CHAPTER 5
SPECIAL-PURPOSE MECHANISMS

NINE DIFFERENT BALL SLIDES FOR LINEAR MOTION

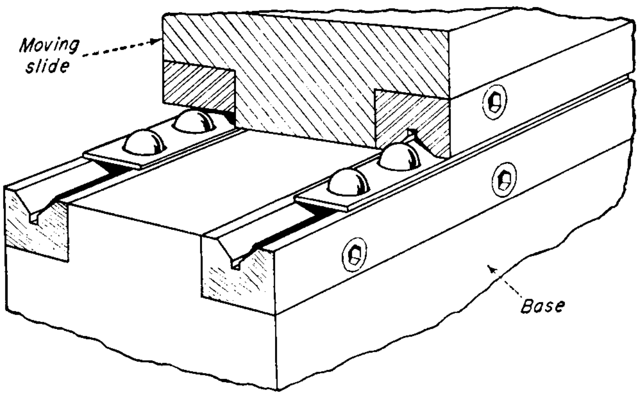


Fig. 1 V-grooves and flat surface make a simple horizontal ball slide for reciprocating motion where no side forces are present and a heavy slide is required to keep the balls in continuous contact. The ball cage ensures the proper spacing of the balls and its contacting surfaces are hardened and lapped.

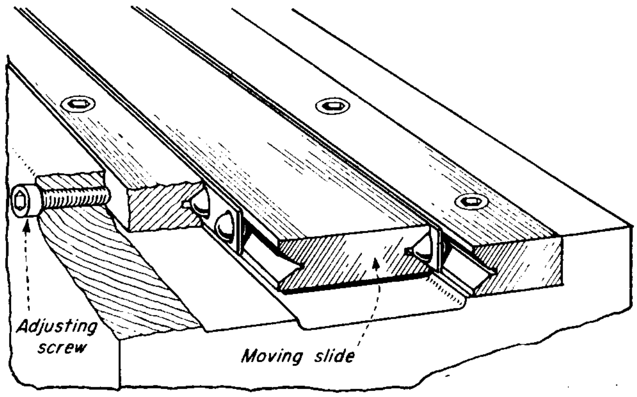


Fig. 2 Double V grooves are necessary where the slide is in a vertical position or when transverse loads are present. Screw adjustment or spring force is required to minimize any looseness in the slide. Metal-to-metal contact between the balls and grooves ensure accurate motion.

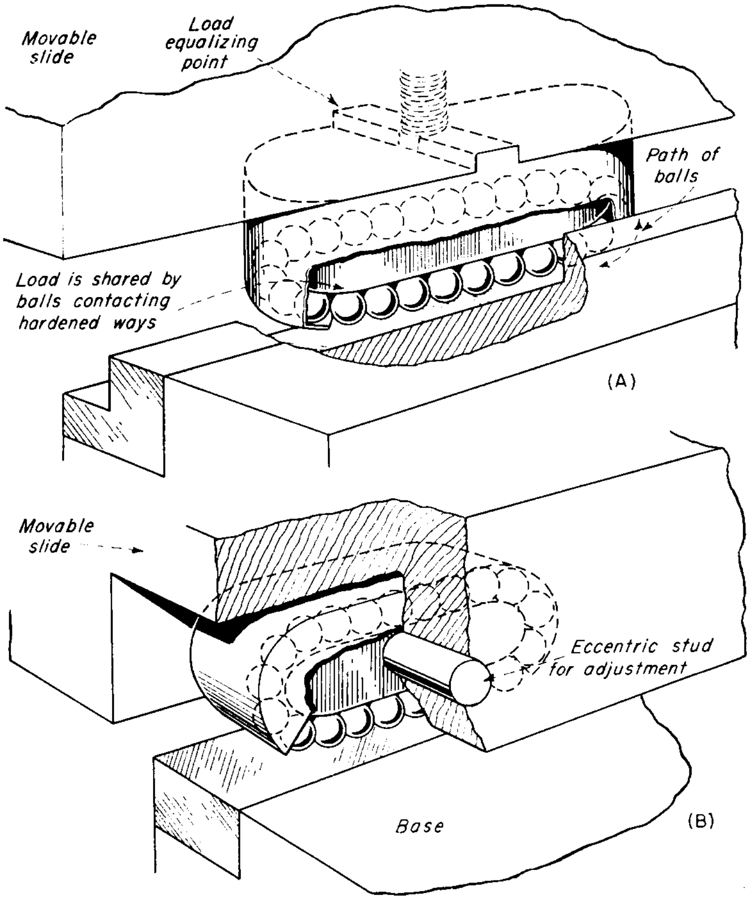


Fig. 3 The ball cartridge has the advantage of unlimited travel because the balls are free to recirculate. Cartridges are best suited for vertical loads. (A) Where lateral restraint is also required, this type is used with a side preload. (B) For flat surfaces the cartridge is easily adjusted.

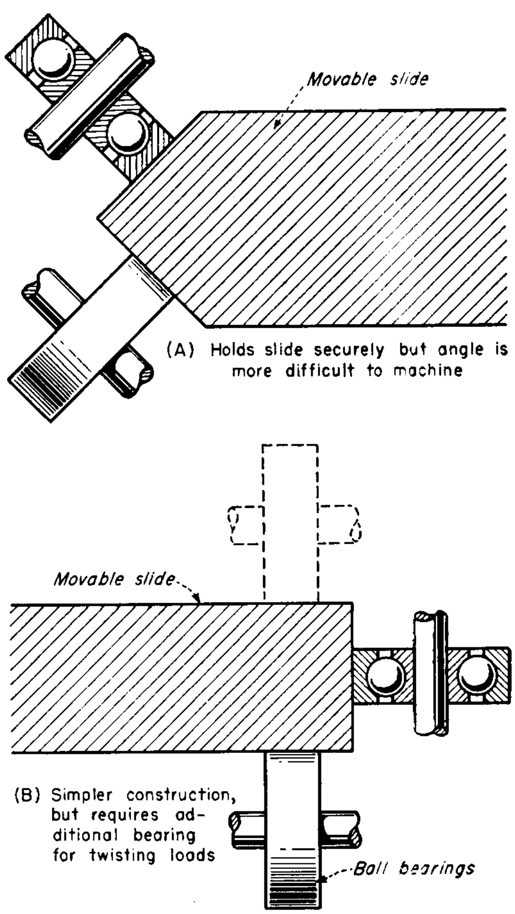


Fig. 4 Commercial ball bearings can be used to make a reciprocating slide. Adjustments are necessary to prevent looseness of the slide. (A) Slide with beveled ends, (B) Rectangular-shaped slide.

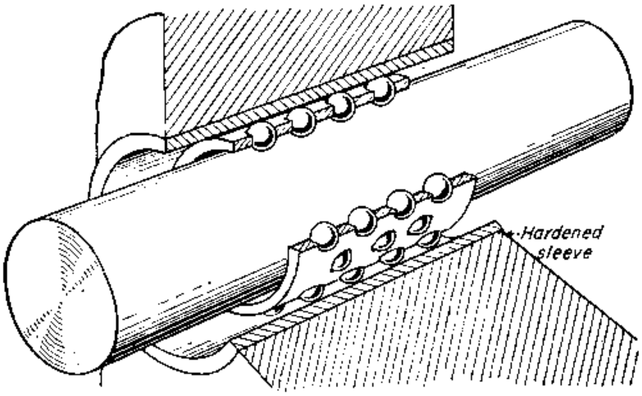


Fig. 5 This sleeve bearing, consisting of a hardened sleeve, balls, and retainer, can be used for reciprocating as well as oscillating motion. Travel is limited in a way similar to that of Fig. 6. This bearing can withstand transverse loads in any direction.

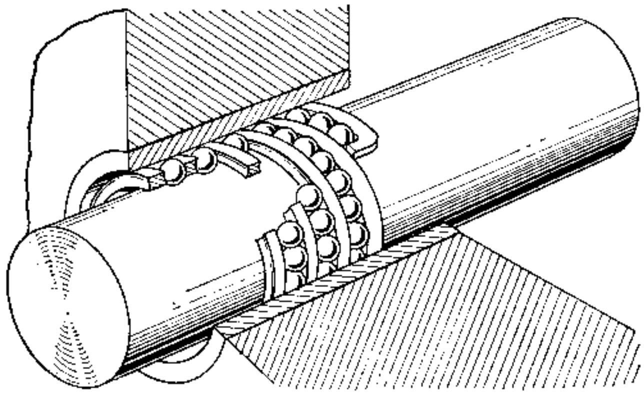


Fig. 6 This ball reciprocating bearing is designed for rotating, reciprocating or oscillating motion. A formed-wire retainer holds the balls in a helical path. The stroke is about equal to twice the difference between the outer sleeve and the retainer length.

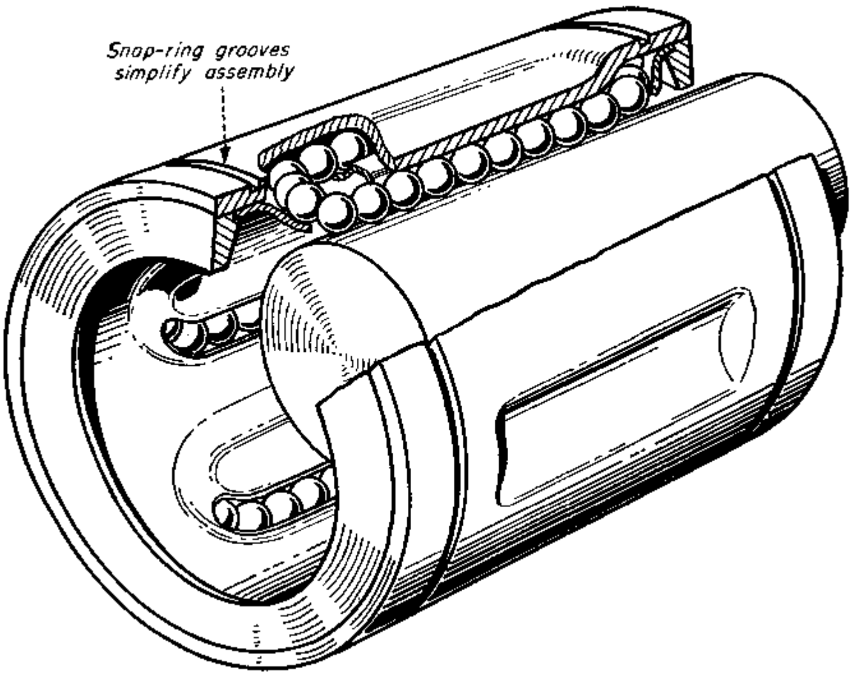


Fig. 7 This ball bushing has several recirculating systems of balls that permit unlimited linear travel. Very compact, this bushing requires only a bored hole for installation. For maximum load capacity, a hardened shaft should be used.

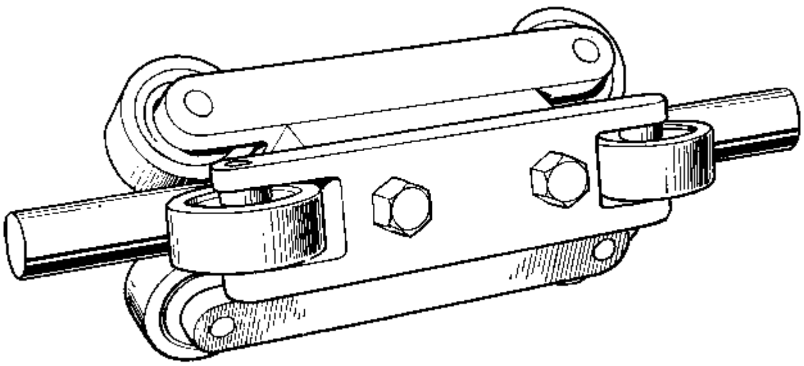


Fig. 8 Cylindrical shafts can be held by commercial ball bearings that are assembled to make a guide. These bearings must be held tightly against the shaft to prevent any looseness.

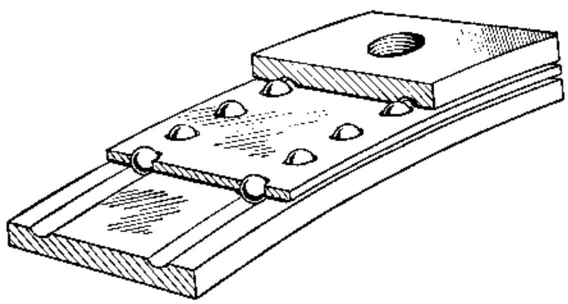
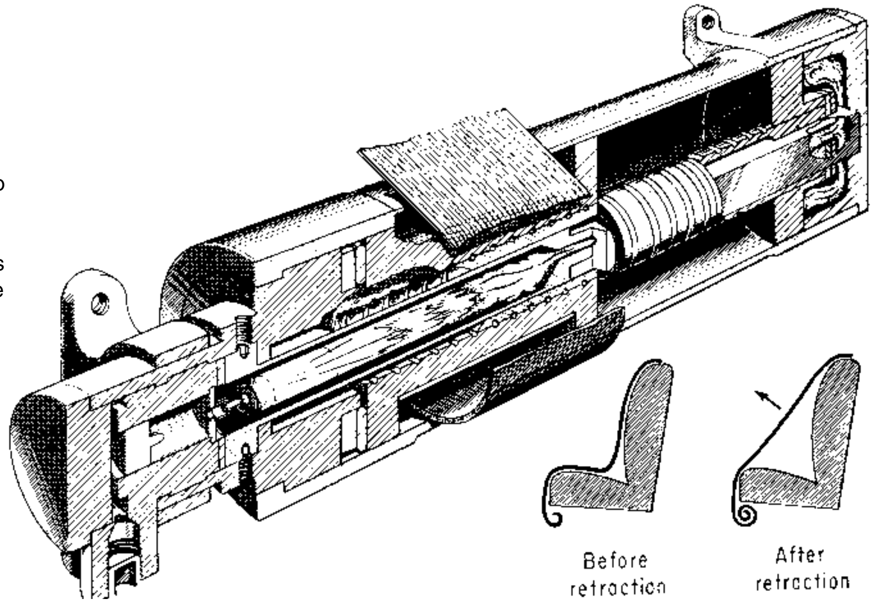
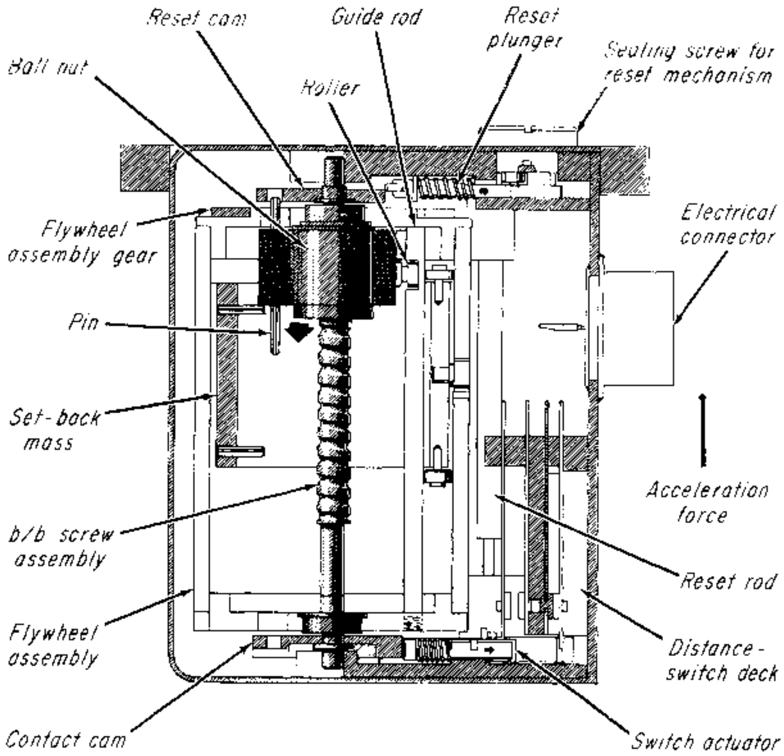


Fig. 9 Curvilinear motion in a plane is possible with this device when the radius of curvature is large. However, uniform spacing between its grooves is important. Circular-sectioned grooves decrease contact stresses.

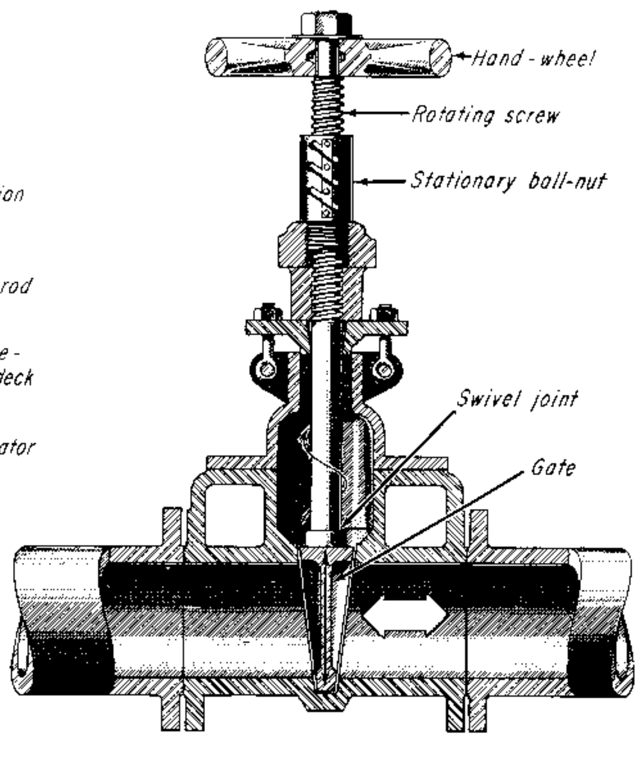
BALL-BEARING SCREWS CONVERT ROTARY TO LINEAR MOTION

This cartridge-operated rotary actuator quickly retracts the webbing to separate a pilot forcibly from his seat as the seat is ejected in emergencies. It eliminates the tendency of both pilot and seat to tumble together after ejection, preventing the opening of the chute. Gas pressure from the ejection device fires the cartridge in the actuator to force the ball-bearing screw to move axially. The linear motion of the screw is translated into the rotary motion of a ball nut. This motion rapidly rolls up the webbing (stretching it as shown) so that the pilot is snapped out of his seat.





This time-delay switching device integrates a time function with a missile's linear travel. Its purpose is to arm the warhead safely. A strict "minimum G-time" system might arm a slow missile too soon for the adequate protection of friendly forces because a fast missile might arrive before the warhead is fused. The weight of the nut plus the inertia under acceleration will rotate the ball-bearing screw which has a flywheel on its end. The screw pitch is selected so that the revolutions of the flywheel represent the distance the missile has traveled.



Fast, easy, and accurate control of fluid flow through a valve is obtained by the rotary motion of a screw in the stationary ball nut. The screw produces linear movement of the gate. The swivel joint eliminates rotary motion between the screw and the gate.

THREE-POINT GEAR/LEADSCREW POSITIONING

The mechanism helps keep the driven plate parallel to a stationary plate.

Lewis Research Center, Cleveland, Ohio

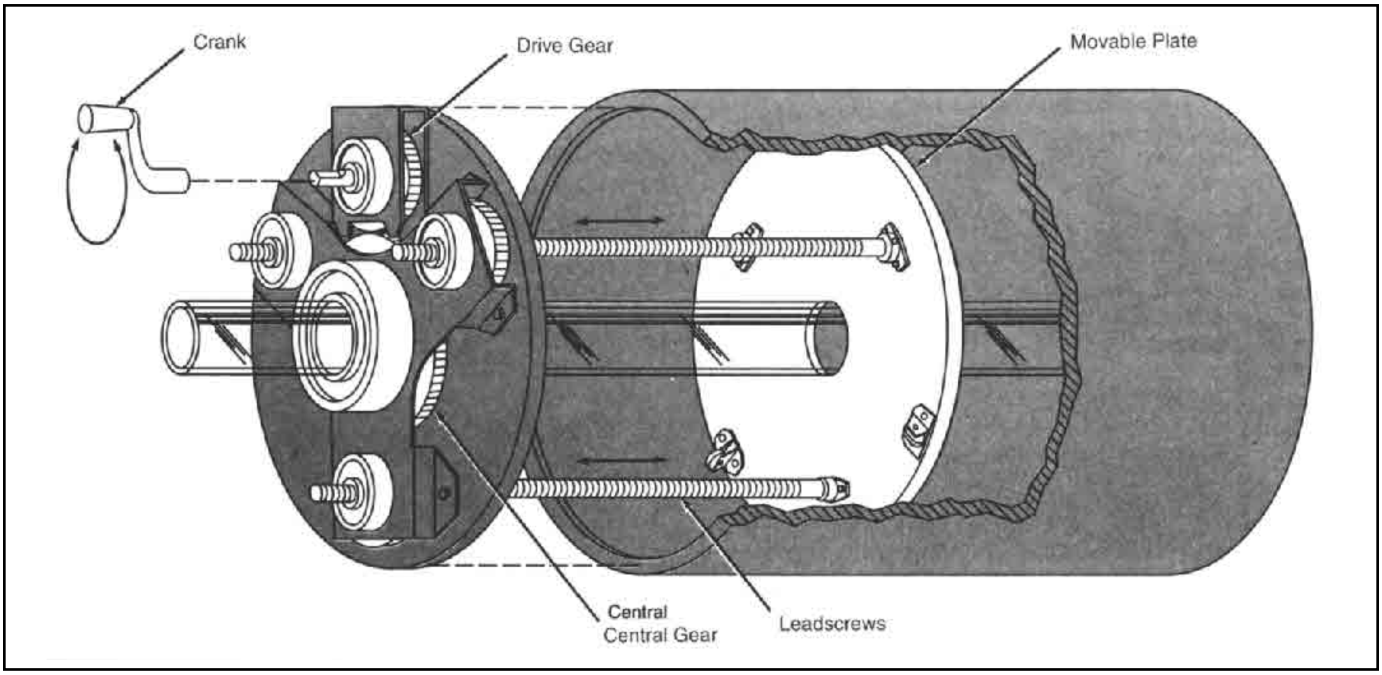
A triple-ganged-leadscrew positioning mechanism drives a movable plate toward or away from a fixed plate and keeps the plates parallel to each other. The mechanism was designed for use in tuning a microwave resonant cavity. The parallel plates are the end walls, and the distance between is the critical dimension to be adjusted. Other potential applications for this or similar mechanisms include adjustable bed plates and cantilever tail stocks in machine tools, adjustable platforms for optical equipment, and lifting platforms.

In the original tunable-microwave-cavity application, the new mechanism replaces a variety of prior mechanisms. Some of those included single-point drives that were subject to backlash (with consequent slight tilting and uncertainty in the distance between the plates). Other prior mechanisms relied on spring loading, differential multiple-point drives and other devices to reduce backlash. In providing three-point drive along a track between the movable and fixed plates, the new mechanism ensures the distance between, and parallelism of, the two plates. It is based on the fundamental geometric principle that three points determine a plane.

The moving parts of the mechanism are mounted on a fixed control bracket that, in turn, is mounted on the same rigid frame that holds the fixed plate and the track along which the movable plate travels (see figure). A large central gear turns on precise ball bearings and drives three identical pinion gears at the corners of an equilateral triangle. The central gear is driven by a hand-cranked or motor-driven drive gear similar to one of the pinion gears.

Each pinion gear is mounted on a hollow shaft that turns on precise ball bearings, and the hollow shaft contains a precise internal thread that mates with one of the leadscrews. One end of each leadscrew is attached to the movable plate. The meshing of the pinions and the central gear is set so that the three leadscrews are aligned with each other and the movable plate is parallel with the fixed plate.

This work was done by Frank S. Calco of **Lewis Research Center.**



The Triple-Ganged-Leadscrew Mechanism, shown here greatly simplified, positions the movable plate along the track while keeping the movable plate parallel to the fixed plate.

UNIQUE LINKAGE PRODUCES PRECISE STRAIGHT-LINE MOTION

A patented family of straight-line mechanisms promises to serve many demands for movement without guideways and with low friction.

A mechanism for producing, without guideways, straight-line motion very close to true has been invented by James A Daniel, Jr., Newton, N.J. A patent has been granted, and the linkage was applied to a camera to replace slides and telescoping devices.

Linkages, with their minimal pivot friction, serve many useful purposes in machinery, replacing sliding and rolling parts that need guideways or one type or another.

James Watt, who developed the first such mechanism in 1784, is said to have been prouder of it than of his steam engine. Other well-known linkage inventors include Evans, Tchebicheff, Roberts, and Scott-Russell.

Four-bar arrangement. Like other mechanisms that aim at straight-line motion, the Daniel design is based on the common four-bar linkage. Usually it is the selection of a certain point on the center link—the "coupler," which can extend past its pivot points—and of the location and proportions of the links that is the key to a straight-line device.

According to Daniel, the deviation of his mechanism from a straight line is "so small it cannot easily be measured." Also, the linkage has the ability to support a weight from the moving point of interest with an equal balance as the point moves along. "This gives the mechanism powers of neutral equilibrium," said Daniel.

Patented action. The basic version of Daniel's mechanism (Fig. 1) consists of the four-bar *ABCD*. The coupler link *BC* is extended to *P* (the proportions of the links must be selected according to a rule). Rotation of link *CD* about *D* (Fig. 2) causes *BA* to rotate about A and point *P* to follow approximately a straight line as it moves to P^1. Another point, *Q*, will move along a straight path to Q^1, also without need for a guide. A weight hung on *P* would be in equilibrium.

"At first glance," said Daniel, "the Evans linkage [Fig. 4] may look similar to mine, but link *CD*, being offset from the perpendicular at *A*, prevents the path of *P* from being a straight line."

Watt's mechanism *EFGD* (Fig. 5) is another four-bar mechanism that will produce a path of *C* that is roughly a straight line as *EF* or *GD* is rotated. Tchebicheff combined the Watt and Evans mechanisms to create a linkage in which point *C* will move almost perpendicularly to the path of *P*.

Steps in layout. Either end of the coupler can be redundant when only one straight-line movement is required (Fig. 6). Relative lengths of the links and placement of the pivots are critical, although different proportions are easily obtained for design purposes (Fig. 7). One proportion, for example, allows the path of *P* to pass below the lower support pivot, giving complete clearance to the traveling member. Any Daniel mechanism can be laid out as follows:

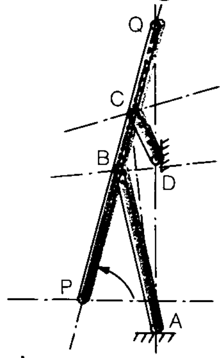
- Lay out any desired right triangle *PQF* (Fig. 3). Best results are with angle A approximately 75 to 80º.
- Pick point *B* on *PQ*. For greatest straight-line motion, *B* should be at or near the midpoint of *PQ*.
- Lay off length *PD* along *FQ* from *F* to find point *E*.
- Draw *BE* and its perpendicular bisector to find point *A*.
- Pick any point *C*. Lay off length *PC* on *FQ* from *F* to find point *G*.
- Draw *CG* and its perpendicular bisector to find *D*. The basic mechanism is *ABCD* with *PQ* as the extension of *BC*.

Multilinked versions. A "gang" arrangement (Fig. 8) can be useful for stamping or punching five evenly spaced holes at one time. Two basic linkages are joined, and the *Q* points will provide short, powerful strokes.

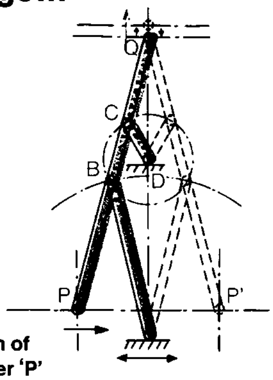
An extended dual arrangement (Fig. 9) can support the traveling point at both ends and can permit a long stroke with no interference. A doubled-up parallel arrangement (Fig. 10) provides a rigid support and two pivot points to obtain the straight-line motion of a horizontal bar.

When the traveling point is allowed to clear the pivot support (Fig. 11), the ultimate path will curve upward to provide a handy "kick" action. A short kick is obtained by adding a stop (Fig. 12) to reverse the direction of the frame links while the long coupler continues its stroke. Daniel suggested that this curved path is useful in engaging or releasing an object on a straight path.

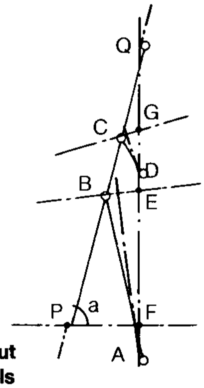
Daniel's basic straight-line linkage...



1. Three bars and four pivots

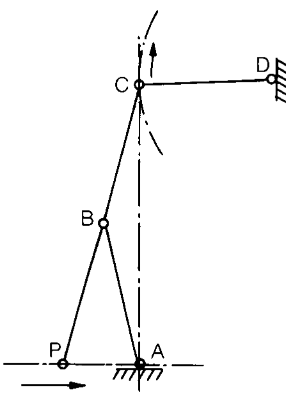


2. Motion of traveler 'P'

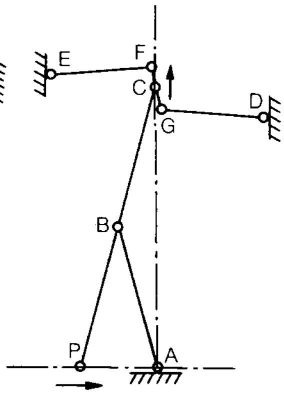


3. Layout details

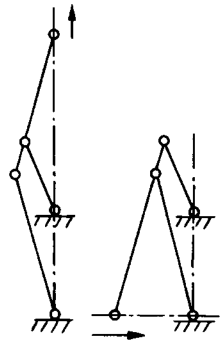
...differs from competing types and leads to many variations



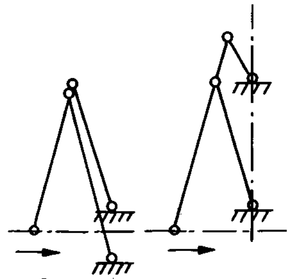
4. Evans' linkage



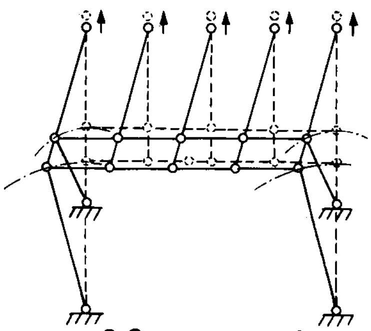
5. Watt-Evans' linkage



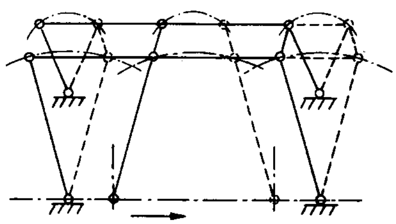
6. Single-extension versions



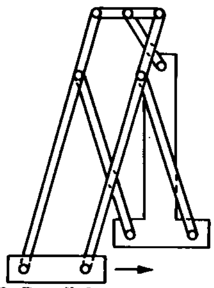
7. Other workable proportions



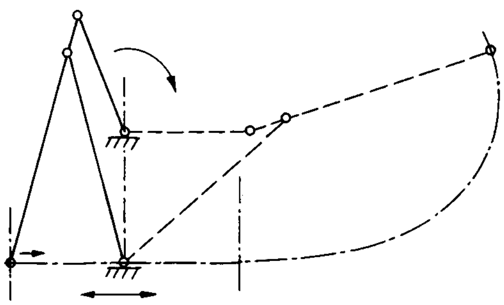
8. Gang arrangement



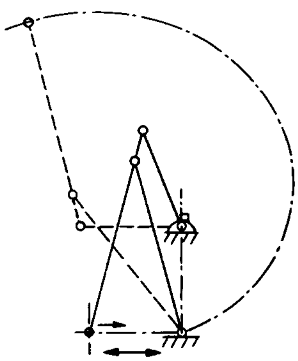
9. Extended dual arrangement



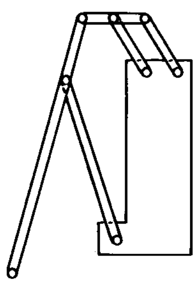
10. Parallel arrangement



11. Long-kick mechanism



12. Short-kick mechanism



13. Multi-link variation

TWELVE EXPANDING AND CONTRACTING DEVICES

Parallel bars, telescoping slides, and other devices that can spark answers to many design problems.

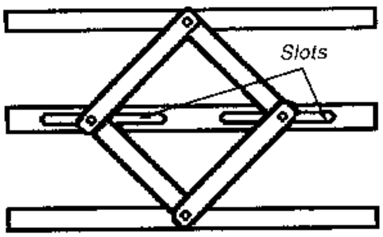


Fig. 1

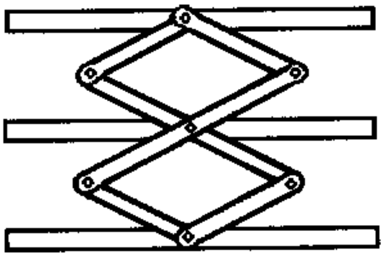


Fig. 2

Figs. 1 and 2 Expanding grilles are often put to work as a safety feature. A single parallelogram (fig. 1) requires slotted bars; a double parallelogram (fig. 2) requires none—but the middle grille-bar must be held parallel by some other method.

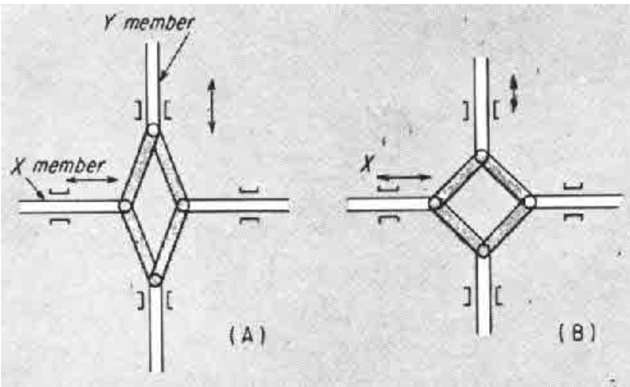


Fig. 3 Variable motion can be produced with this arrangement. In (A) position, the *Y* member is moving faster than the *X* member. In (B), speeds of both members are instantaneously equal. If the motion is continued in the same direction, the speed of *X* will become greater.

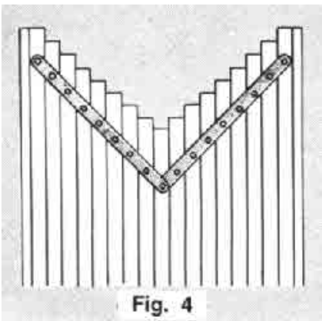


Fig. 4

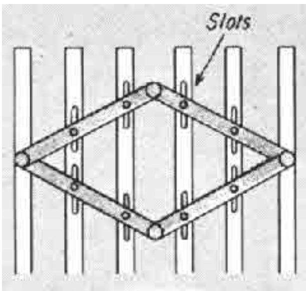


Fig. 5

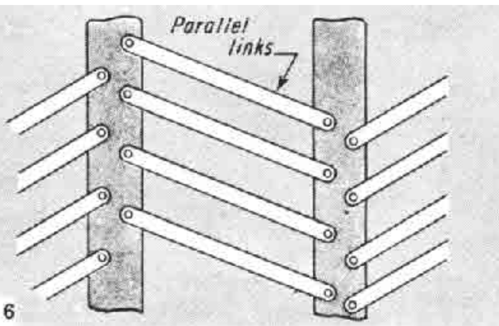


Fig. 6

Figs. 4, 5, and 6 Multibar barriers such as shutters and gates (fig. 4) can take various forms. Slots (fig. 5) allow for vertical adjustment. The space between bars can be made adjustable (fig. 6) by connecting the vertical bars with parallel links.

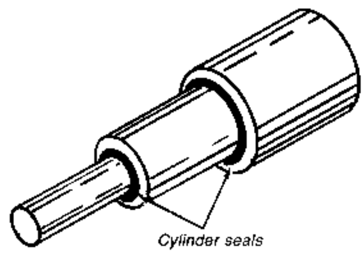


Fig. 7 Telescoping cylinders are the basis for many expanding and contracting mechanisms. In the arrangement shown, nested tubes can be sealed and filled with a highly temperature-responsive medium such as a volatile liquid.

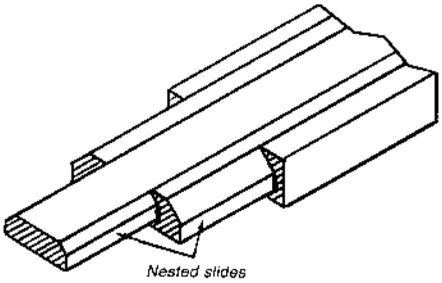


Fig. 8 Nested slides can provide an extension for a machine-tool table or other structure where accurate construction is necessary. In this design, adjustments to obtain smooth sliding must be made first before the table surface is leveled.

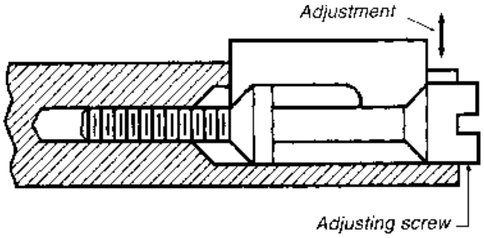


Fig. 9 Circular expanding mandrels are well-known. The example shown here is a less common mandrel-type adjustment. A parallel member, adjusted by two tapered surfaces on the screw, can exert a powerful force if the taper is small.

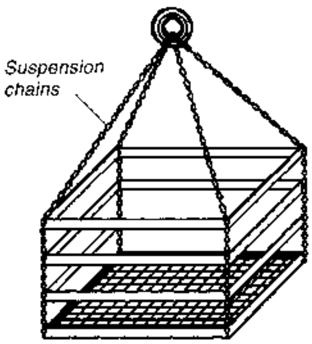


Fig. 10 This expanding basket is opened when suspension chains are lifted. Baskets take up little space when not in use. A typical use for these baskets is for conveyor systems. As tote baskets, they also allow easy removal of their contents because they collapse clear of the load.

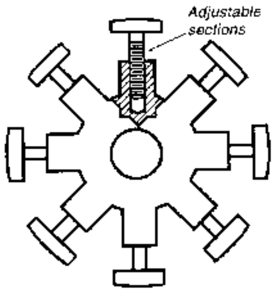


Fig. 11 An expanding wheel has various applications in addition to acting as a pulley or other conventional wheel. Examples include electrical contact on wheel surfaces that allow many repetitive electrical functions to be performed while the wheel turns. Dynamic and static balancing is simplified when an expanding wheel is attached to a nonexpanding main wheel. As a pulley, an expanding wheel can have a steel band fastened to only one section and then passed twice around the circumference to allow for adjustment.

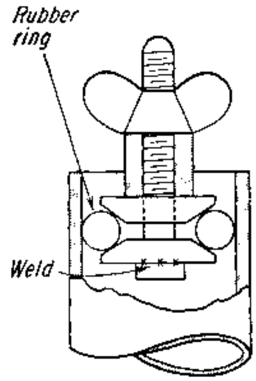


Fig. 12 A pipe stopper depends on a building rubber "O" ring for its action—soft rubber will allow greater conformity than hard rubber. It will also conform more easily to rough pipe surfaces. Hard rubber, however, withstands higher pressures. The screw head is welded to the washer for a leaktight joint.

FIVE LINKAGES FOR STRAIGHT-LINE MOTION

These linkages convert rotary to straight-line motion without the need for guides.

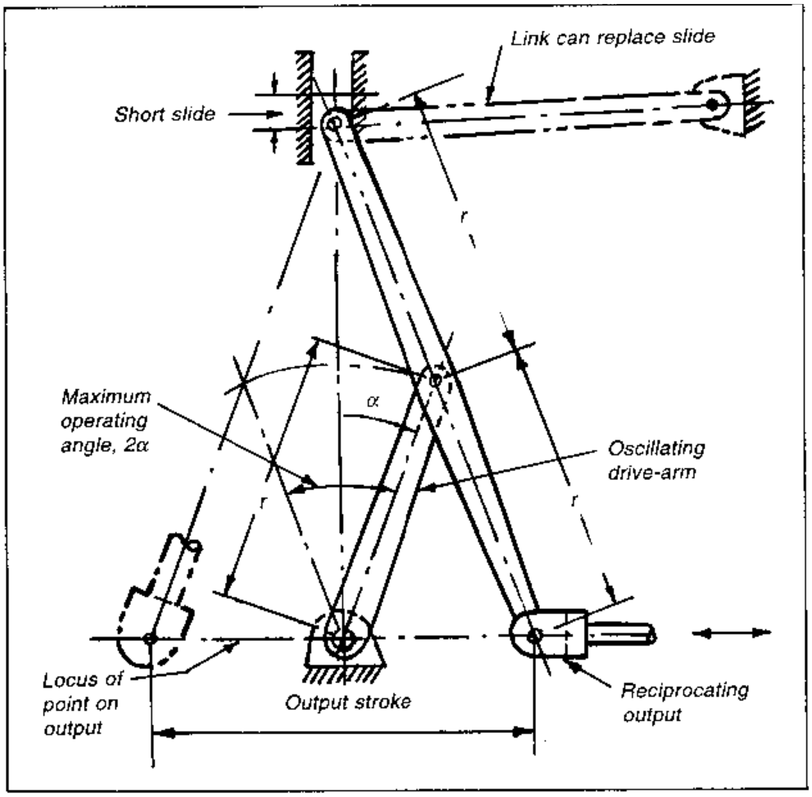


Fig. 1 An Evans' linkage has an oscillating drive-arm that should have a maximum operating angle of about 40º. For a relatively short guideway, the reciprocating output stroke is large. Output motion is on a true straight line in true harmonic motion. If an exact straight-line motion is not required, a link can replace the slide. The longer this link, the closer the output motion approaches that of a true straight line. If the link-length equals the output stroke, deviation from straight-line motion is only 0.03% of the output stroke.

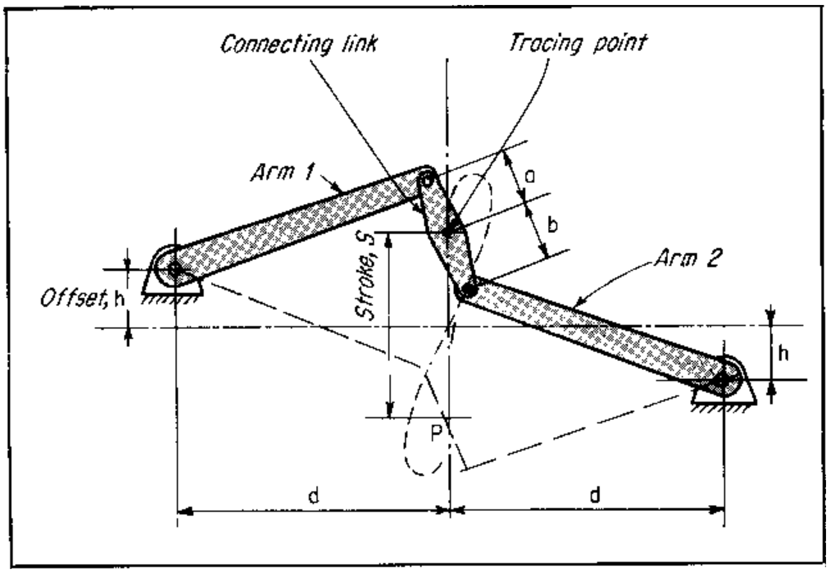


Fig. 2 A simplified Watt's linkage generates an approximate straight-line motion. If the two arms are of equal length, the tracing point describes a symmetrical figure 8 with an almost straight line throughout the stroke length. The straightest and longest stroke occurs when the connecting-link length is about two-thirds of the stroke, and arm length is 1.5 times the stroke length. Offset should equal half the connecting-link length. If the arms are unequal, one branch of the figure-8 curve is straighter than the other. It is straightest when a/b equals (arm 2)/(arm 1).

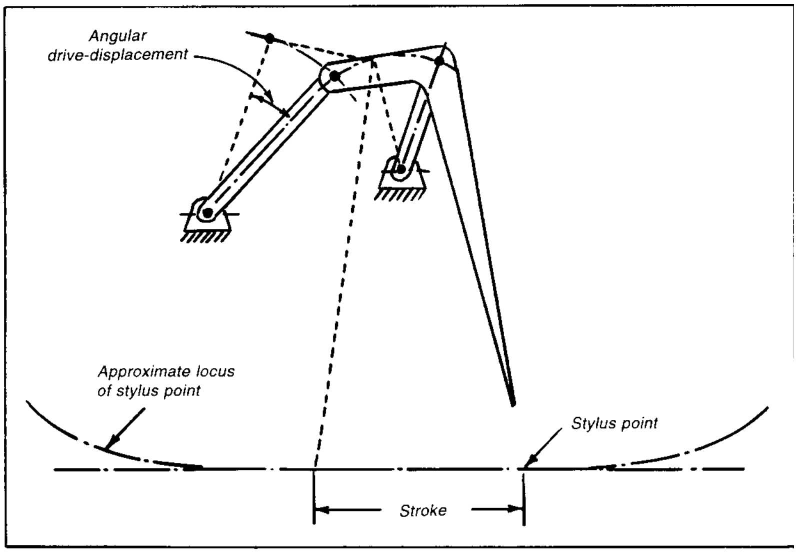


Fig. 3 Four-bar linkage produces an approximately straight-line motion. This arrangement provides motion for the stylus on self-registering measuring instruments. A comparatively small drive displacement results in a long, almost-straight line.

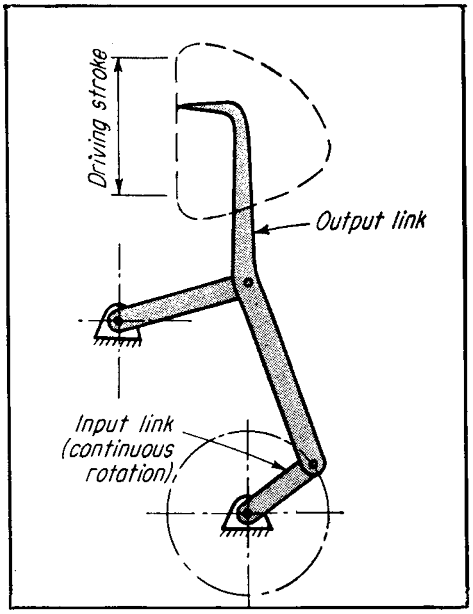


Fig. 4 A D-drive is the result when linkage arms are arranged as shown here. The output-link point describes a path that resembles the letter *D*, so there is a straight part of its cycle. This motion is ideal for quick engagement and disengagement before and after a straight driving stroke.

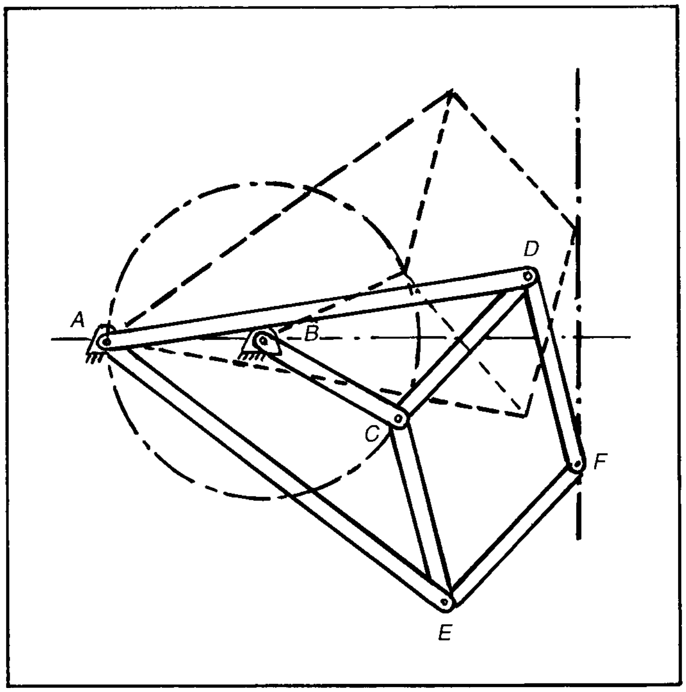


Fig. 5 The "Peaucellier cell" was the first solution to the classical problem of generating a straight line with a linkage. Within the physical limits of the motion, $AC \times AF$ remains constant. The curves described by C and F are, therefore, inverse; if C describes a circle that goes through A, then F will describe a circle of infinite radius—a straight line, perpendicular to AB. The only requirements are that: $AB = BC$; $AD = AE$; and CD, DF, FE, EC be equal. The linkage can be used to generate circular arcs of large radius by locating A outside the circular path of C.

LINKAGE RATIOS FOR STRAIGHT-LINE MECHANISMS

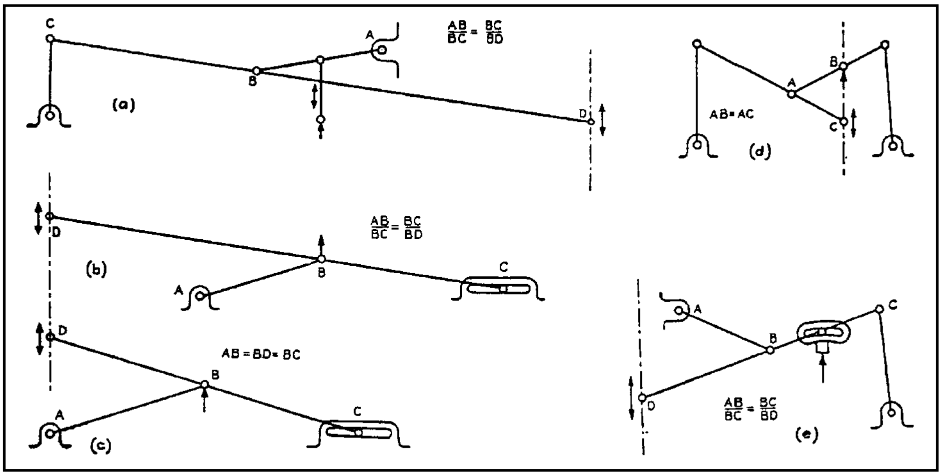


Fig. 1 (a), (b), (c), (d), (e)—Isoceles linkages.

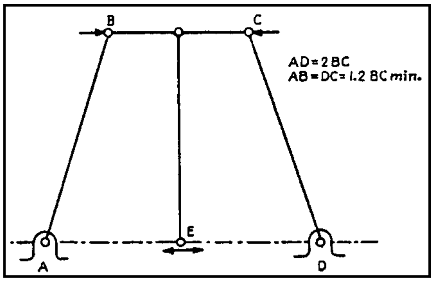


Fig. 2 Robert's linkage.

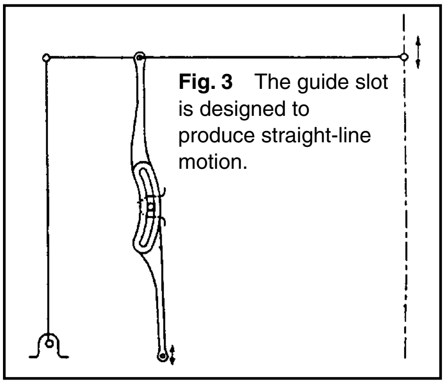


Fig. 3 The guide slot is designed to produce straight-line motion.

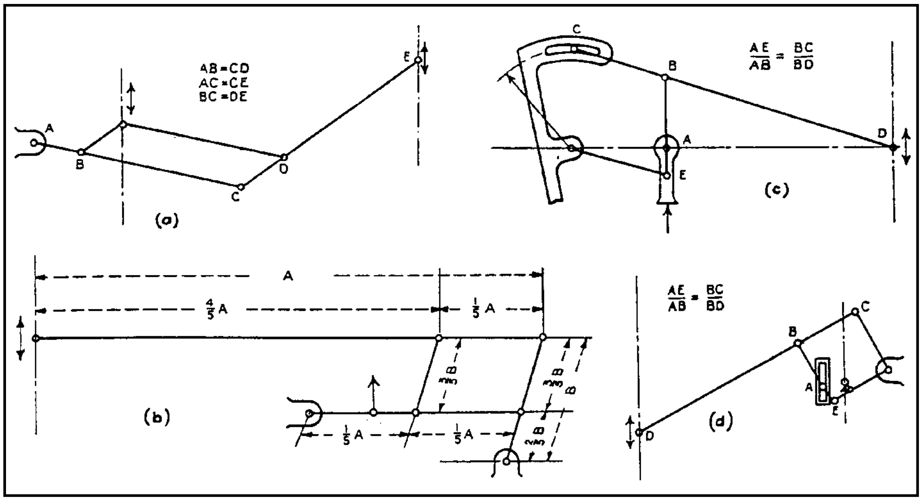


Fig. 4 (a), (b), (c), (d)—Pantograph linkages.

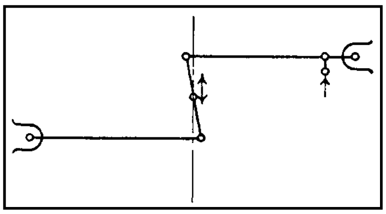


Fig. 5 Watt's linkage.

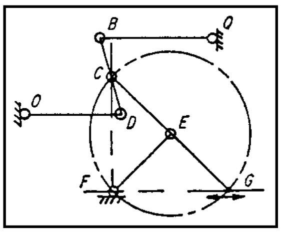


Fig. 6 Tchebicheff combination linkage.

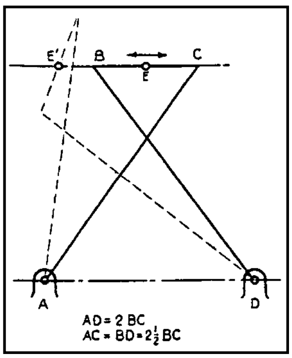


Fig. 7 Tchebicheff's linkage.

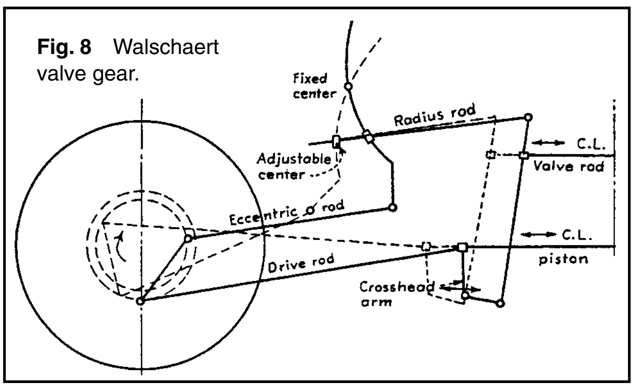


Fig. 8 Walschaert valve gear.

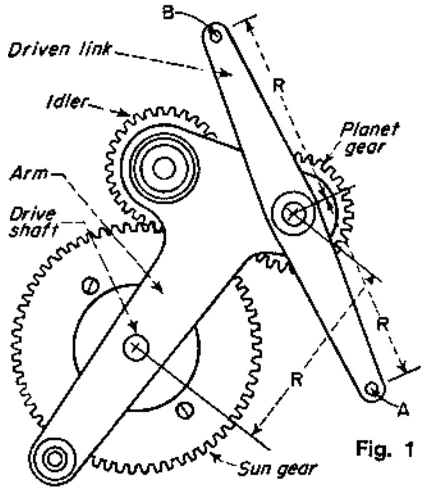


Fig. 1

Fig. 1 No linkages or guides are included in this modified hypocyclic drive which is relatively small in relation to the length of its stroke. The sun gear of pitch diameter *D* is stationary. The drive shaft, which turns the T-shaped arm, is concentric with this gear. The idler and planet gears, with pitch diameters of *D*/2, rotate freely on pivots in the arm extensions. The pitch diameter of the idler has no geometrical significance, although this gear does have an important mechanical function. It reverses the rotation of the planet gear, thus producing true hypocyclic motion with ordinary spur gears only. Such an arrangement occupies only about half as much space as does an equivalent mechanism containing an internal gear. The center distance *R* is the sum of *D*/2, *D*/4, and an arbitrary distance *d*, determined by specific applications. Points *A* and *B* on the driven link, which is fixed to the planet, describe straight-line paths through a stroke of 4*R*. All points between *A* and *B* trace ellipses, while the line *AB* envelopes an astroid.

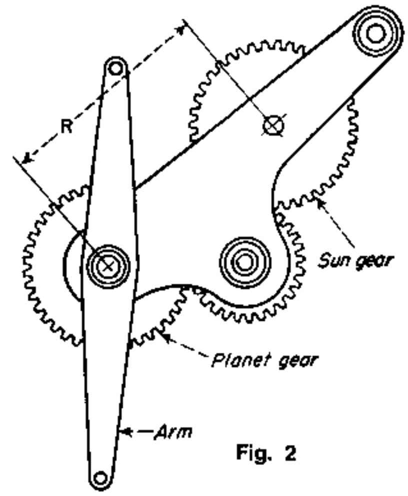


Fig. 2

Fig. 2 A slight modification of the mechanism in Fig. 1 will produce another type of useful motion. If the planet gear has the same diameter as that of the sun gear, the arm will remain parallel to itself throughout the complete cycle. All points on the arm will thereby describe circles of radius *R*. Here again, the position and diameter of the idler gear have no geometrical importance. This mechanism can be used, for example, to cross-perforate a uniformly moving paper web. The value for *R* is chosen so that $2\pi R$, or the circumference of the circle described by the needle carrier, equals the desired distance between successive lines of perforations. If the center distance *R* is made adjustable, the spacing of perforated lines can be varied as desired.

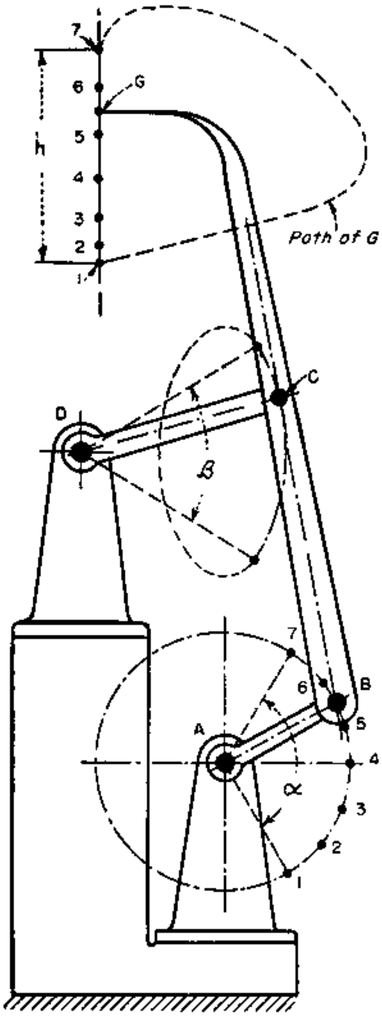


Fig. 3

Fig. 3 To describe a “D” curve, begin at the straight part of path G, and replace the oval arc of C with a circular arc that will set the length of link DC.

Fig. 4 This mechanism can act as a film-strip hook that will describe a nearly straight line. It will engage and disengage the film perforation in a direction approximately normal to the film. Slight changes in the shape of the guiding slot *f* permit the shape of the output curve and the velocity diagram to be varied.

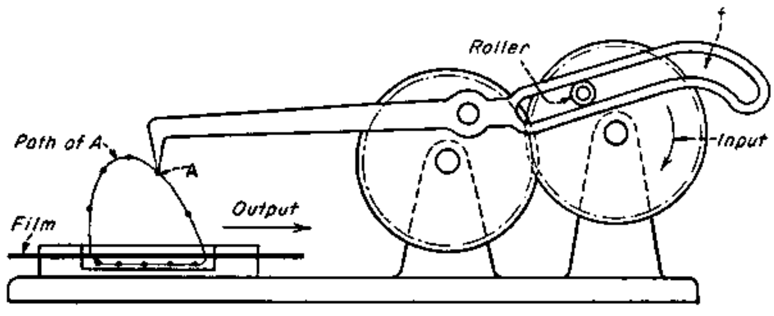


Fig. 4

FIVE CARDAN-GEAR MECHANISMS

These gearing arrangements convert rotary into straight-line motion, without the need for slideways.

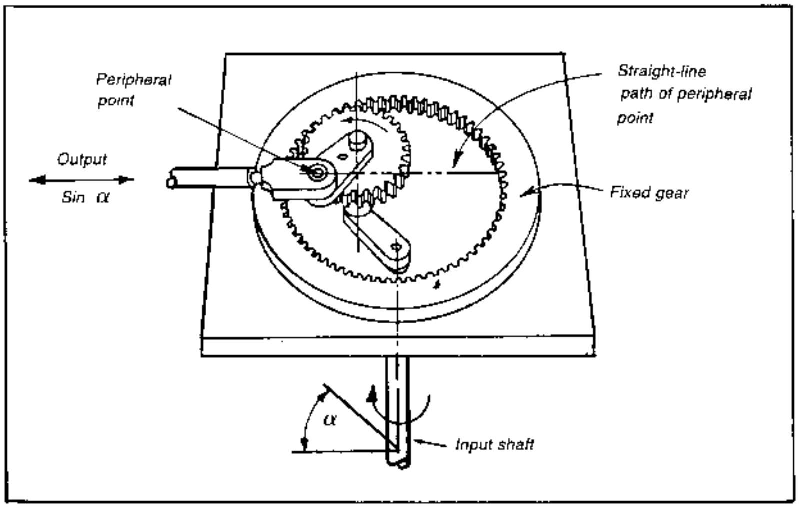


Fig. 1 Cardan gearing works on the principle that any point on the periphery of a circle rolling on the inside of another circle describes, in general, a hypocycloid. This curve degenerates into a true straight line (diameter of the larger circle) if the diameters of both circles are in the ratio of 1:2. The rotation of the input shaft causes a small gear to roll around the inside of the fixed gear. A pin located on the pitch circle of the small gear describes a straight line. Its linear displacement is proportional to the theoretically true sine or cosine of the angel through which the input shaft is rotated.

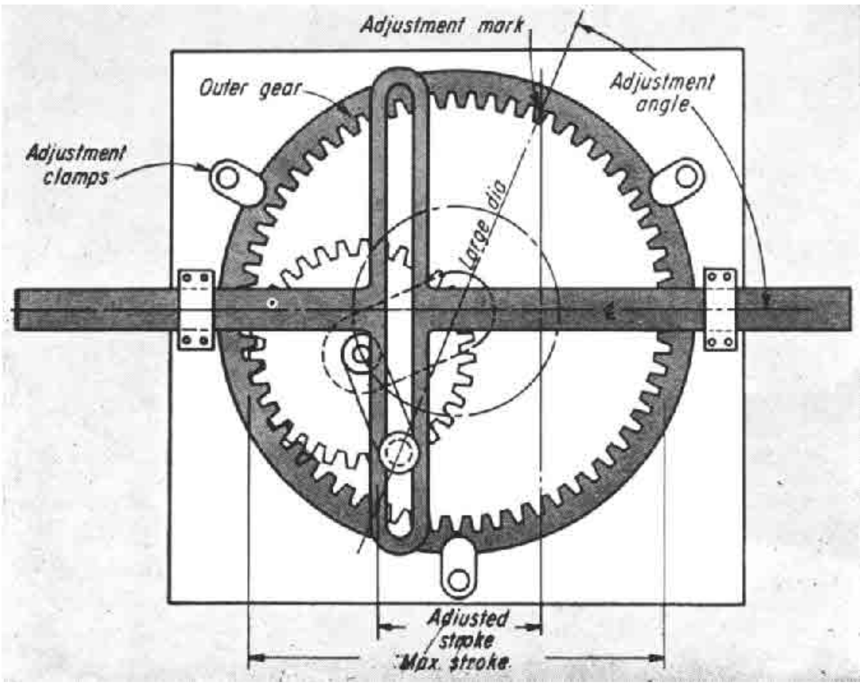


Fig. 2 Cardan gearing and a Scotch yoke in combination provide an adjustable stroke. The angular position of the outer gear is adjustable. The adjusted stroke equals the projection of the large diameter, along which the drive pin travels, on the Scotch-yoke's centerline. The yoke motion is simple harmonic.

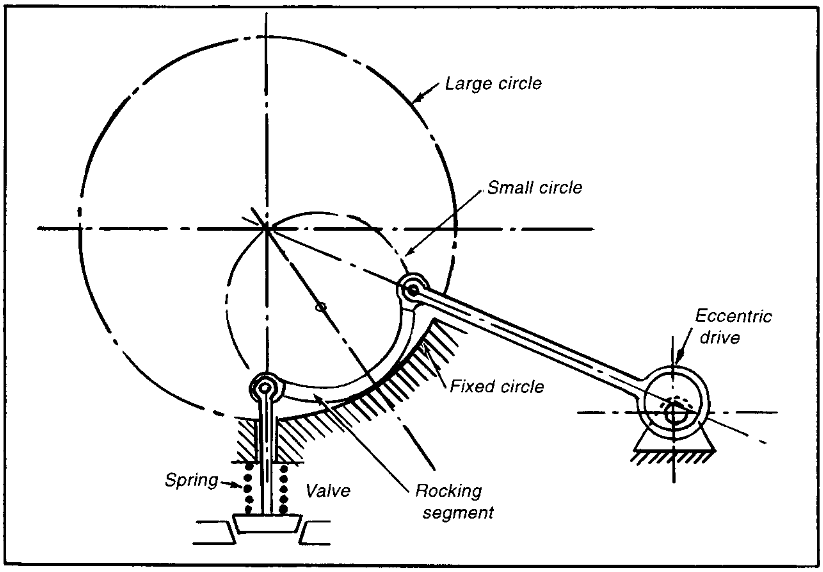


Fig. 3 A valve drive demonstrates how the Cardan principle can be applied. A segment of the smaller circle rocks back and forth on a circular segment whose radius is twice as large. The input and output rods are each attached to points on the small circle. Both these points describe straight lines. The guide of the valve rod prevents the rocking member from slipping.

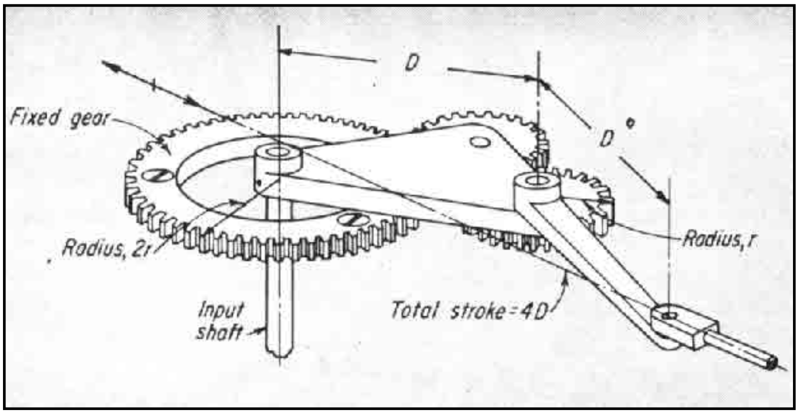
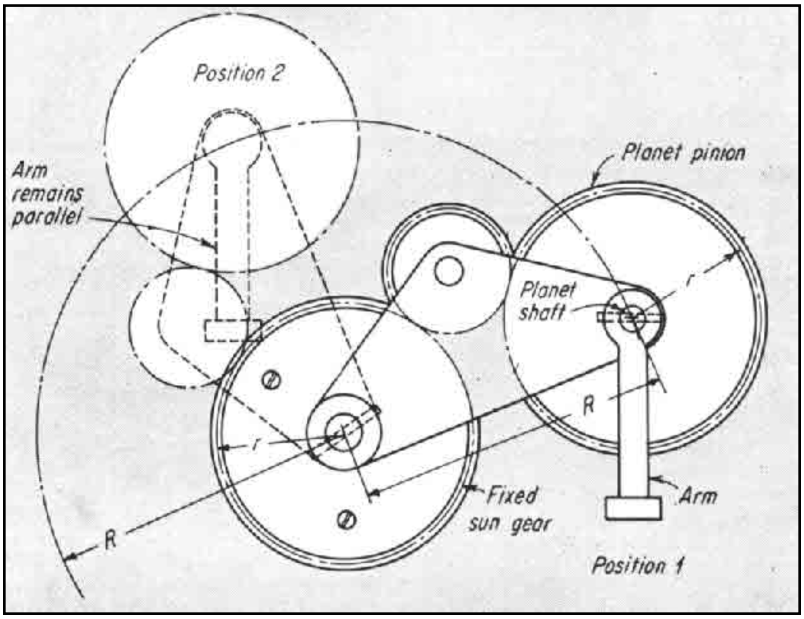


Fig. 4 A simplified Cardan mechanism eliminates the need for the relatively expensive internal gear. Here, only spur gears are used, and the basic requirements must be met, i.e., the 1:2 ratio and the proper direction of rotation. The rotation requirement is met by introducing an idler gear of appropriate size. This drive delivers a large stroke for the comparative size of its gears.

Fig. 5 A rearrangement of gearing in the simplified Cardan mechanism results in another useful motion. If the fixed sun gear and planet pinion are in the ratio of 1:1, an arm fixed to the planet shaft will stay parallel to itself during rotation, while any point on the arm describes a circle of radius *R*. When arranged in conjugate pairs, the mechanism can punch holes on moving webs of paper.



TEN WAYS TO CHANGE STRAIGHT-LINE DIRECTION

These arrangements of linkages, slides, friction drives, and gears can be the basis for many ingenious devices.

LINKAGES

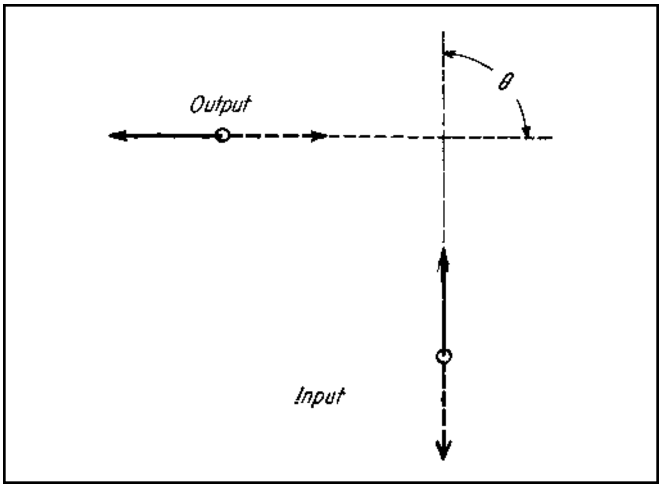


Fig. 1 Basic problem (θ is generally close to 90º).

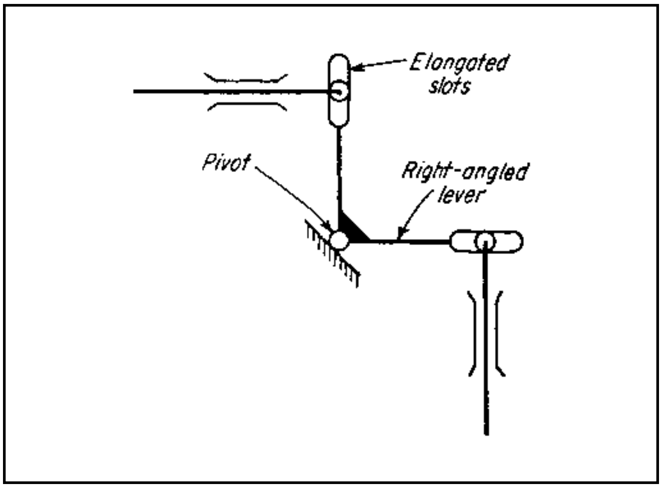


Fig. 2 Slotted lever.

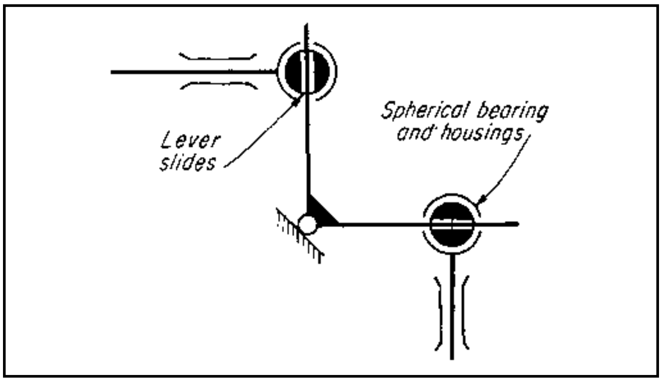


Fig. 3 Spherical bearings.

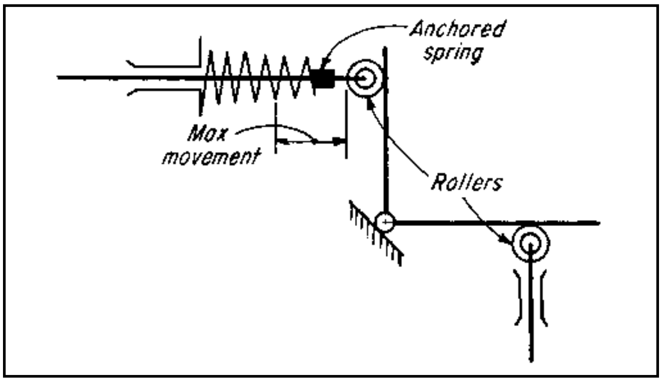


Fig. 4 Spring-loaded lever.

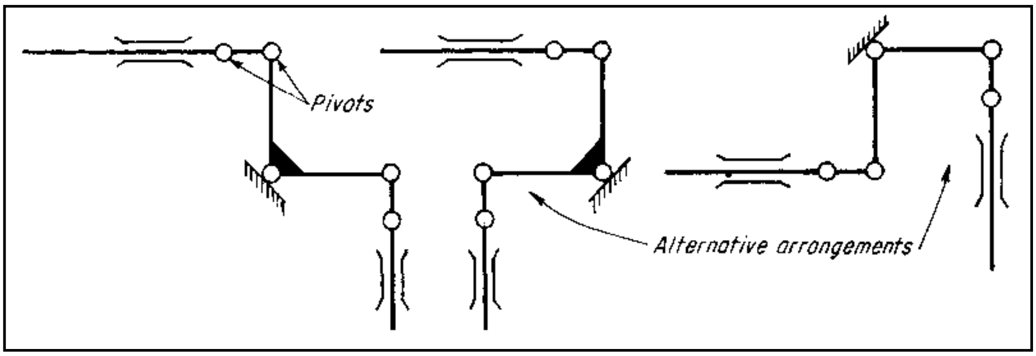


Fig. 5 Pivoted levers with alternative arrangements.

GUIDES

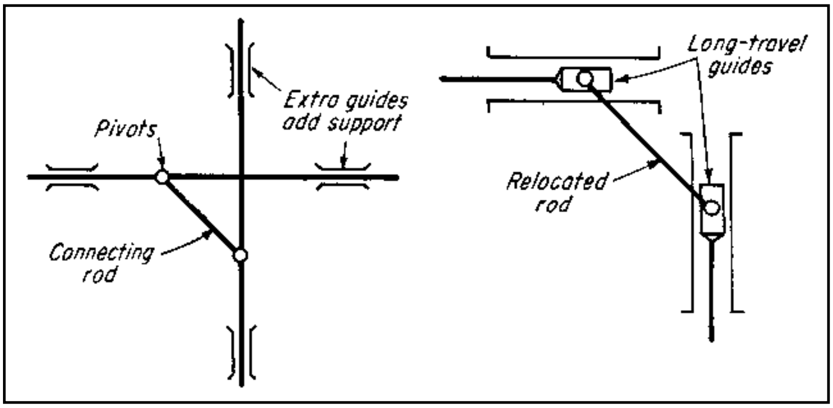


Fig. 6 Single connecting rod (left) is relocated (right) to eliminate the need for extra guides.

FRICTION DRIVES

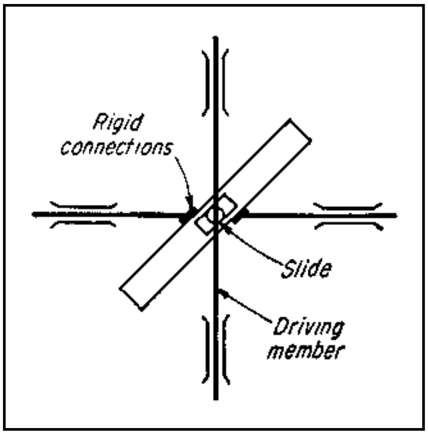


Fig. 7 Inclined bearing-guide.

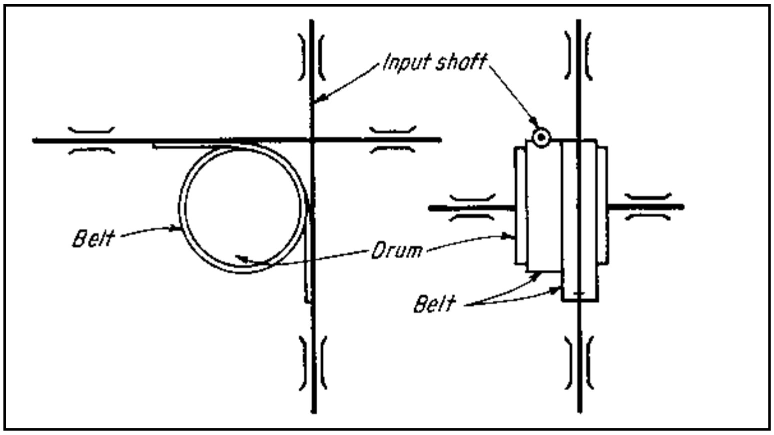


Fig. 8 A belt, steel band, or rope around the drum is fastened to the driving and driven members; sprocket-wheels and chain can replace the drum and belt.

GEARS

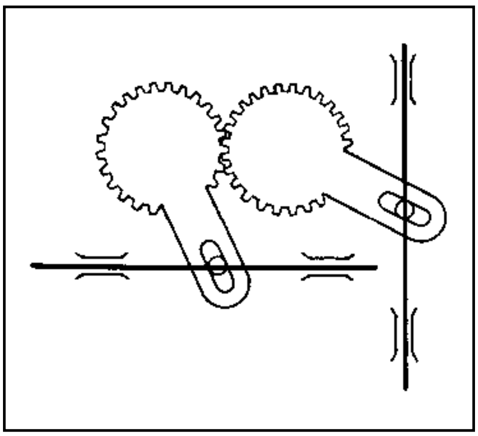


Fig. 9 Matching gear-segments.

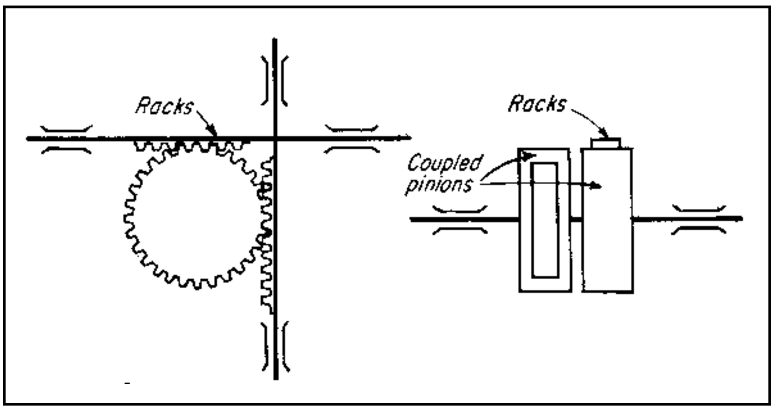


Fig. 10 Racks and coupled pinions (can be substituted as friction surfaces for a low-cost setup).

NINE MORE WAYS TO CHANGE STRAIGHT-LINE DIRECTION

These mechanisms, based on gears, cams, pistons, and solenoids, supplement ten similar arrangements employing linkages, slides, friction drives, and gears.

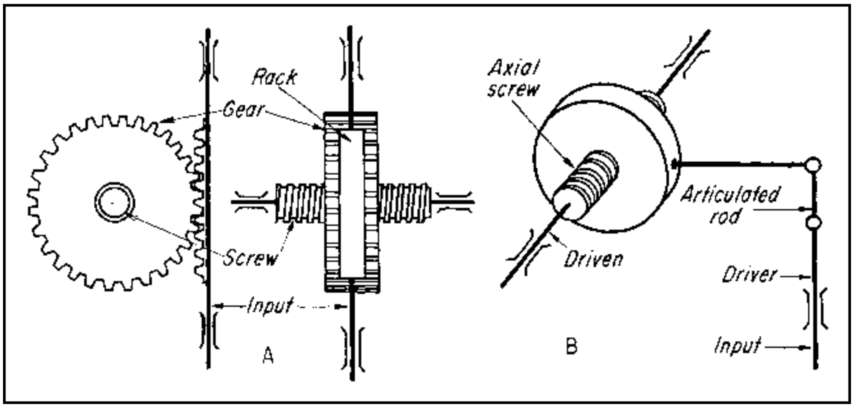


Fig. 1 An axial screw with a rack-actuated gear (A) and an articulated driving rod (B) are both irreversible movements, i.e., the driver must always drive.

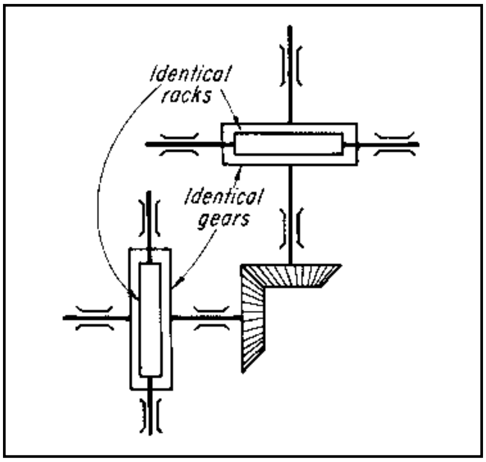


Fig. 2 A rack-actuated gear with associated bevel gears is reversible.

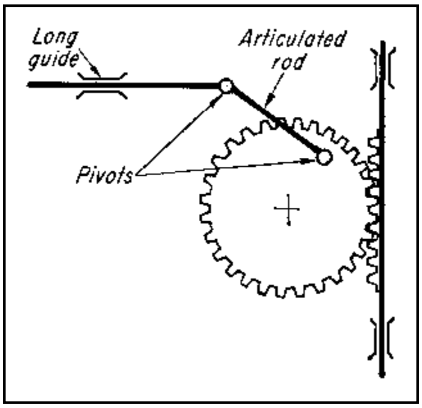


Fig. 3 An articulated rod on a crank-type gear with a rack driver. Its action is restricted to comparatively short movements.

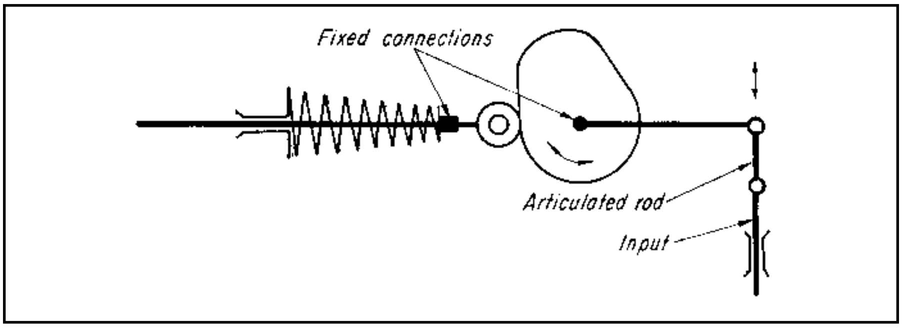


Fig. 4 A cam and spring-loaded follower allows an input/output ratio to be varied according to cam rise. The movement is usually irreversible.

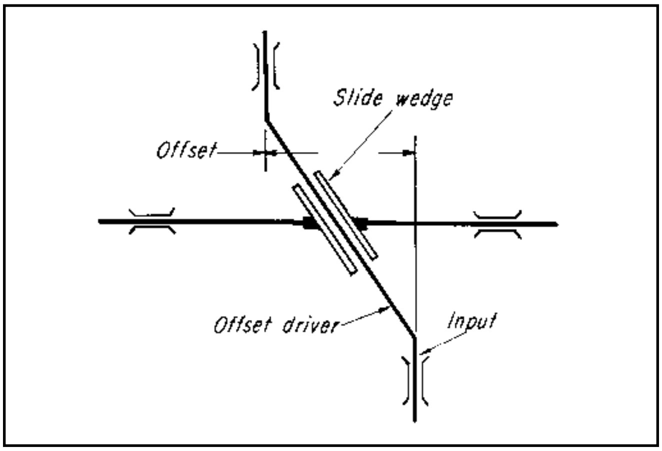


Fig. 5 An offset driver actuates a driven member by wedge action. Lubrication and materials with a low coefficient of friction permit the offset to be maximized.

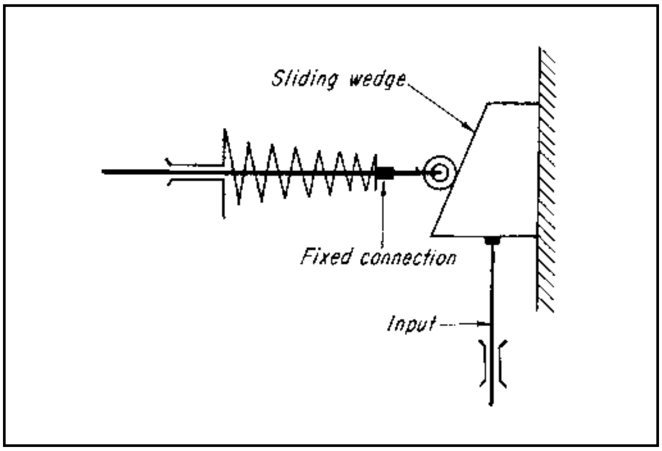


Fig. 6 A sliding wedge is similar to an offset driver but it requires a spring-loaded follower; also, low friction is less critical with a roller follower.

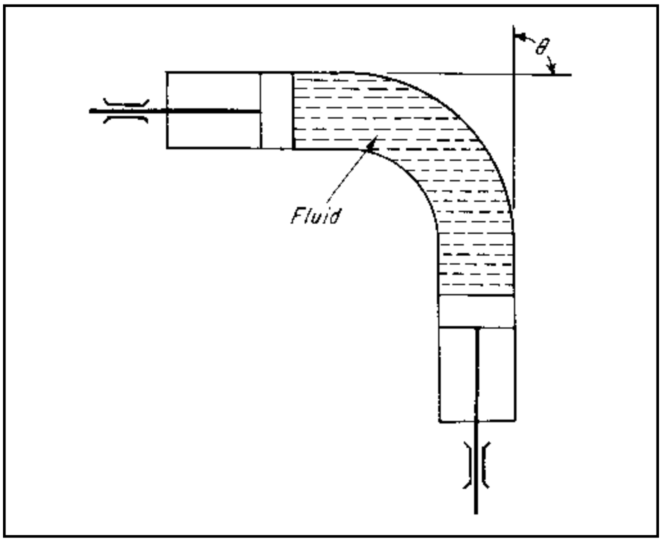


Fig. 7 A fluid coupling allows motion to be transmitted through any angle. Leak problems and accurate piston-fitting can make this method more expensive than it appears to be. Also, although the action is reversible, it must always be compressive for the best results.

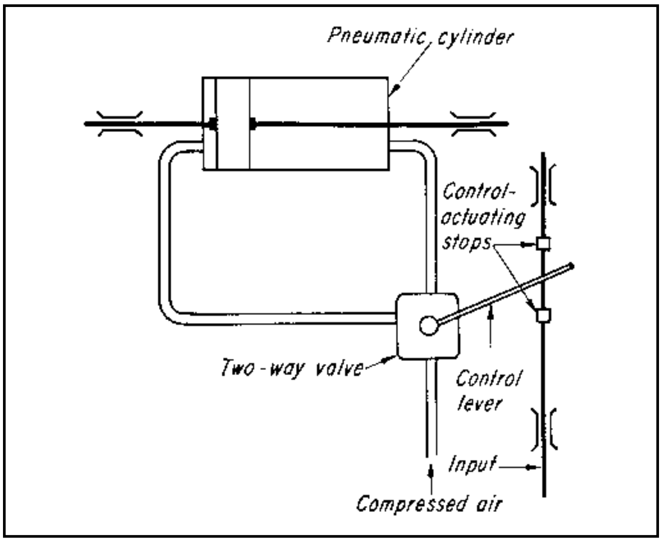


Fig. 8 A pneumatic system with a two-way valve is ideal when only two extreme positions are required. The action is irreversible. The speed of a driven member can be adjusted by controlling the input of air to the cylinder.

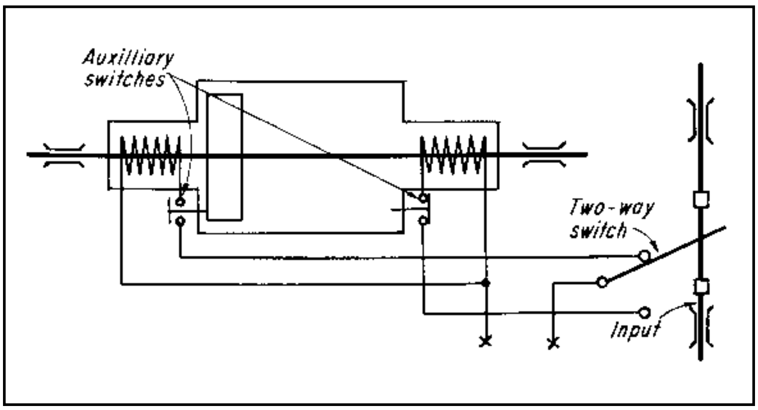


Fig. 9 Solenoids and a two-way switch are organized as an analogy of a pneumatic system. Contact with the energized solenoid is broken at the end of each stroke. The action is irreversible.

LINKAGES FOR ACCELERATING AND DECELERATING LINEAR STROKES

When ordinary rotary cams cannot be conveniently applied, the mechanisms presented here, or adaptations of them, offer a variety of interesting possibilities for obtaining either acceleration or deceleration, or both.

Fig. 1 A slide block with a pinion and shaft and a pin for link B reciprocates at a constant rate. The pinion has a crankpin for mounting link D, and it also engages a stationary rack. The pinion can make one complete revolution at each forward stroke of the slide block and another as the slide block returns in the opposite direction. However, if the slide block is not moved through its normal travel range, the pinion turns only a fraction of a revolution. The mechanism can be made variable by making the connection link for F adjustable along the length of the element that connects links B and D. Alternatively, the crankpin for link D can be made adjustable along the radius of the pinion, or both the connection link and the crankpin can be made adjustable.

Fig. 2 A drive rod, reciprocating at a constant rate, rocks link BC about a pivot on a stationary block. A toggle between arm B and the stationary block contacts an abutment. Motion of the drive rod through the toggle causes deceleration of driven link B. As the drive rod moves toward the right, the toggle is actuated by encountering the abutment. The slotted link BC slides on its pivot while turning. This lengthens arm B and shortens arm C of link BC. The result is deceleration of the driven link. The toggle is returned by a spring (not shown) on the return stroke, and its effect is to accelerate the driven link on its return stroke.

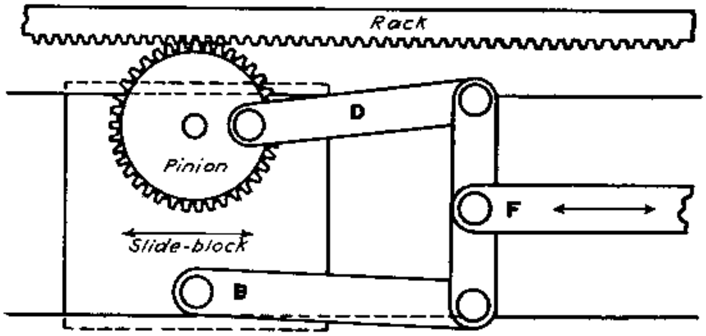


Fig. 1

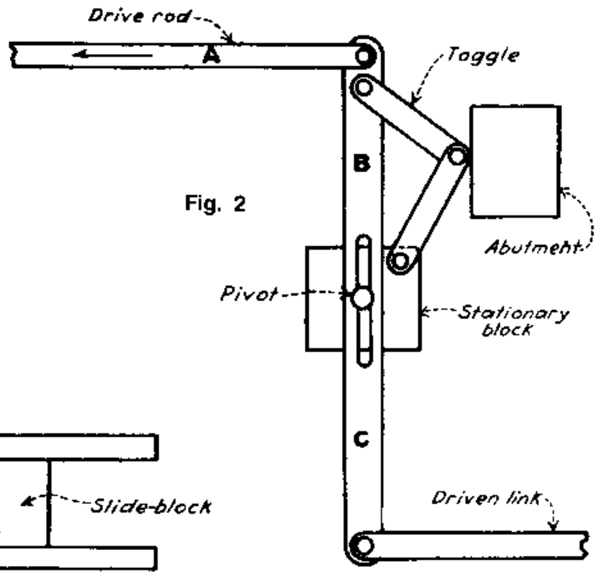


Fig. 2

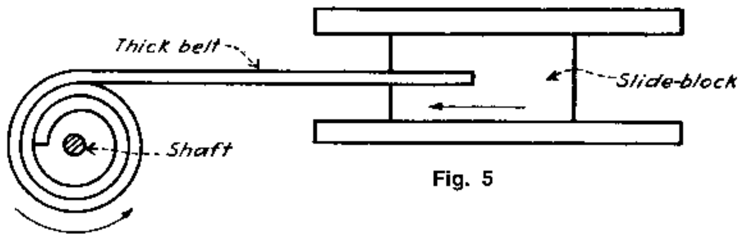


Fig. 5

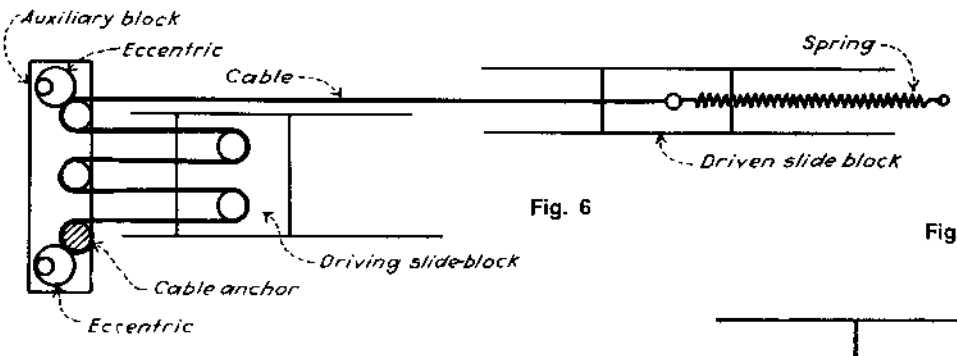


Fig. 6

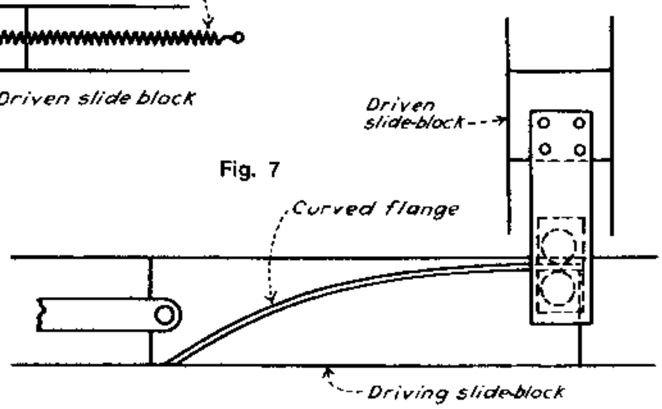


Fig. 7

Fig. 3 The same direction of travel for both the drive rod and the drive link is provided by the variation of the Fig. 2 mechanism. Here, acceleration is in the direction of the arrows, and deceleration occurs on the return stroke. The effect of acceleration decreases as the toggle flattens.

Fig. 4 A bellcrank motion is accelerated as the rollers are spread apart by a curved member on the end of the drive rod, thereby accelerating the motion of the slide block. The driven elements must be returned by spring to close the system.

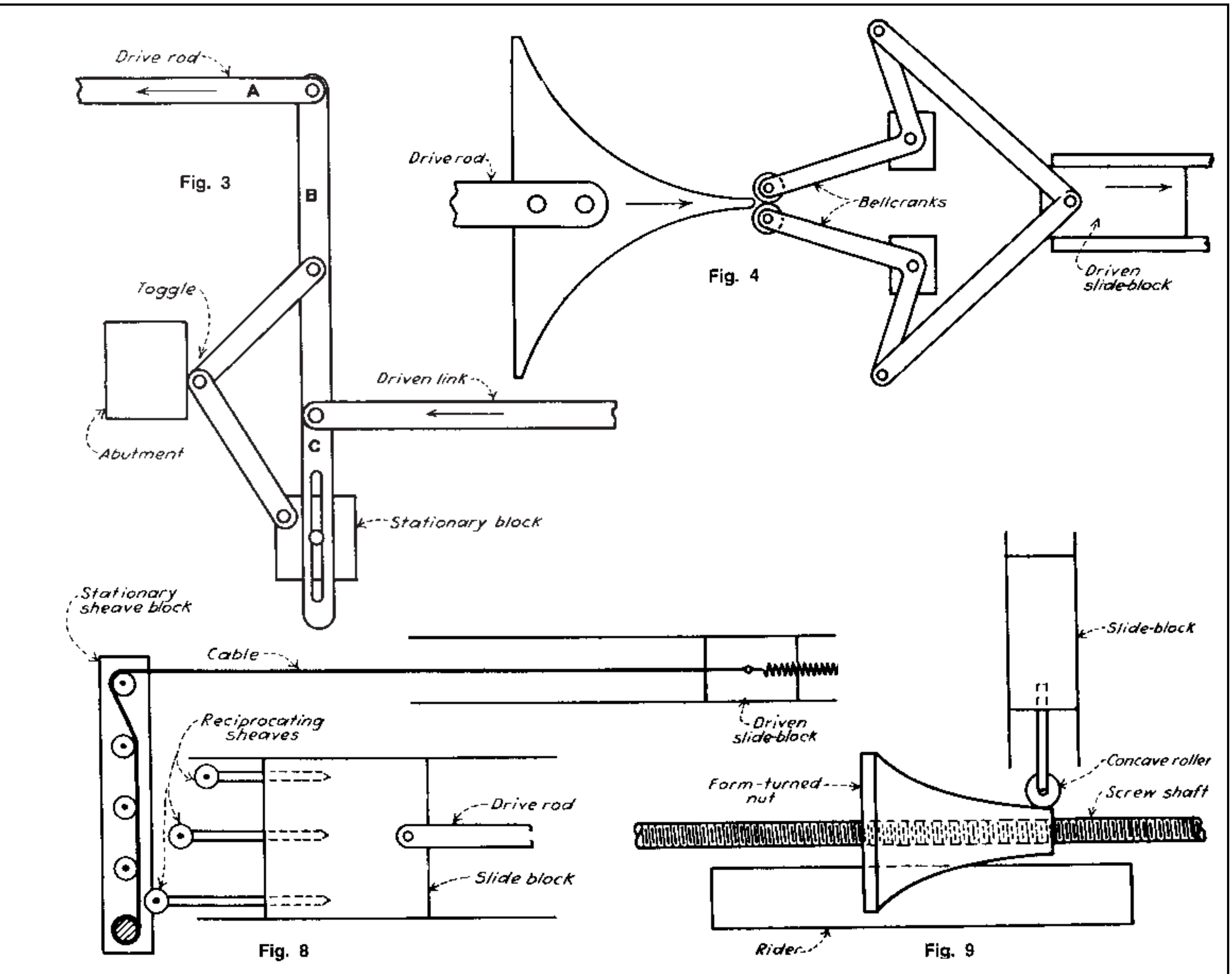
Fig. 5 A constant-speed shaft winds up a thick belt or similar flexible connecting member, and its effective increase in radius causes the slide block to accelerate. It must be returned by a spring or weight on its reversal.

Fig. 6 An auxiliary block that carries sheaves for a cable which runs between the driving and driven slide block is mounted on two synchronized eccentrics. The motion of the driven block is equal to the length of the cable paid out over the sheaves, resulting from the additive motions of the driving and auxiliary blocks.

Fig. 7 A curved flange on the driving slide block is straddled by rollers that are pivotally mounted in a member connected to the driven slide block. The flange can be curved to give the desired acceleration or deceleration, and the mechanism returns by itself.

Fig. 8 The stepped acceleration of the driven block is accomplished as each of the three reciprocating sheaves progressively engages the cable. When the third acceleration step is reached, the driven slide block moves six times faster than the drive rod.

Fig. 9 A form-turned nut, slotted to travel on a rider, is propelled by reversing its screw shaft, thus moving the concave roller up and down to accelerate or decelerate the slide block.



LINKAGES FOR MULTIPLYING SHORT MOTIONS

The accompanying sketches show typical linkages for multiplying short linear motions, usually converting the linear motion into rotation. Although the particular mechanisms shown are designed to multiply the movements of diaphragms or bellows, the same or similar constructions have possible applications wherever it is required to obtain greatly multiplied motions. These transmissions depend on cams, sector gears and pinions, levers and cranks, cord or chain, spiral or screw feed, magnetic attraction, or combinations of these mechanical elements.

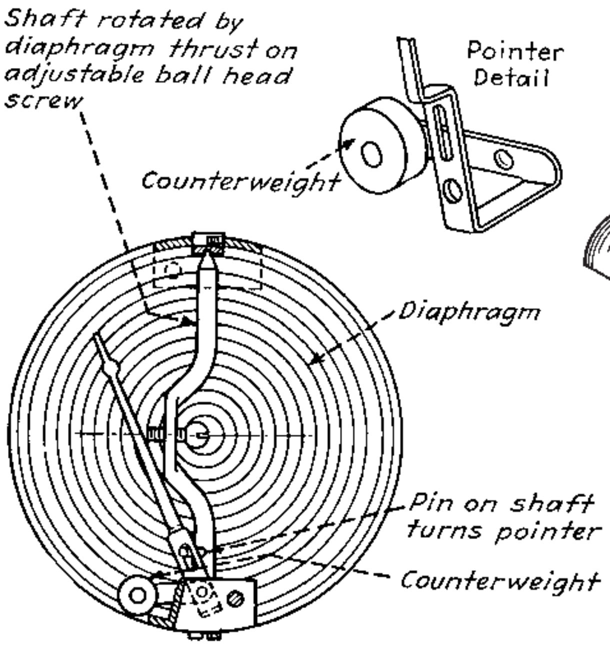


Fig. 1 A lever-type transmission in a pressure gage.

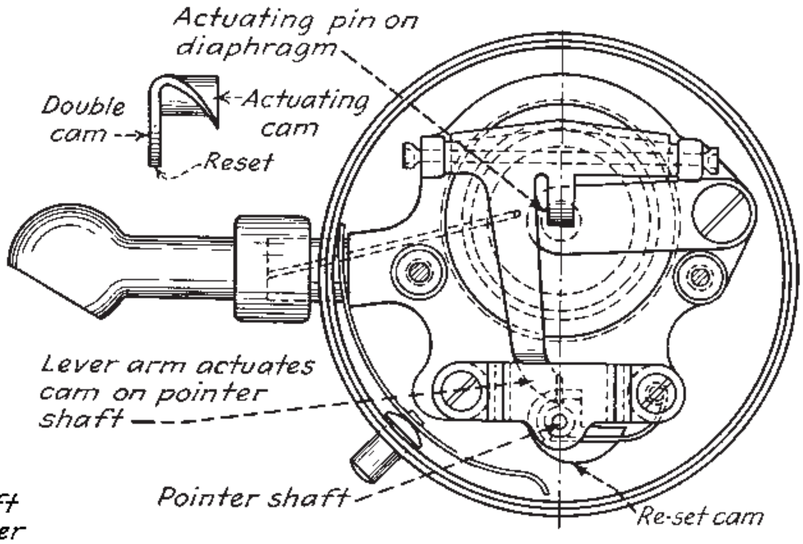


Fig. 2 A lever and cam drive for a tire gage.

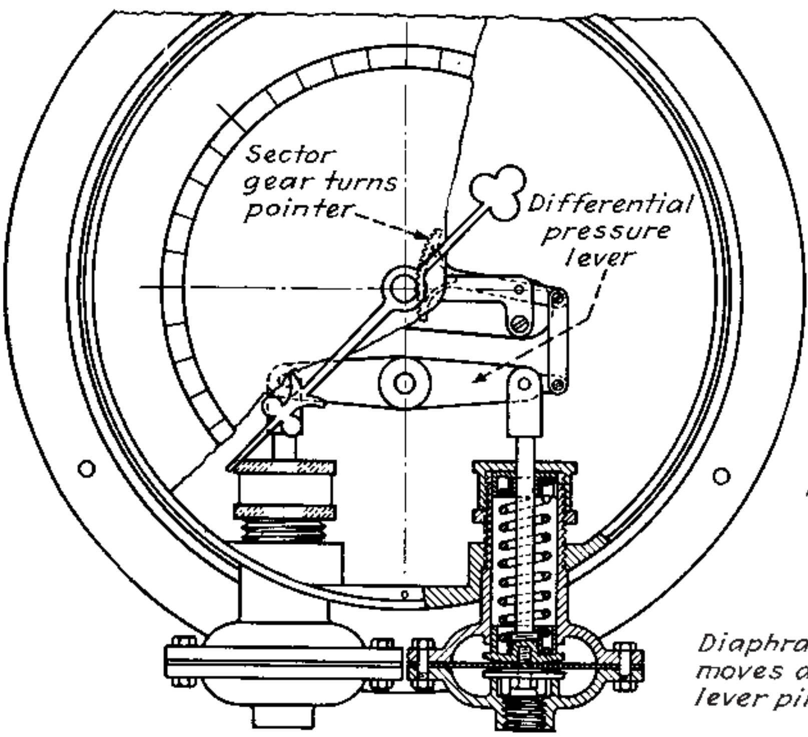


Fig. 3 A lever and sector gear in a differential pressure gage.

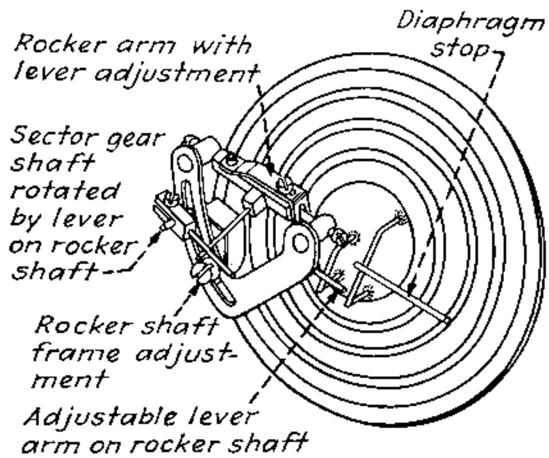


Fig. 4 A sector gear drive for an aircraft air-speed indicator.

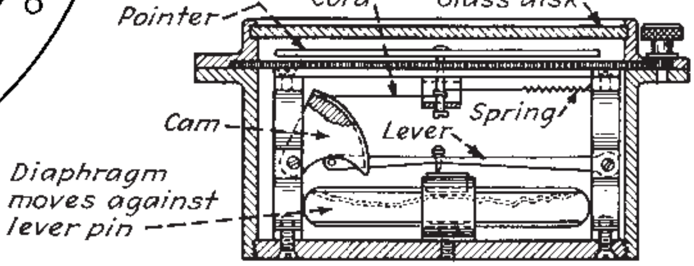


Fig. 5 A lever, cam, and cord transmission in a barometer.

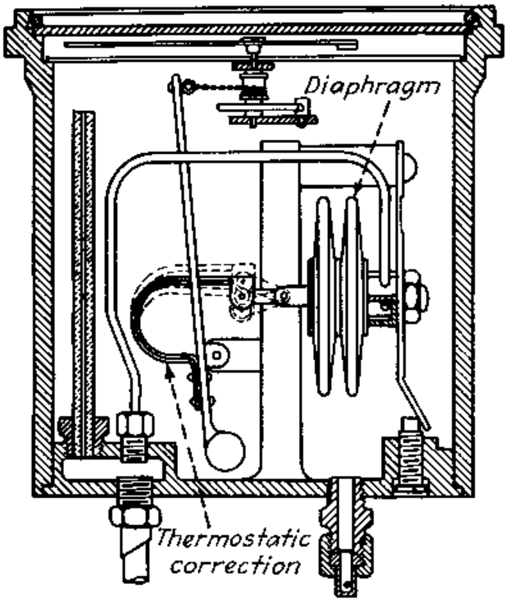


Fig. 6 A link and chain transmission for an aircraft rate of climb instrument.

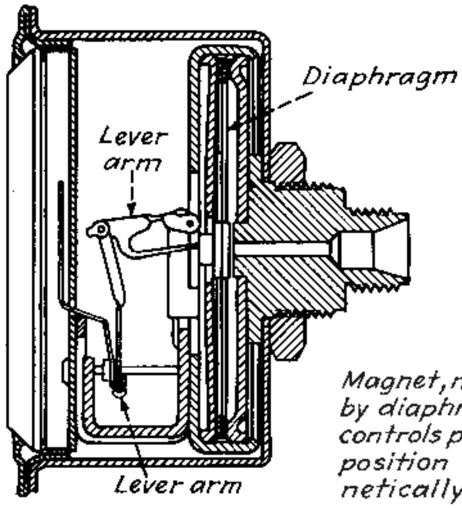


Fig. 7 A lever system in an automobile gasoline tank.

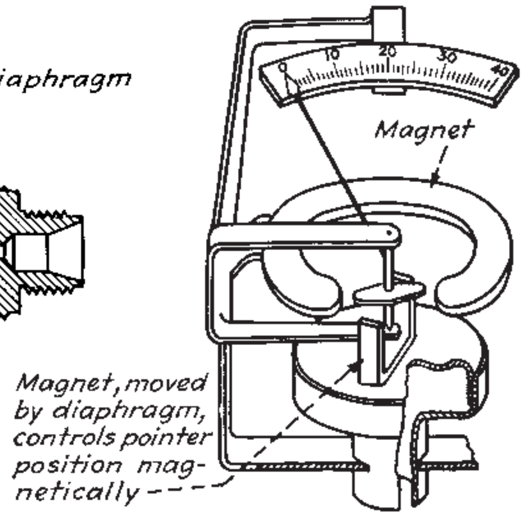


Fig. 8 Interfering magnetic fields for fluid pressure measurement.

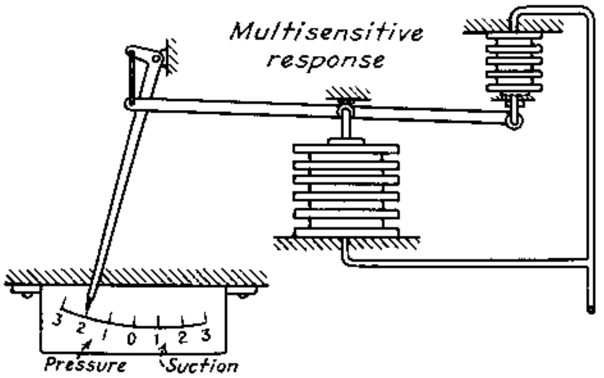


Fig. 9 A lever system for measuring atmospheric pressure variations.

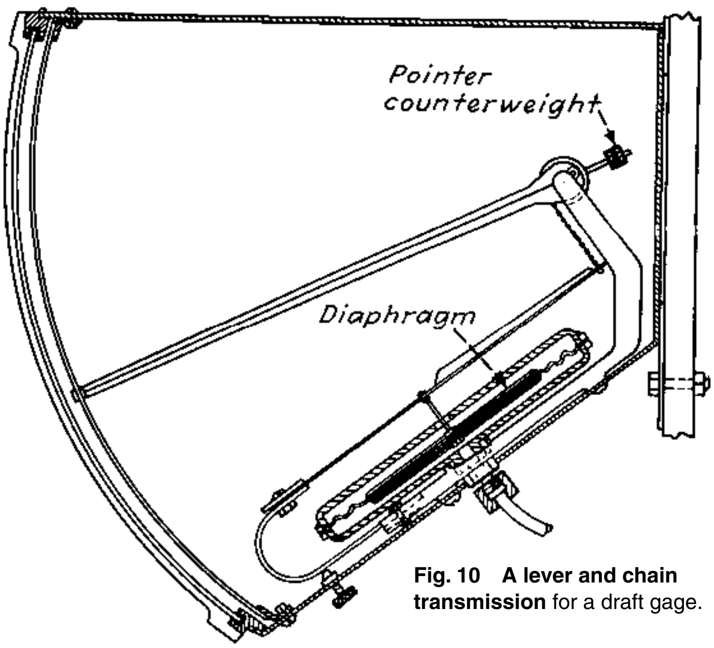


Fig. 10 A lever and chain transmission for a draft gage.

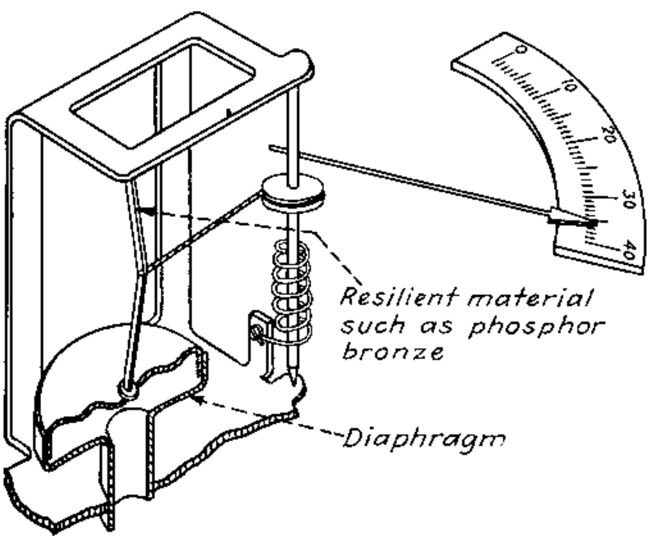


Fig. 11 A toggle and cord drive for a fluid pressure measuring instrument.

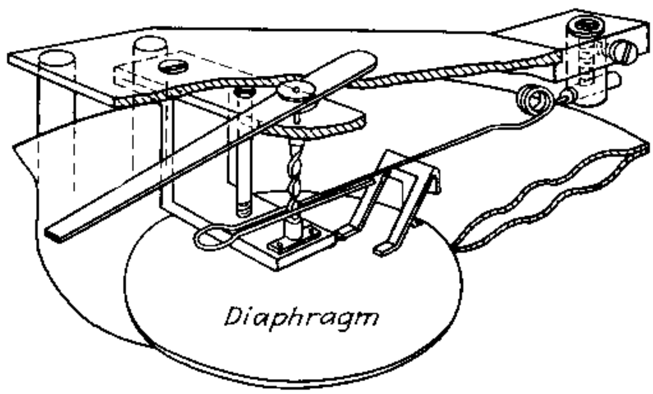
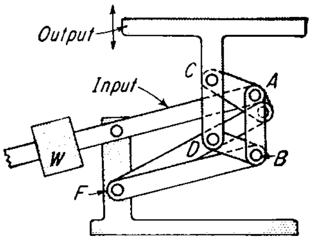


Fig. 12 A spiral feed transmission for a general purpose analog instrument.

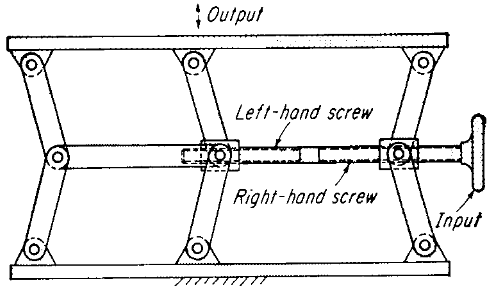
PARALLEL-LINK MECHANISMS

Eight-bar linkage



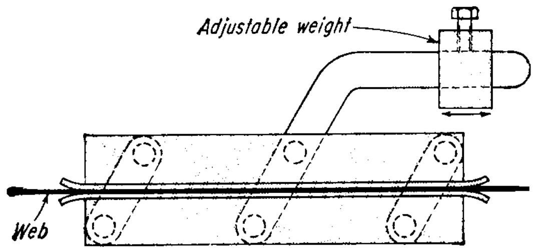
Link AB in this arrangement will always be parallel to EF, and link CD will always be parallel to AB. Hence CD will always be parallel to EF. Also, the linkages are so proportioned that point C moves in an approximately straight line. The final result is that the output plate will remain horizontal while moving almost straight up and down. The weight permitted this device to function as a disappearing platform in a theater stage.

Double-handed screw mechanism



Turning the adjusting screw spreads or contracts the linkage pairs to raise or lower the table. Six parallel links are shown, but the mechanism can be build with four, eight, or more links.

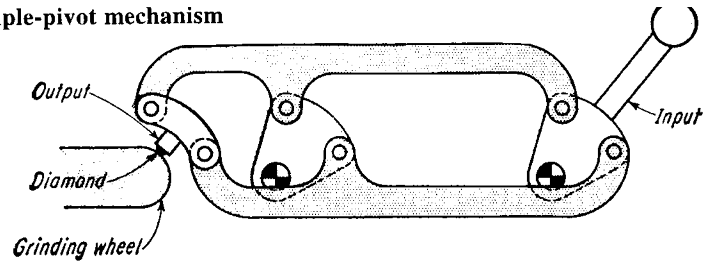
Tensioning mechanism



A simple parallel-link mechanism that produces tension in webs, wires, tapes, and strip steels. Adjusting the weight varies the drag on the material.

Triple-pivot mechanism

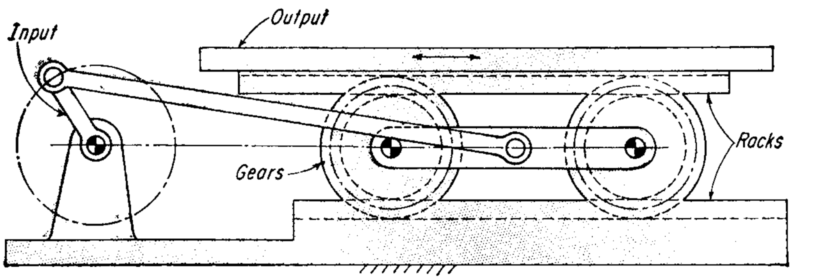
Two triangular plates pivot around fixed points on a machine frame. The output point describes a circular-arc curve. It can round out the cutting surfaces of grinding wheels.



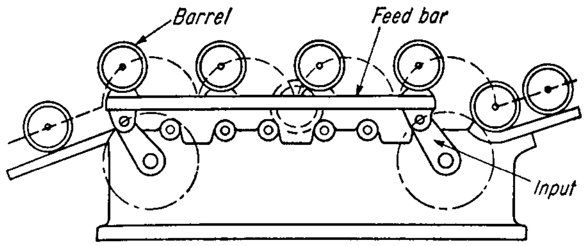
STROKE MULTIPLIER

Reciprocating-table drive

Two gears rolling on a stationary bottom rack drive the movable top rack, which is attached to a printing table. When the input crank rotates, the table will move out to a distance of four times the crank length.

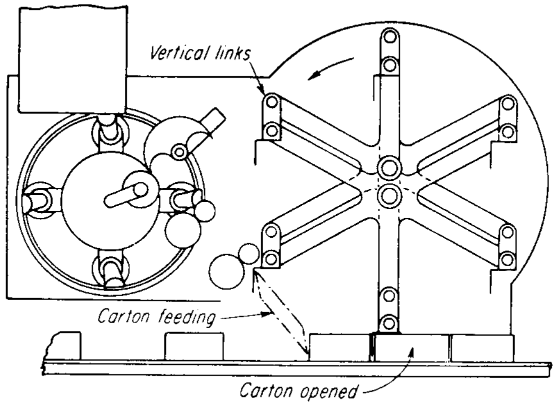


Parallel-link feeder



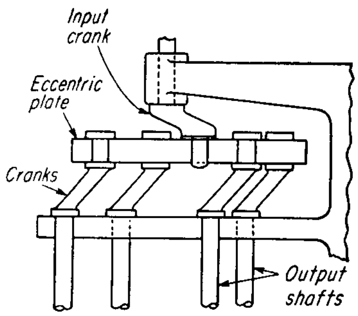
One of the cranks is the input, and the other follows to keep the feeding bar horizontal. The feeder can move barrels from station to station.

Parallelogram linkage



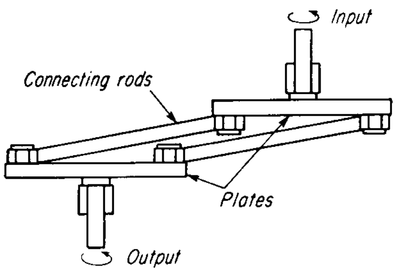
All seven short links are kept in a vertical position while rotating. The center link is the driver. This particular machine feeds and opens cartons, but the mechanism will work in many other applications.

Parallel-link driller



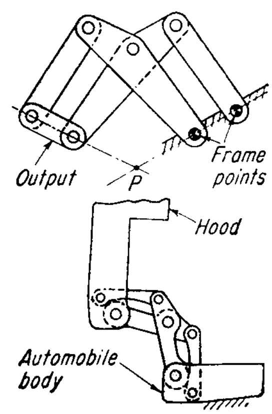
This parallel-link driller powers a group of shafts. The input crank drives the eccentric plate. This, in turn, rotates the output cranks that have the same length at the same speed. Gears would occupy more room between the shafts.

Parallel-plate driver



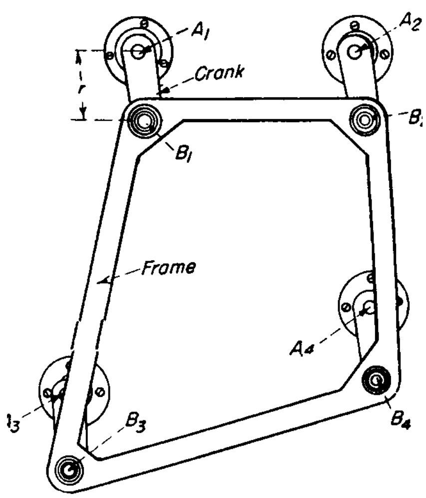
The input and output shafts of this parallel-plate driver rotate with the same angular relationship. The positions of the shafts, however, can vary to suit other requirements without affecting the input-output relationship between the shafts.

Curve-scribing mechanism



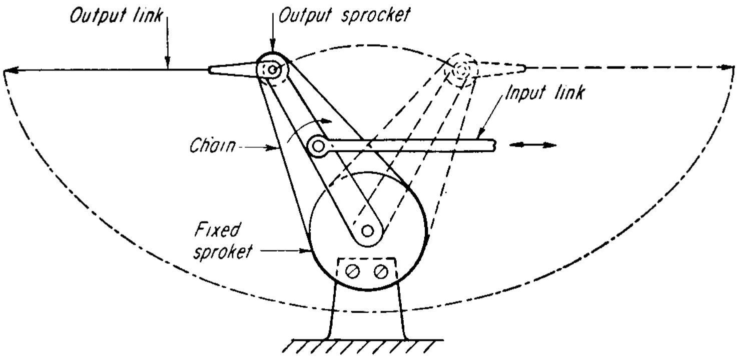
The output link rotates so that it appears to revolve around a point moving in space (P). This avoids the need for hinges at distant or inaccessible spots. The mechanism is suitable for hinging the hoods of automobiles.

Parallel-link coupling



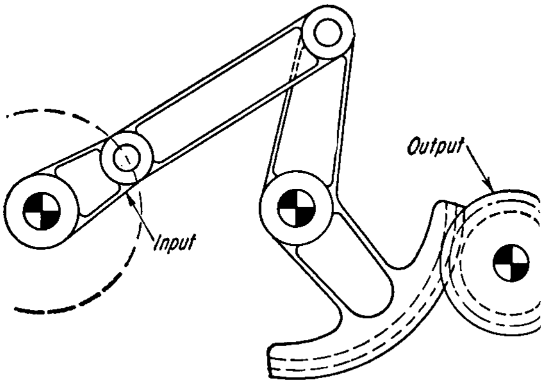
The absence of backlash makes this parallel-link coupling a precision, low-cost replacement for gear or chain drives that can also rotate parallel shafts. Any number of shafts greater than two can be driven from any one of the shafts, provided two conditions are fulfilled: (1) All cranks must have the same length r; and (2) the two polygons formed by the shafts A and frame pivot centers B must be identical. The main disadvantage of this mechanism is its dynamic unbalance, which limits the speed of rotation. To lessen the effect of the vibrations produced, the frame should be made as light as is consistent with strength requirements for the intended application.

Wide-angle oscillator



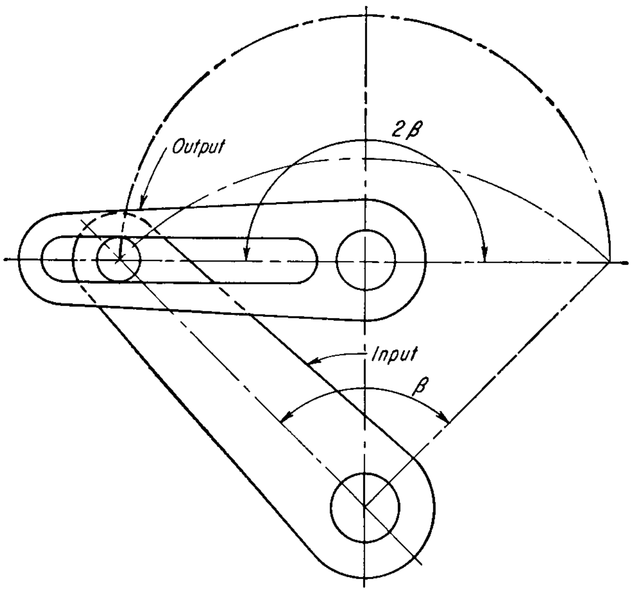
The motion of the input linkage in the diagram is converted into a wide-angle oscillation by the two sprockets and chain. An oscillation of 60º is converted into 180º oscillation.

Gear-sector drive



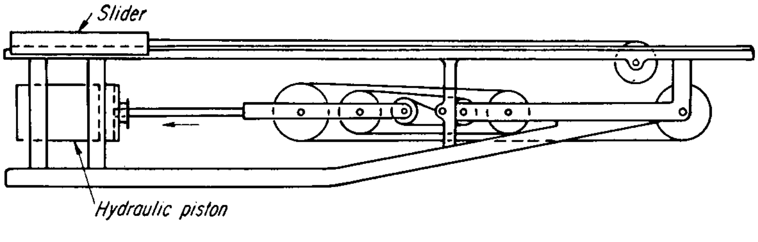
This is actually a four-bar linkage combined with a set of gears. A four-bar linkage usually obtains so more than about 120º of maximum oscillation. The gear segments multiply the oscillation in inverse proportion to the radii of the gears. For the proportions shown, the oscillation is boosted two and one-half times.

Angle-doubling drive



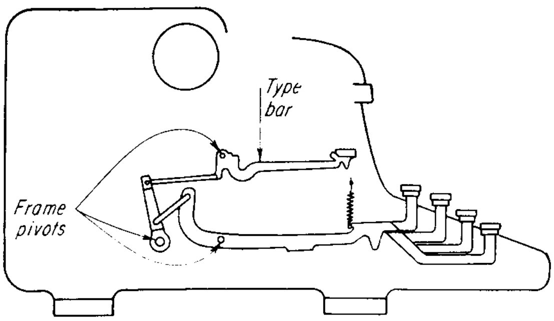
This angle-doubling drive will enlarge the oscillating motion β of one machine member into an output oscillation of 2β. If gears are employed, the direction of rotation cannot be the same unless an idler gear is installed. In that case, the centers of the input and output shafts cannot be too close. Rotating the input link clockwise causes the output to follow in a clockwise direction. For any set of link proportions, the distance between the shafts determines the gain in angle multiplication.

Pulley drive



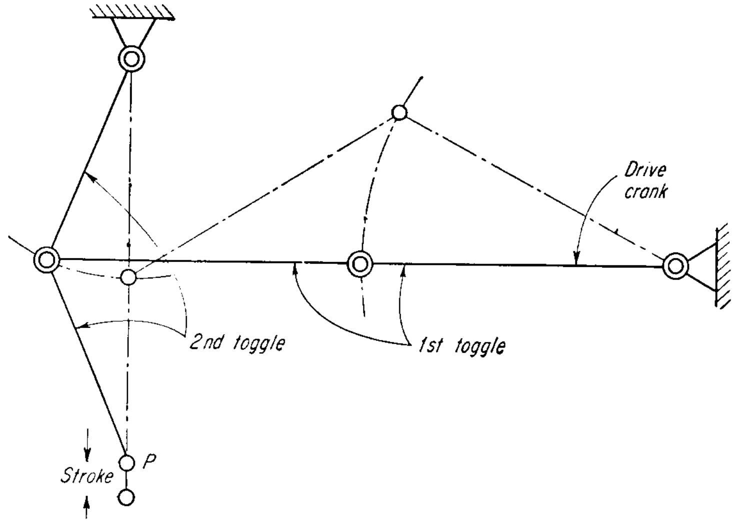
This pulley drive multiplies the stroke of a hydraulic piston, causing the slider to move rapidly to the right for catapulting objects.

Typewriter drive



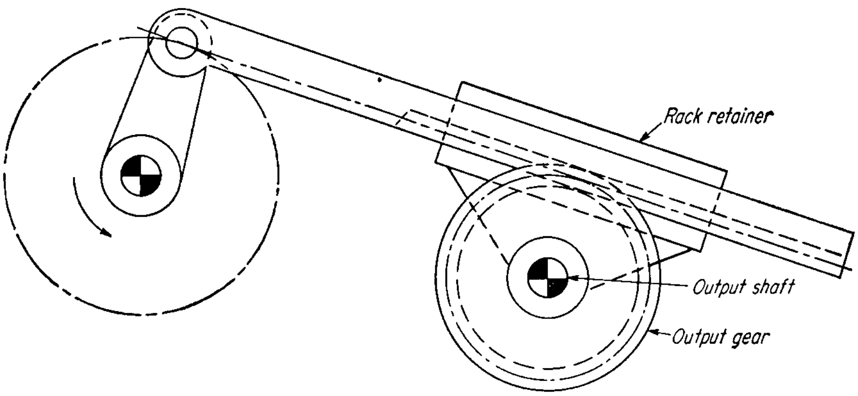
This drive multiplies the finger force of a typewriter, producing a strong hammer action at the roller from a light touch. There are three pivot points attached to the frame. The links are arranged so that the type bar can move in free flight after a key has been struck. The mechanism illustrated is actually two four-bar linkages in series. Some typewriters have as many as four four-bar linkages in a series.

Double-toggle puncher



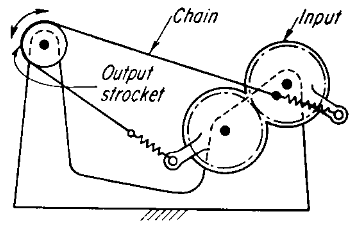
The first toggle of this puncher keeps point *P* in the raised position although its weight can exert a strong downward force (as in a heavy punch weight). When the drive crank rotates clockwise (e.g., driven by a reciprocating mechanism), the second toggle begins to straighten so as to create a strong punching force.

Gear-rack drive



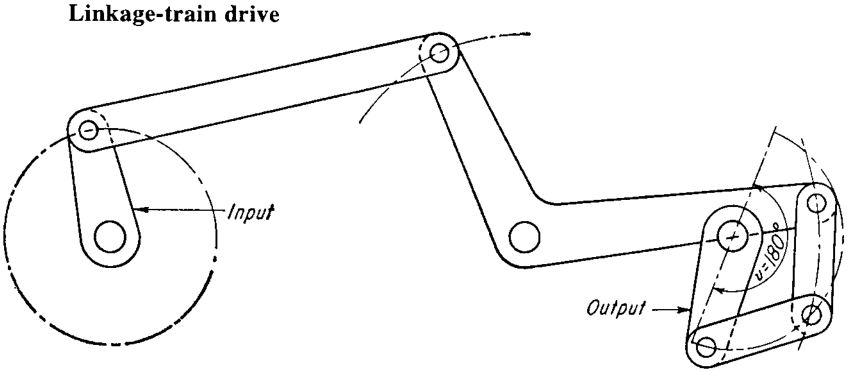
This drive mechanism converts the motion of an input crank into a much larger rotation of the output (from 30° to 360°). The crank drives the slider and gear rack, which in turn rotates the output gear.

Chain drive

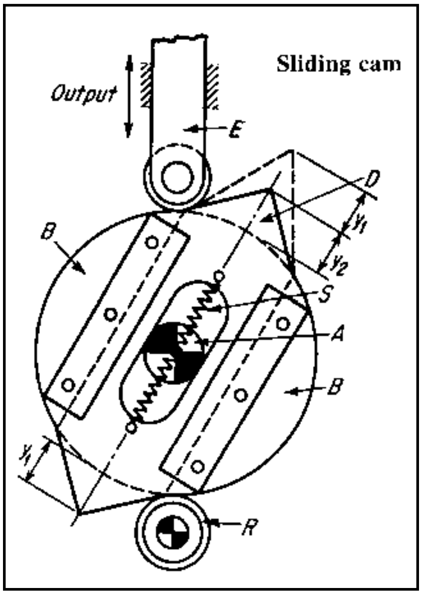


Springs and chains are attached to geared cranks of this drive to operate a sprocket output. Depending on the gear ratio, the output will produce a desired oscillation, e.g., two revolutions of output in each direction for each 360° of input.

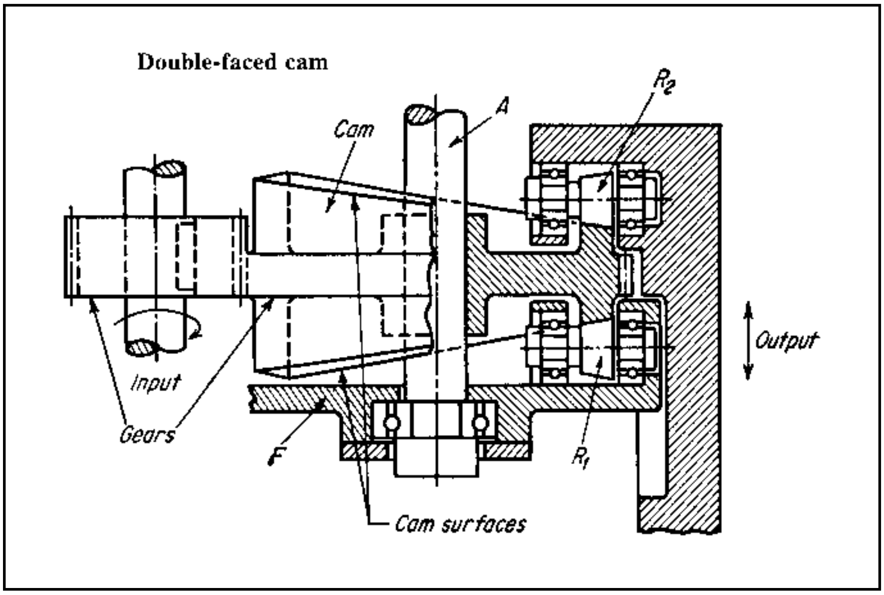
Linkage-train drive



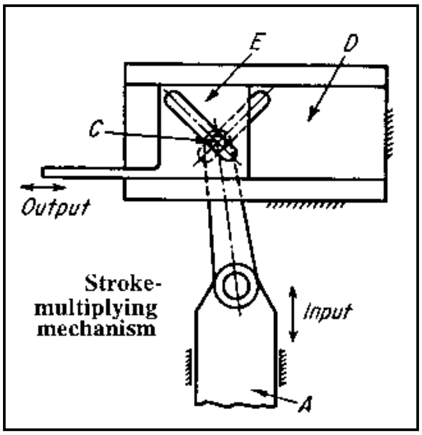
Arranging linkages in series on this drive can increase its angle of oscillation. In the version illustrated, the oscillating motion of the L-shaped rocker is the input for the second linkage. The final oscillation is 180°.



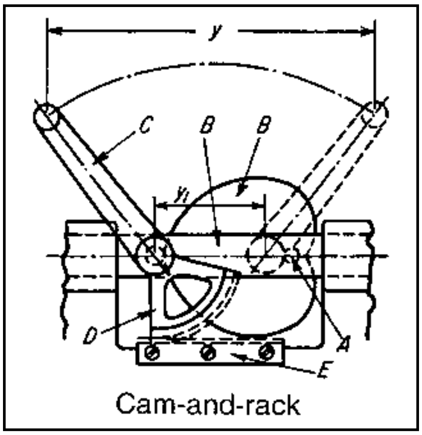
Sliding cam



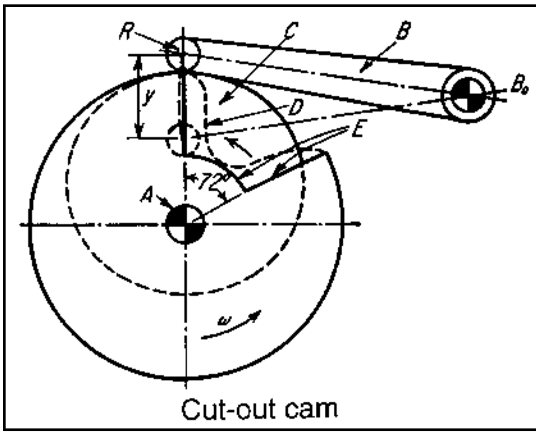
Double-faced cam



Stroke-multiplying mechanism



Cam-and-rack



Cut-out cam

When the pressure angles of stroke-amplifying mechanisms are too high to satisfy the design requirements, and it is undesirable to enlarge the cam size, certain devices can be installed to reduce the pressure angles:

Sliding cam—This mechanism is used on a wire-forming machine. Cam *D* has a pointed shape because of the special motion required for twisting wires. The machine operates at slow speeds, but the principle employed here is also applicable to high-speed cams.

The original stroke desired was ($y_1 + y_2$) but this results in a large pressure angle. The stroke therefore is reduced to y_2 on one side of the cam, and a rise of y_1 is added to the other side. Flanges *B* are attached to cam shaft *A*. Cam *D*, a rectangle with the two cam ends (shaded), is shifted upward as it cams off stationary roller *R* when the cam follower *E* is being cammed upward by the other end of cam *D*.

Stroke-multiplying mechanisms—This mechanism is used in power presses. The opposing slots, the first in a fixed member *D*, and the second in the movable slide *E*, multiply the motion of the input slide *A* driven by the cam. As *A* moves upward, *E* moves rapidly to the right.

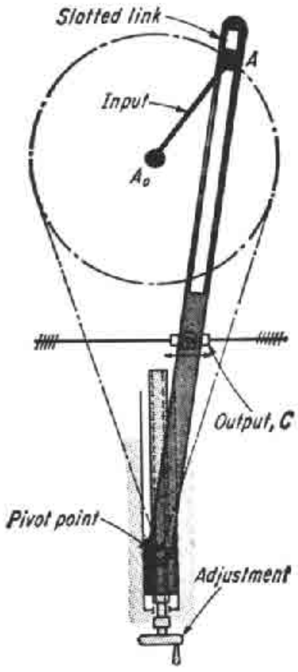
Double-faced cam—This mechanism doubles the stroke, hence reduces the pressure angles to one-half of their original values. Roller R_1 is stationary. When the cam rotates, its bottom surface lifts itself on R_1, while its top surface adds an additional motion to the movable roller R_2. The output is driven linearly by roller R_2 and thus is approximately the sum of the rise of both of these surfaces.

Cam-and-rack—This mechanism increases the throw of a lever. Cam *B* rotates around *A*. The roller follower travels at distances y_1; during this time, gear segment *D* rolls on rack *E*. Thus the output stroke of lever *C* is the sum of transmission and rotation, giving the magnified stoke *y*.

Cut-out cam—A rapid rise and fall within 72º was desired. This originally called for the cam contour, *D*, but produced severe pressure angles. The condition was improved by providing an additional cam *C*. This cam also rotates around the cam center *A*, but at five times the speed of cam *D* because of a 5:1 gearing arrangement (not shown). The original cam was then completely cut away for the 72º (see surfaces *E*). The desired motion, expanded over 360º (because 72º × 5 = 360º), is now designed into cam *C*. This results in the same pressure angle as would occur if the original cam rise occurred over 360º instead of 72º.

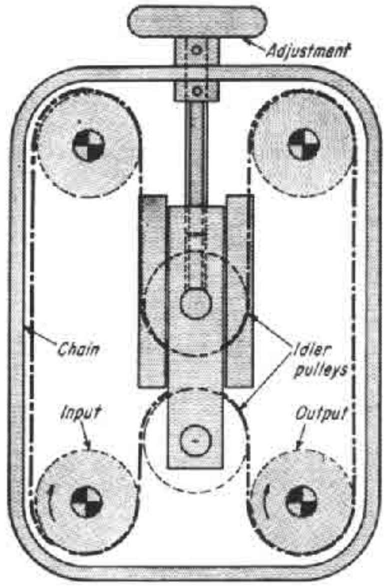
ADJUSTABLE-STROKE MECHANISMS

Adjustable-slider drive



Shifting the pivot point of this drive with the adjusting screw changes the stroke of the output rod.

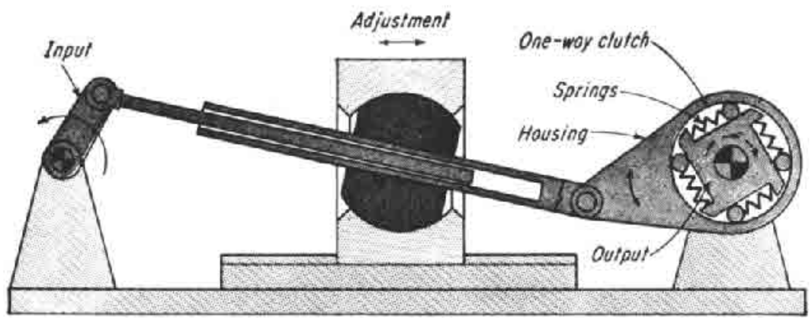
Adjustable-chain drive



Synchronization between input and output shafts of this drive is varied by shifting the two idler pulleys with the adjusting screw.

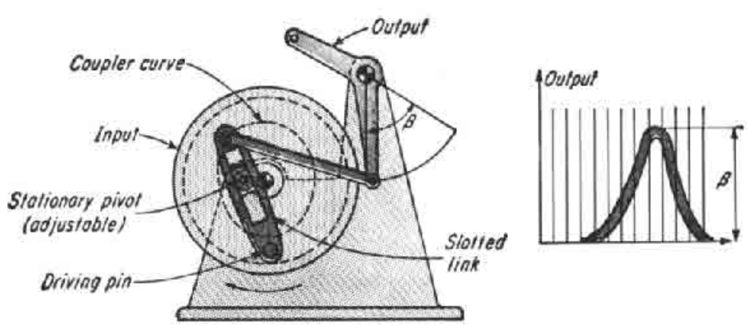
Adjustable-clutch drive

As the input crank of this drive makes a full rotation, the one-way clutch housing oscillates to produce an output rotation consisting of a series of pulse in one direction. Moving the adjusting block to the right or left changes the length of the strokes.



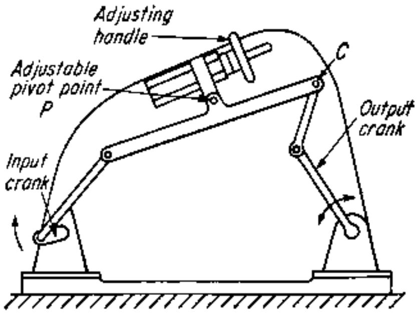
Adjustable-pivot drive

The driving pin of this drive rotates around the input center, but because the pivot is stationary with respect to the frame, the end of the slotted link produces a noncircular coupler curve and a fast advance and slow return in the output link. The stroke is varied by rotating the pivot to another position.



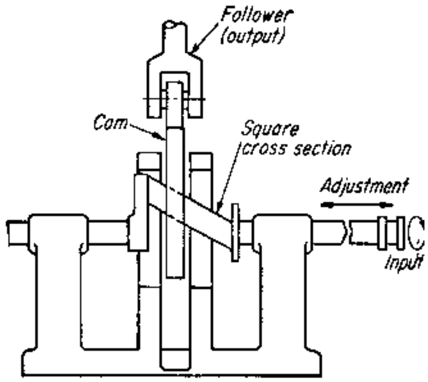
ADJUSTABLE-OUTPUT MECHANISMS

Linkage-motion adjuster



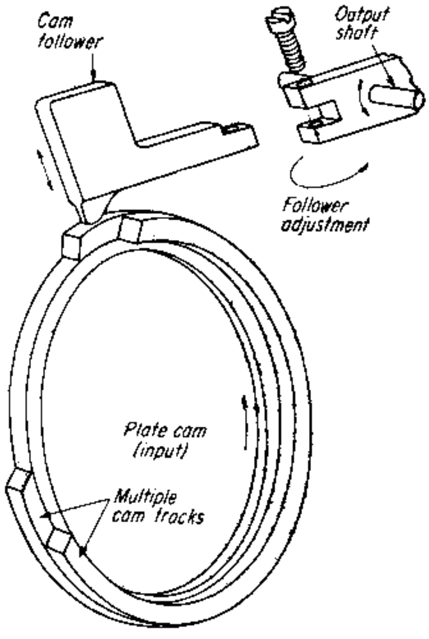
Here the motion and timing of the output link can be varied during its operation by shifting the pivot point of the intermediate link of the six-bar linkage illustrated. Rotation of the input crank causes point *C* to oscillate around the pivot point *P*. This, in turn, imparts an oscillating motion to the output crank. A screw device shifts point *P*.

Cam-motion adjuster



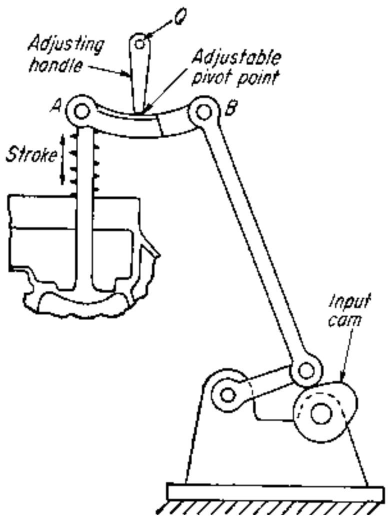
The output motion of the cam follower is varied by linearly shifting the input shaft to the right or left during its operation. The cam has a square hole which fits over the square cross section of the crank shaft. Rotation of the input shaft causes eccentric motion in the cam. Shifting the input shaft to the right, for instance, causes the cam to move radially outward, thus increasing the stroke of the follower.

Double-cam mechanism



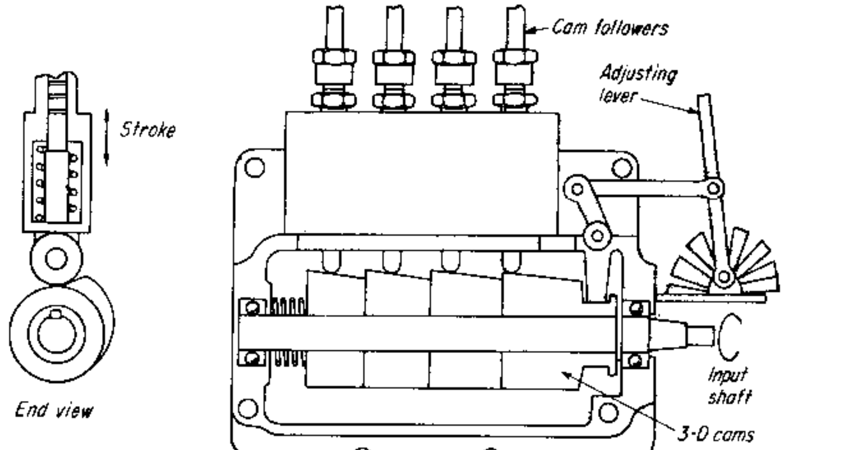
This is a simple but effective mechanism for changing the timing of a cam. The follower can be adjusted in the horizontal plane, but it is restricted in the vertical plane. The plate cam contains two or more cam tracks.

Valve-stroke adjuster



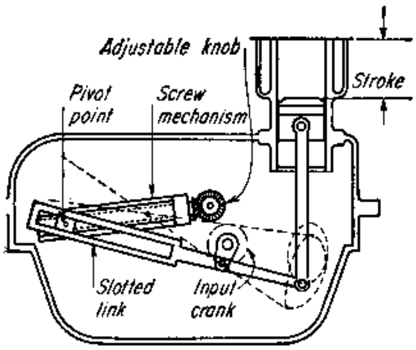
This mechanism adjusts the stroke of valves of combustion engines. One link has a curved surface and pivots around an adjustable pivot point. Rotating the adjusting link changes the proportion of strokes or points *A* and *B* and hence of the valve. The center of curvature of the curve link is at point *Q*.

3-D mechanism



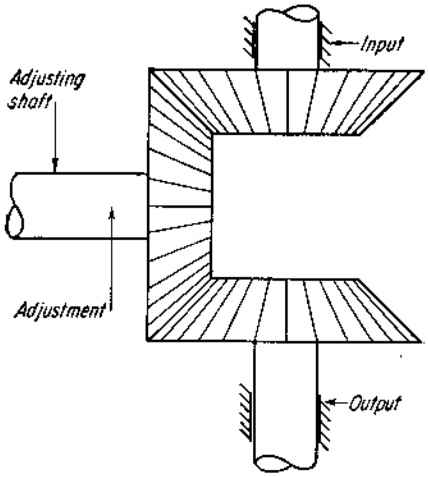
Output motions of four followers can be varied during the rotation by shifting the quadruple 3-D cam to the right or left. A linear shift can be made with the adjustment lever, which can be released in any of the six positions.

Piston-stroke adjuster



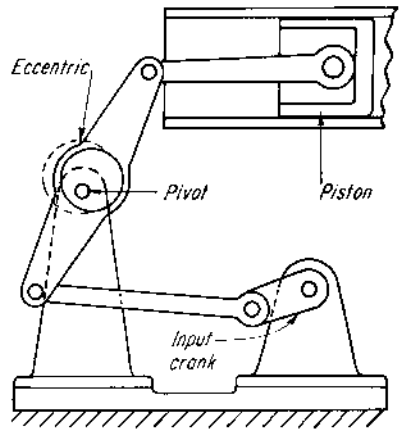
The input crank oscillates the slotted link to drive the piston up and down. The position of the pivot point can be adjusted with the screw mechanism even when the piston is under full load.

Shaft synchronizer



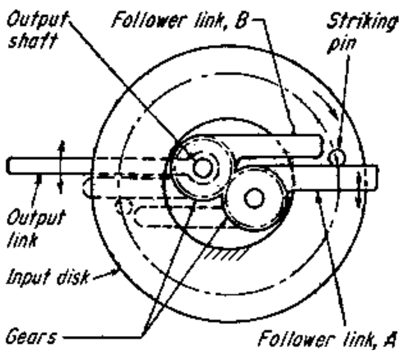
The actual position of the adjusting shaft is normally kept constant. The input then drives the output with the bevel gears. Rotating the adjusting shaft in a plane at right angles to the input-output line changes the relative radial position of the input and output shafts. They introduce a torque into the system while running, synchronizing the input and output shafts, or changing the timing of a cam on the output shaft.

Eccentric pivot point



Rotation of the input crank causes the piston to reciprocate. The stroke length depends on the position of the pivot point which is easily adjusted, even during rotation, by rotating the eccentric shaft.

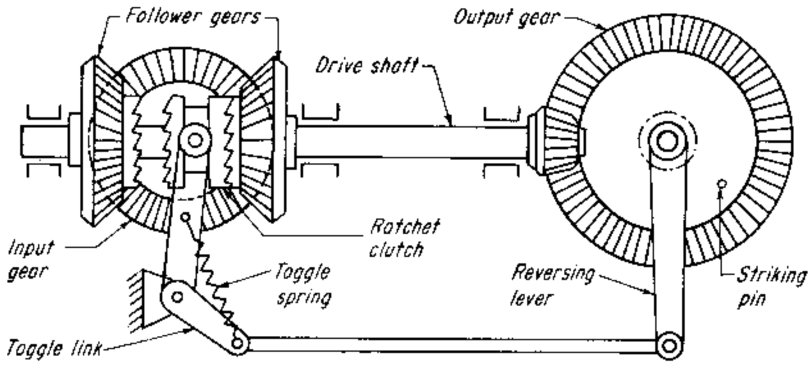
Double-link reverser



This mechanism automatically reverses the output drive for every 180º rotation of the input. The input disk has a press-fit pin which strikes link *A* to drive it clockwise. Link *A* in turn drives link *B* counterclockwise with its gear segments (or gears pinned to the links). The output shaft and the output link (which can be the working member) are connected to link *B*.

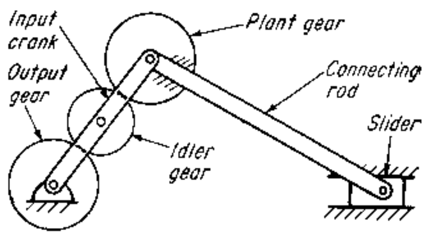
After approximately 180º of rotation, the pin slides past link *A* to strike link *B* coming to meet it—and thus reverses the direction of link *B* (and of the output). Then after another 180º rotation the pin slips past link *B* to strike link *A* and starts the cycle over again.

Toggle-link reverser



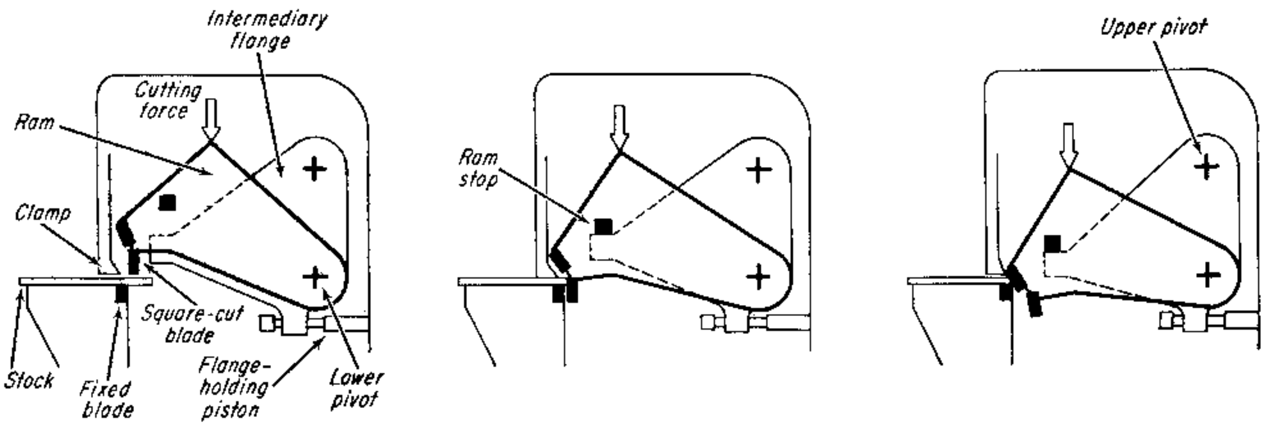
This mechanism also employs a striking pin—but here the pin is on the output member. The input bevel gear drives two follower bevels which are free to rotate on their common shaft. The ratchet clutch, however, is spline-connected to the shaft—although free to slide linearly. As shown, it is the right follower gear that is locked to the drive shaft. Hence the output gear rotates clockwise until the pin strikes the reversing level to shift the toggle to the left. Once past its center, the toggle spring snaps the ratchet to the left to engage the left follower gear. This instantly reverses the output, which now rotates counterclockwise until the pin again strikes the reversing level. Thus the mechanism reverses itself for every 360º rotation of the input.

Modified-Watt's reverser



This is a modification of the well-known Watt crank mechanism. The input crank causes the planet gear to revolve around the output gear. But because the planet gear is fixed to the connecting rod, it causes the output gear to continually reverse itself. If the radii of the two gears are equal, each full rotation of the input link will cause the output gear to oscillate through the same angle as the rod.

Automatically switching from one pivot point to another in midstroke.



Two pivots and the intermediary flange govern the cutting sequence. The flange is connected to the press frame at the upper pivot, and the cutting ram is connected to the flange at the lower pivot. In the first part of the cycle, the ram turns around the lower pivot and shears the plate with the square-cut blade; the motion of the intermediary flange is restrained by the flange-holding piston.

After the shearing cut, the ram stop bottoms on the flange. This overcomes the restraining force of the flange-holding piston, and the ram turns around the upper pivot. This brings the beveling blade into contact with the plate for the bevel cut.

COMPUTING MECHANISMS

Analog computing mechanisms are capable of almost instantaneous response to minute variations in input. Basic units, similar to the examples shown, are combined to form the final mechanism. These mechanisms add, subtract, resolve vectors, or solve special or trigonometric functions.

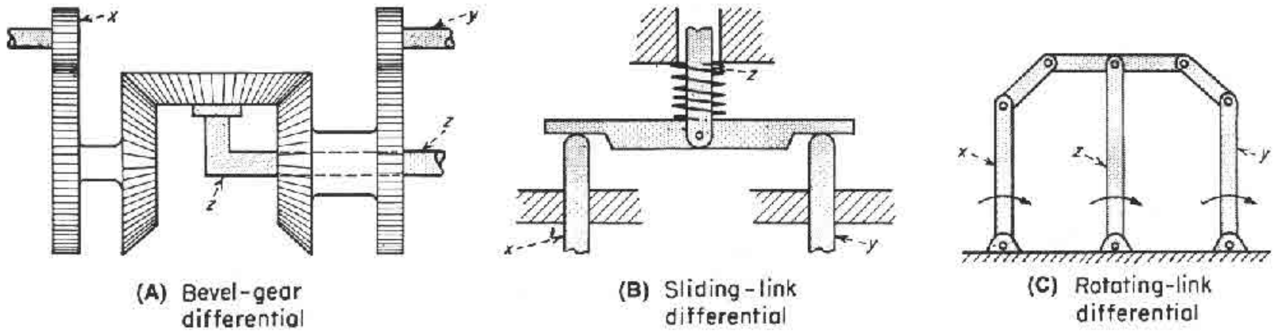


Fig. 1 Addition and subtraction is usually based on the differential principle; variations depend on whether inputs: (A) rotate shafts, (B) translate links, or (C) angularly displace links. Mechanisms can solve the equation: $z = c\,(x \pm y)$, where c is the scale factor, x and y are inputs, and z is the output. The motion of x and y in the same direction performs addition; in the opposite direction it performs subtraction.

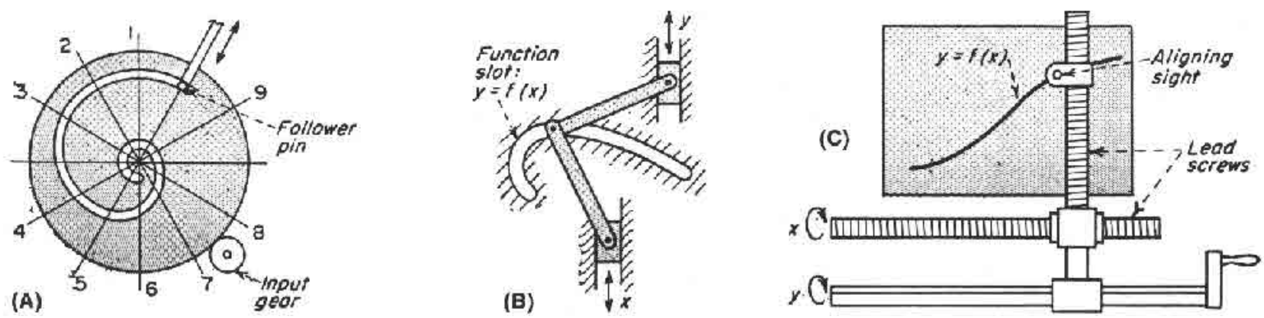


Fig. 2 Functional generators mechanize specific equations. (A) A reciprocal cam converts a number into its reciprocal. This simplifies division by permitting simple multiplication between a numerator and its denominator. The cam is rotated to a position corresponding to the denominator. The distance between the center of the cam to the center of the follower pin corresponds to a reciprocal. (B) A function-slot cam is ideal for performing complex functions involving one variable. (C) A function is plotted on a large sheet attached to a table. The x lead screw is turned at a constant speed by an analyzer. An operator or photoelectric follower turns the y output to keep the aligning sight on the curve.

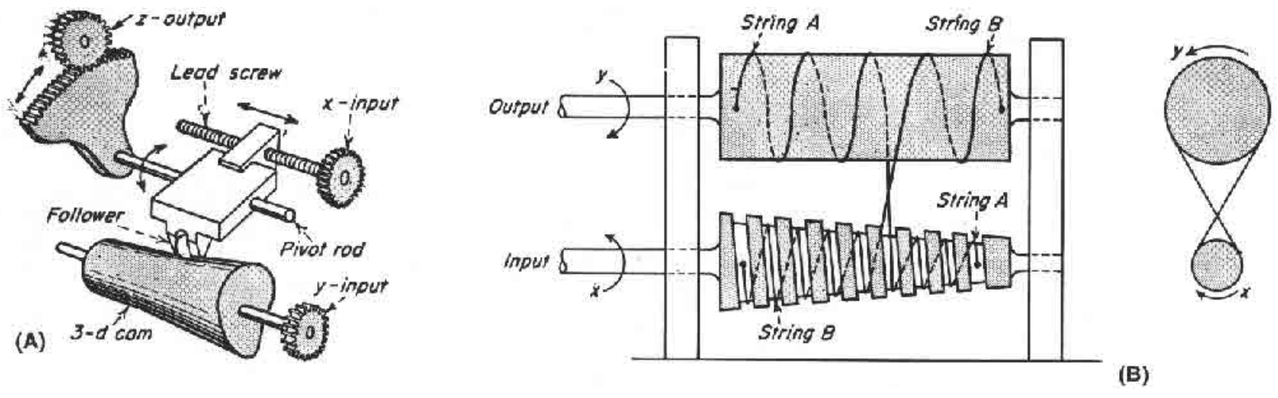


Fig. 3 (A) **A three-dimensional cam** generates functions with two variables: $z = f\,(x, y)$. A cam is rotated by the y-input; the x-input shifts a follower along a pivot rod. The contour of the cam causes a follower to rotate, giving angular displacement to the z-output gear. (B) **A conical cam** for squaring positive or negative inputs: $y = c\,(\pm x)^2$. The radius of a cone at any point is proportional to the length of string to the right of the point; therefore, cylinder rotation is proportional to the square of cone rotation. The output is fed through a gear differential to convert it to a positive number.

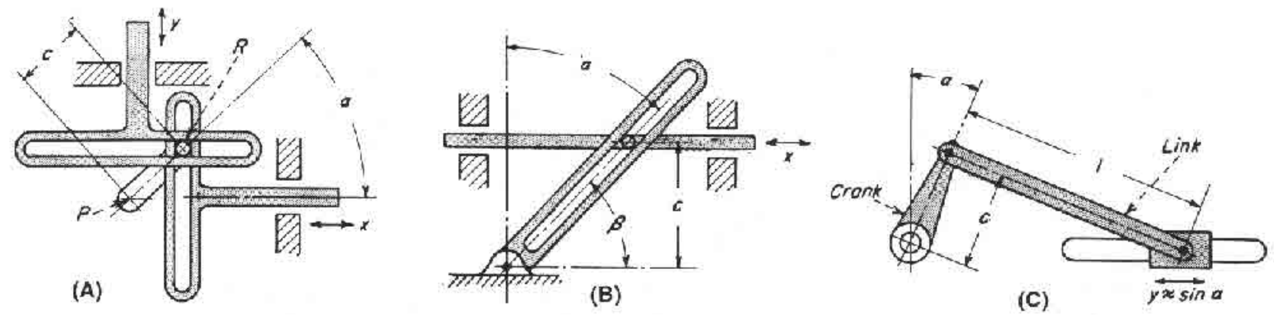


Fig. 4 Trigonometric functions. (A) A Scotch-yoke mechanism for sine and cosine functions. A crank rotates about fixed point *P*, generating angle *a* and giving motion to the arms: $y = c \sin a$; $x = c \cos a$. (B) A tangent-cotangent mechanism generates $x = c \tan a$ or $x = c \cot \beta$. (C) The eccentric and follower is easily manufactured, but sine and cosine functions are approximate. The maximum error is zero at 90° and 270°; *l* is the length of the link, and *c* is the length of the crank.

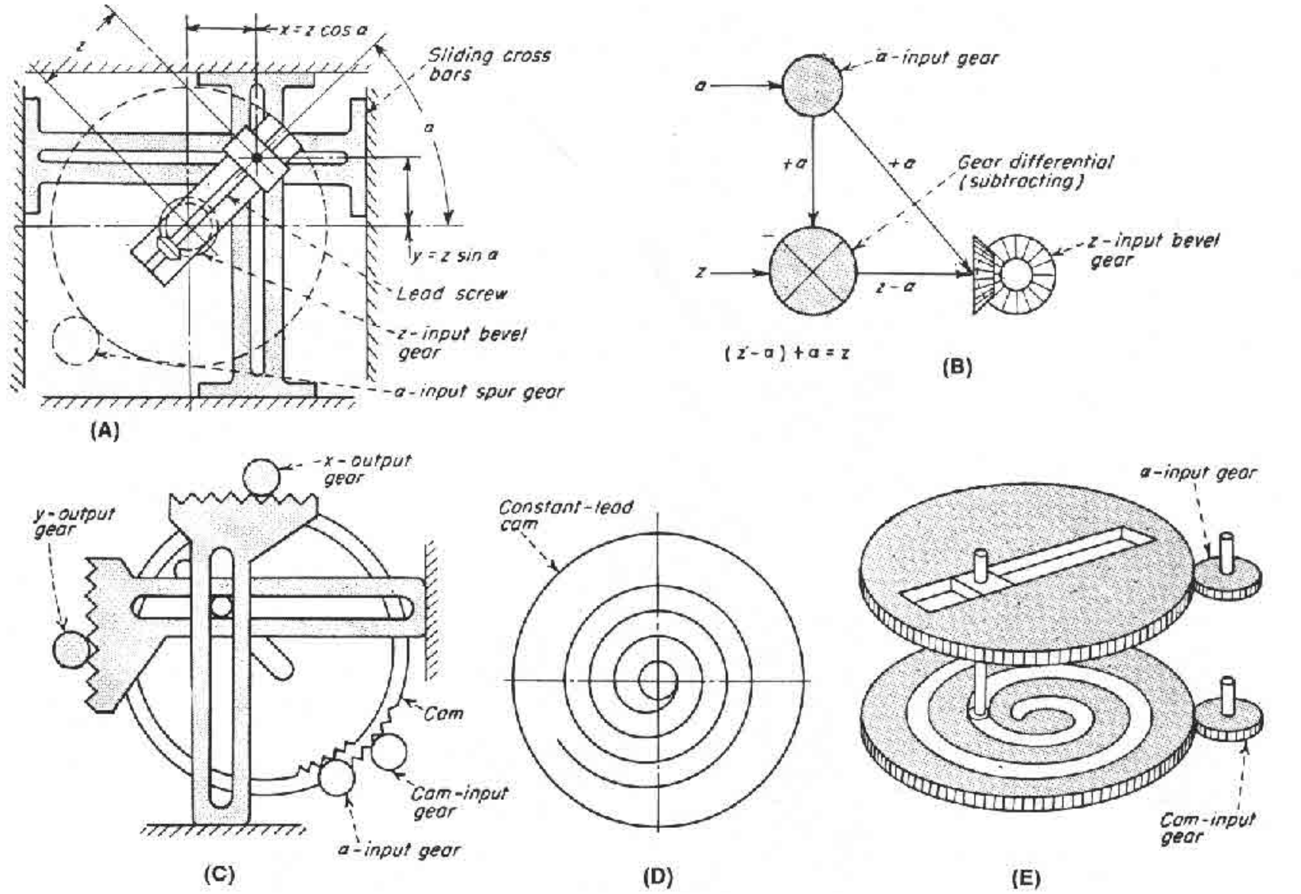


Fig. 5 Component resolvers determine *x* and *y* components of vectors that are continuously changing in both angle and magnitude. Equations are $x = z \cos a$, $y = z \sin a$, where *z* is magnitude of vector, and *a* is vector angle. Mechanisms can also combine components to obtain a resultant. Inputs in (A) are through bevel gears and lead screws for *z*-input, and through spur gears for *a*-input. Compensating gear differential (B) prevents the *a*-input from affecting the *z*-input. This problem is solved in (C) with constant-lead cams (D) and (E).

Typical computing mechanisms for performing the mathematical operations of multiplication, division, differentiation, and integration of variable functions are presented here.

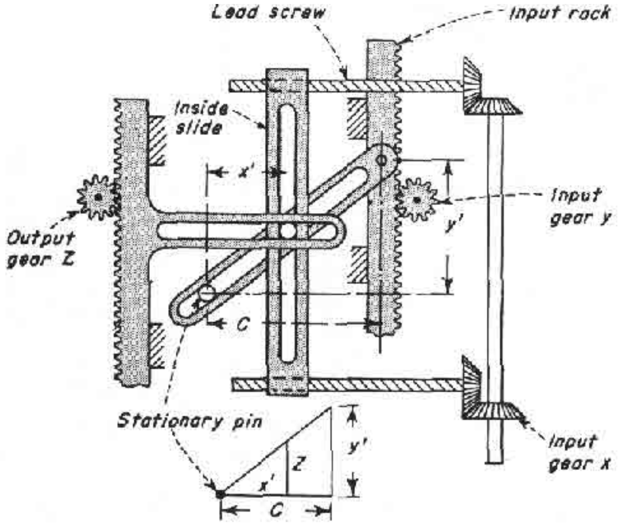


Fig. 1 (A)

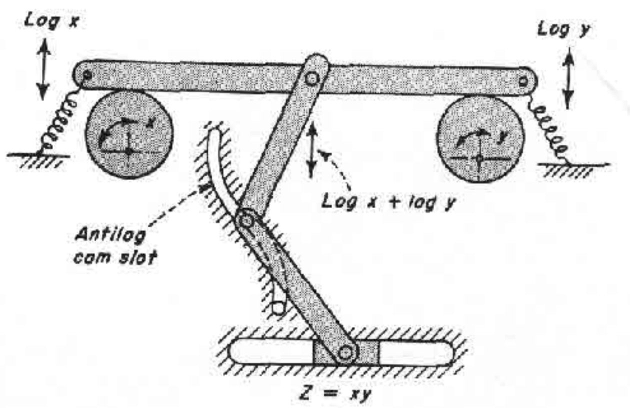


Fig. 1 (B)

Fig. 1 The multiplication of two tables, *x* and *y*, can usually be solved by either: (A) The similar triangle method, or (B) the logarithmic method. In (A), lengths x' and y' are proportional to the rotation of input gears *x* and *y*. Distance *c* is constant. By similar triangles: $z/x = y/c$ or $z = xy/c$, where *z* is vertical displacement of output rack. The mechanism can be modified to accept negative variables. In (B), the input variables are fed through logarithmic cams to give linear displacements of log *x* and log *y*. The functions are then added by a differential link giving $z = \log x + \log xy$ (neglecting scale factors). The result is fed through the antilog cam so that the motion of the follower represents $z = xy$.

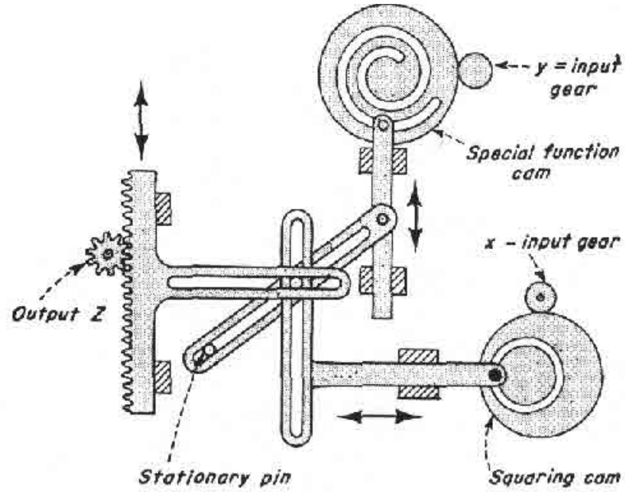


Fig. 2 (A)

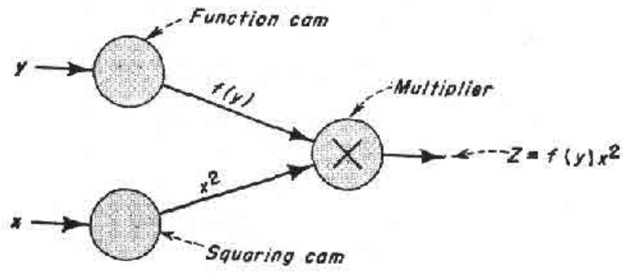


Fig. 2 (B)

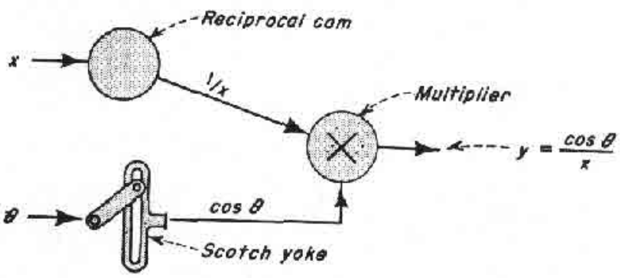
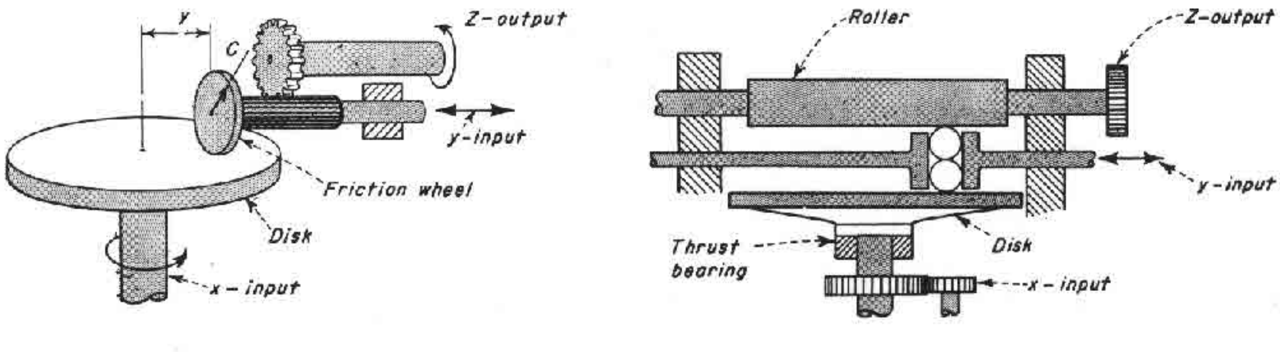


Fig. 2 (C)

Fig. 2 Multiplication of complex functions can be accomplished by substituting cams in place of input slides and racks of the mechanism in Fig. 1. The principle of similar triangles still applies. The mechanism in (A) solves the equation: $z = f(y)x^2$. The schematic is shown in (B). Division of two variables can be done by feeding one of the variables through a reciprocal cam and then multiplying it by the other. The schematic in (C) shows the solution of $y = \cos \theta / x$.



(A) (B)

Fig. 3 Integrators are essentially variable-speed drives. The *x*-input shaft in Fig. 3 (A) rotates the disk which, in turn, rotates the friction wheel on the *y*-input shaft which is perpendicular to the *x*-input shaft. As the friction wheel turns, it rotates a spline on the movable *y*-input shaft. The gear on the end of the parallel *z*-output shaft drives that shaft.

Moving the *y*-input shaft along the radius dimension of the disk changes the rotational speed of the friction wheel from zero at the center of the disk to a maximum at the periphery. The *z*-axis output is thus a function of the rotational speed of the *x*-input, the diameter of the friction wheel, and *y*, the radius distance of the wheel on the disk,

In the integrator shown in Fig. 3 (B), two balls replace the friction wheel and spline of the *y*-input axis, and a roller replaces the gear on the *z*-output shaft to provide a variable-speed output as the *y*-input shaft is moved across the entire diameter of the disk.

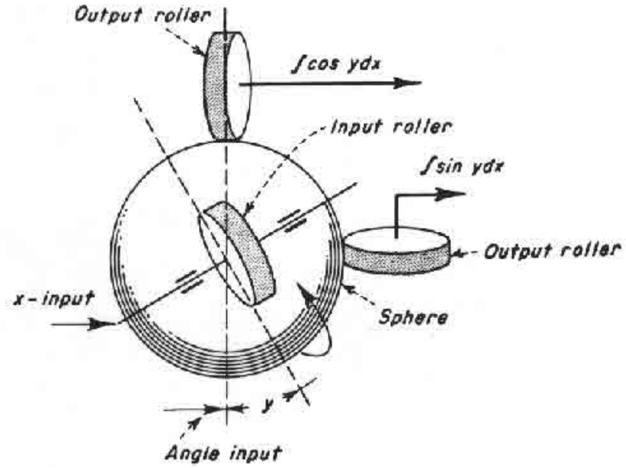


Fig. 4 A component integrator has three disks to obtain the *x* and *y* components of a differential equation. The input roller on the *x*-input shaft spins the sphere, and the *y*-input lever arm changes the angle of the roller with respect to the sphere. The sine and cosine output rollers provide integrals of components that parallel the *x* and *y* axes.

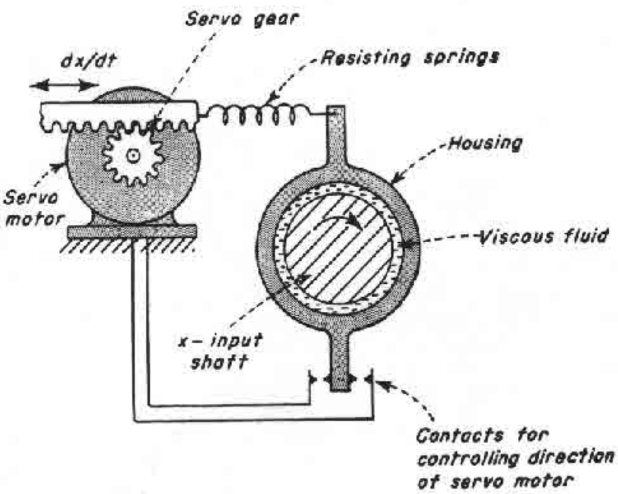
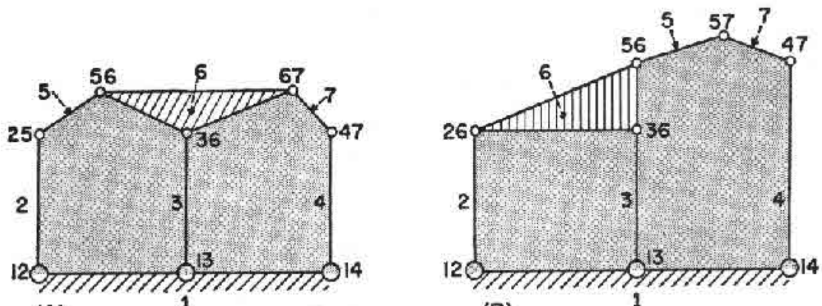


Fig. 5 This differentiator is based on the principle that a viscous drag force in a thin layer of fluid is proportional to the velocity of a rotating *x*-input shaft. The drag force is counteracted by resisting springs in tension. Spring length is regulated by a servomotor controlled by electrical contacts at the base of the housing. A change in shaft velocity causes a change in viscous torque. A shift in the housing closes one set of electrical contacts, causing the motor shaft to turn. This repositions a rack which adjusts the spring tension and balances the system. The total rotation of the servomotor gear is proportional to *dx/dt*.

EIGHTEEN VARIATIONS OF DIFFERENTIAL LINKAGE

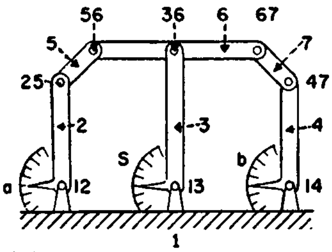
Figure 1 shows the modifications of the differential linkage shown in Fig. 2(A). These are based on the variations in the triple-jointed intermediate link 6. The links are designated as follows: Frame links: links 2, 3 and 4; two-jointed intermediate links: links 5 and 7; three jointed intermediate links: link 6.



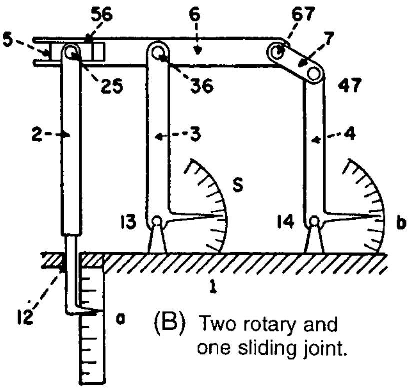
(A) Fig. 2 (B)

The input motions to be added are *a* and *b*; their sum *s* is equal to $c_1a + c_2b$, where c_1 and c_2 are scale factors. The links are numbered in the same way as those in Fig. 2(A).

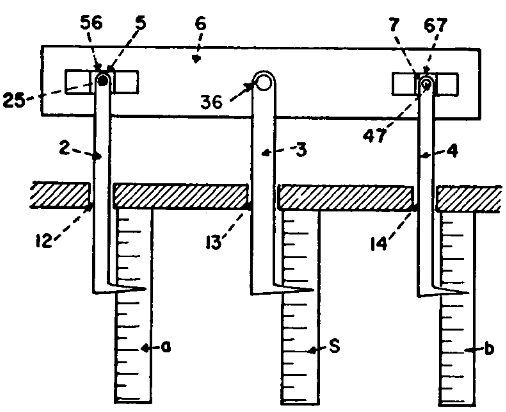
Fig. 1



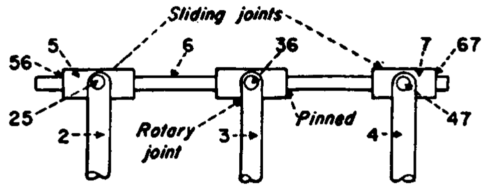
(A) Three rotary joints.



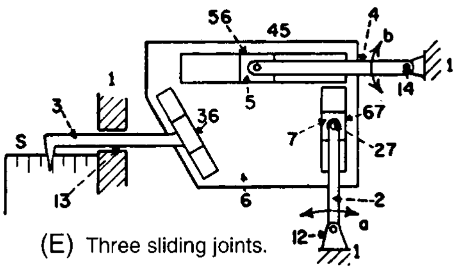
(B) Two rotary and one sliding joint.



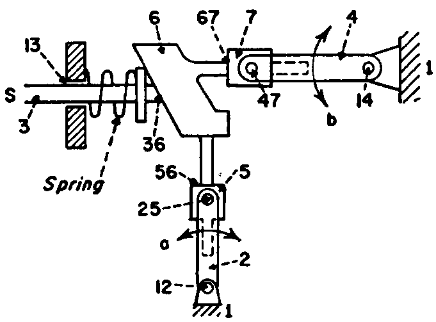
(C) One rotary and two sliding joints.



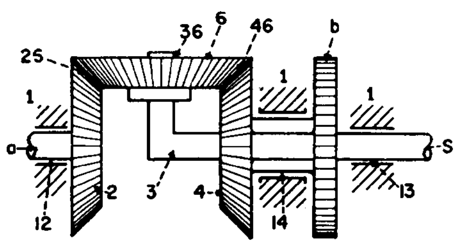
(D) One rotary and two sliding joints.



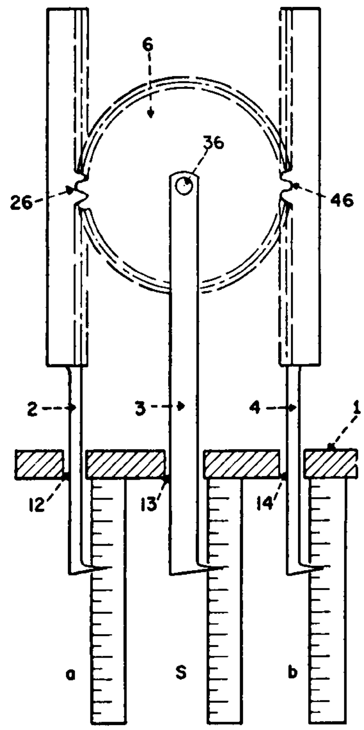
(E) Three sliding joints.



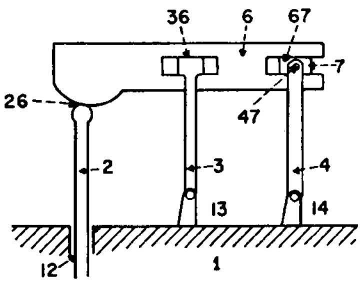
(F) Three sliding joints.



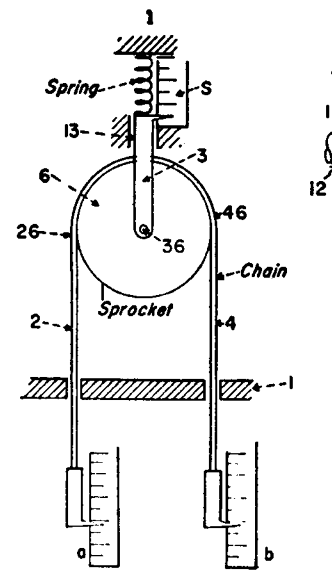
(G) One rotary and two cam joints (planetary gear differential)



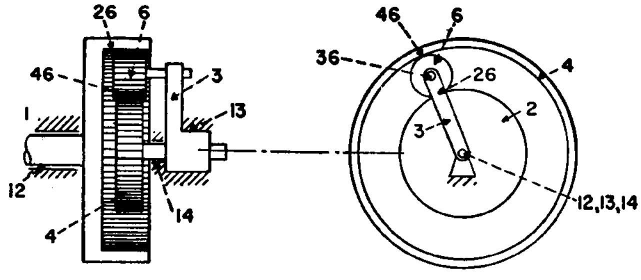
(H) One rotary and two cam joints (rack and gear).



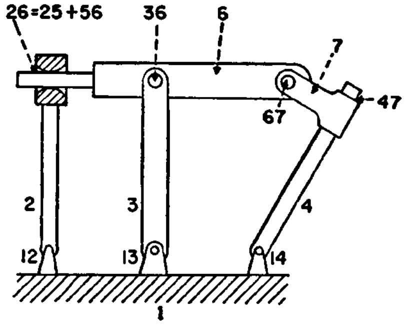
(I) Two sliding and one cam joint.



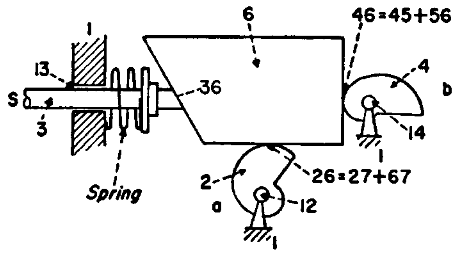
(J) One rotary and two cam joints (chain and sprocket).



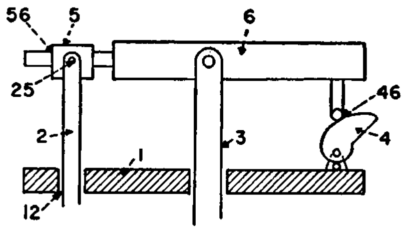
(M) One rotary and two cam joints (planetary gears differential).



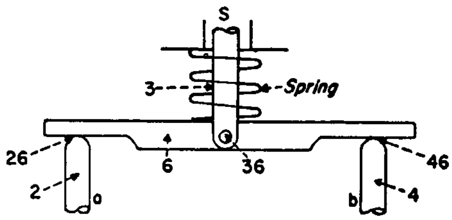
(O) Two rotary and one cam joint.



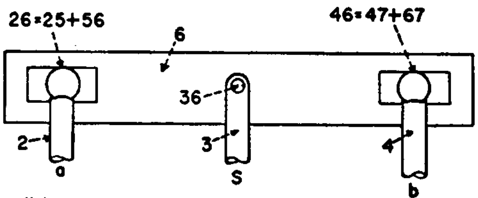
(N) One sliding and two cam joints.



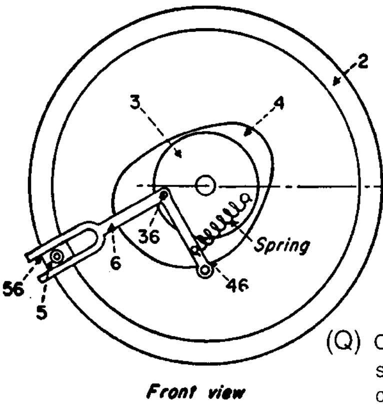
(P) One rotary, one sliding, and one cam joint.

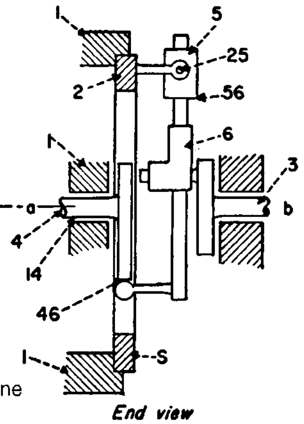


(K) One rotary and two cam joints.

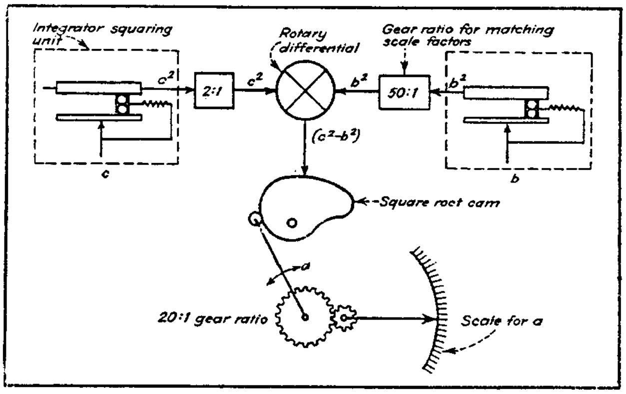


(L) One rotary and two cam joints.

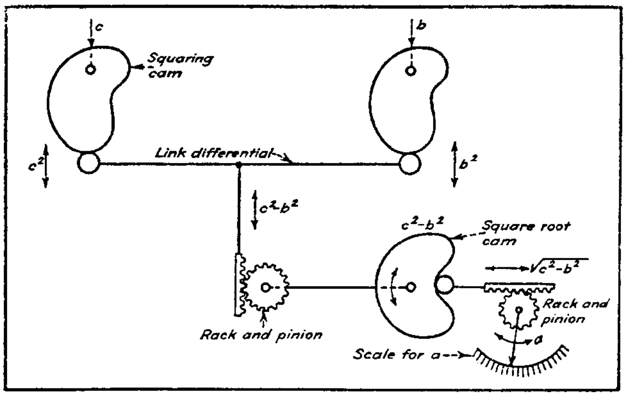




(Q) One rotary, one sliding and and one cam joint.



The intergrator method of mechanizing the equation $a = \sqrt{c^2 - b^2}$ is shown in the schematic form. It requires an excessive number of parts.



The cam method of mechanizing $a = \sqrt{c^2 - b^2}$ uses function generators for squaring and a link differential for subtraction. Note the reduction in parts from the integrator method.

SPACE MECHANISMS

There are potentially hundreds of them, but only a few have been discovered so far. Here are the best of one class—the four-bar space mechanisms.

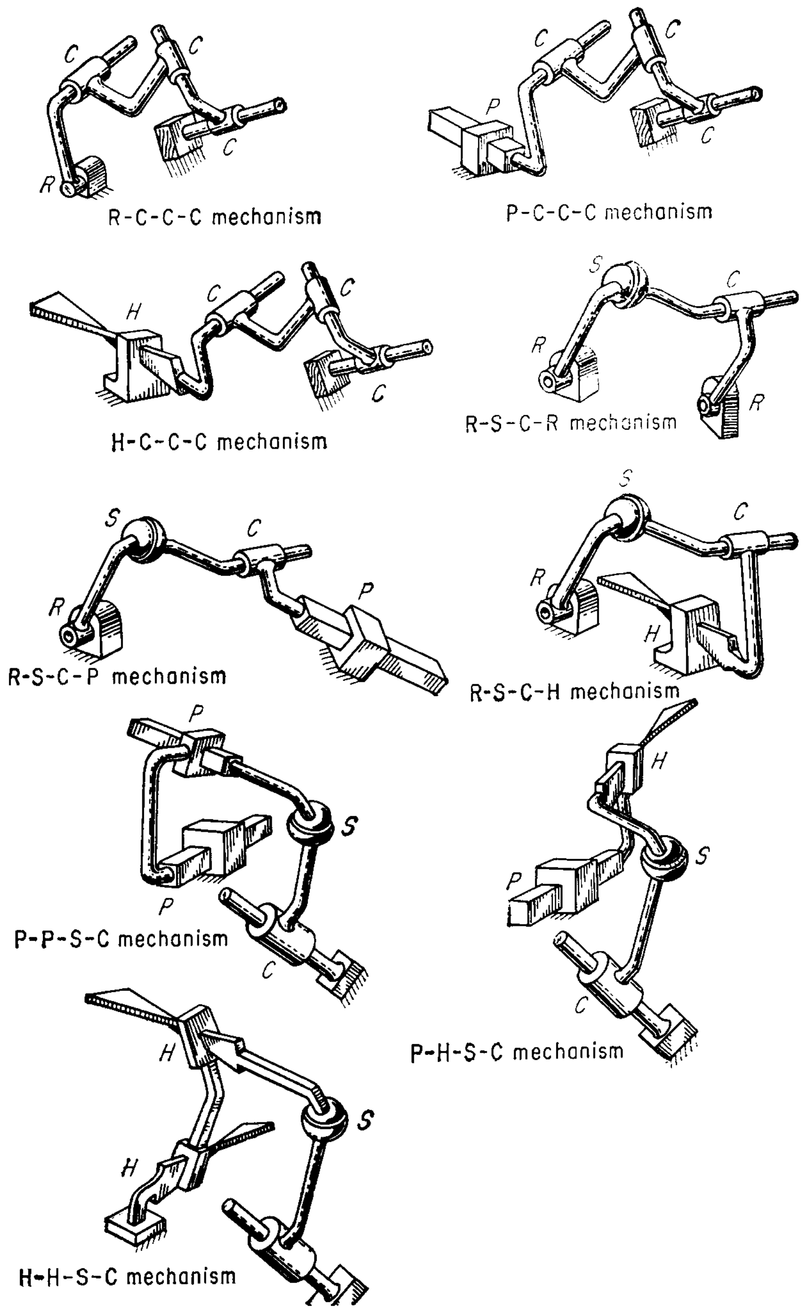


Fig. 1 The nine chosen mechanisms.

A virtually unexplored area of mechanism research is the vast domain of three-dimensional linkage, frequently called space mechanism. Only a comparatively few kinds have been investigated or described, and little has been done to classify those that are known. As a result, many engineers do not know much about them, and applications of space mechanisms have not been as widespread as they could be.

Because a space mechanism can exist with a wide variety of connecting joints or "pair" combinations, it can be identified by the type and sequence of its joints. A listing of all of the physically realizable kinematic pairs has been established, based on the number of degrees-of-freedom of a joint. These pairs are all the known ways of connecting two bodies together for every possible freedom of relative motion between them.

The Practical Nine

The next step was to find the combination of pairs and links that would produce practical mechanisms. Based on the "Kutzbach criterion" (the only known mobility criterion—it determines the degree of freedom of a mechanism due to the constraints imposed by the pairs), 417 different kinds of space mechanisms have been identified. Detailed examination showed many of these to be mechanically complex and of limited adaptability. But the four-link mechanisms had particular appeal because of their mechanical simplicity. A total of 138 different kinds of four-bar mechanisms have been found. Of these, nine have particular merit (Fig. 1).

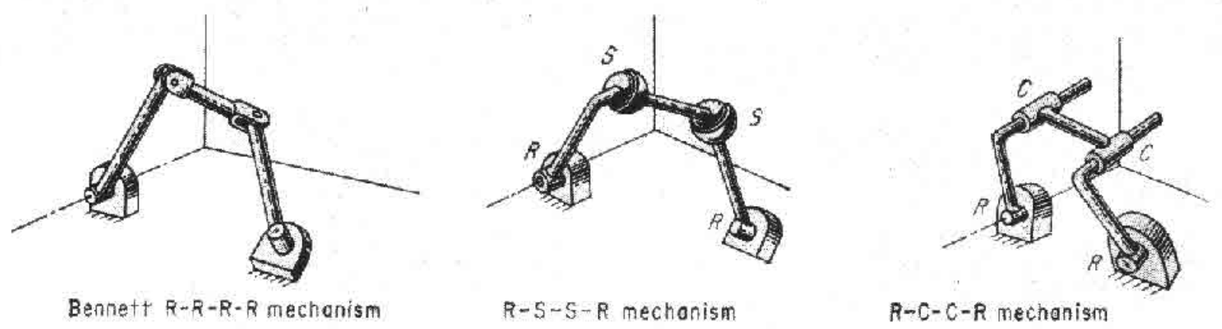


Fig. 2 The three mavericks.

These nine four-link mechanisms are the easiest to build because they contain only those joints that have area contact and are self-connecting. In the table, these joints are the five closed, lower pair types:

R = Revolute joint, which permits rotation only

P = Prism joint, which permits sliding motion only

H = Helix or screw type of joint

C = Cylinder joint, which permits both rotation and sliding (hence has two degrees of freedom)

S = Sphere joint, which is the common ball joint permitting rotation in any direction (three degrees of freedom)

All these mechanisms can produce rotary or sliding output motion from a rotary input—the most common mechanical requirements for which linkage mechanisms are designed.

The type letters of the kinematic pairs in the table identify the mechanism by ordering the letter symbols consecutively around the closed kinematic chain. The first letter identifies the pair connecting the input link and the fixed link; the last letter identifies the output link, or last link, with the fixed link. Thus, a mechanism labeled *R-S-C-R* is a double-crank mechanism with a spherical pair between the input crank and the coupler, and a cylindrical pair between the coupler and the output crank.

The Mavericks

The Kutzbach criterion is inadequate for the job because it cannot predict the existence of such mechanisms as the Bennett *R-R-R-R* mechanism, the double-ball joint *R-S-S-R* mechanism, and the *R-C-C-R* mechanism (Fig. 2). These "special" mechanisms require special geometric conditions to have a single degree of freedom. The *R-R-R-R* mechanism requires a particular orientation of the revolute axes and a particular ratio of link lengths to function as a single degree of freedom space mechanism. The *R-S-S-R* configuration, when functioning as a single degree-of-freedom mechanism, will have a passive degree of freedom of its coupler link. When properly constructed, the configuration *R-C-C-R* will also have a passive degree-of-freedom of its coupler, and it will function as a single-degree space mechanism.

Of these three special four-link mechanisms, the *R-S-S-R* mechanism is seen as the outstanding choice. It is the most versatile and practical configuration for meeting double-crank motion requirements.

Classification of kinematic pairs

Degree of freedom	Type number*	Type of joint	
		Symbol	Name
	100	R	Revolute
1	010	P	Prism
	001	H	Helix
	200	T	Torus
	110	C	Cylinder
2	101	T_H	Torus-helix
	020	..	....
	011	..	....
	300	S	Sphere
	210	S_S	Sphere-slotted cylinder
3	201	S_{SH}	Sphere-slotted helix
	120	P_L	Plane
	021	..	....
	111	..	....
	310	S_G	Sphere-groove
	301	S_{GH}	Sphere-grooved helix .
4	220	C_p	Cylinder-plane
	121	..	....
	211	..	....
	320	S_p	Sphere-plane
5	221	..	....
	311		....

* Number of freedoms, given in the order of N_R, N_T, N_H.

SEVEN POPULAR TYPES OF THREE-DIMENSIONAL DRIVES

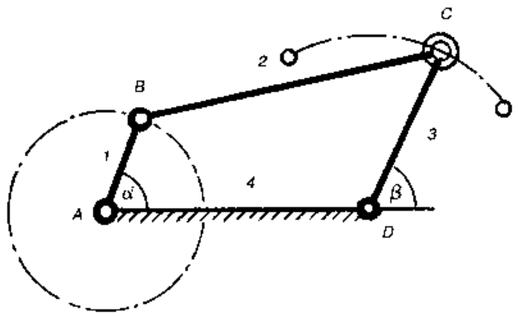
The main advantage of three-dimensional drives is their ability to transmit motion between nonparallel shafts. They can also generate other types of helpful motion. This roundup includes descriptions of seven industrial applications for the drives.

Spherical Crank Drive

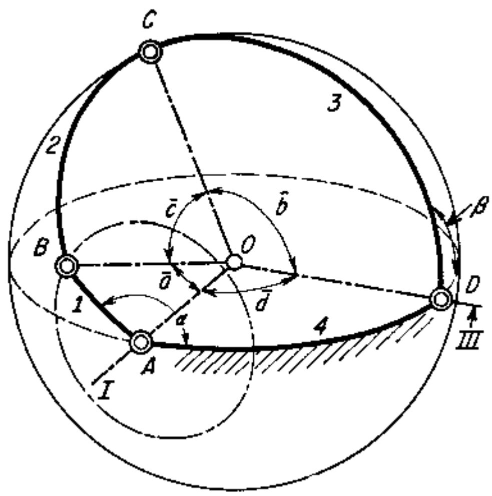
This type of drive is the basis for most three-dimensional linkages, much as the common four-bar linkage is the basis for the two-dimensional field. Both mechanisms operate on similar principles. (In the accompanying sketches, *a* is the input angle, and β the output angle. This notation has been used throughout this section.)

In the four-bar linkage, the rotary motion of driving crank *1* is transformed into an oscillating motion of output link *3*. If the fixed link is made the shortest of all, then it is a double-crank mechanism; both the driving and driven members make full rotations.

The spherical crank drive, link *1* is the input, link *3* the output. The axes of rotation intersect at point *O;* the lines connecting *AB*, *BC*, *CD*, and *DA* can be considered to be parts of great circles of a sphere. The length of the link is best represented by angles *a*, *b*, *c*, and *d*.

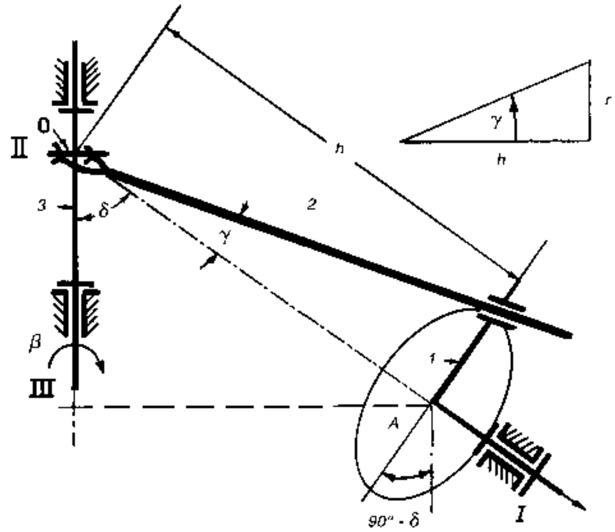


Four-bar linkage



The Spherical Crank

Spherical-Slide Oscillator Drive



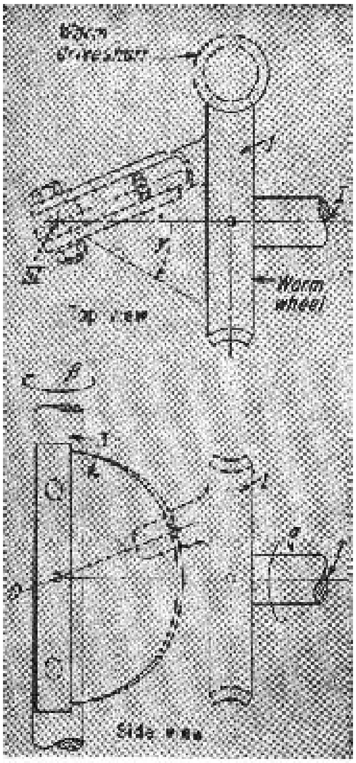
The two-dimensional slider crank is obtained from a four-bar linkage by making the oscillating arm infinitely long. By making an analogous change in the spherical crank, the spherical slider crank is obtained.

The uniform rotation of input shaft *I* is transferred into a nonuniform oscillating or rotating motion of output shaft *III*. These shafts intersect at an angle δ, corresponding to the frame link *4* of the spherical crank. Angle γ corresponds to the length of link *1*, and axis *II* is at right angle to axis *III*.

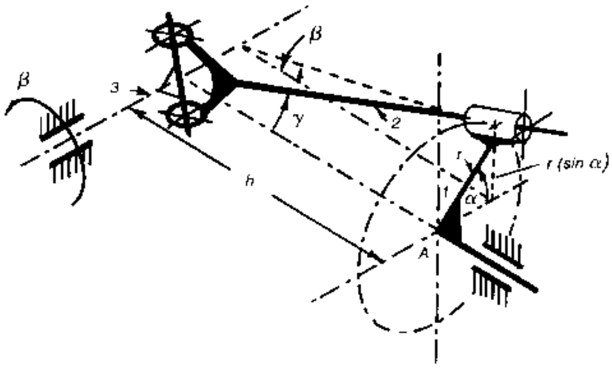
The output oscillates when γ is smaller than δ, but it rotates when γ is larger than δ.

The relation between input angle *a* and output angle β as designated in the skewed Hooke's joint is:

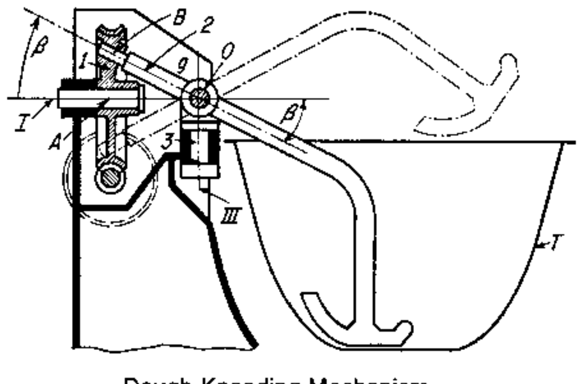
$$\tan\beta = \frac{(\tan\gamma)(\sin\alpha)}{\sin\delta + (\tan\gamma)(\cos\delta)(\cos\alpha)}$$



Washing-Machine Mechanism



Skewed Hooke's Joint



Dough-Kneading Mechanism

Skewed Hooke's Joint Drive

This variation of the spherical crank is specified where an almost linear relation is desired between the input and output angles for a large part of the motion cycle.

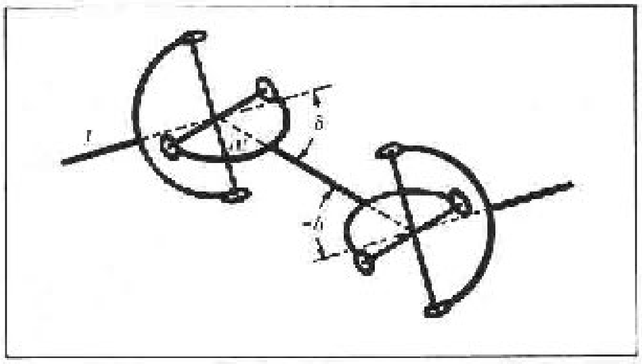
The equation defining the output in terms of the input can be obtained from the skewed Hooke's joint equation by making δ = 90°. Thus, sin δ = 1, cos δ = 0, and

$$\tan\beta = \tan\gamma \sin\alpha$$

The principle of the skewed Hooke's joint has been applied to the drive of a washing machine (see sketch).

Here, the driveshaft drives the worm wheel *1* which has a crank fashioned at an angle γ. The crank rides between two plates and causes the output shaft *III* to oscillate in accordance with the equation.

The dough-kneading drive is also based on the Hooke's joint, but it follows the path of link *2* to give a wobbling motion that kneads dough in the tank.



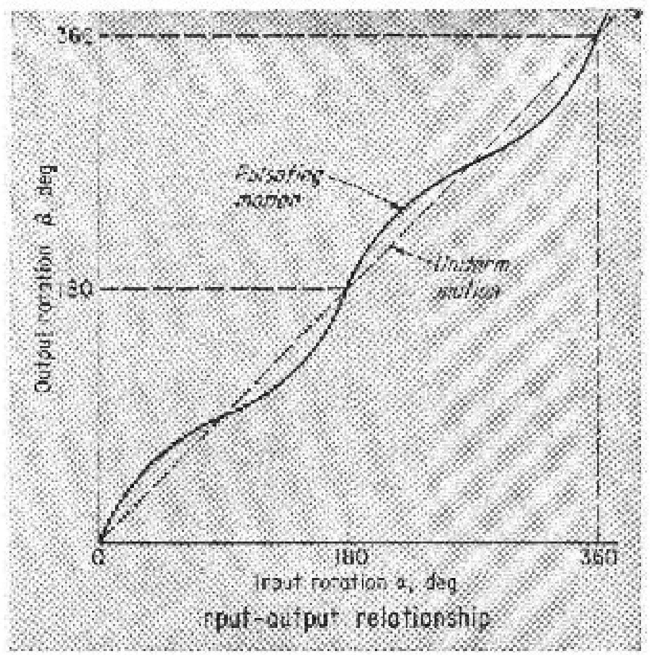
The Universal Joint Drive

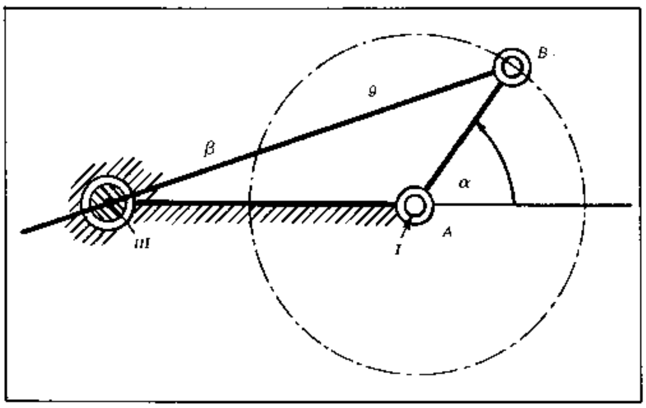
The universal joint is a variation of the spherical-slide oscillator, but with angle $\gamma = 90°$. This drive provides a totally rotating output and can be operated as a pair, as shown in the diagram.

The equation relating input with output for a single universal joint, where γ is the angle between the connecting link and shaft *I*, is:

$$\tan \beta = \tan \alpha \cos \delta$$

The output motion is pulsating (see curve) unless the joints are operates as pairs to provide a uniform motion.

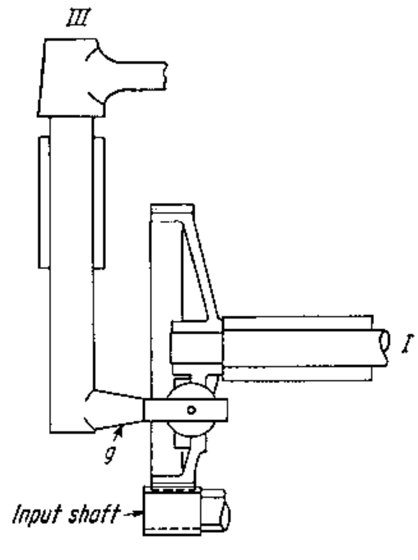




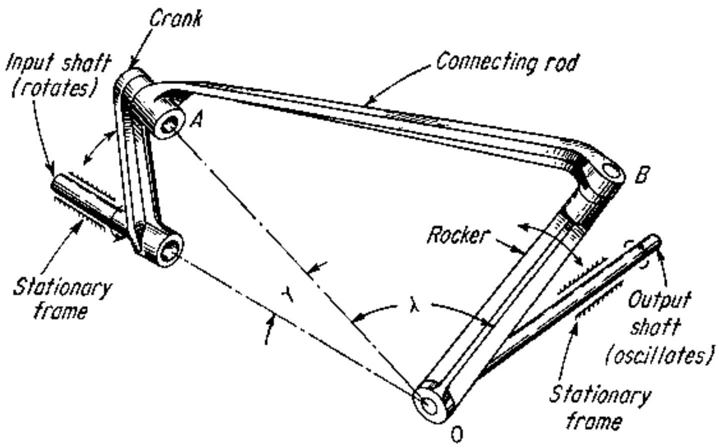
The 3-D Crank Slide Drive

The three-dimensional crank slide is a variation of a plane crank slide (see sketch), with a ball point through which link *g* always slides, while a point B on link *g* describes a circle. A 3-D crank is obtained from this mechanism by shifting output shaft *III* so that it is not normal to the plane of the circle; another way to accomplish this is to make shafts *I* and *III* nonparallel.

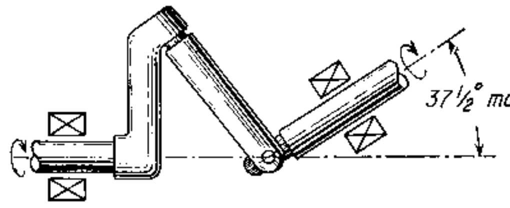
A practical variation of the 3-D crank slide is the agitator mechanism (see sketch). As input gear *I* rotates, link *g* swivels around (and also lifts) shaft *III*. Hence, the vertical link has both an oscillating rotary motion and a sinusoidal harmonic translation in the direction of its axis of rotation. The link performs what is essentially a twisting motion in each cycle.



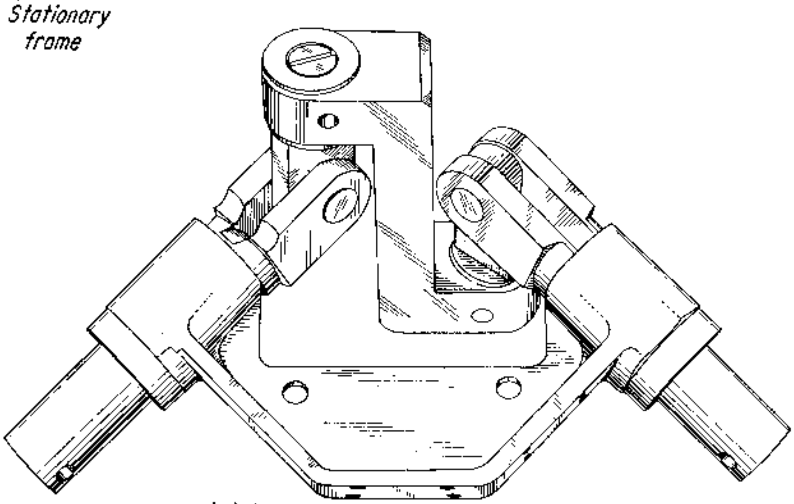
Agitator Mechanism



(A) Basic Configuration



(B) Its Inversion



(C) As a 90° Uniform Motion Transmitter

The Space Crank Drive

One of the more recent developments in 3-D linkages is the space crank shown in (A). It resembles the spherical crank, but has different output characteristics. The relationship between the input and output displacements is:

$$\cos\beta = (\tan\gamma)(\cos\alpha)(\sin\beta) - \frac{\cos\lambda}{\cos\gamma}$$

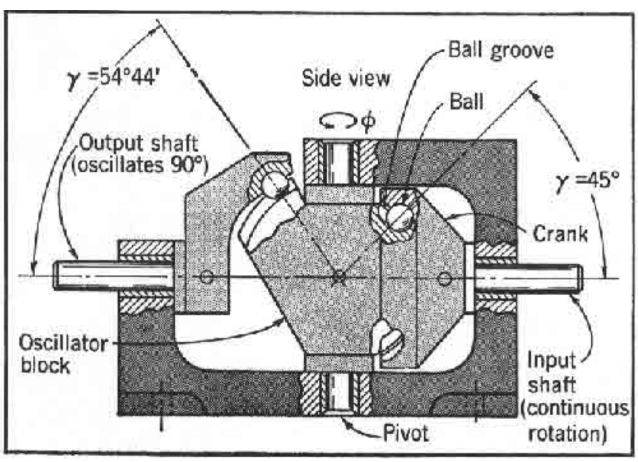
The velocity ratio is:

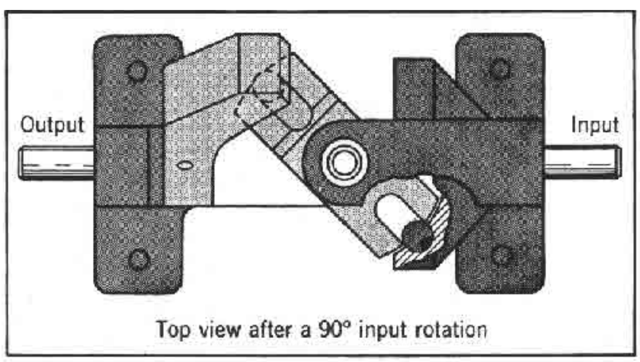
$$\frac{\omega_o}{\omega_i} = \frac{\tan\gamma\sin\alpha}{1+\tan\gamma\cos\alpha\cot\beta}$$

where ω_0 is the output velocity and ω_i is the constant input velocity.

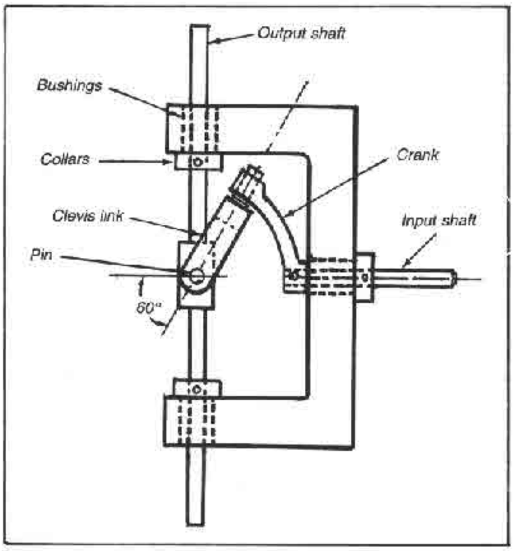
An inversion of the space crank is shown in (B). It can couple intersecting shafts, and permits either shaft to be driven with full rotations. Motion is transmitted up to 37½º misalignment.

By combining two inversions (C), a method for transmitting an exact motion pattern around a 90º bend is obtained. This unit can also act as a coupler or, if the center link is replaced by a gear, it can drive two output shafts; in addition, it can transmit uniform motion around two bends.

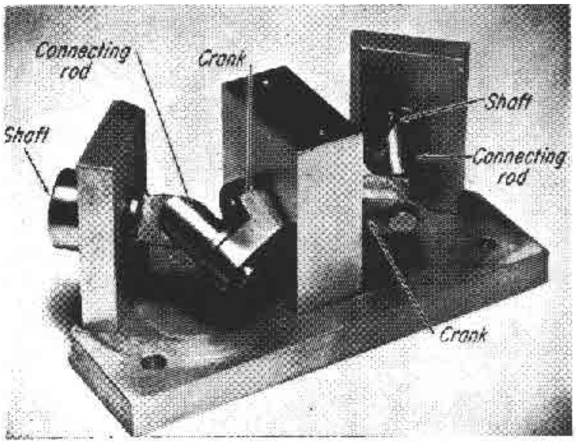




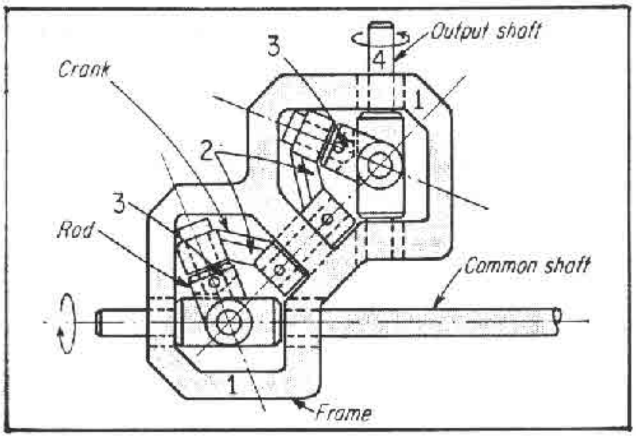
Steel balls riding within spherical grooves convert a continuous rotary input motion into an output that oscillates the shaft back and forth.



The oscillating motion is powered at right angles. The input shaft, in making full rotations, causes the output shaft to oscillate 120°.



A constant-speed-ratio universal is obtained by placing two "inversions" back-to-back. Motion is transmitted up to a 75º misalignment.



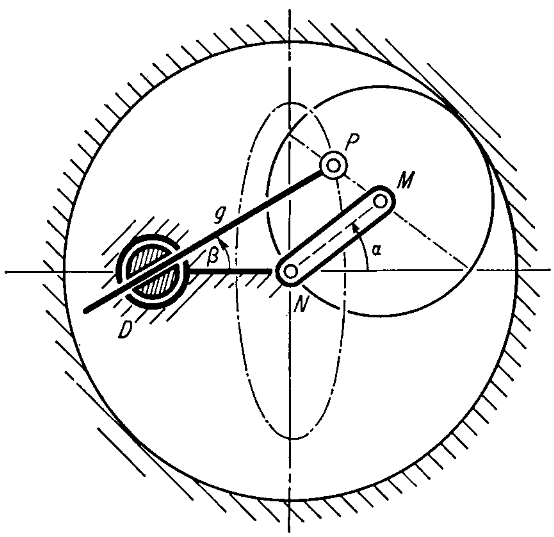
A right-angle limited-stroke drive transmits an exact motion pattern. A multiplicity of fittings can be operated from a common shaft.

The Elliptical Slide Drive

The output motion, β, of a spherical slide oscillator can be duplicated with a two-dimensional "elliptical slide." The mechanism has a link g that slides through a pivot point D and is fastened to a point P moving along an elliptical path. The ellipse can be generated by a Cardan drive, which is a planetary gear system whose planet gear has half the diameter of its internal gear. The center of the planet, point M, describes a circle; any point on its periphery describes a straight line, and any point in between, such as point P, describes an ellipse.

There are special relationships between the dimensions of the 3-D spherical slide and the 2-D elliptical slide: $\tan \gamma/\sin \delta = a/d$ and $\tan \gamma/\cot \delta = b/d$, where a is the major half-axis, b the minor half-axis of the ellipse, and d is the length of the fixed link DN. The minor axis lies along this link.

If point D is moved within the ellipse, a completely rotating output is obtained, corresponding to the rotating spherical crank slide.



INCHWORM ACTUATOR

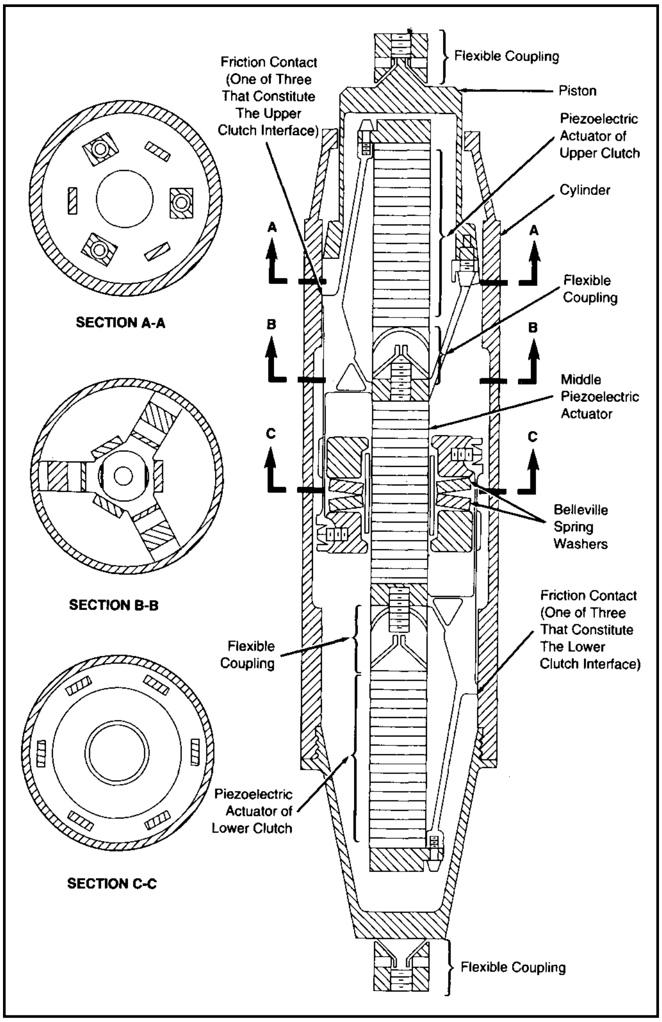
This actuator would serve as an active truss member.
NASA's Jet Propulsion Laboratory, Pasadena, California

A proposed inchworm actuator could be used as an active truss member. Its length of which could be varied slowly to change the configuration of the truss rapidly or simply to counteract vibrations. The overall stroke of the actuator could range from about 3 cm at a frequency of 1 Hz down to 0.002 cm at a frequency of 1 kHz. The length of the stroke could then be controlled with an accuracy of 0.0001 cm.

The inchworm actuator would incorporate three piezoelectric actuators (see figure). The upper and lower piezoelectric actuators would unlock normally locked clutches. The middle piezoelectric actuator would enforce small variations of the distance between the actuator's clutches.

Belleville washers would apply a compression preload to the piezoelectric actuators and the clutches, isolating the piezoelectric devices from tensile stress and keeping the clutches normally locked so that they would maintain the overall length of the actuator without power. A bearing would position the actuator piston laterally in the cylinder. Usually, at least one of the clutches would remain locked. This would prevent the piston from rotating in the cylinder. Flexible couplings and tripod-piston supports in the clutches would accommodate misalignments and fabrication tolerances when the clutch was locked. Any bending loads on the piston would be carried primarily through a direct load path to the cylinder, and only a fraction of the bending load would be carried through the piezoelectric devices. The use of the clutch as a lateral support for the piston would also reduce the cloth stroke needed to accommodate fabrication tolerances in the clutch interfaces.

This work was done by Robert M. Bamford of Caltech for **NASA's Jet Propulsion Laboratory.**



This inchworm actuator would hold its position (that is, it would neither extend nor retract) when electrical energy was not supplied. The maximum end-to-end stroke (extension or retraction of the piston with respect to the cylinder) would be 3 cm.