CHAPTER 7

CAM, TOGGLE, CHAIN, AND BELT MECHANISMS

CAM BASICS

A cam is a mechanical component that is capable of transmitting motion to a follower by direct contact. The driver is called a cam, and the driven member is called the follower. The follower can remain stationary, translate, oscillate, or rotate. The motion is given by $y = f(\theta)$, where

y = cam function (follower) displacement (in.).

f = external force (lb), and

$\theta = w_t$ – cam angle rotation for displacement y, (rad).

Figure 1 illustrates the general form of a plane cam mechanism. It consists of two shaped members A and B with smooth, round, or elongated contact surfaces connected to a third body C. Either body A or body B can be the driver while the other is the follower. These shaped bodies can be replaced by an equivalent mechanism. They are pin-jointed at the instantaneous centers of curvature, 1 and 2, of the contacting surfaces. With any change in relative positions, the points 1 and 2 are shifted and the links of the equivalent mechanism have different lengths.

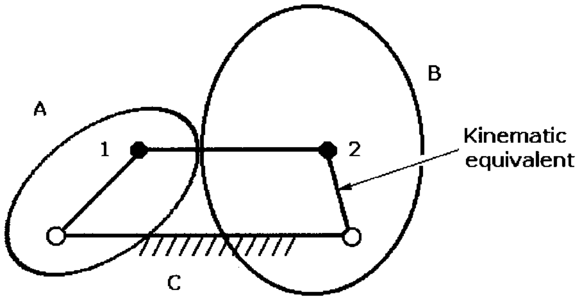


Fig. 1 Basic cam mechanism and its kinematic equivalent (points 1 and 2 are centers of curvature) of the contact point.

Figure 2 shows the two most commonly used cams. Cams can be designed by

- Shaping the cam body to some known curve, such as involutes, spirals, parabolas, or circular arcs.
- Designing the cam mathematically to establish the follower motion and then forming the cam by plotting the tabulated data.
- Establishing the cam contour in parametric form.
- Laying out the cam profile by eye or with the use of appropriately shaped models.

The fourth method is acceptable only if the cam motion is intended for low speeds that will permit the use of a smooth, "bumpless" curve. In situations where higher loads, mass, speed, or elasticity of the members are encountered, a detailed study must be made of both the dynamic aspects of the cam curve and the accuracy of cam fabrication.

The roller follower is most frequently used to distribute and reduce wear between the cam and the follower. The cam and follower must be constrained at all operating speeds. A preloaded compression spring (with an open cam) or a positive drive is used. Positive drive action is accomplished by either a cam groove milled into a cylinder or a conjugate follower or followers in contact with opposite sides of a single or double cam.

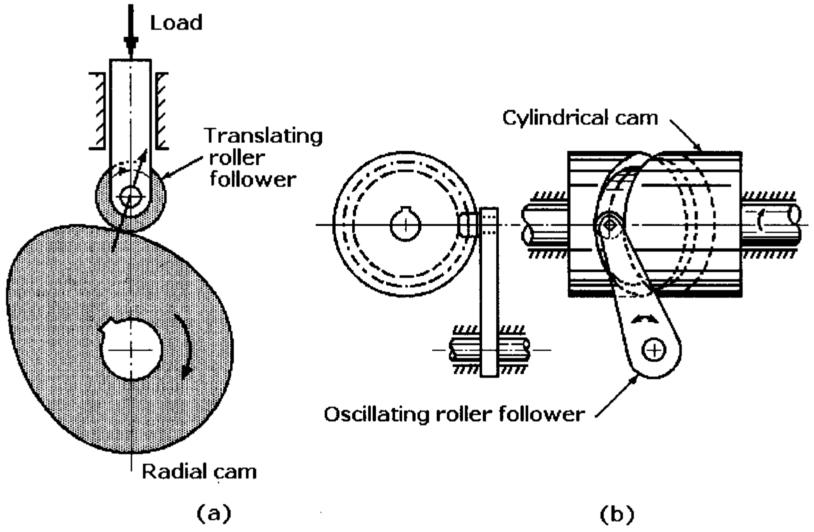


Fig. 2 Popular cams: (a) radial cam with a translating roller follower (open cam), and (b) cylindrical cam with an oscillating roller follower (closed cam).

CAM-CURVE GENERATING MECHANISMS

It usually doesn't pay to design a complex cam curve if it can't be easily machined—so check these mechanisms before starting your cam design.

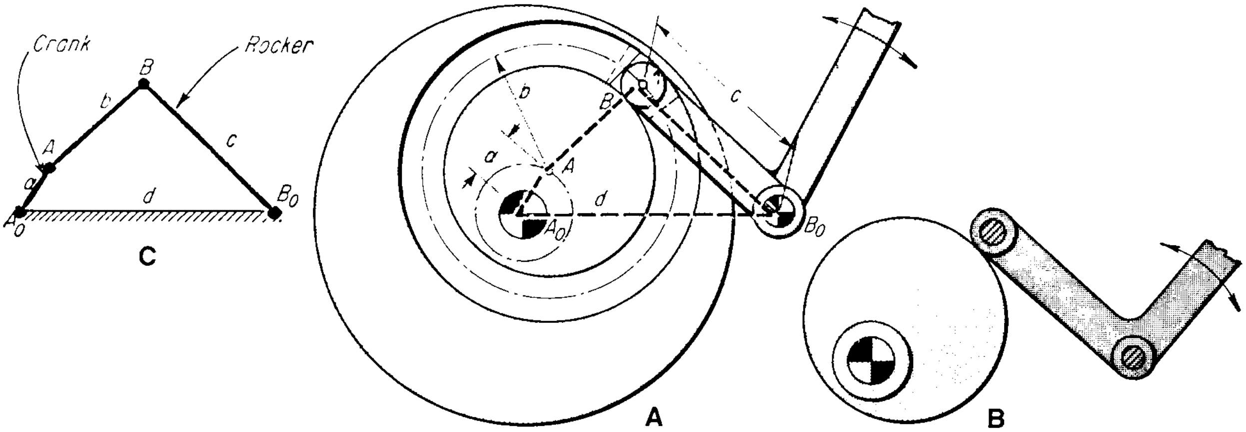


Fig. 1 A circular cam groove is easily machined on a turret lathe by mounting the plate eccentrically onto the truck. The plate cam in **(B)** with a spring-load follower produces the same output motion. Many designers are unaware that this type of cam has the same output motion as four-bar linkage **(C)** with the indicated equivalent link lengths. Thus, it's the easiest curve to pick when substituting a cam for an existing linkage.

If you have to machine a cam curve into the metal blank without a master cam, how accurate can you expect it to be? That depends primarily on how precisely the mechanism you use can feed the cutter into the cam blank. The mechanisms described here have been carefully selected for their practicability. They can be employed directly to machine the cams, or to make master cams for producing other cams.

The cam curves are those frequently employed in automatic-feed mechanisms and screw machines They are the circular, constant-velocity, simple-harmonic, cycloidal, modified cycloidal, and circular-arc cam curve, presented in that order.

Circular Cams

This is popular among machinists because of the ease in cutting the groove. The cam (Fig. 1A) has a circular groove whose center, A, is displaced a distance a from the cam-plate center, A_0, can simply be a plate cam with a spring-loaded follower (Fig. 1B).

Interestingly, with this cam you can easily duplicate the motion of a four-bar linkage (Fig. 1C). Rocker BB_0 in Fig. 1C, therefore, is equivalent to the motion of the swinging follower shown in Fig. 1A.

The cam is machined by mounting the plate eccentrically on a lathe. Consequently, a circular groove can be cut to close tolerances with an excellent surface finish.

If the cam is to operate at low speeds, you can replace the roller with an arc-formed slide. This permits the transmission of high forces. The optimum design of these "power cams" usually requires time-consuming computations.

The disadvantages (or sometimes, the advantage) of the circular-arc cam is that, when traveling from one given point, its follower reaches higher-speed accelerations than with other equivalent cam curves.

Constant-Velocity Cams

A constant-velocity cam profile can be generated by rotating the cam plate and feeding the cutter linearly, both with uniform velocity, along the path the translating roller follower will travel later (Fig. 2A). In the example of a swinging follower, the tracer (cutter) point is placed on an arm whose length is equal to the length of the swinging roller follower, and the arm is rotated with uniform velocity (Fig. 2B).

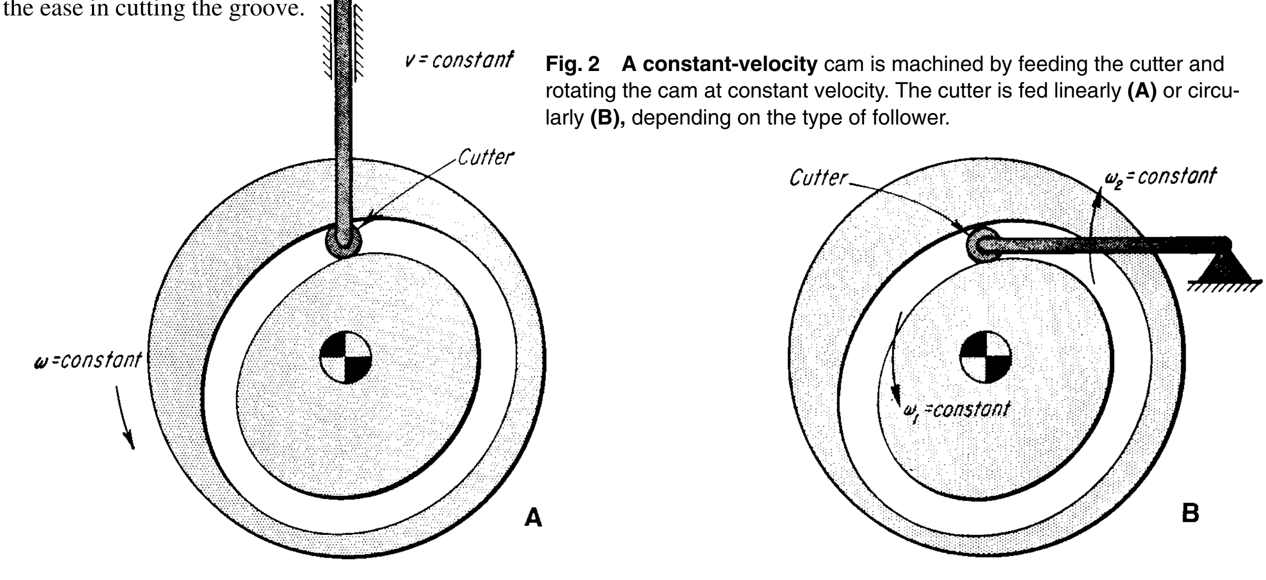


Fig. 2 A constant-velocity cam is machined by feeding the cutter and rotating the cam at constant velocity. The cutter is fed linearly **(A)** or circularly **(B),** depending on the type of follower.

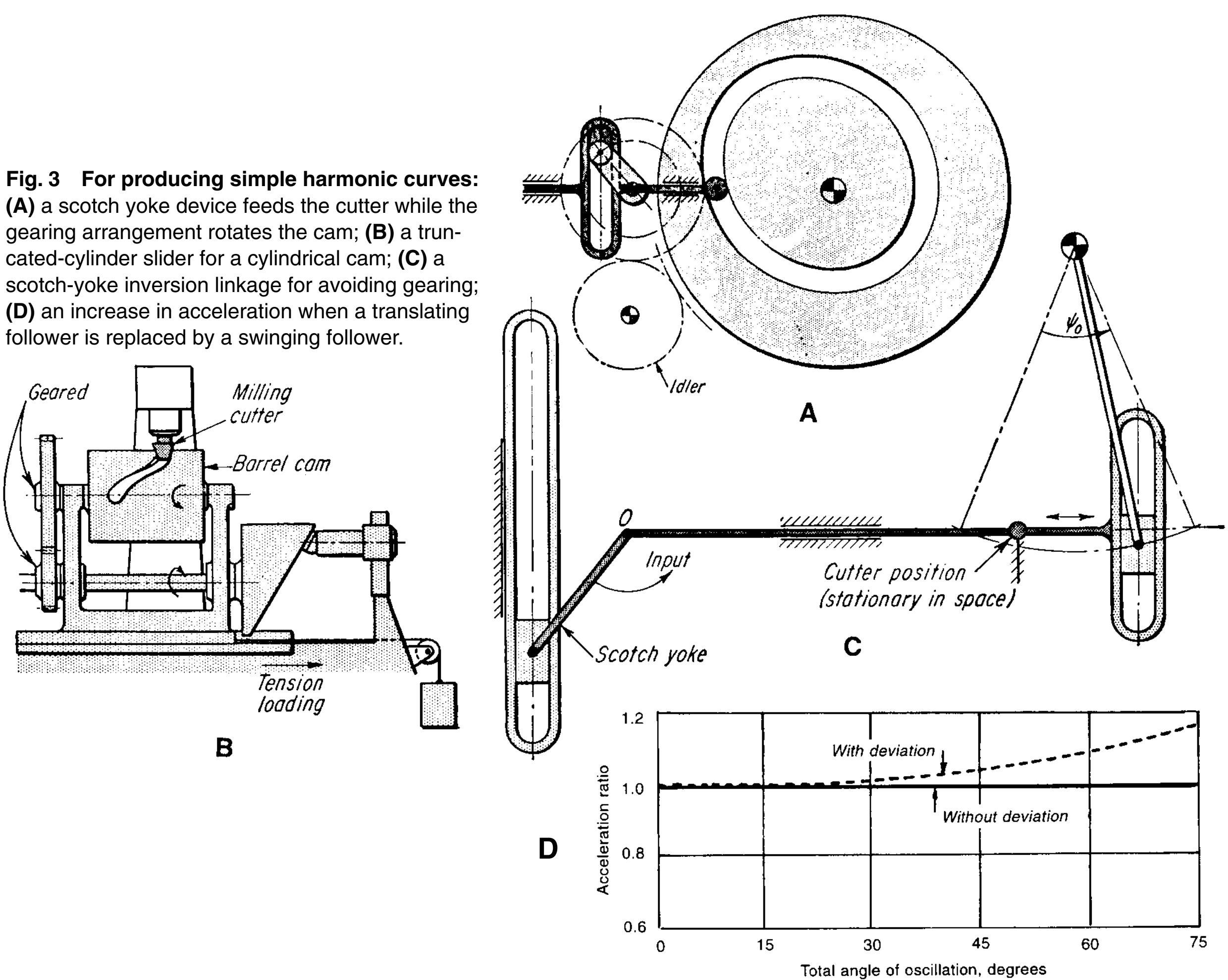


Fig. 3 For producing simple harmonic curves: (A) a scotch yoke device feeds the cutter while the gearing arrangement rotates the cam; **(B)** a truncated-cylinder slider for a cylindrical cam; **(C)** a scotch-yoke inversion linkage for avoiding gearing; **(D)** an increase in acceleration when a translating follower is replaced by a swinging follower.

Simple-Harmonic Cams

The cam is generated by rotating it with uniform velocity and moving the cutter with a scotch yoke geared to the rotary motion of the cam. Fig. 3A shows the principle for a radial translating follower; the same principle is applicable for offset translating and the swinging roller follower. The gear ratios and length of the crank working in the scotch yoke control the pressures angles (the angles for the rise or return strokes).

For barrel cams with harmonic motion, the jig in Fig. 3B can easily be set up to do the machining. Here, the barrel cam is shifted axially by the rotating, weight-loaded (or spring-loaded) truncated cylinder.

The scotch-yoke inversion linkage (Fig. 3C) replaces the gearing called for in Fig. 3A. It will cut an approximate simple-harmonic motion curve when the cam has a swinging roller follower, and an exact curve when the cam has a radial or offset translating roller follower. The slotted member is fixed to the machine frame *1*. Crank 2 is driven around the center *0*. This causes link *4* to oscillate back and forward in simple harmonic motion. The sliding piece *5* carries the cam to be cut, and the cam is rotated around the center of *5* with uniform velocity. The length of arm *6* is made equal to the length of the swinging roller follower of the actual am mechanism and the device adjusted so that the extreme position of the center of *5* lie on the center line of *4*.

The cutter is placed in a stationary spot somewhere along the centerline of member *4*. If a radial or offset translating roller follower is used, sliding piece *5* is fastened to *4*.

The deviation from simple harmonic motion, when the cam has a swinging follower, causes an increase in acceleration ranging from 0 to 18% (Fig. 3D), which depends on the total angle of oscillation of the follower. Note that for a typical total oscillating angle of 45° the increase in acceleration is about 5%.

Cycloidal Motion

This curve is perhaps the most desirable from a designer's viewpoint because of its excellent acceleration characteristic. Luckily, this curve is comparatively easy to generate. Before selecting the mechanism, it is worth looking at the underlying theory of cycloids because it is possible to generate not only cycloidal motion but a whole family of similar curves.

The cycloids are based on an offset sinusoidal wave (Fig. 4). Because the

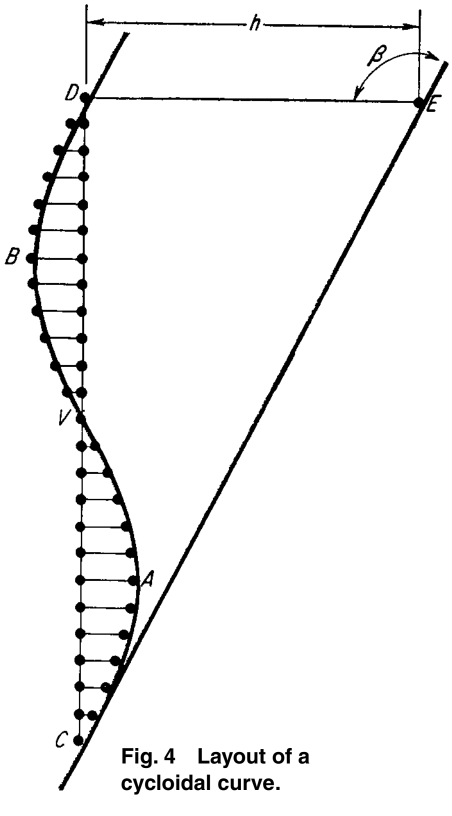


Fig. 4 Layout of a cycloidal curve.

radii of curvatures in points *C*, *V*, and *D* are infinite (the curve is "flat" at these points), if this curve was a cam groove and moved in the direction of line *CVD*, a translating roller follower, actuated by this cam, would have zero acceleration at points *C*, *V*, and *D* no matter in what direction the follower is pointed.

Now, if the cam is moved in the direction of *CE* and the direction of motion of the translating follower is lined up perpendicular to *CE*, the acceleration of the follower in points, *C*, *V*, and *D* would still be zero. This has now become the basic cycloidal curve, and it can be considered as a sinusoidal curve of a certain amplitude (with the amplitude measured perpendicular to the straight line) superimposed on a straight (constant-velocity) line.

The cycloidal is considered to be the best standard cam contour because of its low dynamic loads and low shock and vibration characteristics. One reason for these outstanding attributes is that sudden changes in acceleration are avoided during the cam cycle. But improved performance is obtainable with certain modified cycloidals.

Modified Cycloids

To modify the cycloid, only the direction and magnitude of the amplitude need to be changed, while keeping the radius of curvature infinite at points *C*, *V*, and *D*.

Comparisons are made in Fig. 5 of some of the modified curves used in industry. The true cycloidal is shown in the cam diagram of Fig. 5A. Note that the sine amplitudes to be added to the constant-velocity line are perpendicular to the base. In the Alt modification shown in Fig. 5B (named after Hermann Alt, a German kinematician who first analyzed it), the sine amplitudes are perpendicular to the constant-velocity line. This results in improved (lower) velocity characteristics (Fig. 5D), but higher acceleration magnitudes (Fig. 5E).

The Wildt modified cycloidal (after Paul Wildt) is constructed by selecting a point *w* which is 0.57 the distance *T/2*, and then drawing line *wp* through *yp* which is midway along *OP*. The base of the sine curve is then constructed perpendicular to *yw*. This modification results in a maximum acceleration of 5.88 h/T^2. By contrasts, the standard cycloidal curve has a maximum acceleration of 6.28 h/T^2. This is a 6.8 reduction in acceleration.

(It's a complex task to construct a cycloidal curve to go through a particular point *P*—where *P* might be anywhere within the limits of the box in Fig. 5C—and with a specific scope at *P*. There is a growing demand for this kind of cycloidal modification.

Generating Modified Cycloidals

One of the few methods capable of generating the family of modified cycloidals consists of a double carriage and rack arrangement (Fig. 6A).

The cam blank can pivot around the spindle, which in turn is on the movable carriage I. The cutter center is stationary. If the carriage is now driven at constant speed by the leadscrew in the direction of the arrow, steel bands 1 and 2 will also cause the cam blank to rotate. This rotation-and-translation motion of the cam will cut a spiral groove.

For the modified cycloidals, a second motion must be imposed on the cam to compensate for the deviations from the

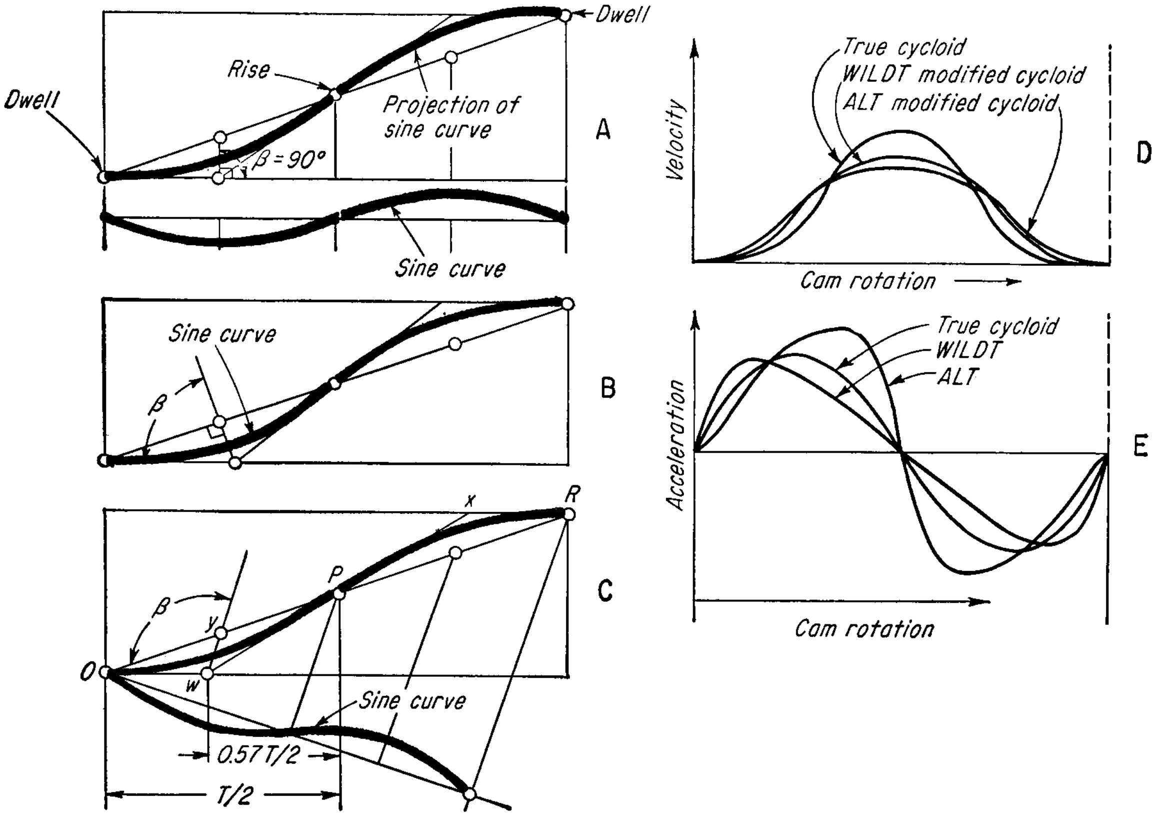


Fig. 5 A family of cycloidal curves: (A) A standard cycloidal motion; **(B)** A modification according to H. Alt; **(C)** A modification according to P. Wildt; **(D)** A comparison of velocity characteristics; **(E)** A comparison of acceleration curves.

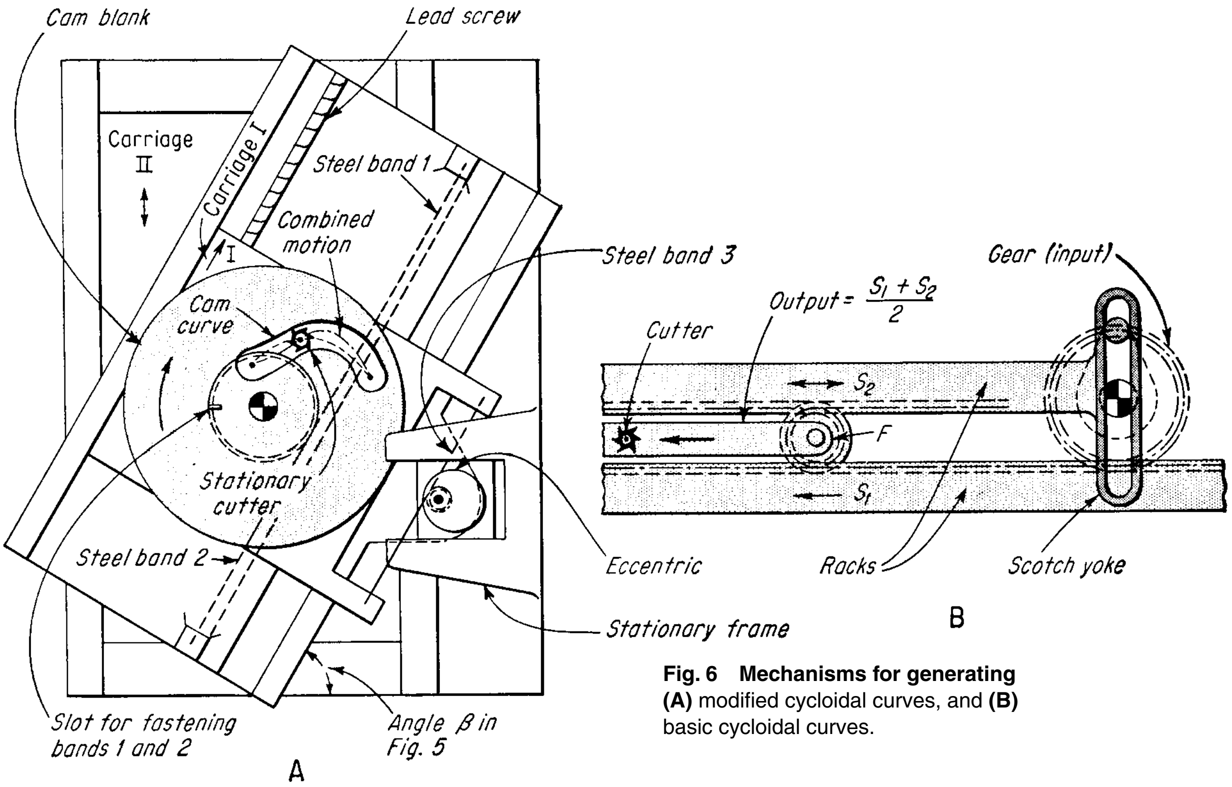


Fig. 6 Mechanisms for generating (A) modified cycloidal curves, and **(B)** basic cycloidal curves.

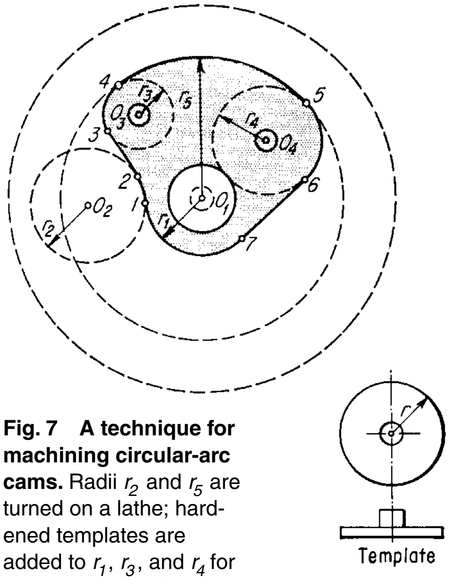


Fig. 7 A technique for machining circular-arc cams. Radii r_2 and r_5 are turned on a lathe; hardened templates are added to r_1, r_3, and r_4 for facilitating hand filing.

true cycloidal. This is done by a second steel-band arrangement. As carriage I moves, bands 3 and 4 cause the eccentric to rotate. Because of the stationary frame, the slide surrounding the eccentric is actuated horizontally. This slide is part of carriage II. As a result, a sinusoidal motion is imposed on the cam.

Carriage I can be set at various angles β to match angle β in Fig. 5B and C. The mechanism can also be modified to cut cams with swinging followers.

Circular-Arc Cams

In recent years it has become customary to turn to the cycloidal and other similar curves even when speeds are low. However, there are still many applications for circular-arc cams. Those cams are composed of circular arcs, or circular arc and straight lines. For comparatively small cams, the cutting technique illustrated in Fig. 7 produces accurate results.

Assume that the contour is composed of circular arc *1-2* with center at 0_2, arc *3-4* with center at 0_3, arc *4*-5 with center at 0_1, arc 5-*6* with center at 0_4, arc *7-1* with center at 0_1, and the straight lines *2-3* and *6-7*. The method calls for a combination of drilling, lathe turning, and template filing.

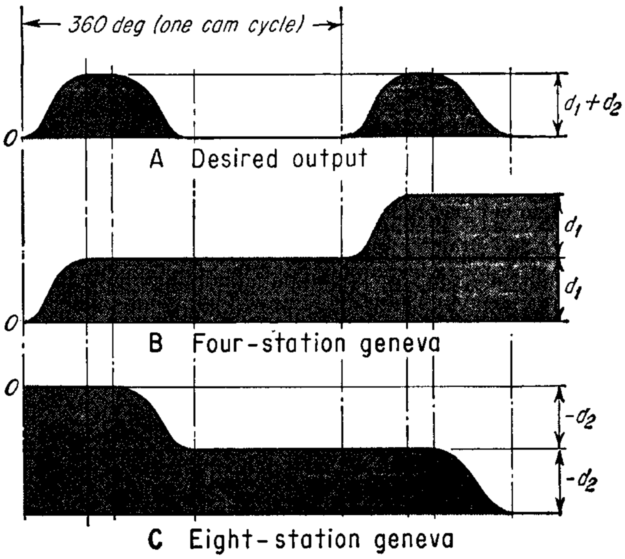
First, small holes about 0.1 in. in diameter are drilled at 0_1, 0_3, and 0_4. Then a hole drilled with the center at 0_2, and radius of r_2. Next the cam is fixed in a turret lathe with the center of rotation at 0_1, and the steel plate is cut until it has a diameter of $2r_5$. This completes the larger convex radius. The straight lines *6-7* and *2-3* are then milled on a milling machine.

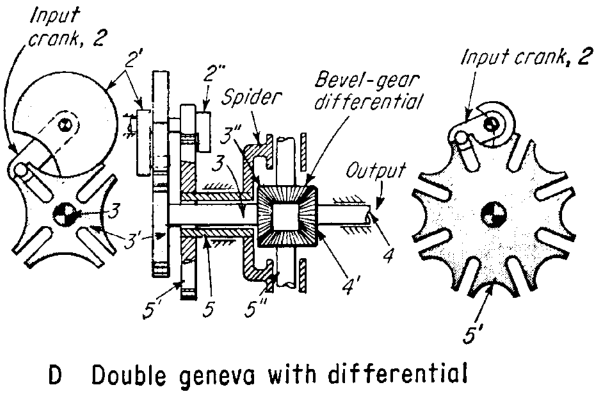
Finally, for the smaller convex arcs, hardened pieces are turned with radii r_1, r_3, and r_4. One such piece is shown in Fig. 7. The templates have hubs that fit into the drilled holes at 0_1, 0_3, and 0_4. Next the arcs *7-1*, *3-4*, and *5-6* are filed with the hardened templates as a guide. The final operation is to drill the enlarged hole at 0_1 to a size that will permit a hub to be fastened to the cam.

This method is usually better than copying from a drawing or filing the scallops from a cam on which a large number of points have been calculated to determine the cam profile.

Compensating for Dwells

One disadvantage with the previous generating machines is that, with the exception of the circular cam, they cannot include a dwell period within the rise-and-fall cam cycle. The mechanisms must be disengaged at the end of the rise, and the cam must be rotated the exact number of degrees to the point where the





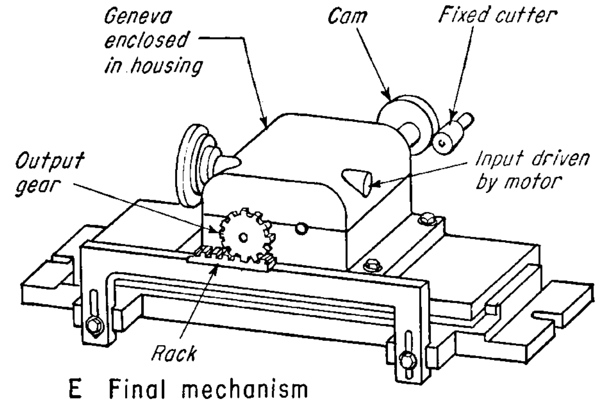


Fig. 8 Double genevas with differentials for obtaining long dwells. The desired output characteristic **(A)** of the cam is obtained by adding the motion **(B)** of a four-station geneva to that of **(C)** an eight-station geneva. The mechanical arrangement of genevas with a differential is shown in **(D)**; the actual device is shown in **(E)**. A wide variety of output dwells **(F)** are obtained by varying the angle between the driving cranks of the genevas.

fall cycle begins. This increases the possibility of inaccuracies and slows down production.

There are two mechanisms, however, that permit automatic cam machining through a specific dwell period: the double-geneva drive and the double eccentric mechanism.

Double-Genevas with Differential

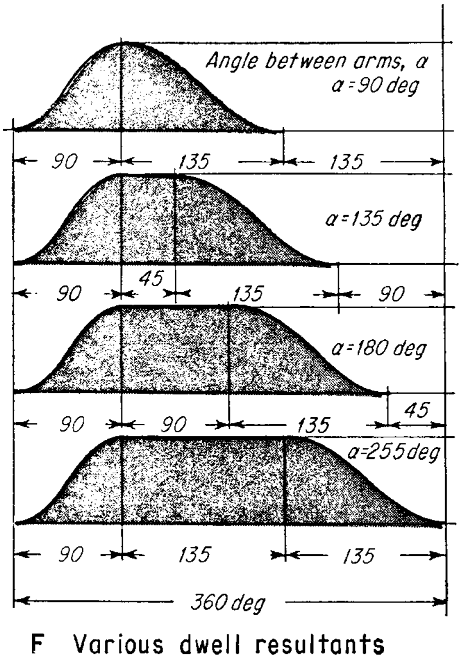
Assume that the desired output contains dells (of specific duration) at both the rise and fall portions, as shown in Fig. 8A. The output of a geneva that is being rotated clockwise will produce an intermittent motion similar to the one shown in Fig. 8B—a rise-dwell-rise-dwell motion. These rise portions are distorted simple-harmonic curves, but are sufficiently close to the pure harmonic to warrant their use in many applications.

If the motion of another geneva, rotating counterclockwise as shown in (Fig. 8C), is added to that of the clockwise geneva by a differential (Fig. 8D), then the sum will be the desired output shown in (Fig. 8A).

The dwell period of this mechanism is varied by shifting the relative positions between the two input cranks of the genevas.

The mechanical arrangement of the mechanism is shown in Fig. 8D. The two driving shafts are driven by gearing (not shown). Input from the four-star geneva to the differential is through shaft *3*; input from the eight-station geneva is through the spider. The output from the differential, which adds the two inputs, is through shaft *4*.

The actual mechanism is shown in Fig. 8E. The cutter is fixed in space. Output is from the gear segment that rides on a fixed rack. The cam is driven by the motor, which also drives the enclosed genevas. Thus, the entire device reciprocates back and forth on the slide to feed the cam properly into the cutter.



F Various dwell resultants

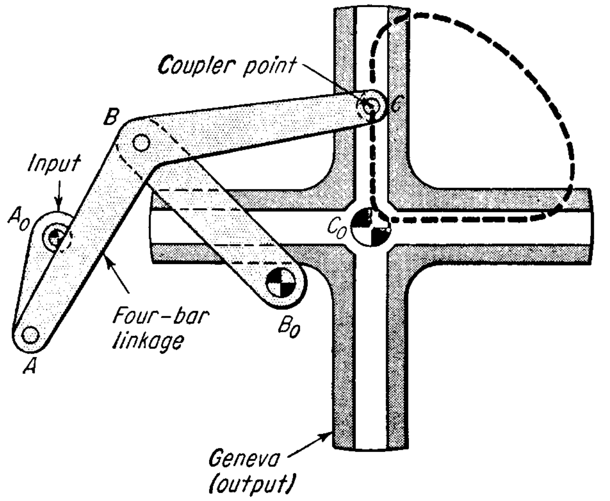


Fig. 9 A four-bar coupler mechanism for replacing the cranks in genevas to obtain smoother acceleration characteristics.

Genevas Driven by Couplers

When a geneva is driven by a constant-speed crank, as shown in Fig. 8D, it has a sudden change in acceleration at the beginning and end of the indexing cycle (as the crank enters or leaves a slot). These abrupt changes can be avoided by employing a four-bar linkage with a coupler in place of the crank. The motion of the coupler point *C* (Fig. 9) permits its smooth entry into the geneva slot

Double Eccentric Drive

This is another machine for automatically cutting cams with dwells. The rotation of crank *A* (Fig. 10) imparts an oscillating motion to the rocker *C* with a prolonged dwell at both extreme positions. The cam, mounted on the rocker, is rotated by the chain drive and then is fed into the cutter with the proper motion. During the dwells of the rocker, for example, a dwell is cut into the cam.

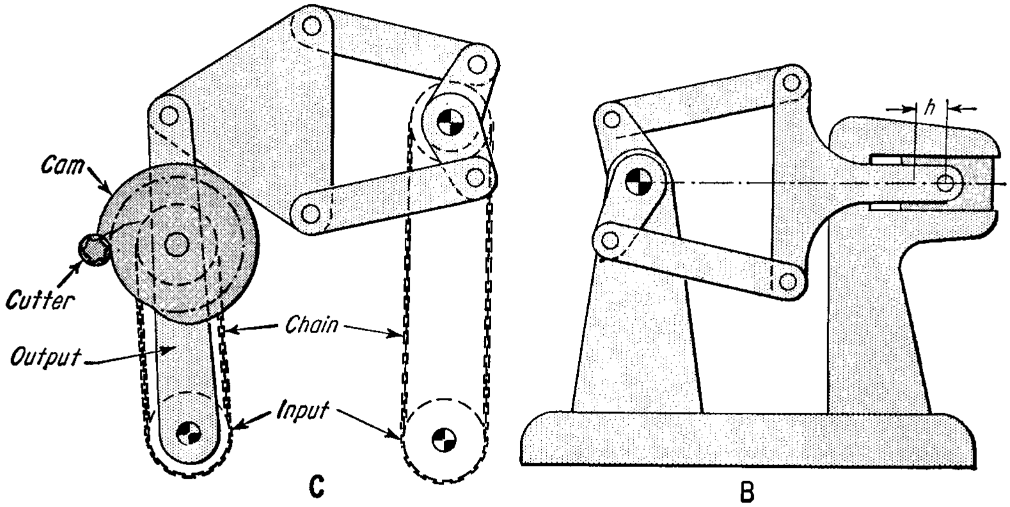
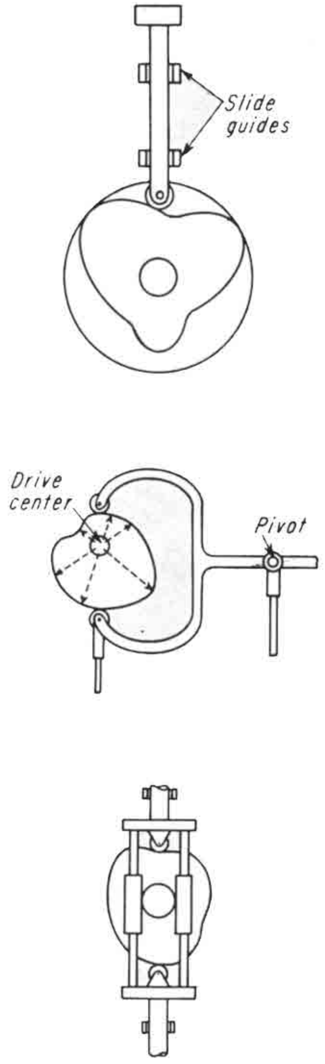


Fig. 10 A double eccentric drive for automatically cutting cams with dwells. The cam is rotated and oscillated, with dwell periods at extreme ends of oscillation corresponding to desired dwell periods in the cam.

FIFTEEN IDEAS FOR CAM MECHANISMS

This assortment of devices reflects the variety of ways in which cams can be put to work.



Figs. 1, 2, and 3 A constant-speed rotary motion is converted into a variable, reciprocating motion (Fig. 1); rocking or vibratory motion of a simple forked follower (Fig. 2); or a more robust follower (Fig. 3), which can provide valve-moving mechanisms for steam engines. Vibratory-motion cams must be designed so that their opposite edges are everywhere equidistant when they are measured through their drive-shaft centers.

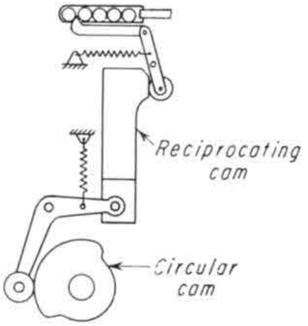


Fig. 4 An automatic feed for automatic machines. There are two cams, one with circular motion, the other with reciprocating motion. This combination eliminates any trouble caused by the irregularity of feeding and lack of positive control over stock feed.

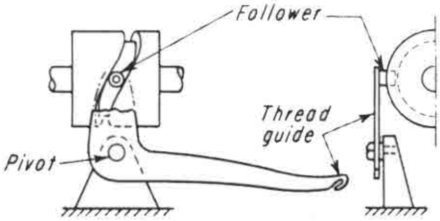


Fig. 5 A barrel cam with milled grooves is used in sewing machines to guide thread. This kind of cam is also used extensively in textile manufacturing machines such as looms and other intricate fabric-making machines.

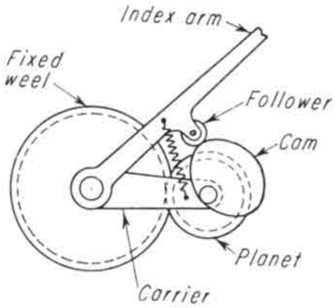


Fig. 6 This indexing mechanism combines an epicyclic gear and cam. A planetary wheel and cam are fixed relative to one another; the carrier is rotated at uniform speed around the fixed wheel. The index arm has a nonuniform motion with dwell periods.

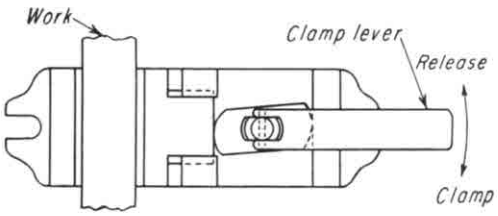


Fig. 7 A double eccentric, actuated by a suitable handle, provides powerful clamping action for a machine-tool holding fixture.

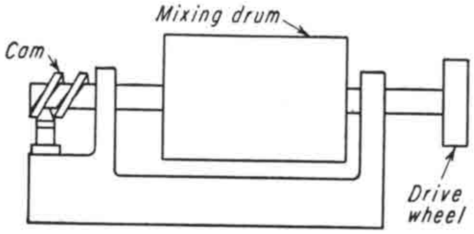


Fig. 8 A mixing roller for paint, candy, or food. A mixing drum has a small oscillating motion while rotating.

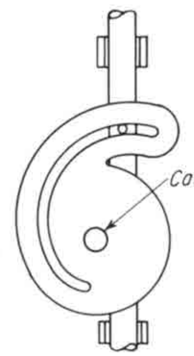


Fig. 9 A slot cam converts the oscillating motion of a camshaft to a variable but straight-line motion of a rod. According to slot shape, rod motion can be made to suit specific design requirements, such as straight-line and logarithmic motion.

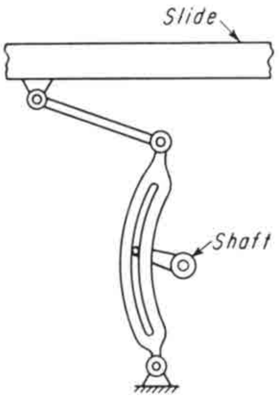


Fig. 10 The continuous rotary motion of a shaft is converted into the reciprocating motion of a slide. This device is used on sewing machines and printing presses.

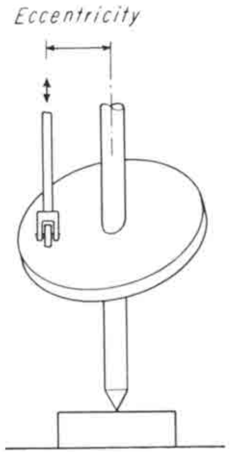


Fig. 11 Swash-plate cams are feasible for light loads only, such as in a pump. The cam's eccentricity produces forces that cause excessive loads. Multiple followers can ride on a plate, thereby providing smooth pumping action for a multipiston pump.

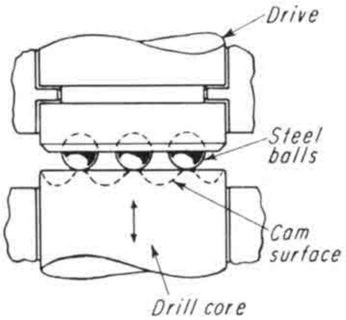


Fig. 12 This steel-ball cam can convert the high-speed rotary motion of an electric drill into high-frequency vibrations that power the drill core for use as a rotary hammer for cutting masonry, and concrete. This attachment can also be designed to fit hand drills.

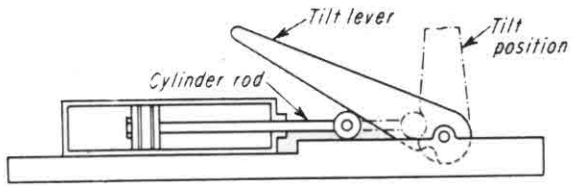


Fig. 13 This tilting device can be designed so that a lever remains in a tilted position when the cylinder rod is withdrawn, or it can be spring-loaded to return with a cylinder rod.

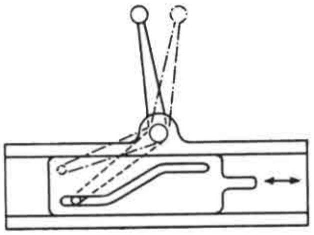


Fig. 14 This sliding cam in a remote control can shift gears in a position that is otherwise inaccessible on most machines.

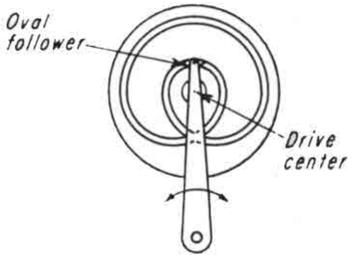


Fig. 15 A groove and oval follower form a device that requires two revolutions of a cam for one complete follower cycle.

SPECIAL-FUNCTION CAMS

Fig. 1—A quick drop of the follower is obtained by permitting the cam to be pushed out of the way by the follower itself as it reaches the edge of the cam. Lugs *C* and *C′* are fixed to the camshaft. The cam is free to turn (float) on the camshaft, limited by lug *C* and the adjusting screw. With the cam rotating clockwise, lug C drives the cam through lug *B*. At the position shown, the roller will drop off the edge of the cam, which is then accelerated clockwise until its cam lug *B* strikes the adjusting screw of lug *C′*.

Fig. 2—Instantaneous drop is obtained by the use of two integral cams and followers. The roller follower rides on cam *1*. Continued rotation will transfer contact to the flat-faced follower, which drops suddenly off the edge of cam *2*. After the desired dwell, the follower is restored to its initial position by cam *1*.

Fig. 3—The dwell period of the cam can be varied by changing the distance between the two rollers in the slot.

Fig. 4—A reciprocating pin (not shown) causes the barrel cam to rotate intermittently. The cam is stationary while a pin moves from *1* to *2*. Groove *2-3* is at a lower level; thus, as the pin retracts, it cams the barrel cam; then it climbs the incline from *2* to the new position of *1*.

Fig. 5—A double-groove cam makes two revolutions for one complete movement of the follower. The cam has movable switches, *A* and *B*, which direct the follower alternately in each groove. At the instant shown, *B* is ready to guide the roller follower from slot *1* to slot *2*.

Figs. 6 and 7—Increased stroke is obtained by permitting the cam to shift on the input shaft. Total displacement of the follower is therefore the sum of the cam displacement on the fixed roller plus the follower displacement relative to the cam.

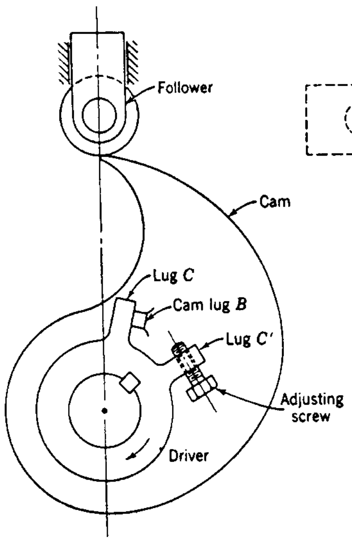


Fig. 1 **A quick-acting floating cam.**

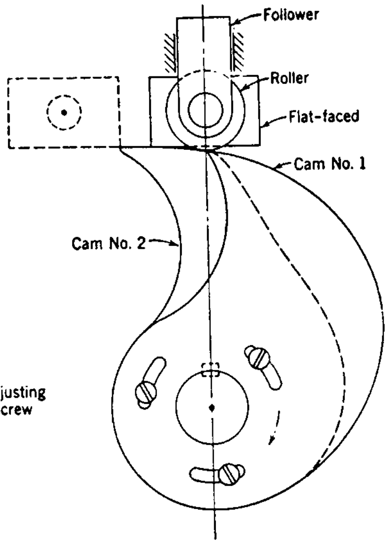


Fig. 2 **A quick-acting dwell cams.**

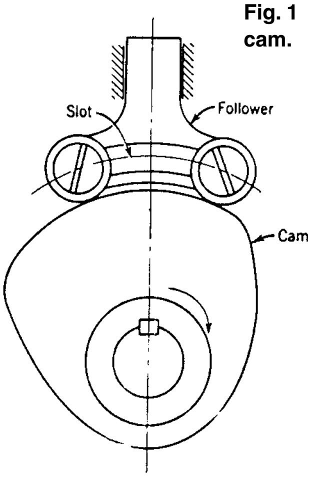


Fig. 3 **An adjustable-dwell cam.**

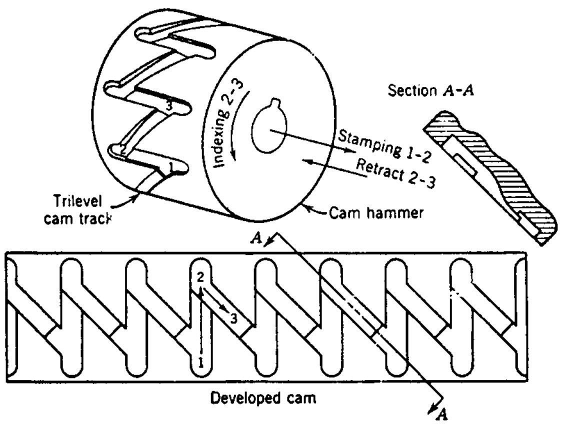


Fig. 4 **An indexing cam.**

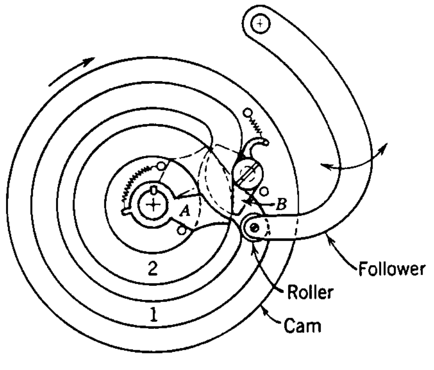


Fig. 5 **A double-revolution cam.**

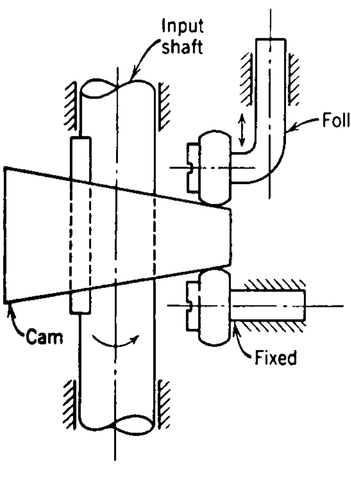


Fig. 6 **An increased-stroke barrel cam.**

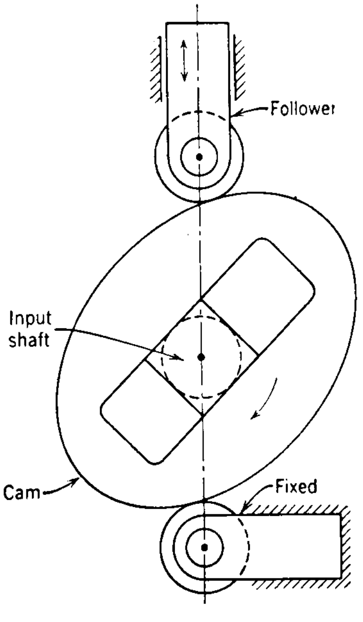


Fig. 7 **An increased-stroke plate cam.**

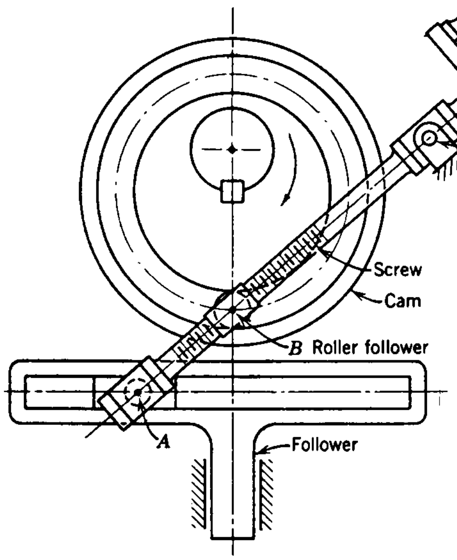
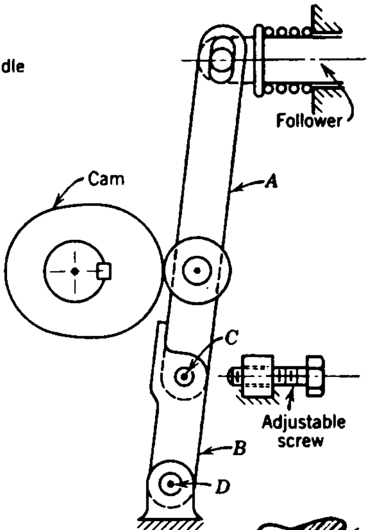


Fig. 8 **An adjustable roller-position cam.**



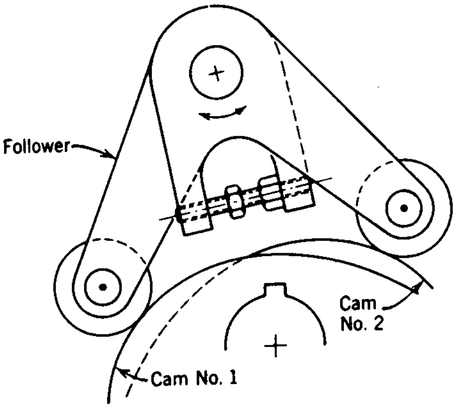


Fig. 9 **An adjustable pivot-point cam.**

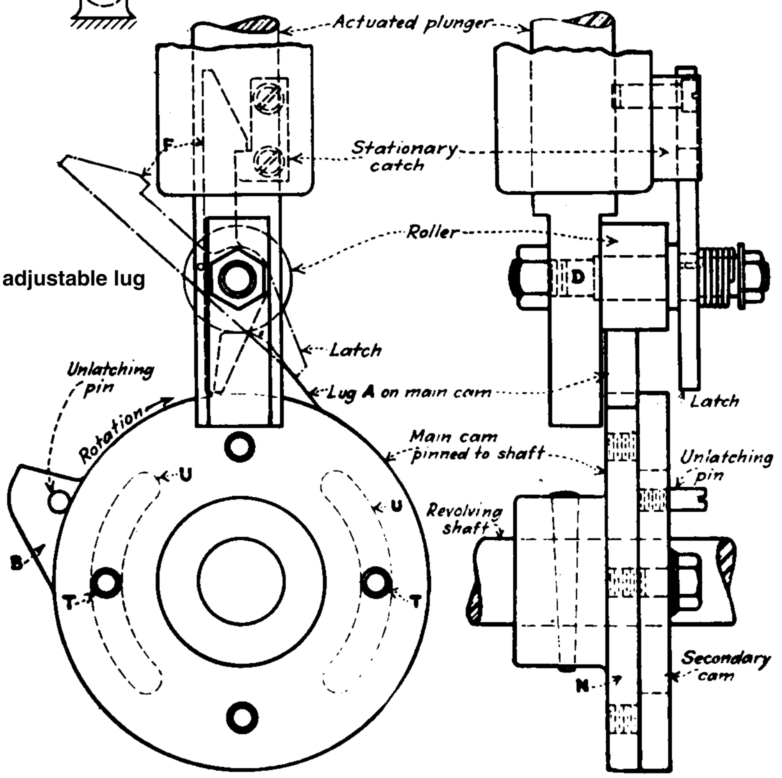


Fig. 10 **An adjustable lug cam.**

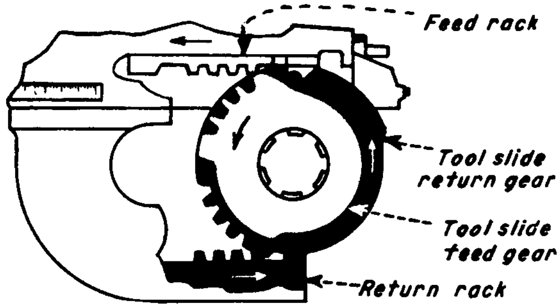
Fig. 8—The stroke of the follower is adjusted by turning the screw handle which changes distance *AB*.

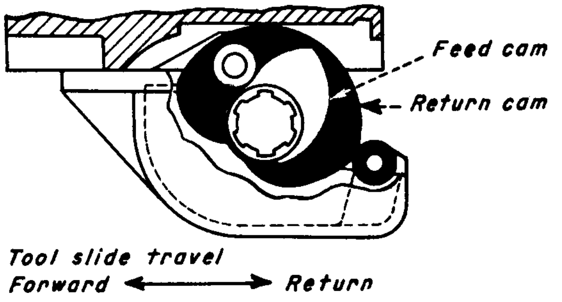
Fig. 9—The pivot point of the connecting link to the follower is changed from point *D* to point *C* by adjusting the screw.

Fig. 10—Adjustable dwell is obtained by having the main cam, with lug *A*, pinned to the revolving shaft. Lug *A* forces the plunger up into the position shown, and allows the latch to hook over the catch, thus holding the plunger in the up position. The plunger is unlatched by lug *B*. The circular slots in the cam plate permit the shifting of lug *B*, thereby varying the time that the plunger is held in the latched position.

REFERENCE: Rothbart, H. A. *Cams—Design, Dynamics, and Accuracy*, John Wiley and Sons, Inc., New York.

CAM DRIVES FOR MACHINE TOOLS

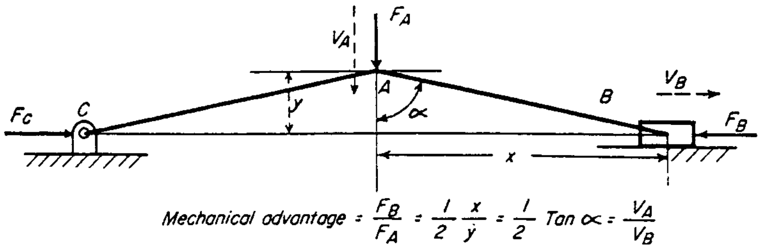




This two-directional rack-and-gear drive for a main tool slide combines accurate, uniform movement and minimum idle time. The mechanism makes a full double stroke each cycle. It approaches fast, shifts smoothly into feed, and returns fast. Its point-of-shift is controlled by an adjustable dog on a calibrated gear. Automatic braking action assures a smooth shift from approach to feed.

A cam drive for a tool-slide mechanism replaces a rack feed when a short stroke is required to get a fast machining cycle on automatic machines. The cams and rollers are shown with the slide in its retracted position.

TOGGLE LINKAGE APPLICATIONS IN DIFFERENT MECHANISMS



$$\text{Mechanical advantage} = \frac{F_B}{F_A} = \frac{1}{2}\frac{x}{y} = \frac{1}{2}\operatorname{Tan}\alpha = \frac{V_A}{V_B}$$

Fig. 1 Many mechanical linkages are based on the simple toggle that consists of two links which tend to line up in a straight line at one point in their motion. The mechanical advantage is the velocity ratio of the input point *A* with respect to the outpoint point *B*: or V_A/V_B. As the angle α approaches 90º, the links come into toggle, and the mechanical advantage and velocity ratio both approach infinity. However, frictional effects reduce the forces to much les than infinity, although they are still quite high.

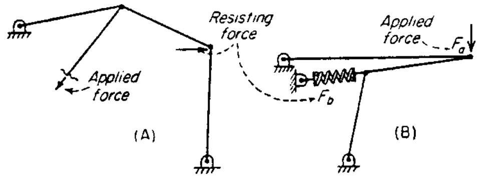


Fig. 2 Forces can be applied through other links, and need not be perpendicular to each other. (A) One toggle link can be attached to another link rather than to a fixed point or slider. (B) Two toggle links can come into toggle by lining up on top of each other rather than as an extension of each other. The resisting force can be a spring.

HIGH MECHANICAL ADVANTAGE

Fig. 3 In punch presses, large forces are needed at the lower end of the work stroke. However, little force is required during the remainder of the stroke. The crank and connecting rod come into toggle at the lower end of the punch stroke, giving a high mechanical advantage at exactly the time it is most needed.

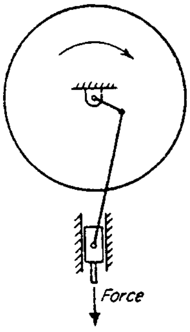
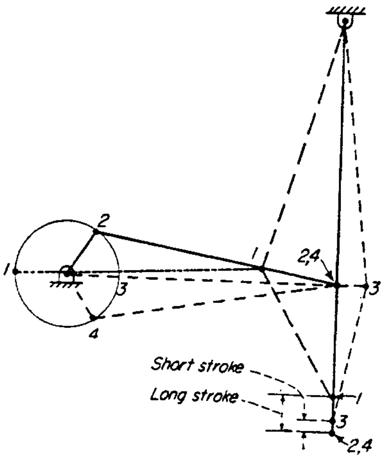


Fig. 4 A cold-heading rivet machine is designed to give each rivet two successive blows. Following the first blow (point 2) the hammer moves upward a short distance (to point 3). Following the second blow (at point 4), the hammer then moves upward a longer distance (to point 1) to provide clearance for moving the workpiece. Both strokes are produced by one revolution of the crank, and at the lowest point of each stroke (points 2 and 4) the links are in toggle.



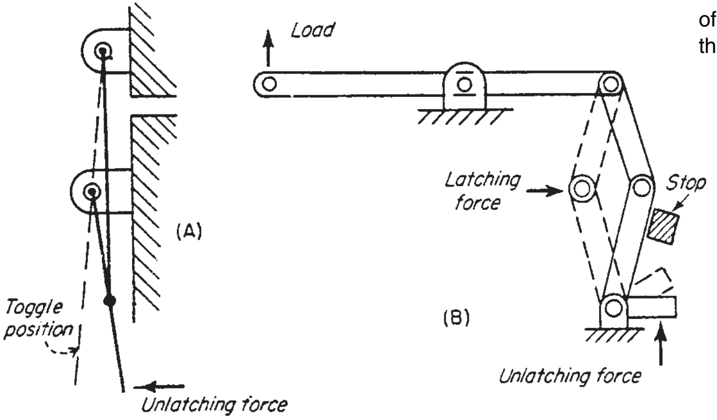


Fig. 5 Locking latches produce a high mechanical advantage when in the toggle portion of the stroke. A simple latch exerts a large force in the locked position (Fig. 5A). For positive locking, the closed position of latch is slightly beyond the toggle position. A small unlatching force opens the linkage (Fig. 5B).

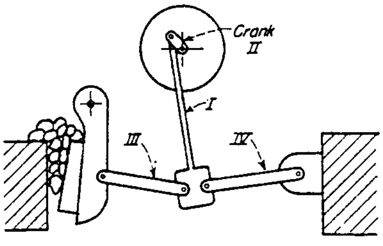
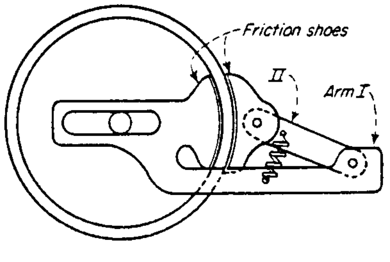
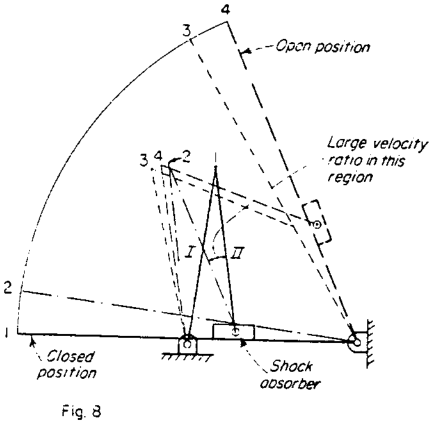


Fig. 6 A stone crusher has two toggle linkages in series to obtain a high mechanical advantage. When the vertical link *I* reaches the top of its stroke, it comes into toggle with the driving crank *II*; at the same time, link *III* comes into toggle and link *IV*. This multiplication results in a very large crushing force.

Fig. 7 A friction ratchet is mounted on a wheel; a light spring keeps the friction shoes in contact with the flange. This device permits clockwise motion of the arm *I*. However, reverse rotation causes friction to force link *II* into toggle with the shoes. This action greatly increases the locking pressure.



HIGH VELOCITY RATIO



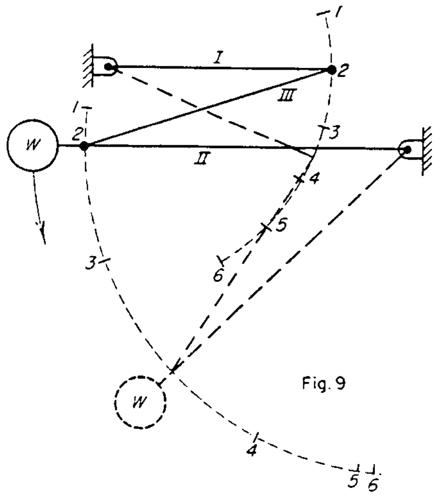


Fig. 8 Door check linkage gives a high velocity ratio during the stroke. As the door swings closed, connecting link *I* comes into toggle with the shock absorber arm *II*, giving it a large angular velocity. The shock absorber is more effective in retarding motion near the closed position.

Fig. 9 An impact reducer is on some large circuit breakers. Crank *I* rotates at constant velocity while the lower crank moves slowly at the beginning and end of the stroke. It moves rapidly at the midstroke when arm *II* and link *III* are in toggle. The accelerated weight absorbs energy and returns it to the system when it slows down.

VARIABLE MECHANICAL ADVANTAGE

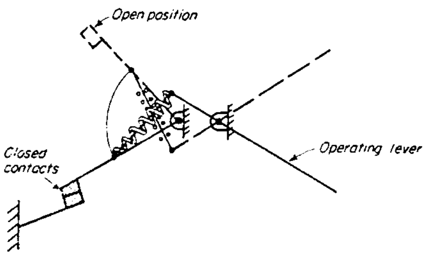


Fig. 10 A toaster switch has an increasing mechanical advantage to aid in compressing a spring. In the closed position, the spring holds the contacts closed and the operating lever in the down position. As the lever is moved upward, the spring is compressed and comes into toggle with both the contact arm and the lever. Little effort is required to move the links through the toggle position; beyond this point, the spring snaps the contacts open. A similar action occurs on closing.

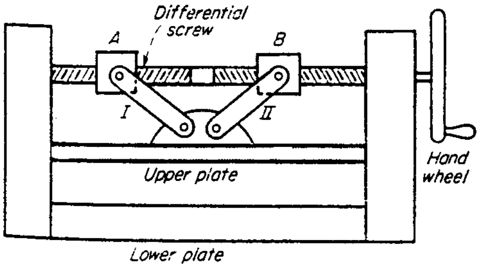


Fig. 11 A toggle press has an increasing mechanical advantage to counteract the resistance of the material being compressed. A rotating handwheel with a differential screw moves nuts *A* and *B* together, and links *I* and *II* are brought into toggle.

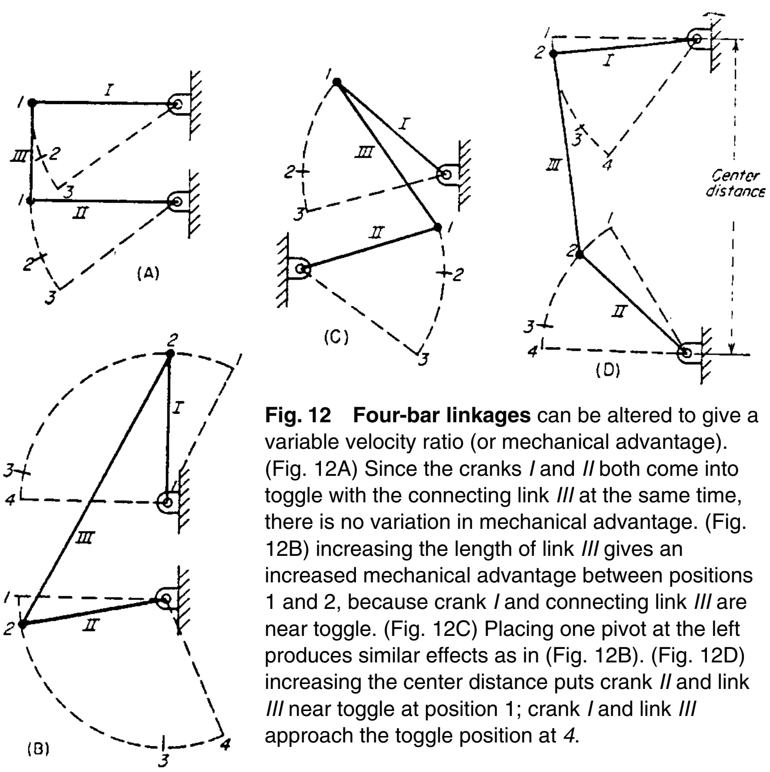


Fig. 12 Four-bar linkages can be altered to give a variable velocity ratio (or mechanical advantage). (Fig. 12A) Since the cranks *I* and *II* both come into toggle with the connecting link *III* at the same time, there is no variation in mechanical advantage. (Fig. 12B) increasing the length of link *III* gives an increased mechanical advantage between positions 1 and 2, because crank *I* and connecting link *III* are near toggle. (Fig. 12C) Placing one pivot at the left produces similar effects as in (Fig. 12B). (Fig. 12D) increasing the center distance puts crank *II* and link *III* near toggle at position 1; crank *I* and link *III* approach the toggle position at *4*.

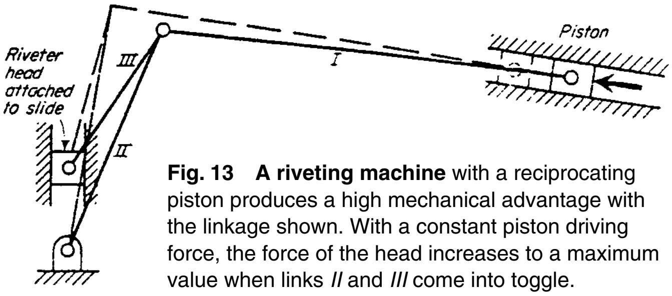


Fig. 13 A riveting machine with a reciprocating piston produces a high mechanical advantage with the linkage shown. With a constant piston driving force, the force of the head increases to a maximum value when links *II* and *III* come into toggle.

SIXTEEN LATCH, TOGGLE, AND TRIGGER DEVICES

Diagrams of basic latching and quick-release mechanisms.

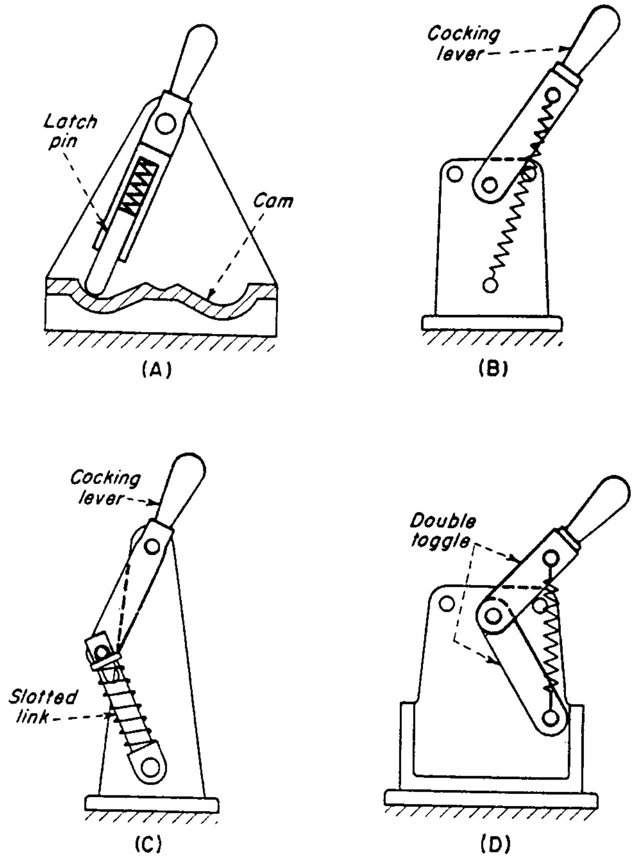


Fig. 1 Cam-guided latch (A) has one cocked, and two relaxed positions, (B) Simple overcenter toggle action. (C) An overcenter toggle with a slotted link. (D) A double toggle action often used in electrical switches.

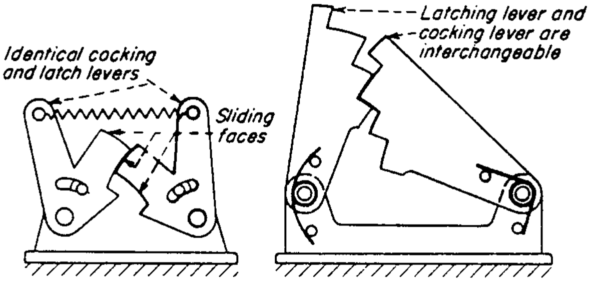


Fig. 2 An identically shaped cocking lever and latch (A) allow their functions to be interchangeable. The radii of the sliding faces must be dimensioned for a mating fit. The stepped latch (B) offers a choice of several locking positions.

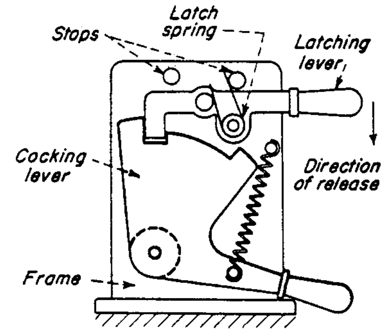


Fig. 3 A latch and cocking lever is spring-loaded so that latch movement releases the cocking lever. The cocked position can be held indefinitely. Studs in the frame provide stops, pivots, or mounts for the springs.

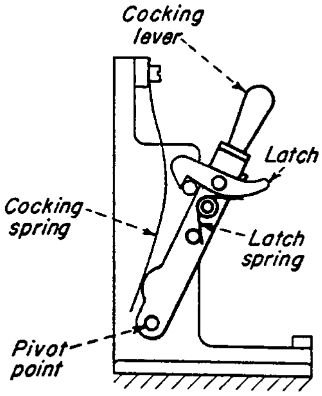


Fig. 4 A latch mounted on a cocking lever allows both levers to be reached at the same time with one hand. After release, the cocking spring initiates clockwise lever movement; then gravity takes over.

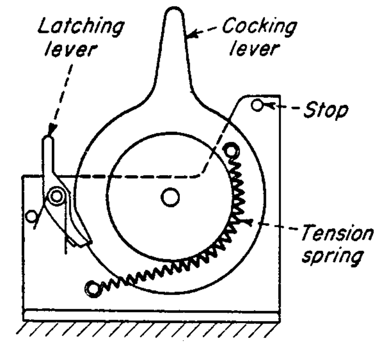


Fig. 5 A disk-shaped cocking has a tension spring resting against the cylindrical hub. Spring force always acts at a constant radius from the lever pivot point.

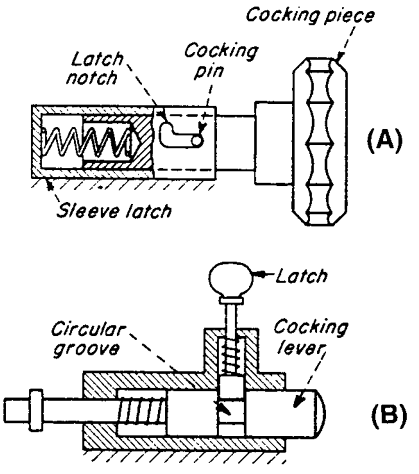


Fig. 6 A sleeve latch (A) as an L-shaped notch. A pin in the shaft rides in a notch. Cocking requires a simple push and twist action. (B) The Latch and plunger depend on axial movement for setting and release. A circular groove is needed if the plunger is to rotate.

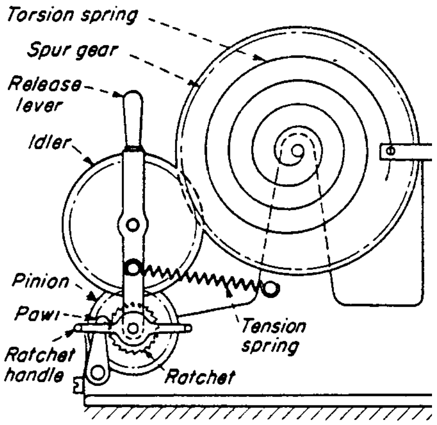
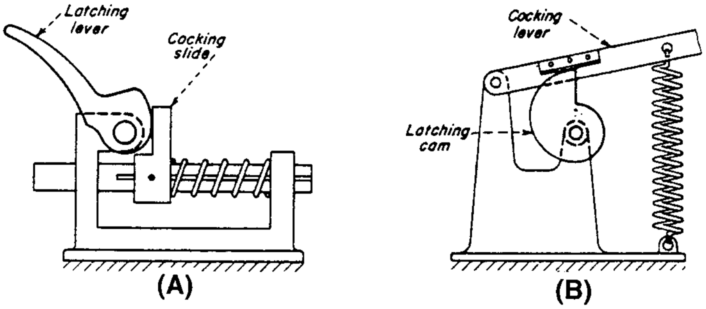


Fig. 7 A geared cocking device has a ratchet fixed to a pinion. A torsion spring exerts clockwise force on the spur gear; a tension spring holds the gar in mesh. The device is wound by turning the ratchet handle counterclockwise, which in turn winds the torsion spring. Moving the release-lever permits the spur gear to unwind to its original position without affecting the ratchet handle.

Fig. 8 In this overcenter lock (A) clockwise movement of the latching lever cocks and locks the slide. A counterclockwise movement is required to release the slide. (B) A latching-cam cocks and releases the cocking lever with the same counterclockwise movement as (A).



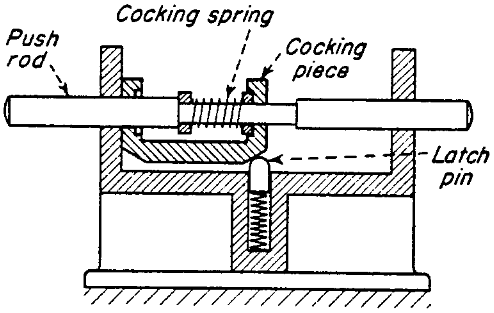


Fig. 9 A spring-loaded cocking piece has chamfered corners. Axial movement of the push-rod forces the cocking piece against a spring-loaded ball or pin set in a frame. When cocking builds up enough force to overcome the latch-spring, the cocking piece snaps over to the right. The action can be repeated in either direction.

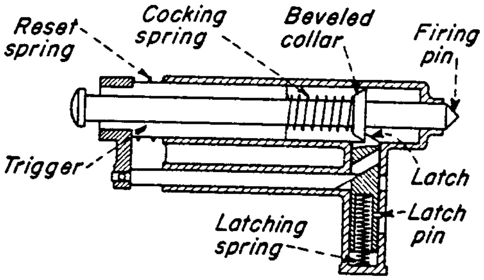


Fig. 10 A firing-pin mechanism has a beveled collar on a pin. Pressure on the trigger forces the latch down until it releases the collar when the pin snaps out, under the force of cocking the spring. A reset spring pulls the trigger and pin back. The latch is forced down by a beveled collar on a pin until it snaps back, after overcoming the force of the latch spring. (A latch pin retains the latch if the trigger and firing pin are removed.)

SIX SNAP-ACTION MECHANISMS

These diagrams show six basic ways to produce mechanical snap action.

Mechanical snap action results when a force is applied to a device over a period of time; buildup of this force to a critical level causes a sudden motion to occur. The ideal snap device would have no motion until the force reached a critical level. This, however, is not possible, and the way in which the mechanism approaches this ideal is a measure of its efficiency as a snap device. Some of the designs shown here approach the ideal closely; others do not, but they have other compensating good features.

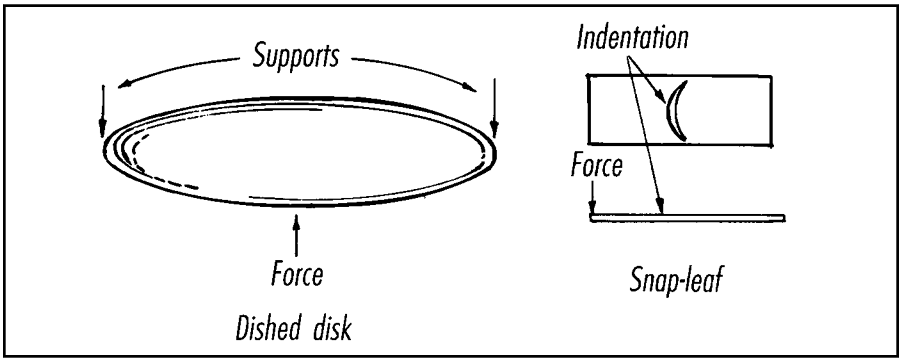


Fig. 1 **A dished disk** is a simple, common method for producing snap action. A snap leaf made from spring material can have various-shaped impressions stamped at the point where the overcentering action occurs. A "Frog clacker" is, of course, a typical applications. A bimetal element made in this way will reverse itself at a predetermined temperature.

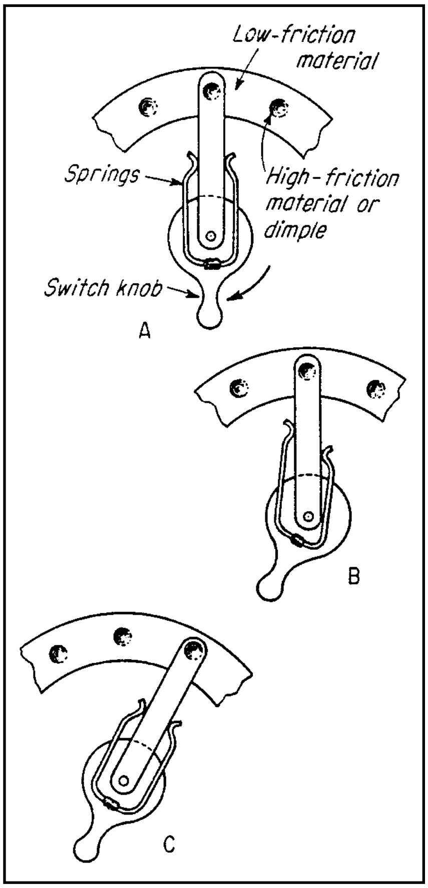


Fig. 2 **Friction override** can hold against an increasing load until friction is suddenly overcome. This is a useful action for small sensitive devices where large forces and movements are undesirable. This is the way we snap our fingers. That action is probably the original snap mechanism.

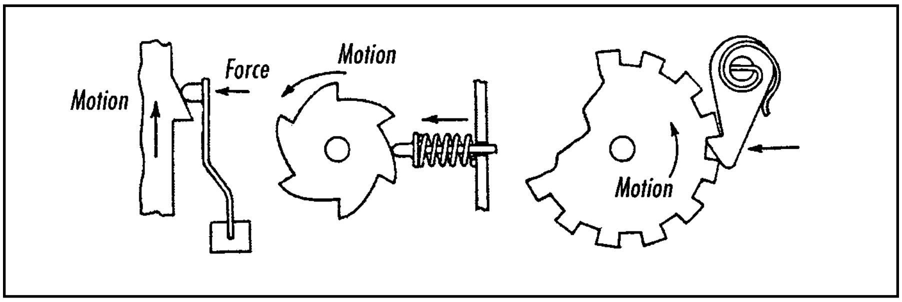


Fig. 3 **A ratchet-and-pawl** combination is probably the most widely used form of snap mechanism. Its many variations are an essential feature in practically every complicated mechanical device. By definition, however, this movement is not true snap-action.

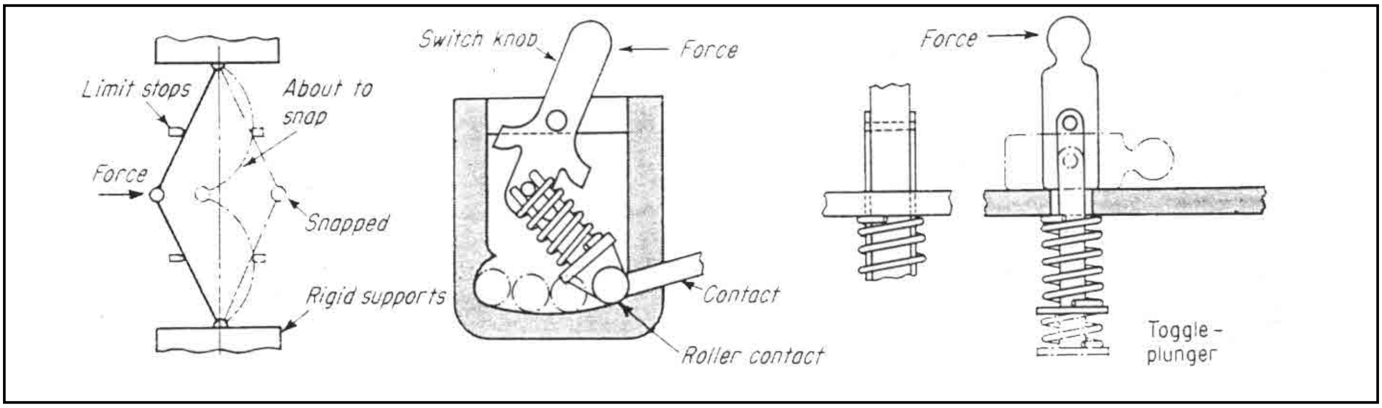


Fig. 4 Over-centering mechanisms find many applications in electrical switches. Considerable design ingenuity has been applied to fit this principle into many different mechanisms. It is the basis of most snap-action devices.

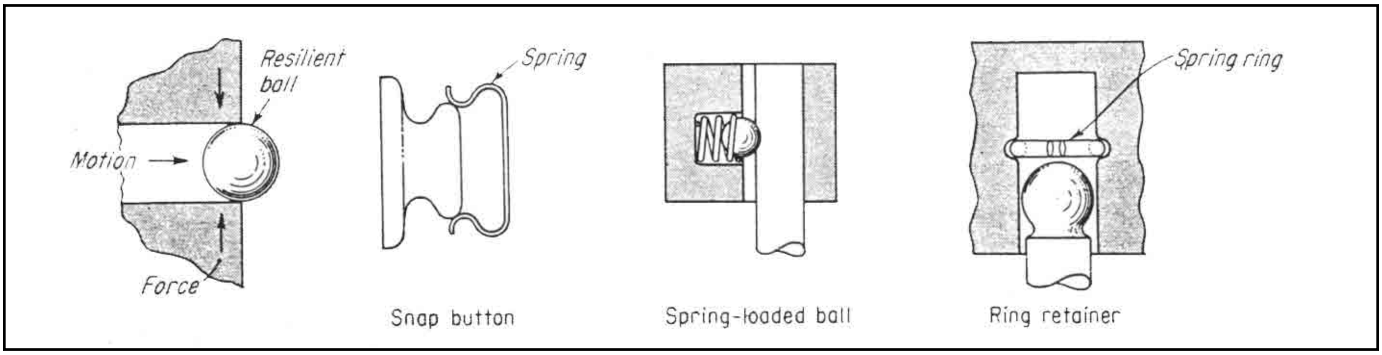


Fig. 5 The sphere ejection principle is based on snap buttons, spring-loaded balls and catches, and retaining-rings for fastening that must withstand repeated use. Their action can be designed to provide either easy or difficult removal. Wear can change the force required.

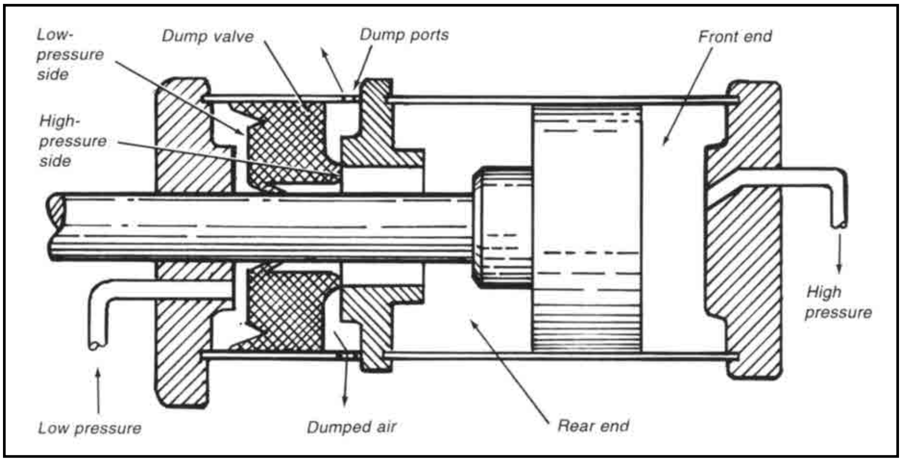


Fig. 6 A pneumatic dump valve produces snap action by preventing piston movement until air pressure has built up in the front end of the cylinder to a relatively high pressure. Dump-valve area in the low-pressure end is six times larger than its area on the high-pressure side. Thus the pressure required on the high-pressure side to dislodge the dump valve from its seat is six times that required on the low-pressure side to keep the valve properly seated.

EIGHT SNAP-ACTION DEVICES

Another selection of basic devices for obtaining sudden motion after a gradual buildup of force.

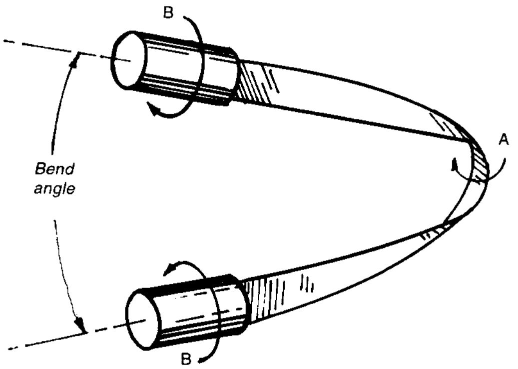


Fig. 1 A torsion ribbon bent as shown will turn “inside out” at A with a snap action when twisted at B. Design factors are ribbon width, thickness, and bend angle.

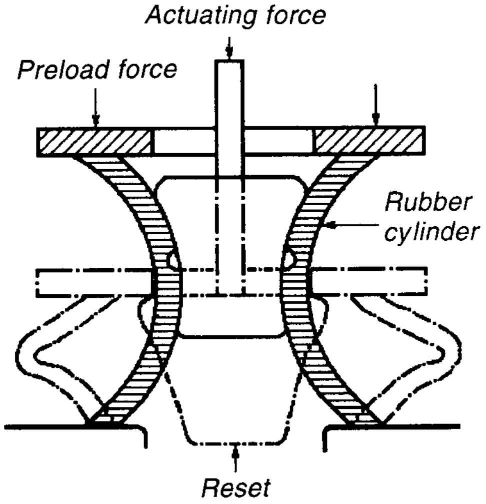


Fig. 2 A collapsing cylinder has elastic walls that can be deformed gradually until their stress changes from compressive to bending, with the resulting collapse of the cylinder.

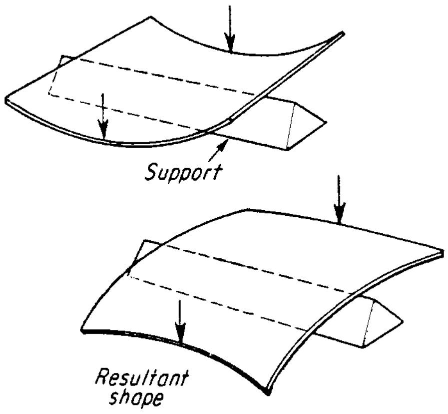


Fig. 3 A bowed spring will collapse into a new shape when it is loaded as shown A. A “push-pull” steel measuring tape illustrates this action; the curved material stiffens the tape so that it can be held out as a cantilever until excessive weight causes it to collapse suddenly.

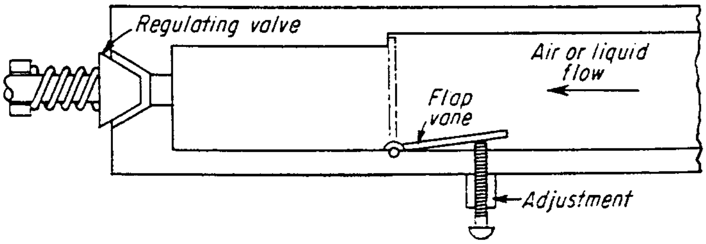


Fig. 4 A flap vane cuts off air or liquid flow at a limiting velocity. With a regulating valve, the vane will snap shut (because of increased velocity) when pressure is reduced below a design value.

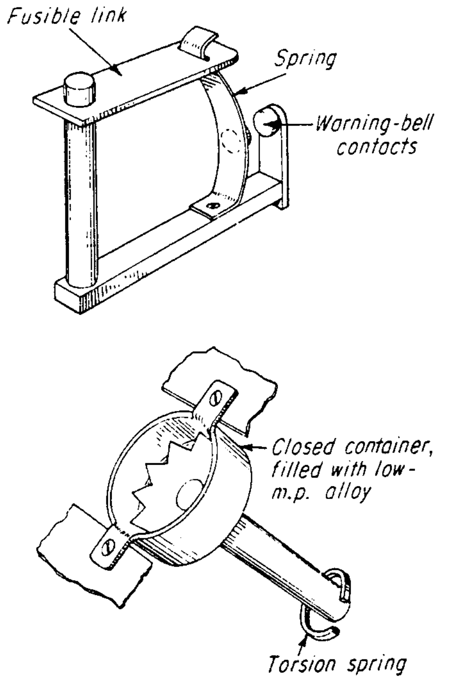


Fig. 5 A sacrificing link is useful where high temperature or corrosive chemicals would be hazardous. If the temperature becomes too high, or atmosphere too corrosive, the link will yield at design conditions. The device usually is required to act only once, although a device like the lower one can be quickly reset. However, it is restricted to temperature control.

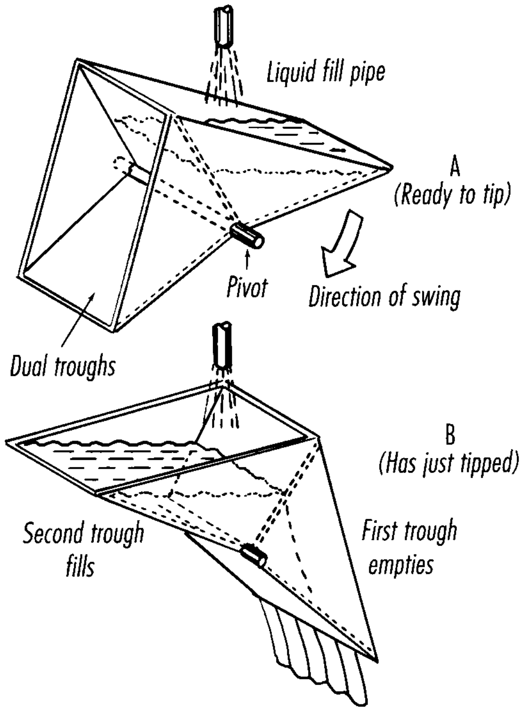


Fig. 6 Gravity-tips, although slower acting than most snap mechanisms, can be called snap mechanisms because they require an accumulation of energy to trigger an automatic release. A tripping trough that spreads sewerage is one example. As shown in A, it is ready to trip. When overbalanced, it trips rapidly, as in B.

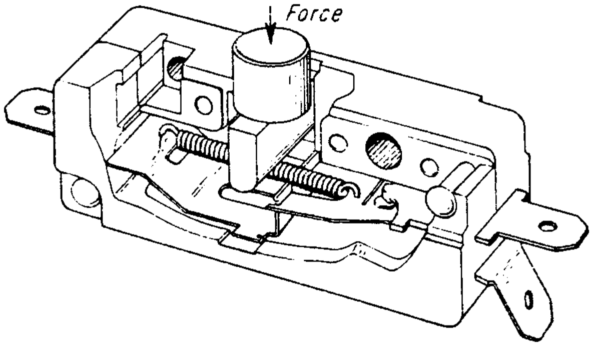


Fig. 7 An overcentering tension spring combined with a pivoted contact-strip is one arrangement used in switches. The example shown here is unusual because the actuating force bears on the spring itself.

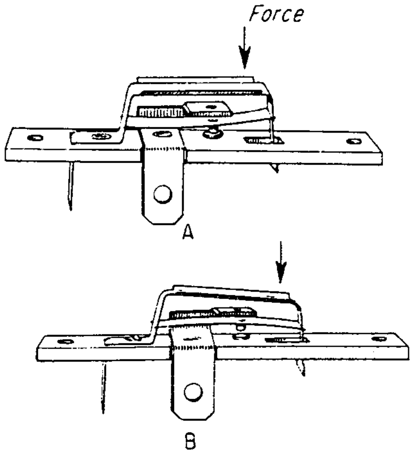


Fig. 8 An overcentering leaf-spring action is also the basis for many ingenious snap-action switches for electrical control. Sometimes spring action is combined with the thermostatic action of a bimetal strip to make the switch respond to heat or cold, either for control purposes or as a safety feature.

APPLICATIONS OF THE DIFFERENTIAL WINCH TO CONTROL SYSTEMS

Known for its mechanical advantage, the differential winch is a control mechanism that can supplement the gear and rack and four-bar linkage systems in changing rotary motion into linear. It can magnify displacement to meet the needs of delicate instruments or be varied almost at will to fulfill uncommon equations of motion.

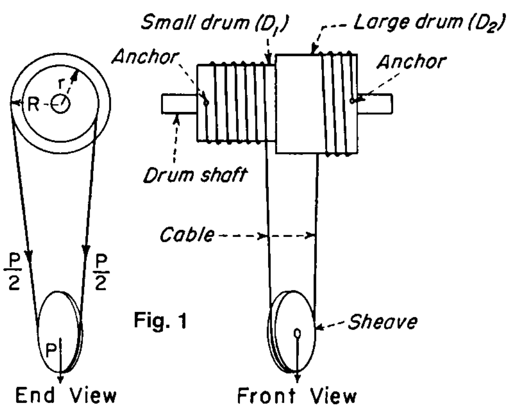


Fig. 1

Fig. 1 A standard differential winch consists of two drums, D_1 and D_2, and a cable or chain which is anchored on both ends and wound clockwise around one drum and counterclockwise around the other. The cable supports a load-carrying sheave, and if the shaft is rotated clockwise, the cable, which unwinds from D_1 on to D_2, will raise the sheave a distance

$$\text{Sheave rise/rev} = \frac{2\pi R - 2\pi r}{2} = \pi(R - r)$$

The winch, which is not in equilibrium exerts a counterclockwise torque.

$$\text{Unbalanced torque} = \frac{P}{2}(R - r)$$

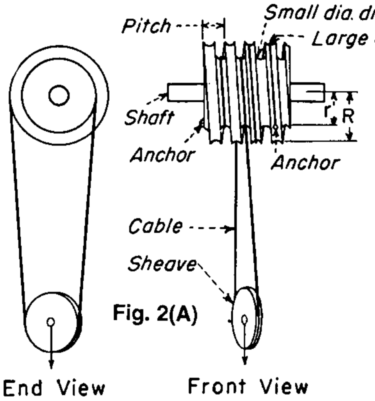


Fig. 2(A)

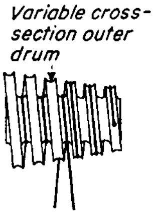


Fig. 2(B)

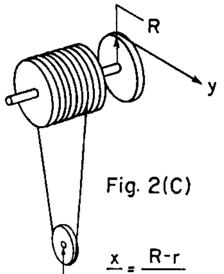


Fig. 2(C)

$$\frac{x}{y} = \frac{R-r}{2R}$$

R	r	x/y
3	$2\frac{15}{16}$	1/96
1	$\frac{63}{64}$	1/128

Fig. 2(A) Hulse Differential Winch*. Two drums, which are in the form of worm threads contoured to guide the cables, concentrically occupy the same logitudinal space. This keeps the cables approximately at right angles to the shaft and eliminates cable shifting and rubbing, especially when used with variable cross sections as in Fig. 2(B). Any equation of motion can be satisfied by choosing suitable cross sections for the drums. Methods for resisting or supporting the axial thrust should be considered in some installations. Fig. 2(C) shows typical reductions in displacement. *Pat. No. 2,590,623

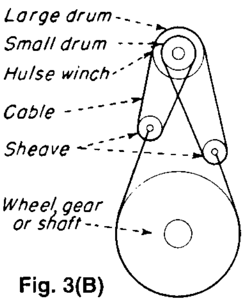


Fig. 3(B)

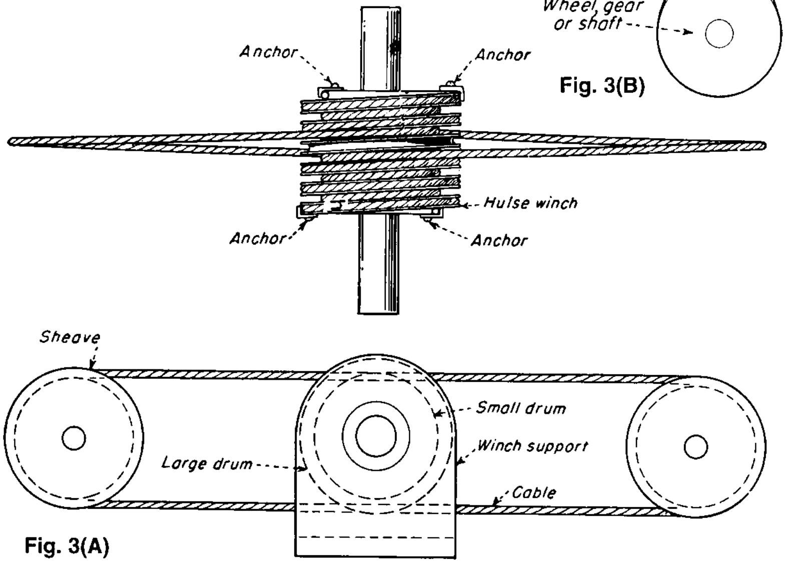


Fig. 3(A)

Fig. 3(A) A Hulse Winch with opposing sheaves. This arrangement, which uses two separate cables and four anchor points, can be considered as two winches back-to-back with one common set of drums. Variations in motion can be obtained by: (1) restraining in the sheaves so that when the system is rotated the drums will travel toward one of the sheaves; (2) restraining the drums and allowing the sheaves to travel. The distance between the sheaves will remain constant and is usually connected by a bar; (3) permitting the drums to move axially while restraining them transversely. When the system is rotated, drums will travel axially one pitch per revolution, and sheaves remain in the same plane perpendicular to the drum axis. This variation can be reversed by allowing sheaves to move axially; and (4) sheaves need not be opposite but can be arranged as in Fig. 3(B) to rotate a wheel.

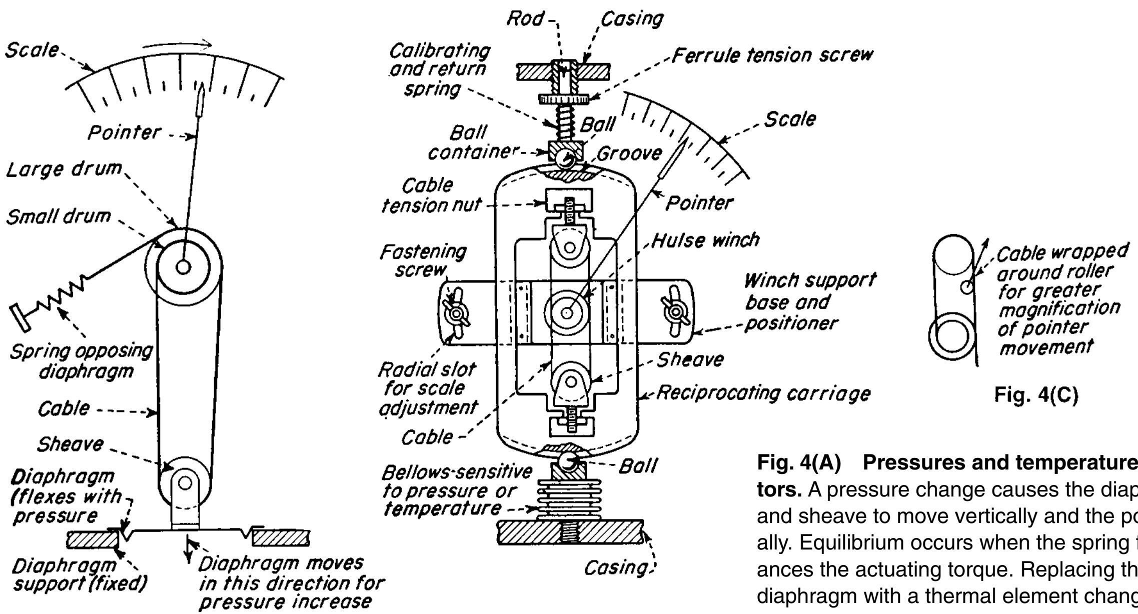


Fig. 4(A) **Fig. 4(B)** **Fig. 4(C)**

Fig. 4(A) Pressures and temperature indicators. A pressure change causes the diaphragm and sheave to move vertically and the pointer radially. Equilibrium occurs when the spring force balances the actuating torque. Replacing the diaphragm with a thermal element changes the instrument into a temperature indicator. Two sheaves and a reciprocating carriage, Fig. 4(B), are based on the principle shown in Fig. 3(A). A carriage is activated by pressure or temperature and is balanced by a spring force in the opposite end. Further magnification can be obtained, Fig. 4(C), by wrapping a cable around the roller to which the pointer is attached.

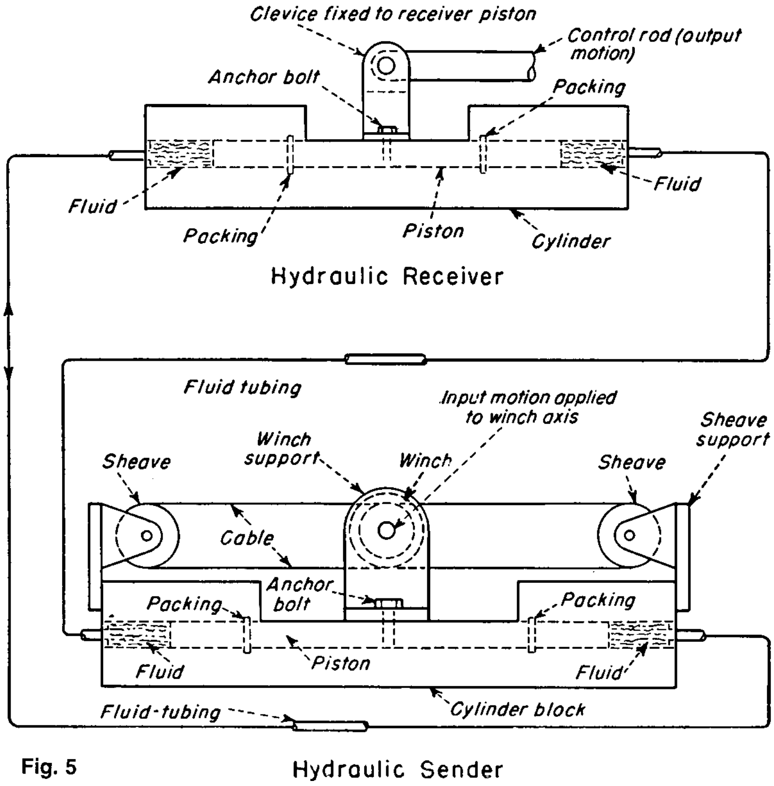


Fig. 5

Fig. 5 A hydraulic control system, actuated by a differential winch, performs remote precision positioning of a control rod with a minimum of applied torque. The sending piston, retained in a cylinder block, reciprocates back and forth from a torque applied to the winch shaft. Fluid is forced out from one end of the cylinder through the pipe lines to displace the receiving piston, which in turn activates a control rod. The receiver simultaneously displaces a similar amount of fluid from the opposite end back to the sender. By suitable valving, the sender can become a double-acting pump.

SIX APPLICATIONS FOR MECHANICAL POWER AMPLIFIERS

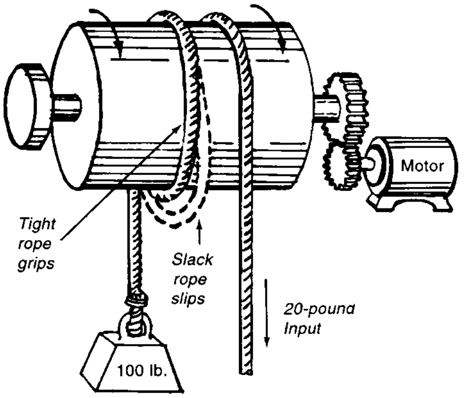
Precise positioning and movement of heavy loads are two basic jobs for this all-mechanical torque booster.

This mechanical power amplifier has a fast response. Power from its continuously rotating drums is instantaneously available. When used for position-control applications, pneumatic, hydraulic, and electrical systems—even those with continuously running power sources—require transducers to change signals from one energy form to another. The mechanical power amplifier, on the other hand, permits direct sensing of the controlled motion.

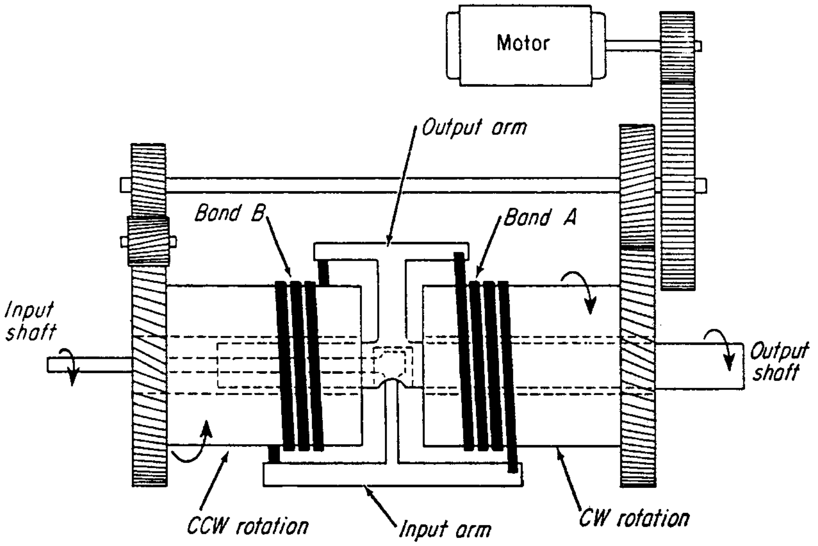
Four major advantages of this all-mechanical device are:

1. Kinetic energy of the power source is continuously available for rapid response.
2. Motion can be duplicated and power amplified without converting energy forms.
3. Position and rate feedback are inherent design characteristics.
4. Zero slip between input and output eliminates the possibility of cumulative error.

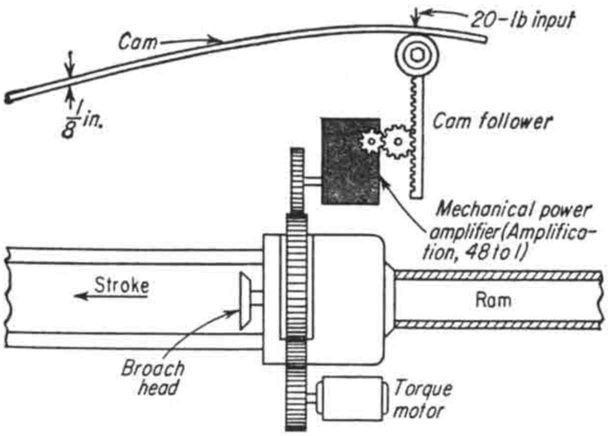
One other important advantage is the ease with which this device can be adapted to perform special functions—jobs for which other types of systems would require the addition of more costly and perhaps less reliable components. The six applications which follow illustrate how those advantages have been put to work in solving widely divergent problems.



The capstan principle is the basis for the mechanical power amplifier described here that combines two counterrotating drums. The drums are continuously rotating but only transmit torque when the input shaft is rotated to tighten the band on drum A. Overrun of output is stopped by drum B, when overrun tightens the band on this drum.



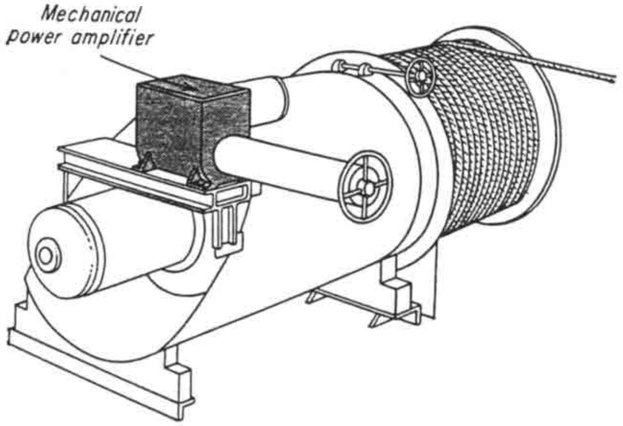
A capstan is a simple mechanical amplifier—rope wound on a motor-driven drum slips until slack is taken up on the free end. The force needed on the free end to lift the load depends on the coefficient of friction and the number of turns of rope. By connecting bands A and B to an input shaft and arm, the power amplifier provides an output in both directions, plus accurate angular positioning. When the input shaft is turned clockwise, the input arm takes up the slack on band A, locking it to its drum. Because the load end of locked band A is connected to the output arm, it transmits the CW motion of the driven drum on which it is wound to the output shaft. Band B therefore slacks off and slips on its drum. When the CW motion of the input shaft stops, tension on band A is released and it slips on its own drum. If the output shaft tries to overrun, the output arm will apply tension to band B, causing it to tighten on the CCW rotating rum and stop the shaft.



1. Nonlinear Broaching

Problem: In broaching large-fore rifles, the twist given to the lands and grooves represents a nonlinear function of barrel length. Development work on such rifles usually requires some experimentation with this function. At present, rotation of the broaching head is performed by a purely mechanical arrangement consisting of a long, heavy wedge-type cam and appropriate gearing. For steep twist angles, however, the forces acting on this mechanism become extremely high.

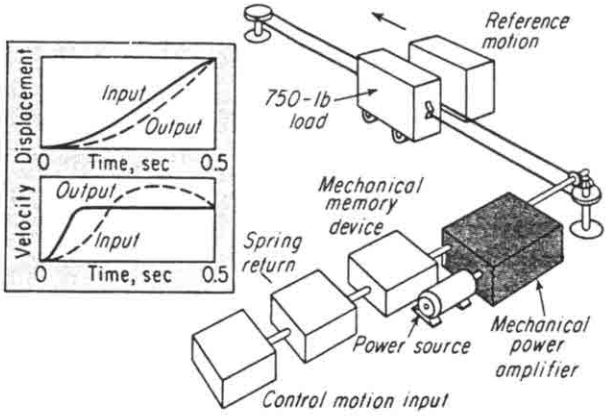
Solution: A suitable mechanical power amplifier, with its inherent position feedback, was added to the existing mechanical arrangement, as shown in Fig. 1. The cam and follower, instead of having to drive the broaching head, simply furnish enough torque to position the input shaft of the amplifier.



2. Hydraulic Winch Control

Problem: Hydraulic pump-motor systems are excellent for controlling position and motion at high power levels. In the 10- to 150-hp range, for example, the usual approach is to vary the output of a positive displacement pump in a closed-loop hydraulic circuit. In many of the systems that might be able to control this displacement, however, a force feedback proportional to system pressure can lead to serious errors or even oscillations.

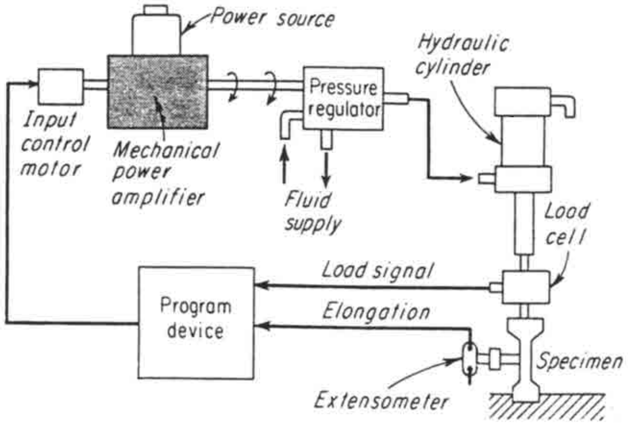
Solution: Figure 2 shows an external view of the complete package. The output shaft of the mechanical power amplifier controls pump displacement, while its input is controlled by hand. In a more recent development requiring remote manual control, a servomotor replaces this local handwheel. Approximately 10 lb-in. torque drives a 600 lb-in. load. If this system had to transmit 600 lb-in., the equipment would be more expensive and more dangerous to operate.



3. Load Positioning

Problem: It was necessary for a 750-lb load to be accelerated from standstill in 0.5 s and brought into speed and position synchronization with a reference linear motion. It was also necessary that the source of control motion be permitted to accelerate more rapidly than the load itself. Torque applied to the load could not be limited by any kind of slipping device.

Solution: A system with a single mechanical power amplifier provided the solution (Fig. 3). A mechanical memory device, preloaded for either rotation, drives the input shaft of the amplifier. This permits the input source to accelerate as rapidly as desired. The total control input travel minus the input travel of the amplifier shaft is temporarily stored. After 0.5 seconds, the load reaches proper speed, and the memory device transmits position information in exact synchronization with the input.



4. Tensile Testing Machine

Problem: On a hydraulic tensile testing machine, the stroke of the power cylinder had to be controlled as a function of two variables: tension in, and extension of, the test specimen. A programming device, designed to provide a control signal proportional to these variables, had an output power level of about 0.001 hp—too low to drive the pressure regulator controlling the flow to the cylinder.

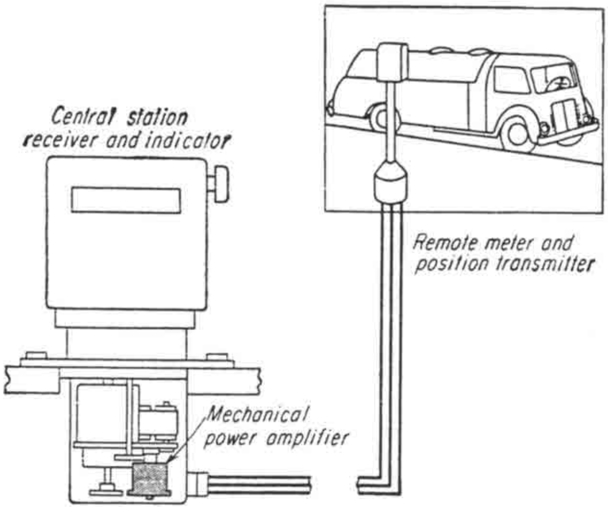
Solution: An analysis of the problem revealed three requirements: the output of the programmer had to be amplified about 60 times, position accuracy had to be within 2º, and acceleration had to be held at a very low value. A mechanical power amplifier satisfied all three requirements. Figure 4 illustrates the completed system. Its design is based principally on steady-state characteristics.

5. Remote Metering and Counting

Problem: For a remote, liquid-metering job, synchro systems had been used to transmit remote meter readings to a central station and repeat this information on local indicating counters. The operation involved a large number of meters and indicators. As new equipment (e.g. ticket printers) was added, the torque requirement also grew.

Solution: Mechanical power amplifiers in the central station indicators not only supplied extra output torque but also made it possible to specify synchros that were even smaller than those originally selected to drive the indicators alone (see Fig. 5).

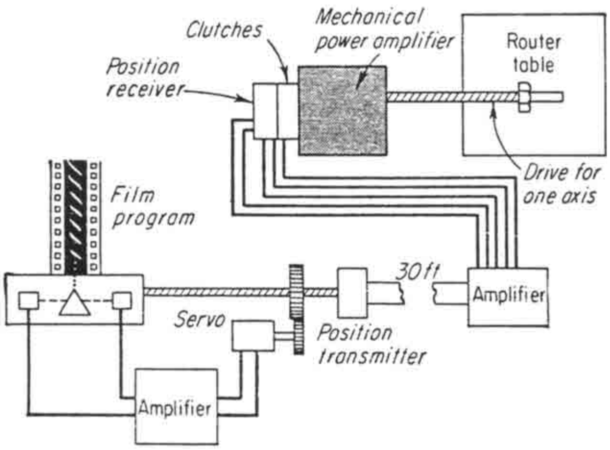
The synchro transmitters selected operate at a maximum speed of 600 rpm and produce only about 3 oz-in. of torque. The mechanical power amplifiers furnish up to 100 lb-in. of torque, and are designed to fit in the bottom of the registers shown in Fig. 5. Total accuracy is within 0.25 gallon, and error is noncumulative.



6. Irregular Routing

Problem: To control remotely the table position of a routing machine from information stored on a film strip. The servoloop developed to interpret this information produced only about 1 oz.-in. of torque. About 20 lb.-ft was required at the table feedscrew.

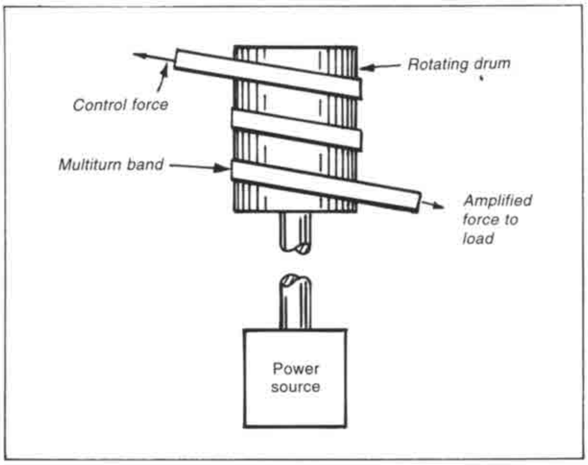
Solution: Figure 6 shows how a mechanical power amplifier supplied the necessary torque at the remote table location. A position transmitter converts the rotary motion output of the servoloop to a proportional electrical signal and sends it to a differential amplifier at the machine location. A position receiver, geared to the output shaft, provides a signal proportional to table position. The differential amplifier compares these, amplifies the difference, and sends a signal t either counterrotating electromagnetic clutch, which drives the input shaft of the mechanical power amplifier.

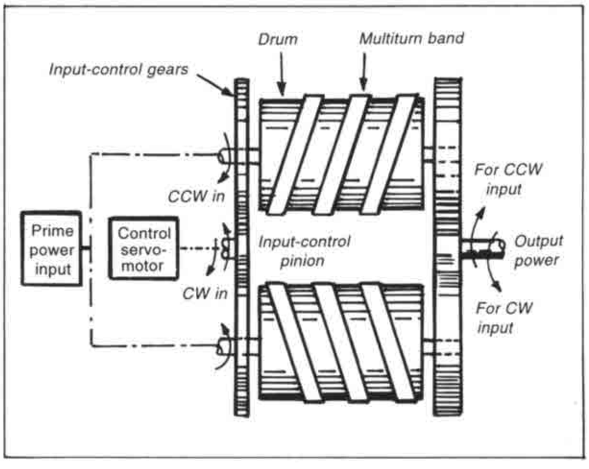


A mechanical power amplifier that drives a crossfeed slide is based on the principle of the windlass. By varying the control force, all or any part of power to the drum can be used.

Two drums mounted back to back supply the bi-directional power needed in servo systems. Replacing the operator with a two-phase induction servomotor permits electronic or magnetic signal amplification. A rotating input avoids a linear input and output of the simple windlass. Control and output ends of the multiturn bands are both connected to gears mounted concentrically with the drum axis.

When the servomotor rotates the control gear, it locks the band-drum combination, forcing output gear to rotate with it. Clockwise rotation of the servomotor produces CW power output while the second drum idles. Varying the servo speed, by changing servo voltage, varies output speed.





VARIABLE-SPEED BELT AND CHAIN DRIVES

Variable-speed drives provide an infinite number of speed ratios within a specific range. They differ from the stepped-pulley drives in that the stepped drives offer only a discrete number of velocity ratios.

Mechanical "all-metal" drives employ friction or preloaded cones, disks, rings, and spheres, which undergo a certain amount of slippage. Belt drives, on the other hand, have little slippage or frictional losses, and chain has none—it maintains a fixed phase relationship between the input and output shafts.

Belt Drives

Belt drives offer high efficiency and are relatively low in price. Most use V-belts, reinforced by steel wires to 3 inches in width.

Speed adjustment in belt drives is obtained through one of the four basic arrangements shown below.

Variable-distance system (Fig. 1). A variable-pitch sheave on the input shaft opposes a solid (fixed-pitch) sheave on the output shaft. To vary the speed, the center distance is varied, usually by an adjustable base, tilting or sliding motor (Fig. 6).

Speed variations up to 4:1 are easily achieved, but torque and horsepower characteristics depend on the location of the variable-diameter sheave.

Fixed-distance system (Fig. 2). Variable-pitch sheaves on both input and output shafts maintain a constant center distance between shafts. The sheaves are controlled by linkage. Either the pitch diameter of one sheave is positively controlled and the disks of the other sheave under spring tension, adjust automatically or the pitch diameters of both sheaves are positively controlled by the linkage system (Fig. 5). Pratt & Whittney has applied the system in Fig. 5 to the spindle drive of numerically controlled machines.

Speed variations up to 11:1 are obtained, which means that with a 1200-rpm motor, the maximum output speed will be $1200\sqrt{11} = 3984$ rpm, and the minimum output speed = 3984/11 = 362 rpm.

Double-reduction system (Fig. 3). Solid sheaves are on both the input and output shafts, but both sheaves on the intermediate shaft are of variable-pitch type. The center distance between input and output is constant.

Coaxial shaft system (Fig. 4). The intermediate shaft in this arrangement permits the output shaft to be coaxial with that of the input shaft. To maintain a fixed center distance, all four sheaves must be of the variable-pitch type and controlled by linkage, similar to the system in Fig. 6. Speed variation up to 16:1 is available.

Packaged belt units (Fig. 7). These combine the motor and variable-pitch transmissions as an integral unit. The belts are usually ribbed, and speed ratios can be dialed by a handle.

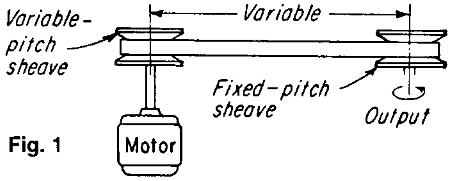


Fig. 1

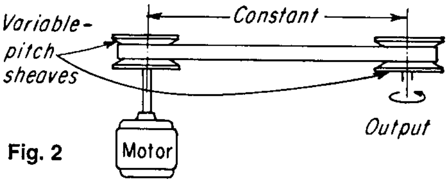


Fig. 2

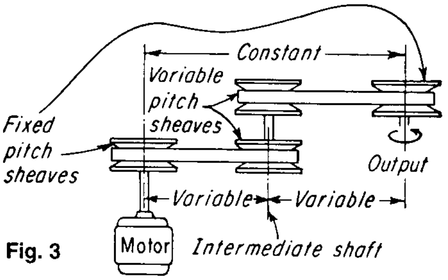


Fig. 3

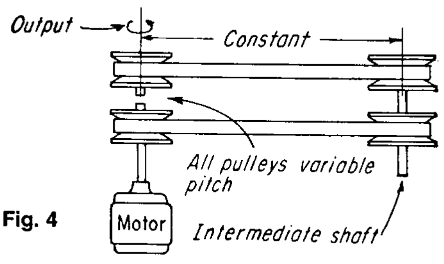


Fig. 4

Four basic belt arrangements for varying output speed.

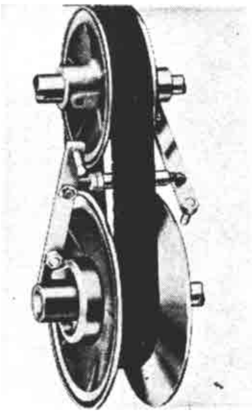


Fig. 5 Linkage controlled pulleys.

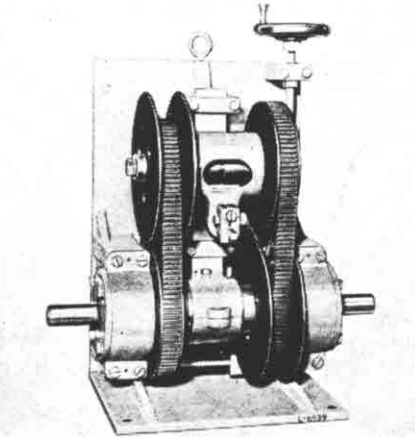
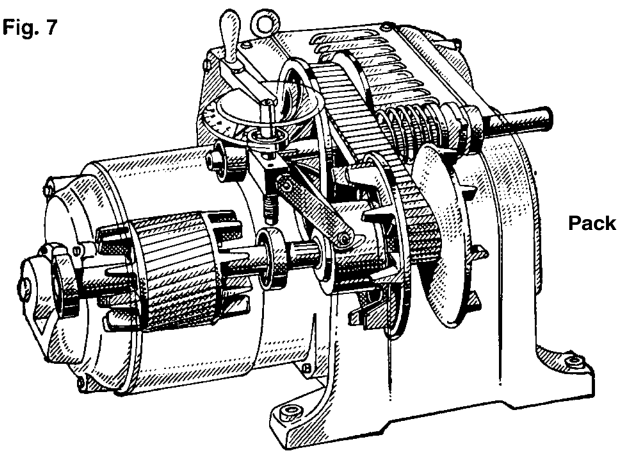


Fig. 6 Tandem arrangement employs dual belt-system to produce high speed-reduction.

Fig. 7



Packaged belt unit.

Sheave Drives

The axial shifting of variable-pitch sheaves is controlled by one of four methods:

Linkage actuation. The sheave assemblies in Fig. 5 are directly controlled by linkages which, in turn, are manually adjusted.

Spring pressure. The cons of the sheaves in Figs. 2 and 4 are axially loaded by spring force. A typical pulley of this type is illustrated in Fig. 8. These pulleys are used in conjunction with directly controlled sheaves, or with variable center-distance arrangements.

Cam-controlled sheave. The cones of this sheave (Fig. 9) are mounted on a floating sheave, free to rotate on the pulley spindle. Belt force rotates the cones, whose surfaces are cammed by the inclined plane of the spring. The camming action wedges the cones against the belt, thus providing sufficient pressure to prevent slippage at the higher speeds, as shown in the curve.

Centrifugal-force actuator. In this unique sheave arrangement (Fig. 10) the pitch diameter of the driving sheave is controlled by the centrifugal force of steel balls. Another variable-pitch pulley mounted on the driven shaft is responsive to the torque. As the drive speed increases, the centrifugal force of the balls forces the sides of the driving sheave together. With a change in load, the movable flange of the driven sheave rotates in relation to the fixed flange. The differential rotation of the sheave flanges cams them together and forces the V-belt to the outer edge of the driven sheave, which has a lower transmission ratio. The driving sheave is also shifted as the load rises with decreasing speed. With a stall load, it is moved to the idling position. When the torque-responsive sheave is the driving member, any increase in drive speed closes its flanges and opens the flanges of the centrifugal member, thus maintaining a constant output speed. The drive has performed well in transmissions with ratings ranging from 2 to 12 hp.

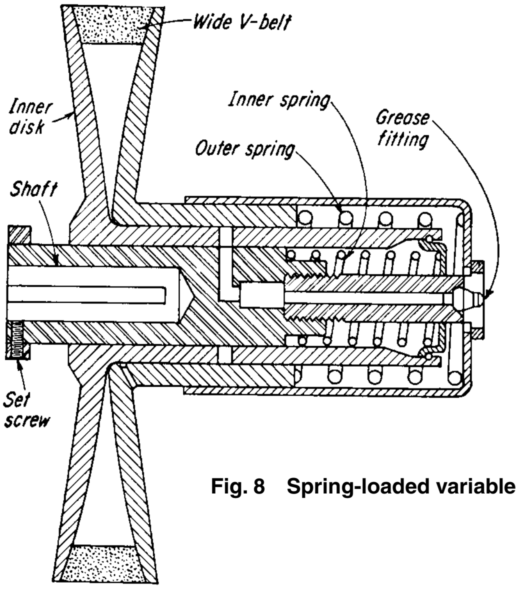


Fig. 8 Spring-loaded variable-pitch pulley.

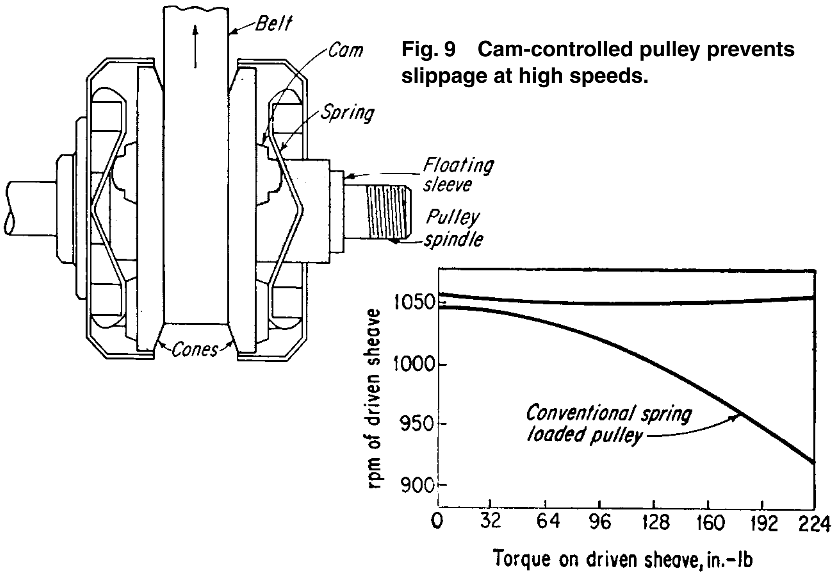
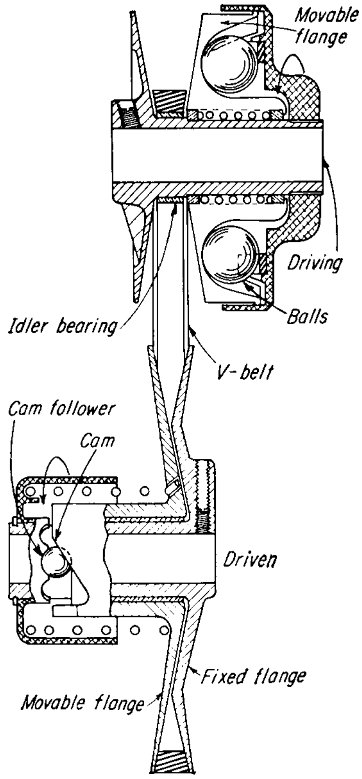


Fig. 9 Cam-controlled pulley prevents slippage at high speeds.

Fig. 10 **A ball-controlled pulley** has its sides pressured by centrifugal force.



Chain Drives

PIV drive (Fig. 11). This chain drive (positive, infinitely variable) eliminates any slippage between the self-forming laminated chain teeth and the chain sheaves. The individual laminations are free to slide laterally to take up the full width of the sheave. The chain runs in radially grooved faces of conical surface sheaves which are located on the input and output shafts. The faces are not straight cones, but have a slight convex curve to maintain proper chain tension at all positions. The pitch diameters of both sheaves are positively controlled by the linkage. Booth action is positive throughout operating range. It is rated to 25 hp with speed variation of 6:1.

Double-roller chain drive (Fig. 12). This specially developed chain is built for capacities to 22 hp. The hardened rollers are wedged between the hardened conical sides of the variable-pitch sheaves. Radial rolling friction results in smooth chain engagement.

Single-roller chain drive (Fig. 13). The double strand of this chain boosts the capacity to 50 hp. The scissor-lever control system maintains the proper proportion of forces at each pair of sheave faces throughout the range.

Fig. 11 A PIV drive chain grips radially grooved faces of a variable-pitch sheave to prevent slippage.

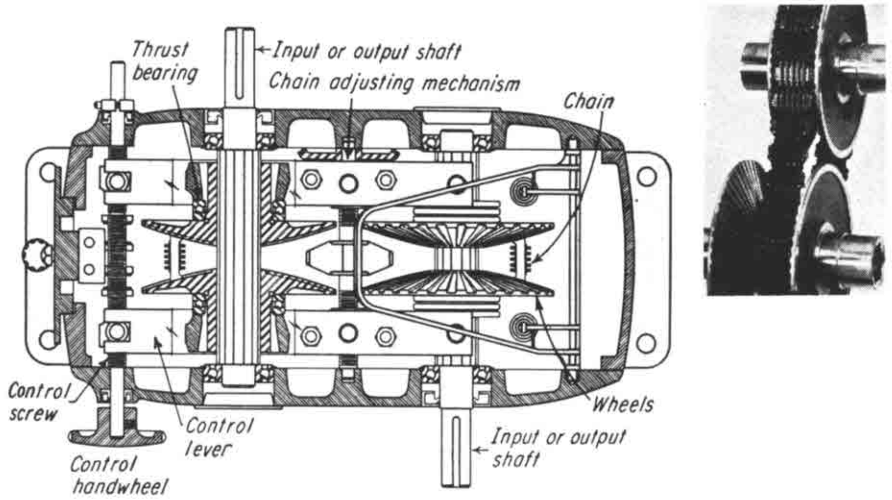
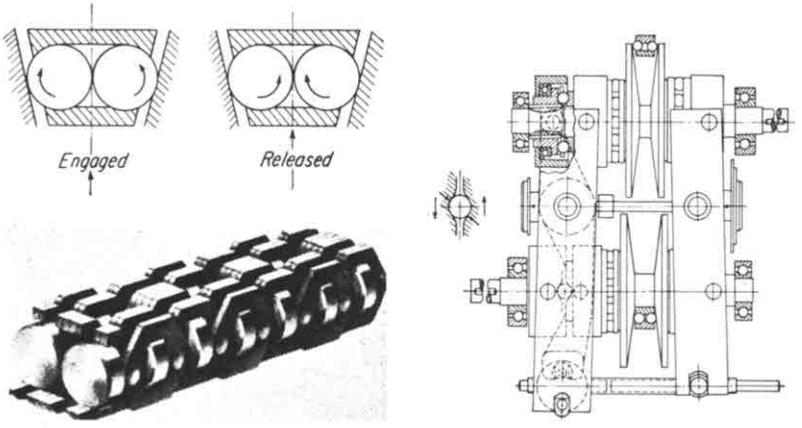


Fig. 12 A double-roller chain drive combines strength with ease in changing speed.



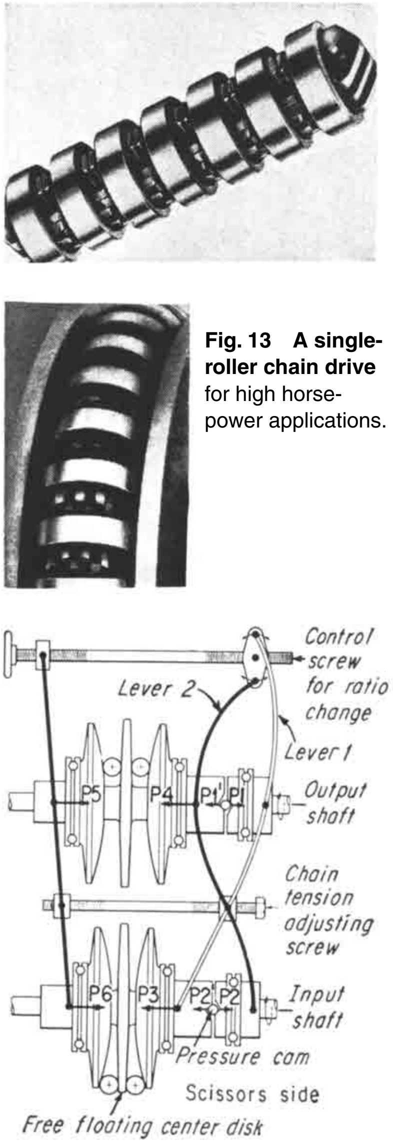


Fig. 13 A single-roller chain drive for high horse-power applications.

GETTING IN STEP WITH HYBRID BELTS

Imaginative fusions of belts, cables, gears and chains are expanding the horizons for light-duty synchronous drives

Belts have long been used for the transfer of mechanical power. Today's familiar flat belts and V-belts are relatively light, quiet, inexpensive, and tolerant of alignment errors. They transmit power solely through frictional contacts. However, they function best at moderate speeds (4000 to 6000 fpm) under static loads. Their efficiencies drop slightly at low speeds, and centrifugal effects limit their capacities at high speeds. Moreover, they are inclined to sip under shock loads or when starting and braking. Even under constant rotation, standard belts tend to creep. Thus, these drives must be kept under tension to function properly, increasing loads on pulley shaft bearings.

Gears and chains, on the other hand, transmit power through bearing forces between positively engaged surfaces. They do not slip or creep, as measured by the relative motions of the driving and driven shafts. But the contacts themselves can slip significantly as the chain rollers and gear teeth move in and out of mesh.

Positive drives are also very sensitive to the geometries of the mating surfaces. A gear's load is borne by one or two teeth, thus magnifying small tooth-to-tooth errors. A chain's load is more widely distributed, but chordal variations in the driving wheel's effective radius produce small oscillations in the chain's velocity.

To withstand these stresses, chains and gears must be carefully made from hard materials and must then be lubri-

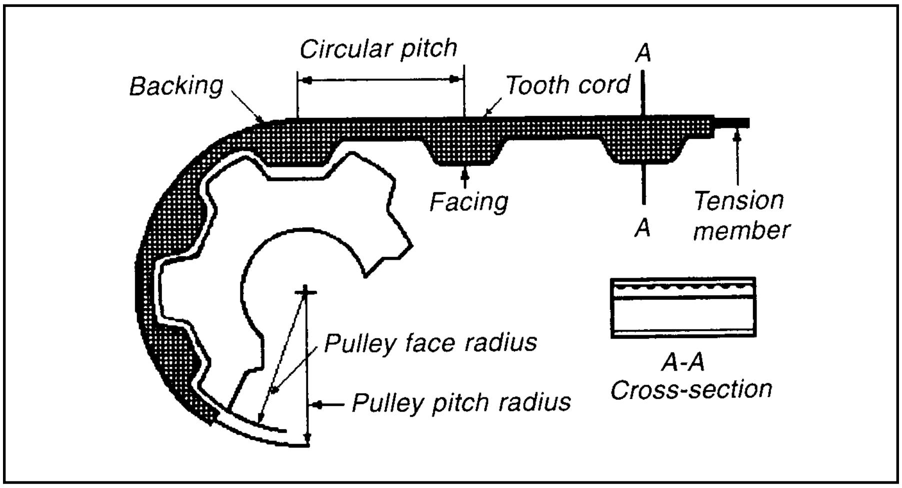


Fig. 1 Conventional timing belts have fiberglass or polyester tension members, bodies of neoprene or polyurethane, and trapezoidal tooth profiles.

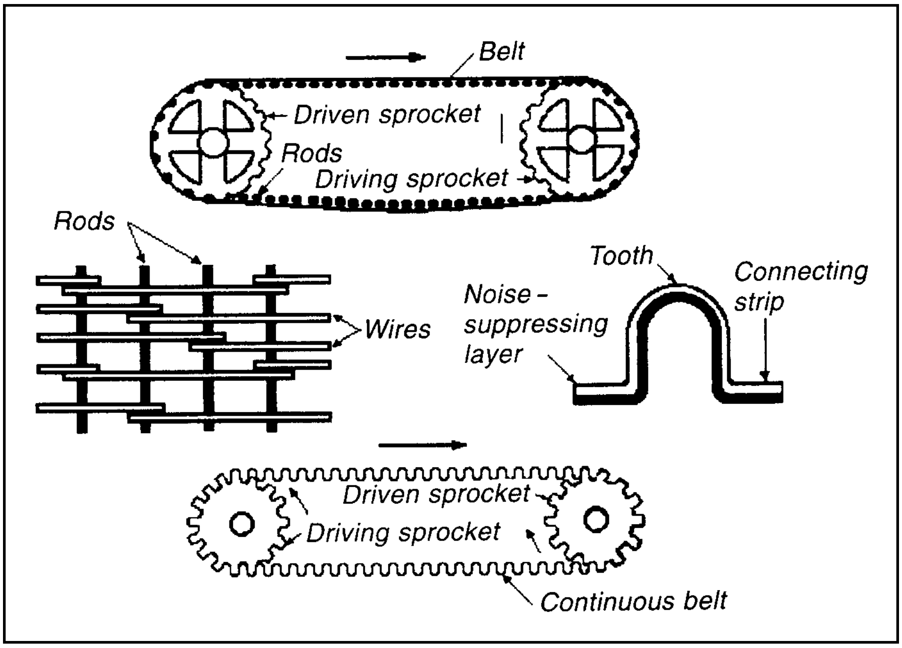


Fig. 2 NASA metal timing belts exploit stainless steel's strength and flexibility, and are coated with sound-and friction-reducing plastic.

cated in operations. Nevertheless, their operating noise betrays sharp impacts and friction between mating surfaces.

The cogged timing belt, with its trapezoidal teeth (Fig. 1), is the best-known fusion of belt, gear, and chain. Though these well-established timing belts can handle high powers (up to 800 hp), many of the newer ideas in synchronous belting have been incorporated into low and fractional horsepower drives for instruments and business machines.

Steel Belts for Reliability

Researchers at NASA's Goddard Space Flight Center (Greenbelt, MD) turned to steel in the construction of long-lived toothed transmission belts for spacecraft instrument drives.

The NASA engineers looked for a belt design that would retain its strength and hold together for long periods of sustained or intermittent operation in hostile environments, including extremes of heat and cold.

Two steel designs emerged. In the more chain-like version (Fig. 2A), wires running along the length of the belt are wrapped at intervals around heavier rods running across the belt. The rods do double duty, serving as link pins and as teeth that mesh with cylindrical recesses cut into the sprocket. The assembled belt is coated with plastic to reduce noise and wear.

In the second design (Fig. 2B), a strip of steel is bent into a series of U-shaped teeth. The steel is supple enough to flex as it runs around the sprocket with its protruding transverse ridges, but the material resists stretching. This belt, too, is plastic-coated to reduce wear and noise.

The V-belt is best formed from a continuous strip of stainless steel "not much thicker than a razor blade," according to the agency, but a variation can be made by welding several segments together.

NASA has patented both belts, which are now available for commercial licensing. Researchers predict that they will be particularly useful in machines that must be dismantled to uncover the belt pulleys, in permanently encased machines, and in machines installed in remote places. In addition, stainless-steel belts might find a place in high-precision instrument drives because they neither stretch nor slip.

Though plastic-and-cable belts don't have the strength or durability of the NASA steel belts, they do offer versatility and production-line economy. One of the least expensive and most adaptable is

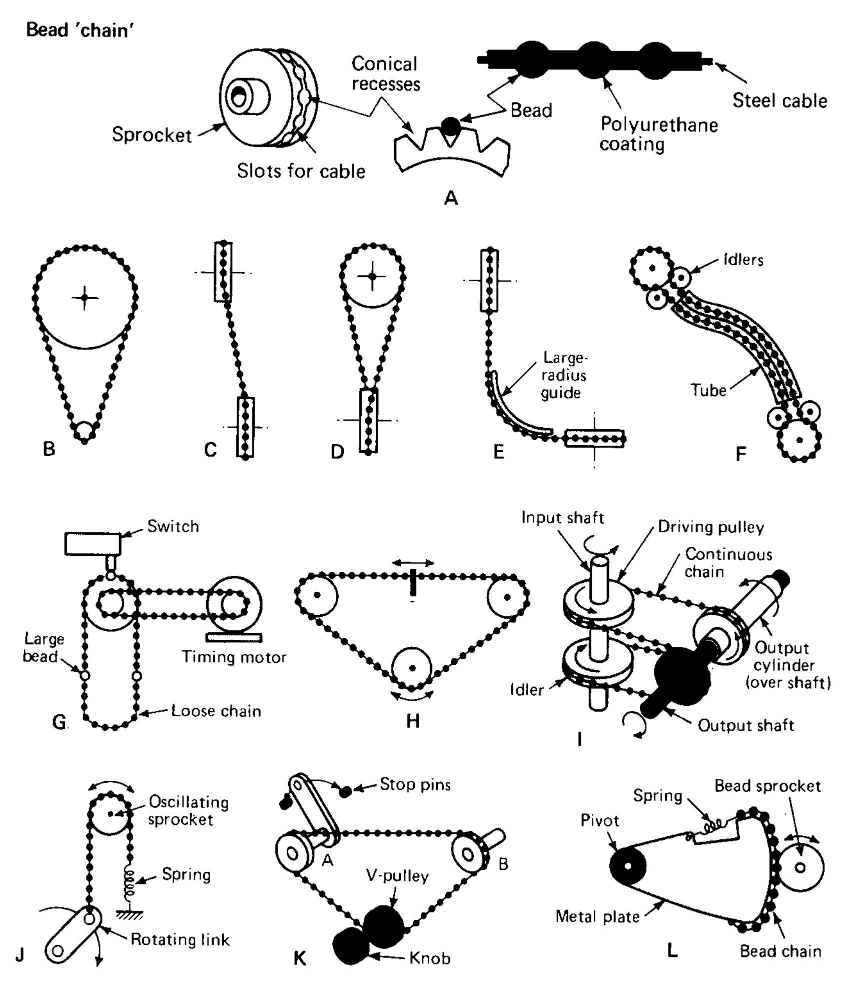


Fig. 3 Polyurethane-coated steel-cable "chains"—both beaded and 4-pinned—can cope with conditions unsuitable for most conventional belts and chains.

Table 1. Conventional Timing Belts

Type	Circular pitch, in.	Wkg. tension lb/in. width	Centr. loss const., K_c
Standard (Fig 1)			
MXL	0.080	32	10×10^{-9}
XL	0.200	41	27×10^{-9}
L	0.375	55	38×10^{-9}
H	0.500	140	53×10^{-9}
40DP	0.0816	13	—
High-torque (Fig 8)			
3 mm	0.1181	60	15×10^{-9}
5 mm	0.1968	100	21×10^{-9}
8 mm	0.3150	138	34×10^{-9}

Courtesy Stock Drive Products

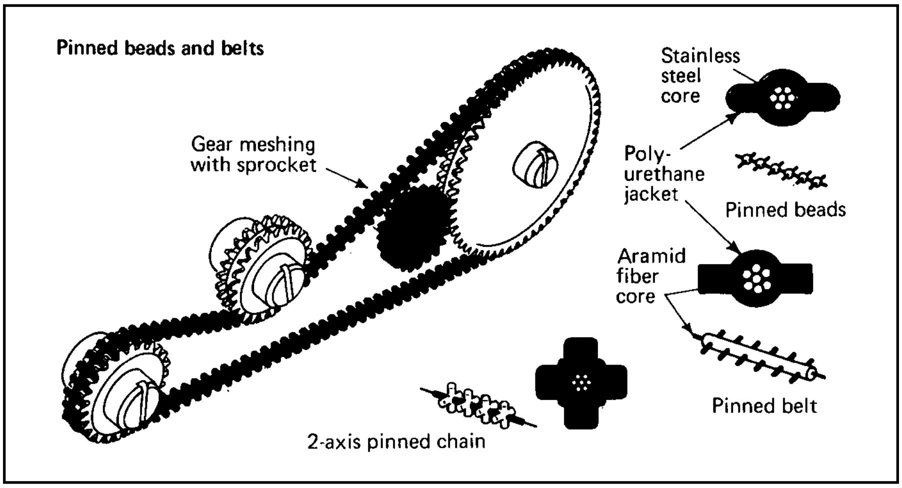


Fig. 4 Plastic pins eliminate the bead chain's tendency to cam out of pulley recesses, and permit greater precision in angular transmission.

the modern version of the bead chain, now common only in key chains and light-switch pull-cords.

The modern bead chain—if chain is the proper word—has no links. It has, instead, a continuous cable of stainless steel or aramid fiber which is covered with polyurethane. At controlled intervals, the plastic coating is molded into a bead (Fig. 3A). The length of the pitches thus formed can be controlled to within 0.001 in.

In operation, the cable runs in a grooved pulley; the beads seat in conical recesses in the pulley face. The flexibility, axial symmetry, and positive drive of bead chain suit a number of applications, both common and uncommon:

- An inexpensive, high-ratio drive that resists slipping and requires no lubrication (Fig. 3B). As with other chains and belts, the bead chain's capacity is limited by its total tensile strength (typically 40 to 80 lb for a single-strand steel-cable chain), by the speed-change ratio, and by the radii of the sprockets or pulleys.
- Connecting misaligned sprockets. If there is play in the sprockets, or if the sprockets are parallel but lie in different planes, the bead chain can compensate for up to 20° of misalignment (Fig. 3C).
- Skewed shafts, up to 90° out of phase (Fig. 3D).
- Right-angle and remote drives using guides or tubes (Figs. 3E and 3F). These methods are suitable only for low-speed, low-torque applications. Otherwise, frictional losses between the guide and the chain are unacceptable.
- Mechanical timing, using oversize beads at intervals to trip a microswitch (Fig. 3G). The chain can be altered or exchanged to give different timing schemes.
- Accurate rotary-to-linear motion conversion (Fig. 3H).
- Driving two counter-rotating outputs from a single input, using just a single belt (Fig. 3I).
- Rotary-to-oscillatory motion conversion (Fig. 3J).
- Clutched adjustment (Fig. 3K). A regular V-belt pulley without recesses permits the chain to slip when it reaches a pre-set limit. At the same time, bead-pulleys keep the output shafts synchronized. Similarly, a pulley or sprocket with shallow recesses permits the chain to slip one bead at a time when overloaded.
- Inexpensive "gears" or gear segments fashioned by wrapping a bead chain round the perimeter of a disk or solid arc of sheet metal (Fig. 3L). The sprocket then acts as a pinion. (Other designs are better for gear fabrication.)

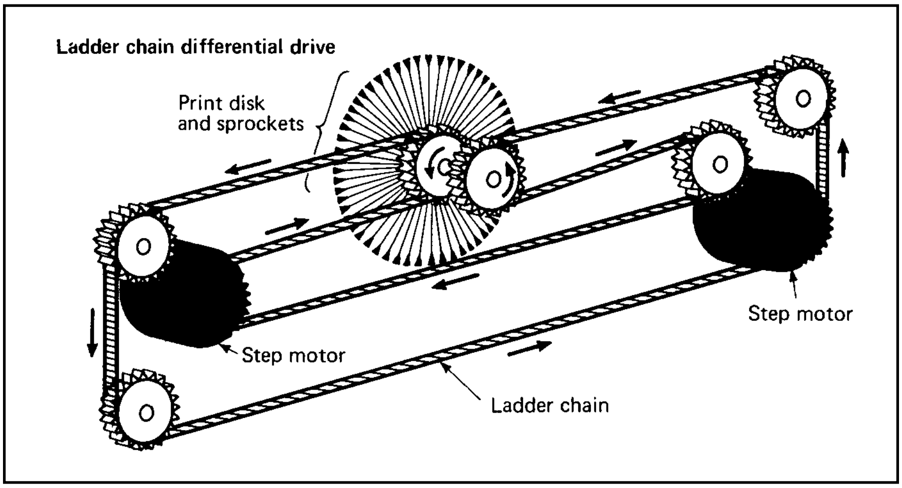


Fig. 5 A plastic-and-cable ladder chain in an impact-printer drive. In extreme conditions, such hybrids can serve many times longer than steel.

A More Stable Approach

Unfortunately, bead chains tend to cam out of deep sprocket recesses under high loads. In its first evolutionary step, the simple spherical bead grew limbs—two pins projecting at right angles to the cable axis (Fig. 4). The pulley or sprocket looks like a spur gear grooved to accommodate the belt; in fact, the pulley can mesh with a conventional spur gear of proper pitch.

Versions of the belt are also available with two sets of pins, one projecting vertically and the other horizontally. This arrangement permits the device to drive a series of perpendicular shafts without twisting the cable, like a bead chain but without the bead chain's load limitations. Reducing twist increases the transmission's lifetime and reliability.

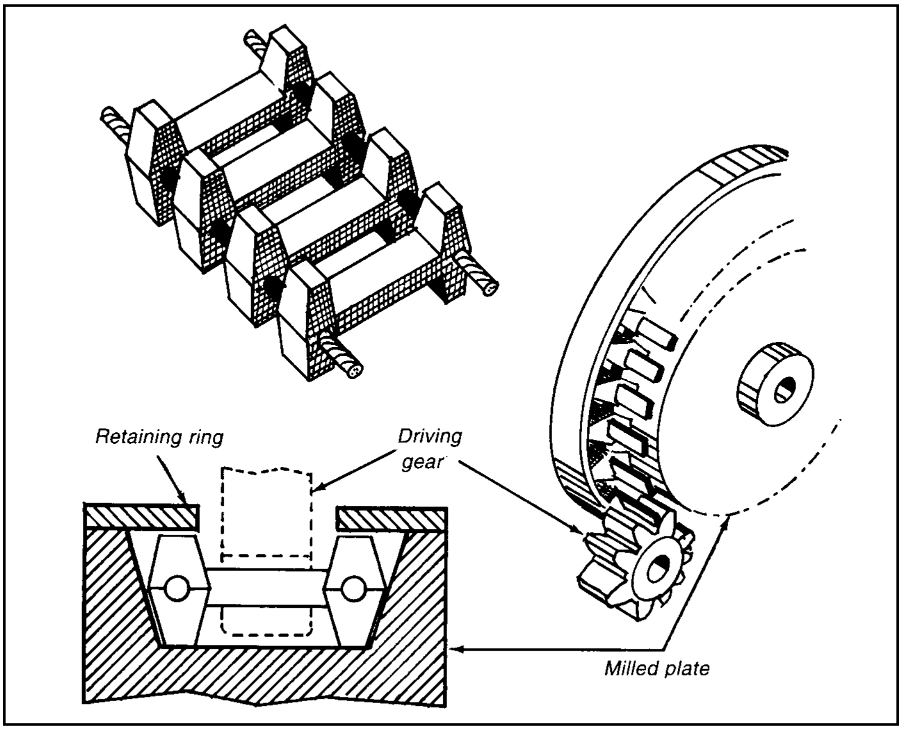


Fig. 6 A gear chain can function as a ladder chain, as a wide V-belt, or, as here, a gear surrogate meshing with a standard pinion.

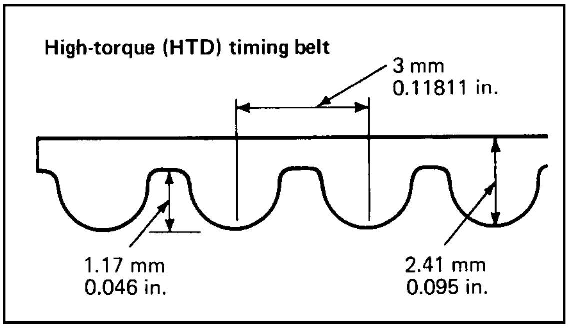


Fig. 7 Curved high-torque tooth profiles (just introduced in 3-mm and 5-mm pitches) increase load capacity of fine-pitch neoprene belts.

These belt-cable-chain hybrids can be sized and connected in the field, using metal crimp-collars. However, nonfactory splices generally reduce the cable's tensile strength by half.

Parallel-Cable Drives

Another species of positive-drive belt uses parallel cables, sacrificing some flexibility for improved stability and greater strength. Here, the cables are connected by rungs molded into the plastic coating, giving the appearance of a ladder (Fig. 6). This "ladder chain" also meshes with toothed pulleys, which need not be grooved.

A cable-and-plastic ladder chain is the basis for the differential drive system in a Hewlett-Packard impact printer (Fig. 5). When the motors rotate in the same direction at the same speed, the carriage moves to the right or left. When they rotate in opposite directions, but at the same speed, the carriage remains stationary and the print-disk rotates. A differential motion of the motors produces a combined translation and rotation of the print-disk.

The hybrid ladder chain is also well suited to laboratory of large spur gears from metal plates or pulleys (Fig. 6). Such a "gear" can run quietly in mesh with a pulley or a standard gear pinion of the proper pitch.

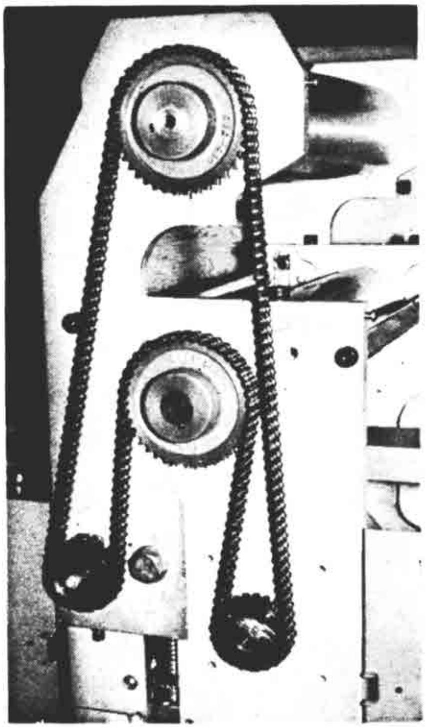
Another type of parallel-cable "chain," which mimics the standard chain, weighs just 1.2 oz/ft, requires no lubrication, and runs almost silently.

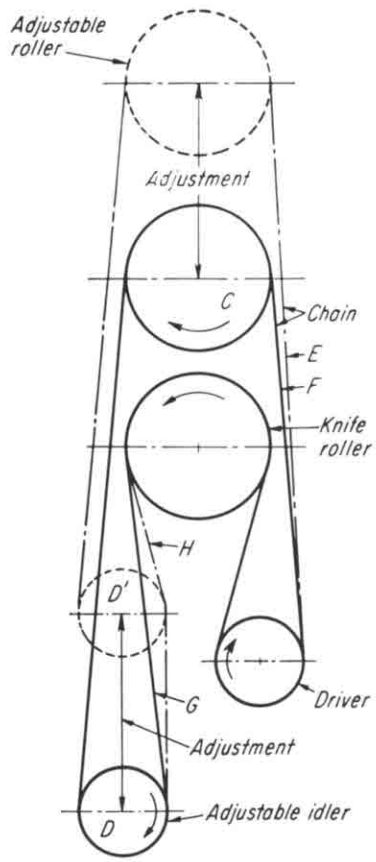
A Traditional Note

A new high-capacity tooth profile has been tested on conventional cogged belts. It has a standard cord and elastic body construction, but instead of the usual trapezoid, it has curved teeth (Fig. 7). Both 3-mm and 5-mm pitch versions have been introduced.

CHANGE CENTER DISTANCE WITHOUT AFFECTING SPEED RATIO

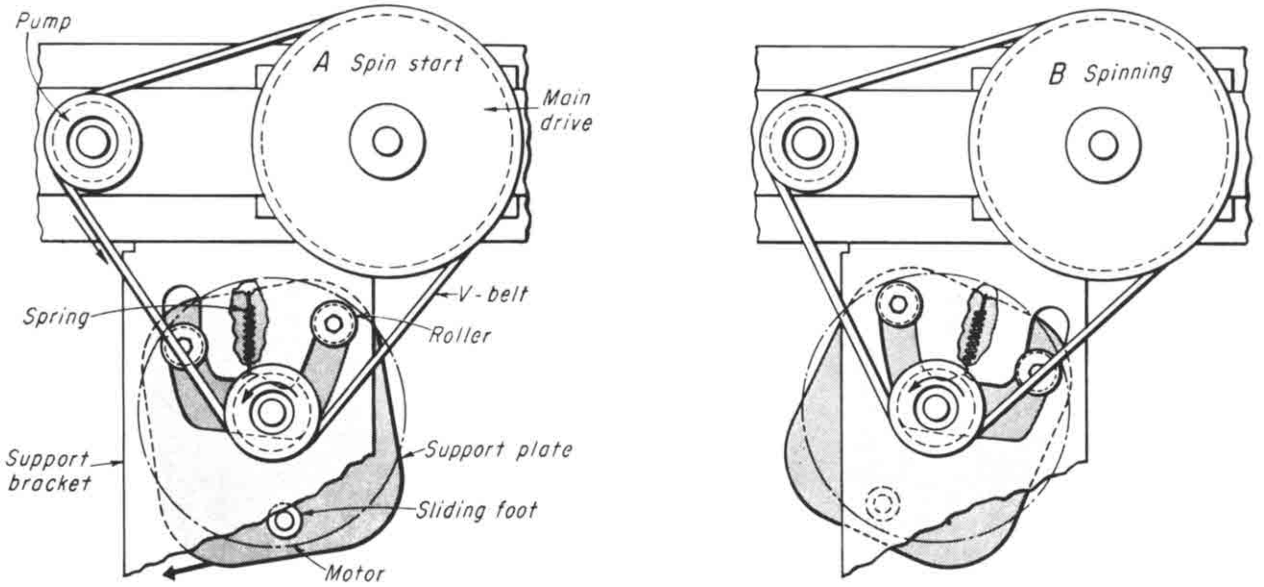
Increasing the gap between the roller and knife changes chain lengths from *F* to *E*. Because the idler moves with the roller sprocket, length *G* changes to *H*. The changes in chain length are similar in value but opposite in direction. Chain lengths *E* minus *F* closely approximate *G* minus *H*. Variations in required chain length occur because the chains do not run parallel. Sprocket offset is required to avoid interference. Slack produced is too minute to affect the drive because it is proportional to changes in the cosine of a small angle (2° to 5°). For the 72-in. chain, variation is 0.020 in.





MOTOR MOUNT PIVOTS FOR CONTROLLED TENSION

Belt tensioning proportional to load



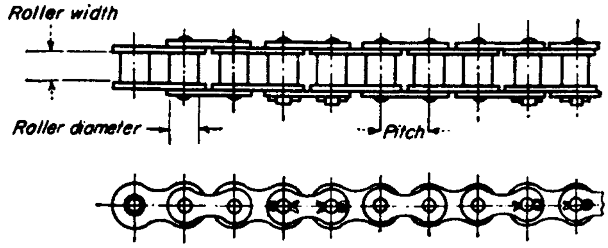
When the agitation cycle is completed, the motor is momentarily idle with the right roller bottomed in the right-hand slot. When spin-dry starts, (A) the starting torque produces a reaction at the stator, pivoting the motor on the bottomed roller. The motor pivots until the opposite roller bottoms in the left-hand slot. The motor now swings out until restrained by the V-belt, which drives the pump and basket.

The motor, momentarily at zero rpm, develops maximum torque and begins to accelerate the load of basket, water, and wash. The motor pivots (B) about the left roller increasing belt tension in proportion to the output torque. When the basket reaches maximum speed, the load is reduced and belt tension relaxes. The agitation cycle produces an identical reaction in the reverse direction.

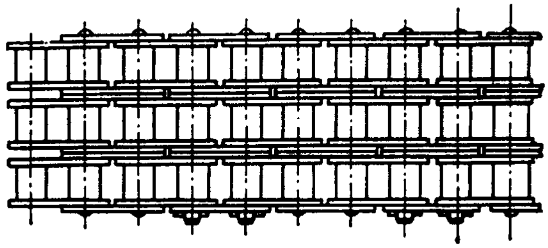
BUSHED ROLLER CHAINS AND THEIR ADAPTATIONS

Various roller, side-plate and pin configurations for power transmissions, conveying, and elevating.

STANDARD ROLLER CHAIN—FOR POWER TRANSMISSION AND CONVEYING

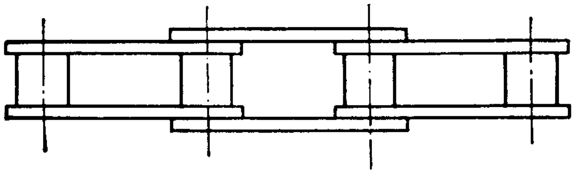


SINGLE WIDTH—Sizes ⅝ in and smaller have a spring-clip connecting link; those ¾ in and larger have a cotter pin.

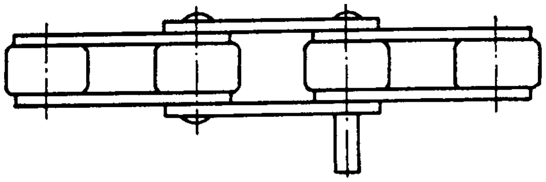


MULTIPLE WIDTH—Similar to single-width chain. It is made in widths up to 12 strands.

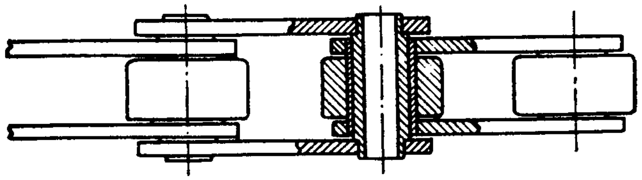
EXTENDED PITCH CHAIN—FOR CONVEYING



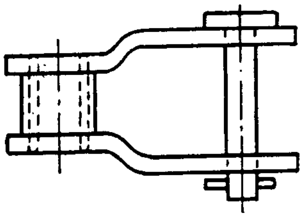
STANDARD ROLLER DIAMETER—made with 1 to 4 in pitch and cotter-pin-type connecting links.



OVERSIZED ROLLER DIAMETER—Same base chain as standard roller type but not made in multiple widths.

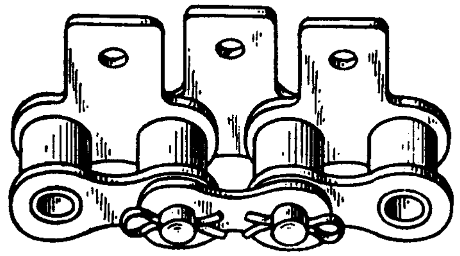


HOLLOW PIN—Made with 1¼ to 15 in pitch. It is adaptable to a variety of bolted attachments.

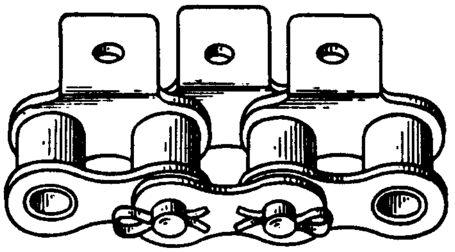


OFFSET LINK—Used when length requires an odd number of pitches and to shorten and lengthen a chain by one pitch.

STANDARD PITCH ADAPTATIONS

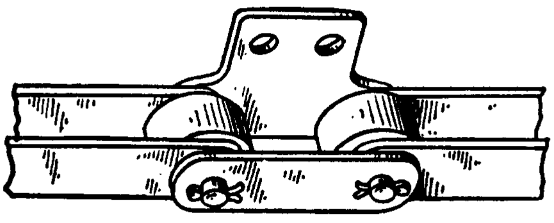


STRAIGHT LUG—Lugs on one or both sides can be spaced as desired. A standard roller is shown.

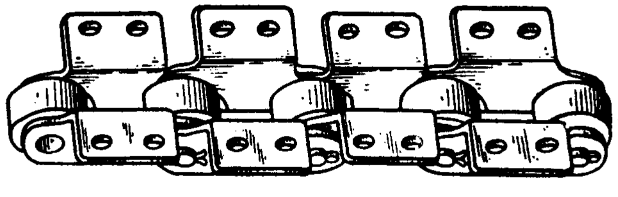


BENT LUG—Similar to straight-lug type for adaptations. A standard roller is shown.

EXTENDED PITCH ADAPTATIONS

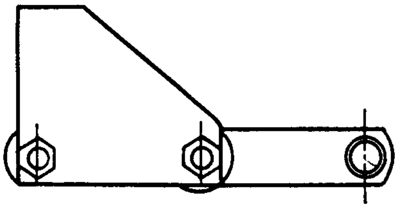


STRAIGHT LUG—An oversized diameter roller is shown.

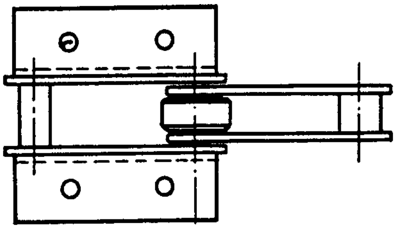


BENT LUG—An oversized diameter roller is shown.

HOLLOW PIN

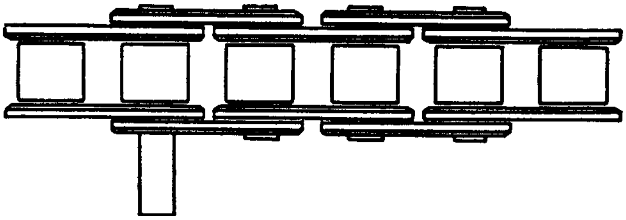


STRAIGHT LUG—Lugs are detachable for field adaptation.

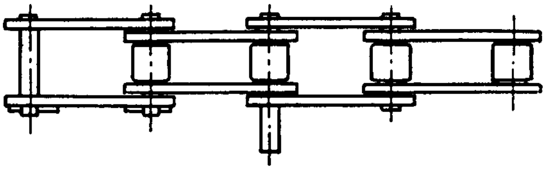


BENT LUG—Similar to straight lug type for adaptations.

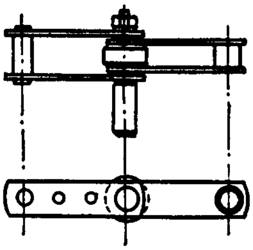
EXTENDED PIN CHAINS



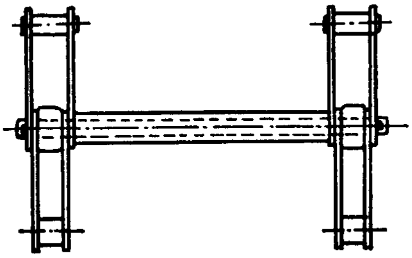
STANDARD PITCH—Pins can be extended on either side.



EXTENDED PITCH—Similar to standard for adaptations.

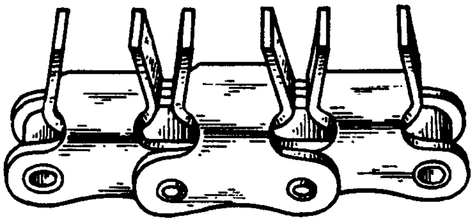


HOLLOW PIN—Pins are designed for field adaptation.

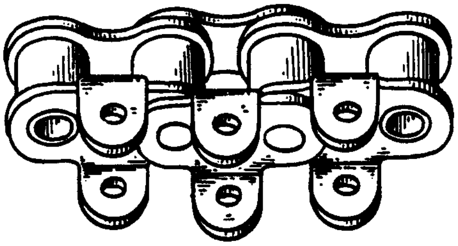


CROSS ROD—The rod can be removed from the hollow pins.

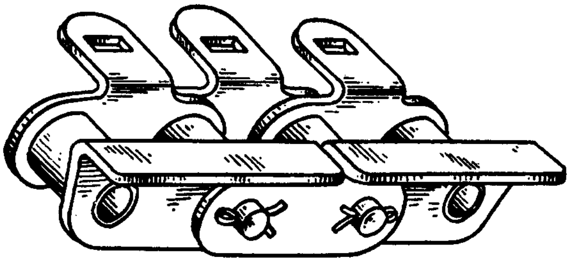
SPECIAL ADAPTATIONS



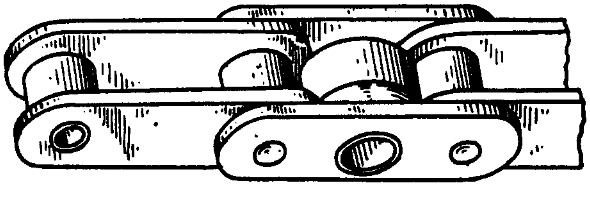
Used for holding conveyed objects.



Used to keep conveyed object on the center-line of the chain.



Used when flexing is desired in one direction only.

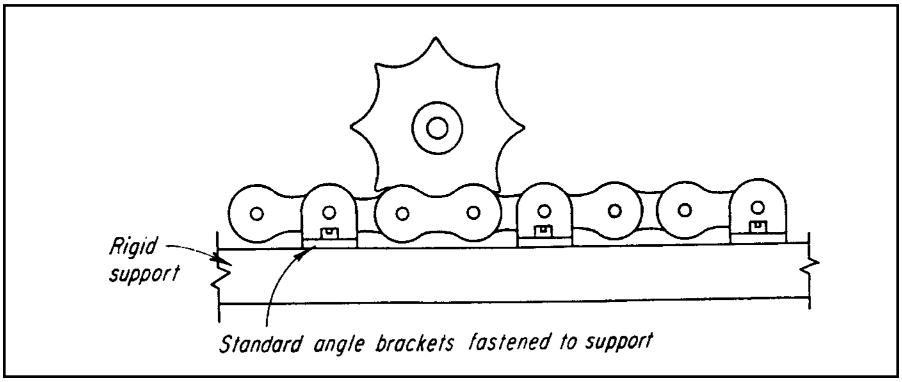


Used for supporting concentrated loads.

SIX INGENIOUS JOBS FOR ROLLER CHAIN

This low-cost industrial chain can be applied in a variety of ways to perform tasks other than simply transmitting power.

Fig. 1 This low-cost rack-and-pinion device is easily assembled from standard parts.



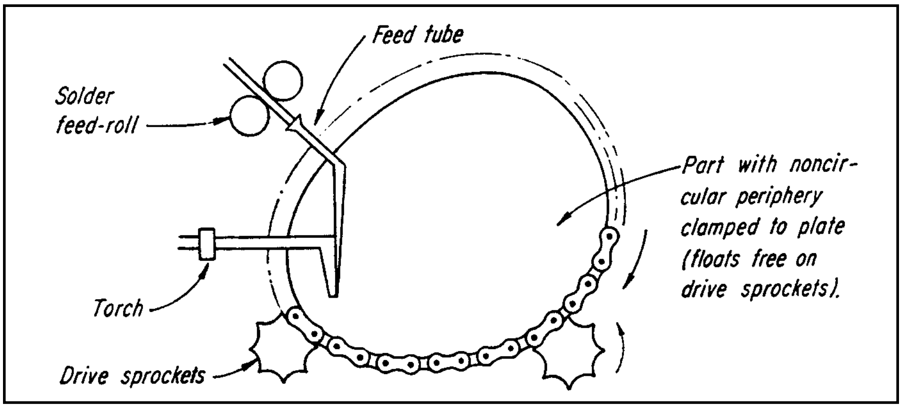
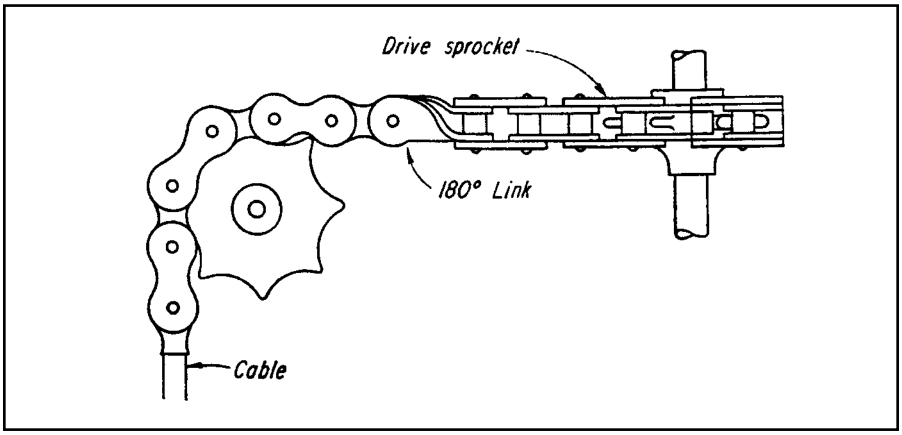


Fig. 2 An extension of the rack-and-pinion principle—This is a soldering fixture for noncircular shells. Positive-action cams can be similarly designed. Standard angle brackets attach the chain to a cam or fixture plate.

Fig. 3 This control-cable dirction-changer is extensively used in aircraft.



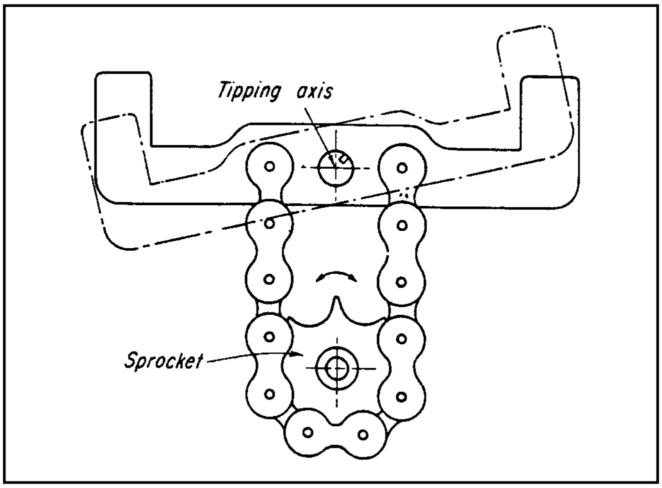


Fig. 4 The transmission of tipping or rocking motion can be combined with the previous example (Fig. 3) to transmit this kind of motion to a remote location and around obstructions. The tipping angle should not exceed 40°.

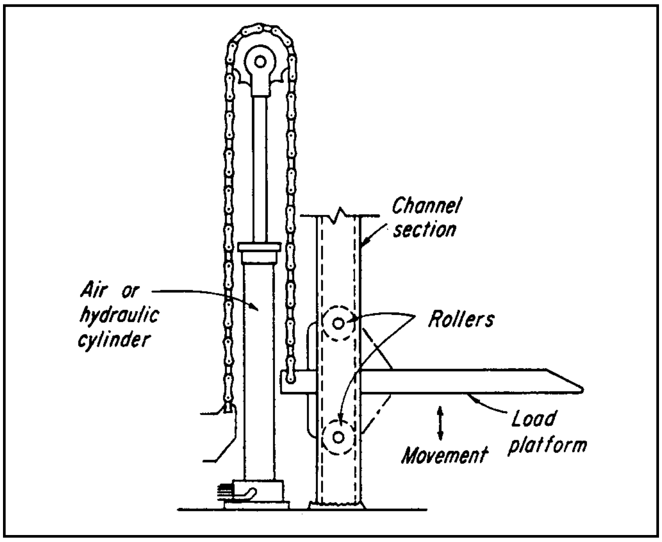


Fig. 5 This lifting device is simplified by roller chain.

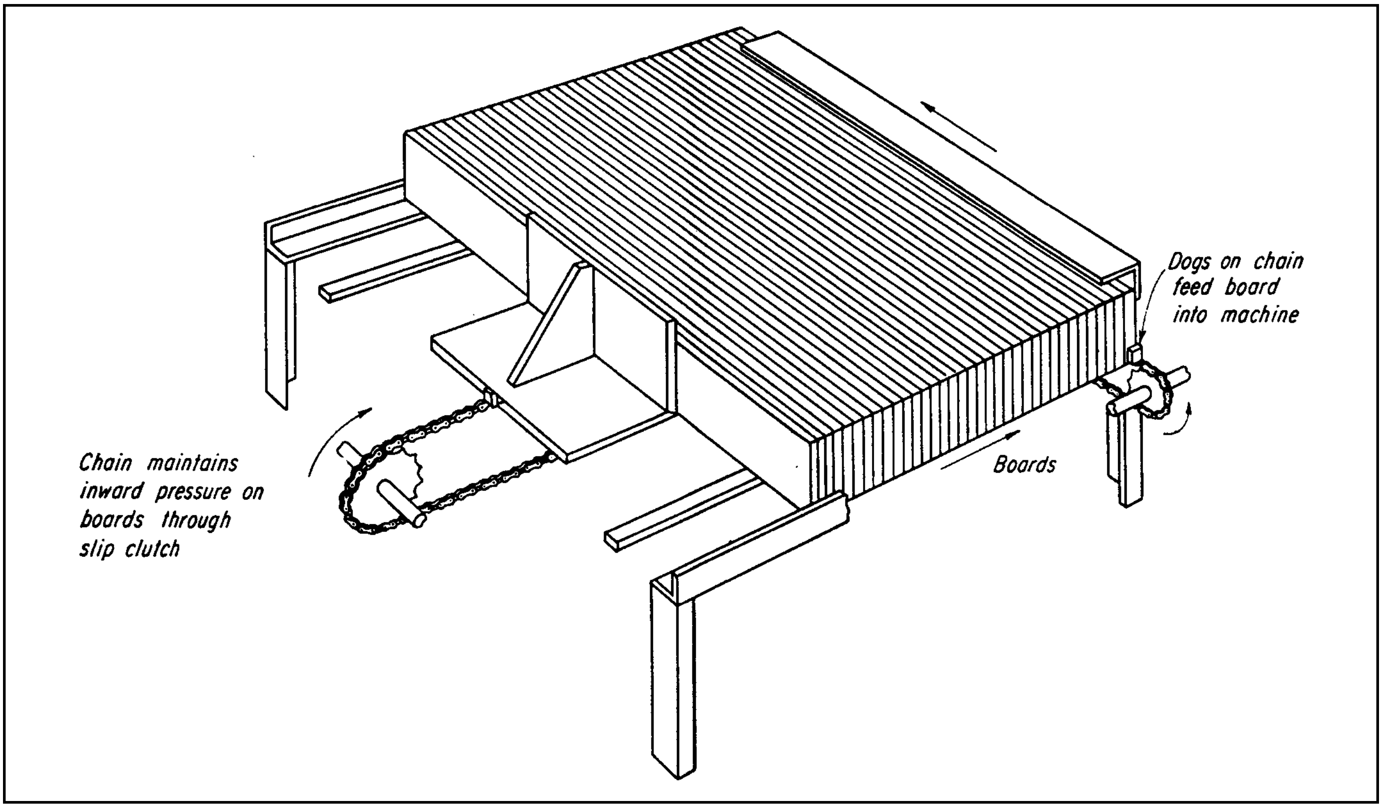


Fig. 6 Two examples of indexing and feeding applications of roller chain are shown here. This setup feeds plywood strips into a machine. The advantages of roller chain as used here are its flexibility and long feed.

SIX MORE JOBS FOR ROLLER CHAIN

Some further examples of how this low-cost but precision-made product can be arranged to do tasks other than transmit power.

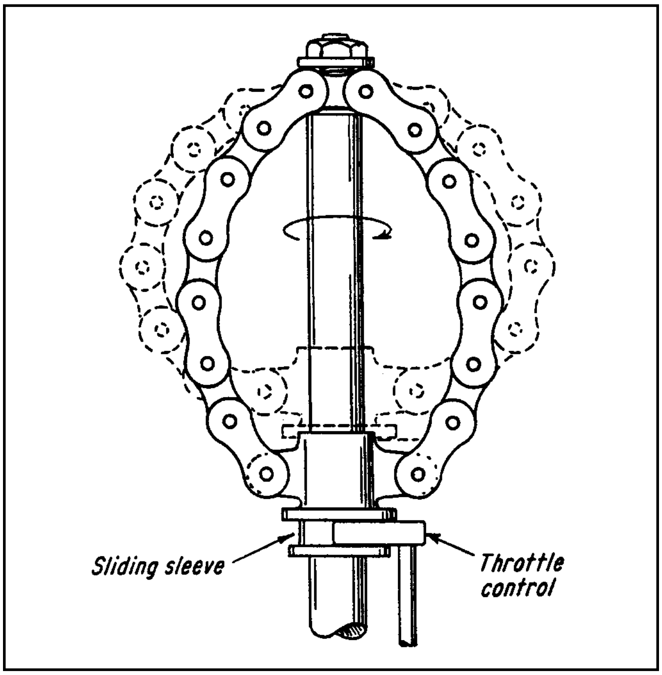


Fig. 1 Simple governor weights can be attached by means of standard brackets to increase response force when rotation speed is slow.

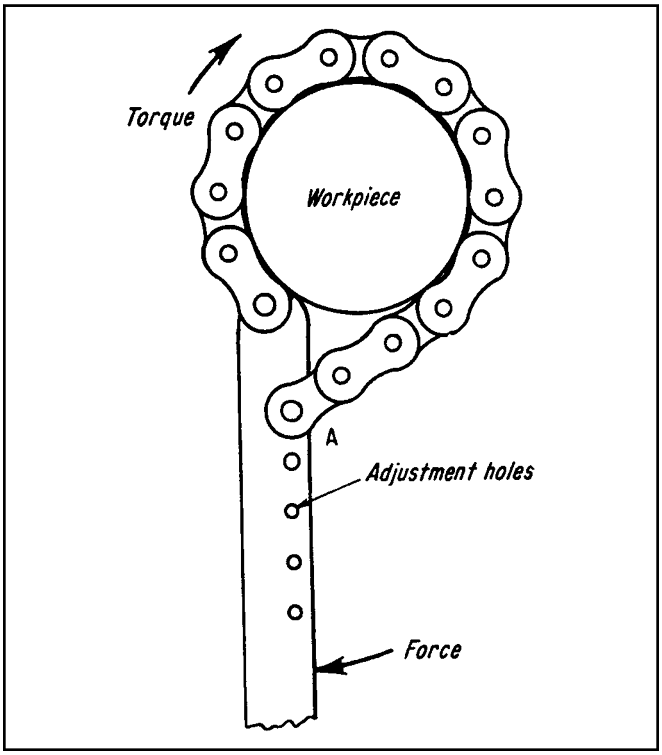
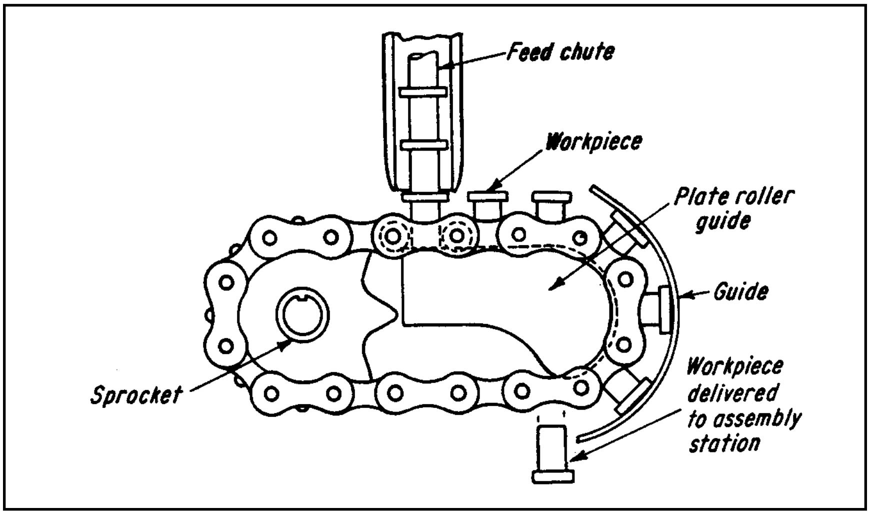


Fig. 2 Wrench pivot A can be adjusted to grip a variety of regularly or irregularly shaped objects.

Fig. 3 Small parts can be conveyed, fed, or oriented between spaces of roller chain.



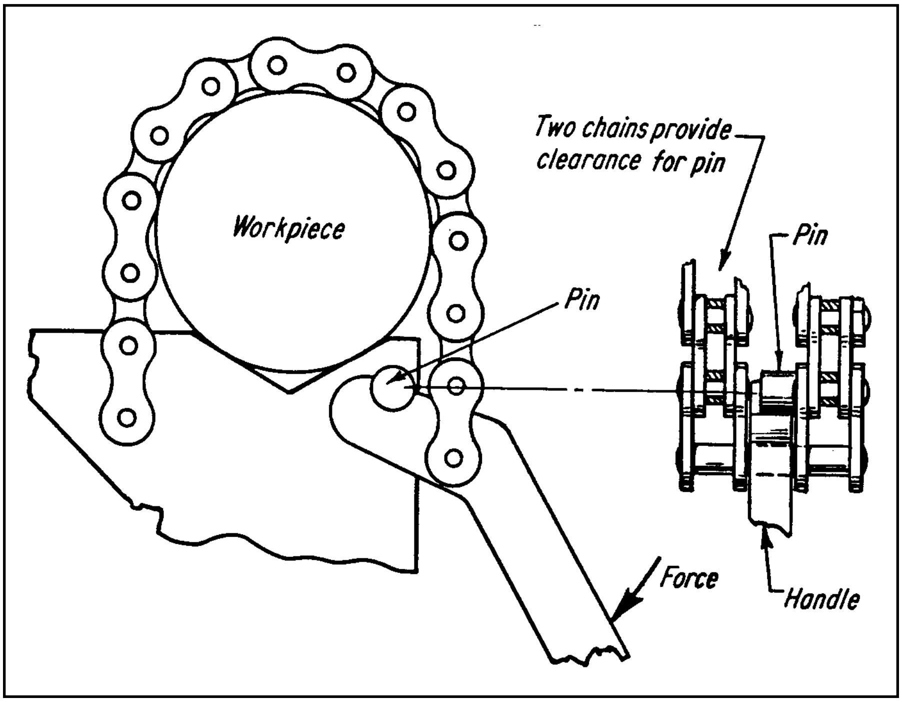


Fig. 4 Clamp toggle action is supplied by two chains, thus clearing pin at fulcrum.

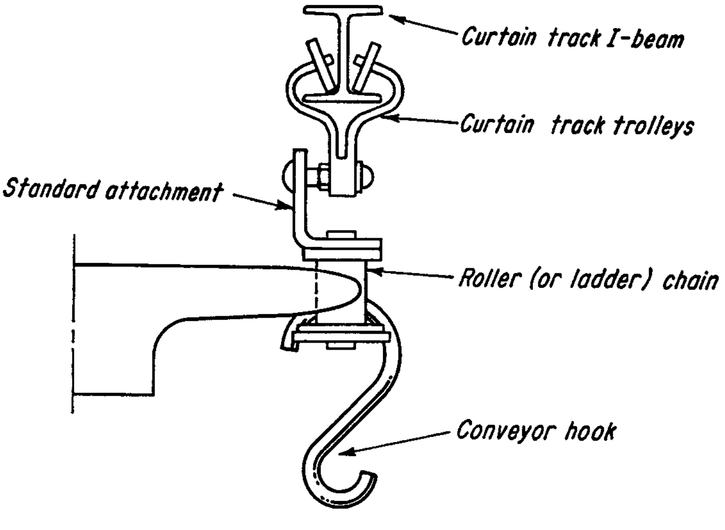


Fig. 5 Light-duty trolley conveyors can be made by combining standard roller-chain components with standard curtain-track components. Small gearmotors are used to drive the conveyor.

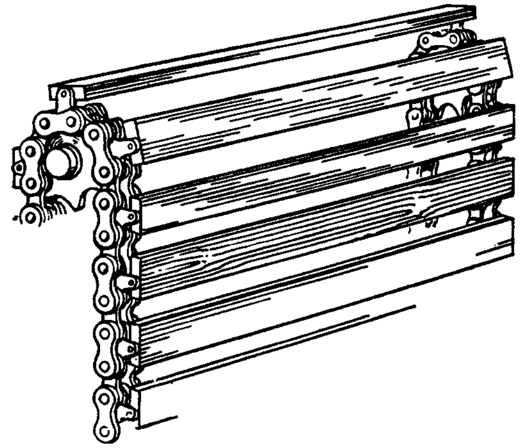


Fig. 6 Slatted belt, made by attaching wood, plastic, or metal slats, can serve as adjustable safety guard, conveyor belt, fast-acting security-wicket window.

MECHANISMS FOR REDUCING PULSATIONS IN CHAIN DRIVES

Pulsations in chain motion created by the chordal action of chain and sprockets can be minimized or avoided by introducing a compensating cyclic motion in the driving sprocket. Mechanisms for reducing fluctuating dynamic loads in chain drives and the pulsations resulting from them include noncircular gears, eccentric gears, and cam-activated intermediate shafts.

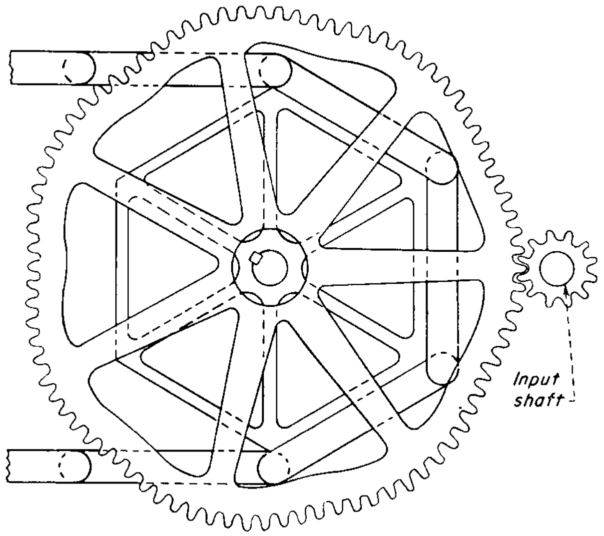


Fig. 1

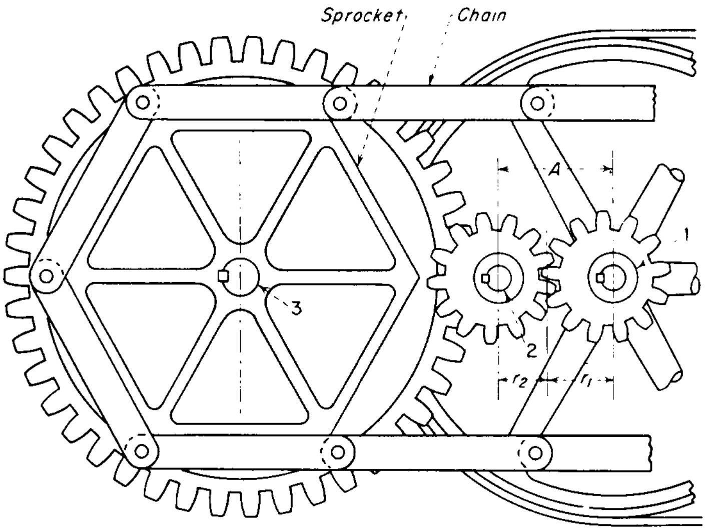


Fig. 2

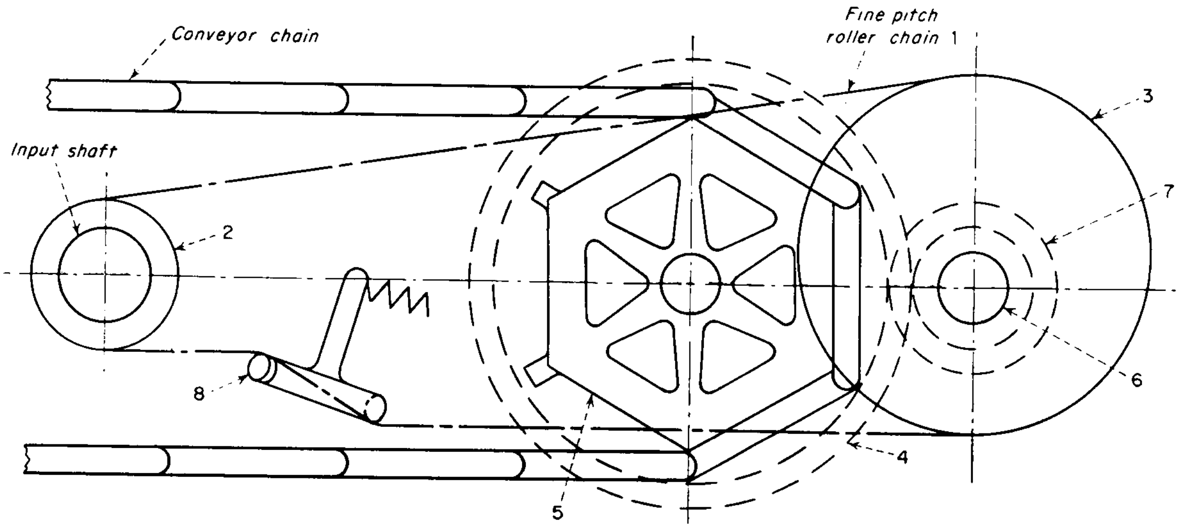


Fig. 3

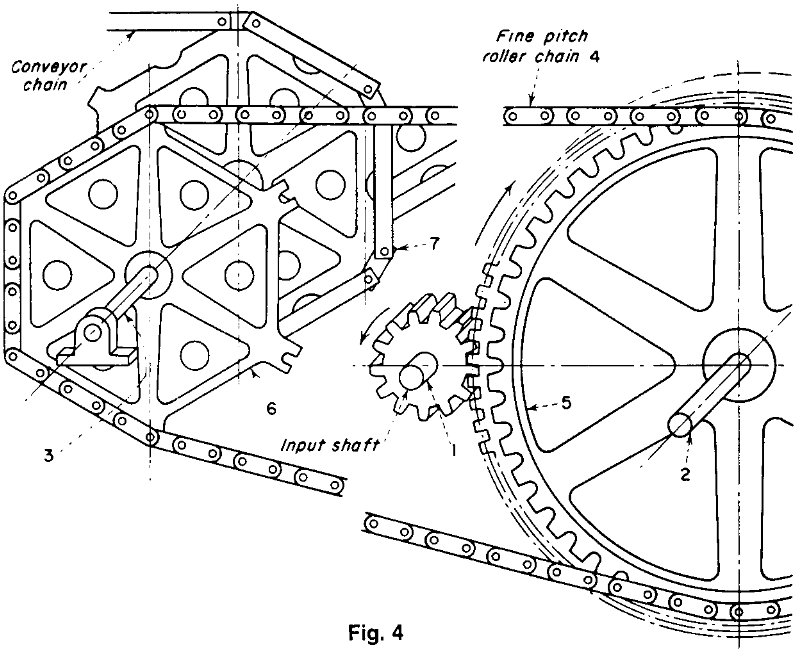


Fig. 4

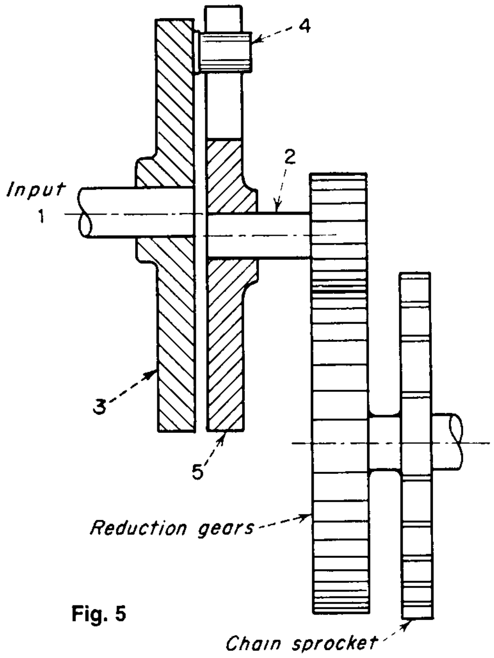


Fig. 5

Fig. 1 The large cast-tooth, noncircular gear, mounted on the chain sprocket shaft, has a wavy outline in which the number of waves equals the number of teeth on a sprocket. The pinion has a corresponding noncircular shape. Although requiring special-shaped gears, the drive completely equalizes the chain pulsations.

Fig. 2 This drive has two eccentrically mounted spur pinions (1 and 2). Input power is through the belt pulley keyed to the same shaft as pinion 1. Pinion 3 (not shown), keyed to the shaft of pinion 2, drives the large gear and sprocket. However, the mechanism does not completely equalize chain velocity unless the pitch lines of pinions 1 and 2 are noncircular instead of eccentric.

Fig. 3 An additional sprocket 2 drives the noncircular sprocket 3 through a fine-pitch chain 1.This imparts pulsating velocity to shaft 6 and to the long-pitch conveyor sprocket 5 through pinion 7 and gear 4. The ratio of the gear pair is made the same as the number of teeth of sprocket 5. Spring-actuated lever and rollers 8 take up the slack. Conveyor motion is equalized, but the mechanism has limited power capacity because the pitch of chain 1 must be kept small. Capacity can be increased by using multiple strands of fine-pitch chain.

Fig. 4 Power is transmitted from shaft 2 to sprocket 6 through chain 4, thus imparting a variable velocity to shaft 3, and through it, to the conveyor sprocket 7. Because chain 4 has a small pitch and sprocket 5 is relatively large, the velocity of 4 is almost constant. This induces an almost constant conveyor velocity. The mechanism requires the rollers to tighten the slack side of the chain, and it has limited power capacity.

Fig. 5 Variable motion to the sprocket is produced by disk 3. It supports pin and roller 4, as well as disk 5, which has a radial slot and is eccentrically mounted on shaft 2. The ratio of rpm of shaft 2 to the sprocket equals the number of teeth in the sprocket. Chain velocity is not completely equalized.

Fig. 6 The integrated "planetary gear" system (gears 4, 5, 6 and 7) is activated by cam 10, and it transmits a variable velocity to the sprocket synchronized with chain pulsations through shaft 2, thus completely equalizing chain velocity. Cam 10 rides on a circular idler roller 11. Because of the equilibrium of the forces, the cam maintains positive contact with the roller. The unit has standard gears, acts simultaneously as a speed reducer, and can transmit high horsepower.

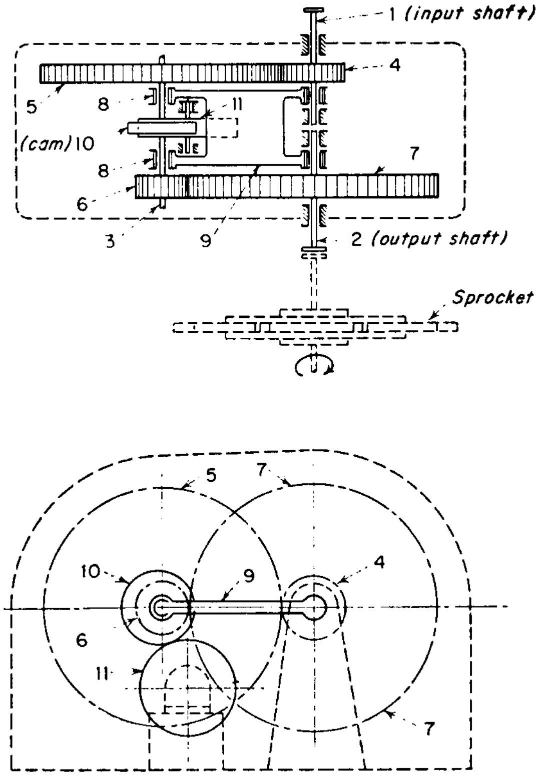


Fig. 6

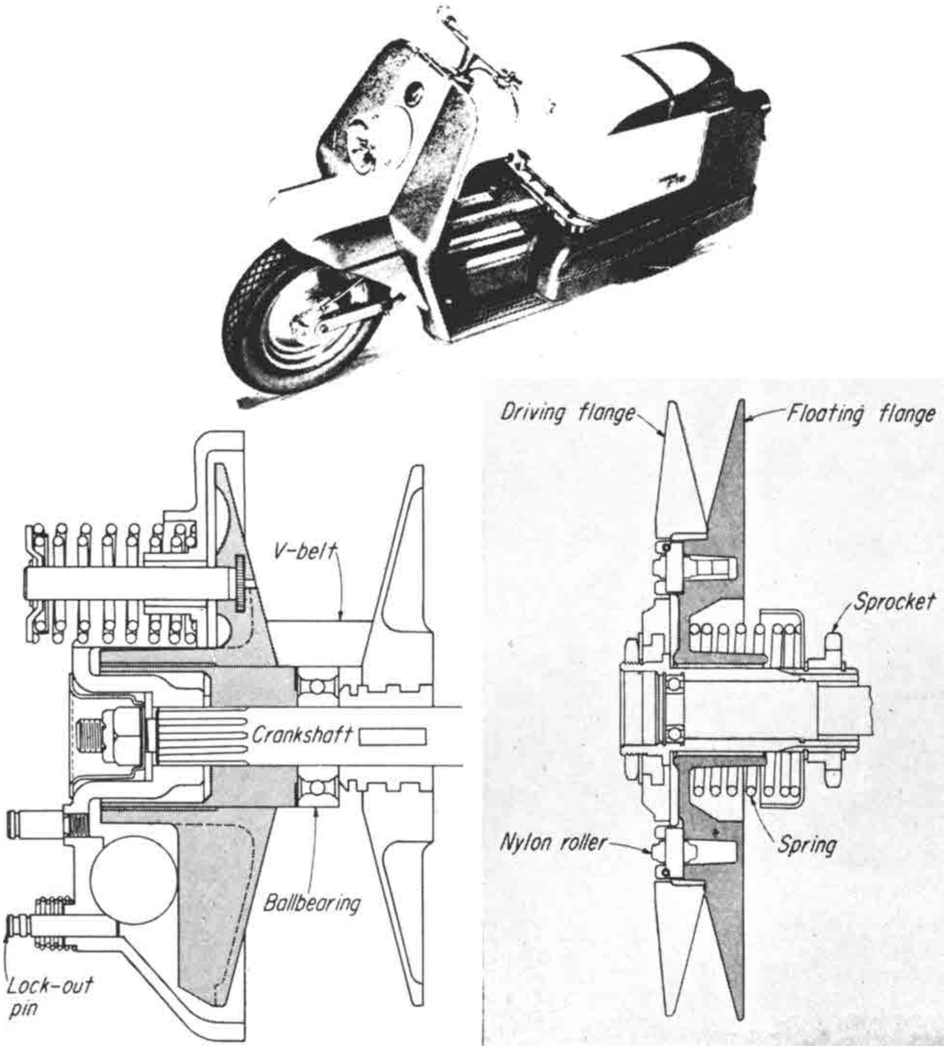
SMOOTHER DRIVE WITHOUT GEARS

The transmission in this motor scoter is torque-sensitive; motor speed controls the continuously variable drive ratio. The operator merely works the throttle and brake.

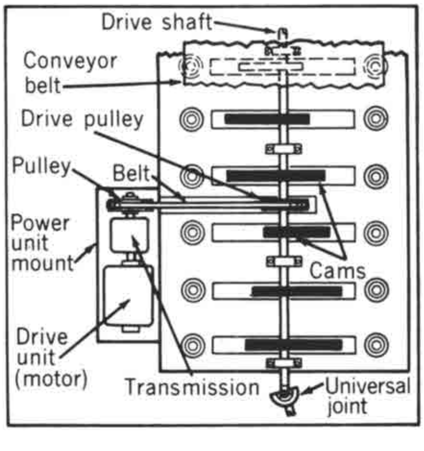
Variable-diameter V-belt pulleys connect the motor and chain drive sprocket to give a wide range of speed reduction. The front pulley incorporates a three-ball centrifugal clutch which forces the flanges together when the engine speeds up. At idle speed the belt rides on a ballbearing between the retracted flanges of the pulley. During starting and warmup, a lockout prevents the forward clutch from operating.

Upon initial engagement, the overall drive ratio is approximately 18:1. As engine speed increases, the belt rides higher up on the forward-pulley flanges until the overall drive ratio becomes approximately 6:1. The resulting variations in belt tension are absorbed by the spring-loaded flanges of the rear pulley. When a clutch is in an idle position, the V-belt is forced to the outer edge of the rear pulley by a spring force. When the clutch engages, the floating half of the front pulley moves inward, increasing its effective diameter and pulling the belt down between the flanges of the rear pulley.

The transmission is torque-responsive. A sudden engine acceleration increases the effective diameter of the rear pulley, lowering the drive ratio. It works this way: An increase in belt tension rotates the floating flange ahead in relation to the driving flange. The belt now slips slightly on its driver. At this time nylon rollers on the floating flange engage cams on the driving flange, pulling the flanges together and increasing the effective diameter of the pulley.



FLEXIBLE CONVEYOR MOVES IN WAVES



Most conventional conveyors used in tunneling and mining can't negotiate curves and can't be powered at different points. They are subject to malfunction because of slight misalignment, and they require time-consuming adjustments to lengthen or shorten them.

Thomas E. Howard of the U.S Bureau of Mines, invented a conveyor belt that does not move forward. That might solve all of these problems.

The conveyor, designed to move broken ore, rock, and coal in mines, moves material along a flexible belt. The belt is given a wave-like movement by the sequenced rising and dropping of supporting yokes beneath it.

The principle. The conveyor incorporates modules built in arcs and Y's in such a way that it can be easily joined with standardized sections to negotiate corners and either merge or separate streams of moving materials. It can be powered at any one point or at several points, and it incorporates automatic controls to actuate only those parts of the belt that are loaded, thereby reducing power consumption.

In tests at the bureau's Pittsburgh Mining Research Center, a simplified mechanical model of the conveyor has moved rock at rates comparable to those of conventional belts.