CHAPTER 10

TORQUE-LIMITING, TENSIONING, AND GOVERNING DEVICES

CALIPER BRAKES HELP MAINTAIN PROPER TENSION IN PRESS FEED

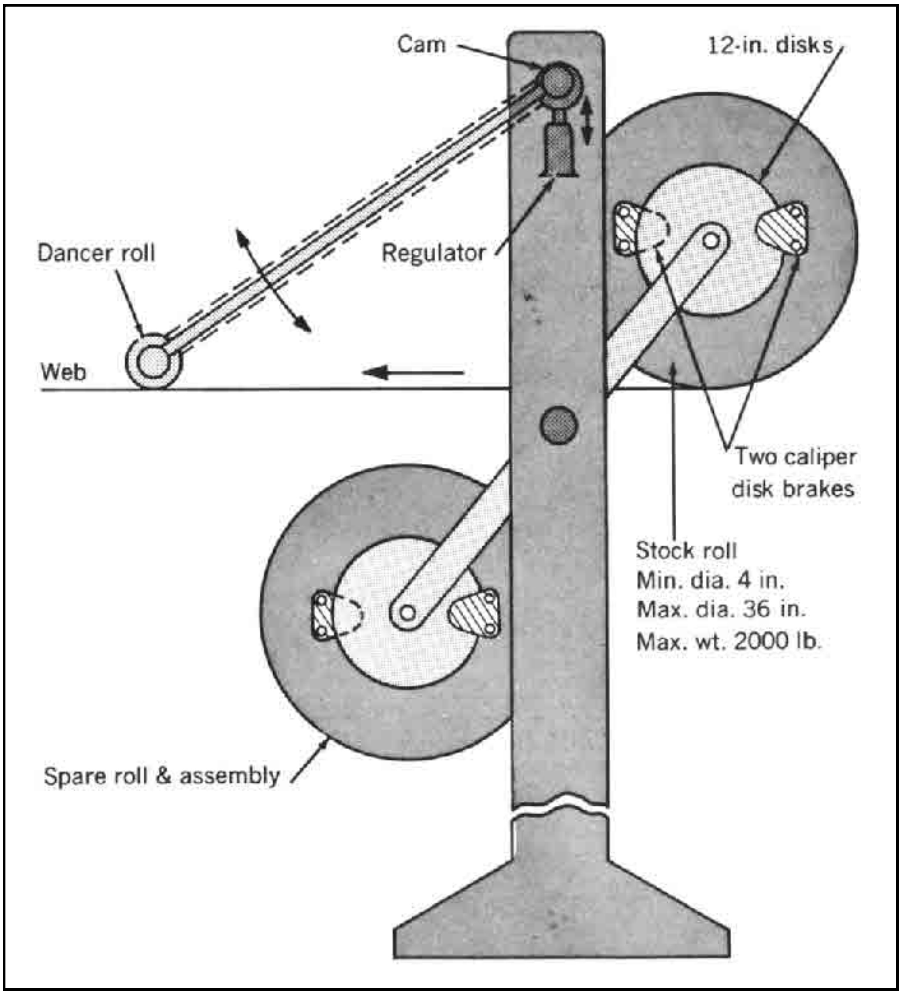
A simple cam-and-linkage arrangement (drawing) works in a team with two caliper disk brakes to provide automatic tension control for paper feeds on a web press.

In the feed system controlled tension must be maintained on the paper that's being drawn off at 1200 fpm from a roll up to 42 in. wide and 36 in. in diameter. Such rolls, when full, weigh 2000 lb. The press must also be able to make nearly instantaneous stops.

Friction-disk brakes are subject to lining wear, but they can make millions of stops before they need relining.

In the system, two pneumatic disk brakes made by Tol-O-Matic, Inc., Minneapolis, were mounted on each roll, gripping two separate 12-in. disks that provide maximum heat dissipation. To provide the desired constant-drag tension on the rolls, the brakes are always under air pressure. A dancer roll riding on the paper web can, however, override the brakes at any time. It operates a cam that adjusts a pressure regulator for controlling brake effort.

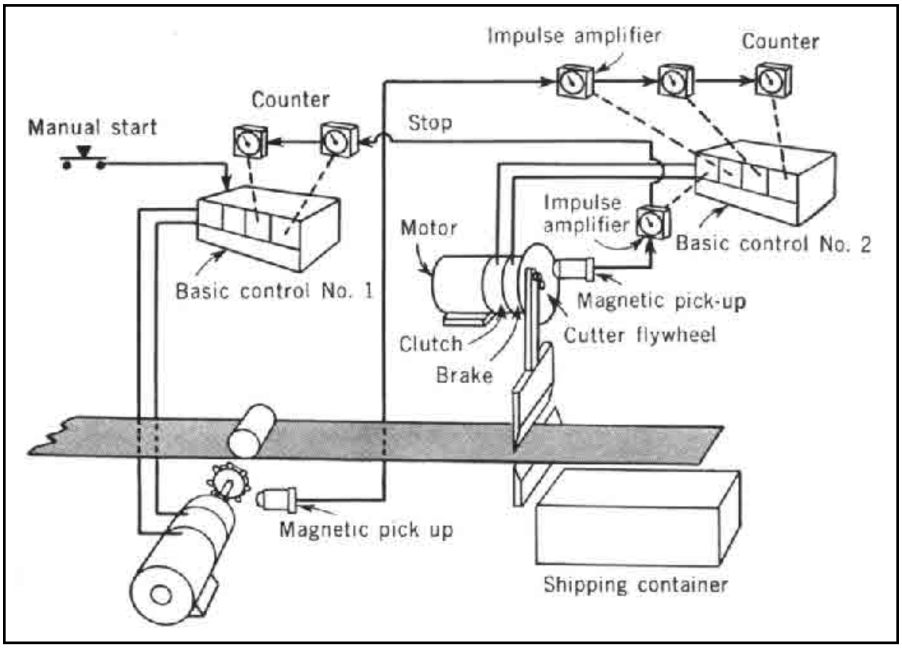
If the web should break or the paper run out on the roll, the dancer roll will allow maximum braking. The press can be stopped in less than one revolution.



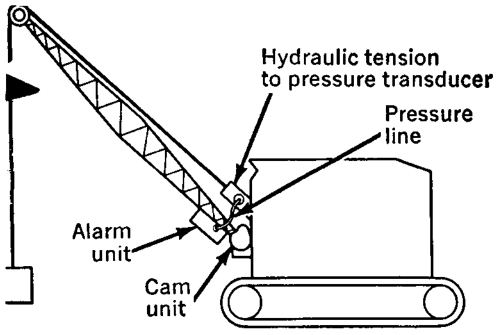
This linkage system works in combination with a regulator and caliper disk brakes to stop a press rapidly from a high speed, if the web should break.

SENSORS AID CLUTCH/ BRAKES

Two clutch/brake systems, teamed with magnetic pickup sensors, cut paper sheets into exact lengths. One magnetic pickup senses the teeth on a rotating sprocket. The resulting pulses, which are related to the paper length, are counted, and a cutter wheel is actuated by the second clutch/brake system. The flywheel on the second system enhances the cutting force.



This control system makes cutting sheets to desired lengths and counting how many cuts are made simpler.



WARNING DEVICE PREVENTS OVERLOADING OF BOOM

Cranes can now be protected against unsafe loading by a device whose movable electrical contacts are shifted by a combination of fluidic power and cam-and-gear arrangement (see drawing).

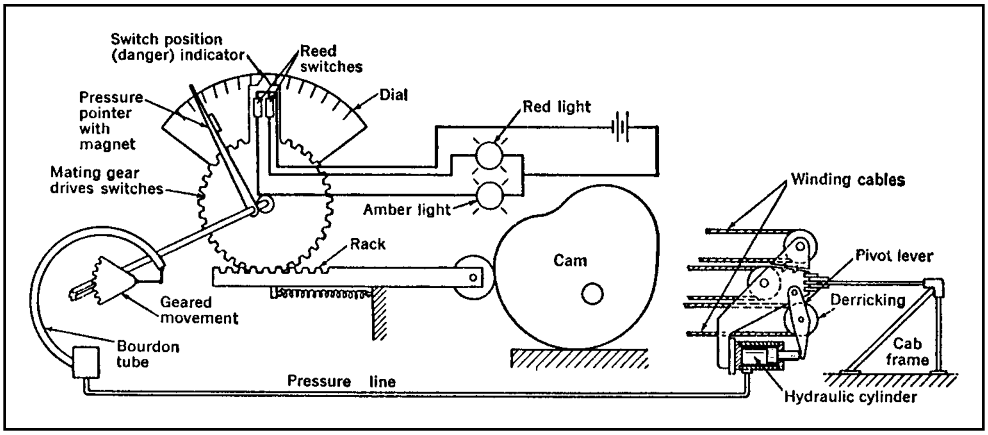
The device takes into consideration the two key factors in the safe loading of a crane boom: the boom angle (low angles create a greater overturning torque than high angles) and the compression load on the boom, which is greatest at high boom angles. Both factors are translated into inputs that are integrated to actuate the electrical warning system, which alerts the crane operator that a load is unsafe to lift.

How it works. In a prototype built for Thew-Lorain Inc. by US Gauge, Sellersville, Pennsylvania, a tension-to-pressure transducer (see drawing) senses the load on the cable and converts it into a hydraulic pressure that is proportional to the tension. This pressure is applied to a Bourdon-tube pressure gage with a rotating pointer that carries a small permanent magnet (see details in drawing). Two miniature magnetic reed switches are carried by another arm that moves on the same center as the pointer.

This arm is positioned by a gear and rack controlled by a cam, with a sinusoidal profile, that is attached to the cab. As the boom is raised or lowered, the cam shifts the position of the reed switches so they will come into close proximity with the magnet on the pointer and, sooner or later, make contact. The timing of this contact depends partly on the movement of the pointer that carries the magnet. On an independent path, the hydraulic pressure representing cable tension is shifting the pointer to the right or left on the dial.

When the magnet contacts the reed switches, the alarm circuit is closed, and it remains closed during a continuing pressure increase without retarding the movement of the point. In the unit built for Thew-Lorain, the switches were arranged in two stages: the first to trigger an amber warning light and second to light a red bulb and also sound an alarm bell.

Over-the-side or over-the-rear loading requires a different setting of the Bourdon pressure-gage unit than does over-the-front loading. A cam built into the cab pivot post actuated a selector switch.



A cam on the cab positions an arm with reed switches according to boom angle; the pressure pointer reacts to cable tension.

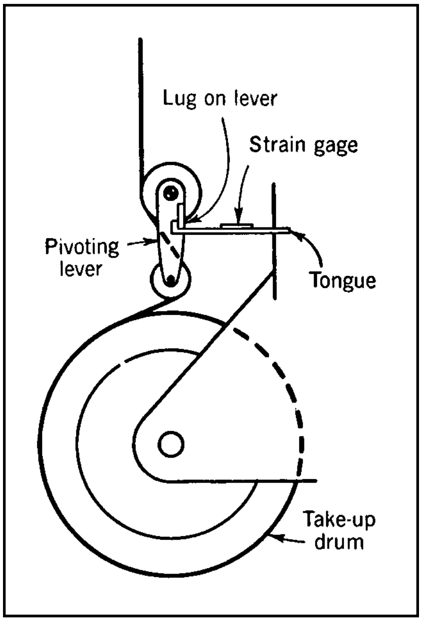
CONSTANT WATCH ON CABLE TENSION

A simple lever system solved the problem of how to keep track of varying tension loads on a cable as it is wound on its drum.

Thomas Grubbs of NASA's Manned Spacecraft Center in Houston devised the system, built around two pulleys mounted on a pivoted lever. The cable is passed between the pulleys (drawing) so an increase in cable tension causes the lever to pivot. This, in turn, pulls linearly on a flat metal tongue to which a strain gage has been cemented. Load on the lower pulley is proportional to tension on the cable. The stretching of the strain gage changes and electrical current that gives a continuous, direct reading of the cable tension.

The two pulleys on the pivoting lever are free to translate on the axes of rotation to allow proper positioning of the cable as it traverses the take-up drum.

A third pulley might be added to the two-pulley assembly to give some degree of adjustment to strain-gage sensitivity. Located in the plane of the other two pulleys, it would be positioned to reduce the strain on the tongue (for heavy loads) or increase the strain (for light loads).



A load on the lower pulley varies with tension on the cable, and the pivoting of the lever gives a direct reading with a strain gage.

TORQUE-LIMITERS PROTECT LIGHT-DUTY DRIVES

Light-duty drives break down when they are overloaded. These eight devices disconnect them from dangerous torque surges.

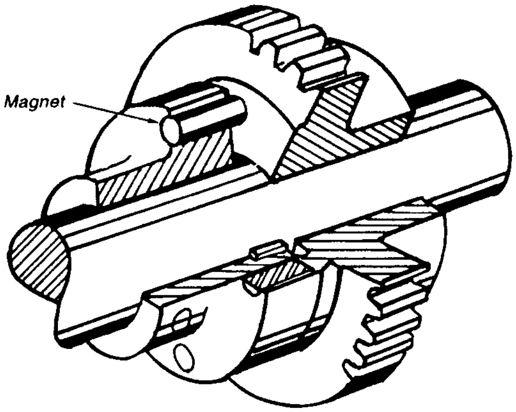


Fig. 1 Permanent magnets transmit torque in accordance with their numbers and size around the circumference of the clutch plate. Control of the drive in place is limited to removing magnets to reduce the drive's torque capacity.

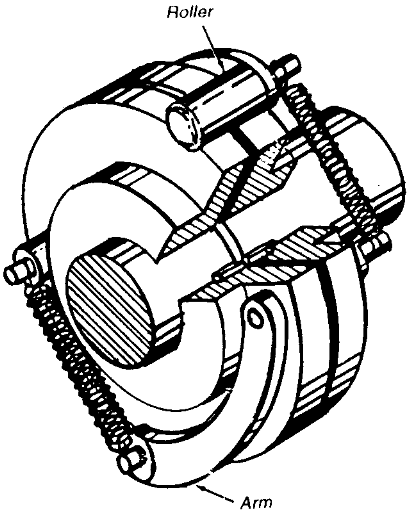


Fig. 2 Arms hold rollers in the slots that are cut across the disks mounted on the ends of butting shafts. Springs keep the roller in the slots, but excessive torque forces them out.

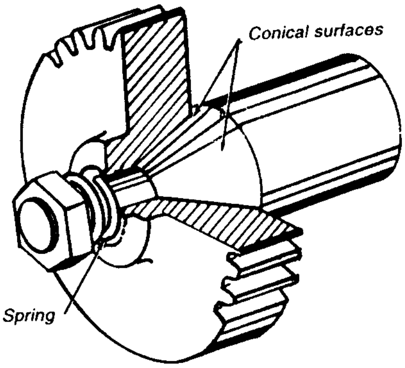


Fig. 3 A cone clutch is formed by mating a taper on the shaft to a beveled central hole in the gear. Increasing compression on the spring by tightening the nut increases the drive's torque capacity.

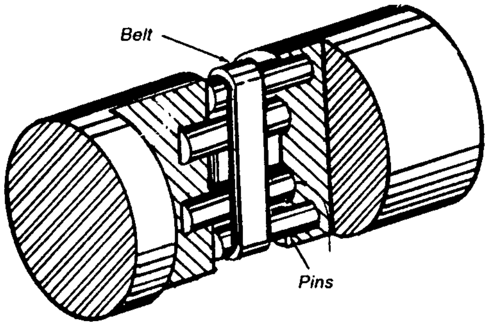


Fig. 4 A flexible belt wrapped around four pins transmits only the lightest loads. The outer pins are smaller than the inner pins to ensure contact.

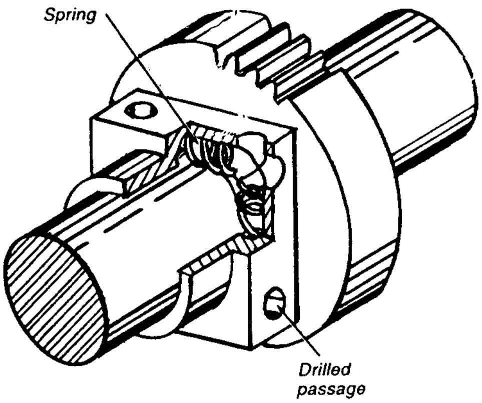


Fig. 5 Springs inside the block grip the shaft because they are distorted when the gear is mounted to the box on the shaft.

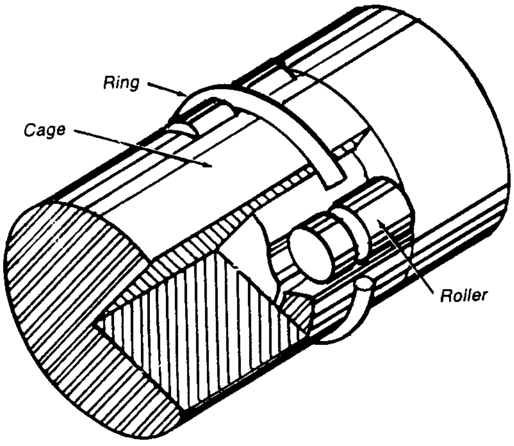


Fig. 6 The ring resists the natural tendency of the rollers to jump out of the grooves in the reduced end of one shaft. The slotted end of the hollow shaft acts as a cage.

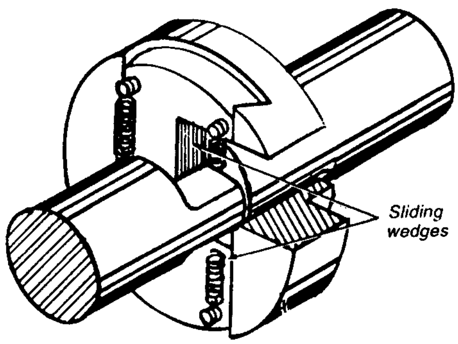


Fig. 7 Sliding wedges clamp down on the flattened end of the shaft. They spread apart when torque becomes excessive. The strength of the springs in tension that hold the wedges together sets the torque limit.

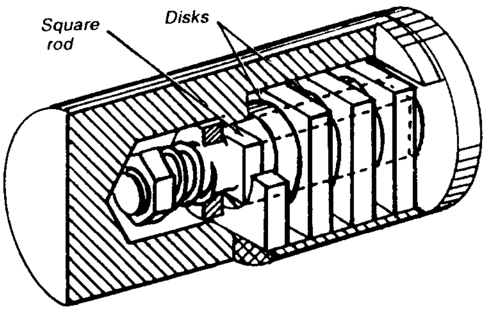


Fig. 8 Friction disks are compressed by an adjustable spring. Square disks lock into the square hole in the left shaft, and round disks lock onto the square rod on the right shaft.

LIMITERS PREVENT OVERLOADING

These 13 "safety valves" give way if machinery jams, thus preventing serious damage.

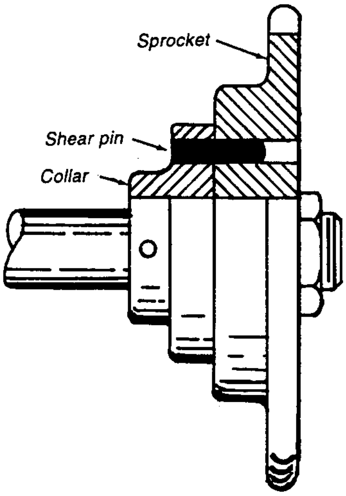


Fig. 1 A shear pin is a simple and reliable torque limiter. However, after an overload, removing the sheared pin stubs and replacing them with a new pin can be time consuming. Be sure that spare shear pins are available in a convenient location.

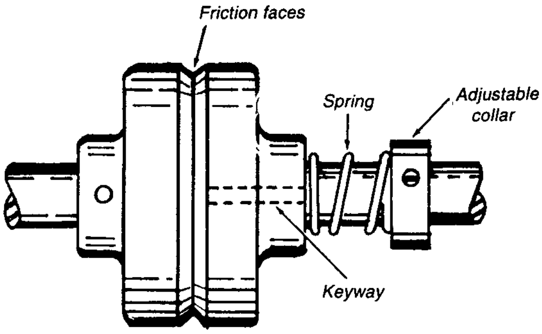


Fig. 2 Friction clutch torque limiter. Adjustable spring tension holds the two friction surfaces together to set the overload limit. As soon as an overload is removed, the clutch reengages. A drawback to this design is that a slipping clutch can destroy itself if it goes undetected.

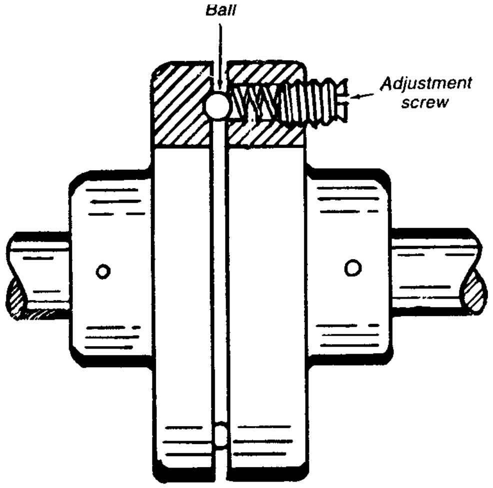


Fig. 3 Mechanical keys. A spring holds a ball in a dimple in the opposite face of this torque limiter until an overload forces it out. Once a slip begins, clutch face wear can be rapid. Thus, this limiter is not recommended for machines where overload is common.

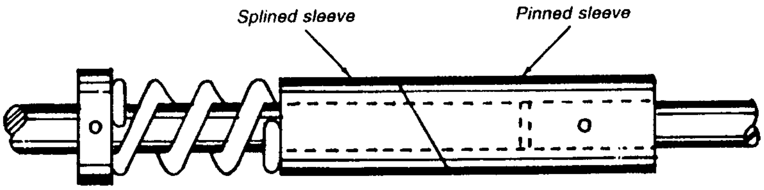


Fig. 4 A cylinder cut at an angle forms a torque limiter. A spring clamps the opposing-angled cylinder faces together, and they separate from angular alignment under overload conditions. The spring tension sets the load limit.

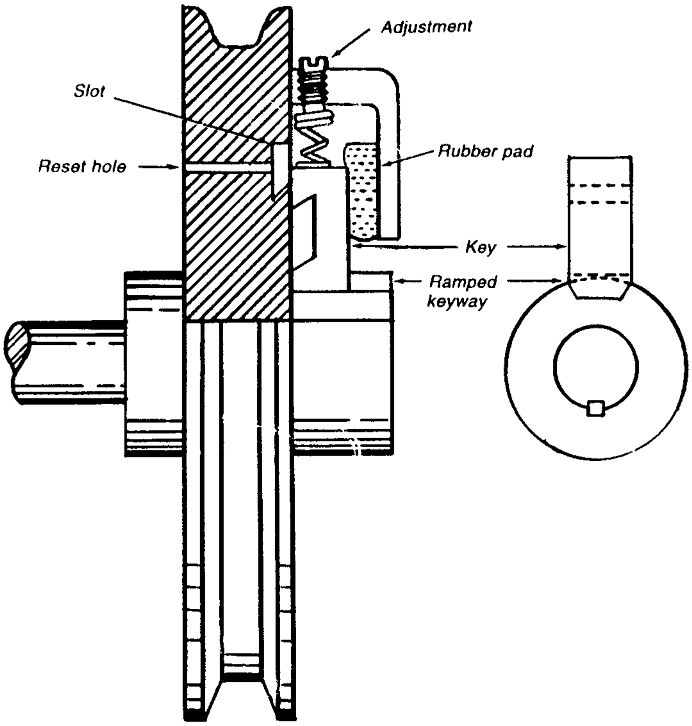


Fig. 5 A retracting key limits the torque in this clutch. The ramped sides of the keyway force the key outward against an adjustable spring. As the key moves outward, a rubber pad or another spring forces the key into a slot in the sheave. This holds the key out of engagement and prevents wear. To reset the mechanism, the key is pushed out of the slot with a tool in the reset hole of the sheave.

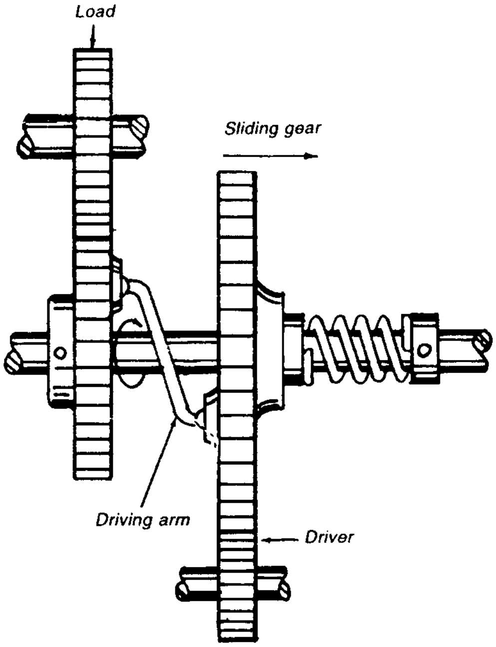


Fig. 6 Disengaging gears. The axial forces of a spring and driving arm are in balance in this torque limiter. An overload condition overcomes the force of the spring to slide the gears out of engagement. After the overload condition is removed, the gears must be held apart to prevent them from being stripped. With the driver off, the gears can safely be reset.

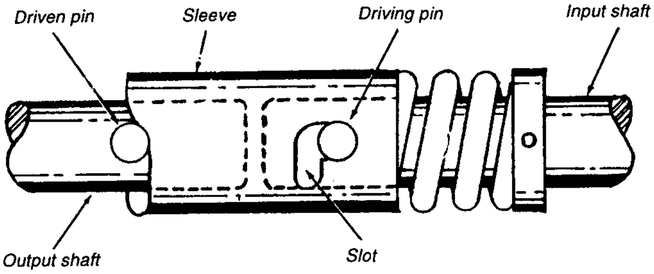


Fig. 7 A cammed sleeve connects the input and output shafts of this torque limiter. A driven pin pushes the sleeve to the right against the spring. When an overload occurs, the driving pin drops into the slot to keep the shaft disengaged. The limiter is reset by turning the output shaft backwards.

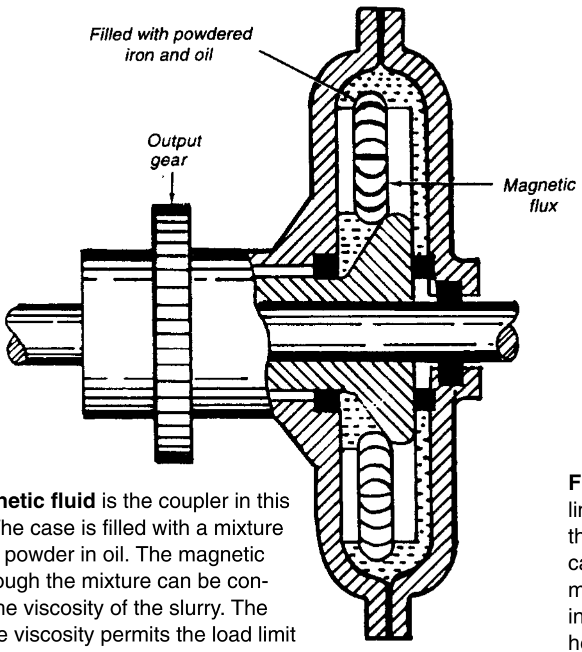


Fig. 8 A magnetic fluid is the coupler in this torque limiter. The case is filled with a mixture of iron or nickel powder in oil. The magnetic flux passed through the mixture can be controlled to vary the viscosity of the slurry. The ability to change viscosity permits the load limit to be varied over a wide range. Slip rings carry electric current to the vanes to create the magnetic field.

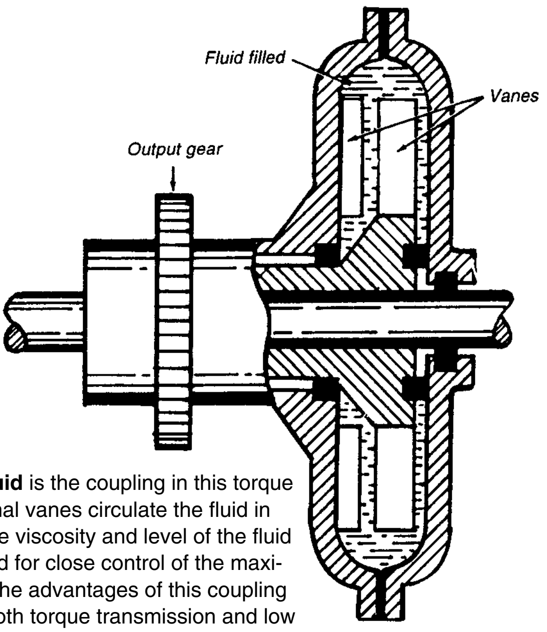


Fig. 9 A fluid is the coupling in this torque limiter. Internal vanes circulate the fluid in the case. The viscosity and level of the fluid can be varied for close control of the maximum load. The advantages of this coupling include smooth torque transmission and low heat rise during slip.

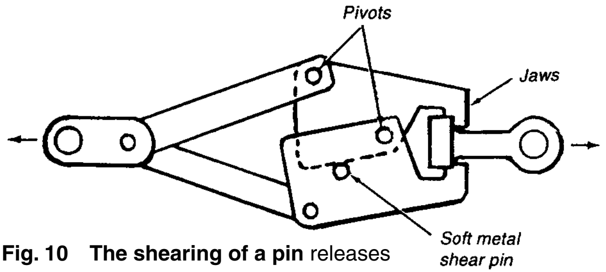


Fig. 10 The shearing of a pin releases tension in this coupling. A toggle-operated blade shears a soft pin so that the jaws open and release an excessive load. In an alternative design, a spring that keeps the jaws from spreading replaces the shear pin.

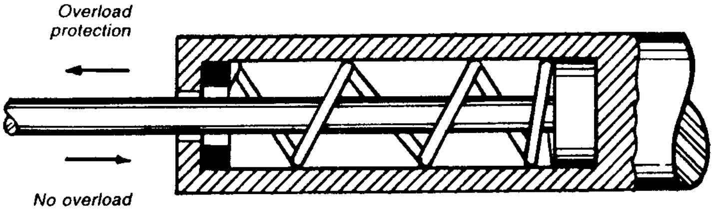


Fig. 11 A spring plunger provides reciprocating motion in this coupling. Overload can occur only when the rod is moving to the left. The spring is compressed under an overload condition.

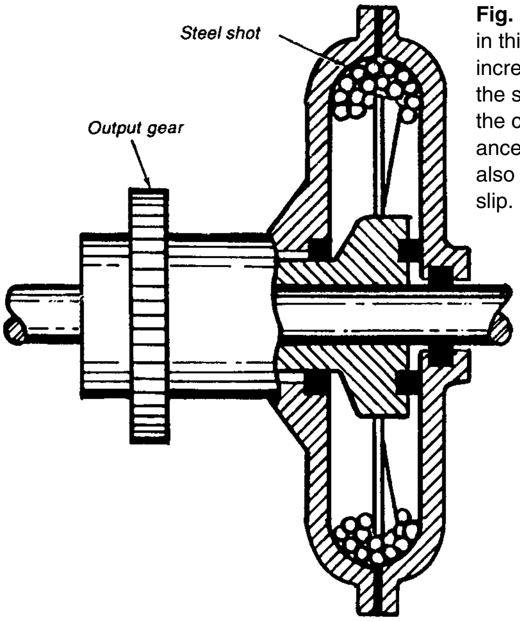


Fig. 12 Steel shot transmits more torque in this coupling as input shaft speed is increased. Centrifugal force compresses the steel shot against the outer surfaces of the case, increasing the coupling's resistance to slip. The addition of more steel shot also increases the coupling's resistance to slip.

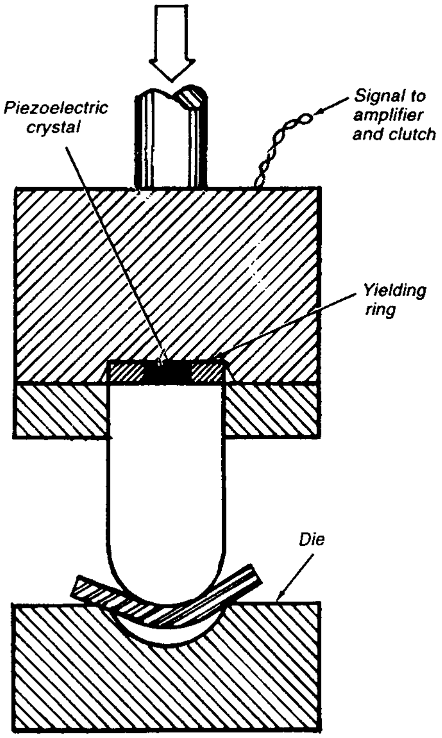


Fig. 13 A piezoelectric crystal produces an electric signal that varies with pressure in this metal-forming press. When the amplified output of the piezoelectric crystal reaches a present value corresponding to the pressure limit, the electric clutch disengages. A yielding ring controls the compression of the piezoelectric crystal.

SEVEN WAYS TO LIMIT SHAFT ROTATION

Traveling nuts, clutch plates, gear fingers, and pinned members form the basis of these ingenious mechanisms.

Mechanical stops are often required in automatic machinery and servomechanisms to limit shaft rotation to a given number of turns. Protection must be provided against excessive forces caused by abrupt stops and large torque requirements when machine rotation is reversed after being stopped.

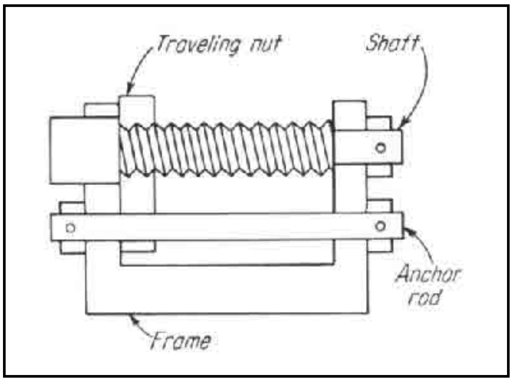


Fig. 1 **A traveling nut** moves along the threaded shaft until the frame prevents further rotation. This is a simple device, but the traveling nut can jam so tightly that a large torque is required to move the shaft from its stopped position. This fault is overcome at the expense of increased device length by providing a stop pin in the traveling nut.

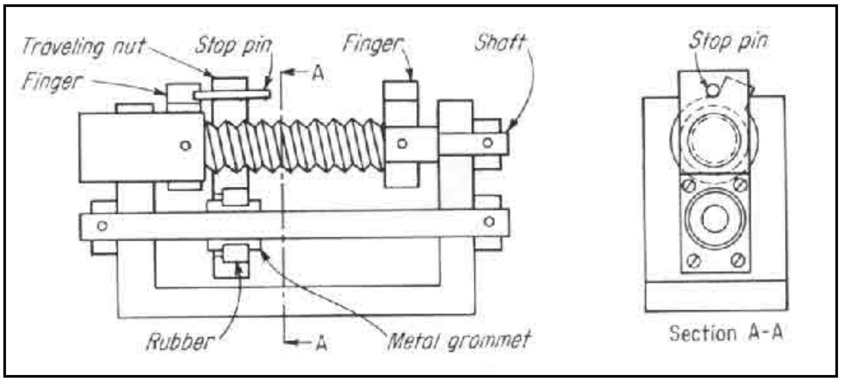
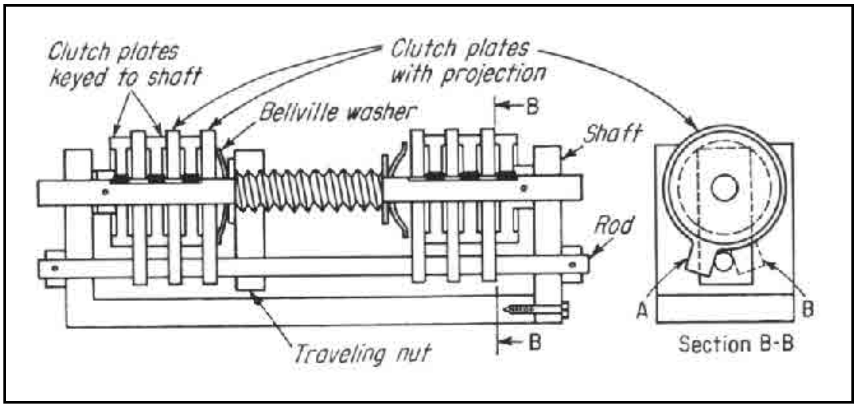


Fig. 2 The engagement between the pin and the rotating finger must be shorter than the thread pitch so the pin can clear the finger on the first reverse-turn. The rubber ring and grommet lessen the impact and provide a sliding surface. The grommet can be oil-impregnated metal.

Fig. 3 **Clutch plates** tighten and stop their rotation as the rotating shaft moves the nut against the washer. When rotation is reversed, the clutch plates can turn with the shaft from A to B. During this movement, comparatively low torque is required to free the nut from the clutch plates. Thereafter, subsequent movement is free of clutch friction until the action is repeated at the other end of the shaft. The device is recommended for large torques because the clutch plates absorb energy well.



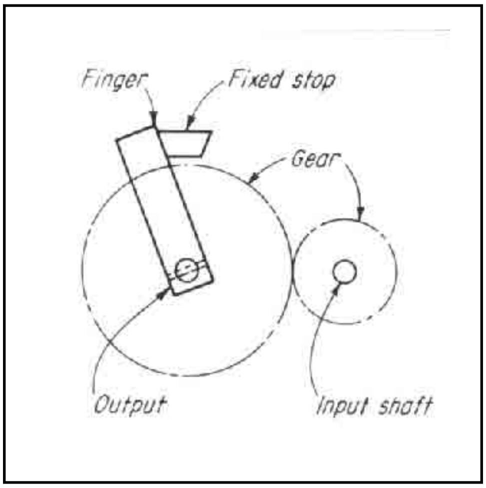


Fig. 4 A shaft finger on the output shaft hits the resilient stop after making less than one revolution. The force on the stop depends upon the gear ratio. The device is, therefore, limited to low ratios and few turns, unless a worm-gear setup is used.

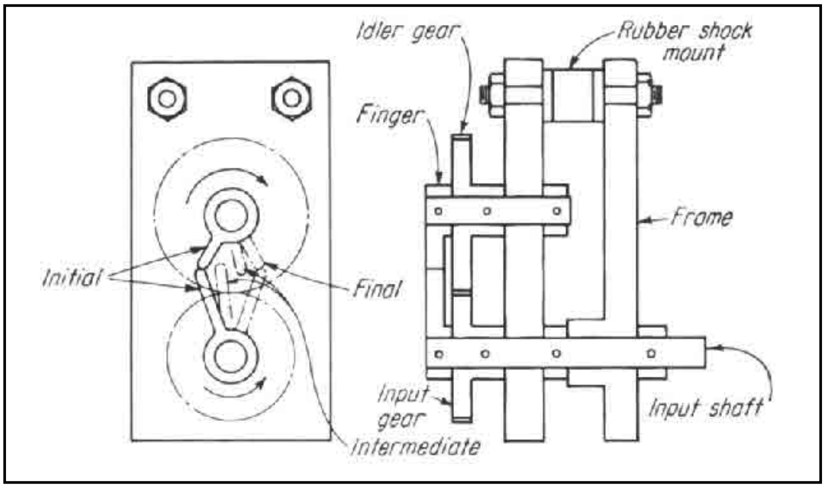


Fig. 5 Two fingers butt together at the initial and final positions to prevent rotation beyond these limits. A rubber shock-mount absorbs the impact load. A gear ratio of almost 1:1 ensures that the fingers will be out-of-phase with one another until they meet on the final turn. Example: Gears with 30 to 32 teeth limit shaft rotation to 25 turns. Space is saved here, but these gears are expensive.

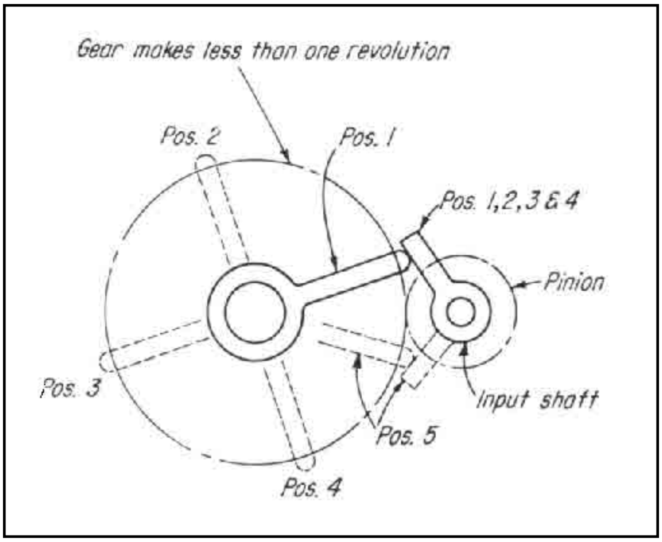


Fig. 6 A large gear ratio limits the idler gear to less than one turn. Stop fingers can be added to the existing gears in a train, making this design the simplest of all. The input gear, however, is limited to maximum of about five turns.

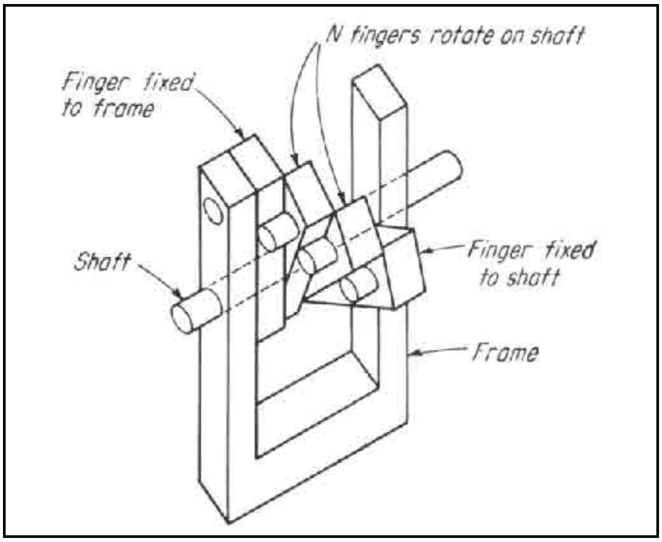
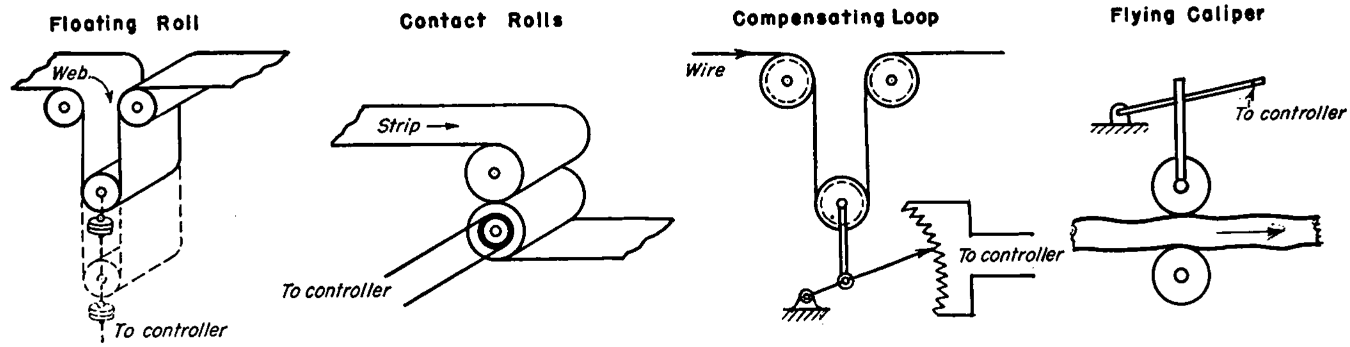


Fig. 7 Pinned fingers limit shaft turns to approximately N + 1 revolutions in either direction. Resilient pin-bushings would help reduce the impact force.

MECHANICAL SYSTEMS FOR CONTROLLING TENSION AND SPEED

The key to the successful operation of any continuous-processing system that is linked together by the material being processed is positive speed synchronization of the individual driving mechanisms. Typical examples of such a system are steel mill strip lines, textile processing equipment, paper machines, rubber and plastic processers, and printing presses. In each of these examples, the material will become wrinkled, marred, stretched or otherwise damaged if precise control is not maintained.

FIG. 1—PRIMARY INDICATORS



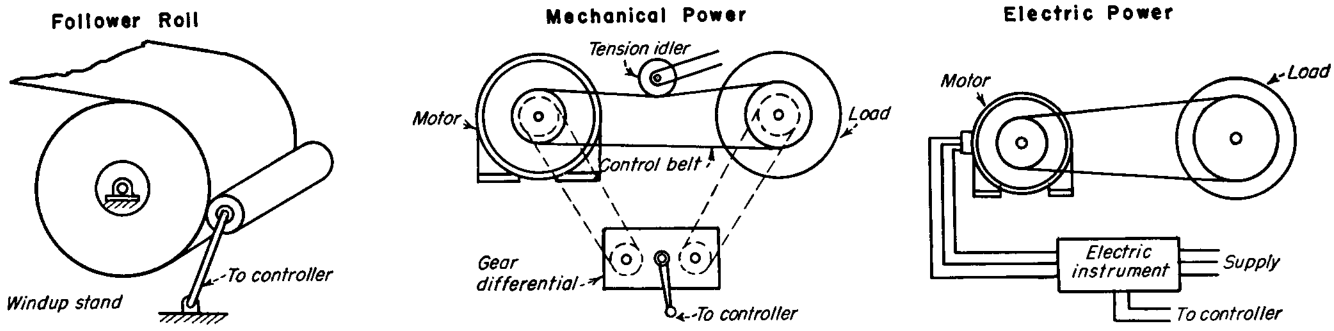
Use: constant tension winding; registry control; section synchronizing.

Use: velocity control; cutter feed control.

Use: line-tension control; section synchronizing.

Use: thickness control; diameter control.

FIG. 2—SECONDARY INDICATORS

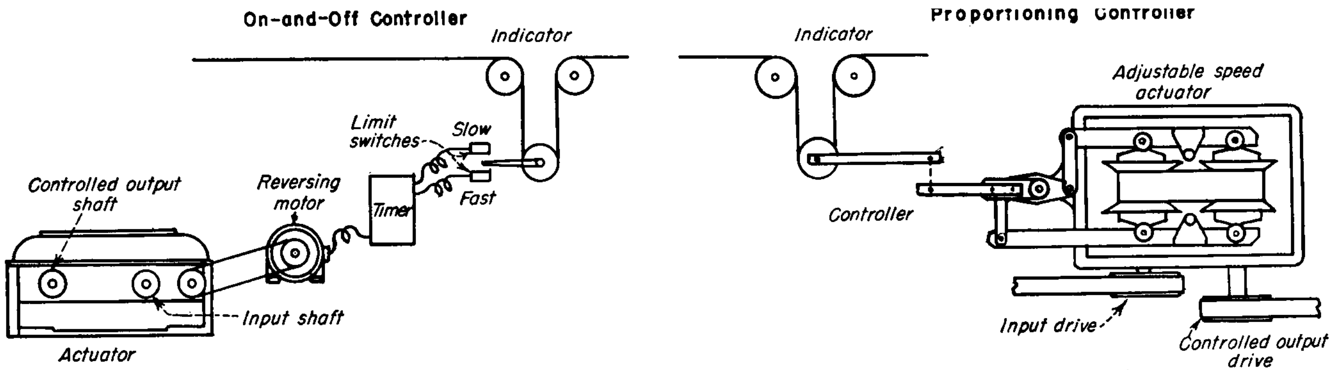


Use: winder control; periphery cutting:

Use: winding tension; feeder control.

Use: winding tension; feeder control.

FIG. 3—CONTROLLERS AND ACTUATORS



Use: delivery rate; storage capacity; rough section synchronizing.

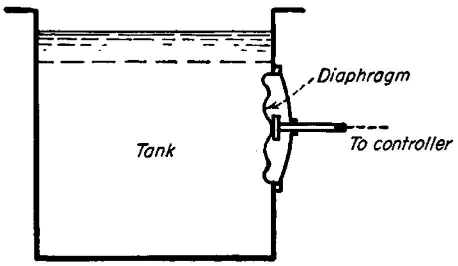
Use: winding controls; section synchronizing; cutter feeds.

The automatic control for such a system contains three basic elements: The *signal device* or *indicator*, which senses the error to be corrected; the *controller*, which interprets the indicator signal and amplifies it, if necessary, to initiate control action; and the *transmission*, which operates from the controller to change the speed of the driving mechanism to correct the error.

Signal indicators for continuous systems are divided in two general classifications: *Primary indicators* that measure the change in speed or tension of the material by direct contact with the material; and *secondary indicators* that measure a change in the material from some reaction in the system that is proportional to the change.

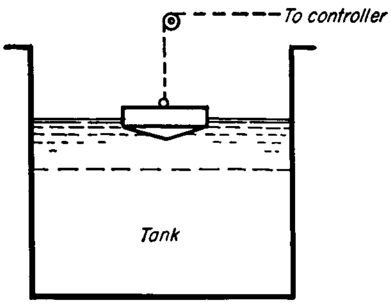
The primary type is inherently more accurate because of its direct contact with the material. These indicators take the form of contact rolls, floating or compensating rolls, resistance bridges and flying calipers, as illustrated in Fig. 1. In each case, any change in the tension, velocity, or pressure of the material is indicated directly and immediately by a displacement or change in position of the indicator element. The primary indicator, therefore, shows deviation from an established norm, regardless of the factors that have caused the change.

Fluid Pressure



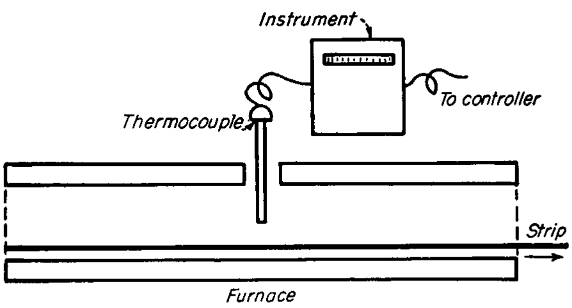
Use: fluid level control; constant pressure control; filtering rate control.

Liquid Level



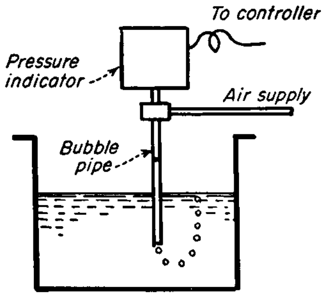
Use: Pumping rate control; system pressure control.

Temperature



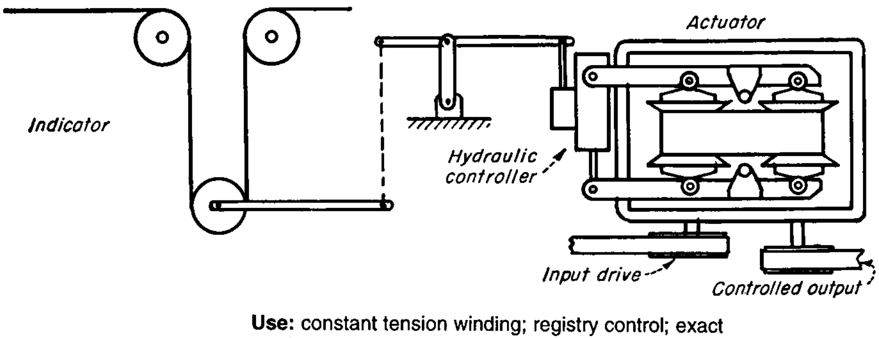
Use: annealing; drum dryers; kilns.

Pressure



Use: fluid density; feeding rate; flow rate.

Proportioning-Throttling Controller



Use: constant tension winding; registry control; exact section synchronizing.

Secondary indicators, shown in Fig. 2, are used in systems where the material cannot be in direct contact with the indicator or when the space limitations of a particular application make their use undesirable. This type of indicator introduces a basic inaccuracy into the control system which is the result of measuring an error in the material from a reaction that is not exactly proportional to the error. The control follows the summation of the errors in the material and the indicator itself.

The controlling devices, which are operated by the indicators, determine the degree of speed change required to correct the error, the rate at which the correction must be made, and the stopping point of the control action after the error has been corrected. The manner in which the corrective action of the controller is stopped determines both the accuracy of the control system and the kind of control equipment required.

Three general types of control action are illustrated in Fig. 3. Their selection for any individual application is based on the degree of control action required, the amount of power available for initiating the control, that is, the torque amplification required, and the space limitations of the equipment.

The on-and-off control with timing action is the simplest of the three types. It functions in this way: when the indicator is displaced, the timer contact energizes the control in the proper direction for correcting the error. The control action continues until the timer stops the action. After a short interval, the timer again energizes the control system and, if the error still exists, control action is continued in the same direction. Thus, the control process is a step-by-step response to make the correction and to stop the operation of the controller.

The proportioning controller corrects an error in the system, as shown by the indicator, by continuously adjusting the actuator to a speed that is in exact proportion to the displacement of the indicator. The diagram in Fig. 3 shows the proportioning controller in its simplest form as a direct link connection between the indicator and the actuating drive. However, the force amplification between the indicator and the drive is rel-

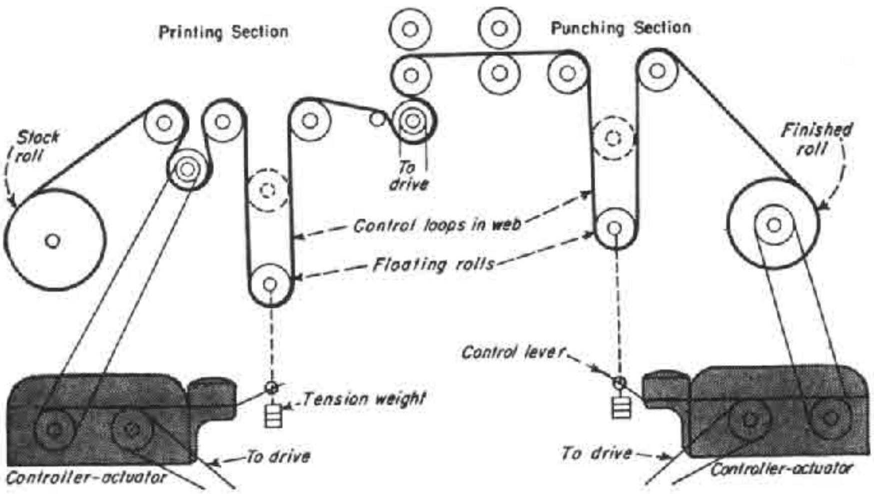


Fig 4 Floating rolls are direct indicators of speed and tension in the paper web. Controller-actuators adjust feed and windup rolls to maintain registry during printing.

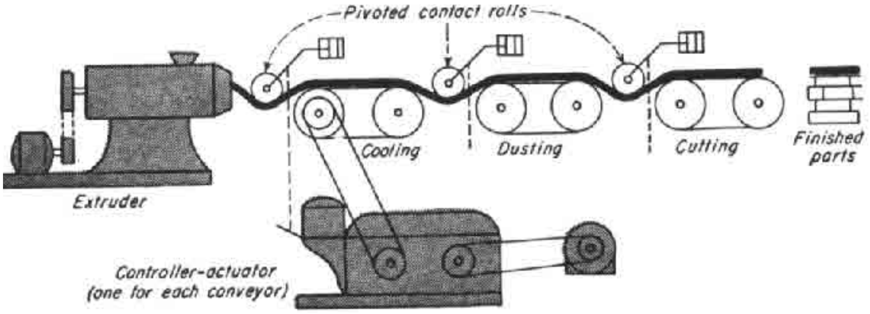


Fig. 5 Dimension control of extruded materials calls for primary indicators like the contact rolls shown. Their movements actuate conveyor control mechanisms.

atively low; thus it limits this controller to applications where the indicator has sufficient operating force to adjust the speed of the variable-speed transmission directly.

The most accurate controller is the proportioning type with throttling action. Here, operation is in response to the rate or error indication. This controller, as shown in Fig. 3, is connected to a throttling valve, which operates a hydraulic servomechanism for adjusting the variable-speed transmission.

The throttling action of the valve provides a slow control action for small error correction or for continuous correction at a slow rate. For following large error, as shown by the indicator, the valve opens to the full position and makes the correction as rapidly as the variable-speed transmission will allow.

Many continuous processing systems can be automatically controlled with a packaged unit consisting of a simple, mechanical, variable-speed transmission and an accurate hydraulic controller.

This controller-transmission package can change the speed relationship at the driving points in the continuous system from any indicator that signals for correction by a displacement. It has anti-hunting characteristics because of the throttling action on the control valve, and is self-neutralizing because the control valve is part of the transmission adjustment system.

The rotary printing press is an example of a continuous processing system that requires automatic control. When making billing forms on a press, the printing plates are rubber, and the forms are printed on a continuous web or paper. The paper varies in texture, moisture content, flatness, elasticity, and finish. In addition, the length of the paper changes as the ink is applied.

In a typical application of this kind, the accuracy required for proper registry of the printing and hole punching must be held to a differential of $1/32$ in. in 15 ft of web. For this degree of accuracy, a floating or compensating roll, as shown in Fig. 4, serves as the indicator because it is the most accurate way to indicate changes in the length of the web by displacement. In this case, two floating rolls are combined with two separate controllers and actuators. The first controls the in-feed speed and tension of the paper stock, and the second controls the wind-up.

The in-feed is controlled by maintaining the turning speed of a set of feeding rolls that pull the paper off the stock roll. The second floating roll controls the speed of the wind-up mandrel. The web of paper is held to an exact value of tension between the feed rolls and the punching cylinder of the press by the in-feed control. It is also held between the punching cylinder and the wind-up roll. Hence, it is possible to control the tension in the web of different grades of paper and also adjust the relative length at these two points, thereby maintaining proper registry.

The secondary function of maintaining exact control of the tension in the paper as it is rewound after printing is to condition the paper and obtain a uniformly wound roll. This makes the web ready for subsequent operations.

The control of dimension or weight by tension and velocity regulation can be illustrated by applying the same general type of controller actuator to the take-odd conveyors in a extruder line such as those used in rubber and plastics processing. Two problems must be solved: First, to set the speed of the take-away conveyor at the extruder to match the variation in extrusion rate; and, second, to set the speeds of the subsequent conveyor sections to match the movement of the stock as it cools and tends to change dimension.

One way to solve these problems is to use the pivoted idlers or contact rolls as indicators, as shown in Fig. 5. The rolls contact the extruded material between each of the conveyor sections and control the speed of the driving mechanism of the following section. The material forms a slight catenary between the stations, and the change in the catenary length indicates errors in driving speeds.

The plasticity of the material prevents the use of a complete control loop. Thus, the contract roll must operate with very little resistance or force through a small operating angle.

The difficulties in winding or coiling a strip of thin steel that has been plated or pre-coated for painting on a continuous basis is typical of processing systems whose primary indicators cannot be used. While it is important that no contact be made with the prepared surface of the steel, it also desirable to rewind the strip after preparation in a coil that is sound and slip-free. An automatic, constant-tension winding control and a secondary indicator initiate the control action.

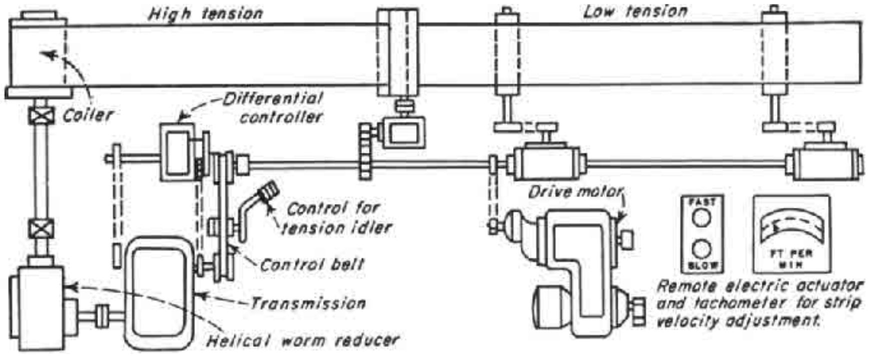


Fig. 6 The differential controller has a third shaft that signals the remote actuator when tension in sheet material changes. Coiler power is a secondary-control index.

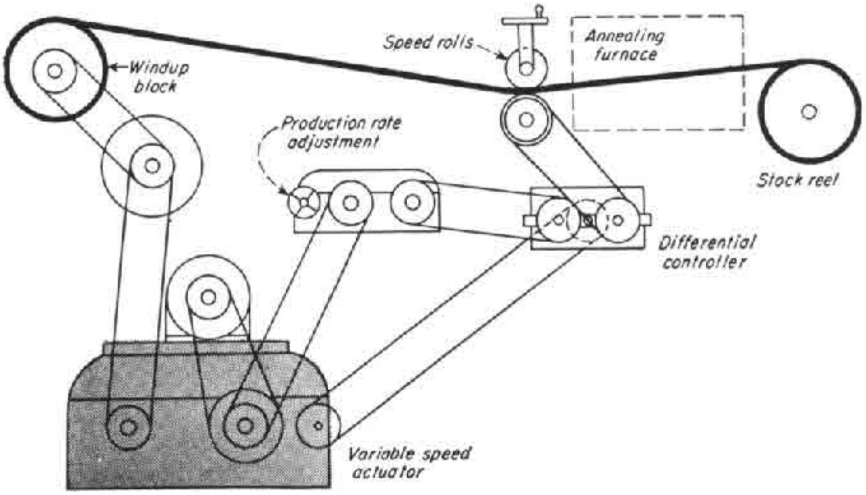


Fig. 7 The movement of wire through the annealing furnace is regulated at constant velocity by continuously retarding the speed of the windup reels to allow for wire build-up.

The control system shown in Fig. 6 is used in winding coils from 16 in. core diameter to 48 in. maximum diameter. The power to wind the coil is the controlling medium because, by maintaining constant winding power as the coil builds up, a constant value of strip tension can be held within the limits required. Actually, this method is so inaccurate that the losses in the driving equipment (which are a factor in the power being measured) are not constant; hence the strip tension changes slightly. This same factor enters into any control system that uses winding power as an index of control.

A torque-measuring belt that operates a differential controller measures the power of the winder. Then, in turn, the controller adjusts the variable-speed transmission. The change in speed between the source of power and the transmission is measured by the three-shaft gear differential, which is driven in tandem with the control belt. Any change in load across the control belt produces a change in speed between the driving and driven ends of the belt. The differential acts as the controller, because any change in speed between the two outside shafts of the differential results in a rotation or displacement of the center or control shaft. By connecting the control shaft of the differential directly to a screw-controlled variable-speed transmission, it is possible to adjust the transmission to correct any change in speed and power delivered by the belt.

This system is made completely automatic by establishing a neutralizing speed between the two input shaft of the differential (within the creep value of the belt). When there is no tension in the strip (e.g., when it is cut), the input speed to the actuator side of the differential is higher on the driven side than it is on the driving side of the differential. This unbalance reverses the rotation of the control shaft of the differential, which in turn resets the transmission to high speed required for starting the next coil on the rewinding mandrel.

In operation, any element in the system that tends to change strip tension causes a change in winding power. This change, in turn, is immediately compensated by the rotation (or tendency to rotate) of the controlling shaft in the differential. Hence, the winding mandrel speed is continuously and automatically corrected to maintain constant tension in the strip.

When the correct speed relationships are established in the controller, the system operates automatically for all conditions of operation. In addition, tension in the strip can be adjusted to any value by moving the tension idler on the control belt to increase or decrease the load capacity of the belt to match a desired strip tension.

There are many continuous processing systems that require constant velocity of the material during processing, yet do not require accurate control of the tension in the material. An example of this process is the annealing of wire that is pulled off stock reels through an annealing furnace and then rewound on a windup block.

The wire must be passed through the furnace at a constant rate so that the annealing time is maintained at a fixed value. Because the wire is pulled through the furnace by the wind-up blocks, shown in Fig. 7, its rate of movement through the furnace would increase as the wire builds up on the reels unless a control slows down the reels.

A constant-velocity control that makes use of the wire as a direct indicator measures the speed of the wire to initiate a control action for adjusting the speed of the wind-up reel. In this case, the wire can be contacted directly, and a primary indicator in the form of a contact roll can register any change in speed. The contact roll drives one input shaft of the differential controller. The second input shaft is connected to the driving shaft of the variable-speed transmission to provide a reference speed. The third, or control, shaft will then rotate when any difference in speed exists between the two input shafts. Thus, if the control shaft is connected to a screw-regulated actuator, an adjustment is obtained for slowing down the wind-up blocks as the coils build up and the wire progresses through the furnace at a constant speed.

DRIVES FOR CONTROLLING TENSION

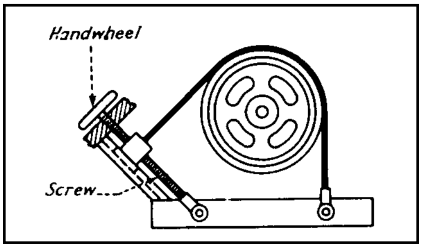
Mechanical, electrical, and hydraulic methods for obtaining controlled tension on winding reels and similar drives, or for driving independent parts of a machine in synchronism.

MECHANICAL DRIVES

A band brake is used on coil winders, insulation winders, and similar machines where maintaining the tension within close limits is not required.

It is simple and economical, but tension will vary considerably. Friction drag at start-up might be several times that which occurs during running because of the difference between the coefficient of friction at the start and the coefficient of sliding friction. Sliding friction will be affected by moisture, foreign matter, and wear of the surfaces.

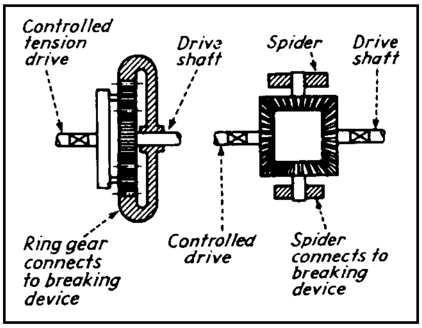
Capacity is limited by the heat radiating capacity of the brake at the maximum permissible running temperature.



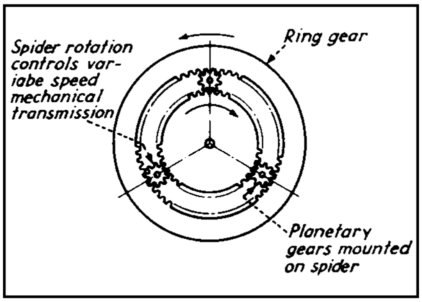
Differential drives can take many different forms, e.g., epicycle spur gears, bevel gear differentials, or worm gear differentials.

The braking device on the ring gear or spider could be a band brake, a fan, an impeller, an electric generator, or an electric drag element such as a copper disk rotating in a powerful magnetic field. A brake will give a drag or tension that is reasonably constant over a wide speed range. The other braking devices mentioned here will exert a torque that will vary widely with speed, but will be definite for any given speed of the ring gear or spider.

A definite advantage of any differential drive is that maximum driving torque can never exceed the torque developed by the braking device.



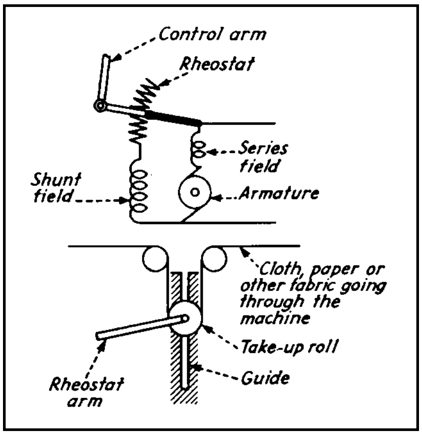
Differential gearing can be used to control a variable-speed transmission. If the ring gear and sun gear are to be driven in opposite directions from their respective shafts which are to be held in synchronism, the gear train can be designed so that the spider on which the planetary gears are mounted will not rotate when the shafts are running at the desired relative speeds. If one or the other of the shafts speeds ahead, the spider rotates correspondingly. The spider rotation changes the ratio of the variable-speed transmission unit.



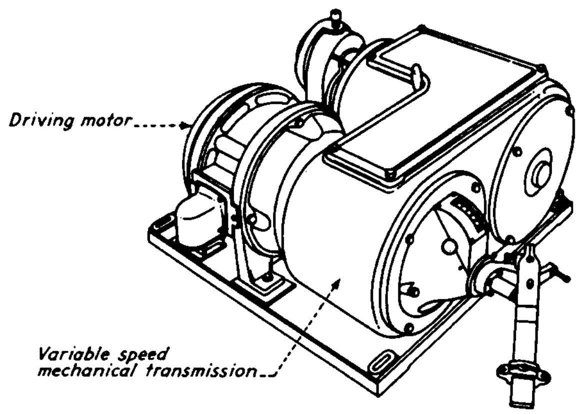
ELECTRICAL DRIVES

The shunt-field rheostat in a DC motor drive can be used to synchronize drives. When connected to a machine for processing paper, cloth, or other sheet material that is passing around a take-up roll, the movement of the take-up roll moves a control arm which is connected to the rheostat. This kind of drive is not suitable for wide changes of speed that exceed about a 2.5 to 1 ratio.

For wide ranges of speed, the rheostat is put in the shunt field of a DC generator that is driven by another motor. The voltage developed by the generator is controlled from zero to full voltage. The generator furnishes the current to the driving motor armature, and the fields of the driving motor are separately excited. Thus, the motor speed is controlled from zero to maximum.

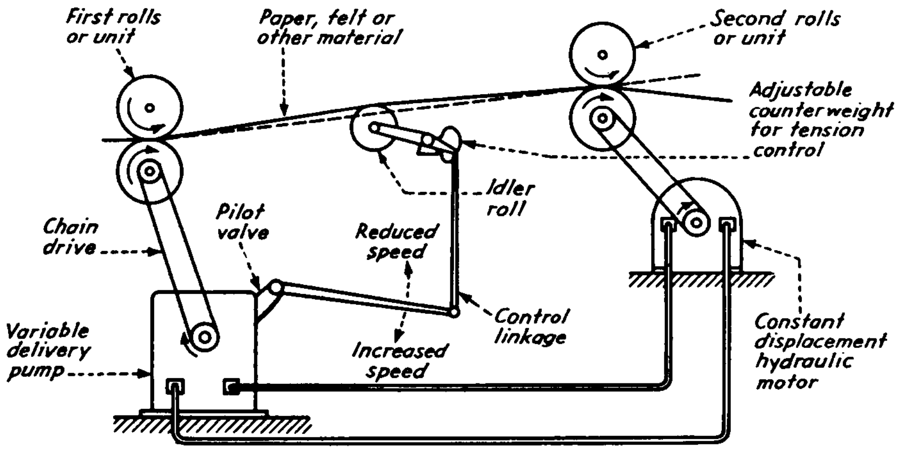


Selsyn motors can directly drive independent units in exact synchronism, provided their inertias are not too great. Regardless of loads and speeds, selsyn motors can be the controlling units. As an example, variable-speed mechanical transmission units with built-in selsyn motors are available for powering constant-tension drives or the synchronous driving of independent units.



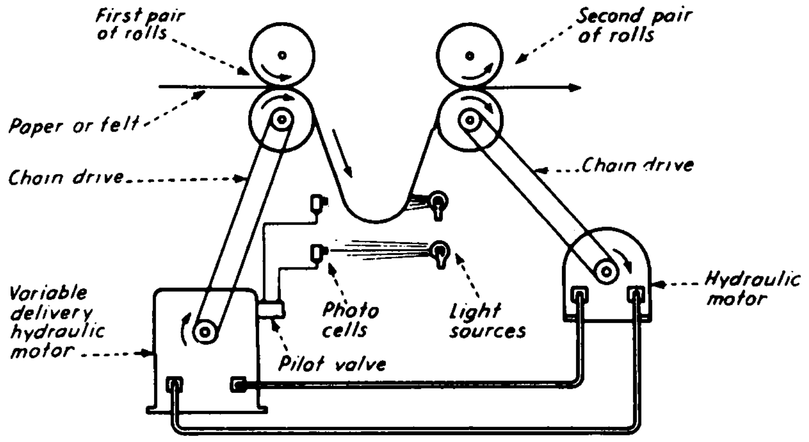
HYDRAULIC DRIVES

Hydraulic Control—Tension between successive pairs of rolls, or synchronism between successive units of a machine can be controlled automatically by hydraulic drives. Driving the variable delivery pump from one of the pairs of rolls automatically maintains an approximately constant relative speed between the two units, at all speeds and loads. The variations caused by oil leakage and similar factors are compensated automatically by the idler roll and linkage. They adjust the pilot valve that controls the displacement of the variable delivery pump.



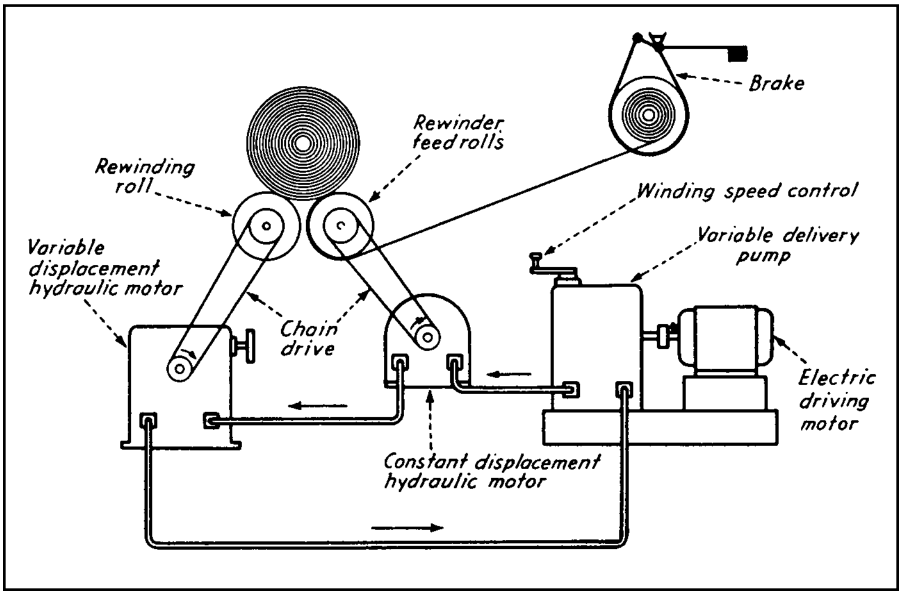
The counterweight on the idler roll is set for the desired tension in the felt, paper, or other material. Increased tension as a result of the high speed of the second pair of rolls depresses the idler roll. The control linkage then moves the pilot valve to decrease pump delivery, which slows the speed of the second pair of rolls. The reverse operations occur when the tension in the paper decreases, allowing the idler roll to move upwards.

If the material passing through the machine is too weak to operate a mechanical linkage, the desired control can be obtained by photoelectric devices. The hydraulic operation is exactly the same as that described for the hydraulic drives.



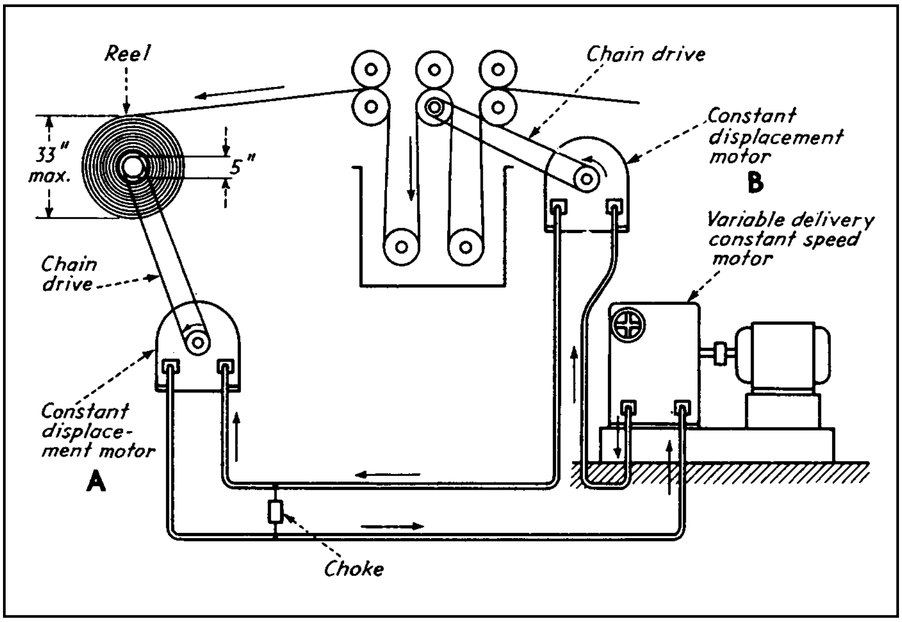
A band brake intended to obtain a friction drag will give variable tension. In this hydraulic drive, the winding tension is determined by the difference in torque exerted on the rewinder feed roll and the winding roll. The brake plays no part in establishing the tension.

The constant displacement hydraulic motor and the variable displacement hydraulic motor are connected in series with the variable delivery pump. Thus, the relative speeds of the two hydraulic motors will always remain substantially the same. The displacement of the variable speed motor is then adjusted to an amount slightly greater than the displacement of the constant-speed motor. This tends to give the winding roll a speed slightly greater than the feed roll speed. This determines the tension, because the winding roll cannot go faster than the feed roll. Both are in contact with the paper roll being wound. The pressure in the hydraulic line between the constant and variable displacement pumps will increase in proportion to the winding tension. For any setting of the winding speed controller on the variable delivery hydraulic pump, the motor speeds are generally constant. Thus, the surface speed of winding will remain substantially constant, regardless of the diameter of the roll being wound.



This is a hydraulic drive for fairly constant tension. The variable-delivery, constant-speed pumping unit supplies the oil to two constant displacement motors. One drives the apparatus that carries the fabric through the bath at a constant speed, and the other drives the winder. The two motors are in series, Motor A drives the winding reel, whose diameter increases from about 5 in. when the reel is empty to about 33 in. when the reel is full. Motor A is geared to the reel so that even when the reel is empty, the surface speed of paper travel will be somewhat faster than the mean rate of paper travel established by motor B, driving the apparatus. Only a small amount of oil will be bypassed through the choke located between the pressure and the return line.

When the roll is full, the revolutions per minute of the reel and its driving motor are only about one-seventh of the revolutions per minute when the reel is empty. More oil is forced through the choke when the reel is full because of the increased pressure in the line between the two motors. The pressure in this line increases as the reel diameter increases because the torque resistance encountered by the reel motor will be directly proportional to the reel diameter and because tension is constant. The larger the diameter of the fabric on the reel, the greater will be the torque exerted by the tension in the fabric. The installation is designed so that the torque developed by the motor driving the reel will be inversely proportional to the revolutions per minute of the reel. Hence, the tension on the fabric will remain fairly constant, regardless of the diameter of the reel. This drive is limited to about 3 hp, and it is relatively inefficient.

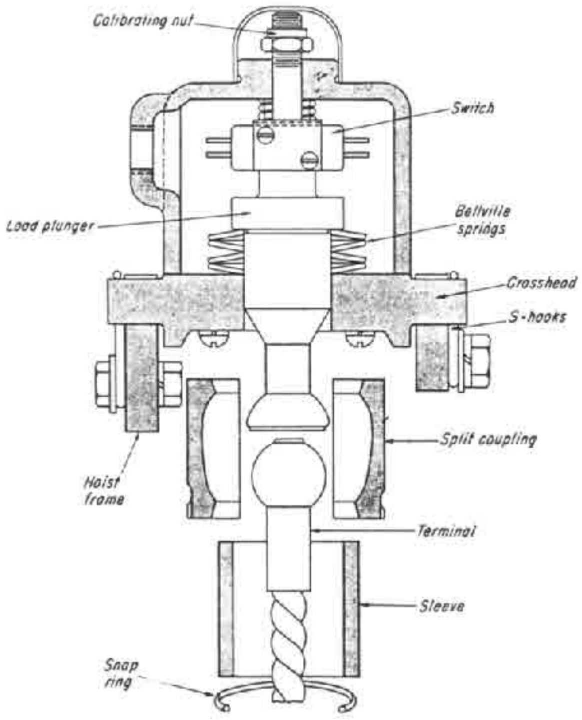


SWITCH PREVENTS OVERLOADING OF A HOIST

A fail-safe switch deactivates a lifting circuit if the load exceeds a preset value. Split coupling permits quick attachment of the cable.

A load plunger is inserted through the belleville springs, which are supported on a swiveling crosshead. The crosshead is mounted on the hoist frame and retained by two S-hooks and bolts. Under load, the belleville springs deflect and permit the load plunger to move axially. The end of the load plunger is connected to a normally closed switch. When the springs deflect beyond a preset value, the load plunger trips the switch, opening the raising-coil circuit of the magnetic hoist-controller. The raising circuit becomes inoperative, but the lowering circuit is not affected. A second contact, normally open, is included in the switch to permit the inclusion of visual or audible overload signal devices.

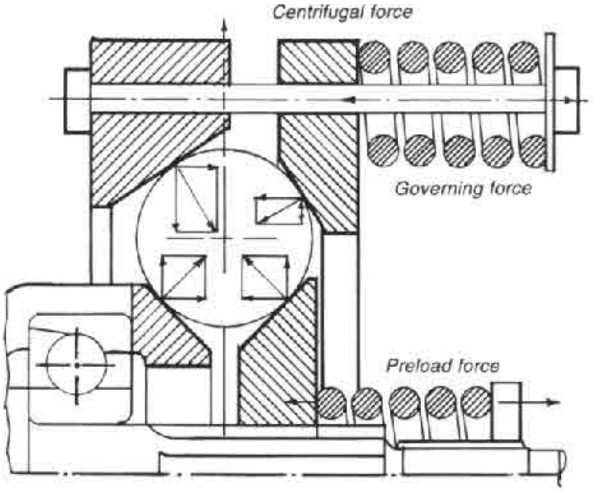
The load plunger and the swaged-on cable termination have ball-and-socket seat sections to permit maximum free cable movement, reducing the possibility of fatigue failure. A split-coupling and sleeve permits quick attachment of the cable-ball terminal to the load plunger.



Ball-Type Transmission Is Self-Governing

The Gerritsen transmission, developed in England at the Tiltman Langley Laboratories Ltd., Redhill Aerodrome, Surrey, governs its own output speed within limits of ±1%. The usual difficulties of speed governing—lack of sensitivity, lag, and hunting—associated with separate governor units are completely eliminated because regulation is effected directly by the driving members through their own centrifugal force. The driving members are precision bearing-steel balls that roll on four hardened-steel, cone-shaped rings. These members can be organized for different ratio arrangements.

The transmission can be used in three different ways: as a fixed “gear,” as an externally controlled variable-speed unit, or as a self-governing drive that produces a constant output speed form varying input speeds.



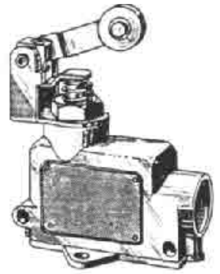
The self-governing action of the transmission is derived from the centrifugal forces of the balls as they rotate. When the balls move outward radially, the input-output ration changes. By properly arranging the rings and springs, the gear ration can be controlled by the movement of the balls to maintain a constant value of output speed.

MECHANICAL, GEARED, AND CAMMED LIMIT SWITCHES

Limit switches are electric current switching devices that are operated by some form or mechanical motion. Limit switches are usually installed in automatic machinery to control a complete operating cycle automatically by closing and opening electrical circuits in the proper sequence.

In addition to interlocking control circuits, limit switches have many other uses. For example, they are important as safety devices to stop a machine, sound a warning signal, or illuminate a warning light when a dangerous operating condition develops. Thus, properly applied switches can both control highly efficient automatic electric machinery and protect it and its operator.

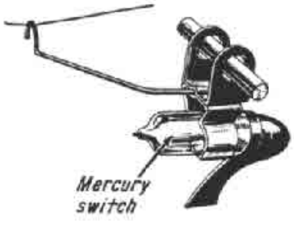
Actuators


BASIC SPRING-RETURN

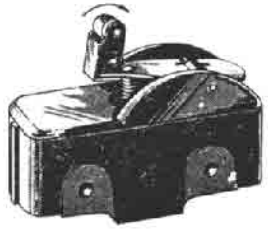

FOUR-POSITION HEAD

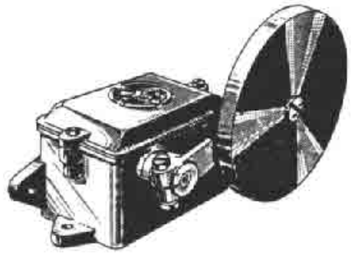

ADJUSTABLE LENGTH

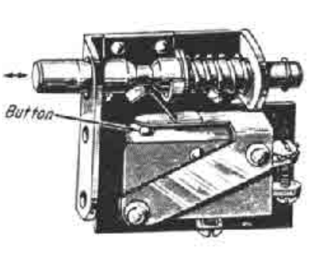

EXTENDED HOUSING



TILT-TO-ACTUATE


ONE-WAY OVERRIDE


OVERSIZE WHEEL



LINEAR CAM

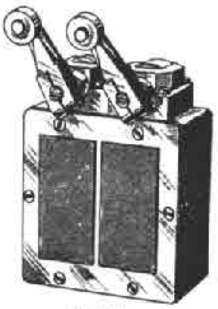
Linear mechanical switches

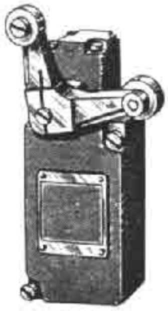

90-POSITION VERNIER

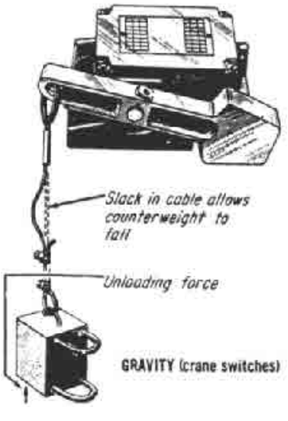

INFINITE-POSITION WORM


THREADED BUSHING

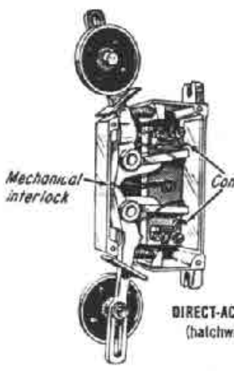
FLEXIBLE ROD


DUPLEX


BELLCRANK

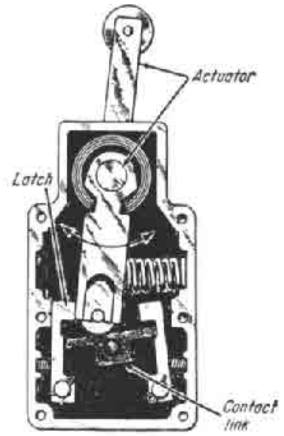


GRAVITY (crane switches)

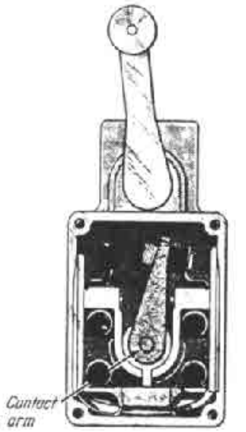


DIRECT-ACTING (hatchway)

Latching switch with contact chamber

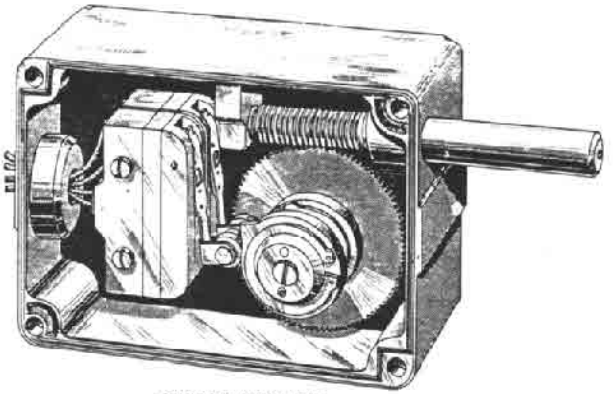


MECHANICAL HALF

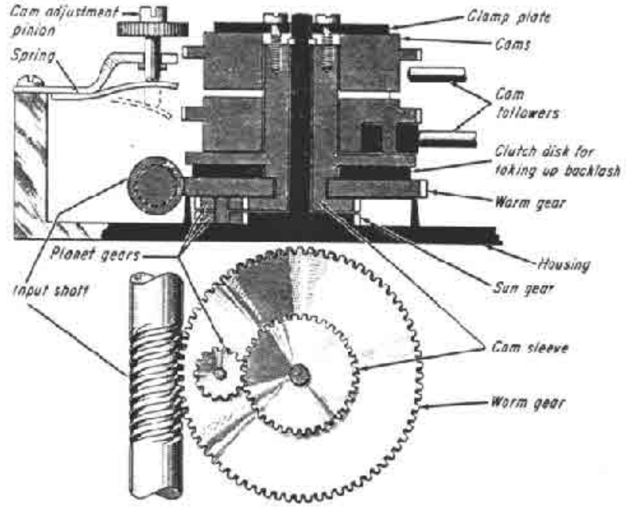


ELECTRICAL HALF

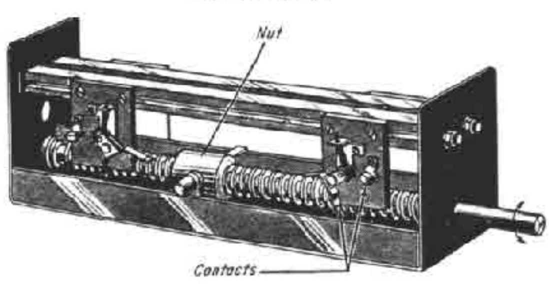
Geared rotary limit switches



SINGLE-REDUCTION WORM

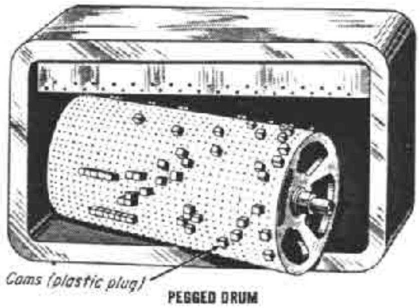


WORM AND PLANETARY

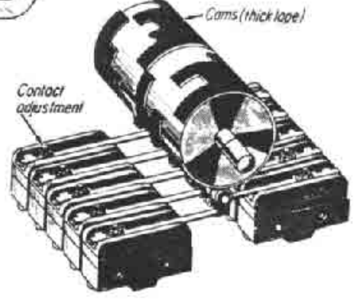


TRAVELING NUT

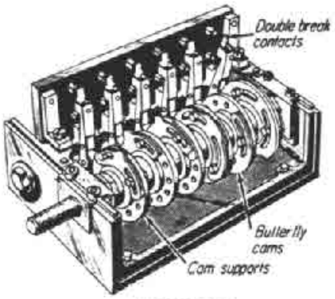
Rotary-cam limit switches



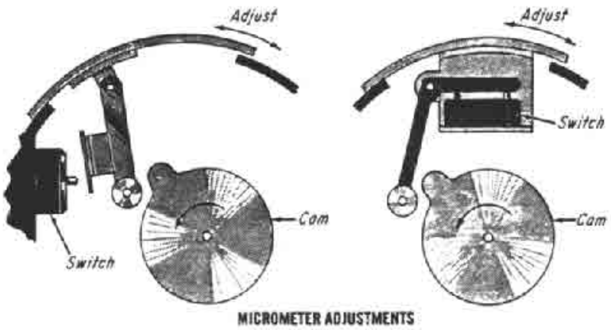
PEGGED DRUM



TAPED DRUM



MULTIPLE CAM



MICROMETER ADJUSTMENTS

LIMIT SWITCHES IN MACHINERY

Limit switches, which confine or restrain the travel or rotation of moving parts within certain predetermined points, are actuated by varying methods. Some of these, such as cams, rollers, push-rods, and traveling nuts, are described and illustrated.

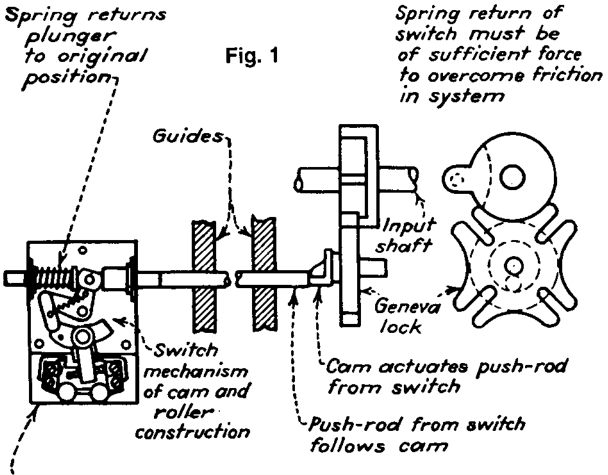


Fig. 1

Spring-return, two-circuit limit switch, stationary mounting

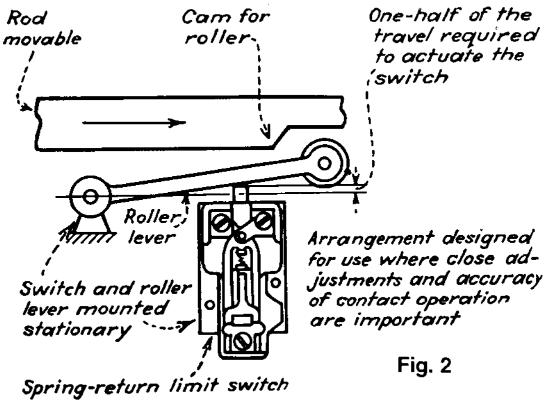


Fig. 2

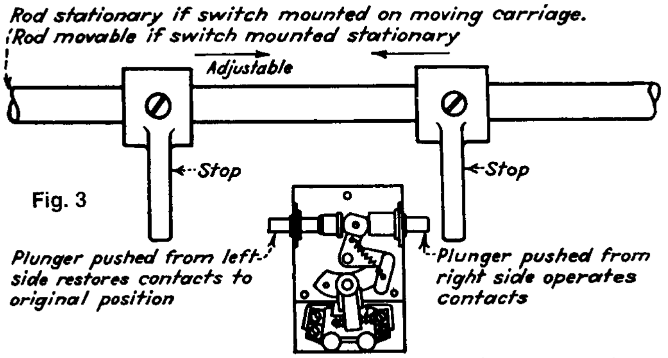
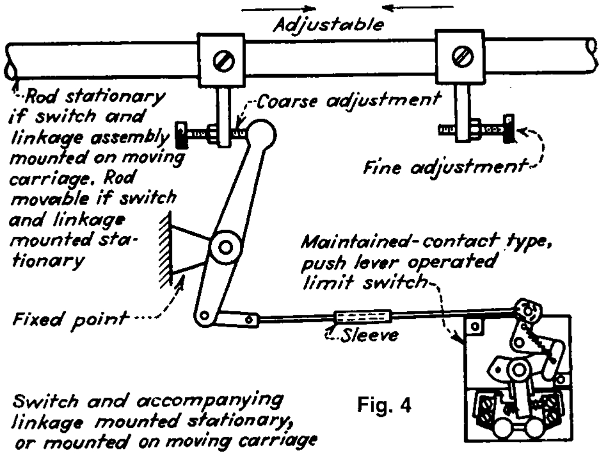


Fig. 3

Two-circuit, maintained-contact type limit switch mounted stationary, or mounted on moving carriage



Switch and accompanying linkage mounted stationary, or mounted on moving carriage

Fig. 4

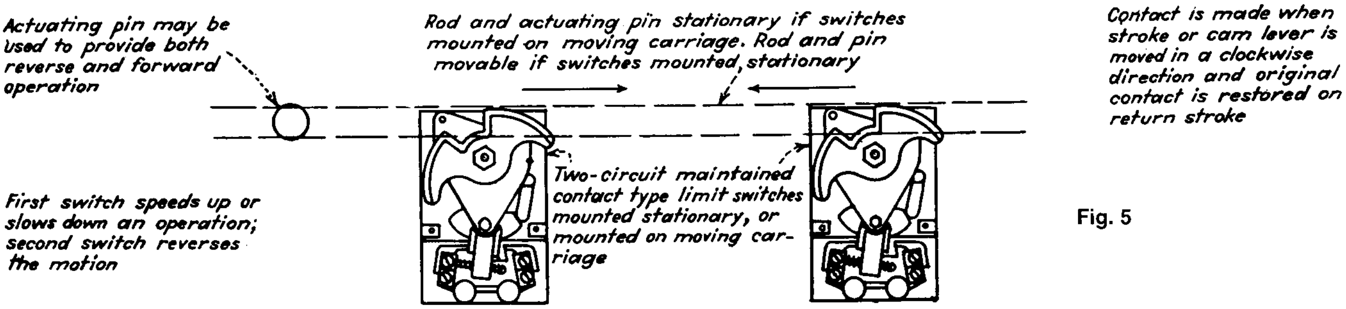
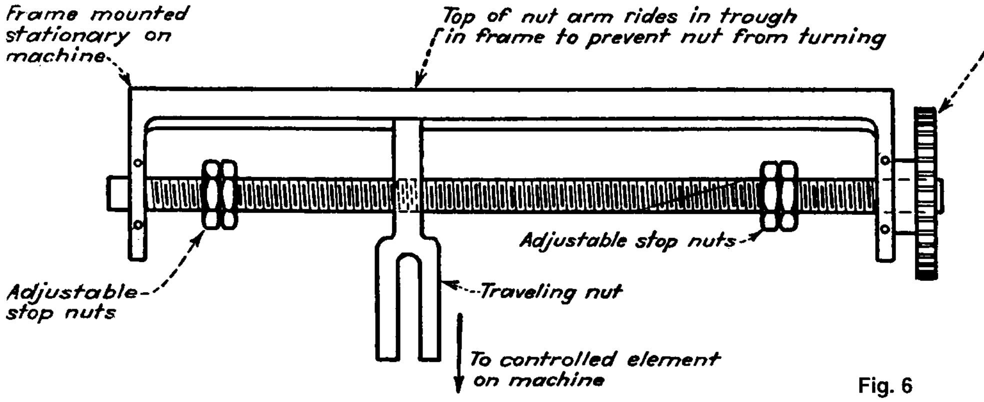
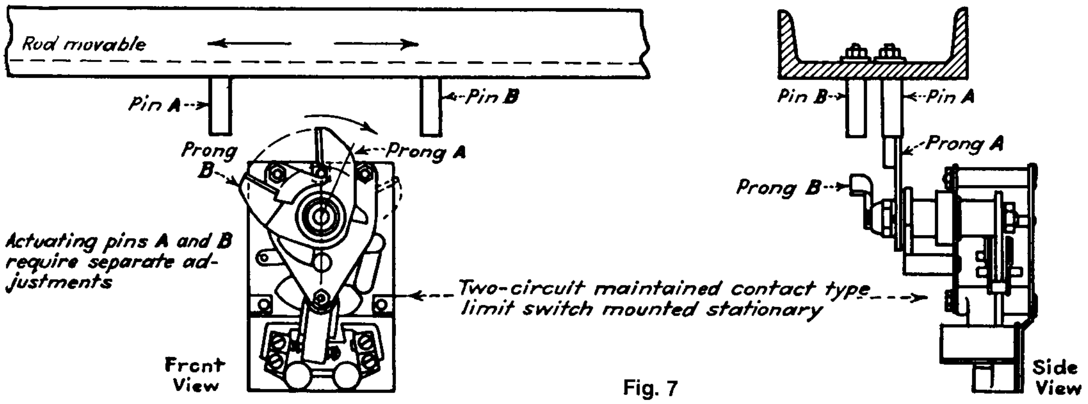


Fig. 5



Screw and nut device to control travel of machine by reversing belt or throwing out a catch after any specified number of revolutions.
Travel is adjusted by stop nuts

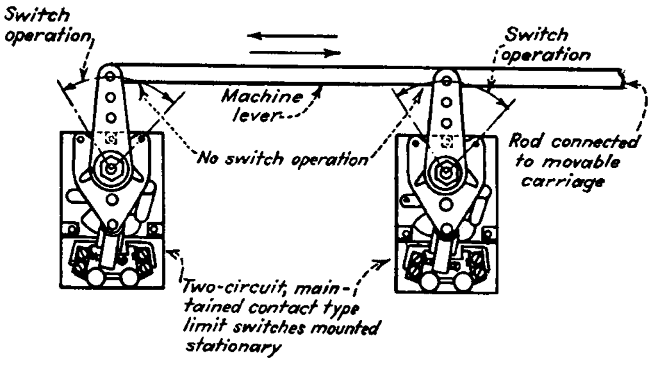
Fig. 6



Contact operation takes place when fork lever is moved. Contacts are restored when fork lever is moved in opposite direction

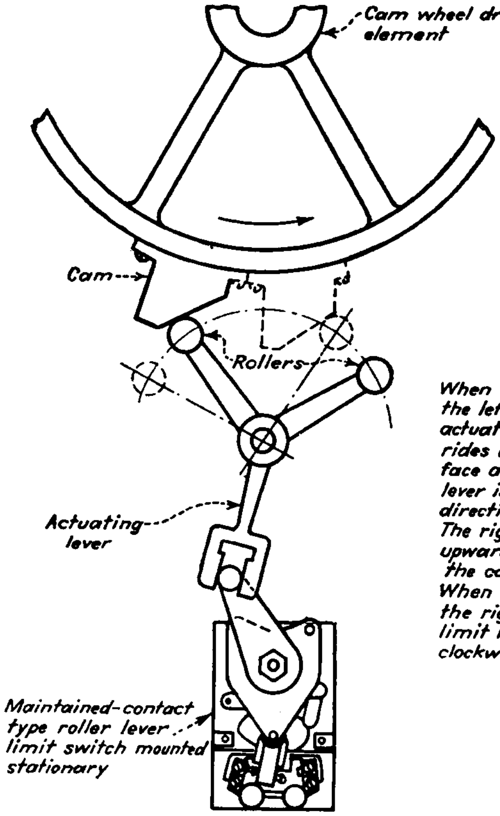
Fig. 7

Movement of the machine lever to the right operates the contacts of the right-hand switch, but no contact takes place in the left-hand switch.
Movement to the left operates the contacts in the left-hand switch, but no contact takes place in the right-hand switch



A spring return mechanism can be used if the weight and friction of the connecting linkage does not offset the power of the return spring

Fig. 8



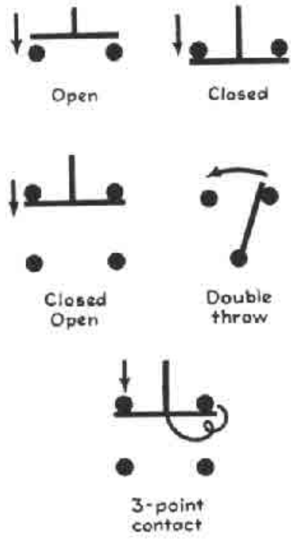
When the cam wheel meets the left-hand roller of the actuating lever, this roller rides up on the inclined surface and the limit switch lever is tripped in a clockwise direction.
The right-hand roller moves upwards into the path of the cam.
When the cam wheel meets the right-hand roller, the limit is tripped in a counterclockwise direction

Fig. 9

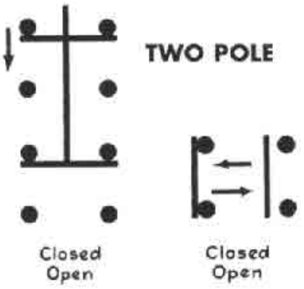
Electrical contact arrangements

All contracts in normal position with limit switch unactuated

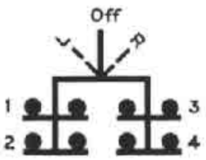
SINGLE POLE



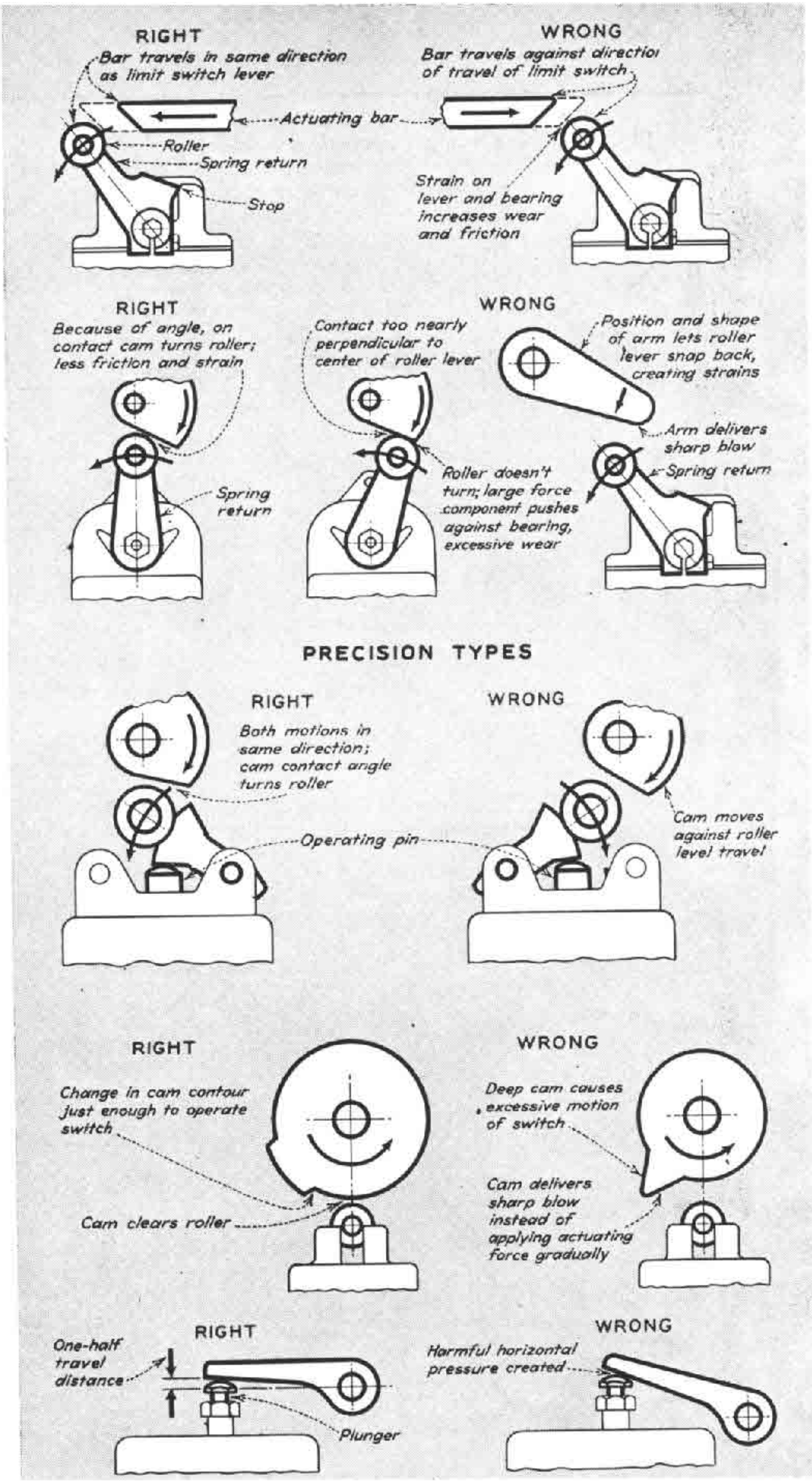
TWO POLE

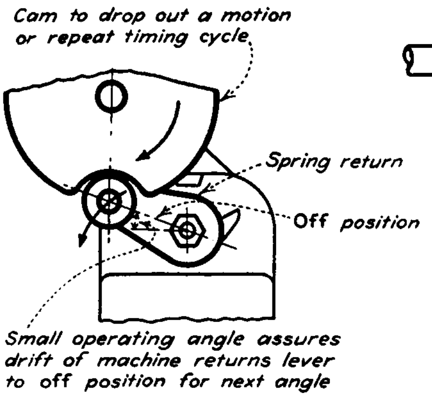


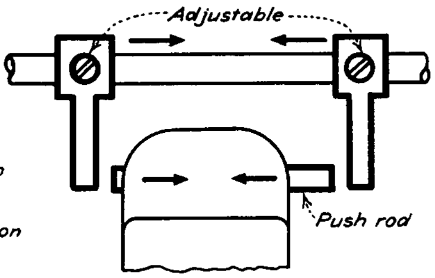
MULTI-CONTACT



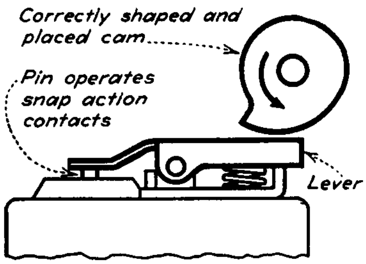
POS.	1	2	3	4
R	C	C	O	O
Off	C	C	C	C
L	O	O	C	C



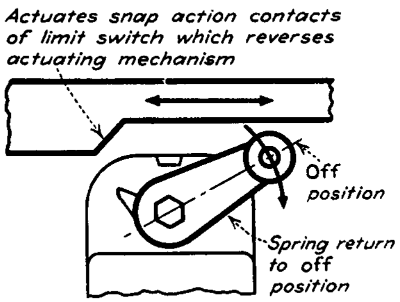


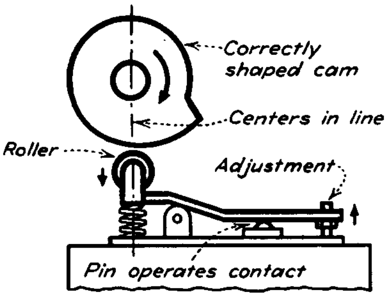


Limit switch with snap action maintained type contacts

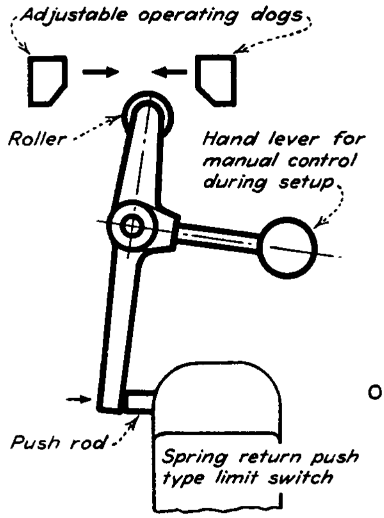


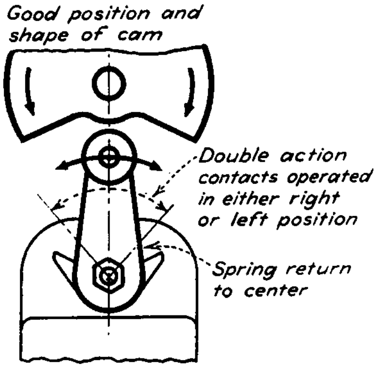
Precision type limit switch

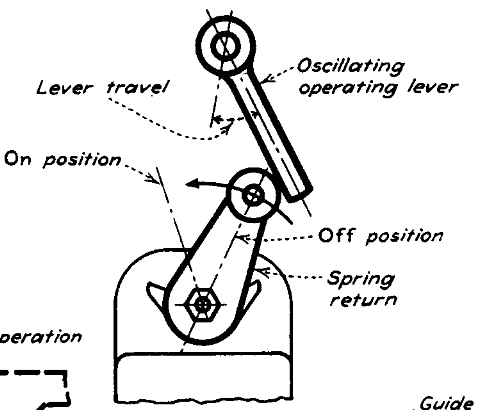


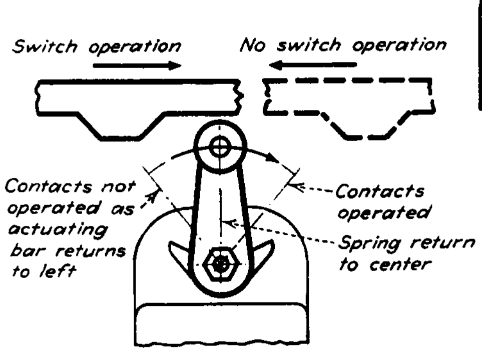


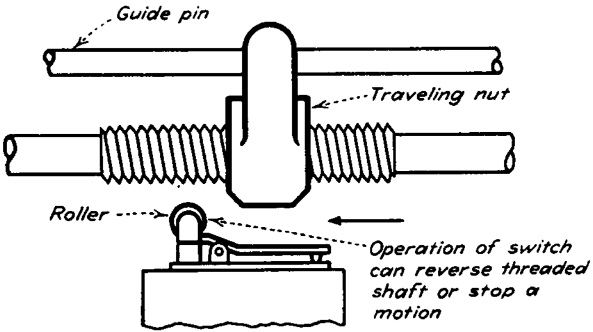
Precision type limit switch



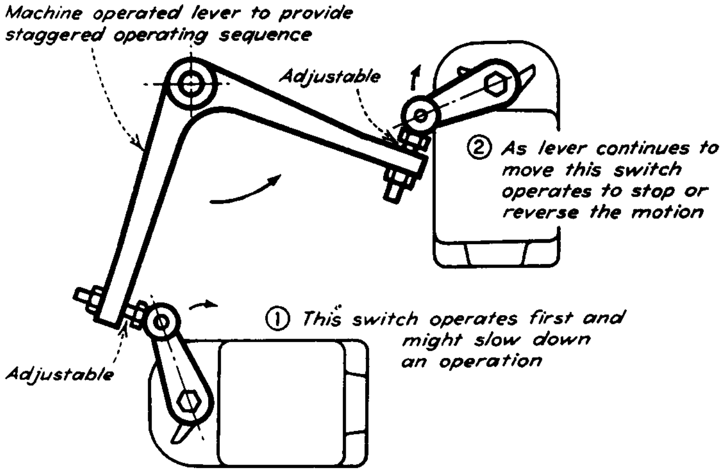


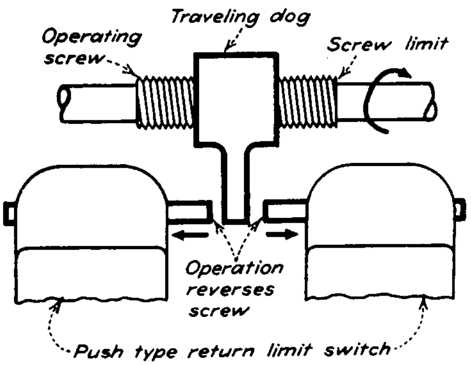






Precision type limit switch





AUTOMATIC SPEED GOVERNORS

Speed governors, designed to maintain the speeds of machines within reasonably constant limits, regardless of loads, depend for their action upon centrifugal force or cam linkages. Other governors depend on pressure differentials and fluid velocities for their actuation.

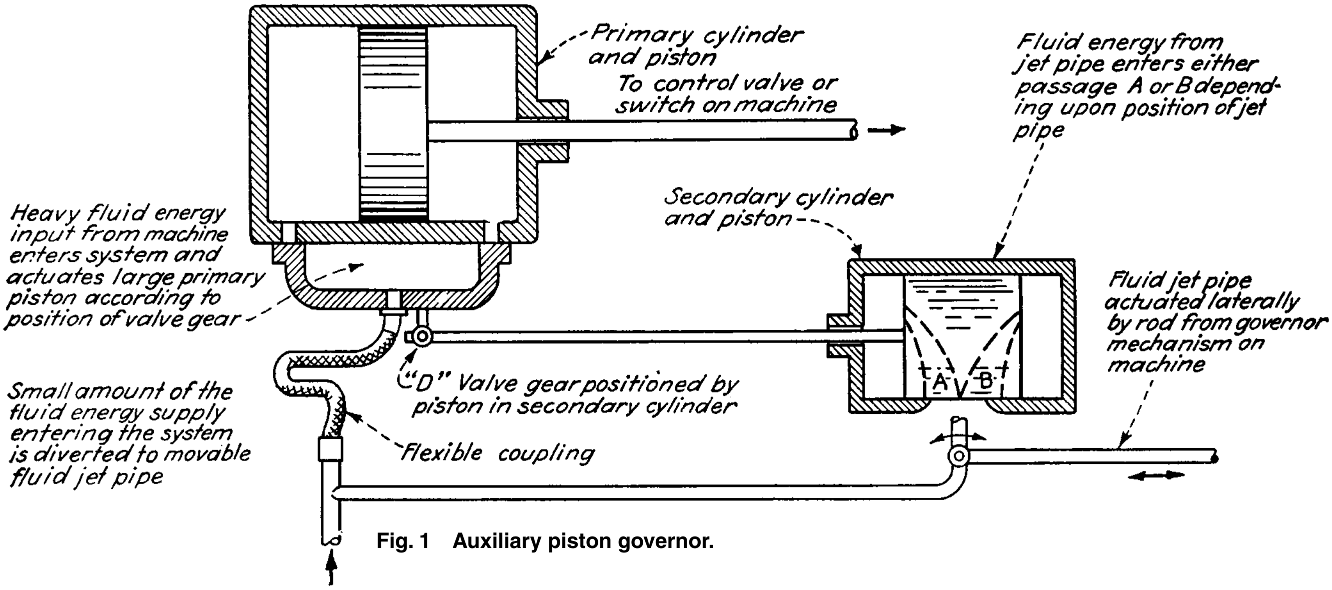


Fig. 1 Auxiliary piston governor.

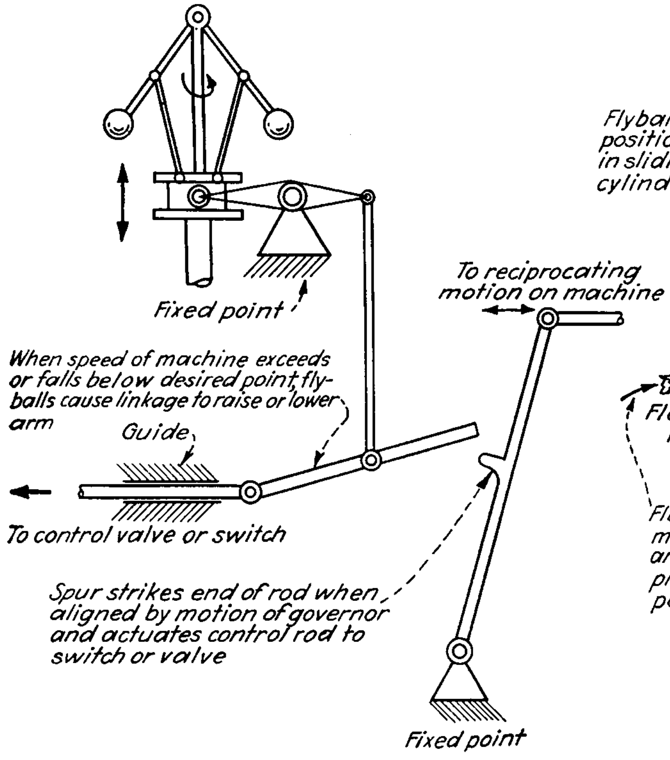


Fig. 2 Hit-and-miss governor.

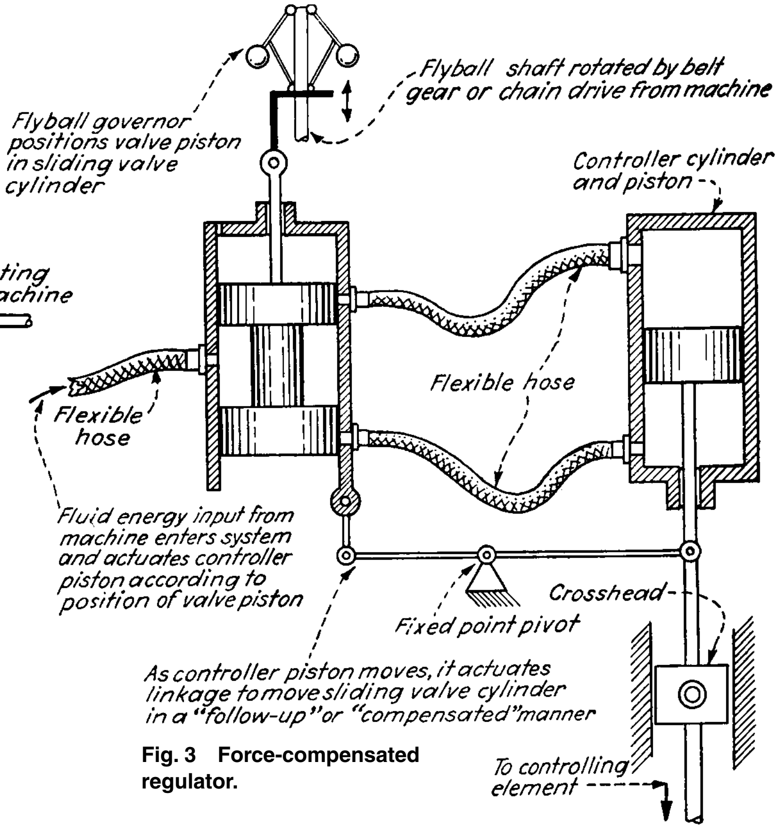


Fig. 3 Force-compensated regulator.

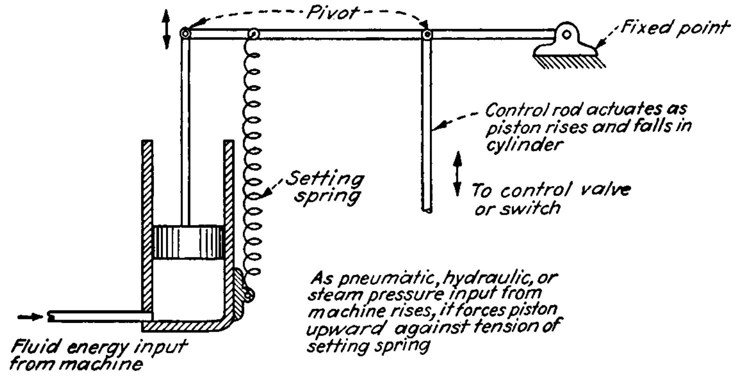


Fig. 4 Pressure-actuated governor.

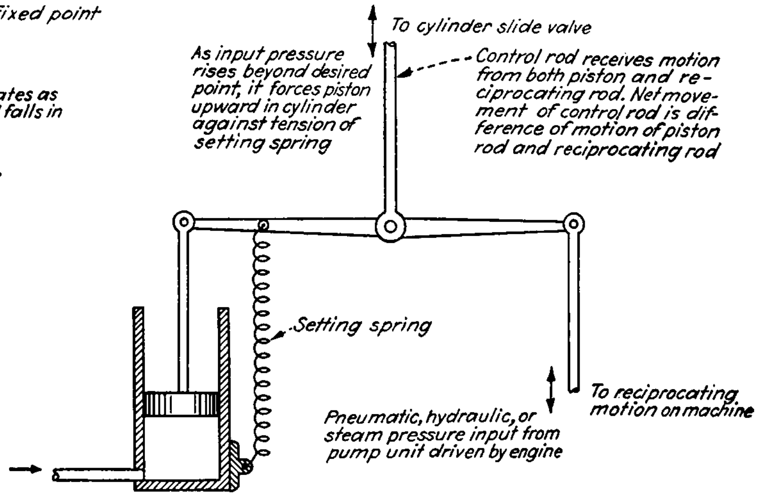


Fig. 5 Varying differential governor.

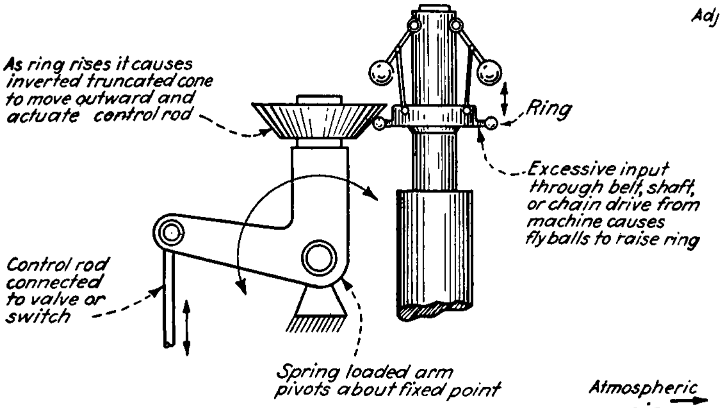


Fig. 6 Centrifugal governor.

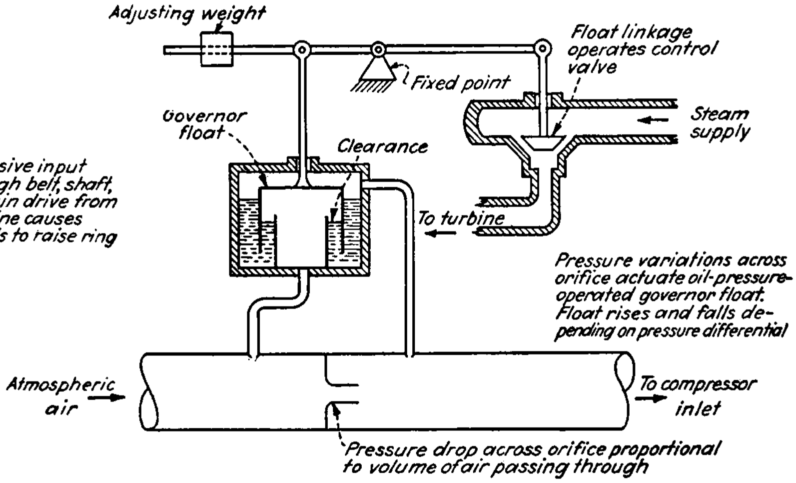


Fig. 7 Constant-volume governor.

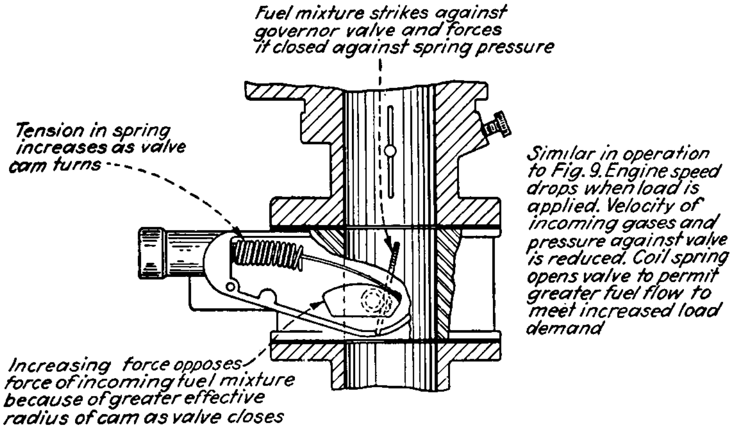


Fig. 8 **Velocity-type governor** (coil spring).

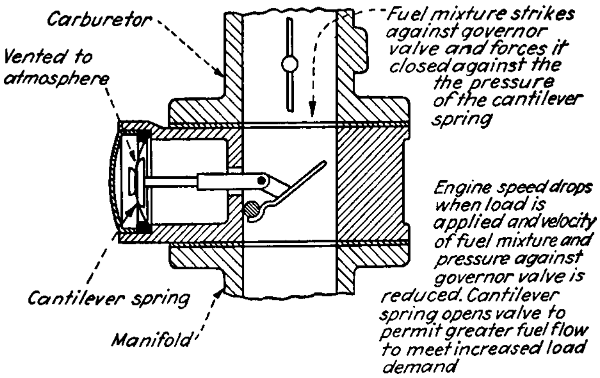


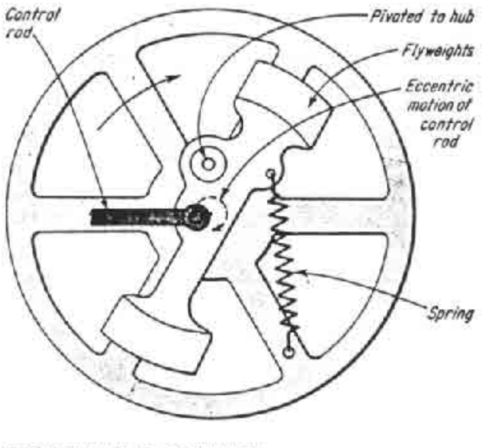
Fig. 9 **Velocity-type governor** (cantilever spring).

CENTRIFUGAL, PNEUMATIC, HYDRAULIC, AND ELECTRIC GOVERNORS

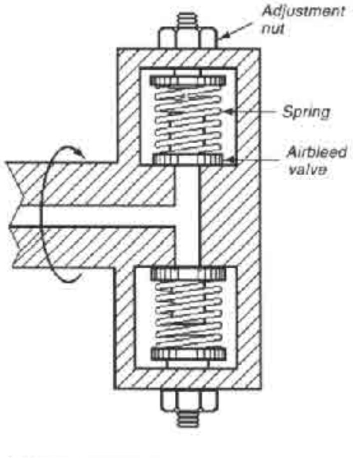
Centrifugal governors are the most common—they are simple and sensitive and have high output force. There is more published information on centrifugal governors than on all other types combined.

In operation, centrifugal flyweights develop a force proportional to the square of the speed, modified by linkages, as required. In small engines the flyweight movement can actuate the fuel throttle directly. Larger engines require amplifiers or relays. This has lead to innumerable combinations of pilot pistons, linear actuators, dashpots, compensators, and gear boxes.

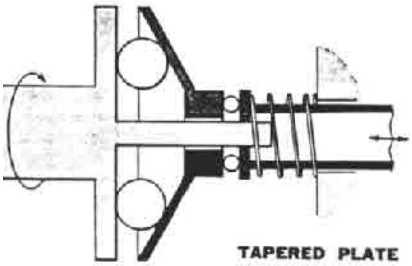
Centrifugal Governors



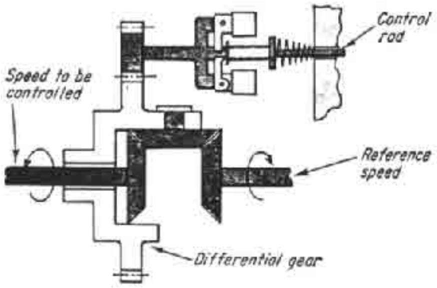
ACCELERATION GOVERNOR
(steam engine)



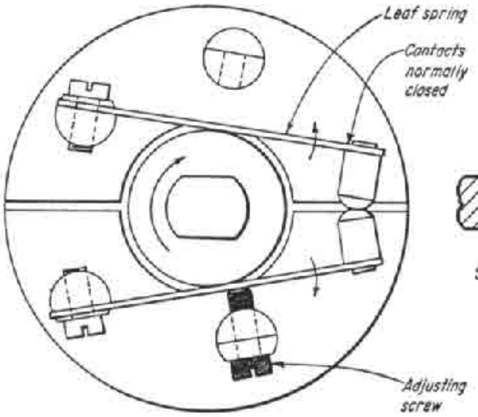
CENTRIFUGAL VALVE



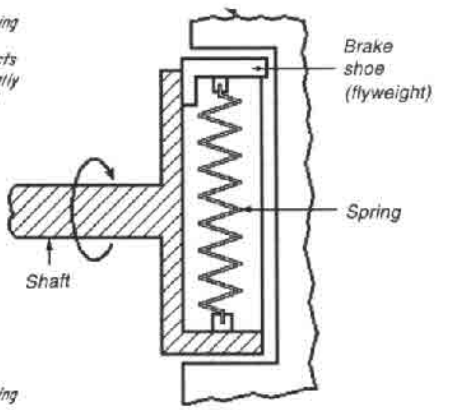
TAPERED PLATE



DIFFERENTIAL CENTRIFUGAL



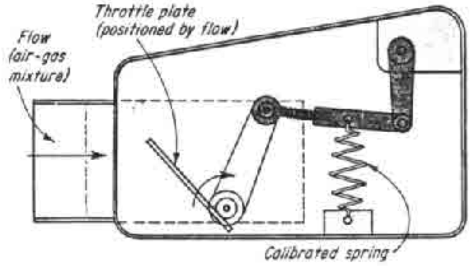
CENTRIFUGAL CONTACTS



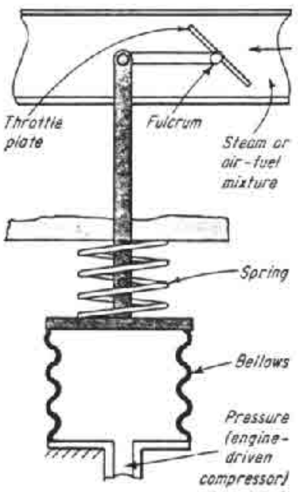
CENTRIFUGAL CLUTCH

Pneumatic sensors are the most inexpensive and also the most inaccurate of all speed-measuring and governing components. Nevertheless, they are entirely adequate for many applications. The pressure or velocity of cooling or combustion air is used to measure and govern the speed of the engine.

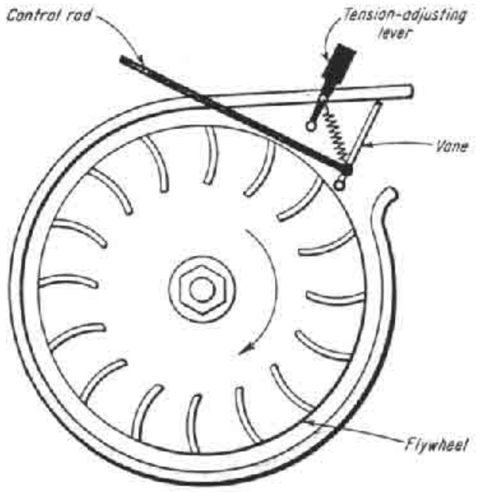
Pneumatic Governors



CARBURETOR-FLOW VELOCITY
(linkage)

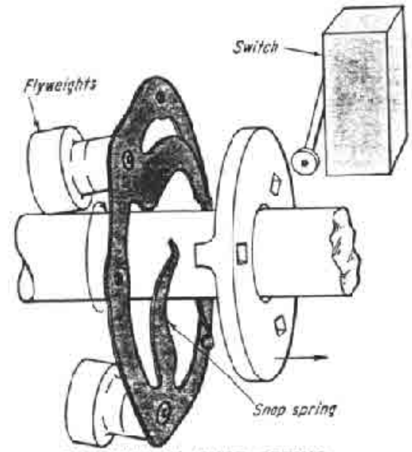


COMPRESSOR PRESSURE
(direct)

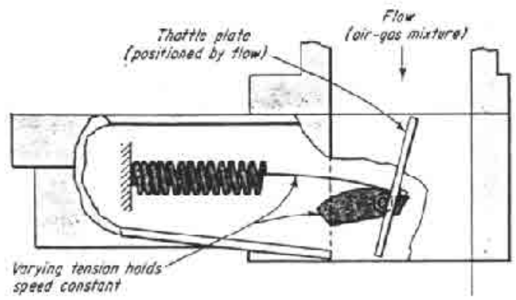


FAN-FLOW VELOCITY

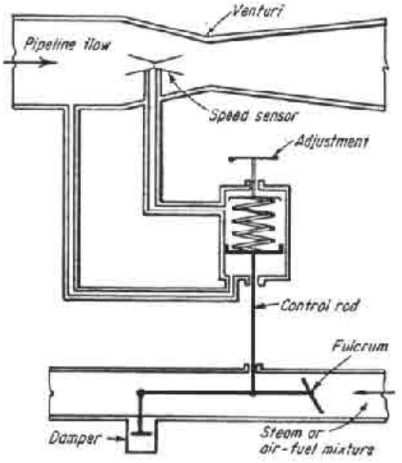
More pneumatic governors



CENTRIFUGAL SNAP SPRING



CARBURETOR-FLOW VELOCITY
(cam)



COMPRESSOR PRESSURE
(differential)

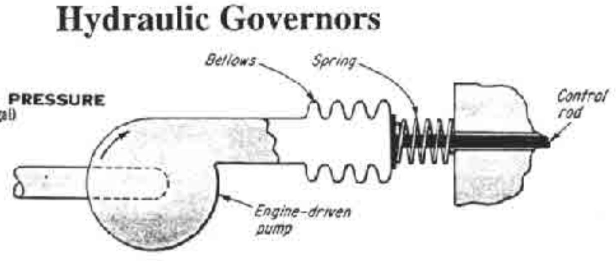
Hydraulic sensors measure the discharge pressure of engine-driven pumps. Pressure is proportional to the square of the speed of most pumps, although some have special impellers with linear pressure-speed characteristics.

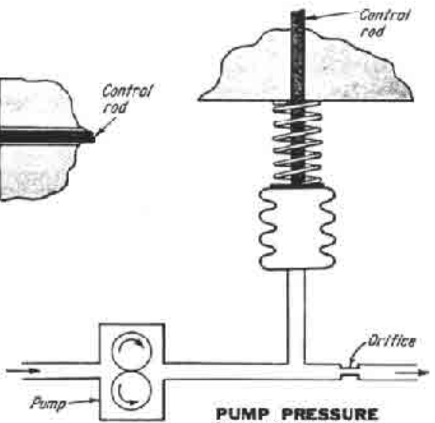
Straight vanes are better than curved vanes because the pressure is less affected by the volume flow. Low pressures are preferred over high pressures because fluid friction is less.

Typical applications for these governors include farm tractors with diesel or gasoline engines, larger diesel engines, and small steam turbines.

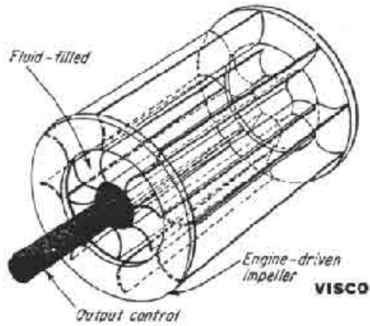
Hydraulic Governors

PUMP PRESSURE
centrifugal)



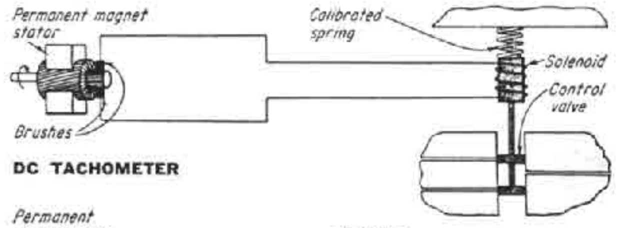


PUMP PRESSURE
(positive displacement)

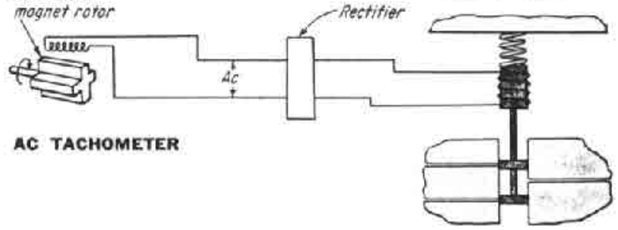


VISCOUS DRAG

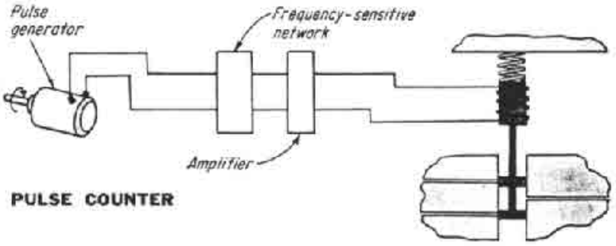
Electric Governors



DC TACHOMETER



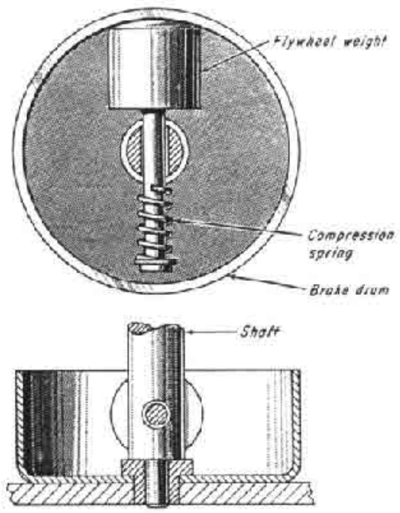
AC TACHOMETER



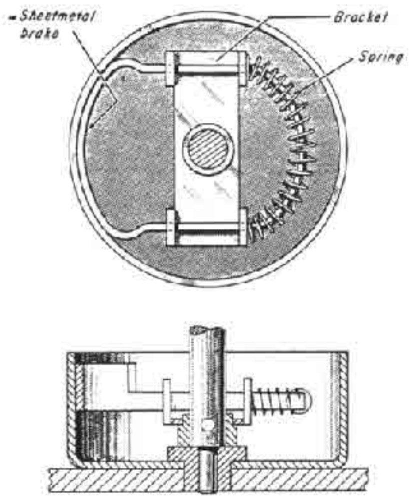
PULSE COUNTER

SPEED CONTROL DEVICES FOR MECHANISMS

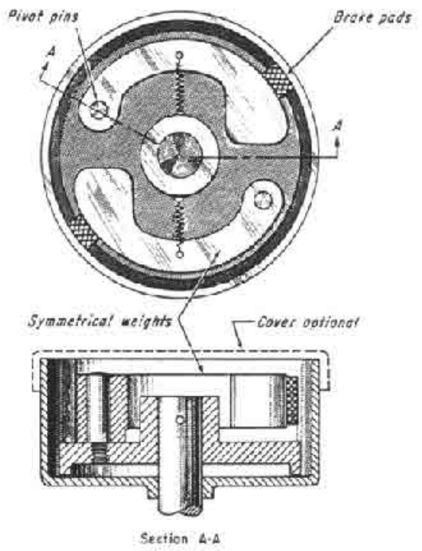
Friction devices, actuated by centrifugal force, automatically keep speed constant regardless of variations of load or driving force.



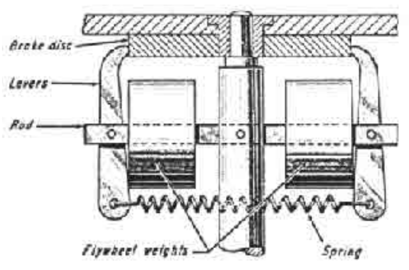
The weight is counterbalanced by a spring that brakes the shaft when the rotation speed becomes too fast. The braking surface is small.



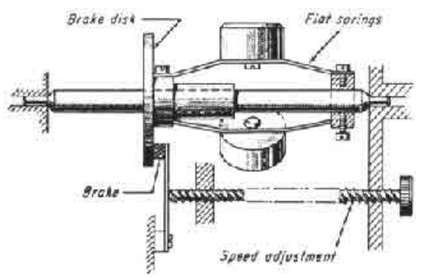
A sheetmetal brake provides a larger braking surface than in the previous brake. Braking is more uniform, and it generates less heat.



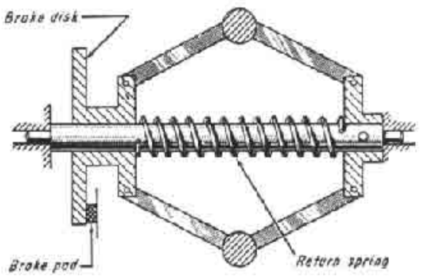
Symmetrical weights give an even braking action when they pivot outward. The entire action can be enclosed in a case.



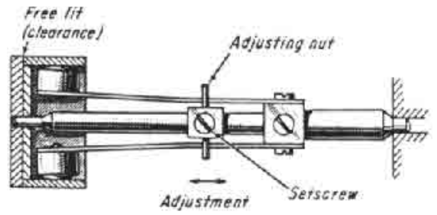
Weight-actuated levers make this arrangement suitable where high braking moments are required.



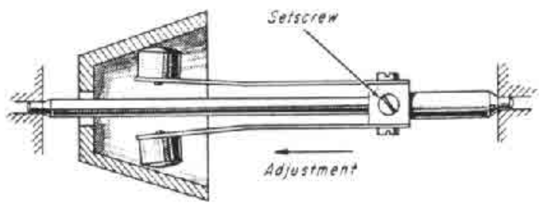
Three flat springs carry weights that provide a brake force upon rotation. A speed adjustment can be included.



The typical governor action of swinging weights is applied here. As in the previous brake, adjustment is optional.



The adjustment of the speed at which this device starts to brake is quick and easy. The adjusting nut is locked in place with a setscrew.



A tapered brake drum is another way to provide for varying speed-control. The adjustment is again locked.

FLOATING-PINION TORQUE SPLITTER

Designed-in looseness at the right locations helps to distribute torques more evenly.

Lewis Research Center, Cleveland, Ohio

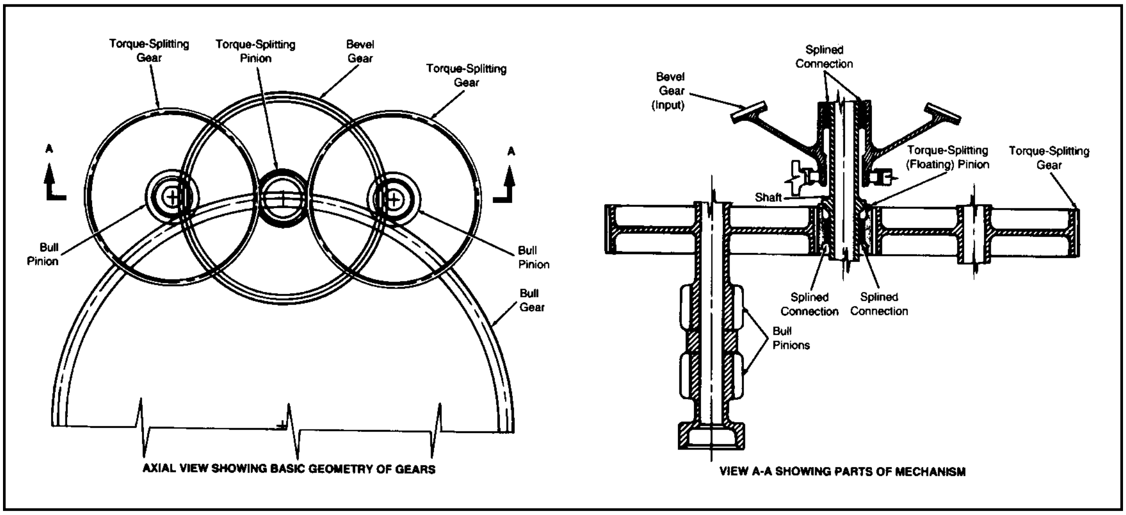
A gear-drive mechanism helps to apportion torques nearly equally along two parallel drive paths from an input bevel gear to an output bull gear. A mechanism of this type could be used, for example, as part of a redundant drive train between the engine and the rotor of a helicopter. The principal advantage of this torque-splitting mechanism is that it weighs less than comparably rated existing torque-splitting mechanisms.

The input torque is supplied to a bevel gear (see figure) from a bevel pinion (not shown) connected to the engine or other source. Overall, the torque is transmitted from the bevel gear through a torque-splitting pinion to two torque-splitting gears, then from the two torque-splitting gears through the associated two bull pinions to the bull gear, then to the output shaft. The purpose of the torque-splitting feature is to distribute the loads as nearly equally as possible to all gear teeth in the two parallel load paths to keep the load on each tooth as nearly equal as possible, thereby prolonging the life of the gear train.

In a redundant drive mechanism of the same basic configuration but without explicit provision for torque splitting, the slightest deviation from precision in machining could cause the entire load to be transmitted along one of the two paths while the gear and pinion in the other path rotate freely. To provide explicitly for torque splitting components made to manufacturing tolerances, elastic deformations, and other deviations from the nominal precise gearing geometry, it is necessary to incorporate a low-spring rate member at one or more critical locations in the mechanism.

In this mechanism, the resultant load on the torque-splitting pinion is zero when the torque is identical on the left and right members. If there is a difference in torque, the resultant load will displace the torque-splitting pinion until the loads are again in balance, thereby ensuring equal loads in each path.

This work was done by Harold W. Melles of United Technologies Corp. for **Lewis Research Center**.



Splined connections permit small angular excursions of a shaft, the bevel gear on its upper end, and the torque-splitting pinion on its lower end. These small excursions are essential for equalization of torques in the presence of machining tolerances and other geometric imperfection of the drive train.