CHAPTER 9
COUPLING, CLUTCHING, AND BRAKING DEVICES

COUPLING OF PARALLEL SHAFTS

Fig. 1 One method of coupling shafts makes use of gears that can replace chains, pulleys, and friction drives. Its major limitation is the need for adequate center distance. However, an idler can be used for close centers, as shown. This can be a plain pinion or an internal gear. Transmission is at a constant velocity and there is axial freedom.

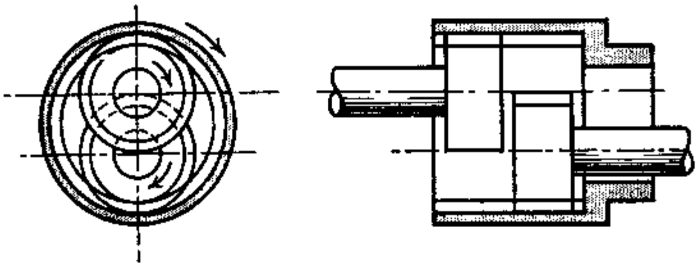


Fig. 2 This coupling consists of two universal joints and a short shaft. Velocity transmission is constant between the input and output shafts if the shafts remain parallel and if the end yokes are arranged symmetrically. The velocity of the central shaft fluctuates during rotation, but high speed and wide angles can cause vibration. The shaft offset can be varied, but axial freedom requires that one shaft be spline mounted.

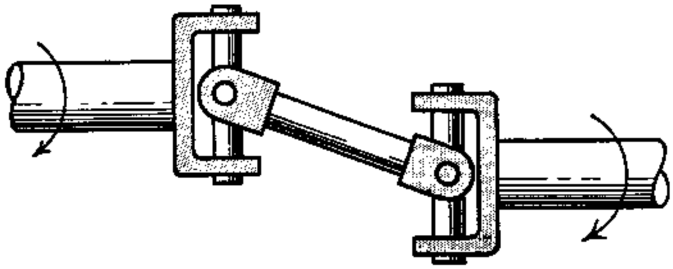


Fig. 3 This crossed-axis yoke coupling is a variation of the mechanism shown in Fig. 2. Each shaft has a yoke connected so that it can slide along the arms of a rigid cross member. Transmission is at a constant velocity, but the shafts must remain parallel, although the offset can vary. There is no axial freedom. The central cross member describes a circle and is thus subjected to centrifugal loads.

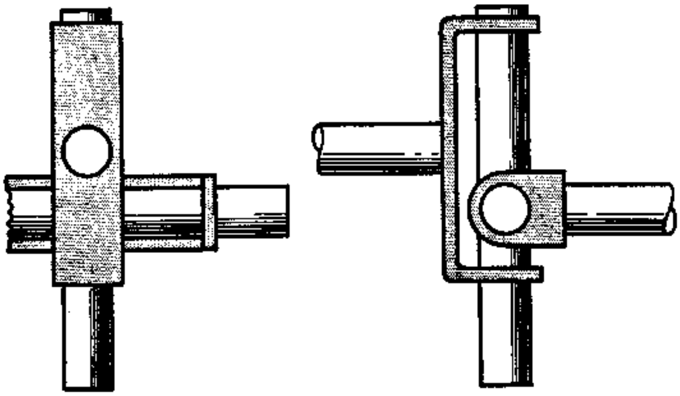
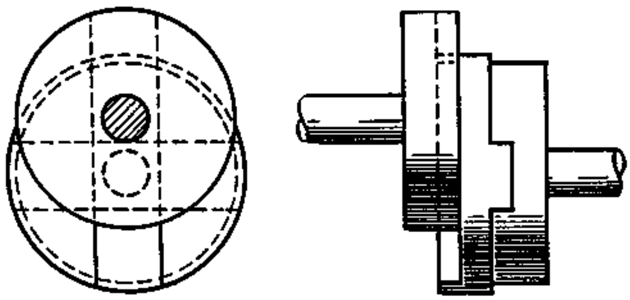


Fig. 4 This Oldham coupling provides motion at a constant velocity as its central member describes a circle. The shaft offset can vary, but the shafts must remain parallel. A small amount of axial freedom is possible. A tilt in the central member can occur because of the offset of the slots. This can be eliminated by enlarging its diameter and milling the slots in the same transverse plane.



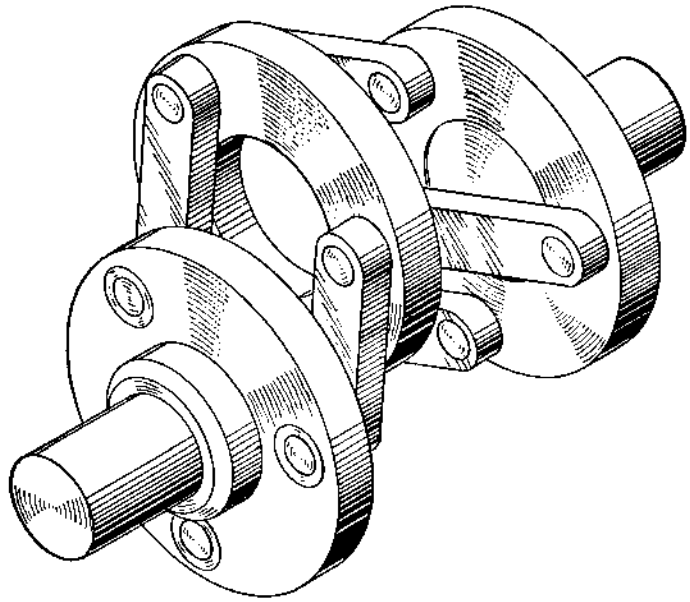
NOVEL LINKAGE COUPLES OFFSET SHAFTS

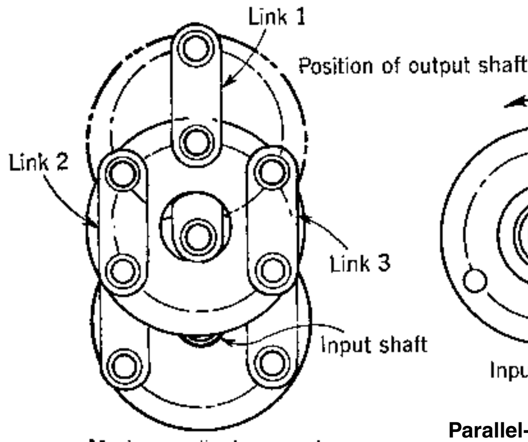
An unorthodox yet remarkably simple arrangement of links and disks forms the basis of a versatile parallel-shaft coupling. This coupling—essentially three disks rotating in unison and interconnected in series by six links (se drawing)—can adapt to wide variations in axial displacement while it is running under load.

Changes in radial displacement do not affect the constant-velocity relationship between the input and output shafts, nor do they affect initial radial reaction forces that might cause imbalance in the system. Those features open up unusual applications for it in automotive, marine, machine-tool, and rolling-mill machinery (see drawings).

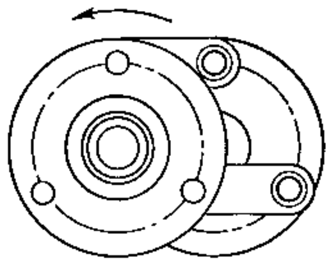
How it works. The inventor of the coupling, Richard Schmidt of Madison, Alabama, said that a similar link arrangement had been known to some German engineers for years. But those engineers were discouraged from applying the theory because they erroneously assumed that the center disk had to be retained by its own bearing. Actually, Schmidt found that the center disk is free to assume its own center of rotation. In operation, all three disks rotate with equal velocity.

The bearing-mounted connections of links to disks are equally spaced at 120º on pitch circles of the same diameter. The distance between shafts can be varied steplessly between zero (when the shafts are in line) and a maximum that is twice the length of the links (see drawings.) There is no phase shift between shafts while the coupling is undulating.



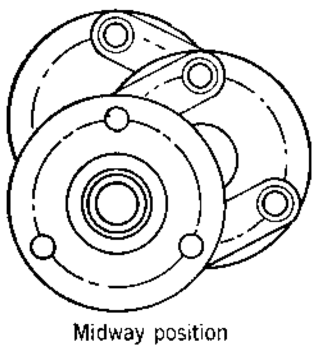


Maximum displacement
(rear view)

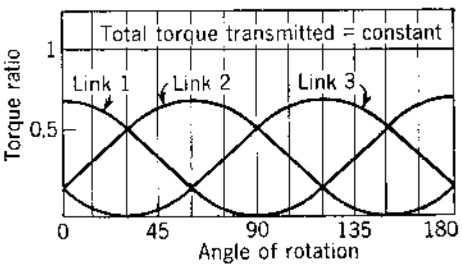


Input and output shafts in line
Zero displacement

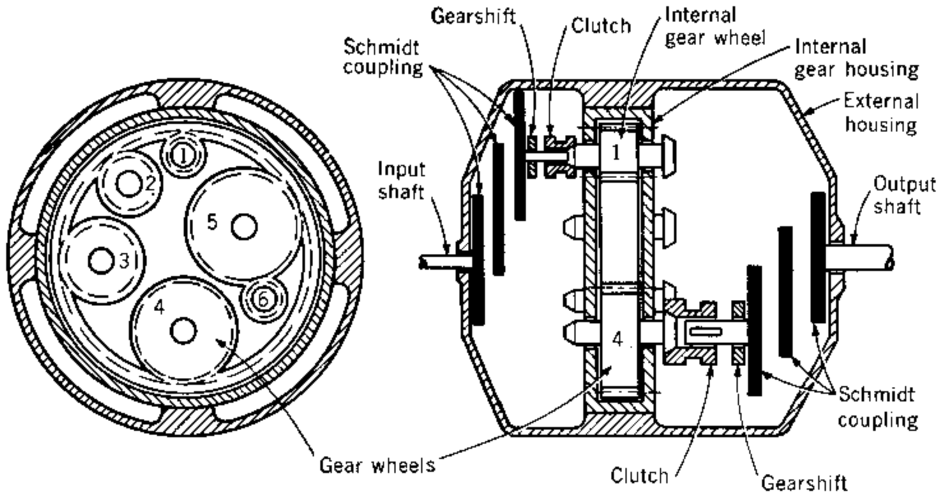
Parallel-link connections between disks (see upper drawing) exactly duplicate the motion between the input and output shafts—the basis of this principle in coupling. The lower diagrams show three positions of the links as one shaft is shifted with respect to the other shaft in the system.

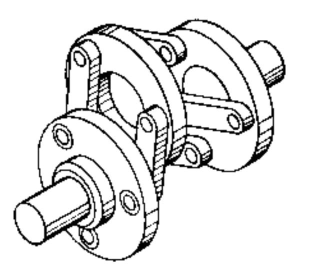


Midway position



Torque transmitted by three links in the group adds up to a constant value, regardless of the angle of rotation.





The parallelgram-type coupling (above) introduces versatility to a gear-transmission design (left) by permitting both the input and output to clutch in directly to any of the six power gears.

DISK-AND-LINK COUPLING SIMPLIFIES TRANSMISSIONS

A unique disk-and-link coupling that can handle large axial displacement between shafts, while the shafts are running under load, has opened up new approaches to transmission design. It was developed by Richard Schmidt of Madison, Alabama.

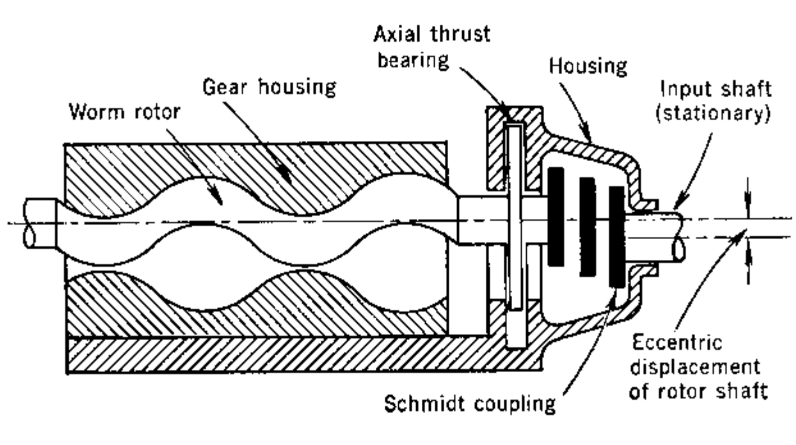
The coupling (drawing, upper right) maintains a constant transmission ratio between input and output shafts while the shafts undergo axial shifts in their relative positions. This permits gear-and-belt transmissions to be designed that need fewer gears and pulleys.

Half as many gears. In the internal-gear transmission shown, a Schmidt coupling on the input side permits the input to be plugged in directly to any one of six gears, all of which are in mesh with the internal gear wheel.

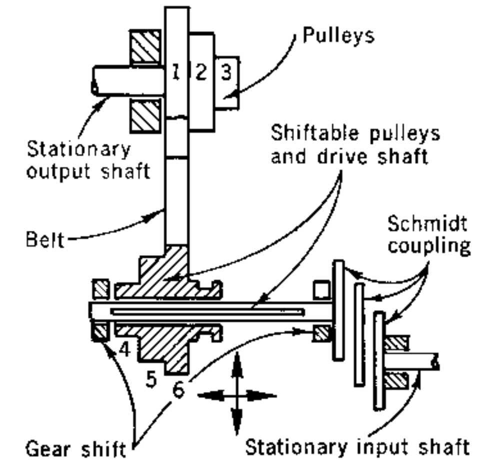
On the output side, after the power flows through the gear wheel, a second Schmidt coupling permits a direct power takeoff from any of the same six gears. Thus, any one of 6×6 minus 5 or 31 different speed ratios can be selected while the unit is running. A more orthodox design would require almost twice as many gears.

Powerful pump. In the worm-type pump (bottom left), as the input shaft rotates clockwise, the worm rotor is forced to roll around the inside of the gear housing, which has a helical groove running from end to end. Thus, the rotor center-line will rotate counterclockwise to produce a powerful pumping action for moving heavy liquids.

In the belt drive (bottom right), the Schmidt coupling permits the belt to be shifted to a different bottom pulley while remaining on the same top pulley. Normally, because of the constant belt length, the top pulley would have to be shifted too, to provide a choice of only three output speeds. With this arrangement, nine different output speeds can be obtained.



The coupling allows a helically-shaped rotor to oscillate for pumping purposes.



This coupling takes up slack when the bottom shifts.

INTERLOCKING SPACE-FRAMES FLEX AS THEY TRANSMIT SHAFT TORQUE

This coupling tolerates unusually high degrees of misalignment, with no variation in the high torque that's being taken from the shaft.

A concept in flexible drive-shaft couplings permits unusually large degrees of misalignment and axial motion during the transmission of high amounts of torque. Moreover, the rotational velocity of the driven member remains constant during transmission at angular misalignments; in other words, cyclic pulsations are not induced as they would be if, say, a universal coupling or a Hooke's joint were employed.

The coupling consists essentially of a series of square space-frames, each bent to provide offsets at the diagonals and each bolted to adjacent members at alternate diagonals. The concept was invented by Robert B. Bossler, Jr. He was granted U.S. Patent No. 3,177,684.

Couplings accommodate the inevitable misalignments between rotating shafts in a driven train. These misalignments are caused by imperfect parts, dimensional variations, temperature changes, and deflections of the supporting structures. The couplings accommodate misalignment either with moving contacts or by flexing.

Most couplings, however, have parts with moving contacts that require lubrication and maintenance. The rubbing parts also absorb power. Moreover, the lubricant and the seals limit the coupling environment and coupling life. Parts wear out, and the coupling can develop a large resistance to movement as the parts deteriorate. Then, too, in many designs, the coupling does not provide true constant velocity.

For flexibility. Bossler studied the various types of couplings n the market and first developed a new one with a moving contact. After exhaustive tests, he became convinced that if there were to be the improvements he wanted, he had to design a coupling that flexed without any sliding or rubbing.

Flexible-coupling behavior, however, is not without design problems. Any flexible coupling can be proportioned with strong, thick, stiff members that easily transmit a design torque and provide the stiffness to operate at design speed. However, misalignment requires flexing of these members. The flexing produces alternating stresses that can limit coupling life. The greater the strength and stiffness of a member, the higher the alternating stress from a given misalignment. Therefore, strength and stiffness provisions that transmit torque at speed will be detrimental to misalignment accommodation capability.

The design problem is to proportion the flexible coupling to accomplish torque transmission and overcome misalignment for the lowest system cost. Bossler looked at a drive shaft, a good example of power transmission—and wondered how he could convert it into one with flexibility.

He began to evolve it by following basic principles. How does a drive shaft transmit torque? By tension and compression. He began paring it down to the important struts that could transmit torque and found that they are curved beams. But a curved beam in tension and compression is not as strong as a straight beam. He ended up with the beams straight in a square space-frame with what might be called a *double helix arrangement.* One helix contained elements in compression; the other helix contained elements in tension.

Flattening the helix. The total number of plates should be an even number to obtain constant velocity characteristics during misalignment. But even with an odd number, the cyclic speed variations are minute, not nearly the magnitude of those in a Hooke's joint.

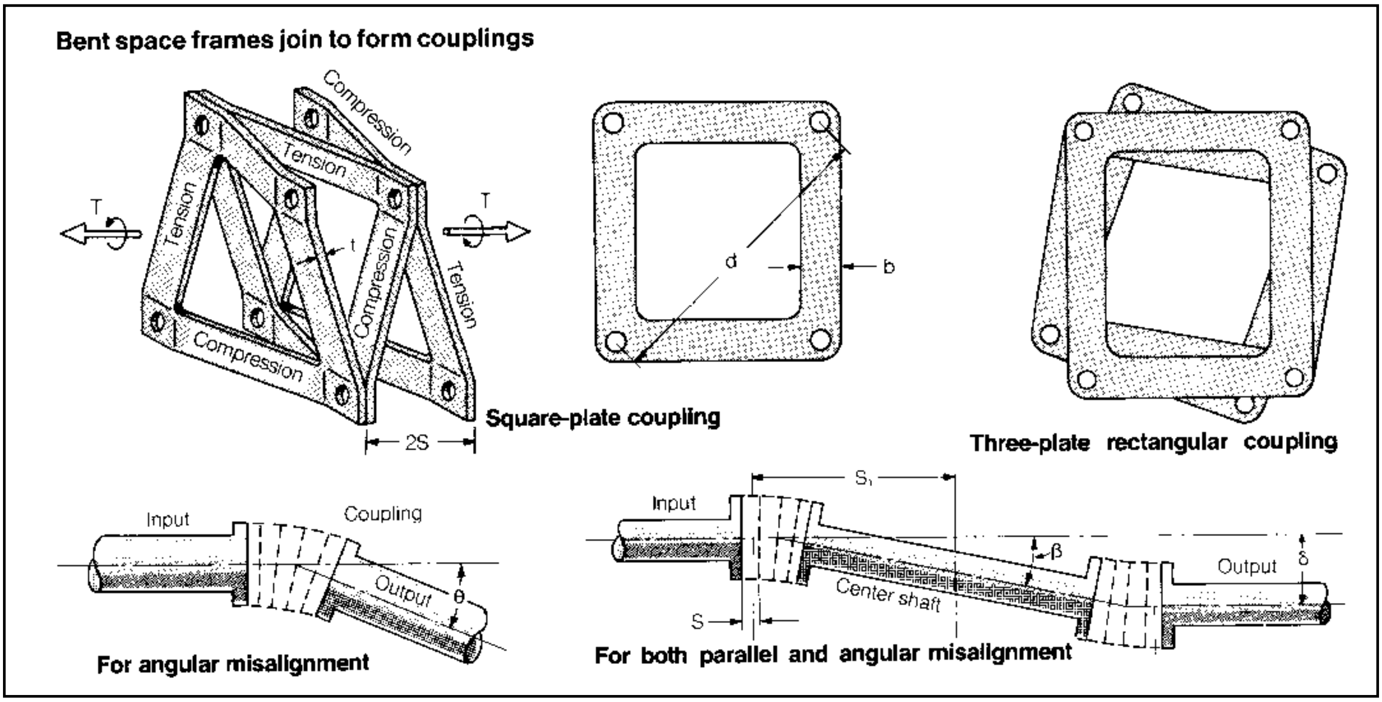
Although the analysis and resulting equations developed by Bossler are based on a square-shaped unit, he concluded that the perfect square is not the ideal for the coupling, because of the position of the mounting holes. The flatter the helix—in other words the smaller the distance S—the more misalignment the coupling will tolerate.

Hence, Bossler began making the space-frames slightly rectangular instead of square. In this design, the bolt-heads that fasten the plates together are offset from adjoining pairs, providing enough clearance for the design of a "flatter" helix. The difference in stresses between a coupling with square-shaped plates and one with slightly rectangular plates is so insignificant that the square-shape equations can be employed with confidence.

Design equations. By making a few key assumptions and approximations, Bossler boiled the complex analytical relationships down to a series of straightforward design equations and charts. The derivation of the equations and the resulting verification from tests are given in the NASA report *The Bossler Coupling,* CR-1241.

Torque capacity. The ultimate torque capacity of the coupling before buckling that might occur in one of the space-frame struts under compression is given by Eq. 1. The designer usually knows or establishes the maximum continuous torque that the coupling must transmit. Then he must allow for possible shock loads and overloads. Thus, the clutch should be designed to have an ultimate torque capacity that is at least twice as much, and perhaps three times as much, as the expected continuous torque, according to Bossler.

Induced stress. At first glance, Eq. 1 seems to allow a lot of leeway in selecting the clutch size. The torque capacity is easily boosted, for example, by picking a smaller bolt-circle diameter, d, which



Design equations for the Bossler coupling

Ultimate torque capacity

(1) $$T = 11.62 \frac{Ebt^3}{dn^{0.9}}$$

Maximum stress per degree of misalignment.

(2) $$\sigma_{max} = 0.0276\ Et/L$$

Minimum thickness to meet required torque strength

(3) $$t = 0.4415 \left(\frac{dT}{bE}\right)^{1/3} n^{0.3}$$

Weight of coupling with minimum-thickness plates

(4) $$W = 1.249w \left(\frac{T}{E}\right)^{1/3} d^{4/3}\, b^{2/3}\, n^{1.3}$$

Maximum permissible misalignment

(5) $$\theta_{max} = 54.7 \left[\frac{bd^2}{TE^2}\right]^{1/3} \sigma_c\, n^{0.7}$$

Maximum permissible misalignment (simplified)

(6) $$\theta/d = 10.9 \frac{n^{0.7}}{T^{1/3}}$$

Maximum permissible offset-angle

(7) $$\beta = 54.7 \left[\frac{bd^2}{TE^2}\right]^{1/3} \frac{\sigma_e C}{n^{0.3}}$$

where: $$\sum_{x=1}^{x=n} \left[1 - (x-1)\frac{S}{S_l}\right]^2$$

Maximum permissible offset-angle (simplified)

(8) $$\beta/d = \frac{10.9\ C}{T^{1/3}\, n^{0.3}}$$

Critical speed frequency

(9) $$f = \frac{60}{2\pi}\left(\frac{k}{M}\right)^{1/2}$$

where: $k = \dfrac{24(EI)_e}{(nS)^3}$ and $(EI)_e = 0.886Ebt^3S/L$

List of symbols

- b = Width of an element
- d = Diameter at the bold circle
- E = Modulus of elasticity
- f = First critical speed, rpm
- I = Flatwise moment of inertia of an element = $bt^3/12$
- k = Spring constant for single degree of freedom
- L = Effective length of an element. This concept is required because joint details tend to stiffen the ends of the elements. L = 0.667 d is recommended
- M = Mass of center shaft plus mass of one coupling with fasteners
- n = Number of plates in each coupling
- S = Offset distance by which a plate is out of plane
- t = Thickness of an element
- T = Torque applied to coupling, useful ultimate, usually taken as lowest critical buckling torque
- w = Weight per unit volume
- W = Total weight of plates in a coupling
- $(EI)_e$ = Flexural stiffness, the moment that causes one radian of flexural angle change per unit length of coupling
- β = Equivalent angle change at each coupling during parallel offset misalignment, deg
- ϑ = Total angular misalignment, deg
- σ_c = Characteristic that limits stress for the material: yield stress for static performance, endurance limit stress for fatigue performance

makes the clutch smaller, or by making the plates thicker. But either solution would also make the clutch stiffer, hence would restrict the misalignment permitted before the clutch becomes overstressed. The stress-misalignment relationship is given in Eq. 2, which shows the maximum flat-wise bending stress produced when a plate is misaligned 1° and is then rotated to transmit torque.

Plate thickness. For optimum misalignment capability, the plates should be selected with the least thickness that will provide the required torque strength. To determine the minimum thickness, Bossler found it expedient to rearrange Eq. 1 into the form shown in Eq. 3. The weight of any coupling designed in accordance to the minimum-thickness equation can be determined from Eq. 4.

Maximum misalignment. Angular misalignment occurs when the centerlines of the input and output shafts intersect at some angle—the angle of misalignment. When the characteristic limiting stress is known for the material selected—and for the coupling's dimensions—the maximum allowable angle of misalignment can be computed from Eq. 5.

If this allowance is not satisfactory, the designer might have to juggle the size factors by, say, adding more plates to the unit. To simplify eq. 5, Bossler made some assumptions in the ratio of endurance limit to modulus and in the ratio of *dsb* to obtain Eq. 6.

Parallel offset. This condition exists when the input and output shafts remain parallel but are displaced laterally. As with Eq. 6, Eq. 7 is a performance equation and can be reduced to design curves. Bossler obtained Eq. 8 by making the same assumptions as in the previous case.

Critical speed. Because of the noncircular configurations of the coupling, it is important that the operating speed of the unit be higher than its critical speed. It should not only be higher but also should avoid an integer relationship.

Bossler worked out a handy relationship for critical speed (Eq. 9) that employs a somewhat idealized value for the spring constant *k*.

Bossler also made other recommendations where weight reduction is vital:

- **Size of plates.** Use the largest *d* consistent with envelope and centrifugal force loading. Usually, centrifugal force loading will not be a problem below 300 ft/s tip speed.
- **Number of plates.** Pick the least *n* consistent with the required performance.
- **Thickness of plates.** Select the smallest *t* consistent with the required ultimate torque.
- **Joint details.** Be conservative; use high-strength tension fasteners with high preload. Provide fretting protection. Make element centerlines and bolt centerlines intersect at a point.
- **Offset distance.** Use the smallest *S* consistent with clearance.

OFF-CENTER PINS CANCEL MISALIGNMENT OF SHAFTS

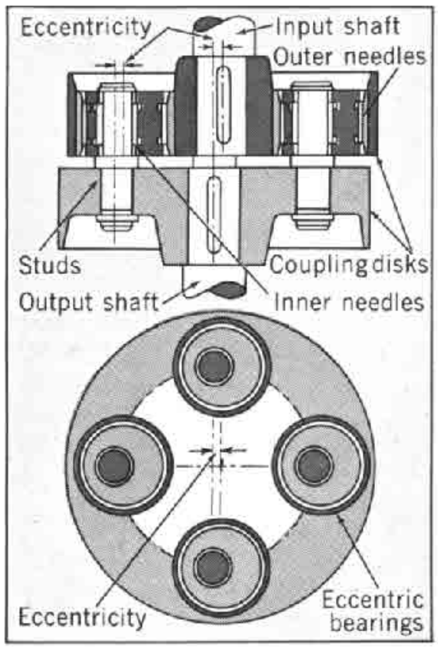
Two Hungarian engineers developed an all-metal coupling (see drawing) for connecting shafts where alignment is not exact—that is, where the degree of misalignment does not exceed the magnitude of the shaft radius.

The coupling is applied to shafts that are being connected for either high-torque or high-speed operation and that must operate at maximum efficiency. Knuckle joints are too expensive, and they have too much play; elastic joints are too vulnerable to the influences of high loads and vibrations.

How it's made. In essence, the coupling consists of two disks, each keyed to a splined shaft. One disk bears four fixed-mounted steel studs at equal spacing; the other disk has large-diameter holes drilled at points facing the studs.

Each large hole is fitted with a bearing that rotates freely inside it on rollers or needles. The bore of the bearings, however, is off-center. The amount of eccentricity of the bearing bore is identical to the deviation of the two shaft center lines.

In operation, input and output shafts can be misaligned, yet they still rotate with the same angular relationship they would have if perfectly aligned.



Eccentrically bored bearings rotate to make up for misalignment between shafts.

HINGED LINKS AND TORSION BUSHINGS GIVE DRIVES A SOFT START

Centrifugal force automatically draws up the linkage legs, while the torsional resistance of the bushings opposes the deflection forces.

A spidery linkage system combined with a rubber torsion bushing system formed a power-transmission coupling. Developed by a British company, Twiflex Couplings Ltd., Twickenham, England, the device (drawing below) provides ultra-soft starting characteristics. In addition to the torsion system, it also depends on centrifugal force to draw up the linkage legs automatically, thus providing additional soft coupling at high speeds to absorb and isolate any torsional vibrations arising from the prime mover.

The TL coupling has been installed to couple marine main engines to gearbox-propeller systems. Here the coupling reduces propeller vibrations to negligible proportions even at high critical speeds. Other applications are also foreseen, including their use in diesel drives, machine tools, and off-the-road construction equipment. The coupling's range is from 100 hp to 4000 rpm to 20,000 hp at 400 rpm.

Articulating links. The key factor in the TL coupling, an improvement over an earlier Twiflex design, is the circular grouping of hinged linkages connecting the driving and driven coupling flanges. The forked or tangential links have resilient precompressed bonded-rubber bushings at the outer flange attachments, while the other pivots ride on bearings.

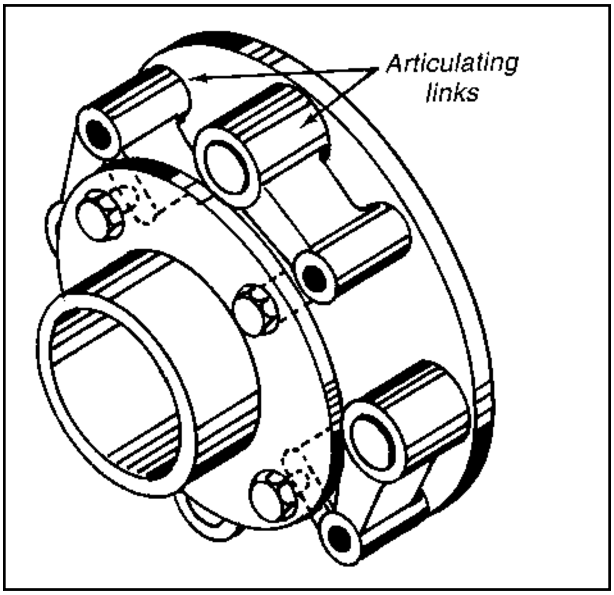
When torque is applied to the coupling, the linkages deflect in a positive or negative direction from the neutral position (drawings, below). Deflection is opposed by the torsional resistance of the rubber bushings at the outer pins. When the coupling is rotating, the masses of the linkage give rise to centrifugal forces that further oppose coupling deflection. Therefore, the working position of the linkages depends both on the applied torque and on the speed of the coupling's rotation.

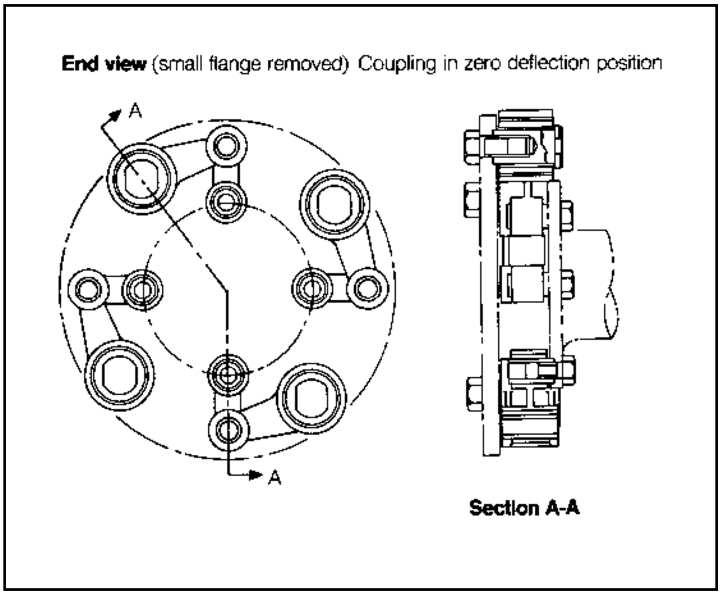
Tests of the coupling's torque/deflection characteristics under load have shown that the torsional stiffness of the coupling increases progressively with speed and with torque when deflected in the positive direction. Although the geometry of the coupling is asymmetrical the torsional characteristics are similar for both directions of drive in the normal working range. Either half of the coupling can act as the driver for either direction of rotation.

The linkage configuration permits the coupling to be tailored to meet the exact stiffness requirements of individual systems or to provide ultra-low torsional stiffness at values substantially softer than other positive-drive couplings. These characteristics enable the Twiflex coupling to perform several tasks:

- It detunes the fundamental mode of torsional vibration in a power-transmission system. The coupling is especially soft at low speeds, which permits complete detuning of the system.
- It decouples the driven machinery from engine-excited torsional vibration. In a typical geared system, the major machine modes driven by the gearboxes are not excited if the ratio of coupling stiffness to transmitted torque is less than about 7:1—a ratio easily provide by the Twiflex coupling.
- It protects the prime mover from impulsive torques generated by driven machinery. Generator short circuits and other causes of impulsive torques are frequently of sufficient duration to cause high response torques in the main shafting.

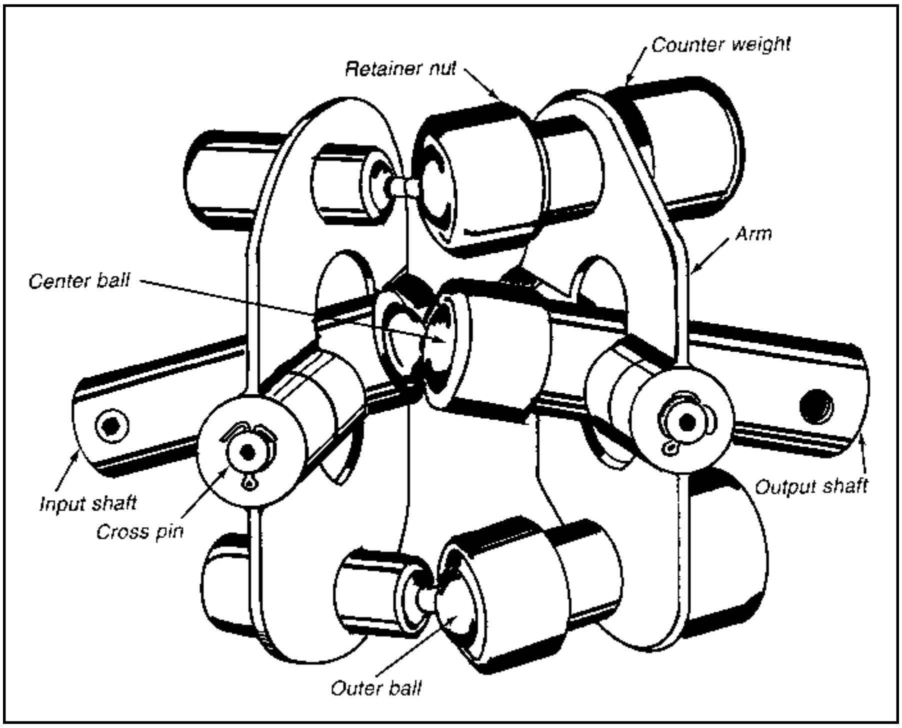
Using the example of the TL 2307G coupling design—which is suitable for 10,000 hp at 525 rpm—the torsional stiffness at working points is largely determined by coupling geometry and is, therefore, affected to a minor extent by the variations in the properties of the rubber bushings. Moreover, the coupling can provide torsional-stiffness values that are accurate within 5.0%.





Articulating links of the new coupling (left) are arranged around the driving flanges. A four-link design (right) can handle torques from a 100-hp prime mover driving at 4000 rpm.

UNIVERSAL JOINT RELAYS POWER 45° AT CONSTANT SPEEDS



A novel arrangement of pivots and ball-socket joints transmits uniform motion.

A universal joint that transmits power at constant speeds through angles up to 45º was designed by Malton Miller of Minnesota.

Models of the true-speed drive that can transmit up to 20 hp have been developed.

It had not been possible to transmit power at constant speeds with only one universal joint. Engineers had to specify an intermediate shaft between two Hooke's joints or use a Rzappa-type joint to get the desired effect.

Ball-and-socket. Basically, the True-Speed joint is a system of ball-and-socket connections with large contact areas (low unit pressure) to transmit torsional forces across the joint. This arrangement minimizes problems when high bearing pressures build up against running surfaces. The low-friction bearings also increase efficiency. The joint is balanced to keep vibration at high speeds to a minimum.

The joint consists of driving and driven halves. Each half has a coupling sleeve at its end of the driveshaft, a pair of driving arms opposite each other and pivoted on a cross pin that extends through the coupling sleeve, and a ball-and-socket coupling at the end of each driving arm.

As the joint rotates, angular flexure in one plane of the joint is accommodated by the swiveling of the all-and-socket couplings and, in the 90º plane, by the oscillation of the driving arms about the transverse pin. As rotation occurs, torsion is transmitted from one half of the joint to the other half through the swiveling ball-and-socket couplings and the oscillating driving arms.

Balancing. Each half of the joint, in effect, rotates about its own center shaft, so each half is considered separate for balancing. The center ball-and-socket coupling serves only to align and secure the intersection point of the two shafts. It does not transmit any forces to the entire drive unit.

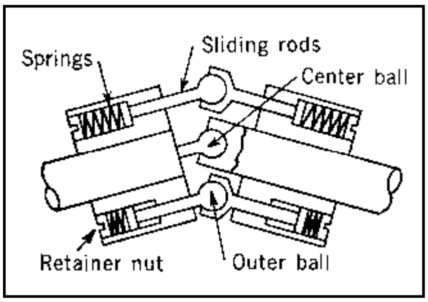
Balancing for rotation is achieved by equalizing the weight of the two driving arms of each half of the joint. Balancing the acceleration forces due to the oscillation of the ball-and-socket couplings, which are offset from their swiveling axes, is achieved by the use of counterweights extending from the opposite side of each driving arm.

The outer ball-and-socket couplings work in two planes of motion, swiveling widely in the plane perpendicular to the main shaft and swiveling slightly about the transverse pin in the plane parallel to the main shaft. In this coupling configuration, the angular displacement of the driving shaft is exactly duplicated in the driven shaft, providing constant rotational velocity and constant torque at all shaft intersection angles.

Bearings. The only bearing parts are the ball-and-socket couplings and the driving arms on the transverse pins. Needle bearings support the driving arms on the transverse pin, which is hardened and ground. A high-pressure grease lubricant coats the bearing surfaces of the ball-and-socket couplings. Under maximum rated loadings of 600 psi on the ball-and-socket surfaces, there is no appreciable heating or power loss due to friction.

Capabilities. Units have been laboratory-tested at all rated angles of drive under dynamometer loadings. Although the first available units were for smaller capacities, a unit designed for 20 hp at 550 rpm, suitable for tractor power take-off drive, has been tested.

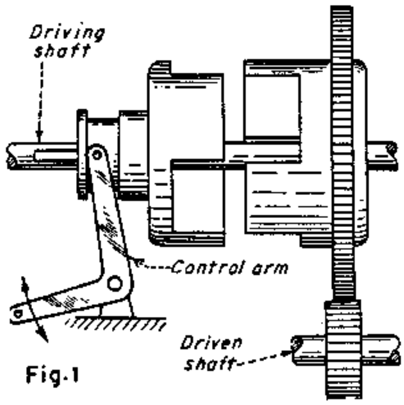
Similar couplings have been designed as pump couplings. But the True-Speed drive differs in that the speed and transfer elements are positive. With the pump coupling, on the other hand, the speed might fluctuate because of spring bounce.

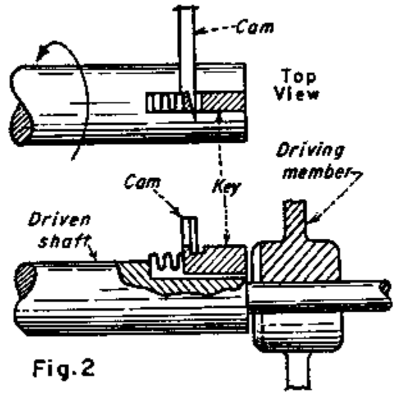


An earlier version for angled shafts required spring-loaded sliding rods.

BASIC MECHANICAL CLUTCHES

Both friction and positive clutches are illustrated here. Figures 1 to 7 show externally controlled clutches, and Figures 8 to 12 show internally controlled clutches which are further divided into overload relief, overriding, and centrifugal versions.





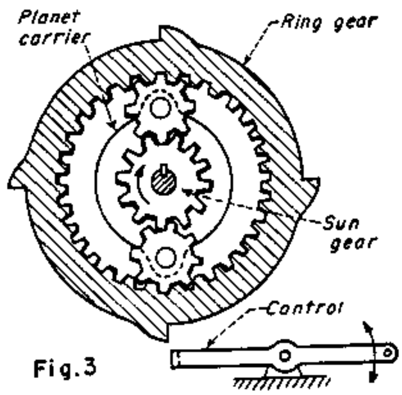
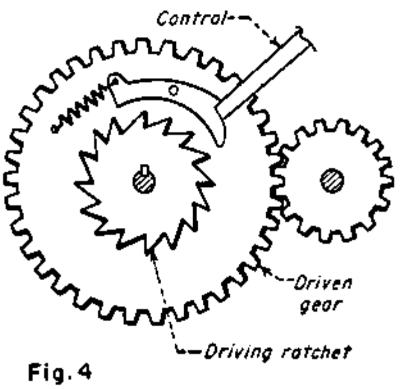
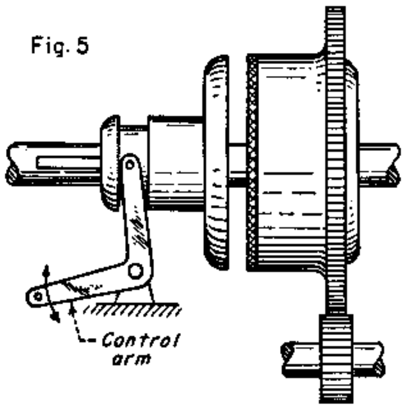


Fig. 1 Jaw Clutch: The left sliding half of this clutch is feathered to the driving shaft while the right half rotates freely. The control arm activates the sliding half to engage or disengage the drive. However, this simple, strong clutch is subject to high shock during engagement and the sliding half exhibits high inertia. Moreover, engagement requires long axial motion.

Fig. 2 Sliding Key Clutch: The driven shaft with a keyway carries the freely rotating member with radial slots along its hub. The sliding key is spring-loaded but is restrained from the engaging slots by the control cam. To engage the clutch, the control cam is raised and the key enters one of the slots. To disengage it, the cam is lowered into the path of the key and the rotation of the driven shaft forces the key out of the slot in the driving member. The step on the control cam limits the axial movement of the key.

Fig. 3 Planetary Transmission Clutch: In the disengaged position shown, the driving sun gear causes the free-wheeling ring gear to idle counter-clockwise while the driven planet carrier remains motionless. If the control arm blocks ring gear motion, a positive clockwise drive to the driven planet carrier is established.





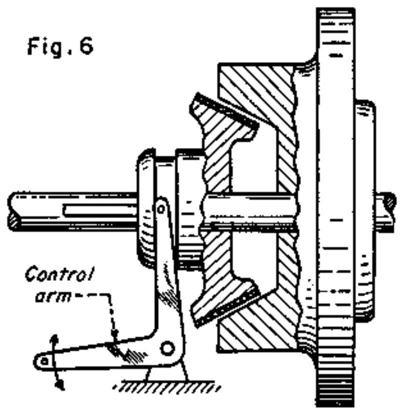
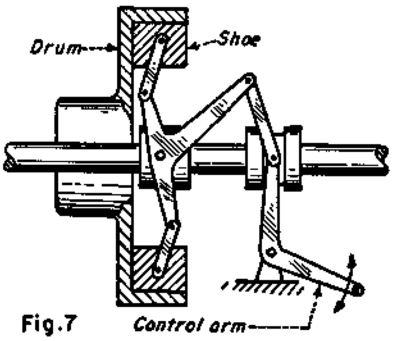
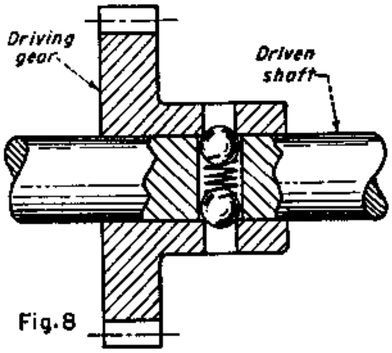


Fig. 4 Pawl and Ratchet Clutch: (External Control) The driving ratchet of this clutch is keyed to the driving shaft, and the pawl is pinned to the driven gear which can rotate freely on the driving shaft. When the control arm is raised, the spring pulls in the pawl to engage the ratchet and drive the gear. To disengage the clutch the control arm is lowered so that driven gear motion will disengage the pawl and stop the driven assembly against the control member.

Fig. 5 Plate Clutch: The plate clutch transmits power through the friction developed between the mating plate faces. The left sliding plate is fitted with a feather key, and the right plate member is free to rotate on the shaft. Clutch torque capacity depends on the axial force exerted by the control half when it engages the sliding half.

Fig. 6 Cone Clutch: The cone clutch, like the plate clutch, requires axial movement for engagement, but less axial force is required because of the increased friction between mating cones. Friction material is usually applied to only one of the mating conical surfaces. The free member is mounted to resist axial thrust.





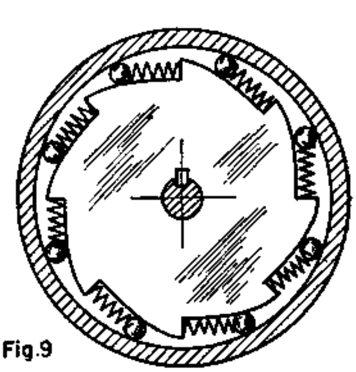
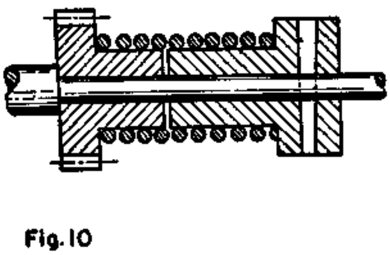
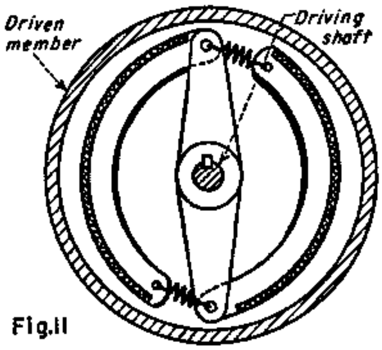


Fig. 7 Expanding Shoe Clutch: This clutch is engaged by the motion of the control arm. It operates linkages that force the friction shoes radially outwards so that they contact the inside surface of the drum.

Fig. 8 Spring and Ball Radial Detent Clutch: This clutch will hold the driving gear and driven gear in a set timing relationship until the torque becomes excessive. At that time the balls will be forced inward against their springs and out of engagement with the holes in the hub. As a result the driving gear will continue rotating while the drive shaft is stationary.

Fig. 9 Cam and Roller Clutch: This over-running clutch is better suited for higher-speed free-wheeling than a pawl-and-ratchet clutch. The inner driving member has cam surfaces on its outer rim that hold light springs that force the rollers to wedge between the cam surfaces and the inner cylindrical face of the driven member. While driving, friction rather than springs force the rollers to wedge tightly between the members to provide positive clockwise drive. The springs ensure fast clutching action. If the driven member should begin to run ahead of the driver, friction will force the rollers out of their tightly wedged positions and the clutch will slip.





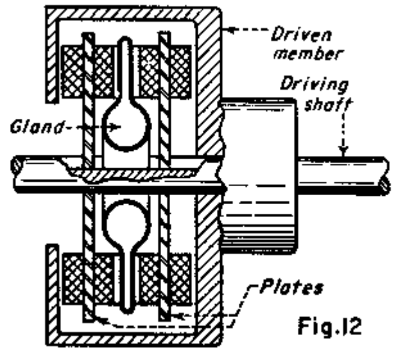


Fig. 10 Wrapped Spring Clutch: This simple unidirectional clutch consists of two rotating hubs connected by a coil spring that is press-fit over both hubs. In the driving direction the spring tightens around the hubs increasing the friction grip, but if driven in the opposite direction the spring unwinds causing the clutch to slip.

Fig. 11 Expanding Shoe Centrifugal Clutch: This clutch performs in a similar manner to the clutch shown in Fig. 7 except that there is no external control. Two friction shoes, attached to the driving member, are held inward by springs until they reach the "clutch-in" speed. At that speed centrifugal force drives the shoes outward into contact with the drum. As the drive shaft rotates faster, pressure between the shoes against the drum increases, thus increasing clutch torque.

Fig. 12 Mercury Gland Clutch: This clutch contains two friction plates and a mercury-filled rubber bladder. At rest, mercury fills a ring-shaped cavity around the shaft, but when rotated at a sufficiently high speed, the mercury is forced outward by centrifugal force. The mercury then spreads the rubber bladder axially, forcing the friction plates into contact with the opposing faces of the housing to drive it.

SPRING-WRAPPED SLIP CLUTCHES

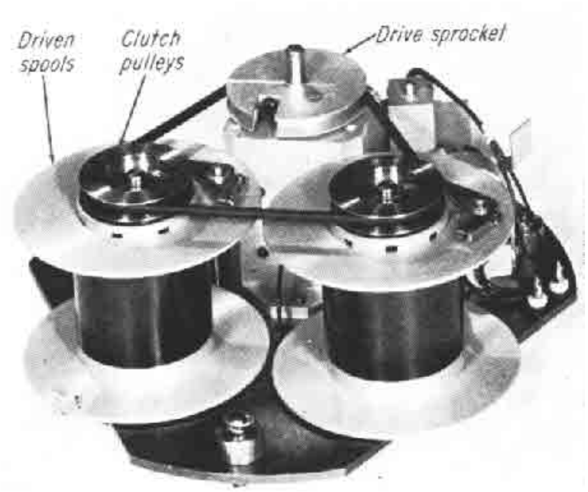


Fig. 1 Two dual-spring clutches are in this tape drive.

The simple spring clutch becomes even more useful when designed to slip at a predetermined torque. Unaffected by temperature extremes or variations in friction, these clutches are simple—they can even be "homemade." Information is provided here on two dual-spring, slip-type clutches. Two of the dual-spring clutches are in the tape drive shown.

Spring clutches are devices for driving a load in one direction and uncoupling it when the output is overdriven or the direction of the input rotation is reversed. A spring clutch was modified to give a predetermined slip in either direction—hence the designation of this type as a "slip clutch." A stepped helical spring was employed to accomplish that modification. Later it was developed further by introducing an intermediate clutch member between two helical springs. This dual-spring innovation was preferred where more output torque accuracy was required.

Most designs employ either a friction-disk clutch or a shoe clutch to obtain a predetermined slip (in which the input drives output without slippage until a certain torque level is reached—then a drag-slippage occurs). But the torque capacity (or slip torque) for friction-disk clutches is the same for both directions of rotation.

By contrast, the stepped-spring slip clutch, pictured on the next page, can be designed to have either the same or different torque capacities for each direction of rotation. Torque levels where slippage occurs are independent of each other, thus providing wide latitude of design.

The element producing slip is the stepped spring. The outside diameter of the large step of the spring is assembled tightly in the bore of the output gear. The inside diameter of the smaller step fits tightly over the shaft. Rotation of the shaft in one direction causes the coils in contact with the shaft to grip tightly, and the coils inside the bore to contract and produce slip. Rotation in the opposite direction reverses the action of the spring parts, and slip is effected on the shaft.

Dual-Spring Slip Clutch

This innovation also permits bi-directional slip and independent torque capacities for the two directions of rotation. It requires two springs, one right-handed and one left-handed, for coupling the input, intermediate and output members. These members are coaxial, with the intermediate and input free to rotate on the output shaft. The rotation of input in one direction causes the spring, which couples the input and intermediate member, to grip tightly. The second spring, which couples the intermediate and output members, is oppositely wound, tends to expand and slip. The rotation in the opposite direction reverses the action of the two springs so that the spring between the input and intermediate members provides the slip. Because this design permits greater independence in the juggling of dimensions, it is preferred where more accurate slip-torque values are required.

Repeatable Performance

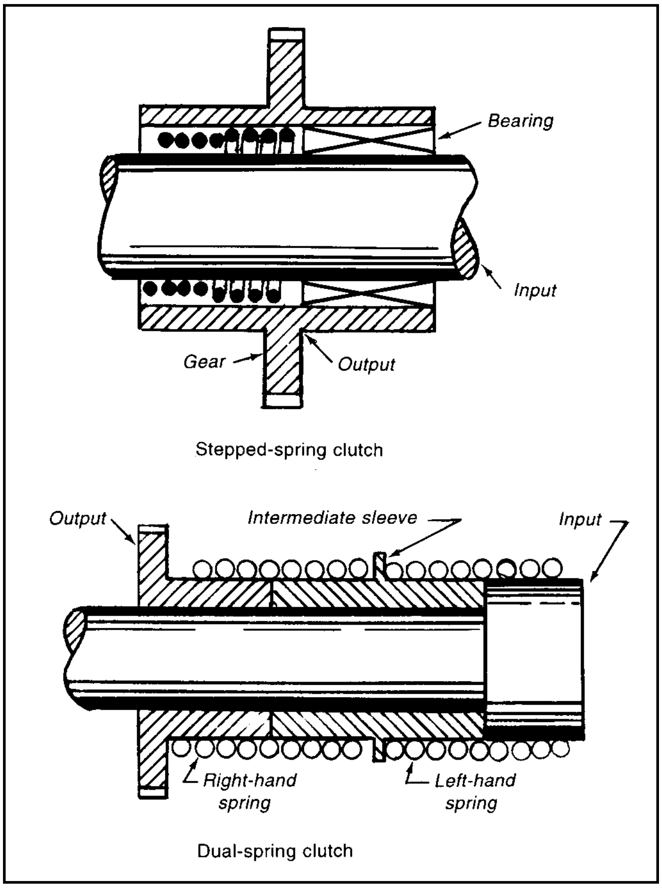
Spring-wrapped slip clutches and brakes have remarkably repeatable slip-torque characteristics which do not change with service temperature. Torque capacity remains constant with or without lubrication, and is unaffected by variations in the coefficient of friction. Thus, break-away torque capacity is equal to the sliding torque capacity. This stability makes it unnecessary to overdesign slip members to obtain reliable operation. These advantages are absent in most slip clutches.

Brake and Clutch Combinations

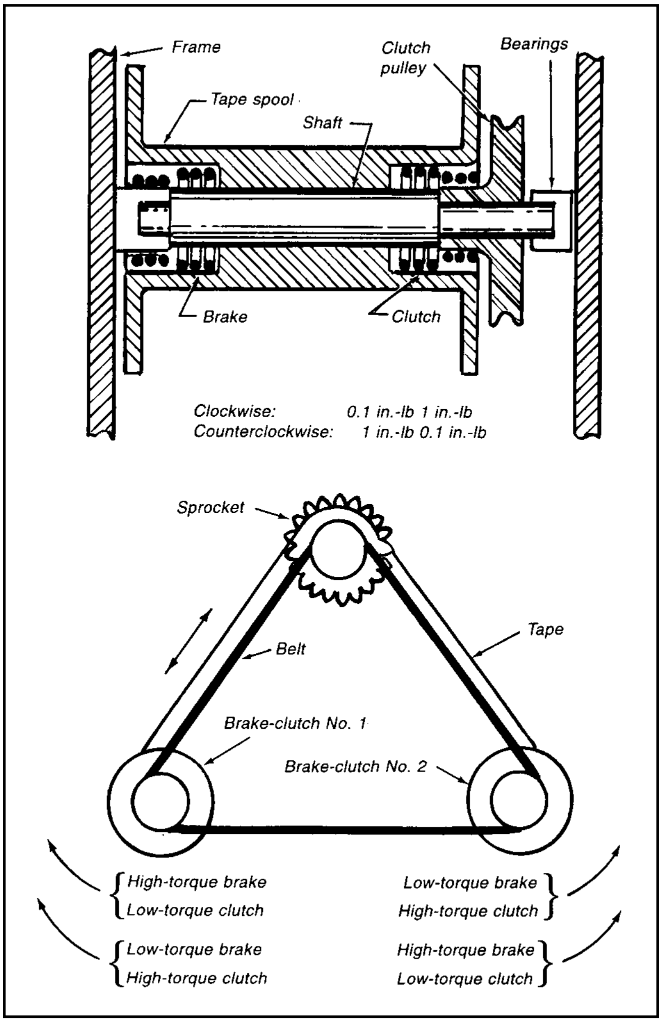
An interesting example of how slip brakes and clutches worked together to maintain proper tension in a tape drive, in either direction of operation, is pictured above and shown schematically on the opposite page. A brake here is simply a slip clutch with one side fastened to the frame of the unit. Stepped-spring clutches and brakes are shown for simplicity although, in the actual drive, dual-spring units were installed.

The sprocket wheel drives both the tape and belt. This allows the linear speed of the tape to be constant (one of the requirements). The angular speed of the spools, however, will vary as they wind or unwind. The task here is to maintain proper tension in the tape at all times and in either direction. This is done with a brake-clutch combination. In a counterclockwise direction, for example, the brake might become a "low-torque brake" that resists with a 0.1 in.-lb. Torque. The clutch in this direction is a "high-torque clutch"—it will provide a 1-in.-lb torque. Thus, the clutch overrides the brake with a net torque of 0.9 in.-lb.

When the drive is reversed, the same brake might now act as a high-torque brake, resisting with a 1 in.-lb torque, while the clutch acts as a low-torque clutch, resisting with 0.1 in.-lb. Thus, in the first direction the clutch drives the spool, in the other direction, the brake overcomes the clutch and provides a steady resist-



These two modifications of spring clutches offer independent slip characteristics in either direction of rotation.



This tape drive requires two slip clutches and two brakes to ensure proper tension for bidirectional rotation. The detail of the spool (above) shows a clutch and brake unit.

ing force to provide tension in the tape. Of course, the clutch also permits the pulley that is driven by the belt to overdrive.

Two brake-clutch units are required. The second unit will provide opposing torque values—as listed in the diagram. The drive necessary to advance the tape only in a clockwise direction would be the slip clutch in unit 2 and the brake in unit 1. Advancing the tape in the other direction calls for use of the clutch in unit 1 and the brake in unit 2.

For all practical purposes, the low torque values in the brakes and clutches can be made negligible by specifying minimum interference between the spring and the bore or shaft. The low torque is amplified in the spring clutch at the level necessary to drive the tensioning torques of the brake and slip clutches.

Action thus produced by the simple arrangement of directional slip clutches and brakes cannot otherwise be duplicated without resorting to more complex designs.

Torque capacities of spring-wrapped slip clutches and brakes with round, rectangular, and square wire are, respectively:

$$T = \frac{\pi E d^4 \delta}{32D^2};\ T = \frac{Ebt^3\delta}{6D^2};\ T = \frac{Et^4\delta}{6D^2}$$

where E = modules of elasticity, psi; d = wire diameter, inches; D = diameter of shaft or bore, inches; ε = diametral interference between spring and shaft, or spring and bore, inches; t = wire thickness, inches; b = width of rectangular wire, inches; and T = slip torque capacity, pound-inches.

Minimum interference moment (on the spring gripping lightly) required to drive the slipping spring is:

$$M = \frac{T}{e^{\mu\theta} - 1}$$

where e = natural logarithmic base (e = 2.716; θ = angle of wrap of spring per shaft, radians, μ = coefficient of friction, M = interference moment between spring and shaft, pound-inches.

Design Example

Required: to design a tape drive similar to the one shown above. The torque requirements for the slip clutches and brakes for the two directions of rotation are:

(1) Slip clutch in normal takeup capacity (active function) is 0.5 to 0.8 in.-lb.

(2) Slip clutch in override direction (passive function) is 0.1 in.-lb (maximum

(3) Brake in normal supply capacity (active function) is 0.7 to 1.0 in.-lb.

(4) Brake in override direction (passive function) is 0.1 in.-lb (maximum).

Assume that the dual-spring design shown previously is to include 0.750-in. drum diameters. Also available is an axial length for each spring, equivalent to 12 coils which are divided equally between the bridged shafts. Assuming round wire, calculate the wire diameter of the springs if 0.025 in. is maximum diametral interference desired for the active functions. For the passive functions use round wire that produces a spring index not more than 25.

Slip clutch, active spring:

$$d = \sqrt[4]{\frac{32D^2T}{\pi E\delta}} = \sqrt[4]{\frac{(32)(0.750)^2(0.8)}{\pi(30\times10^6)(0.025)}} = 0.050 \text{ in.}$$

The minimum diametral interference is (0.025) (0.5)/0.8 = 0.016 in. Consequently, the ID of the spring will vary from 0.725 to 0.734 in.

Slip clutch, passive spring:

$$\text{Wire dia.} = \frac{\text{drum dia.}}{\text{spring index}} = \frac{0.750}{25} = 0.030 \text{ in.}$$

Diametral interference:

$$\delta = \frac{32D^2T}{\pi E d^4} = \frac{(32)(0.750^2)(0.1)}{\pi(30\times10^6)(0.030)^4} = 0.023 \text{ in.}$$

Assuming a minimum coefficient of friction of 0.1, determine the minimum diametral interference for a spring clutch that will drive the maximum slip clutch torque of 0.8 lb-in.

Minimum diametral interference:

$$M = \frac{T}{e^{\mu\theta}-1} = \frac{0.8}{e^{(0.1\pi)(6)}-1}$$

ID of the spring is therefore 0.727 to 0.745 in.

$$\min = 0.023\times\frac{0.019}{0.1} = 0.0044 \text{ in.}$$

Brake springs

By similar computations the wire diameter of the active brake spring is 0.053 in., with an ID that varies from 0.725 and 0.733 in.; wire diameter of the passive brake spring is 0.030 in., with its ID varying from 0.727 to 0.744 in.

CONTROLLED-SLIP CONCEPT ADDS NEW USES FOR SPRING CLUTCHES

A remarkably simple change in spring clutches is solving a persistent problem in tape and film drives—how to keep drag tension on the tape constant, as its spool winds or unwinds. Shaft torque has to be varied directly with the tape diameter so many designers resort to adding electrical control systems, but that calls for additional components; an extra motor makes this an expensive solution. The self-adjusting spring brake (Fig. 1) developed by Joseph Kaplan, Farmingdale, NY, gives a constant drag torque ("slip" torque) that is easily and automatically varied by a simple lever arrangement actuated by the tape spool diameter (Fig. 2). The new brake is also being employed to test the output of motors and solenoids by providing levels of accurate slip torque.

Kaplan used his "controlled-slip" concept in two other products. In the controlled-torque screwdriver (Fig. 3) a stepped spring provides a 1¼-in.-lb slip when turned in either direction. It avoids overtightening machine screws in delicate instrument assemblies. A stepped spring is also the basis for the go/no-go torque gage that permits production inspection of output torques to within 1%.

Interfering spring. The three products were the latest in a series of slip clutches, drag brakes, and slip couplings developed by Kaplan for instrument brake drives. All are actually outgrowths of the spring clutch. The spring in this clutch is normally prevented from gripping the shaft by a detent response. Upon release of the detent, the spring will grip the shaft. If the shaft is turning in the proper direction, it is self-energizing. In the other direction, the spring simply overrides. Thus, the spring clutch is a "one-way" clutch.

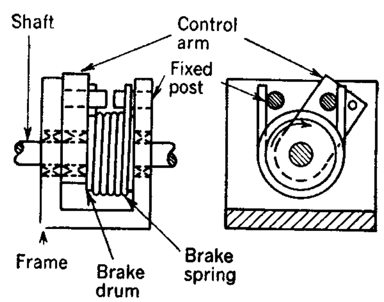


Fig. 1 Variable-torque drag brake . . .

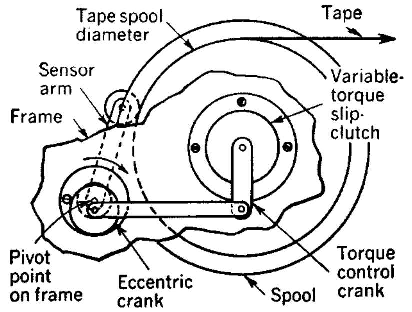


Fig. 2 . . . holds tension constant on tape

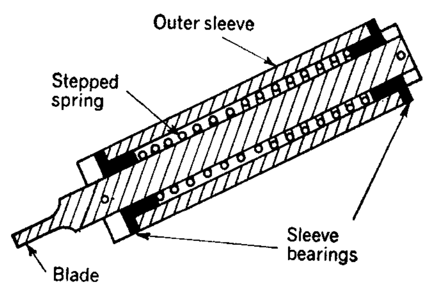


Fig. 3 Constant-torque screwdriver

SPRING BANDS GRIP TIGHTLY TO DRIVE OVERRUNNING CLUTCH

An overrunning clutch that takes up only half the space of most clutches has a series of spiral-wound bands instead of conventional rollers or sprags to transmit high torques. The design (see drawing) also simplifies the assembly, cutting costs as much as 40% by eliminating more than half the parts in conventional clutches.

The key to the savings in cost and space is the clutches' freedom from the need for a hardened outer race. Rollers and sprags must have hardened races because they transmit power by a wedging action between the inner and outer races.

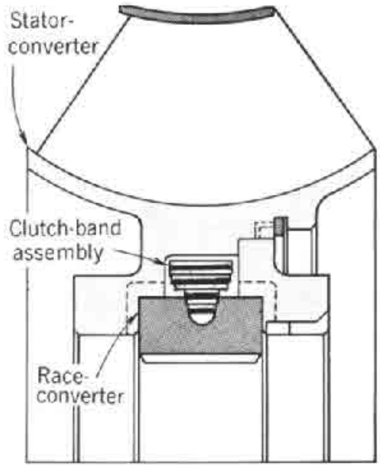
Role of spring bands. Overrunning clutches, including the spiral-band type, slip and overrun when reversed (see drawing). This occurs when the outer member is rotated clockwise and the inner ring is the driven member.

The clutch, developed by National Standard Co., Niles, Michigan, contains a set of high-carbon spring-steel bands (six in the design illustrated) that grip the inner member when the clutch is driving. The outer member simply serves to retain the spring anchors and to play a part in actuating the clutch. Because it isn't subject to wedging action, it can be made of almost any material, and this accounts for much of the cost saving. For example, in the automotive torque converter in the drawing at right, the bands fit into the aluminum die-cast reactor.

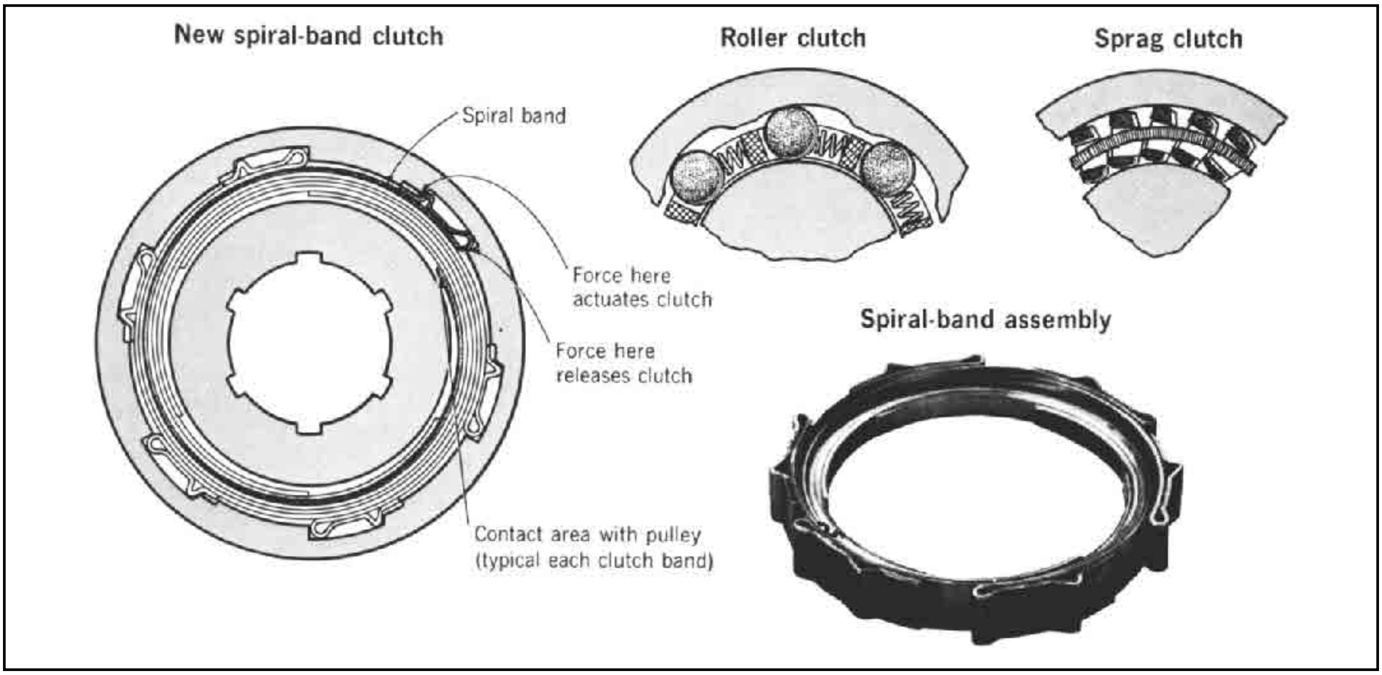
Reduced wear. The bands are spring-loaded over the inner member of the clutch, but they are held and rotated by the outer member. The centrifugal force on the bands then releases much of the force on the inner member and considerably decreases the overrunning torque. Wear is consequently greatly reduced.

The inner portion of the bands fits into a V-groove in the inner member. When the outer member is reversed, the bands wrap, creating a wedging action in this V-groove. This action is similar to that of a spring clutch with a helical-coil spring, but the spiral-band type has very little unwind before it overruns, compared with the coil type. Thus, it responds faster.

Edges of the clutch bands carry the entire load, and there is also a compound action of one band upon another. As the torque builds up, each band pushes down on the band beneath it, so each tip is forced more firmly into the V-groove. The bands are rated for torque capacities from 85 to 400 ft.-lb. Applications include their use in auto transmissions, starters, and industrial machinery.



Spiral clutch bands can be purchased separately to fit the user's assembly.



Spiral bands direct the force inward as an outer ring drives counterclockwise. The rollers and sprags direct the force outward.

SLIP AND BIDIRECTIONAL CLUTCHES COMBINE TO CONTROL TORQUE

A torque-limiting knob includes a dual set of miniature clutches—a detent slip clutch in series with a novel bi-directional-locking clutch—to prevent the driven member from backturning the knob. The bi-directional clutch in the knob locks the shaft from backlash torque originating within the panel, and the slip clutch limits the torque transmitted from outside the panel. The clutch was invented by Ted Chanoux, of Medford, N.Y.

The clutch (see drawing) is the result of an attempt to solve a problem that often plagues design engineers. A mechanism behind a panel such as a precision potentiometer or switch must be operated by a shaft that protrudes from the panel. The mechanism, however, must not be able to turn the shaft. Only the operator in front of the knob can turn the shaft, and he must limit the amount of torque he applies.

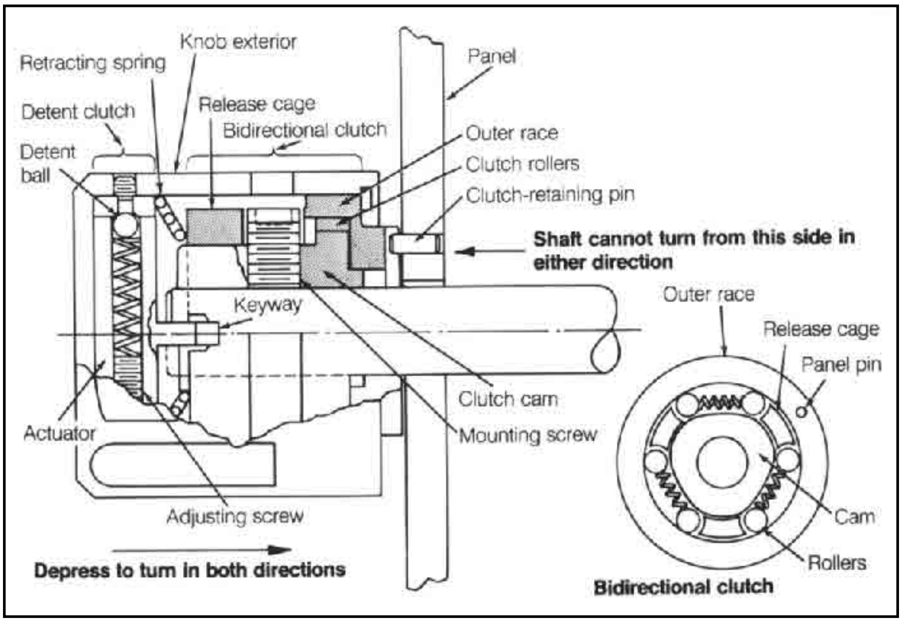
Solving design problem. This problem showed up in the design of a navigational system for aircraft.

The counter gave a longitudinal or latitudinal readout. When the aircraft was ready to take off, the navigator or pilot set a counter to some nominal figure, depending on the location of his starting point, and he energized the system. The computer then accepts the directional information from the gyro, the air speed from instruments in the wings, plus other data, and feeds a readout at the counter.

The entire mechanism was subjected to vibration, acceleration and deceleration, shock, and other high-torque loads, all of which could feed back through the system and might move the counter. The new knob device positively locks the mechanism shaft against the vibration, shock loads, and accidental turning, and it also limits the input torque to the system to a preset value.

Operation. To turn the shaft, the operator depresses the knob 1⁄16 in. and turns it in the desired direction. When it is released, the knob retracts, and the shaft immediately and automatically locks to the panel or frame with zero backlash. Should the shaft torque exceed the preset value because of hitting a mechanical stop after several turns, or should the knob turn in the retracted position, the knob will slip to protect the system mechanism.

Internally, pushing in the knob turns both the detent clutch and the bidirectional-clutch release cage via the keyway. The fingers of the cage extend between the clutch rollers so that the rotation of the cage cams out the rollers, which are usually kept jammed between the clutch cam and the outer race with the roller springs. This action permits rotation of the cam and instrument shaft both clockwise and counterclockwise, but it locks the shaft securely against inside torque up to 30 oz.-in.



Miniature knob is easily operated from outside the panel by pushing it in and turning it in the desired direction. When released, the bi-directional clutch automatically locks the shaft against all conditions of shock and vibration.

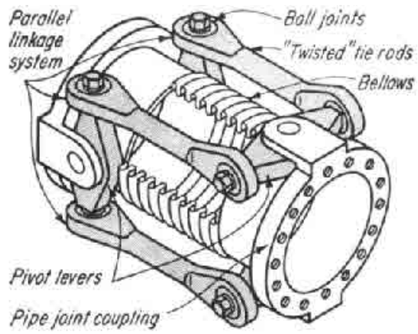
Applications. The detent clutch can be adjusted to limit the input torque to the desired values without removing the knob from the shaft. The outside diameter of the shaft is only 0.900 in., and the total length is 0.940 in. The exterior material of the knob is anodized aluminum, black or gray, and all other parts are stainless steel. The device is designed to meet the military requirements of MIL-E-5400, class 3 and MILK-3926 specifications.

Applications were seen in counter and reset switches and controls for machines and machine tools, radar systems, and precision potentiometers.

Eight-Joint Coupler

A novel coupler combines two parallel linkage systems in a three-dimensional arrangement to provide wide angular and lateral off-set movements in pipe joints. By including a bellows between the connecting pipes, the connector can join high-pressure and high-temperature piping such as is found in refineries, steam plants, and stationary power plants.

The key components in the coupler are four pivot levers (drawing) mounted



in two planes. Each pivot lever has provisions for a ball joint at each end. "Twisted" tie rods, with holes in different planes, connect the pivot levers to complete the system. The arrangement permits each pipe face to twist through an appreciable arc and also to shift orthogonally with respect to the other.

Longer tie rods can be formed by joining several bellows together with center tubes.

The connector was developed by Ralph Kuhm Jr. of El Segundo, California.

WALKING PRESSURE PLATE DELIVERS CONSTANT TORQUE

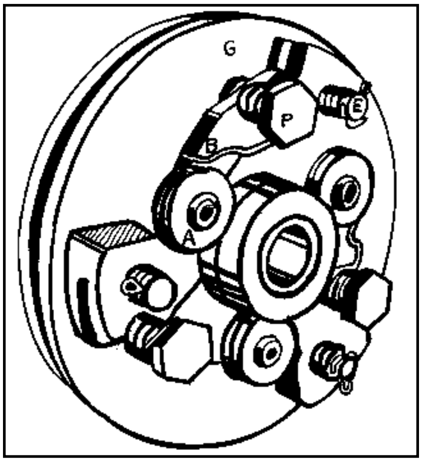
This automatic clutch causes the driving plate to move around the surface of the driven plate to prevent the clutch plates from overheating if the load gets too high. The "walking" action enables the clutch to transmit full engine torque for hours without serious damage to the clutch plates or the engine.

The automatic centrifugal clutch, manufactured by K-M Clutch Co., Van Nuys, California, combines the principles of a governor and a wedge to transmit torque from the engine to the drive shaft (see drawing).

How it works. As the engine builds up speed, the weights attached to the levers have a tendency to move towards the rim of the clutch plate, but they are stopped by retaining springs. When the shaft speed reaches 1600 rpm, however, centrifugal force overcomes the resistance of the springs, and the weights move outward. Simultaneously, the tapered end of the lever wedges itself in a slot in pin E, which is attached to the driving clutch plate. The wedging action forces both the pin and the clutch plate to move into contact with the driven plate.

A pulse of energy is transmitted to the clutch each time a cylinder fires. With every pulse, the lever arm moves outward, and there is an increase in pressure between the faces of the clutch. Before the next cylinder fires, both the lever arm and the driving plate return to their original positions. This pressure fluctuation between the two faces is repeated throughout the firing sequence of the engine.

Plate walks. If the load torque exceeds the engine torque, the clutch immediately slips, but full torque transfer is maintained without serious overheating. The pressure plate then momentarily disengages from the driven plate. However, as the plate rotates and builds up torque, it again comes in contact with the driven plate. In effect, the pressure plate "walks" around the contact surface of the driven plate, enabling the clutch to continuously transmit full engine torque.

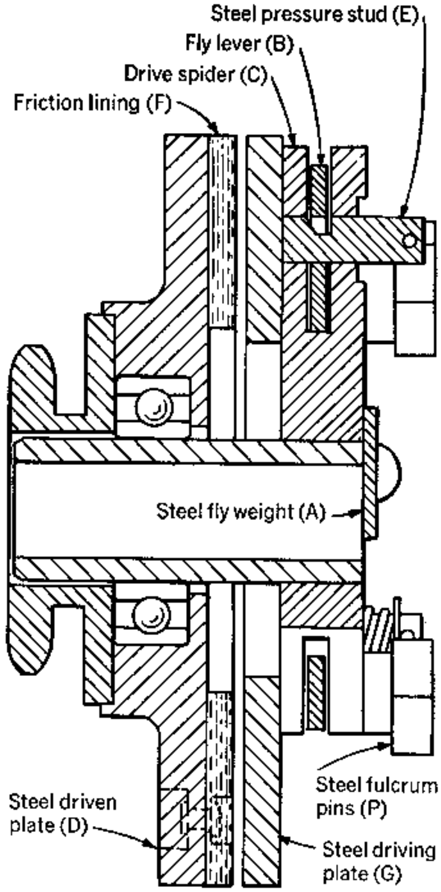


Applications. The clutch has undergone hundreds of hours of development testing on 4-stroke engines that ranged from 5 to 9 hp. According to the K-M Clutch Co., the clutch enables designers to use smaller motors than they previously could because of its no-load starting characteristics.

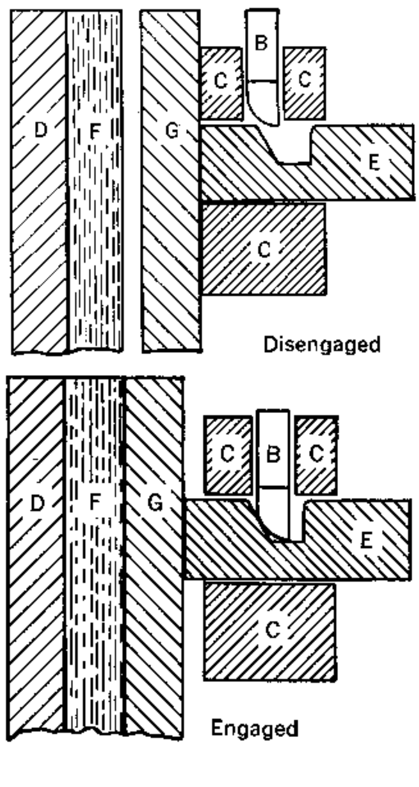
The clutch also acts as a brake to hold engine speeds within safe limits. For example, if the throttle accidentally opens when the driving wheels or driven mechanisms are locked, the clutch will stop.

The clutch can be fitted with sprockets, sheaves, or a stub shaft. It operates in any position, and can be driven in both directions. The clutch could be installed in ships so that the applied torque would come from the direction of the driven plate.

The pressure plate was made of cast iron, and the driven-plate casting was made of magnesium. To prevent too much wear, the steel fly weights and fly levers were pre-hardened.



A driving plate moves to plate D, closing the gap, when speed reaches 1600 rpm.



When a centrifugal force overcomes the resistance of the spring force, the lever action forces the plates together.

CONICAL-ROTOR MOTOR PROVIDES INSTANT CLUTCHING OR BRAKING

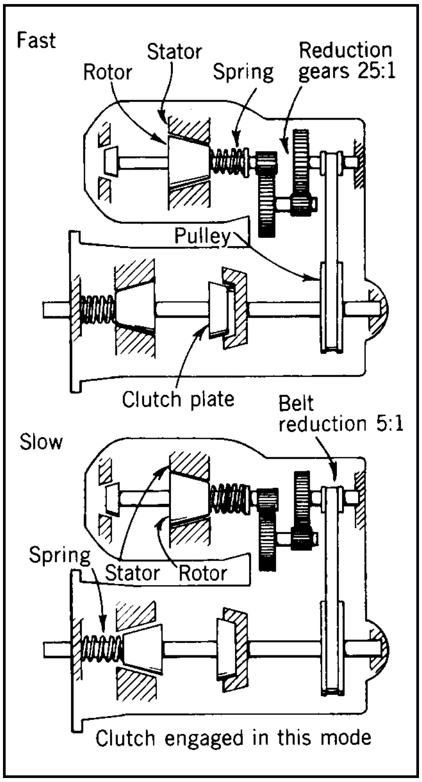
By reshaping the rotor of an ac electric motor, engineers at Demag Brake Motors, Wyandotte, Michigan, found that the axial component of the magnetic forces can be used to act on a clutch or a brake. Moreover, the motors can be arranged in tandem to obtain fast or slow speeds with instant clutching or braking.

As a result, this motor was used in many applications where instant braking is essential—for example, in an elevator when the power supply fails. The principal can also be applied to obtain a vernier effect, which is useful in machine-tool operations.

Operating principles. The Demag brake motor operates on a sliding-rotor principle. When no power is being applied, the rotor is pushed slightly away from the stator in an axial direction by a spring. However, with power the axial vector of the magnetic forces overcomes the spring pressure and causes the rotor to slide forward almost full into the stator. The maximum distance in an axial direction is 0.18 in. This effect permits that a combined fan and clutch, mounted on the rotor shaft, to engage with a brake drum when power is stopped, and disengage when power is applied.

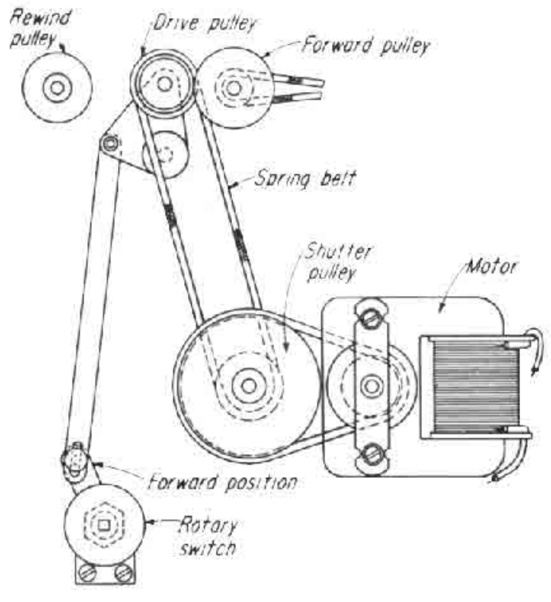
In Europe, the conical-rotored motor is used where rapid braking is essential to overcome time consuming overruns, or where accurate braking and precise angular positioning are critical—such as in packaging machines.

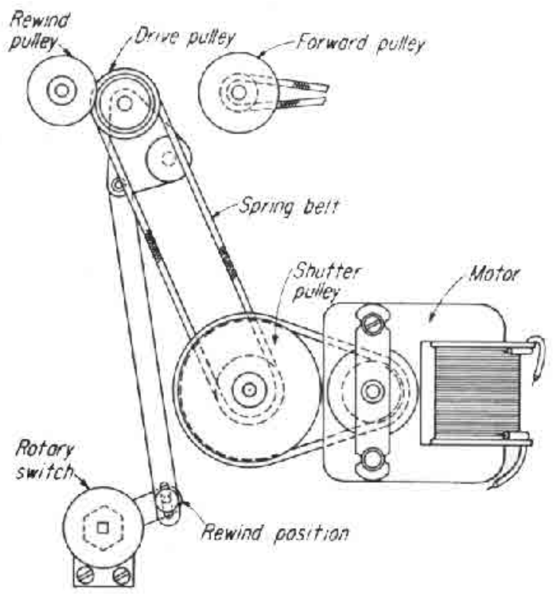
Novel arrangement. For instance, if two motors are installed, one running at 900 rpm and the other at 3600 rpm, the unit can reduce travel at a precise moment from a fats speed to an inching speed. This is achieved in the following way: When the main motor (running at 3600 rpm, and driving a conveyor table at fat speed) is stopped, the rotor slides back, and the clutch plate engages with the other rotating clutch plate, which is being driven through a reduction gear system by the slower running motor.



Because the second motor is running at 900 rpm and the reduction through the gear and belts is 125:1, the speed is greatly reduced.

FAST-REVERSAL REEL DRIVE





A fast-reversal drive for both forward movement and rewind is shifted by the rotary switch; it also controls a lamp and drive motor. A short lever on the switch shaft is linked to an overcenter mechanism on which the drive wheel is mounted. During the shift from forward to rewind, the drive pulley crosses its pivot point so that the spring tension of the drive belt maintains pressure on the driven wheel. The drive from the shutter pulley is 1:1 by the spring belt to the drive pulley and through a reduction when the forward pulley is engaged. When rewind is engaged, the reduction is eliminated and the film rewinds at several times forward speed.

SEVEN OVERRUNNING CLUTCHES

These are simple devices that can be made inexpensively in the workshop.

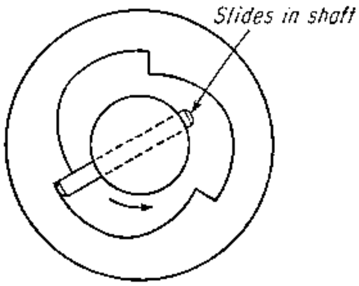


Fig. 1 A lawnmower clutch.

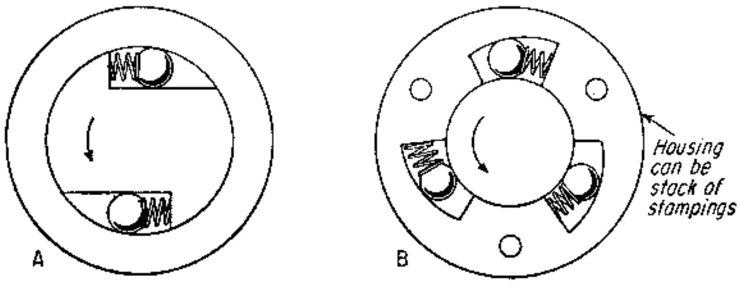


Fig. 2 Wedging balls or rollers: internal (A); external (B) clutches.

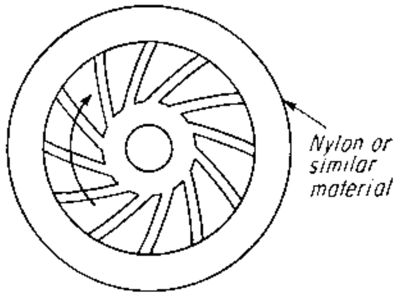


Fig. 3 Molded sprags (for light duty).

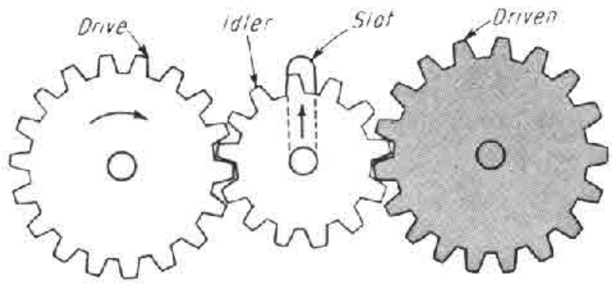


Fig. 4 A disengaging idler rises in a slot when the drive direction is reversed.

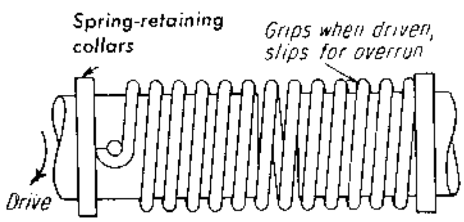


Fig. 5 A slip-spring coupling.

Fig. 6 An internal ratchet and spring-loaded pawls.

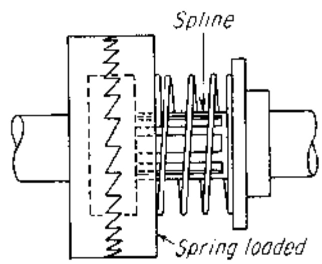
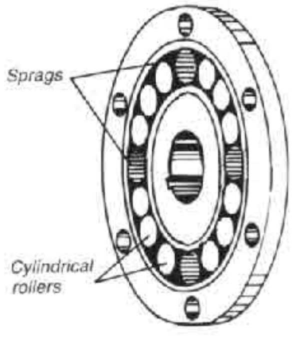


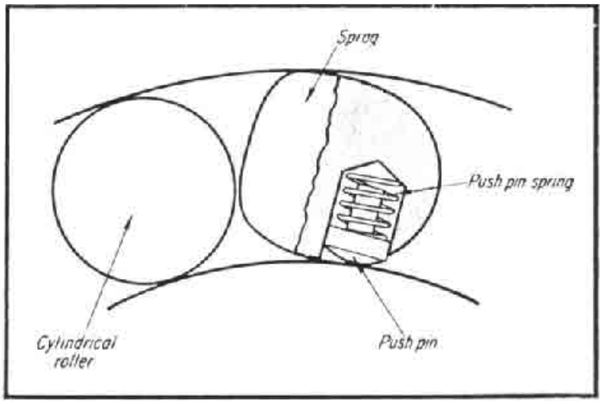
Fig. 7 A one-way dog clutch.

SPRING-LOADED PINS AID SPRAGS IN ONE-WAY CLUTCH

Sprags combined with cylindrical rollers in a bearing assembly can provide a simple, low-cost method for meeting the torque and bearing requirements of most machine applications. Designed and built by Est. Nicot of Paris, this unit gives one-direction-only torque transmission in an overrunning clutch. In addition, it also serves as a roller bearing.

The torque rating of the clutch depends on the number of sprags. A minimum of three, equally spaced around the circumference of the races, is generally necessary to get acceptable distribution of tangential forces on the races.





Races are concentric; a locking ramp is provided by the sprag profile, which is composed of two nonconcentric curves of different radius. A spring-loaded pin holds the sprag in the locked position until the torque is applied in the running direction. A stock roller bearing cannot be converted because the hard-steel races of the bearing are too brittle to handle the locking impact of the sprag. The sprags and rollers can be mixed to give any desired torque value.

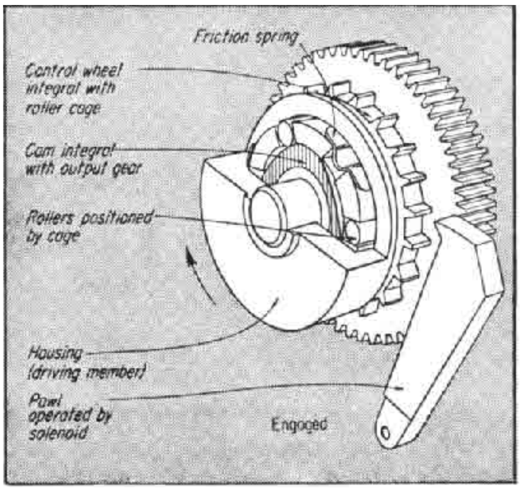
ROLLER-TYPE CLUTCH

This clutch can be adapted for either electrical or mechanical actuation, and will control ½ hp at 1500 rpm with only 7 W of power in the solenoid. The rollers are positioned by a cage (integral with the toothed control wheel —see diagram) between the ID of the driving housing and the cammed hub (integral with the output gear).

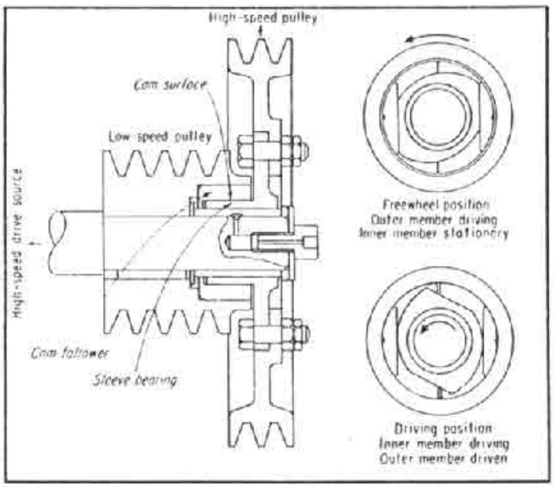
When the pawl is disengaged, the drag of the housing on the friction spring rotates the cage and wedges the rollers into engagement. This permits the housing to drive the gear through the cam.

When the pawl engages the control wheel while the housing is rotating, the friction spring slips inside the housing and the rollers are kicked back, out of engagement. Power is therefore interrupted.

According to the manufacturer, Tiltman Langley Ltd, Surrey, England, the unit operated over the full temperature range of –40º to 200ºF.



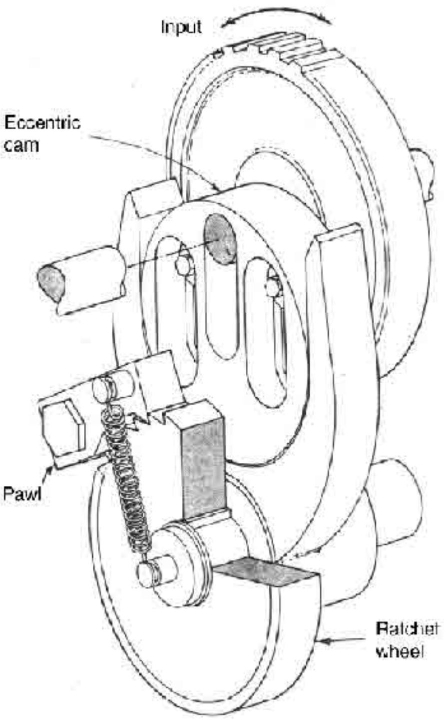
A positive drive is provided by this British roller clutch.



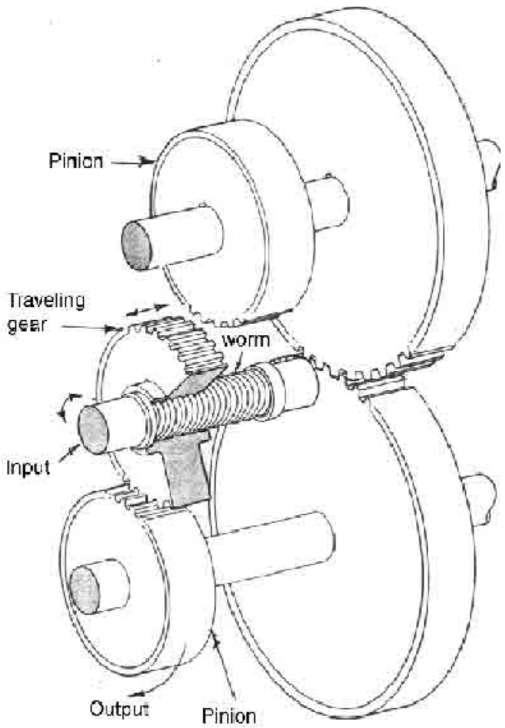
Two-speed operation is provided by the new cam clutch

This clutch consists of two rotary members (see diagrams), arranged so that the outer (follower) member acts on its pulley only when the inner member is driving. When the outer member is driving, the inner member idles. One application was in a dry-cleaning machine. The clutch functions as an intermediary between an ordinary and a high-speed motor to provide two output speeds that are used alternately.

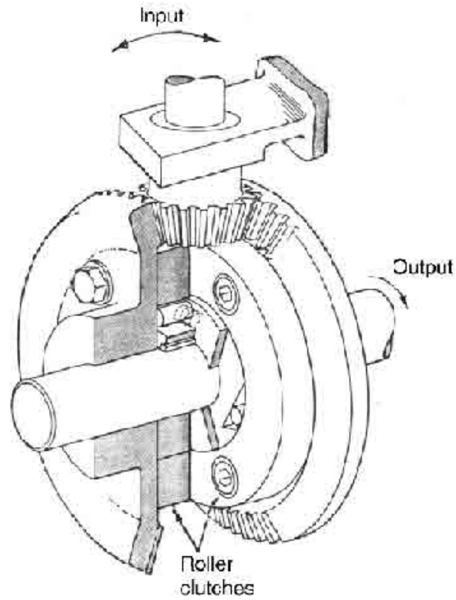
ONE-WAY OUTPUT FROM SPEED REDUCERS



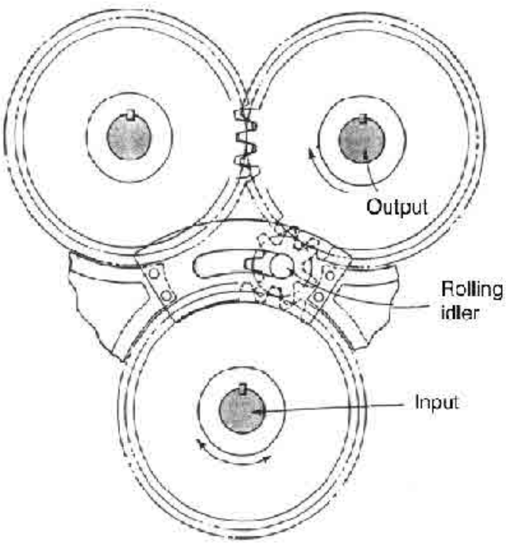
This eccentric cam adjusts over a range of high reduction ratios, but unbalance limits it to low speeds. When its direction of input changes, thee is no lag in output rotation. The output shaft moves in steps because of a ratchet drive through a pawl which is attached to a U follower.



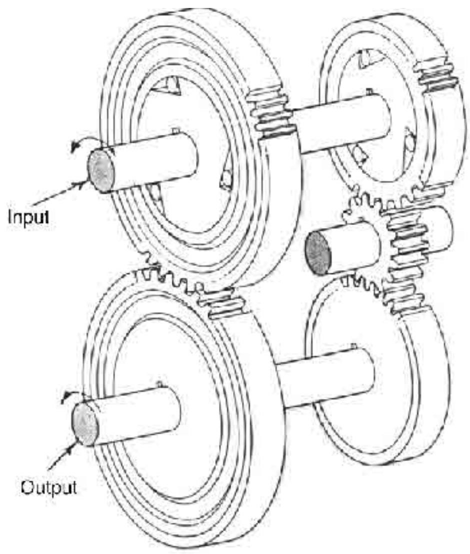
A traveling gear moves along a worm and transfers drive torque to the other pinion when the input rotation changes direction. To ease the gear engagement, the gear teeth are tapered at their ends. Output rotation is smooth, but there is a lag after direction changes as the gear shifts. The gear cannot be wider than the axial offset between pinions or there will be destructive interference.



Two bevel gears drive through roller clutches. One clutch catches in one direction and the other catches in the opposite direction. There is little or no interruption of smooth output rotation when the input direction changes.



This rolling idler also provides a smooth output and a slight lag after its input direction changes. A small drag on the idler is necessary so that it will transfer smoothly into engagement with the other gear and not remain spinning between the gears.



Roller clutches are on the input gears in this drive. These also give smooth output speed and little output lag as the direction changes.

SPRINGS, SHUTTLE PINION, AND SLIDING BALL PERFORM IN ONE-WAY DRIVES

These four drives change oscillating motion into one-way rotation to perform feeding tasks and counting.

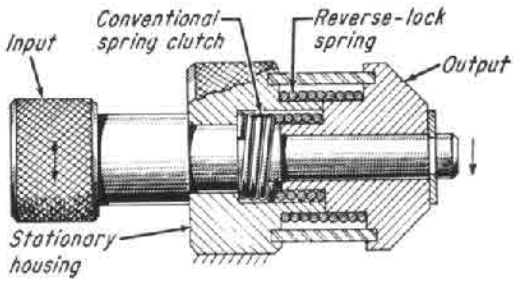


Fig. 1 A double-spring clutch drive.

Fig. 2 A basic spring clutch.

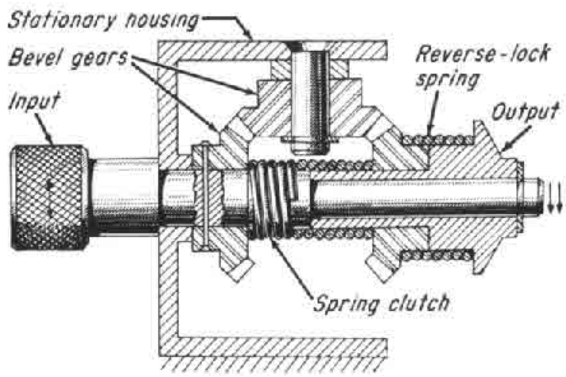


Fig. 3 A full-wave rectification drive.

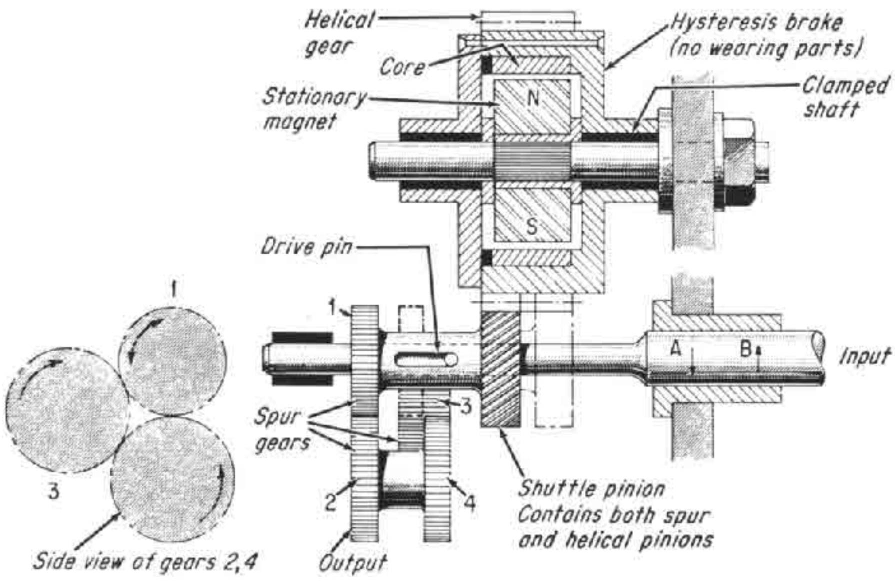


Fig. 4 A shuttle-pinion drive.

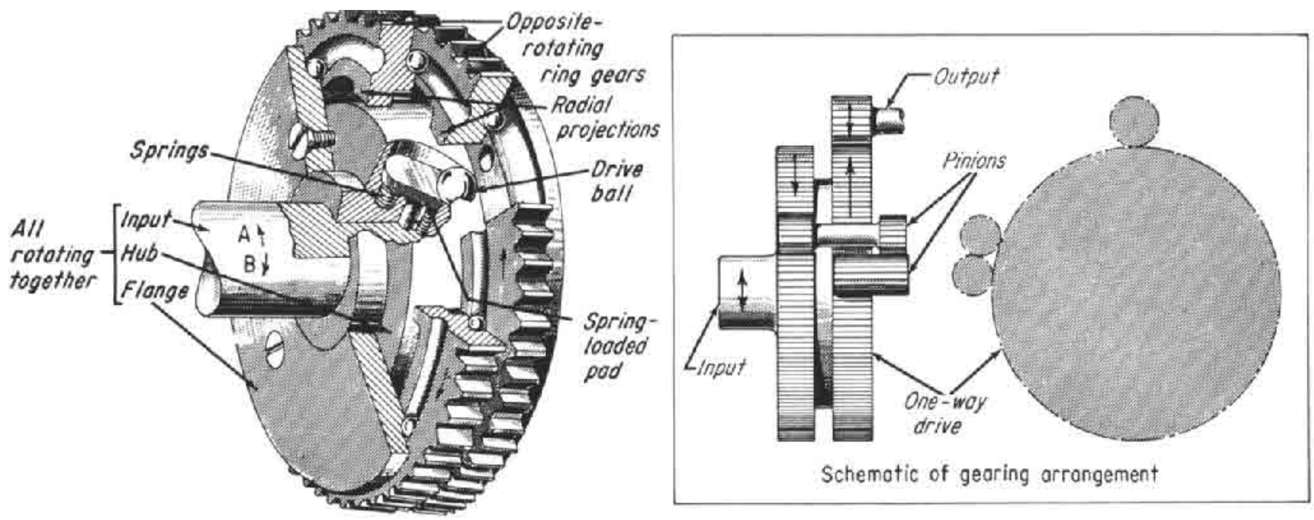


Fig. 5 A reciprocating-ball drive.

The one-way drive, shown in Fig. 1, was invented as a byproduct of the design of a money-order imprinter.

The task was to convert the oscillating motion of the input crank (20º in this case) into a one-way motion to advance an inking ribbon. One of the simplest known devices was used to obtain the one-way drive—a spring clutch which is a helical spring joining two co-linear butting shafts (Fig. 2). The spring is usually made of square or rectangular cross-section wire.

This clutch transmits torque in one direction only because it overrides when it is reversed. The helical spring, which bridges both shafts, need not be fastened at either end; a slight interference fit is acceptable. Rotating the input shaft in the direction tending to wind the spring (direction A in Fig. 2) causes the spring to grip both shafts and then transmit motion from the input to the output shaft. Reversing the input unwinds the spring, and it overrides the output shaft with a drag—but this drag, slight as it was, caused a problem in operation.

Double-Clutch Drive

The spring clutch (Fig. 2) did not provide enough friction in the tape drive to allow the spring clutch to slip on the shafts on the return stroke. Thus the output moved in sympathy with the input, and the desired one-way drive was not achieved.

At first, an attempt was made to add friction artificially to the output, but this resulted in an awkward design. Finally the problem was elegantly solved (Fig. 1) by installing a second helical spring, slightly larger than the first that served exactly the same purpose: transmission of motion in one direction only. This spring, however joined the output shaft and a stationary cylinder. In this way, with the two springs of the same hand, the undesirable return motion of the ribbon drive was immediately arrested, and a positive one-way drive was obtained quite simply.

This compact drive can be considered to be a mechanical *half-wave rectifier* in that it transmits motion in one direction only while it suppresses motion in the reverse direction.

Full-Wave Rectifier

The principles described will also produce a mechanical *full-wave rectifier* by introducing some reversing gears, Fig. 3. In this application the input drive in one direction is directly transmitted to the output, as before, but on the reverse stroke the input is passed through reversing gears so that the output appears in the opposite sense. In other words, the original sense of the output is maintained. Thus, the output moves forward twice for each back-and-forth movement of the input.

Shuttle-Gear Drive

Earlier, a one-way drive was developed that harnessed the axial thrust of a pair of helical gears to shift a pinion, Fig. 4. Although at first glance, it might look somewhat complicated, the drive is inexpensive to make and has been operating successfully with little wear.

When the input rotates in direction *A,* it drives the output through spur gears *1* and *2.* The shuttle pinion is also driving the helical gear whose rotation is resisted by the magnetic flux built up between the stationary permanent magnet and the rotating core. This magnet-core arrangement is actually a hysteresis brake, and its constant resisting torque produces an axial thrust in mesh of the helical pinion acting to the left. Reversing the input reverses the direction of thrust, which shifts the shuttle pinion to the right. The drive then operates through gears *1, 3,* and *4,* which nullifies the reversion to produce output in the same direction.

Reciprocating-Ball Drive

When the input rotates in direction A, Fig. 5, the drive ball trails to the right, and its upper half engages one of the radial projections in the right ring gear to drive it in the same direction as the input. The slot for the ball is milled at 45º to the shaft axes and extends to the flanges on each side.

When the input is reversed, the ball extends to the flanges on each side, trails to the left and deflects to permit the ball to ride over to the left ring gear, and engage its radial projection to drive the gear in the direction of the input.

Each gear, however, is constantly in mesh with a pinion, which in turn is in mesh with the other gear. Thus, regardless of the direction the input is turned, the ball positions itself under one or another ring gear, and the gears will maintain their respective sense of rotation (the rotation shown in Fig. 5). Hence, an output gear in mesh with one of the ring gears will rotate in one direction only.

DETAILS OF OVERRIDING CLUTCHES

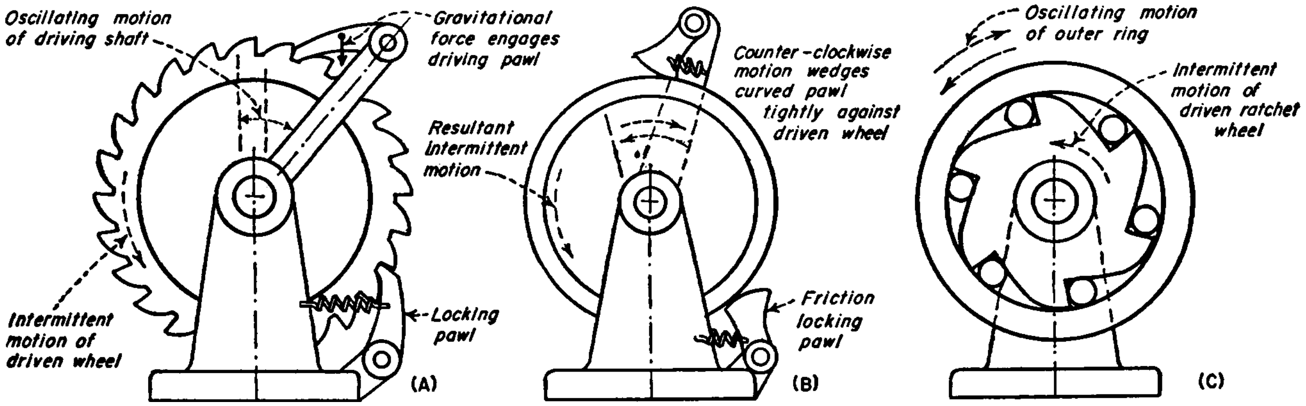


Fig. 1 Elementary overriding clutches: (A) A ratchet and pawl mechanism converts reciprocating or oscillating movement to intermittent rotary motion. This motion is positive but limited to a multiple of the tooth pitch. (B) A friction-type clutch is quieter, but it requires a spring device to keep the eccentric pawl in constant engagement. (C) Balls or rollers replace the pawls in this device. Motion of the outer race wedges the rollers against the inclined surfaces of the ratchet wheel.

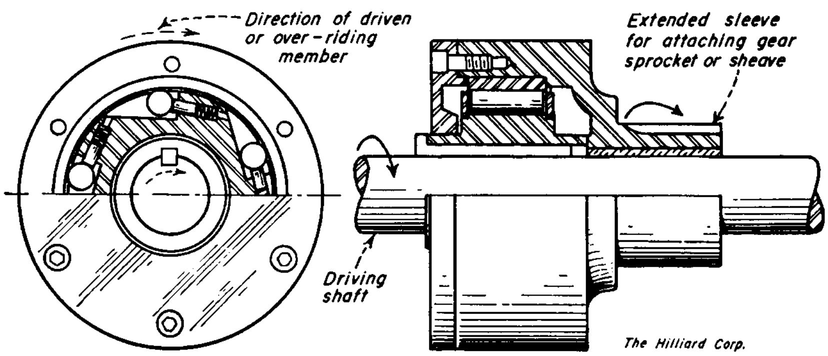


Fig. 2 A commercial overriding clutch has springs that hold the rollers in continuous contact between the cam surfaces and the outer race; thus, there is no backlash or lost motion. This simple design is positive and quiet. For operation in the opposite direction, the roller mechanism can easily be reversed in the housing.

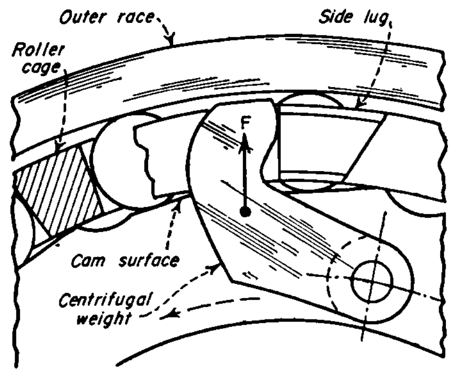


Fig. 3 A centrifugal force can hold the rollers in contact with the cam and outer race. A force is exerted on the lugs of the cage that controls the position of the rollers.

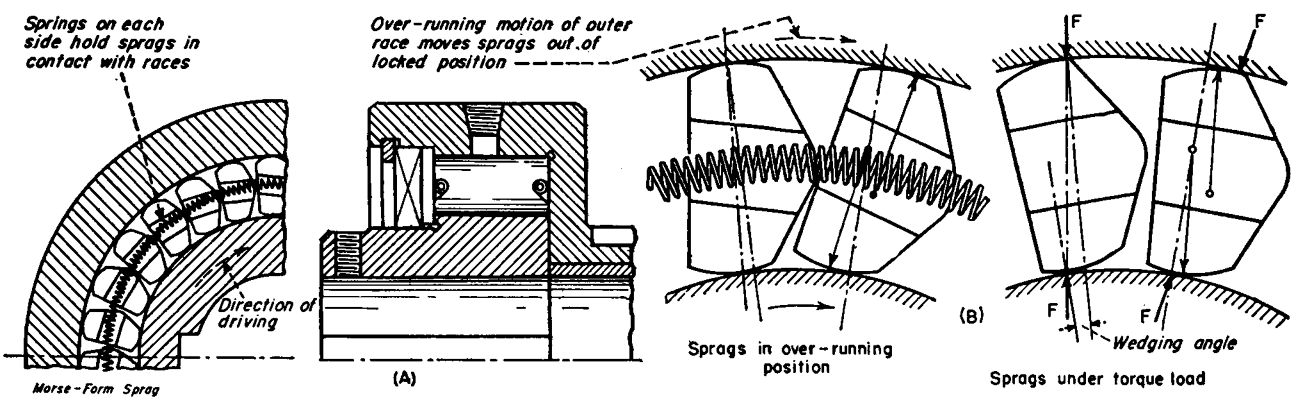


Fig. 4 With cylindrical inner and outer races, sprags can transmit torque. Energizing springs serve as a cage to hold the sprags. (A) Compared to rollers, the shape of a sprag permits a greater number within a limited space; thus higher torque loads are possible. Special cam surfaces are not required, so this version can be installed inside gear or wheel hubs. (B) Rolling action wedges the sprags tightly between the driving and driven members. A relatively large wedging angle ensures positive engagement.

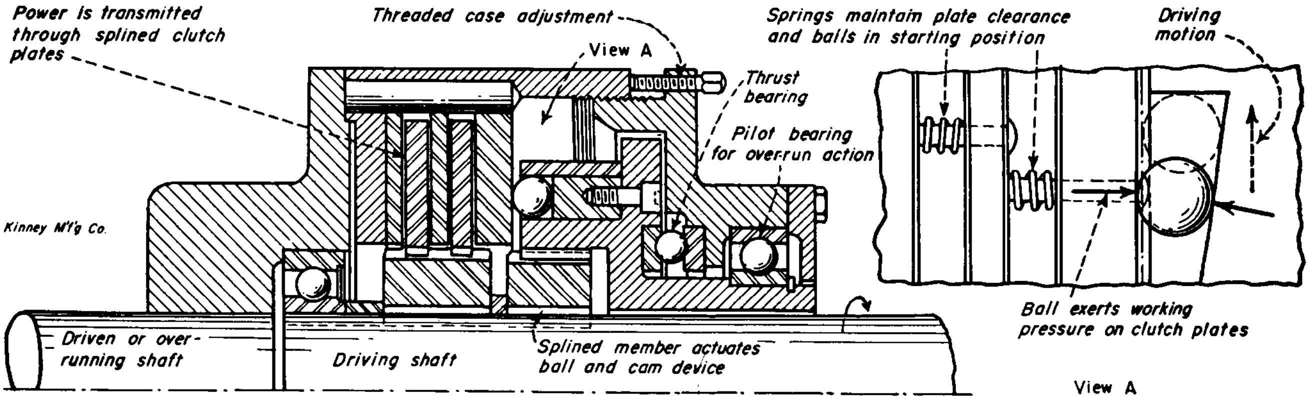


Fig. 5 A multidisk clutch is driven by several sintered-bronze friction surfaces. Pressure is exerted by a cam-actuating device that forces a series of balls against a disk plate. A small part of the transmitted torque is carried by the actuating member, so capacity is not limited by the localized deformation of the contacting balls. The slip of the friction surfaces determines the capacity and prevents rapid shock loads. The slight pressure of disk springs ensures uniform engagement.

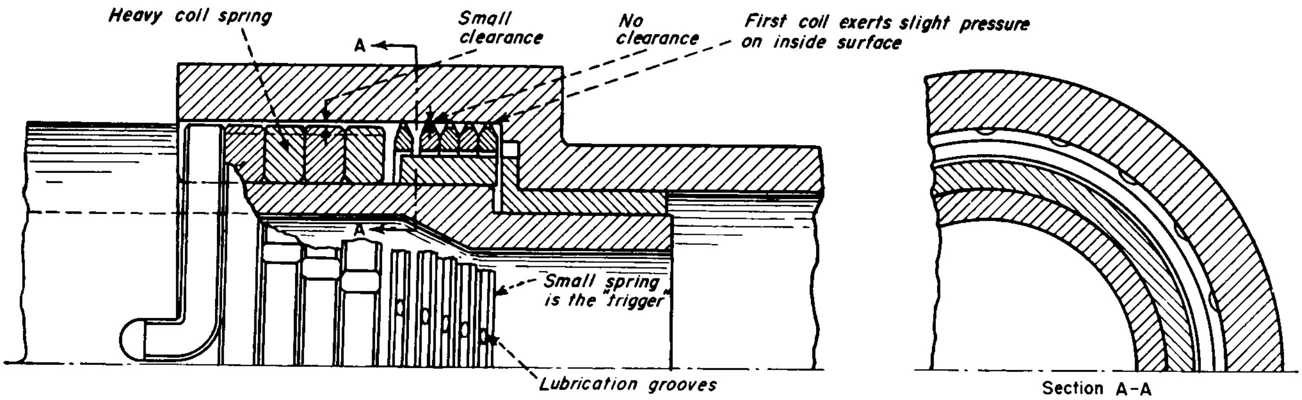


Fig. 6 An engaging device consists of a helical spring that is made up of two sections: a light trigger spring and a heavy coil spring. It is attached to and driven by the inner shaft. The relative motion of the outer member rubbing on the trigger causes this spring to wind up. This action expands the spring diameter, which takes up the small clearance and exerts pressure against the inside surface until the entire spring is tightly engaged. The helix angle of the spring can be changed to reverse the overriding direction.

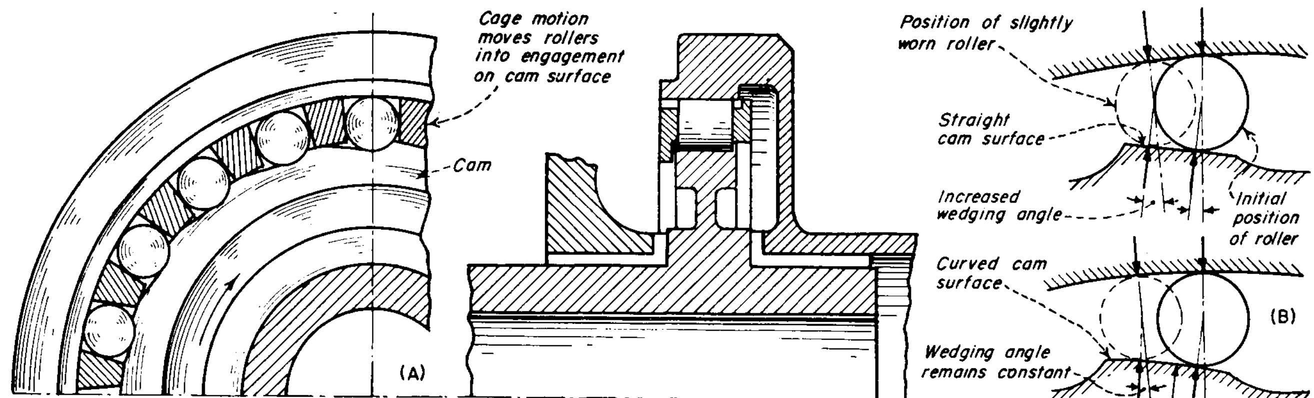


Fig. 7 A free-wheeling clutch widely used in power transmission has a series of straight-sided cam surfaces. An engaging angle of about 3º is used; smaller angles tend to become locked and are difficult to disengage while larger ones are not as effective. (A) The inertia of a floating cage wedges the rollers between the cam and outer race. (B) Continual operation causes the wear of surfaces; 0.001 in. wear alters the angle to 8.5º on straight-sided cams. Curved cam surfaces maintain a constant angle.

TEN WAYS TO APPLY OVERRUNNING CLUTCHES

These clutches allow freewheeling, indexing and backstopping; they will solve many design problems. Here are examples.

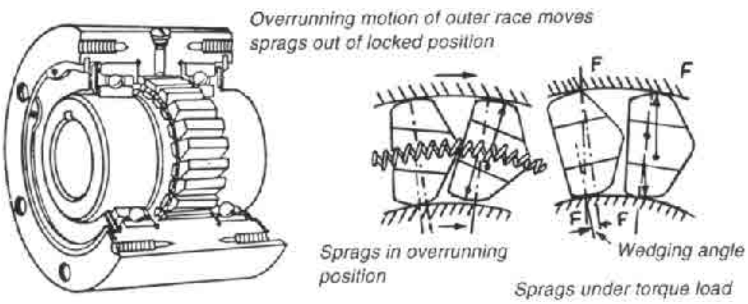


Fig. 1 Precision sprags act as wedges and are made of hardened alloy steel. In the formsprag clutch, torque is transmitted from one race to another by the wedging action of sprags between the races in one direction; in the other direction the clutch freewheels.

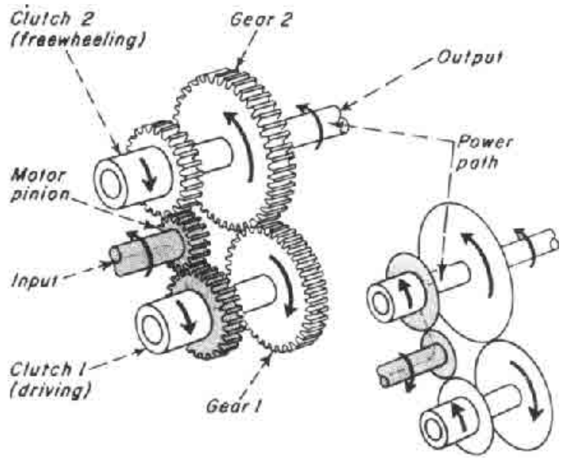


Fig. 2 This speed drive requires input rotation to be reversible. Counterclockwise input (as shown in the diagram) drives gear 1 through clutch 1; the output is counterclockwise; clutch 2 overruns. Clockwise input (schematic) drives gear 2 through clutch 2; the output is still counterclockwise; clutch 1 overruns.

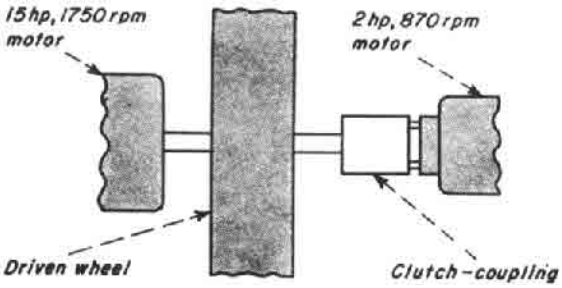


Fig. 3 This speed drive for a grinding wheel can be a simple, in-line assembly if the overrunning clutch couples two motors. The outer race of the clutch is driven by a gearmotor; the inner race is keyed to a grinding-wheel shaft. When the gearmotor drives, the clutch is engaged; when the larger motor drives, the inner race overruns.

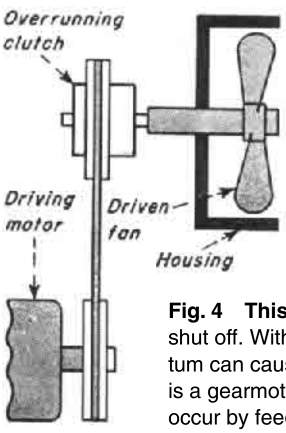


Fig. 4 This fan freewheels when driving power is shut off. Without an overrunning clutch, fan momentum can cause belt breakage. If the driving source is a gearmotor, excessive gear stress can also occur by feedback of kinetic energy from the fan.

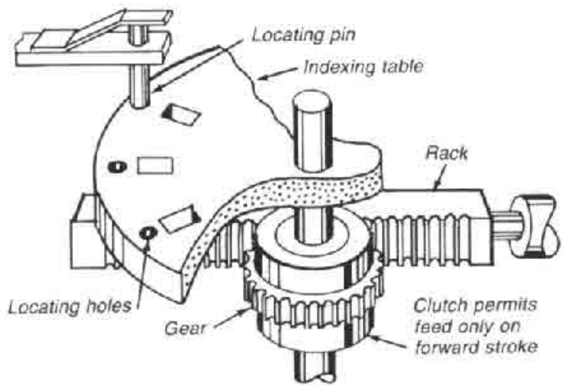


Fig. 5 This indexing table is keyed to a clutch shaft. The table is rotated by the forward stroke of the rack; power is transmitted through the clutch by its outer-ring gear only during this forward stroke. Indexing is slightly short of the position required. The exact position is then located by a spring-loaded pin that draws the table forward to its final positioning. The pin now holds the table until the next power stroke of the hydraulic cylinder.

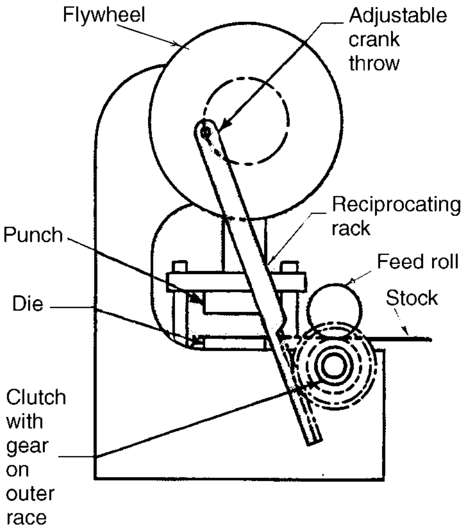


Fig. 6 This punch press feed is arranged so that the strip is stationary on the downstroke of the punch (clutch freewheels); feed occurs during the upstroke when the clutch transmits torque. The feed mechanism can easily be adjusted to vary the feed amount.

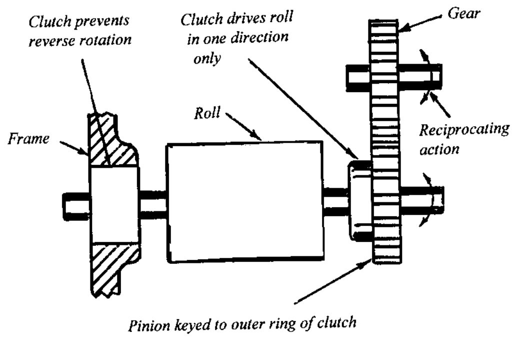


Fig. 7 Indexing and backstopping is done with two clutches arranged so that one drives while the other freewheels. The application shown here is for a capsuling machine; gelatin is fed by the roll and stopped intermittently so the blade can precisely shear the material to form capsules.

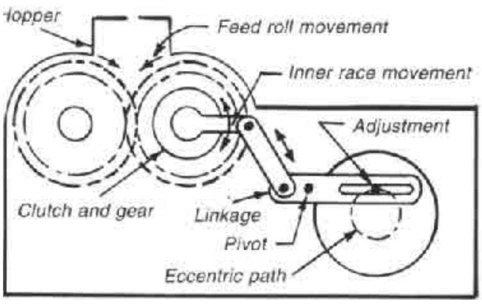


Fig. 8 The intermittent motion of a candy machine is adjustable. The clutch ratchets the feed rolls around. This keeps the material in the hopper agitated.

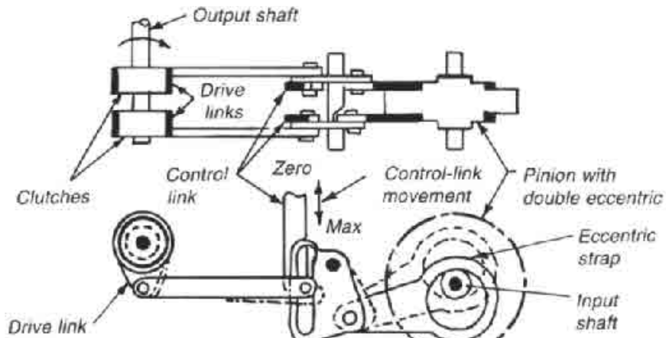


Fig. 9 This double-impulse drive has double eccentrics and drive clutches. Each clutch is indexed 180° out of phase with the other. One revolution of the eccentric produces two drive strokes. Stroke length, and thus the output rotation, can be adjusted from zero to maximum by the control link.

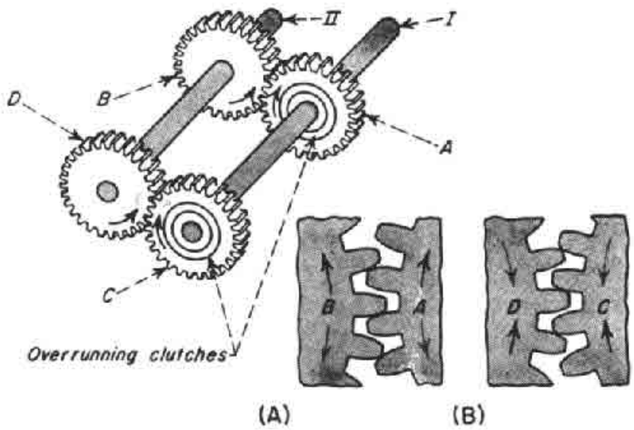


Fig. 10 This anti-backlash device depends on overrunning clutches to ensure that no backlash is left in the unit. Gear *A* drives *B* and shaft *II* with the gear mesh and backlash, as shown in (A). The overrunning clutch in gear *C* permits gear *D* (driven by shaft *II*) to drive gear *C* and results in the mesh and backlash shown in (B). The overrunning clutches never actually overrun. They provide flexible connections (something like split and sprung gears) between shaft *I* and gears *A* and *C* to allow absorption of all backlash.

APPLICATIONS FOR SPRAG-TYPE CLUTCHES

Overrunning sprag clutches transmit torque in one direction and reduce speed, rest, hold, or free-wheel in the reverse direction. Applications include overrunning, backstopping, and indexing. Their selection—similar to other mechanical devices—requires a review of the torque to be transmitted, overrunning speed, type of lubrication, mounting characteristics, environmental conditions, and shock conditions that might be encountered.

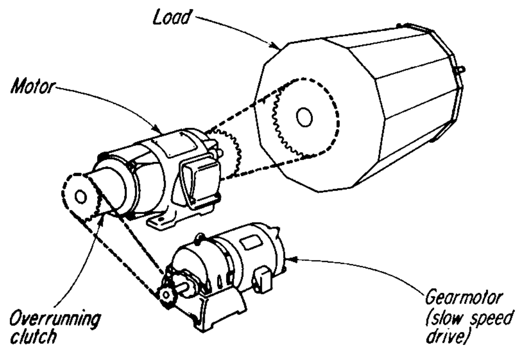


Fig. 1 Overrunning permits torque transmission in one direction and free wheels or overruns in the opposite direction. For example, the gar motor drives the load by transmitting torque through the overrunning clutch and the high-speed shaft. Energizing the high-speed motor causes the inner member to rotate at the rpm of the high-speed motor. The gear motor continues to drive the inner member, but the clutch is free-wheeling.

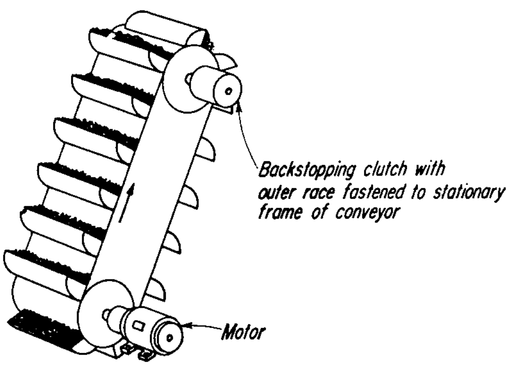


Fig. 2 Backstopping permits rotation in one direction only. The clutch serves as a counter-rotation holding device. An example is a clutch mounted on the headshaft of a conveyor. The outer race is restrained by torque-arming the stationary frame of the conveyor. If, for any reason, power to the conveyor is interrupted, the back-stopping clutch will prevent the buckets from running backwards and dumping the load.

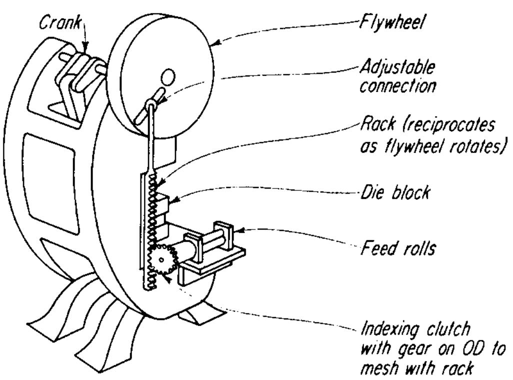


Fig. 3 Indexing is the transmission of intermittent rotary motion in one direction; an example is the feed rolls of a punch press. On each stroke of the press crankshaft, a feed stroke on the feed roll is accomplished by the rack-and-pinion system. The system feeds the material into the dies of the punch press.

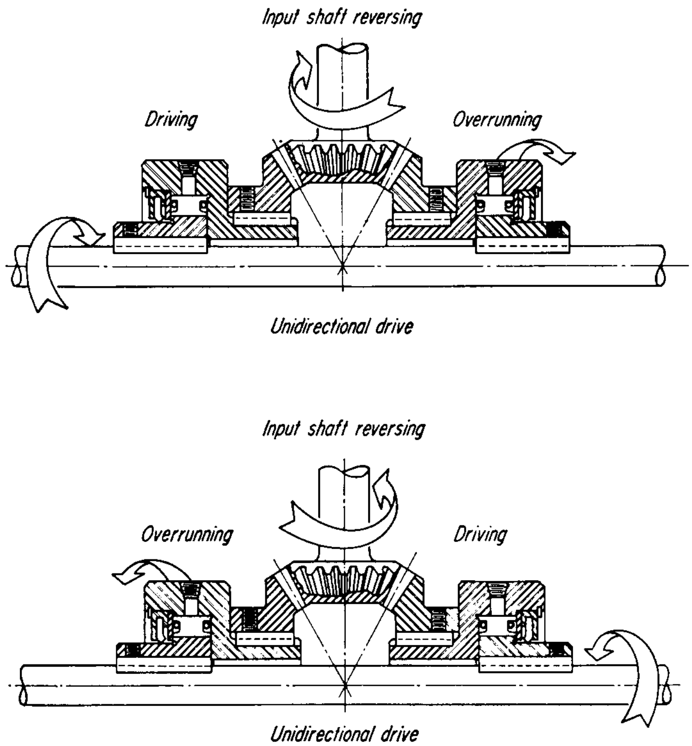


Fig. 4 Unidirectional drives with reverse mechanism incorporate two overrunning clutches into the gears, sheaves, or sprockets. Here, a 1:1 ratio right-angle drive is shown with a reversing input shaft. The output shaft rotates clockwise, regardless of the input shaft direction. By changing gear sizes, combinations of continuous or intermittent unidirectional output relative to the input can be obtained.

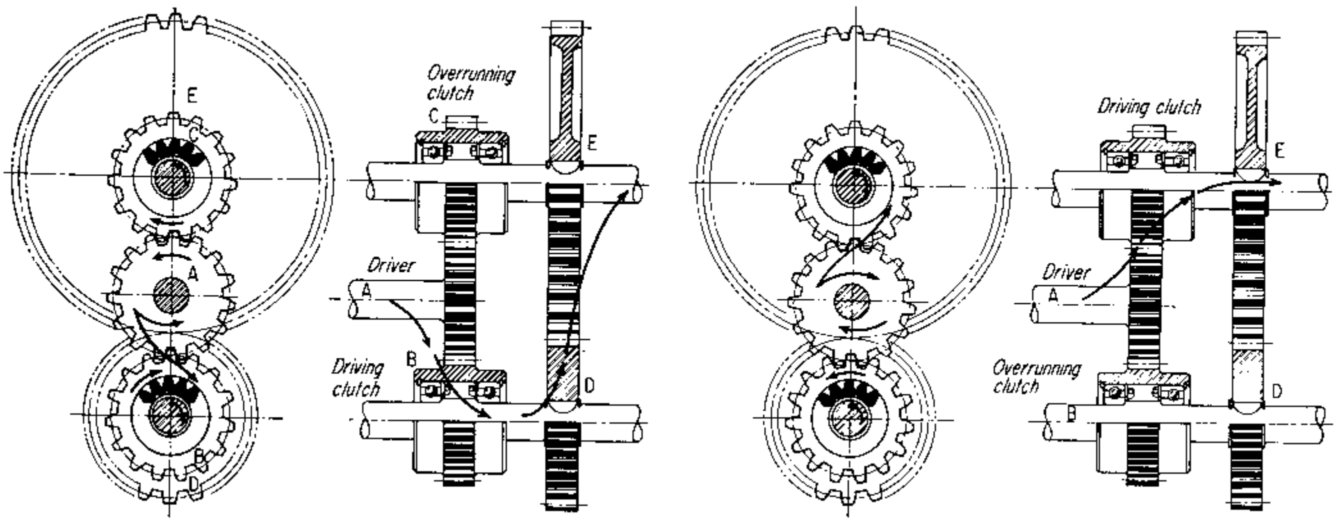


Fig. 5 Two-speed unidirectional output is made possible by using spur gears and reversing the direction of the input shaft. The rotation of shaft A transfers the power of gears B, D, and E to the output. Counterclockwise rotation engages the lower clutch, freewheeling the upper clutch because gear C is traveling at a faster rate than the shaft. This is caused by the reduction between gears B and E. Clockwise rotation of A engages the upper clutch, while the lower clutch freewheels because of the speed increase between gears D and E.

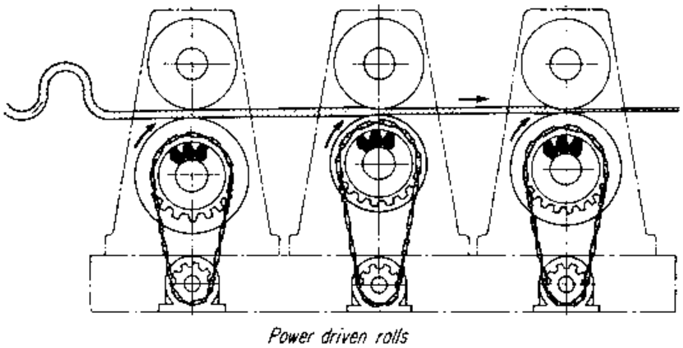


Fig. 6 A speed-differential or compensation is required where a different speed range for a function is desired, while retaining the same basic speed for all other functions. A series of individually driven power rolls can have different surface speeds because of drive or diameter variations of the rolls. An overrunning clutch permits the rolls with slower peripheral speed to overspeed and adjust to the material speed.

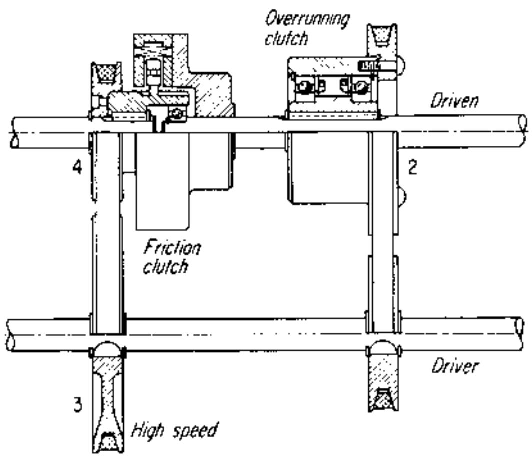


Fig. 7 A speed differential application permits the operation of engine accessories within a narrow speed range while the engine operates over a wide range. Pulley No. 2 contains the overrunning clutch. When the friction or electric clutch is disengaged, the driver pulley drives pulley No. 2 through the overrunning clutch, rotating the driven shaft. The engagement of the friction or electric clutch causes high-speed driven shaft rotation. This causes an overrun condition in the clutch at pulley No. 2.

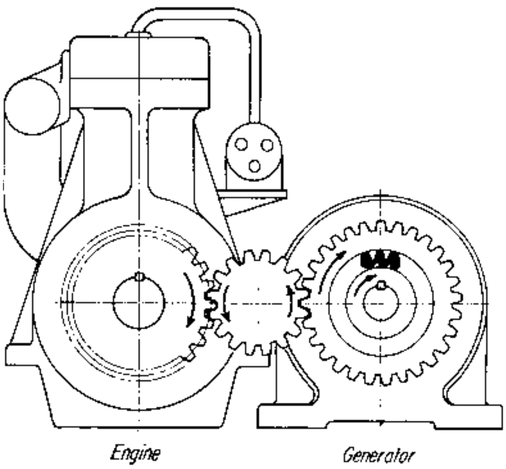


Fig. 8 High inertia dissipation avoids driving back through a power system. In machines with high resistances, it prevents power train damage. If the engine is shut down while the generator is under a no-load condition, it would have a tendency to twist off the generator shaft. The overrunning clutch allows generator deceleration at a slower rate than the engine deceleration.

SMALL MECHANICAL CLUTCHES FOR PRECISE SERVICE

Clutches for small machines must have: (1) Quick response—lightweight moving parts; (2) Flexibility—permit multiple members to control operation; (3) Compactness—for equivalent capacity positive clutches are smaller than friction; (4) Dependability; and (5) Durability.

Fig. 1 A pawl and ratchet, single-cycle Dennis clutch. The primary parts of this clutch are the driving ratchet B, the driven cam plate C, and the connecting pawl D, which is carried by the cam plate. The pawl is normally held disengaged by the lower tooth of clutch arm A. When activated, arm A rocks counterclockwise until it is out of the path of rim F on cam plate C.

This permits pawl D, under the effect of spring E, to engage with ratchet B. Cam plate C then turns clockwise until, near the end of one cycle, pin G on the plate strikes the upper part of arm A, camming it clockwise back to its normal position. The lower part of A then performs two functions: (1) it cams pawl D out of engagement with the driving ratchet B, and (2) it blocks the further motion of rim F and the cam plate.

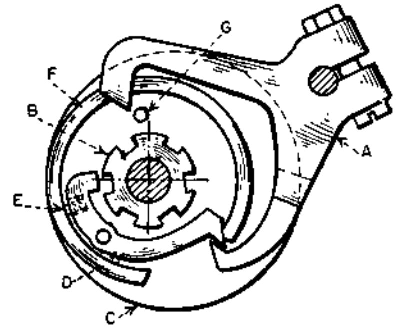


Fig. 1

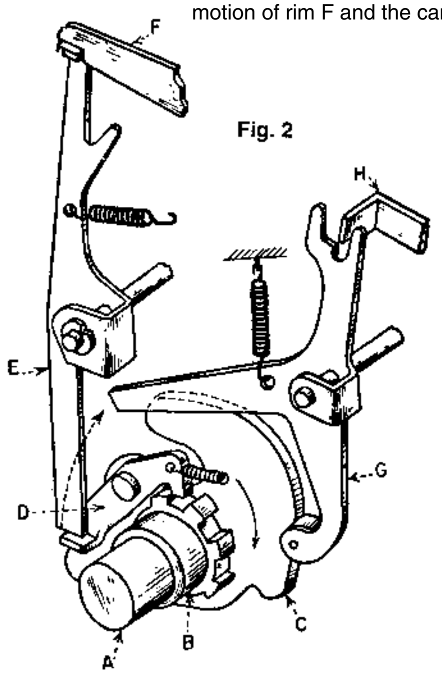


Fig. 2

Fig. 2 A pawl and ratchet, single-cycle, dual-control clutch. The principal parts of this clutch are driving ratchet B, driven crank C, and spring-loaded ratchet pawl D. Driving ratchet B is directly connected to the motor and free to rotate on rod A. Driven crank C is directly connected to the main shaft of the machine and is also free to move on rod A. Spring-loaded ratchet pawl D is carried by crank C, which is normally held disengaged by latch E.

To activate the clutch, arm F is raised, permitting latch E to trip and pawl D to engage with ratchet B. The left arm of clutch latch G, which is in the path of the lug on pawl D, is normally permitted to move out of the way by the rotation of the camming edge of crank C. For certain operations, block H is temporarily lowered. This prevents the motion of latch G, resulting in the disengagement of the clutch after part of the cycle. It remains disengaged until the subsequent raising of block H permits the motion of latch G and the resumption of the cycle.

Fig. 3 Planetary transmission clutch. This is a positive clutch with external control. Two gear trains provide a bi-directional drive to a calculator for cycling the machine and shifting the carriage. Gear A is the driver; gear L, the driven member, is directly connected to the planet carrier F. The planet consists of integral gears B and C. Gear B meshes with free-wheeling gear D. Gears D and G carry projecting lugs E and H, respectively. Those lugs can contact formings on arms J and K of the control yoke.

When the machine is at rest, the yoke is centrally positioned so that arms J and K are out of the path of the projecting lugs, permitting both D and G to free-wheel. To engage the drive, the yoke rocks clockwise, as shown, until the forming on arm K engages lug H, blocking further motion of ring gear G. A solid gear train is thereby established, driving F and L in the same direction as the drive A. At the same time, the gear train alters the speed of D as it continues counterclockwise. A reversing signal rotates the yoke counterclockwise until arm J encounters lug E, blocking further motion of D. This actuates the other gear train with the same ratio.

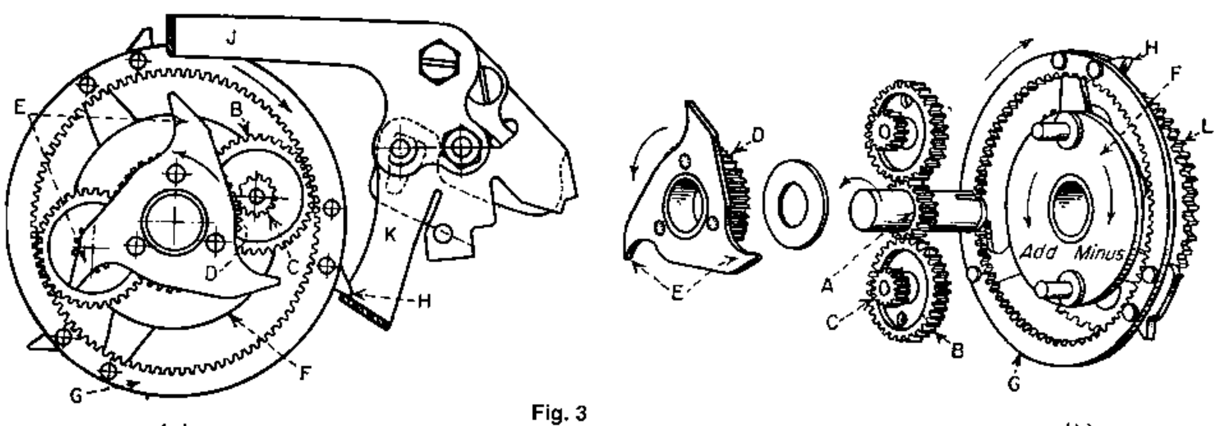


Fig. 3

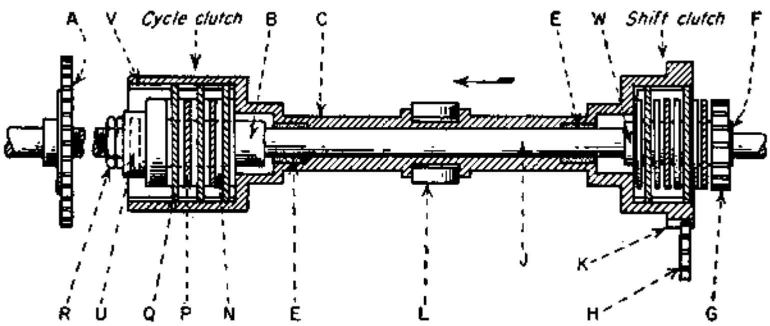


Fig. 4

Fig. 4 A multiple-disk friction clutch. Two multiple-disk friction clutches are combined in a single, two-position unit that is shown shifted to the left. A stepped cylindrical housing, C, encloses both clutches. Internal self-lubricated bearings support the housing on coaxial shaft J that is driven by transmission gear H, meshing with housing gear teeth K. At the other end, the housing carries multiple metal disks Q that engage keyways V and can make frictional contact with phenolic laminate disks N. They, in turn, can contact a set of metal disks P that have slotted openings for couplings with flats located on sleeves B and W.

In the position shown, pressure is exerted through rollers L, forcing the housing to the left, making the left clutch compress against adjusting nuts R. Those nuts drive gear A through sleeve B, which is connected to jack shaft J by pin U. When the carriage is to be shifted, rollers L force the housing to the right. However, it first relieves the pressure between the adjoining disks on the left clutch. Then they pass through a neutral position in which both clutches are disengaged, and they finally compress the right clutch against thrust bearing F. That action drives gear G through sleeve W, which rotates freely on the jack shaft.

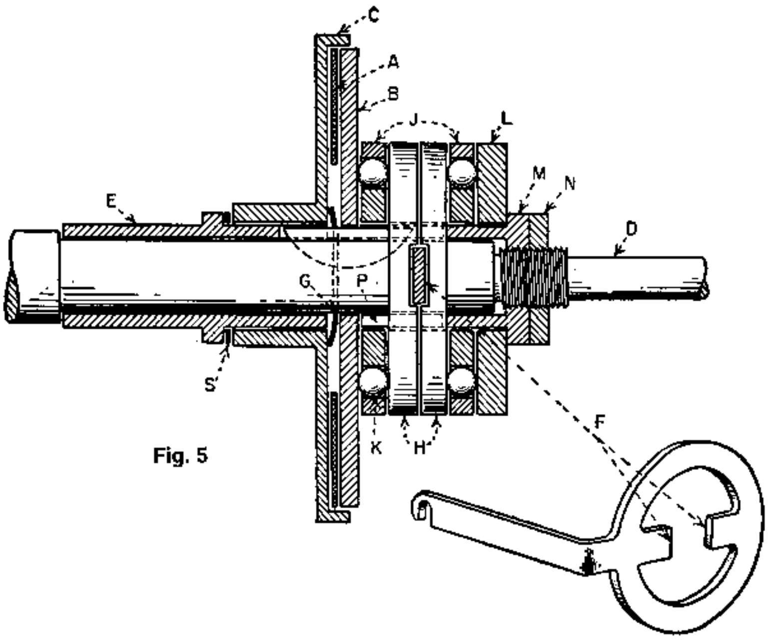


Fig. 5

Fig. 5 A single-plate friction clutch. The basic parts of this clutch are the phenolic laminate clutch disk A, steel disk B, and drum C. They are normally kept separated by spring washer G. To engage the drive, the left end of a control arm is raised, causing ears F, which are located in slots in plates H, to rock clockwise. This action spreads the plates axially along sleeve P. Sleeves E and P and plate B are keyed to the drive shaft; all other members can rotate freely.

The axial motion loads the assembly to the right through the thrust ball bearings K against plate L and adjusting nut M. It also loads them to the left through friction surfaces on A, B, and C to thrust washer S, sleeve E, and against a shoulder on shaft D. This response then permits phemolic laminate disk A to drive drum C.

Fig. 6 An overload relief clutch. This is a simple, double-plate, friction coupling with spring loading. Shaft G drives collar E, which drives slotted plates C and D faced with phenolic laminate disks B. Spring H is held in compression by the two adjusting nuts on the threaded end of collar E. These maintain the unit under axial pressure against the shoulder at the left end of the collar.

This enables the phenolic laminate disks B to drive through friction against both faces of the gar, which is free to turn o the collar. This motion of the gear causes output pinion J to rotate. If the machine to which the clutch is attached should jam and pinion J is prevented from turning, the motor can continue to run without overloading. However, slippage can occur between the phenolic laminate clutch plates B and the large gear.

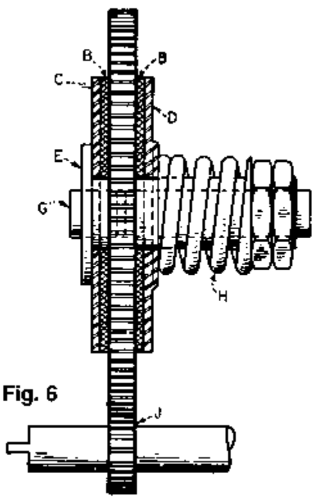
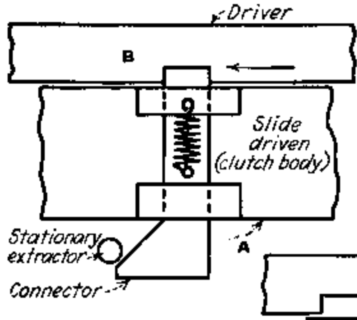


Fig. 6

MECHANISMS FOR STATION CLUTCHES

Innumerable variations of these station clutches can be designed for starting and stopping machines at selected points in their operation cycles.



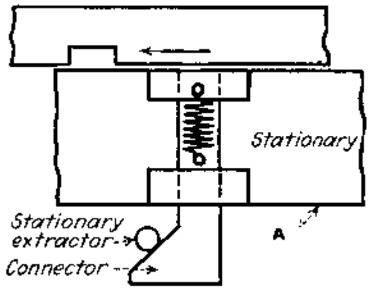


Fig. 1(*a*) The driver and clutch body move in unison with the connector just coming in contact with the extractor. In **Fig. 1(*b*)** continued movement withdraws the connector.

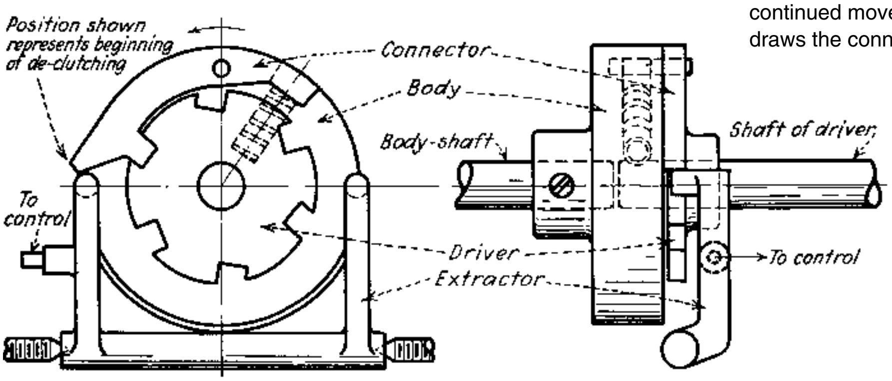


Fig. 2 A two-station clutch whose stations are 180º apart. Because it has only one extractor arm, this mechanism can function as a one-station clutch.

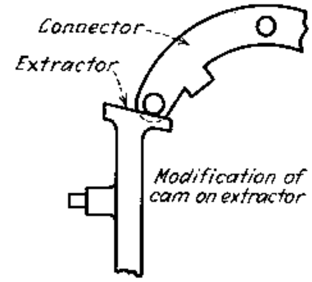


Fig. 3 A modification of the cam extractor shown in Fig. 2.

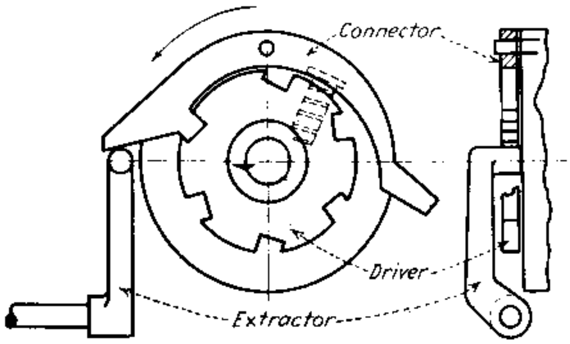


Fig. 4 A single extractor two-station clutch with the stations that are 180º apart. Only one extractor is required because the connector has two cams.

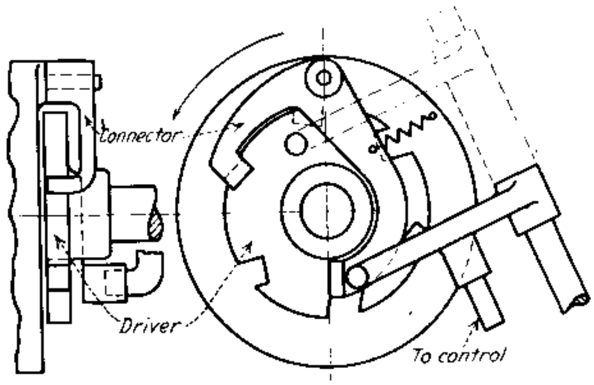


Fig. 5 This one- or two-station clutch with a dual extractor is compact because there are no parts projecting beyond its body.

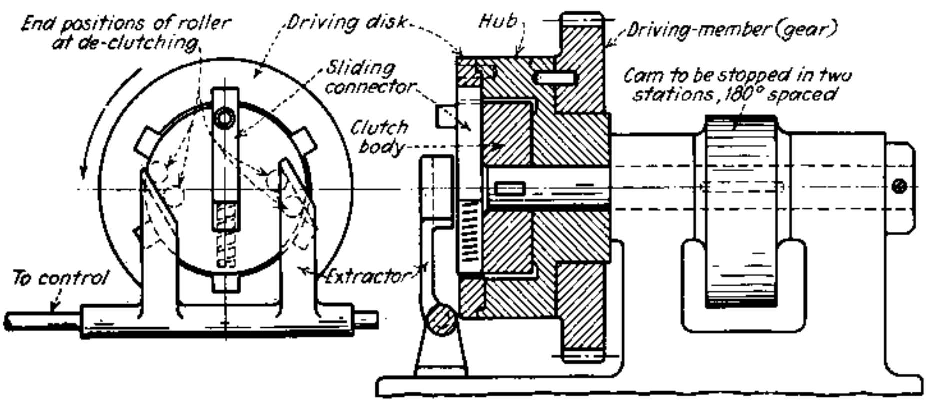


Fig. 6 The end and longitudinal section of a station clutch with internal driving recesses.

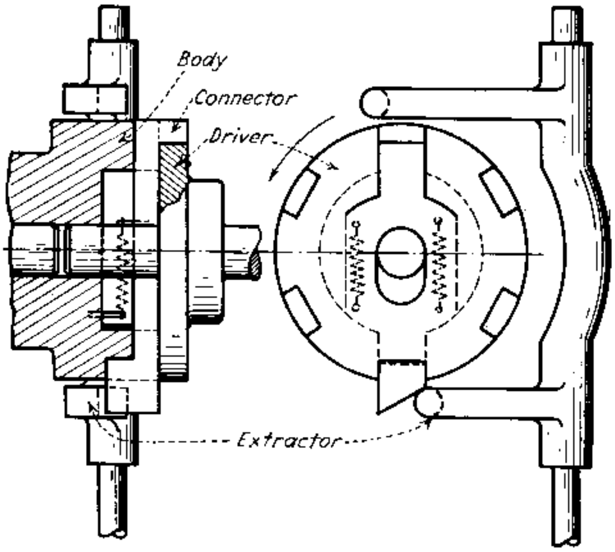


Fig. 7 This one- or two-station clutch depends on a single or a dual extractor. Its stations are spaced 180º apart.

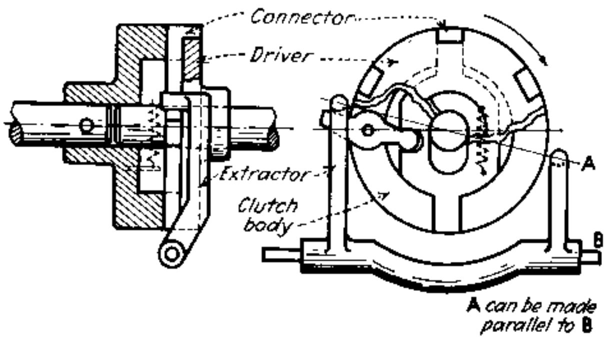


Fig. 8 This is another one- or two-station clutch. It has a single or dual extractor with stations spaced 180º apart.

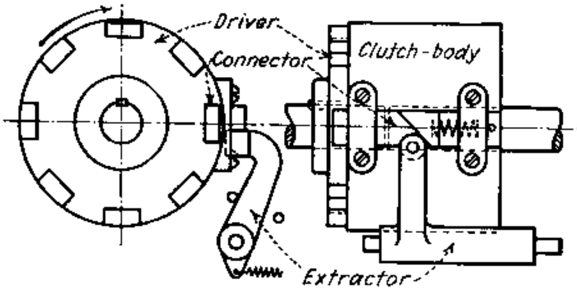
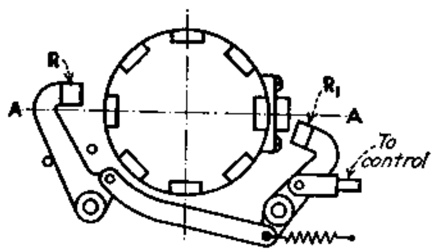


Fig. 9 A one-station axial connector clutch.

Fig. 10 A two-station clutch. The rollers R and R_1 of the extractor can also be arranged on the center-line A-A.



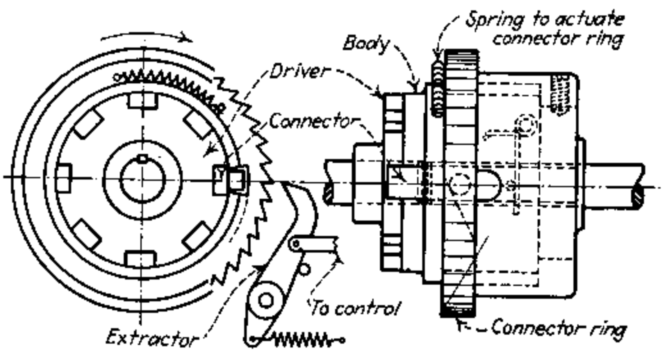
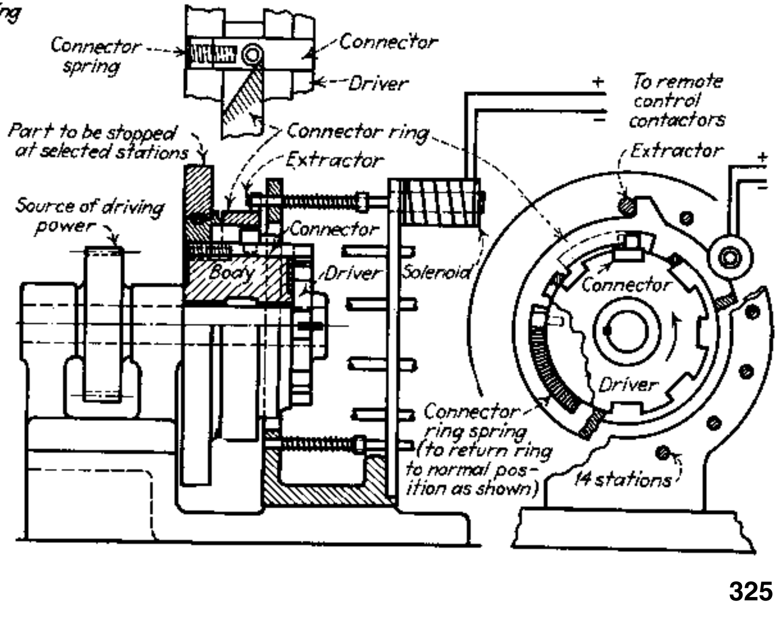


Fig. 11 A nonselective multistation clutch for instantaneous stopping in any position.

Fig. 12 A multistation clutch with remote control. The extractor pins are actuated by solenoids that either hold the extractor pin in position against spring pressure or release the pin.



TWELVE APPLICATIONS FOR ELECTROMAGNETIC CLUTCHES AND BRAKES

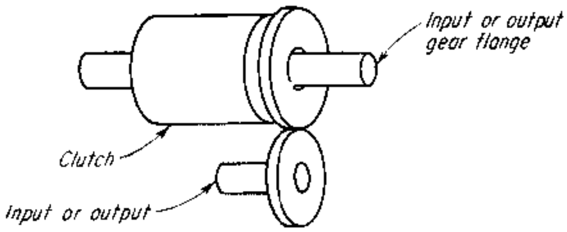


Fig. 1 Coupling or uncoupling power or sensing device.

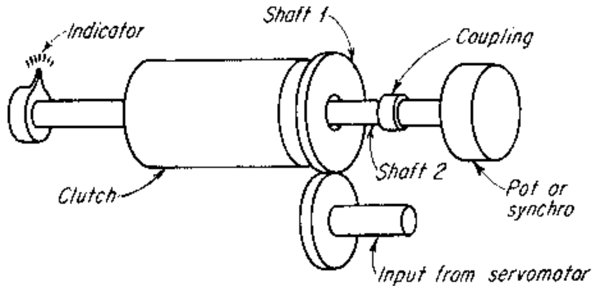
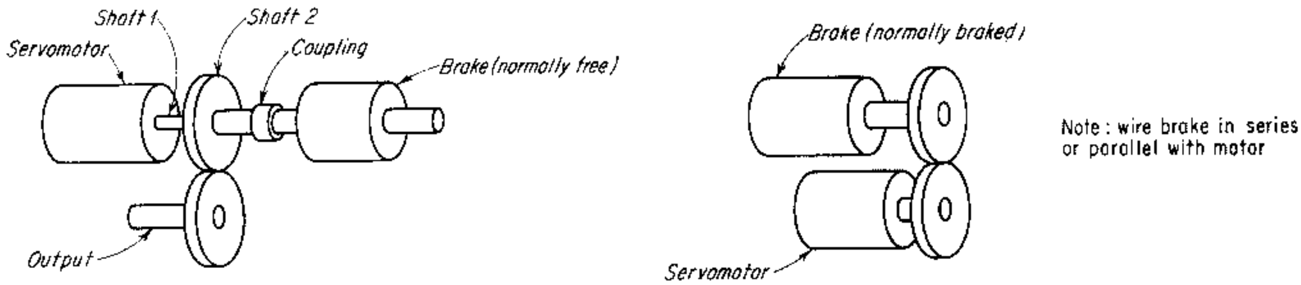


Fig. 2 Calibration protection (energize to adjust).



Figs. 3 & 4 Simple servomotor brakes.

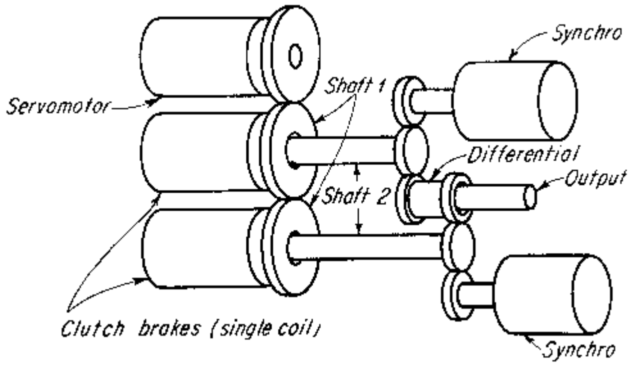


Fig. 5 Adding or subtracting two inputs.

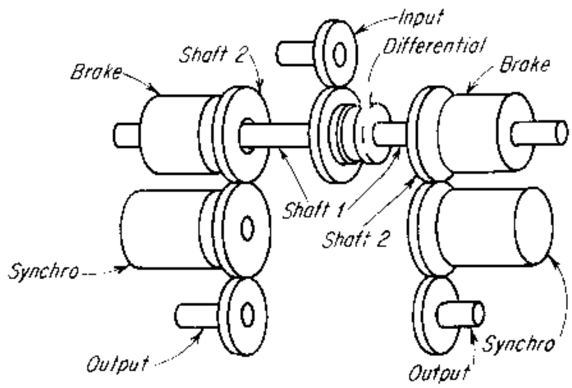


Fig. 6 Controlling output from a differential.

Magnetic Friction Clutches

The simplest and most adaptable electromagnetic control clutch is the magnetic friction clutch. It works on the same principle as a simple solenoid-operated electric relay with a spring return to normal. Like the relay, it is a straightforward automatic switch for controlling the flow of power (in this cases, torque) through a circuit.

Rotating or Fixed Field?

This is a question primarily of magnetic design. Rotating-field clutches include a rotating coil, energized through brushes and slip rings. Fixed-field units have a stationary coil. Rotating-field units are still more common, but there has been a marked trend toward the fixed-field versions.

Generally speaking, a *rotating-field clutch* is a two-member unit, with the coil carried in the driving (input) member. It can be mounted directly on a motor or speed-reducer shaft without loading down the driving motor. In the smaller sizes, it offers a better ratio of size to rated output than the fixed-field type, although the rotating coil increases inertia in the larger models.

A *fixed-field clutch,* on the other hand, is a three-member unit with rotating input and output members and a stationary coil housing. It eliminates the need for brushes and slip rings, but it demands additional bearing supports, and it can require close tolerances in mounting.

Purely Magnetic Clutches

Probably less familiar than the friction types are *hysteresis* and *eddy-current clutches.* They operate on straight magnetic principles and do not depend on mechanical contact between their members. The two styles are almost identical in construction, but the magnetic segments of the hysteresis clutch are electrically isolated, and those of the eddy-current clutch are interconnected. The magnetic analogy of both styles is similar in that the flux is passed between the two clutch members.

Hysteresis Clutches

The hysteresis clutch is a proportional-torque control device. As its name implies, it exploits the hysteresis effect in a permanent-magnet rotor ring to produce a substantially constant torque that is almost completely independent of speed (except for slight,

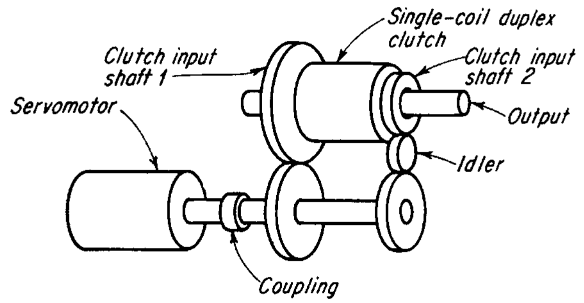


Fig. 7 Simple-speed changing.

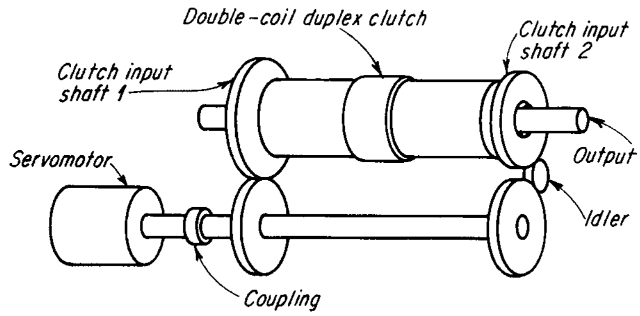


Fig. 8 Speed-changing and uncoupling.

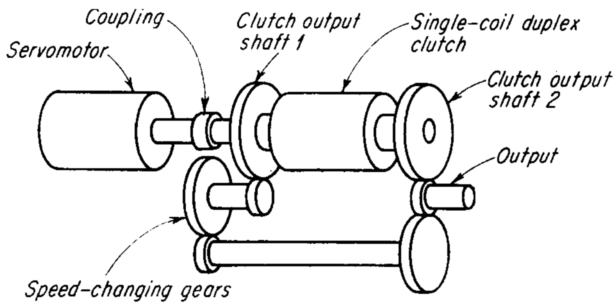


Fig. 9 Simple direction-changing.

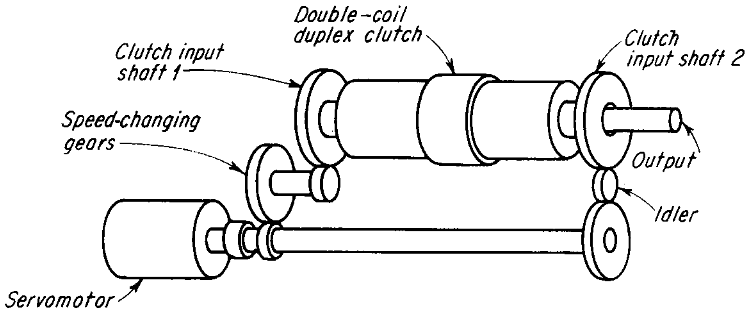


Fig. 10 Direction-changing and uncoupling.

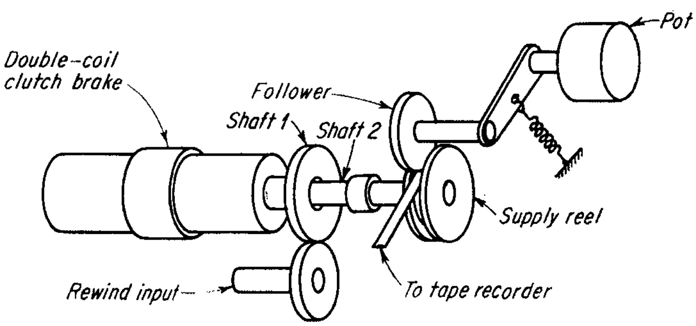


Fig. 11 Constant tensioning.

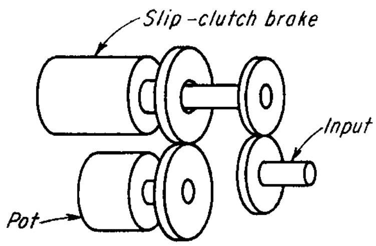


Fig. 12 Potentiometer control.

unavoidable secondary eddy-current torques—which do not seriously reduce performance). It is capable of synchronous driving or continuous slip, with almost no torque variation at any slip differential for a given control current. Its control-power requirement can be met by a transistor drive. Typical applications include wire or tape tensioning, servo-control actuation, and torque control in dynamometers.

Eddy-Current Clutches

Eddy-current clutches on the other hand, are inherently speed-sensitive devices. They exhibit virtually no hysteresis, and develop torque by dissipating eddy currents through the electrical resistance of the rotor ring. This torque is almost a linear function of slip speed. These clutches perform best in speed-control applications, and as oscillation dampers.

Particle and Fluid Magnetic Clutches

There is no real difference between *magnetic-particle* and *magnetic-field clutches.* However, the magnetic medium in the particle clutch is a dry powder; in the fluid clutch it is a powder suspended in oil. In both clutches the ferromagnetic medium is introduced into the airgap between the input and output faces, which do not actually contact one another. When the clutch coil is energized, the particles are excited in the magnetic field between the faces; as they shear against each other, they produce a drag torque between the clutch members.

Theoretically, those clutches can approach the proportional control characteristics of a hysteresis clutch within the small weight and size limits of a comparably rated miniature friction clutch. But in practice, the service life of miniature magnetic-particle clutches has so far been too short for industrial service.

Other Magnetic Clutches

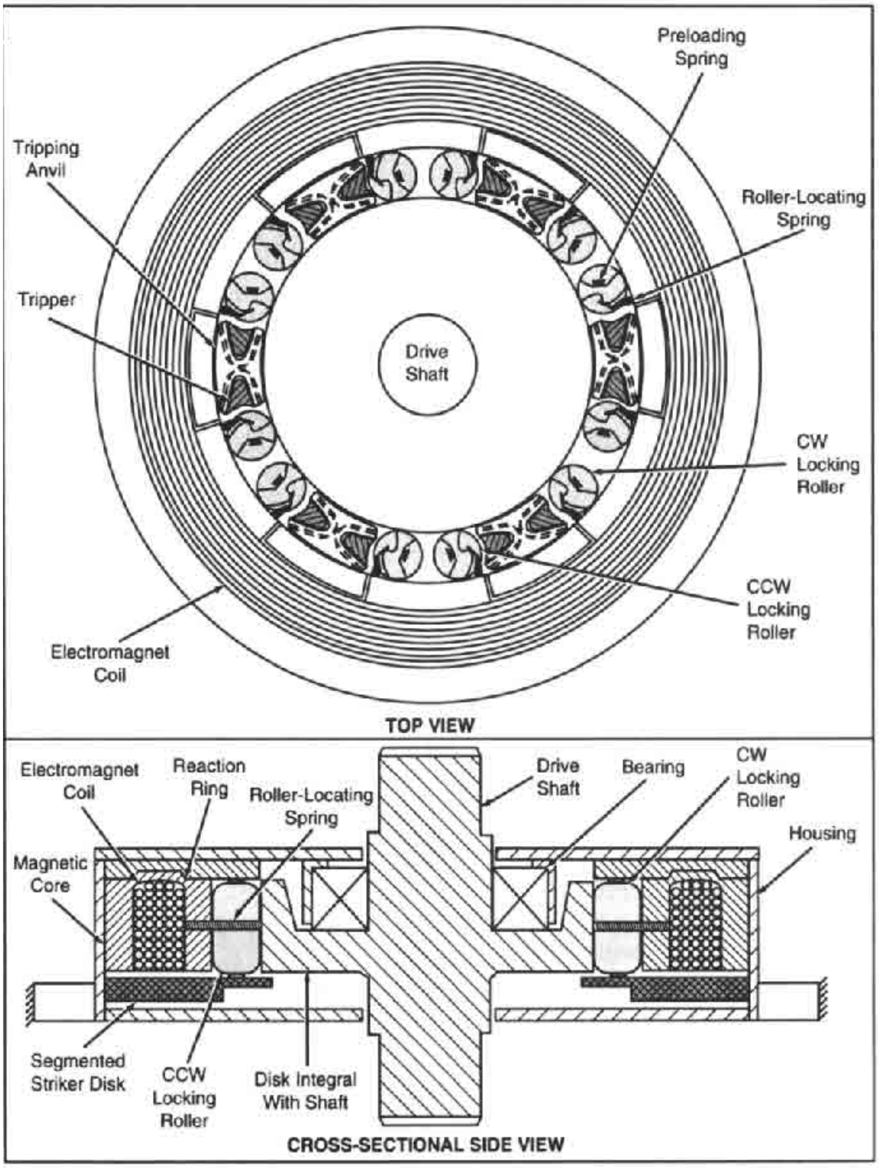
Two sophisticated concepts—neither of them yet developed to the point of practical application—might be of interest to anyone researching this field.

Electrostatic clutches depend on high voltages instead of a magnetic field to create force-producing suspensions.

Magnetostrictive clutches depend on a magnetic force to change the dimensions of a crystal or metal bar poised between two extremely precise facts.

TRIP ROLLER CLUTCH

This bidirectional roller-locking clutch offers the advantages of efficiency and controllability.



The trip roller clutch contains back-to-back roller-locking, overrunning clutches that can be released (tripped) with small magnetic forces.

The figure is a simplified cross-sectional view of an electromagnetically releasable roller-locking mechanism that functions as a brake or clutch in clockwise or counterclockwise rotation. In essence, the mechanism contains two back-to-back overrunning clutches such as those that are commonly used in industry to roll freely in one direction and lock against rolling in the opposite direction. In addition to bidirectionality, the novel design of this mechanism offers advantages of efficiency and controllability over older clutches and brakes.

As in other roller-locking mechanisms, lock is achieved in this mechanism by jamming rollers between a precise surface on one rotating or stationary subassembly (in this case, the inner surface of a reaction ring in a housing) and a precise surface on another rotating or stationary subassembly (in this case, the outer surface of a disk integral with a drive shaft). There are two sets of rollers: CW and CCW locking. They feature cam surfaces that jam against the disk and reaction ring in the event of clockwise and counterclockwise rotation, respectively. The mechanism is called a "trip roller clutch" because of the manner in which the rollers are unjammed or tripped to allow rotation, as explained later.

The rollers are arranged in pairs around the disk and the reaction ring. Each pair contains one CW and one CCW locking roller. A tripping anvil fixed to the reaction ring is located between the rollers in each pair. Each roller is spring-loaded to translate toward a prescribed small distance from the tripping anvil and to rotate toward the incipient-jamming position. In the absence of any tripping or releasing action, the clutch remains in lock; that is, any attempt at clockwise or counterclockwise rotation of the drive shaft result sin jamming of the CW or CCW rollers, respectively.

Release is effected by energizing the electromagnet coil in the housing. The resulting magnetic force pulls a segmented striker disk upward against spring bias. Attached to each segment of the striker disk is a tripper, which slides toward a CW or CCW roller on precisely angled surfaces in the tripping anvil. Each tripper then pushes against its associated CW or CCW locking roller with a small blocking force. But, in blocking the locking roller, the locking cam angles are effectively increased and slipping (followed by release) occurs. Thus, the clutch is "tripped" out of lock into release.

Very little force is needed for this releasing action, even though the forces in lock can be very large. Because the gap between the striker plate and the magnetic core is zero or very small during release, very little magnetic force is needed to maintain release. Thus, the electromagnet coil and the power is consumes can be made smaller than in comparable prior mechanisms, with a corresponding gain in power efficiency and decrease in size and in weight. To lock the clutch, one simply turns off the electromagnet, allowing the springs to retract the trippers and restore the rollers to the incipient-jamming position.

The excellent frequency response and high mechanical efficiency, inherent in roller locking, enable the trip roller clutch to be lockable and releasable precisely at a desired torque under sensory interactive computer control. For the same reasons, the trip roller clutch can be opened and closed repeatedly in a pulsating manner to maintain precise torque(s) or to effect release under impending slip, as in an automative antilock braking system. It operates more predictably than do other friction-based clutches in that its performance is not disturbed when lubricant is dropped on it; indeed, it is designed to operate with lubricant. Also, unlike other friction-based devices, the trip roller clutch remains cool during operation.

This work was done by John M. Vranish of **Goddard Space Flight Center** *and supported by Honeybee Robotics, NY.*

GEARED ELECTROMECHANICAL ROTARY JOINT

Springy planetary gears provide low-noise electrical contact.

The figure illustrates a geared rotary joint that provides low-noise ac or dc electrical contact between electrical subsystems that rotate relative to each other. This joint is designed to overcome some of the disadvantages of older electromechanical interfaces—especially the intermittency (and, consequently, the electrical noise) of sliding-contact and rolling-contact electromechanical joints.

The firs electrical subsystem is mounted on, or at least rotates with, the shaft and the two inner gears attached to the shaft. The inner gears are separated axially by an electrically insulating disk. Each inner gear constitutes one of two electrical terminals through which electrical power is fed to or from the first electrical subsystem.

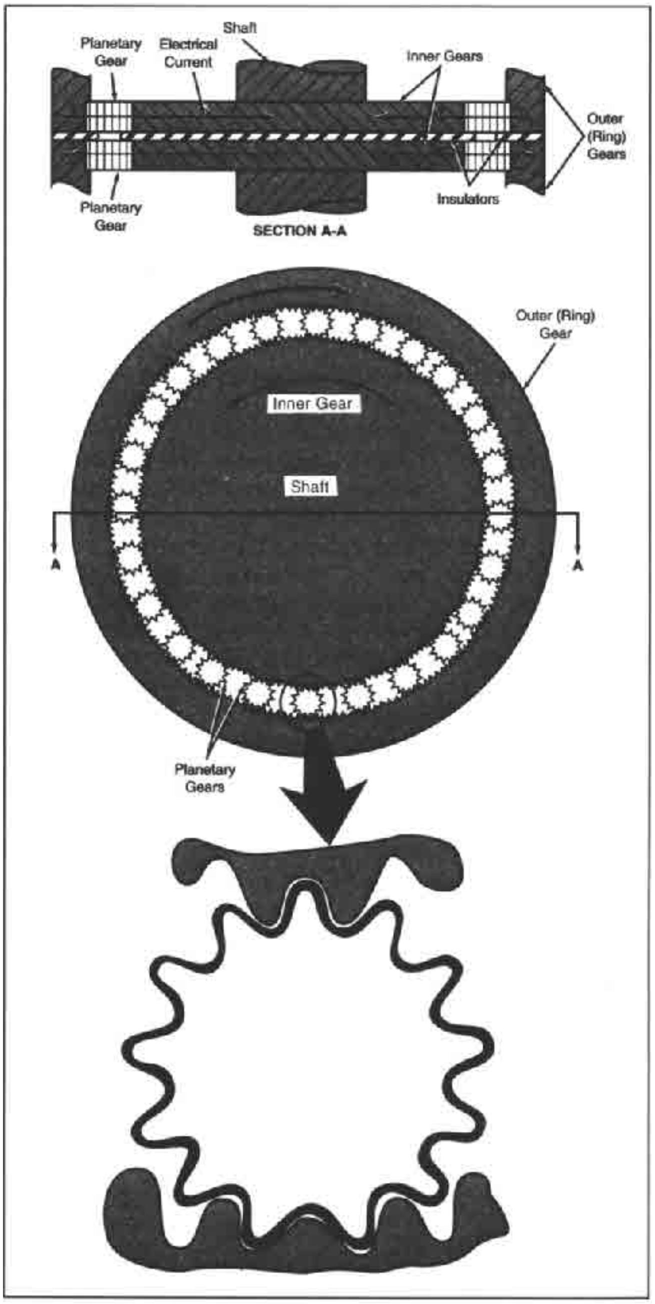
The second electrical subsystem is mounted on, or at least rotates with, the outer (ring) gears. As was done to the inner gears, the ring gears are separated axially by an electrically insulating annular disk. The ring gears act as the electrical terminals through which power is fed from or to the second electrical subsystem.

Electrical contact between the inner and outer (ring) gears is provided by multiple, equally spaced, flexible planetary gears formed as hollow cylinders with thin, fluted walls. These gears mesh with the inner and outer (ring) gears. Those gears are slightly oversize with respect to the gaps between the inner and outer gears, but their flexibility makes it possible to compress them slightly to install them in the gaps. After installation, meshing of the gears maintains the even angular interval between the planetary gears at all rotational speeds.

The planetary gears are made of beryllium copper, which is preferred for electrical contacts because it is a self-cleaning material that exhibits excellent current-carrying characteristics. A typical flexible planetary gear has 13 teeth. Both have an axial length and an average diameter of 0.25 in. (6.35 mm), and a wall thickness of 0.004 in. (0.10 mm). Because each planetary gear is independently sprung into a cylinder-in-socket configuration with respect to the inner and outer gears, it maintains continuous electrical contact between them. The reliability and continuity of the electrical contact is further ensured by the redundancy of the multiple planetary gears. The multiplicity of the contacts also ensures low electrical resistance and large current-carrying capability.

The springiness of the planetary gears automatically compensates for thermal expansion, thermal contraction, and wear; moreover, wear is expected to be minimal. Finally, the springiness of the planetary gears provides an antibacklash capability in a gear system that is simpler and more compact in comparison with conventional antibacklash gear systems.

This work was done by John M. Vranish of **Goddard Space Flight Center**.



Hollow, springy, planetary gears provide continuous, redundant, low-noise electrical contact between the ginner and outer gears.

TEN UNIVERSAL SHAFT COUPLINGS

Hooke's Joints

The commonest form of a universal coupling is a *Hooke's joint.* It can transmit torque efficiently up to a maximum shaft alignment angle of about 36°. At slow speeds, on hand-operated mechanisms, the permissible angle can reach 45°. The simplest arrangement for a Hooke's joint is two forked shaft-ends coupled by a cross-shaped piece. There are many variations and a few of them are included here.

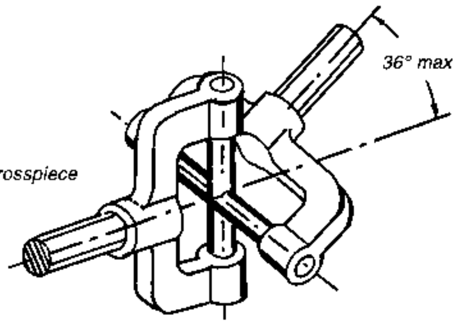


Fig. 1 The Hooke's joint can transmit heavy loads. Anti-friction bearings are a refinement often used.

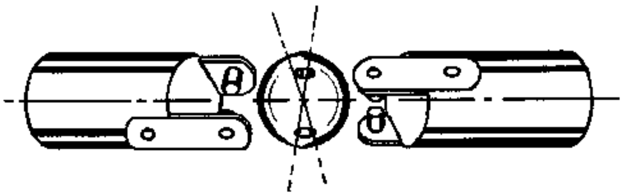


Fig. 2 A pinned sphere shaft coupling replaces a cross-piece. The result is a more compact joint.

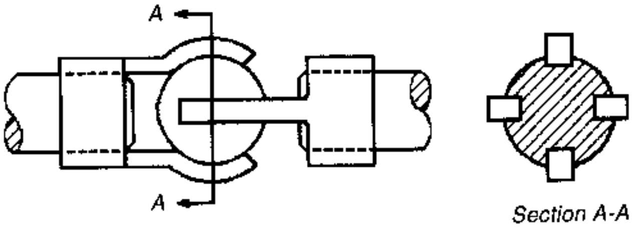


Fig. 3 A grooved-sphere joint is a modification of a pinned sphere. Torques on fastening sleeves are bent over the sphere on the assembly. Greater sliding contact of the torques in grooves makes simple lubrication essential at high torques and alignment angles.

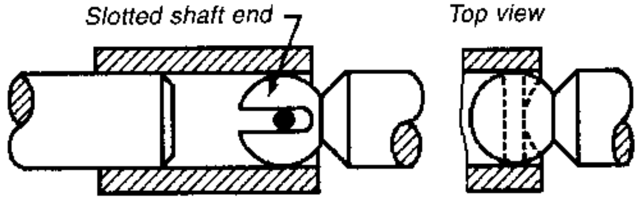


Fig. 4 A pinned-sleeve shaft-coupling is fastened to one shaft that engages the forked, spherical end on the other shaft to provide a joint which also allows for axial shaft movement. In this example, however, the angle between shafts must be small. Also, the joint is only suitable for low torques.

Constant-Velocity Couplings

The disadvantages of a single Hooke's joint is that the velocity of the driven shaft varies. Its maximum velocity can be found by multiplying driving-shaft speed by the secant of the shaft angle; for minimum speed, multiply by the cosine. An example of speed variation: a driving shaft rotates at 100 rpm; the angle between the shafts is 20°. The minimum output is 100×0.9397, which equals 93.9 rpm; the maximum output is 1.0642×100, or 106.4 rpm. Thus, the difference is 12.43 rpm. When output speed is high, output torque is low, and vice versa. This is an objectionable feature in some mechanisms. However, two universal joints connected by an intermediate shaft solve this speed-torque objection.

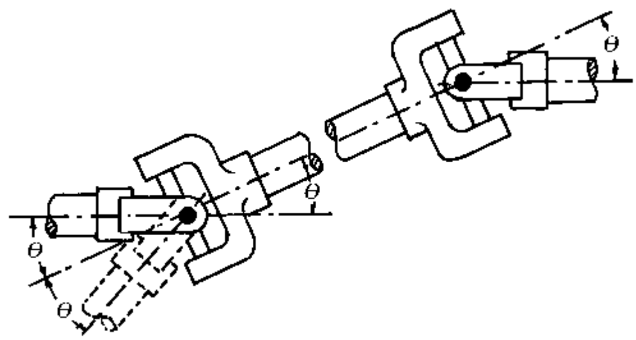


Fig. 5 A constant-velocity joint is made by coupling two Hooke's joints. They must have equal input and output angles to work correctly. Also, the forks must be assembled so that they will always be in the same plane. The shaft-alignment angle can be double that for a single joint.

This single constant-velocity coupling is based on the principle (Fig. 6) that the contact point of the two members must always lie on the homokinetic plane. Their rotation speed will then always be equal because the radius to the contact point of each member will always be equal. Such simple couplings are ideal for toys, instruments, and other light-duty mechanisms. For heavy duty, such as the front-wheel drives of military vehicles, a more complex coupling is shown diagrammatically in Fig. 7A. It has two joints close-coupled with a sliding member between them. The exploded view (Fig. 7B) shows these members. There are other designs for heavy-duty universal couplings; one, known as the *Rzeppa,* consists of a cage that keeps six balls in the homokinetic plane at all times. Another constant-velocity joint, the *Bendix-Weiss,* also incorporates balls.

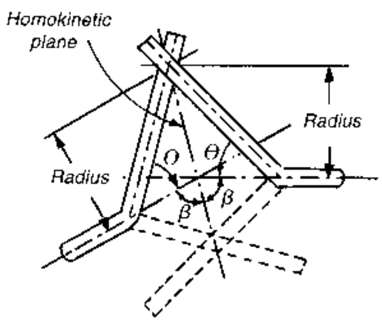


Fig. 6

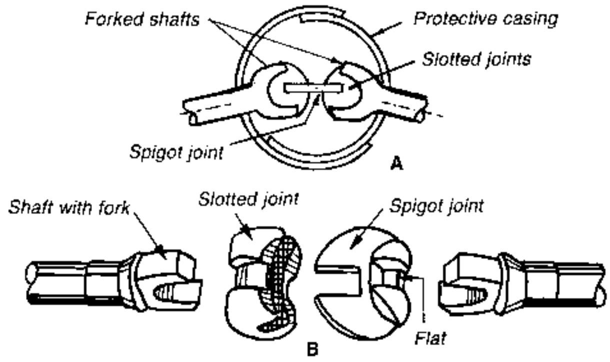


Fig. 7

Fig. 8 This flexible shaft permits any shaft angle. These shafts, if long, should be supported to prevent backlash and coiling.

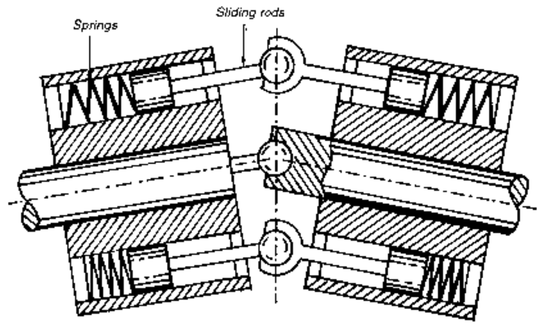


Fig. 9 This pump-type coupling has the reciprocating action of sliding rods that can drive pistons in cylinders.

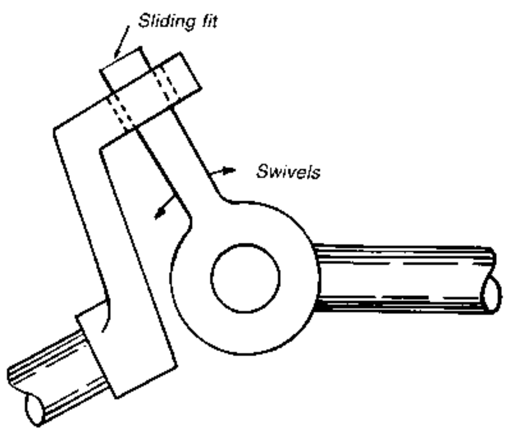


Fig. 10 This light-duty coupling is ideal for many simple, low-cost mechanisms. The sliding swivel-rod must be kept well lubricated at all times.

METHODS FOR COUPLING ROTATING SHAFTS

Methods for coupling rotating shafts vary from simple bolted flange assembles to complex spring and synthetic rubber assembles. Those including chain belts, splines, bands, and rollers are shown here.

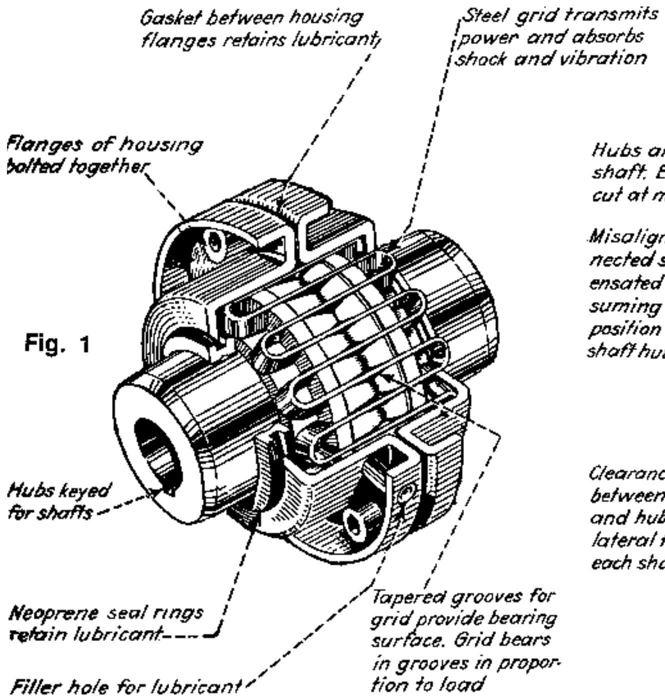


Fig. 1

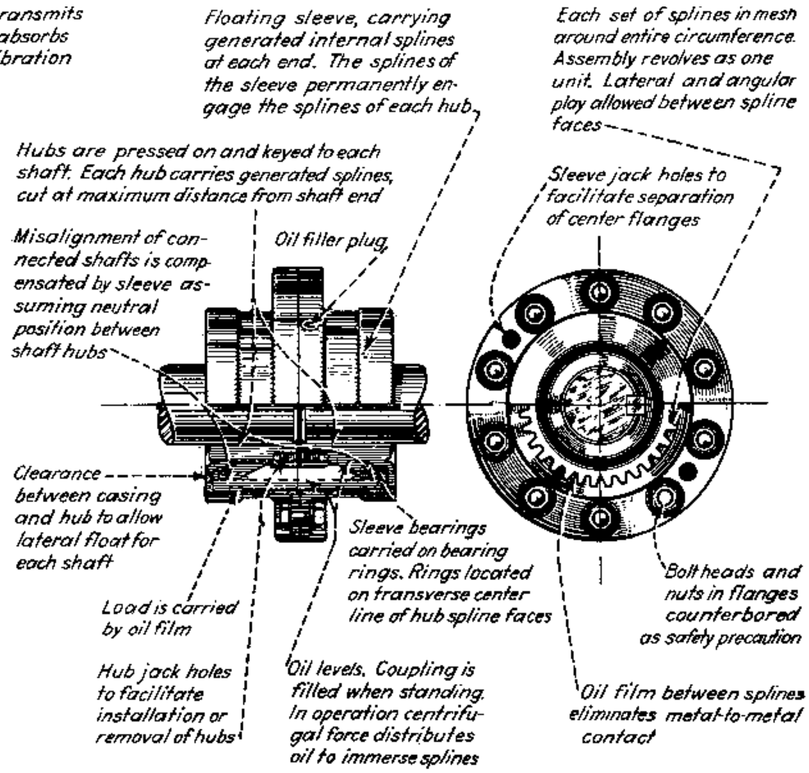


Fig. 2

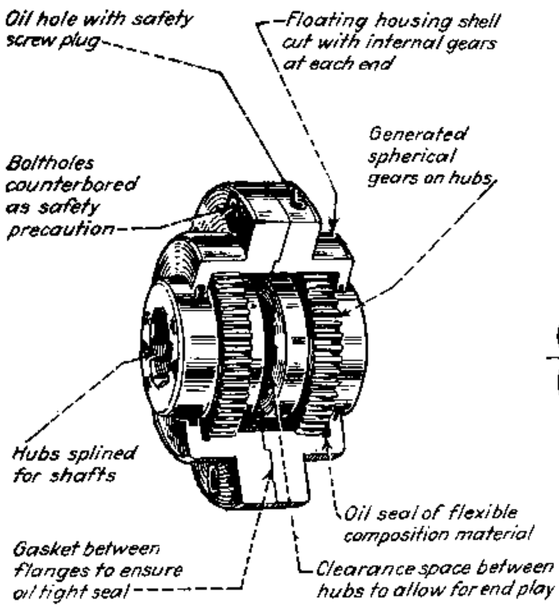


Fig. 3

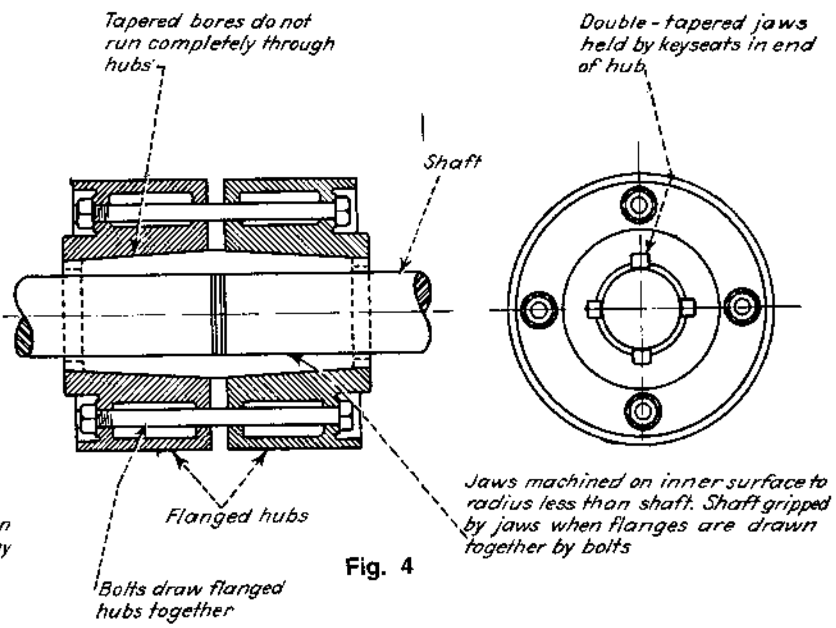


Fig. 4

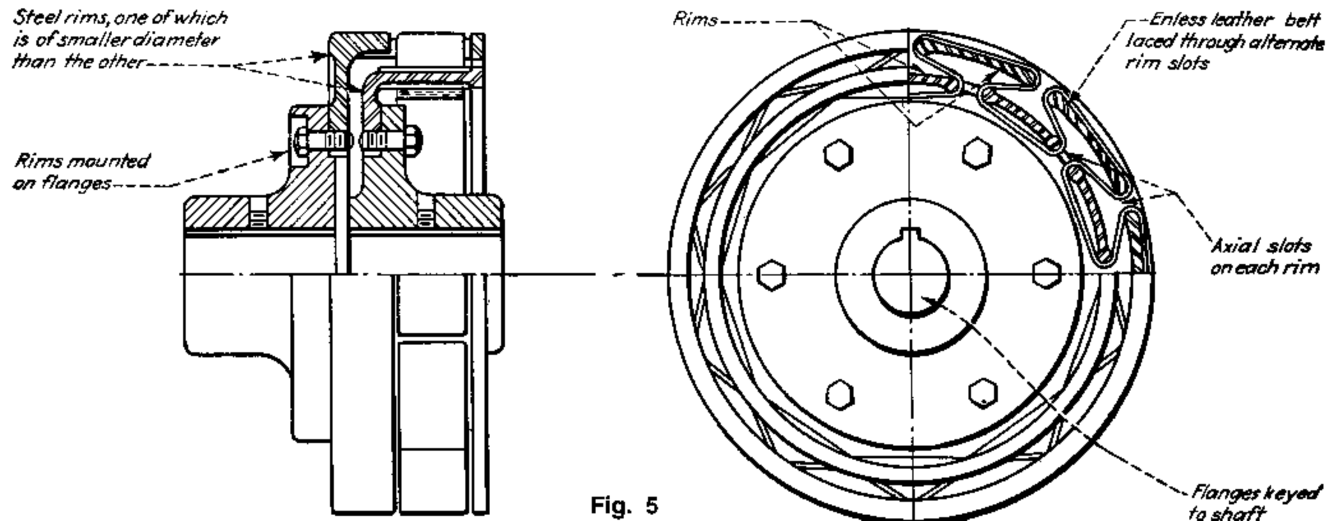


Fig. 5

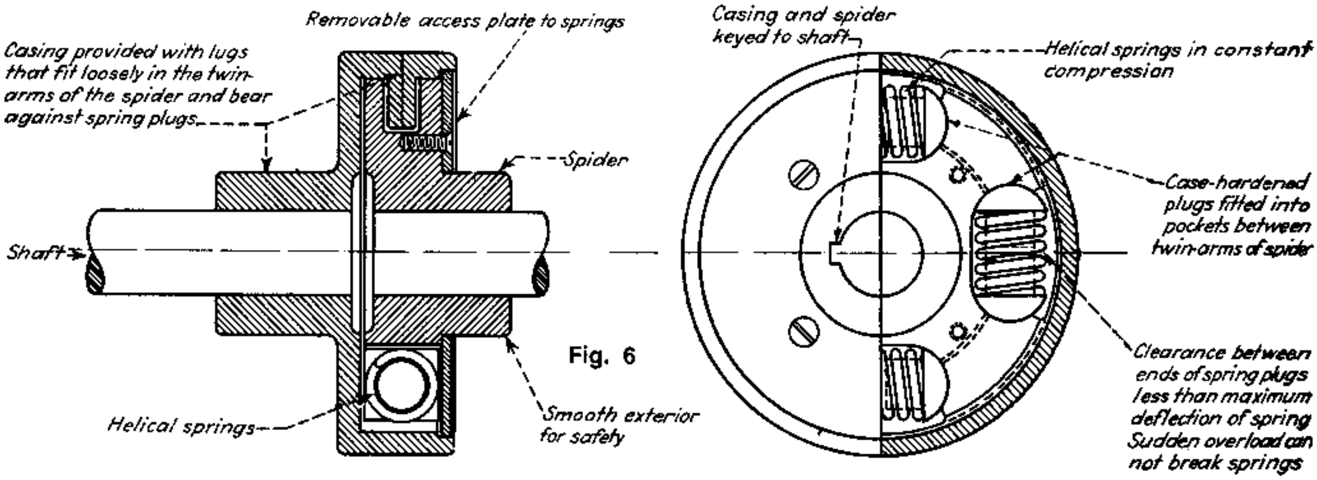


Fig. 6

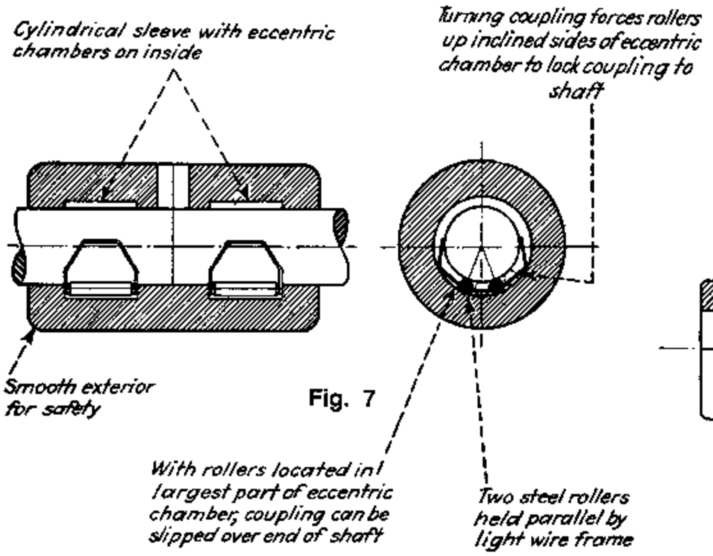


Fig. 7

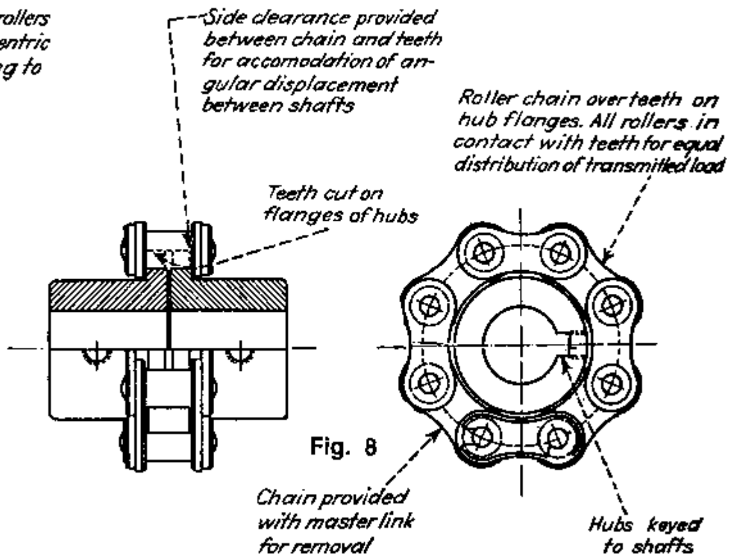


Fig. 8

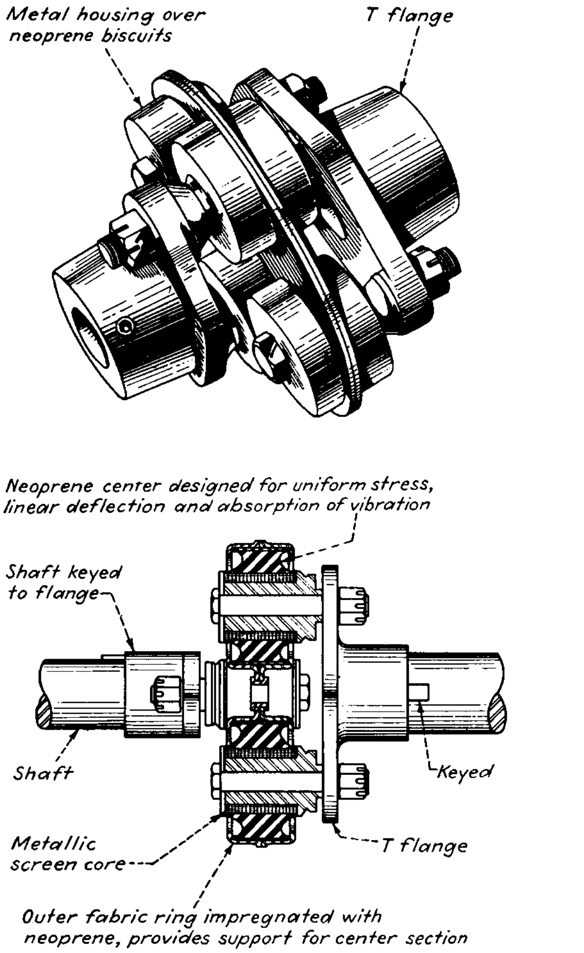


Fig. 9

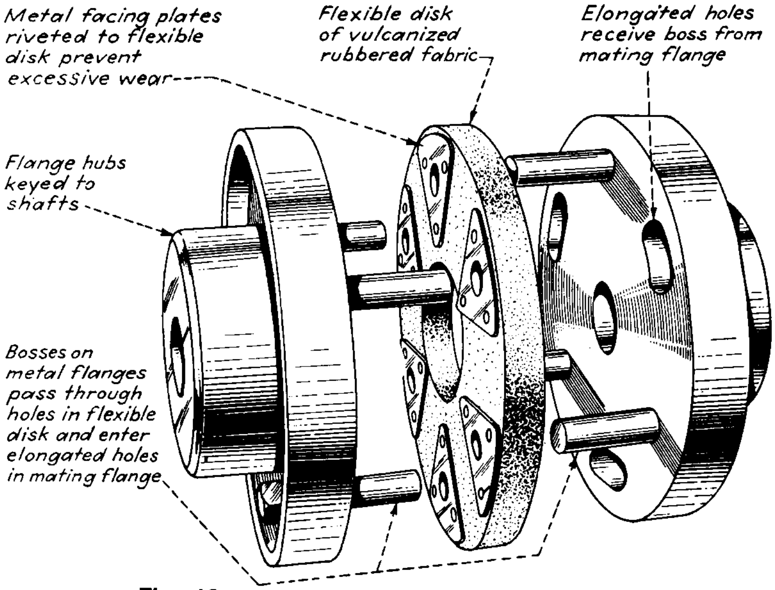


Fig. 10

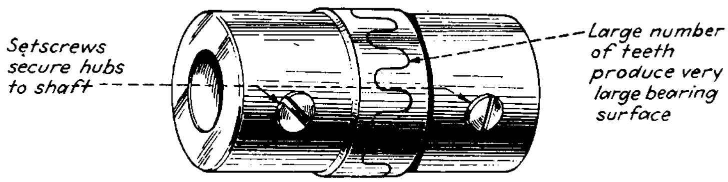


Fig. 11

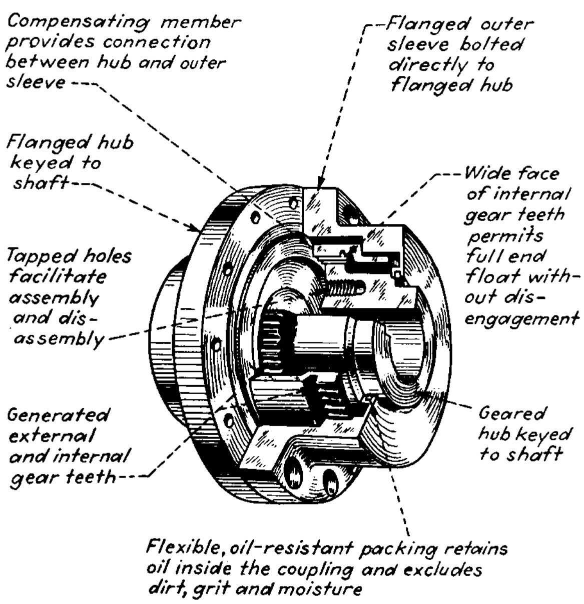


Fig. 12

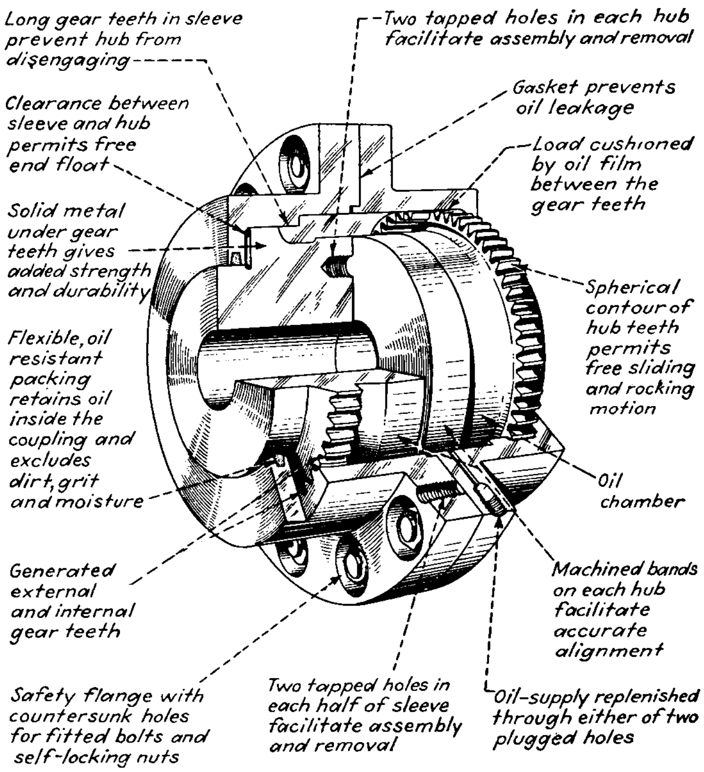


Fig. 13

Shaft couplings that include internal and external gears, balls, pins, and nonmetallic parts to transmit torque are shown here.

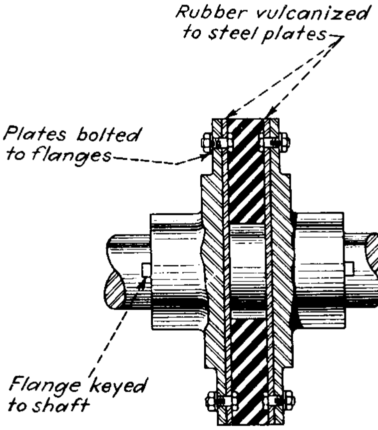


Fig. 14

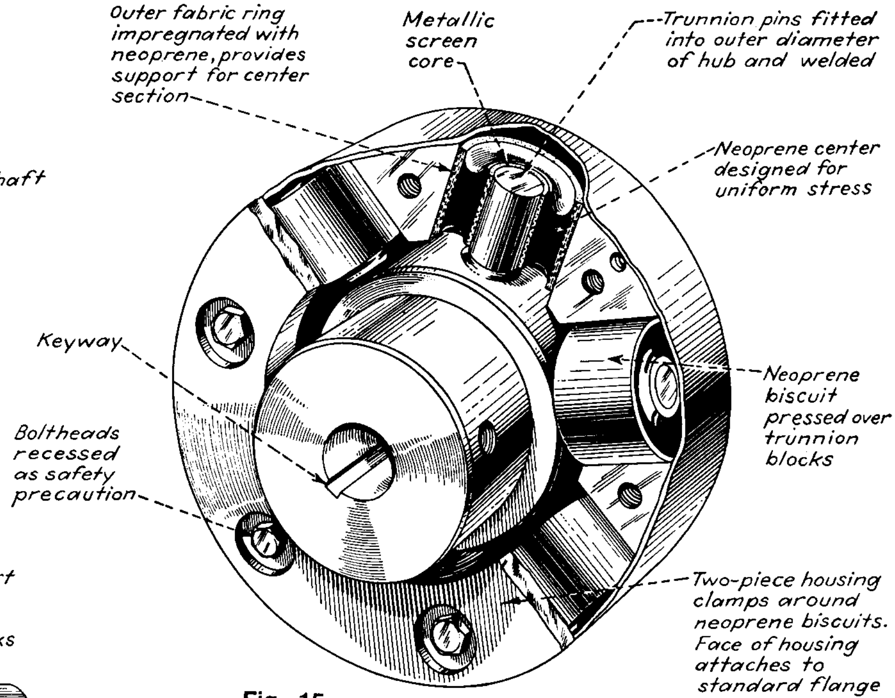


Fig. 15

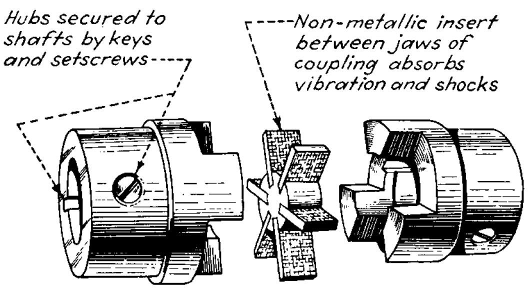


Fig. 16

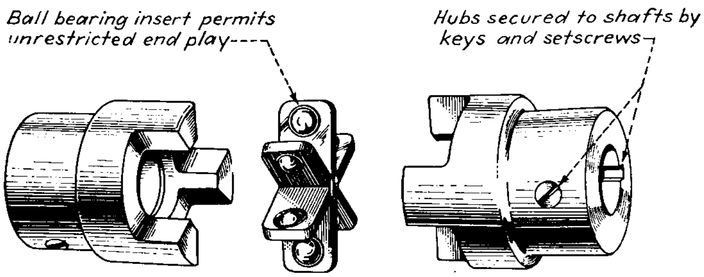


Fig. 17

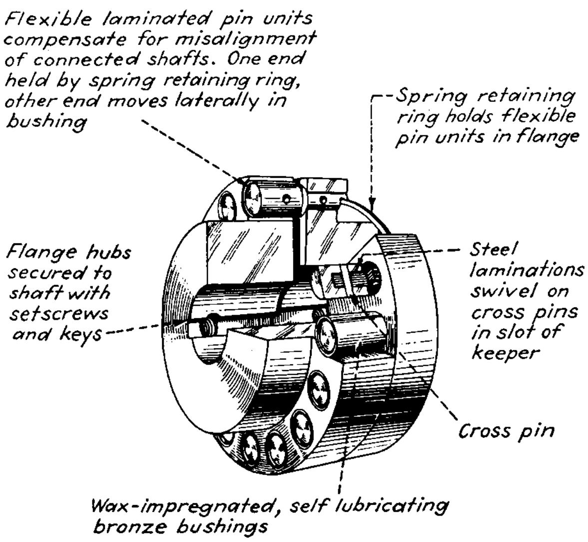


Fig. 18

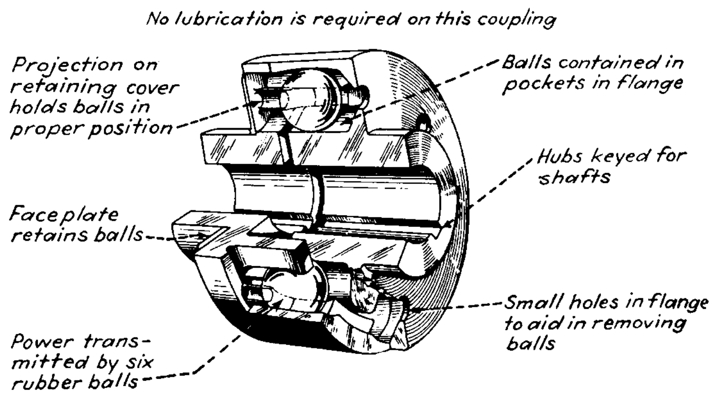


Fig. 19

LINKAGES FOR BAND CLUTCHES AND BRAKES

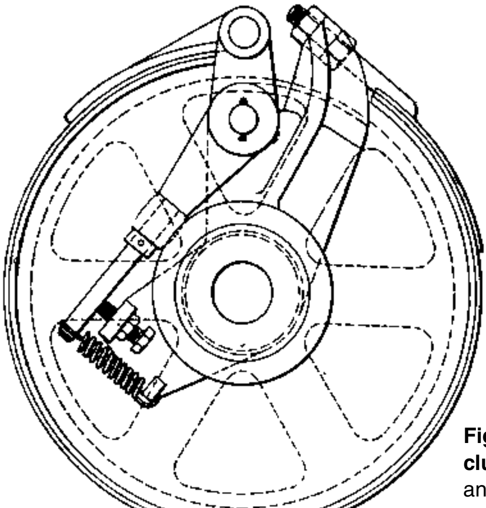


Fig. 1 An outside band clutch operated by a roller and cone.

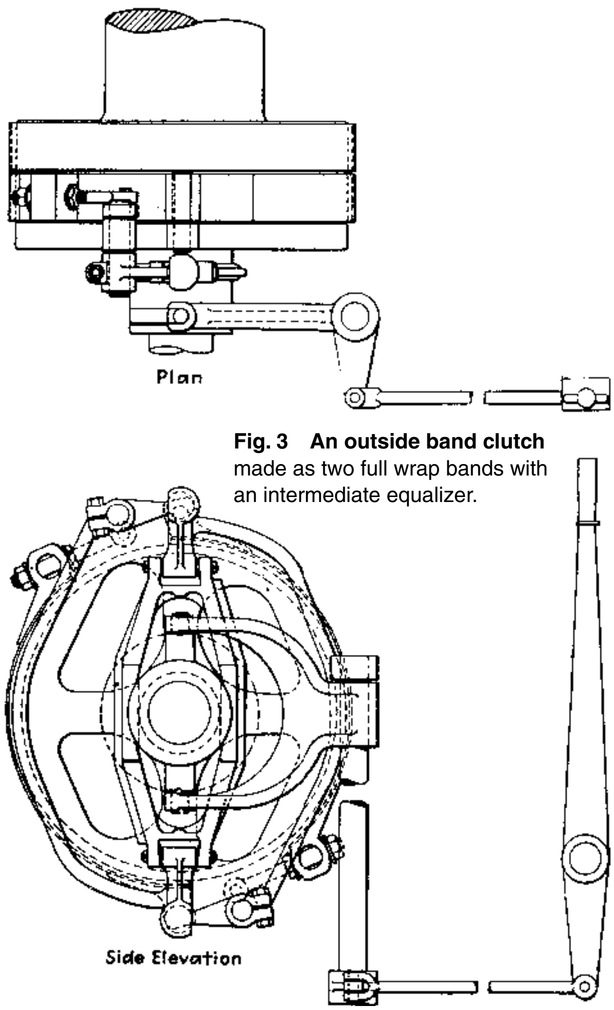


Fig. 3 An outside band clutch made as two full wrap bands with an intermediate equalizer.

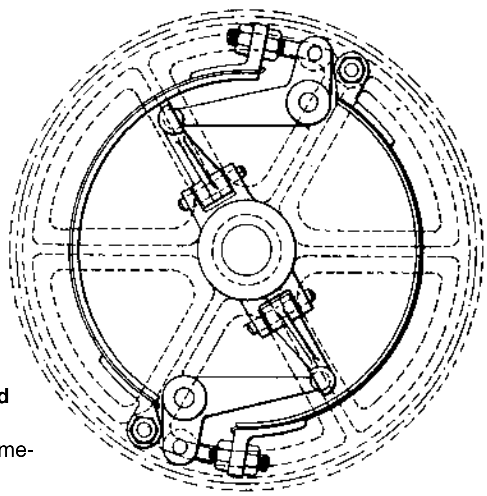


Fig. 2 An outside band clutch made as two half wrap bands with an intermediate equalizer.

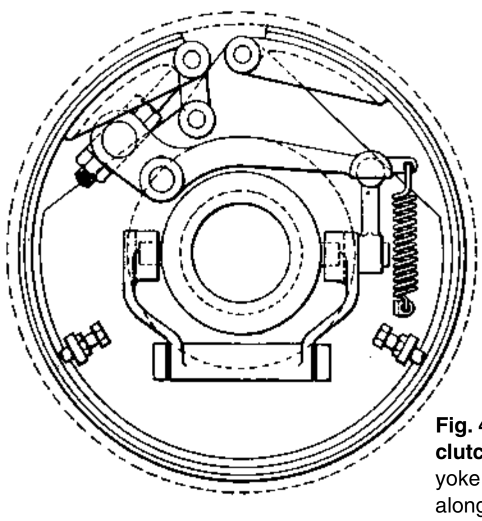


Fig. 4 An inside band clutch operated by a yoke having movement along the shaft.

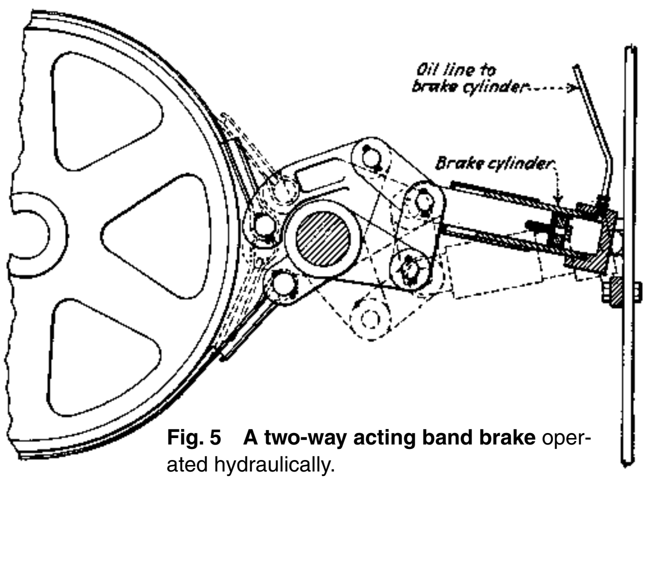


Fig. 5 A two-way acting band brake operated hydraulically.

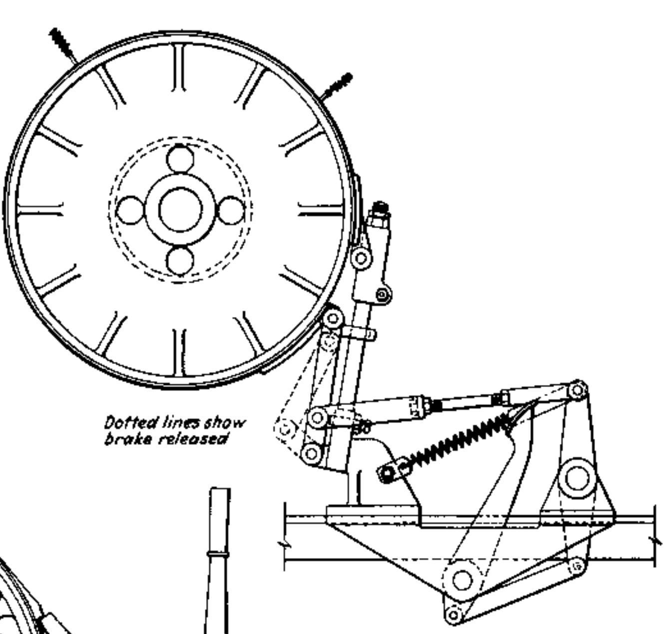


Fig. 6 **A hoist-drum band brake** operated by a foot pedal.

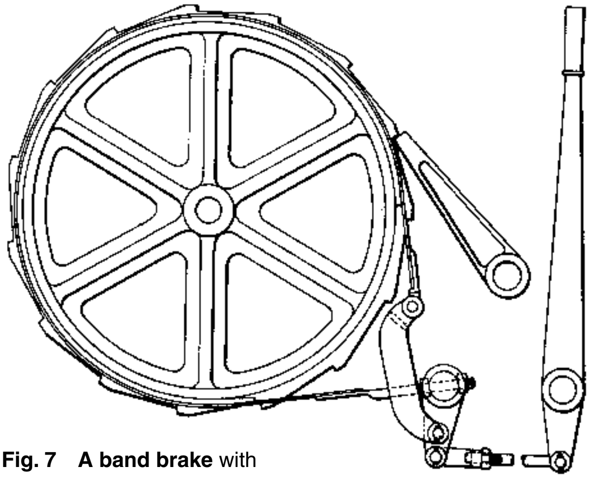


Fig. 7 **A band brake** with a single toggle action.

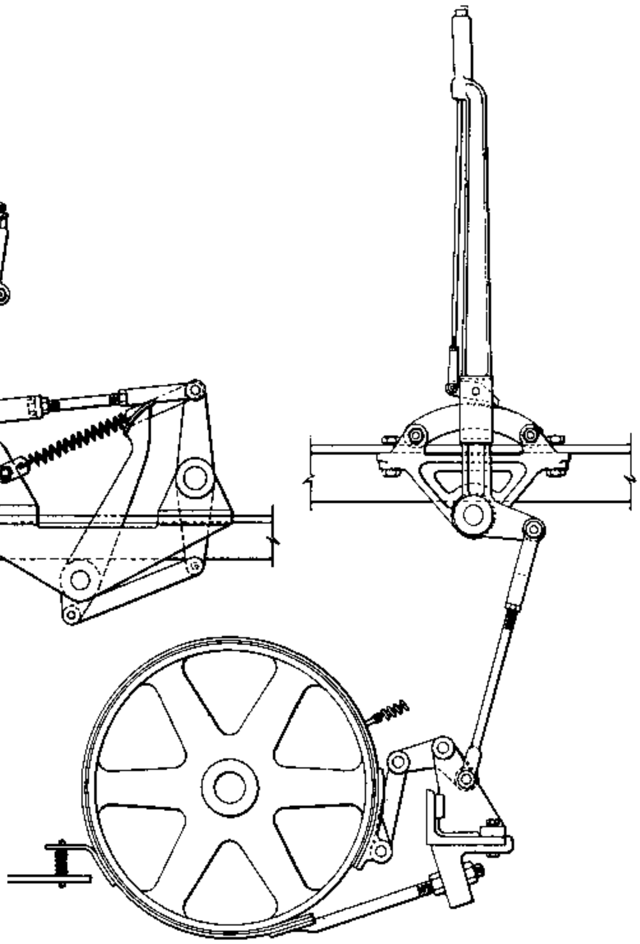


Fig. 8 **A crawler-drive band brake** operated by a ratchet lever.

SPECIAL COUPLING MECHANISMS

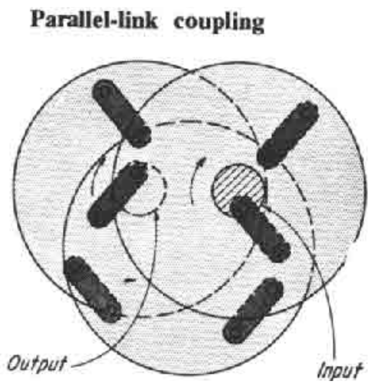


Fig. 1 **Six links and three disks** can synchronize the motion between adjacent, parallel shafts.

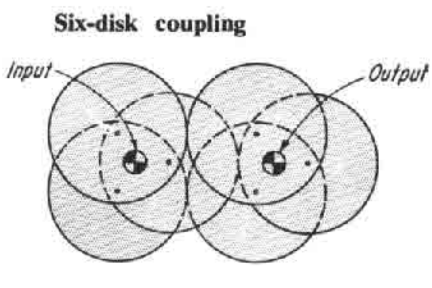


Fig. 2 **An arrangement** to synchronize shafts without the need for links.

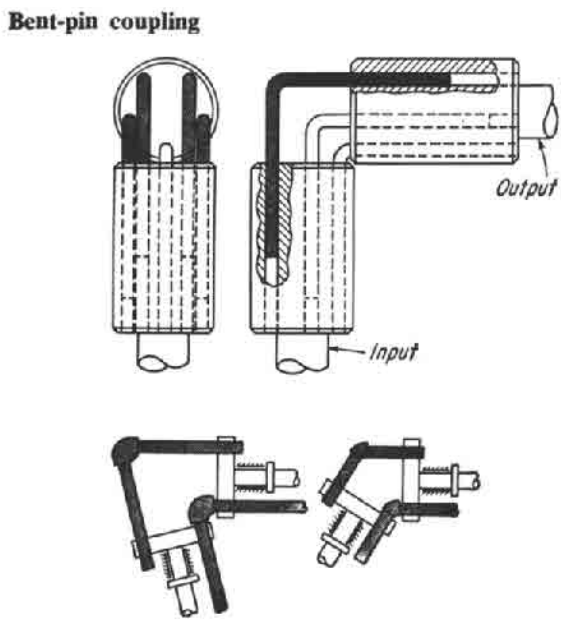


Fig. 3 **As the input rotates,** the five bent pins will move in and out of the drilled holes to impart a constant velocity rotation to the right angle-output shaft. The device can transmit constant velocity at angles other than 90°, as shown.

LINK COUPLING MECHANISMS

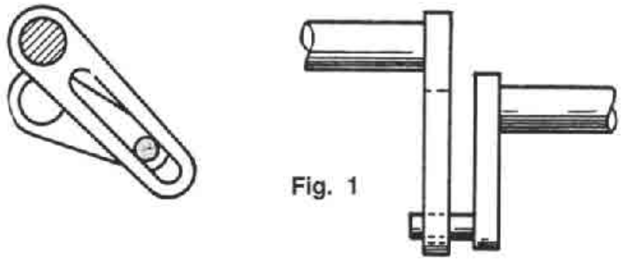


Fig. 1

Fig. 1 If constant velocity is not required, a pin and slot coupling can be used. Velocity transmission is irregular because the effective radius of operation is continually changing. The shafts must remain parallel unless a ball joint is placed between the slot and pin. Axial freedom is possible, but any change in the shaft offset will further affect the fluctuation of velocity transmission.

Fig. 2 This parallel-crank mechanism drives the overhead camshaft on engines. Each shaft has at lest two cranks connected by links. Each must have full symmetry for constant velocity action and to avoid dead points. By attaching ball joints at the ends of the links, displacement between the crank assembles is possible.

Fig. 3 This mechanism is kinematically equivalent to Fig. 2. It can be made by substituting two circular and contacting pins for each link. Each shaft has a disk carrying three or more projecting pins. The sum of the radii of the pins is equal to the eccentricity of offset of the shafts. The center lines between each pair of pins remain parallel as the coupling rotates. The pins need not have equal diameters. Transmission is at a constant velocity, and axial freedom is possible.

Fig. 4 This mechanism is similar to the mechanism shown in Fig. 3. However, holes replace one set of pins. The difference in radii is equal to the eccentricity or offset. Velocity transmission is constant; axial freedom is possible, but as in Fig. 3, the shaft axes must remain fixed. This type of mechanism can be installed in epicyclic reduction gear boxes.

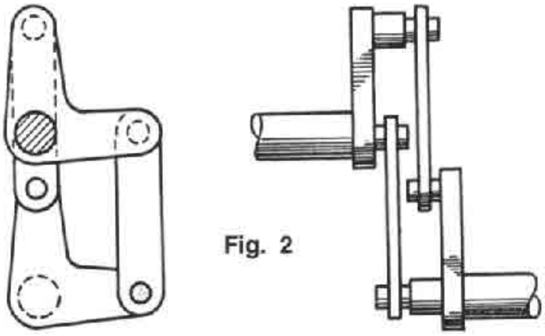


Fig. 2

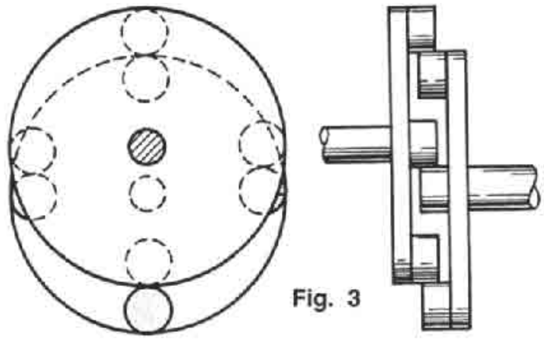


Fig. 3

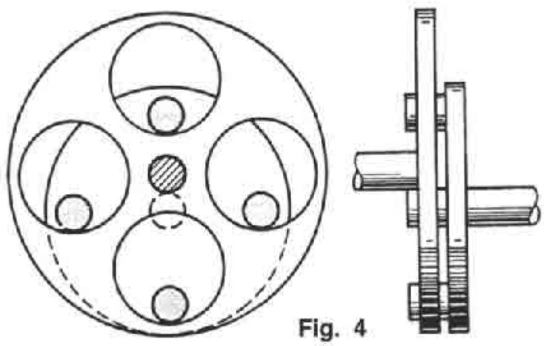


Fig. 4

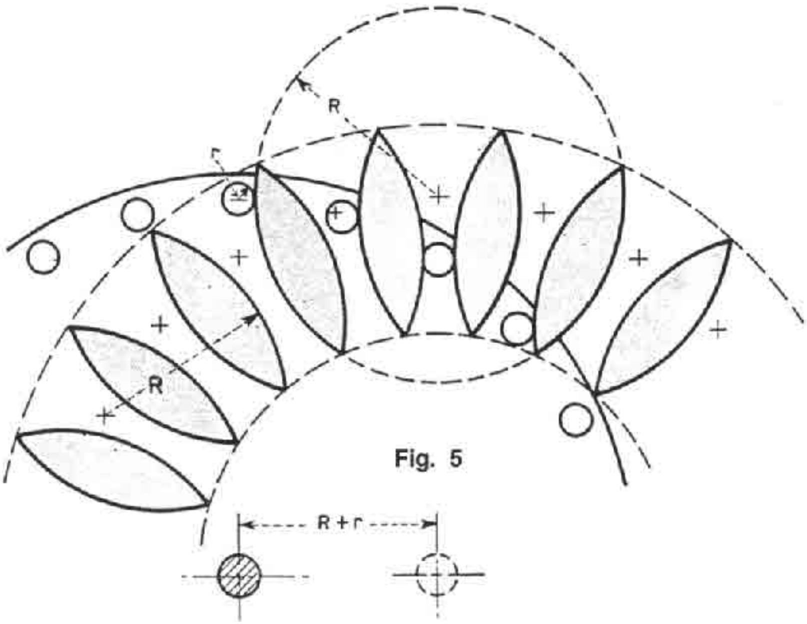


Fig. 5

Fig. 5 An unusual development in pin coupling is shown. A large number of pins engages the lenticular or shield-shaped sections formed from segments of theoretical large pins. The axes forming the lenticular sections are struck from the pith points of the coupling, and the distance $R + r$ is equal to the eccentricity between the shaft centers. Velocity transmission is constant; axial freedom is possible, but the shafts must remain parallel.