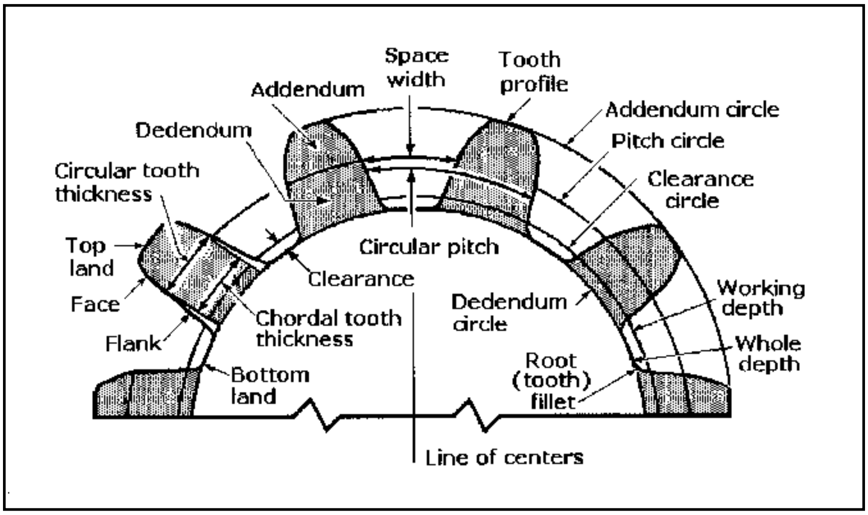
CHAPTER 8

GEARED SYSTEMS AND VARIABLE-SPEED MECHANISMS



Gear tooth terminology

Gears are versatile mechanical components capable of performing many different kinds of power transmission or motion control. Examples of these are

- Changing rotational speed.
- Changing rotational direction.
- Changing the angular orientation of rotational motion.
- Multiplication or division of torque or magnitude of rotation.
- Converting rotational to linear motion and its reverse.
- Offsetting or changing the location of rotating motion.

Gear Tooth Geometry: This is determined primarily by pitch, depth, and pressure angle.

Gear Terminology

addendum: The radial distance between the *top land* and the *pitch circle*.

addendum circle: The circle defining the outer diameter of the gear.

circular pitch: The distance along the pitch circle from a point on one tooth to a corresponding point on an adjacent tooth. It is also the sum of the *tooth thickness* and the space width, measured in inches or millimeters.

clearance: The radial distance between the *bottom land* and the *clearance circle*.

contact ratio: The ratio of the number of teeth in contact to the number of those not in contact.

dedendum circle: The theoretical circle through the *bottom lands* of a gear.

dedendum: The radial distance between the *pitch circle* and the *dedendum circle.*

depth: A number standardized in terms of pitch. Full-depth teeth have a *working depth* of $2/P$. If the teeth have equal *addenda* (as in standard interchangeable gears), the addendum is $1/P$. Full-depth gear teeth have a larger contact ratio than stub teeth, and their working depth is about 20% more than that of stub gear teeth. Gears with a small number of teeth might require *undercutting* to prevent one interfering with another during engagement.

diametral pitch (*P*): The ratio of the number of teeth to the *pitch diameter*. A measure of the coarseness of a gear, it is the index of tooth size when U.S. units are used, expressed as teeth per inch.

pitch: A standard pitch is typically a whole number when measured as a *diametral pitch* (*P*). *Coarse-pitch gears* have teeth larger than a diametral pitch of 20 (typically 0.5 to 19.99). *Fine-pitch gears* usually have teeth of diametral pitch greater than 20. The usual maximum fineness is 120 diametral pitch, but involute-tooth gears can be made with diametral pitches as fine as 200, and cycloidal tooth gears can be made with diametral pitches to 350.

pitch circle: A theoretical circle upon which all calculations are based.

pitch diameter: The diameter of the *pitch circle*, the imaginary circle that rolls without slipping with the pitch circle of the mating gear, measured in inches or millimeters.

pressure angle: The angle between the *tooth profile* and a line perpendicular to the *pitch circle*, usually at the point where the pitch circle and the tooth profile intersect. Standard angles are 20 and 25º. The pressure angle affects the force that tends to separate mating gears. A high pressure angle decreases the *contact ratio*, but it permits the teeth to have higher capacity and it allows gears to have fewer teeth without *undercutting*.

Gear Dynamics Terminology

backlash: The amount by which the width of a tooth space exceeds the thickness of the engaging tooth measured on the pitch circle. It is the shortest distance between the noncontacting surfaces of adjacent teeth.

gear efficiency: The ratio of output power to input power, taking into consideration power losses in the gears and bearings and from windage and churning of lubricant.

gear power: A gear's load and speed capacity, determined by gear dimensions and type. Helical and helical-type gears have capacities to approximately 30,000 hp, spiral bevel gears to about 5000 hp, and worm gears to about 750 hp.

gear ratio: The number of teeth in the *gear* (larger of a pair) divided by the number of teeth in the *pinion* (smaller of a pair). Also, the ratio of the speed of the pinion to the speed of the gear. In reduction gears, the ratio of input to output speeds.

gear speed: A value determined by a specific pitchline velocity. It can be increased by improving the accuracy of the gear teeth and the balance of rotating parts.

undercutting: Recessing in the bases of gear tooth flanks to improve clearance.

Gear Classification

External gears have teeth on the outside surface of a disk or wheel.

Internal gears have teeth on the inside surface of a cylinder.

Spur gears are cylindrical gears with teeth that are straight and parallel to the axis of rotation. They are used to transmit motion between parallel shafts.

Rack gears have teeth on a flat rather than a curved surface that provide straight-line rather than rotary motion.

Helical gears have a cylindrical shape, but their teeth are set at an angle to the axis. They are capable of smoother and quieter action than spur gears. When their axes are parallel, they are called *parallel helical gears,* and when they are at right angles they are called *helical gears.* Herringbone and worm gears are based on helical gear geometry.

Herringbone gears are double helical gears with both right-hand and left-hand helix angles side by side across the face of the gear. This geometry neutralizes axial thrust from helical teeth.

Worm gears are crossed-axis helical gears in which the helix angle of one of the gears (the worm) has a high helix angle, resembling a screw.

Pinions are the smaller of two mating gears; the larger one is called the *gear* or *wheel*.

Bevel gears have teeth on a conical surface that mate on axes that intersect, typically at right angles. They are used in applications where there are right angles between input and output shafts. This class of gears includes the most common straight and spiral bevel as well as the miter and hypoid.

Straight bevel gears are the simplest bevel gears. Their straight teeth produce instantaneous line contact when they mate. These gears provide moderate torque transmission, but they are not as smooth running or quiet as spiral bevel gears because the straight teeth engage with full-line contact. They permit medium load capacity.

Spiral bevel gears have curved oblique teeth. The spiral angle of curvature with respect to the gear axis permits substantial tooth overlap. Consequently, teeth engage gradually and at least two teeth are in contact at the same time. These gears have lower tooth loading than straight bevel gears, and they can turn up to eight times faster. They permit high load capacity.

Miter gears are mating bevel gears with equal numbers of teeth and with their axes at right angles.

Hypoid gears are spiral bevel gears with offset intersecting axes.

Face gears have straight tooth surfaces, but their axes lie in planes perpendicular to shaft axes. They are designed to mate with instantaneous point contact. These gears are used in right-angle drives, but they have low load capacities.

NUTATING-PLATE DRIVE

The Nutation Drive* is a mechanically positive, gearless power transmission that offers high single-stage speed ratios at high efficiencies. A nutating member carries camrollers on its periphery and causes differential rotation between the three major components of the drive: stator, nutator, and rotor. Correctly designed cams on the stator and rotor assure a low-noise engagement and mathematically pure rolling contact between camrollers and cams.

The drive's characteristics include compactness, high speed ratio, and efficiency. Its unique design guarantees rolling contact between the power-transmitting members and even distribution of the load among a large number of these members. Both factors contribute to the drive's inherent low noise level and long, maintenance-free life. The drive has a small number of moving parts; furthermore, commercial grease and solid lubrication provide adequate lubrication for many applications.

Kinetics of the Nutation Drive

Basic components. The three basic components of the Nutation Drive are the stator, nutator, and rotor, as shown in Fig. 1. The nutator carries radially mounted conical camrollers

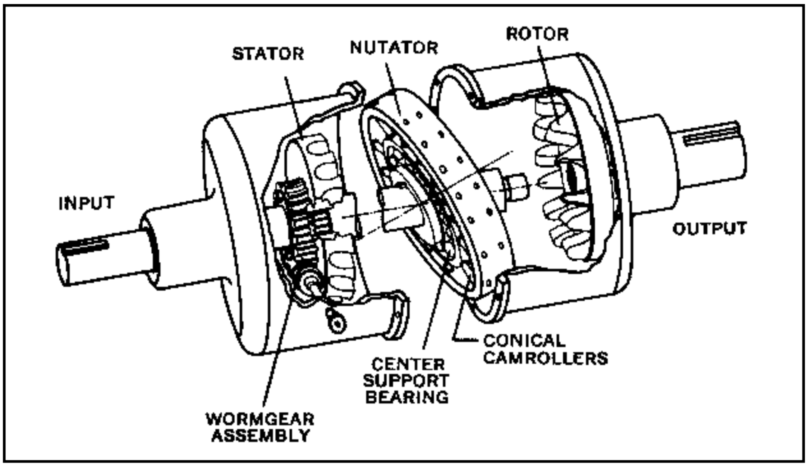


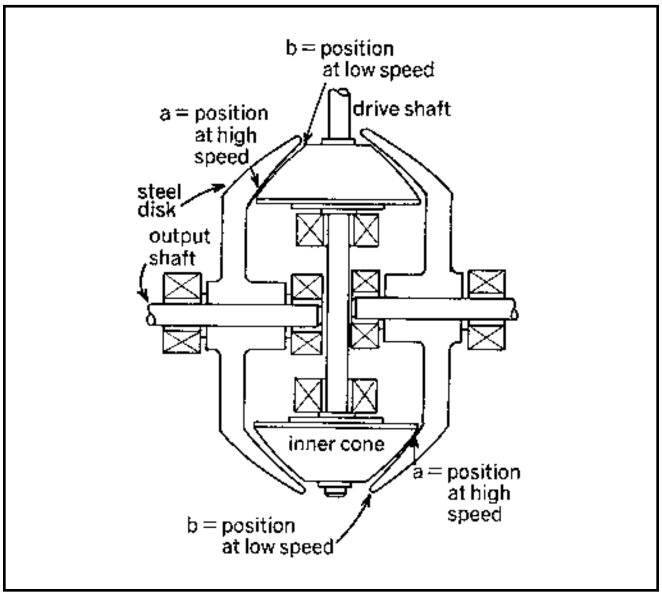
Fig. 1 An exploded view of the Nutation Drive.

that engage between cams on the rotor and stator. Cam surfaces and camrollers have a common vanishing point—the center of the nutator. Therefore, line-contact rolling is assured between the rollers and the cams.

Nutation is imparted to the nutator through the center support bearing by the fixed angle of its mounting on the input shaft. One rotation of the input shaft causes one complete nutation of the nutator. Each nutation cycle advances the rotor by an angle equivalent to the angular spacing of the rotor cams. During nutation the nutator is held from rotating by the stator, which transmits the reaction forces to the housing.

* Four U.S. patents (3,094,880, 3,139,771, 3,139,772, and 3,590,659) have been issued to A. M. Maroth.

CONE DRIVE NEEDS NO GEARS OR PULLEYS



Cone drive operates without lubrication.

A variable-speed-transmission cone drive operates without gears or pulleys. The drive unit has its own limited slip differential and clutch.

As the drawing shows, two cones made of brake lining material are mounted on a shaft directly connected to the engine. These drive two larger steel conical disks mounted on the output shaft. The outer disks are mounted on pivoting frames that can be moved by a simple control rod.

To center the frames and to provide some resistance when the outer disks are moved, two torsion bars attached to the main frame connect and support the disk-support frames. By altering the position of the frames relative to the driving cones, the direction of rotation and speed can be varied.

The unit was invented by Marion H. Davis of Indiana.

VARIABLE-SPEED MECHANICAL DRIVES

CONE DRIVES

The simpler cone drives in this group have a cone or tapered roller in combination with a wheel or belt (Fig. 1). They have evolved from the stepped-pulley system. Even the more sophisticated designs are capable of only a limited (although infinite) speed range, and generally must be spring-loaded to reduce slippage.

Adjustable-cone drive (Fig. 1A). This is perhaps the oldest variable-speed friction system, and is usually custom built. Power from the motor-driven cone is transferred to the output shaft by the friction wheel, which is adjustable along the cone side to change the output speed. The speed depends upon the ratio of diameters at point of contact.

Two-cone drive (Fig. 1B). The adjustable wheel is the power transfer element, but this drive is difficult to preload because both input and output shafts would have to be spring loaded. The second cone, however, doubles the speed reduction range.

Cone-belt drives (Fig. 1C and D). In Fig. 1C the belt envelopes both cones; in Fig. 1D a long-loop endless belt runs between the cones. Stepless speed adjustment is obtained by shifting the belt along the cones. The cross section of the belt must be large enough to transmit the rated force, but the width must be kept to a minimum to avoid a large speed differential over the belt width.

Electrically coupled cones (Fig. 2). This drive is composed of thin laminates of paramagnetic material. The laminates are separated with semidielectric materials which also localize the effect of the inductive field. There is a field generating device within the driving cone. Adjacent to the cone is a positioning motor for the field generating device. The field created in a particular section of the driving cone induces a magnetic effect in the surrounding lamination. This causes the laminate and its opposing lamination to couple and rotate with the drive shaft. The ratio of diameters of the cones, at the point selected by positioning the field-generating component, determines the speed ratio.

Fig. 1

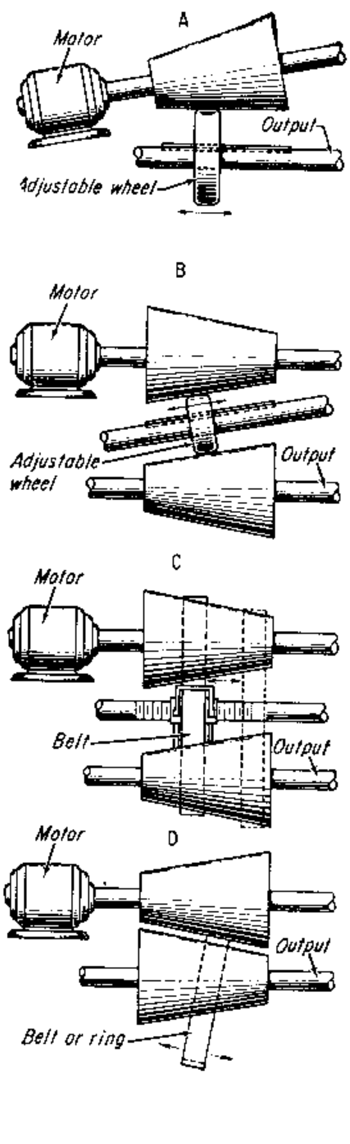


Fig. 2

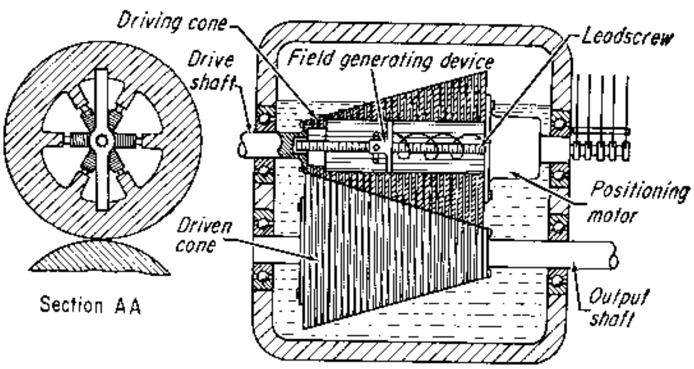
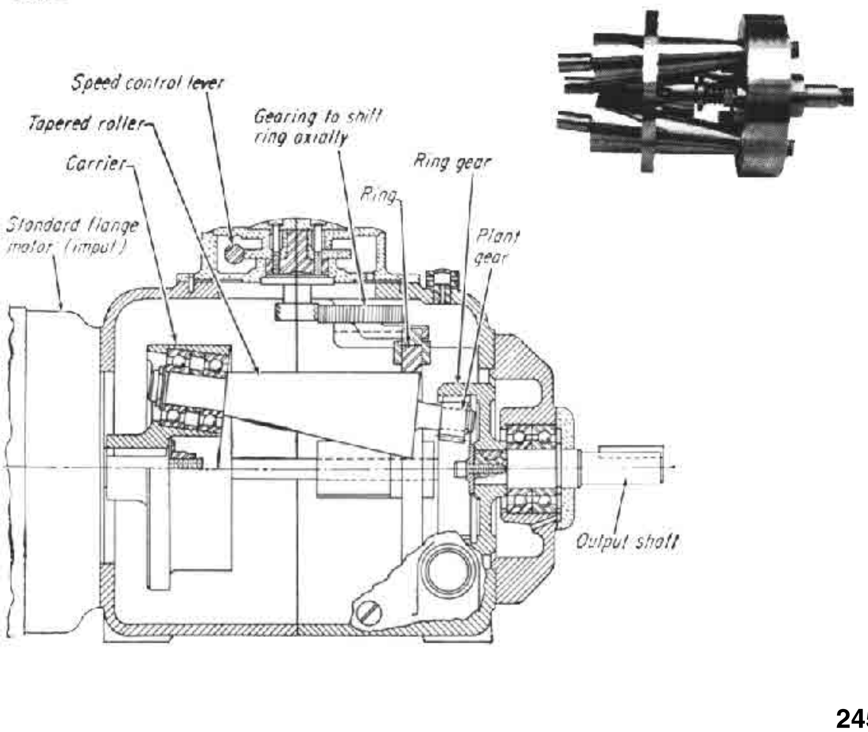


Fig. 3



Graham drive (Fig. 3). This commercial unit combines a planetary-gear set and three tapered rollers (only one of which is shown). The ring is positioned axially by a cam and gear arrangement. The drive shaft rotates the carrier with the tapered rollers, which are inclined at an angle equal to their taper so that their outer edges are parallel to the centerline of the assembly. Traction pressure between the rollers and ring is created by centrifugal force, or spring loading of the rollers. At the end of each roller a pinion meshes with a ring gear. The ring gear is part of the planetary gear system and is coupled to the output shaft.

The speed ratio depends on the ratio of the diameter of the fixed ring to the effective diameter of the roller at the point of contact, and is set by the axial position of the ring. The output speed, even at its maximum, is always reduced to about one-third of input speed because of the differential feature. When the angular speed of the driving motor equals the angular speed of the centers of the tapered rollers around their mutual centerline (which is set by the axial position of the nonrotating friction ring), the output speed is zero. This drive is manufactured in ratings up to 3 hp; efficiency reaches 85%.

Cone-and-ring drive (Fig. 4). Here, two cones are encircled by a preloaded ring. Shifting the ring axially varies the output speed. This principle is similar to that of the cone-and-belt drive (Fig. 1C). In this case, however, the contact pressure between ring and cones increases with load to limit slippage.

Planetary-cone drive (Fig. 5). This is basically a planetary gear system but with cones in place of gears. The planet cones are rotated by the sun cone which, in turn, is driven by the motor. The planet cones are pressed between an outer nonrotating rind and the planet hold. Axial adjustment of the ring varies the rotational speed of the cones around their mutual axis. This varies the speed of the planet holder and the output shaft. Thus, the mechanism resembles that of the Graham drive (Fig. 3).

The speed adjustment range of the unit illustrated if from 4:1 to 24:1. The system is built in Japan in ratings up to 2 hp.

Fig. 4

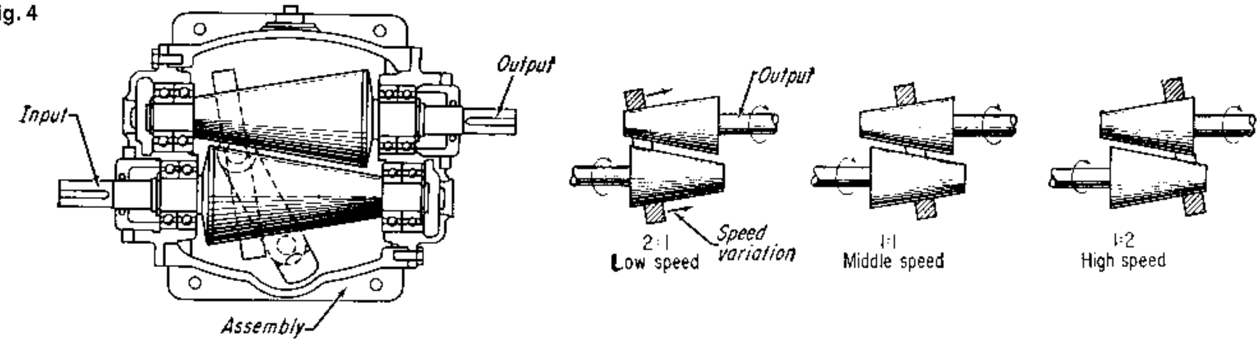
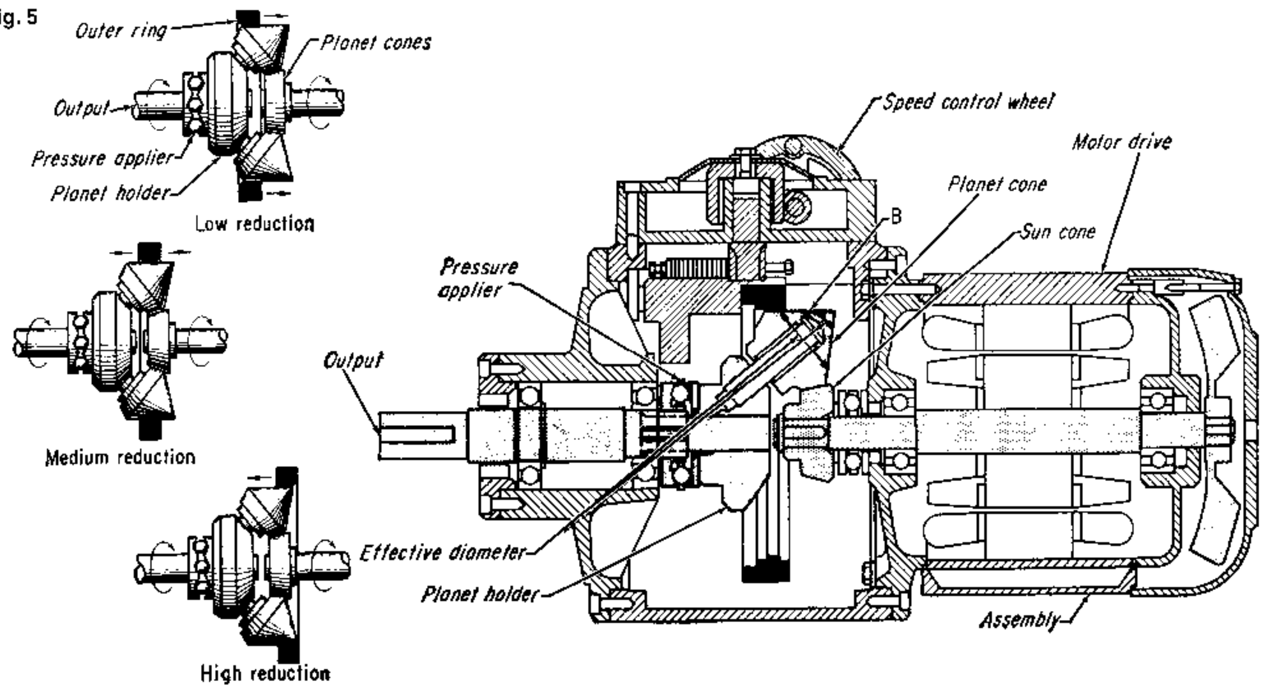


Fig. 5



Adjustable disk drives (Figs. 6A and 6B). The output shaft in Fig. 7A is perpendicular to the input shaft. If the driving power, the friction force, and the efficiency stay constant, the output torque decreases in proportion to increasing output speed. The wheel is made of a high-friction material, and the disk is made of steel. Because of relatively high slippage, only small torques can be transmitted. The wheel can move over the center of the disk because this system has infinite speed adjustment.

To increase the speed, a second disk can be added. This arrangement (Fig. 6B) also makes the input and output shafts parallel.

Spring-loaded disk drive (Fig. 7). To reduce slippage, the contact force between the rolls and disks in this commercial drive is increased with the spring assembly in the output shaft. Speed adjustments are made by rotating the leadscrew to shift the cone roller in the vertical direction. The drive illustrated has a 4-hp capacity. Drives rated up to 20 hp can have a double assembly of rollers. Efficiency can be as high as 92%. Standard speed range is 6:1, but units of 10:1 have been build. The power transferring components, which are made hardened steel, operate in an oil mist, thus minimizing wear.

Planetary disk drive (Fig. 8). Four planet disks replace planet gears in this friction drive. Planets are mounted on levers which control radial position and therefore control the orbit. Ring and sun disks are spring-loaded.

Fig. 6

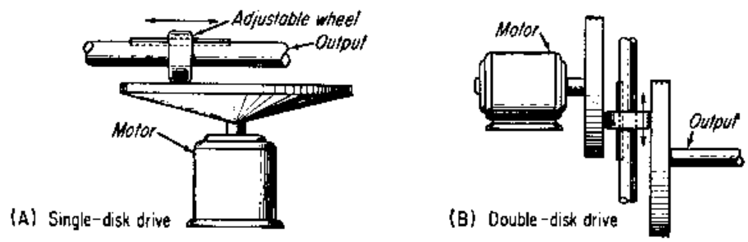


Fig. 7

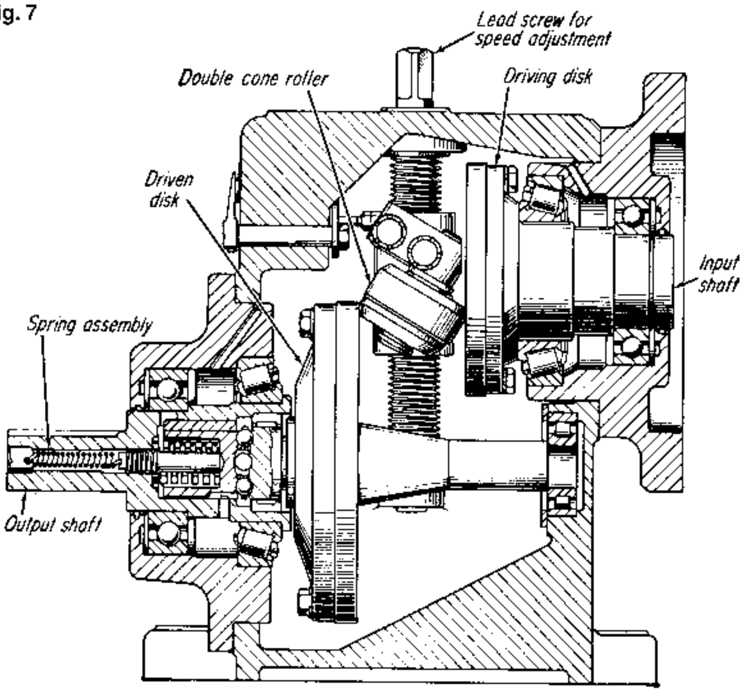
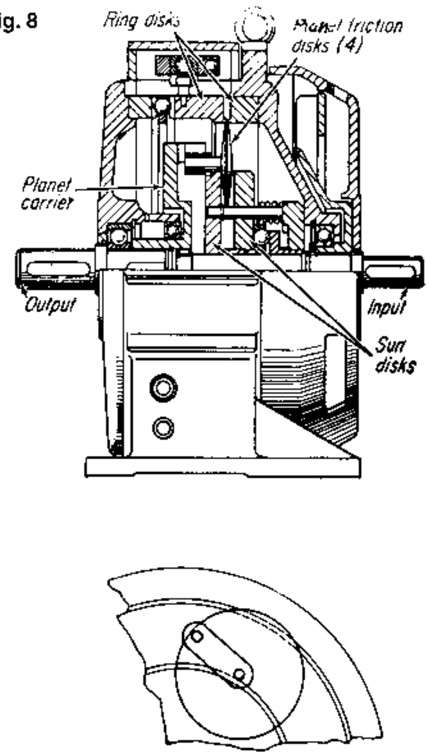


Fig. 8



Ring-and-pulley drive (Fig. 9). A thick steel ring in this drive encircles two variable-pitch (actually variable-width) pulleys. A novel gear-and-linkage system simultaneously changes the width of both pulleys (see Fig. 9B). For example, when the top pulley opens, the sides of the bottom pulley close up. This reduces the effective pitch diameter of the top pulley and increases that of the bottom pulley, thus varying the output speed.

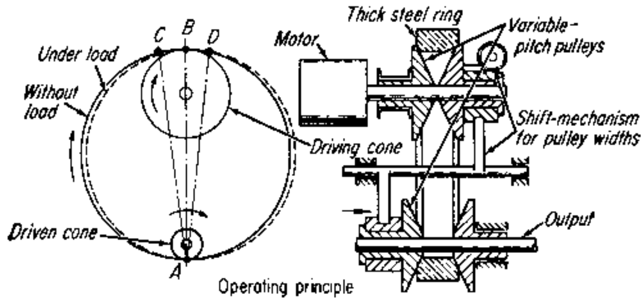
Normally, the ring engages the pulleys at points *A* and *B*. However, under load, the driven pulley resists rotation and the contact point moves from *B* to *D* because of the very small elastic deformation of the ring. The original circular shape of the ring is changed to a slightly oval form, and the distance between points of contact decreases. This wedges the ring between the pulley cones and increases the contact pressure between ring and pulleys in proportion to the load applied, so that constant horsepower at all speeds is obtained. The drive can have up to 3-hp capacity; speed variations can be 16:1, with a practical range of about 8:1.

Some manufacturers install rings with unusual cross sections (Fig. 10) formed by inverting one of the sets of sheaves.

Double-ring drive (Fig. 11). Power transmission is through two steel traction rings that engage two sets of disks mounted on separate shafts. This drive requires that the outer disks be under a compression load by a spring system (not illustrated). The rings are hardened and convex-ground to reduce wear. Speed is changed by tilting the ring support cage, forcing the rings to move to the desired position.

Fig. 9

a



b

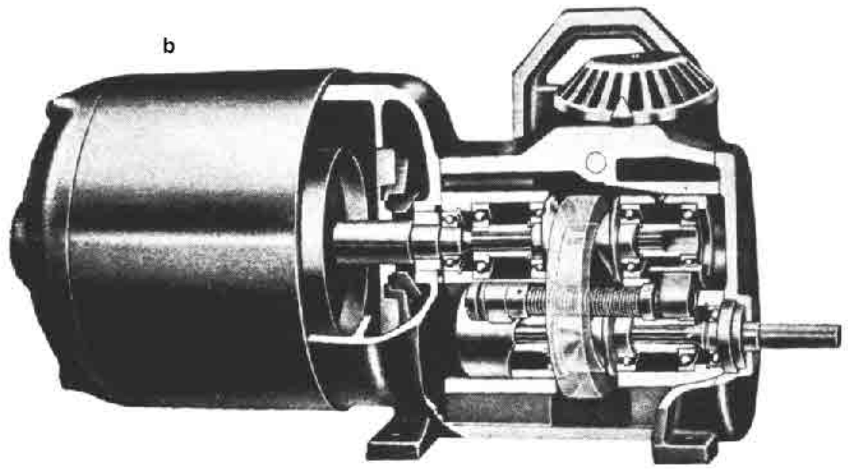


Fig. 10

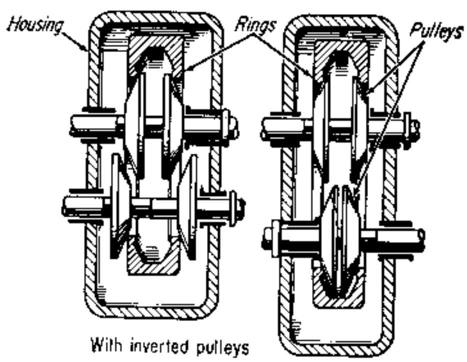
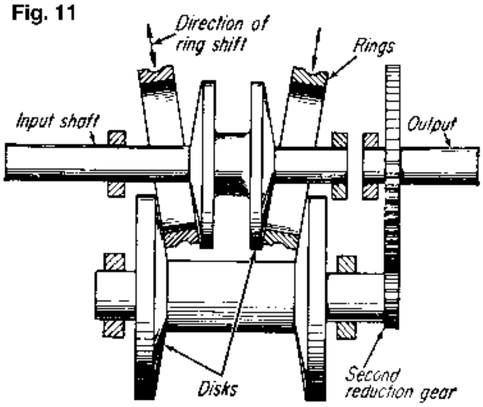


Fig. 11



Sphere-and-disk drives (Figs. 12 and 13). The speed variations in the drive shown in Fig. 12 are obtained by changing the angle that the rollers make in contacting spherical disks. As illustrated, the left spherical disk is keyed to the driving shaft and the right disk contains the output gear. The sheaves are loaded together by a helical spring.

One commercial unit, shown in Fig. 13, is a coaxial input and output shaft-version of the Fig. 12 arrangement. The rollers are free to rotate on bearings and can be adjusted to any speed between the limits of 6:1 and 10:1. An automatic device regulates the contact pressure of the rollers, maintaining the pressure exactly in proportion to the imposed torque load.

Double-sphere drive (Fig. 14). Higher speed reductions are obtained by grouping a second set of spherical disks and rollers. This also reduces operating stresses and wear. The input shaft runs through the unit and carries two opposing spherical disks. The disks drive the double-sided output disk through two sets of three rollers. To change the ratio, the angle of the rollers is varied. The disks are axially loaded by hydraulic pressure.

Tilting-ball drive (Fig. 15). Power is transmitted between disks by steel balls whose rotational axes can be tilted to change the relative lengths of the two contact paths around the balls, and hence the output speed. The ball axes can be tilted uniformly in either direction; the effective rolling radii of balls and disks produce speed variations up to 3:1 increase, or 1:3 decrease, with the total up to 9:1 variation in output speed.

Tilt is controlled by a cam plate through which all ball axes project. To prevent slippage under starting or shock load, torque responsive mechanisms are located on the input and output sides of the drive. The axial pressure created is proportional to the applied torque. A worm drive positions the plate. The drives have been manufactured with capacities to 15-hp. The drive's efficiency is plotted in the chart.

Sphere and roller drive (Fig. 16). The roller, with spherical end surfaces, is

Fig. 12

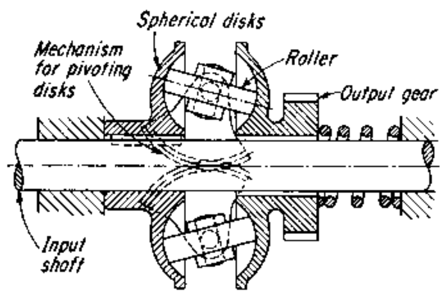


Fig. 13

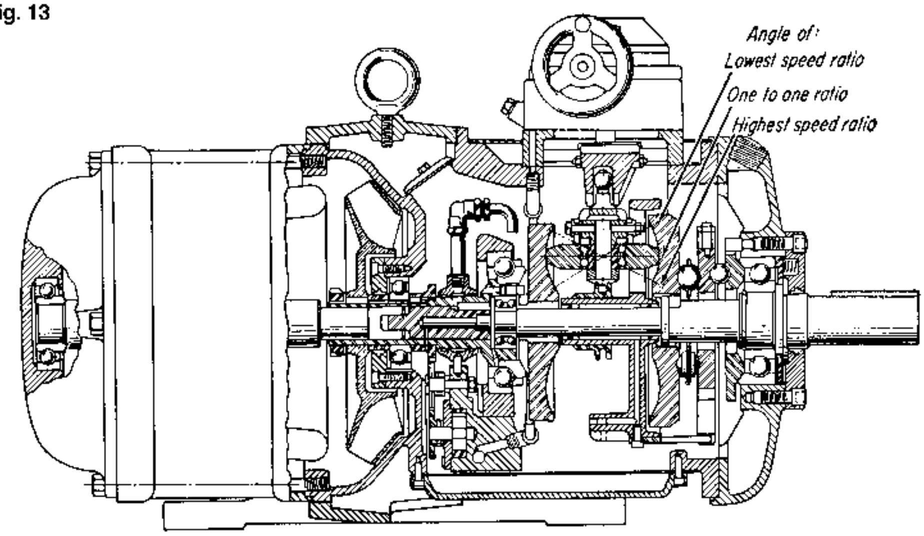


Fig. 14

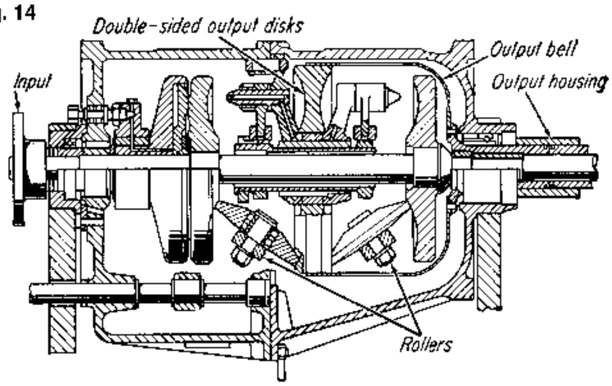
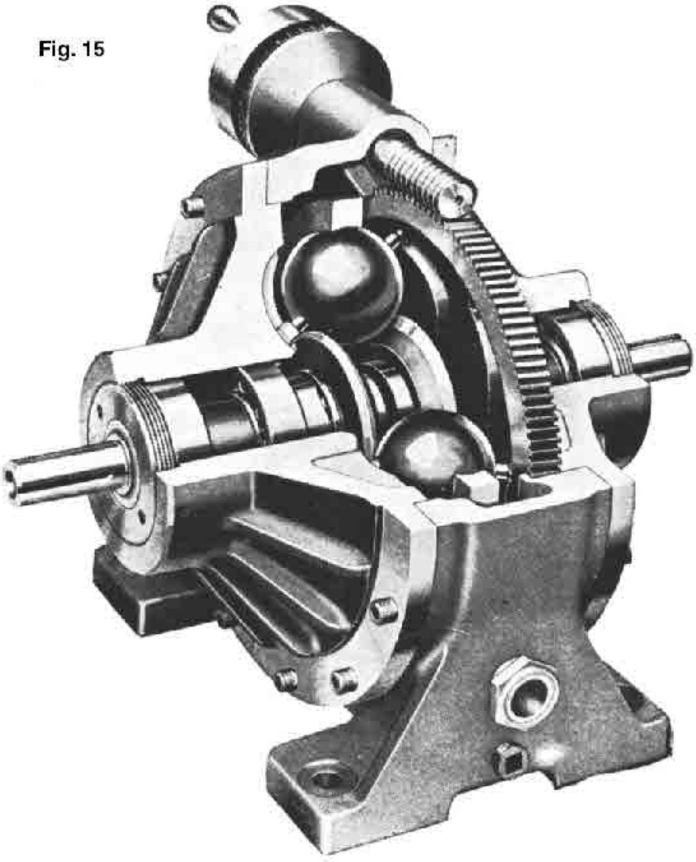
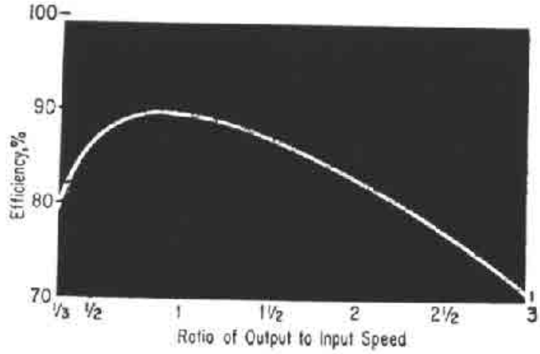


Fig. 15





Efficiency of tilting-ball drive

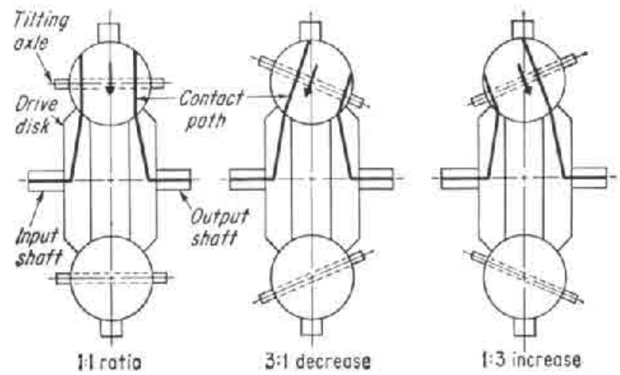
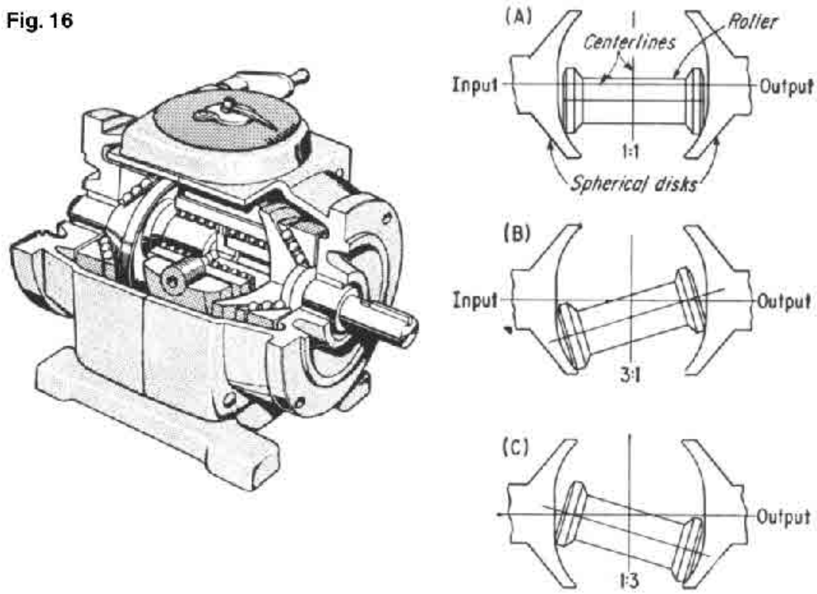


Fig. 16



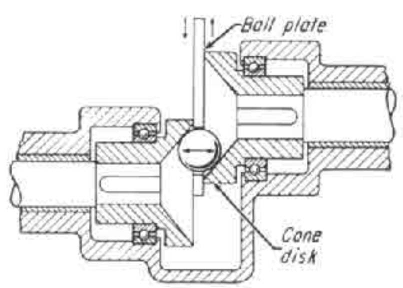
eccentrically mounted between the coaxial input and output spherical disks. Changes in speed ratio are made by changing the angular position of the roller.

The output disk rotates at the same speed as the input disk when the roller centerline is parallel to the disk centerline, as in Fig. 16A. When the contact point is nearer the centerline on the output disk and further from the centerline on the input disk, as in Fig. 16B, the output speed exceeds that of the input. Conversely, when the roller contacts the output disk at a large radius, as in Fig. 16C, the output speed is reduced.

A loading cam maintains the necessary contact force between the disks and power roller. The speed range reaches 9 to 1; efficiency is close to 90%.

Ball-and-cone drive (Fig. 17). In this simple drive the input and output shafts are offset. Two opposing cones with 90º internal vertex angles are fixed to each shaft. The shafts are preloaded against each other. Speed variation is obtained by positioning the ball that contacts the cones. The ball can shift laterally in relation to the ball plate. The conical cavities, as well as the ball, have hardened surfaces, and the drive operates in an oil bath.

Fig. 17



Ball-and-disk drive (Fig. 18). Friction disks are mounted on splined shafts to allow axial movement. The steel balls carried by swing arms rotate on guide rollers, and are in contact with driving and driven disks. Belleville springs provide the loading force between the balls and the disks. The position of the balls controls the ratio of contact radii, and thus the speed.

Only one pair of disks is required to provide the desired speed ratio; the multiple disks increase the torque capacity. If the load changes, a centrifugal loading device increases or decreases the axial pressure in proportion to the speed. The helical gears permit the output shaft to be coaxial with respect to the input shaft. Output to input speed ratios are from 1 to 1 to 1 to 5, and the drive's efficiency can reach 92%. Small ball and disk drives are rated to 9 hp, and large ball and disk drives are rated to 38 hp.

Oil-coated disks (Fig. 19). Power is transmitted without metal-to-metal contact at 85% efficiency. The interleaved disk sets are coated with oil when operating. At their points of contact, axial pressure applied by the rim disks compresses the oil film, increasing its viscosity. The cone disks transmit motion to the rim disks by shearing the molecules of the high-viscosity oil film.

Three stacks of cone disks (only one stack is shown) surround the central rim stack. Speed is changed by moving the cones radially toward the rim disks (output speed increases) or away from the rim disks (output speed decreases). A spring and cam on the output shaft maintain the pressure of the disks at all times.

Drives with ratings in excess of 60 hp have been built. The small drives are cooled, but water cooling is required for the larger units.

Under normal conditions, the drive can transmit its rated power with a 1% slip at high speeds and 3% slip at low speeds.

Fig. 18

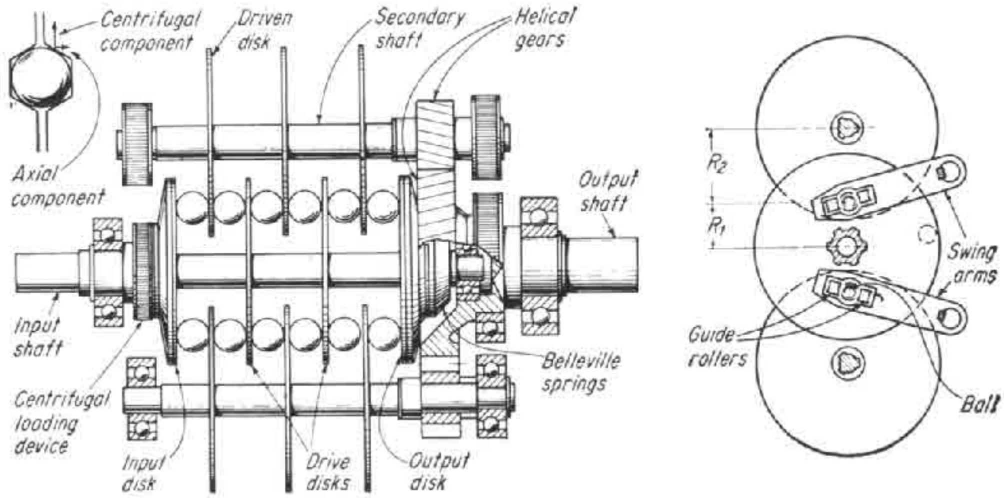
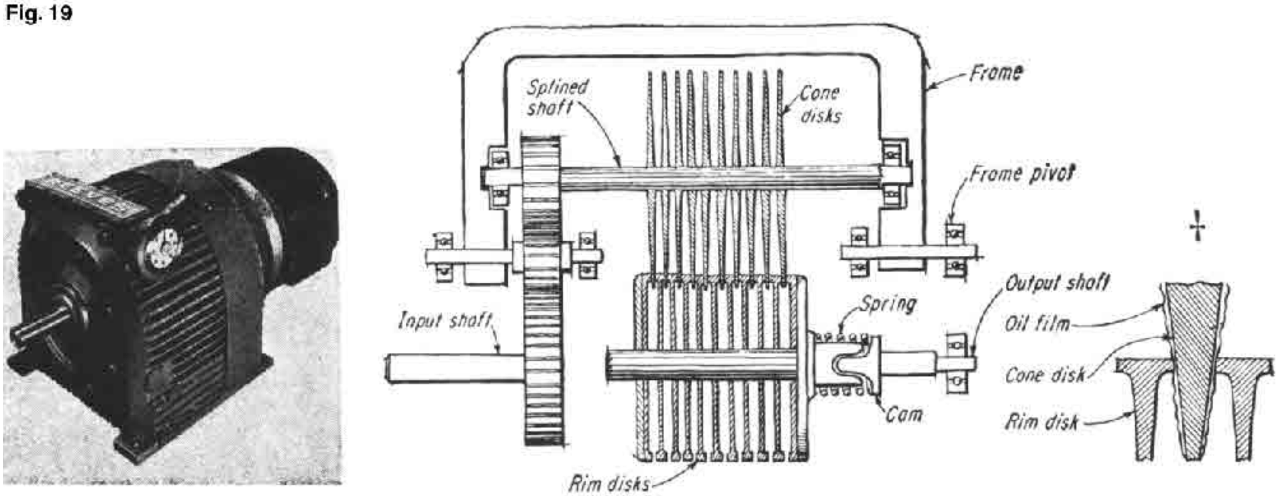


Fig. 19



Variable-stroke drive (Fig. 20). This drive is a combination of a four-bar linkage with a one-way clutch or ratchet. The driving member rotates the eccentric that, through the linkage, causes the output link to rotate a fixed amount. On the return stroke, the output link overrides the output shaft. Thus a pulsating motion is transmitted to the output shaft, which in many applications such as feeders and mixers, is a distinct advantage. Shifting the adjustable pivot varies the speed ratio. By adding eccentrics, cranks, and clutches in the system, the frequency of pulsations per revolution can be increased to produce a smoother drive.

Morse drive (Fig. 21). The oscillating motion of the eccentric on the output shaft imparts motion to the input link, which in turn rotates the output gears. The travel of the input link is regulated by the control link that oscillates around its pivot and carries the roller, which rides in the eccentric cam track. Usually, three linkage systems and gear assemblies overlap the motions: two linkages on return, while the third is driving. Turning the handle repositions the control link and changes the oscillation angles of the input link, intermediate gear, and input gear. This is a constant-torque drive with limited range. The maximum torque output is 175 ft-lb at the maximum input speed of 180 rpm. Speed can be varied between 4.5 to 1 and 120 to 1.

Zero-Max drive (Fig. 22). This drive is also based on the variable-stroke principle. With an 1800-rpm input, it will deliver 7200 or more impulses per minute to the output shaft at all speed ratings above zero. The pulsations of this drive are damped by several parallel sets of mechanisms between the input and output shafts. (Figure 22 shows only one of these sets.)

At zero input speed, the eccentric on the input shaft moves the connecting rod up and down through an arc. The main link has no reciprocating motion. To set the output speed, the pivot is moved (upward in the figure), thus changing the direction of the connecting rod motion and imparting an oscillatory motion to the main link. The one-way clutch mounted on the output shaft provides the ratchet action. Reversing the input shaft rotation does not reverse the output. However, the drive can be reversed in two ways: (1) with a special reversible clutch, or (2) with a bellcrank mechanism in gearhead models.

This drive is classified as an infinite-speed range drive because its output speed passes through zero. Its maximum speed is 2000rpm, and its speed range is from zero to one-quarter of its input speed. It has a maximum rated capacity of ¾ hp.

Fig. 20

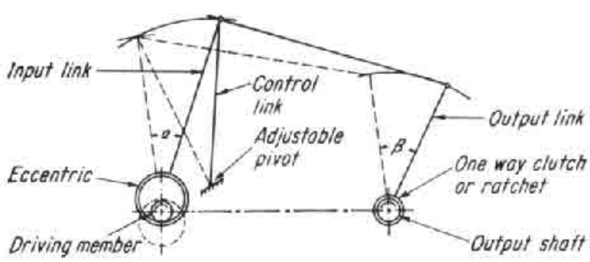


Fig. 21

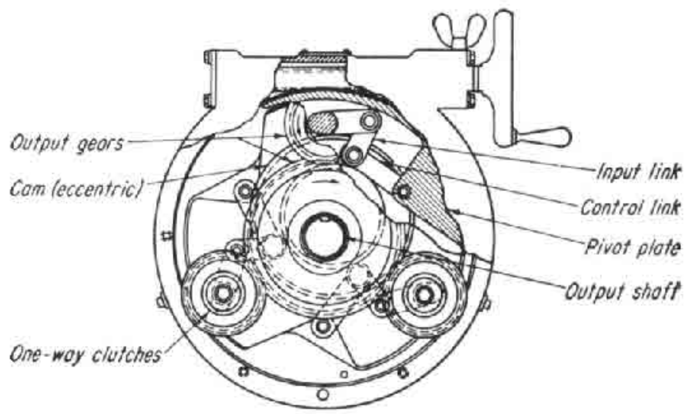
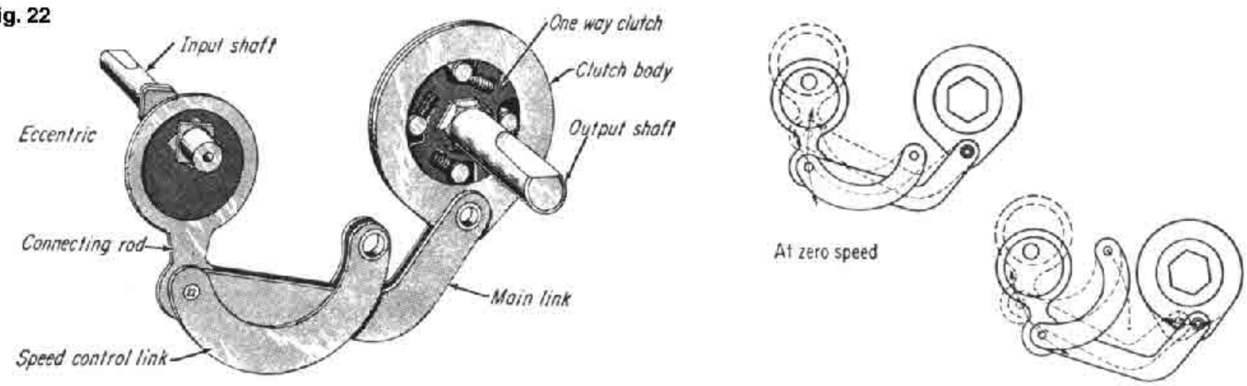


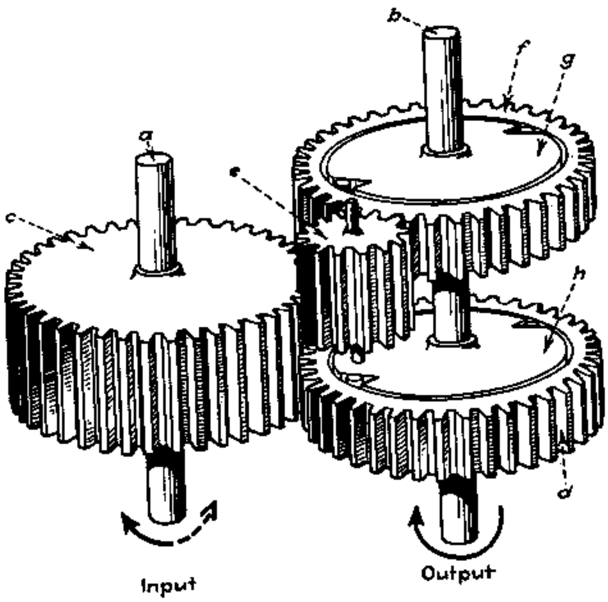
Fig. 22



UNIDIRECTIONAL DRIVE

The output shaft of this unidirectional drive rotates in the same direction at all times, without regard to the direction of the rotation of the input shaft. The angular velocity of the output shaft is directly proportional to the angular velocity of the input shaft. Input shaft *a* carries spur gear *c,* which has approximately twice the face width of spur gears *f* and *d* mounted on output shaft *b.* Spur gear *c* meshes with idler *e* and with spur gear *d.* Idler *e* meshes with spur gears *c* and *f.* The output shaft *b* carries two free-wheel disks *g* and *h,* which are oriented unidirectionally.

When the input shaft rotates clockwise (bold arrow), spur gear *d* rotates counter-clockwise and idles around free-wheel disk *h.* At the same time idler *e,* which is also rotating counter-clockwise, causes spur gear *f* to turn clockwise and engage the rollers on free-wheel disk *g;* thus, shaft *b* is made to rotate clockwise. On the other hand, if the input shaft turns counter-clockwise (dotted arrow), then spur gear *f* will idle while spur gear *d* engages free-wheel disk *h,* again causing shaft *b* to rotate clockwise.



ADDITIONAL VARIATIONS

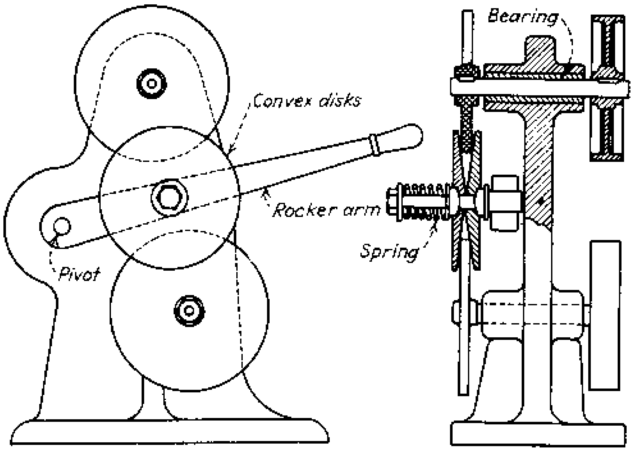


Fig. 1

Fig. 1 The Sellers' disks consist of a mechanism for transmitting power between fixed parallel shafts. Convex disks are mounted freely on a rocker arm, and they are pressed firmly against the flanges of the shaft wheels by a coiled spring to form the intermediate sheave. The speed ratio is changed by moving the rocker lever. No reverse is possible, but the driven shaft can rotate above or below the driver speed. The convex disk must be mounted on self-aligning bearings to ensure good contact in all positions.

Fig. 2 A curved disk device is formed by attaching a motor that is swung on its pivot so that it changes the effective diameters of the contact circles. This forms a compact drive for a small drill press.

Fig. 3 This is another motorized modification of the older mechanism shown in Fig. 2. It works on the principle that is similar that of Fig. 2, but it has only two shafts. Its ratio is changed by sliding the motor in vee guides.

Fig. 4 Two cones mounted close together and making contact through a squeezed belt permit the speed ratio to be changed by shifting the belt longitudinally. The taper on the cones must be moderate to avoid excessive wear on the sides of the belt.

Fig. 5 These cones are mounted at any convenient distance apart. They are connected by a belt whose outside edges consist of an envelope of tough, flexible rubberized fabric that is wear-resistant. It will withstand the wear caused by the belt edge traveling at a slightly different velocity that that part of the cone it actually contacts. The mechanism's speed ratio is changed by sliding the belt longitudinally.

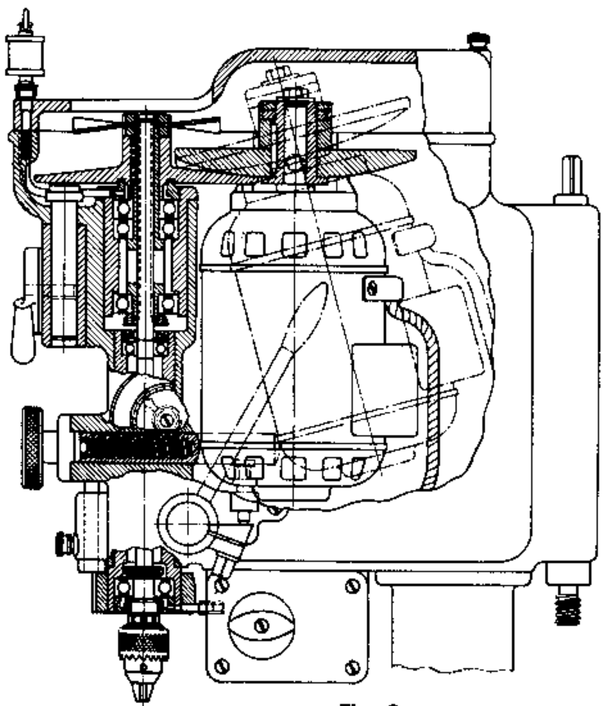


Fig. 2

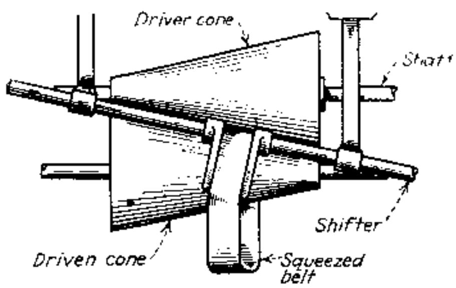


Fig. 4

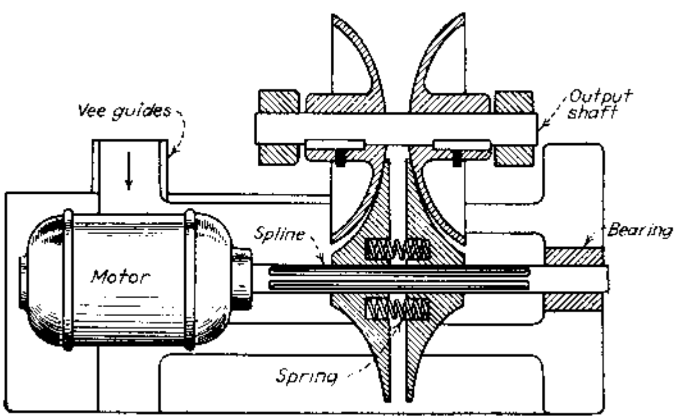


Fig. 3

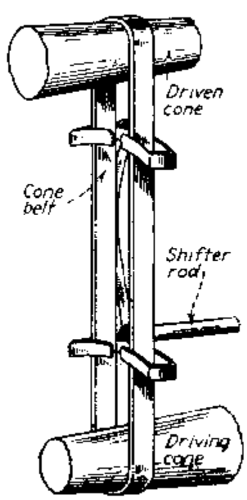


Fig. 5

Fig. 6 This drive avoids belt "creep" and wear in speed-cone transmissions. The inner bands are tapered on the inside, and they present a flat or crowned contact surface for the belt in all positions. The speed ratio is changed by moving the inner bands rather than the main belts.

Fig. 7 This drive avoids belt wear when the drive has speed cones. However, the creeping action of the belt is not eliminated, and the universal joints present ongoing maintenance problems.

Fig. 8 This drive is a modification of the drive shown in Fig. 7. A roller is substituted for the belt, reducing the overall size of the drive.

Fig. 9 The main component of this drive is a hollow internal cone driven by a conical pulley on the motor shaft. Its speed ratio can be changed by sliding the motor and pulley up or down in the vee slide. When the conical pulley on the motor shaft is moved to the center of the driving cone, the motor and cone run at the same speed. This feature makes the system attractive in applications where heavy torque requirements are met at the motor's rated speed and it is useful to have lower speeds for light preliminary operations.

Fig. 10 In this transmission, the driving pulley cone and driven cone are mounted on the same shaft with their small diameters directed toward each other. The driving pulley (at right) is keyed to the common shaft, and the driven cone (at left) is mounted on a sleeve. Power is transmitted by a series of rocking shafts with rollers mounted on their ends. The shafts are free to slide while they are pivoted within sleeves within a disk that is perpendicular to the driven-cone mounting sleeve. The speed ratio can be changed by pivoting the rocking shafts and allowing them to slide across the conical surfaces of the driving pulley and driven cone.

Fig. 11 This transmission has curved surfaces on its planetary rollers and races. The cone shaped inner races revolve with the drive shaft, but are free to slide longitudinally on sliding keys. Strong compression springs keep the races in firm contact with the three planetary rollers.

Fig. 12 This Graham transmission has only five major parts. Three tapered rollers are carried by a spider fastened to the drive shaft. Each roller has a pinion that meshes with a ring gear connected to the output shaft. The speed of the rollers as well as the speed of the output shaft is varied by moving the contact ring longitudinally. This movement changes the ratio of the contacting diameters.

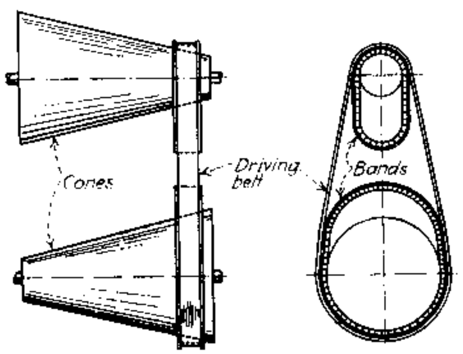


Fig. 6

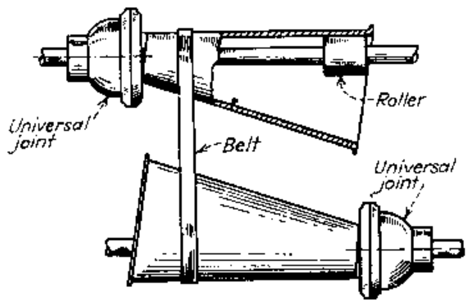


Fig. 7

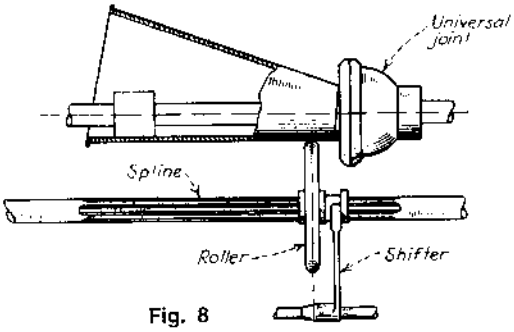


Fig. 8

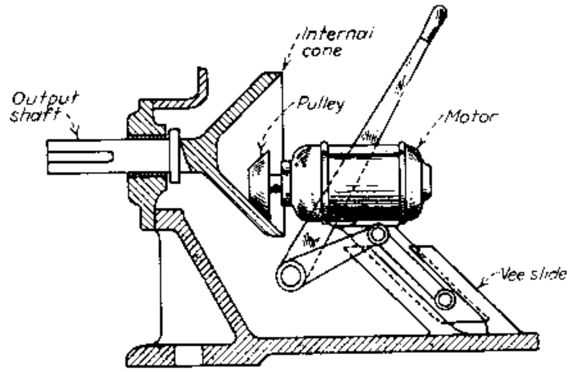


Fig. 9

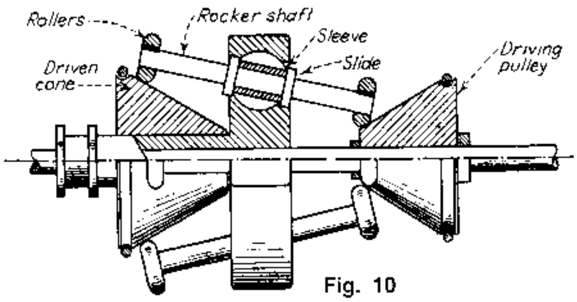
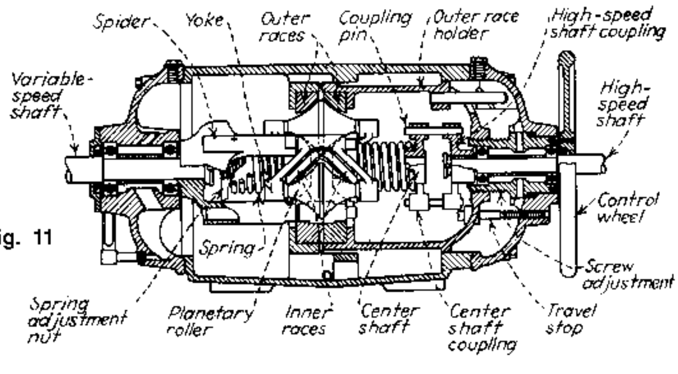


Fig. 10

Fig. 11



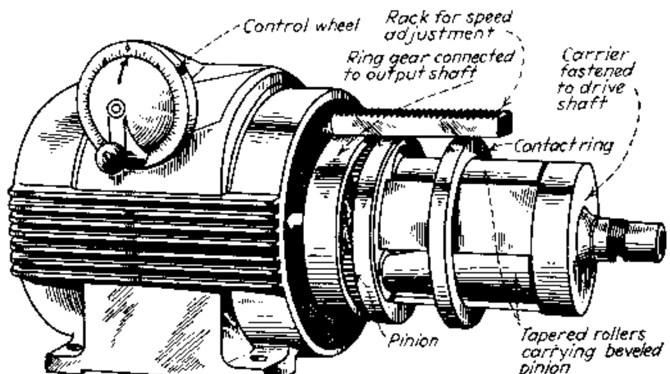


Fig. 12

VARIABLE-SPEED FRICTION DRIVES

These drives can be used to transmit both high torque, as on industrial machines, and low torque, as in laboratory instruments. All perform best if they are used to reduce and not to increase speed. All friction drives have a certain amount of slip due to imperfect rolling of the friction members, but with effective design this slip can be held constant, resulting in constant speed of the driven member. Compensation for variations in load can be achieved by placing inertia masses on the driven end. Springs or similar elastic members can be used to keep the friction parts in constant contact and exert the force necessary to create the friction. In some cases, gravity will take the place of such members. Custom-made friction materials are generally recommended, but neoprene or rubber can be satisfactory. Normally only one of the friction members is made or lined with this material, while the other is metal.

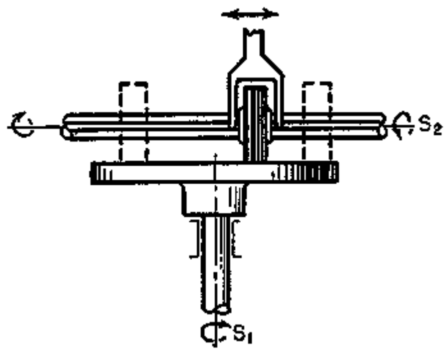


Fig. 1 A disk and roller drive. The roller is moved radially on the disk. Its speed ratio depends upon the operating diameter of the disk. The direction of relative rotation of the shafts is reversed when the roller is moved past the center of the disk, as indicated by dotted lines.

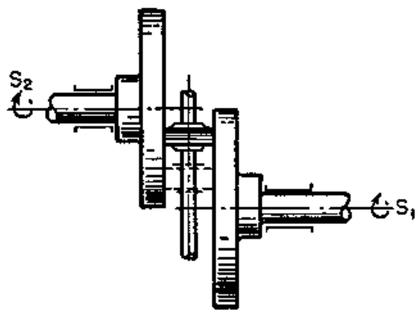


Fig. 2 Two disks have a free-spinning, movable roller between them. This drive can change speed rapidly because the operating diameters of the disks change in an inverse ratio.

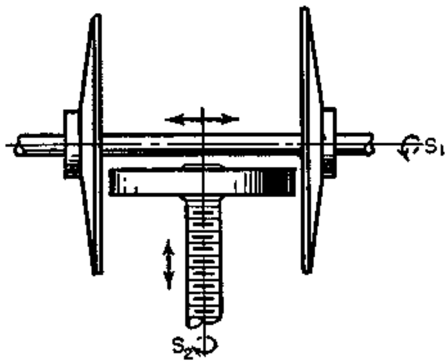


Fig. 3 Two disks are mounted on the same shaft and a roller is mounted on a threaded spindle. Roller contact can be changed from one disk to the other to change the direction of rotation. Rotation can be accelerated or decelerated by moving the screw.

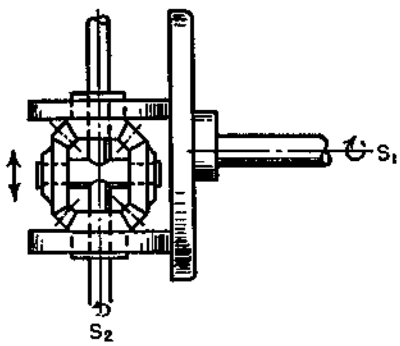


Fig. 4 A disk contacts two differential rollers. The rollers and their bevel gears are free to rotate on shaft S_2. The other two bevel gears are free to rotate on pins connected by S_2. This drive is suitable for the accurate adjustment of speed. S_2 will have the differential speed of the two rollers. The differential assembly is movable across the face of the disk.

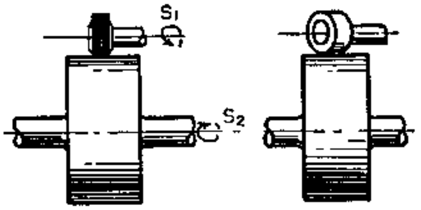


Fig. 5 This drive is a drum and roller. A change of speed is performed by skewing the roller relative to the drum.

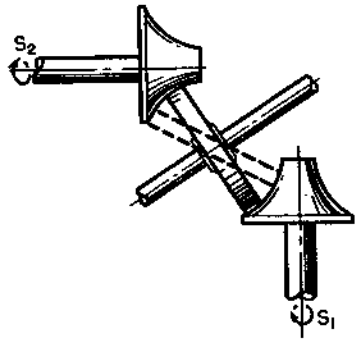


Fig. 6 This drive consists of two spherical cones on intersecting shafts and a free roller.

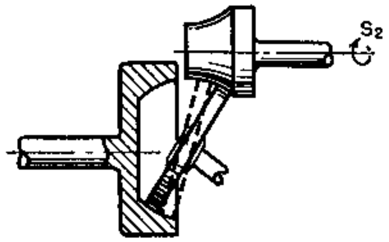


Fig. 7 This drive consists of a spherical cone and groove with a roller. It can be used for small adjustments in speed.

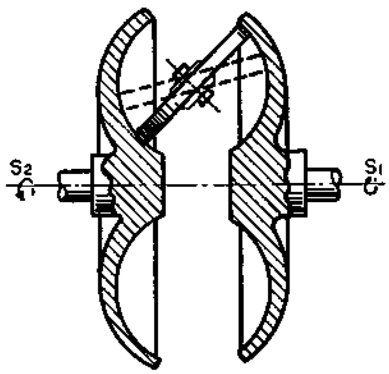


Fig. 8 This drive consists of two disks with torus contours and a free rotating roller.

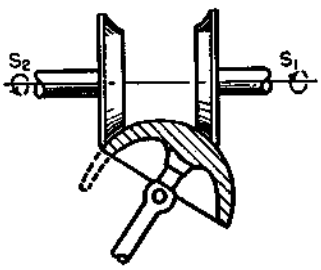


Fig. 9 This drive consists of two disks with a spherical free rotating roller.

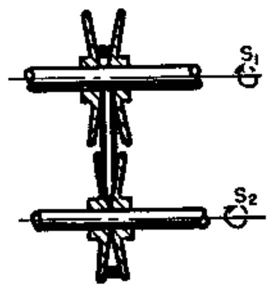


Fig. 10 This drive has split pulleys for V belts. The effective diameter of the belt grip can be adjusted by controlling the distance between the two parts of the pulley.

VARIABLE-SPEED DRIVES AND TRANSMISSIONS

These ratchet and inertial drives provide variable-speed driving of heavy and light loads.

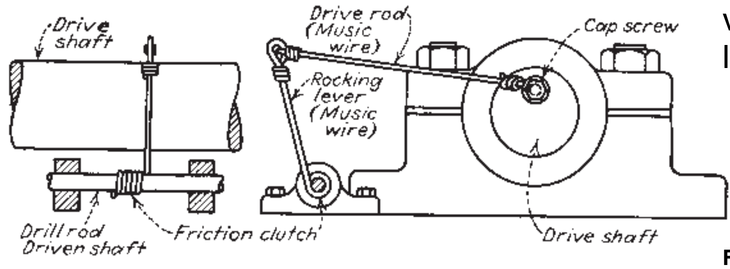


Fig. 1

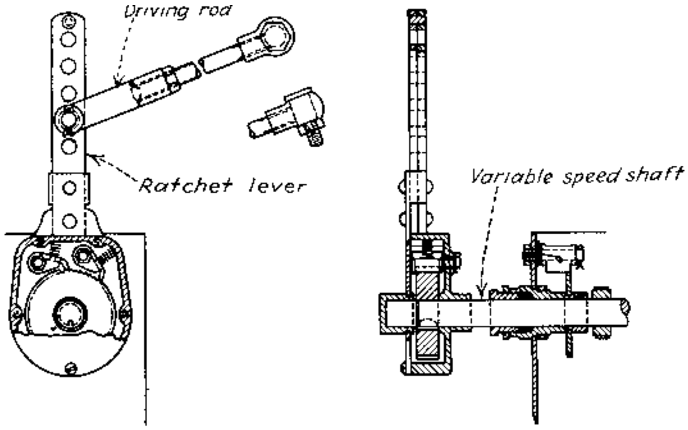


Fig. 2

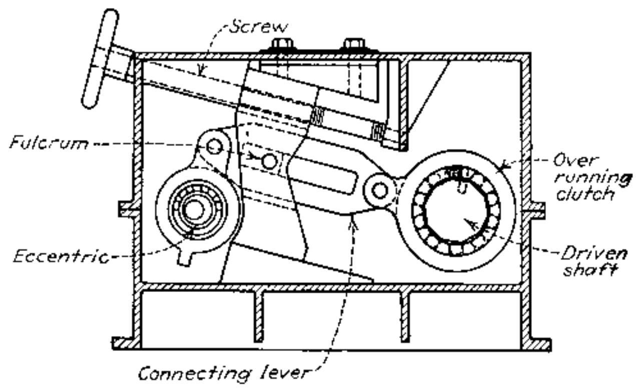


Fig. 3

Fig. 1 This variable-speed drive is suitable only for very light duty in a laboratory or for experimental work. The drive rod receives motion from the drive shaft and it rocks the lever. A friction clutch is formed in a lathe by winding wire around a drill rod whose diameter is slightly larger than the diameter of the driven shaft. The speed ratio can be changed when the drive is stationary by varying the length of the rods or the throw of the eccentric.

Fig. 2 This Torrington lubricator drive illustrates the general principles of ratchet transmission drives. Reciprocating motion from a convenient sliding part, or from an eccentric, rocks the ratchet lever. That motion gives the variable-speed shaft an intermittent unidirectional motion. The speed ratio can be changed only when the unit is stationary. The throw of the ratchet lever can be varied by placing a fork of the driving rod in a different hole.

Fig. 3 This drive is an extension of the principle illustrated in Fig. 2. The Lenney transmission replaces the ratchet with an over-running clutch. The speed of the driven shaft can be varied while the unit is in motion by changing the position of the connecting-lever fulcrum.

Fig. 4 This transmission is based on the principle shown in Fig. 3. The crank disk imparts motion to the connecting rod. The crosshead moves toggle levers which, in turn, give unidirectional motion to the clutch wheel when the friction pawls engage in a groove. The speed ratio is changed by varying the throw of the crank with the aid of a rack and pinion.

Fig. 5 This is a variable speed transmission for gasoline-powered railroad section cars. The connecting rod from the crank, mounted on a constant-speed shaft, rocks the oscillating lever and actuates the over-running clutch. This gives intermittent but unidirectional motion to the variable-speed shaft. The toggle link keeps the oscillating lever within the prescribed path. The speed ratio is changed by swinging the bell crank toward the position shown in the dotted lines, around the pivot attached to the frame. This varies the movement of the over-running clutch. Several units must be out-of-phase with each other for continuous shaft motion.

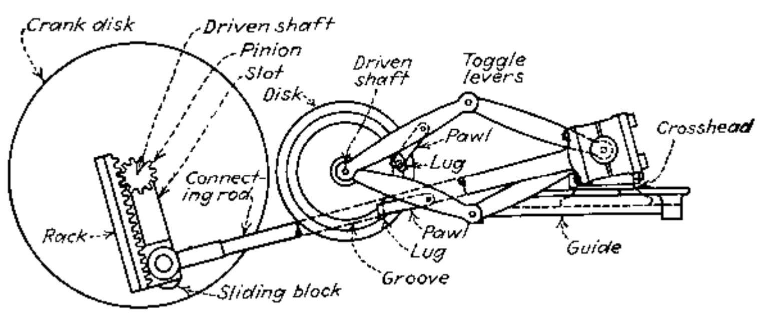


Fig. 4

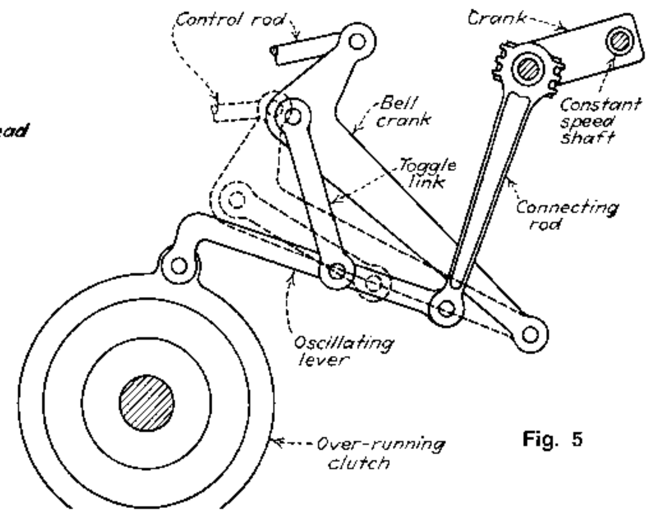


Fig. 5

Fig. 6 This Thomas transmission is an integral part of an automobile engine whose piston motion is transferred by a conventional connecting rod to the long arm of the bellcrank lever oscillating about a fixed fulcrum. A horizontal connecting rod, which rotates the crankshaft, is attached to the short arm of the bellcrank. Crankshaft motion is steadily and continuously maintained by a flywheel. However, no power other than that required to drive auxiliaries is taken from this shaft. The main power output is transferred from the bellcrank lever to the over-running clutch by a third connecting rod. The speed ratio is changed by sliding the top end of the third connecting rod within the bellcrank lever with a crosshead and guide mechanism. The highest ratio is obtained when the crosshead is farthest from the fulcrum, and movement of the crosshead toward the fulcrum reduces the ratio until a "neutral" position is reached. That occurs when the center line of the connecting rod coincides with the fulcrum.

Fig. 7 This Constantino torque converter is another automotive transmission system designed and built as part of the engine. It features an inherently automatic change of speed ratio that tracks the speed and load on the engine. The constant-speed shaft rotates a crank which, in turn, drives two oscillating levers with inertia weights at their ends. The other ends are attached by links to the rocking levers. These rocking levers include over-running clutches. At low engine speeds, the inertia weights oscillate through a wide angle. As a result, the reaction of the inertia force on the other end of the lever is very slight, and the link imparts no motion to the rocker lever. Engine speed increases cause the inertia weight reaction to increase. This rocks the small end of the oscillating lever as the crank rotates. The resulting motion rocks the rocking lever through the link, and the variable shaft is driven in one direction.

Fig. 8 This transmission has a differential gear with an adjustable escapement. This arrangement bypasses a variable portion of the drive-shaft revolutions. A constant-speed shaft rotates a freely mounted worm wheel that carries two pinion shafts. The firmly fixed pinions on these shafts, in turn, rotate the sun gear that meshes with other planetary gears. This mechanism rotates the small worm gear attached to the variable-speed output shaft.

Fig. 9 This Morse transmission has an eccentric cam integral with its constant-speed input shaft. It rocks three ratchet clutches through a series of linkage systems containing three rollers that run in a circular groove cut in the cam face. Unidirectional motion is transmitted to the output shaft from the clutches by planetary gearing. The speed ratio is changed by rotating an anchor ring containing a fulcrum of links, thus varying the stroke of the levers.

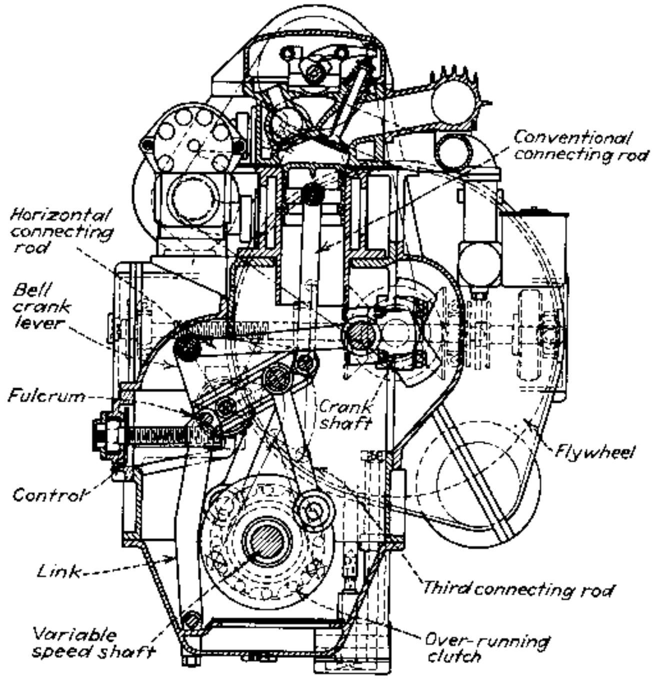


Fig. 6

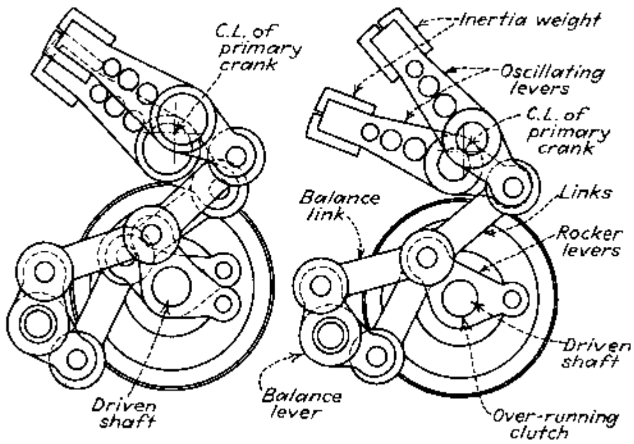


Fig. 7

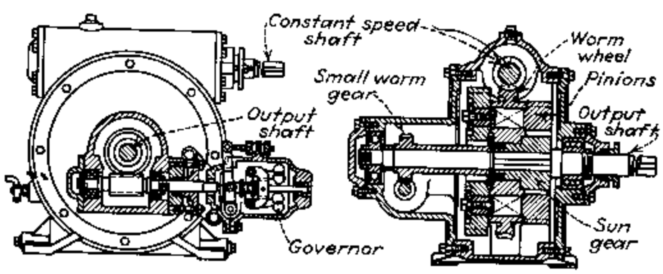


Fig. 8

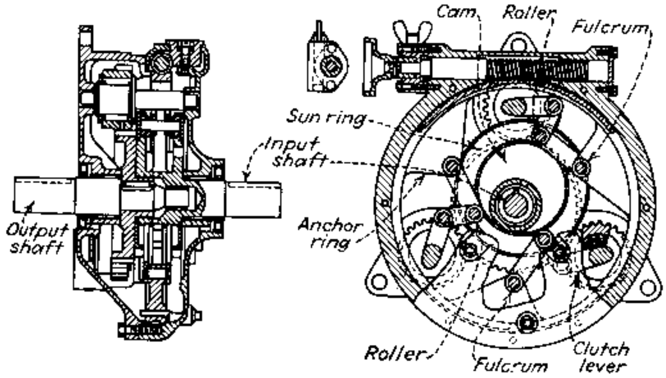


Fig. 9

PRECISION BALL BEARINGS REPLACE GEARS IN TINY SPEED REDUCERS

Miniature bearings can take over the role of gears in speed reducers where a very high speed change, either a speed reduction or speed increase, is desired in a limited space. Ball bearing reducers such as those made by MPB Corp., Keene, N.H. (see drawings), provide speed ratios as high as 300-to-1 in a space ½-in. dia. by ½-in. long.

And at the same time the bearings run quietly, with both the input and output shafts rotating on the same line.

The interest in ball bearing reducers stems from the pressure on mechanical engineers to make their designs more compact to match the miniaturization gains in the electronic fields.

The advantages of the bearing-reducer concept lie in its simplicity. A conventional precision ball bearing functions as an epicyclic or planetary gearing device. The bearing inner ring, outer ring, and ball complement become, in a sense, the sun gear, internal gear, and planet pinions.

Power transmission functions occur with either a single bearing or with two or more in tandem. Contact friction or traction between the bearing components transmits the torque. To prevent slippage, the bearings are preloaded just the right amount to achieve balance between transmitted torque and operating life.

Input and output functions always rotate in the same direction, irrespective of the number of bearings, and different results can be achieved by slight alterations in bearing characteristics. All these factors lead to specific advantages:

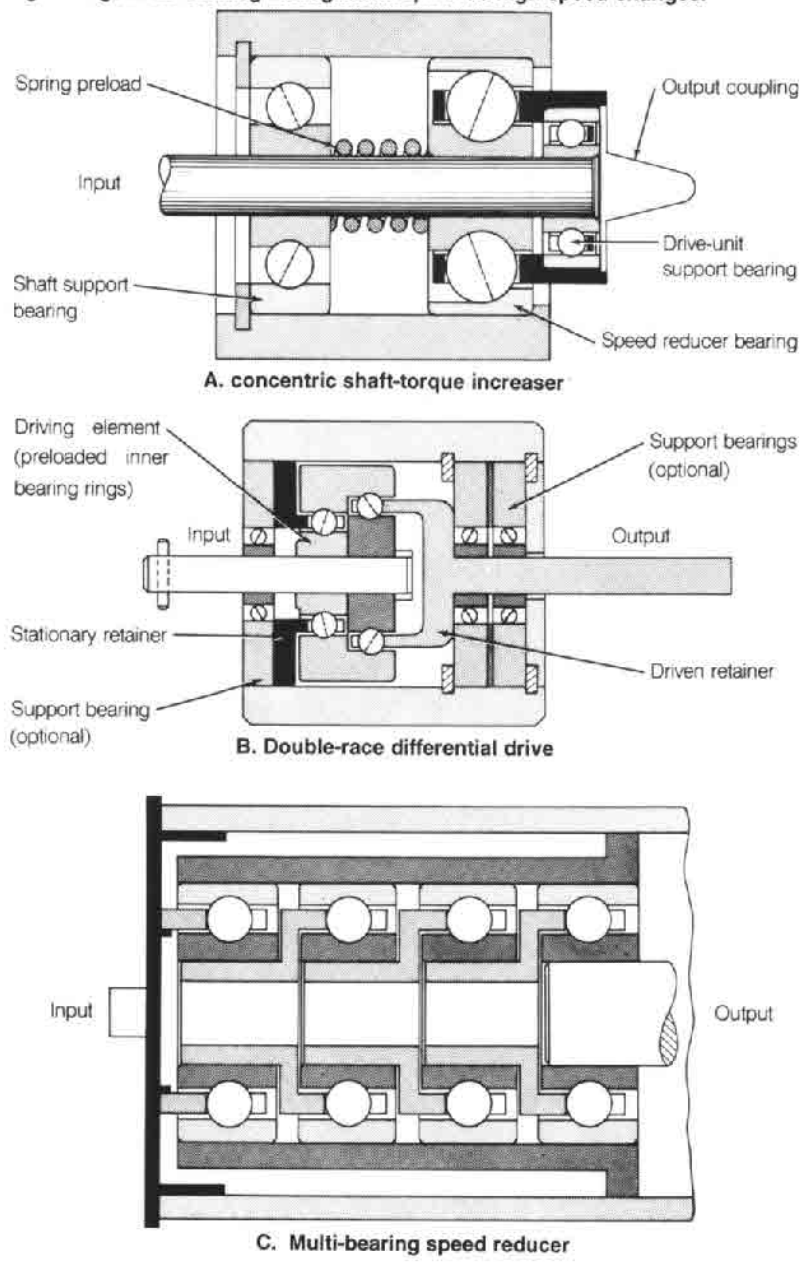
- **Space saving.** The outside diameter, bore, and width of the bearings set the envelope dimensions of the unit. The housing need by only large enough to hold the bearings. In most cases the speed-reducer bearings can be build into the total system, conserving more space.
- **Quiet operation.** The traction drive is between nearly perfect concentric circles with component roundness and concentricity, controlled to precise tolerances of 0.00005 in. or better. Moreover, operation is not independent in any way on conventional gear teeth. Thus quiet operation is inherent.
- **High speed ratios.** As a result of design ingenuity and use of special bearing races, virtually any speed-reducing or speed-increasing ratio can be achieved. MPB studies showed that speed ratios of 100,000-to-1 are theoretically possible with only two bearings installed.
- **Low backlash.** Backlash is restricted mainly to the clearance between backs and ball retainer. Because the balls are preloaded, backlash is almost completely eliminated.

Ball bearing reducers are limited as to the amount of torque that can be transmitted.

The three MPB units (Fig. 1) illustrate the variety of designs possible:

- **Torque increaser** (Fig. 1A). This simple torque increaser boosts the output torque in an air-driven dental handpiece, provide a 2 ½-to-1 speed reduction. The speed reduces as the bearing's outer ring is kept from rotating while the inner ring is driven; the output is taken from a coupling that is integral with the ball retainer.

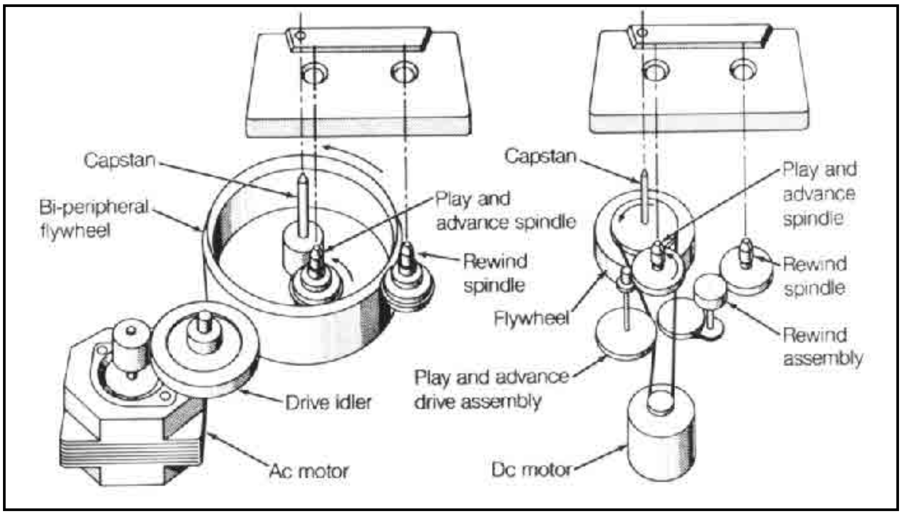
Fig. 1. Ingenious bearing arrangements provide high speed-changes.



The exact speed ratio depends on the bearing's pitch diameter, ball diameter, or contact angle. By stiffening the spring, the amount of torque transmitted increases, thereby increasing the force across the ball's normal line of contact.

- **Differential drive** (Fig. 1B). This experimental reduction drive uses the inner rings of a preloaded pair of bearings as the driving element. The ball retainer of one bearing is the stationary element, and the opposing ball retainer is the driven element. The common outer ring is free to rotate. Keeping the differences between the two bearings small permits extremely high speed reductions. A typical test model has a speed reduction ratio of 200-to-1 and transmits 1 in.-oz of torque.
- **Multi-bearing reducer** (Fig. 1C). This stack of four precision bearings achieves a 26-to-1 speed reduction to drive the recording tape of a dictating machine. Both the drive motor and reduction unit are housed completely within the drive capstan. The balls are preloaded by assembling each bearing with a controlled interference or negative radial play.

MULTIFUNCTION FLYWHEEL SMOOTHES FRICTION IN TAPE CASSETTE DRIVE



A lightweight flywheel in the tape recorder (left) has a higher inertia than in a conventional model (right). Its dual peripheries serve as drives for friction rollers.

A cup-shaped flywheel performs a dual function in tape recorders by acting as a central drive for friction rollers as well as a high inertia wheel. The flywheel is the heart of a drive train in Wollensak cassette audio-visual tape recorders.

The models included record-playback and playback-only portables and decks.

Fixed parameters. The Philips cassette concept has several fixed parameters—the size of the tape cartridge (4 × 2½ in.), the distance between the hubs onto which the tape is wound, and the operating speed. The speed, standardized at 1⅞ ips, made it possible to enclose enough tape in the container for lengthy recordings. Cassettes are available commercially for recording on one side for 30, 35, or 60 min.

The recorders included a motor comparable in size and power to those used in standard reel-to-reel recorders, and a large bi-peripheral flywheel and sturdy capstan that reduces wow and flutter and drives the tape. A patent application was filed for the flywheel design.

The motor drives the flywheel and capstan assemblies. The flywheel moderates or overcomes variations in speed that cause wow and flutter. The accuracy of the tape drive is directly related to the inertia of the flywheel and the accuracy of the flywheel and capstan. The greater the inertia the more uniform is the tape drive, and the less pronounced is the wow and flutter.

The flywheel is nearly twice as large as the flywheel of most portable cassette recorders, which average less than 2 in. dia. Also, a drive idler is used on the Wollensak models while thin rubber bands and pulleys are employed in conventional portable recorders.

Take-up and rewind. In the new tape drive system, the flywheel drives the take-up and rewind spindle. In play or fast-advance mode, the take-up spindle makes contact with the inner surface of the counterclockwise moving flywheel, moving the spindle counterclockwise and winding the tape onto the hub. In the rewind mode, the rewind spindle is brought into contact with the outer periphery of the flywheel, driving it clockwise and winding the tape onto the hub.

According to Wollensak engineers, the larger AC motor had a service life five times that of a DC motor.

The basic performance for all of the models is identical: frequency response is of 50 to 8000 Hz; wow and flutter are less than 0.25%; signal-to-noise ratio is more than 46 db; and each has a 10-watt amplifier.

All the models also have identical operating controls. One simple lever controls fast forward or reverse tape travel. A three-digit, pushbutton-resettable counter permits the user to locate specific portions of recorded programs rapidly.

CONTROLLED DIFFERENTIAL DRIVES

By coupling a differential gear assembly to a variable speed drive, a drive's horsepower capacity can be increased at the expense of its speed range. Alternatively, the speed range can be increased at the expense of the horsepower range. Many combinations of these variables are possible. The features of the differential depend on the manufacturer. Some systems have bevel gears, others have planetary gears. Both single and double differentials are employed. Variable-speed drives with differential gears are available with ratings up to 30 hp.

Horsepower-increasing differential (Fig. 1). The differential is coupled so that the output of the motor is fed into one side and the output of the speed variator is fed into the other side. An additional gear pair is employed as shown in Fig. 1.

Output speed

$$n_4 = \tfrac{1}{2}\left(n_1 + \frac{n_2}{R}\right)$$

Output torque

$$T_4 = 2T_3 = 2RT_2$$

Output hp

$$\text{hp} = \left(\frac{Rn_1 + n_2}{63{,}025}\right)T_2$$

hp increase

$$\Delta\text{hp} = \left(\frac{Rn_1}{63{,}025}\right)T_2$$

Speed variation

$$n_{4\,\max} - n_{4\,\min} = \frac{1}{2R}(n_{2\,\max} - n_{2\,\min})$$

Speed range increase differential (Fig. 2). This arrangement achieves a wide range of speed with the low limit at zero or in the reverse direction.

Fig. 1

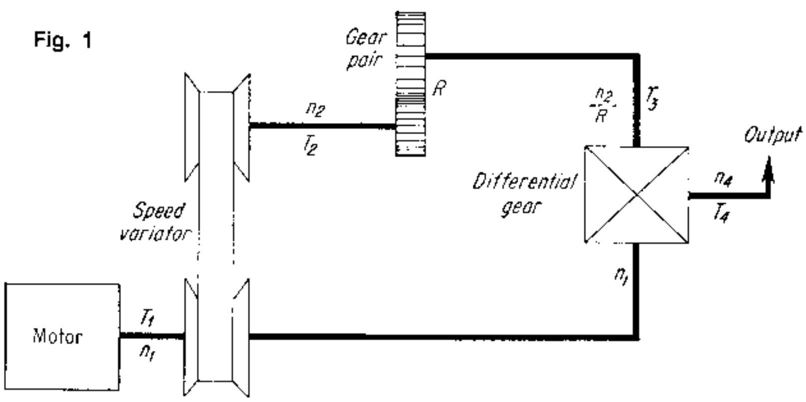


Fig. 2

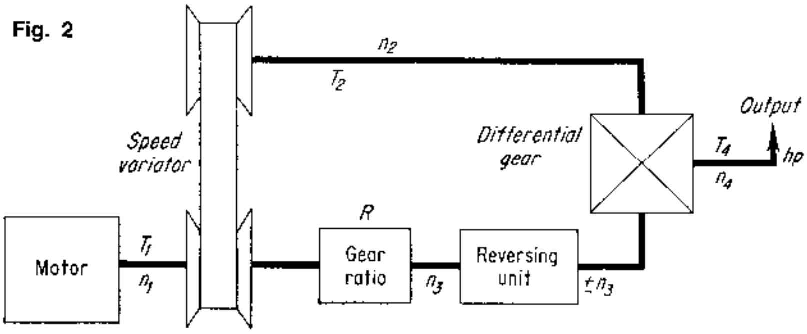


Fig. 3

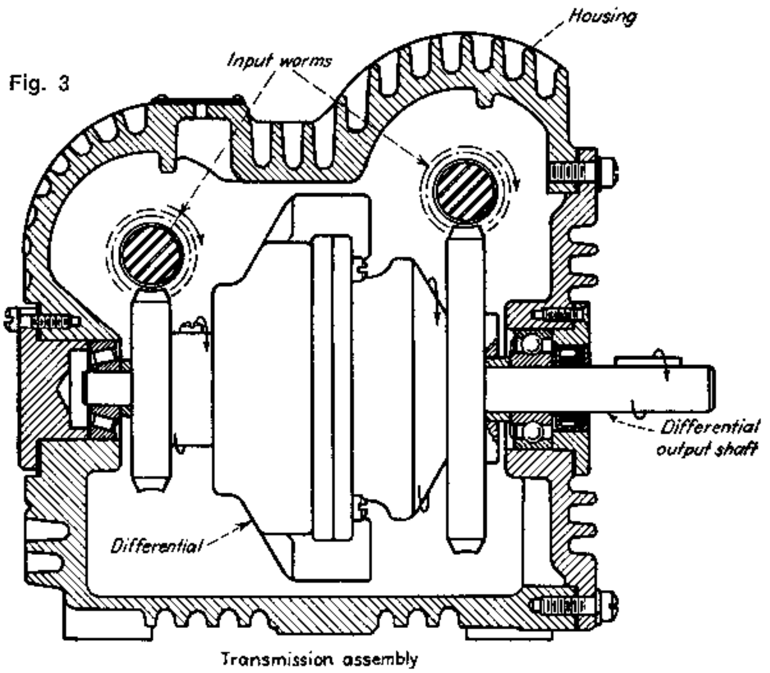


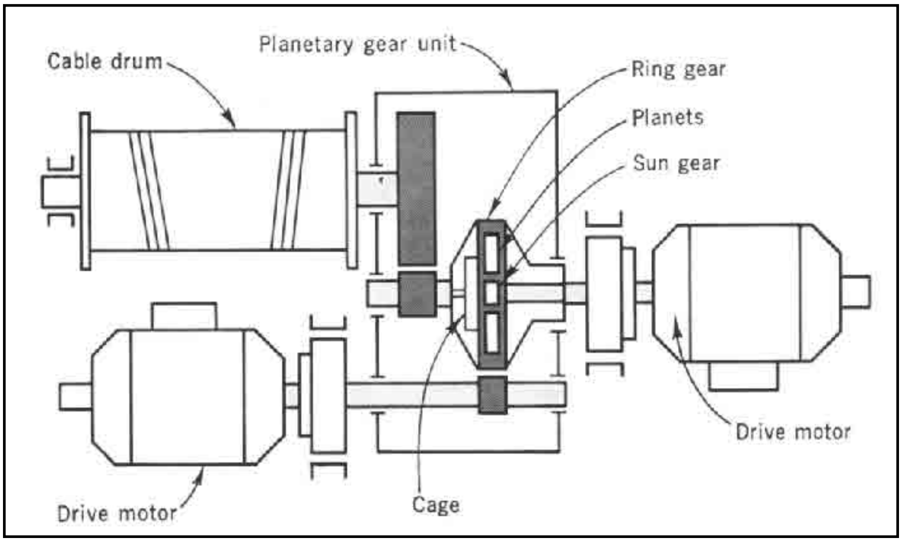
Fig. 3 A variable-speed transmission consists of two sets of worm gears feeding a differential mechanism. The output shaft speed depends on the difference in rpm between the two input worms. When the worm speeds are equal, output is zero. Each worm shaft carries a cone-shaped pulley. These pulley are mounted so that their tapers are in opposite directions. Shifting the position of the drive belt on these pulleys has a compound effect on their output speed.

TWIN-MOTOR PLANETARY GEARS PROVIDE SAFETY PLUS DUAL-SPEED

Many operators and owners of hoists and cranes fear the possible catastrophic damage that can occur if the driving motor of a unit should fail for any reason. One solution to this problem is to feed the power of two motors of equal rating into a planetary gear drive.

Power supply. Each of the motors is selected to supply half the required output power to the hoisting gear (see diagram). One motor drives the ring gear, which has both external and internal teeth. The second motor drives the sun gear directly.

Both the ring gear and sun gear rotate in the same direction. If both gears rotate at the same speed, the planetary cage, which is coupled to the output, will also revolve at the same speed (and in the same direction). It is as if the entire inner works of the planetary were fused together. There would be no relative motion. Then, if one motor fails, the cage will revolve at half its original speed, and the other motor can still lift with undiminished capacity. The same principle holds true when the ring gear rotates more slowly than the sun gear.



Power flow from two motors combine in a planetary that drives the cable drum.

No need to shift gears. Another advantage is that two working speeds are available as a result of a simple switching arrangement. This makes is unnecessary to shift gears to obtain either speed.

The diagram shows an installation for a steel mill crane.

HARMONIC-DRIVE SPEED REDUCERS

The harmonic-drive speed reducer was invented in the 1950s at the Harmonic Drive Division of the United Shoe Machinery Corporation, Beverly, Massachusetts. These drives have been specified in many high-performance motion-control applications. Although the Harmonic Drive Division no longer exists, the manufacturing rights to the drive have been sold to several Japanese manufacturers, so they are still made and sold. Most recently, the drives have been installed in industrial robots, semiconductor manufacturing equipment, and motion controllers in military and aerospace equipment.

The history of speed-reducing drives dates back more than 2000 years. The first record of reducing gears appeared in the writings of the Roman engineer Vitruvius in the first century B.C. He described wooden-tooth gears that coupled the power of water wheel to millstones for grinding corn. Those gears offered about a 5 to 1 reduction. In about 300 B.C., Aristotle, the Greek philosopher and mathematician, wrote about toothed gears made from bronze.

In 1556, the Saxon physician, Agricola, described geared, horse-drawn windlasses for hauling heavy loads out of mines in Bohemia. Heavy-duty cast-iron gear wheels were first introduced in the mid-eighteenth century, but before that time gears made from brass and other metals were included in small machines, clocks, and military equipment.

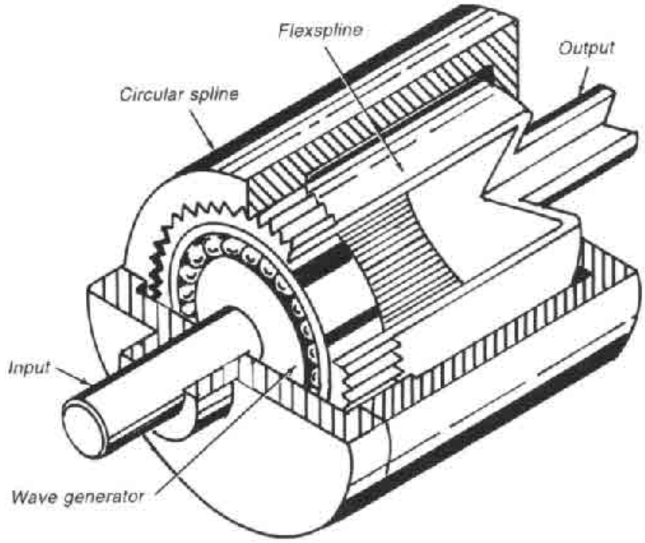


Fig. 1 Exploded view of a typical harmonic drive showing its principal parts. The flexspline has a smaller outside diameter than the inside diameter of the circular spline, so the elliptical wave generator distorts the flexspline so that its teeth, 180º apart, mesh.

The harmonic drive is based on a principle called *strain-wave gearing,* a name derived from the operation of its primary torque-transmitting element, the flexspline. Figure 1 shows the three basic elements of the harmonic drive: the rigid circular spline, the fliexible flexspline, and the ellipse-shaped wave generator.

The *circular spline* is a nonrotating, thick-walled, solid ring with internal teeth. By contrast, a *flexspline* is a thin-walled, flexible metal cup with external teeth. Smaller in external diameter than the inside diameter of the circular spline, the flexspline must be deformed by the wave generator if its external teeth are to engage the internal teeth of the circular spline.

When the *elliptical cam wave generator* is inserted into the bore of the flexspline, it is formed into an elliptical shape. Because the major axis of the wave generator is nearly equal to the inside diameter of the circular spline, external teeth of the flexspline that are 180° apart will engage the internal circular-spline teeth.

Modern wave generators are enclosed in a ball-bearing assembly that functions as the rotating input element. When the wave generator transfers its elliptical shape to the flexspline and the external circular spline teeth have engaged the internal circular spline teeth at two opposing locations, a positive gear mesh occurs at those engagement points. The shaft attached to the flexspline is the rotating output element.

Figure 2 is a schematic presentation of harmonic gearing in a section view. The flexspline typically has two fewer external teeth than the number of internal teeth on the circular spline. The keyway of the input shaft is at its zero-degree or 12 o'clock position. The small circles around the shaft are the ball bearings of the wave generator.

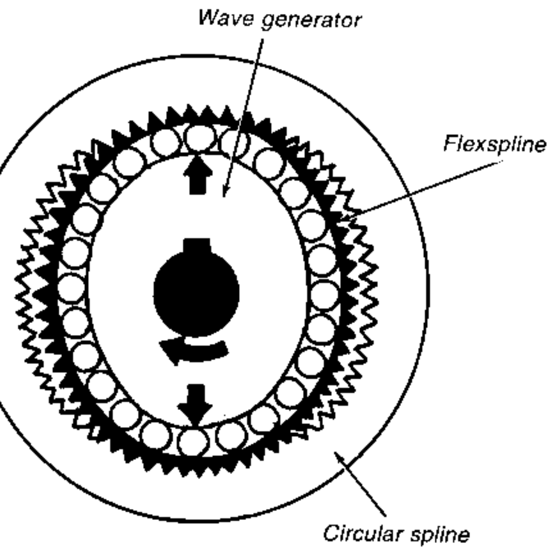


Fig. 2 Schematic of a typical harmonic drive showing the mechanical relationship between the two splines and the wave generator.

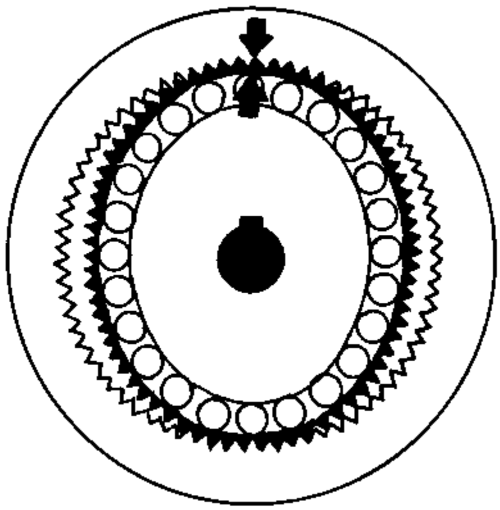


Fig. 3 Three positions of the wave generator: (*A*) the 12 o'clock or zero degree position; (*B*) the 3 o'clock or 90° position; and (*C*) the 360° position showing a two-tooth displacement.

Figure 3 is a schematic view of a harmonic drive in three operating positions. In position 3(*A*), the inside and outside arrows are aligned. The inside arrow indicates that the wave generator is in its 12 o'clock position with respect to the circular spline, prior to its clockwise rotation.

Because of the elliptical shape of the wave generator, full tooth engagement occurs only at the two areas directly in line with the major axis of the ellipse (the vertical axis of the diagram). The teeth in line with the minor axis are completely disengaged.

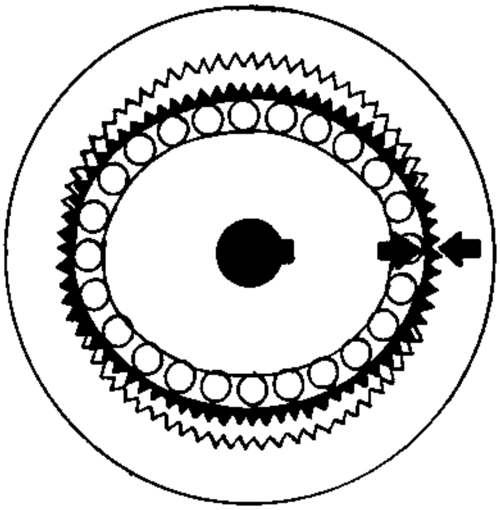
As the wave generator rotates 90° clockwise, as shown in Fig. 3(*B*), the inside arrow is still pointing at the same flexspline tooth, which has begun its counterclockwise rotation. Without full tooth disengagement at the areas of the minor axis, this rotation would not be possible.

At the position shown in Fig. 3(*C*), the wave generator has made one complete revolution and is back at its 12 o'clock position. The inside arrow of the flexspline indicates a two-tooth per revolution displacement counterclockwise. From this one revolution motion the reduction ratio equation can be written as:

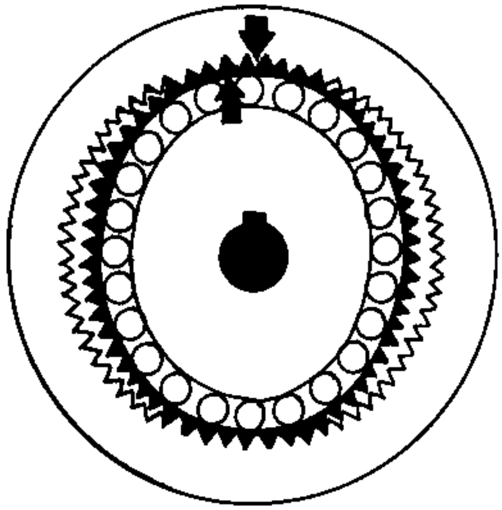
$$GR = \frac{FS}{CS - FS}$$

where:
GR = gear ratio
FS = number of teeth on the flexspline
CS = number of teeth on the circular spline
Example:
FS = 200 teeth
CS = 202 teeth

$$GR = \frac{200}{202 - 200} = 100 : 1 \text{ reduction}$$



B



C

As the wave generator rotates and flexes the thin-walled spline, the teeth move in and out of engagement in a rotating wave motion. As might be expected, any mechanical component that is flexed, such as the flexspline, is subject to stress and strain.

Advantages and Disadvantages

The harmonic drive was accepted as a high-performance speed reducer because of its ability to position moving elements precisely. Moreover, there is no backlash in a harmonic drive reducer. Therefore, when positioning inertial loads, repeatability and resolution are excellent (one arc-minute or less).

Because the harmonic drive has a concentric shaft arrangement, the input and output shafts have the same centerline. This geometry contributes to its compact form factor. The ability of the drive to provide high reduction ratios in a single pass with high torque capacity recommends it for many machine designs. The benefits of high mechanical efficiency are high torque capacity per pound and unit of volume, both attractive performance features.

One disadvantage of the harmonic drive reducer has been its wind-up or torsional spring rate. The design of the drive's tooth form necessary for the proper meshing of the flexspline and the circular spline permits only one tooth to be completely engaged at each end of the major elliptical axis of the generator. This design condition is met only when there is no torsional load. However, as torsional load increases, the teeth bend slightly and the flexspline also distorts slightly, permitting adjacent teeth to engage.

Paradoxically, what could be a disadvantage is turned into an advantage because more teeth share the load. Consequently, with many more teeth engaged, torque capacity is higher, and there is still no backlash. However, this bending and flexing causes torsional wind-up, the major contributor to positional error in harmonic-drive reducers.

At least one manufacturer claims to have overcome this problem with redesigned gear teeth. In a new design, one company replaced the original involute teeth on the flexspline and circular spline with noninvolute teeth. The new design is said to reduce stress concentration, double the fatigue limit, and increase the permissible torque rating.

The new tooth design is a composite of convex and concave arcs that match the loci of engagement points. The new tooth width is less than the width of the tooth space and, as a result of these dimensions and proportions, the root fillet radius is larger.

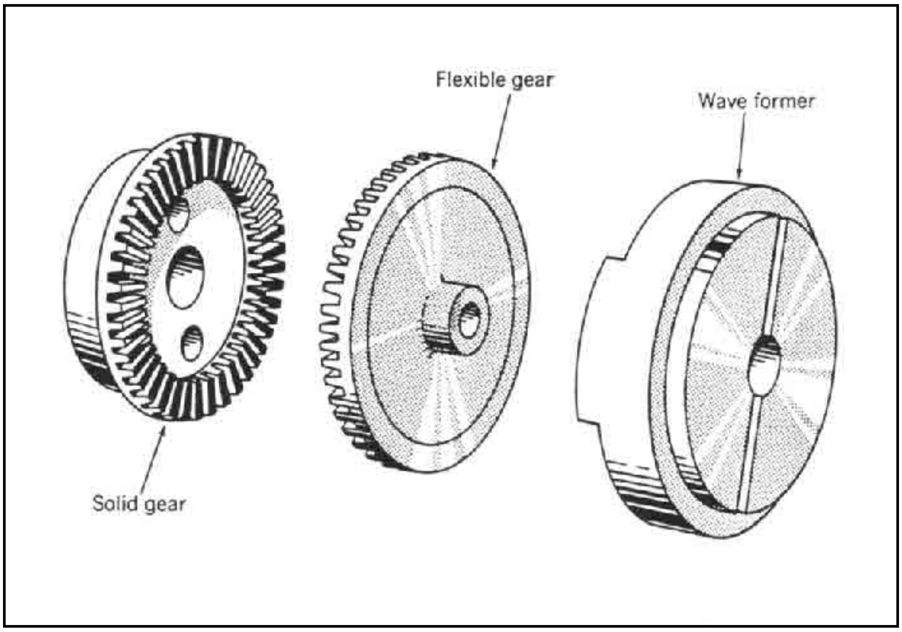
FLEXIBLE FACE-GEARS MAKE EFFICIENT HIGH-REDUCTION DRIVES

A system of flexible face-gearing provides designers with a means for obtaining high-ratio speed reductions in compact trains with concentric input and output shafts.

With this approach, reduction ratios range from 10:1 to 200:1 for single-stage reducers, whereas ratios of millions to one are possible for multi-stage trains. Patents on the flexible face-gear reducers were held by Clarence Slaughter of Grand Rapids, Michigan.

Building blocks. Single-stage gear reducers consist of three basic parts: a flexible face-gear made of plastic or thin metal; a solid, non-flexing face-gear; and a wave former with one or more sliders and rollers to force the flexible gear into mesh with the solid gear at points where the teeth are in phase.

The high-speed input to the system usually drives the wave former. Low-speed output can be derived from either the flexible or the solid face gear; the gear not connected to the output is fixed to the housing.



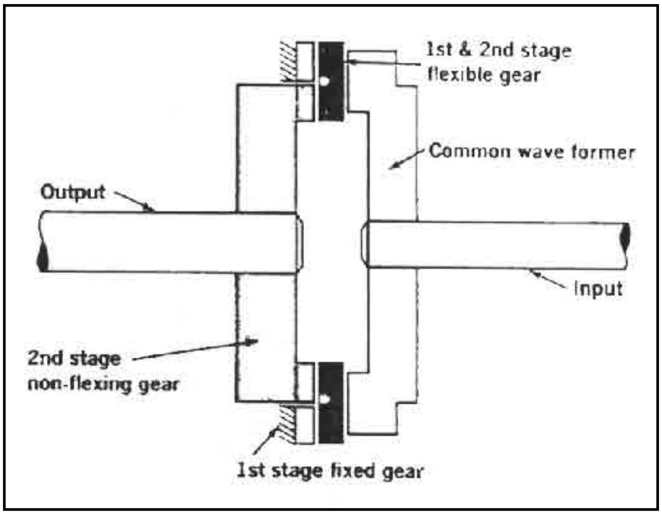
A flexible face-gear is flexed by a rotating wave former into contact with a solid gear at point of mesh. The two gears have slightly different numbers of teeth.

Teeth make the difference. Motion between the two gears depends on a slight difference in their number of teeth (usually one or two teeth). But drives with gears that have up to a difference of 10 teeth have been devised.

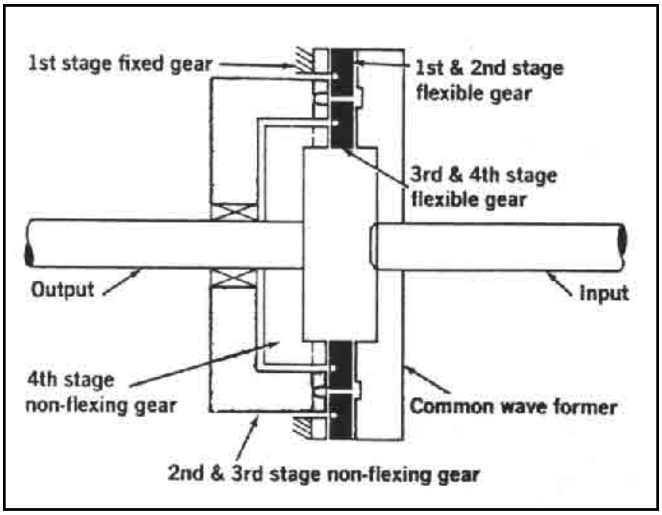
On each revolution of the wave former, there is a relative motion between the two gears that equals the difference in their numbers of teeth. The reduction ratio equals the number of teeth in the output gear divided by the difference in their numbers of teeth.

Two-stage and four-stage gear reducers are made by combining flexible and solid gears with multiple rows of teeth and driving the flexible gears with a common wave former.

Hermetic sealing is accomplished by making the flexible gear serve as a full seal and by taking output rotation from the solid gear.



A two-stage speed reducer is driven by a common-wave former operating against an integral flexible gear for both stages.



A four-stage speed reducer can, theoretically, attain reductions of millions to one. The train is both compact and simple.

COMPACT ROTARY SEQUENCER

Two coaxial rotations, one clockwise and one counterclockwise, are derived from a single clockwise rotation.

A proposed rotary sequencer is assembled from a conventional planetary differential gearset and a latching mechanism. Its single output and two rotary outputs (one clockwise and one counterclockwise) are coaxial, and the output torque is constant over the entire cycle. Housed in a lightweight, compact, cylindrical package, the sequencer requires no bulky ratchets, friction clutches, or cam-and-track followers. Among its possible applications are sequencing in automated production-line equipment, in home appliances, and in vehicles.

The sequencer is shown in Figure 1. A sun gear connects with four planetary gears that engage a ring gear. With the ring gear held stationary, clockwise rotation of the sun gear causes the entire planetary-gear carrier also to rotate clockwise. If the planetary-gear carrier is held fixed, the ring gear will rotate counterclockwise when the sun gear rotates clockwise.

Figure 2 shows the latch. It consists of a hook (the carrier hook) that is rigidly attached to the planetary-gear carrier, a rind that is rigidly attached to the ring gear, and a latch pivot arm with a pair of latch rollers attached to one end. The other end of the pivot arm rotates about a short shaft that extends from the fixed wall of the housing.

The sequencer cycle starts with the ring latch roller resting in a slot in the ring. This locks the ring and causes the planetary-gear carrier to rotate clockwise with the input shaft (Fig. 2a). When the carrier hook has rotated approximately three-quarters of a complete cycle, it begins to engage the planet-carrier latch roller (Fig. 2b), causing the latch pivot arm to rotate and the ring latch roller to slip out of its slot (Fig. 2c). This frees the ring and ring gear for counterclockwise motion, while locking the carrier. After a short interval of concurrent motion, the planetary-gear output shaft ceases its clockwise motion, and the ring-gear output shaft continues its clockwise motion.

When the ring reaches the position in Fig. 2d, the cycle is complete, and the input shaft is stopped. If required, the input can then be rotated counterclockwise, and the sequence will be reversed until the starting position (Fig. 2a) is reached again.

In a modified version of the sequencer, the latch pivot arm is shortened until its length equals the radii of the rollers. This does away with the short overlap of output rotations when both are in motion. For this design, the carrier motion ceases before the ring begins its rotation.

This work was done by Walter T. Appleberry of Rockwell International Corp. for **Johnson Space Center,** *Houston, Texas.*

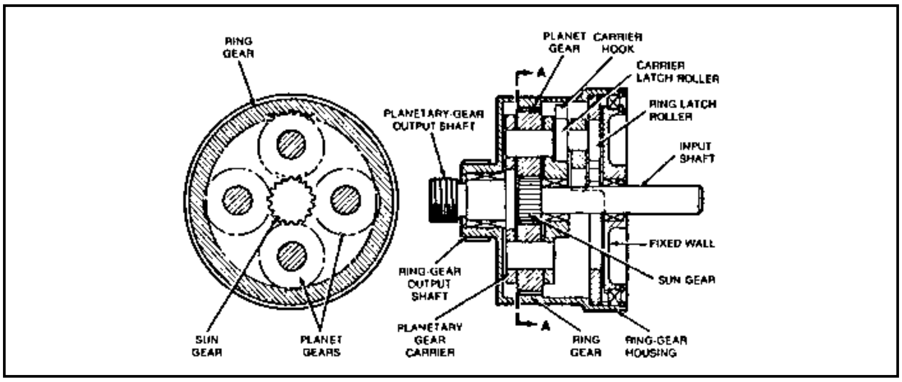


Fig. 1 The Rotary Sequencer has a ring-gear output (in color) that is coaxial with a planetary-gear output (in gray). Clockwise rotation of the input is converted to clockwise rotation of the planetary-gear output followed by counterclockwise rotation of the ring-gear output. The sequence is controlled by the latch action described in Fig. 2.

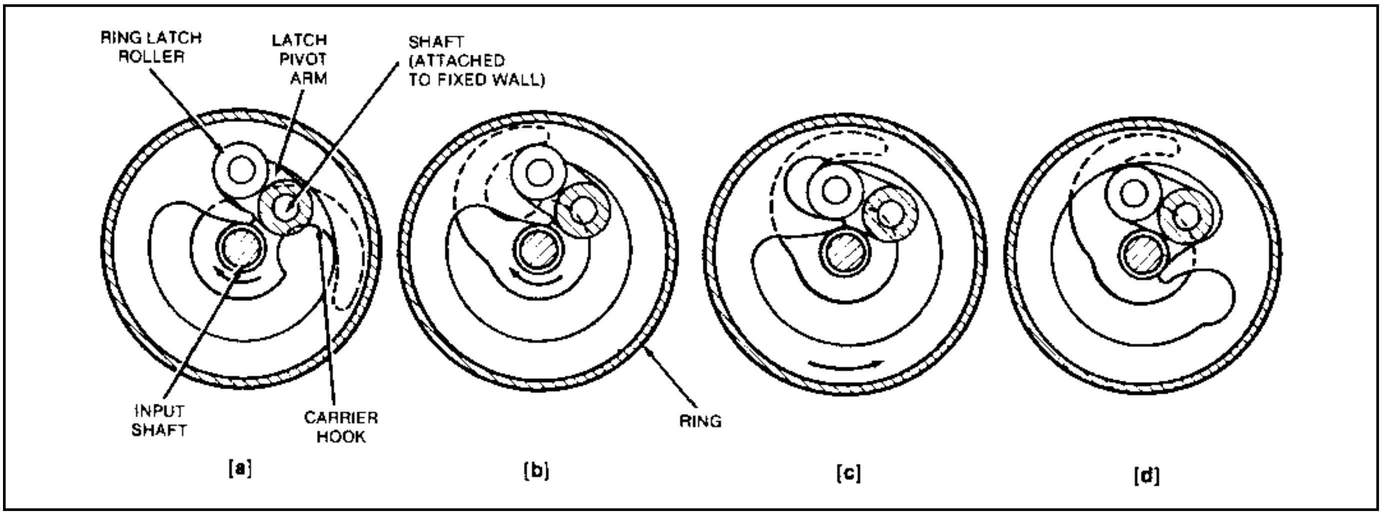
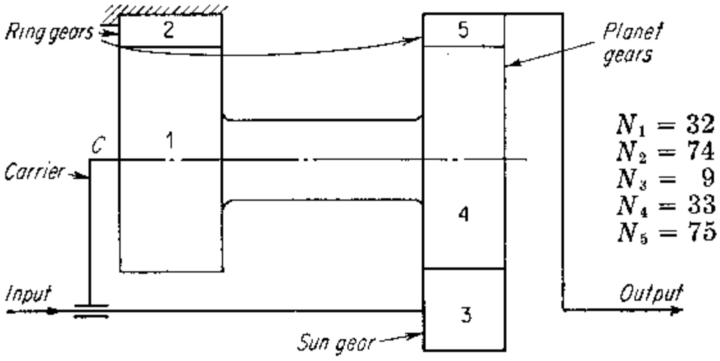


Fig. 2 The Latch Sequence is shown in four steps: (a) The input shaft rotates the carrier clockwise while the ring latch roller holds the ring gear stationary; (b) the carrier hook begins to engage the carrier latch roller; (c) the ring latch roller begins to move out of its slot, and the carrier motion ceases while the ring begins to move; and (d) the sequence has ended with the ring in its final position.

PLANETARY GEAR SYSTEMS

Designers keep finding new and useful planetaries. Forty-eight popular types are given here with their speed-ratio equations.

MISSILE SILO COVER DRIVE



Ring gear 2 fixed; ring gear 5 output

Speed-ratio equation

$$\boxed{R = \frac{1 + \dfrac{N_4 N_2}{N_3 N_1}}{1 - \dfrac{N_4 N_2}{N_5 N_1}}} = \frac{1 + \dfrac{(33)(74)}{(\ 9)(32)}}{1 - \dfrac{(33)(74)}{(75)(32)}} = -541\tfrac{2}{3}$$

Symbols

C = carrier (also called "spider")—a non-gear member of a gear train whose rotation affects gear ratio

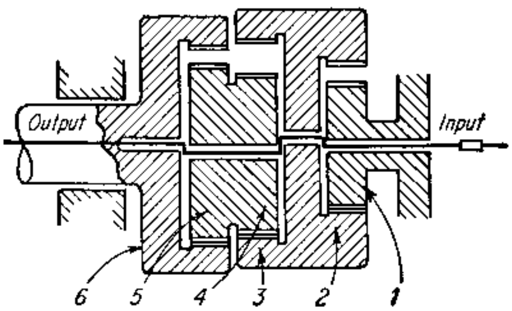
N = number of teeth

R = overall speed reduction ratio

1, 2, 3, etc. = gears in a train (corresponding to labels on schematic diagram)

DOUBLE-ECCENTRIC DRIVE

Input is through double-throw crank (carrier). **Gear 1**

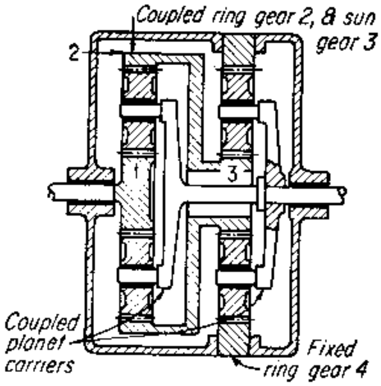


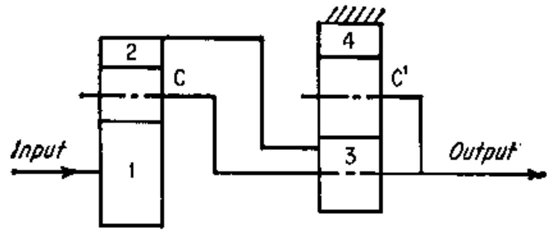
$$R = \frac{1}{1 - \dfrac{N_5 N_3 N_1}{N_6 N_4 N_2}}$$

When $N_1 = 103,\ N_2 = 110,\ N_3 = 109,$
$N_4 = 100,\ N_5 = 94,\ N_6 = 96$

$$R = \frac{1}{1 - \dfrac{(94)(109)(103)}{(96)(100)(110)}} = 1505$$

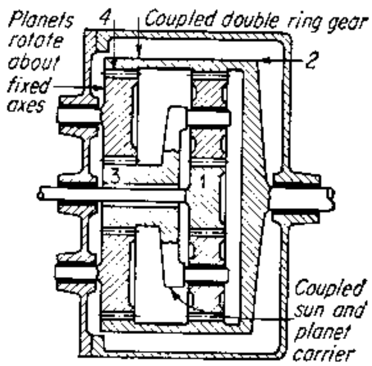
(A)

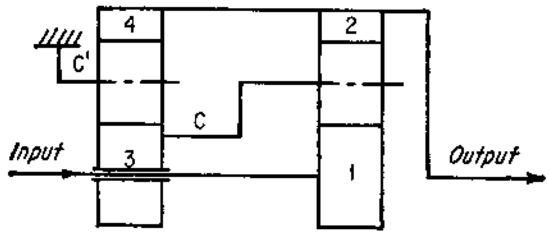




$$R = 1 - \frac{N_2 N_4}{N_1 N_3}$$

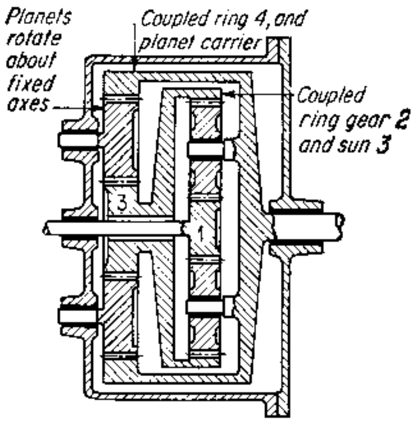
(B)

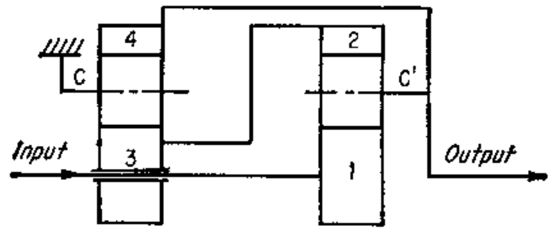




$$R = \left(1 + \frac{N_2}{N_1}\right)\left(-\frac{N_4}{N_3}\right) - \frac{N_2}{N_1}$$

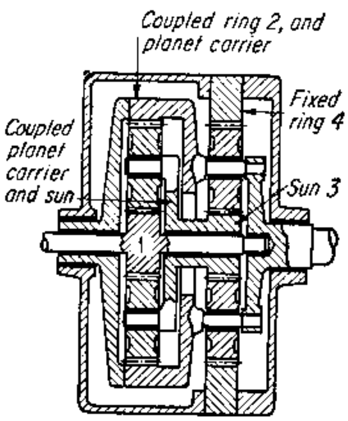
(C)

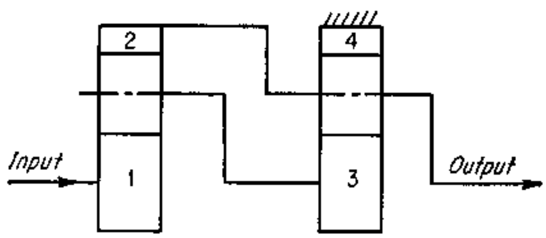




$$R = 1 + \frac{N_2}{N_1}\left(1 + \frac{N_4}{N_3}\right)$$

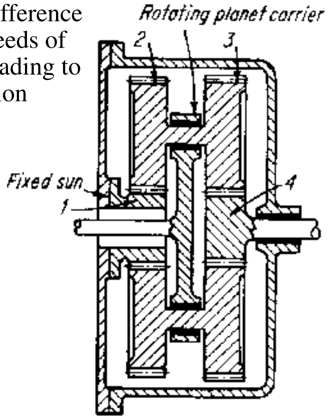
(D)





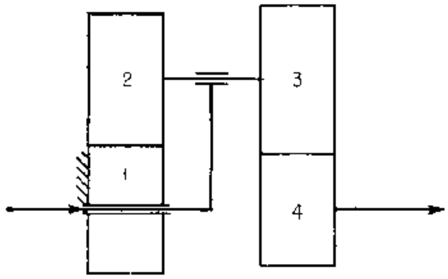
$$R = 1 + \frac{N_4}{N_3}\left(1 + \frac{N_2}{N_1}\right)$$

Output is difference between speeds of two parts leading to high reduction ratios

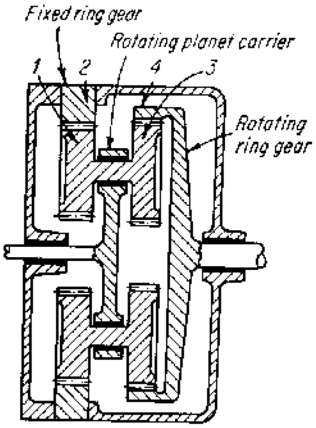


(A)

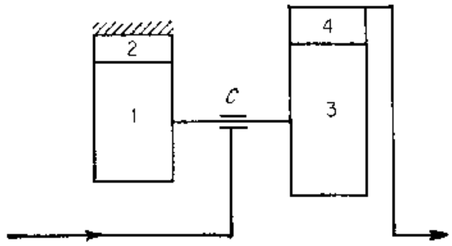
$$N_1 = 20$$
$$N_2 = 31$$
$$N_3 = 32$$
$$N_4 = 19$$



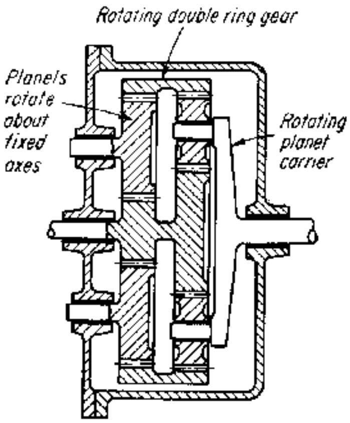
$$R = \frac{1}{1 - \dfrac{N_3 N_1}{N_4 N_2}} = \frac{1}{1 - \dfrac{(32)\,(20)}{(19)\,(31)}} = -11.549$$



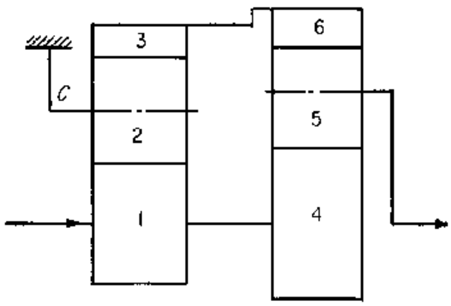
(B)



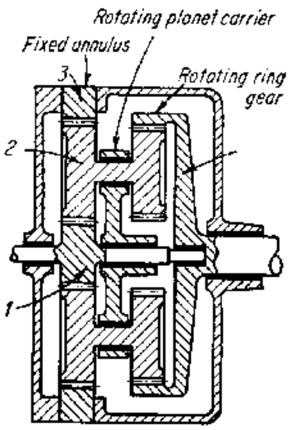
$$R = \frac{1}{1 - \dfrac{N_3 N_2}{N_4 N_1}}$$



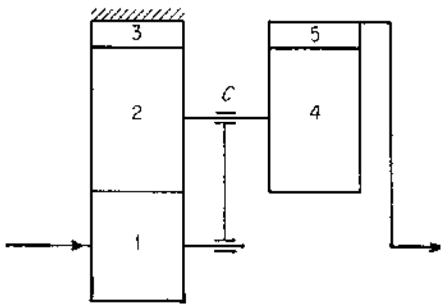
(C)



$$R = \frac{1 + (N_4/N_6)}{(N_4/N_6) - (N_1/N_3)}$$

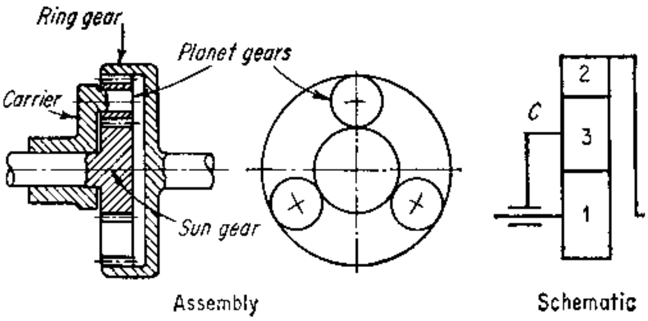


(D)

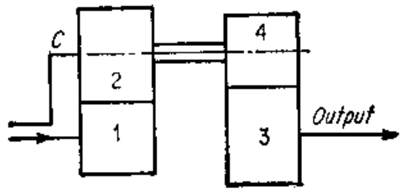


$$R = \frac{1 + (N_3/N_1)}{1 - \dfrac{N_4 N_3}{N_5 N_2}}$$

SIMPLE PLANETARIES AND INVERSIONS



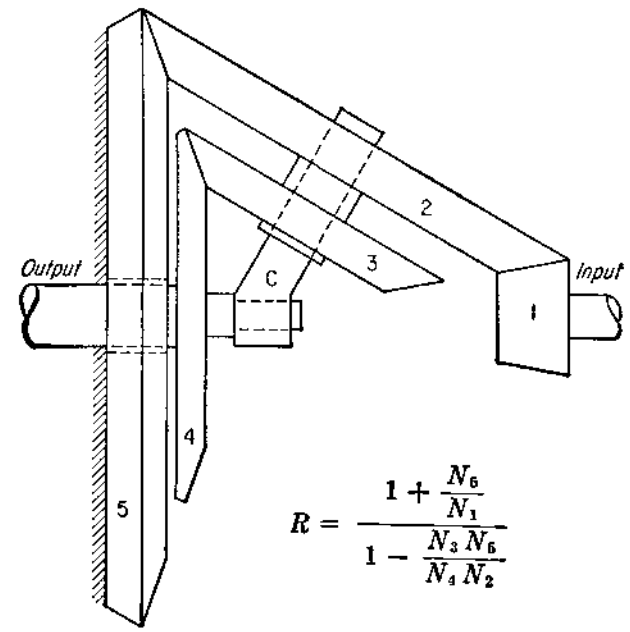
Input member	Fixed member	Output member	Speed-ratio equation
1	C	2	$R = -N_2/N_1$
2	C	1	$R = -N_1/N_2$
1	2	C	$R = 1 + (N_2/N_1)$
2	1	C	$R = 1 + (N_1/N_2)$
C	2	1	$R = \dfrac{1}{1+(N_2/N_1)}$
C	1	2	$R = \dfrac{1}{1+(N_1/N_2)}$



Input member	Fixed member	Output member	Speed-ratio equation
1	C	3	$R = \dfrac{N_2 N_3}{N_1 N_4}$
1	3	C	$R = 1 - \dfrac{N_2 N_3}{N_1 N_4}$
3	1	C	$R = 1 - \dfrac{N_1 N_4}{N_2 N_3}$
3	C	1	$R = \dfrac{N_4 N_1}{N_3 N_2}$
C	1	3	$R = 1\Big/\left(1 - \dfrac{N_1 N_4}{N_2 N_3}\right)$
C	3	1	$R = 1\Big/\left(1 - \dfrac{N_2 N_3}{N_1 N_4}\right)$

HUMPAGE'S BEVEL GEARS

Continued on next page

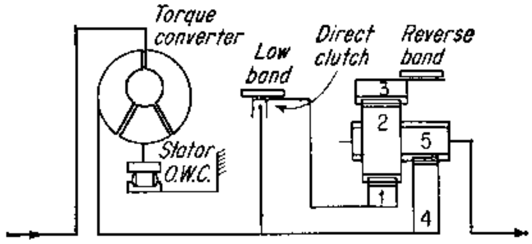


$$R = \frac{1 + \dfrac{N_6}{N_1}}{1 - \dfrac{N_3 N_6}{N_4 N_2}}$$

References

1. D. W. Dudley, ed., *Gear Handbook,* pp. 3-19 to 3-25, McGraw-Hill.

TWO-SPEED FORDOMATIC (Ford Motor Co.)

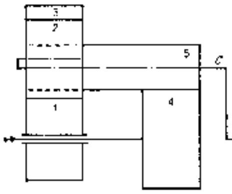


Low gear— gear 1 fixed

$$R = 1 + \frac{N_1}{N_4} = 1.75$$

Reverse gear— gear 3 fixed

$$R = 1 - \frac{N_3}{N_4} = -1.50$$

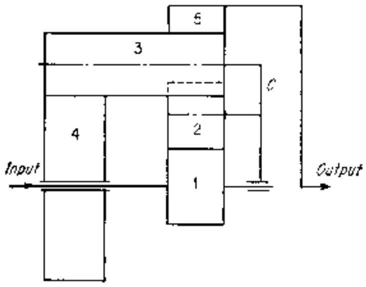


$N_1 = 27$
$N_2 = 31$
$N_3 = 90$
$N_4 = 36$
$N_5 = 19$

Note: Power-Glide Transmission is similar to above, but with $N_1 = 23$, $N_2 = 28$, $N_3 = 79$, $N_4 = 28$, $N_5 = 18$. This produces identical ratios in low and reverse.

$$R = 1 + \frac{23}{28} = 1.82 \qquad R = 1 - \frac{79}{28} = -1.82$$

CRUISE-O-MATIC 3-SPEED TRANSMISSION (Ford Motor Co.)



Long planet, $N_3 = 18$
Short planet, $N_2 = 18$
Sun gears, $N_4 = 36$, $N_1 = 30$
Ring gears, $N_5 = 72$

Low gear—Input to 1 C fixed

$$R = \frac{N_5}{N_1} = 2.4$$

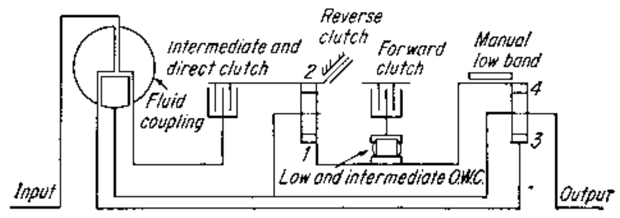
Intermediate gear—
Input to 1, gear 4 fixed

$$R = \frac{1 + \dfrac{N_4}{N_1}}{1 + \dfrac{N_4}{N_5}} = 1.467$$

Reverse gear—
Input to 4, C fixed

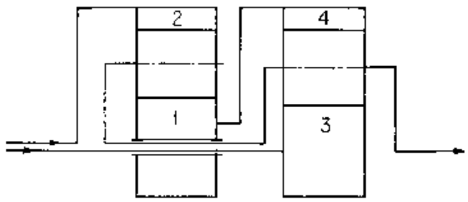
$$R = \frac{N_5}{N_4} = -2.0$$

HYDRAMATIC 3-SPEED TRANSMISSION (General Motors)



Low gear—
Input to 3, 4 fixed

$$R = 1 + \frac{N_4}{N_3} = 2.97$$



$N_1 = 46$
$N_2 = 82$
$N_3 = 39$
$N_4 = 77$

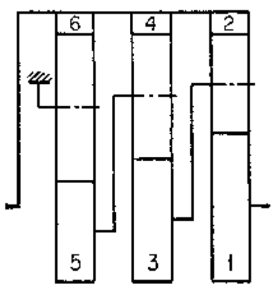
Intermediate gear—Input to 2, 1 fixed

$$= 1 + \frac{N_1}{N_2} = 1.56$$

Reverse gear—Input to 3, 2 fixed

$$R = 1 - \frac{N_4 N_2}{N_3 N_1} = -2.52$$

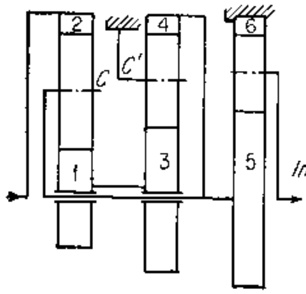
TRIPLE PLANETARY DRIVES

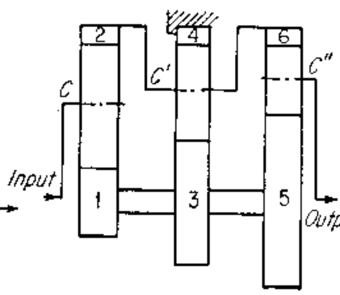


Input to gear 1, output from gear 6

$$R = \left(1 + \frac{N_2}{N_1}\right)\left[\left(1 + \frac{N_4}{N_3}\right)\left(-\frac{N_6}{N_5}\right) - \frac{N_4}{N_3}\right] - \frac{N_2}{N_1}$$

(B)



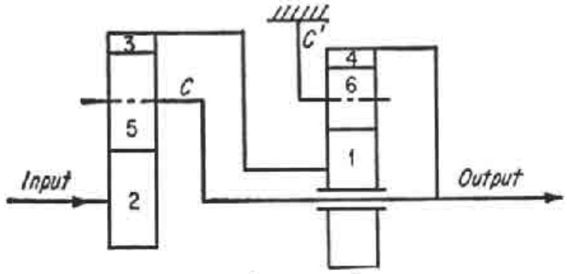


(C)

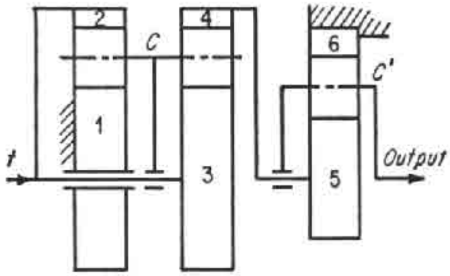
$$\text{(B)} \quad R = \left[1 + \frac{N_1}{N_2}\left(1 + \frac{N_4}{N_3}\right)\right]\left(1 + \frac{N_6}{N_5}\right)$$

$$\text{(C)} \quad R = \left[1 + \frac{N_4/N_3}{1 + (N_2/N_1)}\right] \Bigg/ \left[1 + \frac{N_4/N_3}{1 + (N_6/N_5)}\right]$$

Ring gear 3 coupled to sun gear 1; split output.

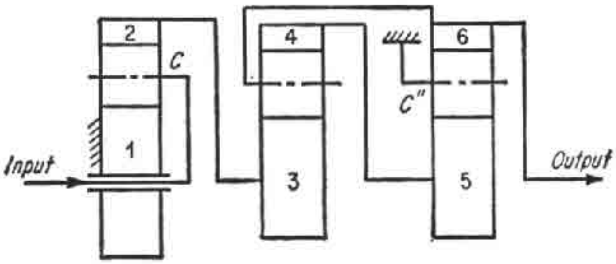


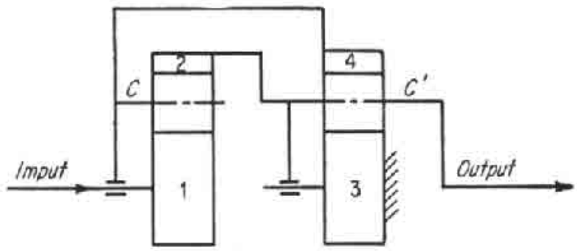
$$N_1 = 46$$
$$N_2 = 77$$
$$N_3 = 29$$
$$N_4 = 85$$



(A) $$R = 1 + \frac{N_3}{N_2}\left(1 + \frac{N_4}{N_1}\right)$$

(B) $$R = \left(\frac{1 + (N_1/N_2)}{1 - \dfrac{N_1 N_3}{N_2 N_4}}\right)\left(1 + \frac{N_6}{N_5}\right)$$



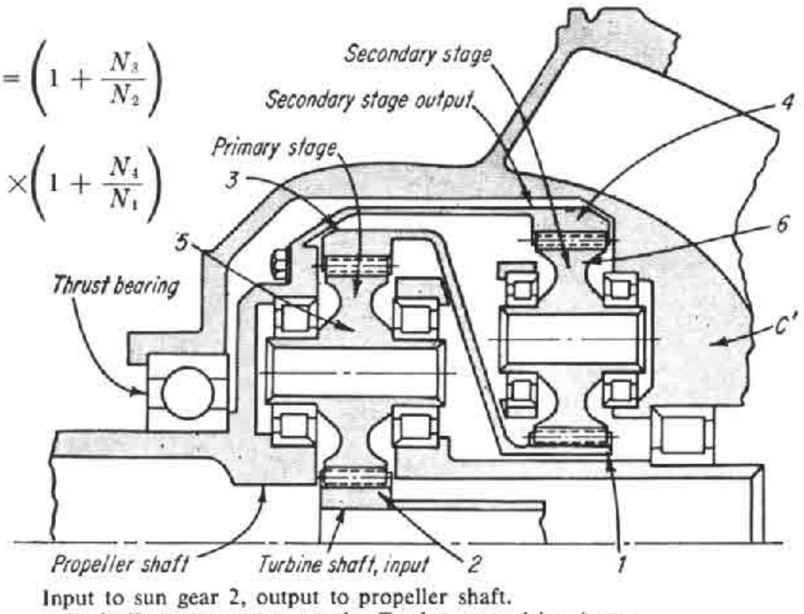


(C) $$R = \frac{1}{1 + \dfrac{N_1}{N_2}}\left[1 + \frac{N_4}{N_3}\left(1 + \frac{N_6}{N_5}\right)\right]$$

(D) $$R = \frac{N_3}{N_4}\left(1 - \frac{N_4}{N_3} + \frac{N_2}{N_1}\right)$$

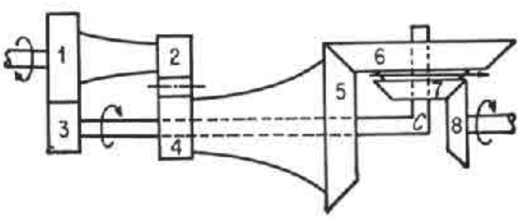
LYCOMING TURBINE DRIVE

$$R = \left(1 + \frac{N_3}{N_2}\right) \times \left(1 + \frac{N_4}{N_1}\right)$$



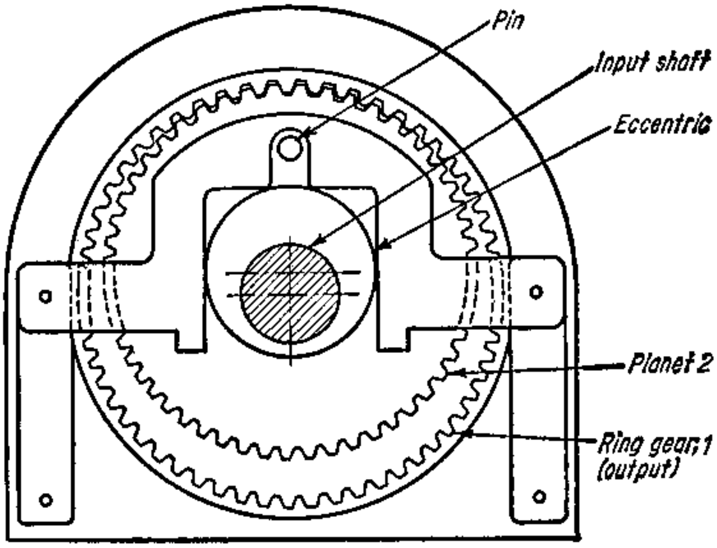
Input to sun gear 2, output to propeller shaft.
Basically same system as the Ford tractor drive, (gears are numbered the same way) and will have the same speed-ratio.

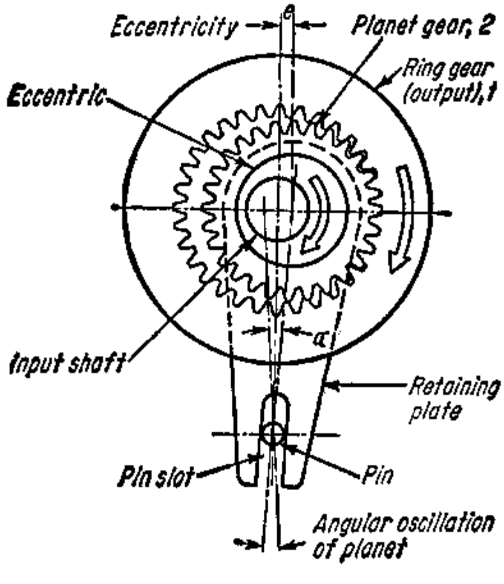
COMPOUND SPUR-BEVEL GEAR DRIVE



$$R = \frac{1}{-\dfrac{N_1}{N_3}\left(1 + \dfrac{N_7 N_5}{N_8 N_6}\right) - \dfrac{N_7 N_5 N_2}{N_8 N_6 N_4}}$$

TWO-GEAR PLANETARY DRIVES

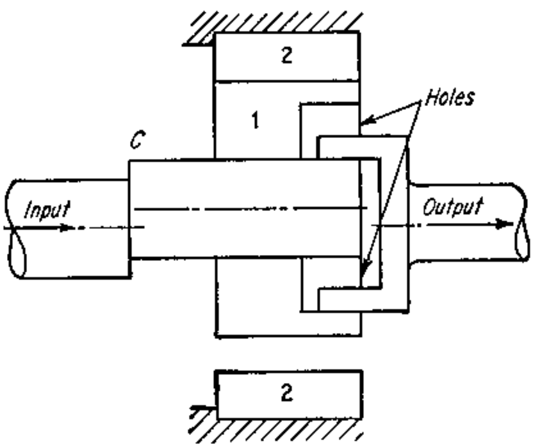




The pin in (A) is fixed to the planet which is mounted on the eccentric hub of the input shaft. The ring gear is the output gear. The system in (B) is simplified, but it produces slight pulsations in output.

$$R = \frac{1}{1 - (N_1/N_2)}$$

PLANOCENTRIC DRIVE

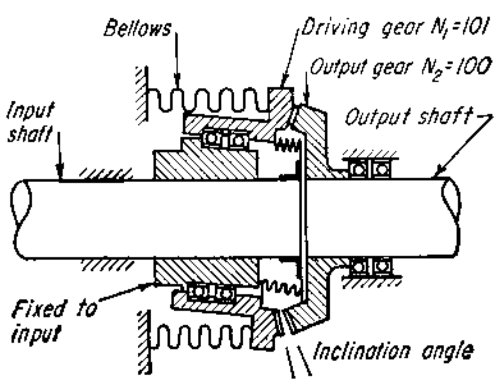


$$N_2 = 65$$
$$N_1 = 64$$

The planet gear 1 is eccentrically mounted to the input gear (planet 1 is not rigidly connected to the eccentric). The output is driven by holes.

$$R = \frac{N_1}{N_1 - N_2} = \frac{64}{64 - 65} = -64$$

WOBBLE-GEAR DRIVE



This drive is a close relative of the harmonic drive. The bevel "wobble" gears mesh at only one point on the circumference because of the slight angle of inclination of the driving gear, N_1, which has one tooth more than output gear, N_2. The driving gear, N_1, does not rotate: it yaws and pitches only.

$$R = R_i = \frac{1}{1 - m_{or}}$$

$$R = \frac{1}{1 - \frac{N_1}{N_2}} = \frac{1}{1 - \frac{101}{100}} = -100$$

NONCIRCULAR GEARS

Noncircular gears generally cost more than competitive components such as linkages and cams. But with the development of modern production methods, such as the computer-controlled gear shaper, cost has gone down considerably. Also, in comparison with linkages, noncircular gears are more compact and balanced—and they can be more easily balanced. These are important considerations in high-speed machinery. Furthermore, the gears can produce continuous, unidirectional cyclic motion—a point in their favor when compared with cams. The disadvantage of cams is that they offer only reciprocating motion.

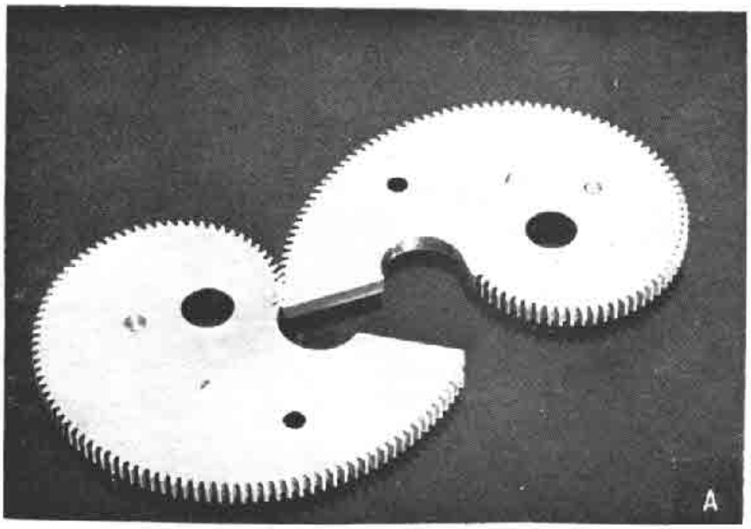
Applications can be classified into two groups:

- Where only an over-all change in angular velocity of the driven member is required, as in quick-return drives, intermittent mechanisms in such machines as printing presses, planers, shears, winding machines, and automatic-feed machines.
- Where precise, nonlinear functions must be generated, as in mechanical computing machines for extracting roots of numbers, raising numbers to any power, or generating trigonometric and logarithmic functions.

Noncircular Gears

It is always possible to design a specially shaped gear to roll and mesh properly with a gear of any shape. The sole requirement is that the distance between the two axes must be constant. However, the pitch line of the mating gear might turn out to be an open curve, and the gears can be rotated only for a portion of a revolution—as with two logarithmic-spiral gears (illustrated in Fig. 1).

True elliptical gears can only be made to mesh properly if they are twins, and if they are rotated about their focal points. However, gears resembling ellipses can be generated from a basic ellipse. These "higher-order" ellipses (see Fig. 2) can be meshed in various interesting combinations to rotate about centers *A, B, C,* or *D.* For example, two second-order elliptical gears can be meshed to rotate about their geometric center; however, they will produce two complete speed cycles per revolution. The difference in contour between a basic ellipse and a second-order ellipse is usually very slight. Note also that the fourth-order "ellipses" resemble square gears (this explains why the square gears, sometimes found as ornaments on tie clasps, illustrated in Fig. 3, actually work).



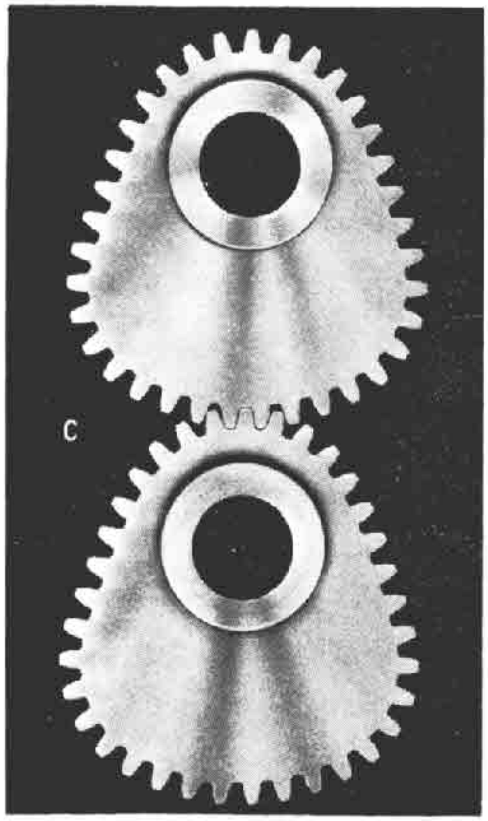


Fig. 1 The logarithmic spiral gears shown in (A), are open-curved. They are usually components in computing devices. The elliptically shaped gears, shown in (B), are closed curved. They are components in automatic machinery. The specially shaped gears, shown in (C), offer a wider range of velocity and acceleration characteristics.

A circular gear, mounted eccentrically, can roll properly only with specially derived curves (shown in Fig. 4). One of the curves, however, closely resembles an ellipse. For proper mesh, it must have twice as many teeth as the eccentric gear. When the radiis *r,* and eccentricity, *e,* are known, the major semiaxis of the elliptically shaped gear becomes $2r + e$, and the minor $2r - e$. Note also that one of the gears in this group must have internal teeth to roll with the eccentric gear. Actually, it is possible to generate internal-tooth shapes to rotate with noncircular gears of any shape (but, again, the curves can be of the open type).

Noncircular gears can also be designed to roll with specially shaped racks (shown in Fig. 5). Combinations include: an elliptical gear and a sinusoid-like rack. A third-order ellipse is illustrated, but any of the elliptical rolling curves can be used in its place. The main advantage of those curves is that when the ellipse rolls, its axis of rotation moves along a straight line; other combinations include a logarithmic spiral and straight rack. The rack, however, must be inclined to its direction of motion by the angle of the spiral.

DESIGN EQUATIONS

Equations for noncircular gears are given here in functional form for three common design requirements. They are valid for any noncircular gear pair. Symbols are defined in the box.

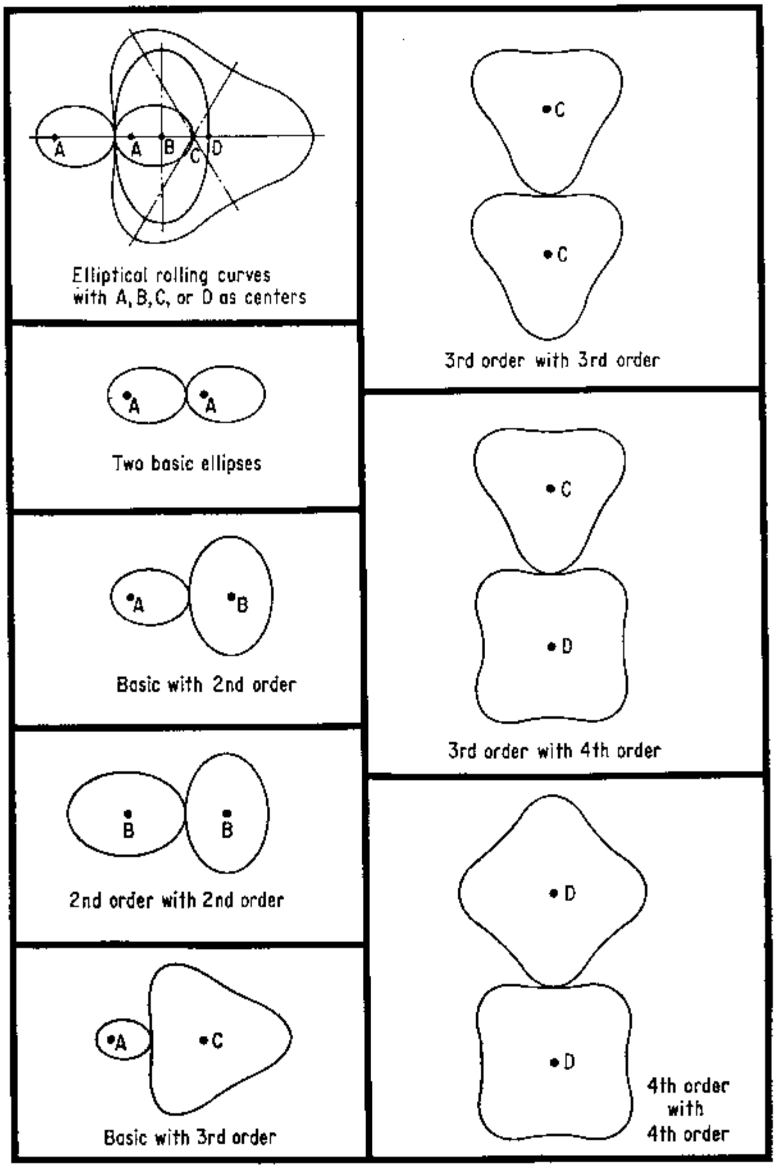


Fig. 2 Basic and High-Order Elliptical Gear Combinations.

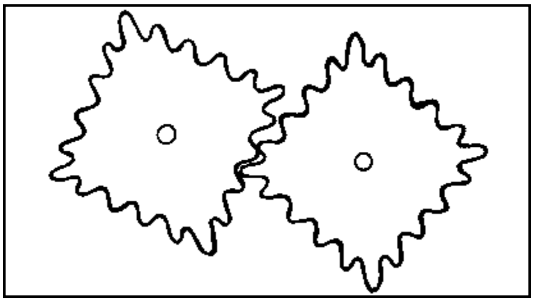


Fig. 3 Square gears seem to defy basic kinematic laws, but they are a takeoff on a pair of fourth-order ellipses.

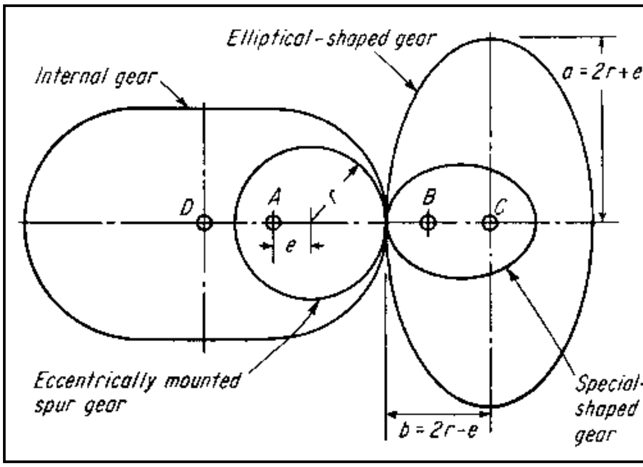


Fig. 4 An eccentric spur gear rotating about point **A,** will mesh properly with any of the three gears shown whose centers are at points B, C and D.

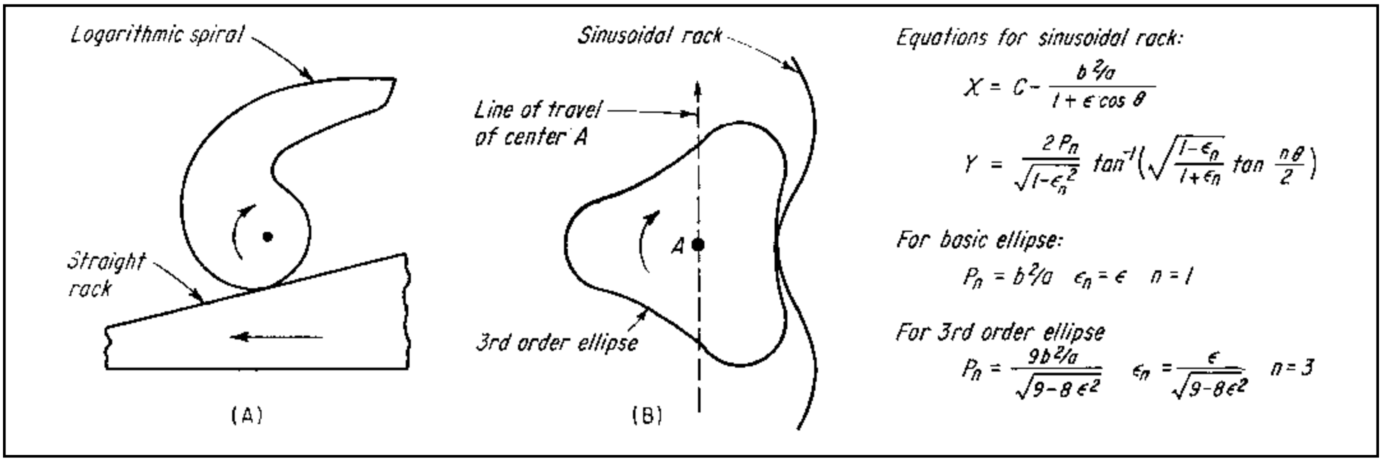
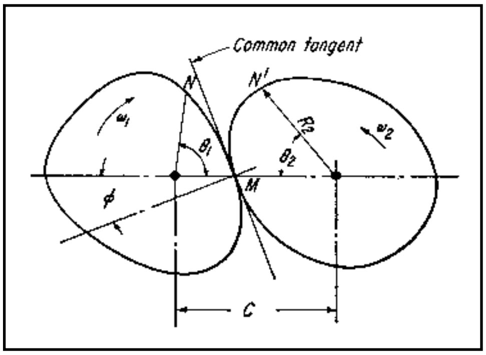


Fig. 5 Rack and gear combinations are possible with noncircular gears. The straight rack for the logarithmic spiral (A) must move obliquely; the center of third-order ellipse (B) follows a straight line.



Symbols

a = semi-major axis of ellipse
b = semi-minor axis of ellipse
C = center distance (see above sketch)
ϵ = eccentricity of an ellipse = $\sqrt{1-(b/a)^2}$
e = eccentricity of an eccentrically mounted spur gear
N = number of teeth
P = diametral pitch
r_c = radius of curvature
R = active pitch radius
S = length of periphery of pitch circle
X, Y = rectangular coordinates
θ = polar angle to R
ϕ = angle of obliquity
ω = angular velocity
$f(\theta), F(\theta), G(\theta)$ = various functions of θ
$f'(\theta), F'(\theta), G'(\theta)$ = first derivatives of functions of θ

CASE I Polar equation of one curve and center distance are known; to find the polar equation of the mating gear:

$$R_1 = f(\theta_1)$$
$$R_2 = C - f(\theta_1)$$
$$\theta_2 = -\theta_1 + C\int \frac{d\theta_1}{C - f(\theta_1)}$$

CASE II The relationship between angular rotation of the two members and the center distance are known; to find the polar equations of both members:

$$\theta_2 = F(\theta_1)$$
$$R_1 = \frac{CF'(\theta_1)}{1 + F'(\theta_1)}$$
$$R_2 = C - R_1 = \frac{C}{1 + F'(\theta_1)}$$

CASE III The relationship between angular velocities of the two members and the center distance are known; to find the polar equations of both members:

$$\omega_2 = \omega_1 G(\theta_1)$$
$$R_1 = \frac{CG(\theta_1)}{1 + G(\theta_1)}$$
$$R_2 = C - R_1$$
$$\theta_1 = \int G(\theta_1)d\theta_1$$

Velocity equations and the characteristics of five types of noncircular gears are listed in the table.

CHECKING FOR CLOSED CURVES

Gears can be quickly analyzed to determine whether their pitch curves are open or close with the following equations:

In case I, if $R = f(\theta) = f(\theta + 2N_\pi)$, the pitch curve is closed.

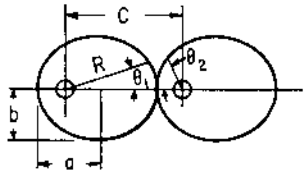
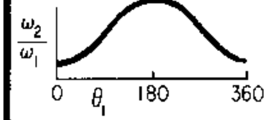
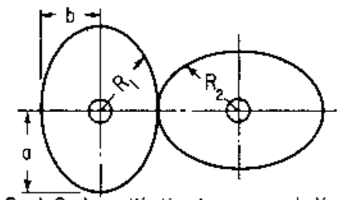
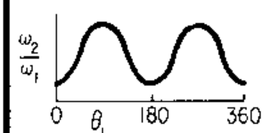
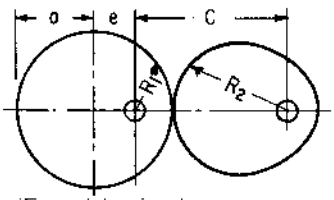
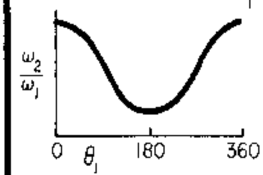
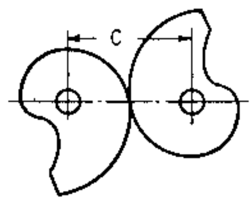
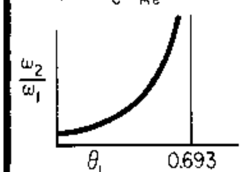
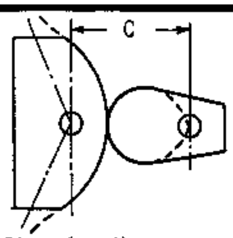
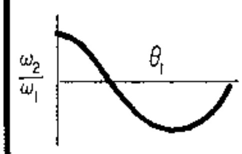
In case II, if $\theta_1 = F(\theta_2)$ and $F(\theta_0) = 0$, the curve is closed with the equation $F(\theta_0 + 2\pi/N_1) = 2_\pi/N_2$ can be satisfied by substituting integers or rational fractions for N_1 and N_2. If fractions must be used to solve this equation, the curve will have double points (intersect itself), which is, or course, an undesirable condition.

In case III, if $\theta_2 = \int G(\theta_1)d\theta_1$, let $G(\theta_1)d\theta_1 = F(\theta_1)$, and use the same method as for Case II, with the subscripts reversed.

With some gear sets, the mating gear will be a closed curve only if the correct center distance is employed. This distance can be found from the equation:

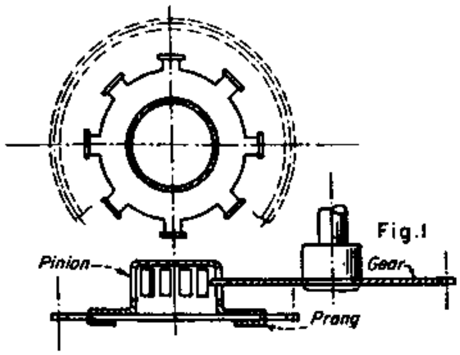
$$4\pi = \int_0^{2\pi} \frac{d\theta_1}{C - f(\theta_1)}$$

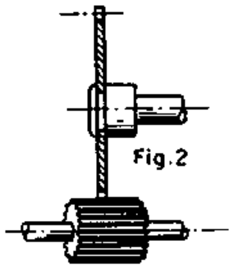
Characteristics of Five Noncircular Gear Systems

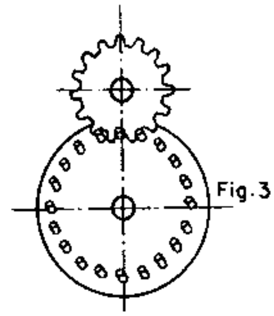
Type	Comments	Basic equations	Velocity equations ω_1 = constant
 Two ellipses rotating about foci	Gears are identical. Comparatively easy to manufacture. Used for quick-return mechanisms, printing presses, automatic machinery	$R = \dfrac{b^2}{a[1+\epsilon\cos\theta]}$ ϵ = eccentricity $= \sqrt{1-\left(\frac{b}{a}\right)^2}$ $a = \frac{1}{2}$ major axis $b = \frac{1}{2}$ minor axis	$\omega_2 = \omega_1\left[\dfrac{r^2+1+(r^2-1)\cos\theta_2}{2r}\right]$ where $r = \dfrac{R\ max}{R\ min}$ 
 2nd Order elliptical gears rotating about their geometric centers	Gears are identical. Geometric properties well known. Better balanced than true elliptical gears. Used where two complete speed cycles are required for one revolution	$R = \dfrac{2ab}{(a+b)-(a-b)\cos 2\theta}$ $C = a+b$ a = maximum radius b = minimum radius	$\omega_2 = \omega_1\left[\dfrac{r^2+1-(r^2-1)\cos 2\theta_2}{2r}\right]$ where $r = \dfrac{a}{b}$ 
 Eccentric circular gear rotating with its conjugate	Standard spur gear can be employed as the eccentric. Mating gear has special shape	$R_1 = \sqrt{a^2+e^2+2ae\cos\theta_1}$ $\theta_2 = \theta_1 + C\displaystyle\int\frac{d\theta_1}{C-R_1}$ $C = R_1 + R_2$	$\dfrac{\omega_2}{\omega_1} = \dfrac{\sqrt{a^2+e^2+2ae\cos\theta_1}}{C-\sqrt{a^2+e^2+2ae\cos\theta_1}}$ 
 Logarithmic spiral gears	Gears can be identical although can be used in conbinations to give variety of functions. Must be open gears	$R_1 = Ae^{k\theta_1}$ $R_2 = C - R_1$ $= Ae^{k\theta_2}$ $\theta_2 = \dfrac{1}{k}\log(C-Ae^{k\theta_1})$ e = natural log base	$\dfrac{\omega_2}{\omega_1} = \dfrac{Ae^{k\theta_1}}{C-Ae^{k\theta_1}}$ 
 Sine-function gears	For producing angular displacement proportional to sine of input angle. Must be open gears	$\theta_2 = \sin^{-1}(k\theta_1)$ $R_2 = \dfrac{C}{1+k\cos\theta_1}$ $R_1 = C - R_2$ $= \dfrac{Ck\cos\theta_1}{1+k\cos\theta_1}$	$\dfrac{\omega_2}{\omega_1} = k\cos\theta_1$ 

SHEET-METAL GEARS, SPROCKETS, WORMS, AND RATCHETS

When a specified motion must be transmitted at intervals rather than continuously, and the loads are light, these mechanisms are ideal because of their low cost and adaptability to mass production. Although not generally considered precision parts, ratchets and gears can be stamped to tolerances of ±0.007 in, and if necessary, shaved to close dimensions.







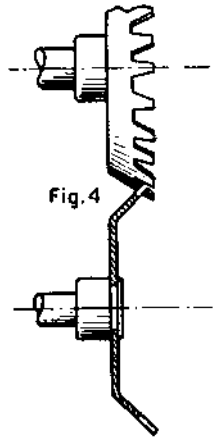
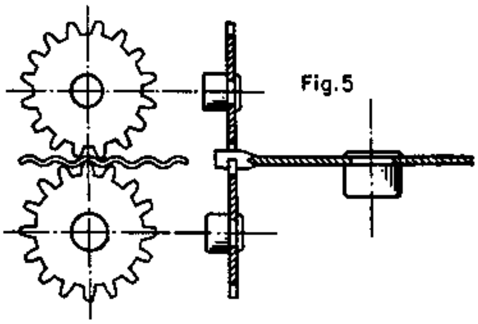


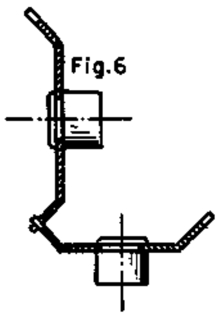
Fig. 1 The pinion is a sheet metal cup with rectangular holes serving as teeth. The meshing gear is sheet metal, blanked with specially formed teeth. The pinion can be attached to another sheet metal wheel by prongs, as shown, to form a gear train.

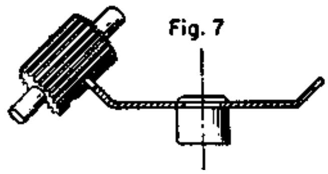
Fig. 2 The sheet-metal wheel gear meshes with a wide-face pinion, which is either extruded or machined. The wheel is blanked with teeth of conventional form.

Fig. 3 The pinion mates with round pins on a circular disk made of metal, plastic or wood. The pins can be attached by staking or with threaded fasteners.

Fig. 4 Two blanked gears, conically formed after blanking, become bevel gears meshing on a parallel axis. Both have specially formed teeth.







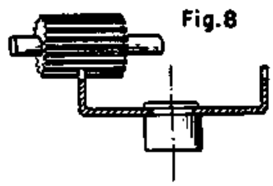


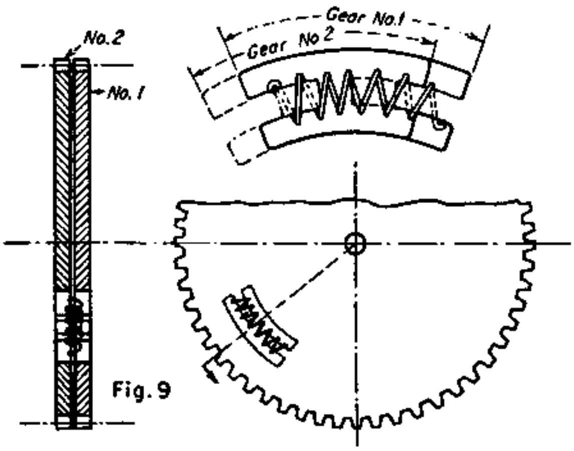
Fig. 5 The horizontal wheel with waves on its out rim replacing teeth, meshes with either one or two sheet-metal pinions. They have specially formed teeth and are mounted on intersecting axes.

Fig. 6 Two bevel-type gears, with specially formed teeth, are mounted on 90° intersecting axes. They can be attached by staking them to hubs.

Fig. 7 The blanked and formed bevel-type gear meshes with a machined or extruded pinion. Conventional teeth can be used on both the gear and pinion.

Fig. 8 The blanked, cup-shaped wheel meshes with a solid pinion on 90° intersecting axes.

Fig. 9 Backlash can be eliminated from stamped gears by stacking two identical gears and displacing them by one tooth. The spring then bears one projection on each gear, taking up lost motion.



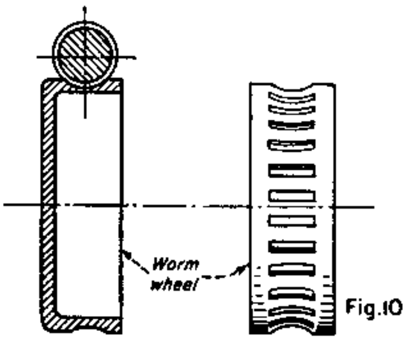


Fig. 10 A sheet metal cup with indentations replacing worm-wheel teeth, meshes with a standard coarse-thread screw.

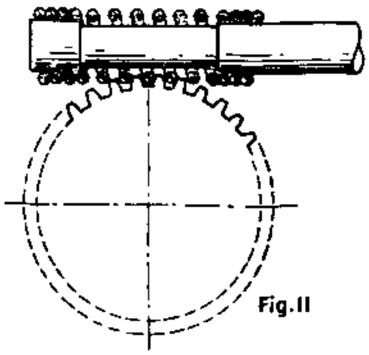


Fig. 11 A blanked wheel, with specially formed teeth, meshes with a helical spring mounted on a shaft, which serves as the worm.

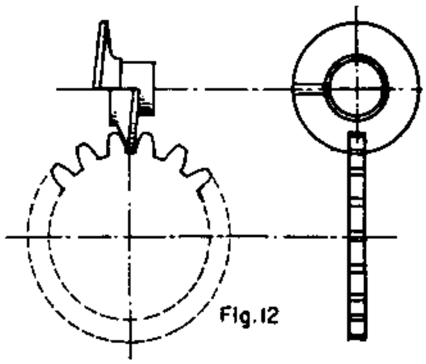
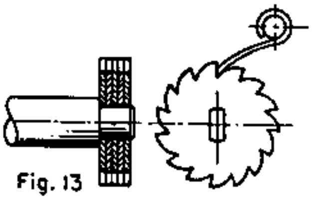


Fig. 12 This worm wheel is blanked from sheet metal with specially formed teeth. The worm is a sheet-metal disk that was split and helically formed.



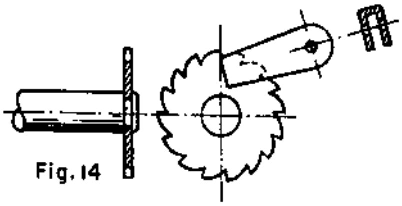


Fig. 13 Blanked ratchets with one-sided teeth are stacked to fit a wide-sheet-metal finger when single thickness is inadequate. The ratchet gears can be spot-welded.

Fig. 14 To avoid stacking, a single ratchet is used with a U-shaped finger, also made of sheet metal.

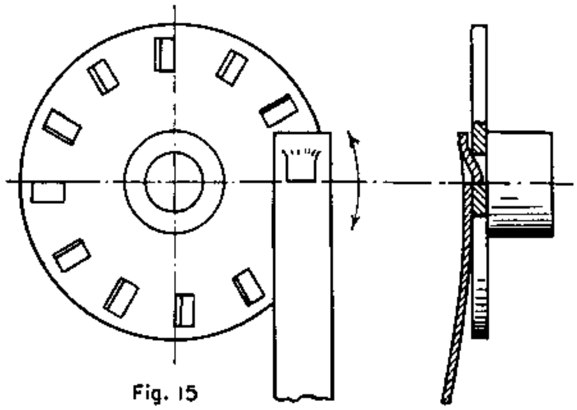


Fig. 15 This wheel is a punched disk with square-punched holes serving as teeth. The pawl is spring steel.

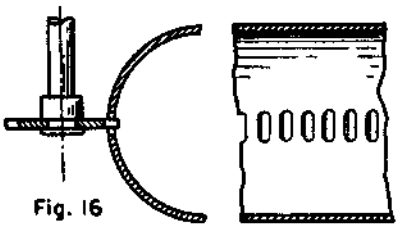
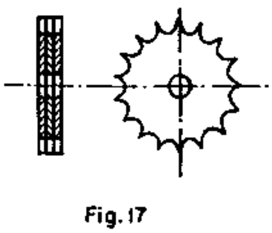


Fig. 16 This sheet-metal blanked pinion, with specially formed teeth, meshes with windows blanked in a sheet metal cylinder. They form a pinion-and-rack assembly.



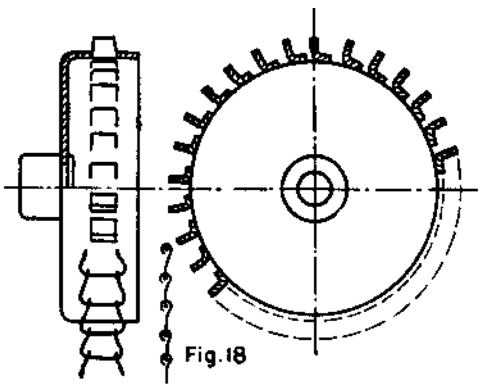


Fig. 17 This sprocket, like that in Fig. 13, can be fabricated from separate stampings.

Fig. 18 For a wire chain as shown, the sprocket is made by bending out punched teeth on a drawn cup.

HOW TO PREVENT REVERSE ROTATION

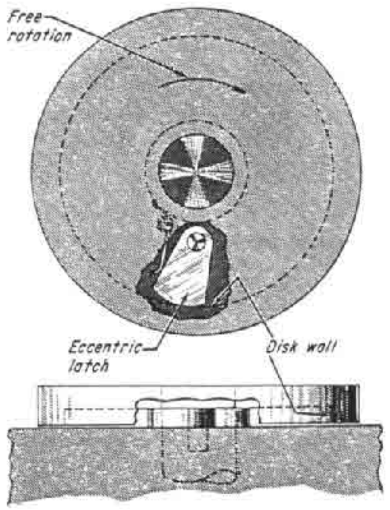


Fig. 1 An eccentric latch allows the shaft to rotate in one direction; any attempted reversal immediately causes the latch to wedge against the disk wall.

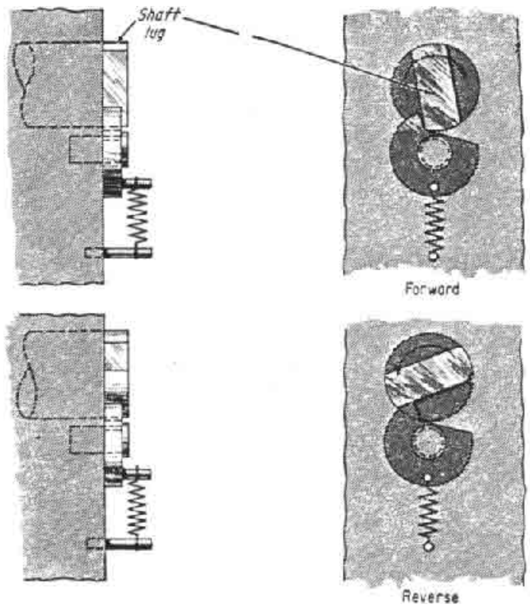


Fig. 2 A lug on a shaft pushes the notched disk free during normal rotation. The disk periphery stops the lug to prevent reverse rotation.

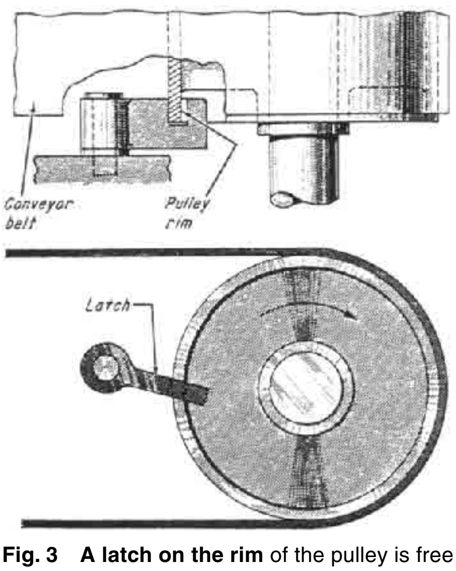


Fig. 3 A latch on the rim of the pulley is free only when the rotation is in the direction shown. This arrangement is ideal for conveyor-belt pulleys.

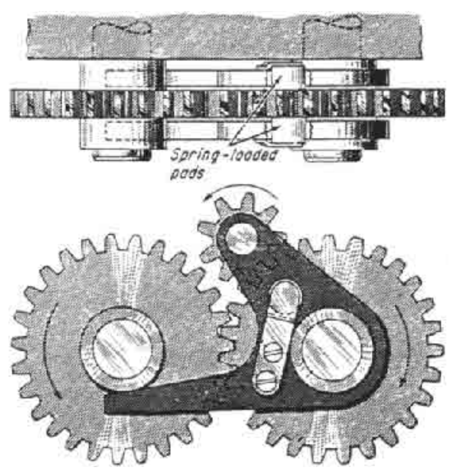


Fig. 4 Spring-loaded friction pads contact the right gear. The idler meshes and locks the gear set when the rotation is reversed.

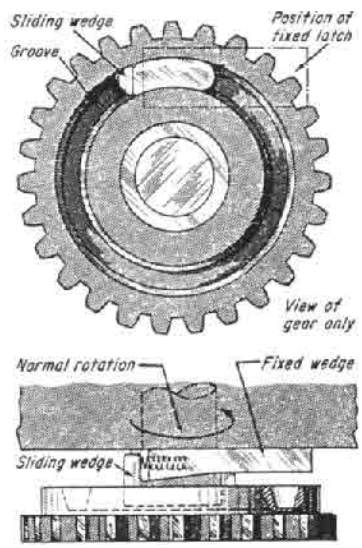


Fig. 5 A fixed wedge and sliding wedge tend to disengage when the gear is turning clockwise. The wedges jam in the reverse direction.

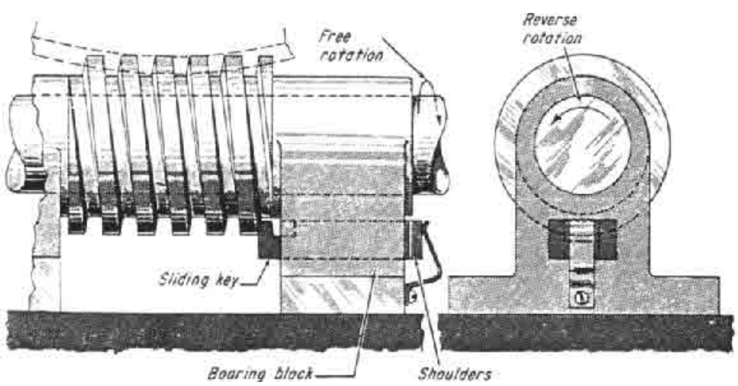


Fig. 6 A sliding key has a tooth which engages the worm threads. In reverse rotation, the key is pulled in until its shoulders contact the block.

GEAR-SHIFT ARRANGEMENTS

13 ways of arranging gears and clutches to obtain changes in speed ratios

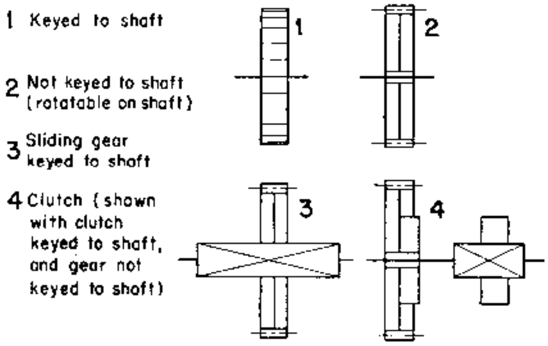


Fig. 1 The schematic symbols used in the following illustrations to represent gears and clutches.

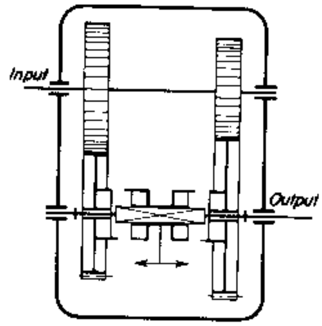


Fig. 2 Double-clutch drive. Two pairs of gears are permanently in mesh. Pair I or II transmits motion to the output shaft depending on the position of the coupling; the other pair idles. The coupling is shown in a neutral position with both gear pairs idle. Herring-bone gears are recommended for quieter running.

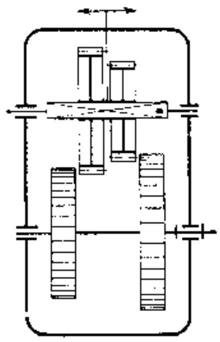


Fig. 3 Sliding-change drive. Gears are meshed by lateral sliding. Up to three gears can be mounted on a sliding sleeve. Only one pair is in mesh in any operating position. This drive is simpler, cheaper, and more extensively used than the drive of Fig. 2. Chamfering the sides of the teeth eases their engagement.

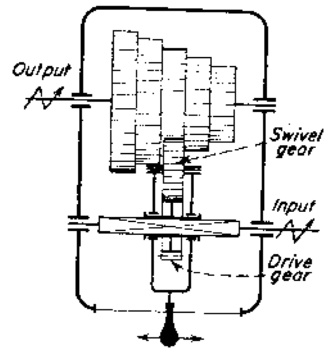


Fig. 4 Swivel-gear drive. Output gears are fastened to the shaft. A handle is pushed down, then shifted laterally to obtain transmission through any output gear. This drive is not suitable for the transmission of large torques because the swivel gear tends to vibrate. Its overall ratio should not exceed 1:3.

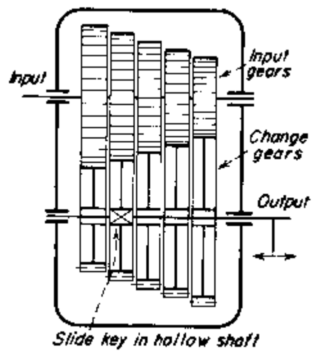


Fig. 5 Slide-key drive. A spring-loaded slide key rides inside a hollow output shaft. The slide key snaps out of the shaft when it is in position to lock a specific change gear to the output shaft. No central position is shown.

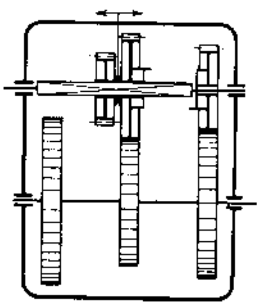


Fig. 6 This is a combination coupling and slide gears. It has three ratios: a direct mesh for ratios I and II; a third ratio is transmitted through gears II and III, which couple together.

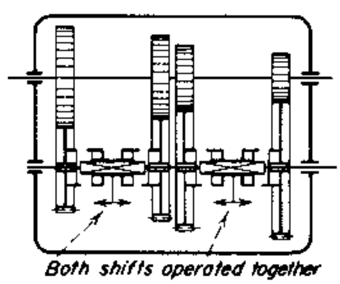
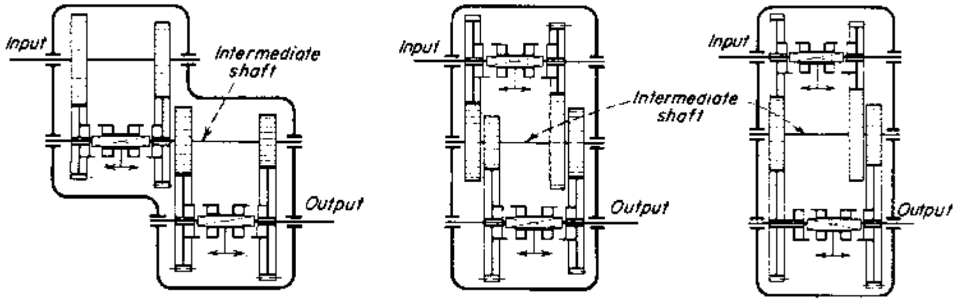
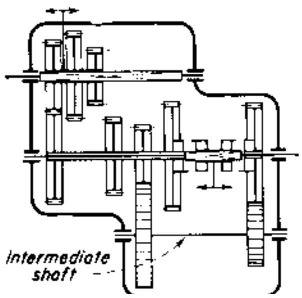


Fig. 7 Double-shift drive. One shift must always be in a neutral position. That might require both levers to be shifted when making a change. However, only two shafts are used to achieve four ratios.



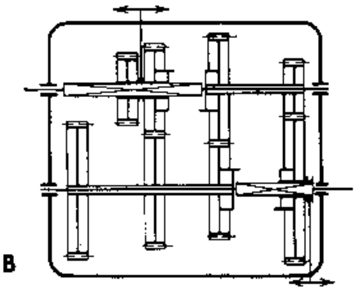
A B C

Fig. 8 A triple shaft drive gives four ratios. (A) The output of the first drive serves as the input for the second. The presence of an intermediate shaft eliminates the requirement for ensuring that one shift is always in the neutral position. A wrong shift-lever position cannot cause damage. (B) A space-saving modification; the coupling is on shaft *A* instead of the intermediate shaft. (C) Still more space is saved if one gear replaces a pair on the intermediate shaft. Ratios can be calculated to allow this.



A

Fig. 9 Six ratios are available with two couplings and (A) ten gears, (B) eight gears. Up to six gears can be in permanent mesh. It is not necessary to ensure that one shift is in neutral.



B

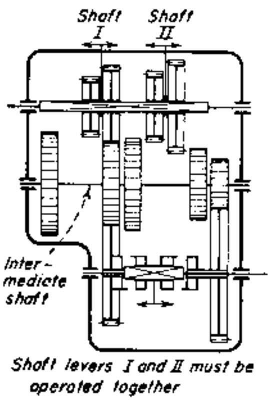


Fig. 10 This eight-ratio drive has two slide gears and a coupling. This arrangement reduces the number of parts and meshes. The position of shifts I and II are interdependent. One shift must be in neutral if the other *is* in mesh.

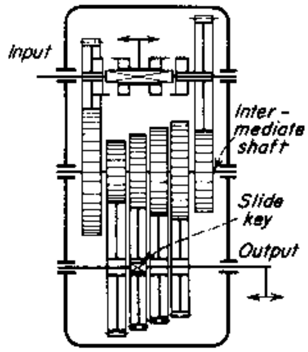
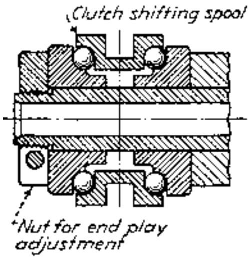
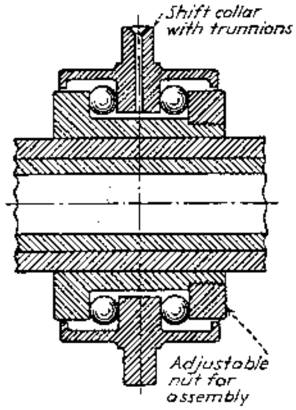
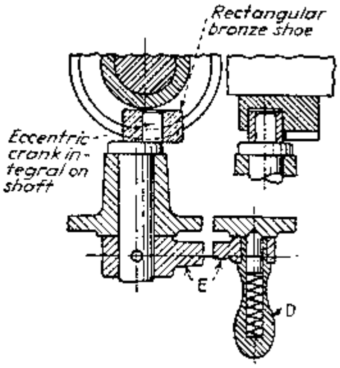


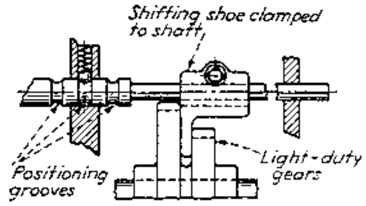
Fig. 11 This drive has eight ratios; a coupled gear drive and slide-key drive are in series. Comparatively low strength of the slide key limits the drive to small torque.

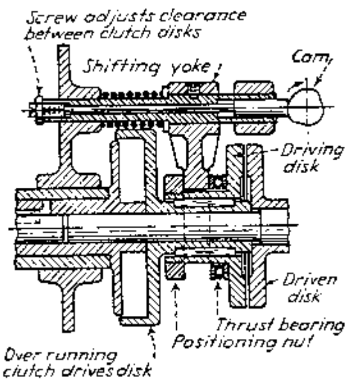
SHIFTING MECHANISMS FOR GEARS AND CLUTCHES

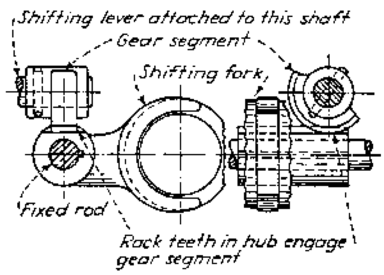


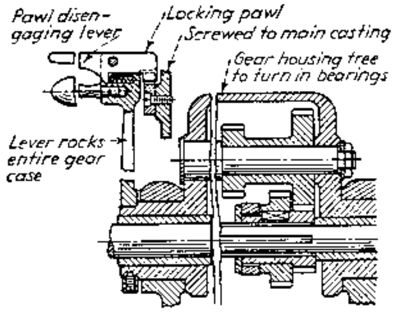


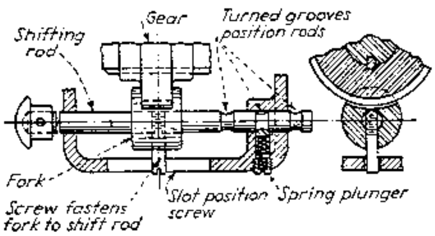


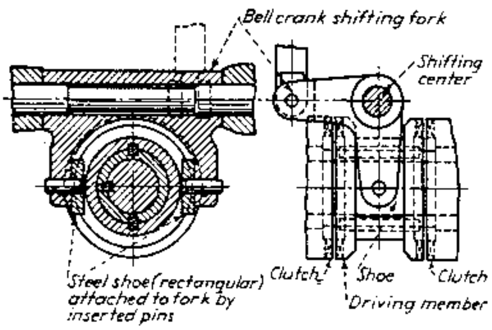


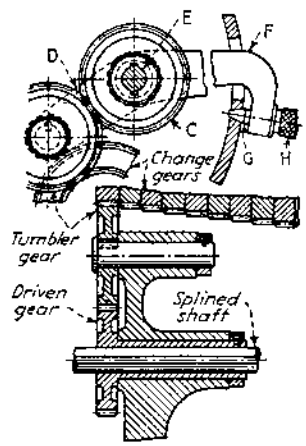


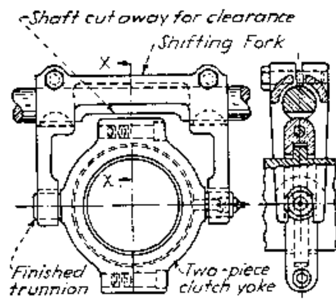


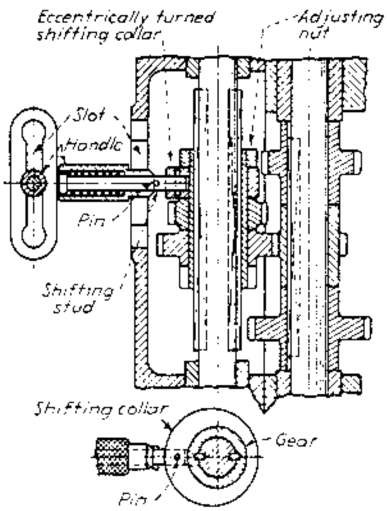


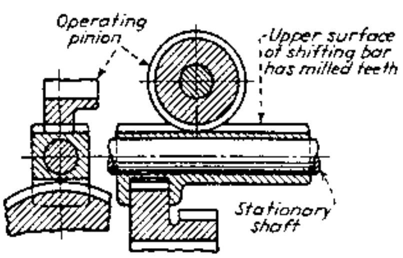


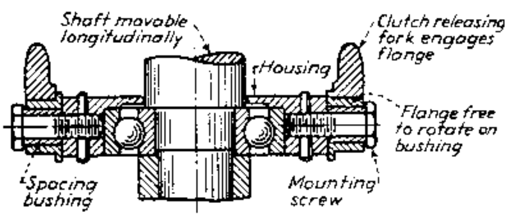


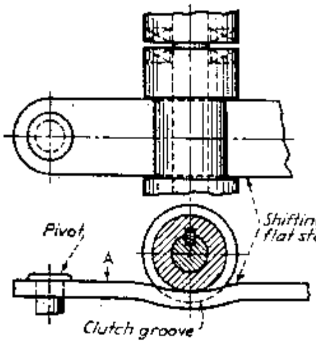


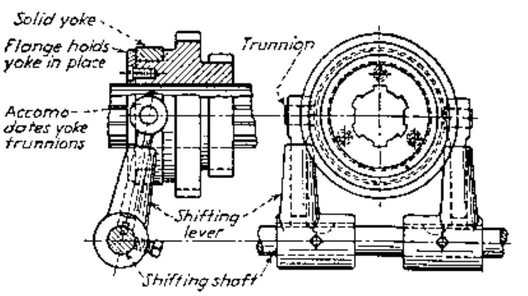


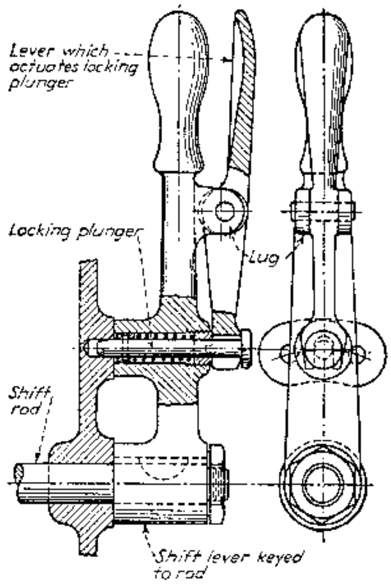


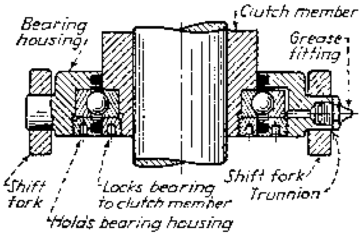


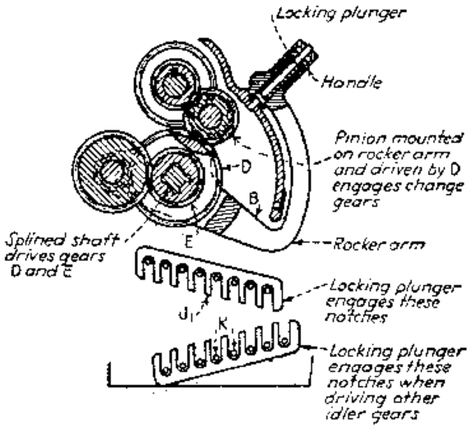












Spiderless differential

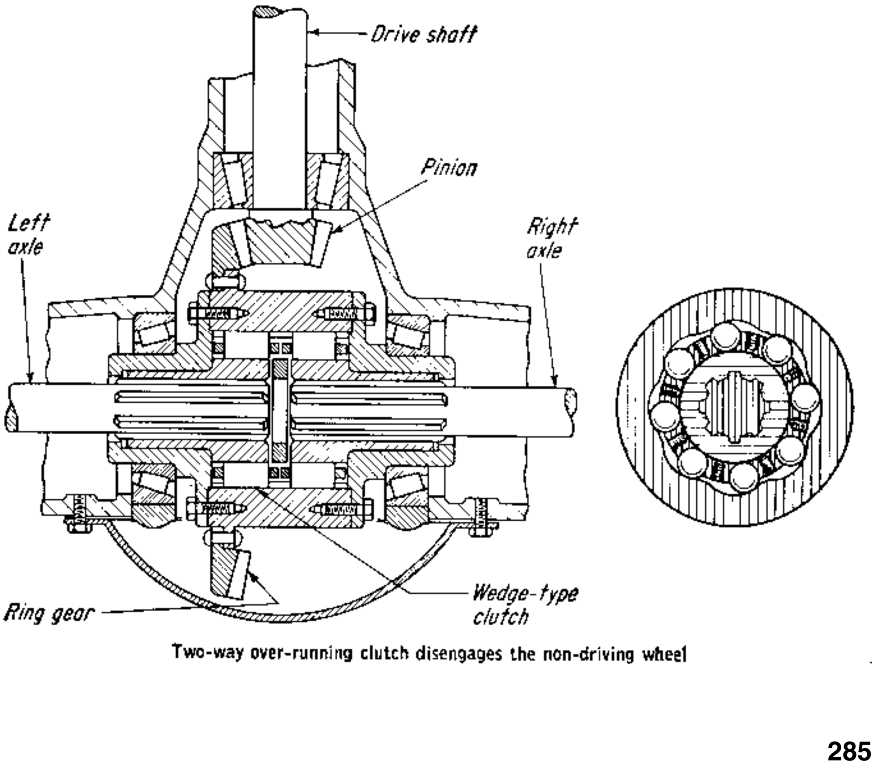
If you've ever been unable to drive your car out of a ditch because one wheel spun uselessly while the other sat torqueless and immobile, you'll thank the inventors (Seliger and Hegar) of the limited-slip differential shown here.

In straight running, it performs as a drive axle, driven by the driveshaft pinion through the ring gear. The differential action occurs only when one wheel loses traction, travels along a different arc, or otherwise attempts to turn at a speed that is different from that of the other. Then the wedge-type, two-way, over-running clutch (second figure) disengages, freeing the wheel to spin without drag.

Variations. Each clutch has three positions: forward drive, idle, and reverse drive. Thus, there are many combinations of drive-idle, depending on road conditions and turn direction. US Patent 3,124,972 describes a few:

- For left turns, the left wheel is driving, and the right wheel is forced to turn faster—thus over-running and disengaging the clutch. A friction ring built into each clutch assembly does the shifting. Wear is negligible.
- If power should be removed from the driveshaft during the left turn, the friction rings will shift each clutch and cause the left wheel to run free and the right wheel to drag in full coupling with the car's driveshaft.
- If your car is on the straightaway, under power and one wheel is lifted out of contact with the road, the other immediately transmits full torque. (The conventional spider differential performs in the opposite manner.)

On or off. Note one limitation, however: There is no gradual division of power. A wheel is either clutched in and turning at exactly the same speed as its opposite, or it is clutched out. It is not the same kind of mechanism as the conventional spider differential, which divides the driving load variably at any ratio of speeds.



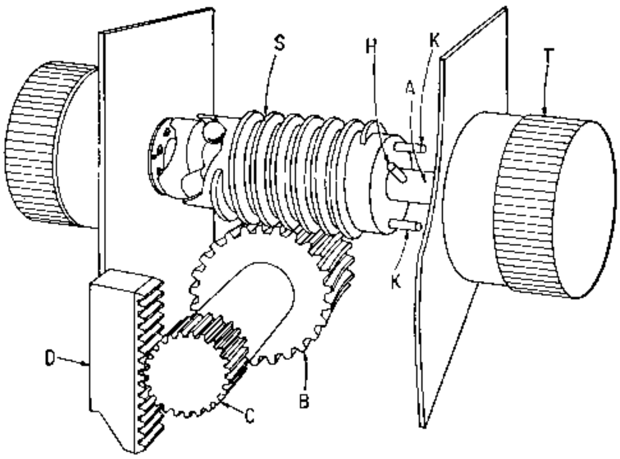
Two-way over-running clutch disengages the non-driving wheel

FINE-FOCUS ADJUSTMENTS

This single knob control for coarse and fine microscope adjustment is available in three series of Leitz microscopes. It is more suitable for high magnification than for low-power work where a greater range of fine adjustment is necessary. The mechanism runs in ball bearings, totally enclosed so lubrication is unnecessary.

Turning the knob continuously in one direction provides the coarse adjustment. When the direction is reversed, the fine adjustment is automatically engaged for about a one-third turn of the knob. Turning beyond this amount at either end shifts it back to coarse adjustment.

Worm S is loosely mounted on shaft A, along which it can move a short distance. Drive knob T is rigidly attached to the shaft. When the drive pin H on shaft A engages one of the stop pins K on the worm, the worm is rotated directly. It, in turn, rotates worm wheel B and pinion C, which, in turn, drives rack D on the table lift. This is the coarse adjustment. But a reversal of the knob disengages the coarse feed and moves the worm gear (S) along the shaft a short distance through a mechanism consisting of an inclined plane and ball. This causes slight rotation of the worm wheel (B) and pinion (C), so movement of rack D is correspondingly limited. This fine feed can be continued, or reversed, within the limits of stops (K).



Achieving fine focus control on high-resolution cameras usually meant that an expensive and intricate gear system must be built. IBM's Research Laboratory, Kingston, NY, designed a simply, low-cost mechanism that can adjust a camera lens to within 10 millionths of an inch.

The ingenious system, called a linear micron-positioner, is based on the differential circumference of connected concentric cylinders of unequal diameters. When flexible bands, in this case shim stock, are fixed to these cylinders, the difference in takeup between the bands occurs when the cylinders are rotated. It is proportional to the differential circumference. If the bands connected to one cylinder are referenced to a fixed frame, and the bands connected to the other cylinder are referenced to a movable member, rotation of the cylinder on the fixed frame will result in relative motion of the movable member equal to the difference in band takeup.

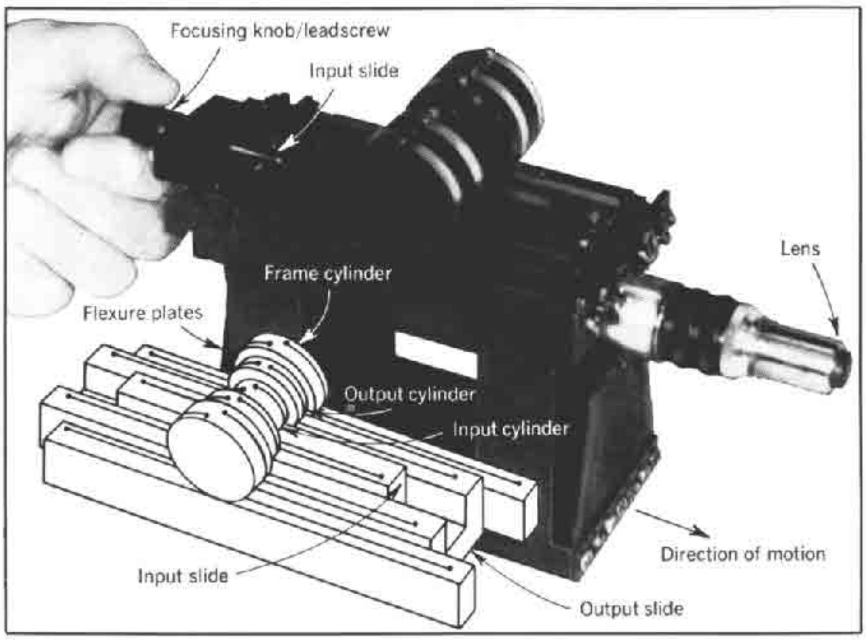
Three cylinders. IBM's mechanism consists of three interconnected, different-sized cylinders to provide reduced displacement. A small input cylinder and the focusing knob are attached to a lead screw.

When the lead screw is turned, the movement between the sliding and fixed frames is very small. When the mechanism was demonstrated, it was necessary to use cylinders whose diameter difference was large enough to show the relative forward motion.

Temperature effects are negligible because all the basic elements, except the leadscrew, have opposing forces. The only element that introduces friction (and is subject to wear in the device) is the leadscrew; the friction here provides a holding force.

Theoretically there is no limit to the reduction ratio obtainable in applying the principle of different sized cylinders. IBM explored the possibility of making a linear positioner with a 10,000:1 ratio.

Applications. In addition to its use in optical systems, the same mechanical principle could be applied to obtain precise adjustments to the axis of an X-Y measurement table; or to position electronic components during manufacture.



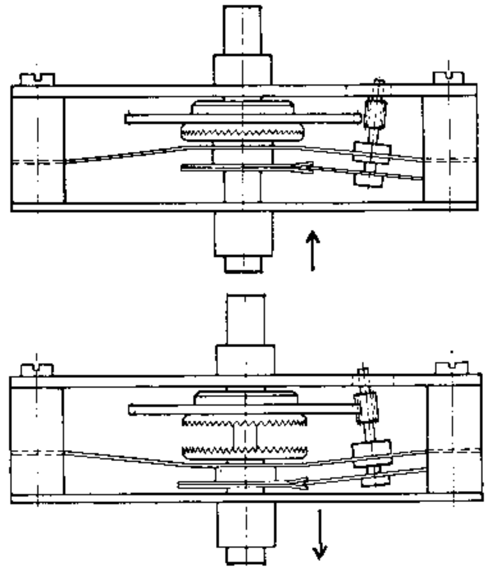
A linear micron-positioner, based on series of connected cylinders of unequal diameters, can adjust a camera lens to within 10 millionths of an inch.

RATCHET-TOOTH SPEED-CHANGE DRIVE

An in-line shaft drive, with reduction ratios of 1:1 and 1:16 or 1:28, combined in a single element, was designed by Telefunken of Germany. It consists basically of friction wheels that drip each other elastically.

Crown wheel with a gear ratio of 1:1 provide the coarse adjustment, and friction spur gearing, with a ratio of 1:16 or 1:28, provides the fine or vernier adjustment.

A spring (see diagram) applies pressure to the fine-adjustment pinion, preventing backlash while the coarse adjustment is in use. It uncouples the coarse adjustment when the vernier is brought into play by forward movement of the front shaft. The spring also ensures that the front shaft is always in gear.



TWINWORM GEAR DRIVE

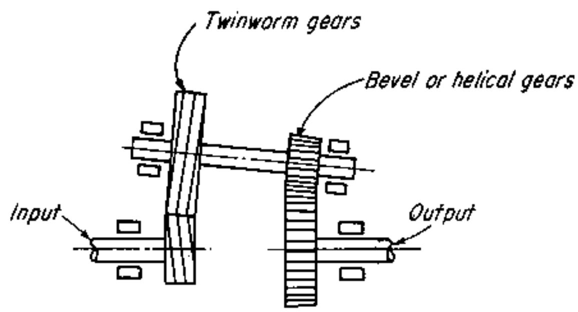
The term “self-locking” as applied to gear systems denotes a drive that gives the input gear freedom to rotate the output gear in either direction. But the output gear locks with the input when an outside torque attempts to rotate the output in either direction. This characteristic is often sought by designers who want to be sure that loads on the output side of the system cannot affect the position of the gears. Worm gears are one of the few gear systems that can be made self-locking, but at the expense of efficiency. It seldom exceeds 40% when the gears are self-locking.

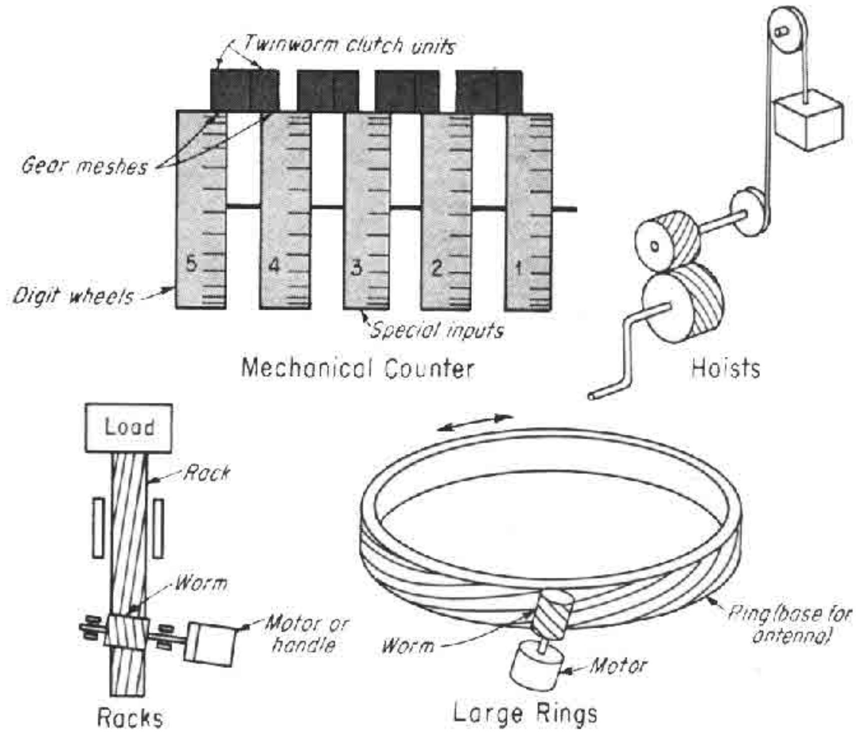
An Israeli engineer, B. Popper, invented a simple dual-worm gear system that not only provided self-locking with over 90% efficiency, but exhibited a phenomenon which the inventor calls “deceleration-locking.”

The “Twinworm” drive has been employed in Israel-designed counters and computers for years with marked success.

The Twinworm drive is simply constructed. Two threaded rods, or “worm” screws, are meshed together. Each worm is wound in a different direction and has a different pitch angle. For proper mesh, the worm axes are not parallel, but slightly skewed. (If both worms had the same pitch angle, a normal, reversible drive would result—similar to helical gears.) But y selecting proper, and different, pitch angles, the drive will exhibit either self-locking, or a combination of self-locking and deceleration-locking characteristics, as desired. Deceleration-locking is a completely new property best described in this way.

When the input gear decelerates (for example, when the power source is shut off, or when an outside force is applied to the output gear in a direction that tends to help the output gear),





the entire transmission immediately locks up and comes to an abrupt stop, moderated only by any elastic "stretch" in the system.

Almost any type of thread will work with the new drive—standard threads, 60º screw threads, Acme threads, or any arbitrary shallow-profile thread. Hence, the worms can be manufactured on standard machine-shop equipment.

JOBS FOR THE NEW DRIVE

Applications for Twinworm can be divided into two groups:

(1) Those employing self-locking characteristics to prevent the load from affecting the system.

(2) Those employing deceleration-locking characteristics to brake the system to an abrupt stop if the input decelerates.

Self-locking occurs as soon as tan ϕ_1 is equal to or smaller than μ, or when

$$\tan\phi_1 = \frac{\mu}{S_1}$$

Angles ϕ_1 and ϕ_2 represent the respective pitch angles of the two worms, and $\phi_2 - \phi_1$ is the angle between the two worm shafts (angle of misalignment). Angle ϕ_1 is quite small (usually in the order of 2º to 5º).

Here, S_1 represents a "safety factor" (selected by the designer). It must be somewhat greater than one to make sure that self-locking is maintained, even if μ should fall below an assumed value. Neither ϕ_2 nor the angle $(\phi_2 - \phi_1)$ affects the self-locking characteristic.

Deceleration-locking occurs as soon as tan ϕ_2 is also equal to or smaller than μ; or, if a second safety factor S_2 is employed (where $S_2 > 1$), when

$$\tan\phi_2 = \frac{\mu}{S_2}$$

For the equations to hold true, ϕ_2 must always be made greater than ϕ_1. Also, μ refers to the idealized case where the worm threads are square. If the threads are inclined (as with Acme-threads or V-threads) then a modified value of μ must be employed, where

$$\mu_{modified} = \frac{\mu_{true}}{\cos\theta}$$

A relationship between the input and output forces during rotation is:

$$\frac{P_1}{P_2} = \frac{\sin\phi_1 + \mu\cos\phi_1}{\sin\phi_2 + \mu\cos\phi_2}$$

Efficiency is determined from the equation:

$$\eta = \frac{1 + \mu/\tan\phi_2}{1 + \mu/\tan\phi_1}$$

COMPLIANT GEARING FOR REDUNDANT TORQUE DRIVE

Elastomeric bearings make torque loads more nearly equal.
Lewis Research Center, Cleveland, Ohio

A set of elastomeric bearings constitutes a springy coupling between a spur gear and a drive shaft. The gear, bearings, and shaft are parts of a split-drive (redundant) mechanical transmission, and the compliance of the coupling helps to distribute torque nearly equally along the load paths of the split drive. Compliance is necessary because without it, even slight deviations in the dimensions of the redundant gears can cause grossly unequal sharing of loads. Indeed, in the absence of compliant coupling, the gears along one load path can assume the entire load while those along another load path can freewheel. Thus, the advantage of reduced loads on gear teeth is lost.

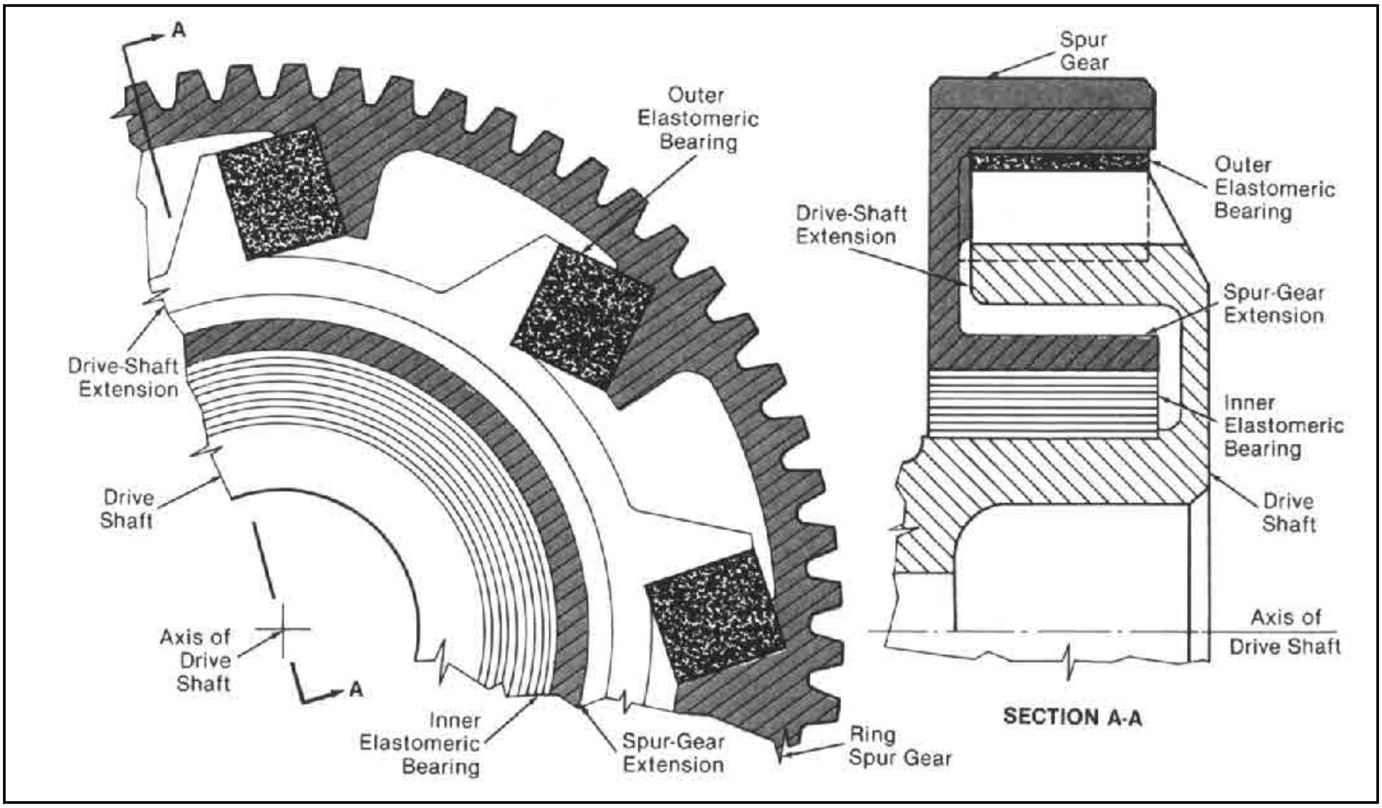
The figure illustrates one version of the shaft/bearing/gear assembly. An inner, concentric elastomeric bearing lies between a central drive shaft and an extension of a ring spur gear. A set of padlike outer elastomeric bearings joins outward protrusions on an extension of the drive shaft with facing inward protrusions on the ring spur gear.

The inner elastomeric bearing has high radial stiffness and low circumferential stiffness. This bearing centers the ring spur gear on the axis of the drive shaft and provides compliance in a circumferential direction. In a representative design of a redundant helicopter transmission, it should be at least 0.5 in. (1.27 cm) thick so that it transmits little torque.

The outer elastomeric bearings, in contrast, have low radial stiffness and high circumferential stiffness. They thus transmit torque effectively between facing protrusions. Nevertheless, they are sufficiently compliant circumferentially to accommodate the desired amount of circumferential displacement [up to 1⁄16 in. (1.6 mm) in the helicopter transmission application].

The process of assembling the compliant gearing begins with the pressing of the inner elastomeric bearing onto the drive shaft. Then, with the help of an alignment tool, the ring spur gear is pressed onto the inner elastomeric bearing. The outer elastomeric bearings are ground to fit the spaces between the protrusions and bonded in place on the protrusions. As an alternative to bonding, the entire assembly can be potted in a soft matrix that holds the outer bearings in place but allows rotation with little restraint.

This work was done by C. Isabelle and J. Kish of United Technologies Corp. for **Lewis Research Center.**



Elastomeric Bearings couple a drive shaft with a ring spur gear. The inner elastomeric bearing is radially stiff and circumferentially compliant, while the outer elastomeric bearings are circumferentially stiff and radially compliant. The combination accommodates minor variations in the dimensions and placements of gears, shafts, and other components.

LIGHTER, MORE-EFFICIENT HELICOPTER TRANSMISSIONS

Redundant gearing transmits torque through an angle or angles.
Lewis Research Center, Cleveland, Ohio

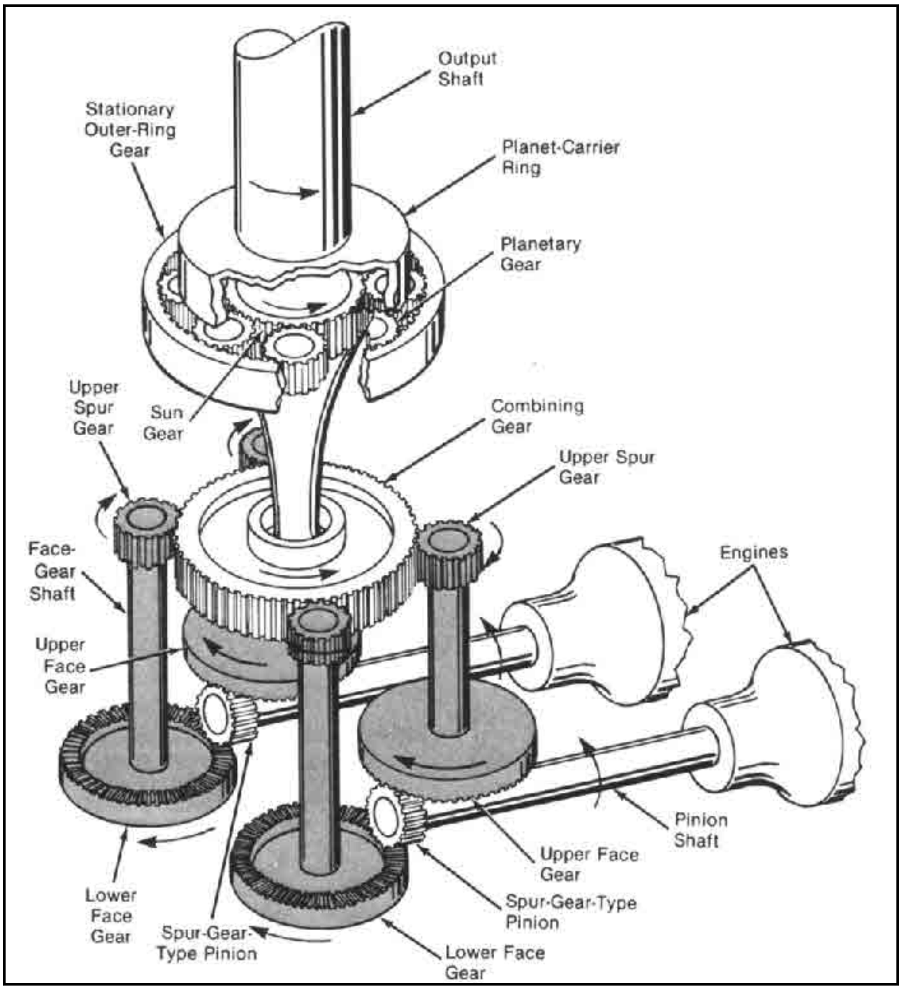
An improved gear system intended primarily for use in a helicopter transmits torque from the horizontal or nearly horizontal shafts of two engines to the vertical output shaft that supports the rotor. The system apportions torques equally along multiple, redundant drive paths, thereby reducing the stresses on individual gear teeth, and it enables one engine to continue to turn the rotor when the other engine fails. The underlying design concept could also be applied to couple two airplane engines to a set of propellers in such a way that both propellers turn as long as at least one engine operates.

The system exploits the special advantages of the geometry of the meshing of a spur-gear-type pinion with a face gear. In comparison with other gear geometries that have been used in helicopter transmissions, this one is much more forgiving of (1) errors in manufacturing and alignment and (2) thermal and vibrational changes in the sizes and positions of the meshing components. One of the benefits is a reduction of gear-tooth-contact noise and vibration. Another benefit is the possibility of achieving a high (> 4) speed-reduction ratio in a single, efficient mesh, and the consequent possibility of reducing the number of parts, the size, the cost, and the weight of the gear system. Of course, the reduction of the number of parts confers yet another benefit by increasing the reliability of the system.

The system is shown schematically in the figure. The output of each engine is coupled by a pinion shaft to a spur-gear-type pinion. Each pinion engages an upper and a lower face gear, and each face gear is coupled by a face-gear shaft to an upper spur gear. The upper spur gears feed torque into a large combining gear. The pinion end of each pinion shaft is lightly spring-loaded in a nominal lateral position and is free to shift laterally through a small distance to take up slack, compensate for misalignments, and apportion torques equally to the two face gears with which it is engaged.

The combining gear is splined to a shaft that flares outwardly to a sun gear. The sun gear operates in conjunction with planetary gears and a stationary outer ring gear. The torque is coupled from the sun gear through the planetary gears to the planet-carrier ring, which is mounted on the output shaft.

This work was done by Robert B. Bossler, Jr., of Lucas Western, Inc., for **Lewis Research Center.**



Torque from each engine is split and transmitted to the combining gear along two redundant paths. Should one engine fail, the other engine could still turn the output shaft.

WORM GEAR WITH HYDROSTATIC ENGAGEMENT

Friction would be reduced greatly.
Lewis Research Center, Cleveland, Ohio

In a proposed worm-gear transmission, oil would be pumped at high pressure through the meshes between the teeth of the gear and the worm coil (see Figure 1). The pressure in the oil would separate the meshing surfaces slightly, and the oil would reduce the friction between these surfaces. Each of the separating forces in the several meshes would contribute to the torque on the gear and to an axial force on the worm. To counteract this axial force and to reduce the friction that it would otherwise cause, oil would also be pumped under pressure into a counterforce hydrostatic bearing at one end of the worm shaft.

This type of worm-gear transmission was conceived for use in the drive train between the gas-turbine engine and the rotor of a helicopter and might be useful in other applications in which weight is critical. Worm gear is attractive for such weight-critical applications because (1) it can transmit torque from a horizontal engine (or other input) shaft to a vertical rotor (or other perpendicular output) shaft, reducing the speed by the desired ratio in one stage, and (2) in principle, a one-stage design can be implemented in a gearbox that weighs less than does a conventional helicopter gearbox.

Heretofore, the high sliding friction between the worm coils and the gear teeth of worm-gear transmissions has reduced efficiency so much that such transmissions could not be used in helicopters. The efficiency of the proposed worm-gear transmission with hydrostatic engagement would depend partly on the remaining friction in the hydrostatic meshes and on the power required to pump the oil. Preliminary calculations show that the efficiency of the proposed transmission could be the same as that of a conventional helicopter gear train.

Figure 2 shows an apparatus that is being used to gather experimental data pertaining to the efficiency of a worm gear with hydrostatic engagement. Two stationary disk sectors with oil pockets represent the gear teeth and are installed in a caliper frame. A disk that represents the worm coil is placed between the disk sectors in the caliper and is rotated rapidly by a motor and gearbox. Oil is pumped at high pressure through the clearances between the rotating disk and the stationary disk sectors. The apparatus is instrumented to measure the frictional force of meshing and the load force.

The stationary disk sectors can be installed with various clearances and at various angles to the rotating disk. The stationary disk sectors can be made in various shapes and with oil pockets at various positions. A flowmeter and pressure gauge will measure the pump power. Oils of various viscosities can be used. The results of the tests are expected to show the experimental dependences of the efficiency of transmission on these factors.

It has been estimated that future research and development will make it possible to make worm-gear helicopter transmission that weigh half as much as conventional helicopter transmissions do. In addition, the new hydrostatic meshes would offer longer service life and less noise. It might even be possible to make the meshing worms and gears, or at least parts of them, out of such lightweight materials as titanium, aluminum, and composites.

This work was done by Lev. I. Chalko of the U.S. Army Propulsion Directorate (AVSCOM) for **Lewis Research Center.**

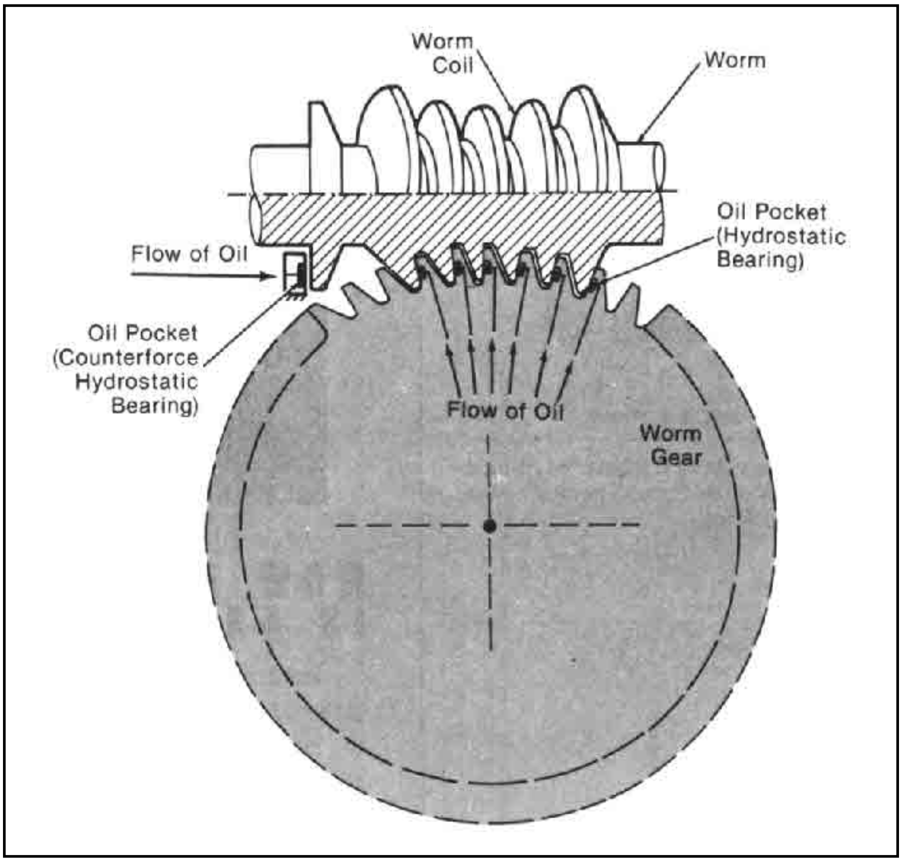


Fig. 1 Oil would be injected at high pressure to reduce friction in critical areas of contact.

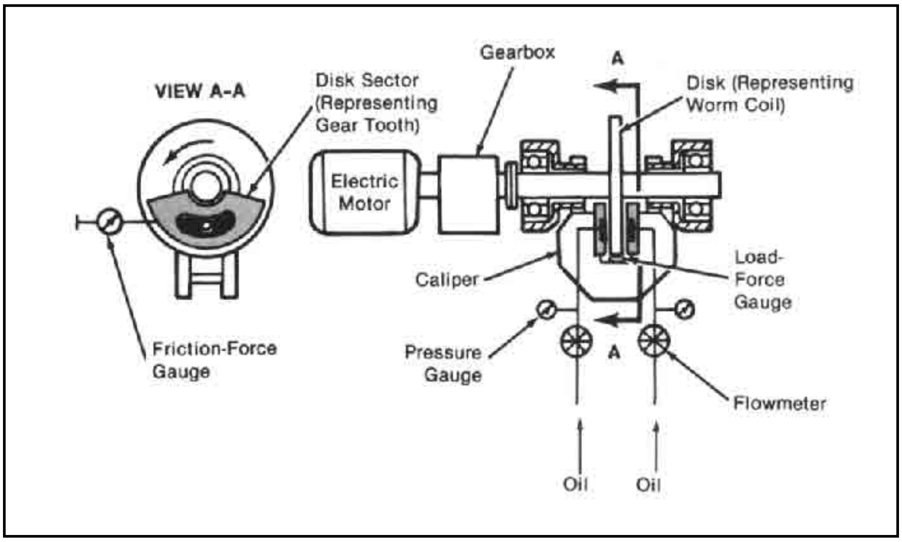


Fig. 2 This test apparatus simulates and measures some of the loading conditions of the proposed worm gear with hydrostatic engagement. The test data will be used to design efficient worm-gear transmissions.

STRADDLE DESIGN OF SPIRAL BEVEL AND HYPOID GEARS

Lengths and radii of shafts can be chosen to prevent undercutting.
Lewis Research Center, Cleveland, Ohio

A computer-assisted method of analysis of straddle designs for spiral bevel and hypoid gears helps to prevent undercutting of gear shafts during cutting of the gear teeth. Figure 1 illustrates a spiral bevel gear or straddle design, in which the shaft extends from both ends of the toothed surface to provide double bearing support. One major problem in such a design is to choose the length and radius of the shaft at the narrow end (equivalently, the radial coordinate r and axial coordinate u) such that the head cutter that generates the gear teeth does not collide with, and thereby undercut, the shaft.

The analytical method and computer program are based on the equations for the surface traced out by the motion of the head cutter, the equation for the cylindrical surface of the shaft, and the equations that express the relationships among the coordinate systems fixed to the various components of the gear-cutting machine tool and to the gear. The location of a collision between the shaft and the cutter is defined as the vector that simultaneously satisfies the equations for head-cutter-traced and shaft surfaces. The solution of these equations yields the u and r coordinates of the point of collision.

Given input parameters in the form of the basic machine-tool settings for cutting the gear, the computer program finds numerical values of r and u at a representative large number of points along the path of the cutter. These computations yield a family of closed curves (see Fig. 2) that are the loci of collision points. The region below the curves is free of collisions: thus, it contains the values of r and u that can be chosen by the designer to avoid collisions between the shaft and the head cutter.

This work was done by Robert F. Handschuh of the U.S. Army Aviation Systems Command; Faydor L. Litvin, Chihping Kuan, and Jonathan Kieffer of the University of Illinois at Chicago; and Robert Bossler of Lucas Western, Inc., for **Lewis Research Center.**

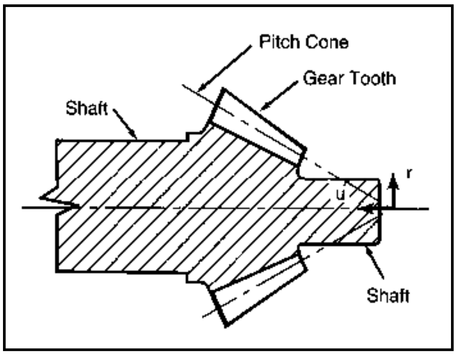


Fig. 1 A straddle-design spiral bevel gear includes two integral shaft extensions. One of these could terminate near or even beyond the apex of the pitch cone.

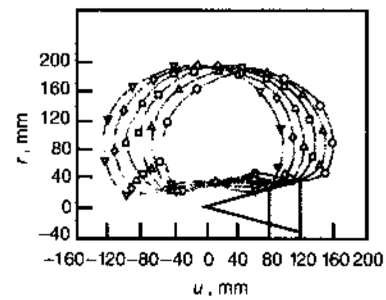


Fig. 2 This family of closed curves applies to a typical hypoid gear. It will help in the selection of the length and radius of the shaft at the narrow end. The region below the curves is free of collisions between the head cutter and the shaft.