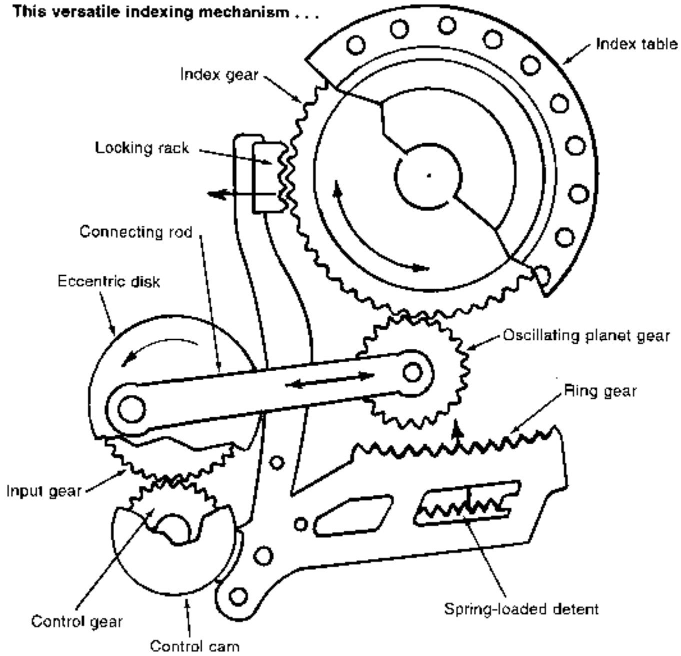
CHAPTER 4

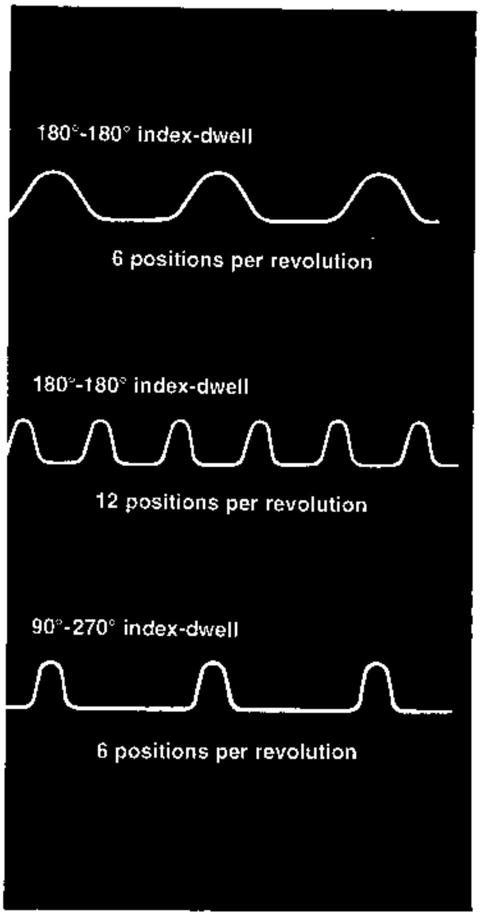
RECIPROCATING AND GENERAL-PURPOSE MECHANISMS

GEARS AND ECCENTRIC DISK COMBINE IN QUICK INDEXING

This versatile indexing mechanism . . .



. . provides choice of indexing modes



Both stops and dwell are adjustable.

An ingenious intermittent mechanism with its multiple gears, gear racks, and levers provides smoothness and flexibility in converting constant rotary motion into a start-and-stop type of indexing.

It works equally well for high-speed operations, as fast as 2 seconds per cycle, including index and dwell, or for slow-speed assembly functions.

The mechanism minimizes shock loads and offers more versatility than the indexing cams and genevas usually employed to convert rotary motion into start-stop indexing. The number of stations (stops) per revolution of the table can easily be changed, as can the period of dwell during each stop.

Advantages. This flexibility broadens the scope of such automatic machine operations as feeding, sorting, packaging, and weighing that the rotary table can perform. But the design offers other advantages, too:

- Gears instead of cams make the mechanism cheaper to manufacture, because gears are simpler to machine.
- The all-mechanical interlocked system achieves an absolute time relationship between motions.
- Gearing is arranged so that the machine automatically goes into a dwell when it is overloaded, preventing damage during jam-ups.
- Its built-in anti-backlash gear system averts rebound effects, play, and lost motion during stops.

How it works. Input from a single motor drives an eccentric disk and connecting rod. In the position shown in the drawing, the indexing gear and table are locked by the rack—the planet gear rides freely across the index gear without imparting any motion to it. Indexing of the table to its next position begins when the control cam simultaneously releases the locking rack from the index gear and causes the spring control ring gear to pivot into mesh with the planet.

This is a planetary gear system containing a stationary ring gear, a driving planet gear, and a "sun" index gear. As the crank keeps moving to the right, it begins to accelerate the index gear with harmonic motion—a desirable type of motion because of its low acceleration-deceleration characteristics—while it is imparting high-speed transfer to the table.

At the end of 180º rotation of the crank, the control cam pivots the ring-gear segment out of mesh and, simultaneously, engages the locking rack. As the connecting rod is drawn back, the planet gear rotates freely over the index gear, which is locked in place.

The cam control is so synchronized that all toothed elements are in full engagement briefly when the crank arm is in full toggle at both the beginning and end of index. The device can be operated just as easily in the other direction.

Overload protection. The ring gear segment includes a spring-load detent mechanism (simplified in the illustration) that will hold the gearing in full engagement under normal indexing forces. If rotation of the table is blocked at any point in index, the detent spring force is overcome and the ring gear pops out of engagement with the planet gear.

A detent roller (not shown) will then snap into a second detent position, which will keep the ring gear free during the remainder of the index portion of the cycle. After that, the detent will automatically reset itself.

Incomplete indexing is detected by an electrical system that stops the machine at the end of the index cycle.

Easy change of settings. To change indexes for a new job setup, the eccentric is simply replaced with one heaving a different crank radius, which gives the proper drive stroke for 6, 8, 12, 16, 24, 32, or 96 positions per table rotation.

Because indexing occurs during one-half revolution of the eccentric disk, the input gear must rotate at two or three times per cycle to accomplish indexing of ½, ¼, or 1⁄16 of the total cycle time (which is the equivalent to index-to-dwell cycles of 180/180º, 90/270º or 60/300º). To change the cycle time, it is only necessary to mount a difference set of change gears between input gear and control cam gear.

TIMING BELTS, FOUR-BAR LINKAGE TEAM UP FOR SMOOTH INDEXING

A class of intermittent mechanisms based on timing belts, pulleys, and linkages (see drawing) instead of the usual genevas or cams is capable of cyclic start-and-stop motions with smooth acceleration and deceleration.

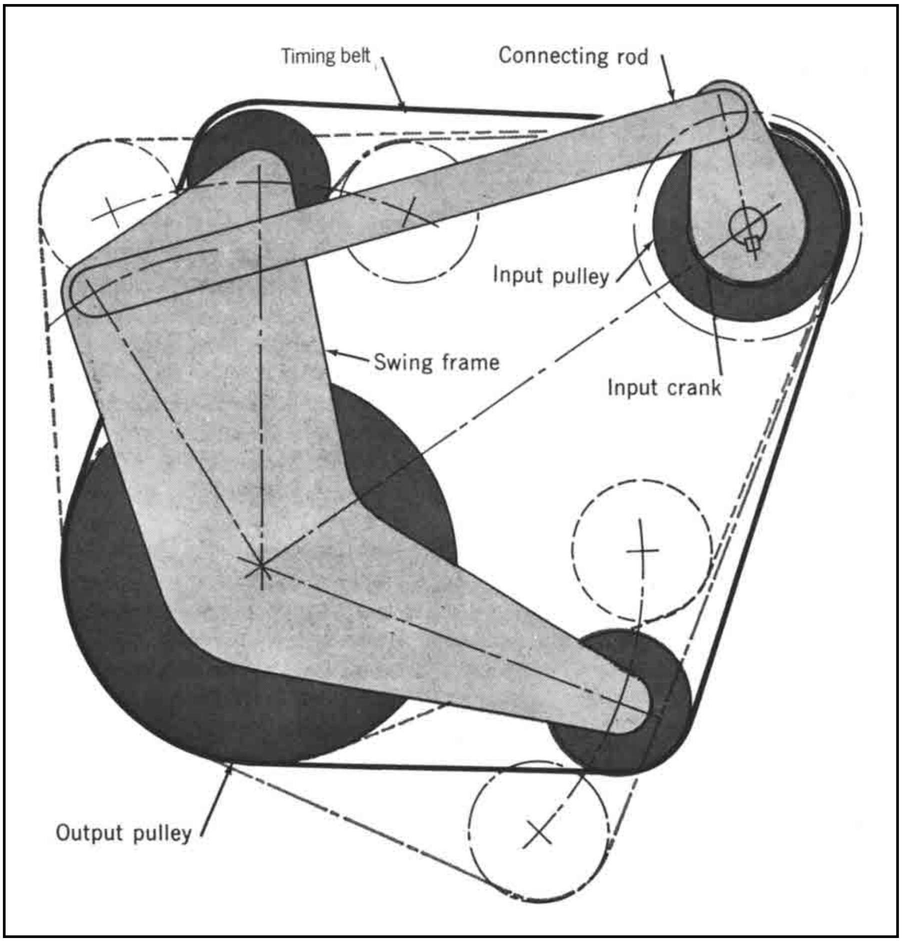
Developed by Eric S. Buhayar and Eugene E. Brown of the Engineering Research Division, Scott Paper Co. (Philadelphia), the mechanisms are employed in automatic assembly lines.

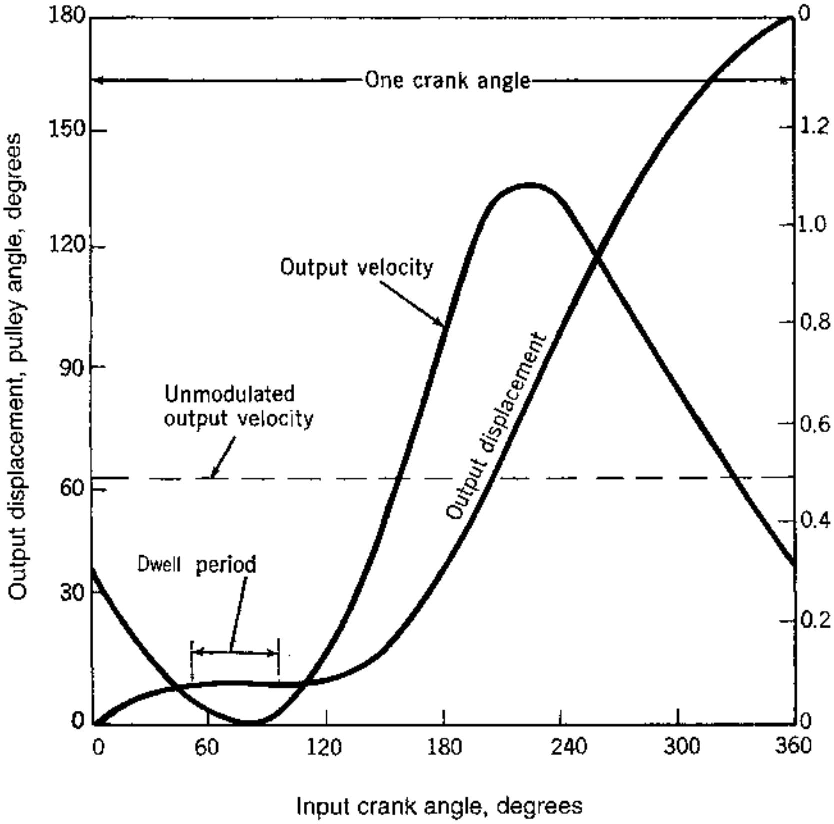
These mechanisms, moreover, can function as phase adjusters in which the rotational position of the input shaft can be shifted as desired in relation to the output shaft. Such phase adjusters have been used in the textile and printing industries to change the "register" of one roll with that of another, when both rolls are driven by the same input.

Outgrowth from chains. Intermittent-motion mechanisms typically have ingenious shapes and configurations. They have been used in watches and in production machines for many years. There has been interest in the chain type of intermittent mechanism (see drawing), which ingeniously routes a chain around four sprockets to produce a dwell-and-index output.

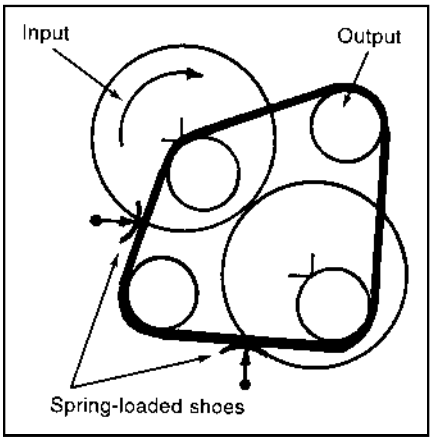
The input shaft of such a device has a sprocket eccentrically fixed to it. The input also drives another shaft through one-to-one gearing. This second shaft mounts a similar eccentric sprocket that is, however, free to rotate. The chain passes first around an idler pulley and then around a second pulley, which is the output.

As the input gear rotates, it also pulls the chain around with it, producing a





modulated output rotation. Two spring-loaded shoes, however, must be employed because the perimeter of the pulleys is not a constant figure, so the drive has varying slack built into it.



Commercial type. A chain also links the elements of a commercial phase-adjuster drive. A handle is moved to change the phase between the input and output shafts. The theoretical chain length is constant.

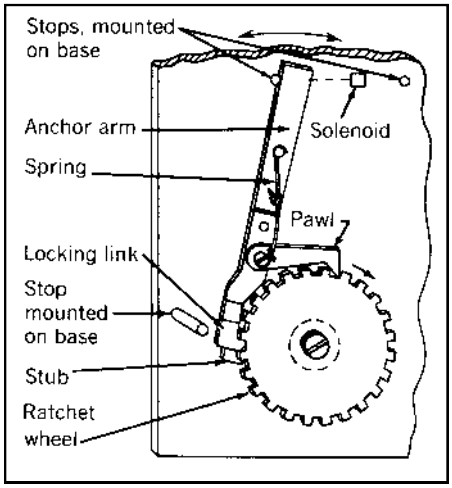
In trying to improve this chain device, Scott engineers decided to keep the input and output pulleys at fixed positions and maintain the two idlers on a swing frame. The variation in wraparound length turned out to be surprisingly little, enabling them to install a timing belt without spring-loaded tensioners instead of a chain.

If the swing frame is held in one position, the intermittent mechanism produces a constant-speed output. Shifting the swing frame to a new position automatically shifts the phase relationship between the input and output.

Computer consulted. To obtain intermittent motion, a four-bar linkage is superimposed on the mechanism by adding a crank to the input shaft and a connecting rod to the swing frame. The developers chose an iterative program on a computer to optimize certain variables of the four-bar version.

In the design of one two-stop drive, a dwell period of approximately 50º is obtained. The output displacement moves slowly at first, coming to a “pseudo dwell,” in which it is virtually stationary. The output then picks up speed smoothly until almost two-thirds of the input rotation has elapsed (240º). After the input crank completes a full circle of rotation, it continues at a slower rate and begins to repeat its slow-down—dwell—speed-up cycle.

MODIFIED RATCHET DRIVE



A ratchet drive was designed to assure movement, one tooth at a time, in only one direction, without overriding. The key element is a small stub that moves along from the bottom of one tooth well, across the top of the tooth, and into an adjacent tooth well, while the pawl remains at the bottom of another tooth well.

The locking link, which carries the stub along with the spring, comprises a system that tends to hold the link and pawl against the outside circumference of the wheel and to push the stub and pawl point toward each other and into differently spaced wells between the teeth. A biasing element, which might be another linkage or solenoid, is provided to move the anchor arm from one side to the other, between the stops, as shown by the double arrow. The pawl will move from one tooth well to the next tooth well only when the stub is at the bottom of a tooth well and is in a position to prevent counter-rotation.

ODD SHAPES IN PLANETARY GIVE SMOOTH STOP AND GO

This intermittent-motion mechanism for automatic processing machinery combines gears with lobes; some pitch curves are circular and some are noncircular.

This intermittent-motion mechanism combines circular gears with noncircular gears in a planetary arrangement, as shown in the drawing.

The mechanism was developed by Ferdinand Freudenstein, a professor of mechanical engineering at Columbia University. Continuous rotation applied to the input shaft produces a smooth, stop-and-go unidirectional rotation in the output shaft, even at high speeds.

This jar-free intermittent motion is sought in machines designed for packaging, production, automatic transfer, and processing.

Varying differential. The basis for Freudenstein's invention is the varying differential motion obtained between two sets of gears. One set has lobular pitch circles whose curves are partly circular and partly noncircular.

The circular portions of the pitch curves cooperate with the remainder of the mechanism to provide a dwell time or stationary phase, or phases, for the output member. The non-circular portions act with the remainder of the mechanism to provide a motion phase, or phases, for the output member.

Competing genevas. The main competitors to Freudenstein's "pulsating planetary" mechanism are external genevas and starwheels. These devices have a number of limitations that include:

- Need for a means, separate from the driving pin, for locking the output member during the dwell phase of the motion. Moreover, accurate manufacture and careful design are required to make a smooth transition from rest to motion and vice versa.
- Kinematic characteristics in the geneva that are not favorable for high-speed operation, except when the number of stations (i.e., the number of slots in the output member) is large. For example, there is a sudden change of acceleration of the output member at the beginning and end of each indexing operation.
- Relatively little flexibility in the design of the geneva mechanism. One factor alone (the number of slots in the output member) determines the characteristics of the motion. As a result, the ratio of the time of motion to the time of dwell cannot exceed one-half, the output motion cannot be uniform for any finite portion of the indexing cycle, and it is always opposite in sense to the sense of input rotation. The output shaft, moreover, must always be offset from the input shaft.

Many modifications of the standard external geneva have been proposed, including multiple and unequally spaced driving pins, double rollers, and separate entrance and exit slots. These proposals have, however, been only partly successful in overcoming these limitations.

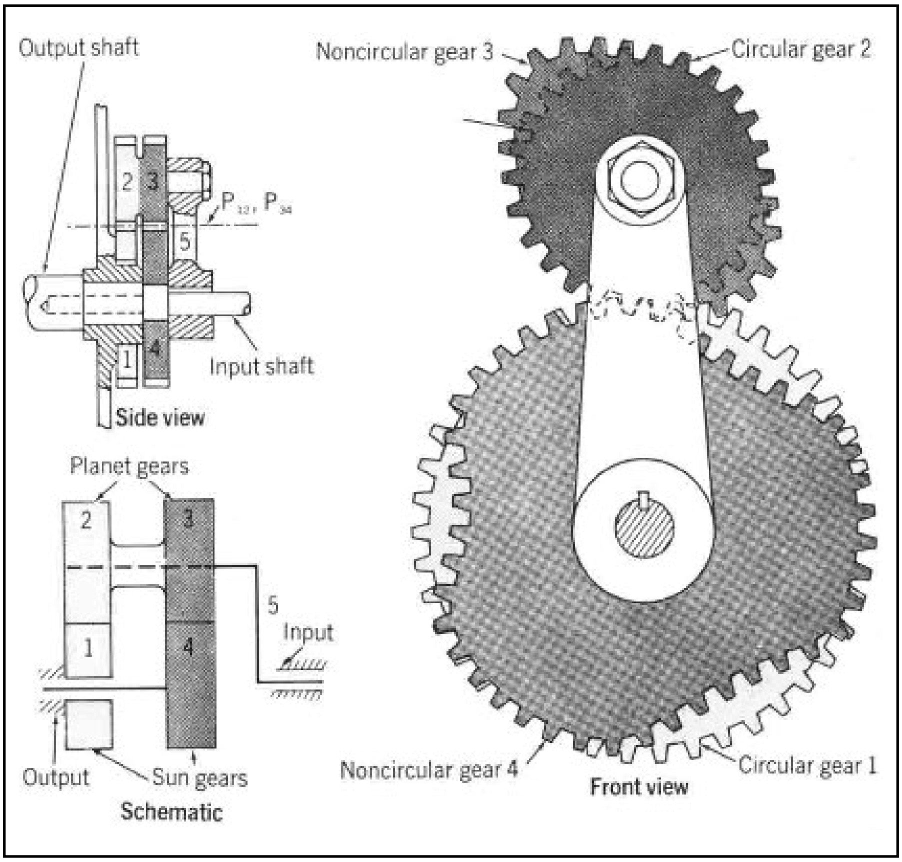
Differential motion. In deriving the operating principle of his mechanism, Freudenstein first considered a conventional epicyclic (planetary) drive in which the input to the cage or arm causes a planet set with gears *2* and *3* to rotate the output "sun," gear *4*, while another sun, gear *1*, is kept fixed (see drawing).

Letting r_1, r_2, r_3, r_4, equal the pitch radii of the circular *1*, *2*, *3*, *4*, then the output ratio, defined as:

$$R = \frac{\text{angular velocity of output gear}}{\text{angular velocity of arm}}$$

is equal to: $R = 1 - \frac{r_1 r_3}{r_2 r_4}$

Now, if $r_1 = r_4$ and $r_2 = r_3$, there is no "differential motion" and the output remains stationary. Thus if one gear pair, say *3* and *4*, is made partly circular and partly noncircular, then where $r_2 = r_3$ and $r_1 = r_4$ for the circular portion, gear *4* dwells. Where $r_2 \neq r_3$ and $r_1 \neq r_4$ for the noncircular portion, gear *4* has motion. The magnitude of this motion depends



At heart of new planetary (in front view, circular set stacked behind noncircular set), two sets of gears when assembled (side view) resemble conventional unit (schematic).

on the difference in radii, in accordance with the previous equation. In this manner, gear *4* undergoes an intermittent motion (see graph).

Advantages. The pulsating planetary approach demonstrates some highly useful characteristics for intermittent-motion machines:

- The gear teeth serve to lock the output member during the dwell as well as to drive that member during motion.
- Superior high-speed characteristics are obtainable. The profiles of the pitch curves of the noncircular gears can be tailored to a wide variety of desired kinematic and dynamic characteristics. There need be no sudden terminal acceleration change of the driven member, so the transition from dwell to motion, and vice versa, will be smooth, with no jarring of machine or payload.
- The ratio of motion to dwell time is adjustable within wide limits. It can even exceed unity, if desired. The number of indexing operations per revolution of the input member also can exceed unity.
- The direction of rotation of the output member can be in the same or opposite sense relative to that of the input member, according to whether the pitch axis P_{34} for the noncircular portions of gears *3* and *4* lies wholly outside or wholly inside the pitch surface of the planetary sun gear *1*.
- Rotation of the output member is coaxial with the rotation of the input member.
- The velocity variation during motion is adjustable within wide limits. Uniform output velocity for part of the indexing cycle is obtainable; by varying the number and shape of the lobes, a variety of other desirable motion characteristics can be obtained.
- The mechanism is compact and has relatively few moving parts, which can be readily dynamically balanced.

Design hints. The design techniques work out surprisingly simply, said Freudenstein. First the designer must select the number of lobes L_3 and L_4 on the gears *3* and *4*. In the drawings, $L_3 = 2$ and $L_4 = 3$. Any two lobes on the two gears (i.e., any two lobes of which one is on one gear and the other on the other gear) that are to mesh together must have the same arc length. Thus, every lobe on gear *3* must mesh with every lobe on gear *4*, and $T_3/T_4 = L_3/L_4 = 2/3$, where T_3 and T_4 are the numbers of teeth on gears *3* and *4*. T_1 and T_2 will denote the numbers of teeth on gears *1* and *2*.

Next, select the ratio S of the time of motion of gear *4* to its dwell time, assuming a uniform rotation of the arm *5*. For the gears shown, $S = 1$. From the geometry,

$$(\theta_{30} + \Delta\theta_{30})L_3 = 360°$$

and

$$S = \Delta\theta_3/\theta_{30}$$

Hence

$$\theta_{30}(1 + S)L_3 = 360°$$

For $S = 1$ and $L_3 + 2$,

$$\theta_{30} = 90°$$

and

$$\Delta\theta_3 = 90°$$

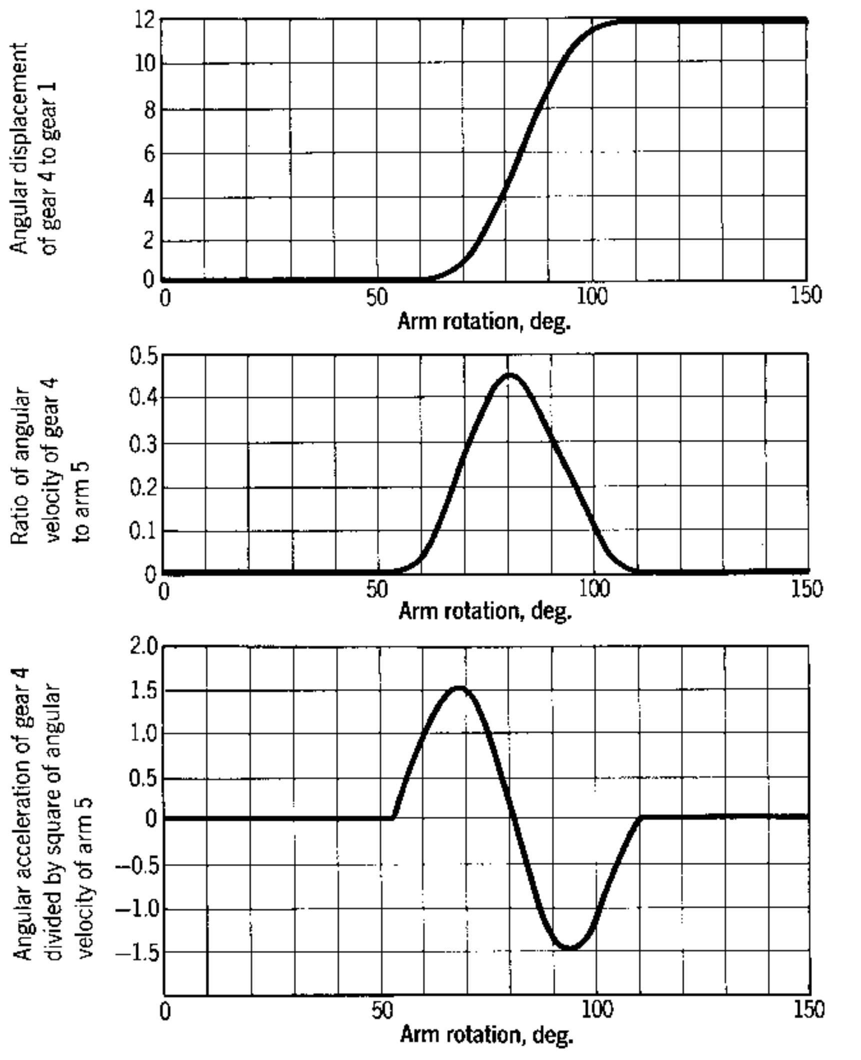
Now select a convenient profile for the noncircular portion of gear *3*. One profile (see the profile drawing) that Freudenstein found to have favorable high-speed characteristics for stop-and-go mechanisms is

$$r_3 = R_3$$

$$\left[1 + \frac{\lambda}{2}\left(1 - \cos\frac{2\pi(\theta_3 - \theta_{30})}{\Delta\theta_3}\right)\right]$$

The profile defined by this equation has, among other properties, the characteristic that, at transition from rest to motion and vice versa, gear *4* will have zero acceleration for the uniform rotation of arm *5*.

In the above equation, λ is the quantity which, when multiplied by R^3, gives the maximum or peak value of $r_3 - R_3$, differing by an amount h' from the radius R^3 of the circular portions of the gear. The noncircular portions of each lobe are, moreover, symmetrical about their midpoints, the midpoints of these portions being indicated by *m*.



Output motion (upper curve) has long dwell periods; velocity curve (center) has smooth transition from zero to peak; acceleration at transition is zero (bottom).

To evaluate the quantity λ, Freudenstein worked out the equation:

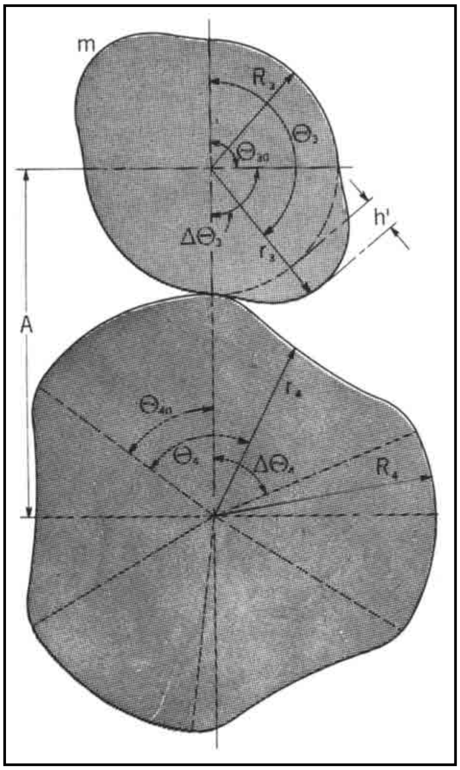
$$\lambda = \frac{1-\mu}{\mu} \times \frac{[S+\alpha-(1+\alpha)\mu][\alpha-S-(1+\alpha)\mu]}{[\alpha-(1+\alpha)\mu]^2}$$

where $R_3\lambda$ = height of lobe

$$\mu = \frac{R_3}{A} = R_3/(R_3+R_4)$$

$$\alpha = S + (1+S)\,L_3/L_4$$

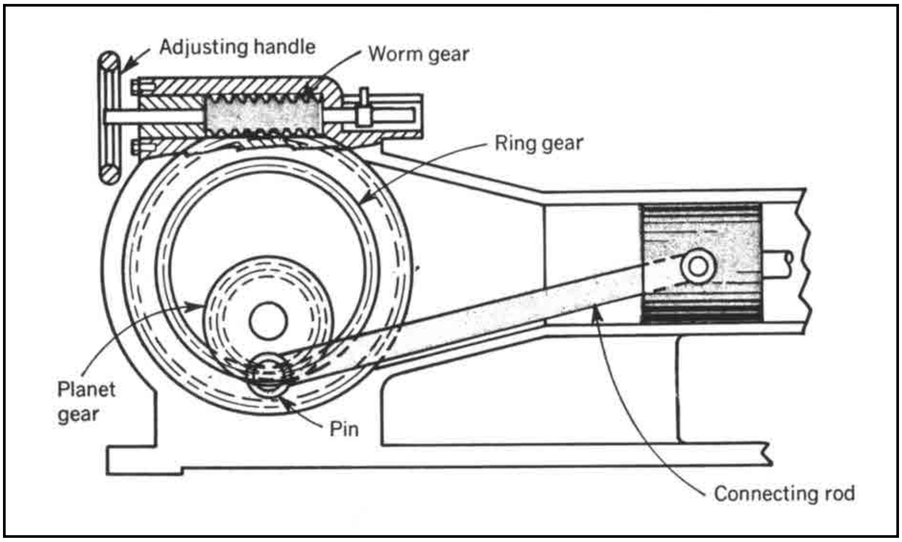
To evaluate the equation, select a suitable value for μ that is a reasonably simple rational fraction, i.e., a fraction such as 3/8 whose numerator and denominator are reasonably small integral numbers.



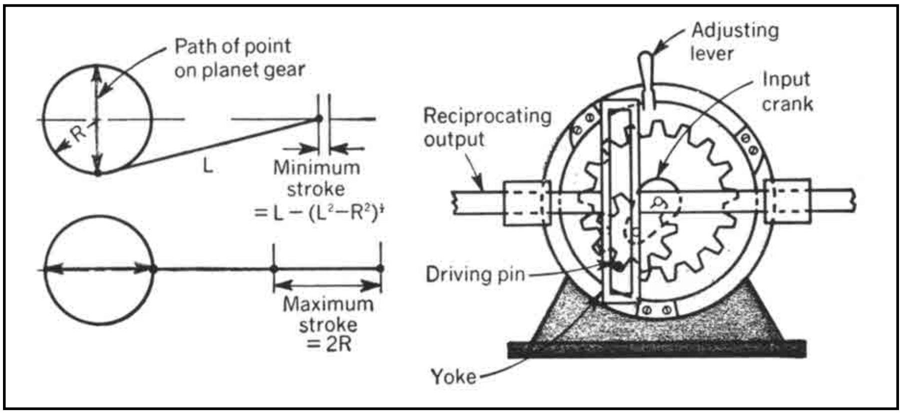
Profiles for noncircular gears are circular arcs blended to special cam curves.

Thus, without a computer or lengthy trial-and-error procedures, the designer can select the configuration that will achieve his objective of smooth intermittent motion.

CYCLOID GEAR MECHANISM CONTROLS STROKE OF PUMP



An adjustable ring gear meshes with a planet gear having half of its diameter to provide an infinitely variable stroke in a pump. The adjustment in the ring gear is made by engaging other teeth. In the design below, a yoke replaces the connecting rod.



A metering pump for liquid or gas has an adjustable ring gear that meshes with a special-size planet gear to provide an infinitely variable stroke in the pump. The stroke can be set manually or automatically when driven by a servomotor. Flow control from 180 to 1200 liter/hr. (48 to 317 gal./hr.) is possible while the pump is at a standstill or running.

Straight-line motion is key. The mechanism makes use of a planet gear whose diameter is half that of the ring gear. As the planet is rotated to roll on the inside of the ring, a point on the pitch diameter of the planet will describe a straight line (instead of the usual hypocycloid curve). This line is a diameter of the ring gear. The left end of the connecting rod is pinned to the planet at this point.

The ring gear can be shifted if a second set of gear teeth is machined in its outer surface. This set can then be meshed with a worm gear for control. Shifting the ring gear alters the slope of the straight-line path. The two extreme positions are shown in the diagram. In the position of the mechanism shown, the pin will reciprocate vertically to produce the minimum stroke for the piston. Rotating the ring gear 90º will cause the pin to reciprocate horizontally to produce the maximum piston stroke.

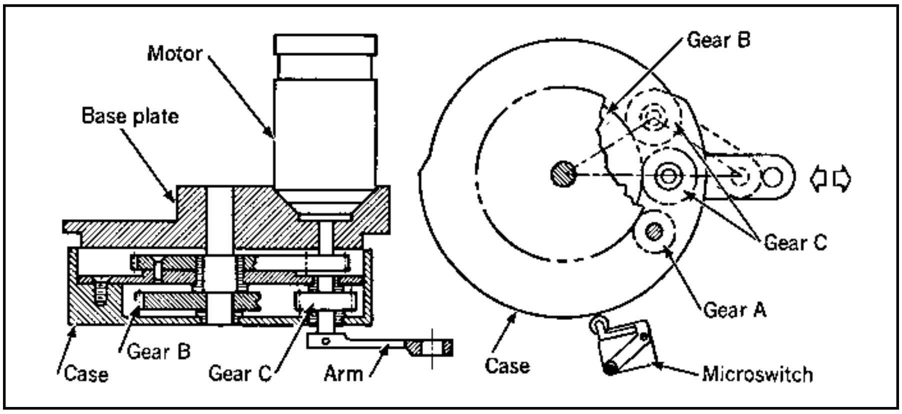
The second diagram illustrates another version that has a yoke instead of a connecting rod. This permits the length of the stroke to be reduced to zero. Also, the length of the pump can be substantially reduced.

CONVERTING ROTARY-TO-LINEAR MOTION

A compact gear system that provides linear motion from a rotating shaft was designed by Allen G. Ford of The Jet Propulsion Laboratory in California. It has a planetary gear system so that the end of an arm attached to the planet gear always moves in a linear path (drawing).

The gear system is set in motion by a motor attached to the base plate. Gear *A*, attached to the motor shaft, turns the case assembly, causing Gear *C* to rotate along Gear *B*, which is fixed. The arm is the same length as the center distance between Gears *B* and *C*. Lines between the centers of Gear *C*, the end of the arm, and the case axle form an isosceles triangle, the base of which is always along the plane through the center of rotation. So the output motion of the arm attached to Gear *C* will be in a straight line.

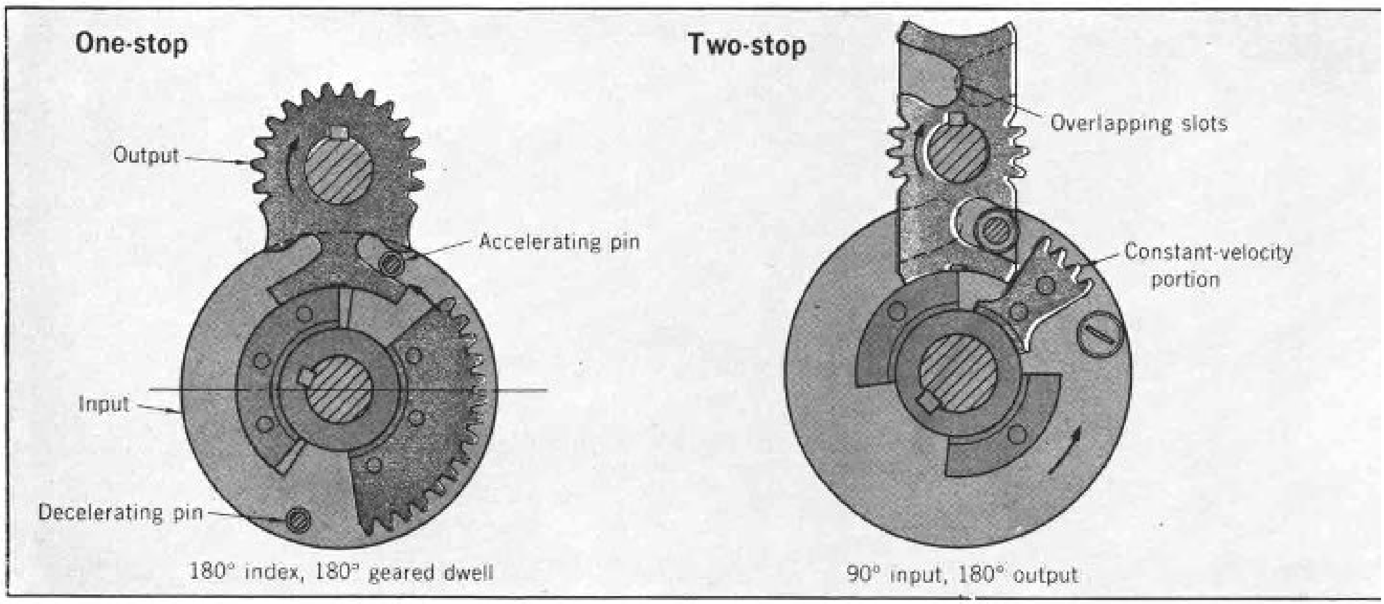
When the end of travel is reached, a switch causes the motor to reverse, returning the arm to its original position.



The end of arm moves in a straight line because of the triangle effect (right).

NEW STAR WHEELS CHALLENGE GENEVA DRIVES FOR INDEXING

Star wheels with circular-arc slots can be analyzed mathematically and manufactured easily.



Star Wheels vary in shape, depending on the degree of indexing that must be done during one input revolution.

A family of star wheels with circular instead of the usual epicyclic slots (see drawings) can produce fast start-and-stop indexing with relatively low acceleration forces.

This rapid, jar-free cycling is important in a wide variety of production machines and automatic assembly lines that move parts from one station to another for drilling, cutting, milling, and other processes.

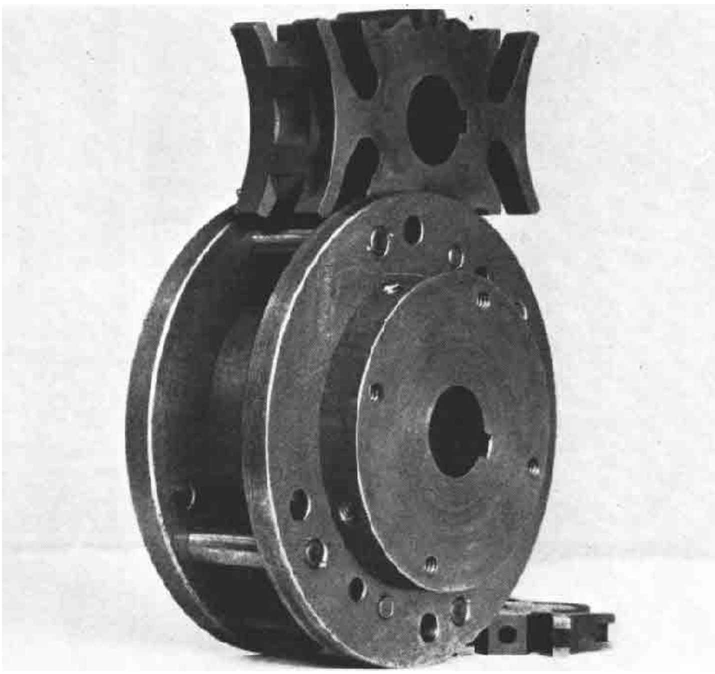
The circular-slot star wheels were invented by Martin Zugel of Cleveland, Ohio.

The motion of older star wheels with epicyclic slots is difficult to analyze and predict, and the wheels are hard to make.

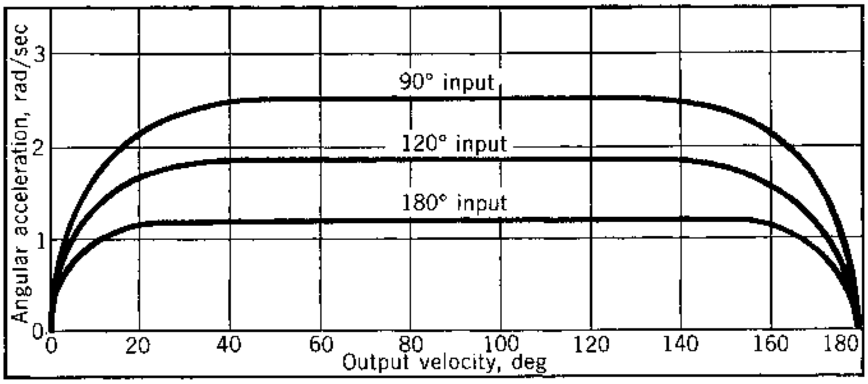
The star wheels with their circular-arc slots are easy to fabricate, and because the slots are true circular arcs, they can be visualized for mathematical analysis as four-bar linkages during the entire period of pin-slot engagement.

Strong points. With this approach, changes in the radius of the slot can be analyzed and the acceleration curve varied to provide inertia loads below those of the genevas for any practical design requirement.

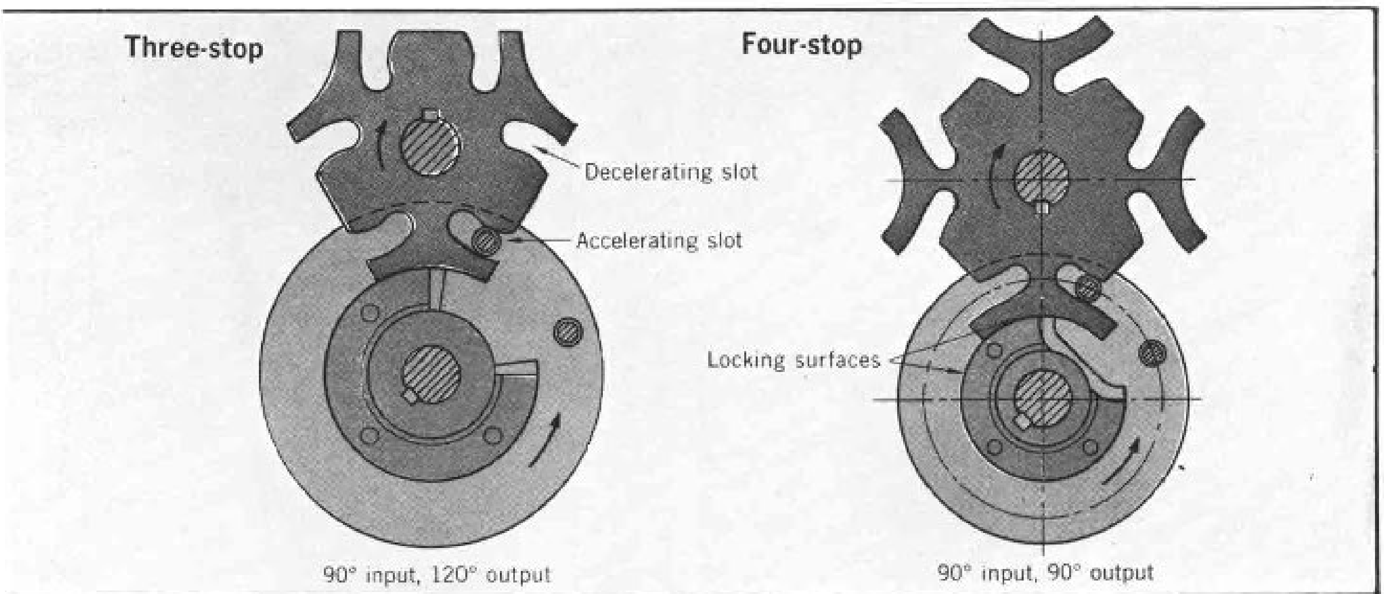
Another advantage of the star wheels is that they can index a full 360° in a relatively short period (180°). Such one-stop operation is not possible with genevas. In fact, genevas cannot do two-stop operations, and they have difficulty producing three stops per index. Most two-stop indexing devices available are cam-operated, which means they require greater input angles for indexing.



Geared star sector indexes smoothly a full 360º during a 180º rotation of the wheel, then it pauses during the other 180º to allow the wheel to catch up.



The one-stop index motion of the unit can be designed to take longer to complete its indexing, thus reducing its index velocity.



An accelerating pin brings the output wheel up to speed. Gear sectors mesh to keep the output rotating beyond 180º.

Operating sequence. In operation, the input wheel rotates continuously. A sequence starts (see drawing) when the accelerating pin engages the curved slot to start indexing the output wheel clockwise. Simultaneously, the locking surface clears the right side of the output wheel to permit the indexing.

Pin C in the drawings continues to accelerate the output wheel past the midpoint, where a geneva wheel would start deceleration. Not until the pins are symmetrical (see drawing) does the acceleration end and the deceleration begin. Pin D then takes the brunt of the deceleration force.

Adaptable. The angular velocity of the output wheel, at this stage of exit of the acceleration roller from Slot 1, can be varied to suit design requirements. At this point, for example, it is possible either to engage the deceleration roller as described or to start the engagement of a constant-velocity portion of the cycle. Many more degrees of output index can be obtained by interposing gear-element segments between the acceleration and deceleration rollers.

The star wheel at left will stop and start four times in making one revolution, while the input turns four times in the same period. In the starting position, the output link has zero angular velocity, which is a prerequisite condition for any star wheel intended to work at speeds above a near standstill.

In the disengaged position, the angular velocity ratio between the output and input shafts (the "gear" ratio) is entirely dependent upon the design angles α and β and independent of the slot radius, r.

Design comparisons. The slot radius, however, plays an important role in the mode of the acceleration forces. A four-stop geneva provides a good basis for comparison with a four-stage "Cyclo-Index" system.

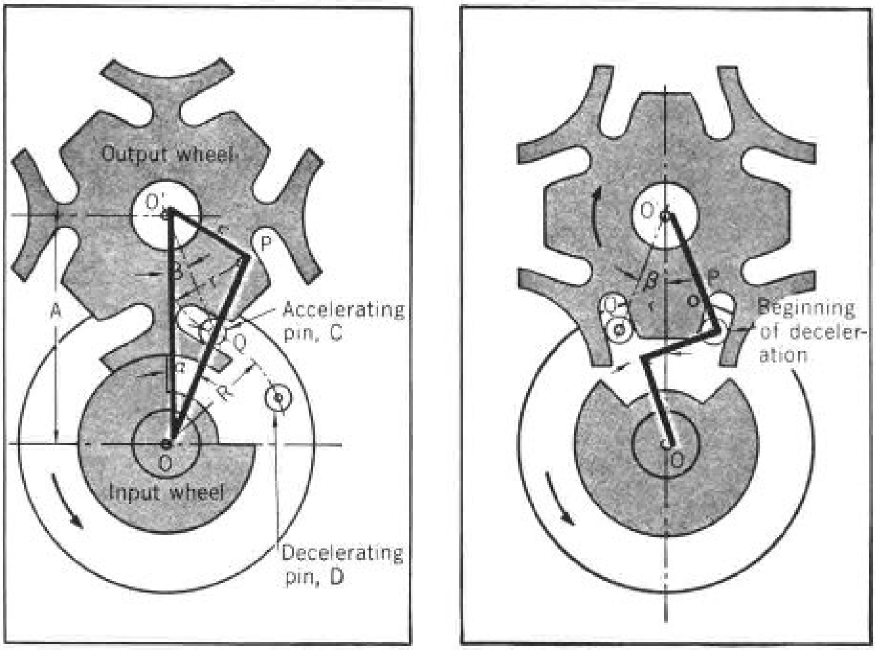
Assume, for example, that $\alpha = \beta = 22.5°$. Application of trigonometry yields:

$$R = A\left[\frac{\sin\beta}{\sin(\alpha+\beta)}\right]$$

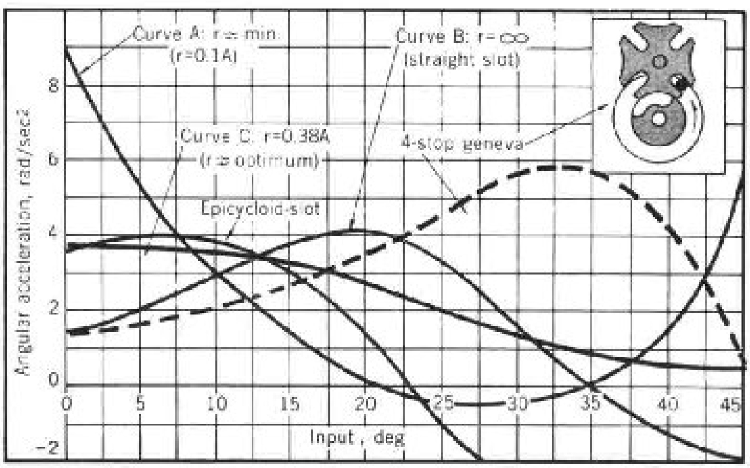
which yields $R = 0.541A$. The only restriction on r is that it be large enough to allow the wheel to pass through its mid-position. This is satisfied if:

$$r > \frac{RA(1-\cos\alpha)}{A - 2R - A\cos\alpha} \approx 0.1A$$

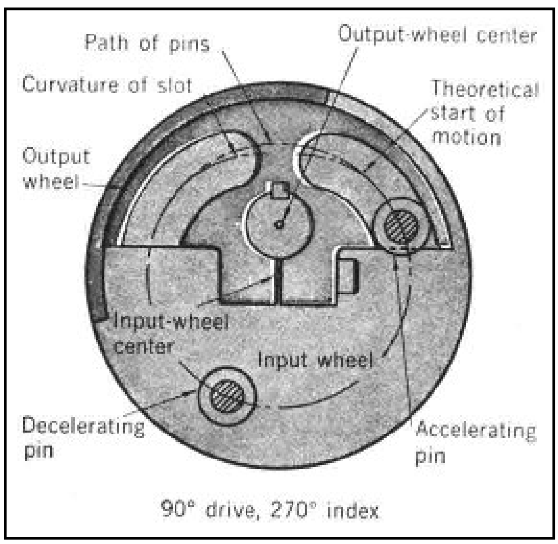
There is no upper limit on r, so that slot can be straight.



Star-wheel action is improved with curved slots over the radius r, centered on the initial-contact line OP. The units then act as four-bar linkages, 00[1]PQ.



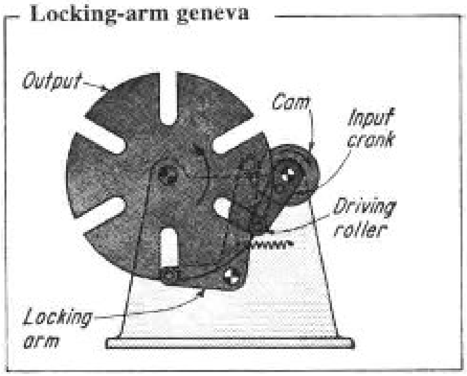
The accelerating force of star wheels (curves A, B, C) varies with input rotation. With an optimum slot (curve C), it is lower than for a four-stop geneva.



This internal star wheel has a radius difference to cushion the indexing shock.

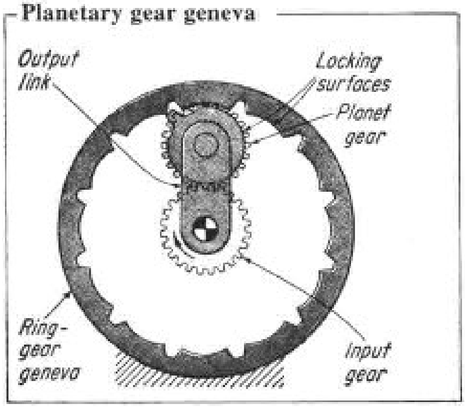
GENEVA MECHANISMS

Locking-arm geneva



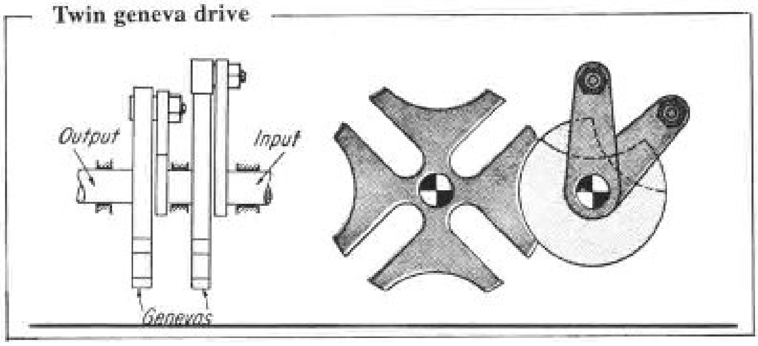
The driving follower on the rotating input crank of this geneva enters a slot and rapidly indexes the output. In this version, the roller of the locking-arm (shown leaving the slot) enters the slot to prevent the geneva from shifting when it is not indexing.

Planetary gear geneva



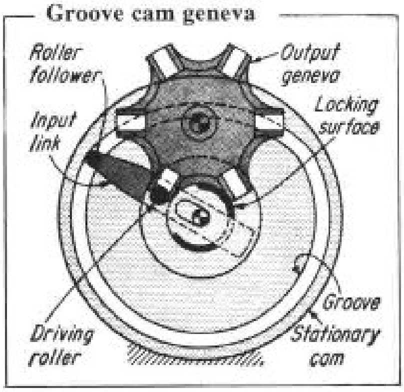
The output link remains stationary while the input gear drives the planet gear with single tooth on the locking disk. The disk is part of the planet gear, and it meshes with the ring-gear geneva to index the output link one position.

Twin geneva drive



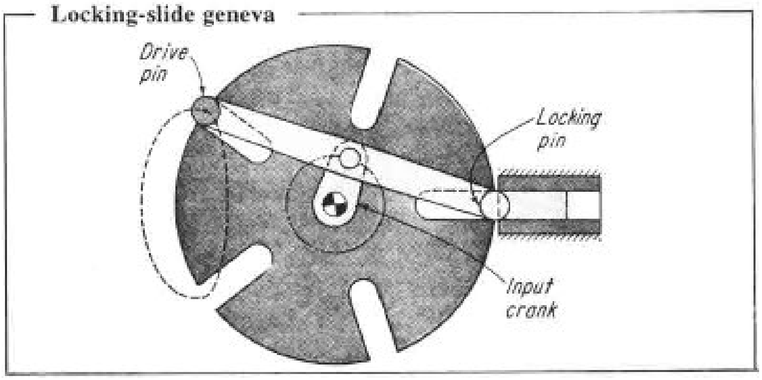
The driven member of the first geneva acts as the driver for the second geneva. This produces a wide variety of output motions including very long dwells between rapid indexes.

Groove cam geneva



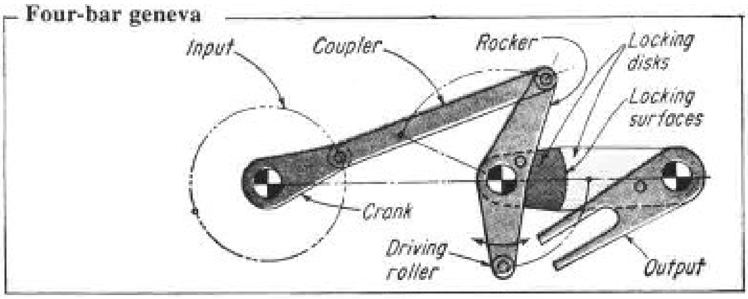
When a geneva is driven by a roller rotating at a constant speed, it tends to have very high acceleration and deceleration characteristics. In this modification, the input link, which contains the driving roller, can move radially while being rotated by the groove cam. Thus, as the driving roller enters the geneva slot, it moves radially inward. This action reduces the geneva acceleration force.

Locking-slide geneva



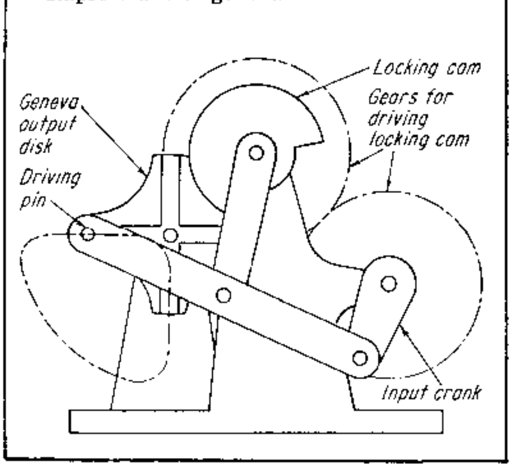
One pin locks and unlocks the geneva; the second pin rotates the geneva during the unlocked phase. In the position shown, the drive pin is about to enter the slot to index the geneva. Simultaneously, the locking pin is just clearing the slot.

Four-bar geneva



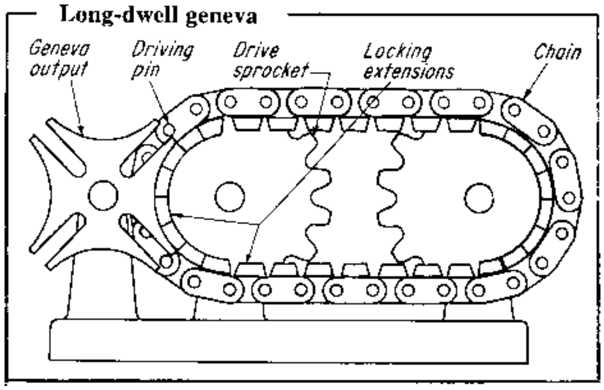
A four-bar geneva produces a long-dwell motion from an oscillating output. The rotation of the input wheel causes a driving roller to reciprocate in and out of the slot of the output link. The two disk surfaces keep the output in the position shown during the dwell period.

Rapid-transfer geneva



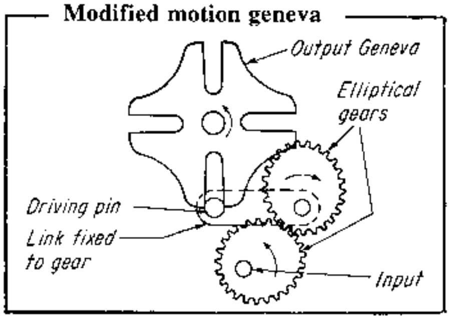
The coupler point at the extension of the connecting link of the four-bar mechanism describes a curve with two approximately straight lines, 90º apart. This provides a favorable entry situation because there is no motion in the geneva while the driving pin moves deeply into the slot. Then there is an extremely rapid index. A locking cam, which prevents the geneva from shifting when it is not indexing, is connected to the input shaft through gears.

Long-dwell geneva



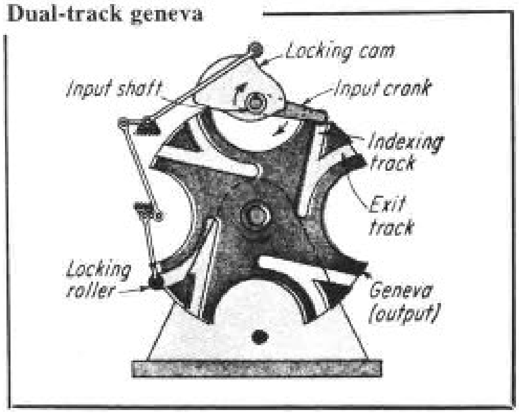
This geneva arrangement has a chain with an extended pin in combination with a standard geneva. This permits a long dwell between each 90º shift in the position of the geneva. The spacing between the sprockets determines the length of dwell. Some of the links have special extensions to lock the geneva in place between stations.

Modified motion geneva



The input link of a normal geneva drive rotates at constant velocity, which restricts flexibility in design. That is, for given dimensions and number of stations, the dwell period is determined by the speed of the input shaft. Elliptical gears produce a varying crank rotation that permits either extending or reducing the dwell period.

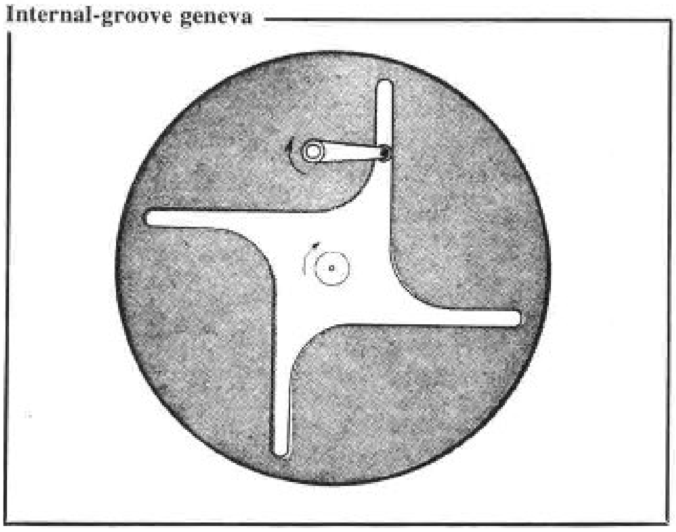
Dual-track geneva



The key consideration in the design of genevas is to have the input roller enter and leave the geneva slots tangentially (as the crank rapidly indexes the output). This is accomplished in the novel mechanism shown with two tracks. The roller enters one track, indexes the geneva 90º (in a four-stage geneva), and then automatically follows the exit slot to leave the geneva.

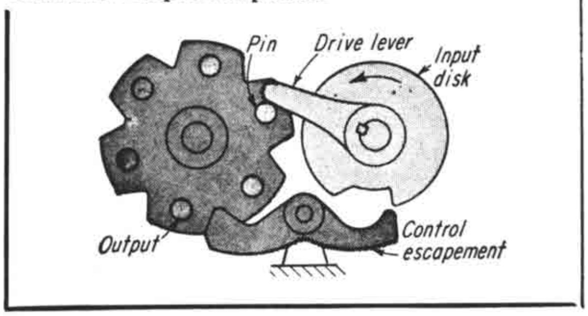
The associated linkage mechanism locks the geneva when it is not indexing. In the position shown, the locking roller is just about to exit from the geneva.

Internal-groove geneva



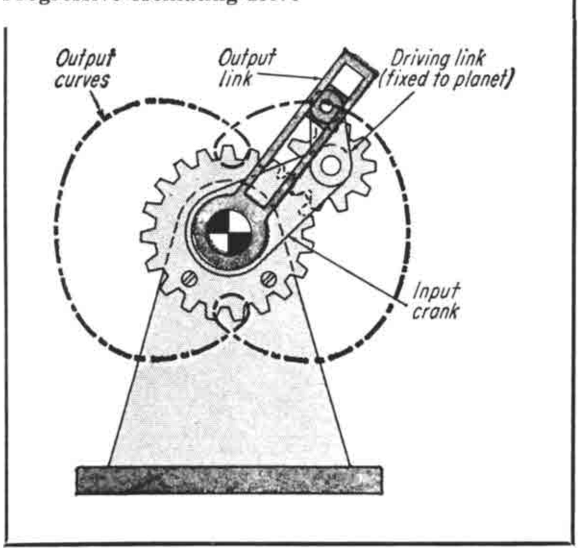
This arrangement permits the roller to exit and enter the driving slots tangentially. In the position shown, the driving roller has just completed indexing the geneva, and it is about to coast for 90º as it goes around the curve. (During this time, a separate locking device might be necessary to prevent an external torque from reversing the geneva.)

Controlled-output escapement



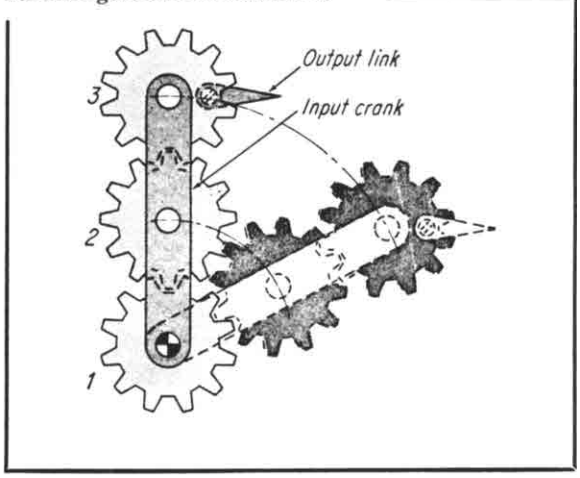
The output in this simple mechanism is prevented from turning in either direction—unless it is actuated by the input motion. In operation, the drive lever indexes the output disk by bearing on the pin. The escapement is cammed out of the way during indexing because the slot in the input disk is positioned to permit the escapement tip to enter it. But as the lever leaves the pin, the input disk forces the escapement tip out of its slot and into the notch. That locks the output in both directions.

Progressive oscillating drive



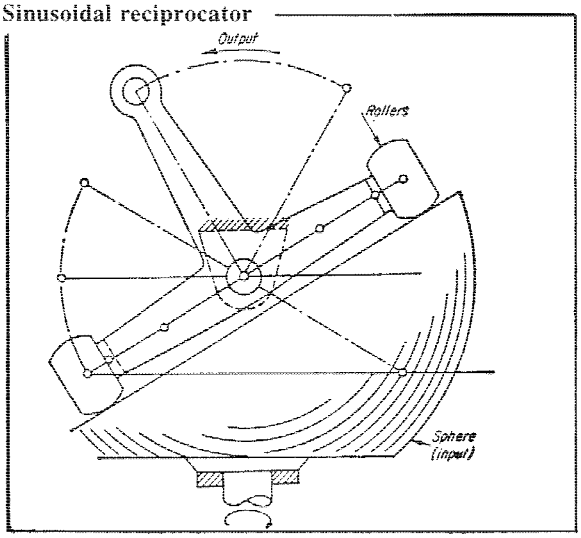
A crank attached to the planet gear can make point *P* describe the double loop curve illustrated. The slotted output crank oscillates briefly at the vertical positions.

Parallel-guidance mechanisms



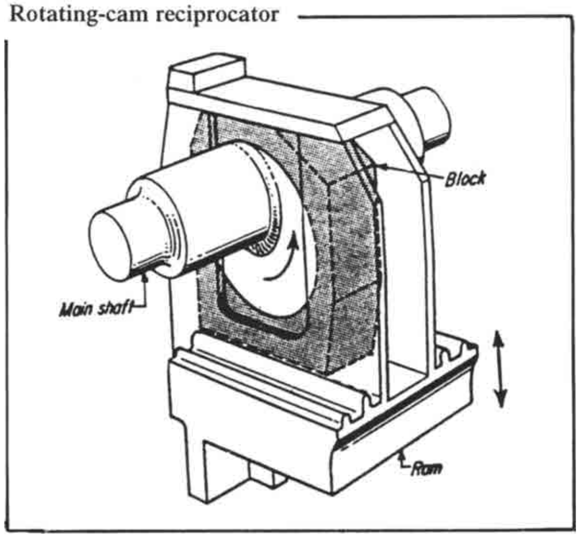
The input crank contains two planet gears. The center sun gear is fixed. By making the three gears equal in diameter and having gear *2* serve as an idler, any member fixed to gear *3* will remain parallel to its previous positions throughout the rotation of the input ring crank.

Sinusoidal reciprocator



This reciprocator transforms rotary motion into a reciprocating motion in which the oscillating output member is in the same plane as the input shaft. The output member has two arms with rollers which contact the surface of the truncated sphere. The rotation of the sphere causes the output to oscillate.

Rotating-cam reciprocator



The high-volume 2500-ton press is designed to shape such parts as connecting rods, tractor track links, and wheel hubs. A simple automatic-feed mechanism makes it possible to produce 2400 forgings per hour.

MODIFIED GENEVA DRIVES

Most of the mechanisms shown here add a varying velocity component to conventional geneva motion.

Fig. 1 With a conventional external geneva drive, a constant-velocity input produces an output consisting of a varying velocity period plus a dwell. The motion period of the modified geneva shown has a constant-velocity interval which can be varied within limits. When spring-loaded driving roller *a* enters the fixed cam *b*, the output-shaft velocity is zero. As the roller travels along the cam path, the output velocity rises to some constant value, which is less than the maximum output of an unmodified geneva with the same number of slots. The duration of constant-velocity output is arbitrary within limits. When the roller leaves the cam, the output velocity is zero. Then the output shaft dwells until the roller re-enters the cam. The spring produces a variable radial distance of the driving roller from the input shaft, which accounts for the described motions. The locus of the roller's path during the constant-velocity output is based on the velocity-ratio desired.

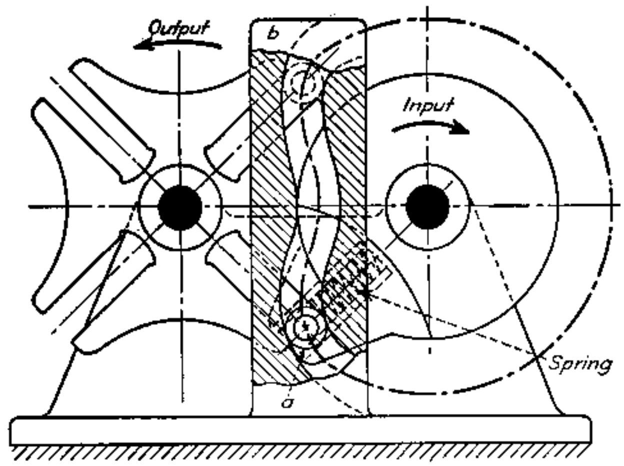


Fig. 1

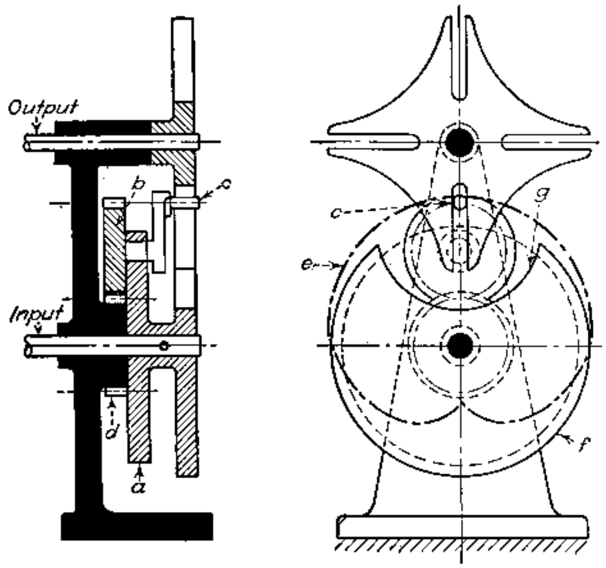


Fig. 2

Fig. 2 This design incorporates a planet gear in the drive mechanism. The motion period of the output shaft is decreased, and the maximum angular velocity is increased over that of an unmodified geneva with the same number of slots. Crank wheel *a* drives the unit composed of planet gear *b* and driving roller *c*. The axis of the driving roller coincides with a point on the pitch circle of the planet gear. Because the planet gear rolls around the fixed sun gear *d*, the axis of roller *c* describes a cardioid *e*. To prevent the roller from interfering with the locking disk *f*, the clearance arc *g* must be larger than is required for unmodified genevas.

Fig. 3 A motion curve similar to that of Fig. 2 can be derived by driving a geneva wheel with a two-crank linkage. Input crank *a* drives crank *b* through link *c*. The variable angular velocity of driving roller *d*, mounted on *b*, depends on the center distance *L*, and on the radii *M* and *N* of the crank arms. This velocity is about equivalent to what would be produced if the input shaft were driven by elliptical gears.

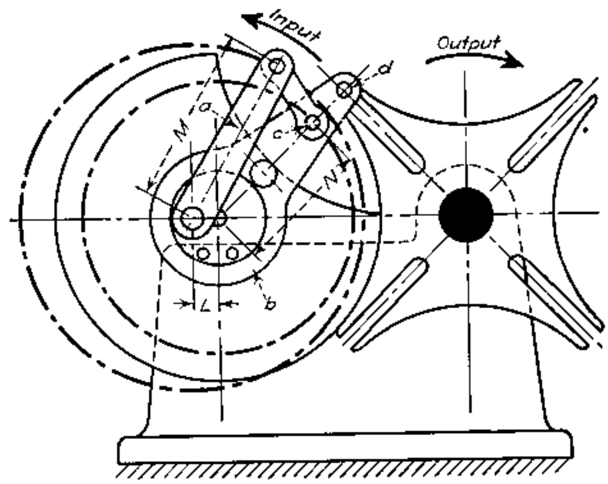


Fig. 3

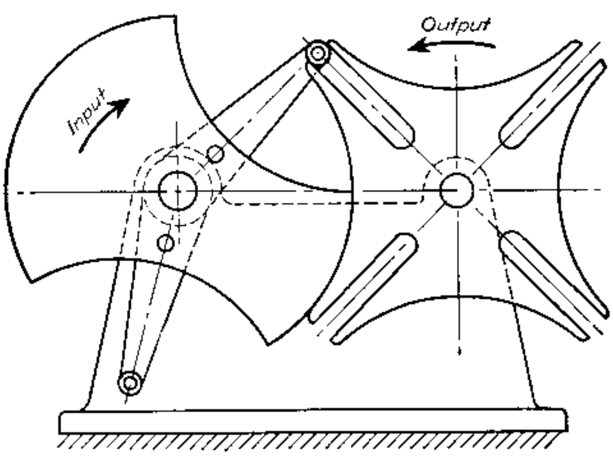


Fig. 4

Fig. 4 The duration of the dwell periods is changed by arranging the driving rollers unsymmetrically around the input shaft. This does not affect the duration of the motion periods. If unequal motion periods and unequal dwell periods are desired, the roller crank-arms must be unequal in length and the star must be suitably modified. This mechanism is called an irregular geneva drive.

Fig. 5 In this intermittent drive, the two rollers drive the output shaft and lock it during dwell periods. For each revolution of the input shaft, the output shaft has two motion periods. The output displacement φ is determined by the number of teeth. The driving angle, ψ, can be chosen within limits. Gear *a* is driven intermittently by two driving rollers mounted on input wheel *b*, which is bearing-mounted on frame *c*. During the dwell period the rollers circle around the top of a tooth. During the motion period, a roller's path *d*, relative to the driven gear, is a straight line inclined towards the output shaft. The tooth profile is a curve parallel to path *d*. The top land of a tooth becomes the arc of a circle of radius *R*, and the arc approximates part of the path of a roller.

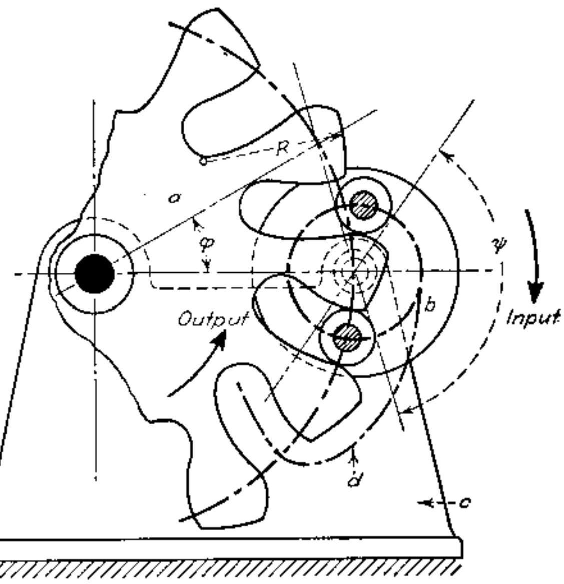


Fig. 5

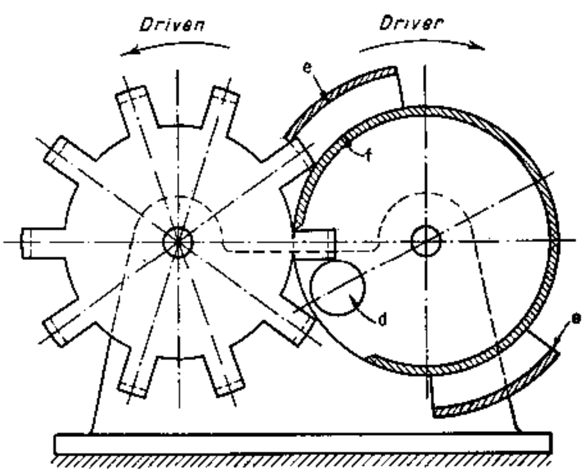
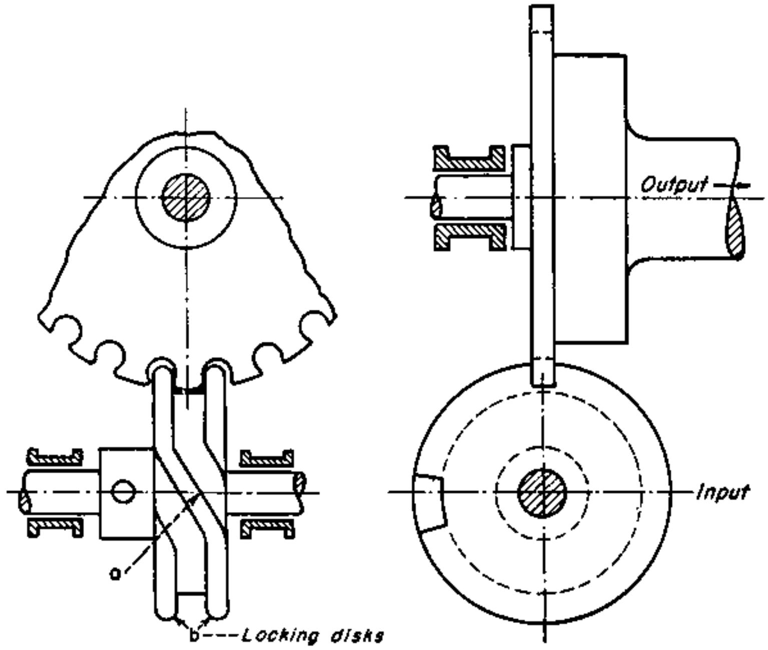


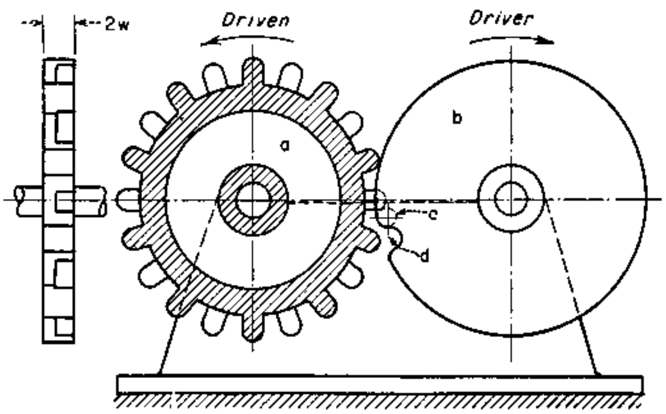
Fig. 6

Fig. 6 An intermittent drive with a cylindrical lock. Shortly before and after the engagement of two teeth with driving pin *d* at the end of the dwell period, the inner cylinder *f* is unable to cause positive locking of the driven gear. Consequently, a concentric auxiliary cylinder *e* is added. Only two segments are necessary to obtain positive locking. Their length is determined by the circular pitch of the driven gear.

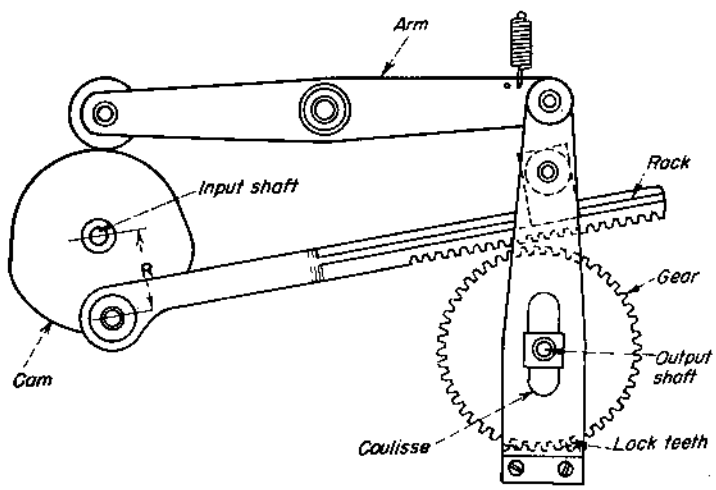
This mechanism transmits intermittent motion between two skewed shafts. The shafts need not be at right angles to one another. Angular displacement of the output shaft per revolution of input shaft equals the circular pitch of the output gear wheel divided by its pitch radius. The duration of the motion period depends on the length of the angular joint *a* of the locking disks *b*.

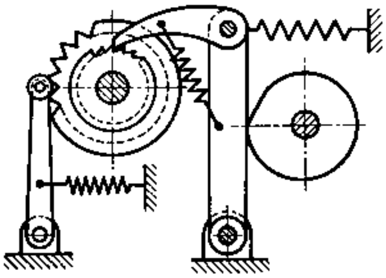


A "mutilated tooth" intermittent drive. Driver *b* is a circular disk of width *w* with a cutout *d* on its circumference. It carries a pin *c* close to the cutout. The driven gear, *a*, of width 2*w* has an even number of standard spur gear teeth. They alternately have full and half-width (mutilated) teeth. During the dwell period, two full-width teeth are in contact with the circumference of the driving disk, thus locking it. The mutilated tooth between them is behind the driver. AT the end of the dwell period, pin *c* contacts the mutilated tooth and turns the driven gear one circular pitch. Then, the full-width tooth engages the cutout *d*, and the driven gear moves one more pitch. Then the dwell period starts again and the cycle is repeated.

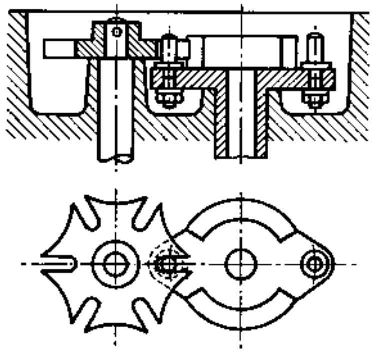


An operating cycle of 180º motion and 180º dwell is produced by this mechanism. The input shaft drives the rack, which is engaged with the output shaft gear during half the cycle. When the rack engages, the lock teeth at the lower end of the coulisse are disengaged and, conversely, when the rack is disengaged, the coulisse teeth are engaged. This action locks the output shaft positively. The changeover points occur at the dead-center positions, so that the motion of the gear is continuously and positively governed. By varying the radius *R* and the diameter of the gear, the number of revolutions made by the output shaft during the operating half of the cycle can be varied to suit many differing requirements.

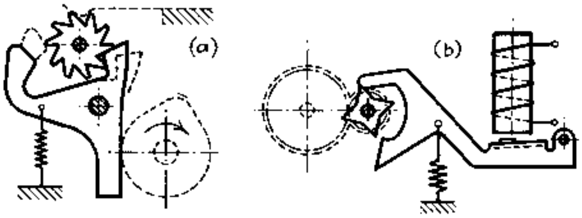




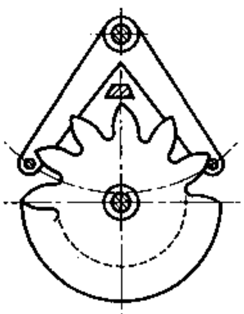
A cam-driven ratchet.



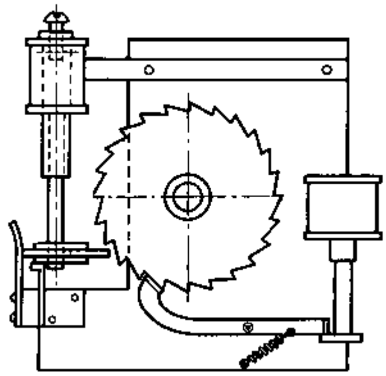
A six-sided Maltese cross and double driver give a 3:1 ratio.



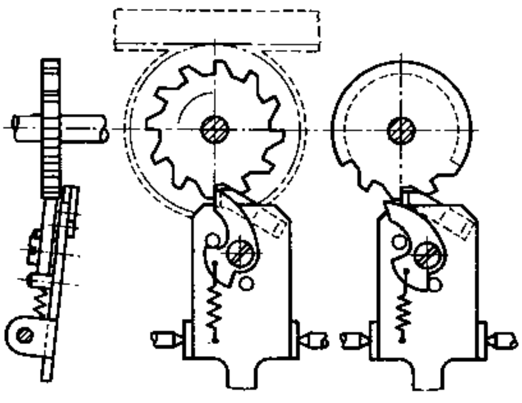
A cam operated escapement on a taximeter (a). A solenoid-operated escapement (b).



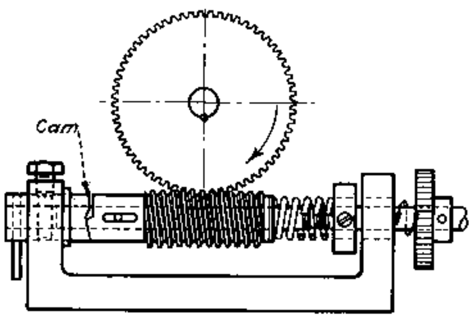
An escapement on an electric meter.



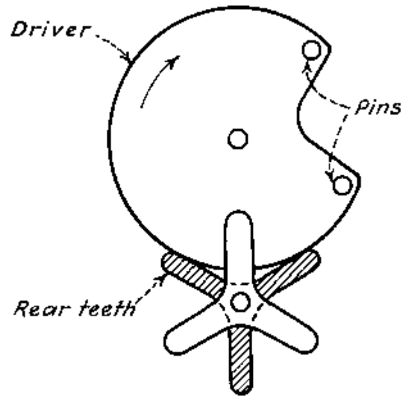
A solenoid-operated ratchet with a solenoid-resetting mechanism A sliding washer engages the teeth.



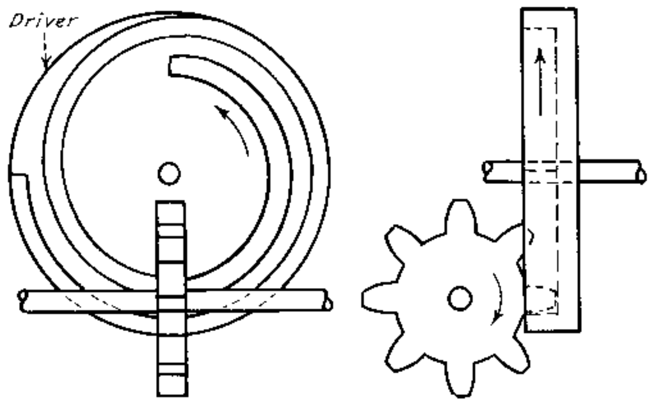
A plate oscillating across the plane of a ratchet-gear escapement carries stationary and spring-held pawls.



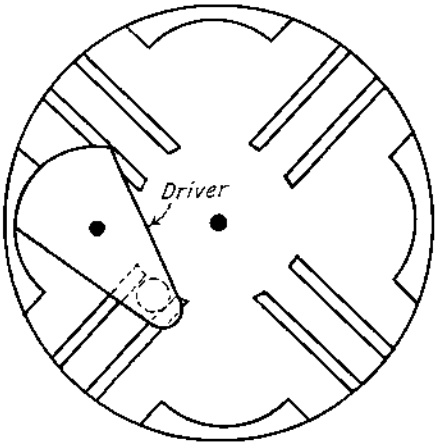
A worm drive, compensated by a cam on a work shaft, produces intermittent motion of the gear.



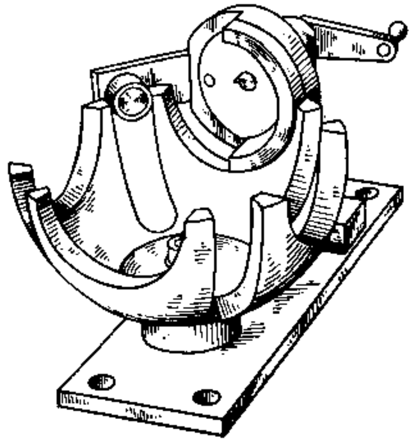
An intermittent counter mechanism. One revolution of the driver advances the driven wheel 120º. The driven-wheel rear teeth are locked on the cam surface during dwell.



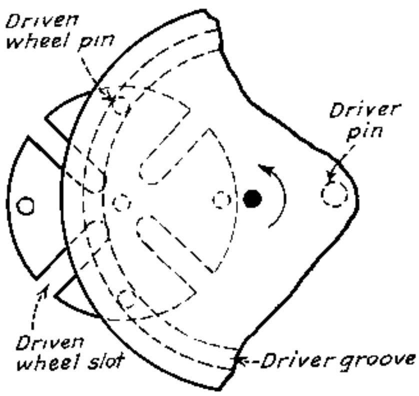
Spiral and wheel. One revolution of the spiral advances the driven wheel one tooth width. The driven-wheel tooth is locked in the driver groove during dwell.



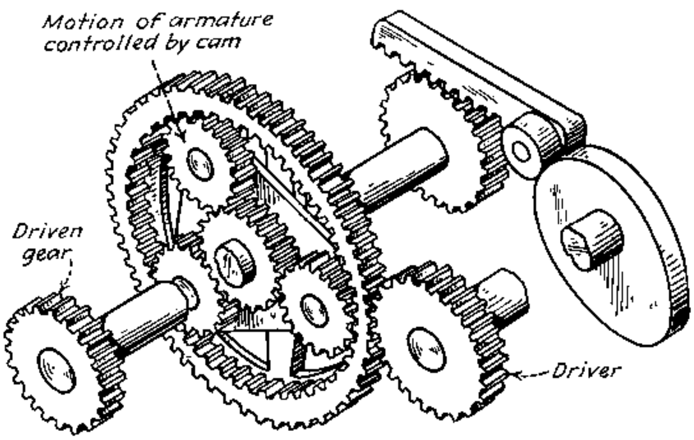
An internal geneva mechanism. The driver and driven wheel rotate in same direction. The duration of dwell is more than 180º of driver rotation.



A spherical geneva mechanism. The driver and driven wheel are on perpendicular shafts. The duration of dwell is exactly 180º of driver rotation.



An external geneva mechanism. The driver grooves lock the driven wheel pins during dwell. During movement, the driver pin mates with the driven-wheel slot.



A special planetary gear mechanism. The principle of relative motion of mating gears illustrated in this method can be applied to spur gears in planetary system. The motion of the central planet gear produces the motion of the summing gear.

The appeal of cycloidal mechanisms is that they can be tailored to provide one of these three common motions:

- **Intermittent**—with either short or long dwells.
- **Rotary with progressive oscillation**—where the output undergoes a cycloidal motion during which the forward motion is greater than the return motion
- **Rotary**-to-linear with a dwell period

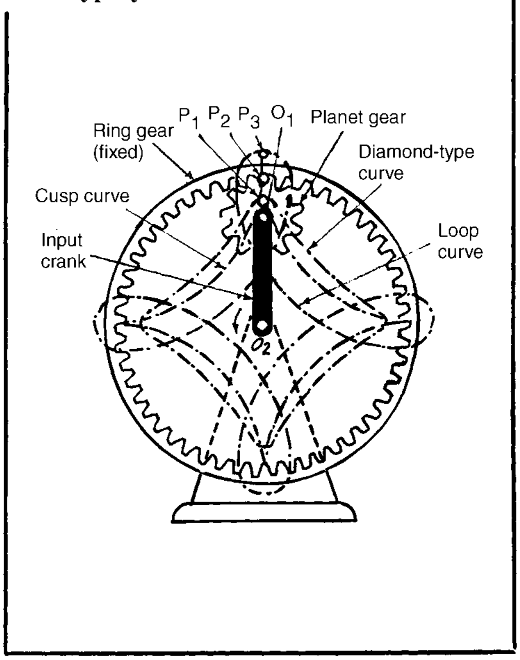
All the cycloidal mechanisms shown here are general. This results in compact positive mechanisms capable of operating at relatively high speeds with little backlash or "slop." These mechanisms can be classified into three groups:

Hypocycloid—the points tracing the cycloidal curves are located on an external gear rolling inside an internal ring gear. This ring gear is usually stationary and fixed to the frame.

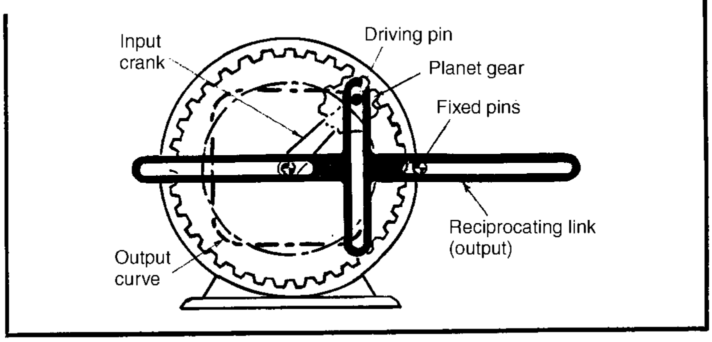
Epicycloid—the tracing points are on an external gear that rolls in another external (stationary) gear.

Pericycloid—the tracing points are located on an internal gear that rolls on a stationary external gear.

Basic hypocycloid curves



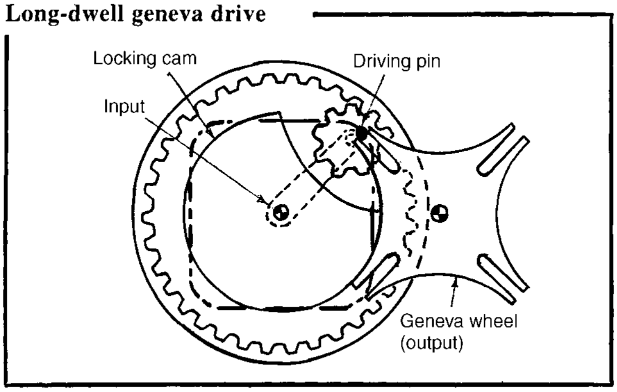
Double-dwell mechanism



Coupling the output pin to a slotted member produces a prolonged dwell in each of the extreme positions. This is another application of the diamond-type hypocycloidal curve.

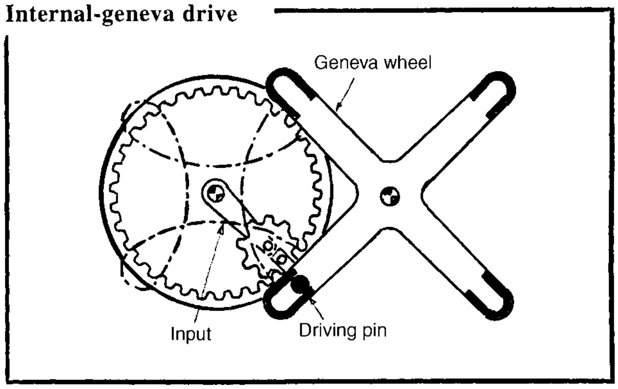
The input drives a planet in mesh with a stationary ring gear. Point P_1 on the planet gear describes a diamond-shape curve, point P_2 on the pitch line of the planet describes the familiar cusp curve, and point P_3, which is on an extension rod fixed to the planet gear, describes a loop-type curve. In one application, an end miller located at P_1 machined a diamond-shaped profile.

Long-dwell geneva drive



In common with standard, four-station genevas, each rotation of the input of this drive indexes the slotted geneva 90º. A pin fastened to the planet gear causes the drive to describe a rectangular-shaped cycloidal curve. This produces a smoother indexing motion because the driving pin moves on a noncircular path.

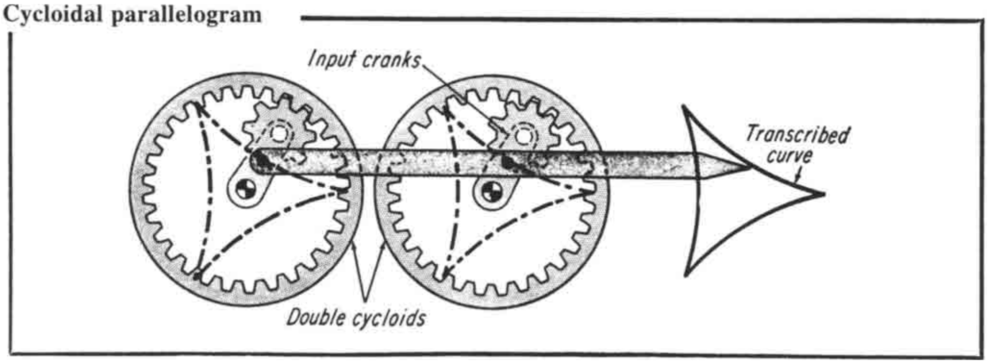
Internal-geneva drive



A loop-type curve permits the driving pin to enter the slot in a direction that is radially outward from the center. The pin then loops over to index the cross member rapidly. As with other genevas, the output rotates 90º before going into a long dwell period during each 270º rotation of the input element.

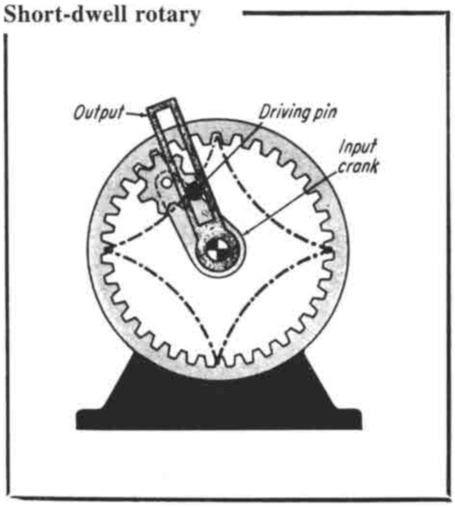
Cycloidal motion is popular for mechanisms in feeders and automatic machines.

Cycloidal parallelogram



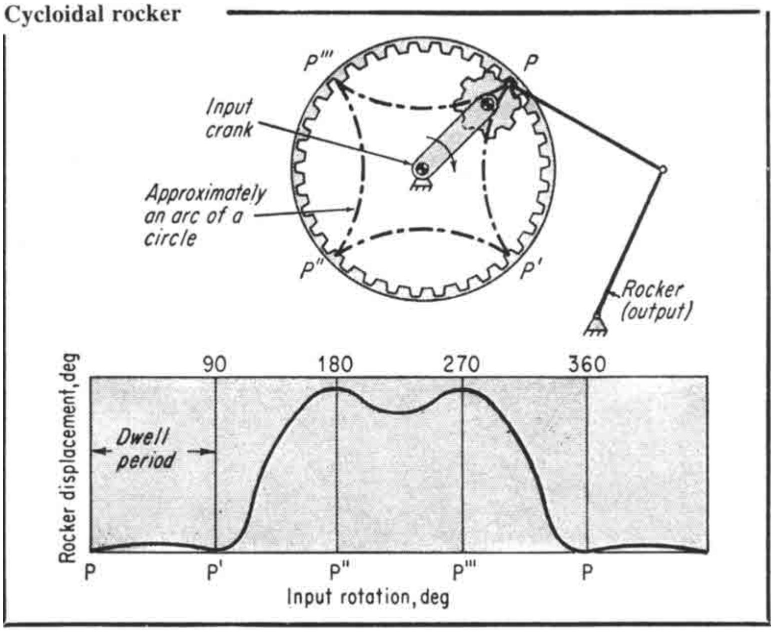
Two identical hypocycloid mechanisms guide the point of the bar along the triangularly shaped path. The mechanisms are useful where space is limited in the area where the curve must be described. These double-cycloid mechanisms can be designed to produce other curve shapes.

Short-dwell rotary



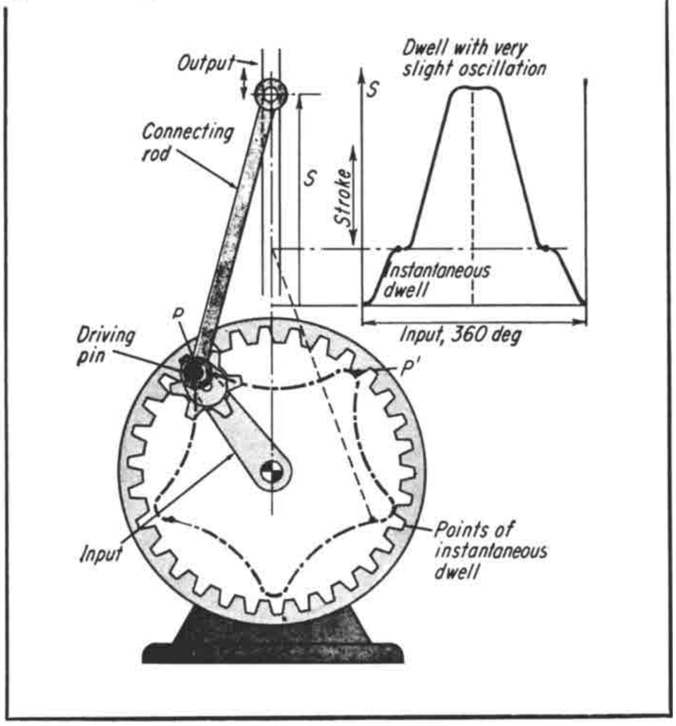
The pitch circle of this planet gear is exactly one-quarter that of the ring gear. A pin on the planet gear will cause the slotted output member to dwell four times during each revolution of the input shaft.

Cycloidal rocker



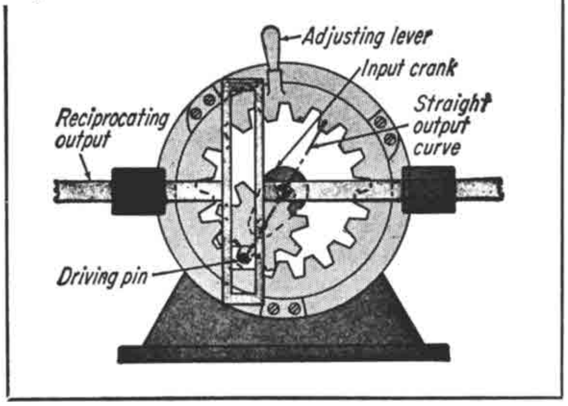
The curvature of the cusp is approximately that of an arc of a circle. Hence the rocker reaches a long dwell at the right extreme position while point P moves to P'. There is then a quick return from P' to P'', with a momentary dwell at the end of this phase. The rocker then undergoes a slight oscillation from point P'' to P''', as shown in the rocker displacement diagram.

Cycloidal reciprocator



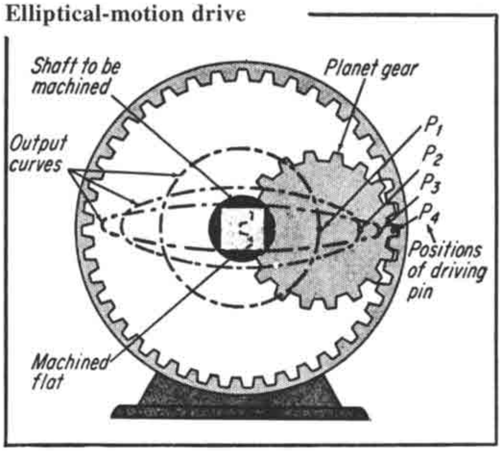
Part of curve P-P' produces a long dwell, but the five-lobe cycloidal curve avoids a marked oscillation at the end of the stroke. There are also two points of instantaneous dwell where the curve is perpendicular to the connecting rod.

Adjustable harmonic drive



By making the planet gear diameter half that of the internal gear, a straight-line output curve can be produced by the driving pin which is fastened to the planet gear. The pin engages the slotted member to cause the output to reciprocate back and forth with harmonic (sinusoidal) motion. The position of the fixed ring gear can be changed by adjusting the lever, which in turn rotates the straight-line output curve. When the curve is horizontal, the stroke is at a maximum; when the curve is vertical, the stroke is zero.

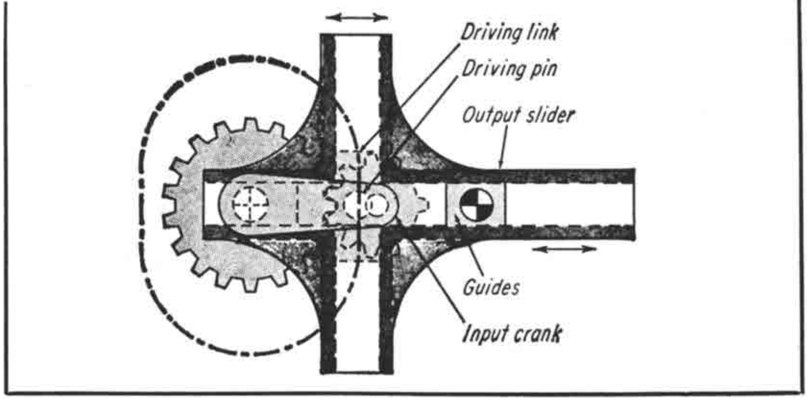
Elliptical-motion drive



By making the pitch diameter of the planet gear equal to half that of the ring gear, every point on the planet gear (such as points P_2 and P_3) will describe elliptical curves which get flatter as the points are selected closer to the pitch circle. Point P_1, at the center of the planet, describes a circle; point P_4, at the pitch circle, describes a straight line. When a cutting tool is placed at P_3, it will cut almost-flat sections from round stock, as when milling flats on a bolt. The other two flats of the bolt can be cut by rotating the bolt or the cutting tool 90º.

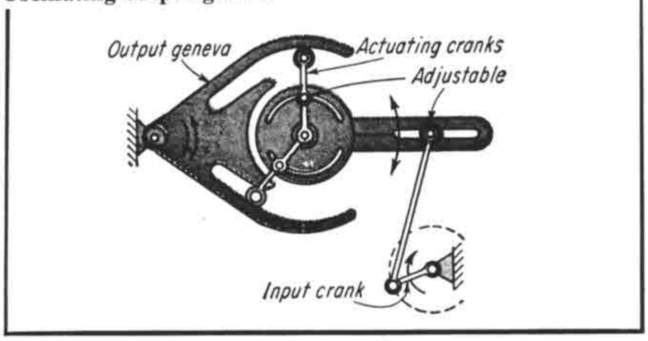
EPICYCLOID MECHANISMS

Epicycloid reciprocator



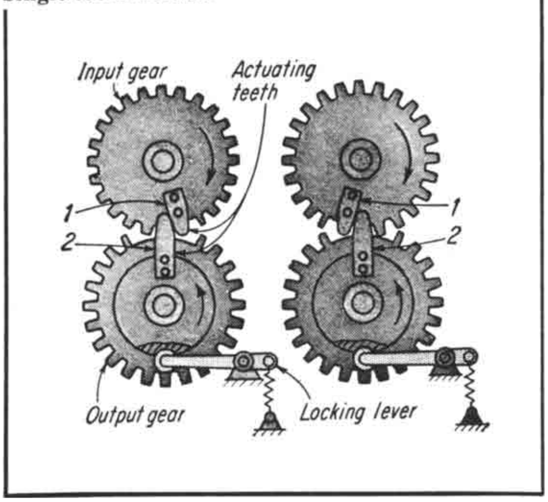
Here the sun gear is fixed and the planet is gear driven around it by the input link. There is no internal ring gear as with the hypocycloid mechanisms. Driving pin P on the planet describes the curve shown, which contains two almost-flat portions. If the pin rides in the slotted yoke, a short dwell is produced at both the extreme positions of the output member. The horizontal slots in the yoke ride the end-guides, as shown.

Oscillating-output geneva



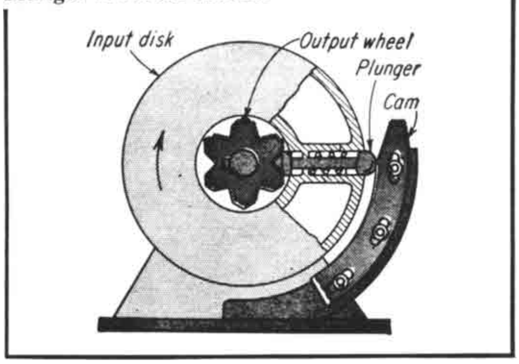
Three adjustable output-links provide a wide variety of oscillating motions. The input crank oscillates the central member that has an adjustable slot to vary the stroke. The oscillation is transferred to the two actuating rollers, which alternately enter the geneva slots to index it, first in one direction and then another. Additional variation in output motion can be obtained by adjusting the angular positions of the output cranks.

Single tooth indexer



The key concept in this indexer is its use of an input gear that is smaller than the output gear. Thus, it can complete its circuit faster than the output gear when both are in mesh. In the left diagram, the actuating tooth of the input gear, tooth 1, strikes that of the output gear, tooth 2, to roll both gears into mesh. After one circuit of the input (right diagram), tooth 1 is now ahead of tooth 2, the gears go out of mesh, and the output gear stops (it is kept in position by the bottom locking detent) for almost 360º of the input gear rotation.

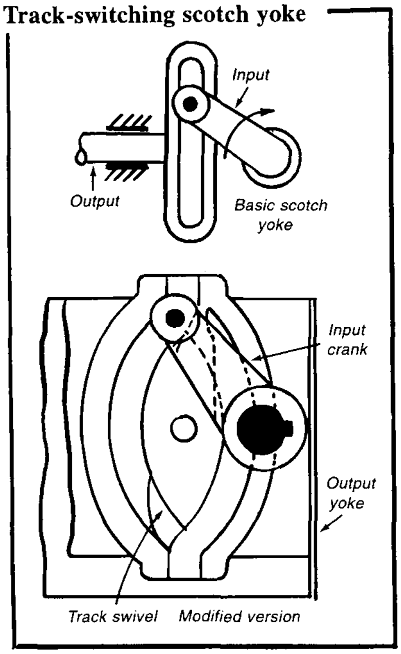
Plunger-actuated indexer



Here the output wheel rotates only when the plunger, which is normally kept in the outer position by its spring, is cammed into the toothed wheel attached to the output. Thus, for every revolution of the input disk, the output wheel is driven approximately 60º, and then it stops for the remaining 300º.

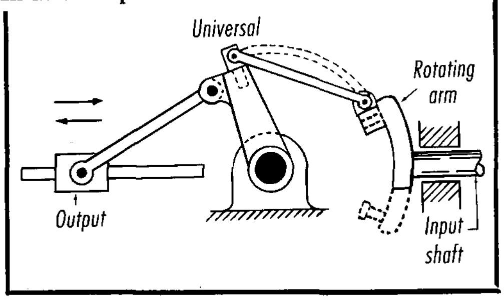
ROTARY-TO-RECIPROCATING DEVICES

Track-switching scotch yoke



In a typical scotch yoke, a, the motion of the rotating input crank is translated into the reciprocating motion of the yoke. But this provides only an instantaneous dwell at each end. To obtain long dwells, the left slot (in the modified version b) is curved with a radius equal to that of the input crank radius. This causes a 90º dwell at the let end of the stroke. For the right end, the crank pushes aside the springloaded track swivel as it comes around the bend, and it is shunted into the second track to provide a 90º dwell at the right end as well.

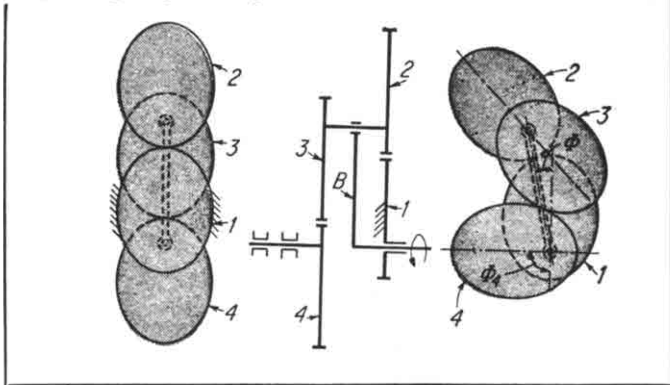
In-line reciprocator



This is a simple way to convert rotary motion to reciprocating motion. Both input and output shafts are in line with each other. The right half of the device is a three-dimensional reciprocator. Rotating the input crank causes its link to oscillate. A second connecting link then converts that oscillation into the desired in-line output motion.

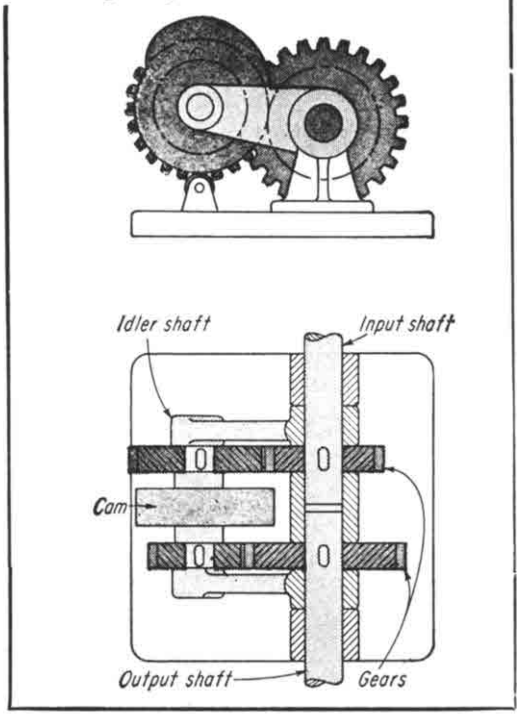
VARIABLE SPEED DEVICES

Elliptical-gear planetary



By substituting elliptical gears for the usual circular gears, a planetary drive is formed. It can provide extra-large variations in the angular speed output.

Cammed-gear speed variator



This is a normal parallel-gear speed reducer, but it has cam actuation to provide a desired variation in the output speed. If the center of the idler shaft were stationary, the output motion would be uniform. But by attaching a cam to the idler shaft, the shaft has an oscillating motion which varies the final output motion.

ADJUSTABLE-SPEED DRIVES

The output of this novel drive can be varied infinitely by changing the distance that the balls will operate from the main shaft line. The drive has multiple disks, free to rotate on a common shaft, except for the extreme left and right disks which are keyed to the input and output shafts, respectively. Every other disk carries three uniformly spaced balls which can be shifted closer to or away from the center by moving the adjustment lever. When disk 1 rotates the first group of balls, disk 3 will rotate slower because of the different radii, r_{x1} and r_{x2}. Disk 3 will then drive disk 5, and disk 5 will drive disk 7, all with the same speed ratios, thus compounding the ratios to get the final speed reduction.

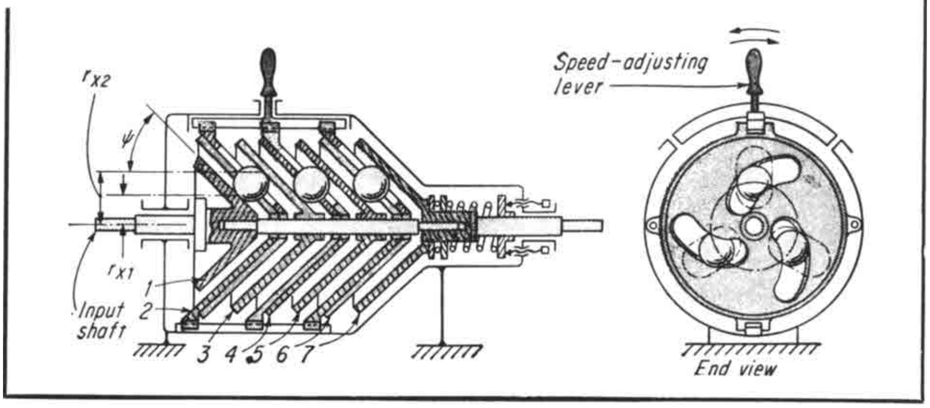
The effective radii can be calculated from

$$r_{x1} = R_x - 1/2\, D \cos \psi$$

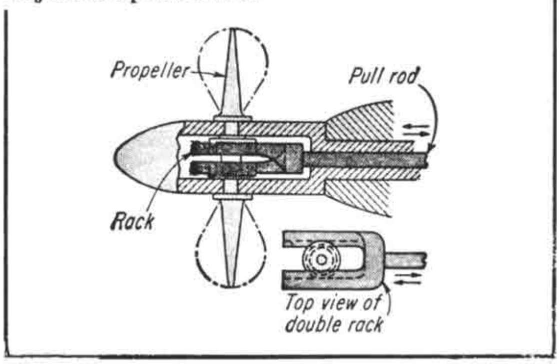
$$r_{x2} = R_x + 1/2\, D \cos \psi$$

where R_x is the distance from the shaft center to the ball center, D is the diameter of the ball, and ψ is one-half the cone angle.

Multicone drive



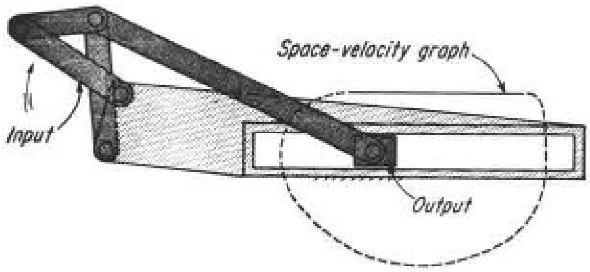
Adjustable-pitch device



Pulling or pushing the axial control rod of this adjustable-pitch propeller linearly twists the propeller blades around on the common axis by moving the rack and gear arrangement. A double rack, one above and on either side of the other, gives the opposing twisting motion required for propeller blades.

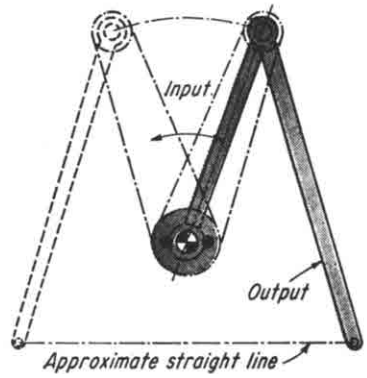
ROTARY-TO-RECIPROCATING MOTION AND DWELL MECHANISMS

Four-bar slider mechanism



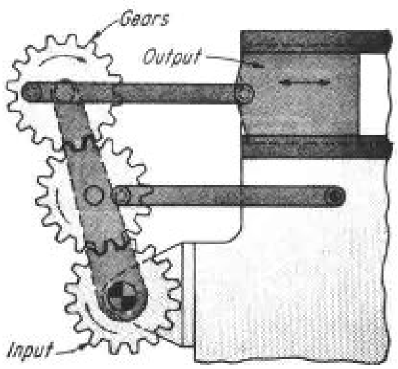
With proper dimensions, the rotation of the input link can impart an almost-constant velocity motion to the slider within the slot.

Oscillating-chain mechanism



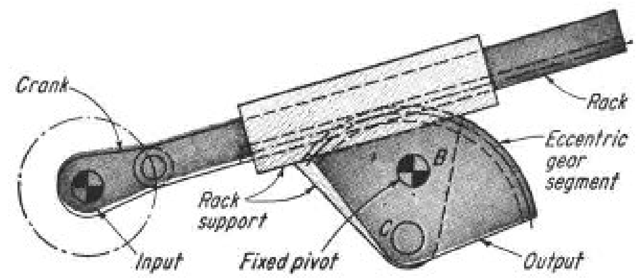
The rotary motion of the input arm is translated into linear motion of the linkage end. The linkage is fixed to the smaller sprocket, and the larger sprocket is fixed to the frame.

Three-gear stroke multiplier



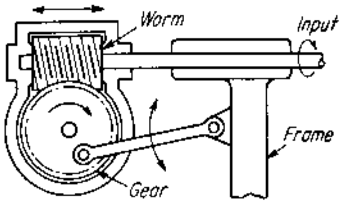
The rotation of the input gear causes the connecting link, attached to the machine frame, to oscillate. This action produces a large-stroke reciprocating motion in the output slider.

Rack and gear sector



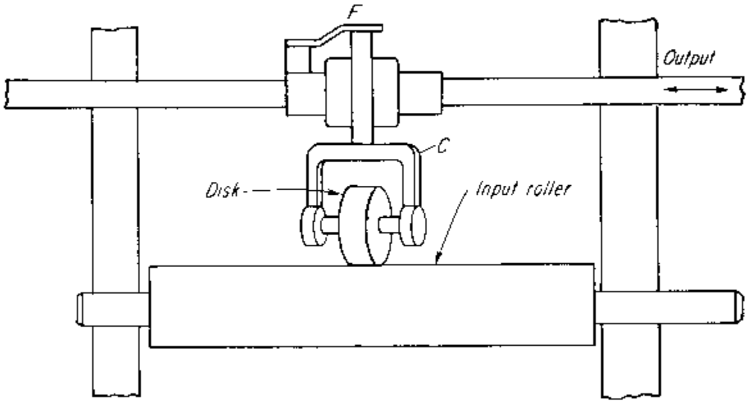
The rotary motion of the input shaft is translated into an oscillating motion of the output gear segment. The rack support and gear sector are pinned at *C* but the gear itself oscillates around *B*.

Linear reciprocator



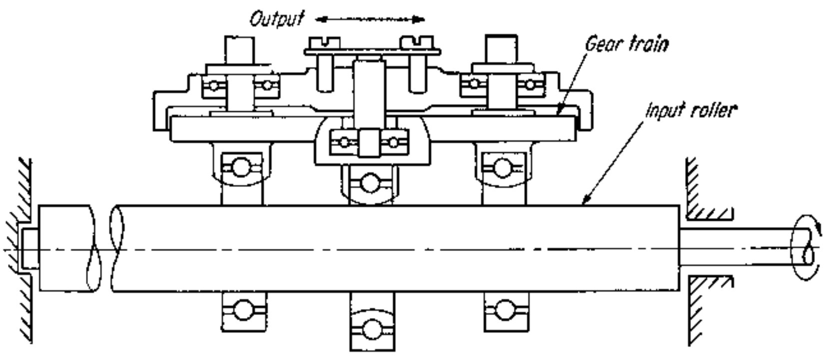
This linear reciprocator converts a rotary motion into a reciprocating motion that is *in line* with the input shaft. Rotation of the shaft drives the worm gear which is attached to the machine frame with a rod. Thus input rotation causes the worm gear to draw itself (and the worm) to the right—thus providing a reciprocating motion.

Disk and roller drive



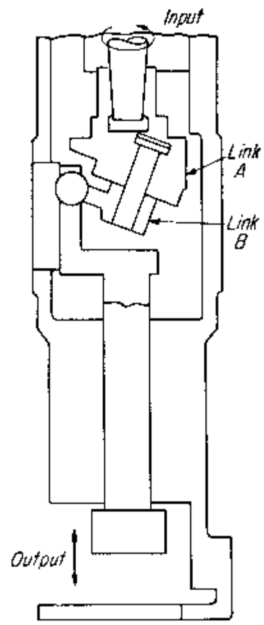
A hardened disk in this drive, riding at an angle to the axis of an input roller, transforms the rotary motion into linear motion parallel to the axis of the input. The roller is pressed against the input shaft by flat spring *F*. The feed rate is easily varied by changing the angle of the disk. This arrangement can produce an extremely slow feed with a built-in safety factor in case of possible jamming.

Bearing and roller drive



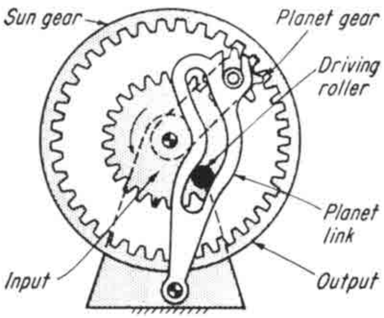
This drive arrangement avoids large Hertzian stresses between the disk and roller by including three ball bearings in place of the single disk. The inner races of the bearings make contact on one side or the other. Hence a gearing arrangement is required to alternate the angle of the bearings. This arrangement also reduces the bending moment on the shaft.

Reciprocating space crank



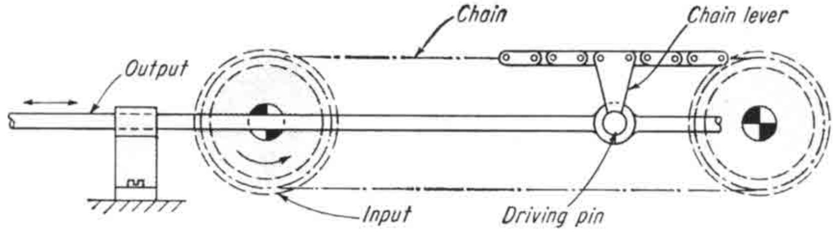
The rotary input of this crank causes the bottom surface of link *A* to wobble with respect to the center link. Link *B* is free of link *A*, but it is restrained from rotating by the slot. This causes the output member to reciprocate linearly.

Oscillating crank and planetary drive



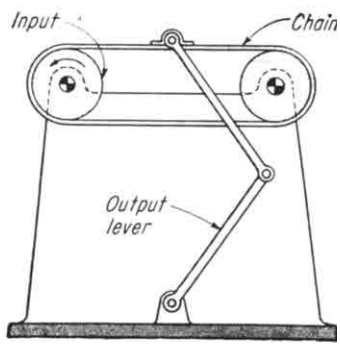
The planet gear is driven with a stop-and-go motion. The driving roller is shown entering the circular-arc slot on the planet link. The link and the planet remain stationary while the roller travels along this section of the slot. As a result, the output sun gear has a rotating output motion with a progressive oscillation.

Chain-slider drive



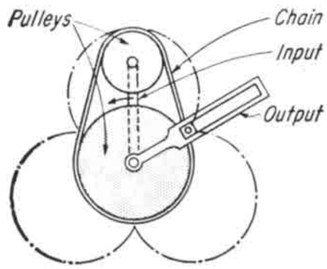
The chain link drives a lever that oscillates. A slowdown-dwell occurs when the chain pin passes around the left sprocket.

Chain-oscillating drive



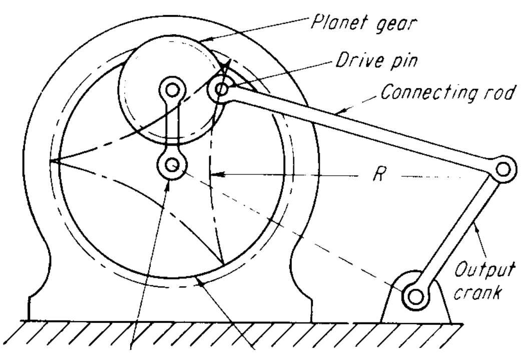
The output shaft reciprocates with a constant velocity, but it reaches a long dwell at both ends as the chain lever, whose length is equal to the radius of the sprockets, goes around both sprockets.

Chain and slider drive



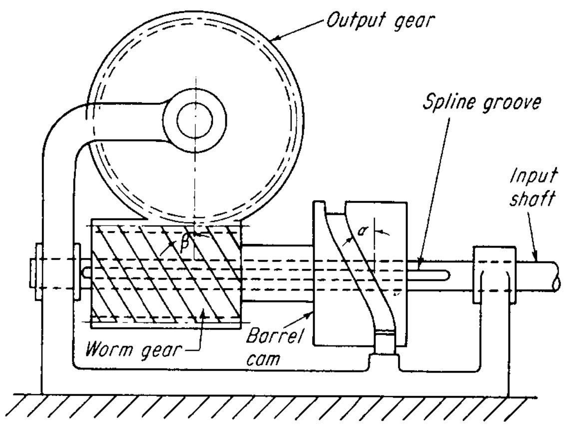
The input crank causes the small pulley to orbit around the stationary larger pulley. A pivot point attached to the chain slides inside the slot of the output link. In the position shown, the output is about to start a long dwell period of about 120º.

Epicyclic dwell mechanism



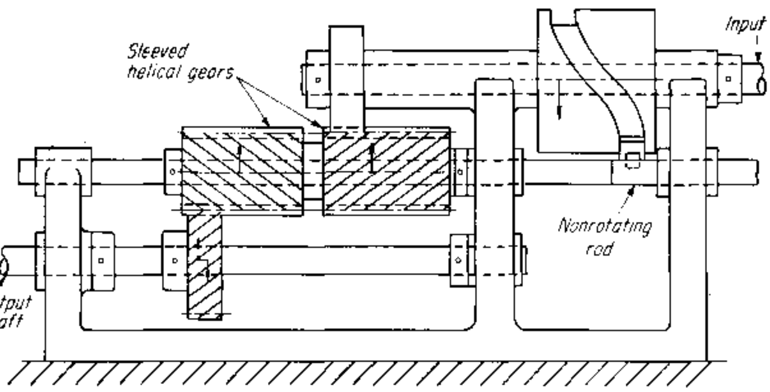
The output crank pulsates back and forth with a long dwell at its extreme right position. The input shaft rotates the planet gear with a crank. The pin on the planet gear traces the epicyclic three-lobe curve shown. The right side of the curve is a near circular arc of radius *R*. If the connecting rod length equals *R*, the output crank reaches a virtual standstill during a third of the total rotation of the input crank. The crank then reverses, stops at its left position, reverses, and repeats its dwell.

Cam-worm dwell mechanism



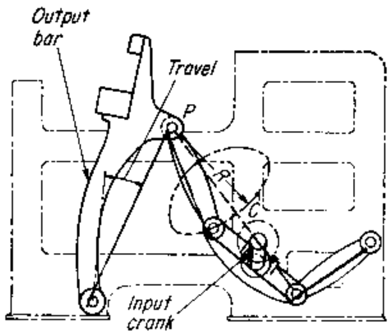
Without the barrel cam, the input shaft would drive the output gear by the worm gear at constant speed. The worm and the barrel cam, however, can slide linearly on the input shaft. The rotation of the input shaft now causes the worm gear to be cammed back and forth, thus adding or subtracting motion to the output. If barrel cam angle α is equal to the worm angle β, the output stops during the limits of rotation shown. It then speeds up to make up for lost time.

Cam-helical dwell mechanism



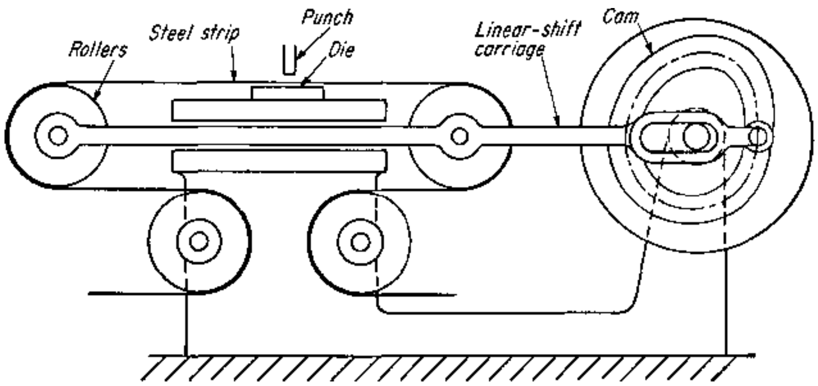
When one helical gear is shifted linearly (but prevented from rotating) it will impart rotary motion to the mating gear because of the helix angle. This principle is applied in the mechanism illustrated. The rotation of the input shaft causes the intermediate shaft to shift to the left, which in turn adds or subtracts from the rotation of the output shaft.

Six-bar dwell mechanism



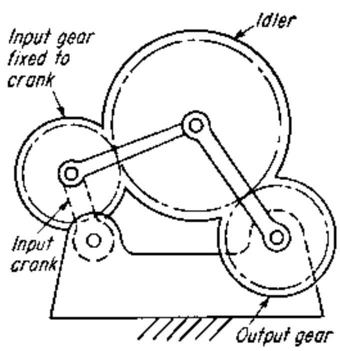
The rotation of the input crank causes the output bar to oscillate with a long dwell at its extreme right position. This occurs because point C describes a curve that is approximately a circular arc (from C to C' with its center at P. The output is almost stationary during that part of the curve.

Cam-roller dwell mechanism



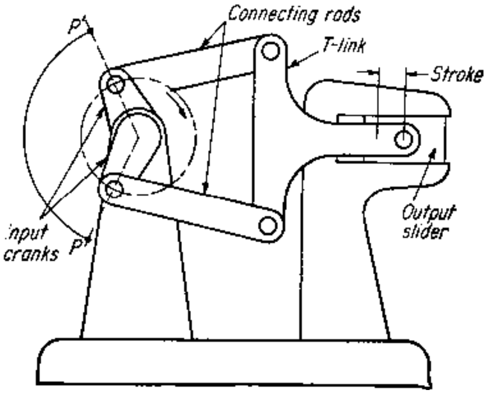
A steel strip is fed at constant linear velocity in this mechanism. But at the die station (illustrated), it is desired to stop the strip so that the punching operation can be performed. The strip passes over movable rollers which, when shifted to the right, cause the strip to move to the right. Since the strip is normally fed to the left, proper design of the cam can nullify the linear feed rates so that the strip stops, and then speeds to catch up to the normal rate.

Three-gear drive

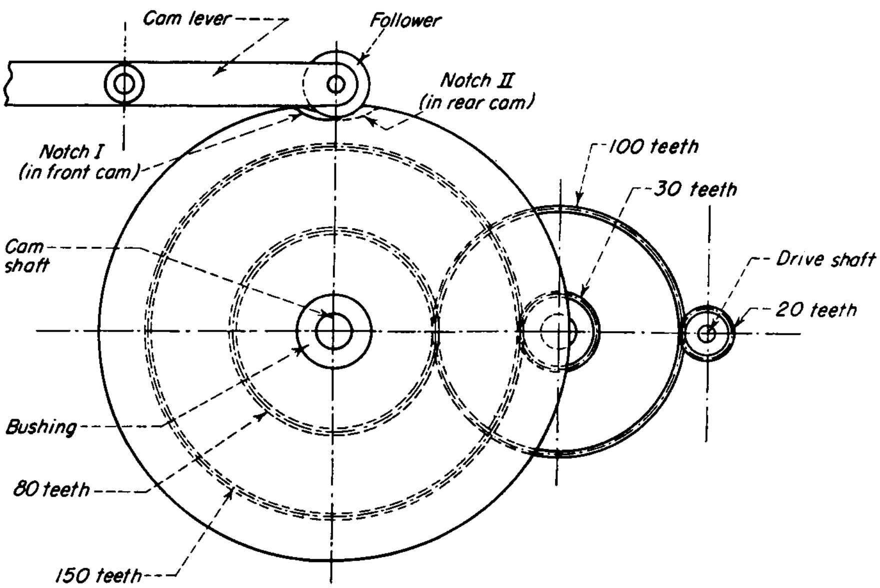


This is actually a four-bar linkage combined with three gears. As the input crank rotates, it turns the input gear which drives the output gear through the idler. Various output motions are possible. Depending on the relative diameters of the gears, the output gear can pulsate, reach a short dwell, or even reverse itself briefly.

Double-crank dwell mechanism



Both cranks are connected to a common shaft which also acts as the input shaft. Thus the cranks always remain a constant distance apart from each other. There are only two frame points—the center of the input shaft and the guide for the output slider. As the output slider reaches the end of its stroke (to the right), it remains at a virtual standstill while one crank rotates through angle PP'.

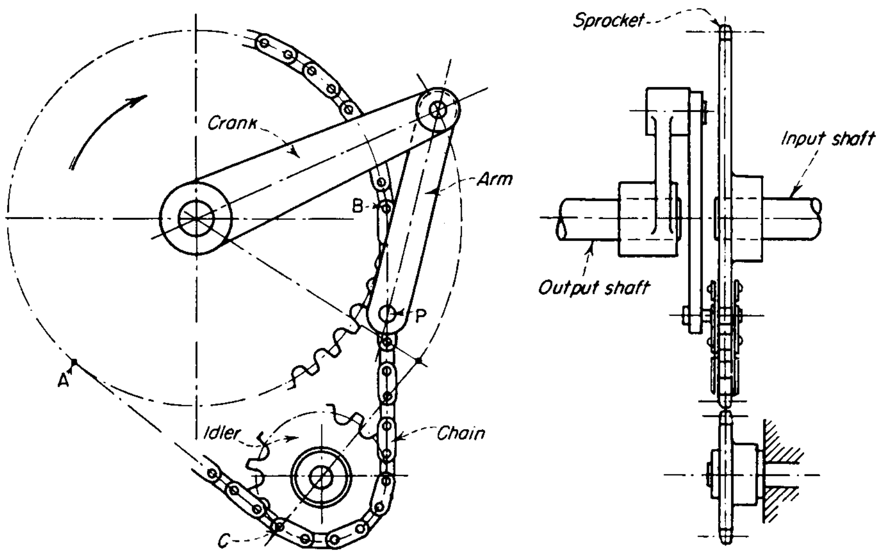


Fast Cam-Follower Motion

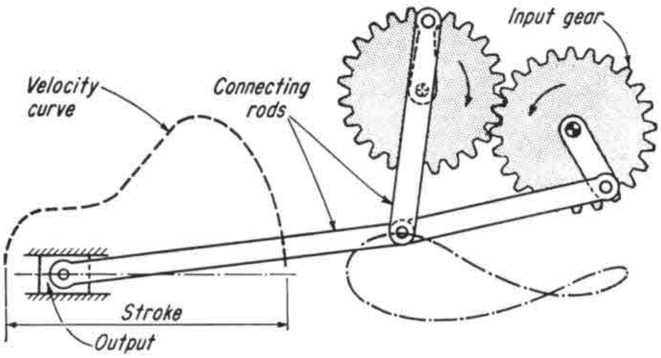
Fast cam action every n cycles (where n is a relatively large number) can be obtained with this manifold cam and gear mechanism. A single notched cam geared $1/n$ to a shaft turning once per cycle moves relatively slowly under the follower. The double notched-cam arrangement shown is designed to operate the lever once in 100 cycles, imparting a rapid movement to it. One of the two identical cams and the 150-tooth gear are keyed to the bushing which turns freely around the cam shaft. The cam shaft carries the second cam and the 80-tooth gear. The 30- and 100-tooth gears are integral, while the 20-tooth gear is attached to the one-cycle drive shaft. One of the cams turns in the ratio of 20/80 or 1/4; the other turns in the ratio 20/100 times 30/150 or 1/25. The notches therefore coincide once every 100 cycles (4×25). Lever movement is the equivalent of a cam turning in a ratio of 1 to 4 in relation to the drive shaft. To obtain fast cam action, n must be reduced to prime factors. For example, if 100 were factored into 5 and 20, the notches would coincide after every 20 cycles.

Intermittent Motion

This mechanism can be adapted to produce a stop, a variable speed without stop, or a variable speed with momentary reverse motion. A uniformly rotating input shaft drives the chain around the sprocket and idler. The arm serves as a link between the chain and the end of the output shaft crank. The sprocket drive must be in the ratio N/n with the cycle of the machine, where n is the number of teeth on the sprocket and N the number of links in the chain. When point P travels around the sprocket from point A to position B, the crank rotates uniformly. Between B and C, P decelerates; between C and A it accelerates; and at C there is a momentary dwell By changing the size and position of the idler, or the lengths of the arm and crank, a variety of motions can be obtained. If the length of the crank is shortened, a brief reverse period will occur in the vicinity of C; if the crank is lengthened, the output velocity will vary between a maximum and minimum without reaching zero.

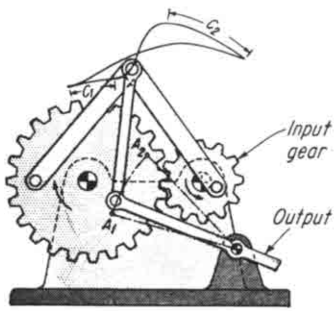


Gear-slider crank



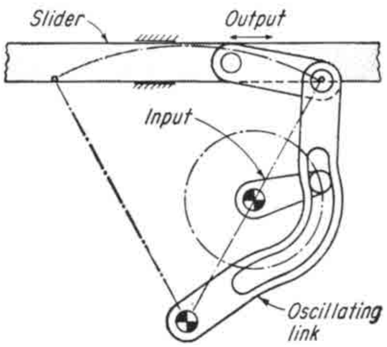
The input shaft drives both gears which, in turn, drive the connecting rods to produce the velocity curve shown. The piston moves with a low constant velocity.

Gear oscillating crank



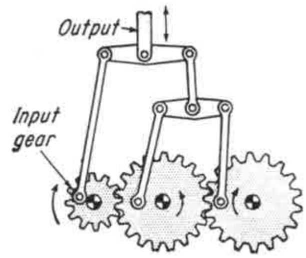
In this arrangement, the curve described by the pin connection has two parts, C_1 and C_2, which are very close to circular arc with its centers at A_1 and A_2. Consequently the driven link will have a dwell at both of its extreme positions.

Curve slider drive



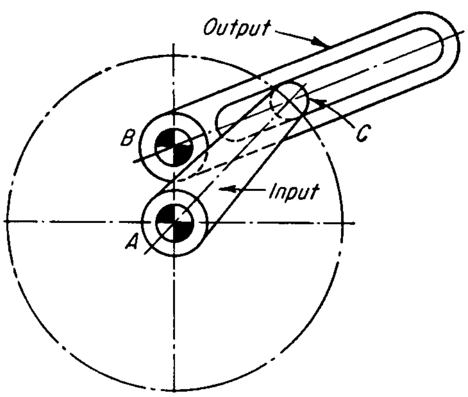
The circular arc on the oscillating link permits the link to reach a dwell during the right position of the output slider.

Triple-harmonic drive



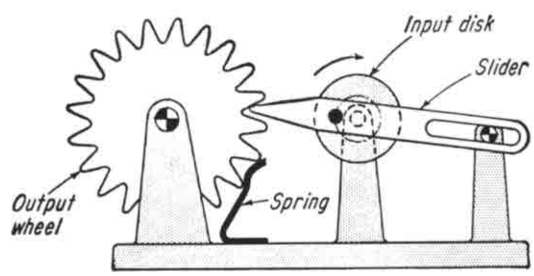
The input shaft drives three gears with connecting rods. A wide variety of reciprocating output motions can be obtained by selecting different lengths for the linkages. In addition, one to several dwells can be obtained per cycle.

Whitworth quick-return drive



Varying motion can be imparted simply to output shaft *B*. However, the axes, *A* and *B*, are not colinear.

Wheel and slider drive



For each revolution of the input disk, the slider moves in to engage the wheel and index it one tooth width. A flat spring keeps the wheel locked while it is stationary.

FRICTION DEVICES FOR INTERMITTENT ROTARY MOTION

Friction devices are free from such common disadvantages inherent in conventional pawl and ratchet drives as: (1) noisy operation; (2) backlash needed for pawl engagement; (3) load concentrated on one tooth of the ratchet; and (4) pawl engagement dependent on an external spring. Each of the five mechanisms presented here converts the reciprocating motion of a connecting rod into an intermittent rotary motion. The connecting rod stroke to the left drives a shaft counterclockwise and that shaft is uncoupled. It remains stationary during the return stroke of the connecting rod to the right.

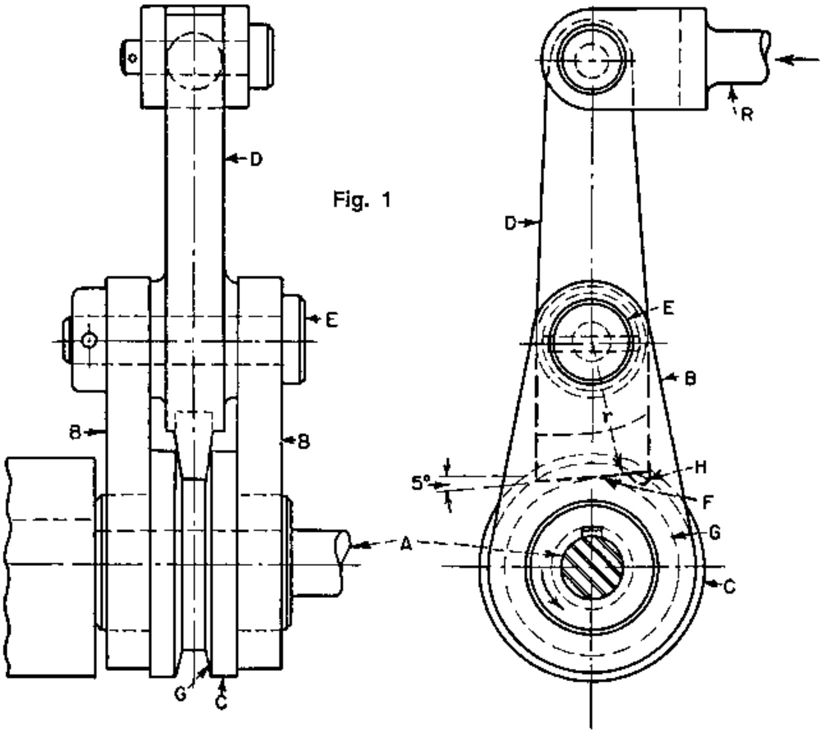


Fig. 1 The wedge and disk mechanism consists of shaft *A* supported in bearing block *J;* ring *C* is keyed to *A* and it contains an annular groove *G;* body *B*, which can pivot around the shoulders of *C;* lever *D*, which can pivot about *E;* and connecting rod *R*, which is driven by an eccentic (not shown). Lever *D* is rotated counterclockwise about *E* by the connecting rod moving to the left until surface *F* wedges into groove *G*. Continued rotation of *D* causes *A*, *B*, and *D* to rotate counterclockwise as a unit about *A*. The reversal of input motion instantly swivels *F* out of *G*, thus unlocking the shaft, which remains stationary during its return stroke because of friction induced by its load. As *D* continues to rotate clockwise about *E*, node *H*, which is hardened and polished to reduce friction, bears against the bottom of *G* to restrain further swiveling. Lever *D* now rotates with *B* around *A* until the end of the stroke.

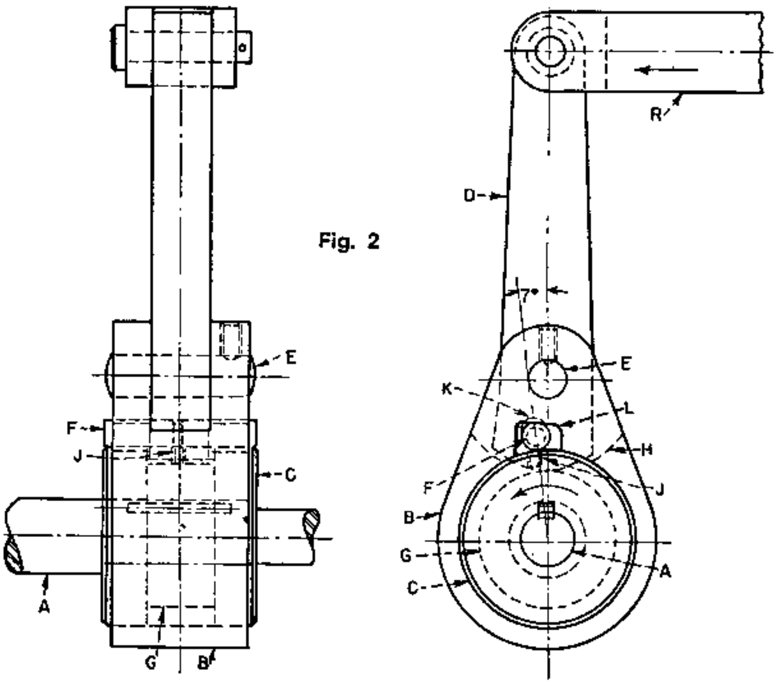


Fig. 2 The pin and disk mechanism: Lever *D*, which pivots around *E*, contains pin *F* in an elongated hole *K*. The hole permits slight vertical movement of the pin, but set screw *J* prevents horizontal movement. Body *B* can rotate freely about shaft *A*. Cut-outs *L* and *H* in body *B* allow clearances for pin *F* and lever *D*, respectively. Ring *C*, which is keyed to shaft *A*, has an annular groove *G* to permit clearance for the tip of lever *D*. Counterclockwise motion of lever *D*, actuated by the connecting rod, jams a pin between *C* and the top of cut-out *L*. This occurs about 7º from the vertical axis. *A*, *B*, and *D* are now locked together and rotate about *A*. The return stroke of *R* pivots *D* clockwise around *E* and unwedges the pin until it strikes the side of *L*. Continued motion of *R* to the right rotates *B* and *D* clockwise around *A*, while the uncoupled shaft remains stationary because of its load.

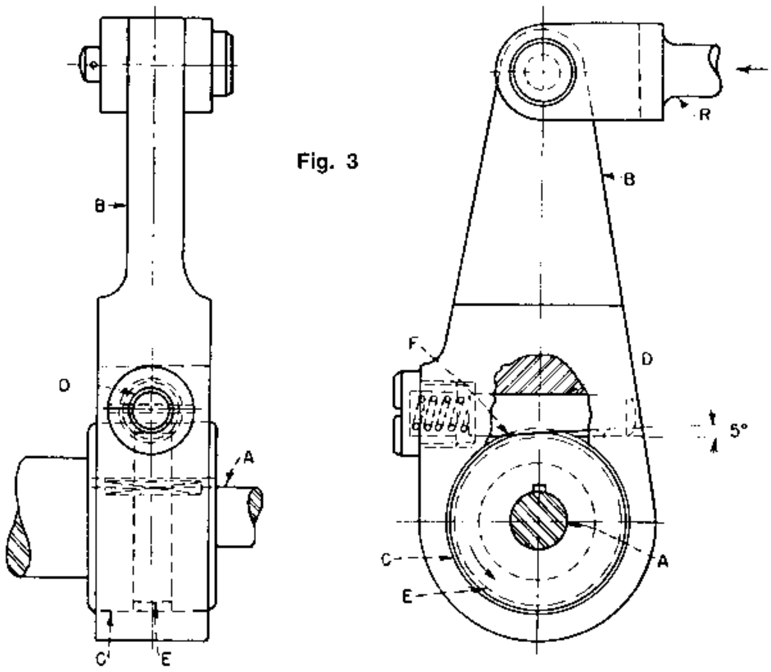


Fig. 3

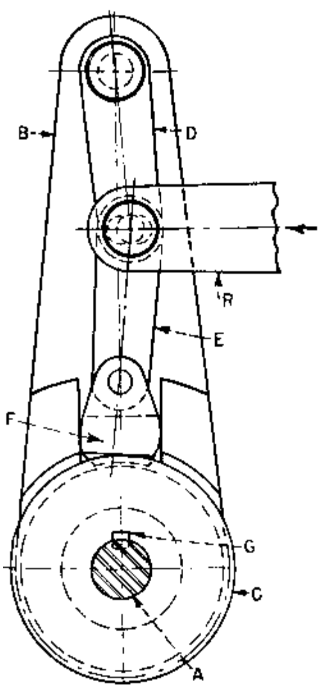


Fig. 4

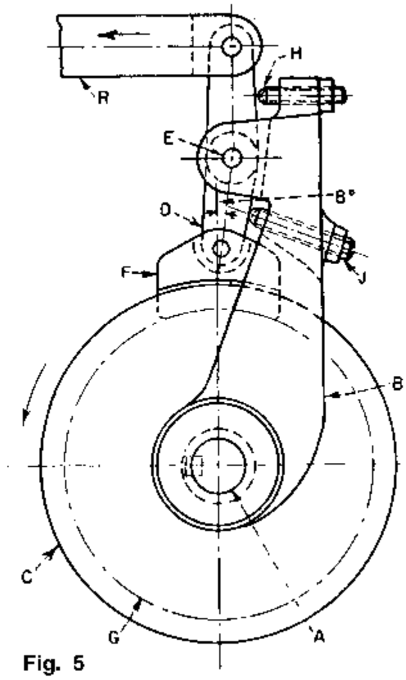


Fig. 5

Fig. 3 The sliding pin and disk mechanism: The counterclockwise movement of body *B* about shaft *A* draws pin *D* to the right with respect to body *B*, aided by spring pressure, until the flat bottom *F* of the pin is wedged against the annular groove *E* of ring *C*. The bottom of the pin is inclined about 5º for optimum wedging action. Ring *C* is keyed to *A*, and parts, *A*, *C*, *D* and *B* now rotate counterclockwise as a unit until the end of the connecting rod's stroke. The reversal of *B* draws the pin out of engagement so that *A* remains stationary while the body completes its clockwise rotation.

Fig. 4 The toggle link and disk mechanism: The input stroke of connecting rod *R* (to the left) wedges block *F* in groove *G* by straightening toggle links *D* and *E*. Body *B*, toggle links, and ring *C*, which is keyed to shaft *A*, rotate counterclockwise together about *A* until the end of the stroke. The reversal of connecting rod motion lifts the block, thus uncoupling the shaft, while body *B* continues clockwise rotation until the end of stroke.

Fig. 5 The rocker arm and disk mechanism: Lever *D*, activated by the reciprocating bar *R* moving to the left, rotates counterclockwise on pivot *E*, thus wedging block *F* into groove *G* of disk *C*. Shaft *A* is keyed to *C* and rotates counterclockwise as a unit with body *B* and lever *D*. The return stroke of *R* to the right pivots *D* clockwise about *E* and withdraws the block from the groove so that shaft is uncoupled while *D*, striking adjusting screw *H*, travels with *B* about *A* until the completion of stroke. Adjusting screw *J* prevents wedging block *F* from jamming in the groove.

NO TEETH ON THESE RATCHETS

Ratchets with springs, rollers, and other devices keep motion going one way.

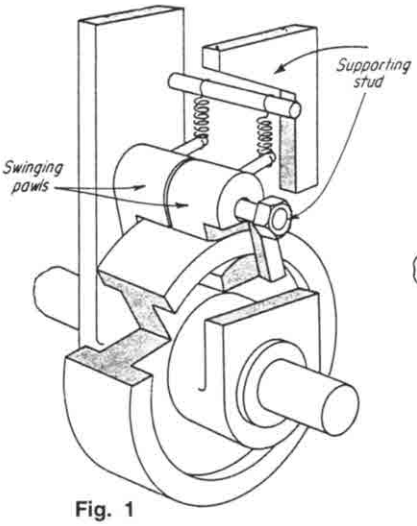


Fig. 1

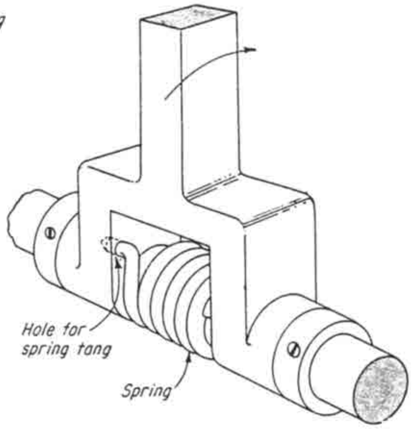


Fig. 2

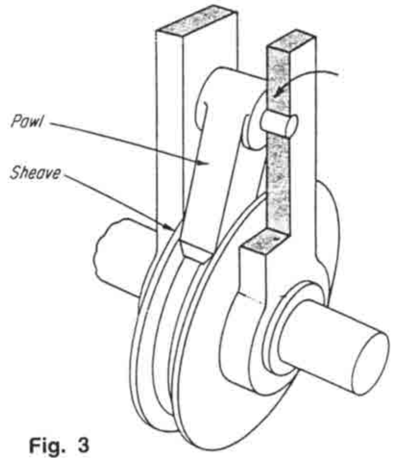


Fig. 3

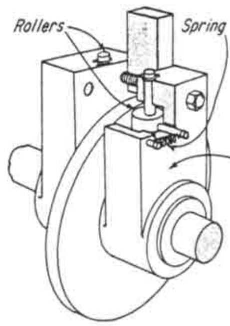


Fig. 4

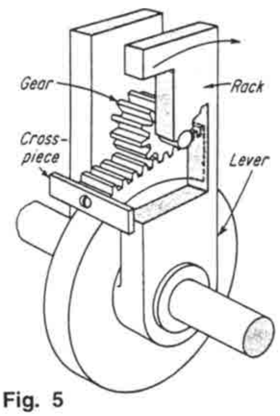


Fig. 5

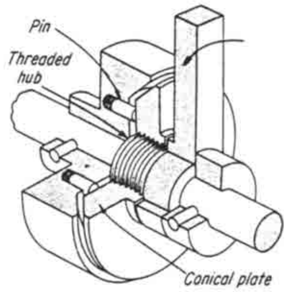


Fig. 6

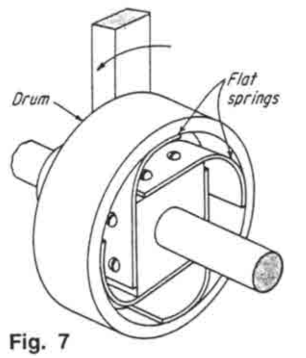


Fig. 7

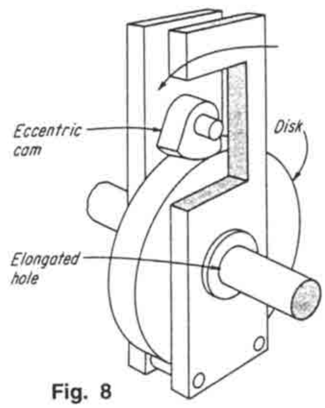


Fig. 8

Fig. 1 Swinging pawls lock on the rim when the lever swings forward, and release on the return stroke. Oversize holes for the supporting stud make sure that both the top and bottom surfaces of the pawls make contact.

Fig. 2 A helical spring grips the shaft because its inner diameter is smaller than the outer diameter of shaft. During the forward stroke, the spring winds tighter; during the return stroke, it expands.

Fig. 3 A V-belt sheave is pushed around when pawl wedges in the groove. For a snug fit, the bottom of the pawl is tapered like a V-belt.

Fig. 4 Eccentric rollers squeeze a disk on its forward stroke. On the return stroke, rollers rotate backwards and release their grip. Springs keep the rollers in contact with the disk.

Fig. 5 A rack is wedge-shaped so that it jams between the rolling gear and the disk, pushing the shaft forward. When the driving lever makes its return stroke, it carries along the unattached rack by the cross-piece.

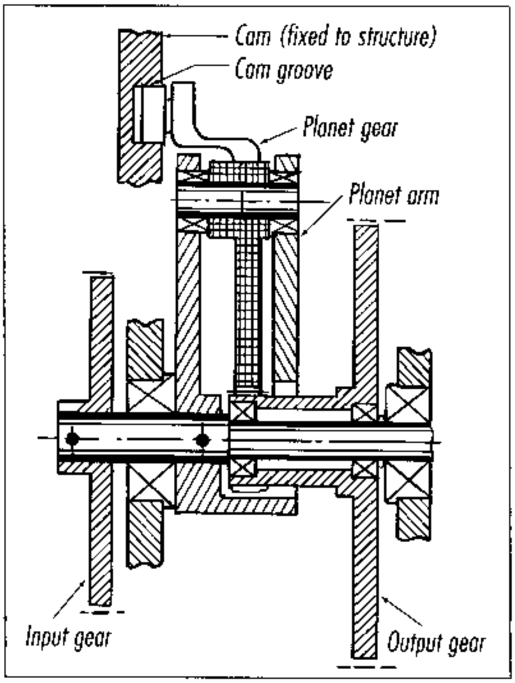
Fig. 6 A conical plate moves like a nut back and forth along the threaded center hub of the lever. The light friction of spring-loaded pins keeps the plate from rotating with the hub.

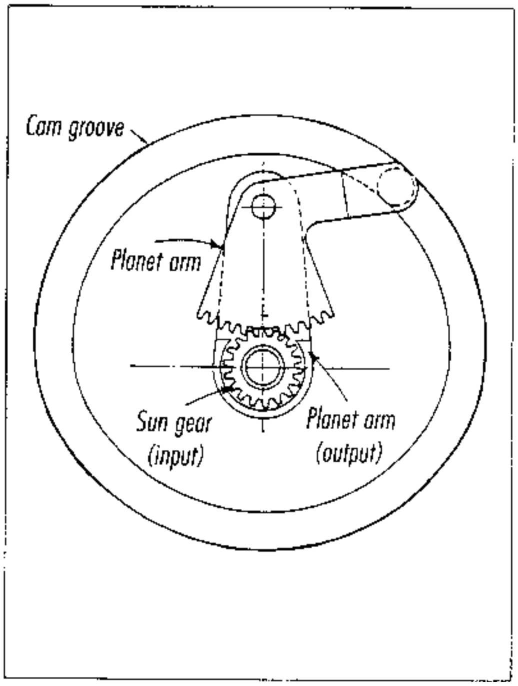
Fig. 7 Flat springs expand against the inside of a drum when a lever moves one way, but they drag loosely when the lever turns the drum in the opposite direction.

Fig. 8 An eccentric cam jams against the disk during the motion half of a cycle. Elongated holes in the levers allow the cam to wedge itself more tightly in place.

CAM-CONTROLLED PLANETARY GEAR SYSTEM

By incorporating a grooved cam a novel mechanism can produce a wide variety of output motions.





Construction details of a cam-planetary mechanism used in a film drive.

Do you want more variety in the kinds of output motion given by a planetary gear system? You can have it by controlling the planet with a grooved cam. The method gives the mechanism these additional features:

- Intermittent motion, with long dwells and minimum acceleration and deceleration.
- Cyclic variations in velocity.
- Two levels, or more, of constant speed during each cycle of the input.

The design is not simple because of need to synchronize the output of the planetary system with the cam contour. However, such mechanisms are now at work in film drives and should prove useful in many automatic machines. Here are equations, tables, and a step-by-step sequence that will make the procedure easier.

How the Mechanism Works

The planet gear need not be cut in full—a gear sector will do because the planet is never permitted to make a full revolution. The sun gear is integral with the output gear. The planet arm is fixed to the input shaft, which is coaxial with the output shaft. Attached to the planet is a follower roller which rides in a cam groove. The cam is fixed to the frame.

The planet arm (input) rotates at constant velocity and makes one revolution with each cycle. Sun gear (output) also makes one revolution during each cycle. Its motion is modified, however, by the oscillatory motion of the planet gear relative to the planet arm. It is this motion that is controlled by the cam (a constant-radius cam would not affect the output, and the drive would give only a constant one-to-one ratio).

Comparison with Other Devices

A main feature of this cam-planetary mechanism is its ability to produce a wide range of nonhomogeneous functions. These functions can be defined by no less than two mathematical expressions, each valid for a discrete portion of the range. This feature is not shared by the more widely known intermittent mechanisms: the external and internal genevas, the three-gear drive, and the cardioid drive.

Either three-gear or cardioid can provide a dwell period—but only for a comparatively short period of the cycle. With the cam-planetary, one can obtain over 180º of dwell during a 360º cycle by employing a 4-to-1 gear ratio between planet and sun.

And what about a cam doing the job by itself? This has the disadvantage of producing reciprocating motion. In other words, the output will always reverse during the cycle—a condition unacceptable in many applications.

Design Procedure

The basic equation for an epicyclic gear train is:

$$d\theta_S = d\,\theta_A - n d\theta_{P-A}$$

where: $d\theta_S$ **= rotation of sun gear (output), deg**
$d\theta_A$ **= rotation of planet arm (input), deg**
$d\theta_{P-A}$ **= rotation of planet gear with respect to arm, deg**
n **= ratio of planet to sun gear.**

The required output of the system is usually specified in the form of kinematic curves. Design procedure then is to:

- Select the proper planet-sun gear ratio
- Develop the equations of the planet motion (which also functions as a cam follower)
- Compute the proper cam contour