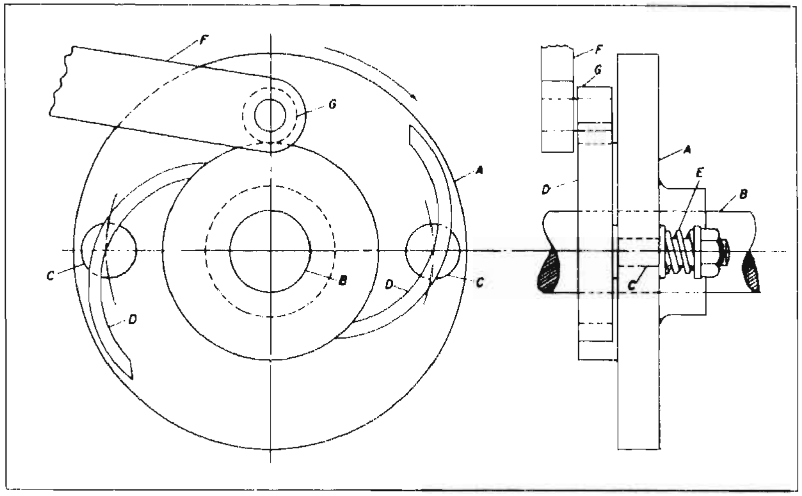


FIG. 1. Cam body A carries two moving cam-bars D. This design imparts two movements to lever F during one revolution, followed by one revolution at rest.

follower-roller G and is held against the cam by a spring (not shown).

Operation of the cam is illustrated in Fig. 2. At W, cam-bar D is in the same position as in Fig. 1, but the entire cam has rotated 90 degrees. Stud C and roller G are now on the same center line, bar D having caused the roller to rise and operate the press through lever F. The cam continues to rotate, as at X, and spring tension on lever F overcomes the frictional resistance of stud C, forcing bar D into the position shown. When the other lobe of the cam comes into position this action is repeated, so that two movements of the lever are produced, 180 degrees apart, in one revolution of shaft B.

No movement of lever F is produced during the next revolution of the shaft. This is because the leading ends of the cam-bars have been lifted from the hub of disc A and now pass over roller G, as shown at Y. As the cam rotates further, and the roller passes the center line of stud C, cam-bar D is forced to pivot as at Z — thus being returned to its original position (Fig. 1). In this manner, each two-revolution cycle of the cam produces two

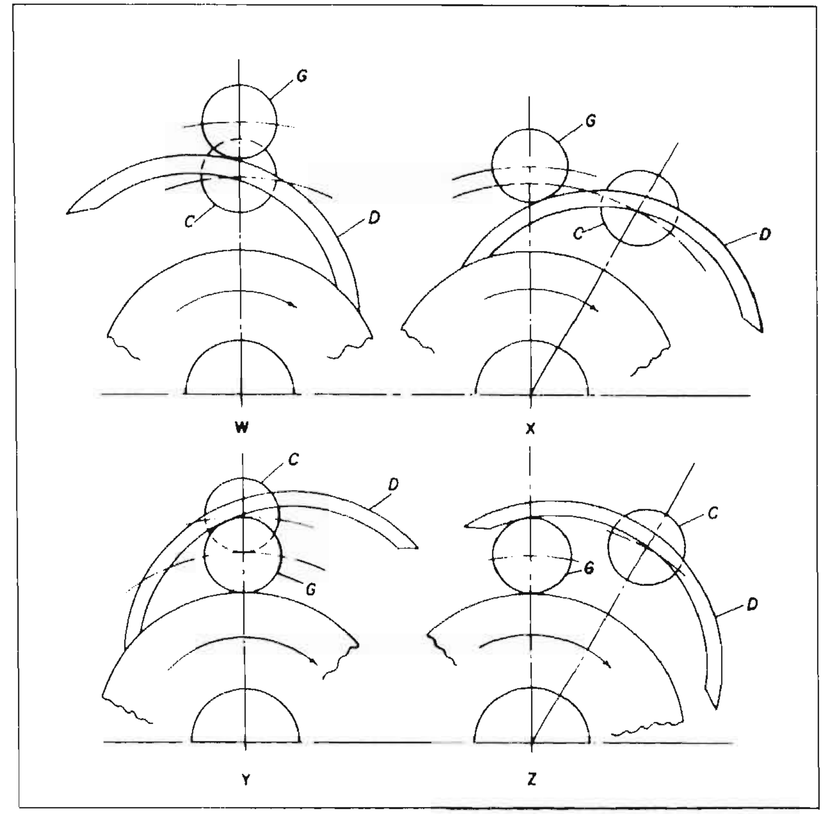


FIG. 2. During the active revolution, follower G rises along the cam-bar D, view W, then forces it to pivot for the downward movement, view X. On the next trip around, the upended cam-bar passes over the follower and is returned to its original position, views Y and Z.

movements of lever F, followed by a rest period of 540 degrees.

There may appear to be an undesirable feature in the design of this cam in that there would be a rapid drop of roller G on the falling side of bar D (view X, Fig. 2). This, however, does not occur, due to the fact that the outer surface of the cam-bar is on a rising angle (view W). Thus, downward movement takes place almost immediately after the center of stud C passes the center of roller G.

Outer surfaces of the cam-bars may be contoured to produce almost any conventional rise and fall pattern. Their inner surfaces must be so dimensioned that there will be sufficient clearance for the passage of roller G, and that full closing of the leading ends will be assured when the roller exits from beneath them.

Four-Lobed Cam Transmits Variable Motion to Follower

On a machine for fabricating a formed wire part, a revision in the product design required a change in a cam which operated a press. Previously, there had been a uniform oscillating motion of the follower with each revolution of the operating shaft. For the new design, it became necessary to transmit a motion of varying magnitude and varying timing for each of four revolutions of the shaft, without any major changes in the machine. The drawing shows the cam that was made to meet the requirements.

In Fig. 3, operating shaft A, rotating in the direction indicated, carries arm B, which is keyed to it. On the arm is pawl C, which

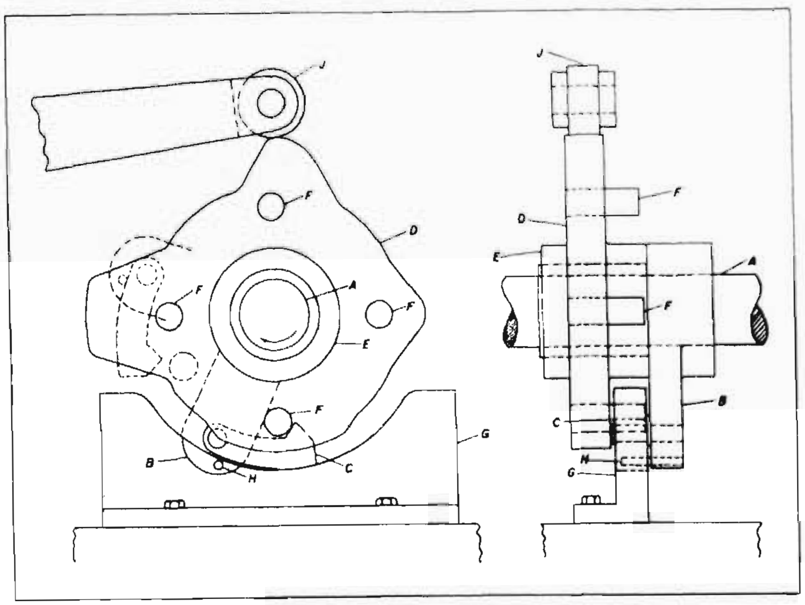


FIG. 3. As long as pawl C remains in contact with angle-plate G, cam D rotates. When the pawl leaves the angle-plate, the cam stops.

can swing on its stud. Four-lobed cam D is free on the extended hub of the arm, and is retained by collar E. Projecting from the face of the cam are four pins F, equally spaced around the center of rotation. Angle-plate G is machined to a true arc of a circle on its upper edge, and is attached to a stationary part of the machine.

The shaft, arm, and pawl rotate as a unit. No motion is transmitted to the cam until the pawl contacts the upper edge of the angle plate, when the pawl is brought into position to engage one of the pins. The drawing shows the position of the components at about the midpoint of the cam movement.

Having engaged one of the pins, the pawl carries the cam with it, until it no longer contacts the angle-plate. At this point, due to the angularity of the contact surface of the pawl, it disengages automatically, and the cam stops. (The position of the pawl at the time of disengagement is shown in broken lines on the left side of the drawing.) Stop H limits the swing of the pawl so that it will be in position to engage the next pin upon rotating to the right side of the angle-plate. Thus, the cam rotates 90 degrees for each revolution of the shaft, and the four lobes of the cam are brought consecutively into position to actuate cam follower J as required.

Intermittent Rotary Motion from a Uniform Reciprocating Drive

Two devices on a machine had to be rotated intermittently, with a rest period at each of eight stations in each cycle. Although the loading devices were widely separated, they could be placed in axial alignment and, be carried on the same shaft. The required movement, shown in Fig. 4, was obtained from a barrel-cam driven by a reciprocating part.

Shaft A, on which the loading devices are mounted, is supported in bearings and carries a barrel-cam B with an irregular groove. Roller C operates in this groove and is carried on a slide-bar D mounted on a stationary part of the machine by gibs E. Member D is given a uniform reciprocating motion by a cam (not shown). Since there are eight stations, eight axial follower

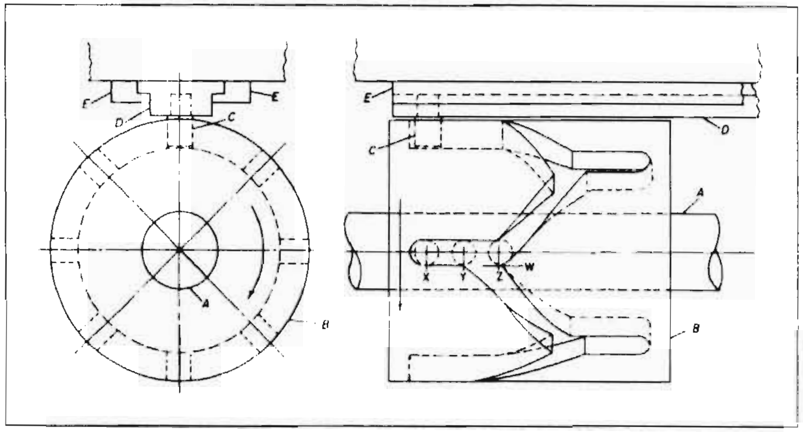


FIG. 4. Cam mechanism for converting uniform reciprocating motion to intermittent rotary motion.

grooves are machined on opposite sides of the cam barrel. These are connected by other grooves milled in the periphery at an angle of about 45 degrees with the axis of the cam. The vertex of the angle formed by any two of the angular grooves is approximately in line with one side of the axial grooves, as indicated by line W.

The assembly is shown with slide-bar D at the extreme left position. To demonstrate the action, three positions of the roller in the center groove are seen at X, Y, and Z. Roller C, in moving from the extreme right position, acts against the angular side of the groove, causing the cam B to rotate in the direction indicated by the arrow. When roller C reaches position Y, rotation of the cam ceases, and it remains at rest during the continued movement of the roller to the extreme left position X.

On the return movement, no rotation of cam B is produced until roller C again contacts the angular groove at position Z. Since the vertex of the angular groove surfaces are not aligned with the centers of the axial grooves, the roller cannot return to the groove previously traversed, but must enter the next one. Continued movement of roller C causes cam B and shaft A to rotate to the next station, and the cycle is repeated.

Cylindrical Cam Positions Wire Guide

Figures 5 and 6 show two views of a mechanism designed to guide a strand of wire through an irregular path in a machine which produces a woven wire product. The purpose of this mechanism is to create a continuously varying pattern in the weave. Position of the wire strand W in the weave pattern must bear a given relationship to other parts of the weave over a required length of the fabric, and then repeat. Figure 5 is a plain view of the mechanism, and Fig. 6 is a front view.

The driving shaft A carries the worm B, which meshes with the worm-gear C on shaft D. Shaft D carries the cylindrical cam E, which imparts a transverse guiding movement to wire W. The two rounded grooves in cam E are identical, although the axes of the grooves are offset from the shaft axis and are 180 degrees apart.

Shaft G receives motion from worm B through worm-gear F, and carries the disc H, which is connected to block J by the pitman I. Block J is attached to the dovetailed slide K, which is

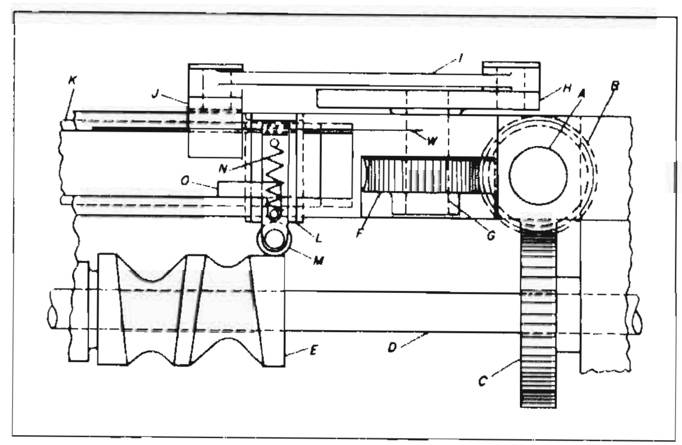


FIG. 5. Plan view of wire-guide mechanism for a metal textile weaving machine designed to impart an intricate transverse motion pattern to wire strand W.

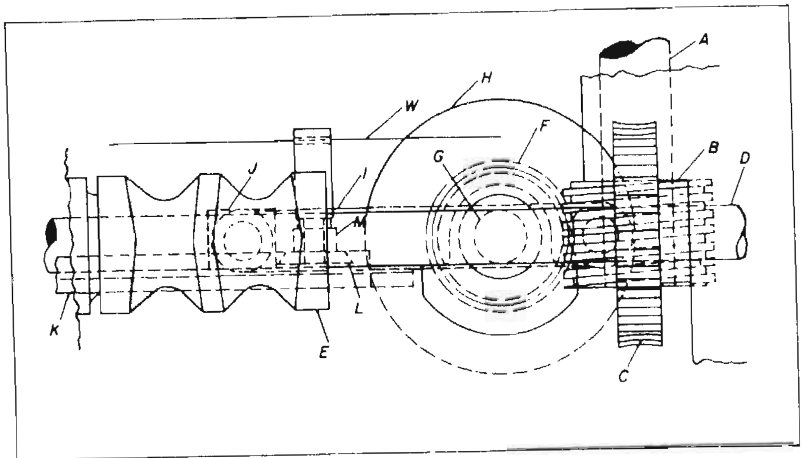


FIG. 6. Sixty turns of shaft A are required to completely cycle follower M through a single out-and-back traverse of rotating cam E.

given a reciprocating motion by the rotation of crank disc H. Slide K carries the dovetailed slide L, which in turn mounts the roller follower M. The follower roll is held in contact with cam E by the spring N attached to a pin in slide L and a pin in bracket O attached to slide K.

In operation, the rotation of worm B transmits rotary motion to cam E through worm-gear C and shaft D, and reciprocating motion to slide K through worm-gear F and disc H. As slide K moves, roller M traverses axially along cam E, following the grooves which are constantly varying in width and depth as a result of the rotation of cam E. The position of slide L in slide K is continually changing except when roller M is in contact with the cylindrical periphery of cam E. As the strand of wire W feeds through a hole in the leg of slide L, its position is guided by the movement of slide L.

In the diagrams, which show the position of the mechanism at the beginning of the cycle, wire W is guided in a straight path until roller M begins to follow the right-hand groove in cam E. Thus the wire is moved from start position. It returns to its starting position when roller M returns to the periphery of cam E. After a short period of rest, slide L again moves as roller M enters

the second groove. This is followed by a period of rest as roller M again reaches the periphery of cam E. Worm-gears C and F are of different pitch diameters. G has thirty teeth and F has twenty teeth. Therefore, the rotation of cam E and the movement of slide K are not synchronized. Thus the path followed by roller M varies as the varying contours of the cam are presented to it. This action results in varying rest periods at the ends of the movement of slide K, as well as a varying timing pattern and positioning of slide L at different points of the cycle, setting up an intricate pattern in the positioning of wire W.

While the diagrams indicate the starting point of the cycle, the completion of the cycle is accomplished only when all of the moving members of the mechanism are returned to this start position. As stated, worm-gears C and F have thirty and twenty teeth, respectively, having a ratio of 3 to 2. Therefore, for a complete cycle, gear F must complete three revolutions, and gear C must complete two revolutions. The complete cycle, therefore, requires sixty revolutions of drive-shaft A.

Combination Cam Controls Stock Feed of Wire-Forming Machine

A combination end and radial cam is the heart of a stock feed for a multiple-slide wire-forming machine. Figure 7 shows the arrangement of the parts.

The combination cam A is carried by the machine's shaft system. It is basically a two-diameter plug, the shoulder having been modified to form an axial cam, and the small diameter to form a radial cam. Follower B in lever C rides on both cam surfaces.

This lever fulcrums at its center on stud D. By being joined to the stud by cross-pin E, the lever can swing both left to right (to follow the end cam surface) and in and out (to follow the radial cam surface).

Directly behind the lower end of the lever is a dovetail slide F containing quill G. The quill has two parts. One part fits the dovetail, and the other has a pin carrying bushing J which fits a slot in the lower end of the lever. There is a semicircular section

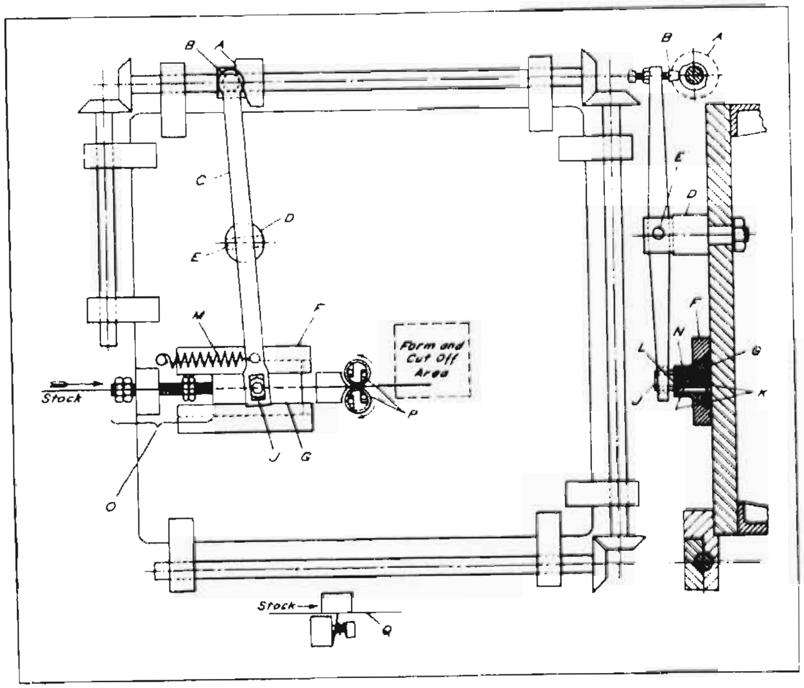


FIG. 7. Cam *A* controls the feed of the wire by having its motion translated to quill *G* through lever *C*.

K in the mating surfaces of the two parts of the quill, through which the stock advances. (The section can be modified to accept whatever stock size or shape is used.) The outer part of the quill is held in position by loose-fitting pin *L*.

The drawing shows the lever position at a point in the feed stroke. At the start of the stroke, spring *M* pulls the bottom of the lever to the left. As the cam starts to rotate, the lobe on its small diameter bears against the follower, causing the bottom of the lever to swing in and thus force the quill to close tightly over the wire. At the same time, the end cam forces the bottom of the lever to the right, thus advancing the wire.

When the lever reaches the end of the forward stroke, the follower no longer has any thrust on it from either cam surface. Spring *N*, contained in a hole extending into both parts of the quill, then allows the quill to release its grip on the wire. Next,

spring *M* operates, pulling the lever to the left, and the feed cycle is completed.

Stop-block assembly *O* provides stroke-length adjustment by controlling the point to which the lever can return. A simple way to prevent the wire from tending to move back on the return stroke is to add a pair of non-reversing rolls *P*. Or, a spring check *Q* can be used.

Mechanism Loops and Twists Wire Ends

Completion of a certain job necessitated the production of wire components having a twisted loop at each end. The soft steel wire parts were required in lengths of 16 inches and longer. A typical twisted loop is shown at X in Fig. 8.

Hollow shaft *A* has a running fit on drive shaft *B*, and is coupled to it by means of a clutch, a portion of which may be

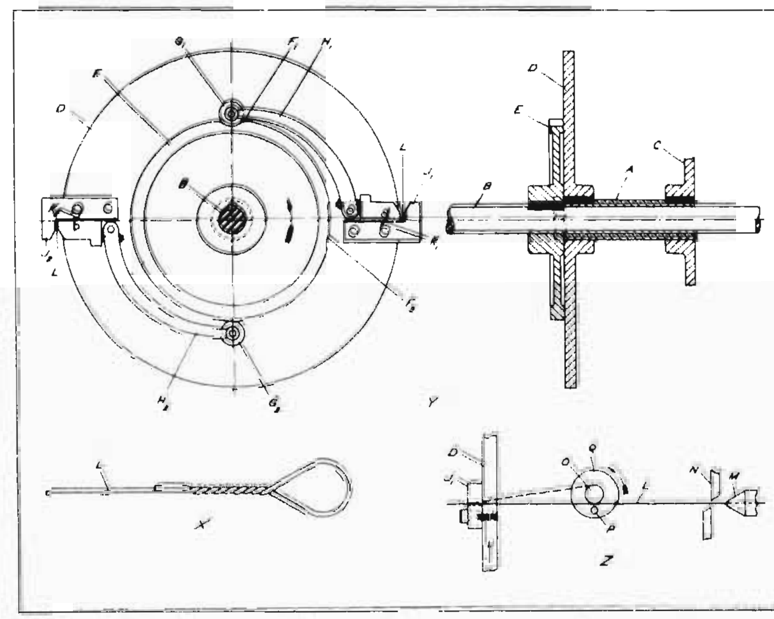


FIG. 8. Device loops and clamps wire component, then transfers it on a rotary disc to a point where the loop is twisted into the form shown at X.

seen at C. This clutch allows shaft A, shown at Y, to rotate one-half revolution for each complete revolution made by shaft B. During the remaining one-half revolution of the drive shaft, shaft A dwells. Keyed to the left-hand end of the hollow shaft is mounting disc D.

Disc cam E, having two notches, F_1 and F_2, is keyed directly to the drive shaft. The purpose of the cam is to actuate follower rollers G_1 and G_2 which, in turn, pivot levers H_1 and H_2. This action either causes the jaws of clamps J_1 and J_2 to close or permits them to open under the influence of springs K_1 and K_2. The two clamps, together with their actuating levers, are mounted on disc D. There is a second complete unit (not shown), identical to the one illustrated, at the left-hand end of shaft B to twist a loop in the opposite end of the wire part.

In operation, the wire L to be formed is fed through guide nozzle M, as shown at Z, between two cutting blades N and then between two pins O and P of looping head Q. When the correct developed length of wire has been fed out, clamp J_1 moves up from below to nest it. At this point clutch C releases, with the result that the clamp remains stationary while cam E continues to rotate.

As this occurs, follower G_1 leaves notch F_1 and rides up on the high portion of the cam, forcing the clamp jaws to close on wire L. Cutting blades N now shear the wire, and looping head Q rotates, thus forming a loop in the wire end around pin O by the action of pin P. The wire end comes to rest in the V-shaped entrance to the clamping jaws and is held in this position by the pin P.

As cam notch F_2 arrives under follower G_1, the clamp jaws are permitted to open a sufficient amount to allow the wire end to drop in place. The passing of the notch once again closes the jaws, locking the looped wire securely in place while the looping head is retracted.

By this time, cam notch F_1 has arrived under follower G_2, allowing the jaws of clamp J_2 to open wide. At this instant clutch C engages, causing disc D and cam E to once again rotate in unison for one-half revolution. During this movement,

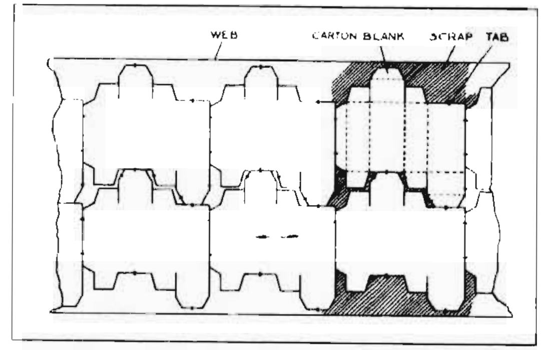


FIG. 9. Cardboard web layout showing die-cut carton blanks and the attached scrap material.

clamp J_2 moves up into contact with a newly fed length of wire, which begins the next cycle.

While in this position, with clamp J_1 dwelling at a location 180 degrees away from its starting point, a twisting head engages the clamped wire loop. This head, not shown, completes the twist as shown in the illustrated example of the workpiece. Following the twisting operation, cam notch F_1 moves beneath follower G_1, allowing the clamp jaws to open and to drop the finished wire components clear of the machine. The cycle of operation then continues.

Rotary Scrap-Stripping Device

One step in the manufacture of cardboard cartons is the die cutting of the developed form. Each printed carton blank is joined to the adjacent carton and to the scrap material surrounding the end flaps by means of small tabs. A typical layout, Fig. 9, shows the arrangement of the die-cut shapes on the web, or continuous cardboard strip; the scrap material, depicted by shading; and the tabs, indicated by two parallel lines.

Upon completion of the die-cutting operation, the scrap pieces must be removed from the web. This procedure, known as stripping, is frequently carried out by hand. However, the rotary scrap stripper shown in Fig. 10 has been designed to replace this manual operation.

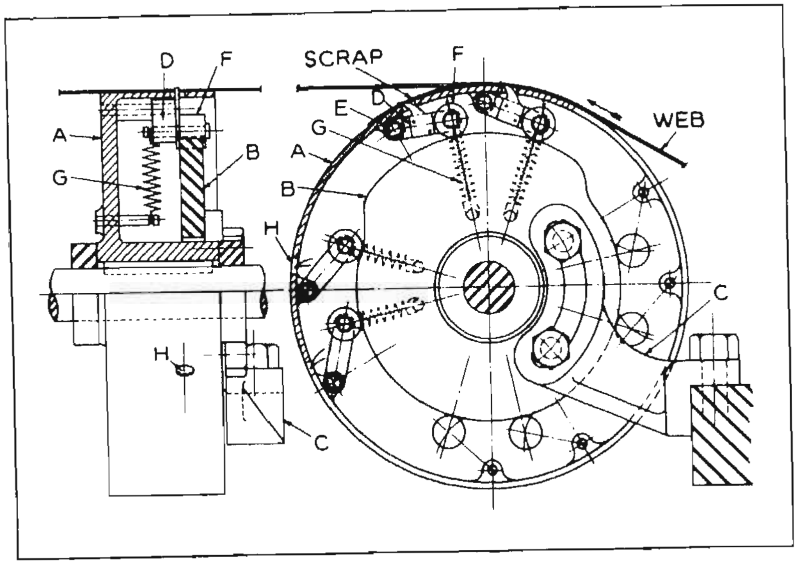


FIG. 10. A rotating unit strips scrap material from a die-cut web on a production line handling cardboard cartons.

The main member of the device is a cylindrical housing *A*. This housing is driven at the same surface speed and is the same diameter as the printing cylinders that are located ahead of the cutting dies. Cam *B* is stationary, being mounted on a fixed portion of the machine by means of bracket *C*. An arcuate slot in the bracket permits adjustment of the cam position.

Scrap pieces are picked out by a hook attached to follower arm *D*. This arm pivots on shaft *E* which fits into a bearing hole bored through an integral housing lug. On the other end of the arm is a cam-follower *F*. Because arm *D* and its associated parts rotate with the cylindrical housing, tension springs *G* are necessary to overcome the centrifugal force developed during normal operation and maintain the cam-followers in contact with the cam surface. A clearance opening *H* is provided for each hook.

It is apparent from the lay-out of the carton blanks on the web that there is a periodic repetition of the scrap pieces and, therefore, that they will always contact the same area on housing *A*. Because of this, one or more hooks are located within each contact area.

During normal operation, follower *G* rides up on the cam lobe as the leading edge of the scrap piece contacts the housing. Follower arm *D* is raised just enough to allow the hook to puncture the scrap piece and remain there while the follower is in contact with the cam lobe. As the web leaves housing *A* in a tangential path, the scrap material is retained by the curved hooks. This division in paths of travel causes the tabs to break, thus effecting a separation between the carton blanks and the unwanted trimmings. After the cam-followers leave the cam lobe, the hooks are withdrawn, and the scrap is free to fall into disposal cans.

Single Closed-Track Cam Drives Glue-Transfer Mechanism

One station of a large machine for packaging material in paper sacks is devoted to the application of glue before the final fold is made to seal the container. An interesting mechanical arrangement is employed to accomplish the various motions required.

At this station of the machine, the package carrier *A* is brought to a momentary halt (see Fig. 11). During this interval, an applicator *B* (carried by slide *C*) must pick up glue from cylinder *D*, turn through an arc of 180 degrees (by means of a stationary cam, not shown), and move to the package fold to deposit a strip of the adhesive.

All movements necessary to advance and retract the applicator slide, rotate the glue cylinder between packages, and synchronize these functions are furnished basically by just one closed-track cam *E*. Follower-roller *F* is carried by follower-lever *G*, which is integral with short lever *H*. The roller rides in cam track *J*.

Long lever *K* pivots freely about the same shaft used by *G* and *H*. However, it is coupled to them as shown in the auxiliary

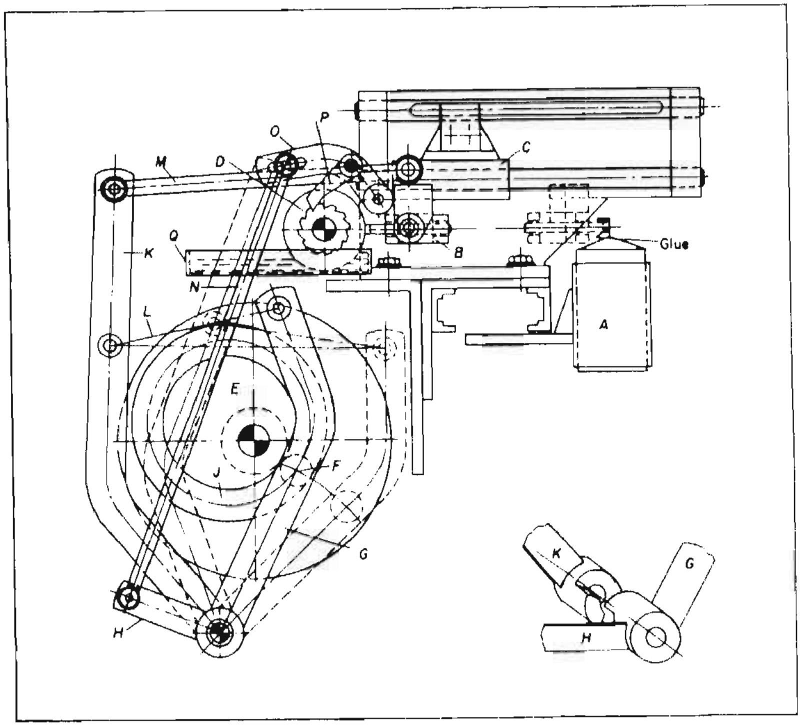


FIG. 11. All movements necessary to the proper functioning of this glue-transfer mechanism are under the control of one closed-track cam.

sketch so that there is over 90 degrees of play between them. The coupling is designed so that lever G will drive lever K to the left only. G and K are shown in their closest position.

As the cam rotates, follower-roller F is forced to the right. Because the coupling does not provide positive drive in this direction, a spring L, attached to levers G and K, causes the latter to follow the movement of the former.

In this way, both desired end motions are achieved. First, lever K, through connecting-rod M, advances and retracts slide C and, in turn, applicator B. Second, lever H, through connecting-rod N, raises and lowers the pivoting support arm O that permits ratchet and pawl P to function, thus rotating the glue cylinder in glue tray Q during each reciprocation of slide C. On

the return movement of lever G the coupling engages and lever K is forced back to its original position under direct drive.

The cam is designed to yield a 180-degree rise and a 180-degree return motion. Although this high-speed mechanism is capable of transferring large forces, its capacity can be increased by cutting a close-tolerance cam groove and by replacing the line-contact follower-roller with a plane-contact sliding piece.

Indexing Attachment that Controls Ratchet Operation

On a machine for producing ornamental wire screening, it is necessary for the movement of the screening through the machine to be interrupted at certain times, depending on the screen pattern. During the idle period of the feeding mechanism, work is performed on the screening. In its movements, the screening is fed by a ratchet mechanism. How this mechanism was designed is shown in Fig. 12.

Drive-shaft A carries a worm C and disc B which rotate with the shaft in a clockwise direction. Disc B is provided with a slide-bar M which operates in a groove against light frictional resistance that is provided by two springs. The ends of bar M are shaped to serve as cam surfaces, and contact roller L which is carried on follower bar K. This latter bar is provided with a return spring, not shown.

Worm C meshes with worm-gear D on shaft E. This shaft also carries disc F, which has twelve holes into which cylindrical buttons are pressed, in various positions, as required by the specified screen pattern. These buttons periodically actuate the swinging lever G as disc F revolves. The worm-gear has twelve teeth, so that there is one rotation of shaft E to twelve turns of shaft A. A cylindrical plunger H mounted in block N carries a roller I at its upper end. A pin extends through vertical slots in plunger H and the wall of block N, and into a horizontal slot in lever G. This lever is mounted to swivel freely on a stud at its left-hand end.

In Fig. 12 slide-bar M is in a position to permit roller L to pass freely between disc B and the contoured offset end of bar M. At

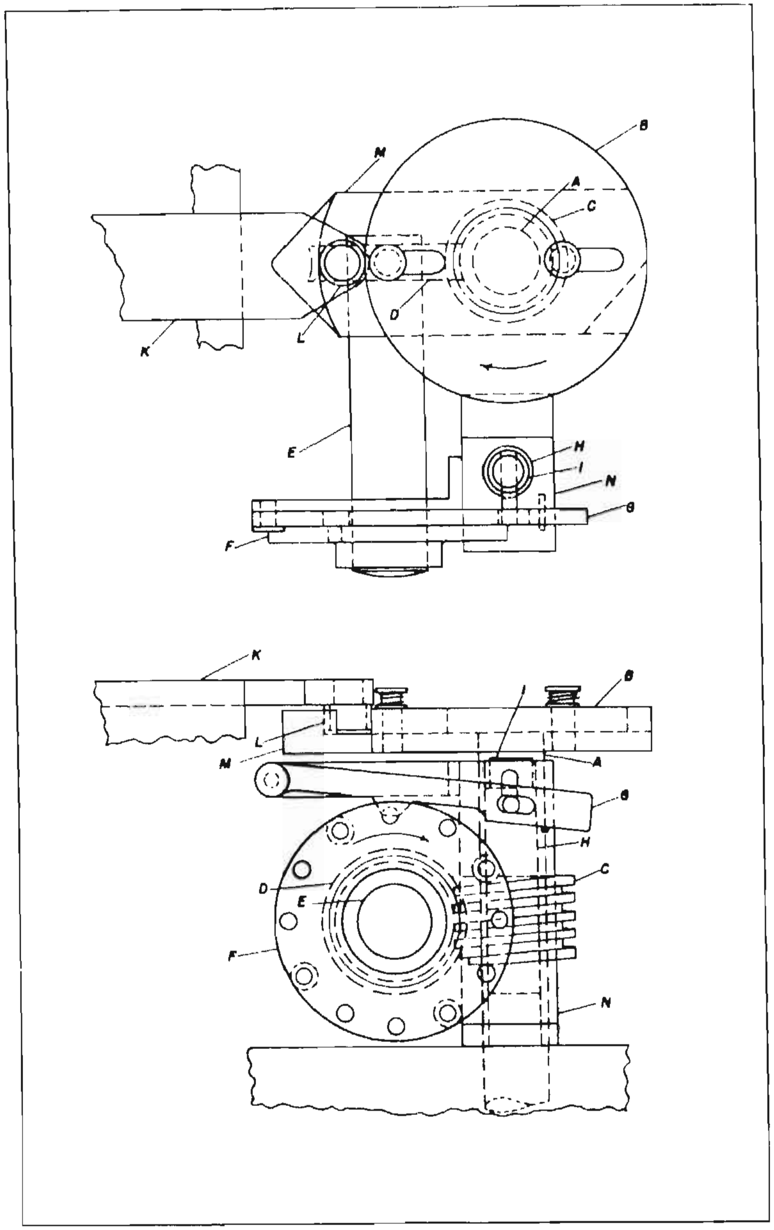


FIG. 12. Mechanism designed to periodically interrupt a feed movement as required for an operation.

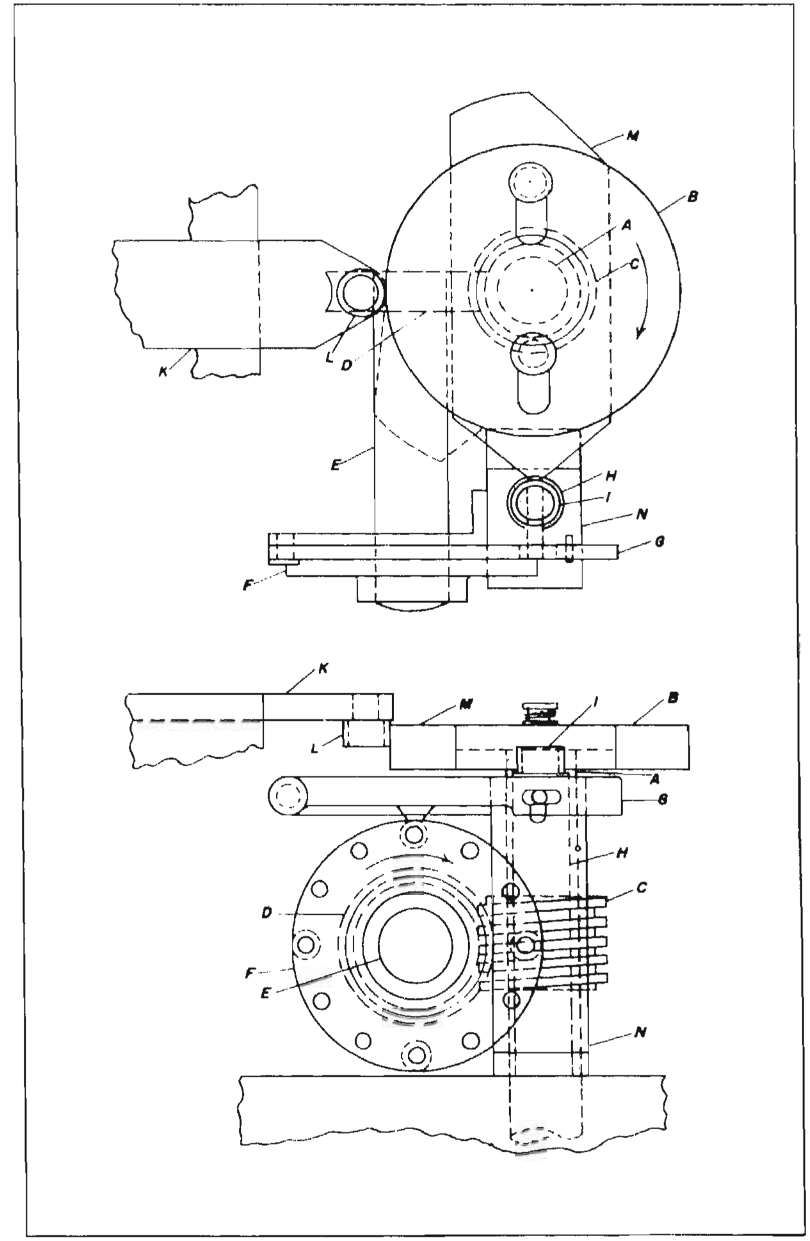


FIG. 13. Position of the various components of the mechanism when screening is being fed.

this point, motion is not transmitted to follower bar K, as roller L remains in contact with the periphery of disc B. The contact lug on the underside of lever G lies between two of the buttons on disc F, and cylinder H is in its lower position. The cylinder remains in this position until one of the buttons on disc F contacts the lug on lever G. Bar M remains in this relative position with disc B as long as cylinder H remains in its lower position, the roller L passing through the opening between the contoured end of bar M and the periphery of disc B.

In Fig. 13, shaft E has rotated sufficiently to cause one of the buttons on disc F to raise cylinder H, through lever G. At this point, roller H lies in the path of the oncoming cam surface on the end of bar M. Continued rotation of disc B causes bar M to be moved so that the offset angular end of bar M coincides with the periphery of disc B, causing the opposite end of bar M to protrude.

As the angular end of bar M passes roller L it causes the follower bar to be moved, thus actuating the ratchet mechanism and feeding the screening to its next position. Continued rotation of disc F causes the button to lose contact with the lug on lever G, and cylinder H again falls to its low position. Further rotation of disc B will bring the protruding end of bar M into contact with roller L, but the spring tension on bar K is sufficient to overcome the frictional resistance of bar M so that it is returned to the position shown in Fig. 12. Bar M remains in this position until cylinder H rises to return it to the position shown in Fig. 13.

Gear Mechanism for Varying Cam Timing

On a machine that manufactures a woven-wire product, it was necessary to provide for varied spacing in the weave. This was accomplished by means of a cam-actuated mechanism which is shown in Fig. 14. The cycle of the mechanism is controlled by shaft A to which gear B is keyed. Gear C is fixed to the hub of cam D which rotates freely on the shaft and is retained by collar

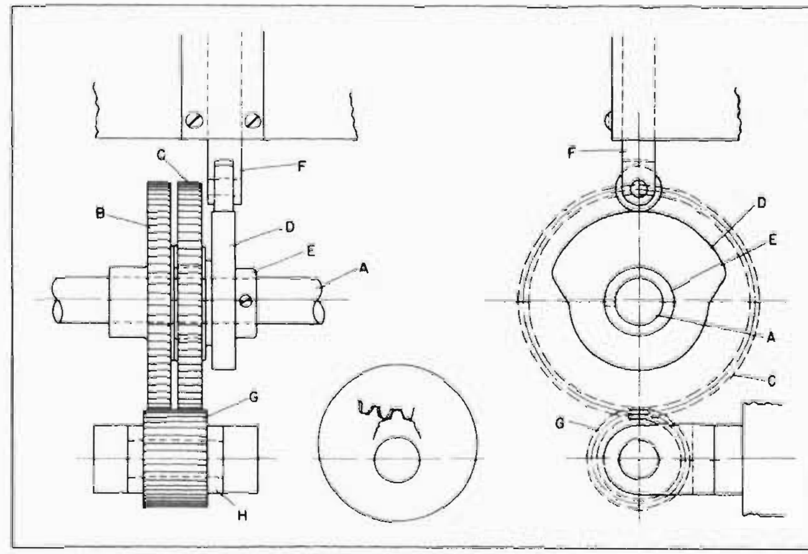


FIG. 14. A cam action which produces varied spacing in a woven product.

E. Cam D operates the follower bar F which actuates the spacing mechanism. Pinion G is supported to rotate freely on bearing bracket H and mesh with gears B and C. Gear B is of standard pitch and has fifty teeth. Gear C has fifty-one teeth but the same outside and pitch diameters as gear B. The additional tooth, however, decreases the circular pitch since the width of the teeth must be narrower than standard.

In operation, shaft A rotates gear B, the motion being transmitted to gear C through pinion G. Since gear C rotates slower than gear B, due to the difference in the number of teeth, cam D, actuated by gear C, will turn less than one revolution in relation to gear B. In the situation described, the loss in radial movement would be approximately 7 degrees, and shaft A would require fifty-one turns to produce a complete timing cycle of cam E. The modification of gear C is restricted by its relation to pinion G. Since the teeth of gears B and C must remain in alignment as they mesh with the pinion gear, the reduction in the thickness of the teeth of gear C must be sufficient to prevent any binding action.

Cam Eliminates Shock in Rack Movement

An arrangement that imparts a rapid intermittent and reciprocating movement to a rack employed in an aluminum foil bag-making machine is shown in Fig. 15. The interesting feature of the mechanism is a cam which slows the motion of a drive-pin *A* each time it is about to contact a disc and move the load. This deceleration prevents undue impact loading and possible breakage of the pin. Since the load, rack *B*, and its drive were added subsequent to the construction of the machine, limited space

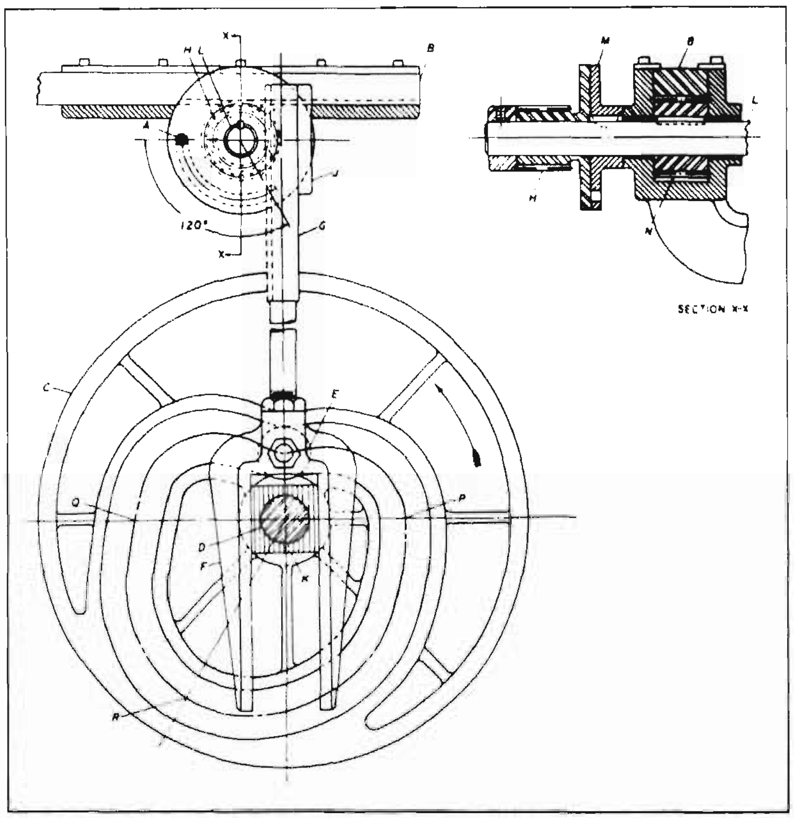


FIG. 15. Arrangement that incorporates a cam to prevent excessive shock in an intermittent, reciprocating rack movement.

and the position of existing members were major considerations governing design of the device.

A face-cam *C* is keyed to a drive-shaft *D* which existed prior to the modification of the machine. Roller follower *E* is free to rotate on a stud secured to a slider guide *F*. One end of the slider guide is turned and threaded for assembly to a rack *G*. A lock-nut secures the assembly in the proper position. Rack *G* is restrained to a vertical linear motion by a pinion *H*, guide *J*, and a stationary slider *K*, the drive-shaft *D* having a running fit in the slider. The pinion *H* is in mesh with rack *G*, and is free to rotate on a shaft *L*.

A disc integral with pinion *H* carries drive-pin *A* which is located in a 120-degree circular slot in an adjacent disc *M*. Disc *M* and a pinion *N*, which is in mesh with the guided horizontal rack *B*, are keyed to shaft *L*. Thus, if pinion *H* is rotated counterclockwise, pin *A* will traverse the circular slot and contact disc *M*. Further motion in the same direction will be transmitted through disc *M*, shaft *L*, and pinion *N* to drive rack *B*. Reversal of rotation of pinion *H* will again cause the drive-pin to traverse the circular slot before contacting disc *M* to drive the load in the other direction.

The rotational speed of pin *A* at impact with disc *M* is controlled by the path of the slot in cam *C*. If the rotation is not reduced to a minimum at the moment of contact, the resulting additional impact loading could cause breakage of the pin. Minimum impact is obtained by making the incremental radial rise of the cam-slot as small as possible in the areas of points *P* and *Q* which are traversed by the follower at the two moments of contact. In addition, cam *C* is designed to impart the movement necessary for rack *B* to complete its function in the machine.

In operation, cam *C* is rotated counterclockwise. The rise imparted to the vertical rack during the first 90 degrees of rotation of the cam traverses pin *A* through the 120-degree slot in disc *M*, rack *B* remaining stationary. When the center of the follower coincides with point *P* on the cam, pin *A* is in contact with disc *M*. For the next 120 degrees of cam rotation pin *A* moves to the horizontal rack to the left. Reversal in the rotation of the pin occurs when point *R* of the cam coincides with the center of the follower. Again, rack *B* is stationary as the pin traverses the slot in disc *M* clockwise during the following 60-degree movement of the cam. Pin *A* contacts the disc when the follower is at point *Q* and returns rack *B* to its initial position in the final 90 degrees of cam rotation. At both points *P* and *Q* the rate of rise or fall of the follower is at a minimum to prevent impact damage to the pin.

Wheel-Dressing Attachment for a Gear-Grinding Screw

A patented attachment for dressing a continuous helical rib on the abrasive wheel of a gear-grinding machine is shown in the accompanying drawing. The attachment, see Fig. 16, is mounted on the compound slide of the machine. During dressing, it is traversed parallel with the axis of the wheel-spindle by a screw driven from the spindle through pick-off gears.

Dressing is carried out simultaneously on both flanks of the helical rib by separate diamonds mounted in holders *A* and *B*. The holders are set to correspond with the pressure angle of the gear to be ground. At the end of the dressing stroke, the cross-slide and attachment are moved away from the spindle by hand, so that the diamonds clear the wheel.

Then, the longitudinal slide is returned to the starting position by power traverse. Next, the cross-slide is brought to the dressing position, and the diamond holders are adjusted lengthwise by means of screws *C* and *D* for applying another cut. In this way, the rib on the grinding wheel is dressed to its full depth in several passes.

The housing for the left-hand diamond holder *A* is fixed to base *E*, which is secured to the compound slide of the grinder. A slide *F* is provided for the right-hand diamond holder *B*. With this arrangement, the right-hand holder can be moved toward or away from the left-hand holder by means of a screw, for setting the distance between the diamonds in accordance with the required width of the helical rib to be dressed.

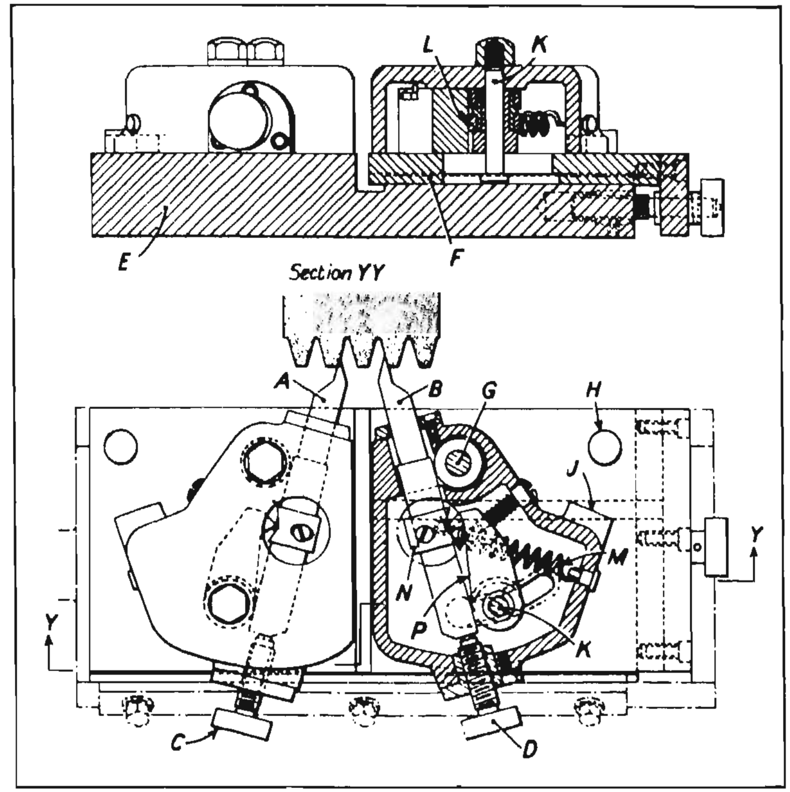


FIG. 16. Attachment for diamond dressing a continuous helical rib on the abrasive wheel of a gear-grinding machine. By changing cam *N*, the wheel may be dressed to various profiles.

Since both diamond-holder assemblies are of similar design, only the right-hand unit will be described. This unit can be pivoted about pin *G*. The diamond holder *B* is set to the required angle by means of gage-blocks, which are placed between the periphery of reference pin *H* and the machined surface *J*. The unit is then secured to slide *F* by nuts on the threaded upper ends of pins *G* and *K*. The enlarged-diameter lower end of pin *K* engages a T-slot in the slide.

An eccentric ring *L* is keyed to a sleeve surrounding the pin *K*. Diamond holder *B* is held in close contact with this ring and a sleeve surrounding pin *G* by a tension spring *M*. When a wheel for grinding gear teeth of modified involute form is to be dressed, a plate type cam N is attached to the diamond holder. This cam is engaged by a follower-arm P, which is also keyed to the sleeve surrounding pin K.

When the diamond holder is adjusted lengthwise at the end of each dressing stroke, the action between the cam and follower-arm causes the eccentric ring L to be rotated through a small angle. As a result, the diamond holder is set in different angular positions during the dressing operation so that a rib with curved flanks is formed on the wheel. By using cams of different shapes, the wheel may be dressed to grind gears having modified profiles over part or full tooth depth.

CHAPTER 2

Intermittent Motions from Gears and Cams

The term "intermittent motion" is applied to mechanisms for obtaining a "dwell" or possibly a series of dwells or moving and stationary periods of equal or unequal lengths. Many different designs of intermittent motions are in use because they are required on so many different types of automatic and semi-automatic machines. The intermittent motions illustrated and described in this and the following chapter supplement those presented in Volumes I, II and III of "Ingenious Mechanisms for Designers and Inventors."

Intermittent Worm-Gear Train

The worm-gear drive shown in Fig. 1 was designed to provide intermittent motion. It consists of a worm-wheel A having teeth in sectors X, Y, and Z; worms B and C, mounted on and keyed to shaft D; and a plunger E. Worm B is firmly attached to the shaft; Worm C is free to slide. The helices of these worms should be continuous.

In order to hold worm C to the right, out of engagement with the worm-wheel, a spring F is located in a longitudinal hole in shaft D. A cross-pin G, extending through a slot in the shaft and engaging a seat in worm C, transmits the spring pressure to this worm. A similar seat is provided in worm B for this pin. The spring is suitably secured at its left end.

The device operates as follows: as shaft D rotates in the direction shown by the arrow, there will be no rotation of the worm-wheel A in the position illustrated, as worm B is rotating in a plain sector of the worm-gear. In order to produce rotation of

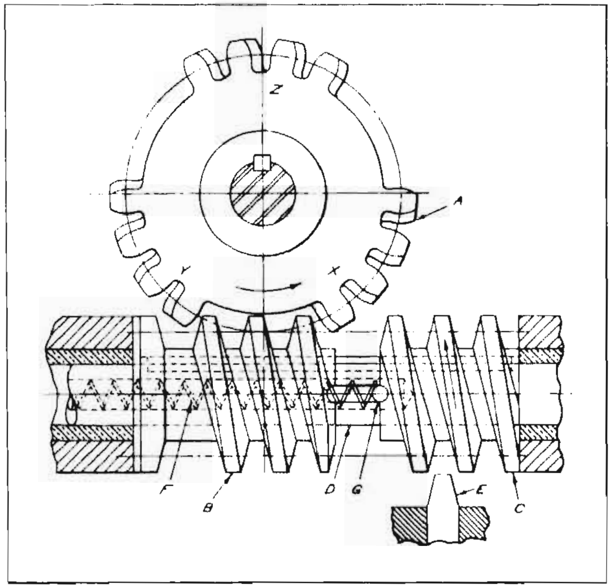


FIG. 1. Worm-gear train for producing intermittent motion. The worm-wheel A is rotated one-third of a revolution, followed by a stationary period, through the action of the two worm-gears B and C.

the worm gear — in this case one-third a complete revolution — plunger E (timed by other parts not shown) rises into engagement with the worm thread. Upon engagement, the rotating worm C moves to the left, as its thread moves past the plunger until the worm engages the last tooth in sector X of the worm-wheel. The plunger must be withdrawn before worm C reaches worm B.

When the end of worm C comes in contact with the end of worm B, the worm-wheel A is rotated in the direction of the arrow, both worms operating as a single unit. After worm-wheel A has rotated sufficiently to bring the last tooth of sector X out of engagement with worm C, spring F pushes the worm to the extreme right, where it cannot engage the teeth of sector Y. When the last tooth in sector Y is disengaged from worm B, worm-wheel A will stop, having made one-third of a revolution about its axis.

Indexing Movement that Starts without Shock

Most indexing mechanisms incorporate either cams, Geneva movements, or other components which present machining problems. An indexing mechanism made up of easily machined components and accurate gears that can be obtained from gear specialists is shown in Fig. 2.

The mechanism, illustrated in the top view of Fig. 2, is a planetary gear device incorporating two eccentrically located spur gears. The bore of each of these gears is machined off center by an amount equal to 20 per cent of its pitch radius. The desired dwell period is realized when the ratio of the number of turns of arm C to the number of turns of sun gear H is 3 to 1; that is, the arm must rotate three times faster than the sun gear, but in the opposite direction.

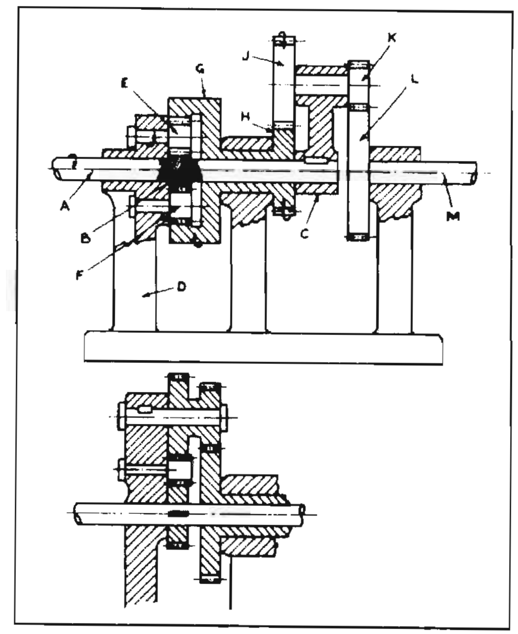


FIG. 2. (Top) Planetary gear type indexing device that provides fixed dwell periods in the movement of follower-shaft M. (Bottom) Alternate gear arrangement permits the elimination of internal gear G.

Drive-shaft A, which carries a 20-tooth pinion B, is keyed to arm C. Frame member D supports two pinions E and F, each having 20 teeth, which mesh with pinion B and also with internal gear G. This internal gear has 60 teeth and is integral with an eccentrically located, 40-tooth sun gear H. Gears G and H revolve around shaft A.

Meshing with the sun gear is another eccentrically located, 40-tooth gear J. Mounted on the same shaft with gear J is a 20-tooth pinion K. This gear meshes with a 60-tooth gear L that is mounted on follower-shaft M.

In operation, drive-shaft A turns clockwise as indicated by the arrow; pinions E and F, internal gear G, and sun gear H turn counterclockwise, while gear J and pinion K turn clockwise. Since gear L is driven in a counterclockwise direction, follower-shaft M receives this motion. One full revolution of the drive-shaft results in several fixed dwell periods in the follower-shaft rotation.

An adaptation of the gear train to the left of sun gear H is shown in the lower view of Fig. 2. In this alternate arrangement only external spur gears are used, thus eliminating internal gear G. The gear ratio, however, remains the same (3 to 1).

Chain Driven Intermittent Rotary Movement

On a wire fabricating machine, a driven shaft was to be given an intermittent rotary movement through a roller chain from a uniformly rotating driving shaft. Both shafts had to begin and complete each revolution together. However, because the driven shaft was to move intermittently, it had to rotate at a higher speed than the driving shaft. The drawing illustrates the mechanism that was designed to obtain the required motion.

Keyed to the driving shaft A, (see Fig. 3), are a sprocket B and a cam C. A bracket L supports a lever J. At its lower end, lever J carries a follower roller K which contacts the cam. The upper end of the lever is joined by a link M to a slide-bar F dovetailed to the frame of the machine. Four idler sprockets D are mounted on the machine, and two idler sprockets E are car-

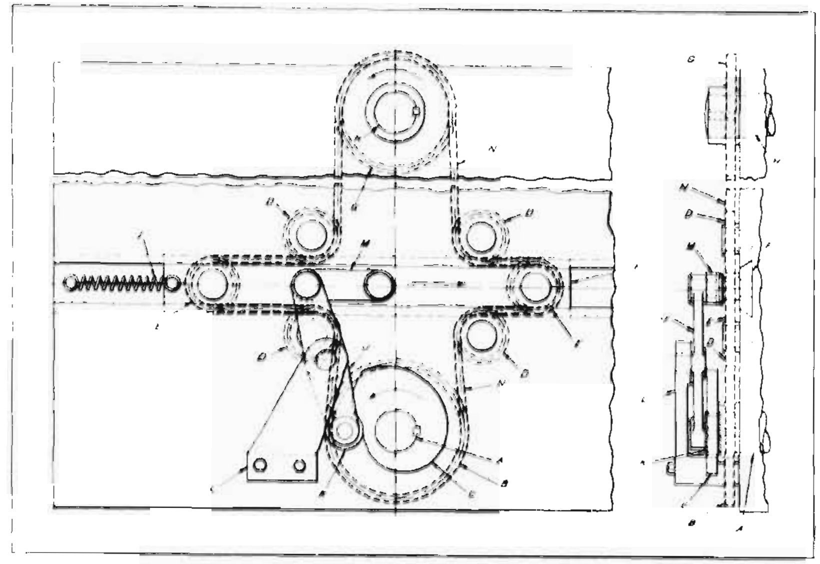


FIG. 3. Practical design for producing intermittent rotation in a driven shaft H from a uniformly rotating driving shaft A.

ried on the slide-bar. A spring I serves to resist any movement of the slide-bar to the right, and maintains the follower roller in contact with the cam. The driven shaft H has a sprocket G keyed to it which, like the sprocket on the driving shaft, is in mesh with a chain N.

In order to explain the operation of the mechanism, let it be assumed that the cam and lever were omitted. The rotation of shafts A and H would then be in the ratio of the number of teeth on sprockets B and G, the idler sprockets D serving merely to direct the chain over the required path. The two idler sprockets E carried on the slide-bar do not affect the motion of sprocket G, provided the slide-bar remains stationary. But if there is a change in the position of the slide-bar, there will also be a change in the relative positions of sprockets B and G. This is because the movement of the slide-bar causes the chain to be let out on one side and taken up on the other side, this action producing a partial rotation of sprocket G.

By referring to the drawing, it can be seen that the cam rotates in the direction indicated by the arrow, and is about to make the lever swing on its fulcrum and move the slide-bar to the right. As a result of this action, the chain is let out on the left side and taken up on the right side. If the take-up speed is equal to the linear speed of the chain, no rotative motion will be transmitted to sprocket G while the slide-bar is in motion. The linear speed of the slide-bar must equal one-half the speed of the chain to produce this condition, because the chain is let out and taken up on both sides of sprockets E.

Intermittent Motion from Two Synchronized Cams

Packaging machines often require mechanisms to transmit a particular motion during each fifth revolution of the main camshaft. Such a need might arise where five packages are to be grouped, then pushed from the machine at the same time. A mechanism that has been arranged to satisfy these particular requirements is shown in Fig. 4.

The principal operating elements of this mechanism are two synchronized cams and one follower-lever. The upper end of single, L-shaped lever A drives the package-ejector unit (not shown). At the opposite end of the lever are two follower-rollers B_1 and B_2 which are held in contact with cams C_1 and C_2, respectively, by a spring D (attached to the upright lever arm).

Cam C_1 is pinned directly to the constantly rotating camshaft E, while the motion for cam C_2 is obtained indirectly from a gear F, also pinned to the camshaft. By means of gears G and H — the latter being keyed to cam C_2 — movement of gear F is reduced to one-fifth by the time it reaches the second cam. Thus, the speed of cam C_2 is only one-fifth that of the camshaft and C_1, although the rotational movement of both of the cams is in the same direction.

Bearing this in mind, and noting the cam configurations and positions in the right-hand view, it can be seen that in one revolution of cam C_1 cam C_2 will rotate a distance equal to the width of its cutout J. This cutout occupies approximately one-fifth of the otherwise circular cam.

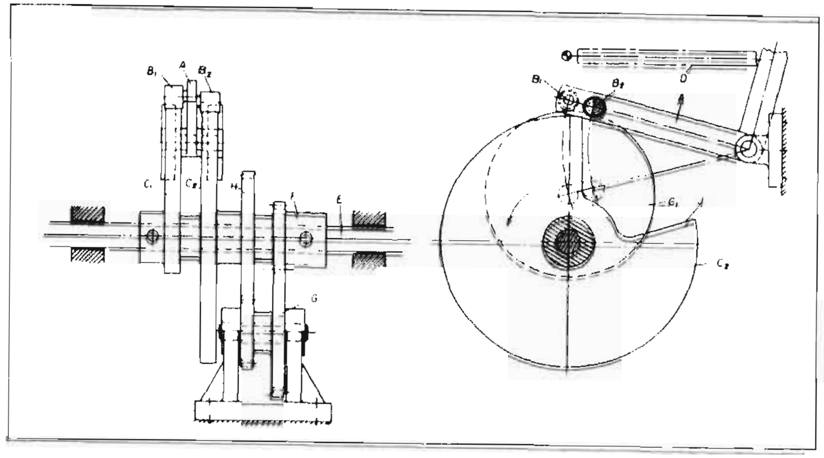


FIG. 4. Lever A is permitted to function only once for each five revolutions of camshaft E. This intermittent movement is controlled by the action of cams C_1 and C_2.

During this rotation of the camshaft, follower-roller B_2 is disengaged from the surface of cam C_2. This permits roller B_1 to track along the entire surface of cam C_1, thus causing lever A to pivot. For the next four rotations of the camshaft, follower-roller B_2 will ride along cam C_2, thereby preventing roller B_1 from being affected by the contour of cam C_1, and causing lever A to remain motionless.

Intermittent Rotary Movement with End-Cycle Reversal

On a machine for forming a product of flat wire, the material is fed intermittently through a twisting clamp. During the cycle, a shaft rotates one revolution, dwells, then rotates another revolution in the same direction. After several such revolutions, the shaft rotates in the reverse direction for a number of revolutions equal to the number of forward, separate revolutions, then stops to end the cycle.

In Fig. 5, tubular spindle shaft A, which carries the twisting clamp, is supported by bearing brackets B. Pinion gear C, brake-drum D, and counterweight drum E are keyed to shaft A. Friction is applied to the brake-drum by leather band F which is

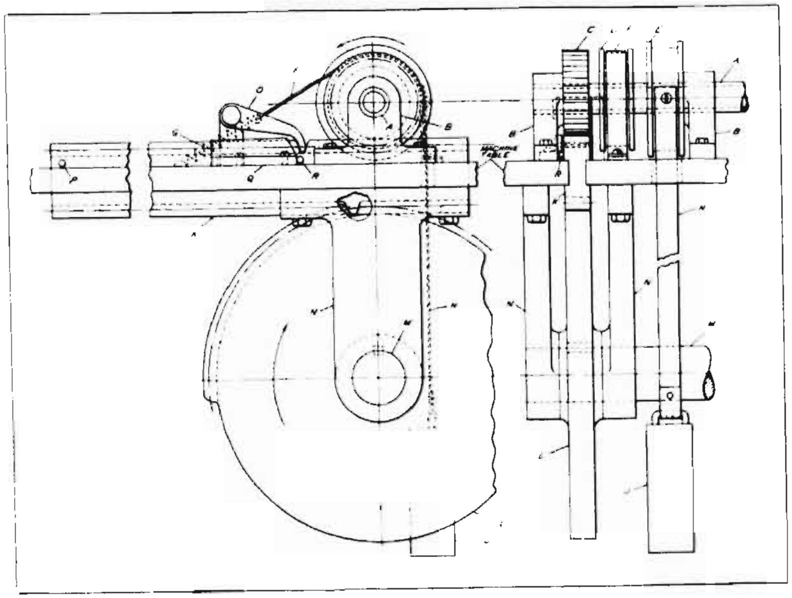


FIG. 5. Mechanism that provides intermittent rotary movement from a rotating drive-shaft, and which has provision for automatic reversal upon completion of operating cycle.

held under tension applied by spring G. The counterweight drum receives steel trap H, to which the counterweight J is fastened. This serves to return the tubular shaft A to its initial position at the end of each cycle. The purpose of the leather band is to control the speed with which the counterweight descends.

A rack K, with teeth cut on both upper and lower surfaces, meshes with pinion gear C and gear segment L. Gear segment L is keyed to the driveshaft M, which, in turn, is supported by bearing bracket N. The ratios of the pitch diameter of gears L and C, and consequently their speed ratios, are 4:1. The gear segment of L extends over 90 degrees and can rotate gear C one revolution.

The diagram shows the mechanism at the beginning of the cycle. Referring to the front elevation at the left, the first tooth in gear L, which is rotating in the direction indicated by the

arrow, is in contact with the gear teeth on the undei side of rack K. This causes the rack to move to the right, thereb*j* rotating gear C in a counter-clockwise direction. Drum E, being keyed to spindle shaft A also rotates, thus causing counterweight J to rise as steel strap H is wrapped around the drum. After the last tooth in gear L has disengaged the teeth in the rack, gear C stops turning and the rack is held in position at this point by pawl O. The pawl drops between two of the upper rack teeth.

The spindle shaft then remains stationary until the first tooth of gear L again contacts the teeth of the under side of rack K. This phase is repeated until pin P, attached to the rack, contacts sliding dog Q, moving it to the right. This raises pawl O out of contact with the teeth in rack K. The action of counterweight J then causes a reverse rotation of pinion gear C, which, in turn, drives the rack to the left. Pawl O remains raised until pin R contacts sliding dog Q, moving it to the left, and allowing the pawl to drop back into contact with rack K, thus completing the cycle.

Fool-Proof Indexing Mechanism

Necessity for the design of an indexing mechanism that will not overshoot the desired position or otherwise inaccurately index, is often encountered. This is especially true when a weighty fixture is to be carried by a large index-table. A lever type indexing mechanism that is fool-proof in operation is shown in Fig. 6.

Lever A pivots about a spindle that is also common to both ratchet wheel B and spring-loaded dog C. The dog remains engaged in a slot milled across a boss integral with the lever. The ratchet-wheel teeth are cut to mesh with mating spaces in the periphery of index-plate D. Pawl E, resting against a flat spring, restricts the motion of the ratchet wheel to a clockwise direction only. The back end of the pawl rests against an adjustable cam-stop.

Slotted lever F pivots freely on the common spindle and is connected to locating plunger G by a headed pin. The locating

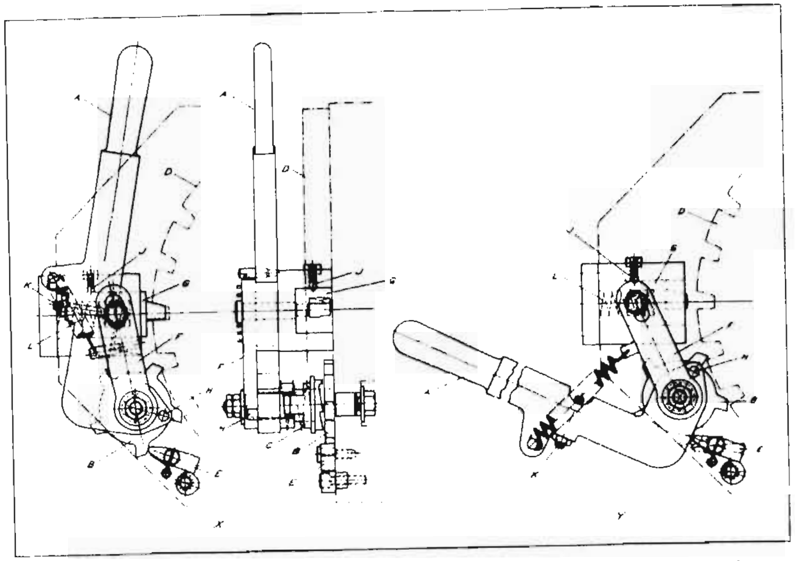


FIG. 6. Index-plate mechanism incorporates locating device and one-directional driving dog to provide fool-proof operation by eliminating the possibility of faulty indexing.

plunger slides in a guide block fastened to the base of the indexing table. A locating tongue on the plunger end is accurately ground to fit within the tooth spaces around index-plate *D*. This tongue is partially relieved on the face parallel to the horizontal center line to clear any burrs that may have been raised in the tooth spaces on index-plate *D* by the action of ratchet wheel *B*.

To move the index-plate from one position to the next, lever *A* is moved from the position shown at X to that shown at Y. During the initial part of this motion, pin *H*, pressed in the short leg of the lever, moves freely in an arc until it contacts slotted lever *F*. The pin, during the remainder of the stroke, forces the slotted lever to pivot around the common spindle, thereby disengaging locating plunger *G* from the index-plate. At the end of the stroke, spring-loaded ball *J* rides into a cone-shaped recess in the top plunger face. The plunger is thus held in the retracted position.

At the same time that these movements are taking place, spring-loaded dog *C* freewheels over the dog teeth that are integral with ratchet wheel *B* and drops into position after having moved a distance equal to one dog tooth. The ratchet wheel, which has the same number of dog teeth as ratchet teeth, is prevented from turning by pawl *E*.

Lever *A* is now moved back to its original position. In doing so, dog *C* drives ratchet wheel *B* one tooth which, in turn, moves the index-plate to the next position. During the first part of this stroke, spring-loaded ball *J* retains locating plunger *G*, allowing the index-plate to move unrestricted. Toward the end of the stroke an adjustable screw *K*, which is threaded through a pad on the lever, contacts the pin connecting slotted lever *F* with the plunger. This drives the plunger forward causing the locating tongue to enter a tooth slot on the index plate, thus locking it firmly in place. The plunger is held in this position by coil spring *L*.

Escapement Provides Regular Intermittent Drive

An escapement mechanism in which a pendulum is applied to control the timing of an intermittently rotating shaft was incorporated in a wire weaving machine to advance strands of wire a required distance at regulated time intervals. It resembles the pin-pallet, or Brocot, escapements used in French and pendulum clocks.

Drive-shaft *A*, see Fig. 7, has gear *B* keyed to it. This gear meshes with gear *C*, which is carried free on driven shaft *E*. Gear *C* carries a series of spring-loaded plungers which contact disc *D*, keyed to shaft *E*, see Fig. 8. Shaft *E* carries, at its outer end, a disc *G*, which is provided with a series of pins. A ring of friction material *F* is carried on the hub of disc *G* where it transmits rotary motion from gear *C* to disc *G* due to the pressure applied by the spring plungers. Bracket *H*, bolted to a stationary part of the machine, carries a pivot stud which supports a pallet *I* and a pendulum *J*, which are locked together by two screws in

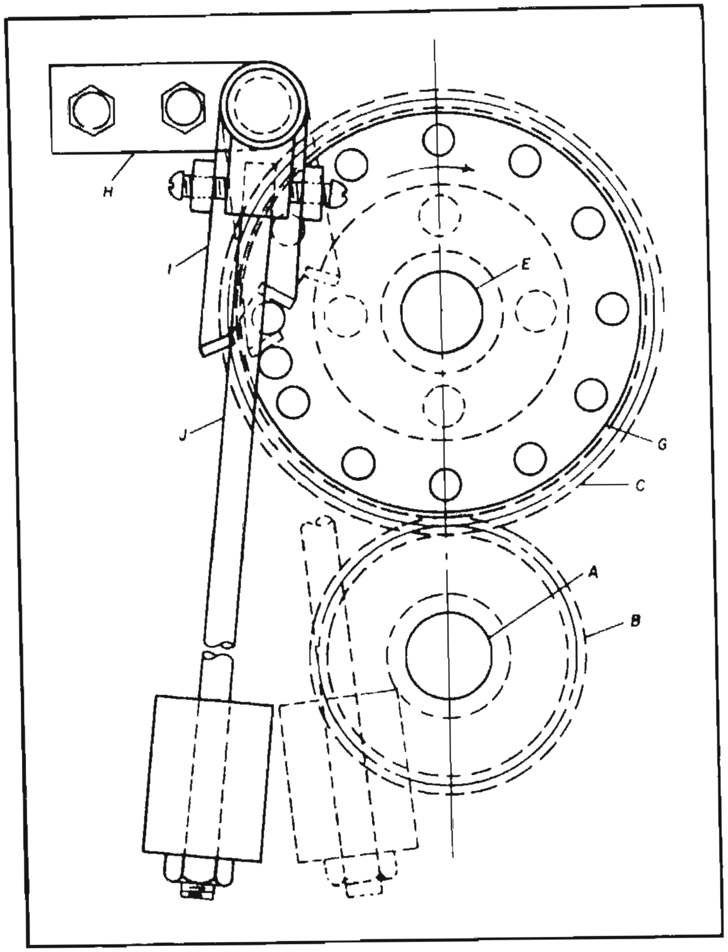


Fig. 7. Pendulum escapement gives disc G intermittent clockwise motion by means of the pin pallet I alternately releasing the pins in the disc as pendulum J swings from side to side.

the side plates attached to pallet I. These screws provide a means of adjusting the position of the pallet relative to the pendulum.

In operation, the continuously rotating shaft A transmits motion to the shaft E through gears B and C, friction ring F, and disc G, where there is no restriction to the movement of disc

G. In Fig. 7, the positions of pendulum J and pallet I are such that the upper foot of pallet I contacting a pin of disc G prevents its rotation. As pendulum J swings to the other end of its arc, as shown dotted, the contacting pallet foot slides off the pin, allowing disc G to rotate. But before the upper foot has completely lost contact with its pin, the lower pallet foot carries into position to catch the succeeding pin. Thus, at the end of the arc

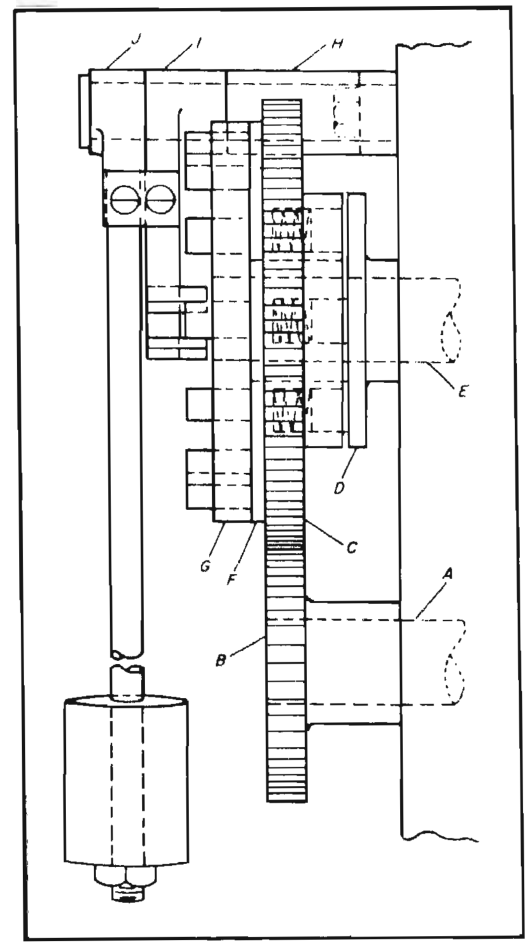


Fig. 8. Driven shaft E is turned intermittently under control of pendulum escapement, receiving power from shaft A.

to the right the pallet I will be in the position shown by the dotted outline, the succeeding pin having moved to contact the lower foot of pallet I. If the pallet feet are correctly located, the angular movement of shaft E will equal one-half the angular spacing between any two consecutive pins on disc G. Therefore, the number of angular movements of shaft E per revolution will equal twice the number of pins in disc G.

The number of movements per minute of shaft E is controlled by the swing of the pendulum regardless of the number of pins on disc G. But in all cases the rotative speed of disc G must be such that the time required for any pin to arrive at the locking position must be less than the time required for the pendulum to swing. The angularity of the feet of pallet I relative to the line of movement of the pins provides an impulse to produce continued motion of pendulum J. However, this angularity is regulated by operating conditions. If the angle is too small, there will be insufficient impulse applied to the pendulum to keep it swinging. But if the angle is too large, free movement of the pendulum will be restricted, particularly when a heavy load is applied by the pins. With a light load and a large angle, there will be a noticeable jerk of disc G as the pallet feet slide off the pins. If this is objectionable, the contact surfaces of the pallet feet must be curved, with the center of the pivot as the center of the arc of curvature. In the latter case, an angle on the leading side of the pallet feet must deliver the impulse. In general, it is advisable to locate the pallet feet at the lowest angle with the line of movement of the pins which provides the necessary impulse.

To determine the length of the pendulum needed to produce the required timing, the formulas applied to a free-swinging weight suspended on a length of cord are applicable. For determining the time of swing, the accepted formula is

$$t = \pi \left(\frac{l}{g} \right)^{\frac{1}{2}}$$

in which: $t =$ time, in seconds; $l =$ length of pendulum in feet; and $g =$ the force of gravity in feet per second^2. The generally applied value of g is 32.2.

To determine the length of pendulum required to produce a specified rate of swing, the foregoing formula is transposed, thus:

$$l = g \left(\frac{t}{\pi} \right)^2$$

In the formulas, the symbol l is the distance from the pivot point to the center of the suspended weight. However, this formula is based on a weight suspended on a cord in which there is no friction influence and negligible weight of the cord. On the other hand, an accurate determination of the length of pendulum by this formula for the escapement application is not accurate because the combined weight of the rod and its fastening produce a distribution of weight which is not easily pinpointed. For all practical purposes, the calculated length is determined by the center of the suspended weight which is provided with a nut for adjustment, as shown. The pound value of the suspended weight in no way controls the timing of the swing. Its value lies in increasing momentum and providing the steadying influence of inertia when the impulse is applied. The lightest weight which will serve this purpose is recommended.

Rotary Work-Table with Mechanism for Automatic Indexing

An indexing work-table that can be used in conjunction with independent cutter-heads to form an automatic multiple-spindle machine is shown in Fig. 9. The table is intended to receive several work-holding fixtures according to the number of indexing stations provided. A variety of machining operations may be performed automatically while the work-pieces are located at these stations.

Referring to sectional view W-W, annular table A rotates on steel balls which surround fixed central disc B. Indexing is carried out by means of gear segment C (section X-X) which is secured to spindle D. The latter component is mounted in ball bearings which are housed in disc B and in the base.

Motion from gear segment C is transmitted by pinion E, which engages gear teeth in the bore of the table. The indexing action is controlled by compound cam F (section W-W) which

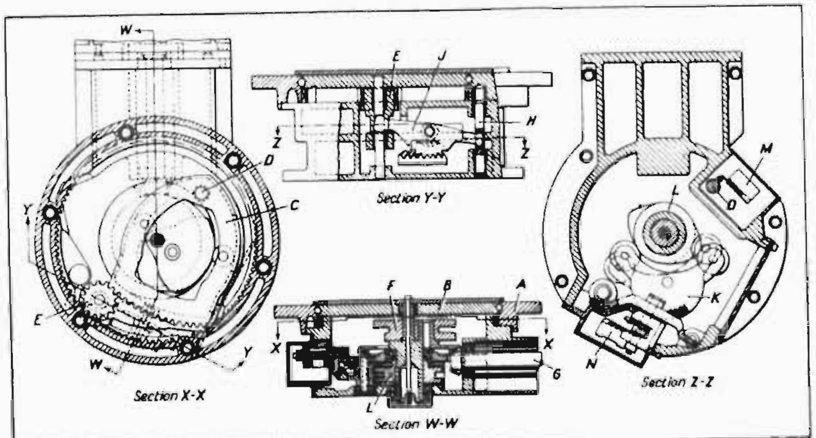


FIG. 9. Sectional views of work-table that can be set up for automatic indexing. Limit switch M stops indexing cycle and switch N starts machining cycle.

engages follower rollers housed in recesses in the gear segment C and is driven through bevel gears by shaft G. This shaft is driven by a motor, through V-belts, an electromagnetic clutch, and brake units. This driving equipment is not shown in the illustration.

At the beginning of the cycle, segment C dwells for a period, and the indexing motion is then completed during a 210-degree angular movement of cam F. Subsequently, segment C is again caused to dwell before it is returned to its original position. This is done in preparation for the next indexing cycle in the course of the final 90-degree angular movement of cam F.

During the dwell periods of the segment C before and after the indexing movement, pinion E and plunger H (section Y-Y) are moved vertically in opposite directions by lever J. At the beginning of the cycle, the plunger is withdrawn from one of a number of holes provided in the under side of the table at the indexing positions. Simultaneously pinion E is brought into engagement with the gear teeth in the table for the indexing movement. After indexing has been completed, the plunger is inserted into the next hole in the table. The latter is, therefore, positively located while the machining operations are being carried out on the work-piece. At the same time, the pinion is withdrawn from the gear teeth in the table in preparation for the return movement of the segment. At certain points in the cycle, the pinion and the plunger are in simultaneous engagement with the table, so that the latter is positively located during the entire indexing operation.

Movement is transmitted to the pinion and the plunger by bevel gear teeth on lever J and on pivoted segment K (section Z-Z). The segment carries two follower rollers which engage simultaneously with compound L. This cam is keyed to the lower end of the shaft which carries cam F. The arrangement may be seen in section W-W of the illustration.

The indexing cycle is started by means of a switch (not shown) which activates the electromagnetic clutch to engage the drive with shaft G. At the end of the indexing cycle, limit switch M (section Z-Z) is operated by means of a detent on the lower end of spindle D, with the result that the clutch and consequently the drive to the shaft G are disengaged. Concurrently, an arm attached to the pivot spindle for the lever J actuates the limit switch N to start the cycle of the cutterheads.

Intermittent and Pressure Applying Mechanism

The gluing of paper watch dials to their metal backings originally required the use of four presses and four operators. In order to reduce labor and equipment costs, a gluing device incorporating an ingenious operating mechanism was developed to do this work. The new device required only one operator, eliminated scrap, and enabled the work to be done with greater safety.

In designing this device, it was necessary to incorporate means for holding and pressing the paper dials and their metal backings together for a sufficient period of time to allow the glue to set. This requirement was met by providing the fixture with eight cam-operated pressing spindles mounted in an intermittently indexed eight-position spindle-carrier, with one position reserved for loading and unloading the work. The spindle-carrier is operated slowly enough to permit proper setting of the glue in one complete revolution of the carrier. A glued dial and its backing is removed from the loading and unloading station designated "0" and replaced by new pieces during the dwell period following each indexing of the spindle-carrier. Thus eight glued dials are completed in one revolution of the spindle-carrier, each dial and its backing being under pressure during one revolution of the carrier.

The essential features of the mechanism designed to operate the gluing device are shown in Fig. 10. The mechanism is driven by a motor through a belt passing over the friction pulley *H* and a clutch operated by lever *J*. When the clutch is engaged, worm *K* on shaft *F* turns the worm-wheel *L*. The upper portion of the worm-wheel carries the indexing pin of a Geneva mechanism. At each revolution of the worm-wheel, the indexing disc *M*, which is fastened rigidly to vertical shaft *N*, is rotated one-eighth revolution. Since the spindle-carrier *A* is keyed to the vertical shaft, it also rotates.

The eight equally spaced pressure spindles *C* are mounted in the carrier *A* as shown. Directly above, and in axial alignment with each pressure spindle *C*, is a spring-backed spindle *B*. It is between the work supporting disc *P* on the upper end of spindle *C* and the disc at the end of spindle *B* that the watch dial and its backing are pressed together to complete the gluing operation. Both spindles *C* and *B* are provided with keys that slide in keyways to prevent them from turning in the carrier *A*. Slots at the bottom ends of spindles *C* accommodate rollers, which are in constant contact with a circular cam-ring *D*.

The cam-ring has a raised portion throughout 246 degrees of its circumference, sloping portions throughout 47 and 44 degrees, and a flat portion throughout 23 degrees, which is located between the sloping portions. The flat portion at the position marked "0" is directly in front of the operator, and it is at this location that a pair of spindles is loaded and unloaded.

Bed *E*, besides acting as a support for cam-ring *D*, houses the Geneva motion and worm-wheel and serves as a support bracket for the worm-shaft *F*. The driving pulley *G* engages the spring-

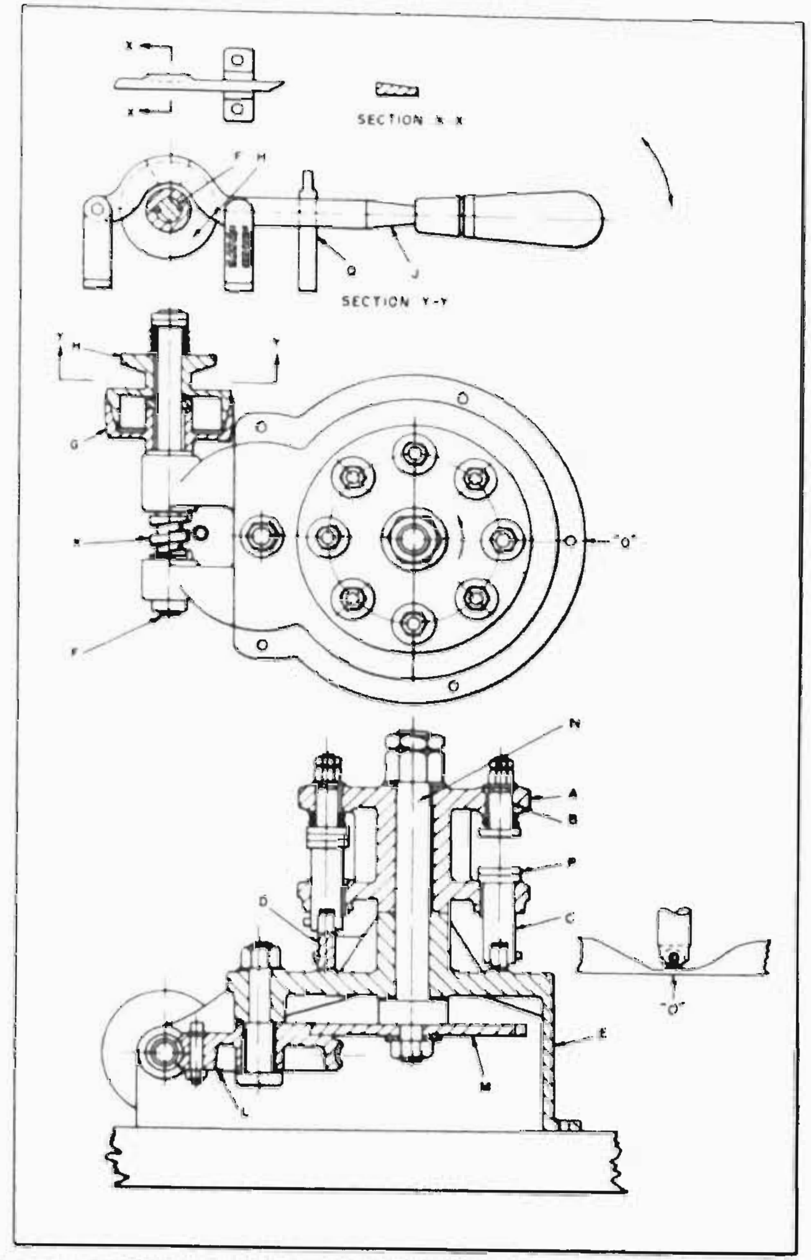


FIG. 10. Mechanism used to glue paper dials to metal discs by applying sufficient pressure for a predetermined length of time.

loaded friction pulley H when the clutch lever J is in the raised position, imparting rotation to shaft F, worm K, and worm-wheel L.

In operation, the parts to be glued are placed by the operator on the disc P of the spindle that is in the "0" position. The correct positioning of the paper dial and metal backing on disc P is facilitated by means of locating pins (not shown). The operator then releases the flat-hooked spring Q, which allows spring-loaded clutch lever J to move upward. This, in turn, releases driven pulley H and engages the train of members that drive the spindle-carrier A.

As the spindle-carrier rotates, the pressure spindle C is moved upward by the action of its roller on the circular cam-ring D. The pressure exerted on this member by the spring-loaded spindle B is sufficient, both in magnitude and duration, to permit setting of the glue. As one spindle is loaded and moved into the position where pressure is exerted, the pressure on the spindle immediately following is relieved. Its roller then follows the descent in the cam-ring under the influence of gravity in moving into position "0." Once in this position, the bottom spindle is quickly unloaded and reloaded.

The mechanism is stopped by depressing lever J, which disengages the driving pulley G. The clutch lever also acts as a friction brake. It is locked in the depressed position by means of flat hooked spring Q.

CHAPTER 3

Intermittent Motions from Ratchet and Geneva Mechanisms

Two methods of producing intermittent motion in which the periods of rest are evenly spaced and of equal length are by means of ratchet gearing and by using some modification of the Geneva motion. In its basic form this motion is obtained by means of a Geneva wheel, acting as a driven member, which has four radial slots located 90 degrees apart that successively engage a roller or pin on the driving member. The Geneva wheel thus turns with the driving member through one-quarter of a revolution and is idle for the remainder of the revolution of the driving member.

A number of ingenious mechanisms in which a ratchet arrangement or a Geneva motion play a prominent part are described in this chapter. For other mechanisms of a similar type, the reader is referred to Volumes I, II, and III of "Ingenious Mechanisms for Designers and Inventors."

Adjustable Intermittent Ratchet Mechanism

The device shown in Fig. 1 was used to give intermittent drive to a mechanism by means of a ratchet. The number of teeth per cycle was to be adjustable, as well as the location of the teeth in the cycle.

The device itself consists of two ratchets of the same diameter and number of teeth. Ratchet A is keyed to shaft B, and transmits the desired motion to this shaft. Ratchet C is free to turn on shaft B. On the extended hub of ratchet C is carried, on one side the pawl arm D, and on the other side the masks E and F.

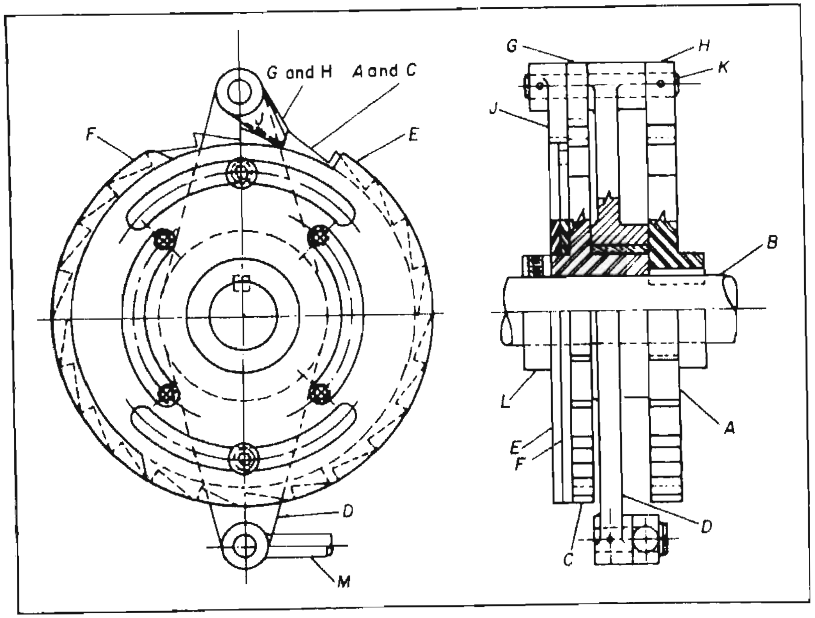


FIG. 1. Pawl G rotates ratchet C and attached masks E and F with every stroke. Masks E and F are capable of lifting pawl J and H, through pin K, thus controlling the rotation of ratchet A.

These masks have stepped diameters, the major equal to that of the ratchets, and the minor slightly less than the root diameter of the ratchet teeth. There is a series of arcuate slots in the masks, and tapped holes in ratchets C are so arranged that a variable number of teeth can be uncovered, and the position of these teeth located anywhere on the circumference of the ratchet. It is obvious that a modification of the profile of these masks will provide for an infinite number of conditions.

On the upper end of the pawl arm are carried two pawls, G, and H, and a pawl-like lever, J. These are carried on a pivot pin K, pawl H, and lever J being pinned on pin K. Pawl G is free on pin K.

Motion is transmitted to the pawl arm D by the connecting rod M by means not shown.

The operation of this device is as follows: As shown, the mechanism is set up to move four teeth per cycle, one tooth having been moved already. The next three movements of the pawl arm will move a tooth each, the whole mechanism rotating as a unit. The fourth backward movement of the pawl arm will cause lever J to ride up on the major diameter of mask F. Both lever J and pawl H being pinned to pivot pin K, this movement outward of lever J will lift pawl H out of engagement with ratchet A. Pawl H will remain out of engagement as long as J remains on the major periphery of the masks F and E. Pawl G, however, will engage ratchet C, moving it forward with its attached masks. This will continue until the lever J will be permitted to move down to the minor radius of the masks, when pawl H will re-engage ratchet A and the next movements of the pawl arm will carry the driven mechanism forward, in this case, four teeth.

Ratchet-Tripping Mechanism Controls Cut-off Length of Sheets

A mechanism employing a feed-ratchet tripped indirectly by a roller chain for pre-setting cut-off lengths was designed for an expanded-metal fabricating machine. The ram type machine has toothed blades attached to a slide that reciprocates across the sheet between strokes. The blades punch and expand openings in a solid steel sheet.

During the first stroke, the slide is laterally situated in one extreme position. Then, after the material has been fed forward a distance equal to one row of perforations, the blade slide moves to the opposite extreme position for a second stroke. Thus the rows of expanded openings are staggered on the sheet. The expanded metal is cut off from the solid sheet each time that the feed mechanism is prevented from functioning by the ratchet-tripping device here described.

The adjustable ratchet arrangement employed for feeding the metal sheets is shown in Fig. 2. Ratchet wheel A is intermittently rotated by feed-pawl B. The feed rate may be controlled by the adjustment of knob C to change the radial location of the driver that imparts motion to the feed pawl.

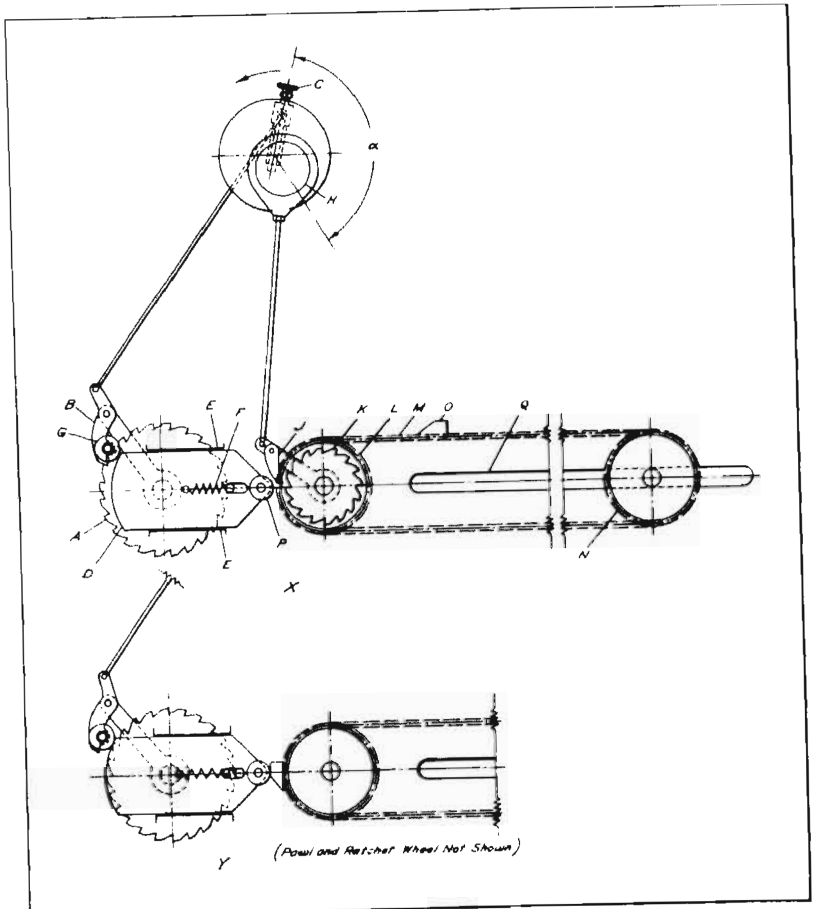


FIG. 2. Ratchet-tripping mechanism for interrupting a press feed at pre-set points, shown in feeding position at X, and non-feeding position at Y.

Slide D, which is retained between two guides E, trips the feed-pawl B directly. The slide is shown in its extreme right-hand position at X, held there by spring F. The left-hand end of the slide is an arc of approximately the same radius as ratchet wheel A. Roller G, attached to the feed-pawl, rides on this rounded end when the slide is shifted to the left.

Slide D is actuated by the ratchet-driven roller chain mechanism at the right in view X. A nonadjustable eccentric H, mounted on the drive-shaft, actuates pawl J. In turn, pawl J

imparts intermittent motion to the ratchet wheel K. The eccentricity of member H was calculated to advance the ratchet wheel one tooth for each revolution of the drive-shaft.

A chain sprocket L is mounted on the same shaft as the ratchet wheel. The sprocket carries a roller chain M which is held in tension by idler sprocket N. Ratchet wheel K and sprocket L have the same number of teeth so that each feed movement of the pawl will advance the chain a distance of one link. Cam O, bolted to one of the chain links, pushes slide D against roller G once for each complete cycle of chain movement.

The arrangement of the mechanism during the feeding phase of the cycle is shown in view X. The slide is located at the right, allowing feed-pawl B to function. As the operation continues, the roller chain moves counterclockwise until the tapered front surface of cam O engages roller P. This forces slide D to the left, against spring pressure, to displace roller G and disengage the feed-pawl B from ratchet wheel A. The mechanism is then in the non-feeding phase of its cycle, as seen at Y. The next revolution of the drive-shaft will index the sprocket K through a distance of one tooth and move cam O past roller P. Spring F then returns slide D to the right-hand position so that feeding of the work will continue.

During the non-feeding portion of the cycle, the fabricated metal is cut off from the solid sheet. For various cut-off lengths, chains of different lengths are used. A slot Q is provided for the purpose of adjusting the idler sprocket either backward or forward to accommodate the various lengths of chain required for different jobs.

Angle α should be selected to make the two pawls operate in opposite directions. This will assure that either a feeding movement will occur or that there will be absolutely no movement; therefore the condition in which a feed movement is partially made before feed-pawl B is disengaged will never arise.

Positive Ratchet Mechanisms Designed for Silent Operation

Silent-operating positive-drive ratchet mechanisms are not too well known. By substituting a brake for the conventional spring, the pawl or finger member is lifted off the ratchet teeth on the idle stroke and made to engage the teeth again on the return stroke. Thus, although still being a positive intermittent mechanism, it works without the usual clicking noise made by the finger in riding or jumping over the teeth on the idle stroke, and therefore reduces the wear on the finger as well as on the teeth.

Referring to Fig. 3, the finger *F* is pivoted on an arm *A* which is, in turn, pivoted on the shaft *S*. The connecting-rod *C* pivots on the finger *F*. A spring-loaded brake *B* — prevented from rotating by a stud in the body of the machine — acts as a brake on a drum which is part of arm *A*. The toothed ratchet wheel *W* is keyed to the shaft *S*.

On the idle stroke (indicated by dotted-line arrow), the connecting-rod *C* will first pivot the finger *F*, thus lifting its point off the tooth on the ratchet wheel *W*. The arm *A* will not turn on the shaft at that time, as it is being restrained by the brake *B* and so offers more resistance to movement than the finger. This finger will pivot only through a certain angle until its short finger hits the stop *T* on the arm. It will then force the arm *A* to turn on the shaft, overcoming the friction of the brake and causing the finger and the arm to pivot on the shaft as one part.

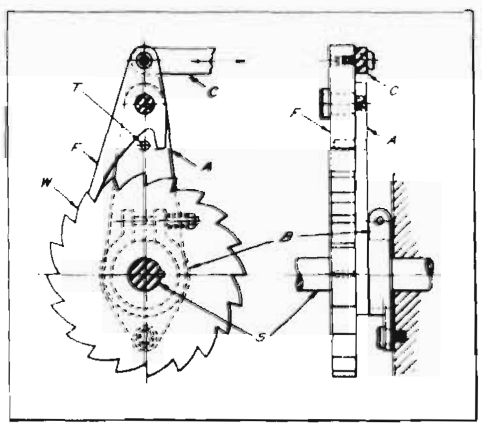


Fig. 3. Silent-operating, positive-drive mechanism with the ratchet pawl pivoted on the oscillating driving arm.

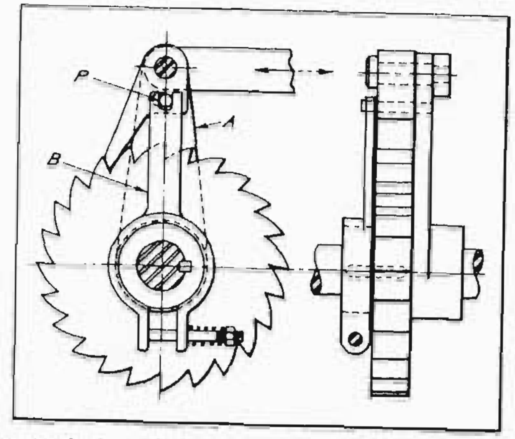


Fig. 4. Alternate design of silent, positive ratchet mechanism with connecting-rod and pawl pivoted on same pin.

On the return stroke, the finger will first pivot to engage a new tooth, and then the whole mechanism will turn as one piece, including the ratchet wheel and the shaft.

Another design of silent ratchet, in which the connecting-rod and the finger are both pivoted on the same pin, is shown in Fig. 4. The arm *A* and the brake arm *B* are arranged on opposite sides of the ratchet wheel. A pin *P* fixed in the finger provides

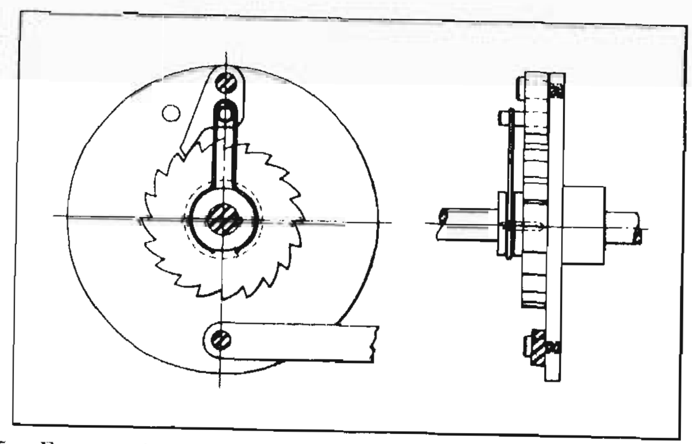


Fig. 5. Front and side views of ratchet designed for use on small-sized mechanisms.

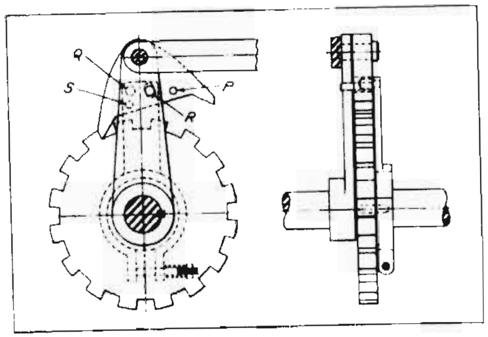


FIG. 6. Reversible ratchet with double-ended pawl.

the necessary stop by engaging an elongated slot in the brake arm on one side and a circular slot in the arm *A* on the other side.

A design that is suitable for small-size mechanisms, and which is similar to the one illustrated in Fig. 4, is shown in Fig. 5. The brake is made of a piece of spring wire. The big disc replaces the arm. The designs shown in Figs. 4 and 5 have the brakes operating on a drum which is attached to the shaft and ratchet wheel. This arrangement calls for a stationary finger to prevent the shaft from reversing on the idle stroke, or else the shaft with all the elements driven by it should offer enough resistance to prevent reversal on the idle stroke resulting from the grip of the brake. Obviously, this arrangement is not absolutely necessary for the design, and the brake drum can be attached to the body of the machine, as in the design shown in Fig. 3.

A reversible silent ratchet mechanism is seen in Fig. 6. The teeth on the wheel are made "square" to have two radial sides, for forward and reverse driving. The finger is made double-pointed. To change the direction of drive, pin *R* on the brake arm should be shifted to position *Q*, and stop-pin *P* on the finger should be shifted to position *S*.

Silent Ratchet Mechanism for Over-Running Drive

Ratchet mechanisms used on over-run drives frequently present problems of noise and wear. Shown in Fig. 7 is a ratchet

mechanism designed to operate silently, with a minimum of wear on its working parts.

Consisting principally of a gear *A* and a ratchet *B*, the over-drive assembly is driven either by shaft *C*, to which the ratchet is keyed, or by the gear. The driving gear is mounted on the hub of the ratchet, and is free to turn on the hub, being retained by collar *D*. A recess is provided in the gear member to accommodate pawl *E*. Although the pawl pivots on pin *F*, it fits the pin loosely, and the actual pressure transmitted by the pawl is borne by the right-hand end of the recess in the gear.

When the shaft is driven by the gear, which revolves counter-clockwise, the pawl drives the ratchet in the usual manner. But when the shaft, which also rotates counter-clockwise, becomes the driver, the gear is stationary, and the pawl over-rides the ratchet. One of the functions of the mechanism at this time is to prevent the pawl from sliding over the ratchet teeth.

This is accomplished in the following manner: When the ratchet rotates counter-clockwise, a brass cam-plate *G* moves with it due to the friction developed by four cork-tipped spring plungers *H* in the ratchet as they ride on the cam-plate. The movement of the cam-plate lifts the pawl from the ratchet as pin *J*, which projects from the pawl, slides up slot *K* in the cam-plate.

When the gear drives the shaft, the ratchet remains stationary until the pawl is engaged. Since the friction generated by the

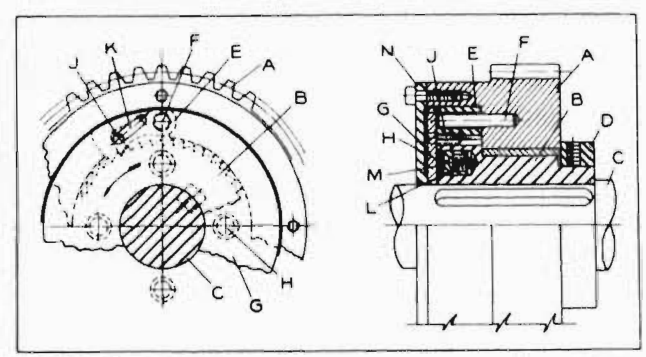


FIG. 7. Ratchet mechanism on an over-run assembly which operates silently and with minimum wear on the parts.

spring plungers in this case will retard the rotation of the cam-plate, pawl pin J is forced down to the left in slot K. Therefore, the pawl engages the ratchet teeth and the entire assembly revolves as a unit.

Spring plungers H are contained in four blind holes bored in the side of the ratchet. Outward pressure of each plunger is exerted by a spring L against a cork friction button M, fitted into a hole bored in the plunger. Covering the ratchet mechanism is a protective plate N, which is relieved to avoid a large area of contact with the cam-plate. To assure the frictional movement of the cam-plate only under the desired circumstances, the area of contact is reduced to a minimum. There is also clearance between the cover plate and the shaft for the same reason.

Additive and Subtractive Ratchet Mechanism

In the operation of a ratchet-driven device it was found desirable to automatically, and frequently, add extra tooth movements. Occasionally, all movement must stop.

The device illustrated in Fig. 8 includes ratchet wheel A which is keyed to the driven shaft B. Mounted on an extended hub of the ratchet is a bushed push-pawl arm C. Push pawl D is pivotally mounted on arm C. The pawl is provided with a projecting pin E that rides on the periphery of mask K. Arm C is provided with gear teeth cut around a portion of its hub for engagement with mating teeth cut on the left end of hook-pawl arm F.

The hook-pawl arm is pivotally mounted on bracket G. The location of this pivot point must be such that the movement of the pawl H on the outer end of arm F is equal to that of pawl D. The hook pawl is provided with a pin J which also rides on mask K. Mask K is bushed and is mounted freely on shaft B. Connecting-rod L has one end attached to the mask. Pawl arm C is reciprocated by another connecting-rod M.

In the operation of this device pawl D does all the driving, ordinarily moving a distance of one tooth on ratchet wheel A

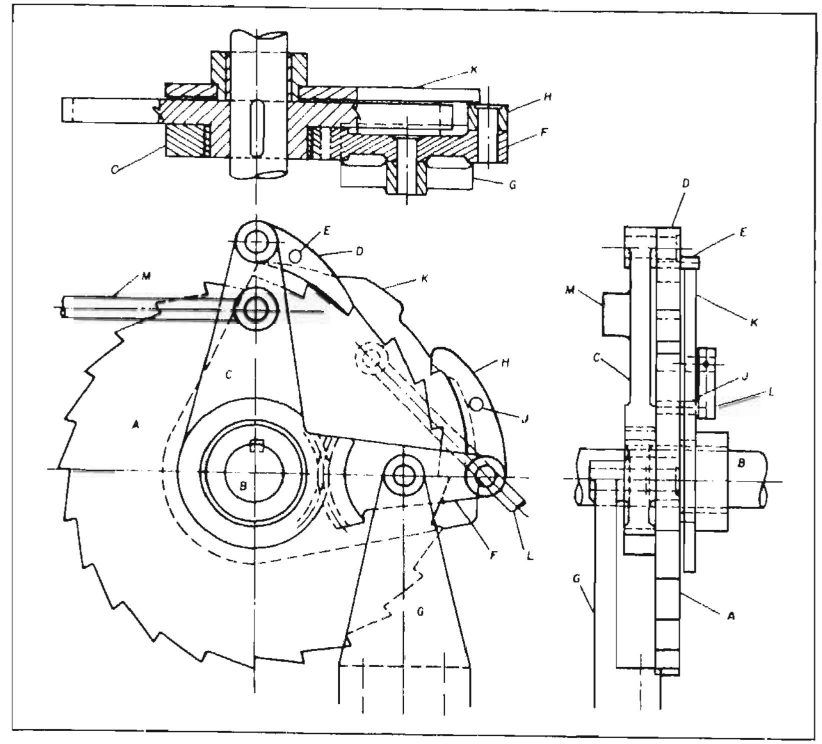


FIG. 8. Ratchet mechanism has a mask that allows automatic changes in ratchet stroke or complete temporary stoppage.

for each reciprocation of arm C, as a simple ratchet movement. In such movement pin J on hook pawl H rests on the lobe of mask K so that pawl H is held out of engagement with the ratchet.

When conditions arise that require increased movement of shaft B, the control mechanism, through connecting-rod L, moves mask K clockwise so as to permit pawl H to function. The mask holds the new setting until changed.

Conversely, if stoppage of shaft B is called for, the mask moves counterclockwise until pawl D is lifted out of engagement. This occurs when pin E rises on the lobe of mask K. Size of the lobe on mask K for pin J permits sufficient counterclockwise movement to hold both pawls out of engagement with ratchet wheel A.

Ratchet Operates on Alternate Strokes

A dual ratchet-wheel system provides the required rotation of a shaft only on alternate strokes of a reciprocating drive lever. Figure 9 shows the mechanism at the end of a power stroke.

Driven shaft *A* and ratchet wheel *B* are keyed together. This wheel has a hub on each side; one side carries one lever *C*, and the other side, pilot wheel *D* and a second lever *C*. Both levers and the pilot wheel are free on the hubs. Pawl *E* is pinned between levers *C*, and is wide enough to engage the teeth of both wheels. Reciprocating drive lever *F* transmits motion to both levers *C*.

Teeth of ratchet wheel *B* are the usual shape, except that there is somewhat greater spacing between them. On the other hand, the teeth of pilot wheel *D* are a special shape, as shown.

With the levers in the position illustrated, at the end of a power stroke, the pawl has engaged one tooth of ratchet wheel *B* and rotated it to the limit of lever movement. It will be noted that the radial contact faces of the teeth of wheels *B* and *D* coincide, and since the pawl is wide enough to engage both wheels, they have been rotated in unison.

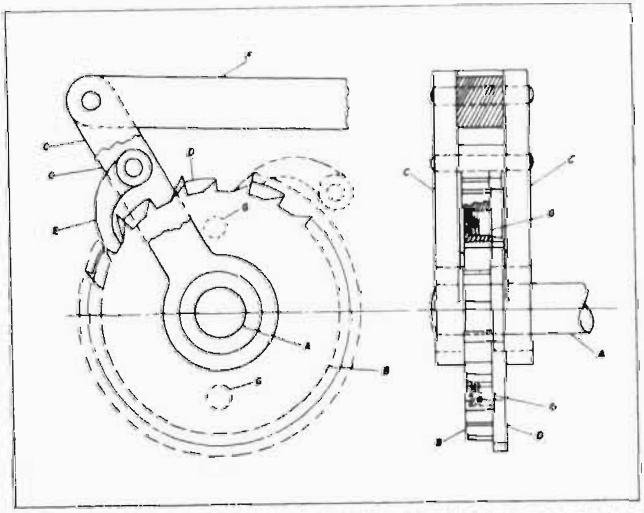


FIG. 9. The design of the teeth of pilot wheel *D* keeps pawl *E* out of engagement with ratchet wheel *B* on alternate oscillations of levers *C*.

The pawl is shown in broken line at the end of the subsequent return stroke. Here, it is in contact with one tooth of pilot wheel *D*, but is raised out of contact with the ratchet wheel *B*. On the next power stroke, the pilot wheel is rotated, but no motion is transmitted to the ratchet wheel, and therefore no motion to the driven shaft. At the end of this power stroke, the pilot wheel will come to rest so that the contact face of the tooth will coincide with the contact face of one of the teeth in the ratchet wheel.

Then, at the end of the next return stroke, the pawl will again be in position to fall into contact with one tooth on both wheels. In this way, the required shaft rotation on alternate reciprocations of the drive lever is obtained: on one power stroke, both wheels are rotated in unison, and the motion is transmitted to the driven shaft; but on the subsequent power stroke, the ratchet wheel is not rotated, since the pawl is held up.

Two spring-loaded plungers *G* are contained in the ratchet wheel, bearing against the adjacent face of the pilot wheel. By applying a light frictional resistance to the pilot wheel, they prevent any backward rotation due to the drag of the pawl on the return stroke.

The mechanism will operate regardless of the angular oscillation of levers *C*, with the limitation that the pawl always must move an uneven number of teeth, such as one, three, or five. If the pawl were to move an even number of teeth, such as two, four, or six, the ratchet wheel would be rotated on each oscillation, rather than on alternate oscillations as required. This, in itself, may be an advantage, in some instances, in that it is possible to vary the driven-shaft rotation from alternate to consecutive action merely by changing the range of oscillation of the levers, to increase or decrease the movement of the ratchet wheel. There must, of course, be an even number of teeth or contact faces on both the ratchet wheel and the pilot wheel of this mechanism.

Variable Intermittent Movement Derived from Gear Drive

An unusual mechanism which provides intermittent movement from a standard gear drive is shown in Fig. 10. The amount of movement of the driven member is variable within wide limits and is easily adjusted to any degree of arc.

In this mechanism, driving gear *A* revolves freely on stationary shaft *B* and is retained in position by collar *C*. Three concentric bores in gear *A* contain the elements of the intermittent movement device. A rectangular flange *D*, mounted integrally on shaft *B*, provides a mounting surface for slide *E*. Two screws *F*, passing through elongated holes in the slide, fasten it to the base of a deep slot machined across the flange face. These holes permit the degree of intermittent movement obtained from the mechanism to be varied.

A lever *G* pivots freely on shoulder-screw *H*. The screw is threaded into slide *E* at a point below the center of gear *A* as indicated by dimension Z. Ratchet wheel *J* is keyed to a shoulder on the end of output shaft *K*.

Motion is transmitted between gear *A* and ratchet wheel *J* by means of pawl *L*. The pawl is secured to gear *A* by shoulder-

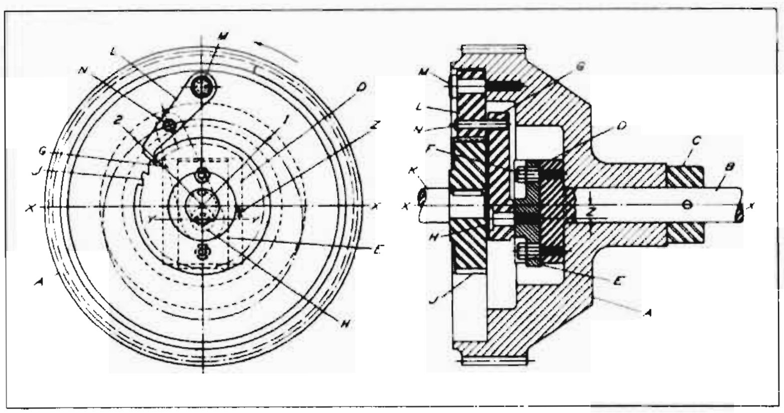


FIG. 10. Drive mechanism designed to convert constant rotary motion into variable intermittent rotary motion.

screw *M*. Driving pin *N* is pressed into the pawl and passes through a short elongated hole in lever *G*. The purpose of this connection is to control the swinging movement of the pawl.

As driving gear *A* rotates in the direction indicated by the arrow, ratchet wheel *J* is forced to revolve in the same direction due to the engagement of pawl *L*. Because driving pin *N* is connected with eccentrically mounted lever *G*, the pin moves in a circular path that is not concentric with shaft *B*. Thus, as gear *A* rotates, the pawl is successively drawn closer to, then farther away from, the teeth of the ratchet wheel. In this way, an intermittent motion is imparted to output shaft *K*.

With slide *E* located so as to provide an offset equal to distance Z, the ratchet wheel will rotate approximately 45 degrees during each revolution of gear *A*. The length of engagement between the pawl and the ratchet wheel is denoted by numbers 1 and 2 in the left-hand view. By adjusting the position of slide *E*, the distance traveled by the ratchet wheel can be varied.

Intermittent Motion Derived from Continuously Rotating Shaft

On a wire-forming machine, it was necessary to interrupt the feed of the wire at certain intervals in the cycle. To accomplish this, the shaft operating the feeding mechanism was cut at one point, and the mechanism illustrated was then installed.

As indicated in Fig. 11, shaft *A*, the driving member, transmits its motion to shaft *B*, which operates the feeding mechanism. Keyed to shaft *A* and rotating with it is a disc *C*. The disc carries a pawl *D* that is normally held in contact with a ratchet *E* by a spring *F*. Ratchet *E* is attached to shaft *B*. There are eight teeth spaced around the ratchet. A ring *G*, which is mounted on a stationary part of the machine, is slotted at eight equally spaced points to receive studs carrying rollers *H*, which contact the tail of the pawl.

For purposes of explanation, the rollers *H* are numbered 1 to 5. As shaft *A* and disc *C* rotate in the direction indicated by the arrow, the pawl engages one of the teeth of the ratchet, causing shaft *B* to rotate in unison until the tail of the pawl contacts

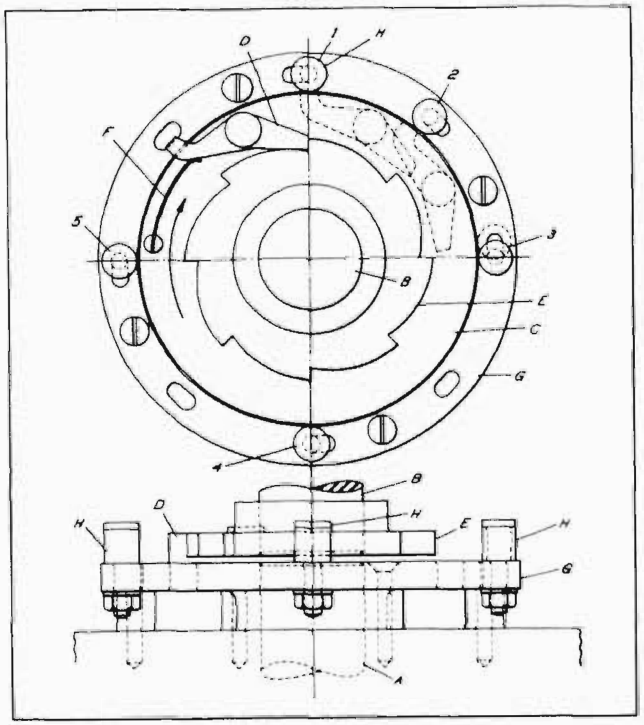


FIG. 11. The pawl D is disengaged from ratchet E by contact with one of the rollers H.

roller No. 1. The pawl at this and subsequent locations is represented in broken line. In contacting the roller, the pawl is released from the ratchet and the motion of shaft B is interrupted until the pawl again engages a ratchet tooth.

Assuming for the present that roller No. 2 has been removed, the movement of the ratchet again begins when the pawl engages ratchet tooth adjacent to roller No. 3. Thus far, shaft A has rotated 90 degrees, but shaft B has rotated only 45 degrees, the other 45 degrees having been lost by the pawl passing over one tooth of the ratchet. Continued movement of the disc causes the ratchet to again be rotated until the tail of the pawl contacts roller No. 3, when movement is once more interrupted. With rollers Nos. 1, 3, 4, and 5 positioned as shown, there are four

movements of shaft B and four rest periods of 45 degrees each in every rotation of shaft A. On the machine involved, this was the particular intermittent motion required.

The design, moreover, lends itself to other variations. Assuming, for example, that roller No. 2 is placed as shown, the pawl is prevented from engaging the ratchet tooth. It will be noted that this roller has been moved to the upper end of its slot, the purpose being to operate the pawl before tooth engagement. If roller No. 3 were moved to the upper end of its slot, disengagement would continue until the tooth adjacent to roller No. 4 is reached, thus producing a rest period of 135 degrees. Likewise, if roller No. 4 were transferred from its present position to the slot immediately to the right, engagement would again be prevented, producing a rest period for shaft B of 180 degrees.

It is evident that various combinations of intermittent motions can be obtained, depending on the number of rollers used and their locations. This mechanism can be adapted to a wide variety of intermittent motions by increasing the number of teeth in the ratchet and providing a continuous slot in ring G, so that rollers are able to be placed in any position.

Cam and Ratchet Intermittent Mechanism

A mechanism that employs a cam, ratchet, and pawl for converting constant rotation into intermittent rotary motion is shown in Fig. 12. Drive-shaft A of this mechanism revolves continuously. Keyed to it is disc B which carries pin C which is the pivot for pawl D. On the driven shaft E there is keyed ratchet wheel F. The inner end of the pawl engages the ratchet teeth while the outer rides around a cam path cut into disc G.

As pivot-pin C is carried around disc G by the rotation of the drive-shaft in the direction indicated by the arrow at M, shaft E is turned by pawl D during the period when end K of the pawl is riding along the high section of the cam P. A spring, not shown, exerts sufficient pressure to hold the cam tooth in engagement with one of the teeth on the ratchet during this period of pawl movement.

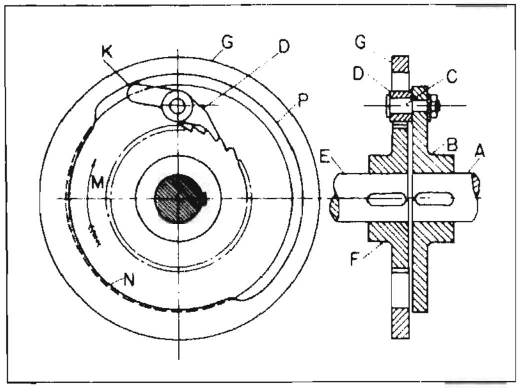


FIG. 12. Mechanism that employs a cam, ratchet, and pawl for converting constant rotation into intermittent rotary motion.

When end K of the pawl reaches the low portion of the cam indicated by letter N, the pawl is swung on its pivot and its tooth is disengaged from the ratchet. The driven shaft E then remains stationary until end K of the pawl again rides on the high section of the cam and the pawl tooth has once more been swung into engagement with a cam tooth.

Geneva Drive in which Gear Ratios Control Motion Time

Geneva drives used to a wide extent in automatic machinery generally consist of a driving roller at the end of a crank and a slotted member which is moved when the driving roller enters into a slot. The conventional Geneva drive has some disadvantages, one of them being that time for motion and dwell of the driven member is usually determined for a given number of slots or stations.

In Fig. 13 is shown a modified Geneva drive of which the time for motion is dependent upon gear ratios. In this drive, input shaft A rotates with uniform velocity and drives gear B, which in turn drives sun gear C. The latter is free to rotate on the shaft D. Shafts A and D are supported in the gear housing. Sun gear C drives planet gear E, and as long as the roller H is outside of the slotted member F, gear E rotates with uniform velocity because the planet block G is detented. The detent device is not shown. Roller H is just starting to enter a slot in Fig. 14, while Fig. 15 shows the mechanism some time after the roller has entered the slot.

At the moment that the roller enters the slot, the planet carrier becomes unlocked. The roller, however, is now in the slot, and because of the angular motion of link J, which is driven by planet gear E, the roller will penetrate deep into the slot and cause shaft K to rotate counterclockwise around shaft D.

In general, the size of an idler gear has no influence on gear ratio, but in this case the size of gear C is of importance because planet gear E rolls on gear C during motion.

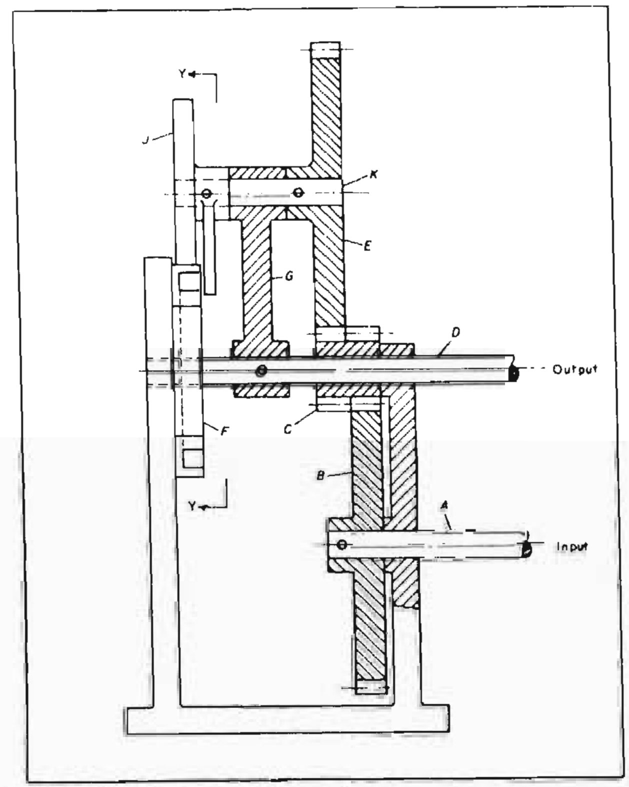


FIG. 13. Modified Geneva drive with motion time controlled by gear ratios.

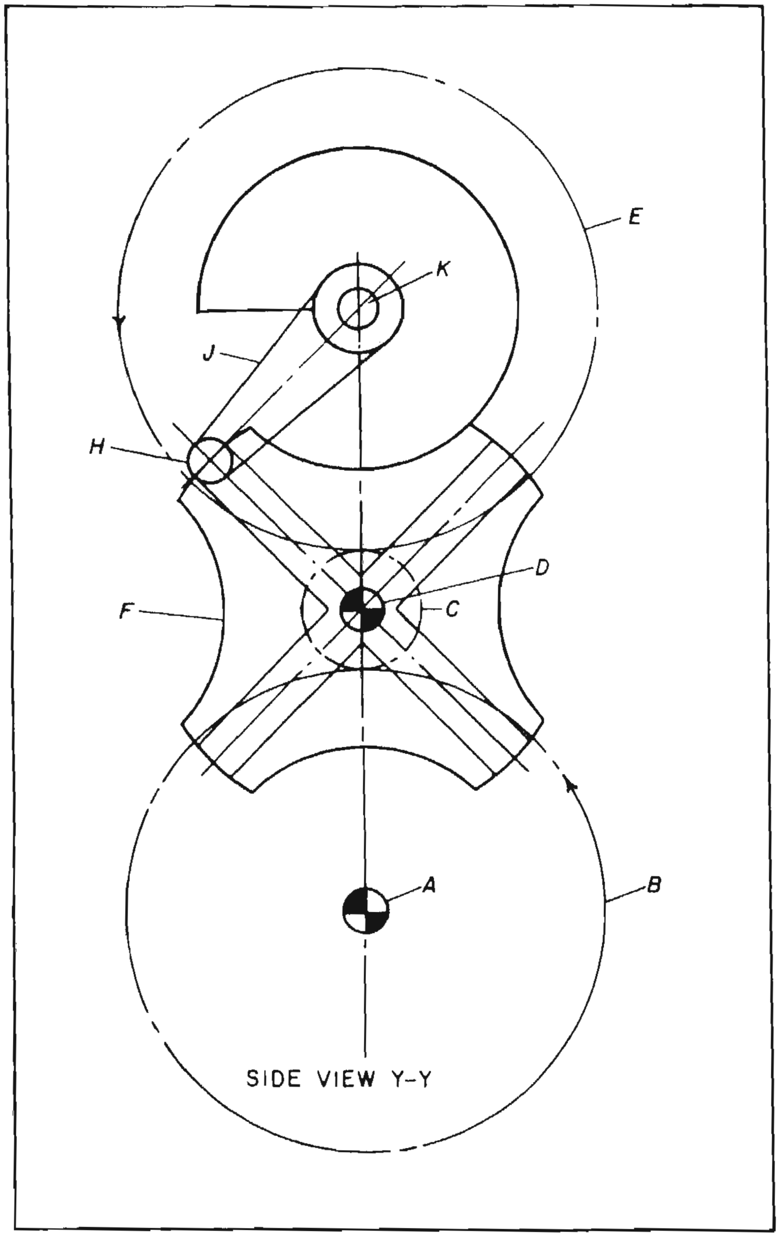


FIG. 14. Diagram showing roller *H* entering slot of planet block at beginning of Geneva motion.

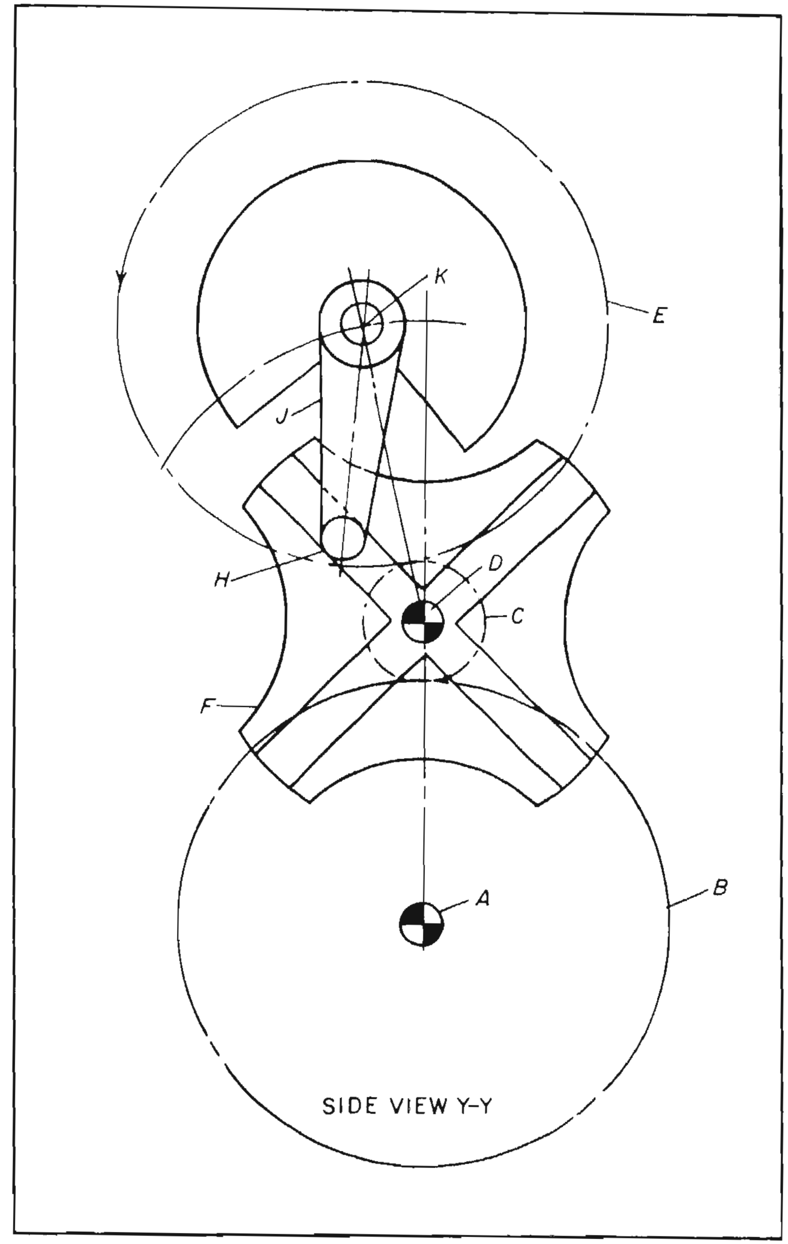


FIG. 15. Diagram showing positions of elements after roller *H* has moved along slot of planet block.

By superimposition of the different motions it can be shown that time for indexing by using a four-slotted member is

$$T = 90^\circ \left(\frac{D_4 - D_3}{D_2}\right)$$

For the mechanism described, $4D_3 = D_2 = D_4$, where D_2, D_3, and D_4 are the pitch diameters of gears B, C, and E, respectively.

$$T = 90^\circ \left(\frac{D_2 - 0.25D_2}{D_2}\right) = 3/4 \times 90^\circ = 67.5$$

degrees, and the time for dwell is $360 - 67.5 = 292.5$ degrees.

The output motion of the planet carrier G by each indexing is 90 degrees.

Automatic Programming by Ratchet Wheel

Figure 16 shows the design of a mechanism which will impart a variable, partial, intermittent rotation to a driven shaft. The

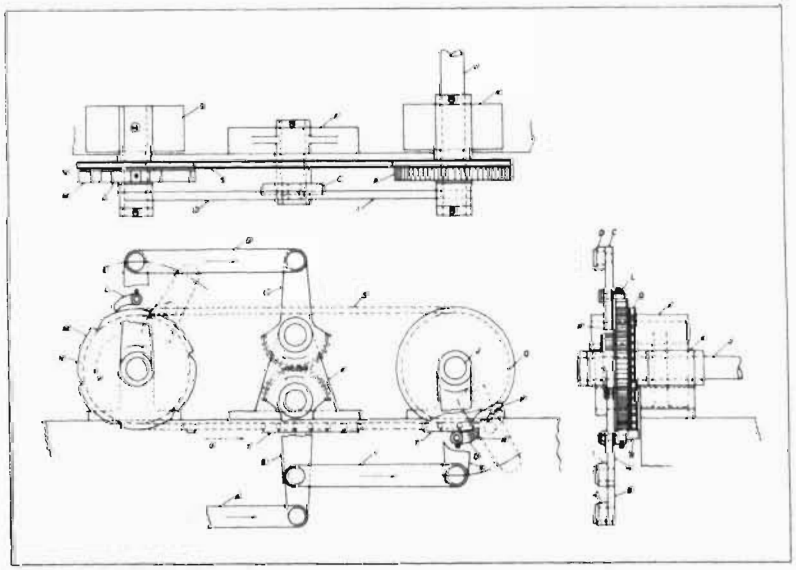


Fig. 16. Three views of the programming mechanism which gives variable, partial, intermittent motion to shaft J as driven by bar A and sequenced by chain S.

purpose is to produce a variable spacing of the strands of woven wire in a pattern.

Referring to Fig. 16, which shows three views of the mechanism, the reciprocating bar link A supplies the mechanism with motion which causes shaft J to deliver the required variable, partial, intermittent rotation. The movement of bar A is transmitted to lever B, which carries a gear sector at its upper end and is keyed to a shaft, which is free to rotate in bearing block F. The gear sector on lever B, in mesh with a mating gear sector on lever C, transmits linear motion to the lever arm of C. Lever C moves lever E through link D. Lever E swings freely on a shaft. The motion of lever B is also transmitted to lever H through link I. Lever H swings freely on the driven output shaft J, which is mounted for free rotation in bearing block K.

Lever E carries pawl L to engage in the notches on the periphery of ratchet disc M. The ratchet wheel is attached to sprocket wheel N, the pair being rotatable on a shaft. The number of notches on ratchet disc M is determined by the angular movement of lever E. Its angular movement, in turn, is governed by the amount of angular movement of lever H needed to produce the maximum partial revolution of shaft J that will give the required spacing of the strands of wire.

Lever H carries pawl O, which engages teeth of ratchet wheel P, keyed to output shaft J. On the side of the pawl is lifter pin R, which overhangs sprocket Q. This sprocket free-wheels on shaft J. Sprockets N and Q are linked by roller chain S. The length of this chain is governed by the number of movements of shaft J in one "repeat" of the required complete program for the wire-mesh pattern. It must be of such length that the number of links will be a multiple of the number of teeth of sprockets N and Q included in the angular movement of levers E and H. Chain S is equipped with special pawl-lifter links T, placed on opposite sides of the chain where necessary. In operation, the high links contact the pawl lifter, which causes pawl O to fail to contact ratchet wheel P. Pawls L and O are equipped with springs (not shown) that normally insure engagement with their ratchet teeth.

In the drawings, the mechanism is shown at the mid-point of its motion. Bar *A*, moving in the direction of the arrow, transmits motion to the various links and levers in the directions indicated by the arrows. At this point, there is no movement of either the sprockets or the ratchet wheels and, therefore, there is no movement of shaft *J* because the pawl is moving opposite to the direction required for engagement. Also, pawl *O* is held out of engagement with ratchet wheel *P* by the three lifter links shown.

Continued movement of bar *A* in the same direction causes the pawl *O* to end contact with the chain links *T*. The pawl then drops into one of the teeth of ratchet *P*. Rotation of shaft *J* starts, and continues until lever *H* reaches the end of its stroke. Levers *E* and *H* finally take the positions shown by the broken lines. At this point, pawl *L* has been brought into position to engage one of the notches in disc *M*.

On the return stroke of bar *A*, disc *M* and sprocket *N* rotate in the direction of the dotted arrow, causing chain *S* to move in direction *U* (dotted arrow) so that any links *T* attached to chain *S* will pass under pawl *O* moving in the opposite direction. In this manner, the movement of chain *S* is brought into position for the next working stroke of lever *H*. The number of chain links which pass under pawl *O* governs the dwell period of shaft *J*. Its rotation can be started only after pawl lifter *R* on pawl *O* has moved off the lifter links.

Adjustable Indexing Mechanism with 180-Degree Dwell

On certain types of printing presses it is often desired to incorporate a variable indexing motion in which a ratchet is moved during 180 degrees of rotation of the driving shaft and then dwells over the rest of the motion. A mechanism designed to accomplish this result is here shown.

In this mechanism gear *A* is half the size of the internal gear *B*; see Fig. 17. Gear *A* is carried in a circle around the internal gear by arm *C* which is attached to driving shaft *D*. This shaft rotates continuously.

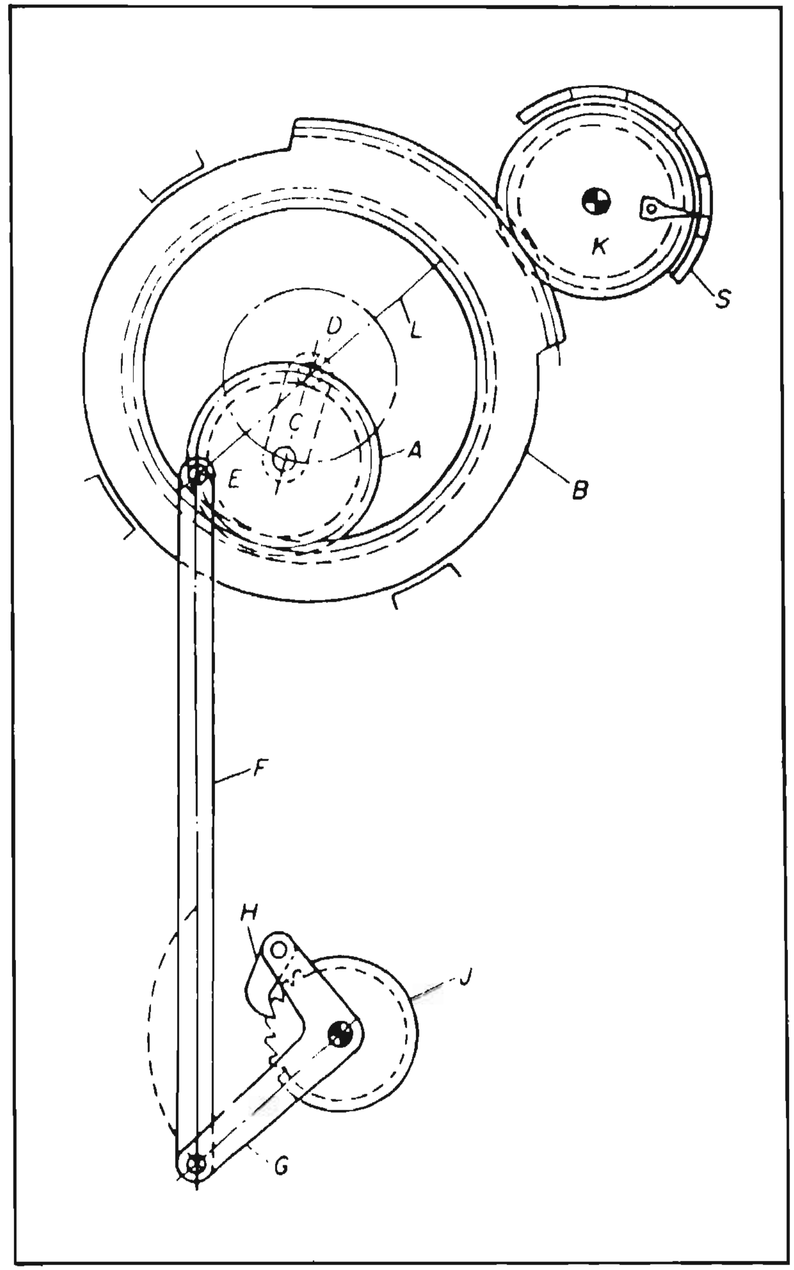


Fig. 17. An adjustable indexing mechanism with 180 degrees of dwell.

As gear A rolls around gear B, each point on the circumference of the small gear describes a straight line. Point E on the circumference describes a path along the straight line L. The motion of point E is transferred to the crank G through link F. Crank G carries pawl H. By the oscillating motion of lever G, pawl H causes ratchet J to move intermittently. The amount of motion is dependent upon the direction in which point E is moving. In order to vary the direction of the path of point E, gear B can be indexed limited amounts by means of gear K. Scale S indicates the amount gear J is indexed.

Counting Device for High Speed Operation

Counters have been required for various applications such as on computers, servo mechanisms, and other similar devices that function at high speeds. Under such usage, the counting unit must impart an absolute minimum of drag, or shock load, to the driving members.

A conventional, low-speed counter, operating with an intermittent motion, builds up momentary shock loads during normal cycling, as can be seen in the graphic illustration at X, Fig. 18. As the operating speed of this type counter increases, so does the acceleration and, as a consequence, the load. This increased load is dissipated in the form of either elastic or plastic

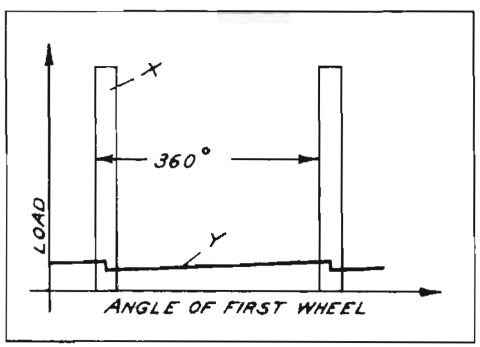


FIG. 18. Load distribution of conventional counter X as compared to the load distribution Y of the high-speed counter.

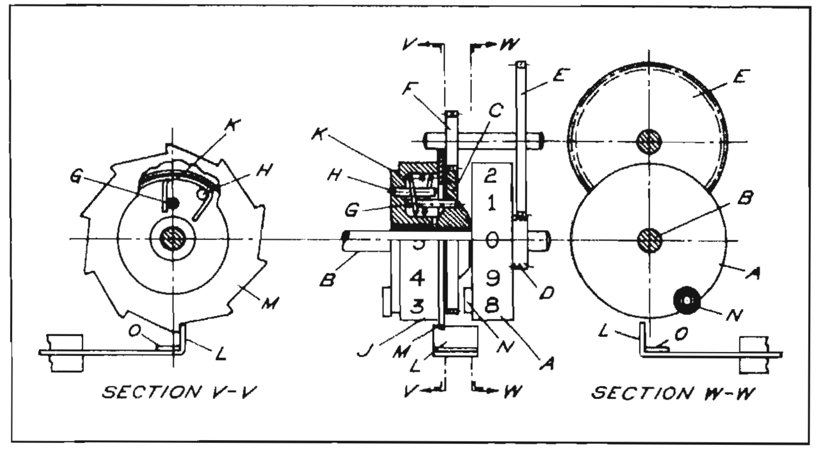


FIG. 19. Spring-loaded wheel J jumps forward one number for each complete revolution of wheel A.

deformation of the counter mechanism. Even if the preferable case of elastic deformation should result, the driving mechanism to which the counter is attached is liable to suffer from the shock of being momentarily halted or rapidly decelerated.

To overcome these problems associated with high-speed operation, ranging up to 12,000 R.P.M. and with a load of approximately 1 ounce-inch, the counter mechanism described was developed. Wheel A, which can be seen in the diagrammatic representation of the counter, Fig. 19, is keyed to input shaft B, thereby making one complete revolution for each made by the shaft. Gear C, which rotates independently of the input shaft, is driven at one-tenth the speed of wheel A by the action of reduction gears D, E, and F.

Pin G, projecting from the hub of gear C, and pin H, projecting from the bottom of a recess in wheel J, engage the bent ends of spiral spring K. As gear C rotates, wheel J tends to rotate in unison through the spiral spring linkage. However, movement of wheel J is prevented by pawl lever L which engages a tooth of ratchet wheel M. The ratchet wheel is attached to the counter wheel. While in this position, energy necessary to turn wheel J is being built up in a potential form in spring K.

A small ball bearing N is mounted on the left-hand face of wheel A. Once during each revolution of this wheel, the ball bearing contacts pawl lever L, depressing it momentarily. This frees wheel J, which then rotates one-tenth of a revolution under the influence of the spiral spring, until its movement is once again arrested by the engagement of the pawl lever with the ratchet wheel. In this way, a comparatively even load distribution, as shown graphically at Y in Fig. 18, is obtained. This advantage is not lost under high-speed operating conditions.

Normally, there is no difficulty in reading the numbers when the mechanism is operating in its proper direction, as all numbers except those on wheel A snap into view the instant "9" of the preceding wheel changes to "0." If the counter is operated in the reverse direction, however, the numbers will no longer swing into position but will move continually. At higher speeds the numbers on wheel A are no longer readable. A small weight O, attached to the pawl lever, retards return of the pawl to the ratchet wheel, thereby smoothing out the movement of wheel J. It is then possible for the operator to satisfactorily interpolate the readings of this wheel.

Multiple-Revolution Ratchet Movement

Generally, a ratchet movement is limited to a partial revolution of the driven shaft, since the rotation of the oscillating lever which carries the pawl is necessarily restricted in order to avoid a dead-center effect. The device shown in Fig. 20, however, incorporates an epicyclic gear train to produce multiple revolutions of the driven shaft with only a conventional magnitude of oscillation of the driving lever.

Gear A and a ratchet wheel B are both mounted and keyed to the driven shaft C. An oscillating lever D pivots on shaft C and carries two gears E and F. These gears are keyed together to rotate as a unit on stud G. Gear F rotates in mesh with gear A, and gear E with an internal gear H, which is free to rotate on shaft C. A pawl J engages with the teeth of ratchet wheel B, preventing rotation of the shaft in a clockwise direction. Another pawl, member K, engages ratchet teeth on the periphery

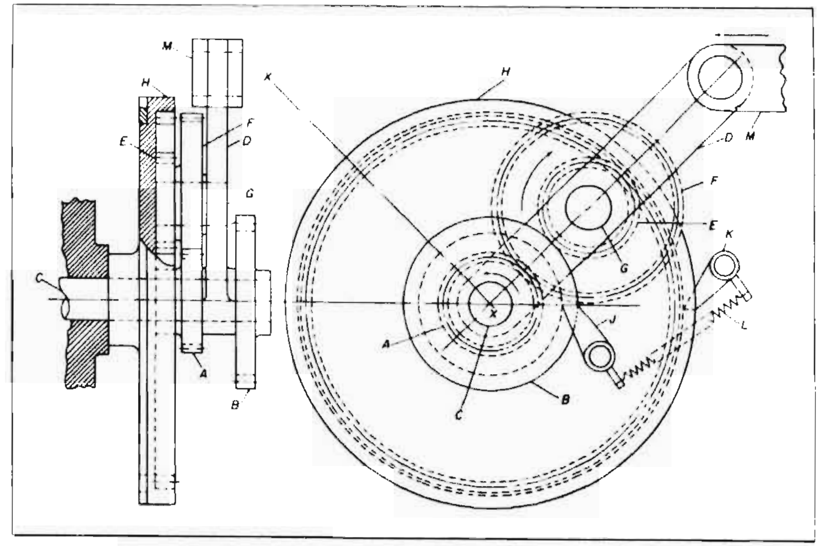


FIG. 20. Ratchet movement that can produce a number of revolutions of the output shaft with each working stroke.

of a ring secured to left side of gear H, preventing counterclockwise rotation. Both pawls are mounted on a stationary part of the machine and springs L hold them in contact with the ratchet teeth.

In the position illustrated, the mechanism is at the beginning of its cycle. Reciprocating rod M, which furnishes the operating power, moves lever D to the left until it occupies a position centered on line X-X. As the lever swings to the left, internal gear H is prevented from rotation in the same direction by pawl K, and gear E, being in mesh with the internal gear, is caused to rotate clockwise. Gear F rotates with gear E as a unit producing a counterclockwise rotation of both gear A and shaft C. On the return stroke of lever D, clockwise movement of the shaft is prevented by pawl J which engages a tooth in the ratchet wheel B. Since both ratchet wheel B and gear A are keyed to the shaft and held stationary, the motion of rod D is transmitted through gears A, F, and E, to produce a clockwise rotation of the internal gear. Thus, by locking gear H, motion is transmitted to the shaft, and by locking the shaft in the reverse direction, motion is transmitted to gear H. No useful work is performed during the return stroke as the shaft remains at rest during this portion of the cycle.

The number of revolutions of the output shaft is a function of the gear ratio and the stroke of lever D. Ratio of the epicyclic gear train R can be obtained by the equation:

$$R = 1 + \frac{F \times H}{E \times A}$$

where A, E, F, and H are the number of teeth or the pitch diameter of gears A, E, F, and H, respectively. Multiplication of this ratio by the magnitude of the stroke, in degrees, divided by 360 will give the number of revolutions of the shaft for each stroke. In the arrangement shown, the ratio of the pitch diameter or gear H to gear E is 4 to 1 and that of gear F to gear A is 2 to 1. The ratio R is, therefore, $1 + \frac{2 \times 4}{1 \times 1}$ or 9. Since the stroke is 90 degrees, the shaft will evolve $9 \times \frac{90}{360}$ or $2\frac{1}{4}$ times in a working stroke.

Two-Speed Double-Action Ratchet Mechanism

A ratchet mechanism had to be designed to move a conveyor belt intermittently so as to carry two parts of an assembly to a number of assembly stations. The two parts vary considerably in size, and so the conveyor belt had to be given a certain movement for the placement of one part and a greater movement for placing the larger part. The mechanism operates the conveyor belt in one direction during two oscillations of a lever and alternately imparts long and short movements to deliver the assembly parts.

In Fig. 21, shaft A, which operates the conveyor belt, carries gear B and ratchet wheel C, both of which are keyed to it. Ratchet wheel D is free on shaft A. Internal ring gear E is fastened to this ratchet wheel. Bracket F, attached to a stationary part of the machine, carries a short rod on which pinion G

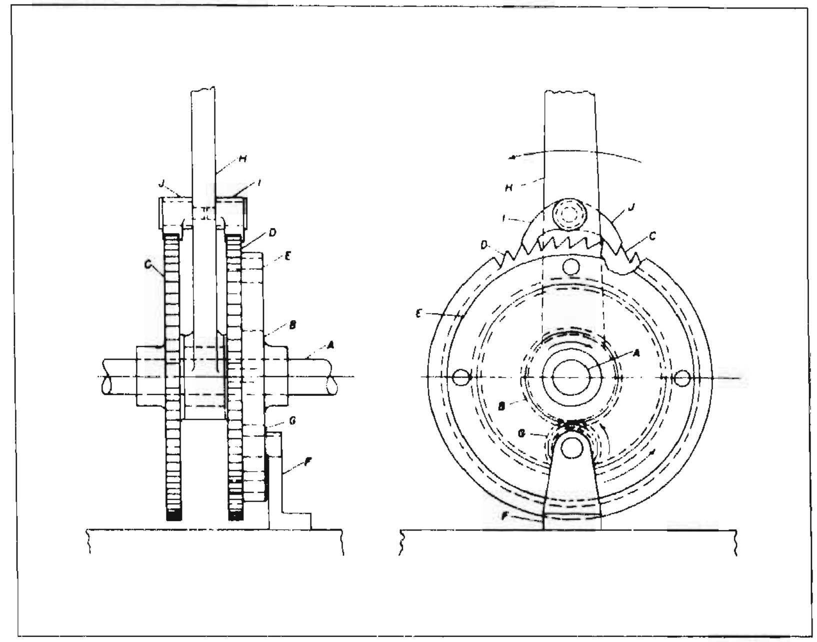


FIG. 21. Ratchet mechanism designed for imparting two rotations of different amounts in the same direction.

rotates freely, meshing with gear B and ring gear E. Ratchet wheel D has a hub on its inner face, on which lever H is free to oscillate. Attached to this lever is pawl J, which engages the teeth of ratchet wheel C, and pawl I, which engages the teeth of ratchet wheel D.

When lever H is moved in the direction indicated by the arrow in the right-hand view, the long stroke is made which produces the longer movement of the conveyor belt. Pawl I turns ratchet wheel D in the direction indicated, and the motion is transmitted to shaft A in the reverse direction through ring gear E, pinion G, and gear B. As the ratio of the tooth count between gear B and ring gear E is 2 to 1, the angular rotation of gear B is twice that of lever H. It will be noted that during this portion of the cycle, the rotation of ratchet wheel C is in the reverse direction to that of ratchet wheel D so that pawl J cannot engage.

When the movement of lever *H* is in the reverse direction, pawl *J* engages the teeth of ratchet wheel *C*, and transmits its motion directly to shaft *A*. During this portion of the cycle, ratchet wheel *D* will rotate in the opposite direction but, being free on shaft *A*, takes no part in transmitting motion.

Blocking Device for a Geneva Wheel

In one particular driving mechanism employing a Geneva wheel, it was found that the wheel was not sufficiently locked. It frequently occurred that the moment pin *A*, shown at Z in Fig. 22, cleared the slot in Geneva wheel *B*, a reverse movement would take place. This was due to a reactive force in the ma-

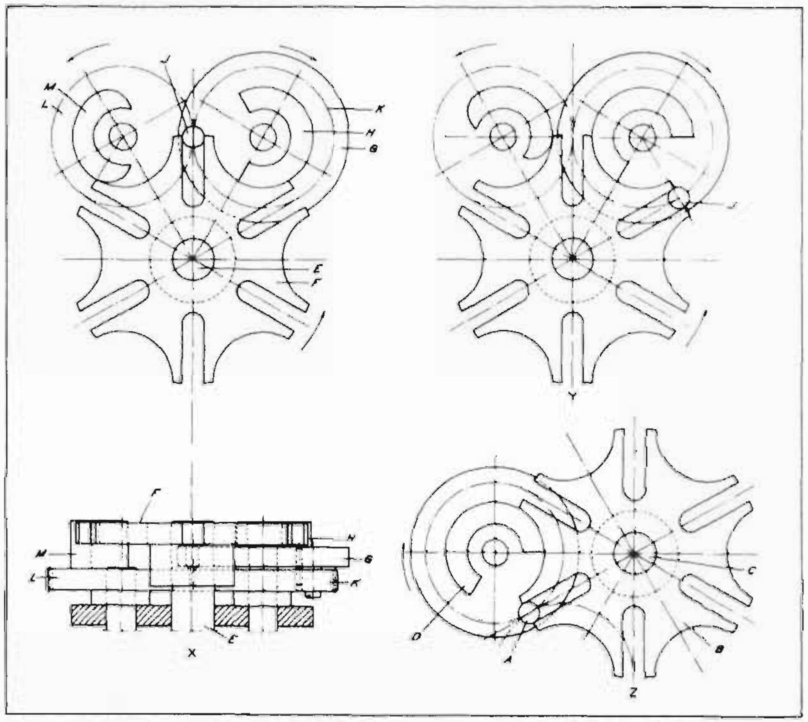


FIG. 22. Geared auxiliary blocking segment prevents reverse movement of Geneva wheel.

chine being driven from shaft *C*. The reverse movement was unchecked because blocking flange *D* is effective in one direction only.

The improved Geneva mechanism shown at X and Y, incorporating a reverse-motion stop, has been designed to eliminate this condition. Shaft *E*, which drives the machine, is fitted with a six-station Geneva wheel *F*. Driving wheel *G*, having a conventional blocking flange *H* and drive-pin *J*, is screwed and doweled to spur gear *K*. Meshing with this gear is a similar spur gear *L* on which is located a crescent-shaped reverse-motion stop *M*.

At X is shown the position of the components at the instant the Geneva wheel has been indexed one station. As drive-pin *J* leaves the slot, wheel *F* is blocked in the forward direction by a portion of flange *H*. The wheel is also blocked in the reverse direction by reverse-motion stop *M*.

The position of the components as the Geneva wheel is about to be indexed another station is shown at Y. Drive-pin *J* enters the appropriate slot in wheel *F* just as crescent-shaped stop *M* is disengaged from the wheel. Due to its shape, stop *M* disengages at a rate that will not impede the forward motion of the Geneva wheel, thus permitting smooth functioning of the mechanism.

Ratchet and Two Pawls Control Movement of Indexing Fixture

Ratchet-controlled positioning and an expanding, work-holding stub-arbor are two features of the unique indexing fixture shown in Fig. 23. From two to eighteen indexing positions can be obtained, depending on the number of notches in ratchet plate *A*.

In this fixture the work-piece is gripped internally on an expanding stub-arbor *B*, shown in the enlarged section X-X. The projecting portion of the arbor has three radial slots, giving it the action of a collet. Knob *C* is prevented from turning on shaft *D* by a full-dog set-screw. The dog point of the set-screw has a sliding fit in a keyway machined in the shaft. When the

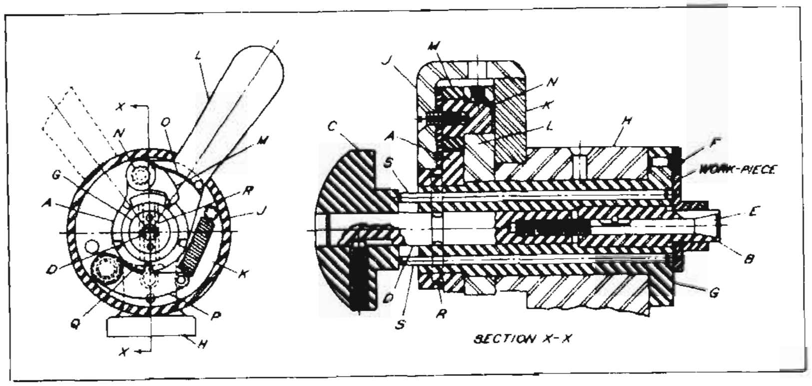


FIG. 23. Indexing fixture functions around the action of a notched-plate type ratchet wheel and two pawls — one driving and one locking.

knob is turned, the threaded end of tapered plug E is drawn into the shaft, causing the arbor to expand. Shaft D is restrained from sliding by dowel-pins R.

The work-piece illustrated is located through a hole in its flange by means of a diamond type locating pin F. This pin is pressed into the flanged face of rotating housing G — the entire subassembly being contained within fixture base H. The complete indexing mechanism is located between moving cover J and stationary plate K, and functions in the following manner.

Indexing of the work-piece is effected by movement of lever L. Pawl M rides on shoulder-stud N which, in turn, is locked to the enclosed part of lever L by a cone-point set-screw (Section X-X). Cover J also is locked to this shoulder-stud by means of a flat-head machine screw as shown.

When the lever is moved to the left, pawl M is disengaged from its notch in ratchet plate A and slides over to the next notch. Flat spring O is brazed to the pawl at one end and backed up by a pin at the other end to maintain downward pressure on the pawl at all times.

During this initial thrust of lever L, the ratchet plate is prevented from rotating by a tooth on the lower, spring-loaded pawl P. However, as the lever moves to the left, a cam surface Q at the lower end of the lever gradually disengages pawl P. Complete

disengagement is timed to occur when pawl M drops into the next notch in the ratchet plate.

At this point, returning the lever to its original position will cause the ratchet plate to rotate clockwise a distance equal to the space between two adjacent notches. This indexing movement is imparted to housing G by two long dowel-pins R that connect the housing to the ratchet plate. As the lever moves to the right, the receding slope of cam surface Q permits spring-loaded pawl P to re-enter a notch in the ratchet plate, thus securing the new position of the work-piece.

After machining operations on the piece have been completed, it is released by first backing off knob C to relieve the expansive forces on stub-arbor B. Then, by striking the knob, ejector-pins S will move to the right and drive the work-piece off the arbor. Altering the number of index positions handled by this fixture would necessitate the replacement of ratchet plate A for one with the appropriate number of notches, and lever L for one with a modified cam surface Q that will effect engagement and disengagement of the lower pawl at the proper moment.

Half Revolution Geneva Mechanism

Three slots are the minimum number that can be used in the conventional type of Geneva mechanism. Or in terms of motion, 120 degrees is the greatest possible angle of rotation of the driven member for each revolution of the driver. It is for this reason that many designers resort to other mechanisms when 180 degrees of intermittent rotation is required of the driven member. For some applications, however, it is possible to use a modified form of the conventional Geneva mechanism, of the type shown in Fig. 24, so that the necessary half-revolution is obtained.

In Fig. 24, driver A is a circular disc to which pin c is bolted. For better performance a roller on a sleeve or an anti-friction bearing, instead of the pin, which is shown for simplicity, should be used. A segment d on driver A serves to lock wheel B in position during the idle period of the cycle.

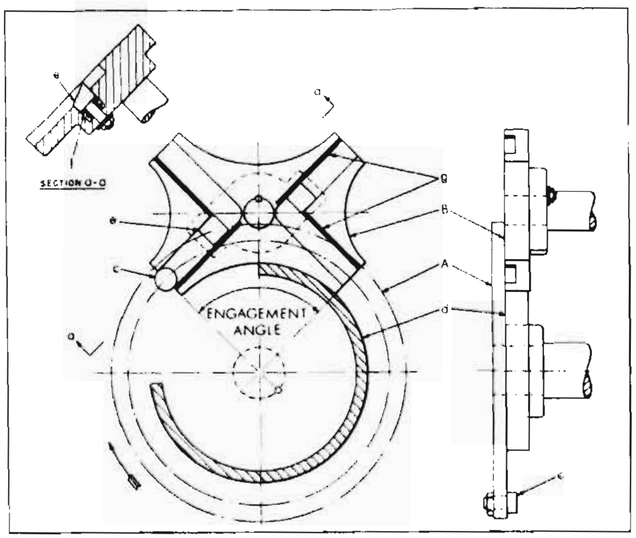


FIG. 24. Diagram of half-revolution Geneva mechanism showing driving pin about to enter slot in driven member.

Wheel *B* has two V-shaped slots. When the wheel is in the idle position, the center line of one leg of each slot is tangential to the circular path of the roller, as shown. Mounted on wheel *B* are two spring-loaded dogs *e*, one in each of the V-slots. These dogs have beveled tops, and, under pressure of driving pin *c*, are forced into a recess in the V-groove so that the pin may pass over them.

When driver *A* rotates, pin *c* enters the slot which lies in its path, and begins to turn wheel *B*. Also, at this point, the segment *d* passes the center line, leaving the wheel free to rotate as the pin continues to enter the slot. Approaching the center of the V-slot, the pin passes over dog *e*, pressing it down as it passes. In Fig. 25 is shown the position of the mechanism when pin *c* has reached the bottom of the V-slot. In this position the pin has passed over the dog and the dog has been returned to its initial extended position by the force exerted through spring *f*, shown in section a-a, Fig. 24.

When the pin is rotated further, it presses against the vertical side of the dog, which now forms an extension of the side of the

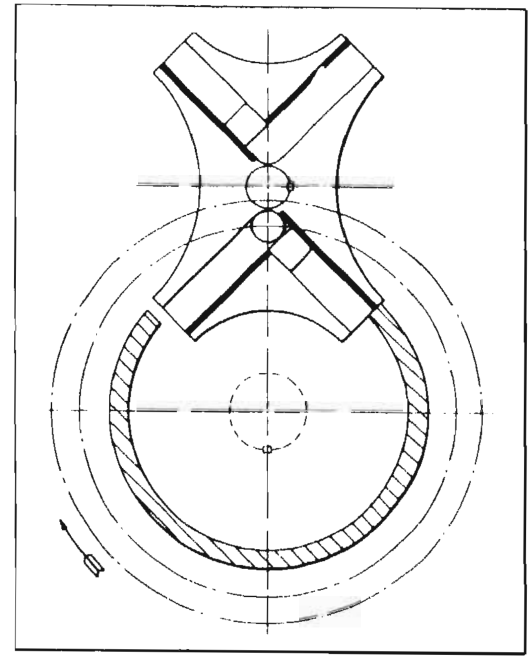


FIG. 25. When the driven wheel has been rotated 90 degrees, as shown, the driving pin has passed over the dog and is at the bottom of the V-slot.

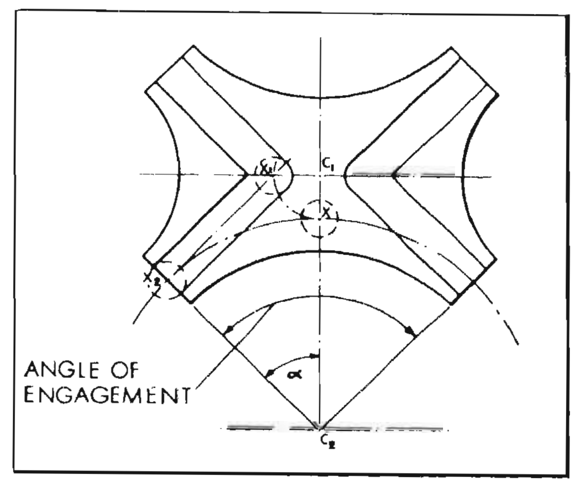


FIG. 26. The correct proportions for the Geneva mechanism involve simple geometric considerations.

V-slot. In practice, the sides of the slot exposed to the pressure of the pin are lined with a hard material which can be replaced when it becomes worn. These linings are shown as heavy lines marked g in Fig. 24.

As shown in Fig. 26, the lay-out of the mechanism may be considered in terms of a simple problem in geometry, namely: Given the centers of the driving and driven wheels, C_1 and C_2, find a point X on the center line C_1C_2 such that $C_1X_1 = C_1X$, $C_2X_2 = C_2X$, and X_1X_2 is perpendicular to C_2X_2. If these conditions are satisfied, the driving pin will enter the V-slot tangentially, which is the most favorable condition, and it will be at the bottom of the V-slot when the driven member has been turned through 90 degrees.

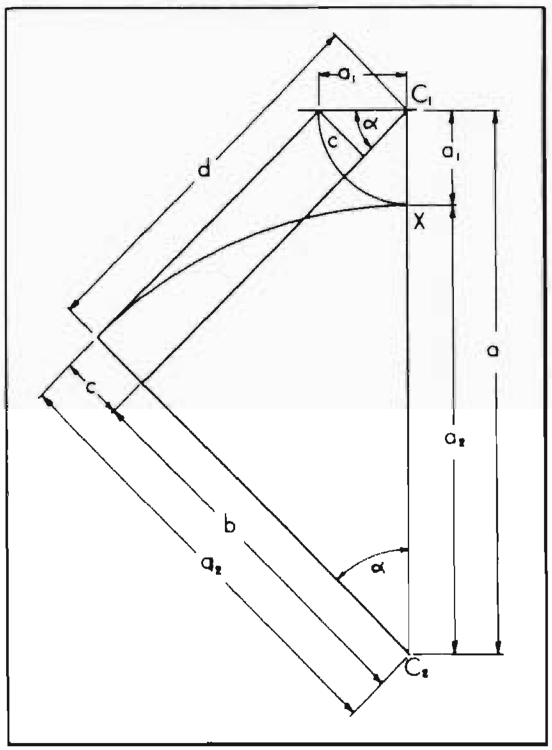


FIG. 27. Simplified lay-out of the mechanism illustrated in Fig. 24, which is employed to determine its proportions.

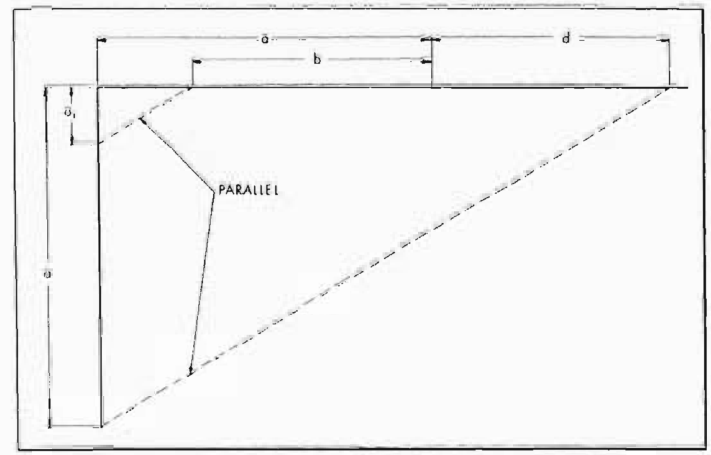


FIG. 28. Graphical method of determining length a_1 shown in Fig. 27.

In Fig. 27, the basic geometric form shown in Fig. 26 has been redrawn with two auxiliary lines added. From the illustration,

$$\frac{d}{a} = \frac{c}{a_1} \tag{1}$$

$$c = a_2 - b = a - a_1 - b \tag{2}$$

Substituting in Equation (1) the value of c from Equation (2), and simplifying,

$$\frac{d}{a} = \frac{a - a_1 - b}{a_1}$$

$$a_1 d = a^2 - aa_1 - ab$$

$$a_1 d + aa_1 = a^2 - ab$$

$$a_1(a + d) = a(a - b)$$

$$\frac{a_1}{a} = \frac{a - b}{a + d} \tag{3}$$

This last equation suggests the graphical method shown in Fig. 28 as a means of finding length a_1 in Fig. 27. Instead of this graphical solution, however, an analytical method may be used: Since $b = a \cos \alpha$ and $d = a \sin \alpha$, these values for b and d may be substituted in Equation (3) to obtain Equation (4):

$$\frac{a_1}{a} = \frac{1 - \cos \alpha}{1 + \sin \alpha} \tag{4}$$

Table 1 is a tabulation of $\dfrac{1 - \cos \alpha}{1 + \sin \alpha}$ for use in Equation (4), and covers angles from 30 to 60 degrees. From general design considerations, an angle $\alpha = 45$ degrees is best.

Table 1. Values of $(1 - \cos \alpha)/(1 + \sin \alpha)$ Corresponding to Various Values of α

α, Degrees	$\dfrac{1 - \cos \alpha}{1 + \sin \alpha}$	α, Degrees	$\dfrac{1 - \cos \alpha}{1 + \sin \alpha}$
30	0.08931	46	0.17759
31	0.09427	47	0.18367
32	0.09932	48	0.18981
33	0.10445	49	0.19601
34	0.10964	50	0.20227
35	0.11493	51	0.20858
36	0.12028	52	0.21495
37	0.12571	53	0.22138
38	0.13121	54	0.22786
39	0.13677	55	0.23441
40	0.14242	56	0.24101
41	0.14812	57	0.24766
42	0.15389	58	0.25436
43	0.15972	59	0.26113
44	0.16561	60	0.26795
45	0.17157		

The angular displacement, angular velocity, and angular acceleration of modified Geneva mechanisms having 60-, 90-, and 120-degree engagement angles are shown in Figs. 29, 30, and 31. The velocity and acceleration curves for each of these mechanisms are based on the driving member having a uniform angular velocity of 1 radian per second (9.55 rpm). The velocity curves shown were obtained by graphical differentiation of the displacement curves, and the acceleration curves by graphical differentiation of the velocity curves, since the equations involved in an analytical solution were not easy to handle.

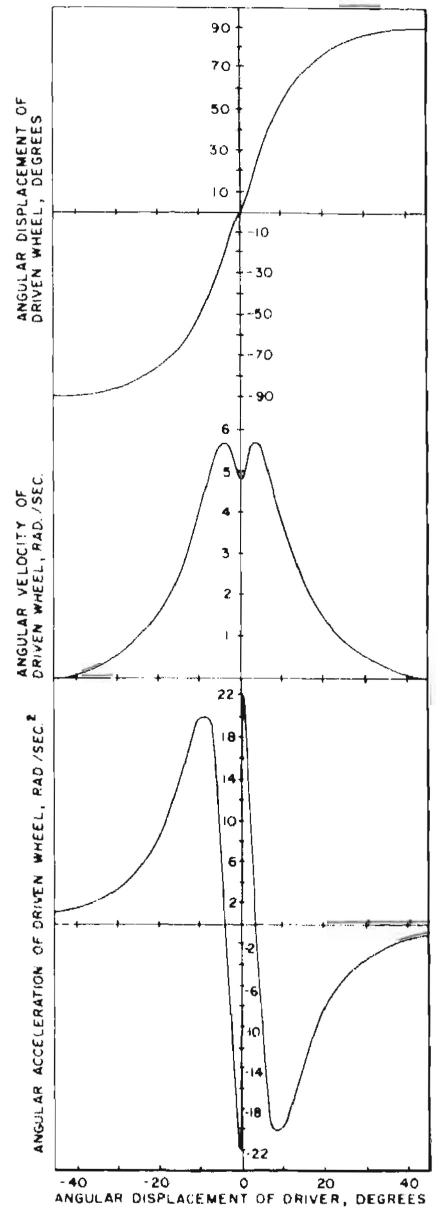


FIG. 29. Displacement, velocity, and acceleration diagrams for a half-revolution Geneva mechanism having an engagement angle of 60 degrees.

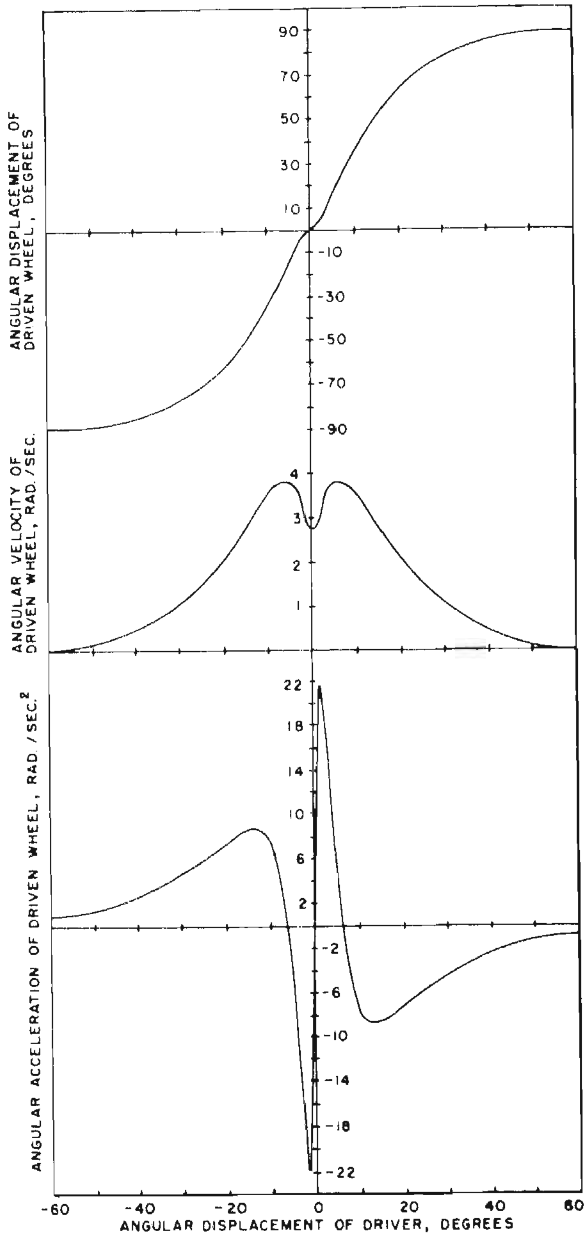


FIG. 30. Displacement, velocity, and acceleration diagrams for a half-revolution Geneva mechanism having a 90-degree engagement angle.

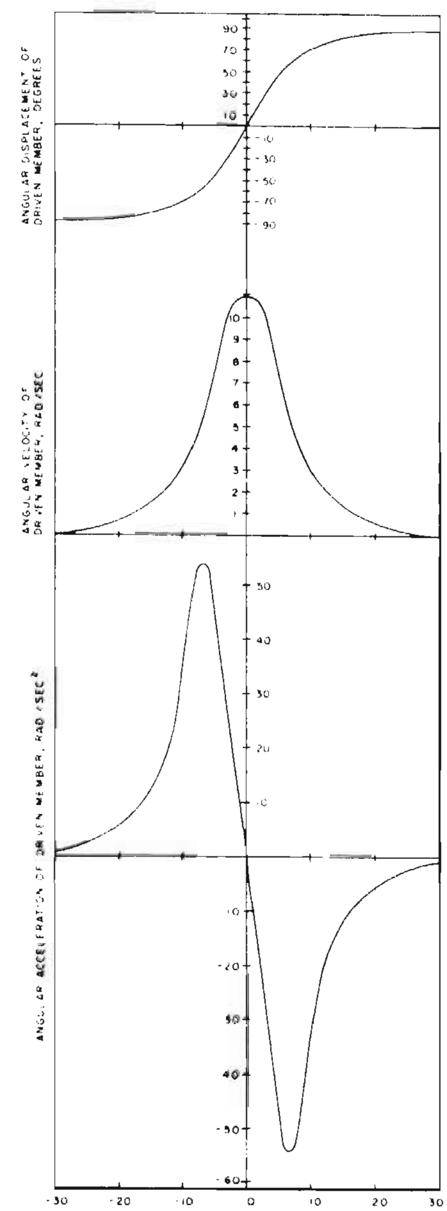


FIG. 31. Displacement, velocity, and acceleration diagrams for a half-revolution Geneva mechanism having an engagement angle of 120 degrees.

The displacement curves have a slight "bend" near the center position which is responsible for two maximums and a minimum in the velocity curves near the center position. The "hump" in the velocity curves decreases with the engagement angle, and disappears entirely when the latter is 60 degrees. In practice, this change in velocity occurring over a very short portion of the cycle will cause some roughness in the operation of the mechanism around the center position.

The curves also show that the maximum velocity of the driven member, for angles of engagement greater than 60 degrees, is not achieved at the center position. The angular velocity in the center position equals a_2/a_1 which value is $[(1 + \sin \alpha)/(1 - \cos \alpha)] - 1$. Table 2 gives values of these velocities for a number of engagement angles from 60 to 120 degrees.

Table 2. Angular Velocity of Driven Member in Center Position Based on 1 Radian per Second Angular Velocity of Driver

Angle of Engagement, Degrees	Velocity, Radians per Second	Angle of Engagement, Degrees	Velocity, Radians per Second
60	10.9695	92	4.6309
62	9.6078	94	4.4445
64	9.0684	96	4.2684
66	8.5739	98	4.1017
68	8.1207	100	3.9438
70	7.7009	102	3.7943
72	7.3139	104	3.6522
74	6.9548	106	3.5171
76	6.6213	108	3.3886
78	6.3115	110	3.2660
80	6.0214	112	3.1492
82	5.7512	114	3.0377
84	5.4981	116	2.9314
86	5.2609	118	2.8295
88	5.0382	120	2.7320
90	4.8285		

The velocity and acceleration curves in Figs. 29, 30, and 31 may be used to obtain the velocity and acceleration values for

N revolutions per minute of the driving member by multiplying the ordinates on these curves by $\left(\frac{\pi N}{30}\right)$ and $\left(\frac{\pi N}{30}\right)^2$, respectively.

Ninety Indexes per Minute

Simple in design, the high-speed indexing mechanism illustrated in Fig. 32 is designed for rotary type transfer machine applica-

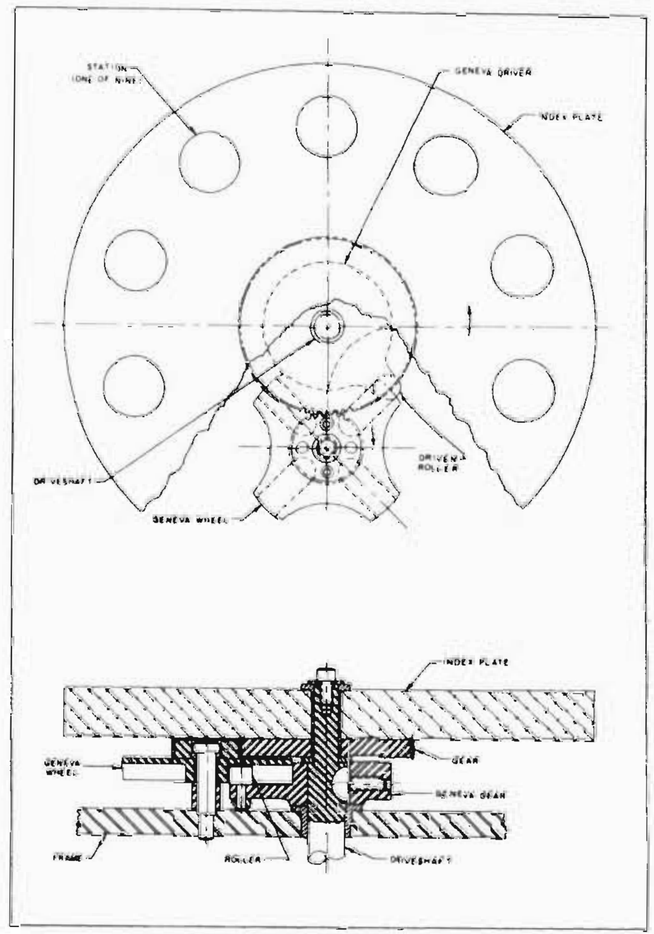


FIG. 32. Driven roller indexes Geneva wheel which rotates the index plate at a slower rate through reducing gears.

tions in which the index plate and the driveshaft are concentric. This arrangement permits tool slides at any or all stations to be actuated from cams mounted on the driveshaft. Each revolution of the driveshaft indexes or cycles the machine to the next station. Operational speeds up to ninety or more indexings per minute can be achieved.

Indexing is in a period equal to one-quarter of the total cycle time for each station regardless of the number of stations.

While this design can be readily adapted for light machining, stamping, assembly and inspection operations, it was first used on an inspection machine. In the original equipment, the part to be inspected is hopper fed and automatically loaded, gaged, rotated, gaged, rotated and gaged again in six consecutive stations. Ejection occurs at either of the four remaining stations depending on the results of gaging. All operations are actuated by three cams mounted on the driveshaft.

A front view of the original mechanism is shown with part of the index plate cut away in the upper drawing on the facing page. The Geneva drive is seen after having just completed an indexing movement. The lower drawing shows the drive at exactly its midpoint in the indexing motion.

All machine functions center around the driveshaft, on which any number of cams can be mounted. Keyed to this shaft, the Geneva driving member moves the driven Geneva wheel through a 90-deg. arc in 90 deg. of its own travel. The concentric diameter of the driver and the concave cutouts in the driven member mesh as shown to locate the Geneva wheel radially during the remainder of the machine cycle time at each station.

A pinion, concentrically mounted on the Geneva wheel with screws and dowels, meshes with a gear similarly attached to the index plate. The index plate and gear are mounted on the driveshaft by means of a single sleeve bearing.

In the original machine, the pinion has 28 teeth and the gear has 70 teeth. As a result, the index plate rotates 36 deg. with every 90-deg. movement of the Geneva wheel. Suitable gears could, of course, be selected to produce the angular rotation necessary for the other members of index plate stations.

Sleeve bearings and the Geneva drive provided sufficient radial accuracy for the purpose of the original machine. Needle roller bearings and an auxiliary shot bolt operated from a cam on the driveshaft could yield greater positioning accuracy.

CHAPTER 4

Overload, Tripping, and Stop Mechanisms

Mechanisms which automatically operate to stop an operation when overload occurs, to trip and start a new sequence or operation when a certain position or part of a cycle is reached, or to bring an operation to a halt at the end of a given cycle or when a given amount of motion has occurred, are described in this chapter. Other mechanisms performing similar functions are described in Volumes I, II and III of "Ingenious Mechanisms for Designers and Inventors."

Shock Absorber for a Rotating Shaft

Shock loads are isolated from the driving gears by the mechanism shown in Fig. 1. Driving gear *C* rotates gear *B* which has a slide fit over shaft *A*. Stud *D* which is rotatably attached to *B* transmits motion to collars *H* which in turn transmit motion to shaft *A* through compression springs *G′* and *G* and bracket *E*. Springs *G′* and *G* are located over rods *F′* and *F* which can slide in retaining holes in bracket *E*. Collars *H* retain the spring.

Should shaft *A* receive a shock, bracket *E* will be caused to rotate with respect to gear *B* and one or the other spring will be compressed. As the springs transmit the motion, the force of the shock will be limited.

An Overload Slipping Ball-Clutch

The diagrams in Fig. 2 show the effective design, construction and operation features of an adjustable slipping type of ball-clutch, which was successfully incorporated in the original drive

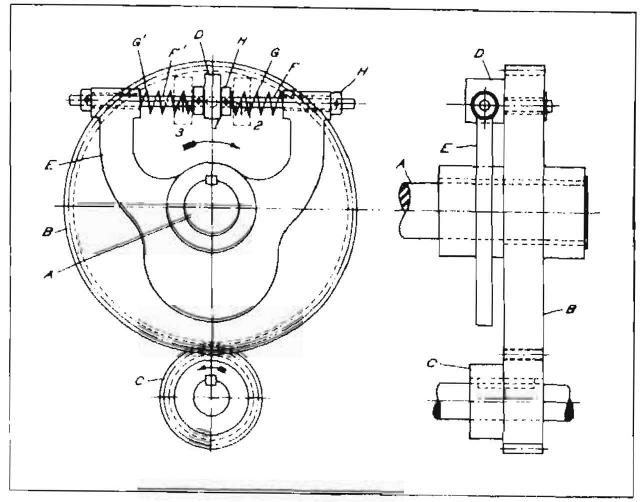


FIG. 1. The mechanism is able to isolate shock loads from the gearing without interfering with the over-all timing of the shaft.

transmission of a machine to safeguard the driven elements against overloading. The clutch was required to replace an existing positive chain-sprocket type clutch for transmitting the drive between two shafts mounted in axial alignment with each other.

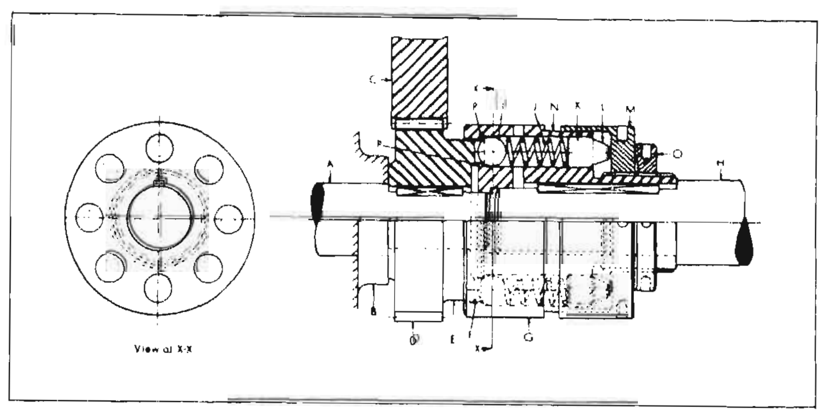


FIG. 2. Contained balls *I* are pressed by spring *J* into notches in a flange of gear *D*. Tension is adjusted by rotating nut *M* which shifts part *K*.

The new clutch had to be easily adjusted to transmit a range of different torques and also for setting to slip at various predetermined loads, which varied within wide limits according to the particular operation carried out on the machine. It was also essential for the clutch to be an entirely self-contained and compact unit so that it could be preset on a special fixture to slip at a given load, before installing the unit in the machine. The clutch had to operate with equal facility and efficiency in either direction of rotation, at different speeds, and to accommodate itself for slight axial floating movements of the driven shaft.

Referring to Fig. 2, *A* is the main shaft and is mounted in the horizontal bearing *B*. It is driven at different forward and reverse speeds by the train gear *C* meshing with pinion *D* keyed and permanently secured to the shouldered end of the shaft. The right-hand side of the pinion has an integral flange *E* which is slightly hollowed out on its end face to leave a narrow annular band at the periphery. A series of radial vee shape serrations, *F*, of identical size and shape and at equal pitch spacings apart, is milled across the annular band as shown.

In the illustrated example, the sides of the serrations are inclined 37½ degrees relative to the centre axis of the pinion that is, with an included angle of 75 degrees. This important dimension can, of course, be varied within certain limits in accordance with the load to be transmitted and the magnitude of the overload at which the clutch is required to slip. The size of the serrations is also determined by the diameter of the driving balls engaging therein.

The cylindrical case-hardened steel body *G* of the ball clutch is keyed to the driven shaft *H*, but it is made slightly longer than the shouldered end of that member so that the short, smaller diameter concentric portion of its bore is a slip fit over the adjoining end of shaft *A* projecting beyond the pinion *D*, as shown. The purpose of this arrangement is to maintain the body of the clutch perfectly concentric with the pinion for ensuring the smooth and accurate engagement of the balls in the pinion serrations. The left-hand end of the body is recessed a small

depth to admit the serrated portion of flange *E*, the outside diameter of which has a tight clearance fit in the recess. This overlapping part of the body serves to enclose the serrations and prevents the ingress of dirt and cuttings.

With this particular example, eight hardened and ground steel balls, *I*, are employed, each being fitted closely in a drilled and reamed hole passing axially through the body. The holes are spaced exactly 45 degrees apart around the same pitch circle, the diameter of which is equal to the pitch diameter of the serrations in the annular band of the flange *E*, thus the balls are disposed radially so as to engage centrally in the width of the serrations, as shown in the half-section diagram. The number and diameter of balls may be varied to suit loading requirements. A stiff coil spring *J* is interposed in each hole behind the ball and backing against the hardened steel plunger *K* fitted in the opposite end of the hole. All the plungers are the same overall length, and the conical portion *L* normally projects about ⅜ inch beyond the body. The end face of each plunger is slightly domed and well polished after hardening.

The right-hand end of the body is reduced in diameter and threaded to receive the hardened steel sleeve *M* of the same outside diameter as the front end of the body. The sleeve is deeply bored at one side to be a close fit over the reduced portion *N* ground on the outside of the body. By fitting the sleeve over the body at that point its correct and accurate location relative to the body is not determined by the fit in the threads. The eight plungers bear simultaneously against the inner left-hand face of the sleeve; thus as that member is adjusted longitudinally, all the springs *J* will be compressed or expanded the same amount. A smaller threaded ring *O* is also screwed on the body behind the sleeve for locking the latter member in any desired setting.

One of the main disadvantages of ordinary type ball-clutches is that when the driving and driven elements are separated, the balls can move completely out of the body of the clutch, and for that reason they generally cannot be preset on the bench or in a fixture, since there is no easy means available for holding the spring-loaded balls in the correct operating position. With the design of clutch described, this limitation is eliminated in a simple yet effective way.

Before machining the shallow recess in the left-hand side of the body, the eight axial holes for receiving the balls and spring were drilled, starting in from the right-hand end of the body and extending to a carefully predetermined depth, namely, to within about 3/8 inch of breaking through the opposite end. The recess was then bored to a controlled depth to ensure breaking into the eight holes a certain amount so as to leave the small lips *P* as shown in the half-sectioned view. When the balls correctly mesh in the serrations for driving purposes, the lips *P* are approximately 1/32 inch clear of the balls, thus allowing them to make full contact. As the body is moved axially away from the pinion, a very small amount, each ball is pressed against the lips and thus cannot move farther out of the body, which member can then be removed without fear of displacing or losing the balls and springs.

To set the removed body for slipping at a different overload, it is simply mounted on a keyed plug fastened in a fixture and sleeve *M* is adjusted in the appropriate direction, during which the balls will remain pressed against the lips *P*; thus the setting operation is simple, rapid and reliable.

To meet another application, this design of ball-clutch was slightly modified for transmitting the drive between two shafts in conditions where the driven shaft was required to operate for certain periods in a slightly axial off-set relationship to the driving shaft. The amount of eccentricity of the shafts varied from zero to 0.050 inch. This requirement prevented the use of any type of rigid clutch or one employing a chain and sprockets.

The modification consisted of shortening the overall length of the clutch body to equal that of the shouldered end of the driven shaft on which it was to be fitted. That eliminated the short, smaller diameter bearing portion at the left-hand end of the body for fitting over the end of the driving shaft. The shallow recess in that end of the body was also machined suit-

ably larger than the diameter of flange *E* to allow for the above-mentioned degree of off-setting.

Safety Devices Protect Slides Against Overloads

In the design of a machine, it may be important to protect a slide and its elements against excessive overloads. Where the slide is driven from an oscillating shaft, either of the two safety devices illustrated in Figs. 3 and 4 proves highly satisfactory. For clarity, the slides themselves have been omitted from each illustration.

In Fig. 3, the movement of the slide, connected to the right end of link *A*, is transmitted from an oscillating drive-shaft *B* through a lever *C*. Lever *C* swings in an arc, an equal distance to each side of vertical. The lever and link are joined by a pin

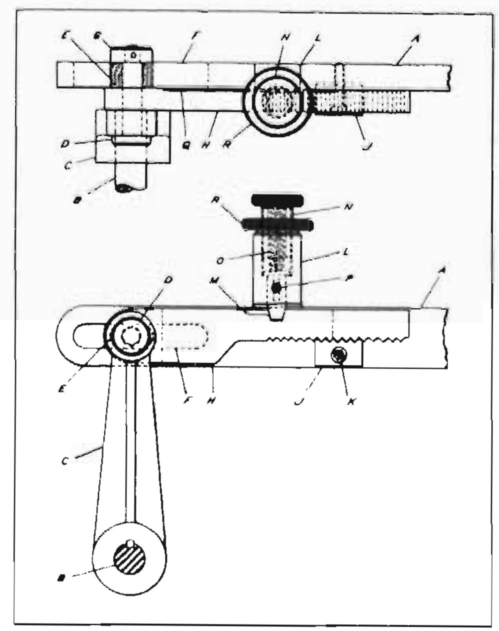


FIG. 3. The original path of transmission through this safety device is resumed automatically when the overload has been removed.

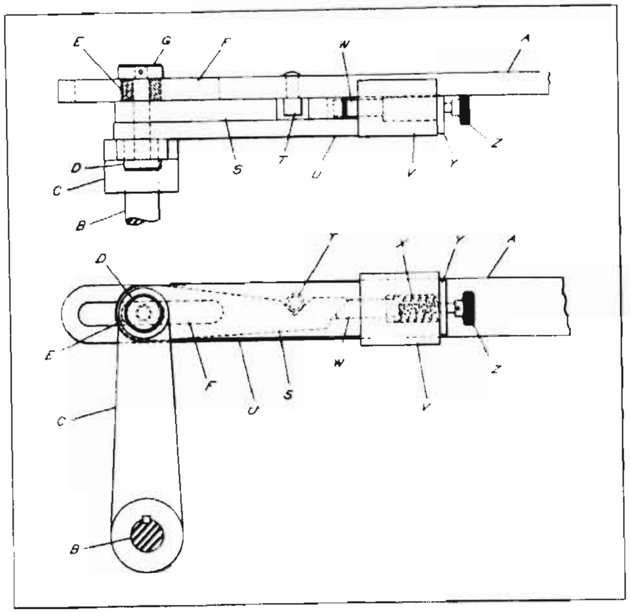


FIG. 4. This safety device can be used where a stud projecting above the links would be impractical.

D which passes through close-fitting holes in the end of the lever and in a bushing E located in a slot F in the link. A collar G is doweled to the protruding end of the pin, keeping the bushing in position.

The pin also passes through a hole in the left-hand end of a second link H. This link is shorter and somewhat narrower than link A. Also, its right half is further reduced in width, and has a fine-pitch tooth rack machined out. The teeth have a 45-degree side angle and engage mating teeth cut along the upper edge of a rectangular block J. This block is thicker than link H so that it can be registered in a shallow depression in the face of link A to which it is secured by a set-screw K.

Welded to the top edge of link A is a hollow stud L. A detent M in the bottom of the stud has two flat surfaces fitting a tapered slot cut across the top edge of link H. The top of the stud is tapped to receive an adjusting screw N. A blind hole in this screw contains a spring O which forces the detent to re-

main in the slot, thus keeping the tooth rack and block in engagement. Within the stud, a key P prevents the detent from rotating.

In the normal operation of the slide, there is no independent travel of the bushing E in the slot F. Instead, the path of transmission from lever C to the slide is through link H and block J to link A. Should the slide become overloaded, further movement of lever C and link H in either direction will raise the detent out of the channel, and the tooth rack will disengage block J.

Lever C and link H are then free to continue their movements, with the bushing E now traveling back and forth in the slot F. (The length of the slot is made slightly more than the stroke of the slide.) Since no movement is transmitted to link A, the slide is protected.

Once the overload is removed, the original path of transmission is resumed automatically. Adjusting screw N is pre-set to have spring O impart just enough pressure to permit the detent to be raised at a specified amount of overload. A jam nut R maintains the setting of adjusting screw N.

The safety device shown in Fig. 4 can be used where, because of space limitations, a stud cannot be located over link A. Here, link A, drive-shaft B, lever C, pin D, bushing E, slot F, and collar G are the same as their counterparts in Fig. 3, and similarly identified.

The left end of link S (corresponding to link H, Fig. 3) fits over pin D. The upper edge of link S tapers to the right, and also forms a 90-degree vee around a hardened pin T pressed into the face of link A. A third link U, also fitting over pin D, is considerably longer than link S, and has an integral block V at its right-hand end. The back of this block straddles the upper and lower edges of link A.

A plunger W, contained in a hole in the block, has a 45-degree point, matching a 45-degree slope on the end of link S. A spring X, which is retained by a cover plate Y, causes the plunger to exert a pressure contact against link S so that the sides of the vee normally bear on pin T.

In operation, the path of transmission from lever C to the slide is through link S and pin T to link A. Should the slide become overloaded, the resistance of A, acting through I pushes S down and temporarily pushes W.

Lever C and link S are then free to continue their movements, with the bushing E now traveling back and forth in the slot F. No movement is transmitted to link A. But unlike the first device, the original path of transmission here is not resumed automatically once the overload is removed. Instead, the knurled knob Z on the end of the plunger shaft must be pulled out while lever S is raised to position. The rear of the knob is milled flat in order to clear link A and keep the plunger point in alignment with the slope on the end of link S.

Safety Overload Mechanism Permits Adjustable Dwell on Reciprocating Drive

Instant, safe, and automatic disengagement of the drive for a reciprocating machine slide when subjected to excessive load is obtained by means of the mechanism illustrated. A useful feature of this safety overload mechanism is that the period of dwell at each end of the reciprocating stroke can be varied by a simple adjustment. Also, the drive is instantaneously and automatically re-engaged when the overload has been removed. The mechanism is smooth and quiet in operation.

Driving lever A (see Fig. 5), which is fastened at its lower end to an oscillating shaft (not shown), swings through a constant arc. The upper end of the lever is bored to be a free swiveling fit on the cylindrical boss B of bracket C. Collar D is pinned to the boss to retain the lever without binding. The rectangular body of bracket C is slotted to hold rectangular sliding member F. Slide F is retained in the slot by a plate G, which is secured to the body by four screws.

Connecting rod H can slide through a hole in F for a distance determined by the positions of lock-nuts J. The length of dwell varies with slide distance. The end of the connecting-rod is attached to the reciprocating slide of the machine (not shown) by shaft K.

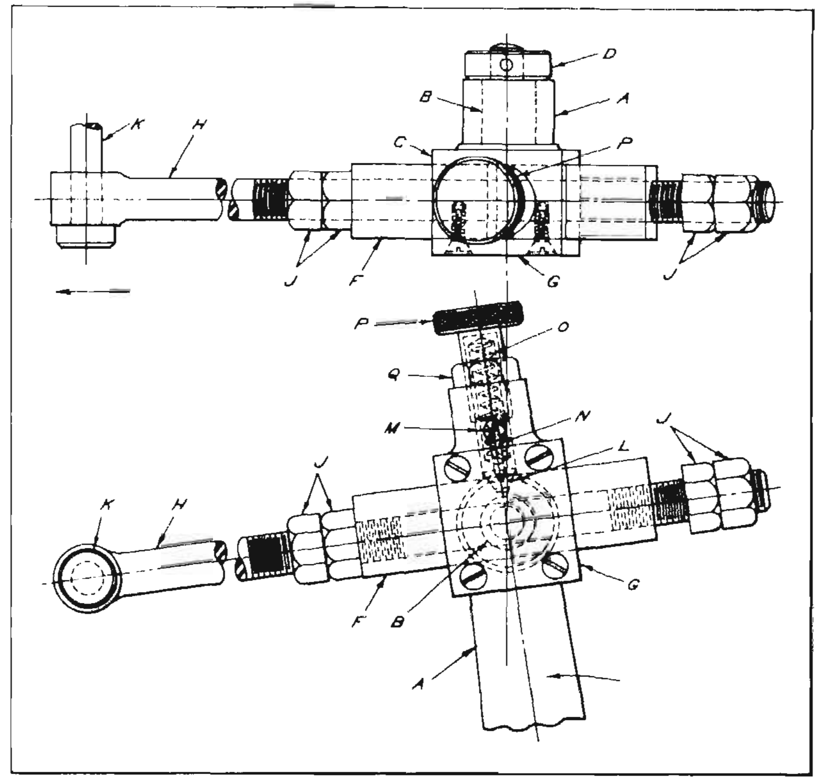


FIG. 5. The drive from oscillating lever A to a reciprocating machine slide which is attached to shaft K is disengaged by a spring-loaded plunger L when overloads are applied to the slide.

Slide F is connected to bracket C by a spring-loaded plunger L, the tapered nose of which fits into a V-notch machined across the slide. The plunger is a sliding fit within a boss on top of the bracket. It is prevented from rotating by a dog-point set-screw M which enters a shallow keyway N cut along the slide of the plunger. Spring O is seated in a blind hole in the plunger, and retained by a knurled-head adjusting screw P, which engages a threaded hole in the bracket boss. Screw P is held in any desired setting by lock-nut Q.

In operation, slide F and connecting-rod H move with driving lever A, thus reciprocating the machine slide. However, when additional resistance is offered to the horizontal movement of the machine slide — whether on its forward or return stroke — the plunger L will be forced out of the notch in slide F, thus disengaging the drive. When the overload is removed, the plunger will snap into the notch again, and the drive will be re-engaged.

By varying the compression of the spring, which is accomplished by screwing P into or out of bracket C, the point of loading at which the drive will be disengaged can be changed. Also, a heavier or a lighter spring can be used to suit requirements.

Device Reduces Initial Acceleration of Flying Shear — *Shockless Startup of Inertia*

A device designed to reduce the force necessary to set a flying shear in motion is shown in Fig. 6. In practice, flying shears employed on cold-roll forming machines may be actuated by any of several methods. One arrangement that results in accurate cutting-to-length of the roll-formed sections allows the shearing mechanism to be started and pulled by the rolled strip. In many cases, however, the strip does not have sufficient stiffness to overcome the inertia of the shear without buckling. By permitting acceleration to occur over a longer period, the mechanism shown in the illustration decreases the initial force required to bring the shear up to the speed of the strip and, thereby, reduces the tendency of the strip to buckle.

The mechanism is arranged as follows: A light flag, or lever, mounted on a track at the outgoing end of the strip runout table, is connected to the flying shear A by a cable to a hinged lever B, see Fig. 6. This lever, in turn, is attached to a second lever C. As the roll-formed section starts the flag moving, a roller D, mounted on lever C, pushes against a bar E attached to the machine base. This setup will gradually accelerate the shear carriage, which is mounted on rollers or on machine ways. The shearing mechanism is pneumatically or hydraulically operated to sever the rolled strip.

When lever C reaches the vertical position, it comes into contact with a stop and the roller leaves bar E. The shear carriage

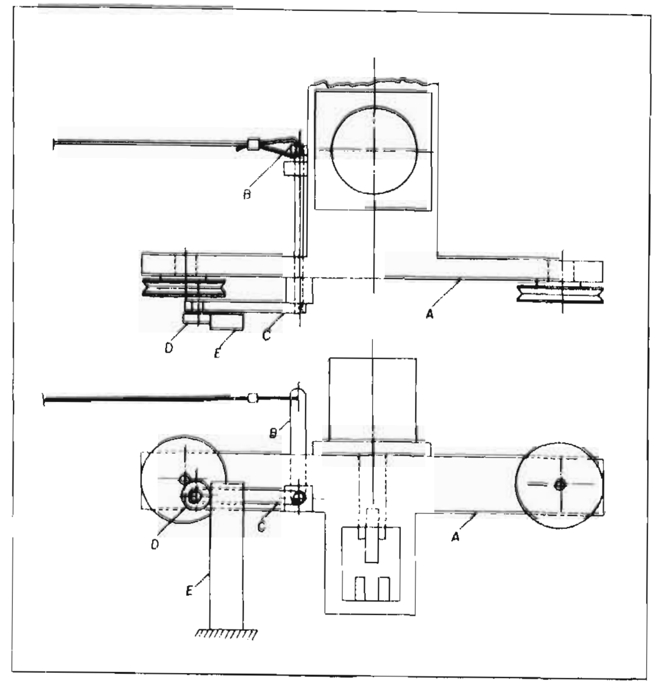


FIG. 6. Device that reduces the initial force necessary to accelerate a flying shear. When lever B reaches a horizontal position the shear A and the strip are moving at the same speed.

has then been accelerated to the same speed as the moving strip and is being pulled by it. At this point, the shearing mechanism is immediately actuated by a micro switch and the flag is triggered, releasing it from the end of the strip. A spring returns the shear to the starting position and the cycle is repeated. Lengths of the lever arms may be varied to suit the acceleration required.

Springs Cushion Shock Loads in Gear Drive

Shock loading of a gear train in either direction can be greatly reduced with the arrangement shown in Fig. 7. The mechanism features a drive that operates under increasing spring

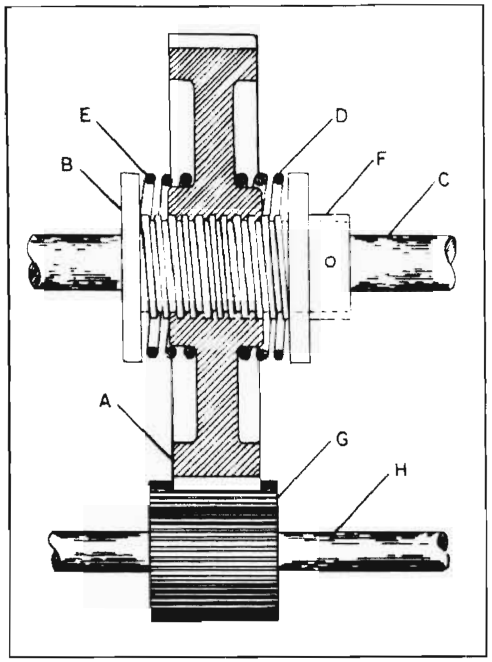


FIG. 7. Gear drive that employs springs to cushion the effects of shock loading in either direction.

pressure as greater angular displacement or slippage occurs between the driving and the driven shafts.

A square thread is machined into the bore of gear *A* to fit a threaded sleeve *B*, which is press-fitted on shaft *C*. A short compression spring *D*, gear *A*, and a second similar spring *E* are mounted on the sleeve in that order. This assembly is held together by a stop-collar *F*, which is pinned in place on the sleeve, the pin passing through the collar, the sleeve, and the shaft.

The pinion *G*, which is mounted on shaft *H*, should be made wide enough to insure full tooth contact with gear *A* as it moves to either the right or the left. To prevent excessive lateral movement of gear *A*, the springs should be under compression when it is centered on the pinion. The springs should be heavy enough to prevent the shock load from causing the gear hub to jam against either the collar or the flange of the sleeve. This

condition would render the shock-absorbing feature of the mechanism inoperative.

In normal operation, when the machine is started under load, the gear will move laterally and compress one spring until the initial loading is overcome. As the machine picks up speed, gear *A* will move back toward the center of the pinion, the distance depending on the running load applied by the machine. Intermittent shock loading of the machine will cause gear *A* to move back and forth on the threaded sleeve. Loading of the drive in the opposite direction will cause the gear to compress the other opposing spring with similar results. If the shock loading is in one direction, the mechanism may be modified to operate with only one spring.

This arrangement is being successfully employed on a drive for a tumbling barrel, the springs having been selected by trial and error. A 0.250-pitch thread and a 10-pitch gear train are used. The driving gear *G* and the driven gear *A* are 3 and 9 inches in diameter, respectively, each gear being mounted on a ⅞-inch-diameter shaft. A ¾-hp, 60-rpm, geared head motor drives the machine.

Torque-Controlled Drive Release for Tapping

A chuck that automatically disengages the drive when a pre-set torque is applied to the tap is shown in Fig. 8. When properly adjusted, this device can effectively reduce tool breakage.

Spindle *A* has equally spaced slots for axial location of three keys, the outer edges of which engage the inclined bases of keyways in sleeve *C*. The flanged lower end of this sleeve is fitted with three bushings with tapered bores for part of their lengths. For driving purposes, these bushings are held in engagement with steel balls *D* by the action of the compression spring *E*. The balls are housed in pockets in the base of the large-diameter bore at the upper end of body *F*.

Spring *E* can be pre-set to release at the maximum torque that can be applied to a tap by adjusting the internally threaded cover *G*, which threads onto the body *F*. When this setting

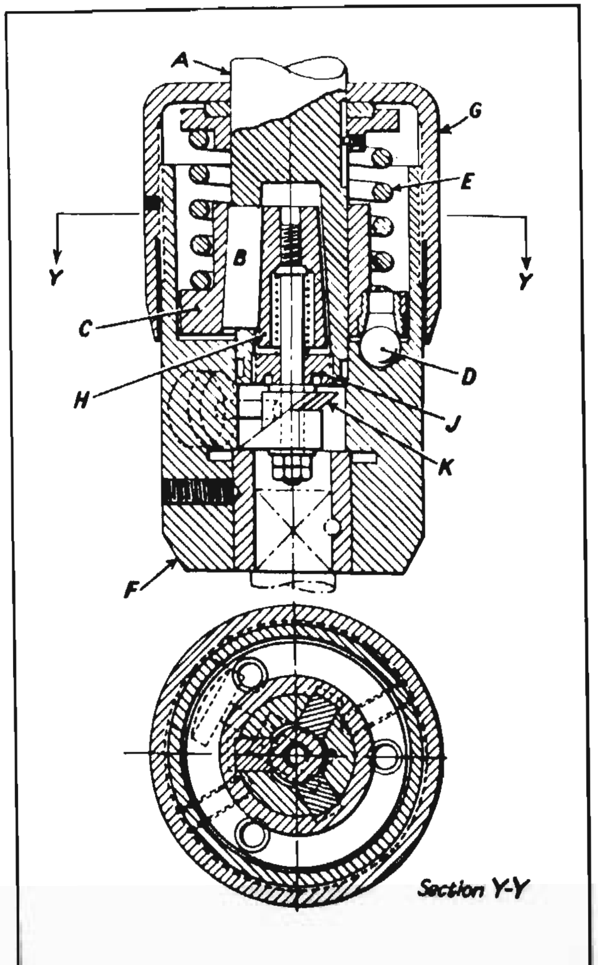


FIG. 8. Excess torque transmission is prevented by sleeve C rising to clear the driving balls D. Re-engagement is prevented by the outward movement of keys B.

has been made, the cover is secured to the body by means of a set-screw.

The inner edges of keys B make contact with cone H, which is housed in the tapered bore of spindle A. When the pre-set torque is applied to the tap, continued rotation of the spindle causes the sleeve C to be moved upward against the action of

spring E. When the bushings rise high enough to clear balls D, the drive is disengaged. At the same time, cone H is caused to move upward by the action of a second compression spring, the lower end of which bears against the threaded plug J in the spindle. As a result, the keys are moved radially outward by the wedging action between their inner edges and the cone. In this way contact is maintained between the keys and the bases of the keyways in sleeve C. The cone and the keys have a taper of 1 in 20 and are self-locking. The result is that any downward movement of the sleeve and, consequently, engagement of the drive are prevented.

When the spindle has been stopped, the sleeve can be released to bring the bushings and the balls D into engagement again by depressing a plunger which passes through a cross-hole in the body F. This action causes wedge K, which is attached to the plunger, to be moved at right angles to the spindle axis and into contact with a mating wedge. The latter is carried on the lower end of a pin attached to member H and is therefore caused to move downward to give the releasing action.

The tap is mounted in a bushing in the lower end of body F. A cross-pin, which passes through a hole formed partly in the square-shaped end of the shank and partly in the bore of the bushing, holds the tap in place.

Mounting Provides Double Action for Compression Spring

A compression spring can be mounted so that either a push or a pull will put the spring under compression. In Fig. 9, spring A is contained between washers B and C, and is secured to the end of shaft D by fillister-head screw E.

While shaft D remains stationary, force is applied to shaft F, which is pinned to one end of step-bored thimble G. The other end of the thimble has an internal thread engaging a slotted externally threaded bushing H which has a slide fit over shaft D.

When shaft F is pulled to the right, thrust is transmitted through the set-screw and washer B against the left end of the spring. Or, when shaft F is pushed to the left, thrust is trans-

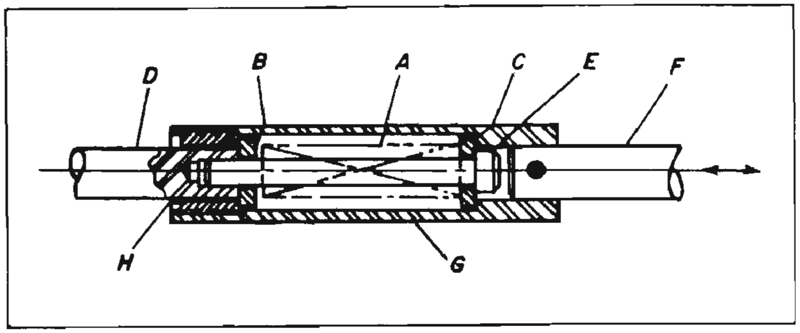


FIG. 9. The mounting permits spring A to be compressed when shaft F is moved in either direction.

mitted through the step in the thimble bore and washer C against the right end of the spring. Thus, the spring is compressed either way shaft F moves.

Linkage for Combined or Independent Lineal Travel

A critical element and an auxiliary element of a mechanical system can be linked to a common actuator in such a way that either the two operate together, or the critical element operates alone, if the auxiliary element is jammed.

Heart of the linkage device is cylinder A, held in a fixed position in bracket B (see Fig. 10). Within the cylinder are two tubular slides — inner slide C and outer slide D. Cable E, entering the cylinder from the left, is joined to plunger F. Around the

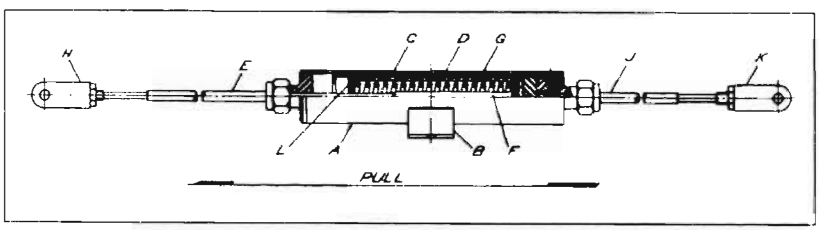


FIG. 10. During the second ¼ inch of travel, slide D normally moves with slide C. If the auxiliary element is jammed, neither slide moves, and spring G is compressed.

plunger is coil spring G. The other end of this cable is connected to the critical element (not shown) through clevis H.

Another cable J, entering the cylinder from the right, is joined directly to slide D. The other end of this cable is connected to the auxiliary element (not shown) through clevis K.

The device functions as follows: When the critical element is operated, it pulls the plunger to the left. The spring, being heavy, resists compression under normal load, and restricted by end plate L, causes the inner slide to move as a unit with the plunger for ¼ inch of free travel. At this point, the end plate contacts the bottom of the outer slide.

Then, for a second ¼ inch of travel, the plunger, inner slide, and outer slide move as a unit. Since cable J is joined to the outer slide, the auxiliary element operates with the critical element during the second ¼ inch of travel as is desired.

On the other hand, assume that the auxiliary element is jammed. Then, in the second ¼ inch of travel, the plunger will travel independently of the inner slide, which has now been immobilized by the stalled outer slide. During this movement, the spring is compressed by the pull on the plunger through the operation of the critical element.

A practical application of this device is found in fighter aircraft. The critical element of the system is a seat ejector, and the auxiliary element, a headrest latch release. In this instance, the spring around the plunger has a 100-pound pre-load. Spring rate is 100 pounds per inch. The additional 25-pound load, when created by the freezing or jamming of the headrest latch release, is reduced to about 1.4 pounds at the input because of a 1 to 18 ratio of the pulling force.

Pressure Governor for Handwheel of Lathe Tailstock

The danger of exerting excessive axial pressure when adjusting a lathe tailstock for a "between centers" operation can be averted with the governor illustrated. It can also be used to advantage where a small drill or reamer must be supported in a relatively large tailstock — increasing the sensitivity of the feed and thus reducing tool breakage. The device is of simple construction and retains all the standard parts of the tailstock.

In Fig. 11, which shows the right-hand end of the tail-stock, the spindle *A* has a sliding fit with the casting *B*. The end of the spindle is threaded and engages the feed-screw *C* which rotates in the cap *D*. A key *E* holds the spindle from turning as it is advanced or retracted. The handwheel *F* is removed from its normal position near the end of the feed-screw and is replaced by a cast-steel disc *G*, feathered to the feed-screw by an existing Woodruff key *H*. Shaft *J*, threaded to the right-hand end of screw *C*, retains disc *G*.

Sleeve *K* clears disc *G* and tight slide fits shaft *J* and is retained on *J* by pinned collar *L*. Handwheel *F* is rigidly attached to sleeve *K*.

At a point on its periphery, the disc *G* has a 90-degree V-notch *O*. A bossed section *P* of the sleeve contains a detent plunger *Q*. This plunger is kept in position in the V-notch by a spring *R*. During normal operation, handwheel *F*, sleeve *K*, disc *G* and feed-screw *C* turn as a unit.

Should the advancing spindle meet with excessive pressure, plunger *Q* will rise out of the V-notch. Thus, if a revolving center is supported by the tailstock spindle, the handwheel advances the spindle until the center engages the conical opening in the end of the work. Further advance is resisted by the center, and the detent plunger automatically rides out of the V-notch.

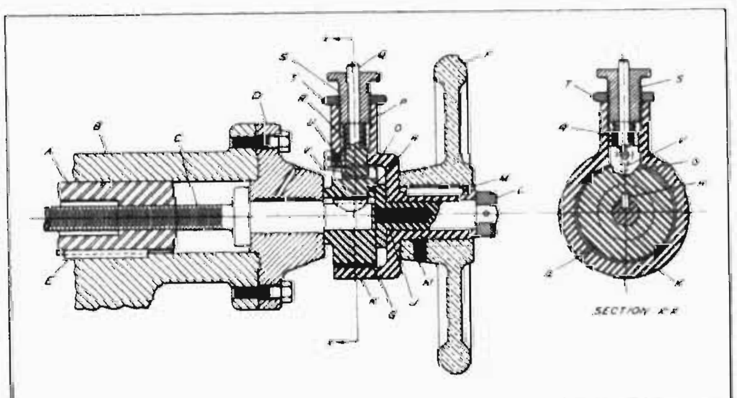


FIG. 11. Transmitting the drive from the handwheel *F* to the feed-screw *C* through a detent plunger *Q* limits the axial pressure that can be exerted.

The load that the detent plunger can carry before it will disengage is adjusted by the vertical setting of a bushing *S* threaded to the inside of the bossed section. A lock-nut *T* serves to maintain the setting. For a revolving center, the mechanism is set at a point that is well below the safe loading on the balls and races, but will permit proper support for the work. This point is readily established by testing the revolving center for free rotation under load.

When the detent plunger is disengaged, its alignment with the V-notch is maintained by a set-screw *U* engaging slot *V* in the plunger. Should it be desired to operate the handwheel without the regulating action of the governor, the bushing *S* can be lowered to fully compress the spring *R*.

Press Clutch Automatically Disengaged after Required Number of Strokes

An indexing die was designed for piercing a number of equally spaced holes in drawn sheet-metal parts or shells. The shells were rotated a partial revolution with each stroke of the press, and the operator disengaged the press clutch when the required number of holes had been pierced. The human element soon became apparent by the number of parts with one or more holes missing. It was then decided to control the number of strokes per shell automatically: the mechanism shown in the accompanying drawing was designed for this purpose.

In the end views seen at the bottom in Fig. 12 and in Fig. 13, the crankshaft bearing and other details have been omitted for clarity. The press used for this operation has a sliding key clutch. Flywheel *A*, rotating in the direction indicated by the arrow, transmits motion to the crankshaft *B* through the sliding key *C*. The spring-loaded key is engaged and disengaged by the action of a wedge-ended lever *D*, which is operated by a foot-pedal through the clevis-rod *E*.

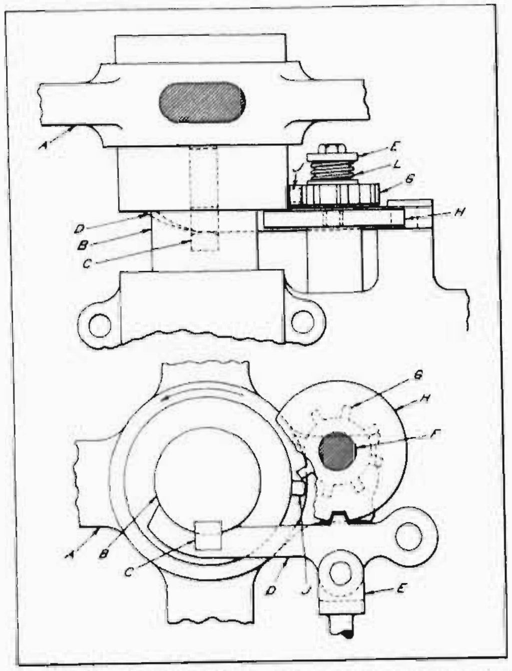


FIG. 12. Mechanism employed on a punch press to disengage the clutch automatically after eight strokes of the ram.

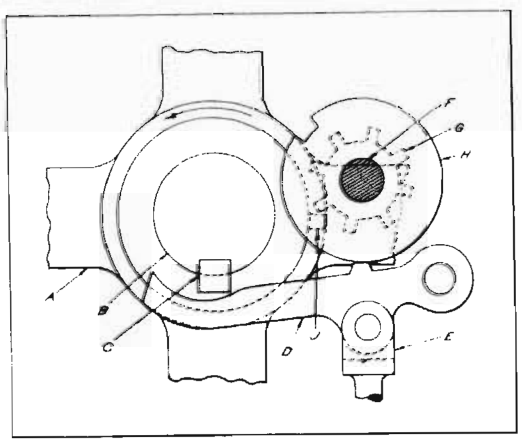


FIG. 13. Spring-loaded sliding key C shown in its engaged position, with the flywheel A rotating crankshaft B. When the ram has completed eight strokes, the projection on top of lever D will enter the groove in disc H, thus disengaging the clutch.

In Fig. 12, the sliding key is shown disengaged from the flywheel, and the press ram is stationary, permitting loading. A toothed disc G and an attached grooved disc H, are free to rotate on a stud F, which is attached to the side of the press frame. Spring L applies frictional resistance to the rotation of discs G and H. Pin J, inserted in the crankshaft flange, contacts one tooth of disc G with each revolution of the crankshaft. A projection on the upper edge of lever D engages the groove in disc H.

When the work-piece has been placed in the die, the foot-pedal is depressed and lever D is withdrawn from the groove in sliding key C, as seen in Fig. 13. The key then engages the rotating flywheel and locks the crankshaft to it. As soon as the crankshaft starts to rotate, pin J contacts one tooth on disc G, causing it and disc H to rotate a partial revolution. The projection on lever D is now in contact with the periphery of disc H, thus preventing the wedge end of lever D from entering the groove in key C. Continued rotation of crankshaft B causes disc G to be rotated, one tooth per revolution, until the projection on lever D again enters the groove in disc H. In this position, lever D is permitted to return to the disengaging position.

The number of strokes per cycle is governed by the number of teeth on disc G and the number of grooves in disc H. It is necessary, however, that the number of teeth on disc G be a multiple of the number of grooves in disc H.

Safety Attachment Designed for a Reciprocating Movement

On a machine for producing a wire product, short lengths of wire are drawn into the machine from a magazine, moved in one direction for one operation, and then in the opposite direction for the next operation. Occasionally, a defective wire fails to release properly from the magazine, resulting in breakage. An attachment designed to eliminate jamming in cases of misfeed is illustrated in the accompanying drawings.

A plan view and front elevation of the attachment during normal operation are shown in Fig. 14. Rod B imparts a re-

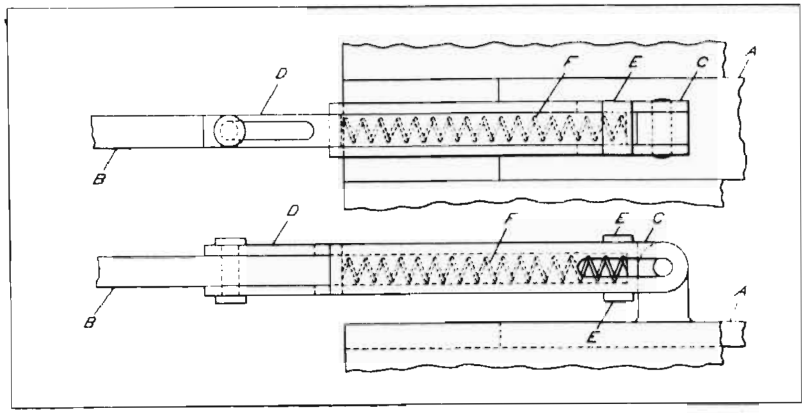


FIG. 14. Plan view and front elevation of attachment for eliminating jamming of the machine in the event of a misfeed.

ciprocating motion to slide *A*, which carries the feeding mechanism, not shown. Part *C* is a U-shaped piece, with its right-hand end slotted and supported on a pin mounted in lugs on slide *A*. Part *D* is another U-shaped piece, also slotted at its left-hand end and supported on a stud in bar *B*. Welded to part *D* are two straps *E*, which support the closed end of part *D* on part *C*. Part *D* is free to slide within part *C*, and both parts *C* and *D* are free to slide on their supporting studs.

A spring *F*, nested in the hollow box section formed by the assembly of parts *C* and *D*, is under compression at all times, so that the closed end of part *D* is held in contact with the lugs on slide *A*. Also, the closed end of part *C* is held in contact with the right-hand end of bar *B*. In the position illustrated, which is normal operation, movement of bar *B* is transmitted to slide *A* through spring *F*. The tension of this spring, determined by trial, must be sufficient to transmit the required movement and still permit further compression without resulting in the breakage of parts.

The operation of the mechanism under abnormal conditions is illustrated in Fig. 15. In the view at the top, slide *A* has been prevented from moving to the right because of a defective wire failing to release from the feeding mechanism. As bar *B* continues its movement to the right, movement of part *D* is prevented by its contact with the lugs on slide *A*, and the stud in bar *B* slides in the slots in part *D*. Also, since the end of bar *B* is in contact with the open end of part *C*, movement of the bar causes this part to move with it, against the compression of the spring. Thus, part *C* slides over the pin mounted in the lugs on slide *A*. As bar *B* again moves to the left, the assembly returns to its normal position, as shown in Fig. 14.

If slide *A* is prevented from moving to the left with bar *B*, a condition such as the one illustrated at the bottom in Fig. 15 is produced. Bar *B*, in moving to the left, draws part *D* with it, thus compressing the spring against the closed ends of parts *C* and *D*. When the abnormal condition has been corrected, the parts again assume the positions shown in Fig. 14.

In a subsequent application of this arrangement having space limitations, it was impossible to provide an attachment long enough to accommodate a compression spring of the required length and strength. Consequently, the design was altered, as shown in Fig. 16, to permit the use of externally supported compression springs. In general, the design has not been changed except by the addition of side extensions on part *D*, and

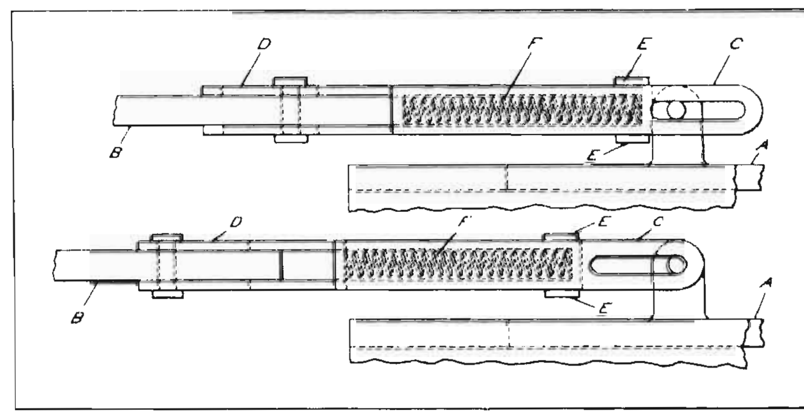


FIG. 15. Operation of mechanism under abnormal conditions. When slide *A* is prevented from moving to right, part *D* contacts lugs on slide *A*, as seen in Fig. 14. When slide *A* is prevented from moving to left, spring *F* is compressed, as illustrated in Fig. 15.

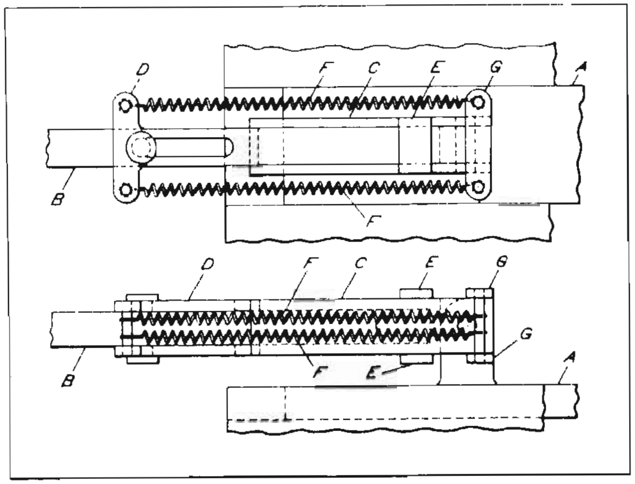


FIG. 16. Modified design of mechanism shown in Fig. 14, necessitated by space limitations that prevented use of long, strong spring.

members G to part C, for attaching springs F. Operation is the same as in the original design.

Although both of these designs will perform equally well, the choice must be governed by space limitations, as the latter design will require greater width, as indicated by the plan view in Fig. 16. However, the fact that the springs are supported externally in this design may prove a definite advantage in that the tension may easily be adjusted to suit the requirements.

CHAPTER 5

Locking, Clamping, and Locating Devices

Means of positively locking a mechanism, clamping a workpiece or part, and locating work in the proper position for some operation to be performed on it, or locating a carriage or table in the correct loading position, are described in this chapter. In some cases, the locking or clamping operation is performed automatically while in others hand operation is required. Similar devices are described in Volumes I, II and III of "Ingenious Mechanisms for Designers and Inventors."

Intermittent Drive with Reverse-Locking Feature

An arrangement that prevents reversal of an intermittent drive during the dwell period is shown in Fig. 1. Compact and quiet in operation, the device was designed for use as a high-speed indexing mechanism in shoe processing machinery.

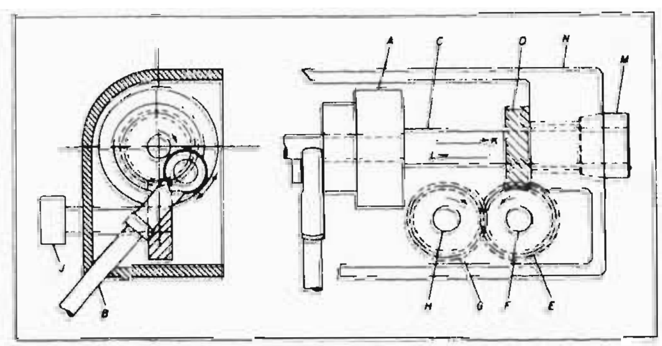


FIG. 1. Cone brake prevents reversal of this intermittent drive during the dwell period.

A roller type indexing clutch *A* is driven with a reciprocating motion by connecting-rod *B*. This causes shaft *C* to rotate clockwise intermittently. Right-hand helical gear *D* is fastened to shaft *C* and engages with a second right-hand helical gear *E* attached to shaft *F*. Left-hand helical gear *G* mounted on shaft *H* also meshes with gear *E*. Two feed rollers *J* are fastened to the ends of shafts *F* and *H*, which rotate intermittently in opposite directions.

When clutch *A* is driving gear *D*, a thrust is produced laterally in shaft *C* in the direction *K*. During the dwell portion of the indexing cycle, any attempt to make gears *G* and *E* the driving gears will produce a lateral thrust and displacement of shaft *C* in the opposite direction *L*.

A cone brake *M* attached to shaft *C* takes advantage of this reversal of thrust to lock the shaft and the rollers during the dwell period. Thrust in the reverse direction *L* causes the cone to be displaced slightly and become tightly held in a mating conical bore in frame *N*. An increase in the reverse thrust only increases the holding power of the cone brake. During the following index cycle, the thrust produced on shaft *C* by helical gear *D* is again in direction *K*, and cone *M* is released from the conical bore. Lateral movement of shaft *C* is held to the minimum displacement necessary to free the cone.

Mechanism for Adjusting Size of "Iris" Drawing Dies

A mechanism designed for adjusting the size oı hexagonal carbide-insert dies used in cold-drawing hexagonal stock is here illustrated. This mechanism enables one die to be used for drawing a large number of sizes. Three master dies cover the range of all the hexagonal sizes drawn in a cold-drawing mill.

The main component of the die is a set of six carbide-insert pieces *A*, (see Fig. 2). As the stock is drawn through these insert surfaces, it is formed into the shape of a hexagon. The other members of the die act to support, adjust, or lock the carbide-insert pieces.

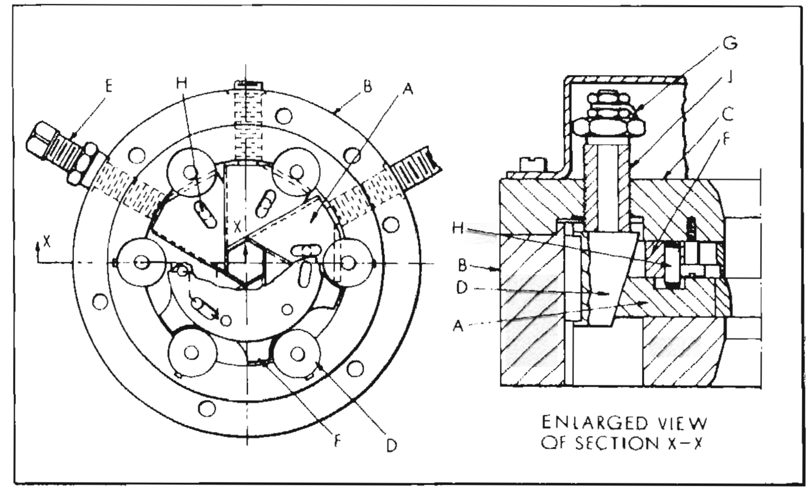


FIG. 2. Plan view and enlarged cross-section of the die showing the arrangement of the various components.

To understand the principle on which the adjustment of the carbide-insert pieces is based, reference should be made to Fig. 3. Here the relative movement of two of the six pieces is indicated by the arrows. Initially, the pieces are in the position indicated by the solid lines, forming two sides of the solid-line

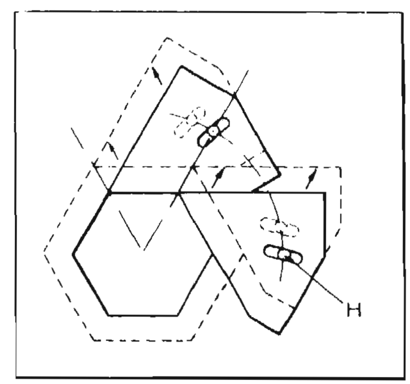


FIG. 3. Carbide-insert pieces are moved in a straight line from the solid- to the dotted-line positions as indicated by the small arrows. Pins acting in elongated slots effect this motion.

hexagon. If they are moved without rotation into the position shown by the dotted lines, they form two sides of a larger hexagon. The movement of these pieces is controlled by the action of pins *H* inside elongated slots in the pieces. The pins move along the circumference of a circle whose center is the center of the hexagon.

Referring to Fig. 2, the other members comprising this adjustable die are the body *B*, cover *C*, keyed bar wedges *D*, lock-screws *E*, and a disc *F* that carries pins *H*. Each of these members has a specific function to perform. Die body *B* holds all the component parts, so that the motion of the carbide-insert pieces can be restricted within required limits. Cover *C* supports disc *F*, which is free to rotate a small distance. Cover *C* also holds the hollow stud and nut arrangement *J*, which positions the wedges *D* laterally.

The purpose of the wedges is to provide a means whereby the carbide-insert pieces can be located properly. Adjustment is necessary when the drawing surfaces of the carbide-insert pieces wear. The lock-screws *E*, provide a locking function.

Once the die has been set, which is usually done in the maximum open position to minimize any error in shape, it is a rather simple operation to adjust it to the required size. Lock-screws *E*, together with their lock-nuts, are loosened. Three alternate keyed bar wedges *D*, the tops of which have indexing pointers *G*, are next loosened to unlock the six carbide-insert pieces. The purpose of the indexing pointers is to insure the proper locating of these three bar wedges for true hexagonal positioning of the insert pieces in the ensuing locking operation.

The other three alternate wedges are left undisturbed, in order that the true hexagonal shape of the die will be maintained in sliding the insert pieces along the flat surfaces of these wedges from one position to another.

Two knurled pins (not shown) rigidly fastened to disc *F* and extending through elongated slots in cover *C* are used to impart rotation to disc *F*. The pins *H* that are engaged in the slots of

the six carbide-insert pieces slide these pieces simultaneously into the new position. When in the required position, the three alternate wedges *D* are properly located by the use of their indexing pointers, and the six lock-screws *E* and their lock-nuts are tightened, thereby locking the die.

Three-Axis Adjusting Mechanism

For applications where a single supporting member must be adjustable in all directions, use can be made of the device shown in Fig. 4. This device consists primarily of a special eyebolt that can be swiveled around a spherical surface and locked in any desired position. The eyebolt can also be adjusted lengthwise.

Bracket *A* serves as an attaching component that is fastened to the object which requires an adjustment feature. Swiveling of the eyebolt *E* is accomplished after backing off nut *B*, thus relaxing washers *C* and allowing spherical washer *D* to slide around the spherical end of housing *F*. The pivot point is about the center of the spherical bushing *G*. Washer *D* carries belt *F*.

Lengthwise adjustment is accomplished by rotating spanner nut *H* to advance or retract housing *F*. Pin *J* rides in a slot of housing *F* and restrains rotation of the housing.

As shown, this mechanism permits adjustment to within any point in the space of a 2-inch cube.

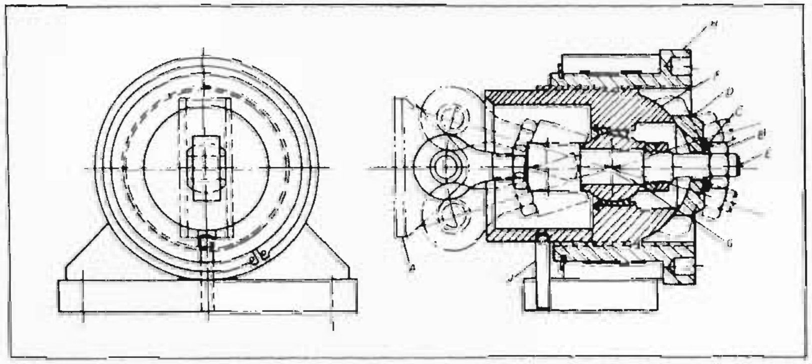


FIG. 4. Device which can be adjusted within any point in the space of a 2-inch cube.

Adjustable Disc-Stacking Magazine

For a special machine designed for an operation on leather discs, a fixture had to be provided to hold discs from 1 to 6 inches in diameter. Figure 5 shows an adjustable "nest" developed to meet this requirement.

An arm from a molding press moves into the position indicated by line P to pick up discs successively and then transfer them, one at a time, to a mold cavity located to the right of the stacking device. Regardless of size, the centers of the discs must always be in the same place.

In adjusting this magazine to suit a change in work diameter it is necessary to swivel arms A, B, and C inward or outward. Movement of the arms is accomplished by rotating threaded

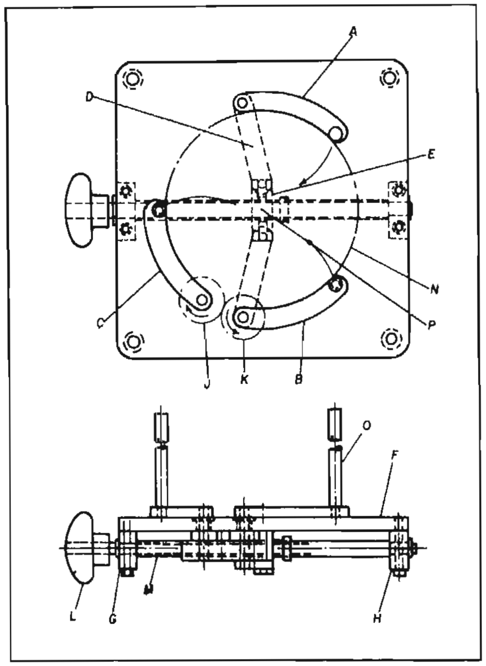


FIG. 5. Adjustable magazine for discs can accommodate work sizes from 1 to 6 inches in diameter.

shaft M to move block E laterally. Arms D are pivoted on this block. Their outer ends are attached to arms A and B. Gear K on the outer end of Arm B meshes with gear J. This gear is mounted on the pivot shaft of arm C.

Movement of arms D and gears J and K causes arms A, B, and C to move toward or away from the center of the stacking unit to suit discs of various diameters. Vertical rods O, mounted on arms A and B, allow for stacking discs on top of each other to convenient heights.

Tape Reel Has Quick Action and Constant Gripping Pressure

Tapes used in programming machine tools and in other operations are wound on spools, which, in turn, are positioned on a reel. Tapes and spools are made in different widths, but the spools necessarily have the same inside diameter.

To grip the spool, most reel designs involve the tightening of a cap-nut on a thread which is part of the reel spindle. The nut presses against a rubber cylinder which expands in the bore of the spool, securing it to the reel. Disadvantages of such action are that engagement and disengagement time is relatively long; the firmness of the grip depends on how much the cap-nut is tightened, which might vary from operator to operator; and some of the components of the reel have to be changed whenever the spool width is changed. An added shortcoming is that the clutching action can only be performed manually, and cannot be made automatic.

On the hand, the reel design proposed in Fig. 6 is quick acting, assures a constant gripping pressure, accommodates spools of different width without adaptation, and can be readily converted to operation by a solenoid.

The device is driven by a motor through a timing belt (not shown) running around pulley A feathered to shaft B. Hub C is pressed on the shaft. Flange E bolted to the frame is lined with a bearing which supports the shaft. This bearing extends through the bore of the hub. The portion of the shaft which

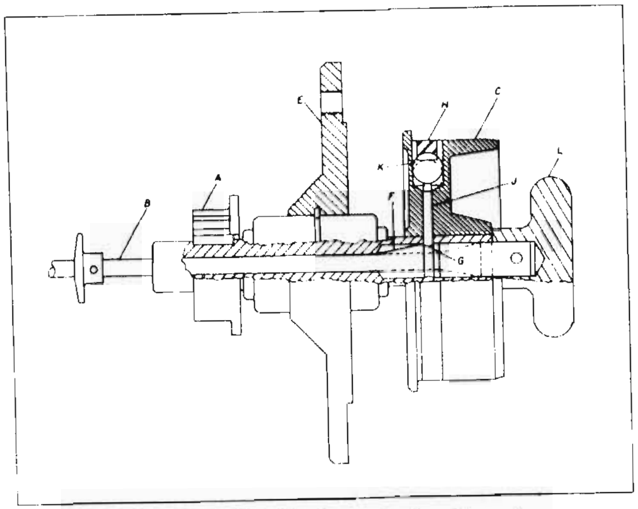


FIG. 6. With ball K up (solid line), part of rubber ring H is displaced; with ball down (broken line), all of rubber ring is contained in neck of hub C.

runs through the hub has a tapered surface F and an annular groove G.

Rubber ring H has a snug fit in a neck in the hub. Pressure is exerted around the hub by the ring, since the inside diameter of the ring is smaller than the diameter of the neck. There are six counterboard holes equally spaced radially in the neck, each containing a dowel-pin J and steel ball K. The rubber ring keeps the balls and pins in pressure contact with shaft B.

Reel spools have a slip fit over hub C. To position a spool on the reel, knob L, pinned to the shaft, is pulled to the right, and the rubber ring forces the balls and dowel-pins down radially on taper F. Since the rubber ring is now completely contained within its neck, the spool is able to slip over it.

To grip the spool, the knob is thrust to the left, the dowel-pins being forced out by the taper, settling in groove G in the shaft. (This is the position illustrated in Fig. 6.) Simultaneously, the balls move out radially, partly displacing the rubber ring in the neck. The displaced rubber fills the clearance between the

outside of the hub and the inside of the spool, and is sufficient to exert a firm grip on the spool. This grip remains constant from spool to spool, since it is outside the control of the operator.

Total radial movement of the dowel-pins is calculated so that the volume of the penetration of the balls in the rubber equals the clearance area. Relatively large tolerances are permissible, because of the compressibility of the rubber, which takes up the variations.

It is possible to connect the left end of shaft B to a two-directional solenoid for automatic operation. The solenoid has to be actuated only when a spool is being positioned or removed. A micro switch can be used to sense the axial position of the shaft, and the machine can be wired in series with the micro switch so that it will not start if the spool is not firmly gripped.

Quick-Acting Clamp with Wide Work Capacity

An unusual gripping and releasing mechanism which enables instant adjustment of a clamping jaw to suit widely different sizes of work is a feature of the special vise-like assembly fixture illustrated in Fig. 7. This clamp was primarily designed

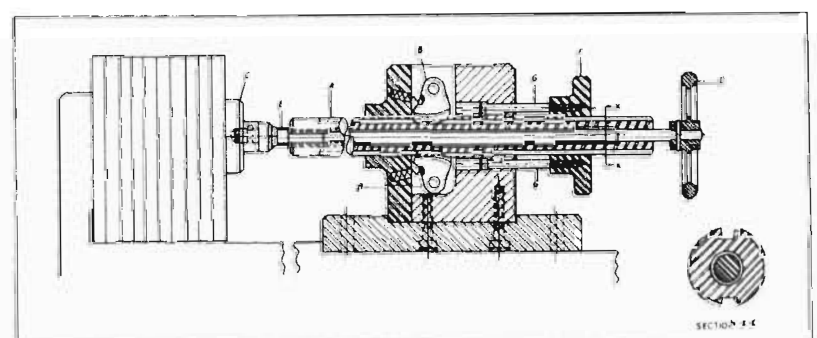


FIG. 7. Ingenious arrangement on this fixture enables quick clamping of work despite large variations in work thickness.

for holding packs of thin sheets that vary considerably in width. Packs vary in over-all width from approximately 5 to 16 inches.

The different pack sizes are made up by the operator in the assembly fixture, according to requirements. To minimize clamp setting time it was desirable to provide a clamp which, although hand-operated, would be capable of being adjusted rapidly. It had to be suitable for imparting a sufficiently powerful grip on packs of all sizes.

The principal working member of this device is a hollow ram *A* which can be readily slid to the right or left except when retarded by the action of pawls *B*. In loading the fixture, the ram is pushed by hand until clamp *C* bears against the pack of work sheets. Then handwheel *D* is revolved to apply pressure through rod *E*. This rod extends completely through hollow ram *A*. At the left-hand end, there is a threaded enlarged diameter on the rod which engages a thread in the ram.

Consequently, after clamp *C* has been positioned against the work, it is positively tightened by revolving handwheel *D*. This action causes an adjustment of the threaded portion of rod *E* in the threaded left-hand end of the ram, and thus exerts pressure on clamp *C* and on the work. This can happen because pawls *B* prevent any right-hand movement of the ram until they are released.

Release of the pawls is effected by striking a sharp blow against ring *F*. This causes ejector pins *G* to strike against the pawls with sufficient force to overcome the pressure of the springs which tend to force the pawls to the right. The pawls operate in grooves cut in ram *A*, as seen in section X-X.

In a modification the two pawls *B* are substantially of the same shape and size as those in Fig. 7, in respect to their contacting peripheries (see Fig. 8). However, each pawl has an integral tail at right angles to the contacting portion. In each case, the pawl tail passes through a clearance slot machined in the right-hand side of the fixture body.

Fastened to the top of the body over both slots is a steel plate which bridges the slot. Screwed into this plate is a fine-pitch

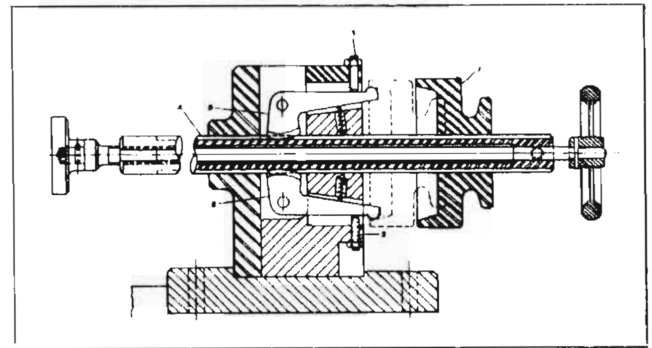


FIG. 8. Safety overload on this modified design guards against work distortion and damage.

headless set-screw, the lower end of which bears against the side of the pawl tail. A lock-nut secures the screw in any desired height setting. The purpose of the two screws *S* is to restrict the amount of swiveling movement of each pawl in the direction of its frictional contact with the ram grooves. A light compression spring bears against the inner side of each pawl tail so as to maintain the contacting peripheries of the pawls in a light frictional engagement with V-grooves of the ram.

Instead of two ejector pins, the ring mechanism *F* has a conical bore which is large enough to pass over the rounded tips of the pawl tails, as shown by dotted lines. When the ring is moved swiftly toward the body, the sides of the conical hole strike the pawl tails and cause the pawls to release.

Cam-Jaw Chucks for Twisting Rod

Two chucks having cam jaws furnish a powerful grip which twists steel rod used in reinforced concrete. One of the chucks, Fig. 9, anchors one end of the rod. The other, Fig. 10, rotates the opposite end of the rod to produce the required twist.

The anchoring chuck has a bracket *A* mounted on the frame of the machine. Rod *B* to be twisted is located against a pad *C* transversely movable within the bracket by means of a toggle arrangement with lever *D*. At two other points on its periphery the rod is under the pressure of knurled cam jaws *E* and *F*.

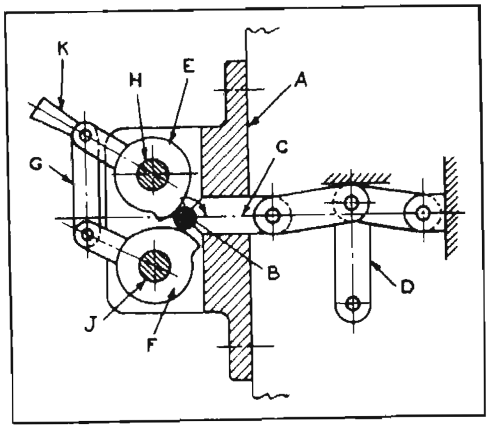


FIG. 9. This anchoring chuck prevents one end of the steel rod from rotating.

A link G joins the jaws so that they can pivot in unison around their respective shafts H and J. Extending from the link is an operating arm K.

When slipping a rod into position in the machine, lever D and arm K occupy the positions illustrated. The arm is then moved down, and the jaws pivot counterclockwise, to lock

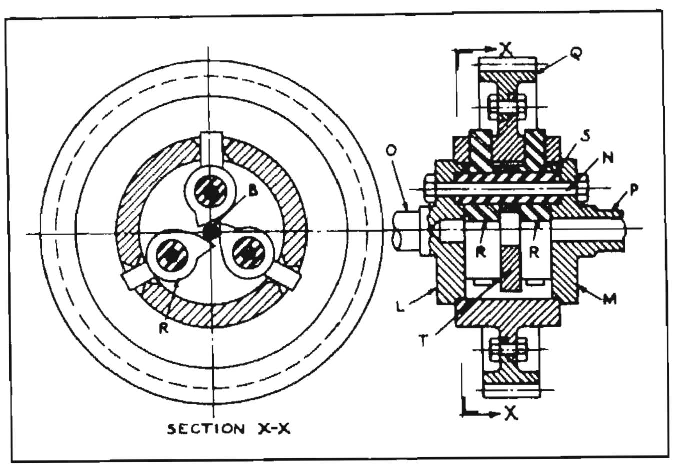


FIG. 10. The driving chuck rotates the opposite end of the rod to produce the required twist.

the end of the rod against the pad. As the machine starts to twist the opposite end of the rod, the cams tend to tighten their grip. To release the rod, lever D is lowered. This action causes locating pad C to retract.

The driving chuck, Fig. 10, has two cover plates L and M, tied together by bolts N. The outside of each plate is turned down to form integral bearing diameters O and P. Diameter P is bored to receive the end of rod B. Located between the cover plates is a large gear Q. This gear has two hubs; each is drilled radially and beveled at three points to receive the lobes of cam jaws R. The jaws pivot on bushings S, which are mounted over the bolts N. Also, the two rows of jaws are separated by a disc T.

When the machine is running, a pinion, which is engaged to the gear, rotates it in the direction indicated by the arrow. The cam jaws immediately pivot in and start twisting the rod, the opposite end of which is fixed in the anchoring chuck. To release the twisted rod, the gear is reversed momentarily. Like the cam jaws in the anchoring chuck, those in the driving chuck have knurled bearing surfaces to provide a better gripping action.

Variable Stroke and Quick-Action Lock for Reciprocating Slides

The device of Fig. 11 was designed to cause two slides to be moved by hand and then be securely locked in a predetermined position.

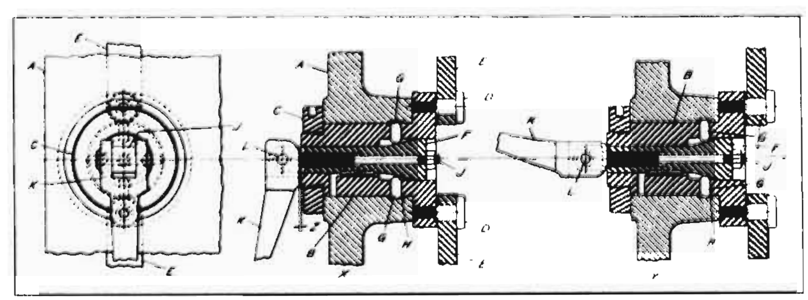


FIG. 11. Manually operated mechanism provides stroke adjustment and rapid locking for two opposed reciprocating slides.

The cross-section at X shows the construction of this manual drive mechanism and the positions occupied by working members when in the unlocked position. A section of the vertical machine wall *A*, on which an integral boss is located, is bored to receive flanged sleeve *B*. The portion of the sleeve extending beyond the machine wall is reduced in diameter and threaded for circular lock-nut *C*, which holds the sleeve in place and may be adjusted by means of a spanner wrench.

Two identical links, *E*, see Fig. 11, can pivot about shoulder studs *D*. The opposite ends of links *E* are attached to the slides.

Sliding within a hole bored through sleeve *B* is cylindrical plug *F* which is keyed in place to prevent independent rotation. The plug has a conical head which rides within a counterbore in the flanged end of the sleeve. The largest diameter of this head is ground with parallel sides for a short distance to provide a close sliding fit within the counterbore. This lends additional support to the head and helps to maintain its accurate alignment during locking movements. The surface of the conical head, which is formed at an angle of 10 to 12 degrees from the axis, should be hardened and polished smooth.

Four equally spaced holes are drilled radially through the side walls of sleeve *B* into the counterbore. These holes are the same diameter and are located in the same plane. Sliding freely within each hole is a pin *G*, both ends of which are rounded. All four pins must be accurately machined to the same over-all length and hardened.

The inner ends of the pins bear against the conical head of the plug while the outer ends extend into a shallow annular groove *H* which is machined concentrically in the bored hole in machine wall *A*. The width of this groove is slightly greater than the diameter of the pins, and the depth need be only about $\frac{1}{16}$ inch. The purpose of this groove is to prevent scoring the surface of the bore in contact with sleeve *B*.

A clevis-pin *J* is threaded into the center of plug *F*, fine-pitch threads are cut on the larger pin diameter at the clevis end. A standard hexagon nut is screwed on the opposite (right-hand) end of the clevis-pin to lock it in position. This arrangement permits both radial and endwise adjustment of the clevis-pin.

Operating lever *K* has a forked end for fitting over the end of pin *J*. The width of the lever end is almost the same as the smaller diameter of sleeve *B*. A cam-like curvature is formed on the upper right-hand corner of the lever as shown at X in Fig. 11. The cam radius increases gradually as the curve approaches the top surface of the fork. The thickness of the forked end of the lever, together with the location of pivot-pin *L*, are carefully determined so that a clearance *Z* of about 0.025 inch will be provided.

To lock sleeve *B* within its bearing hole, lever *K* is simply pivoted upward as shown at Y. This action causes plug *F* to be drawn to the left. The conical head of *F* contacts pins *G* forcing them outward and into annular groove *H*. Sleeve *B* is thus locked within the bearing hole in wall *A*, and further movement cannot be transmitted to the two links *E*.

To operate the machine-slides, lever *K* is depressed to the vertical position, thereby releasing the locking pressure. The lever, clevis, plug, and sleeve members can then be rotated in unison to impart the desired motion to the machine-slides.

Finger Holds Down Paper Stack on Printing Press

On printing presses, some type of suction device generally feeds the paper by lifting the leading edge of the top sheet in the stack and drawing the sheet forward into grippers. To avoid the tendency of the top sheet to pull the next sheet with it — as sometimes happens because of static electricity in the paper — a mechanical hold-down finger can be added to the press. Through a cam and double bellcrank construction, the finger operates in time with the suction device, and separates the top sheet.

Figure 12 shows three positions of the mechanism. In view X, the finger *A* holds down the stack of paper after the leading

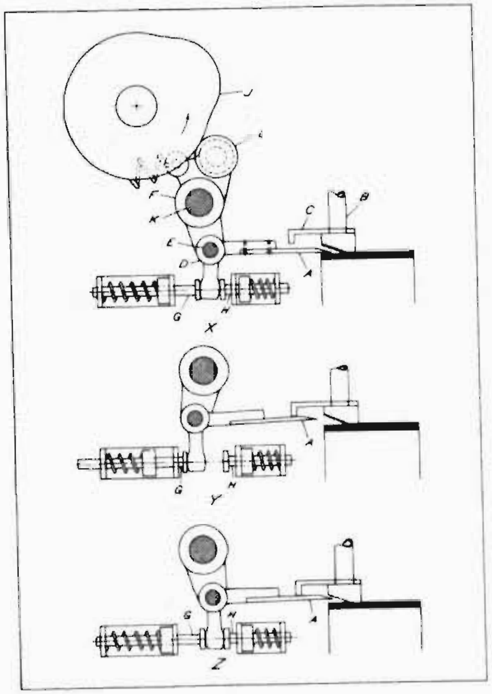


FIG. 12. Hold-down finger A is synchronized with suction cup B through cam J and bellcranks D and F.

edge of the top sheet has been lifted by suction cup B. The bottom of the cup is cut at an angle, as shown, so as to raise the sheet edge sharply. Guide C, fastened to the cup, moves up and down with it.

The finger is attached to one arm of small bellcrank D, which pivots on shaft E in the lower arm of large bellcrank F. Two spring-loaded plungers G and H control the position of the lower arm of the small bellcrank. Cam J, revolving continuously in time with the movement of the suction cup, causes the large bellcrank in turn to pivot on shaft K under the direction of follower L.

When the suction cup comes down on the stack, the finger is pulled back, and under the pressure of plunger G, it is forced upward against the guide, as in view Y. At this point, the lobe

of the cam bears on the follower, and the suction cup lifts the edge of the top sheet.

Then, as the lobe leaves the follower, plunger G loses control of the lower arm of the small bellcrank and plunger H takes over, as in view Z. The guide, meanwhile, forces the finger down, directing it into the lip formed by the raised edge of the top sheet. When the finger is fully in place, as in view X, the suction cup assembly raises the sheet into the grippers which feed it into the press.

Cycling is continuous, with the cam revolving once for each sheet fed. A feature of the mechanism is that the finger is located by the position of the feed cup, so is not affected by variations in the height of the paper stack.

Toggle-Action Drill Jig That Clamps Work at Four Points

In drilling hold-down bolt holes through the steam cylinder heads of duplex piston pumps, it was found that the location of the holes was often inaccurate. The original jigs employed for drilling such holes were simply flat plates of the same shape as the cast heads to be drilled. These bushing plates were equipped with vertical pads around their peripheries to form nests for the castings. However, due to variations in the size of the castings, many of the work-pieces fit loosely in the jigs, resulting in inaccurate location of the drilled holes. To overcome this difficulty, the drill jig seen in Fig. 13 was designed to accurately clamp the work at four points by means of a single toggle action.

The two clamping arms A are slidably mounted on bushing plate B by means of studs C. The central portion of these studs pass through large holes in the arms to permit their free movement. Pins D loosely fit in the centrally located projections of the clamping arms, and their lower, enlarged diameter ends are provided with flats to fit slots milled in the bushing plate. This permits the arms to pivot about these pins and to slide along the slots when operating handle E is rotated.

Cam F, which is rotated by handle E about stud G, is connected to clamping arms A by links H. These links can pivot

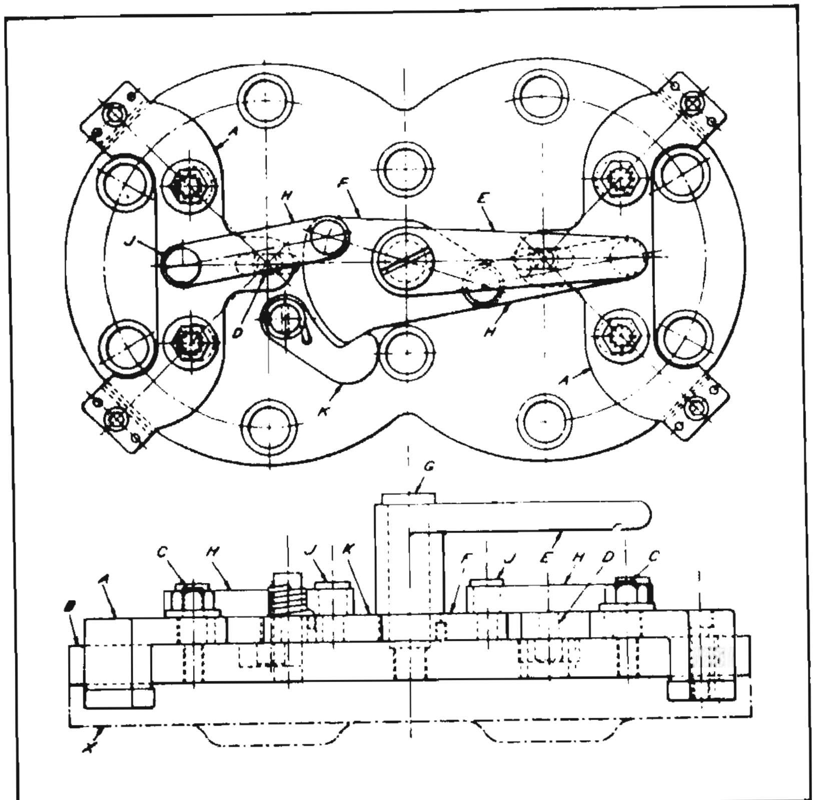


FIG. 13. Work-piece X, which is a cast steam cylinder head for a duplex piston pump, is rigidly clamped at four points in this toggle action drill jig.

about the loose-fitting studs J. A spring-loaded latch K holds the cam, levers, and arms in the work-clamping position shown (the loading position).

As the cam is rotated counter-clockwise, latch K will be rotated clockwise and links H will become aligned with each other. Clamping arms A are moved apart so that the jig can be placed over work-piece X. The cam is then turned clockwise to the position shown, and arms A are pulled together firmly to clamp the work for drilling.

Cam-Operated Stock Clamp for Piercing and Blanking Dies

Difficulty is often encountered in operating piercing and blanking dies if some means is not provided to keep the stock rigidly pressed against the back gages of the die. This is especially true if a high degree of accuracy is desired. The stock will usually weave when being pushed through the gages, or it will jump when struck by the punches. These conditions are particularly aggravating when handling heavy stock.

To overcome such trouble, a mechanism was designed which automatically presses the stock against the back gage without requiring any effort on the part of the operator. This device also has the advantage of reducing the number of scrapped parts.

Shown in Fig. 14 are two hardened and ground slides A mounted in ways at the front of the die-block B. These slides are spaced as far apart as possible. The lever C swivels in a clevis bracket D, which is screwed and doweled to the top shoe. A spring E in the top shoe keeps the lever up against a Z-shaped retainer F when the die set is in the open position.

As the press ram descends, the lower end of the lever C strikes the angular surface on the slide A, forcing the slide up against the stock. Of course, the device must be so designed that the slide is pressed firmly against the stock just before the punches enter the stock. On the up stroke, the slide A is retracted, allowing the operator to move the strip easily through the gages.

Swing Stop for Automatic Lathe

A unique mechanism for operating a swing stop on an automatic lathe is shown in Fig. 15. This mechanism permits the stop to be swung to a position in front of the headstock spindle to stop the axial feed of the bar stock at the beginning of the automatic cycle. Thereafter, the stop is held clear of the work, and, at a predetermined point in the cutting cycle, a mechanism

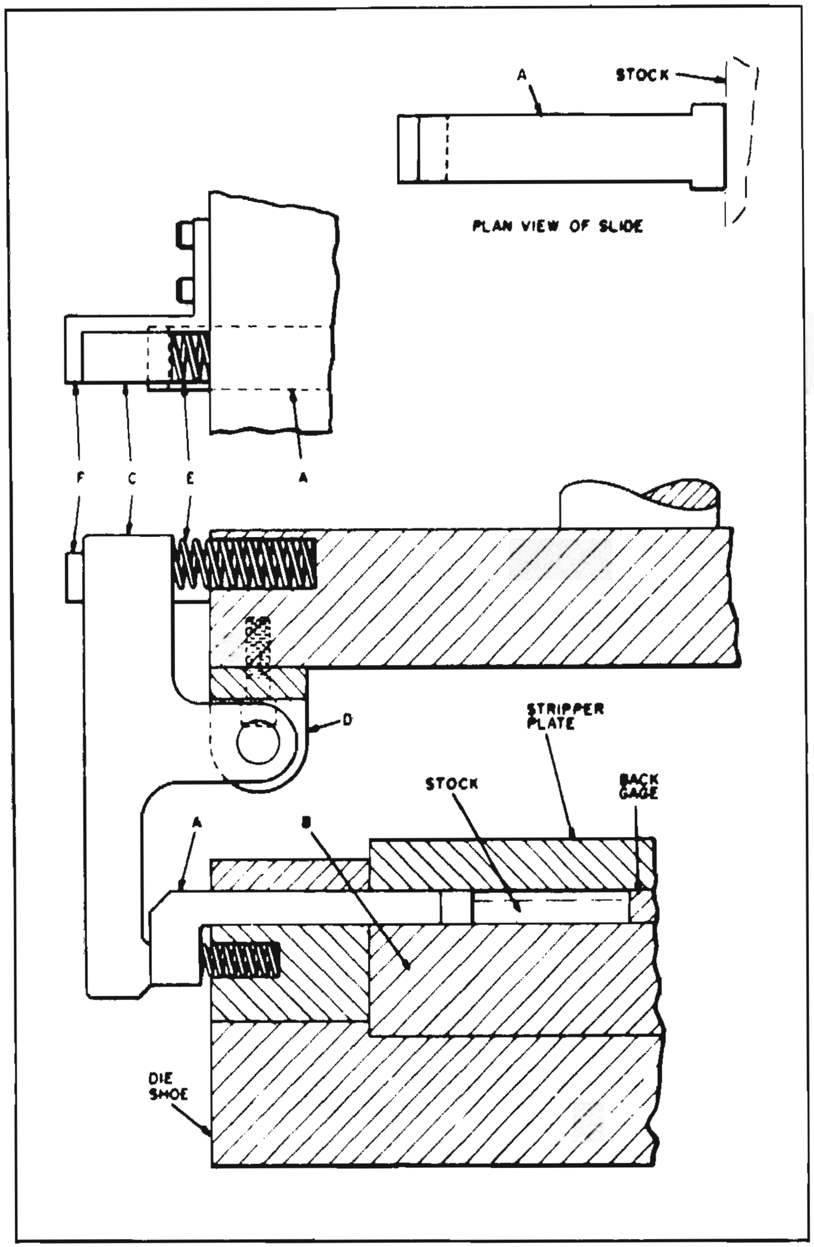


FIG. 14. As the top shoe of a die set descends lever C strikes A, pushing A to the right and clamping the stock.

(not shown) is operated to impart a second feed movement to the stock.

The swing stop A is secured to one end of a shaft which can swivel in a bearing in the headstock. The opposite end of this shaft carries an arm that is connected by link B to the follower arm C, the latter being pivoted at its left-hand end on a pin fitted to the frame. Downward movement of the stop A is imparted by a compression spring D enclosing a pin, the upper end of which makes contact with the arm C. The lower end of this pin is attached to the frame. Upon completion of the initial feed of the bar stock, stop A is swung upward by cam E, which is engaged by a roller on arm C.

The shaft on which cam E is mounted operates the mechanism that feeds the stock through the collet, and is driven intermittently from the back-shaft of the lathe, through a one-

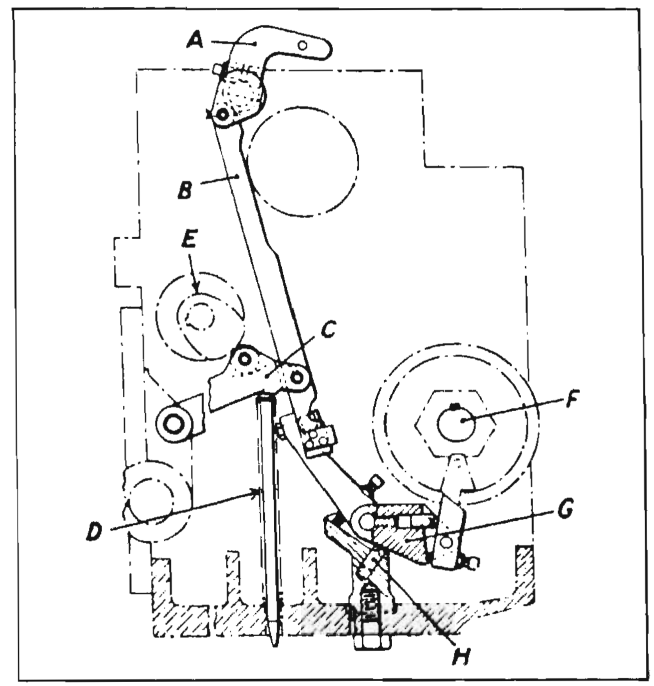


FIG. 15. Mechanism for operating swing stop A on an automatic lathe to control axial feed of bar stock.

revolution-and-stop clutch (not shown). With this arrangement, the cam E is rotated through two revolutions while the continuously driven front camshaft F, which controls the turret and cross-line motions, makes one complete revolution during each cutting cycle.

When the stock is to be fed a second time, a trip-dog, attached to a disc on the camshaft F, engages a spring-loaded pawl fitted to the right-hand end of the bellcrank lever G. The lever, which is carried on a forked bracket secured to the frame, swivels in a clockwise direction against the action of the spring-loaded plunger H. As a result, a stepped pin fitted to the left-hand end of lever G is swung to a position above an angle bracket attached to the lower end of link B, so that movement of the link and of stop A is prevented. Simultaneously, the shaft carrying cam E is rotated, causing the stock to be fed.

Air-Operated Clamping Mechanism for Cylinder Boring Fixture

The cylinder boring fixture shown in Fig. 16 is equipped with an air-operated clamping mechanism designed to hold the work securely without distortion. The work (cylinder C) is located on saddle B which is a close fit over the tongues E' on brackets E. Cylinder C is located or centered by means of the pivoting bracket K, which is shown in the open position by dotted lines K'.

Air cylinder G which actuates the clamping mechanism is slidably mounted on bracket F, so that when air is admitted into the cylinder, the piston rod J will move to the left while the cylinder G will move to the right. The open or non-pressure head end of the air cylinder is connected to the equalizer bar N which actuates the clamping levers D'. The outer end of piston rod J fitted to piston H is connected to lever D.

Air admitted to cylinder G at connection P acts on piston H, compressing the release spring O and moving levers D' and lever D to the clamping positions shown. In these positions the levers exert the required clamping pressure on the work at points S. A long bearing surface at S (as shown in the separate view in the lower right-hand corner of the illustration) distributes

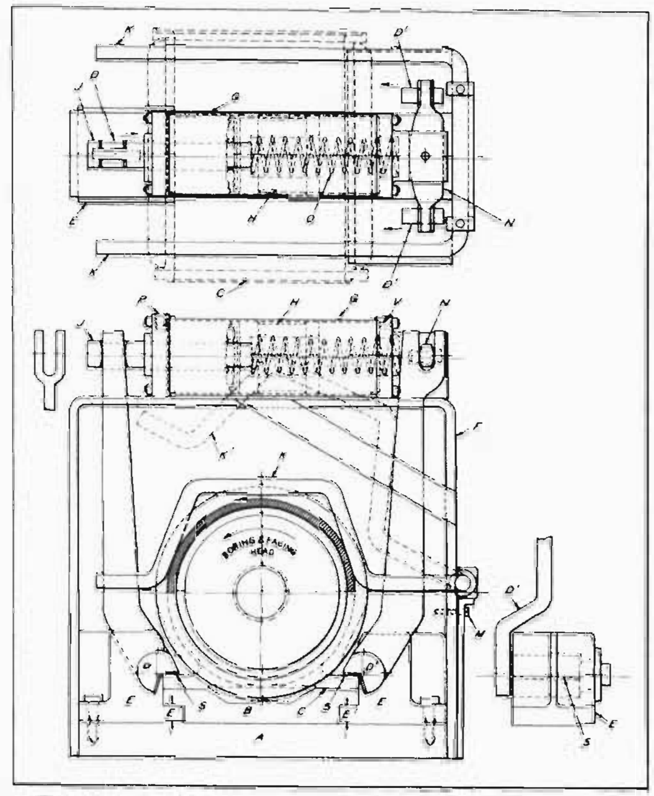


FIG. 16. Cylinder boring fixture equipped with air-operated equalizing clamping levers.

the load over a larger area. The vent V in the non-pressure end of cylinder G permits only atmospheric pressure to act on the right-hand side of piston H. When air pressure is released from the closed end of the cylinder, spring O causes the upper end of lever D' to move to the right and the upper end of lever D to move to the left so that the clamping pressure is released at points S.

Clamping and Indexing Mechanism for Drill Jigs

A multiple-purpose drill jig that incorporates an arrangement to automatically clamp a work-piece simply by lowering a hinged jig plate into position is shown in Fig. 17. In addition, components can be rotated and indexed for drilling a number of radial holes without being unclamped between operations. Although it was originally designed to accommodate collars and pinions in a variety of widths and diameters, the jig can be adapted to handle many other types of cylindrical parts.

Basically, the jig consists of an adjustable V-block *A* to support the work-piece *B*, a hinged jig plate *C* with a replaceable bushing to locate and guide the drill and a means of clamping and rotating the part. The V-block is raised or lowered in guide block *D* by turning the knob end of adjusting screw *E*. A pointer on scale *F* indicates the diameter of collar that can be drilled at each vertical position of the V-block.

Gear *G* transmits the rotary motion of shaft *H* through a gear train to gear nut *J* which moves externally threaded sleeve *K* in an axial direction. The gear nut is axially retained by a bushing and the outer-bearing support plate. Shaft *L* has a sliding fit in the bore of the threaded sleeve. A pin is pressed into shaft *L* and is fitted into slots in clamp *M*. Shaft *L* and clamp *M* must rotate together but can move axially about $\frac{1}{8}$ inch in relation

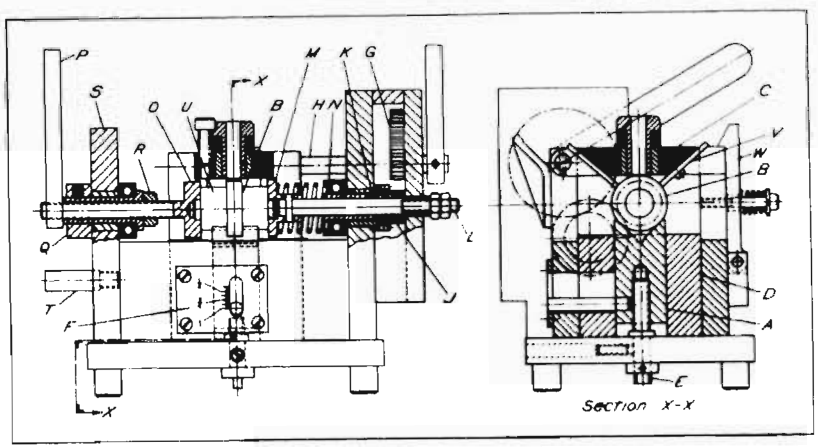


FIG. 17. Drill jig with mechanism for clamping and indexing a variety of cylindrical parts. Clamping action is concurrent with lowering of hinged jig plate.

to each other. A coil spring pushes clamp *M* toward the work-piece and is allowed by thrust bearing *N* to rotate freely with the clamp and the shaft. Forward motion of shaft *L* is restrained either by the work-piece through clamp *M* and its retaining pin or by threaded sleeve *K* through lock-nuts. The latter is the case when clamp *M* is not in contact with the work-piece.

Clamp *O*, lever *P*, and an internally threaded sleeve with its retaining collar *Q* and lock-nut *R* may be rotated as a unit in a fixed axial position. A thrust bearing allows free movement of the parts under any heavy clamping force applied to the work-piece by clamp *M*. Index holes are provided in bearing support plate *S* for insertion of threaded stop *T*. This stop is used to accurately position lever *P* and, therefore, the work-piece for the drilling of radial holes.

Axial position of clamp *O* can be adjusted to suit work of various widths by advancing or retracting this member within the internally threaded sleeve which is held in place by lock-nut *R*. In similar fashion the lock-nuts on the end of shaft *L* provide a means of adjusting the axial position of clamp *M*.

This jig is simple to use. A work-piece is placed on the V-block which is then set to the proper height by means of the adjusting screw. The rotary motion of lowering the jig plate into position is converted to the horizontal translation of threaded sleeve *K*. Clamp *M* (and its retaining shaft) are forced by the compressed spring to duplicate this movement and clamp the work-piece properly. Extension pieces *U* are screwed into the faces of the clamps to increase the range of the jig. Clamp *O* is pre-set axially to locate the work-piece properly. The lock-nuts on shaft *L* should be adjusted so that the pin is not in contact with either end of the slot in clamp *M* when the work-piece is securely gripped. This will prevent friction between the threads in parts *J* and *K* when the work is rotated.

After drilling the first hole in the work-piece, lever *P* is indexed to the stop for the second hole. Additional radial holes can be produced by moving the threaded stop to the next indexing position and repeating this operation. Clamping pressure is maintained on the work-piece since thrust bearing N allows the spring to rotate freely with clamp M. Adjustable work stops V prevent parts from being pulled up when the drill is retracted, and spring-actuated latch W holds the jig plate down in position for drilling.

CHAPTER 6

Reversing Mechanisms of Special Design

Described in this chapter are various arrangements for obtaining reversal of motion. Other reversing mechanisms are described in Chapter 6 of Volumes I and III and Chapter 7 of Volume II of "Ingenious Mechanisms for Designers and Inventors."

Sensitive Feed Arrangement for Coil-Winding Machine

Machines employed for winding fine wires into coils require sensitive feed arrangements that permit extremely quick reversals. For example, in winding wire 0.001 inch in diameter, with the bobbin rotating at 6000 rpm, reversing should be accomplished in a distance of 0.001 inch in a hundredth of a second. Although such quick reversals are usually obtained by fine screw feeds or by friction wheels, the arrangement shown in Fig. 1 is a unique solution to the problem.

A steel band C (see Fig. 1) is driven at constant speed by means of pulleys A and B. The band passes through two electromagnets, M_1 and M_2, which are mounted on wire guide carriage D of the coil-winding machine. When the carriage comes to the right-hand stop E, which is a micro switch, magnet M_1 is de-energized and magnet M_2 energized. The carriage is thus attracted to the lower portion of the band and moved to the left. When the carriage contacts the left-hand stop F (another micro switch), magnet M_2 is de-energized and M_1 energized, thus moving the carriage to the right.

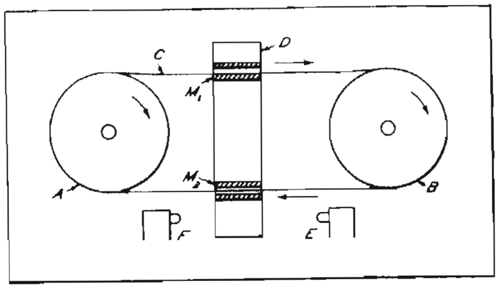


FIG. 1. Wire guide carriage D of a coil-winding machine is reciprocated between micro switches E and F by a steel band C which is alternately attracted to two electromagnets M_1 and M_2.

Speed of the steel band can be varied to obtain different rates of feed, and the location of the micro switches can be altered to change the length of carriage stroke.

Excessive-Torque Reversing Mechanism

A rotating drum type hopper, used to feed small molded parts into a chute, was subject to occasional jamming. The simplest way to free the jam was to reverse the direction of hopper rotation. To do this automatically, the illustrated mechanism was designed.

Two bevel gears A and B (see Fig. 2) are free to turn on drive shaft C. Smaller bevel gear D, which is keyed to the hopper shaft, is in constant mesh with the two larger bevel gears. Keyed to the drive shaft is a central driving member E. Two stepped hubs F and G, each free to turn on the drive shaft, carry pins H and J.

Spiral springs K and L connect the two hubs to their respective bevel gears by means of pin M in the case of spring K, and a similar pin, not shown, in the case of spring L. The primary function of the spiral springs is to absorb any shock load that might occur in the event of the hopper jamming. Driving dog N is secured to member E by means of a shoulder screw on which it is free to pivot. The dog is held against one of the two stop-pins O, by toggle spring P.

During normal operation, drive shaft C rotates in the direction of the arrow. With driving dog N in its left-hand position as shown, motion is transmitted from member E to stepped hub F, and from there, through spiral spring K to pin M. This causes bevel gear A to become the driving gear with respect to driven bevel gear D on the hopper shaft. Under these conditions, bevel gear B merely idles.

Any jamming that occurs during operation of the unit causes an increase in the torque necessary to drive the hopper. As a result, spiral spring K is placed under load. When sufficient pressure is built up between pin H and driving dog N to overcome the initial tension in toggle spring P, the dog will pivot about its mounting screw and come to rest against the right-hand stop-pin O. This releases the driving load from the components on the left.

Continued rotation of drive shaft C in its normal direction causes the right-hand side of the driving dog to engage pin J in stepped hub G. Through spiral spring L, and a pin similar to M, the driving force is now transmitted to bevel gear B. This, of course, causes the hopper to rotate in the opposite direction.

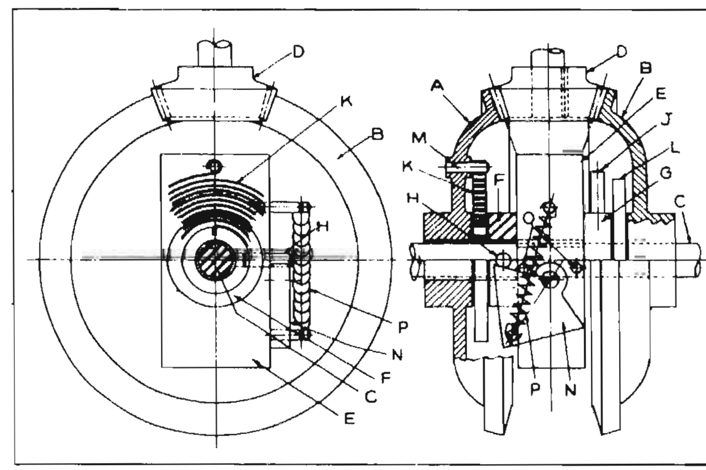


FIG. 2. Direction of hopper rotation, driven by gear D, is reversed by the pivoting action of a driving dog N whenever excessive torque is built up.

Reciprocating Traversing Device with an Adjustable Stroke

A mechanism designed for leading wire onto a spool in uniform layers is shown in Fig. 3. The arrangement incorporates a simple means of producing a smooth, reciprocating motion to the wire guide. In addition, the length of stroke of the guide is easily adjusted to accommodate spools of various widths.

The wire guide *A* is free to slide on a guide rod *B* and is traversed by a lead-screw *C*. Brackets *D* and *E* serve as bearings to support both the lead-screw and the guide rod. A bevel gear *F* is secured on shaft *G*, which is connected to the drive for the wire spool. Two additional bevel gears *H* and *J* are in mesh with gear *F* and rotate in opposite directions on the guide rod. Gears *H* and *J* each have a saw-tooth clutch plate attached to one face. A driving clutch member *K* having teeth on each face is pinned to the lead-screw between the gears. The direction in which the lead-screw is driven depends on the position of the driving clutch member *K*.

In the illustration, part *K* is shown in position to rotate the lead-screw in the direction that will cause the wire guide to move toward a collar *L* on the guide rod. Before reaching the collar, the wire guide compresses a spring *M*. When the spring is compressed, the wire guide stops. However, the lead-screw continues to rotate and moves to the right, pulling sleeve *N* with it, thus lifting a spring-loaded ball *O* out of the right-hand V-notch in the sleeve. A key in bracket *E* keeps the sleeve from rotating with the lead-screw, and two collars *P* pinned to the lead-screw hold the sleeve in place axially.

Once the ball is out of the right-hand V-notch, the pressure of spring *M* on the wire guide will cause both the guide and the lead-screw to move toward the right. This motion will continue until the ball drops into the left-hand V-notch provided in sleeve *N*.

Clutch member *K* will then be engaged with the clutch plate attached to gear *J*, and the lead-screw will rotate in the opposite direction. This will cause the wire guide to move toward collar

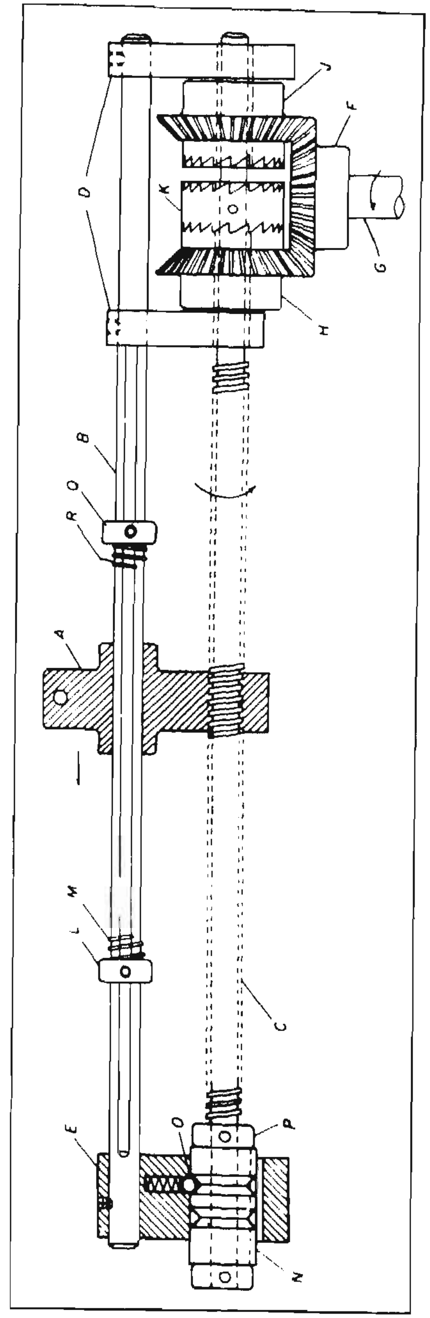


FIG. 3. A reciprocating, traversing arrangement having an adjustable stroke for guiding wire onto spools of various widths.

Q and spring R. On reaching spring R, the lead-screw reversing cycle is repeated. Collars L and Q may be placed at any distance apart within the length of the guide rod to suit various spool widths.

The mechanism operates smoothly with just a slight pause before reversal of the wire guide at the end of each stroke. The rapid motion of the guide when the ball lifts out of the notch compensates in part for the pause.

Reversing Two-Speed Geneva Drive

A proposed aerial camera required a mechanism to drive a prism in a certain series of movements. Specifically, these were: (1) turn 60 degrees counterclockwise, (2) stop momentarily, (3) rotate an additional 60 degrees in the same direction, (4) pause for an instant, (5) turn back clockwise 120 degrees, (6) stop again momentarily, and then repeat the cycle. The device was to be driven by a motor having constant, uniform speed; and the transition between rest and motion had to be shock-free. The compound Geneva mechanism illustrated in Fig. 4 was designed to satisfy all of these requirements.

The driving member, crank A, is fastened to the drive shaft. This crank carries two rollers B and C capable of entering slot D in a Geneva wheel E, which serves as the prism carrier. The length of the crank and the distance between its center of rotation and that of the Geneva wheel are such that the rollers engage and disengage slot D radially. In other words, the angle between slot D and the center line of the crank is 90 degrees at the moment of engagement. This insures smooth, shock-free operation.

Attached to crank A (and the input shaft) is a spur gear F which meshes with a spur gear G. The ratio between these two gears is 2 to 1. Gear G carries two rollers H and J, located 120 degrees apart and capable of entering slots K and L in Geneva wheel E. The distance between the rollers H and J and the center of gear G, and the distance between the center of gear G and Geneva wheel E, are such that the rollers engage slots K

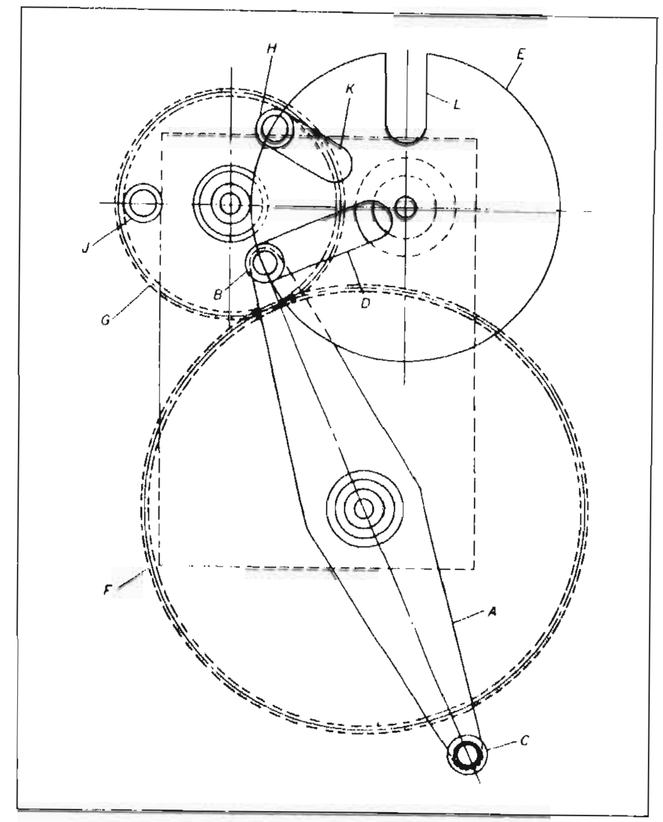


FIG. 4. Plan view showing details of Geneva mechanism. Members are in position to start cycle of movements required for a prism in an aerial camera.

and L radially. These slots are positioned 60 degrees apart on member E.

In operation, the drive starts its cycle in the position shown in Fig. 4 and crank A is rotated counterclockwise. This causes roller B to leave slot D and gear F to drive gear G clockwise. Roller H, mounted on gear G, simultaneously engages slot K and turns wheel E through 60 degrees. At this point, roller H moves out of slot K, wheel E stops momentarily, and roller J enters slot L. Once engaged with slot L, roller J turns wheel E

through an additional 60 degrees. Since the gear ratio is 2 to 1, gear G has, therefore, gone through 240 degrees of rotation and crank A through 120 degrees, placing roller C in a position to enter slot D. Continued turning of crank A revolves wheel E clockwise through 120 degrees and the cycle starts anew.

No locking arcs such as those used in conventional Geneva mechanisms are necessary, since there is always one of the four rollers in engagement with the wheel. This insures a positive correlation between the input and output movements at all times. The device is capable of speeds up to about 250 rpm.

Ratchet with Forward and Reverse Movements

A ratchet mechanism which transmits rotation to a shaft in one direction and, after a period of rest, reverses the motion to a lesser degree is shown in Figs. 5 and 6. This mechanism was designed for use on a machine which produces ornamental wire screening, and operates a mechanism that feeds the sheet of screening through the machine. It is required that the screening be fed through intermittently, advancing a predetermined distance, remaining at rest while a press operation is being performed, and then moving a predetermined distance in the reverse direction to permit withdrawal of the forming punches after the operation has been completed.

Front views of the mechanism at different positions of the cycle are shown in Figs. 5 and 6. In this mechanism, shaft A carries the ratchet wheel B which is keyed to it. Due to the reversing action of the mechanism, the conventional saw-toothed ratchet wheel cannot be used; therefore, the teeth on the ratchet wheel are a series of 90-degree notches. With this design of tooth, it is necessary for the thrust of the pawl to be approximately perpendicular to the contact surfaces, otherwise the pawl will jump out of the notches.

In operation, lever C swings freely on the hub of the ratchet wheel, receiving an oscillating motion from a cam-operated connecting-rod. Lever C carries pawl D, which is heavier on one side in order to be unbalanced. An extending arm of pawl

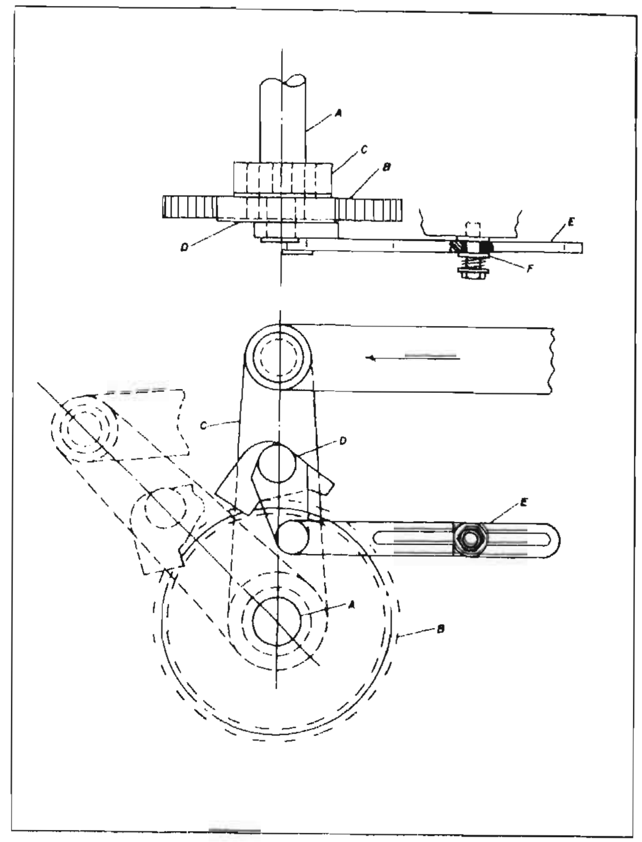


FIG. 5. Ratchet set for movement of slide-bar E to left, in a mechanism for forward and reverse movements.

D is connected to one end of slide-bar E, which is supported on a stud fixed to the machine. A light spring provides frictional resistance to the movement of the slide-bar during part of the cycle. The bar is of reduced thickness at the right-hand end and a flanged bronze bushing applies the frictional resistance under the action of the spring. The length of the bushing shank is such that friction is applied to the thicker section of bar E only, as can be seen in the top view.

Referring to Fig. 5, which shows the mechanism at the midpoint of the cycle, lever C is moving toward the left, and pawl

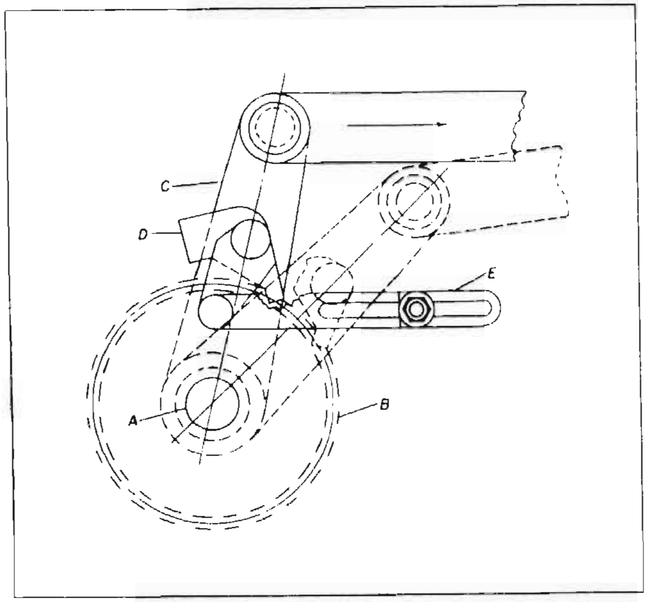


FIG. 6. Mechanism showing ratchet engaged for slide-bar movement to right.

D is engaged with one of the notches in the ratchet wheel. As lever *C* moves to the position shown by the dotted outline, it rotates shaft *A* through the action of the ratchet wheel. The movement of lever *C* draws bar *E* with it, the bar moving without any resistance at this point.

On the return stroke, one end of pawl *D* rides back over ratchet wheel *B* without transmitting motion to it. In Fig. 6, the mechanism is illustrated at the point where the heavier portion of bar *E* has contacted the bronze supporting bushing and the latter now can move only against the resistance created by the spring. This causes pawl *D* to reverse its position so that the other end engages a notch in the ratchet wheel, as shown. The reverse movement of shaft *A* begins at this point.

The resistance applied to the movement of bar *E* holds pawl *D* in engagement until lever *C* has reached the end of its travel to the right, as shown by the dotted outline. When lever *C* begins the forward stroke, the resistance applied to bar *E* causes

pawl *D* to immediately reverse and engage another notch in ratchet wheel *B* for the forward stroke. This mechanism produces a forward rotation of shaft *A* equal to the movement of five ratchet teeth and a reverse movement equal to that of two teeth.

Reversing Linear Feed with Adjustable Stroke

A means was required for guiding wire onto reels by reciprocating a guide head between the flanges. In order to accommodate different flange widths, as well as varying distances between hubs, it was found necessary to provide an adjustment for the linear travel and relative location of the head. Further, to be able to wind different wire sizes on the reel, the linear speed of the guide head had to be variable. The mechanism shown in Figs. 7 and 8 was designed for the application.

The guide head *A*, Fig. 7, is free to travel back and forth on two nonrotating shafts *B* supported by two brackets *C*. A lead-screw *D* engages a full thread in the guide head and is supported

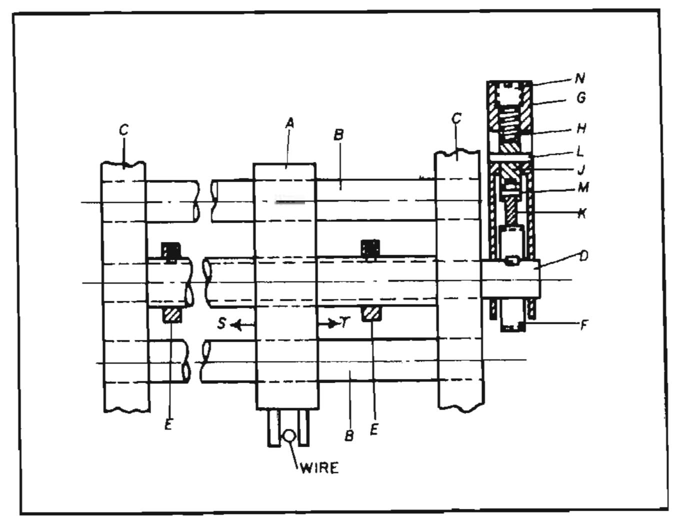


FIG. 7. Front view of feed device which provides a continuously reversing linear movement. Drive wheel and linkage, shown in Fig. 8, are omitted for clarity.

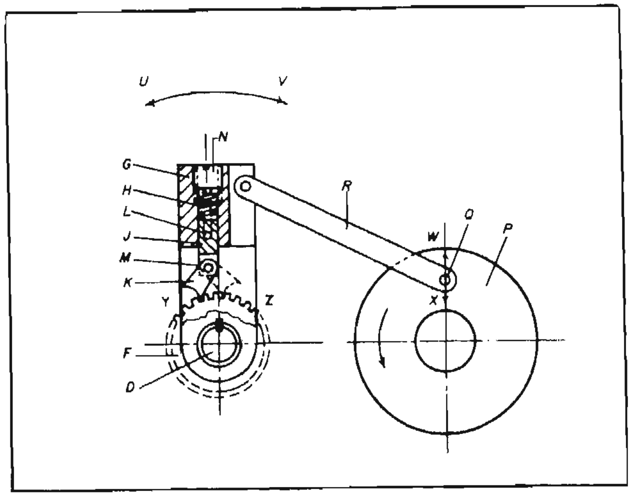


FIG. 8. Partial side view of mechanism shown in Fig. 7. Movement of guide head can be varied by adjusting the position of pin Q.

in journal bearings in brackets C. Stop collars E can be moved along the lead-screw and locked in any required position by means of set-screws.

A ratchet gear F is fastened to the lead-screw. Bracket G, supported on the lead-screw by integral journal bearings, carries a spring H, a slide J, and a pawl K. The slide is restrained from rotating by a pin L, but can move up and down in a slot in bracket G. Pawl K, mounted in the slide, is free to pivot around pin M. A set-screw N is used to adjust spring H.

Driving wheel P (Fig. 8), which rotates only in one direction, is equipped with a radially adjustable pin Q and is connected to the bracket G by the link R. This latter link is free to pivot about both of the connecting pins.

If, in operation, driver P rotates at constant velocity counterclockwise, bracket G will oscillate between points U and V. When moving toward V, pawl K will override the teeth of the ratchet gear and transmit no motion to the lead-screw. In the opposite direction, the pawl will engage the ratchet gear and rotate it counterclockwise. This, in turn, will move the guide head in direction S.

When the linear movement of the guide head is stopped by collar E, lead-screw D is unable to turn, as the guide head is retained by shafts B. If bracket G is still forced to move toward U and the lead-screw cannot rotate, pawl K will force slide J up and compress spring H. This will allow pawl K to flip over from position Y to Z. The oscillating movement of bracket G then will rotate the lead-screw clockwise and reverse the direction of travel of the guide head. When the guide head comes to the opposite stop collar E, the cycle is repeated.

By adjusting the position of pin Q in direction X or W, the oscillating bracket movement can be varied in magnitude. This, in turn, will vary the linear movement of the guide head to accommodate different wire sizes.

Machine Counter with Dwell Interval Operated Through Slotted Discs

On a warp machine used in the textile industry, the counter which keeps tabs on the amount of yarn drawn is operated through a device which incorporates a series of slotted discs to obtain a desired amount of lost motion. The reason for this is that the device must reverse its rotation several turns after drawing a certain length of yarn, then must rotate forward the same number of turns before more yarn is drawn, at which time the counter must pick up where it left off.

A drawing of the device appears in Fig. 9. The main element is a gear A. Within the gear bore is a series of eight discs B. Fastened to the right-hand face of the gear is a flanged plate C. On the left-hand face is a second flanged plate D. The latter is free to rotate, but is restricted from axial movement by a groove in its periphery which engages dog-point set-screws E in the gear. Plate D carries connecting-rod F, the top of which is joined to the counter (not shown). The entire assembly is bolted to the frame G of the machine.

One of the discs is shown in Fig. 10. Each has a large open area in the form of a curved slot H, and a hole J. The width of the slot is made greater than the diameter of the hole. This construction permits the large diameter of a shouldered pin K,

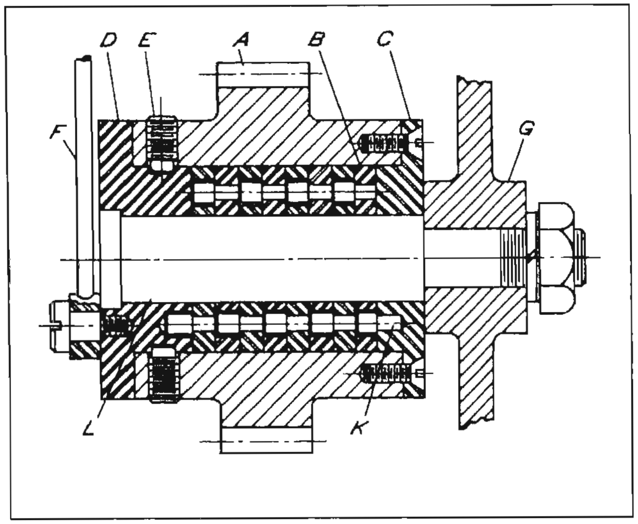


FIG. 9. When gear A is reversed, plate D remains fixed while the lost motion is absorbed.

Fig. 9, to bear in the slot of one disc, and the small diameter of the pin to fit the hole of the preceding disc. (For the first disc, the small diameter fits a hole in plate C. Similarly, the inner face of plate D has a hole accommodating the large diameter of the pin, the small diameter of which bears in the slot of the last disc.)

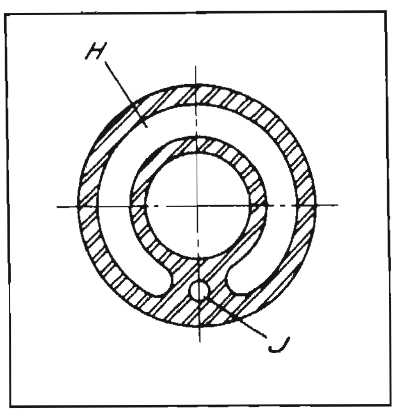


FIG. 10. Slot H in the disc accommodates the large diameter of one pin, and hole J, the small diameter of another pin.

When gear A is running forward and yarn is being drawn, it revolves as a unit with plate D around shaft L, and the connecting-rod operates the counter. The line of transmission extends from the gear to plate C, then to each shouldered pin and disc, and finally, to plate D. Then when the gear must be reversed for several revolutions, plate D remains fixed and the counter does not operate. The reason for this is that as the reversal is set up, the pin in plate C must be dragged from one end of the slot in the first disc to the other before the first disc joins in this reverse rotation. Similarly, there is a delay of almost a complete revolution before each of the following discs in turn starts its reverse rotation. Thus, a total of approximately seven revolutions of backward turning is available if required.

When the gear again runs forward, plate D remains fixed until all the pins in turn have been dragged to the opposite end of their respective disc slots. At that time, yarn again is drawn from the machine and plate D resumes its movement, and the counter again operates.

Direction-Changing Drive

A large number of machine tool components such as lathe saddles, milling and boring machine tables, and boring machine heads, incorporate subassemblies that move at right angles to each other. Usually, each of these subassemblies must be independently reversible. It is also frequently necessary to synchronize the movement of the subassemblies to obtain combined movement of 45 degrees. A patented, compact direction-changing drive, capable of performing all these functions through the actuation of a single control lever, is described.

The outside of the gear-box, from which the gears and control lever have been removed, may be seen at the left in Fig. 11.

Four gears, C, D, E, and F, comprise the train. Two of them, C and F, are sliding gears and are mounted on the two output shafts G and H. The remaining two are fixed gears: gear D is an intermediate gear and gear E is the driving gear, being mounted on input shaft J. Both sliding gears are moved by the action of a single control lever K, view B-B.

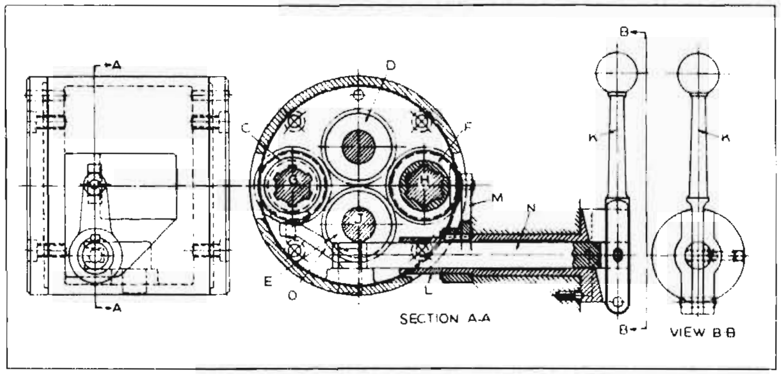


FIG. 11. Compact gear-box provides single-lever selection of eight directional output combinations of shafts *G* and *H* from a unidirectional input shaft *J*.

When the control lever is moved at right angles to the center line of sleeve *L*, sliding gear *F* will be shifted by means of arm *M*. This provides positive control over the engagement of the sliding gear with either intermediate gear *D* or driving gear *E*. This may be more clearly seen in the developed section of the gear train, Fig. 12. Movement of the control lever in a direction parallel to the center line of sleeve *L* will cause the shifting of sliding gear *C* through the action of push-pull rod *N* and offset bellcrank *O*. In this way, through the movement of only one control, eight drive motions can be called upon. A ninth, neutral position is represented by the gear arrangement shown in Fig. 12.

The position of the control lever, together with the positions assumed by the involved gears for each of the eight obtainable output motions, are shown in Fig. 13. In each of these cases input shaft *J* is rotating in a counterclockwise direction as viewed from the output side of the unit. When control lever *K* is pulled away from the gear-box, as seen at P, gear *C* is moved to the left into engagement with driving gear *E*. This causes output shaft *G* to rotate in a clockwise direction as indicated by the arrow. It may be noted that intermediate gear *D* and driving gear *E* are engaged at all times.

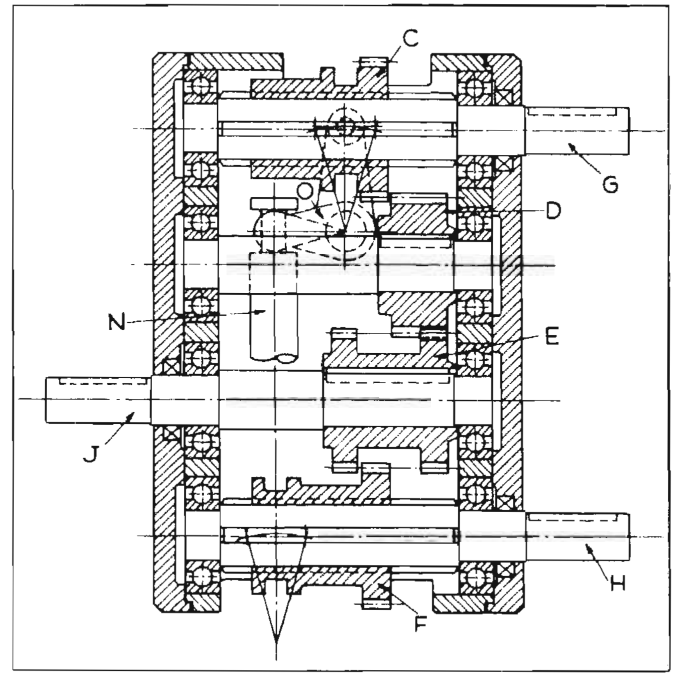


FIG. 12. Developed section of gear train showing types of gears and their position when the control lever is in a neutral position.

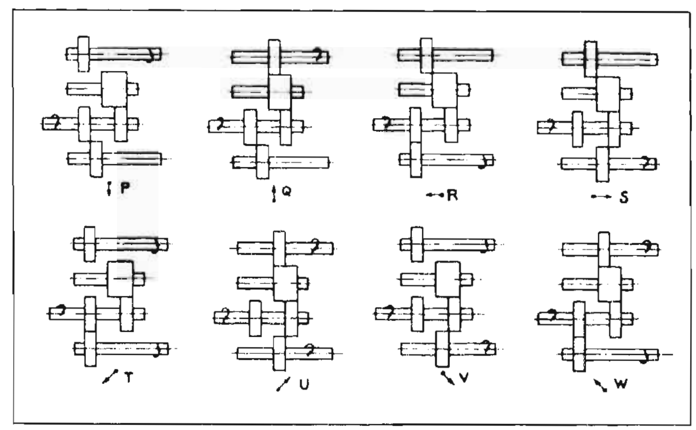


FIG. 13. Diagrammatic representation of the gear and lever positions for each of the eight output variations possible with the direction-changing drive.

At Q is shown the condition resulting from pushing the control lever toward the gear-box. Gear C is now forced to the right into engagement with gear D. The rotation of output shaft G is reversed, rotating in the same direction as the input shaft.

To activate output shaft H only, the control lever is first returned to neutral. When it is then moved to the left, as at R, sliding gear F is engaged with the driving gear and a clockwise rotation is imparted to shaft H. Reversal of this motion is accomplished by moving the control lever to the right, thus engaging gear F with gear D. This arrangement is seen at S in Fig. 13.

Four additional sets of movements may be had by combining the basic settings. An example of this is seen at T where both output shafts are being rotated in a clockwise direction. In this case, the control lever is pulled away from the gear-box to engage gear C with gear E, and then moved to the left so that gear F will also be engaged with gear E.

To reverse both output shafts to a counter-clockwise rotation, as at U, the control lever is pushed toward the gear-box and then moved to the right. This slides both gears C and F into engagement with gear D. The remaining two settings, shown at V and W, illustrate the two additional sets of movements that can be obtained by means of simple two-directional shifting of the single-lever control.

Adjustable Reversing Traverse Mechanism

A mechanism designed for guiding flat wire on reels by traversing a feed unit back and forth between the reel flanges is shown in Fig. 14. The reels were of different widths, and with various distances between the hub faces and the flanges. It was, therefore, necessary to provide an adjustment for the amount of linear movement of the feed unit and also for the relative location of the movement alternations.

The wire-feeding unit reciprocates back and forth along two nonrotating shafts A, Figs. 14 and 15. These shafts are supported by two brackets B, only one of which is shown. Bracket C (the wire-feeding unit) is free to slide on shafts A. This

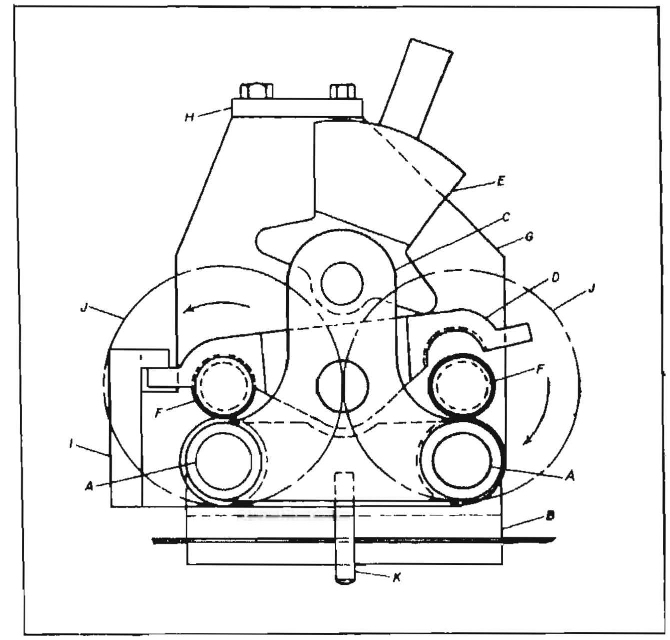


FIG. 14. End view of mechanism that proves an adjustable reversing traverse movement.

bracket carries a swinging lever D, which is provided with two half-thread arms that alternately engage one of two threaded portions of rotating shafts F. Either one of shafts F may be a driver to transmit rotation in the reverse direction through gears J. The threaded portions of shaft F may both be right-hand or both left-hand, depending upon the direction of rotation of gears J. Bracket C also carries a swinging weight E which has lower projections that alternately contact the upper surface of lever D.

Bracket G can be moved along shafts A and locked in any required position by means of a set-screw. This bracket carries a cam-plate H and a bar I. A similar bracket with parts H and I positioned in reverse positions, as shown in the dotted diagrams at the left in Fig. 15, is mounted on the other end of shafts A. Brackets G are positioned to control the distance by the feeding unit and the location of the movement relative to

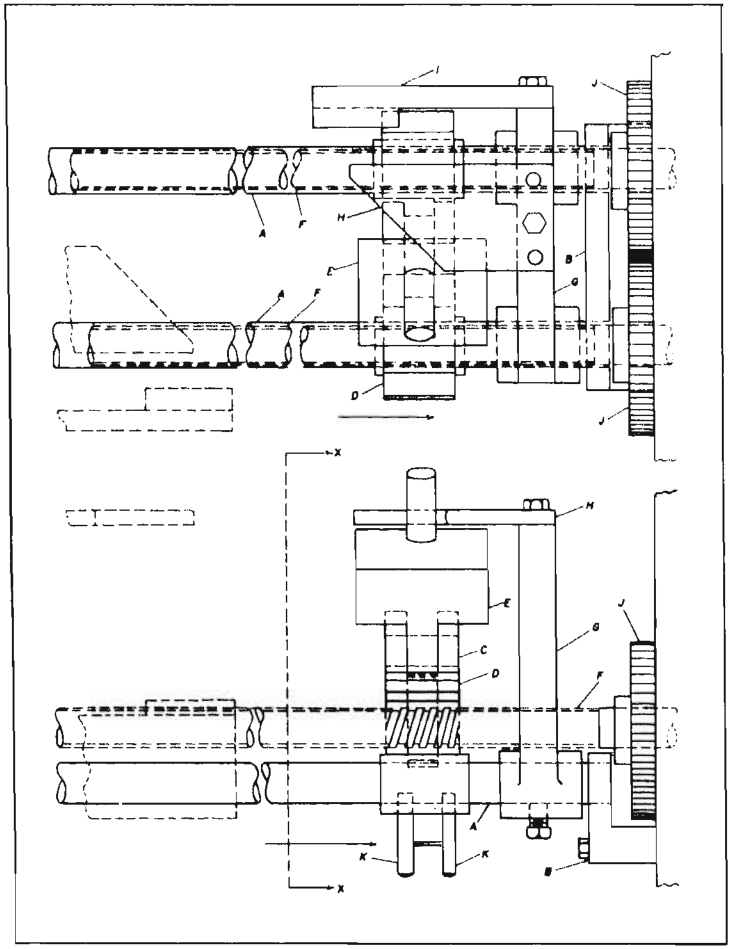


FIG. 15. Top and front views of mechanism that provides a reversing traverse movement which is adjustable for position and length of traverse.

the machine. The wire is guided between two pins K moved on bracket C.

Bracket C is moved in the direction indicated by the arrow when one of the half-threads on lever D engages one of the threaded shafts F. When this occurs the cylindrical projection of weight E has contacted plate H, causing weight E to be tipped

off center, as shown in Fig. 14, so that one of its projections contacts the upper surface of lever D. Lever D, however, cannot pivot at this point because the projection on its left-hand end is retained by the projecting portion of bar I.

In the top view of Fig. 15 bracket C has moved to a point where the projecting end of lever D is about to emerge from under bar I. When bar I has been cleared, lever D will be permitted to swing under the action of weight E. This will cause disengagement of the half-thread from one shaft F and engagement with the other shaft so that the direction of the linear movement of the feeding unit is reversed.

The purpose of bar I is to closely control the timing of the mechanism reversal and to prevent the half-nuts from lifting out of the shaft threads while weight E is swung to the opposite side.

CHAPTER 7

Reciprocating Motions Derived from Cams, Gears, and Levers

Described here are mechanisms which give reciprocating motions derived from cams, gears and levers. Cams, gears and/or levers may be used to vary the stroke in some way or to impart a mechanical movement essential to meet a particular operating requirement.

Other reciprocating mechanisms, based on the action of gears, cams and levers are described in Chapter 9, Volumes I and II, and Chapter 7, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Slow Reciprocation of Grinding Wheel Obtained from High-Speed Drive

Slow reciprocation of the abrasive wheel on a special centerless grinding machine is obtained from the high-speed rotary movement without the need for a reduction gear by means of the mechanism shown in Fig. 1. Reciprocation is accomplished by means of an axial cam drive and an additional pulley on the grinding wheel spindle.

Grinding wheel spindle A can rotate and slide in adjustable bearings B. Sleeve C, on which grinding wheel D is mounted, is pinned to the spindle. The wheel is rotated by belt E, which is mounted on a split pulley (F and G). Half-pulley F is screwed on sleeve C, while adjustable member G is secured to F by set-screws. By varying the axial location of member G, the effective diameter of the pulley and, consequently, its speed can be changed. A second pulley H — this one freely mounted on the spindle — is rotated by belt J. Mounted between this

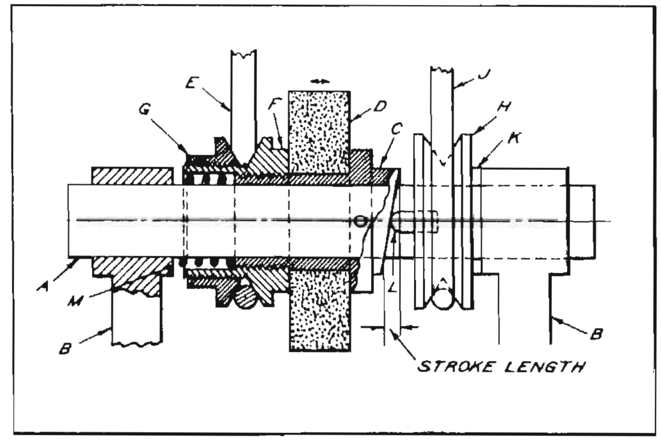


FIG. 1. Slow reciprocation of grinding wheel D is obtained by an axial cam drive mechanism. Freely mounted pulley H must rotate at a different speed than the wheel-driving split pulley F and G.

pulley and the right-hand bearing is a bronze washer K, and a pin L is pressed into the left-hand face of the pulley.

A helical cam surface provided on the right-hand end of sleeve C is kept in contact with pin L by spring M. Belts E and J can both be driven from the same shaft, but they must rotate the split pulley (F and G) and pulley H at different speeds in order to reciprocate the grinding wheel. If these pulleys were rotated at the same speed, there would be no relative motion between the pin and the contact point on the cam surface. However, as soon as there is a difference in the pulley speeds, the grinding wheel is reciprocated. The rate of reciprocation depends upon the difference in pulley speeds, and the length of stroke is controlled by the rise on the cam surface, as indicated on the drawing.

Geared Five-Bar Linkage for Straight-Line Motion

On an automatic machine where it was desired to guide a machine member along a straight line it was not possible to provide a guiding surface for the member. The solution is shown in Fig. 2.

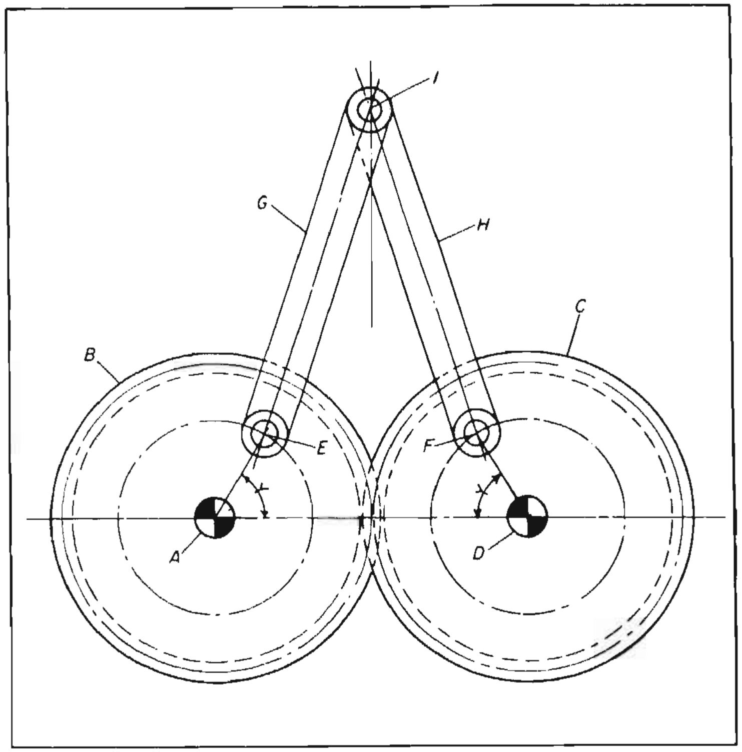


FIG. 2. Simple mechanism for obtaining straight-line motion through gearing.

In this movement driving shaft *A* carries gear *B* which is in mesh with gear *C*. Gear *B* is keyed to shaft *A*, and gear *C*, to shaft *D*. Both gears carry two studs *E* and *F* at equal radii from the centers of their respective shafts. They are also located so that angle *Y* is always the same for both studs. Links *G* and *H*, which are attached to the two studs, are of equal length. When gear *B* rotates, it drives gear *C*, and point *I* moves on a straight vertical line.

Reciprocating Cam with Half-Cycle Dwell

An irregular reciprocating motion was required on a machine used in the production of a formed wire part. This indicated

the necessity of a cam. Due to the absence of a rotating part to which a conventional cam could be attached, it was decided to use a nearby reciprocating member to carry a straight cam. This, however, presented one drawback in that a reciprocating cam transmits motion to the follower on both the forward and the return strokes, the movement being reversed on the latter. As this was not permissible, the modified cam shown in Fig. 3 was designed.

Cam-slide *A* rides in a dovetailed groove machined in a stationary part *B* of the machine. Follower *C* contacts the cam surface of the slide, as shown at X. Adjacent to the cam-slide is a short groove in which rides a small slide *D*. Friction is maintained on the slide, which is slightly higher than the cam sur-

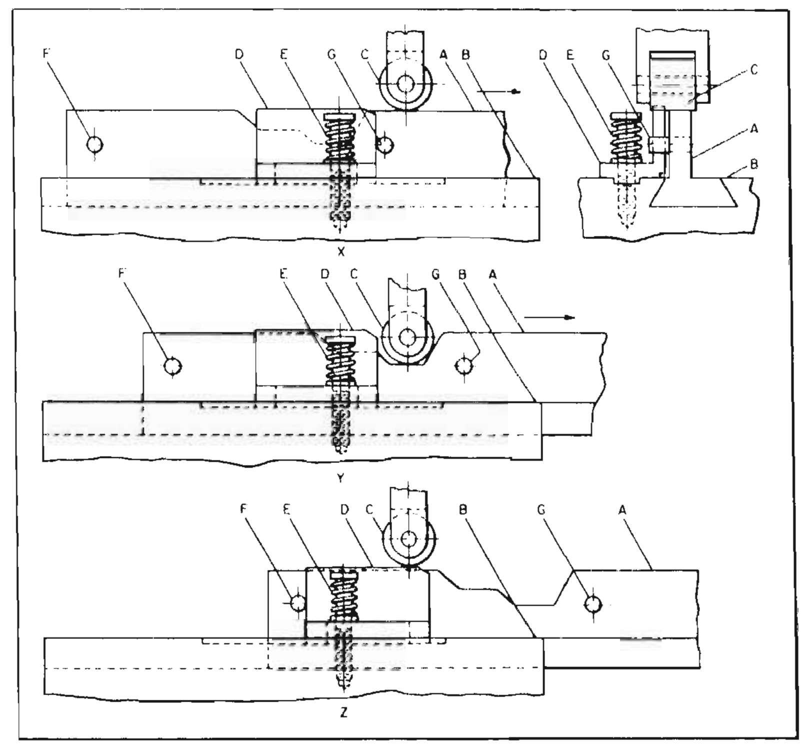


FIG. 3. Follower receives motion from reciprocating cam during forward stroke, but dwells during return stroke.

face, by spring E. The spring is retained by a stud that passes through a slot in the base of the slide. Two pins F and G protrude from the side of the cam.

The components can be seen at X in the position they assume at the beginning of a stroke. As the cam-slide moves in the direction of the arrow, follower C traces a path along the cam outline, as shown at Y. Continued movement of the cam permits pin F to contact slide D and force it under the follower, as illustrated in view Z. Due to the height of the small slide, the follower is lifted out of contact with the cam. This completes the first phase of the cycle of operation during which the desired reciprocating motion was transmitted to the follower linkage.

On the return stroke of cam A, pin F breaks contact with the small slide, which is held stationary by spring friction. The follower, being supported on slide D, also remains stationary. At the end of the return stroke, pin G strikes the opposite side of the small slide, forcing it to the left, as shown again at X. In this manner, movement of the follower takes place during the initial half of the cycle only. During the second half of the cycle a dwell period is substituted for the follower movement.

Mechanism that Develops Two Reciprocations with One Drive-Shaft Revolution

On a machine designed for producing a twisted wire product, it was necessary that two reciprocations back and forth be imparted to a wire guide for each rotation of the driving shaft. It was also required that the twisting spindle be given two cycles during the same period, each cycle consisting of one and one-half rotations in either direction, in synchronization with the guide. To accomplish these movements, the trammel gear mechanism shown in Fig. 4 was developed. The wire guide is indicated at E. Shaft J operates the twisting spindle.

The drive-shaft revolves disc A in the direction indicated by the arrow. Disc A is provided with two perpendicular radial grooves, in each of which one of two blocks B is free to slide. These blocks are pivoted on studs carried on connecting-rod C, which is attached to slide D. Slide D carries the wire guide

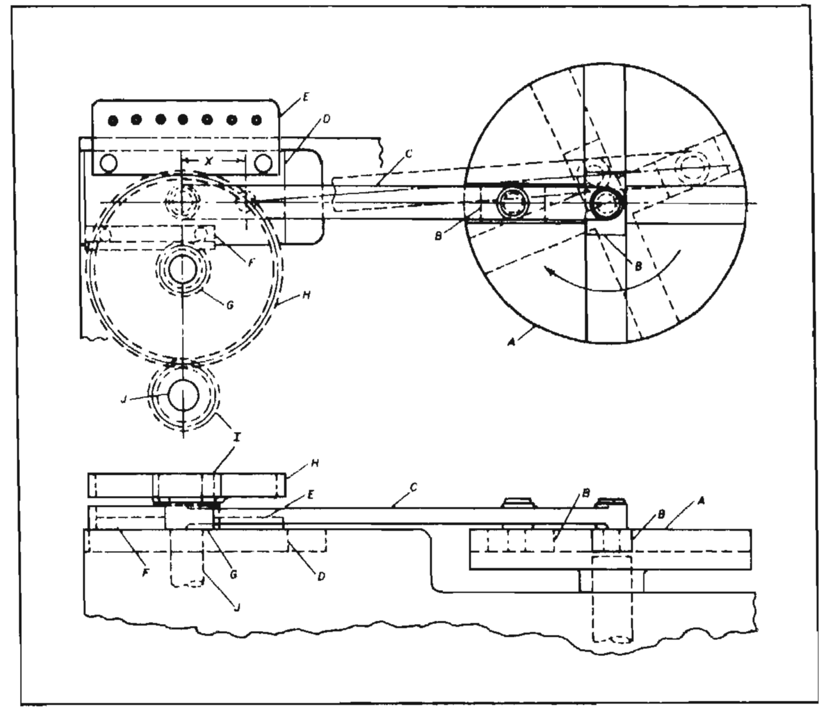


FIG. 4. Mechanism that incorporates a trammel gear for effecting reciprocating and oscillating motions.

and a rack F that meshes with gear G. Gears G and H are fastened together and rotate freely on the stud shaft on which they are mounted. Gear H meshes with gear I, which is attached to the twisting spindle shaft J.

Blocks B, being placed in slots perpendicular to each other, maintain their relative positions regardless of the circumferential position of disc A. As the disc revolves, the outer block B is caused to rise in its groove by the movement of the inner block B. The rotation of disc A acting on the outer block B causes the inner block to be drawn to the right as it rises. The combination of the two movements causes the blocks to move around the path of an ellipse which has its major axis on a center line extending between the center of disc A and the center of the connecting-rod stud on slide D.

The dotted outlines of the slots in disc *A* and of the connecting-rod *C* indicate their relative positions when disc *A* has been rotated about 60 degrees from its original position at the beginning of the cycle. At this point, slide *D* has moved distance *X*. When disc *A* has completed 90 degrees of rotation, the inner block will be on the center of rotation in the groove which is vertical at that time, and the outer block *B* will be on the right-hand side of center in the horizontal groove. This completes the movement of slide *D* in one direction. Continued rotation of disc *A* will reverse the movement of slide *D* and return it to the starting point at 180 degrees of disc rotation. Thus, a one-half rotation of disc *A* produces one complete cycle in the movement of slide *D*.

The length of the stroke given to slide *D* is governed by, and is equal to, the center distance between the studs in blocks *B*. Incidentally, by providing an adjustment for the inner stud, variations in the length of the slide stroke may be accomplished. The action of the movement is not affected except in the degree of shaft *J* rotation, which is governed by the movement of slide *D*.

Modified Eccentric Provides Rest Interval

In the fabrication of a wire product, a slide-actuated mechanism was required to transfer the part being made from a loading station to a work station, and then to an unloading station. The part was to be loaded on the forward stroke of the transfer slide, and unloaded on the return stroke. It appeared that a rotating eccentric disc offered the simplest method of moving the slide. This did not, however, prove entirely satisfactory because the velocity of the slide was too great when passing the unloading station to permit certain discharge of the part.

The difficulty was overcome by modifying the eccentric mechanism to provide a rest interval at the mid-point of the return stroke without otherwise affecting the work cycle. Figure 5 shows a mechanism that was satisfactory. Side views of the device are shown at X and Y and an end view at Z.

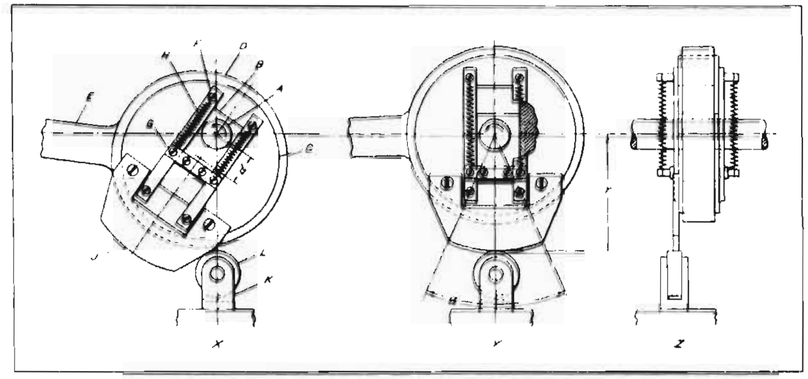


FIG. 5. By having the center of disc *C* momentarily coincide with the axis of shaft *A*, a rest interval is introduced in the action of this eccentric mechanism.

The shaft *A* which drives the mechanism is supported on bearings, not shown, and rotates in the direction indicated by the arrow. Keyed to this shaft is a block *B* fitting a rectangular slot cut through a disc *C*. Encircling the disc is a ring *D* integral with the rod *E* which extends to the transfer slide. Wear plates *F* support the disc. A bar *G* on block *B* carries studs that retain springs *H*. The opposite ends of springs *H* are anchored to studs attached to the wear plates.

A cam-plate *J* is fastened to one face of the disc. The greater part of the periphery of the cam-plate forms an arc that is concentric with the disc. Supported on a stationary bracket *K* is a roller *L* with which the cam-plate comes into intermittent contact during the functioning of the mechanism, and thus sets up a rest interval.

In view X, the disc center is offset from the shaft axis by a distance *d*, being in effect a conventional eccentric arrangement. The springs serve to keep the block at one end of the rectangular slot in the disc. In the position shown, the rise section of the cam-plate has just contacted the roller. Further rotation of the shaft causes the roller to act on this section, creating a movement of the disc center toward the axis of the shaft.

As can be seen in view Y, the curved section of the cam-plate has mounted the roller. (The vertical position of the bracket is such that distance r equals the radius of the cam-plate.) During this period, the axis of the shaft and the disc center coincide, so no movement is transmitted to the eccentric ring. Thus, during the rotation of the shaft, the transfer slide remains a complete rest for an interval determined by the magnitude of angle a.

The radial position of the block and cam-plate J on the disc is, of course, arranged to time the rest interval to the desired point in the return stroke of the slide. As the shaft continues to rotate, the contact of the cam-plate with the roller is terminated, and until contact is again made, the mechanism operates in the manner of an ordinary eccentric.

Two Slides Reciprocated Intermittently from One Source

Figure 6 shows a device used on a machine for producing a woven wire product. Its purpose is to carry strands of wire into the required positions in proper sequence.

Three views show the mechanism at different points in its operating cycle. Referring to view X, spur gear A rotates freely on a stud that is carried by a long bar B, moved by a cam (not shown).

Gear A meshes with two rack slides C and D, one above and one below. Rack C is free to slide and is retained by plate E. Two pins F protrude from this plate and act as stops.

Rack D is retained by plate G. This plate, however, is held against it under the pressure of springs held by a series of studs H, so that the rack must overcome a degree of frictional resistance before it can move. Both racks carry rods J, which are flattened and drilled on their outer ends to form eyes for guiding the wire strands.

View X presents the mechanism at the beginning of the cycle. At this point, rack slide C is in contact with right-hand stop-pin F. When bar B is moved to the left, rack D resists movement

due to spring-loaded plate G. As rack C is not subject to such pressure, the gear is caused to rotate in a counterclockwise direction, rolling on the lower rack while carrying the upper rack with it.

This movement continues until, at the midpoint of the forward stroke, rack C is halted by left-hand stop-pin F — the device then being in the position shown in view Y. At this point, bar B has advanced through distance V, while rod J has traveled twice that distance.

Rack C is now restrained, so that continued movement of the main bar will overcome the friction of rack D. Now, rotation of the gear is reversed as it rolls on the upper rack while carrying the lower rack with it. This movement continues until the end of the forward portion of the cycle, leaving the mechanism as

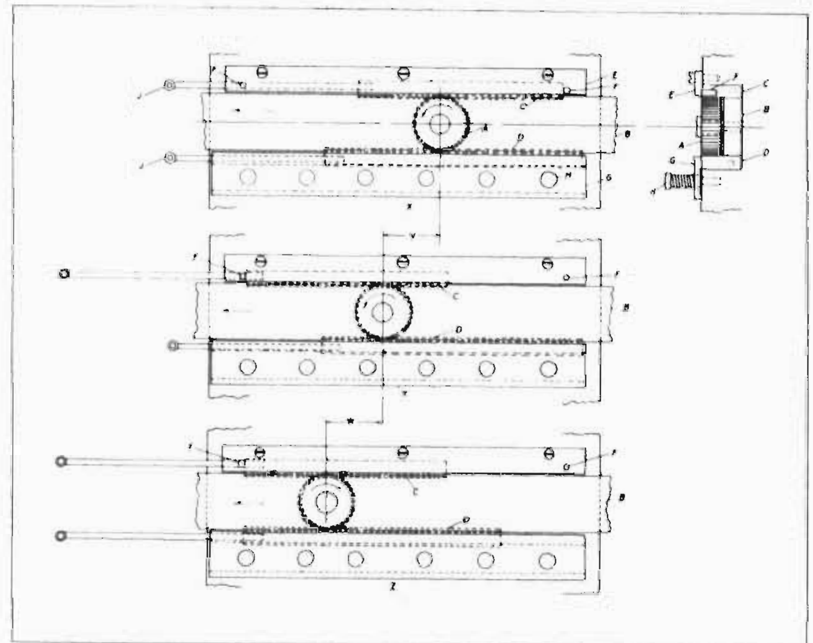


FIG. 6. Eyelet rods, J, are advanced and retracted intermittently by rotation of gear A on reciprocating bar B.

shown in view Z. The gear has advanced through distance *W* from its position in view Y while the lower rod *J* moved twice as far, bringing the eyes of the two rods into alignment in the forward position.

During the return half of the cycle, rack *C* is drawn back to the right-hand stop and rack *D* follows. In this manner, the upper rod is the first to move forward and the first to return. The operating cam is designed to provide a period of rest at each position so that work can be performed on the wire strands.

Mechanism Which Increases Movement by Levers and Chain

On a packaging machine it was necessary to provide a transfer movement for operating a slide. However, the movement of the slide had to be much greater than could be obtained with a direct connection. The drawings show a mechanism designed to provide the necessary increase of movement for the added slide. In Fig. 7 the mechanism is shown by solid lines at the beginning of the movement, and in dotted lines at the end of the movement, and also in an intermediate position. A bottom view of the mechanism is seen in Fig. 8.

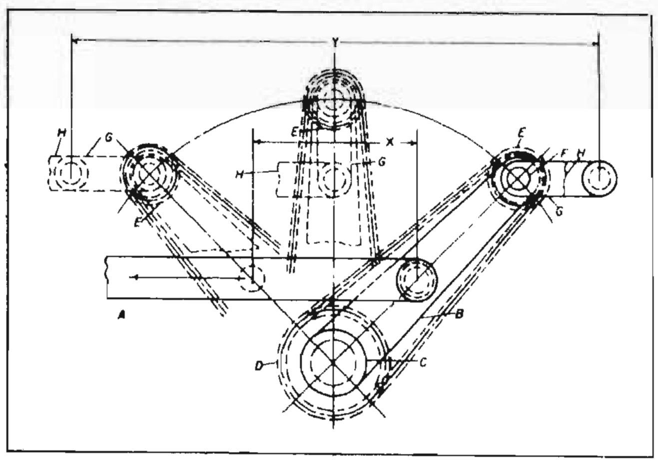


FIG. 7. Lever and chain mechanism designed to impart a traverse much greater than obtainable through a direct connection.

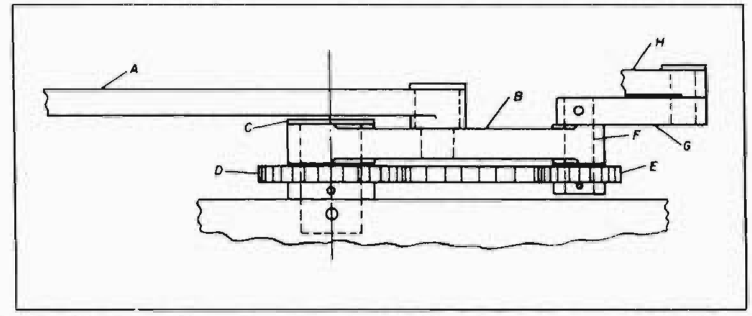


FIG. 8. Bottom view of mechanism which increases movement by levers and chain.

Bar *A* transmits motion from the existing slide to lever *B* of the added mechanism. Lever *B* is free to pivot in the same direction on fixed stud *C*. Sprocket *D* is locked on stud *C*. The sprocket is connected by a roller chain to sprocket *E* on shaft *F*. The shaft is free to turn in its bearing on the upper end of lever *B*. Lever *G* is attached to shaft *F* and carries a stud to which bar *H* is attached. Bar *H* transmits motion to the added slide.

When bar *A* is moved in the direction indicated by the arrow, lever *B* pivots in the same direction. As this occurs there is a change in the wrap-around position of the chain on sprocket *D*. This causes lever *G* to swing on the sprocket axis. Rotation continues until lever *B* reaches the end of its movement, at which position lever *G* has made a half-rotation. Its position is horizontal, as at the beginning of the cycle. However, lever *G* has been reversed relative to lever *B*. Thus, double the length of lever *G* has been added to the horizontal movement of shaft *F*.

Positions of lever *G* at the ends of the movement are controlled by the ratio of sprockets *D* and *E* relative to movement of lever *B*. The increase in the movement of bar *H* over that of bar *A* is indicated by the difference in the dimensions *X* and *Y*. Increase or decrease in the movements of bar *H* may be accomplished by a change in the length of lever *G* or, if adjustment is desired, lever *G* may be slotted to permit relocation of its stud which carries bar *H*.

Bead Chain Drive Turns Shaft on Moving Perpendicular Axes

In a mechanism designed for use on a wire fabricating machine for the purpose of guiding slowly moving strands of wire through a machine, rotary motion is transmitted from a shaft to two elements which continually change their positions relative to the drive-shaft. In addition, the driven shafts are positioned with their axes of rotation perpendicular to each other. The intention is to place the wire strands at various angles off their normal position at selected points in the cycle as determined by the wire pattern requirements. Figure 9 is a plan view of the mechanism; Fig. 10 is a front view; and Fig. 11 is a view of Fig. 9 from the right.

In the sketches the drive shaft *A* carries a bead chain sprocket *B* and may rotate in either direction. Bead chain idler sprocket *C* rotates on a stud mounted to the bed of the machine. Slide

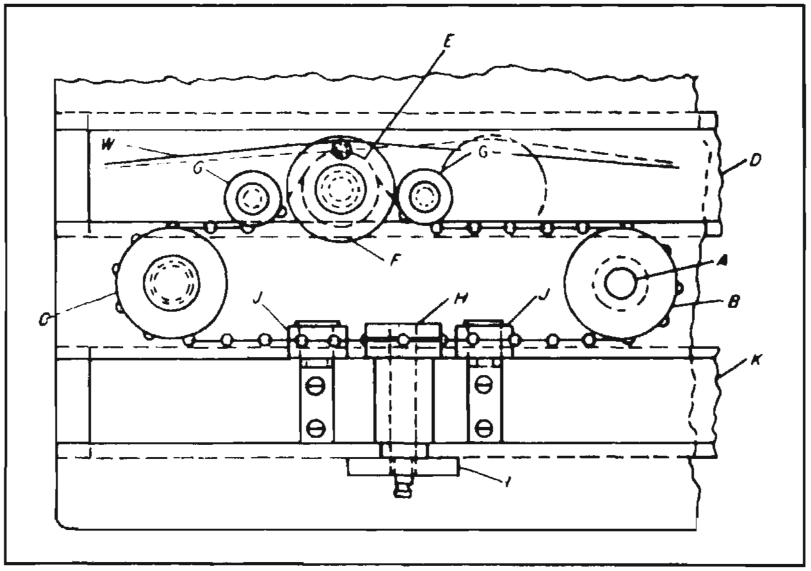


FIG. 9. Plan view of wire guide shows drive *A* and its relationship to the system of motions of the wire *W* as controlled by movement of slides *D* and *K*.

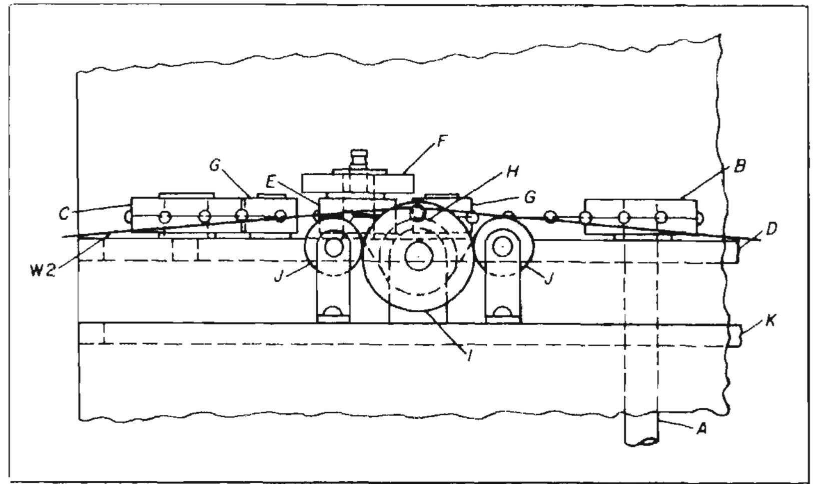


FIG. 10. Bead chain drive of discs *F* and *I* requires but little torque. This is a front view of the mechanism.

D, which is cam-operated, carries the sprocket *E* and disc *F*, keyed together and rotating freely on a stud. Two idler sprockets *G* are applied to maintain chain contact. Disc *F* carries a grooved pin which engages one strand of wire *W* to periodically place it in an angular position relative to its original straight-through position. Slide *K*, also cam-operated, carries a pillow block and shaft on which sprocket *H* and disc *I* are mounted. Disc *I*, like *F*, also has a grooved pin. Idler sprockets *J* are mounted on brackets carried by slide *K*.

In operation, slides *D* and *K* are moved intermittently by individual cams which are designed to provide the motion as programmed by the pattern to be produced in the product. In Fig. 9 the wire *W* is shown moved from its normally straight path. Figure 10 shows all parts in the same relationship as in Fig. 9, but another wire, *W2*, has also moved from its normally straight path. If the pattern required, either or both slides *D* and *K* could remain stationary while the discs *F* and *I* are given a number of rotations. In some wire design sequences, one slide may remain permanently in position while the other slide is moved in a required sequence. Because drive-shaft *A* rotates

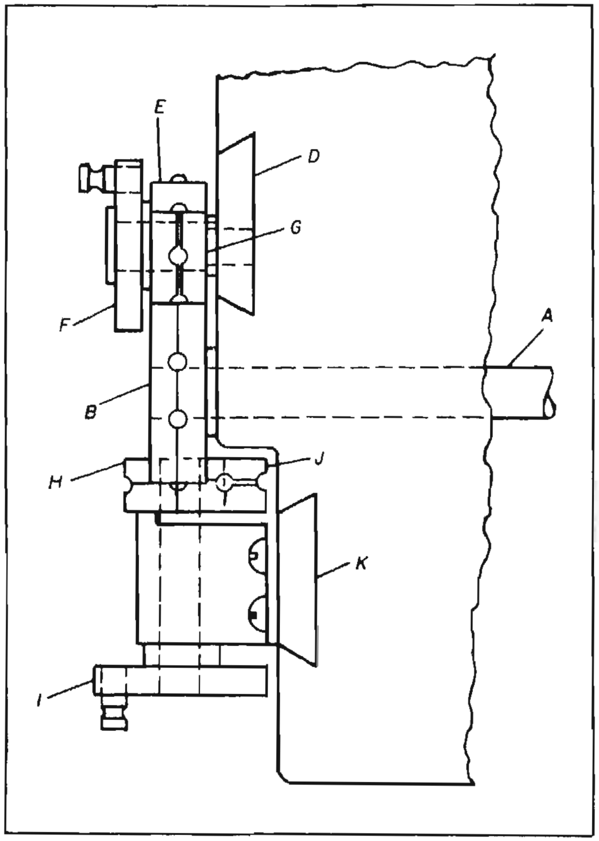


FIG. 11. Continuous rotation of wire guides on discs F and I sets up a "lay" pattern woven in a wire fabric, as shown by this right-hand view of the machine component plan seen in Fig. 9.

continuously, discs F and I also rotate continuously, regardless of the positions or movements of the slides. Any movement of slides D and K varies the angularity of the wires. In Fig. 9, disc F is shown dotted in its extreme position to the right, and indicates the change in the position of the wire as compared with the position represented by the solid outlines.

This mechanism operates at low speed under light load; therefore, a bead chain is ideally suited for transmitting motion from the drive to the perpendicular planes of rotation.

Reciprocating Drive Functions Around Roller Chain

During the development of a new product, the need arose for a simple reciprocating drive capable of operating under heavy loads. A stroke of 24 inches in length was required, but space considerations ruled out the use of conventional crank and lever type drives.

To meet these conditions, the illustrated sprocket and roller chain drive was developed (see Fig. 12). Either one of the two sprockets A or B can be used as the driving member, the remaining sprocket serving as an adjustable idler. Roller chain C is of standard design with one exception, one of the link rivets has been replaced by a long pin D. On each end of this pin is a roller E, held in place by cotter-pins.

In operation, the rollers are located between two follower-plates F, and also drive against them. The followers fit closely within slots in housing G and are held in place by cotter-pins on both the top and bottom. A gap is cut in one leg of the follower-plates to provide clearance over the sprocket hubs when the housing is at either end of the stroke. Bracket H is welded to

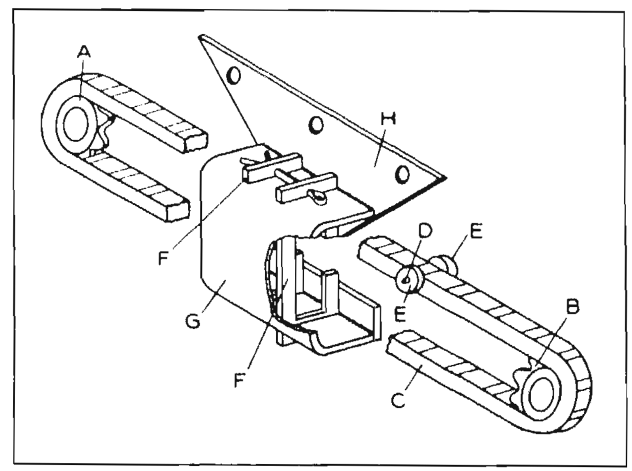


FIG. 12. Roller chain imparts reciprocating driving motion to machine slide (not shown). Rollers E engage follower-plates F which carry slide bracket H with them in both directions of travel.

housing G and bolted to the machine slide to be reciprocated (not shown).

As the roller chain moves with the two rollers E, located between the follower-plates, linear motion is imparted to the housing and then to the machine slide. When the chain link supporting the rollers reaches one of the sprockets, it descends, changing direction and returning on the lower span of the chain. Remaining between the two followers, the rollers now drive the housing and the machine slide in the opposite direction, providing the desired reciprocating motion.

Gear and Clutch Mechanism for Variable Operating Conditions

The various slides of multiple-slide wire-bending machines must be operated at different lengths of stroke, speeds of travel, and dwell periods at points of reversal because of differences in the thickness, springiness, and shape of the material.

The principal operating features embodied in the mechanism are shown in Fig. 13. A short connecting-rod A has one end affixed to an eccentric (not shown), attached to a continuously revolving shaft. The other end of the rod is attached to lever B by means of pin C. Lever B is fastened to shaft D by key E, the shaft being mounted in bearings supported by the machine frame. Also keyed to shaft D and located behind lever B, is a spur gear F.

Gear F meshes with gear G, which is mounted to rotate freely on stationary shaft H which is rigidly attached to the machine frame. Both gears are of the same size and are located on horizontal center lines. Swinging freely on a slightly reduced shouldered portion of shaft H is a lever I. One end of this lever is linked to the connecting-rod J by means of pin K, and the opposite end is linked directly to a former-slide.

The lower end of lever I has a large boss W, in the center of which is a tapered hole. Fitting smoothly in this hole is a circular cone clutch ring L, which is keyed to the stationary shaft H by key M. Shaft H projects beyond the clutch ring, the extension being externally threaded for lock-nuts N. By adjusting

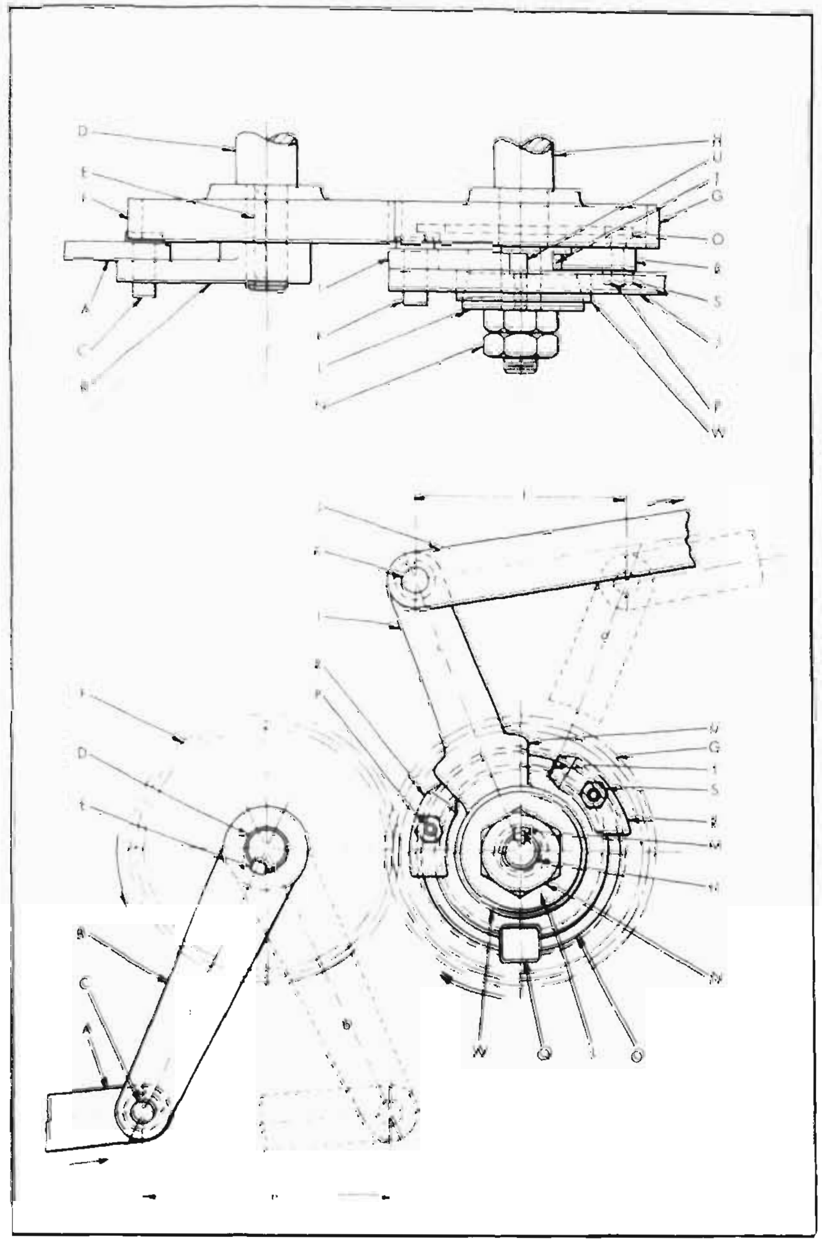


FIG. 13. Gear and clutch mechanism which produces a reciprocating motion with dwell periods. The driving and driven members are connecting-rods A and J, respectively.

the setting of these nuts, variable pressures can be imposed on the clutch ring in its engagement in the tapered recess of lever *I*. With this arrangement a certain degree of frictional restraint can be imparted to the swinging movements of lever *I* and connecting-rod *J*, as well as all other driven members connected to them, at each point of movement reversal.

Lever *I* and gear *G* are connected by two stop-dogs *R* which bear on the sides of lever *I*. The face of the gear has a concentric T-slot *O*. Two T-bolts *P* are placed in this slot through the rectangular opening *Q*. Two hardened steel stop-dogs *R*, provided with tongues to suit slot *O*, are circumferentially adjustable. They may be locked to the face of the gear by nuts *S* on T-bolts *P*.

Each stop-dog has a tapered contact nose *T* for bearing against a flat lug *U* on each side of lever *I*. If the dogs are adjusted to bear directly against lugs *U*, there are no dwell periods. On the other hand, by setting one of the dogs back from its corresponding lug *U* to leave a gap, as indicated, the drive between gear *G* and lever *I* will be interrupted, and there will be a short period of inaction, or dwell, of the lever and the former-slide. By applying sufficient pressure on the clutch ring, any movement of lever *I* and the former-slide will be arrested when the dwell points are reached, yet the lever will be permitted to slip over the clutch mechanism when a positive driving pressure is again imparted to the lever.

In operation, the relative positions occupied by the respective levers and gears at the beginning of the forward working stroke of connecting-rod *A* are shown by the heavy lines in Fig. 13. Rod *A* and lever *B* have, in fact, moved a slight amount toward the right to bring the stop-dog on the left-hand side of lever *I* into direct contact with bearing lug *U* on the same side of the lever. The positions of the driving and driven levers are indicated at *a* and *c*, respectively, and from this position all movement of rod *A* to the right will be transmitted positively to the connecting-rod *J* with all members moving in exact unison. Connecting-rod *J* and the former-slide attached to it will travel at precisely the same speed as connecting-rod *A*, since gears *F* and *G* have a one-to-one ratio and levers *B* and *I* are the same length.

When rod *A* reaches the extreme right-hand limit of its movement, as shown by the dotted lines at *b*, rod *J* and lever *I* will have reached position *d*. With the return movement of rod *A*, however, the driven members will remain stationary for a certain period, or until the stop-dog *R* on the right-hand side of lever *I* is brought to bear against lug *U* on that side of the lever. When this contact takes place, the lever will be drawn toward the left in unison with the movement of rod *A*. At the termination of this return stroke, when rod *A* and lever *B* have been drawn slightly to the left of the positions indicated at *a*, a second dwell period will occur, which lasts until the left-hand dog *R* bears against the adjacent lug of handle *I*.

The dwell periods may be varied by setting the dogs in different positions on the driven gear. Respective movements of levers *B* and *I* will then be different; the latter will have a shorter length of travel. The different stroke lengths are shown at *e* and *f*.

A modification to produce a wider range of stroke and dwell adjustments is illustrated in Fig. 14. This is accomplished by using gears of other than a one-to-one ratio, and by providing a series of connected holes in the end of the driving lever *B* to permit varying the radial setting of rod *A*. The fifteen-tooth intermediate gear shown reverses the stroke direction.

Linear Movement Reduced by Differential Chain Drive Mechanism

A machine producing a woven wire product was required to have two strands of wire traversed across it simultaneously at different rates of travel and over different distances but with the same time cycle. Figure 15 shows a mechanism designed to perform this task.

Bar *A* is slidably dovetail-mounted in a stationary part of the machine, and is caused to reciprocate by a cam, not shown. Bar *B* is slidably dovetail-mounted in bar *A*, and carries two sprockets *C* which are free to rotate on their studs. Block *E* is at-

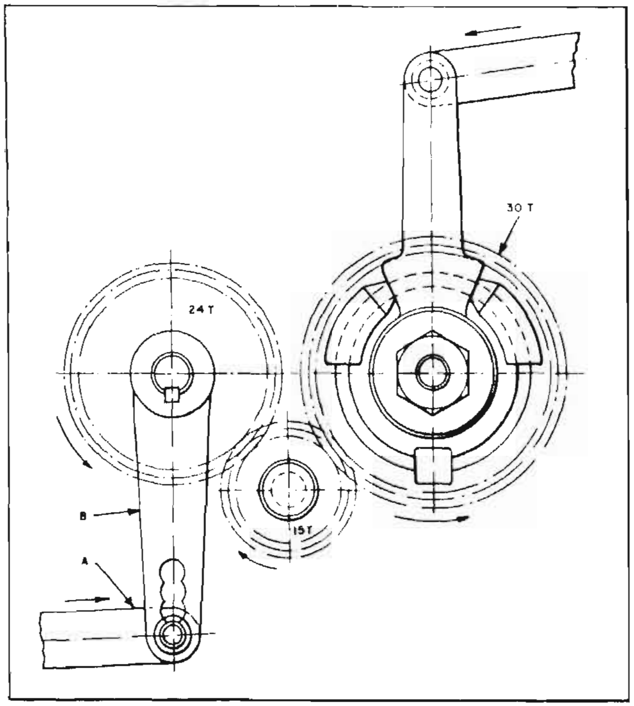


FIG. 14. Modification of mechanism shown in Fig. 13, which provides reversal of stroke direction and various stroke lengths and dwell periods.

tached to a stationary part of the machine. Two chains *D* engage sprockets *C*, one end of each chain being attached to block *E*, the other end being attached to bar *A*. The guides that direct the path of the wire are not shown. One of these guides is attached to bar *A*, and the other to bar *B*.

In the upper view, the assembly is shown at the mid-point of the traverse movement. In the lower view, the assembly is seen at its extreme left-hand position. In operation, the bar *A*, in moving toward the left, carries with it the two lower ends of the chains *D*. This movement results in an increase in the tension of the chain on the right, and a decrease in the tension of the chain on the left, so that motion is transmitted to the bar *B* through the right-hand sprocket *C*. As the left-hand

sprocket *C* travels with bar *B*, it is impossible for slack to develop in the left-hand chain *D*.

Thus, due to the fact that the upper ends of the chains *D* are fixed in position and that the sprockets are movable, a differential motion is produced which results in a 50 per cent reduction in the length of the travel movement of the bar *B*, as indicated by the distances *X* and *Y*, in which *X* represents the movement of bar *A*, and *Y* represents the movement of bar *B*. The bar *B* therefore follows the same motion pattern as bar *A*, but to a reduced magnitude.

Variable Reciprocating Motion Derived from Uniform Rotation

On a machine used in making a formed wire product, the partly completed piece is transferred to various positions for the different operations. One of these transfer movements is accomplished by the use of an eccentric operating a reciprocating member. Owing to variations in the size of the work-pieces, it is necessary that the distance through which the part travels

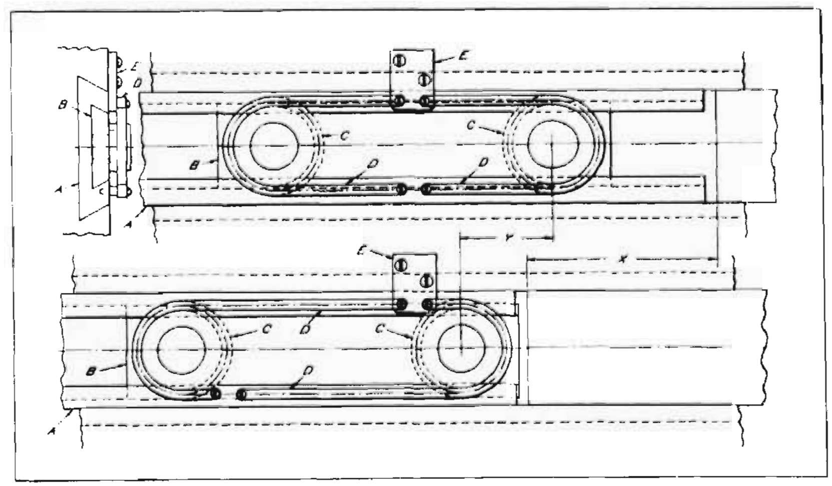


FIG. 15. Differential chain mechanism designed to reduce linear movement of slide.

from the point of transfer be varied without stopping the machine. It is also necessary that the loading position, which is the location at which the cycle is started, remain unchanged, regardless of the adjustment in length of stroke.

Figure 16 shows a mechanism that was designed to obtain the desired variable reciprocating motion from uniform rotation of an eccentric. Eccentric *A*, which rotates at a uniform rate to operate the mechanism, transmits reciprocating motion to slide *B* through the follower *C*. This slide is supported in machine frame *E*, and carries another slide *F*, which is free to reciprocate in the dovetailed slot in slide *B*. Slide *G*, also supported in machine frame *E*, carries a roller *H* which engages an angular slot in slide *F*. At one end of slide *F* is mounted another roller *P*, which engages a groove in the adjustable block *J*. This block is keyed to the vertical shaft *K*, which is supported in bracket *L*. Shaft *K* also carries worm-gear *M* which meshes with worm *N*. The worm, mounted on bracket *L*, is rotated by handwheel *O* to adjust angle of block *J*.

In operation, the rotation of eccentric *A*, which is shown in the plan view (center sketch) with follower *C* in its extreme right-hand position, causes slide *B* to move to the left. As slide *G* is connected to slide *B* by roller *H* in slide *F*, slide *G* is also moved to the left. As roller *P* operates in the groove in block *J* (which is shown set at an angle with relation to the line of motion of slide *B*), the movement of slide *B* causes slide *F* to move in the slot in slide *B*.

This movement of slide *F*, caused by the action of roller *H* in the angular slot in slide *F*, results in motion being transmitted to slide *G*, in addition to the motion that is directly applied by the movement of slide *B*. However, because of the angularity of the slot in slide *F*, any movement of this slide in the direction shown produces motion of slide *G* in the reverse direction, or to the right. This reverse motion subtracts from the movement transmitted to slide *G* by slide *B*, the movement of slide *G* being the resultant of the two motions.

In the partial plan view (top sketch), from which some of the parts have been omitted for clarity, slide *B* has completed

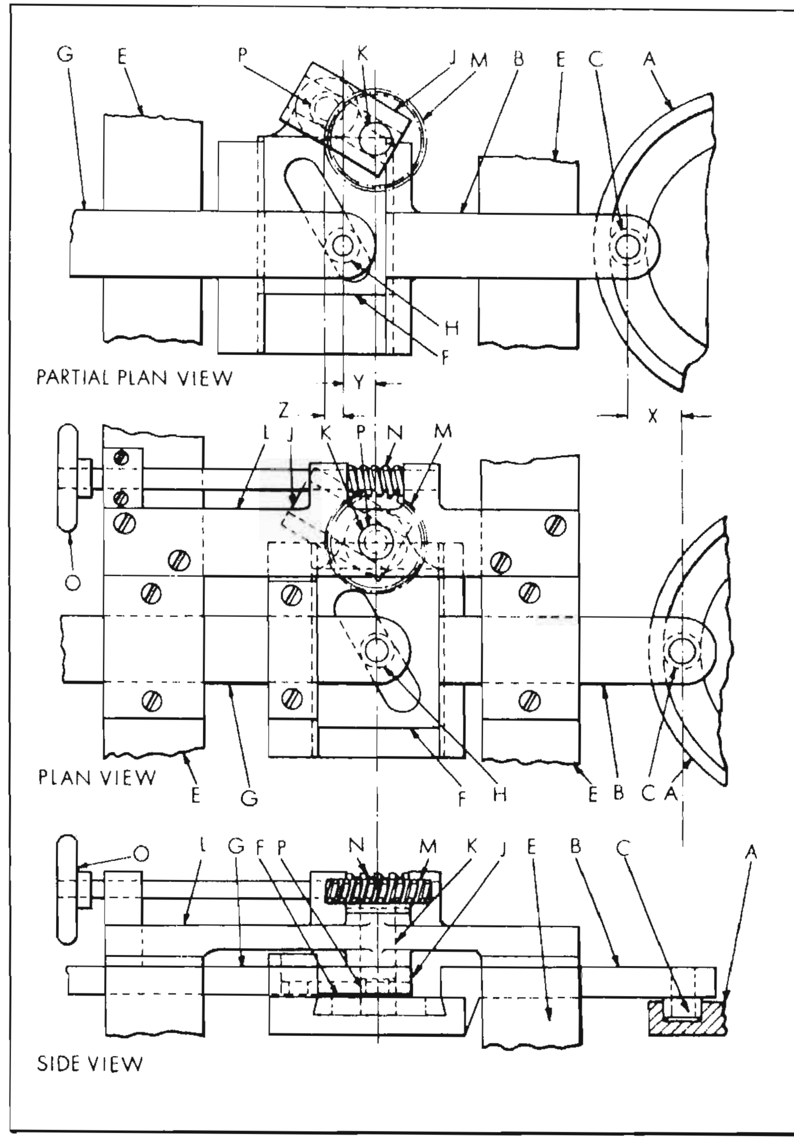


FIG. 16. Mechanism designed to obtain variable reciprocating motion of work-transfer slide *G* on a wire-forming machine from an eccentric *A* that is rotated at a uniform rate. The length of stroke of slide *G* is controlled by the angular setting of block *J*, which can be varied by rotating handwheel *O*.

its movement to the left, roller *C* having reached the high point on eccentric *A*. Also, slide *F* has been moved by the action of roller *P* in the groove in block *J*. Comparing the position of roller *H* in the plan view with its position in the partial plan view, it will be noted that its horizontal position relative to the center line of slide *F* has been moved to the right a distance *Z*, which represents the distance lost in the movement of slide *G* due to the upward movement of slide *F*. The actual movement of slide *G* is distance *Y*, which is equal to the rise of the eccentric *A* (as indicated by distance *X*) minus the distance *Z*.

As previously mentioned, the angularity of block *J* can be adjusted by the handwheel *O* through worm *N* and worm-wheel *M*. If block *J* were set with its groove parallel with the line of motion of slide *B*, there would be no movement of slide *F* within slide *B*. As a result, there would be no lost motion, and the movement of slide *G* would be equal to the movement of slide *B*. If block *J* were rotated counter-clockwise beyond this parallel setting, the resultant downward movement of slide *F* would cause an increase in the movement of slide *G* over that of slide *B*. The setting of block *J* should not be perpendicular or almost perpendicular to the line of motion of slide *B*, since binding would occur.

It should be noted that in the plan view (center sketch) the axes of shaft *K* and roller *P* exactly coincide. This condition will exist at all times when roller *C* is at the low point on eccentric *A*, regardless of the angularity of block *J*. Therefore, the loading point of the transfer movement will remain unchanged, regardless of the stroke adjustment.

Modified Eccentric Provides Adjustable Die Stroke

On a machine employed for producing stamped wire products, one of the dies is carried on a reciprocating slide. In the original design of the reciprocating mechanism, shown at the top in Fig. 17, the slide *C* on which the die was mounted fitted into a dovetail in the bed of the machine, and was reciprocated by an eccentric *B* carried on a rotating shaft *A*. Despite the fact

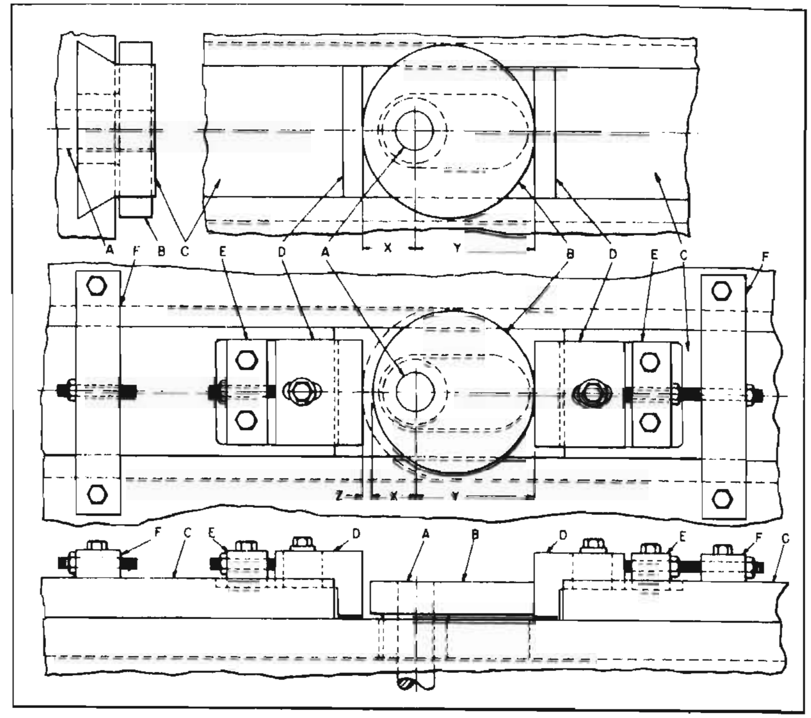


FIG. 17. In original design of reciprocating mechanism (top), the high point of eccentric *A* wore rapidly. In modified design (center and bottom), contact plates *D* are adjustably mounted on slide *C* to provide take-up for wear.

that the contact surfaces of the eccentric and plates *D* were hardened, wear on the high point of the eccentric soon resulted in lost motion.

Since the only means of correcting the lost motion was to refinish the contact surfaces of the eccentric and the wear plates and then to apply shims behind the plates, it was decided to redesign the mechanism to provide take-up for wear. This was accomplished by means of the modifications shown in the center and bottom views of Fig. 17. The redesigned mechanism also provides a means for varying the length of stroke of slide *C*.

In the center view, the outline of the original eccentric is shown in broken lines, while the modified eccentric is shown solid. With this modification it can be seen that dimension X has been shortened, producing a space Z that results in some lost motion before slide C begins moving to the left. As the "throw" of the eccentric is equal to $Y-X$, the reduction in dimension X increases the throw, and the modified eccentric is therefore capable of moving the slide a greater distance. Since, however, the contact plates D are spaced the same distance apart as in the original design, the effective throw of the eccentric is not fully utilized and the travel of slide C is exactly the same.

When the high point of the modified eccentric becomes worn, resulting in a reduction in dimension Y and the travel of the slide, the slide travel can be increased by reducing the distance between the contact faces of plates D. This is easily accomplished since the plates are adjustably mounted on slide C in the modified design.

Blocks E, carrying screws for accurately adjusting the position of plates D, are also mounted on the slide. Bars F, secured to the machine bed, carry adjustable screws which limit the slide stroke.

Adjustable-Stroke Driving Mechanism Reciprocating Slide

The unusual lever driving mechanism shown in Fig. 18 was designed to replace an ordinary connecting-rod arrangement used to impart a reciprocating motion to a machine slide. This slide, indicated at H, was an essential part of a wrapping machine and had to be driven by a constantly revolving shaft in the machine. Adjustments in the position of the reciprocating stroke were frequently necessary.

When making adjustments with the original equipment, it was necessary to stop the machine, sometimes three or four times in succession, before obtaining the correct amount of slide travel or proper positioning of the reciprocating slide in its guideways. With the new drive, provision was made for obtaining adjustments without interfering with the machine drive in any way. The mechanism for the control was designed to accomplish its purpose in a quick and safe manner.

Referring to Fig. 18, member A is the driving shaft which rotates in a horizontal bearing B. Shaft A is driven at a constant speed of rotation in a clockwise direction by a simple spur gear drive (not shown) at the rear of bearing B.

To the front end of shaft A is keyed a cast-iron circular disc C, which carries the projecting crankpin D fastened in place by a lock-nut at the rear of the disc flange. The crankpin is immovable in the disc and cannot be adjusted to obtain variations in the length of stroke as in the original drive. Mounted on the crankpin is the connecting-rod E, this being retained in the correct endwise position by the collar F.

A connecting-rod G, having the same effective length as E, is linked at its right-hand end to the reciprocating slide H by pin K.

Slide H moves in the horizontal plane within the stationary dovetail guide track L. The remaining end of links E and G and

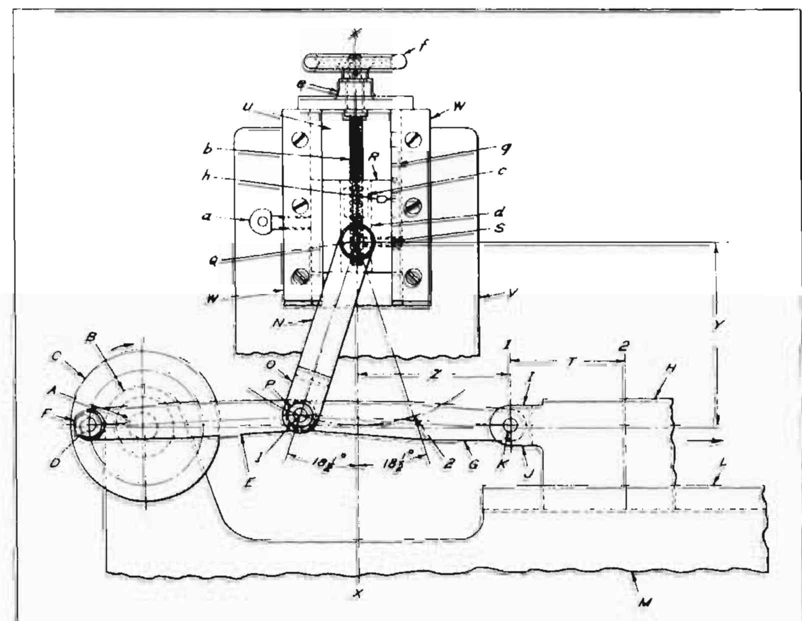


FIG. 18. Adjustable-stroke driving mechanism for reciprocating slide set for minimum stroke length of slide H.

the lower end of link N are linked by and pivot about stud P. The upper end of arm N is attached to the front of the adjustable slide R by a fulcrum stud Q.

Slide R is actuated by a fine-pitch lead-screw b. The upper end of the lead-screw is guided in the fixed bearing plate e fastened to the top of the guide bracket. A handwheel f is keyed to the projecting upper end of the lead-screw, so that it may easily be rotated. The top surface of the retaining plate W at the right-hand side has the accurately measured gradations g. The small pointer h, fastened on top of slide R, is set adjacent to the graduation marks to facilitate setting the slide.

The position of the stroke of H is adjusted by rotating handwheel F to change the position of stud Q. It should be noted that as the length of link N is not infinite, the length of stroke will change a small amount as its position is changed.

Converting Oscillating Circular Motion into Variable Reciprocating Movement

An unusual lever mechanism designed to transform a uniform circular oscillating motion into a periodically varying reciprocating movement is shown in Fig. 19. This simple and inexpensive mechanism was devised to drive the former slide on a special wire-forming machine. The slide had to be reciprocated with a gradually accelerating rate of speed (faster than that of the driving shaft) for a portion of its stroke and an equally decelerating rate of speed for the remaining portion of its stroke. Means also had to be provided for increasing or diminishing the duration of these accelerations and decelerations and for altering the length of stroke without stopping the operation of the mechanism or machine.

The driving shaft and the cast iron disc attached to it are oscillated through 120 degrees by an eccentric, not shown, acting through a short link arm.

An elongated radial slot is machined in the front face of the disc. The length of the slot is determined by the length of stroke required for the driven slide and the permissible outside diameter of the disc. A phosphor-bronze slider block, fitted into this

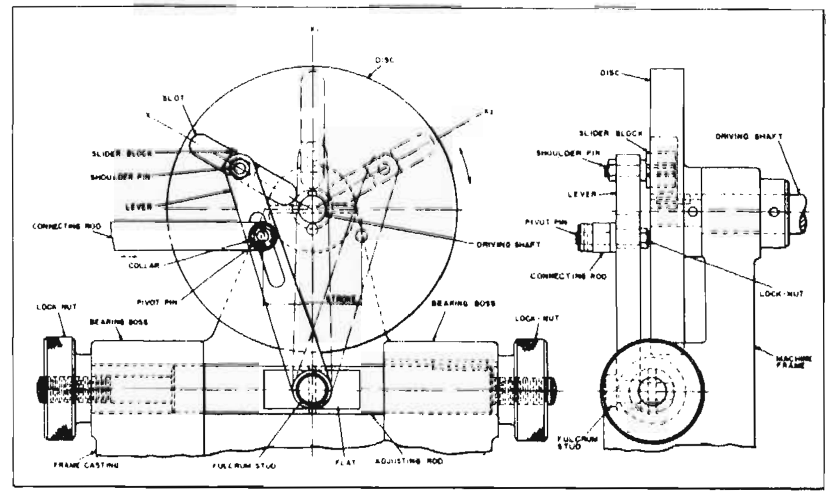


FIG. 19. Mechanism designed to drive a former slide on a special wire-forming machine.

slot, is secured to the end of a lever by means of a threaded shoulder-pin and lock-nut.

The lever is linked to a connecting-rod by means of a pivot pin, located in a slot machined in the center of the lever. A collar pinned to the projecting end of the pivot pin retains the connecting-rod, but allows it to swivel about the pin. The pivot pin may be adjusted to any desired position along the slot, and is locked in place by a nut at the rear of the lever.

The opposite end of the connecting-rod is attached directly to the driven slide (not shown), which has to reciprocate horizontally. The lower end of the lever is mounted on a stationary fulcrum stud that is fixed to the side of a cylindrical adjusting rod. This horizontal rod is carried in two bearing bosses on the machine frame. The lever, thus mounted, can swivel freely upon the stud. A flat on one side of the adjusting rod provides a bearing surface for the location of the stud and the side of the lever.

Each end of the rod is reduced and threaded, and is provided with a knurled lock-nut which bears against the end face of the bearing boss. By adjusting the two lock-nuts, the rod can be moved horizontally to give any desired setting for the fulcrum stud. A key is provided within the right-hand bearing boss to prevent the rod from rotating.

As shown, the rod has been adjusted so as to bring the center of the fulcrum stud exactly in line with the vertical axis of the driving shaft and disc. With this setting, the lever will move an equal distance — 60 degrees — each side of the vertical center line of the mechanism. The disc is shown about to commence its forward oscillation in a clockwise direction. This position of the elongated slot in the disc is denoted by X. Both the lever and the slide attached to the left-hand end of the connecting-rod thus are in their extreme left-hand positions.

As the disc moves clockwise, the lever will be drawn to the right. However, since the lever is fulcrumed at a much greater radius than that of the shoulder-pin on the disc, the slider block will gradually move down the slot and approach the center of the disc. Throughout that portion of the stroke of the disc between positions X and X_1, the connecting-rod and the attached former slide will travel at an increasingly faster rate than the driving shaft.

When the lever has passed position X_1, however, the slider block will gradually move outward in the disc slot, and the driven slide will be correspondingly slowed down. This deceleration will be at exactly the same rate as an acceleration occurring in the first portion of the stroke. When point X_2 has been reached, reversal occurs.

Throughout the return stroke from X_2 to X_1, the speed of the lever and the driven member will be increased; their speed will be decreased when the lever passes point X_1. Thus, for each complete oscillation of the disc and its driving shaft, the driven slide will receive two accelerations and two decelerations of equal duration.

The maximum stroke is obtainable by adjusting the pivot pin so that the connecting-rod is set at the highest point within the slot in the lever. By moving the pin to the opposite end of the slot, so that it lies at the shortest possible distance from the center of the fulcrum stud, the minimum stroke is obtained.

Variations of stroke length ranging between these maximum and minimum values can be readily obtained by changing the position of the fulcrum stud relative to the vertical axis of the mechanism. Such changes can be effected while the mechanism is running merely by adjusting the lock-nuts and the lateral position of the adjusting rod in its bearings.

Unique Pumping Mechanism Applied to a Homogenizer

A mechanism designed to impart pulsations to three plungers has been incorporated in a homogenizer. Hydraulic suction and pressure are alternately obtained in this equipment by rubber pulsators R through the action of pistons P (see Fig. 20).

The pumping portion of the mechanism, with ball suction and discharge valves and a pressure relief value, is shown at the right-hand end of the drawing. The driving mechanism consists of a camshaft D, on which are mounted three cams C_1, C_2, and C_3, which actuate pistons P. There are three sets of parallel pistons, plungers, etc., which extend toward and through the pump housing.

Camshaft D is geared to drive-shaft B. Pistons P, their cylinders, and part of their operating mechanism are constantly immersed in the actuating fluid, which is oil.

Each pulsator R is made of synthetic rubber and bonded to a sleeve E. A metal support S inside the pulsator is connected to cylinder F. Each piston is provided with a collar H that transmits the pressure of spring J.

Rocker arms A pivot on shaft T. Each cam rides against a ball-bearing roller G which is mounted on a stub shaft in a corresponding arm A. A second ball-bearing roller on this arm contacts the left end of the piston. In the position shown in the illustration, cam C_1 has pushed piston P into the discharge position, against the pressure of spring J. When the camshaft makes a half revolution, cam C_1 will have moved 180 degrees to permit piston P to move toward the left and uncover ports O to compensate for any slight oil leakage from the cylinder and prepare for the discharge stroke.

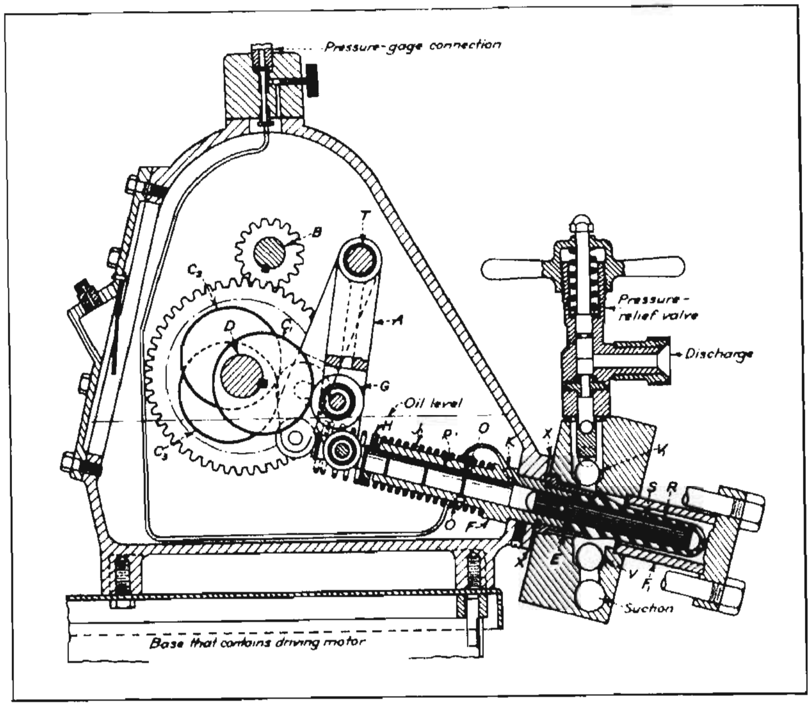


FIG. 20. Triplex type of pumping mechanism which imparts three pulsations to a homogenizer with each revolution of a camshaft.

Pressure inside the rubber pulsator is relieved at this time, and it contracts to create a vacuum inside cylinder F_1. Liquid then flows into this cylinder through the ball suction valve V. When cam C_1 makes another half revolution, arm A moves piston P into the position shown, and forces oil into the rubber pulsator. The pulsator then expands and forces liquid out of cylinder F_1, through the ball discharge valve V_1 and out through the discharge connection.

The pump, being of a triplex design, makes three pulsations for each revolution of the cam-shaft, which insures a relatively constant flow of oil. The discharge pressure is measured by a gage connected to the cylinder at K.

CHAPTER 8

Crank Actuated Reciprocating Mechanisms

The special designs of crank mechanisms described in this chapter are for transmitting motion to slides or other parts having a reciprocating action. These drives may be arranged to produce some special movement, such as, for example, arresting the motion of the slide momentarily during some part of the stroke or providing a quick return movement to reduce the idle period; or the design may be special in that certain parts of the mechanism are to be operated at constant or varying velocities.

Other crank-actuated reciprocating mechanisms are described in Chapter 9, Volume I and Chapter 8, Volumes II and III of "Ingenious Mechanisms for Designers and Inventors."

Oscillating Shaft Driven with Simple Harmonic Motion

In a printing machine it was desired that a shaft be rotated through 180 degrees, and at the same time that the angular motion of the shaft be simple harmonic.

A mechanism that fulfills the requirements is shown in Fig. 1. Its mode of operation is as follows: To the input shaft A is fastened gear B, which is in mesh with gears C and D (both having the same diameter). Gears C and D rotate discs G and H through the shafts E and F, in the same direction and with the same angular velocity. Discs G and H carry crankpins I and K. They, in turn, carry the rack L, which is in mesh with gear M.

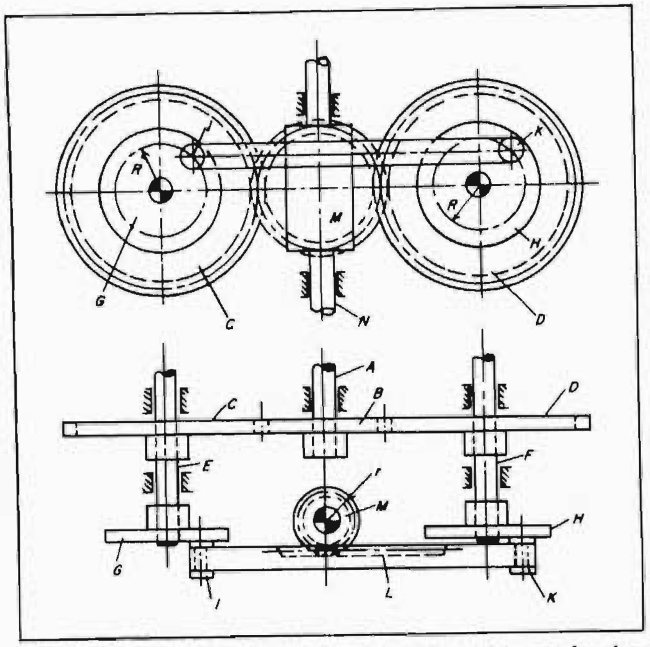


FIG. 1. With drive-shaft A in constant rotation, this mechanism causes output shaft N to oscillate with simple harmonic motion through 180 degrees when proportioned according to the stated formula.

When discs G and H rotate they will impart a simple harmonic motion to gear M and output shaft N. The angle ϕ through which N rotates is determined by

$$\frac{2\pi r \times \phi}{360} = 2R \text{ or } \phi = \frac{360\,R}{\pi r}$$

With geometrical proportions shown in the sketch, $\phi = 180$ degrees.

Rapid Return and Dwell Period Provided by Spring-Loaded Eccentric

A conventional eccentric and connecting-rod were used to apply holding pressure on a wire product during a machine-forming operation. Although this setup was satisfactory for the job at hand, a product design change made it necessary to modify the clamping motion slightly to furnish a quicker release of the wire part. This revised mechanism is shown in Fig. 2.

Eccentric A is free to rotate on hub B of carrier dog C. The carrier dog is, in turn, keyed to shaft D, which rotates in the direction indicated by the arrow. Pin E is the only connection between the eccentric and the dog. The pin, passing through eccentric A, contacts carrier dog C during part of the operating cycle.

The end of pin E that protrudes from the front of the eccentric supports bushing F. An external annular groove cut into the bushing carries one end of tension spring G. A pin H and a bushing J, supported by the machine frame, carry the opposite end of the spring. The machine-slide to be reciprocated (not shown) is attached to the upper end of connecting-rod K.

When the mechanism is functioning, carrier dog C rotates with shaft D in a counterclockwise direction until the dog contacts the end of pin E. Further movement of shaft D will cause eccentric A to rotate, at the same time extending spring G. The eccentric, as shown at X, is in its uppermost position, or point of greatest pressure application to the work. Pin E, at this point, is slightly above the center of shaft D. Continued rotation

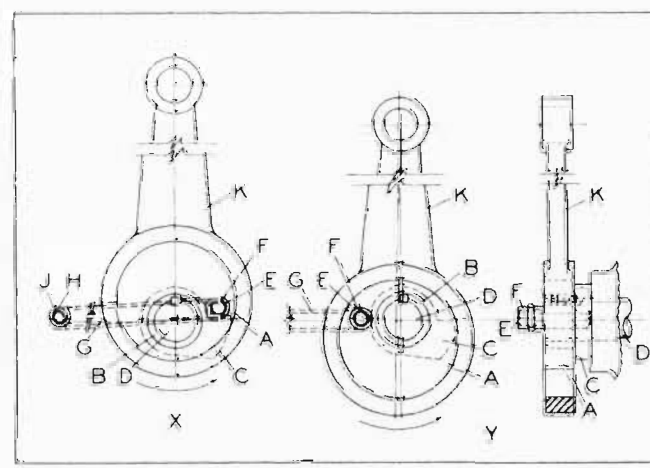


FIG. 2. Modified eccentric provides rapid return of machine-slide followed by a dwell period.

of the eccentric will cause the high-pressure point to be passed, at which time spring G will pull the eccentric sharply in its normal direction of rotation. As a result of this movement, connecting-rod K and its attached machine-slide undergo a rapid partial return.

At Y, in Fig. 2, can be seen the position assumed by eccentric A and pin E at the end of their spring-induced rotation. The tension of spring G prevents additional rotation of the eccentric until carrier dog C once again contacts pin E. Thus, in addition to a rapid partial return of the machine-slide, a dwell period is provided.

Generating Two Reciprocating Strokes for One

An arrangement that produces two strokes for each stroke of a reciprocating driving member is shown in Fig. 3. Alternate movements generated in each direction are of different lengths. This mechanism was designed and used to position a forming head on a machine for processing a wire product.

Lever A is mounted on two studs B and C which are located on a stationary part of the machine. Two curved slots permit movement of the lever of these studs. Each slot is machined to a radius centered on the opposing stud (B or C) when the lever is in the position shown in view X. Driving member D and driven member E are pivoted on lever A by means of studs G and H, respectively. In addition, a spring F resists motion of the lower end of the lever to the left. Member D is actuated by a cam on the wire-forming machine.

View X shows the device at the midpoint of the movement of bar D to the left. In this position studs B and C are in contact with the ends of their respective slots. As driving member D completes its travel to the left, lever A pivots on stud C in the lower slot to the position shown in view Y. This causes member E to move a distance L in the same direction. On reversal of member D, lever A again pivots on stud C returning member E to the center position.

After driving member D passes the midpoint, continued movement to the right causes lever A to pivot on stud B, again mov-

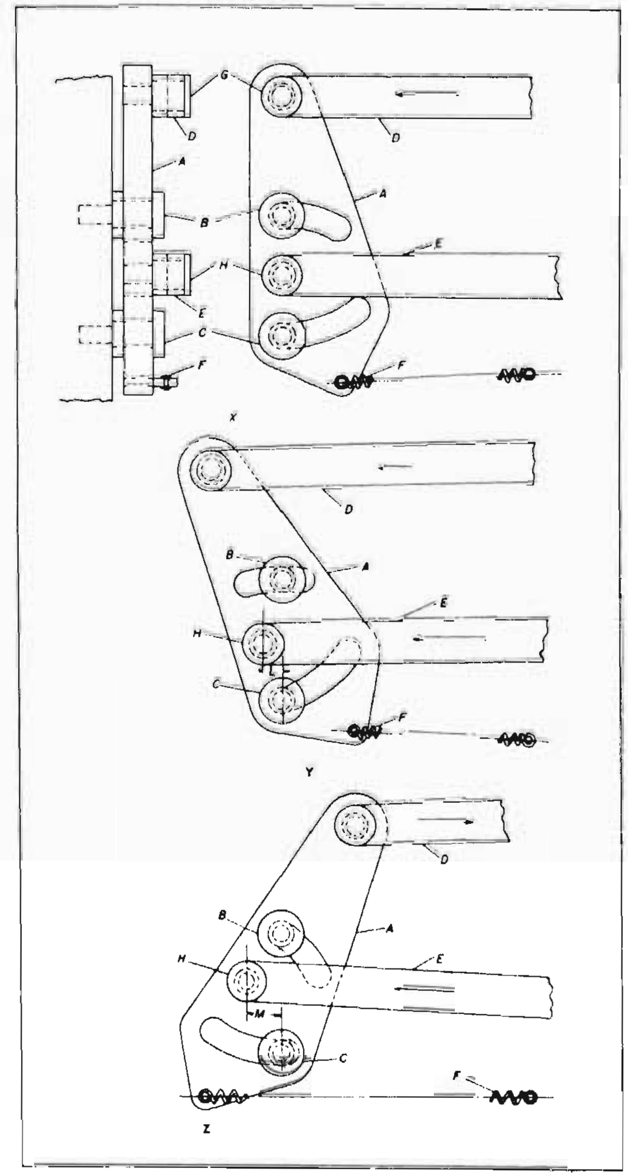


FIG. 3. Lever arrangement that provides two strokes for every stroke of a reciprocating drive rod D. Alternate strokes in each direction are of different lengths.

ing member E to the left. At the end of this stroke, member E is in the position shown in view Z and has traveled a distance M. This is greater than distance L in view Y, due to the change in the fulcrum from stud C to stud B. The location of stud H relative to studs B and C also influences the amount of movement given to member E.

The purpose of spring F is shown in Fig. 3. In view X member D is moving in the direction shown by the arrow and the resistance of member E and its load holds the end of the lower slot in lever A in contact with stud C. When member D returns to the mid-position in the opposite direction, the resistance of member E and its load tends to permit lever A to pivot on stud H. However, sufficient resistance to this movement is applied by spring F to the lower end of lever A, thus insuring that it pivot on stud C. The spring also insures that lever A pivot on stud B in returning to the mid-position from the right.

Rotating Crank that Provides a Dwell in Reciprocating Motion

In a certain mechanism it was necessary to obtain a reciprocating motion with a dwell period at one end of the stroke. This was accomplished by designing the crank and cam mechanism shown in Fig. 4.

The device consists of a crank rotating in the usual fashion but with its crankpin arranged to move radially in a slot provided in the crank-arm. Used in connection with a suitable stationary cam, a cross-head driven by a connecting-rod can be made to follow many different modifications of the reciprocating crank movement. Although the cam-follower in this case is located on the center line of the crankpin, it can be located to suit operating conditions.

Crank A is keyed to shaft B. Crankpin C and cam-follower D are arranged to move radially with a block E which slides in a slot in the crank-arm. The block is retained in the crank-arm by gibs F. The cam-follower and crankpin are secured to block E by means of a screw and nut, as shown in cross-section X-X of the assembly.

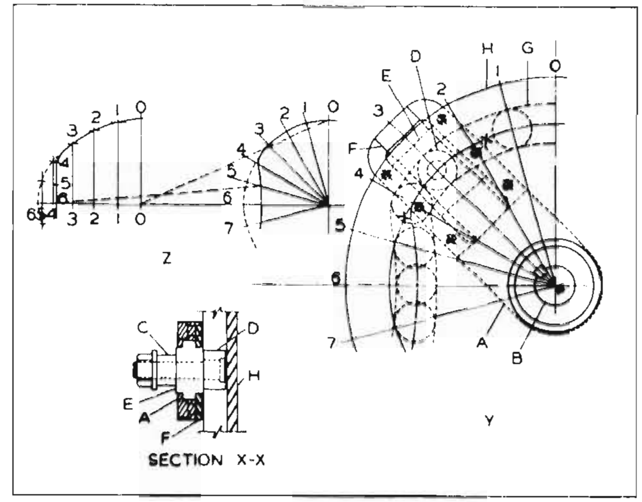


FIG. 4. Rotating crank and cam mechanism produces dwell in reciprocating movement.

The reciprocating motion with a dwell is obtained when cam-follower D moves within a recessed cam-groove G in a stationary plate H during the swing of the crank, as indicated in view Y. By referring to the diagrams showing the movement of the crank device, view Z, it will be apparent that when the follower D passes points numbered 0, 1, 2, and 3, the driven member will follow the normal crank-arm arc. However, since the path through points numbered 5 and 6 is concentric with the point at which the connecting-rod is attached to the cross-head or other reciprocating member, no horizontal movement can occur as the cam-follower moves through this portion of its path. In diagram Z, the diagonal dotted line between points 0-0 and the dotted line between points 5-6 represent two positions of the connecting-rod.

Since part of the horizontal movement of the driven member is lost at the dwell portion of the cycle, the length of the crank-arm radius must be suitable for the desired total movement. The transition from movement to dwell period can be varied to suit conditions.

Crank-Driven Plate Obtains Near-Uniform Velocities Through Compensating Cam

In a certain manufacturing process, sheets arriving on one conveyor section are transferred to another section by means of a suction plate. This plate is crank-driven through a connecting-rod and has a straight-line movement between the ends of the sections. Sheets arrive at regular intervals, and it is essential that the movement of the suction plate by synchronized and approximately uniform during the pick-up and deposit of each sheet, since the transfer cannot be made instantaneously.

The design of the transfer mechanism created a problem, because simple linkage would obviously convert the constant angular velocity of the crank to a continuously varying linear velocity of the suction plate.

The time in the cycling of the plate when a uniform movement was wanted corresponded to 10 degrees of crank rotation during the pick-up of the sheet, and another 10 degrees during its deposit.

A compensating cam solved the problem. As can be seen in Fig. 5, suction plate *A* is joined by connecting-rod *B* and link *C* to crank *D*. The crank is keyed to drive-shaft *E* which rotates at a constant speed. Compensating cam *F*, having a drop *G* and a rise *H*, is rigidly attached to the frame.

At one end, link *C* carries a roll *J* which is spring-loaded against the cam. The drive-shaft rotates clockwise. When the roll contacts the drop on the cam, the effective length of the crank is decreased, and, thus, the velocity of suction plate is reduced. Conversely, the velocity of the suction plate is raised when the roll is on the rise on the cam.

The drop and rise areas are so located on the cam as to produce the near-uniform velocities at the desired points in the cycle of the suction plate. In one instance, an unwanted acceleration of the plate is cancelled by a drop in the cam, and in the other instance, an unwanted deceleration is cancelled by a rise in the cam.

Winding Head for Skeins of Embroidery Floss

Skeins of embroidery floss are formed by winding the thread around two stationary bars. As each skein is formed, it is moved along the bars, away from the winding head, to be cut and labeled. The path of a winding arm swinging around the outside of the two bars must not be restricted by any type bracket, such as might be required for retaining the bars in a stationary position. The illustrated mechanism, utilizing a Scotch yoke, is being used in such a machine.

Floss to be wound passes through a hole drilled in the center of shaft *A* (see Fig. 6). The shaft is supported in bearing bracket *B* and is retained by collar *C*. From there it continues through hollow winding arm *D*. The winding arm is revolved at high speed, wrapping the floss about the two stationary arms *E* mounted in bracket *F*.

Support for this bracket is obtained from pins *G* and *H*, which can slide in holes drilled through integral bracket ears. Two

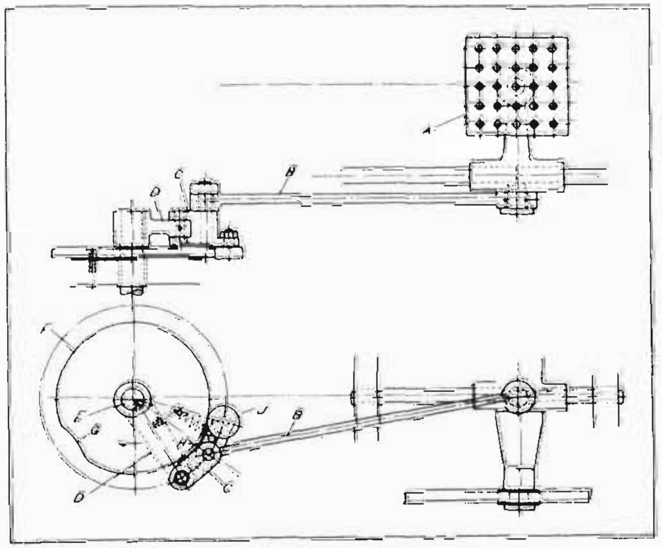


FIG. 5. The net effect of compensating cam *F* is to produce near-uniform velocities at two points in cycle of suction plate *A*.

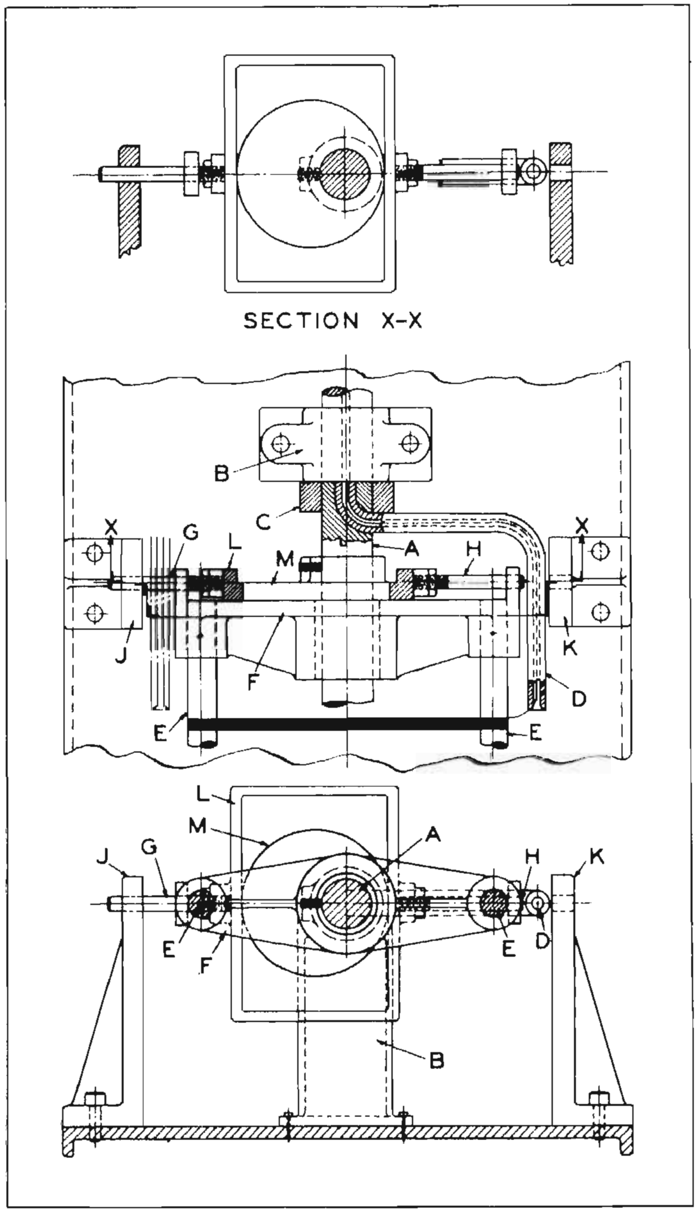


FIG. 6. Head for skein-winding machine features Scotch yoke for reciprocating movable bracket-support pins.

identical angle supports *J* and *K*, screwed to the machine base, are drilled to receive the outer ends of the pins. The inner ends of the two pins are threaded into bosses on either side of Scotch yoke *L*. Cam *M*, mounted on shaft *A*, rides within the yoke.

The cam is timed to move pin *G* into engagement with support *J* when the winding arm is passing between bracket *F* and support *K* as illustrated. As the winding arm approaches the left-hand side of the unit, the yoke and attached pins move to the right. This advances pin *H* into engagement with support *K*, while retracting pin *G* from the opposite support. A path is thus opened between bracket *F* and support *J* to permit unrestricted passage of the winding arm.

Bracket *F* is supported at all times. During the time that winding arm *D* is approaching either vertical position, and just after it leaves that position, both pins are engaged simultaneously. The appropriate pin is completely disengaged only during the instant necessary for the winding arm to pass. The relationship of the working members to each other may be more clearly seen in section X-X.

Oscillating Drive-Shaft Actuates Slide at Variable Speed and Stroke

A slide to move light packages across a wrapping table had to be driven by an oscillating shaft and lever. The position, length, and amplitude of the lever arm were fixed, since the lever also actuated another mechanism synchronized with the motion of the slide. It was not possible to attach the lever directly to the slide because of certain peculiarities of the slide motion.

First, the slide stroke had to have a range of adjustment, from a 9-inch minimum to a 10½-inch maximum (direct attachment of the lever and slide would have produced a 13-inch fixed stroke). Second, in its forward movement the slide initially had to travel in unison with the lever for a distance of 2 inches, then gradually decelerate; and in its return movement, the slide initially had to accelerate gradually to a point 2 inches from the end of the stroke, then travel in unison with the lever. Third, provision had to be made to vary within small limits the distance that the slide traveled in unison with the lever. It was required that the distance of unison slide vary from 2 inches at the maximum slide stroke to 3¼ inches at the minimum stroke.

The mechanism that was devised is shown in Fig. 7. The upper and lower views show, respectively, the position of the elements at the beginning and end of the forward stroke. Slide *A* is dovetailed to and reciprocates along guideway *B* formed in the vertical side of frame *C*. Oscillating drive-shaft *D* and lever *E* are supported on the same frame, but at a considerable distance from the slide.

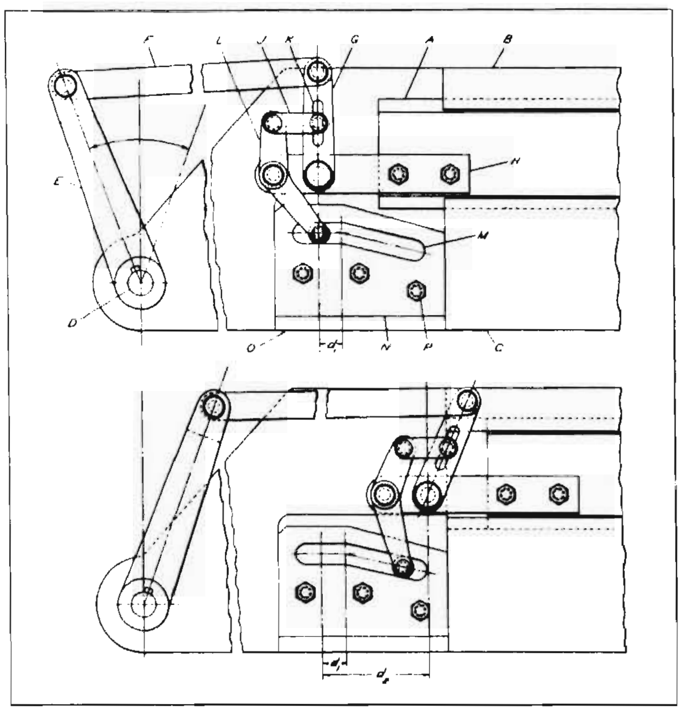


FIG. 7. Converting the oscillation of drive-shaft *D* to a reciprocation of slide *A* through this mechanism permits the stroke to be of adjustable length and made at varying velocity.

The top of lever *E* has a slot through which is pinned a long rod *F*. At one end, rod *F* is joined to arm *G*, which, in turn, pivots on bar *H* fastened to slide *A*. Link *J* pivots around an adjustable point in slot *K* of arm *G*, and the left-hand end of the link is pinned to the upper limb of bellcrank *L*. The bellcrank swings on bar *H*, and its lower limb carries a roller fitting slot *M* in plate *N*. It will be noted that one section of the slot is shorter and parallel to guideway *B*, and the other section is longer and at a decline. The extent of these two sections, as well as the angle of the decline, is carefully determined in accordance with the desired total length of slide travel and range of speed variation.

Providing bell crank *L* is prevented from rotating, arm *F* will push slide *A* in unison. At the start of the stroke, bell crank *L* is prevented from rotating by keeping the distance between the attached roller in *M* and the slide *A* constant. When the roller enters the inclined part of slot *M*, the distance increases causing bell crank *L* to rotate clockwise. The clockwise rotation of *L* causes *A* to move to the left with respect to arm *F*, thus causing deceleration. If the stud connecting *J* and *G* is lowered, rotation of *L* will increase deceleration as it can cause greater rotation of *G*. Lowering of the stud will also change the distance of unison travel. However, the distance of unison travel can be readjusted by changing the position of *N* in slot *O*. Elevating the position of the stud between *J* and *G* will decrease the amount of deceleration.

Drives for Reciprocating Members with Varying Relative Motion

When designing special-purpose machinery for packaging articles of various kinds, it is often necessary to have two elements move in unison for a period of time and then have one travel forward at an accelerated rate. Generally, there is insufficient space to enable each movement to be obtained by the usual method with cams, links, and multiple slides. Such an arrangement may also result in excessive overhang or undue remote operation.

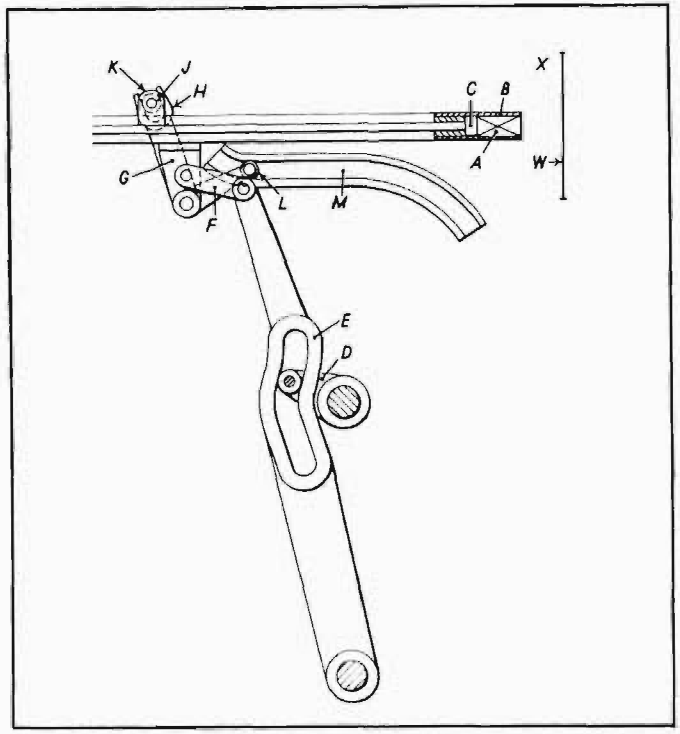


FIG. 8. Transfer device in which an ejector is made to travel at a faster speed during part of the cycle.

A packaging machine may be required to hold the article to be wrapped, transfer it into its wrapper while performing some of the folding operations, and finally eject it into some other receptacle. Alternatively, the machine may be required to apply a hold-down plate to an article while it is being elevated and enveloped on three sides by the wrapper. After this, the wrapper is folded onto the base of the article and then onto the end faces. In the next movement, the article is transferred over a folding plate to complete the wrapping, and the package is carried to the following work station.

The requirements of the first method are met by the arrangement shown in Fig. 8. An article A is temporarily held between a pair of blades B, at the rear of which there is a sliding plunger

C. These components are carried through the various movements by means of a crank D, driving a slotted lever E. A link F connects lever E to a sliding carriage G. For actuating plunger C separately, there is a bellcrank lever H, one end of which is forked to engage a roller J. This roller is carried by a stud supported on a bearing bracket K attached to the bar carrying plunger C. The other arm of bellcrank lever H is fitted with a roller L, which engages the track of a stationary cam M. The pivot stud of lever H is secured to sliding carriage G.

When the article A has been introduced between plates B, the carriage assembly moves to the right. As the article reaches work station X, it is thrust into the wrapper W, since plunger C has by this time moved the trailing edge of the article to the leading edge of blades B, through the action of cam M on bellcrank lever H. At this point, blades B have almost reached the end of their travel, and because of the form of cam M, the plunger is traveling at an increased rate. As a result, the article is pushed from between the blades, and is deposited ready for the following operation.

A mechanism designed for the second set of conditions is illustrated in Fig. 9. It is shown at the completion of its stroke and about to return to the starting position, indicated by the phantom view. With the mechanism in the starting position, the article A is pushed up through well B, together with the wrapper so that the latter is folded into an inverted U-form. The top of the article is held in contact with stop plate C by a platform (not shown). Plate C rests on the article for the duration of the period that the latter is being enveloped by the wrapper, and travels with it until the partially enclosed article has been positioned at the next work station in the packing sequences.

The plate is mounted on a sliding carriage D to which is also attached a pusher E. The latter is fitted with wrapper-folding members (not shown) that make the two rear side folds on the package. Carriage D is reciprocated on guide ways F by means of a link arm G, pivoted on lever H. Member H, in turn, is driven by a crank J by means of a connecting-rod K.

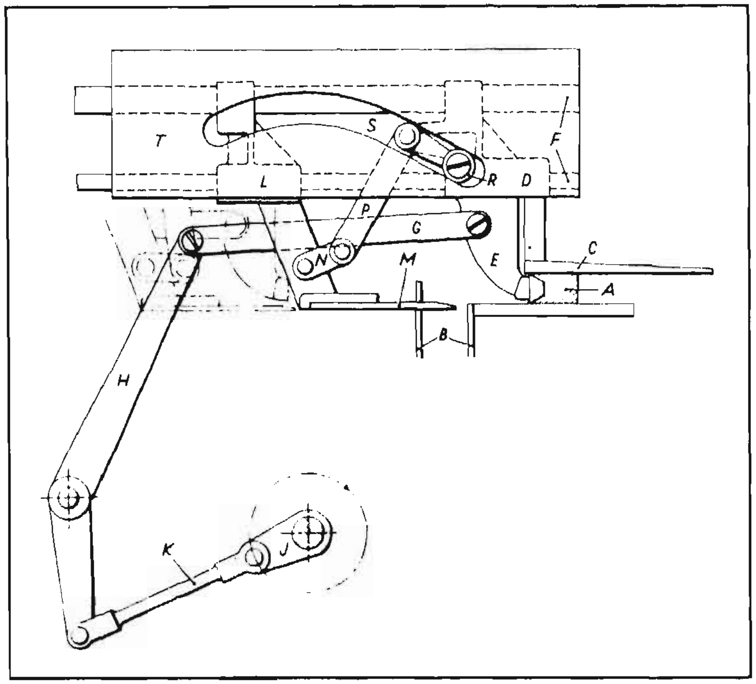


FIG. 9. Crank-driven mechanism with a stationary cam to vary the speed of member *M* for wrapper folding.

A second sliding carriage *L*, to which is attached a bottom folder *M*, is moved on the same guide ways as carriage *D* by means of link *N*. This link is pivoted on the long arm of bellcrank lever *P*. The latter, in turn, is pivoted on a stud on carriage *D*. The short arm of bellcrank *P* carries a roller *R* which engages a cam slot *S* in a stationary plate *T*.

As the stop plate *C* and the plunger *E* are being moved to the right to engage the rear of the partially wrapped article *A*, the bottom folder *M* initially moves forward at a faster rate because of the upward curvature of the cam slot *S*. Thereafter the rate is reduced, and the folder finally dwells while pusher *E* moves the article on to the next work station. At this stage, the article has been enclosed on four surfaces by its wrapper and an inward fold has been made at the rear of each end of the package.

CHAPTER 9

Variable Stroke Reciprocating Mechanisms

Means of adjusting the length, speed or timing of the reciprocating stroke are described in this chapter. Other variable stroke reciprocating mechanisms are described in Chapter 9, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Fixed Stroke Converted into Variable Stroke

The demand for varying widths of a flat wire product where previously only one standard width was processed made it necessary to provide existing machines with a simple mechanism to insure even winding onto spools. Even winding was accomplished without modifying the width of the spools by providing an adjustment for the stroke of the guiding member.

As can be noted in Fig. 1, when double screw *A* is rotated, it transmits reciprocating motion to the slide *C* through the forked, swinging follower *B*.

To provide for adjustment in the length of travel, a few modifications had to be made. A rack-and-pinion and "Scotch yoke" arrangement was devised which consisted of a gear, rack, roller, slide-bar, and support block. Gear *D* rotates freely on a stud in slide *C* and meshes with rack *E* which is attached to a stationary part of the machine. Any linear movement of slide *C* produces rotation of gear *D*, the pitch diameter of the gear *D* being such that it will be given a complete half-revolution as the slide *C* covers its range of travel. The hub *I* of gear *D* has a T-slot containing a T-bolt on which roller *F* rotates freely.

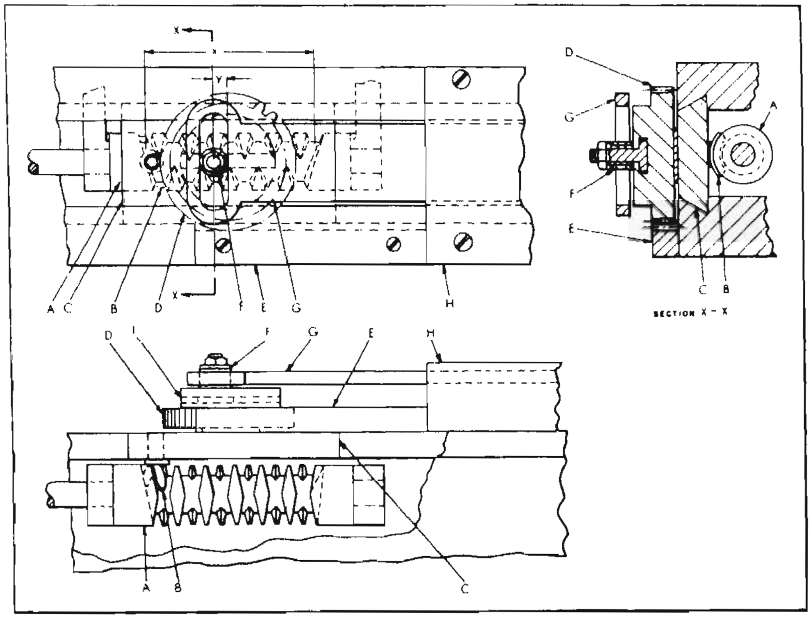


FIG. 1. This mechanism, which originally consisted of a double screw and slide arrangement to give a fixed stroke, was modified by the addition of a rack-and-pinion and a "Scotch yoke" to provide an adjustable stroke.

Movement of the T-bolt in the T-slot permits adjustment of the position of roller F relative to the axis of gear D. Roller F engages a slot in slide-bar G which, in turn, is supported in block H. To the opposite end of bar G is fastened the guide which delivers the product to the spools.

The center of roller F is offset from the center of rotation of gear D, a distance y. As gear D rotates, the center of roller F describes a semi-circle, with radius y, about the axis of gear D. Roller F acts in the slot of slide G, and a half-revolution of gear D rotates to the opposite side of the center of gear D. The relative movement between the bar and the gear is $2y$. Since the center of gear D moves through a distance x due to the linear movement of slide C, the combination of the two movements results in a total length of travel of slide G equal to x plus $2y$.

If the roller F were placed on the opposite side of the center of gear D, with the slide C in the position shown, the total movement of slide G would equal x minus $2y$. Furthermore, if roller F were placed exactly over the center of gear D, roller F would merely rotate on its center, and the total movement of slide G would be the same as that of slide C.

Although the added motion is harmonic it is small enough not to be objectionable.

Variable Straight-line Reciprocation from Uniform Rotary Motion

On a machine producing a woven-wire product, some of the strands of wire are displaced varying amounts to produce a random decorative pattern. This is done by converting a uniformly rotating input to a straight-line reciprocation which can either change length or be the same length from action to action.

Shaft A (see Fig. 2), mounted in a stationary part of the machine, rotates at a uniform rate in the direction indicated by the arrow. The shaft carries and is keyed to a disc B, which in turn carries a pin C that extends from one side.

Ring D is grooved to varying depths on equally spaced radial lines, each of which is numbered in the drawing. The ring is mounted on plate E, and is free to rotate. Two guide bars F are carried by the plate, and slide freely in channels, being retained by strips G. These guide bars are wide enough to support the ring in the back.

Plate E carries an extension on its top, to which the guide for the wire is attached. Stop-pin H, at the bottom, serves to hold the sliding parts in the starting position between movements.

In the position of pin C shown, it is entering the No. 1 groove in the ring, and the ring thus has to rotate in the direction indicated by the arrow. Until the pin contacts the bottom of the groove, the movement of the ring is rotative only. But when the pin reaches the bottom of the groove, it transmits to the ring (and thus, to the plate) a linear movement as well.

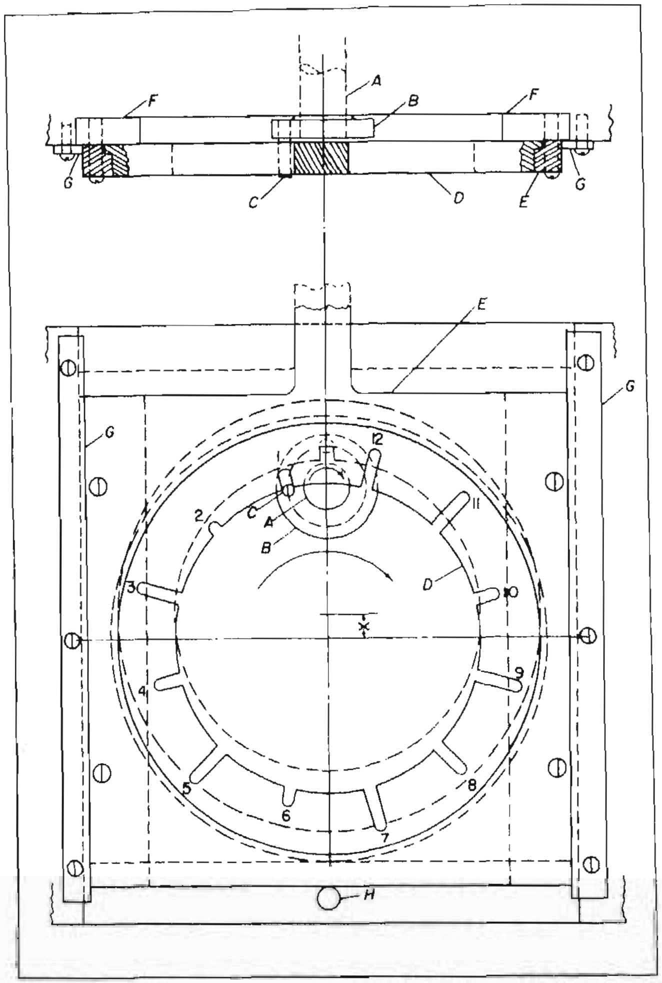


FIG. 2. Length of linear movement of plate E is controlled by the respective depths of the twelve grooves in ring D.

By the time the pin reaches the vertical center line of the mechanism, both the ring and plate E have moved upward a distance equal to x. Continued movement of the pin transfers the No. 1 groove to the position previously occupied by the No. 12 groove, each rotation of disc B indexing the ring to the next groove.

The amount of linear movement given the plate is governed by the depths of the grooves in the ring. Since, in this instance, the No. 2 groove is the shallowest, it will produce the greatest amount of plate movement. Grooves Nos. 3, 5, 7, 9, and 11 are deep enough so that the pin cannot contact their bottoms. Thus, when the pin is engaged in any of these grooves, the ring rotates without any accompanying linear movement of the plate, as is desired.

On the machine to which this mechanism is applied, the axes of the rotating parts are in a horizontal plane, so that the weight of the plate will return it to contact stop-pin H after each reciprocation. In an application where the axes are vertical, a return spring will be required.

Adjustable Eccentric

In the design of various types of machines, an eccentric motion is often needed to operate certain mechanisms. A relatively simple arrangement for obtaining such eccentric movement from a rotary drive is shown in Fig. 3. The resultant drive is positive and can be varied quickly and easily, thus avoiding time-consuming dismantling and reassembly.

In construction, adapter A, made from a rectangular piece of stock, is turned on one end. In addition, a hole is bored

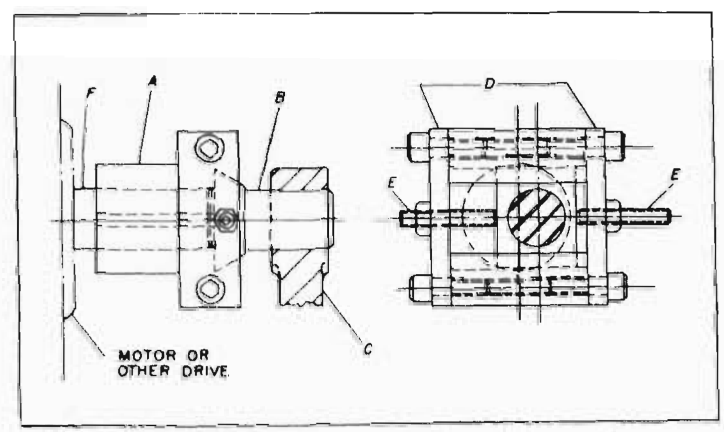


FIG. 3. Simply constructed adjustable eccentric.

through the part and a keyway is provided to fit the driving shaft. A female dovetail is milled in the rectangular section at the opposite end. From a square piece of stock, a male dovetail slide *B* is then machined to fit the adapter. One end of this member is turned to fit a connecting-link arm *C*. Two side-plates *D*, made from flat stock, are bolted to the sides of the adapter with socket-head cap-screws, as shown. Each plate also has a hole tapped to receive headless set-screws *E*, which are used for adjusting the amount of eccentricity and for locking the slide *B* securely in place during operation of the connected drive.

The slide and adapter may be scribed with gage lines, if desired, in case minute adjustment is needed.

Eccentric Driving Mechanism Permits Stroke Adjustment During Operation

Small variations in the stroke length of a reciprocating slide can be made while it is operating, by means of the mechanism shown in Fig. 4. Driving disc *A* has an integral shank revolving in fixed bearing *B*, where it is retained by bearing cap *C*. Driving gear *D*, keyed to the shank, revolves continuously.

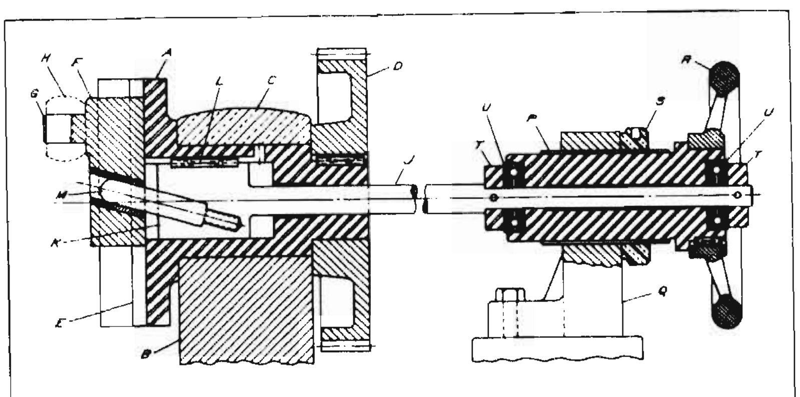


FIG. 4. By revolving handwheel *R*, the throw of crank pin *G* can be varied without stopping the slide.

The disc face contains a dovetail *E* milled across its diameter. Crankpin block *F*, fitting the dovetail, has an integral crankpin *G*, over which is fitted one end of a connecting-rod *H*. At its other end, the rod is attached to the reciprocating slide (not shown).

Rod *J*, by which the device is adjusted, has a shouldered section *K* fitting the bore of disc *A*. Key *L* causes the rod to revolve with the disc yet permits a short axial movement of the rod along the bore. A hard pin *M* is pressed at an angle into the end of section *K*. This pin engages a hole that is drilled in the crankpin block.

Rod *J* extends from any convenient distance to a control point. At its right end, the rod is reduced in diameter and can rotate in sleeve *P*. An external thread on the sleeve engages a threaded hole in angle-bracket *Q*. By revolving handwheel *R*, keyed to the sleeve, the sleeve can be adjusted axially. Threaded ring *S* locks the sleeve, once it has been adjusted. Rod *J* moves axially in unison with the sleeve, by means of the stop collars *T* and thrust bearings *U*.

If the stroke of the slide has to be lengthened, ring *S* is released, and the handwheel revolved counterclockwise. This movement is transmitted to the rod, and pin *M* is retracted a corresponding amount from the crankpin block, causing the block to move radially outward in disc *A*. Thus, crankpin *G* has a greater throw. By revolving the handwheel clockwise, the throw of the crankpin is similarly decreased.

Mechanism for Effecting a Varying Reciprocating Motion

On a wire-forming machine, a reciprocating slide was required to move at a uniform rate of speed during part of the cycle and, at a predetermined point, to increase in speed for the remainder of the cycle. The motive power for operating this slide was taken from a uniformly reciprocating slide and then transformed into varying reciprocating motion by the mechanism here shown.

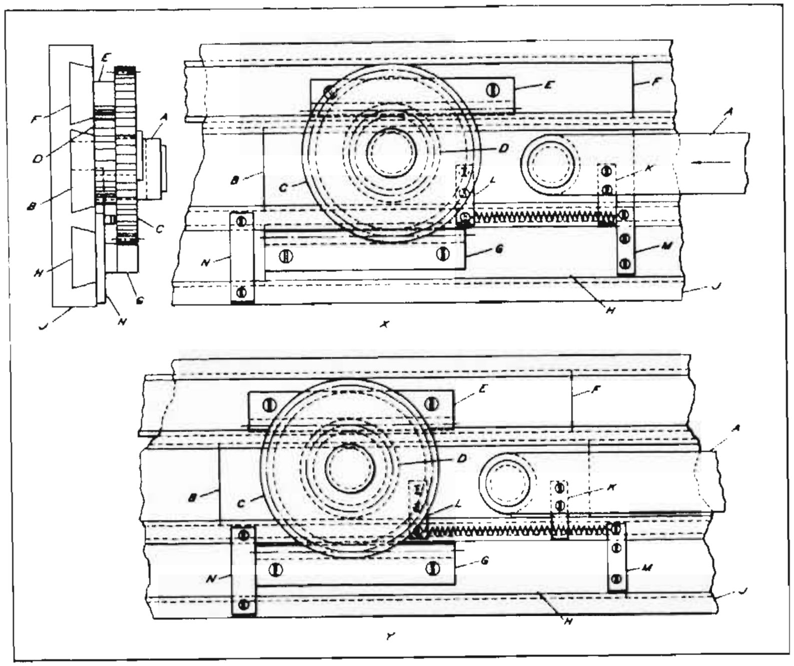


FIG. 5. Three-slide arrangement, utilizing gears in the capacity of a lever, varies motion supplied by a uniformly reciprocating source.

Rod A (see Fig. 5), reciprocates at a uniform rate of speed and drives slide B. This slide carries gears C and D which are keyed together, yet rotate freely on a stud. Gear D meshes with the rack E which is fastened to a second slide F. The larger gear meshes with rack G on a third slide H. All three slides are dovetailed in the stationary slide bracket J.

Dog K and spring retainer L are attached to slide B, while spring retainer M is secured to slide H. A stop N is fastened across the lower dovetailed slot.

In operation the uniform reciprocation is obtained from rod A which moves slide B to the left. As long as contact between parts M and K is maintained by means of the tension spring, the central and lower slides will move in unison. The gears cannot rotate at this point as there is no change in the relative

positions of slides B and H. Therefore the three slides will move together during this phase.

It will be noted that, at Y in the diagram, rack G has contacted stop N, thus preventing further movement of slide H. As slide B continues its movement, gear C is forced to rotate due to the relative movement between it and the now stationary rack G. Gear D, which rotates with the larger gear, transmits this motion to the upper slide through rack E. Slide F now moves at a greater speed than slide B. The ratio of gears C and D governs the extent of the increased speed of slide F as compared with that of slide B.

On the return stroke of rod A, the upper and central slides will move to the right in the same speed ratio with respect to each other as they did on the forward stroke. This will continue until dog K contacts spring retainer M, at which time all three slides will once again move together.

Adjustable Eccentric Produces a Variable Throw

Rotary movement of a driving shaft must be converted frequently into rectilinear movement for the actuation of such members as slides or levers. One of the most common and effective means for accomplishing this conversion is the combination of an eccentric and connecting-rod. In its conventional form, however, an eccentric is applicable only when a fixed length of stroke is desired. If, on a particular machine, it is necessary to vary the stroke length imparted to the driven members, it is advantageous to employ an adjustable eccentric drive mechanism.

Figure 6 shows the design of such an adjustable eccentric. Outer eccentric member A has a running fit within the head of a connecting-rod (not shown) which couples the eccentric to the reciprocating machine member. A large hole is bored through the outer eccentric member, being offset distance X which is determined by the amount of throw required.

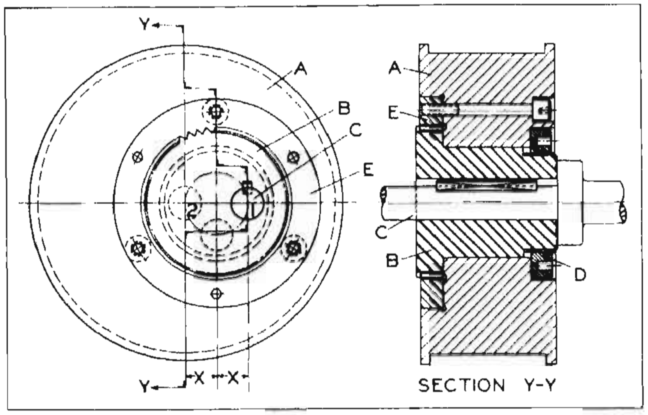


FIG. 6. In this drive, throw of outer eccentric *A* is adjusted by rotating sleeve *B* carrying drive-shaft *C*.

Mounted in the bored hole is flanged sleeve *B*. Sleeve *B* is bored to receive, and is keyed to, drive-shaft *C*. The shaft hole is located eccentrically in the sleeve, the amount of offset provided in the illustrated unit being *X*. Circular lock-nut *D* is threaded on the right-hand end of the flanged sleeve to retain it in place. Spanner holes are drilled in the face of the lock-nut to facilitate removal and replacement.

The flanged end of sleeve *B* has a series of accurately spaced vee serrations machined on its periphery. The serrations mesh with similar internal serrations provided on ring *E*. Cap-screws and dowels, passing through eccentric member *A* from the opposite side, locate and retain ring *E* in the recess provided. The dowels must be of sufficient size in order to bear the main driving load.

With sleeve *B* situated so that shaft *C* is on the horizontal center line in the position shown at 1, maximum throw of the eccentric is obtained. If lock-nut *D* is backed off, and sleeve *B* is rotated 180 degrees in relation to the outer member, shaft *C* will be coincident with the center of the eccentric, Position 2. At this setting, no motion will be delivered to the machine slide. Sleeve *B* may be located at any intermediate position between the neutral and maximum throw settings.

Steplessly Variable Stroke Movement

An arrangement employed on an optical profile-grinding machine that enables the length of stroke of a horizontal reciprocating slide to be steplessly varied is shown in Fig. 7. Of compact design, the unit has provision for bolting directly to a speed reducer.

Body *A* is keyed to the shaft of the reducer and houses a slide *B*. This slide incorporates a crankpin and is retained by a cover plate (not shown in the elevation). From the pin, motion is transmitted to the machine slide by a connecting-rod.

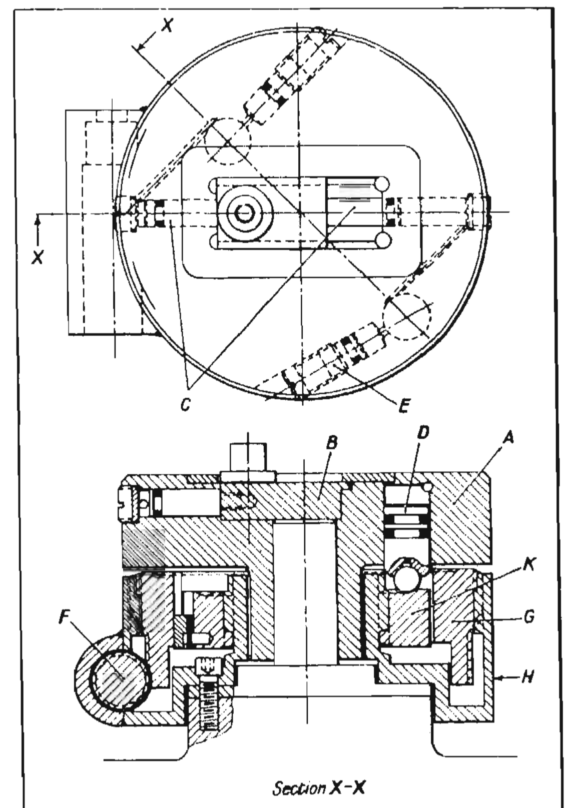


FIG. 7. Device for steplessly varying stroke of a reciprocating drive. Stroke can be changed while drive is in motion.

Two pistons C, forming part of a closed hydraulic system, determine the position of the slide B and, consequently, the throw of the crankpin and the stroke of the machine slide. The pistons C are controlled by pistons D, the cylinder bores being connected by transverse ports as seen in the elevation. To facilitate charging and setting, a piston E with screw adjustment is provided for each branch of the hydraulic circuit in this mechanism.

Adjustment of the stroke can be effected while the machine is running by turning a handwheel connected to the worm F. This member meshes with a worm-wheel G which in turn is threaded into a stationary housing H, permitting the worm-wheel to move into or out of the housing when the worm is rotated. A left-hand stub Acme thread is used for this purpose. From the worm-wheel, drive is also transmitted through a sliding phosphor-bronze key to the inner ring K, which has a right-hand thread. In consequence, this ring is always moved in the housing in the opposite axial direction to the worm-wheel.

Movement is transmitted from the worm-wheel or the inner ring to one of the pistons D, depending on the direction of the stroke adjustment, through a ball which runs in a track on the end face of the appropriate member (K or G). As one piston is raised, the other is permitted to move downward by reason of the right- and left-hand thread arrangement, and a corresponding motion is imparted to the pistons C and slide B which carries the crankpin.

Mechanism That Imparts Variable and Unequal Strokes to Opposed Reciprocating Slides

Incidental to the modification of an existing machine, it was found necessary to actuate two opposed reciprocating slides. Forming tools were to be mounted on each of the slides. The drive for the mechanism had to be powered by a constant-speed shaft that also motivated other machine movements. The length of stroke of one slide had to be variable, and the stroke of each slide had to start and end at the same instant. Also, the slides

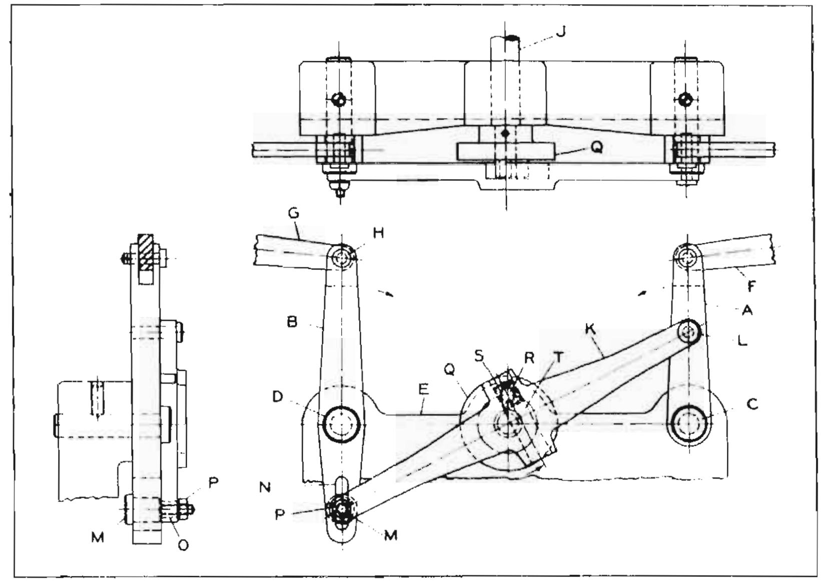


FIG. 8. Drive mechanism for opposed reciprocating slides that permits stroke adjustment of one slide.

were always to move in opposite directions. The mechanism illustrated in Fig. 8 was designed to satisfy these conditions.

The two levers A and B are mounted to pivot freely on headed studs C and D, respectively. The top half of lever B and lever A are of the same length. The levers are mounted along the same horizontal center line in machine frame E. The upper end of each lever is slotted for short connecting links F and G. Each link pivots on a headed stud H. The opposite end (not shown) of each of these connecting links is coupled to a reciprocating slide.

The two levers are mounted at approximately the same distance each side of the driving shaft J which extends from a bossed portion of the machine frame. The levers are constrained to pivot in opposite directions by means of the long steel connecting-rod K. The right-hand end of the connecting-rod is coupled to lever A by the headed stud L secured to the lever at a fixed center distance from the fulcrum stud C. The connecting-rod pivots on stud L. The left-hand end of connecting-rod K is attached to lever B in such a way that its point of pivoting may be changed.

A crank pin R is pressed in disc Q which is keyed to driving shaft J. A slide S pivots about pin R and slides in slot T in connecting-rod K. The rotation of crank pin R slides S in slot T and reciprocates K. K, in turn, reciprocates B and A in opposite directions. The amount of throw of pin R is such that link F moves the required distance. The size of the reciprocation of G can be varied in any instance by altering the position of stud M in slot N.

Lever Type Driving Mechanism Permits Stroke and Dwell Adjustments

The lever type mechanism shown in Fig. 9 was incorporated in the drive of a wrapping machine, and was required to operate a transfer slide which transported wrapped packages from the machine to a conveyor belt alongside. A drive shaft with an oscillating rotary movement of 35 degrees was the source of motion for the driving mechanism. Another factor involved in the reciprocation of the transfer slide was that it had to be readily adjustable to suit the various sizes of packages normally handled by the machine.

Drive shaft A, shown at X in Fig. 9, is mounted horizontally within a bearing hole through the small upright boss B.

Securely keyed to the drive-shaft are two identical levers D, one being situated at each side of boss B. They are retained in position by means of the headed end E of the drive-shaft at the right-hand side, and collar F, cross-pinned to the shaft, on the left-hand side.

Pin G, which supports lever-arm H, passes through in-line holes in the upper end of the twin levers D, and is retained by collar J. The upper limb of the lever-arm H pivots about pin K within the forked end of connecting-rod L. Connecting-rod L is linked directly to the transfer slide of the wrapping machine.

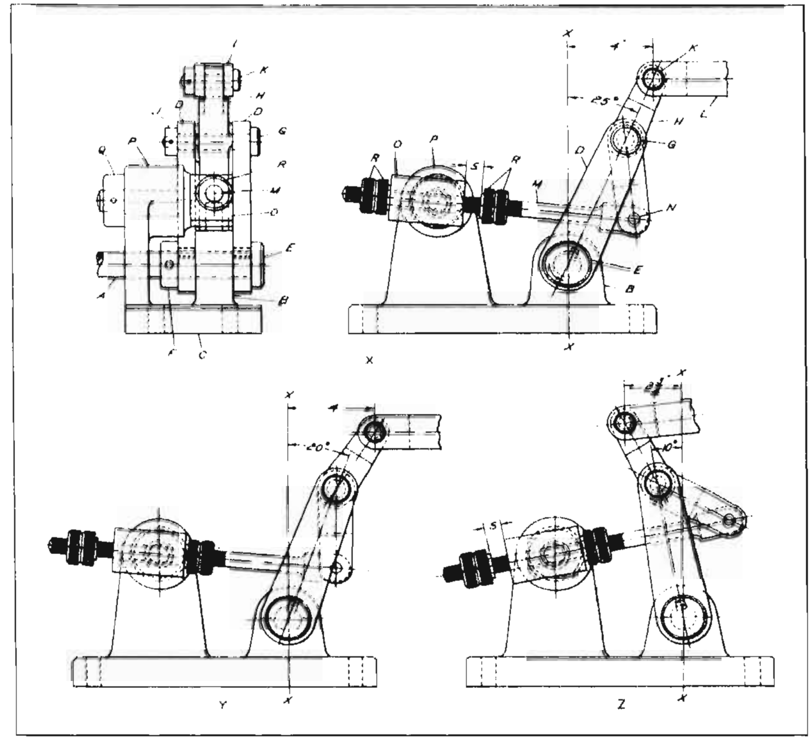


FIG. 9. Knurled nuts of the lever type driving mechanism here shown may be adjusted during operation to vary the dwell duration and also the points of connecting-rod reversal.

The lower limb of the lever-arm lies at a slight angle to the upper limb, and is slotted to receive the end of rod M. A pin N joins these two members so that they are free to pivot.

The opposite end of rod M can slide within trunnion block O. A shaft, machined on one side of the block, rides in a bearing hole in a large boss P. Collar Q is pinned to the shaft to retain the trunnion block in position.

For the major portion of its length, rod M is threaded to receive four knurled nuts R, which are located as shown. The nuts are adjustable, and are normally secured in any desired setting by simply locking them together in pairs.

The diagram at the right-hand side of X shows the relative positions assumed by the various components as drive-shaft *A* starts to move in a counterclockwise direction.

At this point in the cycle, connecting-rod *L* is in its retracted position, that is, situated 4 inches to the right of vertical axis X-X. It will be observed that levers *D* are inclined at an angle of 25 degrees to the right of the same axis.

Distance *S* must be calculated according to the duration of the dwell period required at each end of the stroke. In the illustrated example, this distance is regulated so that the connecting-rod and the transfer-slide will remain inactive at each end of the stroke during a 5-degree travel of shaft *A*.

As the mechanism moves from its starting point in the direction of the arrow, connecting-rod *L* remains stationary due to the resistance of the transfer slide. *L* will remain stationary until the levers *D* have completed the 5 degrees of movement, whereupon knurled nuts *R*, at the right of the trunnion block, will contact that member, as shown at Y in the illustration.

As the drive-shaft continues its movement beyond this point, rod *M* is prevented from sliding any further to the left. Thus connecting-rod *L* receives the combined movements of levers *D* and lever arm *H*. This action may be visualized by referring to diagram Z in the illustration. The diagram shows the relative positions of the components when drive shaft *A* has reached the end of its forward oscillation. At this stage, levers *D* are inclined 10 degrees to the left of vertical axis X-X, while the connecting-rod has moved 2¾ inches to the left of the same axis, a total travel of 6¾ inches.

As the return stroke commences, the connecting-rod again remains stationary, due to the resistance of the transfer slide, for the first 5 degrees of drive shaft travel. After this distance has been covered, the knurled nuts *R*, on the left-hand side of the trunnion block, reach their limit of movement, and motion is once again transmitted through the connecting-rod to the transfer slide of the machine.

Duration of the dwells at each end of the stroke may be altered by adjusting the setting of the knurled nuts on rod *M* while the machine is operating at slow speeds. The nut-setting may also be altered to effect variations in the points of reversal of the stroke. By setting both pairs of nuts in contact with the end faces of the trunnion, the dwell periods will be eliminated and the connecting-rod will then have its maximum length of stroke.

Common Drive for Two Slides with Partially Synchronized Travel

A lever drive mechanism installed in a wire-bending machine now allows a high degree of versatility in the equipment. The mechanism has permitted two existing tool-slides to operate with partially synchronized travel, so that different sizes and forms of wire can be handled, and a greater variety of bent shapes can be produced.

Both tool-slides reciprocate in the same horizontal plane, but are a considerable distance apart in the machine. The mechanism transmits the drive from a common shaft, centrally located, which oscillates through an arc of 40 degrees. The first tool-slide has a short, fixed travel; the other, a considerably longer and adjustable travel. Although both tool-slides start and stop together, the second moves in unison with the first only during the initial portion of the forward travel and the final portion of the return travel. In the interim the second tool-slide increases in speed so as to compensate for its greater length of travel.

A full-scale diagram of the mechanism appears in Fig. 10. The oscillating drive-shaft *A* is carried in the bearing bracket *B*. Keyed to the drive-shaft and oscillating with it is a lever *C*. Through a slot in the top of the lever is cross-pinned a connecting link *D*. The opposite (left-hand) end of this link, not shown, is joined to the first tool-slide, which has the short, fixed travel.

A bellcrank *E* fulcrums on a stud *F* extending from one side of the lever. The second tool-slide with the long, adjustable travel, is joined to another connecting link *G* cross-pinned in a slot in the upper arm of the bellcrank *E*.

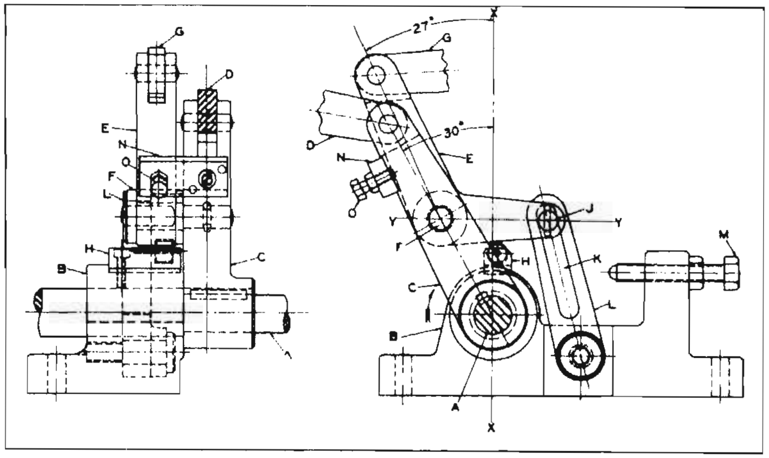


FIG. 10. At the start of the forward stroke, the bellcrank E moves in unison with lever C.

The lower arm of the bellcrank extends laterally, being supported by a roller assembly H mounted on the top of the bracket. The height of the assembly is designed so that the arm is in light contact with the roller when the mechanism is in its starting position, as shown in Fig. 10. The vertical center line of the roller is offset about 1/8 inch to the right of the vertical center line X-X of the drive shaft.

The ratio of the radii of the arms of the bellcrank determines the stroke-length range of the second tool-slide. In this particular instance, the radius of the lower arm is approximately two-thirds that of the upper arm.

At its extremity, the lower arm of the bellcrank contains a short dowel pin J which has a sliding fit in the slot K of a link L. This link pivots in a channel in the bearing bracket, and is a slight distance behind the lower arm of the bellcrank. A stroke adjusting screw M is located in an integral rib in one side of the bearing bracket, to the right of the bellcrank. As will be explained, this screw controls the extent that the second tool-slide travels in unison with the first tool-slide.

To the left side of lever C is doweled and screwed a short rectangular plate N. This plate extends across the adjacent side

of the bellcrank, and serves to transmit the forward motion of the lever to the bellcrank. Because of the slightly different angularity of the bellcrank, its side is somewhat clear of the plate. So that both tool-slides can start in unison, a set-screw O in the plate is adjusted to bear on the bellcrank.

When the drive-shaft starts its forward, or clockwise, oscillation — indicated by the arrow — lever C lies at an angle of 30 degrees to the left of the line X-X, and the upper arm of the bellcrank lies at an angle of 27 degrees; and pin J is on the horizontal center line Y-Y of stud F. Movements of the lever and bellcrank continue in unison until the lever is vertical, as in Fig. 11. At this point, link L has also pivoted clockwise into contact with screw M, and pin J has descended part way down slot K.

Since no further pivoting of the link is possible, continued movement of the lever is not transmitted uniformly to the bellcrank. Instead, the bellcrank fulcrums on stud F for the final 10 degrees of forward oscillation, Fig. 12; pin J descends to the bottom of the slot, and the link pivots to the left. Thus, the upper arm of the bellcrank, with its combined movements, travels in a

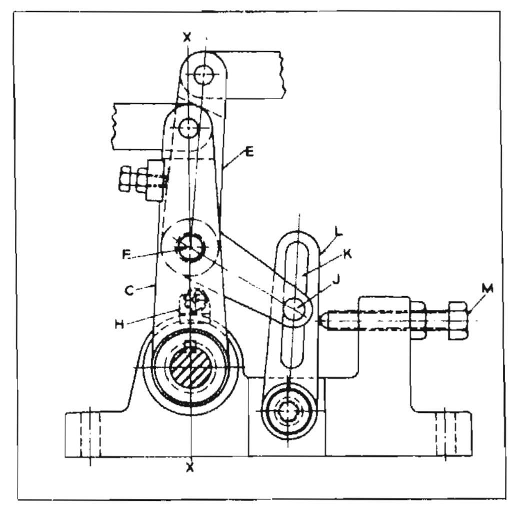


FIG. 11. When link L abuts screw M the bellcrank E starts to fulcrum on stud F.

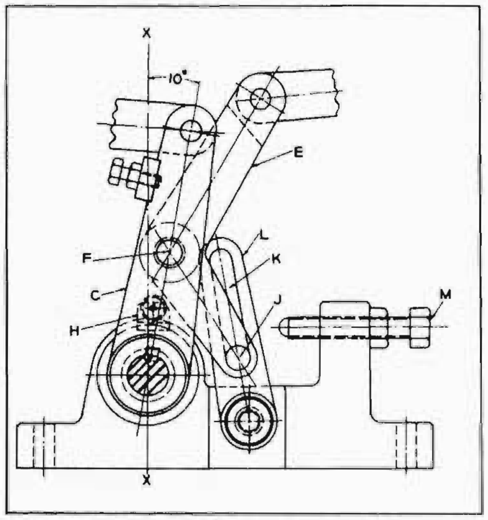


FIG. 12. At the end of the forward stroke, pin J has descended to the bottom of slot K.

considerably longer arc than the lever, causing the second tool-slide to travel a correspondingly greater distance and at a greater speed than the first tool-slide.

As counterclockwise rotation begins, the lower arm of the bellcrank rides over the roller assembly H, forcing pin J to rise in the slot. At the same time, the link pivots to the right into contact with screw M. Then, the bellcrank and lever move in unison for the balance of the return stroke. Thus, during the return stroke, the second tool-slide starts rapidly, then slows down to the speed of the first tool-slide — the reverse of the action during the forward stroke.

Two Rotary Slides Reciprocated in Synchronism on a Single Shaft

Two machine slides were required to rotate on a common shaft and, at the same time, to reciprocate along the shaft in opposite directions. A simple means had to be provided for altering the stroke length of both slides so that they could be adjusted to travel either equal or unequal distances with their reversal points occurring simultaneously. A mechanism incorporating these features is shown in Fig. 13.

Shaft A, shown at V, rotates slowly and carries the two machine slides B and C. The slides are forced to rotate with the shaft by means of keys D and E which are fastened to their respective members. A keyway F is machined along the length of the shaft to provide a sliding fit with the keys.

Reciprocating movements are imparted to slide B by T-shaped lever G, which pivots on stationary headed stud H. Connecting-rod J, providing the main source of motion, is free to pivot on stud K which connects it to the elongated slot at the lower end of the T-shaped lever. The length of the elongated slot is determined by the desired variation in stroke length of member B.

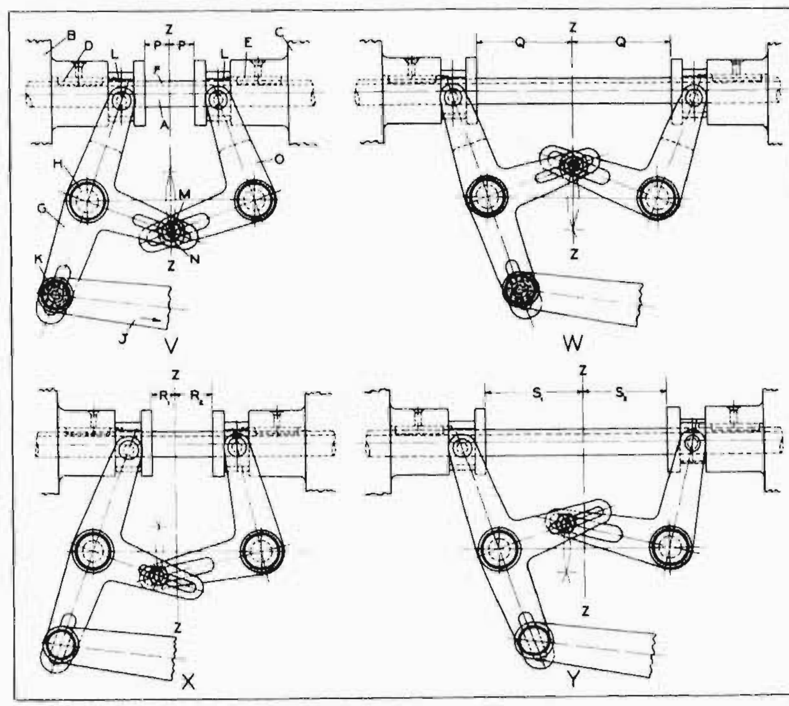


FIG. 13. Adjustable lever type mechanism provides synchronous reciprocation of two opposed, rotating slides.

The upper end of lever G is forked to straddle a cylindrical extension boss on the end of slide B. Each arm of the fork is linked with an annular groove in the extension boss by means of a hardened steel trunnion block L. In this way, the machine slide is free to rotate at the same time that it is being reciprocated.

Centrally located with respect to stud H, in the integral right-hand, or short, limb of lever G, is an elongated slot. Mounted in this slot is headed stud M whose opposite sides are flattened slightly to prevent it from turning in the slot, yet allowing it to slide freely. The stud is secured to the limb by lock-nut N.

A larger diameter of stud M has a sliding fit within an elongated slot cut along the lower limb of bellcrank O. The bellcrank pivots on a stud similar to H. Both of these studs are located the same distance from shaft A, and the same distance each side of vertical axis Z-Z. The uppper end of the bellcrank is forked, and is attached to slide C in the same way that lever G is attached to slide B.

The diagram at V shows the relative positions of the members when connecting-rod J is at its extreme left-hand position. With stud M locked in an appropriate position in the slot of lever G, and with the connecting-rod at its terminal point, slides B and C will be equidistant from vertical axis Z-Z as shown at P. As connecting-rod J moves to the right (arrow) on its return stroke, lever G will move counterclockwise, pivoting on stud H. This motion causes revolving slide B to move to the left. Similarly, bellcrank O will swivel in a clockwise direction, moving slide C to the right in unison with slide B. When connecting-rod J has reached the extent of its travel to the right, the two slides will still be equidistant from vertical axis Z-Z, as can be seen at W.

The illustration at X shows the position of the lever mechanism with connecting-rod J once again at its extreme left-hand position. In this case, stud M is set closer to fixed stud H, thereby conveying a smaller radial movement to the bellcrank. In this way, slide C will be moved through a shorter stroke than that imparted to slide B. The slides will, however, still move in unison, ending their sliding movements at the same instant,

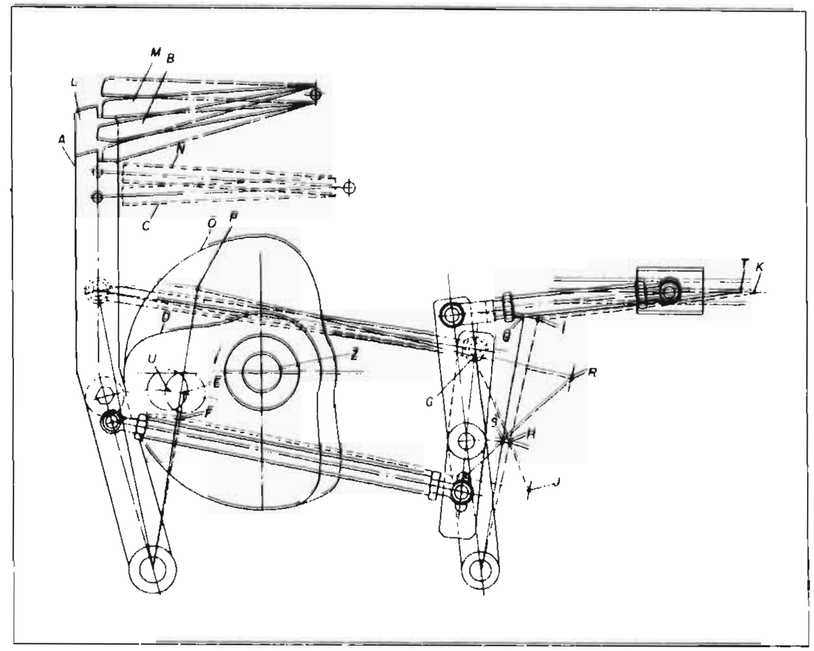


FIG. 14. The output motion to the slide, right, is changed in its stroke length by alternating the use of latches B and M.

although at unequal distances from vertical axis Z-Z, as seen at Y.

Slide Motion Differential

The object of the device is to change the length of slide stroke to position T or K by selection of latches M and B (see Fig. 14). The drive is rotating shaft Z. Levers A and L follow open track cams D and O. This view shows lever L blocked out by latch M. Lever A follows cam D under tension from spring C. Connecting-rod F is driven to the right, carrying the intermediate lever to successive positions G, H, J. The standing lever is thus carried to position I.

When lever A is latched out and lever L is allowed to follow cam O there is the same mode of operation but a different motion. The standing lever moves only to point Q, and the slide goes only as far as T.

Slide with Automatic Reversal and Adjustable Stroke

A half-nut that engages two lead-screws alternately is the heart of a slide mechanism having both automatic-reversal and adjustable-stroke features. Drawings of the principle of the device appear in Fig. 15.

The frame consists of two end plates *A*, square bars *B*, and round bars *C*. Both lead-screws *D* are supported in the end plates and are axially retained by collars *E*. During the operation of the slide mechanism, the lead-screws revolve continuously in opposite directions. The drive for this movement is introduced through the extended left-hand end of one lead-screw and transmitted to the other lead-screw through meshing pinions *F*.

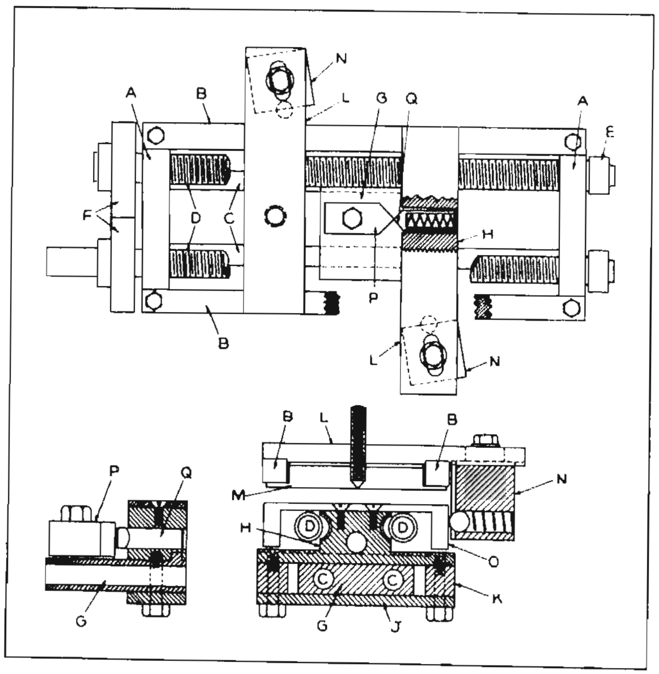


FIG. 15. Half-nut *H* is disengaged from the lead-screw *D* by the thrust of the trip block *N*.

Cross-head *G*, the member required to reciprocate, has a sliding fit on the round bars *C*. The half-nut *H* is splined to the top of the cross-head and is able to move transversely. This half-nut, which has the shape of an inverted T, has two thread sections, one adjacent to each lead-screw.

A keeper plate *J* prevents the half-nut from rising from the cross-head. This plate is joined to the base of the half-nut by cap-screws which run through spacing blocks *K*. Actually, the half-nut never remains in the neutral position illustrated but is engaged to one lead-screw or the other.

To set the stroke area as well as the length of stroke, two stops *L* are fixed along the frame where desired. These stops straddle the square bars *B*, to which they are clamped by keeper plates *M*. One end of each stop extends over an opposite side of the frame, and supports a trip block *N*. The working face of each trip block is set at an angle of 10 degrees to the center line of the device. In the working face is a spring-loaded ball detent.

A shift *O*, with a 10 degree bevel at either end, causes change of engagement of *H*.

In operation, one of the lead-screws moves the half-nut until the shift bar depresses the ball detent in the corresponding trip block, which then thrusts the half-nut out of engagement until it is slightly past a mid-center position over the cross-head.

The completion of the traverse movement of the half-nut *H* is caused by the action of detent *Q* on *P*.

Two Opposed Slides Driven with Rapid Variable Strokes

One phase of the operation of a particular wrapping machine involves the transfer of cartons across a stationary table. Each carton is gripped on two opposite sides by a pair of slides. Figure 16 shows a simple lever type driving mechanism designed to actuate the two slides with a synchronous movement in opposite directions.

Slides *A* and *B* may move freely within their respective dovetailed guide ways which are provided across the top of a cast-

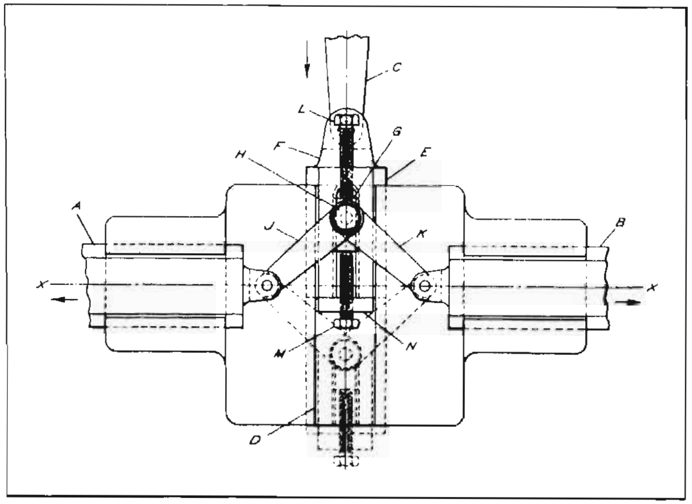


FIG. 16. An adjustable mechanism which drives two opposed slides from a single reciprocating rod.

iron baseplate. This baseplate is bolted to the frame of the machine in line with driving arm C which is a source of reciprocating motion.

Across the center portion of the baseplate is machined a third dovetailed guide way D situated at right angles to the first two. Riding within it is slide E. Lug F, which is integral with the third slide, is slotted to receive the end of the driving arm.

Machined along the top of slide E is a narrow T-shaped slot into which is fitted a steel sliding block G. This member is drilled to receive shoulder stud H. Mounted on the projecting portion of this stud are the ends of two identical levers J and K. The opposite ends of these two levers are connected to slides A and B respectively, as shown.

The sliding block G is locked in position by means of clamp screws L and M. Screw L is threaded through the rear end wall of slide E, while screw M passes through a small plate N. This plate is secured to the front face of the slide. When the lever mechanism is set for normal working conditions, the clamp screws L and M. Screw L is threaded through the rear end wall.

Operation of this mechanism and the manner of its adjustment will be clearly understood by referring to Fig. 16. The solid lines show the driven slides in their innermost position which occurs when slide E is fully retracted. To obtain the most efficient operation of the mechanism, the angle subtended by levers J and K in this position must never be less than 90 degrees. In practice it will be advisable to limit the terminal positions of these levers to a minimum included angle of 100 degrees.

As driving arm C moves forward forcing slide E ahead of it, levers J and K will straighten out. This action will force the driven slides apart equally. The maximum forward stroke imparted to these driven members will occur when the center of shoulder stud H lies on axis X-X along which the slides travel.

Further travel of slide E beyond axis X-X will cause the driven slides to retract in unison until they reach their innermost positions as shown by the light broken lines. A return stroke of the driving arm will produce a duplication of these movements.

Alterations in stroke length of the driven slides are obtained by adjusting the throw of the crankpin on the driving shaft (not shown). Variations in the working positions of the driven slides are obtained by adjusting the position of sliding block G. To facilitate setting of this block, the top surface of slide E may be graduated and the baseplate marked with a zero line.

Dwell periods may be procured at each terminal point in the movements of slides A and B if desired. By allowing block G to have a certain amount of independent sliding movement within slot in slide E, a corresponding motion will be subtracted from the stroke of the driven slides, thus imparting a dwell at the retracted positions of slides A and B. This type of setting is easily possible by merely adjusting clamp screws L and M. Each of these screws should be provided with a simple lock-nut arrangement. Provided slide E is located at a 90-degree angle to axis X-X, and levers J and K are the same size, then driven slides A and B will move in synchronism through identical distances in opposite directions.

Two Slides Operated Longitudinally and One Also Crosswise

For application on a wire fabricating machine, it was necessary to design a mechanism that would guide two strands of wire uniformly back and forth, and at the same time, reciprocate the one wire in a perpendicular plane. The wire to which the two movements are imparted must be stationary during a portion of the cycle. A view of the mechanism at the beginning of its motion, Fig. 17, is a front elevation, while Fig. 18 shows a plan view at an intermediate point of the cycle.

The mechanism includes a slide *A* which is mounted on a stationary part of the machine. The slide receives a uniform motion from a grooved face-cam (not shown) located at its left end. The slide carries a block *B* which is dovetailed to receive

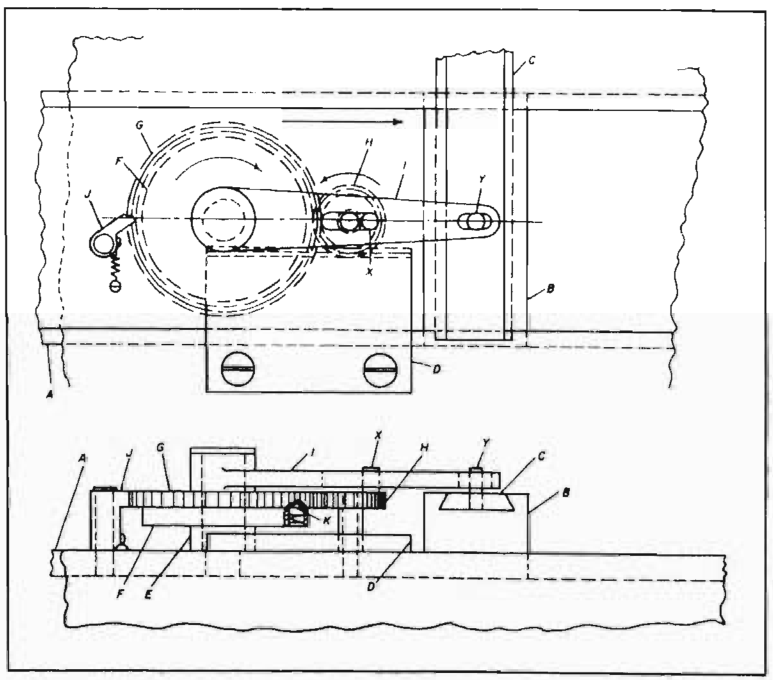


FIG. 17. Mechanism for wire fabricating machine which moves two slides lengthwise and one of them crosswise.

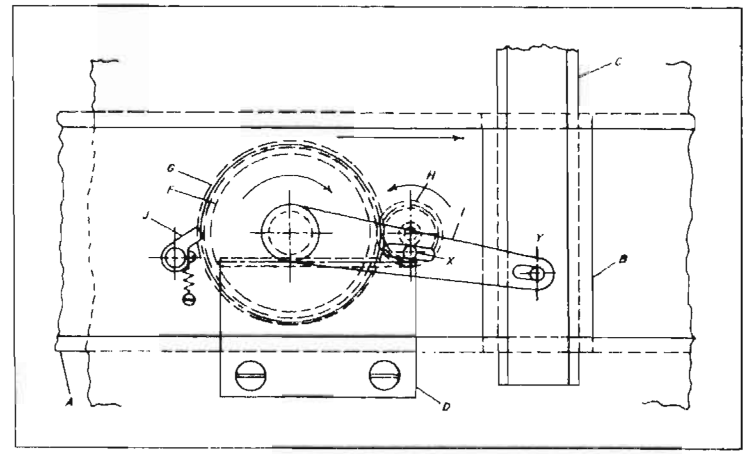


FIG. 18. Mechanism for wire fabricating machine when one-quarter of slide movement has been completed.

slide *C*. Rack plate *D*, also attached to a stationary part of the machine, meshes with pinion *E*. This pinion is attached to disc *F*. Both of them rotate freely on a stud mounted on slide *A*.

Disc *F* carries a series of spring plungers which engage pockets in the bottom side of gear *G* and transmits motion to it. This gear is also free on the stud. It meshes with gear *H*, which carries a pin *X* that engages a slot in lever *I*. The lever is also free on the stud. The end of the lever has a second slot at the right-hand end which engages a pin *Y* on slide *C*. A spring-loaded pawl *J* mounted on slide *A* engages the teeth of gear *G*, permitting rotation of the gear in one direction only.

When slide *A* begins its movement to the right from the position seen in Fig. 17, gear *E* will move with it and rotate in the direction indicated by the arrow, due to its engagement with rack *D*. Disc *F* and gear *E* will rotate as a unit. Gear *H*, meshing with gear *G*, is also caused to rotate. This causes pin *X* to impart an oscillating motion to lever *I*, which is transmitted to slide *C* through *Y*. Slides *A* and *C* carry the wire guides at their outer ends in parallel lines.

In Fig. 18, slide *A* has moved through a part of its motion and has caused the gears to reciprocate slide *C* through lever *I*. At this point, gear *H* has made a partial revolution, which has caused slide *C* to move from its central position, shown in Fig. 17, to its extreme position in one direction. It will now reverse to its extreme position in the opposite direction until the high point of the cam that actuates slide *A* is reached. Then the movement of slide *A* will be reversed.

The reverse movement of slide *A* reverses the rotation of gear *E* and the attached disc *F*, but the motion will not be transmitted to gear *G* because with the engagement of pawl *J* with gear *G*, spring plungers *K* are forced out of their pockets. From this point there is no oscillation of lever *I*. It remains immovable until the return stroke of slide *A*. The number of reciprocations of slide *C* relative to the movement of slide *A* is governed by the ratio of the gear train.

CHAPTER 10

Mechanisms Which Provide Oscillating Motion

Mechanisms which provide oscillating motion are described here. Similar mechanisms are also described in Chapter 10, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Sprocket Operated Geneva Drive Provides Wide Design Possibilities

The normal Geneva drive in which the roller is moved around a circle has limitations which make the mechanism useless for certain purposes. The most severe limitation is that the sum of the time consumed in indexing and the time of dwell corresponds to the time that it takes for the roller to move one revolution around the axis of the driving component of the device.

Instead of moving the roller around a circle, it is possible to move it along a different path and thereby obtain more freedom of design. Figure 1 shows one possible divergence from conventional design. Sprockets *A*, *B*, *C*, and *D* are driven by a chain *E* on which there is fastened roller *F*. Indexing disc *G* has two slots spaced 180 degrees apart.

The roller is shown in a position where it is just about to enter a slot of the indexing disc. With proportions as shown on the drawing, the disc *G* is accelerated during 30 degrees of movement, then moved with a constant velocity through 120 degrees, and decelerated during 30 degrees. The disc then remains in its dwelling position until roller *F* enters the next slot.

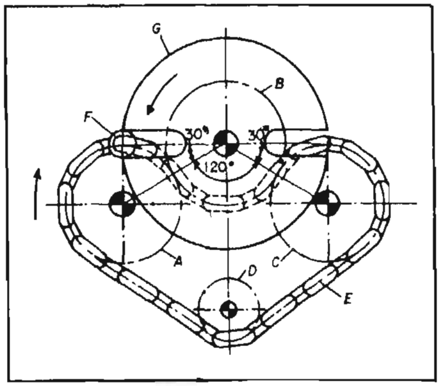


FIG. 1. Geneva drive in which actuating roller does not need to follow circular path.

Flexibility is a feature of this design because the number of slots, the number of rollers, and the length of the chain can be changed to suit various purposes.

Constant Pivot Linkages Replace Ball Bearings

Radar antennas are often constructed so that they rotate about a horizontal axis of no more than 180 degrees (from horizon to horizon). It is necessary to have an unobstructed area around the center of rotation so that the radar beams can pass freely. Therefore, many radar antennas are mounted on ball bearings. One such ball bearing is known to be 13 feet in diameter.

These large, expensive ball bearings can now be replaced by relatively low-cost constant pivot linkages, two forms of which are shown in the sketches. Figure 2 shows a parallelogram linkage. A and B are fixed pivot points and A_1 and B_1 move through circular arms having the same radius. Therefore C_1, too, will move in a circular arc. The same holds true for C_2. Thus, the circular element will rotate around C as center when either arm, A_1A_2 or B_1B_2 is moved.

The system shown in Fig. 3 has more links than the foregoing design but is more compact. In the drawing, B and C are fixed

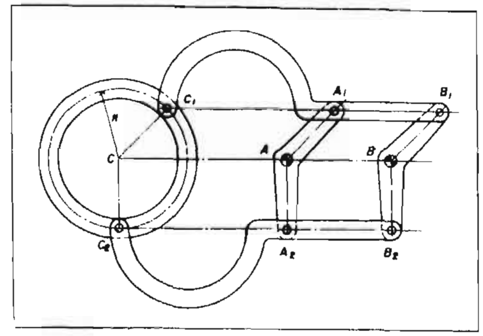


FIG. 2. Constant pivot linkage system rotates circular form C_1C_2 around C when either arm A_1A_2 or B_1B_2 is moved.

points. The triangle B_1BB_2 is equal and similar to triangle B_4AB_3. Similarly, triangle CC_1C_2 is equal and similar to triangle AC_4C_3; also B_1B_4 equals B_2B_3; and finally C_1C_4 equals C_2C_3. When the link CC_1C_2 is moved, the ring-formed piece will rotate around A through 180 degrees of arc.

Converting Rotary Motion from Continuous to Oscillating

A device that transforms a continuous rotary motion into an oscillating rotary motion is shown in Fig. 4. The arrangement is compact and the axis of the rotation of the output is perpendicular to that of the input.

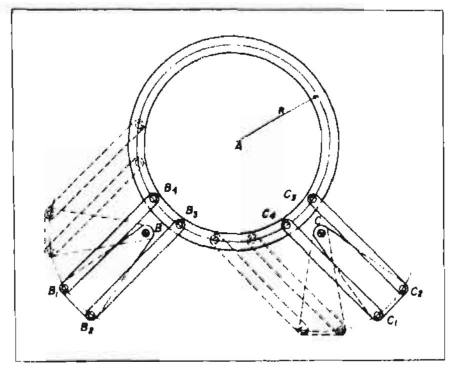


FIG. 3. This pivot system has more levers than that in Fig. 2 but is more compact.

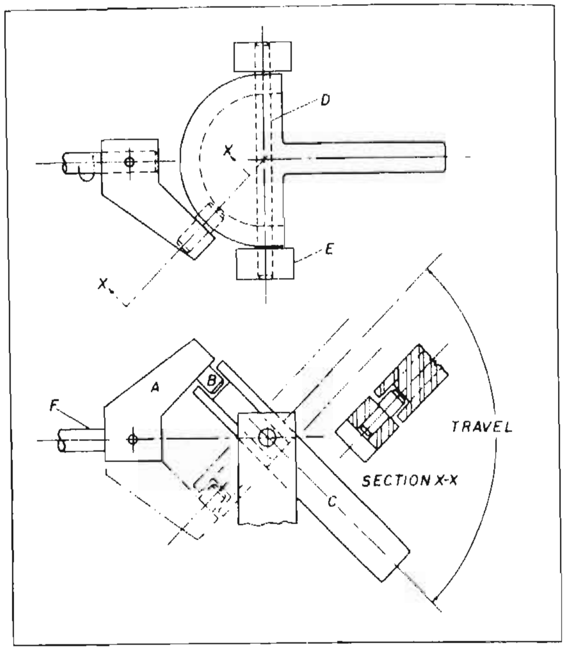


FIG. 4. Arrangement that produces an oscillating rotary output from a continuous rotary input. The mechanism is shown at a different position in each view.

Driving arm A has a drive-pin B which engages in a semicircular groove in a driven member C. This driven member is permitted to pivot freely on pin D which is retained at each end in a support block E. Pin D is press fitted in one block and slip fits in the other. It is important that the axis of pin D intersect the axis of drive shaft F and the axis of drive-pin B at a common point. Oscillating output of the mechanism is derived from the integral arm extending from member C. The total movement of this arm, in degrees, is equal to twice the angle at which the axis of the drive-pin intersects the axis of rotation of arm A.

Adjustable Oscillating Movement Derived from Rotating Shaft

Adjustment of the effective length of a crank may be made with the device in operation by means of the arrangement shown

in Fig. 5. Cranks A' and E', together with their respective shafts, are bored through to receive cylindrical rack F with a sliding fit. Pinion gear G, shown at Y in the illustration, meshes with the cylindrical rack. Gear G is pinned to threaded shaft H. Square nut J, which is threaded on shaft H, rides within a guide-slot in the crank body. This nut is secured to the end of coupler rod C'. The same arrangement is provided in the body of crank E'.

Cylindrical rack F is operated by the mechanism illustrated at Z. Gear wheel K, mounted on shaft L, meshes with the cylindrical rack. Also mounted on shaft L is a worm-gear M that is driven by a worm N. An operating handle and a calibrated dial, both located outside of the crank housing, operate in conjunction with the worm.

When the operating handle is rotated, gear wheel K is moved through the action of the worm and worm-wheel. This rotation of gear wheel K imparts a linear movement to cylindrical rack F.

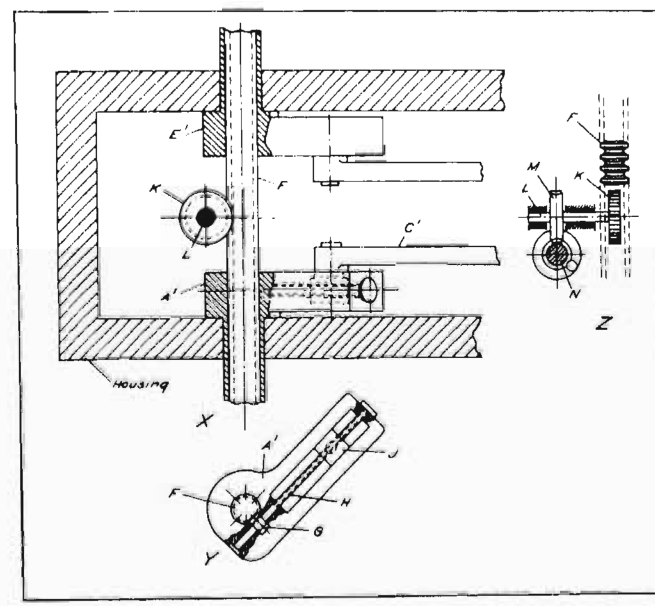


FIG. 5. Effective length of the crank may be altered during operation by means of a gear train.

Pinion gear G and shaft H are then rotated, which, in turn, alters the position of coupler rod C'.

Eccentric Gears Feature of Bag-Folding Device

One functional requirement of folder blades on a machine for making foil bags is that the first fold should not interfere with the second fold. This is necessary to prevent wrinkling and turning of the edge of the first fold. A pair of eccentric gears was arranged as shown in Fig. 6 to accomplish the proper folding action.

Eccentric gear A is secured to the shaft integral with part B which carries folder blade C. Similarly, eccentric gear D and folder blade E are attached to part F. The pitch radii on the gears are indicated by R_A and R_D in the illustration. Bearings in housings integral with the machine frame G supports parts B and F. Eight bag-forms H are mounted on a turret (not shown) having a Geneva motion which intermittently indexes one after the other into position between the two blades for the folding operation. A drive connected to part B rotates blades C and E through an angle of 180 degrees to fold the foil on each form in its turn.

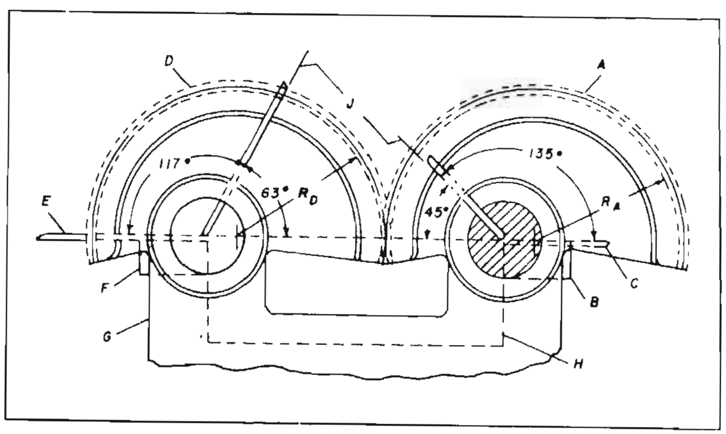


FIG. 6. Eccentric gears arranged to facilitate folding of foil on a form in a bagmaking machine. Interference between the folds is prevented.

In operation, as folder blade C rotates 135 degrees, blade E travels through an angle of 117 degrees. At this instant blade C lags blade E by 18 degrees, and this lag permits the portion of the foil J carried by blade C to clear the portion carried by blade E. During the remaining 45-degree rotation of blade C, blade E gains back the 18 degrees, and both blades complete the 180-degree working cycle simultaneously.

If standard gears were to be used, one gear would have to be retarded slightly to prevent interference between the folds. In this case, both blades would not hold the foil flat against the form at the end of the folding operation.

Simple Linkage Replaces Three Gears

On automatic machines it is sometimes desired to rotate two shafts at a uniform velocity ratio over a limited range of time. In a mechanical calculator being designed this velocity ratio was 2 to 1 and it was required that both shafts should rotate in the same direction. Rotation of the input shaft was through about 180 degrees. Normally this motion would have required three gears because an idler gear would have been necessary in order to operate the two shafts in the same direction. Also, the distance between the two shafts would have been rather large.

A mechanism designed to fulfill all requirements is shown in Fig. 7. As input shaft A turns on its axis, it swings crank C, on the end of which is a roller D. This roller slides in a slot of link E. This link is fastened to output shaft B. Because the radius of crank C equals the center distance between shafts A and B, it can be seen from geometrical considerations that when the shaft A moves through angle Y the output shaft B will move through angle $2Y$.

The total output angle of shaft B is about 120 degrees.

Drilling Parallel Rows of Holes

When drilling a row of holes with a radial drill, a multiple-spindle head is of great advantage, since hole alignment is automatic and a number of holes in the row can be drilled simultaneously.

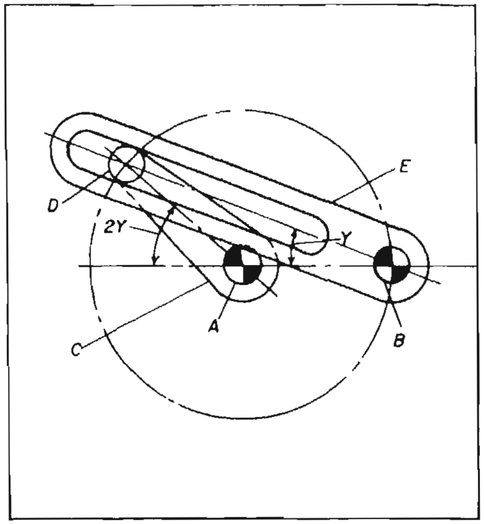


FIG. 7. Rotation of shaft A through $2y$ degrees causes B to rotate y degrees.

If several rows of holes are to be drilled, as in the case of heat exchanger tube sheets, maintaining parallelism between the rows can be something of a problem. A fixed angular relationship between the multiple-spindle head and the workpiece, regardless of the angular position of the radial drill arm, is needed.

The accessory, shown in Fig. 8, solves the problem. It consists of two interconnected double-acting hydraulic cylinders. The piston rod of one of the cylinders is fixed to the outer rotating column of the machine. As the cylinder and piston rod pivot with the external column, a rack integral with the cylinder is driven by a mating gear segment mounted on the stationary inner machine column. This cylinder is thus translated to the left or right, with respect to the piston, as the radial drill arm pivots with the outer column.

Movement of the cylinder varies the volume of hydraulic fluid in the two opposed chambers. The fluid in the cylinder chambers is forced to enter and leave through two ports in the piston rod (one at each side of the piston) passing through the rod and hose connections at either end. If the radial arm is pivoted coun-

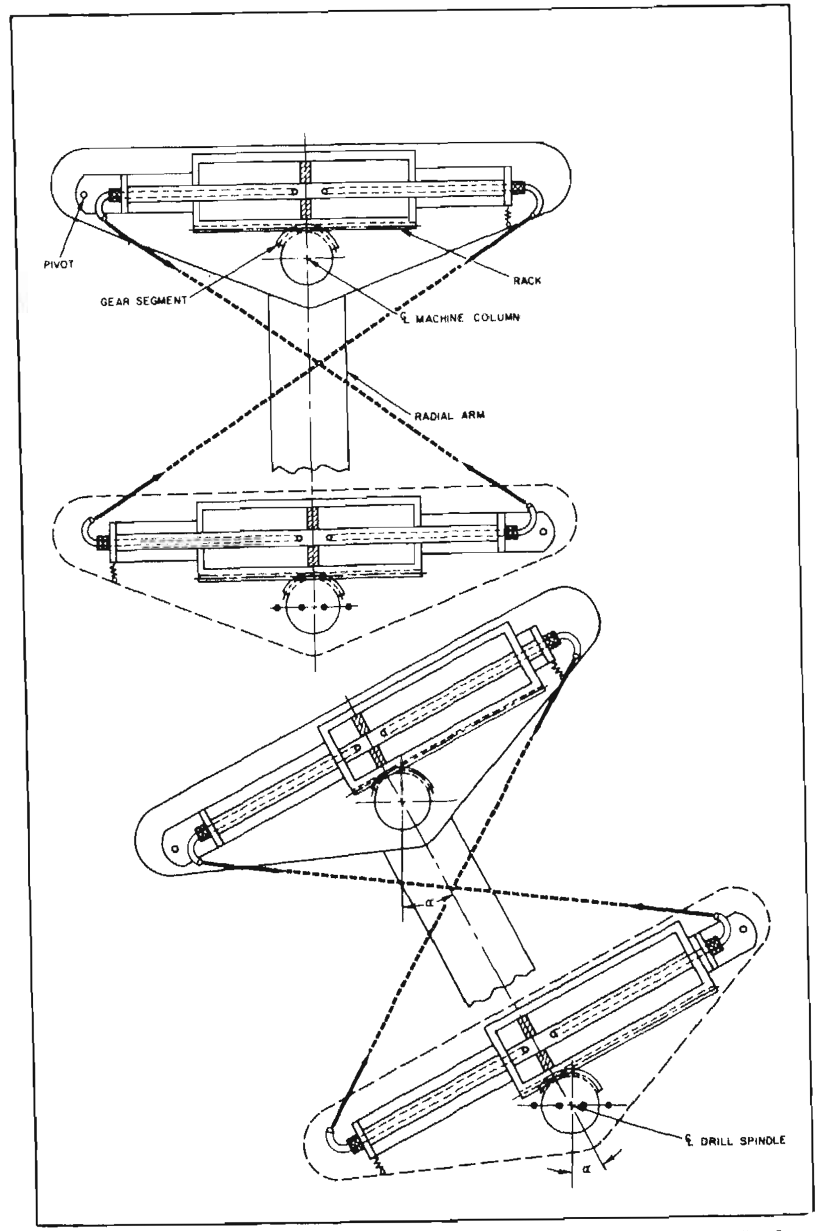


FIG. 8. Spindle located on arm is prevented from rotating by two hydraulic cylinders.

terclockwise, the cylinder moves to the right, decreasing the volume of fluid in the left-hand chamber.

A similar cylinder-and-piston arrangement is fixed to the nonrotating sleeve at the bottom of the radial drill spindle. The rack on the cylinder, in this case, operates a gear segment on which the multiple-spindle head is mounted. By interconnecting the two cylinders so the hose from the left-hand chamber of the cylinder on the machine is connected to the right-hand chamber of the cylinder on the radial drill head, and vice versa, counterclockwise movement of the radial drill arm results in an equal, but clockwise, movement of the multiple-spindle head.

Thus angular rotation of the head due to radial arm pivoting is automatically compensated. Once set, the hole patterns produced by the head will remain parallel regardless of the angular position of the radial arm, within the 90-deg. operating limits of the device.

Gears in Transmission Line Increase Shaft Oscillation

A shaft in a machine for fabricating a wire product oscillated continuously on its axis: rotating 90 degrees in one direction, then 90 degrees in the opposite direction. The movement of the shaft was transmitted from an eccentric through a connecting-rod and link. Subsequently, because of a design change in the product, it became necessary to double the oscillation of the shaft. Space limitations prevented any increase in the throw of the eccentric, and to extend the swing of the link to 180 degrees would have created a dead-center condition that would have rendered the mechanism inoperable. The drawing shows how the problem was solved by introducing gears into the line of transmission.

Originally, the connecting-rod *A* (see Fig. 9), was pinned to the single link *B*, the lower end of which was keyed to the shaft *C* supported by bearings *D*. (The link is shown in solid outline at the midpoint of its movement, and in broken lines at the two extremities of its movement.) In the altered mechanism, two links *B* and *B′* swing freely on the shaft, and straddle a

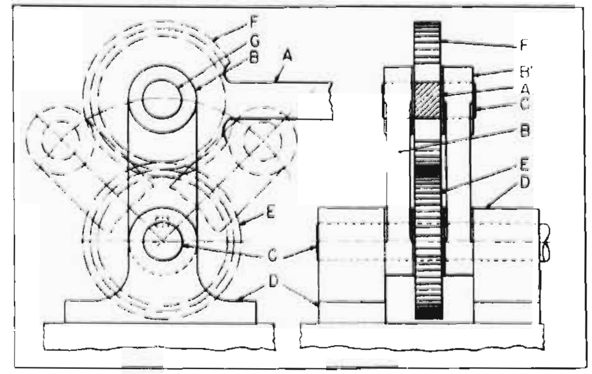


FIG. 9. Driving shaft *C* through two gears doubles the degree of oscillation.

gear *E* keyed to the shaft. Welded to the end of the connecting-rod is a second gear *F*, meshing with gear *E* and free on stud *G*.

In operation, the connecting-rod transmits movement to the links, running 90 degrees, as in the original mechanism. However, the action of gear *F* in rotating around gear *E* causes shaft *C* to oscillate the increased amount that is desired. The reason for the increase is that gear *F* rotates with respect to gear *E* creating an additional movement of gear *E*. Both gears are identical in size. Actually, since gear *F* does not rotate completely around gear *E*, gear segments instead of complete gears would serve the purpose.

Increasing the Movement of an Oscillating Shaft

In fabricating a wire product, it became necessary to increase the angular movement of an oscillating shaft of a machine tool. Because of space limitations, it was impossible to increase the throw of the eccentric controlling the shaft, and some other means of obtaining the additional movement had to be devised, as shown in the accompanying illustration.

In the original design (see Fig. 10), the eccentric-operated rod *A* connected to an arm *B* was keyed to the shaft *C* supported in a bearing *D*. In the present design, the arm is free to rotate on

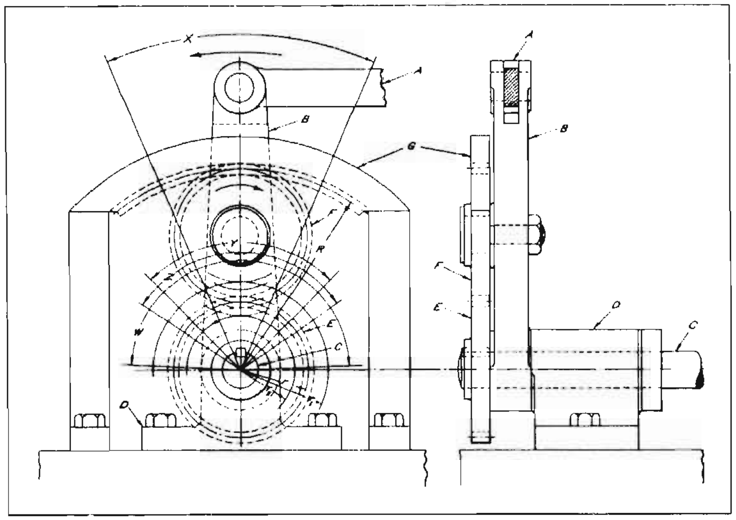


FIG. 10. By introducing an idler gear F and a gear segment G the oscillation of arm B and shaft C can be increased without changing the throw of an eccentric-operated rod A.

the shaft and a gear E is keyed to the shaft. Another gear F rotates freely on a stud carried on the arm and meshes with gear E. Gear F also meshes with a segment of an internal gear G fixed to the bed of the machine. (In the right-hand view the supports for the segment have been omitted.)

The arm is shown at its central position moving in the direction indicated by the arrow. Gear F moves with the arm, and since this gear meshes with the segment, it is caused to rotate on its stud. The rotation of gear F is transmitted to gear E and thus to the shaft. In the illustrated application, gears F and E are of the same pitch diameter, gear F being an idler which has the effect of imparting movement to the shaft in the same direction as that of the arm.

Angle X indicates the magnitude of the oscillation of the arm. Actually, the gear F serves as a lever, with its fulcrum at the pitch line of the segment. In this manner the action of gear F causes an increase in the angular movement of gear E as compared with the movement of the arm.

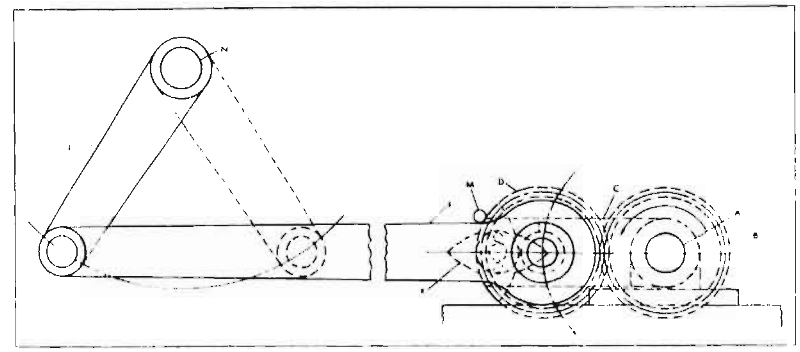


FIG. 11. Front view of intermittent speed-change linkage mechanism in which N is the reciprocating driven shaft and A is the drive. Gear B turns constantly, but at points in a cycle gear is locked by a clutch, causing the whole mechanism to rotate around shaft A.

Sun and Planet Gears Produce Intermittent Speed Change

Figures 11 and 12 illustrate the construction of a mechanism which provides oscillating angular motion to a shaft, with the alternate cycles at different speed relations to the rotation of the driving shaft. This application is used on a machine producing wire textile screening, the object being to produce two different spacings of the wires. Figure 11 is a front elevation and Fig. 12 is a plan view of the intermittent speed-change mechanism.

The driving shaft A, rotating in the direction of the arrow, is keyed to gear B. Gear B meshes with gear D, carried on a pin

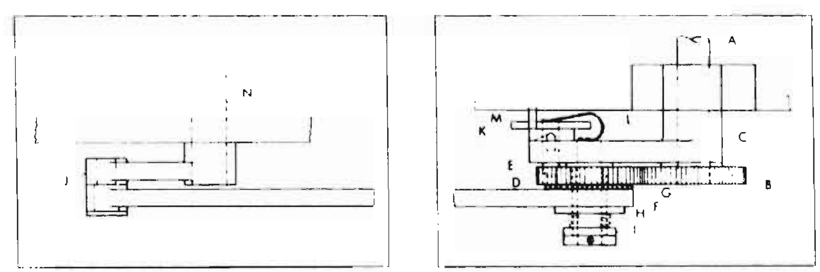


FIG. 12. Details of the clutching device appear in this plan view, with the clutch pin retained by the free end of spring L. The friction material is shown at G.

pressed into the arm of lever C, which is carried free on shaft A. Gear D is pressed onto a flanged sleeve E, which is free to turn on the same pin. The connecting-rod F, which transmits motion to lever J (keyed to shaft N), is rotatably supported by sleeve E. A disc of friction material G is carried between gear D and rod F. A spring H, adjusted by threaded collar I, maintains pressure of gear D and rod F on the friction disc G. A collar on the outside end of the pin retains this assembly.

Another pin with a tapered end carries star-wheel K, and passes freely through lever C. The tapered end of the pin engages a matching hole in gear D on alternate cycles. The hub of star-wheel K is provided with two opposite detent grooves, shaped as shown, which engage two pins in lever C, as controlled by the position of star-wheel K. A flat spring L insures engagement of the pin with the hole in gear D. Another pin M is located in a stationary part of the machine, and imparts rotative motion to star-wheel K with each rotation of lever C.

In the position shown, the mechanism is at the beginning of its cycle, with the lever J at its extreme left-hand position. At this point, the tapered pin carrying star-wheel K is engaged in the hole in gear D, so that gear D and lever C are locked together. Because gear D cannot rotate on its pin, lever C becomes a simple crank revolving on the center of shaft A, thereby transmitting oscillating motion to shaft N through the linkage of rod F and lever J. At this stage of the cycle, rod F is slipping on the friction disc G. Thus the entire assembly delivers a conventional crank motion, imparting an oscillating motion to shaft N with each rotation of shaft A.

The completion of this phase of the cycle occurs when the movement of lever C past tapered pin M causes star-wheel K to make a quarter-turn. This action causes tapered pin M to withdraw from the hole in gear D by the action of the angular surface of the detent groove. Pin M is so placed that complete engagement and disengagement of the tapered pin from gear D takes place when the center lines of lever C and rod F are in alignment as in Fig. 11, so that the speed change will take place at the end of the cycle. Although the tapered pin has now been disengaged from gear D, the latter is not free to rotate because it is frictionally locked to rod F. In this phase gear D now acts as a planet gear revolving about gear B as a sun gear. With this type of reducing gear application, the relative revolution of the planet gear to the rotation of the sun gear will be equal to the ratio of the gear tooth count plus 1. As gears B and D are of the same tooth count, gear B must perform $1/1 + 1$ revolutions, or 2 revolutions, to produce one revolution of the crank whenever the tapered pin is disengaged from gear D.

Producing an Oscillating Movement of Uniform Angular Velocity

The mechanism shown in Fig. 13 was designed to produce an oscillating motion having uniform angular velocity, using a crank A rotating at uniform speed as a driver. With the arrangement shown, an oscillating movement is imparted to lever C by connecting-rod B, but the angular velocity of this movement is not uniform. Although approximately uniform angular velocity could be obtained by arranging a suitable mechanism in front of

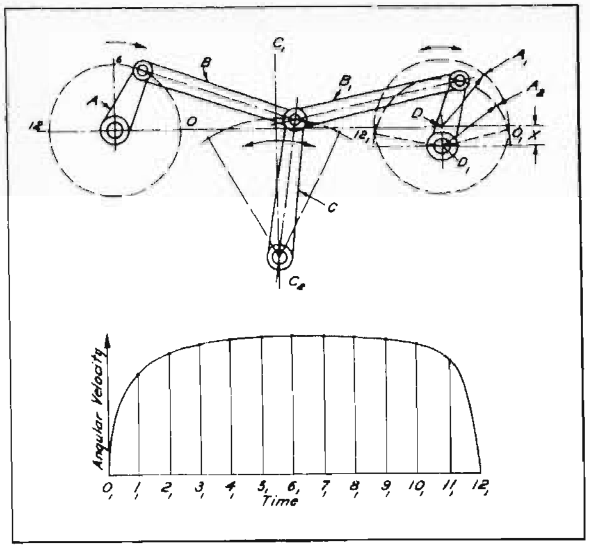


FIG. 13. Diagram showing method of obtaining an oscillating movement of approximately uniform angular velocity from a rotating crank.

the rotary crank, such devices are usually complicated and were not considered applicable in this case.

To achieve the desired results, the center position C_1 C_2 of lever C was taken as the axis of symmetry, and the mechanism on the left-hand side (crank A and connecting-rod B) was exactly duplicated on the right-hand side (crank A_1 and connecting-rod B_1). If crank A_1 were located on the same horizontal center line as crank A the angular velocity of crank A_1 would be uniform, but its oscillating movement would be through an angle of 180 degrees, which is not practical. However, by lowering the center of crank A a distance X (from position D to D_1) keeping radius D_1A_2 equal to DA_1 and leaving the mechanism on the left of the axis of symmetry in its original position, oscillation angles of 120 to 150 degrees, which are permissible, are obtained. Graphical analysis of the angular velocity of crank A_1 will produce a curve like that shown at the bottom of the illustration.

Remote Control Presets Spindle-Quill Travel

An arrangement that can be used to preset the length of travel of a spindle quill, tool-head, or work-table is shown in Fig. 14. Repeated and accurate positioning of one of these machine tool members from a remote point is achieved by the use of synchros.

Linear displacement of a spindle quill A is converted to rotation by means of a rack B and a gear train C. The output shaft of gear train C drives a transmitting synchro D. Components A, B, C, and D are built into the machine tool.

Synchro D is connected electrically to a receiving synchro E located at the remote position from which the machine tool is to be controlled. In this manner, angular displacements which represent the position of the spindle quill are transferred from synchro D to synchro E. The receiving synchro is connected by shaft F to a special dial type indicator G. A pointer H is attached to shaft F, and a second pointer I, which pivots on the same axis and is concentric with dial plate J, is driven through a

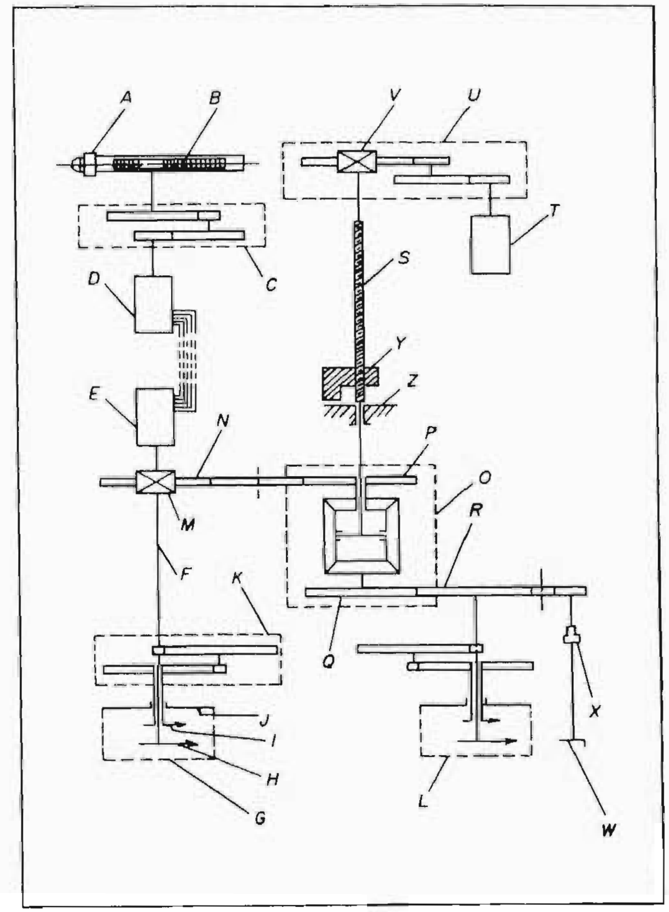


FIG. 14. Arrangement that permits remote presetting of the travel of a spindle quill, tool-head, or work-table.

reduction gear train K. Also driven by shaft F, gear train K reduces the rotation in the ratio of 100 to 1. Since rack B and gear train C are in a ratio such that one turn of shaft F and pointer H represents a 0.1-inch displacement of the spindle quill, one revolution of pointer I will, therefore, represent 10 inches of quill travel.

An electromagnetic clutch M, when actuated, allows shaft F to engage a gear N. This gear, in turn, drives a differential gear train O through an idler and gear P. The differential is connected through gears Q and R to a gear train and a dial indicator L. These are identical in design to members K and G, respectively.

A lead-screw S, which is used for setting quill travel, is also connected to the differential. Thus, if clutch M is engaged and gear Q is held in a fixed position, lead-screw S will drive shaft F through the differential in a ratio of 1 to 1.

An electric motor T, used for presetting quill travel, is connected to lead-screw S by a reduction gear train U and an electromagnetic clutch V. To preset quill travel, gear P is locked in place, gear R is released, and clutch V is actuated. Rotation of lead-screw S by means of the motor will then be transmitted through the differential and gear R to indicator L.

Manual presetting of quill travel is made possible by means of a handle W and a mechanical clutch X, which is geared to gear R. Manual rotation of handle W is in this way transmitted both to indicator L and lead-screw S. Clutch V should be disconnected and gear P locked in place when lead-screw S is preset manually. Actual control of quill travel is accomplished by means of a nut Y on lead-screw S and a plate Z, the movement of which opens a switch in the circuit supplying power to the quill-feed mechanism.

Operation of the remote quill-travel control is as follows: With gear P locked and a nut Y in contact with plate Z, lead-screw S is rotated by actuating motor T or manually by rotating handle W until the desired length of travel is read on the dial of indicator L. This moves nut Y on screw S away from plate Z a distance proportional to the required quill travel. Movement of the quill for the desired travel with gear R locked and gear P released will then cause nut Y to return the same proportional distance in the opposite direction and contact plate Z, thus stopping the forward motion of the quill. During machining, indicator G shows the position of the quill.

Torque Filter Eliminates Backlash

Design for aerospace programs has resulted in some useful solutions to common problems. The mechanism shown in Fig. 15

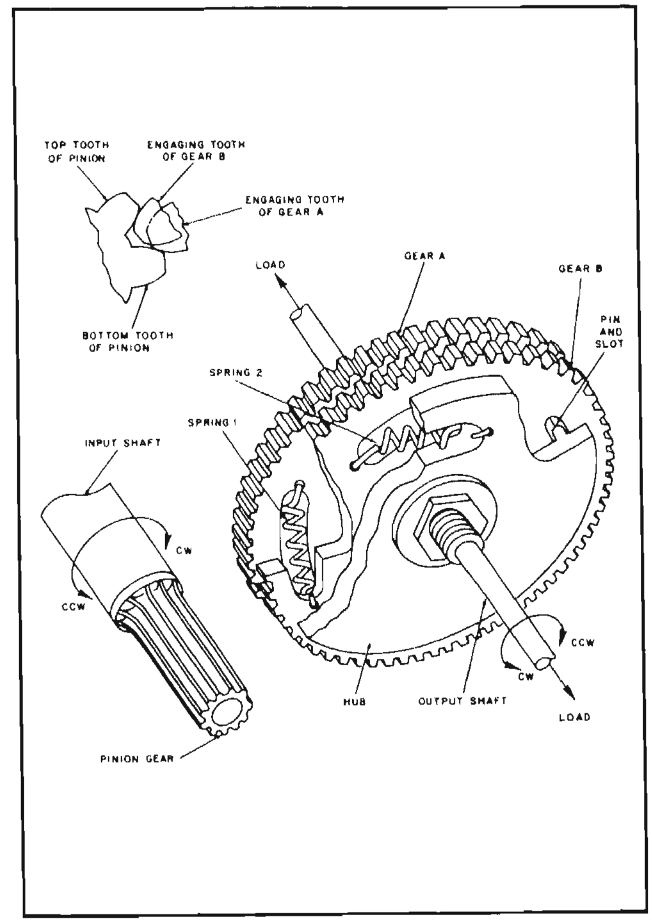


FIG. 15. Smooth output obtained from gears A and B by use of springs 1 and 2.

enables a constant output torque, free of backlash components, to be maintained from a pulsating input torque.

Two elastic components (springs) connecting a hub and two spur gears absorb torque differentials and provide the desired antibacklash characteristic between input and output shafts. The system performs equally well in either direction of rotation.

The hub is securely attached to the output shaft and two spur gears turn freely on the shaft. Spring 1 connects gear A

to gear B. Spring 2 connects gear B to the hub. The input shaft pinion engages both spur gears.

Spring 1 forces the engaging tooth of gear A against a lower tooth of the pinion and the engaging tooth of gear B against a higher tooth of the pinion. This arrangement prevents backlash between the input and output shafts. Spring 1 also serves as a torque filter between the two spur gears.

When the input shaft is rotated counterclockwise, its pulsating torque is transferred directly to gear A, which then drives gear B through spring 1. The torque pulsations are filtered by the spring since the pinion does not directly drive gear B in this direction of rotation and gear B is free to rotate slightly with respect to the pinion. The hub and output shaft are then driven clockwise with a smooth, constant torque by a pin in the hub that engages a slot in gear B.

When the input shaft is rotated clockwise, it drives gear B directly. In this condition, gear B drives the hub and output shaft clockwise through spring 2, which filters out the pulsations in the input torque. This mechanism could be useful in precise control systems. Possible configurations and filter materials are limited only by the application.

CHAPTER 11

Mechanisms Providing Combined Rotary and Linear Motions

Mechanisms which provide combined rotary and linear motions are described in this chapter. Similar mechanisms are described in Chapter 11, Volume III of "Ingenious Mechanisms for Designers and Inventors."

High-Speed Spiral Scanner

In radar, television, and other systems involving the reception of electromagnetic waves, the need frequently arises for a high-speed spiral scanner. In this case, the scanner consists of a mirror held in an adapter actuated by a power-operated mechanism. The mirror is tilted at a constantly changing angle with respect to an axis passing vertically through its center. At the same time, the direction of the tilt continuously changes through a complete circle of 360 degrees. Thus, the motion of the mirror is such that it can "see" any object within the angle of its cone of motion.

The mechanism built to provide the required scanning motion, which has been performing satisfactorily, is shown in Fig. 1. The requirements for this particular instrument were these: operational speed had to top 10,000 revolutions per minute; the mirror or antenna adapter K (see diagram Fig. 2) had to scan a cone opening angle O of 12 degrees, the axis y-y of axle J pivoting about point H while point S at its lower end described a spiral of twenty turns between 0 and 12 degrees — that is, the scanner had to cover its field ten times per second when drive-shaft B, Fig. 1, was operating at 12,000 revolutions per minute; and the center of the conic motion had to be at point H on the upper surface of the adapter K. Finally, it was necessary that the

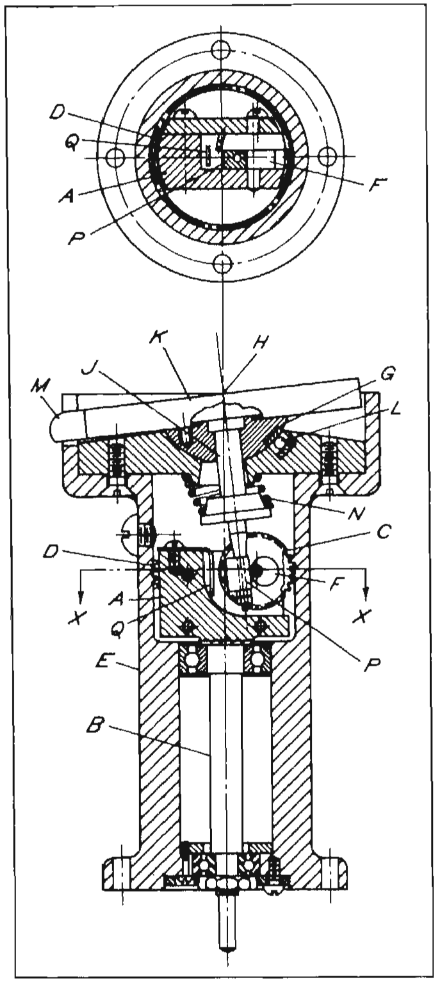


FIG. 1. High-speed spiral scanning mechanism in which the conic motion of the scanner K is set up by gear C turning on its own axis while its carrier A rotates at high speed around the axis of main shaft B.

instrument be simple and easy to construct. Thus, the axle B of the main shaft was designed to (1) revolve at a speed of 12,000 rpm; and (2) move the lower end of axle J outward from its vertical, or zero, position toward the widest cone angle scanning position, and return to the zero position once every forty

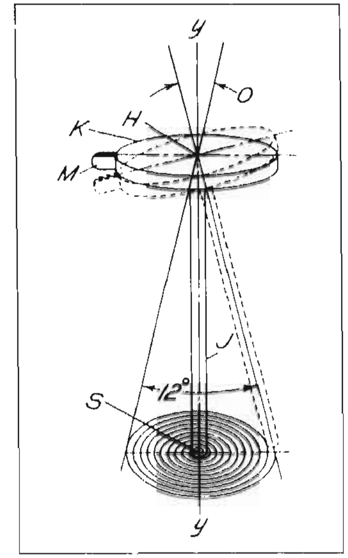


FIG. 2. Diagram illustrating the basic operating principle of the spiral scanning mechanism, Fig. 1.

revolutions of Axle B. In Fig. 2, the axle is shown in vertical position (solid lines) and in widest scanning position (broken lines). The relative positions of the moving parts during this operating cycle are shown by the diagrams in Fig. 3.

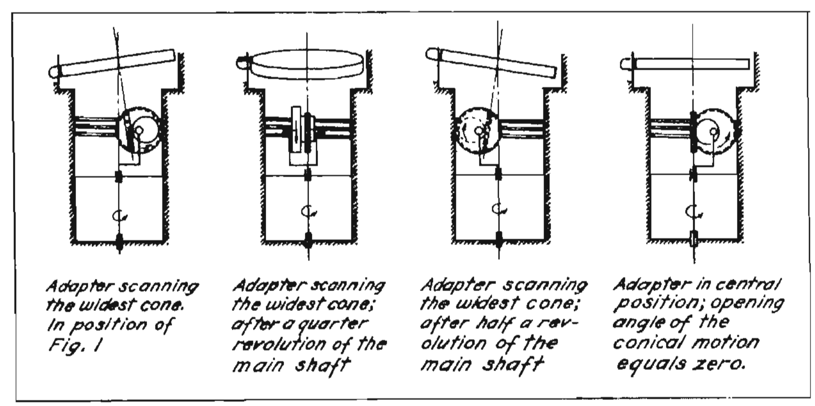


FIG. 3. Diagrams showing relative positions of the moving parts of spiral scanner at different periods of the operation cycle.

Carrier A, Fig. 1, is integral with main shaft B and is balanced. Helical gear C, having forty teeth rotates on a pivot integral with the carrier, and meshes with the internal worm thread D in housing E.

Cam F is fixed to the gear C and revolves with it. Spherical section G, having its center at H, and the axle J are attached to adapter K. Since the spherical section rests on three equally spaced balls L, it can easily be given the continuous tilting or conic motion required. The adapter, spherical section, and axle are prevented from revolving with carrier A by projection M which engages a slot in the housing E. Spring N holds the spherical section in its seat. Slider P, carried on the axle J, is free to rotate around it. The slider is positioned between the gear and the carrier.

In operation, the gear C and cam F move as a unit with the carrier, rotating at high speed with the main shaft B so that the axle J, spherical section, and adapter perform the required conic motion. At the same time, the worm thread D causes the gear C to revolve around its own axis while centrifugal force presses the slider P against the cam F. Since the gear and cam are also revolving around the gear axis, the opening angle of the cone varies continuously as dictated by the shape of the cam, thus producing the required spiral scanning motion.

When the slider reaches the center, flat spring Q thrusts it against the cam, replacing the centrifugal force, and a new scanning cycle begins. The housing is partly filled with lubricating oil, and the high speed of the mechanism serves to create effective mist lubrication.

Simple Device Reciprocates Rotating Printing Roll

An arrangement was required to spread ink evenly between two revolving printing rolls. Although there are many conventional solutions of this problem, the assembly illustrated in Fig. 4 accomplishes the desired results with a minimum number of moving parts which can be easily disassembled and cleaned as needed.

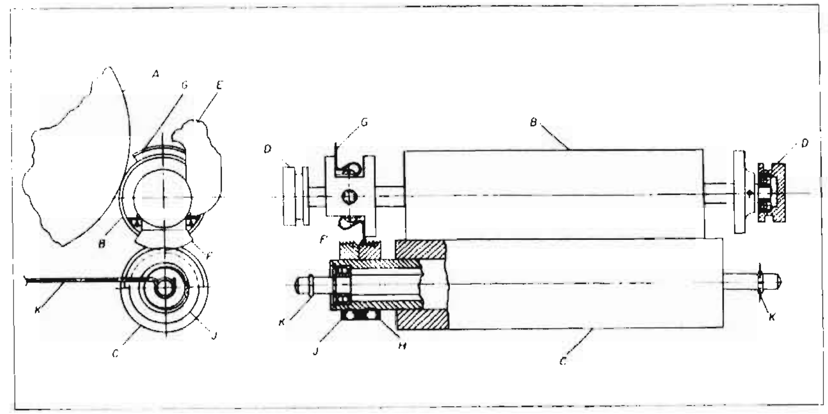


FIG. 4. To spread the ink evenly, roll C is reciprocated as it rotates in contact with roll B. Each stroke is accomplished in short intermittent steps.

Equal distribution of the ink is best obtained by allowing one of the rollers to reciprocate continually in an axial direction during operation. In addition, one roll should be made slightly larger than the other so that the same areas of contact are not repeated on each revolution of the rolls.

The reciprocating mechanism is shown in detail in Fig. 4. In the setup, a drive wheel A rotates printing roll B which, in turn, drives the second printing roll C. Roll A rotates in antifriction bearings which are held in housings D supported freely in slots in machine frame E. Members F and G are leaf-spring segments mounted in diametrically opposed positions on a collar on the shaft of roll B. Two collars H and J are each threaded on one-half of their circumference and relieved on the other half to a diameter less than the minor diameter of the thread. Collars H and J are threaded right-handed and left-handed, respectively, and are mounted side by side on the shaft support roll C. The width of each collar is equal to the length of the total axial stroke to be given to roll C.

As roll B revolves, segment F enters the thread on collar H, moving roll C to the right during the period they remain in mesh. Then, as segment J is rotated into the meshing position, the unthreaded half of the collar is presented to the segment and the roll remains stationary for a period. Later, segment F re-engages collar H and continues moving roll C to the right. This sequence would continue if the rolls were equal in diameter. However, since roll C is slightly larger than roll B, segment G reaches a position where it comes into contact with the threaded part of collar J and moves roll C to the left, as can be seen from the illustration.

The leaf-spring segments are resilient, so that if they hit the crest of the thread, they are pushed aside. Then, because they are straight and the thread is helical, the segments will slip into proper mesh with the thread after roll B has rotated a few degrees. Due to the alternating blank and threaded surfaces on collars H and J, roll C is displaced axially in short intermittent strokes. This type of motion is helpful in spreading the ink.

In the particular arrangement shown, each stroke lasts fourteen revolutions of roll B. The ratio of the roll diameters is 15 to 16, and the length of the axial stroke roll C is ⅜ inch or two and one-half times as large as the largest letter to be printed. Each of the collars has an eight-entry thread, the lead of each thread entry measuring ½ inch and the pitch, therefore, being $1/16$ inch.

The segments displace roll C axially about $1/10$ ipr (which is a little more than the pitch) in order to assure the entry of the segment into the next thread. Consequently, each segment occupies 70 degrees of arc. The effective axial displacement, therefore, measures $1/16$ inch, and six turns of roll B are required to move roll C ⅜ inch. During each seventh and eighth turn of roll B, roll C is idle. In operation, the rotational speed of roll B is 230 rpm.

Members K are two identical hook-shaped springs that hold roll C against roll B and thus cause roll B to press firmly against driving wheel A. With this arrangement, the whole assembly can be taken apart simply by removing hooks K. It is unnecessary to adjust the mechanism during assembly, since the arrangement will automatically start the correct cycle after it has been in operation a few revolutions.

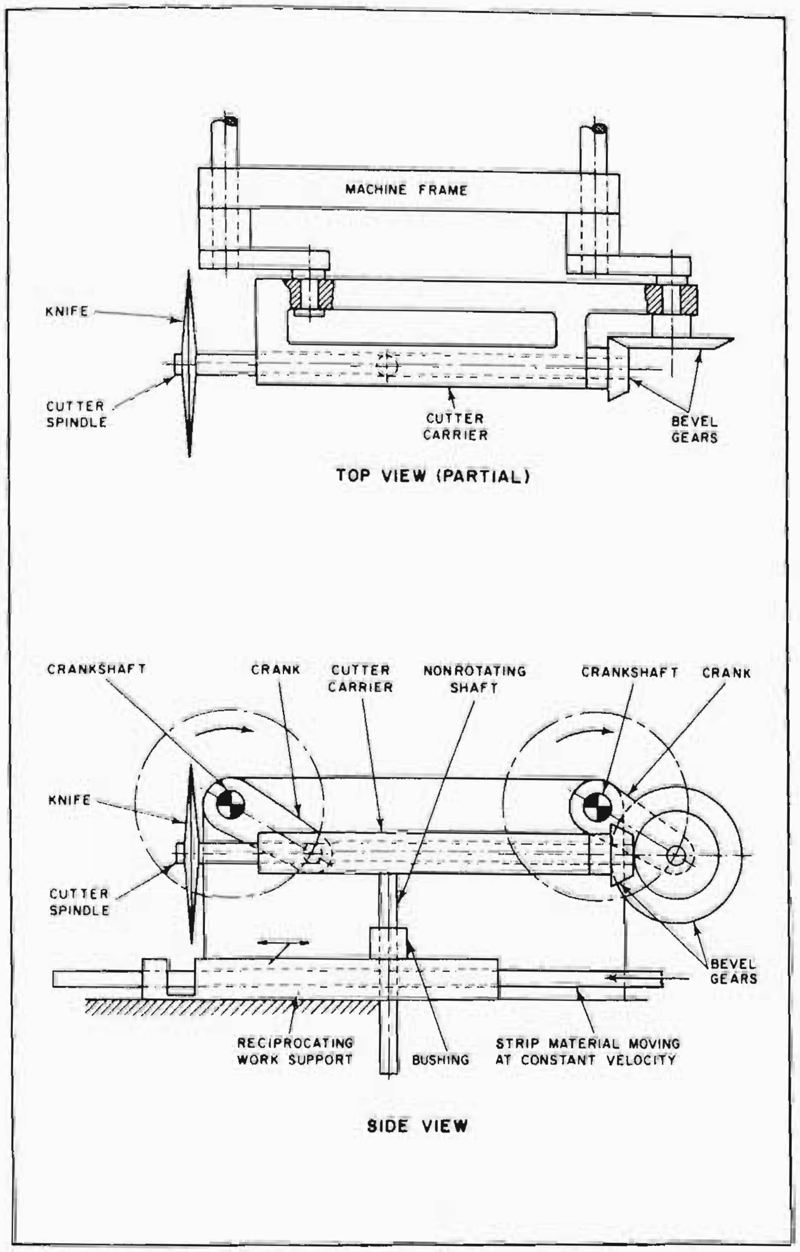


FIG. 5. Stock cut on the fly by knife rotating in a circle.

High-Speed Cutoff Device

Continuously moving strip material can be cut without stopping the strip, thus speeding production. Conventionally, this type of cutting is done while moving the tool at the same velocity as the material and then withdrawing the cutter and returning it to the starting position and repeating the operation.

The principle is illustrated by the device shown in Fig. 5. The material to be cut moves at constant velocity. A circular knife on a shaft is supported by a cutter carrier which swings in a circular path on a pair of equal-length cranks. The cranks are driven clockwise at the same angular velocity by one or both of the connected crankshafts.

When the right-hand crank is rotated, a bevel gear fixed to the crank drives a second gear fixed to the cutter spindle and revolves the knife. The cutter carrier has a nonrotating shaft which imparts a reciprocating motion through a sleeve bushing to a slide supporting the material while it is being cut.

Since, in this arrangement, the cutter and work move in the same direction at the same speed, it is not possible to change the length of the cut-off piece without reproportioning the parts of the device.

Intermittent Rotary and Linear Movement

On a machine designed for producing ornamental woven-wire screening, it was required that two strands of wire be given a twist of two turns while in linear motion, then additional linear moton without a twist, next a twist and linear motion in the opposite direction, and finally a linear motion again without twist, completing the cycle. Figure 6 illustrates a mechanism devised to accomplish these motions.

The bed section of the machine was dovetailed to carry a slide A. This slide is connected at one end with piston-rod B of a hydraulic cylinder that provides the operating power for the mechanism. Slide A carries bearing bracket C, which supports

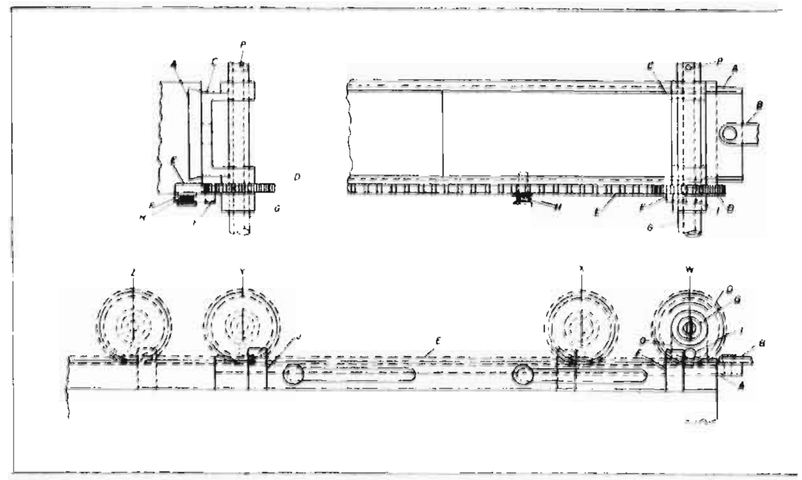


FIG. 6. Mechanism devised for actuating intermittent rotary and linear movements.

a tubular shaft G. The far end of shaft G carries a pin P through its radial center to separate the two strands of wire and apply the twisting force, one strand of wire passing through the tube on either side of the pin.

The outer end of shaft G carries two spools of wire which revolve on the shaft axis. Shaft G also carries gear D, keyed to it. The gear is provided with a pin I. Gear D meshes with rack E, which is mounted on the bed section of the machine by two studs that pass through slots in the rack. Each stud is provided with a spring H that produces frictional resistance to the movement of the rack. Two stops F and J are attached to the ends of the rack.

At the beginning of a cycle, gear D is at the right-hand end of the assembly in position W. As piston-rod B moves to the left and pushes the slide assembly with it, rack E does not move because of the frictional resistance applied by springs H. Gear D is, therefore, caused to rotate counterclockwise, beginning the twisting action on the wires. This action continues until gear D reaches position Y, at which point pin I is in contact with the far side of stop J. As the movement of slide A continues, gear D can no longer rotate. Consequently, rack E is carried along with it until the gear reaches position Z, which is the end of the left-hand movement, and one-half of the cycle has been completed. This half of the cycle, therefore, consists of a linear movement with a counterclockwise twist, followed by a period of linear movement without a twist.

On the return movement of rod B to the right, rack E remains in position, and gear D revolves until it reaches position X, at which point pin I contacts stop F and can no longer rotate. In moving from position X to the starting point W, rack E is returned to its original position, completing the cycle. In the second half of the cycle the linear motion is in the opposite direction to the movement in the first half and the twist is in the clockwise direction.

Rotating and Sliding Mechanism Used in Polishing Rectangular Frames

The mechanism shown in Fig. 7 was designed to provide the required motion for polishing a rectangular metal frame on all four sides, as well as on the faces adjoining those sides. The diagram in the upper right-hand corner indicates the movement of the frame relative to the wheel W during the polishing operation. The frame is indicated by dot-and-dash lines.

In operation, the frame moves along line L_1L_2 in the direction indicated by arrow A until point C_2 reaches point C_1. This movement of the work past the wheel results in polishing surface *1*. The frame then turns 90 degrees in a clockwise direction, as shown by arrow B, and surface *2* passes the fixed point C_1 in the same way that surface *1* did. Surfaces *3* and *4* pass point C_1 in a similar manner. After the entire periphery of the frame has been polished, the mechanism automatically stops for reloading.

The four intersections of the sides of the rectangle — points C_1 to C_4 — are the centers of the 90-degree angle of rotation at the end of each stroke. These points are analogous to the centers C_1 to C_4 shown in section B-B.

Cross-section A-A shows the work in dot-and-dash lines, mounted on a rotating and sliding nest RN. Surfaces *1*, *2*, *3*, and

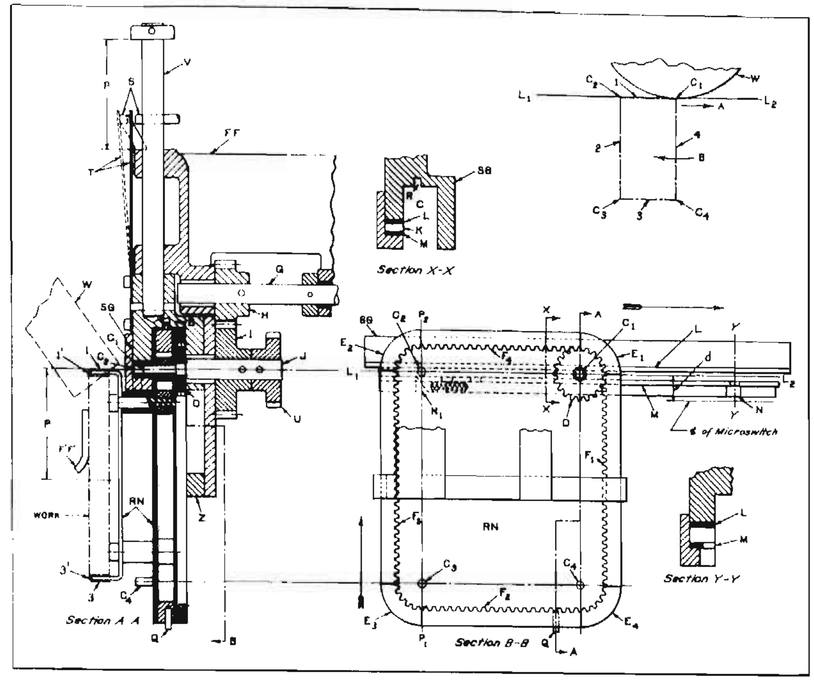


FIG. 7. Rotating and sliding mechanism designed for automatically polishing the periphery and faces of rectangular frames.

4, of which *1* and *3* can be seen in this view, are to be polished. The adjoining faces *1'*, *2'*, *3'*, and *4'* are also to be polished, and for this reason, the wheel W is mounted at an angle, as indicated.

Essentially, the mechanism consists of a stationary frame FF and $F'F'$ in which are bearings for the drive-shaft G to which is fastened a gear H. Rotating slowly and at constant speed, gear H meshes with gear I to drive a shaft J on which is mounted a pinion D. Other members of this assembly include a sliding guide SG and the rotating and sliding nest RN previously mentioned. The thrust of the polishing wheel is absorbed by a support Z. The nest slides (and rotates at the end of each stroke) within the space C section X-X. Four pins C_1, C_2, C_3, and C_4 are tightly pressed in nest RN, pins C_1 and C_3 being shorter than C_2 and C_4.

At the beginning of the polishing cycle, in the position shown, pin C_1 coincides with the center of pinion D, which is in mesh with one-quarter segment E_1 of an internal gear. The number of teeth in this gear is divisible by 4, which provides an equal number of teeth in each of the four segments E_1, E_2, E_3, and E_4 of the internal gear. These segments are connected by racks F_1, F_2, F_3, and F_4, the means of fastening these members being omitted in the drawing for the sake of clarity.

Nest RN slides in the direction of the arrow on two pins (in this position, C_1 and C_2) which ride in a slot K between liners L and M (sections X-X and Y-Y). The upper liner L is a continuous unbroken strip extending along the entire length of the slide guide SG, while liner M is cut through at N and N_1, as may be seen in section B-B. At N (see section Y-Y), this recess is one-half the width of the liner M, so that the longer pins C_2 and C_4 travel past it, while the shorter pins C_1 and C_3 move through it for the 90-degree rotation of the frame. It should be noted that this slot is angular in cross-section, having a separate piece that moves under spring pressure to close the slot after a pin has passed through it. This provides a smooth, unbroken surface for the pin to travel on after entering slot K.

In operation, the engagement of gears H and I rotates pinion D, which, after disengaging gear segment E_1 engages rack F_4. This moves nest RN along line L_1L_2, in the direction indicated by the arrow, until the pinion engages the second gear segment E_2. By that time, the short pin C_1 is at N and moves through the opening, thereby allowing nest RN to turn in a clockwise direction until pin C_3 enters the guide strip through the opening at N_2, which confines the angle of rotation to 90 degrees.

After the rotation of pinion D has rotated segment E_2, it engages rack F_3 and causes the nest to slide along line L_1L_2. A long stroke now takes place, at the end of which C_2 exits from N_3 as pinion D engages E_3 at the end of 90 degrees of rotation. C_4 enters N_1 and movement along L_1L_2 again takes place. This procedure is followed for the remaining side, so that the frame rotates and slides four times in one complete polishing cycle.

At the completion of a cycle, gears H and I are disengaged automatically to stop the movement of the mechanism and permit unloading and reloading of the work. The first step in this automatic action occurs when a pin Q (pressed into the rotating nest at a location farthest from point C_1) contacts a micro-switch (not shown) moving it a distance d. A slot R (section X-X) in the slide guide provides clearance for the pin during its travel along line L_1L_2. The microswitch energizes a solenoid that moves a rod S, disengaging a spring steel latch T which is connected to slide guide SG and nest RN. The slide guide and nest then drop, by reason of their unsupported weight, a distance P, traveling along two guide rods V (only one of which is shown), thereby disengaging gears H and I.

After reloading the nest, the operator raises the nest and guide by means of a knob U, bringing the assembly to the position illustrated. Since gear H rotates very slowly, its reengagement with gear I is a simple matter.

This mechanism can be applied to any polygon. If it is employed for a regular polygon, the short pins C_1 and C_3, as well as slot N, may be eliminated and four (or more) long pins used, which would drop after reaching the end of slot K.

Vibrating Roll Drive for Printing Press Fountains

In the ink-distributing fountain of a printing press, certain rolls have to reciprocate axially as well as rotate. The device shown in Fig. 8 illustrates a simple drive that produces both of these movements in a small-diameter roll A. This roll is in contact with a large-diameter roll B which rotates but does not itself reciprocate. Keyed to the end of each roll shaft is a spur gear. Both gears have pitch diameters that are equal to the outside diameters of the respective rolls to which they are attached.

On each side of the smaller gear C is a shroud D. These shrouds overlap the rim of the larger gear E. Gear E is mounted so that its axis is canted a few degrees to that of roller B.

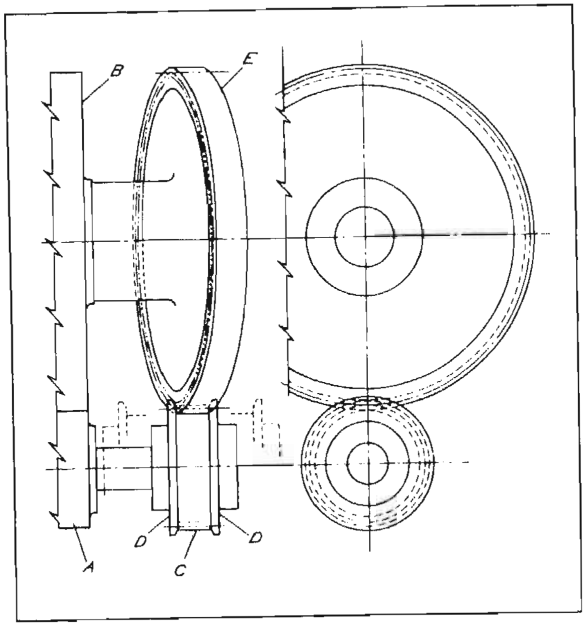


FIG. 8. Shrouds D constrain gear C to the helicoidal path of gear E, causing roll A to vibrate axially as well as rotate.

In operation, gear E drives gear C; the shrouds constrain the smaller gear to follow the face of the larger gear. The result is to produce an axial movement of the roll shaft of the smaller gear equal to the throw of the larger gear, as indicated by the dotted lines.

CHAPTER 12

Speed Changing Mechanisms

Providing a fixed or adjustable speed of rotation of a rotating driven member that is different from the speed of rotation of the driving member can be accomplished in many different ways. Mechanisms described in this chapter illustrate the use of gears, ratchets, friction wheels, cams, pulleys and belts in combinations that are noteworthy for some ingenious feature or special function which they perform.

Other speed-changing mechanisms are described in Chapter 11, Volume I; Chapter 10, of Volume II; and Chapter 12, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Ball Bearing Serves as Planetary Reduction Gear

A light-duty mechanism, driven by a $\frac{1}{10}$-H.P. motor, was found to be running too fast to function properly. Since no space was available within the mechanism housing to permit the use of a larger driven pulley, another means of speed reduction was sought. This took the form of a conventional single-row, heavy-series type ball bearing, which was used as a planetary reducing mechanism.

As shown in Fig. 1, driving pulley A is allowed free rotation on motor shaft B by two ball bearings C. A spacer D is placed between the bearings, and collar E retains them. Steel pins F are pressed into one side of the driving pulley, their free ends projecting into the rivet space between the cage recesses that contain the balls of the heavy-series type ball bearing G.

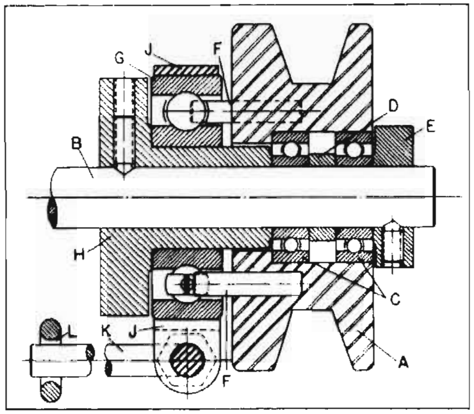


FIG. 1. This drive mechanism makes use of a ball bearing G to serve in the capacity of a reducing planetary gear unit.

The inner race of bearing G is press fitted on sleeve H, which, in turn, is secured to motor shaft B. The outer race of the bearing is held stationary by a split band J. Pressure is applied to the split band by tightening L-shaped bolt K, the end of which passes through eye-bolt L. The eye-bolt is fastened to the frame of the motor.

When the motor shaft rotates, sleeve H and the inner race of ball bearing G move in unison with it. This movement drives the balls and the cage of the bearing, causing the balls to roll along the fixed outer race. Driving pulley A is thus rotated at a reduced speed through the engagement of pins F with the moving cage of bearing G. The slight axial load applied to the balls by pins F tends to increase the power-transmitting capacity of the drive. In practice, no noticeable slip was encountered, even during the instantaneous application of heavy loads to the driving pulley A.

Rotational speed of the inner race of bearing G (also of the motor shaft) is $\left(1 + \frac{D_o}{D_i}\right)$ times the rotational speed of the cage, where D_o is the diameter of the inner-race ball track and D_i is the diameter of the outer-race ball track. With the particular bearing used in the illustrated setup, the speed reduction be-

tween the motor shaft and the driving pulley was approximately 2.5 to 1.

Shaft-Mounted Speed Reducer

Shown in Fig. 2 is a geared speed reducer of unusually compact design. The small cylindrical unit mounts directly on the drive-shaft and transmits its torque to the driven member (not shown) by means of a V-belt. Assembled appearance and construction details can be visualized from the partial section in Fig. 2 and the exploded view in Fig. 3.

The center section of the speed reducer consists of a steel sleeve A, internal gear B, and pinion C. Internal gear B is pressed into the steel sleeve. Pinion C, which is keyed to drive-shaft D, meshes with this gear.

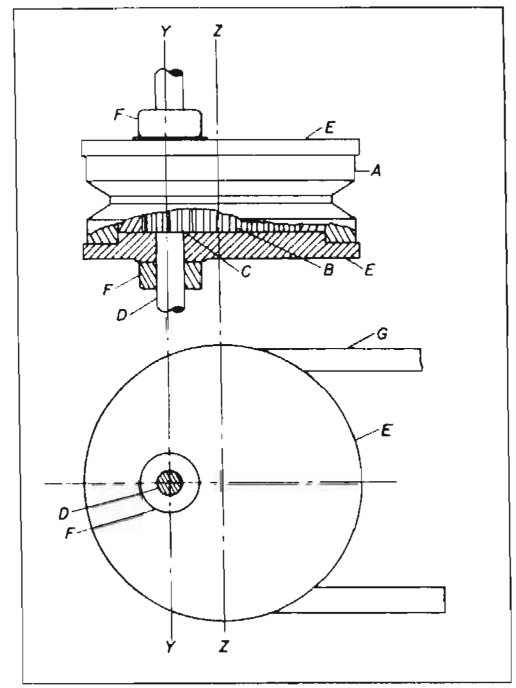


FIG. 2. Compact speed reducer mounts directly on drive-shaft D. Driven sleeve A is coupled to moving machine member by V-belt G.

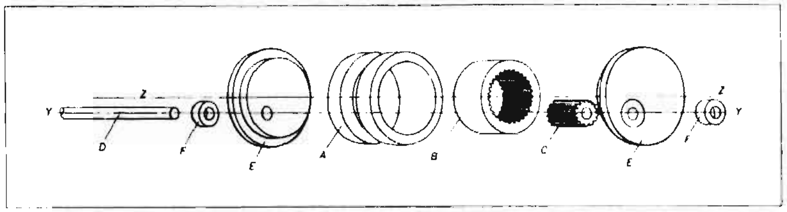


FIG. 3. Exploded view of shaft-mounted speed reducer. For quantity production, internal gear B can be eliminated and the teeth cut directly in sleeve A.

Two bronze end plates E have shoulders that are turned to a running fit with steel sleeve A. The bearing holes through which the drive-shaft passes are located off center by the distance necessary to provide proper engagement between the internal gear and the pinion. Collars F are locked to the shaft adjacent to the end plates and serve to retain the assembly intact. The unit can be packed with grease before assembly and, if desired, a grease fitting can be added for relubrication.

When the speed reducer is in operation there is a tendency for it to rotate with drive-shaft D about axis Y-Y. This proneness toward eccentric rotation is counteracted by the pull of V-belt G which restricts rotation to steel sleeve A about axis Z-Z. A secondary effect of the tendency to rotate about axis Y-Y is that adequate tension is maintained on the V-belt. If it is desired to hold the unit rigid, a support arm can be provided from a point on the machine frame to one of the end plates.

Geared Speed Reducer Changeable Under Load

A geared speed-reducing mechanism that can be regulated to obtain any one of sixteen different ratios without disengaging the input load is here illustrated. Changing of the speed ratio is accomplished with the gears in any position and while they are idle or in motion. Slippage will not occur if the torque transmitted is below a certain predetermined magnitude, but any over-loading of short duration is cushioned by a spring-loaded shock-absorbing arrangement.

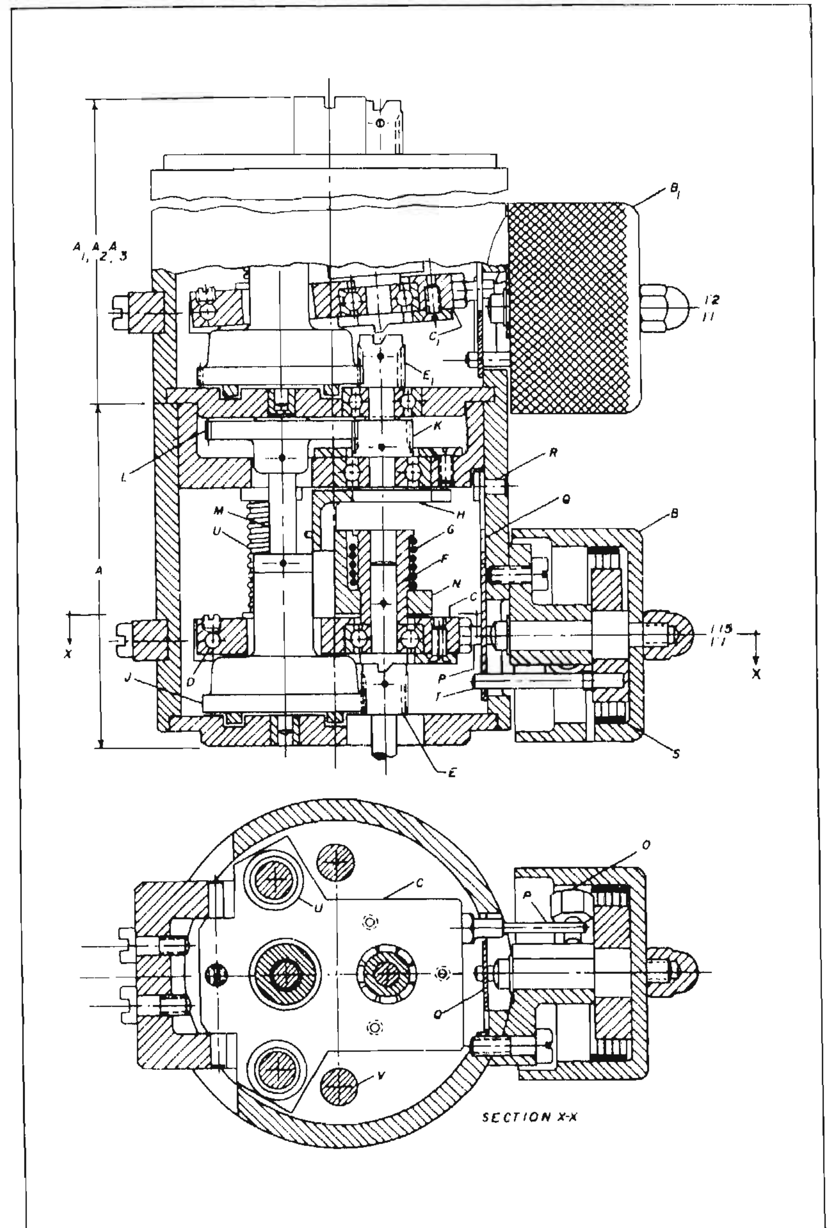


FIG. 4. On the down stroke, cam A pulls slide C into the press. Then punch B pushes the blank through die E.

The mechanism (Fig. 4) consists of four similarly constructed gear-boxes A, A_1, A_2, and A_3 mounted in-line vertically with the output of each unit being the input to the one immediately above. Each gear-box can be operated at either of two speed ratios, one of which is 1 to 1. The second gear ratio for each unit is as follows: 1.5 to 1 for A, 2 to 1 for A_1, 4 to 1 for A_2, and 16 to 1 for A_3. With this choice of speed ratios for the individual gear-boxes, sixteen different speed reductions ranging from 1 to 1 up to 192 to 1 are possible when the units are combined in a single four-stage mechanism. The speed ratio of an individual unit is changed by means of a selector knob B mounted on each gear-box. How the speed ratios of the four individual gear-boxes are combined to obtain the available speed reductions is shown in the accompanying table.

How Speed Ratios of Individual Gear-Boxes are Selected to Obtain Sixteen Speed Reductions

Over-All Speed Reduction	Speed Ratio of Individual Gear-Boxes			
	A	A_1	A_2	A_3
1 to 1	1 to 1	1 to 1	1 to 1	1 to 1
1.5 to 1	1.5 to 1	1 to 1	1 to 1	1 to 1
2 to 1	1 to 1	2 to 1	1 to 1	1 to 1
3 to 1	1.5 to 1	2 to 1	1 to 1	1 to 1
4 to 1	1 to 1	1 to 1	4 to 1	1 to 1
6 to 1	1.5 to 1	1 to 1	4 to 1	1 to 1
8 to 1	1 to 1	2 to 1	4 to 1	1 to 1
12 to 1	1.5 to 1	2 to 1	4 to 1	1 to 1
16 to 1	1 to 1	1 to 1	1 to 1	16 to 1
24 to 1	1.5 to 1	1 to 1	1 to 1	16 to 1
32 to 1	1 to 1	2 to 1	1 to 1	16 to 1
48 to 1	1.5 to 1	2 to 1	1 to 1	16 to 1
64 to 1	1 to 1	1 to 1	4 to 1	16 to 1
96 to 1	1.5 to 1	1 to 1	4 to 1	16 to 1
128 to 1	1 to 1	2 to 1	4 to 1	16 to 1
192 to 1	1.5 to 1	2 to 1	4 to 1	16 to 1

Lower gear-box A is set to operate at a speed ratio of 1 to 1 when lever C, which pivots on pin D, is in the lowered position as shown in Fig. 4. In this case, the motion of the input shaft

and gear E is transmitted through a positive clutch to shaft F. Spring G further transmits the motion to member H which is a section of a cylindrical cup. Member H is mounted on the same shaft as the input gear E_1 for gear-box A_1. In addition, the motion of gear E is transmitted to gear J, and the motion of member H is transmitted through gears K and L to shaft M. An over-running clutch prevents shaft M from becoming coupled to gear J. This arrangement is possible since the shaft rotates faster than the gear.

The over-running clutch, illustrated in Fig. 5, consists of two rollers that revolve with shaft M inside a bushed hole in gear J. (Rotation of both gear and shaft is always in the direction indicated.) If the shaft rotates faster than the gear, the rollers move

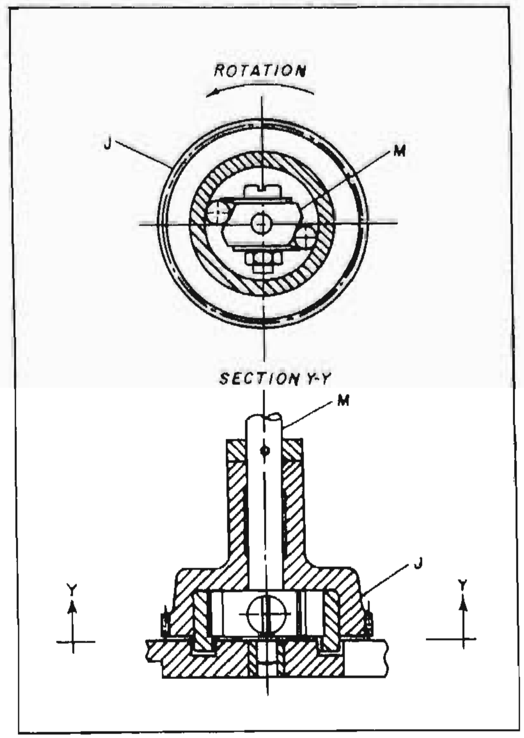


FIG. 5. The over-running clutch that prevents loss of load when speed ratio is changed is seen in detail.

freely with the shaft. Retainers in the form of thin leaf springs are bolted to the shaft to hold the rollers in place. When shaft F is disconnected from member E, the drive slows in rotation until the relative motion between gear J and shaft M becomes zero. At that instant, the angular flats on the shaft force the rollers outward until they become wedged against the gear, which will then become the driving member. This is accomplished immediately and practically no relative motion in the opposite direction is obtained.

When the selector knob is turned to obtain speed reduction, the lever C is pivoted to the raised position and the positive clutch is disengaged. Lever C_1 is seen in the raised position in Fig. 4. With this arrangement, the motion of gear J is transmitted to the output shaft and gear E_1 through the over-running clutch, shaft M, gear L, and gear K. When lever C is lowered to change back to the 1 to 1 ratio, output gear E_1 is again driven through the positive clutch, shaft F, spring G, and member H. The drive through the over-running clutch becomes uncoupled as shaft M again rotates faster than gear J.

The purpose of part N is to tension spring G so as to transmit only a predetermined safe torque without deflecting. Momentary loads greater than this value will cause the spring to deflect and thus cushion the shock of the mechanism. A shock load may occur as lever C is lowered to shift to the 1 to 1 speed ratio. Enough clearance is provided between parts N and H to prevent interference when lever C is in the raised position.

Since gears J and E have a 4 to 1 speed ratio for all four units, the ratio of each gear-box is varied only by the choice of gears K and L. In addition, gears K and L are selected to make the distance between their centers the same in all units.

When knob B is turned counterclockwise, bolt O lifts lever C to the raised position and the drive is immediately shifted to a lower speed. Lever C is secured in this position since an integral arm P is held by latch Q which pivots on headed pin R. Spring S, pin T, and its retaining collar rotate counterclockwise with the knob. Pin T rotates latch Q so that a protrusion on this member is hooked under arm P. Spring S also holds the knob

in the position in which it was set. Two springs U push lever C down and provide the necessary pressure to keep arm P hooked in place.

Shifting back to a higher speed ratio is accomplished by turning knob B clockwise. Bolt O, rotating with the knob, pushes latch Q aside and releases arm P. Springs U, in turn, push lever C to the lowered position, and shaft F becomes coupled to the input gear E by means of the positive clutch.

In operation, response of the mechanism is instantaneous when shifting any individual unit to a lower speed. Response to the changing of any unit to a higher speed (that is to the 1 to 1 ratio) is not instantaneous, as a very slight delay is necessary for the positive clutch and spring G to become driving members. Use of the overrunning clutch, however, keeps the drive operating under load until it is shifted to these members. The mechanism can be driven in only one direction but it is possible to make the output reversible by adding a special gear-box with operating ratios of 1 to 1 and -1 to 1.

Double Clutch Permits Reversal of Driven Shaft

In modern plants, it is frequently desirable to have control over the rotational direction of a driven shaft. A simple set-up that provides this control, without altering the direction of rotation of the driving shaft, is shown in the accompanying illustration.

A double electromagnetic clutch (see Fig. 6), consisting of units A and B, is mounted on driving shaft C. Two drive members, gear D and sprocket wheel E, are mounted on ball bearings and are in contact with clutch units A and B respectively. Driving gear D meshes with driven gear F, and driving sprocket wheel E is joined to driven sprocket wheel G by a link chain, not shown. Both of these driven members are keyed to output shaft H.

Shaft C rotates in the direction shown, arrow *1*, at all times. When it is desired to have shaft H rotate in the same direction, arrow 2, clutch unit B is energized. This results in the transmis-

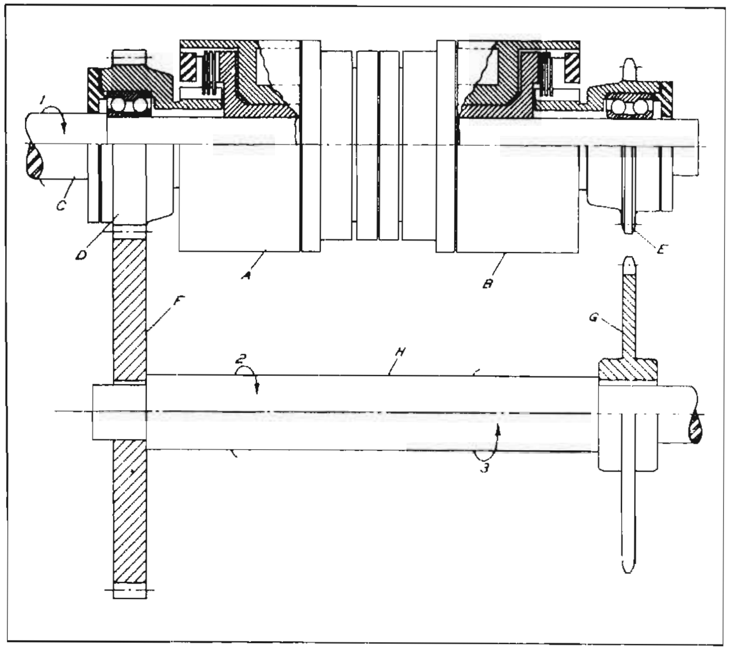


FIG. 6. Rotation of output shaft H can be reversed by the action of a double clutch without altering the rotation of driving shaft C.

sion of power from sprocket wheel E to sprocket wheel G, while gear D idles. If, on the other hand, it is necessary to rotate the output shaft in the direction shown by arrow 3, clutch unit A is energized. Sprocket wheel E then idles, as power is transmitted from gear D to gear F. If desired, the chain drive can be replaced with a belt drive.

Three-Speed Gear Conversion Unit for Bicycle Coaster Brake

The bicycle three-speed mechanism, as shown in Fig. 7, is assembled on a shaft or axle K, which is simply inserted in the regular coaster-brake hub shell F in place of the original plain shaft and sprocket. The regular brake plates or disks H, the ball bearings at each end of the hub shell, and sleeve G remain

undisturbed. The brake can be applied when using any of the three speeds by simply back-pedaling in the regular manner.

Essentially, the "Triplspeed" unit consists of a sprocket A driven by the bicycle chain; a planet-gear carrier B to which the sprocket is attached; four compound or stepped planet gears C journaled on the planet-gear carrier studs; a ring gear D with which the teeth of planet gears C are constantly in mesh (The ring-gear driver D is made with a triple-thread extension E, which serves as a means of driving the hub shell F through the sleeve G or applying the brake by exerting pressure on the brake plates H.); a sliding sun gear I with a larger supplementary sun gear J; and an axle K to which an axle cone L is permanently fixed.

In low gear, the sun gear I rotates freely on the two-piece axle sleeve M. One set of sun-gear teeth I meshes with the teeth at the larger end of planet gears C and the other set of sun-gear teeth engages the internal teeth of supplementary sun-gear J, whose outer teeth mesh with the teeth of the smaller end of the planet gears. Since it is impossible for the stepped planet gears C to revolve about sun gears (I and J) of unequal diam-

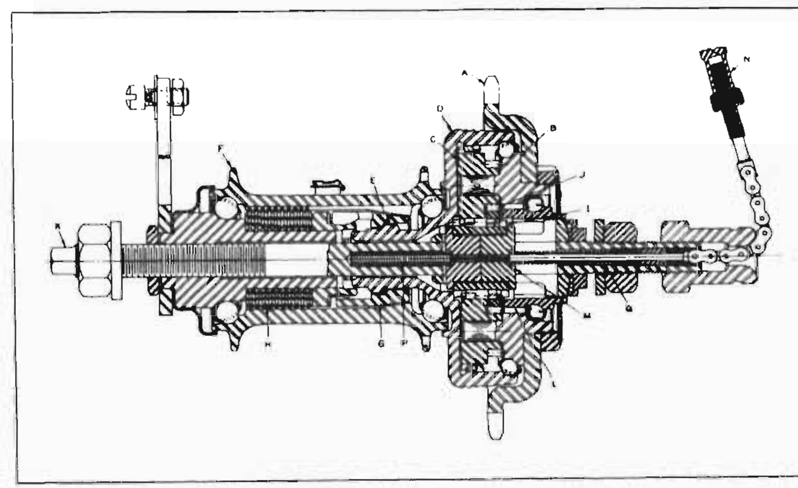


FIG. 7. Cross-section view of "Triplspeed" coaster-brake mechanism.

eters, the mechanism is locked and is driven as a unit about the axle, resulting in direct drive or low gear.

When the sliding sun gear I is moved to the right, it disengages the supplementary sun gear J and immediately engages the internal teeth of the fixed axle cone L. Thus the sun gear I becomes stationary, allowing the planet gears C to revolve about the sun gear and cause ring gear D to revolve. This results in the first over-drive or normal gear.

When the sliding sun gear I is moved farther to the right and deeper into the fixed axle cone L, the teeth on the left-hand end engage the internal teeth of the supplementary sun gear J, allowing the small planet gears C to revolve. The teeth at the larger end of planet gears C, being made integral with the gear on the smaller end, drive the ring gear D. This results in the second overdrive or high gear.

The outward movement of the sliding sun gear I is accomplished by pulling the control cable attached to the coupling N. The cable is operated by means of the control shifting lever mounted in a convenient position on the handle bar. When the cable tension is released, the sliding gear is allowed to move inward, resulting in successive gear changes to normal and low gear. The gear changes are, therefore, from low to normal to high, and vice versa. The brake can be applied at any speed in the conventional manner. This three-speed drive is a true synchro-mesh transmission. The gears cannot clash during gear changes because the sliding sun-gear teeth always leave one set of mating gear teeth before entering another set. The gears are so designed that the sliding gear, regardless of speed, always engages its mating gear without lost motion and without clashing. Therefore, shifting may be done at any time.

Each gear change may be pre-selected. Pre-selecting of any of the three speeds may be accomplished, while driving, by shifting the control lever in advance. When the rider wishes to change gears, he momentarily stops pedaling. This releases the driving pressure, allowing the gear change to be made quickly and automatically by the actuating spring. Pedaling can then be resumed in the pre-selected speed.

Changing from one speed to another is done as follows: The position of the sliding sun gear I is predetermined by the movement of the two-piece sleeve M on which gear I is mounted. The two sections of sleeve M are backed up by two springs P and Q within axle K. With the sliding sun gear I under torque from pedaling, the shifting lever is moved to the desired gear change position. When the torque is removed from the sun gear by momentarily stopping pedal movement, the axle spring moves the sliding sun gear automatically to the predetermined position. Upon resumption of pedaling, the sliding sun gear has assumed its proper position and the unit is in the desired gear. Similarly, pre-selection may be accomplished from high to low, normal to high, or any other desired combination.

One of the most attractive design features of this drive is that the shifting control-lever assembly is located near the handle-bar grip. Shifting of gears is done without the necessity of removing the hand from the grip. Shifting from low to normal to high is done by pulling up on the shifting lever with the fingers. Shifting from high to normal to low is accomplished by pushing down on the release lever with the thumb.

The calculation of the bicycle "gear number" is as follows: The bicycle "gear" is an indication of the distance traveled by the bicycle per revolution of the pedal crank or front sprocket. The "gear number" multiplied by 3.1416 equals the distance covered in one revolution of the front sprocket. Thus a bicycle having a 69 gear travels 216 inches, or 18 feet, along the road for each crank revolution.

The "gear" of a bicycle is the product of the number of teeth in the front sprocket and the number of inches in diameter of the rear wheel divided by the number of teeth in the rear sprocket. The result is the "gear number" in inches. The trade has dropped the dimensional unit of inches and the gear is known as a number. This calculation may be expressed by the following formula:

$$G = \frac{FW}{R}$$

where G = bicycle gear number;
F = number of teeth in front sprocket:
W = rear wheel diameter, in inches; and
R = number of teeth in rear sprocket.

In this new three-speed drive, the gears are reduced 25 per cent and increased $33\frac{1}{3}$ per cent from normal gear, so that the three gears may be computed as follows:

$$\text{Low } G = \frac{FW}{R}$$

$$\text{Normal } G = \frac{4FW}{3R}$$

$$\text{High } G = \frac{16FW}{9R}$$

In a "Triplspeed" equipped bicycle having 26- to $2\frac{1}{8}$-inch balloon tires, a 26-tooth front sprocket, and a 13-tooth rear sprocket, low gear would be 53, normal gear 70, and high gear 94.

Therefore, with the new three-speed coaster brake, a bicycle travels about 14 feet in low gear for each revolution of the pedal, about 18 feet in normal gear, and about 24 feet in high gear.

Moving Supports for Long Boring-Bar

When large forged gun barrels are bored, the boring-bar may be as much as 70 feet long. Such a long bar may easily be bent by its own weight. The consequence is a bore that will weave eccentrically in some sections, or be out of round. To eliminate these inaccuracies an effective traveling support system, Fig. 8, was devised.

The boring-bar is supported at three intermediate points on the tailstock by traveling supports X, Y, and Z. The left-hand support W remains stationary. The housing C contains the motor and the necessary gears and controls to feed the bar to the left into the bore at the desired rate. The right-hand end of the boring-bar is attached to this drive. The three intermediate supports move in the direction of the boring-bar feed at

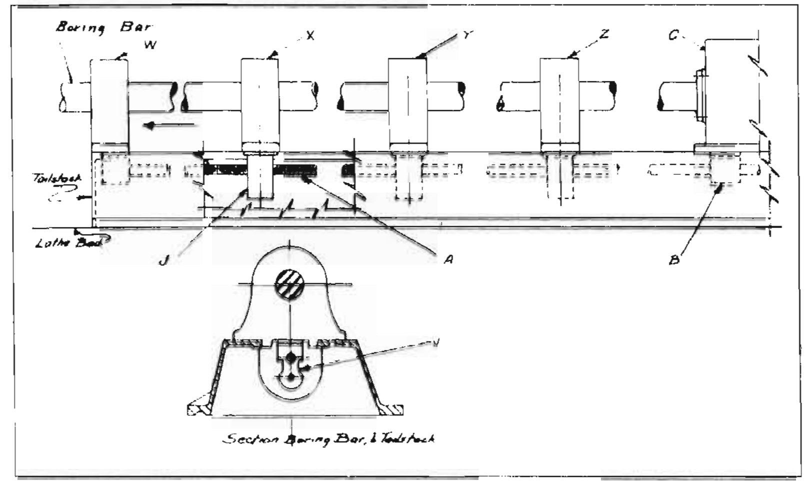


FIG. 8. Movement of boring-bar supports X, Y, and Z toward tailstock W must be proportionate to the total distance traveled by housing C.

speeds arranged so that they are at all times equally spaced between the support W and housing C.

It will be noted that the clear space between supports W and X is one-fourth of the sum of the clear spaces between W and C. Consequently, support X should move one-fourth of the speed of housing C; support Y at one-half the speed of C; and support Z at three-quarters of that speed.

The boring-bar is moved by the lead-screw A which is driven by a gear train in housing C. A nut B carries housing C forward as the screw is rotated. The screw A is provided with a keyway its full length. The bearing at the left end of A is arranged to take axial thrust in both directions. As nut B is rigidly attached to housing C, one revolution of shaft A will move housing C a distance equal to the pitch of the thread. If, however, this nut is rotated in the same direction and speed as the screw, there is no forward motion of housing C.

Figure 9 shows in detail the mechanism used under each support that gives forward motion. The motion consists of a cluster of four gears. Gear D slides on lead-screw A, and is turned by a feather key fastened in the bore of gear D. Gear E,

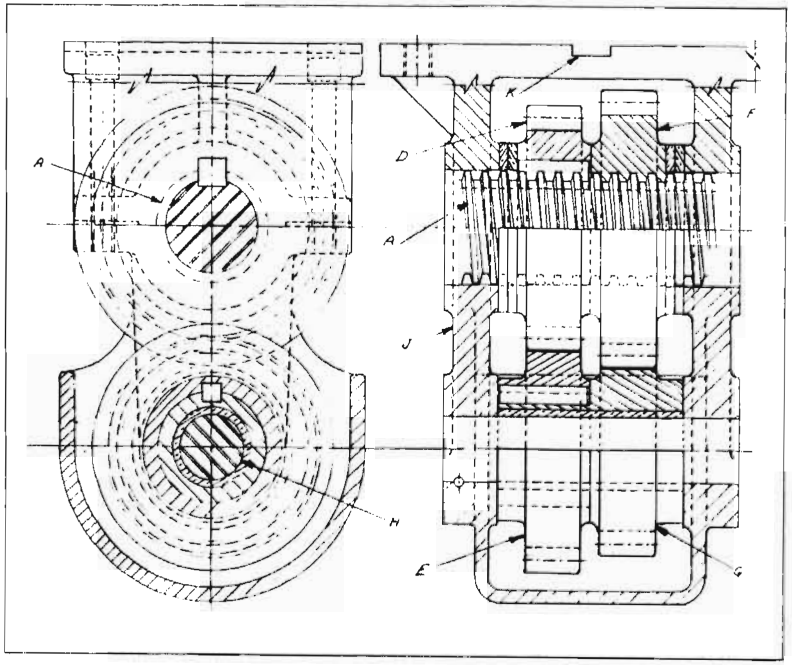


FIG. 9. Each of the boring-bar supports has a reduction gear train driving from the keyway in lead-screw A. Gear D carries the key but only slides over the lead-screw. Gear F drives the nut forward via the reduction gear train from G to E.

which meshes with D, is mounted on the hub of gear G. Gear F is threaded to fit lead-screw A. All four gears are held in a cage J, which is attached to the underside of its particular support, and held from axial-movement keyway K. By changing the ratios between gears D, E, G, and F, the supports can be caused to move toward W at the desired speeds.

Hydraulic Gear-Shift Control for Speed-Change Mechanisms

For controlling modern speed-change mechanisms that incorporate sliding gears or jaw-clutch couplings, hydraulic systems are being widely used. Generally, the gears of such mechanisms run at a speed that is too high to permit changing the speed

under operating conditions. To overcome this difficulty, either slow-motion features are embodied in the design or provision is made to prevent the gears from being shifted prematurely, that is, before the speed has been reduced to a rate suitable for the purpose.

An automatic hydraulic control system that meets the requirements mentioned is shown in Fig. 10. (Note that the views in Fig. 10 are in accordance with the European system of projection.) This system is of conventional type, using an oil pump which serves to lubricate the gearing or the machine driven by it — as for instance, a machine tool. It also supplies oil under the pressure required to effect the control. The system is composed mainly of a small gear type or plunger type auxiliary pump B which has its driving shaft C coupled to a constant-speed shaft of the gearing; a spring-loaded control valve D; and a throttle E.

When the gearing is in operation, oil drawn by the auxiliary pump through bore Q is carried through bore S into cylinder P. The oil lifts control valve D against the action of spring J so that the stream of oil coming from the main pump (not shown) and entering the system at orifice F is permitted to pass through annular space H and flow freely at orifice K to effect the lubrication. The additional amount of oil supplied by pump B is also fed to the lubricating pipe through passages L and M. The oil pressure can be varied by means of set-screw O after removing plug N.

When the gear mechanism is stopped by disengaging a clutch between the driving motor and the gearing, less oil is supplied by pump B and the control valve D will then drop because of the pressure exerted by spring J. This is accomplished at a rate which depends upon the adjustment of spring J and throttle E. Oil coming through cylinder P is then allowed to escape through passage M into the lubricating pipe. In moving downward, control valve D opens the pressure pipe R. As the main pump continues running, the oil it supplies is fed through a distributor to pistons for moving the gears. Thus, smooth shifting is insured.

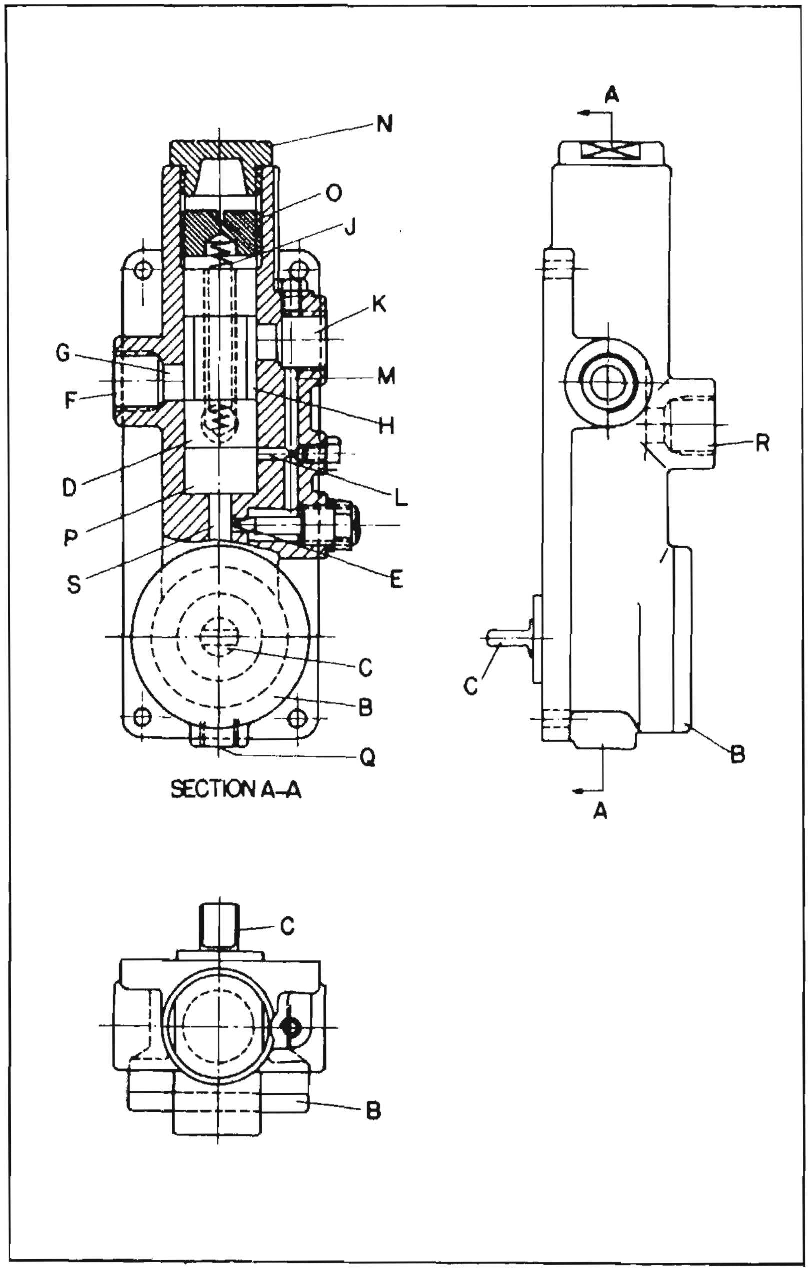


FIG. 10. Hydraulic mechanism assures smooth shifting of gears.

Differential Screw Assembly for a Slide

To enable a slide to travel at a reduced rate during part of its movement, a mechanism consisting of a differential screw assembly was designed. The device, shown in Fig. 11, controls the linear movement of a nut block *A* to which the slide is fastened. The nut block has a 10-pitch internal thread and engages the externally threaded end of a drive-shaft *B*.

At its opposite end, the drive-shaft is square in section, providing a sliding fit for the handwheel *C* which operates the slide. Part of the cylindrical length of the drive-shaft bears in a bushing *D*, which in turn, has a 12-pitch external thread engaging a fixed bracket *E*. Shoulders *F* on the bushing serve to limit its axial movement in the bracket. In addition, the left-hand shoulder is designed as a straight-tooth clutch *G*, the other member of which is developed from the hub end of the handwheel. A spring *H* keeps the two members of the clutch in normal disengagement.

Since another pair of shoulders *J*, fixed on the drive-shaft, prevents its axial movement, turning the handwheel produces a transverse travel of the nut block. When the clutch is disen-

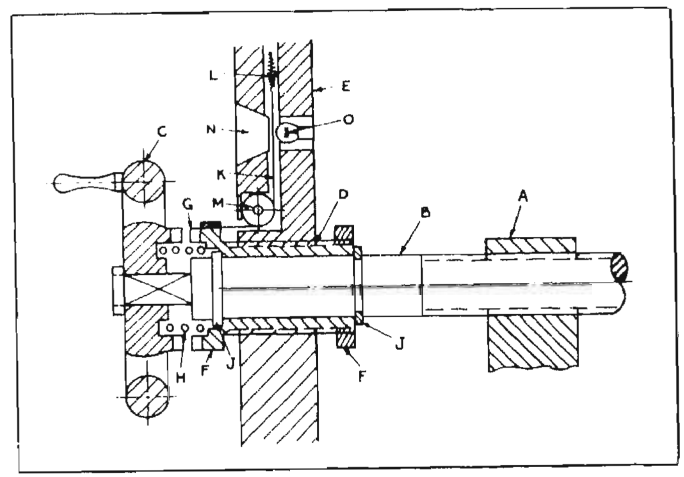


FIG. 11. When clutch *G* is engaged, the rate of travel of the slide is equal to the difference between the pitches of the two threads.

gaged, the bushing remains stationary while the drive-shaft rotates, and the nut block travels $1/10$ inch per revolution.

To produce the speed differential, the clutch is engaged, and the bushing, now rotating in unison with the drive-shaft, also moves axially. This axial movement of the bushing is transmitted to the drive-shaft through the thrust against the shoulders *J*. Movement of the bushing and drive-shaft is opposite to the direction of the nut block. But since the bushing and drive-shaft move only $1/12$ inch per revolution, the net result is to reduce the travel of the nut block to $1/60$ inch ($1/10 - 1/12 = 1/60$) per revolution.

Still finer adjustment of the slide is possible if the difference between the thread pitches is still smaller. Thus, with a 9-pitch nut block and a 10-pitch bushing, the travel of the nut block is only $1/90$ inch per revolution. When equipped with a graduated collar, the handwheel will register the amount of slide travel through any small degree of drive-shaft rotation. For example, if the collar has 110 divisions, controlled fine adjustments of 0.0001 inch can be made.

Difficulties may arise if the bushing is rotated continuously in one direction, since eventually, one of the shoulders *F* will jam against the bracket. For this reason, a simple indicating device consisting of a transparent band *K* has been provided. One end of the band is fixed to the shoulder; the other is held by a spring *L*. The band is guided around a roller *M* in front of a window *N* in the bracket which is illuminated by an electric bulb *O*.

If the shoulders of the bushing come too close to the bracket, red-colored portions appear in the window as a warning. Another method is to fit micro switches or electrical contacts to the shoulders, which can close a circuit to a warning light. Either method will serve to expand the use of differential screws, since an important drawback to their operation is thus eliminated.

CHAPTER 13

Speed Regulating Mechanisms

Machines which wind material such as paper, cloth or metal strip on spools or reels or which form or twist wire may require a synchronous rotation of two shifts with or without an occasional momentary acceleration or retardation of one shaft with respect to the other. In other machines the speed of the driven shaft must be maintained within close limits. The mechanisms described in this chapter have been designed to perform such special speed controlling functions. Similar mechanisms are described in Chapter 13, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Instant Acting Centrifugal Governor

A steam turbine in a chemical plant was required to operate below a certain rotational speed. For the particular application, the use of a conventional centrifugal governor would not be satisfactory, since it would reduce steam gradually by starting the decrease before the turbine reached the maximum allowable speed. This would cause a hunting effect when the speed is adjusted by means of the manual valve and would require the use of an especially complicated automatic setup. The desired effect could be achieved by adding a friction brake to a conventional governor. Such an arrangement, however, would result in hysteresis, since the governor would then stop the steam supply at a turbine speed much higher than that at which the supply is renewed. This, again, would be undesirable. The device shown in Fig. 1 provided the necessary speed control.

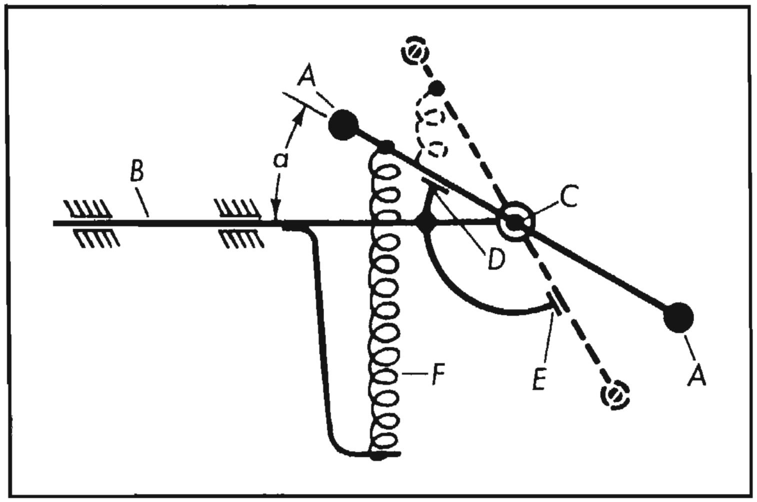


FIG. 1. Schematic diagram illustrating the principal design features of a quick-acting centrifugal governor.

In the schematic diagram of the mechanism (Fig. 1), a rigid bar is shown with weights A attached to each end. The bar is free to turn and is mounted on the main shaft B by means of a short shaft C. This short shaft is carried perpendicular to shaft B. Since a full revolution of the bar about shaft C is unnecessary, stops D and E are employed. A spring F is attached at one end to the bar as shown, and the other end is secured to a rigid support fixed on shaft B.

The centrifugal force f that acts on the mass m of weight A as shaft B rotates at a speed of n rpm is given by the following equation:

$$f = m\ (r \sin a) \left(2\pi \frac{n}{60}\right)^2$$

$$f = mr \sin a \left(\frac{\pi n}{30}\right)^2$$

where

r is the radial distance between the center of rotation of the bar and the center of gravity of the weight, and

a is the included angle between the bar and shaft B.

This force produces a torque on the bar, the lever arm of which is $r \cos a$. Since there are two weights A, the total torque T_A produced is

$$\begin{aligned} T_A &= 2f\ (r \cos a) \\ &= 2mr^2 \sin a \cos a \left(\frac{\pi n}{30}\right)^2 \\ &= mr^2 \left(\frac{\pi}{30}\right)^2 n^2 \sin 2a \end{aligned}$$

In an actual mechanism, the weights and the bar consist of an infinite number of elemental masses, and the total torque T_A will be the integral of the elemental torques these masses produce, or

$$T_A = K_A n^2 \sin 2a$$

where

K_A is a constant depending on the weight and shape of weights A and the bar.

The opposing torque produced on the bar by spring F is obtained in similar fashion. This spring is constructed in such a way that it is not under tension when angle $a = 0$. In Fig. 1, the extension of the spring is

$$r_f \sin a$$

where

r_f is the radial distance between the center of rotation of the bar and the point at which the spring is secured to the bar.

The force exerted by the spring is $cr_f \sin a$ where c is the spring constant of member F. Therefore, the torque T_f applied by the spring on the bar is given by the equation:

$$T_f = cr_f^2 \sin a \cos a$$

In the actual governor (shown in Fig. 2) the spring exerts pressure on two bars. Thus, when each rotates through an angle a, the expansion of the spring will be

$$2r_f \sin a$$

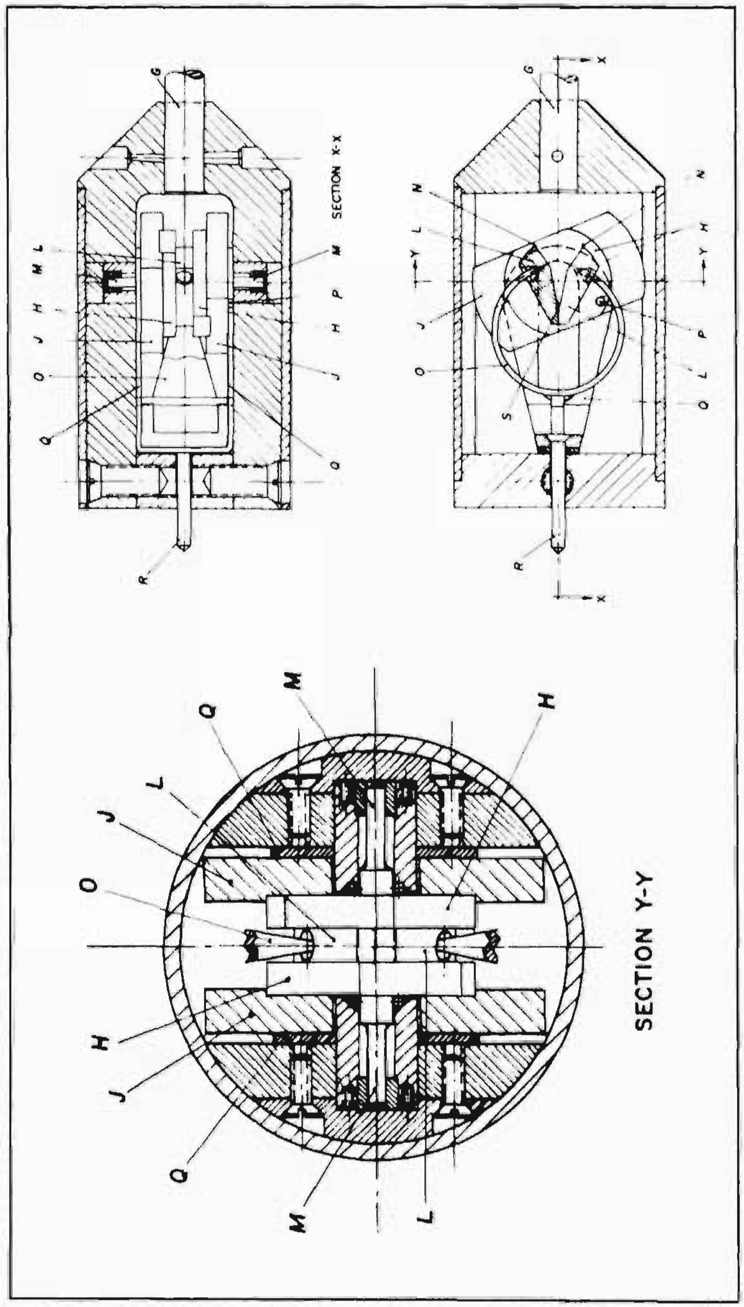


FIG. 2. This governor represents a practical application of the design shown in Fig. 1. In operation, the entire device rotates on the turbine shaft, and pin R provides the output.

and the torque on the bars will be

$$2cr_f^2 \sin a \cos a, \text{ or}$$

$$cr_f^2 \sin 2a$$

since the term cr_f^2 is a constant, it can be expressed as K_f and thus

$$T_f = K_f \sin 2a$$

In this manner the device is arranged so that the torque produced by the spring depends on angle a in the same way as the torque produced by the centrifugal force acting on the weights. Therefore, as long as

$$K_f > K_A n^2$$

the bar is held by the spring against stop D (angle a is at minimum value), but when the rotational speed n increases, $K_A n^2$ increases as the square of the speed until

$$K_f < K_A n^2$$

Then, regardless of the angle, the torque produced by the centrifugal force on weight A will exceed the torque produced by the spring, causing the weighted bar to almost instantly rotate from D to E (maximum value of angle a). When $K_A n^2$ becomes smaller than K_f, the bar will just as rapidly move back to stop D.

The rapid movements are due to the fact that n is squared in the $K_A n^2$ term, and therefore every rise and fall in the speed of the turbine causes a fast increase or decrease of $K_A n^2$ as compared with K_f. The critical speed at which the bar changes position is determined by the shape and weight of the bar and weights A and by the spring constant c. The energy for the movement toward stop E is supplied by the kinetic energy of the turbine, while the return movement is energized by spring F.

When translating the mechanism shown in Fig. 1 into a practical application, three conditions must be met:

1. The weights in the arrangement (Fig. 1) must have opposite counterparts to make it dynamically balanced and

thus prevent the centrifugal force from tending to bend shaft B and causing vibrations.

2. The support for the spring must move parallel to shaft B in order to permit proper application of spring tension to the bar as angle a changes.
3. A means of obtaining the output must be provided.

The actual governor, shown in the actuated position in Fig. 2, is designed so that the entire mechanism rotates on the main shaft G. To achieve dynamic balance, two crossed, weighted bars are employed, each of which is formed of two parts: H and J. Part H is made of steel and has a protrusion L. Member J is made of brass and serves as the weight. These weights revolve opposite each other and are placed in such a way that after a short rotation about shafts M in either direction, the protrusions L contact each other at points N (or S) and do not allow any further rotation in that direction. Parts H, L, and M are machined from one piece of steel.

One end of a spring O is set into a depression in each member L. An imaginary line extending between these depressions will always be perpendicular to shaft G, thus satisfying the second condition.

Spring O is triangular when viewed as shown in section X-X. This shape insures an equal distribution of stress through the circumference of the spring. In this way a spring which has linear characteristics and can withstand large deformations in relation to size is obtained.

To translate the rotation of the weights into a linear movement, a pin P is fixed on each of the weights J. Each pin has a matching groove in a sheet-metal part Q. When the critical speed is exceeded, members H rotate against the torque provided by spring O, moving parts Q outward. Members Q, in turn, move an actuating pin R, welded to them, outward.

Similarly, when the speed of the turbine slows, spring O returns parts H, which then contact each other at points S, and pins P pull parts Q, and thereby pin R, inward. In section X-X

only one of the pins P is seen, as the second is fixed to the underside of the upper weight J. Pins P move parts Q in the same axial direction, whereas weights J revolve in opposite angular directions.

The inner ends of shafts M are supported on ball bearings to reduce friction. Sleeve bearings are used at the other ends, as the loads there are smaller. Shafts M are also provided with hardened steel covers which absorb the centrifugal pressure in the direction of their axis.

Since it was not required, no provisions were made in the governor shown for adjustment of the critical speed. Critical-speed adjustment can be easily accomplished, however, by arranging weights so that their distance from shaft M can be varied.

Synchronizer that Insures Precise Speed Measurement

Accuracy approaching that of an electronic counting device has been obtained by the synchronizing unit shown in Fig. 3. This unit is used to control a brake-test dynamometer. It will

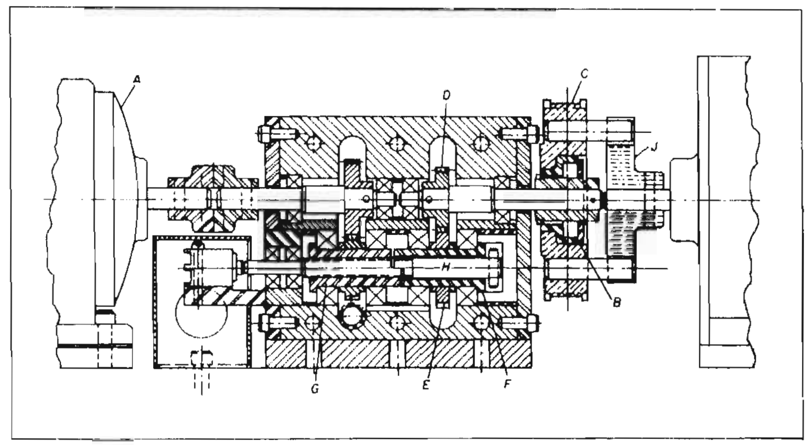


FIG. 3. Speed synchronizer for brake-test dynamometer which is extremely sensitive to speed changes.

sense a speed differential of 1 rpm in a rated speed of 3000 rpm.

The synchronizer receives power input from the synchronous motor *A*, which runs at a speed of 1800 rpm. Through appropriate gearing the power is transmitted at a speed of 3000 rpm to an inner member *B* of an overrunning clutch. Outer member *C* of this clutch is pulley-driven by belt from the flywheel of the dynamometer.

Both members *B* and *C* of the overrunning clutch rotate in the same direction. When the outer member attains a speed of 3000 rpm through the flywheel drive, both the inner and outer members travel at the same relative speed and no locking action takes place within the clutch. A fractional increase in the speed of the outer member, however, will cause the clutch members to lock and the outer member will then be driving the inner member.

This overdrive will cause gears *D* and *E* to run ahead of the synchronous motor input and cause sleeve *F* to advance a fraction of a revolution ahead of sleeve *G*. This advance will cause threads on shaft *H* to screw into threads in sleeve *G*, displacing shaft *H* from its original axial position. Such displacement will actuate a limit switch to trip the flywheel drive motor circuit and cause the flywheel to coast without power. This limit switch also actuates other circuits to apply brakes, timers, and recorders.

As the flywheel loses speed, the inner member *B* of the overrunning clutch is no longer locked to outer member *C*, and the entire train is again driven by the synchronous motor. The original position of sleeves *F* and *G* are re-established, causing threaded shaft *H* to screw back to its starting point and disengage the limit switch.

In the event the flywheel is accelerated too rapidly, no damage can be done to the synchronizer because the gear train will be driven by the flywheel. The synchronous motor will simply be overspeeded for a very short period. An electrical failure of the control circuit will produce the same result.

Sleeves *F* and *G* are so designed that their maximum displacement can never exceed 270 degrees. With 16 threads per inch on the threaded shaft, the axial displacement of the shaft can never exceed 0.047 inch.

Coupling *J* is connected to a zero-speed switch unit which can be used to stop recorders, clocks, and other devices when the pulley has stopped. The synchronizer is made up of stock gears and ball bearings, is oil sealed, and lubricated with light oil. The overrunning clutch is also a standard item designed to have almost zero backlash. A synchronous motor was used to obtain an accurate reference signal.

Constant Horizontal Velocity from a Crank

A paper-converting machine required that an operation be performed on the moving web. The web, however, had to be motionless at the time. Since it was impractical to stop the entire web, the device shown in Fig. 4 was designed for the purpose of stopping a portion of the web.

As illustrated, the web enters from the left, passes under and around roll *A* and over roll *B*. It then passes over table *C*, around and under roll *D*, and returns to the left, to and around roll *E*. At this point, the web leaves the device by moving to the right in the same plane as the web approaching roll *A*. Rolls *A* and *E* are mounted together in a frame *F* which, in turn, is

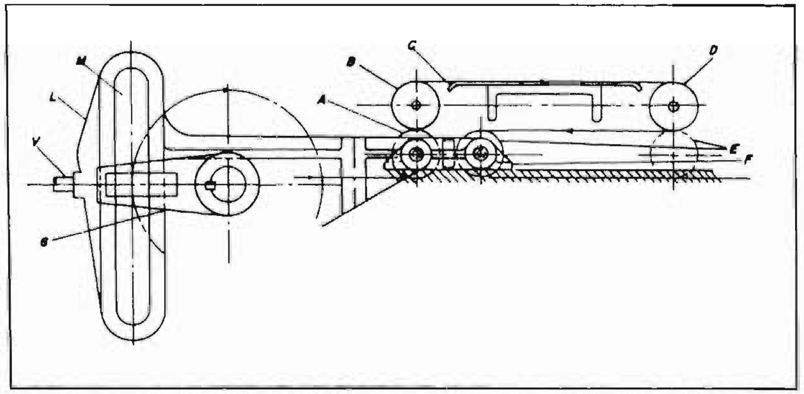


FIG. 4. Device for intermittently stopping a portion of a moving web employs a variable-length crank.

mounted on slides carried in suitable guides in the main frame of the machine.

If frame F with rolls A and E is allowed to move to the right at one-half the speed of the moving web, roll A will take up and pay out the oncoming web at one-half web speed. Since roll E is paying out the web at one-half the absolute web speed and at the same time is moving away from roll B at the same speed (one-half the absolute web speed), the web will remain stationary with respect to rolls B and D and table C. Roll E will receive and pay out the web at one-half web speed, due to the relative linear motion of roll E with respect to stationary roll D. The web, in turn, will be received by the next member of the machine at full web speed, due to the relative motion between that member and roll E.

The velocity of frame F must correspond to one-half the web speed, for if the speed is less, the web will still move forward over the table; or if the speed is greater, that portion of the web between rolls B and D will move backward to the left. Since the web must be stationary for an appreciable length of time, the movement of frame F to the right must remain at a constant speed during this period. A modified crank mechanism gives the frame constant motion.

The horizontal velocity of a crank movement is normally variable through an entire cycle, due to the constant length of the crank arm. If the length of this arm could be continually varied to suit through a portion of the cycle, a constant horizontal velocity would be obtained in that period.

In the arrangement illustrated, the length of the crank arm was varied as needed by means of stationary cam U (Fig. 5). The crank G, driven in the direction shown, has a slot H at the outer end which carries a slider J. Member J is retained in the slot by a cover plate K.

The horizontal motion of frame F is derived from the crank through a yoke L. This member has a slot M which accommodates a second slider N. Washers P and Q retain slider N, and proper clearance is maintained by bushing R. Both sliders are mounted on crankpin S, which also carries a cam roller

follower T. The latter, in turn, engages the groove in stationary cam U.

The profile of the cam groove, Fig. 6, from points 0 to 18 will provide the varying length of the crank arm, whereas the groove

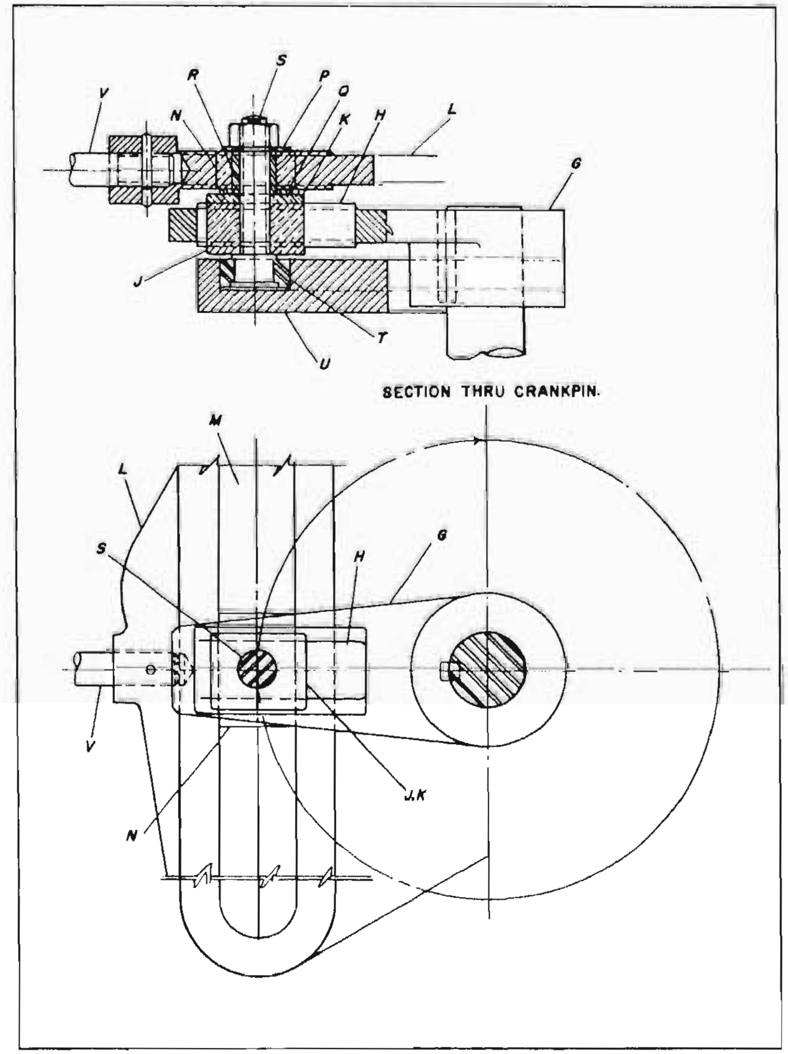


FIG. 5. Enlarged view showing details of mechanism which allows the effective length of the crank arm to be varied.

from points 18 to 0 may be concentric with the center of the cam. The horizontal cam displacements shown above the cam layout, Fig. 6, are variable from points 0 to 4, are equal from points 4 to 14, and vary again from points 14 to 18, but inversely, as from point 4 back to point 0. In this layout, the angular divisions are 10 degrees each.

The radius of the center line of the cam groove from the center of the cam at point 9 is calculated so as to give the required horizontal velocity for the constant-speed portion of the crank stroke. The balance of the groove radii for this portion (points 4 to 14) are then determined. Analytically, the lengths of these various radii would be: $d/\sin 10° = R$; $2d/\sin 20° = R_1$; $3d/\sin$

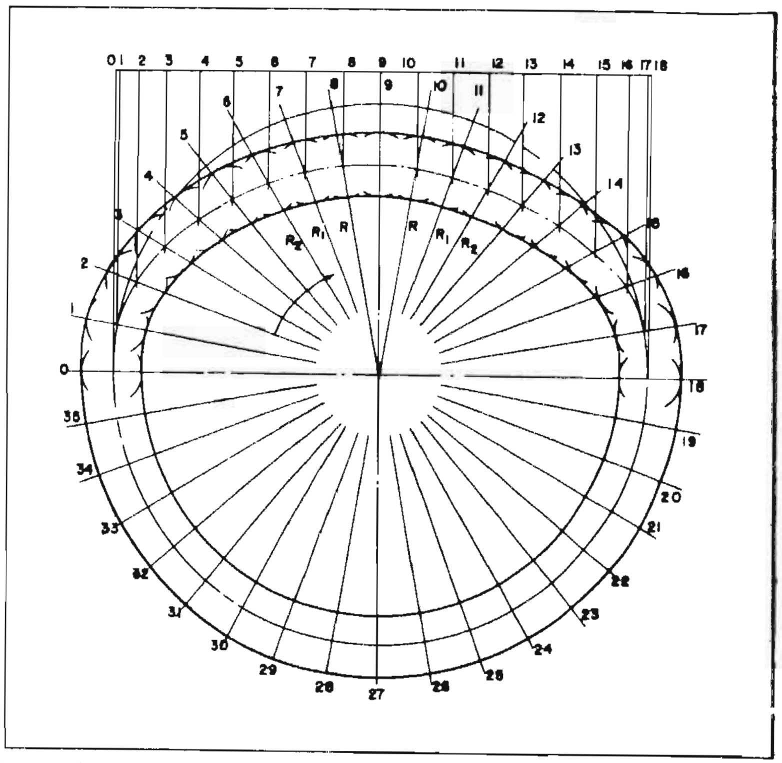


FIG. 6. Layout of the groove in cam U which gives uniform horizontal velocity to follower T, yoke L, and frame F.

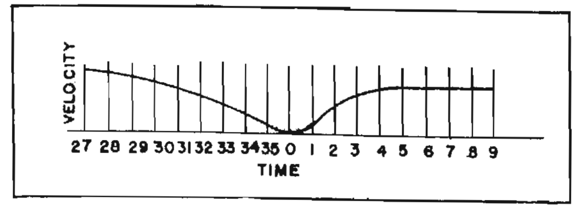


FIG. 7. Time-velocity diagram for one-half of the cam shown in Fig. 6. Constant horizontal velocity of the follower, yoke, and frame is obtained between points 4 and 9, and between points 9 and 14.

$30° = R_2$; etc. where d = total yoke displacement during constant-velocity portion of stroke divided by 10, the number of angular subdivisions in the 100-degree, constant-speed zone of the cam.

As will be seen from the time-velocity diagram, Fig. 7, the horizontal velocity curve from points 27 to 0 is typical of a normal crank motion. From points 0 to 4 the velocity increases until the required value is attained at point 4, and from there to point 14, is constant. Thus, the horizontal velocity of the yoke and frame will be constant through 100 degrees of crank rotation.

Operation of the device is as follows: When the crankpin passes point 27 on the cam, frame F will be moving to the left at maximum velocity. At this instant that portion of the web between rolls B and D will be moving to the right at a velocity much higher than that of the balance of the web. Then, as the crankpin reaches point 0, the frame will have zero velocity, and the entire web will travel at the same velocity.

From point 0 to point 4 the crank radius will be decreasing. Hence the speed of the web between B and D will be modified until, at point 4, the crankpin will have reached a radius that will give the frame a horizontal velocity equal to one-half of the oncoming web speed. At this point, that portion of the web over table C will be traveling at zero velocity and will continue to do so until the crankpin arrives at point 14. During this interval the required operation may be performed on the stationary web which is then supported by table C.

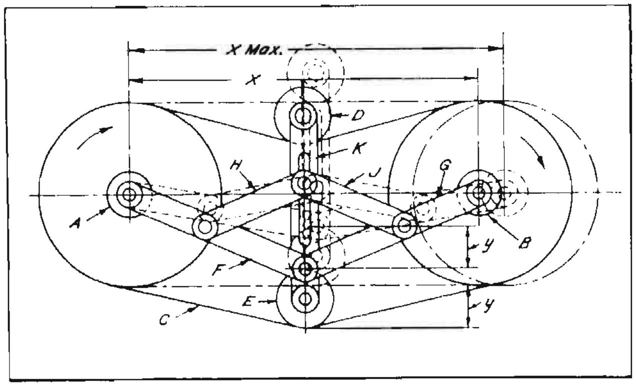


FIG. 8. Idler pulleys D and E, guided by a pantograph linkage mechanism, maintain uniform tension on steel band C when the center distance between shafts A and B is varied.

Transmitting Uniform Speed Between Shafts Having Variable Centers

Driven shafts can be rotated at uniform speeds regardless of variations in the distance from their driving shafts by means of the simple pantograph linkage mechanism seen in Fig. 8. The mechanism maintains uniform tension on a drive belt between shafts having a variable center distance. While a number of plane link mechanisms utilizing as many as eighteen joints and twelve members have been devised for this purpose, the device here described requires only six joints.

Driving shaft A and driven shaft B, having a variable center distance X, are provided with flat-belt pulleys connected by a steel band C. To provide a uniform tension on this band — independent of any changes in the center distance — the drive is equipped with two idler pulleys, D and E, which are guided by a pantograph linkage consisting of levers F, G, H, and J. The long levers F and G are free to pivot about pins pressed into the ends of shafts A and B. Slotted bar K, carrying pulleys E and D, is guided in a direction perpendicular to the common center line of the shafts when the center distance is varied. The relative positions of the mechanism components when shaft B is at its maximum distance from shaft A are shown by broken lines.

CHAPTER 14

Feed Regulating, Shifting, and Stopping Mechanisms

In all machines which perform operations on parts or on material, means must be provided for regulating, shifting and stopping the feed of either a tool or the work. Such mechanisms which provide for this are described here.

Other mechanisms which perform similar functions are described in Chapter 16, Volume I; Chapter 14, Volumes II and III of "Ingenious Mechanisms for Designers and Inventors."

Machine "Stops" Roll Labels Momentarily for High-Speed Die-Cutting

Repetitive operations are sometimes performed on lengths of material that are moving at high speed. Generally, the tool is allowed to move with the work, but the patented mechanism, shown in Fig. 1, momentarily "stops" the moving material long enough for a stationary tool to function. The work, however, passes through the device at a constant, high speed.

The mechanism was designed for die-cutting labels previously printed on rolls of paper stock called "web." Six rows of labels, printed on the web, are cut simultaneously by the die. Enough material is left between the labels within the rows so that the labels can be removed from the machine in rolls. Thin strippings which are left between the rows of labels are separated from them and diverted downward by a stripping mechanism (not shown).

A roll of printed label web A, subsequent to being placed on spindle B of the machine, is unwound a few turns. The loose

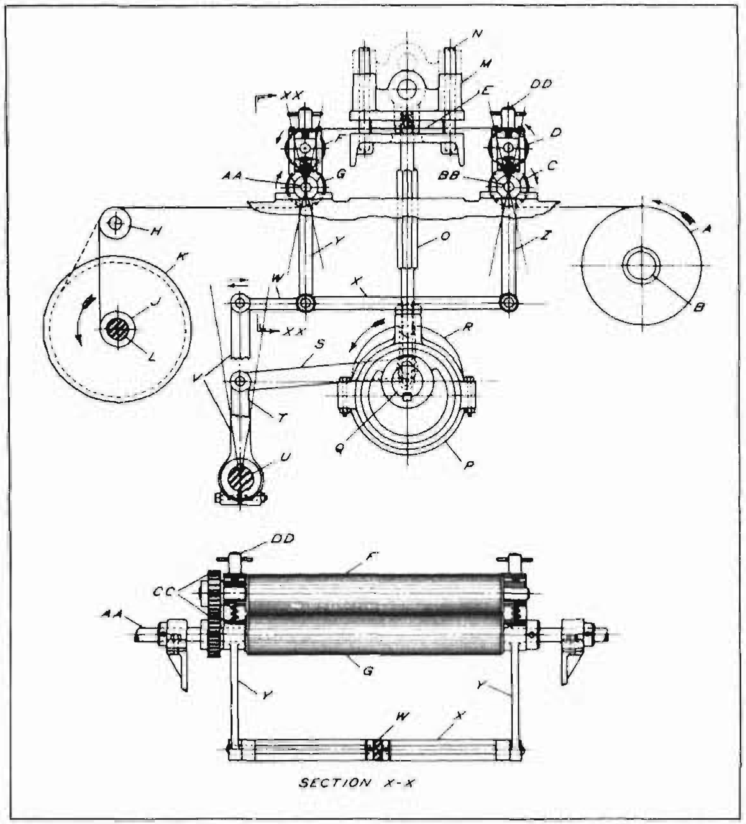


FIG. 1. With this mechanism, printed roll labels are momentarily held stationary and die-cut. The label web enters and leaves the device at a constant high speed.

end of the web is then threaded around rollers C and D, over die-plate E, around rollers F and G, over roller H, and onto cores J. Web is built up on the core by a drive through friction discs K which are held together under the pressure of light springs (not shown). This allows the core to rotate at a variable speed while taking up the web at a constant rate. Spindle L and the friction discs are driven by a separate small motor which runs constantly to keep the labels taut and thus prevent them from being torn by the stripping mechanism.

The upper member M of the cutting die reciprocates vertically on guide posts N, which are secured in the fixed lower die member. Member M is driven by a pair of connecting-rods O, eccentrics P, and a shaft Q. In addition, shaft Q, by means of a third eccentric drive and connected linkages, reciprocates rollers D and F in a short arc.

Connected linkages include driving disc R, connecting-rod S, rocker shaft U, lever V, link W, frame X, and two pairs of parallel levers Y and Z. Driving disc R has a radial T-slot for adjusting the length of stroke of connecting-rod S by repositioning its pivot pin. Levers Y and Z pivot on shafts AA and BB, respectively, but are always parallel.

Each pair of rollers is driven in opposite directions by means of identical gears CC and roller shafts AA and BB. These shafts are rotated at the same speed in the same direction by a roller chain drive. Pillow blocks mounted on the machine frame support the shafts.

In operation, the web is driven at a continuous speed by the rollers, but the rocking action of rollers D and F varies the absolute motion of the section of paper stock located between these rollers and under the die. In order to "stop" the work momentarily for the die-cutting operation, the backward motion of the rollers must be made approximately equal to the forward motion of the web relative to the rollers. This is accomplished by varying the rotational speed of the rollers so that they feed the proper length of web forward for the cutting die. To eliminate any strains in the web the stroke of rollers D and F is then reset by adjusting the stroke of connecting-rod S. On the forward stroke, the absolute speed of the web will be accelerated and will average out to the rate at which it is fed over the rollers.

The eccentrics are timed so that the die cuts the label as the web is "stopped" on the die-plate. In the illustration the mechanism is shown in this position. As the labels are made in various lengths, the speed of the web over rollers and the stroke of rollers D and F must be adjusted for each size.

Table Feed Mechanism Designed to Eliminate Manual Re-Engagement

On a special grinding machine, the work-table was driven by a screw geared to the power source. At the termination of the

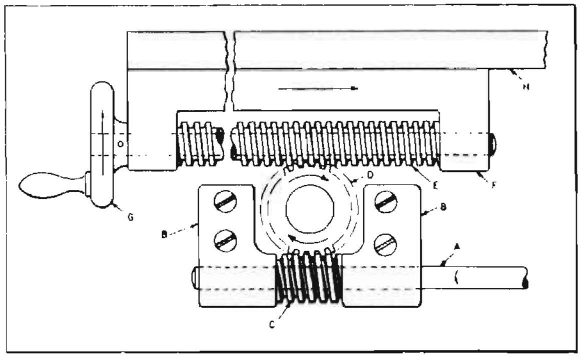


FIG. 2. Table feed mechanism that eliminates manual re-engagement of the clutch and feed-screw.

working cycle, a clutch dog was disengaged to permit the operator to return the table manually by means of a handwheel on the feed-screw. As the table movement was fairly slow, it was frequently impossible for the operator to engage the clutch immediately upon the completion of the loading cycle because the mating teeth were not in position for engagement. A period of several seconds was lost many times an hour in this way, making it advisable to change the feed mechanism to obtain an increase in production.

The design shown in Fig. 2 provided the desired results. A drive-shaft *A*, which rotates in the direction indicated by the arrow and has a worm *C* mounted on it, is supported by two bearings *B* attached to a stationary part of the machine. Worm *C* meshes with a worm-gear *D*, which rotates freely on its supporting stud. The worm-gear also meshes with the screw *E* on the opposite side, screw *E* being supported by member *F* attached to the table *H*.

During the working cycle, shaft *A* transmits rotary motion to the worm-gear *D*, in the direction indicated by the arrows, through the worm *C*. This provides linear motion to table *H*, since screw *E* does not rotate, but acts as a rack. On completion of the working cycle, the screw *E* is rotated by handwheel *G*, in order to move table *H* in the opposite direction. While this is taking place, worm-gear *D* continues to rotate.

While being loaded, the table slowly moves toward the working position, as it is still connected with the drive-shaft through the worm-gearing. Should the loading be completed before the table *H* has reached the working position, the handwheel *G* can be turned in a direction opposite to that in which it was turned previously to accelerate the table movement.

As none of the parts are disengaged at any time, there is no waiting period, resulting in a considerably shortened cycle.

Fine Feed Arrangement for a Surface Grinder

A patented mechanism by which a fine feed can be given to the wheel-head of a vertical spindle surface grinder is shown in the accompanying illustration. Coarse adjustment of the wheel-head slide *A* in relation to the column *B* is effected by rotation of a handwheel attached to the upper end of screw *C*, see Fig. 3.

When the fine feed is to be applied, screw *C* is prevented from rotating by tightening the internally threaded cup-shaped member *D* on the threaded boss that is integral with the column *B*. This is accomplished by means of the attached lever. The

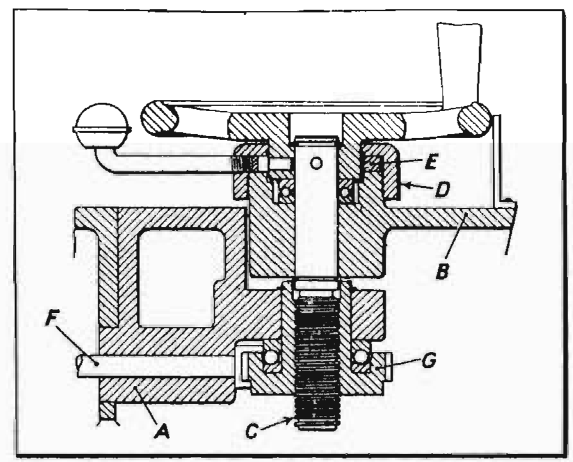


FIG. 3. Tightening threaded cup *D* against ring *E* holds screw *C* in a fixed position. Fine adjustment is then accomplished through the rotation of nut *G* by means of a worm drive on shaft *F*.

action causes ring E, which is keyed to the handwheel, to be clamped between the face of the boss and the bottom of the bore in cup D.

Fine vertical adjustment of the slide A is then effected by manual rotation of shaft F. A worm attached to this shaft drives a worm-wheel integral with nut G which moves along screw C.

An Intermittent Variable-Speed Movement

The device shown in Fig. 4 is used to feed strands of wire at a varying rate of speed through a portion of a machine that produces a woven wire product. A complete feed cycle consisting of a period of movement and an equal period of rest is accomplished by a ratchet and pawl arranged in combination with a pair of levers. The interesting feature of the mechanism is the method of providing the variable-speed motion during the feeding portion of the cycle.

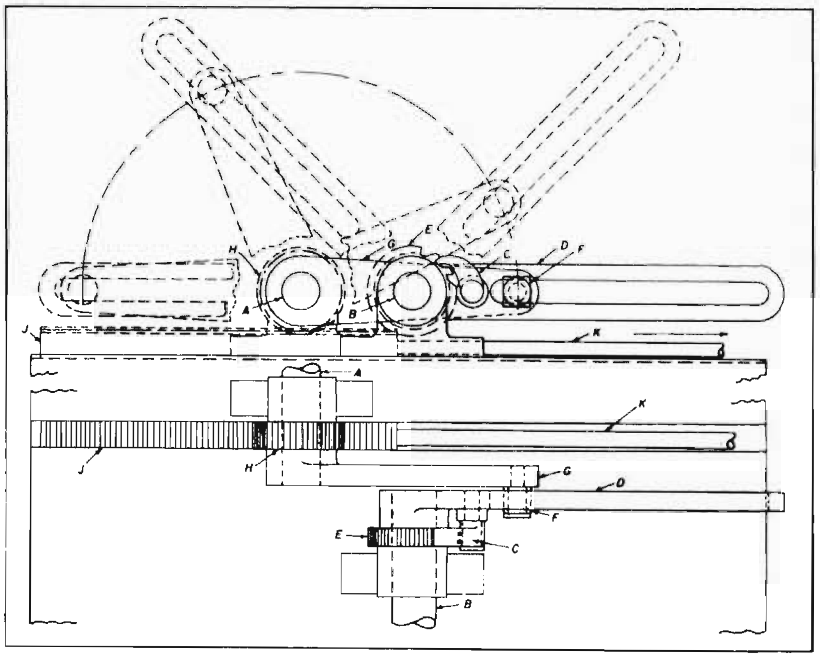


FIG. 4. Device used to convert a reciprocating motion into one that is intermittent and of variable speed.

Shafts A and B are both free to rotate in bearings attached to the machine. Pawl C is mounted on a lever D which, in turn, is pivoted on shaft B. A spring (not shown) holds the pawl in engagement with a ratchet wheel E keyed to shaft B. In addition, lever D is slotted to receive a slide block F. This block, in turn, pivots on a stud secured to the lower end of a lever G keyed to shaft A. A gear H, also keyed to shaft A, is constantly in mesh with a rack J, which is fitted into a groove in the machine table for guiding during its reciprocating motion.

In operation, rod K extending from rack J is given a uniform reciprocating motion by another part of the machine. As seen in Fig. 4, the assembly is at the end of the rest period of the cycle and rack J is about to be moved to the right. This action causes gear H and lever G to rotate counterclockwise, and lever G, through its slide-block and stud, transmits motion to rotate lever D in the same direction. Pawl C then engages the ratchet wheel E and causes shaft B also to rotate in the same counterclockwise direction as levers D and G.

The levers are shown dotted at three positions in their movement. Since they rotate on different axes, there is a continual change in their relative angular positions. This causes slide-block F to move toward the outer end of lever D, thus increasing the length of the effective lever arm. The movement of lever G is uniform throughout the cycle, and therefore, the slide-block transmits a continuously decelerating movement to shaft B until both levers reach the extreme left, where they are in a position of alignment. The rest portion of the cycle is accomplished during the return stroke of rod K by action of the ratchet and pawl arrangement.

Piloted Feed Control Mechanism

Auxiliary tooling for a copying lathe may be carried on an automatic overhead slide. A hydraulically operated mechanism for the independent control of the vertical feed of the slide is shown in Figs. 5 and 6.

Three basic units comprise the complete feed mechanism: a high-pressure hydraulic system to provide the necessary thrust

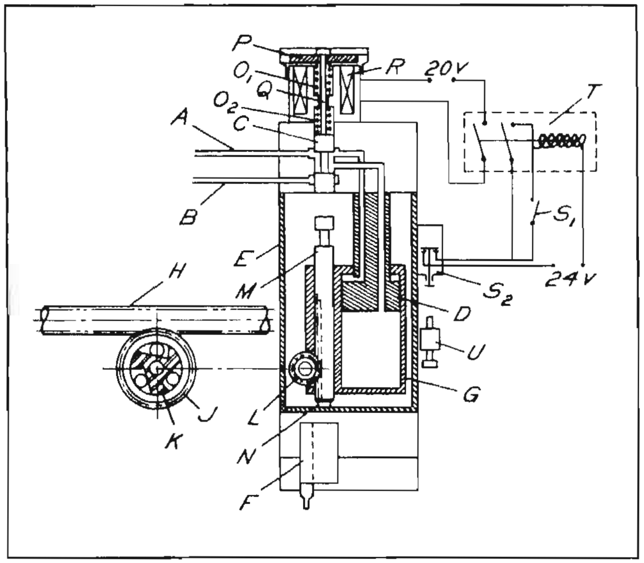


FIG. 5. Mechanism for controlling the feed of a hydraulically operated overhead slide used on a copying lathe.

for rapid approach, working feed, and rapid withdrawal; a lightly loaded mechanical unit to insure precise feed control; and an electrical system to afford positive control over the entire device. Pressure in the hydraulic system is built up by a motor-driven pump that is mounted in a support, forming the hydraulic reservoir, located beneath the lathe headstock. Fluid under pressure enters the slide through line A, Fig. 5, and returns to the reservoir through line B. Fluid flow from these lines to the slide is controlled by a double-acting control valve C.

Differential piston D is attached to moving slide E so that tool-holder F and the piston will move in unison. Cylinder block G, in which the piston rides, is mounted to the frame of the slide unit.

Pilot lead-screw H, the restrictive component of the feed control unit, is driven from the lathe spindle through a separate gear-box, providing it with a selection of eleven feeds. Located within the slide support, and meshing with the lead-screw, is tangential gear J. This gear fits over a roller type clutch-wheel

K. As the lead-screw rotates gear J in a clockwise direction, the rollers in the clutch disengage, providing a free-wheeling condition. It should be noted that the gear and clutch assembly is shown outside of the slide unit for clarity.

Pinion L, mounted on the same shaft as clutch-wheel K, meshes with vertical rack M. The rack is situated along the same axis as is control valve C. When the rack is pushed downward, pinion L rotates. As a result, the clutch-wheel rotates in the same direction as, but faster than, gear J. This causes the clutch rollers to engage, thus restricting the speed of the descending rack to the selected speed of lead-screw H as long as pressure is maintained on the rack. The lower end of the rack is positioned by stop N which is integral with slide E.

When the unit is inactive, spring O_1 forces plate P, which is free to slide on headed valve-stem Q, against the cover of solenoid R. This holds the control valve in a raised position so that the feed-back orifice is closed as shown at X in Fig. 6. In this position the hydraulic fluid under pressure is channeled to both the small chamber above the piston and to the large chamber below the piston. Although the pressures in both chambers are equal, a larger piston-face area is exposed in the lower chamber so that the total force pushing upward is approximately twice

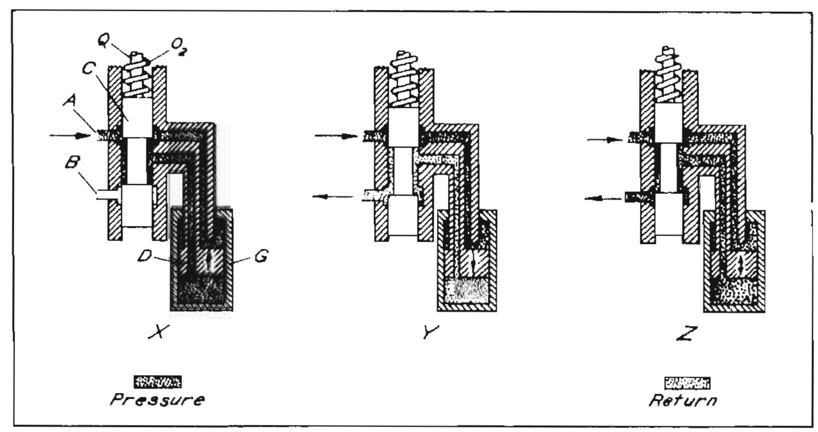


FIG. 6. Positioning of valve C results in either of three movements—rapid approach, rapid withdrawal, or working feed.

that pushing downward. Therefore, the piston, together with the slide, is maintained in a raised position.

With the machine in operation, movement of the overhead slide is initiated either by rotation of the template carrier or by the arrival of the lathe saddle at a chosen longitudinal position. Upon the closing of the switch S_1, relay T is closed, and remains so even when the switch reopens. This energizes solenoid R which attracts plate P to it, compressing spring O_1 and releasing the valve-stem.

Under the influence of spring O_2, the control valve is now forced downward to its lowest position as shown at Y, closing the connecting passage between the two cylinder chambers and opening the return line to the reservoir. Oil delivered by the pump is now directed only to the small chamber above the piston. The piston is thus forced downward imparting a rapid approach to the cutting tool as, at the same time, the oil leaves the large chamber below the piston and flows into return line B.

During ascent of the piston, or rapid withdrawal of the cutting tool, the control valve is in the same position as it is when the unit is inactive. This is the raised position that may be seen by referring back to X, in which the connecting passage between the two cylinder chambers is opened, and the feed-back orifice is blocked. The pressure in each being equal, the greater total force exerted against the large bottom face of the piston forces it to rise at maximum speed. Oil being delivered by the pump joins the oil leaving the upper cylinder chamber and flows into the lower cylinder chamber.

The equilibrium position, or the position assumed by the control valve while the cutting tool is being fed into the work-piece, is illustrated at Z. This position is effected when the slide descends rapidly until the bottom of the control valve contacts the top of rack M. The rack is then forced downward, rotating pinion L and causing clutch-wheel K to rotate in the direction shown. The rotative speed of the clutch-wheel results in its engagement with gear J. Rapid downward movement of the rack is thus checked, it being able to descend only as fast as lead-screw H, through gear J, will permit. This speed is selected by the lathe operator.

As the speed of the rack is reduced, the control valve is pushed upward allowing the feed-back orifice to close. The hydraulic fluid, being now diverted to both sides of the piston, forces the slide in the opposite direction. Because spring O_2 constantly tends to push the control valve downward to effect a rapid approach, a series of valve movements occur until the opposing pressures on the piston are stabilized.

In this position, oil enters the upper annular position of the valve housing and passes immediately to the small chamber. The valve is positioned so as to leave a bleed opening in each of the two annular spaces. Oil bleeds through the first opening to the lower cylinder chamber causing the cutting tool to raise slightly. The oil then bleeds through the second opening and returns to the reservoir, causing the cutting tool to lower slightly. All excess oil supplied by the hydraulic pump by-passes the valve and returns to the reservoir.

At the end of the cutting stroke, microswitch S_2 is actuated by adjustable stop U. When the circuit is broken at this point, relay T is opened with the result that solenoid R is de-energized. Control valve C is pulled upward by the action of spring O_1 against plate P, and a rapid withdrawal is effected. As the slide is raised, it causes the rack to travel with it. The unit is held in this raised position until again activated. If necessary, the solenoid cover can be adjusted to limit valve displacement, thereby varying the speed of withdrawal.

Pi-Ratio Universal Rack-Indexing Attachment

Racks of different pitch can be cut on a milling machine equipped with the indexing attachment shown in Fig. 7, without any change of gears being required. One particular two-gear combination and one or more commercially available index-plates can be used to accurately index a milling machine table for cutting racks in all the commonly employed diametral pitches. Although this gear set is for use on machine tables having a feed-screw with a ¼-inch lead, effective gear arrangements may be set up for other leads.

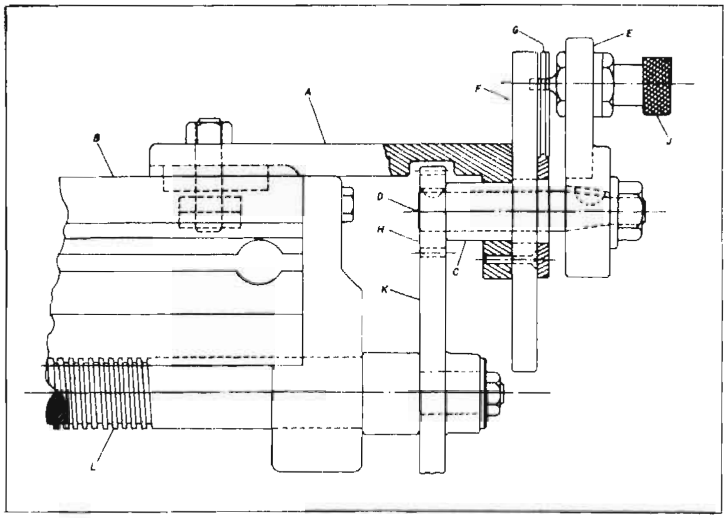


FIG. 7. This attachment for the milling machine facilitates rack-cutting. The same gears can be used to produce racks in all commonly used diametral pitches.

The linear pitch of a rack is equal to π (3.1416 inches) divided by the diametral pitch. Consequently, the number of teeth in 3.1416 inches of rack will be equal to its diametral pitch. A gear set chosen for the indexing attachment must be able to move the milling machine table 3.1416 inches with a number of turns of the crank-handle that can be readily subdivided by the diametral pitch. An ideal gear set is one that can be used with a small selection of index-plates to index the table the exact amount for a rack of any standard diametral pitch.

The ideal condition can be obtained on machines having ¼-inch lead feed-screws with a 71- and 113-tooth gear used in combination. Ideal arrangements or close approximations may be set up for other leads with two- or four-gear combinations.

Construction of the attachment as set up for two-gear operation is illustrated. Bracket *A*, which is keyed and bolted to the machine table *B*, supports bushing *C*, shaft *D*, crank *E*, index-plate *F*, sectors *G*, and gear *H*. A spring-loaded plunger *J* for

Index Settings for the Pi-Ratio Universal Rack-Indexing Attachment Based on a Commercially Available Index-Plate

Diametral Pitch	Number of Complete Index Turns	Fraction of Turn to be Indexed	Number of Index Holes in Circle	Number of Index Holes for Setting	Diametral Pitch	Number of Complete Index Turns	Fraction of Turn to be Indexed	Number of Index Holes in Circle	Number of Index Holes for Setting
128	0	5/32	96	15	8 1/2	2	6/17	68	24
120	0	1/6	54	9	8	2	1/2	66	33
96	0	5/24	72	15	7 1/2	2	2/3	54	36
80	0	1/4	72	18	7	2	6/7	84	72
72	0	5/18	54	15	6 1/2	3	1/13	78	6
64	0	5/16	96	30	6	3	1/3	54	18
56	0	5/14	84	30	5 1/2	3	7/11	66	42
48	0	5/12	72	30	5	4	. . .	any	0
44	0	5/11	66	30	4 1/2	4	4/9	54	36
40	0	1/2	66	33	4	5	. . .	any	0
36	0	5/9	54	30	3 1/2	5	5/7	84	60
32	0	5/8	72	45	3	6	2/3	54	36
28	0	5/7	84	60	2 3/4	7	3/11	66	18
24	0	5/6	54	45	2 1/2	8	. . .	any	0
22	0	10/11	66	60	2 1/4	8	8/9	54	48
20	1	. . .	any	0	2	10	. . .	any	0
19 1/2	1	1/39	78	2	1 7/8	10	2/3	54	36
19	1	1/19	76	4	1 3/4	11	3/7	84	36
18	1	1/9	54	6	1 5/8	12	4/13	78	24
17 1/2	1	1/7	84	12	1 1/2	13	1/3	54	18
17	1	3/17	68	12	1 7/16	13	21/23	92	84
16 1/2	1	7/33	66	14	1 3/8	14	6/11	66	36
16	1	1/4	72	18	1 5/16	15	5/21	84	20
15	1	1/3	54	18	1 1/4	16	. . .	any	0
14 1/2	1	11/29	58	22	1 3/16	16	16/19	76	64
14	1	3/7	84	36	1 1/8	17	7/9	54	42
13 1/2	1	13/27	54	26	1 1/16	18	14/17	68	56
13	1	7/13	78	42	1	20	. . .	any	0
12 1/2	1	3/5	60	36	15/16	21	1/3	54	18
12	1	2/3	54	36	7/8	22	6/7	84	72
11 1/2	1	17/23	92	68	13/16	24	8/13	78	48
11	1	9/11	66	54	3/4	26	2/3	54	36
10 1/2	1	19/21	84	76	11/16	29	1/11	66	6
10	2	. . .	any	0	5/8	32	. . .	any	0
9 1/2	2	2/19	76	8	1/2	40	. . .	any	0
9	2	2/9	54	12					

indexing is mounted on the crank. Gear K is keyed to the feed-screw L of the milling machine table.

If the milling machine screw has an 0.250-inch lead, then 4 times 3.1416, or 12.5664 turns, will be required to move the table 3.1416 inches. An easily subdivided number of turns of the crank should be used to produce this table movement. Twenty revolutions of the crank are required to move the machine table 3.1416 inches when the 71- and 113-tooth gears are used, and the accompanying table shows how commonly used pitches are indexed. This ideal combination will theoretically move the machine table 3.141593 inches or π inches to six places with 20 turns of the crank, as 20 turns times 71/113 gear ratio times ¼-inch lead of feed-screw equals 3.141593 inches of table movement. The 71-tooth gear should be mounted on the crank-shaft D and the 113-tooth gear on the feed-screw L. It should be emphasized that with this arrangement these gears will not have to be changed to produce racks in any of the commonly used diametral pitches. Furthermore, racks based in design on the metric module system may be indexed using only a 127-hole circle.

CHAPTER 15

Automatic Work Feeding and Transfer Mechanisms

This chapter deals with the automatic delivery of workpieces in the proper position for the operation to be performed on them. Other automatic feeding mechanisms are described in Chapter 16, Volume I; Chapter 14, Volume II; and Chapter 15, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Escapement Mechanism Feeds Rods of Various Diameters

Round bar stock of random diameters can be fed one at a time, regardless of the differences in diameter of adjacent bars, by a battery of identical escapement mechanisms that operate from a common drive-shaft. The design and operation of this device are shown in Fig. 1.

The rods are loaded on a feed-table consisting of parallel steel strips A, as can be seen in the plan view at V. A table slope of ½ inch per foot tends to make the rods roll. When the escapement mechanism is in neutral position, as shown in view W, the rods are restrained from rolling by the heel portion of feed-arm B. The center of the radius of the curved surface is coincident with that of square drive-shaft C on which all of the escapements are mounted.

To initiate the delivery cycle, shaft C rotates in a clockwise direction through an arc of 45 degrees, ending up in the position illustrated in view X. This permits the entire stock of rods to roll forward until the first rod strikes the long edge of feed-arm B. A short dwell period is provided to allow all the bars to complete their forward travel. Shaft C then moves in a

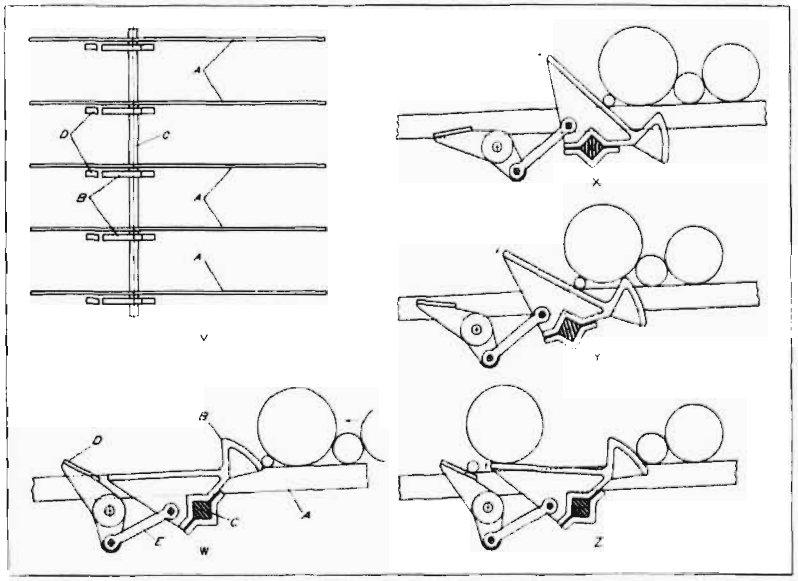


FIG. 1. Escapement mechanism permits feeding of round bar stock of assorted diameters. Arms B and D function together to allow only one rod at a time to be released, regardless of its diameter.

counterclockwise direction, view Y, until the mechanism has returned to its original position, trapping one or more rods in the space between feed-arm B and stop-arm D.

During the next clockwise movement of shaft C, view Z, stop-arm D is lowered, permitting the first rod to roll away. At the same time, feed-arm B begins to rise, causing the second bar to roll backward. This is due to the spacing between the two arms which maintains the center of gravity of the second bar to the right of the end of arm B. Link E that connects arms B and D should be adjustable to facilitate alignment of all the stop-arms across the width of the feed-table.

Escapement Feeds Cylinders One at a Time Down Ramp

In mass-production plants, cylindrical parts are frequently rolled downhill in chutes from one machine location to the next. An example is the gravity handling of automotive pistons in

partly finished condition. Because the force is gravity, the back-up of parts in a chute is a convenient feed magazine. The automatic releasing of one work-piece at a time to feed a machine tool is often a problem.

Figure 2 shows an air-powered escapement device for installation in a gravity feed chute for handling cylindrical parts. Its operation can easily be connected in the electrical system of a machine tool. Compressed air entering cylinder port A swings the cage of rollers E and F in direction C. Rollers E and F are freewheeling. As the roller cage swings, the work cylinder G will roll off to ramp H. At the same time roller F rises to hold cylinder J on ramp K.

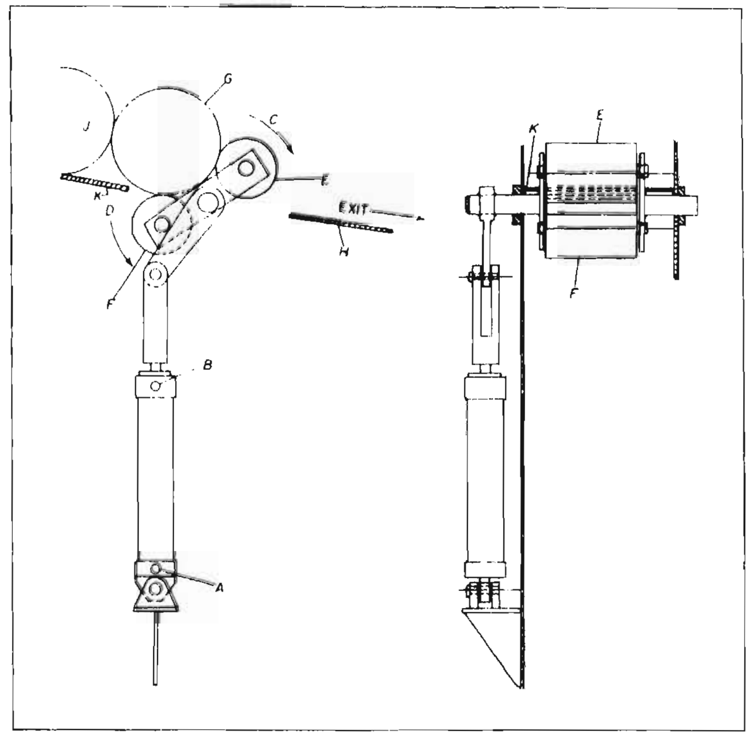


FIG. 2. Cylindrical work parts G from chute K are fed one at a time down ramp H by the rocking of roller cage E-F. The air cylinder has a clevis fastening below port A permitting it to swivel. Flexible air hoses lead to ports A and B.

For the return stroke, compressed air enters port *B*, exhausting air from port *A* and swinging the roller cage back to its original position with cylinder *J* set to be ejected down the line.

Semi-Automatic Feeding Device for Small Headed Parts

For a particular application, it was necessary to form a longitudinal knurl on the shanks of small rivets. One half of the knurling die employed was mounted in a fixed position on the frame of a threading machine, while the other half was mounted on the rotary machine table. It was still necessary to provide a device for introducing a single rivet into the die at each revolution of the table.

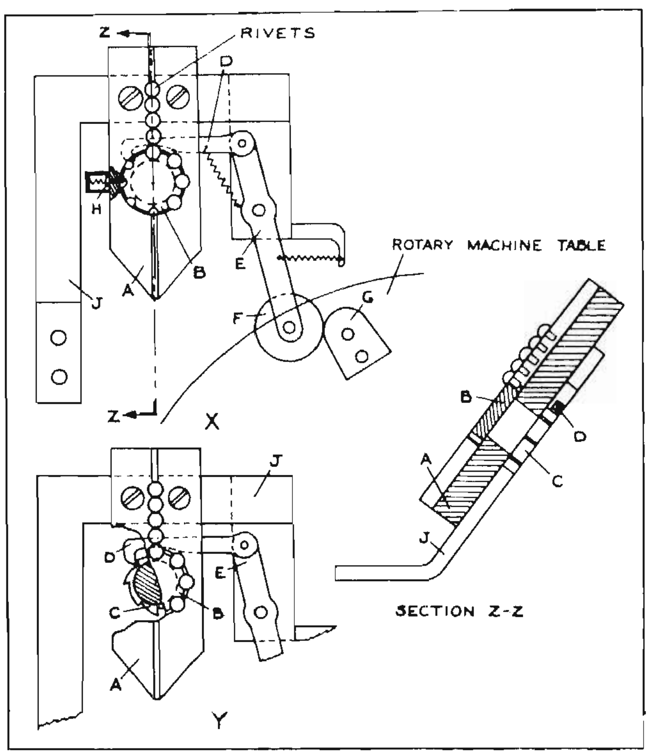


FIG. 3. Feeding device synchronizes rivet flow with rotation of the machine table.

Satisfactory operation was obtained with the feeding device shown in Fig. 3. The main member of the device is central body *A*. A channel, wide enough to accommodate the rivet shanks freely, is milled in the center of the upper surface of the member. With the center of the channel serving as one locating line, a hole is drilled through the body and counterbored to receive transfer wheel *B*, as shown at X in the illustration. Around the periphery of the transfer wheel are machined eight equally spaced slots of a size suitable for carrying the rivet shanks.

A ratchet wheel *C*, which may be seen at Y, is mounted on the underside of the transfer wheel. The remaining parts of the advancing mechanism are ratchet *D*, lever *E*, roller *F*, and actuating finger *G*. Two tension springs are included to insure proper functioning of the lever system. A spring-loaded pawl *H* restricts rotation of the transfer wheel to a clockwise direction. All of these units, with the exception of actuating finger *G*, are mounted on a welded-steel support frame *J*, which is situated at an incline of approximately 35 to 40 degrees from the horizontal. This support frame is bolted directly to the frame of the machine. Actuating finger *G* is screwed to the rotary machine table on which the moving member of the knurling die is mounted.

With the machine functioning, the operator loads the upper portion of the channel in central body *A* with the rivets to be knurled. The rivets are placed with their shanks down as shown in section Z-Z, being supported on the underside of their heads. Normally, four of the eight slots in transfer wheel *B* contain rivets. When actuating finger *G* contacts roller *F*, lever *E* pivots on pin *K*. Ratchet *D* is, in turn, pulled to the right, engaging a tooth on ratchet wheel *C* and rotating the transfer wheel one-eighth of a revolution in a clockwise direction. A rivet is thus aligned with the lower portion of the channel in the central body and, due to the force of gravity, travels downward to the knurling die. As the actuating finger passes by the roller, lever *E* and ratchet *D* are returned to their original position by means of the two tension springs.

Handling Mechanism Turns Strip in Transfer

In processing fiberboard strip for a firelighting device, an interesting materials-transfer mechanism is used. This mechanism picks up the strip as it leaves the saw table, rotates it 90 degrees so that a combustible fluid can be injected into one edge, then rotates it another 90 degrees for ejection.

In Fig. 4, several strips *X* can be seen leaving an extension *A* of the saw table. The strip is advanced manually between raised guides into the fingerlike end of the long leg of a bellcrank *B*.

The bellcrank is keyed to a stud *C* free to revolve on a rectangular slide *D*. Connecting-rod *E*, pivoting at *F*, reciprocates the slide in body casting *G*. The opposite end of the connecting-rod (not shown) is actuated by a conventional eccentric disc. The slide is T-shaped in vertical section, so that it can be retained by keeper plates *H*.

The short leg of the bellcrank forms an angle of 102 degrees with the long leg. At its end, it carries a roller *J* projecting over the front of the body. The roller operates over the upper edge of a guide plate *K* fastened to the front of the body.

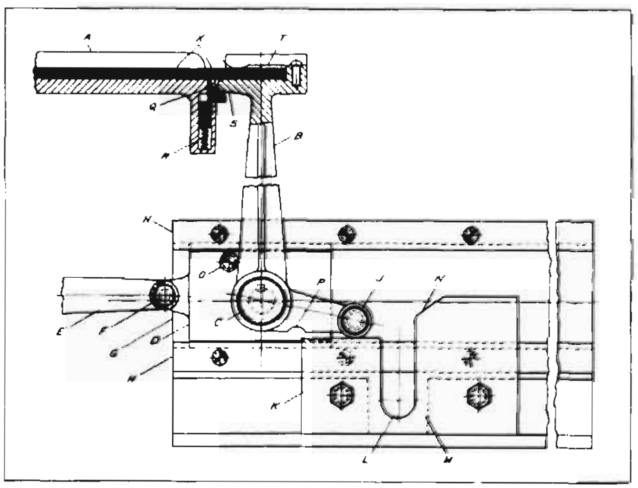


FIG. 4. When bellcrank *B* starts its swing, roller *J* rides over the lower horizontal surface of guide plate *K*.

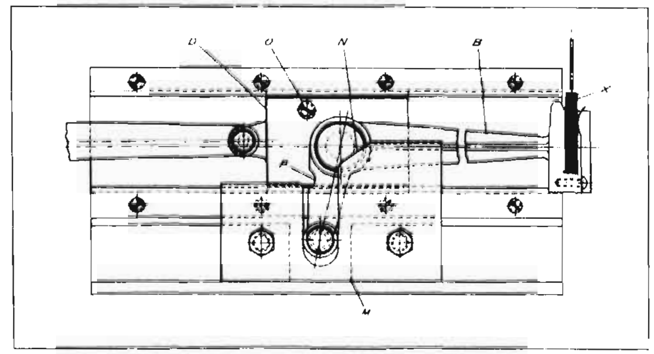


FIG. 5. A momentary dwell of the eccentric disc permits injection of combustible fluid.

When the slide moves to the right, the bellcrank carries along one of the fiber board strips, moving until roller *J* contacts the right-hand wall of slot *L* in the guide plate. The roller then is forced down in the slot, approximately 0.005 inch wider than the roller diameter. Meanwhile the bellcrank, swinging on stud *C*, rotates the strip 90 degrees, Fig. 5. A clearance channel *M* accommodates the short leg of the bellcrank.

Now there is a momentary dwell of the eccentric disc to allow the combustible fluid to be injected. Then, continued movement of the slide in the same direction raises the roller out of the slot, first onto an adjacent 40-degree incline *N*, then onto the higher straight edge of the guide plate. Simultaneously, the strip is rotated downward 90 degrees more, Fig. 6. After the strip is ejected, the slide moves to the left, and the bellcrank returns to its initial position.

Pin *O* (pressed into the slide) offers a positive stop for the bellcrank, bearing against the long leg at the start of the cycle, and against a recess *P* in the short leg when the bellcrank reaches the position shown in Fig. 6. A small, vertical slide *Q*, Fig. 4, prevents the strips remaining on the extension from being pushed off once the long leg is loaded and the bellcrank starts its swing. This slide is in a lip on the extension bottom, and a spring *R* keeps it raised, once the bellcrank swings away, so that the end of the slide slightly intersects the path of the strips.

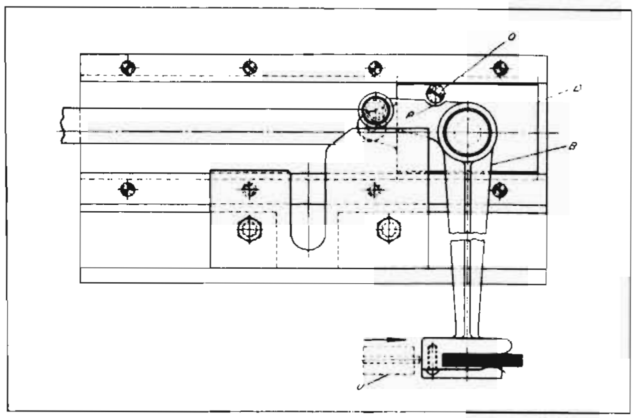


FIG. 6. Recess P in the short leg contacts pin O in rectangular slide D, limiting the movement of bellcrank B.

At the start of the cycle, a projecting surface S of the long leg depresses the slide, and the foremost strip is advanced into the leg. There, the strip is retained by a leaf spring T. To release the strip when it has reached the position shown in Fig 6, a forked ejector plate U is actuated by separate mechanical means at the proper instant.

Mechanism Simultaneously Transfers and Reverses Position of Pad

In the processing of paper pads stuffed with excelsior or shredded paper, they have to be transferred a distance of 36 inches between work stations. The pads, which are rectangular in shape, are picked up by a cam-tripped gripper device (not described) along a folded-over bottom edge. During the transfer, the pad has to be reversed so that the leading edge becomes the trailing edge. The diagram, Fig. 7, shows pad positions during the transfer. It will be noted that the locus of the pad center remains along the line A-A.

The transfer mechanism combines simplicity and smooth operation. Drive-shaft B, Fig. 8, in frame C, rotates at a constant speed of 40 rpm, affecting one transfer per cycle. Lever D, keyed to the drive-shaft, contains at its other end free pin E.

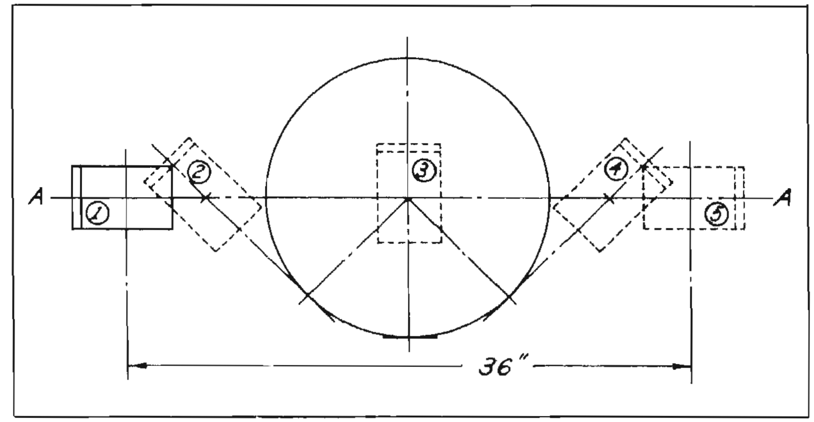


FIG. 7. The transfer mechanism reverses the pad as its center moves along line A-A.

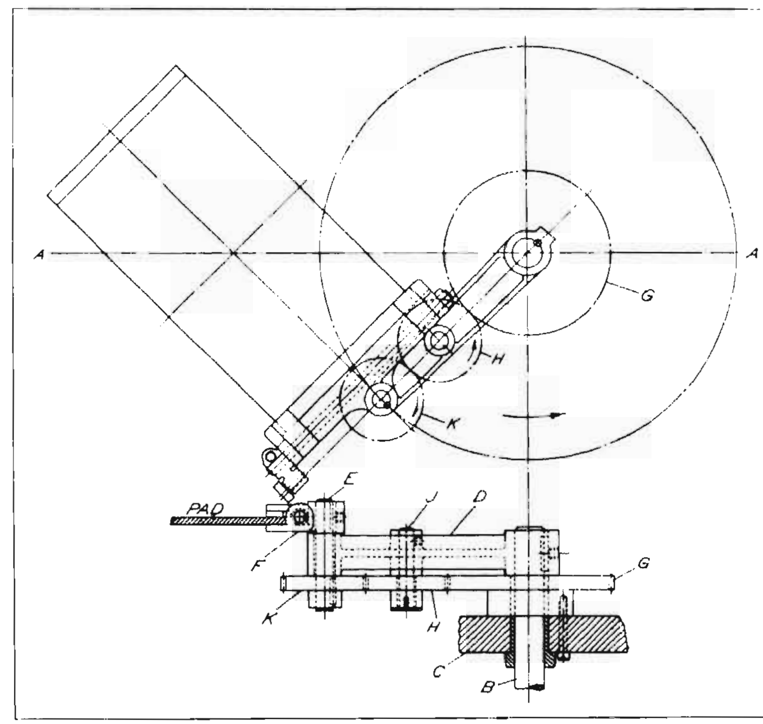


FIG. 8. The counterclockwise movement of lever D and the clockwise movement of gear K act to reverse the pad in its transfer along line A-A.

The gripper device *F* is keyed to the top of the pin. The distances from the pin center to the center of the pad and the center of the drive-shaft are equal.

For the reversal of the pad during the transfer, three gears are provided: gear *G*, having 192 teeth, is fixed to the frame; gear *H*, an idler, is free on stud *J* carried by the lever; and gear *K*, having 96 teeth, is keyed to the bottom of pin *E*.

During the cycle, lever *D* rotates counterclockwise at the drive-shaft speed of 40 rpm. Simultaneously, gear *K* revolves clockwise on its axis. In 180 degrees of movement of the lever, gear *K* has also moved 180 degrees, but in a reverse direction. The result: the leading edge of the pad in Position 1, Fig. 7, becomes the trailing edge in Position 5.

Transfer Device for Cylindrical Parts

Transfer of cylindrical parts from one work station to another can be accomplished by the arrangement seen in Fig. 9. In the particular application shown, a collet-feeding setup introduces

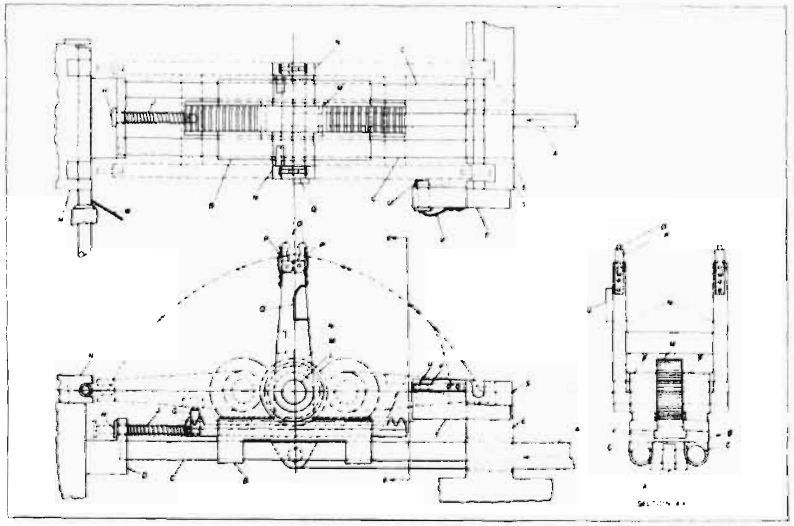


FIG. 9. A mechanism for transferring cylindrical parts from one work-station to another. Drive-bar *A* is given a variable reciprocating motion by a cam.

the work into the first station in the form of tubing. After the tubing is cut to length at the required angle by a slitting saw, the resulting work-piece is picked up by transfer arms which rotate 180 degrees and deposit it at the second station. A cam-operated drive activates the mechanism.

The transfer mechanism is illustrated at the mid-point of its cycle. Drive-bar *A*, which is given a variable reciprocating motion by a cam (not shown), transmits this movement to a slide-block *B* that slides on two rods *C*. These rods, in turn, are supported at the ends by members *D* and *E*. Slide-block *B* is grooved to accommodate a sliding rack *F*, which has a pin *G* and a headed stud *H* at one end. Stud *H* passes through a hole in a projection on member *D* and carries a spring *J*. This spring exerts pressure on the rack, thus tending to hold the head of the stud in contact with member *D*.

Bearings *K*, the bases of which retain the rack in block *B*, support a shaft *L*. Keyed to this shaft is a gear *M* that meshes with the rack. In addition, two transfer arms *N*, connected by a bar, are keyed in alignment on the shaft. Gripping jaws *O*, located on the end of each arm, are held in the closed position by springs *P*. One of the transfer arms carries a small block *Q*.

Member *R* is grooved to guide and back up the entering tubing, and a receiving block *S*, mounted on member *E*, is grooved on the top to accept the severed piece of tubing. Another block *T*, also mounted on member *E*, carries a latch-pin *U*, which is held under tension by a spring *V*.

In operation, drive-bar *A* moves to the left from the center position, carrying slide-block *B*. Since gear *M* is supported on block *B*, and rack *F* cannot move with it due to the resistance of spring *J*, gear *M* is caused to rotate counterclockwise in mesh with rack *F*. Thus, movement continues until the bar connecting the transfer arms *N* contacts pin *G*, as shown in the phantom view of the arms at the left. Pin *G* is so adjusted that arms *N* are then in the horizontal position. At this point in the cycle, there can be no relative motion between block *B* and rack *F*.

Continued movement of block *B* causes spring *J* to compress and the entire assembly to move as a unit farther to the left until the gripper jaws are forced over the tube and it is held under the tension of springs *P*. The driving cam then provides a rest period while the saw *W* cuts the required length from the tubing.

After the saw has completed its operation, bar *A* is reversed and the transfer arms *N* withdraw the work-piece horizontally from guide block *R*. This movement continues until the head of stud *H* again contacts member *D*, at which time rack *F* can no longer move to the right. But as block *B* moves farther in the same direction gear *M* rotates clockwise in mesh with rack *F* until the transfer arms have revolved 180 degrees and deposited the tube in the groove in block *S*. After the transfer arms reach the horizontal position, latch-pin *U* engages the upper side of block *Q*.

The cam then reverses drive-bar *A*, causing the section of tubing to be stripped from between the gripper jaws *O*. This movement of the transfer arms takes place horizontally because they are held in this plant by latch *U*. When the necessary horizontal travel has been completed, block *Q* has moved so that it is no longer under the latch-pin. During this motion, spring *J* is compressed by rack *F*, and in order to prevent sudden expansion of the spring, the upper edge of block *Q* can be shaped to permit a more gradual return. The transfer arms then rotate back to the central position and the cycle is repeated. A plunger (not shown) removes the part from block *S*.

Magazine Feeds Wrappers at Constant Pressure

Wrappers or labels are fed from the top of the magazine shown in Fig. 10. The edge of the wrapper is elevated by vacuum lifter *A* so that it is in position to be picked up by gripper *B* and pulled clear of the magazine.

The most interesting feature of this device is that it maintains a constant pressure between the top of stacked wrappers *C* and the underside of stop-plates *D* and *E*. This is accomplished by means of a rack *F*, pinion *G*, and scroll *H*.

The various parts will assume the positions shown when the magazine is fully loaded. Weight of the wrappers or labels resting on pressure-plate *J* is offset by the magnitude of weights *K*, taking into consideration their distance from the rotational axis of pinion *G*. These weights are supported by a strap passing around the outer surface of the scroll.

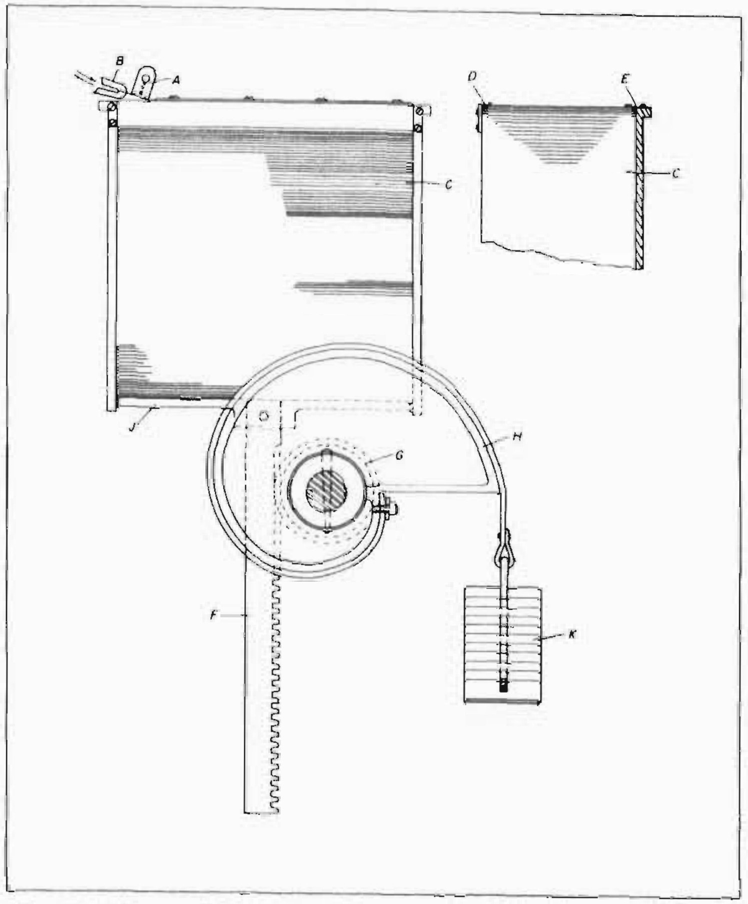


FIG. 10. Rotating scroll *H* alters the moment of the force applied at pinion *G*, thereby reducing upward pressure on the stack as the magazine empties.

As the wrappers are removed from the stack and the magazine begins to empty, the weight resting on pressure-plate *J* diminishes. At the same time, the scroll has been rotating in a clockwise direction. Due to the contour of the scroll, the lever arm of the weight becomes progressively shorter and the total force applied to the rack and pressure-plate is reduced proportionally. In this way, the wrappers are fed upward under an almost-constant pressure from the first to the last.

Materials fed by this system can be paper, foil, light cardboard, or almost any sheet stock. In most cases, only stacked weights K need be made lighter or heavier to accommodate the different materials.

Assembly Operation Mechanized

A rivet is used as an electrical contact on the end of a formed sheet-brass spring, Fig. 11, made in volume. Until recently the contact rivets were manually assembled into the hole in the spring blade and headed under an air hammer. The manual operation was slow and tedious for the operators because of the 0.001-inch clearance of the rivet in the hole.

More recently the assembly has been made semiautomatic, with the operator's duties now restricted to stacking the properly oriented spring blades in a magazine slide and occasionally dumping a boxful of rivets into a Syntron vibratory hopper feeder. The successful assembly mechanism was comparatively simple to build.

The spring blades are automatically placed over the rivet shanks by a pick-off transfer from a Ferris wheel drum, Fig. 12. The main drive is shaft A, which rotates constantly at 100 rpm in a bearing through backing plate C. Keyed to shaft A, but be-

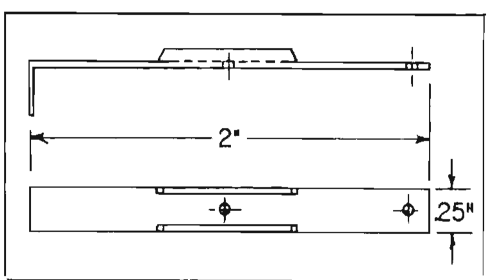


FIG. 11. Work-piece is a formed brass spring. The rivet hole is on the right, and the guide hole in the center.

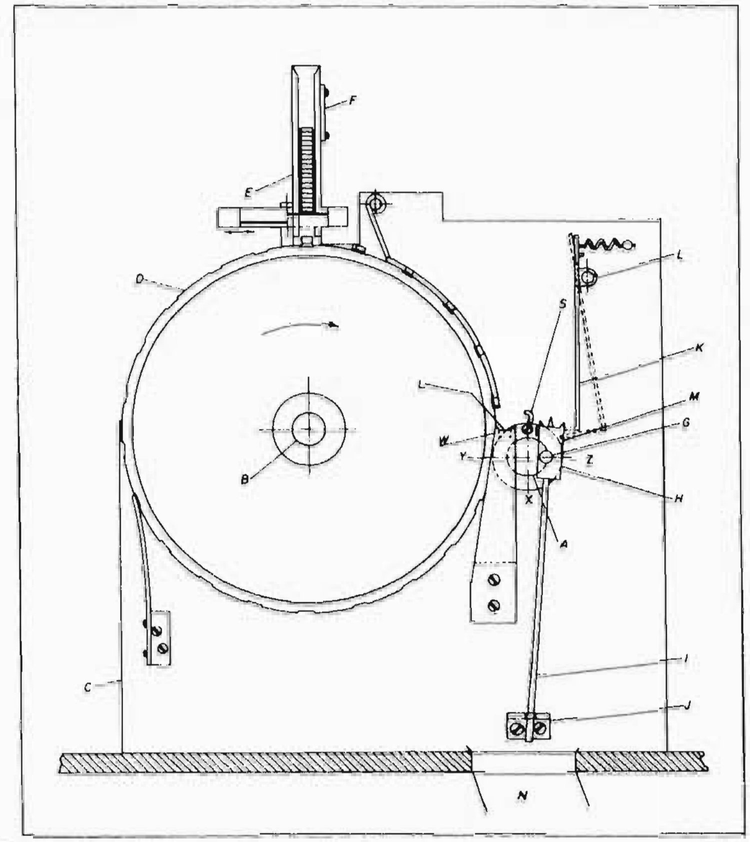


FIG. 12. Positioning and assembly drum indexes by ratchet from shaft A. The riveting anvil is contacted with the pick-off H in Z position, as shown, but is not included in this diagram.

hind plate C, is a cam which, through a lever and pawl linkage, drives a ratchet wheel turning shaft B. The rotation of drum D is thus intermittent, and each of its twenty-two stop indexes per rotation is aligned with the springs that drop, one at a time, from magazine F. The springs lie in the carrier grooves with the bent tab (Fig. 11) pointing upward.

As the drum indexes, the work is held in place by a retaining strap, or guide, that wraps around the outside of the drum. The spring rolls out of the groove, landing horizontal, with the tab

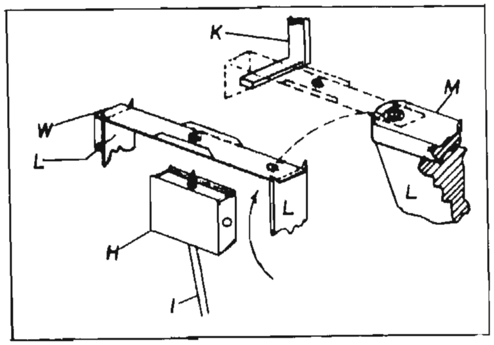


FIG. 13. Lift, carry, and mating steps in the operation of the machine are automatic, with the pick-off H rising between the prongs of dwell cradle L.

pointing down, on the two arms of a dwell cradle L that straddles the drum at this point, Fig. 13.

The spring, as it rests in the dwell cradle, is now positioned for transfer and placement over the shank of the rivet that will be waiting on the anvil M, as in Fig. 14. Transfer is accomplished by a pick-off H, Figs. 12 and 13. The pick-off is a block that hangs on crankpin G, mounted on the end of shaft A. The crankpin rotates once for each index of feed-drum D, carrying the pick-off with it. Stabilizer bar I in the bottom of the pick-off block holds it upright by virtue of a loose fit through a hole in bracket J.

The top of pick-off block H has a groove the width of the spring, and having sloping sides to facilitate seating of the part

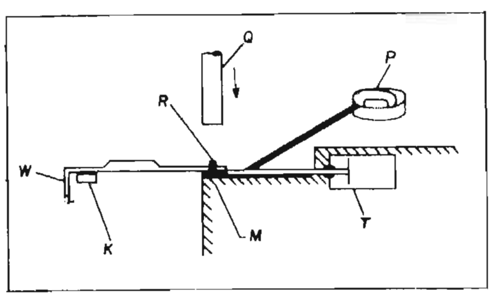


FIG. 14. Mating of spring W with hole finding the rivet R on anvil M. Air cylinder T delivers rivet from track of vibratory hopper P. Hammer Q upsets rivet shank.

by gravity. Near the center of the groove is a pointed stud that "finds" the guide hole in the spring waiting in the dwell cradle.

As the crankpin carrying the pick-off travels upward from position X, Fig. 12, the spring is lifted from the dwell cradle at Y, with the stud through the guide hole and resting in the groove of the pick-off, Fig. 13. As the crank continues to turn, the spring is carried over to riveting position Z, Fig. 12, where the pick-off falls away, leaving the spring with the rivet assembled through the spring's rivet hole and supported by the anvil. The opposite end of the spring sits on rest lever K, Fig. 13.

A rivet, meantime, has been positioned by the arrangement in Fig. 13, which also shows the riveting setup. In Fig. 14, rivet R from hopper P is pushed by air cylinder T into position on anvil M. Air hammer Q upsets a head on the rivet. The hammer is actuated by a micro switch from a cam on shaft A, Fig. 12.

As soon as the rivet has been headed, a knockoff finger S on shaft A swings past, knocking support K out from under the work so that it drops into the discharge chute N, Fig. 12.

Rotary Work-Transfer Device

An arrangement for transferring work-pieces from a horizontal conveyor to a station for vertical stacking is shown in Fig. 15. The parts, in this case papier-mache egg trays, are each conveyed by one of six pivoting carriers borne on a member rotating at constant speed. An interesting feature of the mechanism is the employment of a stationary cam to control the pivoting motion of the carriers. This device replaced a high-speed oscillating movement.

In construction, the mechanism consists of a rotatable member A mounted on a shaft B supported in fixed bearings C. The six carriers D are supported by and pivoted on shafts E mounted in bearings F secured to the periphery of member A. A roller follower G, guided by the large stationary plate-cam H, controls the movement of each shaft E by a connecting arm J.

Thus, as member A is rotated at constant speed by an appropriate drive, the carriers are each brought by the contour of the cam into the proper position for lifting the work-pieces

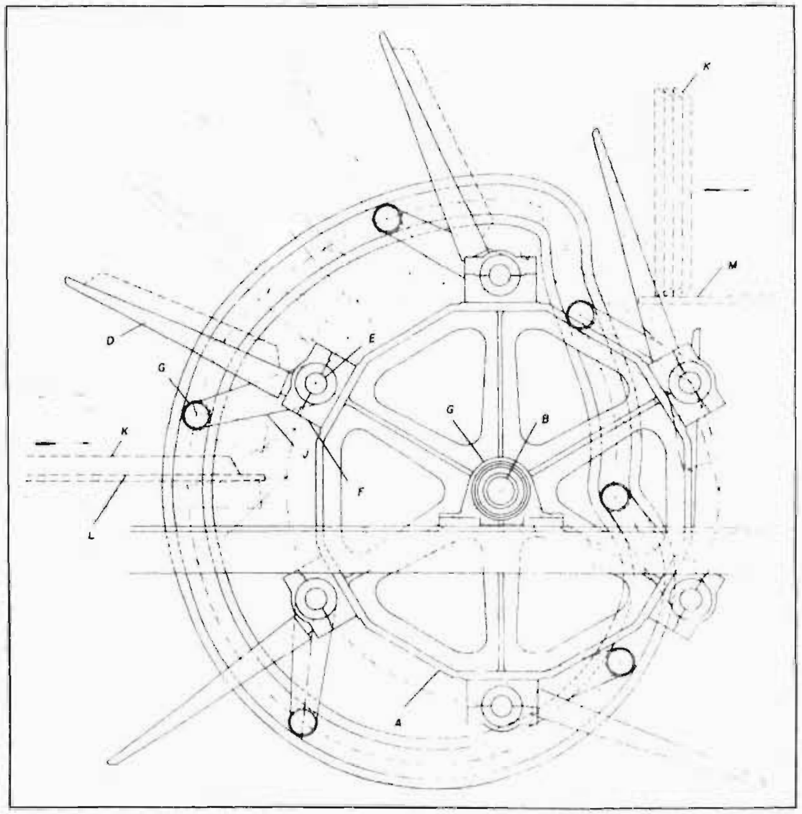


FIG. 15. Work-transfer device featuring stationary cam control of rotating carriers.

K from the conveyor station *L* and depositing them in a vertical position for stacking at station *M*. To assure smooth operation, the drive speed is adjusted to synchronize the carriers with the arrival of the work-pieces at the conveyor station *L*, which should be at a constant interval.

Semi-Automatic Work Feeder Improves Efficiency of Thread Roller

The manual feeding of work-pieces to reciprocating thread rollers has been hazardous both to the operator and the machine. Operators have required considerable skill plus precision

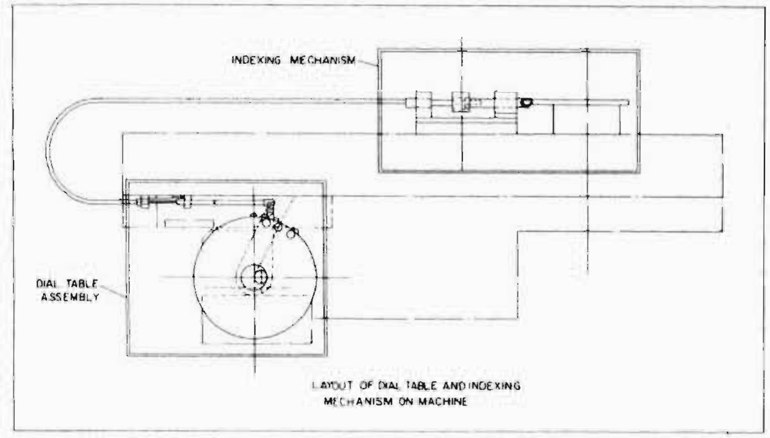


FIG. 16. Thread-rolling machine works on the principle of the automobile choke rod, with the dial-feed mechanism moved by the indexing device.

of hand motion. Careless handling of parts can damage the machine.

Using the attachment (Fig. 16) described and pictured in the illustrations, certain types of straight-shank blanks can now be fed between thread-rolling dies to make the operator's job simply a matter of dropping the work-blanks into holes provided in a dial table. The dial then carries the parts around to loading position above the dies. The construction of the thread roller made necessary the following design for an indexing drive.

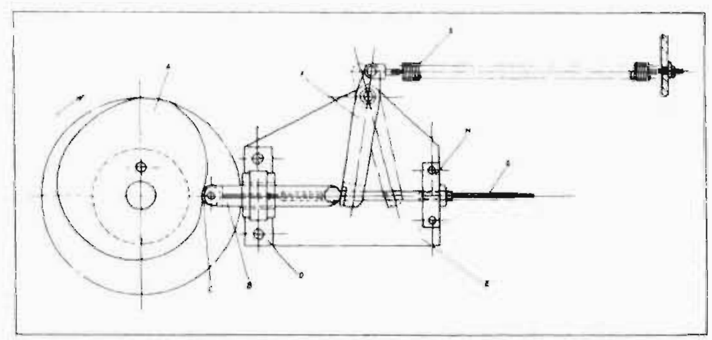


FIG. 17. The indexing mechanism for the thread-roller work feed extends spring *S* which actually drives the dial.

Motion for indexing the dial feed (Fig. 17) is from a cam A mounted on the crankshaft of the thread roller. The roller follower C transmits linear motion to plunger B, which in turn is linked to the center wire of cable G. Alignment is provided by guides D and H on plate E. Also mounted on plate E is lever F, one end of which has a carbide wear strip that is spring-loaded against plunger B under tension from coil spring S. The casing of cable G is anchored to plate E in guide H.

From the crankshaft cam the flexible rod makes a 180-degree bend to the top of the machine, where it actuates the dial feed, Fig. 18. Here the cable casing is secured on block I with adjusting screw and lock J, provided for adjusting the length of the casing with relation to the inner wire, and thereby controlling the stroke of arm X.

The dial-feed assembly consists of rotating dial K, which has thirty equally spaced holes near its outer diameter. Disc L,

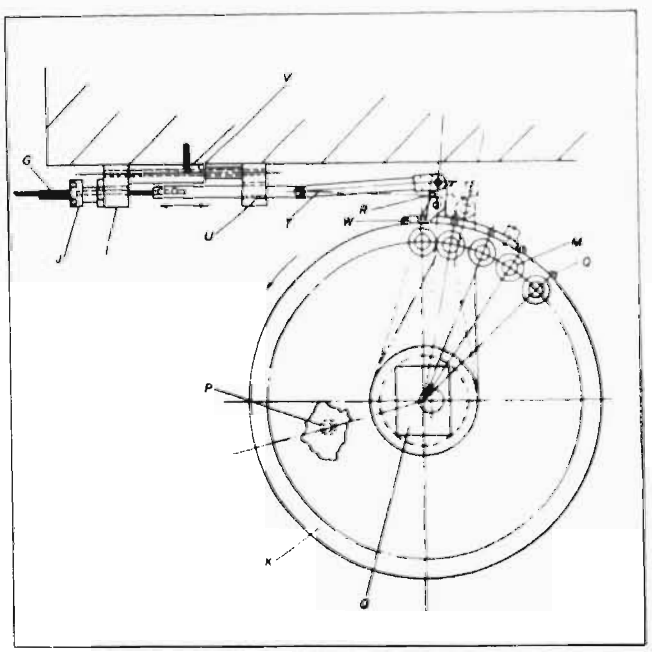


FIG. 18. The dial table feeds blanks to the dies at point M, with the other twenty-nine holes open for loading by the operator.

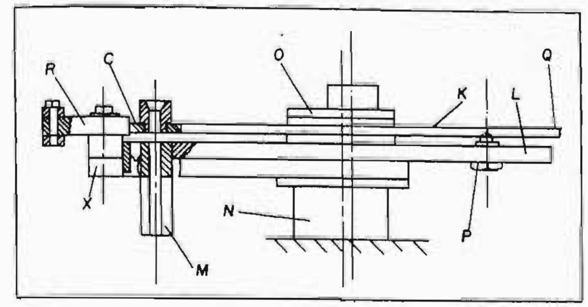


FIG. 19. Elevation view of the dial feed shows feedtube M in fixed disc L beneath the movable dial K.

Fig. 19, is a stationary support for dial K and has just one hole through which feed-tube M leads to the dies at the loading position.

Both dial K and disc L are mounted on column N, which has within it an eccentric bushing O for the dial shift. By turning the eccentric during setup, the position of the feed tube can be varied with relation to the dies to handle a variety of work diameters. Stop location of work-carrying dial K is set by an annular row of thirty spherical recesses on its underside. There is a spring-loaded spherical plunger P on the top side of fixed plate L. The positioning of the recesses is so oriented with relation to the feed tube M that when the plunger shoots home in any recess, one of the thirty work-carrier holes in the bottom of moving dial K is always aligned with feed tube M.

On the periphery of moving dial K, Fig. 18, and opposite each work-carrier hole, is a socket-head cap-screw, as at Q, which functions as a ratchet tooth for the indexing motion. The arm, swinging on shaft O, carries ratchet pawl R and at its free end has a linkage for the clevis and link rod T, which in turn fastens to the cable wire G. Linkage rod T is hinged.

A table stop is provided by lever V, Fig. 20, which has a slot that will accept inner cable wire G. When lever V is raised, rod T can reciprocate a full stroke, but when it is down on the wire its length holds rod T at its full stroke position against the spring S (Fig. 17).

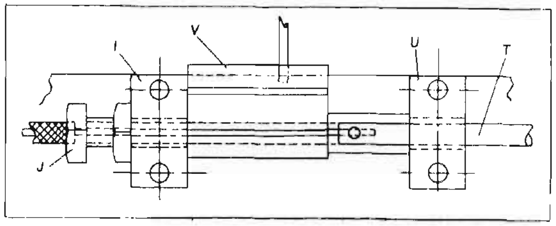


FIG. 20. The indexing stop lever V swings down over the cable wire to hold linkage bar T in extended position.

Indexing table K also has an overshoot control (Fig. 21), which is curved lever W, pivoting on the edge of fixed disc L. The free end rests on indexing arm X. As the indexing arm swings through its arc the contoured bottom of stop lever W causes the far end to rise and fall. The top of its upstroke coincides with the furthest forward motion of the index-arm X and the seating of regular stop positioner P. Thus it catches on its "hook" one of the indexing capscrews Q in a positive mechanical lock.

Straight-Line Motion Through Levers

In processing clutch housings on a transfer machine, movement of the work from station to station is accomplished by the finger type mechanism shown in Fig. 22. Features include simplicity of design, compactness, and a straight-line movement of the work.

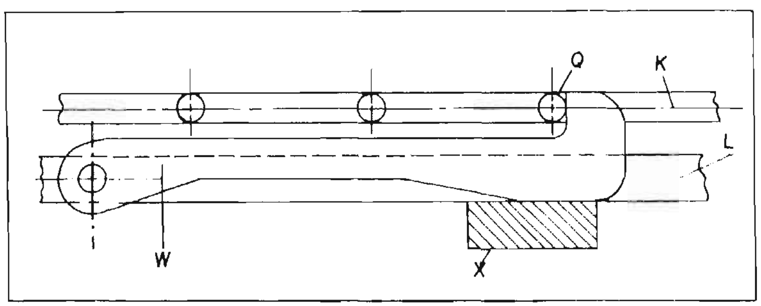


FIG. 21. Table overshoot control lever W has a hooked end that rises to stop bolt head Q when lifted by arm X.

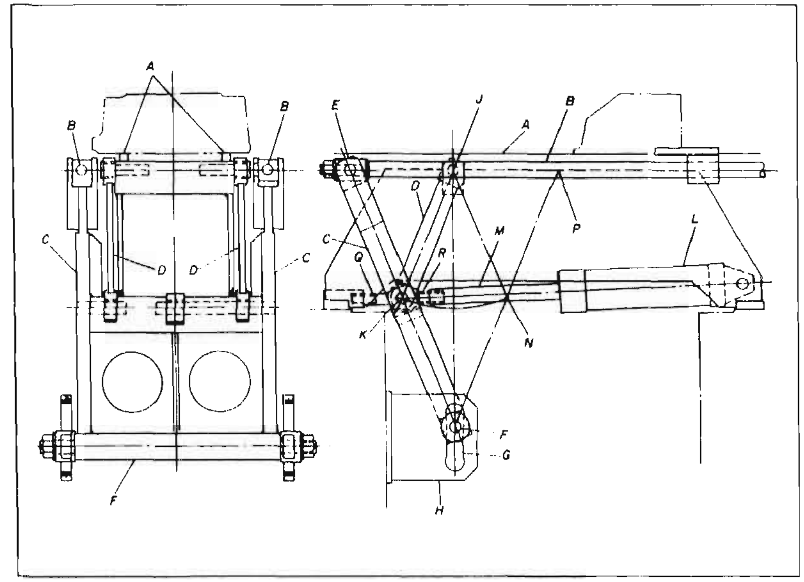


FIG. 22. While the work is indexed in a straight line along rails A, shaft F, attached to transfer levers C, falls and then rises in slots G.

Rails A support the housings. Along each side are a transfer bar B, transfer lever C, and link D. Trunnion blocks E at the top of the levers join them to the bars. The bottoms of the two levers are attached to a common shaft F which is contained at each end in a bushing in a slot G of a bracket H fixed to the frame of the device.

The tops of the two links D are fixed to the frame, by pins J, which are on the center line of slots G. At their bottoms, the links are hinged to the centers of the levers by pins K. Links are one-half as long as the levers, with the distance between K and J equal to the distance between K and the center of trunnion block E.

To operate the mechanism, air cylinder L powers connecting-rod M, attached to the levers at K. (The rod could be attached elsewhere along the levers, its position depending on the desired ratio of piston stroke to transfer movement.)

When the connecting-rod pulls the levers, pins *K* move through an arc to point *N*, causing shaft *F* to fall and then rise in slots *G* as transfer bars *B* travel in a straight line to point *P*. Accurate limits to the indexing stroke are provided by stop-screw *Q* and by threaded clevis *R* on the end of the connecting-rod.

Feed System for a Deep-Hole Drilling Machine

A patented feed system that enables holes to be produced in a deep-hole drilling machine in several stages during an automatic cycle is shown in Fig. 23. At the end of each stage the drill head is automatically withdrawn from the work for clearing of chips. Then, the head is returned to the required drilling position under rapid power traverse.

The feed and rapid traverse motions of the drill head are derived, respectively, from motors *A* and *B*, which drive a screw *C* through gearing. Upon completion of each drilling stage, the feed motor *A* is reversed, and, simultaneously, the rapid-traverse motor *B* is brought into operation by the action of a torque control system which is not shown. As a result, the drill head is moved away from the work under rapid traverse. At the same time, an electromagnetic type clutch *D* is brought into engagement. This causes cam *E* to be driven by means of screw *C* through suitable worm gearing.

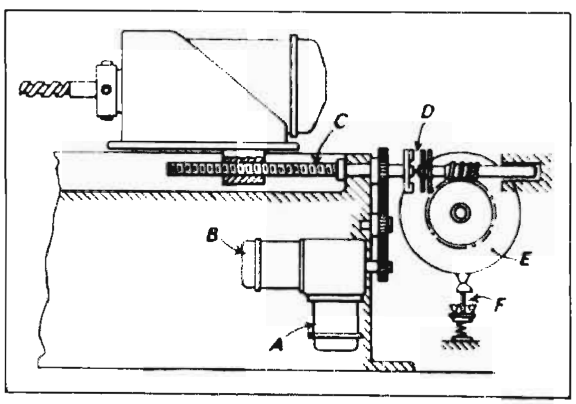


FIG. 23. Feed arrangement for deep-hole drilling that intermittently clears chips.

At the end of the rapid-traverse motion of the drill head, the feed motor *A* is stopped and the motor *B* is reversed, by a means not shown. In consequence the drill head is rapidly traversed towards the work and cam *E* is driven in the opposite direction. When cam *E* has been returned to its original angular position, it operates a switch *F*, with the result that the rapid-traverse motor *B* is stopped, and clutch *D* disengaged. At this point in the cycle, feed motor *A* is again started in order to perform the next stage of the required drilling operation.

Because feed motor *A* is not running during the rapid traverse of the drill head toward the work, when motor *B* is stopped a small clearance exists between the end of the drill and the bottom of the previously drilled portion of the hole in the workpiece. As a result, the risk of drill breakage due to a slight overtravel of the head during the rapid-traverse motion is reduced.

Automatic Feed Mechanism with Quick-Return Motion

Short metal tubes of various lengths and diameters are polished and buffed on centerless grinding machines. In handling this work on the standard type of centerless grinder, the operator inserts an unpolished tube with his right hand, and as soon as the automatic feed is in action, pushes a handle to actuate the feed-belt pusher. After the tube is polished, the operator inserts a wooden stick in it with his left hand to remove it from the machine. In order to provide for automatically feeding the work to the machine and ejecting it, thus eliminating the need for constant attention by the operator, the regulating wheel was replaced by an endless belt and the automatic feed mechanism shown in the illustration was developed. With this arrangement, the operator merely needs to keep the hopper loaded.

The hopper *A* (see Fig. 24), section *X–X*, is designed for tubes *W*, 1½ and 1¼ inches in diameter, which are loaded parallel to each other. A hinged bottom *B* is swung on its pivot by an eccentric *D*, below *B*, which receives its rotary motion from a

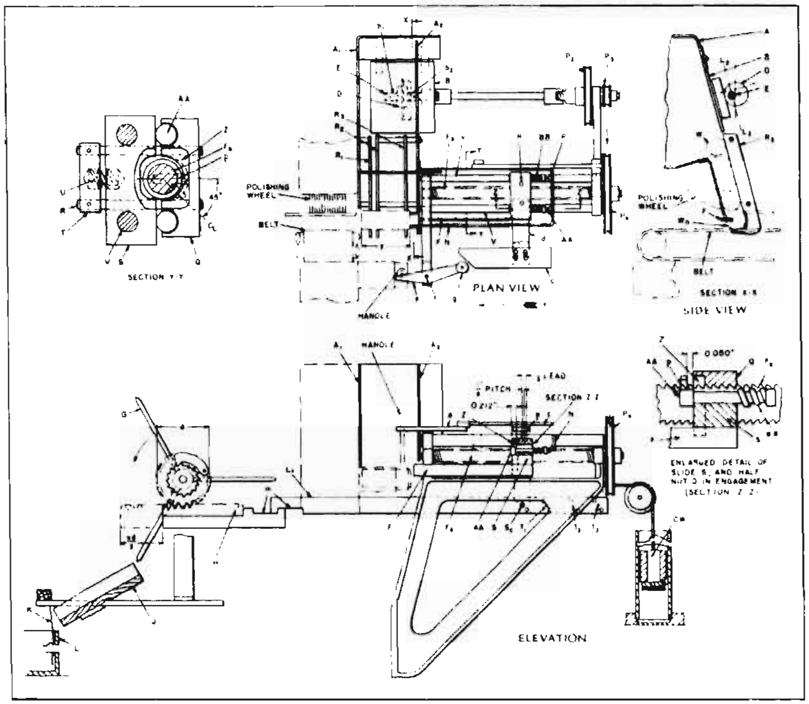


FIG. 24. Feed mechanism used with centerless grinder for automatically loading and unloading tubes during a polishing operation.

constantly rotating shaft E. This shaft is supported in bearings b_1 and b_2 (see plan view), which are adjustable in the direction indicated by line L_2–L_3 (section X–X), so that the amount of movement of hopper bottom B can be regulated. Shaft E is driven by pulley P_2. Pulley P_3, driven by P_2 drives P_4 which drives feed screw F_s.

Referring to the plan view and to the elevation, A_1 is that part of the hopper which is fastened to the machine and has a rail R_1; A_2 is an adjustable section of the hopper, designed to accommodate any length y of tube W, and has rails R_2 and R_3 on which the work rolls into position W_n.

The adjustable wall A_2 has an extension N provided with two stop-buttons P which limit the return stroke of slide S (section Y–Y) as will be described. The slide mechanism, which has a

pusher F, also includes a half-nut Q into which two pins R are pressed. These pins fit loosely in holes through slide S and are fastened to a spring support T. A spring U keeps half-nut Q and slide S in contact, or closed, in which position Q is engaged with feed-screw F_s. The right-hand buttress thread of the feed-screw feeds slide S in the direction of arrow Y, on fixed rods V, when the half-nut and feed-screw are engaged. When they are disengaged, as shown in section Y–Y, where slide S and half-nut Q are separated, slide S returns to the starting position by means of the pull provided by counterweight CW, which is essentially a dashpot, since its outer casing is partly filled with oil.

The disengagement of the half-nut Q and the feed-screw for the return stroke is accomplished by means of pin p in feed-screw F_s, which enters a space Z milled in the half-nut. When members Q and F_s are engaged, pin p is at line C_L, shown in dot-and-dash lines in section Y–Y, and a clearance of 0.050 inch exists between the pin and the front of the half-nut, as shown in the enlarged detail, section Z–Z. The lead of the feed-screw is $\frac{1}{3}$ inch per revolution; hence, as the feed-screw turns through 270 degrees, or three-quarters of a revolution, the nut will advance $\frac{3}{4}\ (\frac{1}{3} - 0.05) = 0.212$ inch.

At this point, the pin will have entered the half-nut a distance of 0.212 inch, as shown at section Z–Z in the front elevation, and will have raised it against spring U to separate it from slide S and the feed-screw. The distance the half-nut is raised is sufficient to clear the heads of bolts AA from slide S so that two springs BB can pull them along the top of the slide, as will be understood by reference to sections Y–Y and Z–Z. The two parts remain disengaged while the bolt heads rest on top of slide S, instead of being engaged with it, as shown in the enlarged detail Z–Z. The counterweight CW then pulls the slide (and the half-nut) in the return-stroke direction.

Since the amount of weight CW is adjusted to overcome friction plus the compression of springs BB for the length of the heads of bolts AA, when the ends of the bolts strike stops P, springs BB are compressed and the bolts back up so that their heads drop down into engagement again with slide S. Spring U

then exerts pressure against slide S, so that the half-nut and feed-screw engage again, thereby repeating the reciprocating motion. The working area between these elements of the feed mechanism is small, but it is sufficient for the light work it is required to do.

This reciprocating movement of any predetermined length y is repeated automatically, as long as the feed-screw rotates. Incidentally, this mechanism also acts as an automatic timer, the time duration of the cycles being proportional to the set length y.

As shown in the plan view, the hopper is set up for a certain length y. If a different length of tube is to be polished with the same equipment, the adjustable section A_2 is reset to the new length y and a screw S_c (front elevation) is reset to any of the tapped holes T_1, T_2, etc. The two holes on each side of S_c or T_1, etc., are for dowel-pins which are not removed.

A ratchet wheel, mounted on a pinion, carries three metal rods G that pick up polished tubes and carry them to a chute J, from which they slide into a receptacle L after passing a cloth apron K. A connecting arm L_1 on slide S engages a rack H during the forward stroke, after an idle period provided by open sections m. This idle movement is used to obtain a fast return stroke while the counterweight CW falls freely before reaching the oil cushion.

On the forward stroke, arm L_1 moves the rack a distance $\frac{\pi d}{3}$ (since the pitch diameter of the pinion is d) to position it for turning the ratchet wheel 120 degrees on the return stroke. During this forward movement, the ratchet wheel and rods or arms G are stationary, due to the action of a pawl that pivots on the pinion and slides over the teeth of the ratchet wheel. This permits the automatic pick-up of a polished tube on one of the stationary arms.

The first part of the return stroke is fast, as previously mentioned, and amounts to the idle motion of the rack plus 55 degrees of the rotation of arms G, when the pawl engages the teeth of the ratchet; the remaining 65 degrees is slowed up by the cushioning effect of the dashpot.

A cam c is fastened to a flat spring d, see plan view; the latter, in turn, is fastened to slide S. A rocker arm e swivels on a fixed pivot f. It has a roller g at one end and pushes the machine handle at the other end. When the slide moves in the direction of arrow Y, cam c exerts pressure on the handle to apply pressure to the belt. The amount of pressure is adjustable by means of two elongated slots in cam c.

CHAPTER 16

Feeding and Ejecting Mechanisms for Power Presses

The use of a properly designed feeding and ejecting mechanism is an important factor in power press operation for maintaining a low percentage of spoiled work and a relatively high production rate. The mechanisms described in this chapter were designed for a wide variety of press functions.

Other similar feeding and ejecting mechanisms will be found in Chapter 16, Volumes I and III and Chapter 15, Volume II of "Ingenious Mechanisms for Designers and Inventors."

Mechanism for Rotary Positioning of Work-Transfer Arm

In order to reduce costs in two stamping operations on a sheet-metal part, existing dies were mounted in tandem on a mechanical press. It was intended that the operator load the work manually in the first die and that the part be transferred automatically to the second die. This transferral is accomplished by the mechanism shown in Fig. 1. The parts are ejected from the second die by air.

The feature of the mechanism is a swinging arm *A* which incorporates a vacuum system. The arm swings through 120 degrees in transferring a part from the first die to the second. In addition, the arm must lower a slight amount on the die to pick up the work, rise to a height that will insure clearance of the work over the two dies, lower a second time to place the part in the second die and again rise to the swinging position. Figure 1 shows the arm in the neutral position it occupies when the ram is at the bottom of its stroke.

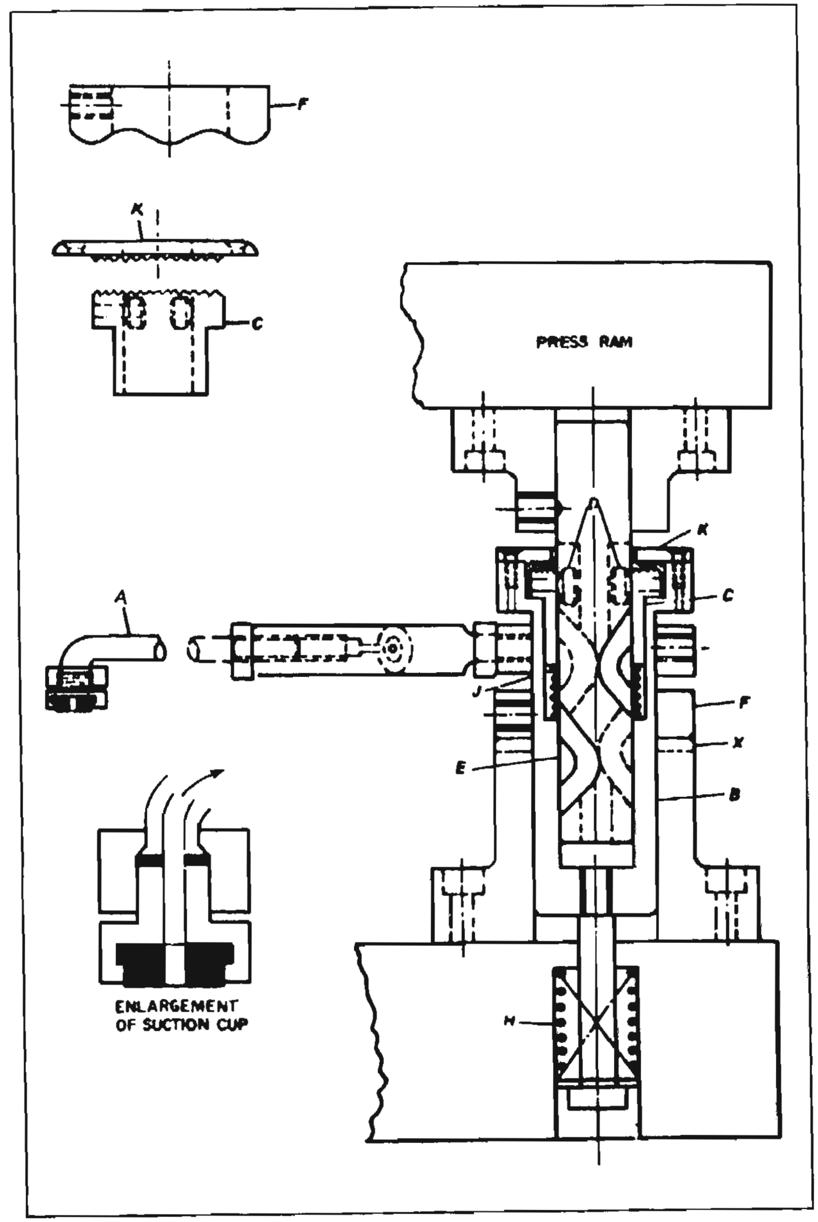


FIG. 1. Swiveling vacuum arm designed for automatically transferring parts from one die to a second die.

When the transfer arm is lowered on the work in the first die, a port is opened to the vacuum system so that the work is picked up by suction. After the arm has lowered the work into the second die, the port is closed to shut off the vacuum and release the work. Operation of the port is controlled by a solenoid valve that is actuated through electrical switches. The transfer arm is constructed of tubing and is adjustable as to length.

The swinging arm is attached to cylinder *B* which can swivel in its housing. Within the upper portion of the cylinder is a floating bushing *C* which carries two rollers *D*. These rollers engage bayonet-like grooves in shaft *E*. As shaft *E* moves up with the operation of the ram, rollers *D* follow the convolutions of the grooves and thus cause arm *A* to index between the two dies.

Raising and lowering of the transfer arm is effected through face cam *F* as it rides over the cam surface *X* on top of the cylinder housing. Cam *F* is fastened to cylinder *B* and turns with it. Tension spring *H* in the press bolster keeps the cam surfaces in close contact with each other.

There is no down movement of the transfer arm *A* during the down stroke of the press because the pressure of the grooves on rollers *D* in shaft *E* compresses the light spring *J*. This causes disengagement of the teeth in clutch plate *K* from mating teeth on the top surface of bushing *C*. During this disengagement there is no swiveling of cylinder *B*.

Vacuum Pickup Extender Obviates Lifting Heavy Stacks

Most vacuum pickups for sheet stock work nicely as long as the stock pile is full. However, when the stack of sheets starts to decrease in height, provision must be made to jack up the stock table so that the vacuum cups can continue to reach the top sheet in the stack.

A simple solution to the problem is the pickup extension device shown in Fig. 2. The shank of the vacuum cup moves downward automatically as the pile is used by the press. The

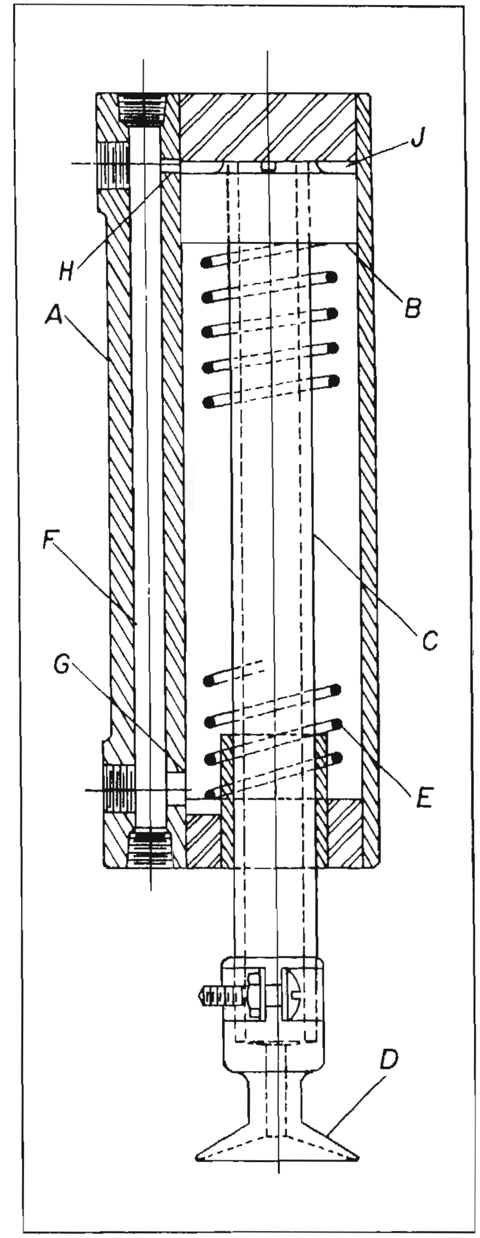


FIG. 2. Vacuum lifter grabs sheet on contact, and will handle the whole of a big stack.

use of these devices allows piles of sheet material over 4 inches high to be fed into presses without lifting the stack.

The lift consists principally of cylinder *A*, piston *B*, hollow piston-rod *C*, vacuum cup *D*, and spring *E*. The entire assembly is secured in the proper location on the feeding device by cap-screws in the tapped holes shown in the side of cylinder *A*. The vacuum line is connected to one of the pipe-tapped holes at either end of duct *F*; the other end is plugged. All other openings, of course, are plugged.

The device functions as follows: when a pickup is desired, the vacuum valve is automatically opened by a cam, or other means. Note that the lower port *G* is much larger than the upper port *H*. Upper port *H* is actually a leak from space *J* to the atmosphere via the hollow channel in the shaft, at this stage of operation. The vacuum formed below the piston *B* causes the piston and rod assembly to move downward. The space *J* above piston *B* is still at approximate atmospheric pressure because there is an air passage through the vacuum cup and the hollow piston-rod to space *J*. Downward movement continues until the vacuum cup *D* contacts the pile of sheets. Upon contact, air passage to space *J* is closed and a vacuum is set up. Because both ends of the cylinder are now in pneumatic balance, the spring *E* is free to lift the piston assembly upward together with the adhering sheet.

At the proper time, the sheet is dropped by releasing the vacuum. The drop may be hastened by the introduction of some air pressure into the system.

Self-Contained Pneumatic Ejection System for Punch Press

A certain shop found it necessary to equip a crank type punch press with a pneumatic ejection system. Because of the absence of a compressed air supply within the shop, a self-contained unit, operated by the normal motion of the press, was installed, as shown in Fig. 3.

Power to operate the air pump *A* is obtained from the drive mechanism of the press. Connecting-rod *B* is extended at its

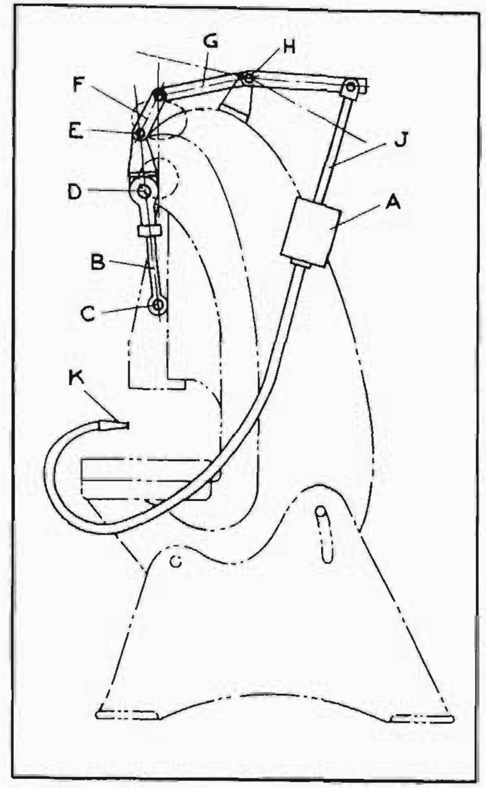


FIG. 3. Ejection system for punch press utilizes air pressure from a self-contained unit.

upper end so that cross-head pin *C*, crankpin *D*, and pin *E* of the extension arm lie in a straight line. The path of pin *E*, which connects the extension arm to free link *F*, is a curve which, for practical purposes, can be considered as an ellipse. The minor axis of this curve is equal in length to the stroke of the crank, and the major axis is equal in length to the stroke of the crank multiplied by $\frac{CE}{CD}$.

To achieve maximum efficiency of the ejection system, the stream of air should be discharged during the latter portion of the up stroke only. This was accomplished by making the length of free link *F*, which actually consists of two parallel links, equal to the approximate radius of curvature of the lowest part of the oval path traversed by pin *E*. Link *G* thus remains motionless during the initial part of the press up stroke, but pivots rapidly around shaft *H* during the final part of the same stroke, as can be seen from Fig. 3.

As link *G* pivots, piston-rod *J* of the air pump, which is secured to the press frame, is depressed. In this way, an air blast is discharged between the punch and the die. A nozzle *K* is attached to the end of a flexible air hose and directs the stream of air to any desired point in the work area of the press.

An Intermittent Feed for Strip Material

An arrangement for intermittently feeding the required length of strip material to a press is shown in Fig. 4. This patented mechanism is actuated by a pin *A*, which is attached to a slotted plate fixed to the bottom of the vertically reciprocating member of a press, or similar machine.

Pin *A* moves in the vertical plane and pivots a bellcrank *B*. This member, in turn, has an open-ended slot which engages with a pin projecting from one side of a sliding block *C*. Block *C* is guided and can only move in the horizontal plane. Thus, vertical movements of pin *A* produce horizontal reciprocation of

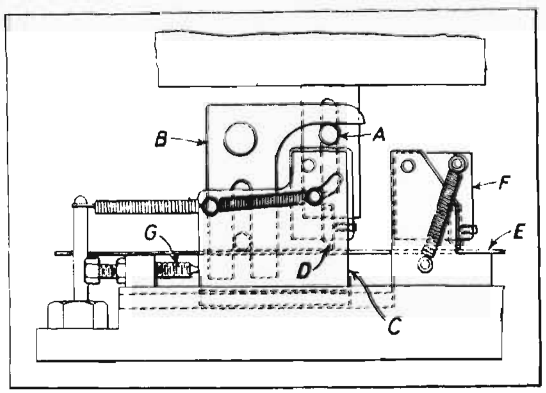


FIG. 4. A device for intermittently feeding lengths of strip stock to a die in a press.

the block. A pivoted holder and blade *D* are housed within block *C*, the blade being held in contact with the strip material *E* by means of a tension spring secured at one end to block *C*.

In operation, as pin *A* travels upward, block *C* moves to the right and, through the gripping action of the blade *D*, the strip material is fed in the same direction. A second spring-loaded and pivoted blade-holder *F* prevents the strip from moving back when block *C* is returned by an associated tension spring seen at the left in the illustration. An adjustable screw *G* limits the return movement of block *C* and the extent of the forward stroke is determined by the position of pin *A* in its slot, permitting the arrangement to be used for various sized parts.

Instant-Release Latch Mechanism

On an automatic machine, a bowl-shaped part must be tipped 90 degrees from a horizontal resting position over transfer bars to a vertical position in a cradle on the same transfer bars. The part is first moved off the bars into position over a gate which picks up the part and rotates it 90 degrees into the bars.

The problem in designing this equipment was to provide some means of latching the part to the tipping gate during rotation and releasing it into the transfer cradle at the precise moment that the part reaches its vertical position. The tipping gate then returns to its starting position. Such a latch mechanism is shown in Fig. 5.

The work-piece *A* is moved from Position 1 over transfer bars to Position 2 above tipping gate *B*. The tipping gate permits work-piece *A* to drop a few degrees below the horizontal plane, so that the work-piece may pass over button *C*, which prevents the part from sliding down the tipping gate during indexing. The tipping gate is rotated by shaft *D*, which passes through a latch operating cam *E*, that is fixed to a frame member. Shaft *D* also passes through elongated slots in the clevis end of the latch-operating link assembly *F*, which straddles cam *E*. The cam follower *G*, located between the clevis legs of link assembly *F*, is allowed to pivot freely on pin *H*, but cannot pivot farther than stop *J* in a clockwise direction.

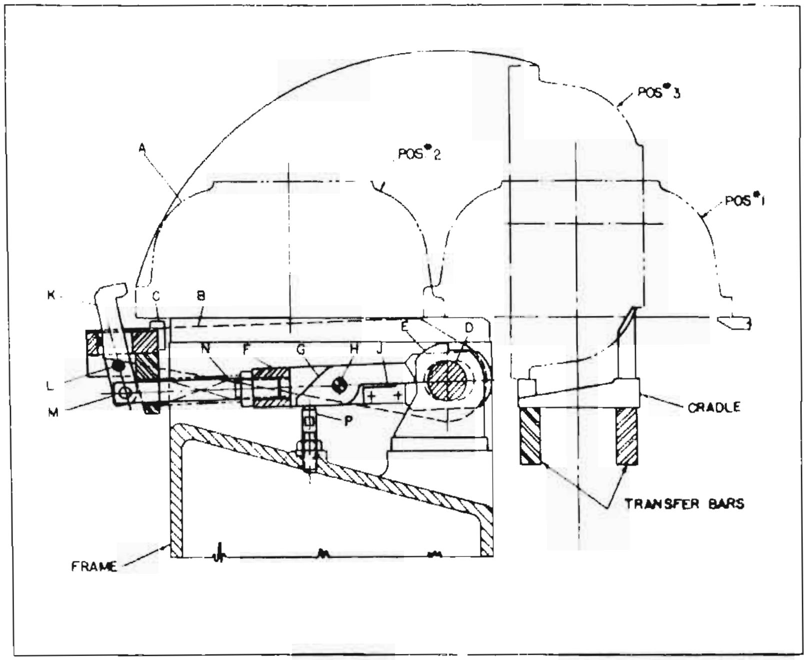


FIG. 5. An instant-release mechanism which facilitates a work-indexing movement.

Latch K pivots on pin L which is fixed to the tipping gate. The end of the link assembly passes through a clearance hole in the tipping gate, is forked to straddle latch K which pivots on pin M. The rotating motion of the tipping gate is imparted to the link assembly through pin M. As the tipping gate rotates, cam follower G backed up by block J rides up on the high lobe of cam E. The resulting motion compresses spring N *and* closes latch K.

Part A is now latched securely to the tipping gate B during the dwell portion of cam E. At the precise moment the part reaches its vertical position, the cam follower reaches a step in cam E, and the compressed spring N forces open latch K, instantly freeing the part from the tipping gate. The part is left resting in its transfer cradle (Position 3), as the tipping gate returns to its starting position. However, the cam follower G, which had

hooked itself over the stop in cam E, pivots on pin H, as the tipping gate rotates. This allows it to pass over the high lobe of the cam. As the tipping gate finally comes to rest at its starting position, the cam follower G is reset by button P. The latch mechanism is now ready to start its next cycle.

High-Speed Punch Press Feed Mechanism

Unlike the conventional feeding mechanisms provided on punch presses that utilize one-half the crankshaft rotation for their operation, the device illustrated in Fig. 6 functions through a span of 240 degrees. This feed mechanism enables a strip-fed, short-stroke punch press to be converted to a high-speed automatic machine.

With power feeding mechanisms, the intermittent motion as well as the necessity of overcoming the inertia of the coiled stock

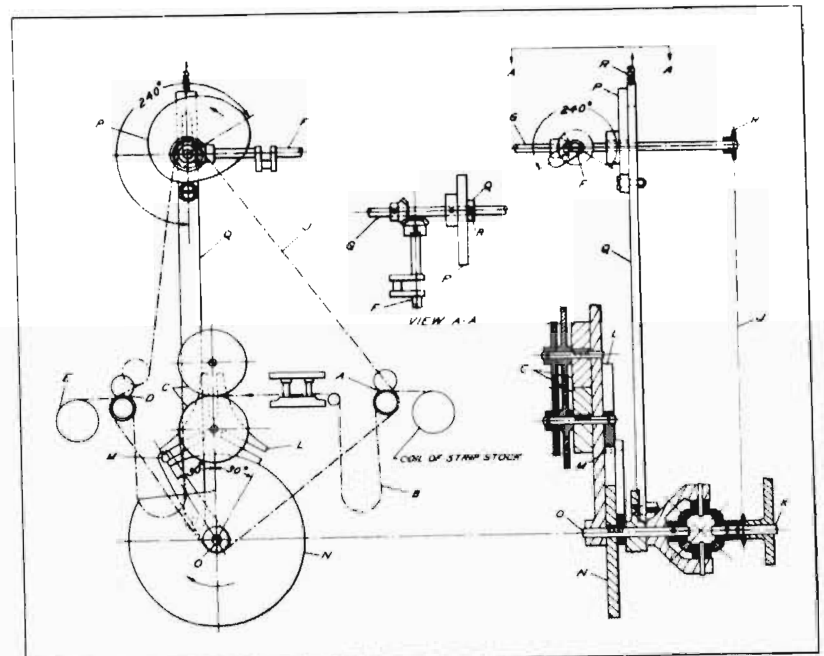


FIG. 6. Cam-controlled differential mechanism combined with Geneva movement to provide a high-speed feed for punch presses.

each time a feed movement occurs, limits the maximum speed at which the material may be fed. These speed-retarding factors have been eliminated in the device here described.

The strip stock is threaded between sprocket-driven rollers *A*, leaving loop *B*, and then over a small idler pulley and through the die (see Fig. 6). From the die, the material passes between feed rollers *C*, leaving a second loop, then through another pair of sprocket-driven rollers *D*, to be finally wound on friction-driven reel *E*. The length of each of the two loops is slightly greater than the required feed per stroke. The circumference of rollers *A* and *D* are equal to the length of feed. The circumference of feed rollers *C* is three times as large.

A pair of miter gears transmits the rotary motion of crankshaft *F* to drive-shaft *G*. This drive-shaft is positively linked, by means of sprocket *H* and chain *J*, to the sprockets on rollers *A* and *D*, and also to shaft *K* of a differential mechanism. All four sprockets are of the same diameter.

The mechanism was designed so that feeding occurred during 240 degrees and punching or forming occurred during 120 degrees of the cycle. Intermittent rotation of the three-branch Geneva wheel *L* actuates the main feed rollers. Movement of the wheel is controlled by arm and roller *M*, and by locking sector *N*. One-third of a revolution, or a 120-degree movement of the Geneva wheel, will feed the stock the required amount.

The roller on driving arm *M* enters a slot of the Geneva wheel at a right angle. Therefore, with the slots spaced at 120-degree intervals, and the roller entering at an angle of 90 degrees, it follows that the driving arm must be situated 30 degrees from the vertical center line as shown. From this it may be noted that a total angular displacement of 60 degrees is all that is necessary of the driving arm and the shaft *O*, to which it is keyed, in order to complete one feeding cycle of the stock. To spread this 60-degree motion over 240 degrees of crankshaft rotation, a differential gear mechanism, coupled to a cam, is employed.

Cam *P* is keyed to the drive-shaft and is designed with a simple harmonic rise through 240 degrees, and a return through the remaining 120 degrees. As the drive-shaft rotates in the direction of the arrow in the illustration, shaft *K* is driven at the same speed. In coordination with this movement, the differential housing is rotated in the same direction as shaft *K* by the action of cam *P* on rod *Q*. The rod is attached directly to the differential housing. By designing the cam to move the housing 90 degrees during a 240-degree rotation on shaft *K*, a total of 180 degrees of rotation will be lost to shaft *O*. In this way, only the 60 degrees necessary to index the Geneva wheel will be transmitted through the differential mechanism during the 240 degrees of crankshaft rotation used for feeding.

The opposite effect is produced by the differential during the last 120 degrees of cam rotation during which the punching or forming operation is being performed. As rod *Q* rises — due to tension spring *R* forcing the roller to follow the cam contour — the housing is moved in the opposite direction and shaft *O* is accelerated. This brings driving arm and roller *M* around to the starting position for the next cycle.

Sorting and Feeding Shells Closed End First

Hollow-drawn cylindrical shells often present handling difficulties when they must be fed rapidly and continuously to an automatic machine with the closed end foremost. This is especially true when the shells are fed to a sorting mechanism from a hopper.

The hole in the press-drawn shell *W* extends approximately two-thirds the length of the piece. Rapid delivery of this shell with its closed end foremost was required to insure economical production. The sorting mechanism for this job had to be of simple design, mechanically actuated, and relatively free from clogging tendencies.

The shells fall unassorted into a large hopper from a drawing press. By simply agitating the hopper, the shells pass down the vertical portion of a chute which feeds them into the short horizontal section *A* leading to the sorting and feeding mechanisms. The hopper and the vertical portion of the chute which holds about twelve shells are not illustrated. The sorting or feeding mechanism is interposed in the horizontal portion of the chute.

Chute *A* (see Fig. 7), is a cylindrical steel tube with an inside diameter of sufficient size to permit an easy flow of parts *W* down its vertical portion and through to the horizontal section. The end of chute *A* is fitted into a hole drilled through a boss in the left-hand wall of the cast-iron sorting body *B*.

Body *B* has flange extensions *D* with hold-down bolt holes *E* for fastening the sorting mechanism to a bracket extension on the press frame. A deep slot *F* is the sorting chamber of the

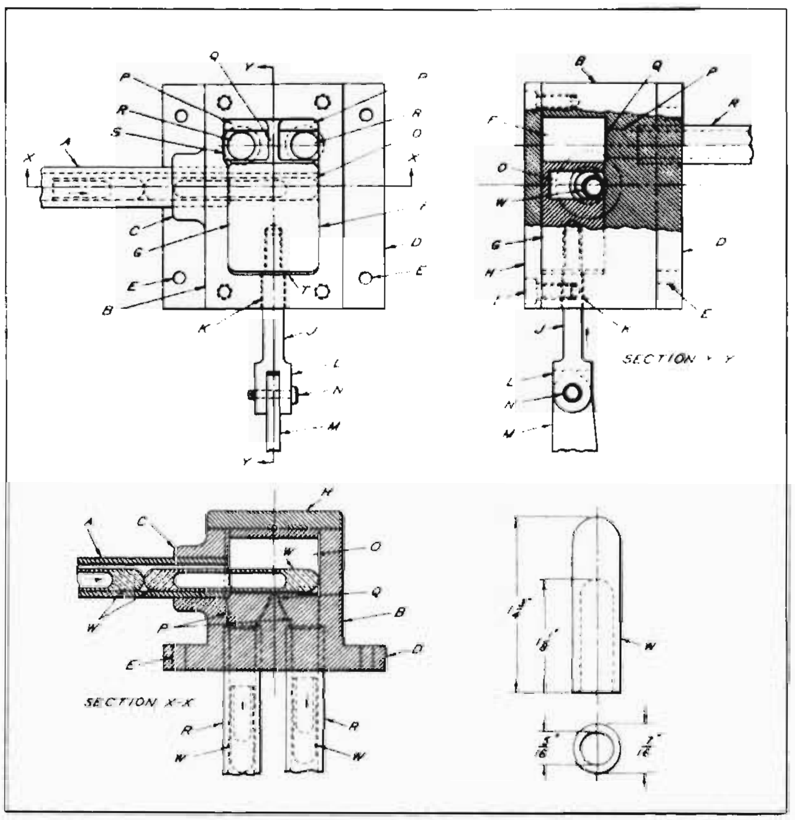


FIG. 7. Mechanism by means of which shells that are fed either open end or closed end first into a feed chute are all sorted and fed closed end first into chute leading to press for succeeding operation.

device into which the shells must pass. The floor of the slot is approximately 1/32 inch below the bottom edge of the bore of chute *A*, as shown in the lower view. Slot *F* is about 0.050 inch wider than the over-all length of the shell *W*.

The walls at each end of slot *F* serve as positive stops to control the sliding movements of the short hardened and ground steel slide *G* which moves easily in the slot. This slide is considerably shorter than the slot length to allow a movement of about 7/8 inch. The depth of slot *F* and the thickness of slide *G* are equally important, and are made not less than three-quarters of the over-all length of the shell.

The slide thickness should be about 0.005 inch less than the depth of the slot to allow a free working fit. The slide is actuated by a simple rotating barrel cam (not shown); linkage between the cam and slide being provided by the short connecting-rod *J*. The shank of rod *J* is screwed permanently into the end of the slide and passes through clearance hole *K* in body *B*.

Each revolution of the barrel cam causes slide *G* to make forward and return movements of 7/8 inch. The cam is designed to provide a short dwell period for the slide at each point of reversal. These reversal points coincide with the positions in which the shell is loaded into slot *O* and ejected from it. The width of slot *O* is slightly greater than the outside diameter of the shell, and the depth is made not less than seven-tenths of the over-all length of the shell, although the diameter may influence the depth dimension.

When slide *G* is in the retracted position, a shell will pass readily from chute *A* into slot *O* so that one end is in contact with the right-hand side of the slot *F*. As slide *G* moves forward within slot *F*, the shell will be carried along, and the plain end of the slide will automatically close off the mouth of chute *A* to prevent another shell from passing into slot *F*.

In the floor of slot *F* are two outlet wells *P*. The sides of these wells are inclined, as shown in heavy broken lines below bridge *Q* in section *X-X*. The minimum width dimensions of each well should be made appreciably greater than the outside diameter of the shell. An allowance of 1/8 inch was found satisfactory.

Bridge Q lies between the wells and central with the width of the slot F.

The two tubes R converge, forming a single tube of slightly larger diameter which feeds the shells directly to the dial mechanism of the press. The junction point of tubes R is omitted from the illustrations.

Slide G is shown in the loading position and near the extreme end of its stroke. In this position, slot O will be in alignment with the outlet of chute A, and since a considerable number of shell components will be carried in the chute, the weight of those in the vertical portion will exert enough pressure to push the first shell into slot O.

The illustration shows the work or shell located with its domed end foremost and lightly pressed against the right-hand wall of slot F. As soon as the shell is positioned in slot O, slide G is moved forward by the link-cam drive, for a distance of approximately 7/8 inch. This movement will also be imparted to the shell contained in the slide. The same movement of slide G serves to close up the end of the chute A, thus preventing the next shell in the bore of the chute from entering slot F. The blank side of slide G passing across the mouth of the chute A will hold the line of shells in position throughout the remainder of the slide movements.

When G is in its most extreme position the shell will be supported only at its mid-point by the very narrow bridge Q, and since the domed end is much heavier, it cannot remain balanced on the bridge Q, and will immediately tilt over in a clockwise direction and pass into the right-hand well P. It then passes down the vertical outlet tube R with its domed end foremost as shown by the broken lines in the section X-X.

A shell advancing to slot F with its open end foremost will enter slot O and be traversed to the opposite end of the sorting chamber in exactly the same manner as a shell entering dome end first. But this time the heavy end of the shell will fall in the left tube R.

CHAPTER 17

Hoppers and Hopper Selector Mechanisms for Automatic Machines

Tool engineers and machine designers are often faced with the problem of designing mechanisms to pick up parts from hoppers for delivery to the assembly machines. By "hopper feeding" is meant the indiscriminate dumping of a load of parts into a hopper of suitable size and shape, from which the parts are picked up, in the proper position, and deposited in a track for feeding to a machine by gravity. Ordinarily, the pick-up member is so shaped that the parts cannot enter the track if they are not in the right position, and therefore are dropped back into the hopper. Occasionally, the shape of the part and the speed requirements of the machine make it necessary to pick up parts that are not all in the same position. In that case, prior to going into the assembly machine, the parts are required to pass through an auxiliary mechanism, or separator, which arranges them all in the required position.

Many types of hoppers have been designed and built with varying degrees of success. One type of hopper may work successfully for a part of a certain shape, but may prove entirely unsuitable for pieces of a different contour. A great deal of thought must be given to the selection of a hopper for any particular job. Every new problem is unique in some respect, and will necessitate variations in the type of hopper selected.

Other hoppers and hopper selector mechanisms are described in Chapter 17, Volume III of "Ingenious Mechanisms for Designers and Inventors."

Designing Hopper Feeds for Square and Hexagonal Nuts

Nuts — both square and hexagonal — are employed in such large quantities as fastenings that the problem of automatically feeding parts of this type to various machines often confronts tool engineers and machine designers. Occasionally, the nut blank must be transferred to machines that perform secondary operations. In other cases, the finished nut must be automatically delivered to a particular location for assembly to other components.

A number of hopper designs have proved successful for handling work of this kind. Among these are the centerboard, the paddle wheel, and the rotary hopper types. Choice of the correct form of hopper to use will depend upon production requirements and, to a certain extent, upon the size of the nuts.

The centerboard type of hopper is widely employed for small and medium size nuts, either square or hexagonal in shape. Because of its low cost, large capacity, and excellent operating efficiency, this type of hopper should receive first consideration when a part delivery problem is met. A centerboard type of hopper and separator mechanism for feeding square or hexagonal nuts, properly positioned, to an automatic machine is illustrated in Fig. 1.

The hopper consists of a body, usually made in two parts as shown in section X–X; and a blade, fastened to an arm, which oscillates through the mass of parts that have been previously placed in the hopper. Parts that happen to be in the correct position drop into a groove machined in the top edge of the blade, and are raised by the blade in its upward travel. At its uppermost position, this groove is in line with a track, down which the nuts slide toward the machine. An actuating rod transmits motion to the centerboard blade from a cam or crank, not shown in the drawing. The centerboard arm is fastened to and pivots about a shaft.

A lever projecting from the opposite side of this shaft operates a knock-out slide. The knock-out slide advances, upon

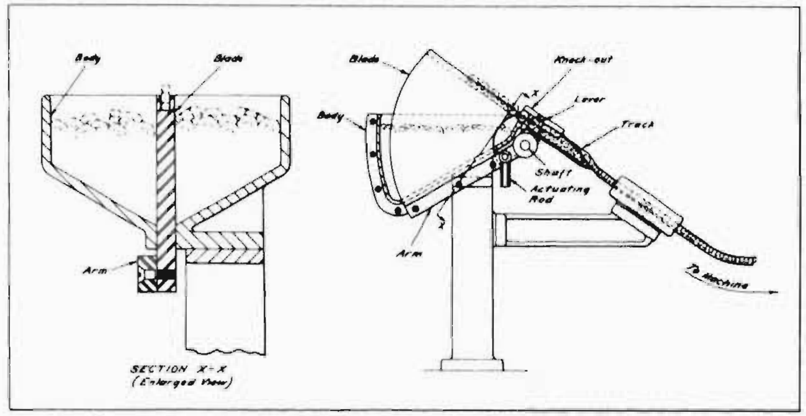


FIG. 1. Centerboard type of hopper and separator mechanism for feeding properly positioned nuts to an automatic machine.

descent of the center-board blade, and clears the mouth of the track of any parts obstructing it. An enlarged sectional view through the knock-out mechanism is seen in Fig. 2.

In cases where it is planned to use the hopper for more than one size of nut, the centerboard blade should be constructed sectionally, as shown in Fig. 3. The two positions in which it

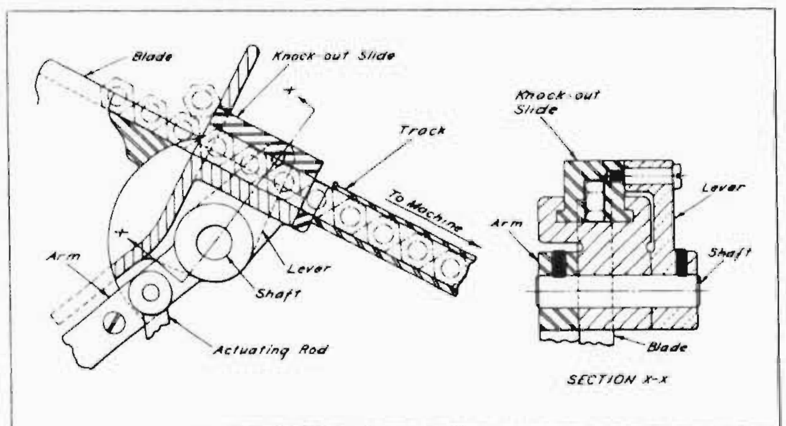


FIG. 2. Enlarged sectional view through the knock-out mechanism of hopper seen in Fig. 1. Oscillation of the knock-out slide clears the track mouth of improperly positioned parts.

is possible for the nuts to be raised by the blade are shown at *A* and *B*. A replaceable cap, keyed to the centerboard blade, is fastened by means of socket-screws as shown. In this manner, the feed mechanism can be changed quickly for a part of another size, as shown at *C*.

One method of constructing the track that delivers the parts to a selector mechanism or to the machine is illustrated at *A* in Fig. 4. The track is a steel bar in which a slot has been milled longitudinally to accommodate the parts. Two rails are fastened to the track by screws, as shown in section *X–X*. This construction allows the operator of the machine to push the parts along with a screwdriver should they jam due to one of the parts being slightly over size.

The end of the track is welded to a fastening plate, which is secured to the hopper with screws. At the opposite end of the track, a similar plate is fastened to the selector mechanism in the same manner. This allows quick removal of the track when changing the setup for other sized nuts.

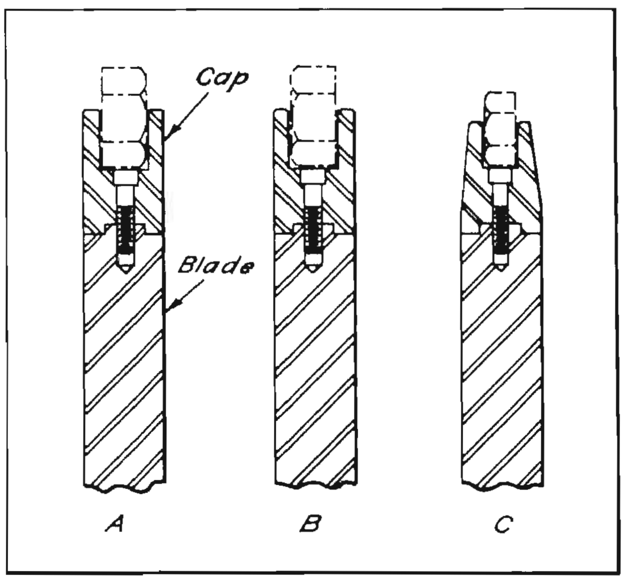


FIG. 3. Sectional construction of the centerboard hopper blade permits rapid change-overs for different sizes of nuts.

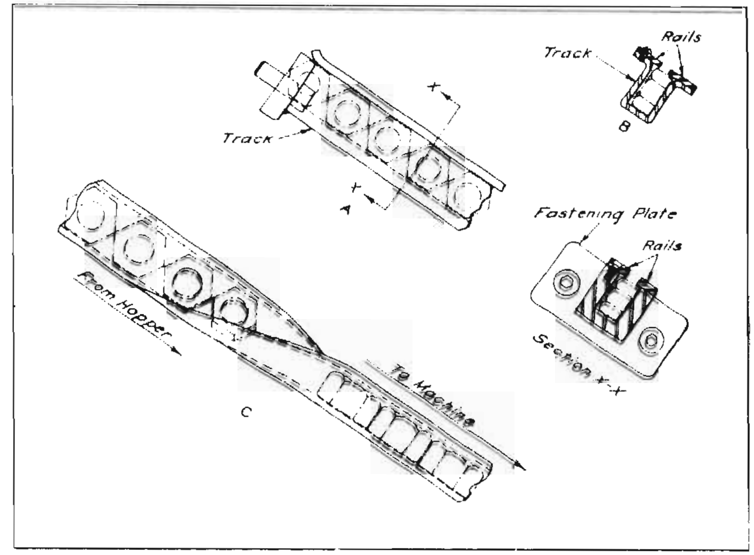


FIG. 4. Details of track construction shown in their position relative to the hopper. The track shown at *A* is machined from bar stock while that at *B* is formed from sheet metal.

A less expensive type of track construction is illustrated at *B*. In this case, the track is made of heavy-gage sheet metal, formed as shown. Rails, fastened to the track with small screws, permit dislodgment of jammed parts through the central gap between them.

A quarter twist may be applied to the track at a position between the hopper and the selector mechanism, or between the hopper and the machine, when the parts must be delivered in a horizontal position. Such a twist in the track is illustrated at *C*. Generally, a considerable amount of hand filing or grinding must be done on the walls of the track at this point to enlarge the sides for proper clearance of the parts.

The construction of one type of selector mechanism employed to turn improperly positioned hexagonal nuts upside down is illustrated in Fig. 5. The sectional view at the right shows the relative position in which the mechanism is mounted on the machine.

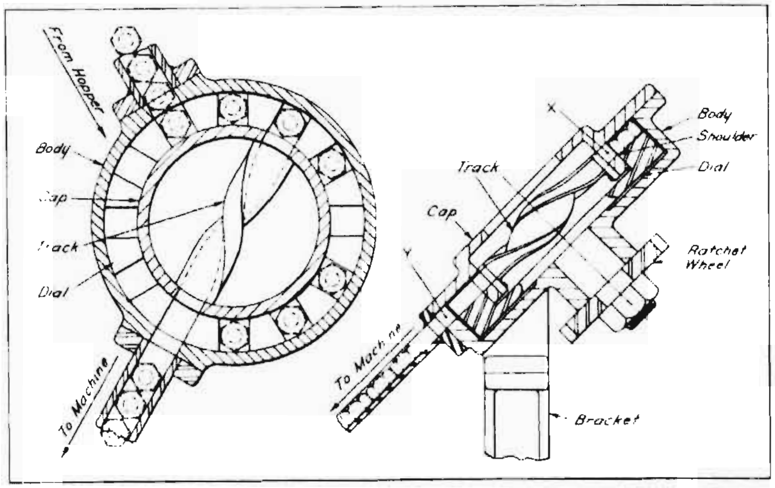


FIG. 5. Selector mechanism for inverting improperly positioned hexagonal nuts prior to delivering them to an automatic machine.

The selector mechanism consists of a cast body, which is fastened in a stationary position to a bracket on the machine; a cap, screwed and doweled to the body, which holds the twisted track of the mechanism; and a dial, free to rotate intermittently within the body and driven by a ratchet wheel.

In operation, the nuts enter the dial from the hopper track, as seen at the left. As the dial indexes, the nuts are carried to a position opposite the mouth of the twisted track. At this point, an opening *X* has been machined in the stationary cap in such a manner that a shoulder is formed between the cap and the body.

If the nuts are properly positioned (that is, if the flat side of the nut is down), this shoulder prevents them from entering the track, and they are carried around by subsequent indexes of the dial to point *Y*. Here they pass through an opening in the body, enter another track, and slide down to the machine.

However, when nuts enter the selector mechanism incorrectly positioned (with their curved side down) and reach the point opposite the track mouth, they slide over the projecting shoulder and enter the twisted track. In sliding through this track, they

are turned over and enter the track leading to the machine correctly positioned for assembly.

Owing to the extremely slight projection of the shoulder, all parts of the selector mechanism must be made sufficiently strong are fitted precisely to eliminate vibration. Improper fitting or vibration might allow properly positioned parts to jump over the shoulder and slide toward the machine resulting, of course, in incorrect assembly. The ratchet that indexes the dial should be driven by a crank. If a cam is employed, it should be of the harmonic motion type to give gradual acceleration and deceleration.

The same basic type of selector mechanism can be used for square nuts, as seen in Fig. 6. However, in this case, the selector shoulder is formed differently, due to the altered contour of the part. It will be noted that the shoulder projects upward at the corners, thereby preventing properly positioned nuts from enter-

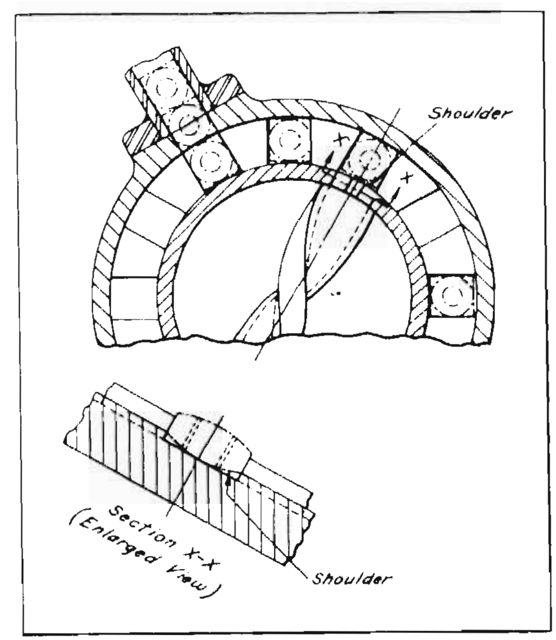


FIG. 6. Selector mechanism for correctly positioning square nuts before delivering them to the machine.

ing the twisted track. Incorrectly delivered nuts, however, slide over this shoulder and are inverted by the twisted track prior to entering the machine.

Another type of hopper frequently employed to feed nut blanks to automatic machines is the paddle wheel type shown in Fig. 7. This consists of a body which is made in two halves that are fastened together with screws and dowels. A spacer, slightly wider than the nuts, is provided between the body halves to allow the nuts to slide freely between the confining portions of the body. When parts are dumped indiscriminately into the hopper, a few will fall into the track formed in the bottom. The rotating paddle wheel pushes the nuts upward, and when they reach the top of the hopper, they enter a track and slide down to the machine.

A retaining dog, which pivots loosely on a pin, prevents the nuts from sliding back every time one of the blades of the paddle wheel clears the track and before the next blade has advanced more blanks to push the row upward. While it is not absolutely necessary, such a retaining dog adds to the smooth operation of the hopper. An offset is provided at the pivoting end of the dog, as shown in view Y-Y, to allow the nuts to slide

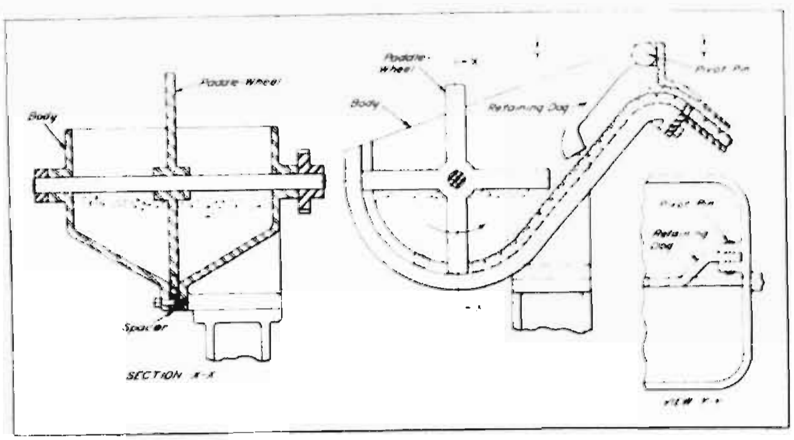


FIG. 7. The paddle wheel type of hopper is an effective means of feeding square or hexagonal nuts to automatic machines.

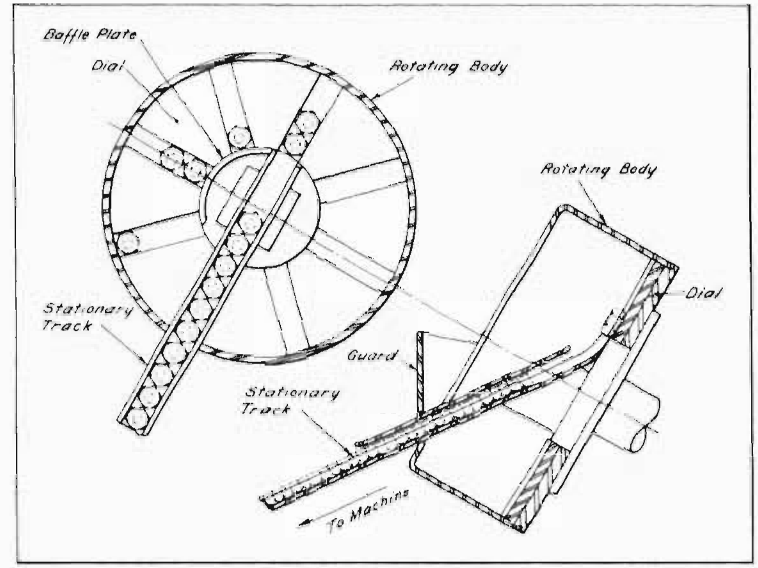


FIG. 8. Rotary type hoppers consist essentially of a rotating body and dial, a stationary track, a baffle plate, and a guard.

up the face of the hopper when the track is full. These parts then fall back into the hopper.

Rotary hoppers, such as the one seen in Fig. 8, can also be arranged to feed nuts to automatic machines. The hopper shown consists of a rotating body attached to a dial. The dial has radial slots milled in it, as shown. A stationary baffle plate prevents the nuts that are being carried upward in the slots from falling back into the hopper. As these parts reach a position opposite the mouth of the stationary track, they enter the track and slide toward the machine. When the track is full, the nuts go past the track mouth and drop back into the hopper.

As in the selector mechanisms described, a shoulder can be arranged at the track mouth to allow only correctly positioned parts to enter, if this feature is desired. In such cases, of course, the track must be given a half twist at some point between the selector and the machine, so that the nuts will be delivered in the correct position for assembly. The guard shown increases the capacity of the hopper.

While both square and hexagonal nuts lend themselves well to hopper feeding, a considerable amount of thought must be given to each individual problem in order to determine the best possible hopper arrangement for each particular case. The component parts of the hopper and selector mechanism must be strong and rigid, and must be machined with smooth finishes and to close tolerances. Sudden jarring motions must be avoided, especially in the selector mechanism.

In other words, the hopper feed must be designed and built with the same care that is given to the machine to which it is applied. Flimsy, poorly designed mechanisms are the principal cause of the difficulties many plants have experienced with hopper feeds.

Designing Hopper Feeds for Bottle Caps – I

The "double-disc rectifying hopper" shown in Fig. 9 is designed to feed bottle caps to automatic machines. It operates on an entirely different principle from the pin type hopper. In the double-disc hopper, the two discs *E* and *F* are fastened to the shaft *G* and kept in continual rotation in their housings *A* and *B*, respectively. The two housings are separated by four spacers *C*. A thrust bearing *H* is provided to give easy and smooth rotation. Each disc is provided with four drivers, each driver consisting of a plunger *O*, a spring *K*, and a split washer forced on the plunger. This unit of the hopper is supported on the column *D*.

As the caps fall on the disc *E*, they are thrown by centrifugal force to the inner side of the housing where they finally take a vertical position. The drivers then push the caps to the slot *I*, through which they are guided to the rectifying chute *L*. This chute is provided with two ribs which direct the caps facing one way down the chute *M*. The caps facing the opposite way fall on disc *F*. Therefore, all the caps falling on disc *F* are in one position; and when the operation of placing these caps in a

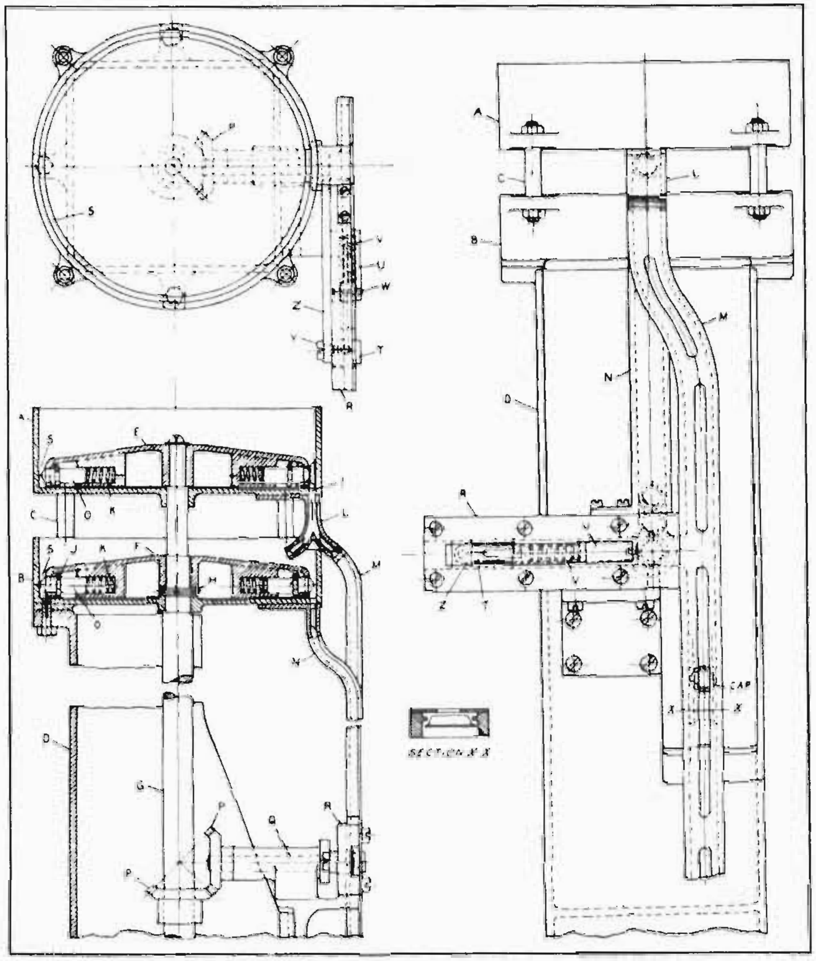


FIG. 9. Double-disc type of hopper for feeding bottle caps to cap making or bottling machines. Caps falling from disc *E* into chute *L* which are facing the desired way, fall into chute *M*. Caps facing the opposite way fall on disc *F*, and their positions are reversed before they enter chute *N*.

vertical plane is repeated (the caps being pushed into chute *N*), they are in the same position as those caps which passed down chute *M*.

Thus there are two lines of caps which must be combined to form one line for feeding the machine below. This is accomplished by a pusher mechanism operated through a pair of bevel gears P from the main shaft G. The mechanism consists of a shaft Q provided with an eccentric plate which operates the driving slide T through the connecting bar Z pivoted on stud Y. A sliding chamber R is provided for the driving slide and push-plate U. These two parts have a flexible connection consisting of a pin W forced into plate U, which is free to operate in a slot provided in the driving slide, and a light compression spring V. This entire mechanism is supported on the bracket which also supports the rotating shaft Q.

The operation of the mechanism needs little explanation. When the chute M is full of caps to a point above the opening from chute N, the caps should not be fed from N to M. This is controlled through the compression of spring V by the driving slide T due to the resistance encountered in trying to push caps from N to M. As soon as there is an opening in chute M, the spring V expands and the full length of the driving slide and push-plate becomes effective, pushing caps from N to M.

This hopper arrangement is provided with a safety slot identical to the one provided in the pin type hopper feed previously described. This slot is even more necessary in the double-disc rectifying hopper here described because of the greater possibility of caps passing into chute M in the wrong position.

Designing Hopper Feeds for Bottle Caps – II

The bottle-cap feeding mechanism shown in Fig. 10 is known as the quarter-turn chute rectifying hopper. The convex disc B is kept in rotation at a moderate speed in the housing A. The disc is supported by a thrust bearing F. A bronze bushing G serves as a bearing for the main shaft E, to which the disc B is fastened.

Four buttons D, pressed into the disc B, serve to agitate the caps until centrifugal force carries them to the inner side of the housing, where they take a vertical position in the annular slot W. The caps pass from slot W down through the opening Z into the rectifying chute C, where they encounter the two ribs

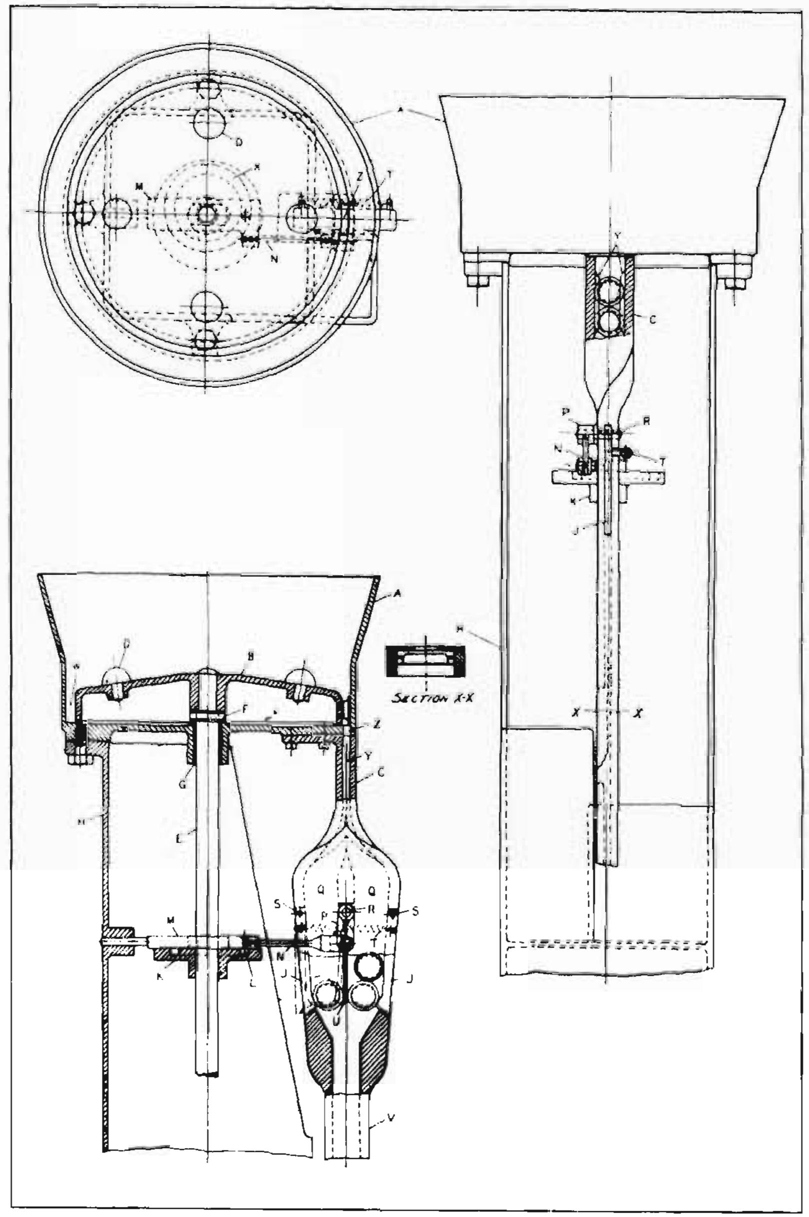


Fig. 10. Bottle-cap feeding mechanism with quarter-turn chute rectifying hopper which is so designed that the delivered caps all face the same way.

Y. These ribs separate the caps facing in opposite directions into two separate lines, which make a quarter-turn so that all the caps will face the same way.

When the caps reach the collecting chambers *Q*, they are checked between the oscillating finger *U* and the gate fingers *J*. The oscillating finger receives its motion from a cam *K* attached to the main shaft *E*. A pin *L*, fastened to the slide *M*, fits into the cam groove. The slide *M* is connected to the rocker arm *P* by the connector *N*. The rocker arm and the oscillating finger are both attached to the pin *R*. The two gate fingers *J* are pivotally mounted on pins *S* and are held together in a closed position by the spring *T*.

As the finger *U* moves to one side, it causes the caps and the gate finger *J* to be pushed outward. The spring *T* is stretched since the opposite gate finger cannot enter into the collecting chamber. The movement of the finger *U* to one side opens up a space big enough for the caps to pass down to the chute *V*. Therefore, the discharge of caps is controlled by the finger *U* oscillating back and forth and releasing caps first from one side of the collecting chamber and then the other.

This hopper is considered very efficient. The principle on which it operates is simple and most important of all, it has practically eliminated the scratching and mutilation of the caps. The one criticism of this mechanism is directed toward the rectifier chamber *C*, where the caps make a quarter turn. This chamber is difficult to make and it must be very carefully designed in order to provide a smooth flow of caps into chute *V* at the lower end.

Designing Hopper Feeds for Bottle Caps – III

The "double-escape rectifying hopper" shown in Fig. 11, for use in feeding bottle caps to production machines, is much the same as the "quarter-turn chute rectifying hopper" with the main exception that the quarter-turn chute has been replaced by double-escape slots *S* in the housing *A*.

Each slot *S* is machined so that it will permit bottle caps to pass through into the chutes *C* and *D* only when they face one

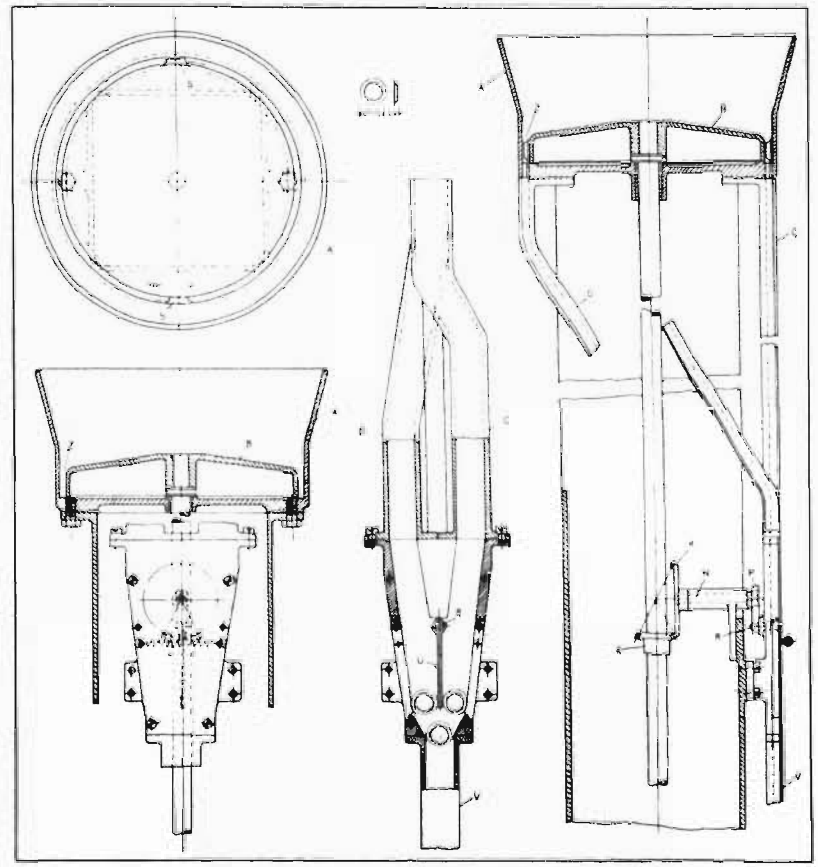


FIG. 11. Hopper feed with "double-escape" slots for automatic feeding of bottle caps.

way. The result is that the caps are divided between chutes *C* and *D* according to the positions in which they drop into the slot *Z* from the rotating disc *B* in the hopper. By placing these escape slots *S* 180 degrees apart, the necessity of making a twist in one of the chutes in order to bring all the caps to the collecting chamber in one position has been eliminated.

The operation of the collecting chamber is the same as that of the quarter-turn rectifying hopper. A slight change is shown in the operating mechanism. This consists of a pair of bevel gears K which rotate a small eccentric plate on the end of the shaft N. The eccentric plate oscillates the rocker arm P which, in turn, oscillates the finger U. The rocker arm P and the finger U are both fastened to the shaft R. The oscillating finger allows bottle caps to drop into chute V alternately from chutes C and D without clogging.

Designing Hopper Feeds for Bottle Caps – IV

A hopper A with the usual plate B for cutting off the cap supply, and the hinged clean-out door C held in place by the

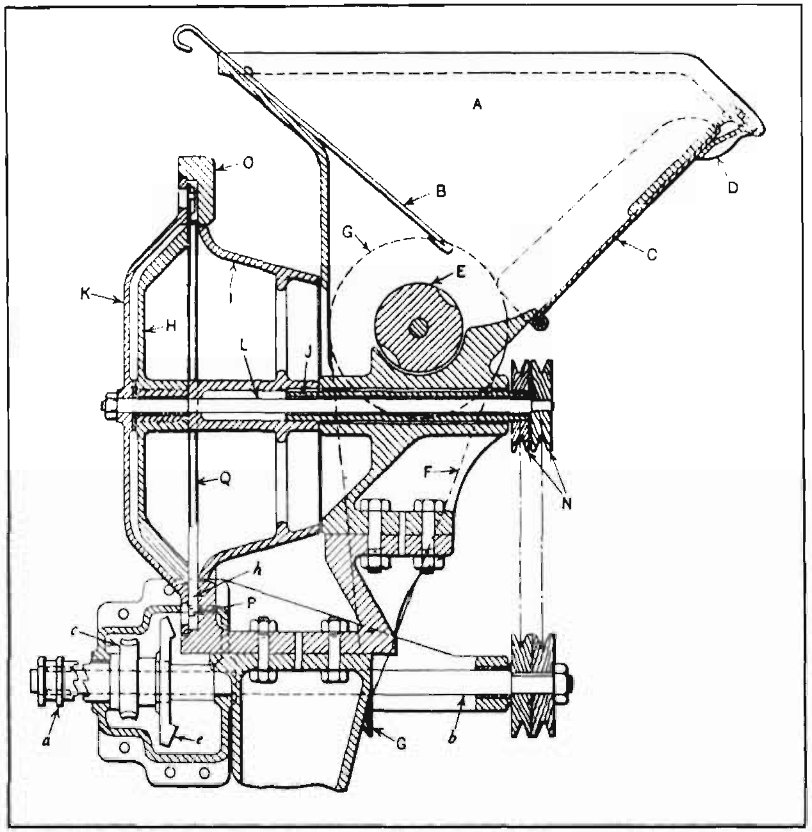


FIG. 12. Cross-section through hopper and driving members of bottle-cap feeding mechanism shown in Fig. 13.

latch D is shown in Fig. 12. A fluted roller E constitutes the first means for obtaining cap control in the hopper. The fluted roller is driven by the belt F which connects pulleys G. The rotation of the roller at a steady speed controls the feeding of caps into the revolving chamber so that the caps are discharged in small numbers.

The revolving chamber consists of two dish-shaped members H and I fastened together and keyed to the hollow shaft J. The pin-wheel K is fastened to an independent shaft L passing through the hollow shaft J. Each shaft is provided with a pulley N, and is so mounted that the two shafts can be rotated independently. This form of construction permits regulation of the speed of the revolving chamber relative to the pin-wheel K, and also permits changing the direction of rotation of the pin-wheel relative to the revolving chamber.

In one case, it was found that best results were obtained when the pin-wheel K was rotated in the opposite direction to the revolving chamber and at the same speed as the chamber. It was observed that there was then a minimum amount of churning and agitation of the caps, and that they were removed rapidly from the chamber by the pin-wheel.

The pin-wheel K is provided with ordinary straight pins P. A housing ring O and the guide ring Q are provided to form a chamber for the caps. The ring Q is fastened to the housing ring O and serves to guide the caps until they pass out to the open cap-rectifying chute R, Fig. 13. This chute is a very clever though simple device. It is provided with ribs which, aided by the fact that the location of the center of gravity of the cap causes it to fall upward, serve to place all caps in one position on the disc S. A cover T prevents the caps from flying out of the chute before they reach the point to which they are being directed.

After reaching the horizontal disc S, which revolves continuously in its housing, the caps have about completed their passage. Centrifugal force throws the caps to the inner edge of the housing. From here they are directed to the chute U, seen in Fig. 14, which conducts them to the machine below.

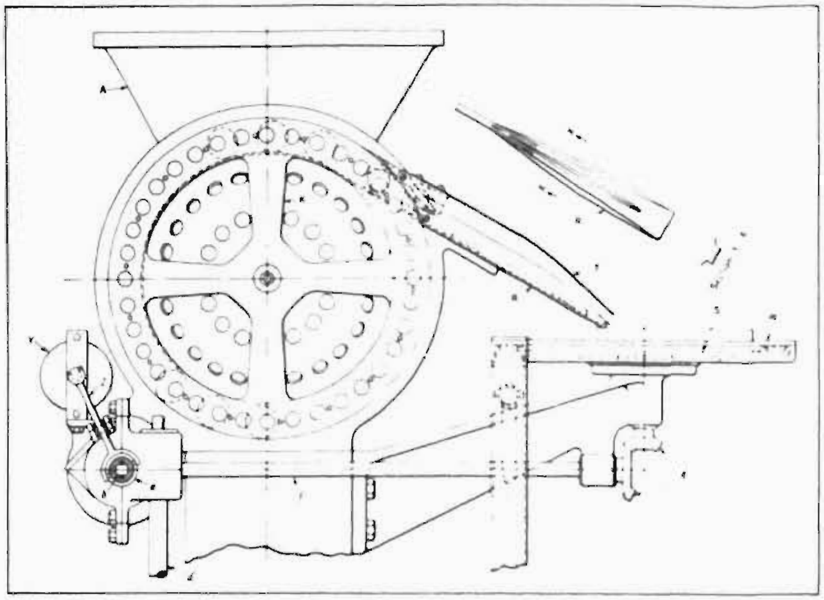


FIG. 13. Hopper mechanism for feeding bottle caps to production machines.

It is apparent that some means must be provided to control the flow of caps from the pinwheel *K* to the disc *S*. This is taken care of by an electrical arrangement consisting of contactors *V*, which are fastened to the disc cover *W*. One of these contactors is provided with the finger *X* which extends through a slot in the disc cover so that it can come in contact with the caps passing under the cover *W*. A solenoid *Y* is shown in Fig. 13.

The plunger of the solenoid *Y* is attached to one end of the rocker arm *Z*. The other end of the rocker arm is shaped into a yoke and provided with two pins which fit into a groove machined in the clutch member *a*, Figs. 12 and 13. This clutch member is attached to the shaft *b* by means of two keys, thus permitting a sliding motion for the clutch *a* under the action of the rocker arm *Z*. The second member of the clutch is part of the worm-wheel *c*.

The entire hopper receives its power from the shaft *d* through a worm not shown in the drawing, and the worm-wheel *c*. The disc *S* is rotated from a pair of bevel gears, one of which (not shown) is in contact with the bevel gear *e*, through the shaft *f* and the pair of bevel gears *g*. The bevel gear *e* is fastened to the worm-wheel *c*, forming a single unit free to rotate on the shaft *b*.

Caps from the hopper *A*, Fig. 12, pass through the fluted roller and, entering the revolving chamber, immediately drop to chamber *h*, where they fall between the pins *P* on the pin-wheel *K*. The pin-wheel carries the caps up between the guide-bar *Q* and the ring *O*, and discharges them onto the open cap-rectifying chute *R*, Fig. 13. Here they are all caused to face upward and

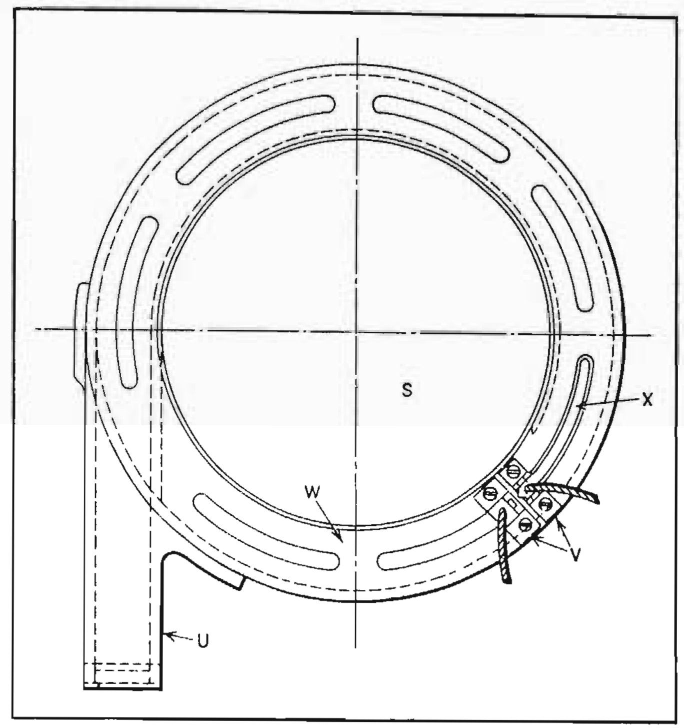


FIG. 14. Plan view of disc *S*, Fig. 13, showing finger *X* that completes electrical circuit, stopping feed when disc *S* becomes overloaded due to clogging of machine.

slide down to the disc S, from which they are discharged to the chute U.

Now, if the disc S is crowded to capacity, the feeler of the electrical contactor V, Fig. 14, is raised so as to complete the circuit and energize the solenoid. If the contactor is allowed to make contact for a short period, nothing happens at the solenoid because of the lag in the time it takes to completely energize the solenoid. However, if the period of contact is continued, as would be the case if there were a full line of caps passing under the feeler X, the solenoid becomes effective.

Upon being energized, the solenoid plunger is drawn in, causing the clutch members to be separated through the rocker arm Z. This immediately stops the rotation of the pin-wheel K, the revolving chamber, and the fluted roller. The disc S remains in rotation because it is actuated by the gear e, which is fastened to the worm-wheel c and is in continual rotation from the shaft d. Churning action in this mechanism was reduced to a great extent in the revolving chamber through the use of reverse rotation and the elimination of the pin arrangement used in the hopper described in the first article.

Designing Hopper Feeds for Bottle Caps – V

Certain cap manufacturers produce a type of bottle cap about twice the size of the ordinary cap used for beer bottles. The machines producing these large caps are relatively slow-speed machines and do not require as elaborate designs of hopper feeding arrangements as for small caps. It was therefore found advantageous to use the type of hopper illustrated in Figs. 15 and 16. This is a simplified form of hopper, the main feature being the rectifying chute, which is quite simple in itself. The rib A, Fig. 15, divides the caps so that they are all discharged with their open face up on the horizontal rotating disc B.

The charge of caps is thrown on the horizontal disc C, and centrifugal force places the caps in a single line between the bar D, Fig. 16, and the inner edge of the housing E. From this disc the caps pass down the rectifying chute F to the second horizontal disc B, where centrifugal force again is used to form

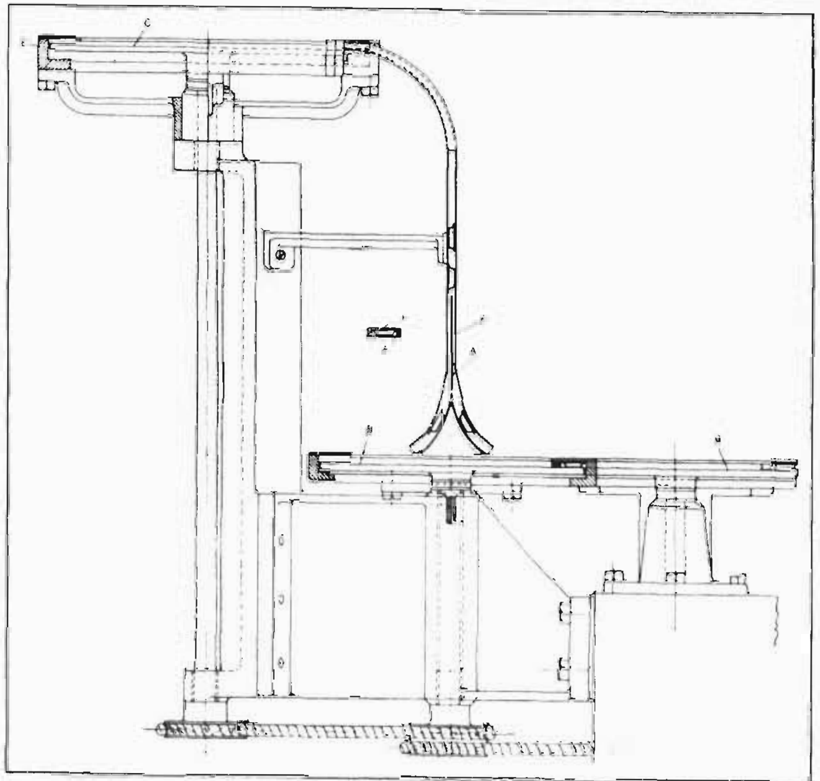


FIG. 15. Type of hopper feed used for large-size bottle caps.

a single line of caps. From disc B, the caps pass to disc G, from which they are fed into the feeding mechanism.

It is obvious that this hopper is very much simplified both in design and construction. Nevertheless, it has maintained excellent performance. However, it is restricted to a slow-speed feeding and has no means for controlling the flow of caps other than that obtained by limiting the number of caps thrown on the first disc from the bin overhead.

Designing Hopper Feeds for Bottle Caps – VI

Bottle cap manufacturing machines and bottling equipment are among the most essential elements of the brewing industry.

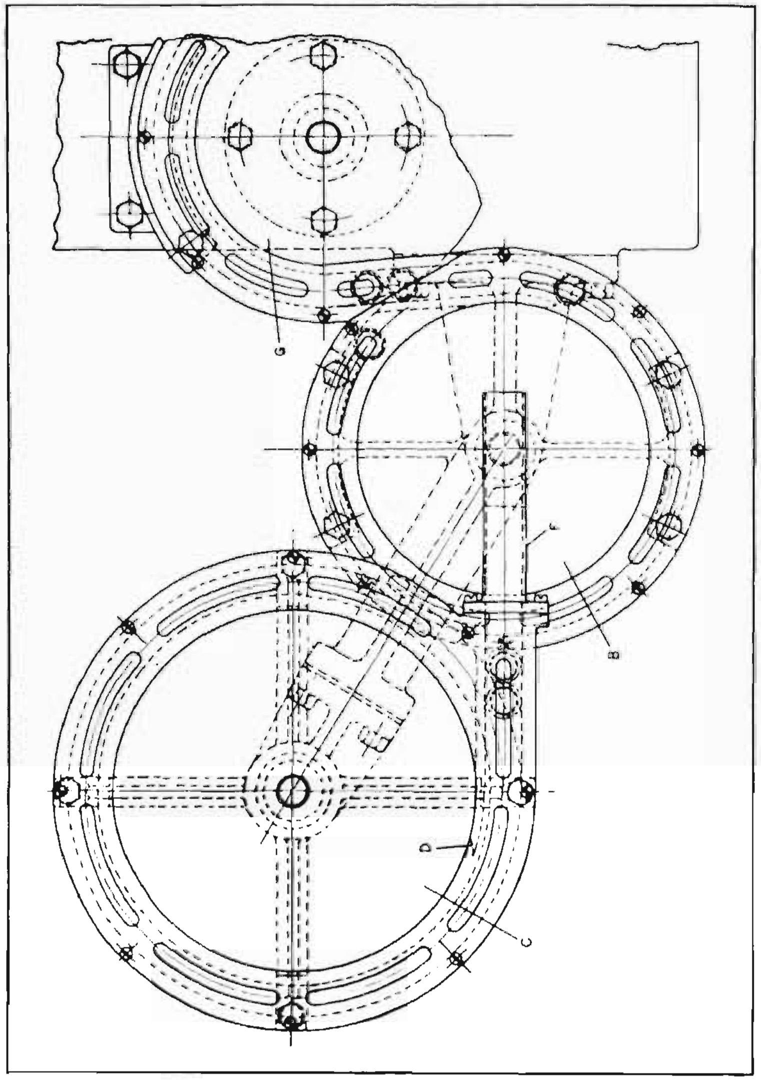


Fig. 16. Plan view of bottle cap feeding equipment shown in Fig. 15.

Remarkable progress has been made in the development of these elements along lines intended to increase production and achieve automatic operation. The hopper feed requirements of the bottle cap manufacturing machines and the bottling machines are very much alike and present designing problems requiring much the same treatment.

The word "hopper" suggests simply a deep receptacle or container for holding a supply of caps for the machine. However, there are many details connected with these hoppers that make them more than a simple receptacle. The most important of these details is the mechanism for arranging the caps so that they will all be face up or all face down, depending upon the kind of machine under consideration. In the bottle cap manufacturing machine, the caps are generally delivered face up, while in the bottling machine, they are usually required to be face down.

The ordinary bottle cap is usually made of sheet tin of a relatively thin gage and can be distorted under light pressure. It is provided with a cork disc, retained in place by an adhesive such as gutta-percha or albumen. An appropriate design is usually lithographed on the top of the cap. Therefore, any device that handles the caps must be so designed that it will not disturb or mutilate the lithographed design.

The general construction of a so-called "pin rectifying hopper" for delivering the caps to a machine in an orderly manner, all with the same side up, is shown in the accompanying illustration. The chute *B*, see Fig. 17, shown by the dot-and-dash lines, supplies caps to the main hopper *A*. The plate *M* serves as a means of checking the flow of caps, in case too large a quantity is placed in the chute. A clean-out door *N* is provided, which is held in place by clip *P* and latch *R*. The rotating member consists of the dish-shaped plate *C* and the ring *D*, which are held together as a single unit by the pins *E*. The ends of these pins are riveted over, as shown in the enlarged section *Y–Y*. The rotating member is fastened to the shaft *F*, supported in bearings *H*, and is kept in continual rotation by a belt drive to the pulley *L*. It is desirable to have a minimum amount of

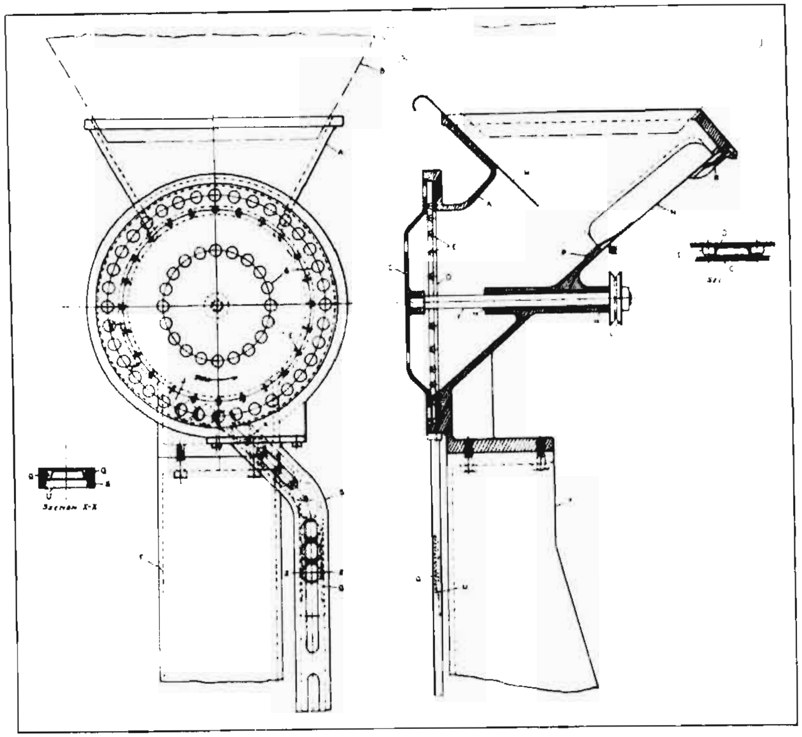


FIG. 17. Hopper feed designed to deliver bottle caps to machine, all with the same side up.

clearance between the ring D and the hopper A. A large number of holes G are provided in the plate C and slots are cut in the cover of chute S for the purpose of observing the movement of the caps. The entire hopper is supported by column T.

When the caps fall into hopper A, they are kept in a state of continual motion by the pins E. From section Y–Y it will be seen that the caps can pass between the pins E only when they are in one position. It is, therefore, the purpose of the rotating member to keep the caps in motion until they are so positioned that they will pass between the pins to the chamber J. From this chamber the caps fall into the slot K and thence down the chute S, which feeds them into the machine in an orderly manner.

As a precaution, chute S is provided with a safety slot which prevents any cap that may fall in the wrong position from passing to the machine below. The safety slot is shown in section X X. It consists of two ribs Q, which hold the cap in the chute if it is in the proper position, or permit it to drop out of the chute through the opening U if the position is reversed. This safety device becomes effective when a cap is crushed, so that it passes between the pins E to chute S.

The main advantage of this arrangement is its simplicity. Its few operating parts and low construction cost make it very desirable from the manufacturing point of view. On the other hand, the rotating member causes the caps to turn around in the hopper and often results in scratching the lithographing. This condition is aggravated by excessive pressure when there is no automatic action available for regulating the quantity of caps admitted to the hopper. Therefore, the main disadvantage is in the churning action, coupled with excessive pressure, which results in serious abrasion and scratching.

"Jamproof" Feeding Mechanism

Figures 18 and 19 present an interesting materials handling system for advancing small parts onto a feed track. The system possesses merit because of the manner in which the feed rate is controlled to prevent any parts from jamming on the track. In this instance, small round-head screw blanks are being fed into a threading machine.

A shallow pan A, Fig. 18, holds a batch of screws. The pan rotates at about 40 rpm around an inclined axis. As the pan rotates, the screws are directed onto a fork B, where some of the screws will align themselves and hang by their heads. At its other end, the fork is hinged to the feed track C, the slot of which forms a continuation with that of the fork.

The fork is intermittently raised to the position shown by the broken lines, where the screws on the fork can slide onto the feed track. This movement of the fork originates from a cam (not shown) rotating on the bottom of a vertical stem D joined to the fork by a link E. As the cam slowly rotates, it intermit-

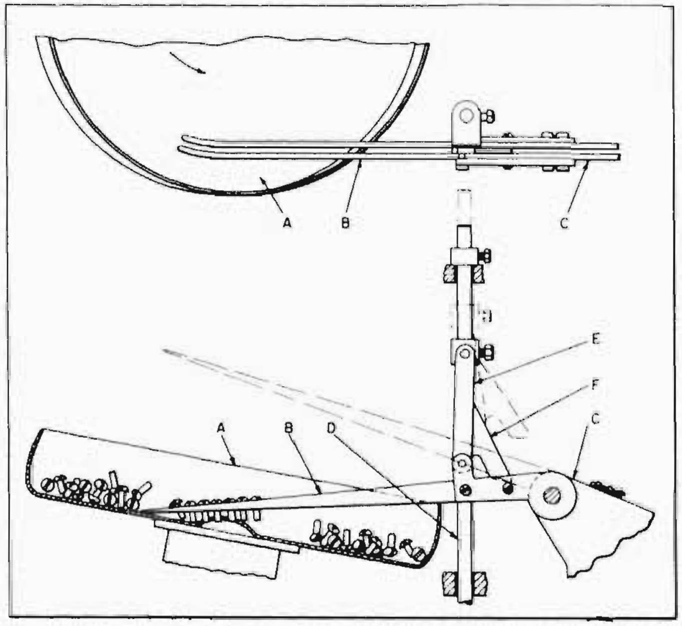


FIG. 18. When the track is not congested, the fork can drop into the pan to pick up screws.

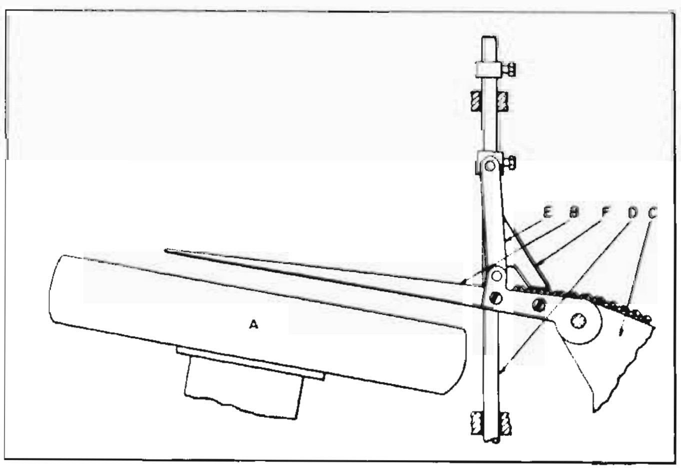


FIG. 19. When the track is congested, the feeling finger prevents the fork from dropping into the pan.

tently raises the stem, then lets the stem fall down. In this manner, the fork is also intermittently raised or lowered.

When the pan is full, each time the fork is raised it may deliver more screws than can be accommodated on the feed track. Consequently, there may be a serious jam on the track, if it were not for a feeling finger *F* which is incorporated in the mechanism. This finger is fixed to the stem, and moves up and down with it.

If the line of screws in the track extends back onto the fork, as in Fig. 19, then the down stroke of the stem is arrested when the finger strikes the heads of the screws. Since the stem and fork are linked together, the fork does not swing completely down into the pan until the congestion on the track has ended. Then, as in Fig. 18, the finger can enter the track during the reciprocation of the stem, and the fork can drop into the pan in order to pick up more screws.

Disc-Stacking Device

The power-operated device shown in Fig. 20 is used for stacking discs like the one shown at *A*. The same arrangement can also be employed for other forms of discs or shallow shells. It is entirely automatic in operation, except for the removal of the complete stack of discs. The particular device shown has been in use for several years and has never given trouble or required repairs of any kind.

Discs *A* are stacked in piles of varying numbers after being punched out of sheet metal on an inclined press. The stacking device consists essentially of two feeding or work-traversing screws *E* and *F* with a special form of square thread. This thread is cut very thin to provide a large space between adjacent threads. The screws are identical in size and shape, with the exception that one has a left-hand thread and the other a right-hand thread.

The screw *E* fastened to the shaft *J*, is driven by the main shaft *L* through a pair of spiral gears *K*. Two spur gears *H* transmit power to the screw *F*, which rotates on stud *G*. The

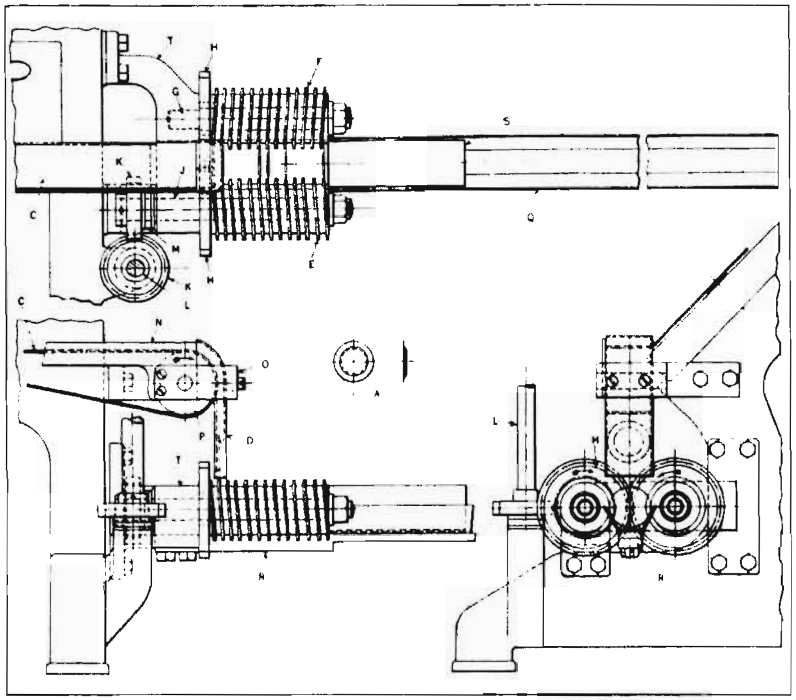


FIG. 20. Device for stacking discs such as shown at *A*.

entire mechanism is carried by a bracket *T* fastened to the press. Thrust bearings *M* are provided for the spiral gears.

A conveying arrangement consisting of a belt *C* and a pulley *P* delivers the disc to the chute *D*. The bracket *O* supports this conveying equipment. A guide strip *N* is located at the side of the belt. The vertical chute *D*, which directs the discs from the belt to the screws *E* and *F*, is fastened to bracket *O*.

The operation of the device is very simple. The press, being inclined, discharges the discs on the conveyor belt *C*, which, in turn, carries them to the vertical chute *D*, through which they drop to a position between the threads of the revolving screws *E* and *F*. The screws advance the discs until they are discharged into the trough *Q*, which is supported by a bar *R* fastened to the

bracket *T*. As the discs are deposited in the trough in an orderly fashion, the slug *S* is gradually pushed back. The trough is graduated so that the attendant can see when the desired number of discs has accumulated. They can then be removed and the slug *S* pushed back to its original position.

As the stacking device receives its supply from a conveying belt, it can be located at any required point. The best results are obtained when the screws *E* and *F* revolve one revolution for each disc produced. At this rate, a smooth and orderly stack is obtained.

Crank Principle Controls Shuttle Between Supply and Discharge Chutes

Figure 21 shows a device for directing small objects — ball screws, rivets and the like — from one chute to two chutes. Initially, nest *E* in slide *D* is located beneath chute *A* allowing an object to fall into it. The nest is then caused to move to one of the chutes *B* or *C* by crank or cam. After the object has fallen into the chute, the nest moves back to chute *A*, picks up another object, then moves to the opposite chute and drops the second object.

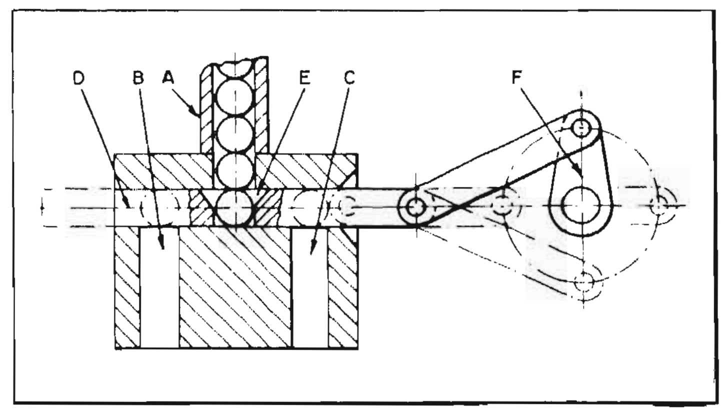


FIG. 21. Transfer of parts from one vertical plane to another is made possible by utilizing the principle of the crank.

Mechanism for Orienting Pins in Assembly Machine

Prior to pressing pins into side plates of a cotter-pin roller chain it is necessary to position them so that all pins going through a tube A leading to the assembly machine will have the cotter-pin hole at the bottom end. These pins are of the type shown at X in Fig. 22. They are ½ inch in diameter by 2¾ inches long.

In the mechanism shown a tube extending from a rotating hopper carries pins fed by gravity. However, the pins are not oriented as to position of the cotter-pin hole. As drum B rotates, gravity forces each pin into a slot on the drum. Each pin is subsequently dropped on rollers C and D, which turn as indicated by the arrows.

The distance between adjacent high points of the rollers is 0.015 to 0.020 inch less than the diameter of the pins. As the rollers revolve the pin also rotates. When the center line of the cotter hole in the pin is horizontal, the hole end of the pin falls through and the opposite end slides off, as indicated by the dotted lines at Y. Thus all pins leave the mechanism with the cotter hole at the bottom end.

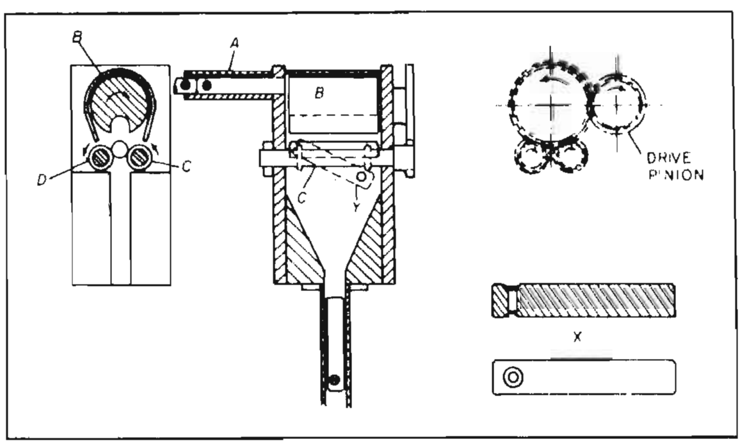


FIG. 22. Mechanism for feeding roller pins to assembly machine with drilled end downward.

CHAPTER 18

Varying Continuously Rotating Output

Some machines require mechanisms that convert uniform rotary motion into variable rotary motion. The mechanisms in this chapter cover this and provide for varying continuously rotating outputs.

Two-Gear Drive Produces Variable Output Motions

Machines often need a drive in which the input shaft turns with uniform angular velocity and the output shaft rotates at a different velocity. Examples are an alternately increasing and slowing of rotation, with stopping; the same type of motion but with the shaft coming to a prolonged dwell; and a varying rotation, interrupted by a reverse oscillation. The two-gear drive shown can produce any of these motions, depending on the motion of a pin that can be readily changed to give the desired output.

Essentially, the mechanism consists of a four-bar linkage A_oABB_o, as shown in Fig. 1. Points A_o and B_o are pivot points on a stationary frame. The input crank rotates around point A_o and carries pivot point A, which is the center of an internal gear. Attached to the internal gear is a pin at point B. A rocker arm — and a pin in the frame at point B_o — pivot on this pin.

The internal gear meshes with an external gear attached to the output shaft.

Center A of the internal gear rotates around A_o, which is the same axis on which the external gear rotates. Consequently, the two gears are always in mesh.

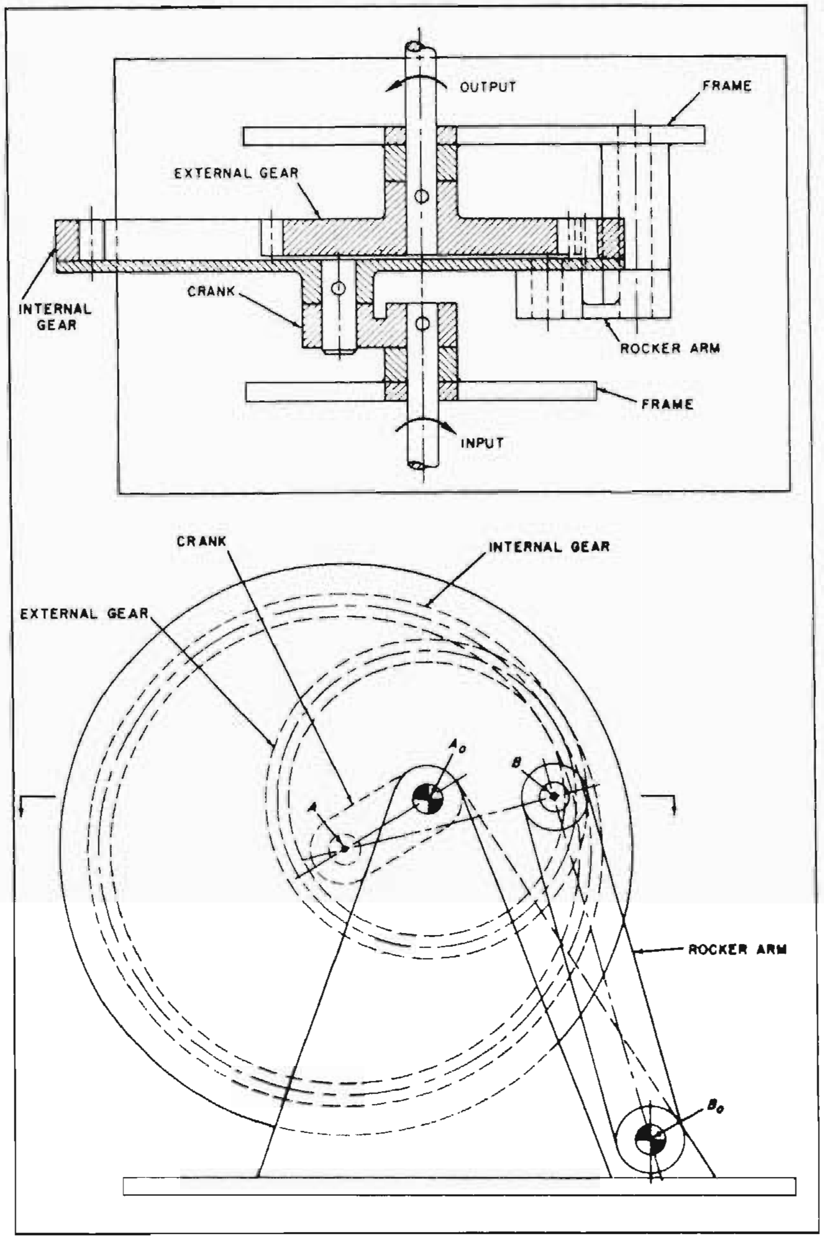


FIG. 1. Crank translates internal gear, causing external gear, with which it meshes, to rotate.

As the crank rotates, the center of the internal gear rotates around point A_0. The pin attached to the internal gear moves in a circular arc around point B_0. The combined motion of points A and B causes the internal gear to rotate and oscillate around the external gear.

Location of the pivot on the internal gear relative to its pitch circle is important. If the pin is outside the pitch circle, the output motion of the external gear is a rotation interrupted by an oscillation. If the pin is on the pitch circle, an instantaneous dwell results. If it is inside the pitch circle, the output motion will still be a rotation but it will alternately speed up and slow down.

Theoretically, dwell is instant. Practically, dwell can be considered prolonged. Two units in series will give an output motion with a larger dwell than one unit. Two units can also give an output with two dwells.

Cam and Link System Produces Varying Rotation Rate

Figure 2 shows two views of a mechanism which converts uniform rotary motion into variable rotary motion, maintaining

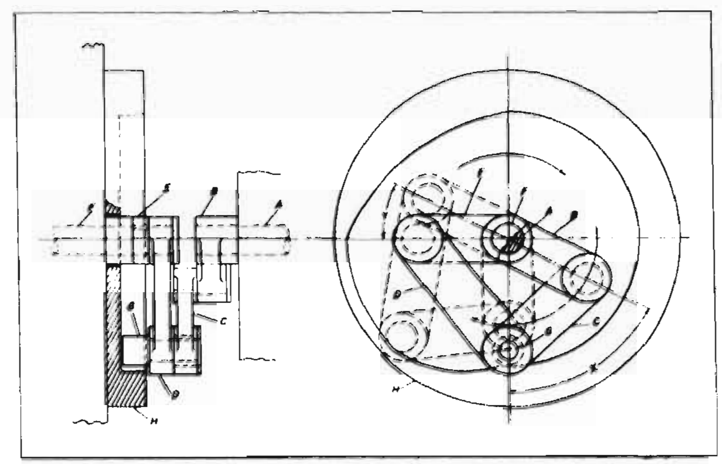


FIG. 2. The result of this link and cam system causes driven shaft F to slow down and then pick up speed with relation to drive-shaft A.

required relationships between the positions of the drive-shaft and the driven shaft. This mechanism was designed for use on a machine on which it is necessary: (1) for the mechanism carried on the driven shaft to have its angular position changed relative to the mechanism carried on the driving shaft; (2) to maintain the changed position for a required portion of the cycle; and then (3) to gradually return the driven shaft to its original relative position.

The driving shaft *A*, rotating uniformly in the direction indicated by the arrow (clockwise), carries the lever *B* keyed to it. Lever *B* carries the link *C*, which is connected to link *D*. Link *D* is connected to lever *E*, which is keyed to the driven shaft *F*. Link *D* carries, at its lower end, the follower roller *G*, which is in contact with the internal cam *H*, attached to a stationary part of the machine with its center coinciding with the centers of shafts *A* and *F*.

Referring to the diagram at the right, the linkage drawn with solid lines indicates the positions of the elements when the rotation and relative angular positions of shafts *A* and *F* coincide. At this point, roller *G* has arrived at the point terminating the concentric surface of the smaller diameter on the inside of cam *H*. Continued rotation of shaft *A* causes roller *G* to ride down the circularly inclined surface of cam *H* to the larger-diameter concentric surface, at which point the links occupy the positions shown by the broken outlines. In so doing, some of the rotary movement applied by lever *B* to lever *E* is lost because of the change in the relative positions of links *C* and *D*. Thus, lever *B*, in rotating the angular distance *X*, has transmitted motion to lever *E* to a lesser extent, as indicated by the angle *Y*.

Shaft *F*, now in a changed position relative to shaft *A*, remains in this position until the rotation of shaft *A* brings roller *G* to the end of the concentric cam surface of the larger diameter of cam *H*, which in this case is on the horizontal center line. From this point on, there is a gradual change in the relative positions of shafts *A* and *F* due to roller *G* riding up the inclined cam surface to the larger-radius concentric surface. This change in shaft positions is in the opposite direction to that previously produced, so that when roller *G* reaches the vertical center line, shafts *A* and *F* are again in their original relative positions, remaining so for a half-cycle when the levers and links will again be in the position shown by the solid outlines.

Adjustable Drive Produces Continually Variable Rotation

A chain-operated drive which converts uniform rotation into variable rotation and can be adjusted while in operation is shown in Fig. 3. The purpose of the drive is to increase the rotative speed of the driven shaft for a portion of each revolution so that the dwell of a cam may be changed.

Driven shaft *A* is keyed to a gear *B* and passes through the bore of a disc *C*. The hub of member *C* is free to rotate in a pillow block *D*. A collar *E* retains disc *C* in the pillow block, and a second collar *F* holds shaft *A* in disc *C*. Mounted on the same disc is a sprocket *G* that is rotated uniformly by a drive chain. In addition, disc *C* has a groove that carries a plate *H*, which is free to slide in this groove.

Gear *B* rotates in a rectangular section removed from plate *H*. One side of this cutout in plate *H* has rack teeth which mesh with gear *B*, while the opposite side of the cutout is machined to clear the gear. Plate *H* also carries a roller follower *J* at one end. A block *K* is mounted on the bed of the machine and is grooved to accommodate a bar *L*. This bar is free to slide in block *K* but is retained in the groove by two blocks *M*.

A stud *N* is secured to bar *L*, and is drilled and tapped to receive a screw *O* which, in turn, is supported by blocks *M*. Bar *L* also carries a ring *P* and a disc *Q* mounted concentric with the ring. These members are so spaced that the roller follower *J* can move freely around the circular path between them. In the position shown, bar *L* with ring *P* and disc *Q* has been moved horizontally to the right by means of screw *O*, placing the center of disc *Q* a distance *X* from the center of shaft *A*.

If disc *Q* and ring *P* are moved to the left so that their common center coincides with the center of shaft *A*, the rotation of sprocket *G* is transmitted through disc *C* and plate *H*, causing

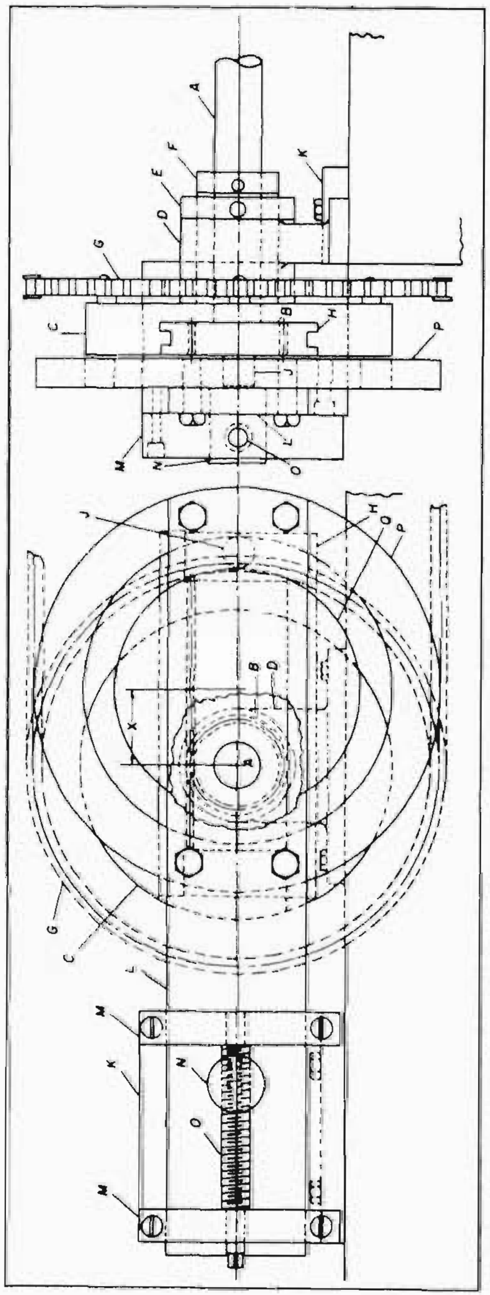


FIG. 3. This drive which produces continually variable rotation can be adjusted while in operation. When center of disc Q coincides with that of shaft A, however, no variation in the drive is obtained.

roller follower J to travel in a circular path concentric with the center of shaft A. The rack teeth on the inside of plate H that mesh with gear B simply act as a key to transfer the rotation to gear B. Thus, the rotation of gear B and shaft A are then synchronized with that of sprocket G.

When plate L is moved horizontally to the right toward the position shown in the illustration, roller J no longer travels in a path concentric with the center of shaft A. Instead, the roller alternately approaches and recedes from the center of shaft A, thus imparting a reciprocating motion to plate H and its rack. The magnitude of this reciprocation is controlled by the adjustment of screw O. Reciprocation of the rack produces a gradual and alternating increase and decrease in the rotational speed of driven gear B relative to that of the driving member. In this manner a continually varying rotational motion is imparted to shaft A by the uniformly rotating sprocket.

This mechanism is not limited to producing a uniformly variable rotation of the driven shaft. By changing the shape of disc Q and ring P, a variety of movements may be obtained in the same manner as from any cam.

Coupling Connects Displaced Shafts

Rotary motion can be transmitted between shafts with a considerable degree of displacement, using the coupling shown in Fig. 4.

This coupling can be used for high-speed and high-torque applications. Inherently, it is dynamically balanced. Angular velocities of the driving shaft are identically reproduced in the driven shaft and no phase shift is produced when shaft displacement is changed.

The drive-shaft is keyed to a disc. Three eccentric, equally spaced pins extend from the face of the disc. The driven shaft has an identical disc and pin arrangement, but is parrallelly displaced from the drive-shaft. A connecting or intermediate disc has three cylindrical pins that extend from each side face and have the same spacing as pins in the other two discs.

Two sets of three links act as connecting members between the discs. These links are of the same length and are less the diam-

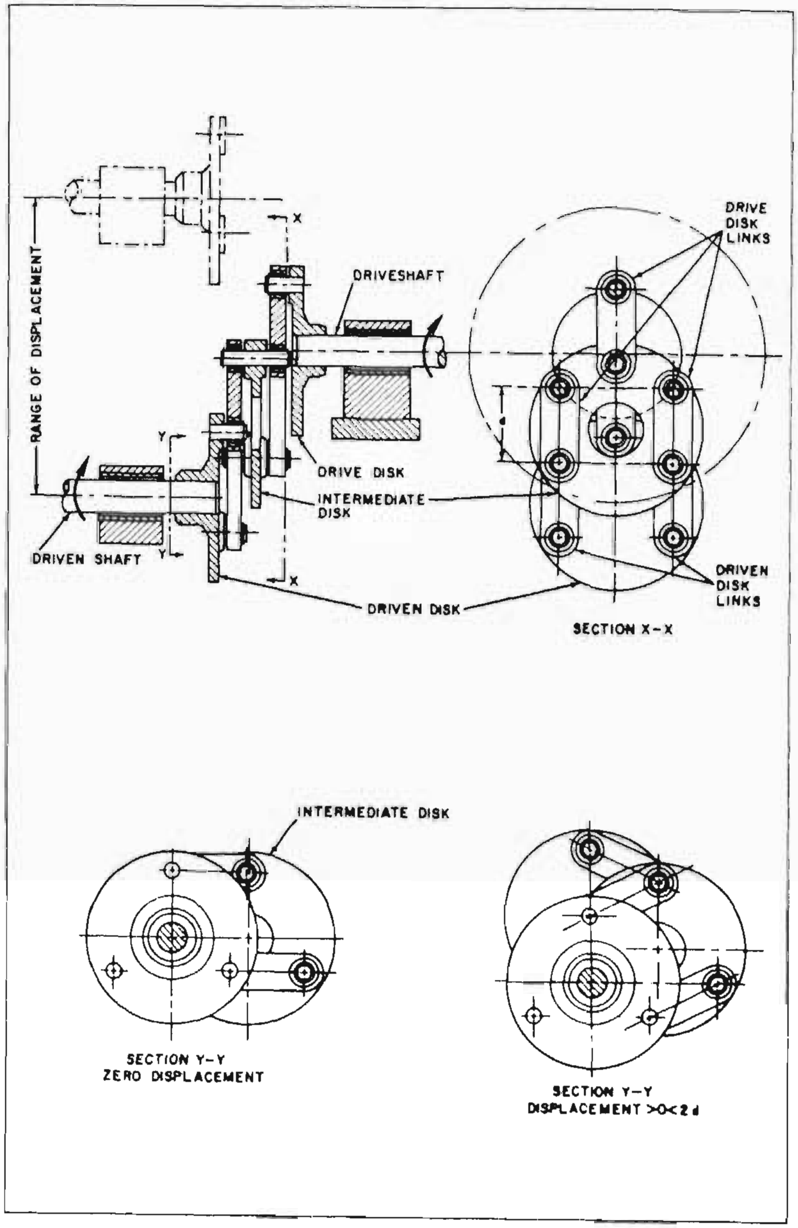


FIG. 4. Constant rotation is transmitted through coupling connecting displaced shafts.

eter of bolt circle formed by the pins. Each has two equally spaced bearings that engage the corresponding pins in adjacent disc faces.

Maximum radial displacement of the shafts is determined by the distances between bearings in the links, multiplied by the number of link sets. Since two sets of three links each are used, the maximum possible radial shaft displacement is $2d$. If the shafts are in the fixed position shown in section Y-Y and the shaft displacement is greater than zero and less than $2d$, the intermediate disc is suspended from the links between the other discs.

The pivot points of the two sets of links form three triangles which are geometrically determined by the length of the links and the fixed parallel displacement of the shafts.

Since the intermediate disc is suspended by three pins located at geometrically determined points, the center of the intermediate disc is also geometrically determined and cannot be moved unless the parallel shaft displacement becomes zero and the centerlines of the pins in the outer disks fall together. In this case the intermediate disc is free to swing about the centers of these pins. For practical applications, the zero-displacement position should be avoided.

When varying the displacement of the shafts, the intermediate disc automatically adjusts its position in compensation. The shafts can be displaced under loads.

Cam-Controlled Differential Mechanism

In a punched-card machine it was necessary to include a mechanism that would convert a uniform rotary input motion into a varying rotary motion. The mechanism shown in Fig. 5 proved very successful in comparison with other mechanisms designed to perform the same job.

In the preferred mechanism, the two shafts are geared together in the ratio of 1 to 1 by an arrangement that is not shown. On shaft A is fastened a cam C and its complementary cam D. These two cams drive arm T through the rollers R and S. Arm T carries gear E, which is driven by gear I. The arm also carries

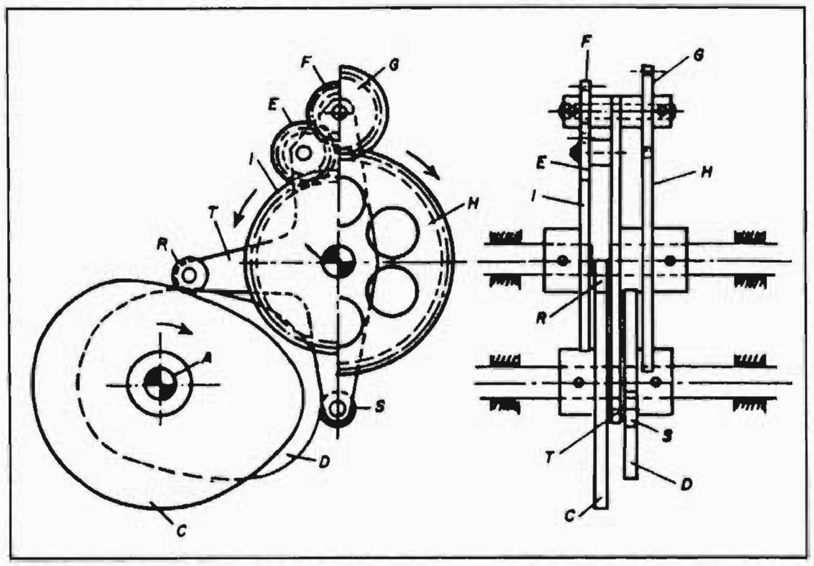


FIG. 5. Derivation of varying rotary motion from uniform rotary motion.

gear F, which is in mesh with gear E, and gear G, which is keyed to the same shaft as gear F.

As gear I rotates, it transmits motion through the gear train E, F, G, and H with a constant angular velocity. However, because of the oscillating motion of arm T, gear E will roll on gear I. The result is that gear H will move with a nonuniform velocity according to the shape of the curves on cams C and D.

Cams C and D impart a constrained movement to arm T. If only one cam were employed, springs would be necessary to hold the follower on arm T in contact with the cam.

Timing of Cam Changed While Machine Is in Motion

It was required that the timing of cam A (see Fig. 6), be changed while in motion. Drive-shaft B rotated the keyed gear F. Gear F rotated pinion G, supported by the normally stationary gear C. Pinion G, in turn, rotated gear L, through pinion J, in the opposite direction to shaft B. The timing of cam A is

adjusted by rotating pinion D, resulting in gear C rotating, say X degrees. The attached pinions will rotate X degrees and the cam A which is attached to gear L, will rotate 2X degrees as gear L has the same pitch diameter as F and the pinions are of equal pitch diameter.

Unidirectional Rotation Regardless of Changes in Drive Direction

Occasionally it is necessary to drive a shaft in only one direction even though the driving member may alter its own direction of rotation. In designing the drive mechanism for a recorder drum, such a situation arose. The drum was required to maintain a single direction of rotation regardless of the fact that its driving member fluctuated between a clockwise and a counterclockwise rotation.

A clutch and bevel gear arrangement, Fig. 7, consists of three bevel gears; gear A being the driven member, and gears B and C being the driving members. Both driving gears are mounted on roller clutches D which are, in turn, mounted on a common drive-shaft E. The clutches provide positive drive when rotated

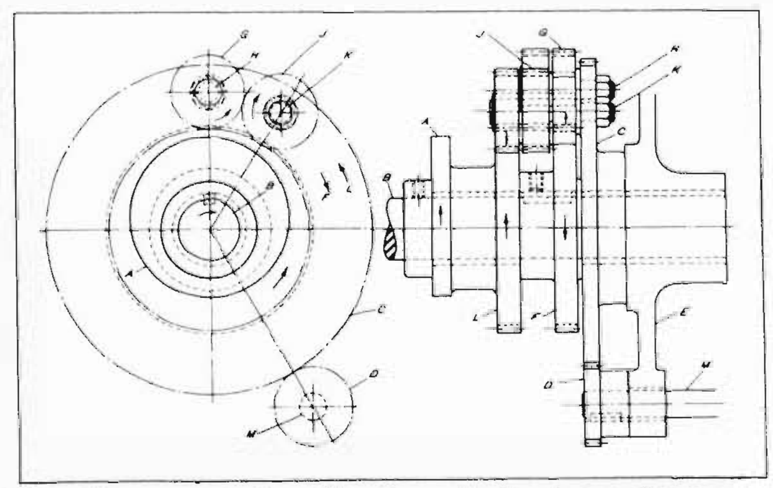


FIG. 6. The gear train from F to L permits the timing of cam A to be changed while it is rotating.

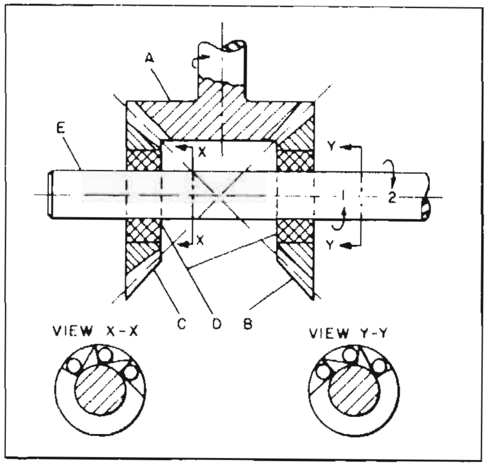


FIG. 7. Two bevel gears mounted on roller clutches *D* provide unidirectional rotation for driven gear *A* regardless of rotational direction of drive-shaft *E*.

in one direction, and overrun, or "free-wheeling," with reverse rotation. Each of the two clutches in this set-up is mounted in opposite directions from one another; hence bevel gears *B* and *C* cannot drive in the same direction.

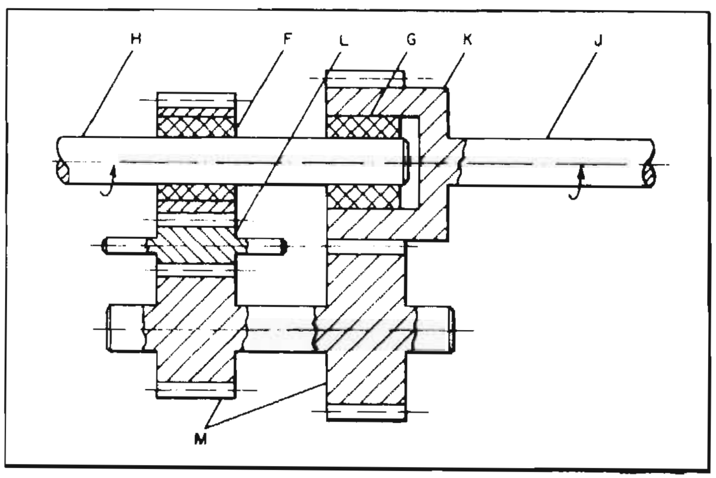


FIG. 8. Spur gear set-up designed to accomplish the same purpose as bevel gearing shown in Fig. 7.

When the drive-shaft rotates in the direction indicated by Arrow 1 (at the right-hand end of drive-shaft), the clutch in gear *B* engages while the clutch in gear *C* slips, thus driving gear *A* as indicated. Upon reversal of the direction of drive-shaft rotation to that indicated by Arrow 2, the operation of the clutches is also reversed. In this way, the direction of gear *A* is unchanged.

A set-up employing spur gears to obtain the same end result may be seen in Fig. 8. Here again, two roller clutches *F* and *G* are mounted on common drive-shaft *H*. In line with this is driven shaft *J*, on the end of which is an integral external spur gear *K*. Gear *K* is counterbored to fit over the outside diameter of clutch *G*.

As the drive-shaft rotates in the direction indicated by the arrow, clutch *F* slips while clutch *G* engages, thus forcing the driven shaft to rotate as shown. When drive-shaft rotation is reversed, clutch *F*, which is mounted within the bore of a spur gear, engages as clutch *G* slips. Motion is transferred through idler gear *L* to intermediate gears *M*, and finally to driven gear *K*, thereby causing the direction of rotation of drive-shaft *J* to remain the same.

Timing of Driven Shaft Changed While Running

A power train through sun gears which rotate within a stationary, but adjustable ring gear permits the timing of a driven shaft to be changed while it is running. The arrangement is used in positioning revenue stamps correctly in a tobacco-packaging machine.

Ordinarily, driven shaft *A* (see Fig. 9), and driving shaft *B* revolve in unison. The transmission is as follows: Pinion *C*, integral with the continuously revolving driving shaft *B*, meshes with two sun gears *D*. These sun gears are the same size as the pinion. They are free on studs *E* fixed in diametrically opposite points in disc *F*. Near the center, the disc forms a hub which is pinned to driven shaft *A*. Both sun gears mesh with stationary ring *G*. Thus, as shaft *B* revolves, the sun gears re-

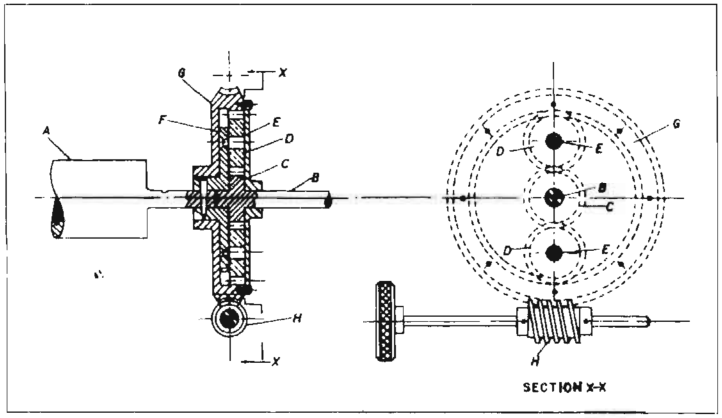


FIG. 9. When the ring gear G is rotated by worm H, the timing of driven shaft A changes with respect to the driving shaft B.

volve on their studs, and in addition they rotate within the ring gear, causing the disc to turn shaft A.

For changing the timing of shaft A with respect to shaft B, the outside of the ring gear is machined to a worm-wheel. The changes of angular displacement of the disc with respect to the pinion, and advances (or retards) the driven shaft the required amount. No stoppage of the mechanism is made, and since the worm is self-locking, it will preserve the timing, once set.

Mechanism for Varying Rotation of Winding Mandrels

The mechanism illustrated in Fig. 10 is designed to vary the rotation of two mandrels on which paper tubes are wound. Paper from the web is wound first on mandrel A and then on mandrel B. The tube thus produced on one mandrel is stripped off while a tube is being wound on the other mandrel. A completed tube consists of a given number of full turns of paper plus a fraction of a turn as indicated at a in diagram X. The means for applying adhesive to the proper portion of the web for securing the lap section of the rolled tube is not shown.

In order to remain synchronous with the other parts of the machine, the mandrels can only make a specified number of full

turns. Therefore, a driving mechanism for the mandrels had to be designed which would give the required number of full turns plus a fraction of a turn a on the winding cycle of each roll without increasing the total number of complete revolutions. To accomplish this, it was necessary for the mandrel being stripped to have its rotation slowed to compensate for the fractional rotation provided for the lap a before starting to wind a new tube.

The mandrels A and B, and their respective worm-wheels C and D, are driven by worms E and F. The worm-wheel C and worm E are right-hand, while the worm-wheel D and worm F are left-hand. The two worms E and F are made integral, and are slidably mounted on their driving shaft M.

The extra rotational movements of the mandrels are obtained by axially reciprocating the worms. A shaft G, which makes one revolution for each complete winding cycle of the two mandrels,

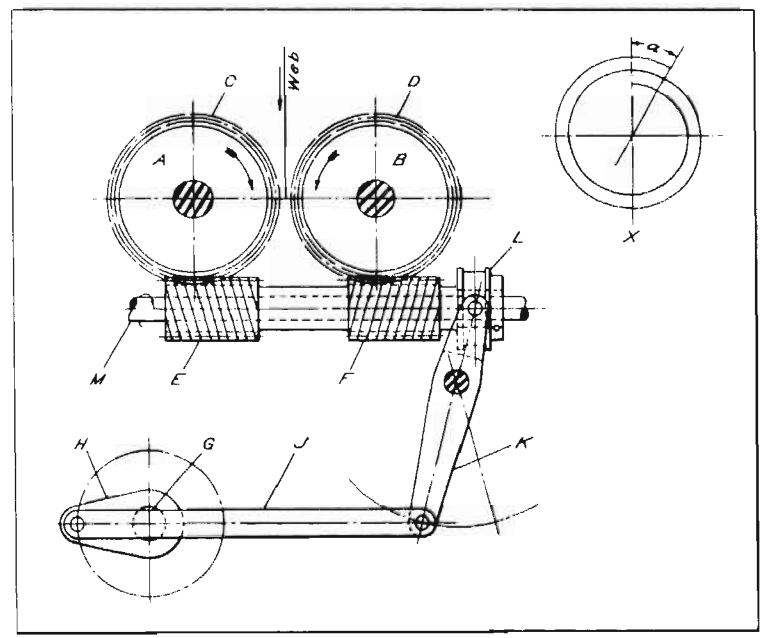


FIG. 10. Mechanism designed to vary rotation of mandrels used to wind paper tubes.

provides the drive for the worm-reciprocating movement. On shaft G is a crank H of appropriate length, which is connected to lever K by link J. The forked end of lever K carries shoes that ride in a groove provided for them in a collar L. Collar L is pinned to a hub at the outer end of worm F.

When the machine is in operation, the web is introduced between the mandrels, as illustrated, and is brought into contact, by means not shown, with, say, mandrel A, to which it is held by vacuum. Mandrel A rotates in the direction indicated, winding the paper around it the required number of laps. At the same time, the worms move to the left a sufficient distance to give the required extra part of a revolution. It should be noted that although mandrel A is given the additional rotational movement, this same amount is subtracted from the rotational movement of mandrel B. Thus it will be apparent that the overlap a is the sum of the under speed of the beginning of the winding cycle and the over speed at the end. This does not affect the winding operation, since B is being stripped of its previously wound tube during this decelerating part of the cycle.

At the proper point in the cycle, the web is cut (by means not shown) and the leading edge then brought into contact with the opposite mandrel. The axial movement of the worms is reversed on the return stroke of link J at the proper time; and the worm F, moving in the opposite direction, gives mandrel B the additional rotation required for the lap joint.

Worm Drive Gives Variable Output Automatically

Figures 11, 12 and 13 illustrate the construction of a mechanism designed for use on a machine that makes a woven wire product in which the strands of wire are spaced to a specific pattern. In order to produce the required pattern, the mechanism which processes some of the wire strands operates at a uniform rate of speed, while other parts of the mechanism must operate at varying speeds. In the mechanism shown, a shaft rotating at a uniform speed transmits motion to another shaft,

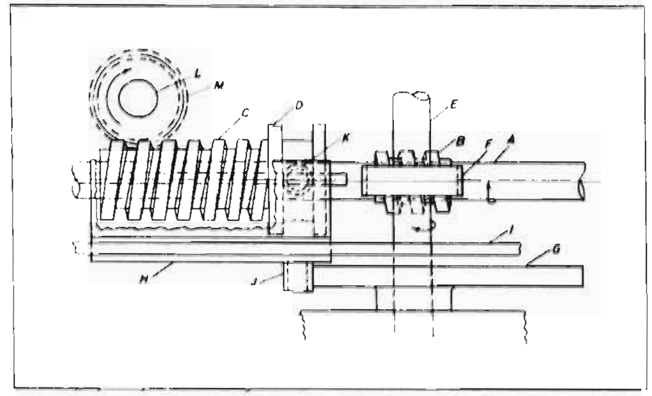


FIG. 11. Front view of speed changer with the cam G at the mid-point of its dwell.

in synchronism during part of the cycle, followed by an increase of speed, and then by a period of rest, which returns the two shafts to the original relative rotation. Figure 11 is a front view, Fig. 12 a plan view, and Fig. 13 an end view of the mechanism.

Drive-shaft A rotates in the direction indicated by the arrow and carries worm B, attached to it. Grooved collar D and worm C are fastened to each other as shown and are arranged to slide axially on shaft A, their rotation being provided by an internal

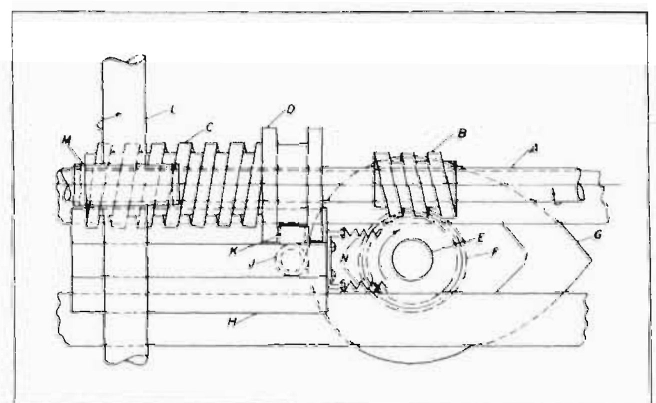


FIG. 12. Variable-speed worm-gear drive, plan view, showing slide-block H, which moves pusher worm C to left in a rack action.

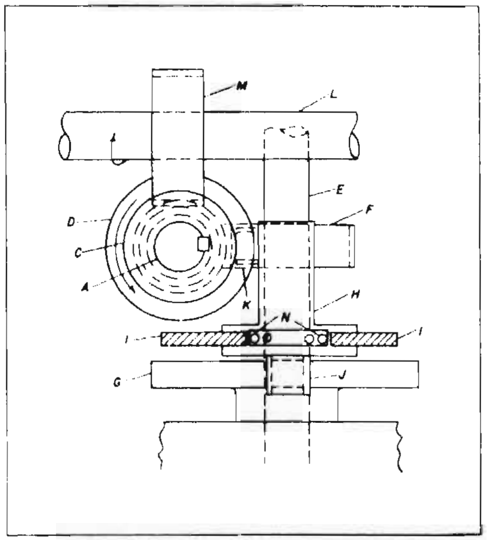


FIG. 13. The relationship of the slide-block H and its springs N is shown in this end view of the device.

key sliding in a groove in shaft A. Worms B and C have the same lead, except that B is right-hand and C is left-hand.

Shaft E, which is free to rotate, is mounted on a stationary part of the machine and carries worm-gear F, in mesh with worm B, and cam G. The contour of cam G is shown in Fig. 12. It is laid out to produce a 240-degree dwell on the base circle, 60-degree rise and 60-degree fall, with a rise equal to three times the lead of worms B and C.

Driven shaft L carries worm-gear M in mesh with worm C. Worm-gears F and M have the same number of teeth, so that shafts E and L have the same rotative speed. Block H has a sliding mount on two bars I, which are fastened to a stationary part of the machine. Block H also carries two rollers. The first is K, which engages the groove in collar D; and the second roller is J, which contacts the cam G. Two coil springs N hold roller J in contact with cam G.

In the position shown in Figs. 11 and 12, roller J is in contact with cam G at the mid-point of the dwell period. Rotation of

shaft A transmits rotation to the shaft E in the direction shown. At this stage in the cycle, no motion is transmitted to slide-block H. Shaft A also transmits rotative motion to shaft L through worm C and gear M, in the ratio of the number of M's teeth.

With the continued rotation of shaft E, the rise section of cam G contacts roller J and motion is thus imparted to the slide-block H. The movement of slide-block H is transmitted to collar D and worm C through the roller K. In addition to transmitting rotary motion to shaft L, the axial movement of worm C causes it to act as a rack to increase the speed of rotation of shaft L. Because the rise of cam G is equal to three times the lead of worm C, gear M turns through three teeth in addition to the normal worm rotation in the same period of time.

After cam G has reached the peak of its rise, springs N draw slide-block H back to the starting point at a rate of speed controlled by the contour of cam G. This movement is at a rate which permits slide-block H to move a distance equal to one lead of worm C with each rotation of shaft A. Therefore, no rotation is transmitted to shaft L on the return movement of block H.

With this mechanism, the relative rotation of shafts A and L remains unchanged over a 240-degree rotation of cam G. The rotation of shaft L is doubled over the 60-degree rise of cam G, and remains stationary over the 60-degree fall of cam G, so that when the cycle is completed drive-shaft A and driven shaft L again assume their original relative rotation.

CHAPTER 19

Clutch and Disconnecting Devices

Clutches and disconnecting devices which play an important role in automatic and semiautomatic machinery are described in this chapter.

Slip Clutch Suitable for Tape Drives

Many applications present themselves for a slip clutch that will not change its rating, can be infinitely adjusted, is not sensitive to speed changes, and will not wear out. One use for such a clutch is in a tape drive. Drives for both punched-paper and magnetic tapes can use this clutch. Clutches of various sizes and ratings can be built in the design shown in Fig. 1.

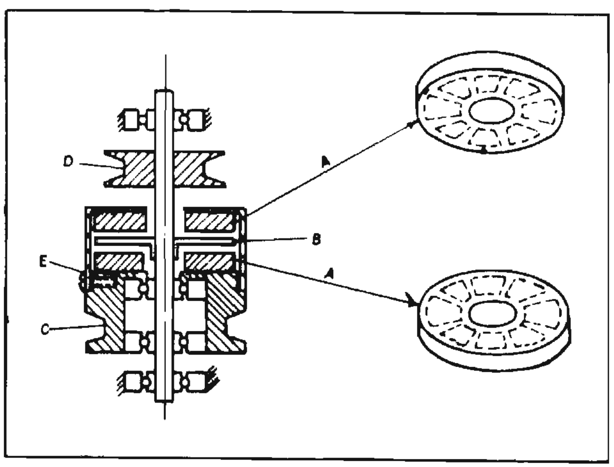


FIG. 1. Nonwearing slip clutch especially suitable for tape drives.

The clutch consists of two ceramic magnets *A* with eight separate poles in each of the opposing faces. North and south poles are alternated on the magnet faces. These magnets are mounted on the driver pulley as indicated. A disc *B* of high-hysteresis cobalt steel runs between the two ceramic magnets on suitable bearings. The output pulley *D* is driven by the disc. Power is, of course, applied through input pulley *C*.

Adjustment is achieved by rotating one of the magnets in relation to the other. Screw *E* passes through a slot in the shell of the input pulley. The highest torque is obtained when unlike poles of the magnets are opposite each other. The drive results from the magnetic path, established in disc *B*, resisting change.

Driving Mechanism Prevents Reverse Movement of Driven Shaft

Feed arrangements for a special wire-forming machine required intermittent, unidirectional rotation of the feed-drive shaft. Any shaft reversal during dwell periods in the feed cycle had to be avoided. To meet these demands, the drive mechanism, shown in Fig. 2, was developed.

Wire feed spool *A* is keyed to driven shaft *B*, as shown at X (section Z-Z). The shaft runs freely within the stationary, circular bearing bracket *C*. A clockwise rotation (indicated by the arrow) is imparted to the spool by pulley *D*, keyed to the shouldered end of the driven shaft. A nut and washer retain the pulley in position.

A counterbored recess is machined in the upper surface of bracket *C*. Cast-iron circular plate *E* is mounted at the base of this recess, a counterbore in the bottom face of the plate forming a running fit over an internal hub on the bracket. The through hole in plate *E* is bored to be a running fit with driven shaft *B*.

A deep slot is cut diametrically across the upper surface of plate *E* to receive sliding bar *F*. Two diagonally opposed sides of this slot are relieved for clearance. A short elongated slot, located approximately in the center of the sliding bar, a slip fit over the driven shaft. Headed steel pin *G* is driven through a

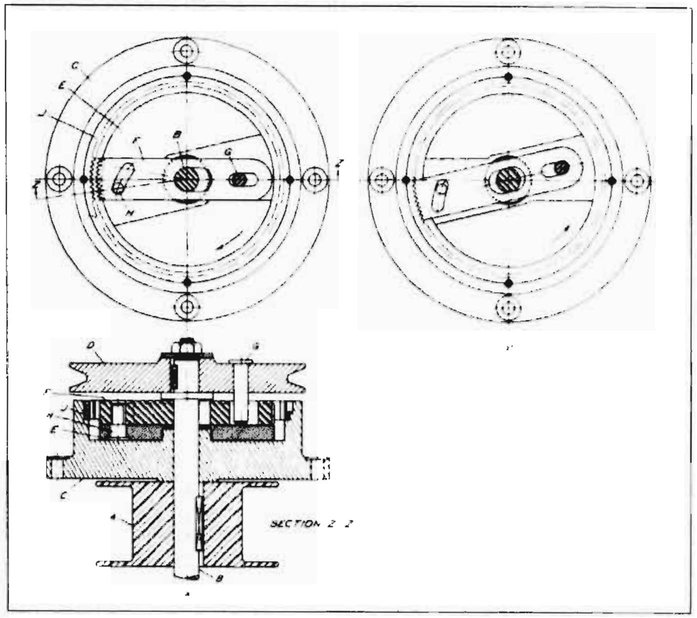


FIG. 2. Drive mechanism arrests any reverse movement of driven shaft B.

hole in the web of the pulley and projects into an elongated slot in the right-hand end of sliding bar F. At the opposite end of the bar is a short arcuate slot. Engaging this slot is the projecting portion of shoulder-pin H. This pin is pressed into a hole that has been drilled in circular plate E.

Cut across the left-hand end of the sliding bar is a series of hardened, fine-pitch vee serrations. A case-hardened steel ring J, having internal vee serrations matching those of the sliding bar, is pressed and pinned into the largest diameter step of the recess in bearing bracket C.

The diagrams at X in the illustration show the relative positions assumed by the various components while shaft B is being driven in its normal (clockwise) direction of rotation. Pulley D, sliding bar F, and circular plate E rotate in unison, being connected by pins G and H. The curved slot bearing against pin H serves to hold the sliding bar to the right, thus allowing free movement of all the components.

When pulley D is at rest, any movement of spool A in the reverse direction will be instantly arrested. Such a movement will be transmitted to the pulley and thus to pin G. The pin, in turn, will cause bar F to rotate counterclockwise for a short distance. As this occurs, and the left-hand slot is caused to slide in contact with pin H, the bar will be forced to the left until it engages the vee serrations on ring J. In this way, the drive mechanism is positively locked, as shown at Y, and will remain so until normal rotation is resumed.

Two-Revolution Clutch

In designing a certain mechanism, provision had to be made to rotate a drive-shaft two complete revolutions and then to stop the shaft. The accompanying drawing shows the means by which this action was obtained. All rotating parts are of low inertia. The full lines in Fig. 3 show all parts in their locations when the mechanism is stopped. However, the driving member A rotates constantly in the direction indicated by the arrow.

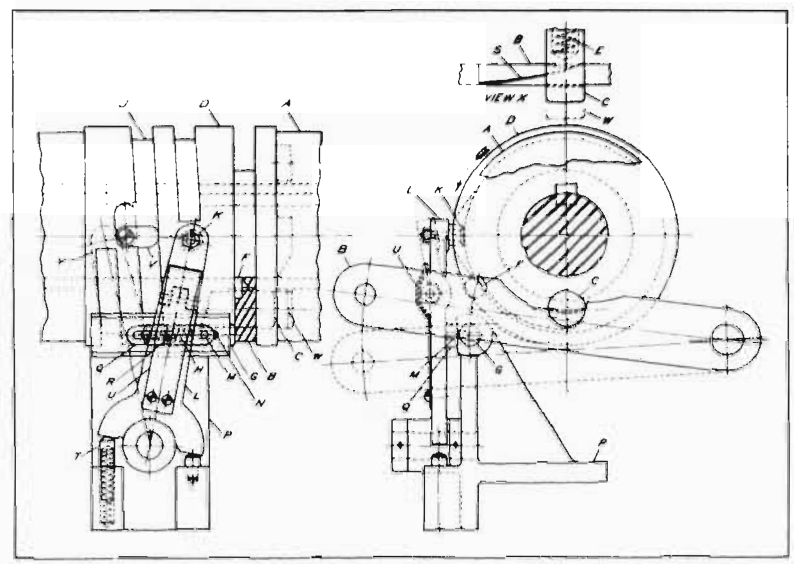


FIG. 3. Clutch which makes two complete revolutions and then stops accurately.

When the mechanism is in motion, a link, which is not shown and which is attached to the left-hand end of stop link B, moves the latter from the full-line position to the dotted location. This movement permits driving plunger C, carried in clutch body D, to move forward under the pressure of spring E (view X) until it bears against the face of driver A. This position is maintained until one of four cam-like recesses registers with plunger C, whereupon spring E forces the plunger to enter the recess as indicated at W. Clutch body D and driving member A then revolve together.

An outstanding feature of the device is the means employed to hold link B out of the path of plunger C until the second revolution has started. To hold link B in the lowered position, a hole F is provided in the side of the link that faces the clutch body. When link B has been lowered to release plunger C, the hole registers with plunger G and the latter is snapped forward into the hole by means of spring H.

Clutch body D now revolves with member A. A right-hand square thread is cut in the outer surface of body D at J. This thread terminates at each end in a slope leading to the circumference of the clutch body, the total thread being somewhat less than two complete turns.

When the device is stopped, roller K on operating lever L rests on the outside surface of body D. When rotation starts, this roller enters thread J and causes lever L to move to the left. (Note: In end view, lever L is shown vertical and does not correspond with the side view. This was done to show the parts more clearly.)

Plunger G carries a pin M which passes through slot N in support P. The pin engages one end of link Q. The other end of this link has a slot which is engaged by pin R on operating lever L. As the operating lever L moves to the left, pin R moves along the slot in link Q.

As the first revolution of body D is completed, lever L assumes a vertical position; driving plunger C is at the bottom center; and stop link B is held in the out position by plunger G. This permits the start of the second revolution.

Sometime after the second revolution starts, pin R will reach the end of the slot in link Q. Further movement of the pin and link to the left will move plunger G until it disengages from hole F. Link B is thereupon returned to its original position by the action of a spring. Then, when plunger C reaches link B, a wide slot on the plunger engages the thin portion of the cam surface S (view X) on the link, forcing the plunger out of engagement with the recess in body A. Inertia will carry the rotating parts until plunger C engages the stop portion of link B.

Upon the clutch body D having completed two turns, operating lever L will have moved to the extreme left as shown by dotted lines at Y, and roller K will have risen to the outside surface of body D. Operating lever L is now free to be moved by means of the spring plunger T to its original position.

In order to permit roller K to rise from the engaged position with the thread to the outside surface of body D, the upper portion of lever L is hinged as shown. This upper portion is forced into its operating position by flat spring U. Swinging block V permits roller K to pass over the central portion of the thread groove in returning to the starting position. This block is held in position across the thread groove by a torsion spring that surrounds its pivot screw. When roller K passes this point, riding in the groove, block V is rotated about its pivot screw into a pocket, as shown by dotted lines, permitting the roller to pass. All parts are now in position for another two-revolution cycle. Stop link B and bracket P are carried on a frame not shown.

Intermittent Rotation for Instrument Pointers

A mechanism that converts the continuous rotation of a shaft to the intermittent rotation of a stop-wheel is shown in Fig. 4. In the application illustrated, the angular position pointer A follows the shaft B with a quick motion each time the shaft completes a 60-degree rotation.

Basically, the mechanism consists of a clutch that engages and disengages automatically by the axial movement of one of

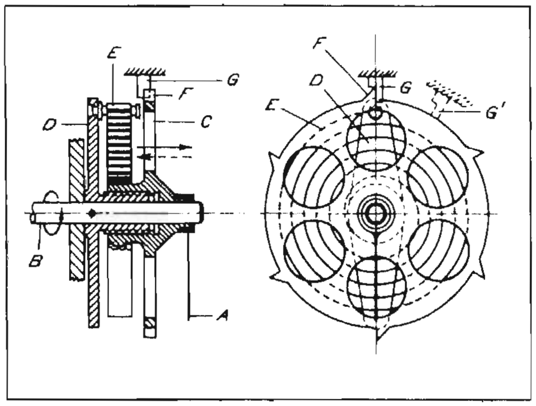


FIG. 4. When shaft B rotates, stop-wheel C moves axially until lobe F clears stop G. The stop-wheel then rotates 60 degrees.

the parts. This axial movement is derived from the continuous rotation of the shaft by a screw and nut action.

Stop-wheel C, carrying the pointer, is mounted freely over shaft B. Also over the shaft, but pinned to it, is an arm D. An integral stud on the arm is threaded externally, and engages an internal thread on the stop-wheel. Fastened to, and wound around the hub of the stop-wheel is a flat coil spring E. The outer end of this spring is connected to the arm. As the shaft and arm rotate (in the direction of the arrow on the shaft) the spring attempts to transmit this motion to the stop-wheel. However, this member is prevented from turning by the blocking of one of the lobes F on the wheel by a stop G. Instead, the wheel moves axially to the right (as indicated by the solid arrow) because of its threaded engagement with the arm. Then, when the stop-wheel has moved out sufficiently to clear the stop, the tension built up by the spring causes the wheel to rotate faster than the shaft and thus reverse its axial direction (the broken-line arrow).

Since this particular stop-wheel has six equally spaced lobes, it will rotate 60 degrees before the next lobe abuts the stop. Double frequency can be obtained by adding a second stop, G'.

The mechanism can be utilized in applications where the rotation of the input shaft itself is intermittent rather than continuous, since there will be a corresponding effect on the stop-wheel. For simplicity, the pointer is shown carried directly by the stop-wheel. However, if the angular movement of the pointer has to be increased or decreased, a gear transmission can be added.

Simple Device Allows Manual Overriding of Remotely Controlled Lever

Rod C (see Fig. 5), is reciprocated by extension B of split bushing A which is clamped to oscillating shaft D. To disengage A from D, arm F, which is attached to bearings G located in a recess in A, is rotated manually. Cam E then forces split bushing A apart, causing disengagement from shaft D. Further rotation of F will cause rotation of A.

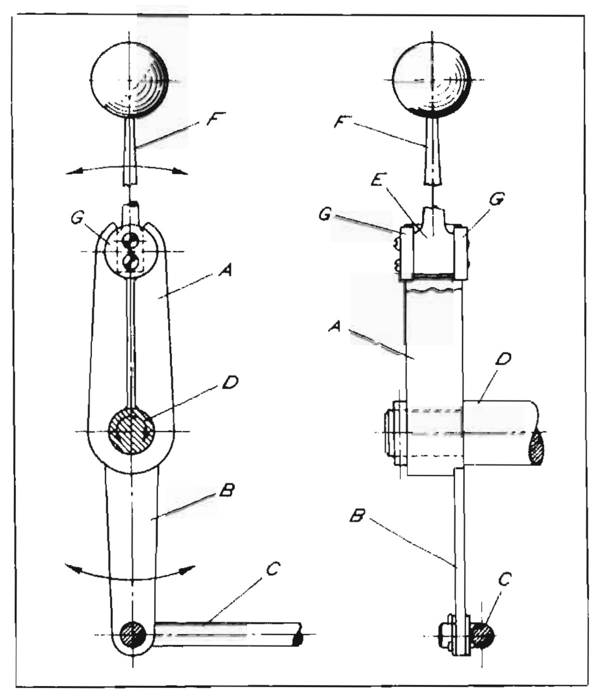


FIG. 5. Split bushing arrangement permits lever actuation by either manual or remote control.

One-Half Revolution — One-Half Pause Mechanism

A one-half turn trip clutch designed to operate two machine heads used for simultaneously twisting eyelets on wires is shown in Fig. 6. Fundamentally, this mechanism is based on a stationary cam and a pivoted latch which control the action of a pair of gears. The mechanism has proved successful in the application mentioned. The capability of the mechanism in driving several units is a decided advantage, and the arrangement has long-wearing characteristics.

The driven shaft makes a one-half revolution with the drive-shaft and then remains idle during the next one-half revolution of the drive-shaft. The mechanism indexes accurately and is suitable for operation at speeds of up to 30 rpm.

The stationary cam *A* is mounted on a short tube *B* which is held tight in housing *C* of the machine. A key prevents rotation of both the tube and the cam. Shaft *E* drives the entire mechanism. Keyed to the drive-shaft is a disc *F* which revolves at all times. Latch *G* is pivoted on this disc. Fastened to the

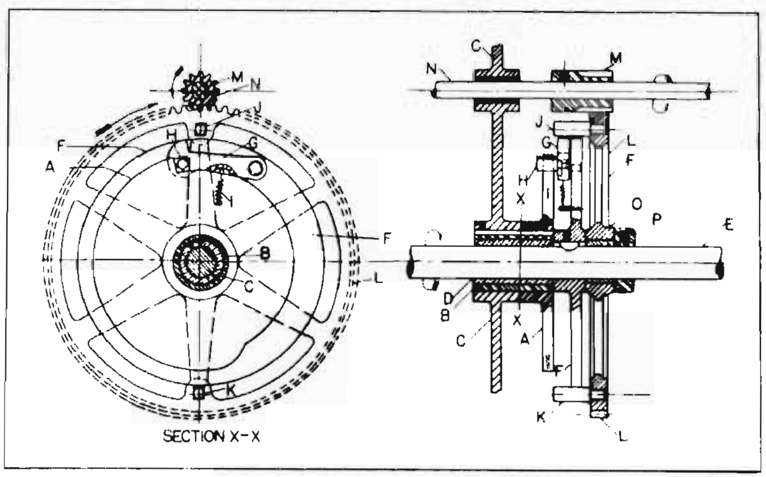


FIG. 6. Mechanism which imparts a one-half revolution to a driven shaft for a one-half revolution of the drive-shaft and then holds the driven shaft stationary as the drive-shaft completes its revolution.

latch is a spring *I* which is also secured to a pin on disc *F*. This spring tends to draw latch *G* toward the center of the mechanism.

Two pins *J* and *K*, respectively, are mounted diametrically opposite to each other on gear *L*. This gear turns one-half revolution during a one-half revolution of shaft *E* and then remains stationary during the remaining one-half revolution of the drive-shaft. Pinion *M* meshes with gear *L*, the gear having 120 teeth and the pinion having 12 teeth, so that in the one-half turn of the gear, the pinion makes five complete turns. The pinion is mounted on the driven shaft and drives it whenever gear *L* is in motion. Bronze bushing *O* is free to rotate on shaft *E*.

The operation of this mechanism is as follows. Shaft *E* drives disc *F* through a key. When latch *G* reaches the bottom of the cam, dog *H* forces the latch to swing radially outward so that the end of the dog engages pin *K* and carries this pin to the position shown occupied by pin *J*. At this point dog *H* rides off the high section of cam *A* and spring *I* pulls the latch inward to clear the pin that has just been indexed to the position shown occupied by pin *J*. The gears are then stationary until the latch is again carried to the bottom of the cam and the dog *H* contacts pin *J*, which has previously been indexed to that position. Gear *L* is then again given a one-half revolution.

Automatic Half-Nut Release for Thread Cutting

A mechanism that permits easy manual engagement and automatic disengagement between the half-nut and the lead-screw of a lathe is illustrated in Fig. 7. With this arrangement incorporated in the lathe apron, screw threads can be cut close to shoulders at a rapid rate.

A single half-nut is carried on the unrestrained end of arm *A*, and is swung downward to engage the lead-screw by an upward movement of hand lever *B*. Any positional change of lever *B*, which is conveniently mounted on the front of the apron, is transmitted to the half-nut by pin *C*. This pin is attached to

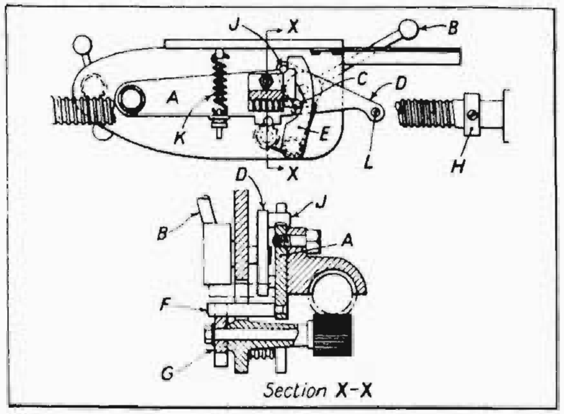


FIG. 7. Lever B is raised manually to engage the half-nut with the lead-screw, and adjustable collar H trips extension L for automatic release.

plate D and engages with a slot in the end of arm A. Plate D is connected to lever B by a shaft that passes through the apron. When the half-nut is engaged with the lead-screw, pawl E is turned counterclockwise by a spring to interlock with the upper end of arm A. In this manner, the half-nut is held in positive engagement with the lead-screw.

Referring to section X-X in the illustration, engagement of the half-nut is controlled by disc G which operates in conjunction with pin F on arm A. Disc G has two diametrically opposite slots, and, when screw-cutting is not actually in progress, is driven by a gear in mesh with the lead-screw. The disc replaces the screw-cutting dial usually provided on the carriage.

With this arrangement, upward movement of lever B causes pin F to make contact with the disc, so that the half-nut is temporarily held out of engagement with the lead-screw. The engagement is made when one of the slots in the disc is presented to pin F. In this way, the engagement of the half-nut is synchronized with the angular position of the work-piece in relation to the cutting tool.

At the end of the cutting stroke, an extension L on plate D makes contact with adjustable collar H mounted on the lead-screw. In consequence, plate D is pivoted in a clockwise direction. A second pin J, attached to this plate, causes pawl E to release arm A. At the same time, this arm is swung upward by the action of pin C, assisted by tension spring K, so that the half-nut is automatically disengaged. The cross-slide is then moved by hand so that the cutting tool is brought clear of the work, and the carriage is traversed toward the tailstock in preparation for taking the next threading cut.

An alternative method of insuring correct timing for engagement of the half-nut depends upon a tripping device, located at the left-hand end of the headstock which is coupled to the apron mechanism by a cable.

One-Revolution Clutch with Positive Action

For a special type of cutting-off machine it was necessary to supply a one-revolution clutch that would be positive in action and trouble-free. Figures 8 and 9 show a clutch designed to meet these requirements. The design is based on the action of a cog wheel and a driving pawl.

The driving member, pulley A, is keyed to drive cog wheel B. The cog wheel has six ratchet-shaped teeth and rotates on shaft C. Pawl D is spring-loaded and free to pivot in assembly E

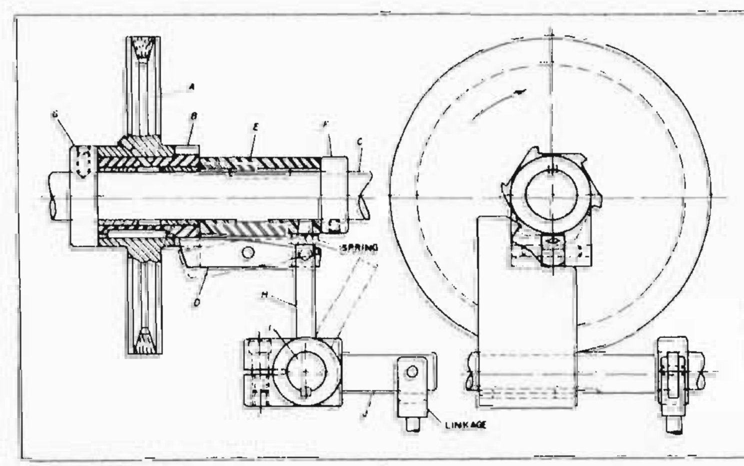


FIG. 8. One-revolution clutch which incorporates a cog wheel and pawl.

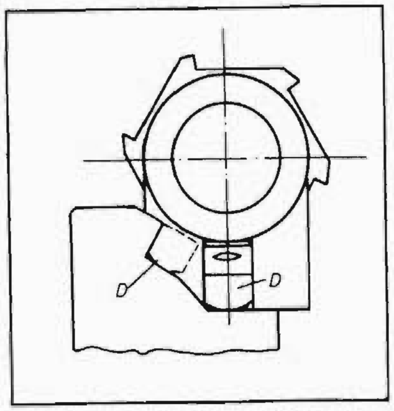


FIG. 9. Diagram showing pawl in entering and stopped positions.

which, in turn, is keyed on shaft *C*. The clutch is held in position on the shaft by means of collars *F* and *G*.

Cam *H* is keyed to the clutch operating shaft *I*. Also keyed to shaft *I* is lever *J*. This lever is used for operating the clutch. With proper linkage it can be manipulated by a foot-pedal.

In operating the clutch, cam *H* is rotated out of position to allow pawl *D* to engage a tooth of cog wheel *B*, as seen in Fig. 9. Cam *H* is then returned to its original position as the pawl and assembly *E* rotate shaft *C*. Continued rotation of assembly *E* will cause the pawl to pivot while in contact with cam *H*, consequently disengaging the drive after one revolution. Cam *H* is swung back to its original position by a coil spring attached to the linkage.

The clutch is intended primarily for a slow operating drive of 10 to 60 rpm.

One-Revolution Clutch Adaptable to Various Dwell Periods

The drive for a wire-twisting mechanism employed in an automatic device for forming eyelets from music wire was required to dwell for five-ninths of the machine cycle and then rotate for the remaining portion. At first a cam-controlled twisting arrangement was tried, but this failed to perform efficiently and was replaced by the one-revolution clutch shown in Fig. 10. The clutch not only provided a smoothly operating means of obtaining interrupted motion but also a method of accurately dividing the constant rotation of the driving member into the proper periods of dwell and movement. One feature of this clutch is that it is versatile and can easily be redesigned for other cycle timing.

Shaft *A*, which is the driven member, is supported in a sleeve bearing pressed in frame *B*. The driving member, sprocket *C*, rotates on shaft *A* and has four ratchet-shaped teeth *D* in the hub of the sprocket. Two pawls *E* pivot on pins *F* secured in disc *G* which, in turn, is keyed to shaft *A*. During periods of drive, each pawl is held in contact with a tooth on sprocket *C* by means of leaf springs *H* and stops *J*. These stops are carried in the bore of a shell *K* which is free to turn on shaft *A*. A collar *L* holds the assembly in place on the shaft.

To disengage shaft *A*, a latch *M*, operated by a tripping cam (not shown), is allowed to contact a tooth *N* on member *K*. (In Fig. 10, tooth *N* has just come into contact with the latch.) The latch stops the rotation of member *K*, but the continued rotation of disc *G* causes the pawls to pivot out of contact with teeth *D* disengaging the drive, and to move clockwise past stops

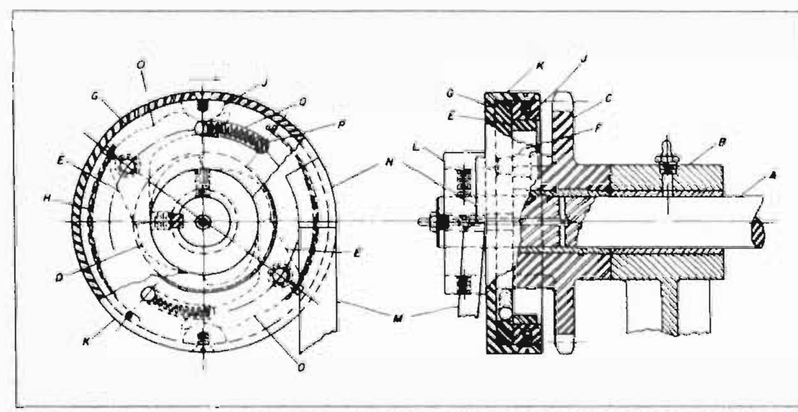


FIG. 10. One-revolution clutch that can be arranged to drive shaft *A* in a dwell-rotate cycle timed in various ratios.

J so that the stops contact the pawls at points *O*. As disc *G* rotates clockwise in relation to shell *K*, due to the friction between the rotating drive and disc *G*, two pins *P* secured in member *K* each compress a spring *Q* carried in an elongated radial slot in member *G*. These springs are heavy enough to rotate shell *K* counterclockwise and return the pawls into contact with the teeth on the drive sprocket when latch *M* releases tooth *N*. They are not heavy enough, however, to rotate disc *G* and shaft *A*.

When the pawls have rotated until they are out of contact with the teeth on the driving sprocket, shaft *A*, disc *G*, shell *K*, and the pawls remain stationary until the latch is tripped to release shell *K*. The release of shell *K* allows shaft *A* to rotate one full revolution before again being stopped by the action of latch *M*. To obtain the proper dwell period, latch *M*, however, is retained in contact with tooth *N* until one and one-quarter turns of the drive sprocket have been completed. The cam then trips latch *M*, and each pawl returns into contact with the adjacent tooth *D* (counterclockwise) on sprocket *C*. Timing of latch *M*, in this case, is accomplished by employing a single-lobe tripping cam geared to the constant speed drive to rotate once in each two and one-quarter turns of sprocket *C*.

Thus the shaft rotates one revolution and then dwells for one and one-quarter revolutions. Since the machine cycle is, therefore, completed in two and one-quarter turns of the drive sprocket, shaft *A* rotates for four-ninths of the cycle and dwells for five-ninths. Other dwell-rotate cycle timings can be obtained by varying the number of teeth *D* on sprocket *C* and/or by changing the timing of the device for tripping latch *M*.

CHAPTER 20

Miscellaneous Mechanisms

The mechanisms described in this chapter are those which were not readily classified in the general groups covered by the preceding chapters. They are included because of some interesting features or ingenious design.

Instantaneous Reversing Shaft-Traversing Device Replaces Troublesome Cam

A "cheeser-twister" in the textile industry is a yarn-winding machine. The machine puts the twist in yarn as it comes from the spinning machine, then winds it (zigzag, like twine) on a cylindrical cardboard core about 5 inches long, making a rough ball about 5 inches in diameter. The balled yarn is known as cheese.

It is important that tension of the yarn in the cheese be uniform and without slack because slackness results in a loose warp thread when the yarn has been woven into cloth. Wear on the drum cam causes the follower and, through it, the thread-guide traverse to dwell at the end of a stroke instead of reversing direction immediately. During the dwell the yarn builds up unevenly on the edges as undesirable slack, that causes loose warp threads.

The described mechanism eliminates the drum cam and its problem of wear. The new traverse therefore provides reliable yarn-tension control, and reduces machine down time and the cost of repairing and replacing worn cams.

In Fig. 1 is shown a cross-section assembly of the new traverse of the cheeser-twister. The cam-shaft has been shortened

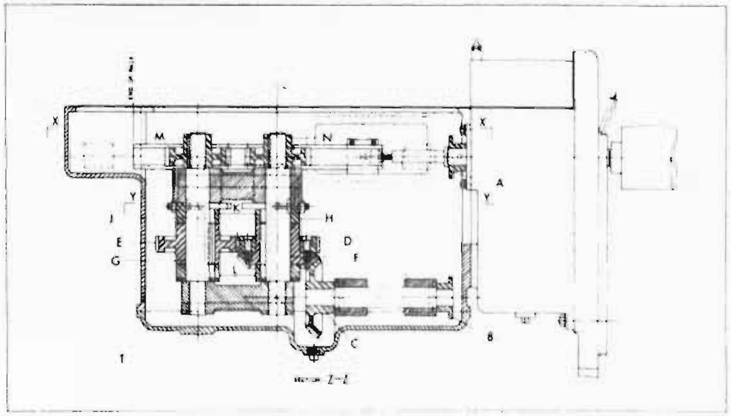


FIG. 1. View of thread-guide traversing mechanism shows design of the pinion-clutching arrangement on shafts H and J.

to provide for a driving sprocket A which transmits the power to the mechanism through the sprocket B by a roller chain.

The principle of the design is a parallel rack movement with alternative driving pinions mounted between the racks. The rotation of each pinion causes one rack to advance and the other to retract, thus creating a balanced motion in the machine.

Bevel gear C drives gears D and E which are free to rotate on shafts F and G. On the ends of the shafts, rack pinions M and N are keyed and always in mesh with the racks O and P (section X-X, Fig. 3).

Each of these shafts is fitted with round clutch keys H and J, each having half the diameter cut away as shown in Fig. 2, section Y-Y, and full-diameter bearings in the collars K and L, shown in section Z-Z.

The length cut away from the clutch keys is the length of the hub on gears D and E. The keys each have one end bent up at right angles to the diameter, providing a means for lever action to rotate them, when contacted by adjustable stop-screws V-1, V-2, V-3 and V-4. Thus, the rotating motion of the key serves as a clutch.

Each rack has a bracket fastened to it for the purpose of moving the traverse bars T and U, section X-X, Fig. 3.

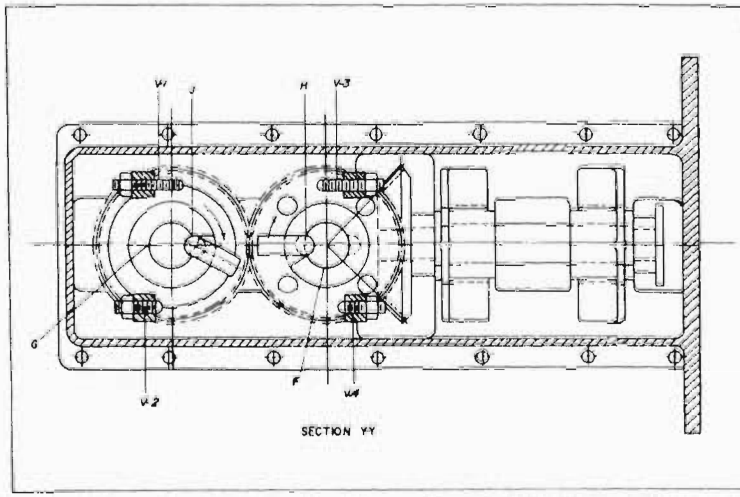


FIG. 2. Alternate clutching of pinions G and F is created by levers on pins J and H striking stops V-1 and V-2, and V-3 and V-4, respectively. (Note section Y-Y, Fig. 1.)

In the operation of this mechanism, as shown in section Y-Y, (Fig. 1) one key J is positive, while the other H is neutral. In this case, shaft G is a positive drive through gear E. At the same time shaft F is driven through the rack pinion N in mesh with

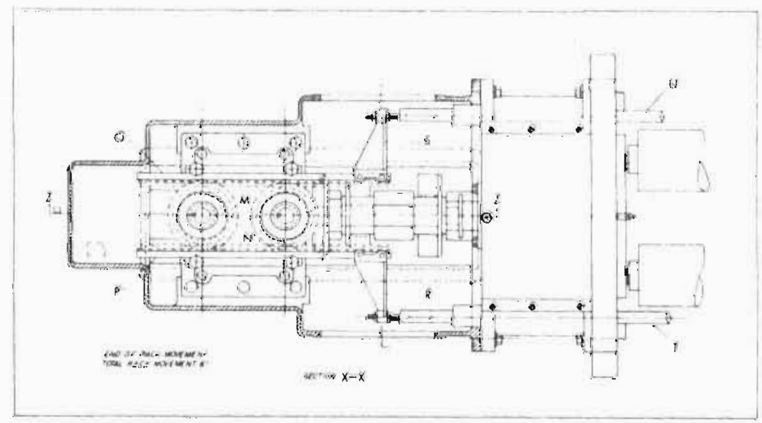


FIG. 3. Section through the shaft-traversing mechanism shows how brackets R and S transmit motion from racks O and P to the yarn guide-shaft traverse bars T and U, respectively.

racks O and P. Reverse rotation of shaft F turns it back to the starting point, when the lever end of the key H is rotated into positive position by stop-screw 3, to do the driving via gear D. At the same time key J has been rotated into neutral position by stop-screw 2, and shaft G backed up to the starting point. Gears D and E are individual drivers, due to the clutch action of the key, in mesh with one another and always rotating in the same direction.

The pinion gears M and N are not in mesh with each other and turn one-half revolution in the same direction, being in mesh with the racks O and P, which produces the reciprocating motion of the traverse bar.

The gears D and E make one-half to one revolution forward and one-half revolution in reversing gears M and N. Hardened chrome-alloy steel is used for the racks, gears, shafts, and keys.

The top of the case is covered with transparent plastic to permit inspection. The bottom has a removable castiron cover which supports the shaft bearings.

Transmission Remotely Controlled by Triple-Action Cylinder

A triple-acting cylinder arranged to remotely actuate a special purpose transmission is illustrated in Fig. 4. Designed for low cost and made of standard components, the device can be pneumatically or hydraulically operated.

Cylinder A contains two piston assemblies B and C (Fig. 5). Assembly B is pivoted on a fixed pin at the rod end, while assembly C is linked to level D but is free to move to the right. Trunnions are mounted on the cylinder shell, and flexible lines are employed to supply and exhaust the pressure chambers.

Input shaft E is integral with a gear F. This gear has a recess in the right-hand face that serves as the front bearing for an output shaft G. Gears H, J, and K idle on shaft G but are restrained from axial movement. Dual face clutch L and a rear clutch M are splined to this shaft but are free to float axially. The gears on shaft N are secured to it and are in positive drive with those on shafts E and G.

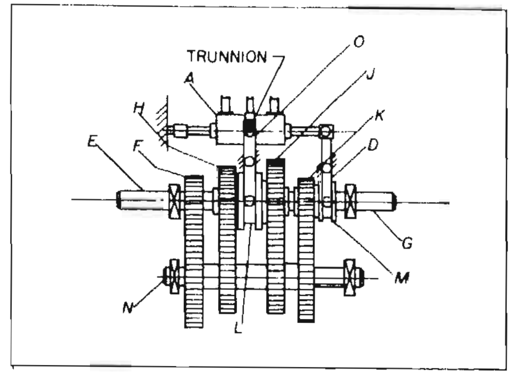


FIG. 4. Triple-acting cylinder A permits remote control of this gear transmission. Either pneumatic or hydraulic operation is possible.

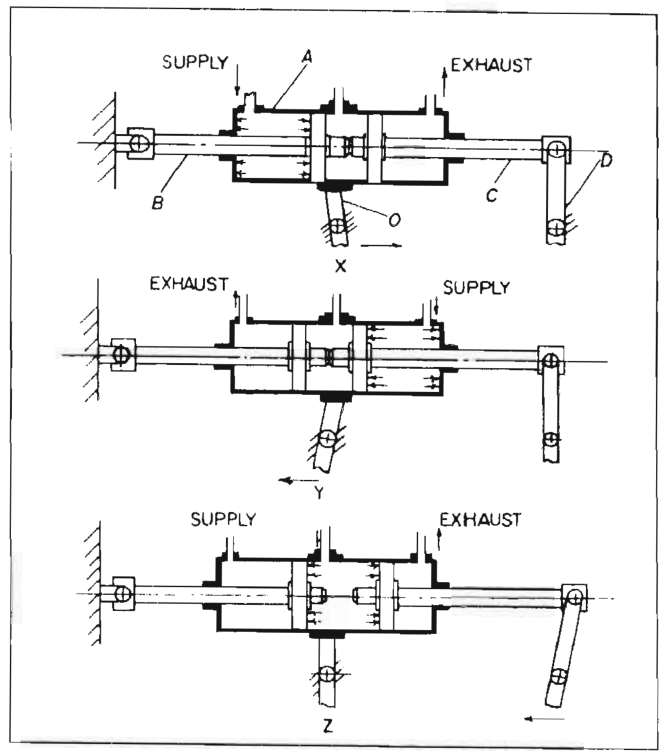


FIG. 5. Lever movements that result when a fluid is introduced under pressure into each section of the cylinder. A different section is vented in each case.

When the left-hand chamber (view X, Fig. 5) is pressurized, the cylinder body moves to the left, and the trunnions tilt a pivoting forked lever *O*. This lever, in turn, slides the dual face clutch *L* to the right, thus engaging gear *J* to output shaft *G* for drive. If the right-hand chamber is pressurized (view Y, Fig. 5) the cylinder body moves in the reverse direction and the left-hand face of clutch *L* engages gear *H* to the output shaft. Pressurizing the center chamber (view Z, Fig. 5) will force the rear piston-rod to the right, pivoting forked lever *D* so that it engages the rear clutch *M* with gear *K*.

Cone type clutches with meshing teeth provide a positive engagement with the gears, thus allowing a low pressure to actuate the system. Coil springs are mounted on shaft *G* between the clutches and gears *H*, *J*, and *K*. These springs overcome any piston-seal friction and disengage the clutches when the pressure actuating the cylinder is released.

Gyroscopic Grinding Setup

Unique application of gyroscopic procession makes possible the grinding of diamond phonograph needles by the device illustrated in Fig. 6. In use, the setup needs little manual control and can be employed for similar operations on work-pieces requiring rounded ends.

The arrangement, which resembles a toy gyroscopic top, consists essentially of a flywheel on a shaft that is rotated at high speed and allowed to precess around a given point. If, while the shaft is revolving, an object having an arbitrary profile is positioned along the precession axis and brought into contact with the precessing end of the shaft, the shaft will follow and press against the surface of the object.

To utilize this principle for grinding diamond phonograph styluses, the object is replaced by a conical grinding wheel *A* driven by a motor *B*, and the work-piece *C* is mounted in a chuck on the shaft *D* driven by a motor *E* through a flexible coupling *F*. This coupling can be a gear type or any other kind which will transmit a constant velocity. Motor *B* and the grinding wheel are mounted so that their axis is at an angle to that of motor *E*.

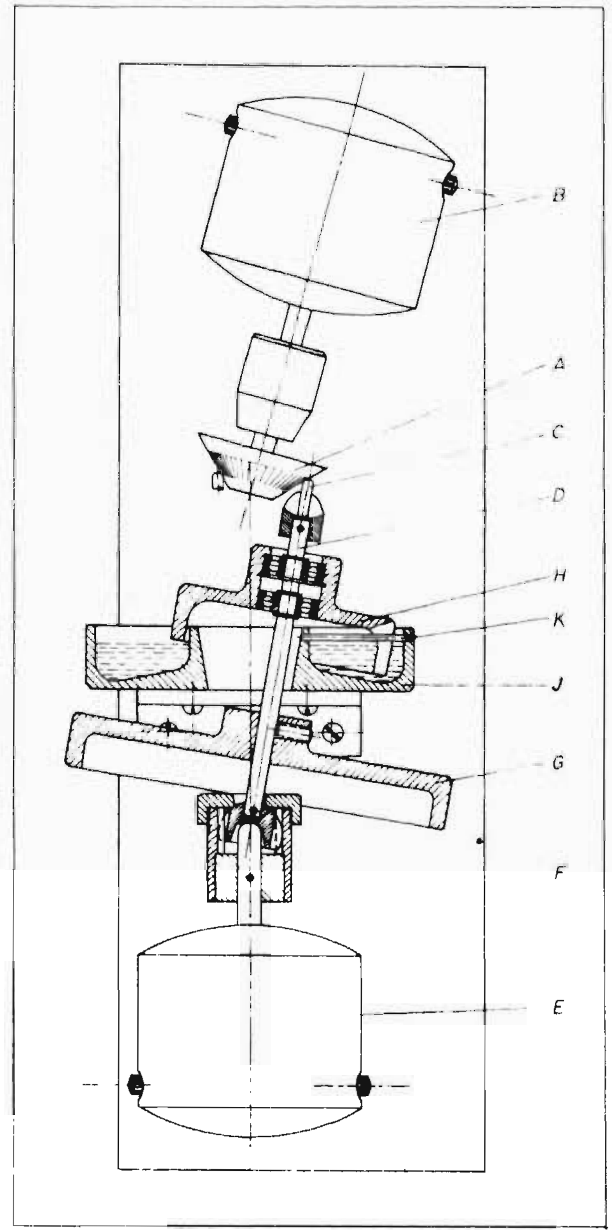


FIG. 6. A device for grinding diamond phonograph styluses. Gyroscopic precession causes the rotating stylus to press against and travel around the rotating conical grinding wheel *A*.

The flywheel G is mounted on shaft D and a braking wheel H is supported on antifriction bearings on the same shaft. The braking wheel is immersed in heavy oil in a fixed container J. A pin K prevents the braking wheel from completely revolving on its axis but allows the stylus and shaft to circle or precess around the rotating grinding wheel. Thus, work-piece C revolves about its own axis, and, because of the gyroscopic precession effect, also moves around cone-shaped grinding wheel A. Since the stylus contacts the wheel at different heights and angles successively, a rounded head is obtained. The speed of grinding is increased by running the two motors in the same direction, thereby adding their relative tangential speeds.

The pressure of the work-piece against the wheel is a function of the speed of precession — the velocity with which the shaft moves around the wheel. Another factor is the height of the center of gravity of the shaft. This latter can be adjusted by raising or lowering the flywheel. The centrifugal force of the flywheel is constantly opposing its gyroscopic tendency to press against the wheel. Therefore, by raising the flywheel, the lever arm of the centrifugal force is lengthened, increasing its effect.

Relative motion between the work-piece and grinding wheel causes a moment tending to accelerate precession, resulting in higher pressure and diminution of the relative grinding speed. To prevent this, the braking wheel H is mounted on the shaft and immersed in the heavy oil contained in member J, which is attached to the frame of the device. Since it is mounted on ball bearings, the braking wheel does not disturb the constant rotation of the shaft around its axis, but the oil's viscosity does prevent too fast a precession of the shaft around the grinding wheel. This also permits the possibility of obtaining elongated and flattened forms by inclining one side of the braking wheel more into the oil, consequently slowing the work-piece as it passes over any desired section of the wheel. The pressure of the work-piece against the wheel is also raised by increasing the speed of motor E and the moment of inertia of the flywheel.

The stylus is contacted by all of the area shaded on the grinding wheel as seen in Fig. 6. This is important for even wear

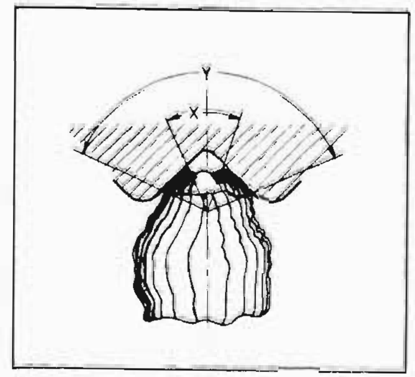


FIG. 7. Each stylus is required to be ground between the angles X (=60 degrees) and Y (=120 degrees) or for a spherical zone equivalent to 90 degrees. Included angle of grinding wheel cone (90 degrees) and axis inclination (15 degrees) is determined by this relationship.

of the wheel. Motor B can also be moved in the direction of its axis, thus changing the portion of the grinding wheel being used without affecting the adjustment of the whole assembly.

In the case under discussion, the stylus has to be ground only in the area between the angles X (= 60 degrees) and Y (= 120 degrees) as illustrated in Fig. 7, because the actual working angle of the diamond point in the record groove is equal to only 90 degrees. This means that one flank of the grinding wheel has to be inclined at an angle of 30 degrees from the vertical, and the other side inclined by $\frac{X}{2}$ or 30 degrees from the horizontal. Consequently, the required cone angle of the grinding wheel is 90 degrees and the inclination of motor B from the vertical is 15 degrees. The actual setup was built for grinding a diamond-stylus radius of 0.0008 inch.

Dial's Moving Witness Mark Compensates for Backlash

A moving witness mark on a dial controlling the angular position of gears, feed-screws, or levers solves the problem of discounting the influence of any backlash. In Fig. 8, the device is shown from the front and side. Dial A is fixed to shaft B which

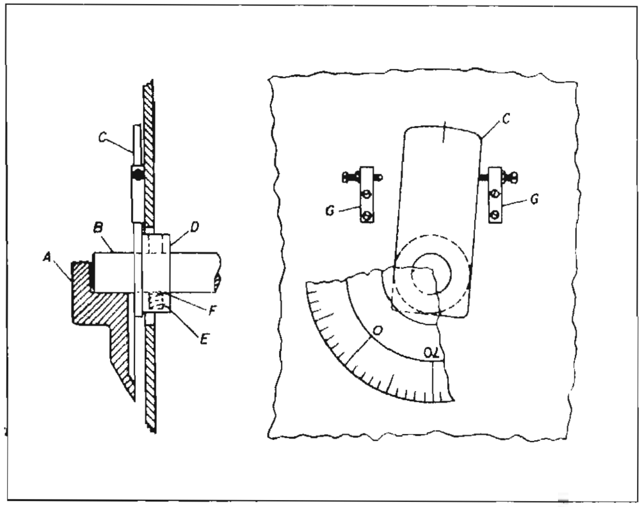


FIG. 8. The free movement of plate C between the dogs G equals the amount of backlash in the system that has to be taken up.

feeds into the machine or mechanism involved. The witness mark is scribed on plate C, which in turn is attached to hub D mounted on, but not fixed to, shaft B. A blind hole cross-drilled in the hub contains spring E and friction pad F. The pad drags against the shaft, transmitting the torque from the shaft to the hub. Adjustable stops G limit plate travel.

In each view, Fig. 9, the dial is set at 30. In view A, the setting was made by rotating the dial counterclockwise, while in view B, the dial has been rotated clockwise. Although the set-

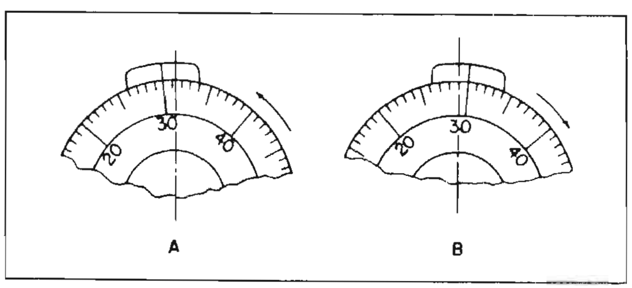


FIG. 9. Backlash in the system accounts for the slight difference in the angular disposition of the two dials and witness marks.

tings are the same, the dial in each instance is in a slightly different position, as is the witness mark. This difference is due to the backlash in the system between the dial (the input) and the gears, feed-screws, or levers (the output).

If the dial in view A is further rotated counterclockwise, the output motion will follow the input motion directly and immediately because the backlash has been taken up in this direction. But if the same dial is rotated clockwise, the input motion is not transmitted until the backlash has been taken up. As the dial moves clockwise, the witness mark moves with it until plate C abuts the right-hand stop G. Backlash has now been taken up, the output starts to move, and the dial reading changes accordingly.

Pneumatically Actuated Mechanical Oscillator

With the device shown in Fig. 10, a reciprocating mechanical motion can be generated from a constant gas pressure source. The unit is capable of functioning at frequencies as high as 100 cps and its oscillating rate for most operating conditions is independent of fluctuations in the applied pressure. Its inherent reliability is enhanced by the fact that it contains only one moving element.

Essentially, the mechanism consists of an oscillating piston and a cylinder. One end of the oscillating element is turned down to form an output shaft and the piston end is drilled with a number of air passages. The stationary member has a cylinder bore and three annular grooves with threaded external connections.

The two outermost cylinder connections are inputs fed by an air pressure source. The center connection is vented to the atmosphere. With the piston at the left, air from the left input enters the left end of the cylinder through the piston passages. Meanwhile, air previously under pressure in the right end of the cylinder is vented through other piston passages to the atmosphere. The right-hand input is closed by the piston. As a result the net force on the piston causes it to move to the right.

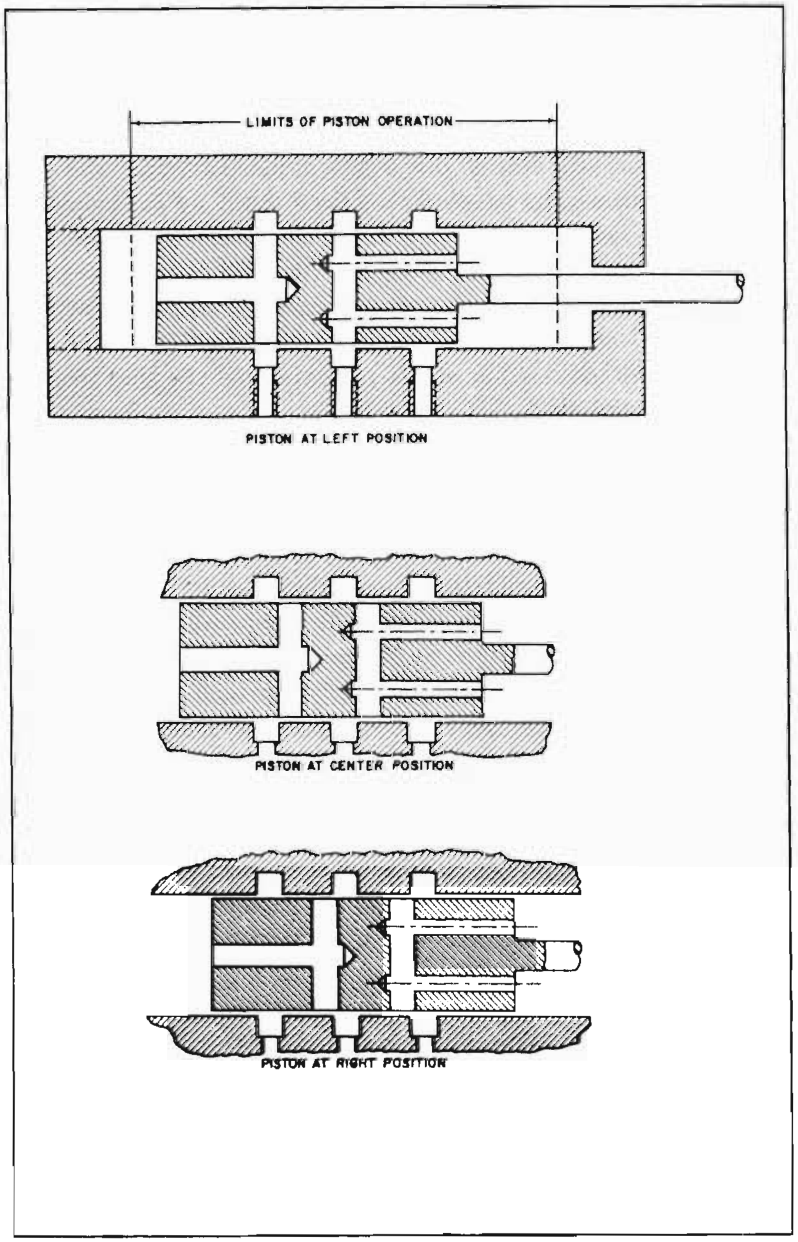


FIG. 10. Piston caused to vibrate by air pressure.

When the piston reaches the center position, it closes the left input and the vent connection to the right cylinder chamber. Further motion to the right allows the air in the left chamber to expand as the residual air in the right chamber is compressed by the piston.

With the piston at the right, the air in the left chamber starts escaping to the atmosphere and the right chamber is opened to the right-hand air supply. As the pressure in the right chamber rises, the piston first decelerates and then reverses its direction. On return of the piston to the left, the cycle repeats.

Reciprocating piston motion can be started from any initial position between the limits shown. The amount of pressure required for operating the mechanism depends upon the weight of the piston and the load attached to the output shaft. Initial pressure also has to overcome the initial friction of the piston in the bore.

Depending on the application, the mechanism can be built with one or two output shafts, one at either end. If it is desired to simply sense the motion of the piston electromagnetically, it can be built without an output shaft. Another application is to use the pulsating output air flow from the vent for pneumatic timing.

Plotting Circular Arcs with Inaccessible Centers

Plotting of circular arcs with inaccessible centers is possible, using a theorem of geometry that states that the locus of the vertex of a constant angle whose legs pass through two fixed points is a circular arc.

If points A and B in Fig. 11 are fixed and angle ACB is held constant, any position of point C, such as at C_1, C_2, or C_3, lies on a circular arc. Also, if angle ABF equals angle ACB, line EBF is tangent to the arc at point B.

This geometry principle provides a means of laying out a circular arc when access to the center point is inconvenient or impossible. Where the arc is to be drawn through three points, or to be drawn through one point and tangent to the line at an-

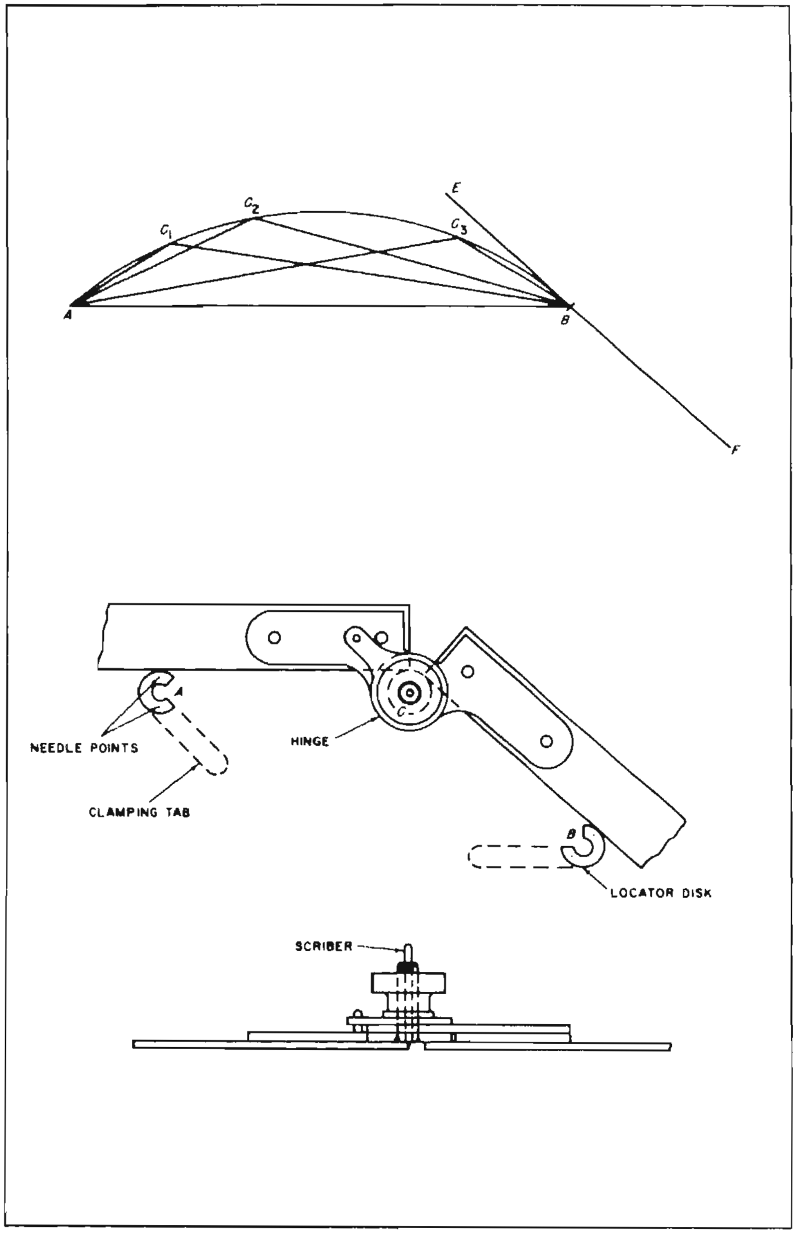


FIG. 11. Instrument to draw arcs with inaccessible centers.

other point, the angle involved can be reproduced on tracing paper and the resulting layout used as a template to mark off successive points of the required arc.

Where the problem is met frequently, construction of an instrument of the type illustrated may be justified. It affords additional accuracy by permitting a continuous drawing of the arc instead of the approximation achieved from plotted points.

The device consists of a pair of straight-edge blades connected by a clamp type hinge with its center offset from the blade edges. A hole is provided through the hinge center for the purpose of mounting a pencil lead or a steel scriber. Two discs with radii equal to the hinged offset are used as locators for the fixed points.

Center holes and cutaway segments in the discs allow the arc to be drawn through the fixed points. The locator discs can carry needle points for use on a drafting board, or clamping tabs (shown in dash lines) for part layout.

High-Speed Cutoff Mechanism

Length of the cut-off piece can be varied when using the simple but improved high-speed cutoff mechanism illustrated in Fig. 12. The device consists of a cutter blade attached to a circular disc on a shaft supported by two bearing brackets. The cutter blade is oriented so that when it is in its lowest position, as shown, it is perpendicular to the center line of the material being cut. The strip is supported by a work guide.

Angle α between the center line of the material and that of the cutter shaft can be determined by the equation:

$$\frac{2\pi RN}{60}\sin\alpha = V, \text{ or } \sin\alpha = \frac{30V}{\pi RN}$$

where R is the active cutting radius of the blade in inches, N is the rotary speed of the disc in rpm, and V the velocity of the material in inches per second.

In order to cut various lengths of strip material at different strip velocities V, the following principle is applied: The knife is rotated with a speed N equal to the number of pieces to be

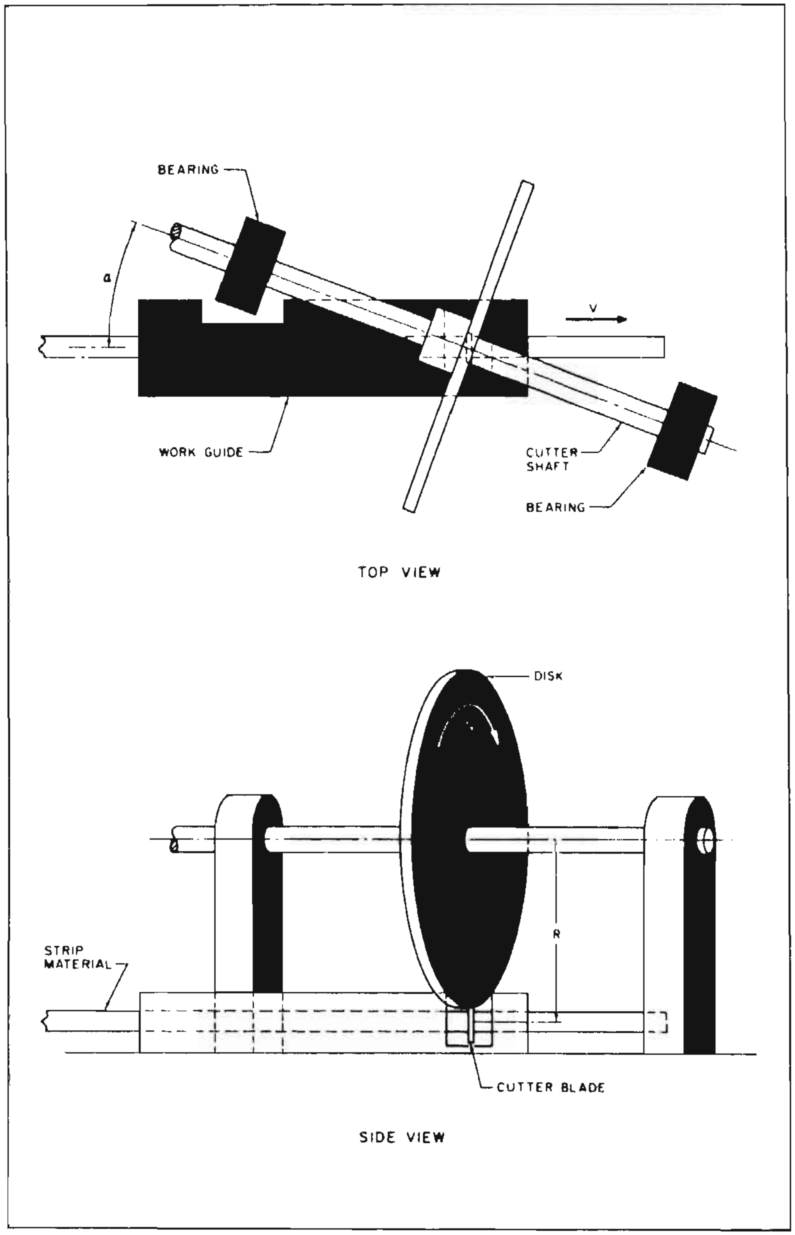


FIG. 12. Stock cut on the fly by shear on a disc.

cut off per minute. Angle α is then adjusted according to the equation and the blade angle is reset to cut perpendicular to the moving strip.

This device has been successful in cutting preslit tubes for can manufacture at a production rate of 1000-2000 cans per min. Actual deviation from an exact perpendicular cut is about 0.002 inch for an active cutting radius R of 2.5 inches.

Mechanism Compensates for Lead-Screw Pitch Errors

Pitch errors in the lead-screws of a thread-cutting or a thread-grinding machine can be compensated for by the mechanism shown in Fig. 13. The drawing shows, in view X, a sectional view of the lead-screw drive in which motion is transmitted through gearing and a differential unit. If the outer casing of the differential is held stationary, the differential acts as a coupling, and drive from the spur gear A is transmitted to the lead-screw in a ratio of 1 to 1, but in the opposite direction of rotation.

A non-self-locking worm B meshes with teeth cut on the periphery of the differential casing. On the outer end of this worm is keyed a plain spur gear C, as can be seen in view Y. Meshing with gear C are two rack-toothed plungers, each of which has a roller at the lower end making contact with one of a pair of templates D and E.

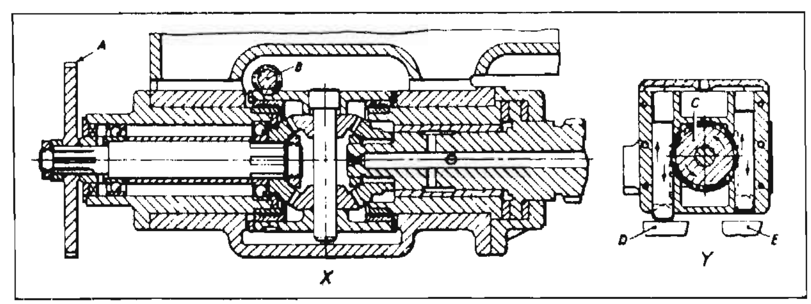


FIG. 13. The surfaces of templates D and E are convex or concave at different points to correspond to pitch errors in the lead-screw.

As gear A starts to rotate, the outer casing of the differential will turn in the same direction, rotating worm B and spur gear C until one of the plungers makes contact with a template. When this occurs, worm B can no longer rotate, and the outer casing of the differential is held stationary. Drive to the lead-screw is now transmitted in the manner described in the first paragraph.

As the lead-screw moves axially, the plunger travels along the upper edge of the template, and as long as the template is flat and parallel with the machine guide ways, the differential casing will be held stationary. However, the edge of the template is arranged with concave or convex curves, which correspond to the known pitch errors in the lead-screw, and when the plunger encounters one of these curves it is raised or lowered accordingly. These movements rotate gear C, which, in turn, rotates worm B, and thus the outer casing of the differential is also caused to rotate.

This rotation has the effect of adding to, or subtracting from, the rotary movement of the lead-screw, and correspondingly reducing or increasing the axial traverse of the carriage, thus compensating for pitch errors. In view Y, one of the plungers has contacted a concave area on template D, which has resulted in a counterclockwise rotation of gear C.

Because of the backlash between the lead-screw and its nut, only one flank of the thread is critical for each direction of carriage traverse. Two templates are necessary, therefore, to compensate for the pitch errors present on each flank of the lead-screw.

Dual Gear Train Diminishes Backlash

Backlash in gears that function to amplify mechanical movements, as in servomechanisms, analog computers, and dial indicators, distorts the movements. The mechanism here described involves a dual gear train that greatly minimizes the amount of backlash in a servomechanism.

In this particular instance, motion has to be transmitted with a ratio of 160:1. Backlash permissible for the slower pinion must

not exceed 0.0025 degree. This accuracy is particularly important when the mechanism is in motion. The servo has worked for over one thousand hours and has backlash too small to be measured by normally available means.

A diagrammatic representation of the dual gear train principle appears in Fig. 14. The line of transmission from driving gear A to driven gear B extends both through intermediate gears C and D and through intermediate gears E and F. Through gears C and D, the ratio is 160:1, as required; but through gears E and F, the ratio is stepped up to 190:1.

For all practical purposes, gears C and D may be considered as a single compound gear. Gears E and F, on the other hand, are entirely separate. Gear E, free on shaft G, carries a pawl H. This pawl, in conjunction with a ratchet J pinned to the shaft, represents a one-direction clutch. On the other end of the shaft

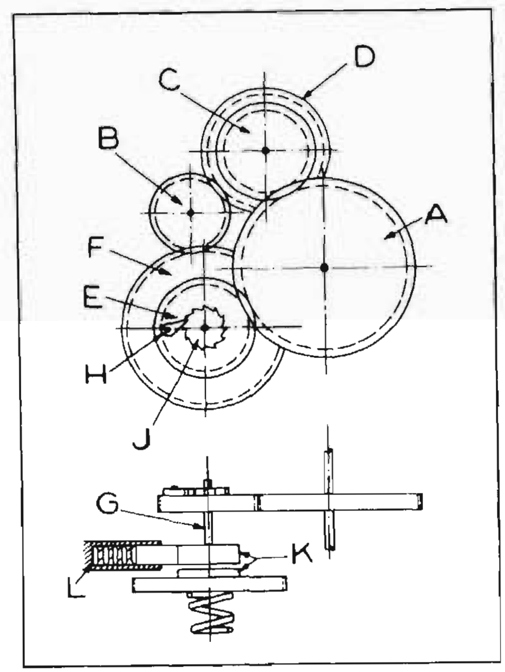


FIG. 14. Intentional slippage in the friction coupling K compensates for the differences in the ratios of the two gear trains.

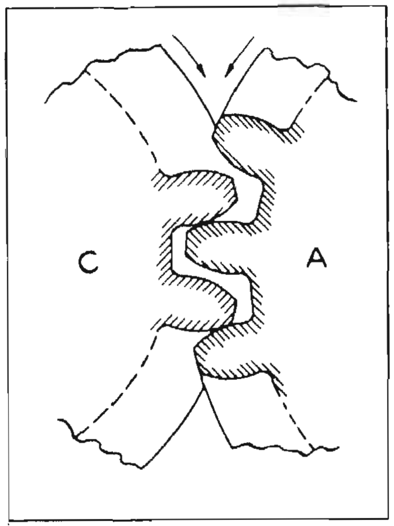


FIG. 15. When gear A rotates counterclockwise, the tooth-flank contact with gear C is as shown.

is a friction coupling K through which the drive is transmitted to gear F.

When the driving gear A rotates counterclockwise, gears E and F tend to rotate the driven gear B at a 190:1 ratio. However, since the actual speed of gear B is restricted by gears C and D to a 160:1 ratio with gear A, two things happen. First, a certain amount of slippage occurs in the friction coupling. Secondly, gear C tends to "drive" gear A, so that the tooth-flank contact between them is as illustrated in Fig. 15. Similarly, gear B tends to "drive" gear D.

Then, when gear A rotates clockwise, the drive is through the compound gears, and theoretically, no backlash has to be absorbed. Gears E and F will rotate by reason of their engagement with gears A and B, respectively, but the one-direction clutch prevents motion from being transmitted.

Since, during the original counterclockwise rotation of gear A, gear F drove gear B, no backlash exists between gears F and B when the rotation is reversed and gear B becomes the driver of gear F which now idles.

Also, the mild drag of a brake L, Fig. 14, on shaft G prevents gear C from running ahead of gear A when the latter is rotating clockwise. Since the brake exerts pressure on the shaft at all times, the friction moment of the brake must be stronger than the remaining moment transmitted by the one-direction clutch when rotating in its free direction. In order to attain the greatest accuracy in the mechanism, the friction moment of the brake should correspond to the following equation:

$$M_{\mathrm{br}} + M_{\mathrm{set}} - M_{\mathrm{odc}} = M_{\mathrm{coup}} - M_{\mathrm{set}}$$

where,

M_{br} = friction moment of brake
M_{set} = moment needed to rotate output shaft and overcome friction in mechanism
M_{odc} = remaining moment of one-direction clutch in free direction
M_{coup} = moment transmitted by coupling

(All of these moments have to be measured on the same axis.)

By thus maintaining a pressure between the teeth of the gears that is equal in both directions, the moment of the mechanism

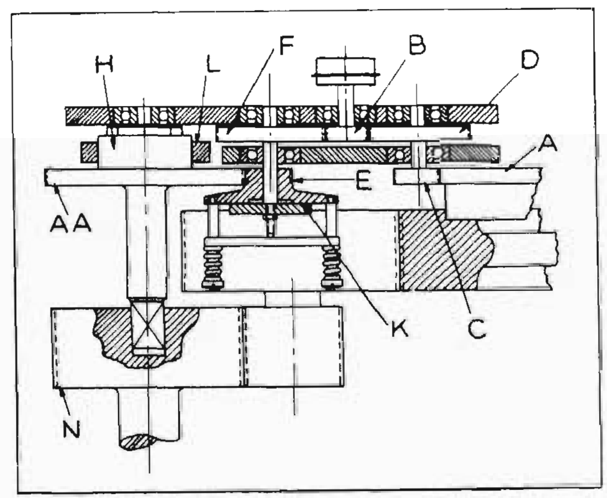


FIG. 16. Cross-sectional view of the servomechanism built around the principle in Fig. 14.

is assured of being constant. This, incidentally, is a feature not found in spring type backlash eliminators.

The actual mechanism that was built is shown in cross section in Fig. 16. Identifying letters on the drawing correspond to those for the counterparts in the diagrammatic representation, Fig. 14. In arranging the transmission, it was found practical to include two driving gears, A and AA. Both are powered by the same motor pinion N. From gear A to gear B, the train runs through gears C and D. And from gear AA to gear B, the train runs through gears E and F.

The one-direction clutch H, the brake L, and the friction coupling K function as previously described. Details of the

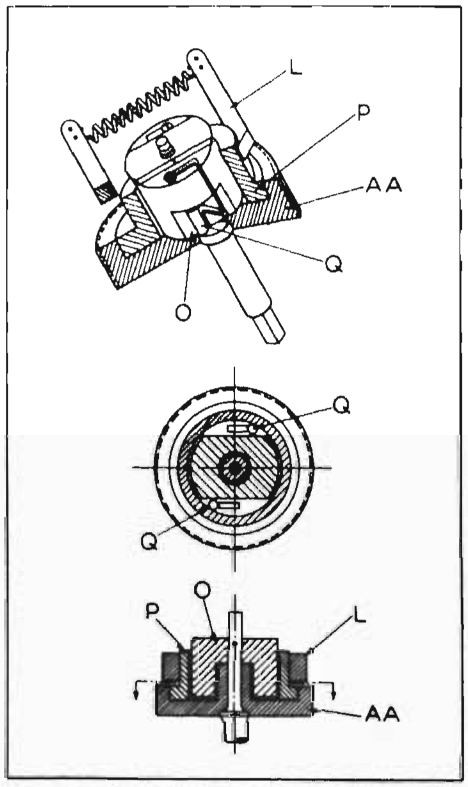


FIG. 17. Rollers Q serve to transmit motion between drum O and sleeve P in one direction only.

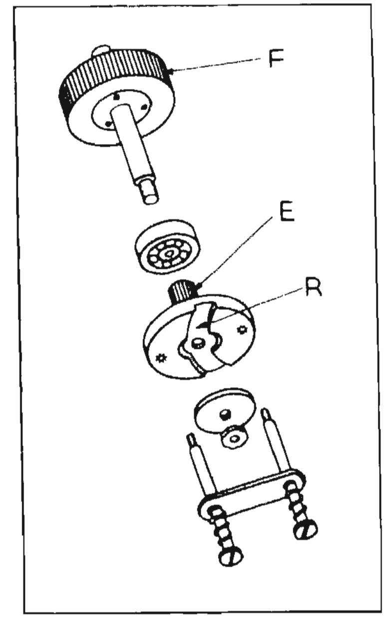


FIG. 18. Exploded view shows the friction coupling between gears E and F.

clutch and brake appear in Fig. 17. The clutch consists of a drum O and sleeve P. A pair of rollers Q transmit motion in but one direction.

An exploded view of the friction coupling is shown in Fig. 18. Although the relative motion between the two discs of the coupling is small, together they revolve rapidly. A groove R in the larger disc produces a flow of air that serves to cool the coupling.

Hydraulically Operated Cross-Slide for a Copying Lathe

A hydraulically operated cross-slide is shown in Fig. 19. Two templates of the part to be turned, as indicated by the broken lines at A and B, are secured to a fixed member of the lathe. This arrangement enables shoulders up to an angle of 90 degrees to be reproduced on both the headstock and tailstock ends of the work.

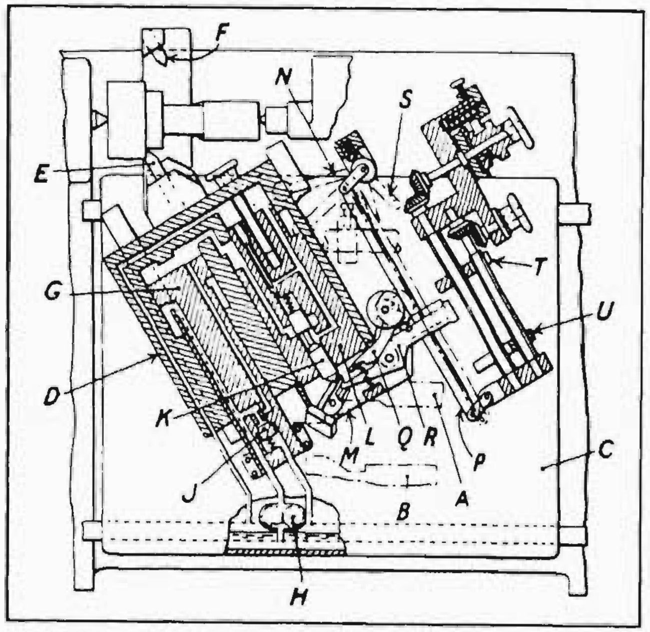


FIG. 19. Sectional view of a hydraulically operated cross-slide for a copying lathe, employing two templates, A and B, of the part to be turned.

Saddle C, mounted on ways beneath the lathe centers, has angular guide ways on which the cross-slide D can be moved hydraulically toward and away from the work-piece. A tool-holder of the upper end of cross-slide D carries the cutting tools E and F. Tool E is used in conjunction with template A for turning stepped portions on the tailstock end of the piece. Tool F is employed, together with template B, when profiles are to be reproduced on the headstock end of the part. A bore is provided in cross-slide D to form a hydraulic cylinder which surrounds the fixed differential piston G carried on saddle C.

In operation, saddle C is traversed on the ways, and oil is delivered to both sides of piston G by the pump H. The pressure in the lower end of the cylinder is maintained constant by a spring-loaded relief valve J. When cross-slide D is to be held stationary or moved away from the work axis, oil is permitted to escape from the upper end of the cylinder and return to the

reservoir by way of the spring-loaded piston type valve K (fitted to body L), and through passages formed in the body and in cross-slide D.

Alternatively, when the cross-slide is to be traversed toward the work axis, valve K is moved downward by spring action so that the oil flow is interrupted and the pressure in the upper end of the cylinder is increased. Movement of valve K and, consequently, of cross-slide D toward and away from the work, is controlled by a stylus which engages template A or B. The stylus is pivoted on a pin fitted to the bracket M, the latter being attached to the lower end of valve body L.

Cutting tools E and F are brought into use, as required, by movement of a lever N, which together with its associated control mechanism, is carried on brackets on the cross-slide. When this lever is turned to its lowest position, bar P (connected to it and to a bracket at its lower end by swivel links) is caused to move in a curved path to the right. The arm Q, carried on a cross-pin fitted to the bracket M, is then swiveled by spring action so that a pin attached to it makes contact with the lever R. Lever R is carried on the same pin as arm Q and its left-hand end engages a slot in the stylus.

Due to the swivel action of arm Q, and consequently of the lever R, valve K is moved upward causing the cross-slide D to be moved away from the work. Thus, cutting tool F is brought into use, and the stylus makes contact with the template B.

Conversely, when lever N is set in its upper position, bar P moves to the left to engage the roller mounted on arm Q, thereby causing the pin fitted to the latter to be brought clear of lever R. Thereupon, valve K is moved downward by spring action with the result that cross-slide D is moved toward the work. In this way, cutting tool E is brought into use and the stylus engages template A.

If desired, cutting tool F can be automatically brought into use when the saddle reaches the extreme left-hand position by means of the lever S. This lever is mounted on the same spindle as lever N and is arranged to make contact with a horizontal adjustable stop (indicated by broken lines), whereupon bar P

and the stylus mechanism are actuated. The sliding motion of saddle C is then reversed.

The movement of cross-slide D in each direction can be limited independently of templates A and B by the stops T and U. These stops are engaged by the right-hand end of lever R and can be adjusted toward and away from each other by handwheels, through bevel gears and screw systems. The handwheel controlling stop T is provided with a circular scale to enable parts to be turned by tool E without the need for a template.

Valve body L can be adjusted axially in a second bore provided in cross-slide D so that the distance between one of the cutting tools and the stylus can be set to produce stepped parts of the required diameters.

Tension Regulating Device for Coiling Machines

One of the final phases in the fabrication of wire is the winding of the product around a reel as it leaves the annealing furnace. Winding is done by rotating the reel and guiding the wire back and forth along the axis of the reel until layers of wire have been built up.

If the reel rotates at a fixed speed, the tension on the wire increases for each layer, since the wire is forced to feed faster as the diameter of the reel body is built up. Eventually, the wire may snap, or some other failure take place. The accompanying diagram shows the principle of a mechanism designed to maintain a state of equilibrium in which the tension on the wire is constant.

In advancing from the annealing furnace to the reel A (see Fig. 20), the wire is directed through a control unit B. The reel is driven by a motor C through a drive-shaft D. A weight E produces sufficient drag in the control unit to keep the wire taut as it is wound on the reel. The tendency for the wire to speed up, and for the tension on the wire to increase, is countered by a movement of the control unit. This unit rotates through a small angle after each layer has been wound, to a total of a degrees.

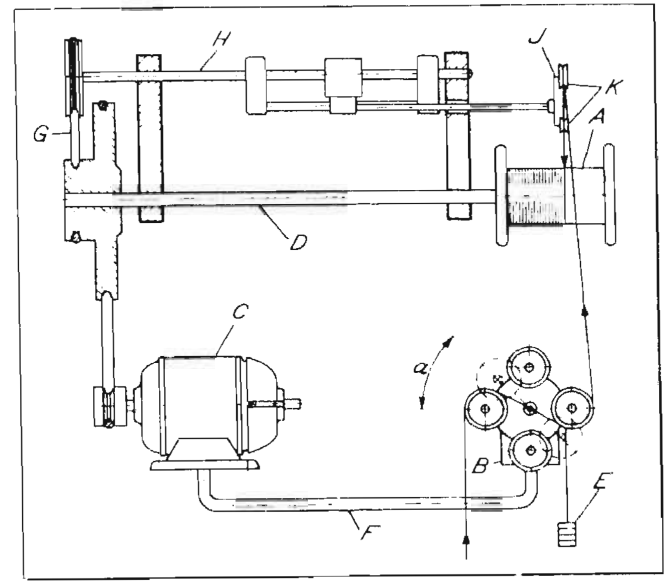


FIG. 20. The control unit B maintains constant tension on the wire by decelerating the motor C.

In pivoting, the control unit actuates a rheostat which, through a cable F, decelerates the motor. Thus, by driving the reel at a progressively slower rate, the feed and tension of the wire are held constant. For any particular wire size, the tension can be set by varying the weight E in the control unit.

A V-belt G causes the wire-guide shaft H to turn with the drive-shaft so that the lineal movement of the wire-guide roller J is synchronized. This roller has a friction contact with two sheaves K, the axes of which are able to float out from a horizontal position to permit the wire to be properly wound around the reel.

Hydraulic Copying System Controls Two Lathe Tools

A patented hydraulic copying system for a railroad car wheel lathe is shown in Figs. 21 and 22. This device permits simultaneous turning on the tread and flange portions of the car wheel

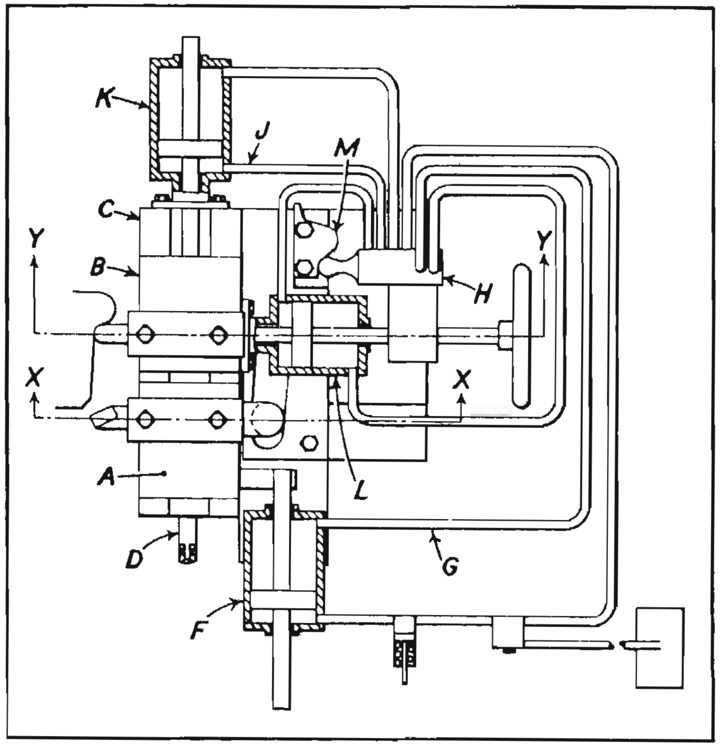


FIG. 21. Plan view of a hydraulic copying system for a railroad car wheel lathe that permits turning tread and flange portions simultaneously.

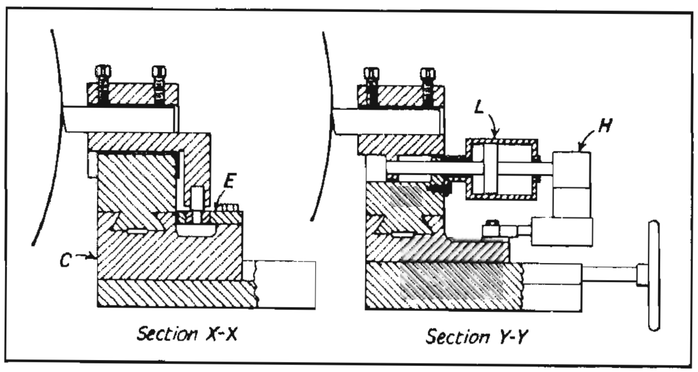


FIG. 22. Sectional views of copying system illustrated in Fig. 21 show template E for tread form, and spoolvalve H attached to piston-rod of cylinder L.

by separate, single-point cutting tools. Thus, cycle time can be reduced and production increased.

As seen in Fig. 21, the cutting tools are mounted on compound slides A and B, which can be traversed parallel with the work axis on way C. The entire assembly can be adjusted toward or away from the work on ways provided on the base.

During the machining cycle, rotation of shaft D causes slide A only to be traversed parallel with the work axis for turning the tread portion of the wheel. The required tapered form of the tread is produced by the action between a follower roll attached to the slide and the template E (section X-X in Fig. 22) which is secured to way C.

This traversing motion causes a piston-rod attached to slide A to be moved axially in cylinder F. As a result, hydraulic fluid is discharged from the inner end of this cylinder, and through pipe G to a spool-valve H. When profile-turning is not being performed on the wheel flange, the fluid is directed from the valve through pipe J to the inner end of cylinder K. The piston-rod of this cylinder is attached to slide B. Thus, slides A and B are traversed in the same direction at similar speeds.

As shown in section Y-Y of Fig. 22, valve H is attached to one end of a piston-rod. The opposite end of this rod is coupled to the slide carrying the cutting tool for profile-turning the wheel flange. Cylinder L is fixed to slide B. When profile-turning the flange, a spring-loaded follower (mounted on one end of the valve-spool) is held in contact with template M, which is fixed to way C.

When the valve-spool is moved outward, due to the action of the template on the follower, the hydraulic fluid from cylinder F is directed simultaneously to the inner ends of cylinders K and L. As a result, the traverse rate of slide B is reduced, and the profiling tool is moved away from the work axis to turn half the wheel flange. When the largest diameter of the flange has been turned, the valve-spool is moved in the opposite direction by spring action. This keeps the follower roll in contact with template M as slide B continues to move in the same direction, as can be seen from Fig. 21.

The hydraulic fluid is then directed to the outer end of cylinder L, as well as to the inner end of cylinder K. In this way, the profiling tool is moved toward the work to turn the other half of the wheel flange, reproducing the shape of template M on the work-piece.

Pitch-Error Compensating Device for Height Gage

An unusual height gage is equipped with an adjustable scale for direct reading and a measuring head that is positioned on the guide column by means of a lead-screw. A vernier, with a dial that is coupled to the lead-screw, permits setting of the instrument to within 0.0001 inch, normally without the use of an optical aid. This accuracy, however, is only made possible by an arrangement that automatically compensates for small pitch errors along the full length of the lead-screw.

The height gage has a tubular column A (Fig. 23) mounted on a hollow base B. A 10-pitch lead-screw C, 8½ inches in length, is supported upright inside of the column by a special anti-friction bearing D (Fig. 24) attached to the top of the base. The screw thread is of modified Acme form with a root diameter of only 0.190 inch. A plain bushing, lightly pressed in a suitable cap that is screwed on to the top of the column, retains the lead-screw and keeps it in axial alignment.

A measuring head E has a slide fit over the column and is fastened to a tongue integral with a nut F that has a close fit on the lead-screw. For nearly its entire length, the column is provided with a slot 0.200 inch wide to permit passage of the tongue.

Drive for the lead-screw is simply a chain of 50 diametral pitch gears (not shown). There are four gears in all: one is attached to the bottom of the lead-screw; one is an intermediate; and the other two are drivers chosen to give a 2-to-1 speed ratio for rapid traverse, and a 3-to-1 speed reduction for fine adjustment. The setting of the measuring head is accomplished by two knurled knobs protruding from the base of the height gage.

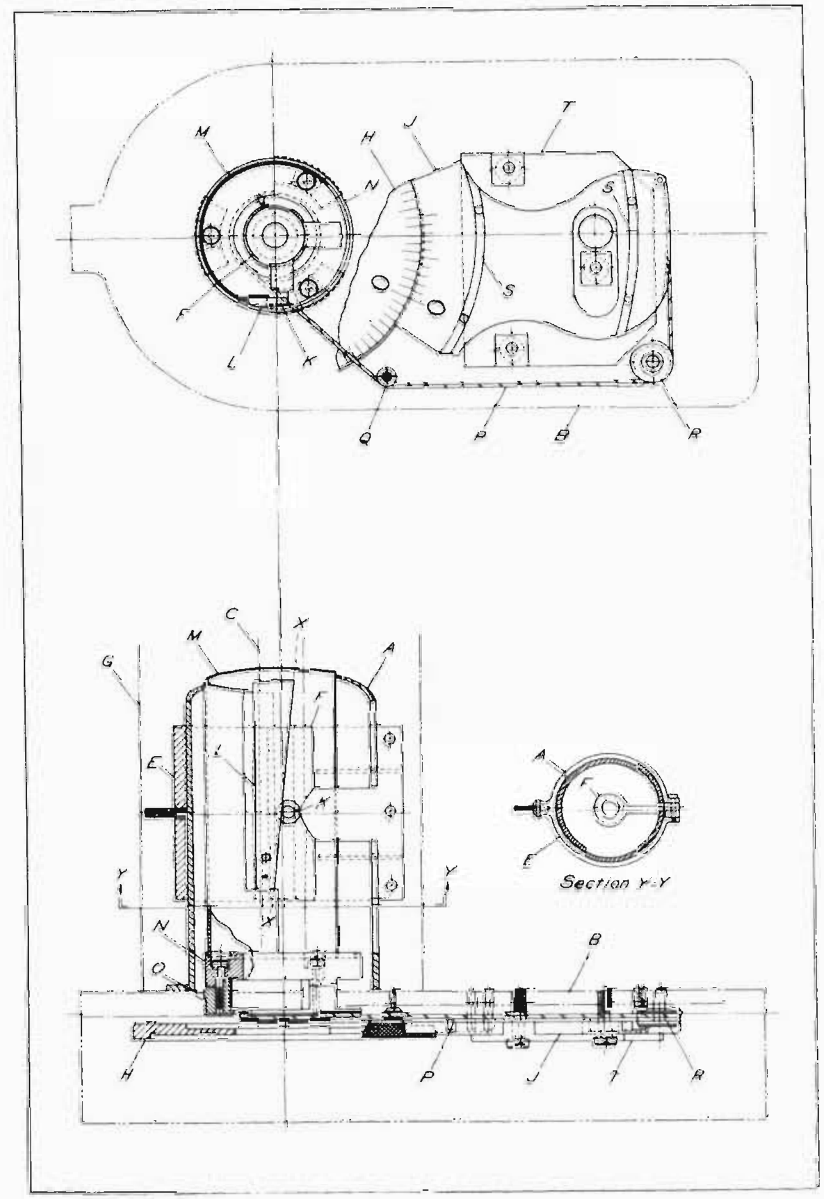


FIG. 23. This mechanism automatically resets vernier scale to compensate for pitch error in the lead-screw of special height gage. For clarity, components not used for this function have been eliminated.

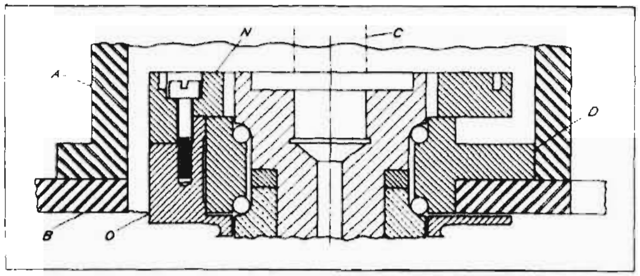


FIG. 24. Enlarged view shows details of main bearing *D* and parts *N* and *O*. Outer race of the bearing has a flange that is cut away at three points to permit passage and movement of part *O*.

A sheet-metal cover *G* encloses the column except for the scale and the slot that provides clearance for the measuring head.

A dial *H*, coupled to the lead-screw, is visible through a window in the base of the gage and is used in conjunction with a vernier scale *J* to set the measuring head. The main scale is vertical. Also, exerting light pressure on a button on top of the base frees the dial for readjusting to zero and simultaneously locks the lead-screw in place. This arrangement permits the scale and the dial to be quickly and easily set to zero, regardless of the position of the head within its full range of 7 inches. The height gage is, therefore, direct reading, and consequently the usual paper work required for setting the gage is greatly reduced. Downward measurements, although involving subtraction, present no problem since the dial is simply read backwards, as is the micrometer dial on the feed-screws of many machine tools.

Construction of the measuring head is shown in section Y-Y, Fig. 23. The center hole in the tongue of the nut is so accommodate a knurled knob for clamping the measuring head to the column. Scriber clamping arrangement (not shown) is attached to the measuring head by means of the stud located opposite the tongue. The accuracy of measurements made with this height gage would normally depend on the accuracy of the lead-screw were it not that the arrangement incorporates an automatic pitch error compensating device. This device advances or retracts the vernier-plate according to the position of the measuring head. Amount and direction of vernier movement are controlled by the contour of a compensating bar. The pitch-error compensating arrangement and associated parts are illustrated in detail in Fig. 23.

Actuating pin *K* is lightly pressed in a boss on nut *F*, and is always in contact with the working edge of compensating bar *L*. The bar is attached to a member *M*, which is made of light tubing. Member *M* stands upright between the inner wall of the column and the screw. Sufficient clearance prevents interference between these parts. The bushing forming the top bearing for the screw also serves as the top pivot for member *M*, the bottom end of which fits in the annular groove in a part *N*. A small screw, not shown, restrains member *M* from turning in the groove. Three screws fasten member *N* to three bosses milled on the upper face of a member *O*. The shape of these bosses may be seen in Figs. 23 and 24. Before assembly, the bosses on member *O* are inserted into openings in the top of the base. These openings surround main bearing *D*, and are shaped as shown to permit members *N*, *O*, and attached parts to have a limited rotational movement.

The body of the main bearing fits snugly in a reamed hole in the top of the base and is flanged. Three screws secure the flange to the top of the base. In addition, the flange is cut away the same as the base to allow clearance for the bosses on part *O*. Member *N* rests on the top face of the bearing body which is somewhat higher than the top of the flange. The bosses on member *O* are a free fit on the periphery of the outer bearing race. This arrangement permits members *N* and *O* to straddle the bearing support flange and to rotate a few degrees on the bearing.

A groove of semicircular cross-section is machined into the periphery of member *O* to accommodate a cord *P*. One end of this cord is attached to member *O*, and the other end is attached to vernier-plate *J*. In addition, cord *P* passes over pulleys *Q* and *R*. Two grooves of rectangular cross-section are cut in the vernier-plate along arcs centered on the axis of the main bearing.

These grooves accommodate two phospher bronze guides S of the same radii. Each guide is attached to the base with two 0.063-inch diameter pins that are a light press fit in locating holes. The guides allow the vernier-plate to move in an arc. Retaining plate T, held in place with three screws, supports the vernier-plate. A suitable tension spring (not shown) is attached to the vernier-plate to exert a constant pull on the related parts and keep the operating edge of compensating bar L in contact with actuating pin K. Dial H rotates with the lead-screw.

As an illustration, let X-X represent the operating edge of compensating bar L. Assuming the measuring head is being raised for a distance of 0.5000 inch, actuating pin K, moving with the nut, slides along the bar. Since the bar is inclined in relation to the axis of the column, the sliding action of pin K imparts a clockwise movement to members M, N, and O, and the pull of the cord advances the vernier accordingly. Advancement of the vernier is necessitated by the fact, previously established, that a little more than five turns of the 10-pitch screw is required to raise the measuring head exactly 0.5000 inch. If the vernier remained stationary, the zero on the dial would be slightly ahead of the zero on the vernier. Disregarding the pitch error and coinciding the zeros of the dial would, of course, result in a false setting. This has been avoided by the automatic adjustment of the vernier-plate setting.

Finding the pitch errors of the lead-screw and bringing the compensating bar to the desired shape are simple undertakings carried out at the time of assembly. Pitch errors are first ascertained by recording the dial readings of the gage when measuring the height of gage-blocks. Then the compensating bar is detached from the easily removed member M and is filed to a shape that will automatically position the vernier scale for accurate readings. Since the back edge of the bar is straight, the contour may be easily measured with a 1-inch micrometer. A contour filed to within plus or minus 0.001 inch of the desired shape will meet practical requirements, for movement of the pin K in relation to the change in the reading on the vernier scale of the height gage is in the ratio of over 50 to 1.

Mechanism for Checking the Torque of Tappet Screws

Tappet screws have to fit a tapped hole properly for an assembly to pass inspection. If the fit is too loose (30 inch-pounds is the minimum torque specified), the screws may become slack during operation; and if too tight (120 inch-pounds is the maximum allowed), adjustment for making service repairs difficult.

A tapping machine — equipped with a socket wrench instead of a tap — is used to drive the screws after starting them by hand.

To check the torque applied, the machine is equipped with a special mechanism so designed that if the torque needed at any point in driving the screw to a specified depth is outside the limits set, the assembly is rejected. Assemblies coming within the limits are automatically dropped into a tote box below the machine.

To check the torque applied to the screw A, seen in Fig. 25, rocker arm B is mounted on a plunger C. The cylinder D holding this plunger is mounted on a turntable E. The turntable is thus free to rock horizontally through a certain angle without significant friction when torque is applied to the screw. The screw is located directly under the socket wrench held in the vertical spindle of the machine.

Before loading the rocker arm over the plunger, the operator starts the screw by hand. As the rocker is mounted on the plunger, it pushes pin F inward to actuate limit switch O and automatically start the machine. The wrench then grips the screw and feeds it downward into the arm, applying whatever torque is required.

Projecting radially from the turntable is an arm G that carries a roller H in contact with a piston J, which is pressed outward by a spring. The spring presses the roller against a fixed stop-screw K with sufficient pressure to resist a torque of 30 inch-pounds. If the torque applied to the screw does not exceed this amount, the roller will not leave the stop and the plunger C

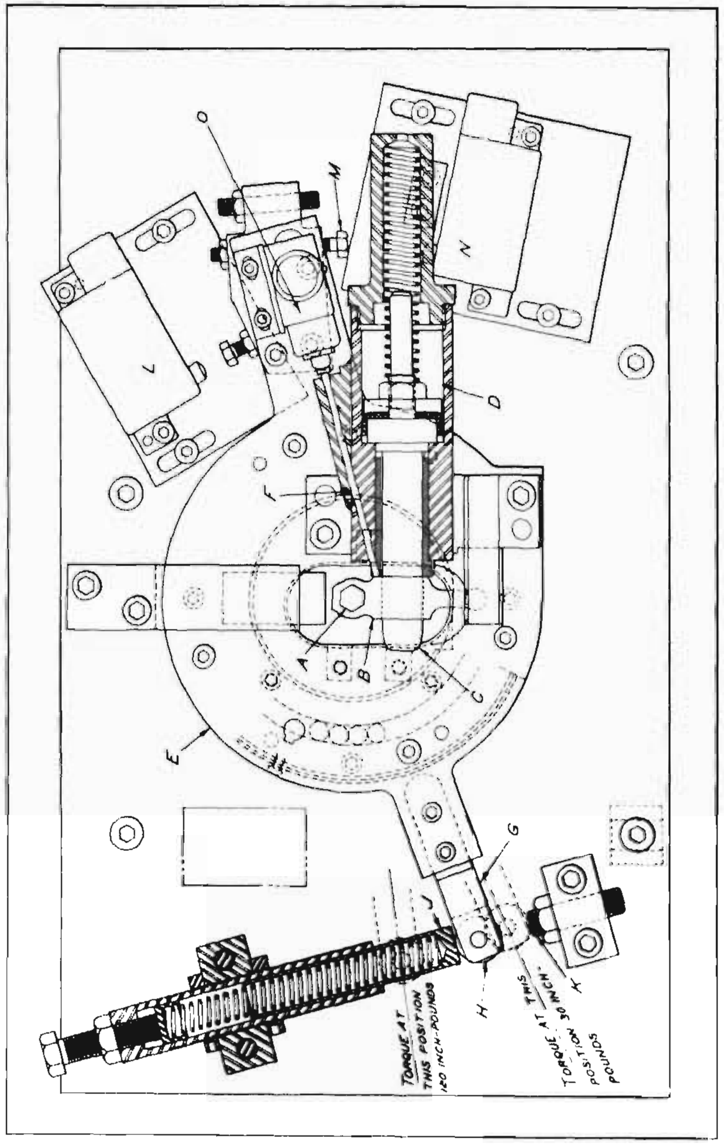


FIG. 25. Assembly drawing of the torque inspection mechanism. The rocker arm *B* is applied over plunger *C* in a cylinder mounted on turntable *E*. Torque is measured, in effect, by the resistance of a spring in back of piston *J*.

will not retract at the end of the cycle because switch *L* remains open. In this event, the operator knows that the screw fits too loosely. Consequently, he must remove the rocker arm from the plunger by hand and place it in a reject box.

If the torque exceeds 30 inch-pounds, roller *H* will be swung clockwise against steadily increasing pressure of the spring behind piston *J*. If the torque reaches 120 inch-pounds, the roller will reach the position indicated in Fig. 25. When this occurs, a stop-screw *M* on the opposite side of the turntable contacts switch *N*, and by tripping this switch prevents plunger *C* from retracting at the end of the cycle. This occurrence warns the operator that the torque is above the limit set. He consequently removes the assembly by hand and puts it in a reject box.

In case the torque applied is not outside the specified limits, the tapping machine spindle feeds down a specified amount in driving the screw and then automatically retracts. As the spindle reaches its upper limit, it trips a switch that operates a solenoid. This, in turn, opens an air valve, admitting air to the cylinder whose piston retracts plunger *C*. This retraction does not occur, however, if roller *H* does not leave the stop-screw *K*, or if the roller moves far enough to trip the switch *N*.

When the applied torque is within the limits set, retraction of the plunger *C* releases the assembly and it drops through a hole in the base of the machine and into a box containing acceptable parts. A spring then returns the plunger, ready to repeat the cycle. With this set-up, it is possible to apply screws to 362 rocker arms per hour on each machine.

All the operator does is to start each screw and then load the rocker arm on the plunger. The remainder of the cycle is automatic and the operator does not even have to remove the workpiece unless it is unacceptable.

As the cylinder carrying plunger *C* is mounted on a turntable that must move freely, the air lines to the cylinders are made from flexible rubber tires. The resistance of these tubes to bending as the turntable rocks has a negligible effect on the torque applied.

INDEX