MECHANISMS & MECHANICAL DEVICES SOURCEBOOK

Third Edition

NEIL SCLATER
NICHOLAS P. CHIRONIS

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ABOUT THE EDITORS

Neil Sclater began his career as an engineer in the military/aerospace industry and a Boston engineering consulting firm before changing his career path to writing and editing on electronics and electromechanical subjects. He was a staff editor for engineering publications in electronic design, instrumentation, and product engineering, including McGraw-Hill's *Product Engineering* magazine, before starting his own business as a consultant and contributing editor in technical communication.

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PREFACE

This is the third edition of *Mechanisms & Mechanical Devices Sourcebook*, a well illustrated reference book containing a wide range of information on both classical and modern mechanisms and mechanical devices. This edition retains a large core of the contents from both the first and second editions, (published in 1991 and 1996, respectively), that has been supplemented by new and revised articles reflecting present and future trends in mechanical engineering and machine design.

The new articles in this edition address topics that are covered regularly in mechanical engineering and science magazines as well as being the subjects of technical papers presented at engineering conferences. Among these new articles is an overview of motion control systems, highlighting the influence of programmable computer and digital technology on those systems. Other articles discuss servomotors, actuators, solenoids, and feedback sensors—important electromechanical, and electronic components used in motion control systems. Also included are articles on gearheads, single-axis motion guides, and X-Y motion systems assembled from stock mechanical components.

Other articles in this edition describe commercially available 2D and 3D CAD (computer-aided design) software and update previous articles on industrial robots and rapid prototyping (RP) systems. Another article reviews recent research in MEMS (microelectromechanical systems) and recent spinoffs of that technology. All of these subjects are continuing to influence the direction of mechanical engineering, and they are having a profound impact on engineering education and practice.

Since the publication of the second edition, the term *mechatronics* has gained wider acceptance as a word that identifies an ongoing trend in mechanical engineering—the merging of mechanics, electronics, and computer science. Coined in Japan in the 1970s, mechatronics describes the synergistic blend of technologies that has led to the creation of many new functional and adaptable products that could not have been produced with a purely mechanical approach. While there is no formal definition of mechatronics, most mechanical engineers agree on its meaning.

The concept of mechatronics has been illustrated as a Venn diagram showing four overlapping circles representing the fields of *mechanics, electronics, computers,* and *controls*. Over the years, this convergence has spawned the more specialized disciplines of *electromechanics, computer-aided design, control electronics,* and *digital control systems*, all considered to be within the purview of mechatronics. These specialties have, in turn, fostered the creation of the even more focused technologies of *system analysis, transducers, simulation,* and *microcontrollers*.

Some of the important consumer products that have been identified as resulting from the practice of mechatronics are the computer hard-disk drive, the inkjet printer, the digital video disk (DVD) player, and the camcorder. Examination of these products reveals that they are eclectic assemblies of different kinds of mechanical devices, motors, electronic circuits, and in some of them, optics.

The inclusion of such classical mechanical elements as gears, levers, clutches, cams, leadscrews, springs, and motors in those advanced products is evidence that they still perform valuable functions, making it quite likely that they will continue to be included in the new and different products to be developed in this century.

A major attraction of the earlier editions of this book has been their cores of illustrations and descriptions of basic mechanisms and mechanical devices, accompanied by useful applications information. This material has been culled from a wide range of books and magazines that were published during the last half century. In an era of rapidly changing technology, most of this hardware has retained its universal value. As a result, this book has become recognized worldwide as a unique repository of historical engineering drawings and data not available in other more formal books. These earlier editions have served as a convenient technical reference and even as inspirational "mind-joggers" for seasoned professional machine designers as well as learning aids for engineering students.

Readers trying to arrive at new and different solutions for machine design problems can thumb through these pages, study their many illustrations, and consider adapting some of the successful mechanical inventions of the past to their new applications. Thus, proven solutions from the past can be recycled to perform new duties in the present. An old invention might be transferred without modification, or perhaps it could be improved if made from newer materials by newer manufacturing methods. What is old can be new again! For those unable to find instant solutions, this book contains a chapter of tutorial text and formulas for the design of certain basic mechanisms from scratch.

It is assumed that the reader is familiar with the basics of mechanics gained from formal education, practical experience, or both. This book is expected to be of most value to practicing machine designers and mechanical engineers, but its contents should also be of use to machine design instructors at the college and vocational school level, amateur and professional inventors, and students of all of the engineering disciplines and physical sciences. Last but not least, it is hoped that the book will be attractive to those who simply enjoy looking at illustrations of machines and figuring out how they work.

The drawings in this edition have stood the test of time. Certain material published in the previous editions has been deleted because reader feedback suggested that important design details were missing or unclear. Also, some material considered to be obsolete and unsuitable for new designs was deleted. For example, clockwork mechanisms for timing, control, and display have almost universally been replaced in contemporary designs by more cost-effective electronic modules that perform the same functions.

References to manufacturers or publications that no longer exist were deleted so that readers will not waste time trying to contact them for further information. However, the names of the inventors, where previously given, have been retained to help the reader who may want to do further research on any patents now or once held by those individuals.

Many of the mechanisms illustrated in this book were invented by anonymous artisans, millwrights, instrument makers, and mechanics over the past centuries. They left behind the sketches, formal drawings, and even the working models on which many of the illustrations in this book are based. It is also worth noting that many of the most influential machines from the water pump, steam engine, and chronometer, to the cotton gin, and airplane were invented by self-trained engineers, scientists, and technicians.

By themselves, many of the mechanisms and devices described in this book are just mechanical curiosities, but when integrated by creative minds with others, they can perform new and different functions. One need only consider the role of basic mechanisms in the crucial inventions of the past century—the airplane, the helicopter, the jet engine, the programmable robot, and most of our familiar home appliances.

Have you noticed how the size of objects is both increasing and decreasing? There is now a 142,000-ton cruise ship that can accommodate more than 5000 passengers, and plans have been announced for building jumbo jet aircraft capable of carrying more than 500 passengers. Moreover, laptop computers now have more computing power than mainframe computers that filled a room a quarter century ago. Work is progressing in efforts to combine the functions of computer, cellular telephone, personal digital assistant (PDA), and Internet-access terminal in a single wireless handheld device.

MEMS are expected to evolve beyond their current prime roles as sensors to become security locks for computers, optical switches, and practical micromachines. Meanwhile, scientists are studying the feasibility of microminiature, self-propelled capsules made with even smaller-scale nanotechnology that could navigate through the human body and seek out, diagnose, and treat diseases at their source.

—Neil Sclater

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CHAPTER 1
MOTION CONTROL SYSTEMS

MOTION CONTROL SYSTEMS OVERVIEW

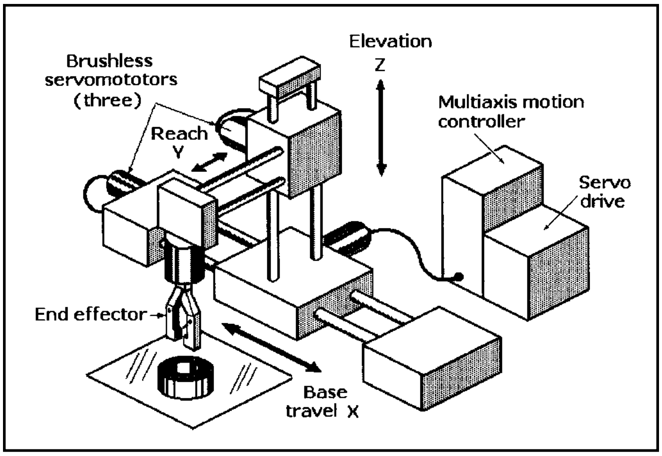


Fig. 1 This multiaxis X-Y-Z motion platform is an example of a motion control system.

Introduction

A modern motion control system typically consists of a motion controller, a motor drive or amplifier, an electric motor, and feedback sensors. The system might also contain other components such as one or more belt-, ballscrew-, or leadscrew-driven linear guides or axis stages. A motion controller today can be a standalone programmable controller, a personal computer containing a motion control card, or a programmable logic controller (PLC).

All of the components of a motion control system must work together seamlessly to perform their assigned functions. Their selection must be based on both engineering and economic considerations. Figure 1 illustrates a typical multiaxis X-Y-Z motion platform that includes the three linear axes required to move a load, tool, or end effector precisely through three degrees of freedom. With additional mechanical or electromechanical components on each axis, rotation about the three axes can provide up to six degrees of freedom, as shown in Fig. 2.

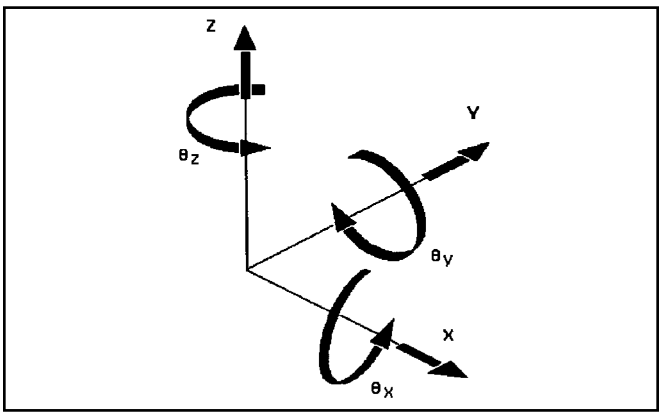


Fig. 2 The right-handed coordinate system showing six degrees of freedom.

Motion control systems today can be found in such diverse applications as materials handling equipment, machine tool centers, chemical and pharmaceutical process lines, inspection stations, robots, and injection molding machines.

Merits of Electric Systems

Most motion control systems today are powered by electric motors rather than hydraulic or pneumatic motors or actuators because of the many benefits they offer:

- More precise load or tool positioning, resulting in fewer product or process defects and lower material costs.
- Quicker changeovers for higher flexibility and easier product customizing.
- Increased throughput for higher efficiency and capacity.
- Simpler system design for easier installation, programming, and training.
- Lower downtime and maintenance costs.
- Cleaner, quieter operation without oil or air leakage.

Electric-powered motion control systems do not require pumps or air compressors, and they do not have hoses or piping that can leak hydraulic fluids or air. This discussion of motion control is limited to electric-powered systems.

Motion Control Classification

Motion control systems can be classified as *open-loop* or *closed-loop*. An open-loop system does not require that measurements of any output variables be made to produce error-correcting signals; by contrast, a closed-loop system requires one or more feedback sensors that measure and respond to errors in output variables.

Closed-Loop System

A *closed-loop motion control system*, as shown in block diagram Fig. 3, has one or more feedback loops that continuously compare the system's response with input commands or settings to correct errors in motor and/or load speed, load position, or motor torque. Feedback sensors provide the electronic signals for correcting deviations from the desired input commands. Closed-loop systems are also called servosystems.

Each motor in a servosystem requires its own feedback sensors, typically encoders, resolvers, or tachometers that close

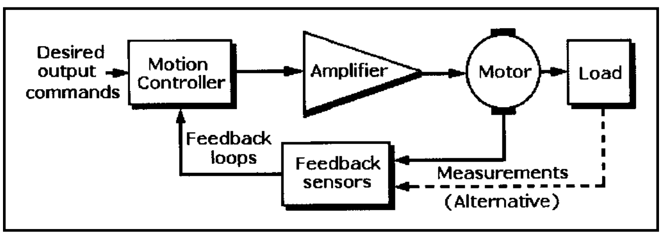


Fig. 3 Block diagram of a basic closed-loop control system.

loops around the motor and load. Variations in velocity, position, and torque are typically caused by variations in load conditions, but changes in ambient temperature and humidity can also affect load conditions.

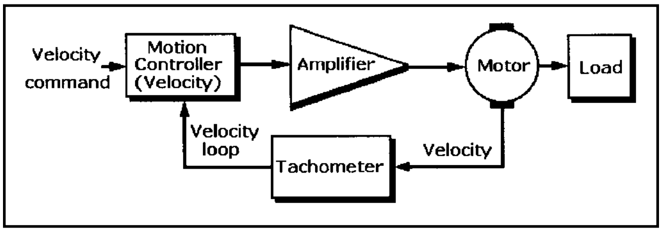


Fig. 4 Block diagram of a velocity-control system.

A *velocity control loop,* as shown in block diagram Fig. 4, typically contains a tachometer that is able to detect changes in motor speed. This sensor produces error signals that are proportional to the positive or negative deviations of motor speed from its preset value. These signals are sent to the motion controller so that it can compute a corrective signal for the amplifier to keep motor speed within those preset limits despite load changes.

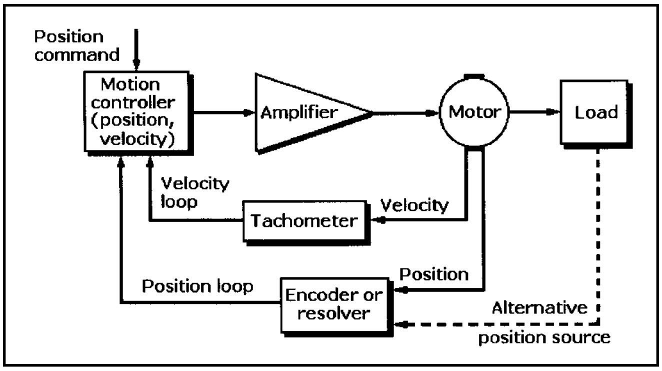


Fig. 5 Block diagram of a position-control system.

A *position-control loop,* as shown in block diagram Fig. 5, typically contains either an encoder or resolver capable of direct or indirect measurements of load position. These sensors generate error signals that are sent to the motion controller, which produces a corrective signal for amplifier. The output of the amplifier causes the motor to speed up or slow down to correct the position of the load. Most position control closed-loop systems also include a velocity-control loop.

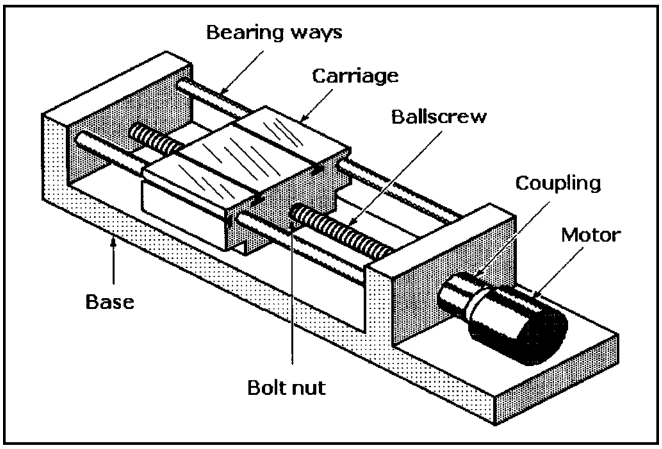


Fig. 6 Ballscrew-driven single-axis slide mechanism without position feedback sensors.

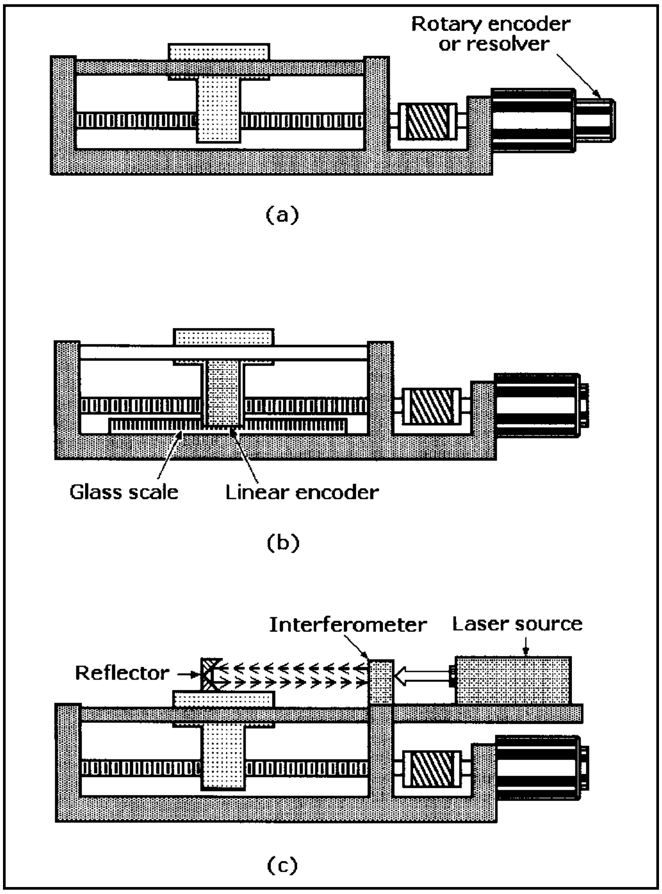


Fig. 7 Examples of position feedback sensors installed on a ballscrew-driven slide mechanism: (a) rotary encoder, (b) linear encoder, and (c) laser interferometer.

The *ballscrew slide mechanism*, shown in Fig. 6, is an example of a mechanical system that carries a load whose position must be controlled in a closed-loop servosystem because it is not equipped with position sensors. Three examples of feedback sensors mounted on the ballscrew mechanism that can provide position feedback are shown in Fig. 7: (a) is a rotary optical encoder mounted on the motor housing with its shaft coupled to the motor shaft; (b) is an optical linear encoder with its graduated scale mounted on the base of the mechanism; and (c) is the less commonly used but more accurate and expensive laser interferometer.

A *torque-control loop* contains electronic circuitry that measures the input current applied to the motor and compares it with a value proportional to the torque required to perform the desired task. An error signal from the circuit is sent to the motion controller, which computes a corrective signal for the motor amplifier to keep motor current, and hence torque, constant. Torque-control loops are widely used in machine tools where the load can change due to variations in the density of the material being machined or the sharpness of the cutting tools.

Trapezoidal Velocity Profile

If a motion control system is to achieve smooth, high-speed motion without overstressing the servomotor, the motion controller must command the motor amplifier to ramp up motor velocity gradually until it reaches the desired speed and then ramp it down gradually until it stops after the task is complete. This keeps motor acceleration and deceleration within limits.

The trapezoidal profile, shown in Fig. 8, is widely used because it accelerates motor velocity along a positive linear "up-ramp" until the desired constant velocity is reached. When the

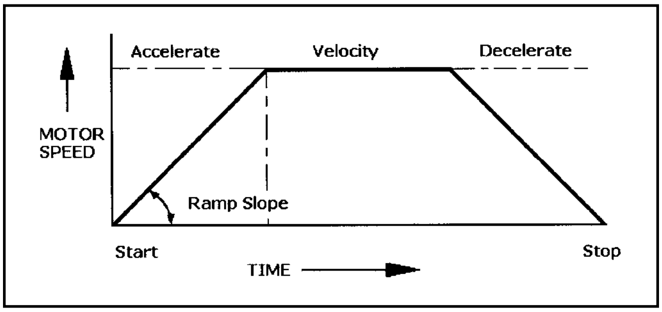


Fig. 8 Servomotors are accelerated to constant velocity and decelerated along a trapezoidal profile to assure efficient operation.

motor is shut down from the constant velocity setting, the profile decelerates velocity along a negative "down ramp" until the motor stops. Amplifier current and output voltage reach maximum values during acceleration, then step down to lower values during constant velocity and switch to negative values during deceleration.

Closed-Loop Control Techniques

The simplest form of feedback is *proportional control*, but there are also *derivative* and *integral control* techniques, which compensate for certain steady-state errors that cannot be eliminated from proportional control. All three of these techniques can be combined to form *proportional-integral-derivative (PID) control.*

- In *proportional control* the signal that drives the motor or actuator is directly proportional to the linear difference between the input command for the desired output and the measured actual output.
- In *integral control* the signal driving the motor equals the *time integral* of the difference between the input command and the measured actual output.
- In *derivative control* the signal that drives the motor is proportional to the *time derivative* of the difference between the input command and the measured actual output.
- In *proportional-integral-derivative (PID) control* the signal that drives the motor equals the weighted sum of the difference, the time integral of the difference, and the time derivative of the difference between the input command and the measured actual output.

Open-Loop Motion Control Systems

A typical *open-loop motion control system* includes a stepper motor with a programmable indexer or pulse generator and motor driver, as shown in Fig. 9. This system does not need feedback sensors because load position and velocity are controlled by the predetermined number and direction of input digital pulses sent to the motor driver from the controller. Because load position is not continuously sampled by a feedback sensor (as in a closed-loop servosystem), load positioning accuracy is lower and position errors (commonly called step errors) accumulate over time. For these reasons open-loop systems are most often specified in applications where the load remains constant, load motion is simple, and low positioning speed is acceptable.

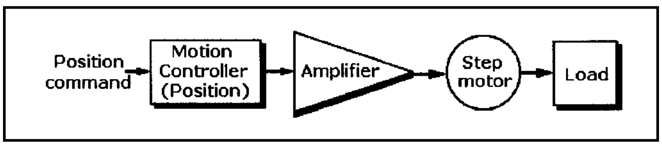


Fig. 9 Block diagram of an open-loop motion control system.

Kinds of Controlled Motion

There are five different kinds of motion control: *point-to-point, sequencing, speed, torque,* and *incremental.*

- In *point-to-point motion control* the load is moved between a sequence of numerically defined positions where it is stopped before it is moved to the next position. This is done at a constant speed, with both velocity and distance monitored by the motion controller. Point-to-point positioning can be performed in single-axis or multiaxis systems with servomotors in closed loops or stepping motors in open loops. X-Y tables and milling machines position their loads by multiaxis point-to-point control.
- *Sequencing control* is the control of such functions as opening and closing valves in a preset sequence or starting and stopping a conveyor belt at specified stations in a specific order.
- *Speed control* is the control of the velocity of the motor or actuator in a system.
- *Torque control* is the control of motor or actuator current so that torque remains constant despite load changes.
- *Incremental motion control* is the simultaneous control of two or more variables such as load location, motor speed, or torque.

Motion Interpolation

When a load under control must follow a specific path to get from its starting point to its stopping point, the movements of the axes must be coordinated or interpolated. There are three kinds of interpolation: *linear, circular, and contouring.*

Linear interpolation is the ability of a motion control system having two or more axes to move the load from one point to another in a straight line. The motion controller must determine the speed of each axis so that it can coordinate their movements. True linear interpolation requires that the motion controller modify axis acceleration, but some controllers approximate true linear interpolation with programmed acceleration profiles. The path can lie in one plane or be three dimensional.

Circular interpolation is the ability of a motion control system having two or more axes to move the load around a circular trajectory. It requires that the motion controller modify load acceleration while it is in transit. Again the circle can lie in one plane or be three dimensional.

Contouring is the path followed by the load, tool, or end-effector under the coordinated control of two or more axes. It requires that the motion controller change the speeds on different axes so that their trajectories pass through a set of predefined points. Load speed is determined along the trajectory, and it can be constant except during starting and stopping.

Computer-Aided Emulation

Several important types of programmed computer-aided motion control can emulate mechanical motion and eliminate the need for actual gears or cams. *Electronic gearing* is the control by software of one or more axes to impart motion to a load, tool, or end effector that simulates the speed changes that can be performed by actual gears. *Electronic camming* is the control by software of one or more axes to impart a motion to a load, tool, or end effector that simulates the motion changes that are typically performed by actual cams.

Mechanical Components

The mechanical components in a motion control system can be more influential in the design of the system than the electronic circuitry used to control it. Product flow and throughput, human

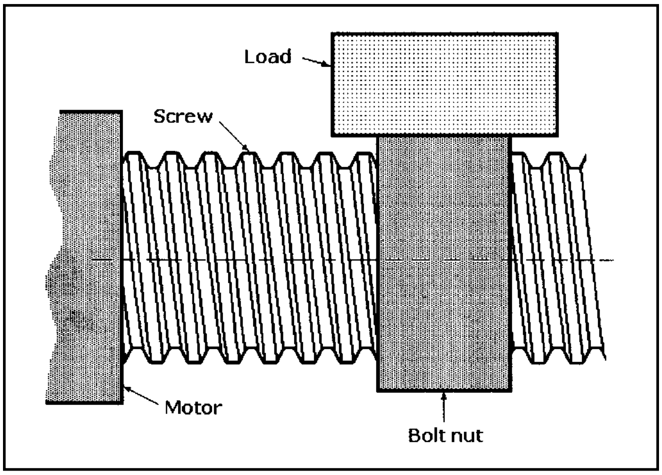


Fig. 10 Leadscrew drive: As the leadscrew rotates, the load is translated in the axial direction of the screw.

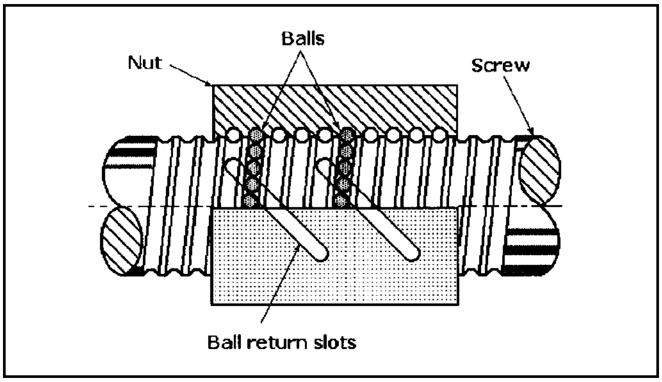


Fig. 11 Ballscrew drive: Ballscrews use recirculating balls to reduce friction and gain higher efficiency than conventional leadscrews.

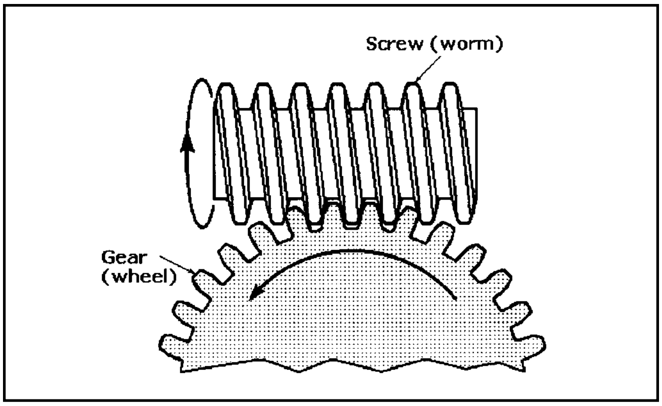


Fig. 12 Worm-drive systems can provide high speed and high torque.

operator requirements, and maintenance issues help to determine the mechanics, which in turn influence the motion controller and software requirements.

Mechanical actuators convert a motor's rotary motion into linear motion. Mechanical methods for accomplishing this include the use of leadscrews, shown in Fig. 10, ballscrews, shown in Fig. 11, worm-drive gearing, shown in Fig. 12, and belt, cable, or chain drives. Method selection is based on the relative costs of the alternatives and consideration for the possible effects of backlash. All actuators have finite levels of torsional and axial stiffness that can affect the system's frequency response characteristics.

Linear guides or stages constrain a translating load to a single degree of freedom. The linear stage supports the mass of the load to be actuated and assures smooth, straight-line motion while minimizing friction. A common example of a linear stage is a ballscrew-driven single-axis stage, illustrated in Fig. 13. The motor turns the ballscrew, and its rotary motion is translated into the linear motion that moves the carriage and load by the stage's bolt nut. The bearing ways act as linear guides. As shown in Fig. 7, these stages can be equipped with sensors such as a rotary or linear encoder or a laser interferometer for feedback.

A ballscrew-driven single-axis stage with a rotary encoder coupled to the motor shaft provides an indirect measurement. This method ignores the tolerance, wear, and compliance in the mechanical components between the carriage and the position encoder that can cause deviations between the desired and true positions. Consequently, this feedback method limits position accuracy to ballscrew accuracy, typically ±5 to 10 μm per 300 mm.

Other kinds of single-axis stages include those containing antifriction rolling elements such as recirculating and nonrecirculating balls or rollers, sliding (friction contact) units, air-bearing units, hydrostatic units, and magnetic levitation (Maglev) units.

A single-axis air-bearing guide or stage is shown in Fig. 14. Some models being offered are 3.9 ft (1.2 m) long and include a carriage for mounting loads. When driven by a linear servomotors the loads can reach velocities of 9.8 ft/s (3 m/s). As shown in Fig. 7, these stages can be equipped with feedback devices such

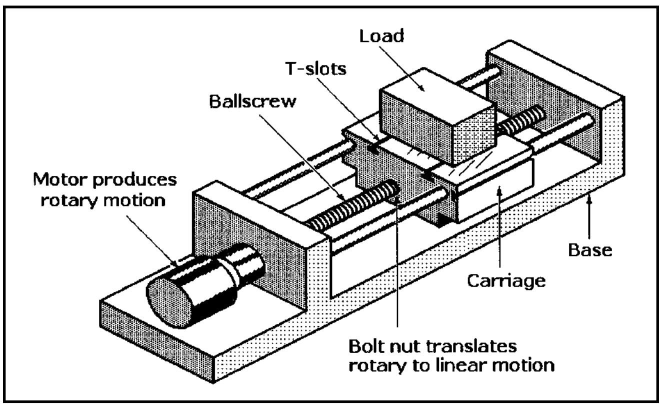


Fig. 13 Ballscrew-driven single-axis slide mechanism translates rotary motion into linear motion.

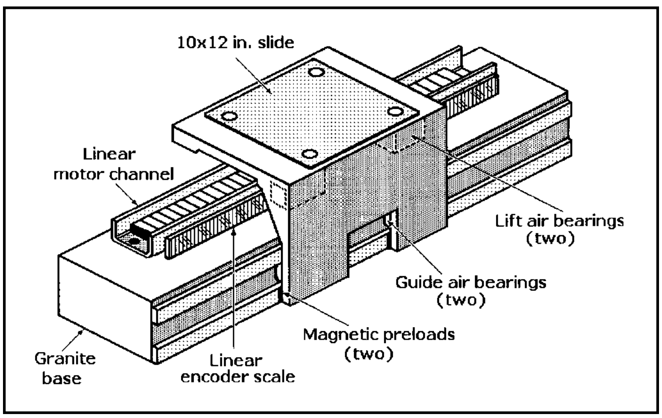


Fig. 14 This single-axis linear guide for load positioning is supported by air bearings as it moves along a granite base.

as cost-effective linear encoders or ultra-high-resolution laser interferometers. The resolution of this type of stage with a noncontact linear encoder can be as fine as 20 nm and accuracy can be ±1 µm. However, these values can be increased to 0.3 nm resolution and submicron accuracy if a laser interferometer is installed.

The pitch, roll, and yaw of air-bearing stages can affect their resolution and accuracy. Some manufacturers claim ±1 arc-s per 100 mm as the limits for each of these characteristics. Large air-bearing surfaces provide excellent stiffness and permit large load-carrying capability.

The important attributes of all these stages are their dynamic and static friction, rigidity, stiffness, straightness, flatness, smoothness, and load capacity. Also considered is the amount of work needed to prepare the host machine's mounting surface for their installation.

The structure on which the motion control system is mounted directly affects the system's performance. A properly designed base or host machine will be highly damped and act as a compliant barrier to isolate the motion system from its environment and minimize the impact of external disturbances. The structure must be stiff enough and sufficiently damped to avoid resonance problems. A high static mass to reciprocating mass ratio can also prevent the motion control system from exciting its host structure to harmful resonance.

Any components that move will affect a system's response by changing the amount of inertia, damping, friction, stiffness, or resonance. For example, a flexible shaft coupling, as shown in Fig. 15, will compensate for minor parallel (a) and angular (b) misalignment between rotating shafts. Flexible couplings are available in other configurations such as bellows and helixes, as shown in Fig. 16. The bellows configuration (a) is acceptable for light-duty applications where misalignments can be as great as 9º angular or ¼ in. parallel. By contrast, helical couplings (b) prevent backlash at constant velocity with some misalignment, and they can also be run at high speed.

Other moving mechanical components include cable carriers that retain moving cables, end stops that restrict travel, shock absorbers to dissipate energy during a collision, and way covers to keep out dust and dirt.

Electronic System Components

The motion controller is the "brain" of the motion control system and performs all of the required computations for motion path planning, servo-loop closure, and sequence execution. It is essentially a computer dedicated to motion control that has been programmed by the end user for the performance of assigned tasks. The motion controller produces a low-power motor command signal in either a digital or analog format for the motor driver or amplifier.

Significant technical developments have led to the increased acceptance of programmable motion controllers over the past five to ten years: These include the rapid decrease in the cost of microprocessors as well as dramatic increases in their computing power. Added to that are the decreasing cost of more advanced semiconductor and disk memories. During the past five to ten years, the capability of

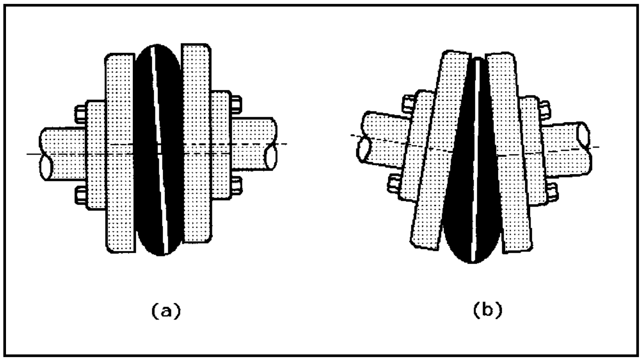


Fig. 15 Flexible shaft couplings adjust for and accommodate parallel misalignment (a) and angular misalignment between rotating shafts (b).

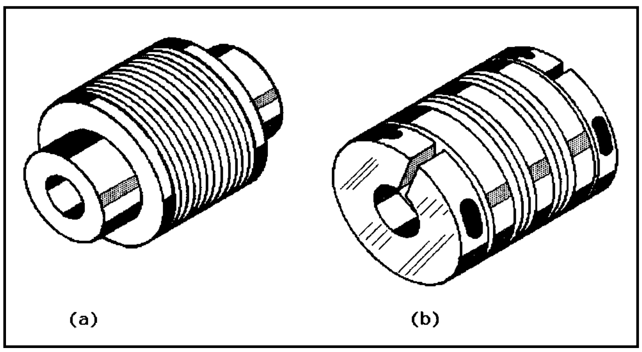


Fig. 16 Bellows couplings (a) are acceptable for light-duty applications. Misalignments can be 9º angular or ¼ in. parallel. Helical couplings (b) prevent backlash and can operate at constant velocity with misalignment and be run at high speed.

these systems to improve product quality, increase throughput, and provide just-in-time delivery has improved has improved significantly.

The motion controller is the most critical component in the system because of its dependence on software. By contrast, the selection of most motors, drivers, feedback sensors, and associated mechanisms is less critical because they can usually be changed during the design phase or even later in the field with less impact on the characteristics of the intended system. However, making field changes can be costly in terms of lost productivity.

The decision to install any of the three kinds of motion controllers should be based on their ability to control both the number and types of motors required for the application as well as the availability of the software that will provide the optimum performance for the specific application. Also to be considered are the system's multitasking capabilities, the number of input/output (I/O) ports required, and the need for such features as linear and circular interpolation and electronic gearing and camming.

In general, a motion controller receives a set of operator instructions from a host or operator interface and it responds with corresponding command signals for the motor driver or drivers that control the motor or motors driving the load.

Motor Selection

The most popular motors for motion control systems are stepping or stepper motors and permanent-magnet (PM) DC brush-type and brushless DC servomotors. Stepper motors are selected for systems because they can run open-loop without feedback sensors. These motors are indexed or partially rotated by digital pulses that turn their rotors a fixed fraction or a revolution where they will be clamped securely by their inherent holding torque. Stepper motors are cost-effective and reliable choices for many applications that do not require the rapid acceleration, high speed, and position accuracy of a servomotor.

However, a feedback loop can improve the positioning accuracy of a stepper motor without incurring the higher costs of a complete servosystem. Some stepper motor motion controllers can accommodate a closed loop.

Brush and brushless PM DC servomotors are usually selected for applications that require more precise positioning. Both of these motors can reach higher speeds and offer smoother low-speed operation with finer position resolution than stepper motors, but both require one or more feedback sensors in closed loops, adding to system cost and complexity.

Brush-type permanent-magnet (PM) DC servomotors have wound armatures or rotors that rotate within the magnetic field produced by a PM stator. As the rotor turns, current is applied sequentially to the appropriate armature windings by a mechanical commutator consisting of two or more brushes sliding on a ring of insulated copper segments. These motors are quite mature, and modern versions can provide very high performance for very low cost.

There are variations of the brush-type DC servomotor with its iron-core rotor that permit more rapid acceleration and deceleration because of their low-inertia, lightweight cup- or disk-type armatures. The disk-type armature of the pancake-frame motor, for example, has its mass concentrated close to the motor's faceplate permitting a short, flat cylindrical housing. This configuration makes the motor suitable for faceplate mounting in restricted space, a feature particularly useful in industrial robots or other applications where space does not permit the installation of brackets for mounting a motor with a longer length dimension.

The brush-type DC motor with a cup-type armature also offers lower weight and inertia than conventional DC servomotors. However, the tradeoff in the use of these motors is the restriction on their duty cycles because the epoxy-encapsulated armatures are unable to dissipate heat buildup as easily as iron-core armatures and are therefore subject to damage or destruction if overheated.

However, any servomotor with brush commutation can be unsuitable for some applications due to the electromagnetic interference (EMI) caused by brush arcing or the possibility that the arcing can ignite nearby flammable fluids, airborne dust, or vapor, posing a fire or explosion hazard. The EMI generated can adversely affect nearby electronic circuitry. In addition, motor brushes wear down and leave a gritty residue that can contaminate nearby sensitive instruments or precisely ground surfaces. Thus brush-type motors must be cleaned constantly to prevent the spread of the residue from the motor. Also, brushes must be replaced periodically, causing unproductive downtime.

Brushless DC PM motors overcome these problems and offer the benefits of electronic rather than mechanical commutation. Built as inside-out DC motors, typical brushless motors have PM rotors and wound stator coils. Commutation is performed by internal noncontact Hall-effect devices (HEDs) positioned within the stator windings. The HEDs are wired to power transistor switching circuitry, which is mounted externally in separate modules for some motors but is mounted internally on circuit cards in other motors. Alternatively, commutation can be performed by a commutating encoder or by commutation software resident in the motion controller or motor drive.

Brushless DC motors exhibit low rotor inertia and lower winding thermal resistance than brush-type motors because their high-efficiency magnets permit the use of shorter rotors with smaller diameters. Moreover, because they are not burdened with sliding brush-type mechanical contacts, they can run at higher speeds (50,000 rpm or greater), provide higher continuous torque, and accelerate faster than brush-type motors. Nevertheless, brushless motors still cost more than comparably rated brush-type motors (although that price gap continues to narrow) and their installation adds to overall motion control system cost and complexity. Table 1 summarizes some of the outstanding characteristics of stepper, PM brush, and PM brushless DC motors.

The linear motor, another drive alternative, can move the load directly, eliminating the need for intermediate motion translation mechanism. These motors can accelerate rapidly and position loads accurately at high speed because they have no moving parts in contact with each other. Essentially rotary motors that have been sliced open and unrolled, they have many of the characteristics of conventional motors. They can replace conventional rotary motors driving leadscrew-, ballscrew-, or belt-driven single-axis stages, but they cannot be coupled to gears that could change their drive characteristics. If increased performance is required from a linear motor, the existing motor must be replaced with a larger one.

Table 1. Stepping and Permanent-Magnet DC Servomotors Compared.

	Stepping	PM Brush	PM Brushless
Cost	Low	Medium	High
Smoothness	Low to	Good to excellent	Good to excellent
Speed range	0-1500 rpm (typical)	0-6000 rpm	0-10,000 rpm
Torque	High- (falls off with speed)	Medium	High
Required feedback	None	Position or velocity	Commutation and position or velocity
Maintenance	None	Yes	None
Cleanliness	Excellent	Brush dust	Excellent

Linear motors must operate in closed feedback loops, and they typically require more costly feedback sensors than rotary motors. In addition, space must be allowed for the free movement of the motor's power cable as it tracks back and forth along a linear path. Moreover, their applications are also limited because of their inability to dissipate heat as readily as rotary motors with metal frames and cooling fins, and the exposed magnetic fields of some models can attract loose ferrous objects, creating a safety hazard.

Motor Drivers (Amplifiers)

Motor drivers or amplifiers must be capable of driving their associated motors—stepper, brush, brushless, or linear. A drive circuit for a stepper motor can be fairly simple because it needs only several power transistors to sequentially energize the motor phases according to the number of digital step pulses received from the motion controller. However, more advanced stepping motor drivers can control phase current to permit "microstepping," a technique that allows the motor to position the load more precisely.

Servodrive amplifiers for brush and brushless motors typically receive analog voltages of ±10-VDC signals from the motion controller. These signals correspond to current or voltage commands. When amplified, the signals control both the direction and magnitude of the current in the motor windings. Two types of amplifiers are generally used in closed-loop servosystems: *linear* and *pulse-width modulated* (PWM).

Pulse-width modulated amplifiers predominate because they are more efficient than linear amplifiers and can provide up to 100 W. The transistors in PWM amplifiers (as in PWM power supplies) are optimized for switchmode operation, and they are capable of switching amplifier output voltage at frequencies up to 20 kHz. When the power transistors are switched on (on state), they saturate, but when they are off, no current is drawn. This operating mode reduces transistor power dissipation and boosts amplifier efficiency. Because of their higher operating frequencies, the magnetic components in PWM amplifiers can be smaller and lighter than those in linear amplifiers. Thus the entire drive module can be packaged in a smaller, lighter case.

By contrast, the power transistors in linear amplifiers are continuously in the on state although output power requirements can be varied. This operating mode wastes power, resulting in lower amplifier efficiency while subjecting the power transistors to thermal stress. However, linear amplifiers permit smoother motor operation, a requirement for some sensitive motion control systems. In addition linear amplifiers are better at driving low-inductance motors. Moreover, these amplifiers generate less EMI than PWM amplifiers, so they do not require the same degree of filtering. By contrast, linear amplifiers typically have lower maximum power ratings than PWM amplifiers.

Feedback Sensors

Position feedback is the most common requirement in closed-loop motion control systems, and the most popular sensor for providing this information is the rotary optical encoder. The axial shafts of these encoders are mechanically coupled to the drive shafts of the motor. They generate either sine waves or pulses that can be counted by the motion controller to determine the motor or load position and direction of travel at any time to permit precise positioning. Analog encoders produce sine waves that must be conditioned by external circuitry for counting, but digital encoders include circuitry for translating sine waves into pulses.

Absolute rotary optical encoders produce binary words for the motion controller that provide precise position information. If they are stopped accidentally due to power failure, these encoders preserve the binary word because the last position of the encoder code wheel acts as a memory.

Linear optical encoders, by contrast, produce pulses that are proportional to the actual linear distance of load movement. They work on the same principles as the rotary encoders, but the graduations are engraved on a stationary glass or metal scale while the read head moves along the scale.

Tachometers are generators that provide analog signals that are directly proportional to motor shaft speed. They are mechanically coupled to the motor shaft and can be located within the motor frame. After tachometer output is converted to a digital format by the motion controller, a feedback signal is generated for the driver to keep motor speed within preset limits.

Other common feedback sensors include resolvers, linear variable differential transformers (LVDTs), Inductosyns, and potentiometers. Less common are the more accurate laser interferometers. Feedback sensor selection is based on an evaluation of the sensor's accuracy, repeatability, ruggedness, temperature limits, size, weight, mounting requirements, and cost, with the relative importance of each determined by the application.

Installation and Operation of the System

The design and implementation of a cost-effective motion-control system require a high degree of expertise on the part of the person or persons responsible for system integration. It is rare that a diverse group of components can be removed from their boxes, installed, and interconnected to form an instantly effective system. Each servosystem (and many stepper systems) must be tuned (stabilized) to the load and environmental conditions. However, installation and development time can be minimized if the customer's requirements are accurately defined, optimum components are selected, and the tuning and debugging tools are applied correctly. Moreover, operators must be properly trained in formal classes or, at the very least, must have a clear understanding of the information in the manufacturers' technical manuals gained by careful reading.

GLOSSARY OF MOTION CONTROL TERMS

Abbe error: A linear error caused by a combination of an underlying angular error along the line of motion and a dimensional offset between the position of the object being measured and the accuracy-determining element such as a leadscrew or encoder.

acceleration: The change in velocity per unit time.

accuracy: (1) absolute accuracy: The motion control system output compared with the commanded input. It is actually a measurement of inaccuracy and it is typically measured in millimeters. (2) motion accuracy: The maximum expected difference between the actual and the intended position of an object or load for a given input. Its value depends on the method used for measuring the actual position. (3) on-axis accuracy: The uncertainty of load position after all linear errors are eliminated. These include such factors as inaccuracy of leadscrew pitch, the angular deviation effect at the measuring point, and thermal expansion of materials.

backlash: The maximum magnitude of an input that produces no measurable output when the direction of motion is reversed. It can result from insufficient preloading or poor meshing of gear teeth in a gear-coupled drive train.

error: (1) The difference between the actual result of an input command and the ideal or theoretical result. (2) following error: The instantaneous difference between the actual position as reported by the position feedback loop and the ideal position, as commanded by the controller. (3) steady-state error: The difference between the actual and commanded position after all corrections have been applied by the controller

hysteresis: The difference in the absolute position of the load for a commanded input when motion is from opposite directions.

inertia: The measure of a load's resistance to changes in velocity or speed. It is a function of the load's mass and shape. The torque required to accelerate or decelerate the load is proportional to inertia.

overshoot: The amount of overcorrection in an underdamped control system.

play: The uncontrolled movement due to the looseness of mechanical parts. It is typically caused by wear, overloading the system, or improper system operation.

precision: See *repeatability.*

repeatability: The ability of a motion control system to return repeatedly to the commanded position. It is influenced by the presence of *backlash* and *hysteresis*. Consequently, *bidirectional repeatability*, a more precise specification, is the ability of the system to achieve the commanded position repeatedly regardless of the direction from which the intended position is approached. It is synonymous with *precision.* However, accuracy and precision are not the same.

resolution: The smallest position increment that the motion control system can detect. It is typically considered to be display or encoder resolution because it is not necessarily the smallest motion the system is capable of delivering reliably.

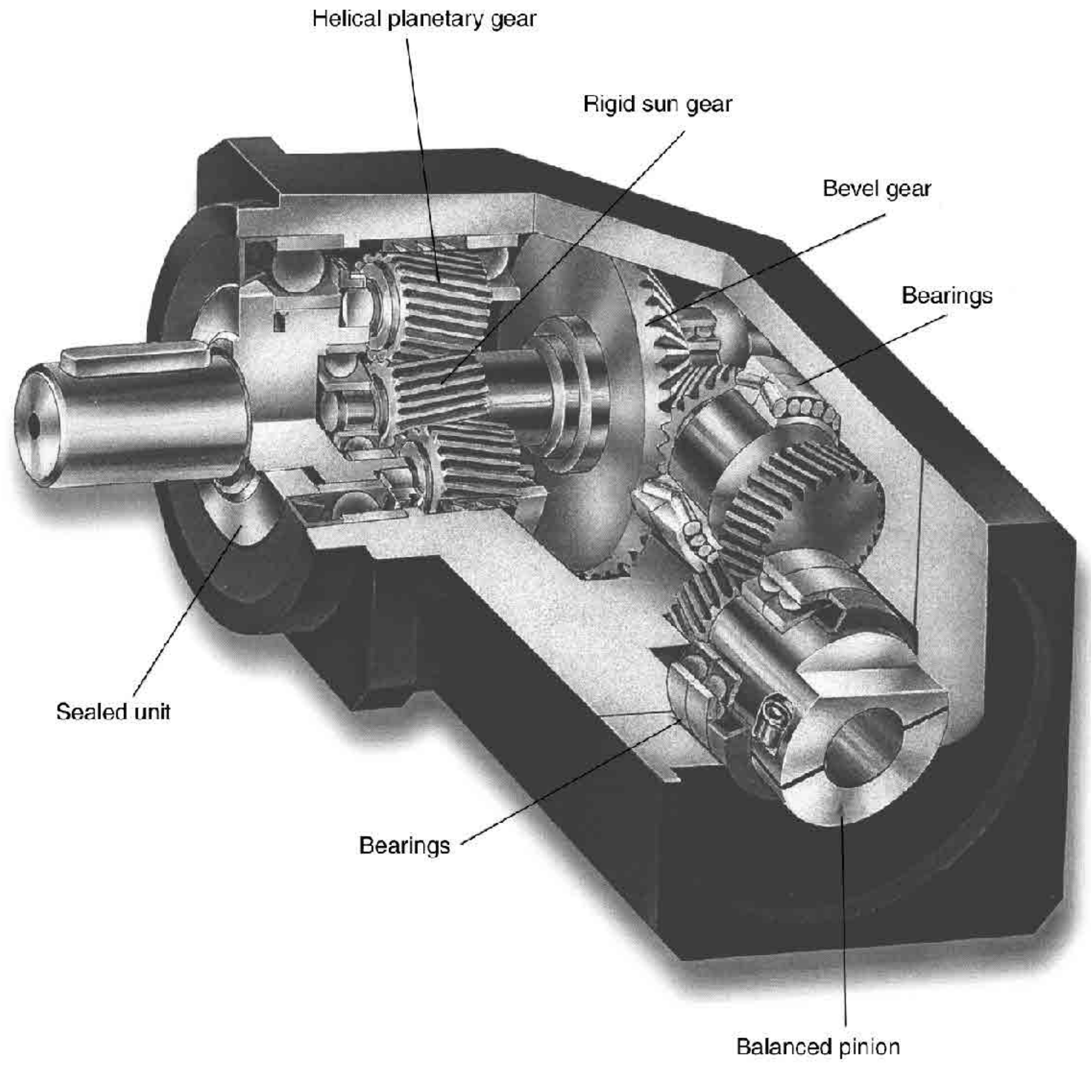
runout: The deviation between ideal linear (straight-line) motion and the actual measured motion.

sensitivity: The minimum input capable of producing output motion. It is also the ratio of the output motion to the input drive. This term should not be used in place of resolution.

settling time: The time elapsed between the entry of a command to a system and the instant the system first reaches the commanded position and maintains that position within the specified error value.

velocity: The change in distance per unit time. Velocity is a *vector* and speed is a *scalar*, but the terms can be used interchangeably.

HIGH-SPEED GEARHEADS IMPROVE SMALL SERVO PERFORMANCE



This right-angle gearhead is designed for high-performance servo applications. It includes helical planetary output gears, a rigid sun gear, spiral bevel gears, and a balanced input pinion. *Courtesy of Bayside Controls Inc.*

The factory-made precision gearheads now available for installation in the latest smaller-sized servosystems can improve their performance while eliminating the external gears, belts, and pulleys commonly used in earlier larger servosystems. The gearheads can be coupled to the smaller, higher-speed servomotors, resulting in simpler systems with lower power consumption and operating costs.

Gearheads, now being made in both in-line and right-angle configurations, can be mounted directly to the drive motor shafts. They can convert high-speed, low-torque rotary motion to a low-speed, high-torque output. The latest models are smaller and more accurate than their predecessors, and they have been designed to be compatible with the smaller, more precise servomotors being offered today.

Gearheads have often been selected for driving long trains of mechanisms in machines that perform such tasks as feeding wire, wood, or metal for further processing. However, the use of an in-line gearhead adds to the space occupied by these machines, and this can be a problem where factory floor space is restricted. One way to avoid this problem is to choose a right-angle gearhead. It can be mounted vertically beneath the host machine or even horizontally on the machine bed. Horizontal mounting can save space because the gearheads and motors can be positioned behind the machine, away from the operator.

Bevel gears are commonly used in right-angle drives because they can provide precise motion. Conically shaped bevel gears with straight- or spiral-cut teeth allow mating shafts to intersect at 90º angles. Straight-cut bevel gears typically have contact ratios of about 1.4, but the simultaneous mating of straight teeth along their entire lengths causes more vibration and noise than the mating of spiral-bevel gear teeth. By contrast, spiral-bevel gear teeth engage and disengage gradually and precisely with contact ratios of 2.0 to 3.0, making little noise. The higher contact ratios of spiral-bevel gears permit them to drive loads that are 20 to 30% greater than those possible with straight bevel gears. Moreover, the spiral-bevel teeth mesh with a rolling action that increases their precision and also reduces friction. As a result, operating efficiencies can exceed 90%.

Simplify the Mounting

The smaller servomotors now available force gearheads to operate at higher speeds, making vibrations more likely. Inadvertent misalignment between servomotors and gearboxes, which often occurs during installation, is a common source of vibration. The mounting of conventional motors with gearboxes requires several precise connections. The output shaft of the motor must be attached to the pinion gear that slips into a set of planetary gears in the end of the gearbox, and an adapter plate must joint the motor to the gearbox. Unfortunately, each of these connections can introduce slight alignment errors that accumulate to cause overall motor/gearbox misalignment.

The pinion is the key to smooth operation because it must be aligned exactly with the motor shaft and gearbox. Until recently it has been standard practice to mount pinions in the field when the motors were connected to the gearboxes. This procedure often caused the assembly to vibrate. Engineers realized that the integration of gearheads into the servomotor package would solve this problem, but the drawback to the integrated unit is that failure of either component would require replacement of the whole unit.

A more practical solution is to make the pinion part of the gearhead assembly because gearheads with built-in pinions are easier to mount to servomotors than gearheads with field-installed pinions. It is only necessary to insert the motor shaft into the collar that extends from the gearhead's rear housing, tighten the clamp with a wrench, and bolt the motor to the gearhead.

Pinions installed at the factory ensure smooth-running gearheads because they are balanced before they are mounted. This procedure permits them to spin at high speed without wobbling. As a result, the balanced pinions minimize friction and thus cause less wear, noise, and vibration than field-installed pinions.

However, the factory-installed pinion requires a floating bearing to support the shaft with a pinion on one end. The Bayside Motion Group of Bayside Controls Inc., Port Washington, New York, developed a self-aligning bearing for this purpose. Bayside gearheads with these pinions are rated for input speeds up to 5000 rpm. A collar on the pinion shaft's other end mounts to the motor shaft. The bearing holds the pinion in place until it is mounted. At that time a pair of bearings in the servomotor support the coupled shaft. The self-aligning feature of the floating bearing lets the motor bearing support the shaft after installation.

The pinion and floating bearing help to seal the unit during its operation. The pinion rests in a blind hole and seals the rear of the gearhead. This seal keeps out dirt while retaining the lubricants within the housing. Consequently, light grease and semifluid lubricants can replace heavy grease.

Cost-Effective Addition

The installation of gearheads can smooth the operation of servosystems as well as reduce system costs. The addition of a gearhead to the system does not necessarily add to overall operating costs because its purchase price can be offset by reductions in operating costs. Smaller servomotors inherently draw less current than larger ones, thus reducing operating costs, but those power savings are greatest in applications calling for low speed and high torque because direct-drive servomotors must be considerably larger than servomotors coupled to gearheads to perform the same work.

Small direct-drive servomotors assigned to high-speed/low-torque applications might be able to perform the work satisfactorily without a gearhead. In those instances servo/gearhead combinations might not be as cost-effective because power consumption will be comparable. Nevertheless, gearheads will still improve efficiency and, over time, even small decreases in power consumption due to the use of smaller-sized servos will result in reduced operating costs.

The decision to purchase a precision gearhead should be evaluated on a case-by-case basis. The first step is to determine speed and torque requirements. Then keep in mind that although in high-speed/low-torque applications a direct-drive system might be satisfactory, low-speed/high-torque applications almost always require gearheads. Then a decision can be made after weighing the purchase price of the gearhead against anticipated servosystem operating expenses in either operating mode to estimate savings.

MODULAR SINGLE-AXIS MOTION SYSTEMS

Modular single-axis motion systems are motion control modules capable of translating rotary motion, typically from servomotors or stepper motors, into linear motion. Two different kinds of single-axis modules are illustrated here: ballscrew-driven and belt-driven.

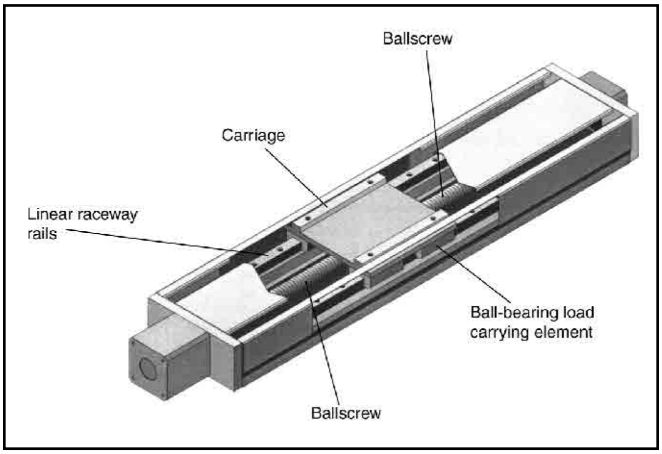


Fig. 1 This commercial ballscrew-actuated system offers position accuracy of 0.025 mm per 300 mm with repeatability of ±0.005 mm. Its carriage can move at speeds up to 1 m/s. It has T-slots in its base mounting system and is designed to be continuously supported. *Courtesy of Thomson Industries, Inc.*

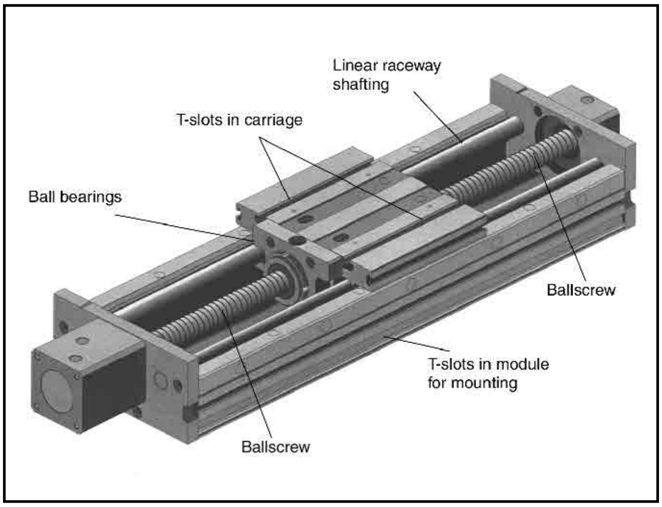


Fig. 2 This commercial ballscrew-driven system also offers position accuracy of 0.025 mm per 300 mm with repeatability of ±0.005 mm. It has T-slots in both its carriage and base mounting system, and is also designed to be continuously supported. *Courtesy of Thomson Industries, Inc.*

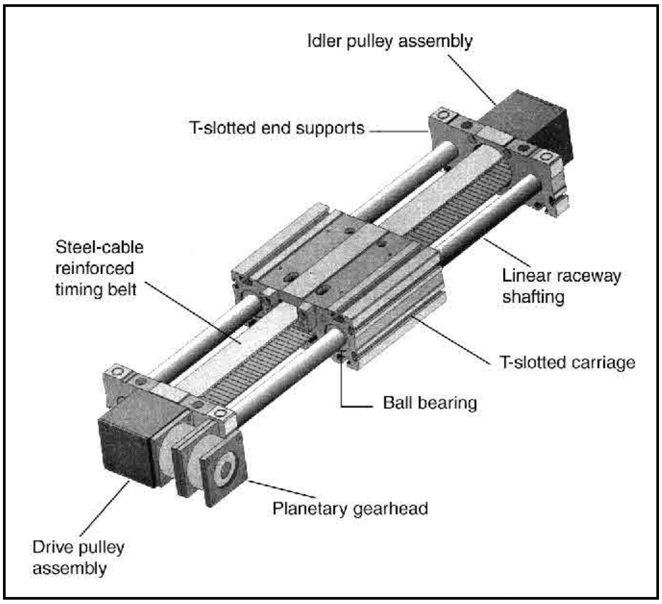


Fig. 3 This modular single-axis belt-driven system is built to bridge a gap between its supporting surfaces. Position accuracy is better than ±0.15 mm and speed can reach 5 m/s, both higher than for a ballscrew-driven system. A precision gearhead matches the inertia between system payload, and the servomotor provides thrust to 1400 N-m at speeds up to 4000 rpm. *Courtesy of Thomson Industries, Inc.*

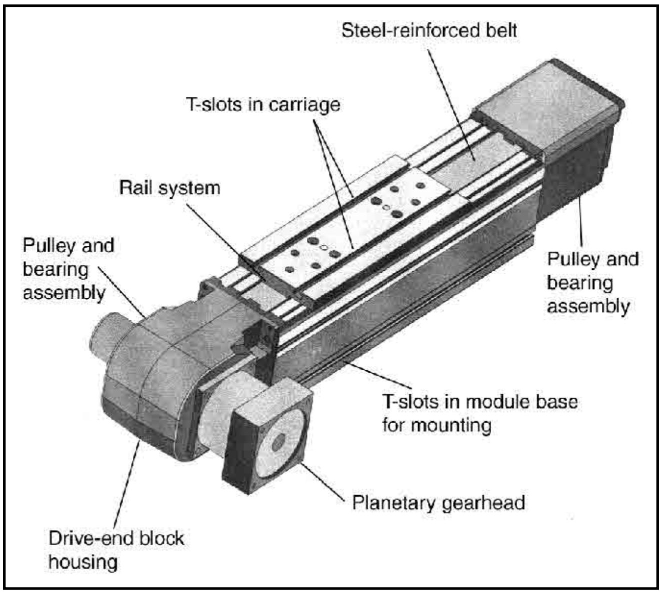


Fig. 4 This modular single-axis belt-driven system is built more ruggedly for applications where a rigid, continuously supported module is required. With a planetary gearhead its mechanical characteristics match those of the module in Fig. 3. *Courtesy of Thomson Industries, Inc.*

MECHANICAL COMPONENTS FORM SPECIALIZED MOTION-CONTROL SYSTEMS

Many different kinds of mechanical components are listed in manufacturers' catalogs for speeding the design and assembly of motion control systems. These drawings illustrate what, where, and how one manufacturer's components were used to build specialized systems.

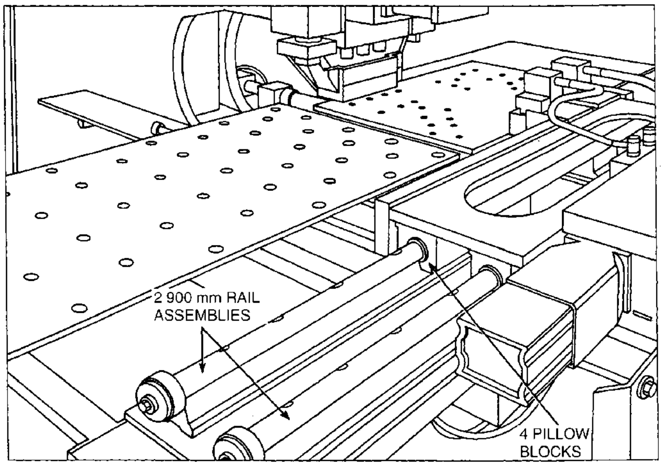


Fig. 1 Punch Press: Catalog pillow blocks and rail assemblies were installed in this system for reducing the deflection of a punch press plate loader to minimize scrap and improve its cycle speed. *Courtesy of Thomson Industries, Inc.*

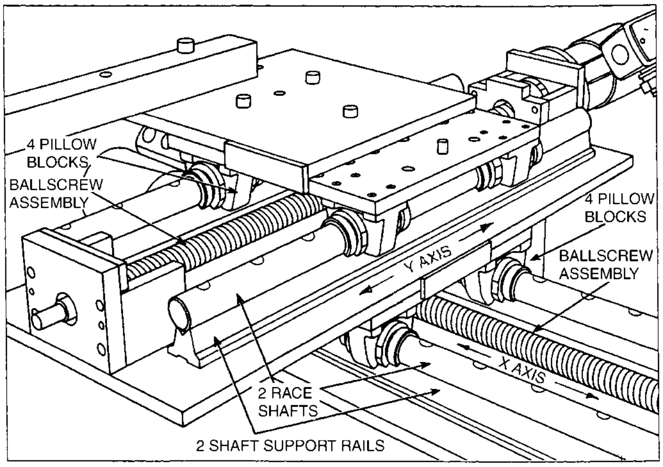


Fig. 2 Microcomputer-Controlled X-Y Table: Catalog pillow blocks, rail guides, and ballscrew assemblies were installed in this rigid system that positions workpieces accurately for precise milling and drilling on a vertical milling machine. *Courtesy of Thomson Industries, Inc.*

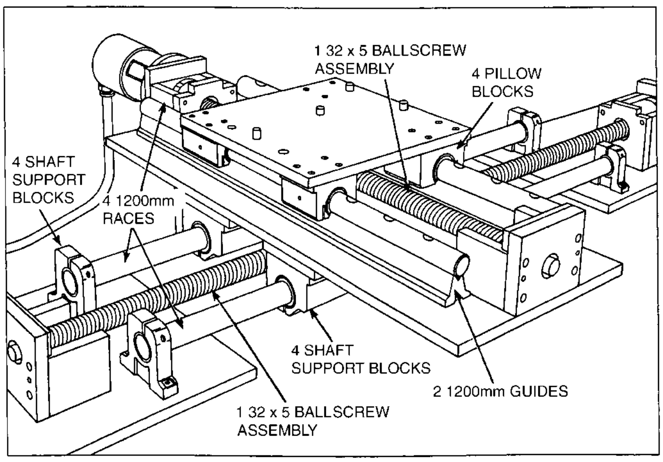


Fig. 3 Pick and Place X-Y System: Catalog support and pillow blocks, ballscrew assemblies, races, and guides were in the assembly of this X-Y system that transfers workpieces between two separate machining stations. *Courtesy of Thomson Industries, Inc.*

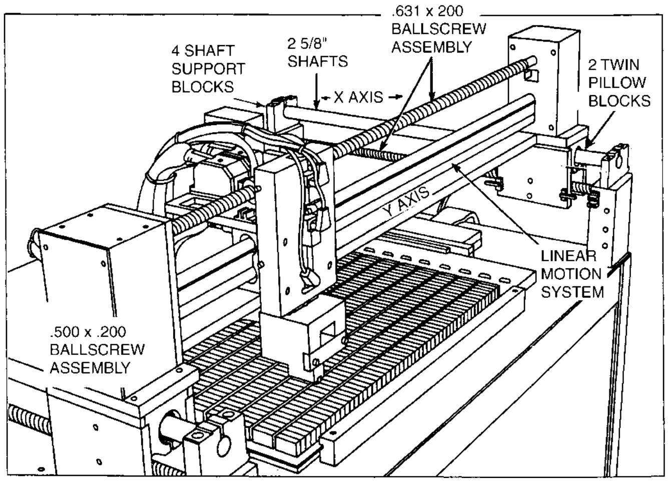


Fig. 4 X-Y Inspection System: Catalog pillow and shaft-support blocks, ballscrew assemblies, and a preassembled motion system were used to build this system, which accurately positions an inspection probe over small electronic components. *Courtesy of Thomson Industries, Inc.*

SERVOMOTORS, STEPPER MOTORS, AND ACTUATORS FOR MOTION CONTROL

Many different kinds of electric motors have been adapted for use in motion control systems because of their linear characteristics. These include both conventional rotary and linear alternating current (AC) and direct current (DC) motors. These motors can be further classified into those that must be operated in closed-loop servosystems and those that can be operated open-loop.

The most popular servomotors are permanent magnet (PM) rotary DC servomotors that have been adapted from conventional PM DC motors. These servomotors are typically classified as brush-type and brushless. The brush-type PM DC servomotors include those with wound rotors and those with lighter weight, lower inertia cup- and disk coil-type armatures. Brushless servomotors have PM rotors and wound stators.

Some motion control systems are driven by two-part linear servomotors that move along tracks or ways. They are popular in applications where errors introduced by mechanical coupling between the rotary motors and the load can introduce unwanted errors in positioning. Linear motors require closed loops for their operation, and provision must be made to accommodate the back-and-forth movement of the attached data and power cable.

Stepper or stepping motors are generally used in less demanding motion control systems, where positioning the load by stepper motors is not critical for the application. Increased position accuracy can be obtained by enclosing the motors in control loops.

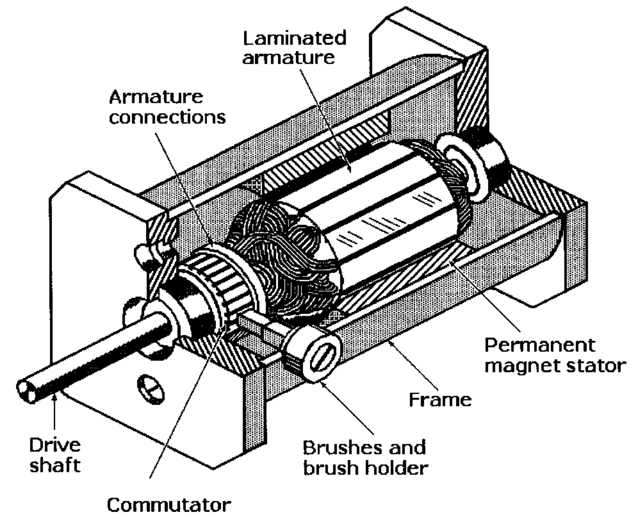


Fig. 1 Cutaway view of a fractional horsepower permanent-magnet DC servomotor.

Permanent-Magnet DC Servomotors

Permanent-magnet (PM) field DC rotary motors have proven to be reliable drives for motion control applications where high efficiency, high starting torque, and linear speed–torque curves are desirable characteristics. While they share many of the characteristics of conventional rotary series, shunt, and compound-wound brush-type DC motors, PM DC servomotors increased in popularity with the introduction of stronger ceramic and rare-earth magnets made from such materials as neodymium–iron–boron and the fact that these motors can be driven easily by microprocessor-based controllers.

The replacement of a wound field with permanent magnets eliminates both the need for separate field excitation and the electrical losses that occur in those field windings. Because there are both brush-type and brushless DC servomotors, the term *DC motor* implies that it is brush-type or requires mechanical commutation unless it is modified by the term *brushless.* Permanent-magnet DC brush-type servomotors can also have armatures formed as laminated coils in disk or cup shapes. They are lightweight, low-inertia armatures that permit the motors to accelerate faster than the heavier conventional wound armatures.

The increased field strength of the ceramic and rare-earth magnets permitted the construction of DC motors that are both smaller and lighter than earlier generation comparably rated DC motors with alnico (aluminum–nickel–cobalt or AlNiCo) magnets. Moreover, integrated circuitry and microprocessors have increased the reliability and cost-effectiveness of digital motion controllers and motor drivers or amplifiers while permitting them to be packaged in smaller and lighter cases, thus reducing the size and weight of complete, integrated motion-control systems.

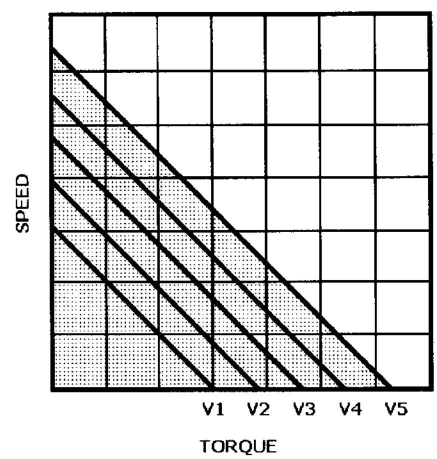


Fig. 2 A typical family of speed/torque curves for a permanent-magnet DC servomotor at different voltage inputs, with voltage increasing from left to right (V1 to V5).

Brush-Type PM DC Servomotors

The design feature that distinguishes the brush-type PM DC servomotor, as shown in Fig. 1, from other brush-type DC motors is the use of a permanent-magnet field to replace the wound field. As previously stated, this eliminates both the need for separate field excitation and the electrical losses that typically occur in field windings.

Permanent-magnet DC motors, like all other mechanically commutated DC motors, are energized through brushes and a multisegment commutator. While all DC motors operate on the same principles, only PM DC motors have the linear speed–torque curves shown in Fig. 2, making them ideal for closed-loop and variable-speed servomotor applications. These linear characteristics conveniently describe the full range of motor performance. It can be seen that both speed and torque increase linearly with applied voltage, indicated in the diagram as increasing from V1 to V5.

The stators of brush-type PM DC motors are magnetic pole pairs. When the motor is powered, the opposite polarities of the energized windings and the stator magnets attract, and the rotor rotates to align itself with the stator. Just as the rotor reaches alignment, the brushes move across the commutator segments and energize the next winding. This sequence continues as long as power is applied, keeping the rotor in continuous motion. The commutator is staggered from the rotor poles, and the number of its segments is directly proportional to the number of windings. If the connections of a PM DC motor are reversed, the motor will change direction, but it might not operate as efficiently in the reversed direction.

Disk-Type PM DC Motors

The disk-type motor shown exploded view in Fig. 3 has a disk-shaped armature with stamped and laminated windings. This nonferrous laminated disk is made as a copper stamping bonded between epoxy–glass insulated layers and fastened to an axial shaft. The stator field can either be a ring of many individual ceramic magnet cylinders, as shown, or a ring-type ceramic magnet attached to the dish-shaped end bell, which completes the magnetic circuit. The spring-loaded brushes ride directly on stamped commutator bars.

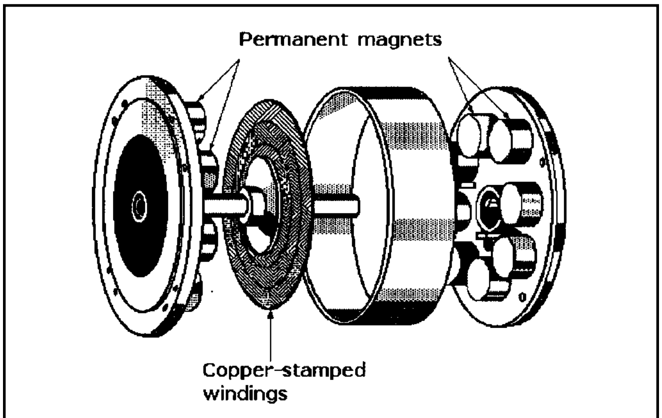


Fig. 3 Exploded view of a permanent-magnet DC servomotor with a disk-type armature.

These motors are also called *pancake motors* because they are housed in cases with thin, flat form factors whose diameters exceed their lengths, suggesting pancakes. Earlier generations of these motors were called *printed-circuit motors* because the armature disks were made by a printed-circuit fabrication process that has been superseded. The flat motor case concentrates the motor's center of mass close to the mounting plate, permitting it to be easily surface mounted. This eliminates the awkward motor overhang and the need for supporting braces if a conventional motor frame is to be surface mounted. Their disk-type motor form factor has made these motors popular as axis drivers for industrial robots where space is limited.

The principal disadvantage of the disk-type motor is the relatively fragile construction of its armature and its inability to dissipate heat as rapidly as iron-core wound rotors. Consequently, these motors are usually limited to applications where the motor can be run under controlled conditions and a shorter duty cycle allows enough time for armature heat buildup to be dissipated.

Cup- or Shell-Type PM DC Motors

Cup- or shell-type PM DC motors offer low inertia and low inductance as well as high acceleration characteristics, making

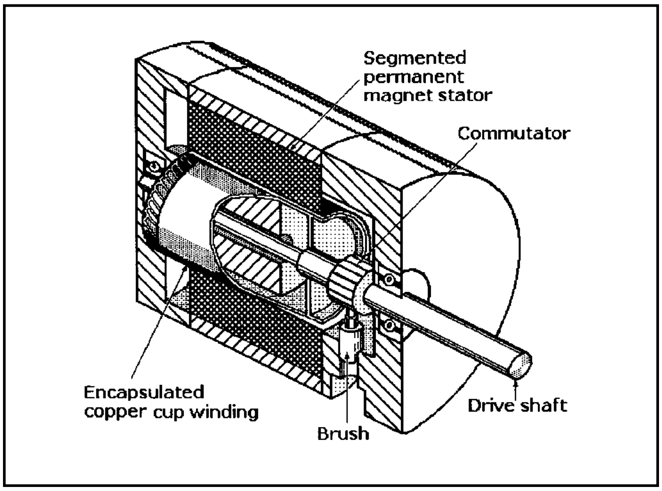


Fig. 4 Cutaway view of a permanent-magnet DC servomotor with a cup-type armature.

them useful in many servo applications. They have hollow cylindrical armatures made as aluminum or copper coils bonded by polymer resin and fiberglass to form a rigid "ironless cup," which is fastened to an axial shaft. A cutaway view of this class of servomotor is illustrated in Fig. 4.

Because the armature has no iron core, it, like the disk motor, has extremely low inertia and a very high torque-to-inertia ratio. This permits the motor to accelerate rapidly for the quick response required in many motion-control applications. The armature rotates in an air gap within very high magnetic flux density. The magnetic field from the stationary magnets is completed through the cup-type armature and a stationary ferrous cylindrical core connected to the motor frame. The shaft rotates within the core, which extends into the rotating cup. Spring-brushes commutate these motors.

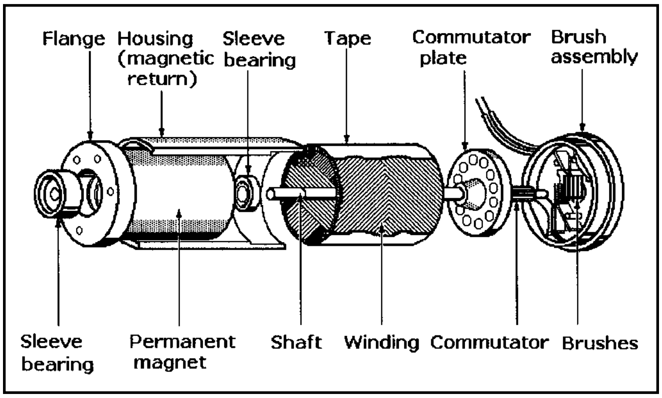


Fig. 5 Exploded view of a fractional horsepower brush-type DC servomotor.

Another version of a cup-type PM DC motor is shown in the exploded view in Fig. 5. The cup type armature is rigidly fastened to the shaft by a disk at the right end of the winding, and the magnetic field is also returned through a ferrous metal housing. The brush assembly of this motor is built into its end cap or flange, shown at the far right.

The principal disadvantage of this motor is also the inability of its bonded armature to dissipate internal heat buildup rapidly because of its low thermal conductivity. Without proper cooling and sensitive control circuitry, the armature could be heated to destructive temperatures in seconds.

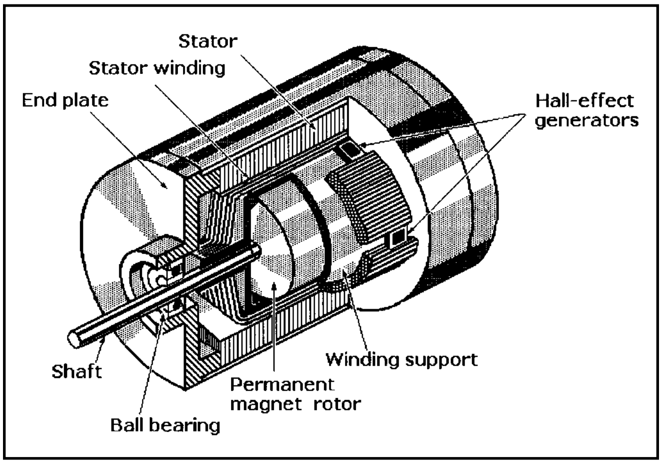


Fig. 6 Cutaway view of a brushless DC motor.

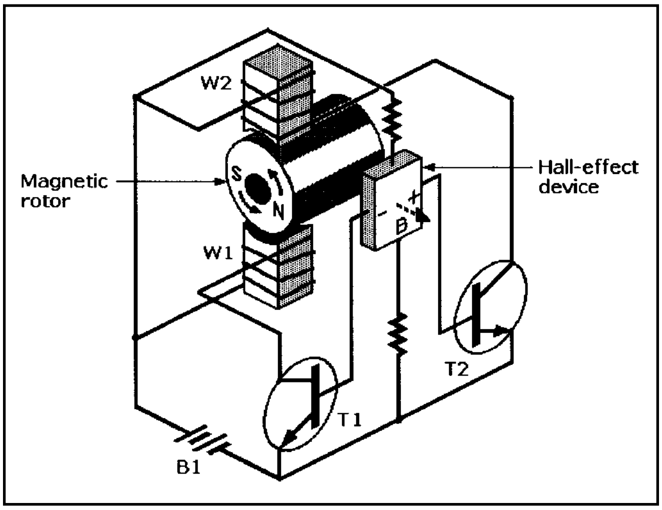


Fig. 7 Simplified diagram of Hall-effect device (HED) commutation of a brushless DC motor.

Brushless PM DC Motors

Brushless DC motors exhibit the same linear speed–torque characteristics as the brush-type PM DC motors, but they are electronically commutated. The construction of these motors, as shown in Fig. 6, differs from that of a typical brush-type DC motor in that they are "inside-out." In other words, they have permanent magnet rotors instead of stators, and the stators rather than the rotors are wound. Although this geometry is required for brushless DC motors, some manufacturers have adapted this design for brush-type DC motors.

The mechanical brush and bar commutator of the brushless DC motor is replaced by electronic sensors, typically Hall-effect devices (HEDs). They are located within the stator windings and wired to solid-state transistor switching circuitry located either on circuit cards mounted within the motor housings or in external packages. Generally, only fractional horsepower brushless motors have switching circuitry within their housings.

The cylindrical magnet rotors of brushless DC motors are magnetized laterally to form opposing north and south poles across the rotor's diameter. These rotors are typically made from neodymium–iron–boron or samarium–cobalt rare-earth magnetic materials, which offer higher flux densities than alnico magnets. These materials permit motors offering higher performance to be packaged in the same frame sizes as earlier motor designs or those with the same ratings to be packaged in smaller frames than the earlier designs. Moreover, rare-earth or ceramic magnet rotors can be made with smaller diameters than those earlier models with alnico magnets, thus reducing their inertia.

A simplified diagram of a DC brushless motor control with one Hall-effect device (HED) for the electronic commutator is shown in Fig. 7. The HED is a Hall-effect sensor integrated with an amplifier in a silicon chip. This IC is capable of sensing the polarity of the rotor's magnetic field and then sending appropriate signals to power transistors T1 and T2 to cause the motor's rotor to rotate continuously. This is accomplished as follows:

(1) With the rotor motionless, the HED detects the rotor's north magnetic pole, causing it to generate a signal that turns on transistor T2. This causes current to flow, energizing winding W2 to form a south-seeking electromagnetic rotor pole. This pole then attracts the rotor's north pole to drive the rotor in a counterclockwise (CCW) direction.

(2) The inertia of the rotor causes it to rotate past its neutral position so that the HED can then sense the rotor's south magnetic pole. It then switches on transistor T1, causing current to flow in winding W1, thus forming a north-seeking stator pole that attracts the rotor's south pole, causing it to continue to rotate in the CCW direction.

The transistors conduct in the proper sequence to ensure that the excitation in the stator windings W2 and W1 always leads the PM rotor field to produce the torque necessary keep the rotor in constant rotation. The windings are energized in a pattern that rotates around the stator.

There are usually two or three HEDs in practical brushless motors that are spaced apart by 90 or 120º around the motor's rotor. They send the signals to the motion controller that actually triggers the power transistors, which drive the armature windings at a specified motor current and voltage level.

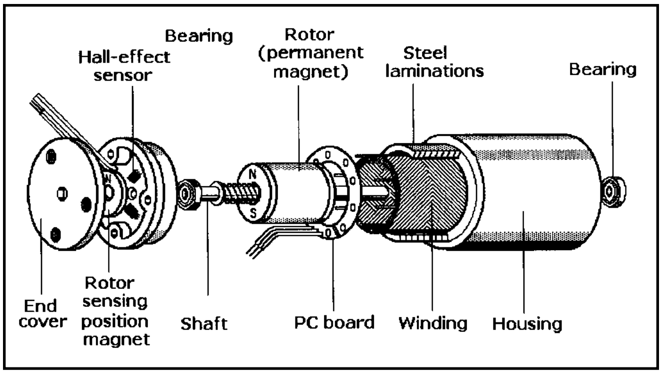


Fig. 8 Exploded view of a brushless DC motor with Hall-effect device (HED) commutation.

The brushless motor in the exploded view Fig. 8 illustrates a design for a miniature brushless DC motor that includes Hall-effect commutation. The stator is formed as an ironless sleeve of copper coils bonded together in polymer resin and fiberglass to form a rigid structure similar to cup-type rotors. However, it is fastened inside the steel laminations within the motor housing.

This method of construction permits a range of values for starting current and specific speed (rpm/V) depending on wire gauge and the number of turns. Various terminal resistances can be obtained, permitting the user to select the optimum motor for a specific application. The Hall-effect sensors and a small magnet disk that is magnetized widthwise are mounted on a disk-shaped partition within the motor housing.

Position Sensing in Brushless Motors

Both magnetic sensors and resolvers can sense rotor position in brushless motors. The diagram in Fig. 9 shows how three mag-

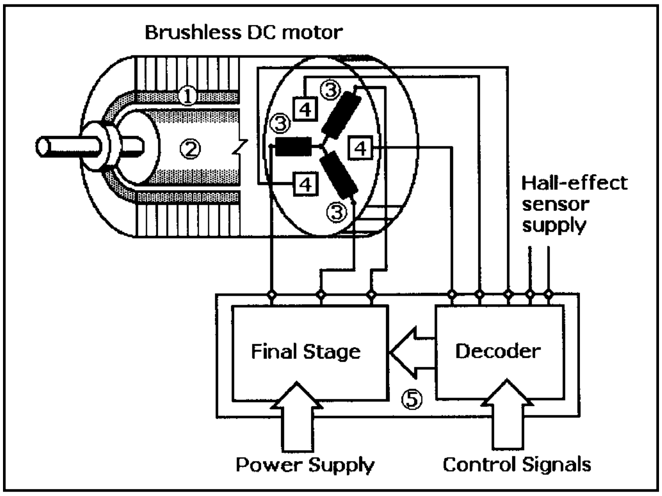


Fig. 9 A magnetic sensor as a rotor position indicator: stationary brushless motor winding (1), permanent-magnet motor rotor (2), three-phase electronically commutated field (3), three magnetic sensors (4), and the electronic circuit board (5).

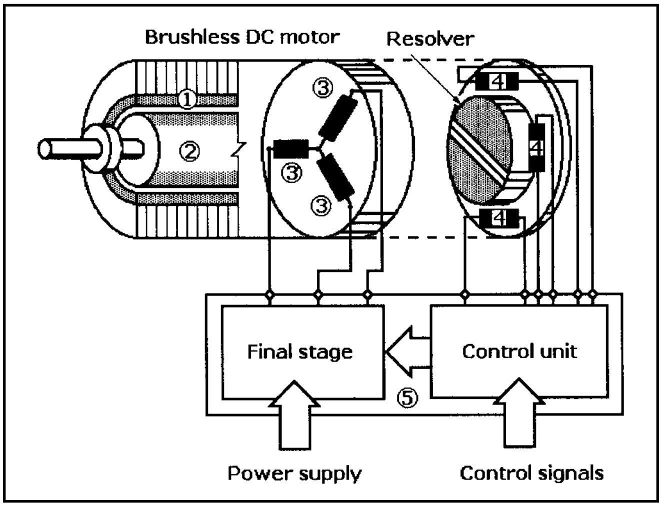


Fig. 10 A resolver as a rotor position indicator: stationary motor winding (1), permanent-magnet motor rotor (2), three-phase electronically commutated field (3), three magnetic sensors (4), and the electronic circuit board (5).

netic sensors can sense rotor position in a three-phase electronically commutated brushless DC motor. In this example the magnetic sensors are located inside the end-bell of the motor. This inexpensive version is adequate for simple controls.

In the alternate design shown in Fig. 10, a resolver on the end cap of the motor is used to sense rotor position when greater positioning accuracy is required. The high-resolution signals from the resolver can be used to generate sinusoidal motor currents within the motor controller. The currents through the three motor windings are position independent and respectively 120º phase shifted.

Brushless Motor Advantages

Brushless DC motors have at least four distinct advantages over brush-type DC motors that are attributable to the replacement of mechanical commutation by electronic commutation.

- There is no need to replace brushes or remove the gritty residue caused by brush wear from the motor.
- Without brushes to cause electrical arcing, brushless motors do not present fire or explosion hazards in an environment where flammable or explosive vapors, dust, or liquids are present.
- Electromagnetic interference (EMI) is minimized by replacing mechanical commutation, the source of unwanted radio frequencies, with electronic commutation.
- Brushless motors can run faster and more efficiently with electronic commutation. Speeds of up to 50,000 rpm can be achieved vs. the upper limit of about 5000 rpm for brush-type DC motors.

Brushless DC Motor Disadvantages

There are at least four disadvantages of brushless DC servomotors.

- Brushless PM DC servomotors cannot be reversed by simply reversing the polarity of the power source. The order in which the current is fed to the field coil must be reversed.
- Brushless DC servomotors cost more than comparably rated brush-type DC servomotors.
- Additional system wiring is required to power the electronic commutation circuitry.
- The motion controller and driver electronics needed to operate a brushless DC servomotor are more complex and expensive than those required for a conventional DC servomotor.

Consequently, the selection of a brushless motor is generally justified on a basis of specific application requirements or its hazardous operating environment.

Characteristics of Brushless Rotary Servomotors

It is difficult to generalize about the characteristics of DC rotary servomotors because of the wide range of products available commercially. However, they typically offer continuous torque ratings of 0.62 lb-ft (0.84 N-m) to 5.0 lb-ft (6.8 N-m), peak torque ratings of 1.9 lb-ft (2.6 N-m) to 14 lb-ft (19 N-m), and continuous power ratings of 0.73 hp (0.54 kW) to 2.76 hp (2.06 kW). Maximum speeds can vary from 1400 to 7500 rpm, and the weight of these motors can be from 5.0 lb (2.3 kg) to 23 lb (10 kg). Feedback typically can be either by resolver or encoder.

Linear Servomotors

A linear motor is essentially a rotary motor that has been opened out into a flat plane, but it operates on the same principles. A permanent-magnet DC linear motor is similar to a permanent-magnet rotary motor, and an AC induction squirrel cage motor is similar to an induction linear motor. The same electromagnetic force that produces torque in a rotary motor also produces torque in a linear motor. Linear motors use the same controls and programmable position controllers as rotary motors.

Before the invention of linear motors, the only way to produce linear motion was to use pneumatic or hydraulic cylinders, or to translate rotary motion to linear motion with ballscrews or belts and pulleys.

A linear motor consists of two mechanical assemblies: *coil* and *magnet*, as shown in Fig. 11. Current flowing in a winding in a magnetic flux field produces a force. The copper windings conduct current (I), and the assembly generates magnetic flux density (B). When the current and flux density interact, a force (F) is generated in the direction shown in Fig. 11, where $F = I \times B$.

Even a small motor will run efficiently, and large forces can be created if a large number of turns are wound in the coil and the magnets are powerful rare-earth magnets. The windings are

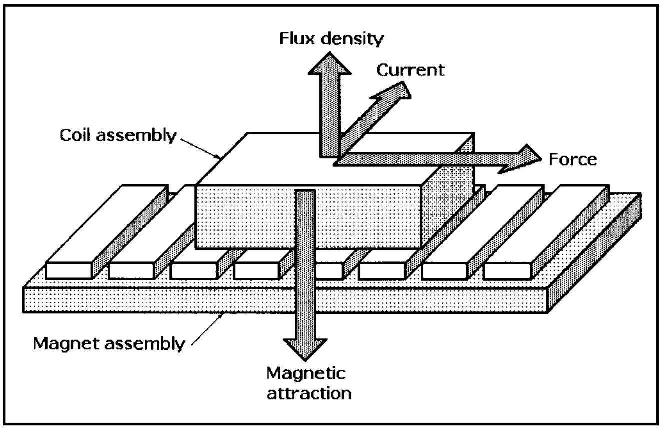


Fig. 11 Operating principles of a linear servomotor.

phased 120 electrical degrees apart, and they must be continually switched or commutated to sustain motion.

Only brushless linear motors for closed-loop servomotor applications are discussed here. Two types of these motors are available commercially—*steel-core* (also called *iron-core*) and *epoxy-core* (also called *ironless*). Each of these linear servomotors has characteristics and features that are optimal in different applications

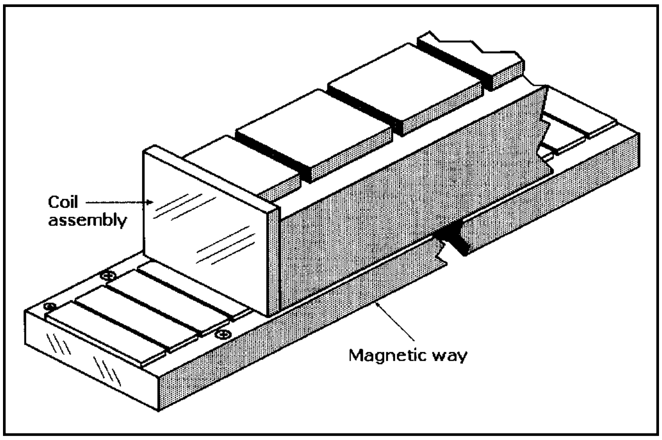


Fig. 12 A linear iron-core linear servomotor consists of a magnetic way and a mating coil assembly.

The coils of steel-core motors are wound on silicon steel to maximize the generated force available with a single-sided magnet assembly or way. Figure 12 shows a steel-core brushless linear motor. The steel in these motors focuses the magnetic flux to produce very high force density. The magnet assembly consists of rare-earth bar magnets mounted on the upper surface of a steel base plate arranged to have alternating polarities (i.e., N, S, N, S)

The steel in the cores is attracted to the permanent magnets in a direction that is perpendicular (normal) to the operating motor force. The magnetic flux density within the air gap of linear motors is typically several thousand gauss. A constant magnetic force is present whether or not the motor is energized. The normal force of the magnetic attraction can be up to ten times the continuous force rating of the motor. This flux rapidly diminishes to a few gauss as the measuring point is moved a few centimeters away from the magnets.

Cogging is a form of magnetic "detenting" that occurs in both linear and rotary motors when the motor coil's steel laminations cross the alternating poles of the motor's magnets. Because it can occur in steel-core motors, manufacturers include features that minimize cogging. The high thrust forces attainable with steel-core linear motors permit them to accelerate and move heavy masses while maintaining stiffness during machining or process operations.

The features of epoxy-core or ironless-core motors differ from those of the steel-core motors. For example, their coil assemblies are wound and encapsulated within epoxy to form a thin plate that is inserted in the air gap between the two permanent-magnet strips fastened inside the magnet assembly, as shown in Fig. 13. Because the coil assemblies do not contain steel cores, epoxy-core motors are lighter than steel-core motors and less subject to cogging.

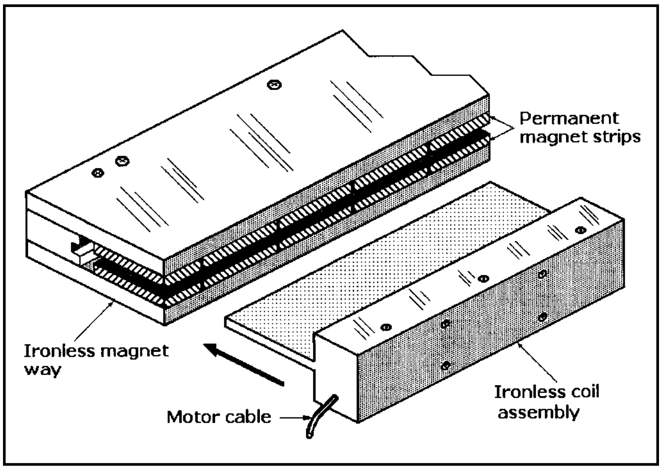


Fig. 13 A linear ironless servomotor consists of an ironless magnetic way and an ironless coil assembly.

The strip magnets are separated to form the air gap into which the coil assembly is inserted. This design maximizes the generated thrust force and also provides a flux return path for the magnetic circuit. Consequently, very little magnetic flux exists outside the motor, thus minimizing residual magnetic attraction.

Epoxy-core motors provide exceptionally smooth motion, making them suitable for applications requiring very low bearing friction and high acceleration of light loads. They also permit constant velocity to be maintained, even at very low speeds.

Linear servomotors can achieve accuracies of 0.1 µm. Normal accelerations are 2 to 3 *g*, but some motors can reach 15 *g*. Velocities are limited by the encoder data rate and the amplifier voltage. Normal peak velocities are from 0.04 in./s (1 mm/s) to about 6.6 ft/s (2 m/s), but the velocity of some models can exceed 26 ft/s (8 m/s).

Ironless linear motors can have continuous force ratings from about 5 to 55 lbf (22 to 245 N) and peak force ratings from about 25 to 180 lbf (110 to 800 N). By contrast, iron-core linear motors are available with continuous force ratings of about 30 to 1100 lbf (130 to 4900 N) and peak force ratings of about 60 to 1800 lbf (270 to 8000 N).

Commutation

The linear motor windings that are phased 120º apart must be continually switched or commutated to sustain motion. There are two ways to commutate linear motors: *sinusoidal* and *Hall-effect device (HED)*, or *trapezoidal*. The highest motor efficiency is achieved with sinusoidal commutation, while HED commutation is about 10 to 15% less efficient.

In sinusoidal commutation, the linear encoder that provides position feedback in the servosystem is also used to commutate the motor. A process called "phase finding" is required when the motor is turned on, and the motor phases are then incrementally advanced with each encoder pulse. This produces extremely smooth motion. In HED commutation a circuit board containing Hall-effect ICs is embedded in the coil assembly. The HED sensors detect the polarity change in the magnet track and switch the motor phases every 60º.

Sinusoidal commutation is more efficient than HED commutation because the coil windings in motors designed for this commutation method are configured to provide a sinusoidally shaped back EMF waveform. As a result, the motors produce a constant force output when the driving voltage on each phase matches the characteristic back EMF waveform.

Installation of Linear Motors

In a typical linear motor application the coil assembly is attached to the moving member of the host machine and the magnet assembly is mounted on the nonmoving base or frame. These motors can be mounted vertically, but if they are they typically require a counterbalance system to prevent the load from dropping if power temporarily fails or is routinely shut off. The counterbalance system, typically formed from pulleys and weights, springs, or air cylinders, supports the load against the force of gravity.

If power is lost, servo control is interrupted. Stages in motion tend to stay in motion while those at rest tend to stay at rest. The stopping time and distance depend on the stage's initial velocity and system friction. The motor's back EMF can provide dynamic braking, and friction brakes can be used to attenuate motion rapidly. However, positive stops and travel limits can be built into the motion stage to prevent damage in situations where power or feedback might be lost or the controller or servo driver fail.

Linear servomotors are supplied to the customer in kit form for mounting on the host machine. The host machine structure must include bearings capable of supporting the mass of the motor parts while maintaining the specified air gap between the assemblies and also resisting the normal force of any residual magnetic attraction.

Linear servomotors must be used in closed loop positioning systems because they do not include built-in means for position sensing. Feedback is typically supplied by such sensors as linear encoders, laser interferometers, LVDTs, or linear Inductosyns.

Advantages of Linear vs. Rotary Servomotors

The advantages of linear servomotors over rotary servomotors include:

- *High stiffness:* The linear motor is connected directly to the moving load, so there is no backlash and practically no compliance between the motor and the load. The load moves instantly in response to motor motion.
- *Mechanical simplicity:* The coil assembly is the only moving part of the motor, and its magnet assembly is rigidly mounted to a stationary structure on the host machine. Some linear motor manufacturers offer modular magnetic assemblies in various modular lengths. This permits the user to form a track of any desired length by stacking the modules end to end, allowing virtually unlimited travel. The force produced by the motor is applied directly to the load without any couplings, bearings, or other conversion mechanisms. The only alignments required are for the air gaps, which typically are from 0.039 in. (1 mm) to 0.020 in. (0.5 mm).
- *High accelerations and velocities:* Because there is no physical contact between the coil and magnet assemblies, high accelerations and velocities are possible. Large motors are capable of accelerations of 3 to 5 *g*, but smaller motors are capable of more than 10 *g*.
- *High velocities:* Velocities are limited by feedback encoder data rate and amplifier bus voltage. Normal peak velocities are up to 6.6 ft/s (2 m/s), although some models can reach 26 ft/s (8 m/s). This compares with typical linear speeds of ballscrew transmissions, which are commonly limited to 20 to 30 in./s (0.5 to 0.7 m/s) because of resonances and wear.
- *High accuracy and repeatability:* Linear motors with position feedback encoders can achieve positioning accuracies of ±1 encoder cycle or submicrometer dimensions, limited only by encoder feedback resolution.
- *No backlash or wear:* With no contact between moving parts, linear motors do not wear out. This minimizes maintenance and makes them suitable for applications where long life and long-term peak performance are required.
- *System size reduction:* With the coil assembly attached to the load, no additional space is required. By contrast, rotary motors typically require ballscrews, rack-and-pinion gearing, or timing belt drives.
- *Clean room compatibility:* Linear motors can be used in clean rooms because they do not need lubrication and do not produce carbon brush grit.

Coil Assembly Heat Dissipation

Heat control is more critical in linear motors than in rotary motors because they do not have the metal frames or cases that can act as large heat-dissipating surfaces. Some rotary motors also have radiating fins on their frames that serve as heatsinks to augment the heat dissipation capability of the frames. Linear motors must rely on a combination of high motor efficiency and good thermal conduction from the windings to a heat-conductive, electrically isolated mass. For example, an aluminum attachment bar placed in close contact with the windings can aid in heat dissipation. Moreover, the carriage plate to which the coil assembly is attached must have effective heat-sinking capability.

Stepper Motors

A *stepper* or *stepping motor* is an AC motor whose shaft is indexed through part of a revolution or *step angle* for each DC pulse sent to it. Trains of pulses provide input current to the motor in increments that can "step" the motor through 360º, and the actual angular rotation of the shaft is directly related to the number of pulses introduced. The position of the load can be determined with reasonable accuracy by counting the pulses entered.

The stepper motors suitable for most open-loop motion control applications have wound stator fields (electromagnetic coils) and iron or permanent magnet (PM) rotors. Unlike PM DC servomotors with mechanical brush-type commutators, stepper motors depend on external controllers to provide the switching pulses for commutation. Stepper motor operation is based on the same electromagnetic principles of attraction and repulsion as other motors, but their commutation provides only the torque required to turn their rotors.

Pulses from the external motor controller determine the amplitude and direction of current flow in the stator's field windings, and they can turn the motor's rotor either clockwise or counterclockwise, stop and start it quickly, and hold it securely at desired positions. Rotational shaft speed depends on the frequency of the pulses. Because controllers can step most motors at audio frequencies, their rotors can turn rapidly.

Between the application of pulses when the rotor is at rest, its armature will not drift from its stationary position because of the stepper motor's inherent holding ability or *detent torque*. These motors generate very little heat while at rest, making them suitable for many different instrument drive-motor applications in which power is limited.

The three basic kinds of stepper motors are *permanent magnet, variable reluctance,* and *hybrid.* The same controller circuit can drive both hybrid and PM stepping motors.

Permanent-Magnet (PM) Stepper Motors

Permanent-magnet stepper motors have smooth armatures and include a permanent magnet core that is magnetized widthwise or perpendicular to its rotation axis. These motors usually have two independent windings, with or without center taps. The most common step angles for PM motors are 45 and 90º, but motors with step angles as fine as 1.8º per step as well as 7.5, 15, and 30º per step are generally available. Armature rotation occurs when the stator poles are alternately energized and deenergized to create torque. A 90º stepper has four poles and a 45º stepper has eight poles, and these poles must be energized in sequence. Permanent-magnet steppers step at relatively low rates, but they can produce high torques and they offer very good damping characteristics.

Variable Reluctance Stepper Motors

Variable reluctance (VR) stepper motors have multitooth armatures with each tooth effectively an individual magnet. At rest these magnets align themselves in a natural detent position to provide larger holding torque than can be obtained with a comparably rated PM stepper. Typical VR motor step angles are 15 and 30º per step. The 30º angle is obtained with a 4-tooth rotor and a 6-pole stator, and the 15º angle is achieved with an 8-tooth rotor and a 12-pole stator. These motors typically have three windings with a common return, but they are also available with four or five windings. To obtain continuous rotation, power must be applied to the windings in a coordinated sequence of alternately deenergizing and energizing the poles.

If just one winding of either a PM or VR stepper motor is energized, the rotor (under no load) will snap to a fixed angle and hold that angle until external torque exceeds the holding torque of the motor. At that point, the rotor will turn, but it will still try to hold its new position at each successive equilibrium point.

Hybrid Stepper Motors

The hybrid stepper motor combines the best features of VR and PM stepper motors. A cutaway view of a typical industrial-grade hybrid stepper motor with a multitoothed armature is shown in Fig. 14. The armature is built in two sections, with the teeth in the second section offset from those in the first section. These motors also have multitoothed stator poles that are not visible in the figure. Hybrid stepper motors can achieve high stepping rates, and they offer high detent torque and excellent dynamic and static torque.

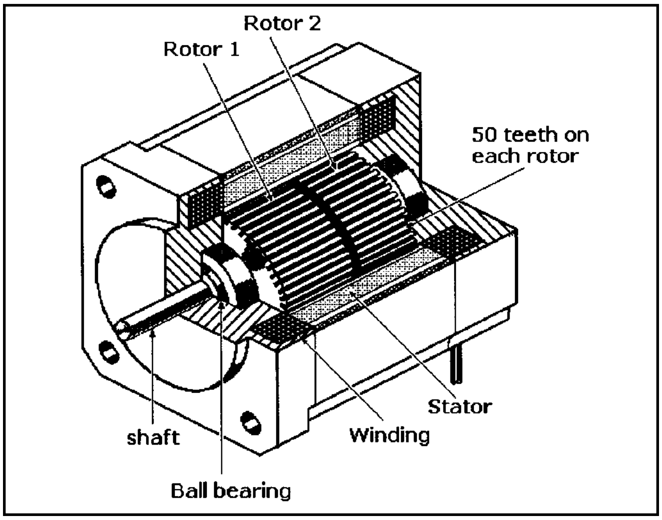


Fig. 14 Cutaway view of a 5-phase hybrid stepping motor. A permanent magnet is within the rotor assembly, and the rotor segments are offset from each other by 3.5°.

Hybrid steppers typically have two windings on each stator pole so that each pole can become either magnetic north or south, depending on current flow. A cross-sectional view of a hybrid stepper motor illustrating the multitoothed poles with dual windings per pole and the multitoothed rotor is illustrated in Fig. 15. The shaft is represented by the central circle in the diagram.

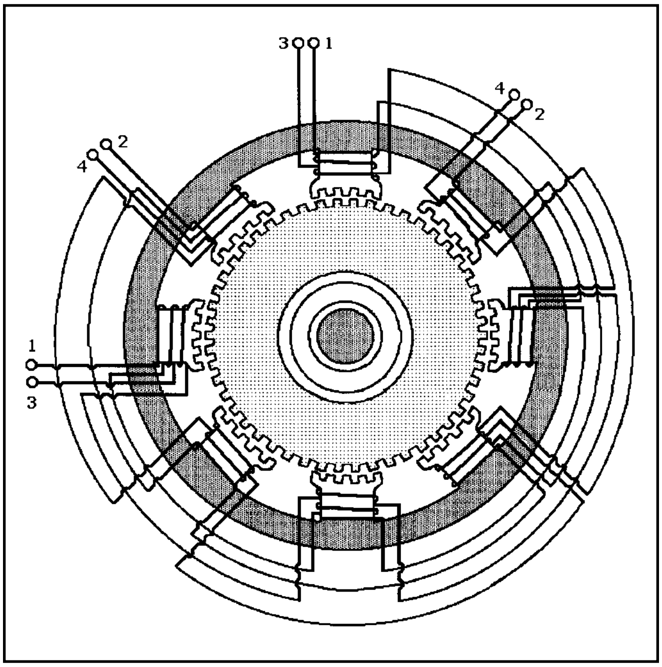


Fig. 15 Cross-section of a hybrid stepping motor showing the segments of the magnetic-core rotor and stator poles with its wiring diagram.

The most popular hybrid steppers have 3- and 5-phase wiring, and step angles of 1.8 and 3.6º per step. These motors can provide more torque from a given frame size than other stepper types because either all or all but one of the motor windings are energized at every point in the drive cycle. Some 5-phase motors have high resolutions of 0.72° per step (500 steps per revolution). With a compatible controller, most PM and hybrid motors can be run in half-steps, and some controllers are designed to provide smaller fractional steps, or *microsteps*. Hybrid stepper motors capable of a wide range of torque values are available commercially. This range is achieved by scaling length and diameter dimensions. Hybrid stepper motors are available in NEMA size 17 to 42 frames, and output power can be as high as 1000 W peak.

Stepper Motor Applications

Many different technical and economic factors must be considered in selecting a hybrid stepper motor. For example, the ability of the stepper motor to repeat the positioning of its multitoothed rotor depends on its geometry. A disadvantage of the hybrid stepper motor operating open-loop is that, if overtorqued, its position "memory" is lost and the system must be reinitialized. Stepper motors can perform precise positioning in simple open-loop control systems if they operate at low acceleration rates with static loads. However, if higher acceleration values are required for driving variable loads, the stepper motor must be operated in a closed loop with a position sensor.

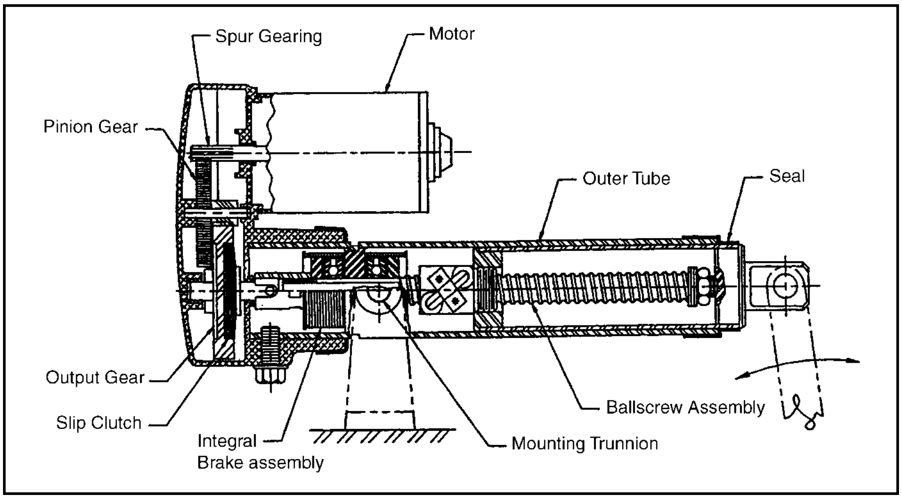


Fig. 16 This linear actuator can be powered by either an AC or DC motor. It contains ballscrew, reduction gear, clutch, and brake assemblies. *Courtesy of Thomson Saginaw.*

DC and AC Motor Linear Actuators

Actuators for motion control systems are available in many different forms, including both linear and rotary versions. One popular configuration is that of a Thomson Saginaw PPA, shown in section view in Fig. 16. It consists of an AC or DC motor mounted parallel to either a ballscrew or Acme screw assembly through a reduction gear assembly with a slip clutch and integral brake assembly. Linear actuators of this type can perform a wide range of commercial, industrial, and institutional applications.

One version designed for mobile applications can be powered by a 12-, 24-, or 36-VDC permanent-magnet motor. These motors are capable of performing such tasks as positioning antenna reflectors, opening and closing security gates, handling materials, and raising and lowering scissors-type lift tables, machine hoods, and light-duty jib crane arms.

Other linear actuators are designed for use in fixed locations where either 120- or 220-VAC line power is available. They can have either AC or DC motors. Those with 120-VAC motors can be equipped with optional electric brakes that virtually eliminate coasting, thus permitting point-to-point travel along the stroke.

Where variable speed is desired and 120-VAC power is available, a linear actuator with a 90-VDC motor can be equipped with a solid-state rectifier/speed controller. Closed-loop feedback provides speed regulation down to one tenth of the maximum travel rate. This feedback system can maintain its selected travel rate despite load changes.

Thomson Saginaw also offers its linear actuators with either Hall-effect or potentiometer sensors for applications where it is necessary or desirable to control actuator positioning. With Hall-effect sensing, six pulses are generated with each turn of the output shaft during which the stroke travels approximately $\frac{1}{32}$ in. (0.033 in. or 0.84 mm). These pulses can be counted by a separate control unit and added or subtracted from the stored pulse count in the unit's memory. The actuator can be stopped at any 0.033-in. increment of travel along the stroke selected by programming. A limit switch can be used together with this sensor.

If a 10-turn, 10,000-ohm potentiometer is used as a sensor, it can be driven by the output shaft through a spur gear. The gear ratio is established to change the resistance from 0 to 10,000 ohms over the length of the actuator stroke. A separate control unit measures the resistance (or voltage) across the potentiometer, which varies continuously and linearly with stroke travel. The actuator can be stopped at any position along its stroke.

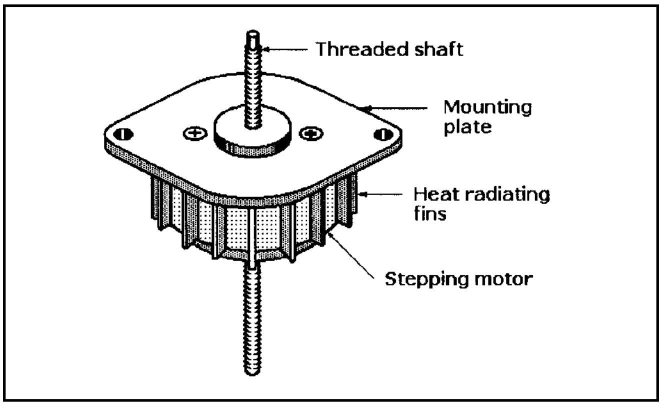


Fig. 17 This light-duty linear actuator based on a permanent-magnet stepping motor has a shaft that advances or retracts.

Stepper-Motor Based Linear Actuators

Linear actuators are available with axial integral threaded shafts and bolt nuts that convert rotary motion to linear motion. Powered by fractional horsepower permanent-magnet stepper motors, these linear actuators are capable of positioning light loads. Digital pulses fed to the actuator cause the threaded shaft to rotate, advancing or retracting it so that a load coupled to the shaft can be moved backward or forward. The bidirectional digital linear actuator shown in Fig. 17 can provide linear resolution as fine as 0.001 in. per pulse. Travel per step is determined by the pitch of the leadscrew and step angle of the motor. The maximum linear force for the model shown is 75 oz.

SERVOSYSTEM FEEDBACK SENSORS

A servosystem feedback sensor in a motion control system transforms a physical variable into an electrical signal for use by the motion controller. Common feedback sensors are encoders, resolvers, and linear variable differential transformers (LVDTs) for motion and position feedback, and tachometers for velocity feedback. Less common but also in use as feedback devices are potentiometers, linear velocity transducers (LVTs), angular displacement transducers (ADTs), laser interferometers, and potentiometers. Generally speaking, the closer the feedback sensor is to the variable being controlled, the more accurate it will be in assisting the system to correct velocity and position errors.

For example, direct measurement of the linear position of the carriage carrying the load or tool on a single-axis linear guide will provide more accurate feedback than an indirect measurement determined from the angular position of the guide's leadscrew and knowledge of the drivetrain geometry between the sensor and the carriage. Thus, direct position measurement avoids drivetrain errors caused by backlash, hysteresis, and leadscrew wear that can adversely affect indirect measurement.

Rotary Encoders

Rotary encoders, also called *rotary shaft encoders* or *rotary shaft-angle* encoders, are electromechanical transducers that convert shaft rotation into output pulses, which can be counted to measure shaft revolutions or shaft angle. They provide rate and positioning information in servo feedback loops. A rotary encoder can sense a number of discrete positions per revolution. The number is called *points per revolution* and is analogous to the *steps per revolution* of a stepper motor. The speed of an encoder is in units of counts per second. Rotary encoders can measure the motor-shaft or leadscrew angle to report position indirectly, but they can also measure the response of rotating machines directly.

The most popular rotary encoders are *incremental optical shaft-angle encoders* and the *absolute optical shaft-angle encoders*. There are also *direct contact* or *brush-type* and *magnetic rotary encoders*, but they are not as widely used in motion control systems.

Commercial rotary encoders are available as standard or catalog units, or they can be custom made for unusual applications or survival in extreme environments. Standard rotary encoders are packaged in cylindrical cases with diameters from 1.5 to 3.5 in. Resolutions range from 50 cycles per shaft revolution to 2,304,000 counts per revolution. A variation of the conventional configuration, the *hollow-shaft encoder,* eliminates problems associated with the installation and shaft runout of conventional models. Models with hollow shafts are available for mounting on shafts with diameters of 0.04 to 1.6 in. (1 to 40 mm).

Incremental Encoders

The basic parts of an incremental optical shaft-angle encoder are shown in Fig. 1. A glass or plastic code disk mounted on the encoder shaft rotates between an internal light source, typically a light-emitting diode (LED), on one side and a mask and matching photodetector assembly on the other side. The incremental code disk contains a pattern of equally spaced opaque and transparent segments or spokes that radiate out from its center as shown. The electronic signals that are generated by the encoder's electronics board are fed into a motion controller that calculates position and velocity information for feedback purposes. An exploded view of an industrial-grade incremental encoder is shown in Fig. 2.

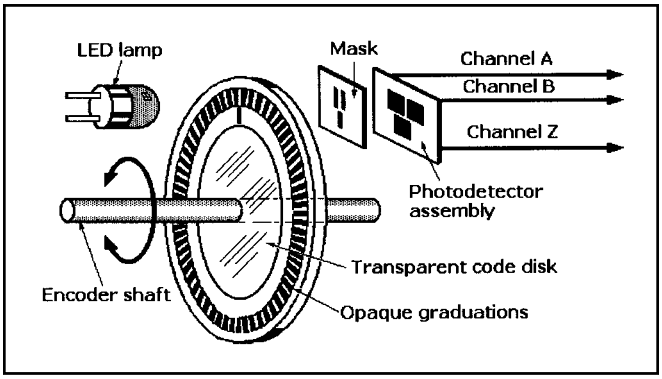


Fig. 1 Basic elements of an incremental optical rotary encoder.

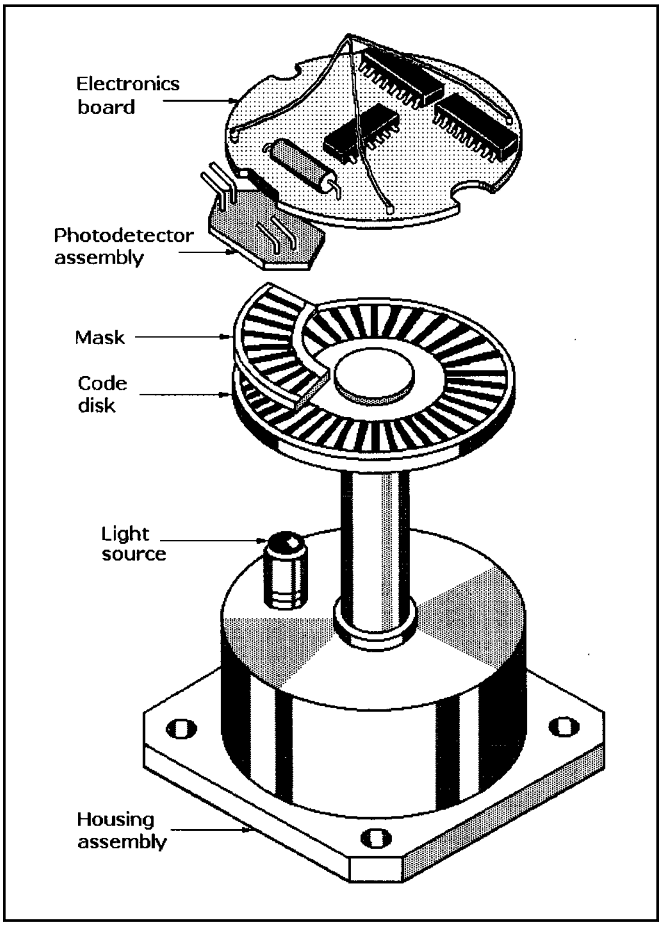


Fig. 2 Exploded view of an incremental optical rotary encoder showing the stationary mask between the code wheel and the photodetector assembly.

Glass code disks containing finer graduations capable of 11- to more than 16-bit resolution are used in high-resolution encoders, and plastic (Mylar) disks capable of 8- to 10-bit resolution are used in the more rugged encoders that are subject to shock and vibration.

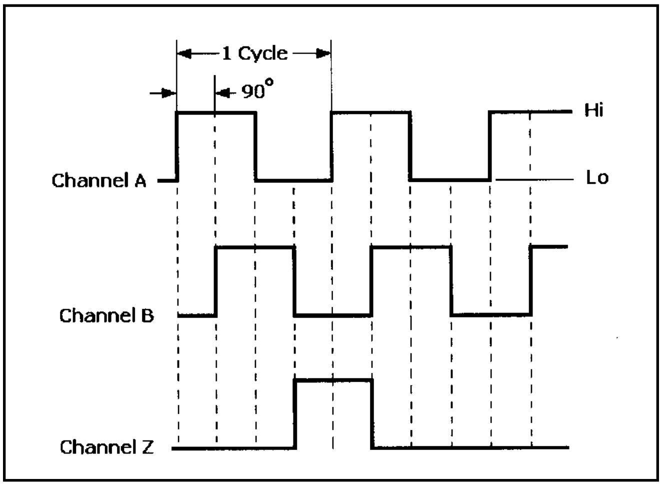


Fig. 3 Channels A and B provide bidirectional position sensing. If channel A leads channel B, the direction is clockwise; if channel B leads channel A, the direction is counterclockwise. Channel Z provides a zero reference for determining the number of disk rotations.

The quadrature encoder is the most common type of incremental encoder. Light from the LED passing through the rotating code disk and mask is "chopped" before it strikes the photodetector assembly. The output signals from the assembly are converted into two channels of square pulses (A and B) as shown in Fig. 3. The number of square pulses in each channel is equal to the number of code disk segments that pass the photodetectors as the disk rotates, but the waveforms are 90º out of phase. If, for example, the pulses in channel A lead those in channel B, the disk is rotating in a clockwise direction, but if the pulses in channel A lag those in channel B lead, the disk is rotating counterclockwise. By monitoring both the number of pulses and the relative phases of signals A and B, both position and direction of rotation can be determined.

Many incremental quadrature encoders also include a third output Z channel to obtain a zero reference or index signal that occurs once per revolution. This channel can be gated to the A and B quadrature channels and used to trigger certain events accurately within the system. The signal can also be used to align the encoder shaft to a mechanical reference.

Absolute Encoders

An *absolute shaft-angle optical encoder* contains multiple light sources and photodetectors, and a code disk with up to 20 tracks of segmented patterns arranged as annular rings, as shown in Fig. 4. The code disk provides a binary output that uniquely defines each shaft angle, thus providing an absolute measurement. This type of encoder is organized in essentially the same way as the incremental encoder shown in Fig. 2, but the code disk rotates between linear arrays of LEDs and photodetectors arranged radially, and a LED opposes a photodetector for each track or annular ring.

The arc lengths of the opaque and transparent sectors decrease with respect to the radial distance from the shaft. These disks, also made of glass or plastic, produce either the natural binary or Gray code. Shaft position accuracy is proportional to the number of annular rings or tracks on the disk. When the code disk rotates, light passing through each track or annular ring generates a continuous stream of signals from the detector array. The electronics

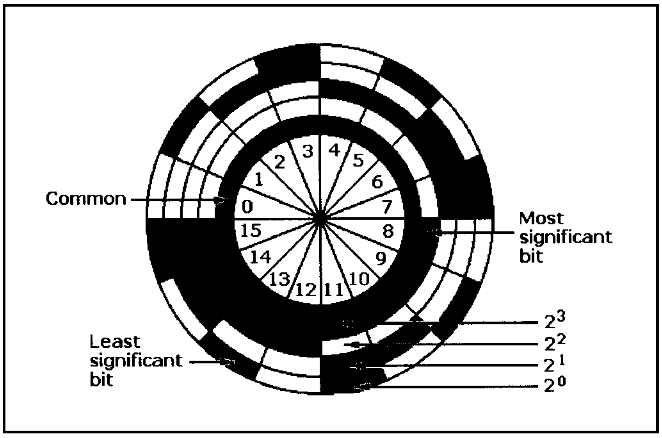


Fig. 4 Binary-code disk for an absolute optical rotary encoder. Opaque sectors represent a binary value of 1, and the transparent sectors represent binary 0. This four-bit binary-code disk can count from 1 to 15.

board converts that output into a binary word. The value of the output code word is read radially from the most significant bit (MSB) on the inner ring of the disk to the least significant bit (LSB) on the outer ring of the disk.

The principal reason for selecting an absolute encoder over an incremental encoder is that its code disk retains the last angular position of the encoder shaft whenever it stops moving, whether the system is shut down deliberately or as a result of power failure. This means that the last readout is preserved, an important feature for many applications.

Linear Encoders

Linear encoders can make direct accurate measurements of unidirectional and reciprocating motions of mechanisms with high resolution and repeatability. Figure 5 illustrates the basic parts of an optical linear encoder. A movable scanning unit contains the light source, lens, graduated glass scanning reticule, and an array of photocells. The scale, typically made as a strip of glass with opaque graduations, is bonded to a supporting structure on the host machine.

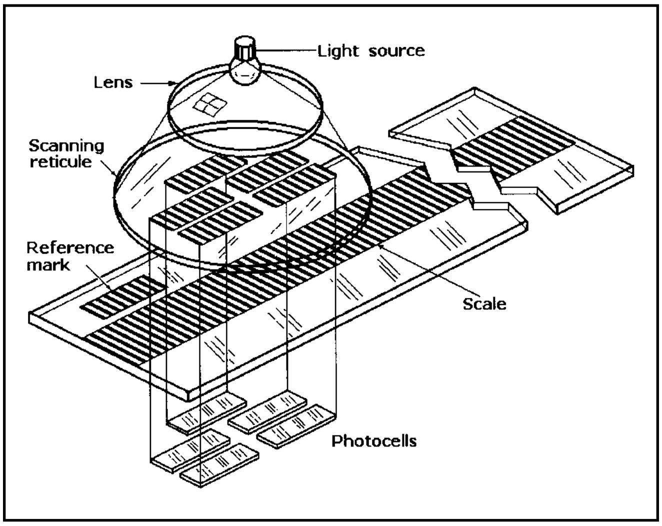


Fig. 5 Optical linear encoders direct light through a moving glass scale with accurately etched graduations to photocells on the opposite side for conversion to a distance value.

A beam of light from the light source passes through the lens, four windows of the scanning reticule, and the glass scale to the array of photocells. When the scanning unit moves, the scale modulates the light beam so that the photocells generate sinusoidal signals.

The four windows in the scanning reticule are each 90º apart in phase. The encoder combines the phase-shifted signal to produce two symmetrical sinusoidal outputs that are phase shifted by 90º. A fifth pattern on the scanning reticule has a random graduation that, when aligned with an identical reference mark on the scale, generates a reference signal.

A fine-scale pitch provides high resolution. The spacing between the scanning reticule and the fixed scale must be narrow and constant to eliminate undesirable diffraction effects of the scale grating. The complete scanning unit is mounted on a carriage that moves on ball bearings along the glass scale. The scanning unit is connected to the host machine slide by a coupling that compensates for any alignment errors between the scale and the machine guideways.

External electronic circuitry interpolates the sinusoidal signals from the encoder head to subdivide the line spacing on the scale so that it can measure even smaller motion increments. The practical maximum length of linear encoder scales is about 10 ft (3 m), but commercial catalog models are typically limited to about 6 ft (2 m). If longer distances are to be measured, the encoder scale is made of steel tape with reflective graduations that are sensed by an appropriate photoelectric scanning unit.

Linear encoders can make direct measurements that overcome the inaccuracies inherent in mechanical stages due to backlash, hysteresis, and leadscrew error. However, the scale's susceptibility to damage from metallic chips, grit oil, and other contaminants, together with its relatively large space requirements, limits applications for these encoders.

Commercial linear encoders are available as standard catalog models, or they can be custom made for specific applications or extreme environmental conditions. There are both fully enclosed and open linear encoders with travel distances from 2 in. to 6 ft (50 mm to 1.8 m). Some commercial models are available with resolutions down to 0.07 µm, and others can operate at speeds of up to 16.7 ft/s (5 m/s).

Magnetic Encoders

Magnetic encoders can be made by placing a transversely polarized permanent magnet in close proximity to a Hall-effect device sensor. Figure 6 shows a magnet mounted on a motor shaft in close proximity to a two-channel HED array which detects changes in magnetic flux density as the magnet rotates. The output signals from the sensors are transmitted to the motion controller. The encoder output, either a square wave or a quasi sine wave (depending on the type of magnetic sensing device) can be used to count revolutions per minute (rpm) or determine motor shaft accurately. The phase shift between channels A and B permits them to be compared by the motion controller to determine the direction of motor shaft rotation.

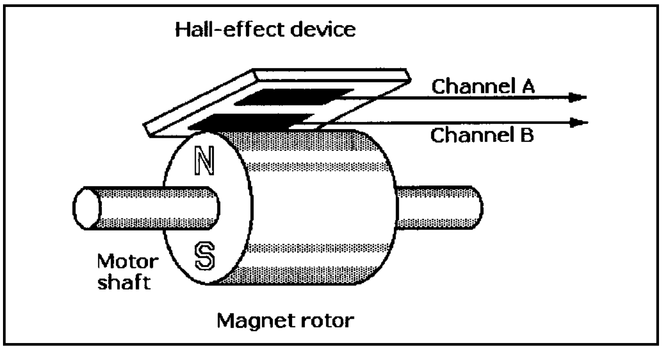


Fig. 6 Basic parts of a magnetic encoder.

Resolvers

A resolver is essentially a rotary transformer that can provide position feedback in a servosystem as an alternative to an encoder. Resolvers resemble small AC motors, as shown in Fig. 7, and generate an electrical signal for each revolution of their shaft. Resolvers that sense position in closed-loop motion control applications have one winding on the rotor and a pair of windings on the stator, oriented at 90º. The stator is made by winding copper wire in a stack of iron laminations fastened to the housing, and the rotor is made by winding copper wire in a stack of laminations mounted on the resolver's shaft.

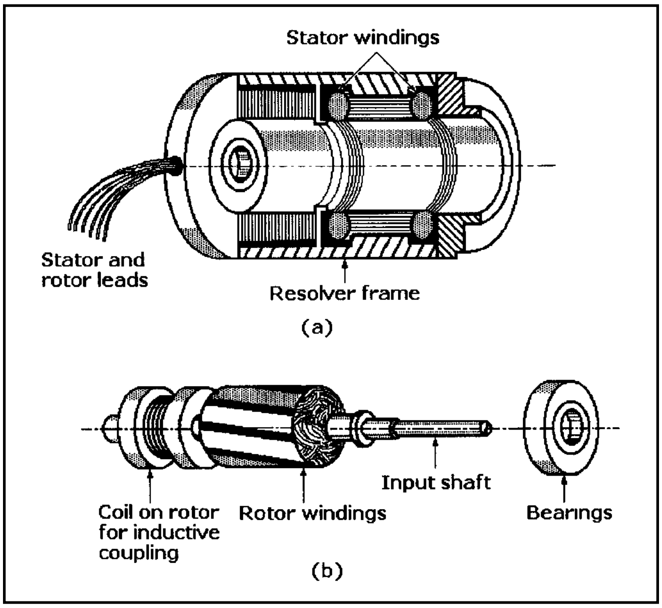


Fig. 7 Exploded view of a brushless resolver frame (a), and rotor and bearings (b). The coil on the rotor couples speed data inductively to the frame for processing.

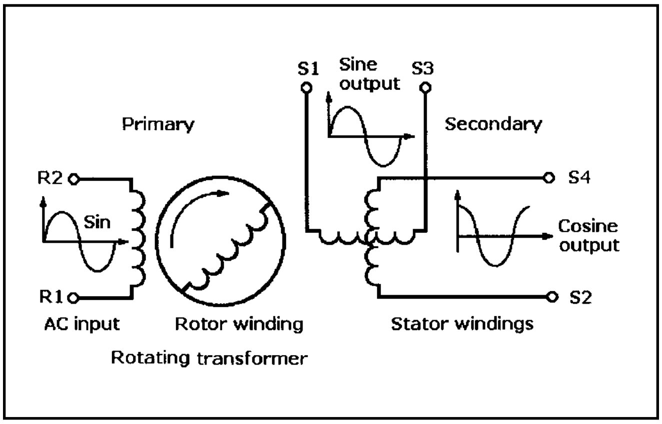


Fig. 8 Schematic for a resolver shows how rotor position is transformed into sine and cosine outputs that measure rotor position.

Figure 8 is an electrical schematic for a brushless resolver showing the single rotor winding and the two stator windings 90º apart. In a servosystem, the resolver's rotor is mechanically coupled to the drive motor and load. When a rotor winding is excited by an AC reference signal, it produces an AC voltage output that varies in amplitude according to the sine and cosine of shaft position. If the phase shift between the applied signal to the rotor and the induced signal appearing on the stator coil is measured, that

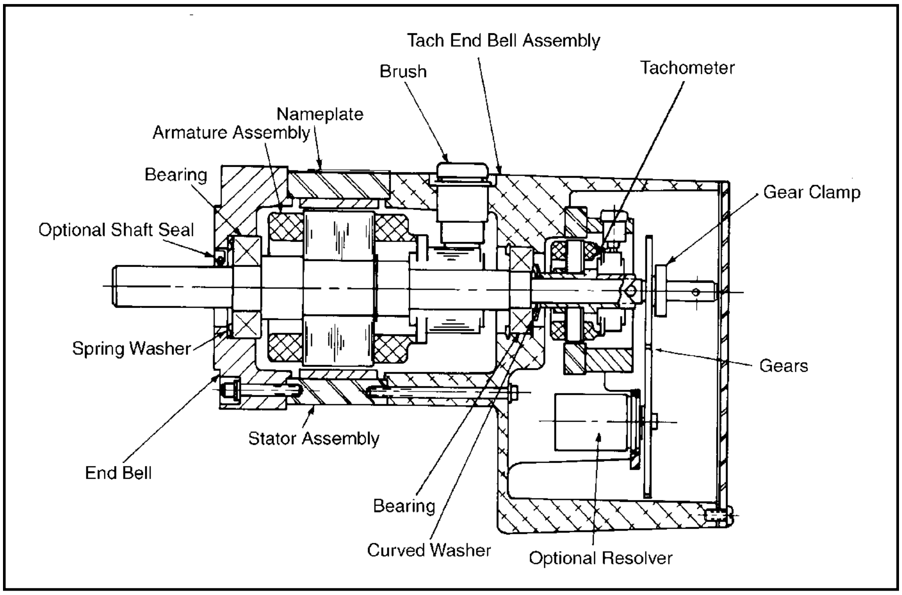


Fig. 9 Section view of a resolver and tachometer in the same frame as the servomotor.

angle is an analog of rotor position. The absolute position of the load being driven can be determined by the ratio of the sine output amplitude to the cosine output amplitude as the resolver shaft turns through one revolution. (A single-speed resolver produces one sine and one cosine wave as the output for each revolution.)

Connections to the rotor of some resolvers can be made by brushes and slip rings, but resolvers for motion control applications are typically brushless. A rotating transformer on the rotor couples the signal to the rotor inductively. Because brushless resolvers have no slip rings or brushes, they are more rugged than encoders and have operating lives that are up to ten times those of brush-type resolvers. Bearing failure is the most likely cause of resolver failure. The absence of brushes in these resolvers makes them insensitive to vibration and contaminants. Typical brushless resolvers have diameters from 0.8 to 3.7 in. Rotor shafts are typically threaded and splined.

Most brushless resolvers can operate over a 2- to 40-volt range, and their winding are excited by an AC reference voltage at frequencies from 400 to 10,000 Hz. The magnitude of the voltage induced in any stator winding is proportional to the cosine of the angle, *q*, between the rotor coil axis and the stator coil axis. The voltage induced across any pair of stator terminals will be the vector sum of the voltages across the two connected coils. Accuracies of ±1 arc-minute can be achieved.

In feedback loop applications, the stator's sinusoidal output signals are transmitted to a resolver-to-digital converter (RDC), a specialized analog-to-digital converter (ADC) that converts the signals to a digital representation of the actual angle required as an input to the motion controller.

Tachometers

A tachometer is a DC generator that can provide velocity feedback for a servosystem. The tachometer's output voltage is directly proportional to the rotational speed of the armature shaft that drives it. In a typical servosystem application, it is mechanically coupled to the DC motor and feeds its output voltage back to the controller and amplifier to control drive motor and load speed. A cross-sectional drawing of a tachometer built into the same housing as the DC motor and a resolver is shown in Fig. 9. Encoders or resolvers are part of separate loops that provide position feedback.

As the tachometer's armature coils rotate through the stator's magnetic field, lines of force are cut so that an electromotive force is induced in each of its coils. This emf is directly proportional to the rate at which the magnetic lines of force are cut as well as being directly proportional to the velocity of the motor's drive shaft. The direction of the emf is determined by Fleming's generator rule.

The AC generated by the armature coil is converted to DC by the tachometer's commutator, and its value is directly proportional to shaft rotation speed while its polarity depends on the direction of shaft rotation.

There are two basic types of DC tachometer: *shunt wound* and *permanent magnet* (PM), but PM tachometers are more widely used in servosystems today. There are also moving-coil tachometers which, like motors, have no iron in their armatures. The armature windings are wound from fine copper wire and bonded with glass fibers and polyester resins into a rigid cup, which is bonded to its coaxial shaft. Because this armature contains no iron, it has lower inertia than conventional copper and iron armatures, and it exhibits low inductance. As a result, the moving-coil tachometer is more responsive to speed changes and provides a DC output with very low ripple amplitudes.

Tachometers are available as standalone machines. They can be rigidly mounted to the servomotor housings, and their shafts can be mechanically coupled to the servomotor's shafts. If the DC servomotor is either a brushless or moving-coil motor, the standalone tachometer will typically be brushless and, although they are housed separately, a common armature shaft will be shared.

A brush-type DC motor with feedback furnished by a brush-type tachometer is shown in Fig. 10. Both tachometer and motor rotor coils are mounted on a common shaft. This arrangement

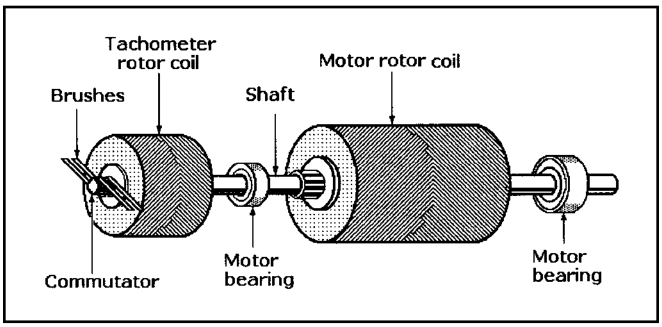


Fig. 10 The rotors of the DC motor and tachometer share a common shaft.

provides a high resonance frequency. Moreover, the need for separate tachometer bearings is eliminated.

In applications where precise positioning is required in addition to speed regulation, an incremental encoder can be added on the same shaft, as shown in Fig. 11.

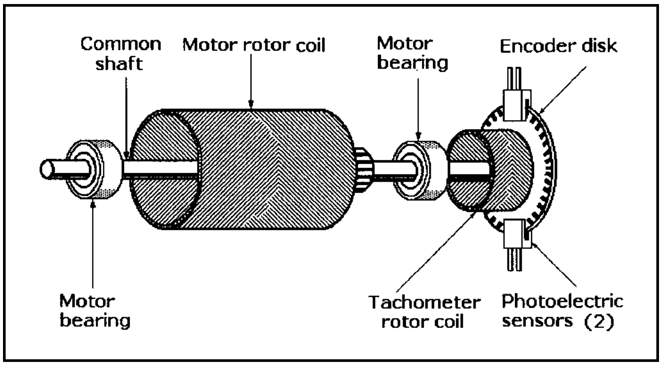


Fig. 11 This coil-type DC motor obtains velocity feedback from a tachometer whose rotor coil is mounted on a common shaft and position feedback from a two-channel photoelectric encoder whose code disk is also mounted on the same shaft.

Linear Variable Differential Transformers (LVDTs)

A linear variable differential transformer (LVDT) is a sensing transformer consisting of a primary winding, two adjacent secondary windings, and a ferromagnetic core that can be moved axially within the windings, as shown in the cutaway view Fig. 12. LVDTs are capable of measuring position, acceleration, force, or pressure, depending on how they are installed. In motion control systems, LVDTs provide position feedback by measuring the variation in mutual inductance between their primary and secondary windings caused by the linear movement of the ferromagnetic core.

The core is attached to a spring-loaded sensing shaft. When depressed, the shaft moves the core axially within the windings, coupling the excitation voltage in the primary (middle) winding P1 to the two adjacent secondary windings S1 and S2.

Figure 13 is a schematic diagram of an LVDT. When the core is centered between S1 and S2, the voltages induced in S1 and S2 have equal amplitudes and are 180º out of phase. With a series-opposed connection, as shown, the net voltage across the secondaries is zero because both voltages cancel. This is called the *null position* of the core.

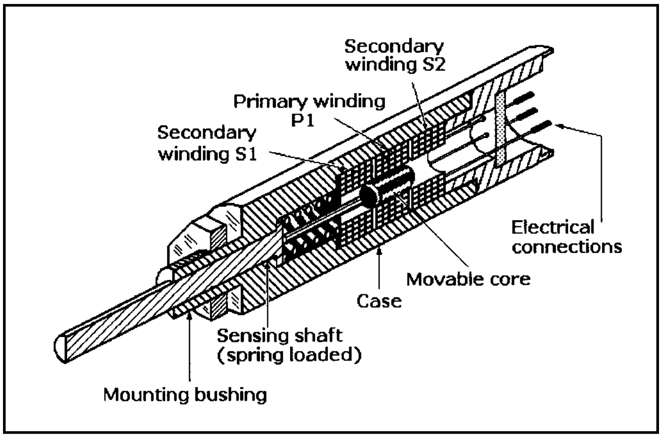


Fig. 12 Cutaway view of a linear variable displacement transformer (LVDT).

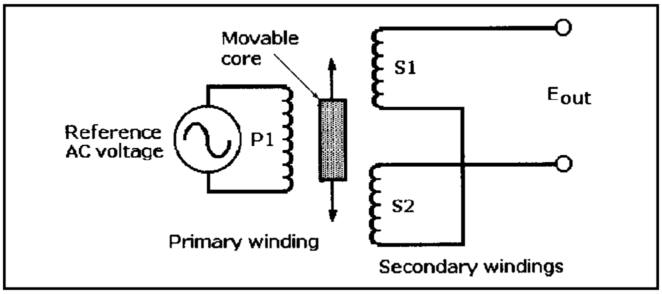


Fig. 13 Schematic for a linear variable differential transformer (LVDT) showing how the movable core interacts with the primary and secondary windings.

However, if the core is moved to the left, secondary winding S1 is more strongly coupled to primary winding P1 than secondary winding S2, and an output sine wave in phase with the primary voltage is induced. Similarly, if the core is moved to the right and winding S2 is more strongly coupled to primary winding P1, an output sine wave that is 180º out-of-phase with the primary voltage is induced. The amplitudes of the output sine waves of the LVDT vary symmetrically with core displacement, either to the left or right of the null position.

Linear variable differential transformers require signal conditioning circuitry that includes a stable sine wave oscillator to excite the primary winding P1, a demodulator to convert secondary AC voltage signals to DC, a low-pass filter, and an amplifier to buffer the DC output signal. The amplitude of the resulting DC voltage output is proportional to the magnitude of core displacement, either to the left or right of the null position. The phase of the DC voltage indicates the position of the core relative to the null (left or right). An LVDT containing an integral oscillator/demodulator is a DC-to-DC LVDT, also known as a DCDT.

Linear variable differential transformers can make linear displacement (position) measurements as precise as 0.005 in. (0.127 mm). Output voltage linearity is an important LVDT characteristic, and it can be plotted as a straight line within a specified range. Linearity is the characteristic that largely determines the LVDT's absolute accuracy.

Linear Velocity Transducers (LVTs)

A linear velocity transducer (LVT) consists of a magnet positioned axially within a two wire coils. When the magnet is moved through the coils, it induces a voltage within the coils in accordance with the Faraday and Lenz laws. The output voltage from the coils is directly proportional to the magnet's field strength and axial velocity over its working range.

When the magnet is functioning as a transducer, both of its ends are within the two adjacent coils, and when it is moved axially, its north pole will induce a voltage in one coil and its south pole will induce a voltage in the other coil. The two coils can be connected in series or parallel, depending on the application. In both configurations the DC output voltage from the coils is proportional to magnet velocity. (A single coil would only produce zero voltage because the voltage generated by the north pole would be canceled by the voltage generated by the south pole.)

The characteristics of the LVT depend on how the two coils are connected. If they are connected in series opposition, the output is added and maximum sensitivity is obtained. Also, noise generated in one coil will be canceled by the noise generated in the other coil. However, if the coils are connected in parallel, both sensitivity and source impedance are reduced. Reduced sensitivity improves high-frequency response for measuring high velocities, and the lower output impedance improves the LVT's compatibility with its signal-conditioning electronics.

Angular Displacement Transducers (ATDs)

An angular displacement transducer is an air-core variable differential capacitor that can sense angular displacement. As shown in exploded view Fig. 14 it has a movable metal rotor sandwiched between a single stator plate and segmented stator plates. When a high-frequency AC signal from an oscillator is placed across the plates, it is modulated by the change in capacitance value due to the position of the rotor with respect to the segmented stator plates. The angular displacement of the rotor can then be determined accurately from the demodulated AC signal.

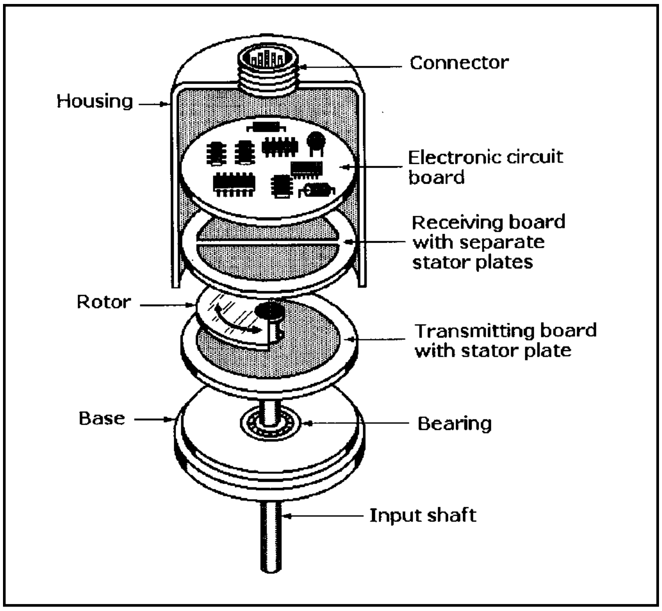


Fig. 14 Exploded view of an angular displacement transducer (ADT) based on a differential variable capacitor.

The base is the mounting platform for the transducer assembly. It contains the axial ball bearing that supports the shaft to which the rotor is fastened. The base also supports the transmitting board, which contains a metal surface that forms the lower plate of the differential capacitor. The semicircular metal rotor mounted on the shaft is the variable plate or rotor of the capacitor. Positioned above the rotor is the receiving board containing two separate semicircular metal sectors on its lower surface. The board acts as the receiver for the AC signal that has been modulated by the capacitance difference between the plates caused by rotor rotation.

An electronics circuit board mounted on top of the assembly contains the oscillator, demodulator, and filtering circuitry. The ADT is powered by DC, and its output is a DC signal that is proportional to angular displacement. The cup-shaped housing encloses the entire assembly, and the base forms a secure cap.

DC voltage is applied to the input terminals of the ADT to power the oscillator, which generates a 400- to 500-kHz voltage that is applied across the transmitting and receiving stator plates. The receiving plates are at virtual ground, and the rotor is at true ground. The capacitance value between the transmitting and receiving plates remains constant, but the capacitance between the separate receiving plates varies with rotor position.

A null point is obtained when the rotor is positioned under equal areas of the receiving stator plates. In that position, the capacitance between the transmitting stator plate and the receiving stator plates will be equal, and there will be no output voltage. However, as the rotor moves clockwise or counterclockwise, the capacitance between the transmitting plate and one of the receiving plates will be greater than it is between the other receiving plate. As a result, after demodulation, the differential output DC voltage will be proportional to the angular distance the rotor moved from the null point.

Inductosyns

The Inductosyn is a proprietary AC sensor that generates position feedback signals that are similar to those from a resolver. There are rotary and linear Inductosyns. Much smaller than a resolver, a rotary Inductosyn is an assembly of a scale and slider on insulating substrates in a loop. When the scale is energized with AC, the voltage couples into the two slider windings and induces voltages proportional to the sine and cosine of the slider spacing within a cyclic pitch.

An Inductosyn-to-digital (I/D) converter, similar to a resolver-to-digital (R/D) converter, is needed to convert these signals into a digital format. A typical rotary Inductosyn with 360 cyclic pitches per rotation can resolve a total of 1,474,560 sectors for each resolution. This corresponds to an angular rotation of less than 0.9 arc-s. This angular information in a digital format is sent to the motion controller.

Laser Interferometers

Laser interferometers provide the most accurate position feedback for servosystems. They offer very high resolution (to 1.24 nm), noncontact measurement, a high update rate, and intrinsic accuracies of up to 0.02 ppm. They can be used in servosystems either as passive position readouts or as active feedback sensors in a position servo loop. The laser beam path can be precisely aligned to coincide with the load or a specific point being measured, eliminating or greatly reducing Abbe error.

A single-axis system based on the Michaelson interferometer is illustrated in Fig. 15. It consists of a helium–neon laser, a polarizing beam splitter with a stationary retroreflector, a moving retroreflector that can be mounted on the object whose position is to be measured, and a photodetector, typically a photodiode.

Light from the laser is directed toward the polarizing beam splitter, which contains a partially reflecting mirror. Part of the laser beam goes straight through the polarizing beam splitter, and part of the laser beam is reflected. The part that goes straight through the beam splitter reaches the moving reflectometer, which reflects it back to the beam splitter, that passes it on to the photodetector. The part of the beam that is reflected by the beam splitter reaches the stationary retroreflector, a fixed distance away. The retroreflector reflects it back to the beam splitter before it is also reflected into the photodetector.

As a result, the two reflected laser beams strike the photodetector, which converts the combination of the two light beams into an electrical signal. Because of the way laser light beams interact, the output of the detector depends on a *difference* in the distances traveled by the two laser beams. Because both light beams travel the same distance from the laser to the beam splitter and from the beam splitter to the photodetector, these distances

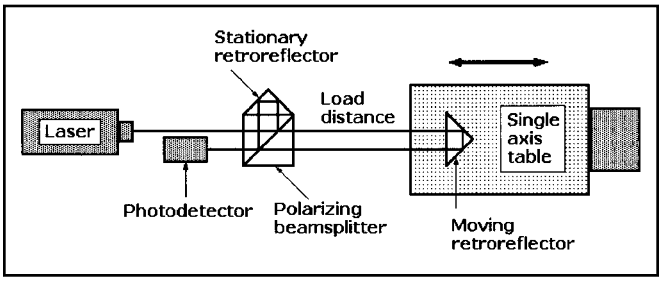


Fig. 15 Diagram of a laser interferometer for position feedback that combines high resolution with noncontact sensing, high update rates, and accuracies of 0.02 ppm.

are not involved in position measurement. The laser interferometer measurement depends only on the difference in distance between the round trip laser beam travel from the beam splitter to the moving retroreflector and the fixed round trip distance of laser beam travel from the beam splitter to the stationary retroreflector.

If these two distances are exactly the same, the two light beams will recombine in phase at the photodetector, which will produce a high electrical output. This event can be viewed on a video display as a bright *light fringe*. However, if the difference between the distances is as short as one-quarter of the laser's wavelength, the light beams will combine out-of-phase, interfering with each other so that there will be no electrical output from the photodetector and no video output on the display, a condition called a *dark fringe*.

As the moving retroreflector mounted on the load moves farther away from the beam splitter, the laser beam path length will increase and a pattern of light and dark fringes will repeat uniformly. This will result in electrical signals that can be counted and converted to a distance measurement to provide an accurate position of the load. The spacing between the light and dark fringes and the resulting electrical pulse rate is determined by the wavelength of the light from the laser. For example, the wavelength of the light beam emitted by a helium–neon (He–Ne) laser, widely used in laser interferometers, is 0.63 µm, or about 0.000025 in.

Thus the accuracy of load position measurement depends primarily on the known stabilized wavelength of the laser beam. However, that accuracy can be degraded by changes in humidity and temperature as well as airborne contaminants such as smoke or dust in the air between the beam splitter and the moving retroreflector.

Precision Multiturn Potentiometers

The rotary precision multiturn potentiometer shown in the cutaway in Fig 16 is a simple, low-cost feedback instrument. Originally developed for use in analog computers, precision potentiometers can provide absolute position data in analog form as a resistance value or voltage. Precise and resettable voltages correspond to each setting of the rotary control shaft. If a potentiometer is used in a servosystem, the analog data will usually be converted to digital data by an integrated circuit analog-to-digital converter (ADC). Accuracies of 0.05% can be obtained from an instrument-quality precision multiturn potentiometer, and resolutions can exceed 0.005° if the output signal is converted with a 16-bit ADC.

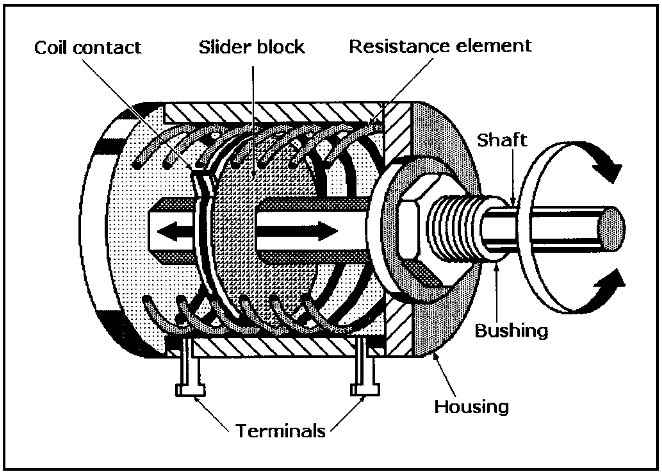


Fig. 16 A precision potentiometer is a low-cost, reliable feedback sensor for servosystems.

Precision multiturn potentiometers have wirewound or hybrid resistive elements. Hybrid elements are wirewound elements coated with resistive plastic to improve their resolution. To obtain an output from a potentiometer, a conductive wiper must be in contact with the resistive element. During its service life wear on the resistive element caused by the wiper can degrade the precision of the precision potentiometer.

SOLENOIDS AND THEIR APPLICATIONS

Solenoids: An Economical Choice for Linear or Rotary Motion

A solenoid is an electromechanical device that converts electrical energy into linear or rotary mechanical motion. All solenoids include a coil for conducting current and generating a magnetic field, an iron or steel shell or case to complete the magnetic circuit, and a plunger or armature for translating motion. Solenoids can be actuated by either direct current (DC) or rectified alternating current (AC).

Solenoids are built with conductive paths that transmit maximum magnetic flux density with minimum electrical energy input. The mechanical action performed by the solenoid depends on the design of the plunger in a linear solenoid or the armature in a rotary solenoid. Linear solenoid plungers are either spring-loaded or use external methods to restrain axial movement caused by the magnetic flux when the coil is energized and restore it to its initial position when the current is switched off.

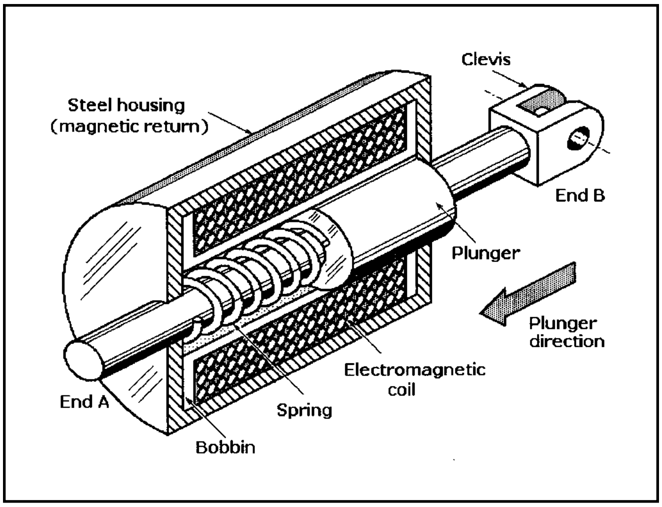


Fig. 1 The pull-in and push-out functions of a solenoid are shown. End A of the plunger pushes out when the solenoid is energized while the clevis-end B pulls in.

Cutaway drawing Fig. 1 illustrates how pull-in and push-out actions are performed by a linear solenoid. When the coil is energized, the plunger pulls in against the spring, and this motion can be translated into either a "pull-in" or a "push-out" response. All solenoids are basically pull-in-type actuators, but the location of the plunger extension with respect to the coil and spring determines its function. For example, the plunger extension on the left end (end A) provides "push-out" motion against the load, while a plunger extension on the right end terminated by a clevis (end B) provides "pull-in" motion. Commercial solenoids perform only one of these functions. Figure 2 is a cross-sectional view of a typical pull-in commercial linear solenoid.

Rotary solenoids operate on the same principle as linear solenoids except that the axial movement of the armature is converted into rotary movement by various mechanical devices. One of these is the use of internal lands or ball bearings and slots or races that convert a pull-in stroke to rotary or twisting motion.

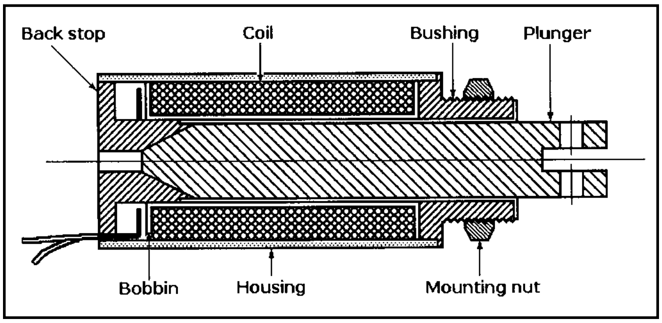


Fig. 2 Cross-section view of a commercial linear pull-type solenoid with a clevis. The conical end of the plunger increases its efficiency. The solenoid is mounted with its threaded bushing and nut.

Motion control and process automation systems use many different kinds of solenoids to provide motions ranging from simply turning an event on or off to the performance of extremely complex sequencing. When there are requirements for linear or rotary motion, solenoids should be considered because of their relatively small size and low cost when compared with alternatives such as motors or actuators. Solenoids are easy to install and use, and they are both versatile and reliable.

Technical Considerations

Important factors to consider when selecting solenoids are their rated torque/force, duty cycles, estimated working lives, performance curves, ambient temperature range, and temperature rise. The solenoid must have a magnetic return path capable of transmitting the maximum amount of magnetic flux density with minimum energy input. Magnetic flux lines are transmitted to the plunger or armature through the bobbin and air gap back through the iron or steel shell. A ferrous metal path is more efficient than air, but the air gap is needed to permit plunger or armature movement. The force or torque of a solenoid is inversely proportional to the square of the distance between pole faces. By optimizing the ferrous path area, the shape of the plunger or armature, and the magnetic circuit material, the output torque/force can be increased.

The torque/force characteristic is an important solenoid specification. In most applications the force can be a minimum at the start of the plunger or armature stroke but must increase at a rapid rate to reach the maximum value before the plunger or armature reaches the backstop.

The magnetizing force of the solenoid is proportional to the number of copper wire turns in its coil, the magnitude of the current, and the permeance of the magnetic circuit. The pull force required by the load must not be greater than the force developed by the solenoid during any portion of its required stroke, or the plunger or armature will not pull in completely. As a result, the load will not be moved the required distance.

Heat buildup in a solenoid is a function of power and the length of time the power is applied. The permissible temperature rise limits the magnitude of the input power. If constant voltage is applied, heat buildup can degrade the efficiency of the coil by effectively reducing its number of ampere turns. This, in turn, reduces flux density and torque/force output. If the temperature of the coil is permitted to rise above the temperature rating of its insulation, performance will suffer and the solenoid could fail prematurely. Ambient temperature in excess of the specified limits will limit the solenoid cooling expected by convection and conduction.

Heat can be dissipated by cooling the solenoid with forced air from a fan or blower, mounting the solenoid on a heat sink, or circulating a liquid coolant through a heat sink. Alternatively, a larger solenoid than the one actually needed could be used.

The heating of the solenoid is affected by the duty cycle, which is specified from 10 to 100%, and is directly proportional to solenoid *on* time. The highest starting and ending torque are obtained with the lowest duty cycle and *on* time. Duty cycle is defined as the ratio of *on* time to the sum of *on* time and *off* time. For example, if a solenoid is energized for 30 s and then turned off for 90 s, its duty cycle is $^{30}/_{120} = ^1/_4$, or 25%.

The amount of work performed by a solenoid is directly related to its size. A large solenoid can develop more force at a given stroke than a small one with the same coil current because it has more turns of wire in its coil.

Open-frame Solenoids

Open-frame solenoids are the simplest and least expensive models. They have open steel frames, exposed coils, and movable plungers centered in their coils. Their simple design permits them to be made inexpensively in high-volume production runs so that they can be sold at low cost. The two forms of open-frame solenoid are the *C-frame solenoid* and the *box-frame solenoid*. They are usually specified for applications where very long life and precise positioning are not critical requirements.

C-Frame Solenoids

C-frame solenoids are low-cost commercial solenoids intended for light-duty applications. The frames are typically laminated steel formed in the shape of the letter C to complete the magnetic circuit through the core, but they leave the coil windings without a complete protective cover. The plungers are typically made as laminated steel bars. However, the coils are usually potted to resist airborne and liquid contaminants. These solenoids can be found in appliances, printers, coin dispensers, security door locks, cameras, and vending machines. They can be powered with either AC or DC current. Nevertheless, C-frame solenoids can have operational lives of millions of cycles, and some standard catalog models are capable of strokes up to 0.5 in. (13 mm).

Box-Frame Solenoids

Box-frame solenoids have steel frames that enclose their coils on two sides, improving their mechanical strength. The coils are wound on phenolic bobbins, and the plungers are typically made from solid bar stock. The frames of some box-type solenoids are made from stacks of thin insulated sheets of steel to control eddy currents as well as keep stray circulating currents confined in solenoids powered by AC. Box-frame solenoids are specified for higher-end applications such as tape decks, industrial controls, tape recorders, and business machines because they offer mechanical and electrical performance that is superior to those of C-frame solenoids. Standard catalog commercial box-frame solenoids can be powered by AC or DC current, and can have strokes that exceed 0.5 in. (13 mm).

Tubular Solenoids

The coils of *tubular solenoids* have coils that are completely enclosed in cylindrical metal cases that provide improved magnetic circuit return and better protection against accidental damage or liquid spillage. These DC solenoids offer the highest volumetric efficiency of any commercial solenoids, and they are specified for industrial and military/aerospace equipment where the space permitted for their installation is restricted. These solenoids are specified for printers, computer disk-and tape drives, and military weapons systems; both pull-in and push-out styles are available. Some commercial tubular linear solenoids in this class have strokes up to 1.5 in. (38 mm), and some can provide 30 lbf (14 kgf) from a unit less than 2.25 in (57 mm) long. Linear solenoids find applications in vending machines, photocopy machines, door locks, pumps, coin-changing mechanisms, and film processors.

Rotary Solenoids

Rotary solenoid operation is based on the same electromagnetic principles as linear solenoids except that their input electrical energy is converted to rotary or twisting rather than linear motion. Rotary actuators should be considered if controlled speed is a requirement in a rotary stroke application. One style of rotary solenoid is shown in the exploded view Fig. 3. It includes an armature-plate assembly that rotates when it is pulled into the housing by magnetic flux from the coil. Axial stroke is the linear distance that the armature travels to the center of the coil as the solenoid is energized. The three ball bearings travel to the lower ends of the races in which they are positioned.

The operation of this rotary solenoid is shown in Fig. 4. The rotary solenoid armature is supported by three ball bearings that travel around and down the three inclined ball races. The de-

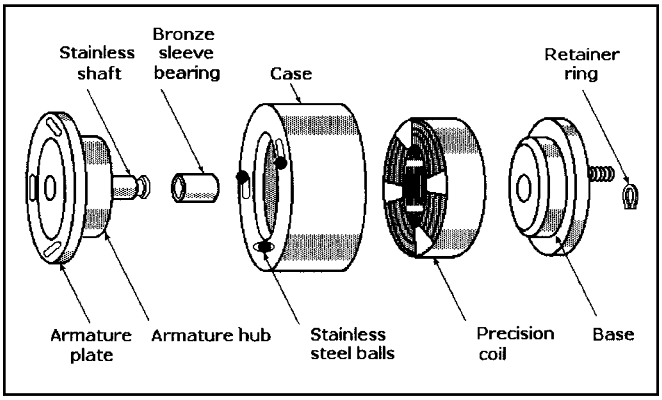


Fig. 3 Exploded view of a rotary solenoid showing its principal components.

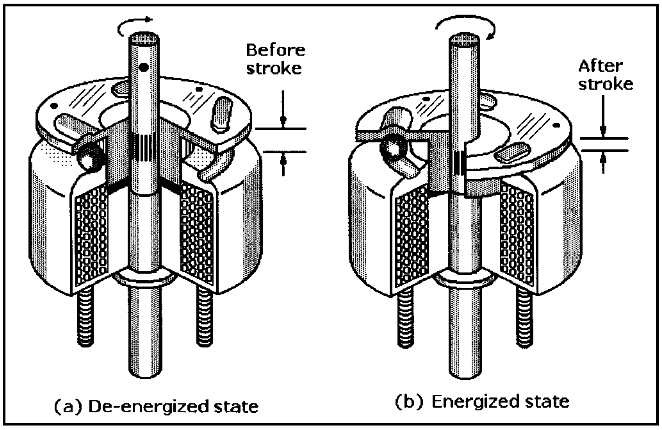


Fig. 4 Cutaway views of a rotary solenoid de-energized (a) and energized (b). When energized, the solenoid armature pulls in, causing the three ball bearings to roll into the deeper ends of the lateral slots on the faceplate, translating linear to rotary motion.

energized state is shown in (a). When power is applied, a linear electromagnetic force pulls in the armature and twists the armature plate, as shown in (b). Rotation continues until the balls have traveled to the deep ends of the races, completing the conversion of linear to rotary motion.

This type of rotary solenoid has a steel case that surrounds and protects the coil, and the coil is wound so that the maximum amount of copper wire is located in the allowed space. The steel housing provides the high permeability path and low residual flux needed for the efficient conversion of electrical energy to mechanical motion.

Rotary solenoids can provide well over 100 lb-in. (115 kgf-cm) of torque from a unit less than 2.25 in. (57 mm) long. Rotary solenoids are found in counters, circuit breakers, electronic component pick-and-place machines, ATM machines, machine tools, ticket-dispensing machines, and photocopiers.

Rotary Actuators

The rotary actuator shown in Fig. 5 operates on the principle of attraction and repulsion of opposite and like magnetic poles as a motor. In this case the electromagnetic flux from the actuator's solenoid interacts with the permanent magnetic field of a neodymium–iron disk magnet attached to the armature but free to rotate.

The patented Ultimag rotary actuator from the Ledex product group of TRW, Vandalia, Ohio, was developed to meet the need for a bidirectional actuator with a limited working stroke of less than 360° but capable of offering higher speed and torque than a rotary solenoid. This fast, short-stroke actuator is finding applications in industrial, office automation, and medical equipment as well as automotive applications

The PM armature has twice as many poles (magnetized sectors) as the stator. When the actuator is not energized, as shown in (a), the armature poles each share half of a stator pole, causing the shaft to seek and hold mid-stroke.

When power is applied to the stator coil, as shown in (b), its associated poles are polarized north above the PM disk and south beneath it. The resulting flux interaction attracts half of the armature's PM poles while repelling the other half. This causes the shaft to rotate in the direction shown.

When the stator voltage is reversed, its poles are reversed so that the north pole is above the PM disk and south pole is below it. Consequently, the opposite poles of the actuator armature are attracted and repelled, causing the armature to reverse its direction of rotation.

According to the manufacturer, Ultimag rotary actuators are rated for speeds over 100 Hz and peak torques over 100 oz-in. Typical actuators offer a 45° stroke, but the design permits a maximum stroke of 160°. These actuators can be operated in an *on/off* mode or proportionally, and they can be operated either open- or closed-loop. Gears, belts, and pulleys can amplify the stroke, but this results in reducing actuator torque.

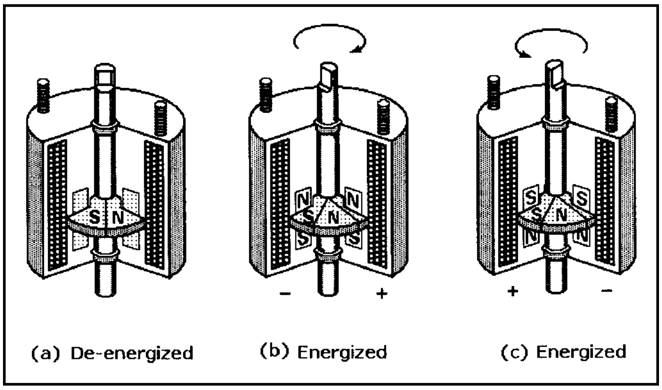
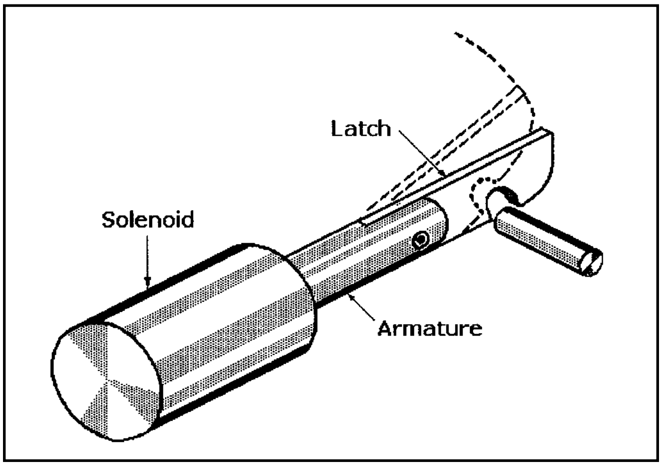
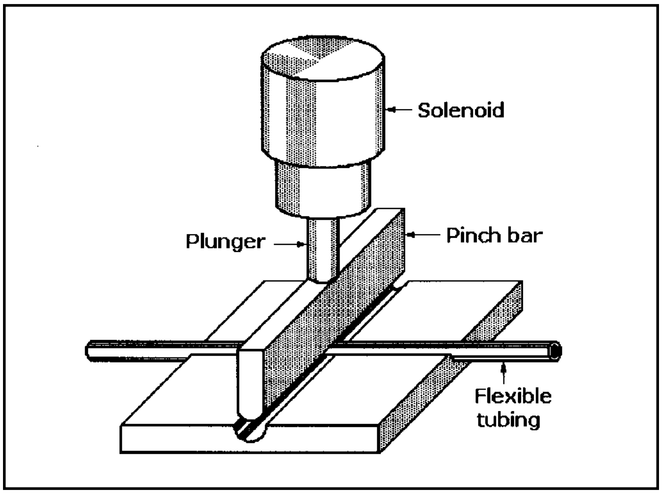


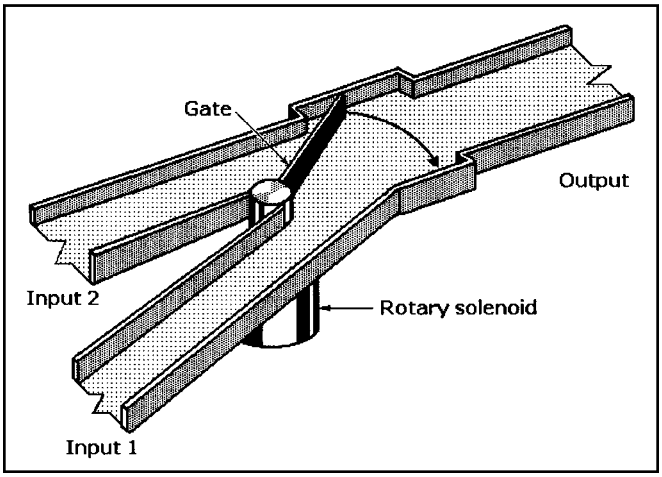
Fig. 5 This bidirectional rotary actuator has a permanent magnet disk mounted on its armature that interacts with the solenoid poles. When the solenoid is deenergized (a), the armature seeks and holds a neutral position, but when the solenoid is energized, the armature rotates in the direction shown. If the input voltage is reversed, armature rotation is reversed (c).



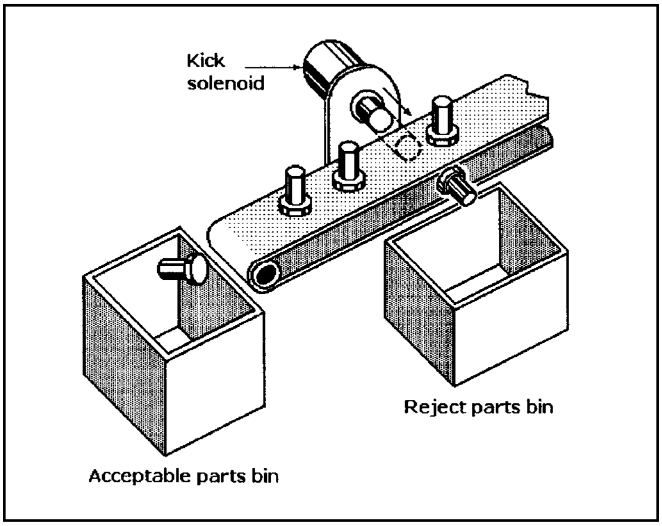
Latching: Linear solenoid push-out or pull-in motion can be used in a wide variety of latching applications such as locking vault doors, safe deposit boxes, secure files, computers, and machine tools, depending on how the movable latch is designed.



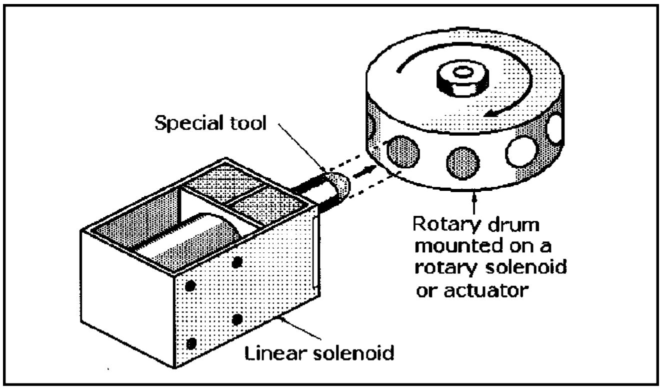
Pinchoff of Flexible Tubing: This push-out linear solenoid with an attached blade can control or pinch off liquid flowing in flexible tubing when energized by a remote operator. This arrangement can eliminate valves or other devices that could leak or admit contaminants. It can be used in medical, chemical, and scientific laboratories where fluid flow must be accurately regulated.



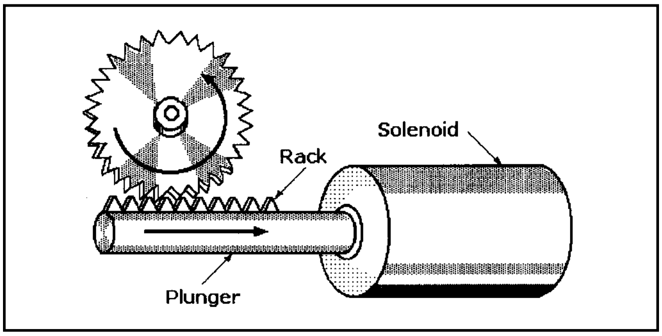
Parts or Material Diversion: This diverter arrangement consists of a rotary solenoid with a gate attached to its armature. The gate can swing to either of two alternate positions under pushbutton or automatic control to regulate the flow of parts or materials moving on belts or by gravity feed.



Parts Rejection: A push-out linear solenoid can rapidly expel or reject parts that are moving past it into a bin when triggered. An electronic video or proximity sensing system is required to energize the solenoid at the right time.



Rotary Positioning: A linear push-out solenoid is paired with a multi-station drum containing objects that are indexed by a linear solenoid or actuator. This arrangement would permit the automatic assembly of parts to those objects or the application of adhesives to them as the drum is indexed.



Ratcheting Mechanism: A pull-in solenoid with a rack mounted on its plunger becomes a ratcheting mechanism capable of turning a gear for the precise positioning of objects under operator or automated control.

CHAPTER 2
ROBOT MECHANISMS

The programmability of the industrial robot using computer software makes it both flexible in the way it works and versatile in the range of tasks it can accomplish. The most generally accepted definition of a *robot* is a reprogrammable, multifunction manipulator designed to move material, parts, tools, or specialized devices through variable programmed motions to perform a variety of tasks. Robots can be floor-standing, benchtop, or mobile.

Robots are classified in ways that relate to the characteristics of their control systems, manipulator or arm geometry, and modes of operation. There is no common agreement on or standardizations of these designations in the literature or among robot specialists around the world.

A basic robot classification relates to overall performance and distinguishes between limited and unlimited sequence control. Four classes are generally recognized: limited sequence and three forms of unlimited sequence—point-to-point, continuous path, and controlled path. These designations refer to the path taken by the end effector, or tool, at the end of the robot arm as it moves between operations.

Another classification related to control is *nonservoed* versus *servoed*. Nonservoed implies open-loop control, or no closed-loop feedback, in the system. By contrast, servoed means that some form of closed-loop feedback is used in the system, typically based on sensing velocity, position, or both. Limited sequence also implies nonservoed control while unlimited sequence can be achieved with point-to-point, continuous-path, or controlled-path modes of operation.

Robots are powered by electric, hydraulic, or pneumatic motors or actuators. Electric motor power is most popular for the major axes of floor-standing industrial robots today. Hydraulic-drive robots are generally assigned to heavy-duty lifting applications. Some electric and hydraulic robots are equipped with pneumatic-controlled tools or end effectors.

The number of degrees of freedom is equal to the number of axes of a robot, and is an important indicator of its capability. Limited-sequence robots typically have only two or three degrees of freedom, but point-to-point, continuous-path, and controlled-path robots typically have five or six. Two or three of those may be in the wrist or end effector.

Most heavy-duty industrial robots are floor-standing. Figure 1 shows a typical floor-standing robot system whose principal axes are powered by responsive electric motors. Others in the same size range are powered by hydraulic motors. The console contains a digital computer that has been programmed with an operating system and applications software so that it can perform the tasks assigned to it. Some robot systems also include training pendants—handheld pushbutton panels connected by cable to the console that permit direct control of the robot.

The operator or programmer can control the movements of the robot arm or manipulator with pushbuttons or other data input devices so that it is run manually through its complete task

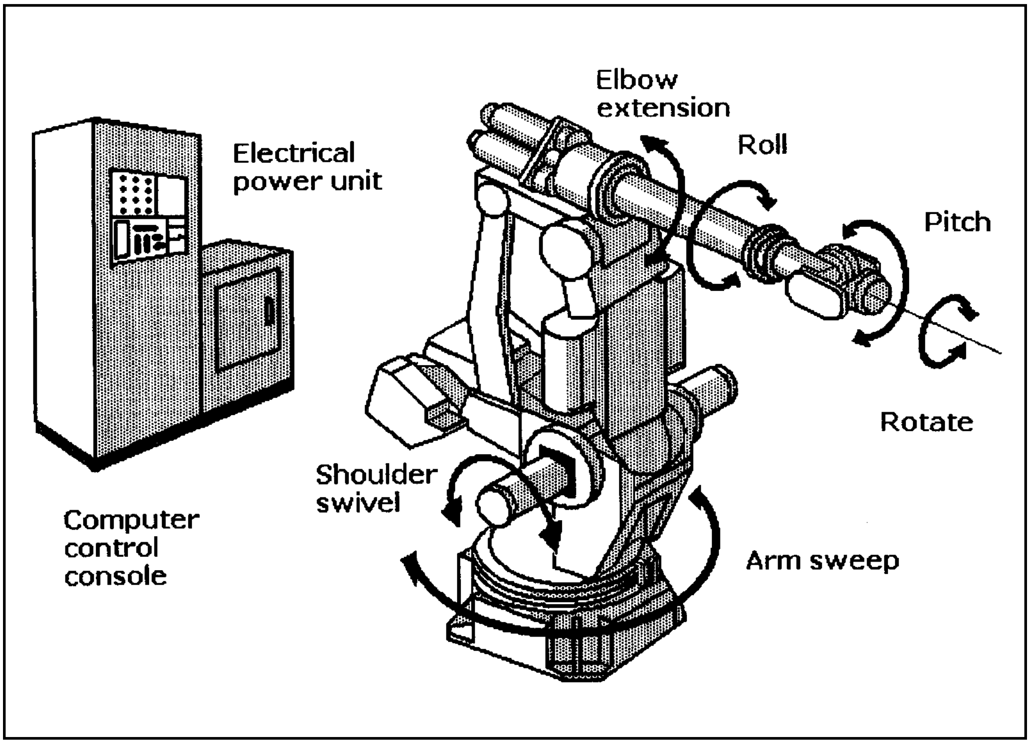


Fig. 1 Components of a floor-standing, six-degree-of-freedom industrial robot. The principal axes are driven by servo-controlled electric motors. The digital computer and remote-control pendant are located in the computer control console.

sequence to program it. At this time adjustments can be made to prevent any part of the robot from colliding with nearby objects.

There are also many different kinds of light-duty assembly or pick-and-place robots that can be located on a bench. Some of these are programmed with electromechanical relays, and others are programmed by setting mechanical stops on pneumatic motors.

Robot versus Telecheric

The true robot should be distinguished from the manually controlled manipulator or *telecheric*, which is remotely controlled by human operators and not programmed to operate automatically and unattended. These machines are mistakenly called robots because some look like robots or are equipped with similar components. Telecherics are usually controlled from a remote location by signals sent over cable or radio link.

Typical examples of telecherics are manually controlled manipulators used in laboratories for assembling products that contain radioactive materials or for mixing or analyzing radioactive materials. The operator is shielded from radiation or hazardous fumes by protective walls, airlocks, special windows, or a combination of these. Closed-circuit television permits the operator to view the workplace so that precise or sensitive work can be performed. Telecherics are also fitted to deep-diving submersibles or extraterrestrial landing platforms for gathering specimens in hostile or inaccessible environments.

Telecherics can be mobile machines equipped with tanklike treads that can propel it over rough terrain and with an arm that can move in three or more degrees of freedom. Depending on its mission, this kind of vehicle can be equipped with handlike grippers or other specialized tools for performing various tasks in environments where hazardous materials have been spilled or where fires are burning. Other missions might include bomb disposal, firefighting, or gathering information on armed criminals or persons trapped in confined spaces following earthquakes or explosions. Again, a TV camera gives the operator information for guidance.

Robot Advantages

The industrial robot can be programmed to perform a wider range of tasks than dedicated automatic machines, even those that can accept a wide selection of different tools. However, the full benefits of a robot can be realized only if it is properly integrated with the other machines human operators, and processes. It must be evaluated in terms of cost-effectiveness of the performance or arduous, repetitious, or dangerous tasks, particularly in hostile environments. These might include high temperatures, high humidity, the presence of noxious or toxic fumes, and proximity to molten metals, welding arcs, flames, or high-voltage sources.

The modern industrial robot is the product of developments made in many different engineering and scientific disciplines, with an emphasis on mechanical, electrical, and electronic technology as well as computer science. Other technical specialties that have contributed to robot development include servomechanisms, hydraulics, and machine design. The latest and most advanced industrial robots include dedicated digital computers.

The largest number of robots in the world are limited-sequence machines, but the trend has been toward the electric-motor powered, servo-controlled robots that typically are floor-standing machines. Those robots have proved to be the most cost-effective because they are the most versatile.

Trends in Robots

There is evidence that the worldwide demand for robots has yet to reach the numbers predicted by industrial experts and visionaries some ten years ago. The early industrial robots were expensive and temperamental, and they required a lot of maintenance. Moreover, the software was frequently inadequate for the assigned tasks, and many robots were ill-suited to the tasks assigned them.

Many early industrial customers in the 1970s and 1980s were disappointed because their expectations had been unrealistic; they had underestimated the costs involved in operator training, the preparation of applications software, and the integration of the robots with other machines and processes in the workplace.

By the late 1980s, the decline in orders for robots drove most American companies producing them to go out of business, leaving only a few small, generally unrecognized manufacturers. Such industrial giants as General Motors, Cincinnati Milacron, General Electric, International Business Machines, and Westinghouse entered and left the field. However, the Japanese electrical equipment manufacturer Fanuc Robotics North America and the Swedish-Swiss corporation Asea Brown Boveri (ABB) remain active in the U.S. robotics market today.

However, sales are now booming for less expensive robots that are stronger, faster, and smarter than their predecessors. Industrial robots are now spot-welding car bodies, installing windshields, and doing spray painting on automobile assembly lines. They also place and remove parts from annealing furnaces and punch presses, and they assemble and test electrical and mechanical products. Benchtop robots pick and place electronic components on circuit boards in electronics plants, while mobile robots on tracks store and retrieve merchandise in warehouses.

The dire predictions that robots would replace workers in record numbers have never been realized. It turns out that the most cost-effective robots are those that have replaced human beings in dangerous, monotonous, or strenuous tasks that humans do not want to do. These activities frequently take place in spaces that are poorly ventilated, poorly lighted, or filled with noxious or toxic fumes. They might also take place in areas with high relative humidity or temperatures that are either excessively hot or cold. Such places would include mines, foundries, chemical processing plants, or paint-spray facilities.

Management in factories where robots were purchased and installed for the first time gave many reasons why they did this despite the disappointments of the past ten years. The most frequent reasons were the decreasing cost of powerful computers as well as the simplification of both the controls and methods for programming the computers. This has been due, in large measure, to the declining costs of more powerful microprocessors, solid-state and disk memory, and applications software.

However, overall system costs have not declined, and there have been no significant changes in the mechanical design of industrial robots during the industrial robot's ten-year "learning curve" and maturation period.

The shakeout of American robot manufacturers has led to the near domination of the world market for robots by the Japanese manufacturers who have been in the market for most of the past ten years. However, this has led to de facto standardization in robot geometry and philosophy along the lines established by the Japanese manufacturers. Nevertheless, robots are still available in the same configurations that were available five to ten years ago, and there have been few changes in the design of the end-use tools that mount on the robot's "hand" for the performance of specific tasks (e.g., parts handling, welding, painting).

Robot Characteristics

Load-handling capability is one of the most important factors in a robot purchasing decision. Some can now handle payloads of as much as 200 pounds. However, most applications do not require the handling of parts that are as heavy as 200 pounds. High on the list of other requirements are "stiffness"—the ability of the robot to perform the task without flexing or shifting; accuracy—the ability to perform repetitive tasks without deviating from the programmed dimensional tolerances; and high rates of acceleration and deceleration.

The size of the manipulator or arm influences accessibility to the assigned floor space. Movement is a key consideration in choosing a robot. The robot must be able to reach all the parts or tools needed for its application. Thus the robot's working range or envelope is a critical factor in determining robot size.

Most versatile robots are capable of moving in at least five degrees of freedom, which means they have five axes. Although most tasks suitable for robots today can be performed by robots with at least five axes, robots with six axes (or degrees of freedom) are quite common. Rotary base movement and both radial and vertical arm movement are universal. Rotary wrist movement and wrist bend are also widely available. These movements have been designated as roll and pitch by some robot manufacturers. Wrist yaw is another available degree of freedom.

More degrees of freedom or axes can be added externally by installing parts-handling equipment or mounting the robot on tracks or rails so that it can move from place to place. To be most effective, all axes should be servo-driven and controlled by the robot's computer system.

Principal Robot Categories

There are four principal geometries for robot manipulators: (1) articulated, revolute, or jointed-arm (Figs. 2 and 3); (2) polar coordinate (Fig. 4); (3) Cartesian (Fig. 5); and (4) cylindrical (Fig. 6). However, there are many variations possible on these basic designs, including vertically jointed (Fig. 7), horizontally jointed, and gantry or overhead-configured.

The robot "wrist" is mounted on the end of the robot's arm and serves as a tool holder. It can also provide additional axes or degrees of freedom, which is particularly desirable when the end effector, such as welding electrodes or a paint spray gun, must be maneuvered within confined spaces. Three common forms of end effector are illustrated in Figs. 8, 9, and 10.

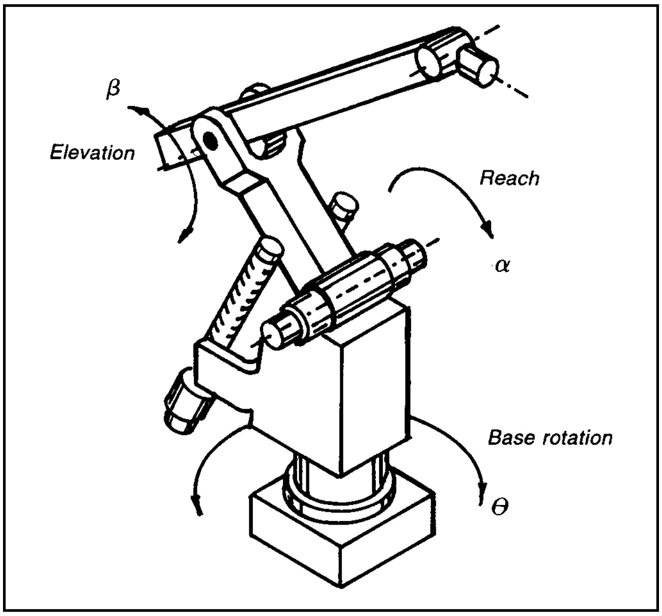


Fig. 2 A low-shoulder, articulated, revolute, or jointed-geometry robot has a base or waist, an upper arm extending from the shoulder to the elbow, and a forearm extending from the elbow to the wrist. This robot can rotate at the waist, and both upper and lower arms can move independently through angles in the vertical plane. The angle of rotation is θ (theta), the angle of elevation is β (beta), and the angle of forearm movement is α (alpha).

There are many different kind of end effectors, but among the most common are hand-like grippers that can pick up, move, and release objects. Some are general purpose, but others are specially machined to fit around specific objects. Crude in comparison with a human hand, the grippers must be able to pick up an object and hold it securely without damaging or dropping it. Three of the most common designs are illustrated in Figs. 11, 12, and 13.

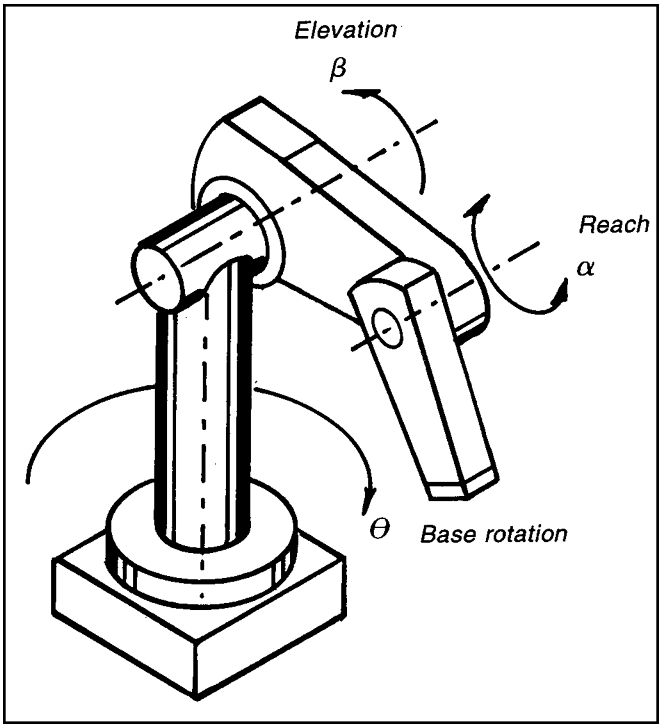


Fig. 3 A high-shoulder articulated, revolute, or jointed-geometry robot has a base or waist, an upper arm extending from the shoulder to the elbow, and a forearm extending from the elbow to the wrist. This robot can also rotate at the waist, and both upper and lower arms can move independently through angles in the vertical plane. As in Fig. 2, the angle of rotation is θ (theta), the angle of elevation is β (beta), and the angle of forearm movement is α (alpha).

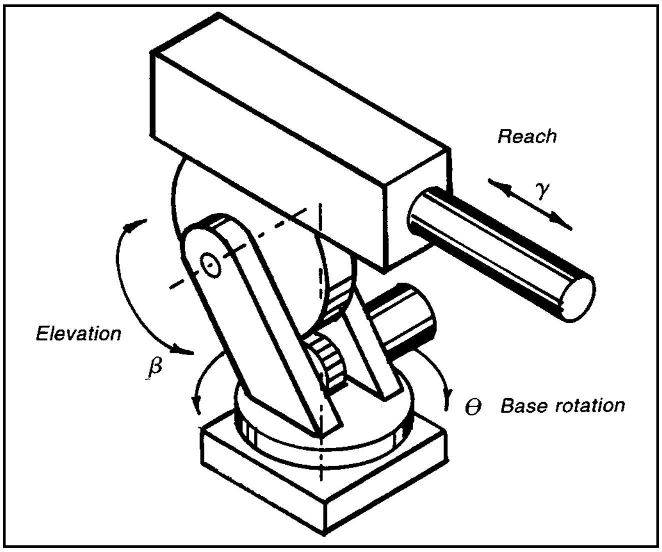


Fig. 4 A polar coordinate or gun-turret-geometry robot has a main body or waist that rotates while the arm can move in elevation like a gun barrel. The arm is also able to extend or reach. The angle of rotation in this robot is θ (theta), the angle of elevation is β (beta), and the reciprocal motion of the arm is γ (gamma).

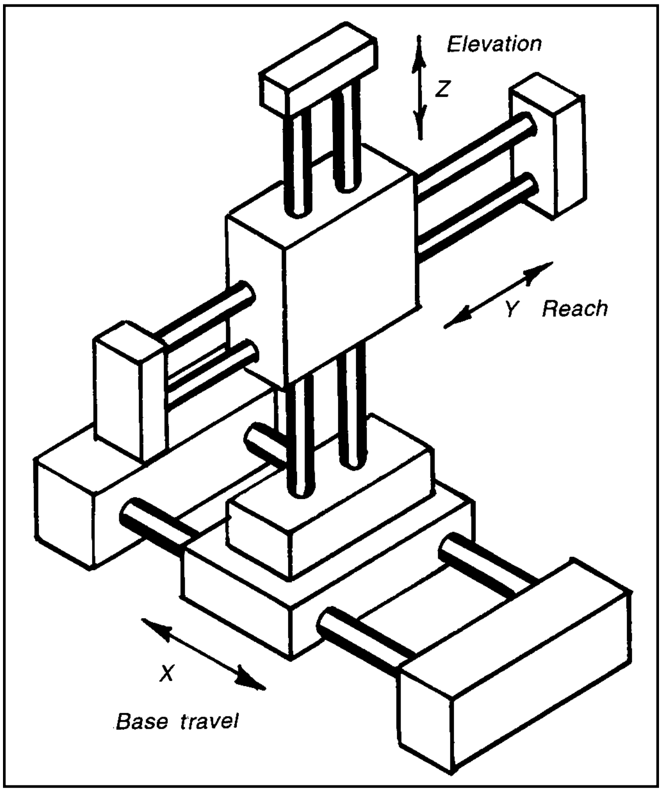


Fig. 5 The Cartesian-coordinate-geometry robot has three linear axes, X, Y, and Z. A moving arm mounted on a vertical post moves along a linear track. The base or X axis is usually the longest; the vertical axis is the Z axis; and the horizontal axis, mounted on the vertical posts, is the Y axis. This geometry is effective for high-speed, low-weight robots.

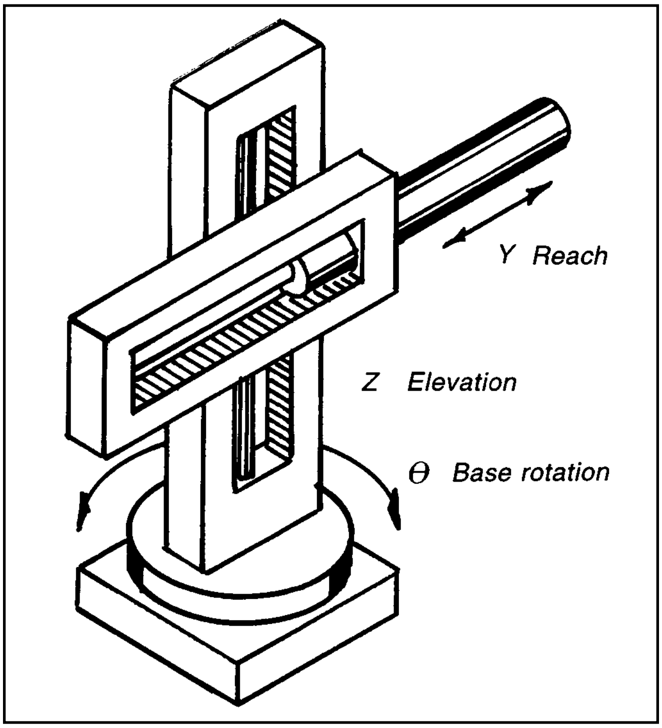


Fig. 6 The cylindrical-coordinate-geometry robot can have the same geometry as the Cartesian-coordinate robot (Fig. 5) except that its forearm is free to rotate. Alternatively, it can have a rotating waist like the polar-coordinate robot (Fig. 4) or the revolute-coordinate-geometry robot (Figs. 2 and 3). The Z axis defines vertical movement of the arm, and the Y axis defines traverse motion. Again, the angle of rotation is defined by θ (theta).

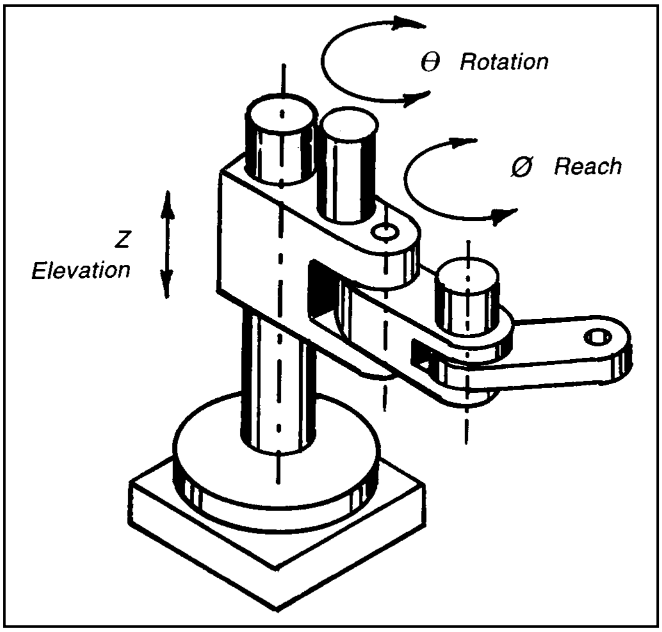


Fig. 7 A vertically-jointed robot is similar to an articulated robot, except that the mechanism is turned on its side, and the axes of rotation are vertical. The mechanism is then mounted on a vertical post or linear side, as shown. In another variation, the horizontally jointed robot, the mechanism is turned so that the slide is horizontal.

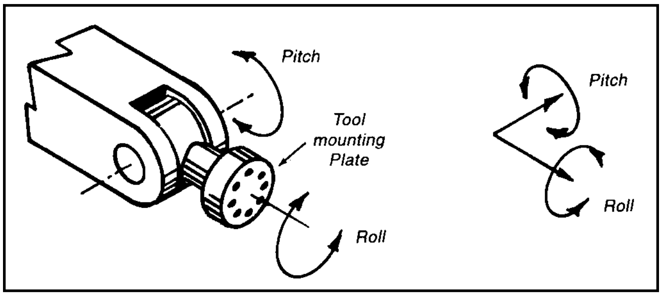


Fig. 8 A two-degree-of-freedom robot wrist can move a tool on its mounting plate around both pitch and roll axes.

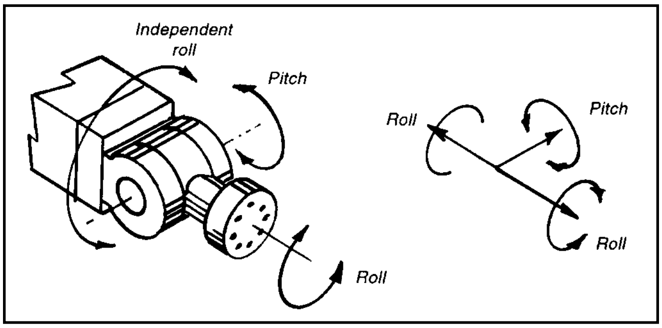


Fig. 9 This two-degree-of-freedom robot wrist can move a tool on its mounting plate around the pitch and two independent roll axes.

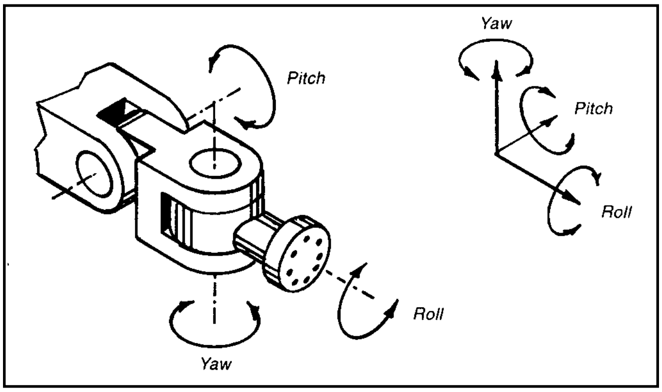


Fig. 10 A three-degree-of-freedom robot wrist can move a tool on its mounting plate around the pitch, roll, and yaw axes.

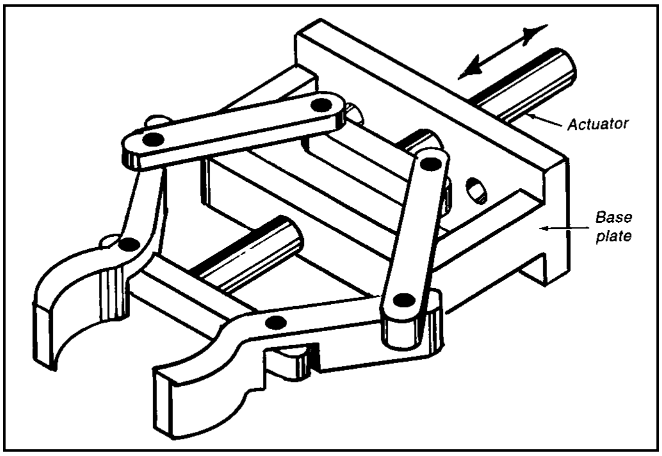


Fig. 11 A reciprocating lever mechanism opens and closes the jaws of this robot gripper, permitting it to grasp and release objects.

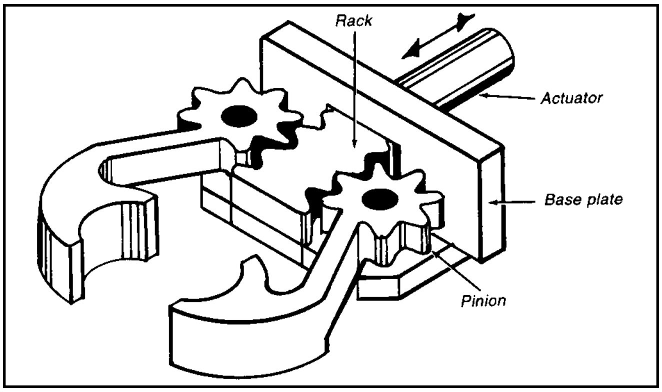


Fig. 12 A rack and pinion mechanism opens and closes the jaws of this robot gripper, permitting it to grasp and release objects.

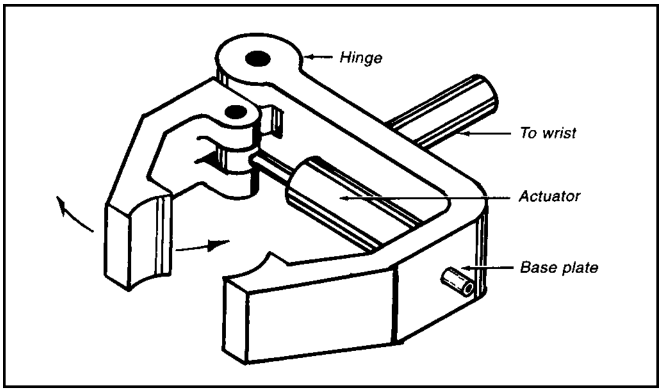


Fig. 13 A hydraulic or pneumatic piston opens and closes the jaws of this robot gripper, permitting it to grasp and release objects.

FANUC ROBOT SPECIFICATIONS

The data sheets for three robots from FANUC Robotics North America, Inc., Rochester Hills, Michigan, have been reproduced on the following pages to illustrate the range of capabilities of industrial robots now in production. These specifications include the manufacturer's ratings for the key characteristics: motion range and speed, wrist load moments and inertias, repeatability, reach, payload, and weight.

S-900*i*H/*i*L/*i*W Robots

There are three robots in the S-900*i* family: S-900*i*H, S-900*i*L, and S-900*i*W. They are floor-standing, 6-axis, heavy-duty robots with reaches of between 8 and 10 ft, (2.5 and 3.0 m) and maximum payloads of 441 to 880 lb (200 to 400 kg). S-900*i* robots can perform such tasks as materials handling and removal, loading and unloading machines, heavy-duty spot welding, and participation in casting operations.

These high-speed robots are controlled by FANUC R-J3 controllers, which provide point-to-point positioning and smooth controlled motion. S-900*i* robots have high-inertia wrists with large allowable moments that make them suitable for heavy-duty work in harsh environments. Their slim J3 outer arms and wrist profiles permit these robots to work in restricted space, and their small footprints and small i-size controllers conserve factory floor space. Many attachment points are provided on their wrists for process-specific tools, and axes J5 and J6 have precision gear drives. All process and application cables are routed through the arm, and there are brakes on all axes.

S-900*i* robots support standard I/O networks and have standard Ethernet ports. Process-specific software packages are available for various applications. Options include B-size controller cabinets, additional protection for harsh environments, a precision baseplate for quick robot exchanges, and integrated auxiliary axes packages.

S-500 Robot

The S-500 is a 6-axis robot with a reach of 9 ft (2.7 m) and a load capacity of 33 lb (15 kg). Equipped with high-speed electric servo-drives, the S-500 can perform a wide range of manufacturing and processing tasks such as materials handling, loading and unloading machines, welding, waterjet cutting, dispensing, and parts transfer.

The S-500 can be mounted upright, inverted, or on walls without modification, and it can operate in harsh uninhabited locations as well as on populated factory floors. Absolute serial encoders eliminate the need for calibration at power-up. Repeatability is ±0.010 in. (±0.25 mm), and axes 3 to 6 can reach speeds of 320°/s.

Features for increasing reliability include mechanical brakes on all axes and grease fittings on all lubrication points for quick and easy maintenance. RV speed reducers provide smooth motion at all speeds. Bearings and drives are sealed for protection, and cables are routed through hollow joints to eliminate snagging. Brushless AC servo motors minimize motor maintenance.

An optional drive for axis 6 is capable of speeds up to 600°/s. Other options include a 3.5-in. floppy-disk drive for storing data off-line and a printer for printing out data and programs. Also available are an RS-232C communication port and integrated auxiliary axes.

LR Mate 100*i* Robot

The LR Mate 100*i* is a 5-axis benchtop robot suitable for performing a wide range of tasks in environments ranging from clean rooms to harsh industrial sites. It has a nominal payload capacity of 6.6 to 8.8 lb (3 to 4 kg) and a 24.4-in. (620 mm) reach. Its payload can be increased to 11 lb with a shorter reach of 23.6 in. (600 mm) This modular electric servo-driven robot can perform such tasks as machine loading and unloading, materials handling and removal, testing and sampling, assembly, welding, dispensing, and parts cleaning.

The 100*i* robot can be mounted upright or inverted without modification, and its small footprint allows it to be mounted on machine tools. Repeatability is ±0.002 in. (±0.04 mm), and the axis 5 speed can reach 272°/s. Two integral double solenoid valves and the end effector connector are in the wrist. It is able to "double back" on itself for increased access, and axes 2 and 3 have fail-safe brakes. Standard software permits 3D palletizing and depalletizing of rows, columns, and layers simply by teaching the robot three points.

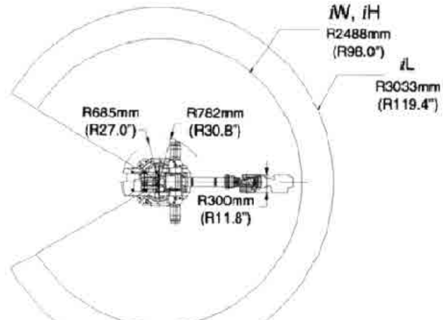
The FANUC R-J2 Mate *i*-Controller is easy to install, start up, troubleshoot, and maintain. The controller weighs approximately 110 lb (50 kg) and is housed in a small case measuring 14.9 in. wide by 18.5 in. high and 12.6 in. deep (380 × 470 × 320 mm). Its low-voltage I/O has 20 inputs (8 dedicated), 16 outputs (4 dedicated), and 4 inputs at the end-of-arm connector.

Reliability is increased and maintenance is reduced with brushless AC servo motors and harmonic drives on all axes. Only two types of motors are used to simplify servicing and reduce spare parts requirements. Bearings and drives are sealed for protection against harsh factory environments. There are grease fittings on all lubrication points for quick and easy maintenance, and easily removable service panels give fast access to the robot's drive train. A standard IP65 dust and liquid intrusion package is included.

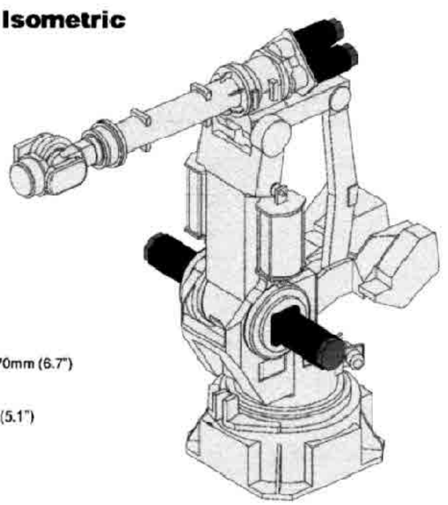
As options for the LR Mate 100*i*, Class 100 cleanroom and high-speed versions are offered. The cleanroom version can serve in biomedical research labs and high-precision production and testing facilities. A high-speed version with an axis 5 speed of 480°/s and a payload of 6.6 lb (3 kg) is available. Other options include additional integral valve packages, brakes for axis 1, and a higher-speed CPU to speed up path and cycle times. FANUC's Sensor Interface serial communications software allows the robot to exchange data with third-party equipment such as bar code readers, vision systems, and personal computers, while its Data Transfer Function serial communications software allows two-way data exchange between the robot and a PC. This permits the robot to be controlled through a VB graphical interface.

S-900 *i*H/*i*L/*i*W Dimensions

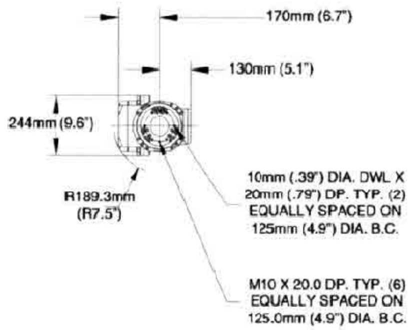
Top



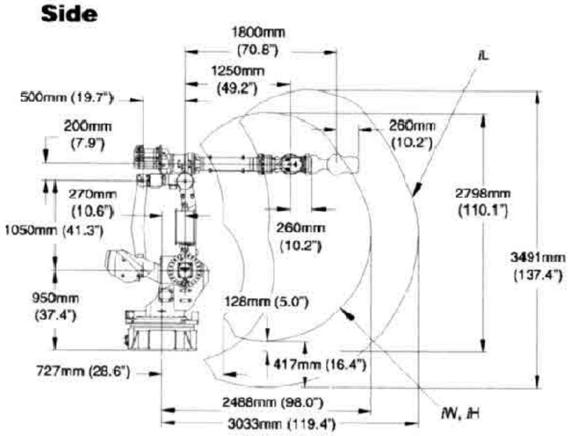
Isometric



Wrist ISO Flange



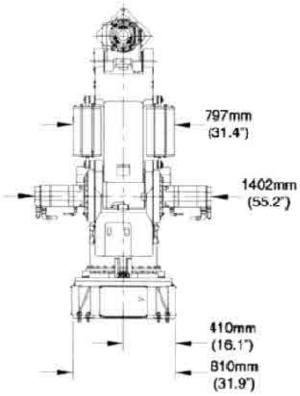
Side



S-900 *i*H/*i*L/*i*W Specifications

Items		S-900*i*H		S-900*i*L		S-900*i*W	
Motion range and speed	Axis 1	300°	120°/sec	300°	95°/sec	300°	95°/sec
	Axis 2	115°	120°/sec	115°	95°/sec	115°	95°/sec
	Axis 3	145°	125°/sec	145°	95°/sec	145°	95°/sec
	Axis 4	720°	115°/sec	720°	110°/sec	720°	100°/sec
	Axis 5	250°	115°/sec	250°	110°/sec	250°	100°/sec
	Axis 6	720°	200°/sec	720°	165°/sec	720°	160°/sec
Wrist load moments	Axis 4	120kg • m		130kg • m		140kg • m	
	Axis 5	120kg • m		130kg • m		140kg • m	
	Axis 6	65kg • m		70kg • m		70kg • m	
Wrist load inertias	Axis 4	1200kg • cm • sec^2		1200kg • cm • sec^2		1200kg • cm • sec^2	
	Axis 5	1200kg • cm • sec^2		1200kg • cm • sec^2		1200kg • cm • sec^2	
	Axis 6	300kg • cm • sec^2		600kg • cm • sec^2		600kg • cm • sec^2	
Reach		2488		3033		2488	
Max payload at J6		200kg (441lbs)		220kg (484 lbs)		400kg (880 lbs)	
Mounting method		Floor		Floor		Floor	
Failsafe mech brakes		All axes		All axes		All axes	
Mech unit weight		1920kg (4230 lbs)		2030kg (4466 lbs)		1995kg (4395 lbs)	

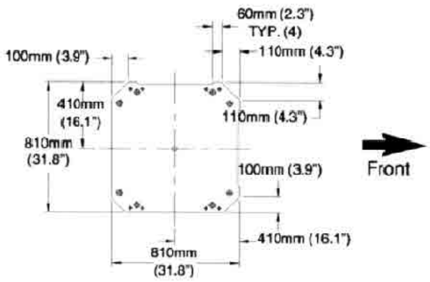
Front



FANUC Robotics

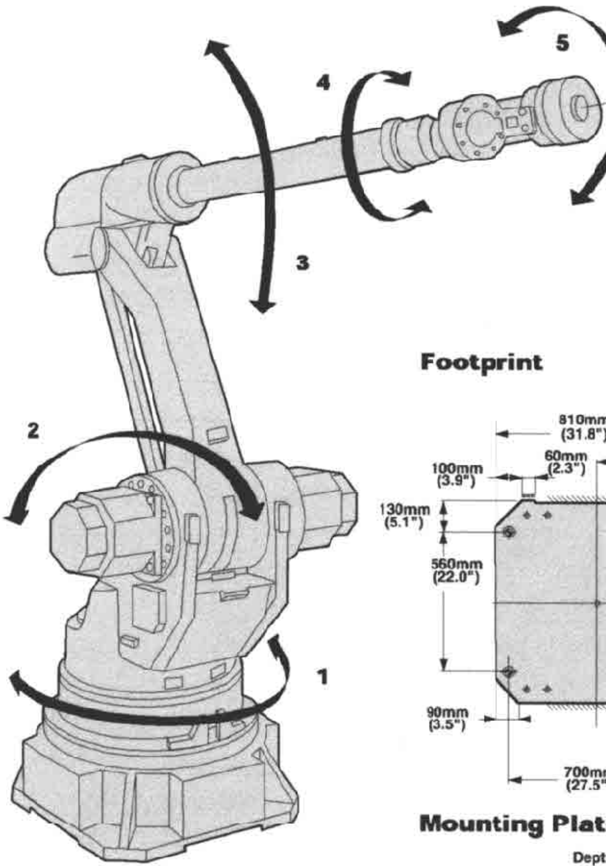
A New Age of Industry

Footprint

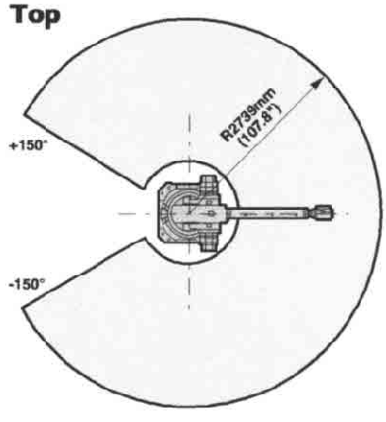


FANUC Robotics North America
3900 W. Hamlin Road
Rochester Hills, MI 48309-3253

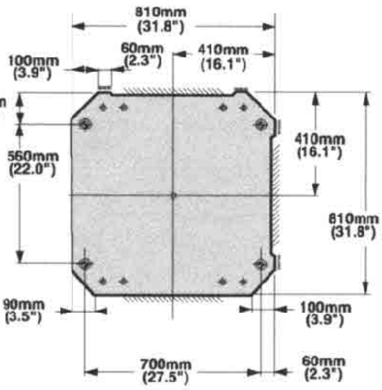
S-500 Dimensions



Top



Footprint



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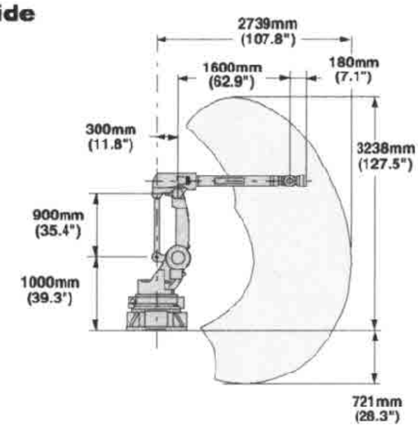
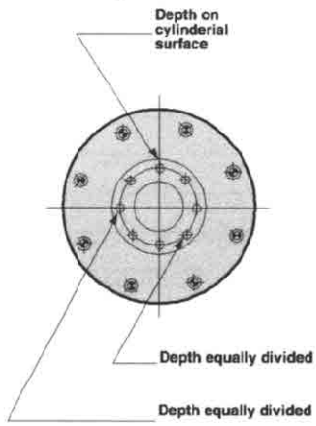
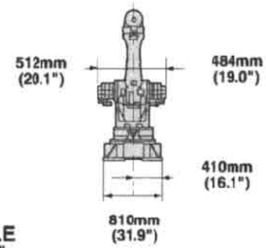


Mounting Plate



Front



SCALE
1/100"

Items			
		Range	Speed
Motion range and speed	Axis 1	300°	90°/sec.
	Axis 2	160°	90°/sec.
	Axis 3	160°	320°/sec.
	Axis 4	480°	320°/sec.
	Axis 5	240°	320°/sec.
	Axis 6	900°	320°/sec.
	Axis 1	2.0 kgf • cm	
	Axis 2	3.0 kgf • cm	
	Axis 3	3.0 kgf • cm	
Load	Axis 1	2.2 kgf • cm • sec^2	
	Axis 2	6.2 kgf • cm • sec^2	
	Axis 3	6.2 kgf • cm • sec^2	
		±0.25mm(±0.010")	
Max. load capacity		15kg (33 lbs)	
Mounting method			
Mechanical brakes		All axes	
Mechanical weight		900 kg (1985 lbs)	

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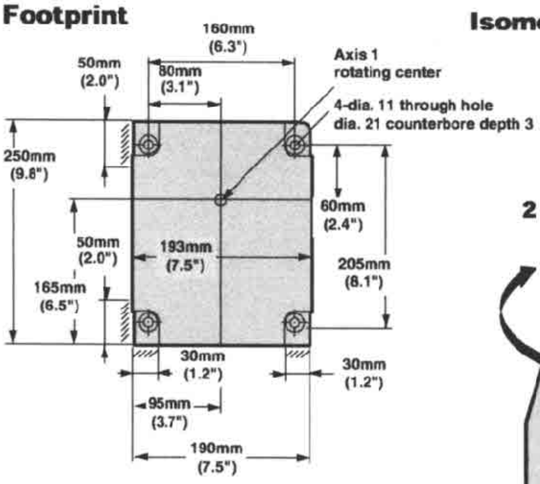
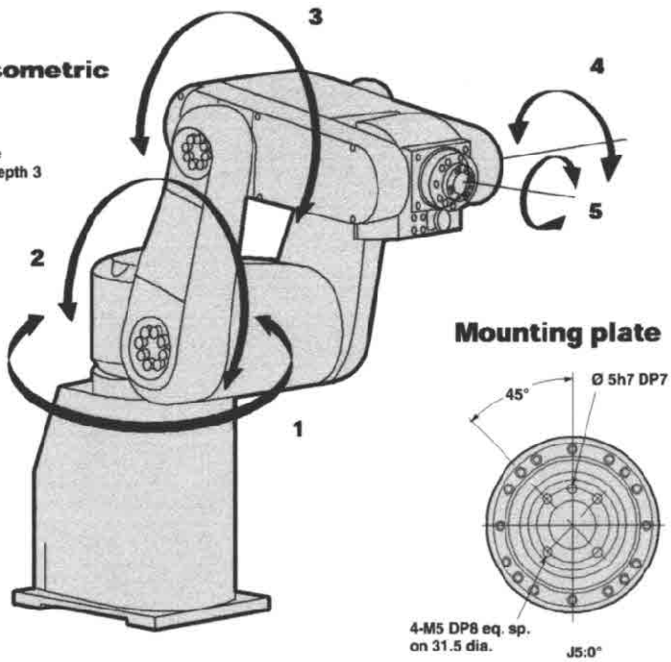
LR Mate 100*i* Dimensions

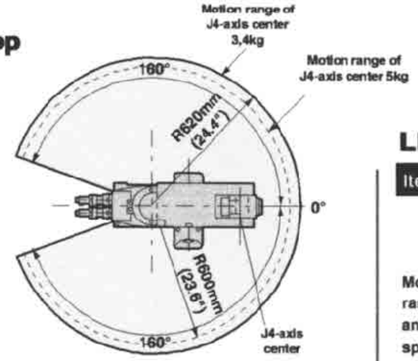
Footprint



Isometric



Top



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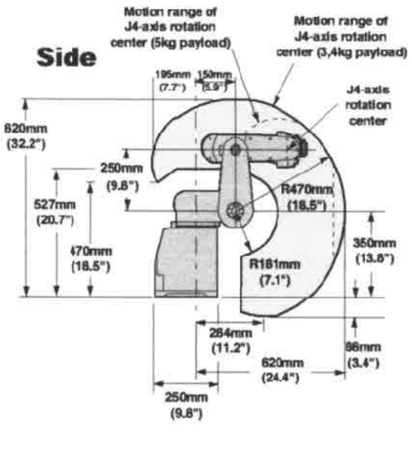
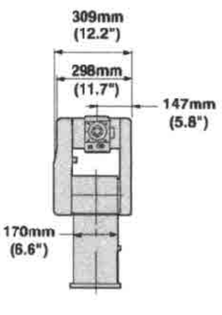


Front



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LR Mate 100*i* Specifications

Items		3kg (6.6 lbs) payload			4kg (8.8 lbs) payload		5kg (11 lbs) payload	
		Standard & Cleanroom Versions		High Speed Version	Standard & Cleanroom Versions		Standard & Cleanroom Versions	
		Range	Speed	Speed	Range	Speed	Range	Speed
Motion range and speed	Axis 1	320°	180°/sec	240°/sec	320°	150°/sec	Same as 4kg (note 1)	
	Axis 2	185°	180°/sec	285°/sec	185°	150°/sec		
	Axis 3	365°	225°/sec	360°/sec	365°	180°/sec		
	Axis 4	240°	216°/sec	330°/sec	240°	100°/sec		
	Axis 5	400°	272°/sec	480°/sec	400°	250°/sec		
Moment	Axis 4	55.5 kgf • cm			74.0 kgf • cm		50.0 kgf • cm	
	Axis 5	40.0 kgf • cm			40.0 kgf • cm		4.4 kgf • cm (note 2)	
Load inertia	Axis 4	1.1 kgf • cm • s^2			1.4 kgf • cm • s^2		2.25 kgf • cm • s^2	
	Axis 5	0.41 kgf • cm • s^2			0.41 kgf • cm • s^2		0.51 kgf • cm • s^2	
Repeatability		±0.04mm (±0.002") based on JISB8432						
Mounting method		Upright/inverted						
Mechanical brakes		Axis 2, axis 3 (axis 1 option)						
Mechanical weight		32kg (70.5 lbs)						
Dust/water intrusion protection		Conforms to the IP65 standard for dust and liquid intrusion protection (seals may need periodic replacement if used with chlorine or gasoline based coolants)						
Notes		(1) In the case of 5kg wrist payload, the motion area and the wrist direction are limited (2) In the case of 5kg wrist payload, the wrist direction is limited to downward						

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MECHANISM FOR PLANAR MANIPULATION WITH SIMPLIFIED KINEMATICS

Simple combinations of actuator motions yield purely radial or purely tangential end-effector motions.

Goddard Space Flight Center, Greenbelt, Maryland

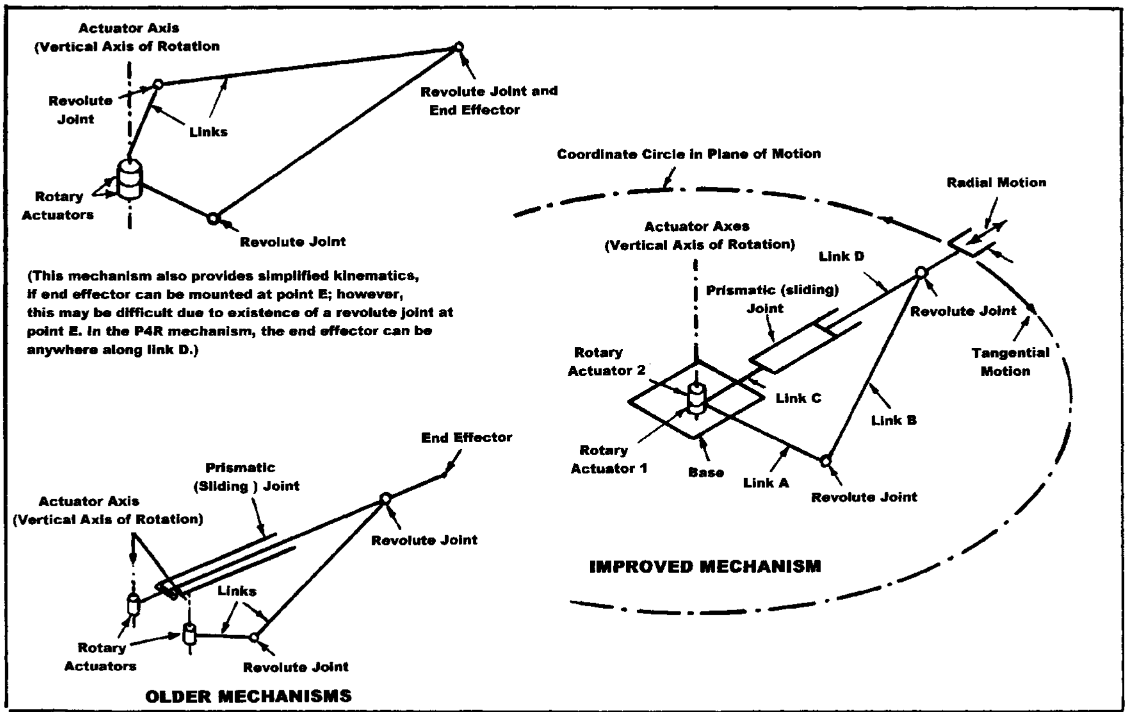
The figure schematically illustrates three manipulator mechanisms for positioning an end effector (a robot hand or other object) in a plane (which would ordinarily be horizontal). One of these is a newer, improved mechanism that includes two coaxial, base-mounted rotary actuators incorporated into a linkage that is classified as “P4R” in the discipline of kinematics of mechanisms because it includes one prismatic (P) joint and four revolute (R) joints. The improved mechanism combines the advantages of coaxial base mounting (as opposed to noncoaxial and/or nonbase mounting) of actuators, plus the advantages of closed-loop (as opposed to open-loop) linkages in such a way as to afford a simplification (in comparison with other linkages) of inverse kinematics. Simplification of the kinematics reduces the computational burden incurred in controlling the manipulator.

In the general case of a two-degree-of-freedom manipulator with two rotary actuators, the inverse kinematic problem is to find the rotary-actuator angles needed to place the end effector at a specified location, velocity, and acceleration in the plane of motion. In the case of a typical older manipulator mechanism of this type, the solution of the inverse kinematic problem involves much computation because what one seeks is the coordinated positions, velocities, and accelerations of the two manipulators, and these coordinates are kinematically related to each other and to the required motion in a complex way.

In the improved mechanism, the task of coordination is greatly simplified by simplification of the inverse kinematics; the motion of the end effector is easily resolved into a component that is radial and a component that is tangential to a circle that runs through the end effector and is concentric with the rotary actuators.

If rotary actuator 2 is held stationary, while rotary actuator 1 is turned, then link D slides radially in the prismatic joint, causing the end effector to move radially. If both rotary actuators are turned together, then there is no radial motion; instead, the entire linkage simply rotates as a rigid body about the actuator axis, so that the end effector moves tangentially. Thus, the task of coordination is reduced to a simple decision to (a) rotate actuator 1 only to obtain radial motion, (b) rotate both actuators together to obtain tangential motion, or (c) rotate the actuators differentially according to a straightforward kinematic relationship to obtain a combination of radial and axial motion.

This work was done by Farhad Tahmasebi of **Goddard Space Flight Center**.



The Improved Mechanism affords a simplification of kinematics: Whereas the coordination of actuator motions necessary to obtain specified end-effector motions in the older mechanisms is a complex task, it is a relatively simple task in the improved mechanism.

TOOL-CHANGING MECHANISM FOR ROBOT

A tool is handed off securely between an end effector and a holster.

Goddard Space flight Center, Greenbelt, Maryland

Figure 1 is a partially exploded view of a tool-changing mechanism for robotic applications. The mechanism effects secure handoff of the tool between the end effector of the robot and a yoke in which the tool is stowed when not in use. The mechanism can be operated in any orientation in normal or low gravitation. Unlike some other robotic tool-changing mechanisms, this one imposes fewer constraints on the design of the robot and on the tool because it is relatively compact. Moreover, it does not require the large insertion forces and the large actuators that would be needed to produce them. Also, it can be stored in zero g and can survive launch loads.

A tool interface assembly is affixed to each tool and contains part of the tool-changing mechanism. The tool is stowed by (1) approximately aligning the tips of the yoke arms with flared openings of the holster guides on the tool interface assembly, (2) sliding the assembly onto the yoke arms, which automatically enforce fine alignment because of the geometric relationship between the mating surfaces of the yoke-arm wheels and the holster guide, (3) locking the assembly on the holster by pushing wing segments of a captured nut (this is described more fully later) into chamfered notches in the yoke arms, and (4) releasing the end effector from the tool interface assembly.

The end effector includes a male splined shaft (not shown in Fig. 1) that is spring-loaded to protrude downward. A motor rotates the male splined shaft via a splined drive shaft that mates with a splined bore in the shank of the male splined shaft. The sequence of movements in which the end effector takes the tool from the holster begins with the movement of the end effector into a position in which its alignment recesses can engage the mating blocks on the tool interface assembly. The end effector is then pushed downward into contact with the tool interface assembly. Meanwhile, the male splined shaft is rotated until the spring force can push it through the opening in the splined female end of a driven bolt, and an alignment cone at the end of the splined male shaft bottoms in a conical hole in the female end of the driven bolt (see Fig. 2)

Assuming that the thread on the driven bolt is right handed, the male splined shaft is rotated clockwise until a vertical spline on this shaft engages a tab in the driven bolt. At that location the shaft and bolt rotate together. As the rotation continues, the driven bolt moves downward in a captive nut until the mating splined surfaces on the male splined shaft and driven bolt make contact. This prevents further downward movement of the driven bolt.

As the rotation continues, the captive nut moves upward. The wing segments mentioned previously are then pulled up, out of the chamfered slots on the yoke

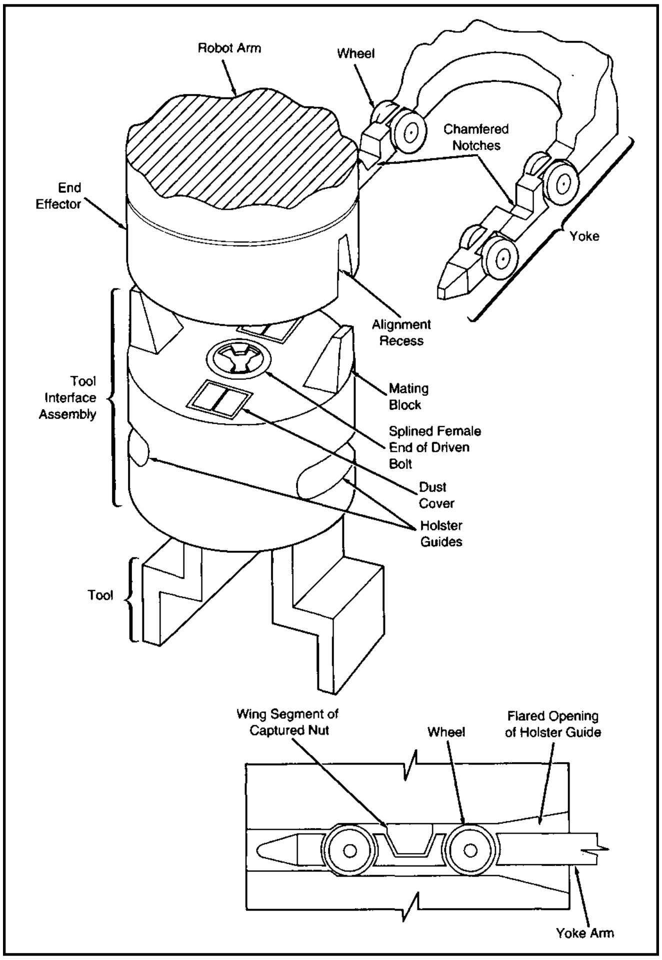


Fig. 1 This tool-changing mechanism operates with relatively small contact forces and is relatively compact.

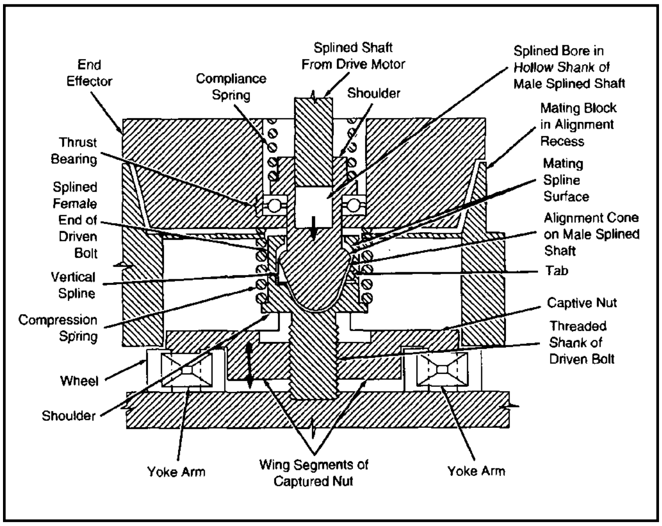


Fig. 2 This end effector and tool interface assembly is shown in its initial mating configuration, immediately before the beginning of the sequence of motions that release the tool from the yoke and secure it to the end effector.

arms, so that the tool interface plate can then be slid freely off of the yoke. Simultaneously, two other wing segments of the captured nut (not shown) push up sets of electrical connectors, through the dust covers, to mate with electrical connectors in the end effector. Once this motion is completed, the tool is fully engaged with the end effector and can be slid off the yoke. To release the tool from the end effector and lock it on the yoke (steps 3 and 4 in the second paragraph), this sequence of motions is simply reversed.

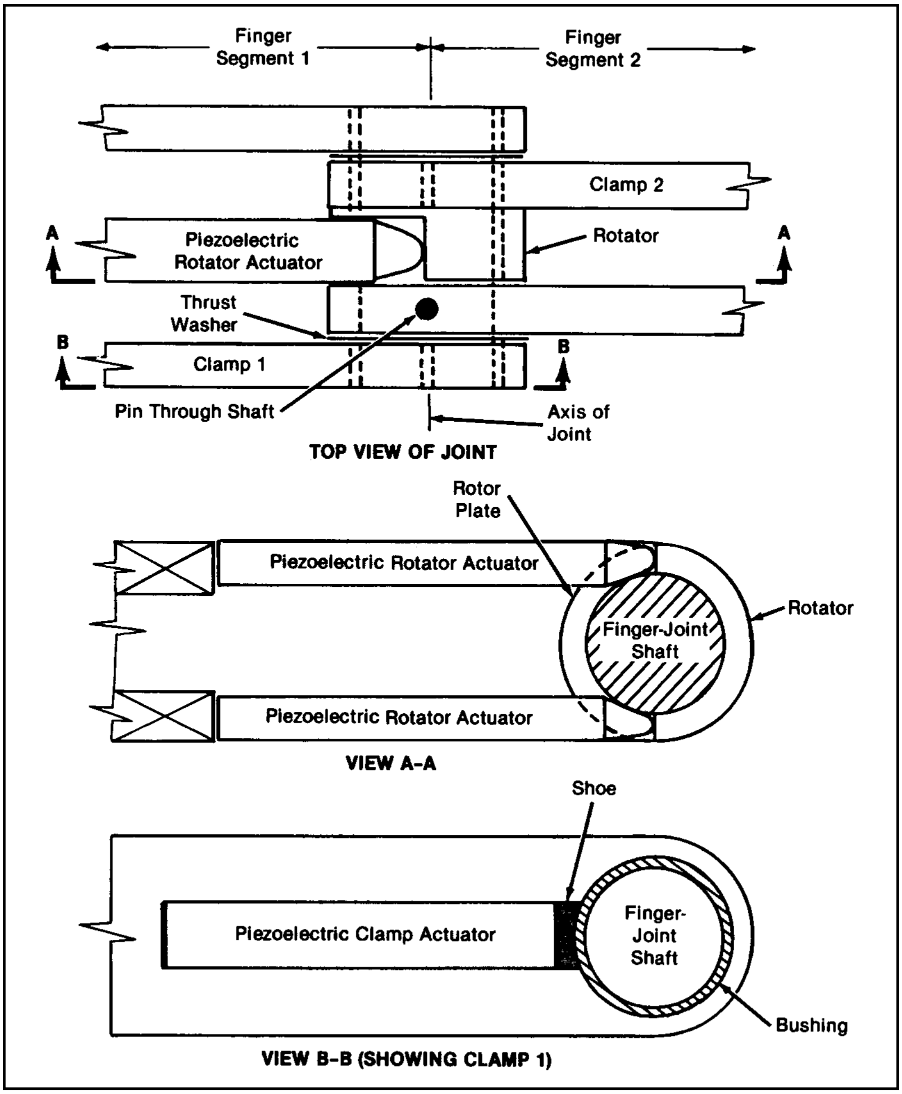
This work was done by John M. Vranish of **Goddard Space Flight Center**.

PIEZOELECTRIC MOTOR IN ROBOT FINGER JOINT

A direct drive unit replaces a remote electromagnetic motor.
Marshall Space Flight Center, Alabama

A robotic finger contains an integral piezoelectric motor. In comparison with a robotic finger actuated by remote motors via tendonlike cables, this robotic finger is simpler and can therefore be assembled, disassembled, and repaired more easily. It is also more reliable and contains more internal space that can be allocated for additional sensors and control circuitry.

The finger (see figure) includes two piezoelectric clamps and a piezoelectric-rotator subassembly. Each clamp is composed of a piezoelectric actuator, a concave shoe, and a thin bushing with an axial slit. A finger-joint shaft fits in the bushing. When the actuator in a clamp is de-energized, the shaft is free to rotate in the bushing. When the same actuator is energized, it expands and pushes the shoe against the bushing. This action clamps the shaft. (The slit in the bushing allows it to flex so that more actuator force acts on the shaft and is not wasted in deforming the bushing.)



Each piezoelectric clamp grasps a shaft when energized. The piezoelectric rotor turns the shaft in small increments as it is alternately clamped and unclamped.

The piezoelectric-rotor subassembly includes a pair of piezoelectric actuators and a component simply called the rotator, which is attached to the bushing in clamp 2. The upper rotator actuator, when energized, pushes the rotator a fraction of a degree clockwise. Similarly, when the lower rotator is energized, it pushes the rotator a fraction of a degree counterclockwise. The finger-joint shaft extends through the rotator. The two clamps are also mounted on the same shaft, on opposite sides of the rotator. The rotator actuators are energized alternately to impart a small back-and-forth motion to the rotator. At the same time, the clamp actuators are energized alternately in such a sequence that the small oscillations of therotator accumulate into a net motion of the shaft (and the finger segment attached to it), clockwise or counterclockwise, depending on whether the shaft is clamped during clockwise or counterclockwise movement of the rotator.

The piezoelectric motor, including lead wires, rotator-actuator supports, and actuator retainers, ads a mass of less than 10 grams to the joint. The power density of the piezoelectric motor is much grater than that of the electromagnetic motor that would be needed to effect similar motion. The piezoelectric motor operates at low speed and high torque—characteristics that are especially suitable for robots.

This work was done by Allen R Grahn of Bonneville Scientific, Inc., for **Marshall Space Flight Center**.

SIX-DEGREE-OF-FREEDOM PARALLEL MINIMANIPULATOR

Advantages include greater stiffness and relative simplicity.

Goddard Space Flight Center, *Greenbelt*, *Maryland*

Figure 1 illustrates schematically a six-degree-of-freedom manipulator that produces small, precise motions and that includes only three inextensible limbs with universal joints at their ends. The limbs have equal lengths and can be said to act in parallel in that they share the load on a manipulated platform. The mechanism is therefore called a "six-degree-of-freedom parallel minimanipulator." The minimanipulator is designed to provide high resolution and high stiffness (relative to the other mechanisms) for fine control of position and force in a hybrid form of serial/parallel-manipulator system.

Most of the six-degree-of-freedom parallel manipulators that have been proposed in the past contain six limbs, and their direct kinematic analyses are very complicated. In contrast, the equations of the direct kinematics of the present minimanipulator can be solved in closed form. Furthermore, in comparison with a typical six-degree-of-freedom parallel manipulator, the present minimanipulator can be made of fewer parts, the probability of mechanical interference between its limbs is smaller, its payload capacity can be made greater, and its actuators, which are base-mounted, can be made smaller.

The upper ends of the limbs are connected to the manipulated platform by universal (two-degree-of-freedom) joints. The lower end of each limb is connected via a universal (two-degree-of-freedom) rotary joint to a two-degree-of-freedom driver. The drivers are mounted directly on the baseplate, without any intervening power-transmission devices, like gears or belts, that could reduce stiffness and precision.

The position and orientation of the manipulated platform is governed uniquely, in all six degrees of freedom, by the positions of the drivers on the baseplate. Examples of two-degree-of-freedom drivers include bi-directional linear stepping motors, *x-y* positioning tables, five-bar linkages driven by rotary actuators, and pantographs. Figure 2 shows an example of a baseplate equipped with pantograph drivers. The position of each universal joint C_i (where i = 1, 2, or 3) is controlled by moving either or both of sliders A_i and B_i in their respective guide slots. The displacement reduction provided by the pantograph linkage and the inextensible limbs is equivalent to an increase in mechanical advantage; it increases the stiffness and resolution available at the manipulated platform.

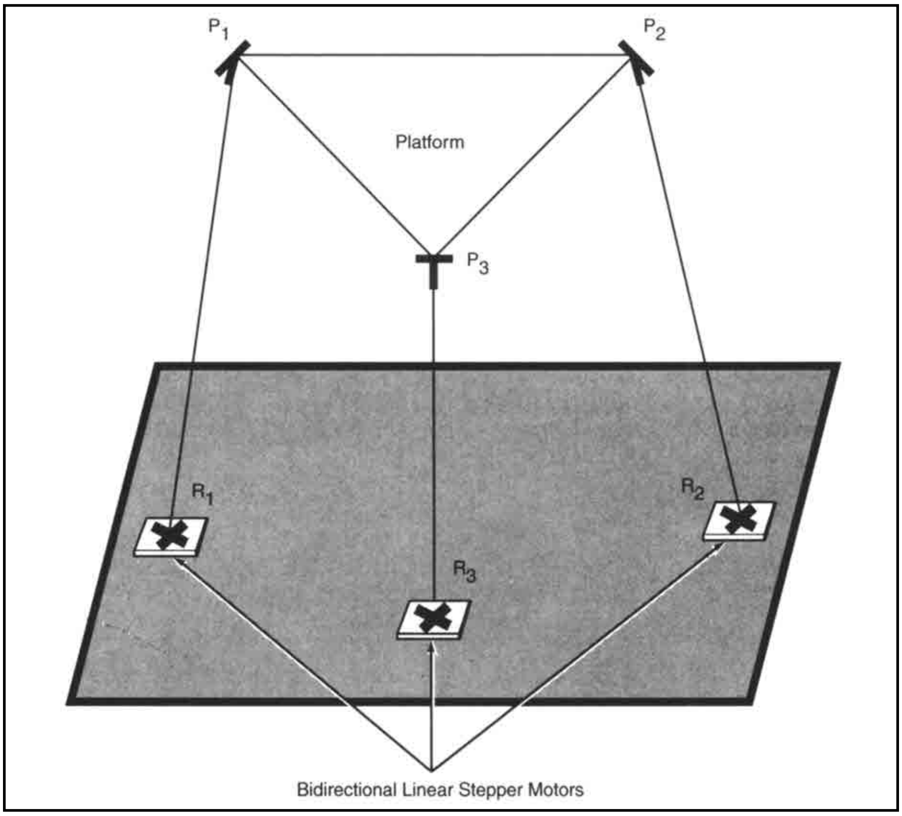


Fig. 1 The **six-degree-of-freedom parallel minimanipulator** is stiffer and simpler than earlier six-degree-of-freedom manipulators, partly because it includes only three inextensible limbs.

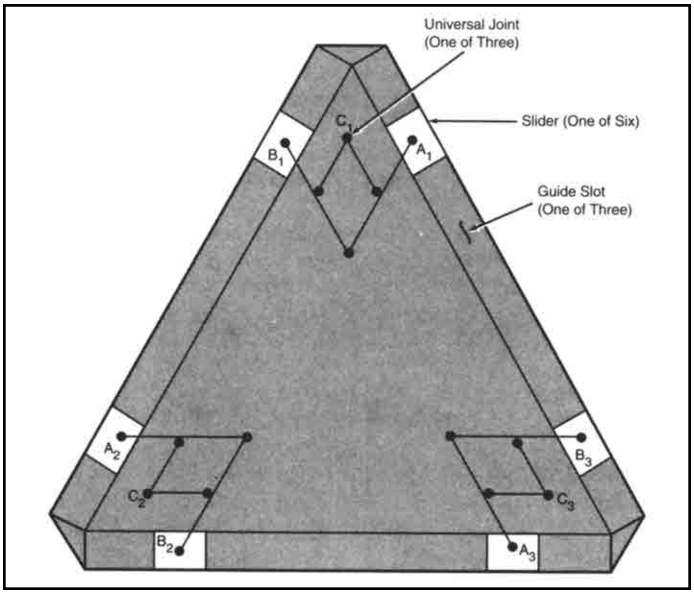
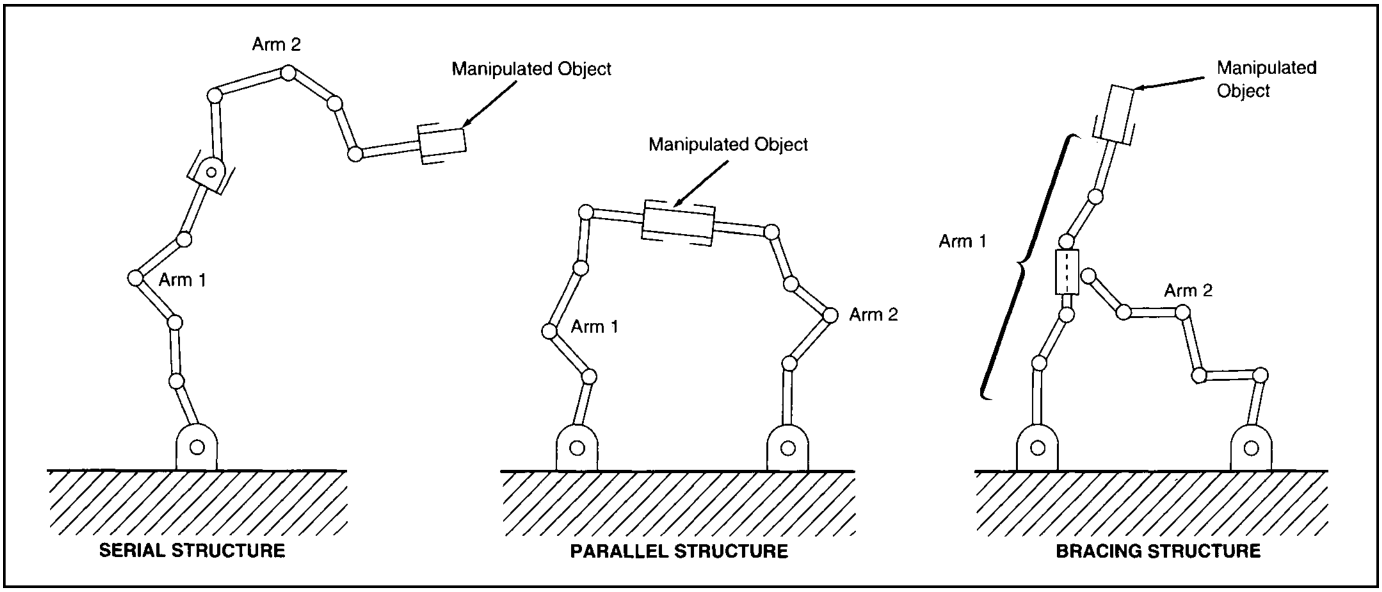


Fig. 2 Three pantographs on the baseplate control the positions of the universal joints at C_1 and thereby control the position and orientation of the manipulated platform.

This work was done by Farhad Tahmasebi and Lung-Web Tsai of **Goddard Space Flight Center**.

SELF-RECONFIGURABLE, TWO-ARM MANIPULATOR WITH BRACING

Structure can be altered dynamically to suit changing tasks.
NASA's Jet Propulsion Laboratory, Pasadena, California



Alternative structures of cooperating manipulator arms can be selected to suit changing tasks.

A proposed two-arm robotic manipulator would be capable of changing its mechanical structure to fit a given task. Heretofore, the structures of reconfigurable robots have been changed by replacement and/or reassembly of modular links. In the proposed manipulator, there would be no reassembly or replacement in the conventional sense: instead, the arms would be commanded during operation to assume any of a number of alternative configurations.

The configurations (see figure) are generally classified as follows: (1) serial structure, in which the base of arm 1 is stationary, the tip of arm 1 holds the base of arm 2, and the tip of arm 2 holds the manipulated object; (2) parallel structure, in which the bases of both arms are stationary and the tips of both arms make contact with the manipulated object at two different points; and (3) the bracing structure, in which the basis of both arms are stationary and the tip of arm 2 grasps some intermediate point along the length of arm 1. the serial and parallel structures can be regarded as special cases of the bracing structure. Optionally, each configuration could involve locking one or more joints of either or both arms, and the bracing contact between the two arms could be at a fixed position of arm 1 or else allowed to slide along a link of arm 1.

The performances of the various configurations can be quantified in terms of quantities called "dual-arm manipulabilities," and "dual-arm resistivities." Dual-arm manipulabilities are defined on the basis of kinematic and dynamic constraints; dual-arm resistivities are defined on the basis of static-force constraints. These quantities serve as measures of how well such dextrous-bracing actions as relocation of the bracing point, sliding contact, and locking of joints affect the ability of the dual-arm manipulator to generate motions and to apply static forces.

Theoretical study and computer simulation have shown that dextrous bracing yields performance characteristics that vary continuously and widely as the bracing point is moved along the braced arm. In general, performance characteristics lie between those of the serial and parallel structures. Thus, one can select configurations dynamically, according to their performance characteristics, to suit the changing requirements of changing tasks.

This work was done by Sukhan Lee and Sungbok Kim of Caltech for **NASA's Jet Propulsion Laboratory**.

IMPROVED ROLLER AND GEAR DRIVES FOR ROBOTS AND VEHICLES

One type is designed to eliminate stick/slip, another to eliminate reaction torque.

Lewis Research Center, Cleveland, Ohio

Two types of gear drives have been devised to improve the performances of robotic mechanisms. One type features a dual-input/single-output differential-drive configuration intended to eliminate stick/slip motions; the other type features a single-input/dual-angular-momentum-balanced-output configuration intended to eliminate reaction torques.

Stick/slip motion can degrade the performance of a robot because a robotic control system cannot instantaneously correct for a sudden change between static and dynamic friction. Reaction torque arises in a structure that supports a mechanism coupled to a conventional gear drive, and can adversely affect the structure, the mechanism, or other equipment connected to the structure or mechanism.

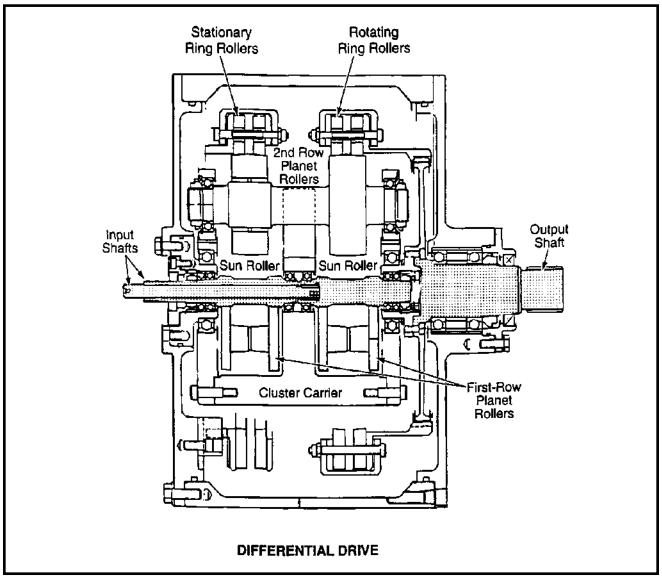
In a drive of the differential type, the two input shafts can be turned at different speeds and, if necessary, in opposite directions, to make the output shaft turn in the forward or reverse direction at a desired speed. This is done without stopping rotation of either input shaft, so that stick/slip does not occur. In a drive of the angular-momentum-balanced type, turning the single input shaft causes the two output shafts to rotate at equal speeds in opposite directions.

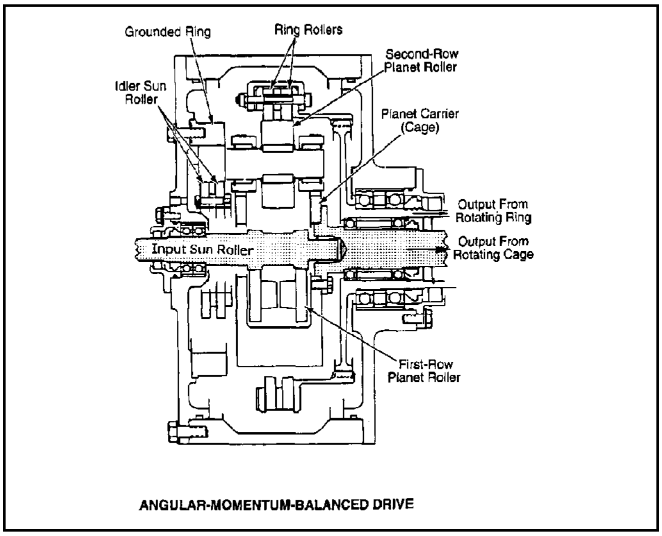
The figure schematically illustrates one of two drives of the differential type and one drive of the angular-momentum-balanced type that have been built and tested. Each of the differential drives is rated at input speeds up to 295 radians per second (2,800 r/min), output torque up to 450 N·m (4,000 lb-in.), and power up to 5.6 kW (7.5 hp). The maximum ratings of the angular-momentum-balanced drive are input speed of 302 radians per second (2,880 r/min), dual output torques of 434 N·m (3,840 lb-in.) each, and power of 10.9 kW (14.6 hp).

Each differential drive features either (as explained in the next two sentences) a dual roller-gear or a roller arrangement with a sun gear, four first-row planet gears, four second-row planet gears, and a ring gear. One of the differential drives contains a planetary roller-gear system with a reduction ratio (measured with one input driving the output while the other input shaft remains stationary) of 29.23:1. The other differential drive (the one shown in the figure) contains a planetary roller system with a reduction ratio of 24:1. The angular-momentum-balanced drive features a planetary roller system with five first- and second-row planet gears and a reduction ratio (the input to each of the two outputs) of 24:1. The three drives were subjected to a broad spectrum of tests to measure linearity, cogging, friction, and efficiency. All three drives operated as expected kinematically, exhibiting efficiencies as high as 95 percent.

Drives of the angular-momentum-balanced type could provide a reaction-free actuation when applied with proper combinations of torques and inertias coupled to output shafts. Drives of the differential type could provide improvements over present robotic transmissions for applications in which there are requirements for extremely smooth and accurate torque and position control, without inaccuracies that accompany stick/slip. Drives of the differential type could also offer viable alternatives to variable-ratio transmissions in applications in which output shafts are required to be driven both forward and in reverse, with an intervening stop. A differential transmission with two input drive motors could be augmented by a control system to optimize input speeds for any requested output speed; such a transmission could be useful in an electric car.

This work was done by William J. Anderson and William Shipitalo of Nastec, Inc., and Wyatt Newman of Case Western Reserve University for **Lewis Research Center**.

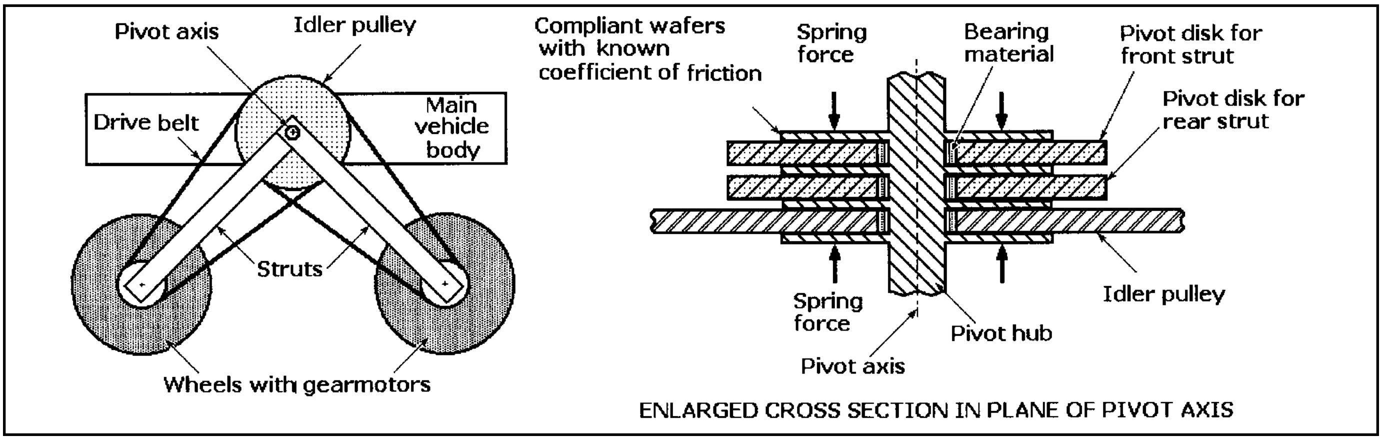




These Improved Gear Drives offer advantages for control of traction and rotary actuation in robots. In addition, drives of the differential type could be used in variable-speed transmissions in automobiles.

ALL-TERRAIN VEHICLE WITH SELF-RIGHTING AND POSE CONTROL

Wheels driven by gearmotors are mounted on pivoting struts.
NASA's Jet Propulsion Laboratory, Pasadena, California



Each wheel Is driven by a dedicated gearmotor and is coupled to the idler pulley. The pivot assembly imposes a constant frictional torque T, so that it is possible to (a) turn both wheels in unison while both struts remain locked, (b) pivot one strut, or (c) pivot both struts in opposite directions by energizing the gearmotors to apply various combinations of torques T/2 or T.

A small prototype robotic all-terrain vehicle features a unique drive and suspension system that affords capabilities for self righting, pose control, and enhanced maneuverability for passing over obstacles. The vehicle is designed for exploration of planets and asteroids, and could just as well be used on Earth to carry scientific instruments to remote, hostile, or otherwise inaccessible locations on the ground. The drive and suspension system enable the vehicle to perform such diverse maneuvers as flipping itself over, traveling normal side up or upside down, orienting the main vehicle body in a specified direction in all three dimensions, or setting the main vehicle body down onto the ground, to name a few. Another maneuver enables the vehicle to overcome a common weakness of traditional all-terrain vehicles—a limitation on traction and drive force that makes it difficult or impossible to push wheels over some obstacles: This vehicle can simply lift a wheel onto the top of an obstacle.

The basic mode of operation of the vehicle can be characterized as four-wheel drive with skid steering. Each wheel is driven individually by a dedicated gearmotor. Each wheel and its gearmotor are mounted at the free end of a strut that pivots about a lateral axis through the center of gravity of the vehicle (see figure). Through pulleys or other mechanism attached to their wheels, both gearmotors on each side of the vehicle drive a single idler disk or pulley that turns about the pivot axis.

The design of the pivot assembly is crucial to the unique capabilities of this system. The idler pulley and the pivot disks of the struts are made of suitably chosen materials and spring-loaded together along the pivot axis in such a way as to resist turning with a static frictional torque T; in other words, it is necessary to apply a torque of T to rotate the idler pulley or either strut with respect to each other or the vehicle body.

During ordinary backward or forward motion along the ground, both wheels are turned in unison by their gearmotors, and the belt couplings make the idler pulley turn along with the wheels. In this operational mode, each gearmotor contributes a torque T/2 so that together, both gearmotors provide torque T to overcome the locking friction on the idler pulley. Each strut remains locked at its preset angle because the torque T/2 supplied by its motor is not sufficient to overcome its locking friction T.

If it is desired to change the angle between one strut and the main vehicle body, then the gearmotor on that strut only is energized. In general, a gearmotor acts as a brake when not energized. Since the gearmotor on the other strut is not energized and since it is coupled to the idler pulley, a torque greater than T would be needed to turn the idler pulley. However, as soon as the gearmotor on the strut that one desires to turn is energized, it develops enough torque (T) to begin pivoting the strut with respect to the vehicle body.

It is also possible to pivot both struts simultaneously in opposite directions to change the angle between them. To accomplish this, one energizes the gearmotors to apply equal and opposite torques of magnitude T: The net torque on the idler pulley balances out to zero, so that the idler pulley and body remain locked, while the applied torques are just sufficient to turn the struts against locking friction. If it is desired to pivot the struts through unequal angles, then the gearmotor speeds are adjusted accordingly.

The prototype vehicle has performed successfully in tests. Current and future work is focused on designing a simple hub mechanism, which is not sensitive to dust or other contamination, and on active control techniques to allow autonomous planetary rovers to take advantage of the flexibility of the mechanism.

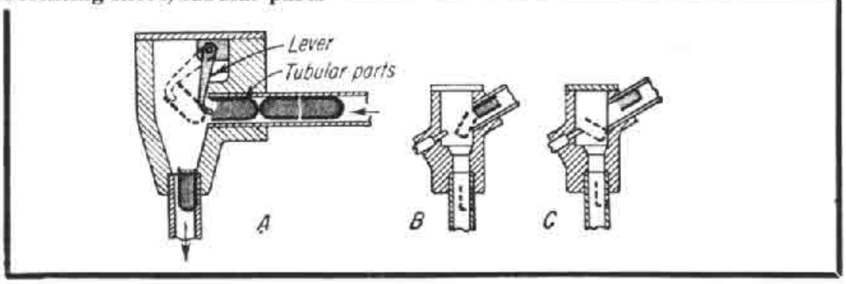
This work was done by Brian H. Wilcox and Annette K. Nasif of Caltech for **NASA's Jet Propulsion Laboratory**.

CHAPTER 3
PARTS-HANDLING MECHANISMS

MECHANISMS THAT SORT, FEED, OR WEIGH

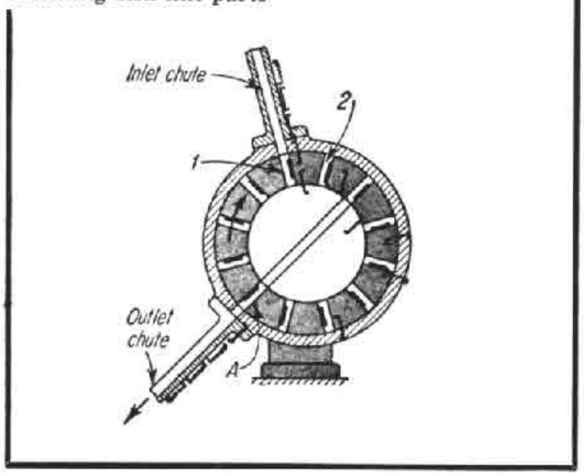
ORIENTING DEVICES

Orienting short, tubular parts



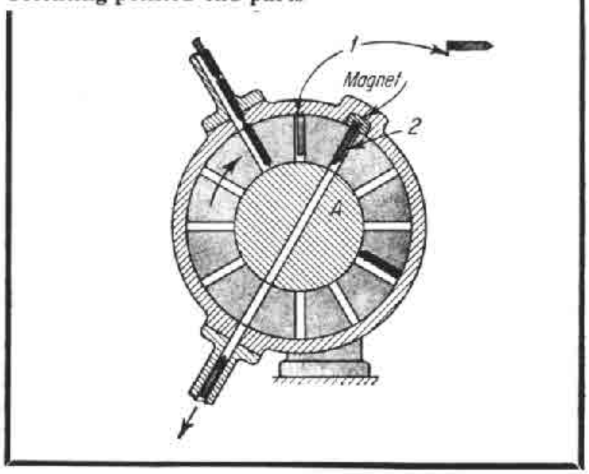
Here's a common problem; Parts arrive in either open-end or closed-end first; you need a device that will orient all the parts so they feed out facing the same way. In Fig. A. when a part comes in open-end first, it is pivoted by the swinging lever so that the open end is up. When it comes in closed-end first, the part brushes away the lever to flip over headfirst. Fig. B and C show a simpler arrangement with pin in place of lever.

Orienting dish-like parts



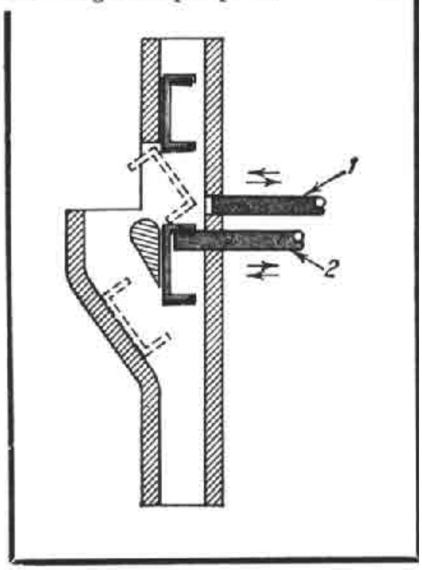
A part with its open-end facing to the right (part 1) falls on a matching projection as the indexing wheel begins to rotate clockwise. The projection retains the part for 230º to point A where it falls away from the projection to slide down the outlet chute, open-end up. An incoming part facing the other way (2) is not retained by the projection, hence it slides *through* the indexing wheel so that it too, passes through the outlet with its open-end up.

Orienting pointed-end parts



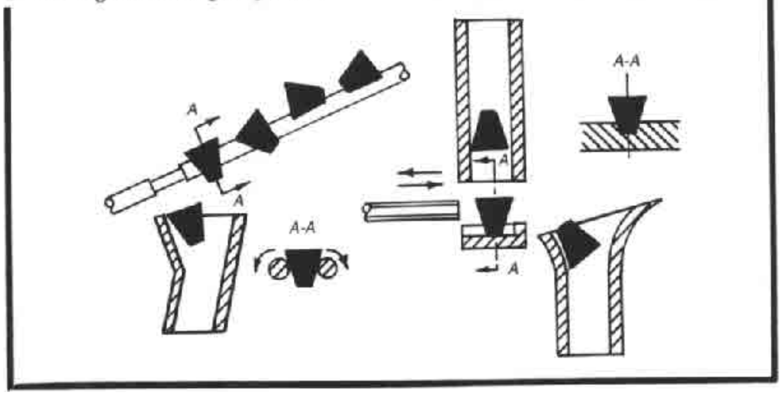
The important point here is that the built-in magnet cannot hold on to a part as it passes by if the part has its pointed end facing the magnet. Such a correctly oriented part (part 1) will fall through the chute as the wheel indexes to a stop. An incorrectly oriented part (part 2) is briefly held by the magnet until the indexing wheel continues on past the magnet position. The wheel and the core with the slot must be made from some nonmagnetic material.

Orienting U-shaped parts



The key to this device is two pins that reciprocate one after another in the horizontal direction. The parts come down the chute with the bottom of the "U" facing either to the right or left. All pieces first strike and rest on pin 2. Pin 1 now moves into the passage way, and if the bottom of the "U" is facing to the right, the pin would kick over the part as shown by the dotted lines. If, on the other hand, the bottom of the "U" had been to the left, the motion of pin 1 would have no effect, and as pin 2 withdrew to the right, the part would be allowed to pass down through the main chute.

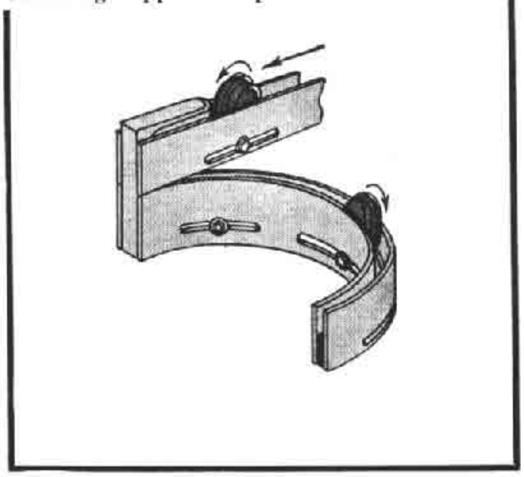
Orienting cone-shaped parts



Regardless of which end of the cone faces forward as the cones slide down the cylindrical rods, the fact that both rods rotate in opposite directions causes the cones to assume the position shown in section *A-A* (above). When the cones reach the thinned-down section of the rods, they fall down into the chute, as illustrated.

In the second method of orienting cone-shaped parts (left), if the part comes down small end first, it will fit into the recess. The reciprocating rod, moving to the right, will then kick the cone over into the exit chute. But if the cone comes down with its large end first, it sits on top of the plate (instead of inside the recess), and the rod simply pushes it into the chute without turning it over.

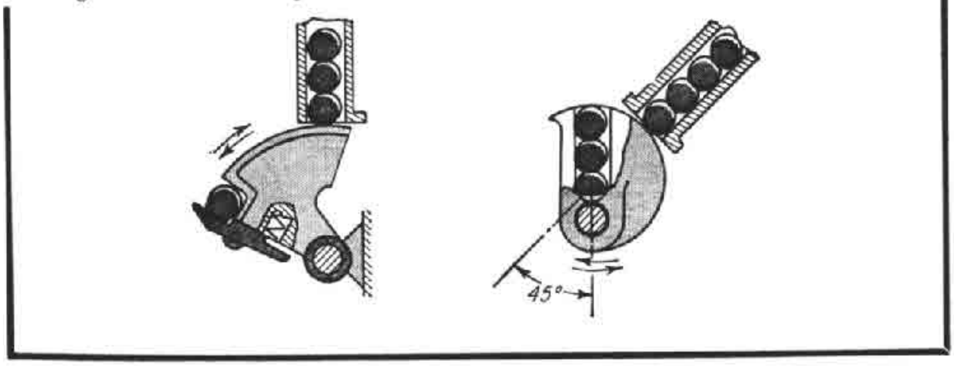
Orienting stepped-disk parts



Parts rolling down the top rail to the left drop to the next rail which has a circular segment. The part, therefore, continue to roll on in the original direction, but their faces have now been rotated 180º. The idea of dropping one level might seem oversimplified, but it avoids the cam-based mechanisms more commonly used for accomplishing this job.

SIMPLE FEEDING DEVICES

Feeding a fixed number of parts

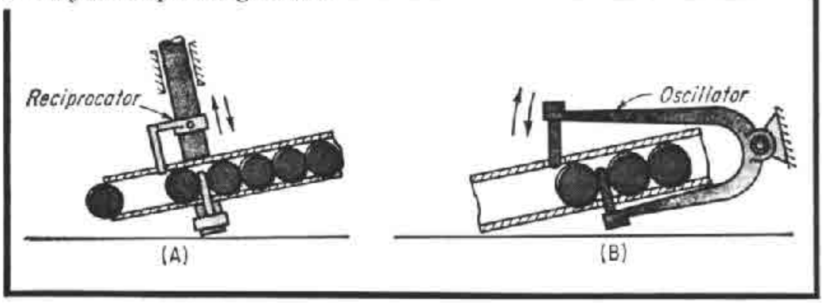


The oscillating sector picks up the desired number of parts, left diagram, and feeds them by pivoting the required number of degrees. The device for oscillating the sector must be able to produce dwells at both ends of the stroke to allow sufficient time for the parts to fall in and out of the sector.

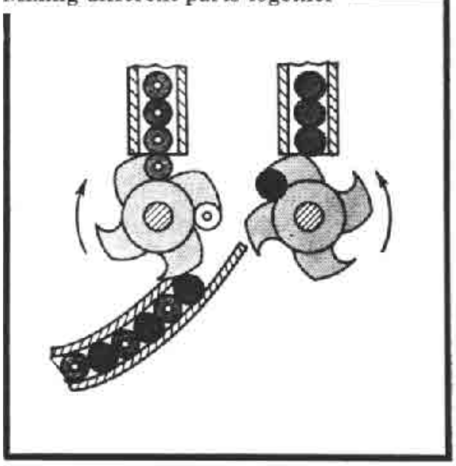
The circular parts feed down the chute by gravity, and they are separated by the reciprocating rod. The parts first roll to station 3 during the downward stroke of the reciprocator, then to station 1 during the upward stroke; hence the time span between parts is almost equivalent to the time it takes for the reciprocator to make one complot oscillation.

The device in Fig. B is similar to the one in Fig. A, except that the reciprocator is replaced by an oscillating member.

One-by-one separating device

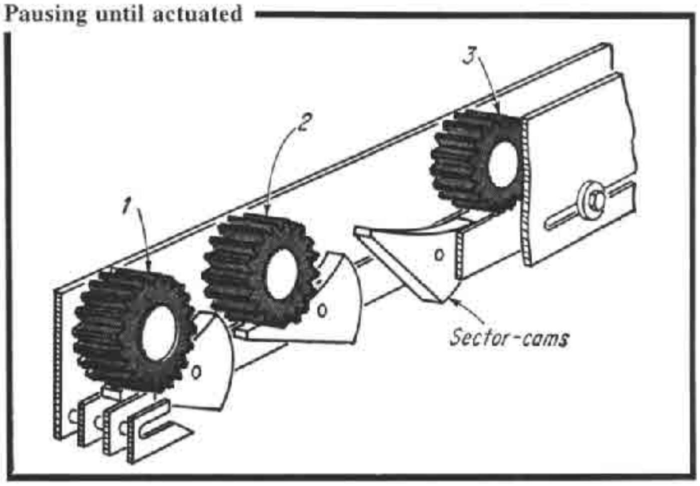


Mixing different parts together



Two counter rotating wheels form a simple device for alternating the feed of two different workpieces.

Pausing until actuated



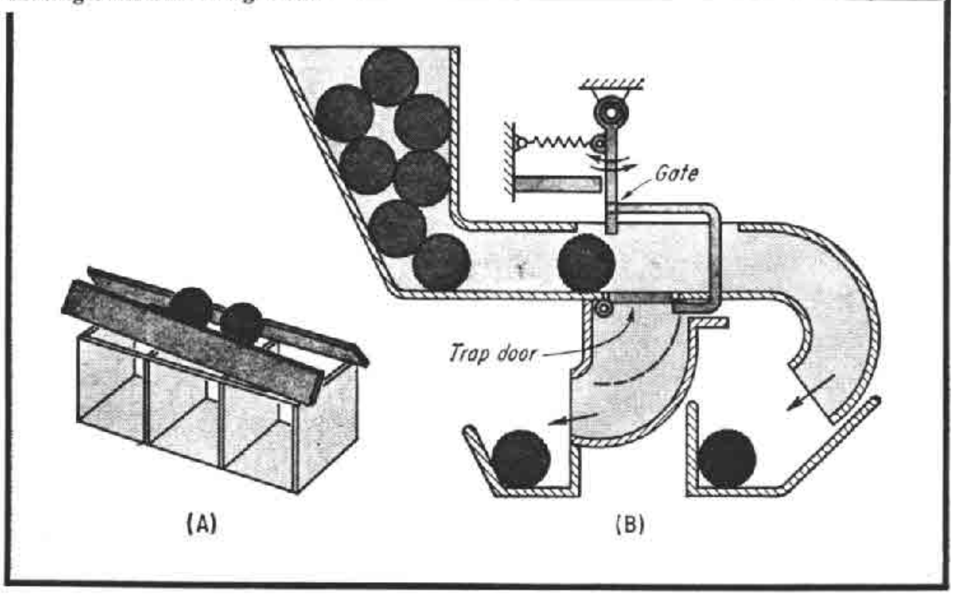
Each gear in this device is held up by a pivotable cam sector until the gear ahead of it moves forward. Thus, gear 3, rolling down the chute, kicks down its sector cam but is held up by the previous cam. When gear 1 is picked off (either manually, or mechanically), its sector cam pivots clockwise because of its own weight. This permits gear 2 to move into place of gear 1—and frees cam 2 to pivot clockwise. Thus, all gears in the row move forward one station.

SORTING DEVICES

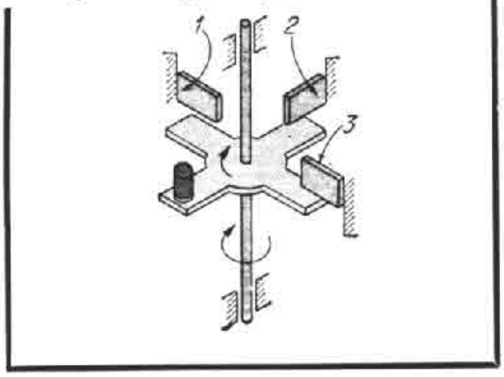
In the simple device (A) the balls run down two inclined and slightly divergent rails. The smallest balls, therefore, will fall into the left chamber, the medium-size ones into the middle-size chamber, and the largest ones into the right chamber.

In the more complicated arrangement (B), the balls come down the hopper and must pass a gate which also acts as a latch for the trapdoor. The proper-size balls pass through without touching (actuating) the gate. Larger balls, however, brush against the gate which releases the catch on the bottom of the trapdoor, and fall through into the special trough for the rejects.

Sorting balls according to size



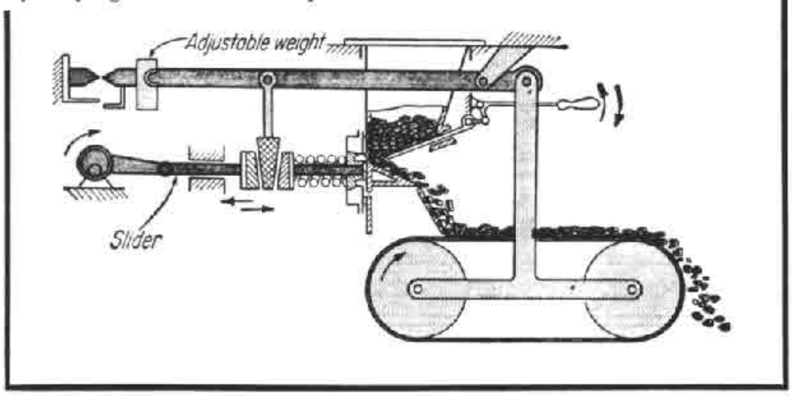
Sorting according to height



Workpieces of varying heights are placed on this slowly rotating cross-platform. Bars 1, 2, and 3 have been set at decreasing heights beginning with the highest bar (bar 1), down to the lowest bar (bar 3). The workpiece is therefore knocked off the platform at either station 1, 2, or 3, depending on its height.

WEIGHT-REGULATING ARRANGEMENTS

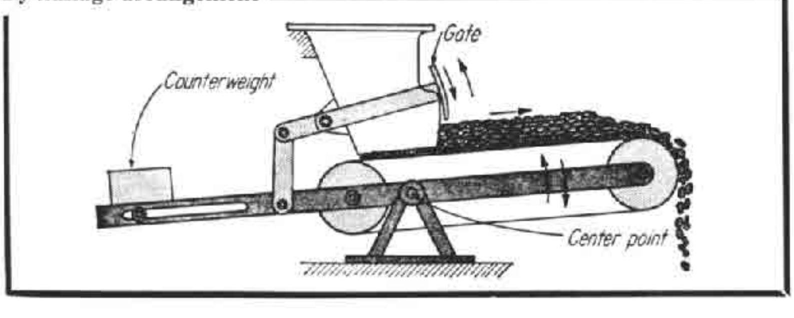
By varying the vibration amplitudes



The material in the hopper is fed to a conveyor by the vibration of the reciprocating slider. The pulsating force of the slider is transmitted through the rubber wedge and on to the actuating rod. The amplitude of this force can be varied by moving the wedge up or down. This is done automatically by making the conveyor pivot around a central point. As the conveyor becomes overloaded, it pivots clockwise to raise the wedge, which reduces the amplitude of the force and slows the feed rate of the material.

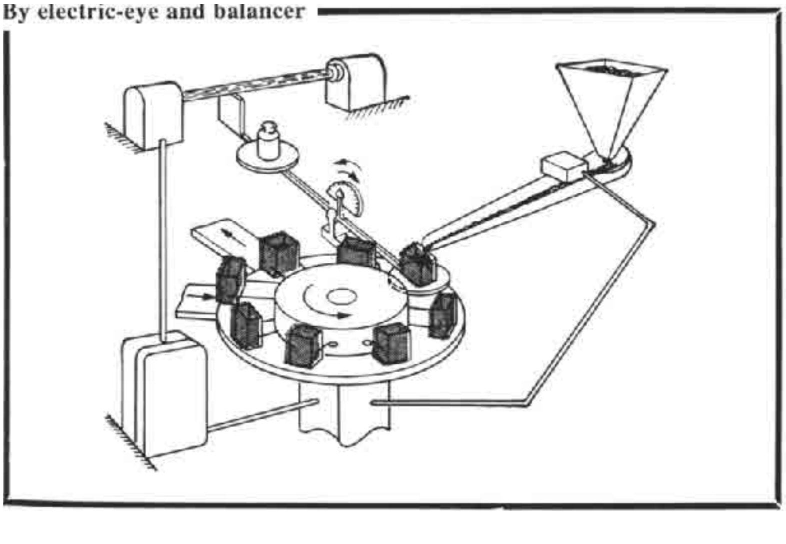
Further adjustments in feed rate can be made by shifting the adjustable weight or by changing the speed of the conveyor belt.

By linkage arrangement



The loose material falls down the hopper and is fed to the right by the conveyor system which can pivot about the center point. The frame of the conveyor system also actuates the hopper gate so that if the amount of material on the belt exceeds the required amount, the conveyor pivots clockwise and closes the gate. The position of the counterweight on a frame determines the feed rate of the system.

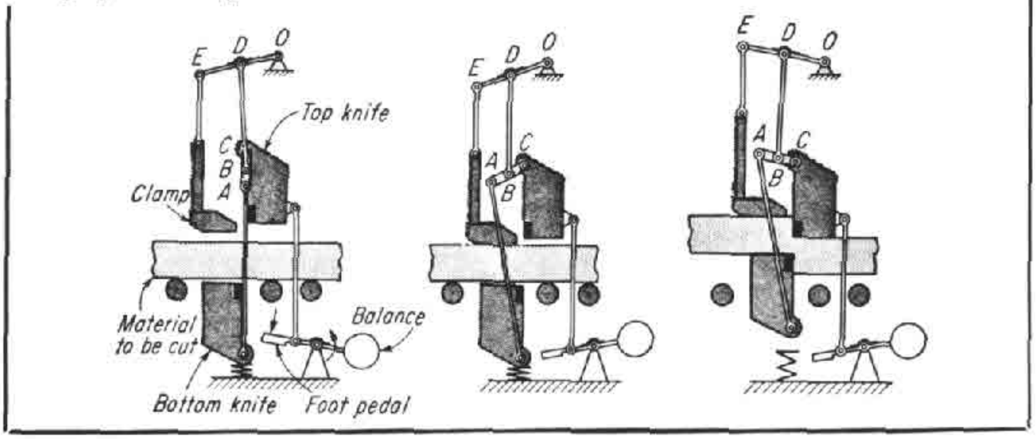
By electric-eye and balancer



The indexing table automatically stops at the feed station. As the material drops into the container, its weight pivots the screen upward to cut off the light beam to the photocell relay. This in turn shuts the feed gate. The reactuation of the indexing table can be automatic after a time delay or by the cutoff response of the electric eye.

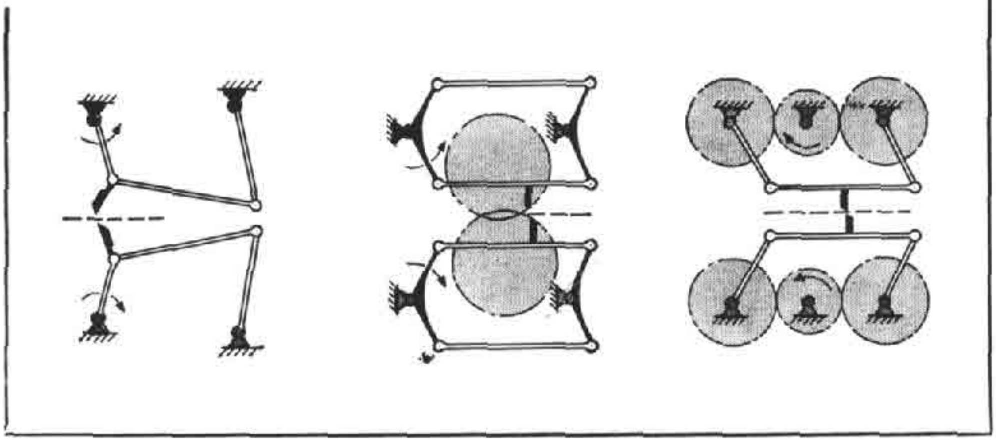
CUTTING MECHANISMS

Clamping and cutting device



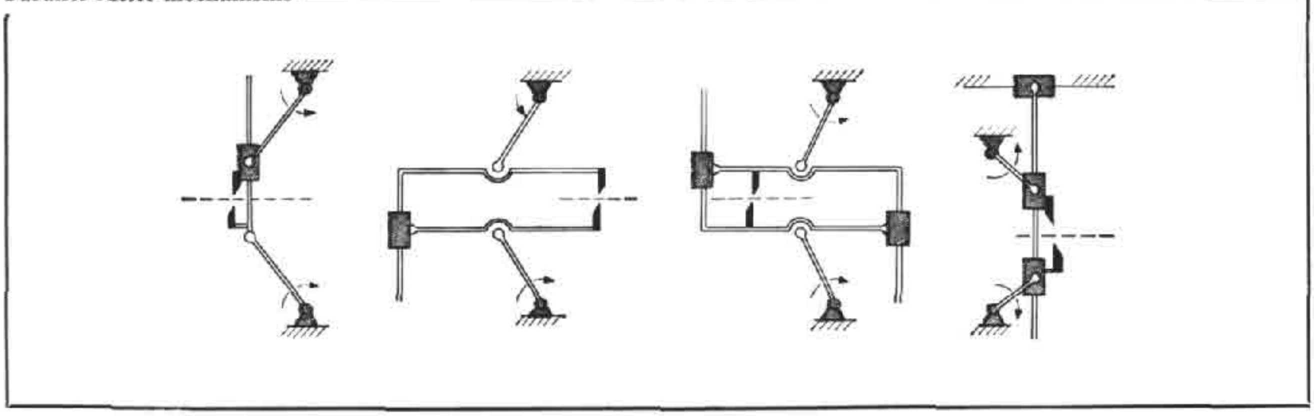
By pressing down on the foot pedal of this mechanism, the top knife and the clamp will be moved downward. However, when the clamp presses on the material, both it and link *EDO* will be unable to move further. Link *AC* will now begin to pivot around point *B*, drawing the lower knife up to begin the cutting action.

Four-bar cutter device



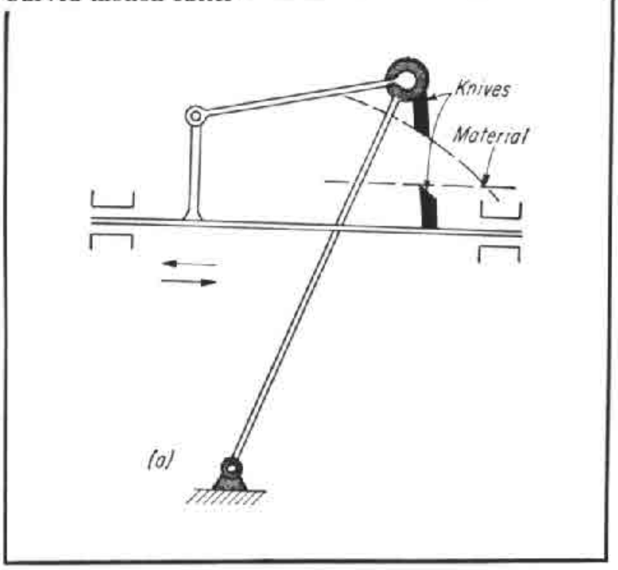
These 3 four-bar cutters provide a stable, strong, cutting action by coupling two sets of links to chain four-bar arrangements.

Parallel cutter mechanisms



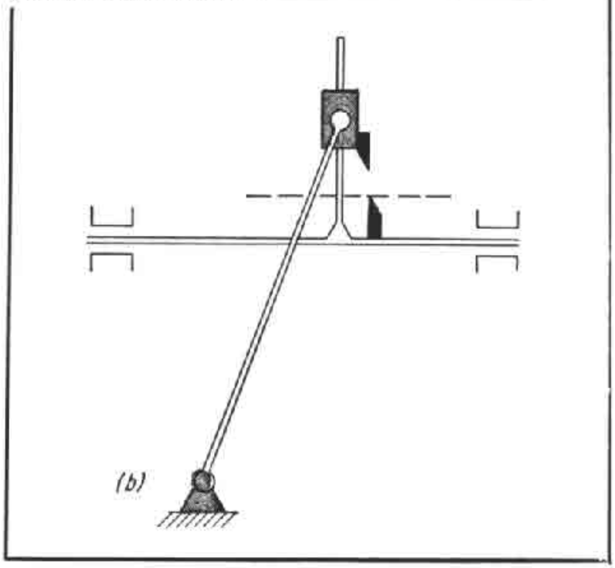
The cutting edges of the knives in the four mechanisms move parallel to each other, and they also remain vertical at all times to cut the material while it is in motion. The two cranks are rotated with constant velocity by a 1 to 1 gear system (not shown), which also feeds the material through the mechanism.

Curved-motion cutter



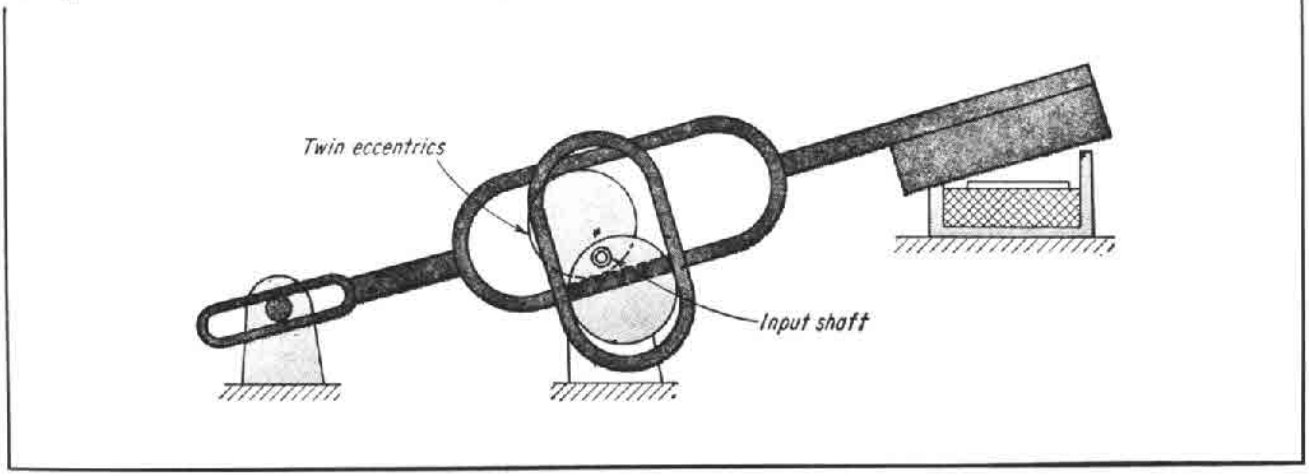
The material is cut while in motion by the reciprocating action of the horizontal bar. As the bar with the bottom knife moves to the right, the top knife will arc downward to perform the cutting operation.

Vertical cutter motion



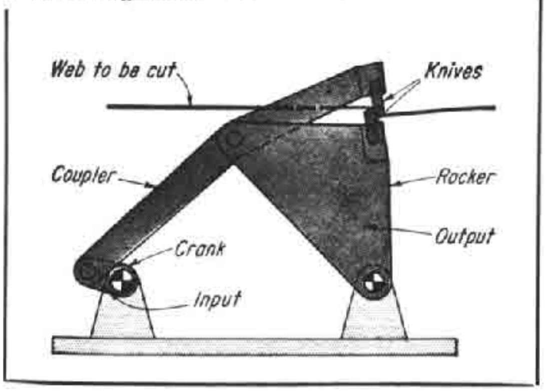
The top knife in this arrangement remains parallel to the bottom knife at all times during cutting to provide a true scissor-like action, but friction in the sliding member can limit the cutting force.

Slicing mechanism



Slicing motion is obtained from the synchronized effort of two eccentric disks. The two looped rings actuated by the disks are welded together. In the position shown, the bottom eccentric disk provides the horizontal cutting movement, and the top disk provides the up-and-down force necessary for the cutting action.

Web-cutting mechanism

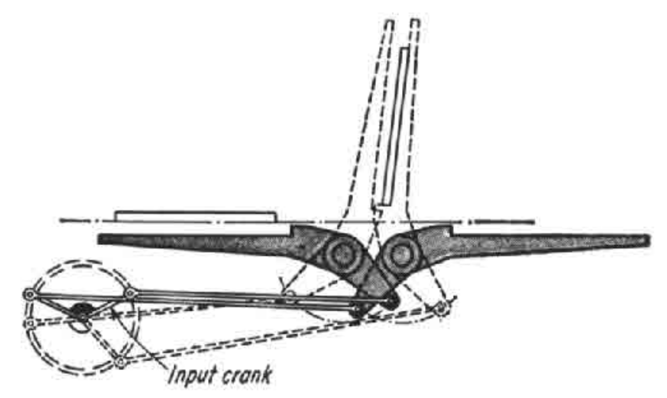


This four-bar linkage with an extended coupler can cut a web on the run at high speeds. The four-bar linkage shown is dimensioned to give the knife a velocity during the cutting operation that is equal to the linear velocity of the web.

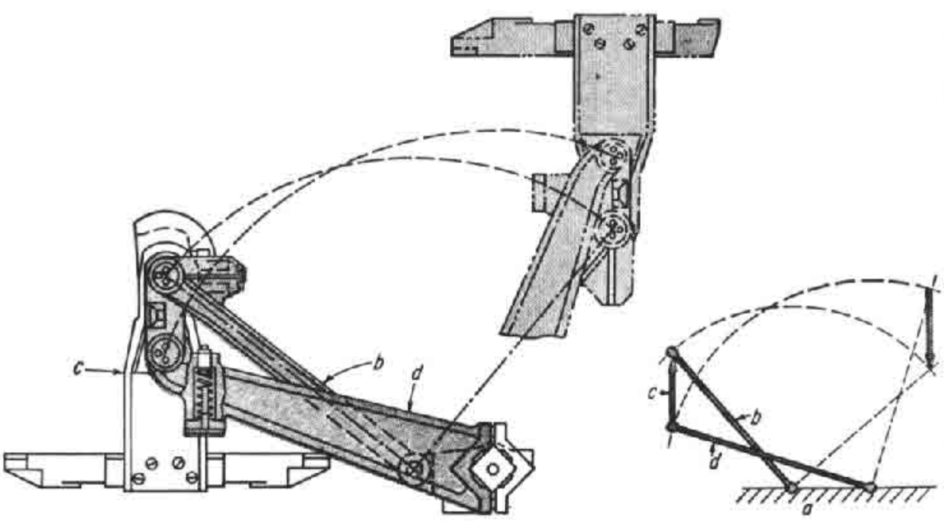
FLIPPING MECHANISMS

This mechanism can turn over a flat piece by driving two four-bar linkages from one double crank. The two flippers are actually extensions of the fourth members of the four-bar linkages. Link proportions are selected so that both flippers rise up at the same time to meet a line slightly off the vertical to transfer the piece from one flipper to the other by the momentum of the piece.

Turnover device



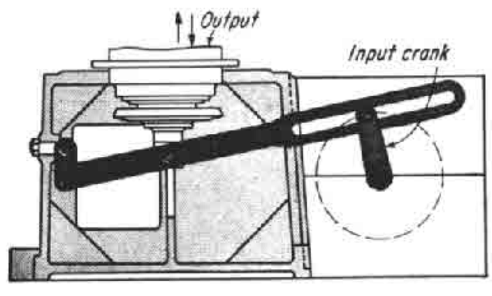
Upside-down flipper



This is a four-bar linkage (links a, b, c, d) in which the part to be turned over is coupler c of the linkage. For the proportions shown, the 180º rotation of link c is accomplished during the 90º rotation of the input link.

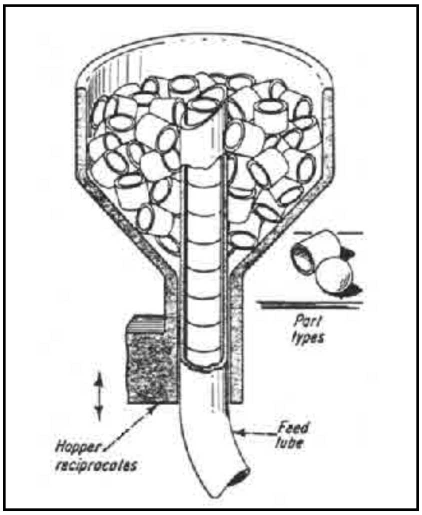
VIBRATING MECHANISM

Vibrating mechanism

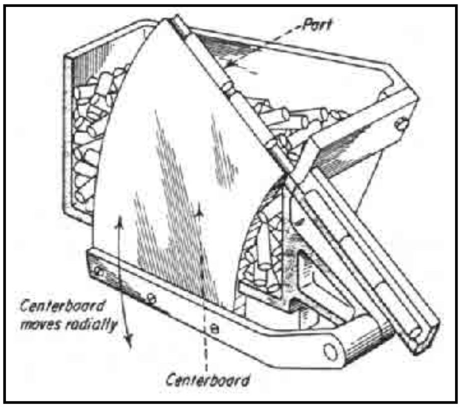


As the input crank rotates, the slotted link, which is fastened to the frame with an intermediate link, oscillates to vibrate the output table up and down.

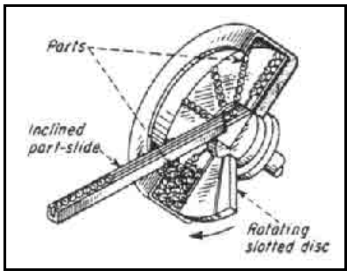
SEVEN BASIC PARTS SELECTORS



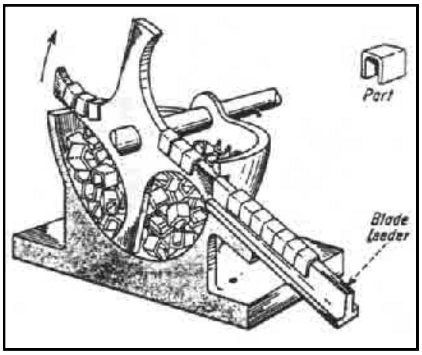
A **reciprocating feed** for spheres or short cyclinders is one of the simplest feed mehanisms. Either the hopper or the tube reciprocates. The hopper must be kept topped-up with parts unless the tube can be adjusted to the parts level.



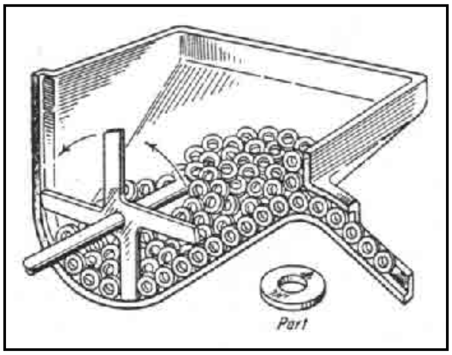
A centerboard selector is similar to reciprocating feed. The centerboard top can be milled to various section shapes to pick up moderately complex parts. I works best, however, with cylinders that are too long to be led with the reciprocating hopper. The feed can be continuos or as required.



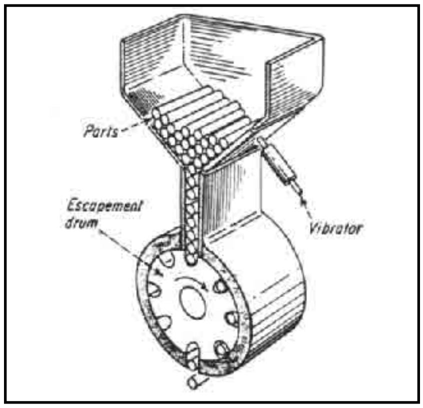
A rotary screw-feed handles screws, headed pings, shouldered shafts, and similar parts in most hopper feeds, random selection of chance-oriented parts calls for additional machinery if the parts must be fed in only one specific position. Here, however, all screws are fed in the same orientation) except for slot position) without separate machinery.



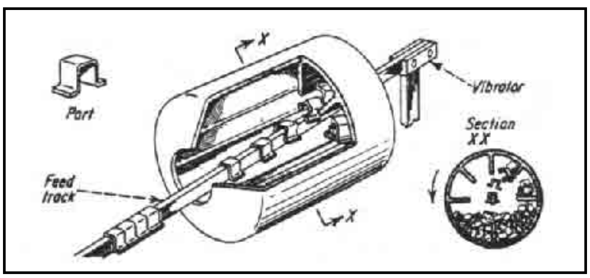
Rotary centerblades catch small U-shaped parts effectively if their legs are not too long. The parts must also be resilient enough to resist permanent set from displacement forces as the blades cut through a pile of parts. The feed is usual continuous.



A paddle wheel is effective for feeding disk-shaped parts if they are stable enough. Thin, weak parts would bend and jam. Avoid these designs, if possible—Especially if automatic assembly methods will be employed.

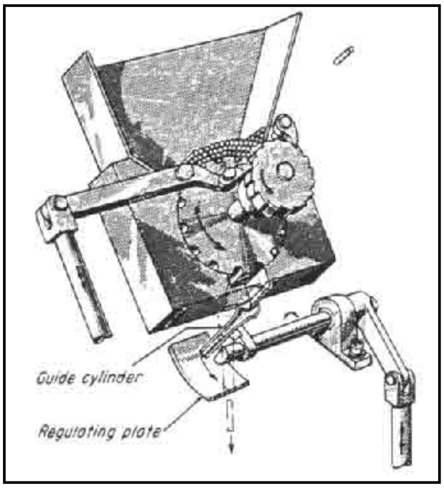


A long-cylinder feeder is a variation of the first two hoppers. If the cylinders have similar ends, the parts can be fed without proposition, thus assisting automatic assembly. A cylinder with differently shaped ends requires extra machinery to orientated the part before it can be assembled.

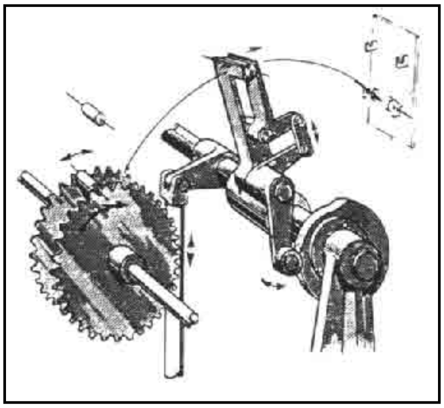


A barrel hopper is useful if parts lend to become entangled. The parts drop free of the rotating-barrel sides. By chance selection, some of them fall onto the vibrating rack and are fed out of the barrel. The parts should be stiff enough to resist excessive bending because the tumbling action can subject them to relatively severe loads. The tumbling can help to remove sharp burrs.

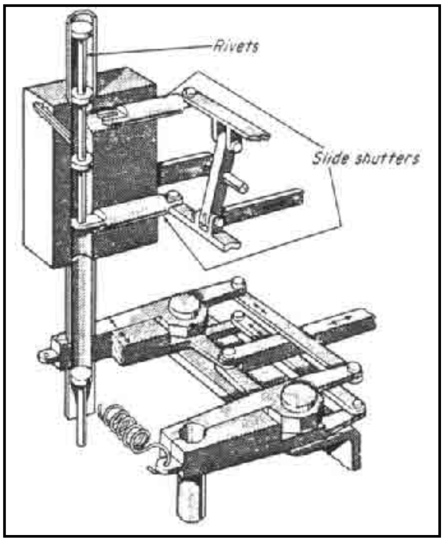
ELEVEN PARTS-HANDLING MECHANISMS



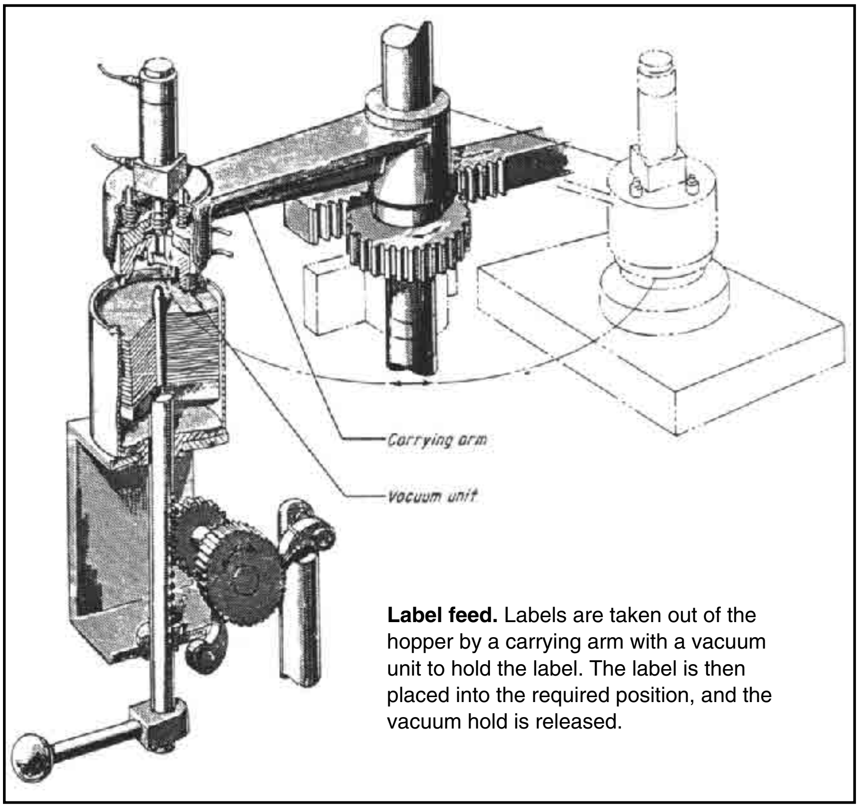
Gravity feed for rods. Single rods of a given length are transferred from the hopper to the lower guide cylinder by means of an intermittently rotating disk with a notched circumference. The guide cylinder, moved by a lever, delivers the rod when the outlet moves free of the regulating plate.



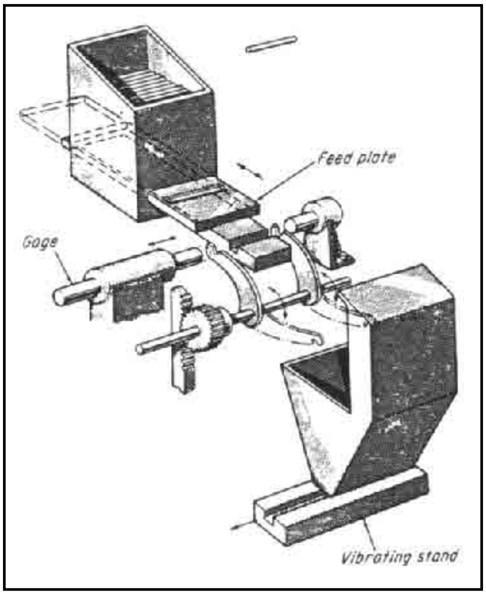
Feeding electronic components. Capacitors, for example, can be delivered by a pair of intermittently rotating gearlike disks with notched circumferences. Then a pick-up arm lifts the capacitor and it is carried to the required position by the action of a cam and follower.



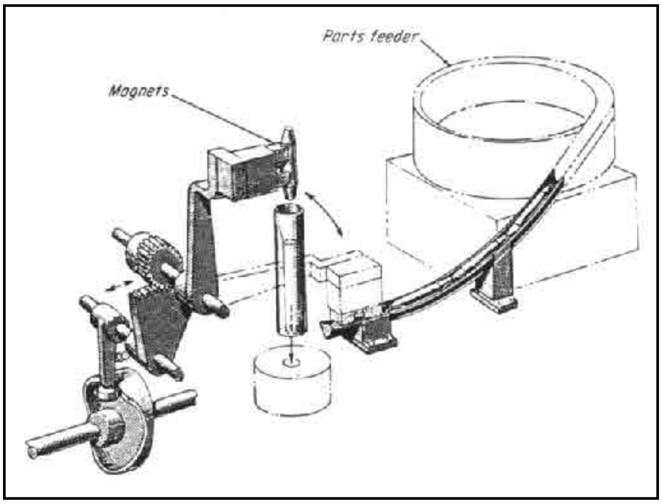
Feeding headed rivets. Headed rivets, correctly oriented, are supplied from a parts-feeder in a given direction. They are dropped, one by one, by the relative movement of a pair of slide shutters. Then the rivet falls through a guide cylinder to a clamp. Clamp pairs drop two rivets into corresponding holes.



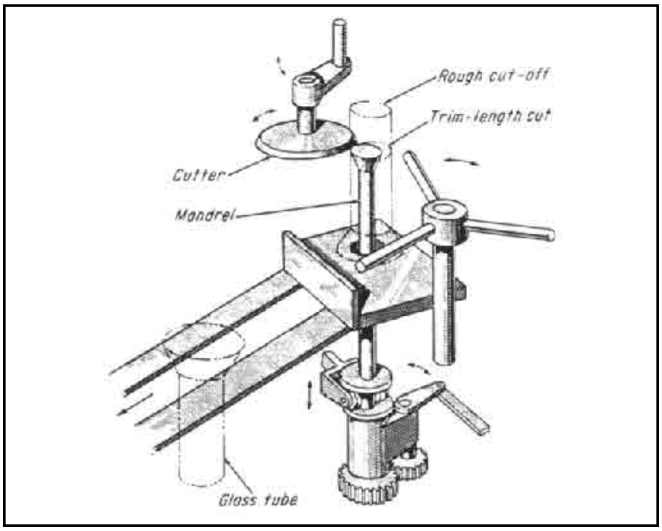
Label feed. Labels are taken out of the hopper by a carrying arm with a vacuum unit to hold the label. The label is then placed into the required position, and the vacuum hold is released.



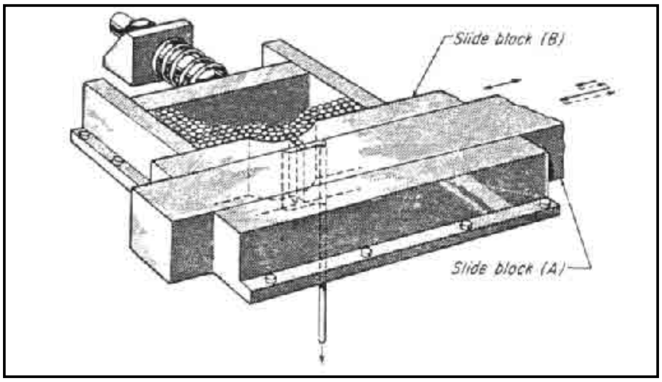
Horizontal feed for fixed-length rods. Single rods of a given length are brought from the hopper to the slot of a fixed plate by a moving plate. After being gauged in the notched portion of the fixed plate, each rod is moved to the chute by means of a lever, and is removed from the chute by a vibrating table.



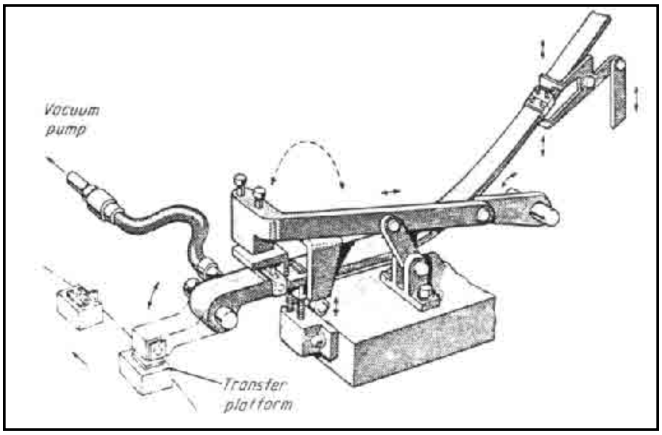
Pin inserter. Pins, supplied from the parts-feeder, are raised to a vertical position by a magnet arm. The pin drops through a guide cylinder when the electromagnet is turned off.



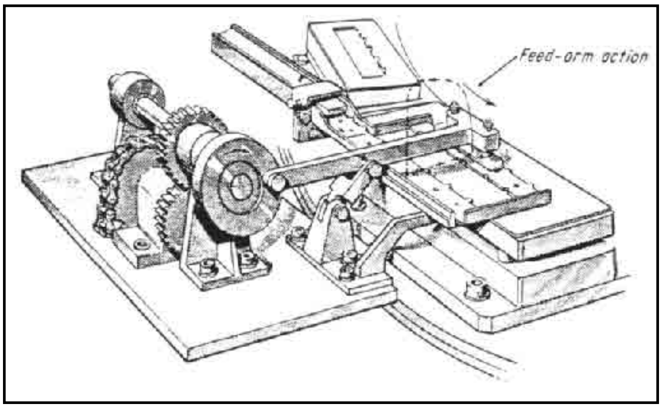
Cutoff and transfer devices for glass tubes. The upper part of a rotating glass tube is held by a chuck (not shown). When the cutter cuts the tube to a given length, the mandrel comes down and a spring member (not shown) drops the tube on the chute.



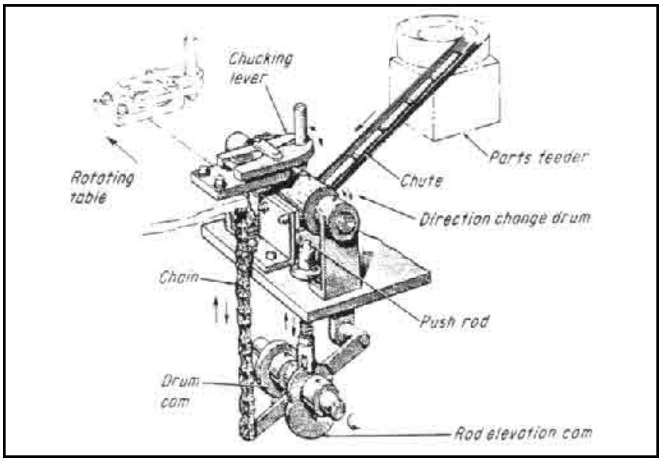
Vertical feed for wires. Wires of fixed length are stacked vertically, as illustrated. They are removed, one by one, as blocks A and B are slid by a cam and lever (not shown) while the wires are pressed into the hopper by a spring.



Feeding special-shaped parts. Parts of such special shapes as shown are removed, one by one, in a given direction, and are then moved individually into the corresponding indents on transfer platforms.



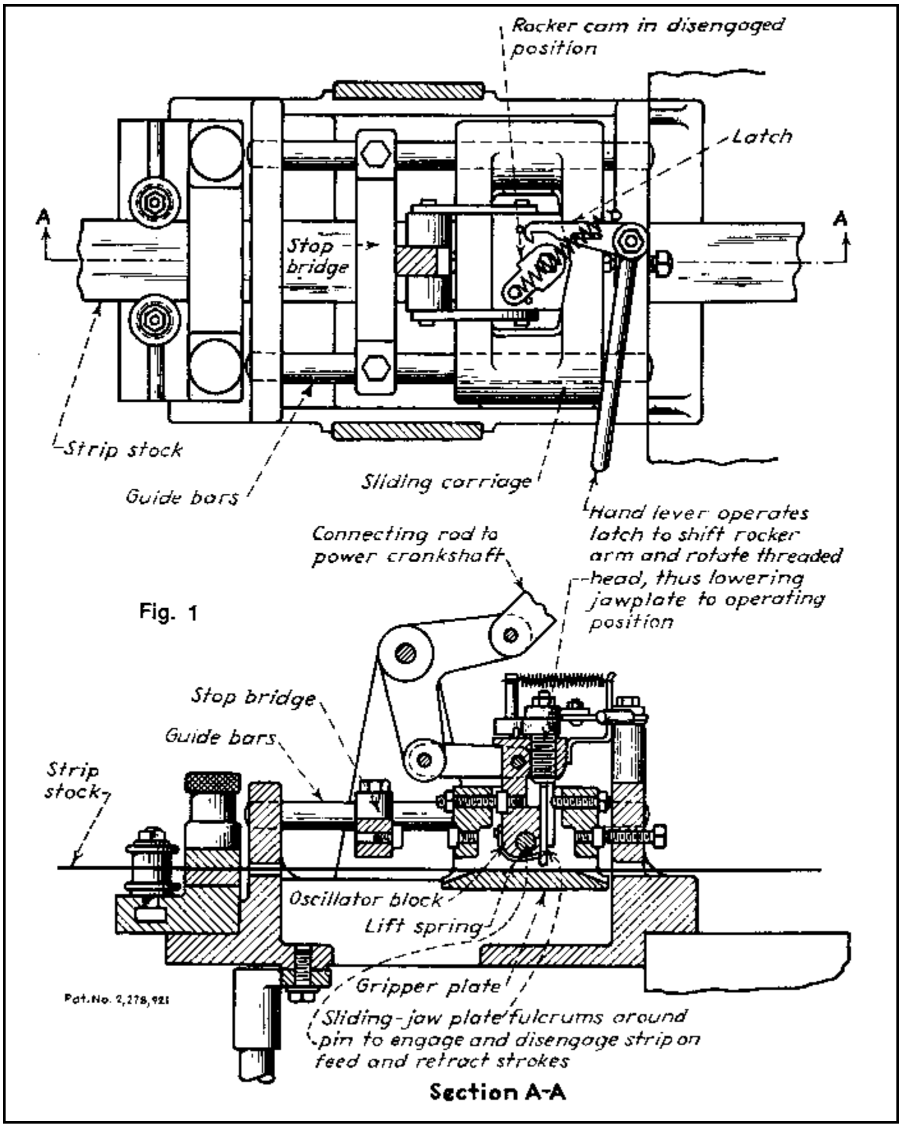
Lateral feed for plain strips. Strips supplied from the parts-feeder are put into the required position, one by one, by an arm that is part of a D-drive linkage.

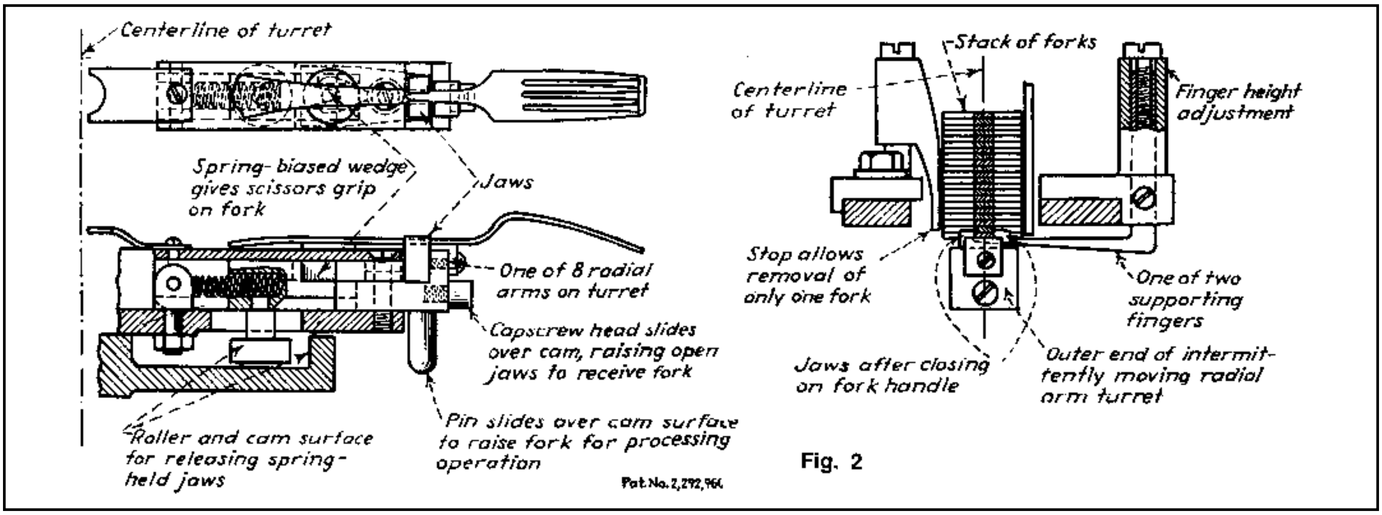


Vertical feed for rods. Rods supplied from the parts-feeder are fed vertically by a direction drum and a pushing bar. The rod is then drawn away by a chucking lever.

SEVEN AUTOMATIC-FEED MECHANISMS

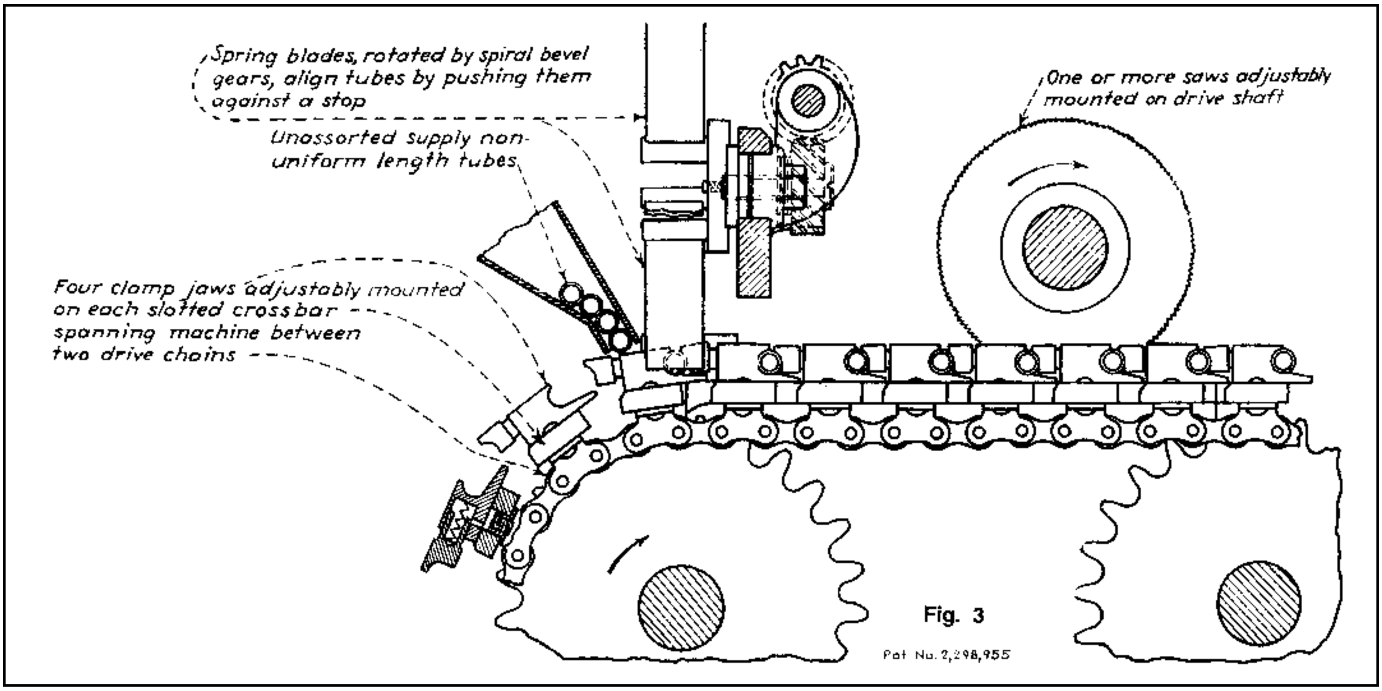
The design of feed mechanisms for automatic or semiautomatic machines depends largely upon such factors as size, shape, and character of the materials or parts that are to be fed into a machine, and upon the kinds of operation to be performed. Feed mechanisms can be simple conveyors that give positive guidance, or they might include secure holding devices if the parts are subjected to processing operations while being fed through a machine. One of the functions of a feed mechanism is to extract single pieces from a stack or unassorted supply of stock. If the stock is a continuous strip of metal, roll of paper, long bar, or tube, the mechanism must maintain intermittent motion between processing operations. These conditions are illustrated in the accompanying drawings of feed mechanisms.





Centerline of turret
Spring-biased wedge gives scissors grip on fork
Jaws
One of 8 radial arms on turret
Capscrew head slides over cam, raising open jaws to receive fork
Roller and cam surface for releasing spring-held jaws
Pin slides over cam surface to raise fork for processing operation
Pat. No. 2,292,966
Stack of forks
Centerline of turret
Finger height adjustment
Stop allows removal of only one fork
One of two supporting fingers
Jaws after closing on fork handle
Outer end of intermittently moving radial arm turret

Fig. 2



Spring blades, rotated by spiral bevel gears, align tubes by pushing them against a stop
Unassorted supply non-uniform length tubes
One or more saws adjustably mounted on drive shaft
Four clamp jaws adjustably mounted on each slotted crossbar spanning machine between two drive chains
Pat. No. 2,298,955

Fig. 3

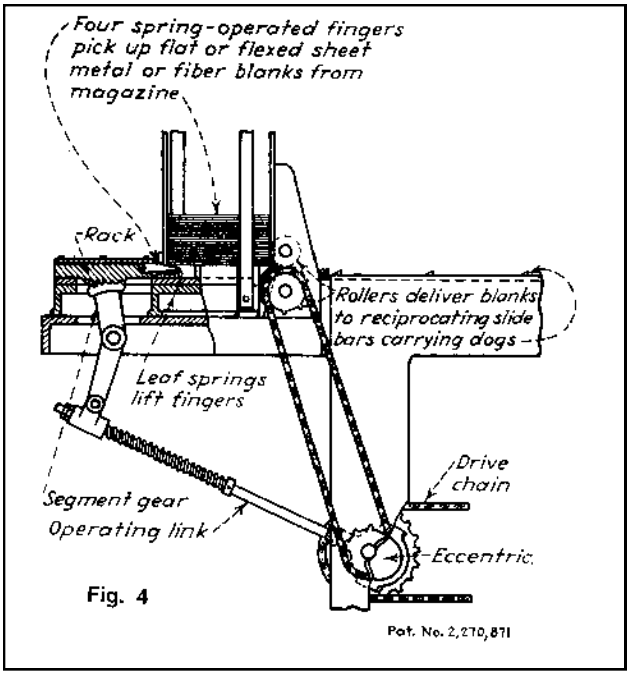
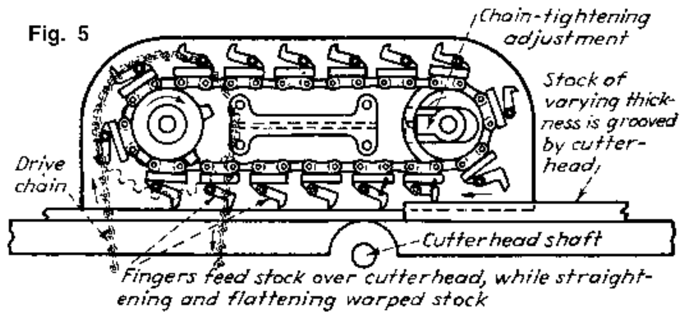


Fig. 4

Fig. 5



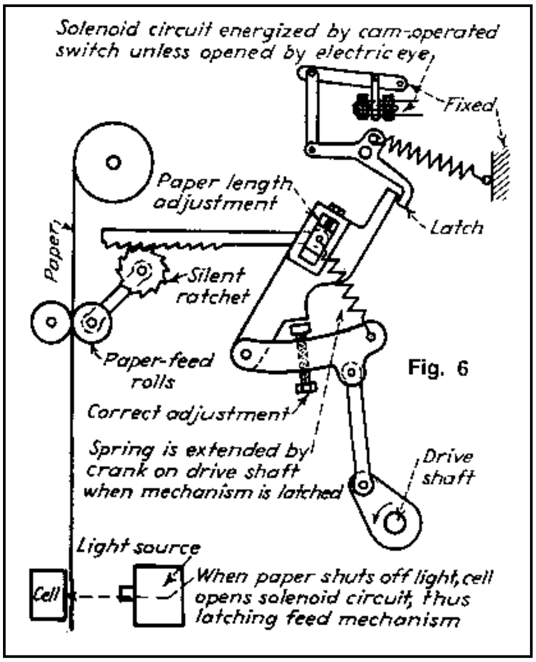


Fig. 6

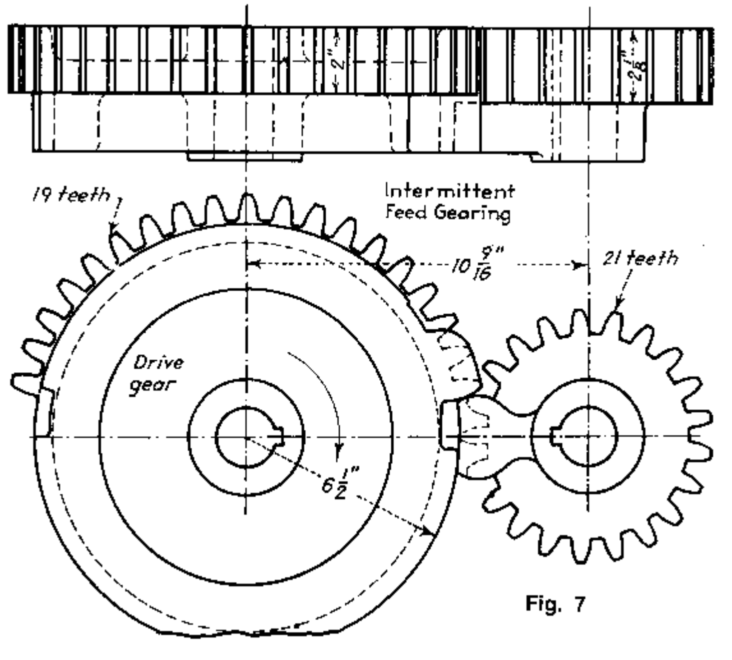


Fig. 7

SEVEN LINKAGES FOR TRANSPORT MECHANISMS

Transport mechanisms generally move material. The motion, although unidirectional, gives an intermittent advancement to the material being conveyed. The essential characteristic of such a motion is that all points in the main moving members follow similar and equal paths. This is necessary so that the members can be subdivided into sections with projecting parts. The purpose of the projections is to push the articles during the forward motion of the material being transported. The transport returns by a different path from the one it follow in its advancement, and the material is left undisturbed until the next cycle begins. During this period of rest, while the transport is returning to its starting position, various operations can be performed sequentially. The selection of the particular transport mechanism best suited to any situation depends, to some degree, on the arrangement that can be obtained for driving the materials and the path desired. A slight amount of overtravel is always required so that the projection on the transport can clear the material when it is going into position for the advancing stroke.

The designs illustrated here have been selected from many sources and are typical of the simplest solutions of such problems. The paths, as indicated in these illustrations, can be varied by changes in the cams, levers, and associated parts. Nevertheless, the customary cut-and-try method might still lead to the best solution.

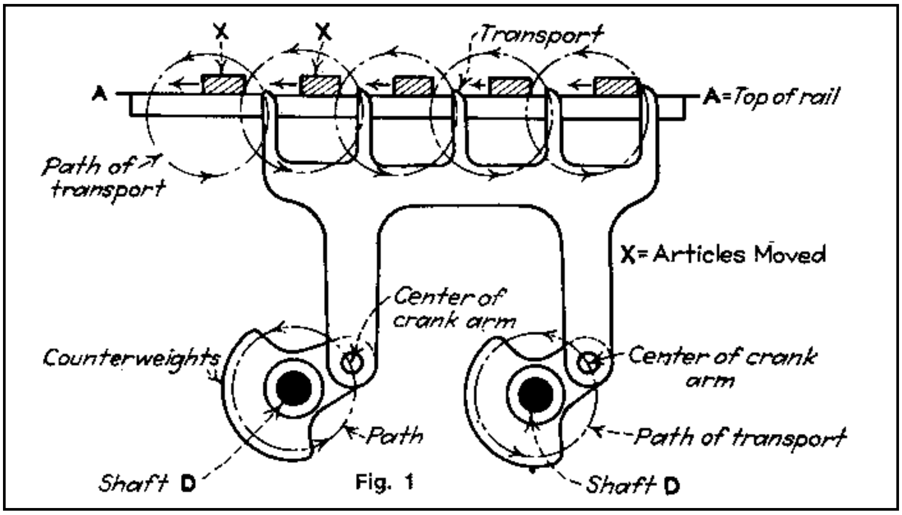


Fig. 1 In this design a rotary action is used. The shafts *D* rotate in unison and also support the main moving member. The shafts are carried in the frame of the machine and can be connected by either a link, a chain and sprocket, or by an intermediate idler gear between two equal gears keyed on the shafts. The rail *A-A* is fixed rigidly on the machine. A pressure or friction plate can hold the material against the top of the rail and prevent any movement during the period of rest.

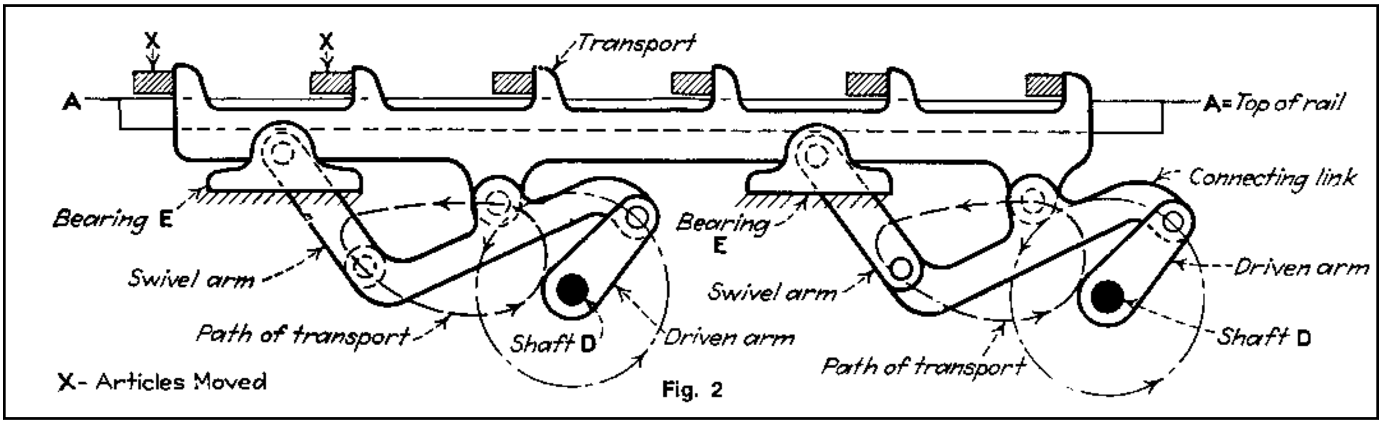


Fig. 2 Here is a simple form of linkage that imparts a somewhat "egg-shaped" motion to the transport. The forward stroke is almost a straight line. The transport is carried on the connecting links. As in the design of Fig. 1, the shafts *D* are driven in unison and are supported in the frame of the machine. Bearings *E* are also supported by the frame of the machine and the rail *A-A* is fixed.

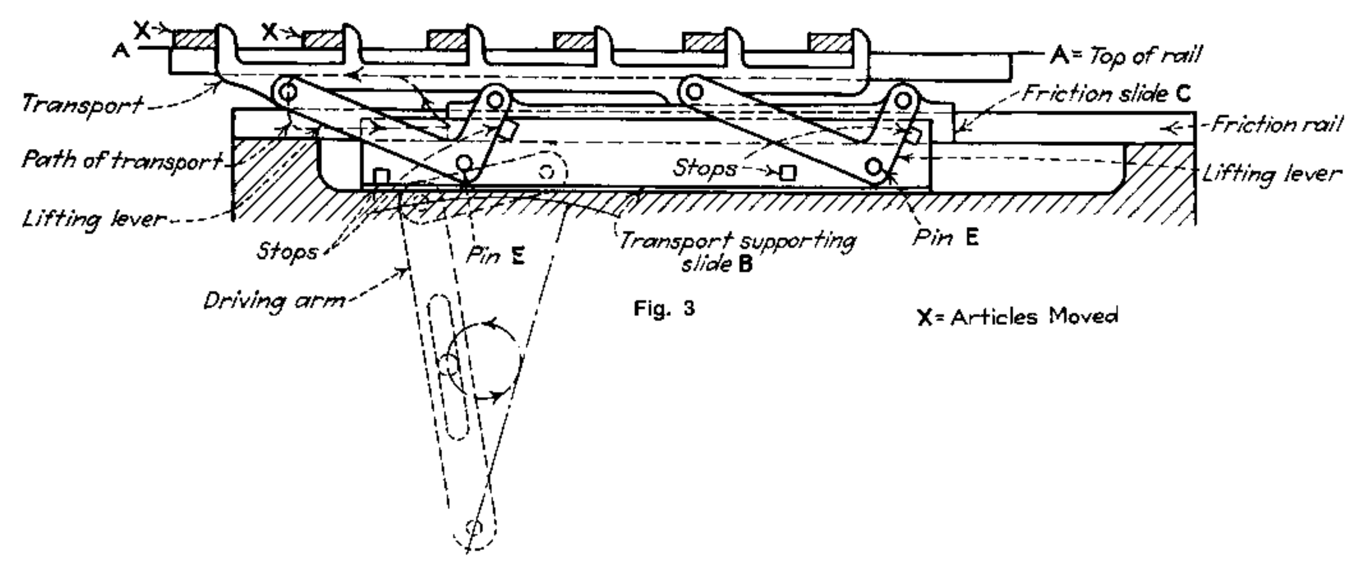


Fig. 3

Fig. 3 In another type of action, the forward and return strokes are accomplished by a suitable mechanism, while the raising and lowering is imparted by a friction slide. Thus it can be seen that as the transport supporting slide *B* starts to move to the left, the friction slide *C,* which rests on the friction rail, tends to remain at rest. As a result, the lifting lever starts to turn in a clockwise direction. This motion raises the transport which remains in its raised position against stops until the return stroke starts. At that time the reverse action begins. An adjustment should be provided to compensate for the friction between the slide and its rail. It can readily be seen that this motion imparts a long straight path to the transport.

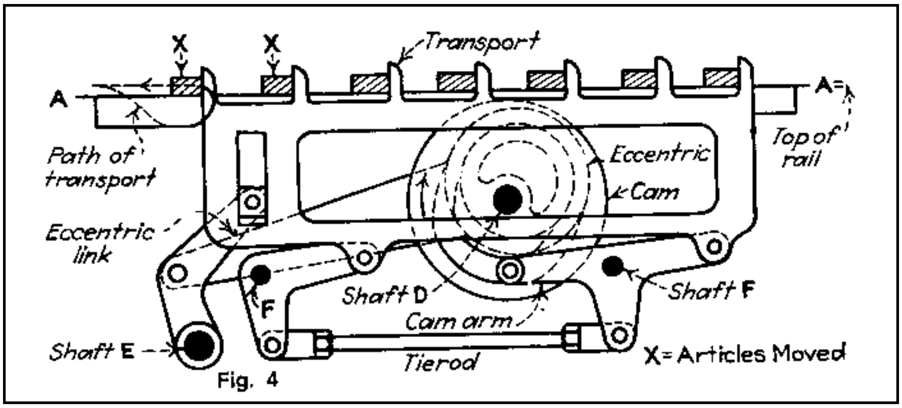


Fig. 4

Fig. 4 This drawing illustrates an action in which the forward motion is imparted by an eccentric while the raising and lowering of the transport is accomplished by a cam. The shafts, *F, E,* and *D* are positioned by the frame of the machine. Special bell cranks support the transport and are interconnected by a tierod.

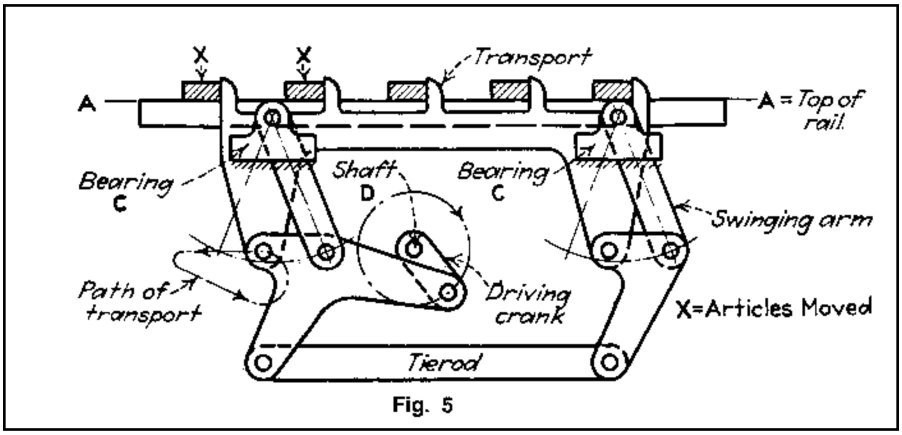


Fig. 5

Fig. 5 This is another form of transport mechanism based on a link motion. The bearings *C* are supported by the frame as is the driving shaft *D.*

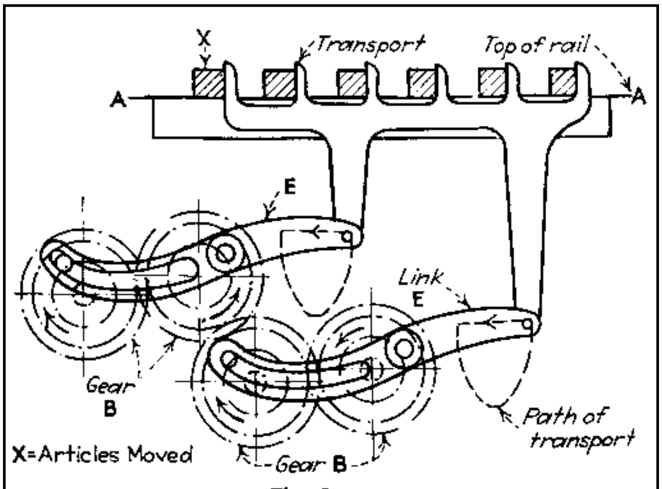


Fig. 6 An arrangement of interconnected gears with equal diameters that will impart a transport motion to a mechanism. The gear and link mechanism imparts both the forward motion and the raising and lowering motions. The gear shafts are supported in the frame of the machine.

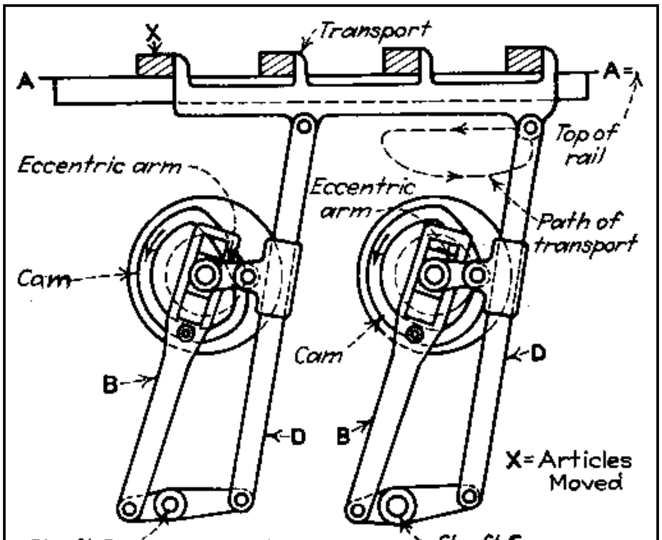
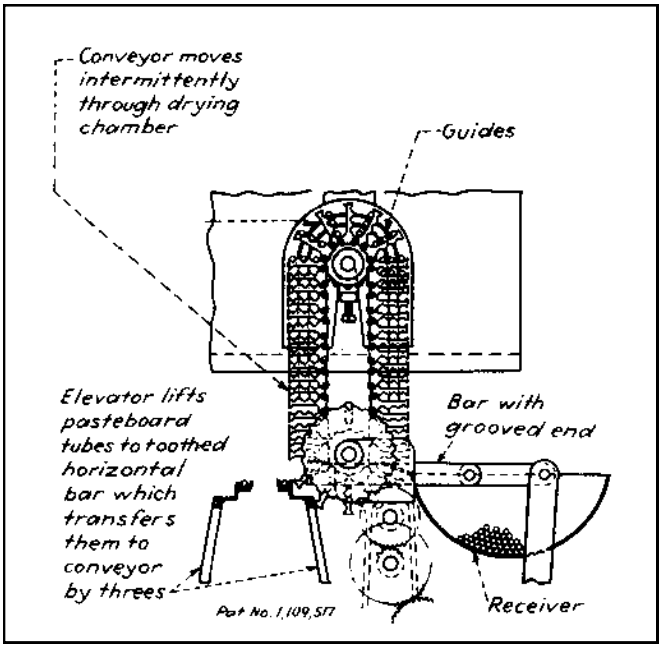


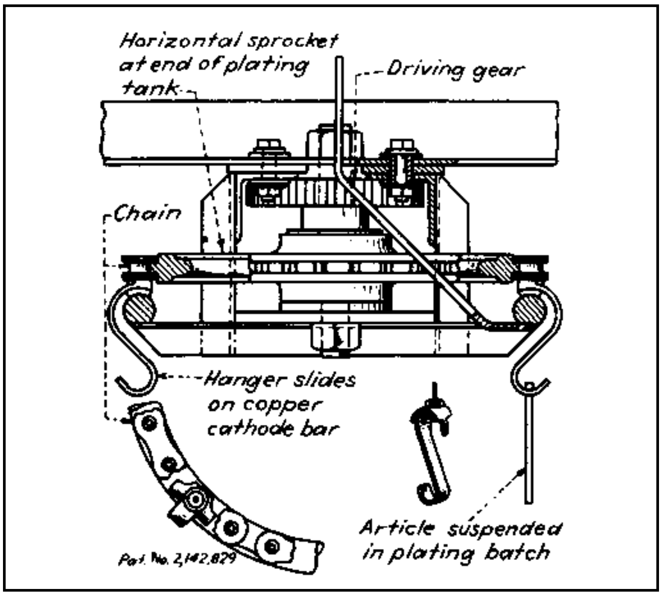
Fig. 7 In this transport mechanism, the forward an return strokes are accomplished by the eccentric arms, while the vertical motion is performed by the cams.

CONVEYOR SYSTEMS FOR PRODUCTION MACHINES

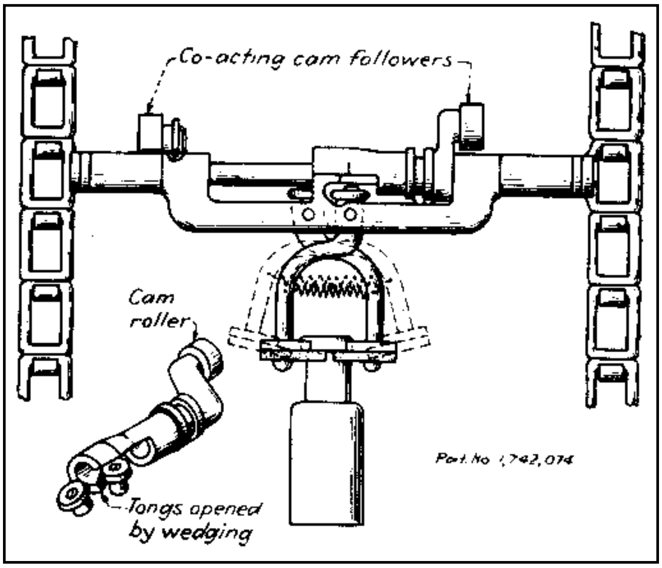
Conveyor systems can be divided into two classes: those that are a part of a machine for processing a product, and those that move products in various stages of fabrication. The movement might be from one worker to another or from one part of a plant to another. Most of the conveyors shown here are components in processing machines. Both continuous and intermittently moving equipment are illustrated.



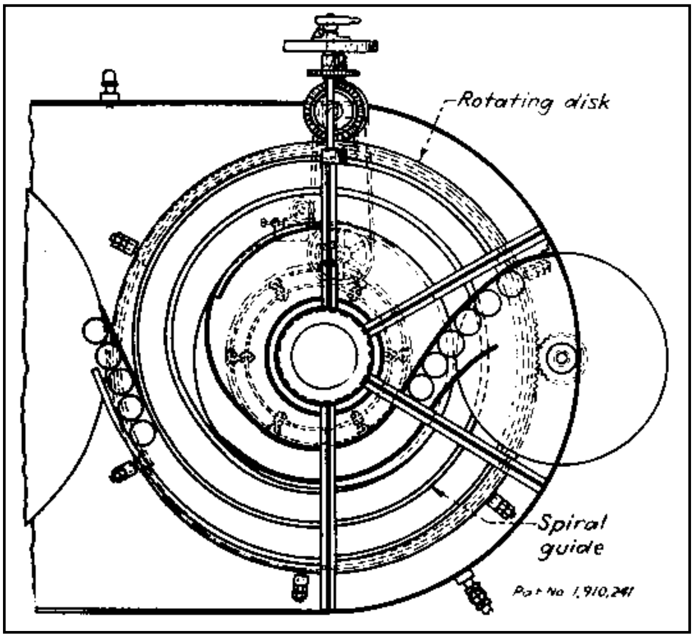
Intermittently moving grooved bar links convey pasteboard tubes through a drying chamber.



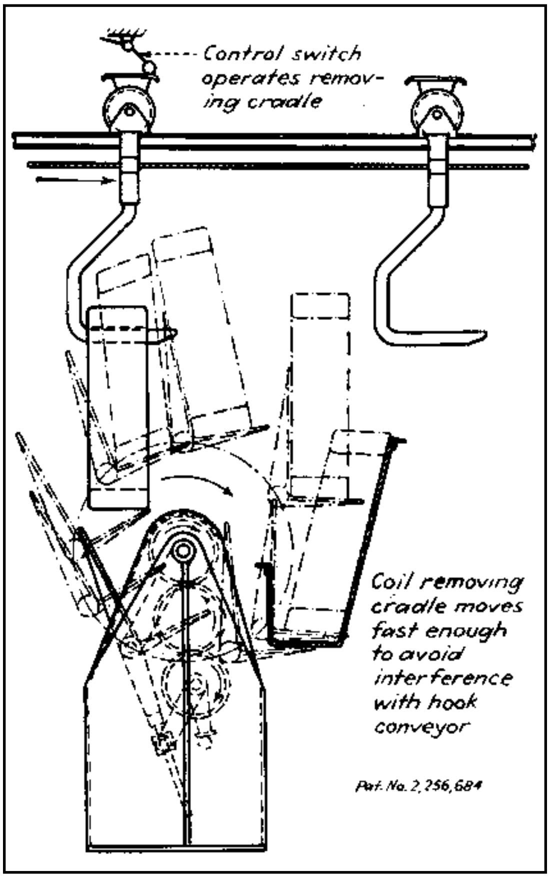
Hooks on a chain-driven conveyor move articles through a plating bath.



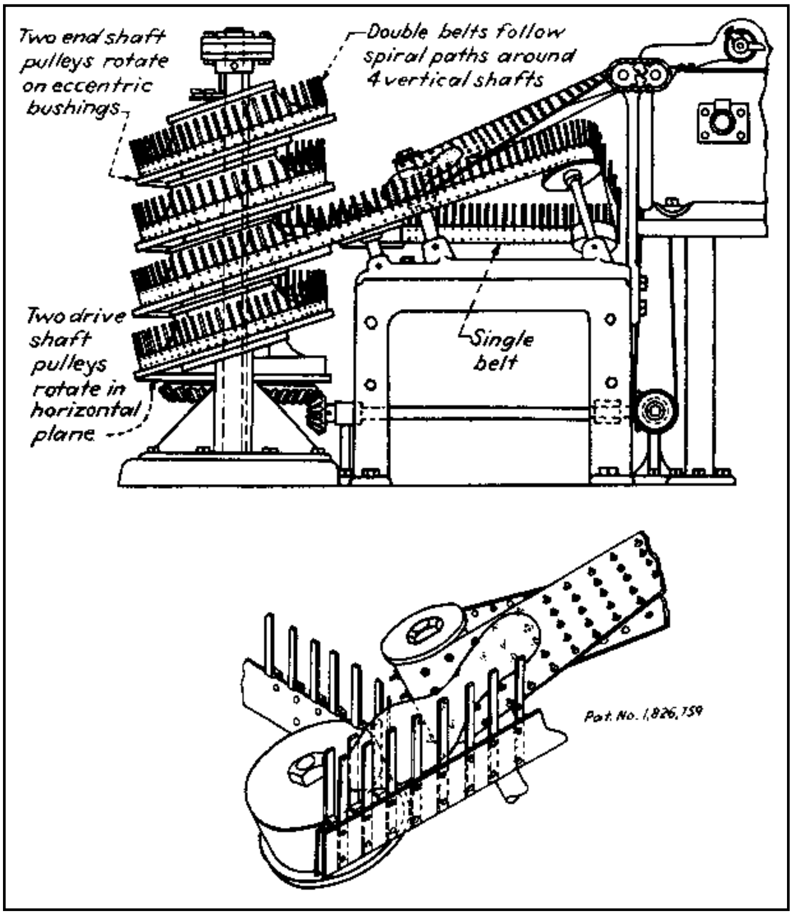
Co-acting cams in the paths of follower rollers open and close tongs over bottlenecks by a wedging action.



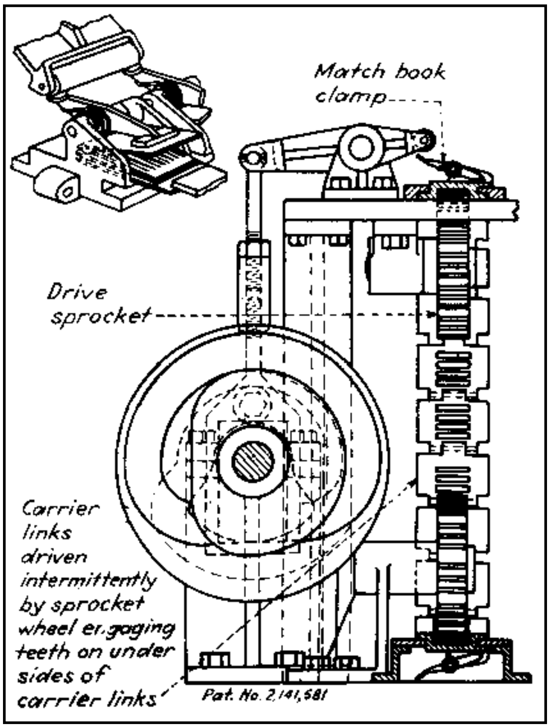
A rotating disk carries food cans in a spiral path between stationary guides for presealing heat treatment.



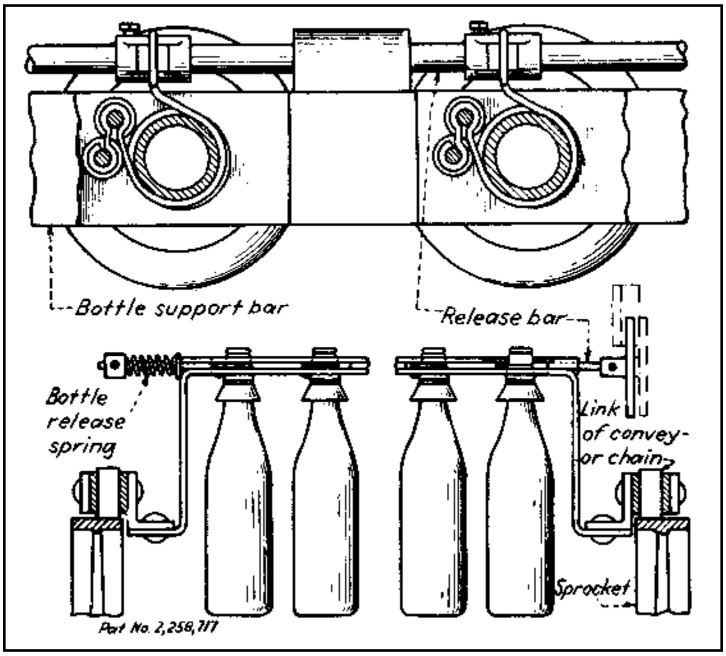
Hooks on a cable-driven conveyor and an automatic cradle for removing coils.



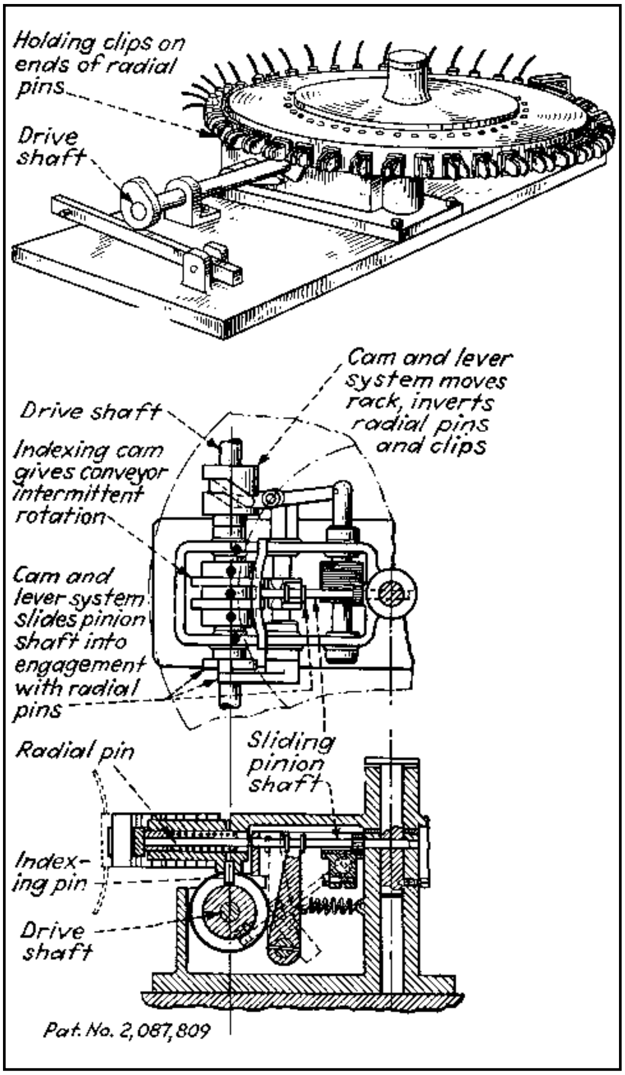
A double belt sandwiches shoe soles during their cycle around a spiral system and then separates to discharge the soles.



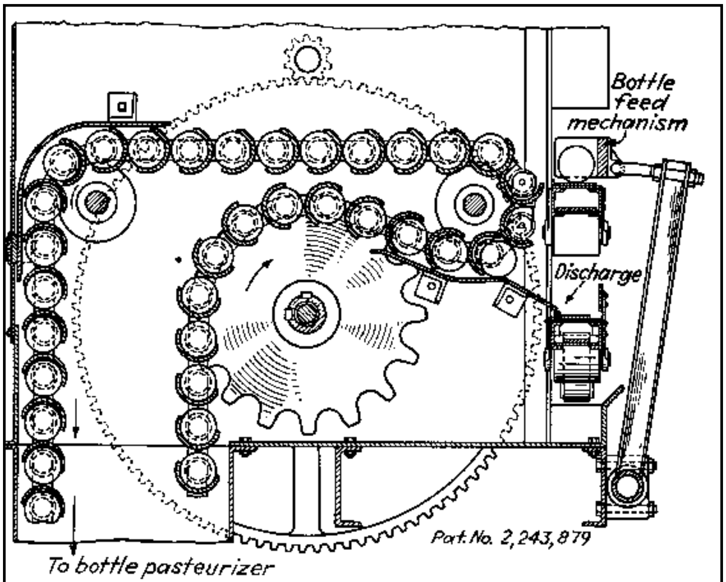
A matchbook carrier links with holding clips that are moved intermittently by sprockets.

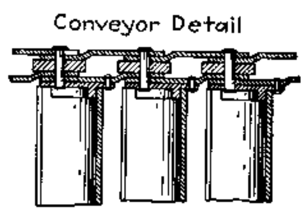


One of several possible kinds of bottle clips with release bars for automatic operation.

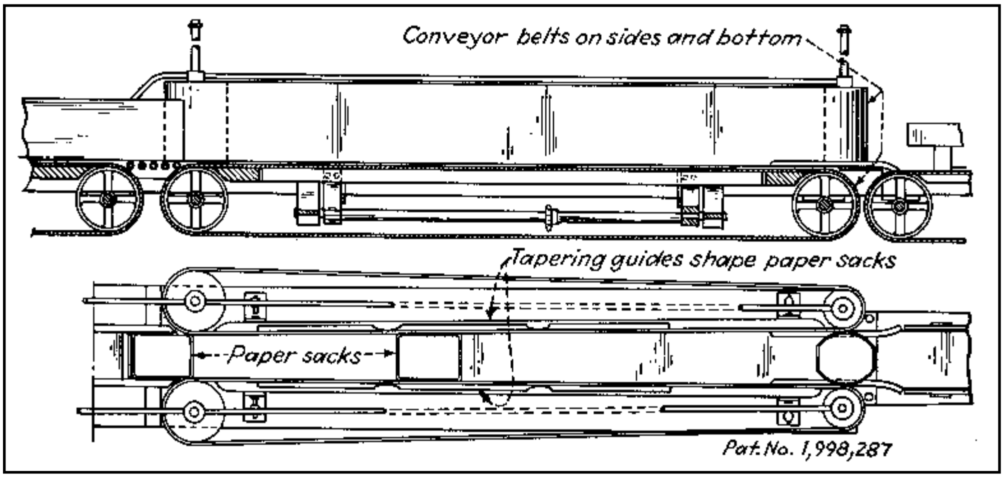


An intermittent rotary conveyor inverts electrical capacitors that are to be sealed at both ends by engaging radial pins which have holding clips attached.

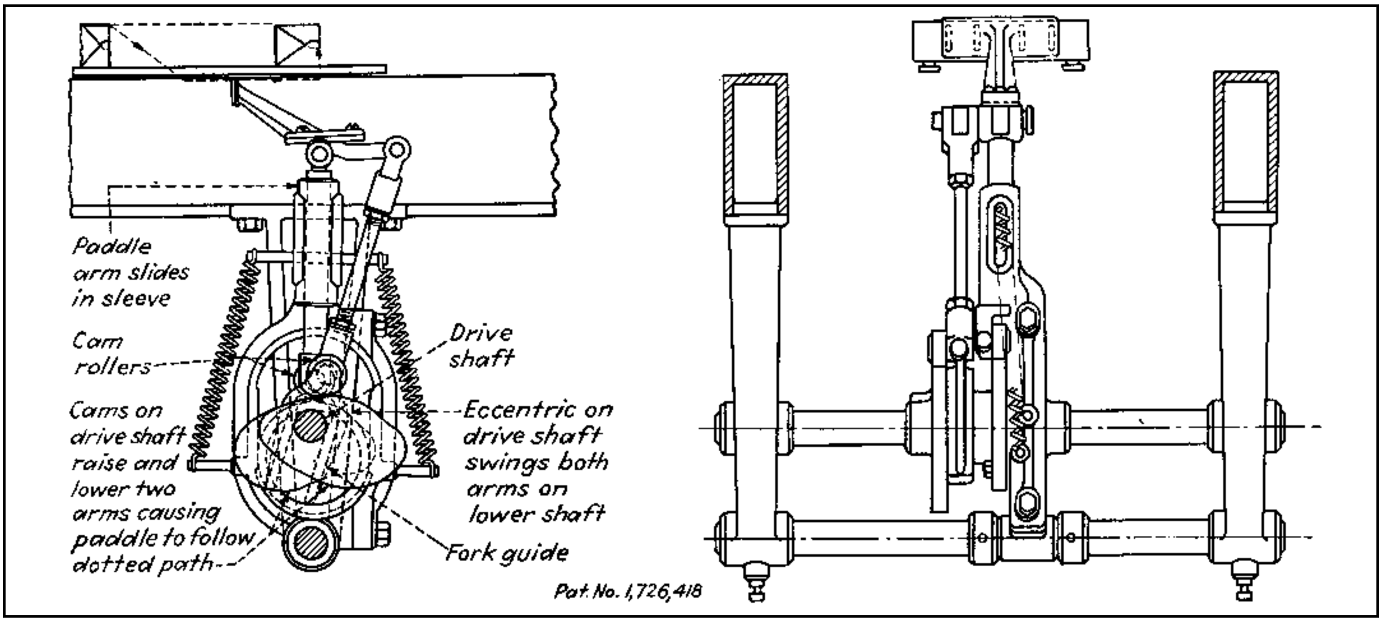




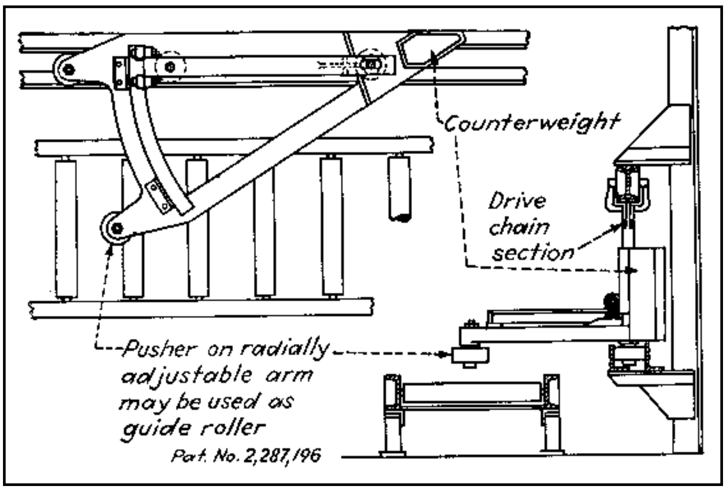
This pasteurizer carrier links lock bottles in place on straight ways.



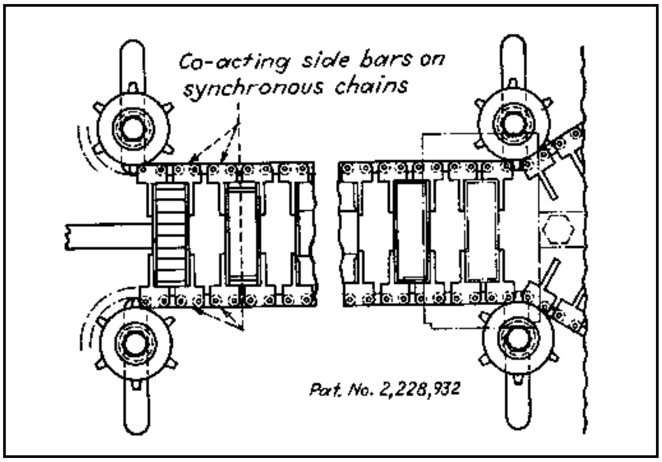
The wedging action of the side belts shapes paper sacks for wrapping an packing.



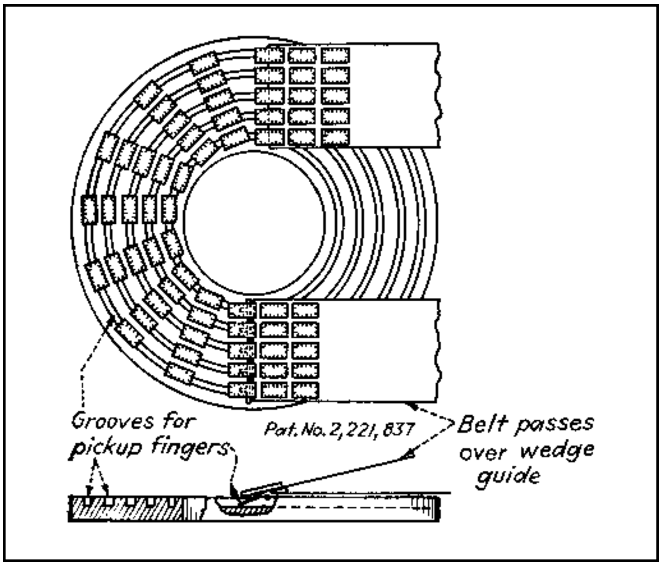
A reciprocating pusher plate is activated by an eccentric disk and two cams on a drive shaft.



A pusher-type conveyor can have a drive on either side.



Synchronous chains with side arms grasp and move packages.



A rotary conveyor transfers articles from one belt conveyor to another without disturbing their relative positions.

TRAVERSING MECHANISMS FOR WINDING MACHINES

The seven mechanisms shown are parts of different yarn- and coil-winding machines. Their fundamentals, however, might be applicable to other machines that require similar changes of motion. Except for the leadscrews found on lathes, these seven represent the operating principles of all well-known, mechanical traversing devices.

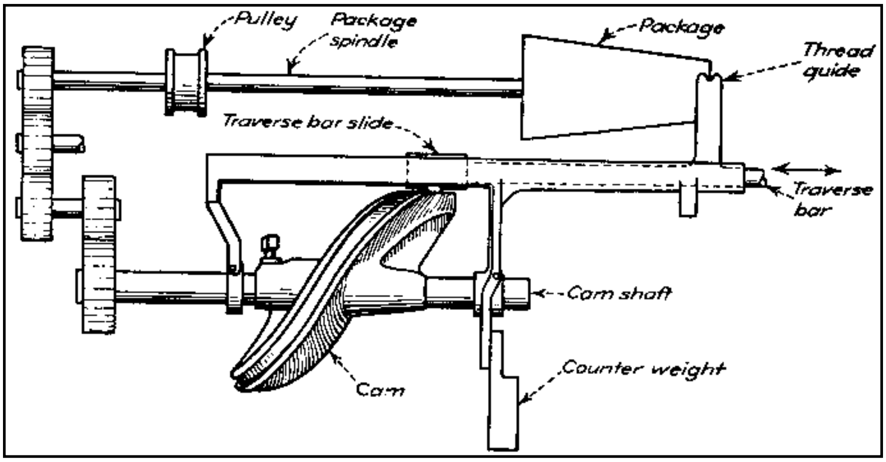


Fig. 1 A package is mounted on a belt-driven shaft on this precision winding mechanism. A camshaft imparts reciprocating motion to a traverse bar with a cam roll that runs in a cam groove. Gears determine the speed ratio between the cam and package. A thread guide is attached to the traverse bar, and a counterweight keeps the thread guide against the package.

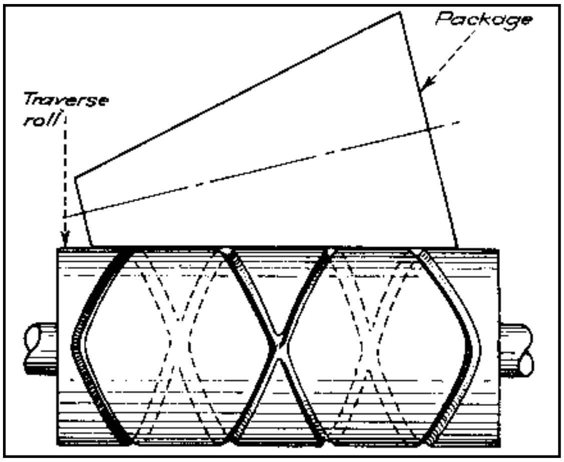
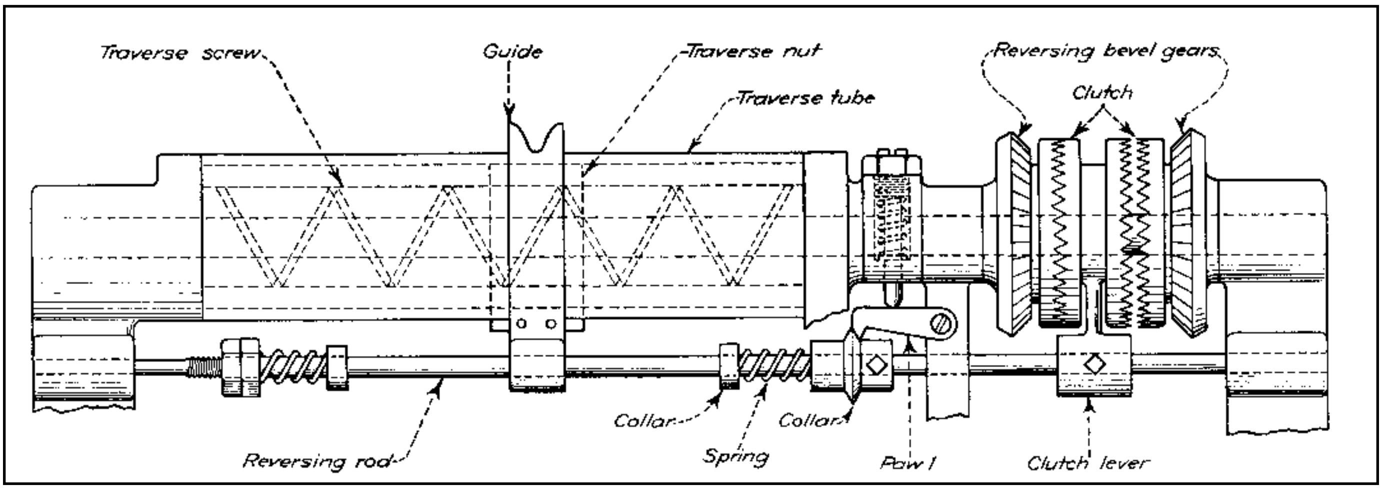


Fig. 2 A package is friction-driven from a traverse roll. Yarn is drawn from the supply source by traverse roll and is transferred to a package from the continuous groove in the roll. Different winds are obtained by varying the grooved path.

Fig. 3 Reversing bevel gears that are driven by a common bevel gear drive the shaft carrying the traverse screw. A traverse nut mates with this screw and is connected to the yarn guide. The guide slides along the reversing rod. When the nut reaches the end of its travel, the thread guide compresses the spring that actuates the pawl and the reversing lever. This action engages the clutch that rotates the traverse screw in the opposite direction. As indicated by the large pitch on the screw, this mechanism is limited to low speeds, but it permits longer lengths of traverse than most of the others shown.



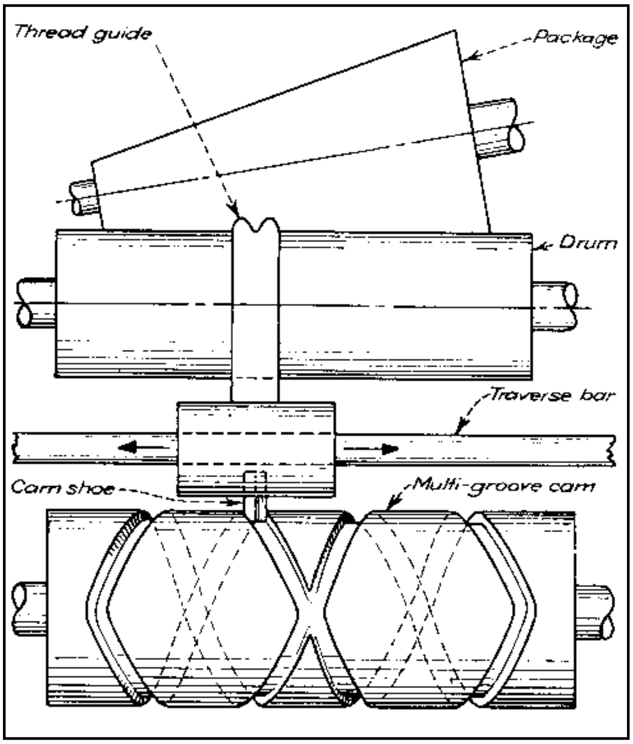


Fig. 4 A drum drives the package by friction. A pointed cam shoe, which pivots in the bottom side of the thread guide assembly, rides in cam grooves and produces a reciprocating motion of the thread guide assembly on the traverse bar. Plastic cams have proved to be satisfactory even with fast traverse speeds. Interchangeable cams permit a wide variety of winding methods.

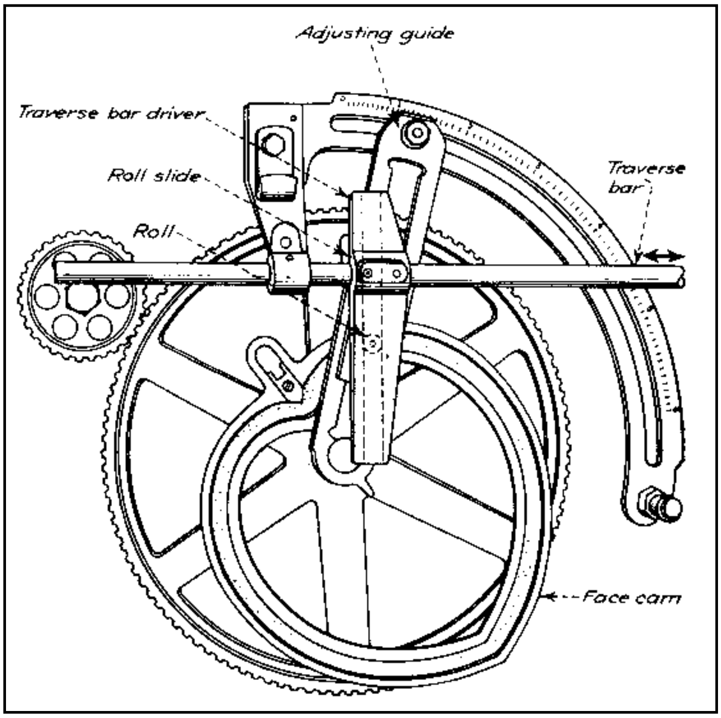


Fig. 5 A roll that rides in a heart-shaped cam groove engages a slot in a traverse bar driver which is attached to the traverse bar. Maximum traverse is obtained when the adjusting guide is perpendicular to the driver. As the angle between the guide and driver is decreased, traverse decreases proportionately. Inertia effects limit this mechanism to slow speeds.

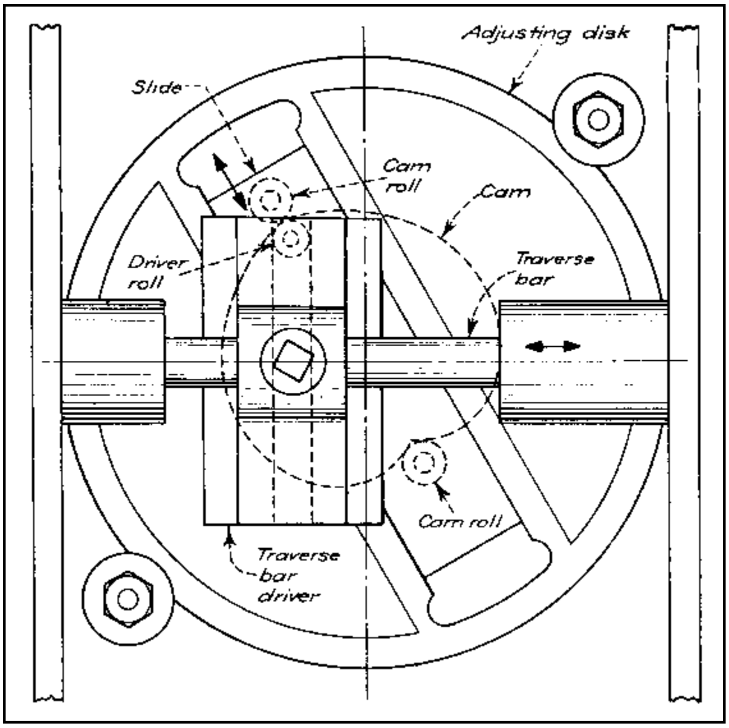


Fig. 6 The two cam rolls that engage this heart-shaped cam are attached to the slide. The slide has a driver roll that engages a slot in the traverse bar driver. Maximum traverse (to the capacity of the cam) occurs when the adjusting disk is set so the slide is parallel to the traverse bar. As the angle between the traverse bar and slide increases, traverse decreases. At 90º traverse is zero.

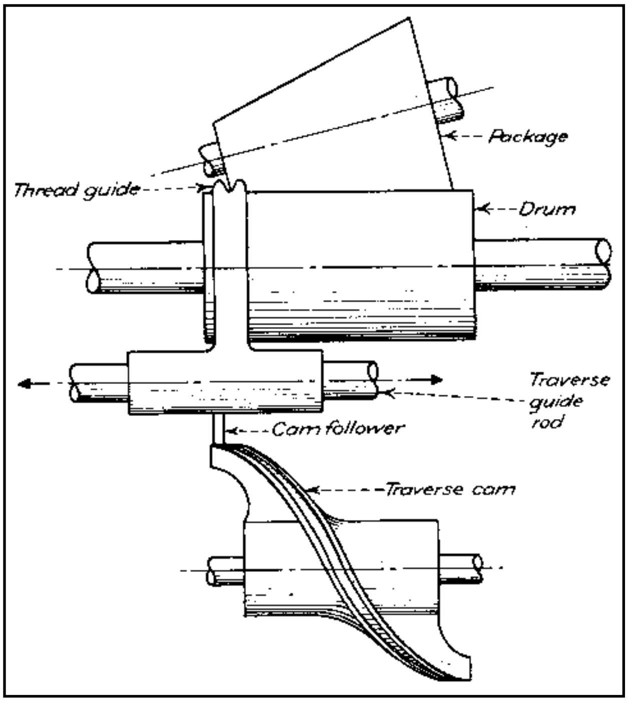
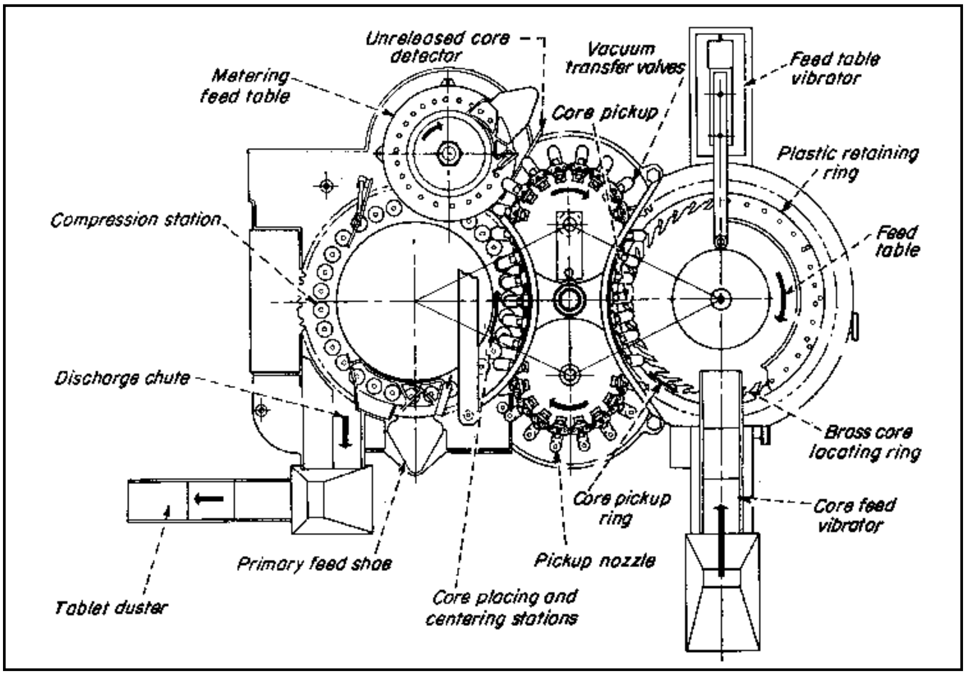


Fig. 7 A traverse cam imparts reciprocating motion to a cam follower that drives thread guides on traverse guide rods. The package is friction driven from the drum. Yarn is drawn from the supply source through a thread guide and transferred to the drum-driven package. The speed of this mechanism is determined by the weight of its reciprocating parts.

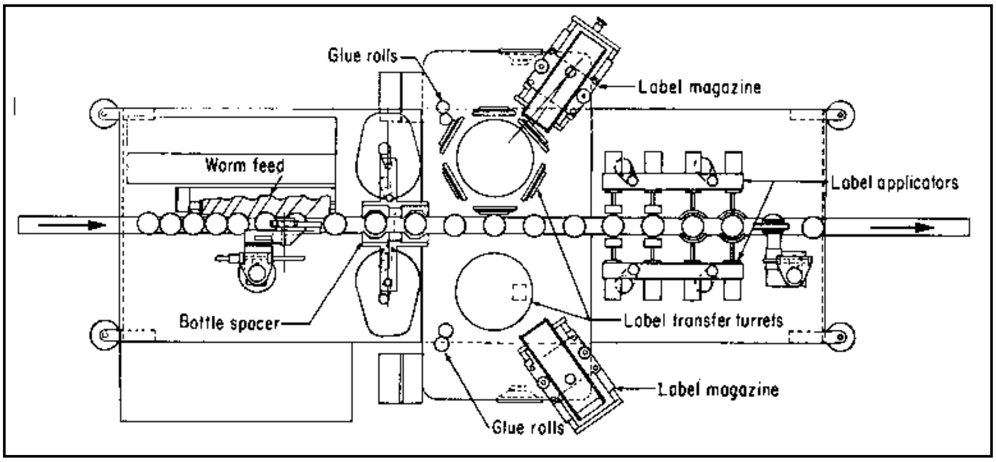
VACUUM PICKUP POSITIONS PILLS

This pickup carries tablet cores to moving dies, places cores accurately in coating granulation, and prevents the formation of tablets without cores.



Cores are hopper fed to a rotating feeder disk through a tablet duster. This disk is vibrated clockwise under a slotted pick-up ring which rotates counter-clockwise. Each slot in the pickup ring holds two cores and lets broken tablets fall through to an area under the feeder table. Cores are picked from ring slots, carried to tablet press dies, and deposited in dies by vacuum nozzles fastened to a chain driven by the press die table. This chain also drives the pickup ring to synchronize the motion of ring slots and pickup nozzles. Coating granulation is fed into the dies ahead of and after the station where a vacuum pickup deposits a core in each die. Compressing rolls are at the left side of the machine. The principal design objective here was to develop a machine to apply dry coatings at speeds that lowered costs below those of liquid coating techniques.

MACHINE APPLIES LABELS FROM STACKS OR ROLLERS



The flow of containers through this labeler is shown by the top-view drawing of the machine. Bottle spacers ensure that containers remain 7½ in. apart on the conveyor. Dual label-transfer turrets allow for the simultaneous application of front and back labels.

This labeling machine can perform either conventional glue-label application or it can heat-seal labels in cut or roll form. The machine labels the front and back of round or odd-shaped containers at speeds of 60 to 160 containers per minute. The containers handled range from 1 in. diameter or thickness to 4¼ in. diameter by 5½ in. wide. Container height can vary from 2 to 14 inches. The unit handles labels ranging from ⅞ to 5½ in. wide and ⅞ to 6½ in. high. The label hopper is designed for labels that are generally rectangular in shape, although it can be modified to handle irregular shapes. Provision has been made in design of the unit, according to the manufacturer, to allow labels to be placed at varying heights on the containers. The unit's cut-and-stacked label capacity is 4,500. An electric eye is provided for cutting labels in web-roll form.

HIGH-SPEED MACHINES FOR ADHESIVE APPLICATIONS

Viscous liquid adhesives are used to glue fabrics and paper, apply paper labels, make cardboard and wooden boxes and shoes, and bind books. Specially designed machines are required if the application of adhesives with different characteristics is to be satisfactorily controlled. The methods and machines shown here have been adapted to the application of adhesives in mass production. They might also work well for the application of liquid finishes such as primers, paint, and lacquer.

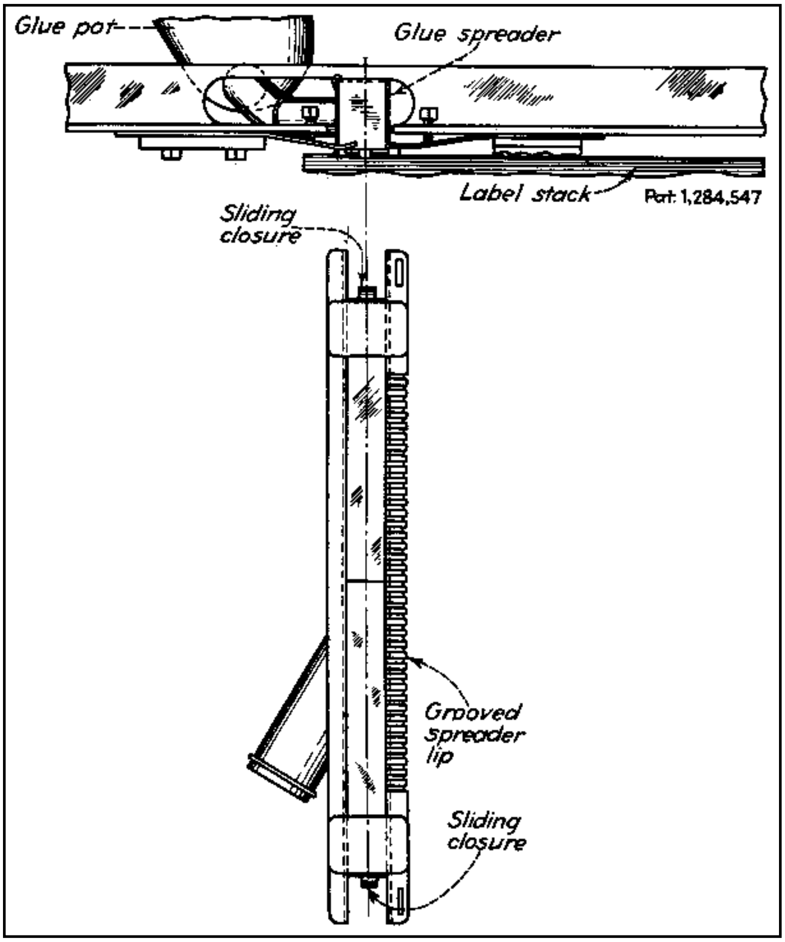


Fig. 1 A gravity spreader has an open bottom and a grooved lip.

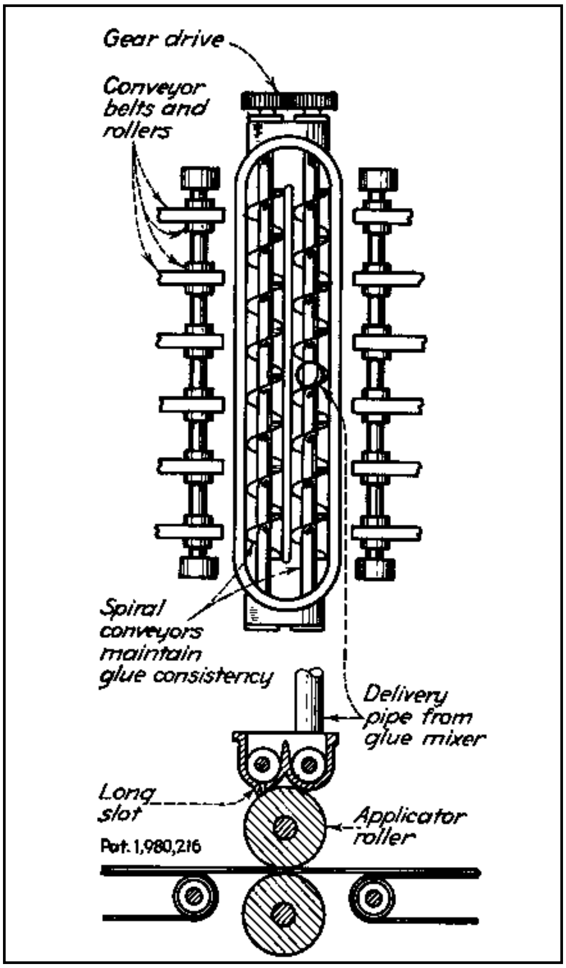


Fig. 2 Spiral conveyors feed the applicator roller by the force or gravity.

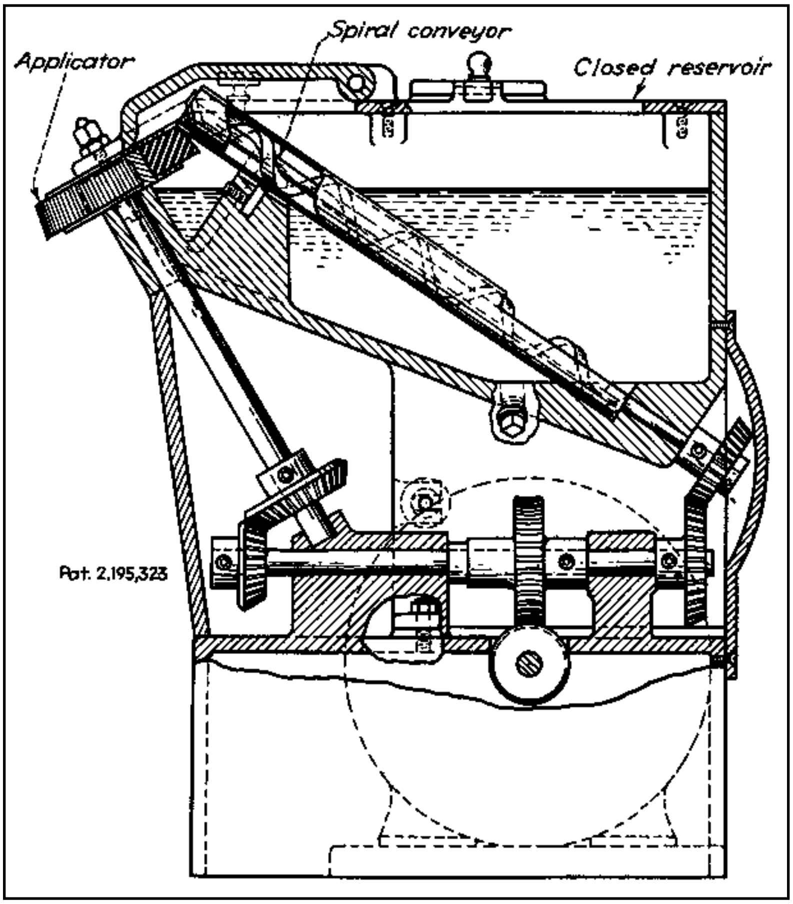


Fig. 3 An applicator wheel is fed by a spiral conveyor.

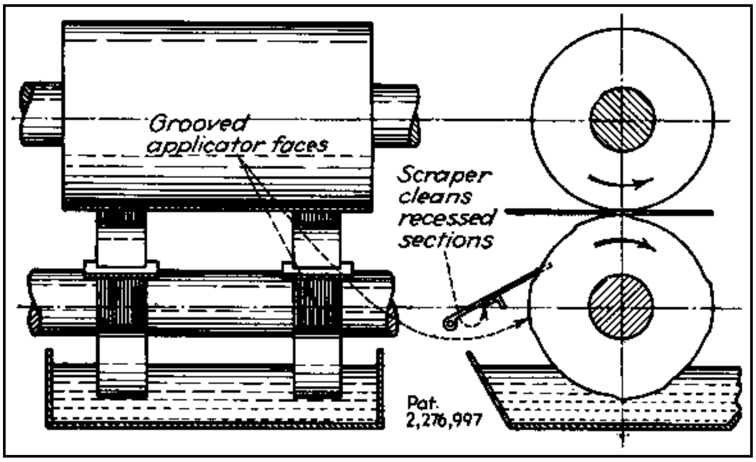


Fig. 4 An adhesive pattern produced by raised faces on the applicator roll.

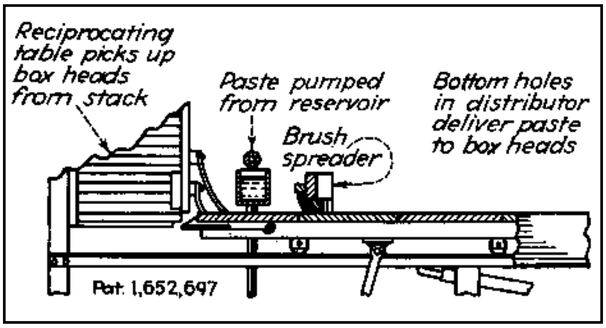


Fig. 5 A gravity spreader with flow from its bottom holes.

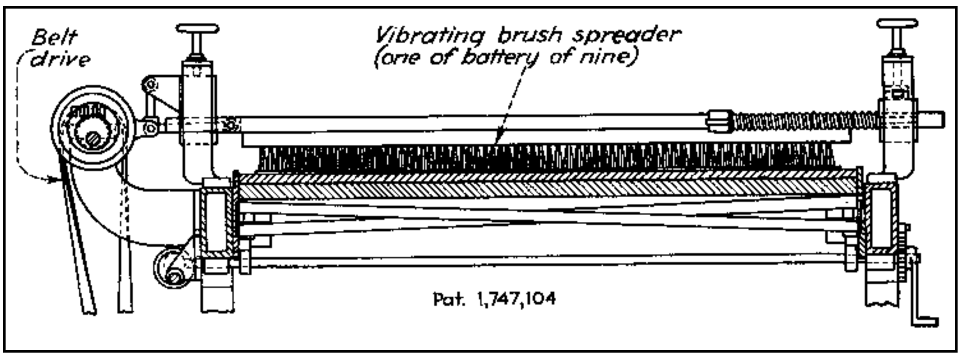


Fig. 6 Vibrating brushes spread the coating after the application by a cylindrical brush.

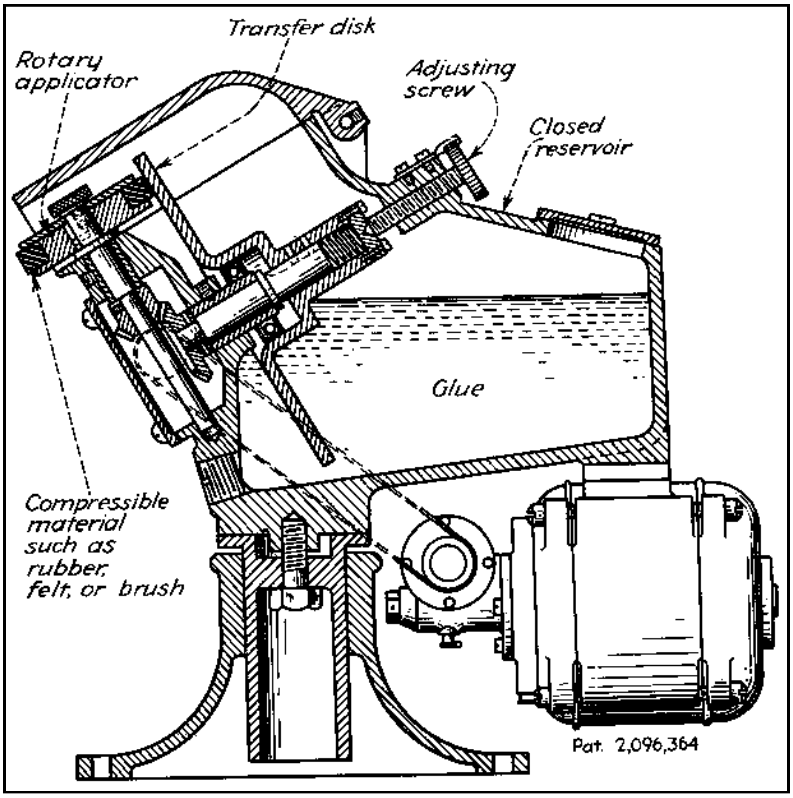


Fig. 7 This applicator wheel is fed by a transfer disk.

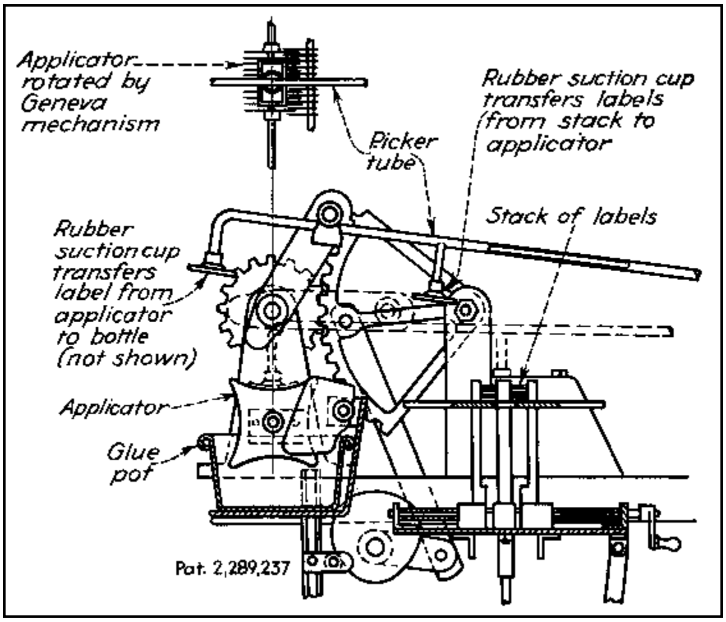


Fig. 8 This applicator surface, consisting of a series of plate edges, is rotated by a Geneva mechanism in the glue pot.

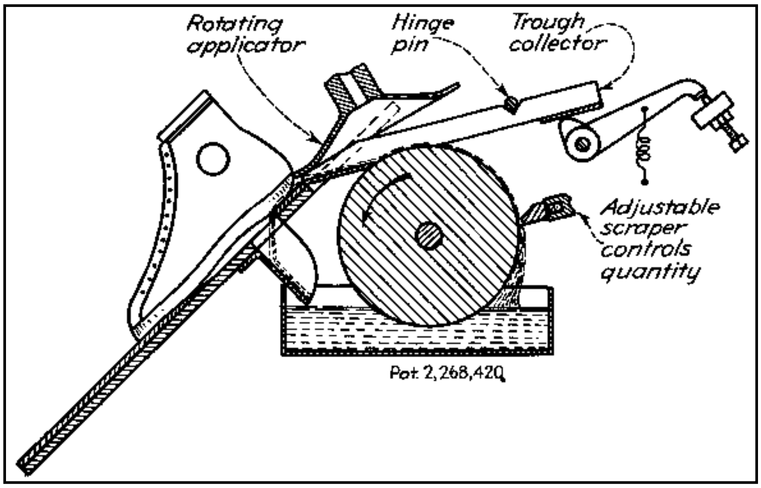


Fig. 9 A rotating applicator disk fed by a trough collector on a transfer drum.

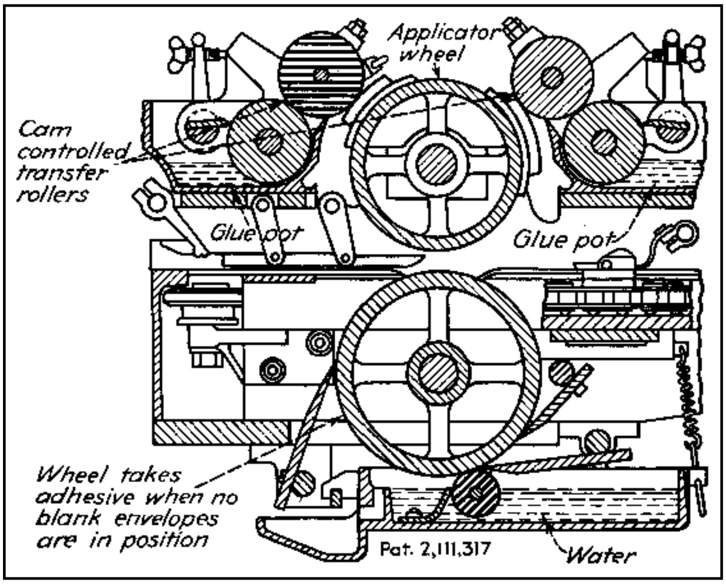


Fig. 10 Cam controlled transfer rollers supply applicator wheel pads with two kinds of adhesive.

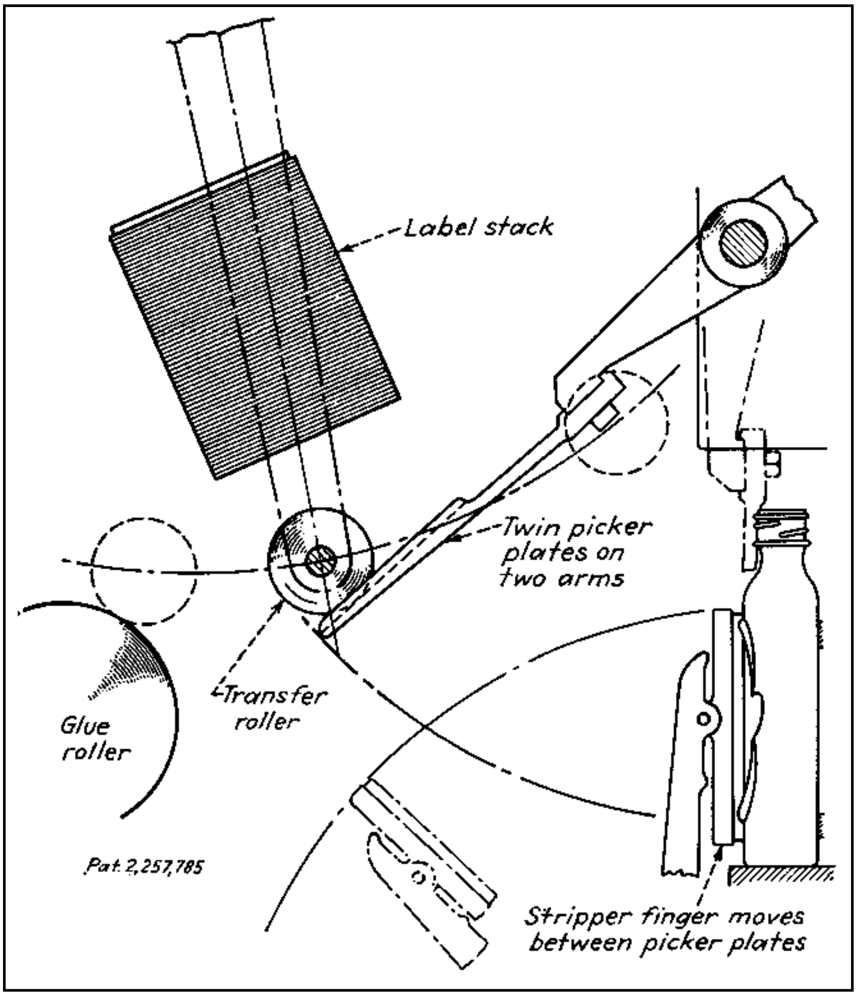


Fig. 11 The bottom label is spread with glue by two abutting glue-coated picker plates, which separate during contact with label stack, then carry the label to the bottle.

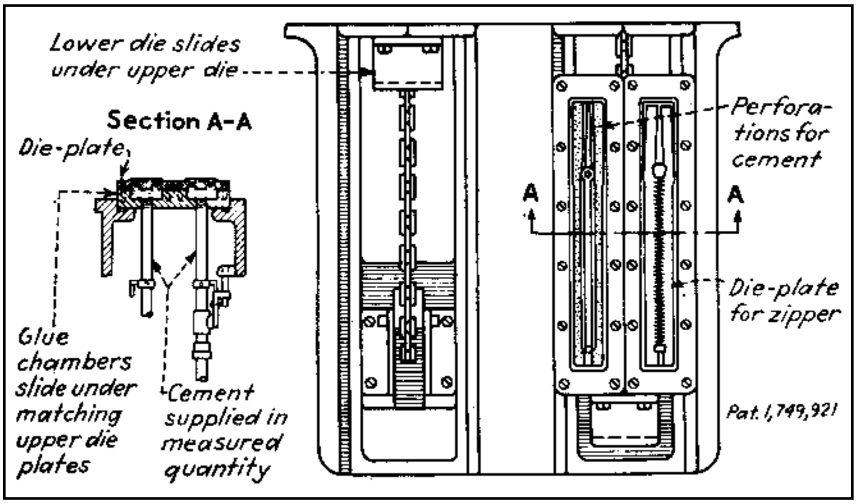


Fig. 12 Measured quantities of cement are forced through perforations in specially designed upper and lower die plates, which are closed hydraulically over zippers. Only the lower die is shown.

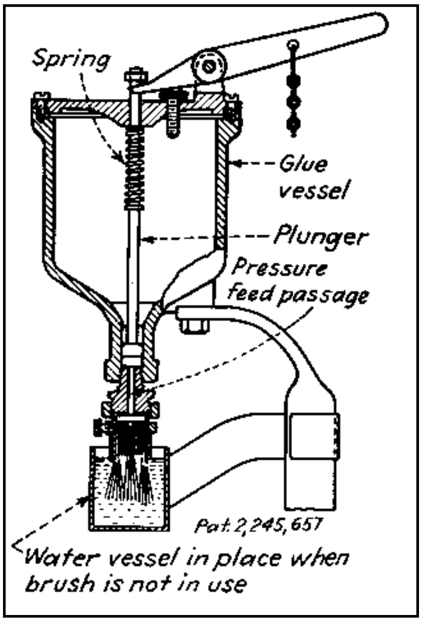


Fig. 13 A brush applicator is fed through passages between bristle tufts by a spring-operated plunger.

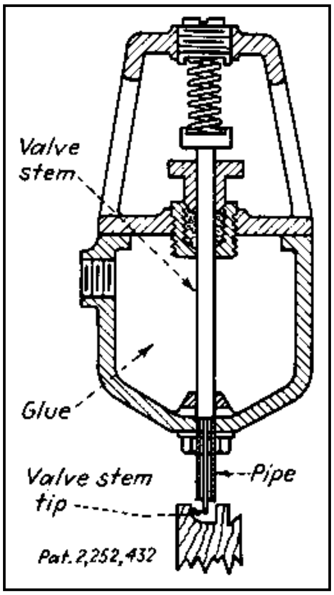


Fig. 14 A shoulder on a valve stem in a glue chamber retains glue until pressure on the tip opens the bottom valve.

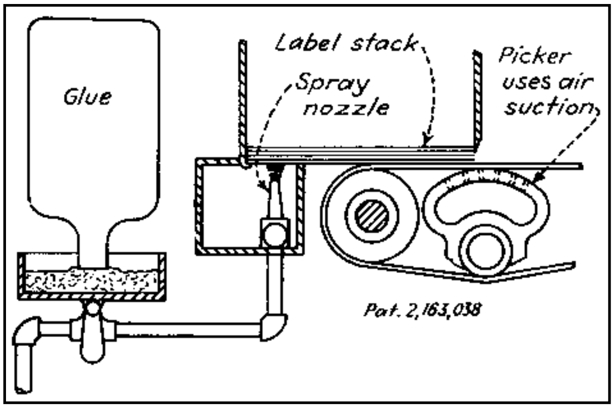


Fig. 15 Glue is applied to envelopes by a spray nozzle.

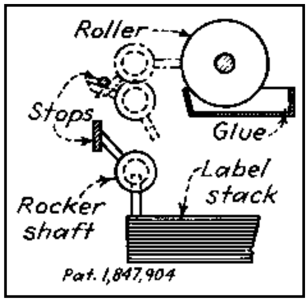


Fig. 16 A rocker shaft on a rack, which is moved vertically by a sector gear, carries glue on a contact bar from the roll to the label stack.

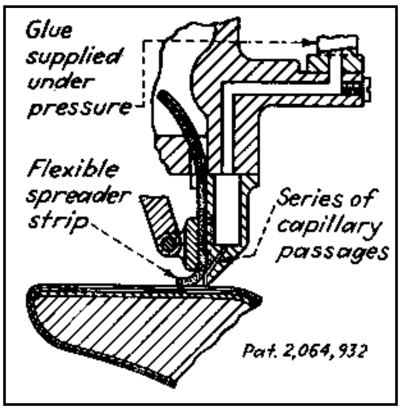


Fig. 17 Glue is extruded through nozzle on work.

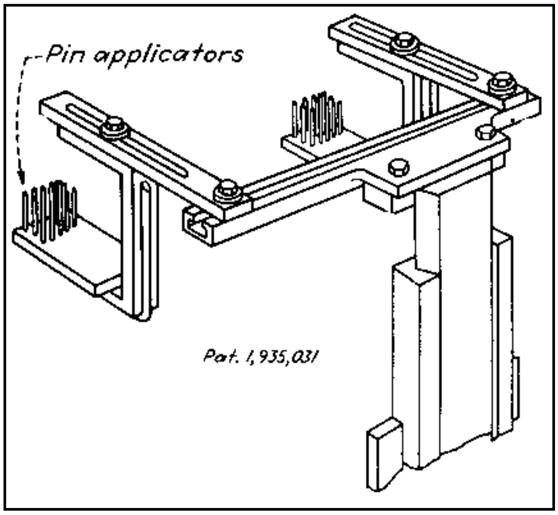


Fig. 18 Pin applicators reciprocate vertically, first immersing themselves in glue, then contacting the undersides of carton flaps in a desired pattern.

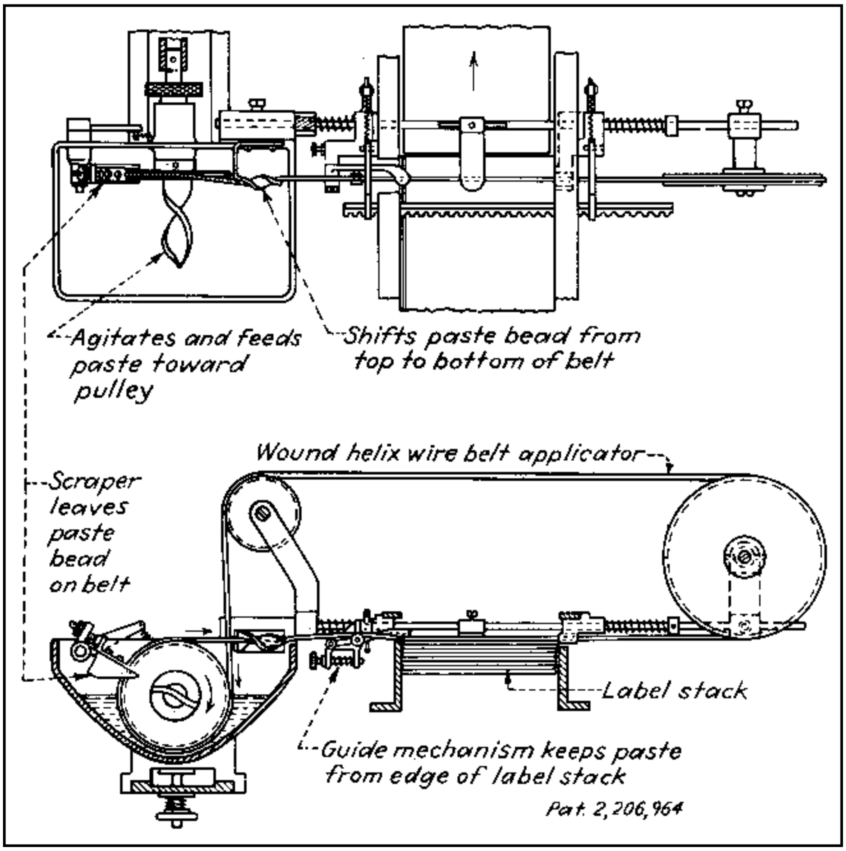


Fig. 19 A paste belt applicator passes around the pulley in a pastepot and slides over the label stack.

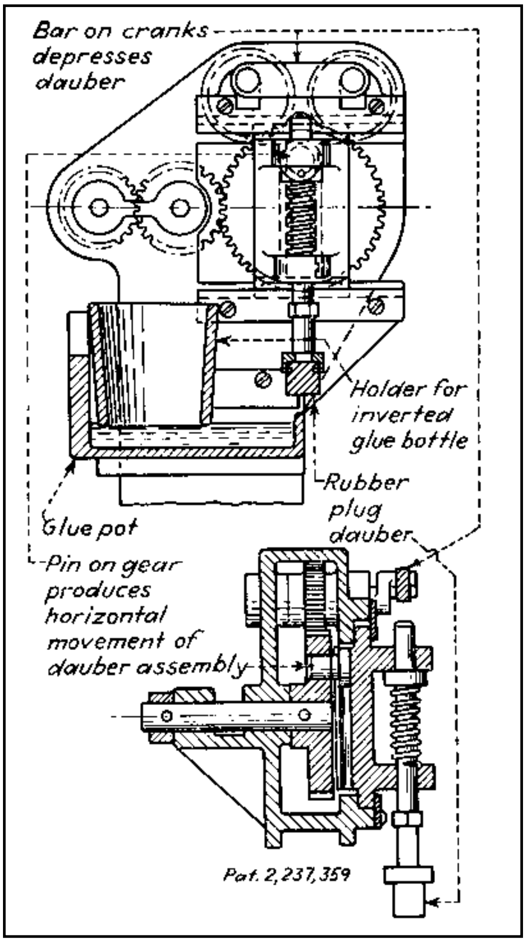


Fig. 20 A dauber assembly is moved horizontally between a glue pot and work by an eccentric pin on a gear. Vertical movements are produced by a crank-operated bar over the dauber shaft.

AUTOMATIC STOPPING MECHANISMS FOR FAULTY MACHINE OPERATION

Automatic stopping mechanisms that prevent machines from damaging themselves or destroying work in process are based on the principles of mechanics, electricity, hydraulics, and pneumatics.

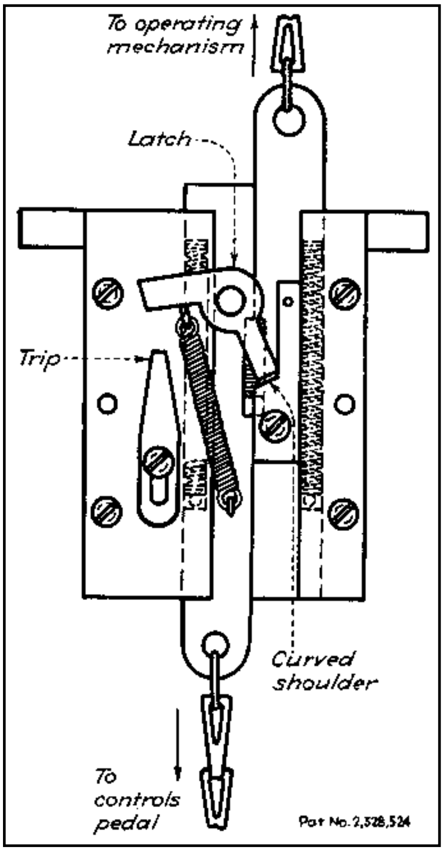


Fig. 1 A repetition of the machine cycle is prevented if a pedal remains depressed. The latch carried by the left slide pushes the right slide downward with a curved shoulder until the latch is disengaged by trip member.

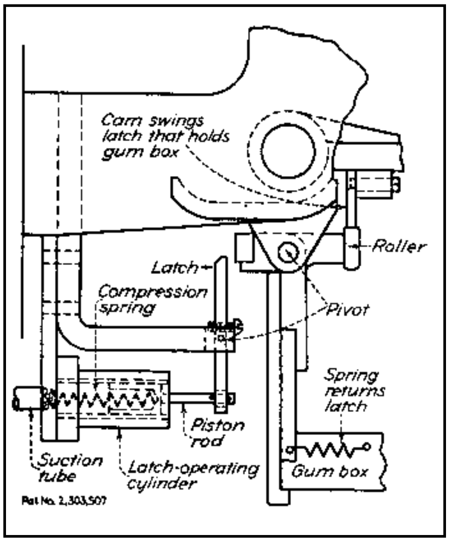


Fig. 2 The gumming of the suction picker and label carrier when the label is not picked up by the suction is prevented by insufficient suction on a latch-operating cylinder, caused by open suction holes on the picker. When a latch-operating cylinder does not operate, the gum box holding latch returns to its holding position after cyclic removal by the cam and roller, thus preventing the gum box and rolls from rocking to make contact with the picker face.

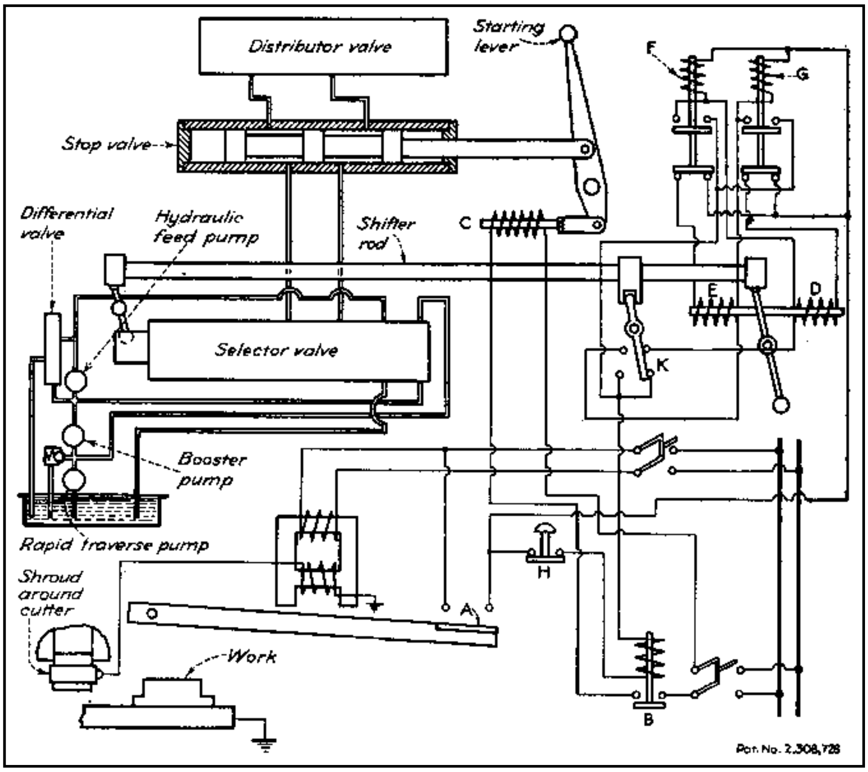


Fig. 3 Damage to the milling cutter, work, or fixtures is prevented by the shroud around the cutter. Upon contact with the work, the shroud closes the electric circuit through the relay, thus closing contact A. This causes contact B to close, thus energizing relay C to operate a stop valve. It also closes a circuit through relay D, thus reversing the selector valve by means of a shifter rod so that bed travel will reverse on starting. Simultaneously, relay F opens the circuit of relay E and closes a holding circuit that was broken by the shifter lever at K. Relay G also closes a holding circuit and opens a circuit through relay D. The starting lever, released by pushbutton H, releases contact A and returns the circuit to normal. If contact is made with the shroud when the bed travel is reversed, interchange the positions of D and E, with F and G in the sequence of operations.

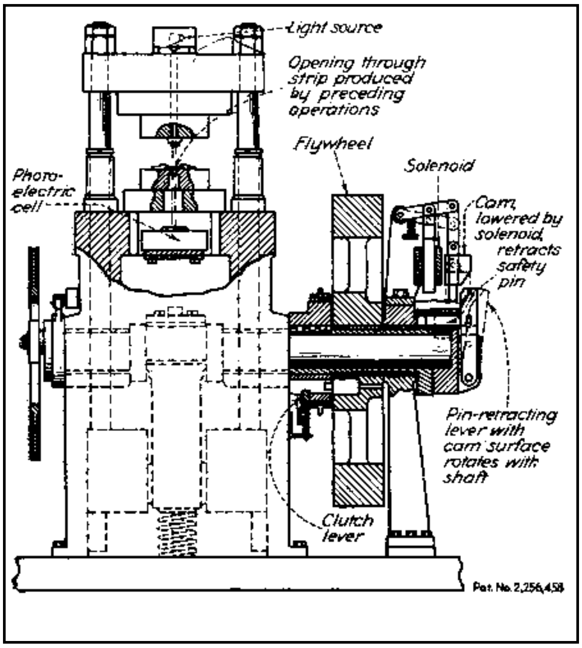


Fig. 4 A high-speed press is stopped when a metal strip advances improperly. A hole punched in the strip fails to match while the opening in the die block to permit a light beam to pass. When the light beam to the photoelectric cell is blocked, the solenoid which withdraws the clutch pin is activated.

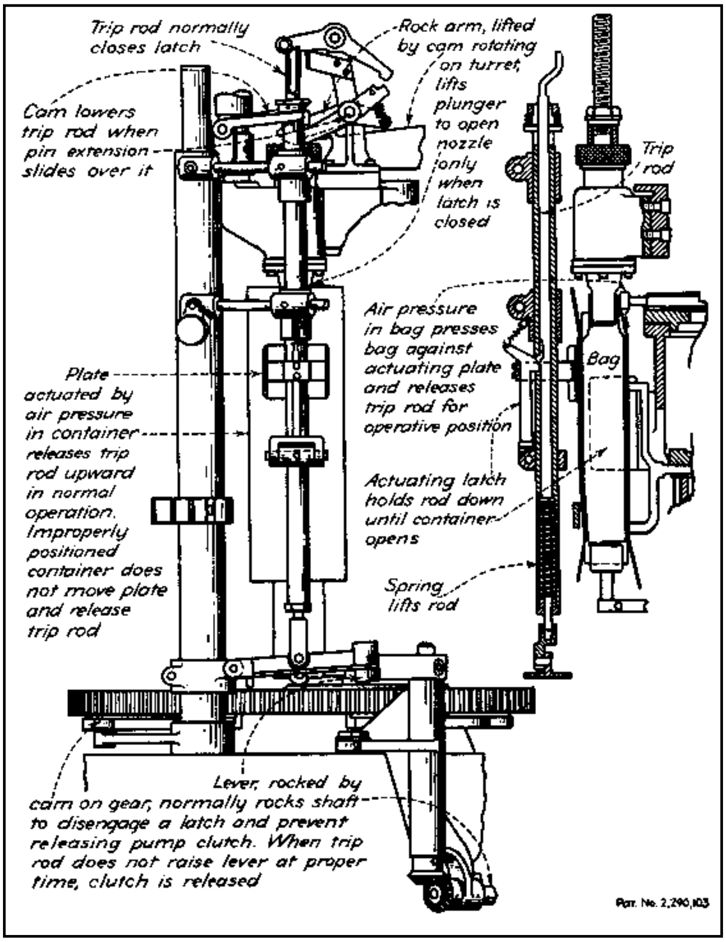


Fig. 6 A nozzle on the packaging machine does not open when the container is improperly positioned.

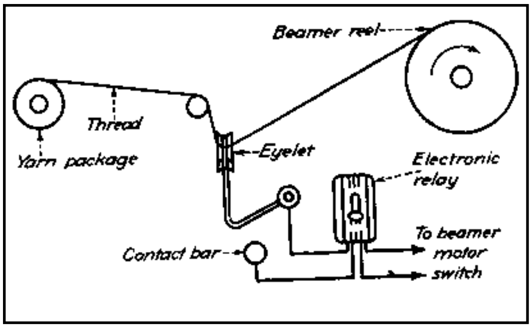


Fig. 5 A broken thread allows the contact bar to drop, thereby closing the electronic relay circuit; this stops the beamer reeling equipment.

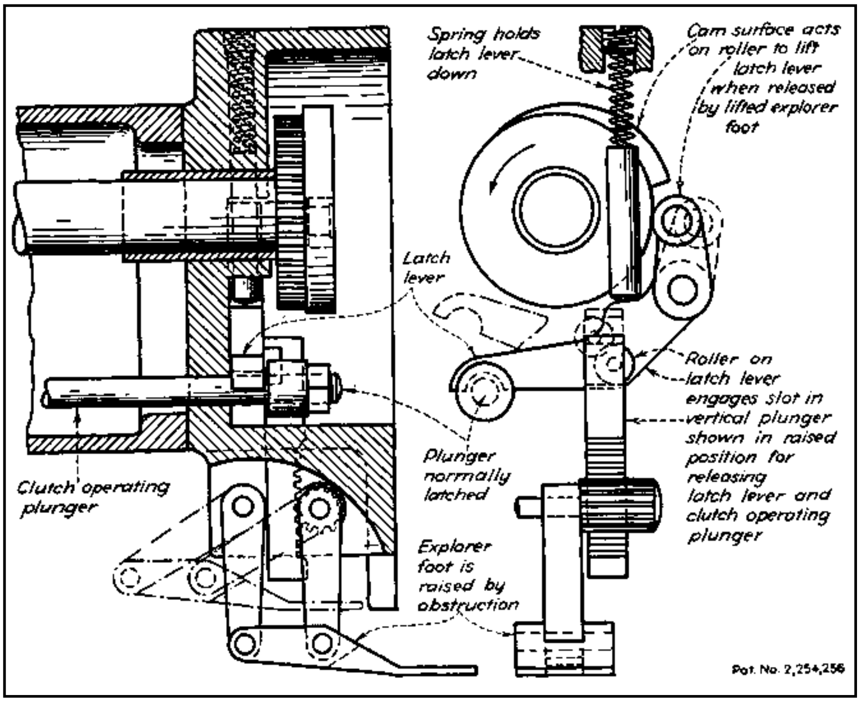


Fig. 7 An obstruction under the explorer foot of a wire-stitching machine raises a vertical plunger which releases a latch lever. A rotary cam then raises a lever that retains the clutch-operating plunger.

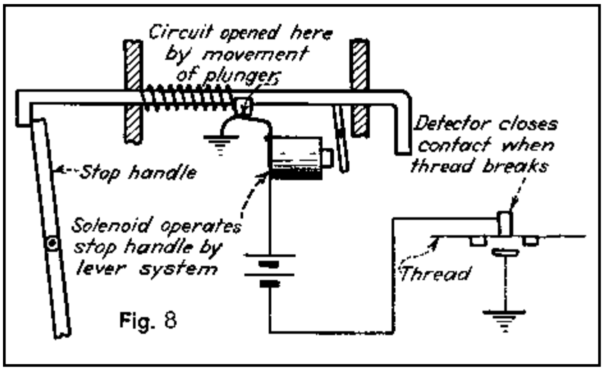


Fig. 8

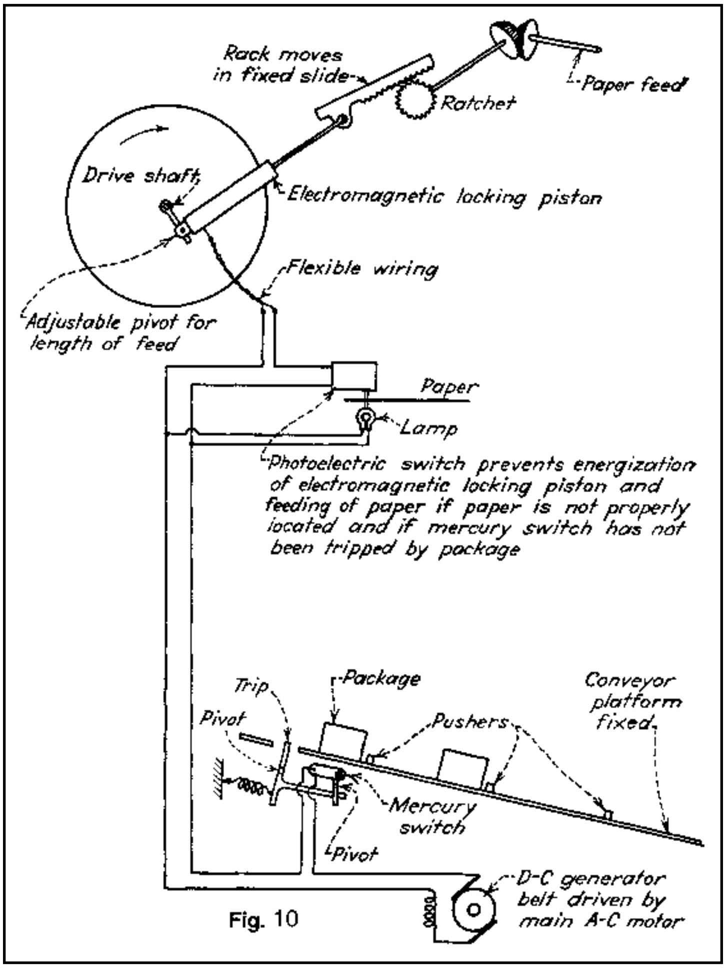


Fig. 10

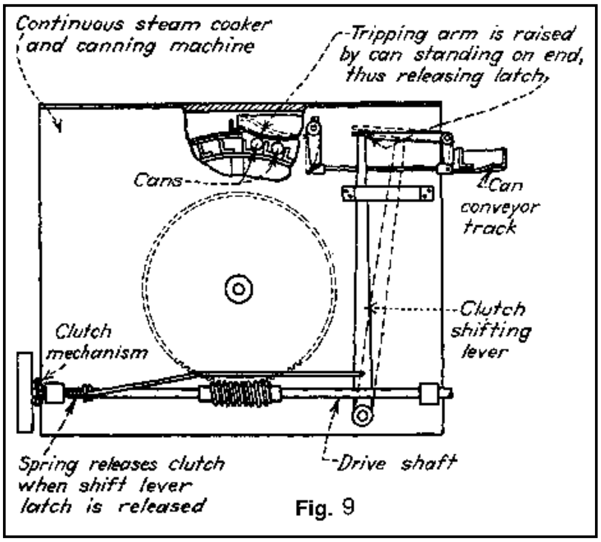


Fig. 9

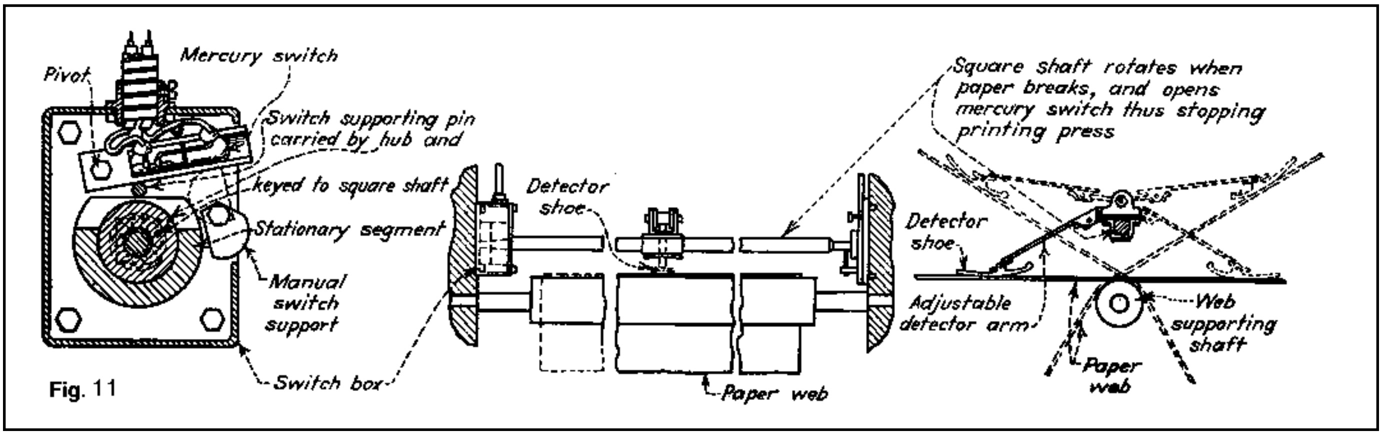


Fig. 11

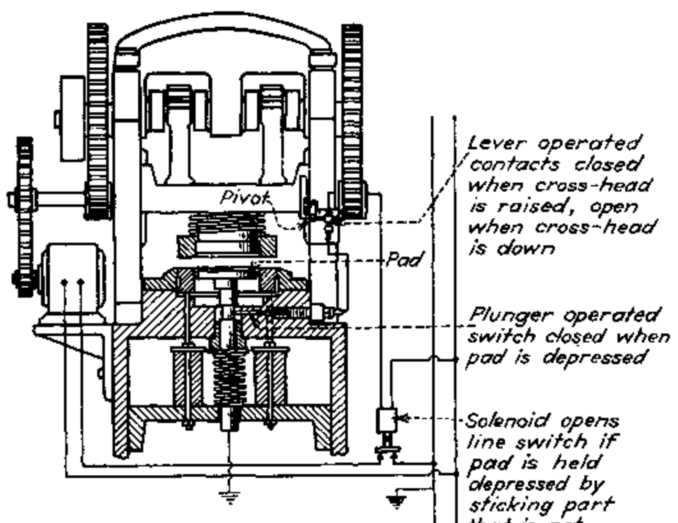


Fig. 12

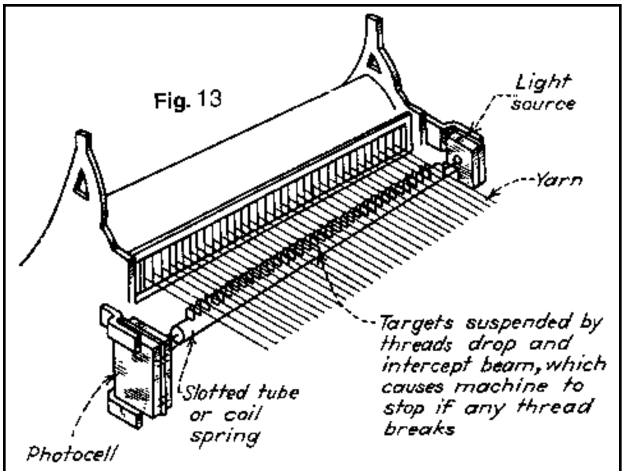


Fig. 13

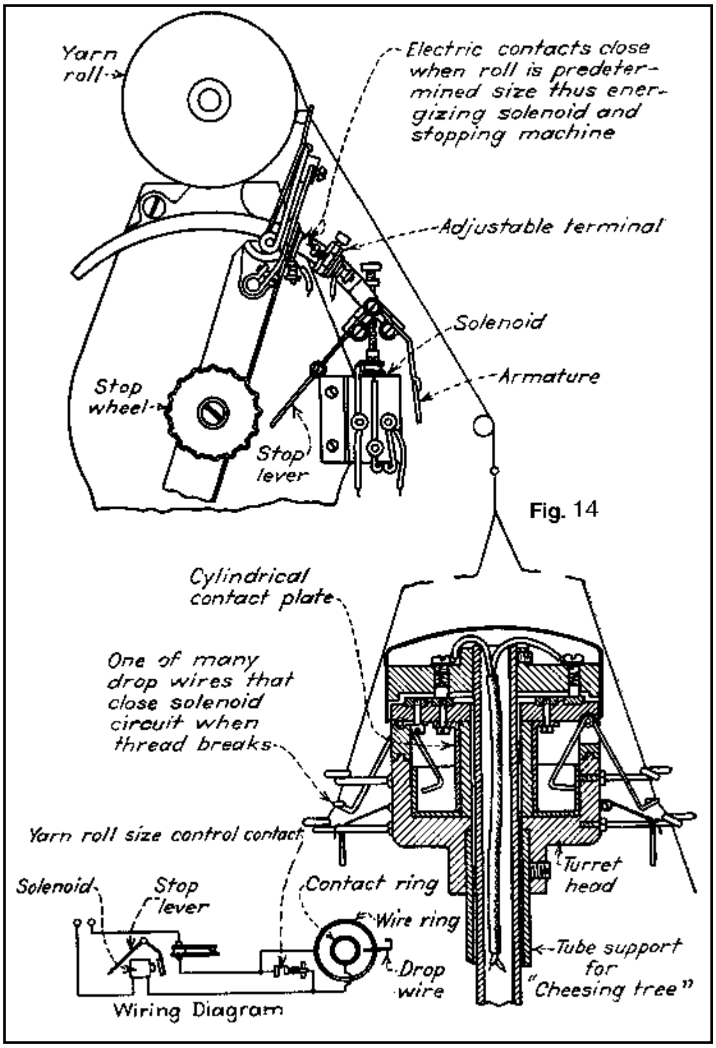


Fig. 14

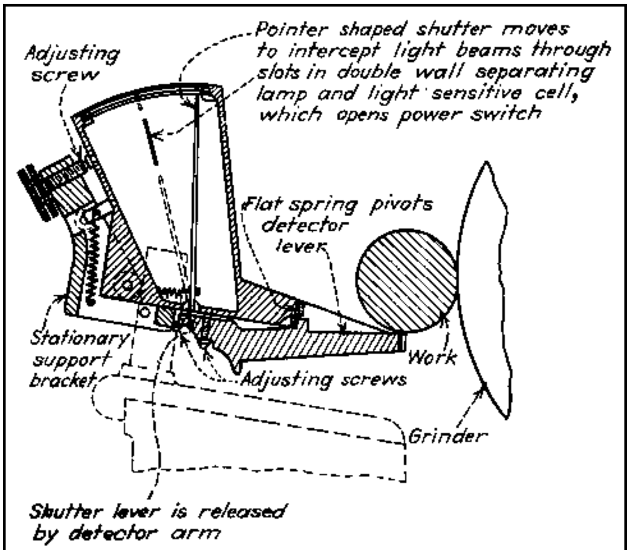


Fig. 15

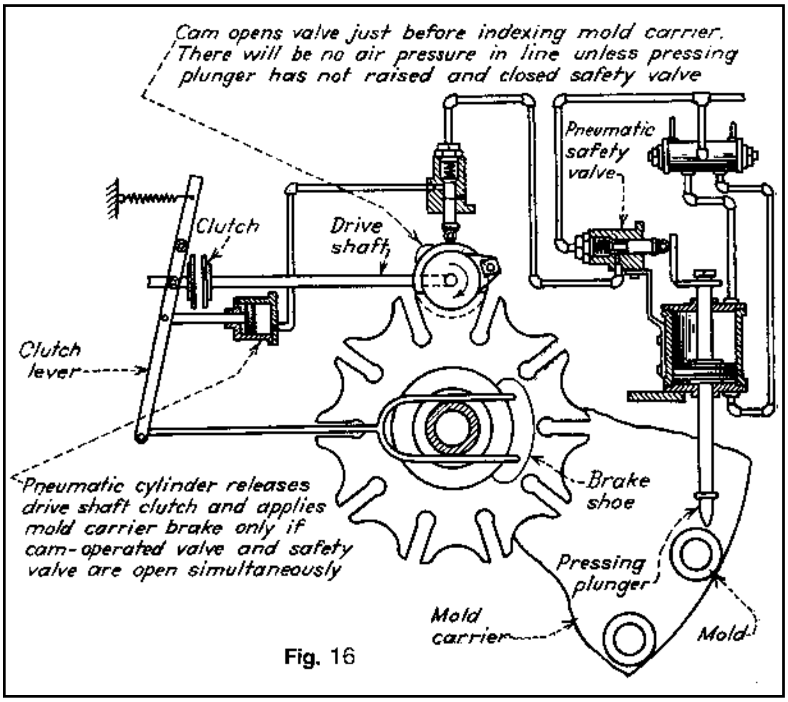


Fig. 16

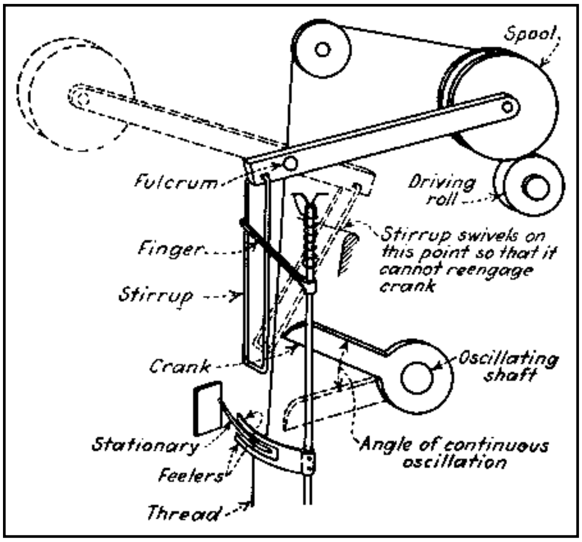


Fig. 17 A mechanism on a spooler. When a thread breaks, the feelers are released and the spiral spring causes the spindle with finger to rotate. The finger throws the stirrup into the path of the oscillating crank, which on its downward stroke throws the spool into the position shown dotted. The stirrup is then thrown out of the path of the oscillating crank.

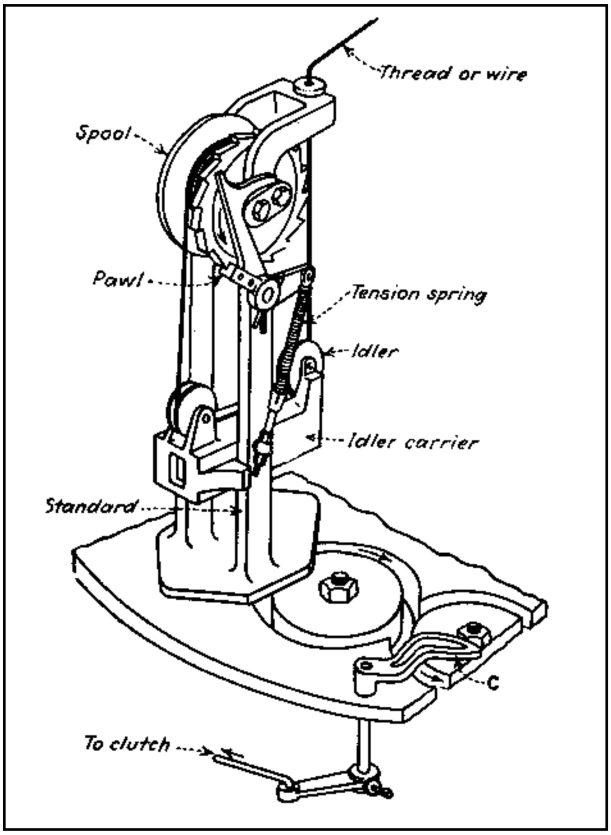


Fig. 18 A mechanism in use on tubular braiding machines. When the machine is braiding, tension on the wire or thread lifts the idler carrier which then releases the pawl from the ratchet on the spool flange and allows the spool to turn and unwind. When the machine stops, the tension on the wire is decreases, allowing the idler carrier to fall so that the pawl can engage the ratchet. If a wire breaks while the machine is running, the unsupported idler carrier falls to the base of the standard, and when the standard arrives at the station in the raceway adjacent to the cam *C,* the lug *L* on the idler carrier strikes the cam *C,* rotating it far enough to disengage a clutch on the driving shaft, thereby stopping the machine.

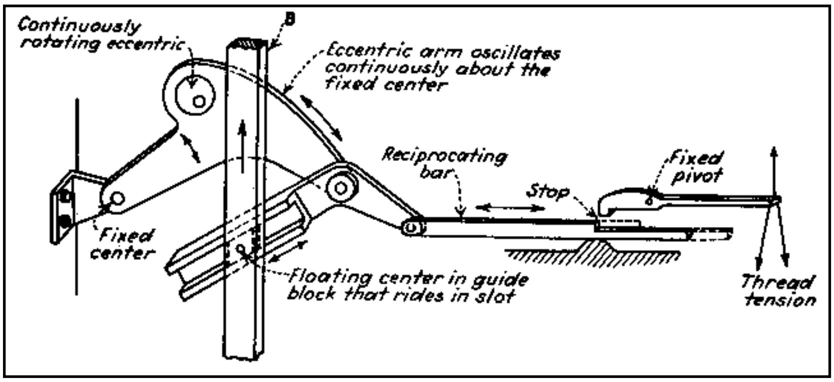


Fig. 19 When thread breaks, the stop drops and intercepts the reciprocating bar. On the next counter-clockwise oscillation of the eccentric arm, the bar *B* is raised. A feature of this design is that it permits the arm *B* to move up or down independently for a limited distance.

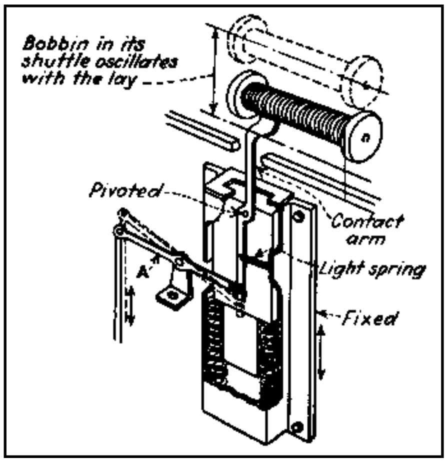


Fig. 20 A diagram of a mechanism that causes a bobbin changer to operate. If the contact arm does not slip on the bobbin, lever *A* will rotate to the position shown. But if contact with the bobbin center slips, when the bobbin is empty, lever *A* will not rotate to the position indicated by the dashed line. This will cause the bobbin changer to operate.

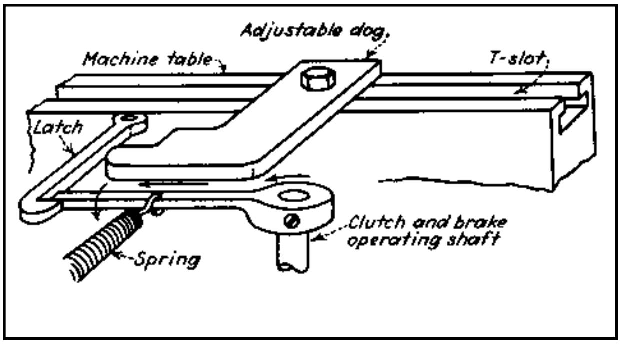


Fig. 21 A simple stop mechanism for limiting the stroke of a reciprocating machine member. Arrows indicate the direction of movements.

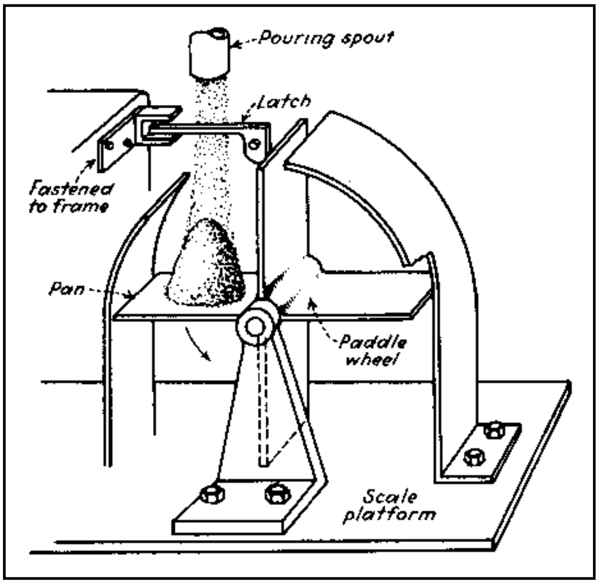


Fig. 22 When the predetermined weight of material has been poured on the pan, the movement of the scale beam pushes the latch out of engagement. This allows the paddle wheel to rotate and dump the load. The scale beam drops, thereby returning the latch to the holding position and stopping the wheel when the next vane hits the latch.

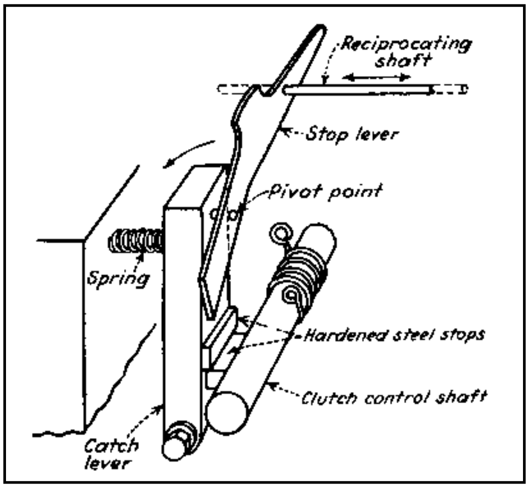


Fig. 23 In this textile machine, any movement that will rotate the stop lever counter-clockwise will move it into the path of the continuously reciprocating shaft. This will cause the catch lever to be pushed counter-clockwise, freeing the hardened steel stop on the clutch control shaft. A spiral spring then impels the clutch control shaft to rotate clockwise. That movement throws out the clutch and applies the brake. The initial movement of the stop lever can be caused by a breaking thread or a moving dog.

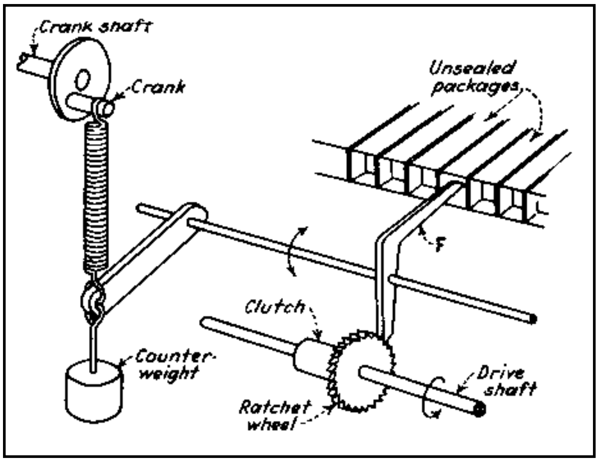


Fig. 24 some package-loading machines have provisions to stop the machine if a package passes the loading station without receiving an insert. Pawl finger *F* has a rocking motion imparted by the crankshaft, timed so that it enters the unsealed packages and is stopped against the contents. If the box is not filled, the finger enters a long distance. The pawl end at the bottom engages and holds a ratchet wheel on the driving clutch which disengages the machine-driving shaft.

ELECTRICAL AUTOMATIC STOPPING MECHANISMS

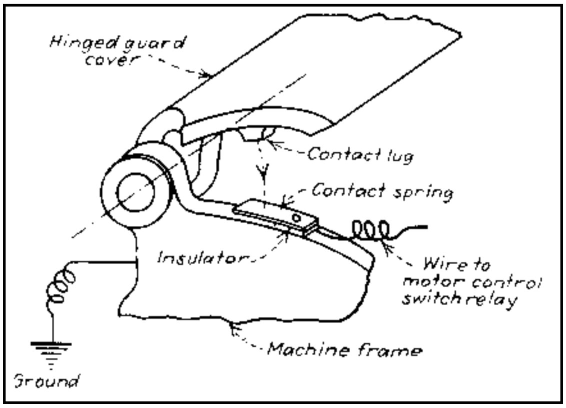


Fig. 1 A safety mechanism on some machines stops the motor when a guard cover is lifted. The circuit is complete only when the cover is down. In that position a contact lug establishes a metal-to-metal connection with the contact spring, completing the relay circuit.

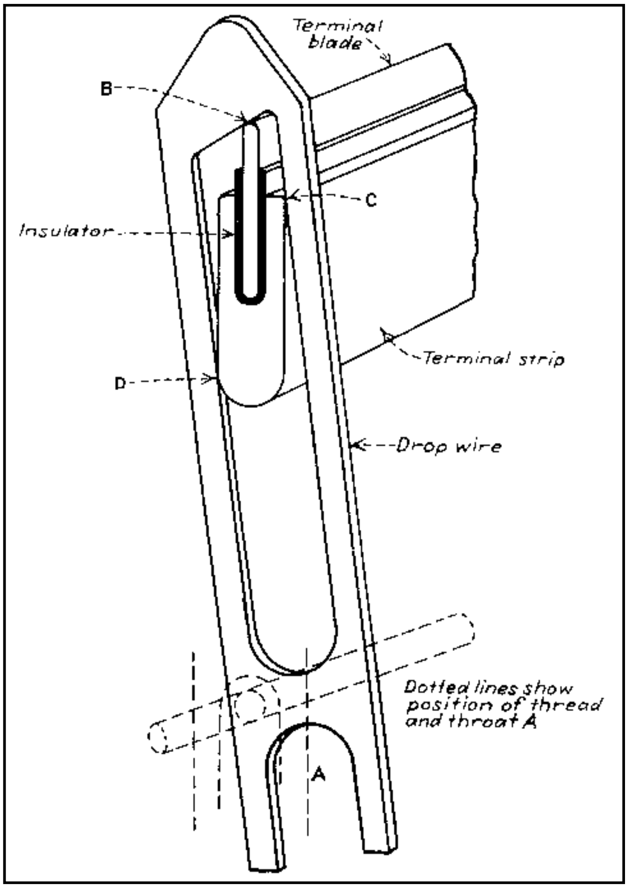


Fig. 2 An electrical three-point wedging "warp stop" is shown after a thread has broken and a drop wire has fallen and tilted to close the circuit. Dotted lines indicate the normal position of the drop wire when it is riding on a thread. When a thread breaks, the drop wire falls and strikes the top of the terminal blade B, the inclined top of the slot. This causes a wedging action that tilts the drop wire against the terminal strip at C and D, reinforcing the circuit-closing action.

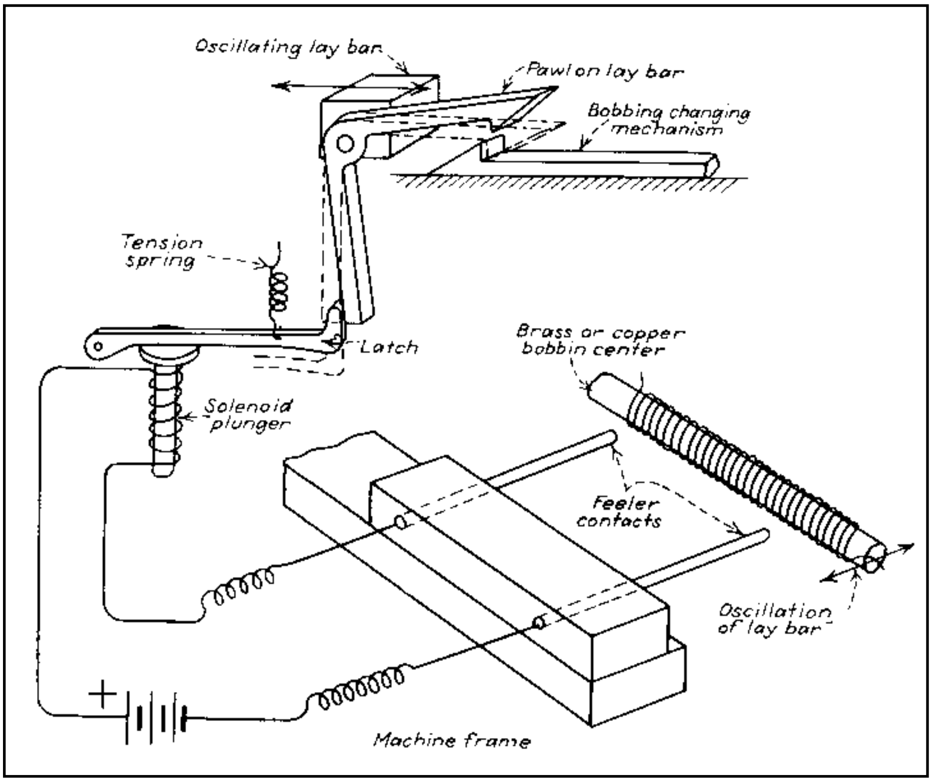


Fig. 3 Bobbin changer. When a bobbin is empty, the feeders contact the metal bobbin center, completing the circuit through a solenoid which pulls a latch. That causes the bobbin-changing mechanism to operator and put a new bobbin in the shuttle. As long as the solenoid remains deenergized, the pawl on the lay bar is raised clear of the hook on the bobbin-changing mechanism.

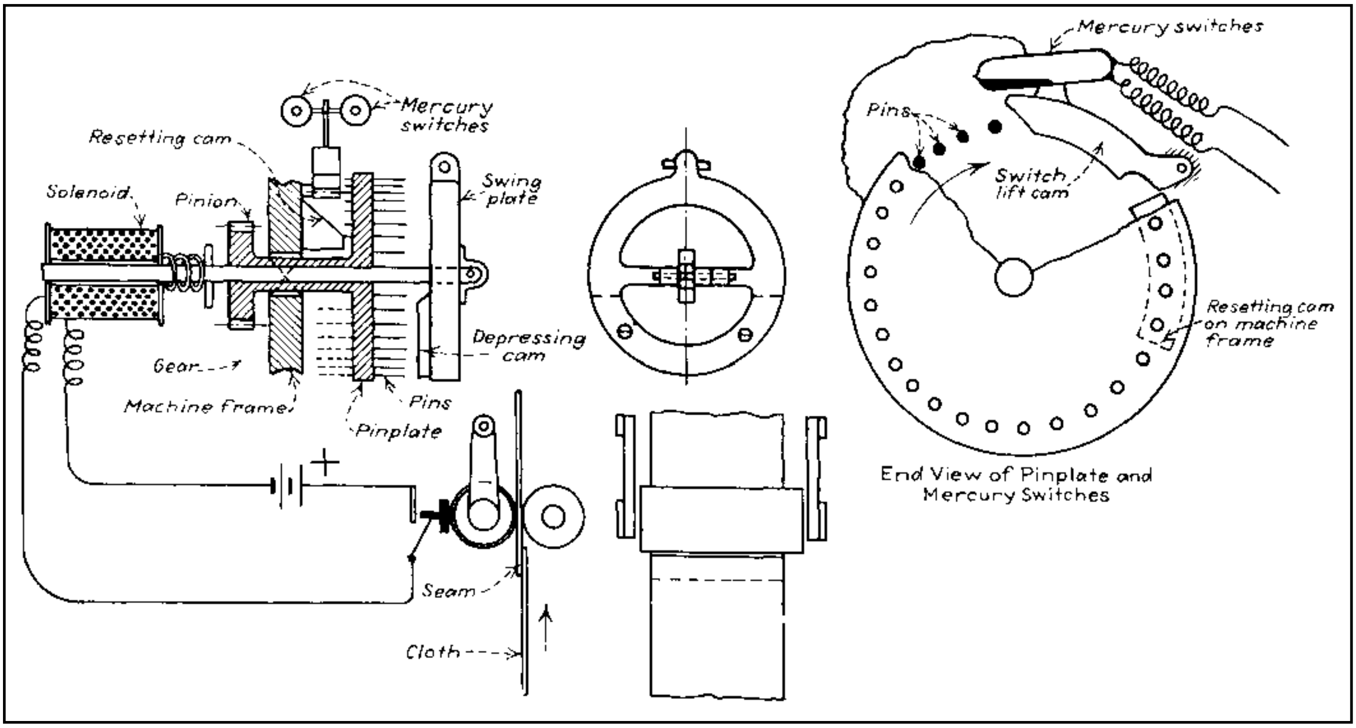


Fig. 4 Control for the automatic shear. When a seam of two thicknesses of cloth passes between the rolls, the swing roller is moved outward and closes a sensitive switch which energizes a solenoid. The solenoid pulls in an armature whose outer end is attached to the hinged ring where a cam plate is also fastened. The cam plate depresses the pins in a rotating plate. As the plate rotates, the depressed pins lift a hinged cam arm on which two mercury switches are mounted. When tilted, the switches complete circuits in two motor controls. A resetting cam for pushing the depressed pins back to their original position is fastened on the machine frame. The two motors are stopped and reversed until the seam has passed through rollers before they are stopped and reversed again.

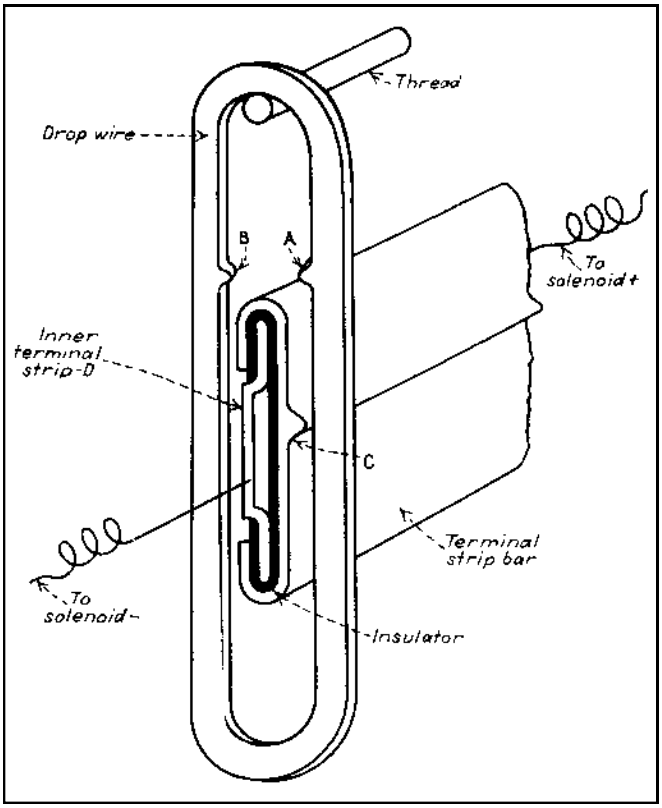


Fig. 5 Electric stop for a loom. When a thread breaks or slackens, the drop wire falls and contact A rides on contact C. The drop wire, supported off-center, swings so that contact B is pulled against the inner terminal strip D, completing the solenoid circuit.

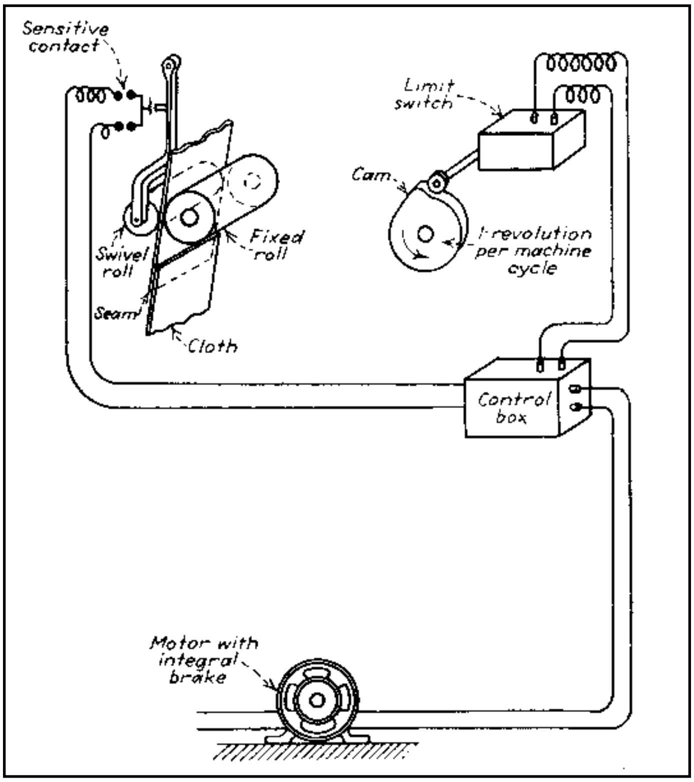
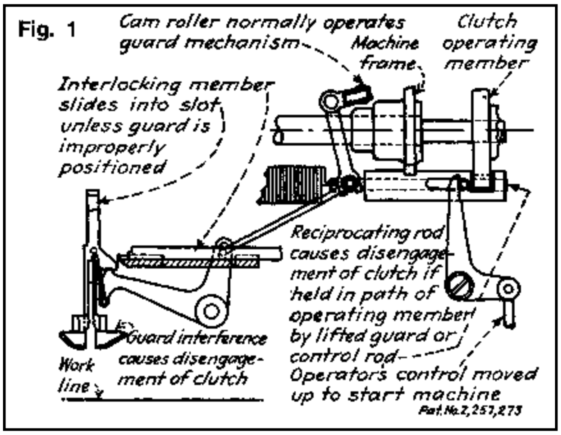


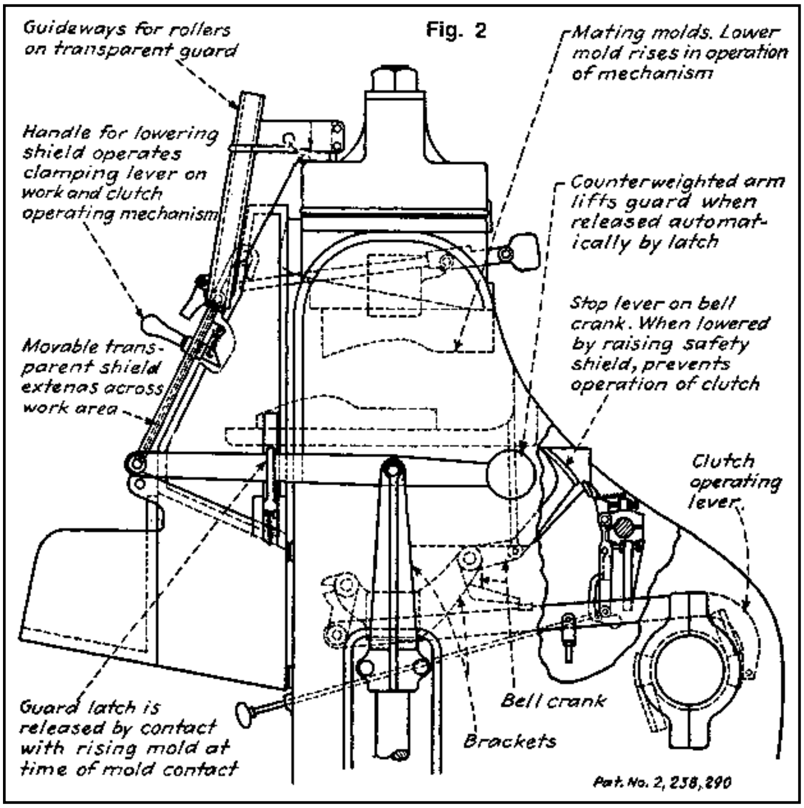
Fig. 6 This automatic stop for a folder or yarder machine always stops the machine in the same position when a seam in the cloth passes between the rolls. A seam passing between the rolls causes the swivel-mounted roll to lift slightly. This motion closes contacts in a sensitive switch that opens a relay in the control box. The next time the cam closes the limit switch, the power of the motor with the integral magnetic brake is shut off. The brake always stops the machine in the same place.

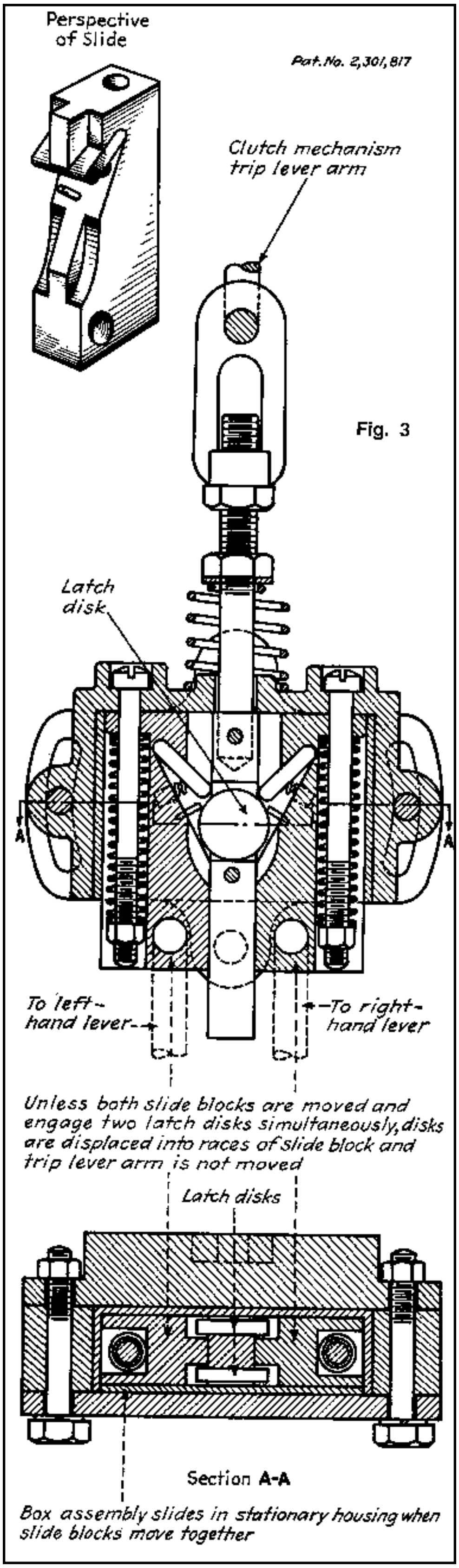
AUTOMATIC SAFETY MECHANISMS FOR OPERATING MACHINES

The best automatic safety mechanisms are those that have been designed specifically for the machine on which they will be installed. When properly designed, automatic safety devices (1) do not reduce the operator's visibility, (2) do not interfere with the operator's performance, (3) do not make physical contact with the operator to prevent injury (e.g., by knocking his hand out of the way), (4) are fail-safe, (5) are sensitive enough to operate instantly, and (6) render the machine inoperative if anyone attempts to tamper with them or remove them.

Safety mechanisms can range from those that keep both of the operator's hands occupied on controls away from the work area to shields that completely enclose the work in progress on the machine and prevent machine operation unless the shield is securely in place. Many modern safety systems are triggered if any person or object breaks a light beam between a photoemitter and photoreceiver.







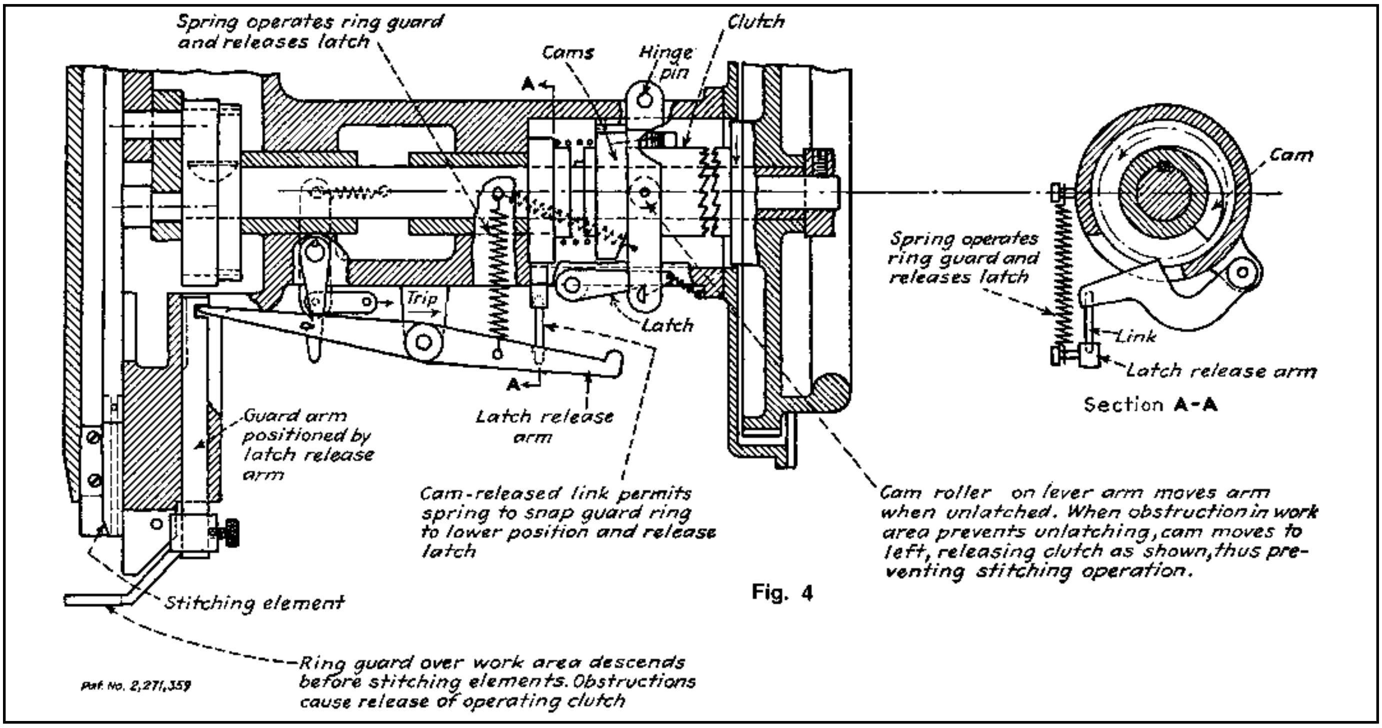


Fig. 4

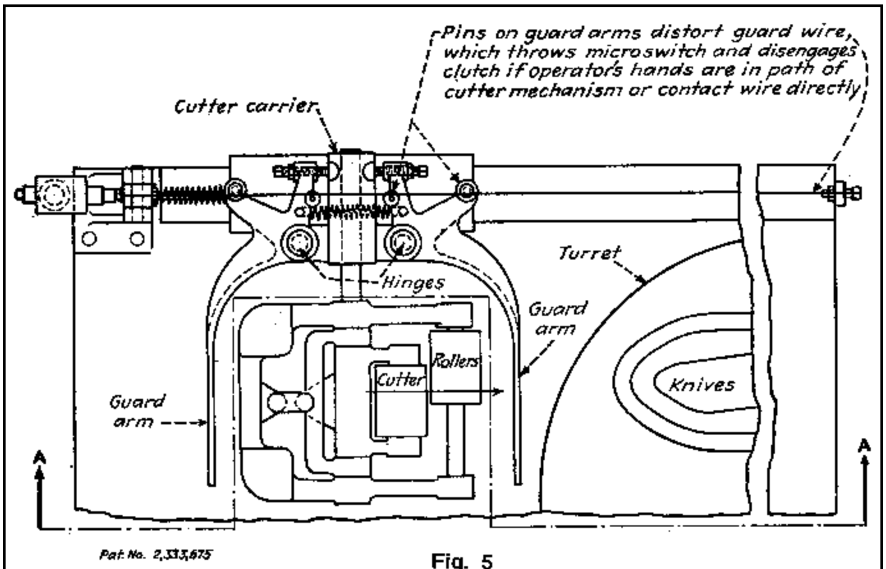


Fig. 5

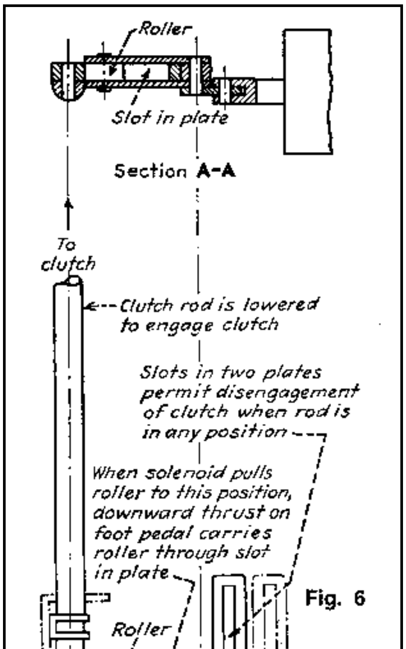
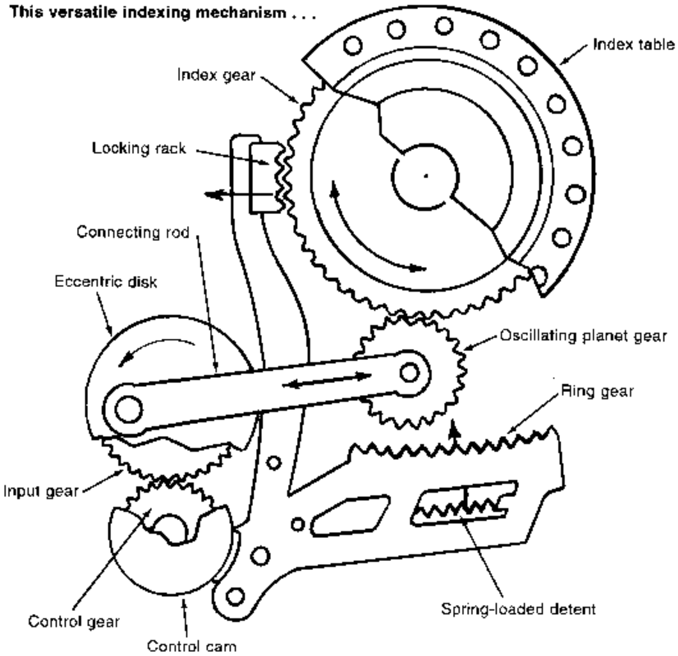


Fig. 6

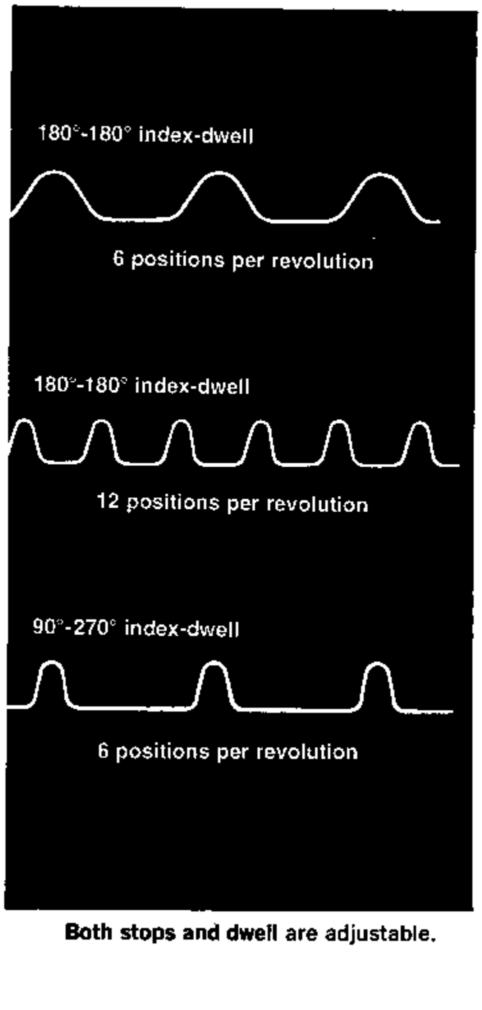
CHAPTER 4

RECIPROCATING AND GENERAL-PURPOSE MECHANISMS

GEARS AND ECCENTRIC DISK COMBINE IN QUICK INDEXING



. . provides choice of indexing modes



Both stops and dwell are adjustable.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

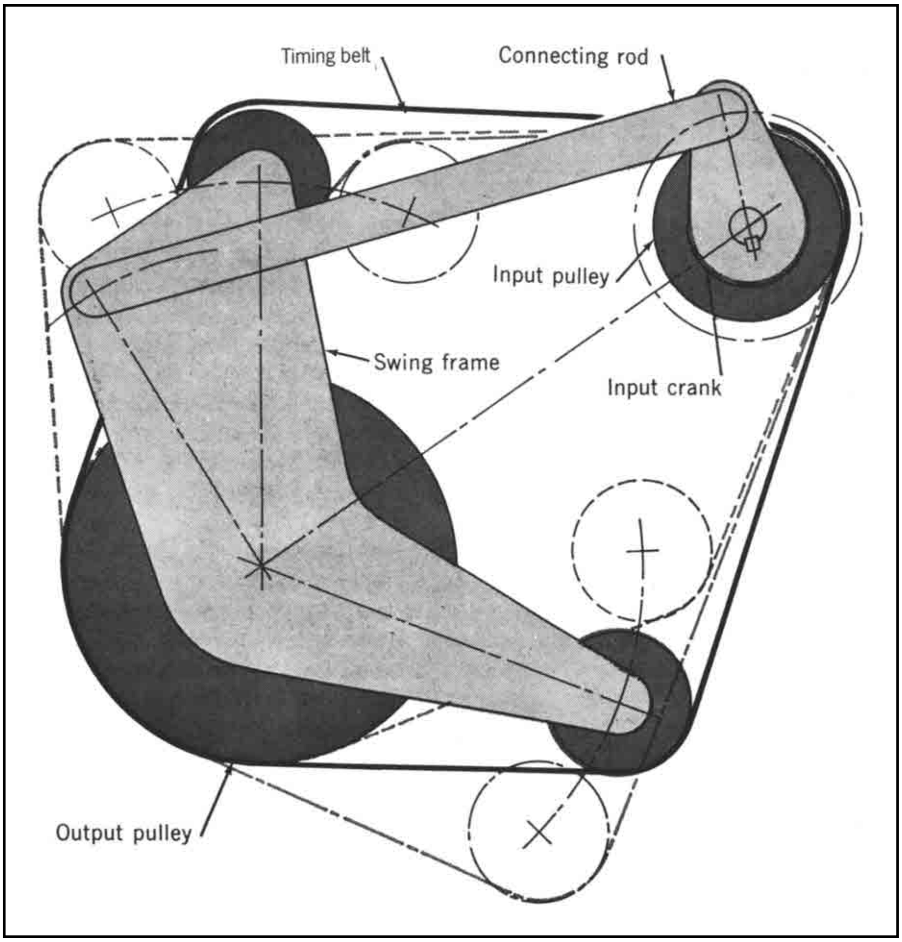
Developed by Eric S. Buhayar and Eugene E. Brown of the Engineering Research Division, Scott Paper Co. (Philadelphia), the mechanisms are employed in automatic assembly lines.

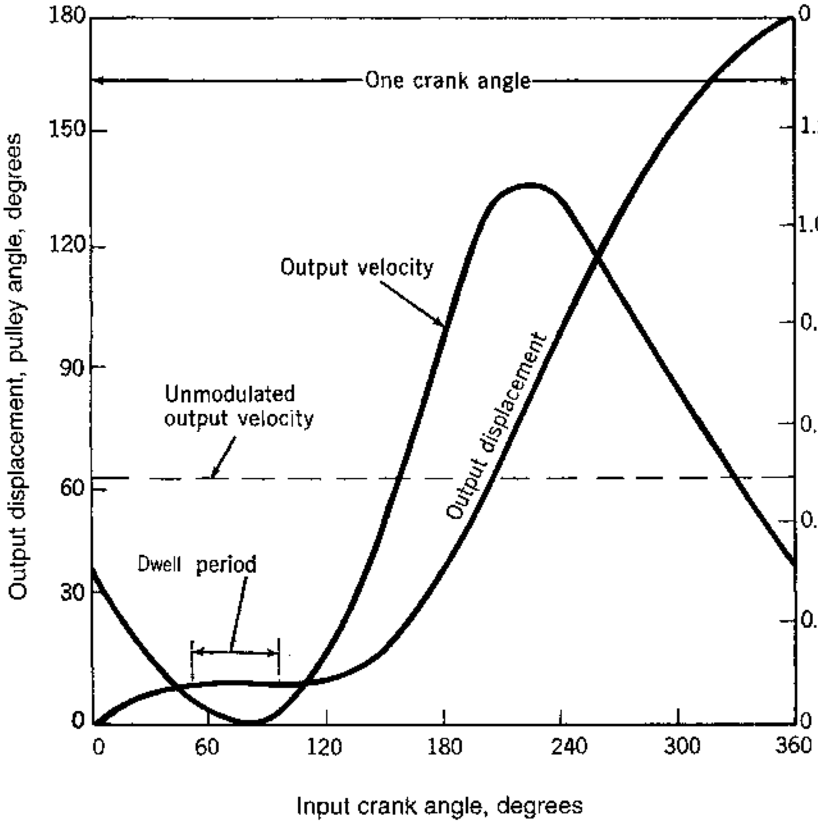
These mechanisms, moreover, can function as phase adjusters in which the rotational position of the input shaft can be shifted as desired in relation to the output shaft. Such phase adjusters have been used in the textile and printing industries to change the "register" of one roll with that of another,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

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

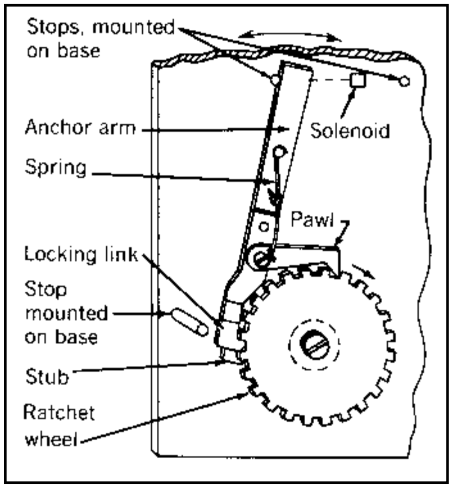
The input shaft of such a device has a sprocket eccentrically fixed to it. The input also drives another shaft through one-to-one gearing. This second shaft mounts a similar eccentric sprocket that is, however, free to rotate. The chain passes first around an idler pulley and then around a second pulley,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

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

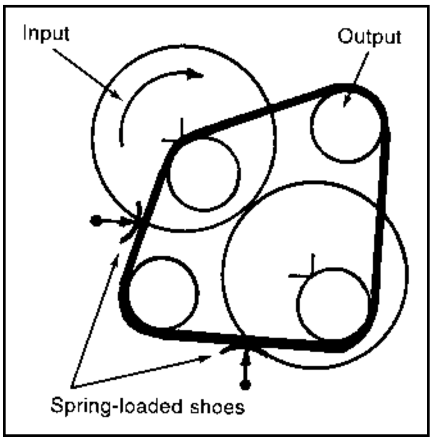




MODIFIED RATCHET DRIVE



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



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

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

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

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

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

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

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

ODD SHAPES IN PLANETARY GIVE SMOOTH STOP AND GO

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

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

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

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

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

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

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

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

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

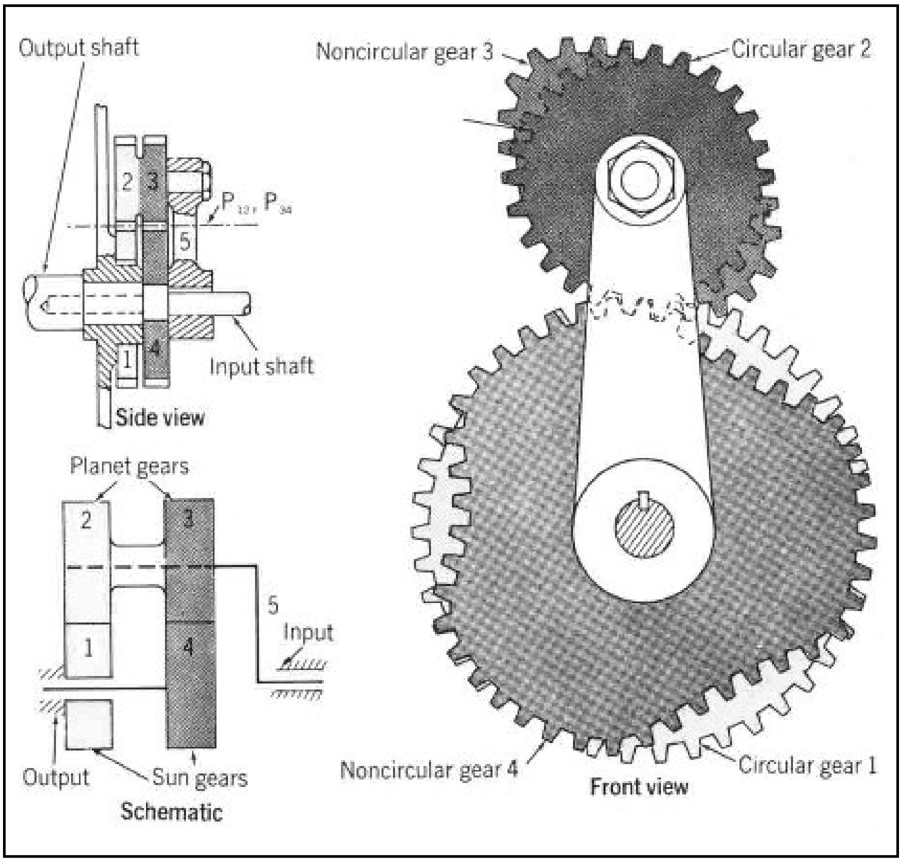
Differential motion. In deriving the operating principle of his mechanism, Freudenstein first considered a conventional epicyclic (planetary) drive in which the input to the cage or arm causes a planet set with gears *2* and *3* to rotate the output "sun," gear *4*, while another sun, gear *1*, is kept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

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

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

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

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



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

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

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

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

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

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

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

and

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

Hence

$$\theta_{30}(1 + S)L_3 = 360^\circ$$

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

$$\theta_{30} = 90^\circ$$

and

$$\Delta\theta_3 = 90^\circ$$

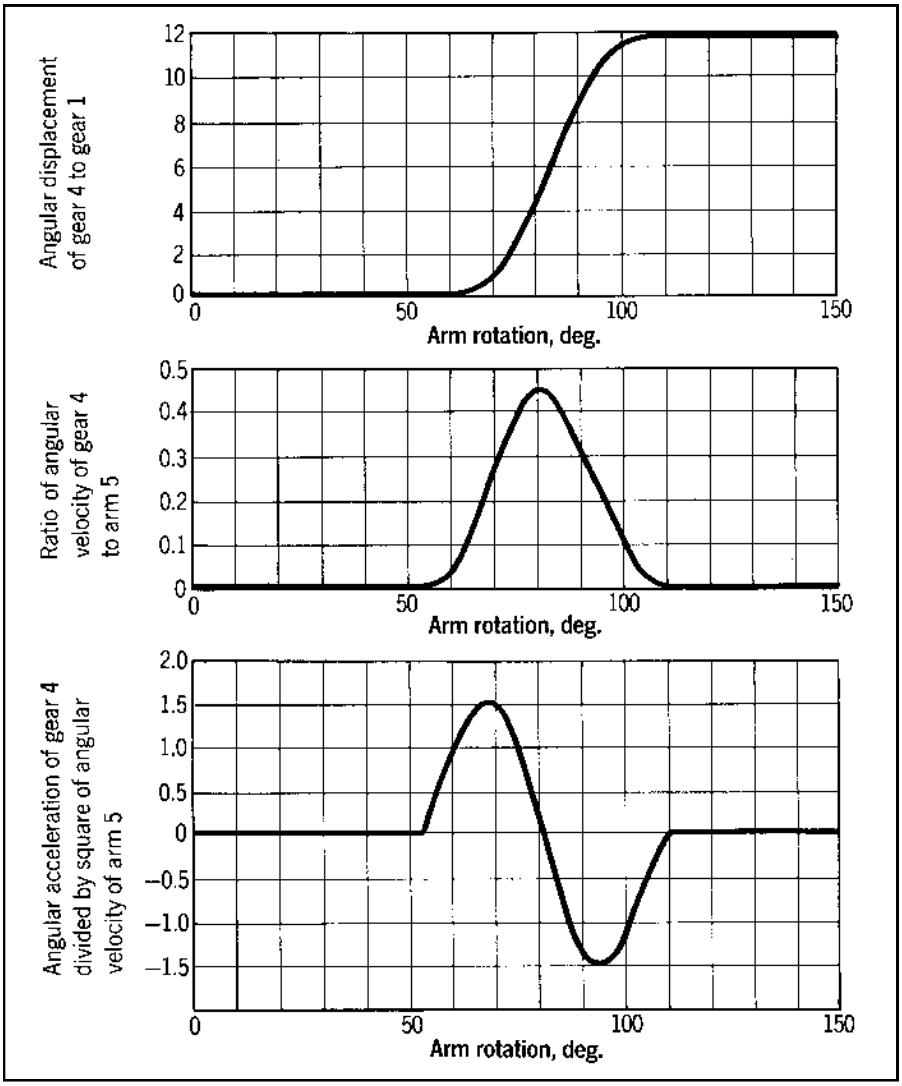
Now select a convenient profile for the noncircular portion of gear *3*. One profile (see the profile drawing) that Freudenstein found to have favorable high-speed characteristics for stop-and-go mechanisms is

$$r_3 = R_3$$

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

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

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



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

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

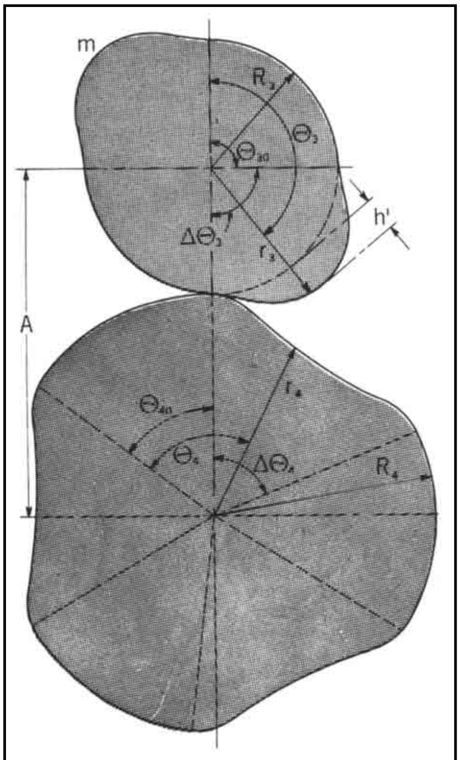
$$\lambda = \frac{1-\mu}{\mu} \times \frac{[S+\alpha-(1+\alpha)\mu][\alpha-S-(1+\alpha)\mu]}{[\alpha-(1+\alpha)\mu]^2}$$

where $R_3\lambda$ = height of lobe

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

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

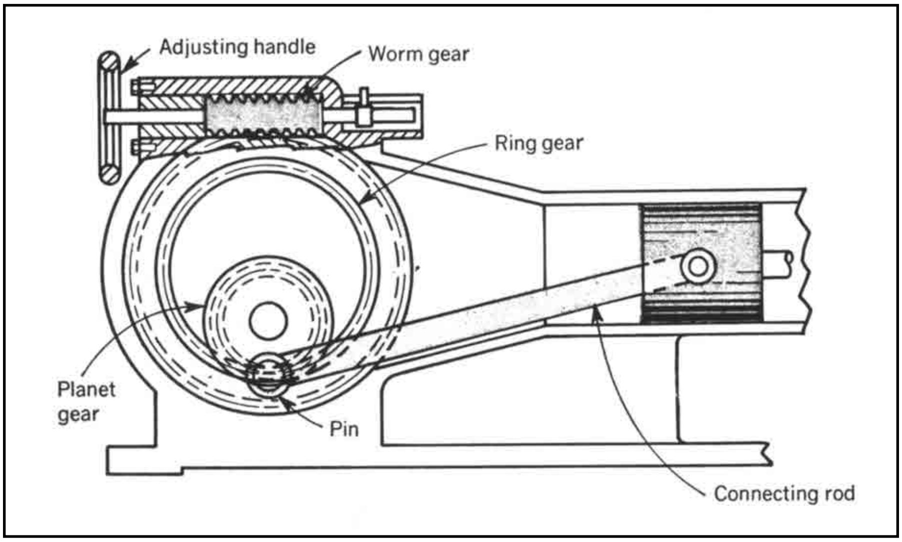
To evaluate the equation, select a suitable value for μ that is a reasonably simple rational fraction, i.e., a fraction such as ⅜ whose numerator and denominator are reasonably small integral numbers.



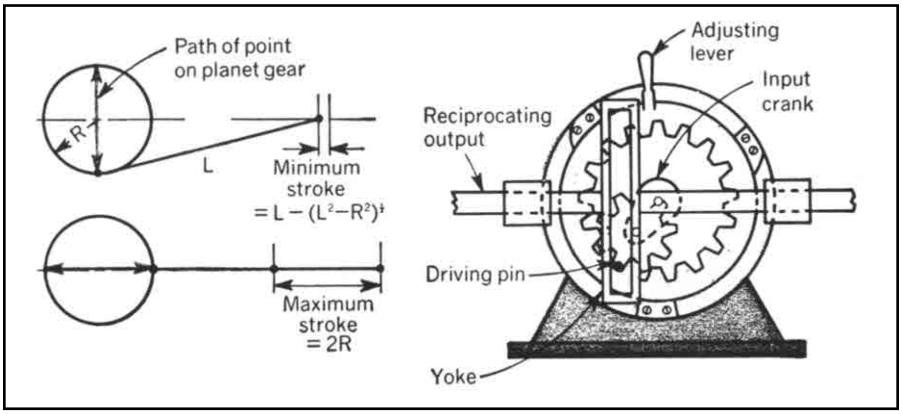
Profiles for noncircular gears are circular arcs blended to special cam curves.

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

CYCLOID GEAR MECHANISM CONTROLS STROKE OF PUMP



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



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

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

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

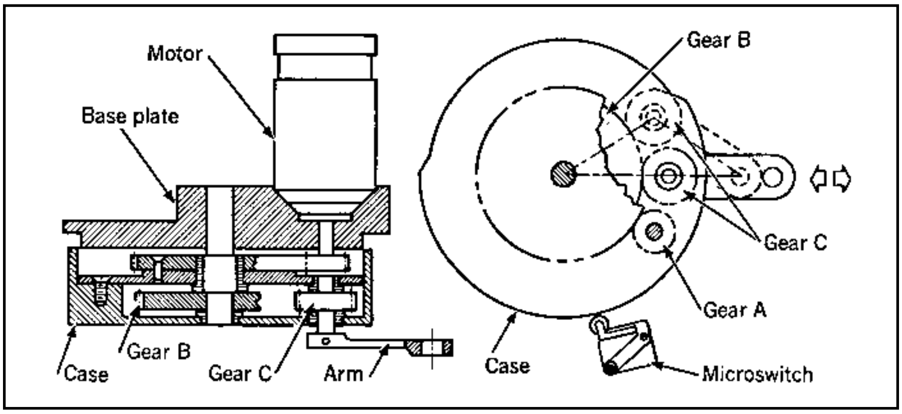
The second diagram illustrates another version that has a yoke instead of a connecting rod. This permits the length of the stroke to be reduced to zero. Also, the length of the pump can be substantially reduced.

CONVERTING ROTARY-TO-LINEAR MOTION

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

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

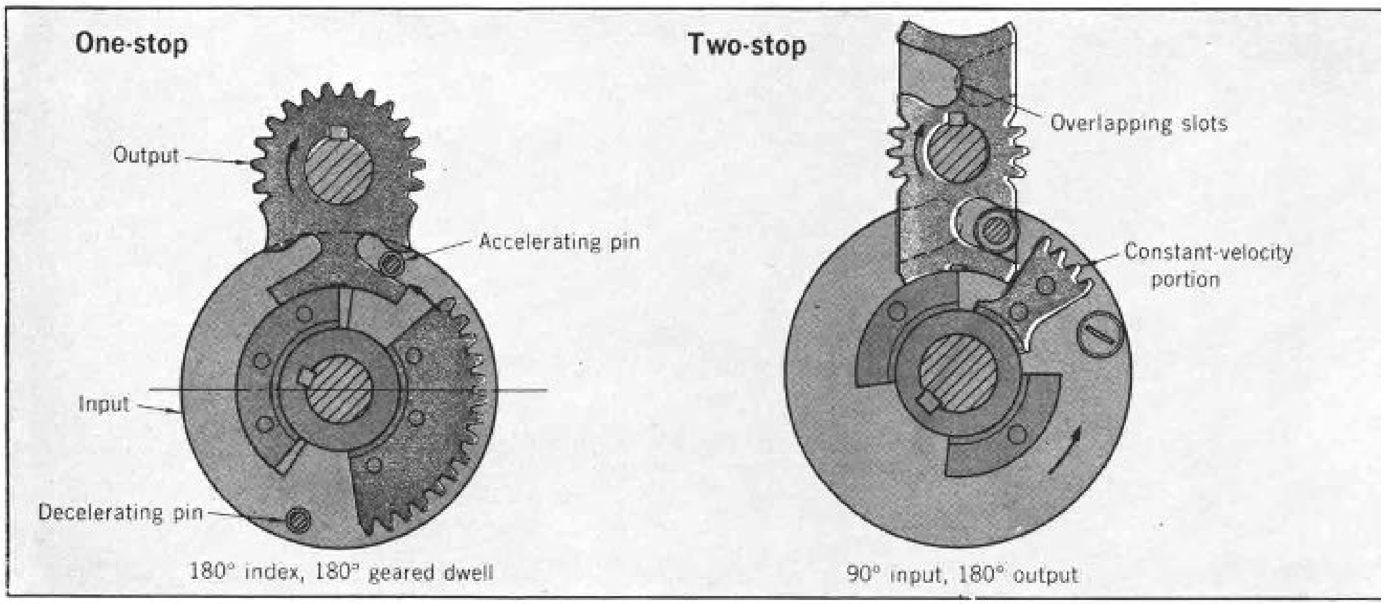
When the end of travel is reached, a switch causes the motor to reverse, returning the arm to its original position.



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

NEW STAR WHEELS CHALLENGE GENEVA DRIVES FOR INDEXING

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



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

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

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

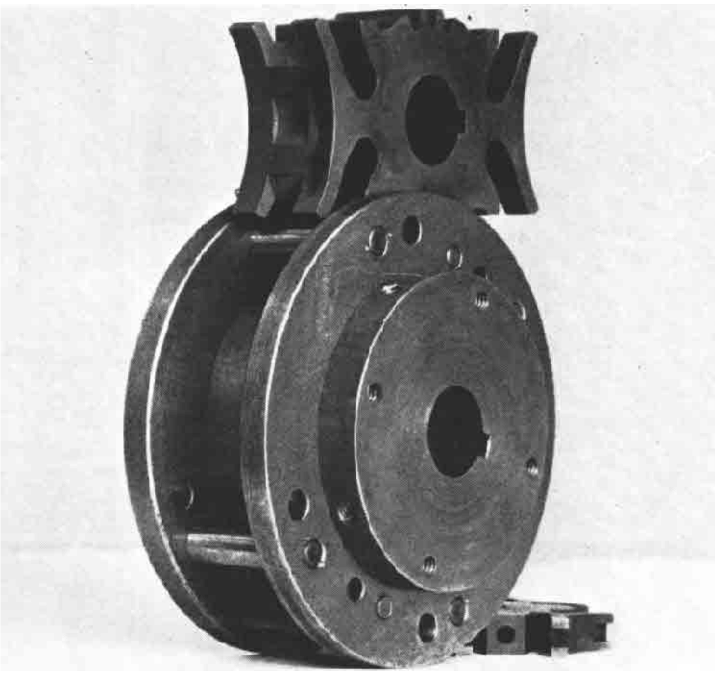
The circular-slot star wheels were invented by Martin Zugel of Cleveland, Ohio.

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

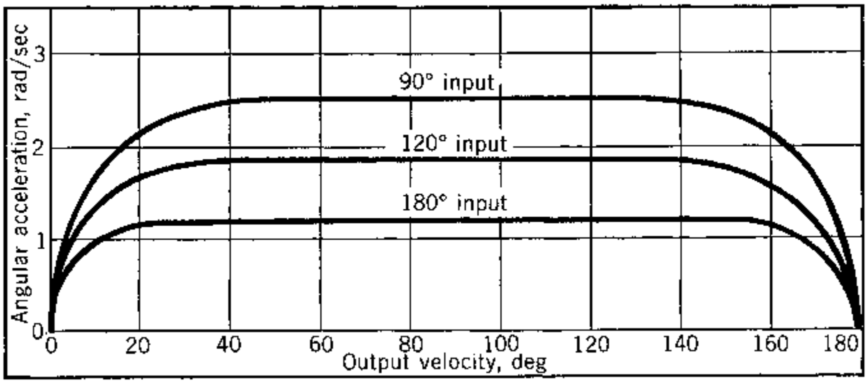
The star wheels with their circular-arc slots are easy to fabricate, and because the slots are true circular arcs, they can be visualized for mathematical analysis as four-bar linkages during the entire period of pin-slot engagement.

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

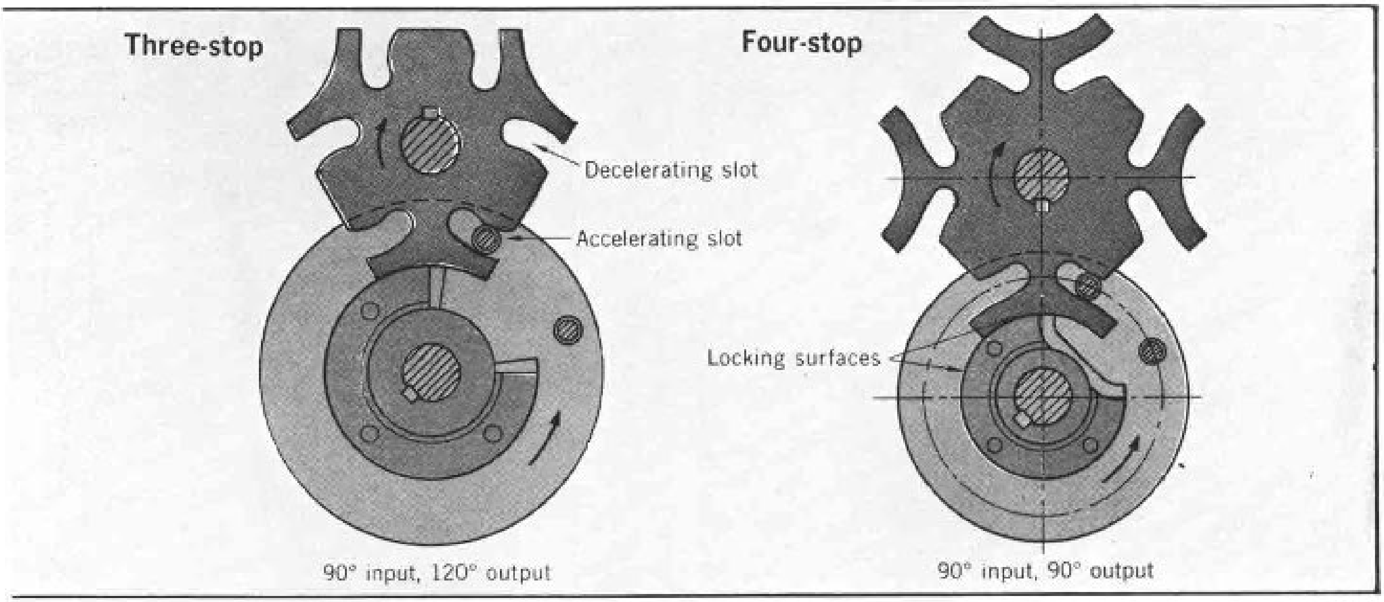
Another advantage of the star wheels is that they can index a full 360º in a relatively short period (180º). Such one-stop operation is not possible with genevas. In fact, genevas cannot do two-stop operations, and they have difficulty producing three stops per index. Most two-stop indexing devices available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



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



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



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

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

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

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

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

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

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

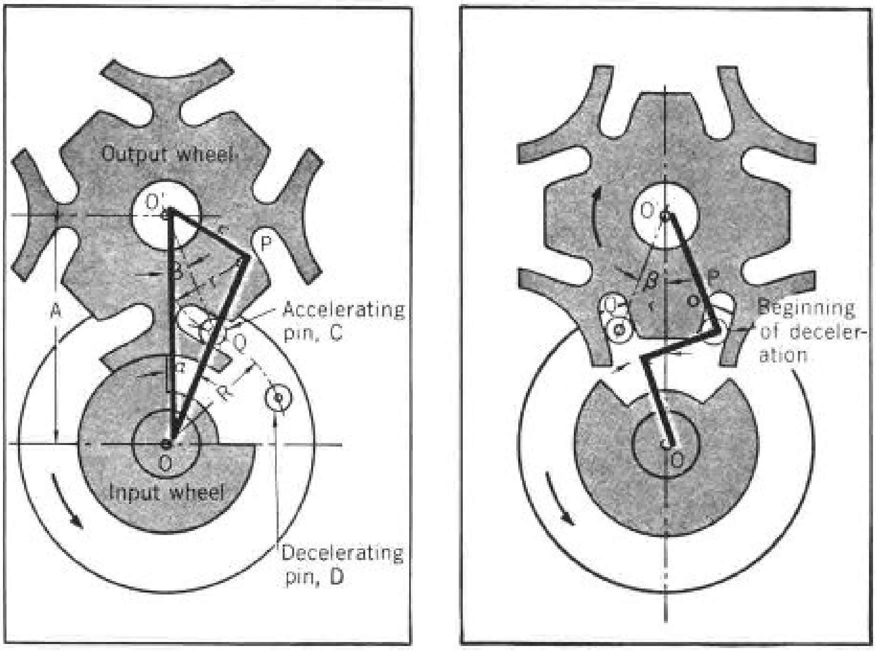
Assume, for example, that $\alpha = \beta = 22.5°$. Application of trigonometry yields:

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

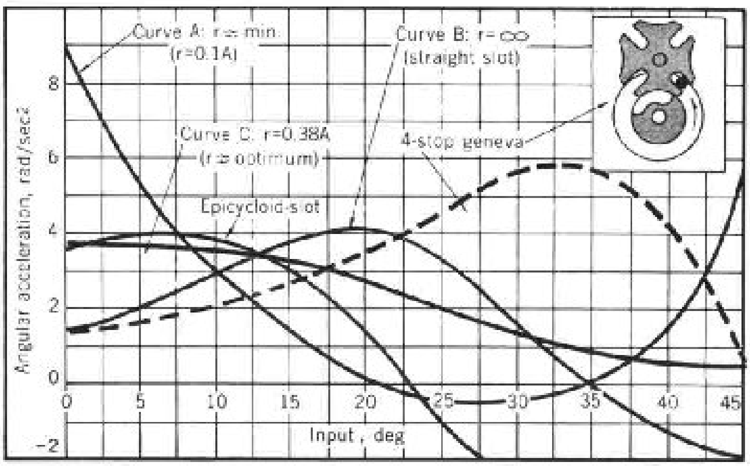
which yields $R = 0.541A$. The only restriction on r is that it be large enough to allow the wheel to pass through its mid-position. This is satisfied if:

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

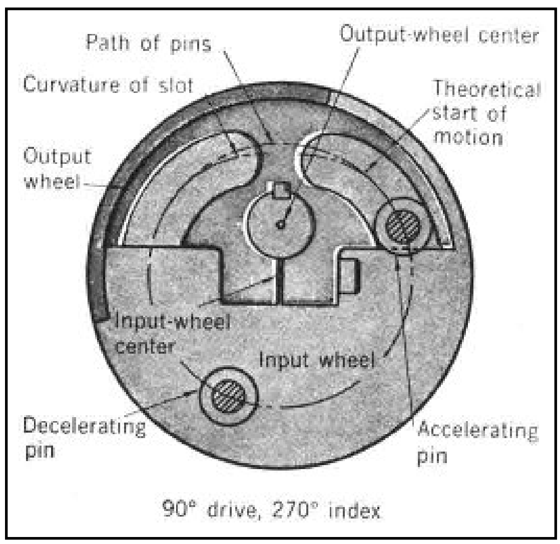
There is no upper limit on r, so that slot can be straight.



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



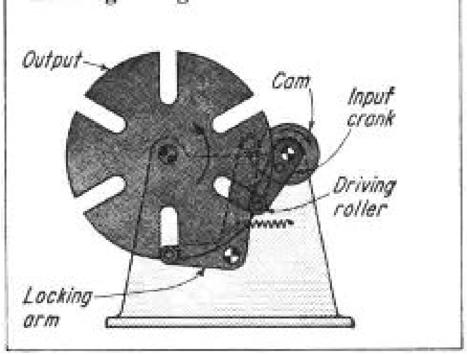
The accelerating force of star wheels (curves A, B, C) varies with input rotation. With an optimum slot (curve C), it is lower than for a four-stop geneva.



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

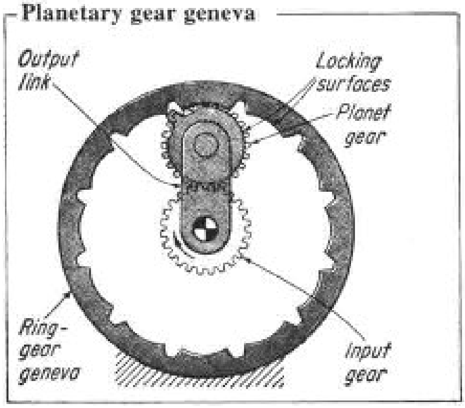
GENEVA MECHANISMS

Locking-arm geneva



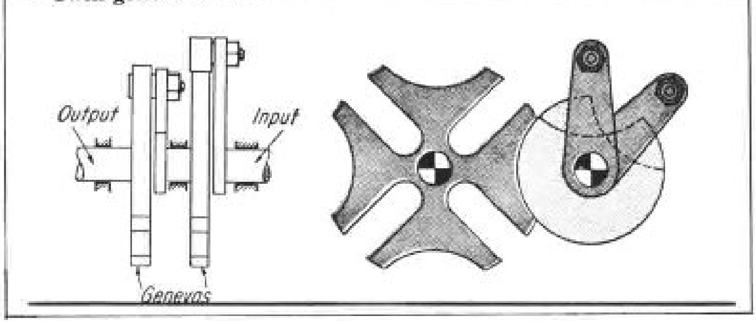
The driving follower on the rotating input crank of this geneva enters a slot and rapidly indexes the output. In this version, the roller of the locking-arm (shown leaving the slot) enters the slot to prevent the geneva from shifting when it is not indexing.

Planetary gear geneva



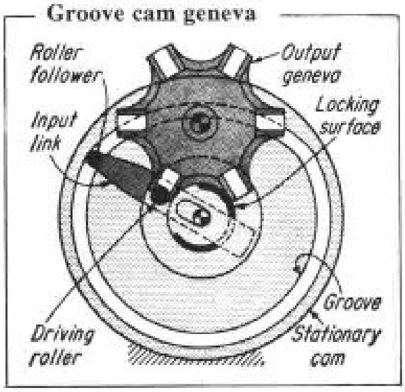
The output link remains stationary while the input gear drives the planet gear with single tooth on the locking disk. The disk is part of the planet gear, and it meshes with the ring-gear geneva to index the output link one position.

Twin geneva drive



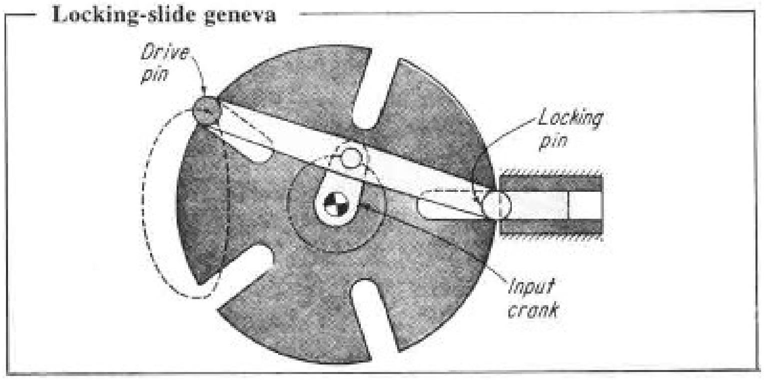
The driven member of the first geneva acts as the driver for the second geneva. This produces a wide variety of output motions including very long dwells between rapid indexes.

Groove cam geneva



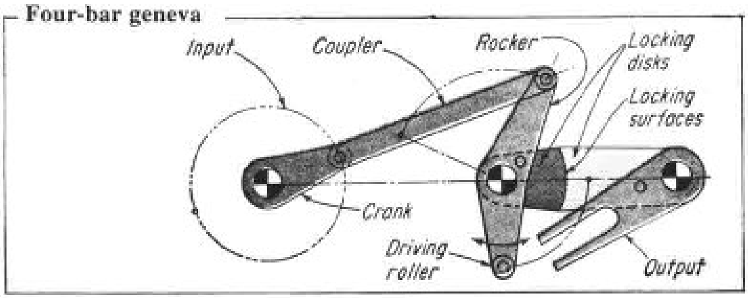
When a geneva is driven by a roller rotating at a constant speed, it tends to have very high acceleration and deceleration characteristics. In this modification, the input link, which contains the driving roller, can move radially while being rotated by the groove cam. Thus, as the driving roller enters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

Locking-slide geneva

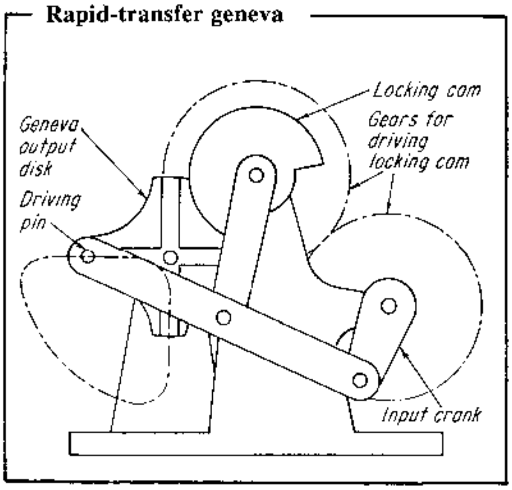


One pin locks and unlocks the geneva; the second pin rotates the geneva during the unlocked phase. In the position shown, the drive pin is about to enter the slot to index the geneva. Simultaneously, the locking pin is just clearing the slot.

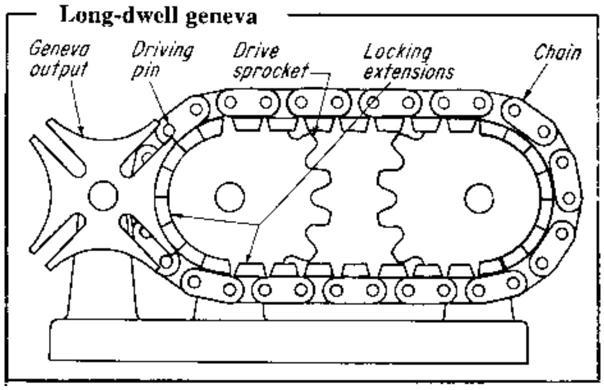
Four-bar geneva



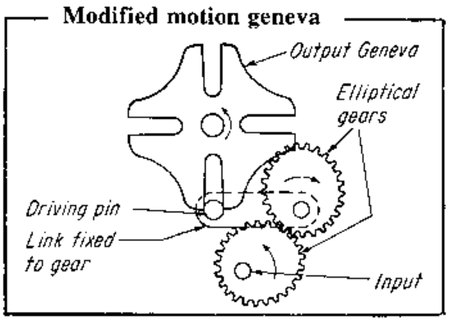
A four-bar geneva produces a long-dwell motion from an oscillating output. The rotation of the input wheel causes a driving roller to reciprocate in and out of the slot of the output link. The two disk surfaces keep the output in the position shown during the dwell period.



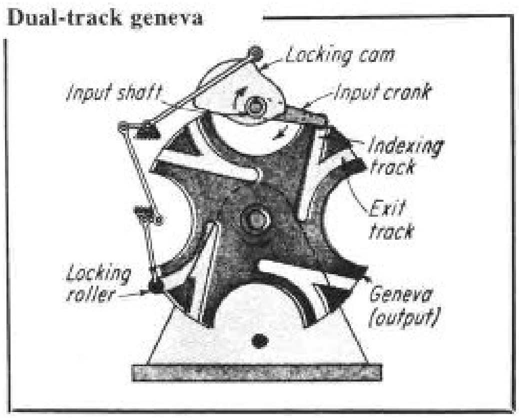
The coupler point at the extension of the connecting link of the four-bar mechanism describes a curve with two approximately straight lines, 90º apart. This provides a favorable entry situation because there is no motion in the geneva while the driving pin moves deeply into the slot. Then there is an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



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

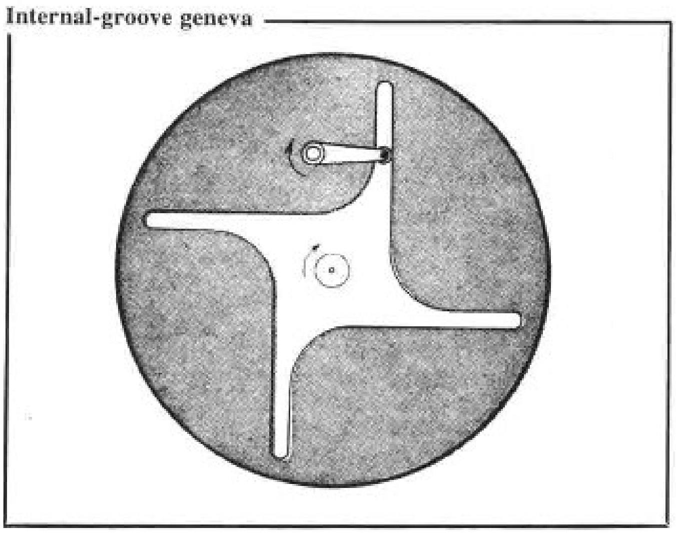


The input link of a normal geneva drive rotates at constant velocity, which restricts flexibility in design. That is, for given dimensions and number of stations, the dwell period is determined by the speed of the input shaft. Elliptical gears produce a varying crank rotation that permits either extending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



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

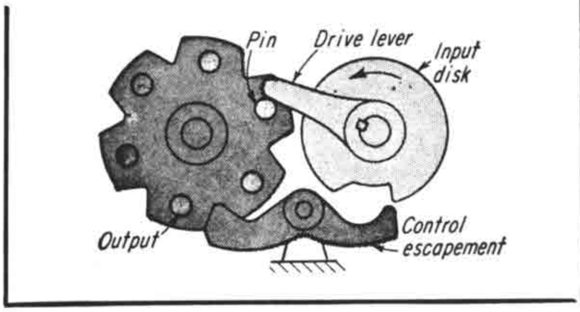
The associated linkage mechanism locks the geneva when it is not indexing. In the position shown, the locking roller is just about to exit from the geneva.



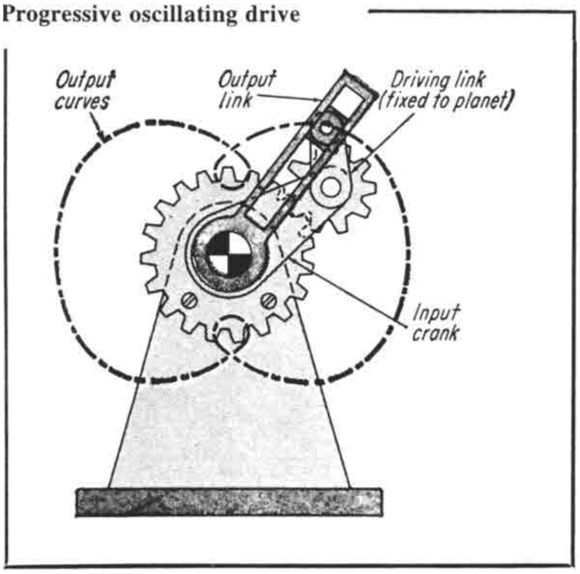
This arrangement permits the roller to exit and enter the driving slots tangentially. In the position shown, the driving roller has just completed indexing the geneva, and it is about to coast for 90º as it goes around the curve. (During this time, a separate locking device might be necessary to prevent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

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

Controlled-output escapement

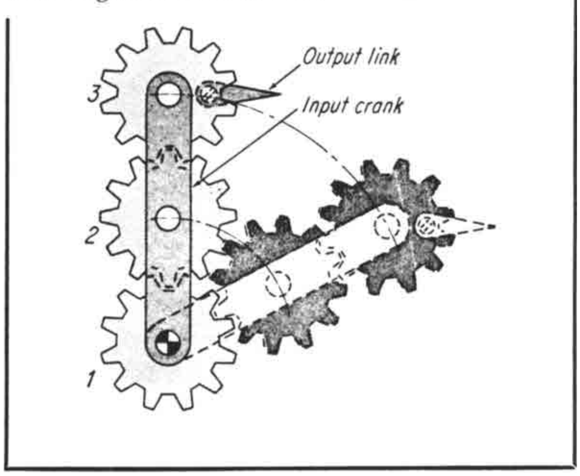


Progressive oscillating drive



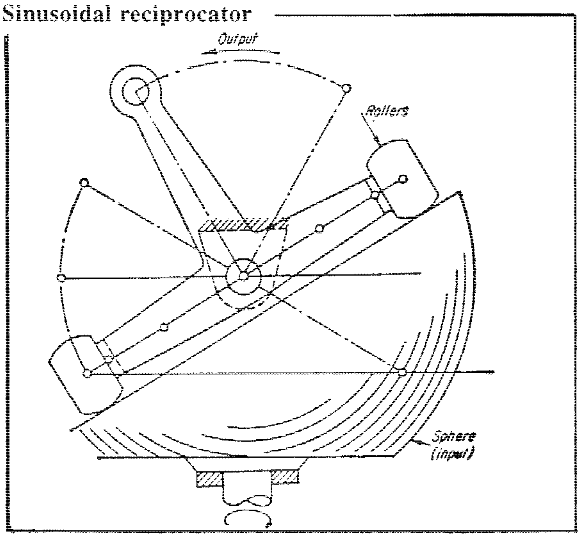
A crank attached to the planet gear can make point *P* describe the double loop curve illustrated. The slotted output crank oscillates briefly at the vertical positions.

Parallel-guidance mechanisms



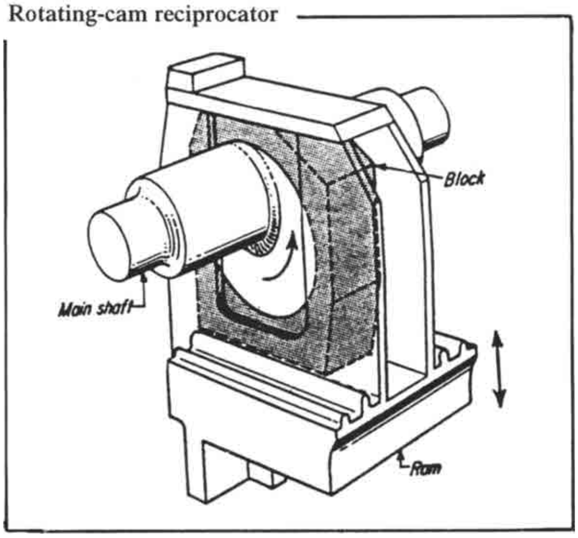
The input crank contains two planet gears. The center sun gear is fixed. By making the three gears equal in diameter and having gear *2* serve as an idler, any member fixed to gear *3* will remain parallel to its previous positions throughout the rotation of the input ring crank.

Sinusoidal reciprocator



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

Rotating-cam reciprocator



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

MODIFIED GENEVA DRIVES

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

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

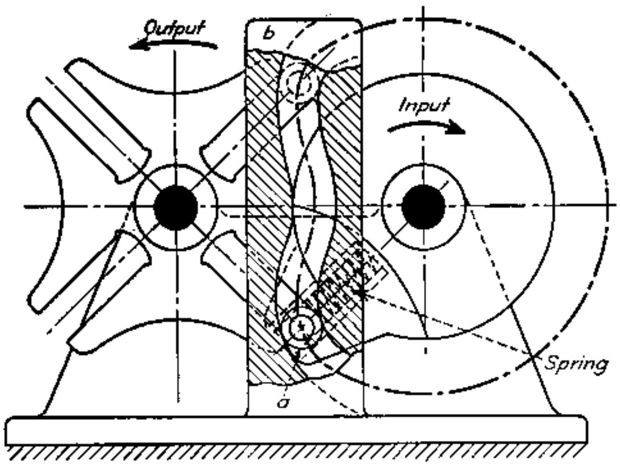


Fig. 1

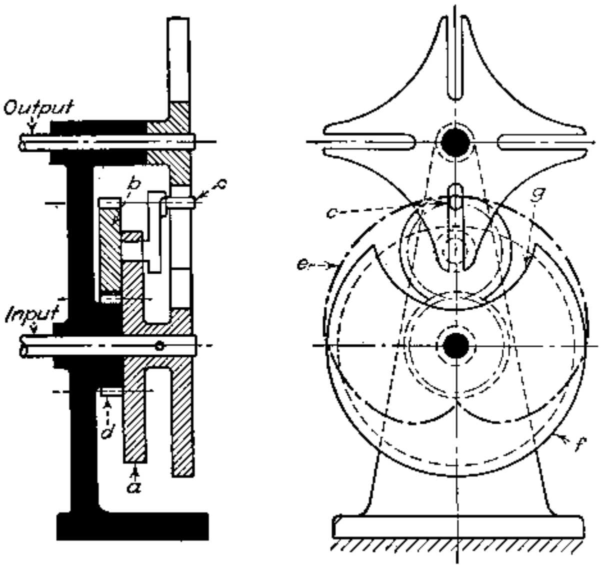


Fig. 2

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

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

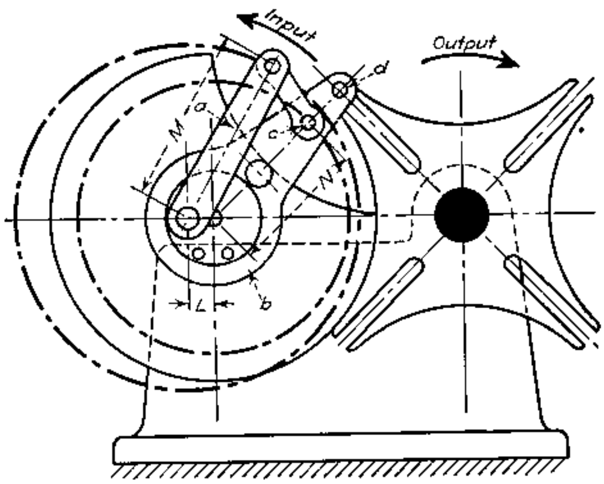


Fig. 3

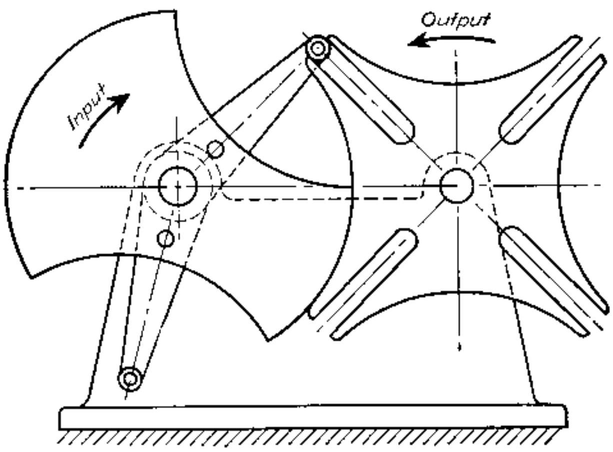


Fig. 4

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

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

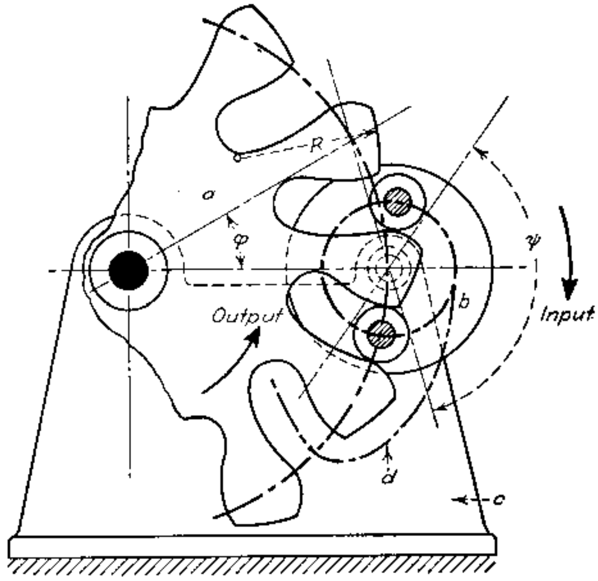


Fig. 5

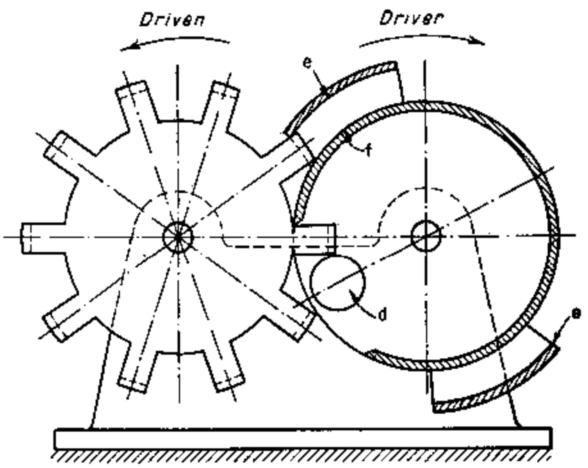
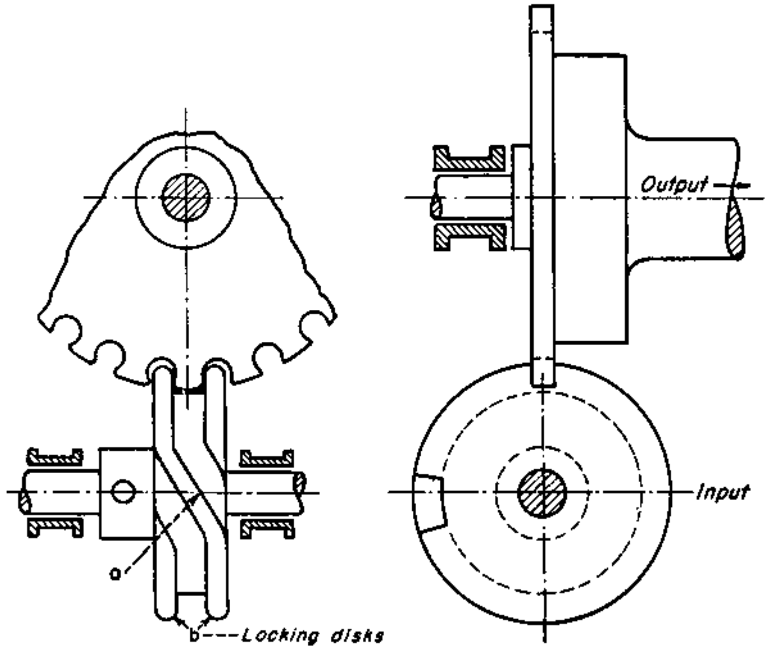


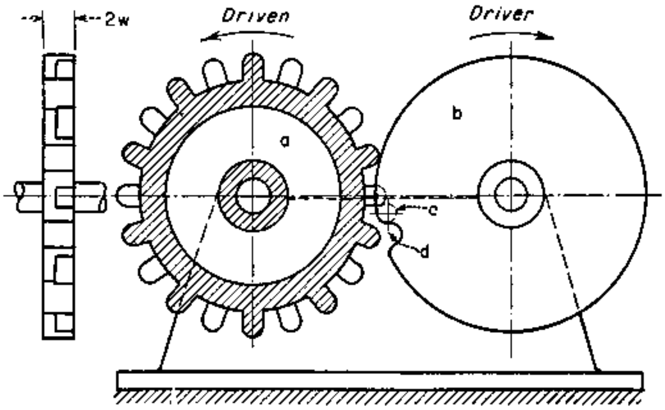
Fig. 6

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

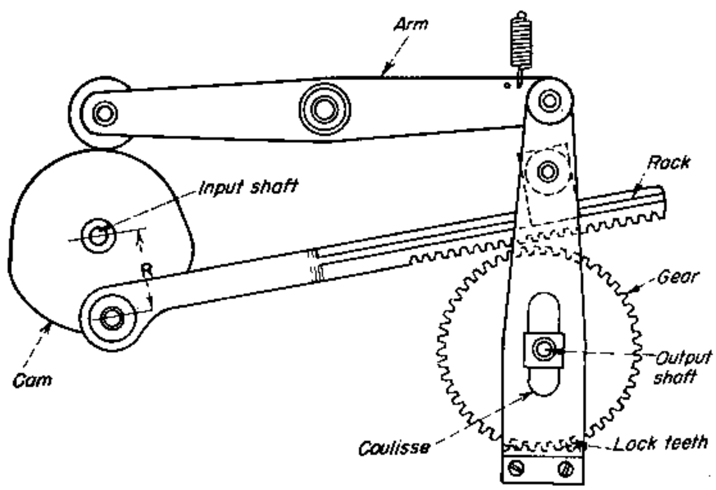
This mechanism transmits intermittent motion between two skewed shafts. The shafts need not be at right angles to one another. Angular displacement of the output shaft per revolution of input shaft equals the circular pitch of the output gear wheel divided by its pitch radius. The duration of the motion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

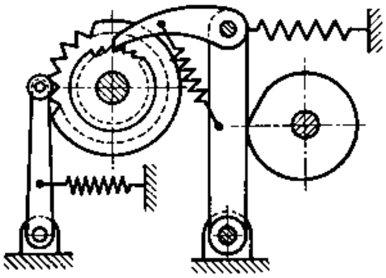


A "mutilated tooth" intermittent drive. Driver *b* is a circular disk of width *w* with a cutout *d* on its circumference. It carries a pin *c* close to the cutout. The driven gear, *a*, of width 2*w* has an even number of standard spur gear teeth. They alternately have full and half-width (mutilated)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

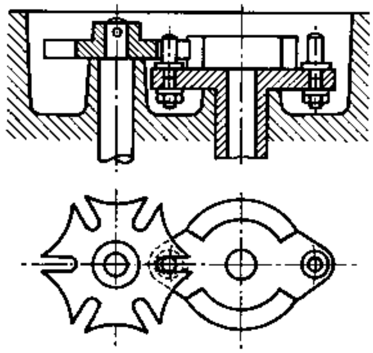


An operating cycle of 180º motion and 180º dwell is produced by this mechanism. The input shaft drives the rack, which is engaged with the output shaft gear during half the cycle. When the rack engages, the lock teeth at the lower end of the coulisse are disengaged and, conversely, when the rack is disengaged,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

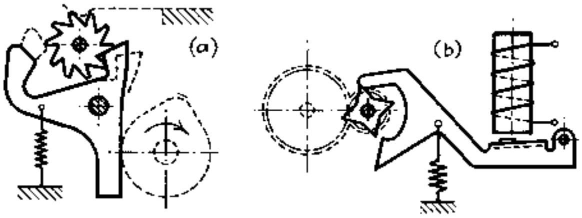




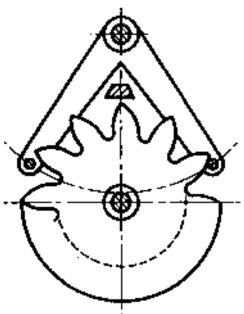
A cam-driven ratchet.



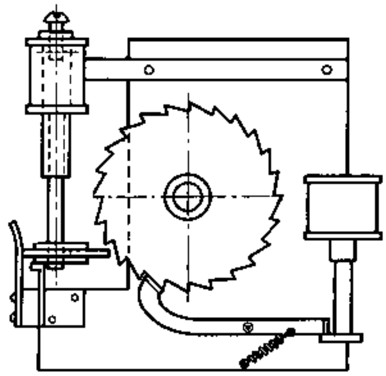
A six-sided Maltese cross and double driver give a 3:1 ratio.



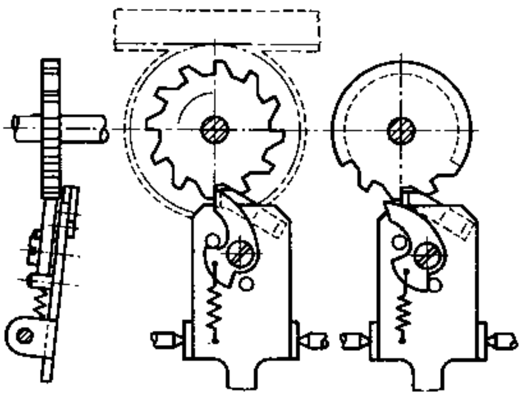
A cam operated escapement on a taximeter (a). A solenoid-operated escapement (b).



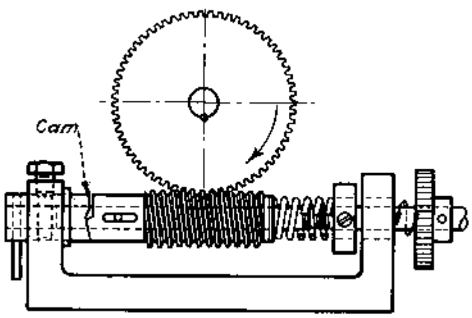
An escapement on an electric meter.



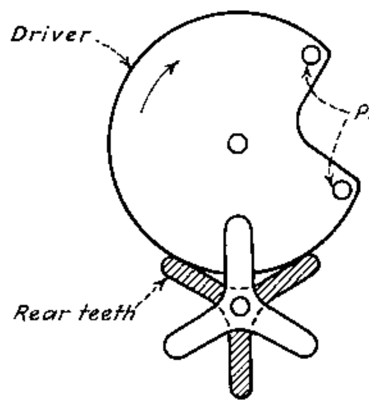
A solenoid-operated ratchet with a solenoid-resetting mechanism A sliding washer engages the teeth.



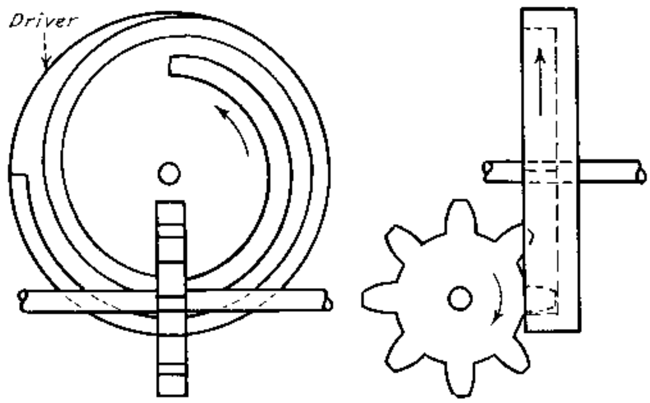
A plate oscillating across the plane of a ratchet-gear escapement carries stationary and spring-held pawls.



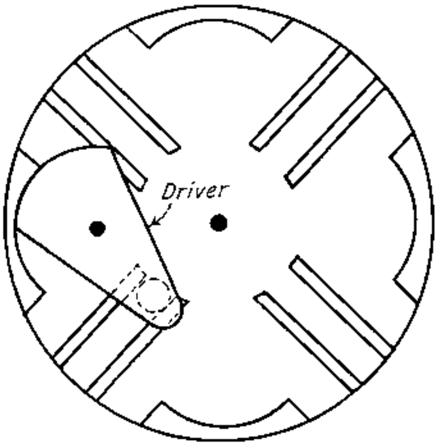
A worm drive, compensated by a cam on a work shaft, produces intermittent motion of the gear.



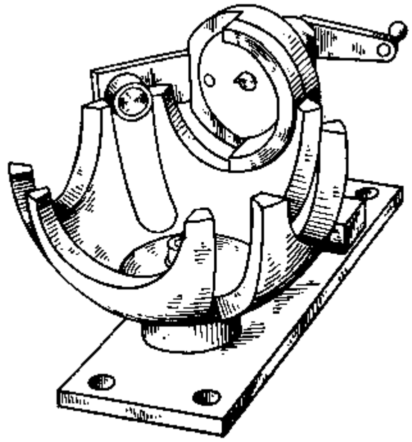
An intermittent counter mechanism. One revolution of the driver advances the driven wheel 120º. The driven-wheel rear teeth are locked on the cam surface during dwell.



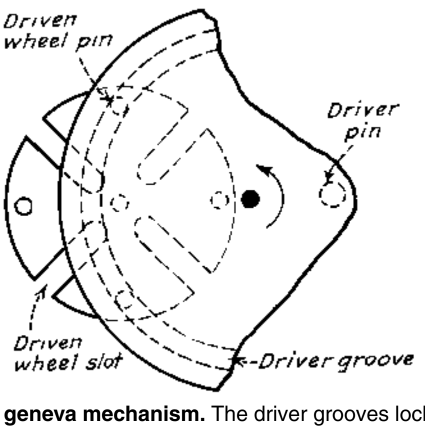
Spiral and wheel. One revolution of the spiral advances the driven wheel one tooth width. The driven-wheel tooth is locked in the driver groove during dwell.



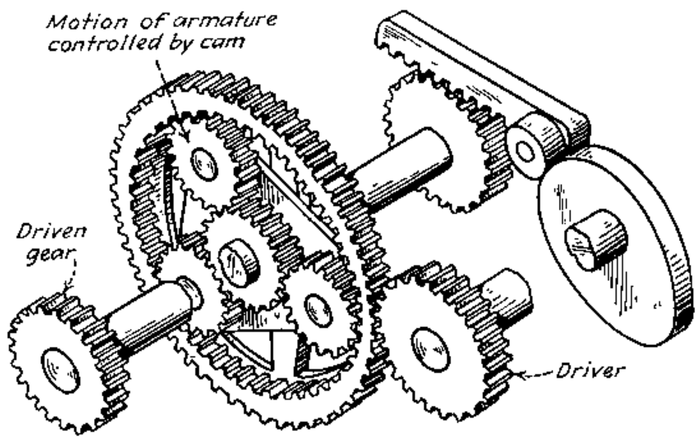
An internal geneva mechanism. The driver and driven wheel rotate in same direction. The duration of dwell is more than 180º of driver rotation.



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



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



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

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

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

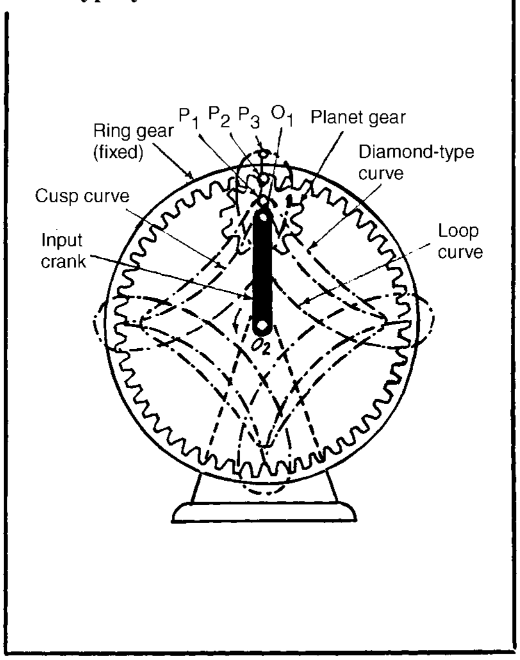
All the cycloidal mechanisms shown here are general. This results in compact positive mechanisms capable of operating at relatively high speeds with little backlash or "slop." These mechanisms can be classified into three groups:

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

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

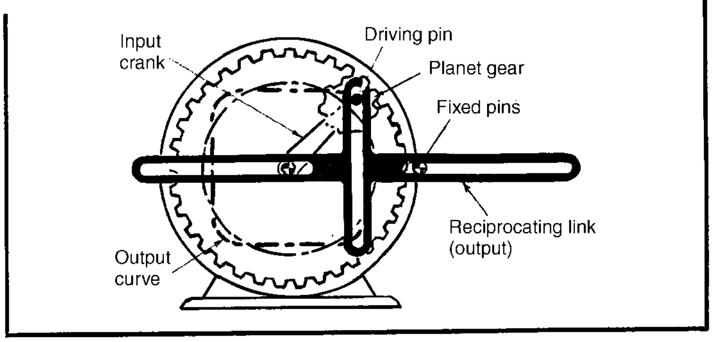
Pericycloid—the tracing points are located on an internal gear that rolls on a stationary external gear.

Basic hypocycloid curves



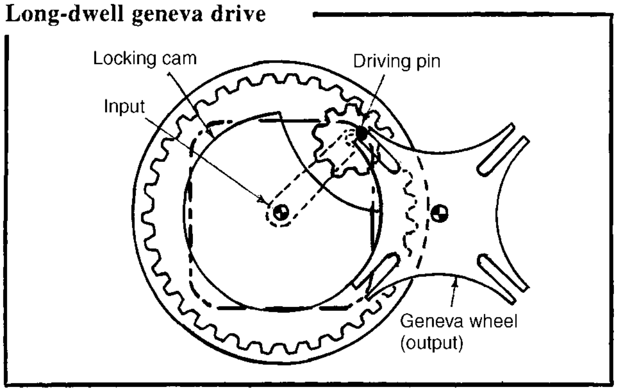
The input drives a planet in mesh with a stationary ring gear. Point P_1 on the planet gear describes a diamond-shape curve, point P_2 on the pitch line of the planet describes the familiar cusp curve, and point P_3, which is on an extension rod fixed to the planet gear, describes a loop-type curve.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

Double-dwell mechanism



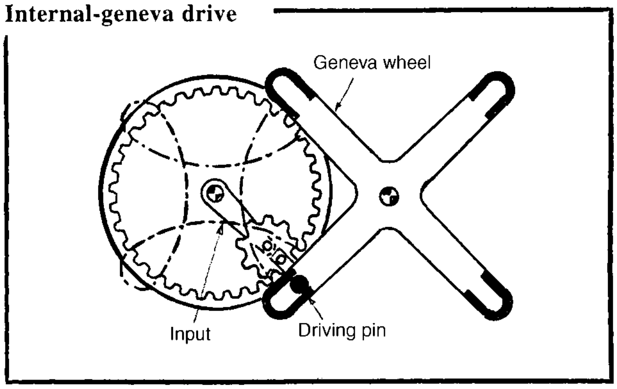
Coupling the output pin to a slotted member produces a prolonged dwell in each of the extreme positions. This is another application of the diamond-type hypocycloidal curve.

Long-dwell geneva drive



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

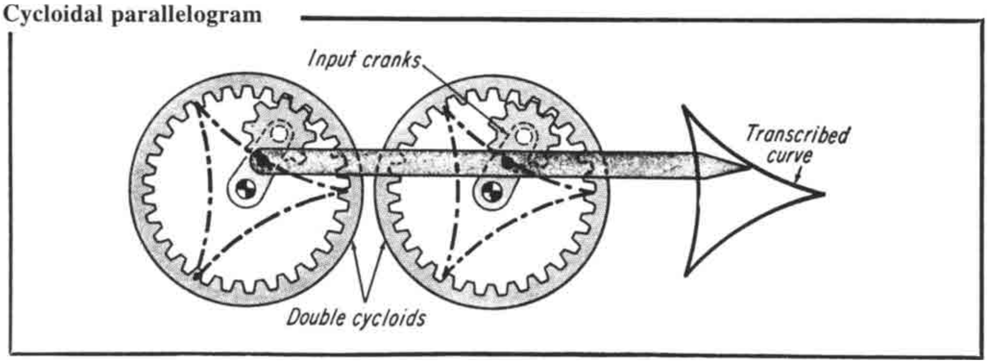
Internal-geneva drive



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

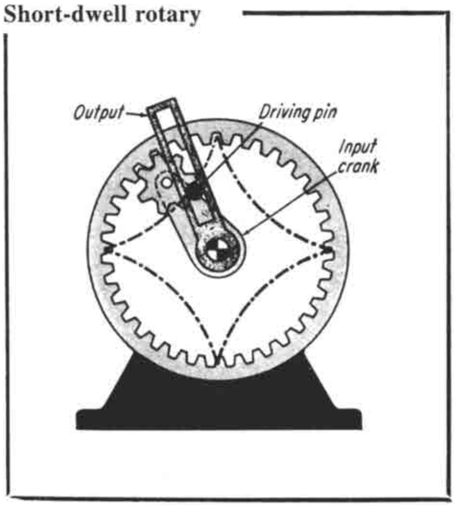
Cycloidal motion is popular for mechanisms in feeders and automatic machines.

Cycloidal parallelogram



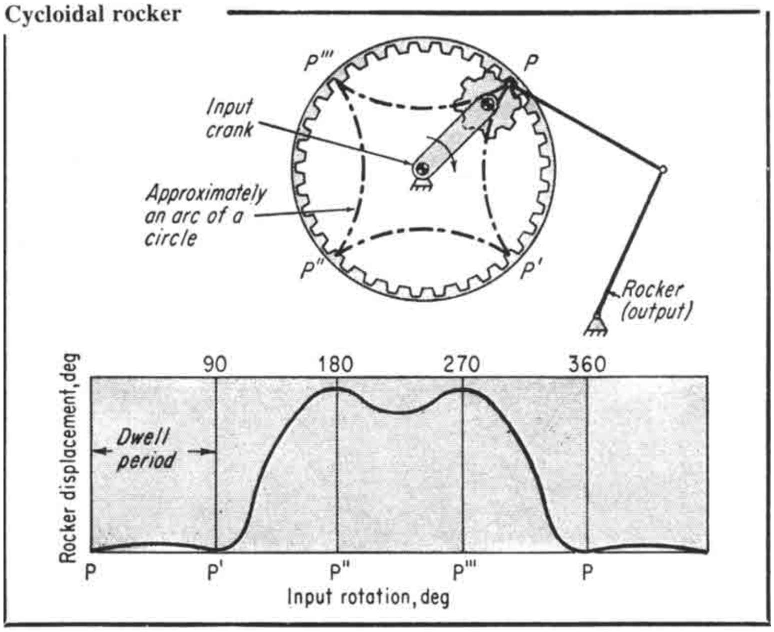
Two identical hypocycloid mechanisms guide the point of the bar along the triangularly shaped path. The mechanisms are useful where space is limited in the area where the curve must be described. These double-cycloid mechanisms can be designed to produce other curve shapes.

Short-dwell rotary



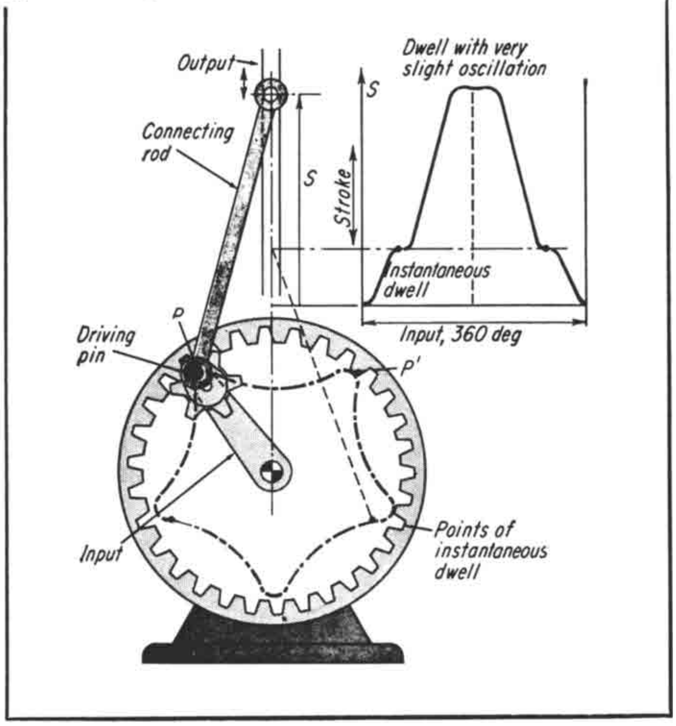
The pitch circle of this planet gear is exactly one-quarter that of the ring gear. A pin on the planet gear will cause the slotted output member to dwell four times during each revolution of the input shaft.

Cycloidal rocker



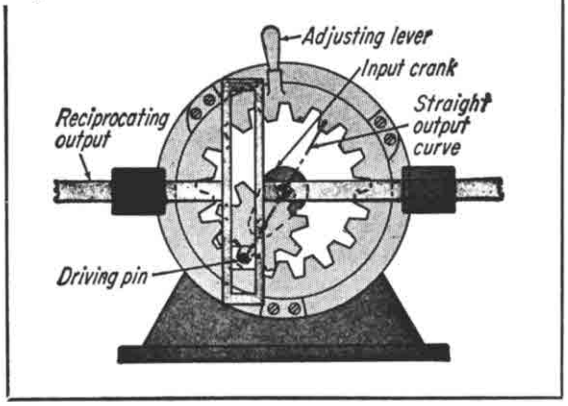
The curvature of the cusp is approximately that of an arc of a circle. Hence the rocker reaches a long dwell at the right extreme position while point P moves to P'. There is then a quick return from P' to P'', with a momentary dwell at the end of this phase. The rocker then undergoes a slight oscillation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

Cycloidal reciprocator



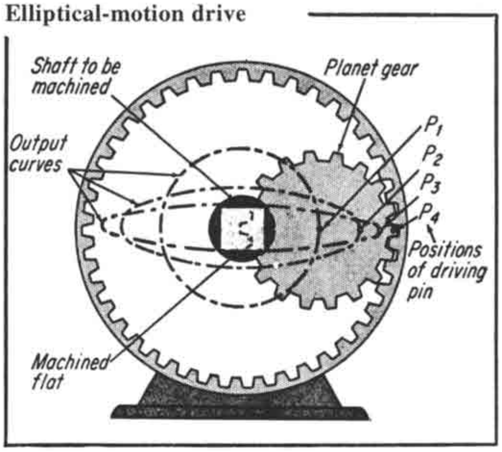
Part of curve P-P' produces a long dwell, but the five-lobe cycloidal curve avoids a marked oscillation at the end of the stroke. There are also two points of instantaneous dwell where the curve is perpendicular to the connecting rod.

Adjustable harmonic drive



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

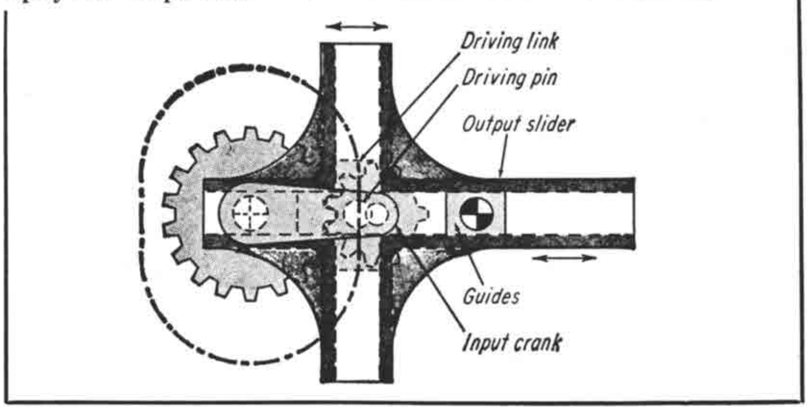
Elliptical-motion drive



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

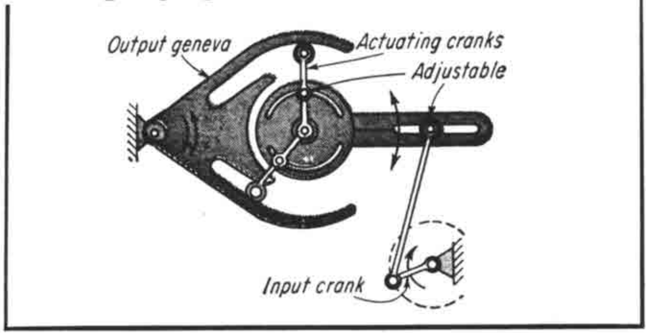
EPICYCLOID MECHANISMS

Epicycloid reciprocator



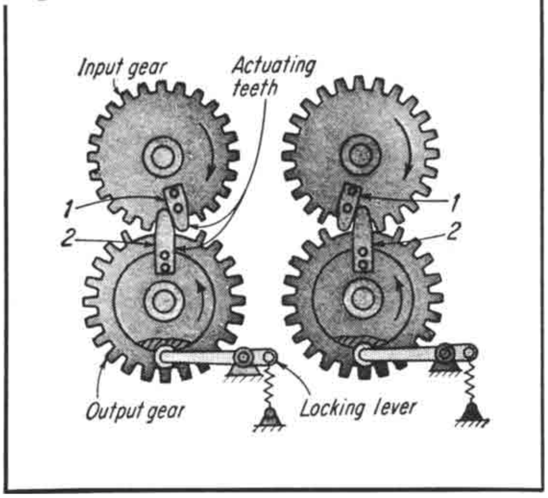
Here the sun gear is fixed and the planet is gear driven around it by the input link. There is no internal ring gear as with the hypocycloid mechanisms. Driving pin P on the planet describes the curve shown, which contains two almost-flat portions. If the pin rides in the slotted yoke, a short dwell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

Oscillating-output geneva



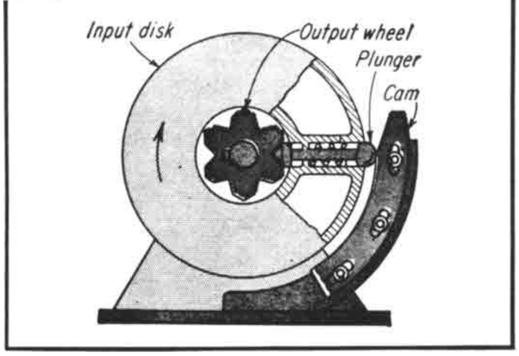
Three adjustable output-links provide a wide variety of oscillating motions. The input crank oscillates the central member that has an adjustable slot to vary the stroke. The oscillation is transferred to the two actuating rollers, which alternately enter the geneva slots to index it, first in one direction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

Single tooth indexer



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

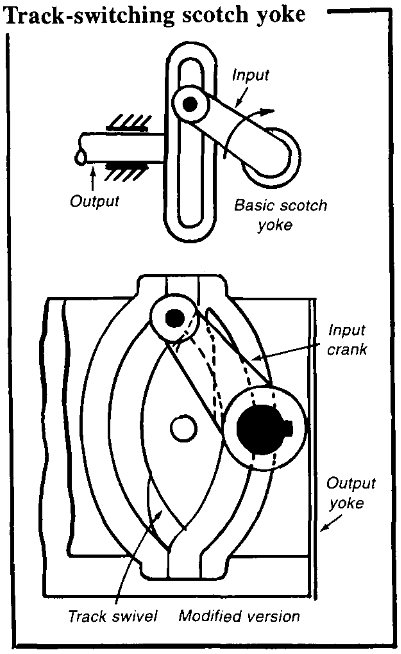
Plunger-actuated indexer



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

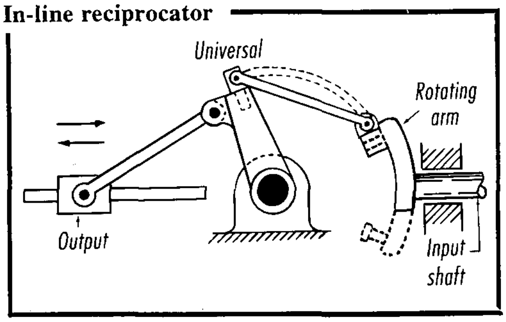
ROTARY-TO-RECIPROCATING DEVICES

Track-switching scotch yoke



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

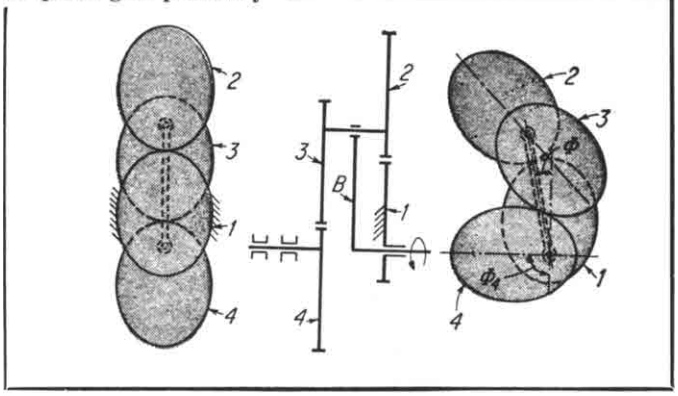
In-line reciprocator



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

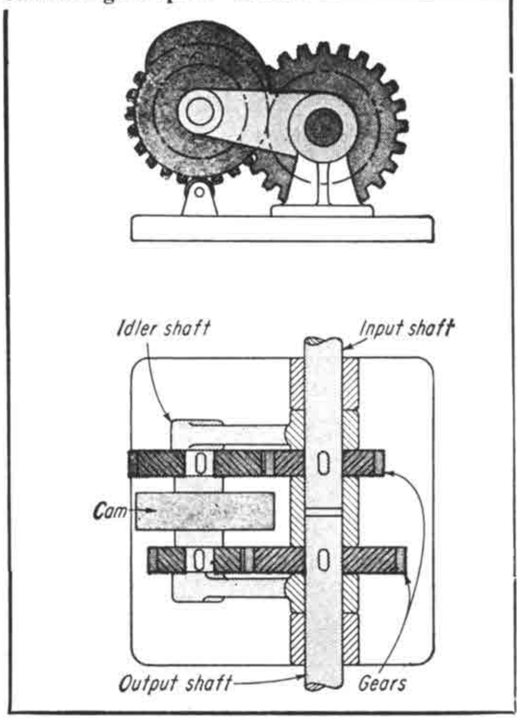
VARIABLE SPEED DEVICES

Elliptical-gear planetary



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

Cammed-gear speed variator



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

ADJUSTABLE-SPEED DRIVES

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

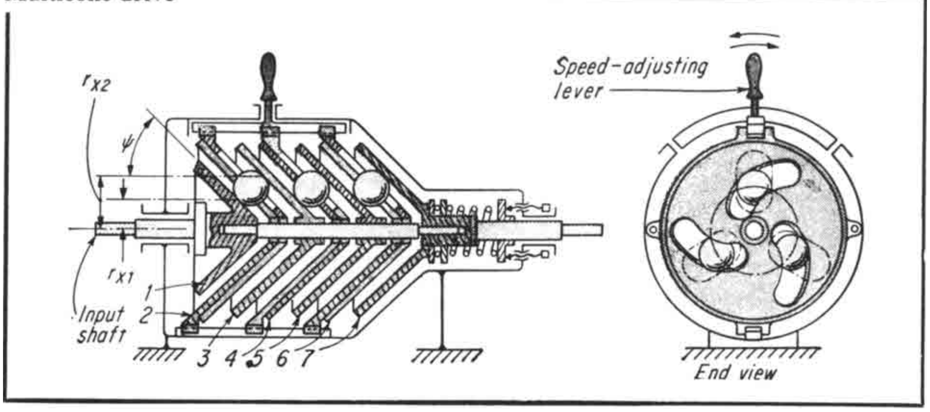
The effective radii can be calculated from

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

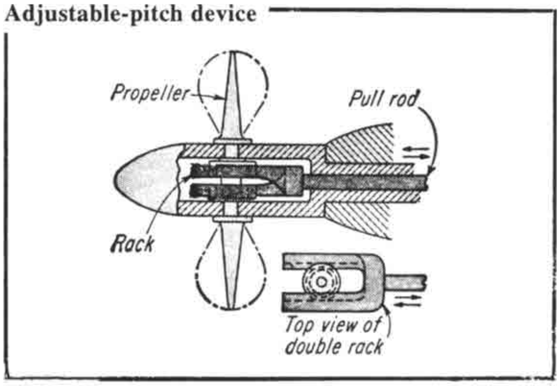
$$r_{x2} = R_x + 1/2\, D \cos \psi$$

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

Multicone drive



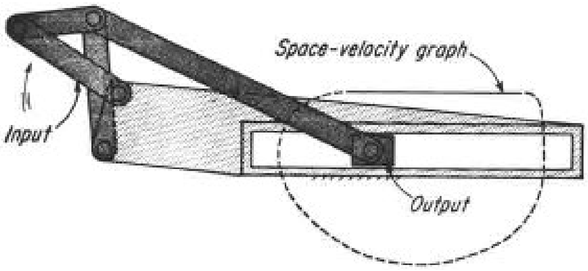
Adjustable-pitch device



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

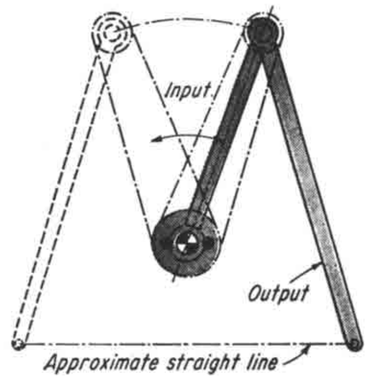
ROTARY-TO-RECIPROCATING MOTION AND DWELL MECHANISMS

Four-bar slider mechanism



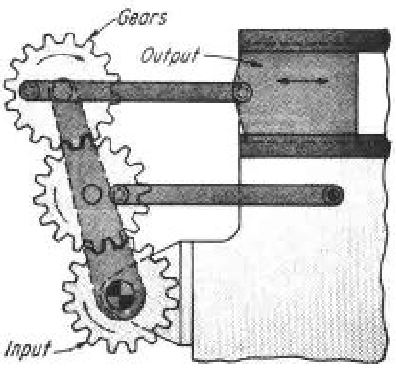
With proper dimensions, the rotation of the input link can impart an almost-constant velocity motion to the slider within the slot.

Oscillating-chain mechanism



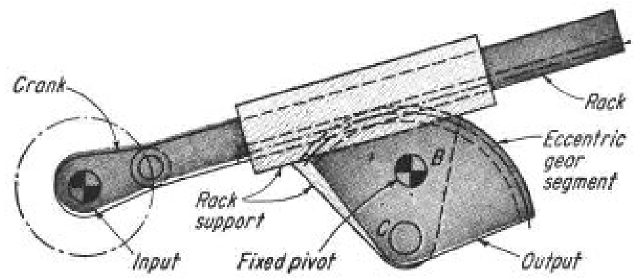
The rotary motion of the input arm is translated into linear motion of the linkage end. The linkage is fixed to the smaller sprocket, and the larger sprocket is fixed to the frame.

Three-gear stroke multiplier



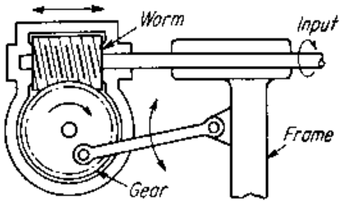
The rotation of the input gear causes the connecting link, attached to the machine frame, to oscillate. This action produces a large-stroke reciprocating motion in the output slider.

Rack and gear sector



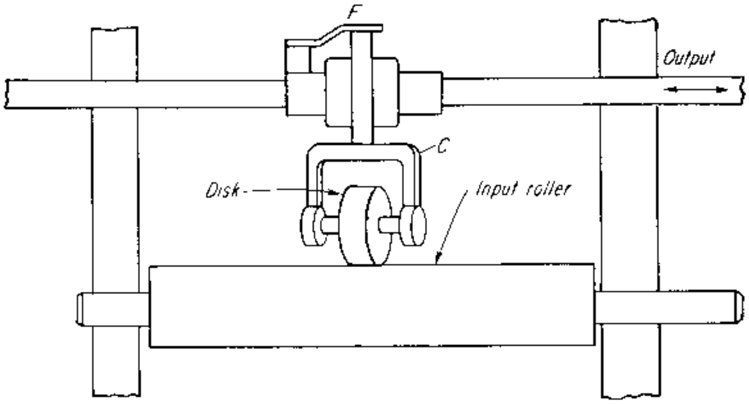
The rotary motion of the input shaft is translated into an oscillating motion of the output gear segment. The rack support and gear sector are pinned at *C* but the gear itself oscillates around *B*.

Linear reciprocator



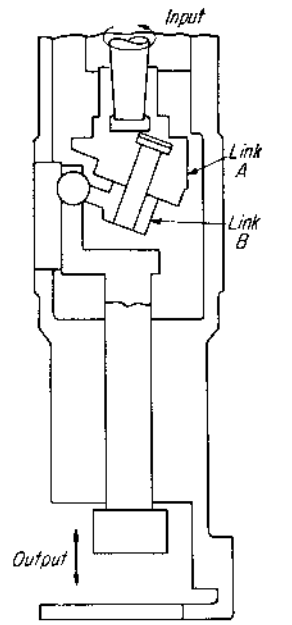
This linear reciprocator converts a rotary motion into a reciprocating motion that is *in line* with the input shaft. Rotation of the shaft drives the worm gear which is attached to the machine frame with a rod. Thus input rotation causes the worm gear to draw itself (and the worm) to the right—thus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

Disk and roller drive



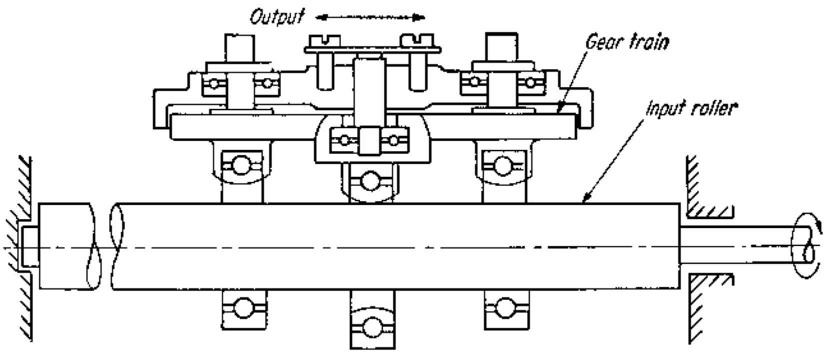
A hardened disk in this drive, riding at an angle to the axis of an input roller, transforms the rotary motion into linear motion parallel to the axis of the input. The roller is pressed against the input shaft by flat spring *F*. The feed rate is easily varied by changing the angle of the disk. This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

Reciprocating space crank



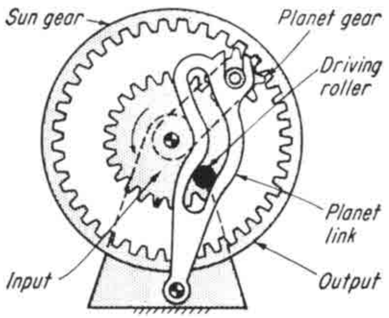
The rotary input of this crank causes the bottom surface of link *A* to wobble with respect to the center link. Link *B* is free of link *A*, but it is restrained from rotating by the slot. This causes the output member to reciprocate linearly.

Bearing and roller drive



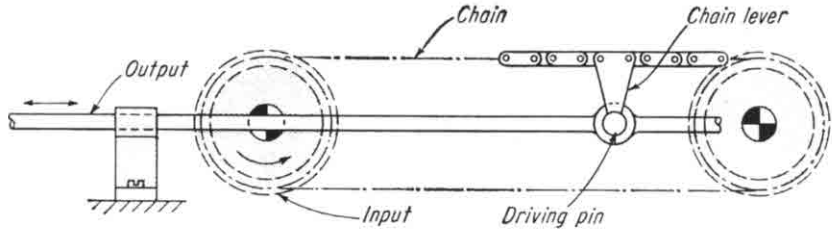
This drive arrangement avoids large Hertzian stresses between the disk and roller by including three ball bearings in place of the single disk. The inner races of the bearings make contact on one side or the other. Hence a gearing arrangement is required to alternate the angle of the bearings. This arrangement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

Oscillating crank and planetary drive



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

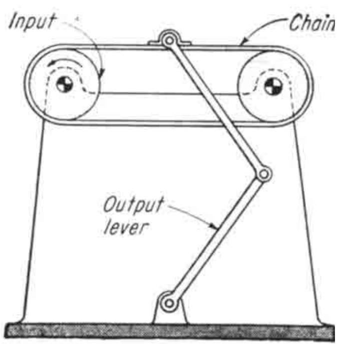
Chain-slider drive



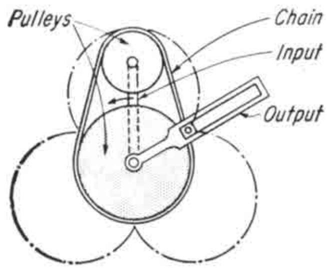
The chain link drives a lever that oscillates. A slowdown-dwell occurs when the chain pin passes around the left sprocket.

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

Chain-oscillating drive

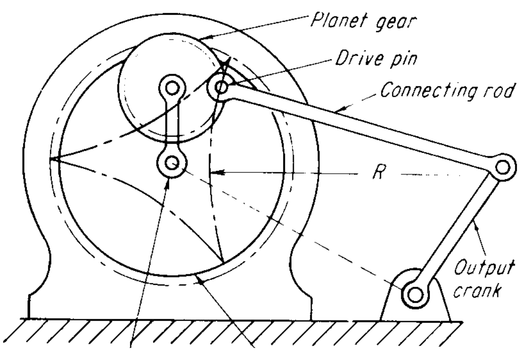


Chain and slider drive



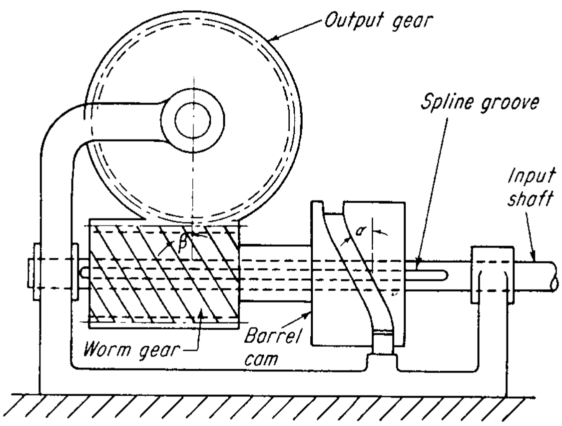
The input crank causes the small pulley to orbit around the stationary larger pulley. A pivot point attached to the chain slides inside the slot of the output link. In the position shown, the output is about to start a long dwell period of about 120º.

Epicyclic dwell mechanism



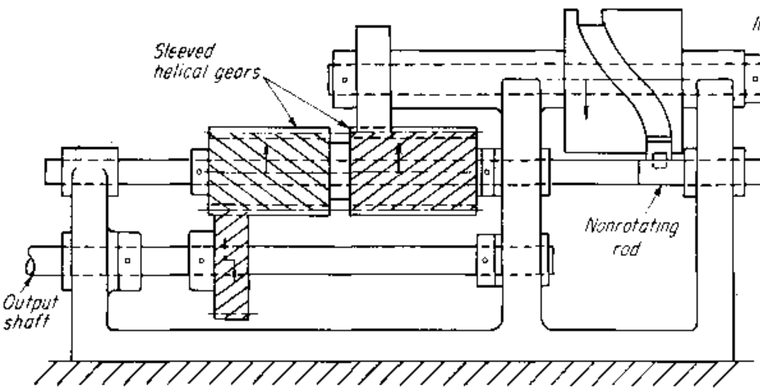
The output crank pulsates back and forth with a long dwell at its extreme right position. The input shaft rotates the planet gear with a crank. The pin on the planet gear traces the epicyclic three-lobe curve shown. The right side of the curve is a near circular arc of radius *R*. If the connecting rod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

Cam-worm dwell mechanism



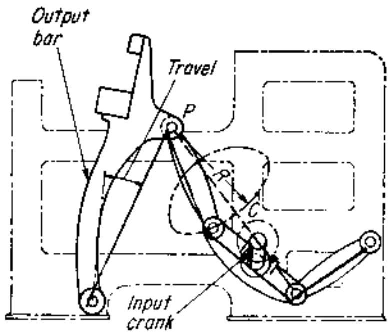
Without the barrel cam, the input shaft would drive the output gear by the worm gear at constant speed. The worm and the barrel cam, however, can slide linearly on the input shaft. The rotation of the input shaft now causes the worm gear to be cammed back and forth, thus adding or subtracting motion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

Cam-helical dwell mechanism



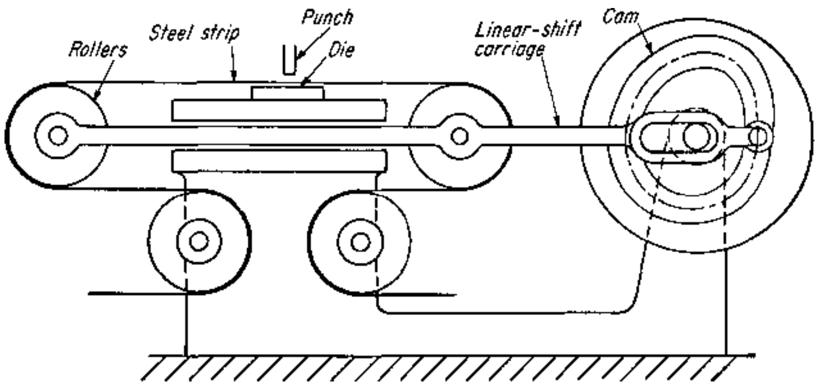
When one helical gear is shifted linearly (but prevented from rotating) it will impart rotary motion to the mating gear because of the helix angle. This principle is applied in the mechanism illustrated. The rotation of the input shaft causes the intermediate shaft to shift to the left, which in turn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

Six-bar dwell mechanism



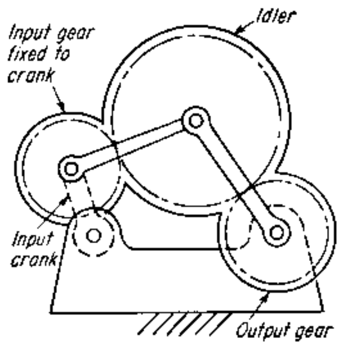
The rotation of the input crank causes the output bar to oscillate with a long dwell at its extreme right position. This occurs because point *C* describes a curve that is approximately a circular arc (from *C* to *C′* with its center at *P*. The output is almost stationary during that part of the curve.

Cam-roller dwell mechanism



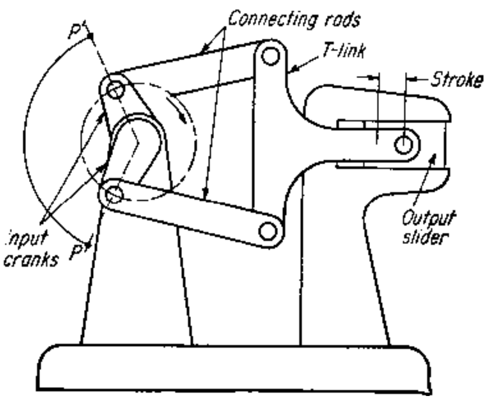
A steel strip is fed at constant linear velocity in this mechanism. But at the die station (illustrated), it is desired to stop the strip so that the punching operation can be performed. The strip passes over movable rollers which, when shifted to the right, cause the strip to move to the right. Since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

Three-gear drive

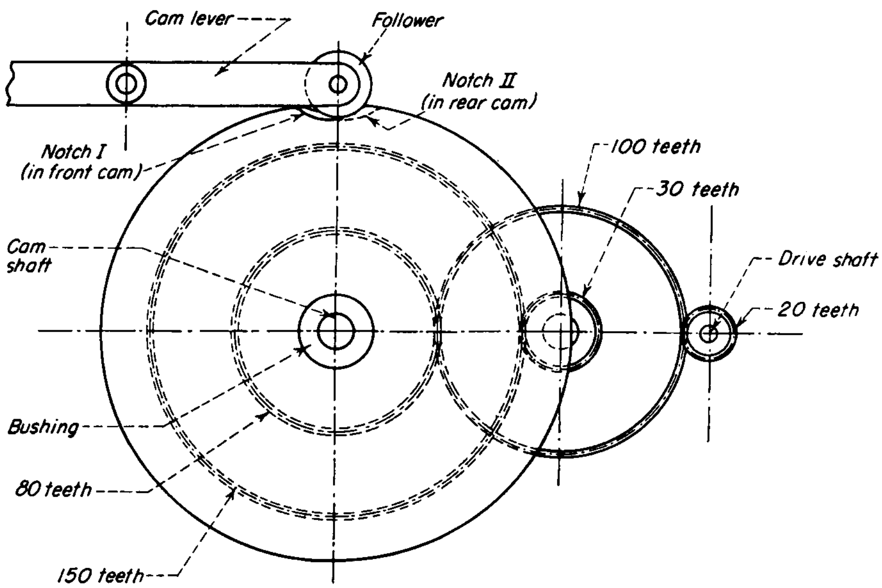


This is actually a four-bar linkage combined with three gears. As the input crank rotates, it turns the input gear which drives the output gear through the idler. Various output motions are possible. Depending on the relative diameters of the gears, the output gear can pulsate, reach a short dwell, or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

Double-crank dwell mechanism



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

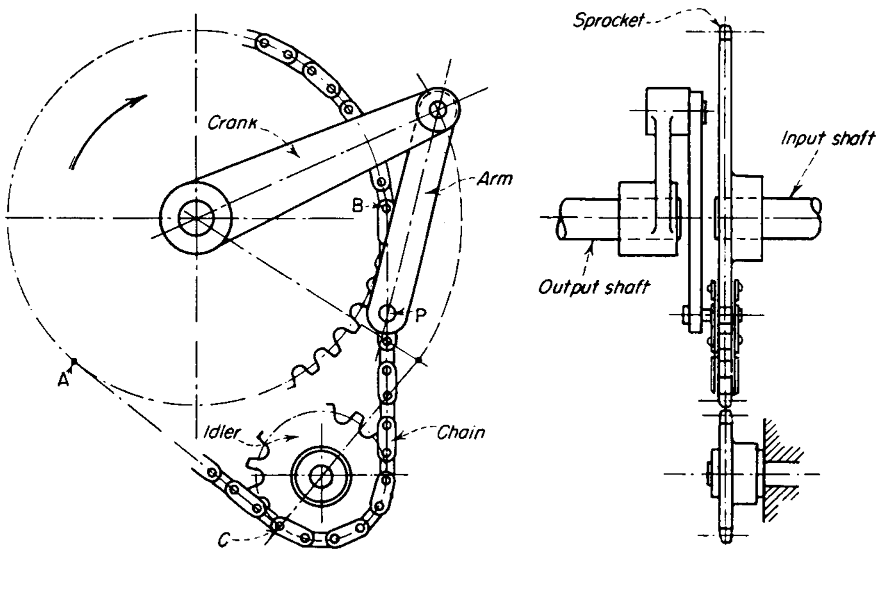


Fast Cam-Follower Motion

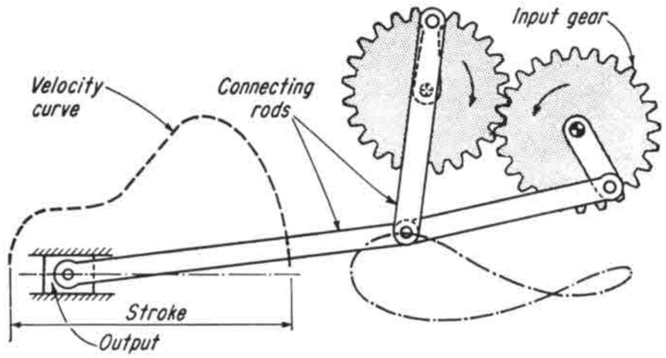
Fast cam action every n cycles (where n is a relatively large number) can be obtained with this manifold cam and gear mechanism. A single notched cam geared $1/n$ to a shaft turning once per cycle moves relatively slowly under the follower. The double notched-cam arrangement shown is designed to operate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

Intermittent Motion

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

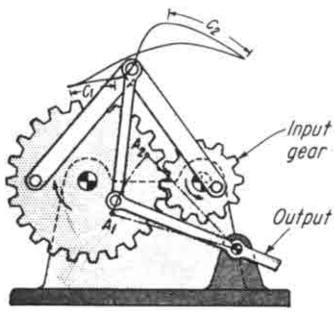


Gear-slider crank



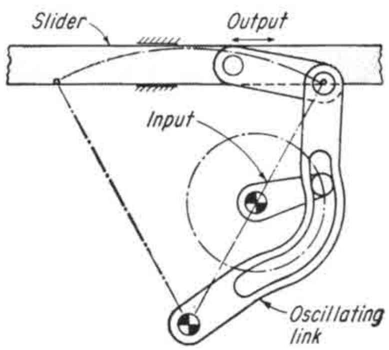
The input shaft drives both gears which, in turn, drive the connecting rods to produce the velocity curve shown. The piston moves with a low constant velocity.

Gear oscillating crank



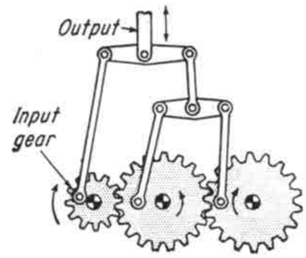
In this arrangement, the curve described by the pin connection has two parts, C_1 and C_2, which are very close to circular arc with its centers at A_1 and A_2. Consequently the driven link will have a dwell at both of its extreme positions.

Curve slider drive



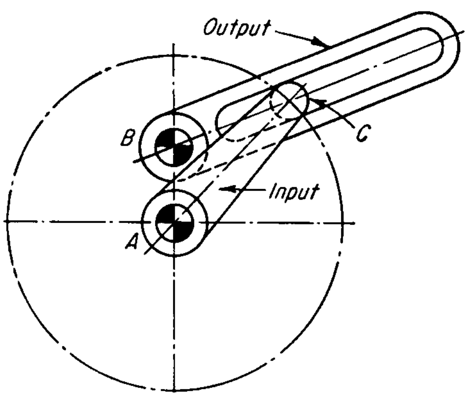
The circular arc on the oscillating link permits the link to reach a dwell during the right position of the output slider.

Triple-harmonic drive



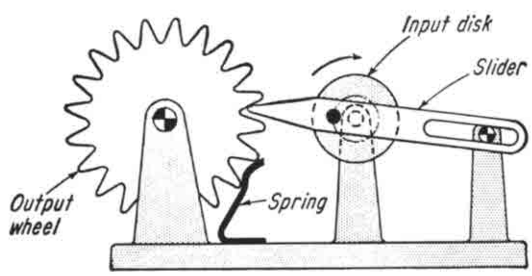
The input shaft drives three gears with connecting rods. A wide variety of reciprocating output motions can be obtained by selecting different lengths for the linkages. In addition, one to several dwells can be obtained per cycle.

Whitworth quick-return drive



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

Wheel and slider drive



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

FRICTION DEVICES FOR INTERMITTENT ROTARY MOTION

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

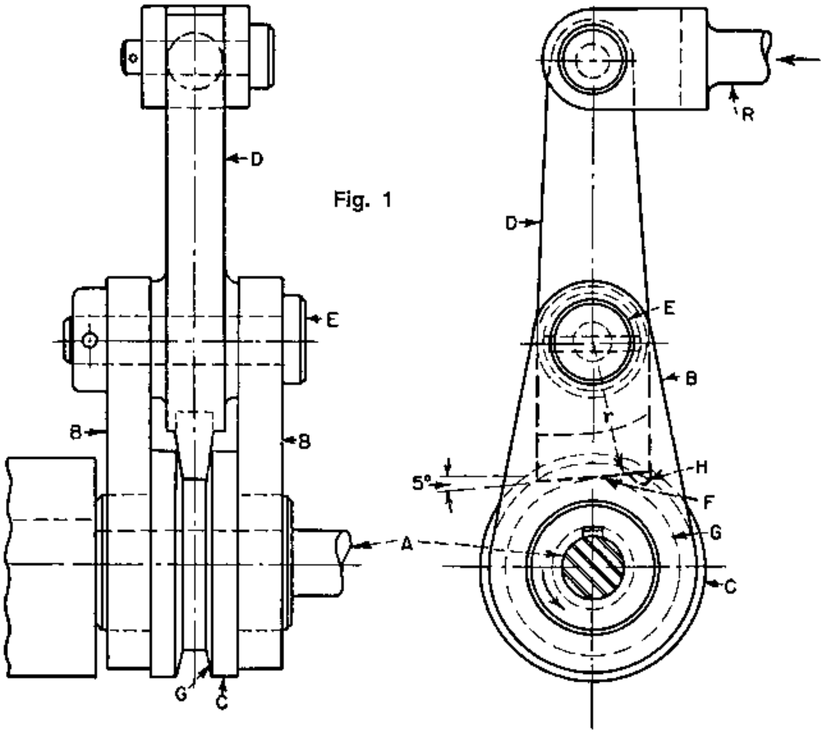


Fig. 1

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

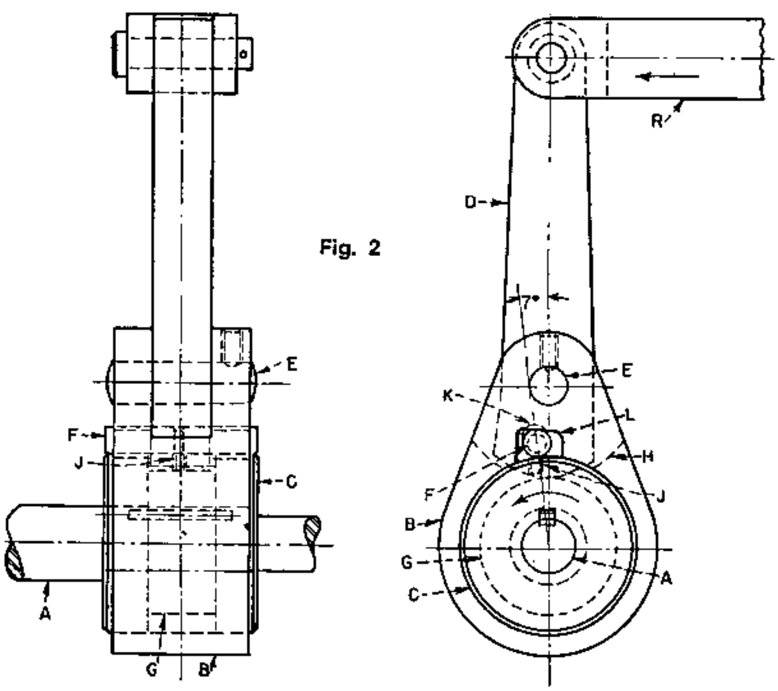
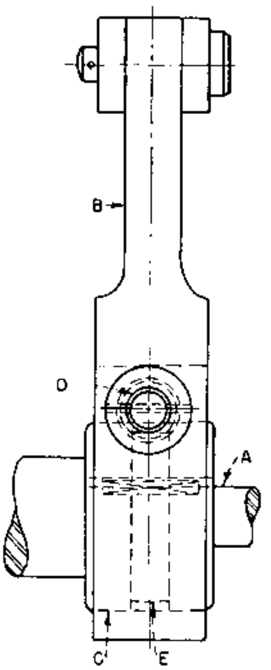


Fig. 2

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

Fig. 3



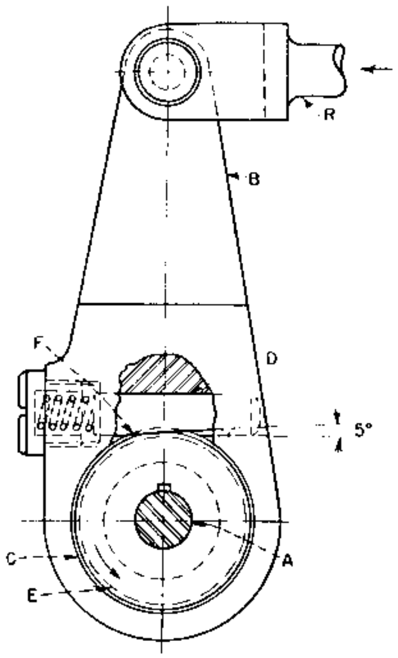


Fig. 3 The sliding pin and disk mechanism: The counterclockwise movement of body *B* about shaft *A* draws pin *D* to the right with respect to body *B*, aided by spring pressure, until the flat bottom *F* of the pin is wedged against the annular groove *E* of ring *C*. The bottom of the pin is inclined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

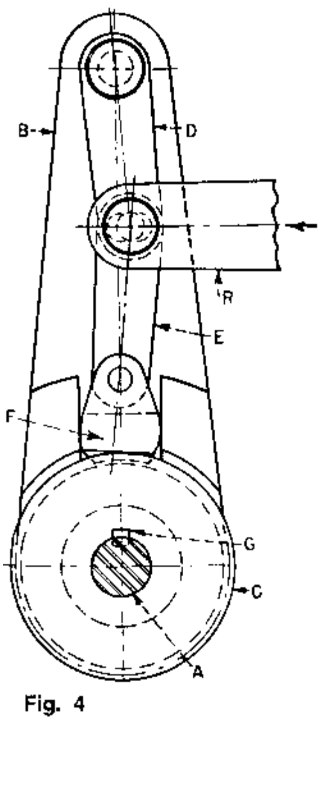


Fig. 4

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

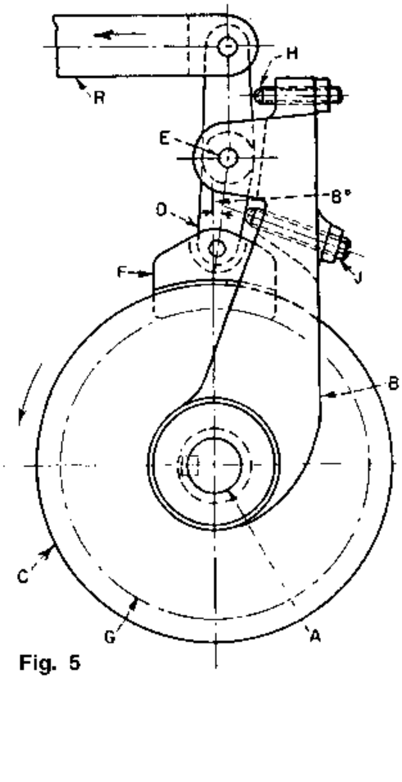


Fig. 5

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

NO TEETH ON THESE RATCHETS

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

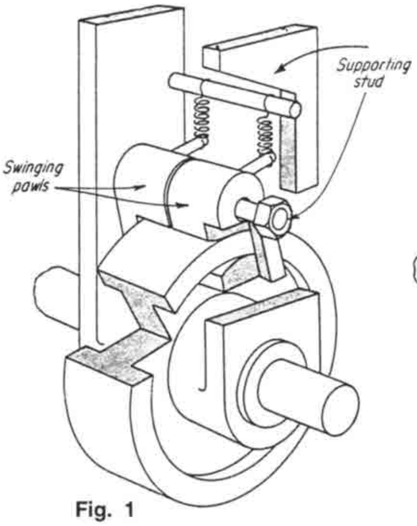


Fig. 1

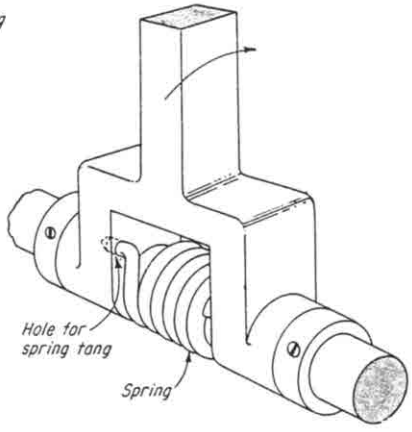


Fig. 2

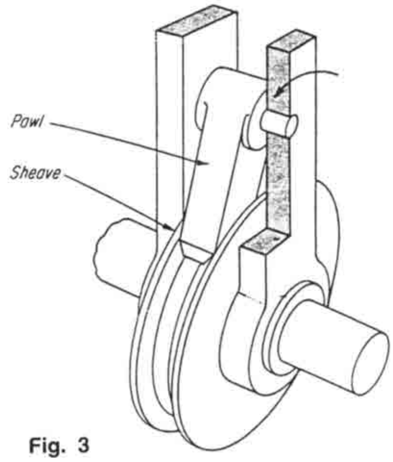


Fig. 3

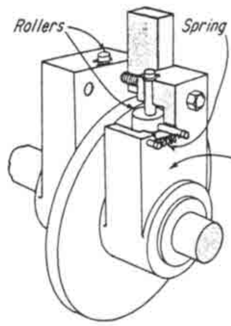


Fig. 4

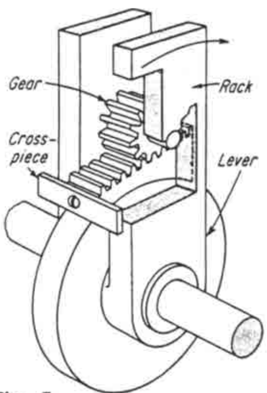


Fig. 5

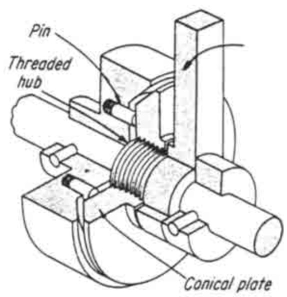


Fig. 6

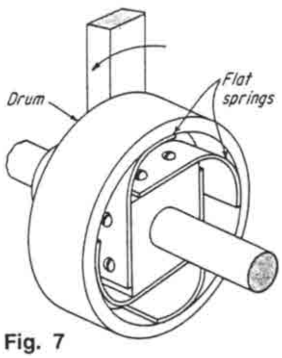


Fig. 7

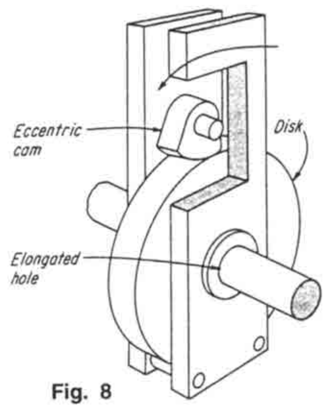


Fig. 8

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

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

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

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

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

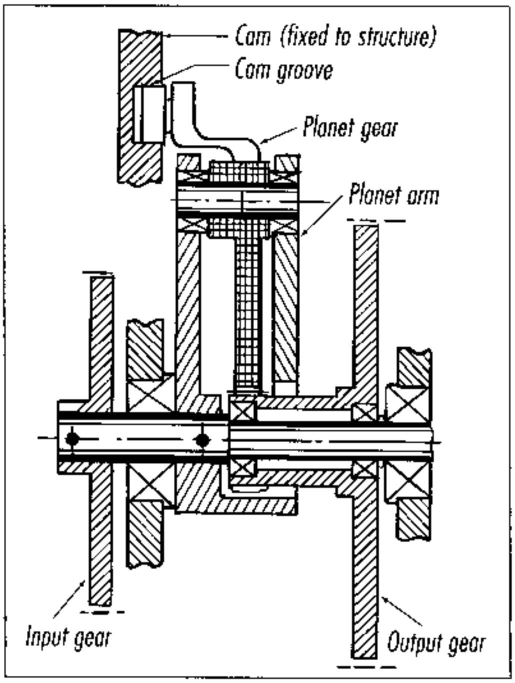
Fig. 6 A conical plate moves like a nut back and forth along the threaded center hub of the lever. The light friction of spring-loaded pins keeps the plate from rotating with the hub.

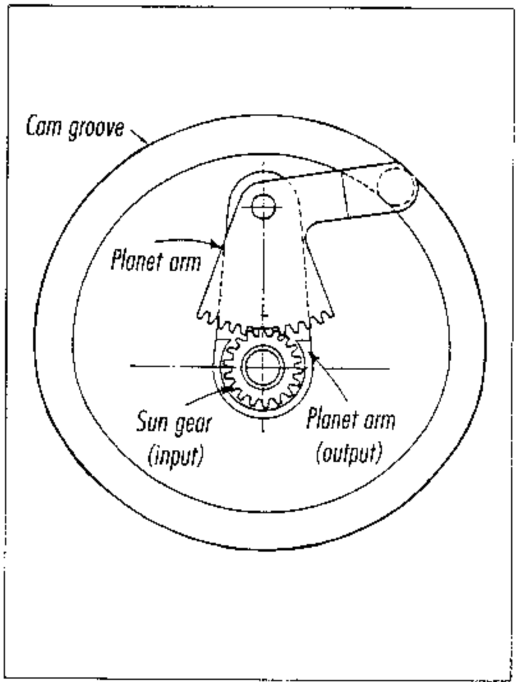
Fig. 7 Flat springs expand against the inside of a drum when a lever moves one way, but they drag loosely when the lever turns the drum in the opposite direction.

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

CAM-CONTROLLED PLANETARY GEAR SYSTEM

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





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

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

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

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

How the Mechanism Works

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

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

Comparison with Other Devices

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

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

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

Design Procedure

The basic equation for an epicyclic gear train is:

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

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

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

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

CHAPTER 5
SPECIAL-PURPOSE MECHANISMS

NINE DIFFERENT BALL SLIDES FOR LINEAR MOTION

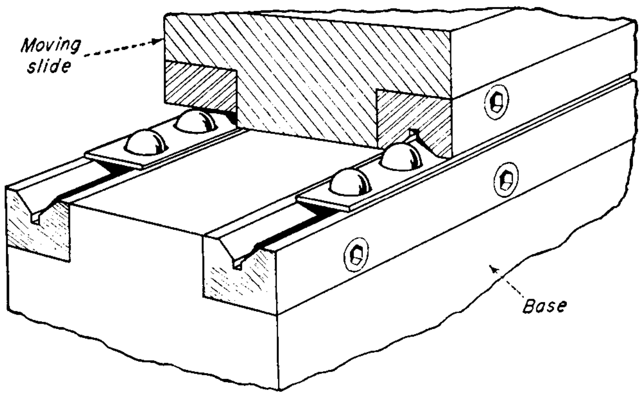


Fig. 1 V-grooves and flat surface make a simple horizontal ball slide for reciprocating motion where no side forces are present and a heavy slide is required to keep the balls in continuous contact. The ball cage ensures the proper spacing of the balls and its contacting surfaces are hardened and lapped.

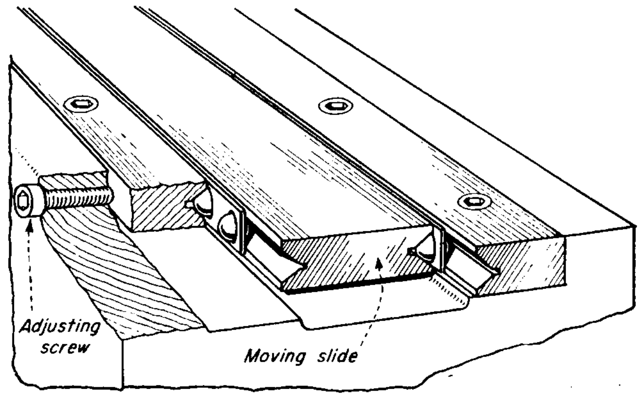


Fig. 2 Double V grooves are necessary where the slide is in a vertical position or when transverse loads are present. Screw adjustment or spring force is required to minimize any looseness in the slide. Metal-to-metal contact between the balls and grooves ensure accurate motion.

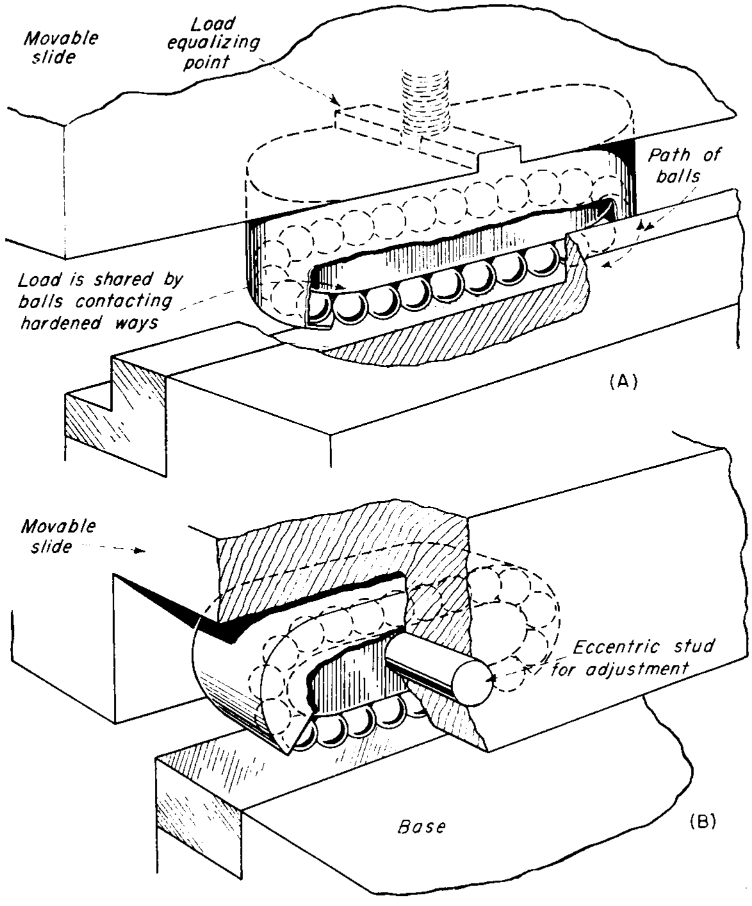


Fig. 3 The ball cartridge has the advantage of unlimited travel because the balls are free to recirculate. Cartridges are best suited for vertical loads. (A) Where lateral restraint is also required, this type is used with a side preload. (B) For flat surfaces the cartridge is easily adjusted.

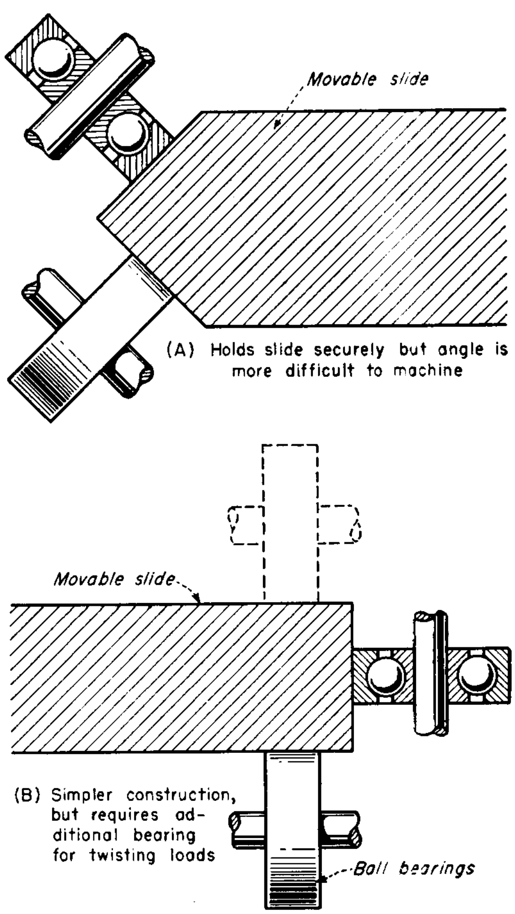


Fig. 4 Commercial ball bearings can be used to make a reciprocating slide. Adjustments are necessary to prevent looseness of the slide. (A) Slide with beveled ends, (B) Rectangular-shaped slide.

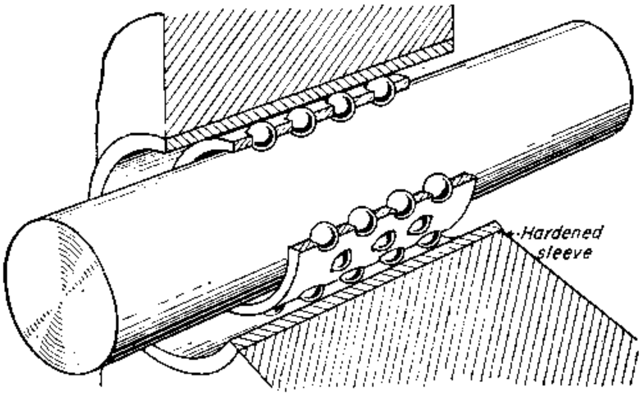


Fig. 5 This sleeve bearing, consisting of a hardened sleeve, balls, and retainer, can be used for reciprocating as well as oscillating motion. Travel is limited in a way similar to that of Fig. 6. This bearing can withstand transverse loads in any direction.

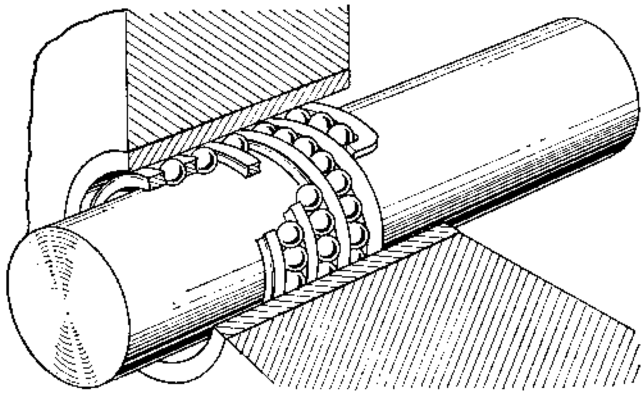


Fig. 6 This ball reciprocating bearing is designed for rotating, reciprocating or oscillating motion. A formed-wire retainer holds the balls in a helical path. The stroke is about equal to twice the difference between the outer sleeve and the retainer length.

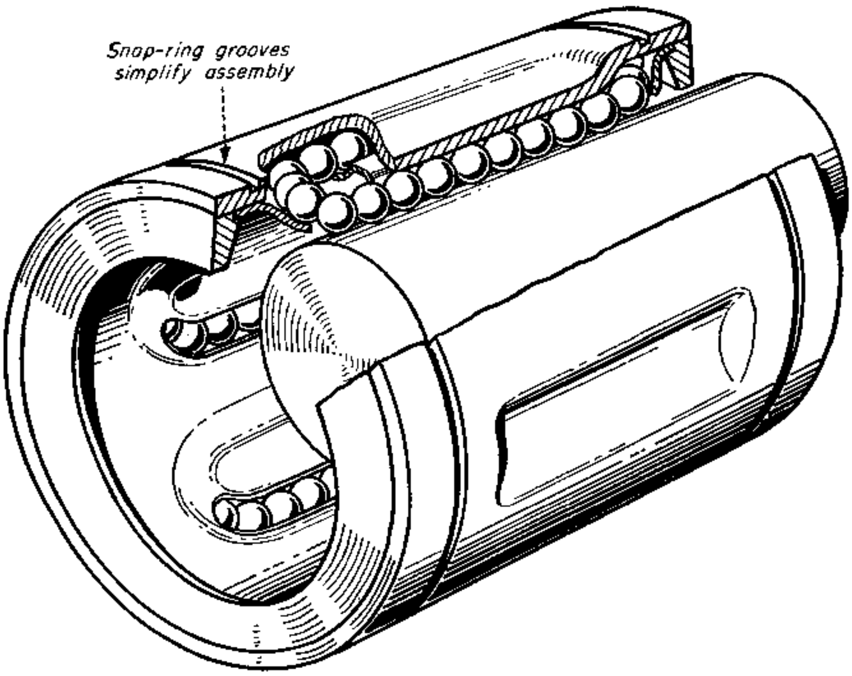


Fig. 7 This ball bushing has several recirculating systems of balls that permit unlimited linear travel. Very compact, this bushing requires only a bored hole for installation. For maximum load capacity, a hardened shaft should be used.

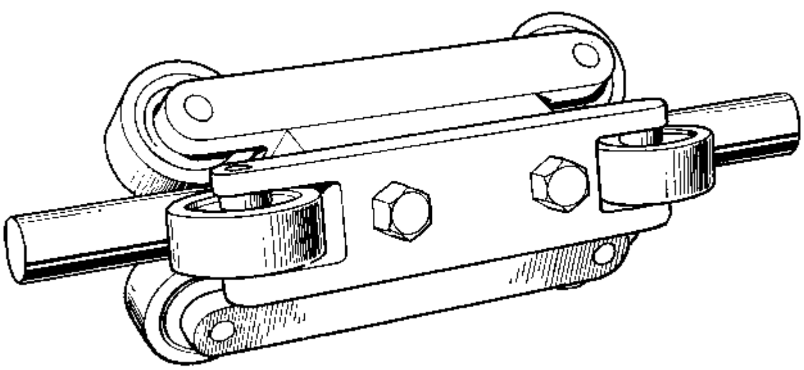


Fig. 8 Cylindrical shafts can be held by commercial ball bearings that are assembled to make a guide. These bearings must be held tightly against the shaft to prevent any looseness.

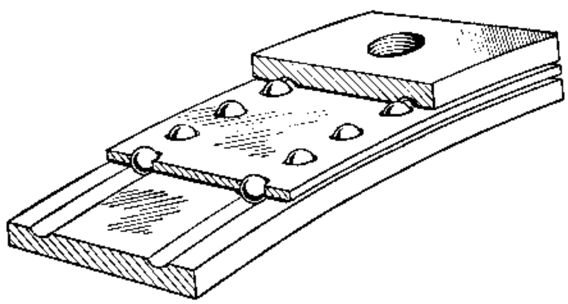
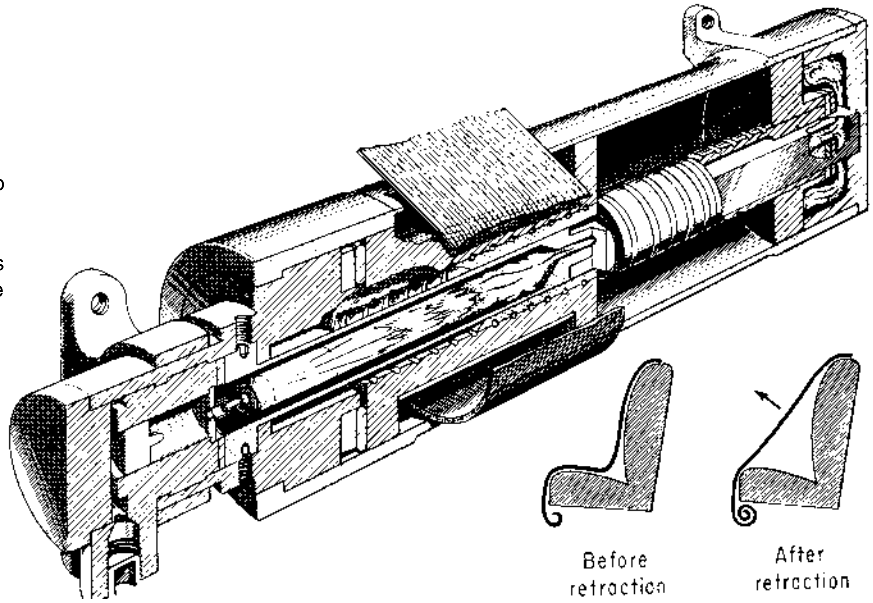
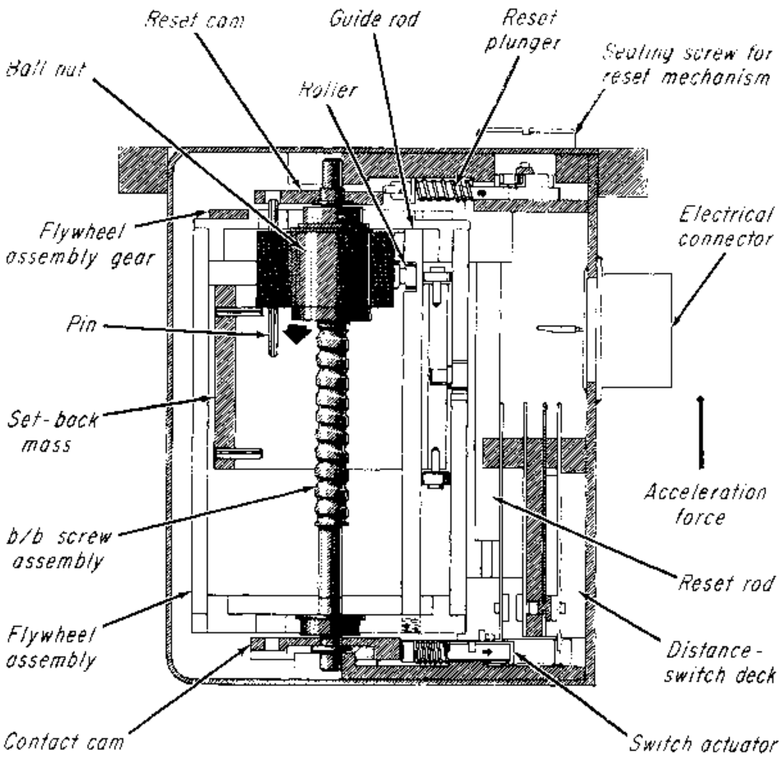


Fig. 9 Curvilinear motion in a plane is possible with this device when the radius of curvature is large. However, uniform spacing between its grooves is important. Circular-sectioned grooves decrease contact stresses.

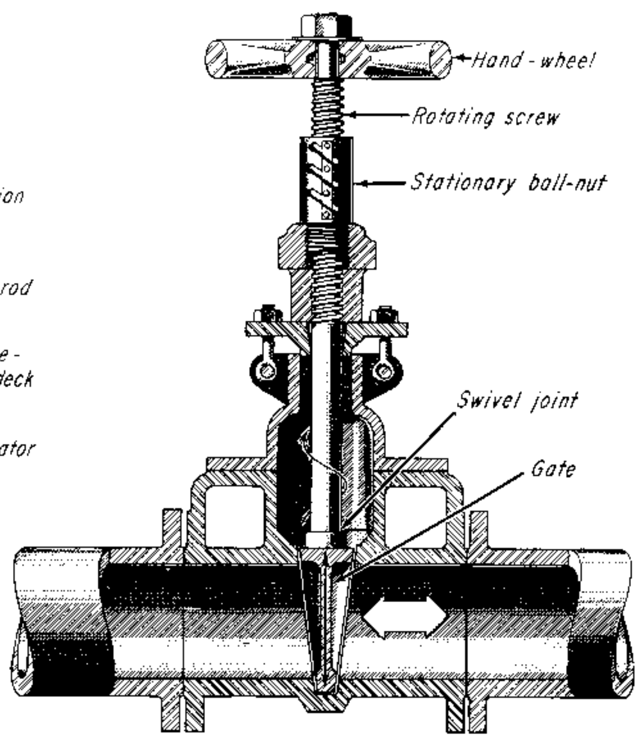
BALL-BEARING SCREWS CONVERT ROTARY TO LINEAR MOTION

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





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



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

THREE-POINT GEAR/LEADSCREW POSITIONING

The mechanism helps keep the driven plate parallel to a stationary plate.
Lewis Research Center, Cleveland, Ohio

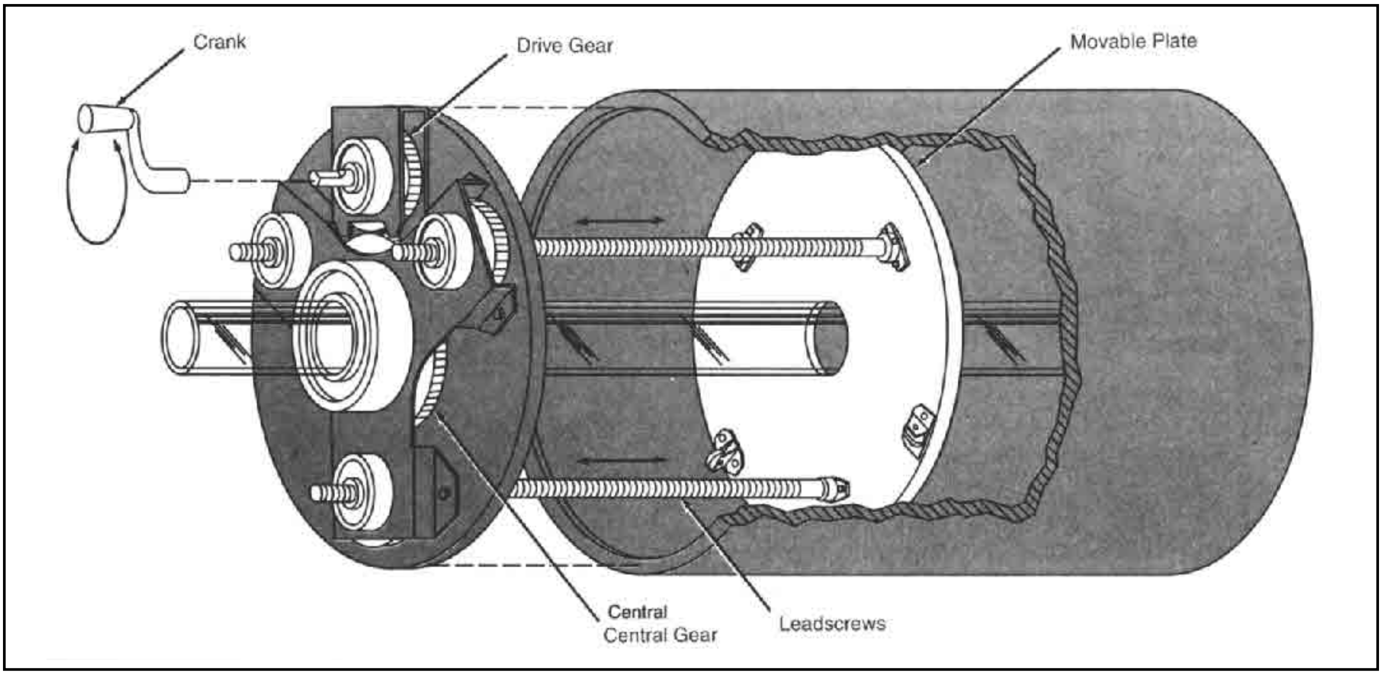
A triple-ganged-leadscrew positioning mechanism drives a movable plate toward or away from a fixed plate and keeps the plates parallel to each other. The mechanism was designed for use in tuning a microwave resonant cavity. The parallel plates are the end walls, and the distance between is the critical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

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

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

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

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



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

UNIQUE LINKAGE PRODUCES PRECISE STRAIGHT-LINE MOTION

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

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

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

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

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

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

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

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

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

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

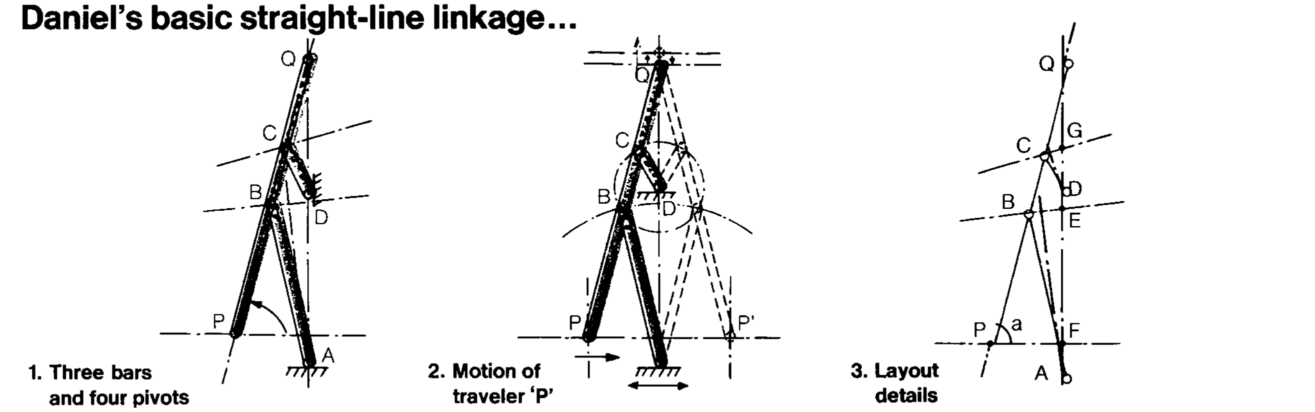
- Lay out any desired right triangle *PQF* (Fig. 3). Best results are with angle A approximately 75 to 80º.
- Pick point *B* on *PQ*. For greatest straight-line motion, *B* should be at or near the midpoint of *PQ*.
- Lay off length *PD* along *FQ* from *F* to find point *E*.
- Draw *BE* and its perpendicular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

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

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

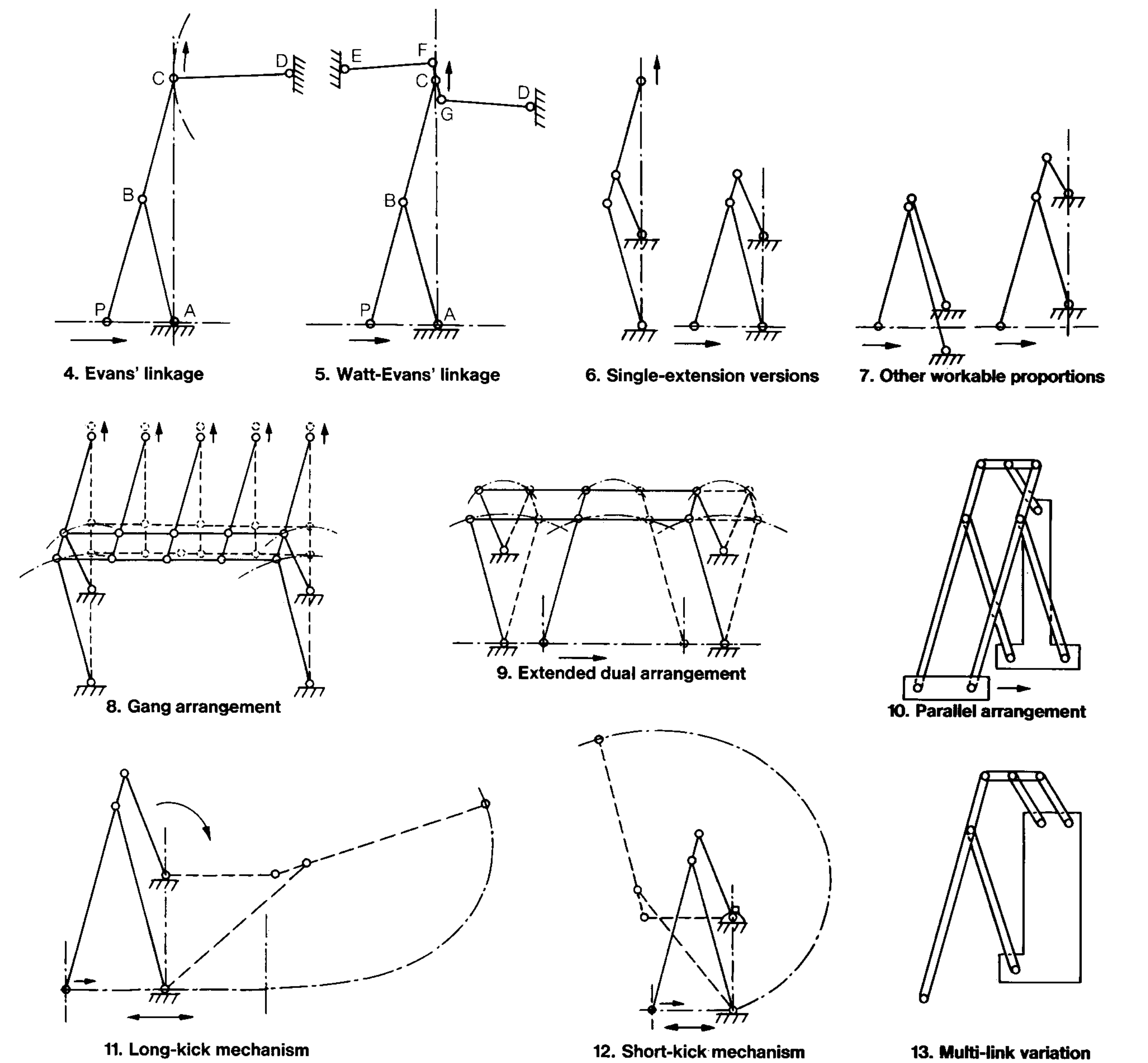
When the traveling point is allowed to clear the pivot support (Fig. 11), the ultimate path will curve upward to provide a handy "kick" action. A short kick is obtained by adding a stop (Fig. 12) to reverse the direction of the frame links while the long coupler continues its stroke. Daniel suggested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

Daniel's basic straight-line linkage...



1. Three bars and four pivots
2. Motion of traveler 'P'
3. Layout details

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



4. Evans' linkage
5. Watt-Evans' linkage
6. Single-extension versions
7. Other workable proportions
8. Gang arrangement
9. Extended dual arrangement
10. Parallel arrangement
11. Long-kick mechanism
12. Short-kick mechanism
13. Multi-link variation

TWELVE EXPANDING AND CONTRACTING DEVICES

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

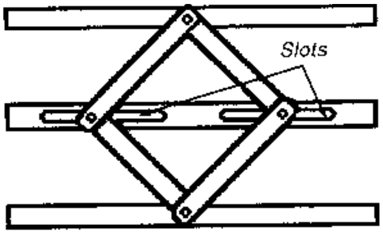


Fig. 1

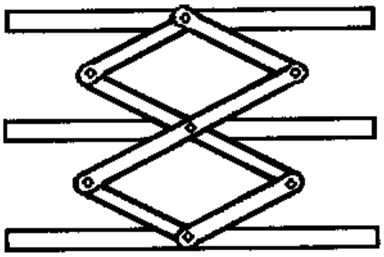


Fig. 2

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

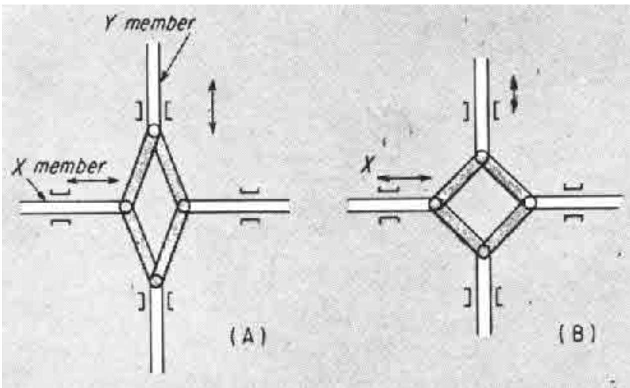
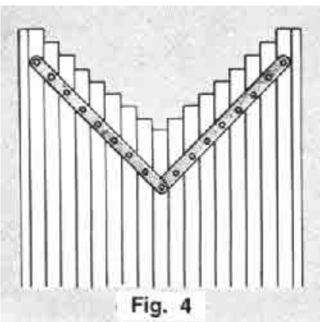
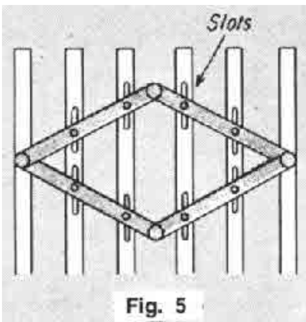
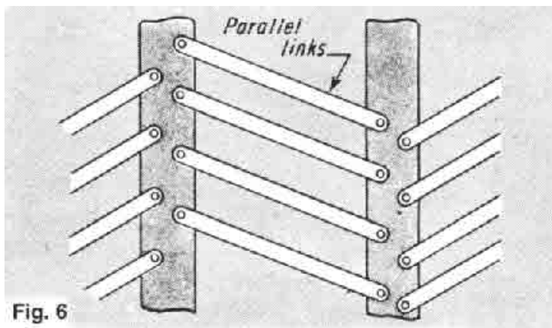


Fig. 3 Variable motion can be produced with this arrangement. In (A) position, the *Y* member is moving faster than the *X* member. In (B), speeds of both members are instantaneously equal. If the motion is continued in the same direction, the speed of *X* will become greater.







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

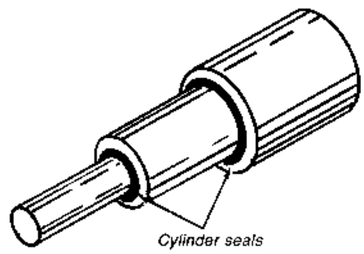


Fig. 7 Telescoping cylinders are the basis for many expanding and contracting mechanisms. In the arrangement shown, nested tubes can be sealed and filled with a highly temperature-responsive medium such as a volatile liquid.

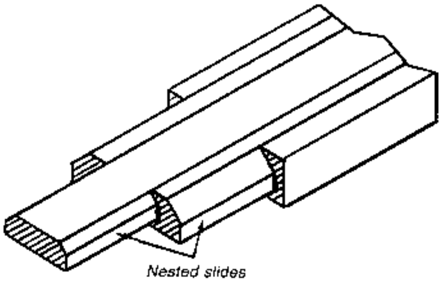


Fig. 8 Nested slides can provide an extension for a machine-tool table or other structure where accurate construction is necessary. In this design, adjustments to obtain smooth sliding must be made first before the table surface is leveled.

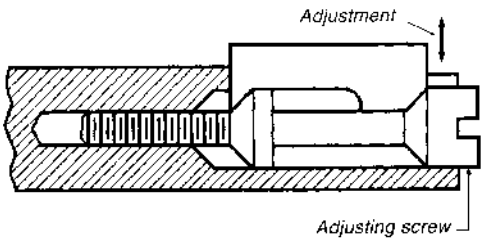


Fig. 9 Circular expanding mandrels are well-known. The example shown here is a less common mandrel-type adjustment. A parallel member, adjusted by two tapered surfaces on the screw, can exert a powerful force if the taper is small.

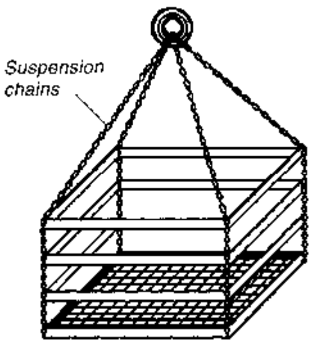


Fig. 10 This expanding basket is opened when suspension chains are lifted. Baskets take up little space when not in use. A typical use for these baskets is for conveyor systems. As tote baskets, they also allow easy removal of their contents because they collapse clear of the load.

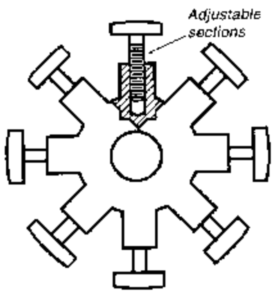


Fig. 11 An expanding wheel has various applications in addition to acting as a pulley or other conventional wheel. Examples include electrical contact on wheel surfaces that allow many repetitive electrical functions to be performed while the wheel turns. Dynamic and static balancing is simplified when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

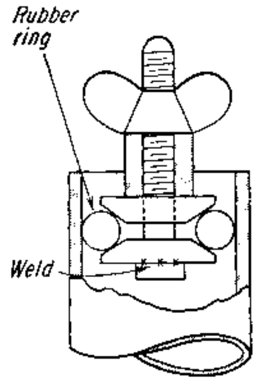


Fig. 12 A pipe stopper depends on a building rubber "O" ring for its action—soft rubber will allow greater conformity than hard rubber. It will also conform more easily to rough pipe surfaces. Hard rubber, however, withstands higher pressures. The screw head is welded to the washer for a leaktight joint.

FIVE LINKAGES FOR STRAIGHT-LINE MOTION

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

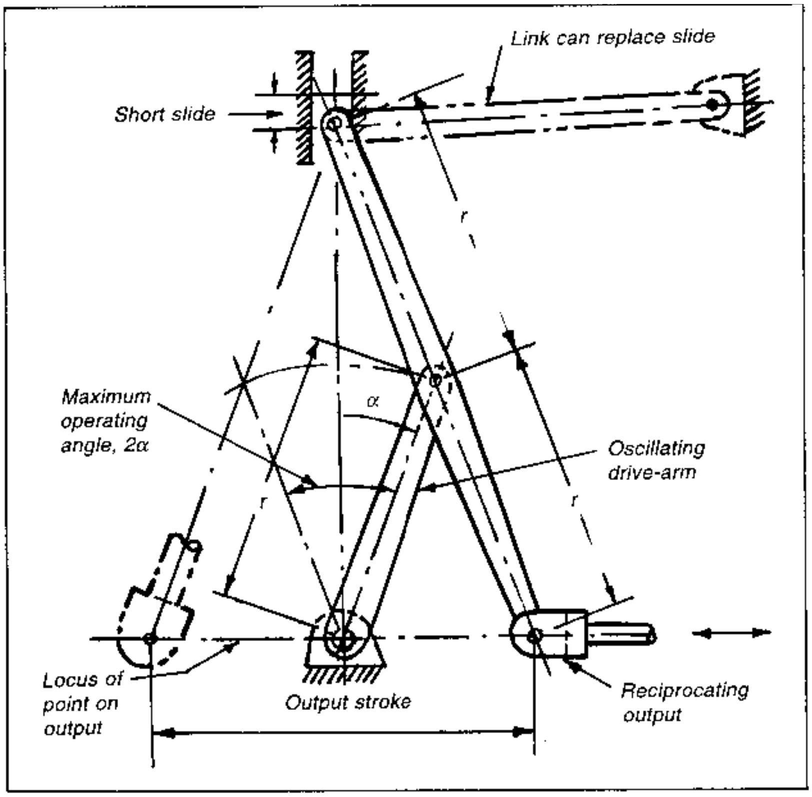


Fig. 1 An Evans' linkage has an oscillating drive-arm that should have a maximum operating angle of about 40º. For a relatively short guideway, the reciprocating output stroke is large. Output motion is on a true straight line in true harmonic motion. If an exact straight-line motion is not required,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

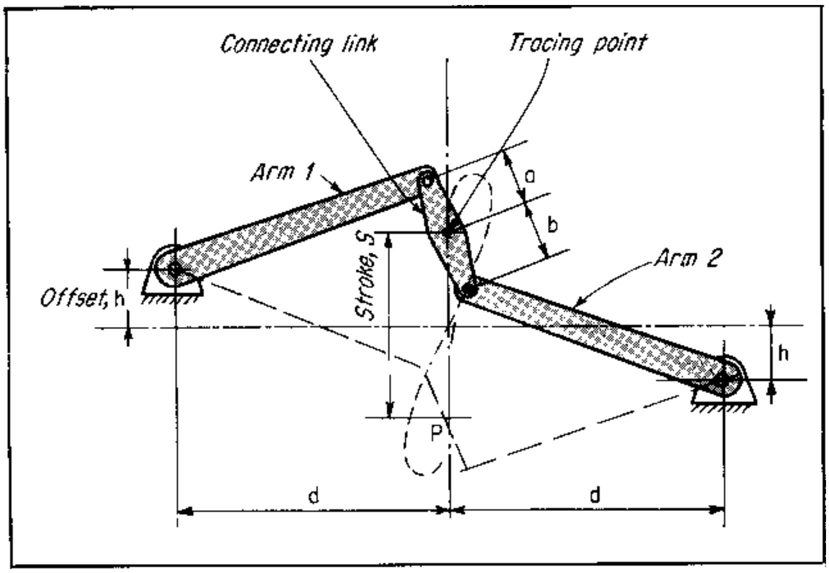


Fig. 2 A simplified Watt's linkage generates an approximate straight-line motion. If the two arms are of equal length, the tracing point describes a symmetrical figure 8 with an almost straight line throughout the stroke length. The straightest and longest stroke occurs when the connecting-link length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

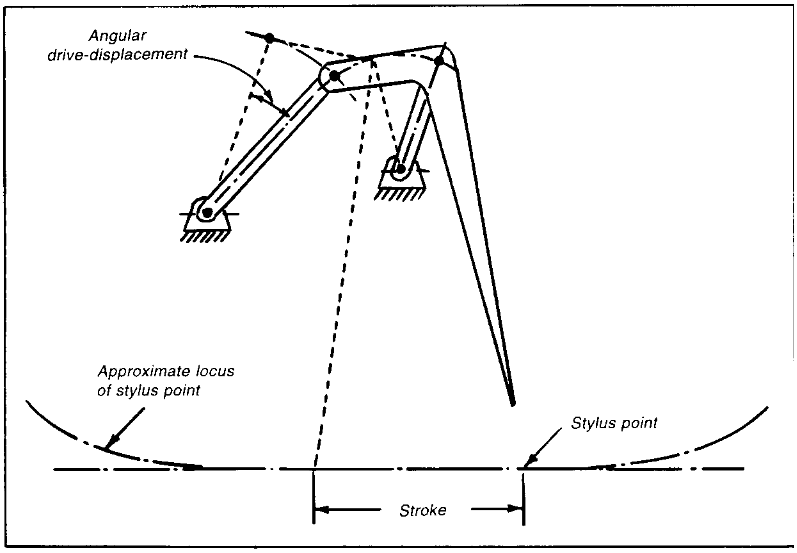


Fig. 3 Four-bar linkage produces an approximately straight-line motion. This arrangement provides motion for the stylus on self-registering measuring instruments. A comparatively small drive displacement results in a long, almost-straight line.

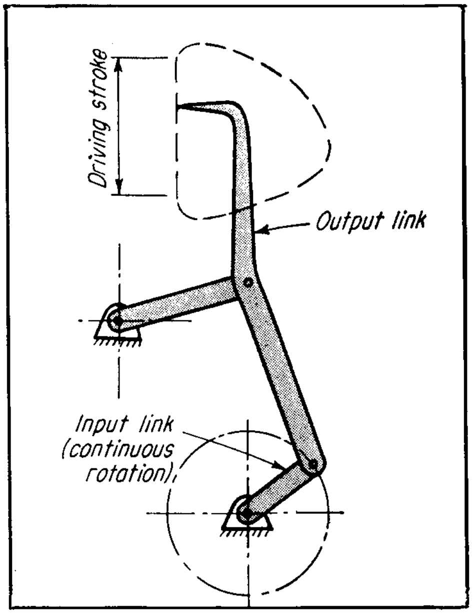


Fig. 4 A D-drive is the result when linkage arms are arranged as shown here. The output-link point describes a path that resembles the letter *D*, so there is a straight part of its cycle. This motion is ideal for quick engagement and disengagement before and after a straight driving stroke.

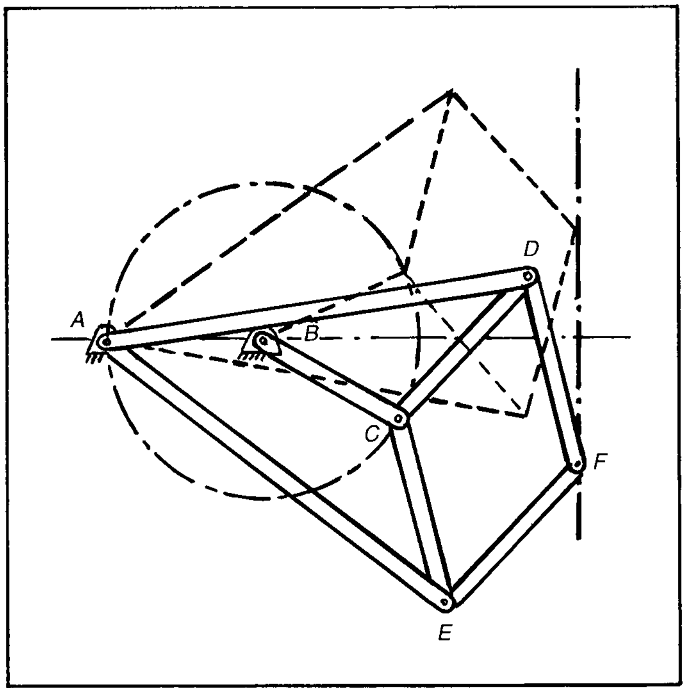


Fig. 5 The "Peaucellier cell" was the first solution to the classical problem of generating a straight line with a linkage. Within the physical limits of the motion, $AC \times AF$ remains constant. The curves described by C and F are, therefore, inverse; if C describes a circle that goes through A,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

LINKAGE RATIOS FOR STRAIGHT-LINE MECHANISMS

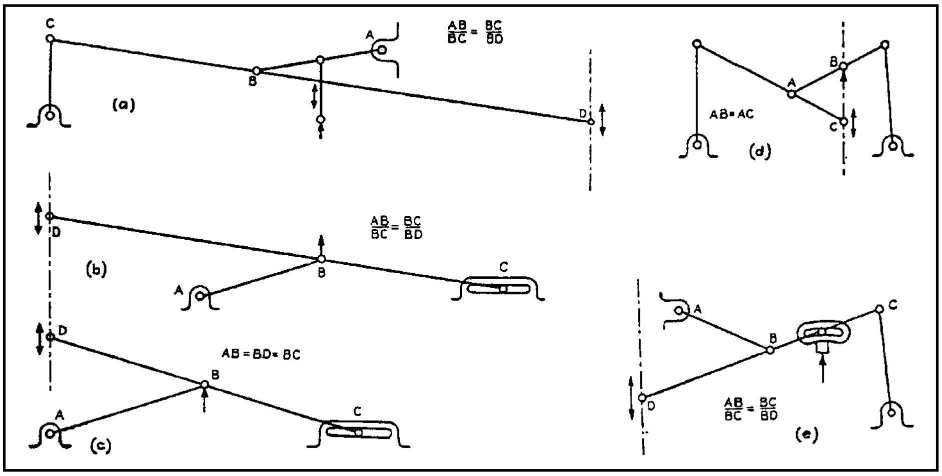


Fig. 1 (a), (b), (c), (d), (e)—Isoceles linkages.

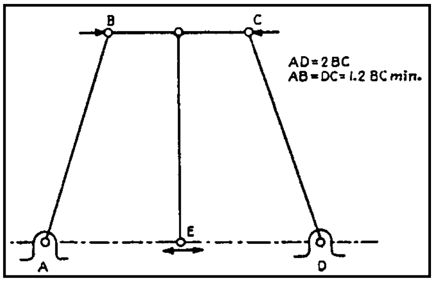


Fig. 2 Robert's linkage.

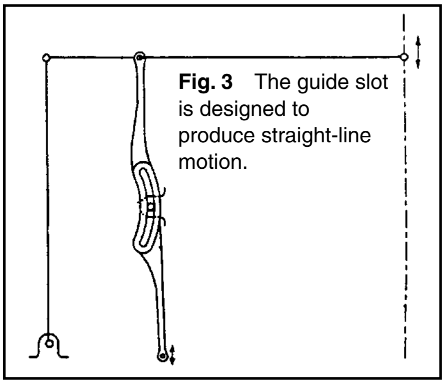


Fig. 3 The guide slot is designed to produce straight-line motion.

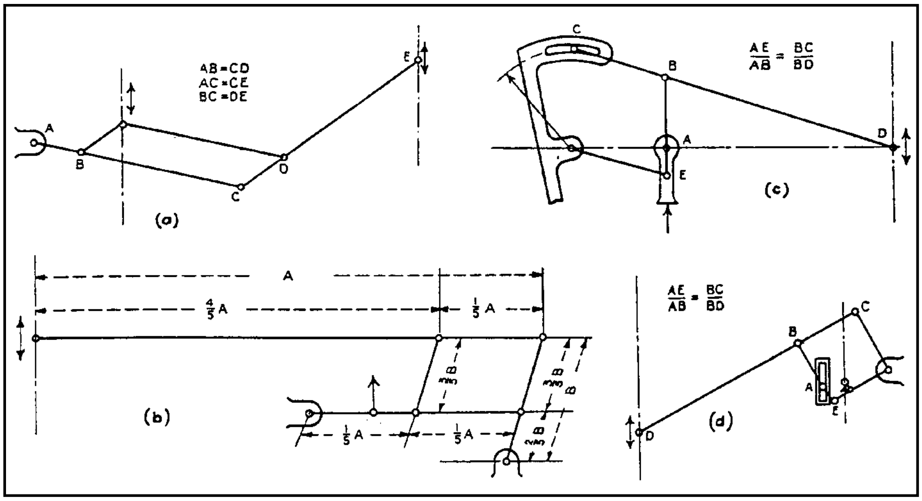


Fig. 4 (a), (b), (c), (d)—Pantograph linkages.

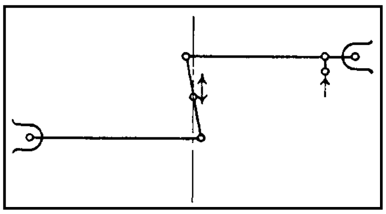


Fig. 5 Watt's linkage.

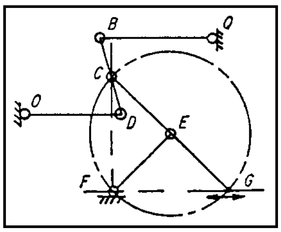


Fig. 6 Tchebicheff combination linkage.

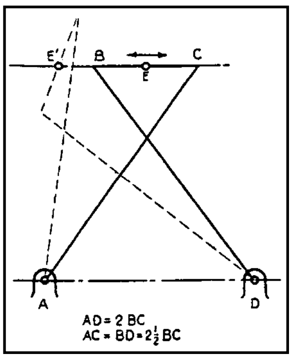


Fig. 7 Tchebicheff's linkage.

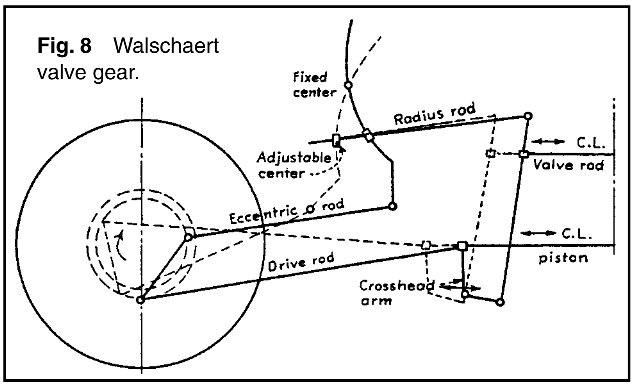


Fig. 8 Walschaert valve gear.

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

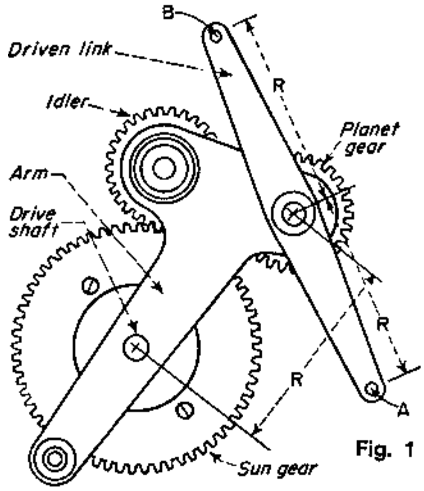


Fig. 1

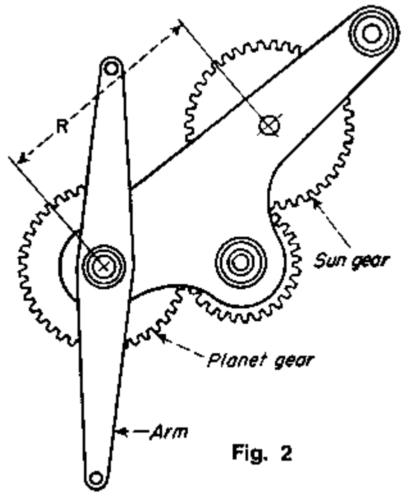


Fig. 2

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

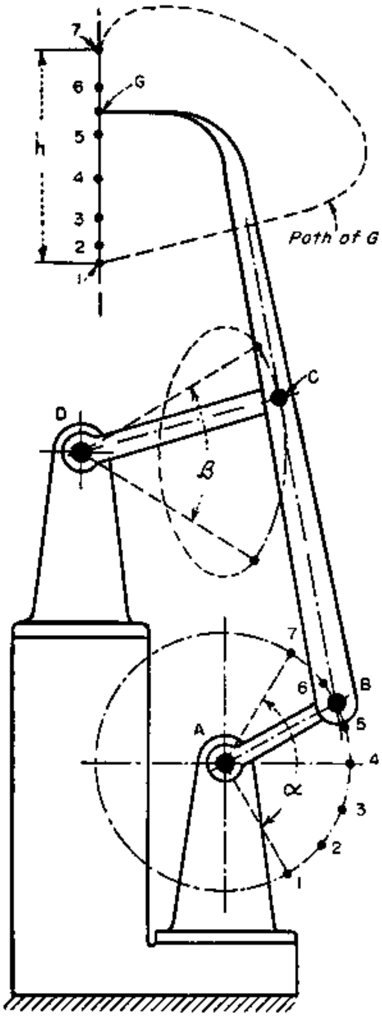


Fig. 3

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

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

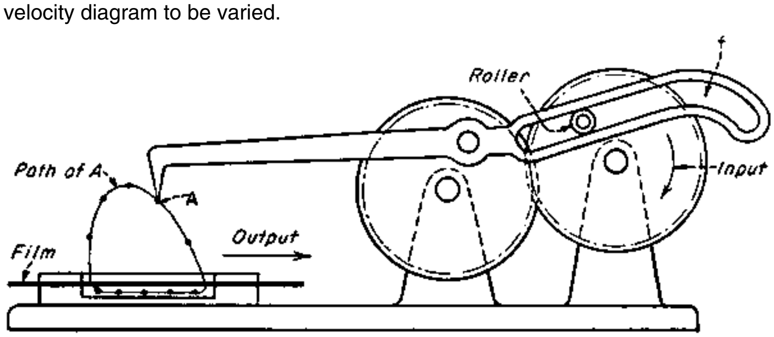


Fig. 4

FIVE CARDAN-GEAR MECHANISMS

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

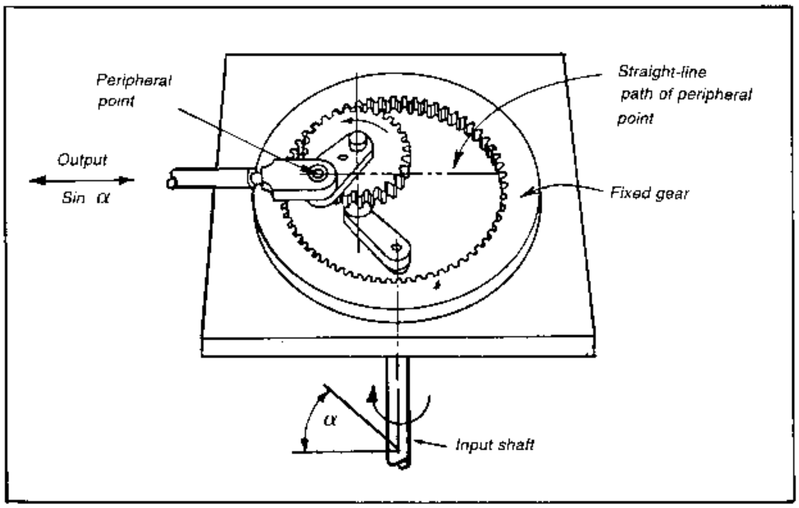


Fig. 1 Cardan gearing works on the principle that any point on the periphery of a circle rolling on the inside of another circle describes, in general, a hypocycloid. This curve degenerates into a true straight line (diameter of the larger circle) if the diameters of both circles are in the ratio of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

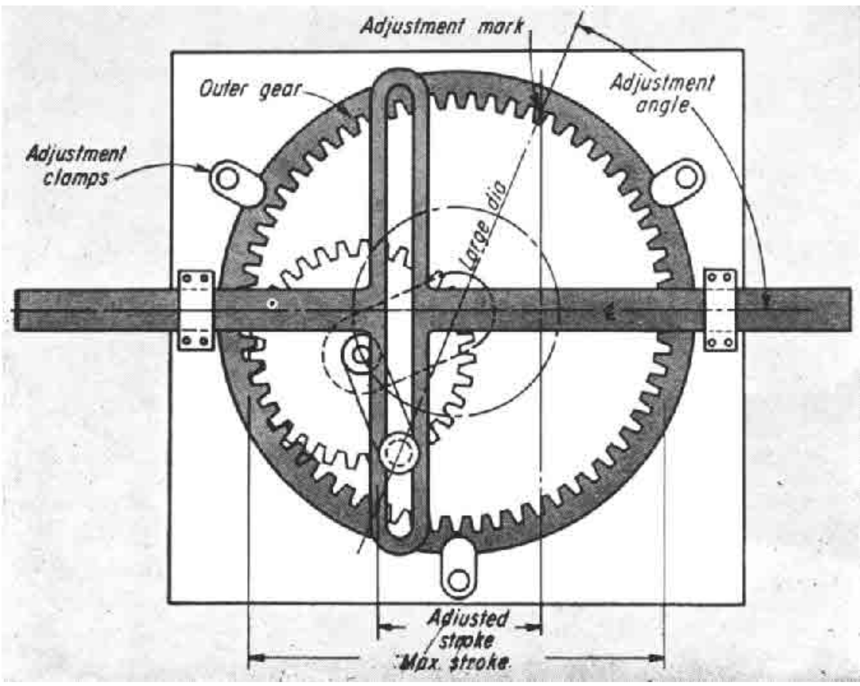


Fig. 2 Cardan gearing and a Scotch yoke in combination provide an adjustable stroke. The angular position of the outer gear is adjustable. The adjusted stroke equals the projection of the large diameter, along which the drive pin travels, on the Scotch-yoke's centerline. The yoke motion is simple harmonic.

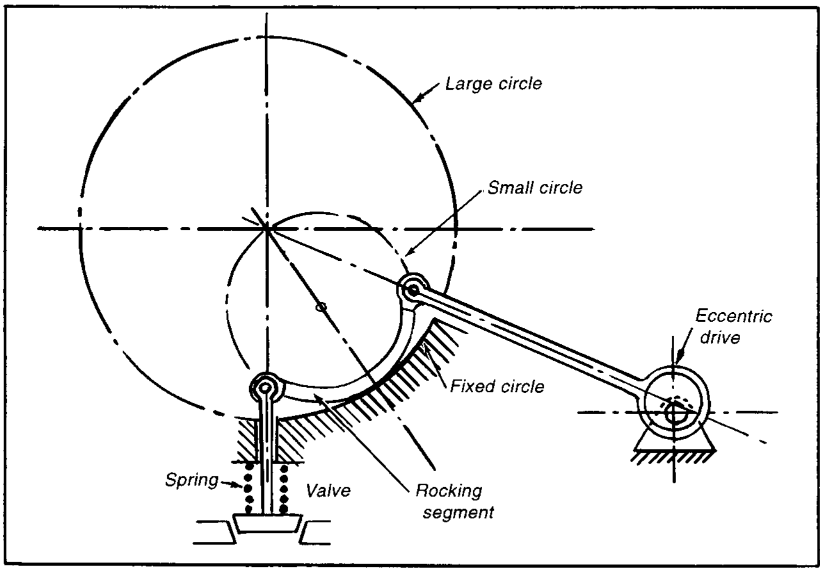


Fig. 3 A valve drive demonstrates how the Cardan principle can be applied. A segment of the smaller circle rocks back and forth on a circular segment whose radius is twice as large. The input and output rods are each attached to points on the small circle. Both these points describe straight lines. The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

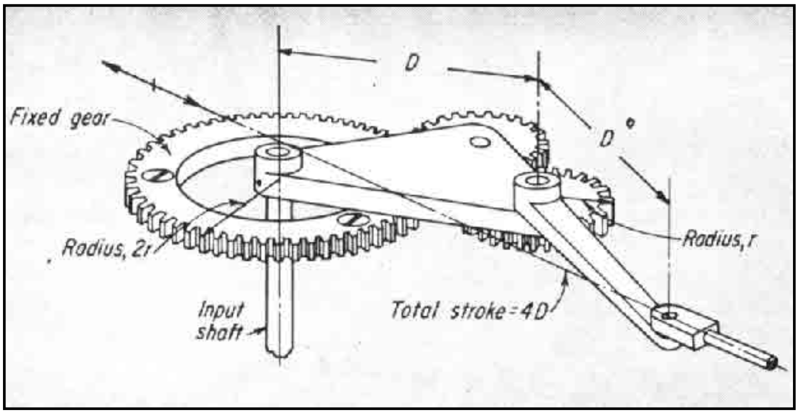


Fig. 4 A simplified Cardan mechanism eliminates the need for the relatively expensive internal gear. Here, only spur gears are used, and the basic requirements must be met, i.e., the 1:2 ratio and the proper direction of rotation. The rotation requirement is met by introducing an idler gear of appropriate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

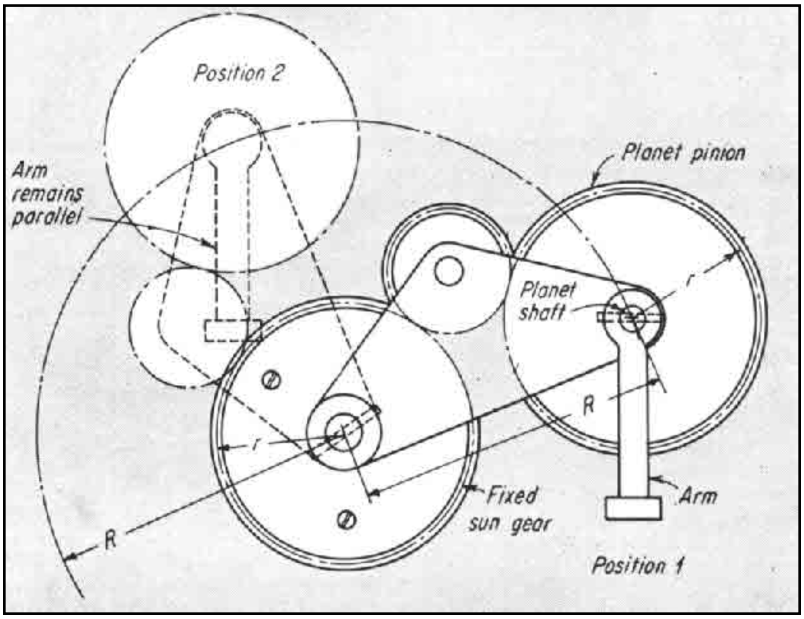


Fig. 5 A rearrangement of gearing in the simplified Cardan mechanism results in another useful motion. If the fixed sun gear and planet pinion are in the ratio of 1:1, an arm fixed to the planet shaft will stay parallel to itself during rotation, while any point on the arm describes a circle of radius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

TEN WAYS TO CHANGE STRAIGHT-LINE DIRECTION

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

LINKAGES

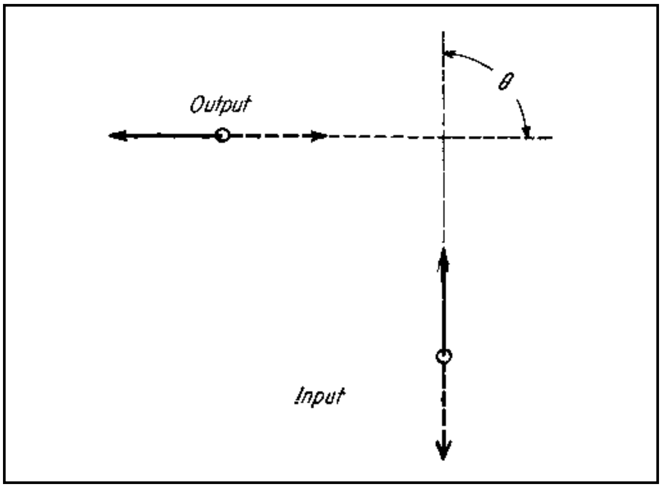


Fig. 1 Basic problem (θ is generally close to 90º).

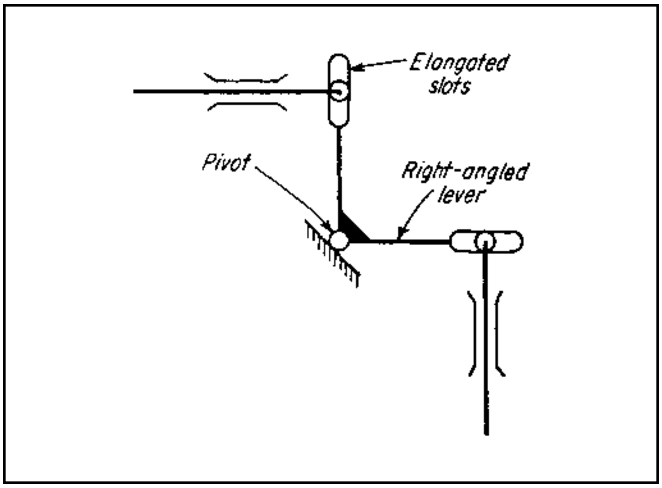


Fig. 2 Slotted lever.

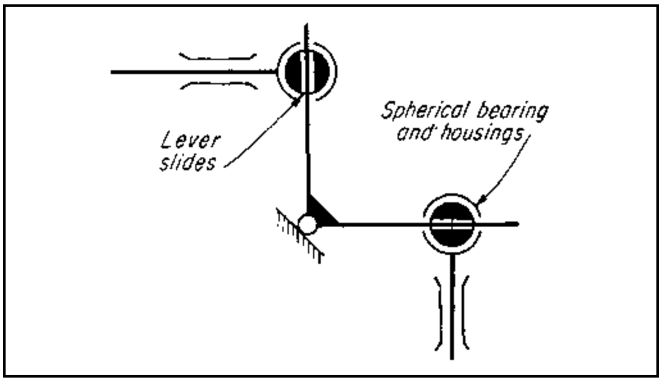


Fig. 3 Spherical bearings.

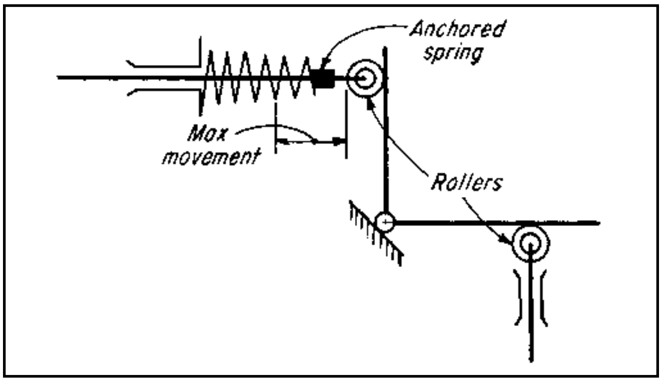


Fig. 4 Spring-loaded lever.

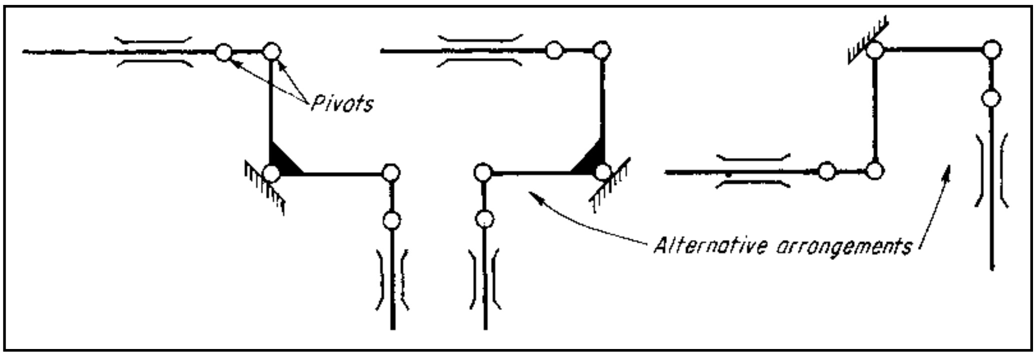


Fig. 5 Pivoted levers with alternative arrangements.

GUIDES

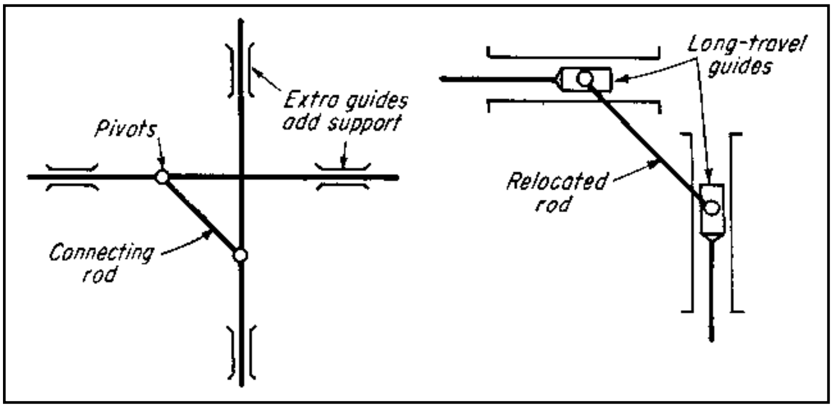


Fig. 6 Single connecting rod (left) is relocated (right) to eliminate the need for extra guides.

FRICTION DRIVES

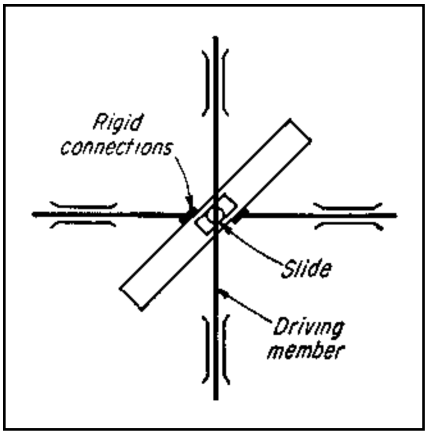


Fig. 7 Inclined bearing-guide.

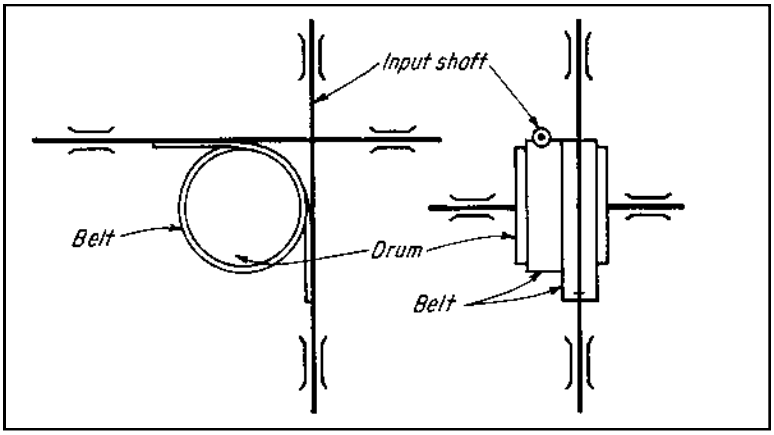


Fig. 8 A belt, steel band, or rope around the drum is fastened to the driving and driven members; sprocket-wheels and chain can replace the drum and belt.

GEARS

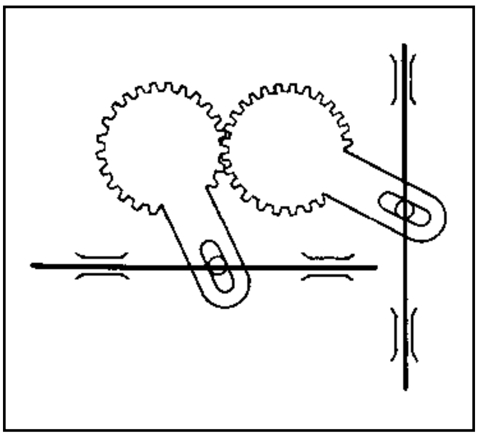


Fig. 9 Matching gear-segments.

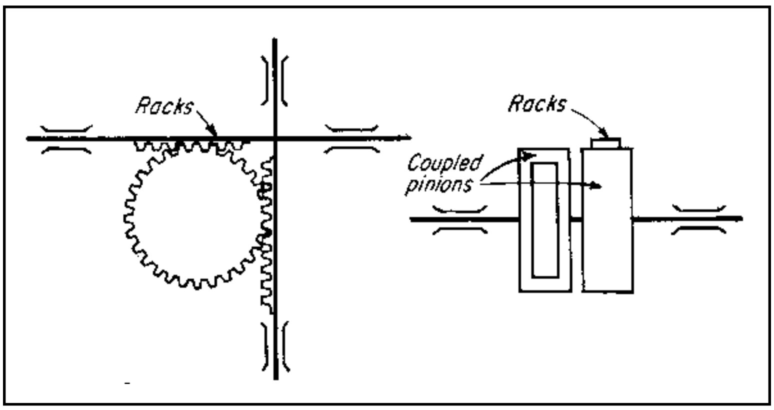


Fig. 10 Racks and coupled pinions (can be substituted as friction surfaces for a low-cost setup).

NINE MORE WAYS TO CHANGE STRAIGHT-LINE DIRECTION

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

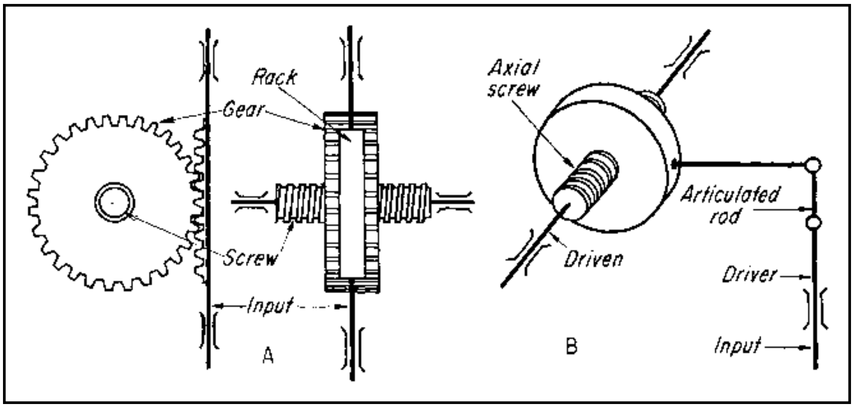


Fig. 1 An axial screw with a rack-actuated gear (A) and an articulated driving rod (B) are both irreversible movements, i.e., the driver must always drive.

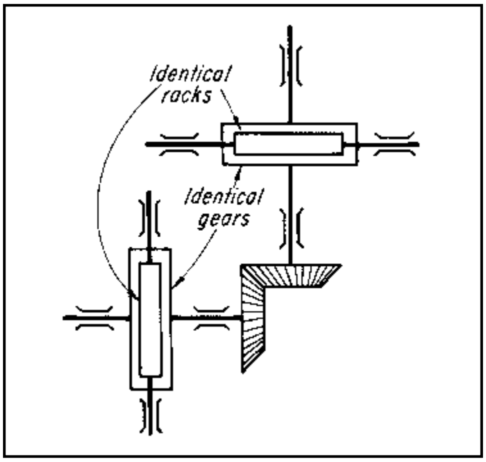


Fig. 2 A rack-actuated gear with associated bevel gears is reversible.

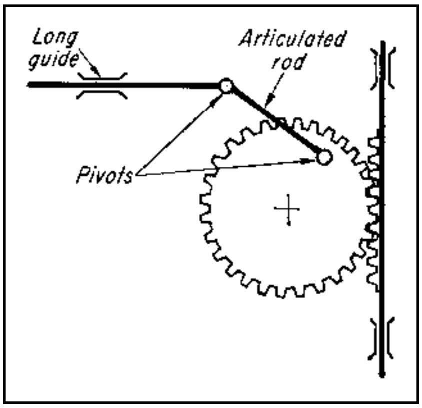


Fig. 3 An articulated rod on a crank-type gear with a rack driver. Its action is restricted to comparatively short movements.

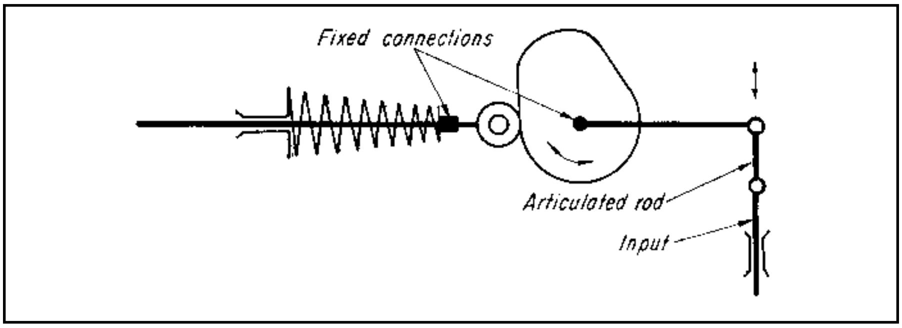


Fig. 4 A cam and spring-loaded follower allows an input/output ratio to be varied according to cam rise. The movement is usually irreversible.

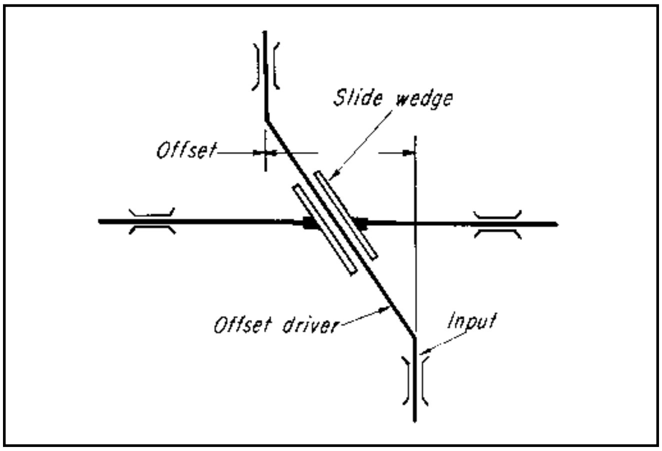


Fig. 5 An offset driver actuates a driven member by wedge action. Lubrication and materials with a low coefficient of friction permit the offset to be maximized.

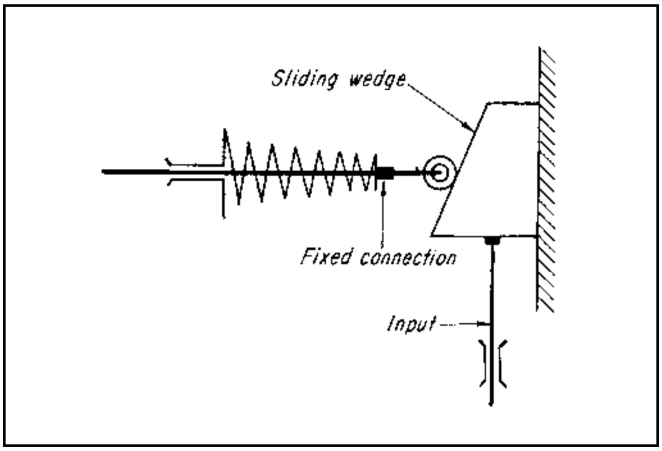


Fig. 6 A sliding wedge is similar to an offset driver but it requires a spring-loaded follower; also, low friction is less critical with a roller follower.

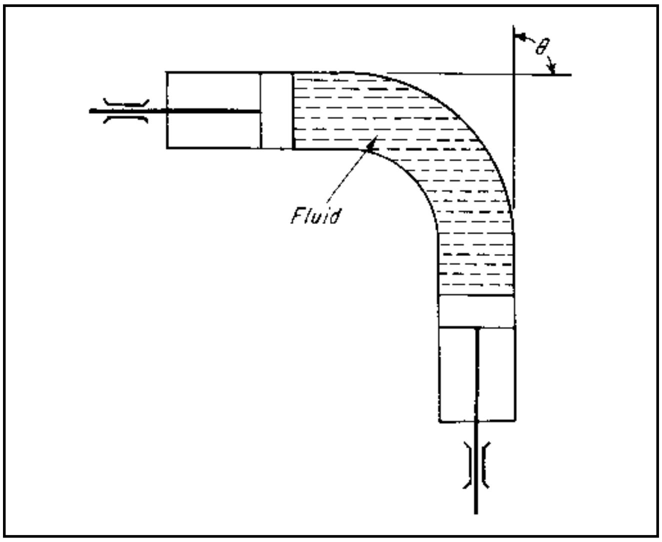


Fig. 7 A fluid coupling allows motion to be transmitted through any angle. Leak problems and accurate piston-fitting can make this method more expensive than it appears to be. Also, although the action is reversible, it must always be compressive for the best results.

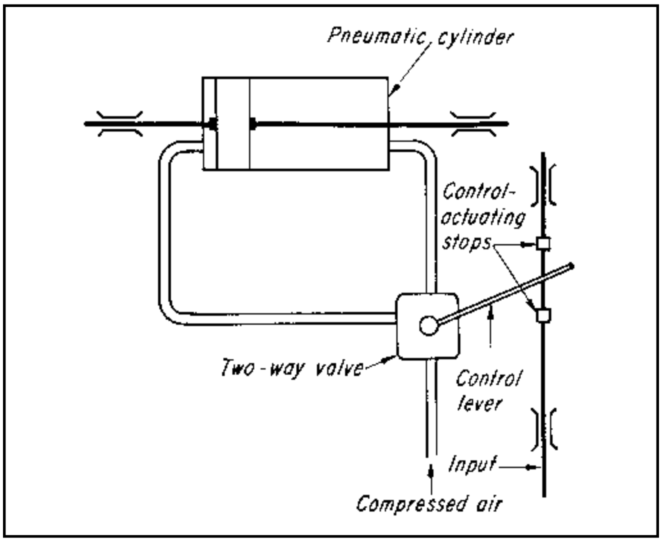


Fig. 8 A pneumatic system with a two-way valve is ideal when only two extreme positions are required. The action is irreversible. The speed of a driven member can be adjusted by controlling the input of air to the cylinder.

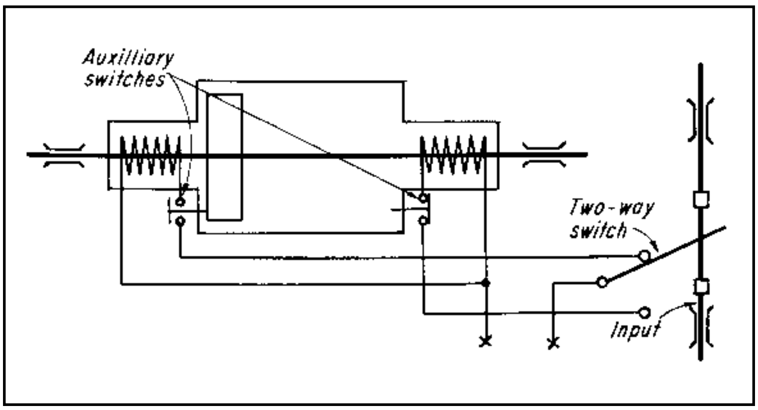


Fig. 9 Solenoids and a two-way switch are organized as an analogy of a pneumatic system. Contact with the energized solenoid is broken at the end of each stroke. The action is irreversible.

LINKAGES FOR ACCELERATING AND DECELERATING LINEAR STROKES

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

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

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

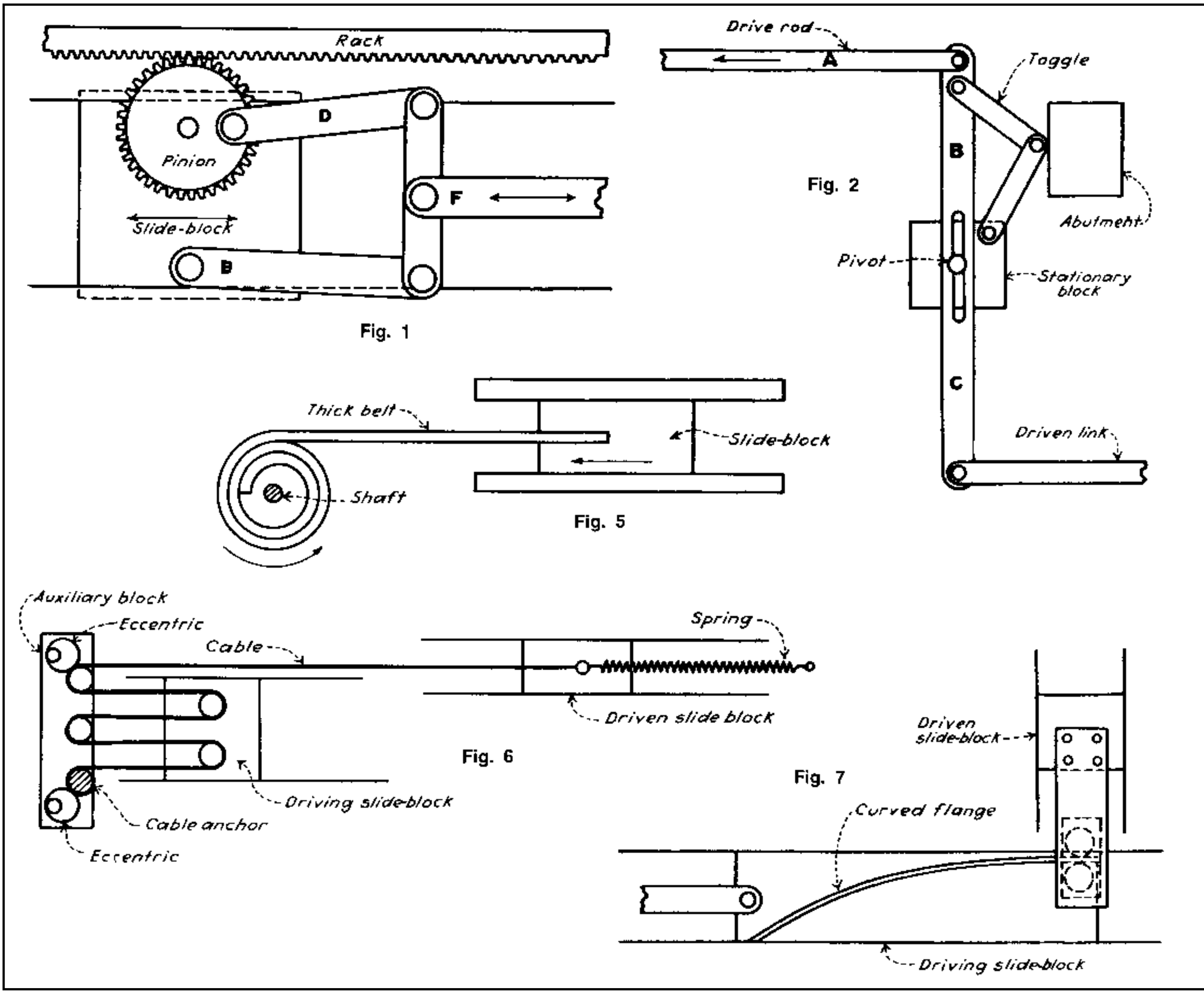


Fig. 3 **The same direction** of travel for both the drive rod and the drive link is provided by the variation of the Fig. 2 mechanism. Here, acceleration is in the direction of the arrows, and deceleration occurs on the return stroke. The effect of acceleration decreases as the toggle flattens.

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

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

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

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

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

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

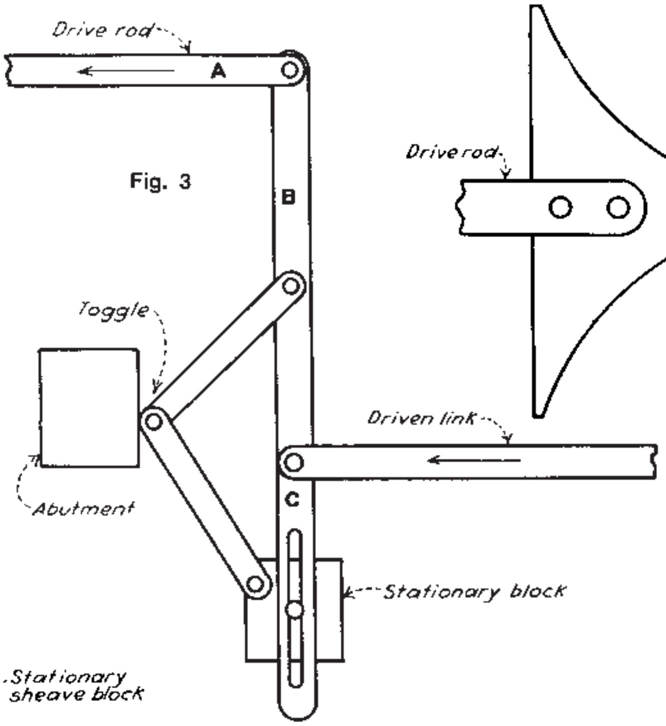


Fig. 3

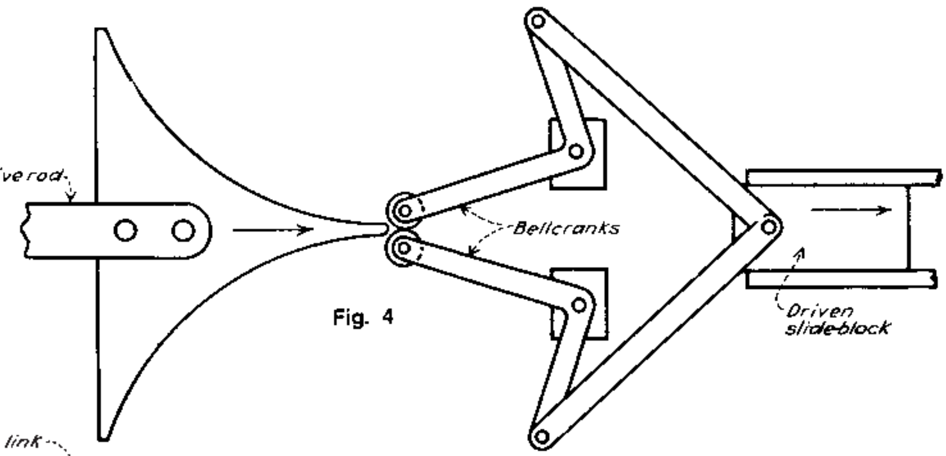


Fig. 4

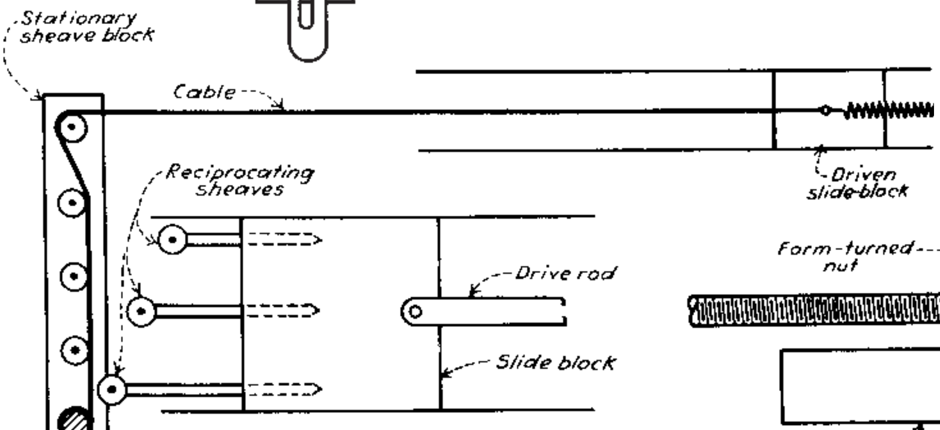


Fig. 8

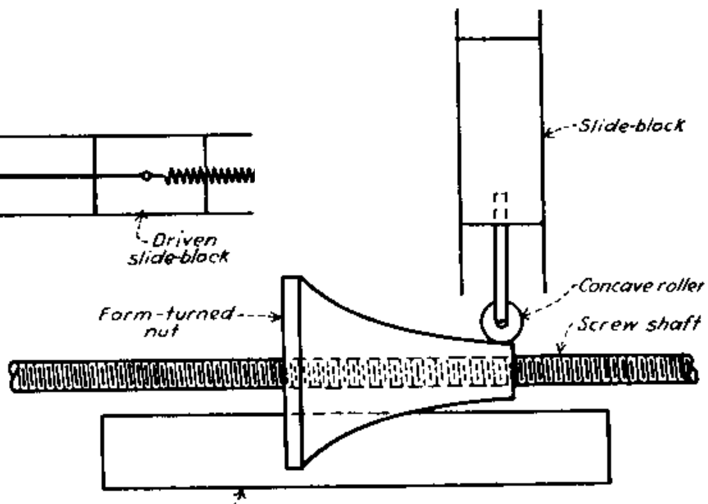


Fig. 9

LINKAGES FOR MULTIPLYING SHORT MOTIONS

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

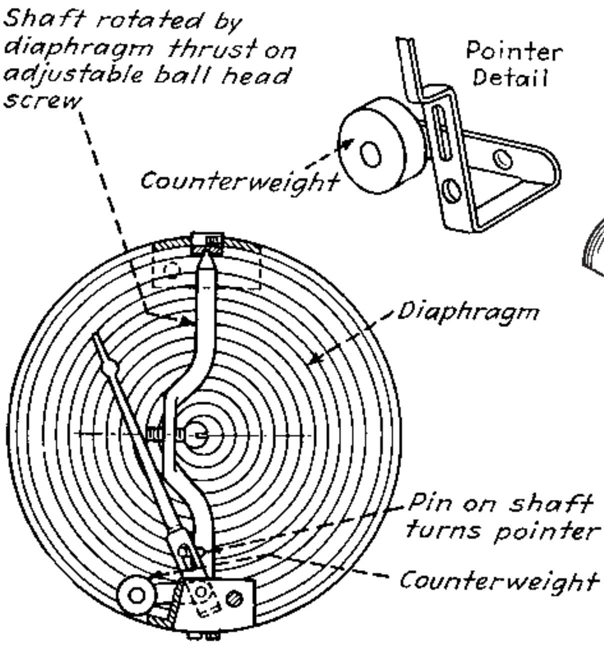


Fig. 1 A lever-type transmission in a pressure gage.

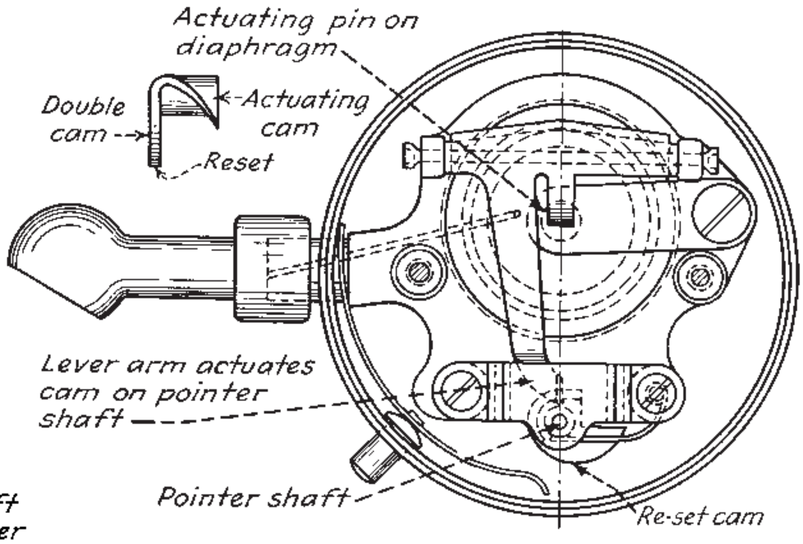


Fig. 2 A lever and cam drive for a tire gage.

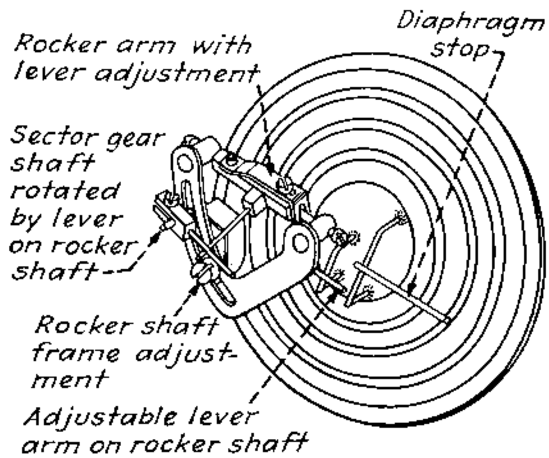


Fig. 4 A sector gear drive for an aircraft air-speed indicator.

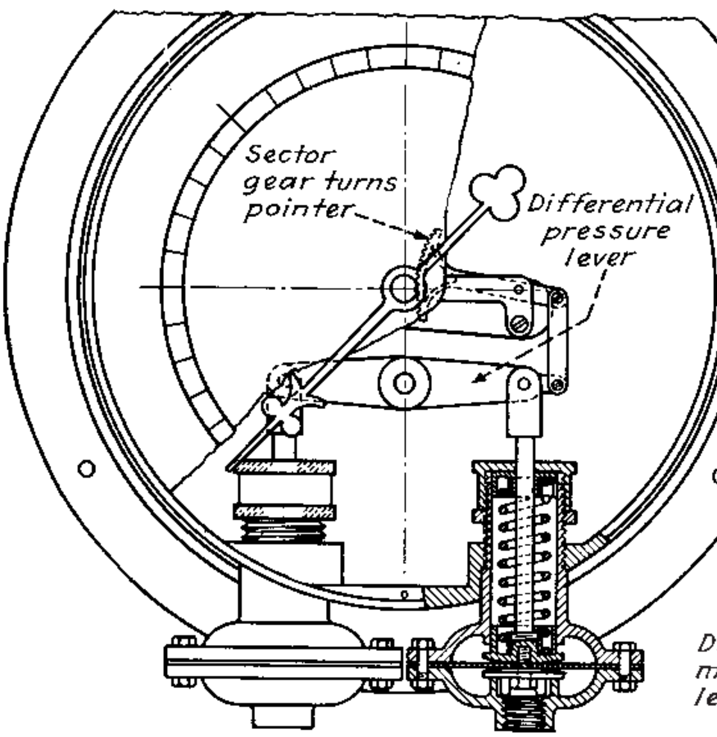


Fig. 3 A lever and sector gear in a differential pressure gage.

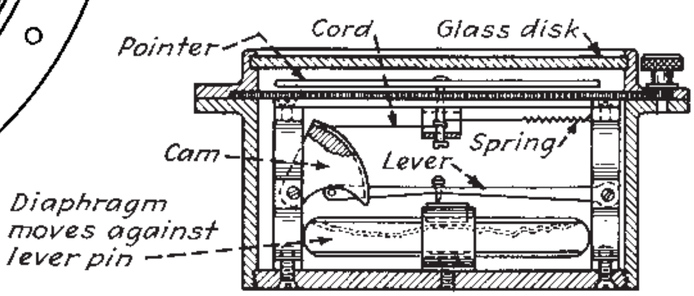


Fig. 5 A lever, cam, and cord transmission in a barometer.

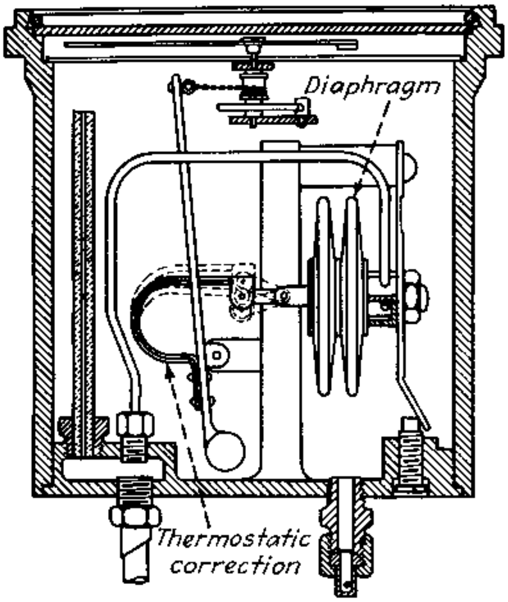


Fig. 6 A link and chain transmission for an aircraft rate of climb instrument.

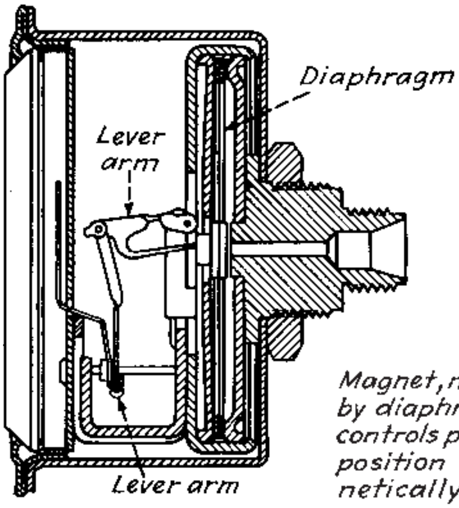


Fig. 7 A lever system in an automobile gasoline tank.

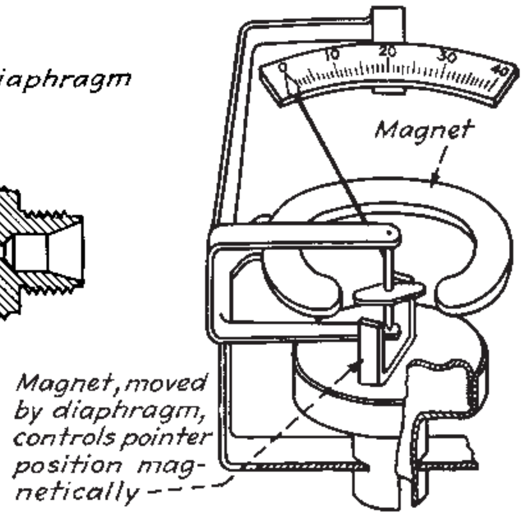


Fig. 8 Interfering magnetic fields for fluid pressure measurement.

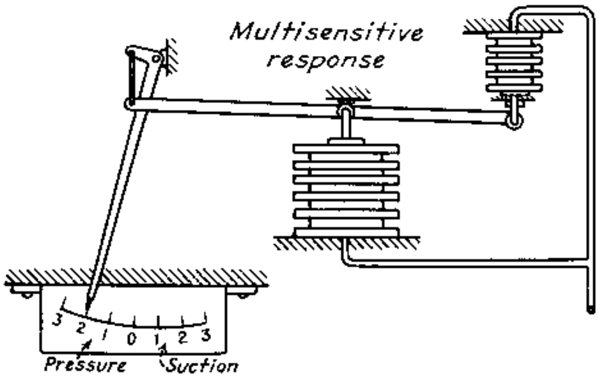


Fig. 9 A lever system for measuring atmospheric pressure variations.

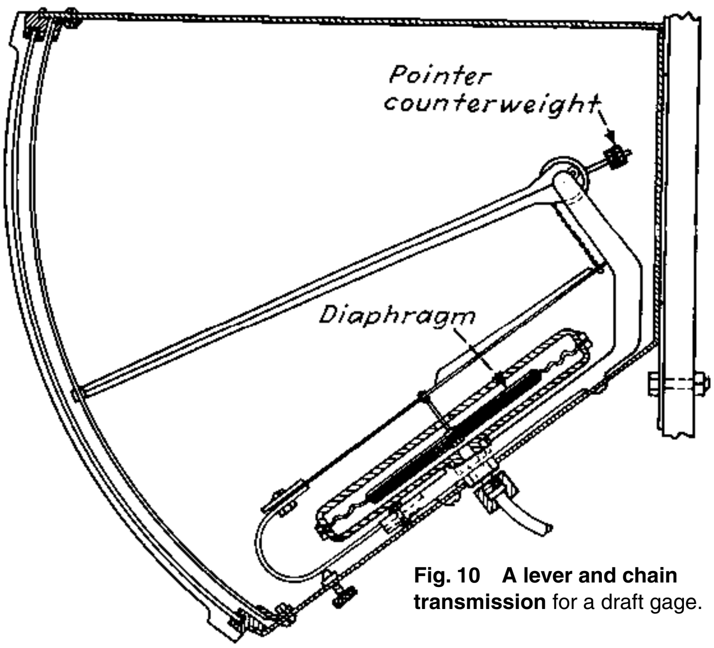


Fig. 10 A lever and chain transmission for a draft gage.

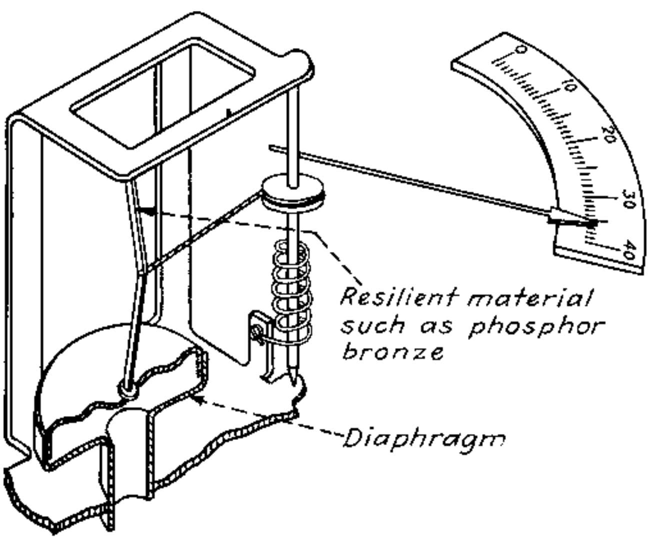


Fig. 11 A toggle and cord drive for a fluid pressure measuring instrument.

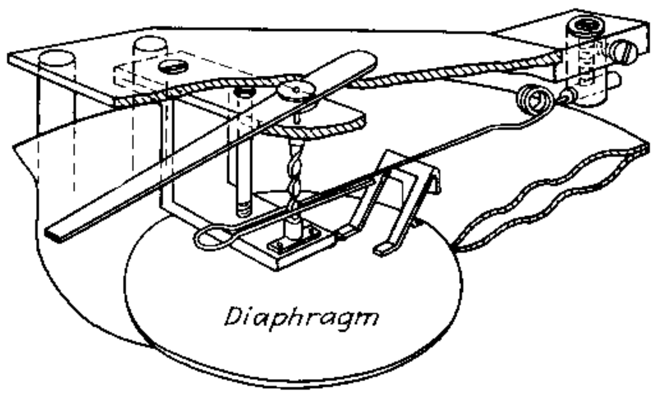
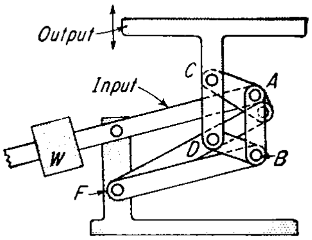


Fig. 12 A spiral feed transmission for a general purpose analog instrument.

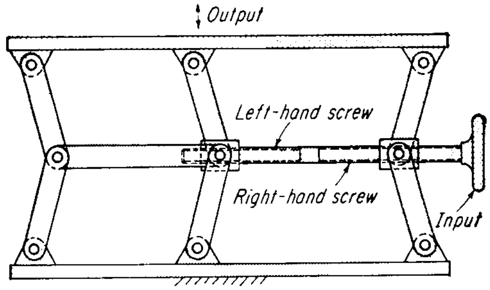
PARALLEL-LINK MECHANISMS

Eight-bar linkage



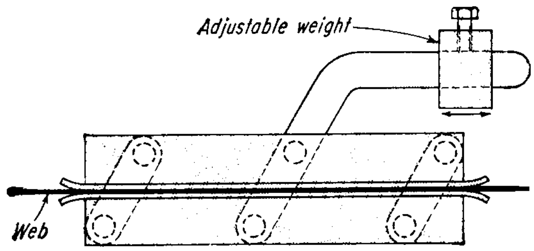
Link *AB* in this arrangement will always be parallel to *EF*, and link *CD* will always be parallel to *AB*. Hence *CD* will always be parallel to *EF*. Also, the linkages are so proportioned that point *C* moves in an approximately straight line. The final result is that the output plate will remain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

Double-handed screw mechanism



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

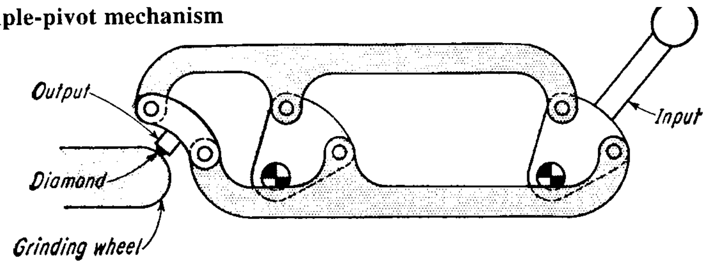
Tensioning mechanism



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

Triple-pivot mechanism

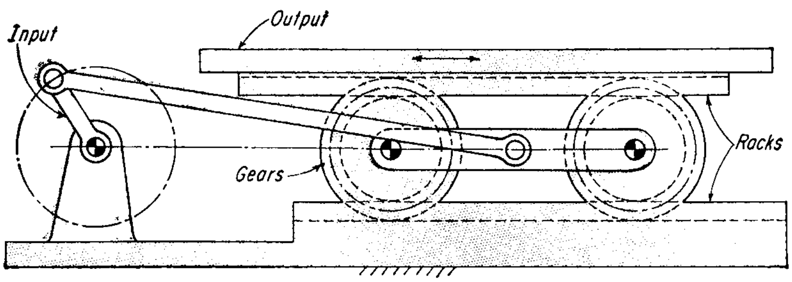
Two triangular plates pivot around fixed points on a machine frame. The output point describes a circular-arc curve. It can round out the cutting surfaces of grinding wheels.



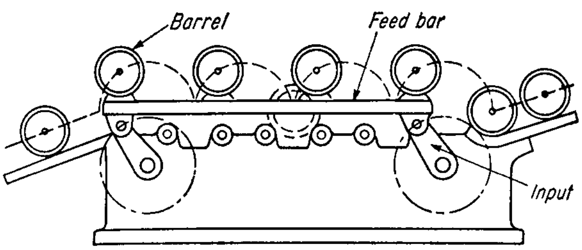
STROKE MULTIPLIER

Reciprocating-table drive

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

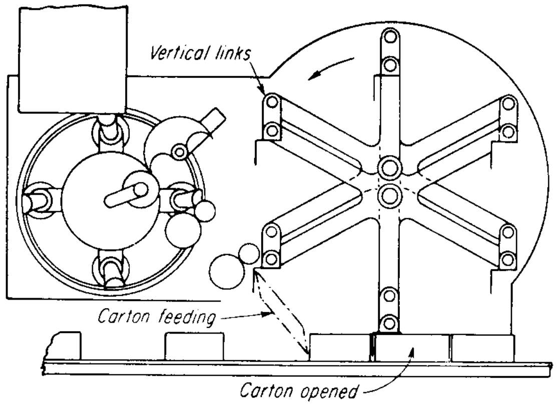


Parallel-link feeder



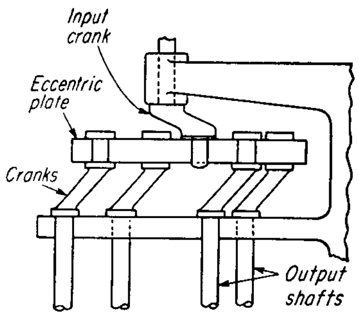
One of the cranks is the input, and the other follows to keep the feeding bar horizontal. The feeder can move barrels from station to station.

Parallelogram linkage



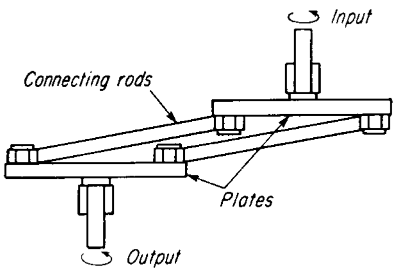
All seven short links are kept in a vertical position while rotating. The center link is the driver. This particular machine feeds and opens cartons, but the mechanism will work in many other applications.

Parallel-link driller



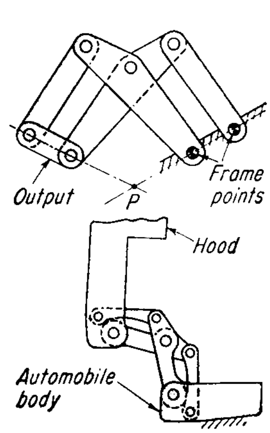
This parallel-link driller powers a group of shafts. The input crank drives the eccentric plate. This, in turn, rotates the output cranks that have the same length at the same speed. Gears would occupy more room between the shafts.

Parallel-plate driver



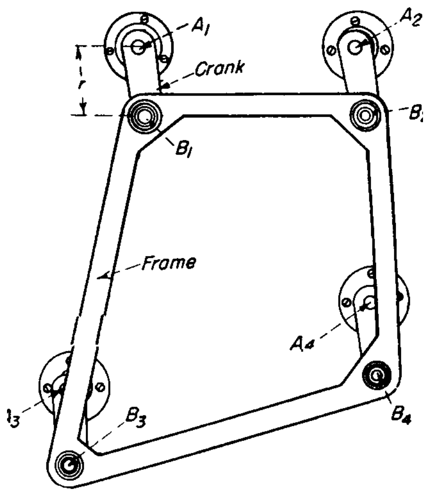
The input and output shafts of this parallel-plate driver rotate with the same angular relationship. The positions of the shafts, however, can vary to suit other requirements without affecting the input-output relationship between the shafts.

Curve-scribing mechanism



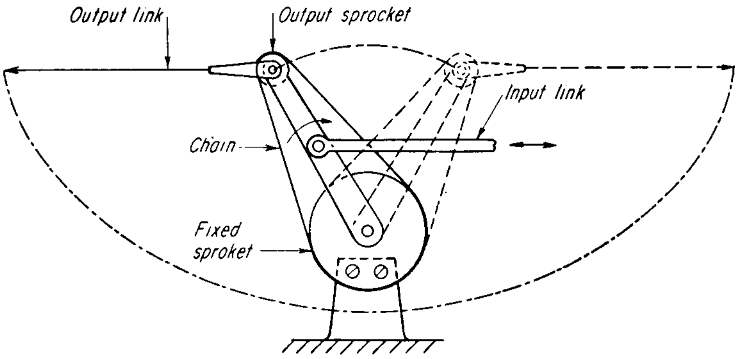
The output link rotates so that it appears to revolve around a point moving in space (P). This avoids the need for hinges at distant or inaccessible spots. The mechanism is suitable for hinging the hoods of automobiles.

Parallel-link coupling



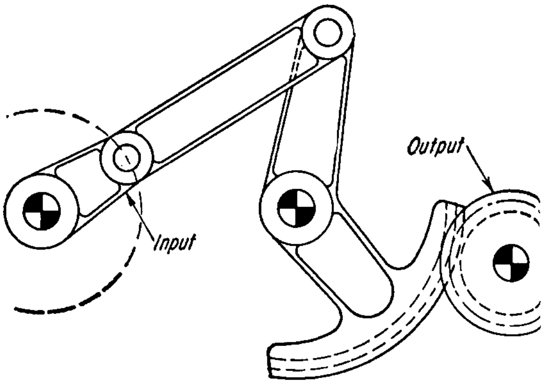
The absence of backlash makes this parallel-link coupling a precision, low-cost replacement for gear or chain drives that can also rotate parallel shafts. Any number of shafts greater than two can be driven from any one of the shafts, provided two conditions are fulfilled: (1) All cranks must have the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

Wide-angle oscillator



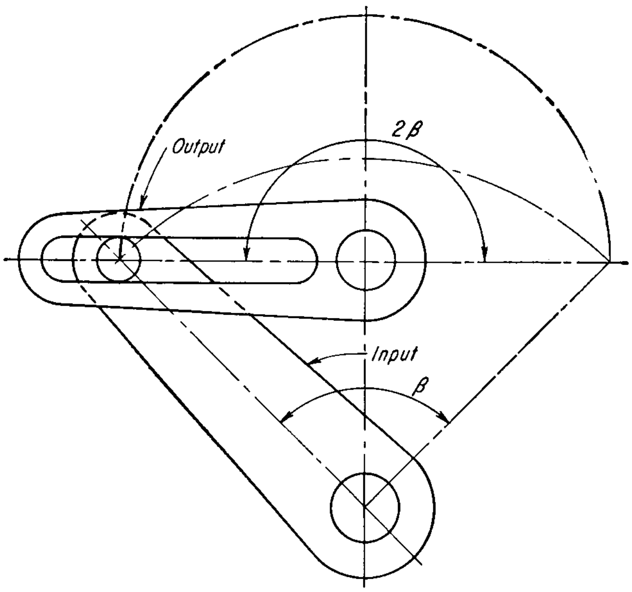
The motion of the input linkage in the diagram is converted into a wide-angle oscillation by the two sprockets and chain. An oscillation of 60º is converted into 180º oscillation.

Gear-sector drive



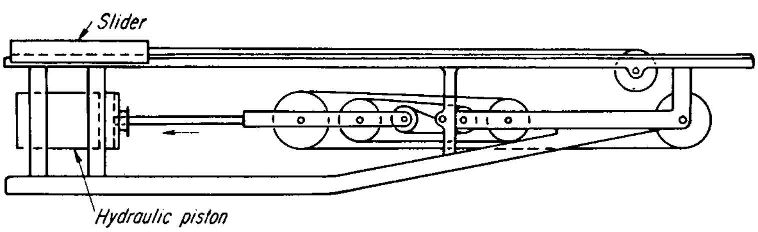
This is actually a four-bar linkage combined with a set of gears. A four-bar linkage usually obtains so more than about 120º of maximum oscillation. The gear segments multiply the oscillation in inverse proportion to the radii of the gears. For the proportions shown, the oscillation is boosted two and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

Angle-doubling drive



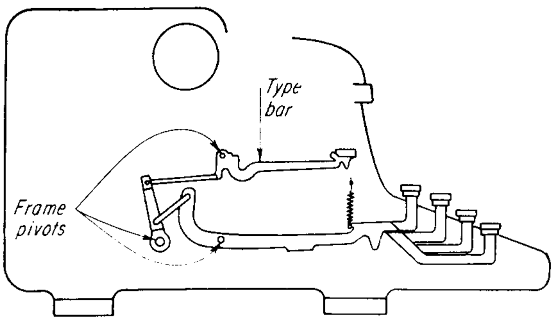
This angle-doubling drive will enlarge the oscillating motion β of one machine member into an output oscillation of 2β. If gears are employed, the direction of rotation cannot be the same unless an idler gear is installed. In that case, the centers of the input and output shafts cannot be too close.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

Pulley drive



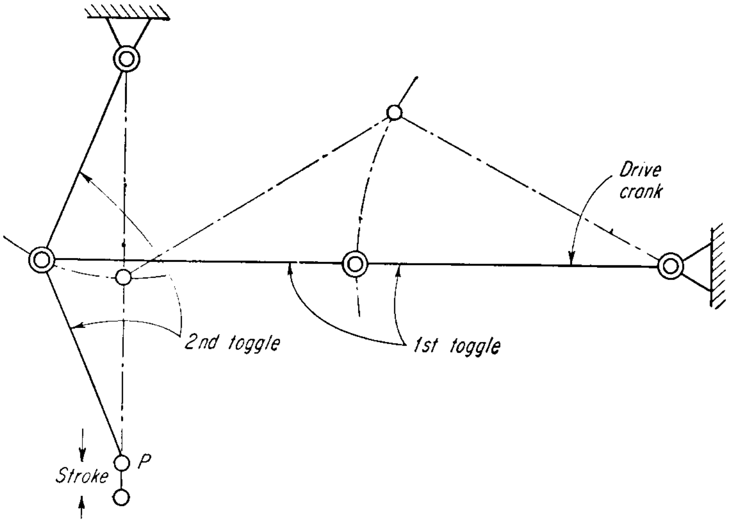
This pulley drive multiplies the stroke of a hydraulic piston, causing the slider to move rapidly to the right for catapulting objects.

Typewriter drive



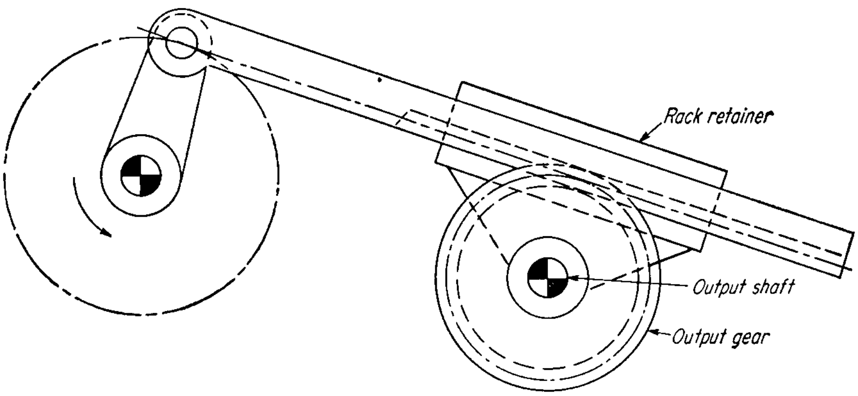
This drive multiplies the finger force of a typewriter, producing a strong hammer action at the roller from a light touch. There are three pivot points attached to the frame. The links are arranged so that the type bar can move in free flight after a key has been struck. The mechanism illustrated is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

Double-toggle puncher



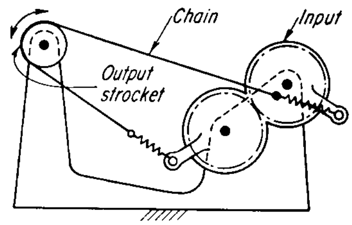
The first toggle of this puncher keeps point *P* in the raised position although its weight can exert a strong downward force (as in a heavy punch weight). When the drive crank rotates clockwise (e.g., driven by a reciprocating mechanism), the second toggle begins to straighten so as to create a strong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

Gear-rack drive



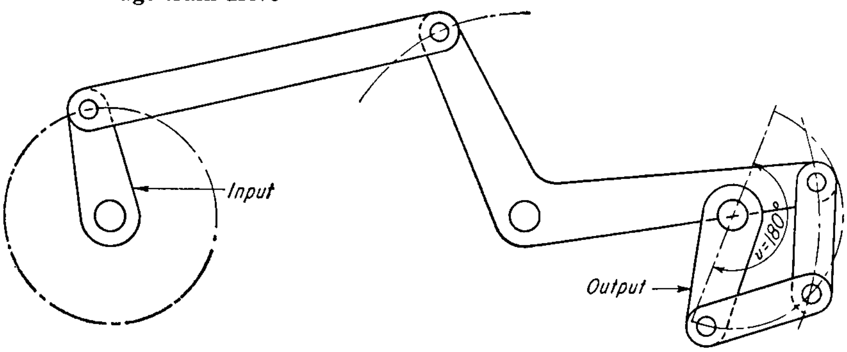
This drive mechanism converts the motion of an input crank into a much larger rotation of the output (from 30° to 360°). The crank drives the slider and gear rack, which in turn rotates the output gear.

Chain drive

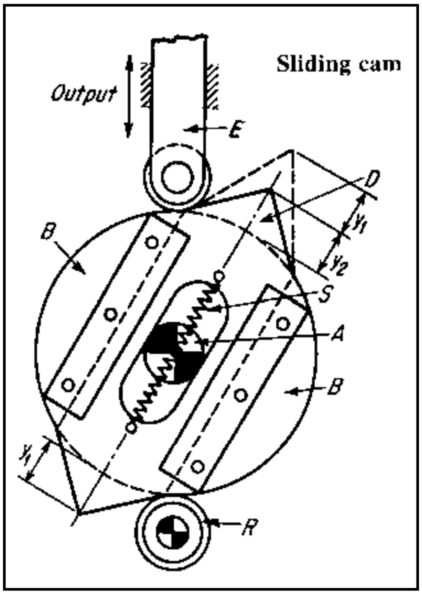


Springs and chains are attached to geared cranks of this drive to operate a sprocket output. Depending on the gear ratio, the output will produce a desired oscillation, e.g., two revolutions of output in each direction for each 360° of input.

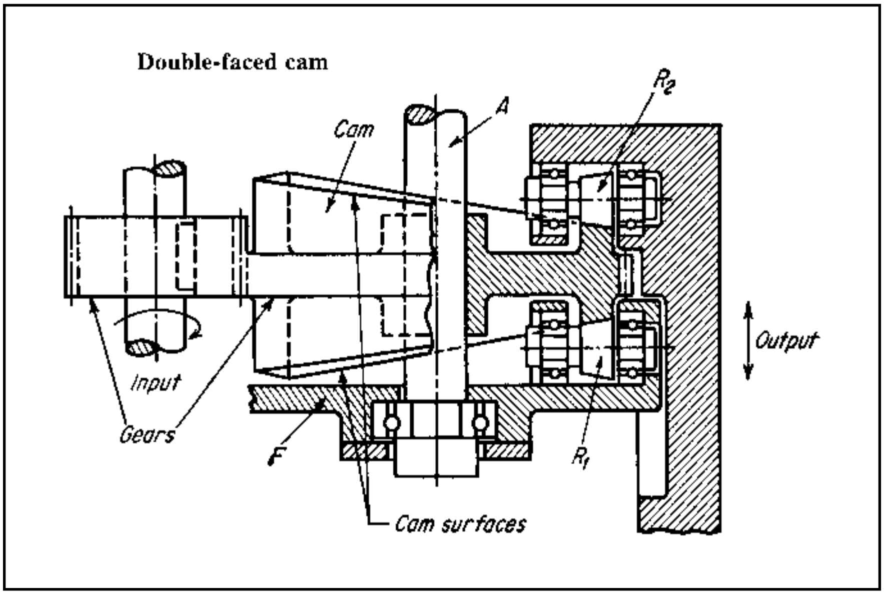
Linkage-train drive



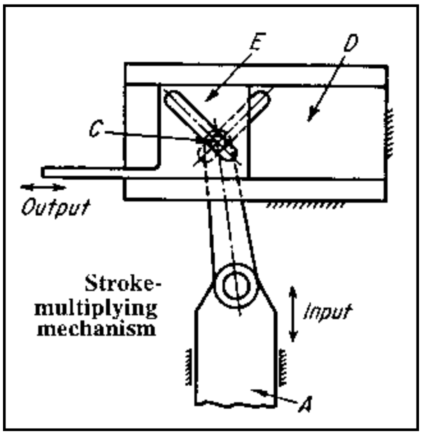
Arranging linkages in series on this drive can increase its angle of oscillation. In the version illustrated, the oscillating motion of the L-shaped rocker is the input for the second linkage. The final oscillation is 180°.



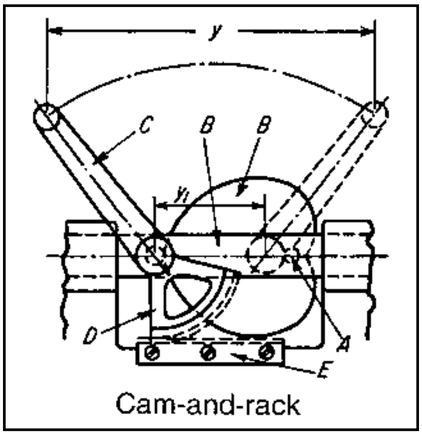
Sliding cam



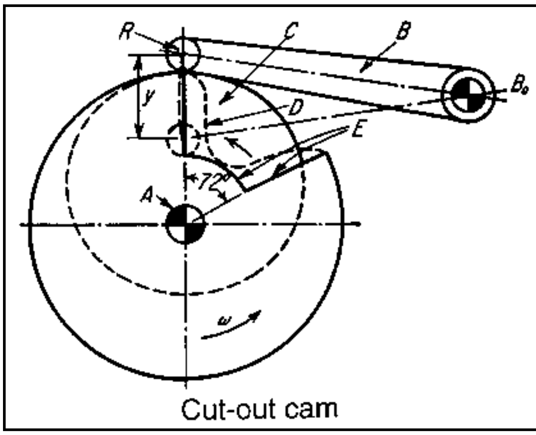
Double-faced cam



Stroke-multiplying mechanism



Cam-and-rack



Cut-out cam

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

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

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

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

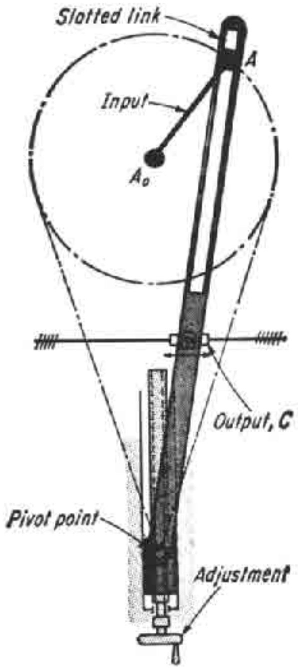
Double-faced cam—This mechanism doubles the stroke, hence reduces the pressure angles to one-half of their original values. Roller R_1 is stationary. When the cam rotates, its bottom surface lifts itself on R_1, while its top surface adds an additional motion to the movable roller R_2. The output is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

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

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

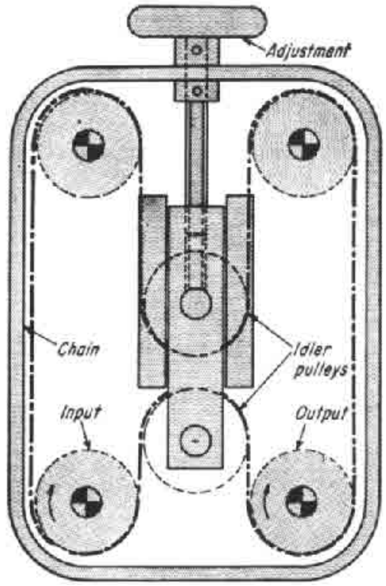
ADJUSTABLE-STROKE MECHANISMS

Adjustable-slider drive



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

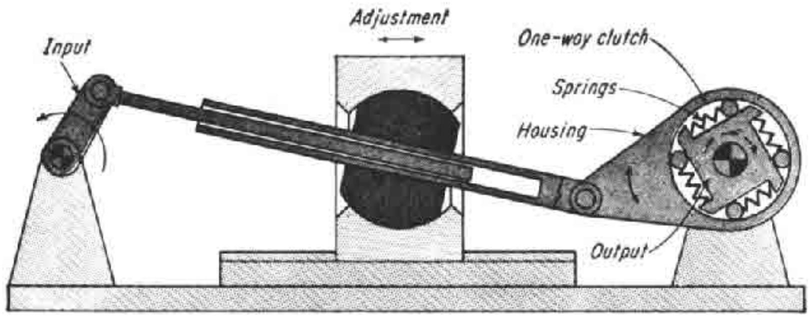
Adjustable-chain drive



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

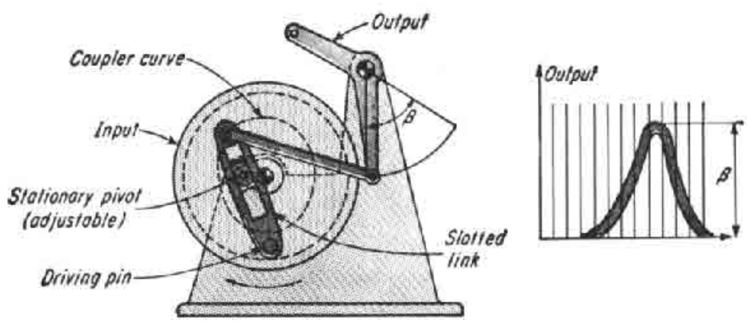
Adjustable-clutch drive

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

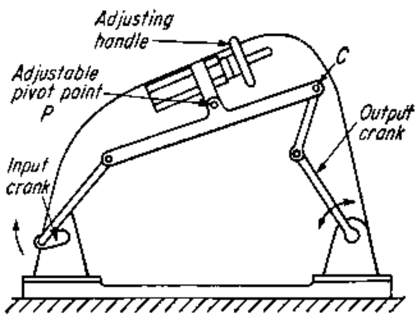


Adjustable-pivot drive

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

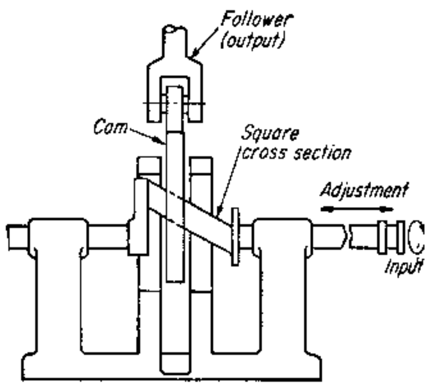


Linkage-motion adjuster



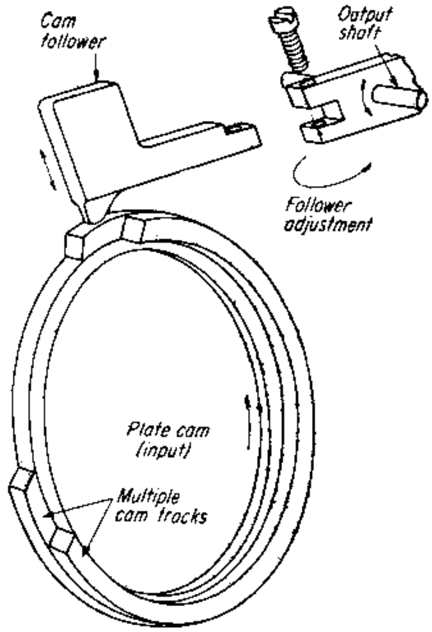
Here the motion and timing of the output link can be varied during its operation by shifting the pivot point of the intermediate link of the six-bar linkage illustrated. Rotation of the input crank causes point *C* to oscillate around the pivot point *P*. This, in turn, imparts an oscillating motion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

Cam-motion adjuster



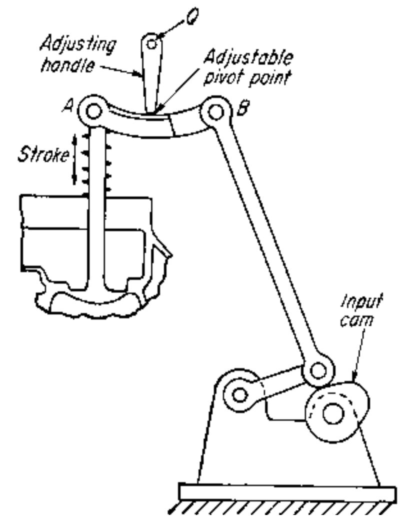
The output motion of the cam follower is varied by linearly shifting the input shaft to the right or left during its operation. The cam has a square hole which fits over the square cross section of the crank shaft. Rotation of the input shaft causes eccentric motion in the cam. Shifting the input shaft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

Double-cam mechanism



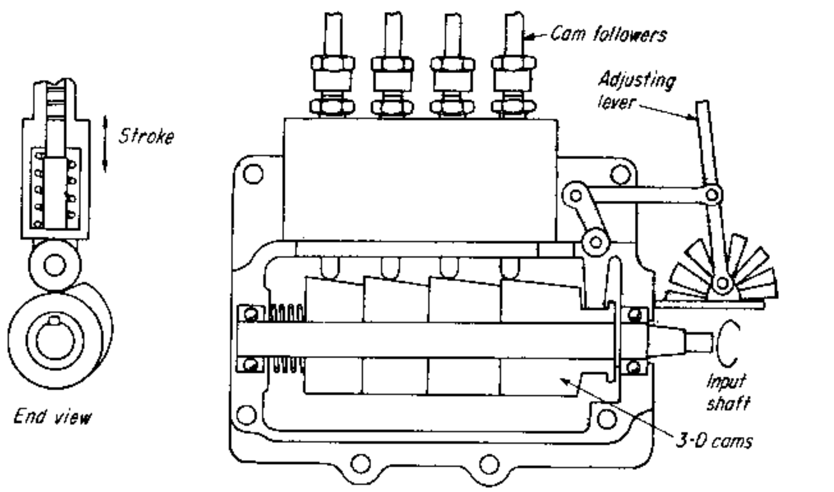
This is a simple but effective mechanism for changing the timing of a cam. The follower can be adjusted in the horizontal plane, but it is restricted in the vertical plane. The plate cam contains two or more cam tracks.

Valve-stroke adjuster



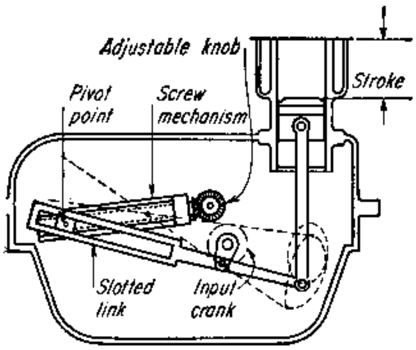
This mechanism adjusts the stroke of valves of combustion engines. One link has a curved surface and pivots around an adjustable pivot point. Rotating the adjusting link changes the proportion of strokes or points *A* and *B* and hence of the valve. The center of curvature of the curve link is at point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

3-D mechanism



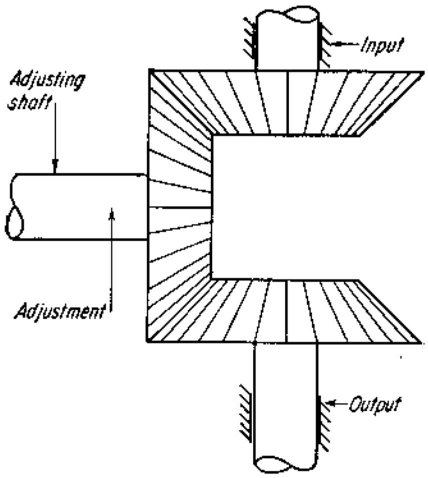
Output motions of four followers can be varied during the rotation by shifting the quadruple 3-D cam to the right or left. A linear shift can be made with the adjustment lever, which can be released in any of the six positions.

Piston-stroke adjuster



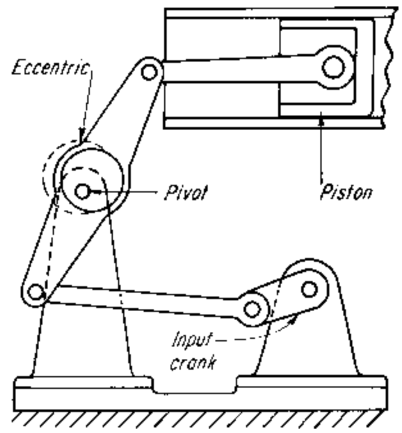
The input crank oscillates the slotted link to drive the piston up and down. The position of the pivot point can be adjusted with the screw mechanism even when the piston is under full load.

Shaft synchronizer



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

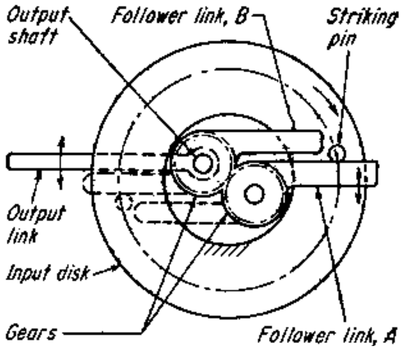
Eccentric pivot point



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

REVERSING MECHANISMS

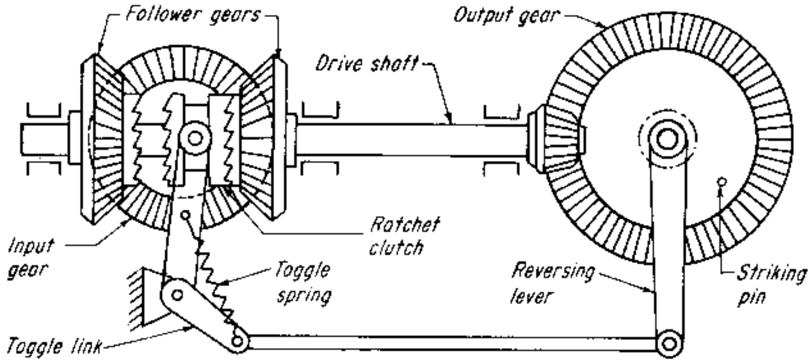
Double-link reverser



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

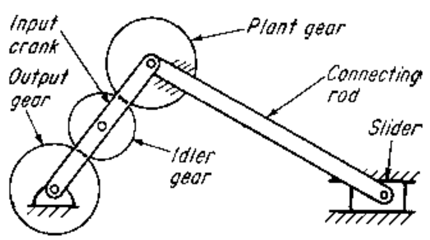
After approximately 180º of rotation, the pin slides past link *A* to strike link *B* coming to meet it—and thus reverses the direction of link *B* (and of the output). Then after another 180º rotation the pin slips past link *B* to strike link *A* and starts the cycle over again.

Toggle-link reverser



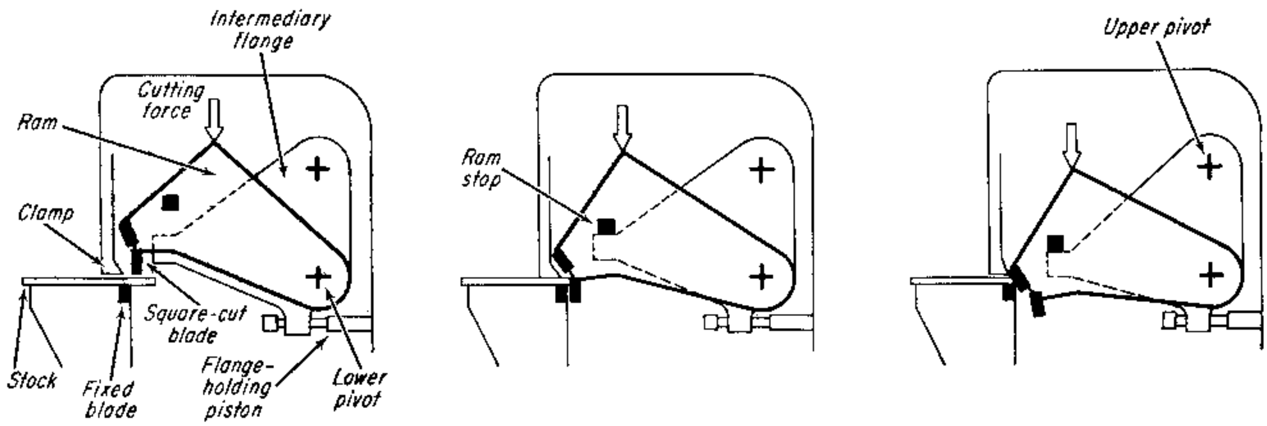
This mechanism also employs a striking pin—but here the pin is on the output member. The input bevel gear drives two follower bevels which are free to rotate on their common shaft. The ratchet clutch, however, is spline-connected to the shaft—although free to slide linearly. As shown, it is the right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

Modified-Watt's reverser



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

Automatically switching from one pivot point to another in midstroke.



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

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

COMPUTING MECHANISMS

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

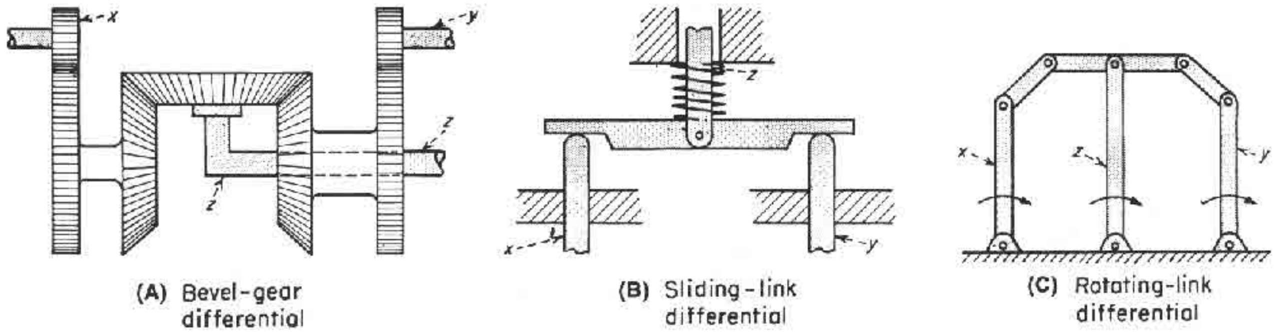


Fig. 1 Addition and subtraction is usually based on the differential principle; variations depend on whether inputs: (A) rotate shafts, (B) translate links, or (C) angularly displace links. Mechanisms can solve the equation: $z = c(x \pm y)$, where c is the scale factor, x and y are inputs, and z is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

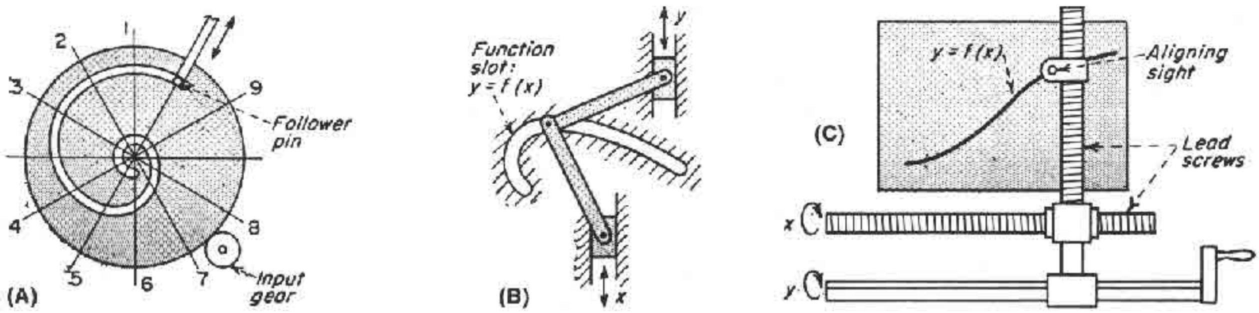


Fig. 2 Functional generators mechanize specific equations. (A) A reciprocal cam converts a number into its reciprocal. This simplifies division by permitting simple multiplication between a numerator and its denominator. The cam is rotated to a position corresponding to the denominator. The distance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

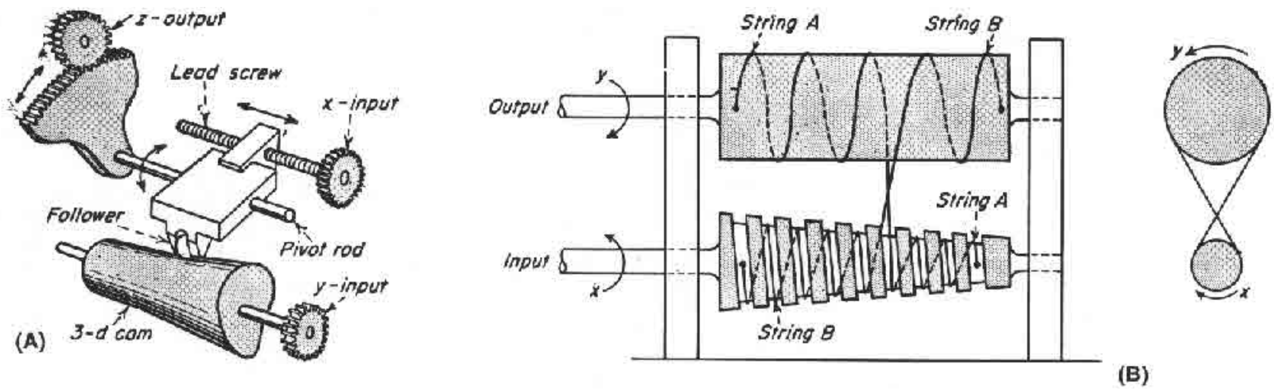


Fig. 3 (A) **A three-dimensional cam** generates functions with two variables: $z = f(x, y)$. A cam is rotated by the y-input; the x-input shifts a follower along a pivot rod. The contour of the cam causes a follower to rotate, giving angular displacement to the z-output gear. (B) **A conical cam** for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

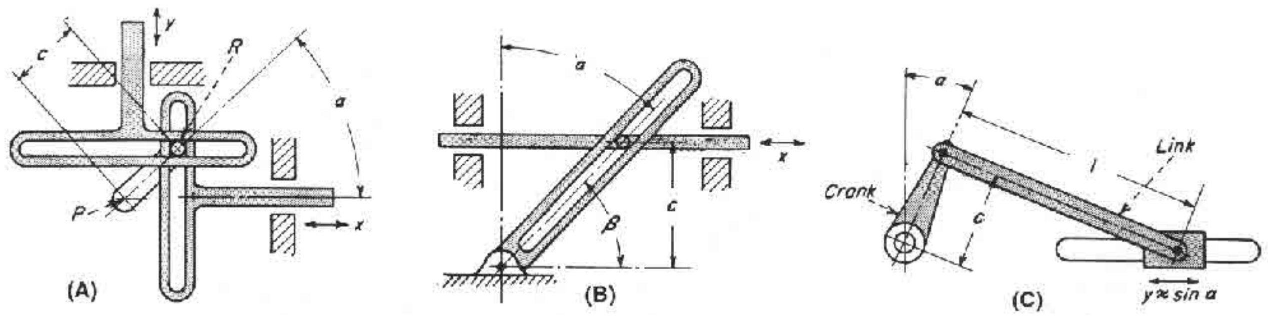


Fig. 4 Trigonometric functions. (A) A Scotch-yoke mechanism for sine and cosine functions. A crank rotates about fixed point P, generating angle a and giving motion to the arms: $y = c \sin a$; $x = c \cos a$. (B) A tangent-cotangent mechanism generates $x = c \tan a$ or $x = c \cot \beta$. (C) The eccentric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

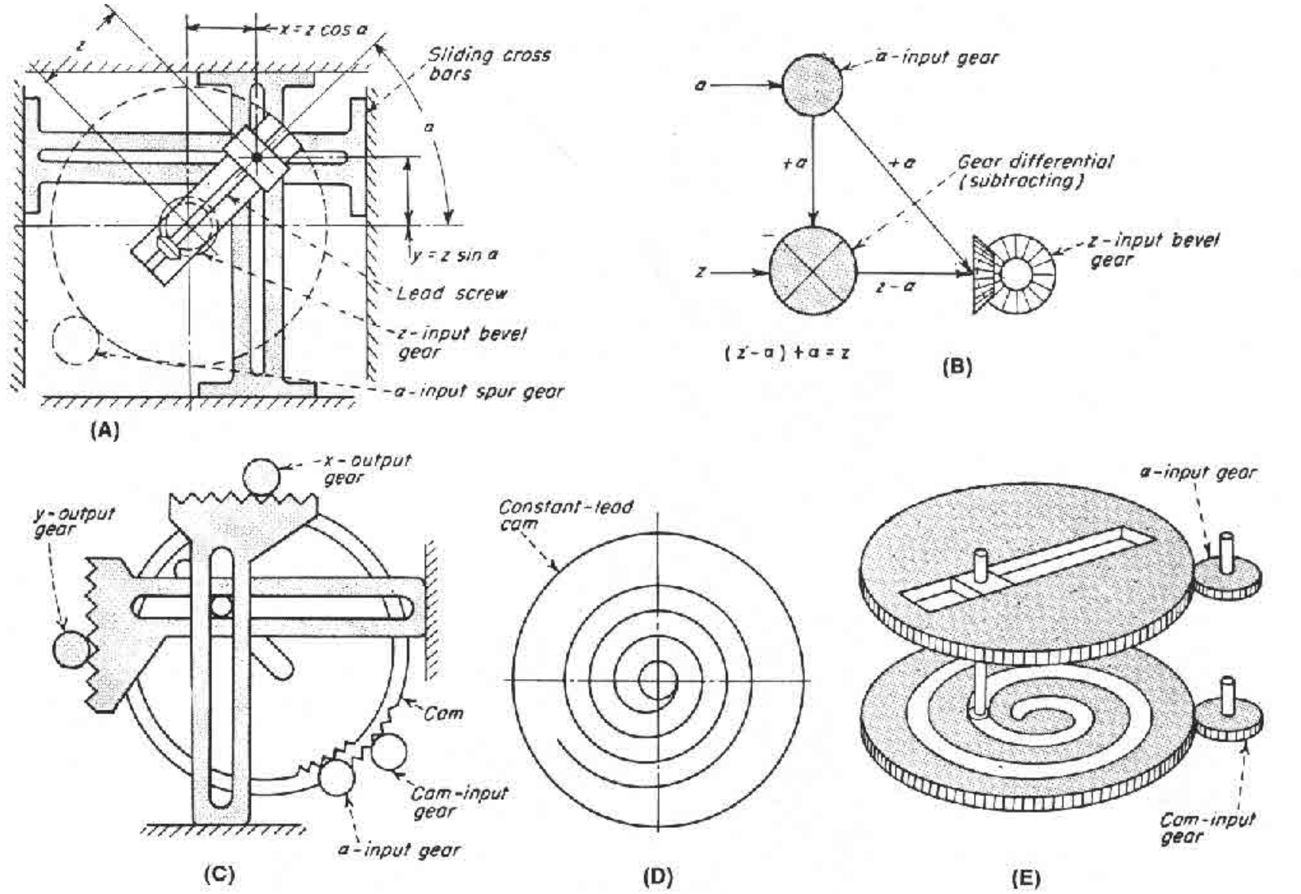


Fig. 5 Component resolvers determine x and y components of vectors that are continuously changing in both angle and magnitude. Equations are $x = z \cos a$, $y = z \sin a$, where z is magnitude of vector, and a is vector angle. Mechanisms can also combine components to obtain a resultant. Inputs in (A)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

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

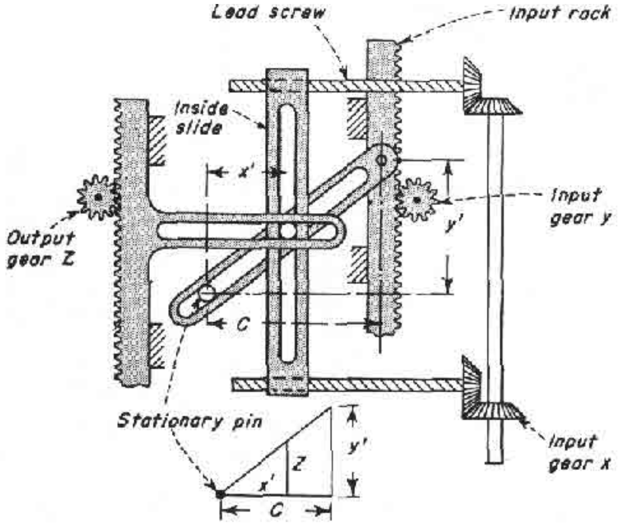


Fig. 1 (A)

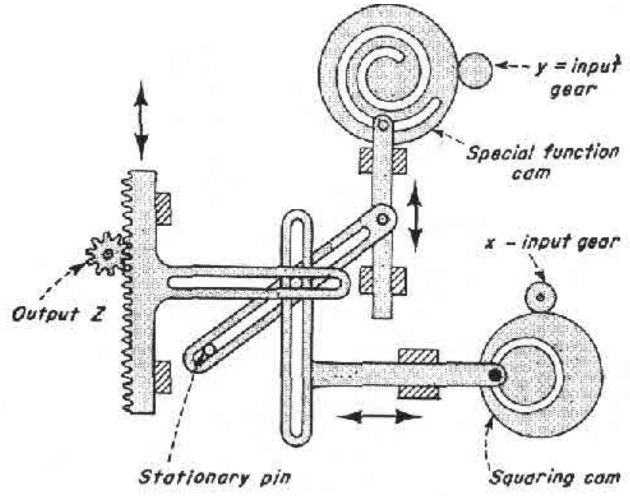


Fig. 2 (A)

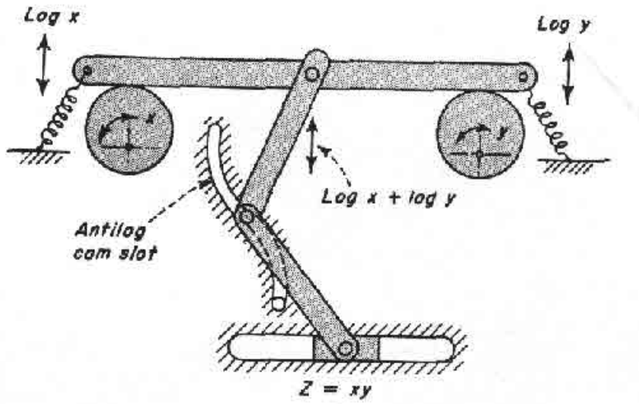


Fig. 1 (B)

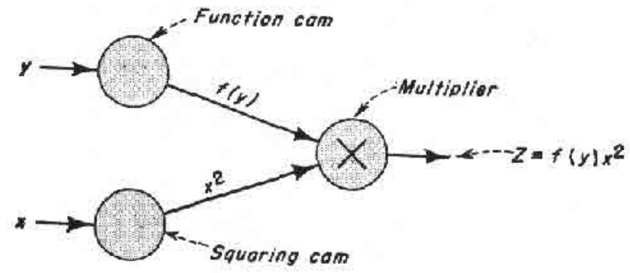


Fig. 2 (B)

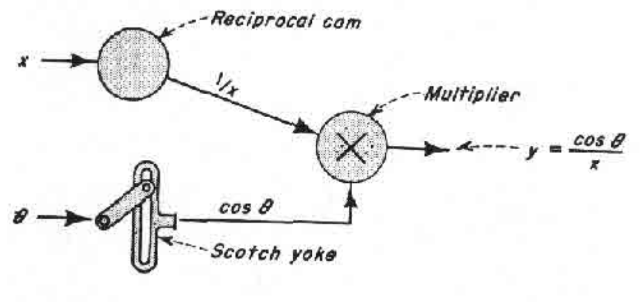
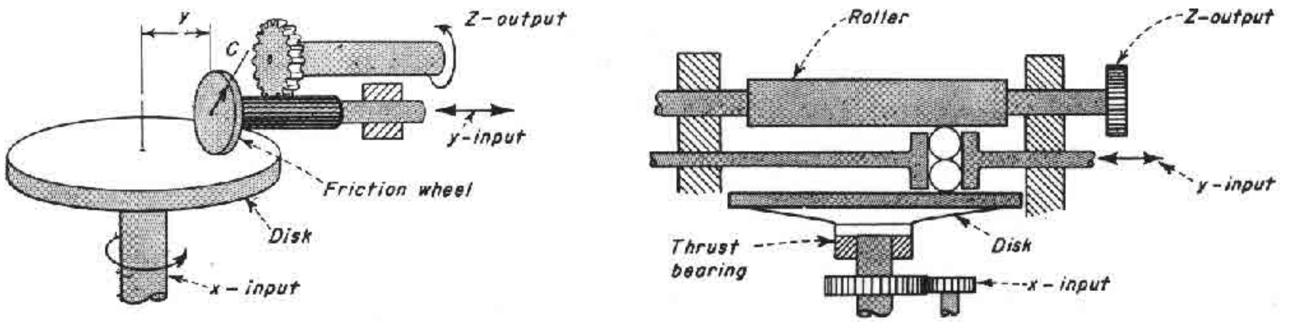


Fig. 2 (C)

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

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



(A) (B)

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

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

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

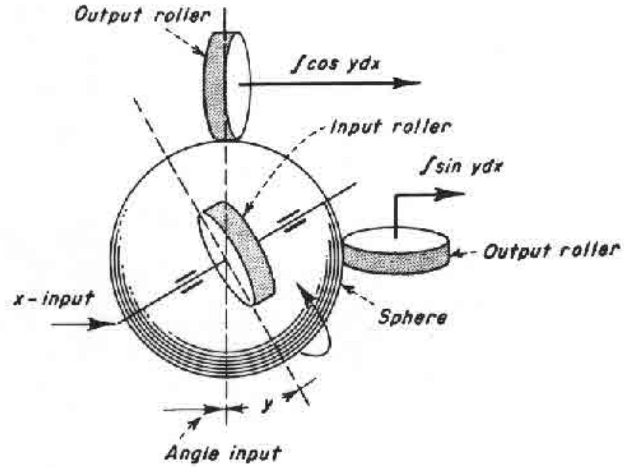


Fig. 4 A component integrator has three disks to obtain the *x* and *y* components of a differential equation. The input roller on the *x*-input shaft spins the sphere, and the *y*-input lever arm changes the angle of the roller with respect to the sphere. The sine and cosine output rollers provide integrals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

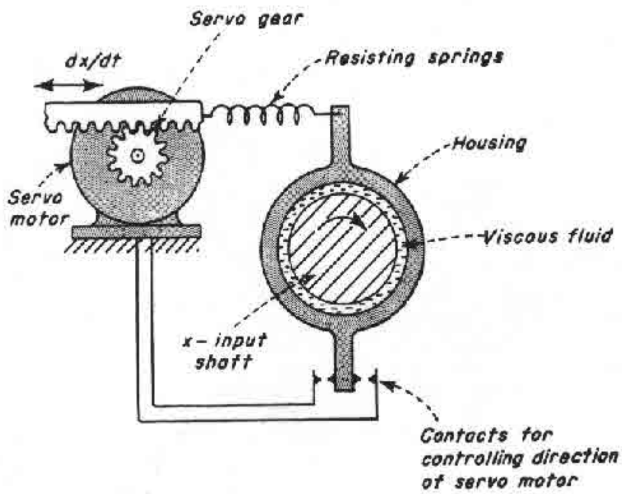
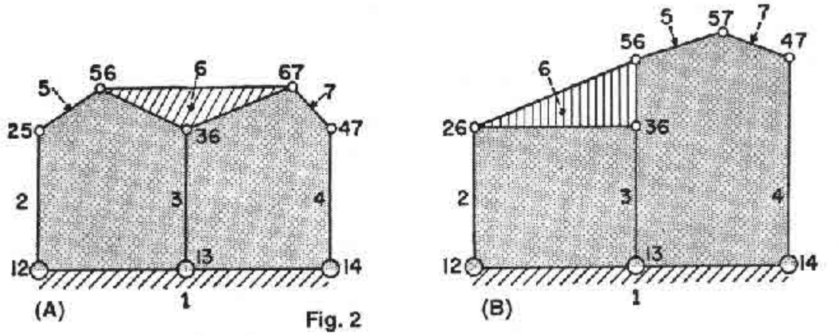


Fig. 5 This differentiator is based on the principle that a viscous drag force in a thin layer of fluid is proportional to the velocity of a rotating *x*-input shaft. The drag force is counteracted by resisting springs in tension. Spring length is regulated by a servomotor controlled by electrical contacts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

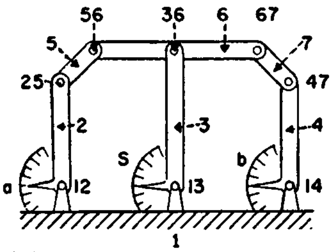
EIGHTEEN VARIATIONS OF DIFFERENTIAL LINKAGE

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

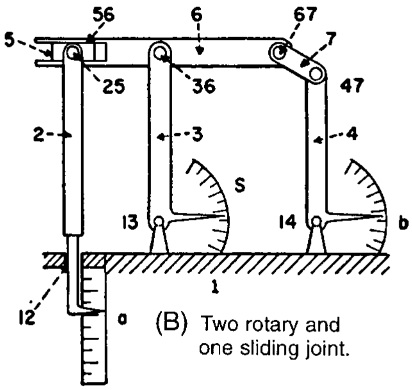


The input motions to be added are *a* and *b*; their sum *s* is equal to $c_1a + c_2b$, where c_1 and c_2 are scale factors. The links are numbered in the same way as those in Fig. 2(A).

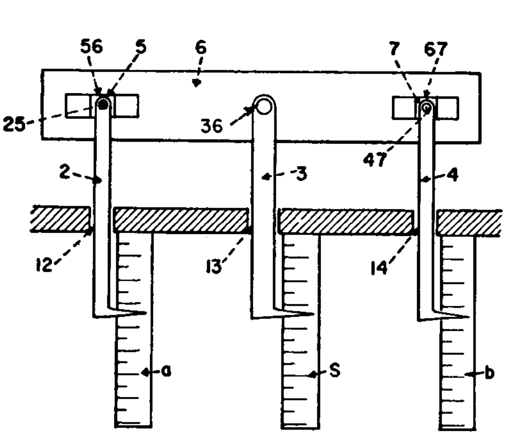
Fig. 1



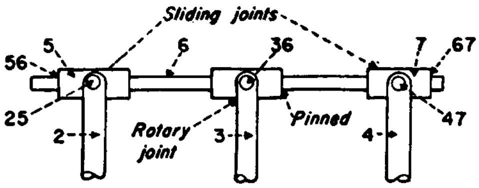
(A) Three rotary joints.



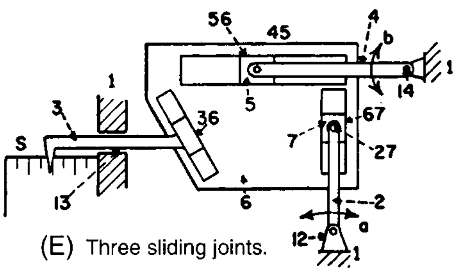
(B) Two rotary and one sliding joint.



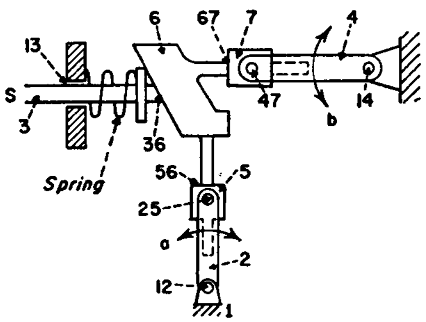
(C) One rotary and two sliding joints.



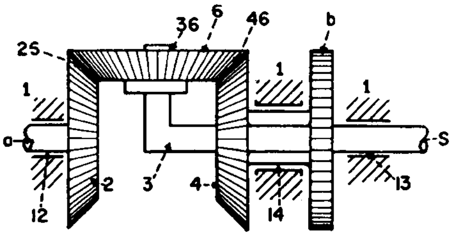
(D) One rotary and two sliding joints.



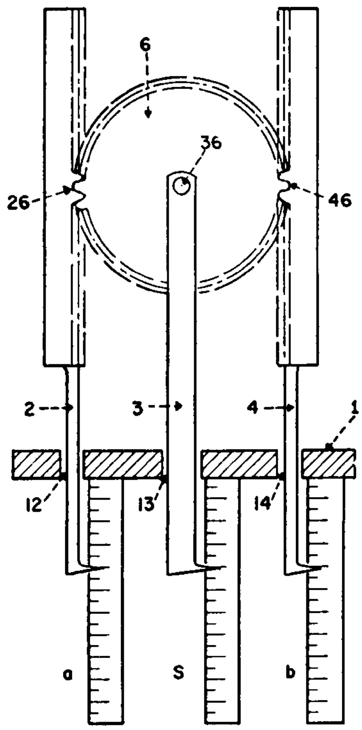
(E) Three sliding joints.



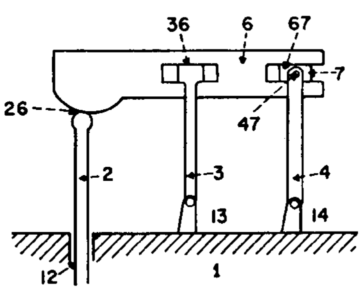
(F) Three sliding joints.



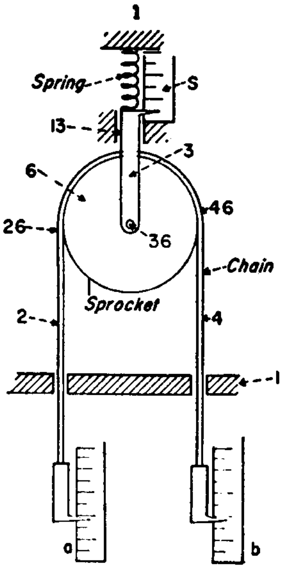
(G) One rotary and two cam joints (planetary gear differential)



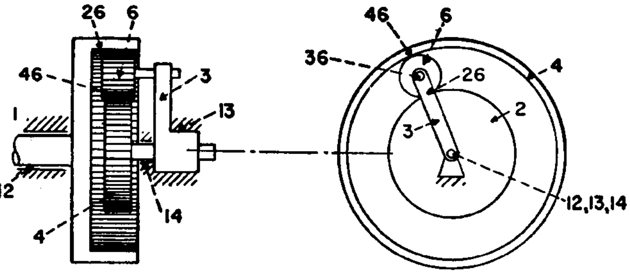
(H) One rotary and two cam joints (rack and gear).



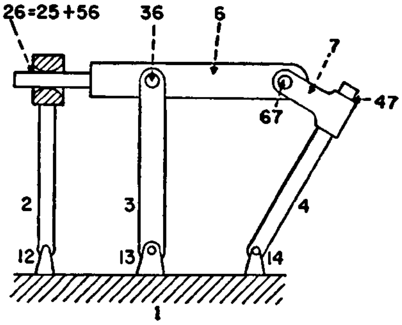
(I) Two sliding and one cam joint.



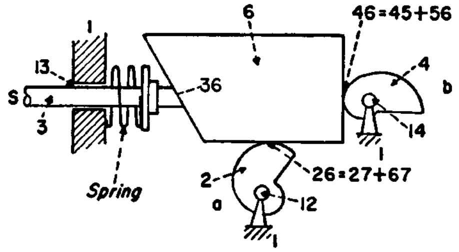
(J) One rotary and two cam joints
(chain and sprocket).



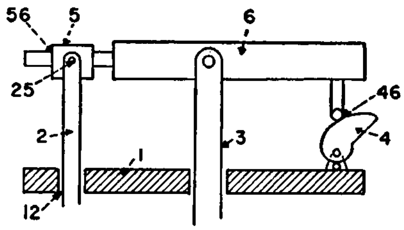
(M) One rotary and two cam joints
(planetary gears differential).



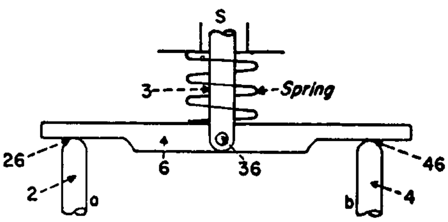
(O) Two rotary and one cam joint.



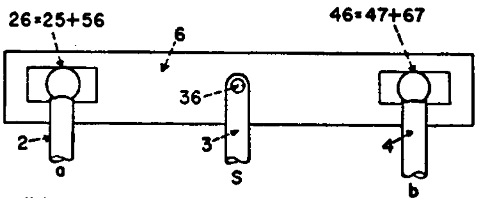
(N) One sliding and two cam joints.



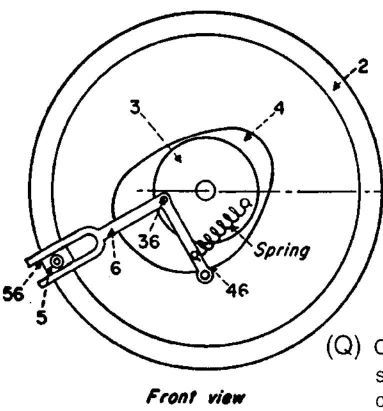
(P) One rotary, one sliding,
and one cam joint.

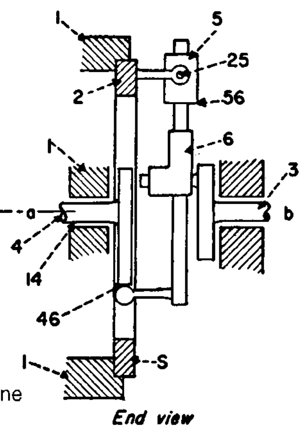


(K) One rotary and two cam joints.

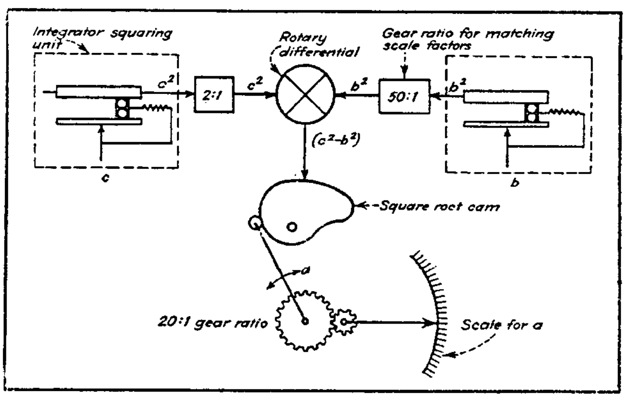


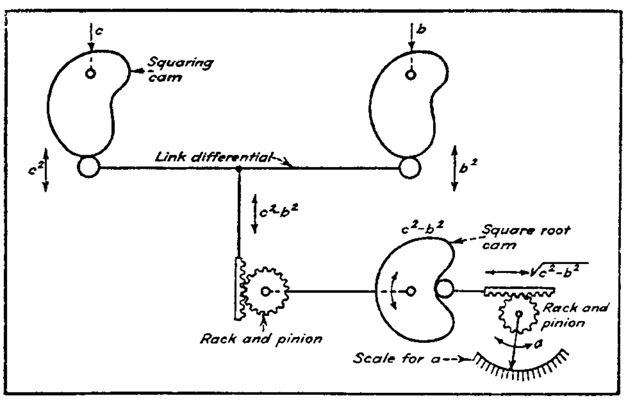
(L) One rotary and two cam joints.





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





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

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

SPACE MECHANISMS

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

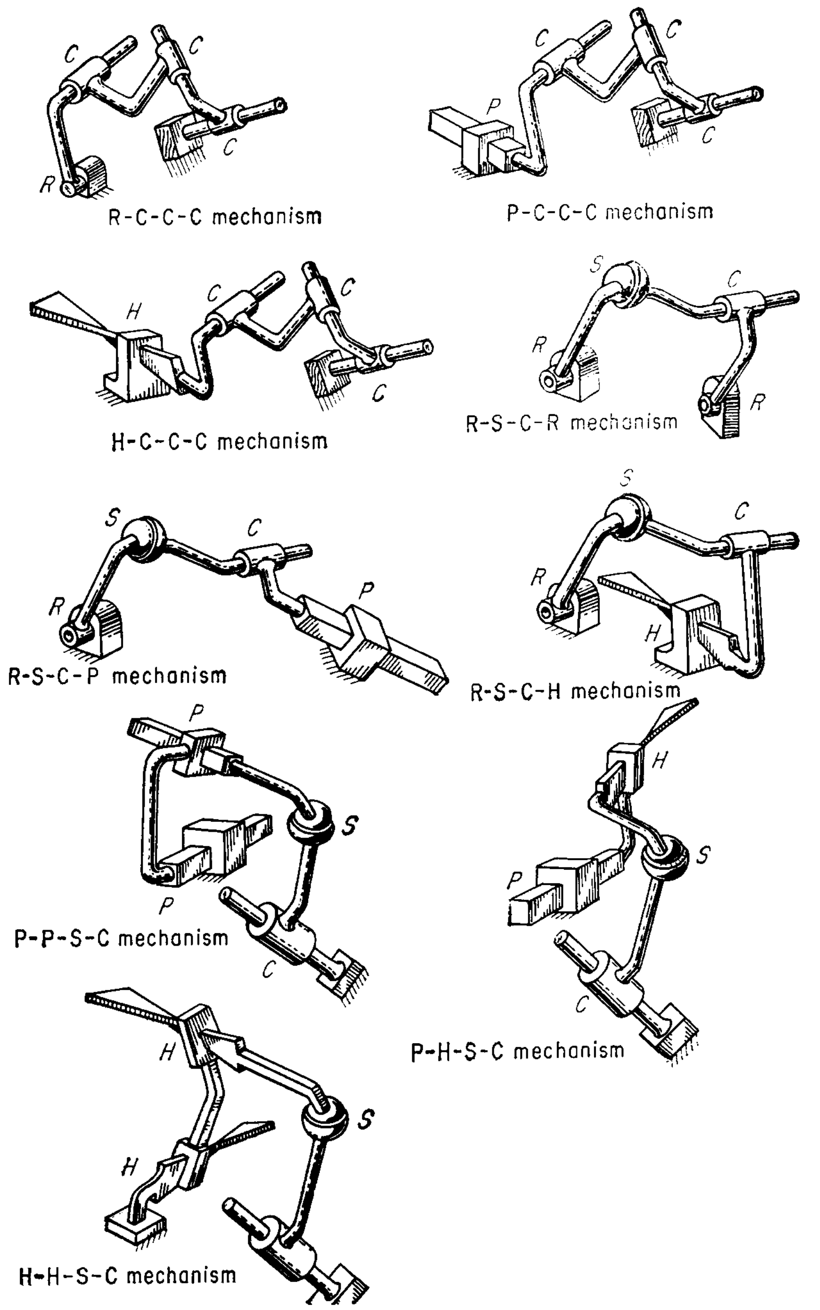


Fig. 1 The nine chosen mechanisms.

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

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

The Practical Nine

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

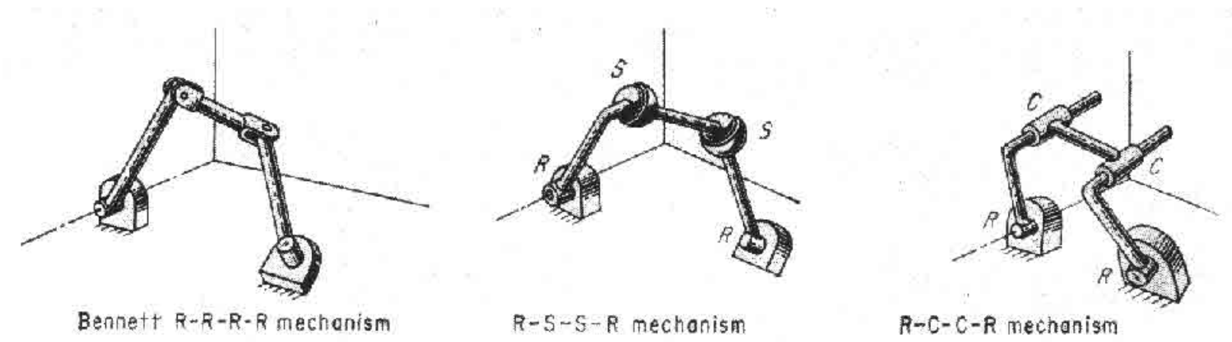


Fig. 2 The three mavericks.

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

R = Revolute joint, which permits rotation only

P = Prism joint, which permits sliding motion only

H = Helix or screw type of joint

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

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

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

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

The Mavericks

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

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

Classification of kinematic pairs

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

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

SEVEN POPULAR TYPES OF THREE-DIMENSIONAL DRIVES

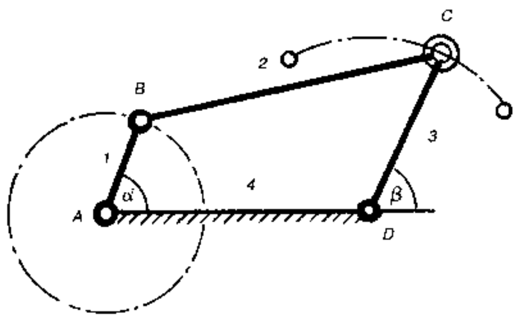
The main advantage of three-dimensional drives is their ability to transmit motion between nonparallel shafts. They can also generate other types of helpful motion. This roundup includes descriptions of seven industrial applications for the drives.

Spherical Crank Drive

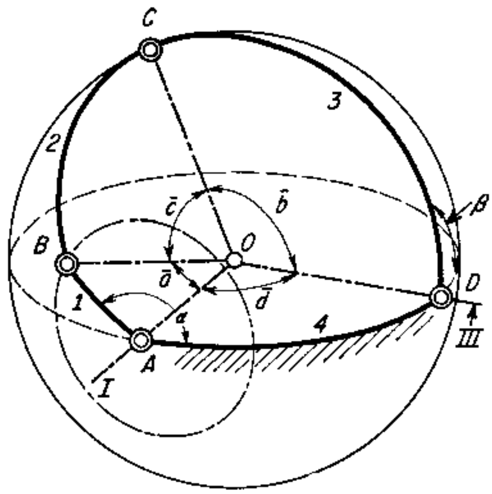
This type of drive is the basis for most three-dimensional linkages, much as the common four-bar linkage is the basis for the two-dimensional field. Both mechanisms operate on similar principles. (In the accompanying sketches, *a* is the input angle, and β the output angle. This notation has been used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

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

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



Four-bar linkage



The Spherical Crank

Spherical-Slide Oscillator Drive

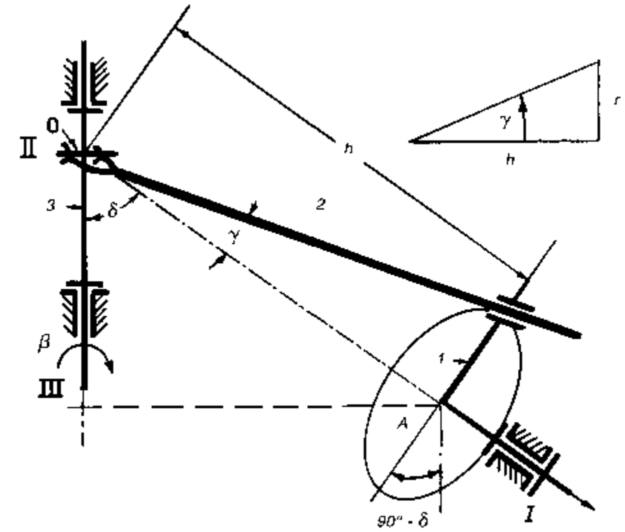
The two-dimensional slider crank is obtained from a four-bar linkage by making the oscillating arm infinitely long. By making an analogous change in the spherical crank, the spherical slider crank is obtained.

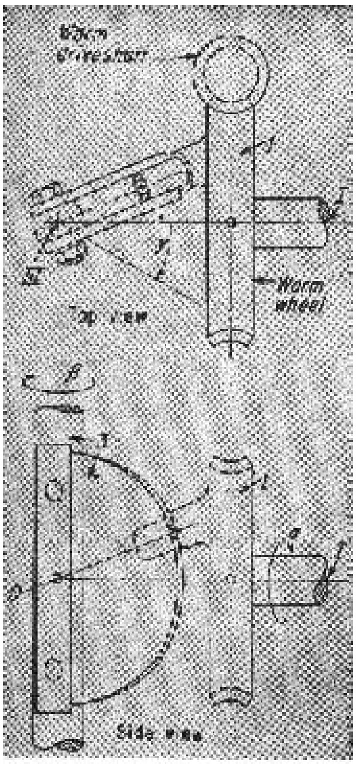
The uniform rotation of input shaft *I* is transferred into a nonuniform oscillating or rotating motion of output shaft *III*. These shafts intersect at an angle δ, corresponding to the frame link *4* of the spherical crank. Angle γ corresponds to the length of link *1*, and axis *II* is at right angle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

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

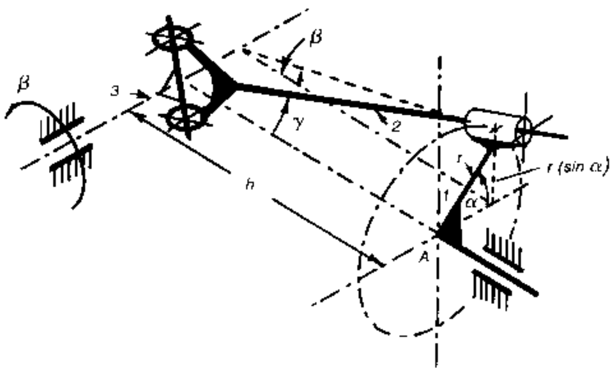
The relation between input angle *a* and output angle β as designated in the skewed Hooke's joint is:

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

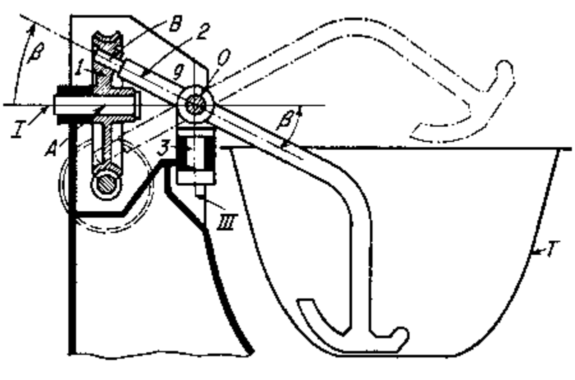




Washing-Machine Mechanism



Skewed Hooke's Joint



Dough-Kneading Mechanism

Skewed Hooke's Joint Drive

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

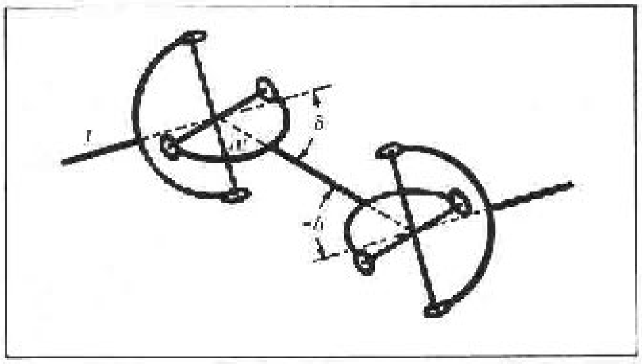
The equation defining the output in terms of the input can be obtained from the skewed Hooke's joint equation by making δ = 90°. Thus, sin δ = 1, cos δ = 0, and

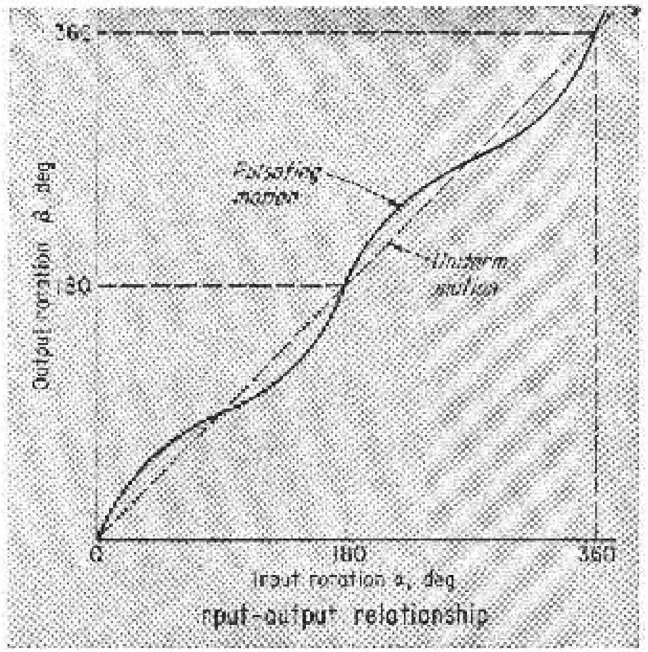
$$\tan\beta = \tan\gamma\sin\alpha$$

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

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

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





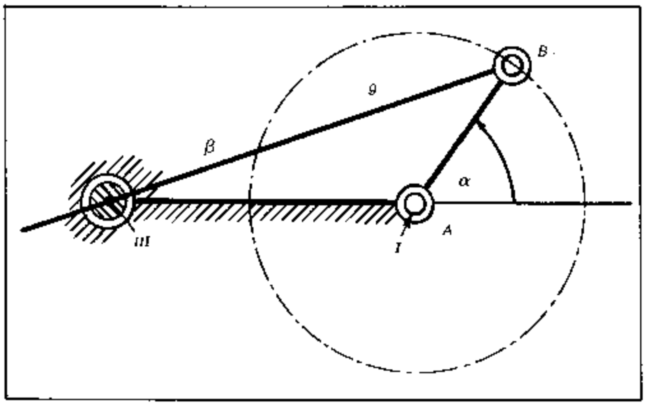
The Universal Joint Drive

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

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

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

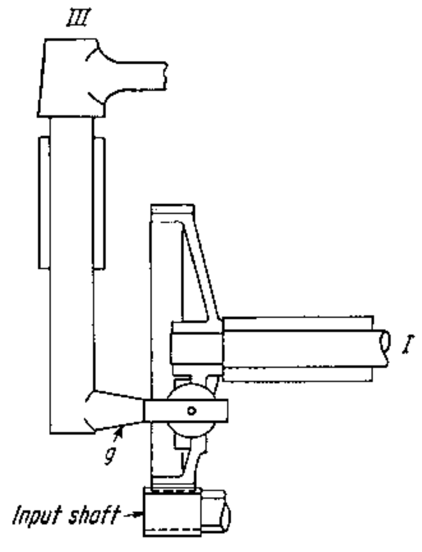
The output motion is pulsating (see curve) unless the joints are operates as pairs to provide a uniform motion.



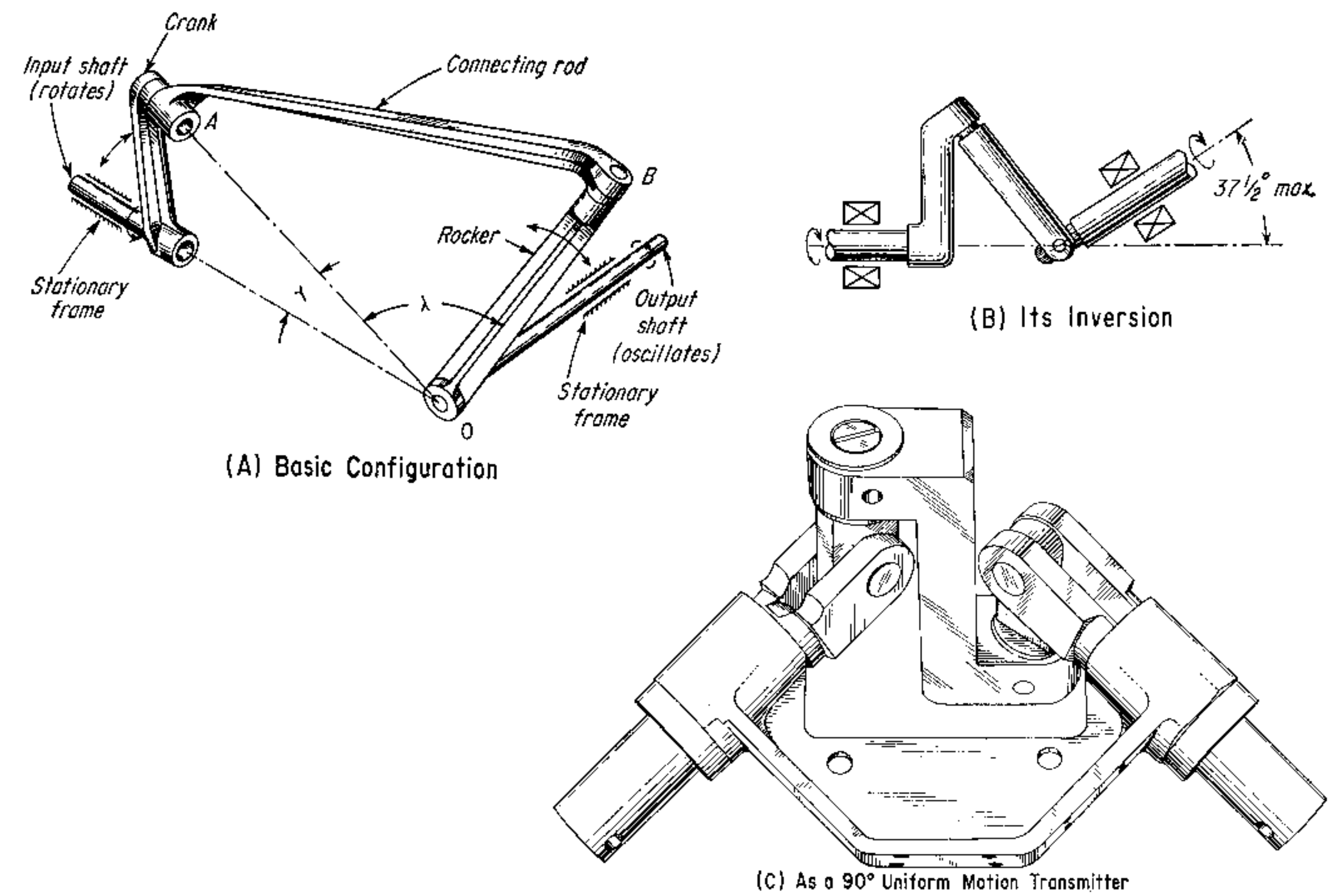
The 3-D Crank Slide Drive

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

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



Agitator Mechanism



The Space Crank Drive

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

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

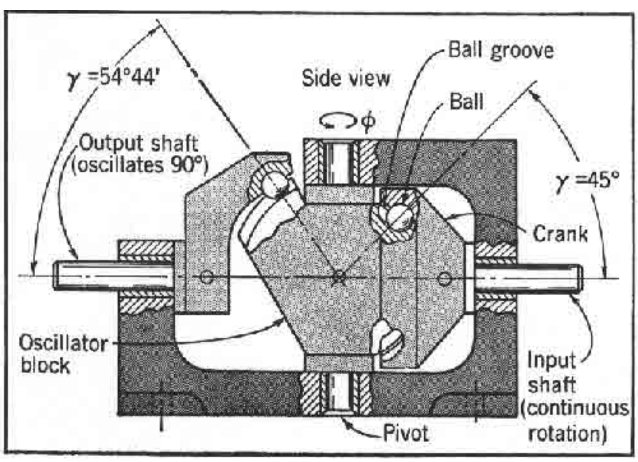
The velocity ratio is:

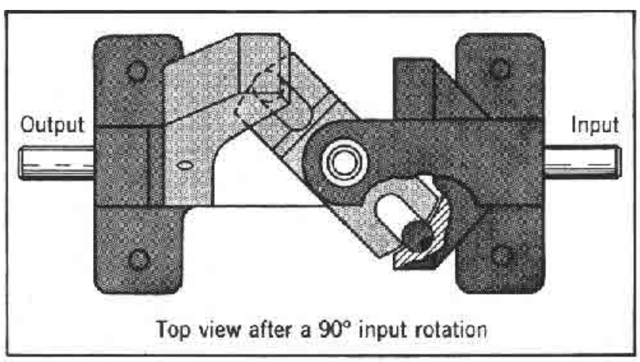
$$\frac{\omega_o}{\omega_i} = \frac{\tan\gamma\sin\alpha}{1+\tan\gamma\cos\alpha\cot\beta}$$

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

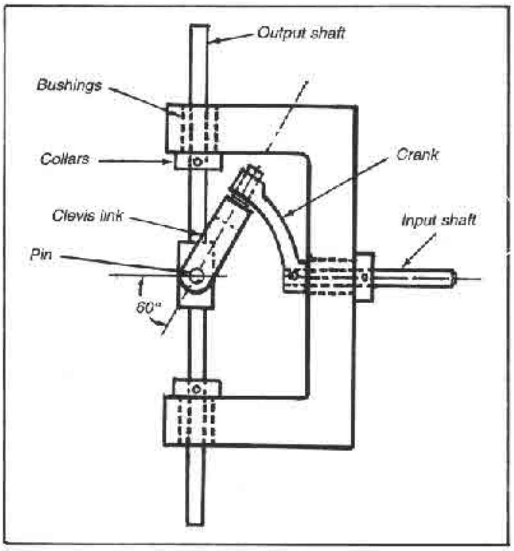
An inversion of the space crank is shown in (B). It can couple intersecting shafts, and permits either shaft to be driven with full rotations. Motion is transmitted up to 37½º misalignment.

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

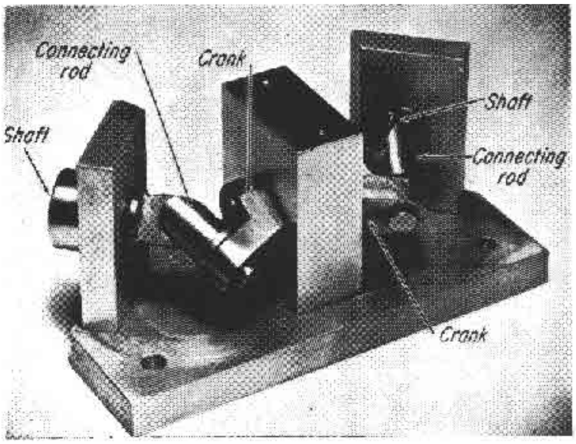




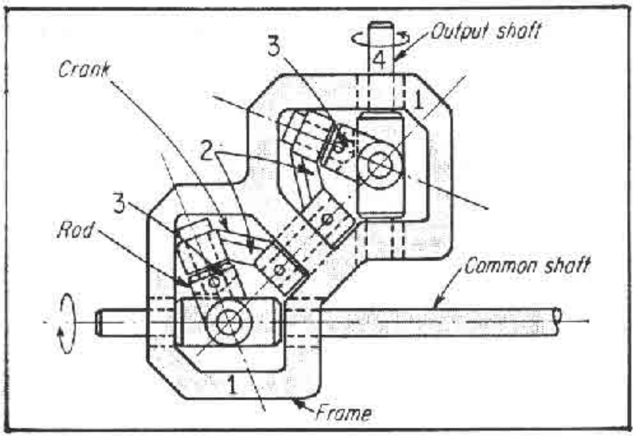
Steel balls riding within spherical grooves convert a continuous rotary input motion into an output that oscillates the shaft back and forth.



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



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



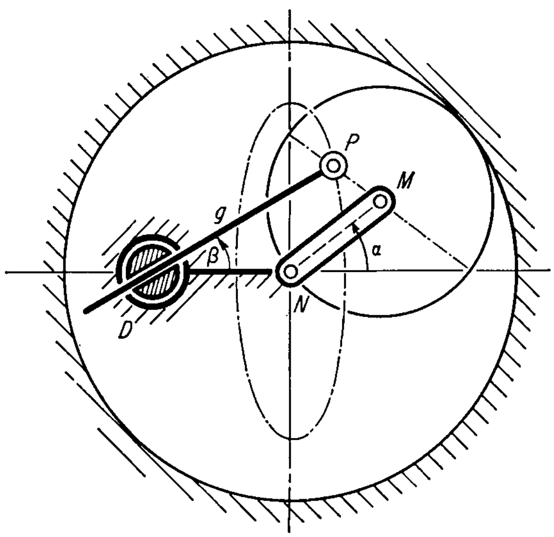
A right-angle limited-stroke drive transmits an exact motion pattern. A multiplicity of fittings can be operated from a common shaft.

The Elliptical Slide Drive

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

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

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



INCHWORM ACTUATOR

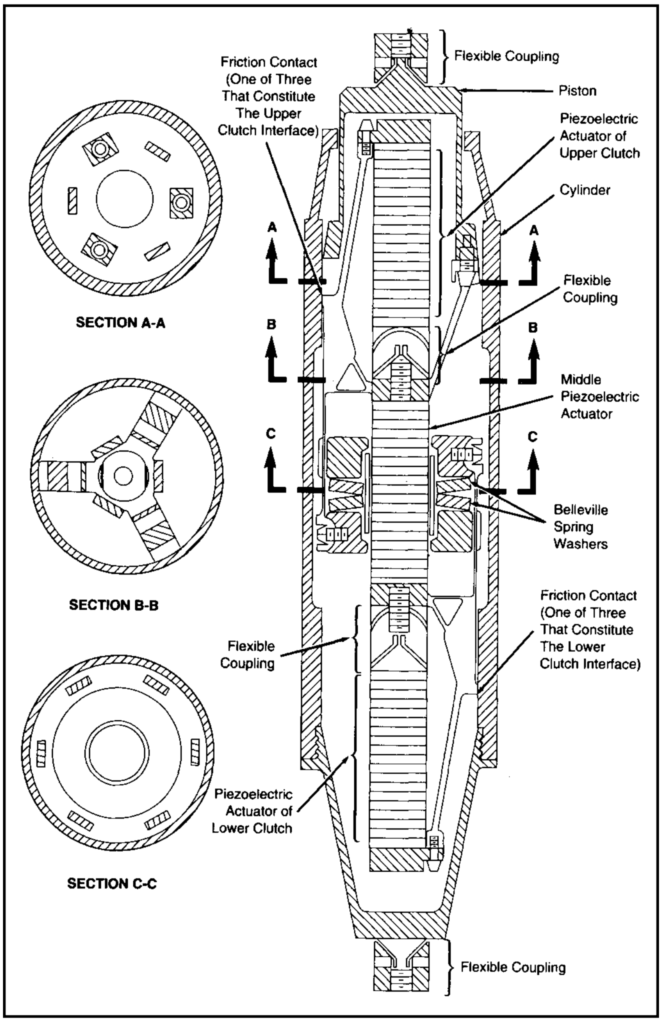
This actuator would serve as an active truss member.
NASA's Jet Propulsion Laboratory, Pasadena, California

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

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

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

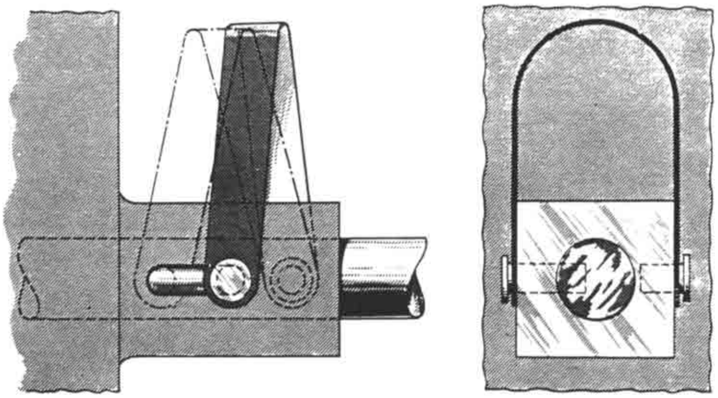
This work was done by Robert M. Bamford of Caltech for **NASA's Jet Propulsion Laboratory.**



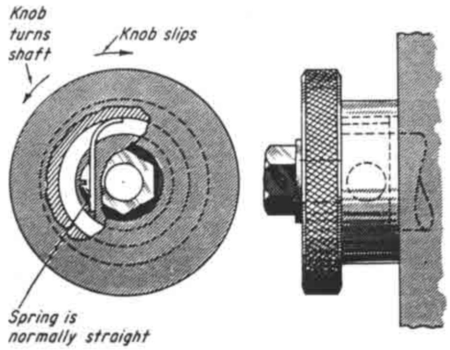
This inchworm actuator would hold its position (that is, it would neither extend nor retract) when electrical energy was not supplied. The maximum end-to-end stroke (extension or retraction of the piston with respect to the cylinder) would be 3 cm.

CHAPTER 6
SPRING, BELLOW, FLEXURE, SCREW, AND BALL DEVICES

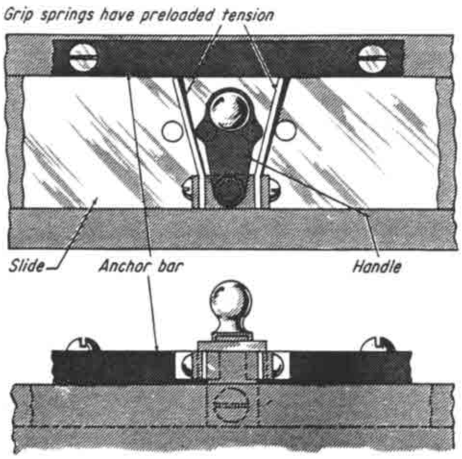
FLAT SPRINGS IN MECHANISMS



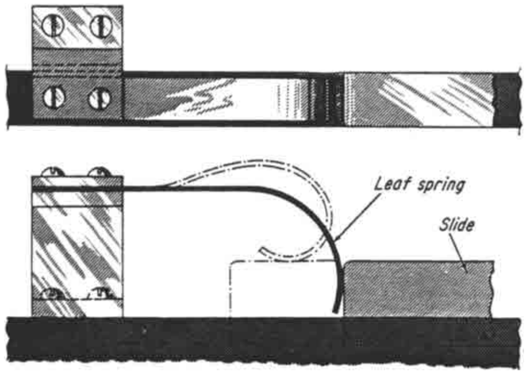
Constant force is approached because of the length of this U-spring. Don't align the studs or the spring will fall.



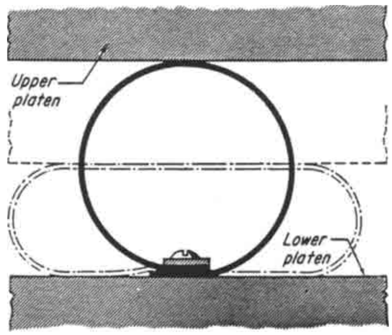
A flat-wire sprag is straight until the knob is assembled: thus tension helps the sprag to grip for one-way clutching.



Easy positioning of the slide is possible when the handle pins move a grip spring out of contact with the anchor bar.

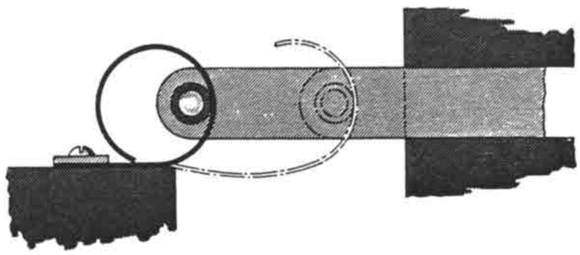


A spring-loaded slide will always return to its original position unless it is pushed until the spring kicks out.

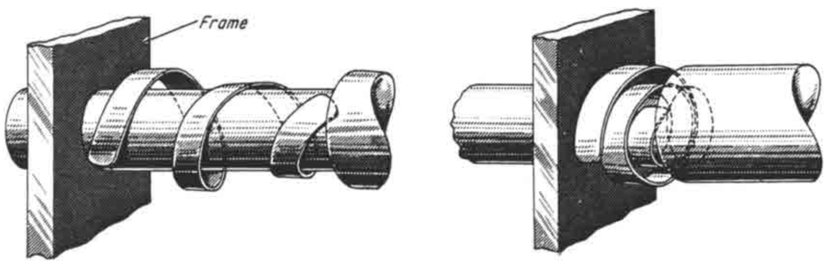


Increasing support area as the load increases on both upper and lower platens is provided by a circular spring.

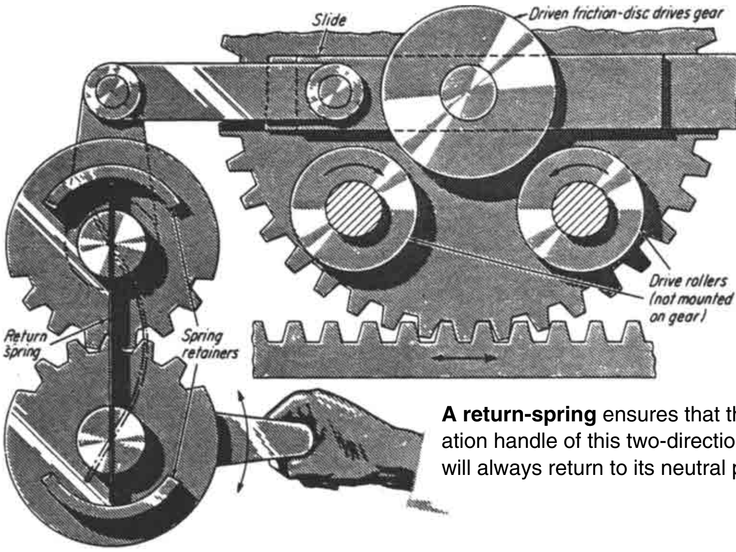
Nearly constant tension in the spring, as well as the force to activate the slide, is provided by this single coil.



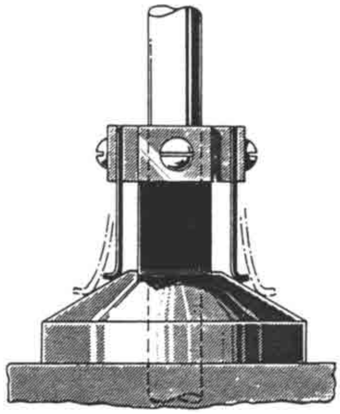
This volute spring lets the shaft be moved closer to the frame, thus allowing maximum axial movement.



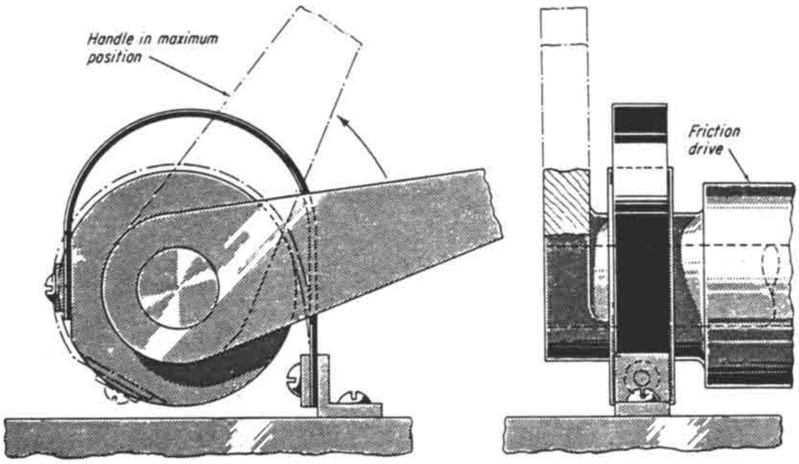
These mechanisms rely on a flat spring for their efficient actions.



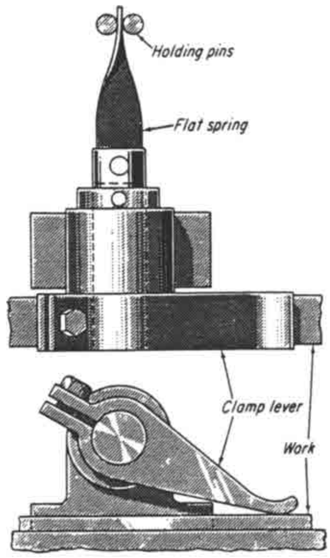
A return-spring ensures that the operation handle of this two-direction drive will always return to its neutral position.



This cushioning device imparts rapid increase of spring tension because of the small pyramid angle. Its rebound is minimum.

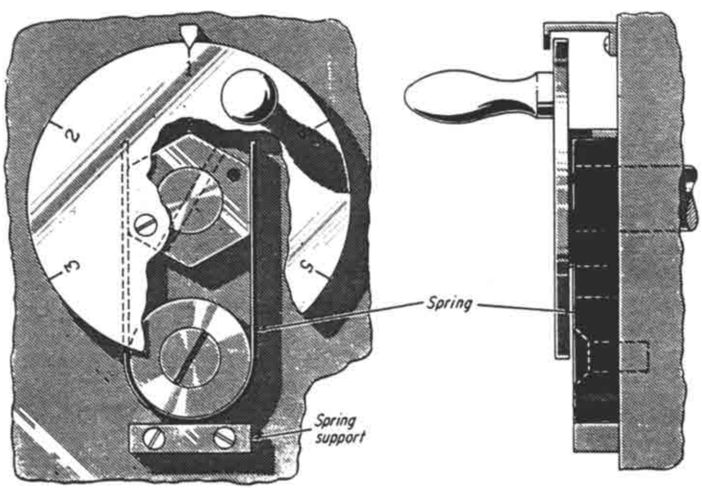


This spring-mounted disk changes its center position as the handle is rotated to move the friction drive. It also acts as a built-in limit stop.

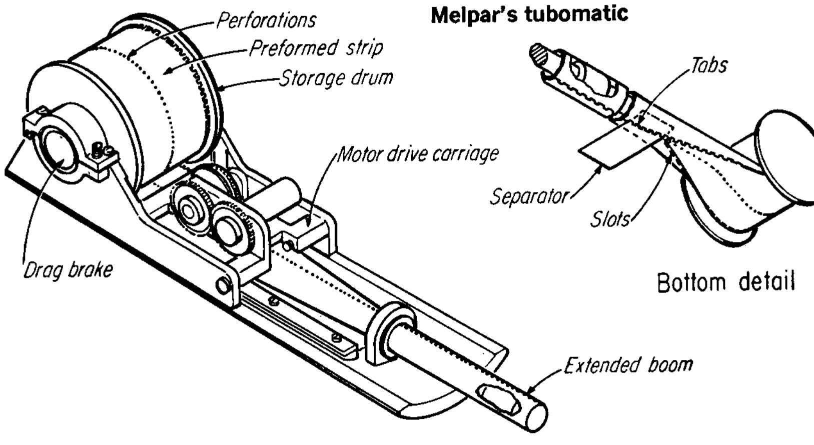


This hold-down clamp has its flat spring assembled with an initial twist to provide a clamping force for thin material.

Indexing is accomplished simply, efficiently, and at low cost by flat-spring arrangement shown here.



POP-UP SPRINGS GET NEW BACKBONE



An addition to the family of retractable coil springs, initially popular for use as antennas, holds promise of solving one problem in such applications: lack of torsional and flexural rigidity when extended. A pop-up boom that locks itself into a stiffer tube has been made.

In two previous versions—De Havilland Aircraft's Stem and Hunter Springs's Helix—rigidity was obtained by permitting the material to overlap. In Melpar's design, the strip that unrolls from the drum to form the cylindrical mast has tabs and slots that interlock to produce a strong tube.

Melpar has also added a row of perforations along the center of the strip to aid in accurate control of the spring's length during extension or contraction. This adds to the spring's attractiveness as a positioning device, besides its established uses as antennas for spacecraft and portable equipment and as gravity gradient booms and sensing probes.

Curled by heat. Retractable, prestressed coil springs have been in the technical news for many years, yet most manufacturers have been rather close-mouthed about exactly how they covert a strip of beryllium copper or stainless steel into such a spring.

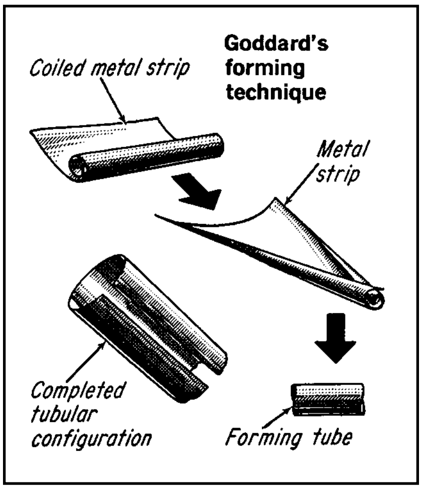
In its Helix, Hunter induced the prestressing at an angle to the axis of the strip, so the spring uncoils helically; De Havilland and Melpar prestress the material along the axis of the strip.

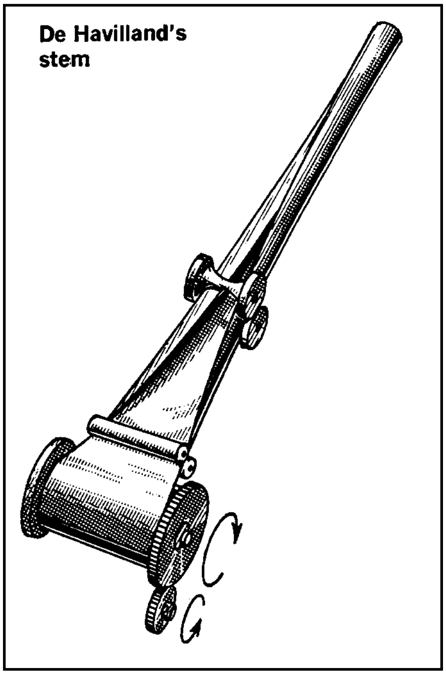
A prestressing technique was worked out by John J. Park of the NASA Goddard Center. Park found early in his assignment that technical papers were lacking on just how a metal strip can be given a new "memory" that makes it curl longitudinally unless restrained.

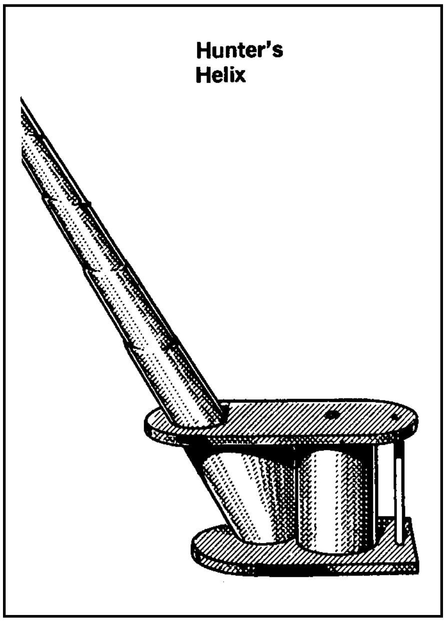
Starting from scratch, Park ran a series of experiments using a glass tube, 0.65 in. ID, and strips of beryllium copper allow, 2 in. wide and 0.002 in. thick. He found it effective to roll the alloy strip lengthwise into the glass tube and then to heat it in a furnace. Test strips were then allowed to cool down to room temperature.

It was shown that the longer the treatment and the hotter the furnace time, the more tightly the strip would curl along its length, producing a smaller tube. For example, a test strip heated at 920° F for 5 min would produce a tube that remained at the 0.65-in. inside diameter of the glass holder; at 770 F, heating for even 15 min produced a tube that would expand to an 0.68-in. diameter.

By proper correlation of time and temperature in the furnace, Park suggested that a continuous tube-forming process could be set up and segments of the completed tube could be cut off at the lengths desired.



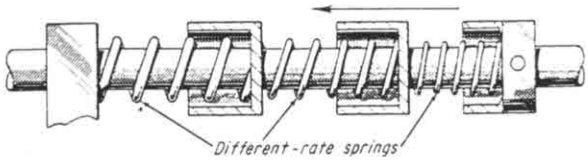




TWELVE WAYS TO PUT SPRINGS TO WORK

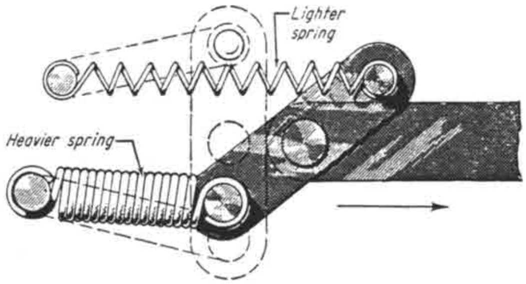
Variable-rate arrangements, roller positioning, space saving, and other ingenious ways to get the most from springs.

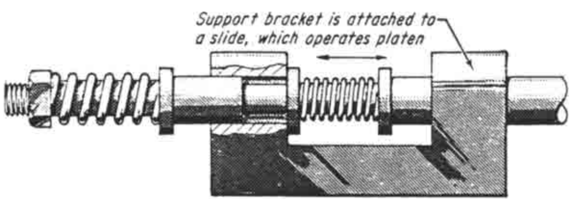
This setup provides a **variable rate** with a sudden change from a light load to a heavy load by limiting the low-rate extension with a spring.



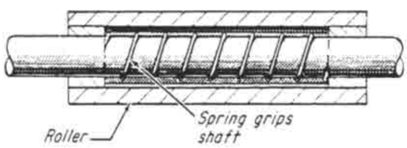
This mechanism provides a **three-step rate** change at predetermined positions. The lighter springs will always compress first, regardless of their position.

This differential-rate linkage sets the actuator stroke under light tension at the start, then allows a gradual transition to heavier tension.

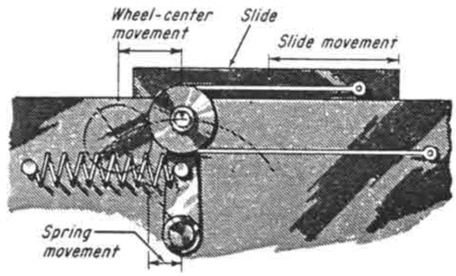




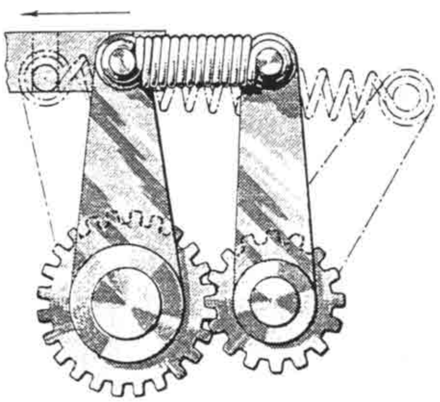
This compressing mechanism has a dual rate for double-action compacting. In one direction pressure is high, but in the reverse direction pressure is low.



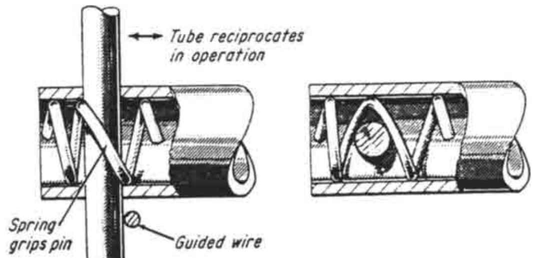
Roller positioning by a tightly wound spring on the shaft is provided by this assembly. The roller will slide under excess end thrust.



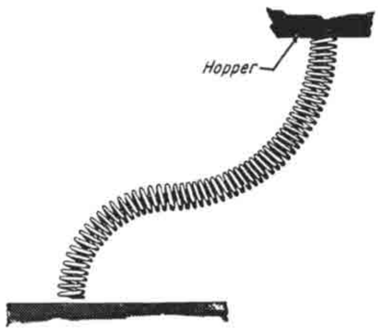
A short extension of the spring for a long movement of the slide keeps the tension change between maximum and minimum low.



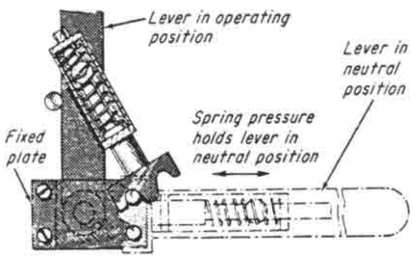
Increased tension for the same movement is gained by providing a movable spring mount and gearing it to the other movable lever.



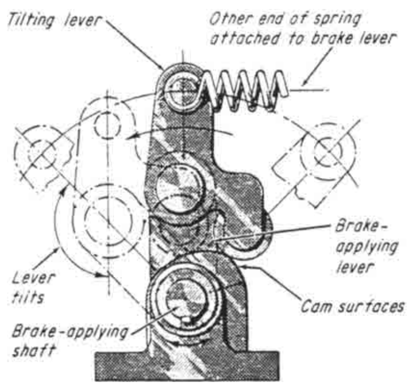
This pin grip is a spring that holds a pin by friction against end movement or rotation, but lets the pin be repositioned without tools.



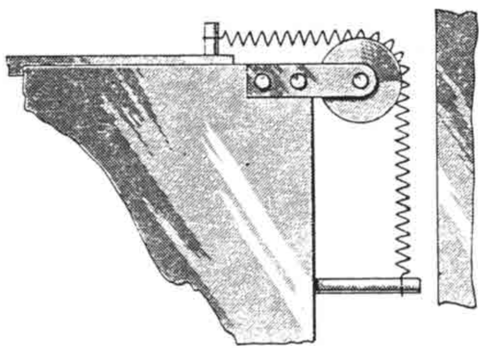
A close-wound spring is attached to a hopper, and it will not buckle when it is used as a movable feed-duct for nongranular material.



Toggle action here ensures that the gear-shift lever will not inadvertently be thrown past its neutral position.



Tension varies at a different rate when the brake-applying lever reaches the position shown. The rate is reduced when the tilting lever tilts.



The spring wheel helps to distribute deflection over more coils that if the spring rested on the corner. The result is less fatigue and longer life.

OVERRIDING SPRING MECHANISMS FOR LOW-TORQUE DRIVES

Overriding spring mechanisms are widely used in the design of instruments and controls. All of the arrangements illustrated allow an incoming motion to override the outgoing motion whose limit has been reached. In an instrument, for example, the spring mechanism can be placed between the sensing and indicating elements to provide overrange protection. The dial pointer is driven positively up to its limit before it stops while the input shaft is free to continue its travel. Six of the mechanisms described here are for rotary motion of varying amounts. The last is for small linear movements.

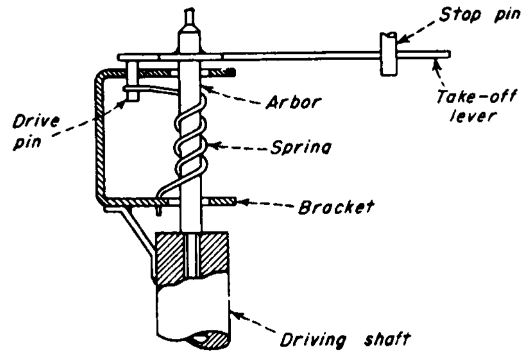


Fig. 1 Unidirectional override. The take-off lever of this mechanism can rotate nearly 360°. Its movement is limited only by one stop pin. In one direction, motion of the driving shaft is also impeded by the stop pin. But in the reverse direction the driving shaft is capable or rotating approximately 270° past the stop pin. In operation, as the driving shaft is turned clockwise, motion is transmitted through the bracket to the take-off lever. The spring holds the bracket against the drive pin. When the take-off lever has traveled the desired limit, it strikes the adjustable stop pin. However, the drive pin can continue its rotation by moving the bracket away from the drive pin and winding up the spring. An overriding mechanism is essential in instruments employing powerful driving elements, such as bimetallic elements, to prevent damage in the overrange regions.

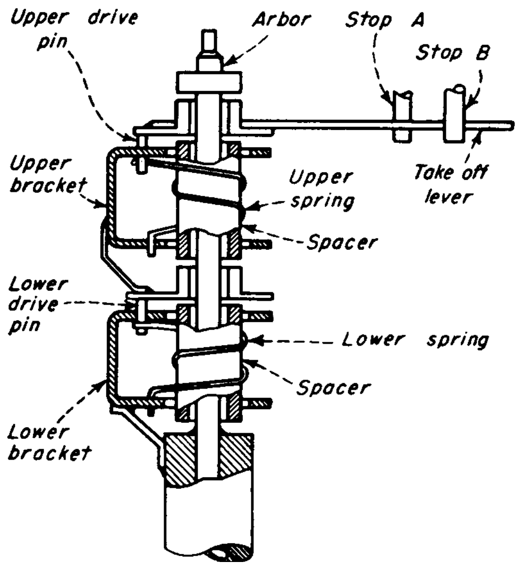


Fig. 2 Two-directional override. This mechanism is similar to that described under Fig. 1, except that two stop pins limit the travel of the take-off lever. Also, the incoming motion can override the outgoing motion in either direction. With this device, only a small part of the total rotation of the driving shaft need be transmitted to the take-off lever, and this small part can be anywhere in the range. The motion of the deriving shaft is transmitted through the lower bracket to the lower drive pin, which is held against the bracket by the spring. In turn, the lower drive pin transfers the motion through the upper bracket to the upper drive pin. A second spring holds this pin against the upper drive bracket. Because the upper drive pin is attached to the take-off lever, any rotation of the drive shaft is transmitted to the lever, provided it is not against either stop *A* or *B*. When the driving shaft turns in a counterclockwise direction, the take-off lever finally strikes against the adjustable stop *A*. The upper bracket then moves away from the upper drive pin, and the upper spring starts to wind up. When the driving shaft is rotated in a clockwise direction, the take-off lever hits adjustable stop *B,* and the lower bracket moves away from the lower drive pin, winding up the other spring. Although the principal applications for overriding spring arrangements are in instrumentation, it is feasible to apply these devices in the drives of heavy-duty machines by strengthening the springs and other load-bearing members.

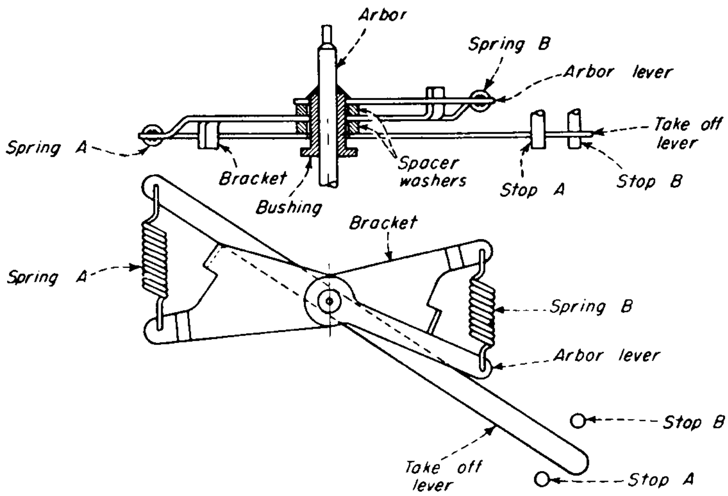


Fig. 3 Two-directional, limited-travel override. This mechanism performs the same function as that shown in Fig. 2, except that the maximum override in either direction is limited to about 40°**.** By contrast, the unit shown in Fig. 2 is capable of 270° movement. This device is suited for applications where most of the incoming motion is to be used, and only a small amount of travel past the stops in either direction is required. As the arbor is rotated, the motion is transmitted through the arbor lever to the bracket.. The arbor lever and the bracket are held in contact by spring *B*. The motion of the bracket is then transmitted to the take-off lever in a similar manner, with spring *A* holding the take-off lever until the lever engages either stops *A* or *B*. When the arbor is rotated in a counterclockwise direction, the take-off lever eventually comes up against the stop *B*. If the arbor lever continues to drive the bracket, spring *A* will be put in tension.

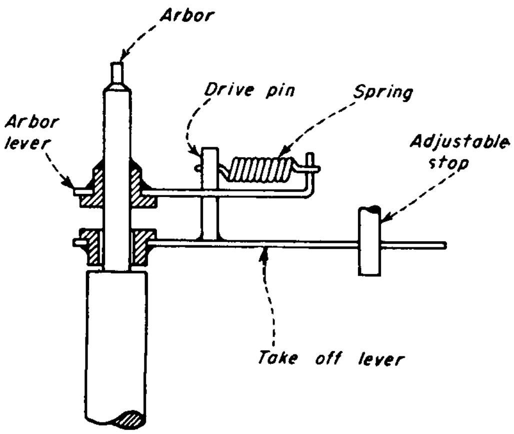


Fig. 4 Unidirectional, 90° override. This is a single overriding unit that allows a maximum travel of 90° past its stop. The unit, as shown, is arranged for overtravel in a clockwise direction, but it can also be made for a counterclockwise override. The arbor lever, which is secured to the arbor, transmits the rotation of the arbor to the take-off lever. The spring holds the drive pin against the arbor lever until the take-off lever hits the adjustable stop. Then, if the arbor lever continues to rotate, the spring will be placed in tension. In the counterclockwise direction, the drive pin is in direct contact with the arbor lever so that no overriding is possible.

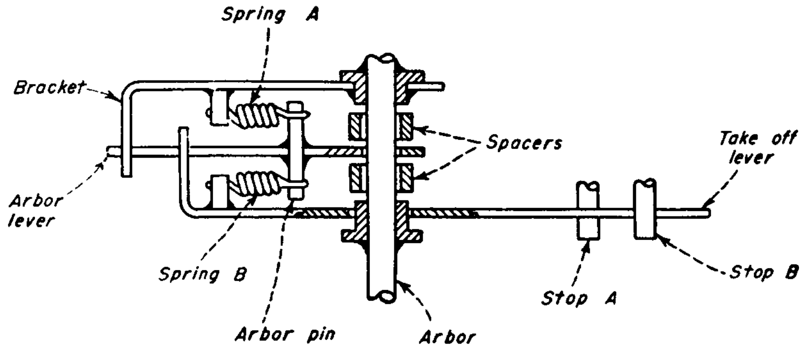


Fig. 5 Two-directional, 90° override. This double-overriding mechanism allows a maximum overtravel of 90° in either direction. As the arbor turns, the motion is carried from the bracket to the arbor lever, then to the take-off lever. Both the bracket and the take-off lever are held against the arbor lever by spring *A* and *B*. When the arbor is rotated counterclockwise, the takeoff lever hits stop *A*. The arbor lever is held stationary in contact with the take-off lever. The bracket, which is fastened to the arbor, rotates away from the arbor lever, putting spring *A* in tension. When the arbor is rotated n a clockwise direction, the take-off lever comes against stop *B*, and the bracket picks up the arbor lever, putting spring *B* in tension.

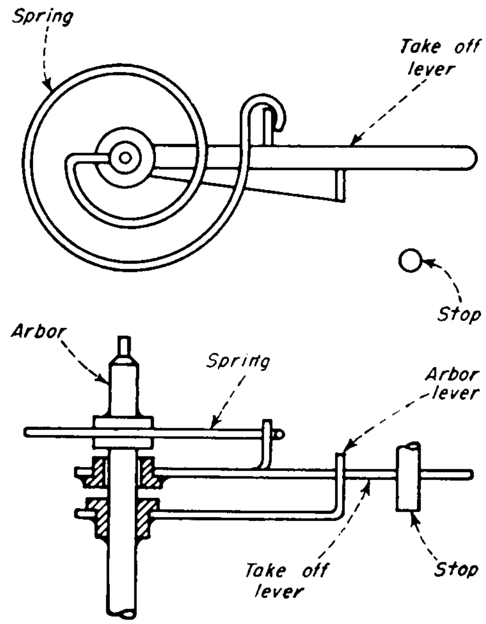


Fig. 6 Unidirectional, 90° override. This mechanism operates exactly the same as that shown in Fig. 4. However, it is equipped with a flat spiral spring in place of the helical coil spring used in the previous version. The advantage of the flat spiral spring is that it allows for a greater override and minimizes the space required. The spring holds the take-off lever in contact with the arbor lever. When the take-off lever comes in contact with the stop, the arbor lever can continue to rotate and the arbor winds up the spring.

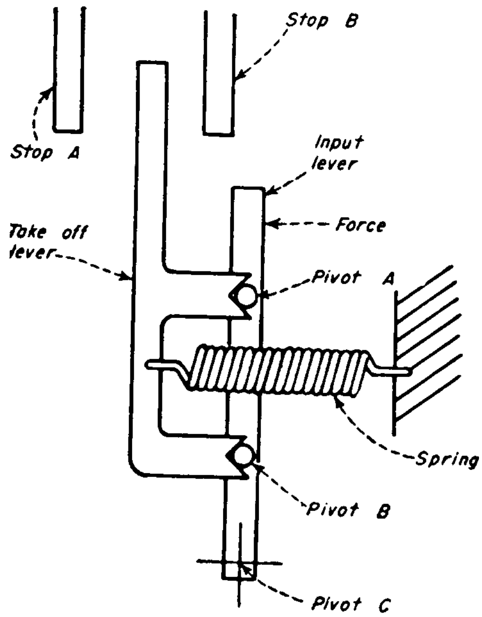


Fig. 7 Two-directional override, linear motion. The previous mechanisms were overrides for rotary motion. The device in Fig. 7 is primarily a double override for small linear travel, although it could be used on rotary motion. When a force is applied to the input lever, which pivots about point *C*, the motion is transmitted directly to the take-off lever through the two pivot posts, *A* and *B*. The take-off lever is held against these posts by the spring. When the travel causes the take-off lever to hit the adjustable stop *A*, the take-off lever revolves about pivot post *A*, pulling away from pivot post *B*, and putting additional tension in the spring. When the force is diminished, the input lever moves in the opposite direction until the take-off lever contacts the stop *B*. This causes the take-off lever to rotate about pivot post *B*, and pivot post *A* is moved away from the take-off lever.

SPRING MOTORS AND TYPICAL ASSOCIATED MECHANISMS

Many applications of spring motors in clocks, motion picture cameras, game machines, and other mechanisms offer practical ideas for adaptation to any mechanism that is intended to operate for an appreciable length of time. While spring motors are usually limited to comparatively small power application where other sources of power are unavailable or impracticable, they might also be useful for intermittent operation requiring comparatively high torque or high speed, using a low-power electric motor or other means for building up energy.

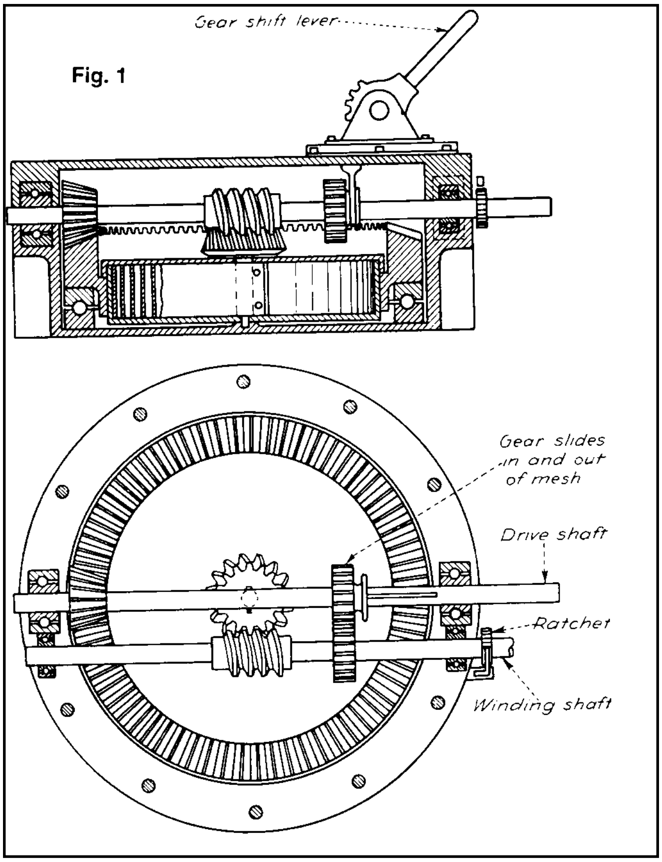


Fig. 1

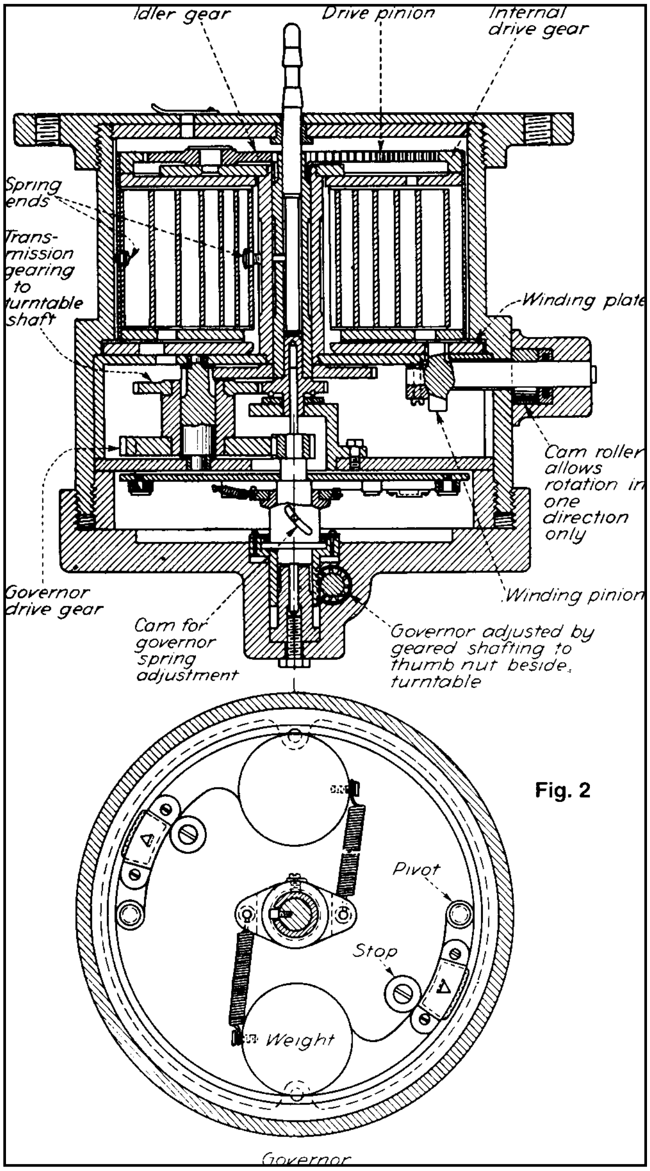


Fig. 2

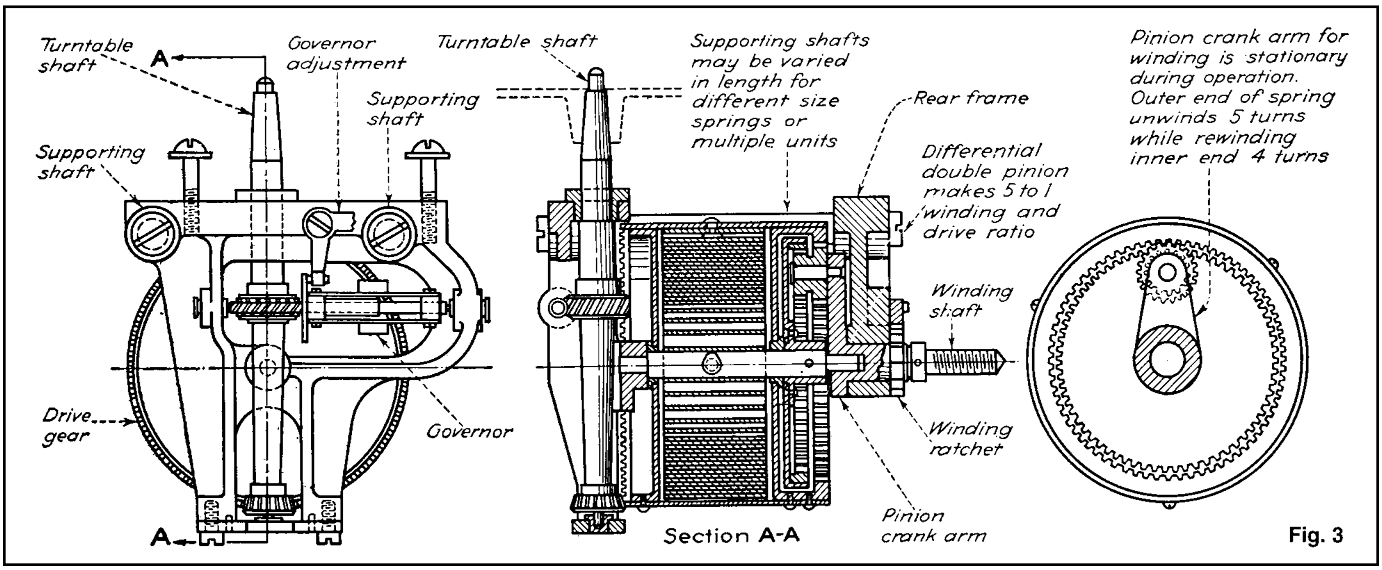


Fig. 3

The accompanying patented spring motor designs show various methods for the transmission and control of spring-motor power. Flat-coil springs, confined in drums, are most widely used because they are compact, produce torque directly, and permit long angular displacement. Gear trains and feedback mechanisms reduce excess power drain so that power can be applied for a longer time. Governors are commonly used to regulate speed.

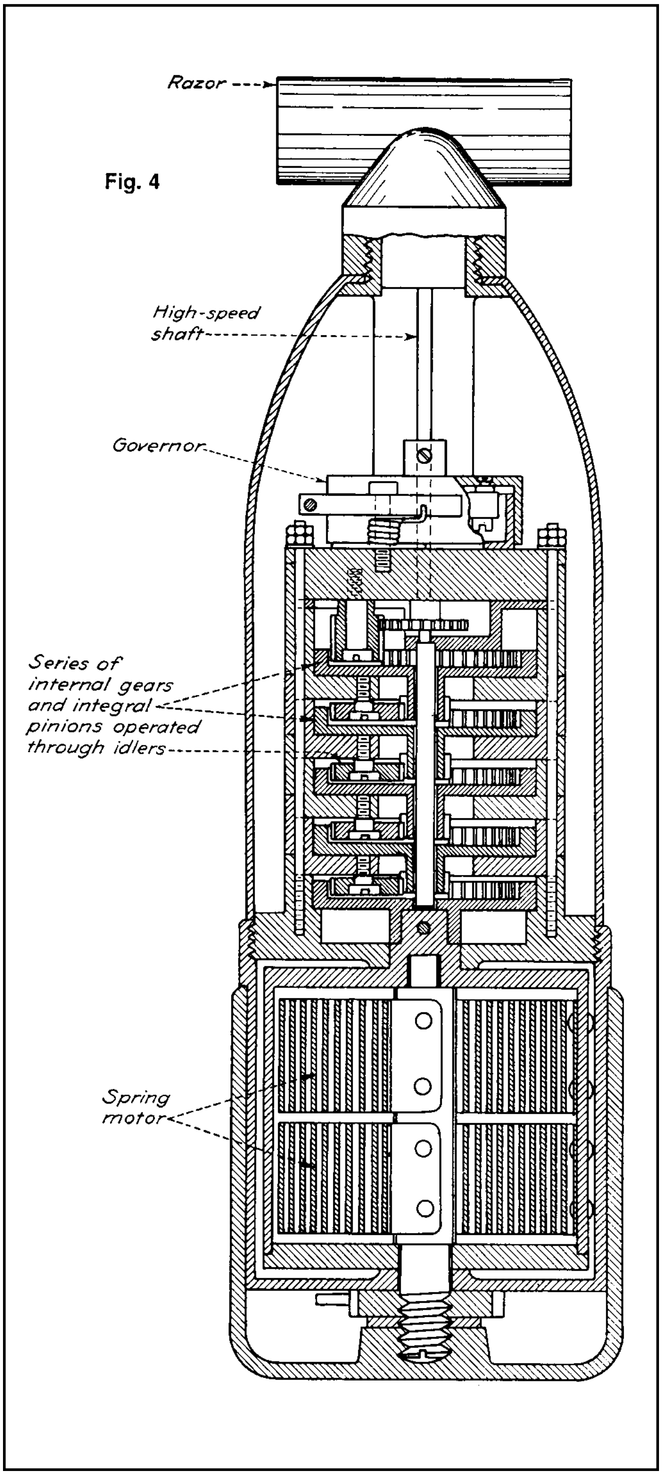


Fig. 4

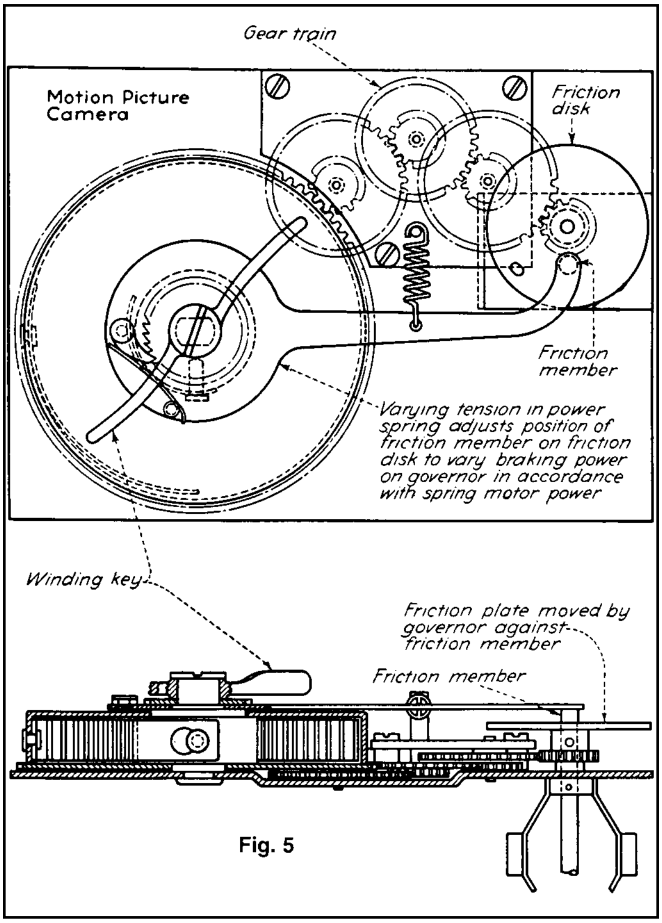


Fig. 5

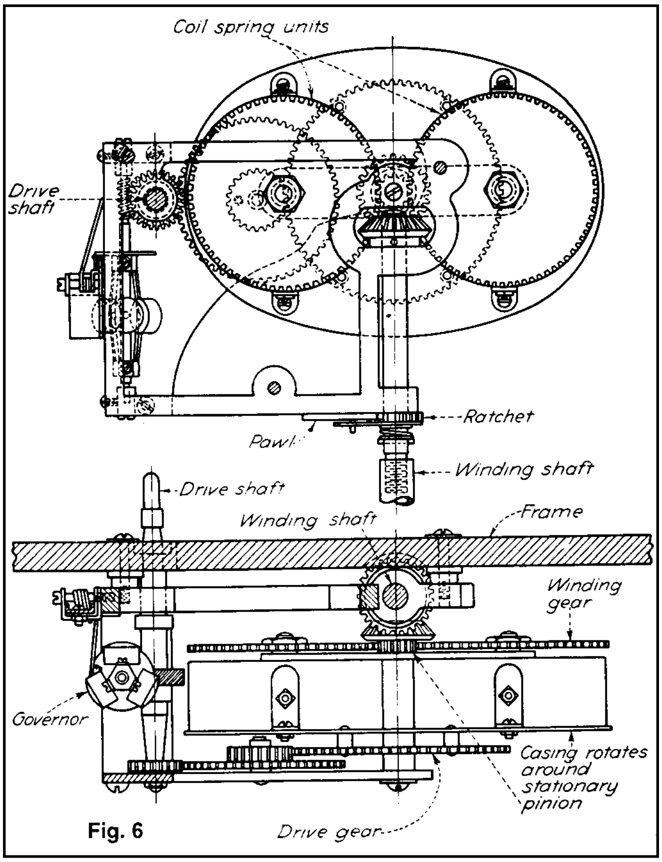


Fig. 6

FLEXURES ACCURATELY SUPPORT PIVOTING MECHANISMS AND INSTRUMENTS

Flexures, often bypassed by various rolling bearing, have been making steady progress—often getting the nod for applications in space and industry where their many assets outweigh the fact that they cannot give the full rotation that bearings offer.

Flexures, or flexible suspensions as they are usually called, lie between the worlds of rolling bearings—such as the ball and roller bearings—and of sliding bearings—which include sleeve and hydrostatic bearings. Neither rolling nor sliding, flexures simply cross-suspend a part and flex to allow the necessary movement.

There are many applications for parts of components that must reciprocate or oscillate, so flexure are becoming more readily available as the off-the-shelf part with precise characteristics.

Flexures for space. Flexures have been selected over bearings in space applications because they do not wear out, have simpler lubrication requirements, and are less subject to backlash.

One aerospace flexure—scarcely more than 2 in. high—was used for a key task on the Apollo Applications Program (AAP), in which Apollo spacecraft and hardware were employed for scientific research. The flexures' job was to keep a 5000-lb telescope pointed at the sun with unprecedented accuracy so that solar phenomena could be viewed.

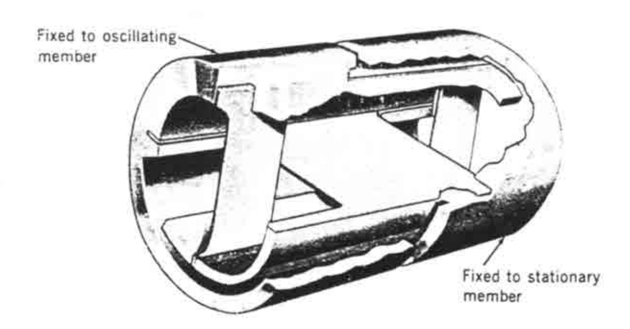
The flexure pivot selected contained thin connecting beams that had flexing action so they performed like a combination spring and bearing.

Unlike a true bearing, however, it had no rubbing surfaces. Unloaded, or with a small load, a flexure pivot acts as a positive—or center-seeking—spring; loaded above a certain amount, it acts as a negative spring.

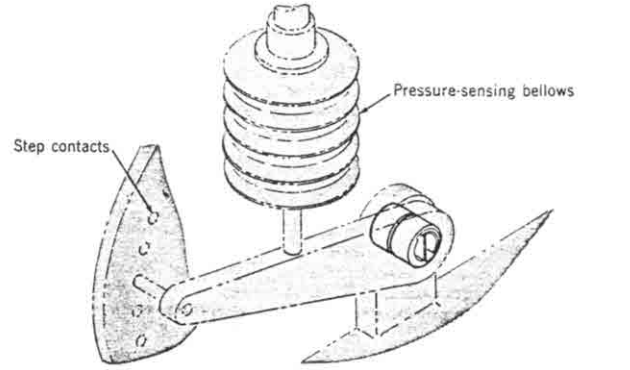
A consequence of this duality is that in space, the AAP telescope always returned to a central position, while during ground testing it drifted away from center. The Lockheed design took advantage of this phenomenon of flexure pivots: By attaching a balancing weight to the telescope during ground tests, Lockheed closely simulated the dynamic conditions of space.

Potential of flexures. Lockheed adapted flexure pivots to other situations as well. In one case, a flexure was used for a gimbal mount in a submarine. Another operated a safety shutter to protect delicate sensors in a satellite.

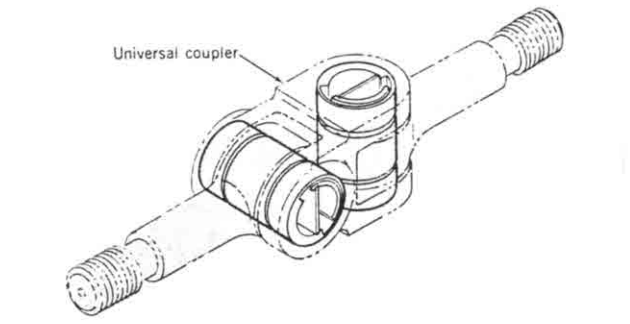
Realizing the potential of flexure pivots, Bendix Corp. (Utica, N.Y.) developed an improved type of bearing flexure, commonly known as "flexure pivot." It was designed to be compliant around one axis and rigid around the cross axes. The flexure pivots have the same kind of flat, crossed springs as the rectangular kind, but they were designed as a simple package that could be easily



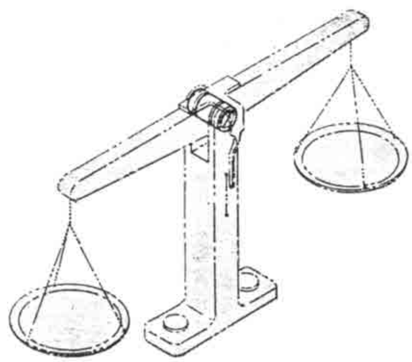
A frictionless flexure pivot, which resembles a bearing, is made of flat, angular crossed springs that support rotating sleeves in a variety of structural designs.



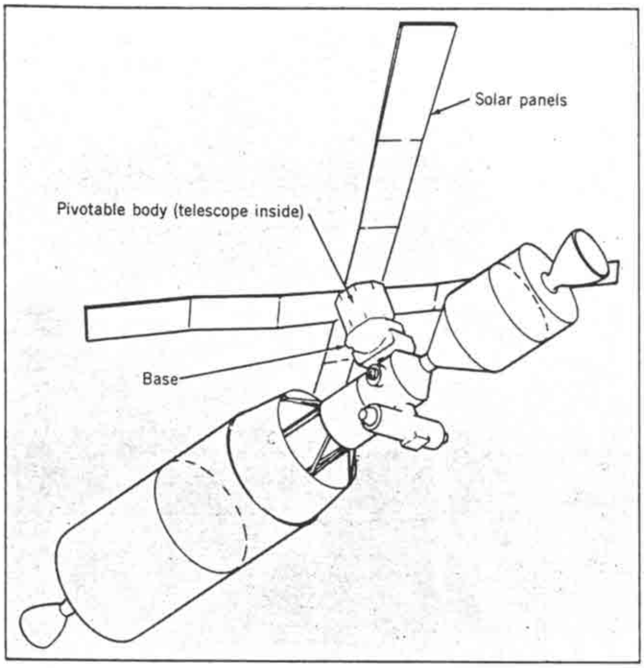
A pressure transducer with a flexure pivot can oscillate 30º to translate the movements of bellows expansion and contraction into electrical signals.

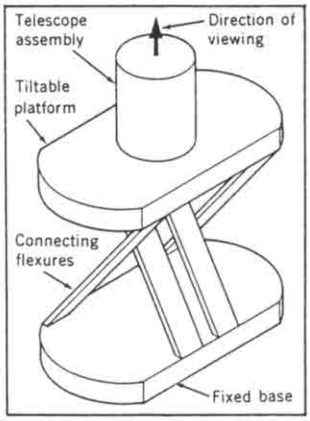


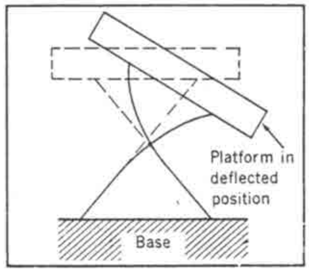
A universal joint has flexure pivots so there is no need for lubrication. There is also a two-directional pivot made with integral housing.

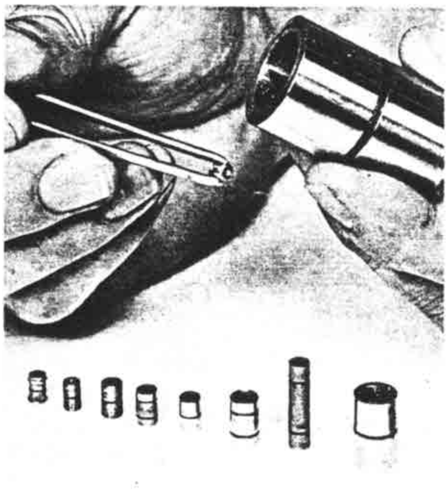


A balance scale substitutes flexure pivots in place of a knife edge, which can be affected by dirt, dust, and sometimes even by the lubricants themselves.









The Apollo telescope-mount cluster (top left) had flexures for tilting an X-ray telescope. The platform (top right) is tilted without break-away torque. The photo above shows typical range of flexure sizes.

installed and integrated into a design (see photo). The compactness of the flexure pivot make it suitable to replace ordinary bearings in many oscillating applications (see drawings).

The Bendix units were built around three elements: flexures, a core or inner housing, and an outer housing or mounting case. They permit angular deflections of 7½°, 15°, or 30°.

The cantilever type (see drawing) can support an overhung load. There is also a double-ended kind that supports central loads. The width of each cross member of the outer flexure is equal to one-half that of the inner flexure, so that when assembled at 90° from each other, the total flexure width in each plane is the same.

Key point. The heart of any flexure pivot is the flexure itself.

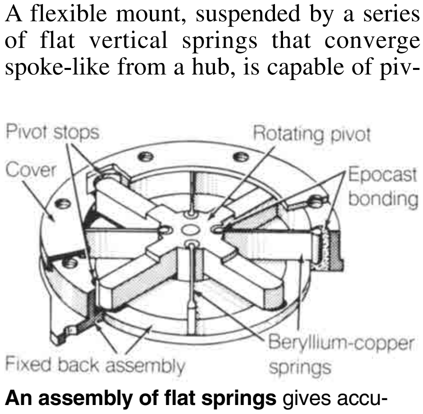
A key factor in applying a flexure is the torsional-spring constant of the assembly—in other words, the resisting restoring torque per angle of twist, which can be predicted from the following equation:

$$K = C\frac{NEbt^3}{12L}$$

where K = spring constant, in.-lb/deg
N = number of flexures of width b
E = modulus of elasticity, lb/in.^{2}
b = flexure width, in.
t = flexure thickness, in.
L = flexure length, in.
C = summation of constants resulting from variations in tolerances and flexure shape.

Flat Springs Serve as a Frictionless Pivot

A flexible mount, suspended by a series of flat vertical springs that converge spoke-like from a hub, is capable of pivoting through small angles without any friction. The device, developed by C. O. Highman of Ball Bros. Research Corp. under contract to Marshall Space Flight Center, Huntsville, Ala., is also free of any hysteresis when rotated (it will return exactly to its position before being pivoted). Moreover, its rotation is smooth and linearly proportional to torque.



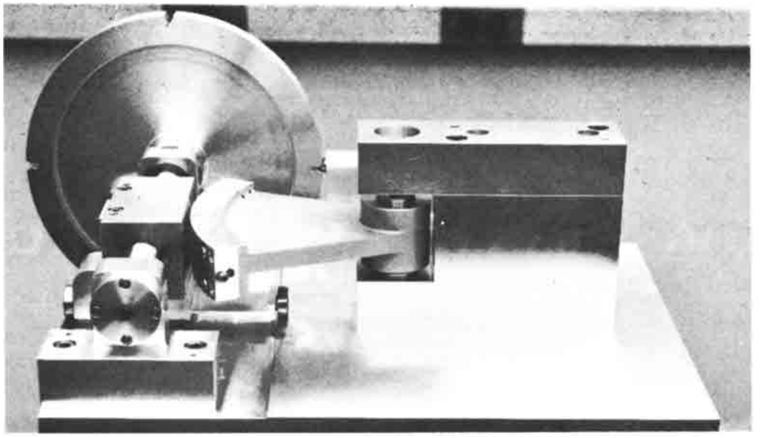
An assembly of flat springs gives accurate, smooth pivoting with no starting friction.

The pivot mount, which in a true sense acts as a pivot bearing without need for any lubrication, was developed with the aim of improving the pointing accuracies of telescopes, radar antennas, and laser ranging systems. It has other interesting potential applications, however. When the pivot mount is supported by springs that have different thermal expansion coefficients, for example, heat applied to one spring segment produces an angular rotation independent of external drive.

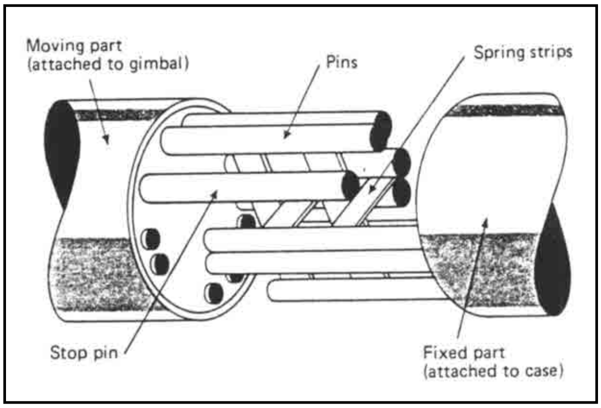
Flexing springs. The steel pivot mount is supported by beryllium-copper springs attached to the outer frame. Stops limit the thrust load. The flexure spring constant is about 4 ft-lb/radian.

The flexible pivot mount can be made in tiny sizes, and it can be driven by a dc torque motor or a mechanical linkage. In general, the mount can be used in any application requiring small rotary motion with zero chatter.

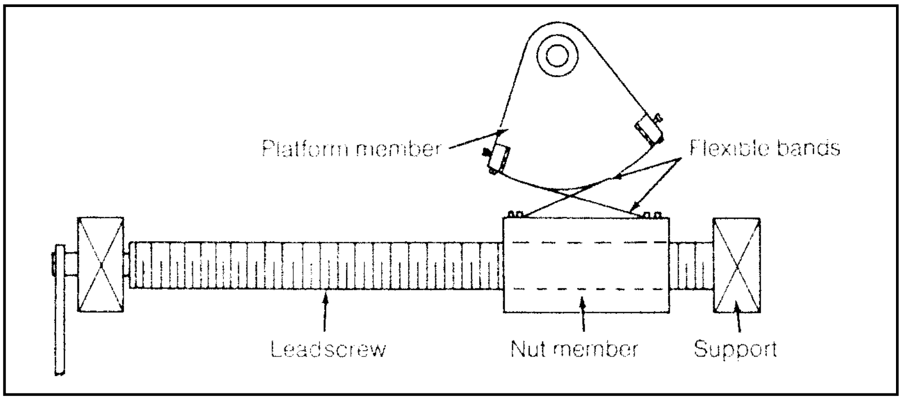
TAUT BANDS AND LEADSCREW PROVIDE ACCURATE ROTARY MOTION



Flexible bands substitute for a worm gear in a precisely repeatable rotary mechanism used as a star tracker. The tracker instrumentation, mounted on the platform, is rotated by an input motion to the leadscrew.



A flexure pivot boasts high mechanical stability for use in precision instruments.



A pair of opposed, taut, flexible bands in combination with a leadscrew provides an extremely accurate technique for converting rotary motion in one plane to rotary motion in another plane. Normally, a worm-gear set would be employed for such motion. The technique, however, developed by Kenneth G. Johnson of Jet Propulsion Laboratory, Pasadena, California, under a NASA sponsored project, provided repeatable, precise positioning within two seconds of an arc for a star tracker mechanism (drawing, photo).

Crossed bands. In the mechanism, a precision-finished leadscrew and a fitted mating nut member produce linear translatory motion. This motion is then transformed to a rotary movement of a pivotal platform member. The transformation was achieved by coupling the nut member and the platform member through a pair of crossed flexible phosphor-bronze bands.

The precision leadscrew is journaled at its ends in the two supports.

With the bands drawn taut, the leadscrew is rotated to translate the nut member. The platform member will be drawn about its pivot without any lost motion or play. Because the nut member is accurately fitted to the leadscrew, and because precision-ground leadscrews have a minimum of lead error, the uniform linear translation produced by rotation of the lead screw resulted in a uniform angular rotation of the platform member.

Points on the radial periphery of the sector are governed by the relationship $S = R\Theta$, which means that rotation is directly proportional to distance as measured at the circumference. The nut that translates on the leadscrew was directly related to the rotary input because the leadscrew was accurately ground and lapped. Also, 360° of rotation of the leadscrew translates the saddle nut a distance of one thread pitch. This translation result in rotation of the sector through an angle equal to S/R.

The relationship is true at any point within the operating rang of the instrument, provided that R remains constant. Two other necessary conditions for maintaining relationship are that the saddle nut be constrained against rotation, and that there be a zero gap between sector and saddle nut.

Pivots with a Twist

A multipin flexure-type pivot, developed by Smiths Industries in England, combined high radial and axial stiffness with the inherent advantages of a cross-spring pivot—which it is.

The pivot provides non-sliding, non-rolling radial and axial support without the need for lubrication. The design combines high radial and axial stiffness with a relatively low and controlled angular stiffness. Considerable attention was given to solving the practical problems of mounting the pivot in a precise and controlled way.

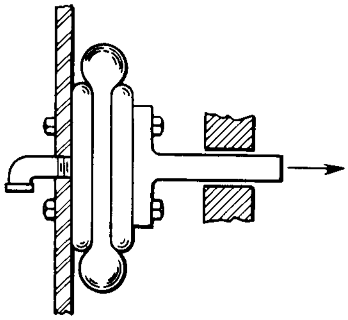
The finished pivot is substantially free from residual mechanical stress to achieve stability in service. Maraging steel is used throughout the assembly to avoid any differential expansion due to material mismatch. The blades of the flexure pivot are free from residual braze t o avoid any bimetallic movements when the temperature of the pivot changes.

The comparatively open construction of the pivot made it less susceptible to jamming caused by any loose particles. Furthermore, the simple geometric arrangement of the support pins and flexure blade allowed blade anchor points to be defined with greater accuracy. The precision ground integral mounting flanges simplified installation.

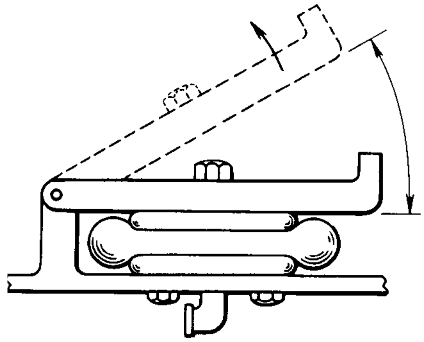
Advantages, according to its designer, include frictionless, stictionless and negligible hysteresis characteristics. The bearing is radiation-resistant and can be used in high vacuum conditions or in environments where there is dirt and contamination.

AIR SPRING MECHANISMS

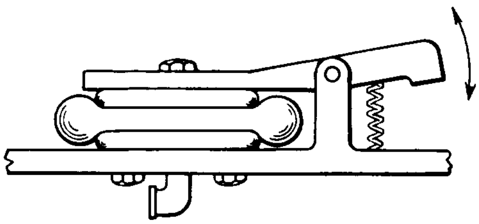
EIGHT WAYS TO ACTUATE MECHANISMS WITH AIR SPRINGS



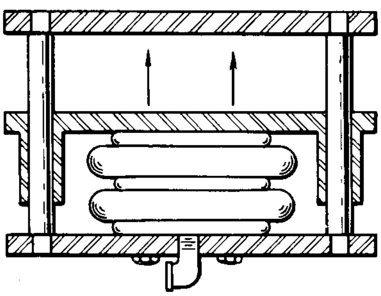
Linear force link: A one- or two-convolution air spring drives the guide rod. The rod is returned by gravity, opposing force, metal spring or, at times, internal stiffness of an air spring.



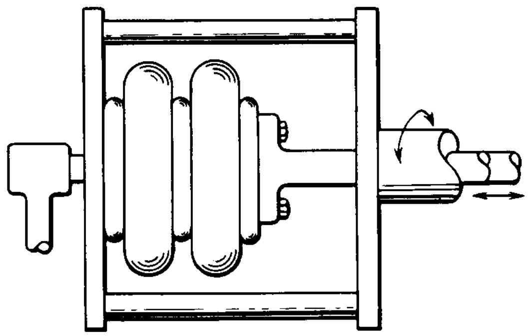
Rotary force link: A pivoted plate can be driven by a one-convolution or two-convolution spring to 30° of rotation. The limitation on the angle is based on permissible spring misalignment.



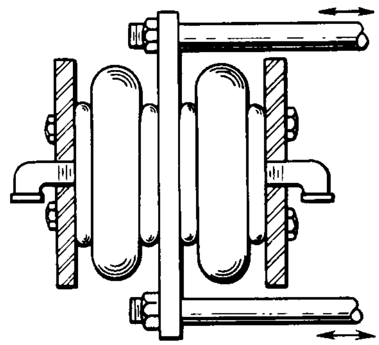
Clamp: A jaw is normally held open by a metal spring. Actuation of the air spring then closes the clamp. The amount of opening in the jaws of the clamp can be up to 30° of arc.



Direct-acting press: One-, two-, or three-convolution air springs are assembled singly or in gangs. They are naturally stable when used in groups. Gravity returns the platform to its starting position.



Rotary shaft actuator: The activator shifts the shaft longitudinally while the shaft is rotating. Air springs with one, two, or three convolutions can be used. A standard rotating-air fitting is required.

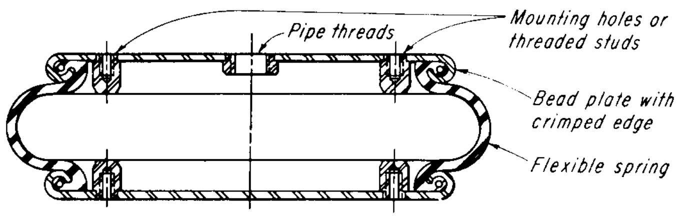


Reciprocating linear force link: It reciprocates with one-, two-, or three-convolution air springs in a back-to-back arrangement. Two- and three-convolution springs might need guides for their force rods.

POPULAR TYPES OF AIR SPRINGS

Air is an ideal load-carrying medium. It is highly elastic, its spring rate can be easily varied, and it is not subject to permanent set.

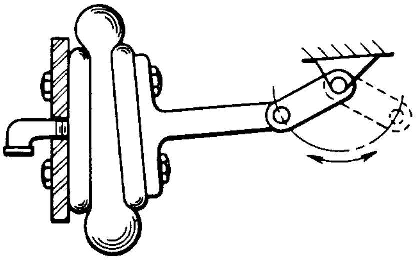
Air springs are elastic devices that employ compressed air as the spring element. They maintain a soft ride and a constant vehicle height under varying load. In industrial applications they control vibration (isolate or amplify it) and actuate linkages to provide either rotary or linear movement. Three kinds of air springs (bellows, rolling sleeve, and rolling diaphragm) are illustrated.



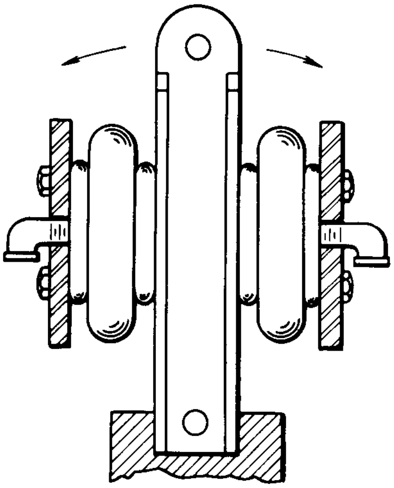
ONE-CONVOLUTION BELLOW

Bellows Type

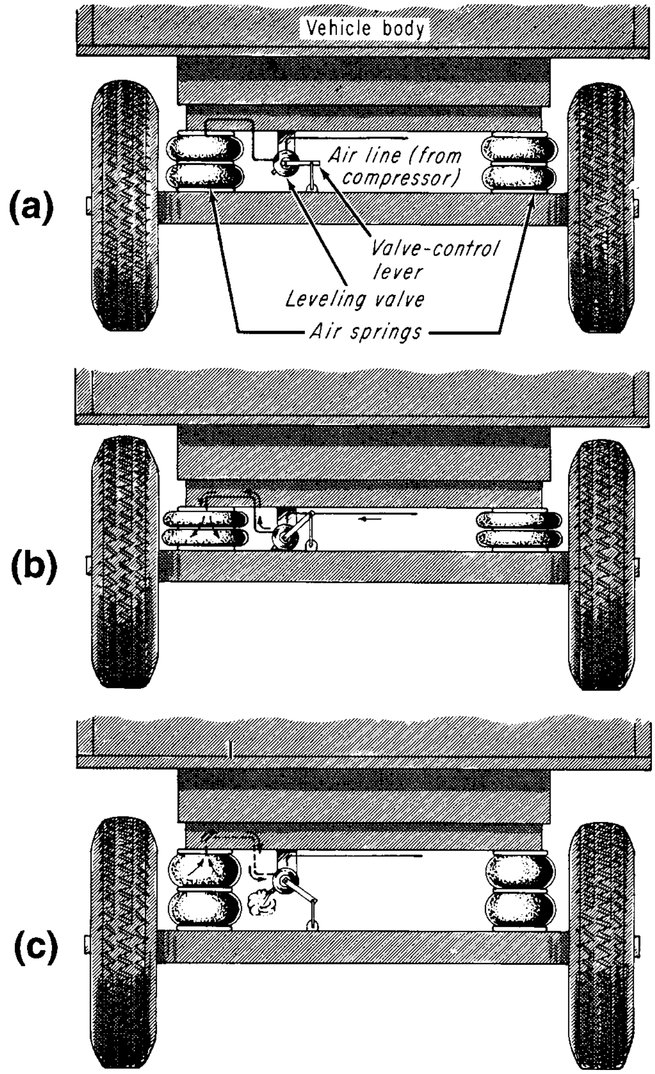
A single-convolution spring looks like a tire lying on its side. It has a limited stroke and a relatively high spring rate. Its natural frequency is about 150 cpm without auxiliary volume for most sizes, and as high as 240 cpm for the smallest size. Lateral stiffness is high (about half the vertical rate); therefore the spring is quite stable laterally when used for industrial vibration isolation. It can be filled manually or kept inflated to a constant height if is connected to factory air



Pivot mechanism: It rotates a rod through 145° of rotation. It can accept a 30° misalignment because of the circular path of its connecting-link pin. A metal spring or opposing force retracts the link.



Reciprocating rotary motion with one-convolution and two-convolution springs. An arc up to 30° is possible. It can pair a large air spring with a smaller one or a lengthen lever.



Air suspension on vehicle: A view of normal static conditions—air springs at desired height and height-control valve closed (a). When a load is added to the vehicle—the valve opens to admit air to the springs and restore height, but at higher pressure (b). With load removed from the vehicle—valve permits bleeding off excess air pressure to atmosphere and restores its design height (c).

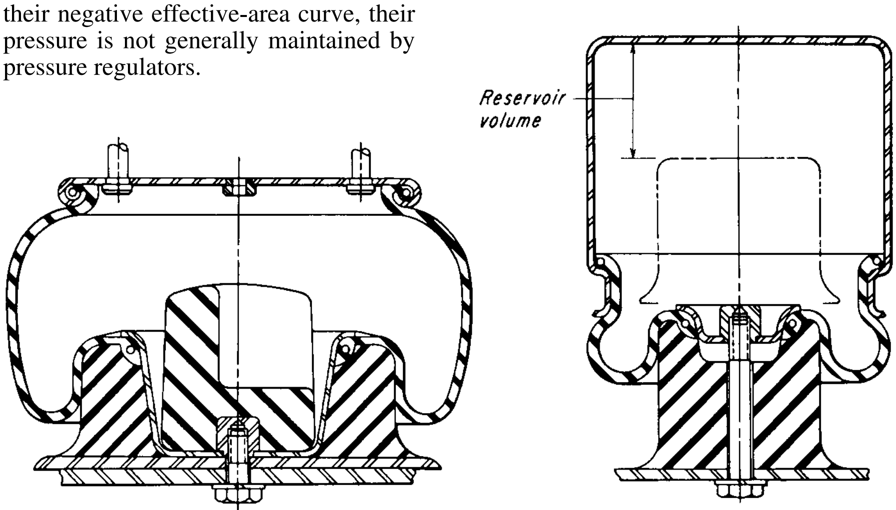
supply through a pressure regulator. This spring will also actuate linkages where short axial length is desirable. It is seldom used in vehicle suspension systems.

Rolling-Sleeve Type

This spring is sometimes called the reversible-sleeve or rolling-lobe type. It has a telescoping action—the lobe at the bottom of the air spring rolls up and down along the piston. The spring is used primarily in vehicle suspensions because lateral stiffness is almost zero.

Rolling-Diaphragm Type

These are laterally stable and can be used as vibration isolators, actuators, or constant-force spring. But because of their negative effective-area curve, their pressure is not generally maintained by pressure regulators.

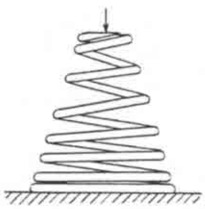


ROLLING-SLEEVE

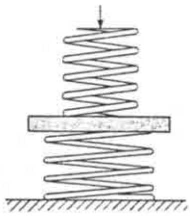
OBTAINING VARIABLE RATES FROM SPRINGS

How stops, cams, linkages, and other arrangements can vary the load/deflection ratio during extension or compression

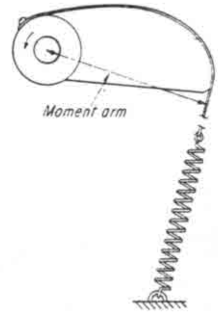
With tapered-pitch spring the number of effective coils changes with deflection—the coils "bottom" progressively.



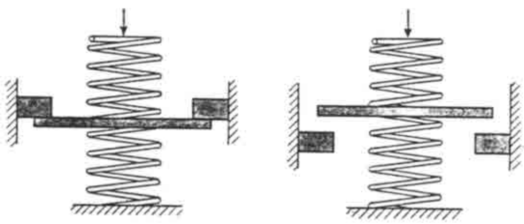
A tapered outside diameter and pitch combine to produce a similar effect except that the spring with tapered O.D. will have a shorter solid height.



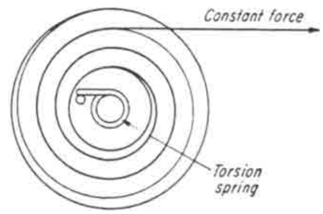
In dual springs, one spring closes completely before the other.



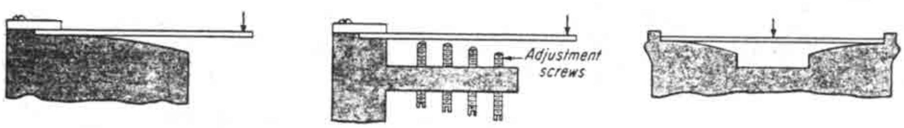
A cam-and-spring device causes the torque relationship to vary during rotation as the moment arm changes.



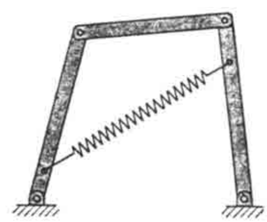
Stops can be used with either compression or extension springs.



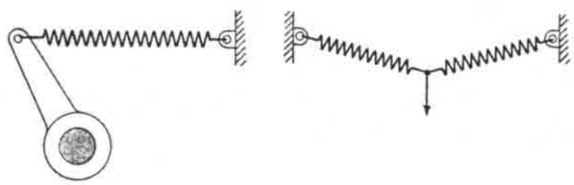
Torsion spring combined with a variable-radius pulley gives a constant force.



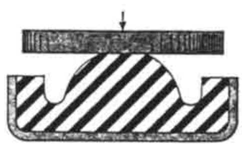
Leaf springs can be arranges so that their effective lengths change with deflection.



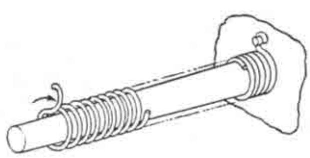
A four-bar mechanism in conjunction with a spring has a wide variety of load/deflection characteristics.



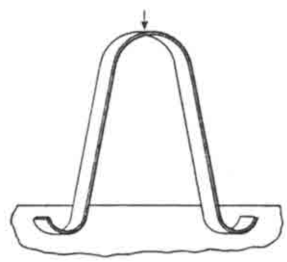
These linkage-type arrangements are used in instruments where torque control or anti-vibration suspension is required.



A molded-rubber spring has deflection characteristics that vary with its shape.



With a tapered mandrel and torsion spring the effective number or coils decreases with torsional deflection.



An arched leaf-spring gives an almost constant force when it is shaped like the one illustrated.

BELLEVILLE SPRINGS

Belleville springs are low-profile conical rings with differing height (h) to thickness (t) ratios, as shown in Fig. 1. Four way to stack them are shown in Fig. 2.

Belleville springs lend themselves to a wide variety; of applications:

For height to spring ratios of about 0.4—A linear spring rate and high load resistance with small deflections.

For height to spring ratios between 0.8 and 1.0—An almost linear spring rate for fasteners and bearing and in stacks.

For rations of around 1.6—A constant (flat) spring rate starting at about 60% of the deflection (relative to the fully compressed flat position) and proceeding to the flat position and, if desired, on to the flipped side to a deflection of about 140%. In most applications, the flat position is the limit of travel, and for deflections beyond the flat, the contact elements must be allowed unrestricted travel

One application of bellevilles with constant spring rate is on live spindles on the tailpiece of a lathe. The work can be loaded on the lathe, and as the piece heats up and begins to expand, the belleville will absorb this change in length without adding any appreciable load.

For high height to spring ratios exceeding about 2.5—The spring is stiff, and as the stability point (high point on the curve) is passed the spring rate becomes negative causing resistance to drop rapidly. If allowed, the belleville will snap through the flat position. In other words, it will turn itself inside out.

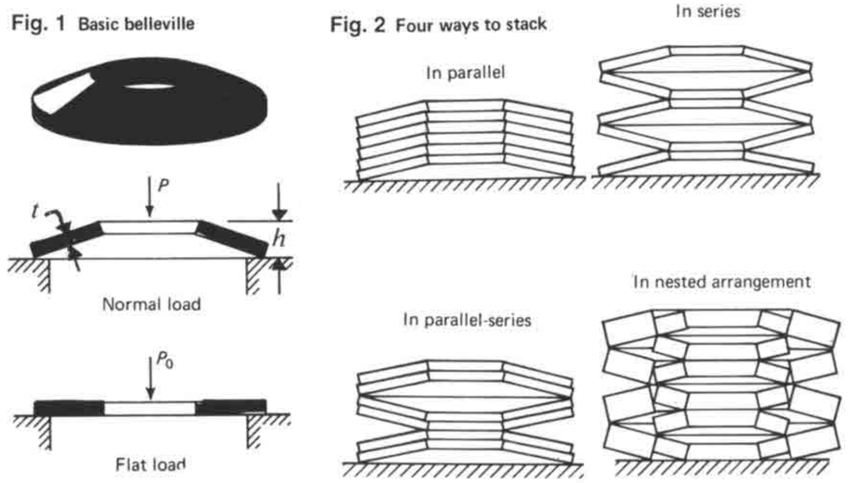
Working in groups. Belleville washers stacked in the parallel arrangement have been used successfully in a variety of applications.

One is a pistol or rifle buffer mechanism (Fig. 3) designed to absorb repeated, high-energy shock loads. A preload nut predeflects the washers to stiffen their resistance. The stacked washers are guided by a central shaft, an outside guide cylinder, guide rings, or a combination of these.

A wind-up starter mechanism for diesel engines (shown in Fig. 4) replaces a heavy-duty electric starter or auxiliary gas engine. To turn over the engine, energy is manually stored in a stack of bellevilles compressed by a hand crank. When released, the expanding spring pack rotates a pinion meshed with the flywheel ring gear to start the engine.

Figure 5 shows a belleville as a loading spring for a clutch.

Popular arrangements



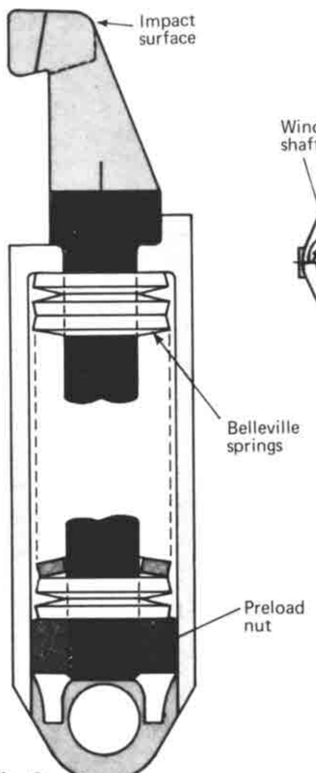


Fig. 3 Buffer mechanism for high-impact energy

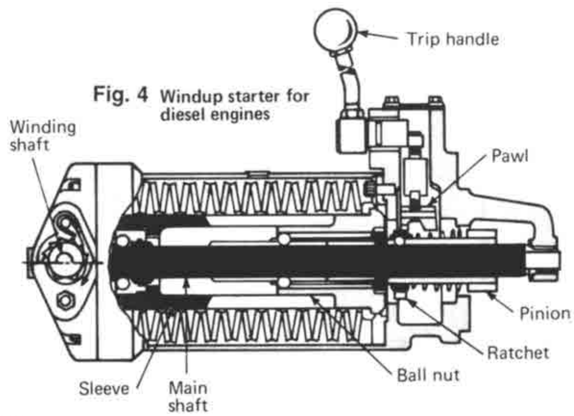


Fig. 4 Windup starter for diesel engines

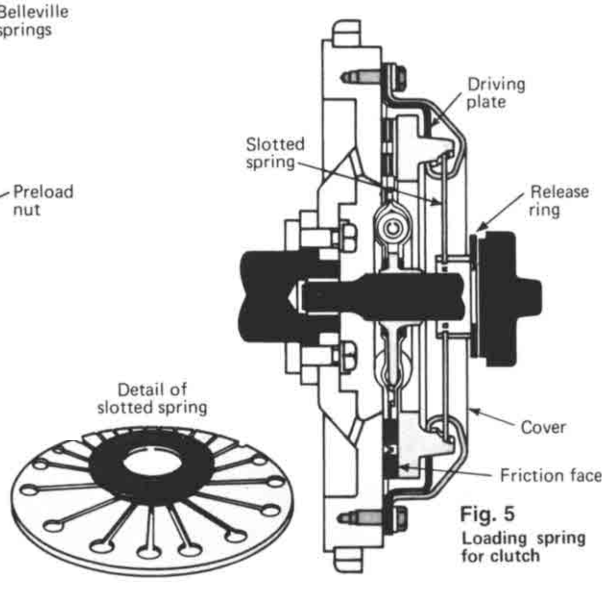


Fig. 5 Loading spring for clutch

SPRING-TYPE LINKAGE FOR VIBRATION CONTROL

Do you need a buffer between vibrating machinery and the surrounding structure? These isolators, like capable fighters, absorb the light jabs and stand firm against the forces that inflict powerful haymakers

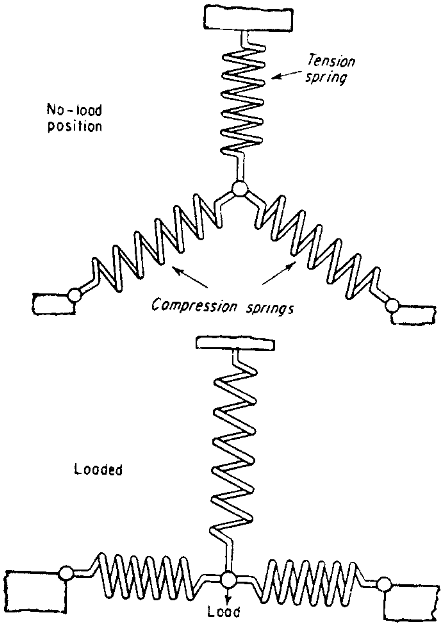


Fig. 1 This basic spring arrangement has zero stiffness, and is as "soft as a cloud" when compression springs are in line, as illustrated in the loaded position. But change the weight or compression-spring alignment, and stiffness increases greatly. This support is adequate for vibration isolation because zero stiffness give a greater range or movement than the vibration amplitude generally in the hundredths-of-an-inch range.

Arrangements shown here are highly absorbent when required, yet provide a firm support when large force changes occur. By contrast, isolators that depend upon very "soft" springs, such as the sine spring, are unsatisfactory in many applications; they allow a large movement of the supported load with any slight weight change or large-amplitude displacing force.

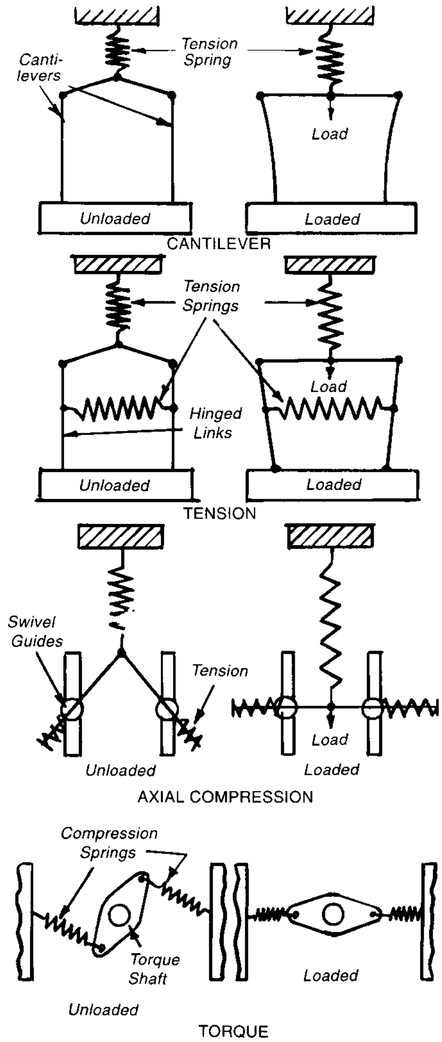


Fig. 2 Alternative arrangements illustrate adaptability of basic design. Here, instead of the inclined, helical compression springs, wither tension or cantilever springs can serve. Similarly, different type of springs can replace the axial, tension spring. Zero torsional stiffness can also be provided.

Various applications of the principle of vibration isolation show how versatile the design is. Coil spring (Fig. 4) as well as cantilever and torsion-bar suspension of automobiles can all be reduced in stiffness by adding an inclined spring; stiffness of the tractor seat (Fig. 5) and, consequently, transmitted shocks can be similarly reduced. Mechanical tension meter (Fig. 6) provides a sensitive indication of small variations in tension. A weighing scale, for example, could detect small variations in nominally identical objects. A nonlinear torque mete (Fig. 7) provides a sensitive indication of torque variations about a predetermined level.

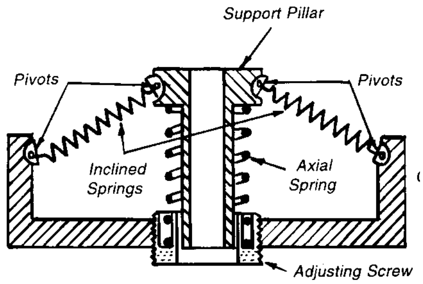


Fig. 3 A general-purpose support is based on basic spring arrangement, except that an axial compression spring is substituted for a tension spring. Inclined compression springs, spaced around a central pillar, carry the component to be isolated. When a load is applied, adjustment might be necessary to bring the inclined springs to zero inclination. Load range that can be supported with zero stiffness on a specific support is determined by the adjustment range and physical limitations of the axial spring.

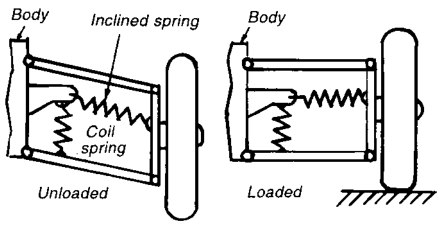


Fig. 4

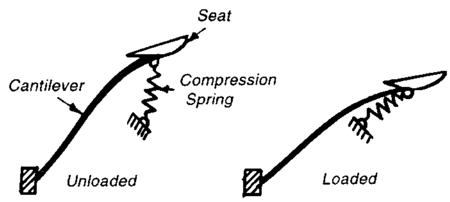


Fig. 5

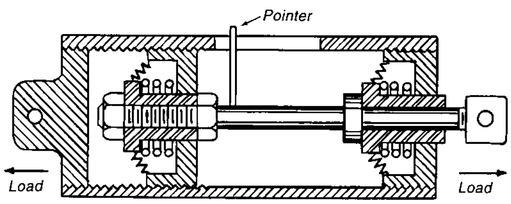


Fig. 6

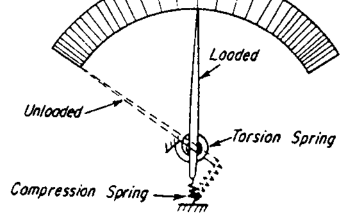


Fig. 7

TWENTY SCREW DEVICES

A threaded shaft and a nut plus some way to make one of these members rotate without translating and the other to translate without rotating are about all you need to do practically all of the adjusting, setting, or locking in a machine design.

Most of these applications have low-precision requirements. That's why the thread might be a coiled wire or a twisted strip; the nut might be a notched ear on a shaft or a slotted disk. Standard screws and nuts from hardware store shelves can often serve at very low cost.

Here are the basic motion transformations possible with screw threads (Fig. 1):

- Transform rotation into linear motion or reverse (A),
- Transform helical motion into linear motion or reverse (B),
- Transform rotation into helical motion or reverse (C).

Of course the screw thread can be combined with other components: in a four-bar linkage (Fig. 2), or with multiple screw elements for force or motion amplification.

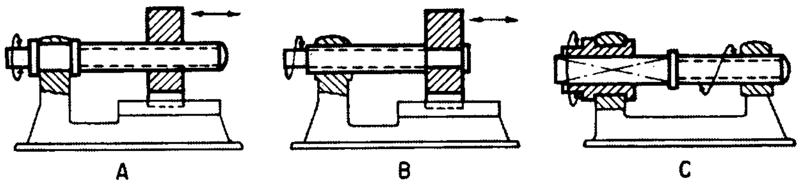


Fig. 1 Motion transformations of a screw thread include: rotation to translation (A), helical to translation (B), rotation to helical (C). These are reversible if the thread is not self-locking. (The thread is reversible when its efficiency is over 50%.)

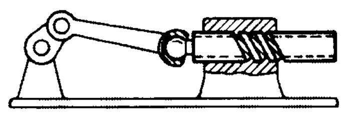


Fig. 2 Standard four-bar linkage has a screw thread substituted for a slider. The output is helical rather than linear.

Rotation to Translation

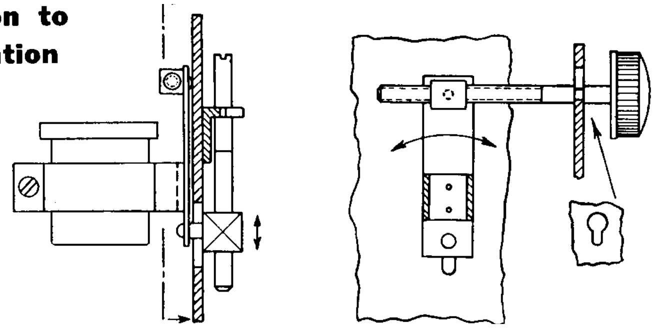


Fig. 3 A two-directional lamp adjustment with screwdriver will move a lamp up and down. A knob adjust (right) rotates the lamp about a pivot.

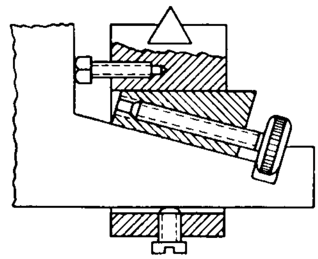


Fig. 4 A knife-edge bearing is raised or lowered by a screw-driven wedge. Two additional screws position the knife edge laterally and lock it.

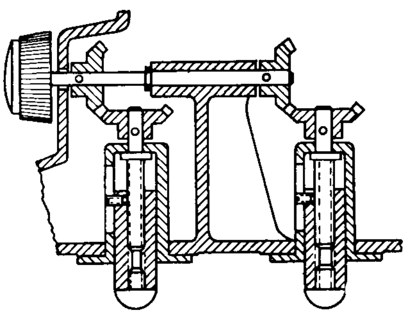


Fig. 5 A parallel arrangement of tandem screw threads raises the projector evenly.

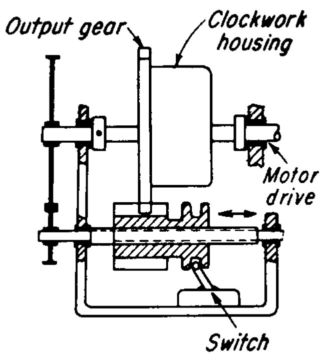


Fig. 6 Automatic clockwork is kept would taut by an electric motor turned on and off by a screw thread and nut. The motor drive must be self-locking or it will permit the clock to unwind as soon as the switch is turned off.

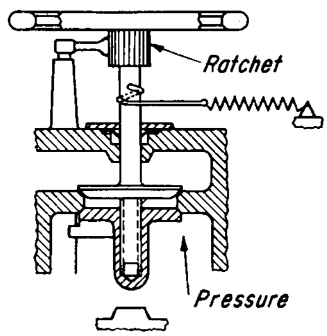


Fig. 7 A valve stem has two oppositely moving valve cones. When opening, the upper cone moves up first, until it contacts its stop. Further turning of the valve wheel forces the lower cone out of its seat. The spring is wound up at the same time. When the ratchet is released, the spring pulls both cones into their seats.

TRANSLATION TO ROTATION

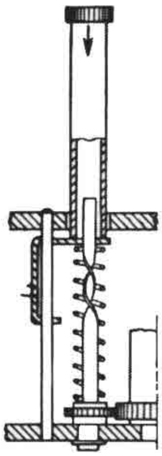


Fig. 8 A metal strip or square rod can be twisted to make a long-lead thread. It is ideal for transforming linear into rotary motion. Here a pushbutton mechanism winds a camera. The number of turns or dwell of the output gear is easily altered by changing (or even reversing) the twist of the strip.

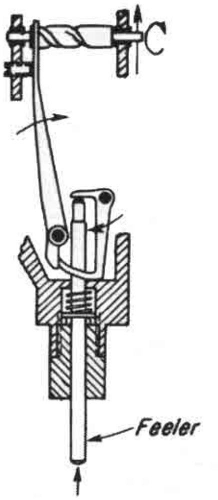


Fig. 9 A feeler gage has its motion amplified through a double linkage and then transformed to rotation for moving a dial needle.

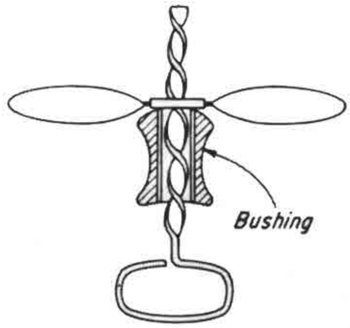


Fig. 10 The familiar flying propeller-toy is operated by pushing the bushing straight up and off the thread.

SELF-LOCKING

Fig. 11 A hairline adjustment for a telescope with two alternative methods for drive and spring return.

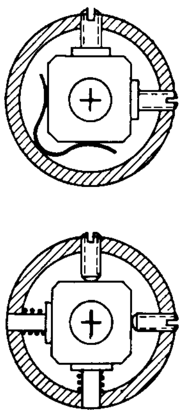
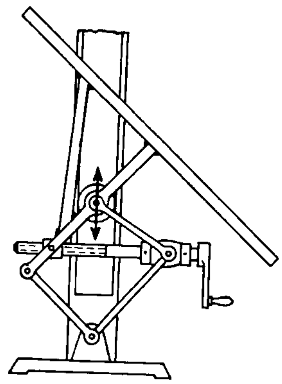


Fig. 12 This screw and nut form a self-locking drive for a complex linkage.



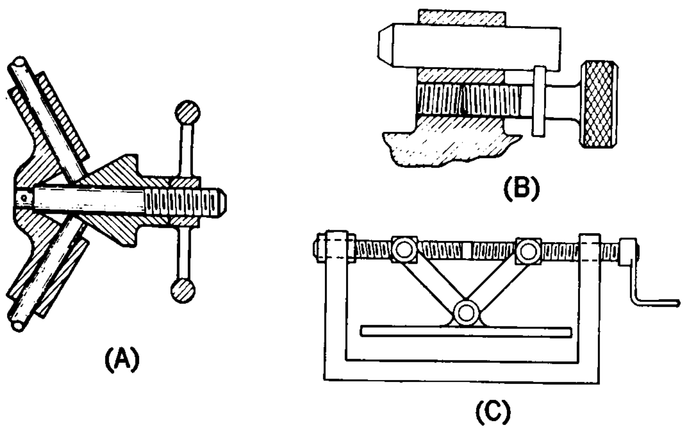


Fig. 13 Force translation. The threaded handle in (A) drives a coned bushing that thrusts rods outwardly for balanced pressure. The screw in (B) retains and drives a dowel pin for locking purposes. A right- and left-handed shaft (C) actuates a press.

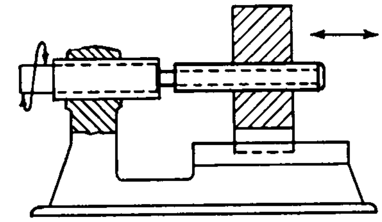


Fig. 14 Double-threaded screws, when used as differentials, permit very fine adjustment for precision equipment at relatively low cost.

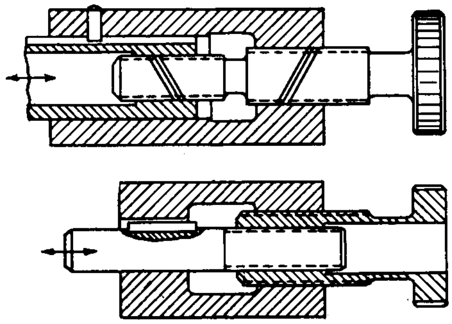


Fig. 15 Differential screws can be made in dozens of forms. Here are two methods: in the upper figure, two opposite-hand threads on a single shaft; in the lower figure, same-hand threads on independent shafts.

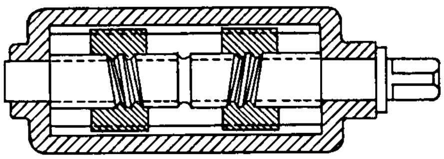


Fig. 16 Opposite-hand threads make a high-speed centering clamp out of two moving nuts.

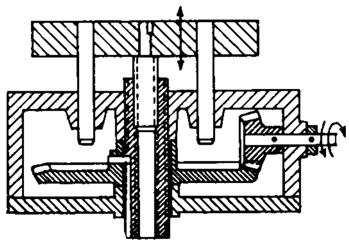


Fig. 17 A measuring table rises very slowly for many turns of the input bevel gear. If the two threads are 1½ to 12 and ¾ to 16, in the fine-thread series, the table will rise approximately 0.004 in. per input-gear revolution.

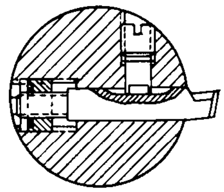


Fig. 18 A lathe turning tool in a drill rod is adjusted by a differential screw. A special double-pin wrench turns the intermediate nut, advancing the nut and retracting the threaded tool simultaneously. The tool is then clamped by a setscrew.

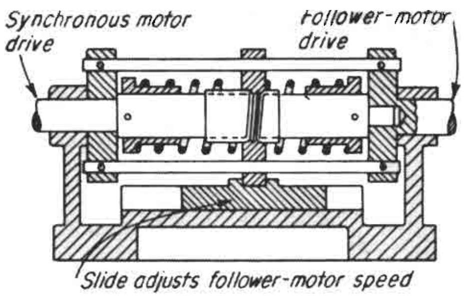


Fig. 19 Any variable-speed motor can be made to follow a small synchronous motor by connecting them to the two shafts of this differential screw. Differences in the number of revolutions between the two motors appear as motion of the traveling nut and slide, thus providing electrical speed compensation.

Fig. 20 A wire fork is the nut in this simple tube-and-screw device.

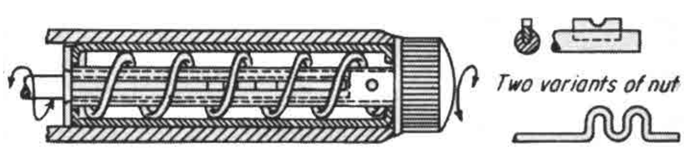
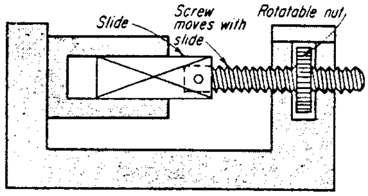


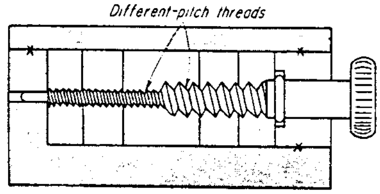
Fig. 21 A mechanical pencil includes a spring as the screw thread and a notched ear or a bent wire as the nut.

TEN WAYS TO EMPLOY SCREW MECHANISMS

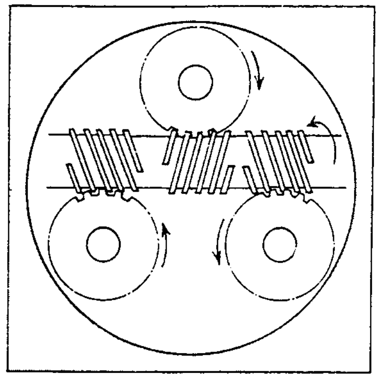
Three basic components of screw mechanisms are: actuating member (knob, wheel, handle), threaded device (screw-nut set), and sliding device (plunger-guide set).



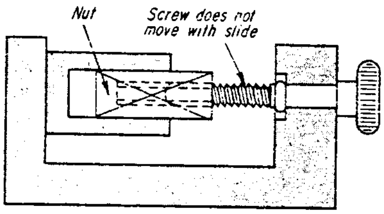
A nut can rotate but will not move longitudinally. Typical applications: screw jacks, heavy vertically moved doors; floodgates, opera-glass focusing, vernier gages, and Stillson wrenches.



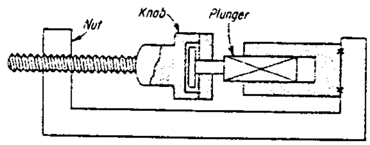
A differential movement is given by threads of different pitch. When the screw is rotated, the nuts move in the same direction but at different speeds.



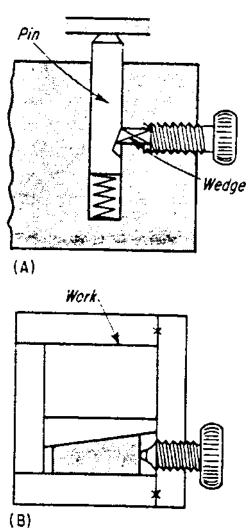
One screw actuates three gears simultaneously. The axes of gears are at right angles to that of the screw. This mechanism can replace more expensive gear setups there speed reduction and multiple output from a single input is required.



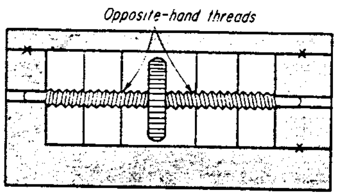
A screw can rotate but only the nut moves longitudinally. Typical applications: lathe tailstock feed, vises, lathe apron.



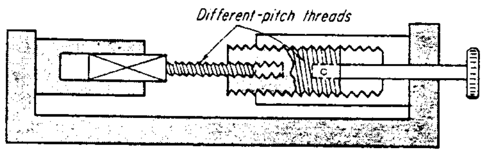
A screw and plunger are attached to a knob. The nut and guide are stationary. It is used on: screw presses, lathe steady-rest jaws for adjustment, and shaper slide regulation.



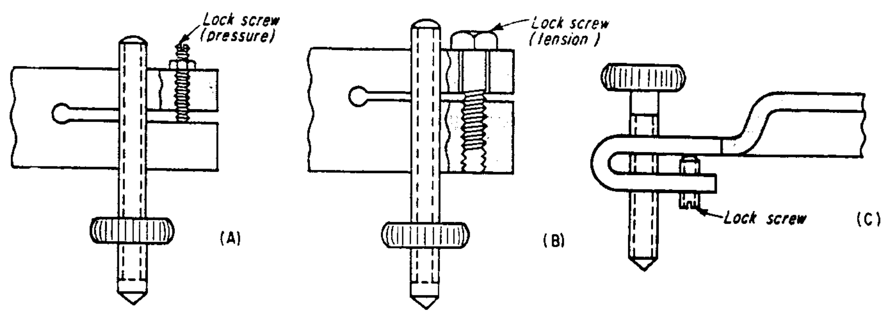
Screw-actuated wedges lock locating pin A and hold the work in fixture (B). These are just two of the many tool and diemaking applications for these screw actions.



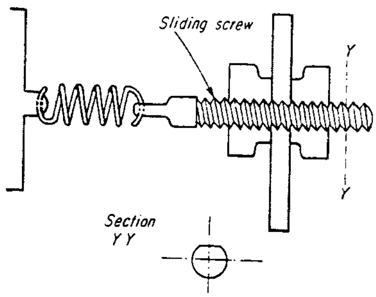
Opposing movement of lateral slides; adjusting members or other screw-actuated parts can be achieved with opposite-hand threads.



Concentric threading also gives differential movement. Such movements are useful wherever rotary mechanical action is required. A typical example is a gas-bottle valve, where slow opening is combined with easy control.



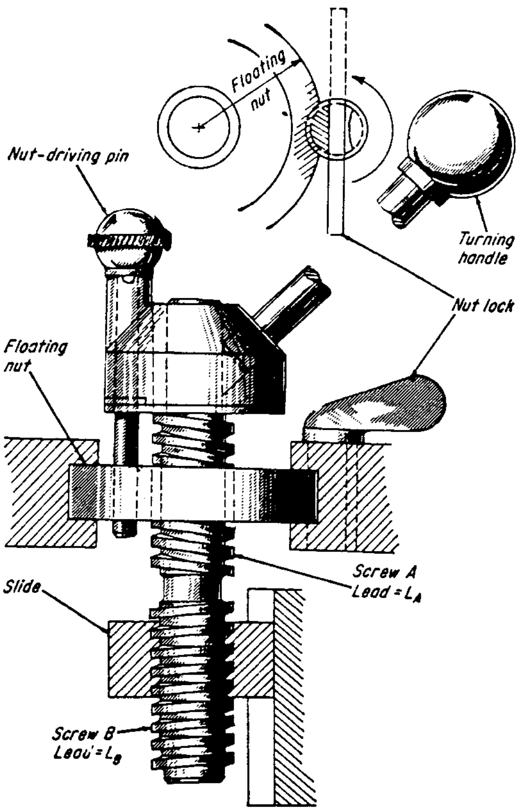
Adjustment screws are effectively locked by either a pressure screw (A) or tension screw (B). If the adjusting screw is threaded into a formed sheet-metal component (C), a setscrew can be used to lock the adjustment.



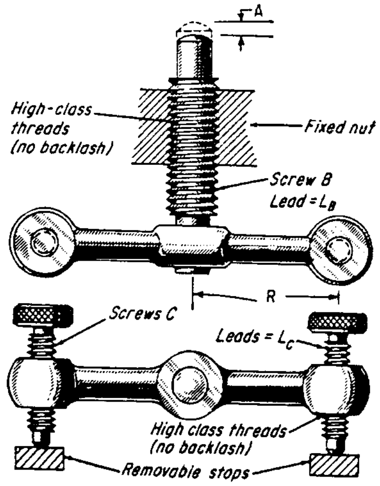
Locking nuts can be placed on opposite sides of a panel to prevent axial screw movement and simultaneously lock against vibrations. Drill-press depth stops and adjustable stops for shearing and cutoff dies are some examples.

SEVEN SPECIAL SCREW ARRANGEMENTS

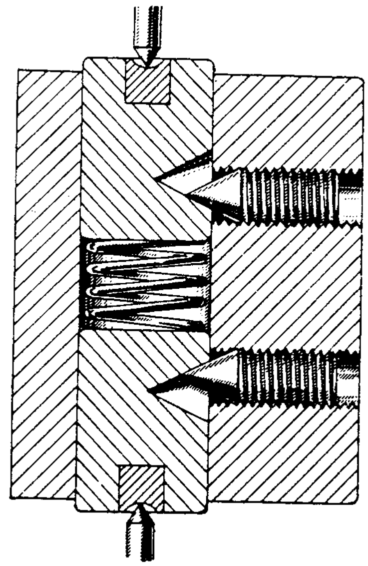
Differential, duplex, and other types of screws can provide slow and fast feeds, minute adjustments, and strong clamping action.



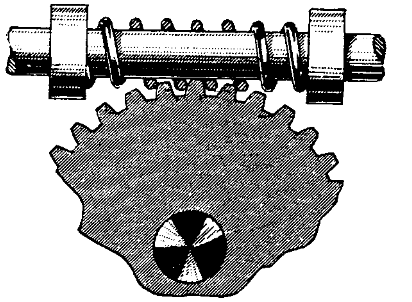
Rapid and slow feed. With left- and right-hand threads, slide motion with the nut locked equals L_A plus L_B per turn; with the nut floating, slide motion per turn equals L_B. Extremely fine feed with a rapid return motion is obtained when the threads are differential.



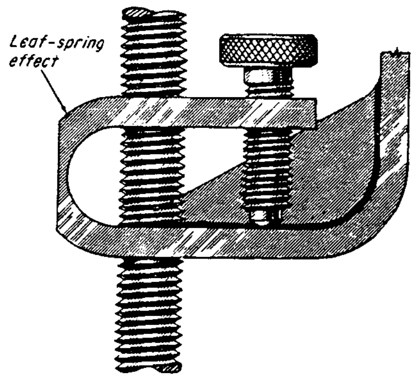
Extremely small movements. Microscopic measurements, for example, are characteristic of this arrangement. Movement A is equal to $N(L_B \times L_t)12\pi R$, where N equals the number of turns of screw C.



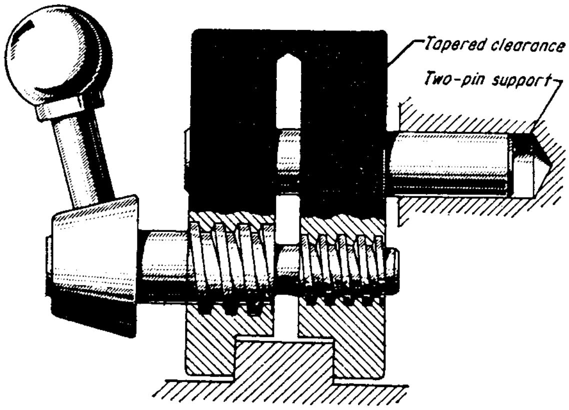
Bearing adjustment. This screw arrangement is a handy way for providing bearing adjustment and overload protection.



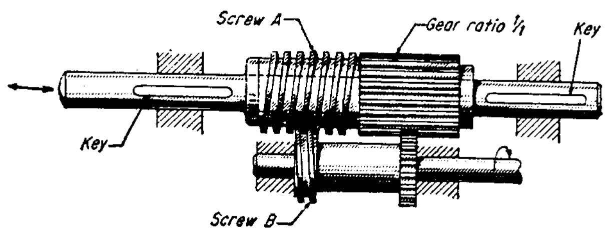
Shock absorbent screw. When the springs coiled as shown are used as worm drives for light loads, they have the advantage of being able to absorb heavy shocks.



Backlash elimination. The large screw is locked and all backlash is eliminated when the knurled screw is tightened; finger torque is sufficient.



Differential clamp. This method of using a differential screw to tighten clamp jaws combines rugged threads with high clamping power. Clamping pressure, $P = Te[R(\tan\phi + \tan\alpha]$, where T = torque at handle, R = mean radius of screw threads, ϕ = angle of friction (approx. 0.1), α = mean pitch angle or screw, and e = efficiency of screw generally about 0.8).



High reduction of rotary motion to fine linear motion is possible here. This arrangement is for low forces. Screws are left and right hand. $L_A = L_B$ plus or minus a small increment. When L_B **=** 1/10 and L_A = 1/10.5, the linear motion f screw A will be 0.05 in. per turn. When screws are the same hand, linear motion equals $L_A + L_B$.

FOURTEEN ADJUSTING DEVICES

Here is a selection of some basic devices that provide and hold mechanical adjustment.

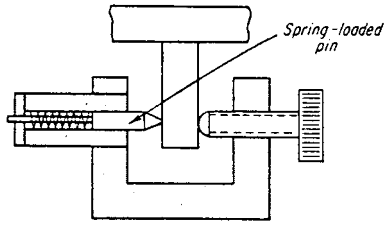


Fig. 1 A **spring-loaded pin** supplies a counterforce against which an adjustment force must always act. A leveling foot would work against gravity, but for most other setups a spring is needed to give a counter-force.

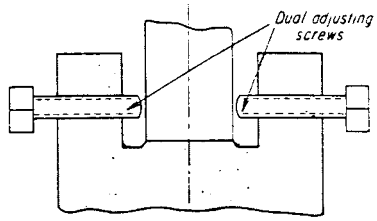


Fig. 2 Dual screws provide an inelastic counterforce. Backing-off one screw and tightening the other allows extremely small adjustments to be made. Also, once adjusted, the position remains solid against any forces tending to move the device out of adjustment.

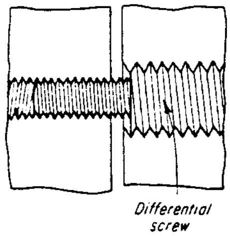


Fig. 3 A differential screw has same-hand threads but with different pitches. The relative distance between the two components can be adjusted with high precision by differential screws.

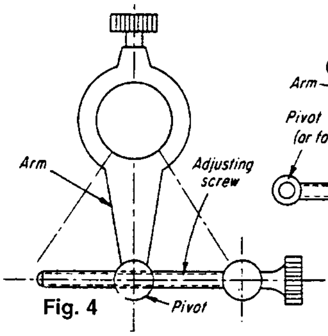


Fig. 4

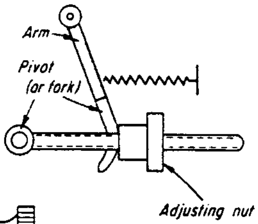


Fig. 5

Figs. 4 and 5 Swivel motion is necessary in (Fig. 4) between the adjusting screw and arm because of a circular locus of female threads in the actuated member. Similar action (Fig. 5) requires either the screw to be pivoted or the arm to be forked.

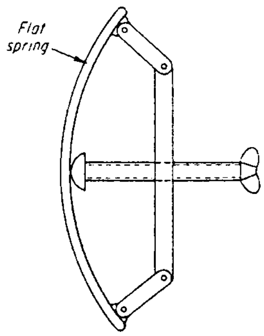


Fig. 6 This arc-drafting guide is an example of an adjusting device. One of its components, the flat spring, both supplies the counterforce and performs the mechanism's main function—guiding the pencil.

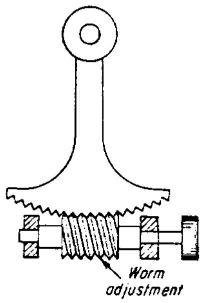


Fig. 7 The worm adjustment shown here is in a device for varying the position of an arm. Measuring instruments, and other tools requiring fine adjustments, include this adjusting device.

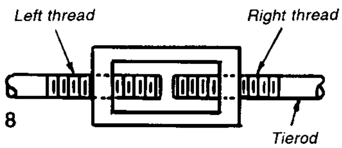


Fig. 8

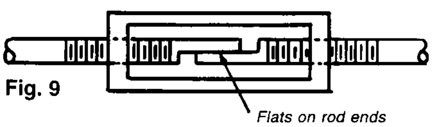


Fig. 9

Figs. 8 and 9 Tierods with opposite-hand threads at their ends (Fig. 8) require only a similarly threaded nut to provide simple, axial adjustment. Flats on the rod ends (Fig. 9) make it unnecessary to restrain both the rods against rotation when the adjusting screw is turned; restraining one rod is enough.

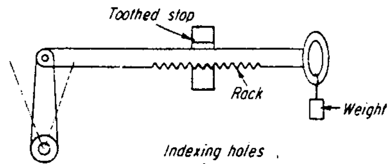


Fig. 13

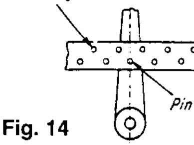


Fig. 14

Figs 13 and 14 Rack and toothed stops (Fig. 13) are frequently used to adjust heavy louvers, boiler doors and similar equipment. The adjustment is not continuous; it depends on the rack pitch. Large counter-adjustment forces might require a weighted rack to prevent tooth disengagement. Indexing holes (Fig. 14) provide a similar adjustment to the rack. The pin locks the members together.

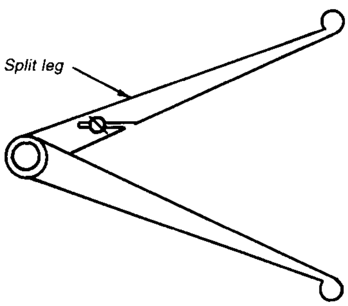


Fig. 10 A split-leg caliper is an example of a simple but highly efficient adjusting device. A tapered screw forces the split leg part, thus enlarging the opening between the two legs.

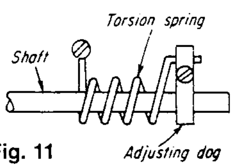


Fig. 11

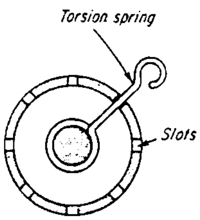


Fig. 12

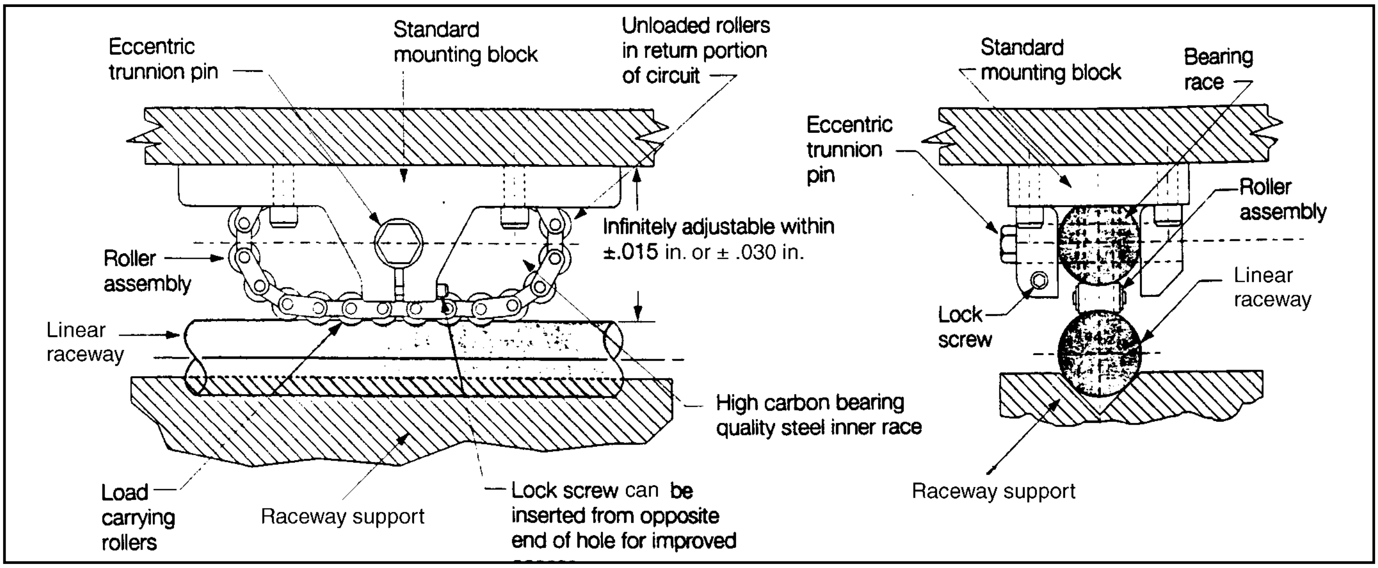
Figs. 11 and 12 Shaft torque is adjusted (Fig. 11) by rotating the spring-holding collar relative to the shaft, and locking the collar at a position of desired torque. Adjusting slots (Fig. 12) accommodate the torsion-spring arm after the spring is wound to the desired torque.

LINEAR ROLLER BEARINGS ARE SUITED FOR HIGH-LOAD, HEAVY-DUTY TASKS

The patented Roundway linear roller bearings from Thomson Industries, Inc., Port Washington, NY, can carry heavy loads on supported parallel cylindrical rails where rigidity and stiffness is required. The Roundway linear roller bearing consists of a cylindrical inner bearing race with rounded-ends that is fastened to a mounting block by a trunnion pin. It is enclosed by a linked chain of concave rollers that circulate around the race. The rollers and the inner race are made from hardened and ground high-carbon bearing steel, and the mounting block is cast from malleable iron. The load on the mounting block is transferred through the trunnion pin, race, and roller chain assembly to the supporting rail, which functions as the external raceway.

The height of the bearing can be adjusted with the eccentric trunnion pin to compensate for variations in the mounting surfaces. The pin can also be used to preload the bearing by eliminating internal bearing clearance. After the trunnion pin has been adjusted, it can be held in place by tightening the lock screw.

Because a single Roundway linear roller bearing does not resist side loads, a dual version of the Roundway bearing capable of resisting those loads is available. It has two race and roller assemblies mounted on a wider iron block so that the bearings contact the raceway support at angles of 45° from the centerline. In typical motion control installation, two single-bearing units are mounted in tandem on one of the parallel rails and two dual-bearing units are mounted in tandem on the other rail to withstand any sideloads.



The concave steel rollers in this linear bearing are linked in a chain assembly as they circulate around the inner race.

CHAPTER 7

CAM, TOGGLE, CHAIN, AND BELT MECHANISMS

CAM BASICS

A cam is a mechanical component that is capable of transmitting motion to a follower by direct contact. The driver is called a cam, and the driven member is called the follower. The follower can remain stationary, translate, oscillate, or rotate. The motion is given by $y = f(\theta)$, where

y = cam function (follower) displacement (in.).

f = external force (lb), and

$\theta = w_t$ – cam angle rotation for displacement y, (rad).

Figure 1 illustrates the general form of a plane cam mechanism. It consists of two shaped members A and B with smooth, round, or elongated contact surfaces connected to a third body C. Either body A or body B can be the driver while the other is the follower. These shaped bodies can be replaced by an equivalent mechanism. They are pin-jointed at the instantaneous centers of curvature, 1 and 2, of the contacting surfaces. With any change in relative positions, the points 1 and 2 are shifted and the links of the equivalent mechanism have different lengths.

Figure 2 shows the two most commonly used cams. Cams can be designed by

- Shaping the cam body to some known curve, such as involutes, spirals, parabolas, or circular arcs.
- Designing the cam mathematically to establish the follower motion and then forming the cam by plotting the tabulated data.
- Establishing the cam contour in parametric form.
- Laying out the cam profile by eye or with the use of appropriately shaped models.

The fourth method is acceptable only if the cam motion is intended for low speeds that will permit the use of a smooth, "bumpless" curve. In situations where higher loads, mass, speed, or elasticity of the members are encountered, a detailed study must be made of both the dynamic aspects of the cam curve and the accuracy of cam fabrication.

The roller follower is most frequently used to distribute and reduce wear between the cam and the follower. The cam and follower must be constrained at all operating speeds. A preloaded compression spring (with an open cam) or a positive drive is used. Positive drive action is accomplished by either a cam groove milled into a cylinder or a conjugate follower or followers in contact with opposite sides of a single or double cam.

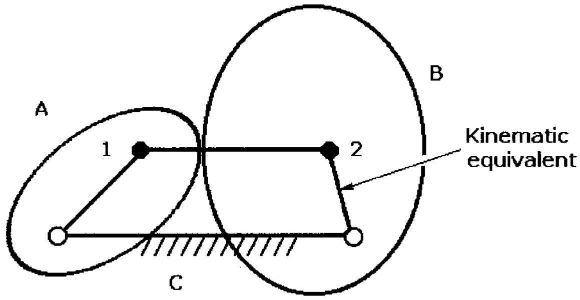


Fig. 1 Basic cam mechanism and its kinematic equivalent (points 1 and 2 are centers of curvature) of the contact point.

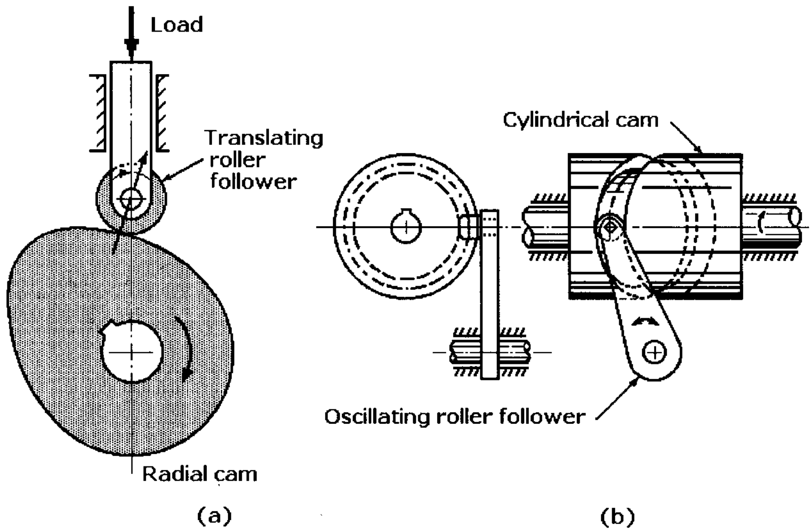


Fig. 2 Popular cams: (a) radial cam with a translating roller follower (open cam), and (b) cylindrical cam with an oscillating roller follower (closed cam).

CAM-CURVE GENERATING MECHANISMS

It usually doesn't pay to design a complex cam curve if it can't be easily machined—so check these mechanisms before starting your cam design.

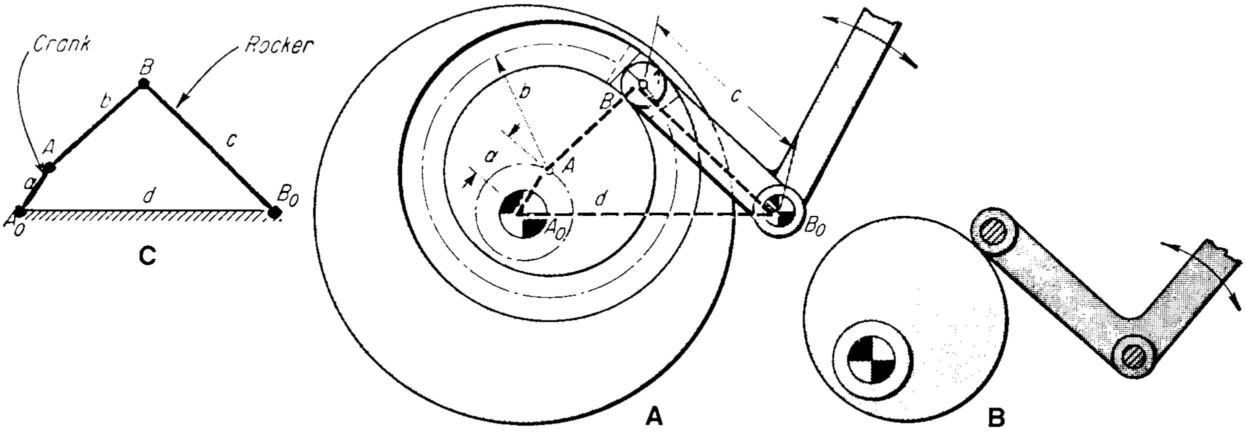


Fig. 1 A circular cam groove is easily machined on a turret lathe by mounting the plate eccentrically onto the truck. The plate cam in **(B)** with a spring-load follower produces the same output motion. Many designers are unaware that this type of cam has the same output motion as four-bar linkage **(C)** with the indicated equivalent link lengths. Thus, it's the easiest curve to pick when substituting a cam for an existing linkage.

If you have to machine a cam curve into the metal blank without a master cam, how accurate can you expect it to be? That depends primarily on how precisely the mechanism you use can feed the cutter into the cam blank. The mechanisms described here have been carefully selected for their practicability. They can be employed directly to machine the cams, or to make master cams for producing other cams.

The cam curves are those frequently employed in automatic-feed mechanisms and screw machines They are the circular, constant-velocity, simple-harmonic, cycloidal, modified cycloidal, and circular-arc cam curve, presented in that order.

Circular Cams

This is popular among machinists because of the ease in cutting the groove. The cam (Fig. 1A) has a circular groove whose center, A, is displaced a distance a from the cam-plate center, A_0, can simply be a plate cam with a spring-loaded follower (Fig. 1B).

Interestingly, with this cam you can easily duplicate the motion of a four-bar linkage (Fig. 1C). Rocker BB_0 in Fig. 1C, therefore, is equivalent to the motion of the swinging follower shown in Fig. 1A.

The cam is machined by mounting the plate eccentrically on a lathe. Consequently, a circular groove can be cut to close tolerances with an excellent surface finish.

If the cam is to operate at low speeds, you can replace the roller with an arc-formed slide. This permits the transmission of high forces. The optimum design of these "power cams" usually requires time-consuming computations.

The disadvantages (or sometimes, the advantage) of the circular-arc cam is that, when traveling from one given point, its follower reaches higher-speed accelerations than with other equivalent cam curves.

Constant-Velocity Cams

A constant-velocity cam profile can be generated by rotating the cam plate and feeding the cutter linearly, both with uniform velocity, along the path the translating roller follower will travel later (Fig. 2A). In the example of a swinging follower, the tracer (cutter) point is placed on an arm whose length is equal to the length of the swinging roller follower, and the arm is rotated with uniform velocity (Fig. 2B).

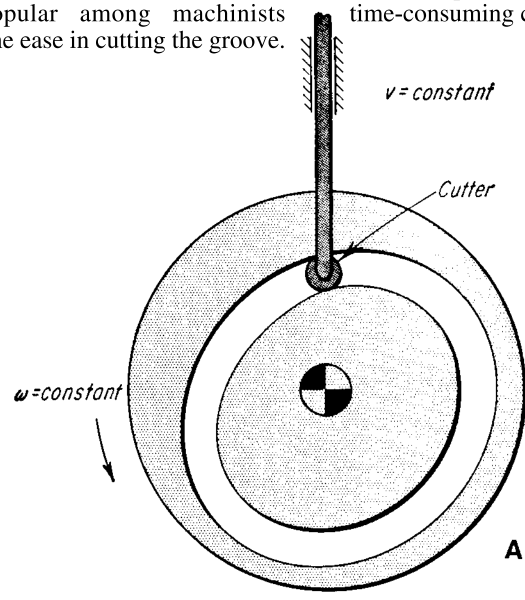
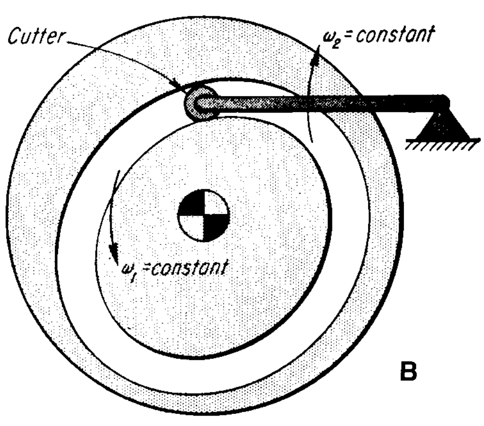


Fig. 2 A constant-velocity cam is machined by feeding the cutter and rotating the cam at constant velocity. The cutter is fed linearly **(A)** or circularly **(B)**, depending on the type of follower.



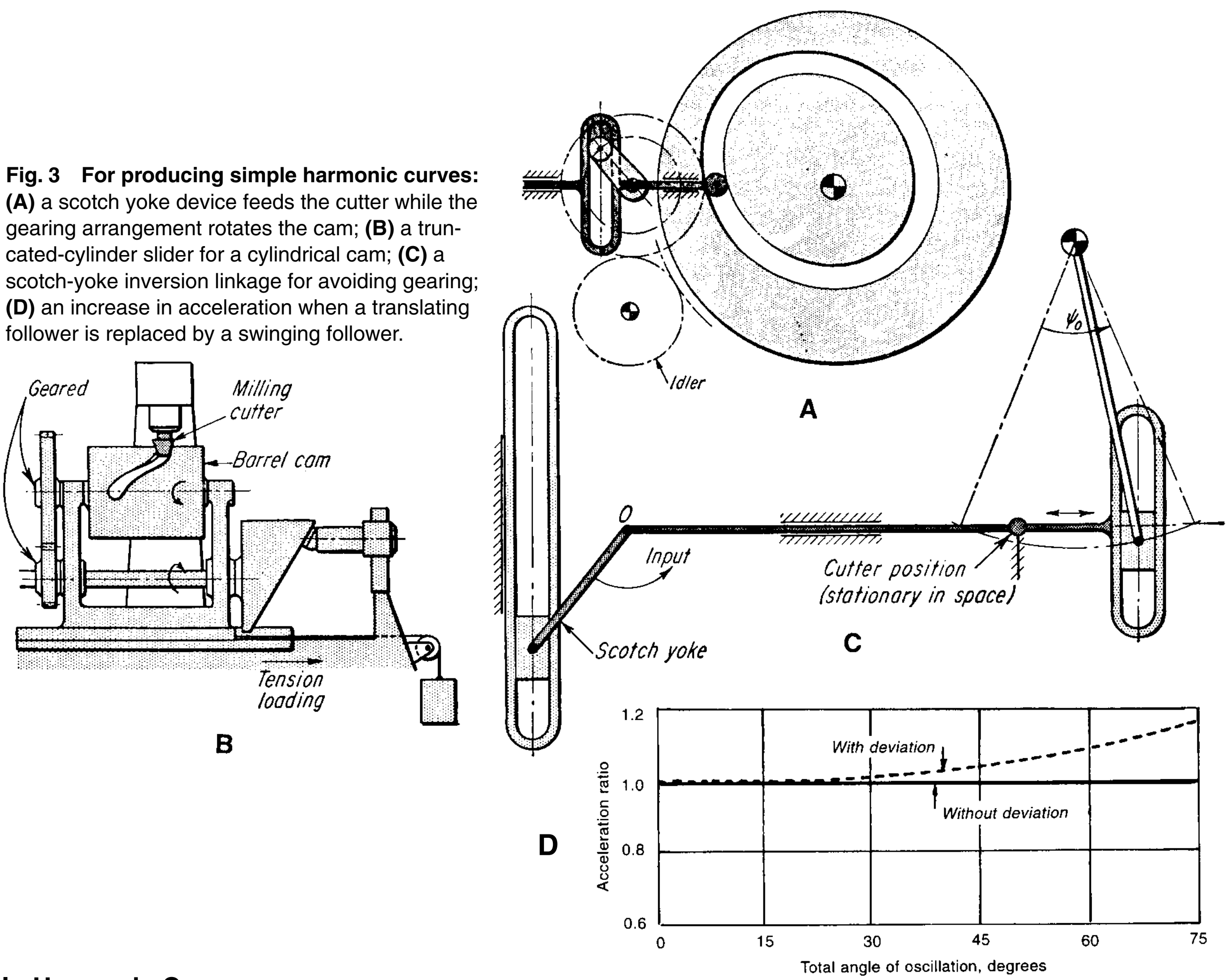


Fig. 3 For producing simple harmonic curves: **(A)** a scotch yoke device feeds the cutter while the gearing arrangement rotates the cam; **(B)** a truncated-cylinder slider for a cylindrical cam; **(C)** a scotch-yoke inversion linkage for avoiding gearing; **(D)** an increase in acceleration when a translating follower is replaced by a swinging follower.

Simple-Harmonic Cams

The cam is generated by rotating it with uniform velocity and moving the cutter with a scotch yoke geared to the rotary motion of the cam. Fig. 3A shows the principle for a radial translating follower; the same principle is applicable for offset translating and the swinging roller follower. The gear ratios and length of the crank working in the scotch yoke control the pressures angles (the angles for the rise or return strokes).

For barrel cams with harmonic motion, the jig in Fig. 3B can easily be set up to do the machining. Here, the barrel cam is shifted axially by the rotating, weight-loaded (or spring-loaded) truncated cylinder.

The scotch-yoke inversion linkage (Fig. 3C) replaces the gearing called for in Fig. 3A. It will cut an approximate simple-harmonic motion curve when the cam has a swinging roller follower, and an exact curve when the cam has a radial or offset translating roller follower. The slotted member is fixed to the machine frame *1*. Crank 2 is driven around the center *0*. This causes link *4* to oscillate back and forward in simple harmonic motion. The sliding piece *5* carries the cam to be cut, and the cam is rotated around the center of *5* with uniform velocity. The length of arm *6* is made equal to the length of the swinging roller follower of the actual am mechanism and the device adjusted so that the extreme position of the center of *5* lie on the center line of *4*.

The cutter is placed in a stationary spot somewhere along the centerline of member *4*. If a radial or offset translating roller follower is used, sliding piece *5* is fastened to *4*.

The deviation from simple harmonic motion, when the cam has a swinging follower, causes an increase in acceleration ranging from 0 to 18% (Fig. 3D), which depends on the total angle of oscillation of the follower. Note that for a typical total oscillating angle of 45° the increase in acceleration is about 5%.

Cycloidal Motion

This curve is perhaps the most desirable from a designer's viewpoint because of its excellent acceleration characteristic. Luckily, this curve is comparatively easy to generate. Before selecting the mechanism, it is worth looking at the underlying theory of cycloids because it is possible to generate not only cycloidal motion but a whole family of similar curves.

The cycloids are based on an offset sinusoidal wave (Fig. 4). Because the

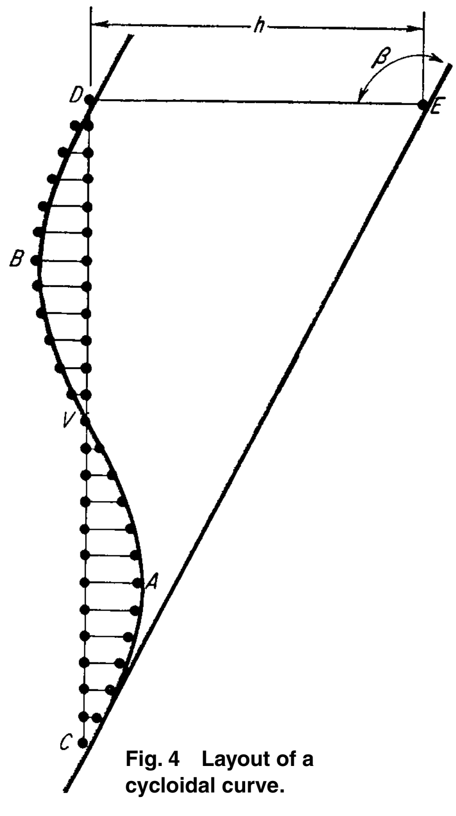


Fig. 4 Layout of a cycloidal curve.

radii of curvatures in points C, V, and D are infinite (the curve is "flat" at these points), if this curve was a cam groove and moved in the direction of line CVD, a translating roller follower, actuated by this cam, would have zero acceleration at points C, V, and D no matter in what direction the follower is pointed.

Now, if the cam is moved in the direction of CE and the direction of motion of the translating follower is lined up perpendicular to CE, the acceleration of the follower in points, C, V, and D would still be zero. This has now become the basic cycloidal curve, and it can be considered as a sinusoidal curve of a certain amplitude (with the amplitude measured perpendicular to the straight line) superimposed on a straight (constant-velocity) line.

The cycloidal is considered to be the best standard cam contour because of its low dynamic loads and low shock and vibration characteristics. One reason for these outstanding attributes is that sudden changes in acceleration are avoided during the cam cycle. But improved performance is obtainable with certain modified cycloidals.

Modified Cycloids

To modify the cycloid, only the direction and magnitude of the amplitude need to be changed, while keeping the radius of curvature infinite at points C, V, and D.

Comparisons are made in Fig. 5 of some of the modified curves used in industry. The true cycloidal is shown in the cam diagram of Fig. 5A. Note that the sine amplitudes to be added to the constant-velocity line are perpendicular to the base. In the Alt modification shown in Fig. 5B (named after Hermann Alt, a German kinematician who first analyzed it), the sine amplitudes are perpendicular to the constant-velocity line. This results in improved (lower) velocity characteristics (Fig. 5D), but higher acceleration magnitudes (Fig. 5E).

The Wildt modified cycloidal (after Paul Wildt) is constructed by selecting a point w which is 0.57 the distance $T/2$, and then drawing line wp through yp which is midway along OP. The base of the sine curve is then constructed perpendicular to yw. This modification results in a maximum acceleration of 5.88 h/T^2. By contrasts, the standard cycloidal curve has a maximum acceleration of 6.28 h/T^2. This is a 6.8 reduction in acceleration.

(It's a complex task to construct a cycloidal curve to go through a particular point P—where P might be anywhere within the limits of the box in Fig. 5C—and with a specific scope at P. There is a growing demand for this kind of cycloidal modification.

Generating Modified Cycloidals

One of the few methods capable of generating the family of modified cycloidals consists of a double carriage and rack arrangement (Fig. 6A).

The cam blank can pivot around the spindle, which in turn is on the movable carriage I. The cutter center is stationary. If the carriage is now driven at constant speed by the leadscrew in the direction of the arrow, steel bands 1 and 2 will also cause the cam blank to rotate. This rotation-and-translation motion of the cam will cut a spiral groove.

For the modified cycloidals, a second motion must be imposed on the cam to compensate for the deviations from the

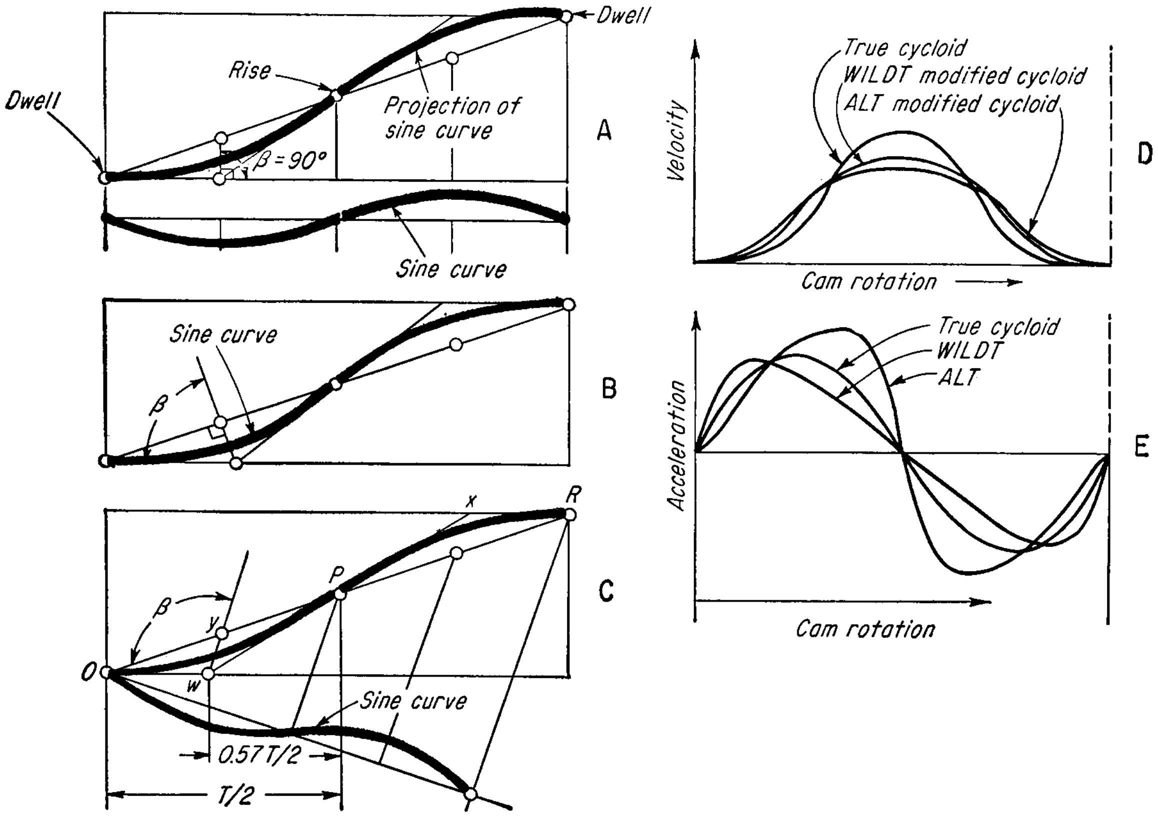


Fig. 5 A family of cycloidal curves: (A) A standard cycloidal motion; **(B)** A modification according to H. Alt; **(C)** A modification according to P. Wildt; **(D)** A comparison of velocity characteristics; **(E)** A comparison of acceleration curves.

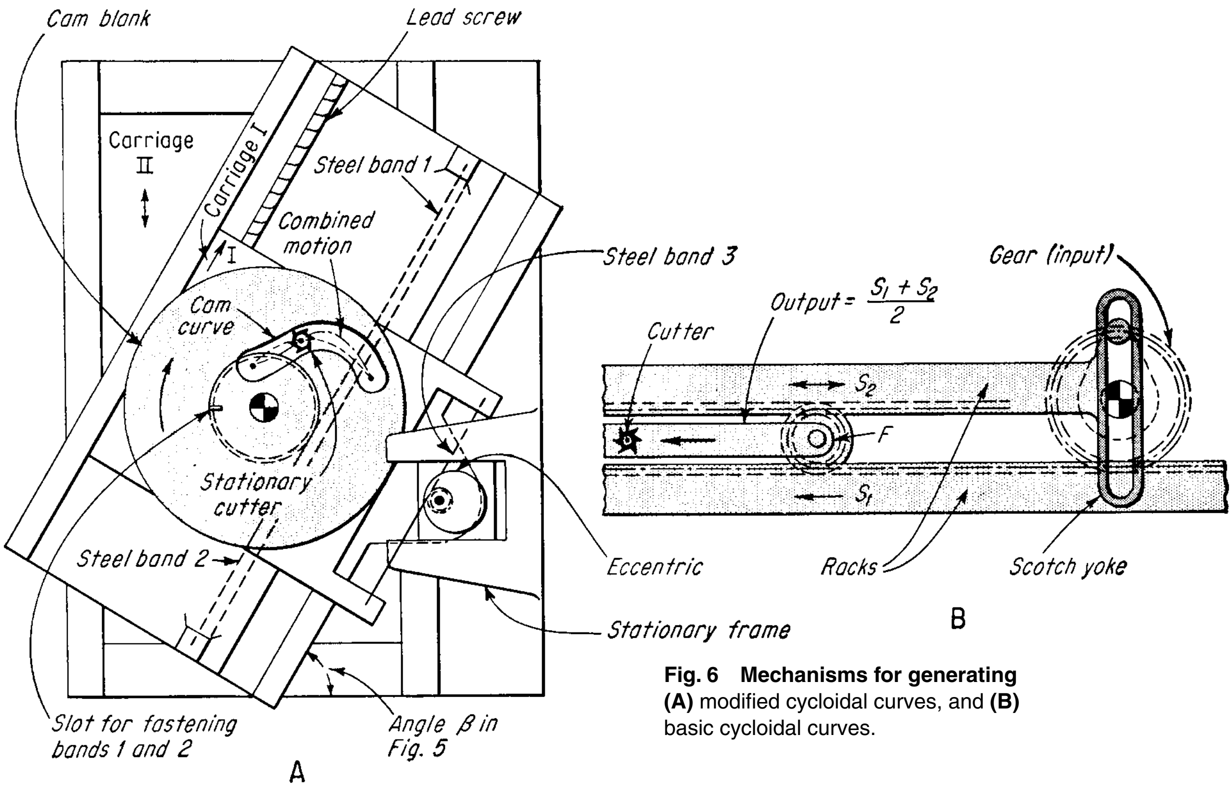


Fig. 6 Mechanisms for generating (A) modified cycloidal curves, and **(B)** basic cycloidal curves.

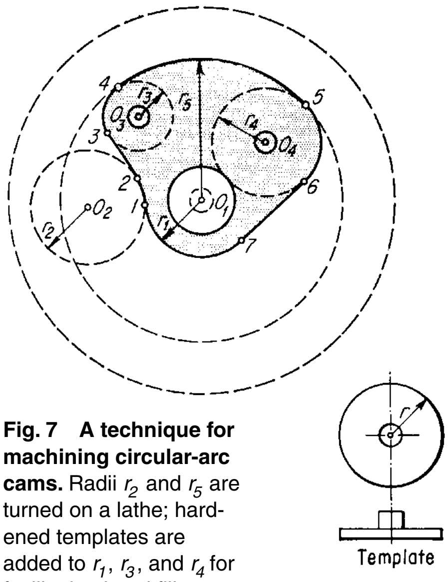


Fig. 7 A technique for machining circular-arc cams. Radii r_2 and r_5 are turned on a lathe; hardened templates are added to r_1, r_3, and r_4 for facilitating hand filing.

true cycloidal. This is done by a second steel-band arrangement. As carriage I moves, bands 3 and 4 cause the eccentric to rotate. Because of the stationary frame, the slide surrounding the eccentric is actuated horizontally. This slide is part of carriage II. As a result, a sinusoidal motion is imposed on the cam.

Carriage I can be set at various angles β to match angle β in Fig. 5B and C. The mechanism can also be modified to cut cams with swinging followers.

Circular-Arc Cams

In recent years it has become customary to turn to the cycloidal and other similar curves even when speeds are low. However, there are still many applications for circular-arc cams. Those cams are composed of circular arcs, or circular arc and straight lines. For comparatively small cams, the cutting technique illustrated in Fig. 7 produces accurate results.

Assume that the contour is composed of circular arc *1-2* with center at 0_2, arc *3-*4 with center at 0_3, arc *4-*5 with center at 0_1, arc 5-*6* with center at 0_4, arc *7-1* with center at 0_1, and the straight lines *2-3* and *6-7*. The method calls for a combination of drilling, lathe turning, and template filing.

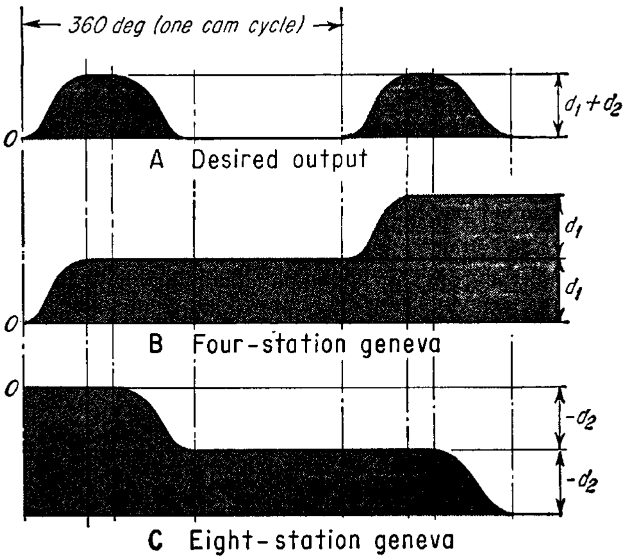
First, small holes about 0.1 in. in diameter are drilled at 0_1, 0_3, and 0_4. Then a hole drilled with the center at 0_2, and radius of r_2. Next the cam is fixed in a turret lathe with the center of rotation at 0_1, and the steel plate is cut until it has a diameter of $2r_5$. This completes the larger convex radius. The straight lines *6-7* and *2-3* are then milled on a milling machine.

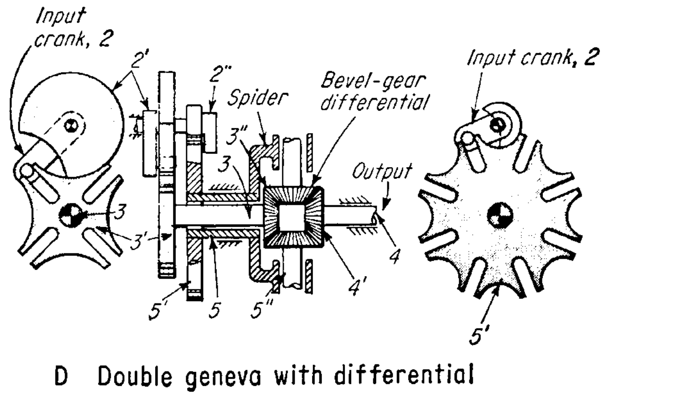
Finally, for the smaller convex arcs, hardened pieces are turned with radii r_1, r_3, and r_4. One such piece is shown in Fig. 7. The templates have hubs that fit into the drilled holes at 0_1, 0_3, and 0_4. Next the arcs *7-1*, *3-4*, and *5-6* are filed with the hardened templates as a guide. The final operation is to drill the enlarged hole at 0_1 to a size that will permit a hub to be fastened to the cam.

This method is usually better than copying from a drawing or filing the scallops from a cam on which a large number of points have been calculated to determine the cam profile.

Compensating for Dwells

One disadvantage with the previous generating machines is that, with the exception of the circular cam, they cannot include a dwell period within the rise-and-fall cam cycle. The mechanisms must be disengaged at the end of the rise, and the cam must be rotated the exact number of degrees to the point where the





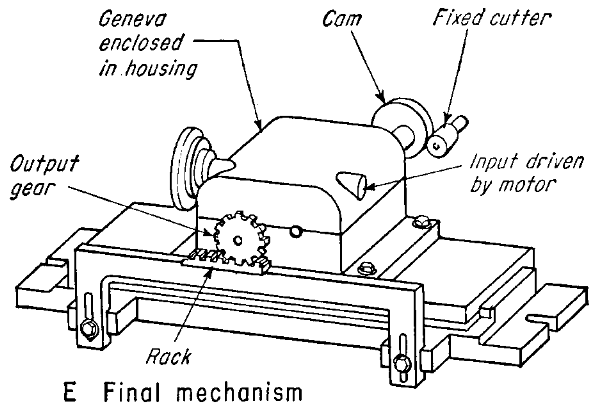


Fig. 8 Double genevas with differentials for obtaining long dwells. The desired output characteristic **(A)** of the cam is obtained by adding the motion **(B)** of a four-station geneva to that of **(C)** an eight-station geneva. The mechanical arrangement of genevas with a differential is shown in **(D);** the actual device is shown in **(E).** A wide variety of output dwells **(F)** are obtained by varying the angle between the driving cranks of the genevas.

fall cycle begins. This increases the possibility of inaccuracies and slows down production.

There are two mechanisms, however, that permit automatic cam machining through a specific dwell period: the double-geneva drive and the double eccentric mechanism.

Double-Genevas with Differential

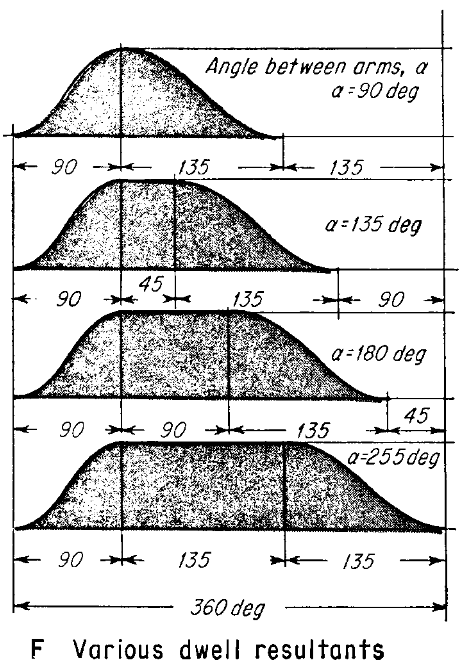
Assume that the desired output contains dells (of specific duration) at both the rise and fall portions, as shown in Fig. 8A. The output of a geneva that is being rotated clockwise will produce an intermittent motion similar to the one shown in Fig. 8B—a rise-dwell-rise-dwell motion. These rise portions are distorted simple-harmonic curves, but are sufficiently close to the pure harmonic to warrant their use in many applications.

If the motion of another geneva, rotating counterclockwise as shown in (Fig. 8C), is added to that of the clockwise geneva by a differential (Fig. 8D), then the sum will be the desired output shown in (Fig. 8A).

The dwell period of this mechanism is varied by shifting the relative positions between the two input cranks of the genevas.

The mechanical arrangement of the mechanism is shown in Fig. 8D. The two driving shafts are driven by gearing (not shown). Input from the four-star geneva to the differential is through shaft *3*; input from the eight-station geneva is through the spider. The output from the differential, which adds the two inputs, is through shaft *4*.

The actual mechanism is shown in Fig. 8E. The cutter is fixed in space. Output is from the gear segment that rides on a fixed rack. The cam is driven by the motor, which also drives the enclosed genevas. Thus, the entire device reciprocates back and forth on the slide to feed the cam properly into the cutter.



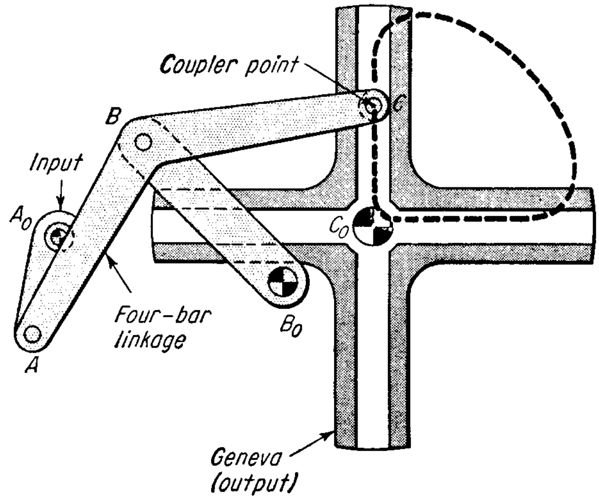


Fig. 9 A four-bar coupler mechanism for replacing the cranks in genevas to obtain smoother acceleration characteristics.

Genevas Driven by Couplers

When a geneva is driven by a constant-speed crank, as shown in Fig. 8D, it has a sudden change in acceleration at the beginning and end of the indexing cycle (as the crank enters or leaves a slot). These abrupt changes can be avoided by employing a four-bar linkage with a coupler in place of the crank. The motion of the coupler point *C* (Fig. 9) permits its smooth entry into the geneva slot

Double Eccentric Drive

This is another machine for automatically cutting cams with dwells. The rotation of crank *A* (Fig. 10) imparts an oscillating motion to the rocker *C* with a prolonged dwell at both extreme positions. The cam, mounted on the rocker, is rotated by the chain drive and then is fed into the cutter with the proper motion. During the dwells of the rocker, for example, a dwell is cut into the cam.

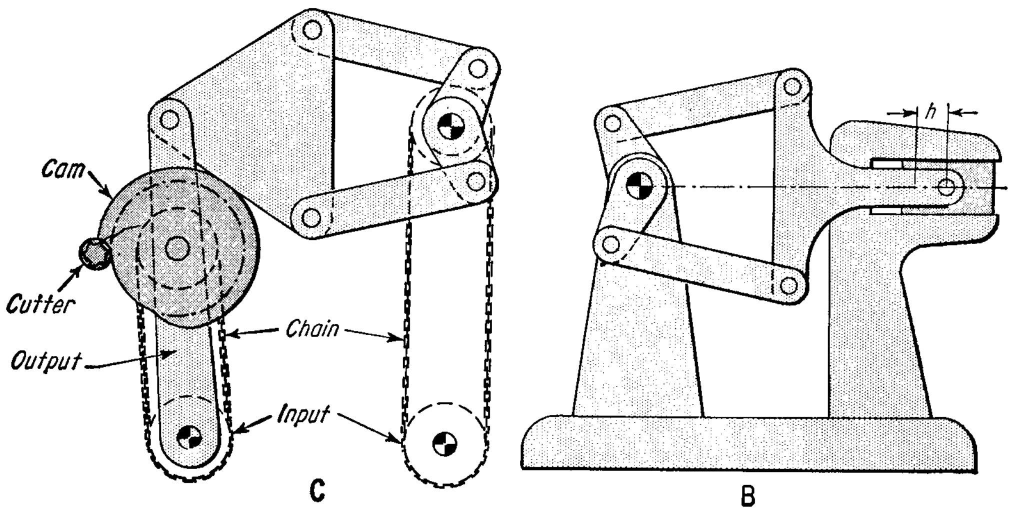
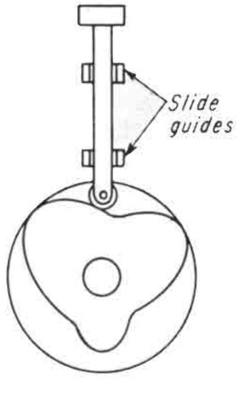


Fig. 10 A double eccentric drive for automatically cutting cams with dwells. The cam is rotated and oscillated, with dwell periods at extreme ends of oscillation corresponding to desired dwell periods in the cam.

FIFTEEN IDEAS FOR CAM MECHANISMS

This assortment of devices reflects the variety of ways in which cams can be put to work.



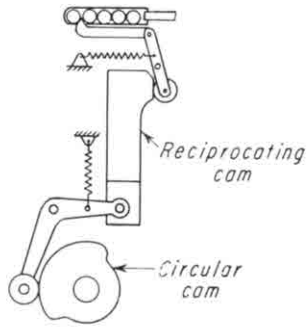


Fig. 4 An automatic feed for automatic machines. There are two cams, one with circular motion, the other with reciprocating motion. This combination eliminates any trouble caused by the irregularity of feeding and lack of positive control over stock feed.

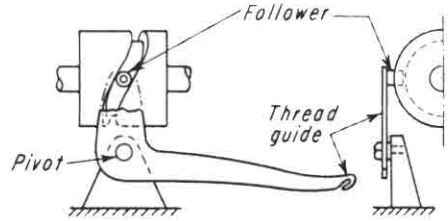
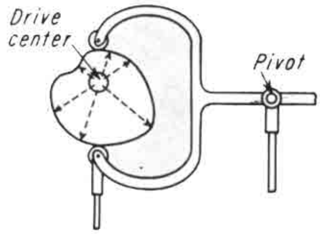
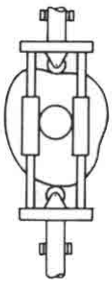


Fig. 5 A barrel cam with milled grooves is used in sewing machines to guide thread. This kind of cam is also used extensively in textile manufacturing machines such as looms and other intricate fabric-making machines.





Figs. 1, 2, and 3 A constant-speed rotary motion is converted into a variable, reciprocating motion (Fig. 1); rocking or vibratory motion of a simple forked follower (Fig. 2); or a more robust follower (Fig. 3), which can provide valve-moving mechanisms for steam engines. Vibratory-motion cams must be designed so that their opposite edges are everywhere equidistant when they are measured through their drive-shaft centers.

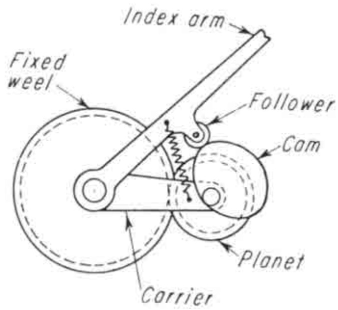


Fig. 6 This indexing mechanism combines an epicyclic gear and cam. A planetary wheel and cam are fixed relative to one another; the carrier is rotated at uniform speed around the fixed wheel. The index arm has a nonuniform motion with dwell periods.

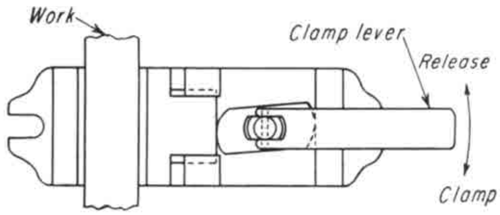
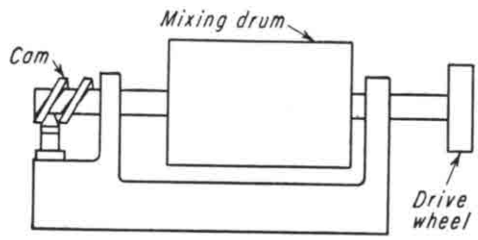


Fig. 7 A double eccentric, actuated by a suitable handle, provides powerful clamping action for a machine-tool holding fixture.

Fig. 8 A mixing roller for paint, candy, or food. A mixing drum has a small oscillating motion while rotating.



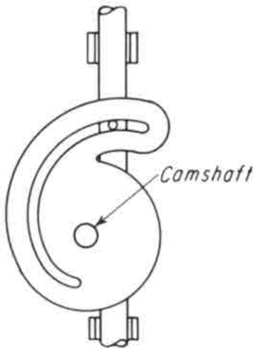


Fig. 9 A slot cam converts the oscillating motion of a camshaft to a variable but straight-line motion of a rod. According to slot shape, rod motion can be made to suit specific design requirements, such as straight-line and logarithmic motion.

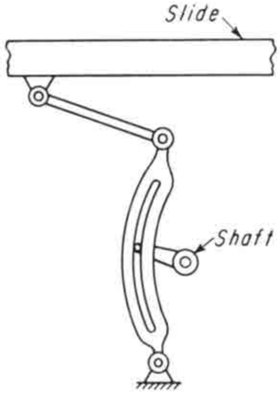


Fig. 10 The continuous rotary motion of a shaft is converted into the reciprocating motion of a slide. This device is used on sewing machines and printing presses.

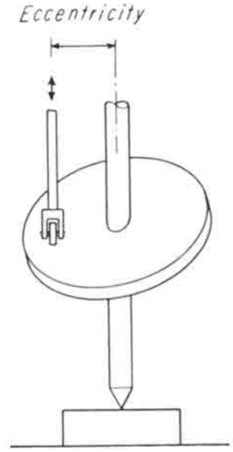


Fig. 11 Swash-plate cams are feasible for light loads only, such as in a pump. The cam's eccentricity produces forces that cause excessive loads. Multiple followers can ride on a plate, thereby providing smooth pumping action for a multipiston pump.

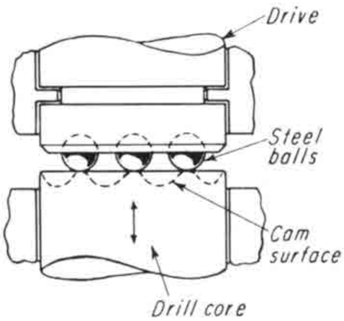


Fig. 12 This steel-ball cam can convert the high-speed rotary motion of an electric drill into high-frequency vibrations that power the drill core for use as a rotary hammer for cutting masonry, and concrete. This attachment can also be designed to fit hand drills.

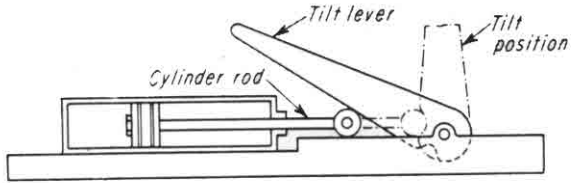


Fig. 13 This tilting device can be designed so that a lever remains in a tilted position when the cylinder rod is withdrawn, or it can be spring-loaded to return with a cylinder rod.

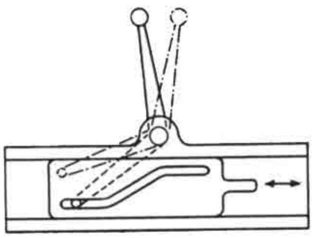


Fig. 14 This sliding cam in a remote control can shift gears in a position that is otherwise inaccessible on most machines.

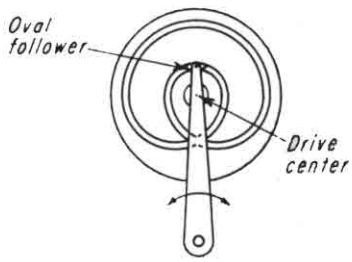


Fig. 15 A groove and oval follower form a device that requires two revolutions of a cam for one complete follower cycle.

SPECIAL-FUNCTION CAMS

Fig. 1—A quick drop of the follower is obtained by permitting the cam to be pushed out of the way by the follower itself as it reaches the edge of the cam. Lugs *C* and *C′* are fixed to the camshaft. The cam is free to turn (float) on the camshaft, limited by lug *C* and the adjusting screw. With the cam rotating clockwise, lug C drives the cam through lug *B*. At the position shown, the roller will drop off the edge of the cam, which is then accelerated clockwise until its cam lug *B* strikes the adjusting screw of lug *C′*.

Fig. 2—Instantaneous drop is obtained by the use of two integral cams and followers. The roller follower rides on cam *1*. Continued rotation will transfer contact to the flat-faced follower, which drops suddenly off the edge of cam *2*. After the desired dwell, the follower is restored to its initial position by cam *1*.

Fig. 3—The dwell period of the cam can be varied by changing the distance between the two rollers in the slot.

Fig. 4—A reciprocating pin (not shown) causes the barrel cam to rotate intermittently. The cam is stationary while a pin moves from *1* to *2*. Groove *2-3* is at a lower level; thus, as the pin retracts, it cams the barrel cam; then it climbs the incline from *2* to the new position of *1*.

Fig. 5—A double-groove cam makes two revolutions for one complete movement of the follower. The cam has movable switches, *A* and *B*, which direct the follower alternately in each groove. At the instant shown, *B* is ready to guide the roller follower from slot *1* to slot *2*.

Figs. 6 and 7—Increased stroke is obtained by permitting the cam to shift on the input shaft. Total displacement of the follower is therefore the sum of the cam displacement on the fixed roller plus the follower displacement relative to the cam.

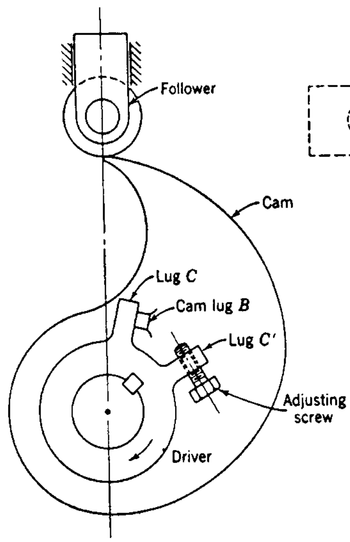


Fig. 1 **A quick-acting floating cam.**

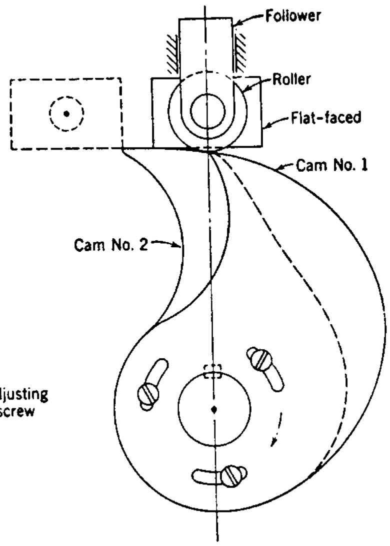


Fig. 2 **A quick-acting dwell cams.**

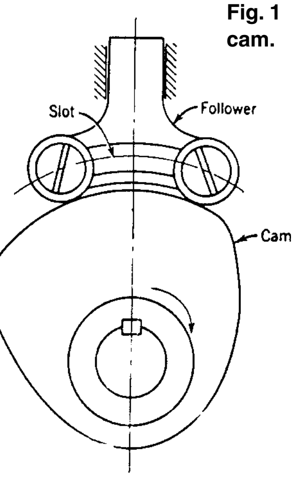


Fig. 3 **An adjustable-dwell cam.**

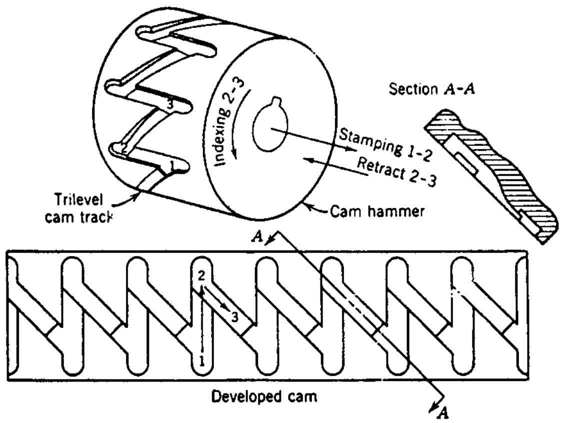


Fig. 4 **An indexing cam.**

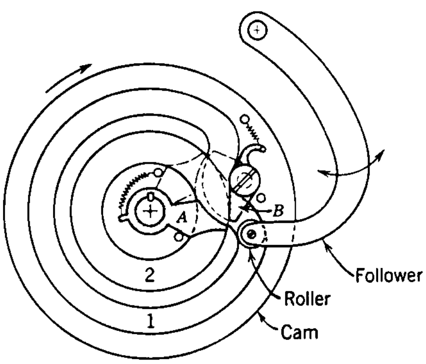


Fig. 5 **A double-revolution cam.**

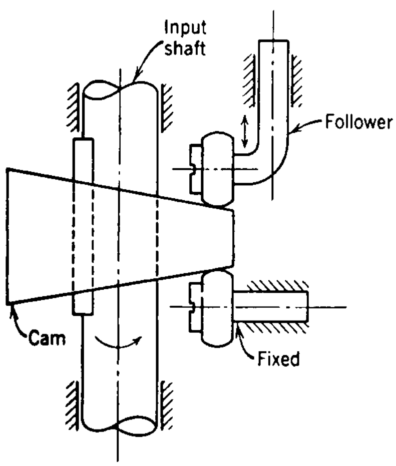


Fig. 6 **An increased-stroke barrel cam.**

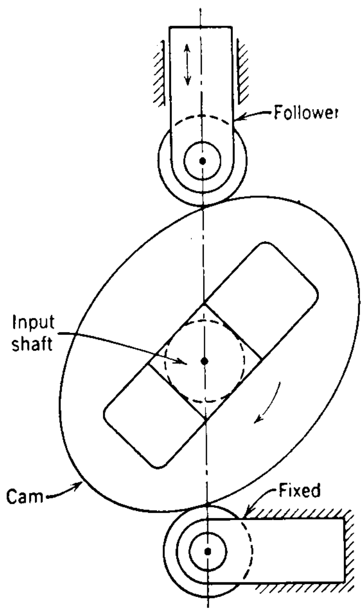


Fig. 7 **An increased-stroke plate cam.**

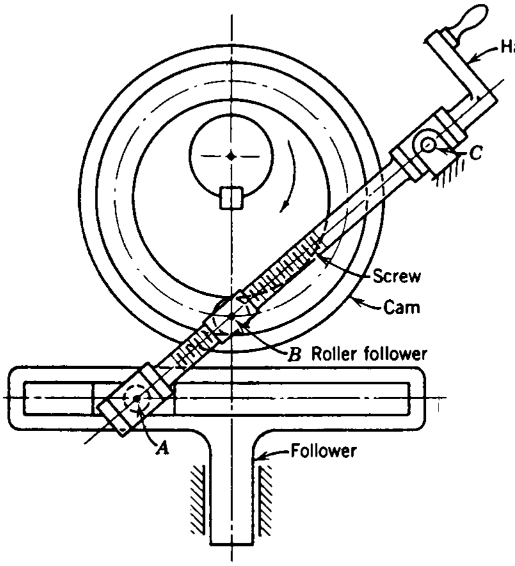
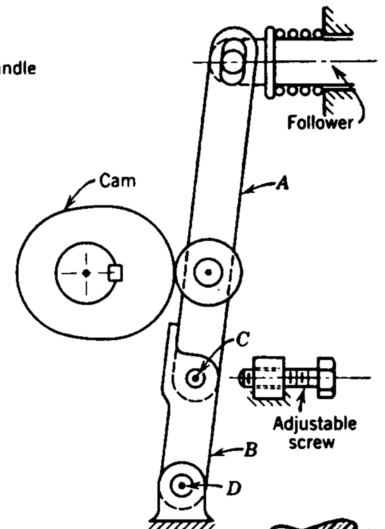


Fig. 8 An adjustable roller-position cam.



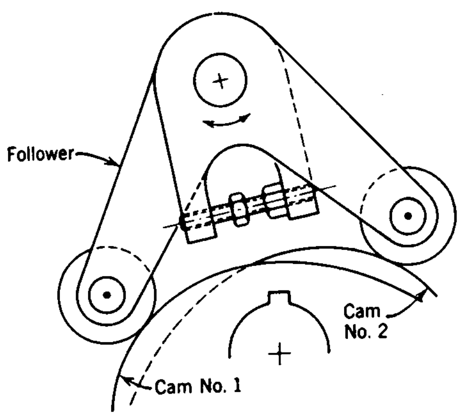


Fig. 9 An adjustable pivot-point cam.

Fig. 8—The stroke of the follower is adjusted by turning the screw handle which changes distance *AB*.

Fig. 9—The pivot point of the connecting link to the follower is changed from point *D* to point *C* by adjusting the screw.

Fig. 10—Adjustable dwell is obtained by having the main cam, with lug *A*, pinned to the revolving shaft. Lug *A* forces the plunger up into the position shown, and allows the latch to hook over the catch, thus holding the plunger in the up position. The plunger is unlatched by lug *B*. The circular slots in the cam plate permit the shifting of lug *B*, thereby varying the time that the plunger is held in the latched position.

REFERENCE: Rothbart, H. A. *Cams—Design, Dynamics, and Accuracy*, John Wiley and Sons, Inc., New York.

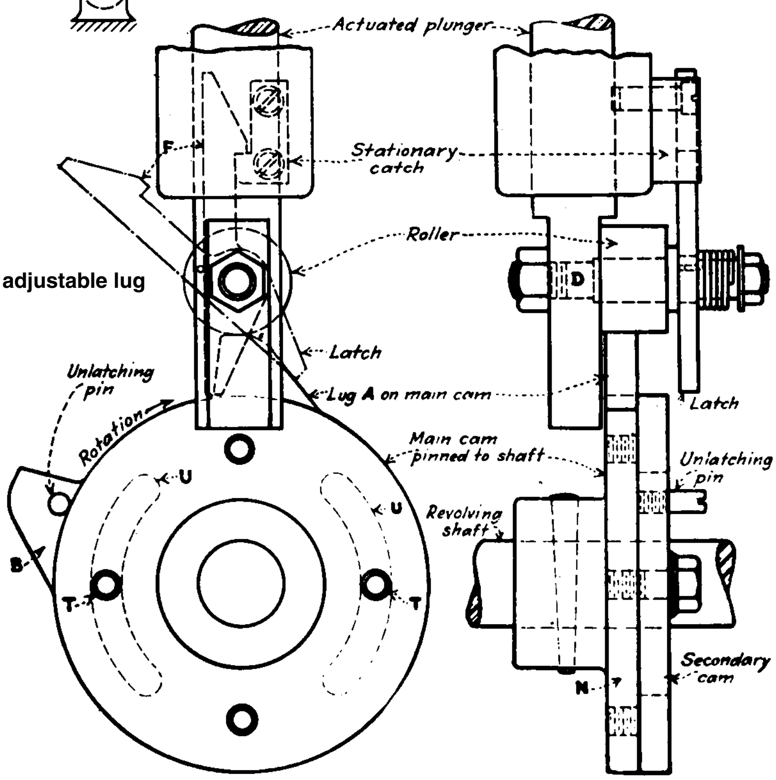
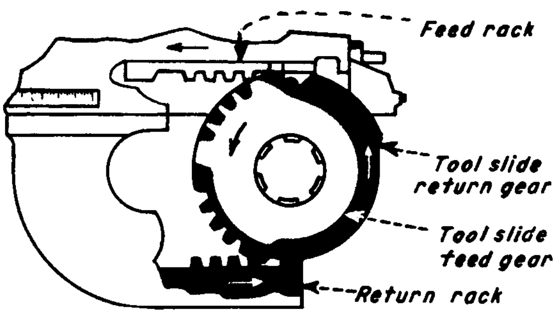
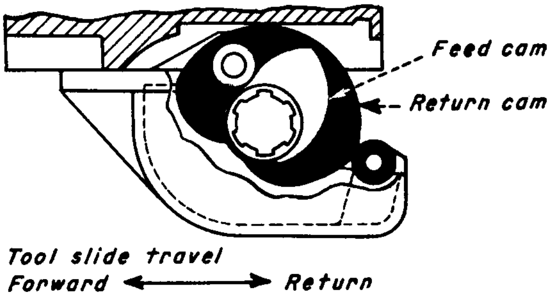


Fig. 10 An adjustable lug cam.

CAM DRIVES FOR MACHINE TOOLS

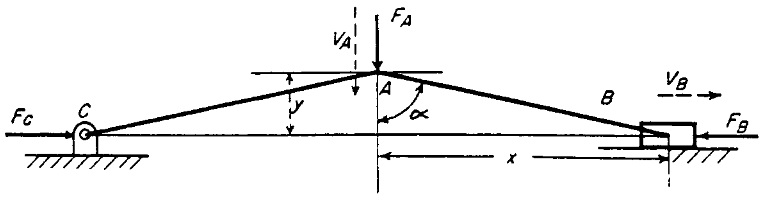




This two-directional rack-and-gear drive for a main tool slide combines accurate, uniform movement and minimum idle time. The mechanism makes a full double stroke each cycle. It approaches fast, shifts smoothly into feed, and returns fast. Its point-of-shift is controlled by an adjustable dog on a calibrated gear. Automatic braking action assures a smooth shift from approach to feed.

A cam drive for a tool-slide mechanism replaces a rack feed when a short stroke is required to get a fast machining cycle on automatic machines. The cams and rollers are shown with the slide in its retracted position.

TOGGLE LINKAGE APPLICATIONS IN DIFFERENT MECHANISMS



$$\text{Mechanical advantage} = \frac{F_B}{F_A} = \frac{1}{2}\frac{x}{y} = \frac{1}{2}\,\text{Tan}\,\alpha = \frac{V_A}{V_B}$$

Fig. 1 Many mechanical linkages are based on the simple toggle that consists of two links which tend to line up in a straight line at one point in their motion. The mechanical advantage is the velocity ratio of the input point *A* with respect to the outpoint point *B*: or V_A/V_B. As the angle α approaches 90º, the links come into toggle, and the mechanical advantage and velocity ratio both approach infinity. However, frictional effects reduce the forces to much les than infinity, although they are still quite high.

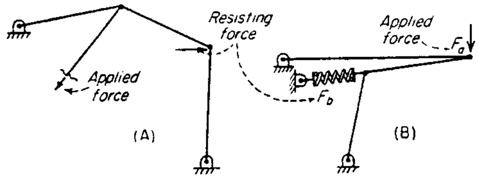


Fig. 2 Forces can be applied through other links, and need not be perpendicular to each other. (A) One toggle link can be attached to another link rather than to a fixed point or slider. (B) Two toggle links can come into toggle by lining up on top of each other rather than as an extension of each other. The resisting force can be a spring.

HIGH MECHANICAL ADVANTAGE

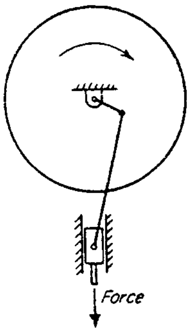


Fig. 3 In punch presses, large forces are needed at the lower end of the work stroke. However, little force is required during the remainder of the stroke. The crank and connecting rod come into toggle at the lower end of the punch stroke, giving a high mechanical advantage at exactly the time it is most needed.

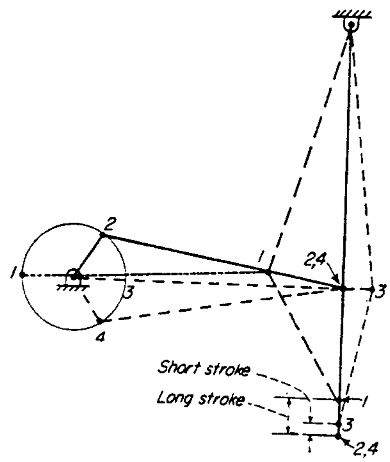


Fig. 4 A cold-heading rivet machine is designed to give each rivet two successive blows. Following the first blow (point 2) the hammer moves upward a short distance (to point 3). Following the second blow (at point 4), the hammer then moves upward a longer distance (to point 1) to provide clearance for moving the workpiece. Both strokes are produced by one revolution of the crank, and at the lowest point of each stroke (points 2 and 4) the links are in toggle.

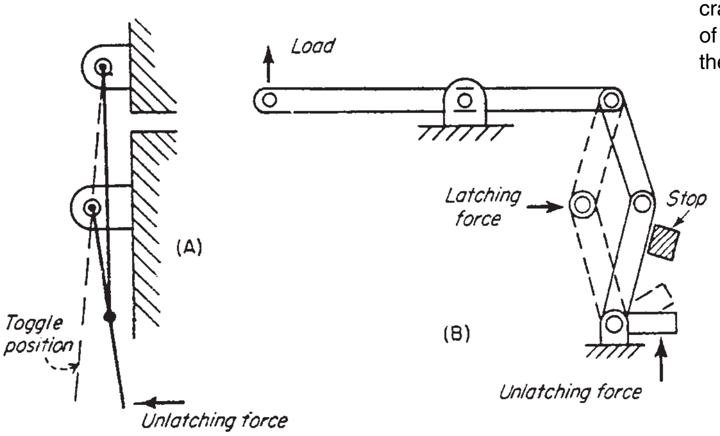


Fig. 5 Locking latches produce a high mechanical advantage when in the toggle portion of the stroke. A simple latch exerts a large force in the locked position (Fig. 5A). For positive locking, the closed position of latch is slightly beyond the toggle position. A small unlatching force opens the linkage (Fig. 5B).

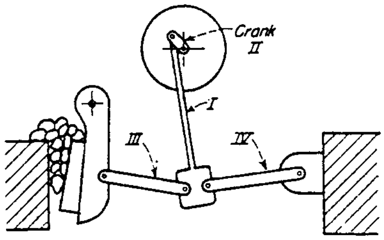


Fig. 6 A stone crusher has two toggle linkages in series to obtain a high mechanical advantage. When the vertical link *I* reaches the top of its stroke, it comes into toggle with the driving crank *II*; at the same time, link *III* comes into toggle and link *IV*. This multiplication results in a very large crushing force.

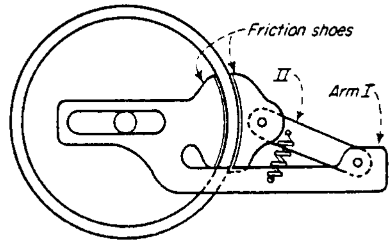
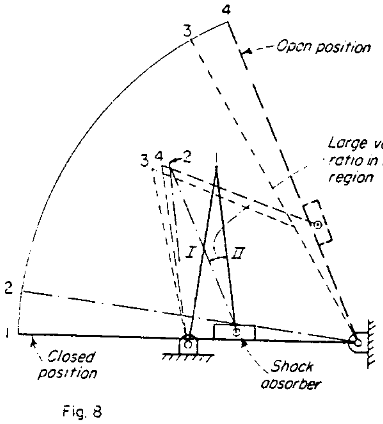


Fig. 7 A friction ratchet is mounted on a wheel; a light spring keeps the friction shoes in contact with the flange. This device permits clockwise motion of the arm *I*. However, reverse rotation causes friction to force link *II* into toggle with the shoes. This action greatly increases the locking pressure.

HIGH VELOCITY RATIO



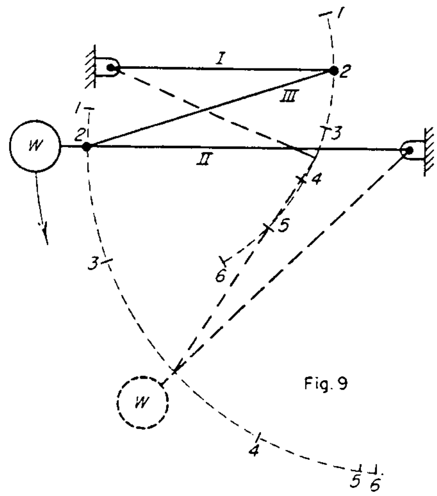


Fig. 8 Door check linkage gives a high velocity ratio during the stroke. As the door swings closed, connecting link *I* comes into toggle with the shock absorber arm *II*, giving it a large angular velocity. The shock absorber is more effective in retarding motion near the closed position.

Fig. 9 An impact reducer is on some large circuit breakers. Crank *I* rotates at constant velocity while the lower crank moves slowly at the beginning and end of the stroke. It moves rapidly at the midstroke when arm *II* and link *III* are in toggle. The accelerated weight absorbs energy and returns it to the system when it slows down.

VARIABLE MECHANICAL ADVANTAGE

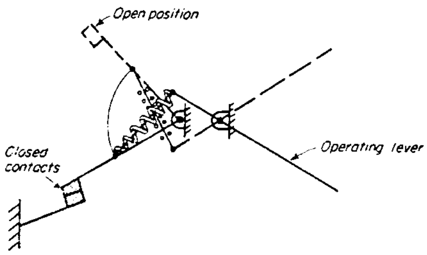


Fig. 10 A toaster switch has an increasing mechanical advantage to aid in compressing a spring. In the closed position, the spring holds the contacts closed and the operating lever in the down position. As the lever is moved upward, the spring is compressed and comes into toggle with both the contact arm and the lever. Little effort is required to move the links through the toggle position; beyond this point, the spring snaps the contacts open. A similar action occurs on closing.

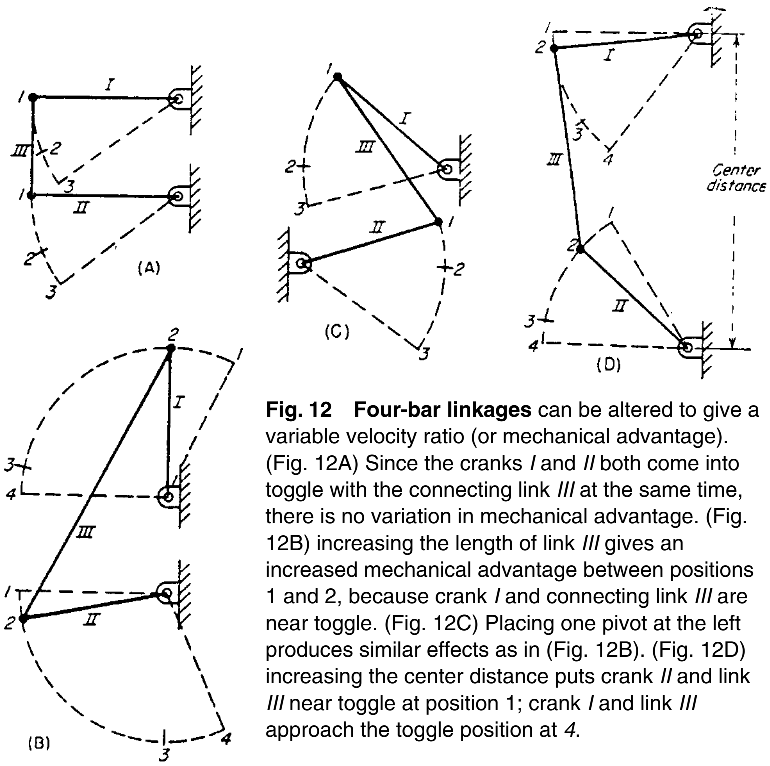


Fig. 12 Four-bar linkages can be altered to give a variable velocity ratio (or mechanical advantage). (Fig. 12A) Since the cranks *I* and *II* both come into toggle with the connecting link *III* at the same time, there is no variation in mechanical advantage. (Fig. 12B) increasing the length of link *III* gives an increased mechanical advantage between positions 1 and 2, because crank *I* and connecting link *III* are near toggle. (Fig. 12C) Placing one pivot at the left produces similar effects as in (Fig. 12B). (Fig. 12D) increasing the center distance puts crank *II* and link *III* near toggle at position 1; crank *I* and link *III* approach the toggle position at *4*.

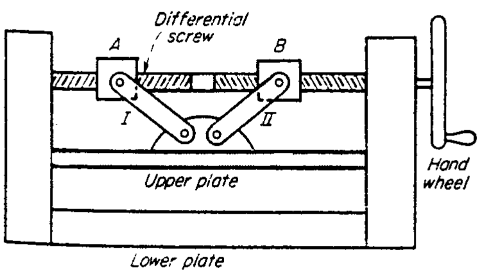


Fig. 11 A toggle press has an increasing mechanical advantage to counteract the resistance of the material being compressed. A rotating handwheel with a differential screw moves nuts *A* and *B* together, and links *I* and *II* are brought into toggle.

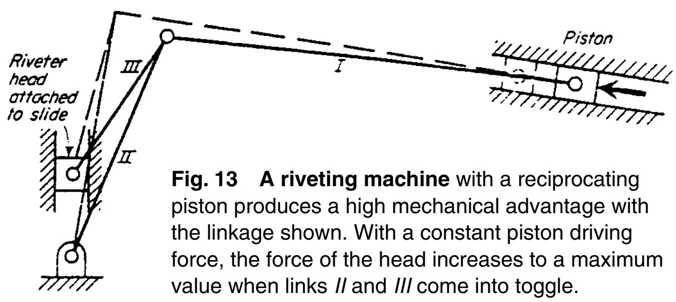


Fig. 13 A riveting machine with a reciprocating piston produces a high mechanical advantage with the linkage shown. With a constant piston driving force, the force of the head increases to a maximum value when links *II* and *III* come into toggle.

SIXTEEN LATCH, TOGGLE, AND TRIGGER DEVICES

Diagrams of basic latching and quick-release mechanisms.

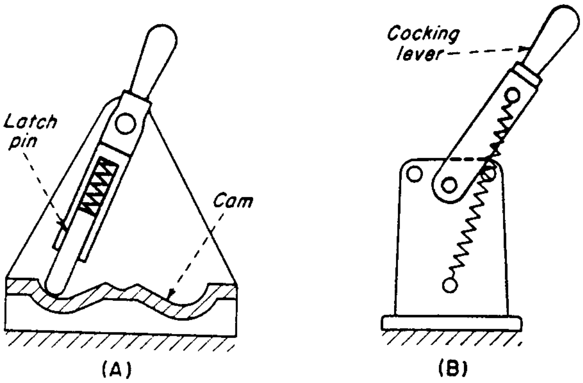
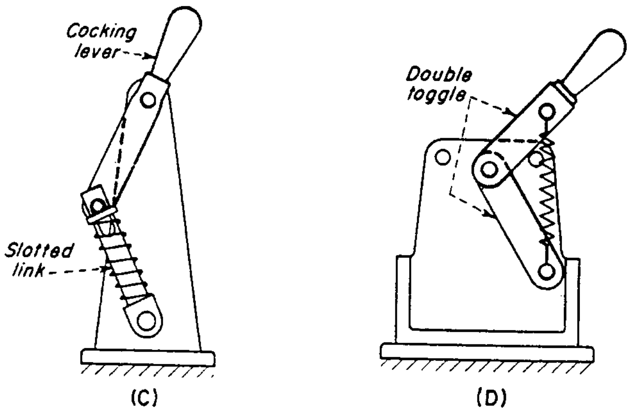


Fig. 1 Cam-guided latch (A) has one cocked, and two relaxed positions, (B) Simple overcenter toggle action. (C) An overcenter toggle with a slotted link. (D) A double toggle action often used in electrical switches.



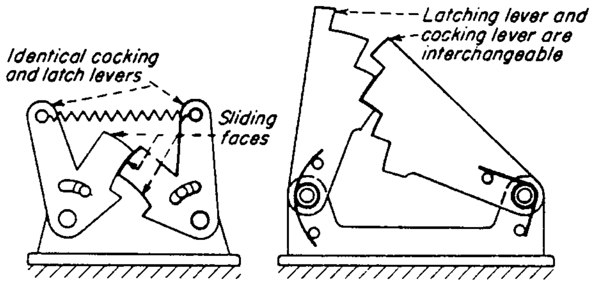


Fig. 2 An identically shaped cocking lever and latch (A) allow their functions to be interchangeable. The radii of the sliding faces must be dimensioned for a mating fit. The stepped latch (B) offers a choice of several locking positions.

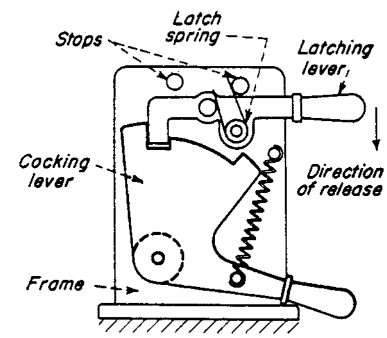


Fig. 3 A latch and cocking lever is spring-loaded so that latch movement releases the cocking lever. The cocked position can be held indefinitely. Studs in the frame provide stops, pivots, or mounts for the springs.

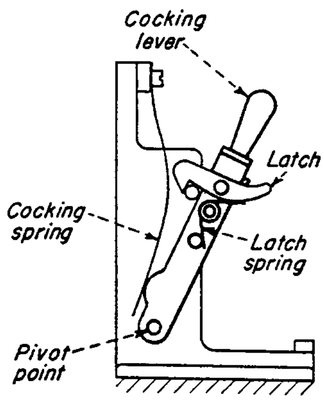


Fig. 4 A latch mounted on a cocking lever allows both levers to be reached at the same time with one hand. After release, the cocking spring initiates clockwise lever movement; then gravity takes over.

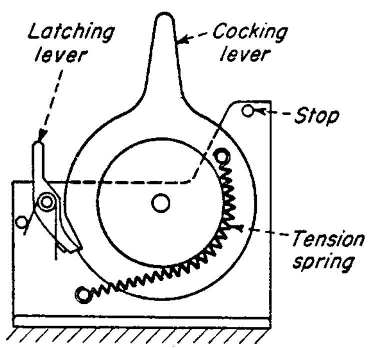


Fig. 5 A disk-shaped cocking has a tension spring resting against the cylindrical hub. Spring force always acts at a constant radius from the lever pivot point.

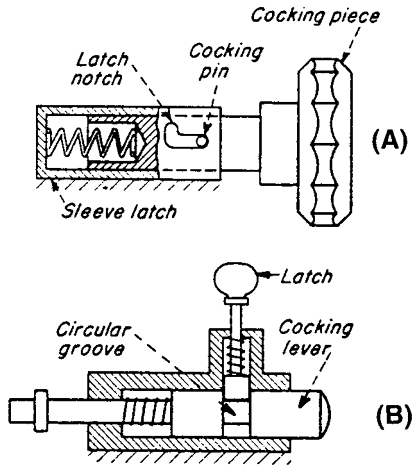


Fig. 6 A sleeve latch (A) as an L-shaped notch. A pin in the shaft rides in a notch. Cocking requires a simple push and twist action. (B) The Latch and plunger depend on axial movement for setting and release. A circular groove is needed if the plunger is to rotate.

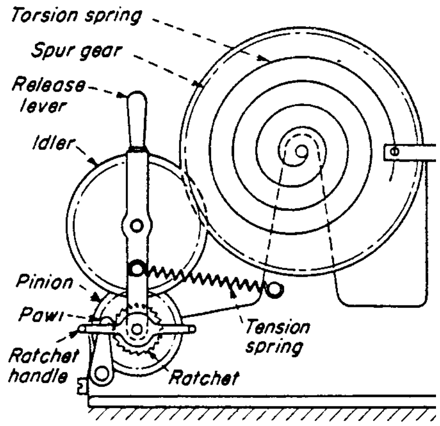
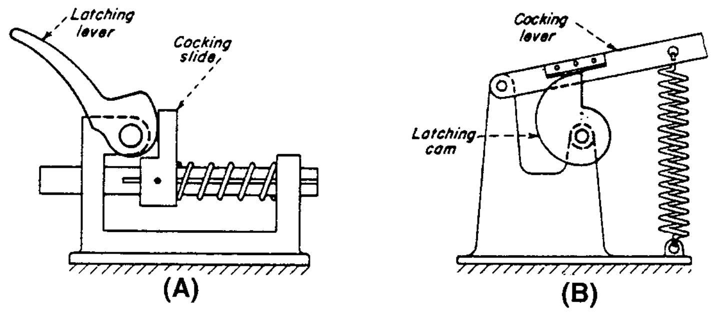


Fig. 7 A geared cocking device has a ratchet fixed to a pinion. A torsion spring exerts clockwise force on the spur gear; a tension spring holds the gar in mesh. The device is wound by turning the ratchet handle counterclockwise, which in turn winds the torsion spring. Moving the release-lever permits the spur gear to unwind to its original position without affecting the ratchet handle.

Fig. 8 In this overcenter lock (A) clockwise movement of the latching lever cocks and locks the slide. A counterclockwise movement is required to release the slide. (B) A latching-cam cocks and releases the cocking lever with the same counterclockwise movement as (A).



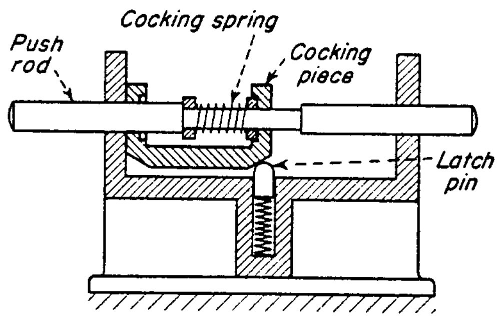


Fig. 9 A spring-loaded cocking piece has chamfered corners. Axial movement of the push-rod forces the cocking piece against a spring-loaded ball or pin set in a frame. When cocking builds up enough force to overcome the latch-spring, the cocking piece snaps over to the right. The action can be repeated in either direction.

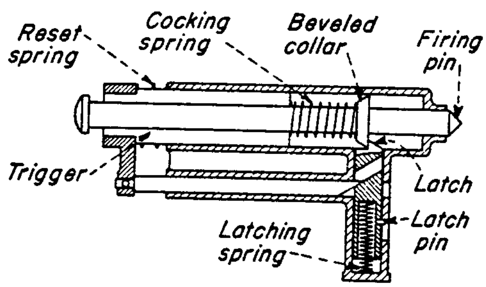


Fig. 10 A firing-pin mechanism has a beveled collar on a pin. Pressure on the trigger forces the latch down until it releases the collar when the pin snaps out, under the force of cocking the spring. A reset spring pulls the trigger and pin back. The latch is forced down by a beveled collar on a pin until it snaps back, after overcoming the force of the latch spring. (A latch pin retains the latch if the trigger and firing pin are removed.)

SIX SNAP-ACTION MECHANISMS

These diagrams show six basic ways to produce mechanical snap action.

Mechanical snap action results when a force is applied to a device over a period of time; buildup of this force to a critical level causes a sudden motion to occur. The ideal snap device would have no motion until the force reached a critical level. This, however, is not possible, and the way in which the mechanism approaches this ideal is a measure of its efficiency as a snap device. Some of the designs shown here approach the ideal closely; others do not, but they have other compensating good features.

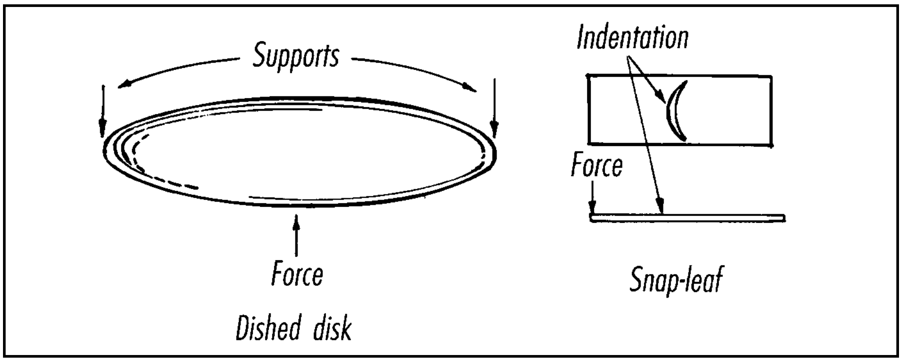


Fig. 1 A dished disk is a simple, common method for producing snap action. A snap leaf made from spring material can have various-shaped impressions stamped at the point where the overcentering action occurs. A "Frog clacker" is, of course, a typical applications. A bimetal element made in this way will reverse itself at a predetermined temperature.

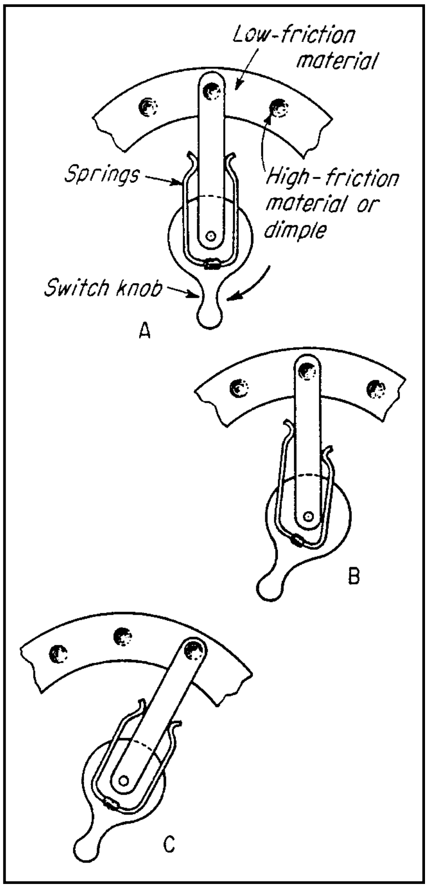


Fig. 2 Friction override can hold against an increasing load until friction is suddenly overcome. This is a useful action for small sensitive devices where large forces and movements are undesirable. This is the way we snap our fingers. That action is probably the original snap mechanism.

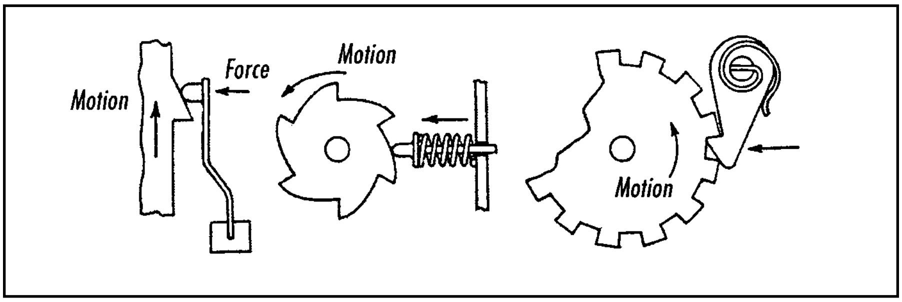


Fig. 3 A ratchet-and-pawl combination is probably the most widely used form of snap mechanism. Its many variations are an essential feature in practically every complicated mechanical device. By definition, however, this movement is not true snap-action.

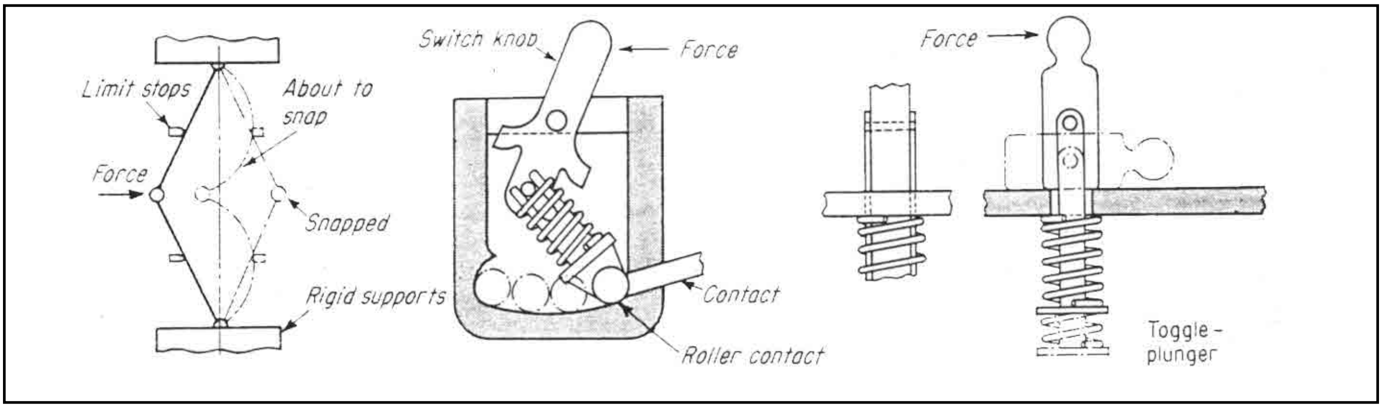


Fig. 4 Over-centering mechanisms find many applications in electrical switches. Considerable design ingenuity has been applied to fit this principle into many different mechanisms. It is the basis of most snap-action devices.

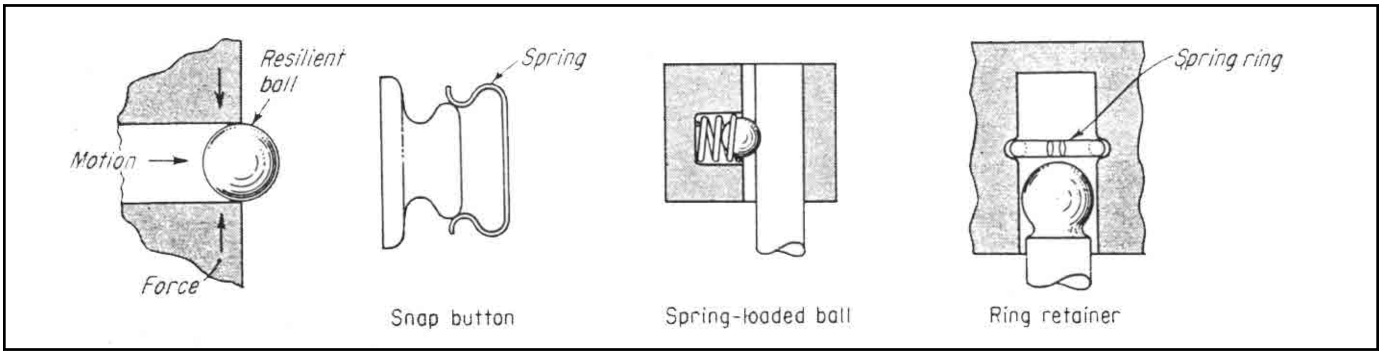
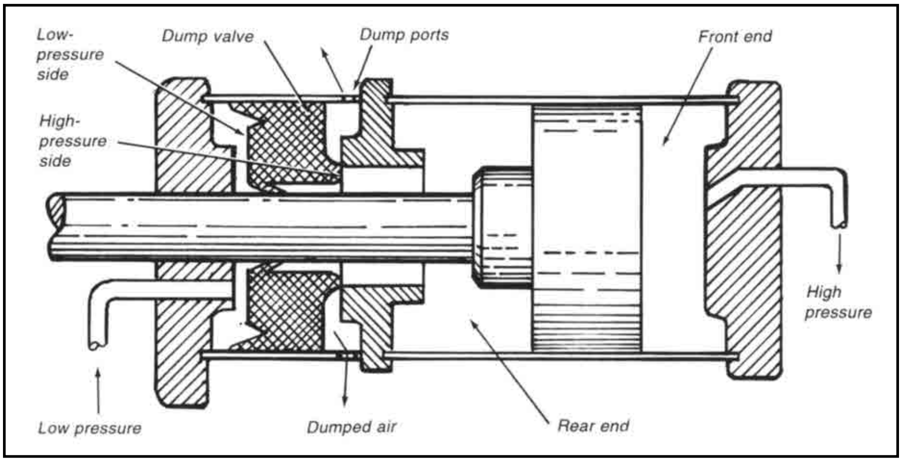


Fig. 5 The sphere ejection principle is based on snap buttons, spring-loaded balls and catches, and retaining-rings for fastening that must withstand repeated use. Their action can be designed to provide either easy or difficult removal. Wear can change the force required.

Fig. 6 A pneumatic dump valve produces snap action by preventing piston movement until air pressure has built up in the front end of the cylinder to a relatively high pressure. Dump-valve area in the low-pressure end is six times larger than its area on the high-pressure side. Thus the pressure required on the high-pressure side to dislodge the dump valve from its seat is six times that required on the low-pressure side to keep the valve properly seated.



EIGHT SNAP-ACTION DEVICES

Another selection of basic devices for obtaining sudden motion after a gradual buildup of force.

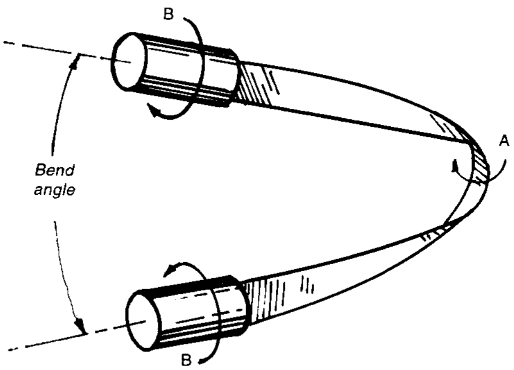


Fig. 1 A torsion ribbon bent as shown will turn “inside out” at A with a snap action when twisted at B. Design factors are ribbon width, thickness, and bend angle.

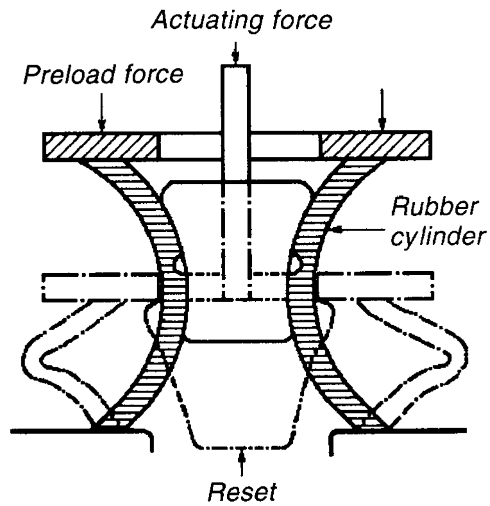


Fig. 2 A collapsing cylinder has elastic walls that can be deformed gradually until their stress changes from compressive to bending, with the resulting collapse of the cylinder.

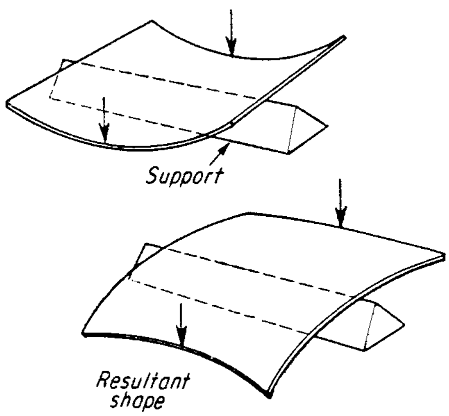


Fig. 3 A bowed spring will collapse into a new shape when it is loaded as shown A. A “push-pull” steel measuring tape illustrates this action; the curved material stiffens the tape so that it can be held out as a cantilever until excessive weight causes it to collapse suddenly.

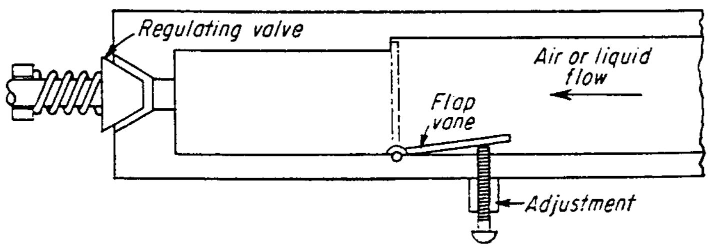


Fig. 4 A flap vane cuts off air or liquid flow at a limiting velocity. With a regulating valve, the vane will snap shut (because of increased velocity) when pressure is reduced below a design value.

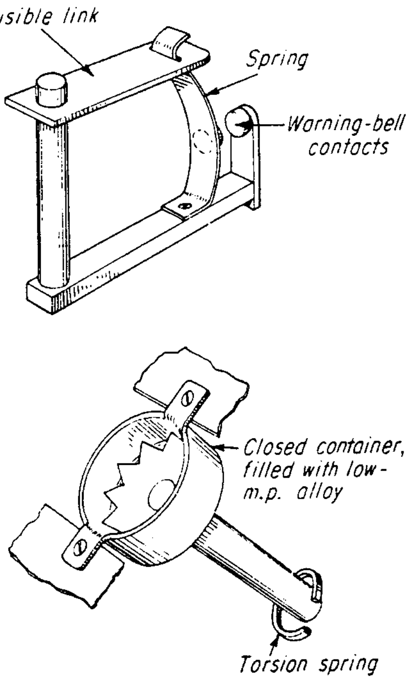


Fig. 5 A sacrificing link is useful where high temperature or corrosive chemicals would be hazardous. If the temperature becomes too high, or atmosphere too corrosive, the link will yield at design conditions. The device usually is required to act only once, although a device like the lower one can be quickly reset. However, it is restricted to temperature control.

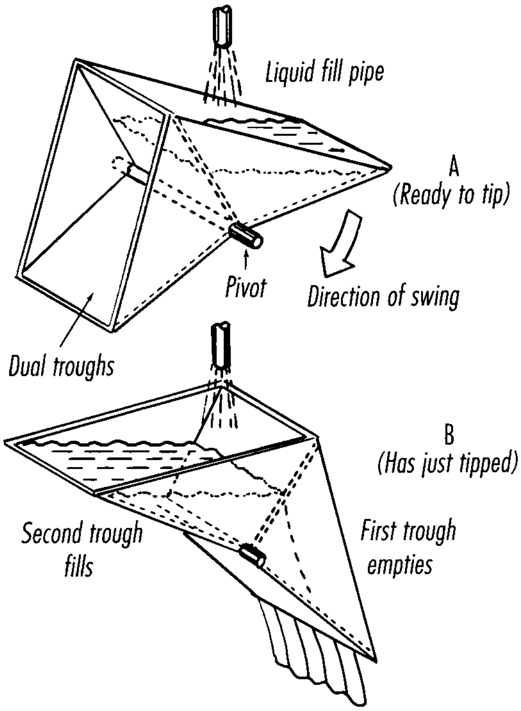


Fig. 6 Gravity-tips, although slower acting than most snap mechanisms, can be called snap mechanisms because they require an accumulation of energy to trigger an automatic release. A tripping trough that spreads sewerage is one example. As shown in A, it is ready to trip. When overbalanced, it trips rapidly, as in B.

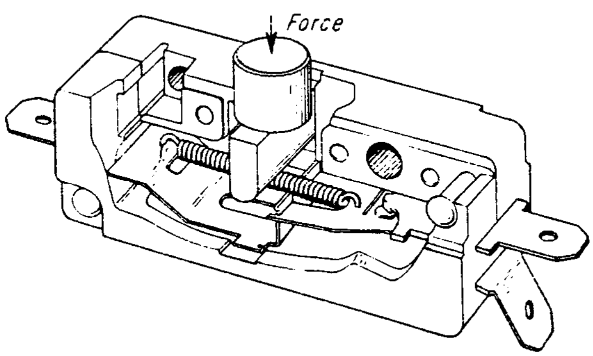


Fig. 7 An overcentering tension spring combined with a pivoted contact-strip is one arrangement used in switches. The example shown here is unusual because the actuating force bears on the spring itself.

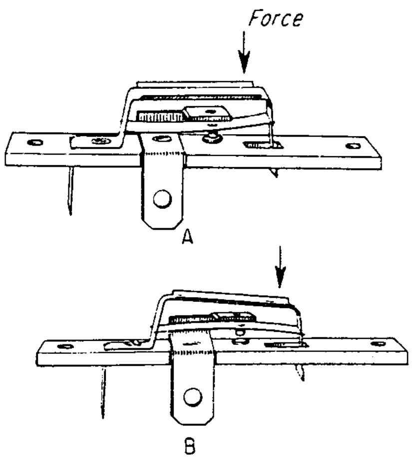


Fig. 8 An overcentering leaf-spring action is also the basis for many ingenious snap-action switches for electrical control. Sometimes spring action is combined with the thermostatic action of a bimetal strip to make the switch respond to heat or cold, either for control purposes or as a safety feature.

APPLICATIONS OF THE DIFFERENTIAL WINCH TO CONTROL SYSTEMS

Known for its mechanical advantage, the differential winch is a control mechanism that can supplement the gear and rack and four-bar linkage systems in changing rotary motion into linear. It can magnify displacement to meet the needs of delicate instruments or be varied almost at will to fulfill uncommon equations of motion.

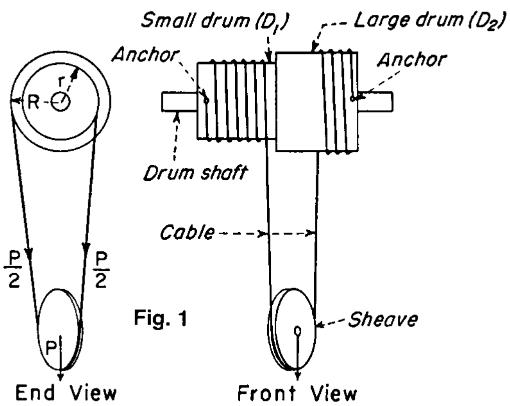
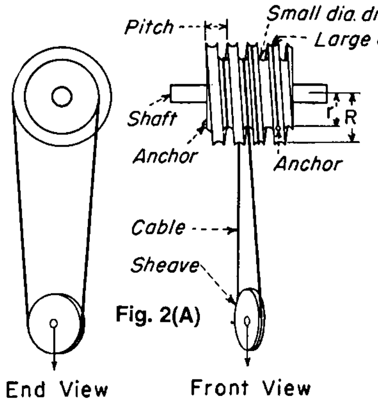


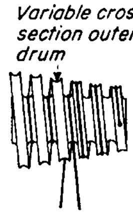
Fig. 1 A standard differential winch consists of two drums, D_1 and D_2, and a cable or chain which is anchored on both ends and wound clockwise around one drum and counterclockwise around the other. The cable supports a load-carrying sheave, and if the shaft is rotated clockwise, the cable, which unwinds from D_1 on to D_2, will raise the sheave a distance

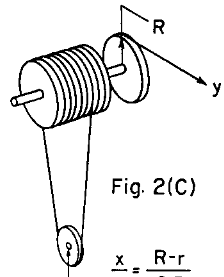
$$\text{Sheave rise/rev} = \frac{2\pi R - 2\pi r}{2} = \pi(R - r)$$

The winch, which is not in equilibrium exerts a counterclockwise torque.

$$\text{Unbalanced torque} = \frac{P}{2}(R - r)$$







$$\frac{x}{y} = \frac{R-r}{2R}$$

R	r	x/y
3	$2\frac{15}{16}$	1/96
1	$\frac{63}{64}$	1/128

Fig. 2(A) Hulse Differential Winch*. Two drums, which are in the form of worm threads contoured to guide the cables, concentrically occupy the same logitudinal space. This keeps the cables approximately at right angles to the shaft and eliminates cable shifting and rubbing, especially when used with variable cross sections as in Fig. 2(B). Any equation of motion can be satisfied by choosing suitable cross sections for the drums. Methods for resisting or supporting the axial thrust should be considered in some installations. Fig. 2(C) shows typical reductions in displacement. *Pat. No. 2,590,623

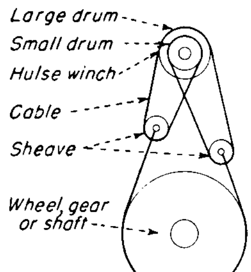


Fig. 3(B)

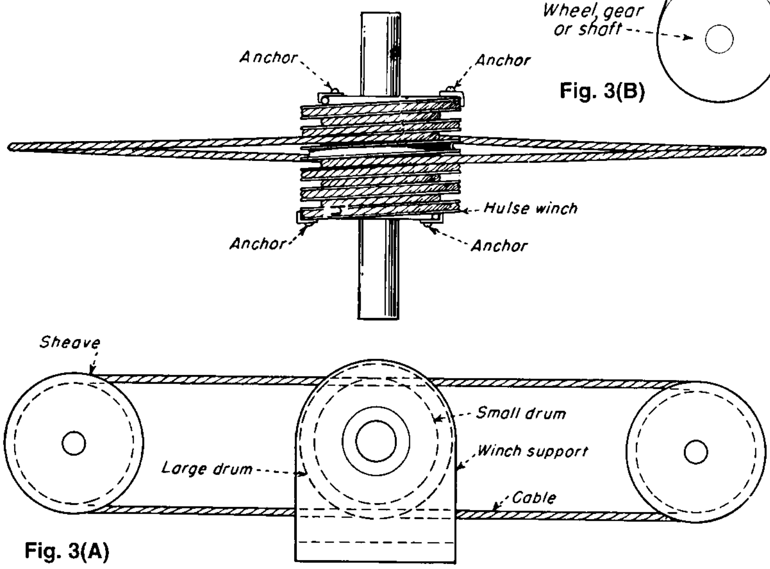


Fig. 3(A)

Fig. 3(A) A Hulse Winch with opposing sheaves. This arrangement, which uses two separate cables and four anchor points, can be considered as two winches back-to-back with one common set of drums. Variations in motion can be obtained by: (1) restraining in the sheaves so that when the system is rotated the drums will travel toward one of the sheaves; (2) restraining the drums and allowing the sheaves to travel. The distance between the sheaves will remain constant and is usually connected by a bar; (3) permitting the drums to move axially while restraining them transversely. When the system is rotated, drums will travel axially one pitch per revolution, and sheaves remain in the same plane perpendicular to the drum axis. This variation can be reversed by allowing sheaves to move axially; and (4) sheaves need not be opposite but can be arranged as in Fig. 3(B) to rotate a wheel.

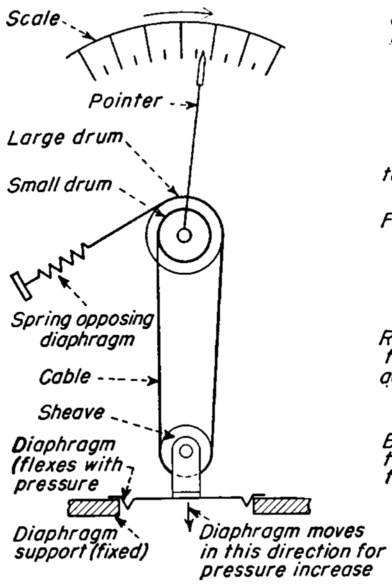


Fig. 4(A)

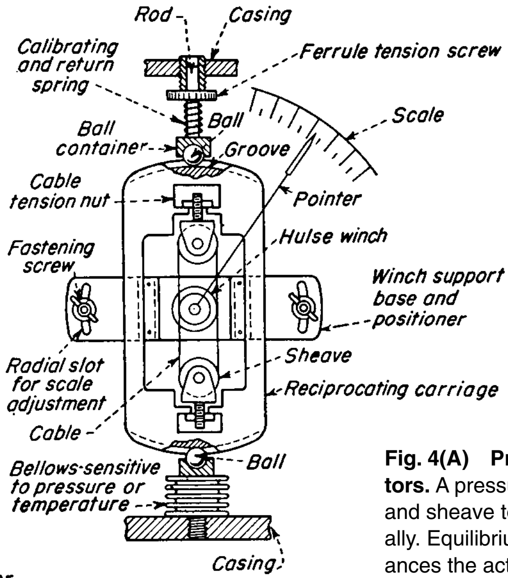


Fig. 4(B)

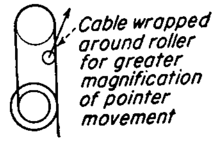


Fig. 4(C)

Fig. 4(A) Pressures and temperature indicators. A pressure change causes the diaphragm and sheave to move vertically and the pointer radially. Equilibrium occurs when the spring force balances the actuating torque. Replacing the diaphragm with a thermal element changes the instrument into a temperature indicator. Two sheaves and a reciprocating carriage, Fig. 4(B), are based on the principle shown in Fig. 3(A). A carriage is activated by pressure or temperature and is balanced by a spring force in the opposite end. Further magnification can be obtained, Fig. 4(C), by wrapping a cable around the roller to which the pointer is attached.

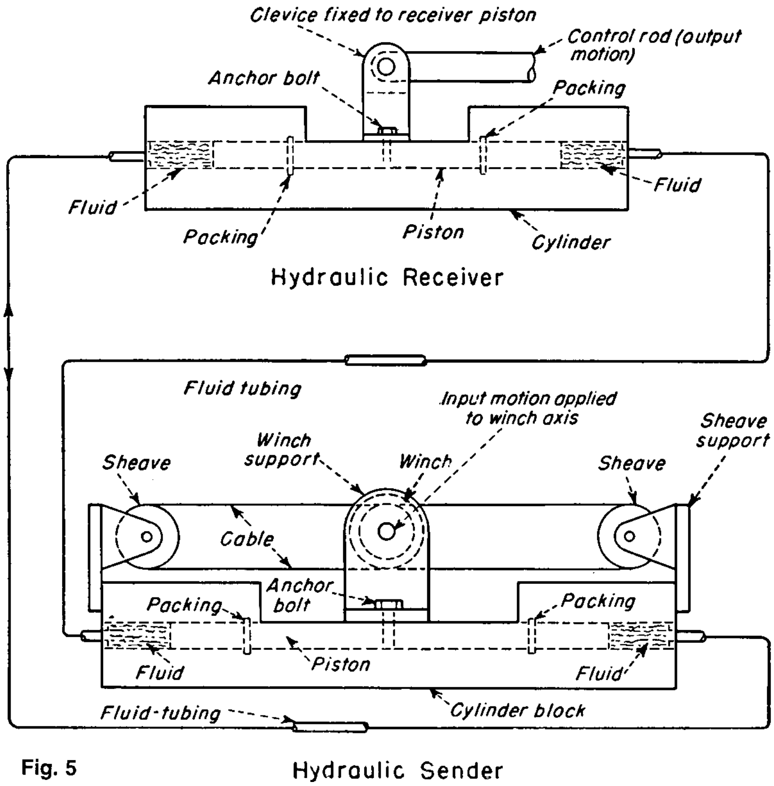


Fig. 5

Fig. 5 A hydraulic control system, actuated by a differential winch, performs remote precision positioning of a control rod with a minimum of applied torque. The sending piston, retained in a cylinder block, reciprocates back and forth from a torque applied to the winch shaft. Fluid is forced out from one end of the cylinder through the pipe lines to displace the receiving piston, which in turn activates a control rod. The receiver simultaneously displaces a similar amount of fluid from the opposite end back to the sender. By suitable valving, the sender can become a double-acting pump.

SIX APPLICATIONS FOR MECHANICAL POWER AMPLIFIERS

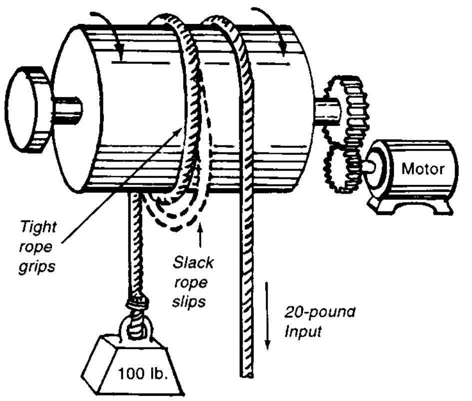
Precise positioning and movement of heavy loads are two basic jobs for this all-mechanical torque booster.

This mechanical power amplifier has a fast response. Power from its continuously rotating drums is instantaneously available. When used for position-control applications, pneumatic, hydraulic, and electrical systems—even those with continuously running power sources—require transducers to change signals from one energy form to another. The mechanical power amplifier, on the other hand, permits direct sensing of the controlled motion.

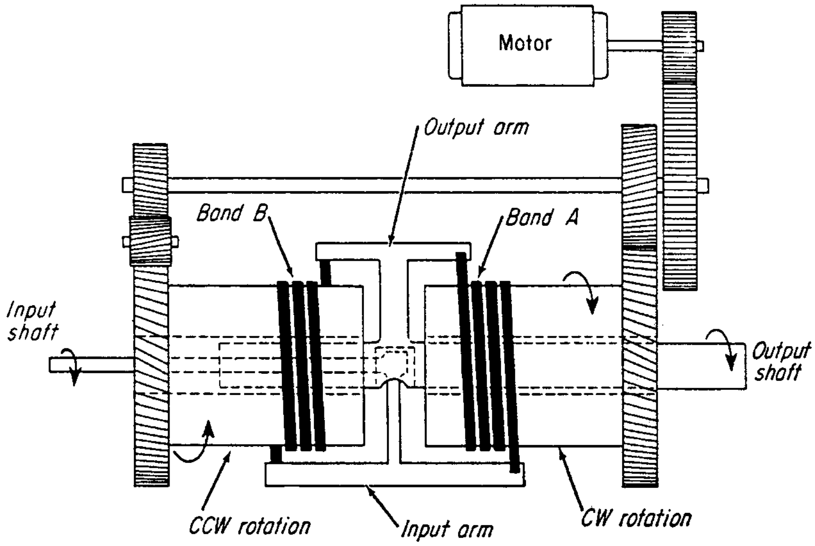
Four major advantages of this all-mechanical device are:

1. Kinetic energy of the power source is continuously available for rapid response.
2. Motion can be duplicated and power amplified without converting energy forms.
3. Position and rate feedback are inherent design characteristics.
4. Zero slip between input and output eliminates the possibility of cumulative error.

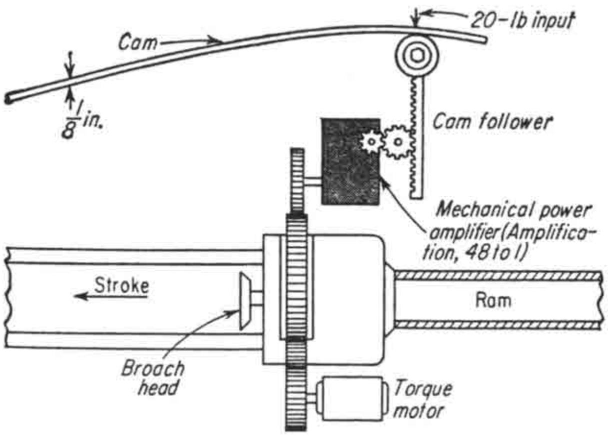
One other important advantage is the ease with which this device can be adapted to perform special functions—jobs for which other types of systems would require the addition of more costly and perhaps less reliable components. The six applications which follow illustrate how those advantages have been put to work in solving widely divergent problems.



The capstan principle is the basis for the mechanical power amplifier described here that combines two counterrotating drums. The drums are continuously rotating but only transmit torque when the input shaft is rotated to tighten the band on drum A. Overrun of output is stopped by drum B, when overrun tightens the band on this drum.



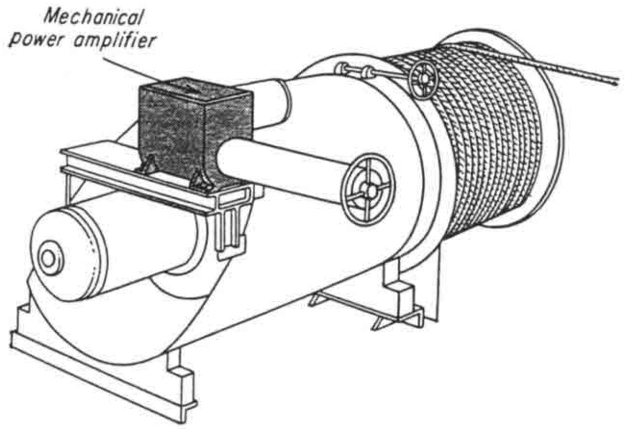
A capstan is a simple mechanical amplifier—rope wound on a motor-driven drum slips until slack is taken up on the free end. The force needed on the free end to lift the load depends on the coefficient of friction and the number of turns of rope. By connecting bands A and B to an input shaft and arm, the power amplifier provides an output in both directions, plus accurate angular positioning. When the input shaft is turned clockwise, the input arm takes up the slack on band A, locking it to its drum. Because the load end of locked band A is connected to the output arm, it transmits the CW motion of the driven drum on which it is wound to the output shaft. Band B therefore slacks off and slips on its drum. When the CW motion of the input shaft stops, tension on band A is released and it slips on its own drum. If the output shaft tries to overrun, the output arm will apply tension to band B, causing it to tighten on the CCW rotating rum and stop the shaft.



1. Nonlinear Broaching

Problem: In broaching large-fore rifles, the twist given to the lands and grooves represents a nonlinear function of barrel length. Development work on such rifles usually requires some experimentation with this function. At present, rotation of the broaching head is performed by a purely mechanical arrangement consisting of a long, heavy wedge-type cam and appropriate gearing. For steep twist angles, however, the forces acting on this mechanism become extremely high.

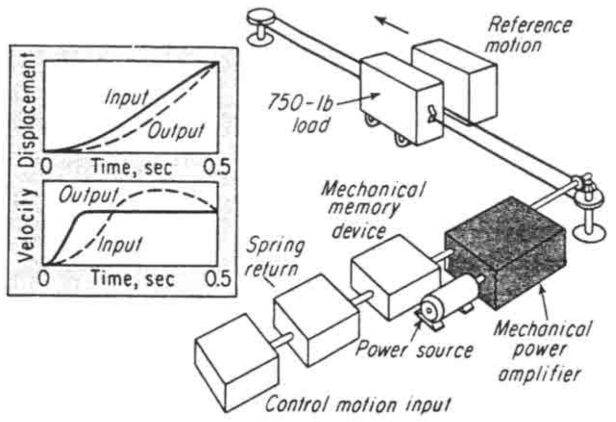
Solution: A suitable mechanical power amplifier, with its inherent position feedback, was added to the existing mechanical arrangement, as shown in Fig. 1. The cam and follower, instead of having to drive the broaching head, simply furnish enough torque to position the input shaft of the amplifier.



2. Hydraulic Winch Control

Problem: Hydraulic pump-motor systems are excellent for controlling position and motion at high power levels. In the 10- to 150-hp range, for example, the usual approach is to vary the output of a positive displacement pump in a closed-loop hydraulic circuit. In many of the systems that might be able to control this displacement, however, a force feedback proportional to system pressure can lead to serious errors or even oscillations.

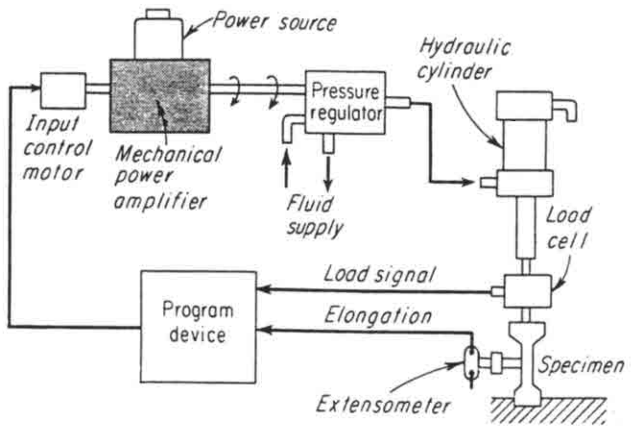
Solution: Figure 2 shows an external view of the complete package. The output shaft of the mechanical power amplifier controls pump displacement, while its input is controlled by hand. In a more recent development requiring remote manual control, a servomotor replaces this local handwheel. Approximately 10 lb-in. torque drives a 600 lb-in. load. If this system had to transmit 600 lb-in., the equipment would be more expensive and more dangerous to operate.



3. Load Positioning

Problem: It was necessary for a 750-lb load to be accelerated from standstill in 0.5 s and brought into speed and position synchronization with a reference linear motion. It was also necessary that the source of control motion be permitted to accelerate more rapidly than the load itself. Torque applied to the load could not be limited by any kind of slipping device.

Solution: A system with a single mechanical power amplifier provided the solution (Fig. 3). A mechanical memory device, preloaded for either rotation, drives the input shaft of the amplifier. This permits the input source to accelerate as rapidly as desired. The total control input travel minus the input travel of the amplifier shaft is temporarily stored. After 0.5 seconds, the load reaches proper speed, and the memory device transmits position information in exact synchronization with the input.



4. Tensile Testing Machine

Problem: On a hydraulic tensile testing machine, the stroke of the power cylinder had to be controlled as a function of two variables: tension in, and extension of, the test specimen. A programming device, designed to provide a control signal proportional to these variables, had an output power level of about 0.001 hp—too low to drive the pressure regulator controlling the flow to the cylinder.

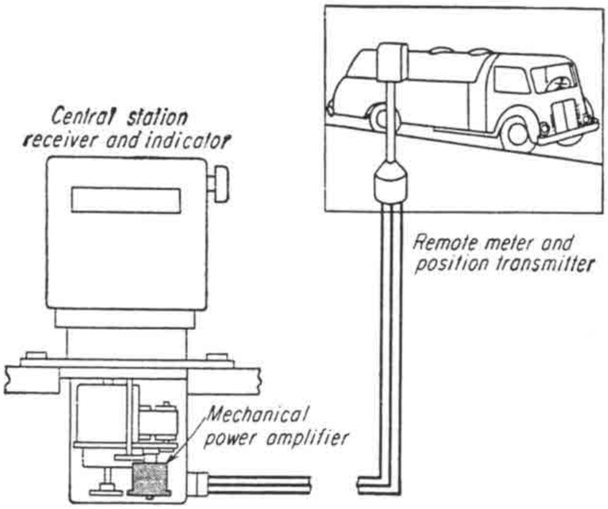
Solution: An analysis of the problem revealed three requirements: the output of the programmer had to be amplified about 60 times, position accuracy had to be within 2º, and acceleration had to be held at a very low value. A mechanical power amplifier satisfied all three requirements. Figure 4 illustrates the completed system. Its design is based principally on steady-state characteristics.

5. Remote Metering and Counting

Problem: For a remote, liquid-metering job, synchro systems had been used to transmit remote meter readings to a central station and repeat this information on local indicating counters. The operation involved a large number of meters and indicators. As new equipment (e.g. ticket printers) was added, the torque requirement also grew.

Solution: Mechanical power amplifiers in the central station indicators not only supplied extra output torque but also made it possible to specify synchros that were even smaller than those originally selected to drive the indicators alone (see Fig. 5).

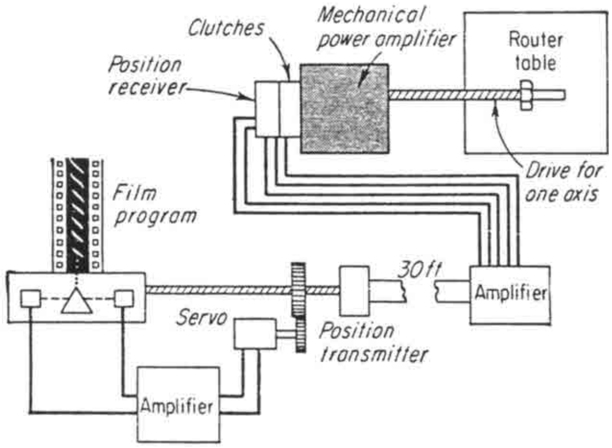
The synchro transmitters selected operate at a maximum speed of 600 rpm and produce only about 3 oz-in. of torque. The mechanical power amplifiers furnish up to 100 lb-in. of torque, and are designed to fit in the bottom of the registers shown in Fig. 5. Total accuracy is within 0.25 gallon, and error is noncumulative.



6. Irregular Routing

Problem: To control remotely the table position of a routing machine from information stored on a film strip. The servoloop developed to interpret this information produced only about 1 oz.-in. of torque. About 20 lb.-ft was required at the table feedscrew.

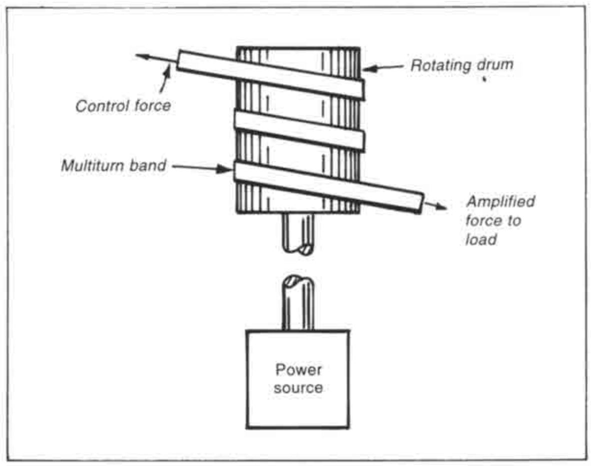
Solution: Figure 6 shows how a mechanical power amplifier supplied the necessary torque at the remote table location. A position transmitter converts the rotary motion output of the servoloop to a proportional electrical signal and sends it to a differential amplifier at the machine location. A position receiver, geared to the output shaft, provides a signal proportional to table position. The differential amplifier compares these, amplifies the difference, and sends a signal t either counterrotating electromagnetic clutch, which drives the input shaft of the mechanical power amplifier.

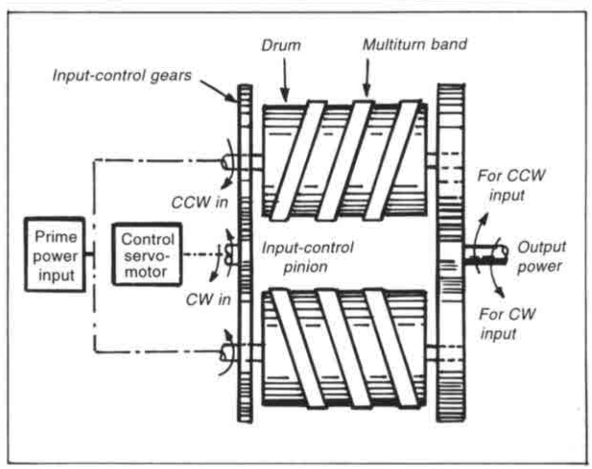


A mechanical power amplifier that drives a crossfeed slide is based on the principle of the windlass. By varying the control force, all or any part of power to the drum can be used.

Two drums mounted back to back supply the bi-directional power needed in servo systems. Replacing the operator with a two-phase induction servomotor permits electronic or magnetic signal amplification. A rotating input avoids a linear input and output of the simple windlass. Control and output ends of the multiturn bands are both connected to gears mounted concentrically with the drum axis.

When the servomotor rotates the control gear, it locks the band-drum combination, forcing output gear to rotate with it. Clockwise rotation of the servomotor produces CW power output while the second drum idles. Varying the servo speed, by changing servo voltage, varies output speed.





VARIABLE-SPEED BELT AND CHAIN DRIVES

Variable-speed drives provide an infinite number of speed ratios within a specific range. They differ from the stepped-pulley drives in that the stepped drives offer only a discrete number of velocity ratios.

Mechanical "all-metal" drives employ friction or preloaded cones, disks, rings, and spheres, which undergo a certain amount of slippage. Belt drives, on the other hand, have little slippage or frictional losses, and chain has none—it maintains a fixed phase relationship between the input and output shafts.

Belt Drives

Belt drives offer high efficiency and are relatively low in price. Most use V-belts, reinforced by steel wires to 3 inches in width.

Speed adjustment in belt drives is obtained through one of the four basic arrangements shown below.

Variable-distance system (Fig. 1). A variable-pitch sheave on the input shaft opposes a solid (fixed-pitch) sheave on the output shaft. To vary the speed, the center distance is varied, usually by an adjustable base, tilting or sliding motor (Fig. 6).

Speed variations up to 4:1 are easily achieved, but torque and horsepower characteristics depend on the location of the variable-diameter sheave.

Fixed-distance system (Fig. 2). Variable-pitch sheaves on both input and output shafts maintain a constant center distance between shafts. The sheaves are controlled by linkage. Either the pitch diameter of one sheave is positively controlled and the disks of the other sheave under spring tension, adjust automatically or the pitch diameters of both sheaves are positively controlled by the linkage system (Fig. 5). Pratt & Whittney has applied the system in Fig. 5 to the spindle drive of numerically controlled machines.

Speed variations up to 11:1 are obtained, which means that with a 1200-rpm motor, the maximum output speed will be $1200\sqrt{11} = 3984$ rpm, and the minimum output speed = 3984/11 = 362 rpm.

Double-reduction system (Fig. 3). Solid sheaves are on both the input and output shafts, but both sheaves on the intermediate shaft are of variable-pitch type. The center distance between input and output is constant.

Coaxial shaft system (Fig. 4). The intermediate shaft in this arrangement permits the output shaft to be coaxial with that of the input shaft. To maintain a fixed center distance, all four sheaves must be of the variable-pitch type and controlled by linkage, similar to the system in Fig. 6. Speed variation up to 16:1 is available.

Packaged belt units (Fig. 7). These combine the motor and variable-pitch transmissions as an integral unit. The belts are usually ribbed, and speed ratios can be dialed by a handle.

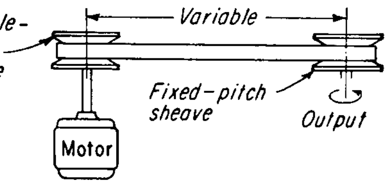


Fig. 1

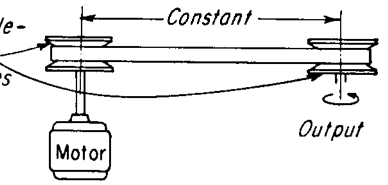


Fig. 2

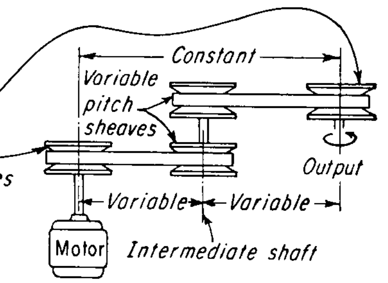


Fig. 3

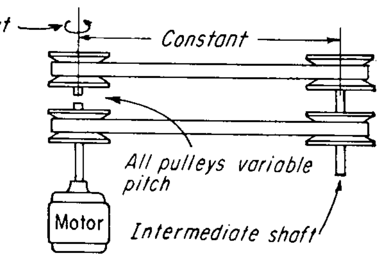


Fig. 4

Four basic belt arrangements for varying output speed.

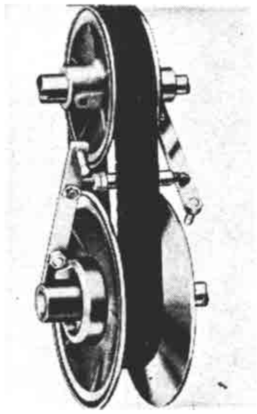


Fig. 5 Linkage controlled pulleys.

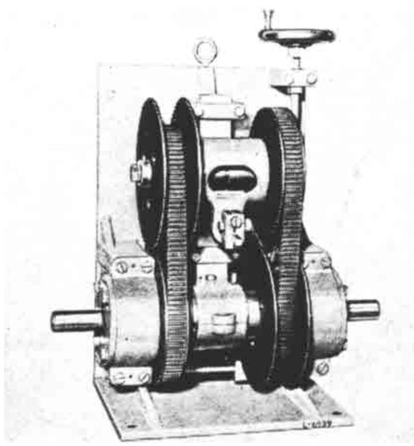
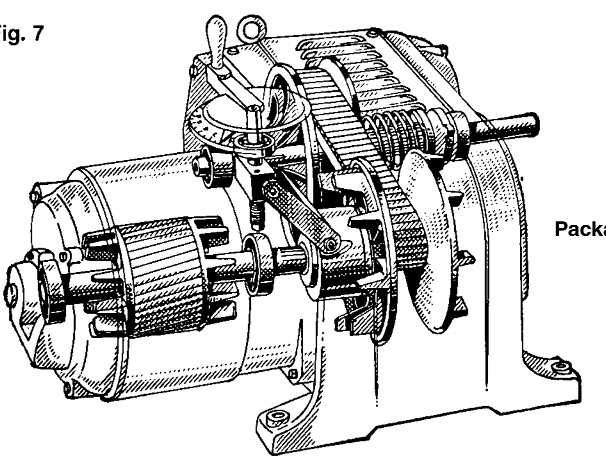


Fig. 6 Tandem arrangement employs dual belt-system to produce high speed-reduction.

Fig. 7



Packaged belt unit.

Sheave Drives

The axial shifting of variable-pitch sheaves is controlled by one of four methods:

Linkage actuation. The sheave assemblies in Fig. 5 are directly controlled by linkages which, in turn, are manually adjusted.

Spring pressure. The cons of the sheaves in Figs. 2 and 4 are axially loaded by spring force. A typical pulley of this type is illustrated in Fig. 8. These pulleys are used in conjunction with directly controlled sheaves, or with variable center-distance arrangements.

Cam-controlled sheave. The cones of this sheave (Fig. 9) are mounted on a floating sheave, free to rotate on the pulley spindle. Belt force rotates the cones, whose surfaces are cammed by the inclined plane of the spring. The camming action wedges the cones against the belt, thus providing sufficient pressure to prevent slippage at the higher speeds, as shown in the curve.

Centrifugal-force actuator. In this unique sheave arrangement (Fig. 10) the pitch diameter of the driving sheave is controlled by the centrifugal force of steel balls. Another variable-pitch pulley mounted on the driven shaft is responsive to the torque. As the drive speed increases, the centrifugal force of the balls forces the sides of the driving sheave together. With a change in load, the movable flange of the driven sheave rotates in relation to the fixed flange. The differential rotation of the sheave flanges cams them together and forces the V-belt to the outer edge of the driven sheave, which has a lower transmission ratio. The driving sheave is also shifted as the load rises with decreasing speed. With a stall load, it is moved to the idling position. When the torque-responsive sheave is the driving member, any increase in drive speed closes its flanges and opens the flanges of the centrifugal member, thus maintaining a constant output speed. The drive has performed well in transmissions with ratings ranging from 2 to 12 hp.

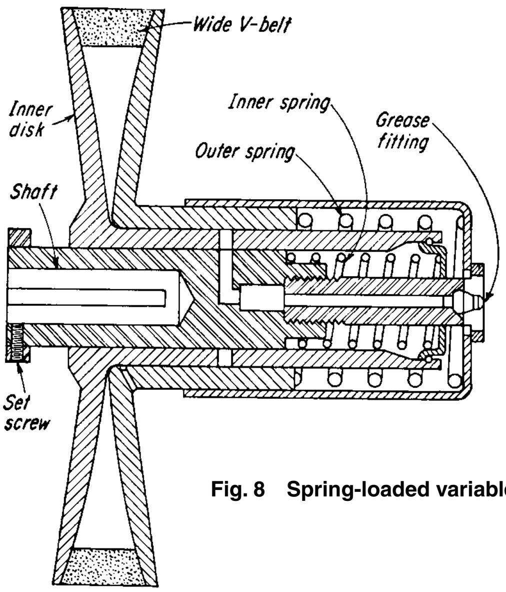


Fig. 8 Spring-loaded variable-pitch pulley.

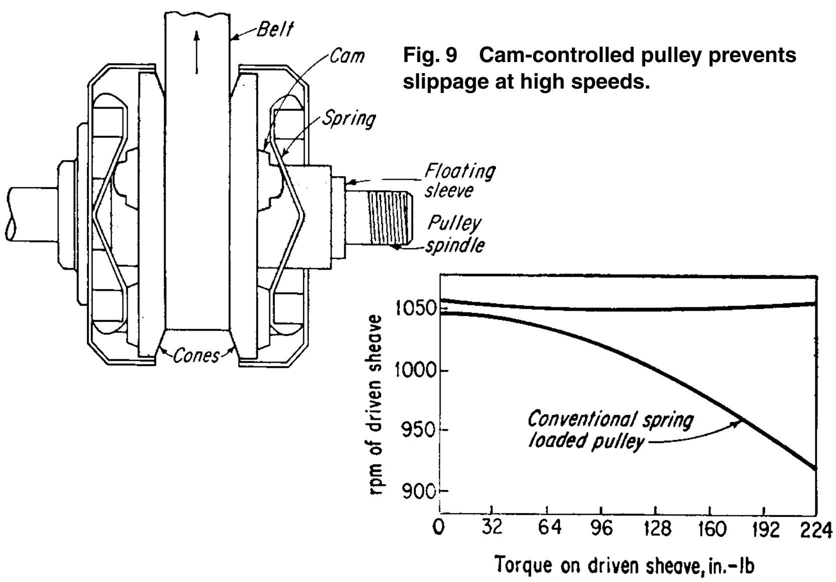
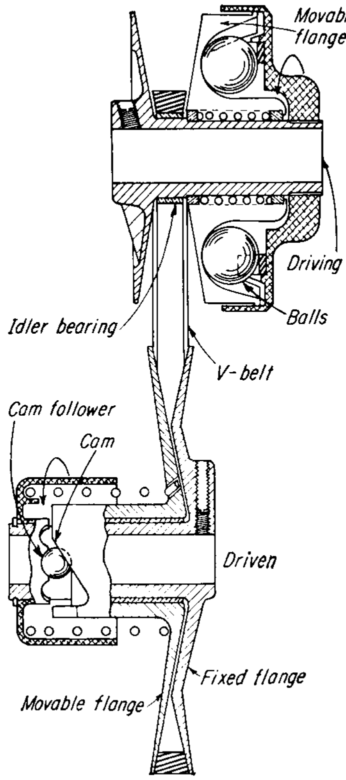


Fig. 9 Cam-controlled pulley prevents slippage at high speeds.

Fig. 10 A ball-controlled pulley has its sides pressured by centrifugal force.



Chain Drives

PIV drive (Fig. 11). This chain drive (positive, infinitely variable) eliminates any slippage between the self-forming laminated chain teeth and the chain sheaves. The individual laminations are free to slide laterally to take up the full width of the sheave. The chain runs in radially grooved faces of conical surface sheaves which are located on the input and output shafts. The faces are not straight cones, but have a slight convex curve to maintain proper chain tension at all positions. The pitch diameters of both sheaves are positively controlled by the linkage. Booth action is positive throughout operating range. It is rated to 25 hp with speed variation of 6:1.

Double-roller chain drive (Fig. 12). This specially developed chain is built for capacities to 22 hp. The hardened rollers are wedged between the hardened conical sides of the variable-pitch sheaves. Radial rolling friction results in smooth chain engagement.

Single-roller chain drive (Fig. 13). The double strand of this chain boosts the capacity to 50 hp. The scissor-lever control system maintains the proper proportion of forces at each pair of sheave faces throughout the range.

Fig. 11 A PIV drive chain grips radially grooved faces of a variable-pitch sheave to prevent slippage.

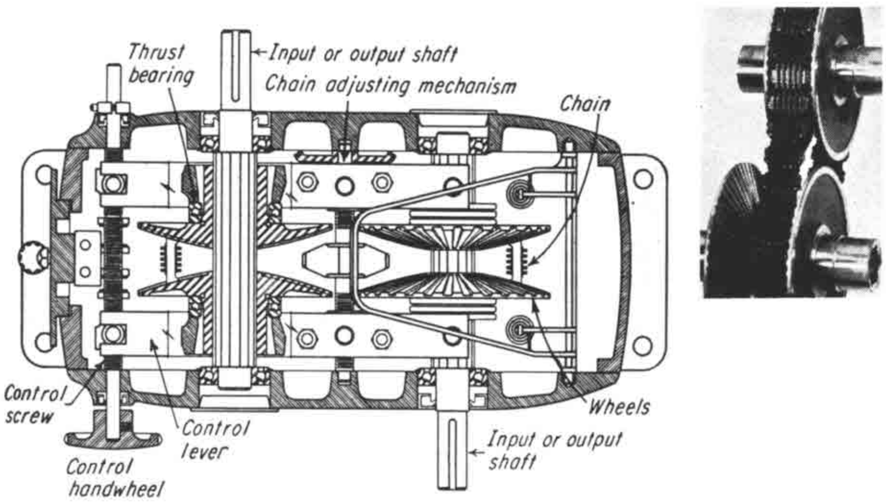
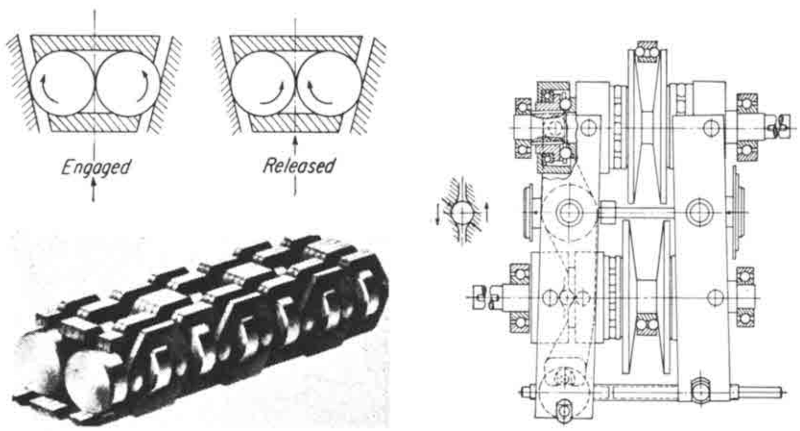
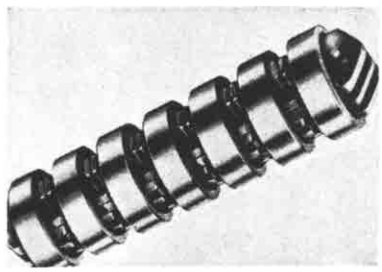


Fig. 12 A double-roller chain drive combines strength with ease in changing speed.





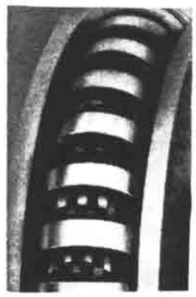
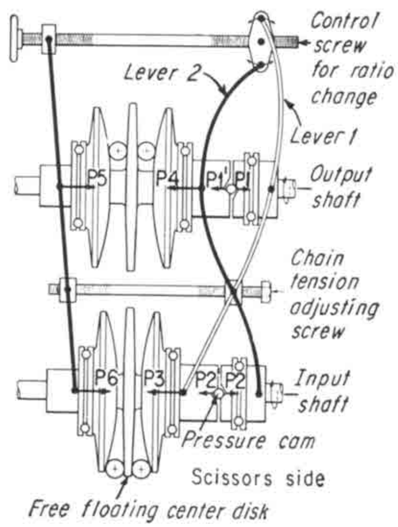


Fig. 13 A single-roller chain drive for high horse-power applications.



GETTING IN STEP WITH HYBRID BELTS

Imaginative fusions of belts, cables, gears and chains are expanding the horizons for light-duty synchronous drives

Belts have long been used for the transfer of mechanical power. Today's familiar flat belts and V-belts are relatively light, quiet, inexpensive, and tolerant of alignment errors. They transmit power solely through frictional contacts. However, they function best at moderate speeds (4000 to 6000 fpm) under static loads. Their efficiencies drop slightly at low speeds, and centrifugal effects limit their capacities at high speeds. Moreover, they are inclined to sip under shock loads or when starting and braking. Even under constant rotation, standard belts tend to creep. Thus, these drives must be kept under tension to function properly, increasing loads on pulley shaft bearings.

Gears and chains, on the other hand, transmit power through bearing forces between positively engaged surfaces. They do not slip or creep, as measured by the relative motions of the driving and driven shafts. But the contacts themselves can slip significantly as the chain rollers and gear teeth move in and out of mesh.

Positive drives are also very sensitive to the geometries of the mating surfaces. A gear's load is borne by one or two teeth, thus magnifying small tooth-to-tooth errors. A chain's load is more widely distributed, but chordal variations in the driving wheel's effective radius produce small oscillations in the chain's velocity.

To withstand these stresses, chains and gears must be carefully made from hard materials and must then be lubri-

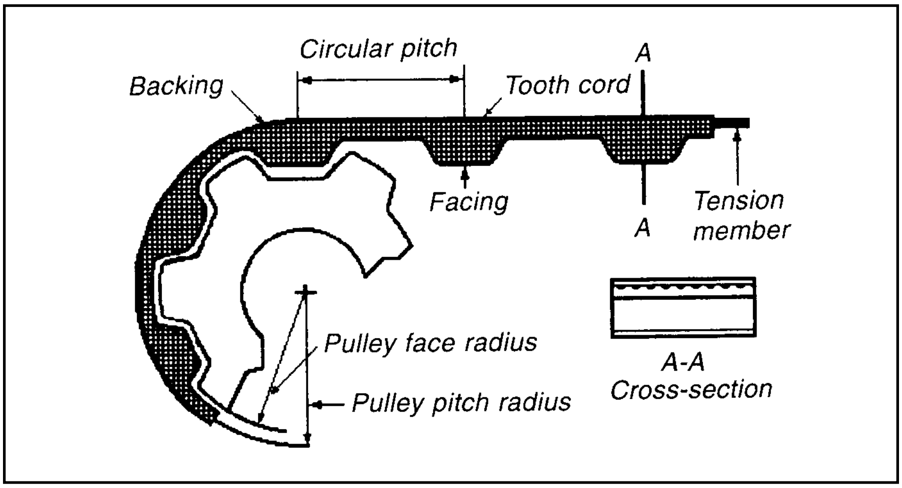


Fig. 1 Conventional timing belts have fiberglass or polyester tension members, bodies of neoprene or polyurethane, and trapezoidal tooth profiles.

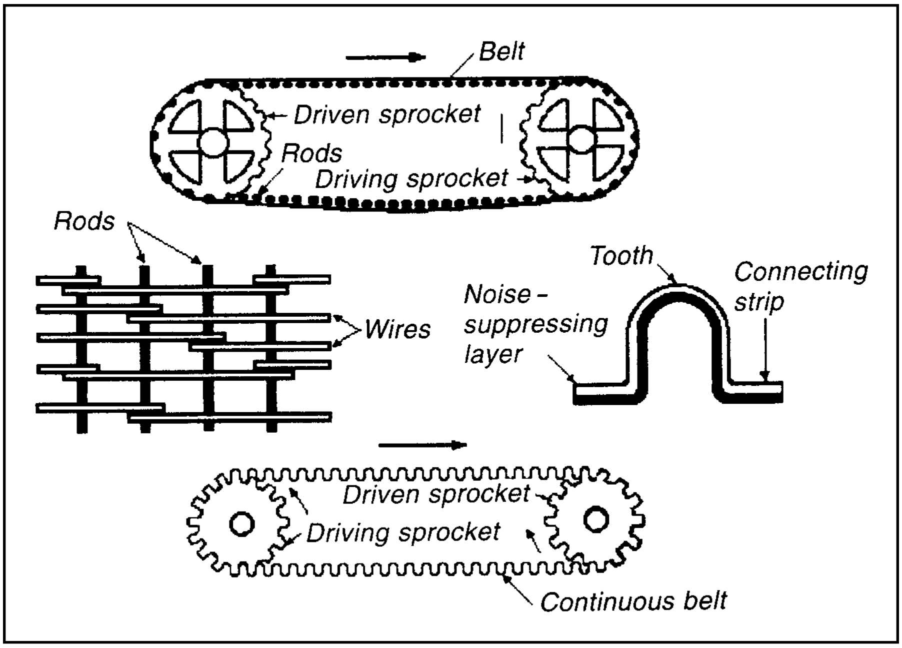


Fig. 2 NASA metal timing belts exploit stainless steel's strength and flexibility, and are coated with sound-and friction-reducing plastic.

cated in operations. Nevertheless, their operating noise betrays sharp impacts and friction between mating surfaces.

The cogged timing belt, with its trapezoidal teeth (Fig. 1), is the best-known fusion of belt, gear, and chain. Though these well-established timing belts can handle high powers (up to 800 hp), many of the newer ideas in synchronous belting have been incorporated into low and fractional horsepower drives for instruments and business machines.

Steel Belts for Reliability

Researchers at NASA's Goddard Space Flight Center (Greenbelt, MD) turned to steel in the construction of long-lived toothed transmission belts for spacecraft instrument drives.

The NASA engineers looked for a belt design that would retain its strength and hold together for long periods of sustained or intermittent operation in hostile environments, including extremes of heat and cold.

Two steel designs emerged. In the more chain-like version (Fig. 2A), wires running along the length of the belt are wrapped at intervals around heavier rods running across the belt. The rods do double duty, serving as link pins and as teeth that mesh with cylindrical recesses cut into the sprocket. The assembled belt is coated with plastic to reduce noise and wear.

In the second design (Fig. 2B), a strip of steel is bent into a series of U-shaped teeth. The steel is supple enough to flex as it runs around the sprocket with its protruding transverse ridges, but the material resists stretching. This belt, too, is plastic-coated to reduce wear and noise.

The V-belt is best formed from a continuous strip of stainless steel "not much thicker than a razor blade," according to the agency, but a variation can be made by welding several segments together.

NASA has patented both belts, which are now available for commercial licensing. Researchers predict that they will be particularly useful in machines that must be dismantled to uncover the belt pulleys, in permanently encased machines, and in machines installed in remote places. In addition, stainless-steel belts might find a place in high-precision instrument drives because they neither stretch nor slip.

Though plastic-and-cable belts don't have the strength or durability of the NASA steel belts, they do offer versatility and production-line economy. One of the least expensive and most adaptable is

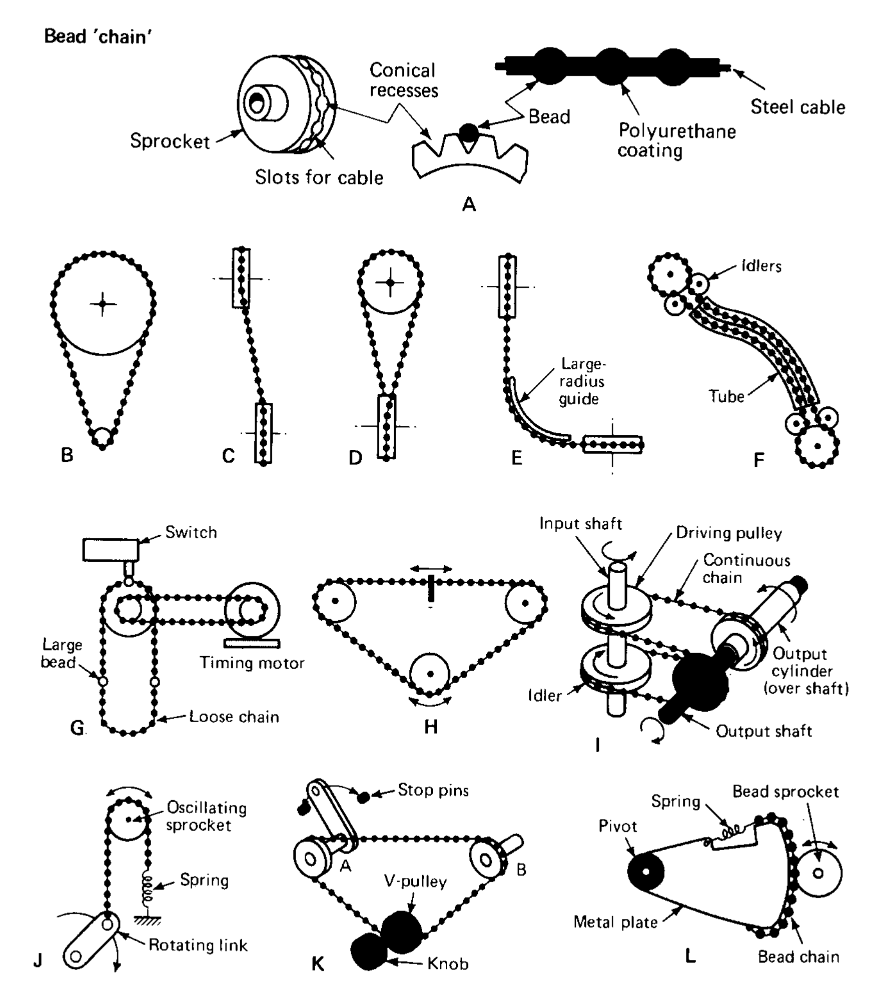


Fig. 3 Polyurethane-coated steel-cable "chains"—both beaded and 4-pinned—can cope with conditions unsuitable for most conventional belts and chains.

Table 1. Conventional Timing Belts

Type	Circular pitch, in.	Wkg. tension lb/in. width	Centr. loss const., K_c
Standard (Fig 1)			
MXL	0.080	32	10×10^{-9}
XL	0.200	41	27×10^{-9}
L	0.375	55	38×10^{-9}
H	0.500	140	53×10^{-9}
40DP	0.0816	13	—
High-torque (Fig 8)			
3 mm	0.1181	60	15×10^{-9}
5 mm	0.1968	100	21×10^{-9}
8 mm	0.3150	138	34×10^{-9}

Courtesy Stock Drive Products

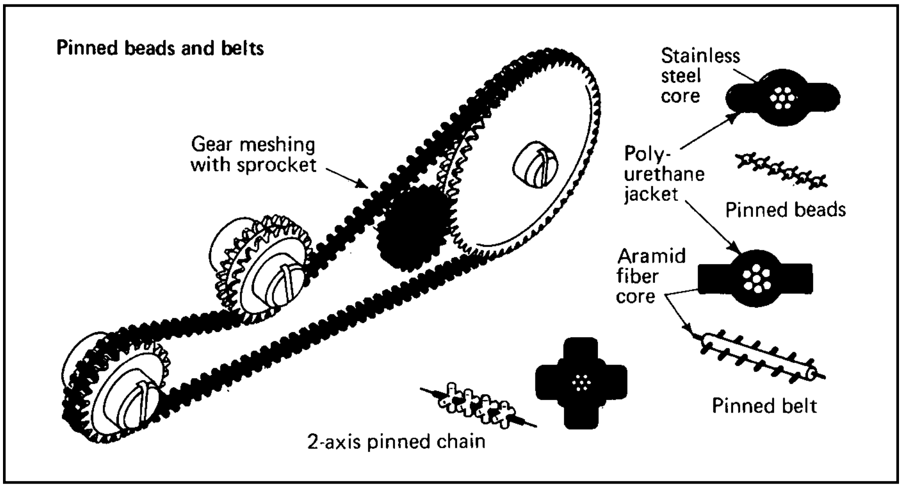


Fig. 4 Plastic pins eliminate the bead chain's tendency to cam out of pulley recesses, and permit greater precision in angular transmission.

the modern version of the bead chain, now common only in key chains and light-switch pull-cords.

The modern bead chain—if chain is the proper word—has no links. It has, instead, a continuous cable of stainless steel or aramid fiber which is covered with polyurethane. At controlled intervals, the plastic coating is molded into a bead (Fig. 3A). The length of the pitches thus formed can be controlled to within 0.001 in.

In operation, the cable runs in a grooved pulley; the beads seat in conical recesses in the pulley face. The flexibility, axial symmetry, and positive drive of bead chain suit a number of applications, both common and uncommon:

- An inexpensive, high-ratio drive that resists slipping and requires no lubrication (Fig. 3B). As with other chains and belts, the bead chain's capacity is limited by its total tensile strength (typically 40 to 80 lb for a single-strand steel-cable chain), by the speed-change ratio, and by the radii of the sprockets or pulleys.
- Connecting misaligned sprockets. If there is play in the sprockets, or if the sprockets are parallel but lie in different planes, the bead chain can compensate for up to 20º of misalignment (Fig. 3C).
- Skewed shafts, up to 90º out of phase (Fig. 3D).
- Right-angle and remote drives using guides or tubes (Figs. 3E and 3F). These methods are suitable only for low-speed, low-torque applications. Otherwise, frictional losses between the guide and the chain are unacceptable.
- Mechanical timing, using oversize beads at intervals to trip a microswitch (Fig. 3G). The chain can be altered or exchanged to give different timing schemes.
- Accurate rotary-to-linear motion conversion (Fig. 3H).
- Driving two counter-rotating outputs from a single input, using just a single belt (Fig. 3I).
- Rotary-to-oscillatory motion conversion (Fig. 3J).
- Clutched adjustment (Fig. 3K). A regular V-belt pulley without recesses permits the chain to slip when it reaches a pre-set limit. At the same time, bead-pulleys keep the output shafts synchronized. Similarly, a pulley or sprocket with shallow recesses permits the chain to slip one bead at a time when overloaded.
- Inexpensive "gears" or gear segments fashioned by wrapping a bead chain round the perimeter of a disk or solid arc of sheet metal (Fig. 3L). The sprocket then acts as a pinion. (Other designs are better for gear fabrication.)

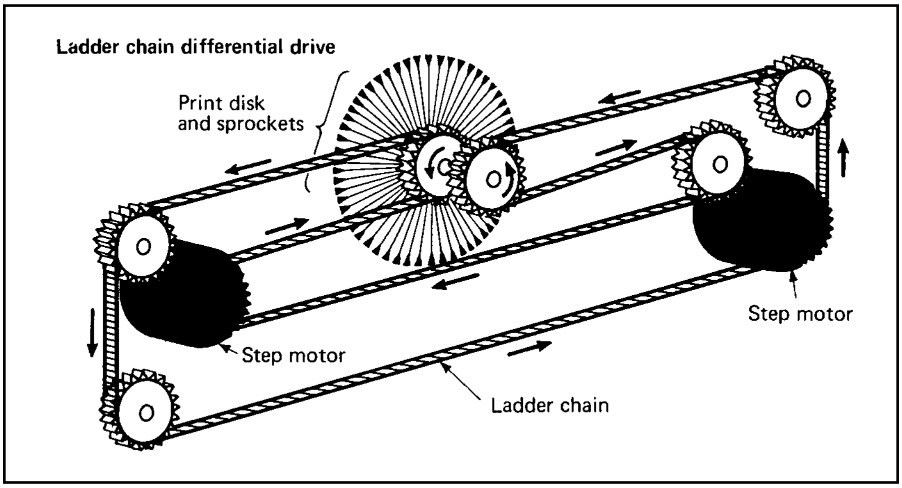


Fig. 5 A plastic-and-cable ladder chain in an impact-printer drive. In extreme conditions, such hybrids can serve many times longer than steel.

A More Stable Approach

Unfortunately, bead chains tend to cam out of deep sprocket recesses under high loads. In its first evolutionary step, the simple spherical bead grew limbs—two pins projecting at right angles to the cable axis (Fig. 4). The pulley or sprocket looks like a spur gear grooved to accommodate the belt; in fact, the pulley can mesh with a conventional spur gear of proper pitch.

Versions of the belt are also available with two sets of pins, one projecting vertically and the other horizontally. This arrangement permits the device to drive a series of perpendicular shafts without twisting the cable, like a bead chain but without the bead chain's load limitations. Reducing twist increases the transmission's lifetime and reliability.

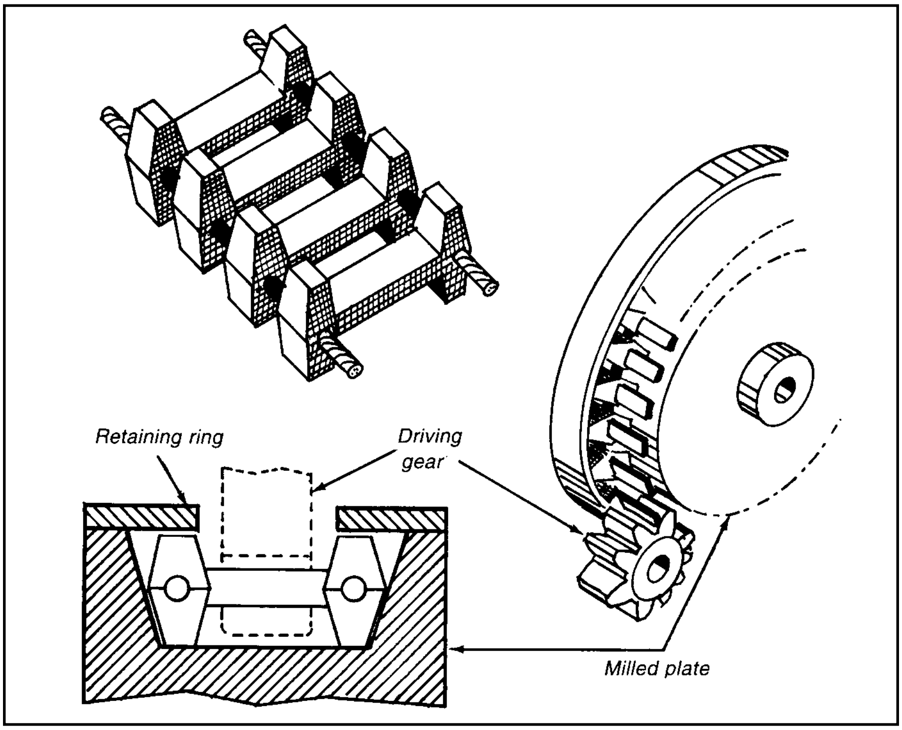


Fig. 6 A gear chain can function as a ladder chain, as a wide V-belt, or, as here, a gear surrogate meshing with a standard pinion.

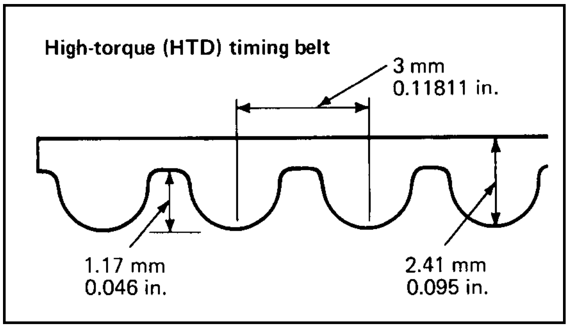


Fig. 7 Curved high-torque tooth profiles (just introduced in 3-mm and 5-mm pitches) increase load capacity of fine-pitch neoprene belts.

These belt-cable-chain hybrids can be sized and connected in the field, using metal crimp-collars. However, nonfactory splices generally reduce the cable's tensile strength by half.

Parallel-Cable Drives

Another species of positive-drive belt uses parallel cables, sacrificing some flexibility for improved stability and greater strength. Here, the cables are connected by rungs molded into the plastic coating, giving the appearance of a ladder (Fig. 6). This "ladder chain" also meshes with toothed pulleys, which need not be grooved.

A cable-and-plastic ladder chain is the basis for the differential drive system in a Hewlett-Packard impact printer (Fig. 5). When the motors rotate in the same direction at the same speed, the carriage moves to the right or left. When they rotate in opposite directions, but at the same speed, the carriage remains stationary and the print-disk rotates. A differential motion of the motors produces a combined translation and rotation of the print-disk.

The hybrid ladder chain is also well suited to laboratory of large spur gears from metal plates or pulleys (Fig. 6). Such a "gear" can run quietly in mesh with a pulley or a standard gear pinion of the proper pitch.

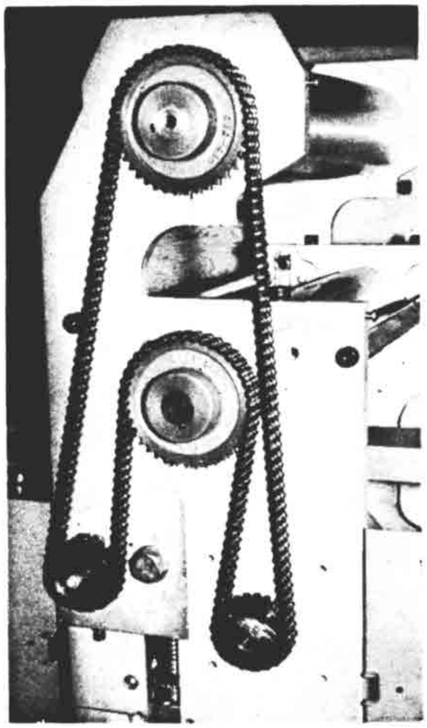
Another type of parallel-cable "chain," which mimics the standard chain, weighs just 1.2 oz/ft, requires no lubrication, and runs almost silently.

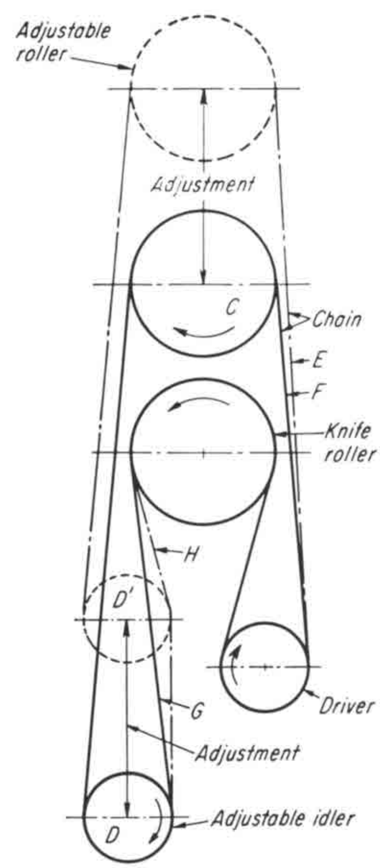
A Traditional Note

A new high-capacity tooth profile has been tested on conventional cogged belts. It has a standard cord and elastic body construction, but instead of the usual trapezoid, it has curved teeth (Fig. 7). Both 3-mm and 5-mm pitch versions have been introduced.

CHANGE CENTER DISTANCE WITHOUT AFFECTING SPEED RATIO

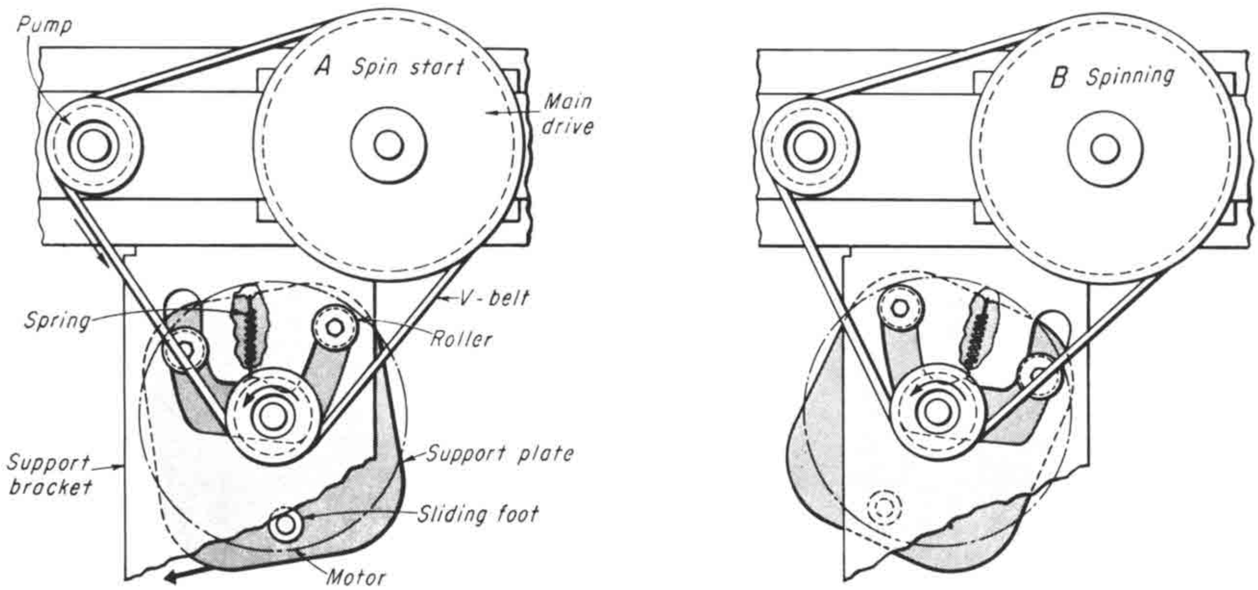
Increasing the gap between the roller and knife changes chain lengths from *F* to *E*. Because the idler moves with the roller sprocket, length *G* changes to *H*. The changes in chain length are similar in value but opposite in direction. Chain lengths *E* minus *F* closely approximate *G* minus *H*. Variations in required chain length occur because the chains do not run parallel. Sprocket offset is required to avoid interference. Slack produced is too minute to affect the drive because it is proportional to changes in the cosine of a small angle (2° to 5°). For the 72-in. chain, variation is 0.020 in.





MOTOR MOUNT PIVOTS FOR CONTROLLED TENSION

Belt tensioning proportional to load



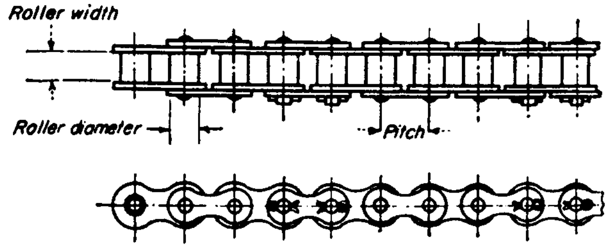
When the agitation cycle is completed, the motor is momentarily idle with the right roller bottomed in the right-hand slot. When spin-dry starts, (A) the starting torque produces a reaction at the stator, pivoting the motor on the bottomed roller. The motor pivots until the opposite roller bottoms in the left-hand slot. The motor now swings out until restrained by the V-belt, which drives the pump and basket.

The motor, momentarily at zero rpm, develops maximum torque and begins to accelerate the load of basket, water, and wash. The motor pivots (B) about the left roller increasing belt tension in proportion to the output torque. When the basket reaches maximum speed, the load is reduced and belt tension relaxes. The agitation cycle produces an identical reaction in the reverse direction.

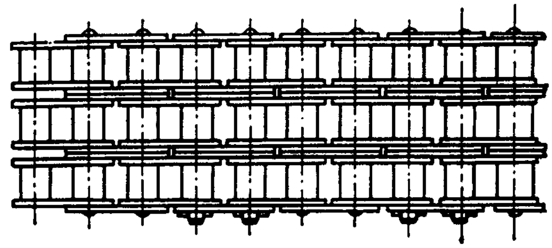
BUSHED ROLLER CHAINS AND THEIR ADAPTATIONS

Various roller, side-plate and pin configurations for power transmissions, conveying, and elevating.

STANDARD ROLLER CHAIN—FOR POWER TRANSMISSION AND CONVEYING

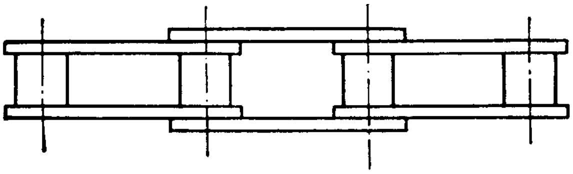


SINGLE WIDTH—Sizes ⅝ in and smaller have a spring-clip connecting link; those ¾ in and larger have a cotter pin.

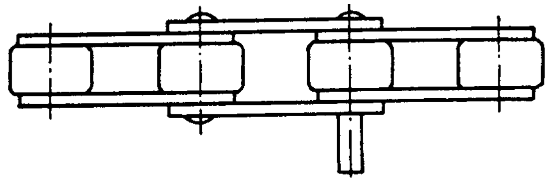


MULTIPLE WIDTH—Similar to single-width chain. It is made in widths up to 12 strands.

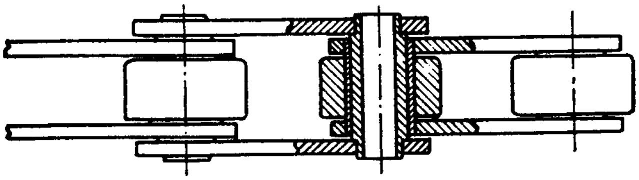
EXTENDED PITCH CHAIN—FOR CONVEYING



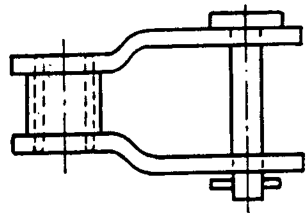
STANDARD ROLLER DIAMETER—made with 1 to 4 in pitch and cotter-pin-type connecting links.



OVERSIZED ROLLER DIAMETER—Same base chain as standard roller type but not made in multiple widths.

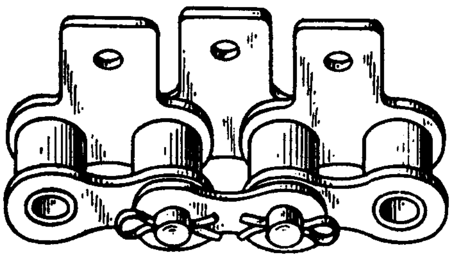


HOLLOW PIN—Made with 1¼ to 15 in pitch. It is adaptable to a variety of bolted attachments.

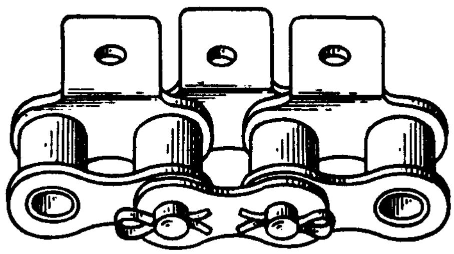


OFFSET LINK—Used when length requires an odd number of pitches and to shorten and lengthen a chain by one pitch.

STANDARD PITCH ADAPTATIONS

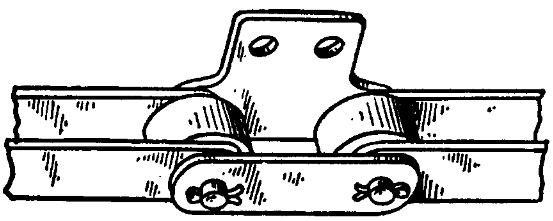


STRAIGHT LUG—Lugs on one or both sides can be spaced as desired. A standard roller is shown.

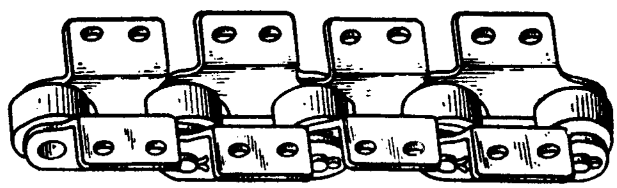


BENT LUG—Similar to straight-lug type for adaptations. A standard roller is shown.

EXTENDED PITCH ADAPTATIONS

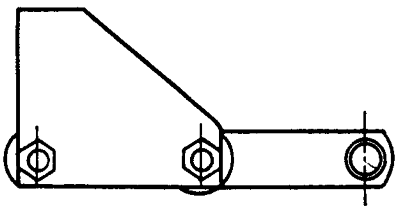


STRAIGHT LUG—An oversized diameter roller is shown.

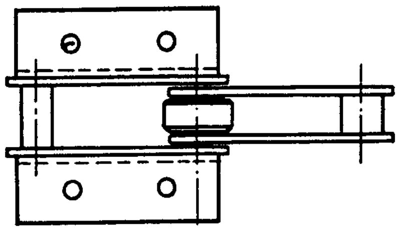


BENT LUG—An oversized diameter roller is shown.

HOLLOW PIN

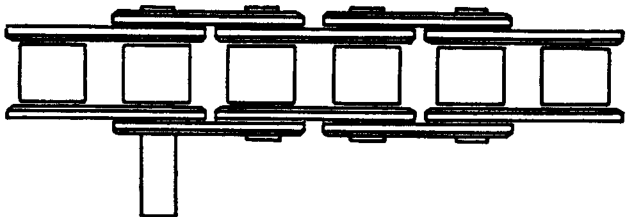


STRAIGHT LUG—Lugs are detachable for field adaptation.

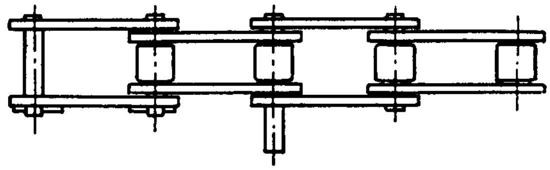


BENT LUG—Similar to straight lug type for adaptations.

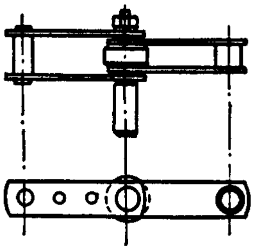
EXTENDED PIN CHAINS



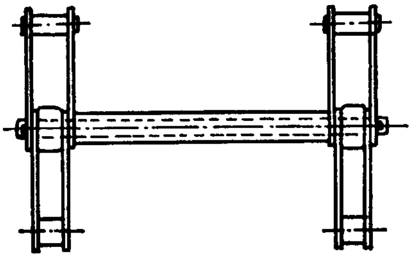
STANDARD PITCH—Pins can be extended on either side.



EXTENDED PITCH—Similar to standard for adaptations.

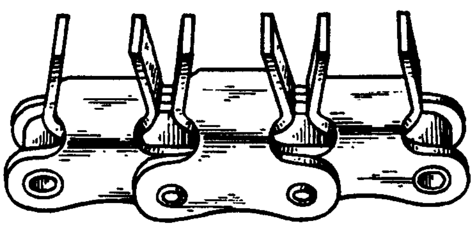


HOLLOW PIN—Pins are designed for field adaptation.

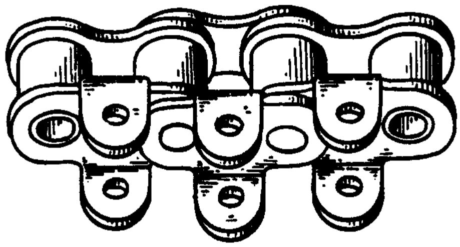


CROSS ROD—The rod can be removed from the hollow pins.

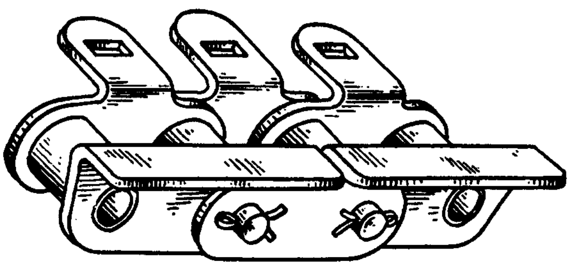
SPECIAL ADAPTATIONS



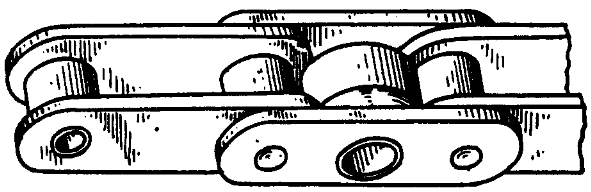
Used for holding conveyed objects.



Used to keep conveyed object on the center-line of the chain.



Used when flexing is desired in one direction only.



Used for supporting concentrated loads.

SIX INGENIOUS JOBS FOR ROLLER CHAIN

This low-cost industrial chain can be applied in a variety of ways to perform tasks other than simply transmitting power.

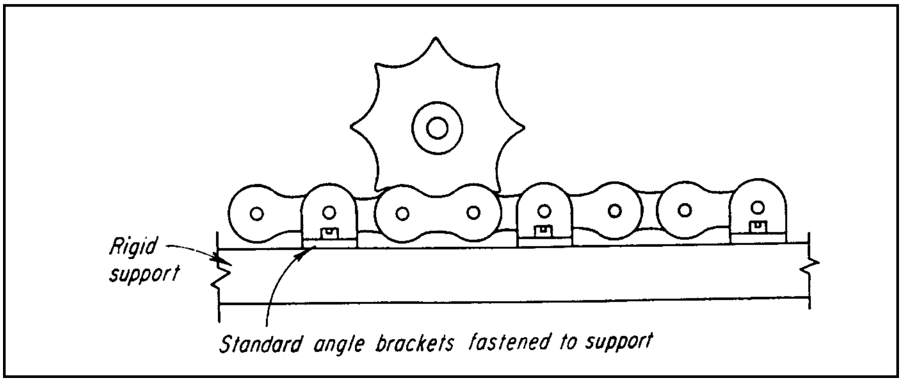


Fig. 1 This low-cost rack-and-pinion device is easily assembled from standard parts.

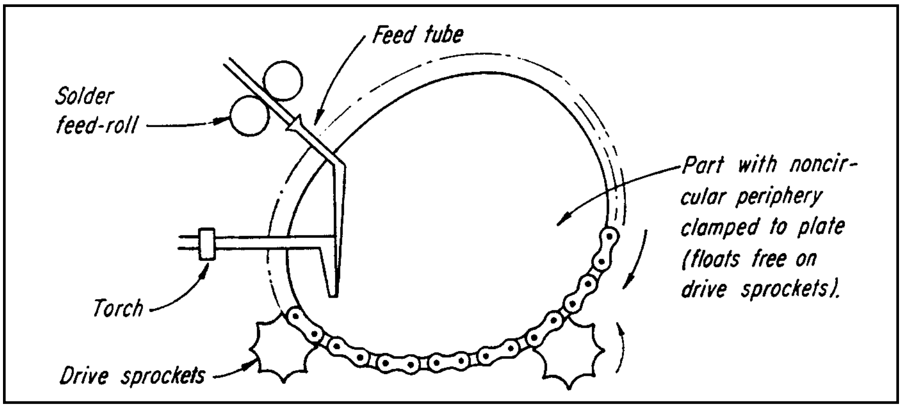


Fig. 2 An extension of the rack-and-pinion principle—This is a soldering fixture for noncircular shells. Positive-action cams can be similarly designed. Standard angle brackets attach the chain to a cam or fixture plate.

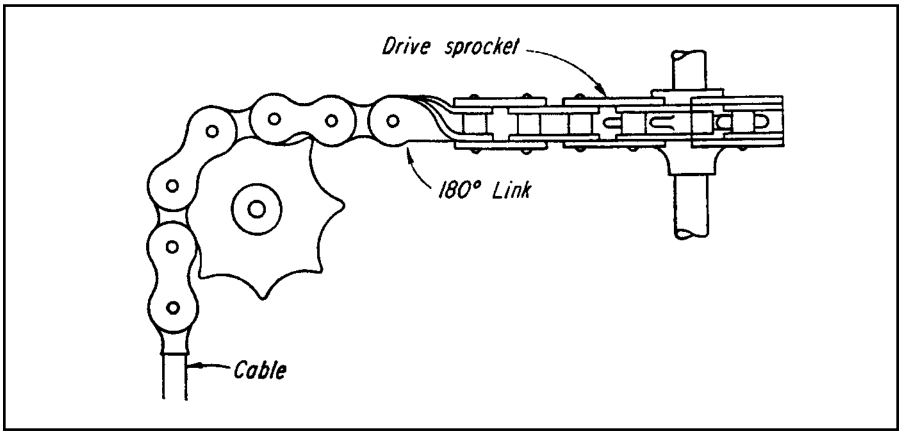


Fig. 3 This control-cable dirction-changer is extensively used in aircraft.

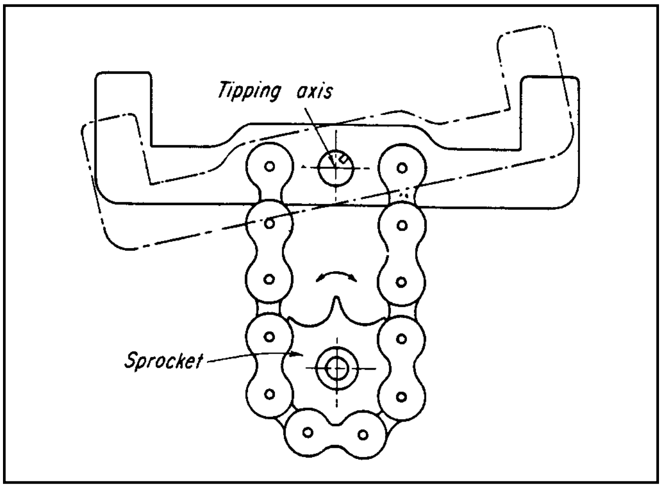


Fig. 4 The transmission of tipping or rocking motion can be combined with the previous example (Fig. 3) to transmit this kind of motion to a remote location and around obstructions. The tipping angle should not exceed 40°.

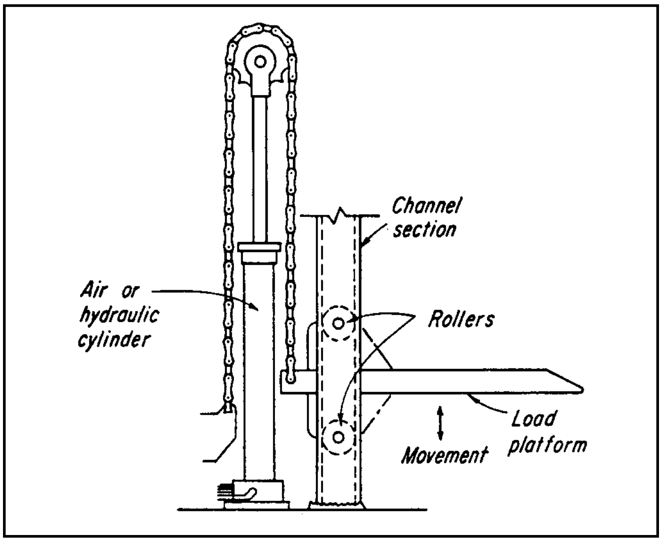


Fig. 5 This lifting device is simplified by roller chain.

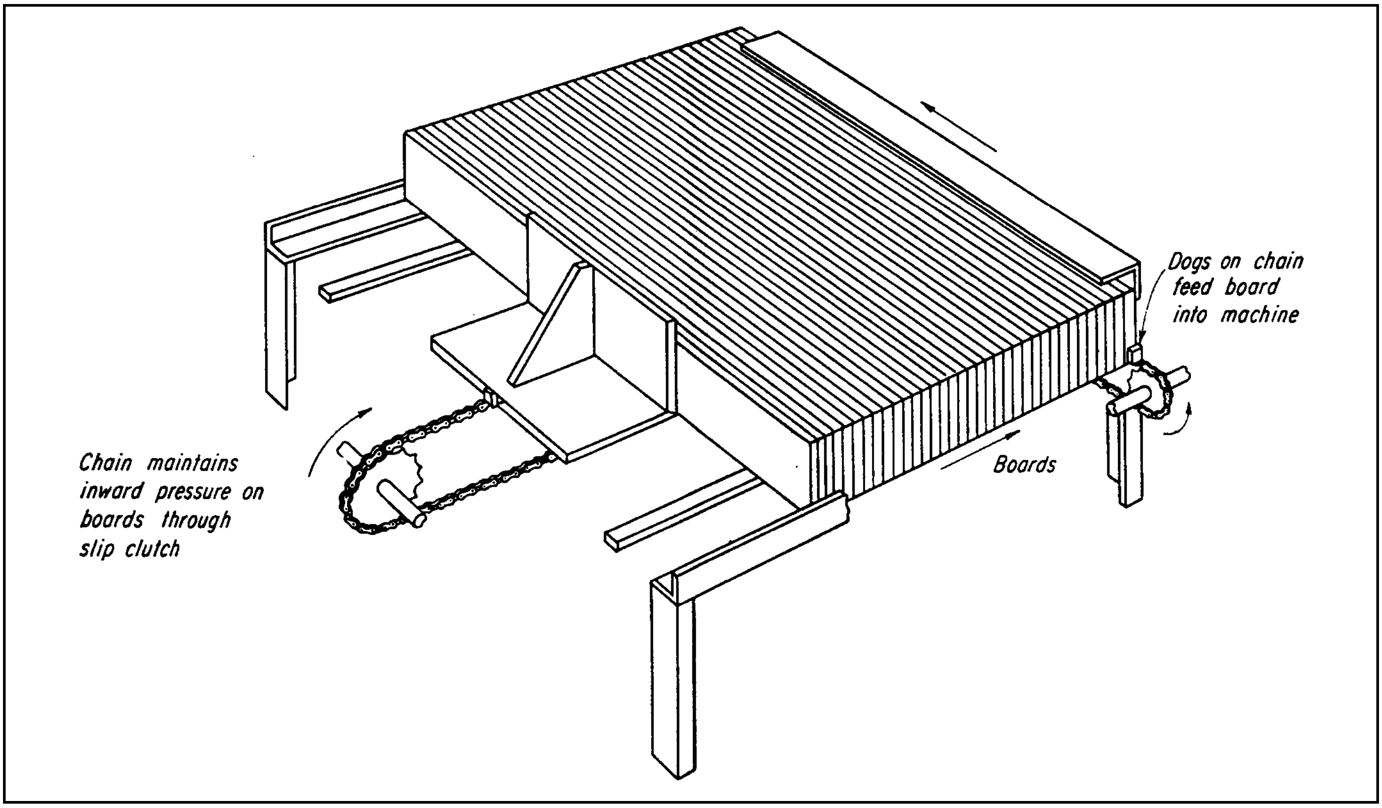


Fig. 6 Two examples of indexing and feeding applications of roller chain are shown here. This setup feeds plywood strips into a machine. The advantages of roller chain as used here are its flexibility and long feed.

SIX MORE JOBS FOR ROLLER CHAIN

Some further examples of how this low-cost but precision-made product can be arranged to do tasks other than transmit power.

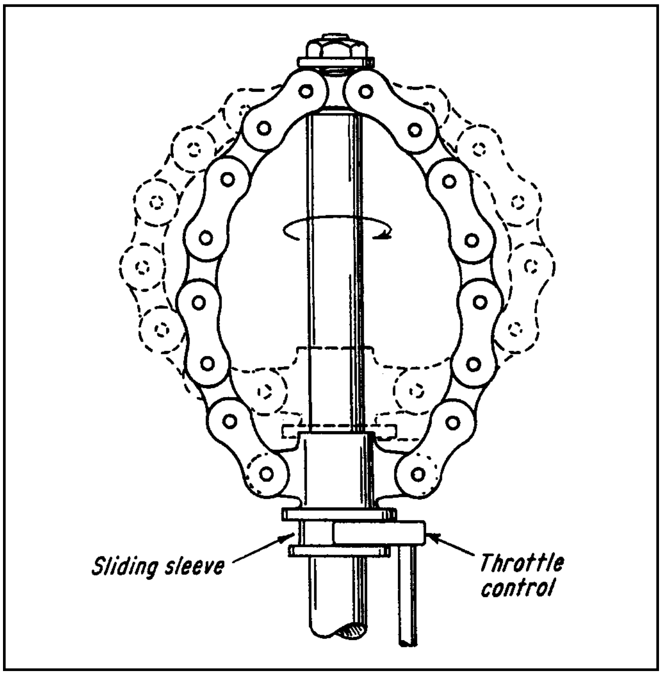


Fig. 1 Simple governor weights can be attached by means of standard brackets to increase response force when rotation speed is slow.

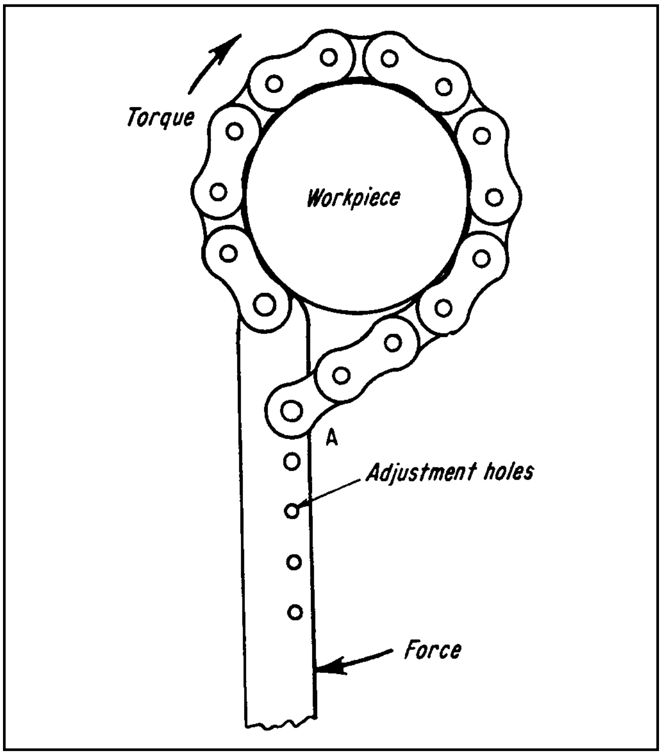
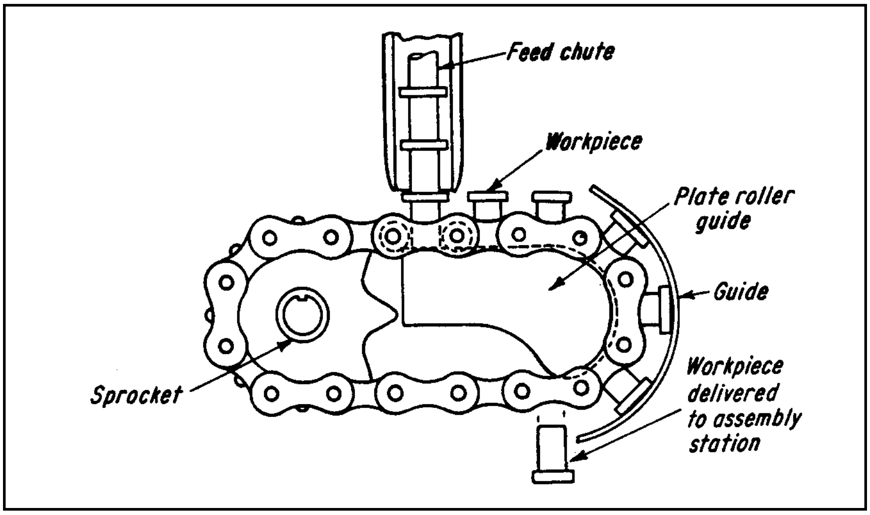


Fig. 2 Wrench pivot A can be adjusted to grip a variety of regularly or irregularly shaped objects.

Fig. 3 Small parts can be conveyed, fed, or oriented between spaces of roller chain.



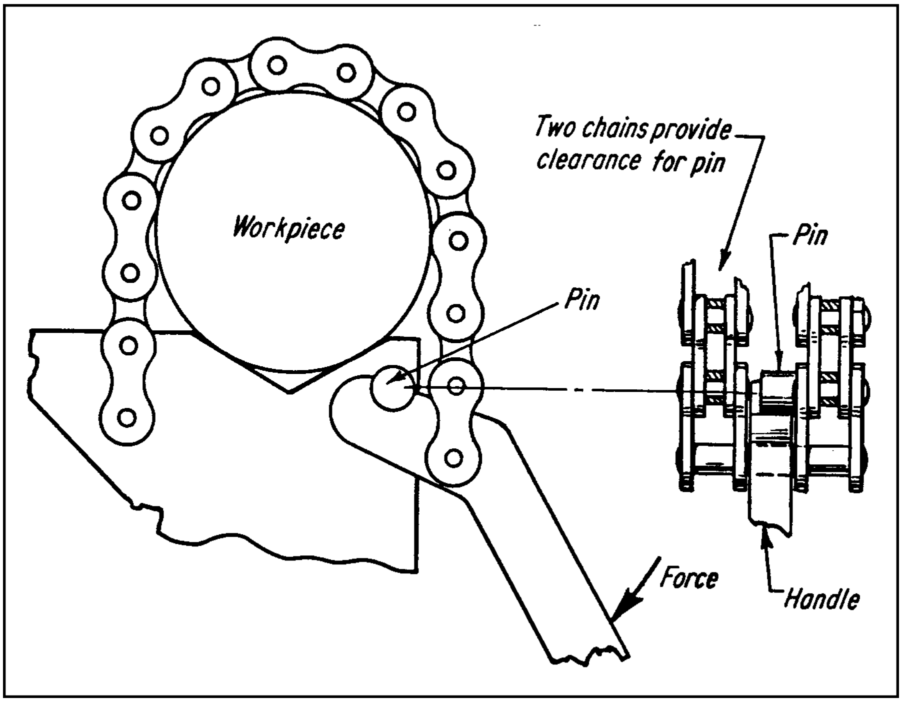
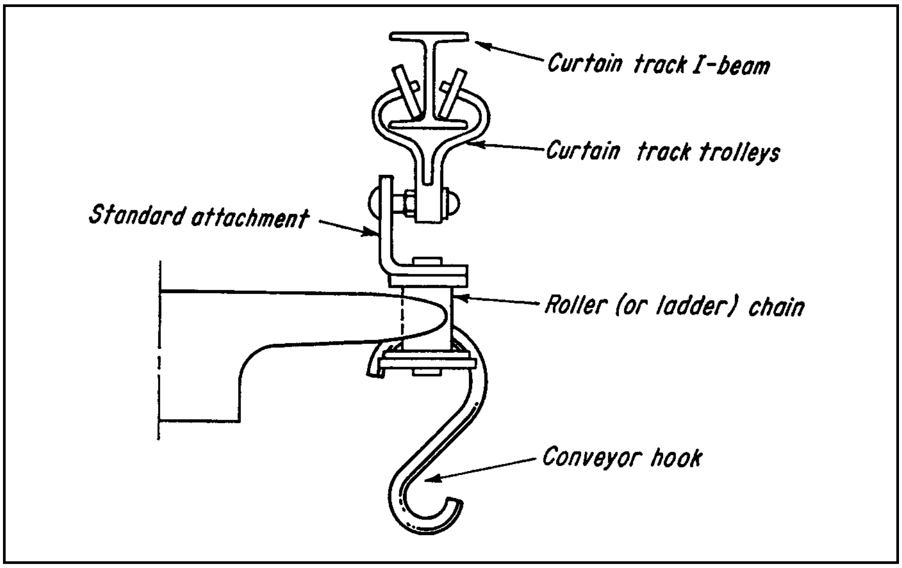


Fig. 4 Clamp toggle action is supplied by two chains, thus clearing pin at fulcrum.

Fig. 5 Light-duty trolley conveyors can be made by combining standard roller-chain components with standard curtain-track components. Small gearmotors are used to drive the conveyor.



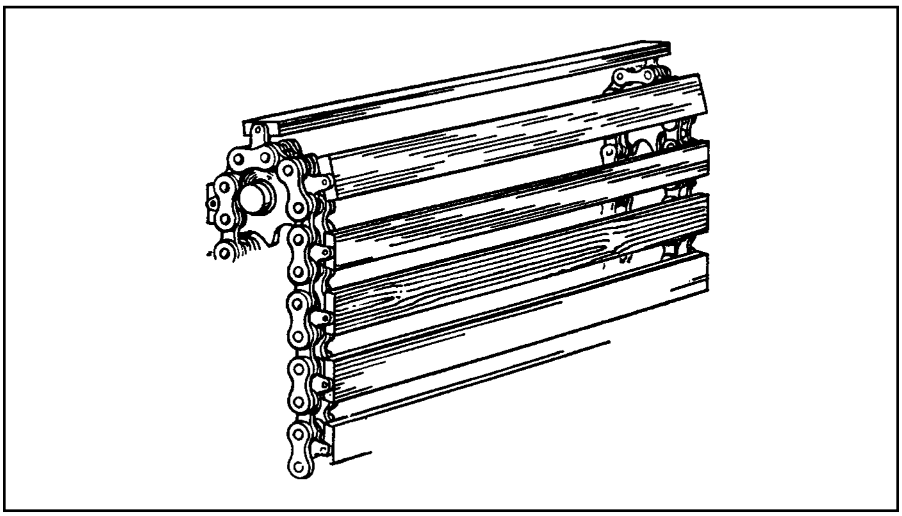


Fig. 6 Slatted belt, made by attaching wood, plastic, or metal slats, can serve as adjustable safety guard, conveyor belt, fast-acting security-wicket window.

MECHANISMS FOR REDUCING PULSATIONS IN CHAIN DRIVES

Pulsations in chain motion created by the chordal action of chain and sprockets can be minimized or avoided by introducing a compensating cyclic motion in the driving sprocket. Mechanisms for reducing fluctuating dynamic loads in chain drives and the pulsations resulting from them include noncircular gears, eccentric gears, and cam-activated intermediate shafts.

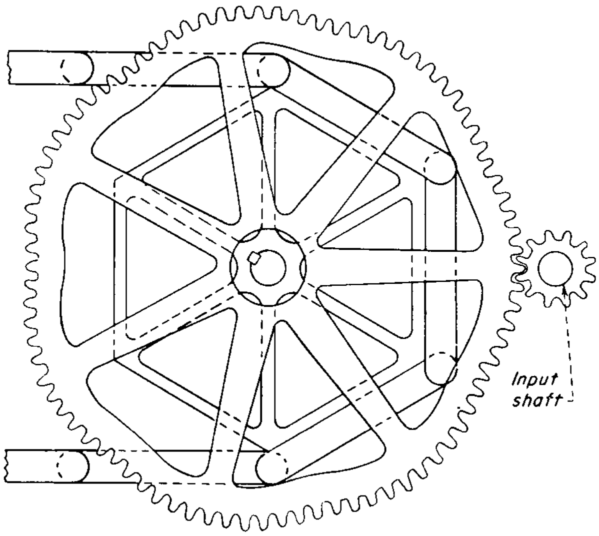


Fig. 1

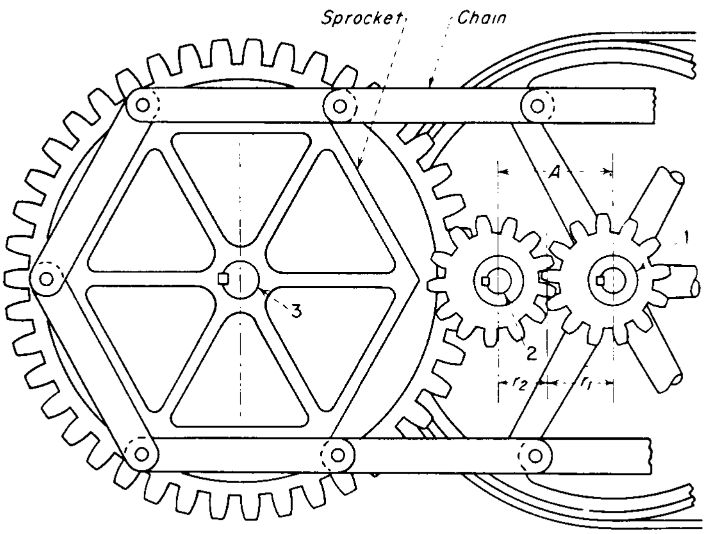


Fig. 2

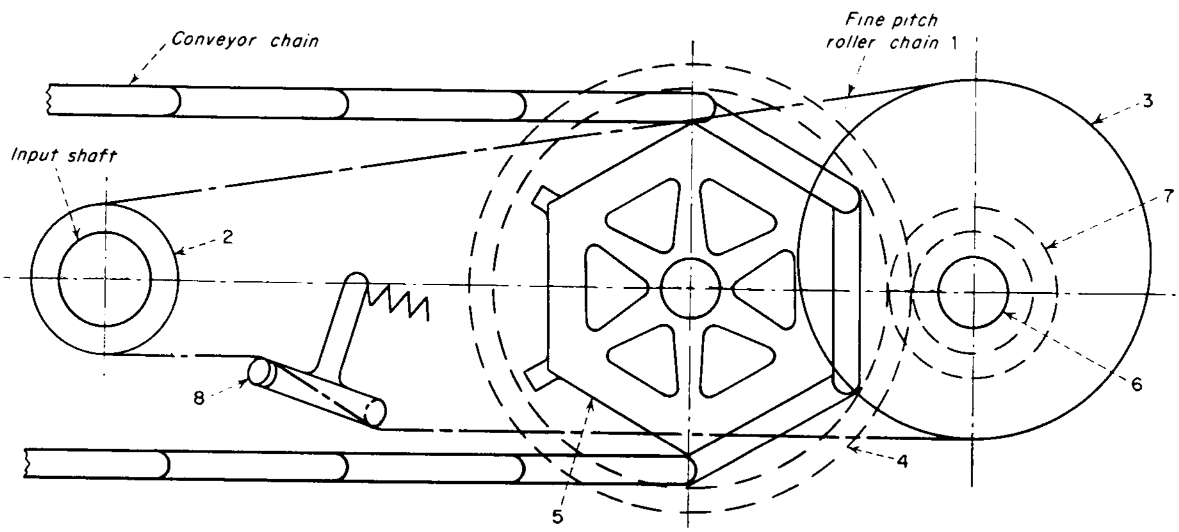


Fig. 3

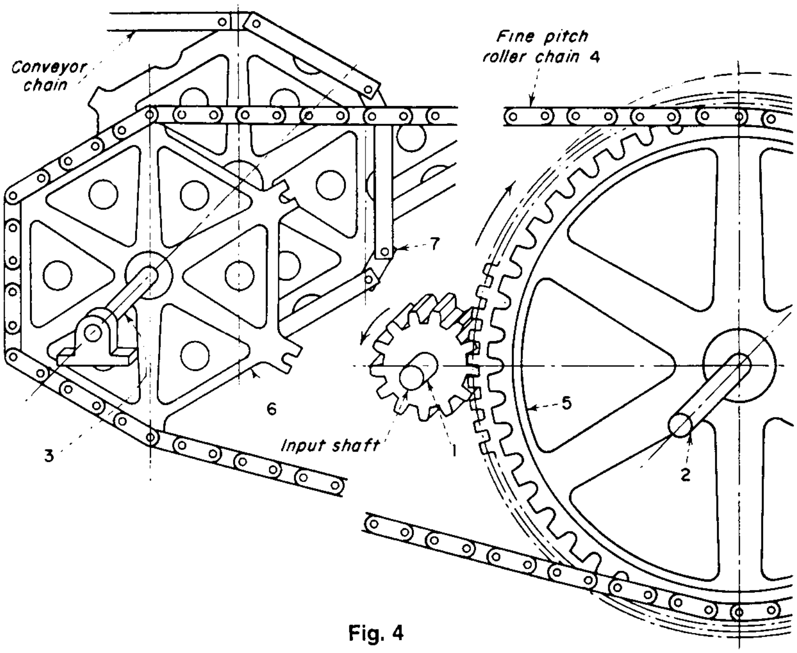


Fig. 4

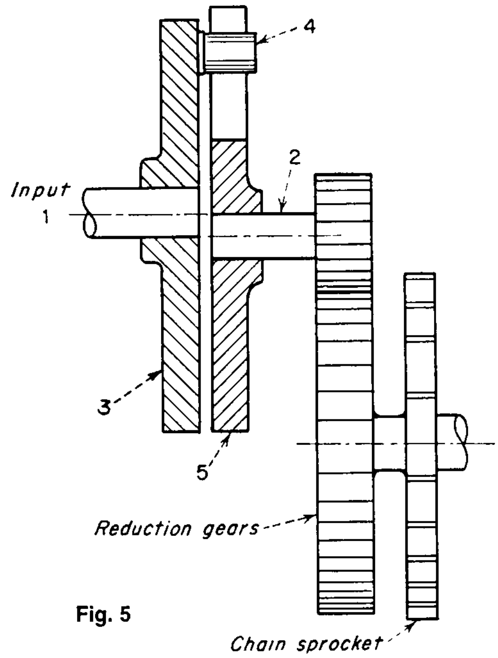


Fig. 5

Fig. 1 The large cast-tooth, noncircular gear, mounted on the chain sprocket shaft, has a wavy outline in which the number of waves equals the number of teeth on a sprocket. The pinion has a corresponding noncircular shape. Although requiring special-shaped gears, the drive completely equalizes the chain pulsations.

Fig. 2 This drive has two eccentrically mounted spur pinions (1 and 2). Input power is through the belt pulley keyed to the same shaft as pinion 1. Pinion 3 (not shown), keyed to the shaft of pinion 2, drives the large gear and sprocket. However, the mechanism does not completely equalize chain velocity unless the pitch lines of pinions 1 and 2 are noncircular instead of eccentric.

Fig. 3 An additional sprocket 2 drives the noncircular sprocket 3 through a fine-pitch chain 1.This imparts pulsating velocity to shaft 6 and to the long-pitch conveyor sprocket 5 through pinion 7 and gear 4. The ratio of the gear pair is made the same as the number of teeth of sprocket 5. Spring-actuated lever and rollers 8 take up the slack. Conveyor motion is equalized, but the mechanism has limited power capacity because the pitch of chain 1 must be kept small. Capacity can be increased by using multiple strands of fine-pitch chain.

Fig. 4 Power is transmitted from shaft 2 to sprocket 6 through chain 4, thus imparting a variable velocity to shaft 3, and through it, to the conveyor sprocket 7. Because chain 4 has a small pitch and sprocket 5 is relatively large, the velocity of 4 is almost constant. This induces an almost constant conveyor velocity. The mechanism requires the rollers to tighten the slack side of the chain, and it has limited power capacity.

Fig. 5 Variable motion to the sprocket is produced by disk 3. It supports pin and roller 4, as well as disk 5, which has a radial slot and is eccentrically mounted on shaft 2. The ratio of rpm of shaft 2 to the sprocket equals the number of teeth in the sprocket. Chain velocity is not completely equalized.

Fig. 6 The integrated "planetary gear" system (gears 4, 5, 6 and 7) is activated by cam 10, and it transmits a variable velocity to the sprocket synchronized with chain pulsations through shaft 2, thus completely equalizing chain velocity. Cam 10 rides on a circular idler roller 11. Because of the equilibrium of the forces, the cam maintains positive contact with the roller. The unit has standard gears, acts simultaneously as a speed reducer, and can transmit high horsepower.

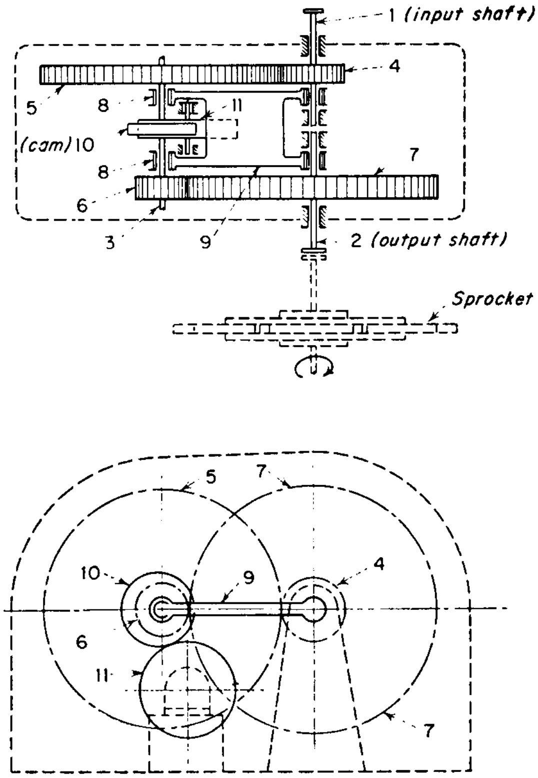


Fig. 6

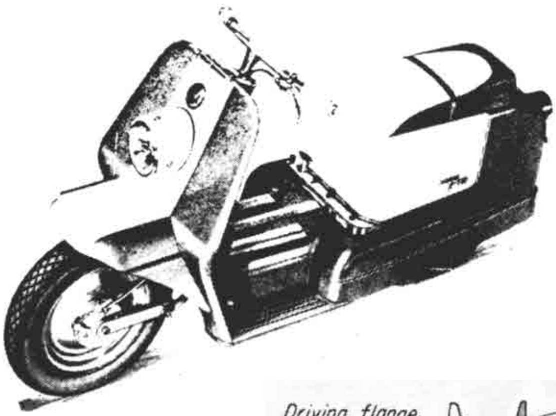
SMOOTHER DRIVE WITHOUT GEARS

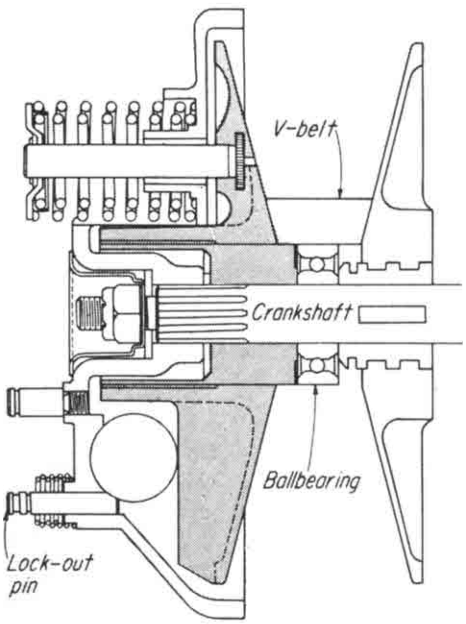
The transmission in this motor scoter is torque-sensitive; motor speed controls the continuously variable drive ratio. The operator merely works the throttle and brake.

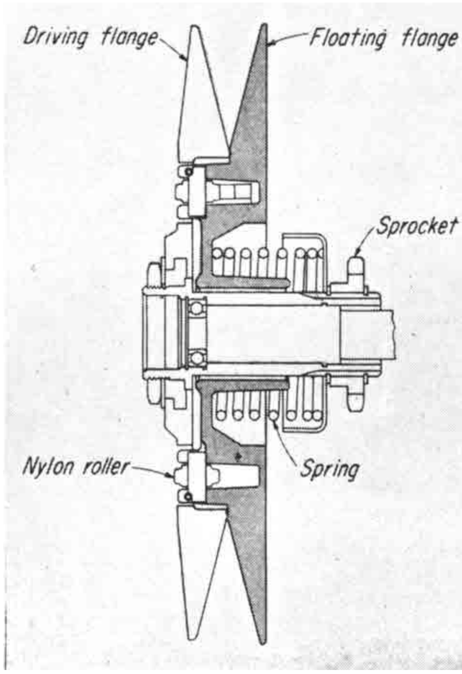
Variable-diameter V-belt pulleys connect the motor and chain drive sprocket to give a wide range of speed reduction. The front pulley incorporates a three-ball centrifugal clutch which forces the flanges together when the engine speeds up. At idle speed the belt rides on a ball-bearing between the retracted flanges of the pulley. During starting and warmup, a lockout prevents the forward clutch from operating.

Upon initial engagement, the overall drive ratio is approximately 18:1. As engine speed increases, the belt rides higher up on the forward-pulley flanges until the overall drive ratio becomes approximately 6:1. The resulting variations in belt tension are absorbed by the spring-loaded flanges of the rear pulley. When a clutch is in an idle position, the V-belt is forced to the outer edge of the rear pulley by a spring force. When the clutch engages, the floating half of the front pulley moves inward, increasing its effective diameter and pulling the belt down between the flanges of the rear pulley.

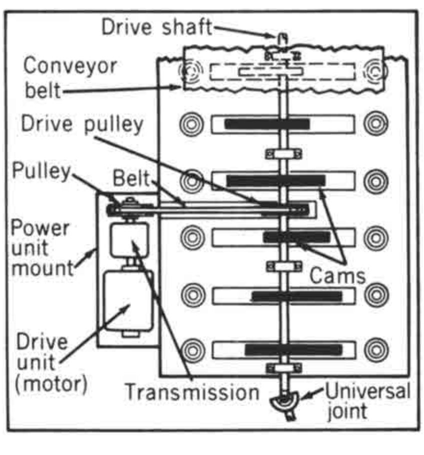
The transmission is torque-responsive. A sudden engine acceleration increases the effective diameter of the rear pulley, lowering the drive ratio. It works this way: An increase in belt tension rotates the floating flange ahead in relation to the driving flange. The belt now slips slightly on its driver. At this time nylon rollers on the floating flange engage cams on the driving flange, pulling the flanges together and increasing the effective diameter of the pulley.







FLEXIBLE CONVEYOR MOVES IN WAVES



Most conventional conveyors used in tunneling and mining can't negotiate curves and can't be powered at different points. They are subject to malfunction because of slight misalignment, and they require time-consuming adjustments to lengthen or shorten them.

Thomas E. Howard of the U.S Bureau of Mines, invented a conveyor belt that does not move forward. That might solve all of these problems.

The conveyor, designed to move broken ore, rock, and coal in mines, moves material along a flexible belt. The belt is given a wave-like movement by the sequenced rising and dropping of supporting yokes beneath it.

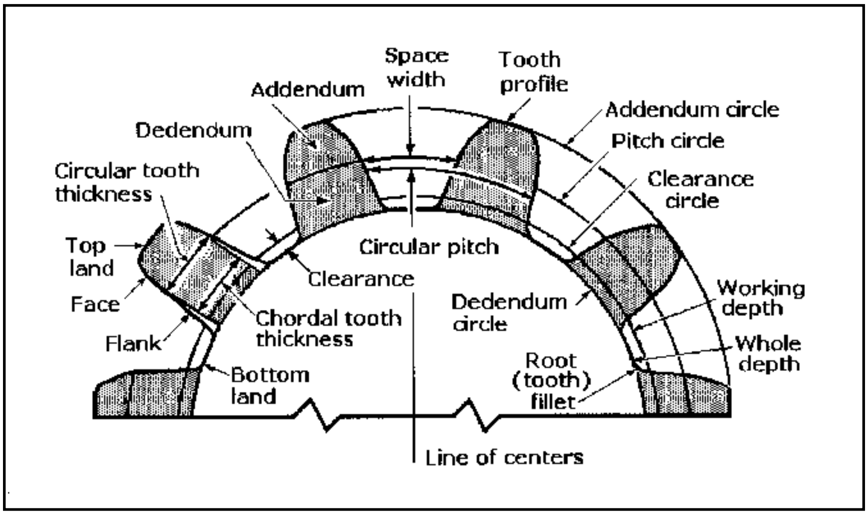
The principle. The conveyor incorporates modules built in arcs and Y's in such a way that it can be easily joined with standardized sections to negotiate corners and either merge or separate streams of moving materials. It can be powered at any one point or at several points, and it incorporates automatic controls to actuate only those parts of the belt that are loaded, thereby reducing power consumption.

In tests at the bureau's Pittsburgh Mining Research Center, a simplified mechanical model of the conveyor has moved rock at rates comparable to those of conventional belts.

CHAPTER 8

GEARED SYSTEMS AND VARIABLE-SPEED MECHANISMS

GEARS AND GEARING



Gear tooth terminology

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

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

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

Gear Terminology

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

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

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

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

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

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

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

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

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

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

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

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

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

Gear Dynamics Terminology

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

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

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

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

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

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

Gear Classification

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

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

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

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

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

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

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

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

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

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

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

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

Hypoid gears are spiral bevel gears with offset intersecting axes.

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

NUTATING-PLATE DRIVE

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

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

Kinetics of the Nutation Drive

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

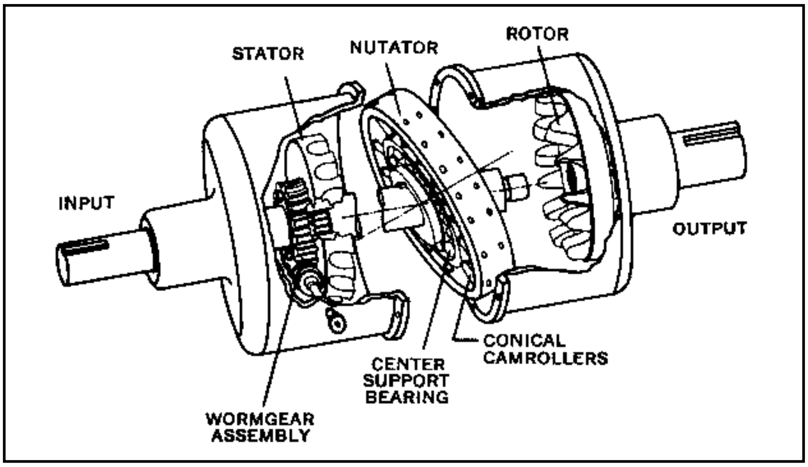


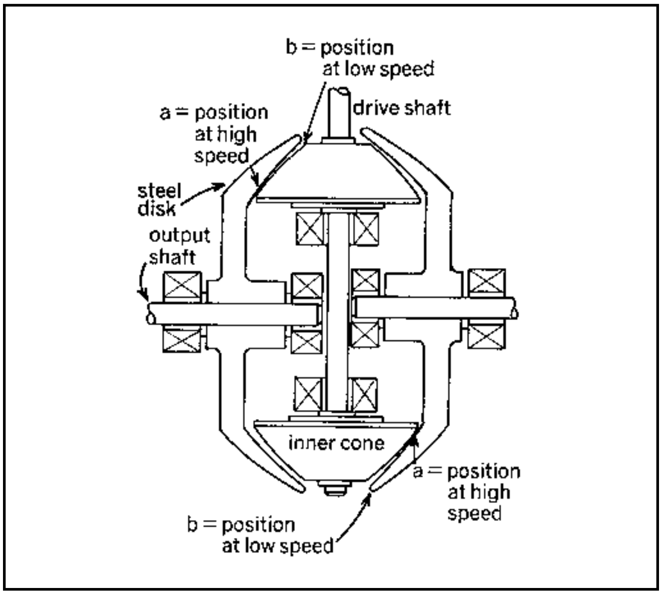
Fig. 1 An exploded view of the Nutation Drive.

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

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

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

CONE DRIVE NEEDS NO GEARS OR PULLEYS



Cone drive operates without lubrication.

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

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

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

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

VARIABLE-SPEED MECHANICAL DRIVES

CONE DRIVES

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

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

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

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

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

Fig. 1

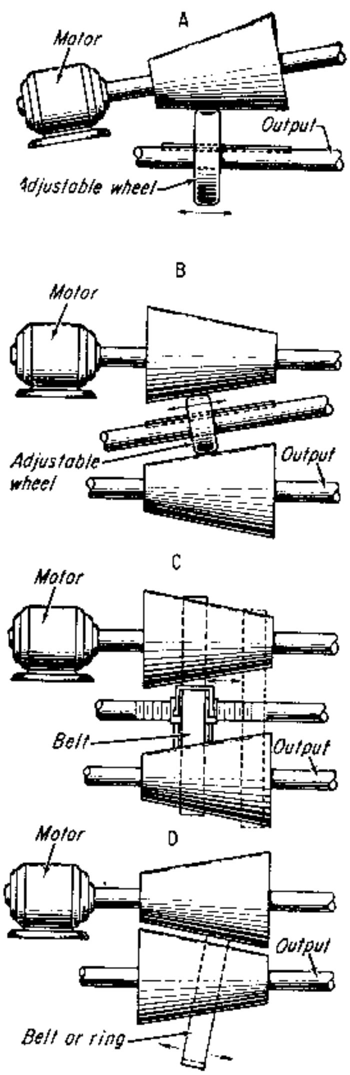


Fig. 2

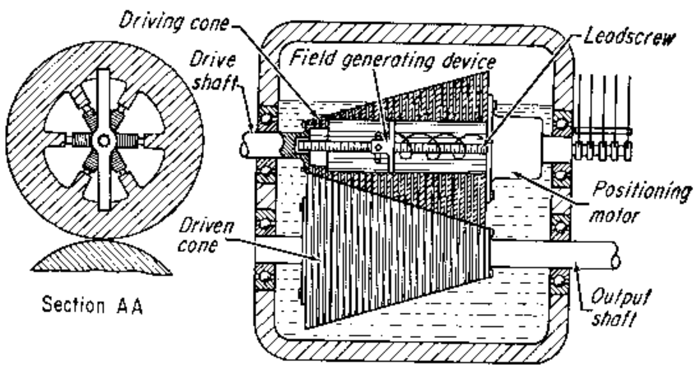
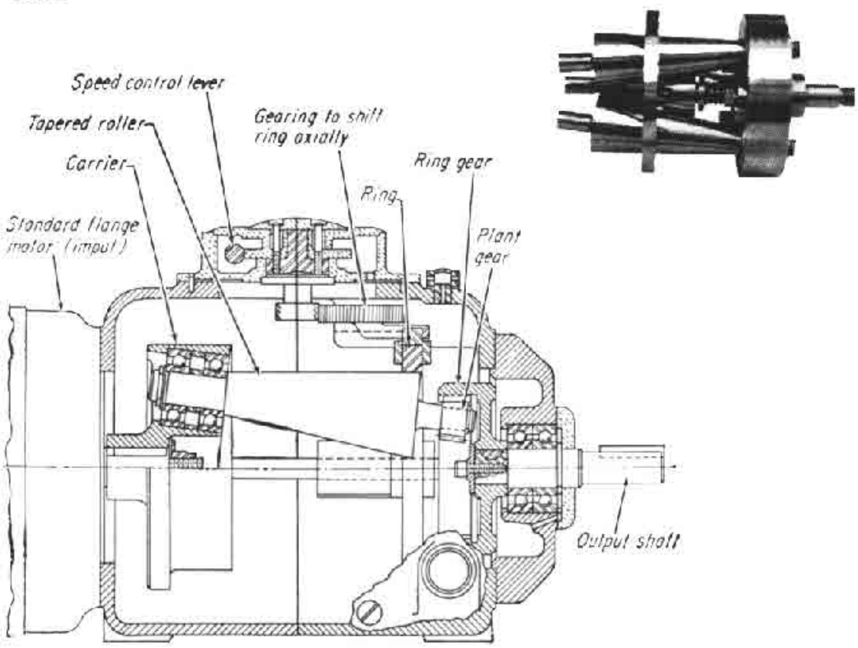


Fig. 3



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

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

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

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

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

Fig. 4

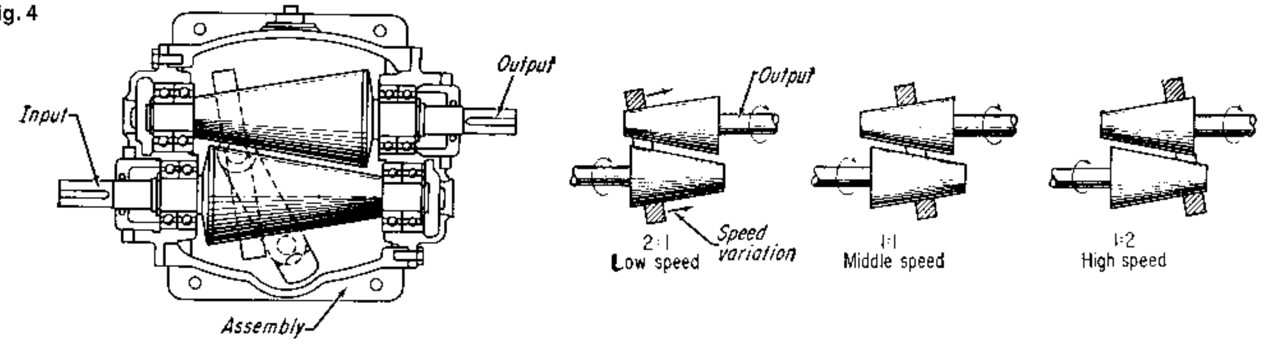
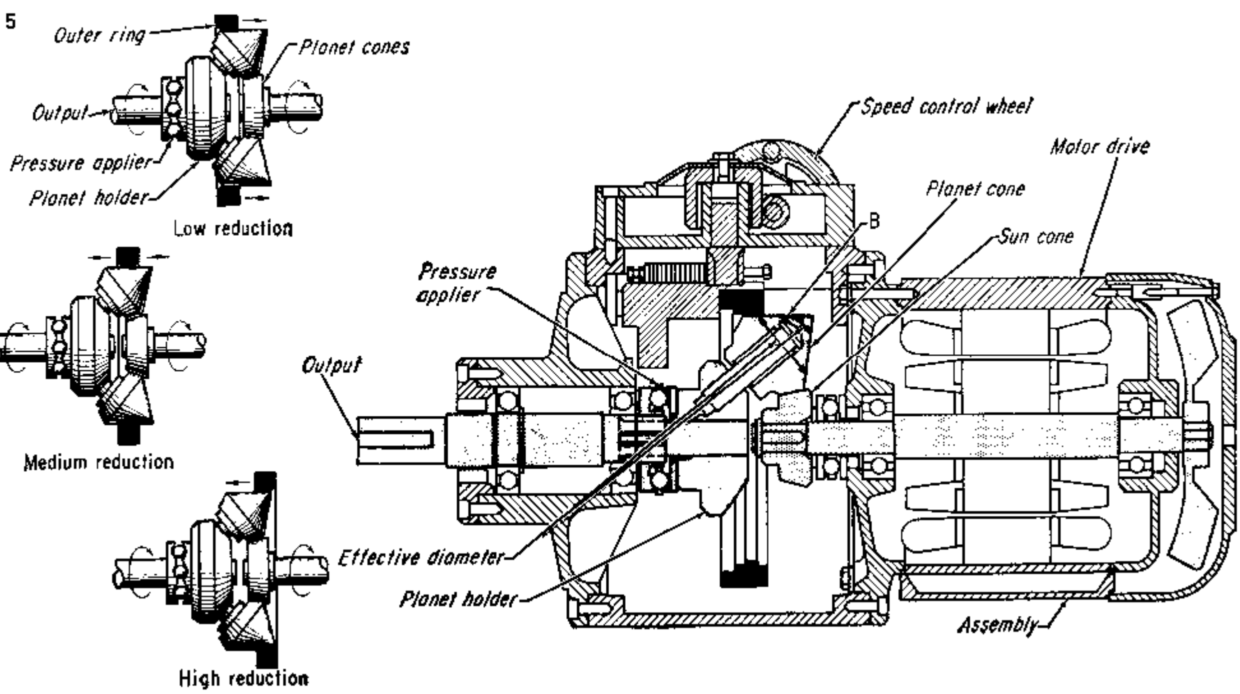


Fig. 5



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

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

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

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

Fig. 6

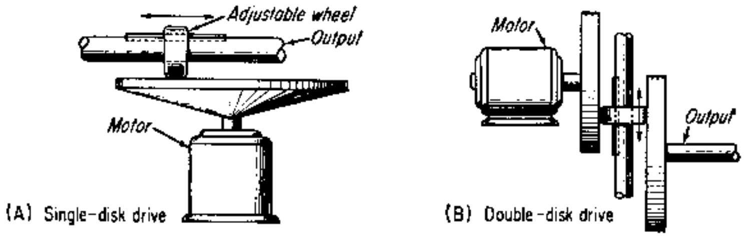


Fig. 7

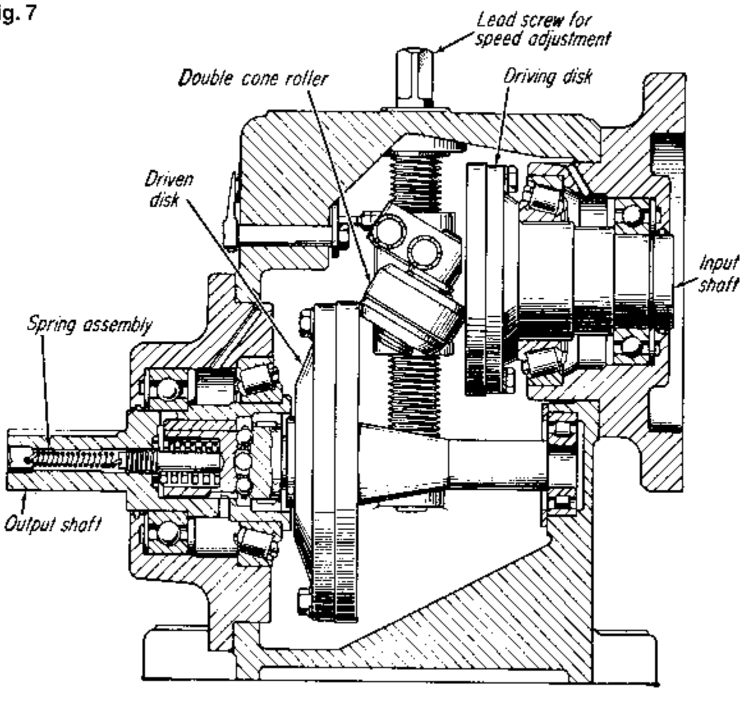
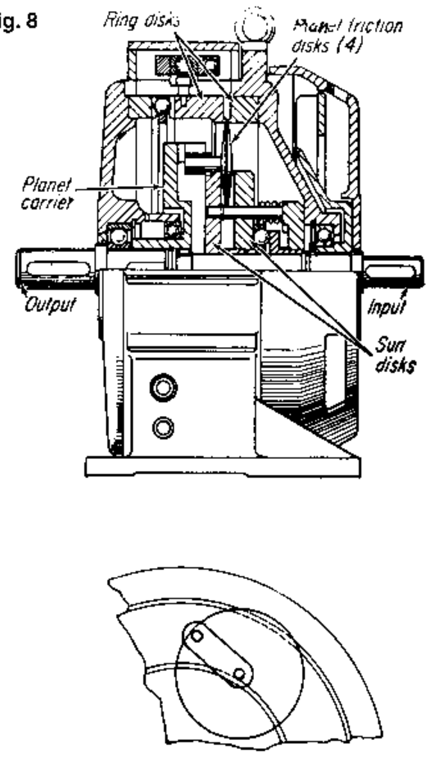


Fig. 8



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

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

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

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

Fig. 9

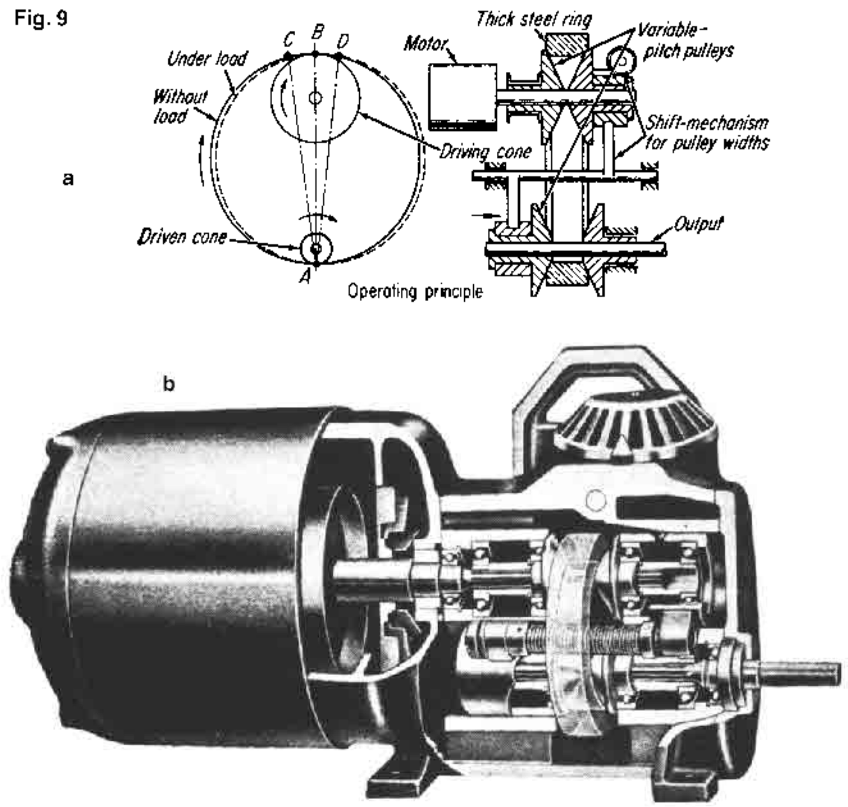


Fig. 10

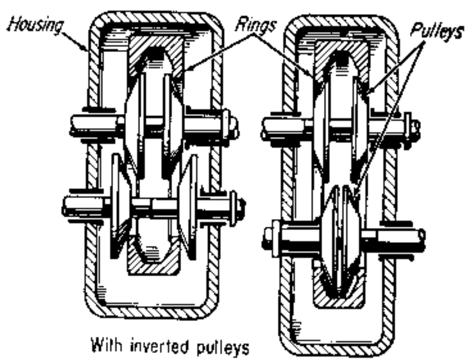
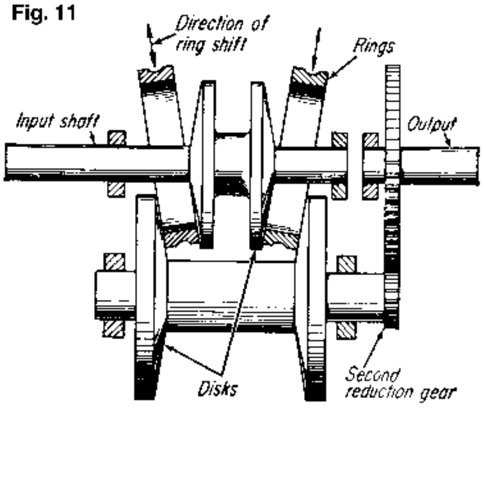


Fig. 11



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

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

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

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

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

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

Fig. 12

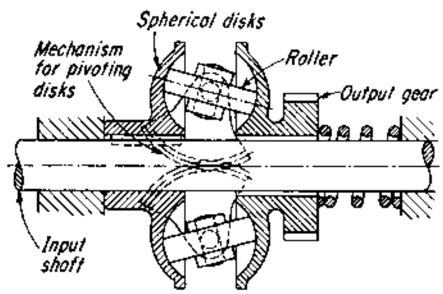


Fig. 13

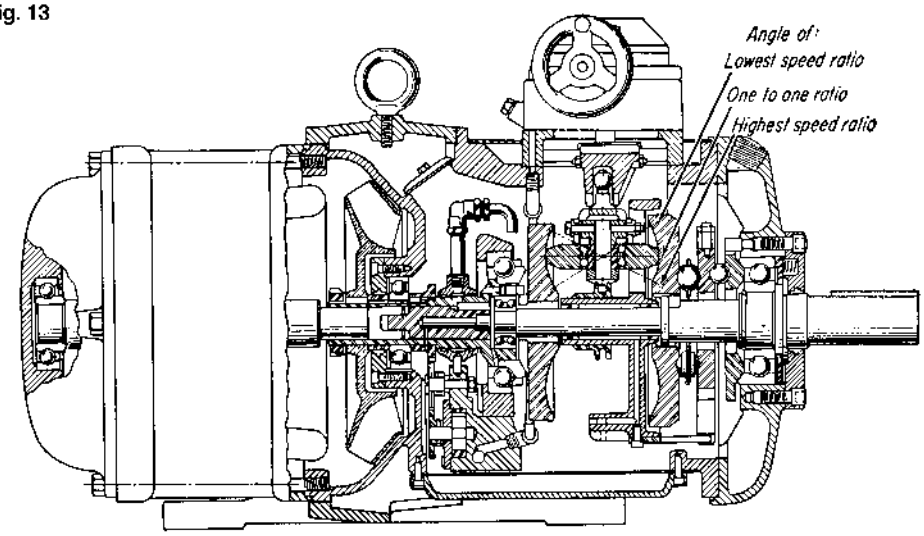


Fig. 14

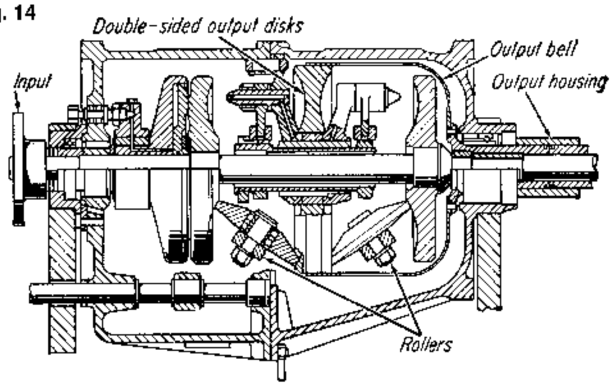
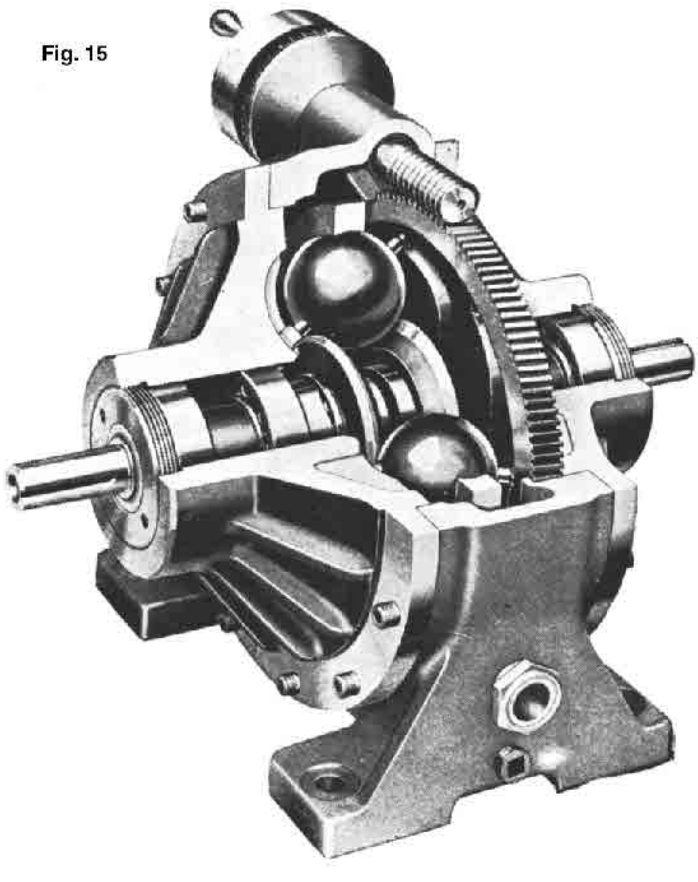
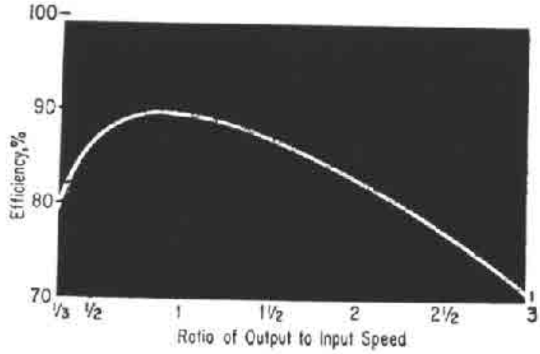


Fig. 15





Efficiency of tilting-ball drive

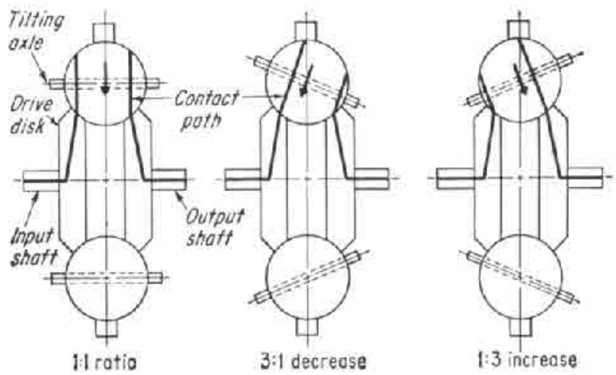
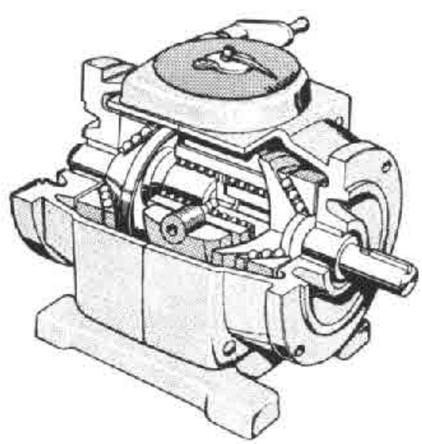
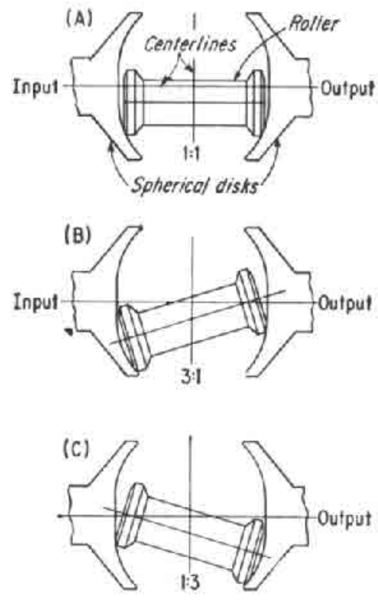


Fig. 16





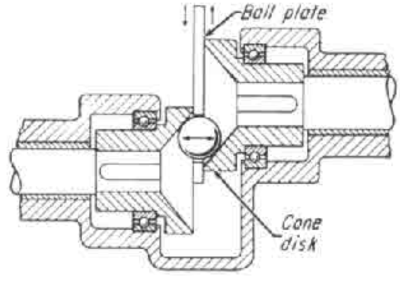
eccentrically mounted between the coaxial input and output spherical disks. Changes in speed ratio are made by changing the angular position of the roller.

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

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

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

Fig. 17



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

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

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

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

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

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

Fig. 18

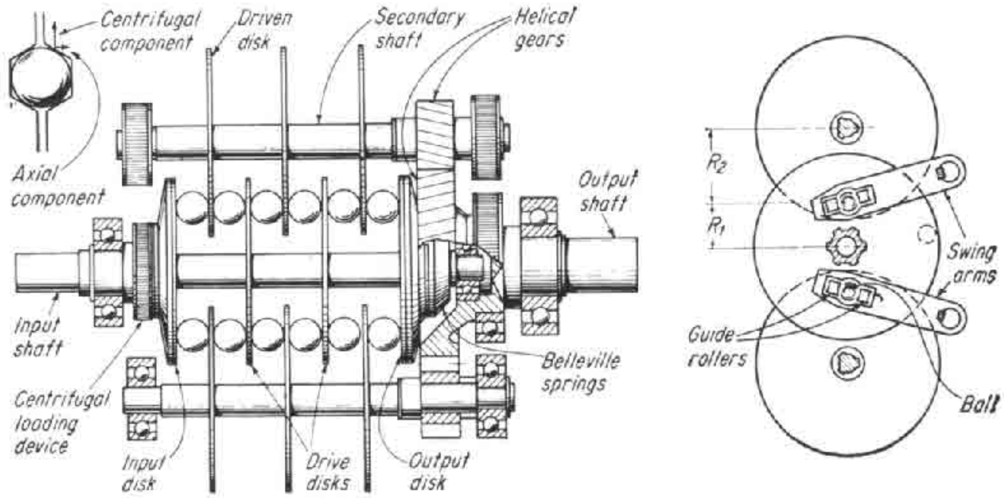
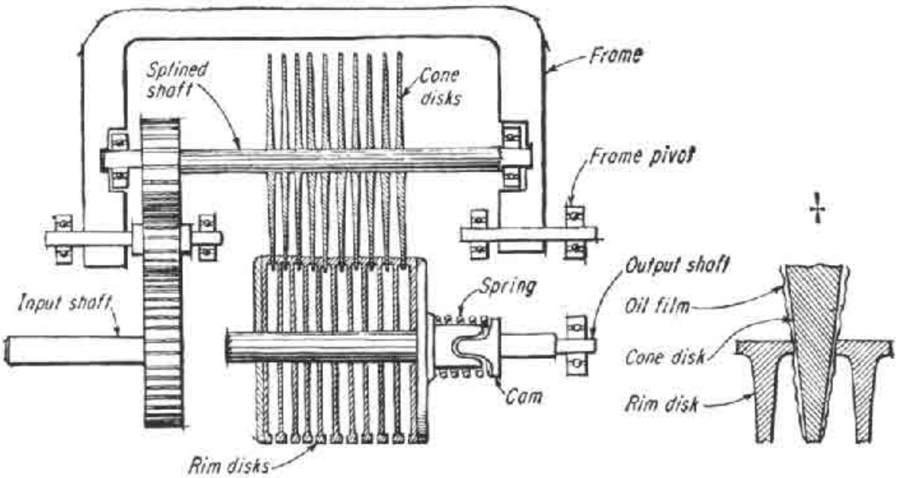


Fig. 19



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

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

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

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

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

Fig. 20

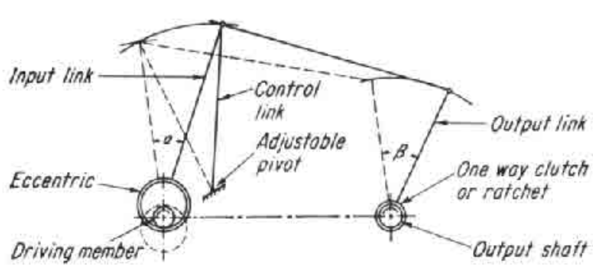


Fig. 21

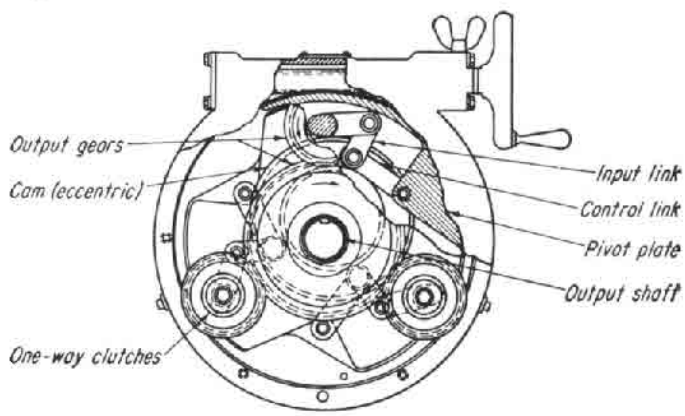
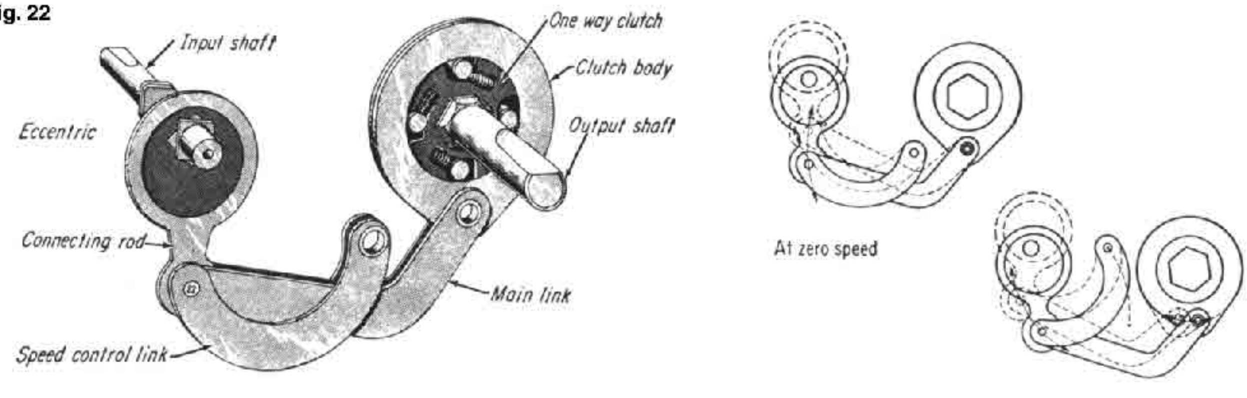


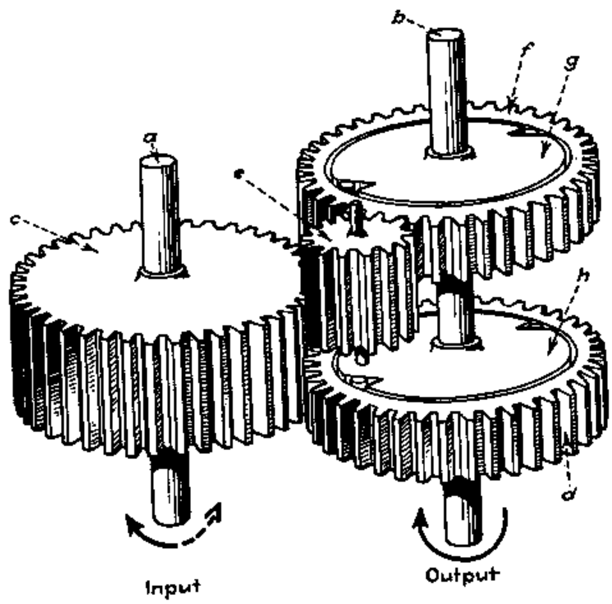
Fig. 22



UNIDIRECTIONAL DRIVE

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

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



ADDITIONAL VARIATIONS

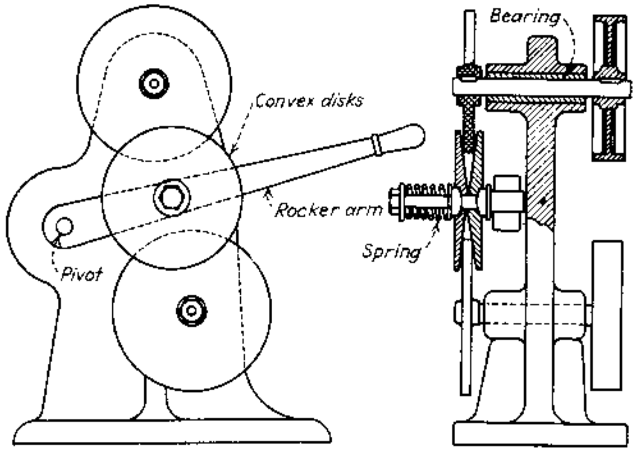


Fig. 1

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

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

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

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

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

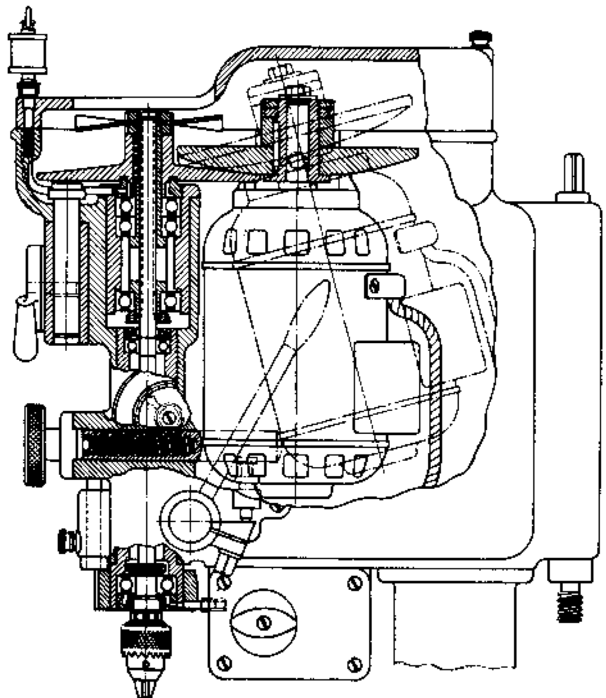


Fig. 2

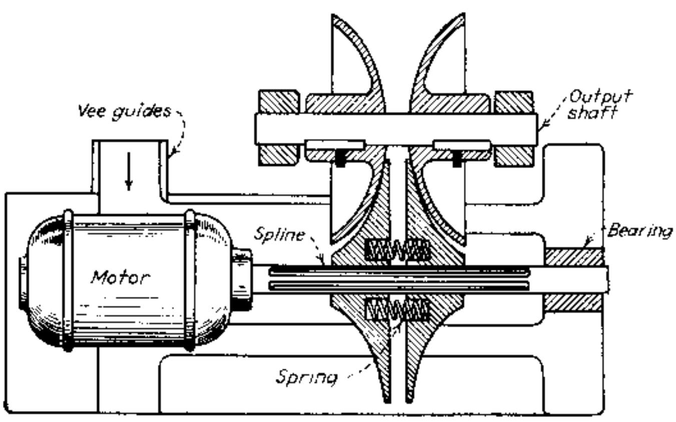


Fig. 3

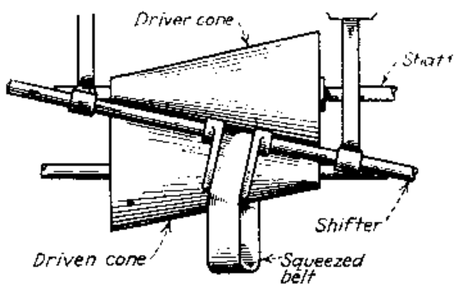


Fig. 4

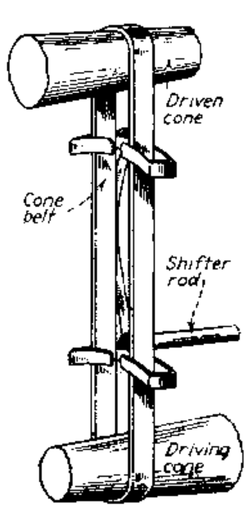


Fig. 5

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

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

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

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

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

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

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

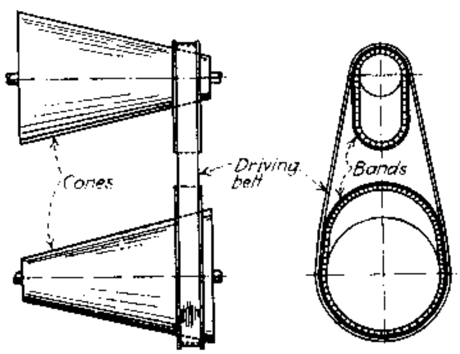


Fig. 6

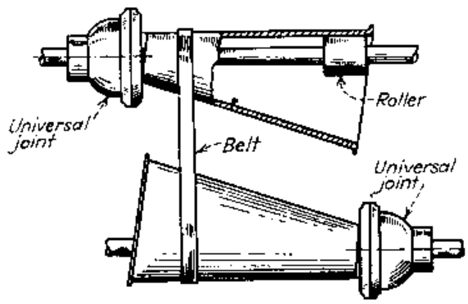


Fig. 7

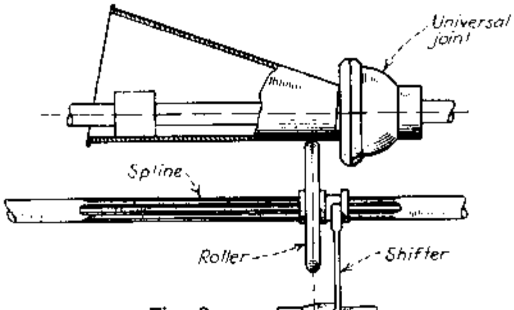


Fig. 8

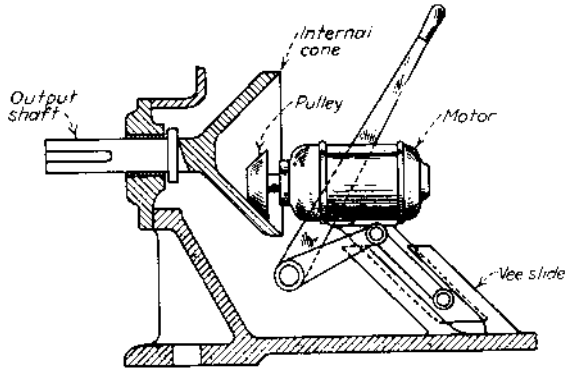


Fig. 9

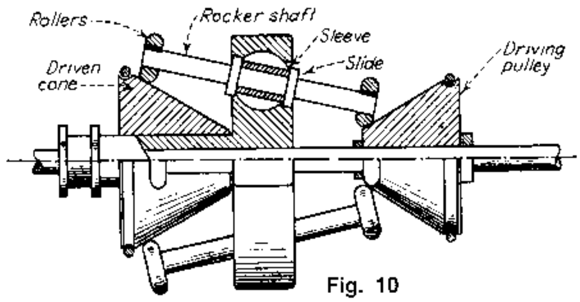


Fig. 10

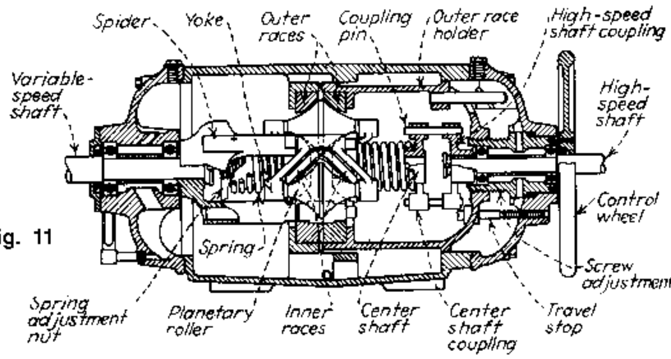


Fig. 11

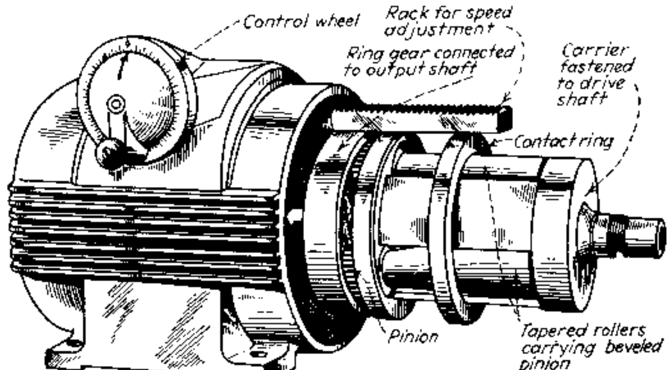


Fig. 12

VARIABLE-SPEED FRICTION DRIVES

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

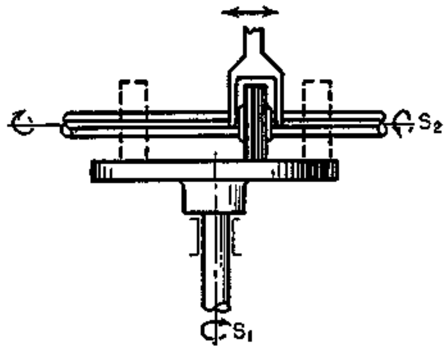


Fig. 1 A disk and roller drive. The roller is moved radially on the disk. Its speed ratio depends upon the operating diameter of the disk. The direction of relative rotation of the shafts is reversed when the roller is moved past the center of the disk, as indicated by dotted lines.

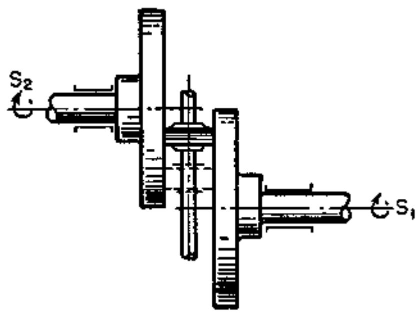


Fig. 2 Two disks have a free-spinning, movable roller between them. This drive can change speed rapidly because the operating diameters of the disks change in an inverse ratio.

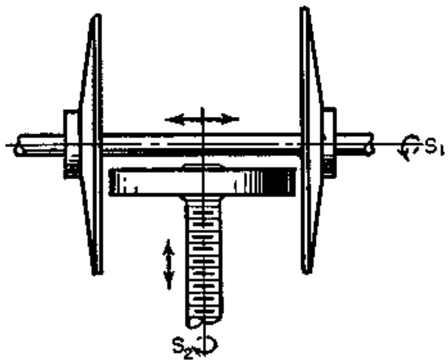


Fig. 3 Two disks are mounted on the same shaft and a roller is mounted on a threaded spindle. Roller contact can be changed from one disk to the other to change the direction of rotation. Rotation can be accelerated or decelerated by moving the screw.

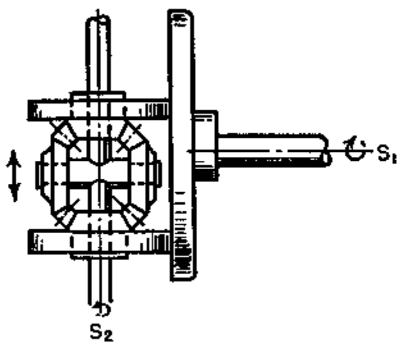


Fig. 4 A disk contacts two differential rollers. The rollers and their bevel gears are free to rotate on shaft S_2. The other two bevel gears are free to rotate on pins connected by S_2. This drive is suitable for the accurate adjustment of speed. S_2 will have the differential speed of the two rollers.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

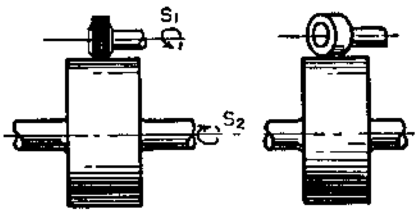


Fig. 5 This drive is a drum and roller. A change of speed is performed by skewing the roller relative to the drum.

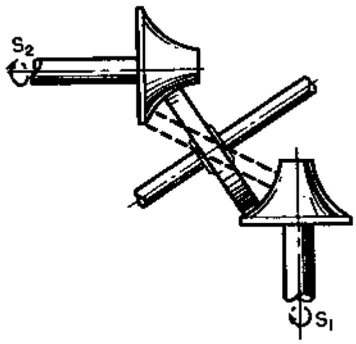


Fig. 6 This drive consists of two spherical cones on intersecting shafts and a free roller.

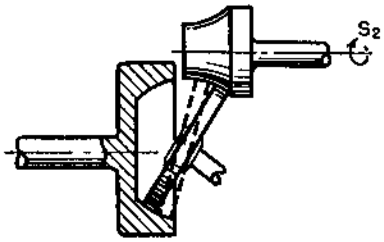


Fig. 7 This drive consists of a spherical cone and groove with a roller. It can be used for small adjustments in speed.

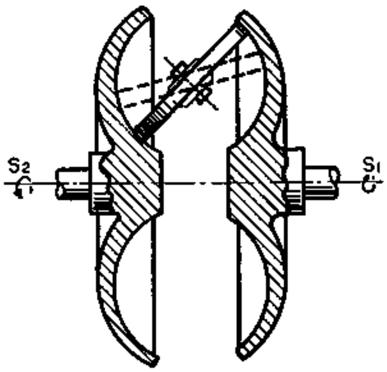


Fig. 8 This drive consists of two disks with torus contours and a free rotating roller.

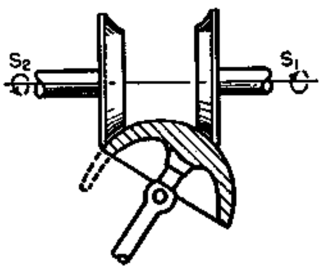


Fig. 9 This drive consists of two disks with a spherical free rotating roller.

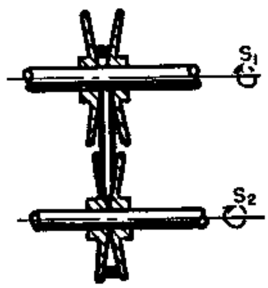


Fig. 10 This drive has split pulleys for V belts. The effective diameter of the belt grip can be adjusted by controlling the distance between the two parts of the pulley.

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

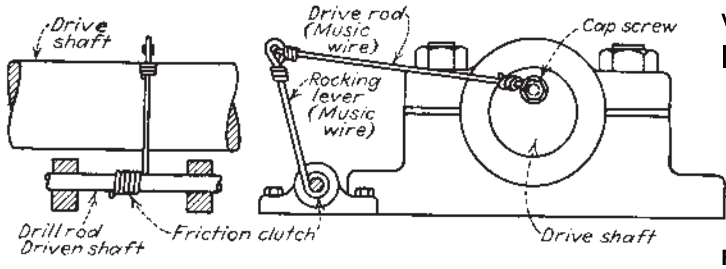


Fig. 1

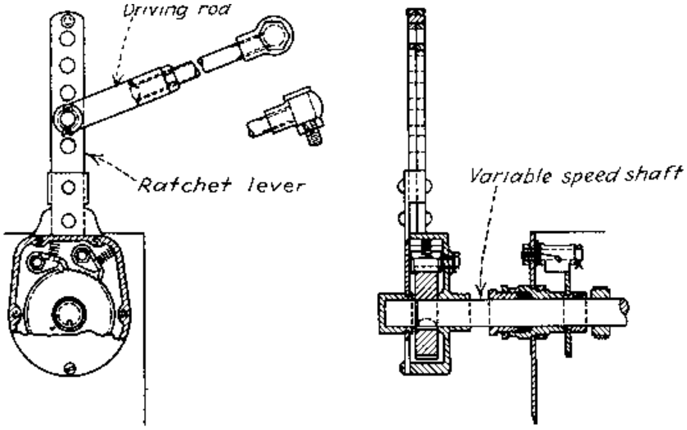


Fig. 2

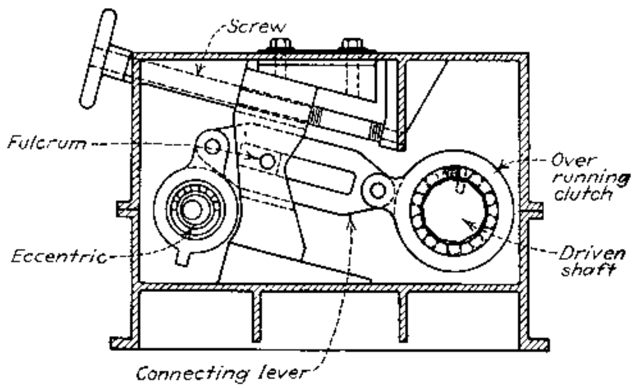


Fig. 3

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

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

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

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

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

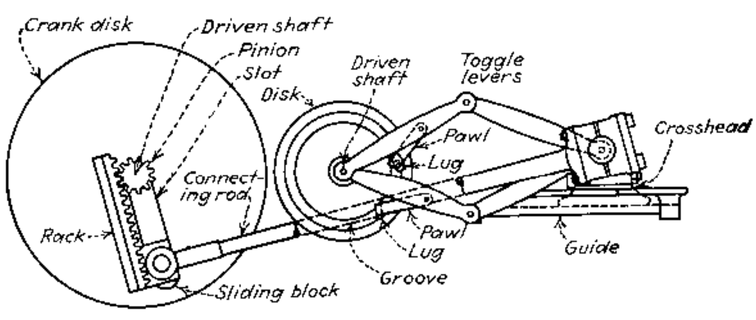


Fig. 4

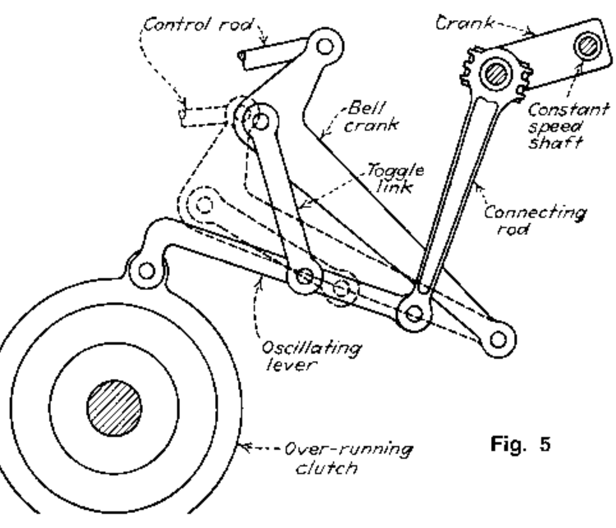


Fig. 5

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

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

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

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

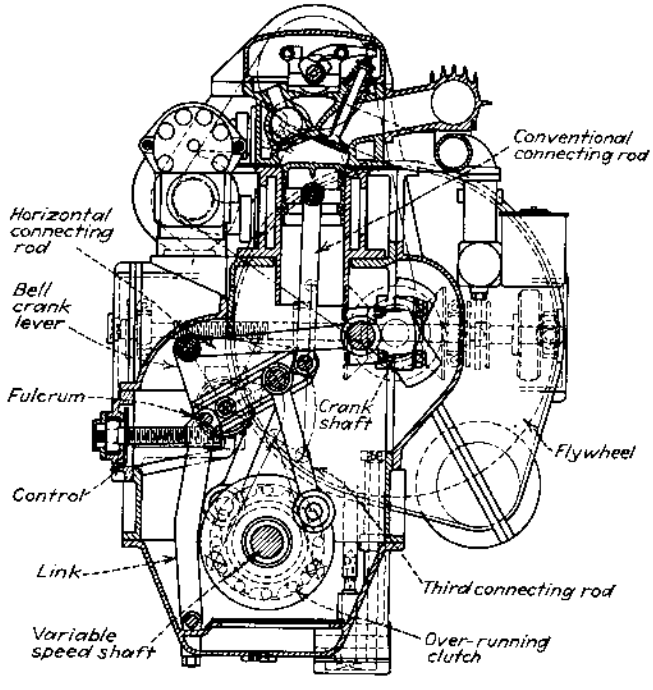


Fig. 6

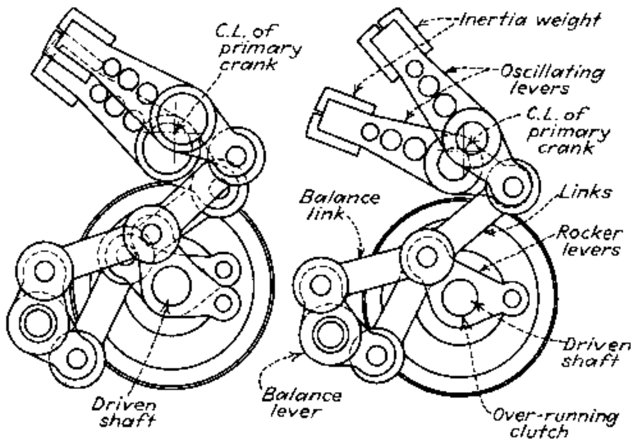


Fig. 7

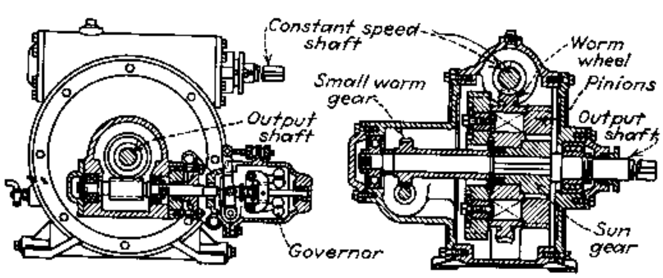


Fig. 8

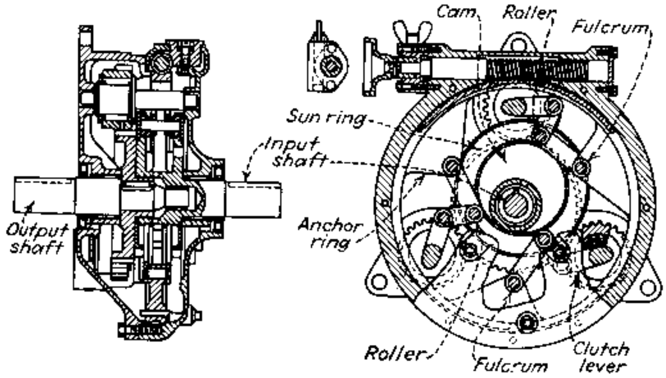


Fig. 9

PRECISION BALL BEARINGS REPLACE GEARS IN TINY SPEED REDUCERS

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

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

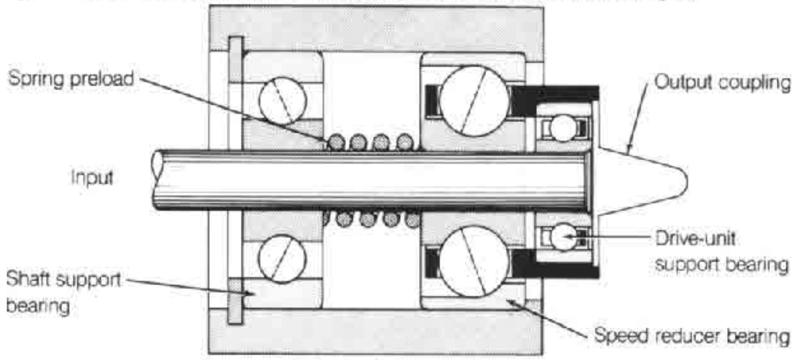
The interest in ball bearing reducers stems from the pressure on mechanical engineers to make their designs more compact to match the miniaturization gains in the electronic fields.

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

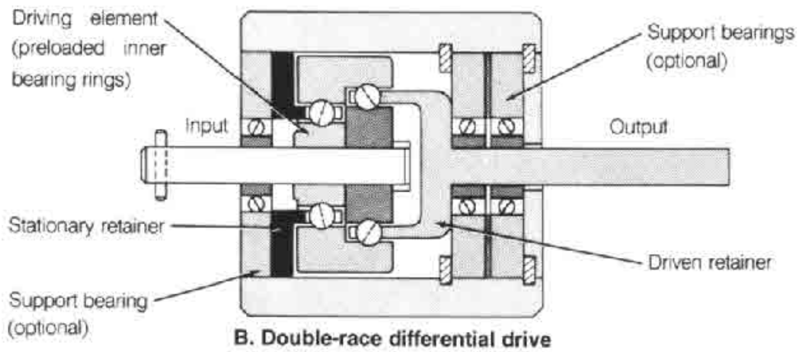
Power transmission functions occur with either a single bearing or with two or more in tandem. Contact friction or traction between the bearing components transmits the torque. To prevent slippage, the bearings are preloaded just the right amount to achieve balance between transmitted torque and operating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

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

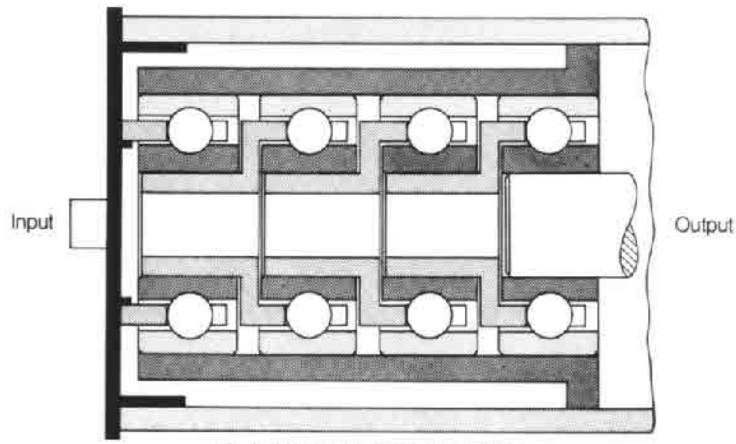
Fig. 1. Ingenious bearing arrangements provide high speed-changes.



A. concentric shaft-torque increaser



B. Double-race differential drive



C. Multi-bearing speed reducer

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

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

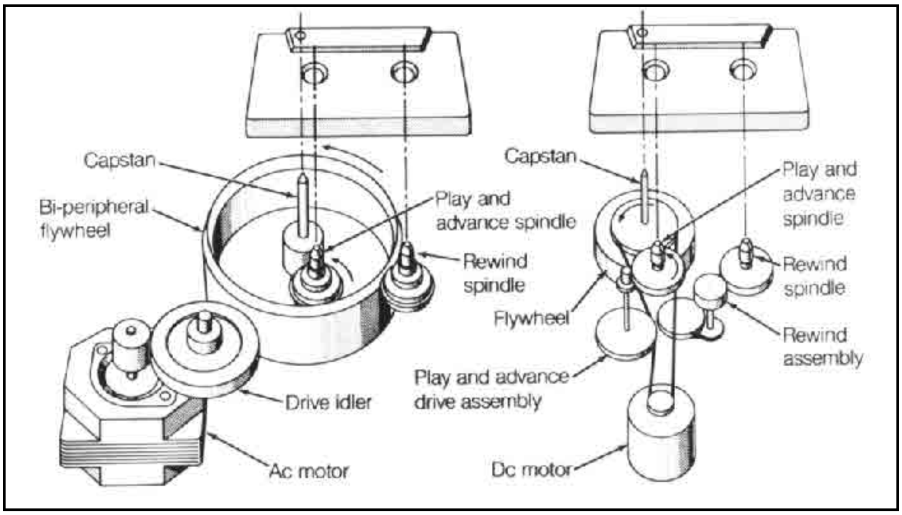
The three MPB units (Fig. 1) illustrate the variety of designs possible:

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

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

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

MULTIFUNCTION FLYWHEEL SMOOTHES FRICTION IN TAPE CASSETTE DRIVE



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

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

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

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

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

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

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

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

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

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

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

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

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

Output speed

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

Output torque

$$T_4 = 2T_3 = 2RT_2$$

Output hp

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

hp increase

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

Speed variation

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

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

Fig. 1

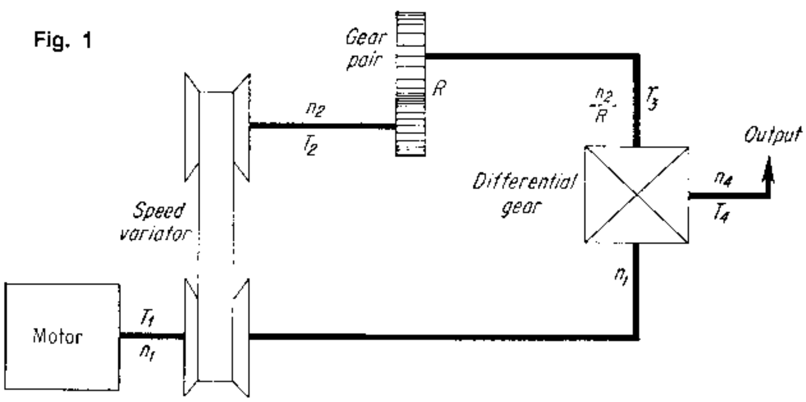


Fig. 2

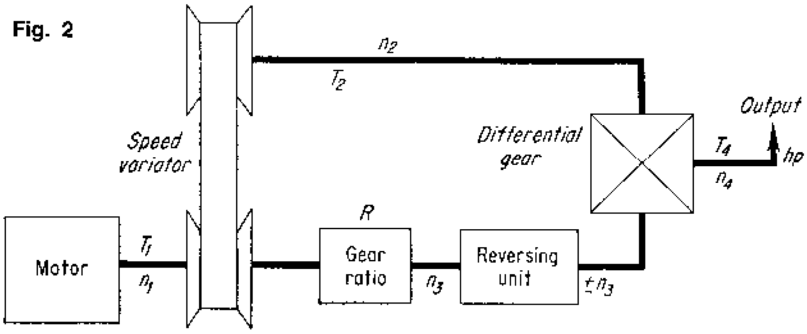


Fig. 3

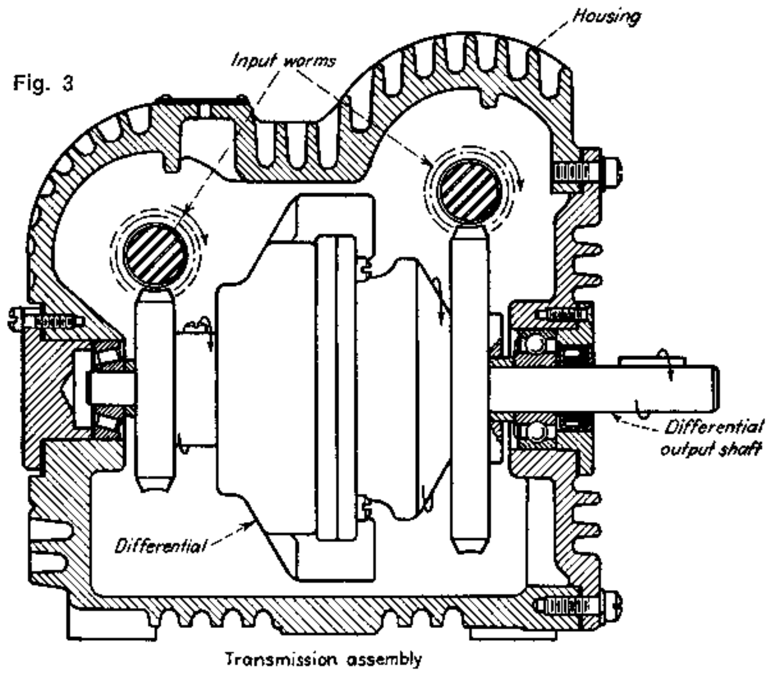


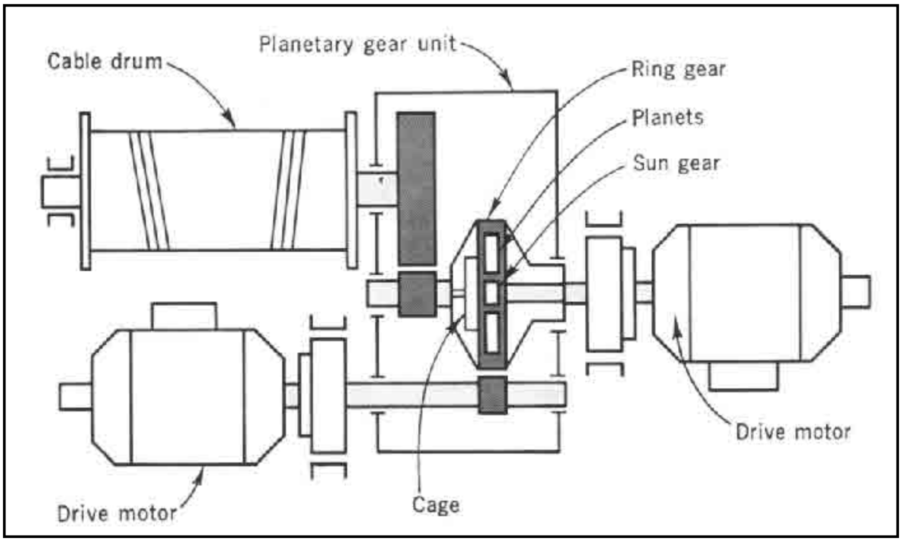
Fig. 3 A variable-speed transmission consists of two sets of worm gears feeding a differential mechanism. The output shaft speed depends on the difference in rpm between the two input worms. When the worm speeds are equal, output is zero. Each worm shaft carries a cone-shaped pulley. These pulley are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

TWIN-MOTOR PLANETARY GEARS PROVIDE SAFETY PLUS DUAL-SPEED

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

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

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



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

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

The diagram shows an installation for a steel mill crane.

HARMONIC-DRIVE SPEED REDUCERS

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

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

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

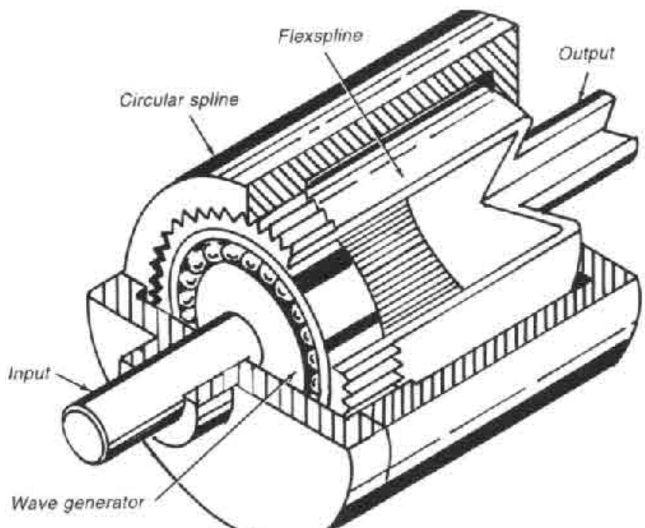


Fig. 1 Exploded view of a typical harmonic drive showing its principal parts. The flexspline has a smaller outside diameter than the inside diameter of the circular spline, so the elliptical wave generator distorts the flexspline so that its teeth, 180º apart, mesh.

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

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

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

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

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

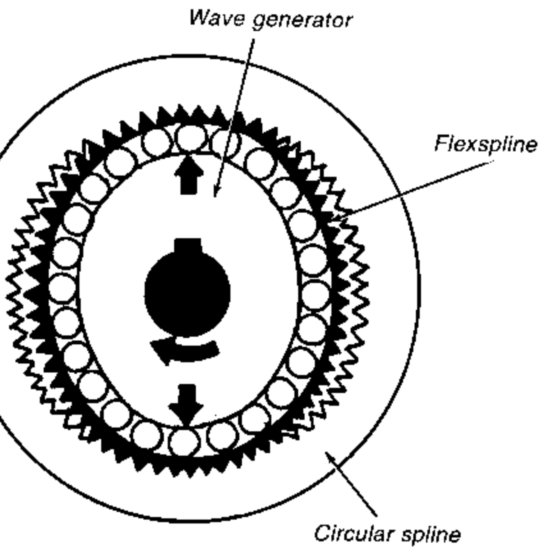
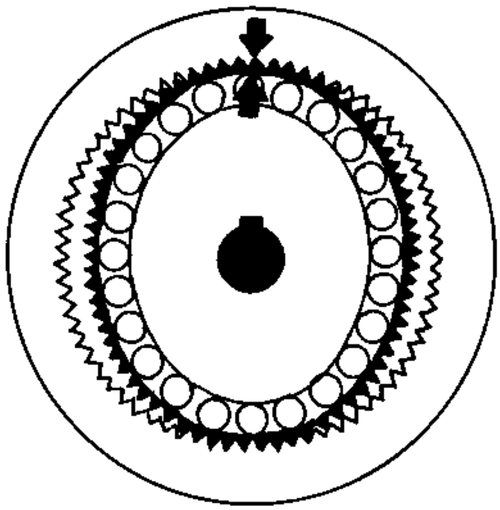


Fig. 2 Schematic of a typical harmonic drive showing the mechanical relationship between the two splines and the wave generator.



A

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

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

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

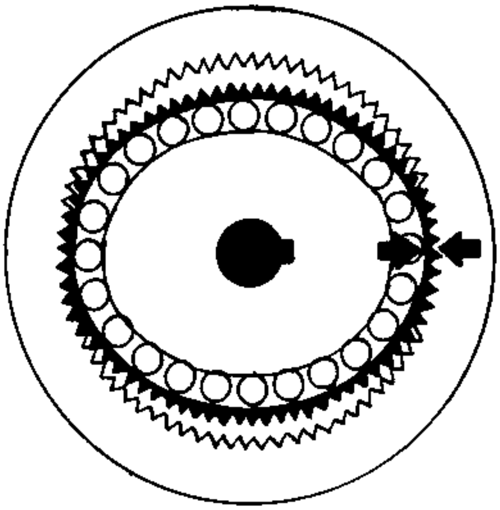
As the wave generator rotates 90° clockwise, as shown in Fig. 3(*B*), the inside arrow is still pointing at the same flexspline tooth, which has begun its counterclockwise rotation. Without full tooth disengagement at the areas of the minor axis, this rotation would not be possible.

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

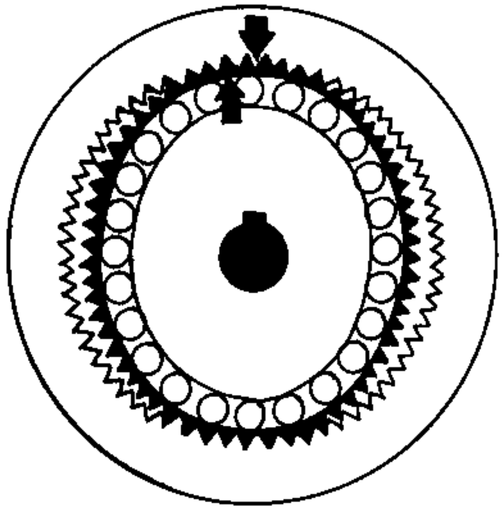
$$GR = \frac{FS}{CS - FS}$$

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

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



B



C

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

Advantages and Disadvantages

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

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

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

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

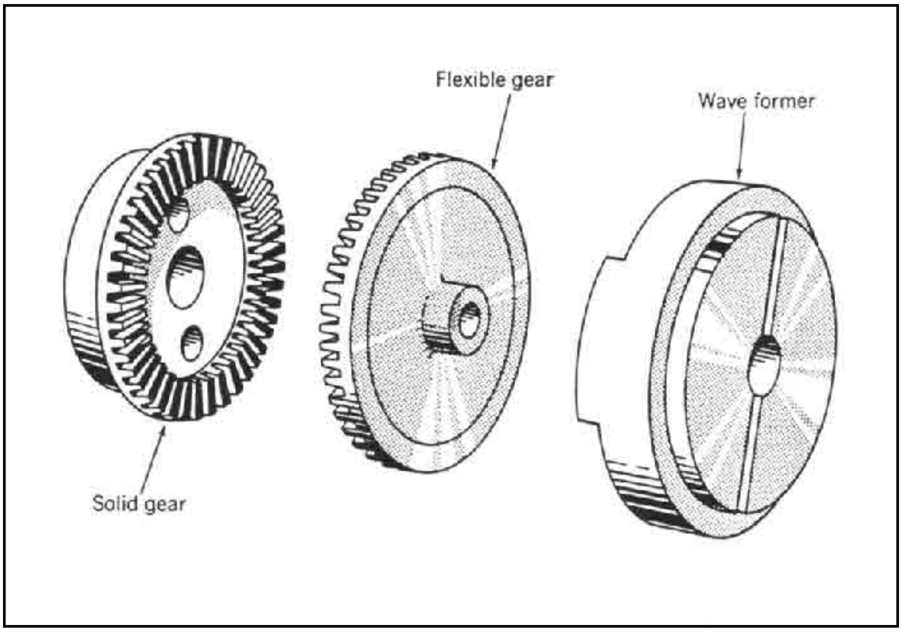
At least one manufacturer claims to have overcome this problem with redesigned gear teeth. In a new design, one company replaced the original involute teeth on the flexspline and circular spline with noninvolute teeth. The new design is said to reduce stress concentration, double the fatigue limit, and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

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

FLEXIBLE FACE-GEARS MAKE EFFICIENT HIGH-REDUCTION DRIVES

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

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



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

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

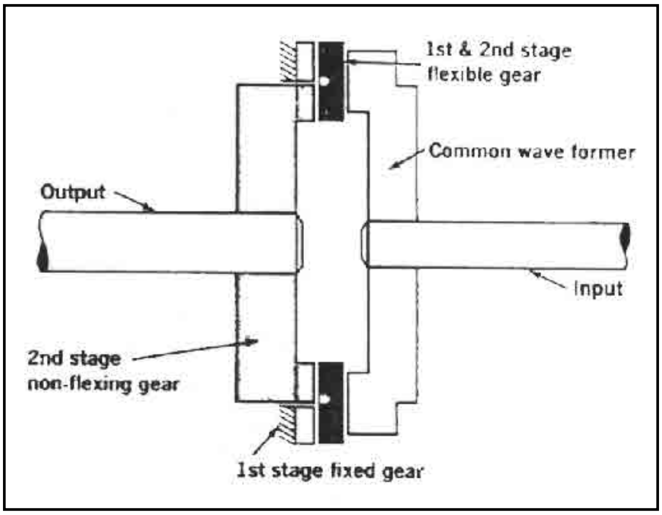
The high-speed input to the system usually drives the wave former. Low-speed output can be derived from either the flexible or the solid face gear; the gear not connected to the output is fixed to the housing.

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

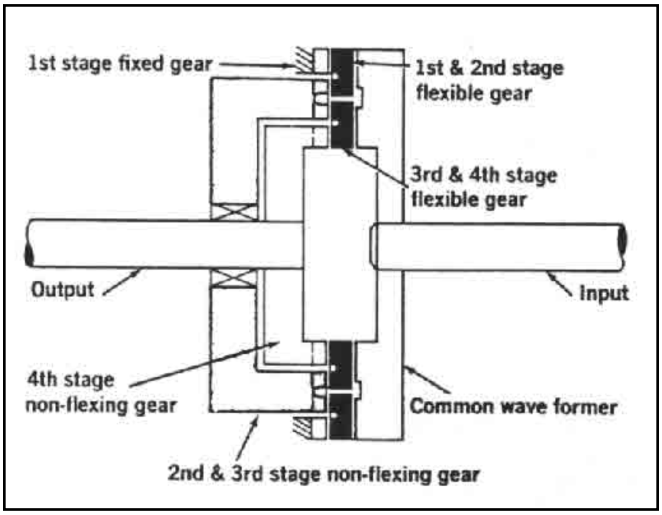
On each revolution of the wave former, there is a relative motion between the two gears that equals the difference in their numbers of teeth. The reduction ratio equals the number of teeth in the output gear divided by the difference in their numbers of teeth.

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

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



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



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

COMPACT ROTARY SEQUENCER

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

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

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

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

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

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

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

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

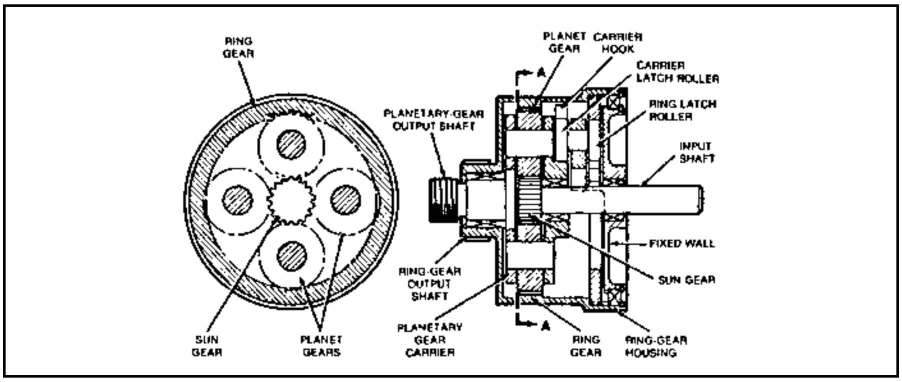


Fig. 1 The Rotary Sequencer has a ring-gear output (in color) that is coaxial with a planetary-gear output (in gray). Clockwise rotation of the input is converted to clockwise rotation of the planetary-gear output followed by counterclockwise rotation of the ring-gear output. The sequence is controlled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

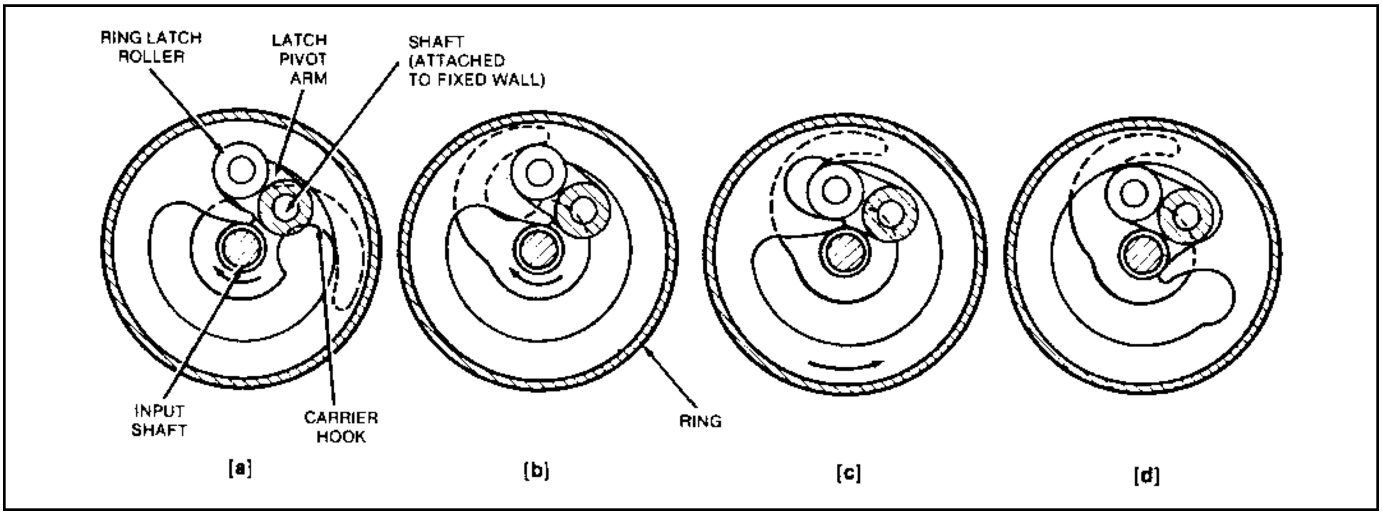
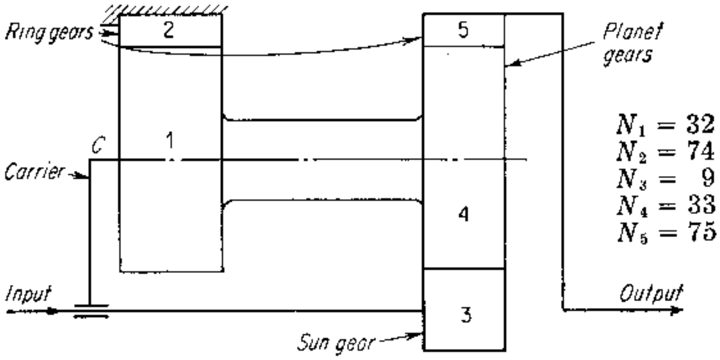


Fig. 2 The Latch Sequence is shown in four steps: (a) The input shaft rotates the carrier clockwise while the ring latch roller holds the ring gear stationary; (b) the carrier hook begins to engage the carrier latch roller; (c) the ring latch roller begins to move out of its slot, and the carrier motion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

PLANETARY GEAR SYSTEMS

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

MISSILE SILO COVER DRIVE



Ring gear 2 fixed; ring gear 5 output

Speed-ratio equation

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

Symbols

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

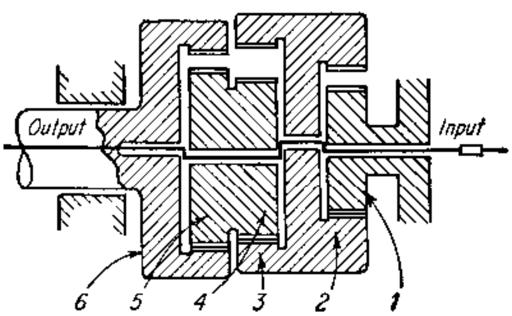
N = number of teeth

R = overall speed reduction ratio

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

DOUBLE-ECCENTRIC DRIVE

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

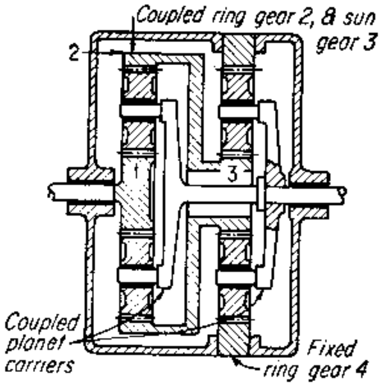


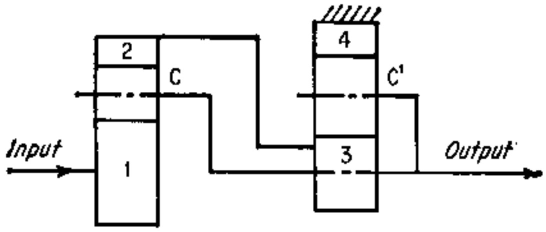
$$R = \frac{1}{1 - \dfrac{N_5 N_3 N_1}{N_6 N_4 N_2}}$$

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

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

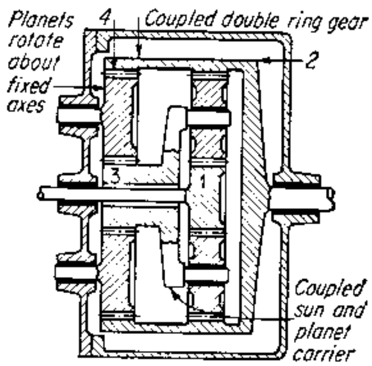
(A)

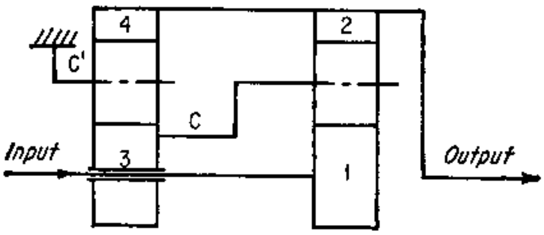




$$R = 1 - \frac{N_2 N_4}{N_1 N_3}$$

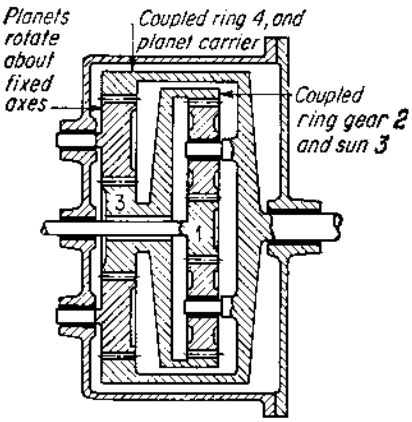
(B)

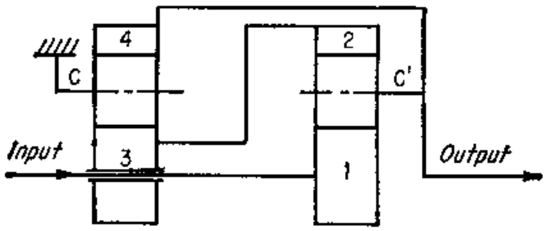




$$R = \left(1 + \frac{N_2}{N_1}\right)\left(-\frac{N_4}{N_3}\right) - \frac{N_2}{N_1}$$

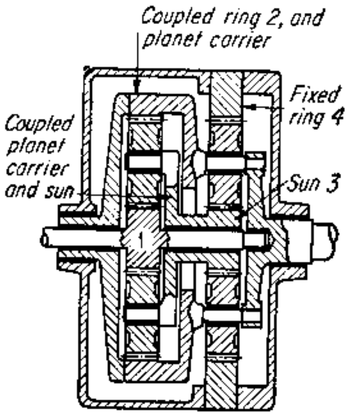
(C)

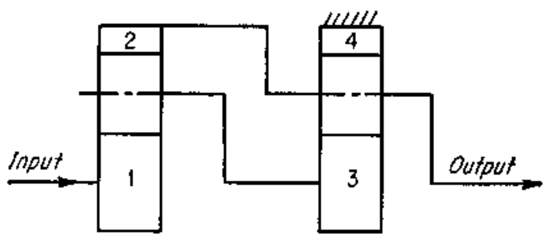




$$R = 1 + \frac{N_2}{N_1}\left(1 + \frac{N_4}{N_3}\right)$$

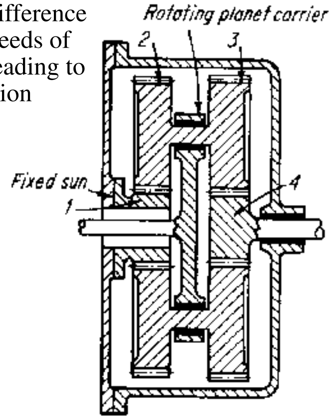
(D)





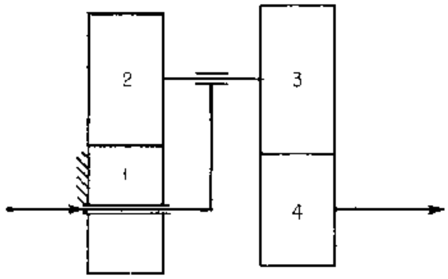
$$R = 1 + \frac{N_4}{N_3}\left(1 + \frac{N_2}{N_1}\right)$$

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

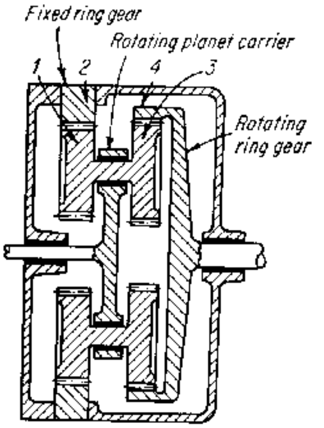


(A)

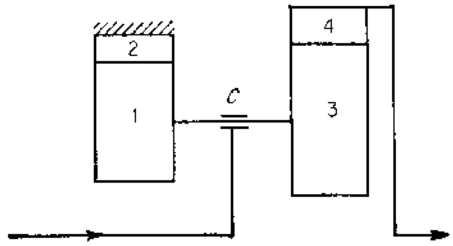
$N_1 = 20$
$N_2 = 31$
$N_3 = 32$
$N_4 = 19$



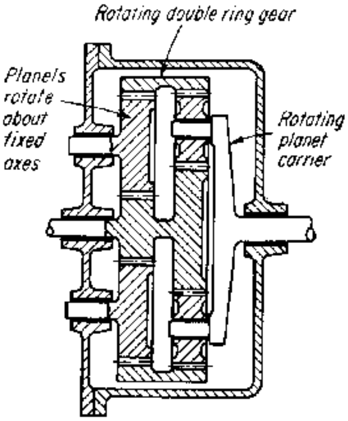
$$R = \frac{1}{1 - \dfrac{N_3 N_1}{N_4 N_2}} = \frac{1}{1 - \dfrac{(32)\,(20)}{(19)\,(31)}} = -11.549$$



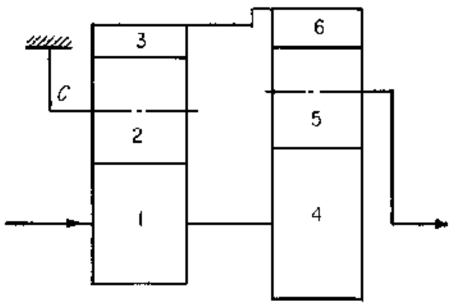
(B)



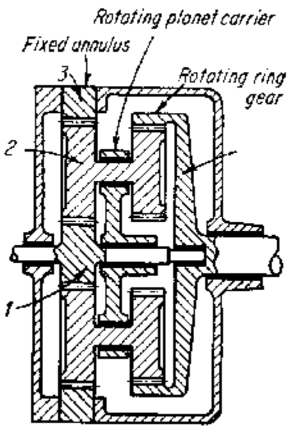
$$R = \frac{1}{1 - \dfrac{N_3 N_2}{N_4 N_1}}$$



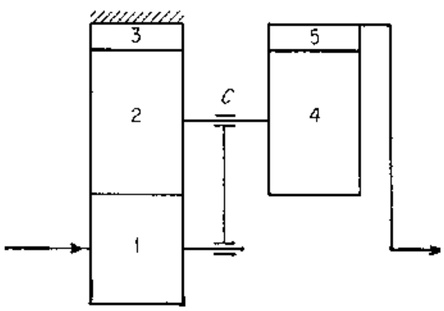
(C)



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

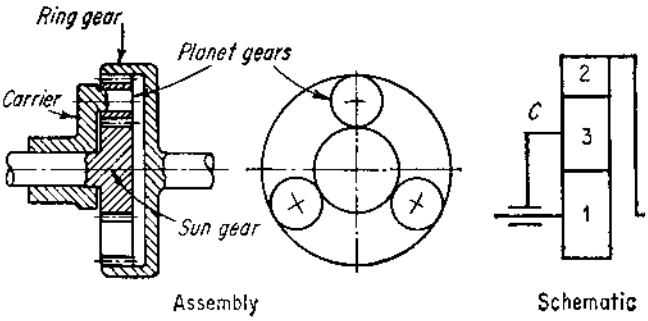


(D)

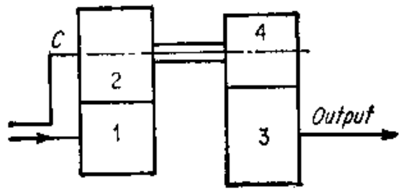


$$R = \frac{1 + (N_3/N_1)}{1 - \dfrac{N_4 N_3}{N_5 N_2}}$$

SIMPLE PLANETARIES AND INVERSIONS



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



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

HUMPAGE'S BEVEL GEARS

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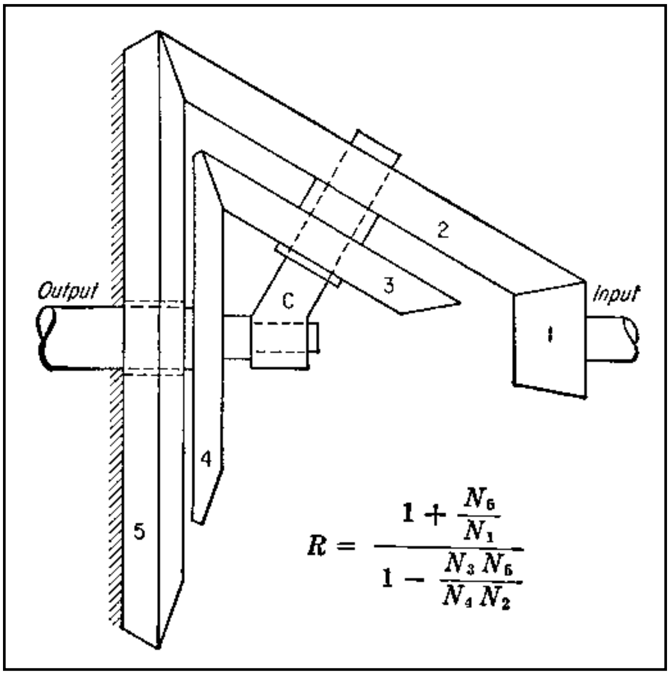
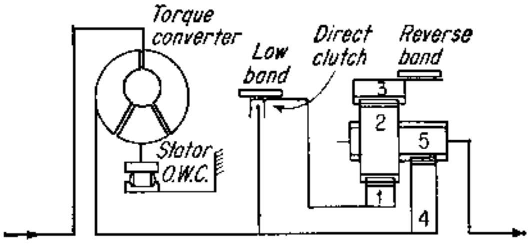


$$R = \frac{1 + \dfrac{N_6}{N_1}}{1 - \dfrac{N_3 N_5}{N_4 N_2}}$$

References

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

TWO-SPEED FORDOMATIC (Ford Motor Co.)

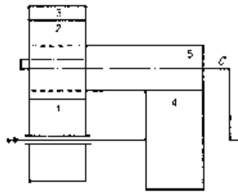


Low gear— gear 1 fixed

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

Reverse gear— gear 3 fixed

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

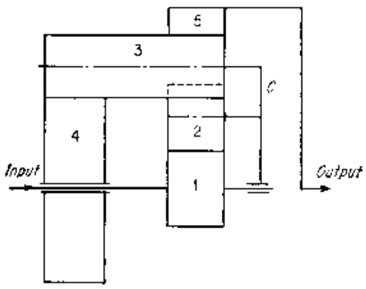


$N_1 = 27$
$N_2 = 31$
$N_3 = 90$
$N_4 = 36$
$N_5 = 19$

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

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

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



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

Low gear—Input to 1 C fixed

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

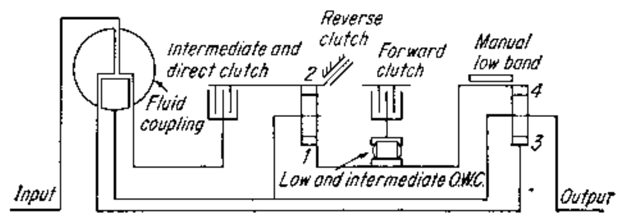
Intermediate gear—
Input to 1, gear 4 fixed

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

Reverse gear—
Input to 4, C fixed

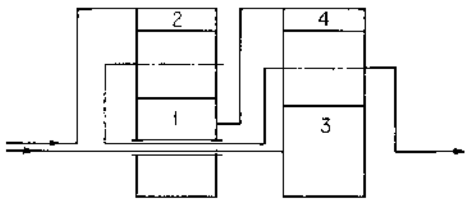
$$R = \frac{N_5}{N_4} = -2.0$$

HYDRAMATIC 3-SPEED TRANSMISSION (General Motors)



Low gear—
Input to 3, 4 fixed

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



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

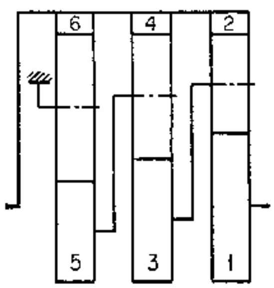
Intermediate gear—Input to 2, 1 fixed

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

Reverse gear—Input to 3, 2 fixed

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

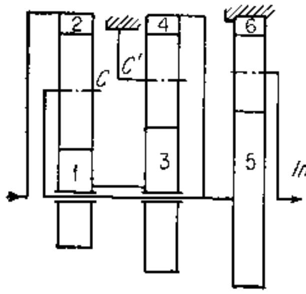
TRIPLE PLANETARY DRIVES

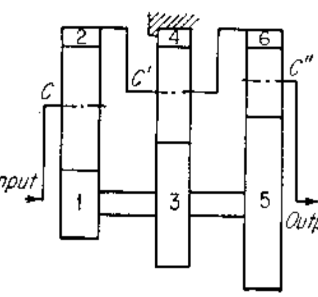


Input to gear 1, output from gear 6

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

(B)





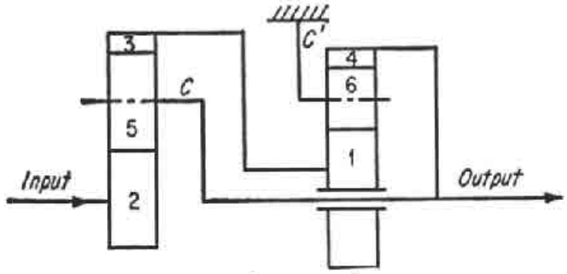
(C)

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

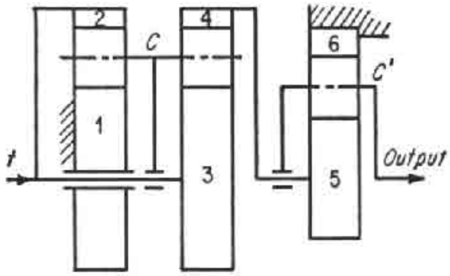
(C) $$R = \left[1 + \frac{N_4/N_3}{1 + (N_2/N_1)}\right] \bigg/ \left[1 + \frac{N_4/N_3}{1 + (N_6/N_5)}\right]$$

FORD TRACTOR DRIVES

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

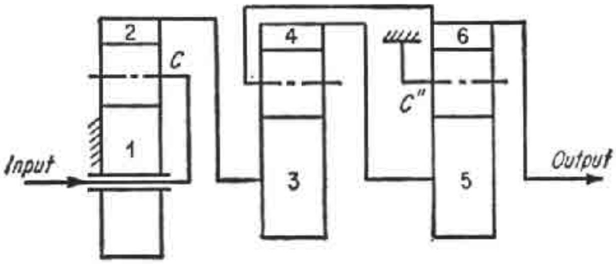


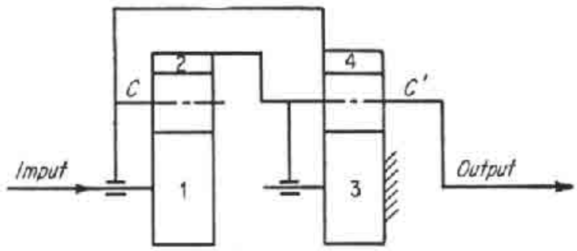
$$N_1 = 46$$
$$N_2 = 77$$
$$N_3 = 29$$
$$N_4 = 85$$



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

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



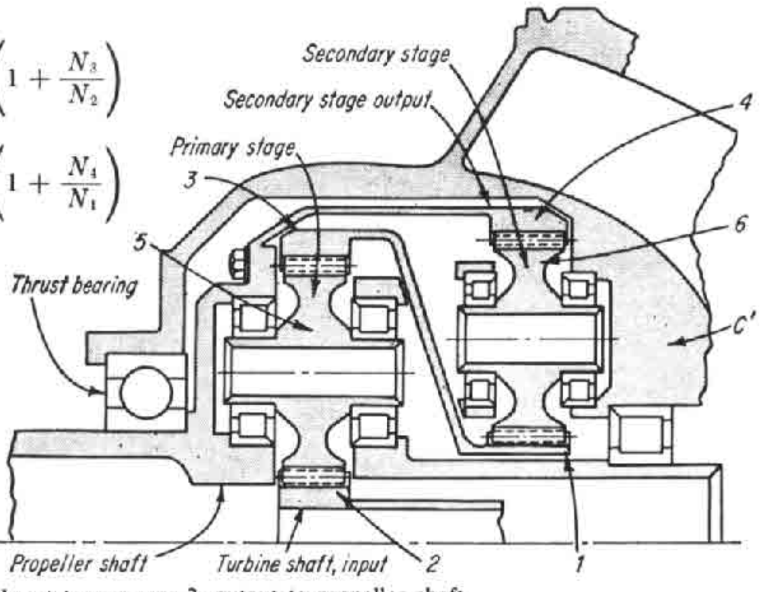


(C) $$R = \frac{1}{1 + \dfrac{N_1}{N_2}}\left[1 + \frac{N_4}{N_3}\left(1 + \frac{N_6}{N_5}\right)\right]$$

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

LYCOMING TURBINE DRIVE

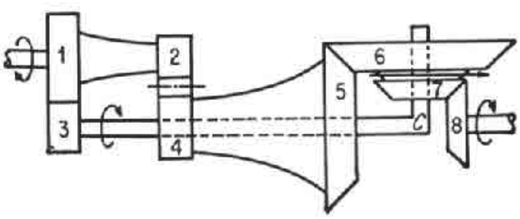
$$R = \left(1 + \frac{N_3}{N_2}\right) \times \left(1 + \frac{N_4}{N_1}\right)$$



Input to sun gear 2, output to propeller shaft.

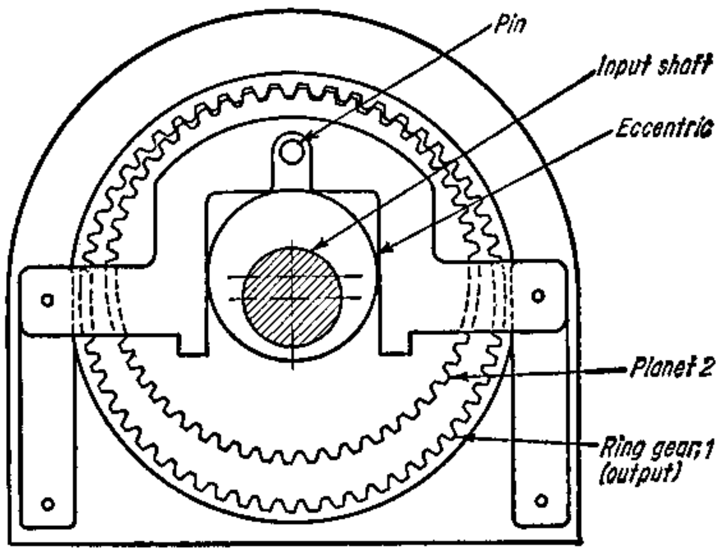
Basically same system as the Ford tractor drive, (gears are numbered the same way) and will have the same speed-ratio.

COMPOUND SPUR-BEVEL GEAR DRIVE

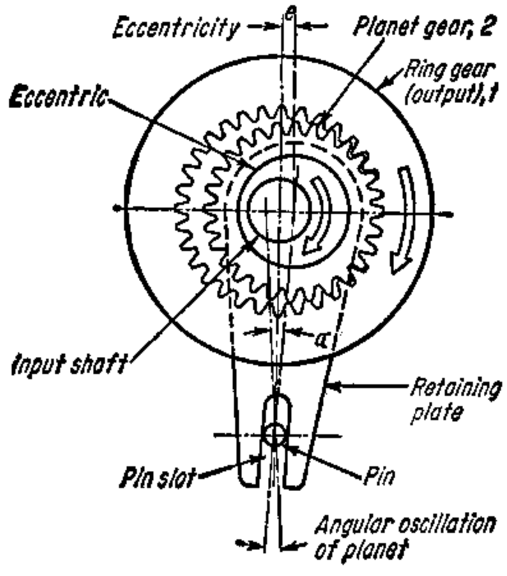


$$R = \frac{1}{-\dfrac{N_1}{N_3}\left(1 + \dfrac{N_7 N_5}{N_8 N_6}\right) - \dfrac{N_7 N_5 N_2}{N_8 N_6 N_4}}$$

TWO-GEAR PLANETARY DRIVES



(A)

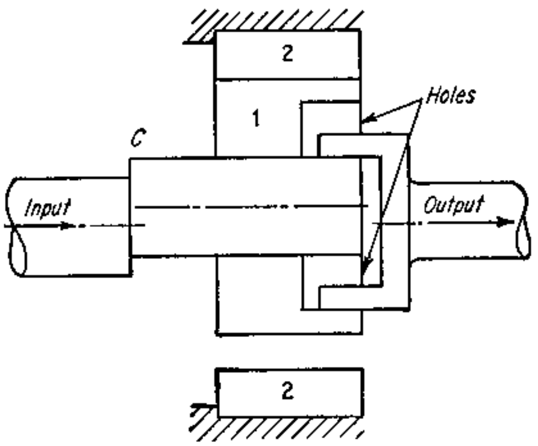


(B)

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

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

PLANOCENTRIC DRIVE

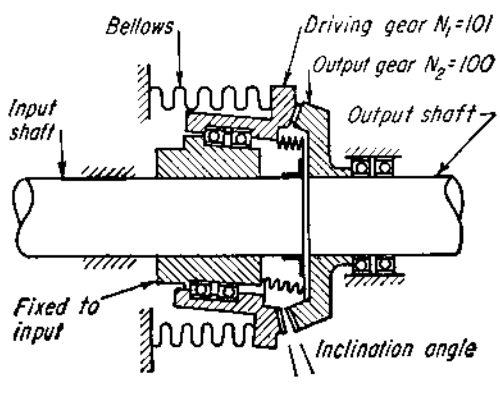


$N_2 = 65$
$N_1 = 64$

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

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

WOBBLE-GEAR DRIVE



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

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

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

NONCIRCULAR GEARS

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

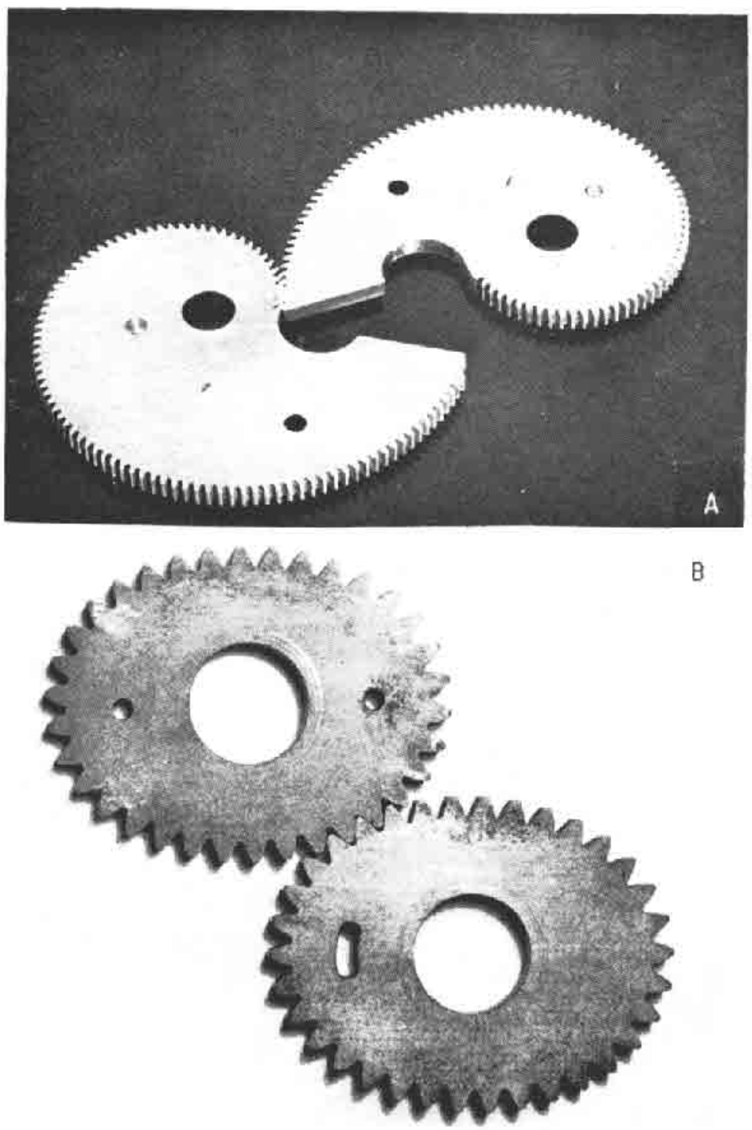
Applications can be classified into two groups:

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

Noncircular Gears

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

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



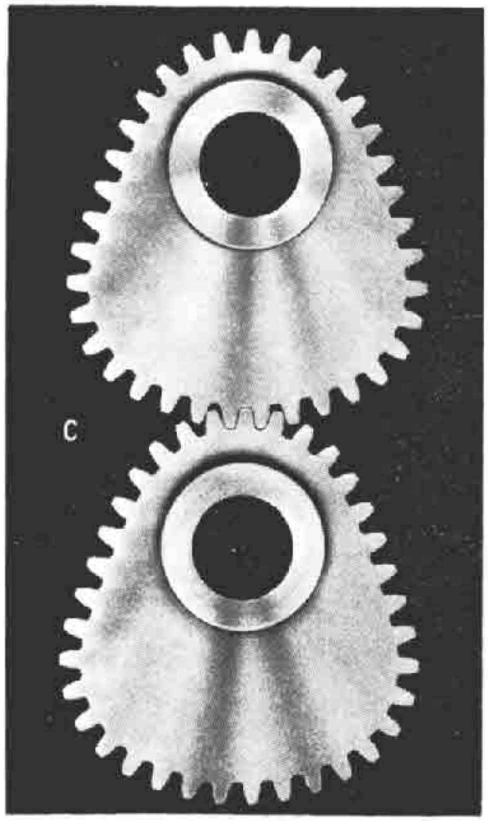


Fig. 1 The logarithmic spiral gears shown in (A), are open-curved. They are usually components in computing devices. The elliptically shaped gears, shown in (B), are closed curved. They are components in automatic machinery. The specially shaped gears, shown in (C), offer a wider range of velocity and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

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

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

DESIGN EQUATIONS

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

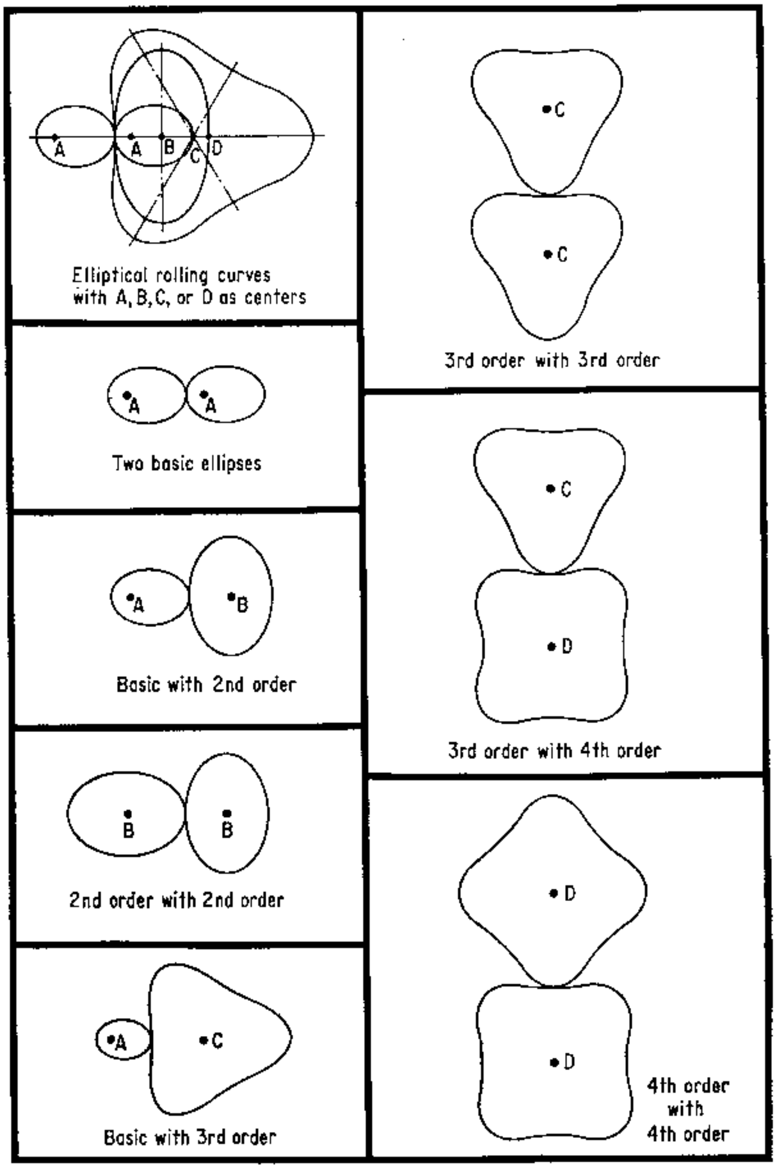


Fig. 2 Basic and High-Order Elliptical Gear Combinations.

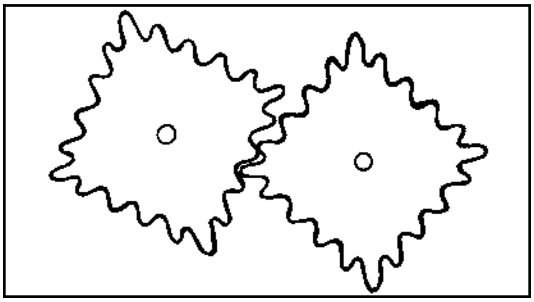


Fig. 3 Square gears seem to defy basic kinematic laws, but they are a takeoff on a pair of fourth-order ellipses.

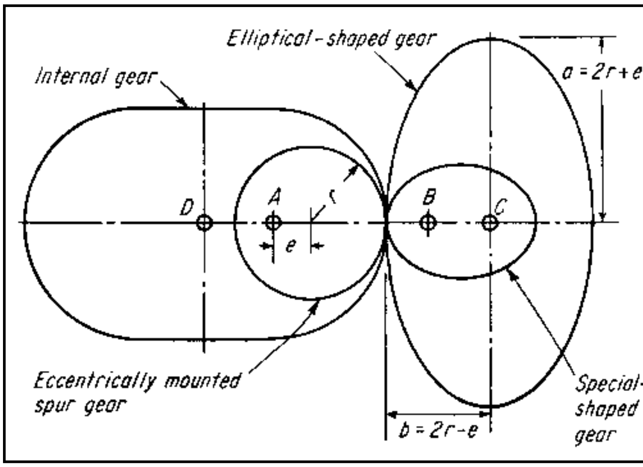


Fig. 4 An eccentric spur gear rotating about point **A,** will mesh properly with any of the three gears shown whose centers are at points B, C and D.

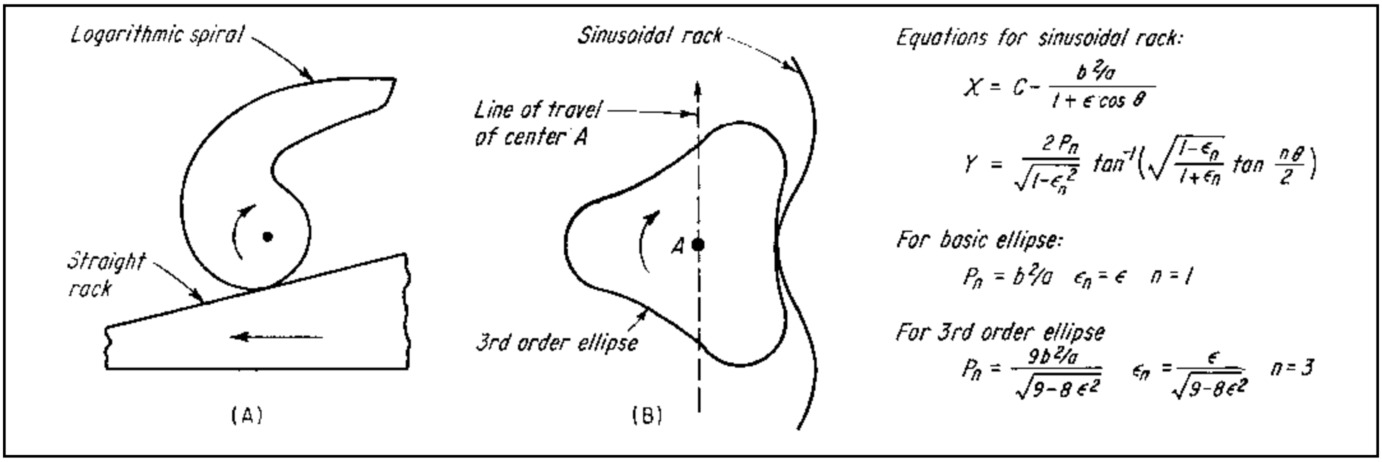
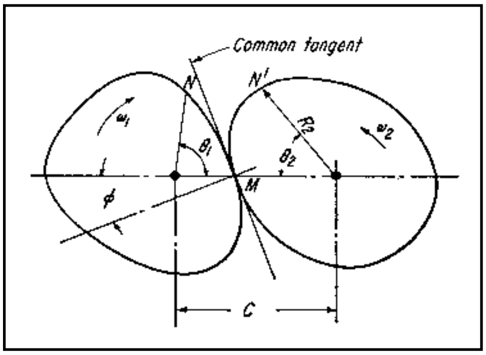


Fig. 5 Rack and gear combinations are possible with noncircular gears. The straight rack for the logarithmic spiral (A) must move obliquely; the center of third-order ellipse (B) follows a straight line.



Symbols

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

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

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

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

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

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

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

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

CHECKING FOR CLOSED CURVES

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

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

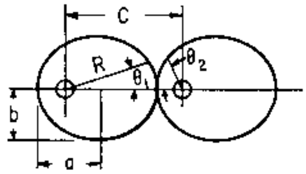
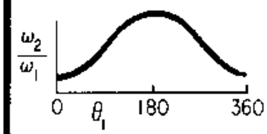
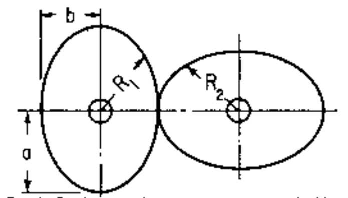
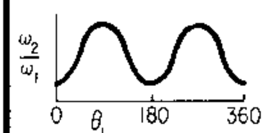
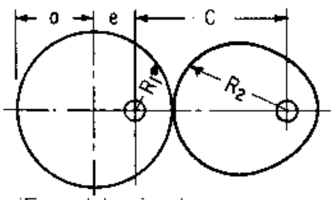
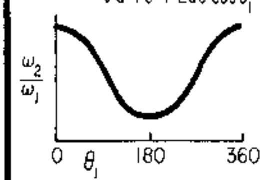
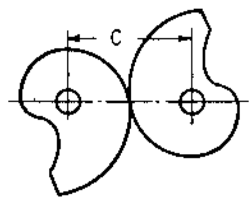
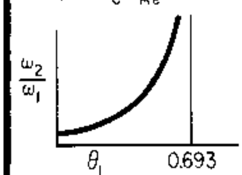
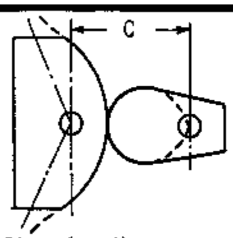
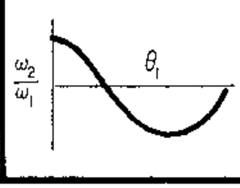
In case II, if $\theta_1 = F(\theta_2)$ and $F(\theta_0) = 0$, the curve is closed with the equation $F(\theta_0 + 2\pi/N_1) = 2_\pi/N_2$ can be satisfied by substituting integers or rational fractions for N_1 and N_2. If fractions must be used to solve this equation, the curve will have double points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

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

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

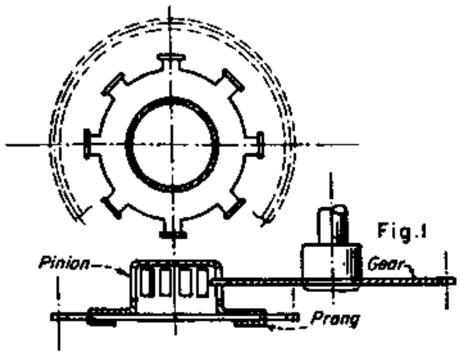
$$4\pi = \int_0^{2\pi} \frac{d\theta_1}{C - f(\theta_1)}$$

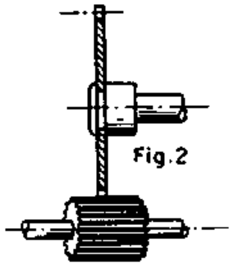
Characteristics of Five Noncircular Gear Systems

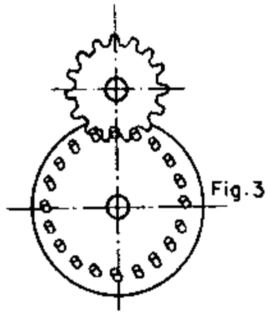
Type	Comments	Basic equations	Velocity equations ω_1 = constant
 Two ellipses rotating about foci	Gears are identical. Comparatively easy to manufacture. Used for quick-return mechanisms, printing presses, automatic machinery	$R = \frac{b^2}{a[1+\epsilon\cos\theta]}$ ϵ = eccentricity $= \sqrt{1-\left(\frac{b}{a}\right)^2}$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

SHEET-METAL GEARS, SPROCKETS, WORMS, AND RATCHETS

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







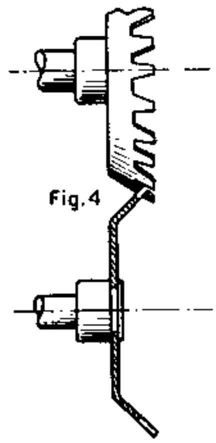
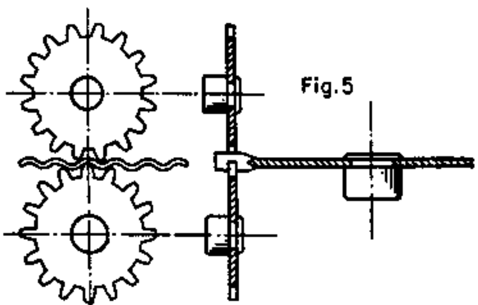


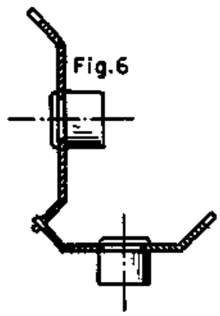
Fig. 1 The pinion is a sheet metal cup with rectangular holes serving as teeth. The meshing gear is sheet metal, blanked with specially formed teeth. The pinion can be attached to another sheet metal wheel by prongs, as shown, to form a gear train.

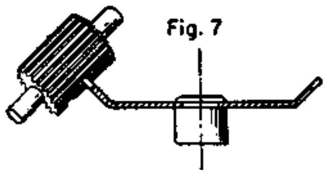
Fig. 2 The sheet-metal wheel gear meshes with a wide-face pinion, which is either extruded or machined. The wheel is blanked with teeth of conventional form.

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

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







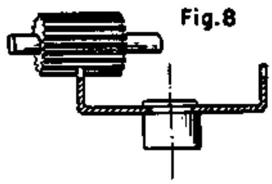


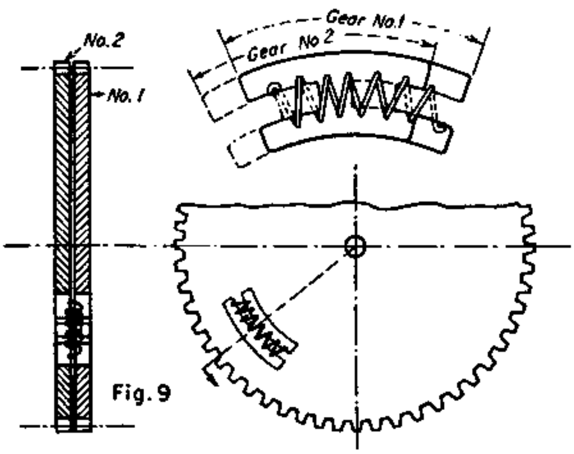
Fig. 5 The horizontal wheel with waves on its out rim replacing teeth, meshes with either one or two sheet-metal pinions. They have specially formed teeth and are mounted on intersecting axes.

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

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

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

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



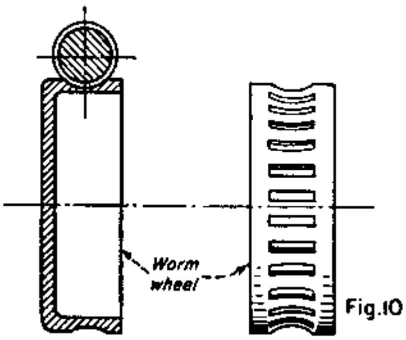


Fig. 10 A sheet metal cup with indentations replacing worm-wheel teeth, meshes with a standard coarse-thread screw.

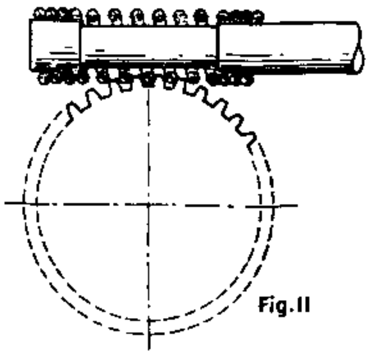


Fig. 11 A blanked wheel, with specially formed teeth, meshes with a helical spring mounted on a shaft, which serves as the worm.

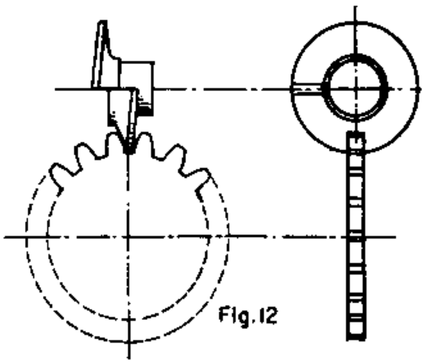
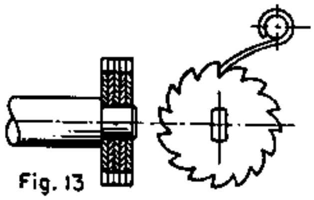


Fig. 12 This worm wheel is blanked from sheet metal with specially formed teeth. The worm is a sheet-metal disk that was split and helically formed.



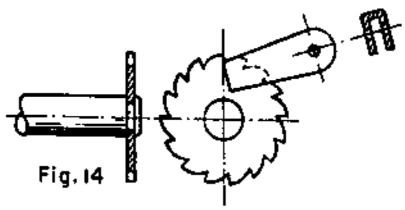


Fig. 13 Blanked ratchets with one-sided teeth are stacked to fit a wide-sheet-metal finger when single thickness is inadequate. The ratchet gears can be spot-welded.

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

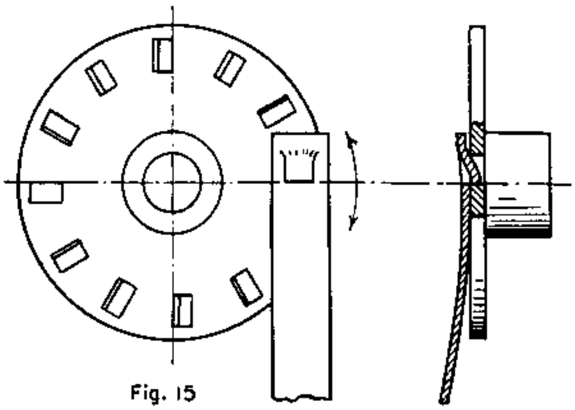


Fig. 15 This wheel is a punched disk with square-punched holes serving as teeth. The pawl is spring steel.

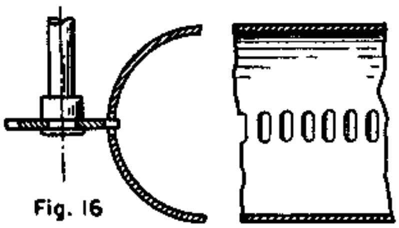
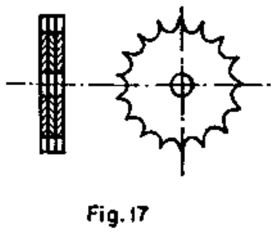


Fig. 16 This sheet-metal blanked pinion, with specially formed teeth, meshes with windows blanked in a sheet metal cylinder. They form a pinion-and-rack assembly.



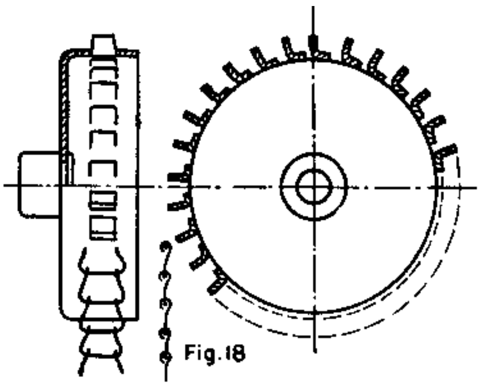


Fig. 17 This sprocket, like that in Fig. 13, can be fabricated from separate stampings.

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

HOW TO PREVENT REVERSE ROTATION

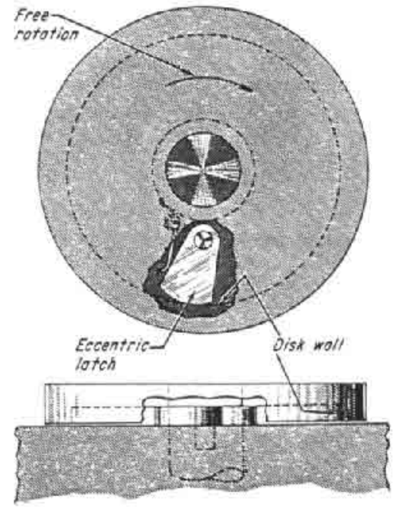


Fig. 1 An eccentric latch allows the shaft to rotate in one direction; any attempted reversal immediately causes the latch to wedge against the disk wall.

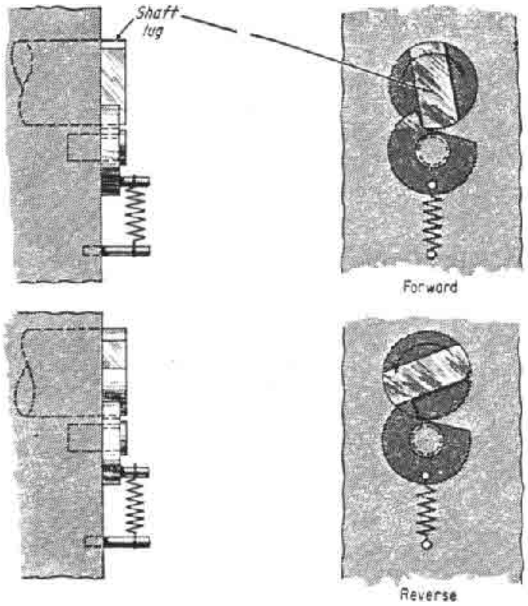


Fig. 2 A lug on a shaft pushes the notched disk free during normal rotation. The disk periphery stops the lug to prevent reverse rotation.

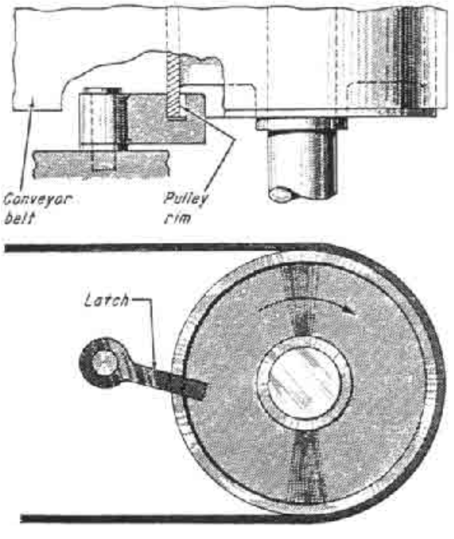


Fig. 3 A latch on the rim of the pulley is free only when the rotation is in the direction shown. This arrangement is ideal for conveyor-belt pulleys.

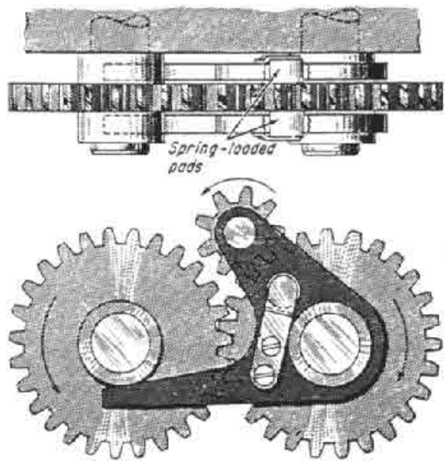


Fig. 4 Spring-loaded friction pads contact the right gear. The idler meshes and locks the gear set when the rotation is reversed.

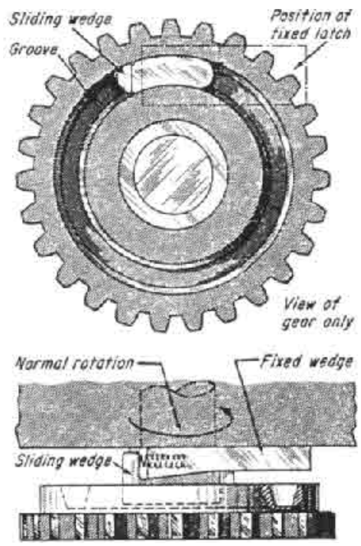


Fig. 5 A fixed wedge and sliding wedge tend to disengage when the gear is turning clockwise. The wedges jam in the reverse direction.

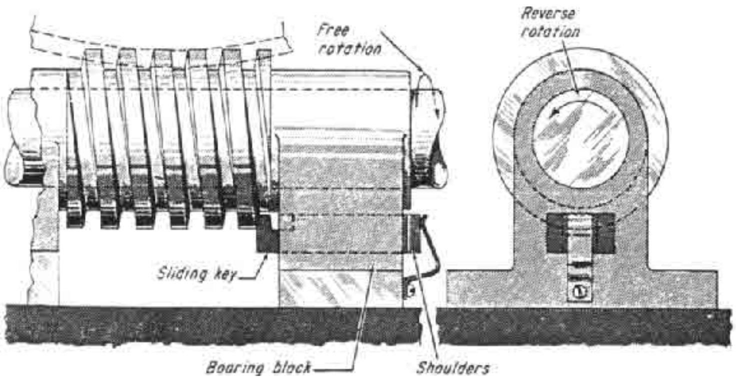


Fig. 6 A sliding key has a tooth which engages the worm threads. In reverse rotation, the key is pulled in until its shoulders contact the block.

GEAR-SHIFT ARRANGEMENTS

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

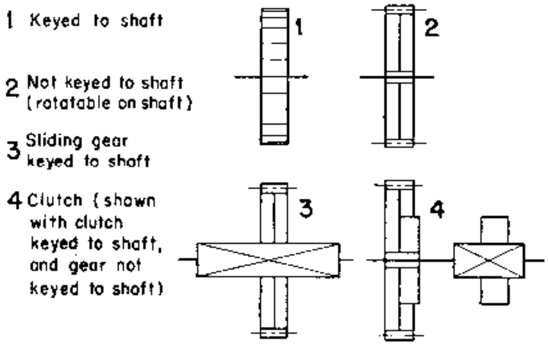


Fig. 1 The schematic symbols used in the following illustrations to represent gears and clutches.

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

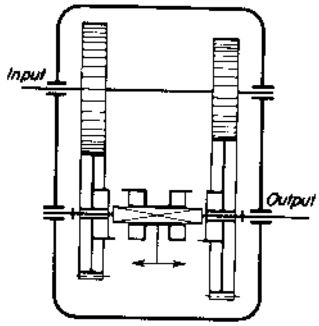


Fig. 3 Sliding-change drive. Gears are meshed by lateral sliding. Up to three gears can be mounted on a sliding sleeve. Only one pair is in mesh in any operating position. This drive is simpler, cheaper, and more extensively used than the drive of Fig. 2. Chamfering the sides of the teeth eases their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

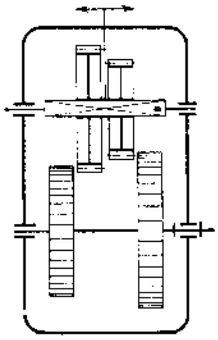


Fig. 4 Swivel-gear drive. Output gears are fastened to the shaft. A handle is pushed down, then shifted laterally to obtain transmission through any output gear. This drive is not suitable for the transmission of large torques because the swivel gear tends to vibrate. Its overall ratio should not exceed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

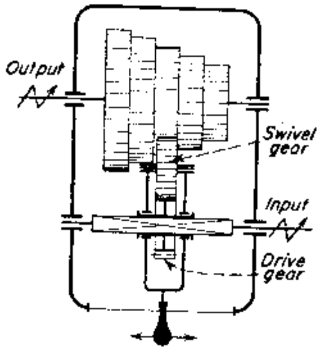


Fig. 5 Slide-key drive. A spring-loaded slide key rides inside a hollow output shaft. The slide key snaps out of the shaft when it is in position to lock a specific change gear to the output shaft. No central position is shown.

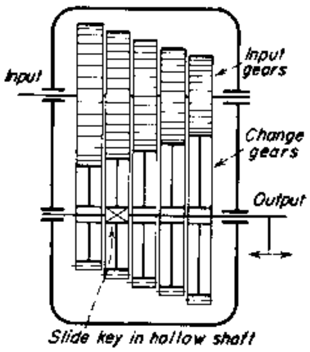


Fig. 6 This is a combination coupling and slide gears. It has three ratios: a direct mesh for ratios I and II; a third ratio is transmitted through gears II and III, which couple together.

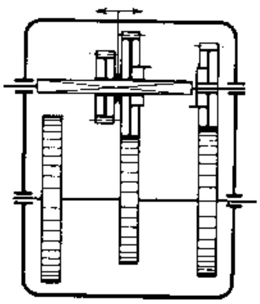
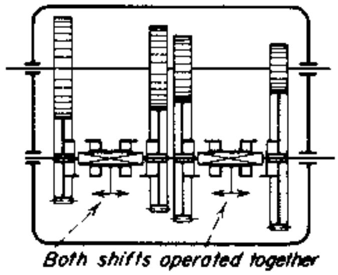


Fig. 7 Double-shift drive. One shift must always be in a neutral position. That might require both levers to be shifted when making a change. However, only two shafts are used to achieve four ratios.



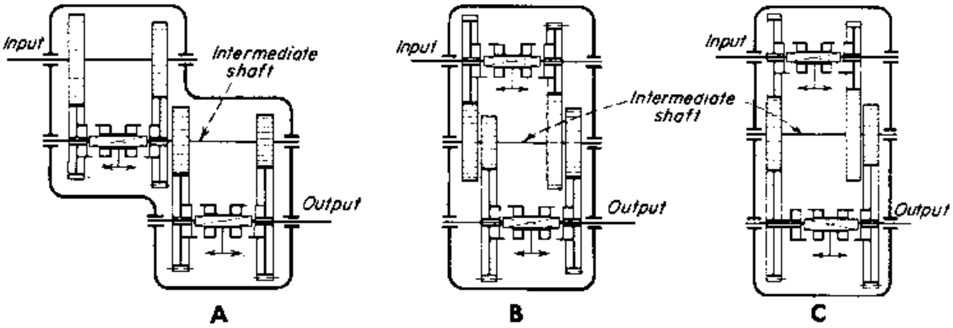


Fig. 8 A triple shaft drive gives four ratios. (A) The output of the first drive serves as the input for the second. The presence of an intermediate shaft eliminates the requirement for ensuring that one shift is always in the neutral position. A wrong shift-lever position cannot cause damage. (B) A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

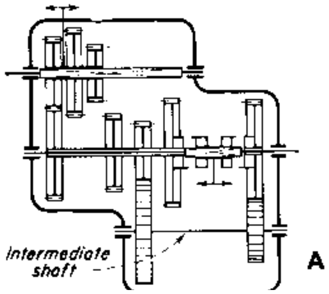
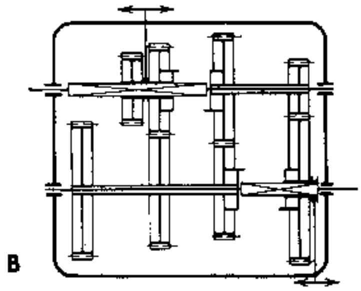


Fig. 9 Six ratios are available with two couplings and (A) ten gears, (B) eight gears. Up to six gears can be in permanent mesh. It is not necessary to ensure that one shift is in neutral.



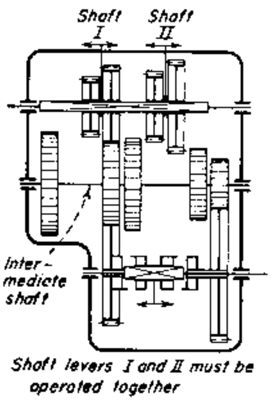


Fig. 10 This eight-ratio drive has two slide gears and a coupling. This arrangement reduces the number of parts and meshes. The position of shifts I and II are interdependent. One shift must be in neutral if the other *is* in mesh.

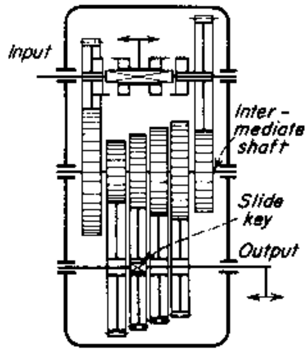
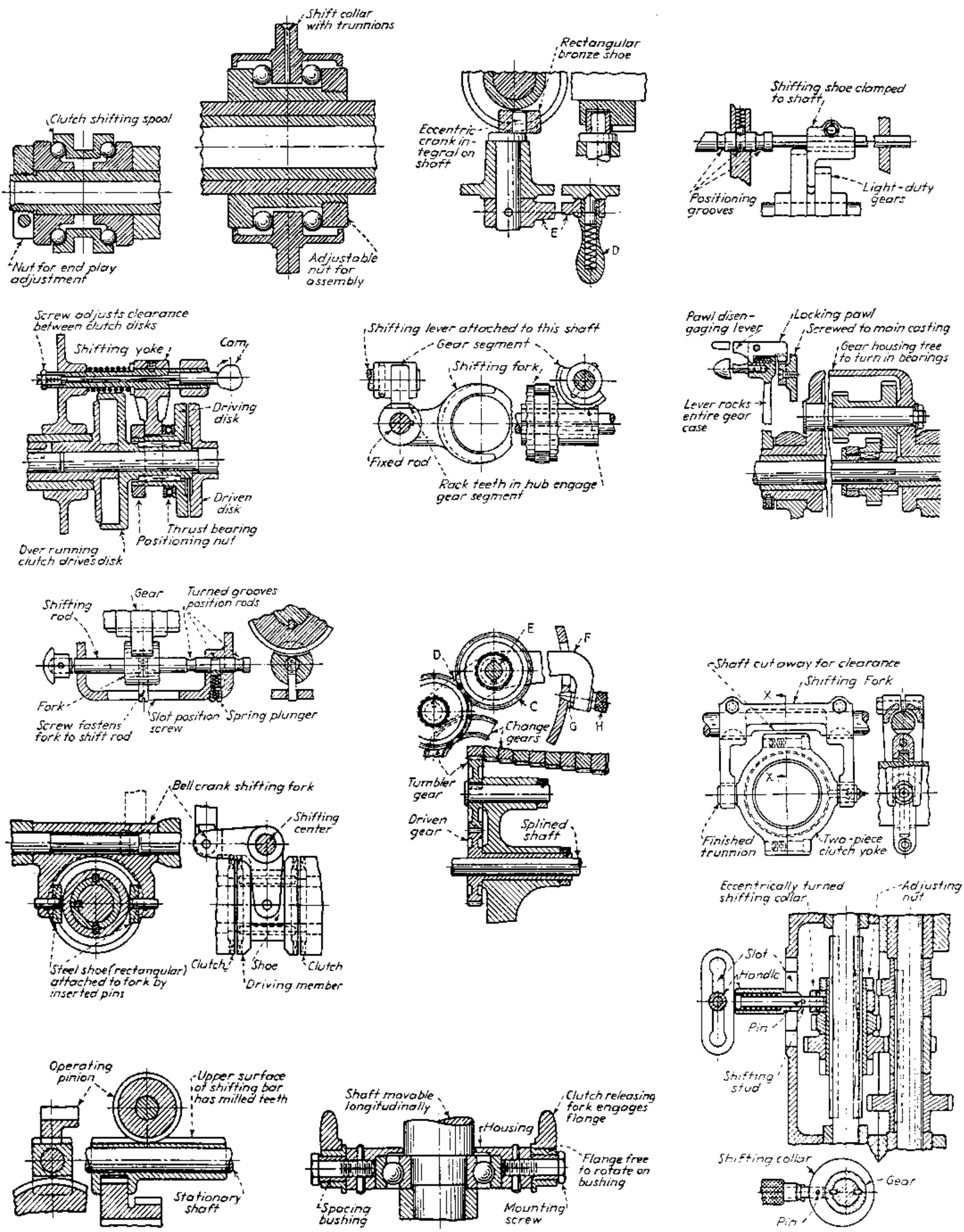
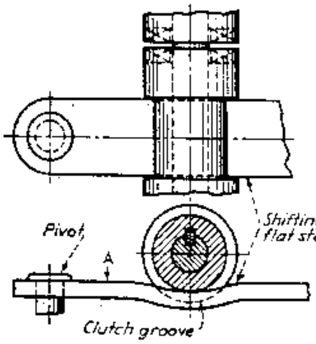
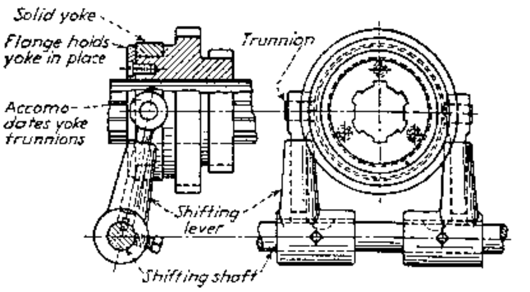


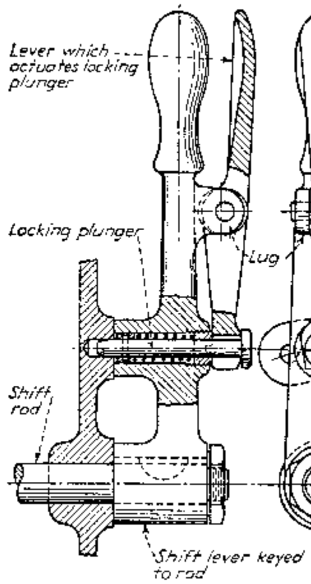
Fig. 11 This drive has eight ratios; a coupled gear drive and slide-key drive are in series. Comparatively low strength of the slide key limits the drive to small torque.

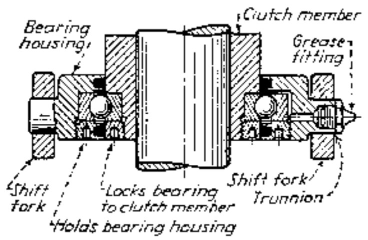
SHIFTING MECHANISMS FOR GEARS AND CLUTCHES

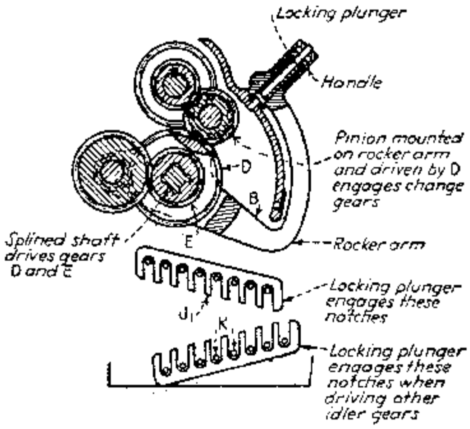












Spiderless differential

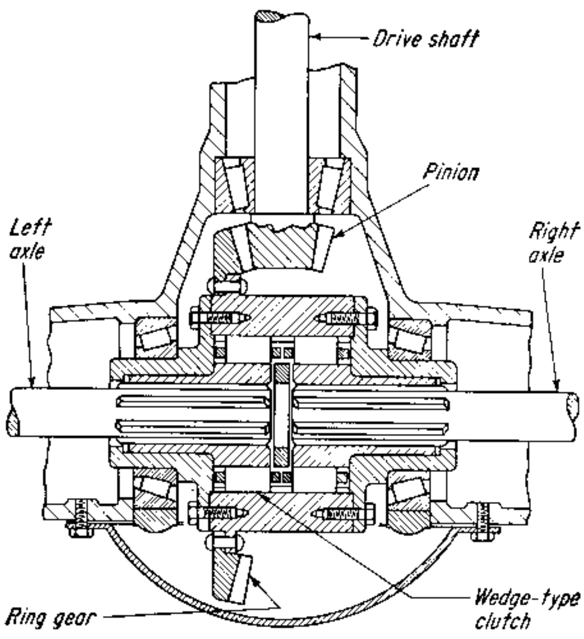
If you've ever been unable to drive your car out of a ditch because one wheel spun uselessly while the other sat torqueless and immobile, you'll thank the inventors (Seliger and Hegar) of the limited-slip differential shown here.

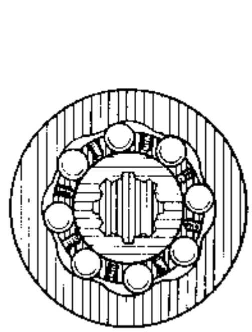
In straight running, it performs as a drive axle, driven by the driveshaft pinion through the ring gear. The differential action occurs only when one wheel loses traction, travels along a different arc, or otherwise attempts to turn at a speed that is different from that of the other. Then the wedge-type,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

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

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

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





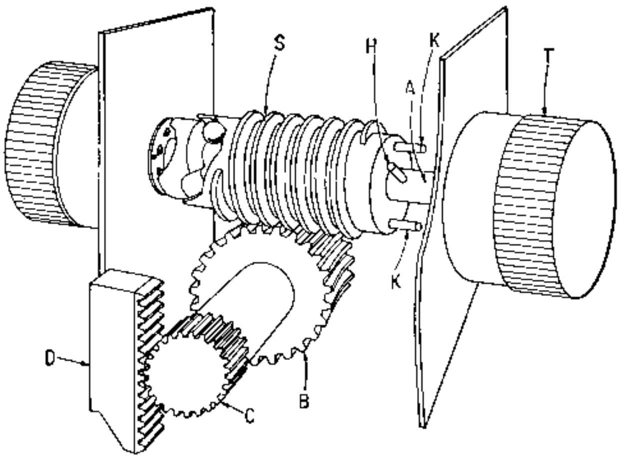
Two-way over-running clutch disengages the non-driving wheel

FINE-FOCUS ADJUSTMENTS

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

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

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



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

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

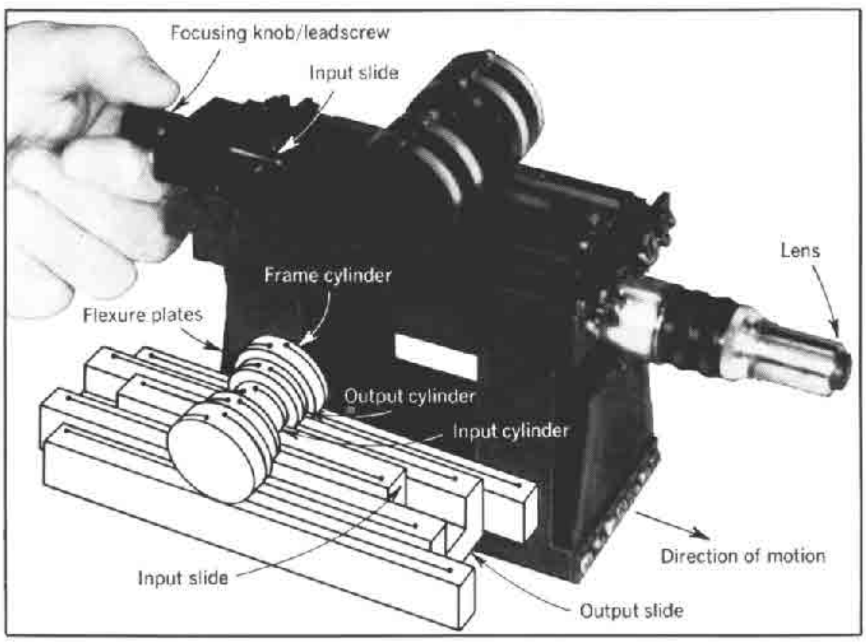
Three cylinders. IBM's mechanism consists of three interconnected, different-sized cylinders to provide reduced displacement. A small input cylinder and the focusing knob are attached to a lead screw.

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

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

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

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



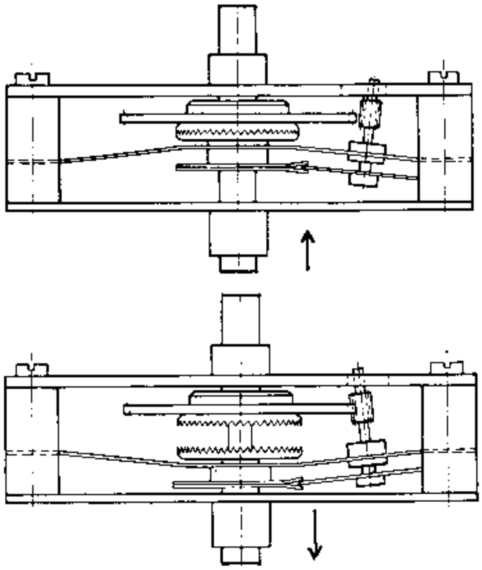
A linear micron-positioner, based on series of connected cylinders of unequal diameters, can adjust a camera lens to within 10 millionths of an inch.

RATCHET-TOOTH SPEED-CHANGE DRIVE

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

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

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



TWINWORM GEAR DRIVE

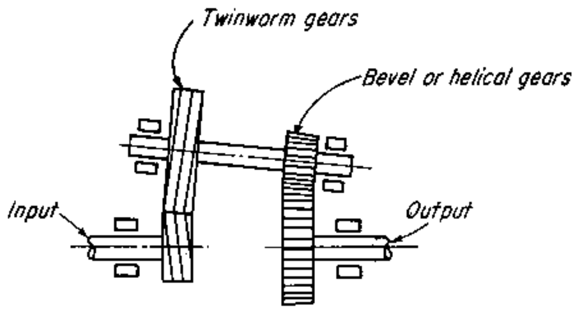
The term “self-locking” as applied to gear systems denotes a drive that gives the input gear freedom to rotate the output gear in either direction. But the output gear locks with the input when an outside torque attempts to rotate the output in either direction. This characteristic is often sought by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

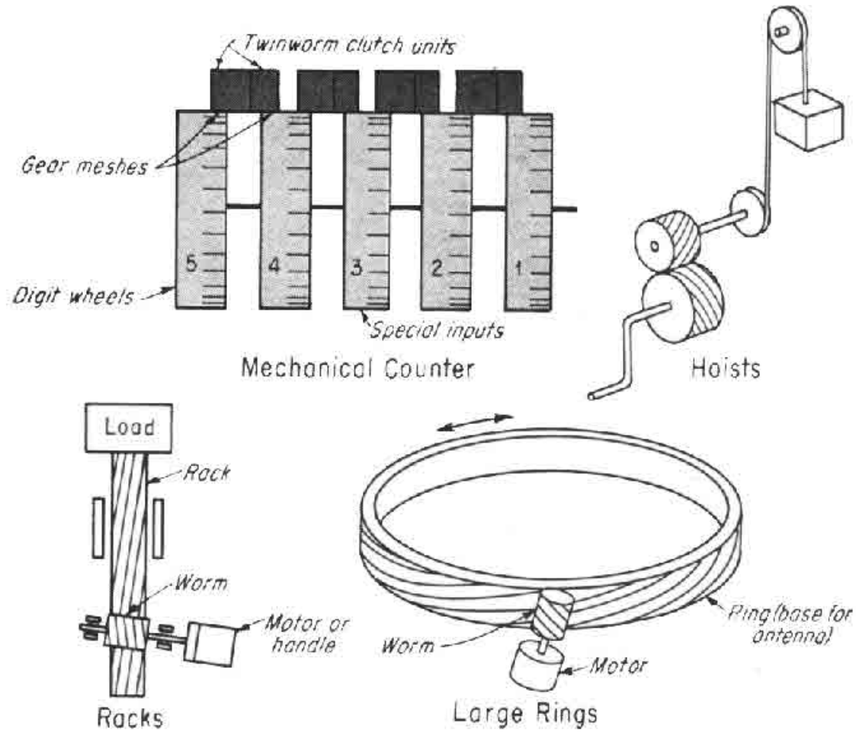
An Israeli engineer, B. Popper, invented a simple dual-worm gear system that not only provided self-locking with over 90% efficiency, but exhibited a phenomenon which the inventor calls “deceleration-locking.”

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

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

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





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

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

JOBS FOR THE NEW DRIVE

Applications for Twinworm can be divided into two groups:

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

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

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

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

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

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

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

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

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

$$\mu_{modified} = \frac{\mu_{true}}{\cos\theta}$$

A relationship between the input and output forces during rotation is:

$$\frac{P_1}{P_2} = \frac{\sin\phi_1 + \mu\cos\phi_1}{\sin\phi_2 + \mu\cos\phi_2}$$

Efficiency is determined from the equation:

$$\eta = \frac{1 + \mu/\tan\phi_2}{1 + \mu/\tan\phi_1}$$

COMPLIANT GEARING FOR REDUNDANT TORQUE DRIVE

Elastomeric bearings make torque loads more nearly equal.
Lewis Research Center, Cleveland, Ohio

A set of elastomeric bearings constitutes a springy coupling between a spur gear and a drive shaft. The gear, bearings, and shaft are parts of a split-drive (redundant) mechanical transmission, and the compliance of the coupling helps to distribute torque nearly equally along the load paths of the split drive. Compliance is necessary because without it, even slight deviations in the dimensions of the redundant gears can cause grossly unequal sharing of loads. Indeed, in the absence of compliant coupling, the gears along one load path can assume the entire load while those along another load path can freewheel. Thus, the advantage of reduced loads on gear teeth is lost.

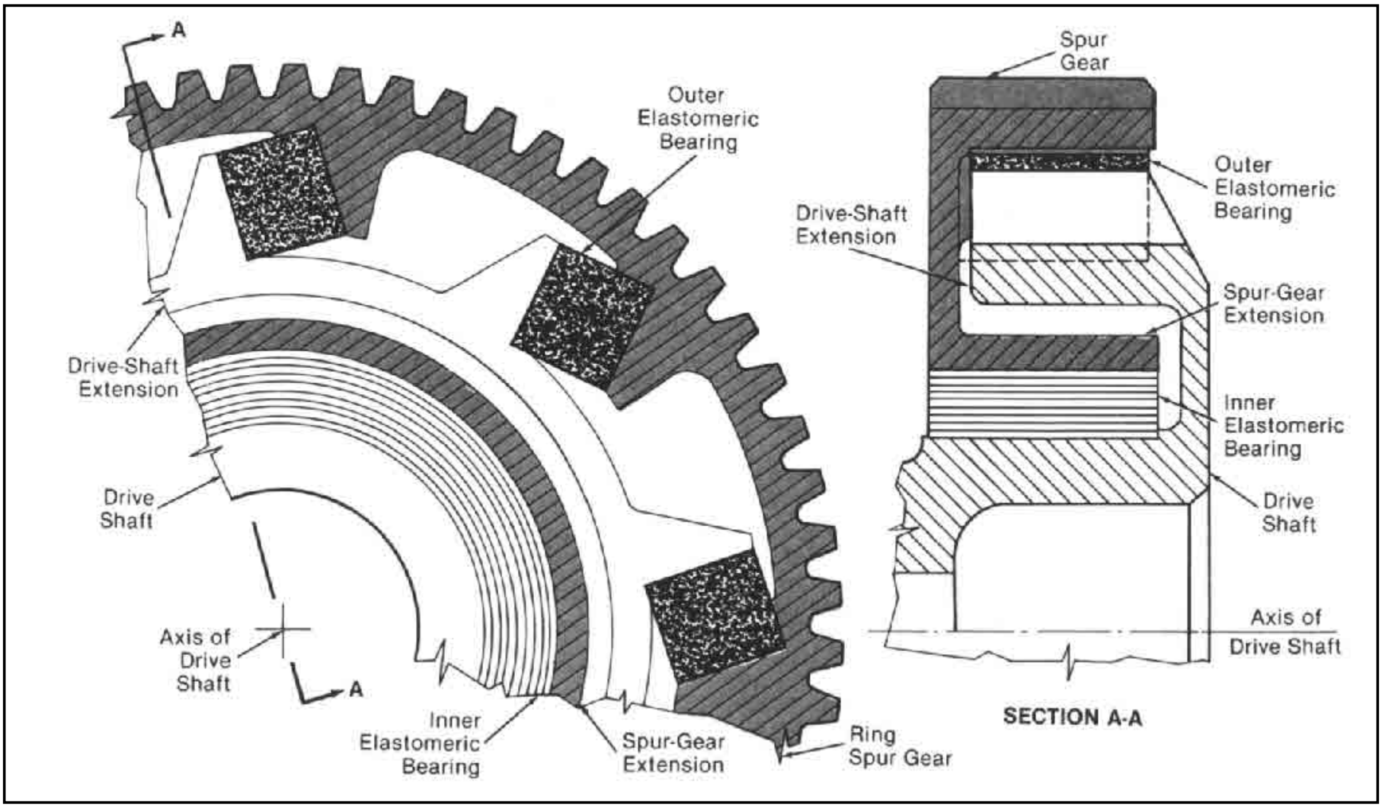
The figure illustrates one version of the shaft/bearing/gear assembly. An inner, concentric elastomeric bearing lies between a central drive shaft and an extension of a ring spur gear. A set of padlike outer elastomeric bearings joins outward protrusions on an extension of the drive shaft with facing inward protrusions on the ring spur gear.

The inner elastomeric bearing has high radial stiffness and low circumferential stiffness. This bearing centers the ring spur gear on the axis of the drive shaft and provides compliance in a circumferential direction. In a representative design of a redundant helicopter transmission, it should be at least 0.5 in. (1.27 cm) thick so that it transmits little torque.

The outer elastomeric bearings, in contrast, have low radial stiffness and high circumferential stiffness. They thus transmit torque effectively between facing protrusions. Nevertheless, they are sufficiently compliant circumferentially to accommodate the desired amount of circumferential displacement [up to 1⁄16 in. (1.6 mm) in the helicopter transmission application].

The process of assembling the compliant gearing begins with the pressing of the inner elastomeric bearing onto the drive shaft. Then, with the help of an alignment tool, the ring spur gear is pressed onto the inner elastomeric bearing. The outer elastomeric bearings are ground to fit the spaces between the protrusions and bonded in place on the protrusions. As an alternative to bonding, the entire assembly can be potted in a soft matrix that holds the outer bearings in place but allows rotation with little restraint.

This work was done by C. Isabelle and J. Kish of United Technologies Corp. for **Lewis Research Center.**



Elastomeric Bearings couple a drive shaft with a ring spur gear. The inner elastomeric bearing is radially stiff and circumferentially compliant, while the outer elastomeric bearings are circumferentially stiff and radially compliant. The combination accommodates minor variations in the dimensions and placements of gears, shafts, and other components.

LIGHTER, MORE-EFFICIENT HELICOPTER TRANSMISSIONS

Redundant gearing transmits torque through an angle or angles.
Lewis Research Center, Cleveland, Ohio

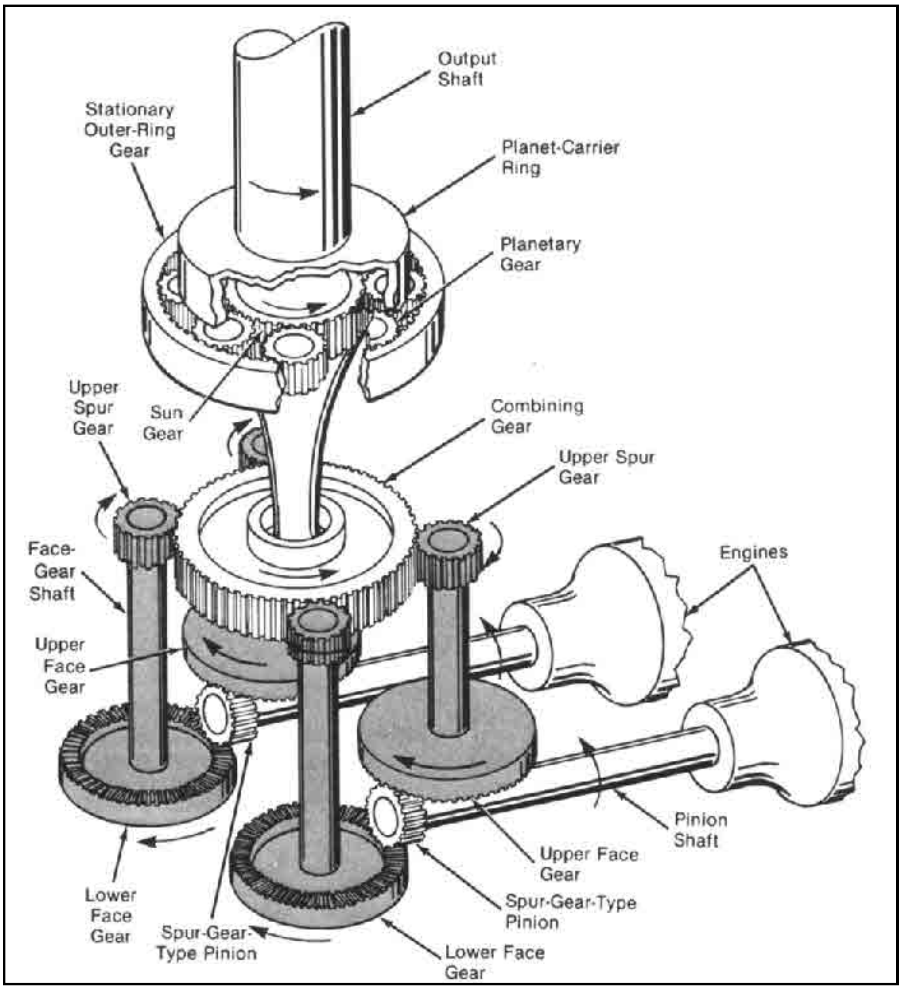
An improved gear system intended primarily for use in a helicopter transmits torque from the horizontal or nearly horizontal shafts of two engines to the vertical output shaft that supports the rotor. The system apportions torques equally along multiple, redundant drive paths, thereby reducing the stresses on individual gear teeth, and it enables one engine to continue to turn the rotor when the other engine fails. The underlying design concept could also be applied to couple two airplane engines to a set of propellers in such a way that both propellers turn as long as at least one engine operates.

The system exploits the special advantages of the geometry of the meshing of a spur-gear-type pinion with a face gear. In comparison with other gear geometries that have been used in helicopter transmissions, this one is much more forgiving of (1) errors in manufacturing and alignment and (2) thermal and vibrational changes in the sizes and positions of the meshing components. One of the benefits is a reduction of gear-tooth-contact noise and vibration. Another benefit is the possibility of achieving a high (> 4) speed-reduction ratio in a single, efficient mesh, and the consequent possibility of reducing the number of parts, the size, the cost, and the weight of the gear system. Of course, the reduction of the number of parts confers yet another benefit by increasing the reliability of the system.

The system is shown schematically in the figure. The output of each engine is coupled by a pinion shaft to a spur-gear-type pinion. Each pinion engages an upper and a lower face gear, and each face gear is coupled by a face-gear shaft to an upper spur gear. The upper spur gears feed torque into a large combining gear. The pinion end of each pinion shaft is lightly spring-loaded in a nominal lateral position and is free to shift laterally through a small distance to take up slack, compensate for misalignments, and apportion torques equally to the two face gears with which it is engaged.

The combining gear is splined to a shaft that flares outwardly to a sun gear. The sun gear operates in conjunction with planetary gears and a stationary outer ring gear. The torque is coupled from the sun gear through the planetary gears to the planet-carrier ring, which is mounted on the output shaft.

This work was done by Robert B. Bossler, Jr., of Lucas Western, Inc., for **Lewis Research Center.**



Torque from each engine is split and transmitted to the combining gear along two redundant paths. Should one engine fail, the other engine could still turn the output shaft.

WORM GEAR WITH HYDROSTATIC ENGAGEMENT

Friction would be reduced greatly.
Lewis Research Center, Cleveland, Ohio

In a proposed worm-gear transmission, oil would be pumped at high pressure through the meshes between the teeth of the gear and the worm coil (see Figure 1). The pressure in the oil would separate the meshing surfaces slightly, and the oil would reduce the friction between these surfaces. Each of the separating forces in the several meshes would contribute to the torque on the gear and to an axial force on the worm. To counteract this axial force and to reduce the friction that it would otherwise cause, oil would also be pumped under pressure into a counterforce hydrostatic bearing at one end of the worm shaft.

This type of worm-gear transmission was conceived for use in the drive train between the gas-turbine engine and the rotor of a helicopter and might be useful in other applications in which weight is critical. Worm gear is attractive for such weight-critical applications because (1) it can transmit torque from a horizontal engine (or other input) shaft to a vertical rotor (or other perpendicular output) shaft, reducing the speed by the desired ratio in one stage, and (2) in principle, a one-stage design can be implemented in a gearbox that weighs less than does a conventional helicopter gearbox.

Heretofore, the high sliding friction between the worm coils and the gear teeth of worm-gear transmissions has reduced efficiency so much that such transmissions could not be used in helicopters. The efficiency of the proposed worm-gear transmission with hydrostatic engagement would depend partly on the remaining friction in the hydrostatic meshes and on the power required to pump the oil. Preliminary calculations show that the efficiency of the proposed transmission could be the same as that of a conventional helicopter gear train.

Figure 2 shows an apparatus that is being used to gather experimental data pertaining to the efficiency of a worm gear with hydrostatic engagement. Two stationary disk sectors with oil pockets represent the gear teeth and are installed in a caliper frame. A disk that represents the worm coil is placed between the disk sectors in the caliper and is rotated rapidly by a motor and gearbox. Oil is pumped at high pressure through the clearances between the rotating disk and the stationary disk sectors. The apparatus is instrumented to measure the frictional force of meshing and the load force.

The stationary disk sectors can be installed with various clearances and at various angles to the rotating disk. The stationary disk sectors can be made in various shapes and with oil pockets at various positions. A flowmeter and pressure gauge will measure the pump power. Oils of various viscosities can be used. The results of the tests are expected to show the experimental dependences of the efficiency of transmission on these factors.

It has been estimated that future research and development will make it possible to make worm-gear helicopter transmission that weigh half as much as conventional helicopter transmissions do. In addition, the new hydrostatic meshes would offer longer service life and less noise. It might even be possible to make the meshing worms and gears, or at least parts of them, out of such lightweight materials as titanium, aluminum, and composites.

This work was done by Lev. I. Chalko of the U.S. Army Propulsion Directorate (AVSCOM) for **Lewis Research Center.**

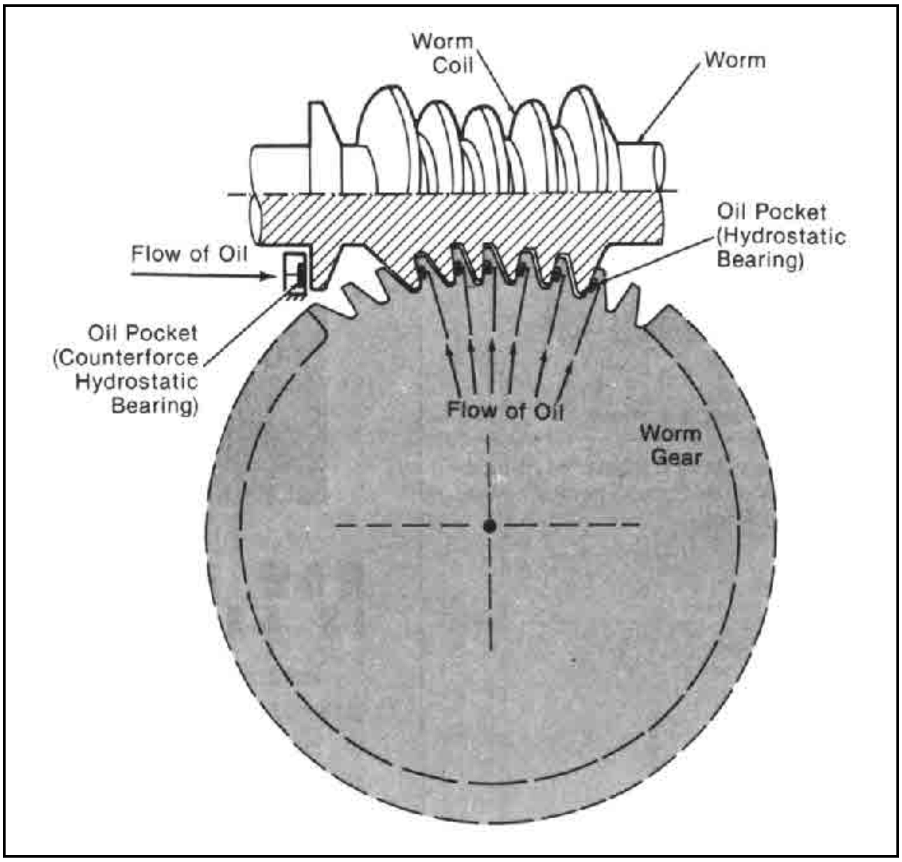


Fig. 1 Oil would be injected at high pressure to reduce friction in critical areas of contact.

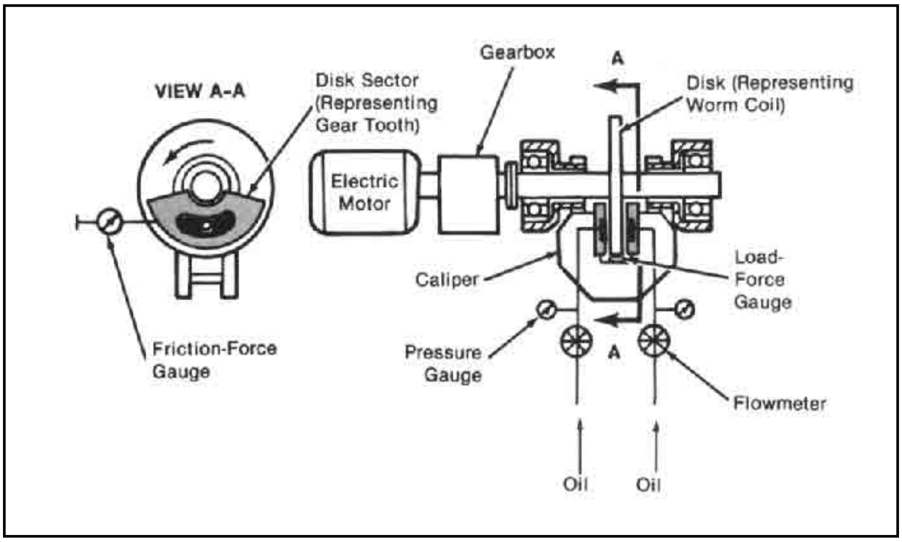


Fig. 2 This test apparatus simulates and measures some of the loading conditions of the proposed worm gear with hydrostatic engagement. The test data will be used to design efficient worm-gear transmissions.

STRADDLE DESIGN OF SPIRAL BEVEL AND HYPOID GEARS

Lengths and radii of shafts can be chosen to prevent undercutting.
Lewis Research Center, Cleveland, Ohio

A computer-assisted method of analysis of straddle designs for spiral bevel and hypoid gears helps to prevent undercutting of gear shafts during cutting of the gear teeth. Figure 1 illustrates a spiral bevel gear or straddle design, in which the shaft extends from both ends of the toothed surface to provide double bearing support. One major problem in such a design is to choose the length and radius of the shaft at the narrow end (equivalently, the radial coordinate r and axial coordinate u) such that the head cutter that generates the gear teeth does not collide with, and thereby undercut, the shaft.

The analytical method and computer program are based on the equations for the surface traced out by the motion of the head cutter, the equation for the cylindrical surface of the shaft, and the equations that express the relationships among the coordinate systems fixed to the various components of the gear-cutting machine tool and to the gear. The location of a collision between the shaft and the cutter is defined as the vector that simultaneously satisfies the equations for head-cutter-traced and shaft surfaces. The solution of these equations yields the u and r coordinates of the point of collision.

Given input parameters in the form of the basic machine-tool settings for cutting the gear, the computer program finds numerical values of r and u at a representative large number of points along the path of the cutter. These computations yield a family of closed curves (see Fig. 2) that are the loci of collision points. The region below the curves is free of collisions: thus, it contains the values of r and u that can be chosen by the designer to avoid collisions between the shaft and the head cutter.

This work was done by Robert F. Handschuh of the U.S. Army Aviation Systems Command; Faydor L. Litvin, Chihping Kuan, and Jonathan Kieffer of the University of Illinois at Chicago; and Robert Bossler of Lucas Western, Inc., for **Lewis Research Center.**

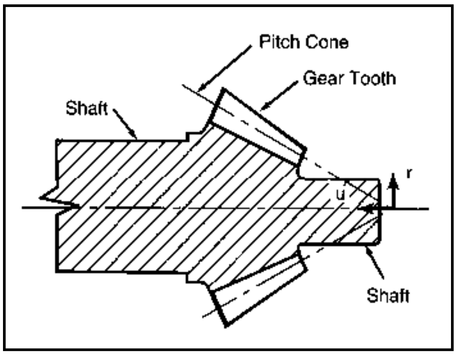


Fig. 1 A straddle-design spiral bevel gear includes two integral shaft extensions. One of these could terminate near or even beyond the apex of the pitch cone.

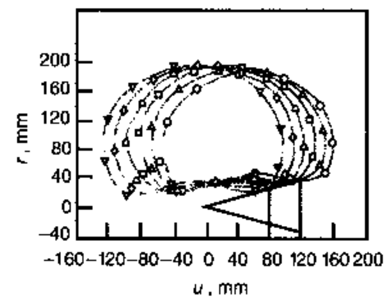


Fig. 2 This family of closed curves applies to a typical hypoid gear. It will help in the selection of the length and radius of the shaft at the narrow end. The region below the curves is free of collisions between the head cutter and the shaft.

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.

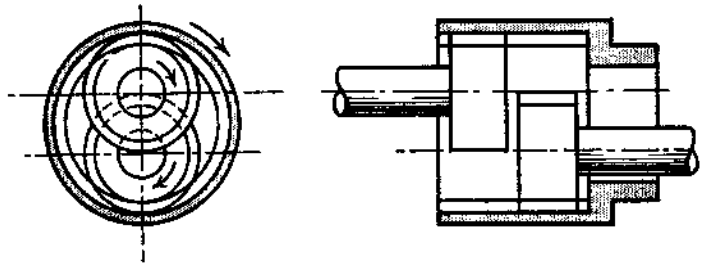


Fig. 2 This coupling consists of two universal joints and a short shaft. Velocity transmission is constant between the input and output shafts if the shafts remain parallel and if the end yokes are arranged symmetrically. The velocity of the central shaft fluctuates during rotation, but high speed and wide angles can cause vibration. The shaft offset can be varied, but axial freedom requires that one shaft be spline mounted.

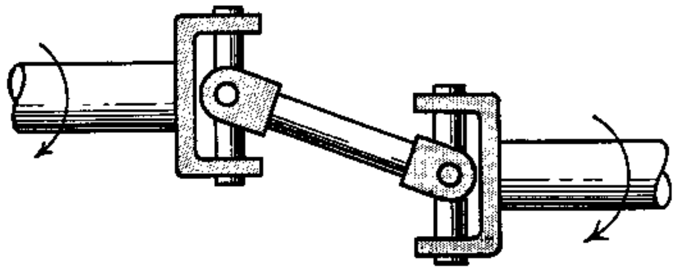


Fig. 3 This crossed-axis yoke coupling is a variation of the mechanism shown in Fig. 2. Each shaft has a yoke connected so that it can slide along the arms of a rigid cross member. Transmission is at a constant velocity, but the shafts must remain parallel, although the offset can vary. There is no axial freedom. The central cross member describes a circle and is thus subjected to centrifugal loads.

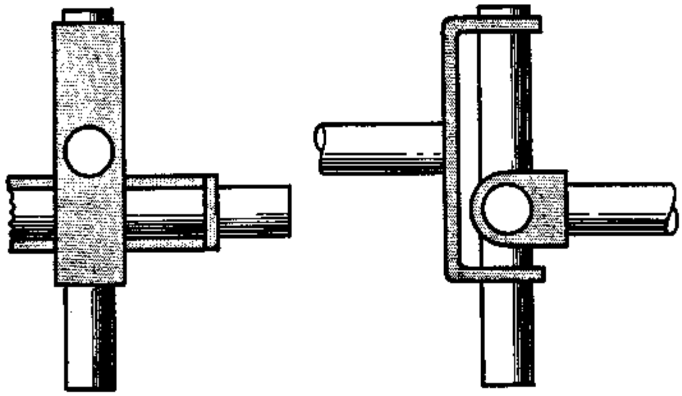
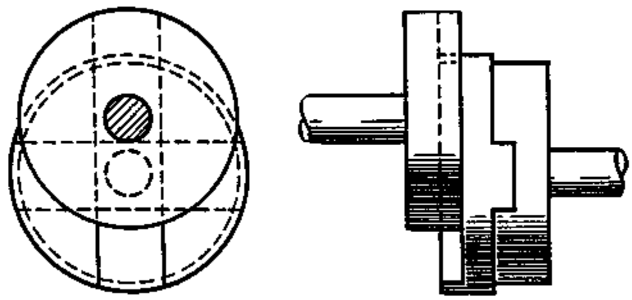


Fig. 4 This Oldham coupling provides motion at a constant velocity as its central member describes a circle. The shaft offset can vary, but the shafts must remain parallel. A small amount of axial freedom is possible. A tilt in the central member can occur because of the offset of the slots. This can be eliminated by enlarging its diameter and milling the slots in the same transverse plane.



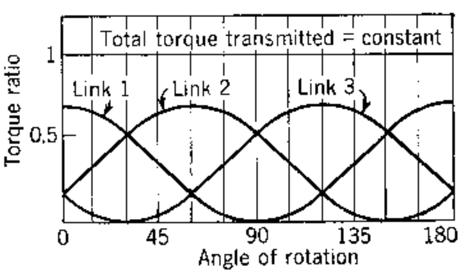
NOVEL LINKAGE COUPLES OFFSET SHAFTS

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.

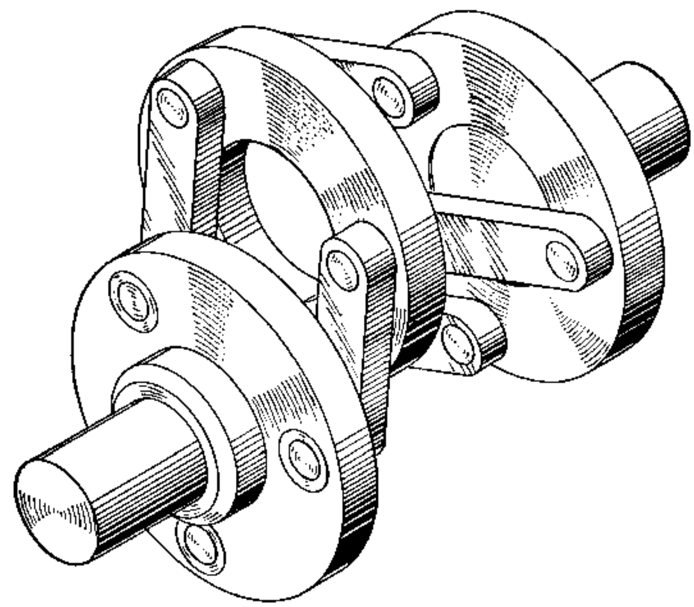
Changes in radial displacement do not affect the constant-velocity relationship between the input and output shafts, nor do they affect initial radial reaction forces that might cause imbalance in the system. Those features open up unusual applications for it in automotive, marine, machine-tool, and rolling-mill machinery (see drawings).

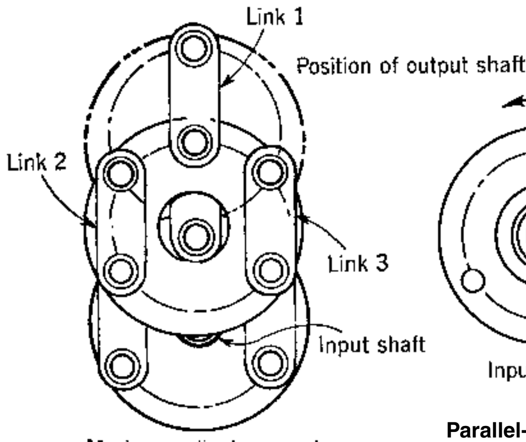
How it works. The inventor of the coupling, Richard Schmidt of Madison, Alabama, said that a similar link arrangement had been known to some German engineers for years. But those engineers were discouraged from applying the theory because they erroneously assumed that the center disk had to be retained 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.

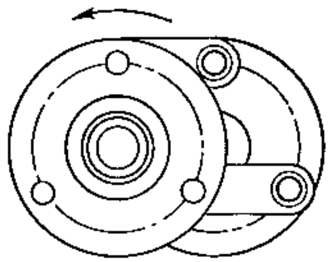


Torque transmitted by three links in the group adds up to a constant value, regardless of the angle of rotation.



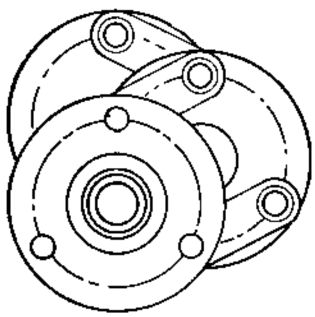


Maximum displacement
(rear view)

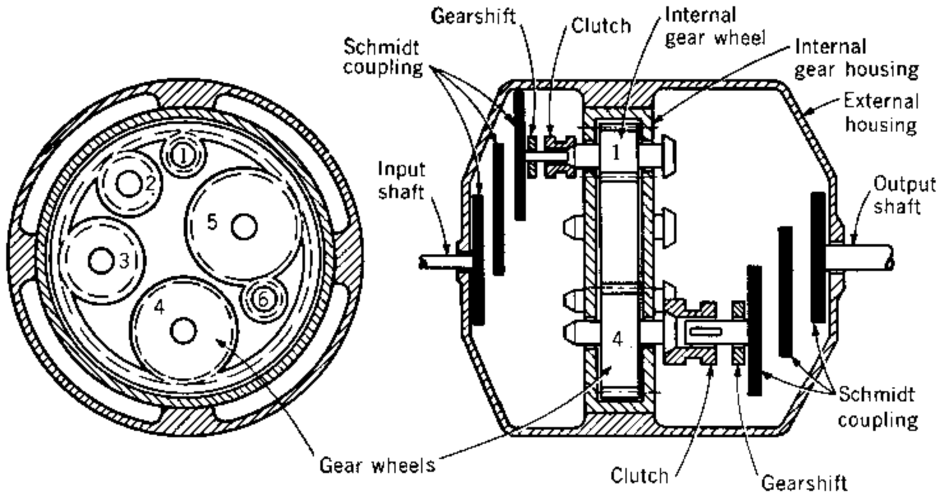


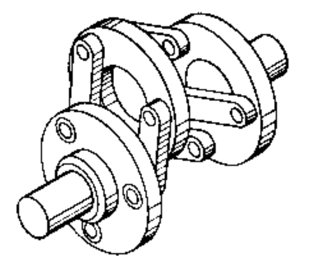
Input and output shafts in line
Zero displacement

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





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.

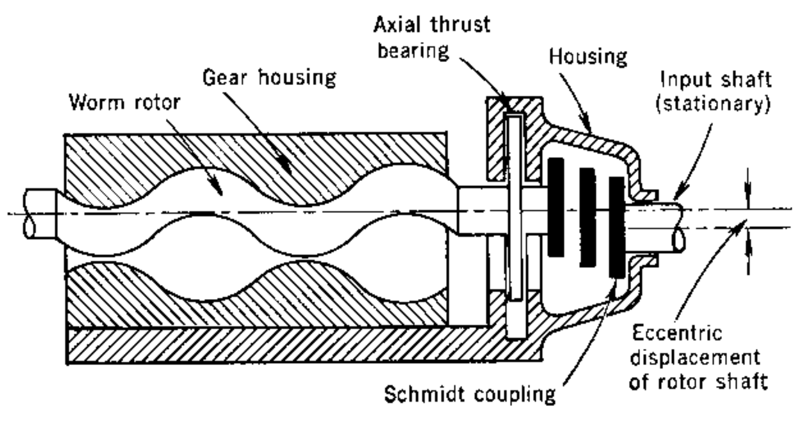
The coupling (drawing, upper right) maintains a constant transmission ratio between input and output shafts while the shafts undergo axial shifts in their relative positions. This permits gear-and-belt transmissions to be designed that need fewer gears and pulleys.

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.

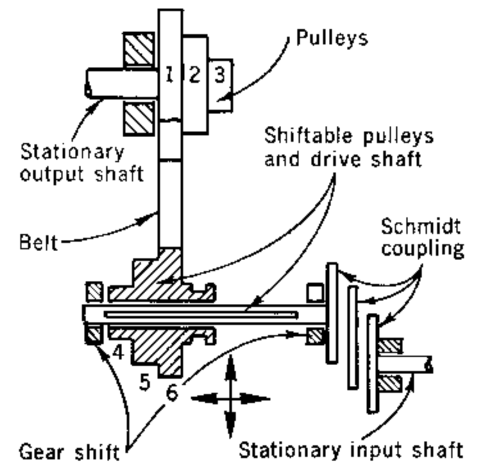
On the output side, after the power flows through the gear wheel, a second Schmidt coupling permits a direct power takeoff from any of the same six gears. Thus, any one of 6×6 minus 5 or 31 different speed ratios can be selected while the unit is running. A more orthodox design would require almost twice 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.

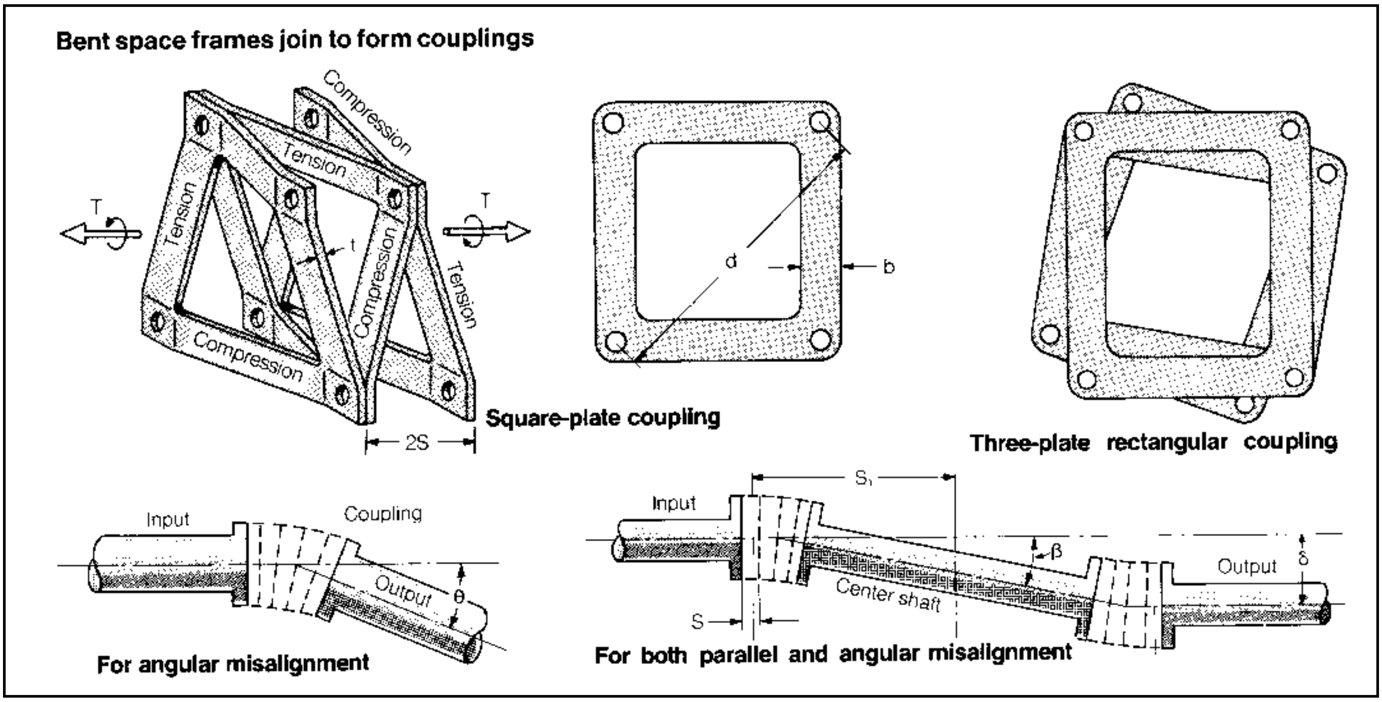
Although the analysis and resulting equations developed by Bossler are based on a square-shaped unit, he concluded that the perfect square is not the ideal for the coupling, because of the position of the mounting holes. The flatter the helix—in other words the smaller the distance *S*—the more misalignment 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) $T = 11.62 \dfrac{Ebt^3}{dn^{0.9}}$

Maximum stress per degree of misalignment.

(2) $\sigma_{max} = 0.0276\ Et/L$

Minimum thickness to meet required torque strength

(3) $t = 0.4415 \left(\dfrac{dT}{bE}\right)^{1/3} n^{0.3}$

Weight of coupling with minimum-thickness plates

(4) $W = 1.249w \left(\dfrac{T}{E}\right)^{1/3} d^{4/3}\, b^{2/3}\, n^{1.3}$

Maximum permissible misalignment

(5) $\theta_{max} = 54.7 \left[\dfrac{bd^2}{TE^2}\right]^{1/3} \sigma_c\, n^{0.7}$

Maximum permissible misalignment (simplified)

(6) $\theta/d = 10.9 \dfrac{n^{0.7}}{T^{1/3}}$

Maximum permissible offset-angle

(7) $\beta = 54.7 \left[\dfrac{bd^2}{TE^2}\right]^{1/3} \dfrac{\sigma_e C}{n^{0.3}}$

where: $\displaystyle\sum_{x=1}^{x=n} \left[1 - (x-1)\frac{S}{S_l}\right]^2$

Maximum permissible offset-angle (simplified)

(8) $\beta/d = \dfrac{10.9\ C}{T^{1/3}\, n^{0.3}}$

Critical speed frequency

(9) $f = \dfrac{60}{2\pi}\left(\dfrac{k}{M}\right)^{1/2}$

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 1º 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

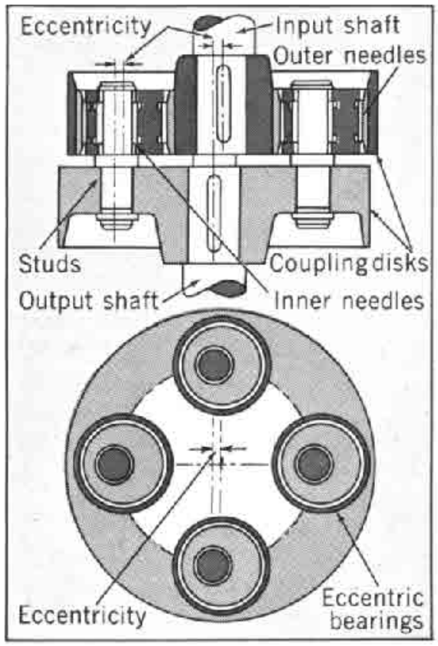
Two Hungarian engineers developed an all-metal coupling (see drawing) for connecting shafts where alignment is not exact—that is, where the degree of misalignment does not exceed the magnitude of the shaft radius.

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.

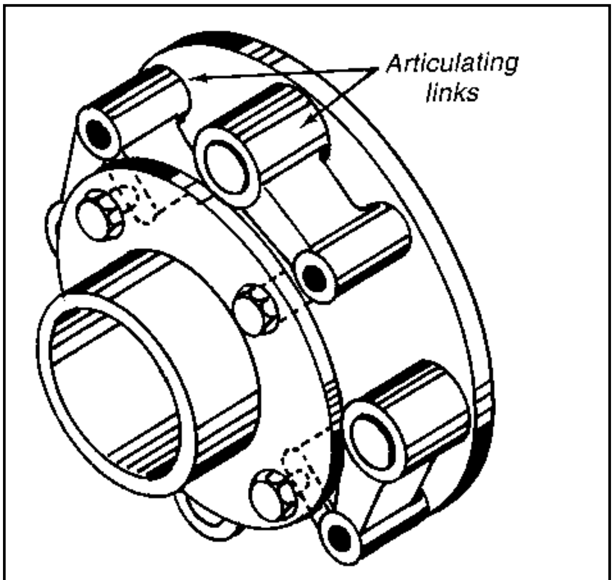
When torque is applied to the coupling, the linkages deflect in a positive or negative direction from the neutral position (drawings, below). Deflection is opposed by the torsional resistance of the rubber bushings at the outer pins. When the coupling is rotating, the masses of the linkage give rise 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.

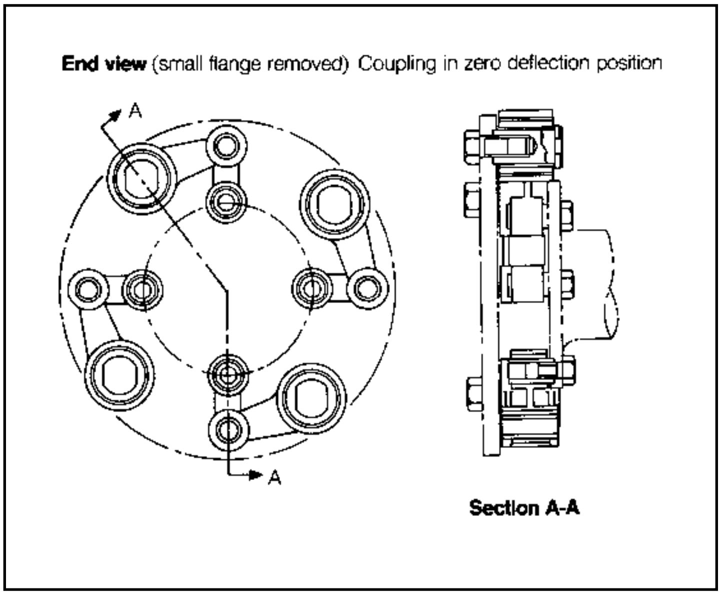
Tests of the coupling's torque/deflection characteristics under load have shown that the torsional stiffness of the coupling increases progressively with speed and with torque when deflected in the positive direction. Although the geometry of the coupling is asymmetrical the torsional characteristics 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.

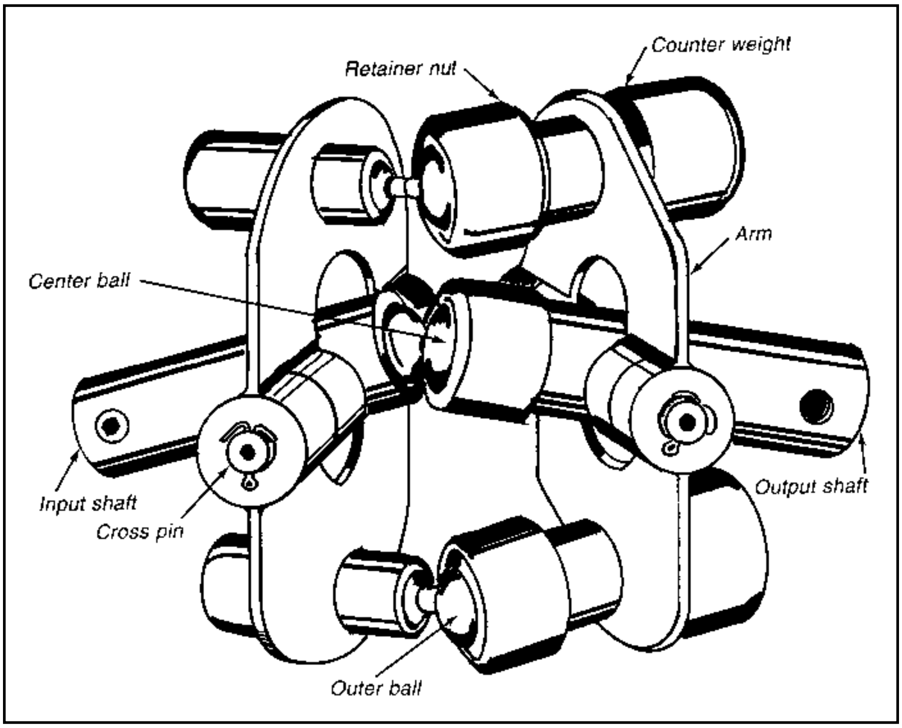
Using the example of the TL 2307G coupling design—which is suitable for 10,000 hp at 525 rpm—the torsional stiffness at working points is largely determined by coupling geometry and is, therefore, affected to a minor extent by the variations in the properties of the rubber bushings. Moreover, the coupling 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.

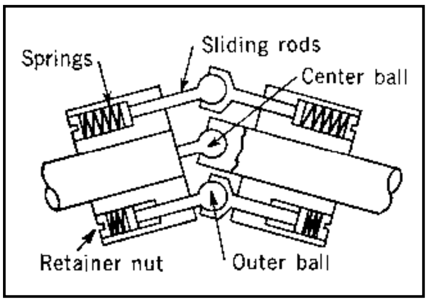
Balancing for rotation is achieved by equalizing the weight of the two driving arms of each half of the joint. Balancing the acceleration forces due to the oscillation of the ball-and-socket couplings, which are offset from their swiveling axes, is achieved by the use of counterweights extending from 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.

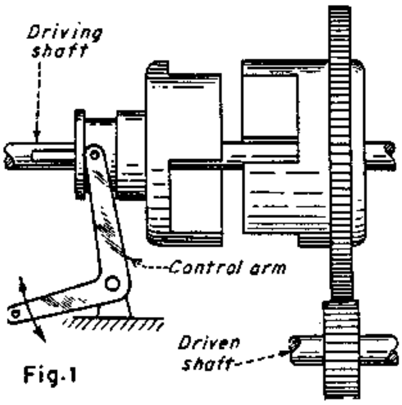


Fig. 1

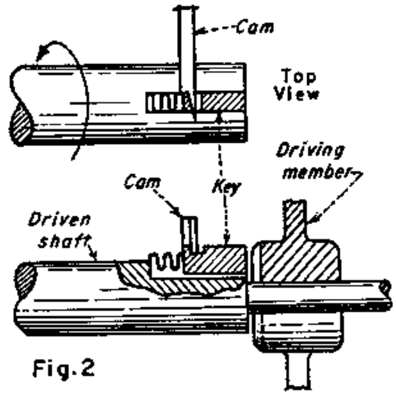


Fig. 2

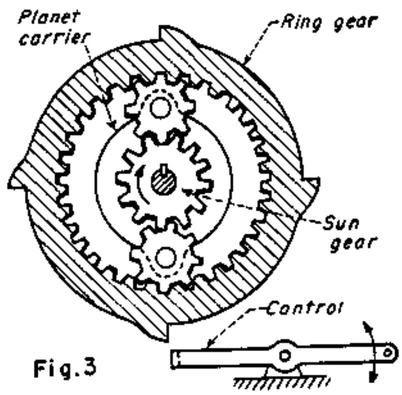


Fig. 3

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.

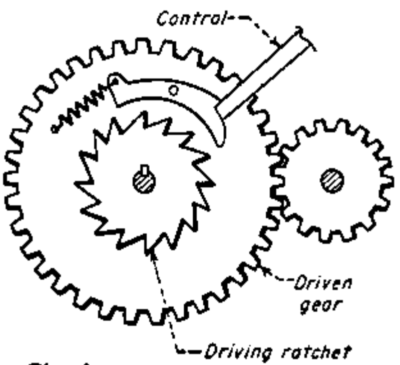


Fig. 4

Fig. 5

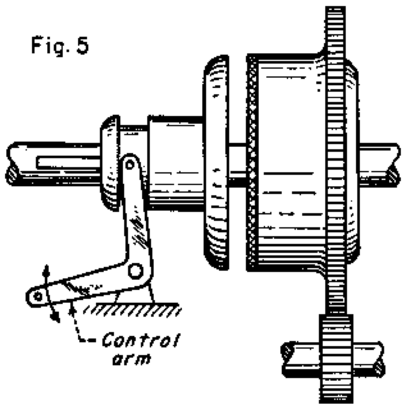


Fig. 6

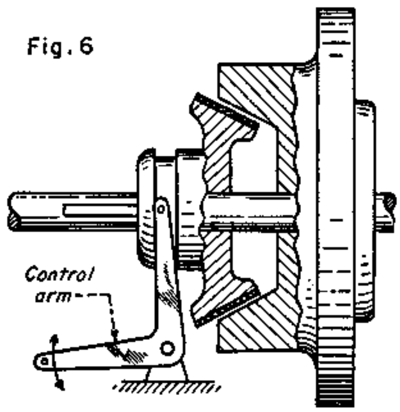


Fig. 4 Pawl and Ratchet Clutch: (External Control) The driving ratchet of this clutch is keyed to the driving shaft, and the pawl is pinned to the driven gear which can rotate freely on the driving shaft. When the control arm is raised, the spring pulls in the pawl to engage the ratchet and drive the 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.

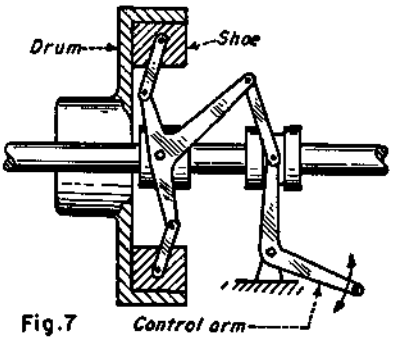


Fig.7

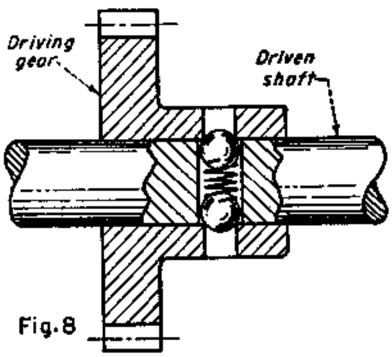


Fig.8

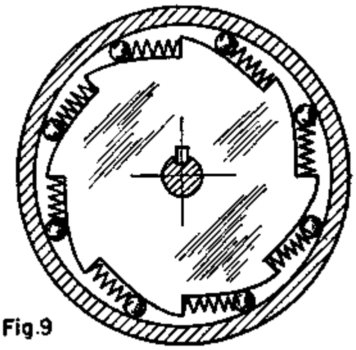


Fig.9

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.

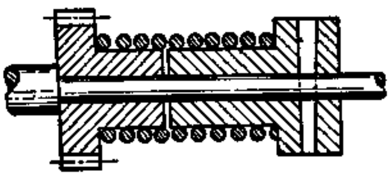


Fig.10

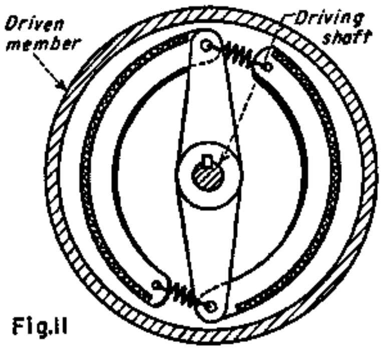


Fig.11

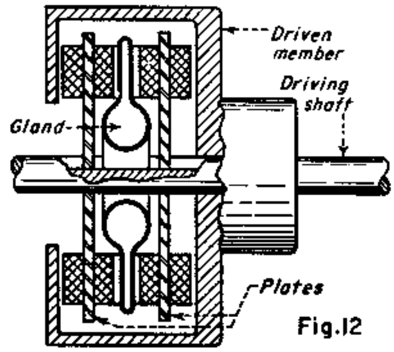


Fig.12

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

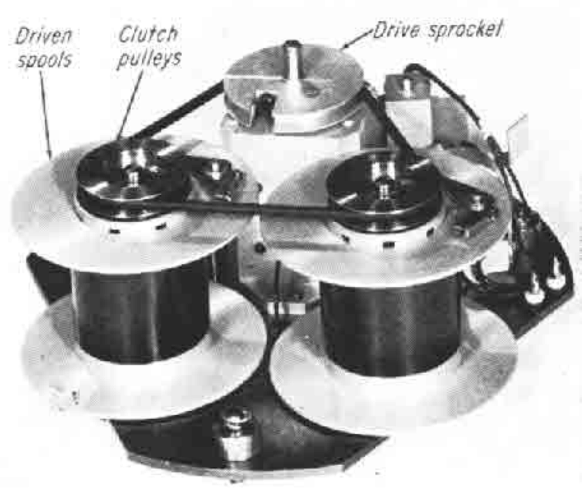


Fig. 1 Two dual-spring clutches are in this tape drive.

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

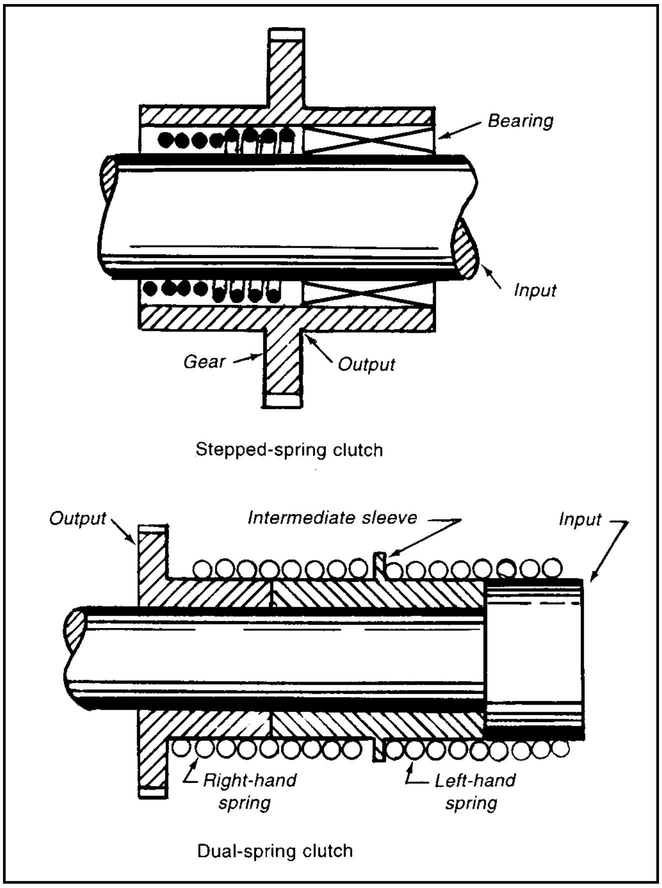
Spring-wrapped slip clutches and brakes have remarkably repeatable slip-torque characteristics which do not change with service temperature. Torque capacity remains constant with or without lubrication, and is unaffected by variations in the coefficient of friction. Thus, break-away torque capacity is 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

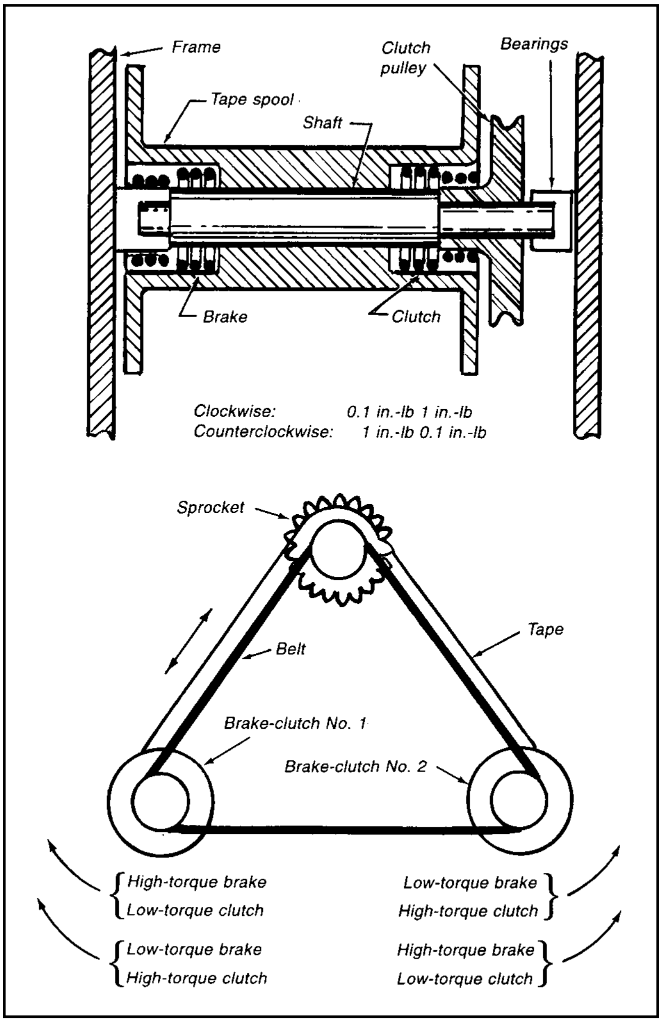
An interesting example of how slip brakes and clutches worked together to maintain proper tension in a tape drive, in either direction of operation, is pictured above and shown schematically on the opposite page. A brake here is simply a slip clutch with one side fastened to the frame of the unit. Stepped-spring 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:

$$d = \sqrt[4]{\frac{32D^2T}{\pi E\delta}} = \sqrt[4]{\frac{(32)(0.750)^2(0.8)}{\pi(30\times 10^6)(0.025)}} = 0.050 \text{ in.}$$

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:

$$\delta = \frac{32D^2T}{\pi Ed^4} = \frac{(32)(0.750^2)(0.1)}{\pi(30\times 10^6)(0.030)^4} = 0.023 \text{ in.}$$

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.

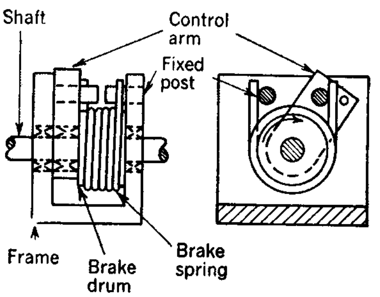


Fig. 1 Variable-torque drag brake . . .

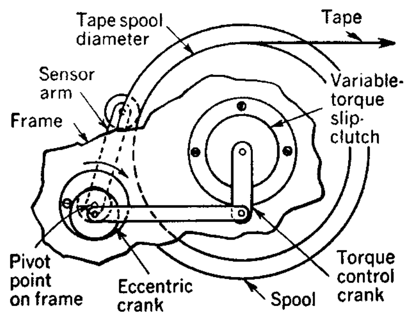


Fig. 2 . . . holds tension constant on tape

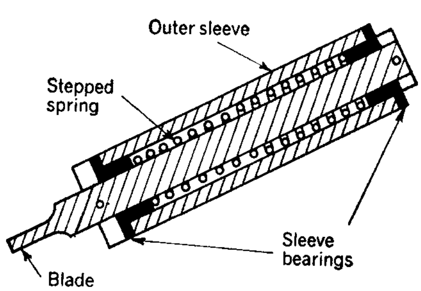


Fig. 3 Constant-torque screwdriver

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.

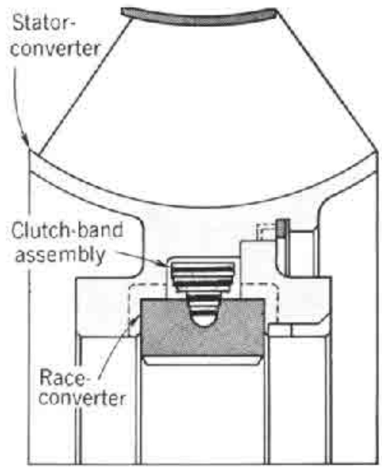
Role of spring bands. Overrunning clutches, including the spiral-band type, slip and overrun when reversed (see drawing). This occurs when the outer member is rotated clockwise and the inner ring is the driven member.

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.

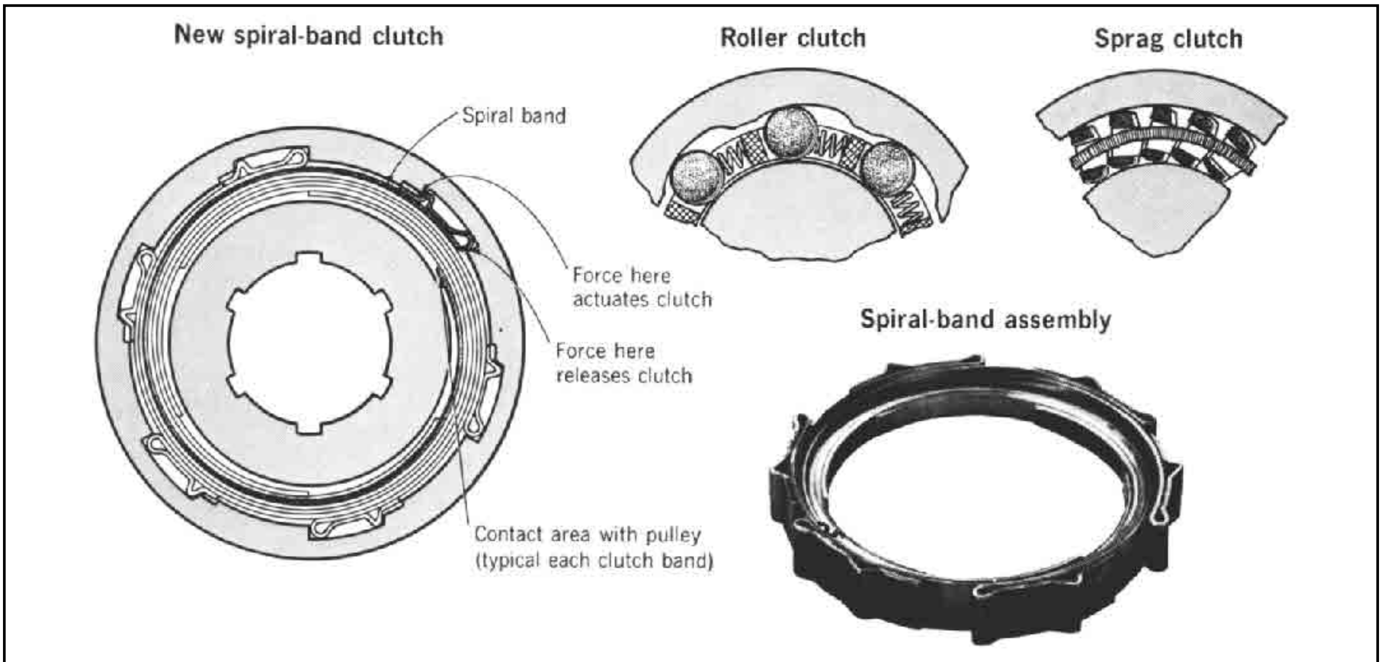
Reduced wear. The bands are spring-loaded over the inner member of the clutch, but they are held and rotated by the outer member. The centrifugal force on the bands then releases much of the force on the inner member and considerably decreases the overrunning torque. Wear is consequently greatly reduced.

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.

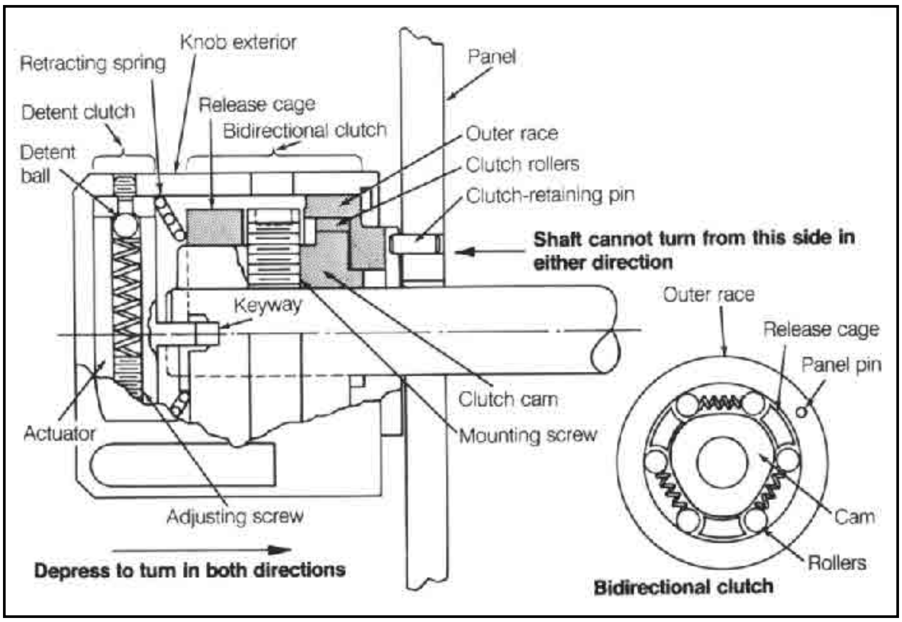
The entire mechanism was subjected to vibration, acceleration and deceleration, shock, and other high-torque loads, all of which could feed back through the system and might move the counter. The new knob device positively locks the mechanism shaft against the vibration, shock loads, and accidental turning, 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.

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.

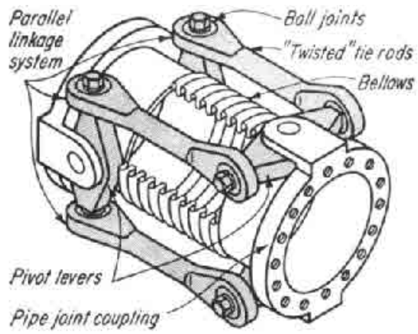


Miniature knob is easily operated from outside the panel by pushing it in and turning it in the desired direction. When released, the bi-directional clutch automatically locks the shaft against all conditions of shock and vibration.

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

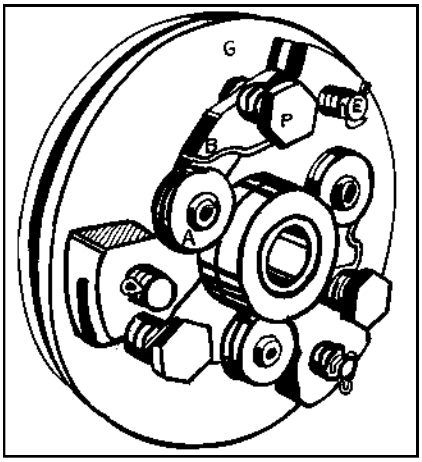
This automatic clutch causes the driving plate to move around the surface of the driven plate to prevent the clutch plates from overheating if the load gets too high. The “walking” action enables the clutch to transmit full engine torque for hours without serious damage to the clutch plates or the engine.

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.

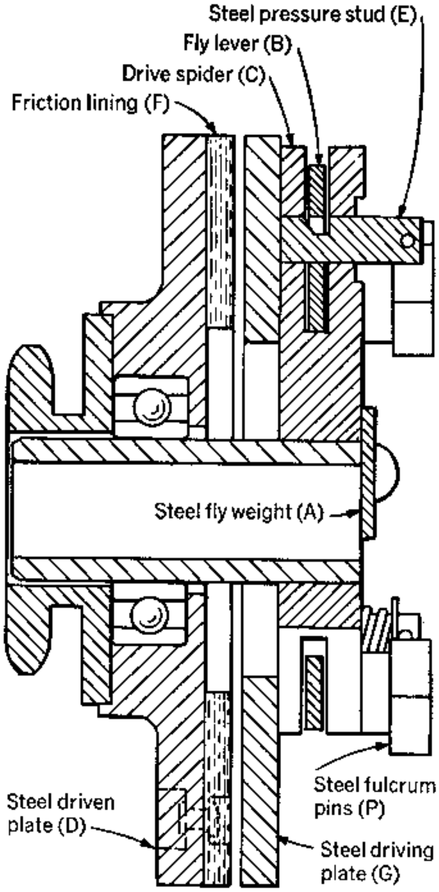


Applications. The clutch has undergone hundreds of hours of development testing on 4-stroke engines that ranged from 5 to 9 hp. According to the K-M Clutch Co., the clutch enables designers to use smaller motors than they previously could because of its no-load starting characteristics.

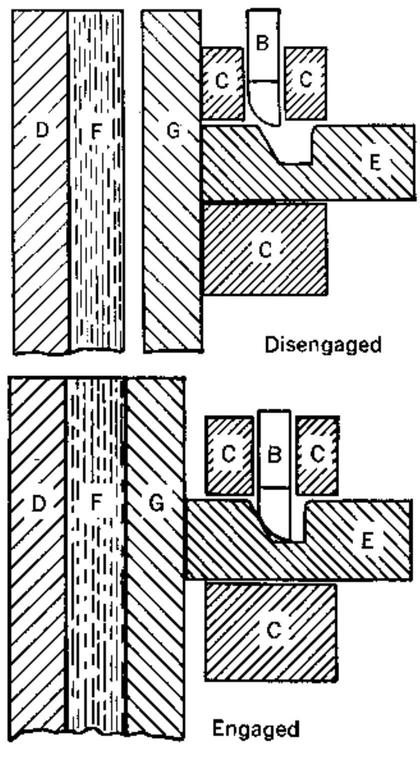
The clutch also acts as a brake to hold engine speeds within safe limits. For example, if the throttle accidentally opens when the driving wheels or driven mechanisms are locked, the clutch will stop.

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

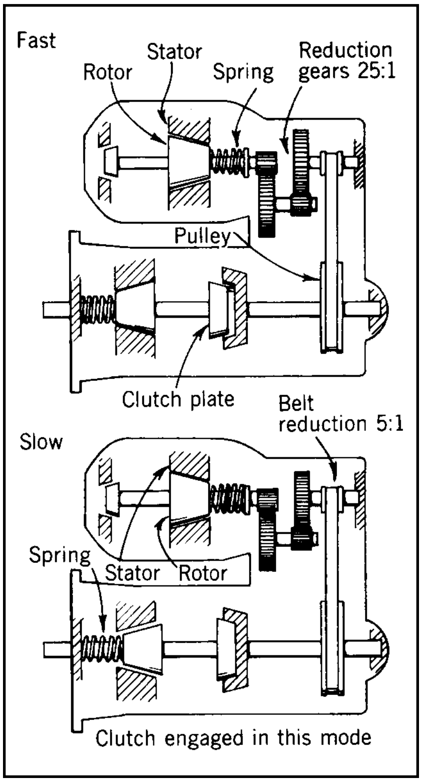
By reshaping the rotor of an ac electric motor, engineers at Demag Brake Motors, Wyandotte, Michigan, found that the axial component of the magnetic forces can be used to act on a clutch or a brake. Moreover, the motors can be arranged in tandem to obtain fast or slow speeds with instant clutching or 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.

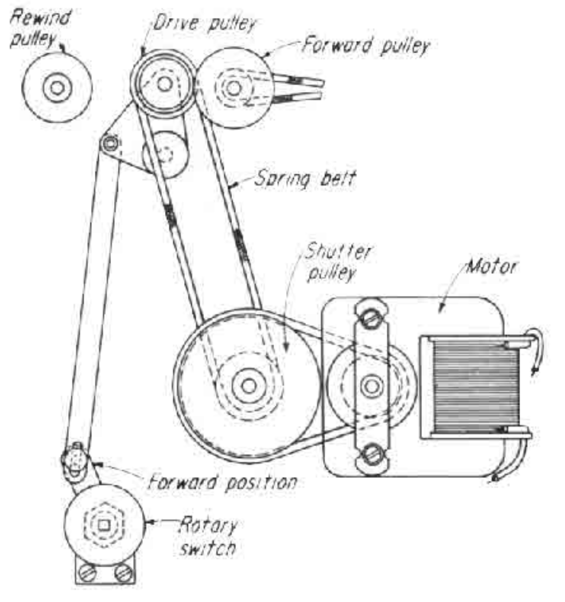
In Europe, the conical-rotored motor is used where rapid braking is essential to overcome time consuming overruns, or where accurate braking and precise angular positioning are critical—such as in packaging machines.

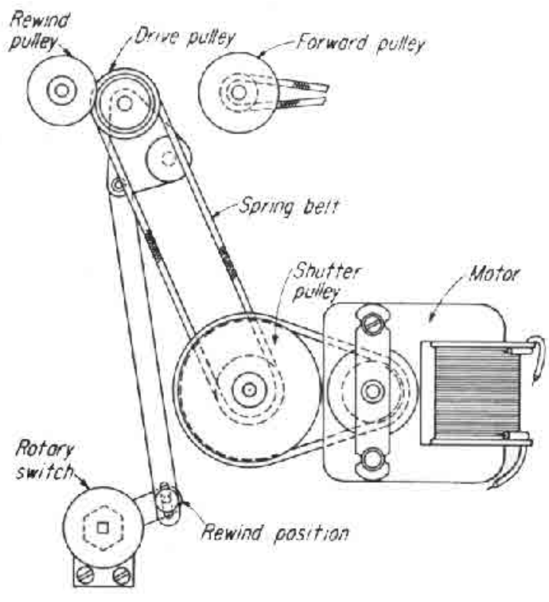
Novel arrangement. For instance, if two motors are installed, one running at 900 rpm and the other at 3600 rpm, the unit can reduce travel at a precise moment from a fats speed to an inching speed. This is achieved in the following way: When the main motor (running at 3600 rpm, and driving a conveyor 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.

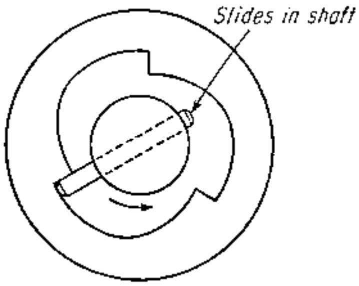


Fig. 1 A lawnmower clutch.

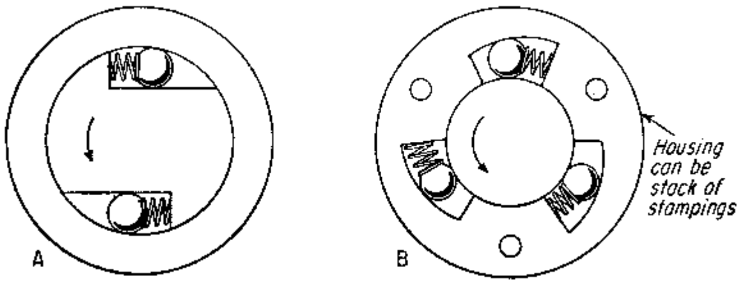


Fig. 2 Wedging balls or rollers: internal (A); external (B) clutches.

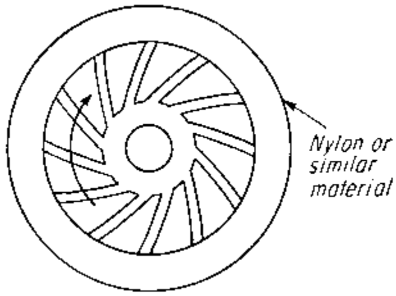


Fig. 3 Molded sprags (for light duty).

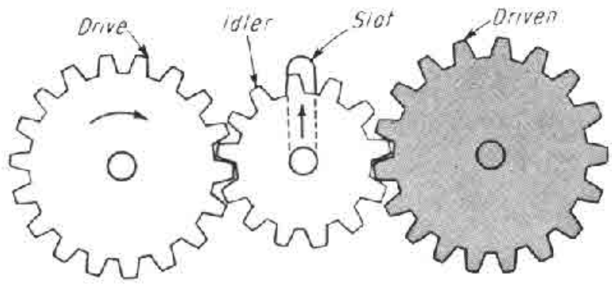


Fig. 4 A disengaging idler rises in a slot when the drive direction is reversed.

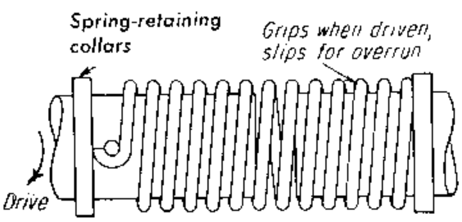


Fig. 5 A slip-spring coupling.

Fig. 6 An internal ratchet and spring-loaded pawls.

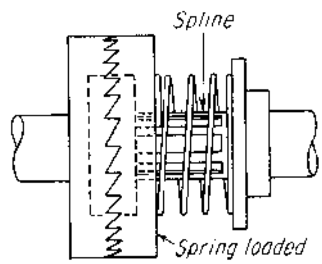
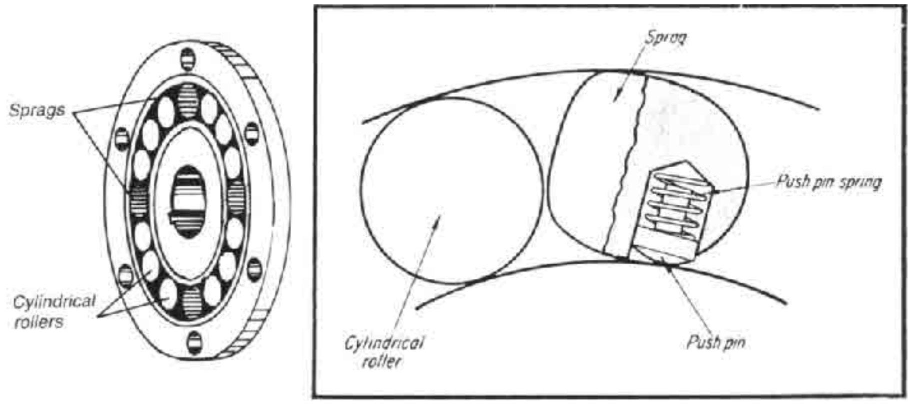


Fig. 7 A one-way dog clutch.

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.

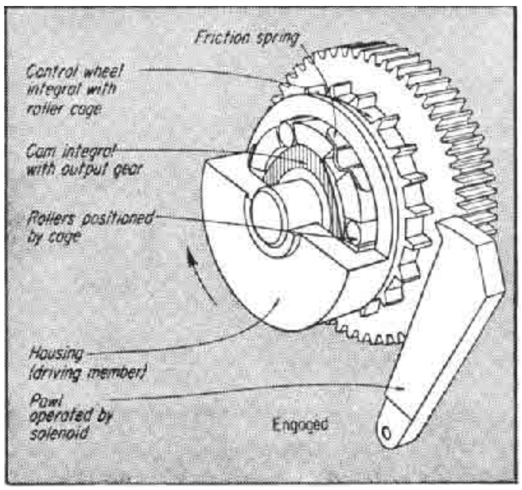
ROLLER-TYPE CLUTCH

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).

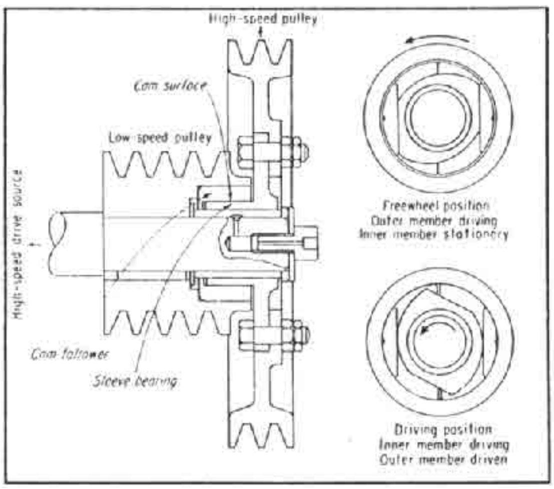
When the pawl is disengaged, the drag of the housing on the friction spring rotates the cage and wedges the rollers into engagement. This permits the housing to drive the gear through the cam.

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.



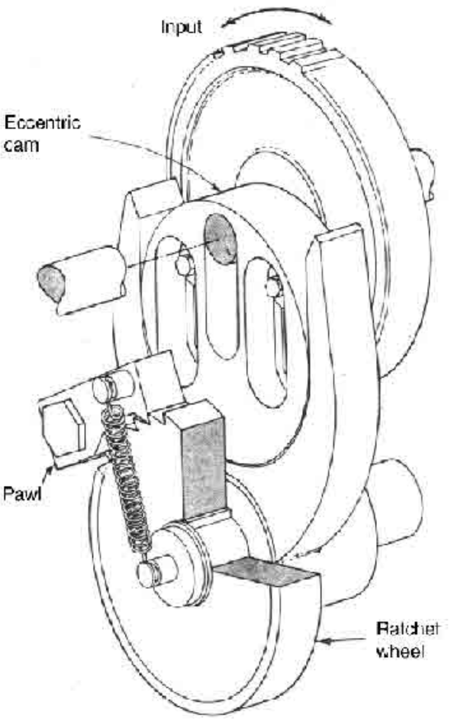
A positive drive is provided by this British roller clutch.



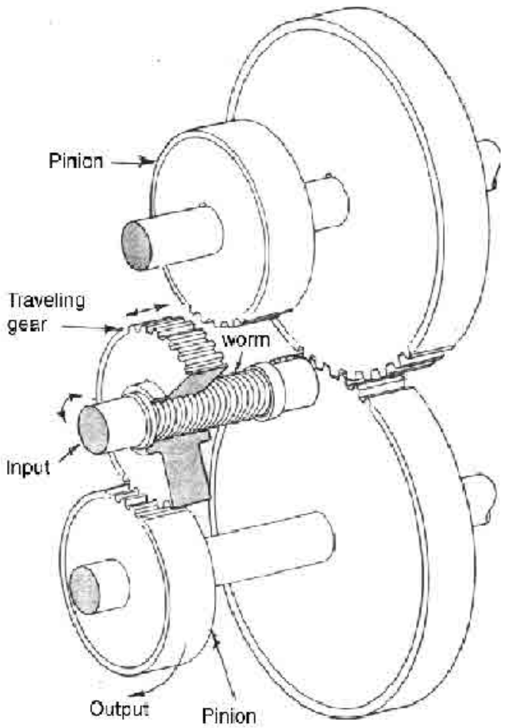
Two-speed operation is provided by the new cam clutch

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.

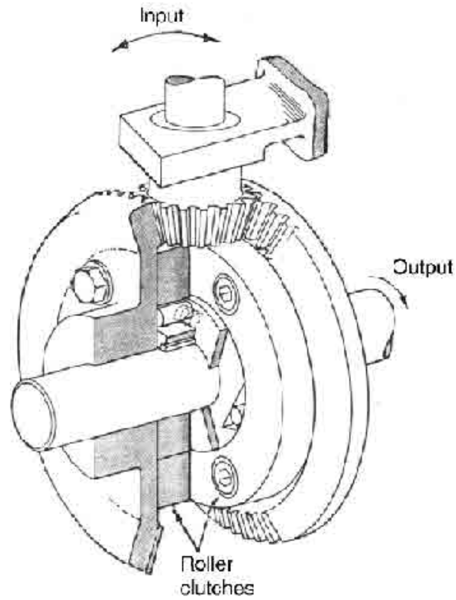
ONE-WAY OUTPUT FROM SPEED REDUCERS



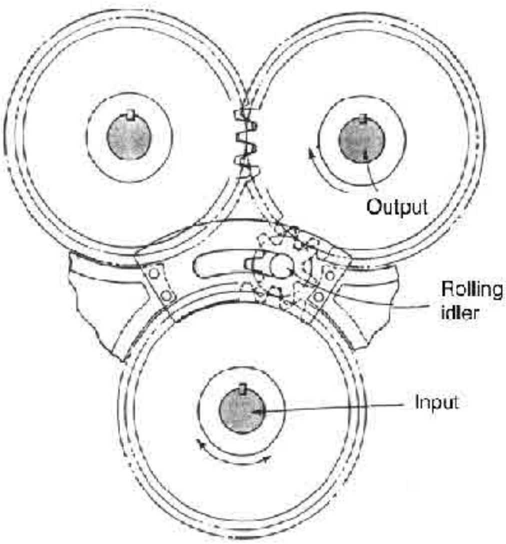
This eccentric cam adjusts over a range of high reduction ratios, but unbalance limits it to low speeds. When its direction of input changes, thee is no lag in output rotation. The output shaft moves in steps because of a ratchet drive through a pawl which is attached to a U follower.



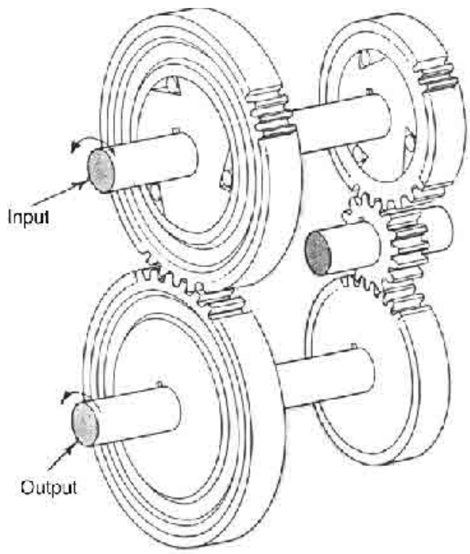
A traveling gear moves along a worm and transfers drive torque to the other pinion when the input rotation changes direction. To ease the gear engagement, the gear teeth are tapered at their ends. Output rotation is smooth, but there is a lag after direction changes as the gear shifts. The gear cannot 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.

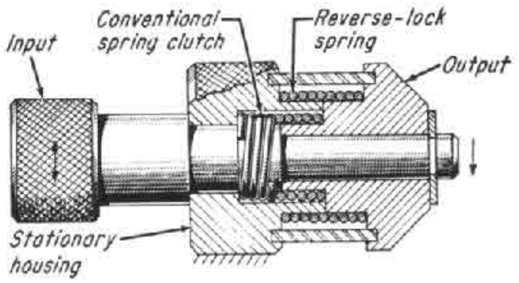


Fig. 1 A double-spring clutch drive.

Fig. 2 A basic spring clutch.

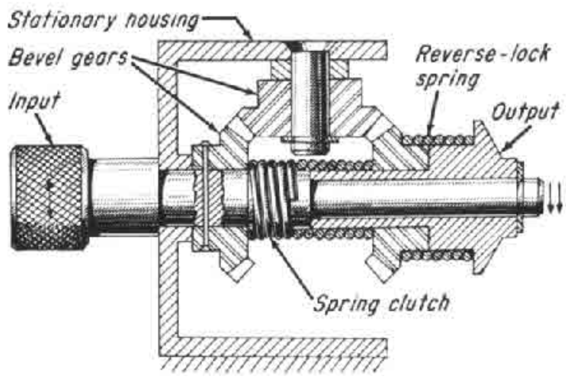


Fig. 3 A full-wave rectification drive.

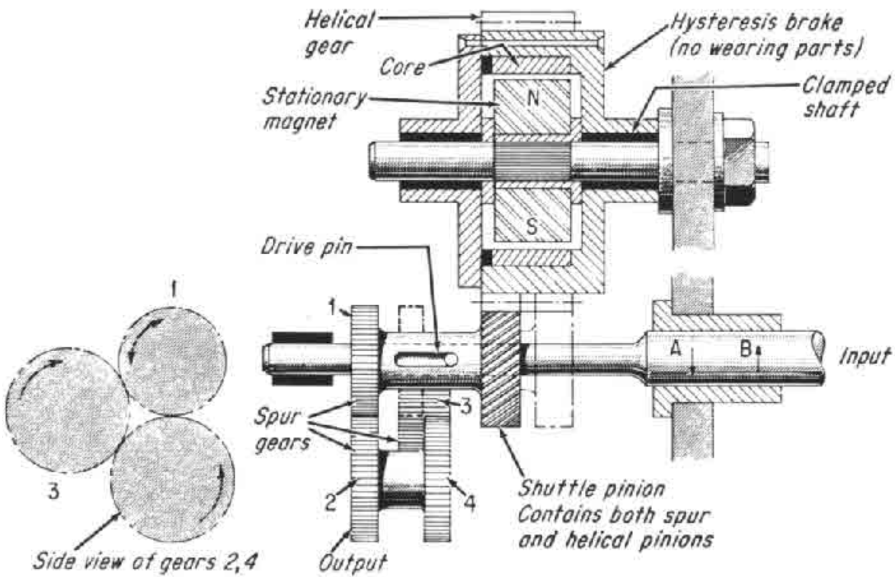
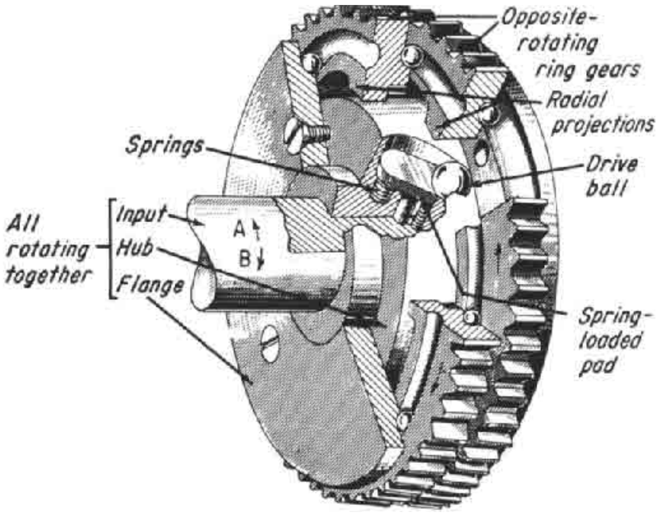


Fig. 4 A shuttle-pinion drive.



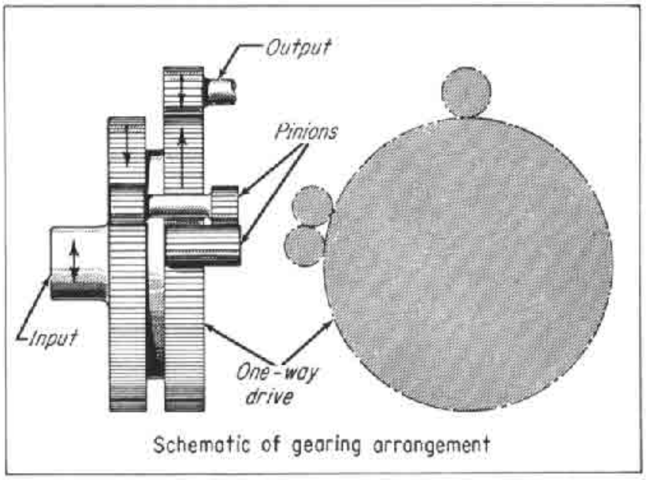


Fig. 5 A reciprocating-ball drive.

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

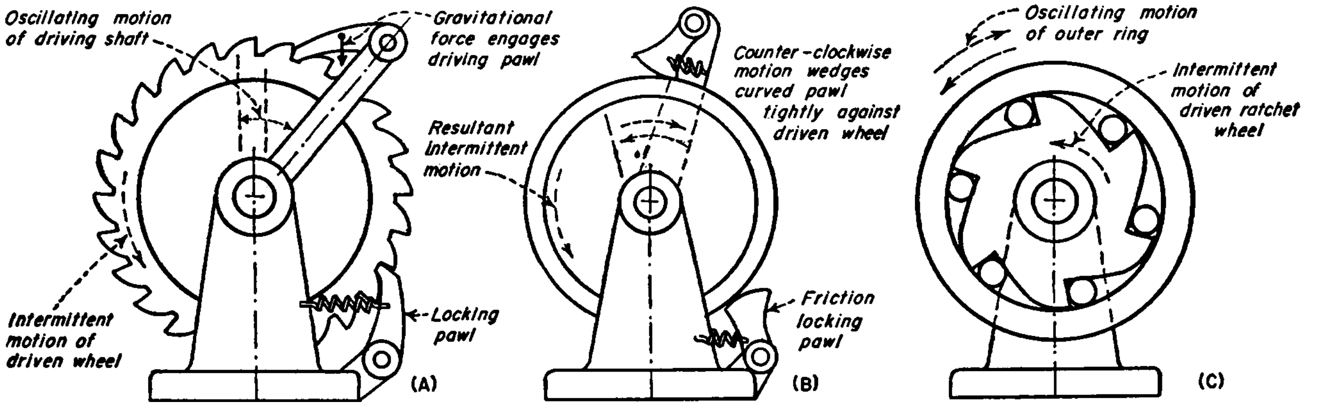


Fig. 1 Elementary overriding clutches: (A) A ratchet and pawl mechanism converts reciprocating or oscillating movement to intermittent rotary motion. This motion is positive but limited to a multiple of the tooth pitch. (B) A friction-type clutch is quieter, but it requires a spring device to keep the 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.

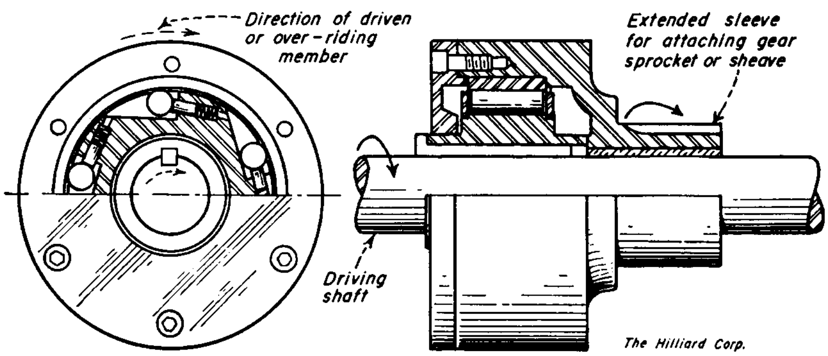


Fig. 2 A commercial overriding clutch has springs that hold the rollers in continuous contact between the cam surfaces and the outer race; thus, there is no backlash or lost motion. This simple design is positive and quiet. For operation in the opposite direction, the roller mechanism can easily be reversed in the housing.

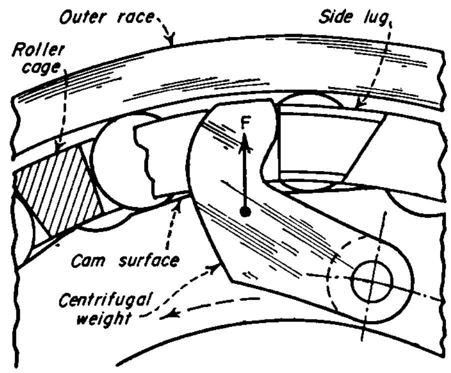


Fig. 3 A centrifugal force can hold the rollers in contact with the cam and outer race. A force is exerted on the lugs of the cage that controls the position of the rollers.

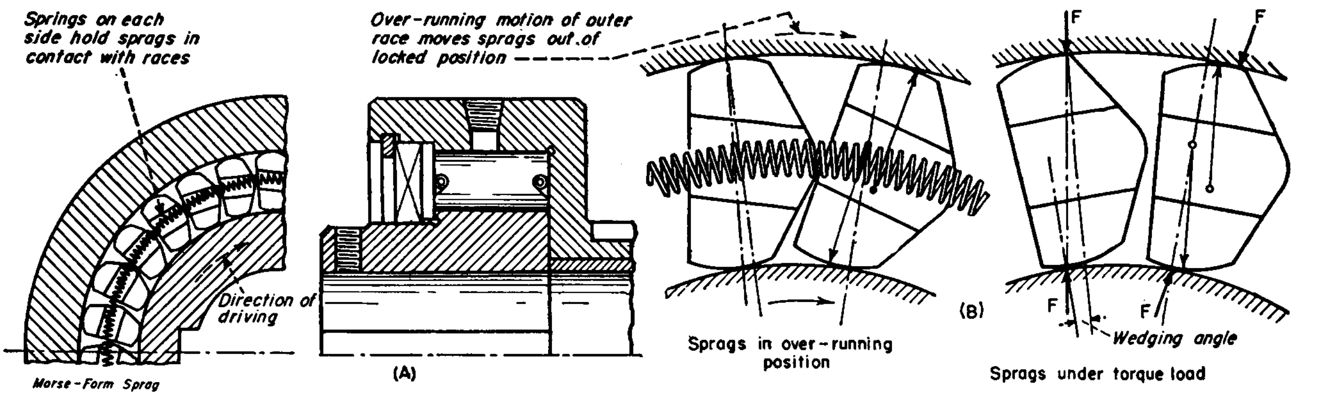


Fig. 4 With cylindrical inner and outer races, sprags can transmit torque. Energizing springs serve as a cage to hold the sprags. (A) Compared to rollers, the shape of a sprag permits a greater number within a limited space; thus higher torque loads are possible. Special cam surfaces are not required, 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.

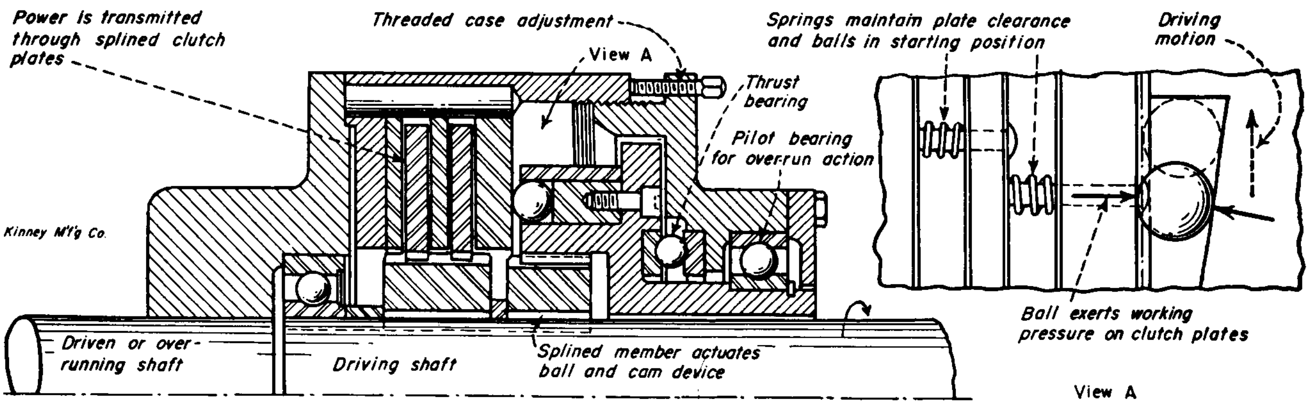


Fig. 5 A multidisk clutch is driven by several sintered-bronze friction surfaces. Pressure is exerted by a cam-actuating device that forces a series of balls against a disk plate. A small part of the transmitted torque is carried by the actuating member, so capacity is not limited by the localized deformation 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.

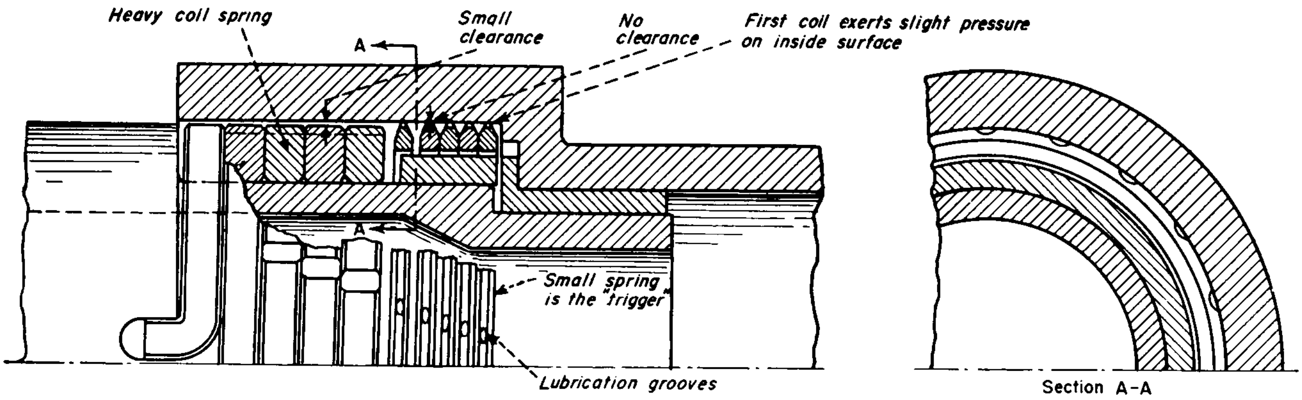


Fig. 6 An engaging device consists of a helical spring that is made up of two sections: a light trigger spring and a heavy coil spring. It is attached to and driven by the inner shaft. The relative motion of the outer member rubbing on the trigger causes this spring to wind up. This action expands the 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.

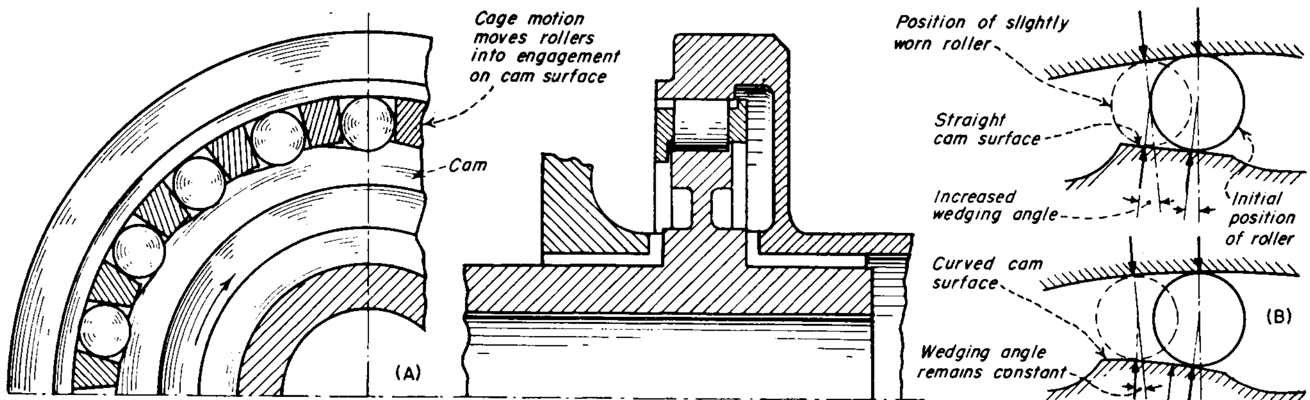


Fig. 7 A free-wheeling clutch widely used in power transmission has a series of straight-sided cam surfaces. An engaging angle of about 3º is used; smaller angles tend to become locked and are difficult to disengage while larger ones are not as effective. (A) The inertia of a floating cage wedges the 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.

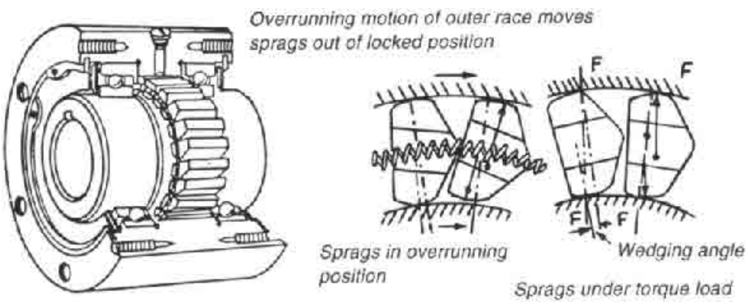


Fig. 1 Precision sprags act as wedges and are made of hardened alloy steel. In the formsprag clutch, torque is transmitted from one race to another by the wedging action of sprags between the races in one direction; in the other direction the clutch freewheels.

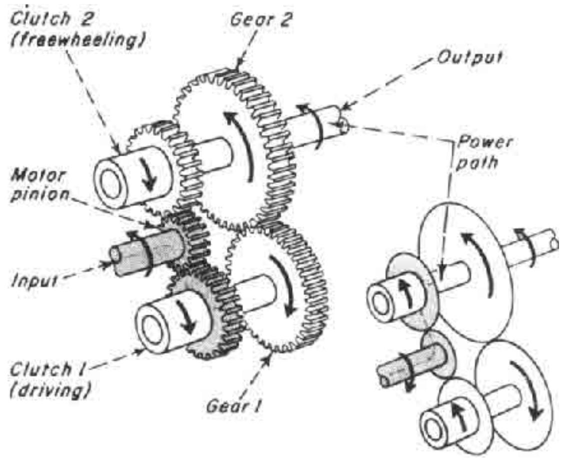


Fig. 2 This speed drive requires input rotation to be reversible. Counterclockwise input (as shown in the diagram) drives gear 1 through clutch 1; the output is counterclockwise; clutch 2 overruns. Clockwise input (schematic) drives gear 2 through clutch 2; the output is still counterclockwise; clutch 1 overruns.

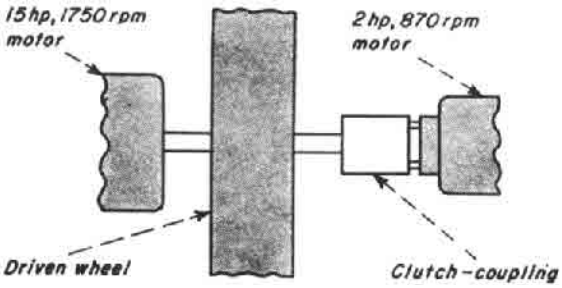


Fig. 3 This speed drive for a grinding wheel can be a simple, in-line assembly if the overrunning clutch couples two motors. The outer race of the clutch is driven by a gearmotor; the inner race is keyed to a grinding-wheel shaft. When the gearmotor drives, the clutch is engaged; when the larger motor drives, the inner race overruns.

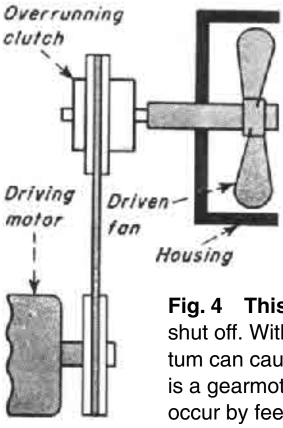


Fig. 4 This fan freewheels when driving power is shut off. Without an overrunning clutch, fan momentum can cause belt breakage. If the driving source is a gearmotor, excessive gear stress can also occur by feedback of kinetic energy from the fan.

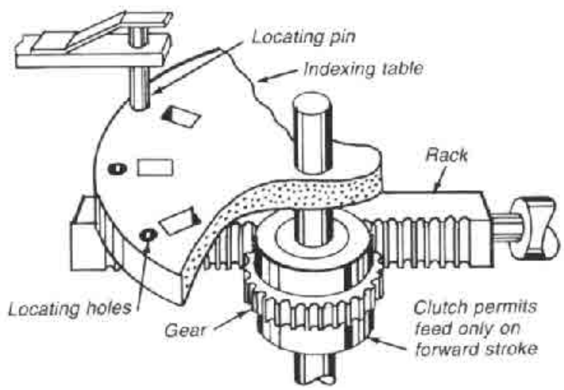


Fig. 5 This indexing table is keyed to a clutch shaft. The table is rotated by the forward stroke of the rack; power is transmitted through the clutch by its outer-ring gear only during this forward stroke. Indexing is slightly short of the position required. The exact position is then located by a spring-loaded 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.

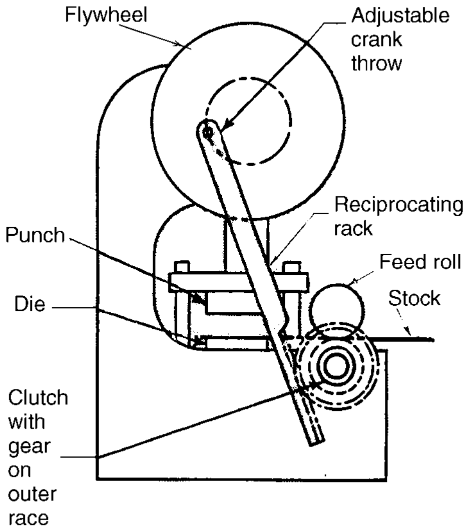


Fig. 6 This punch press feed is arranged so that the strip is stationary on the downstroke of the punch (clutch freewheels); feed occurs during the upstroke when the clutch transmits torque. The feed mechanism can easily be adjusted to vary the feed amount.

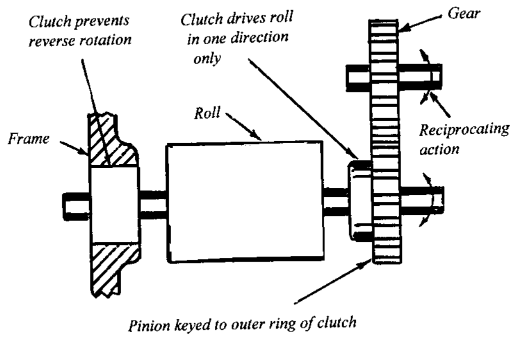


Fig. 7 Indexing and backstopping is done with two clutches arranged so that one drives while the other freewheels. The application shown here is for a capsuling machine; gelatin is fed by the roll and stopped intermittently so the blade can precisely shear the material to form capsules.

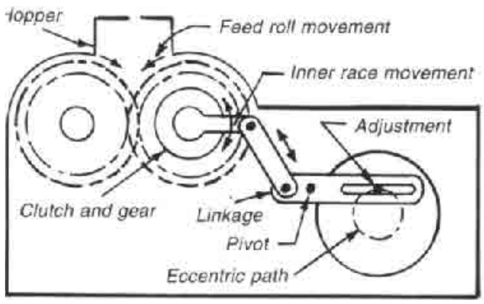


Fig. 8 The intermittent motion of a candy machine is adjustable. The clutch ratchets the feed rolls around. This keeps the material in the hopper agitated.

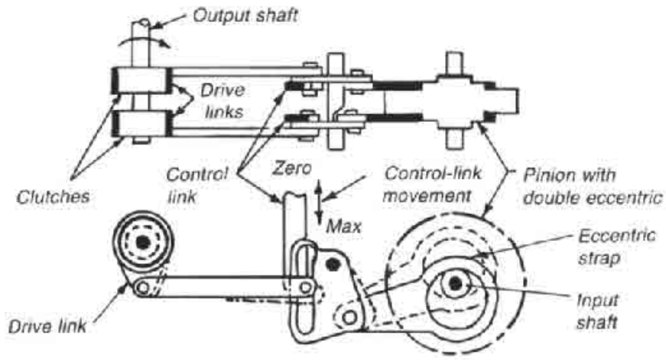


Fig. 9 This double-impulse drive has double eccentrics and drive clutches. Each clutch is indexed 180° out of phase with the other. One revolution of the eccentric produces two drive strokes. Stroke length, and thus the output rotation, can be adjusted from zero to maximum by the control link.

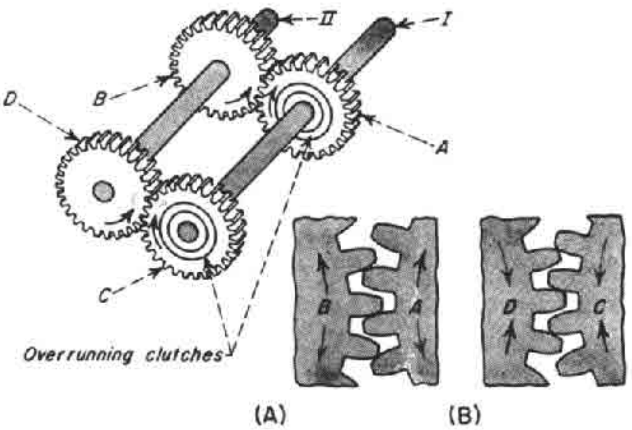


Fig. 10 This anti-backlash device depends on overrunning clutches to ensure that no backlash is left in the unit. Gear *A* drives *B* and shaft *II* with the gear mesh and backlash, as shown in (A). The overrunning clutch in gear *C* permits gear *D* (driven by shaft *II*) to drive gear *C* and results 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.

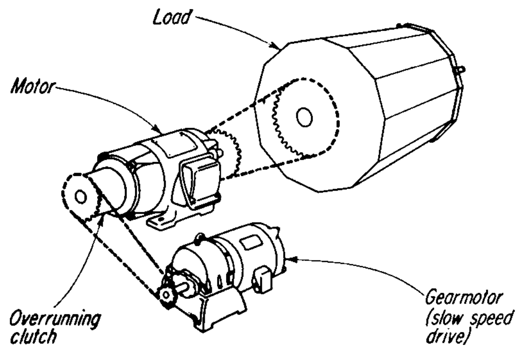


Fig. 1 Overrunning permits torque transmission in one direction and free wheels or overruns in the opposite direction. For example, the gar motor drives the load by transmitting torque through the overrunning clutch and the high-speed shaft. Energizing the high-speed motor causes the inner member to rotate at the rpm of the high-speed motor. The gear motor continues to drive the inner member, but the clutch is free-wheeling.

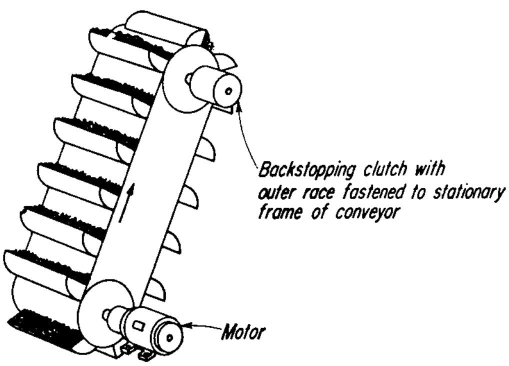


Fig. 2 Backstopping permits rotation in one direction only. The clutch serves as a counter-rotation holding device. An example is a clutch mounted on the headshaft of a conveyor. The outer race is restrained by torque-arming the stationary frame of the conveyor. If, for any reason, power to the conveyor is interrupted, the back-stopping clutch will prevent the buckets from running backwards and dumping the load.

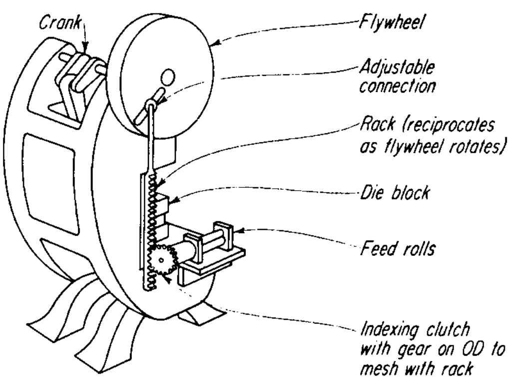


Fig. 3 Indexing is the transmission of intermittent rotary motion in one direction; an example is the feed rolls of a punch press. On each stroke of the press crankshaft, a feed stroke on the feed roll is accomplished by the rack-and-pinion system. The system feeds the material into the dies of the punch press.

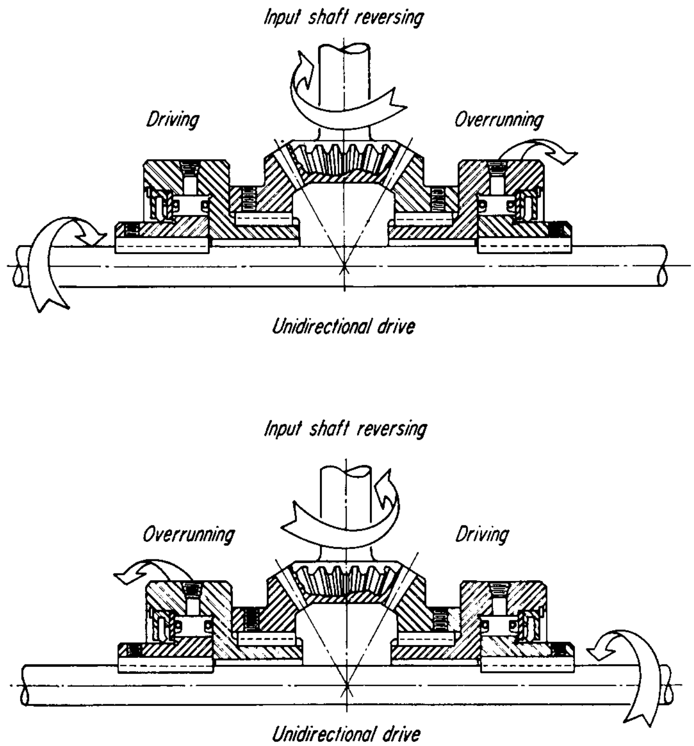


Fig. 4 Unidirectional drives with reverse mechanism incorporate two overrunning clutches into the gears, sheaves, or sprockets. Here, a 1:1 ratio right-angle drive is shown with a reversing input shaft. The output shaft rotates clockwise, regardless of the input shaft direction. By changing gear sizes, combinations of continuous or intermittent unidirectional output relative to the input can be obtained.

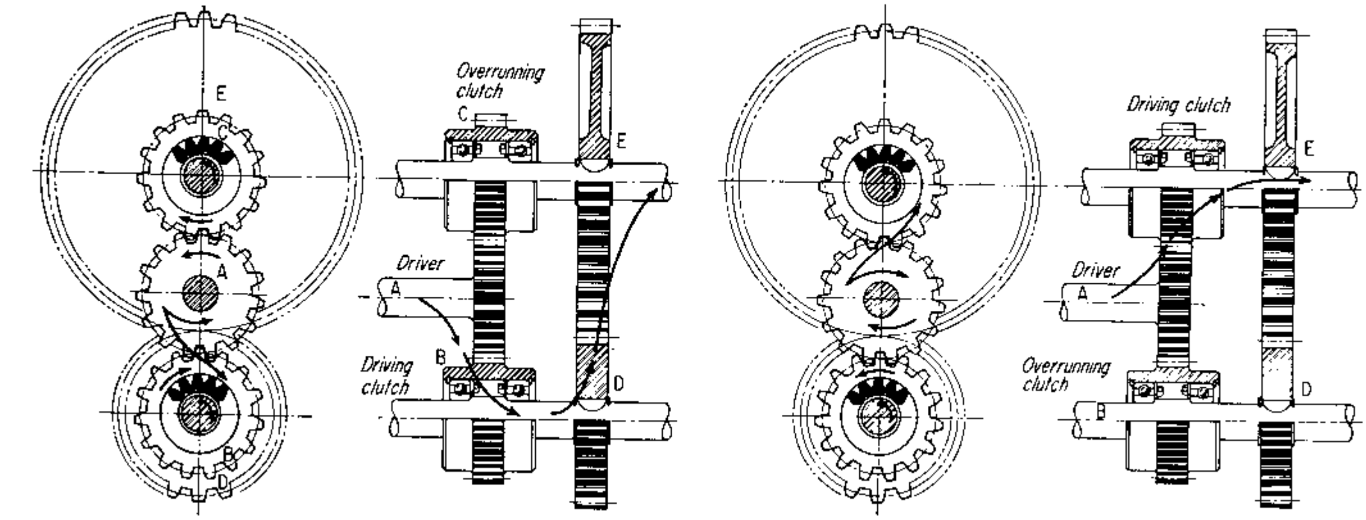


Fig. 5 Two-speed unidirectional output is made possible by using spur gears and reversing the direction of the input shaft. The rotation of shaft A transfers the power of gears B, D, and E to the output. Counterclockwise rotation engages the lower clutch, freewheeling the upper clutch because gear C 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 free-wheels because of the speed increase between gears D and E.

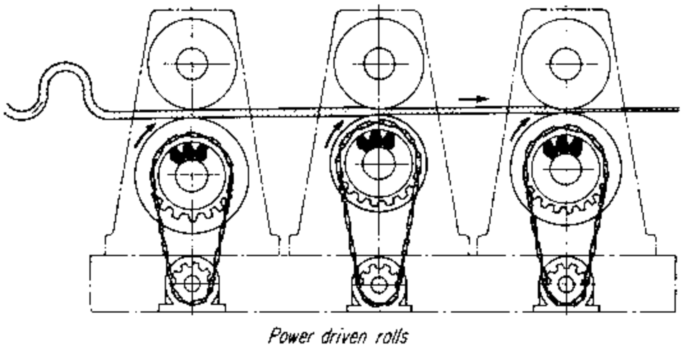


Fig. 6 A speed-differential or compensation is required where a different speed range for a function is desired, while retaining the same basic speed for all other functions. A series of individually driven power rolls can have different surface speeds because of drive or diameter variations of the rolls. An overrunning clutch permits the rolls with slower peripheral speed to overspeed and adjust to the material speed.

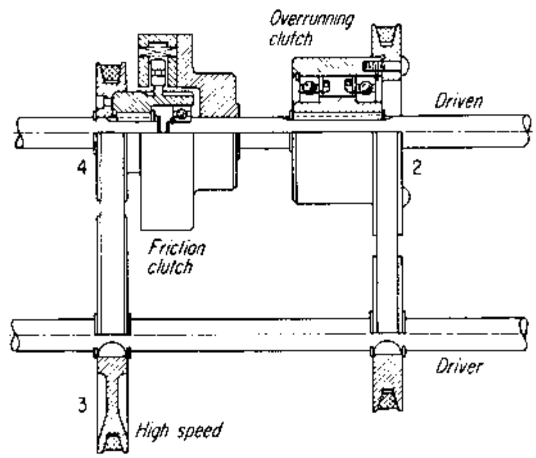


Fig. 7 A speed differential application permits the operation of engine accessories within a narrow speed range while the engine operates over a wide range. Pulley No. 2 contains the overrunning clutch. When the friction or electric clutch is disengaged, the driver pulley drives pulley No. 2 through 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.

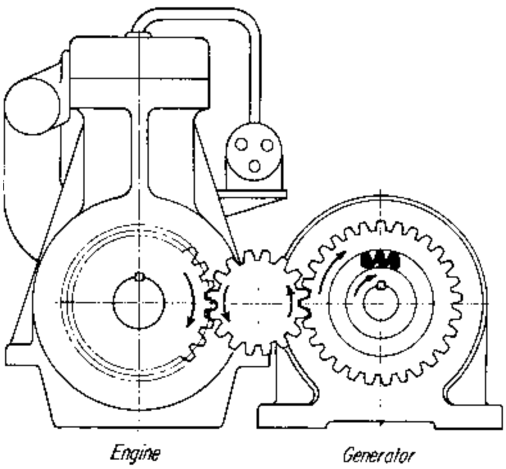


Fig. 8 High inertia dissipation avoids driving back through a power system. In machines with high resistances, it prevents power train damage. If the engine is shut down while the generator is under a no-load condition, it would have a tendency to twist off the generator shaft. The overrunning clutch 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.

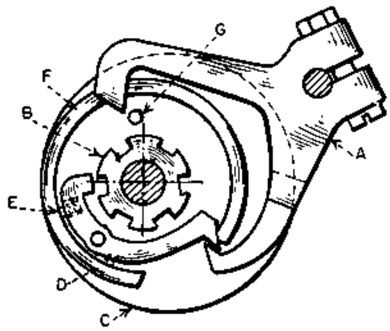


Fig. 1

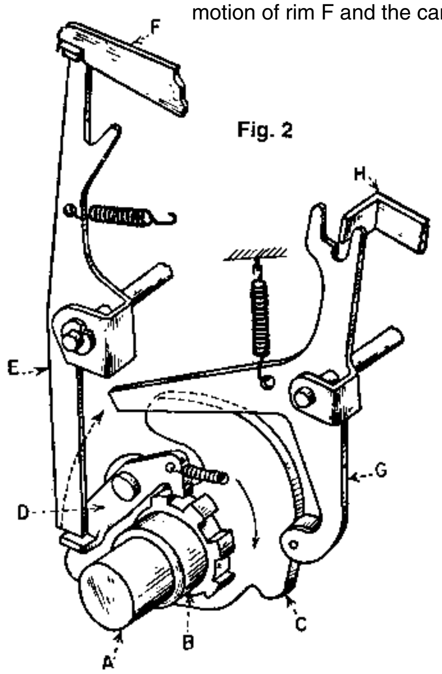


Fig. 2

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.

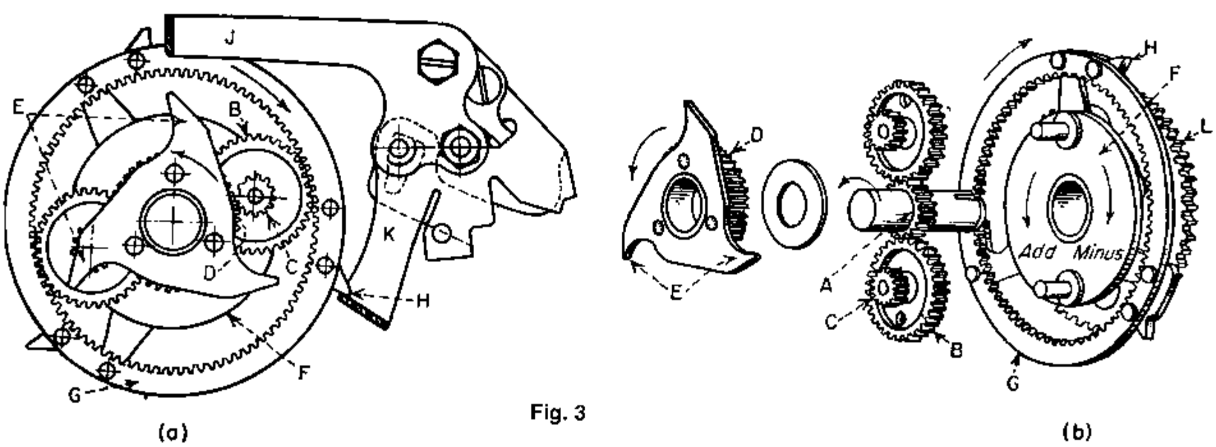


Fig. 3

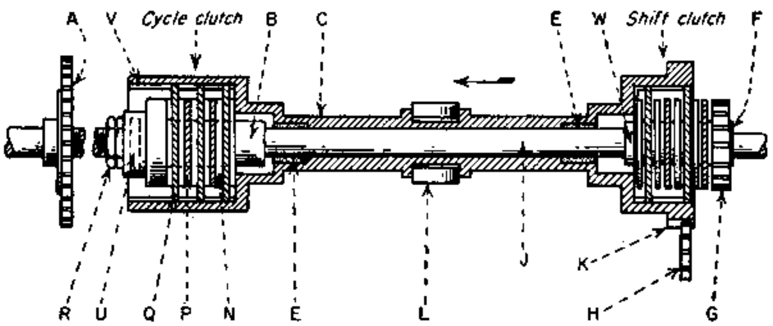


Fig. 4

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.

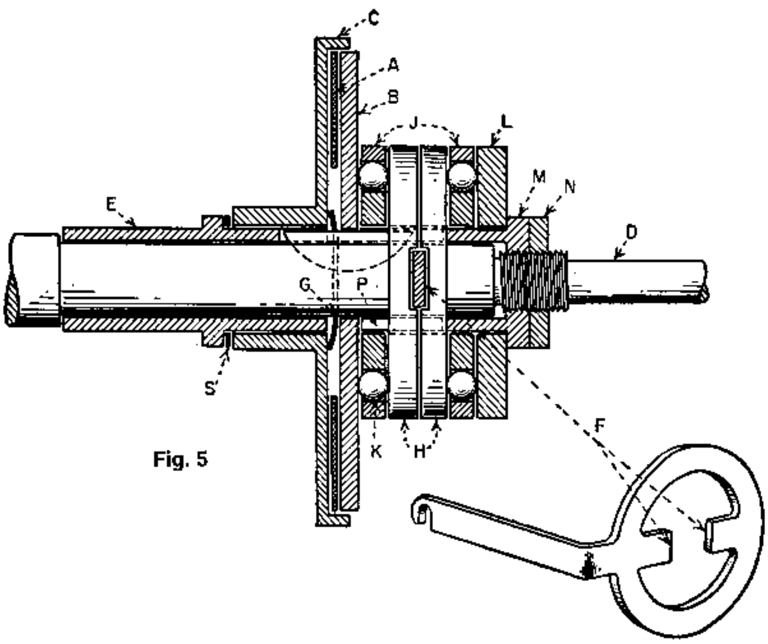


Fig. 5

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.

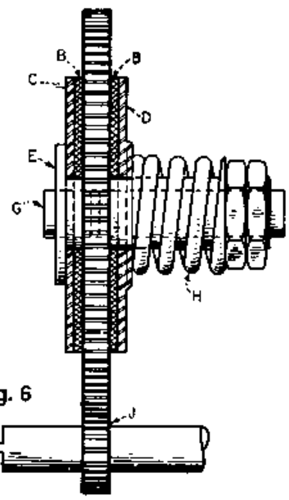


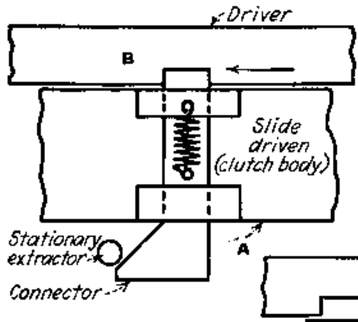
Fig. 6

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.

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.



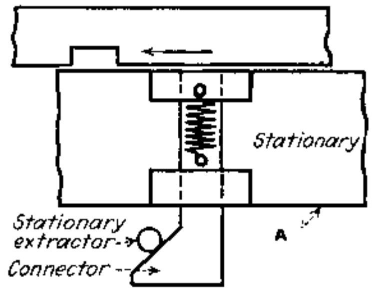


Fig. 1(*a*) The driver and clutch body move in unison with the connector just coming in contact with the extractor. In **Fig. 1(*b*)** continued movement withdraws the connector.

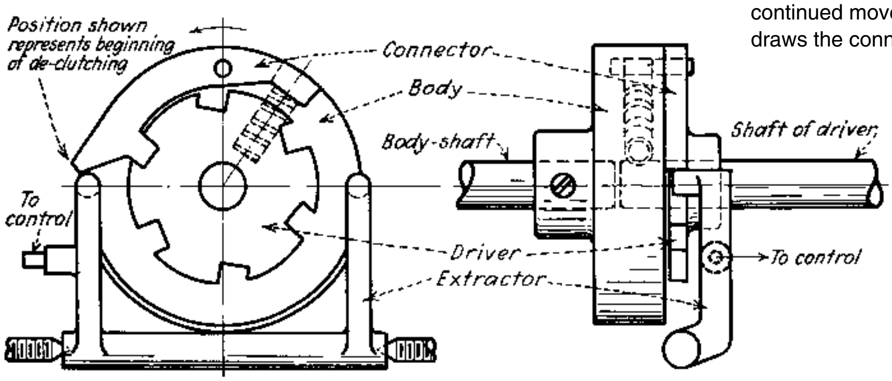


Fig. 2 A two-station clutch whose stations are 180º apart. Because it has only one extractor arm, this mechanism can function as a one-station clutch.

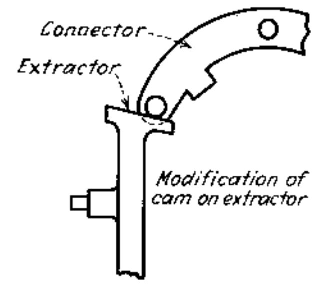


Fig. 3 A modification of the cam extractor shown in Fig. 2.

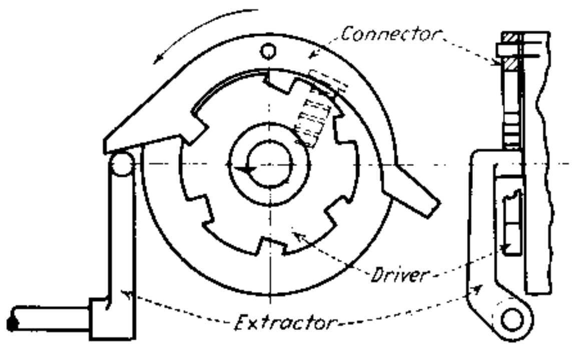


Fig. 4 A single extractor two-station clutch with the stations that are 180º apart. Only one extractor is required because the connector has two cams.

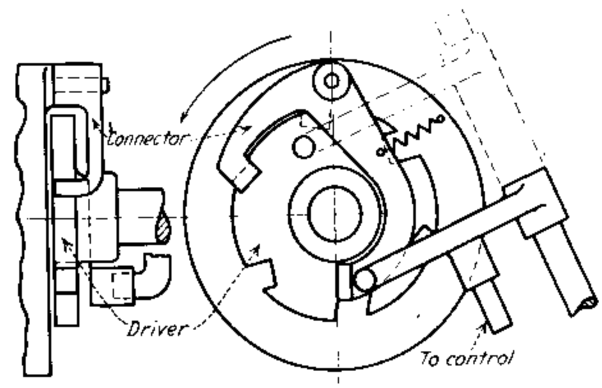


Fig. 5 This one- or two-station clutch with a dual extractor is compact because there are no parts projecting beyond its body.

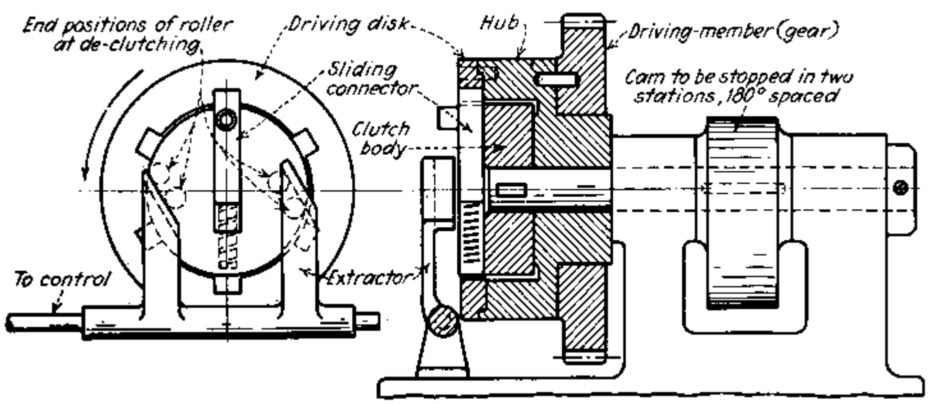


Fig. 6 The end and longitudinal section of a station clutch with internal driving recesses.

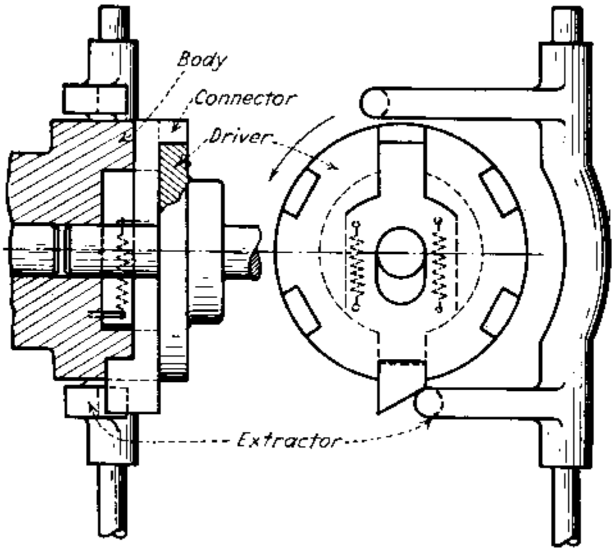


Fig. 7 This one- or two-station clutch depends on a single or a dual extractor. Its stations are spaced 180º apart.

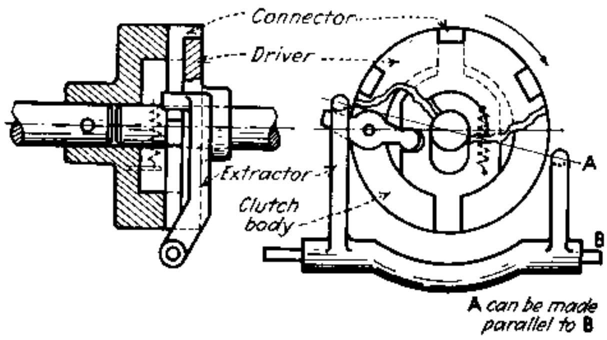


Fig. 8 This is another one- or two-station clutch. It has a single or dual extractor with stations spaced 180º apart.

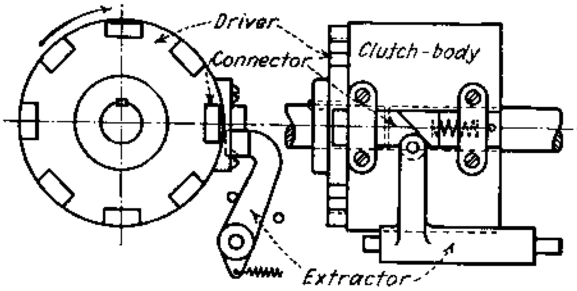
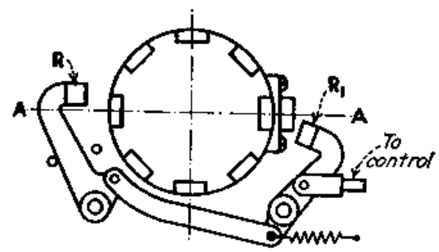


Fig. 9 A one-station axial connector clutch.

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.



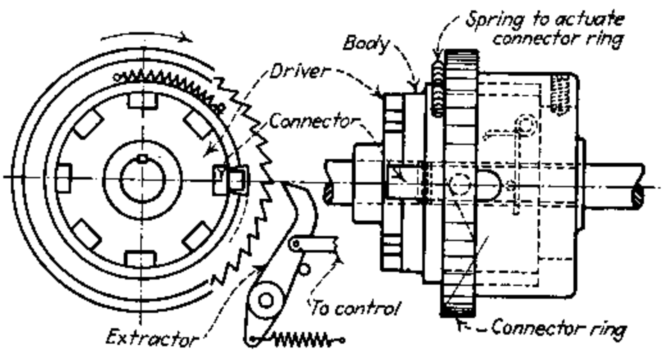
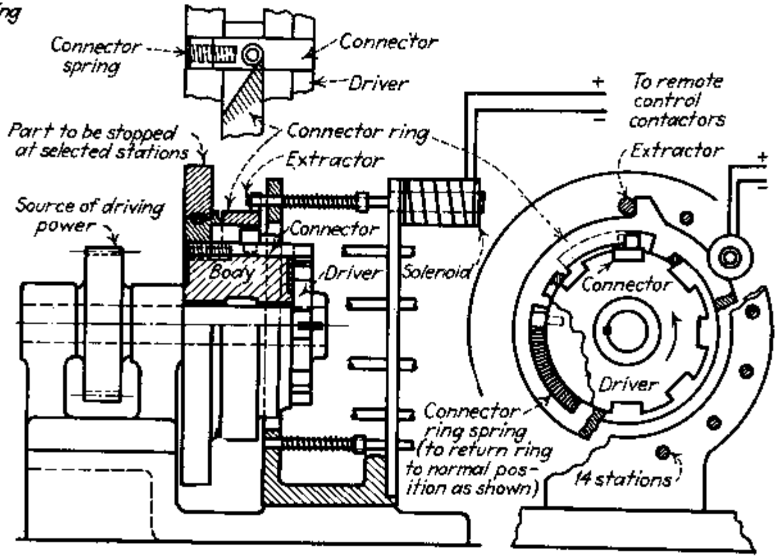


Fig. 11 A nonselective multistation clutch for instantaneous stopping in any position.

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

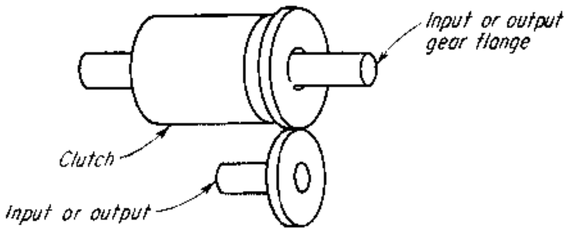


Fig. 1 Coupling or uncoupling power or sensing device.

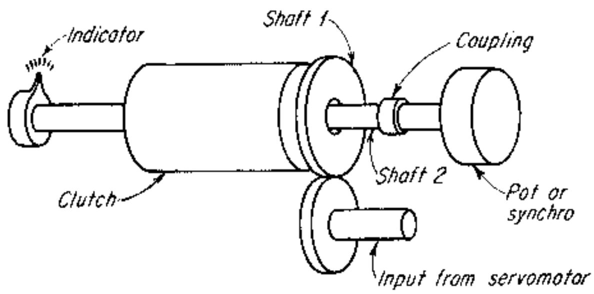
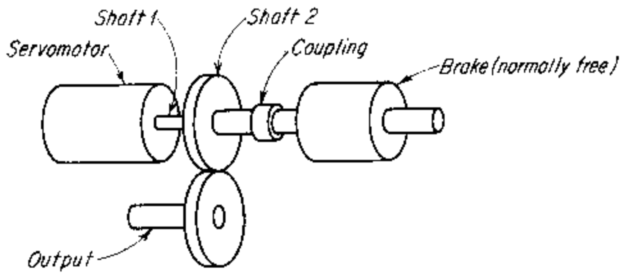
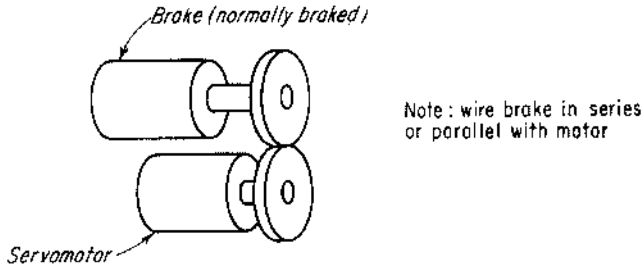


Fig. 2 Calibration protection (energize to adjust).





Figs. 3 & 4 Simple servomotor brakes.

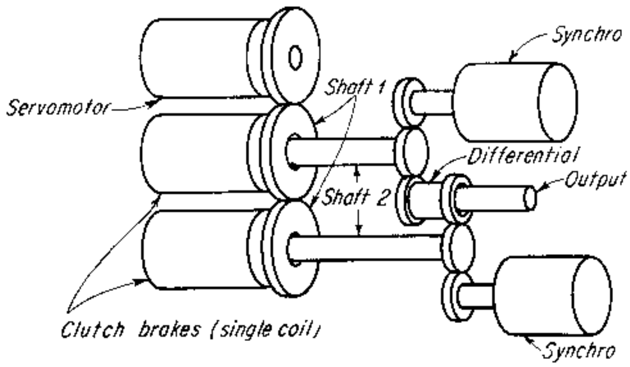


Fig. 5 Adding or subtracting two inputs.

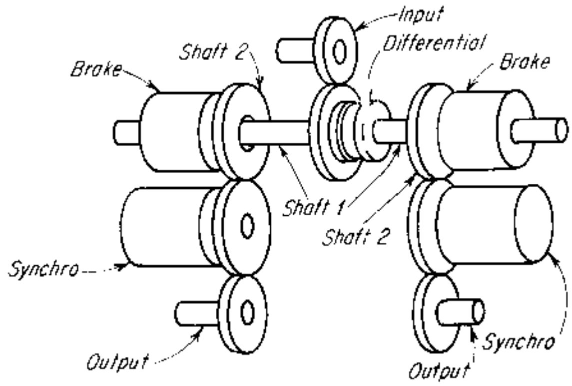


Fig. 6 Controlling output from a differential.

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,

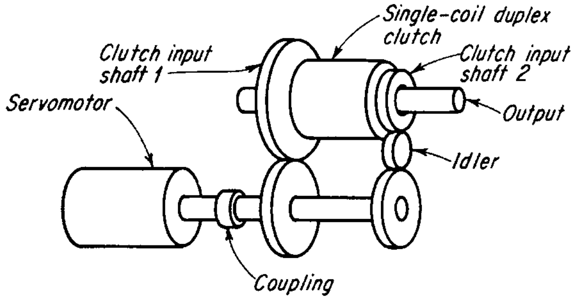


Fig. 7 Simple-speed changing.

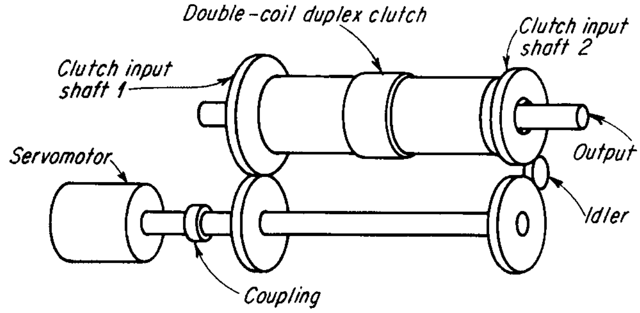


Fig. 8 Speed-changing and uncoupling.

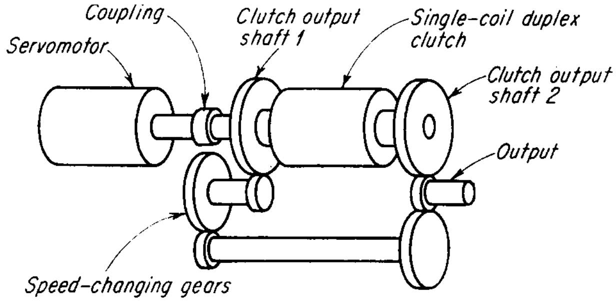


Fig. 9 Simple direction-changing.

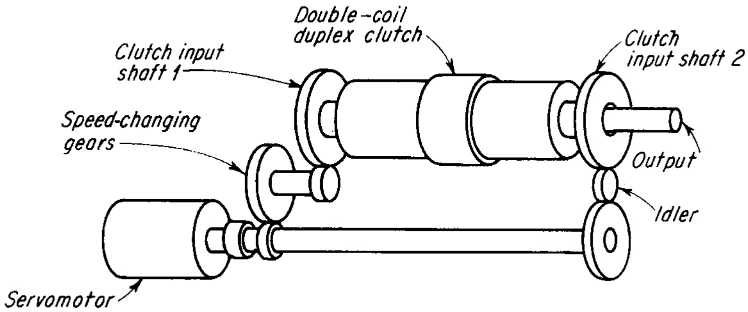


Fig. 10 Direction-changing and uncoupling.

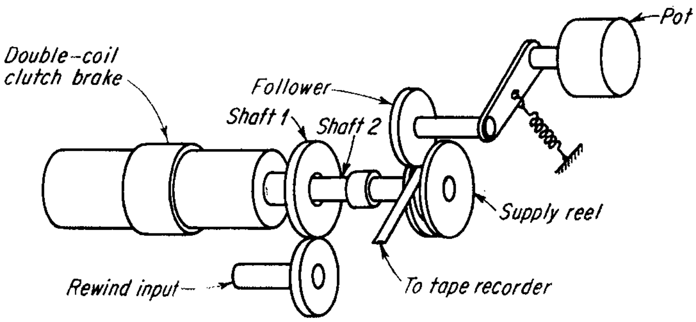


Fig. 11 Constant tensioning.

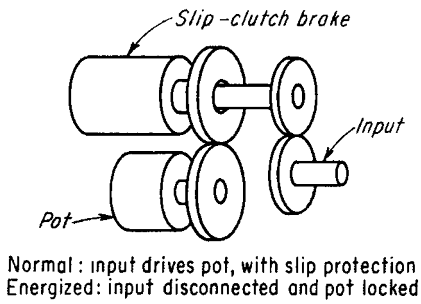


Fig. 12 Potentiometer control.

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

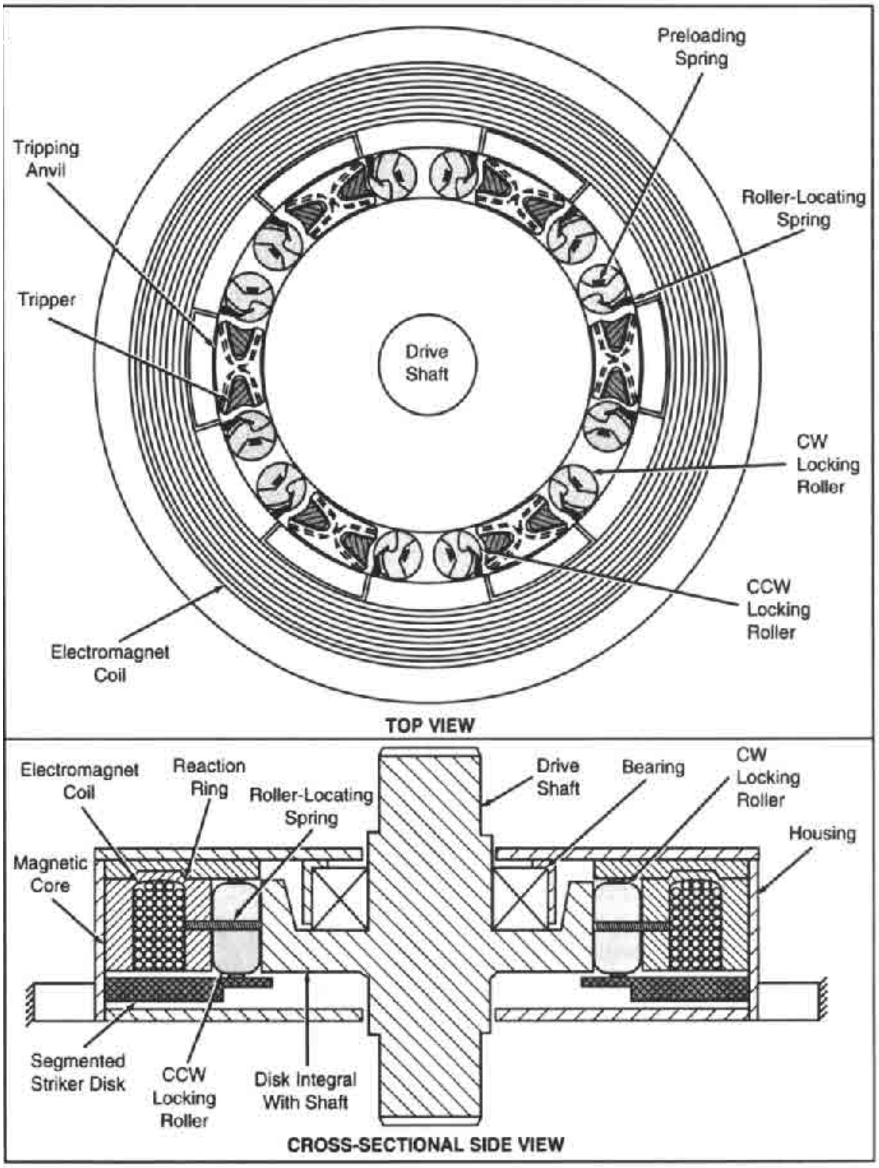
This bidirectional roller-locking clutch offers the advantages of efficiency and controllability.

The 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.



The trip roller clutch contains back-to-back roller-locking, overrunning clutches that can be released (tripped) with small magnetic forces.

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.

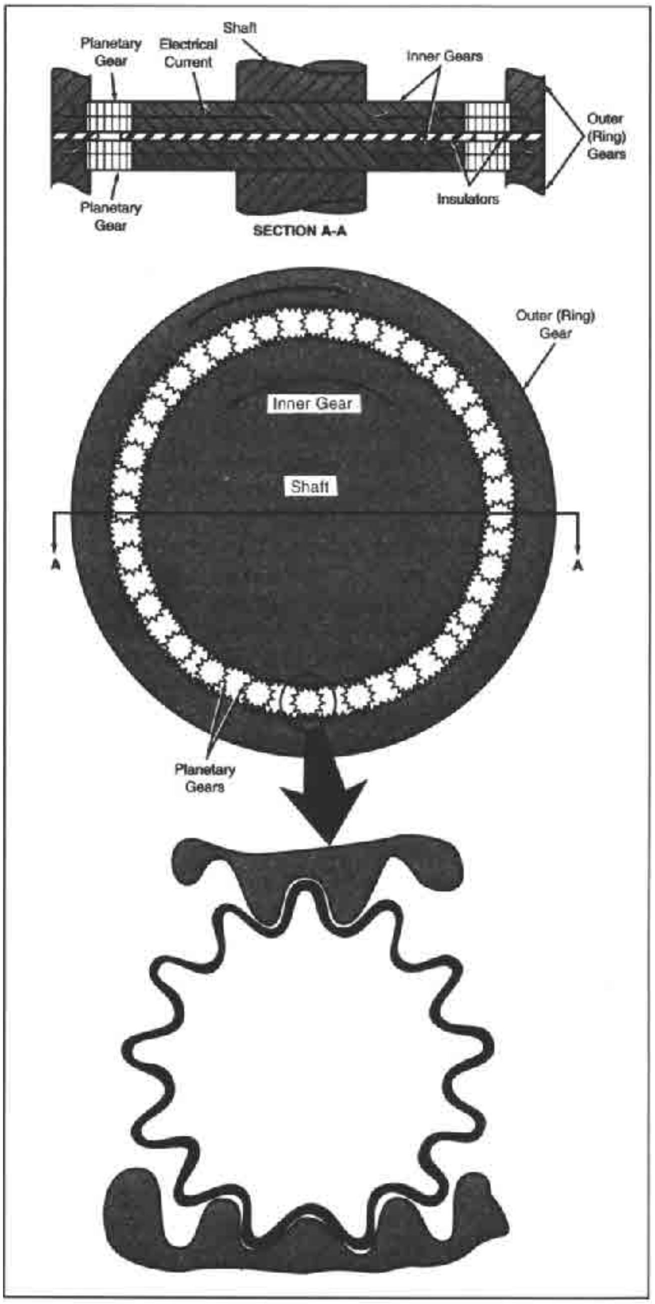
The second electrical subsystem is mounted on, or at least rotates with, the outer (ring) gears. As was done to the inner gears, the ring gears are separated axially by an electrically insulating annular disk. The ring gears act as the electrical terminals through which power is fed from or to the second 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.

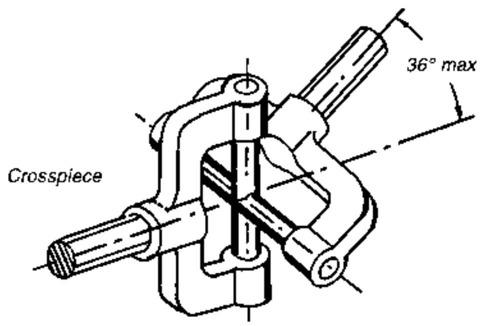


Fig. 1 The Hooke's joint can transmit heavy loads. Anti-friction bearings are a refinement often used.

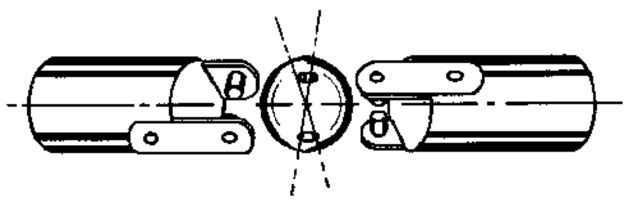


Fig. 2 A pinned sphere shaft coupling replaces a cross-piece. The result is a more compact joint.

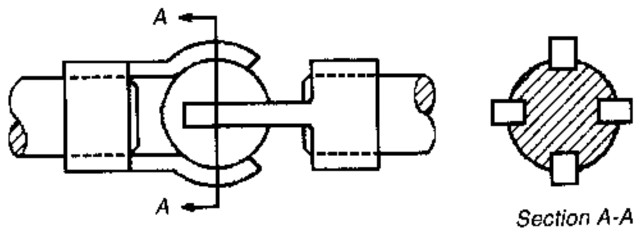


Fig. 3 A grooved-sphere joint is a modification of a pinned sphere. Torques on fastening sleeves are bent over the sphere on the assembly. Greater sliding contact of the torques in grooves makes simple lubrication essential at high torques and alignment angles.

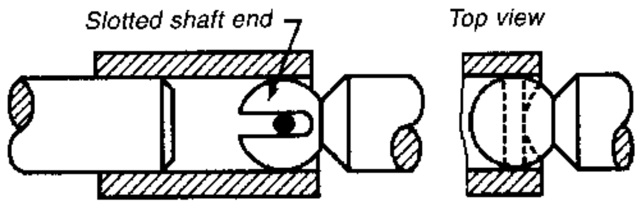


Fig. 4 A pinned-sleeve shaft-coupling is fastened to one shaft that engages the forked, spherical end on the other shaft to provide a joint which also allows for axial shaft movement. In this example, however, the angle between shafts must be small. Also, the joint is only suitable for low torques.

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.

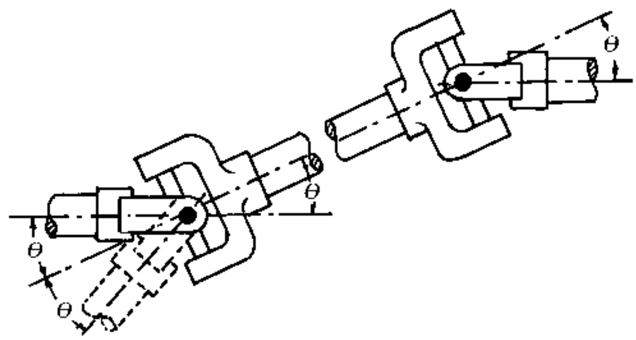


Fig. 5 A constant-velocity joint is made by coupling two Hooke's joints. They must have equal input and output angles to work correctly. Also, the forks must be assembled so that they will always be in the same plane. The shaft-alignment angle can be double that for a single joint.

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.

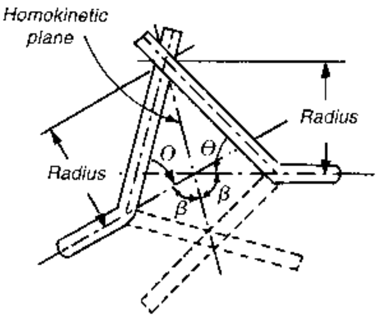


Fig. 6

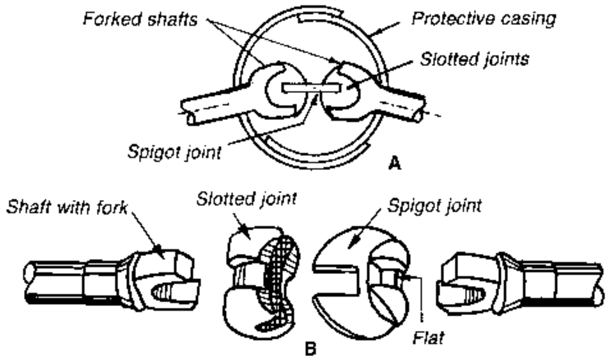


Fig. 7

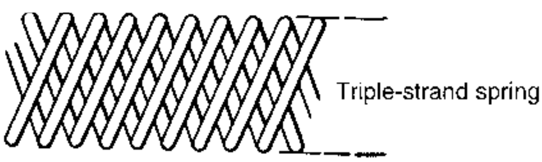


Fig. 8 This flexible shaft permits any shaft angle. These shafts, if long, should be supported to prevent backlash and coiling.

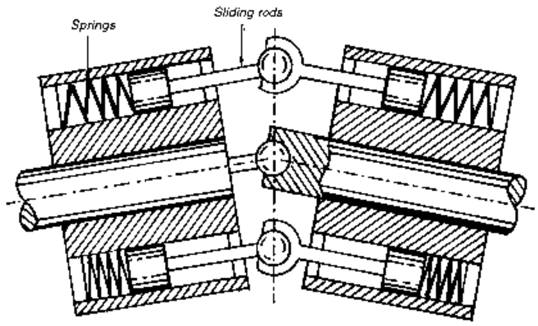


Fig. 9 This pump-type coupling has the reciprocating action of sliding rods that can drive pistons in cylinders.

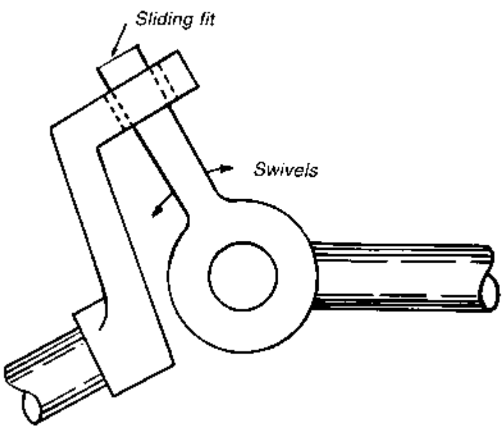


Fig. 10 This light-duty coupling is ideal for many simple, low-cost mechanisms. The sliding swivel-rod must be kept well lubricated at all times.

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.

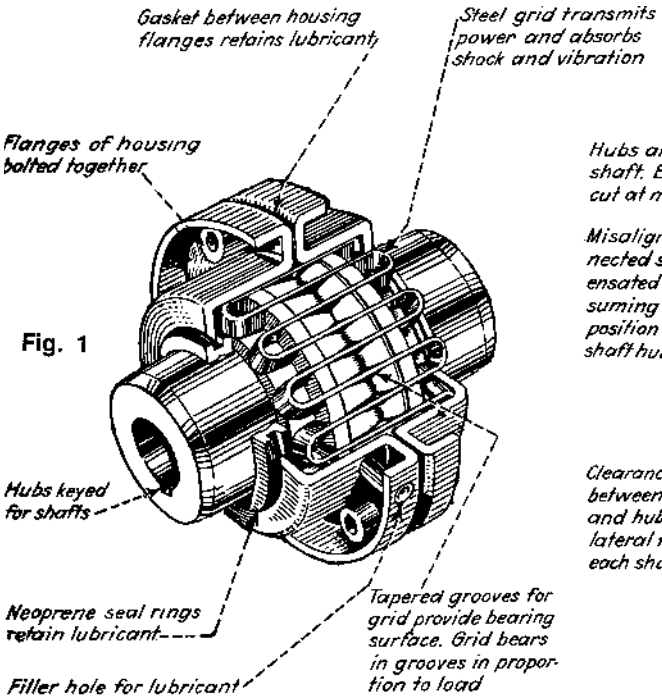


Fig. 1

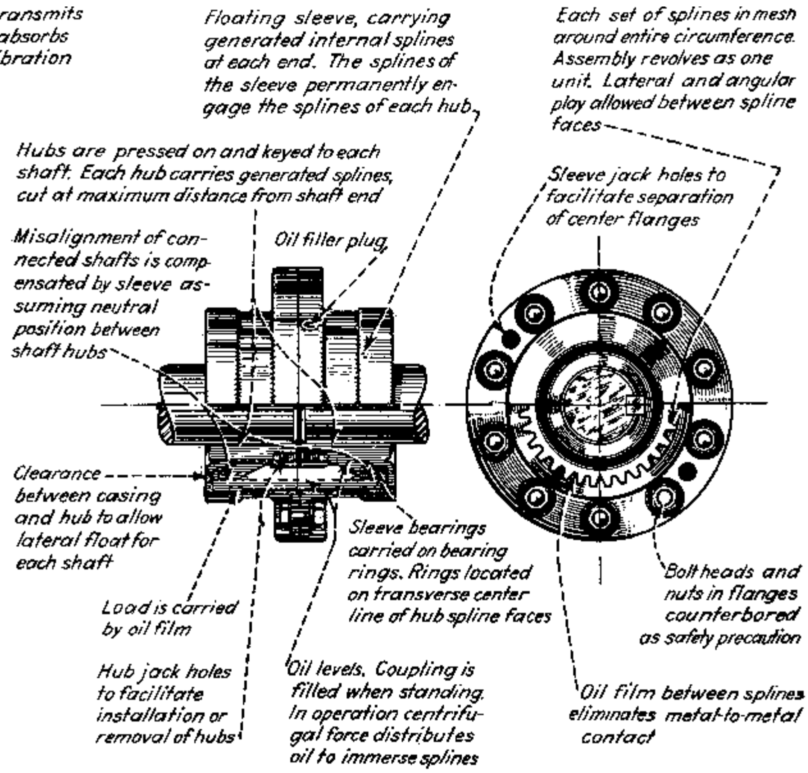


Fig. 2

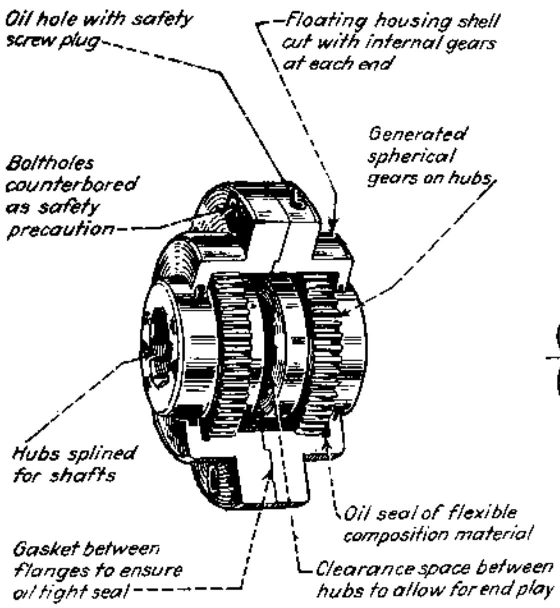


Fig. 3

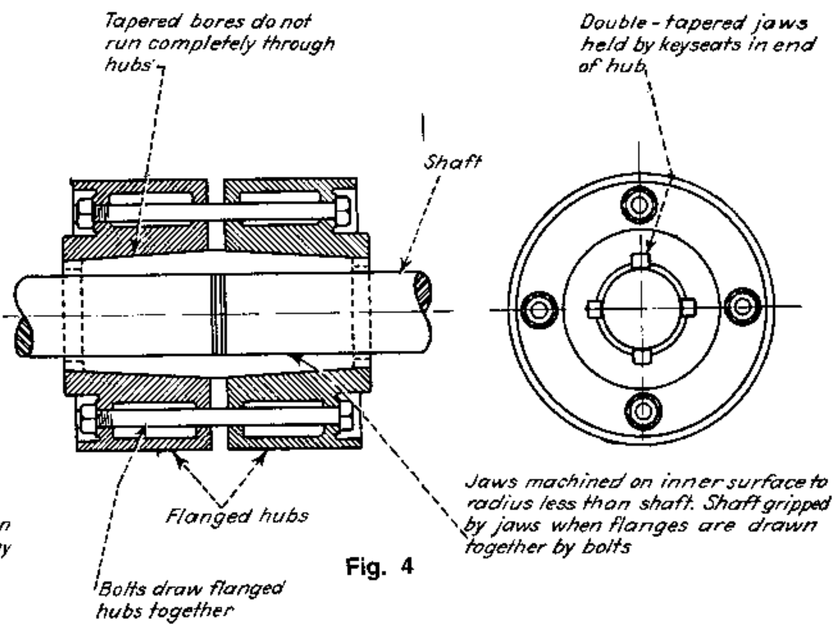


Fig. 4

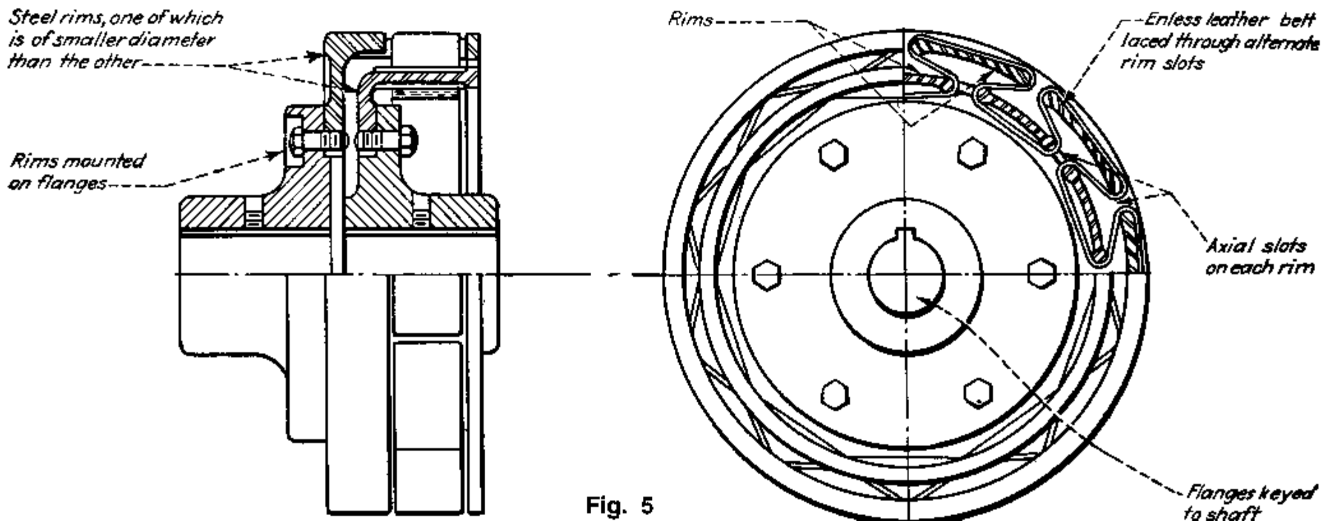


Fig. 5

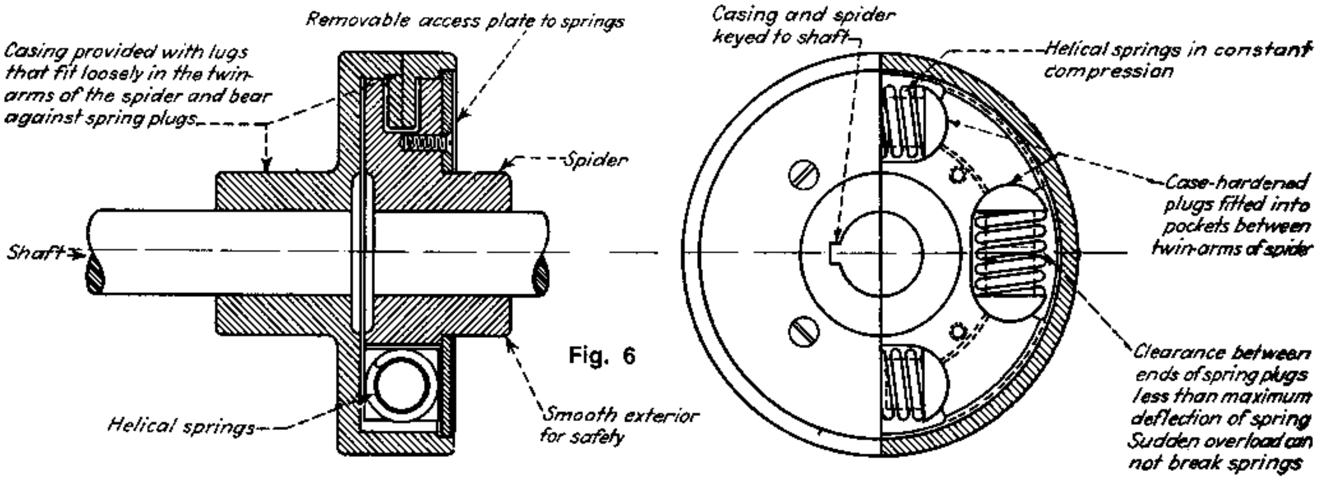


Fig. 6

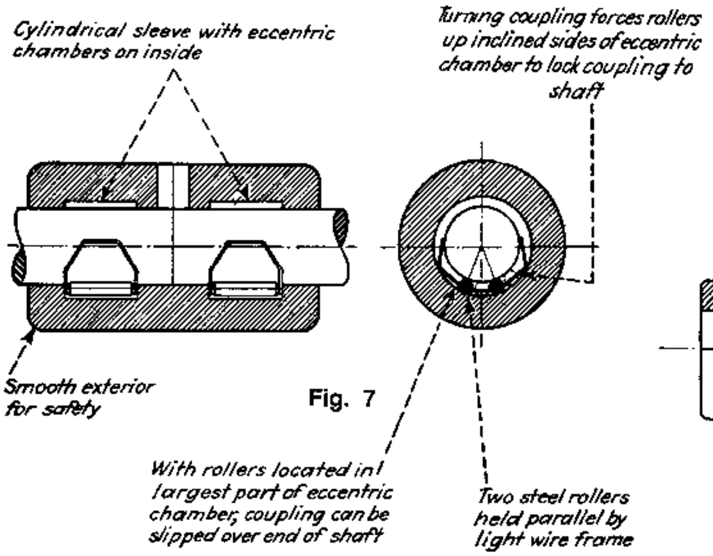


Fig. 7

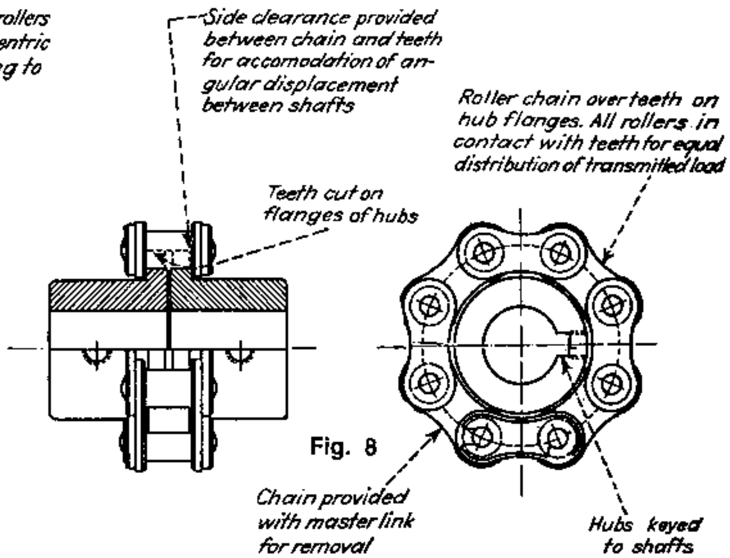


Fig. 8

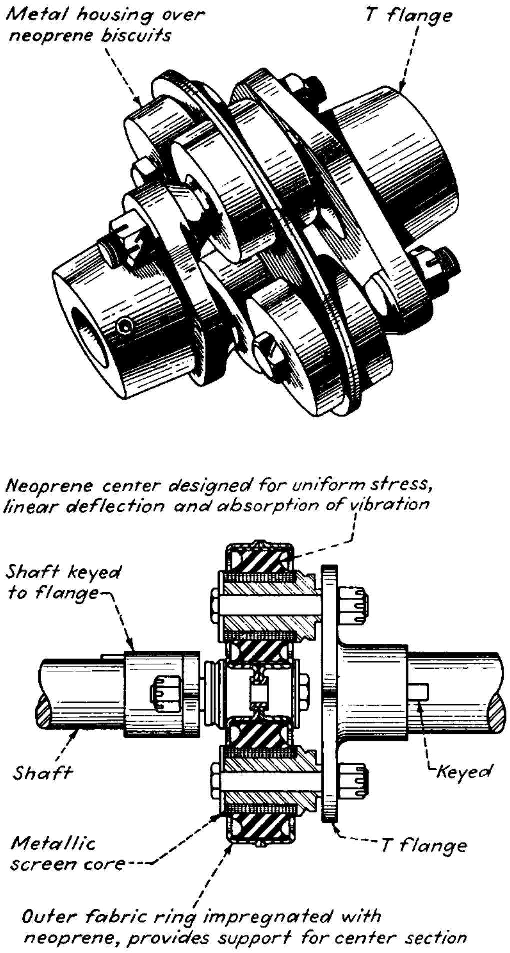


Fig. 9

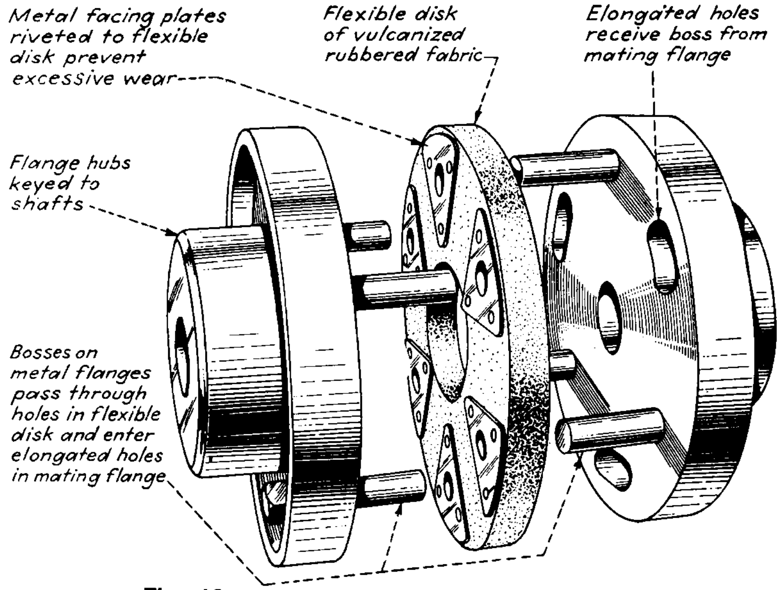


Fig. 10

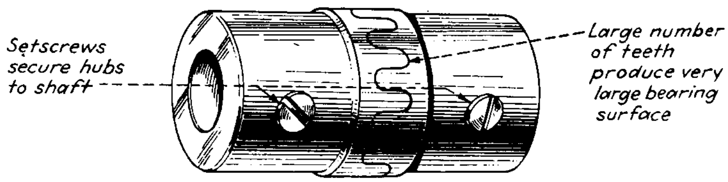


Fig. 11

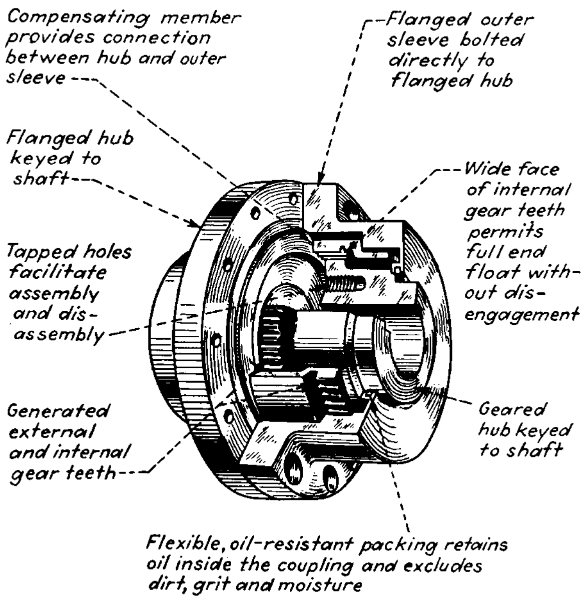


Fig. 12

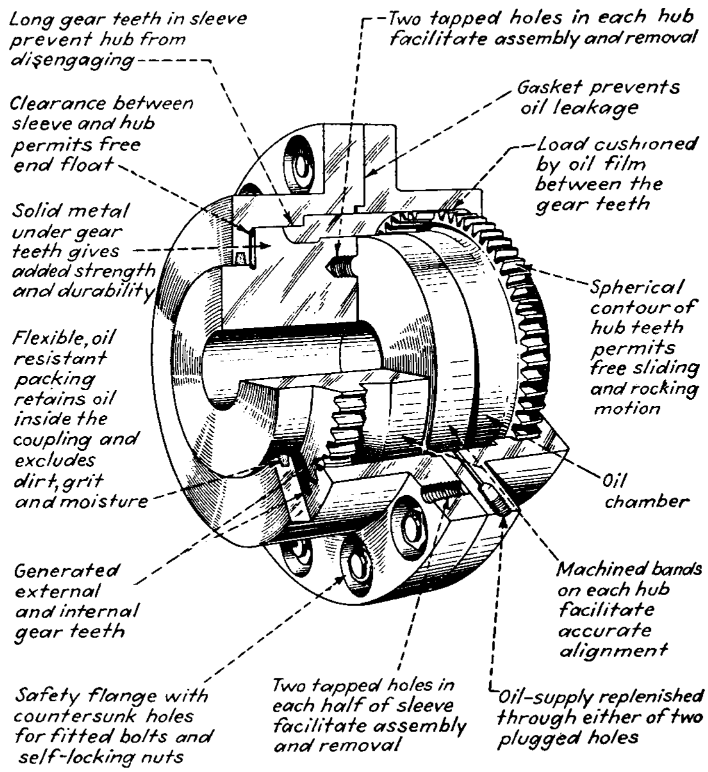


Fig. 13

Shaft couplings that include internal and external gears, balls, pins, and nonmetallic parts to transmit torque are shown here.

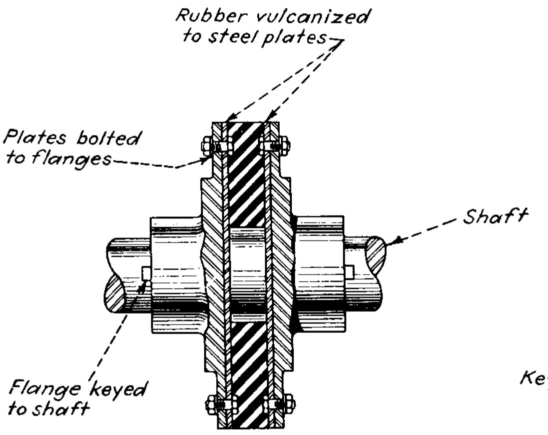


Fig. 14

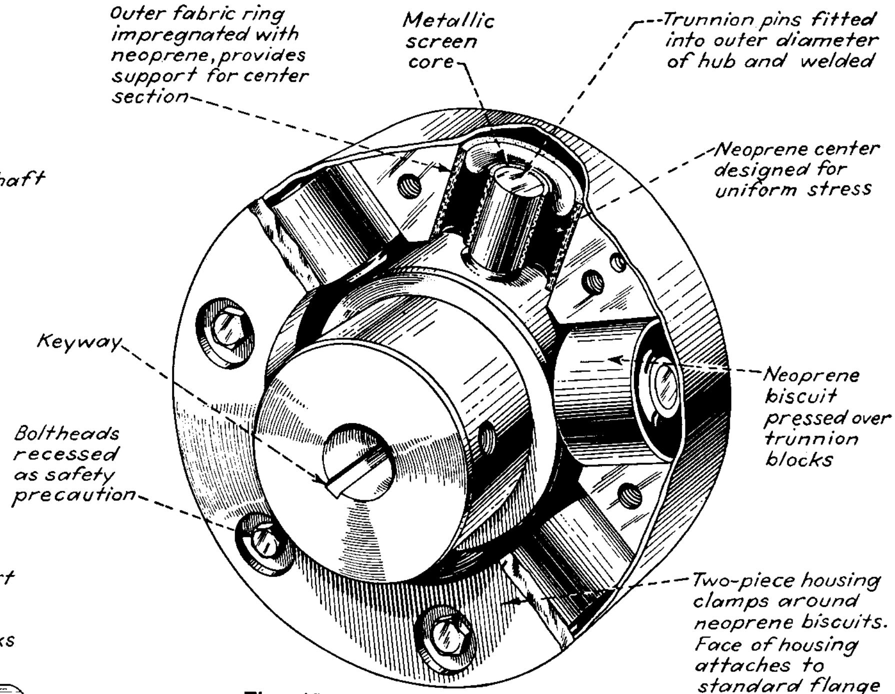


Fig. 15

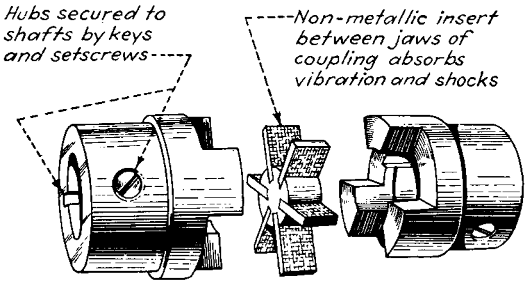


Fig. 16

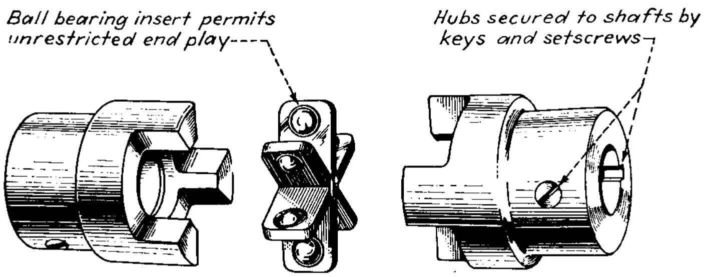


Fig. 17

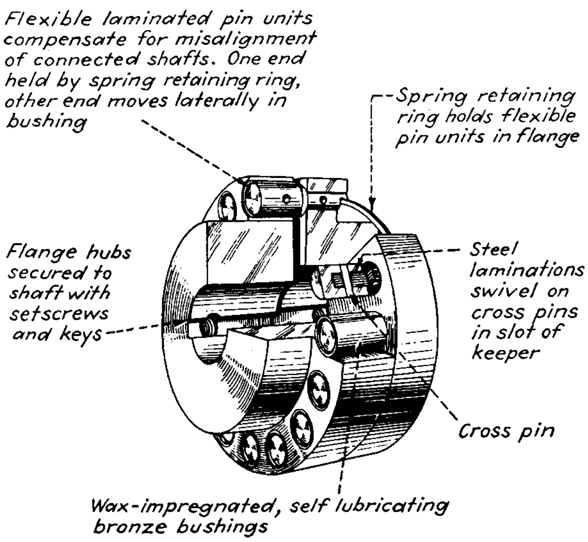


Fig. 18

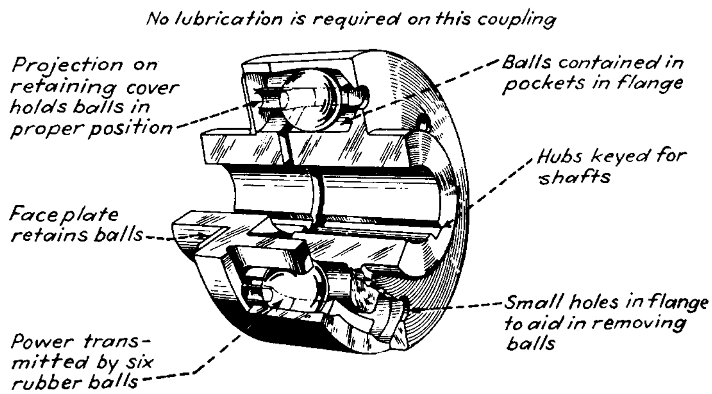


Fig. 19

LINKAGES FOR BAND CLUTCHES AND BRAKES

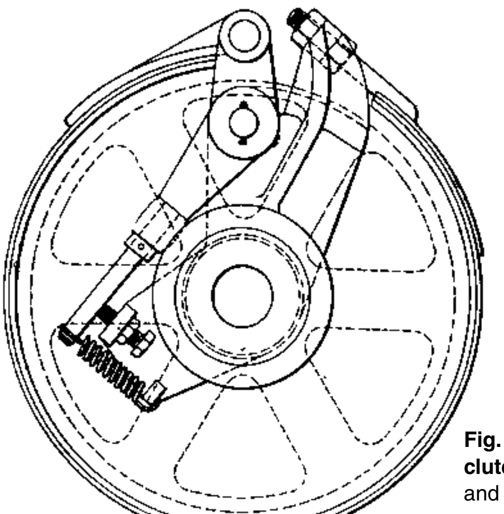


Fig. 1 An outside band clutch operated by a roller and cone.

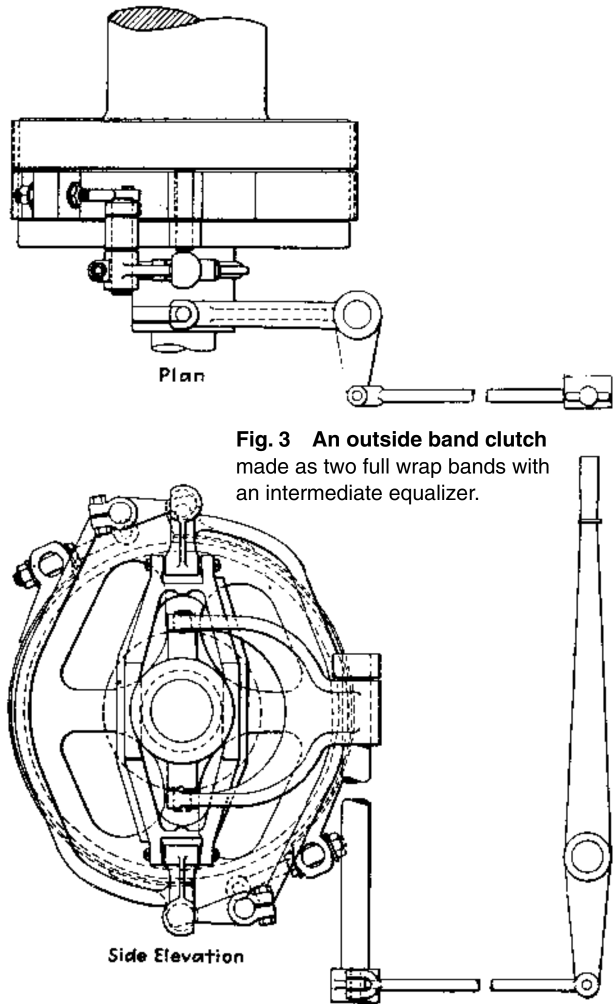


Fig. 3 An outside band clutch made as two full wrap bands with an intermediate equalizer.

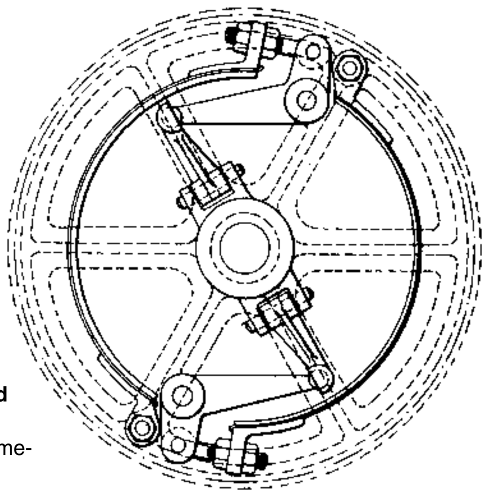


Fig. 2 An outside band clutch made as two half wrap bands with an intermediate equalizer.

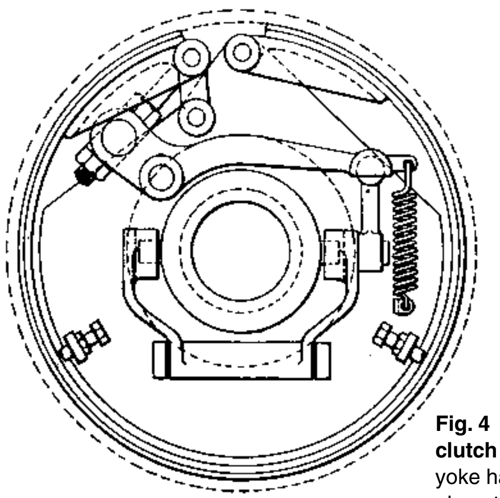


Fig. 4 An inside band clutch operated by a yoke having movement along the shaft.

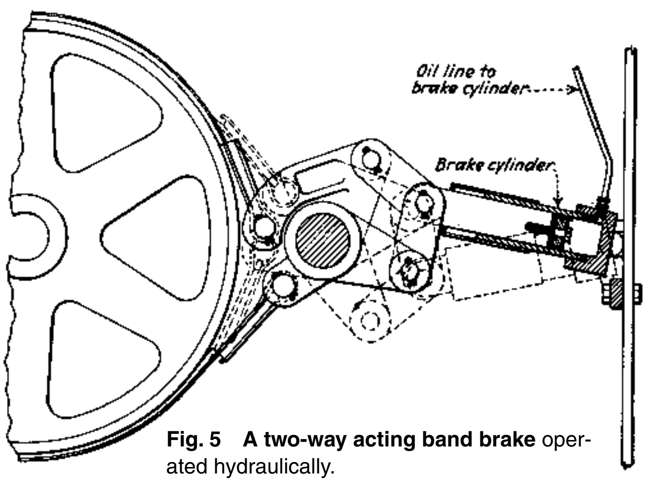
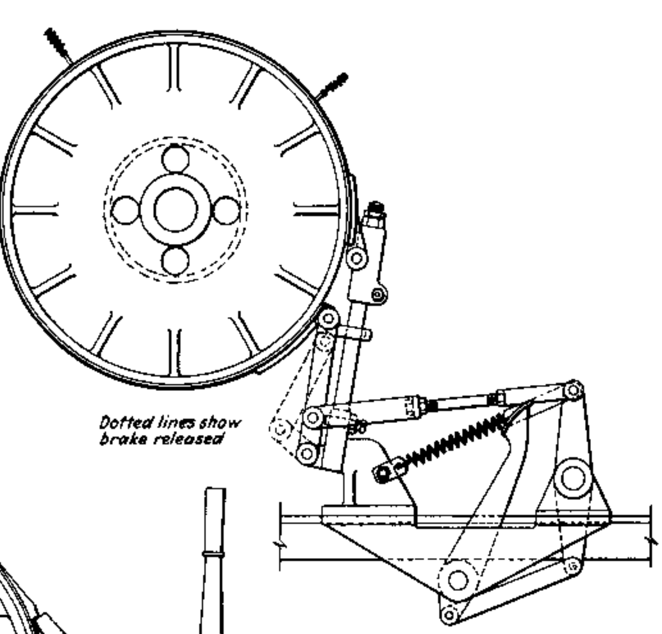


Fig. 5 A two-way acting band brake operated hydraulically.

Fig. 6 **A hoist-drum band brake** operated by a foot pedal.



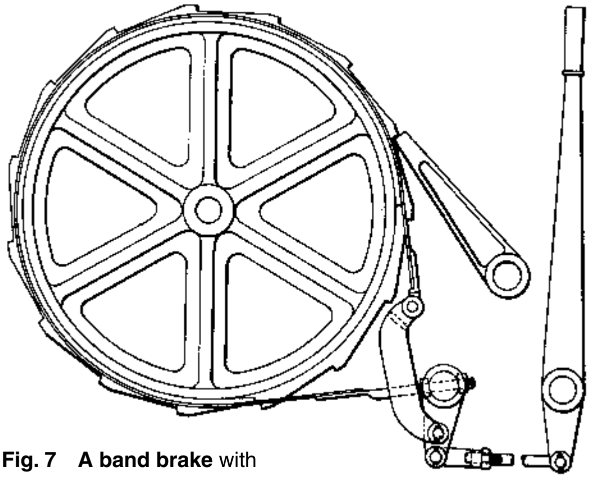


Fig. 7 **A band brake** with a single toggle action.

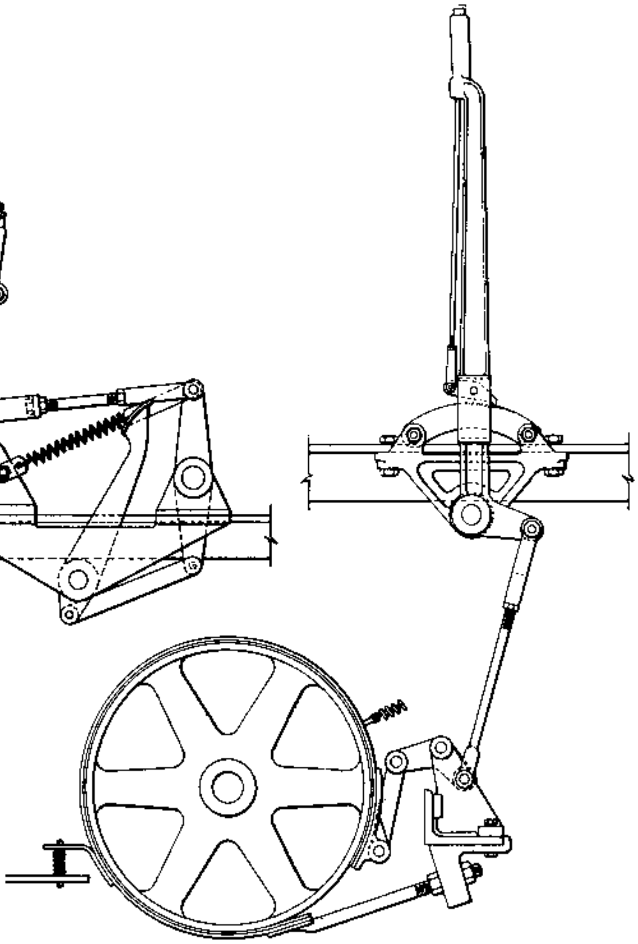


Fig. 8 **A crawler-drive band brake** operated by a ratchet lever.

SPECIAL COUPLING MECHANISMS

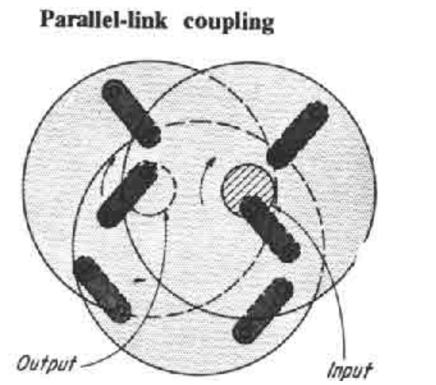


Fig. 1 **Six links and three disks** can synchronize the motion between adjacent, parallel shafts.

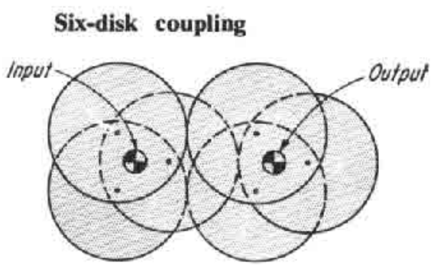


Fig. 2 **An arrangement** to synchronize shafts without the need for links.

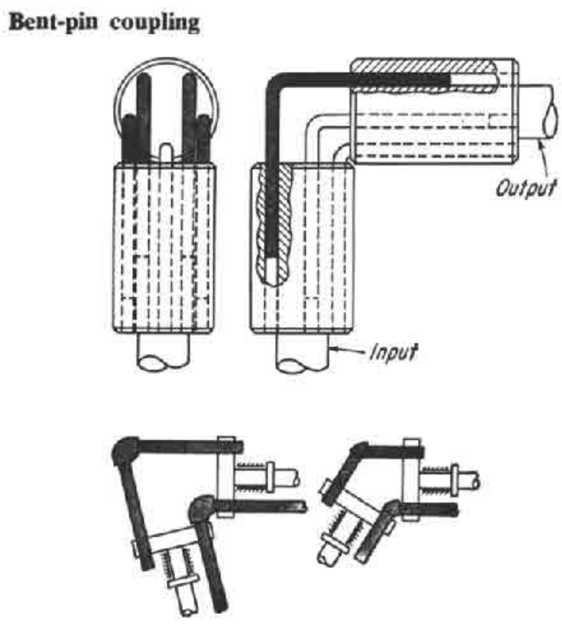


Fig. 3 **As the input rotates,** the five bent pins will move in and out of the drilled holes to impart a constant velocity rotation to the right angle-output shaft. The device can transmit constant velocity at angles other than 90°, as shown.

LINK COUPLING MECHANISMS

Fig. 1 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.

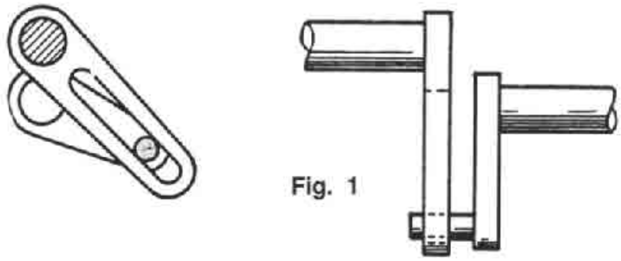


Fig. 1

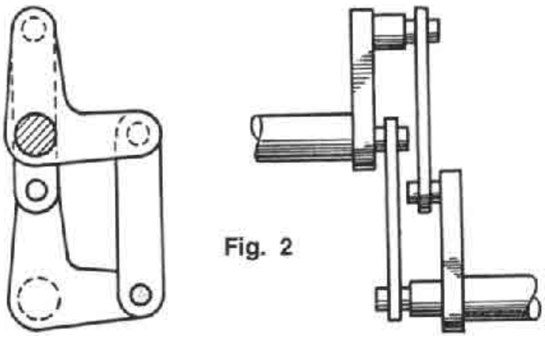


Fig. 2

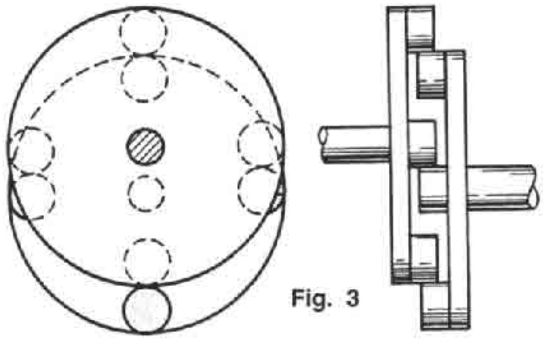


Fig. 3

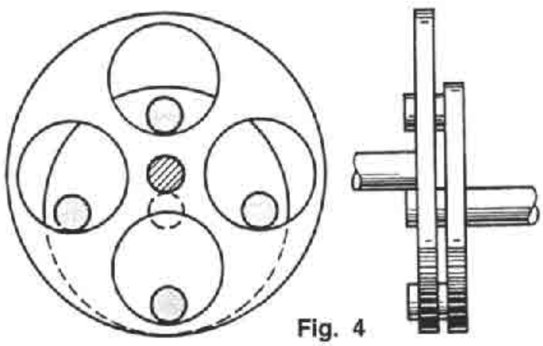


Fig. 4

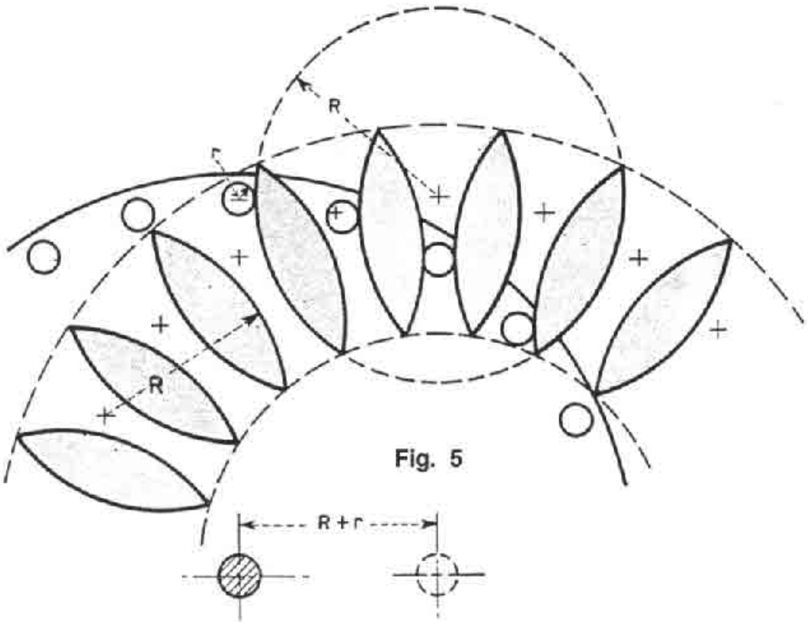


Fig. 5

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.

CHAPTER 10

TORQUE-LIMITING, TENSIONING, AND GOVERNING DEVICES

CALIPER BRAKES HELP MAINTAIN PROPER TENSION IN PRESS FEED

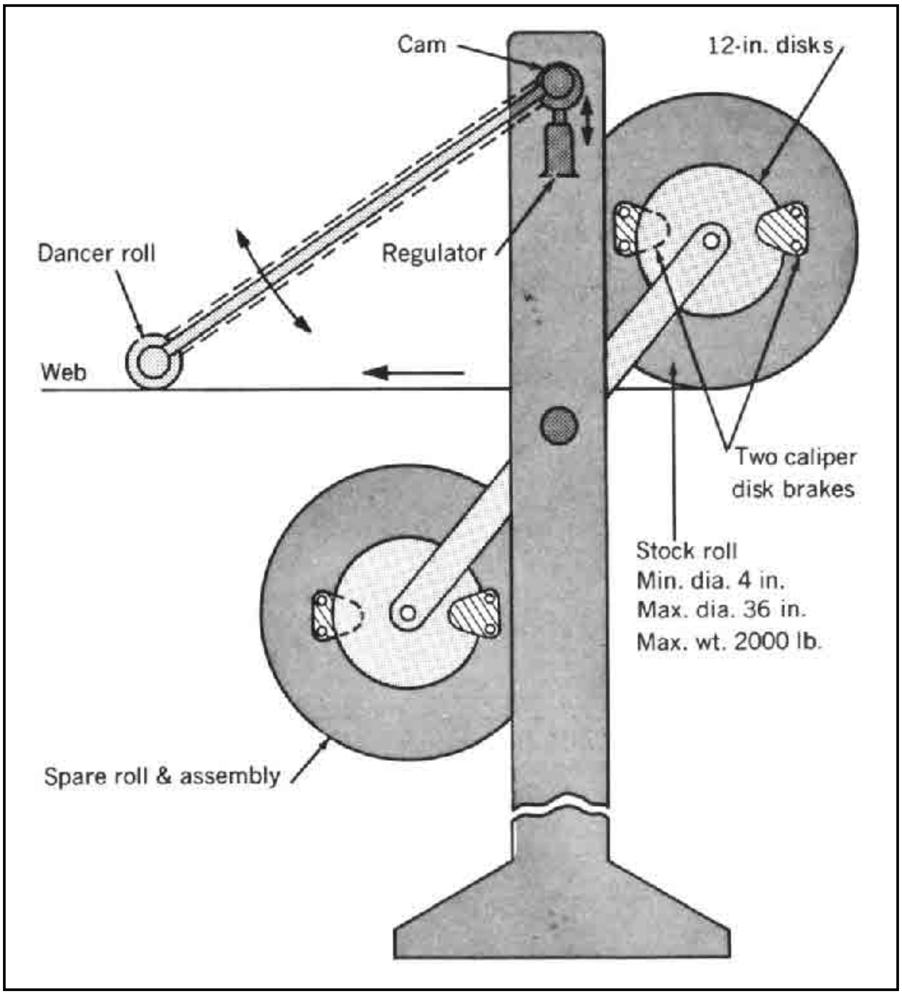
A simple cam-and-linkage arrangement (drawing) works in a team with two caliper disk brakes to provide automatic tension control for paper feeds on a web press.

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.

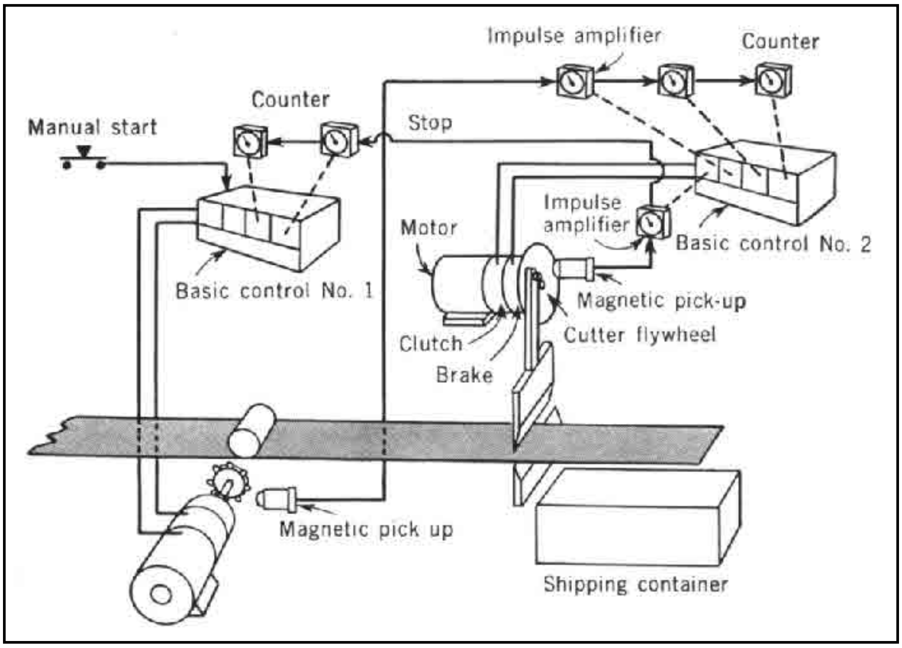
If the web should break or the paper run out on the roll, the dancer roll will allow maximum braking. The press can be stopped in less than one revolution.



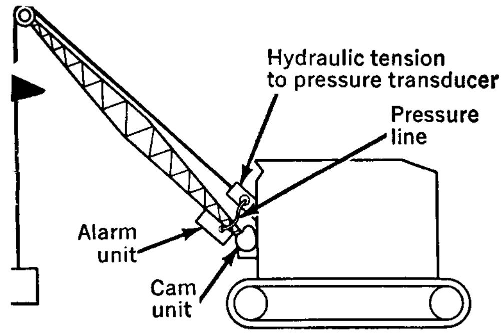
This linkage system works in combination with a regulator and caliper disk brakes to stop a press rapidly from a high speed, if the web should break.

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).

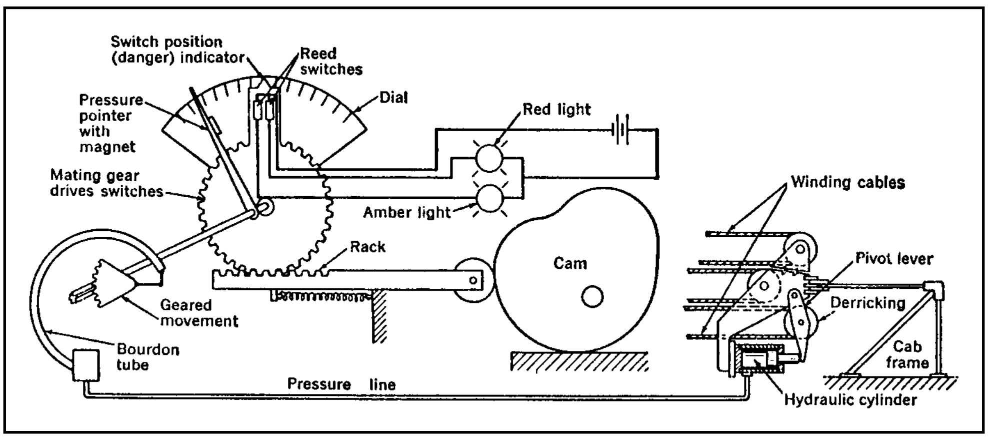
The device takes into consideration the two key factors in the safe loading of a crane boom: the boom angle (low angles create a greater overturning torque than high angles) and the compression load on the boom, which is greatest at high boom angles. Both factors are translated into inputs that are integrated 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.

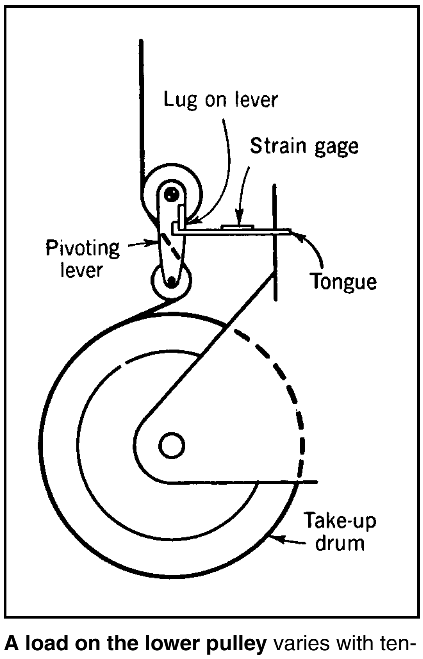
CONSTANT WATCH ON CABLE TENSION

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.

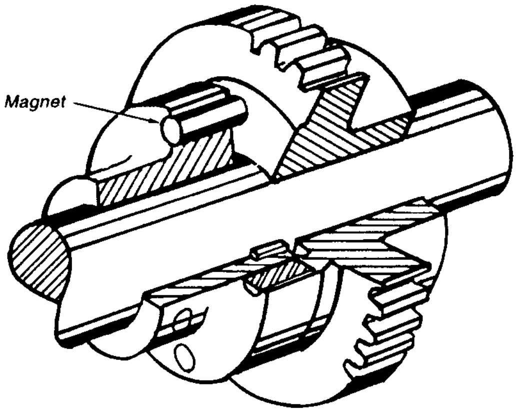


Fig. 1 Permanent magnets transmit torque in accordance with their numbers and size around the circumference of the clutch plate. Control of the drive in place is limited to removing magnets to reduce the drive's torque capacity.

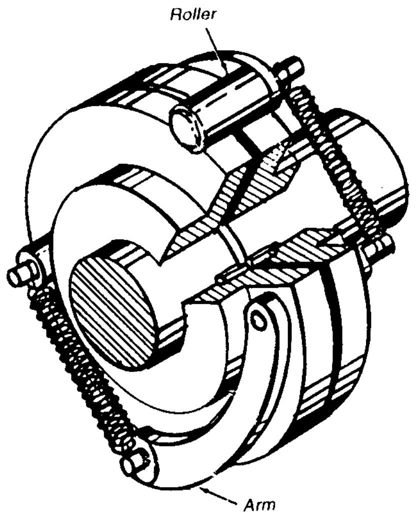


Fig. 2 Arms hold rollers in the slots that are cut across the disks mounted on the ends of butting shafts. Springs keep the roller in the slots, but excessive torque forces them out.

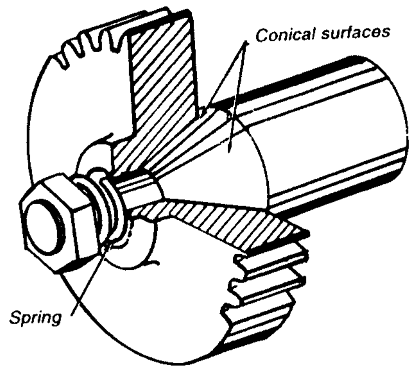


Fig. 3 A cone clutch is formed by mating a taper on the shaft to a beveled central hole in the gear. Increasing compression on the spring by tightening the nut increases the drive's torque capacity.

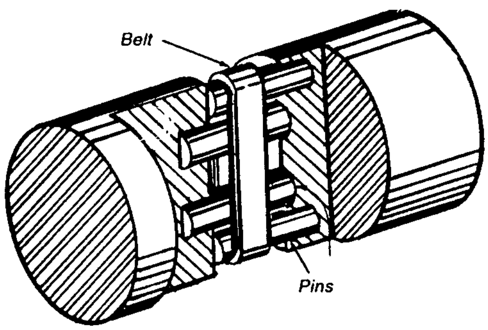


Fig. 4 A flexible belt wrapped around four pins transmits only the lightest loads. The outer pins are smaller than the inner pins to ensure contact.

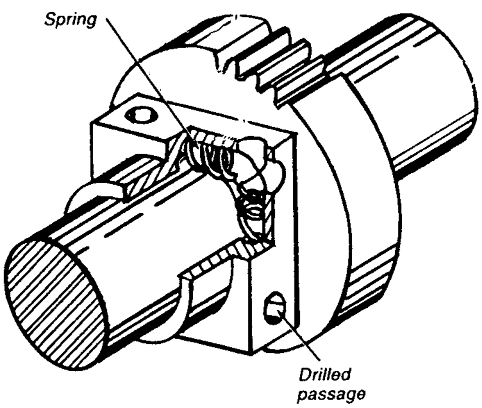


Fig. 5 Springs inside the block grip the shaft because they are distorted when the gear is mounted to the box on the shaft.

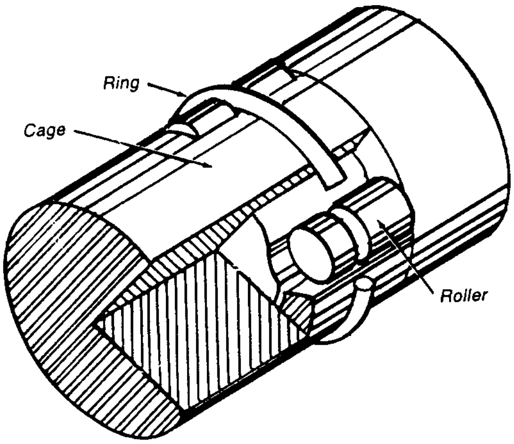


Fig. 6 The ring resists the natural tendency of the rollers to jump out of the grooves in the reduced end of one shaft. The slotted end of the hollow shaft acts as a cage.

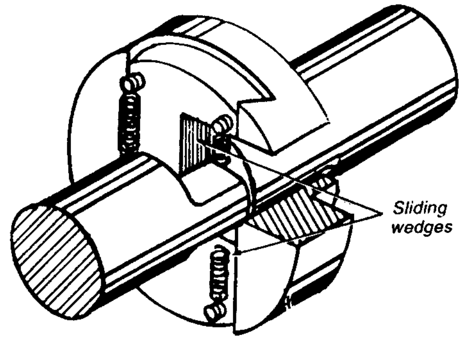


Fig. 7 Sliding wedges clamp down on the flattened end of the shaft. They spread apart when torque becomes excessive. The strength of the springs in tension that hold the wedges together sets the torque limit.

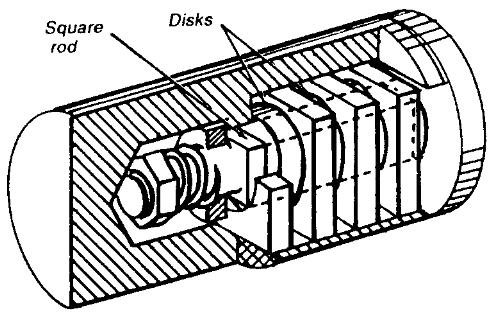


Fig. 8 Friction disks are compressed by an adjustable spring. Square disks lock into the square hole in the left shaft, and round disks lock onto the square rod on the right shaft.

LIMITERS PREVENT OVERLOADING

These 13 "safety valves" give way if machinery jams, thus preventing serious damage.

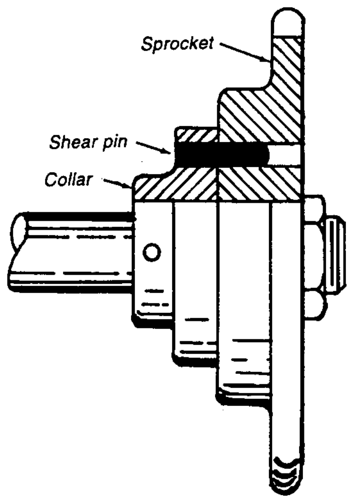


Fig. 1 A shear pin is a simple and reliable torque limiter. However, after an overload, removing the sheared pin stubs and replacing them with a new pin can be time consuming. Be sure that spare shear pins are available in a convenient location.

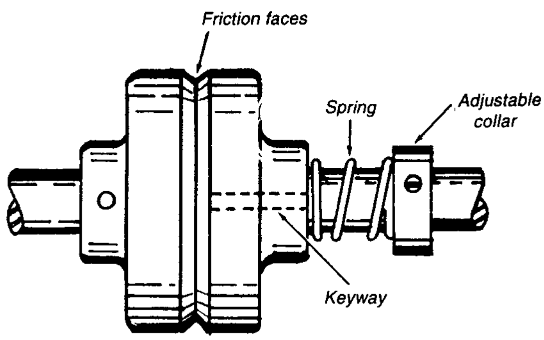


Fig. 2 Friction clutch torque limiter. Adjustable spring tension holds the two friction surfaces together to set the overload limit. As soon as an overload is removed, the clutch reengages. A drawback to this design is that a slipping clutch can destroy itself if it goes undetected.

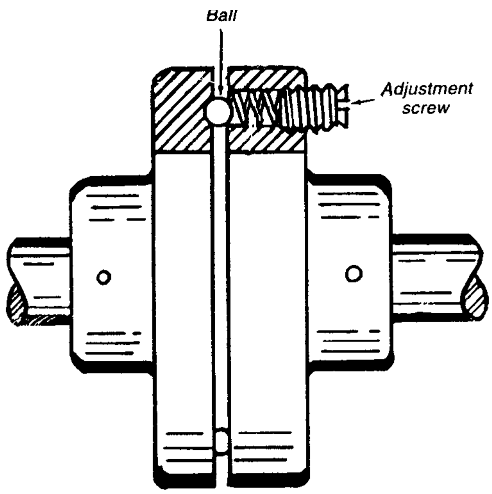


Fig. 3 Mechanical keys. A spring holds a ball in a dimple in the opposite face of this torque limiter until an overload forces it out. Once a slip begins, clutch face wear can be rapid. Thus, this limiter is not recommended for machines where overload is common.

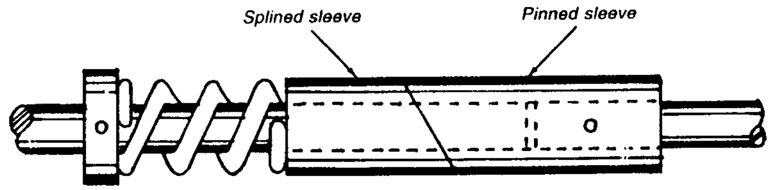


Fig. 4 A cylinder cut at an angle forms a torque limiter. A spring clamps the opposing-angled cylinder faces together, and they separate from angular alignment under overload conditions. The spring tension sets the load limit.

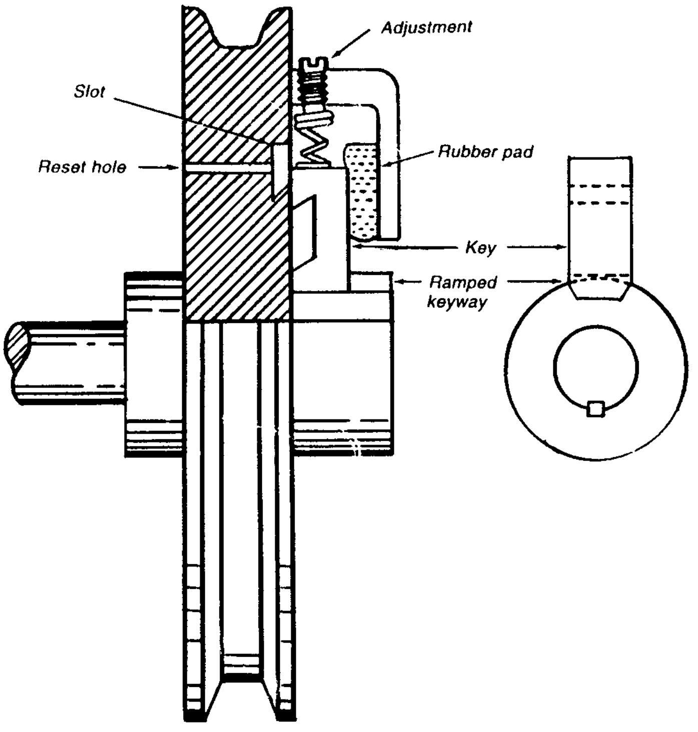


Fig. 5 A retracting key limits the torque in this clutch. The ramped sides of the keyway force the key outward against an adjustable spring. As the key moves outward, a rubber pad or another spring forces the key into a slot in the sheave. This holds the key out of engagement and prevents wear. To reset the mechanism, the key is pushed out of the slot with a tool in the reset hole of the sheave.

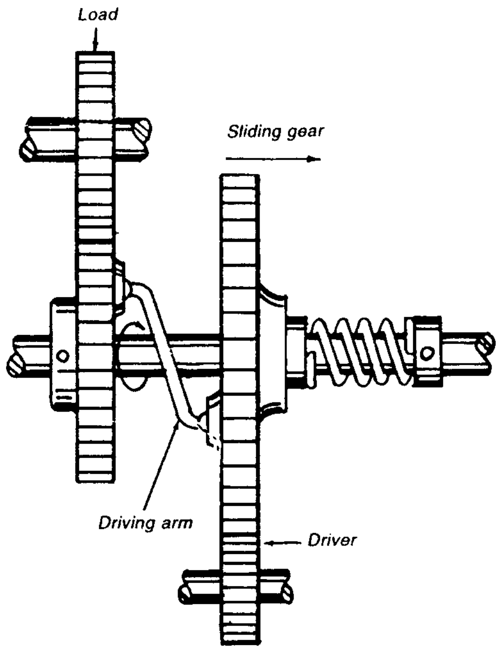


Fig. 6 Disengaging gears. The axial forces of a spring and driving arm are in balance in this torque limiter. An overload condition overcomes the force of the spring to slide the gears out of engagement. After the overload condition is removed, the gears must be held apart to prevent them from being stripped. With the driver off, the gears can safely be reset.

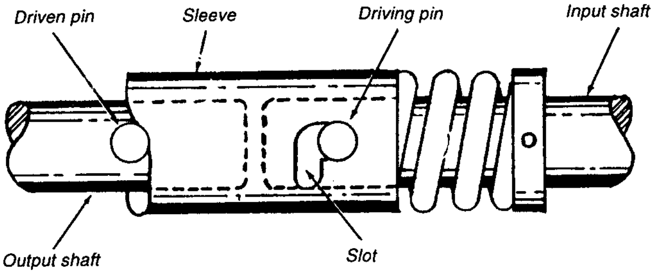


Fig. 7 A cammed sleeve connects the input and output shafts of this torque limiter. A driven pin pushes the sleeve to the right against the spring. When an overload occurs, the driving pin drops into the slot to keep the shaft disengaged. The limiter is reset by turning the output shaft backwards.

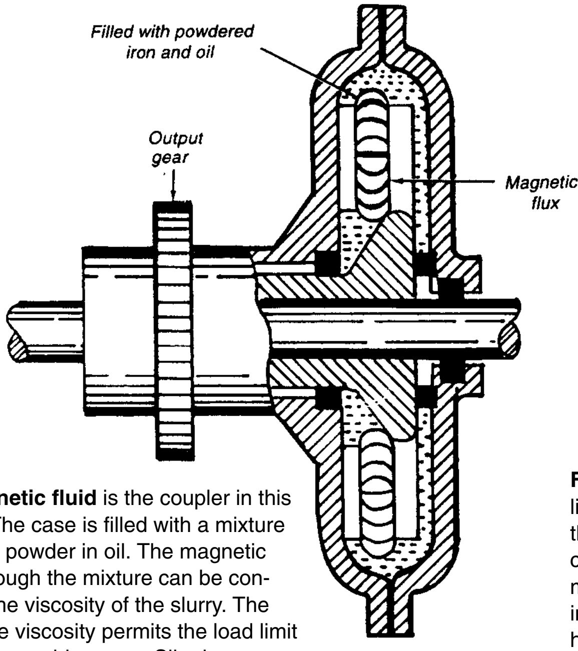


Fig. 8 A magnetic fluid is the coupler in this torque limiter. The case is filled with a mixture of iron or nickel powder in oil. The magnetic flux passed through the mixture can be controlled to vary the viscosity of the slurry. The ability to change viscosity permits the load limit to be varied over a wide range. Slip rings carry electric current to the vanes to create the magnetic field.

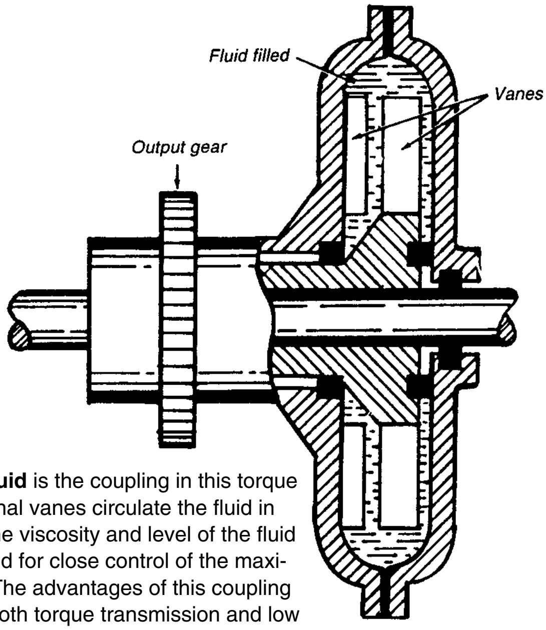


Fig. 9 A fluid is the coupling in this torque limiter. Internal vanes circulate the fluid in the case. The viscosity and level of the fluid can be varied for close control of the maximum load. The advantages of this coupling include smooth torque transmission and low heat rise during slip.

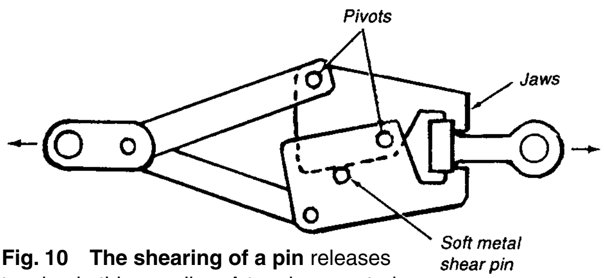


Fig. 10 The shearing of a pin releases tension in this coupling. A toggle-operated blade shears a soft pin so that the jaws open and release an excessive load. In an alternative design, a spring that keeps the jaws from spreading replaces the shear pin.

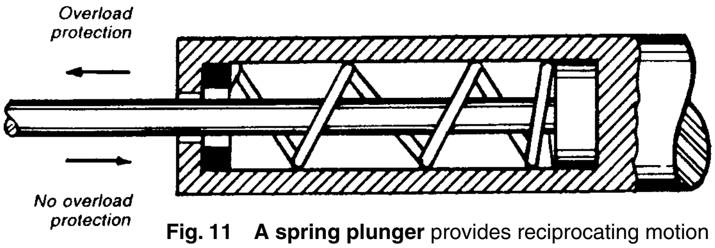


Fig. 11 A spring plunger provides reciprocating motion in this coupling. Overload can occur only when the rod is moving to the left. The spring is compressed under an overload condition.

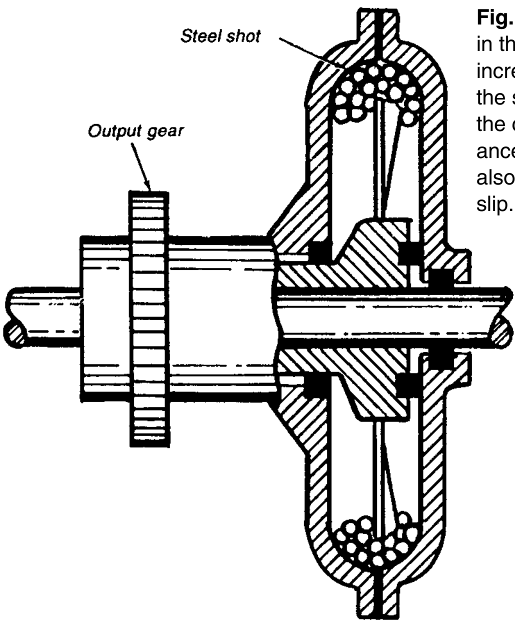


Fig. 12 Steel shot transmits more torque in this coupling as input shaft speed is increased. Centrifugal force compresses the steel shot against the outer surfaces of the case, increasing the coupling's resistance to slip. The addition of more steel shot also increases the coupling's resistance to slip.

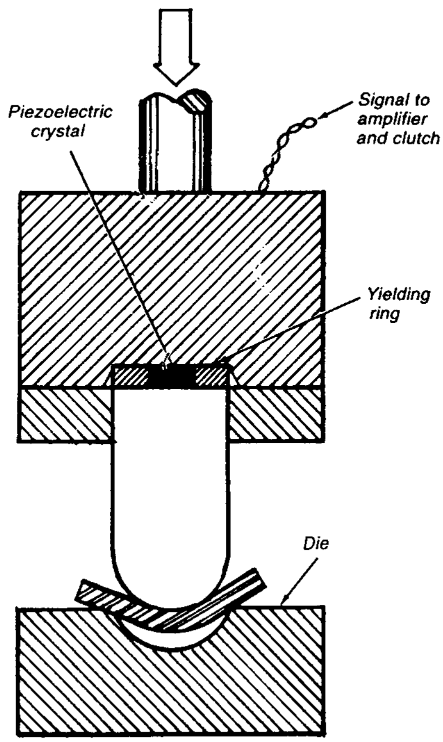


Fig. 13 A piezoelectric crystal produces an electric signal that varies with pressure in this metal-forming press. When the amplified output of the piezoelectric crystal reaches a present value corresponding to the pressure limit, the electric clutch disengages. A yielding ring controls the compression 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.

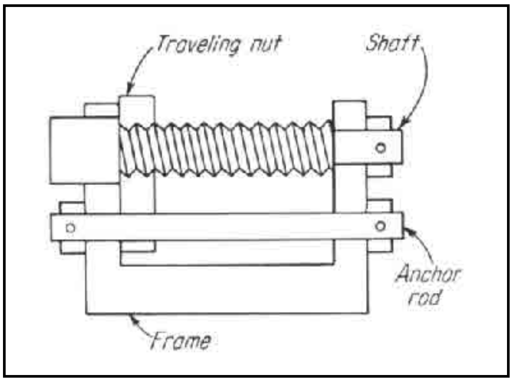


Fig. 1 A traveling nut moves along the threaded shaft until the frame prevents further rotation. This is a simple device, but the traveling nut can jam so tightly that a large torque is required to move the shaft from its stopped position. This fault is overcome at the expense of increased device length by providing a stop pin in the traveling nut.

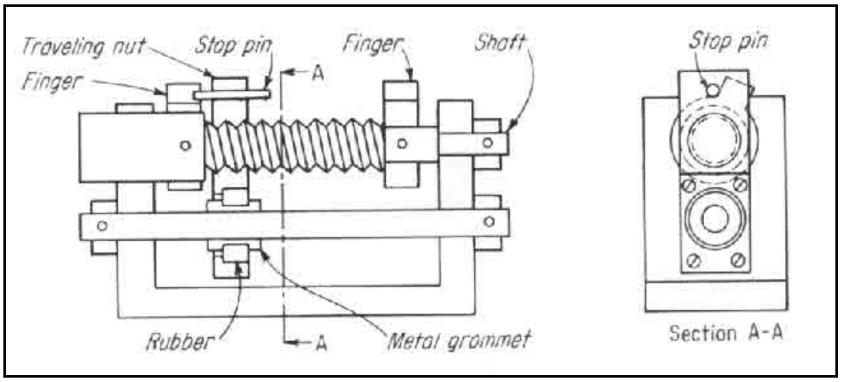
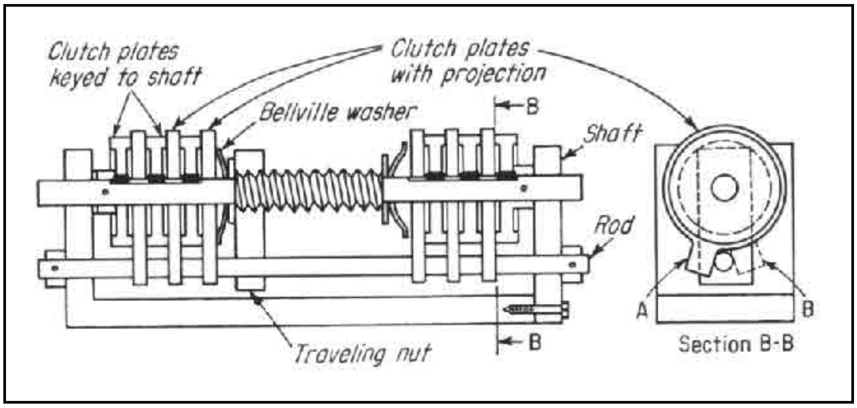


Fig. 2 The engagement between the pin and the rotating finger must be shorter than the thread pitch so the pin can clear the finger on the first reverse-turn. The rubber ring and grommet lessen the impact and provide a sliding surface. The grommet can be oil-impregnated metal.

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.



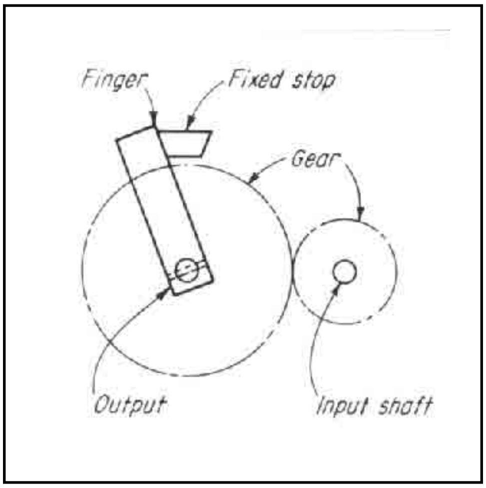


Fig. 4 A shaft finger on the output shaft hits the resilient stop after making less than one revolution. The force on the stop depends upon the gear ratio. The device is, therefore, limited to low ratios and few turns, unless a worm-gear setup is used.

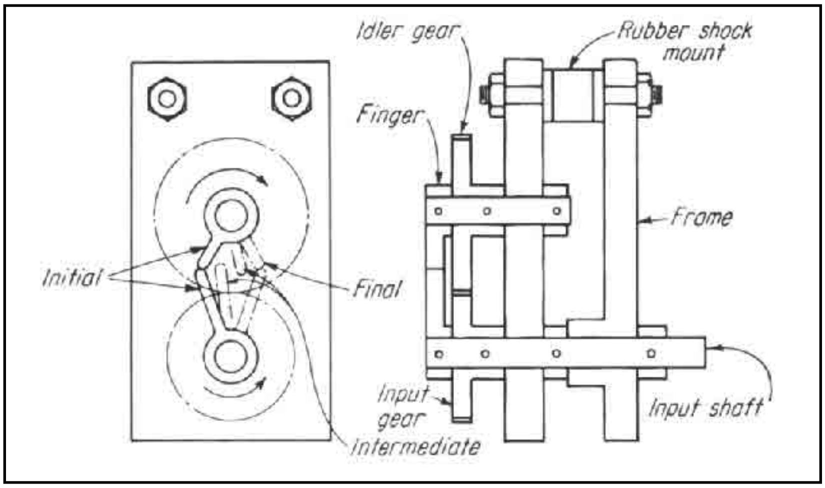


Fig. 5 Two fingers butt together at the initial and final positions to prevent rotation beyond these limits. A rubber shock-mount absorbs the impact load. A gear ratio of almost 1:1 ensures that the fingers will be out-of-phase with one another until they meet on the final turn. Example: Gears with 30 to 32 teeth limit shaft rotation to 25 turns. Space is saved here, but these gears are expensive.

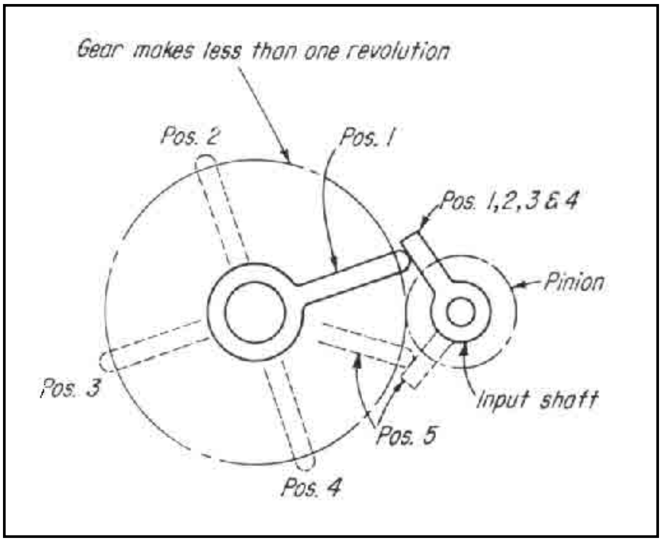


Fig. 6 A large gear ratio limits the idler gear to less than one turn. Stop fingers can be added to the existing gears in a train, making this design the simplest of all. The input gear, however, is limited to maximum of about five turns.

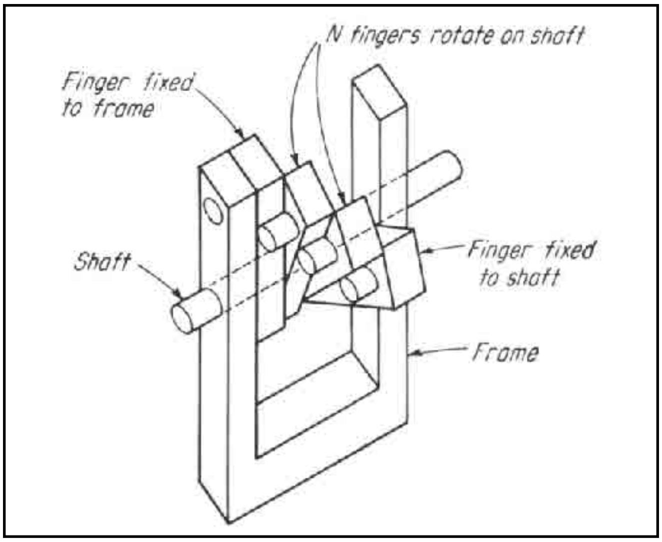
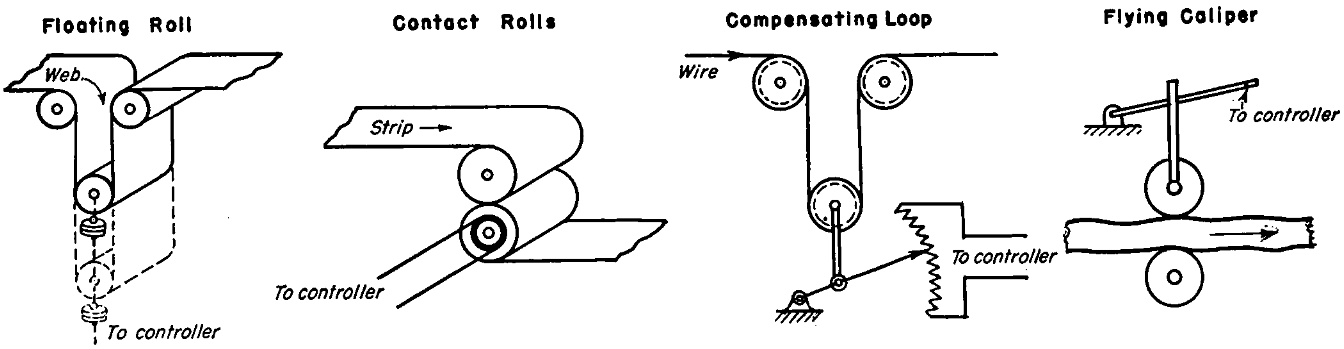


Fig. 7 Pinned fingers limit shaft turns to approximately N + 1 revolutions in either direction. Resilient pin-bushings would help reduce the impact force.

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



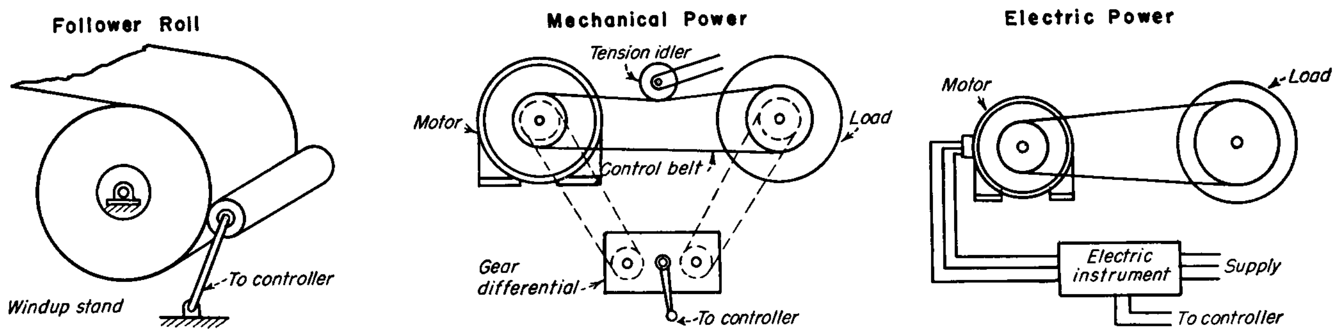
Use: constant tension winding; registry control; section synchronizing.

Use: velocity control; cutter feed control.

Use: line-tension control; section synchronizing.

Use: thickness control; diameter control.

FIG. 2—SECONDARY INDICATORS

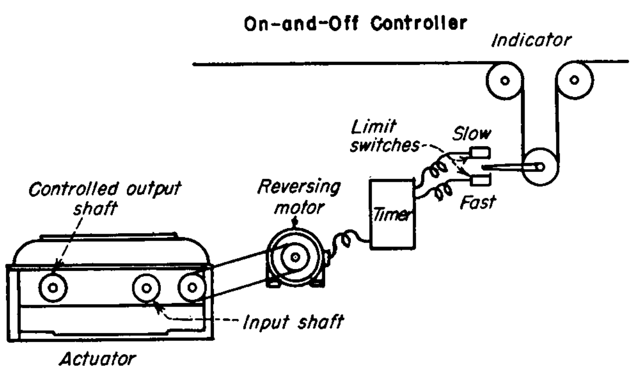


Use: winder control; periphery cutting:

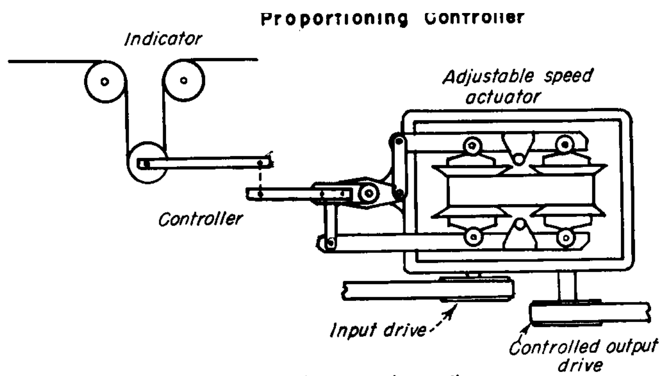
Use: winding tension; feeder control.

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.

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.

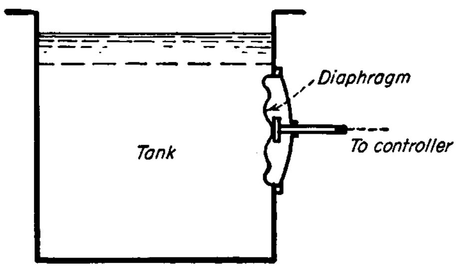
The controlling devices, which are operated by the indicators, determine the degree of speed change required to correct the error, the rate at which the correction must be made, and the stopping point of the control action after the error has been corrected. The manner in which the corrective action 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.

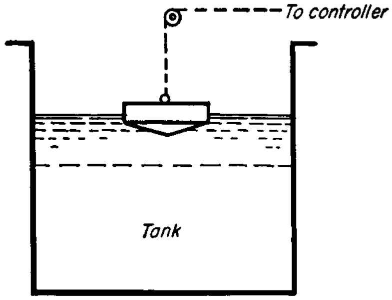
The proportioning controller corrects an error in the system, as shown by the indicator, by continuously adjusting the actuator to a speed that is in exact proportion to the displacement of the indicator. The diagram in Fig. 3 shows the proportioning controller in its simplest form as a direct link connection between the indicator and the actuating drive. However, the force amplification between the indicator and the drive is rel-

Fluid Pressure



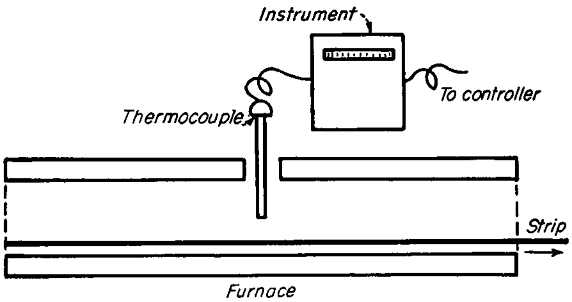
Use: fluid level control; constant pressure control; filtering rate control.

Liquid Level



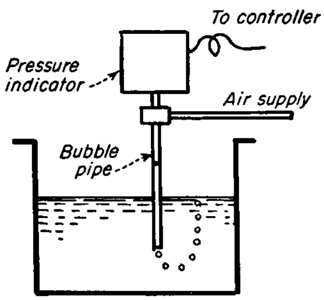
Use: Pumping rate control; system pressure control.

Temperature



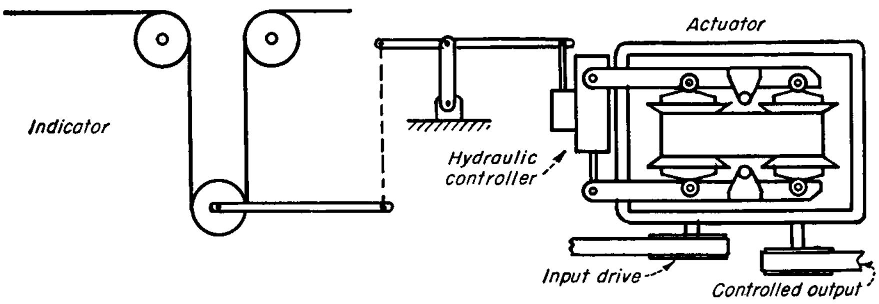
Use: annealing; drum dryers; kilns.

Pressure



Use: fluid density; feeding rate; flow rate.

Proportioning-Throttling Controller



Use: constant tension winding; registry control; exact section synchronizing.

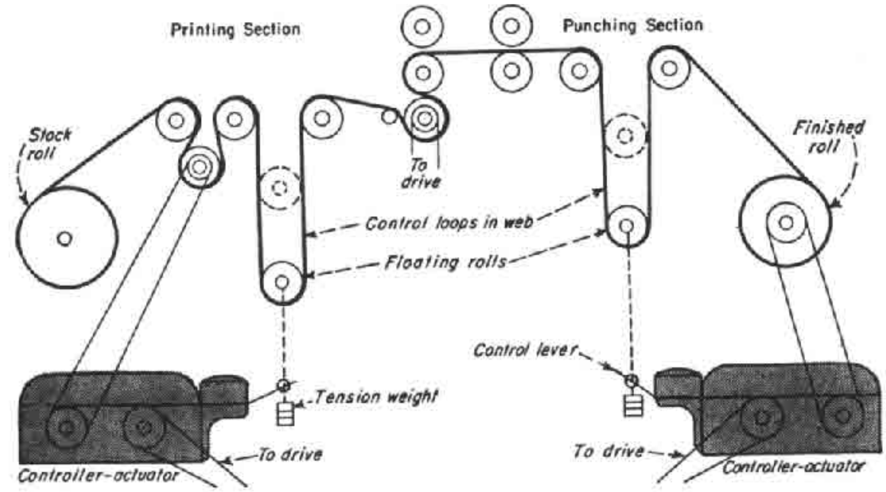


Fig 4 Floating rolls are direct indicators of speed and tension in the paper web. Controller-actuators adjust feed and windup rolls to maintain registry during printing.

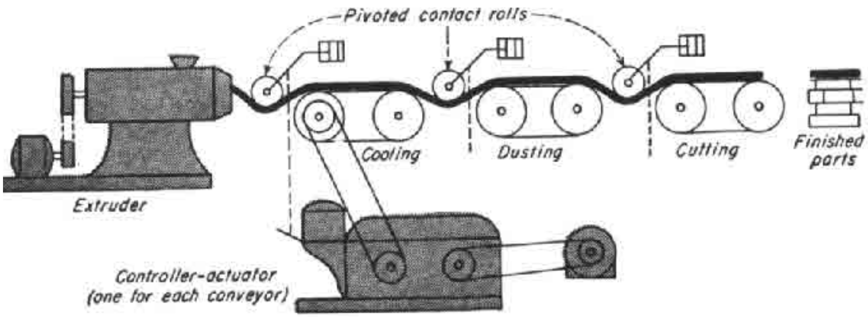


Fig. 5 Dimension control of extruded materials calls for primary indicators like the contact rolls shown. Their movements actuate conveyor control mechanisms.

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 $\frac{1}{32}$ 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.

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.

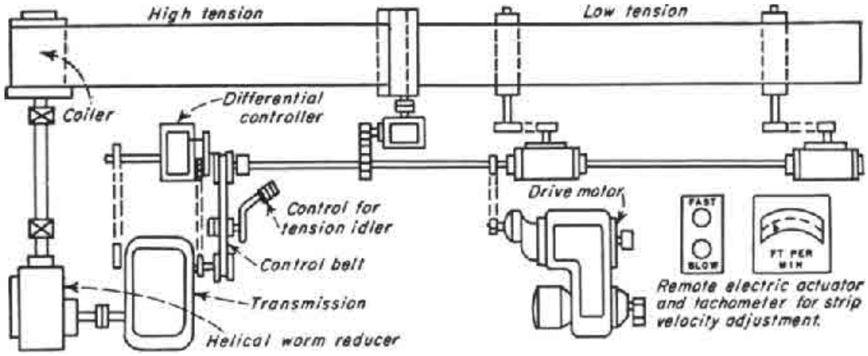


Fig. 6 The differential controller has a third shaft that signals the remote actuator when tension in sheet material changes. Coiler power is a secondary-control index.

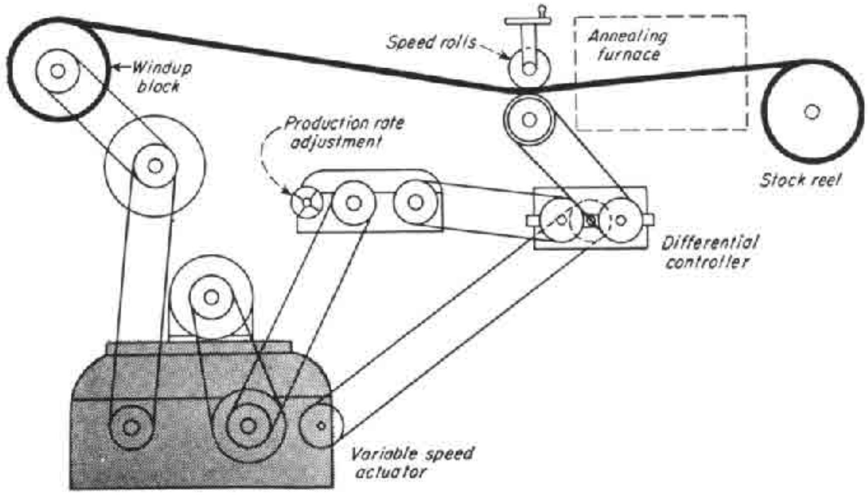


Fig. 7 The movement of wire through the annealing furnace is regulated at constant velocity by continuously retarding the speed of the windup reels to allow for wire build-up.

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 wind-up 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

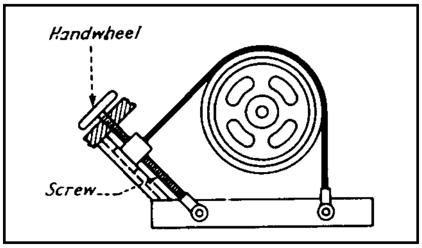
Mechanical, electrical, and hydraulic methods for obtaining controlled tension on winding reels and similar drives, or for driving independent parts of a machine in synchronism.

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.

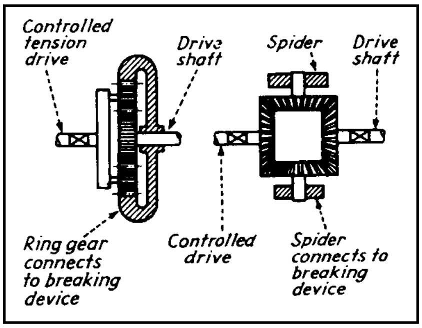
Capacity is limited by the heat radiating capacity of the brake at the maximum permissible running temperature.



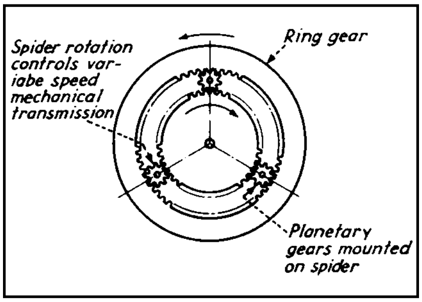
Differential drives can take many different forms, e.g., epicycle spur gears, bevel gear differentials, or worm gear differentials.

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.



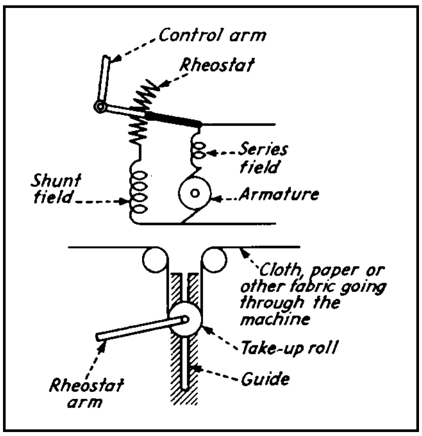
Differential gearing can be used to control a variable-speed transmission. If the ring gear and sun gear are to be driven in opposite directions from their respective shafts which are to be held in synchronism, the gear train can be designed so that the spider on which the planetary gears are mounted 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.



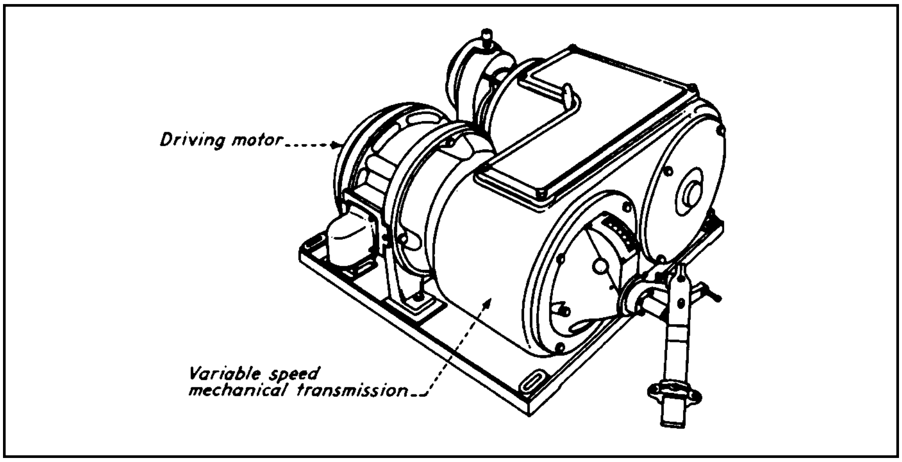
ELECTRICAL DRIVES

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.



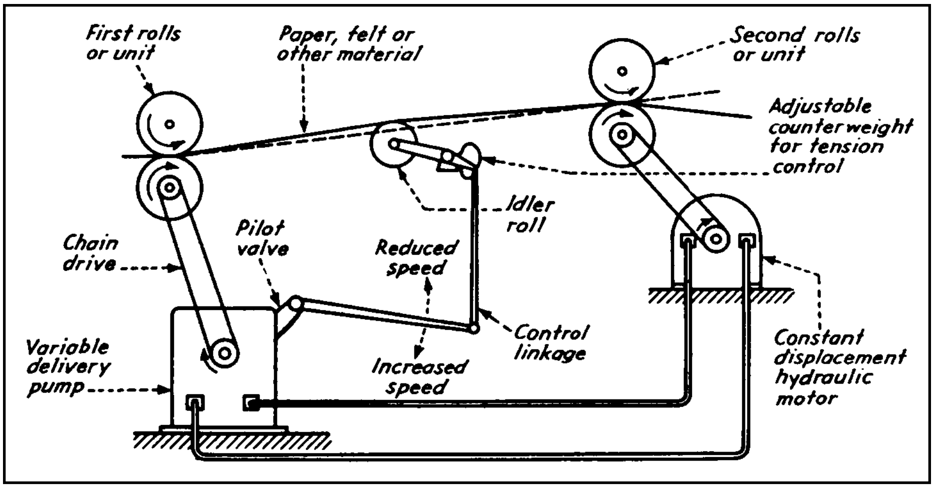
Selsyn motors can directly drive independent units in exact synchronism, provided their inertias are not too great. Regardless of loads and speeds, selsyn motors can be the controlling units. As an example, variable-speed mechanical transmission units with built-in selsyn motors are available for powering constant-tension drives or the synchronous driving of independent units.



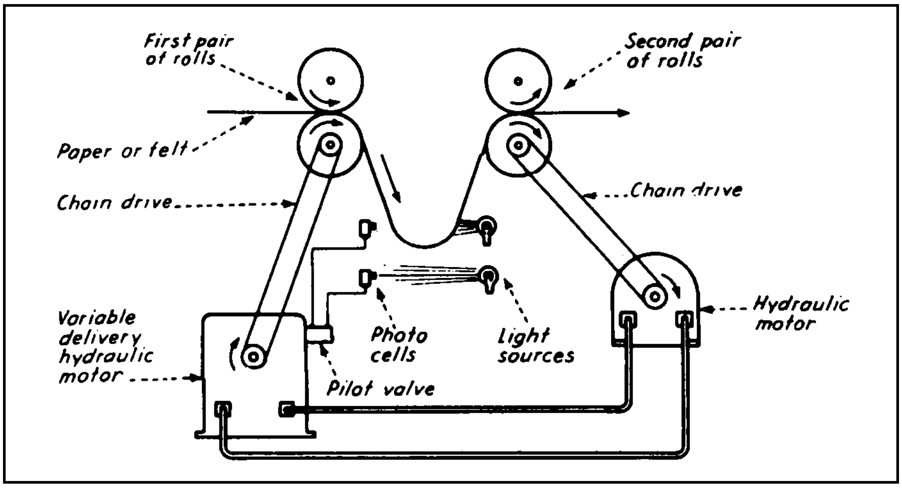
HYDRAULIC DRIVES

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.

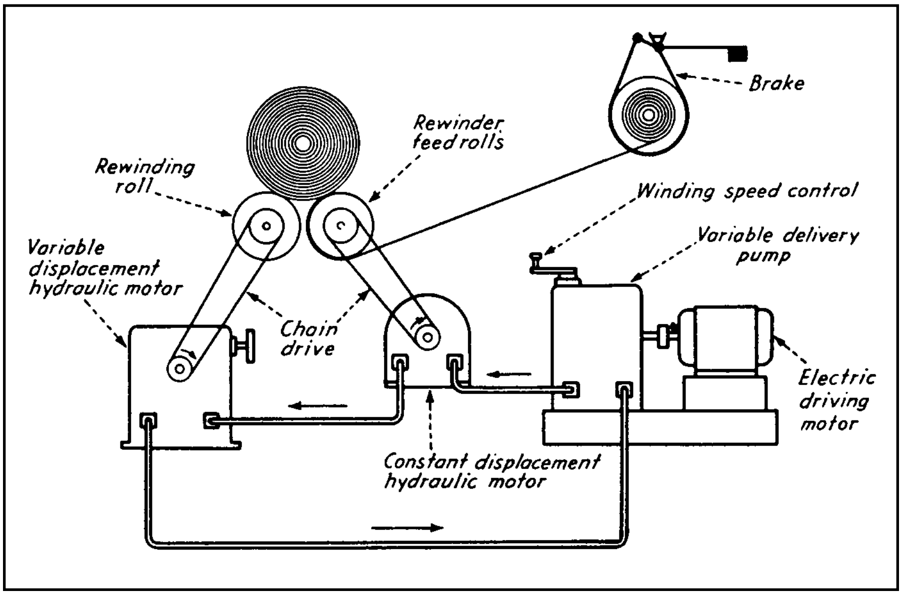


If the material passing through the machine is too weak to operate a mechanical linkage, the desired control can be obtained by photoelectric devices. The hydraulic operation is exactly the same as that described for the hydraulic drives.



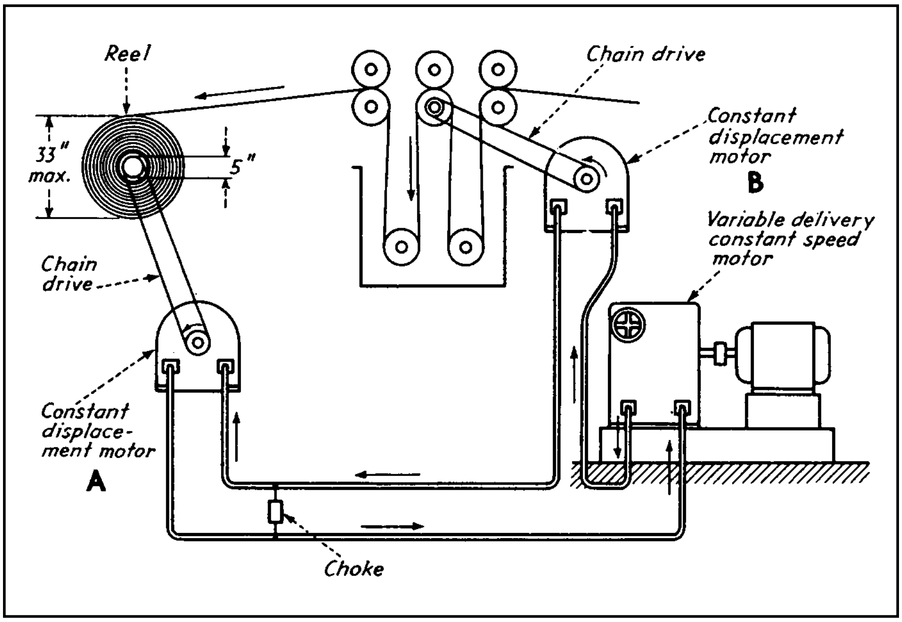
A band brake intended to obtain a friction drag will give variable tension. In this hydraulic drive, the winding tension is determined by the difference in torque exerted on the rewinder feed roll and the winding roll. The brake plays no part in establishing the tension.

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.



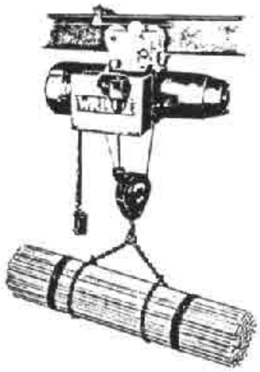
This is a hydraulic drive for fairly constant tension. The variable-delivery, constant-speed pumping unit supplies the oil to two constant displacement motors. One drives the apparatus that carries the fabric through the bath at a constant speed, and the other drives the winder. The two motors are in 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.

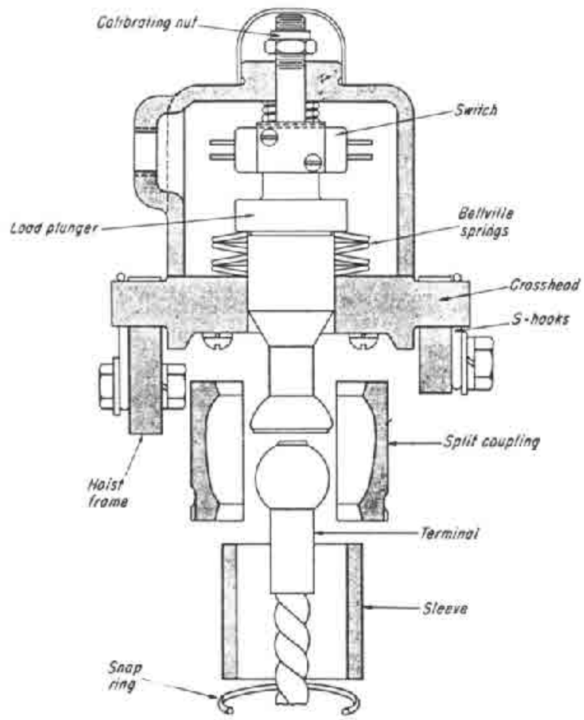
When the roll is full, the revolutions per minute of the reel and its driving motor are only about one-seventh of the revolutions per minute when the reel is empty. More oil is forced through the choke when the reel is full because of the increased pressure in the line between the two motors. The pressure 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.





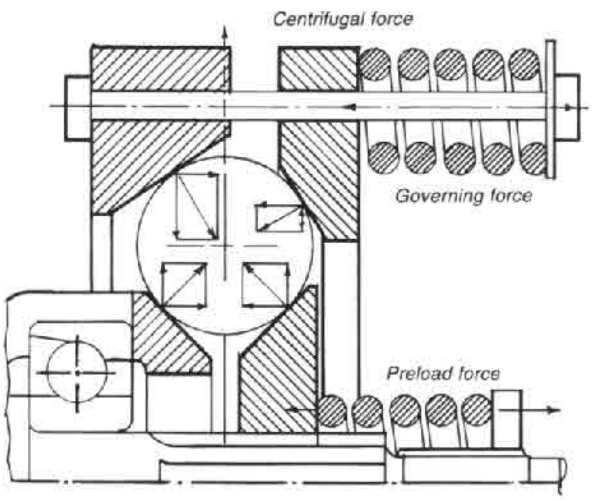
A load plunger is inserted through the belleville springs, which are supported on a swiveling crosshead. The crosshead is mounted on the hoist frame and retained by two S-hooks and bolts. Under load, the belleville springs deflect and permit the load plunger to move axially. The end of the load plunger 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.

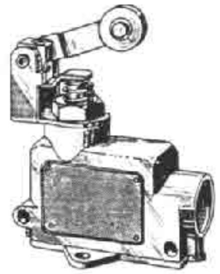
MECHANICAL, GEARED, AND CAMMED LIMIT SWITCHES

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.

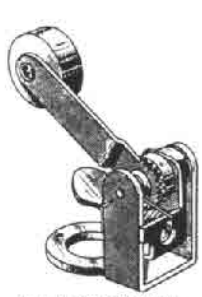
Actuators

Linear mechanical switches


BASIC SPRING-RETURN


FOUR-POSITION HEAD


90-POSITION VERNIER

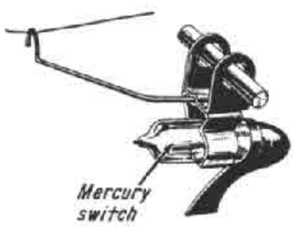

INFINITE-POSITION WORM


ADJUSTABLE LENGTH

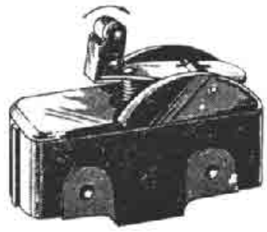

EXTENDED HOUSING

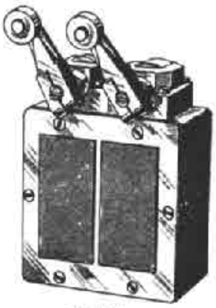

THREADED BUSHING

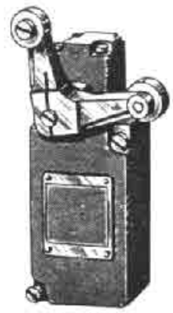
FLEXIBLE ROD

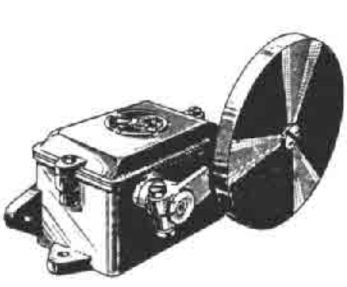


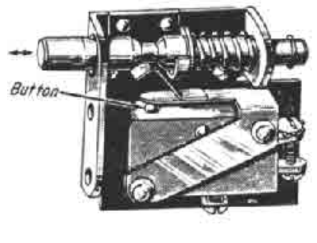
TILT-TO-ACTUATE


ONE-WAY OVERRIDE

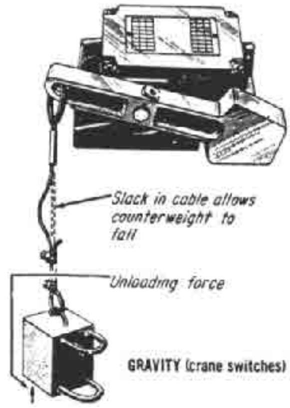

DUPLEX


BELLCRANK


OVERSIZE WHEEL



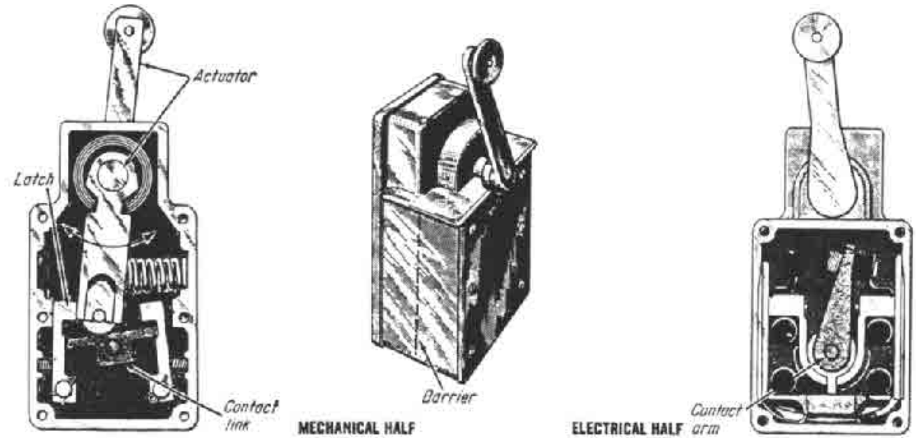
LINEAR CAM



GRAVITY (crane switches)

DIRECT-ACTING (hatchway)

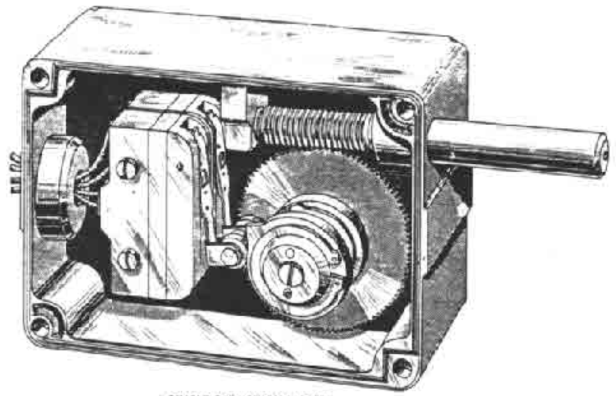
Latching switch with contact chamber



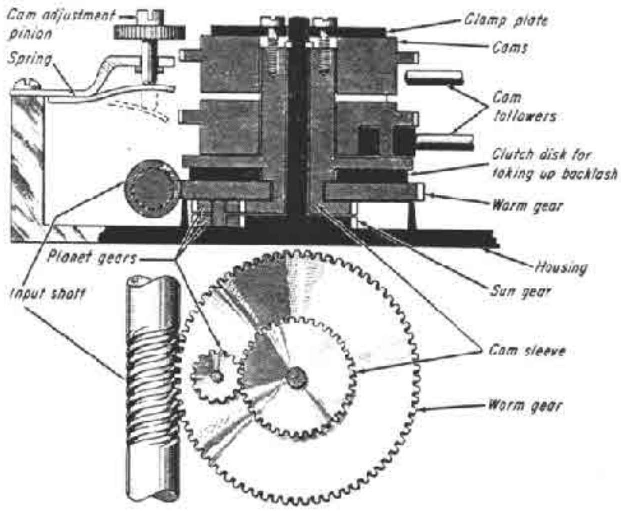
MECHANICAL HALF

ELECTRICAL HALF

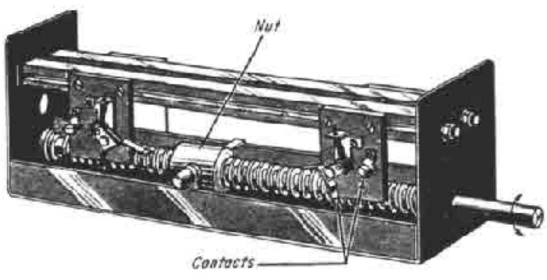
Geared rotary limit switches



SINGLE-REDUCTION WORM

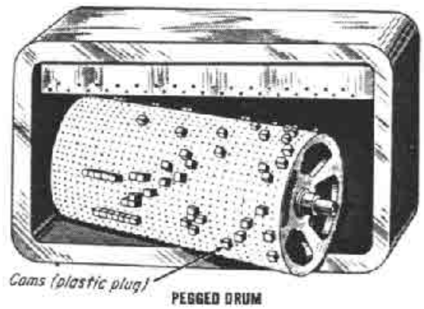


WORM AND PLANETARY

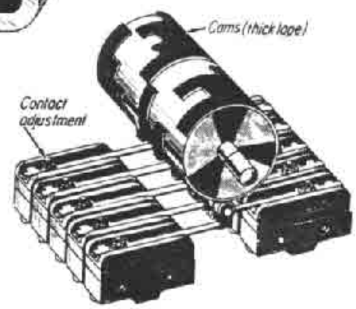


TRAVELING NUT

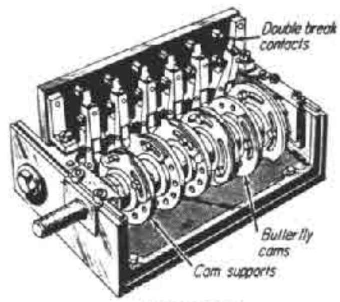
Rotary-cam limit switches



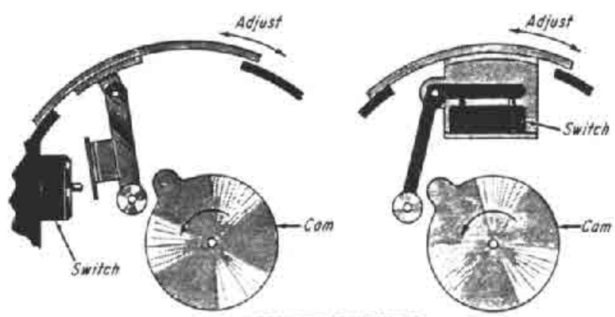
PEGGED DRUM



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.

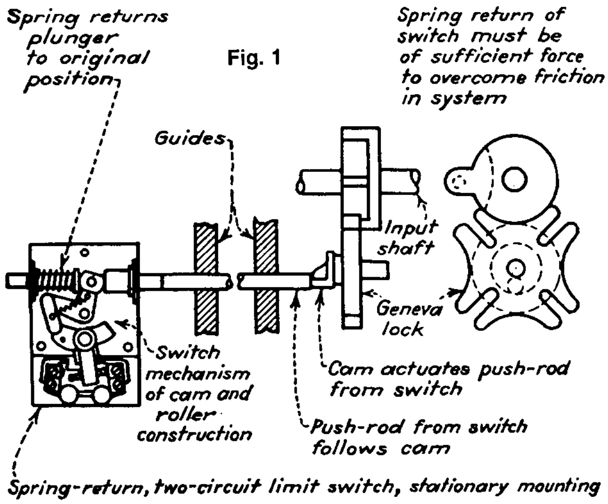


Fig. 1

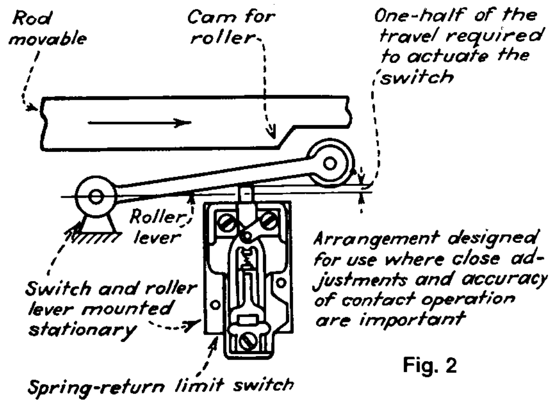


Fig. 2

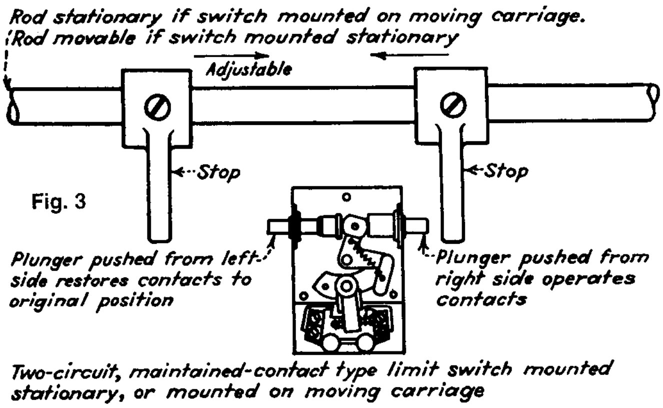


Fig. 3

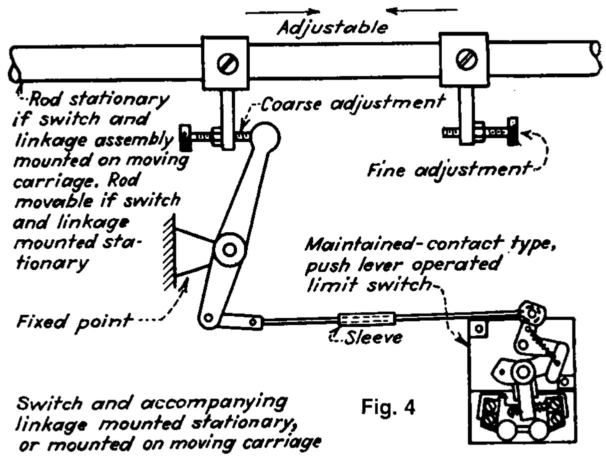


Fig. 4

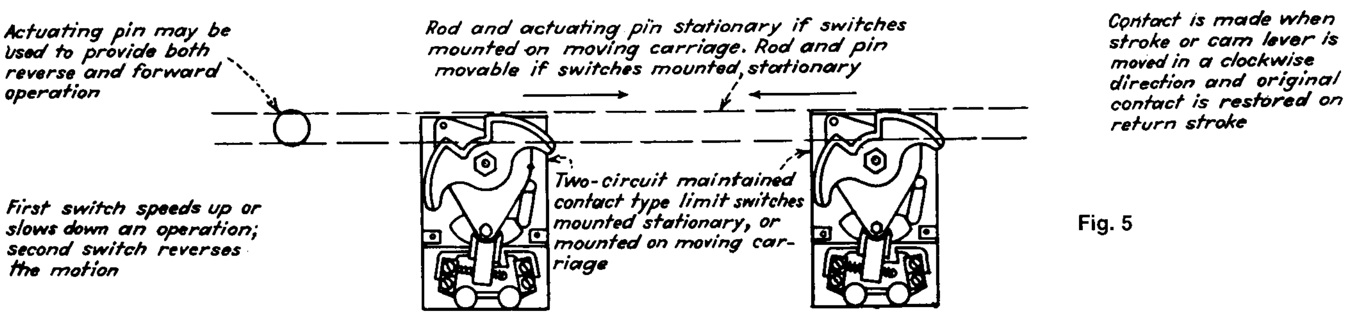
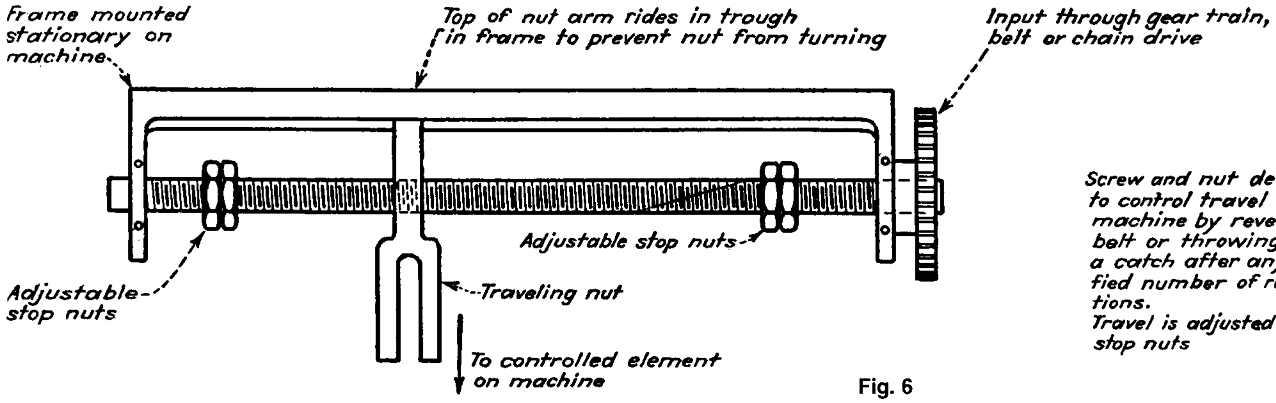
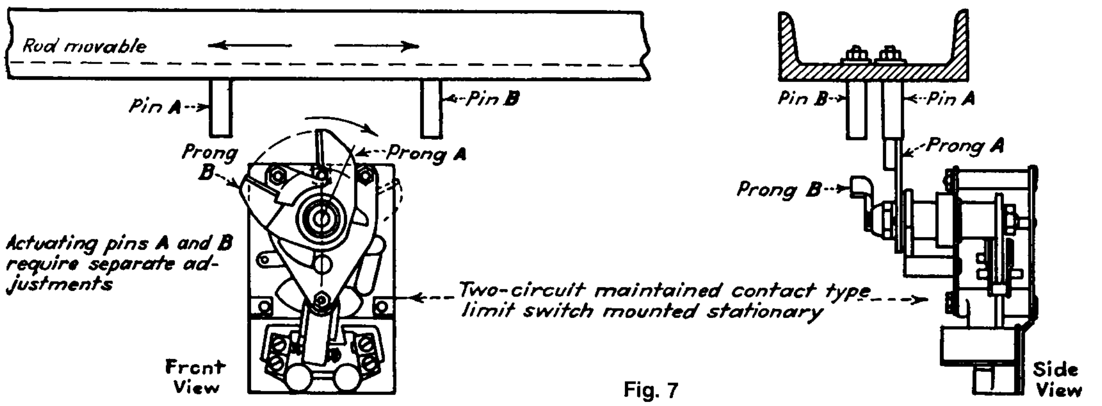


Fig. 5



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

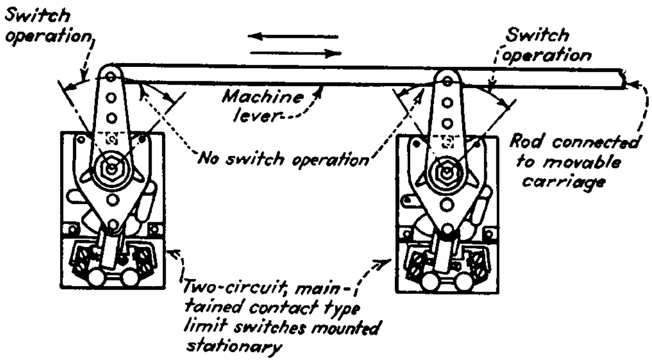
Fig. 6



Contact operation takes place when fork lever is moved. Contacts are restored when fork lever is moved in opposite direction

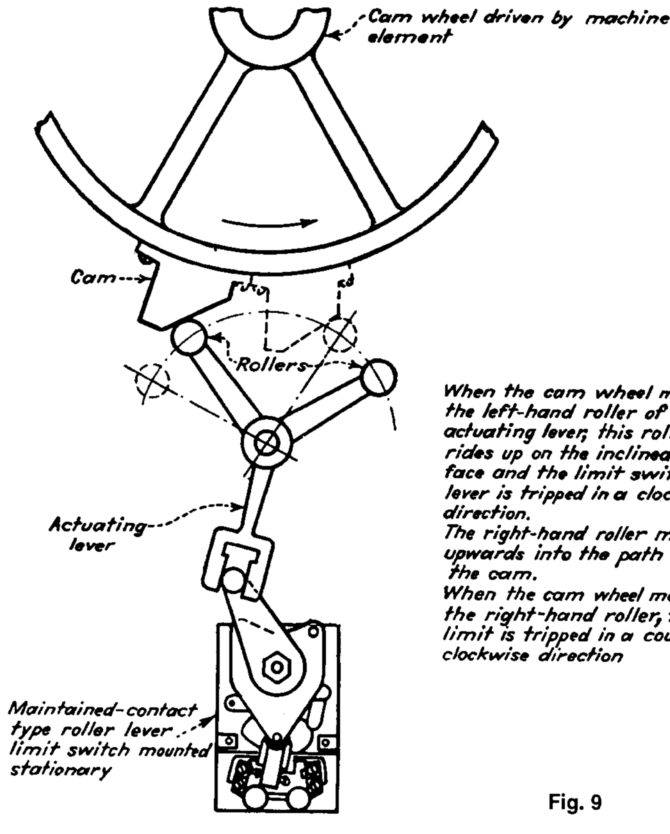
Fig. 7

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

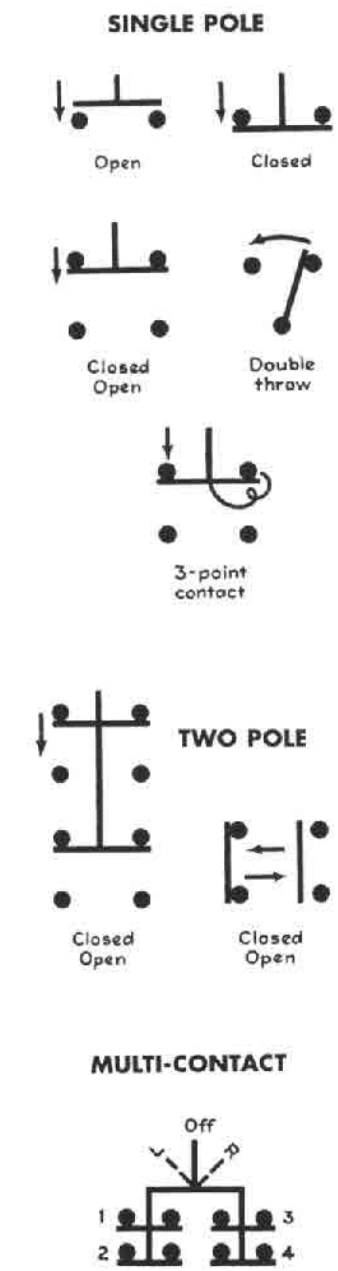


When the cam wheel meets the left-hand roller of the actuating lever, this roller rides up on the inclined surface and the limit switch lever is tripped in a clockwise direction.
The right-hand roller moves upwards into the path of the cam.
When the cam wheel meets the right-hand roller, the limit is tripped in a counterclockwise direction

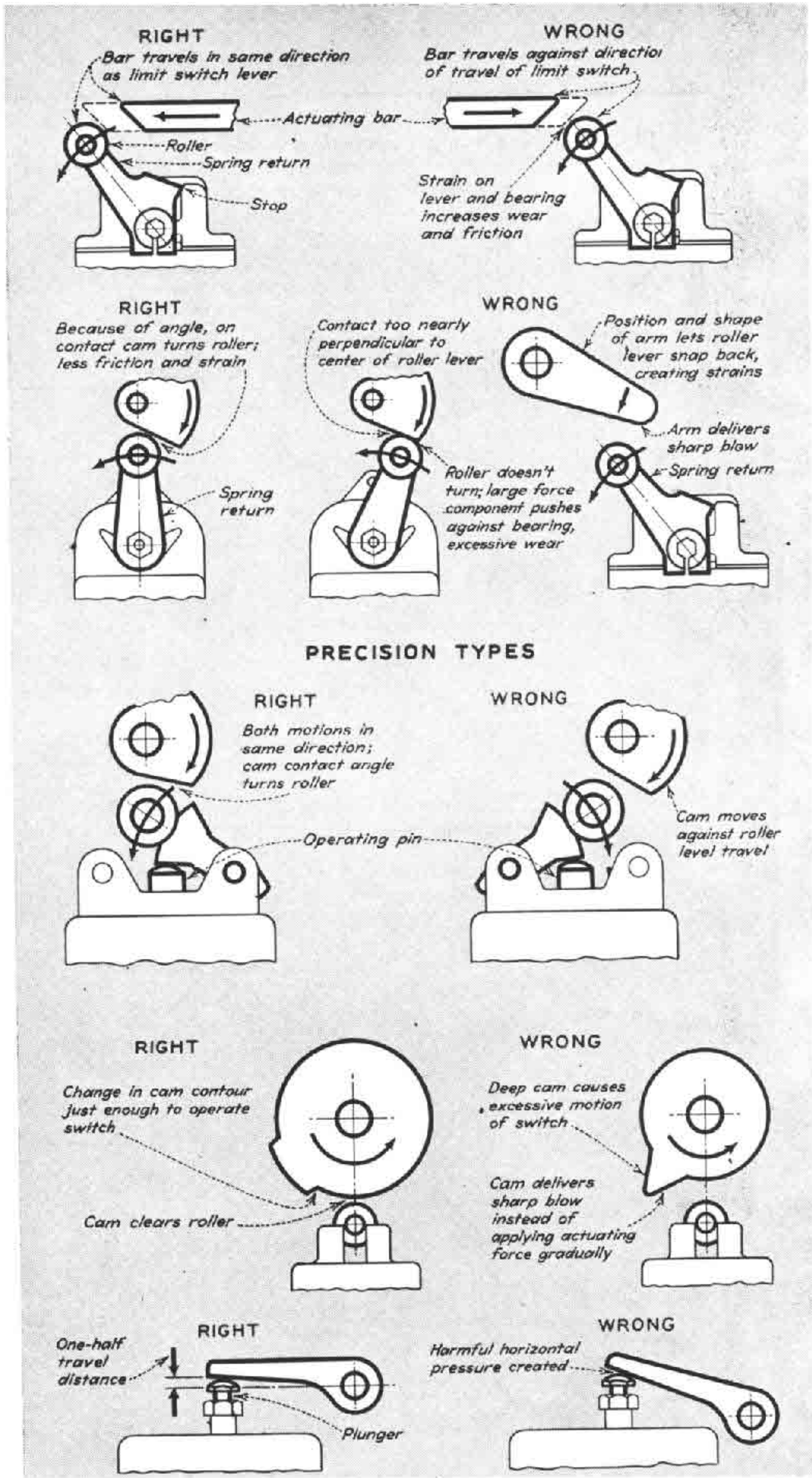
Fig. 9

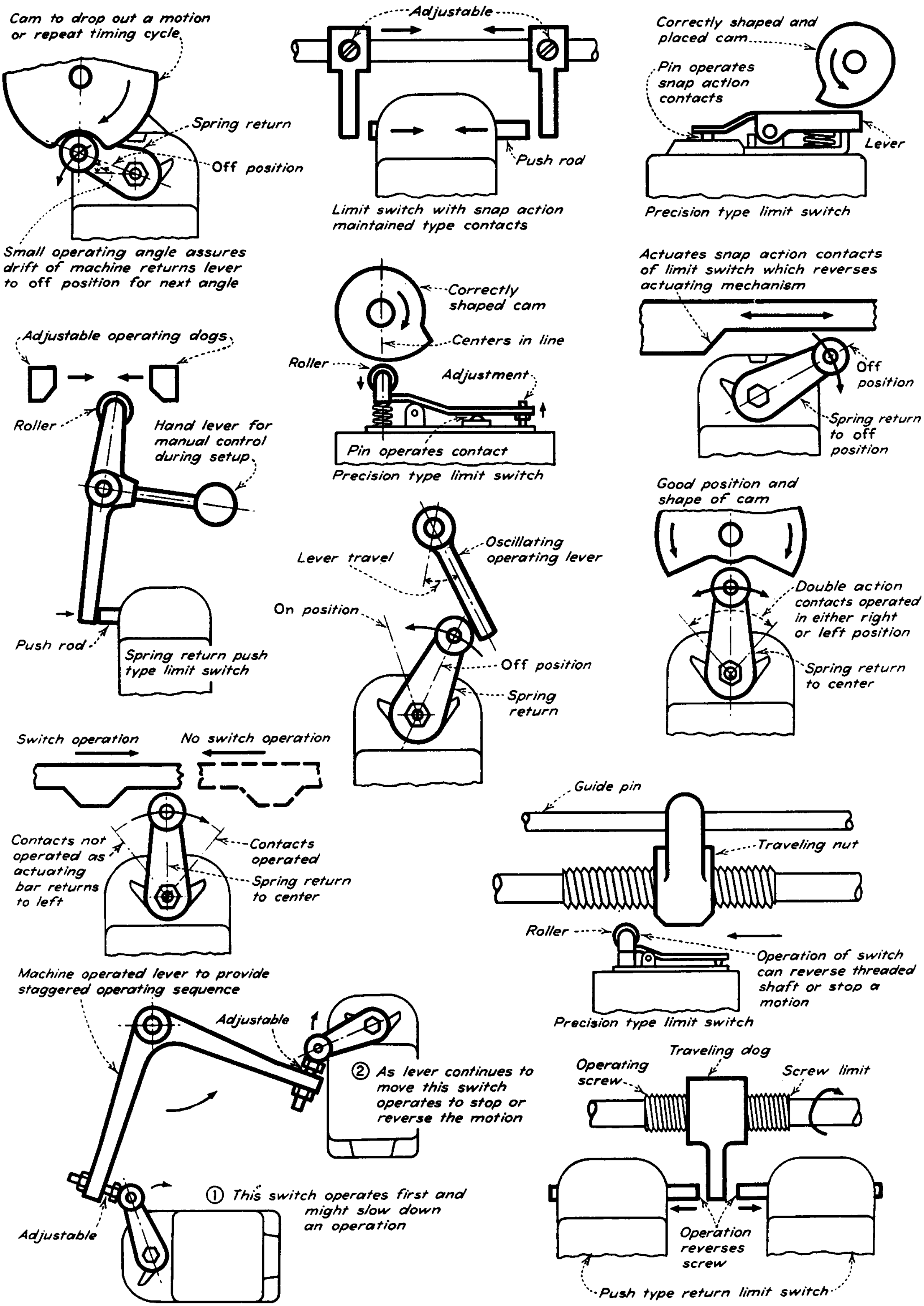
Electrical contact arrangements

All contracts in normal position with limit switch unactuated



POS.	1	2	3	4
R	C	C	O	O
Off	C	C	C	C
L	O	O	C	C




Cam to drop out a motion or repeat timing cycle
Spring return
Off position
Small operating angle assures drift of machine returns lever to off position for next angle
Adjustable
Push rod
Limit switch with snap action maintained type contacts
Correctly shaped and placed cam
Pin operates snap action contacts
Lever
Precision type limit switch
Adjustable operating dogs
Roller
Hand lever for manual control during setup
Push rod
Spring return push type limit switch
Correctly shaped cam
Centers in line
Roller
Adjustment
Pin operates contact
Precision type limit switch
Actuates snap action contacts of limit switch which reverses actuating mechanism
Off position
Spring return to off position
Lever travel
Oscillating operating lever
On position
Off position
Spring return
Good position and shape of cam
Double action contacts operated in either right or left position
Spring return to center
Switch operation
No switch operation
Contacts not operated as actuating bar returns to left
Contacts operated
Spring return to center
Guide pin
Traveling nut
Roller
Operation of switch can reverse threaded shaft or stop a motion
Precision type limit switch
Machine operated lever to provide staggered operating sequence
Adjustable
② As lever continues to move this switch operates to stop or reverse the motion
① This switch operates first and might slow down an operation
Adjustable
Traveling dog
Operating screw
Screw limit
Operation reverses screw
Push type return 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.

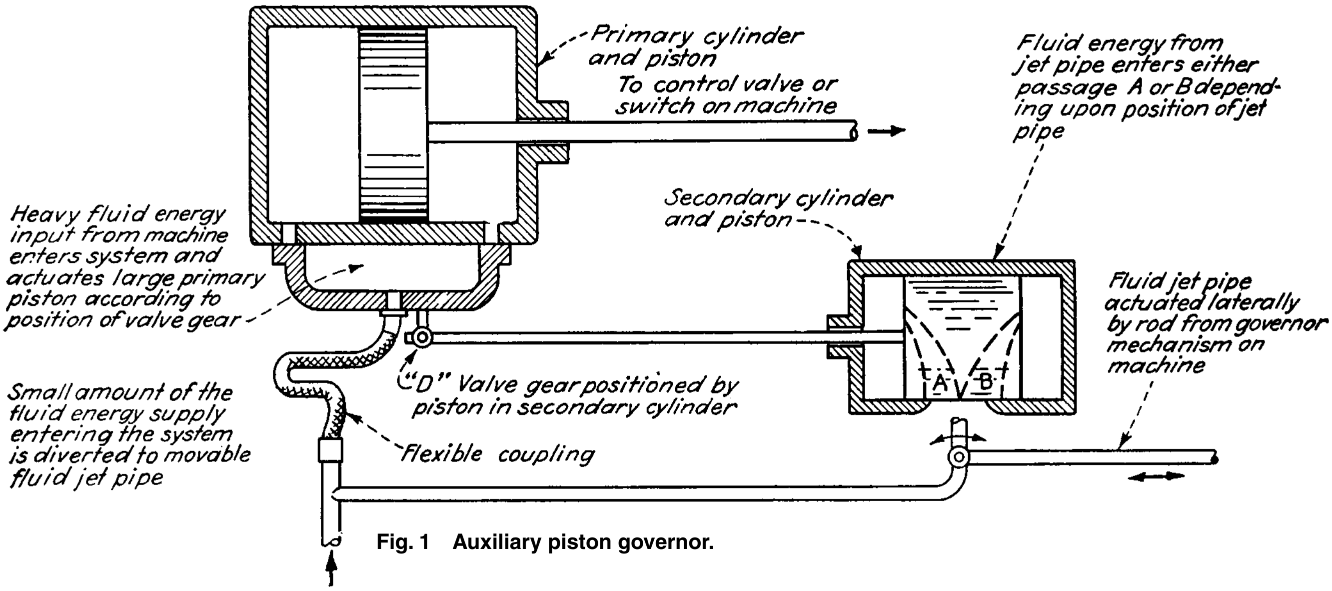


Fig. 1 Auxiliary piston governor.

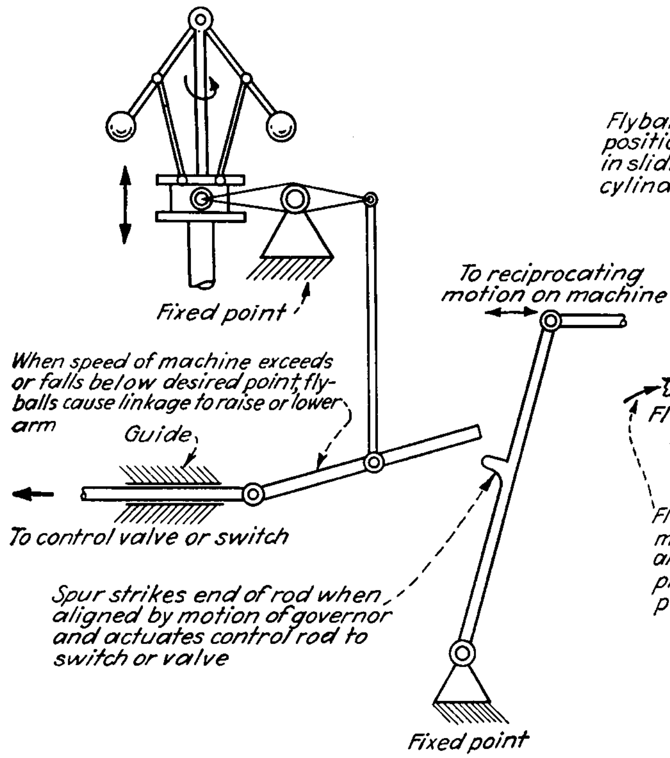


Fig. 2 Hit-and-miss governor.

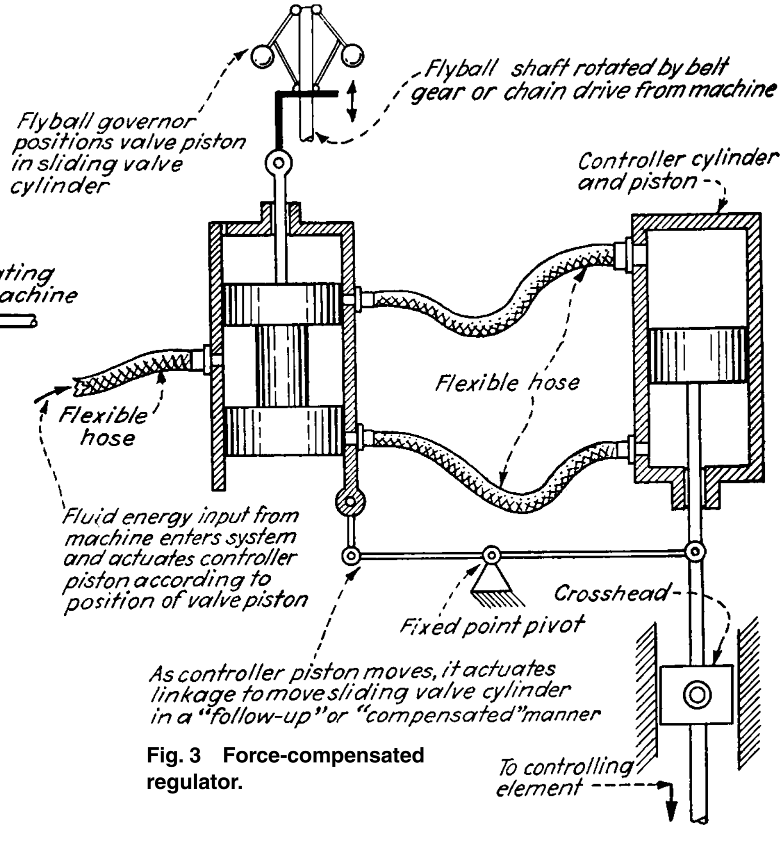


Fig. 3 Force-compensated regulator.

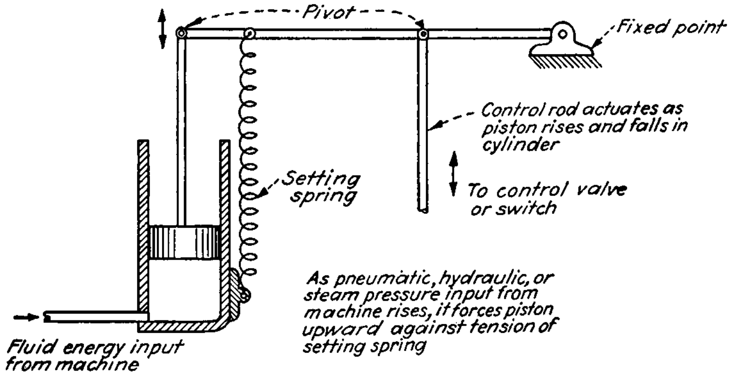


Fig. 4 Pressure-actuated governor.

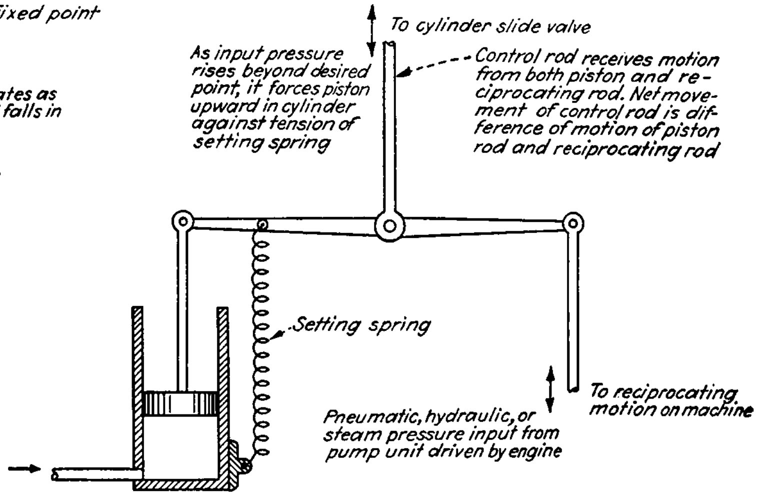


Fig. 5 Varying differential governor.

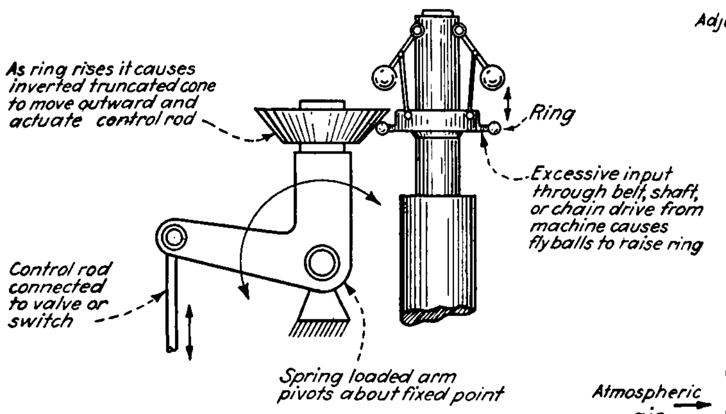


Fig. 6 Centrifugal governor.

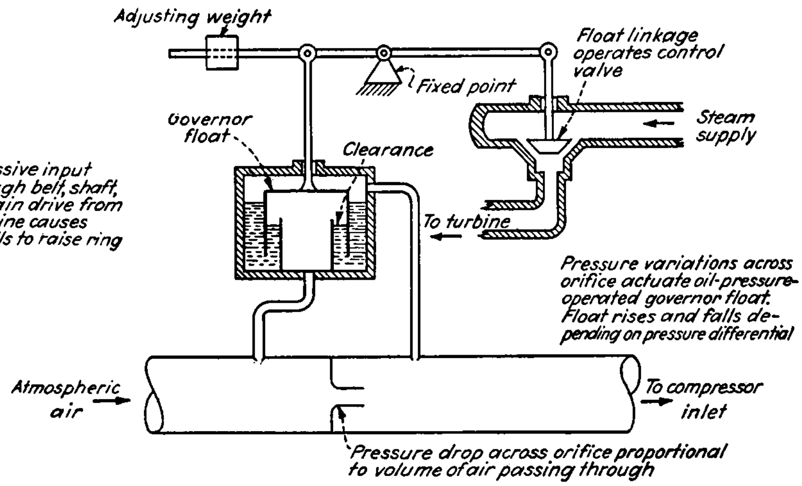


Fig. 7 Constant-volume governor.

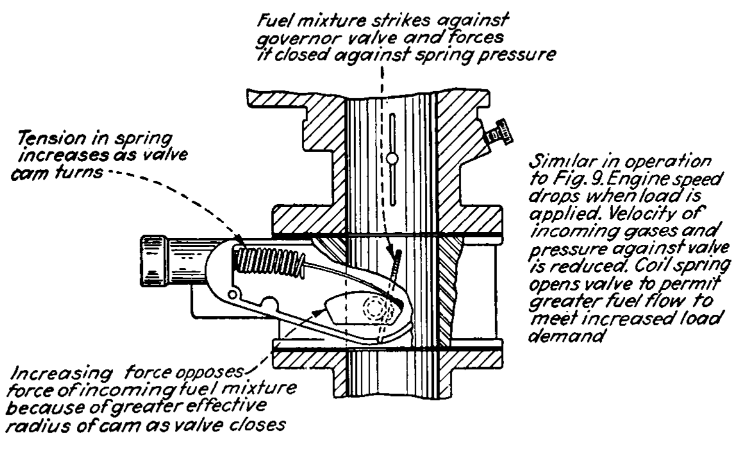


Fig. 8 Velocity-type governor (coil spring).

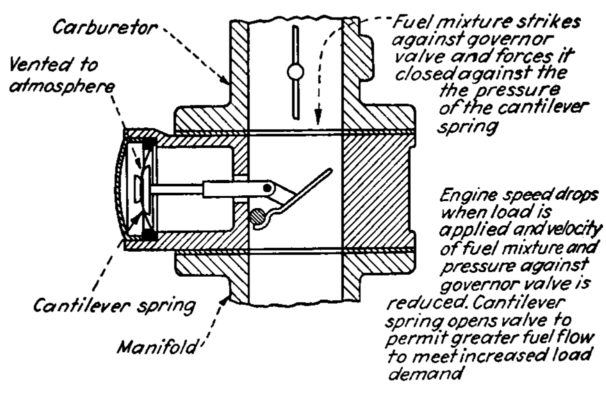


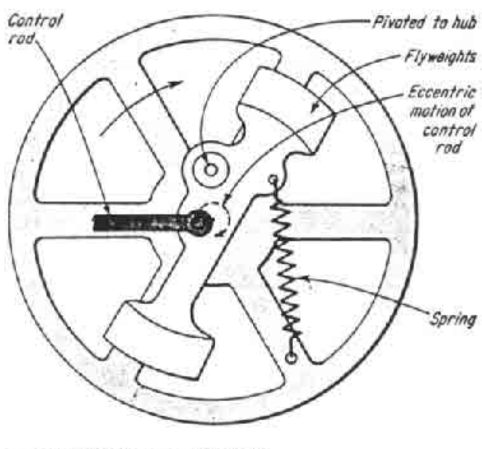
Fig. 9 Velocity-type governor (cantilever spring).

CENTRIFUGAL, PNEUMATIC, HYDRAULIC, AND ELECTRIC GOVERNORS

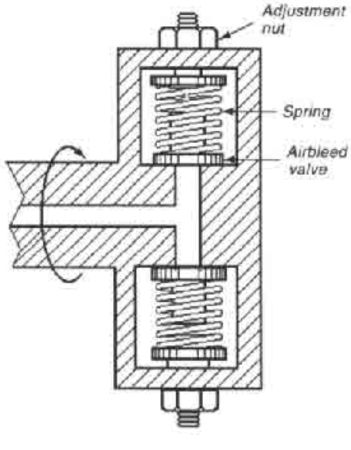
Centrifugal governors are the most common—they are simple and sensitive and have high output force. There is more published information on centrifugal governors than on all other types combined.

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.

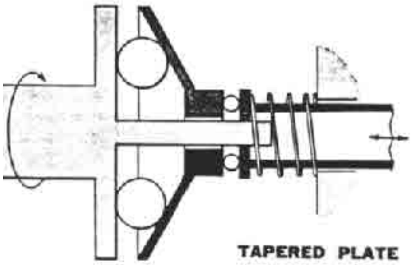
Centrifugal Governors



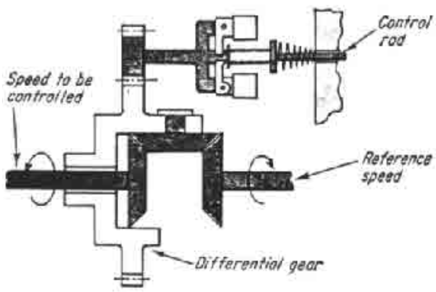
ACCELERATION GOVERNOR (steam engine)



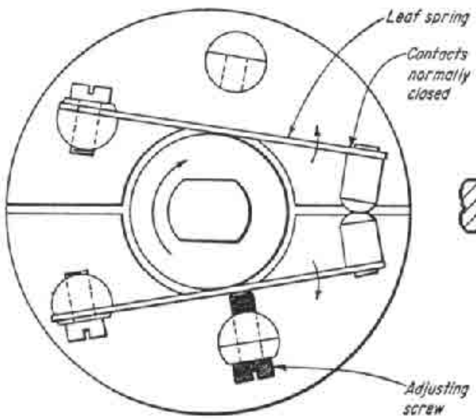
CENTRIFUGAL VALVE



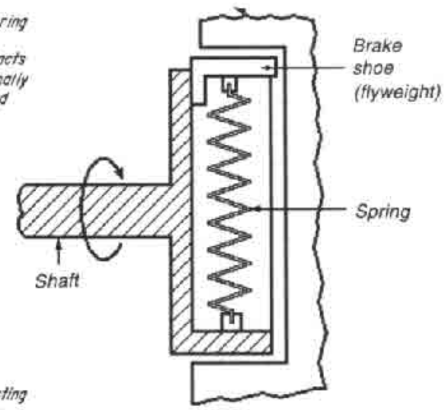
TAPERED PLATE



DIFFERENTIAL CENTRIFUGAL



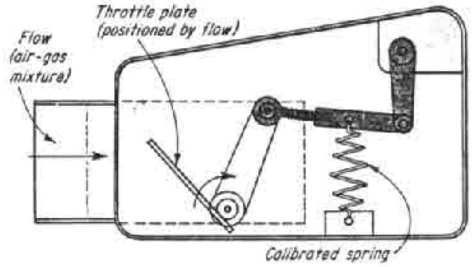
CENTRIFUGAL CONTACTS



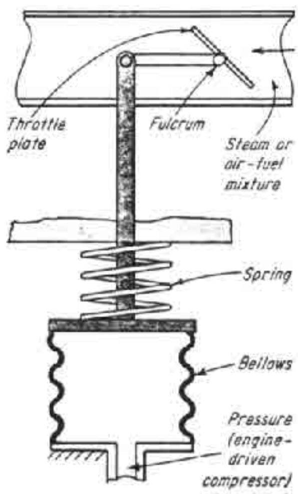
CENTRIFUGAL CLUTCH

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.

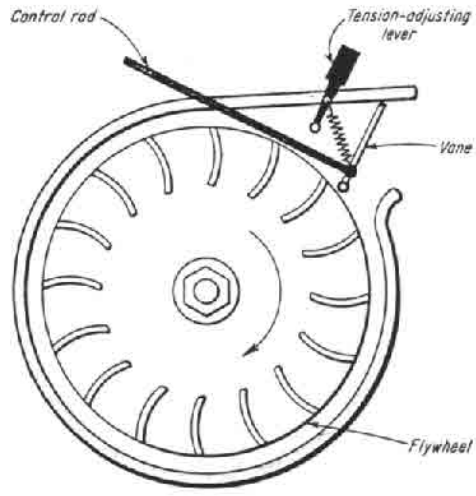
Pneumatic Governors



CARBURETOR-FLOW VELOCITY (linkage)

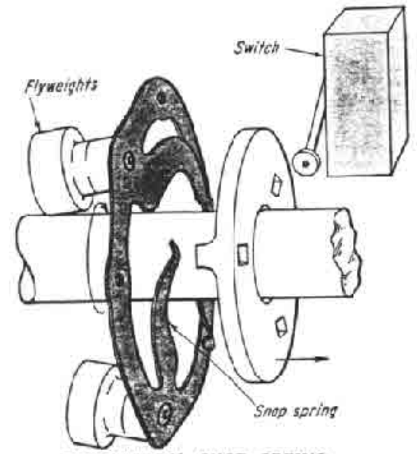


COMPRESSOR PRESSURE (direct)

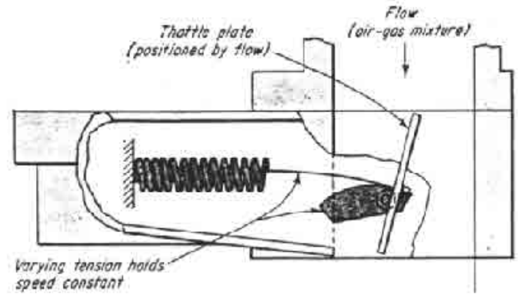


FAN-FLOW VELOCITY

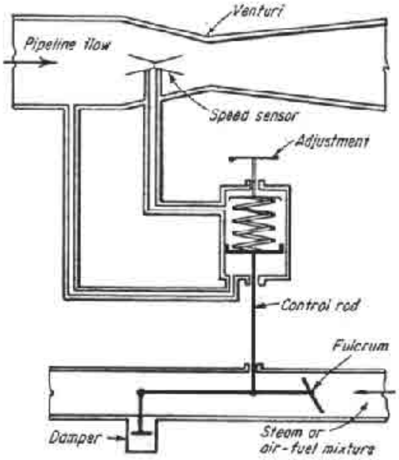
More pneumatic governors



CENTRIFUGAL SNAP SPRING



CARBURETOR-FLOW VELOCITY
(cam)



COMPRESSOR PRESSURE
(differential)

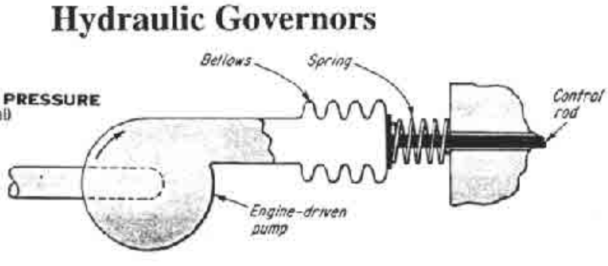
Hydraulic sensors measure the discharge pressure of engine-driven pumps. Pressure is proportional to the square of the speed of most pumps, although some have special impellers with linear pressure-speed characteristics.

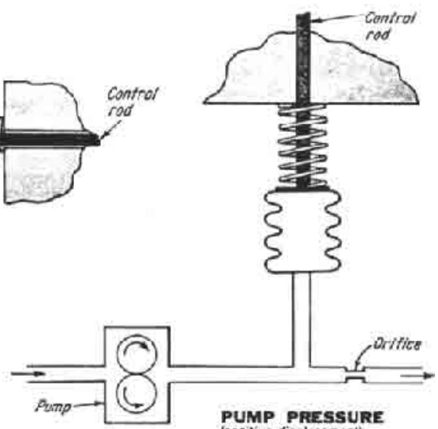
Straight vanes are better than curved vanes because the pressure is less affected by the volume flow. Low pressures are preferred over high pressures because fluid friction is less.

Typical applications for these governors include farm tractors with diesel or gasoline engines, larger diesel engines, and small steam turbines.

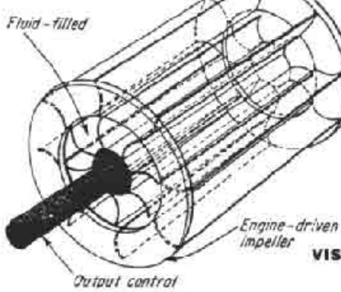
Hydraulic Governors

PUMP PRESSURE
centrifugal



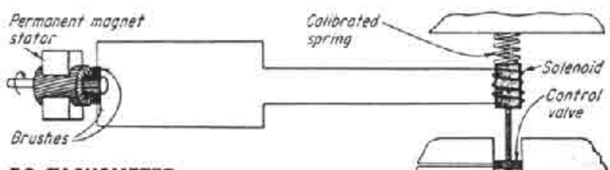


PUMP PRESSURE
(positive displacement)

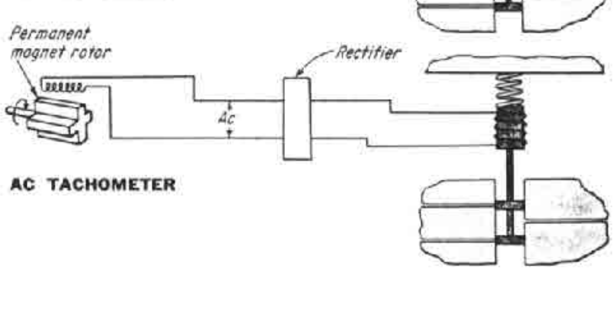


VISCOUS DRAG

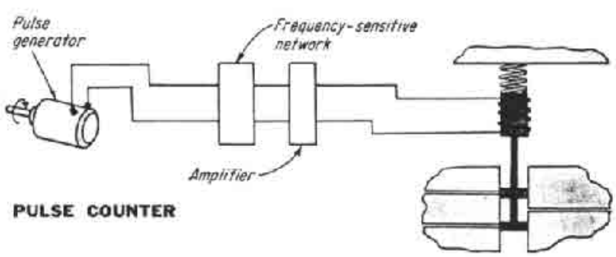
Electric Governors



DC TACHOMETER



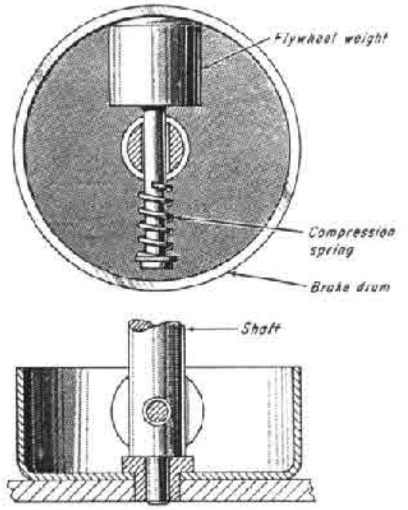
AC TACHOMETER



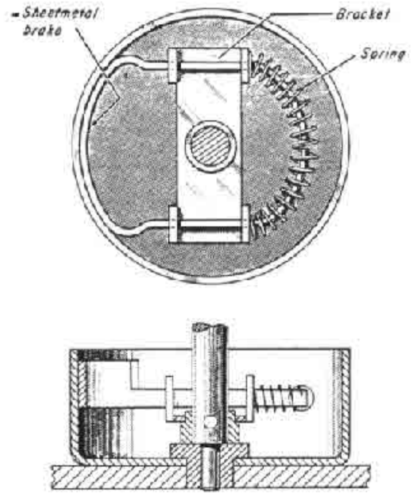
PULSE COUNTER

SPEED CONTROL DEVICES FOR MECHANISMS

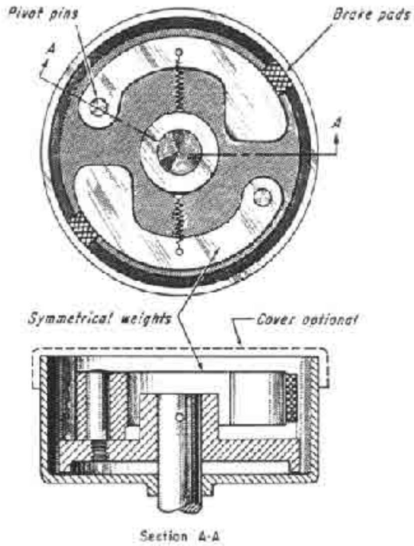
Friction devices, actuated by centrifugal force, automatically keep speed constant regardless of variations of load or driving force.



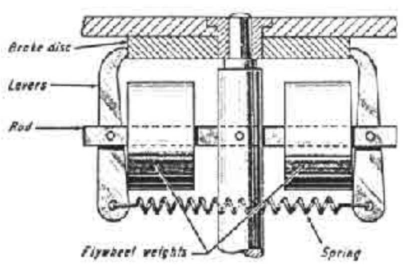
The weight is counterbalanced by a spring that brakes the shaft when the rotation speed becomes too fast. The braking surface is small.



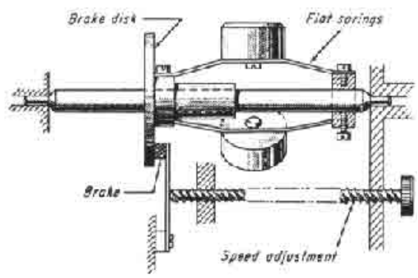
A sheetmetal brake provides a larger braking surface than in the previous brake. Braking is more uniform, and it generates less heat.



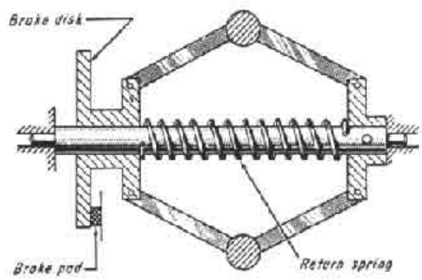
Symmetrical weights give an even braking action when they pivot outward. The entire action can be enclosed in a case.



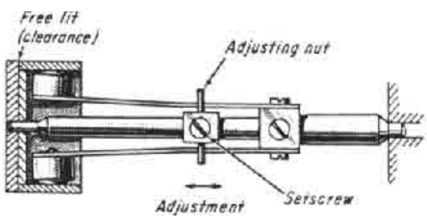
Weight-actuated levers make this arrangement suitable where high braking moments are required.



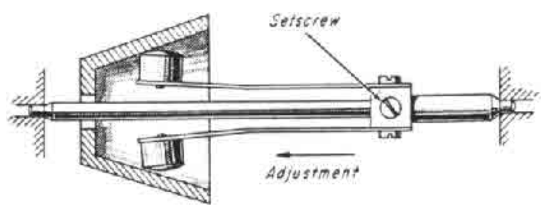
Three flat springs carry weights that provide a brake force upon rotation. A speed adjustment can be included.



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

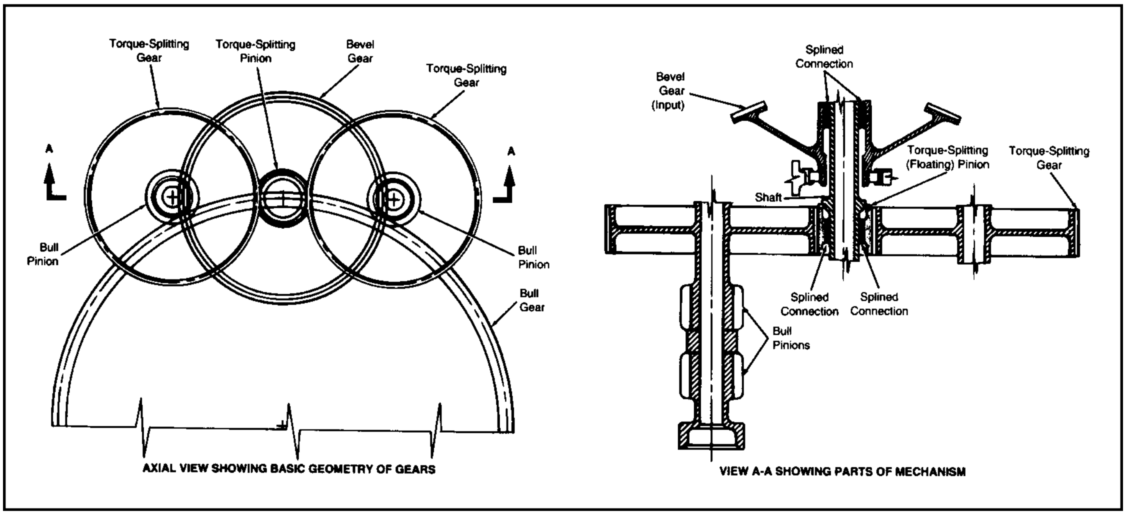
A gear-drive mechanism helps to apportion torques nearly equally along two parallel drive paths from an input bevel gear to an output bull gear. A mechanism of this type could be used, for example, as part of a redundant drive train between the engine and the rotor of a helicopter. The principal advantage 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**.



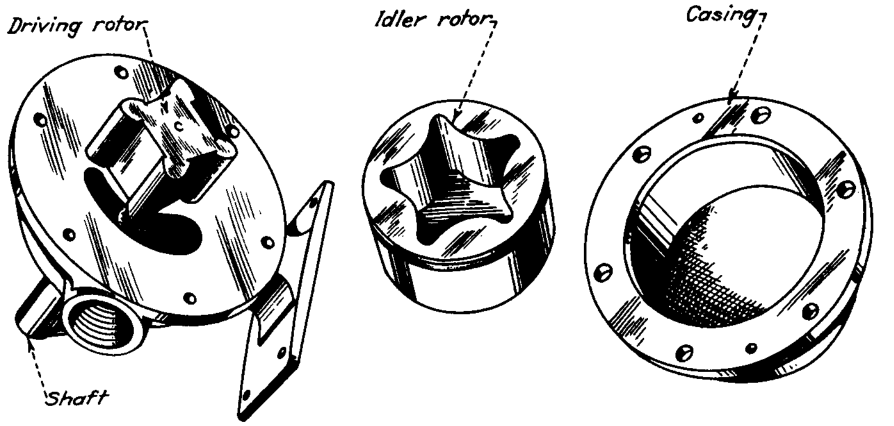
Splined connections permit small angular excursions of a shaft, the bevel gear on its upper end, and the torque-splitting pinion on its lower end. These small excursions are essential for equalization of torques in the presence of machining tolerances and other geometric imperfection of the drive train.

CHAPTER 11

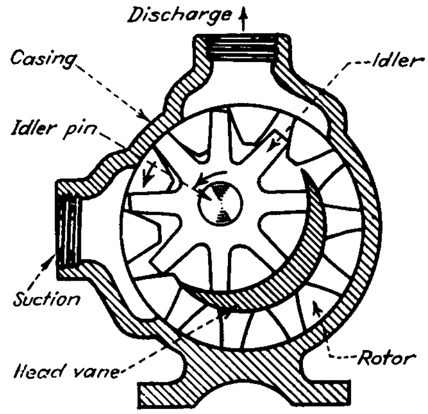
PNEUMATIC AND HYDRAULIC MACHINE AND MECHANISM CONTROL

DESIGNS AND OPERATING PRINCIPLES OF TYPICAL PUMPS

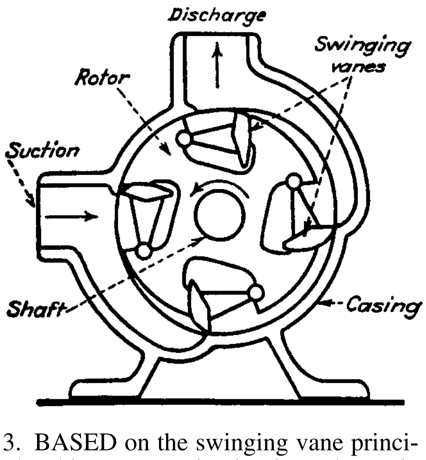
These pumps are used to transfer liquids and supply hydraulic power.



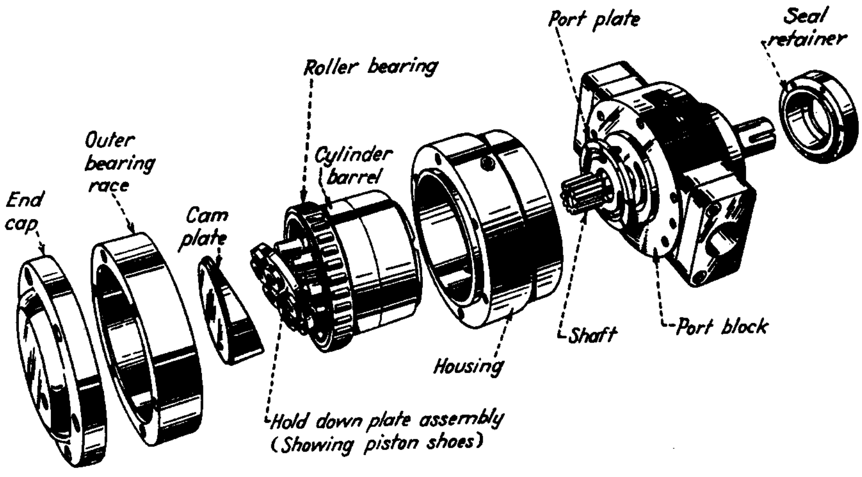
1. WITH BUT TWO MOVING PARTS, the rotors that turn in the same direction, this rotary pump has reduced friction to a minimum. The rotors rotate against flexible synthetic rubber cushions that allow sand, grit and other abrasives to flow freely through the pump without damage. It is a positive displacement pump that develops a constant pressure and will deliver a uniform flow at any given speed. The pump is reversible and can be driven by a gasoline engine or electric motor. The rubber cushions withstand the action of oil, kerosene, and gasoline, and the pump operates at any angle. It has been used in circulating water systems, cutting tool coolant oil systems and general applications.



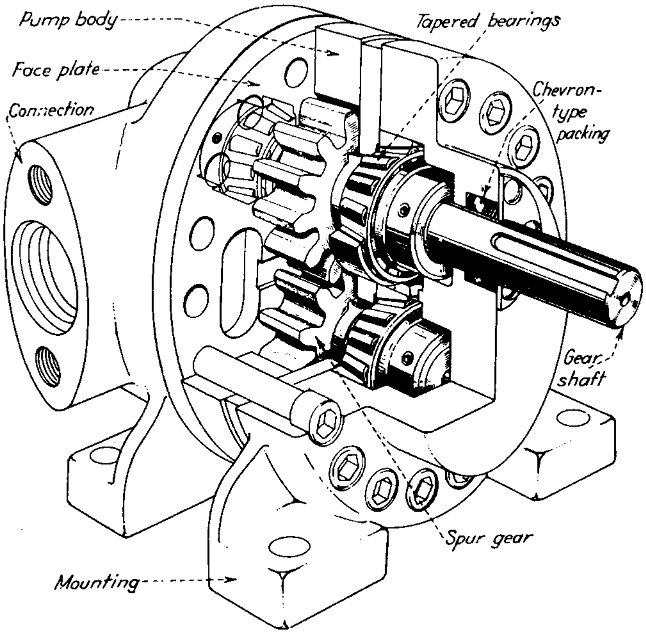
2. PUMPING ACTION is produced by the meshing of the idler and rotor teeth in this rotary pump. The idler is pin-mounted to the head and the rotor operates in either direction. This pump will not splash, foam, churn or cause pounding. Liquids of any viscosity that do not contain grit can be transferred by this pump which is made of iron and bronze.



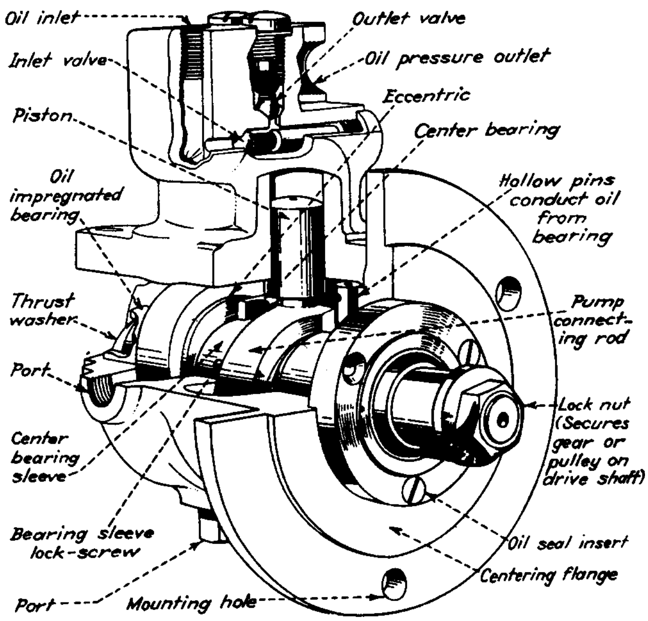
3. BASED on the swinging vane principle, this pump maintains its volumetric efficiency automatically. The action of the buckets, fitted loosely into recesses in the rotor, compensates for wear. In operation, the tip of the bucket is in light contact with the casing wall. Liquids are moved by sucking and pushing actions and are not churned or foamed.



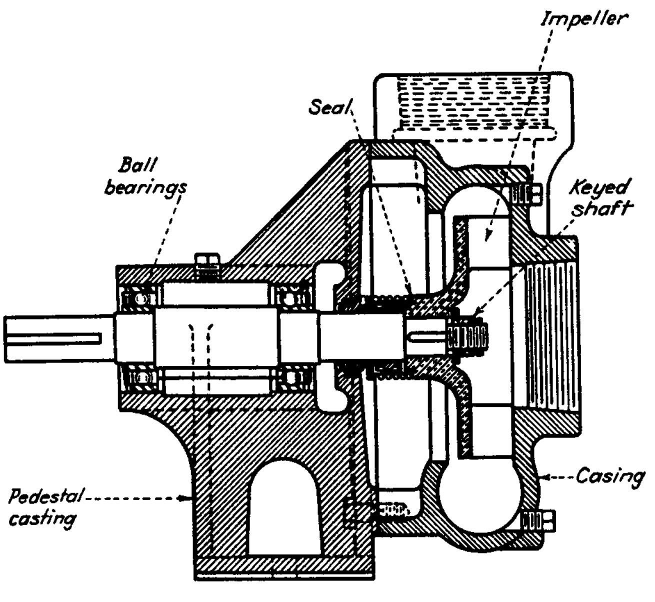
4. HIGH-PRESSURE, high-volume pumps of the axial piston, constant displacement type are rated at 3,500 psi for continuous duty operation; higher pressure is permissible for intermittent operation. A pressure-balanced piston shoe lubricates the cam plate and prevents direct contact between the shoe and cam plate. The use of the pressure balanced system removes the need for thrust bearings. The two-piece shaft absorbs deflection and minimizes bearing wear. The pump and electric driving motor are connected by a flexible coupling. The revolving cylinder barrel causes the axial reciprocation of the pumping pistons. These pumps only pump hydraulic fluids.



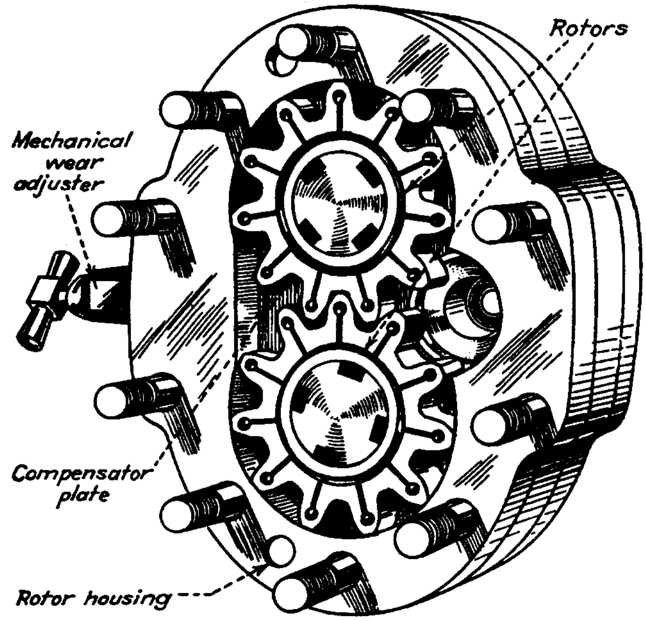
5. THE GEAR SHAFTS of this hydraulic gear pump are mounted on tapered roller bearings that accurately position the gears, decrease end play, and reduce wear to a minimum. This heavy-duty gear pump can be used at pressures up to 1,000 psi. These pumps were made with either single- or double-end shafts and can be foot- or flange-mounted. The drive shaft entrance packing is made from oil-resistant material, and the gear shafts are made from hardened molybdenum steel.



6. THIS HIGH-PRESSURE hydraulic pump has twin pistons that build pressures from 100 to 4,000 psi at speeds from 600 to 1,200 rpm. This pump can be operated continuously at 900 rpm and 2,500 psi with 1.37 gpm delivered. Because it can be mounted at any angle, and because it is used with small oil lines, small diameter rams and compact valves, the pump is suitable for installation in new equipment. This pump contains a pressure adjusting valve that is factory set to bypass at a predetermined pressure.

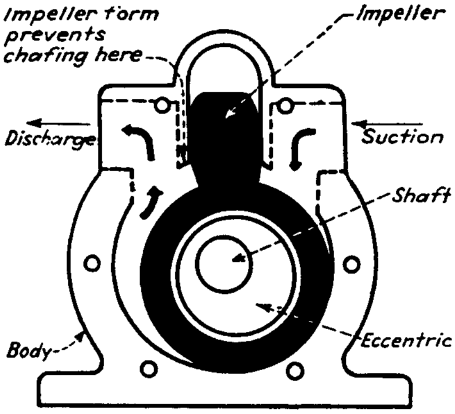


7. This pump is characterized by its pedestal mounting. The only non-critical fit is between the pedestal casting and the casing. Positive alignment is obtained because the sealed ball bearings and the shaft are supported in the single casting. The five-vaned, open, bronze impeller will move liquids that contain a high volume of solids. The pump is not for use with corrosive liquids. The five models of this pump, with ratings up to 500 gpm, are identical except for impeller and casing sizes.

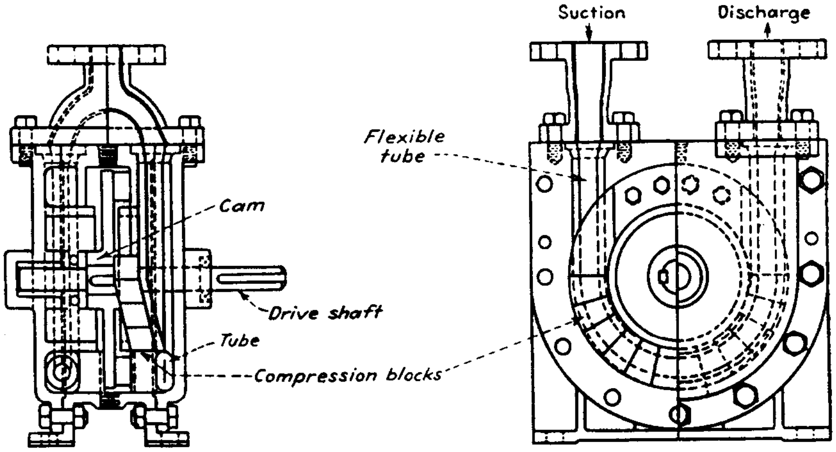


8. USED TO TRANSFER, meter, or proportion liquids of high or low viscosity, this pump is a positive displacement gear pump. It is made of stainless steel with a stainless steel armored, automatic take-up, shaft seal of the single-gland type. Automatic wear control compensates for normal wear and maintains volumetric efficiency. This pump will handle 5 to 300 gph without churning or foaming. It needs no lubrication and operates against high or low pressure.

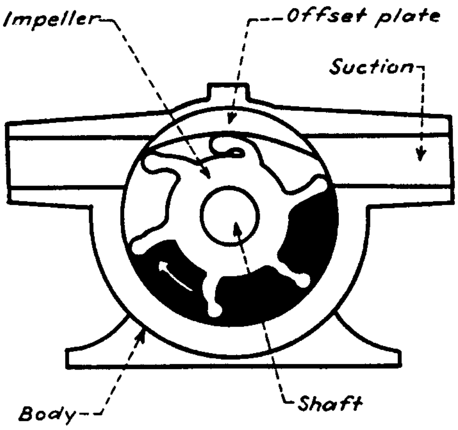
These pumps are used to transfer liquids and semisolids, pump vacuums, and boost oil pressure



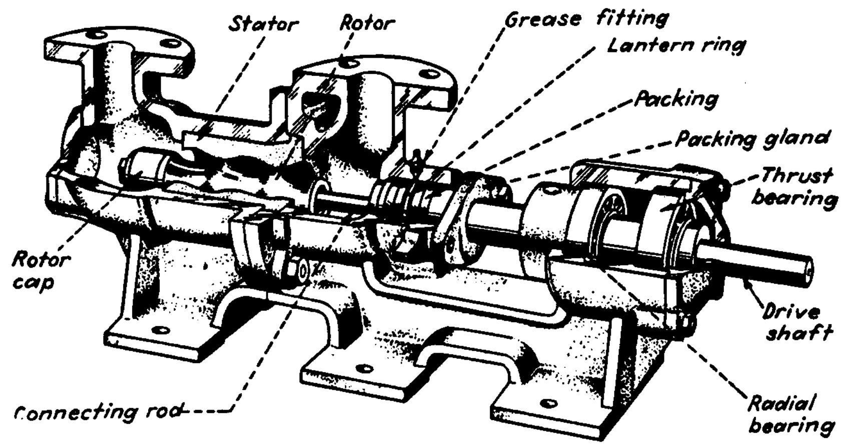
9. ORIGINALLY DESIGNED for use in the marine field, this gearless pump was made from stainless steel, monel, and bronze for handling acids, oils, and solvents. The impeller is made of pressure-vulcanized laminated layers of Hycar, 85 to 90 percent hard. Sand, grit, scale and fibrous materials will pass through. With capacities from 1 to 12 gpm and speeds from 200 to 3,500 rpm, these pumps will deliver against pressures up to 60 psi. Not self-priming, it can be installed with a reservoir. It operates in either direction and is self-lubricating.



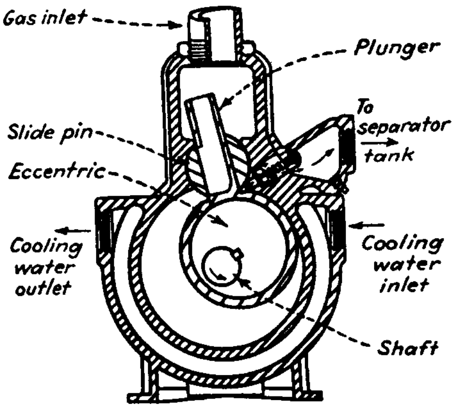
10. THE SQUEEGEE PUMP consists of a U-shaped flexible tube made of rubber, neoprene, or other flexible material. Acids and corrosive liquids or gases pass through the tube and do not contact working parts or lubricating oil. This prevents contamination of the liquid and avoids corrosion of metal parts. In operation, the tube is compressed progressively from the intake side to the discharge side by cams mounted on a driven shaft. Compression blocks move against the tube, closing the tube gradually and firmly from block to block, which forces the liquid out. As the cam passes the compression blocks, the tube returns to its original diameter. This creates a high vacuum on the intake side and causes the tube to be filled rapidly. The pump can be driven clockwise or counter-clockwise. The tube is completely encased and cannot expand beyond its original diameter. The standard pump is made of bronze and will handle volumes to 15 gpm. The Squeegee develops a vacuum of 25 in. of mercury and will work against pressures of 50 lb/in^2.



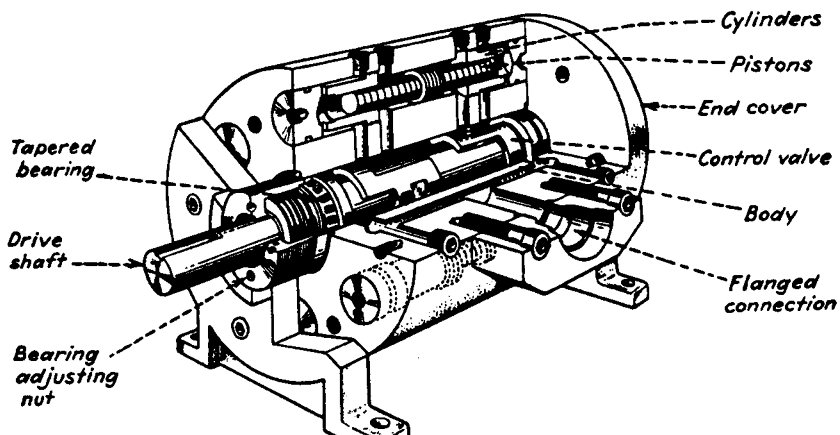
11. DEFLECTED BLADES of the flexible neoprene impeller straighten as they leave contact with offset plate. The high suction created draws fluid into pump, filling space between the blades. It handles animal, vegetable, and mineral oils but not napthas, gasoline, ordinary cleaning solvents, or paint thinners. The pump operates in either direction and can be mounted at any angle. It runs at 100 to 2,000 rpm, can deliver up to 55 gpm, and will operate against 25 psi. It operates at temperatures between 0 and 160 F.



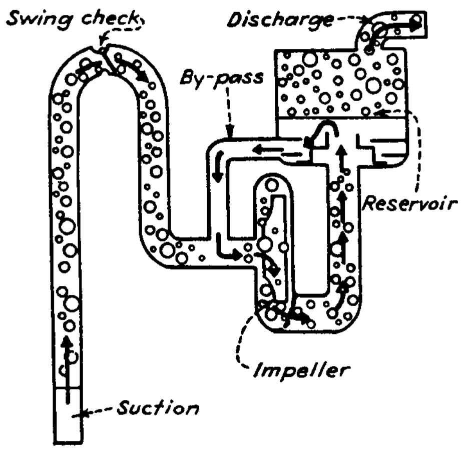
12. THIS PUMP CAN TRANSFER free-flowing liquids, non-pourable pastes, clean or contaminated with abrasives, chemically inert or active, homogeneous or containing solids in suspension. It is a positive displacement pump that delivers continuous, uniform flow. The one moving part, a low-alloy or tool-steel rotor, is a single helix, and the Hycar or natural rubber stator is a double internal helix. Pumping action is similar to that of a piston moving through a cylinder of infinite length. Containing no valves, this pump will self-prime at 28 ft of suction lift. The head developed is independent of speed, and capacity is proportional to speed. Slippage is a function of viscosity and pressure, and is predictable for any operating condition. The pump passes particles up to ⅞ in. diameter through its largest pump. Pumping action can be in either direction. The largest standard pump, with two continuous seal lines, handles 150 gpm up to 200 psi.



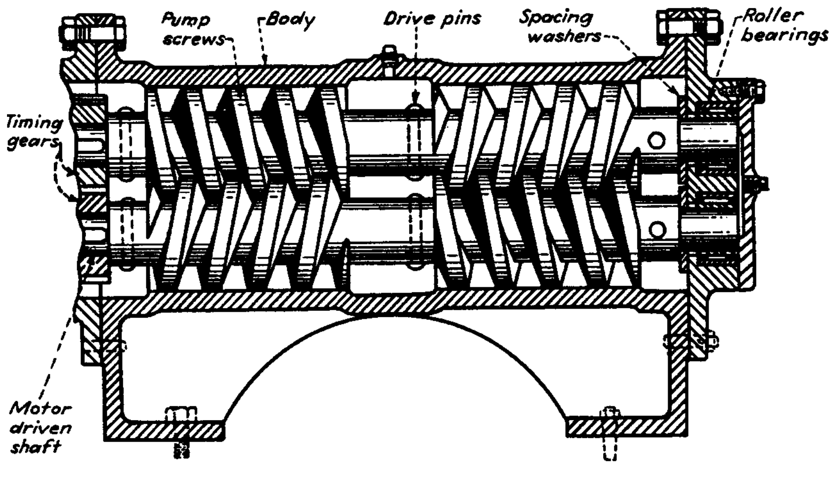
13. HIGH-VACUUM PUMPS operate with the rotating plunger action of liquid pumps. Sealing oil lubricates the three moving parts. Parts are accessible without disturbing connected piping. These pumps are used to rough pump a vacuum before connecting a diffusion pump; to evacute light bulbs and electronic tubes, and to vacuum dry and distill. Single pumps draw vacuums from 2 to 5 microns; in series to 0.5 micron, and compound pumps draw to 0.1 micron. They can be run in reverse for transferring liquids. Diagonal cored slots, closed by a slide pin, form the passageway and inlet valve. Popper or feature outlet valves are used.



14. A COMPACT MULTI-PLUNGER INTENSIFIER, this hydraulic booster is designed to convert low pressure to high pressure in any oil-hydraulic circuit. No additional pumps are required. Because of its six plungers, the pressure flow from the booster is both smooth and uninterrupted. High-pressure pumps are not required, and no operating valves are needed to control the high-pressure system. Small cylinder and ram assemblies can be used on operating equipment because the pressure is high. Operating costs can be low because of the efficient use of connected horsepower. The inertia effects of the small operating rams are low, so high speed operations can be attained. These boosters were built in two standard sizes, each of which was available in two pressure ranges: 2 to 1 and 3 to 1. Volumetric output is in inverse proportion to the pressure ratio. All units have a maximum 7,500 psi discharge pressure. Pistons are double-acting, and the central valve admits oil to pistons in sequence and is always hydraulically balanced.



15. THIS SELF-PRIMING PUMP gives a rapid and smooth transition from priming cycle to centrifugal pumping. The pump starts with its priming chamber full. Liquid is recirculated through the impeller until the pump is primed. As priming liquid circulates, air is drawn through impeller and expelled through the discharge. When all air is evacuated, discharge velocity closes the priming valve completely. These pumps can have open or closed impellers. Solids up to 1 in. can be passed through a 3 in. size pump with an open impeller.



16. INTERNAL SCREW PUMPS can easily transfer high-viscosity petroleum products. They can be used as boiler fuel pumps because they deliver a pulseless flow of oil. For marine or stationary systems, the characteristic low vibration of screw pumps has allowed them to be mounted on light foundations. The absence of vibration and pulsing flow reduces strain on pipes, hose and fittings. The pumping screws are mounted on shafts and take in liquid at both ends of the pump body and move it to the center for discharge. This balanced pumping action makes it unnecessary to use thrust bearings except in installations where the pump is mounted at a high angle. The pumps can be used at any angle up to vertical. Where thrust bearings are needed, antifriction bearings capable of supporting the load of the shaft and screws are used. The intermeshing pumping screws are timed by a pair of precision-cut herringbone gears. These are self-centering, and do not allow the side wear of the screws while they are pumping. The pump is most efficient when driven less than 1,200 rpm by an electric motor and 1,300 rpm by a steam turbine.

ROTARY-PUMP MECHANISMS

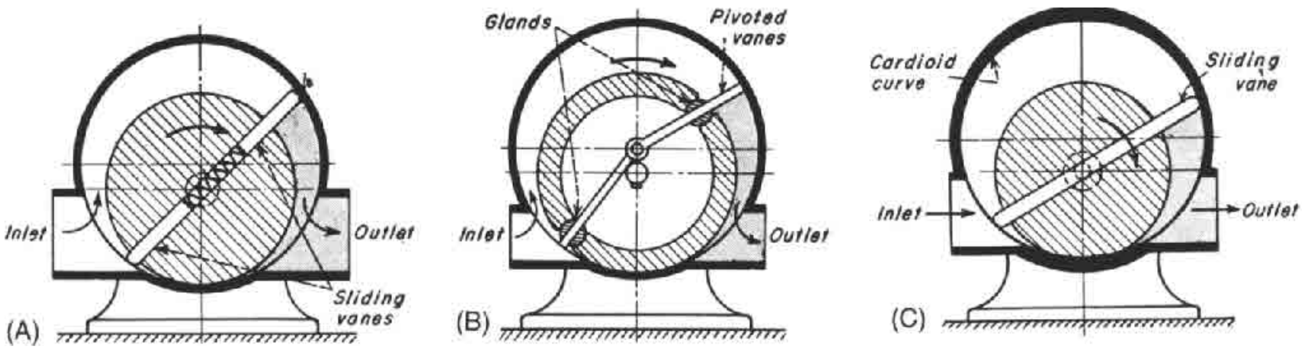


Fig. 1 (A) **A Ramelli pump** with spring-loaded vanes to ensure contact with the wall; vane ends are rounded for line contact. (B) Two vanes pivot in the housing and are driven by an eccentrically mounted disk; vanes slide in glands and are always radial to the housing, thus providing surface contact. (C) A housing with a cardioid curve allows the use of a single vane because opposing points on the housing in line with the disk center are equidistant.

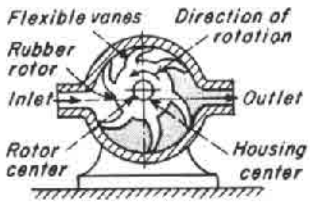


Fig. 2 **Flexible vanes** on an eccentric rubber rotor displace liquid as in sliding-vane pumps. Instead of the vanes sliding in and out, they bend against the casing to perform pumping.

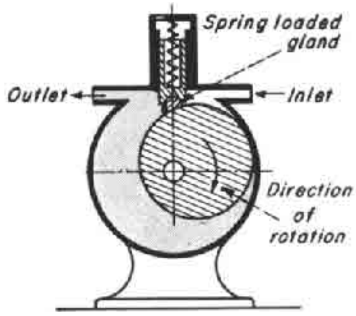


Fig. 3 **A disk** mounted eccentrically on the drive shaft displaces liquid in continuous flow. A spring-loaded gland separates the inlet from the outlet except when the disk is at the top of stroke.

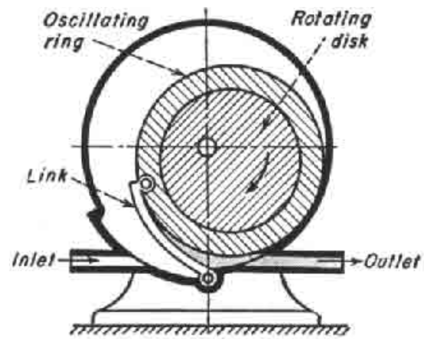


Fig. 4 **A rotary compressor pump** has a link separating its suction and compression sides. The link is hinged to a ring which oscillates while it is driven by the disk. Oscillating action pumps the liquid in a continuous flow.

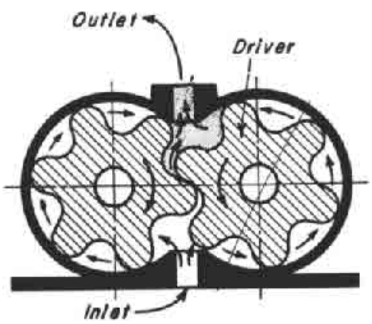


Fig. 5 **A gear pump** transports liquid between the tooth spaces and the housing wall. A circular tooth shape ha sonly one tooth making contact, and it is more efficient than an involute shape which might enclose a pocket between two adjoining teeth, recirculating part of the liquid. The pump has helical teeth.

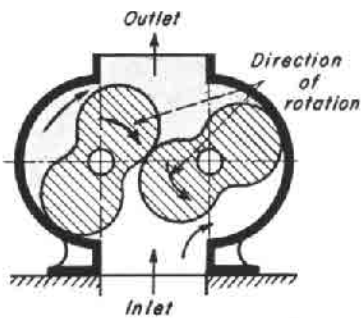
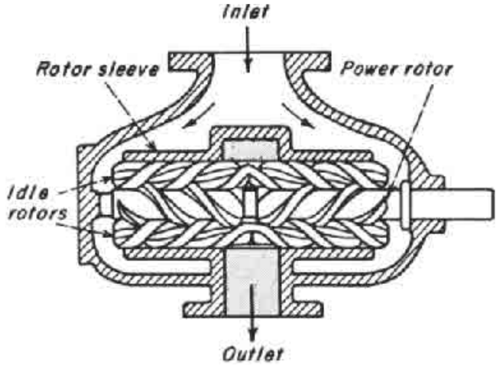


Fig. 6 **A Roots compressor** has two identical impellers with specially shaped teeth. The shafts are connected by external gearing to ensure constant contact between the impellers.

Fig. 7 **A three-screw pump** drives liquid between the screw threads along the axis of the screws. The idle rotors are driven by fluid pressure, not by metallic contact with the power rotor.



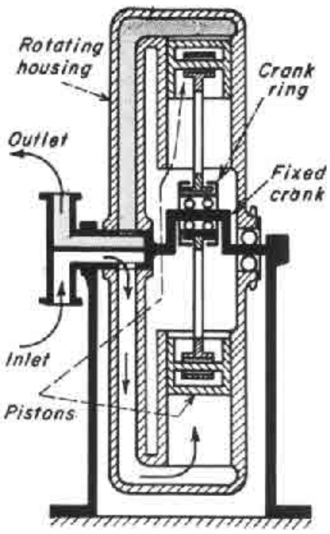


Fig. 8 The housing of the Hele-Shaw-Beachum pump rotates the round-cranked shaft. Connecting rods attached to the crank ring cause the pistons to oscillate as the housing rotates. No valves are necessary because the fixed hollow shaft, divided by a wall, has suction and compression sides that are always in correct register with the inlet and outlet ports.

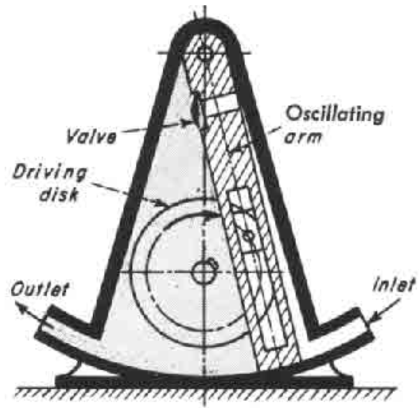


Fig. 9 A disk drives the oscillating arm which acts as piston. The velocity of the arm varies because of its quick-return mechanism. Liquid is slowly drawn in and expelled during the clockwise rotation of the arm; the return stroke transfers the liquid rapidly.

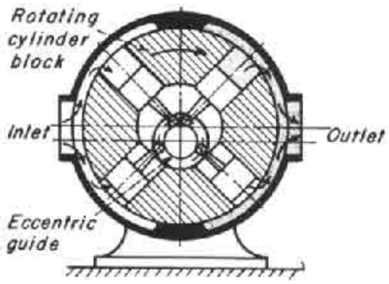


Fig. 10 A rotating cylinder block is mounted concentrically in a housing. Connecting-rod ends slide around an eccentric guide as the cylinders rotate and cause the pistons to reciprocate. The housing is divided into suction and compression compartments.

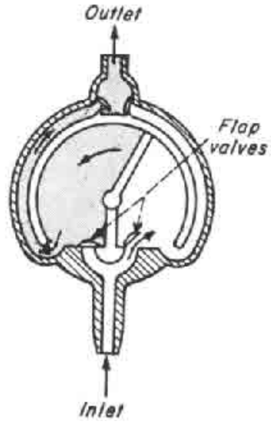
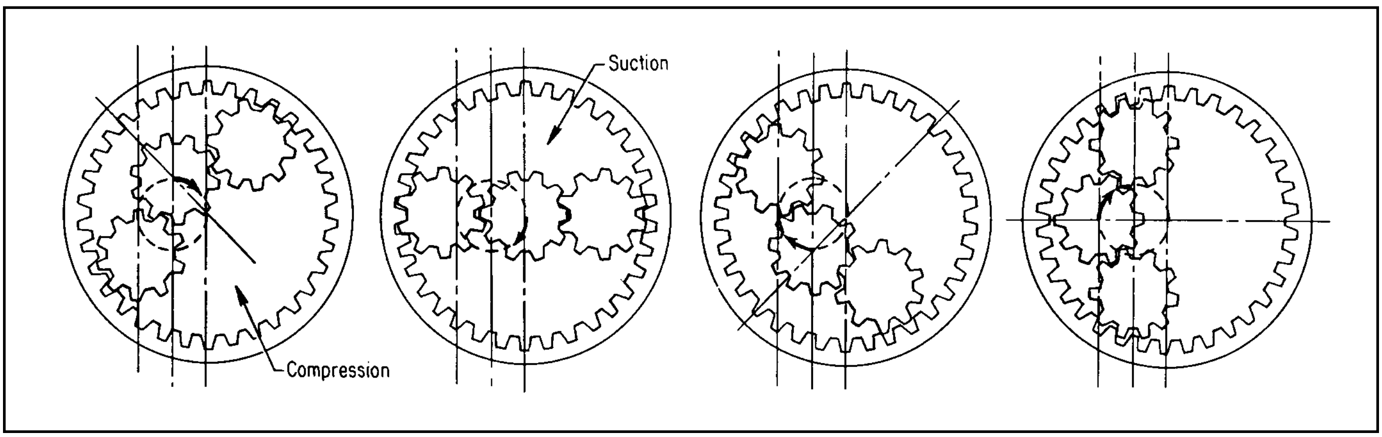


Fig. 11 A rotary-reciprocating pump that is normally operated manually to pump high-viscosity liquids such as oil.

OFFSET PLANETARY GEARS INDUCE ROTARY-PUMP ACTION



Two planetary gears are driven by an offset sun gear to provide the pumping action in this positive-displacement pump. A successively increasing/decreasing (suction/compression) is formed on either side of the sun and planet gears.

MECHANISMS ACTUATED BY PNEUMATIC OR HYDRAULIC CYLINDERS

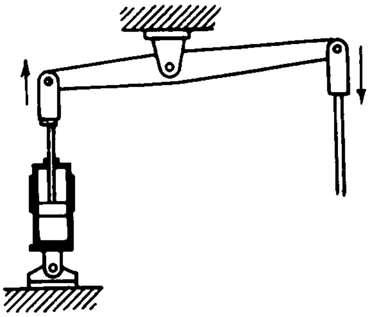


Fig. 1 A cylinder can be used with a first-class lever.

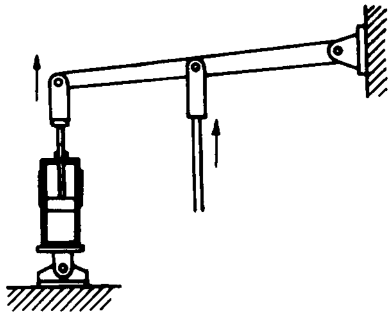


Fig. 2 A cylinder can be used with a second-class lever.

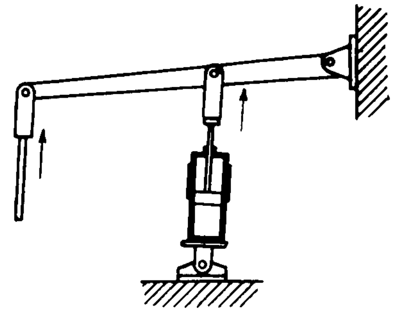


Fig. 3 A cylinder can be used with a third-class lever.

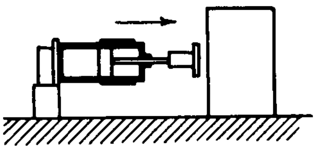


Fig. 4 A cylinder can be linked up directly to the load.

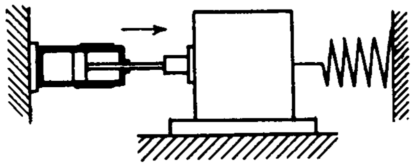


Fig. 5 A spring reduces the thrust at the end of the stroke.

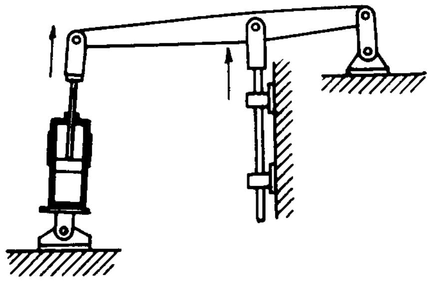


Fig. 6 The point of application of force follows the direction of thrust.

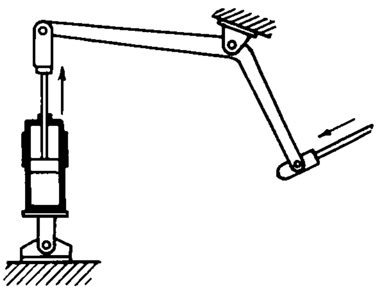


Fig. 7 A cylinder can be used with a bent lever.

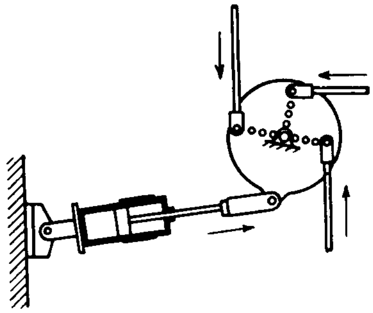


Fig. 8 A cylinder can be used with a trammel plate.

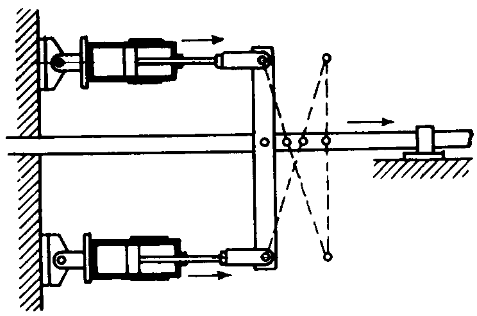


Fig. 9 Two pistons with fixed strokes position the load in any of four stations.

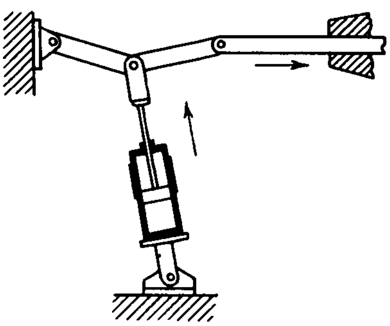


Fig. 10 A toggle can be actuated by the cylinder.

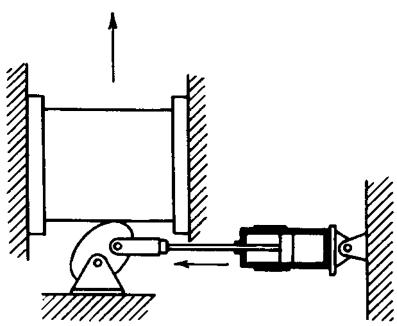


Fig. 11 The cam supports the load after the completion of the stroke.

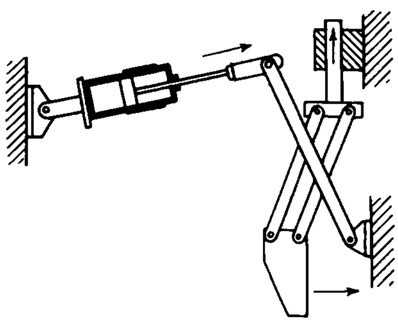


Fig. 12 Simultaneous thrusts in two different directions are obtained.

(Note: In place of cylinders, electrically powered thrust units or solenoids can be used.)

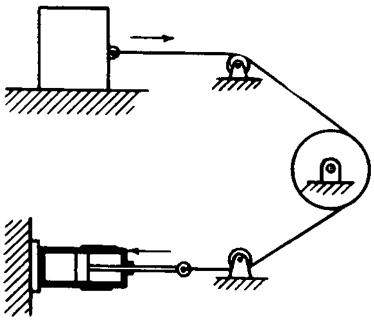


Fig. 13 **A force** is transmitted by a cable.

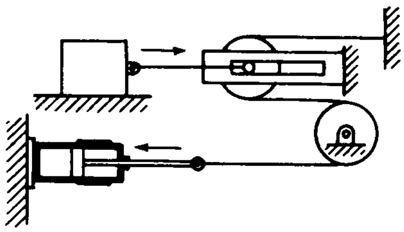


Fig. 14 **A force** can be modified by a system of pulleys.

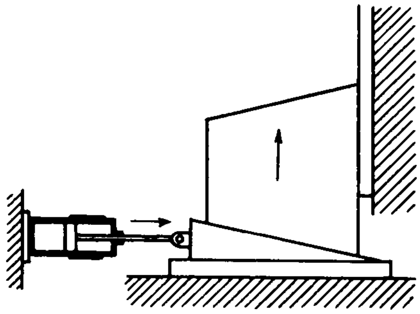


Fig. 15 A force can be modified by wedges.

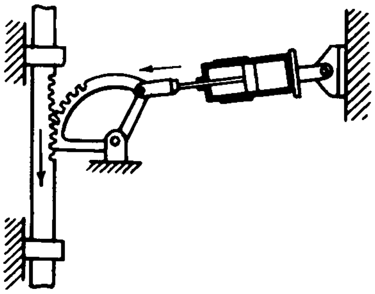


Fig. 16 **A gear sector** moves the rack perpendicular to the piston stroke.

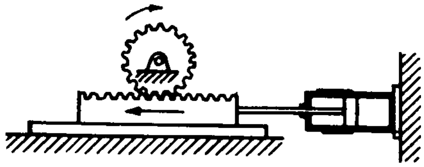


Fig. 17 **A rack** turns the gear sector.

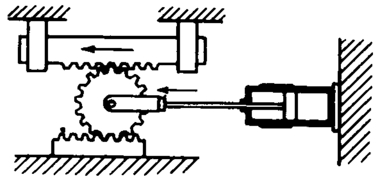


Fig. 18 **The motion** of a movable rack is twice that of the piston.

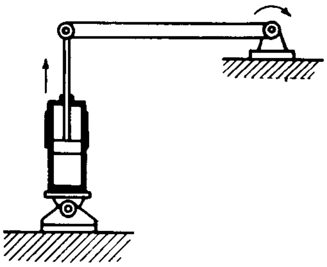


Fig. 19 **A torque** applied to the shaft can be transmitted to a distant point.

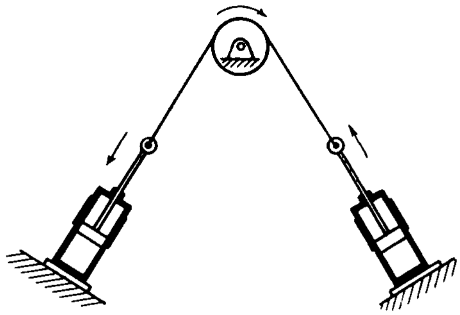


Fig. 20 **A torque** can also be applied to a shaft by a belt and pulley.

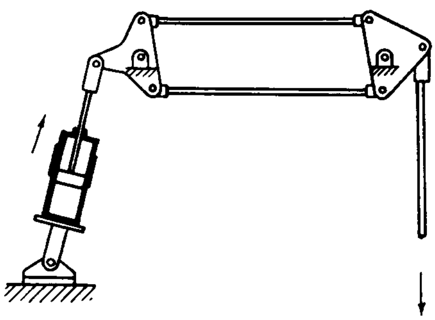


Fig. 21 **A motion** is transmitted to a distant point in the plane of motion.

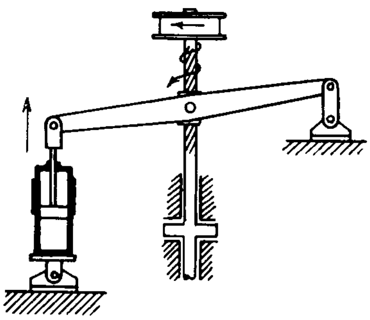


Fig. 22 **A steep screw nut** produces a rotation of the shaft.

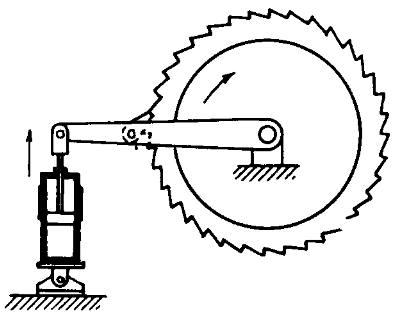


Fig. 23 **A single-sprocket wheel** produces rotation in the plane of motion.

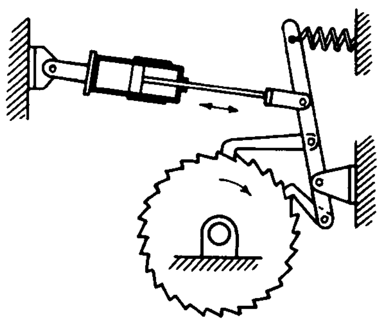
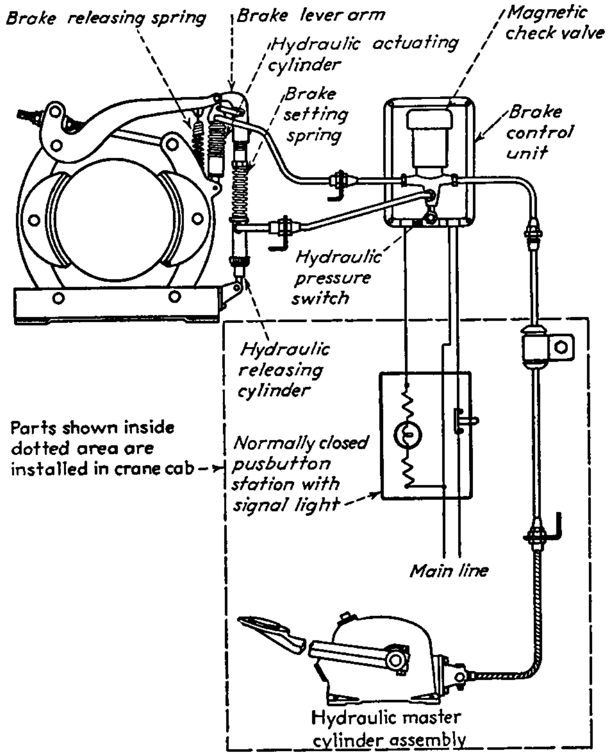


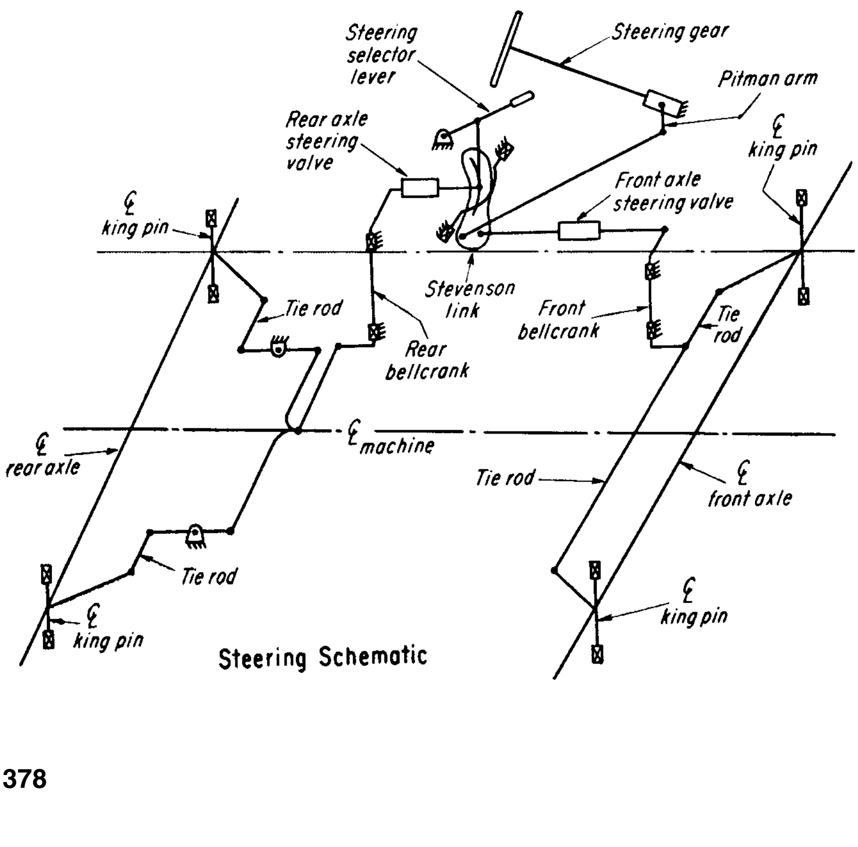
Fig. 24 **A double-sprocket wheel** makes the rotation more nearly continuous.

FOOT-CONTROLLED BRAKING SYSTEM

This crane braking system (see figure) operates when the main line switch closes. The full depression of the master-cylinder foot-pedal compresses the brake-setting spring mounted on the hydraulic releasing cylinder. After the setting spring is fully compressed, the hydraulic pressure switch closes, completing the electric circuit and energizing the magnetic check valve. The setting spring remains compressed as long as the magnetic check valve is energized because the check valve traps the fluid in the hydraulic-releasing cylinder. Upon release of the foot peal, the brake lever arm is pulled down by the brake releasing spring, thus releasing the brake shoes.



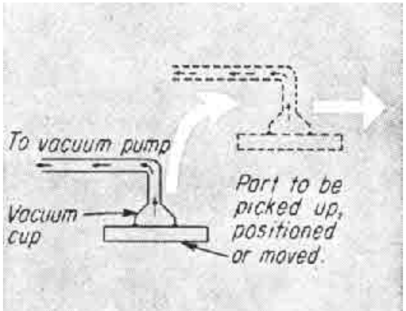
LINKAGES ACTUATE STEERING IN A TRACTOR



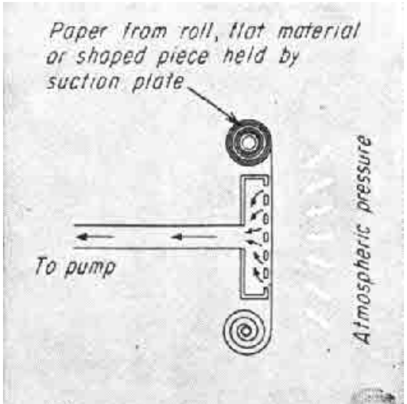
Hydraulic power for operating the brakes, clutch, and steering of a 300 hp diesel-powered tractor is supplied by an engine-driven pump delivering 55 gpm at 1200 psi. The system is designed to give a 15-gpm preference to the steering system. The steering drive to each wheel is mechanical for synchronization, with mechanical selection of the front-wheel, four-wheel or crab-steering hookup; hydraulic power amplifies the manual steering effort.

FIFTEEN JOBS FOR PNEUMATIC POWER

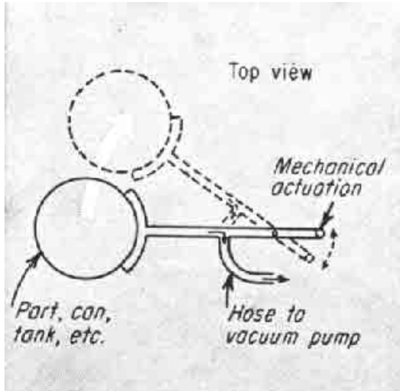
Suction can feed, hold, position, and lift parts, form plastic sheets, sample gases, test for leaks, convey solids, and de-aerate liquids. Compressed air can convey materials, atomize and agitate liquids, speed heat transfer, support combustion, and protect cable.



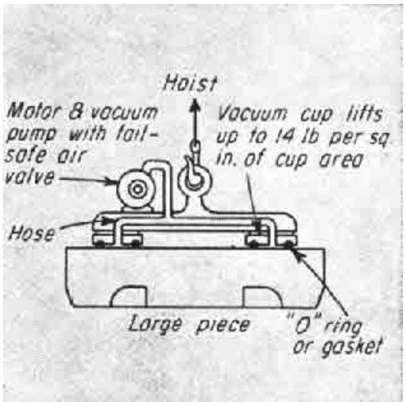
FEEDING PARTS



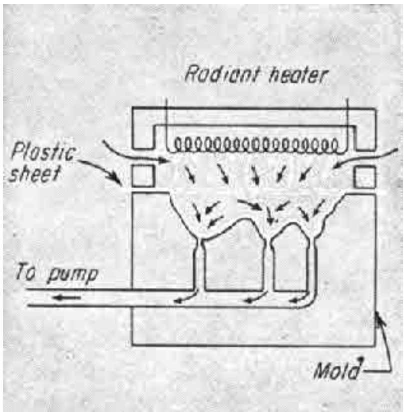
HOLDING PARTS



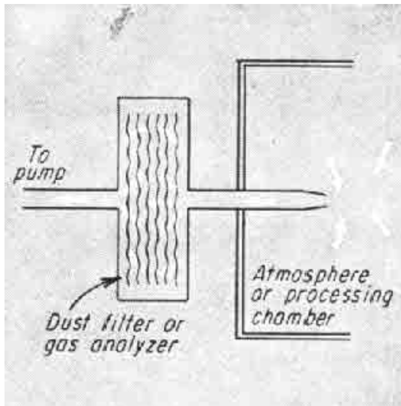
POSITIONING PARTS



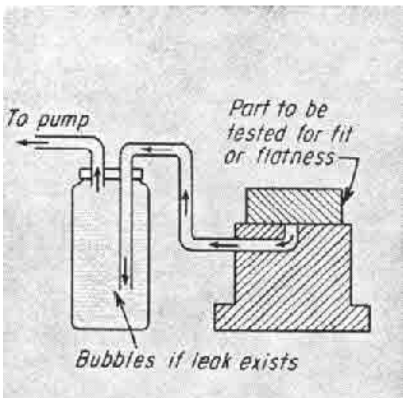
LIFTING PARTS



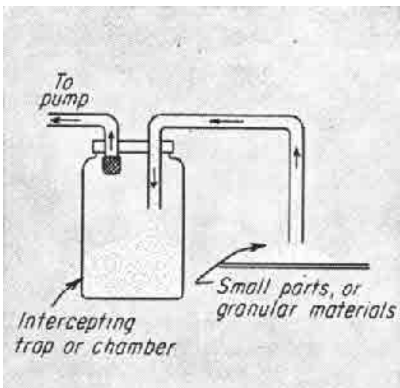
VACUUM FORMING



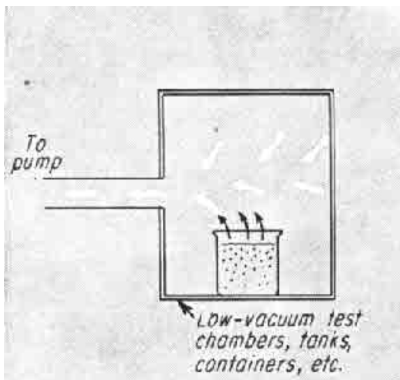
GAS SAMPLING



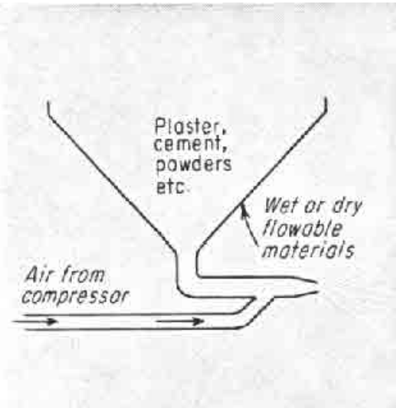
LEAK TESTING



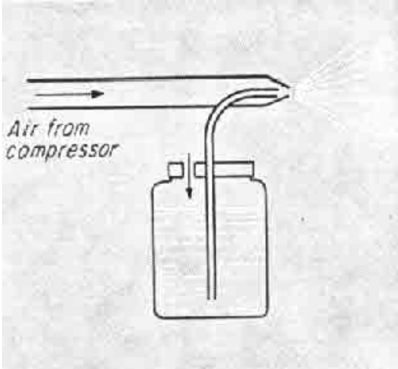
CONVEYING SOLIDS



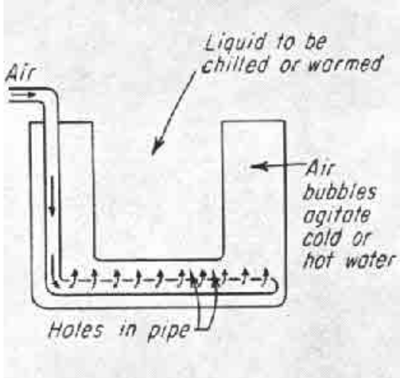
DE-AERATING HIGH-VISCOSITY LIQUIDS



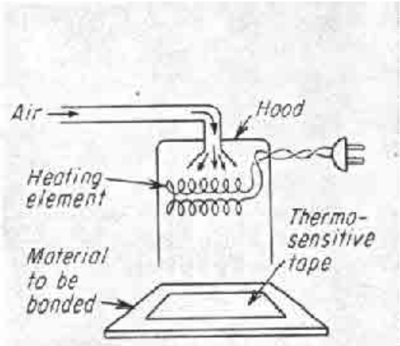
CONVEYING MATERIALS



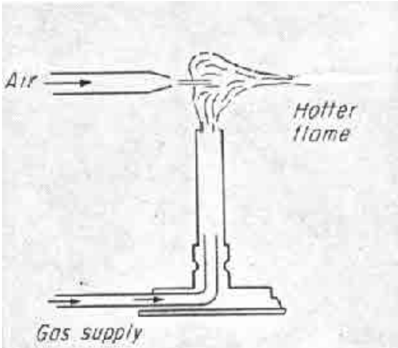
ATOMIZING LIQUIDS



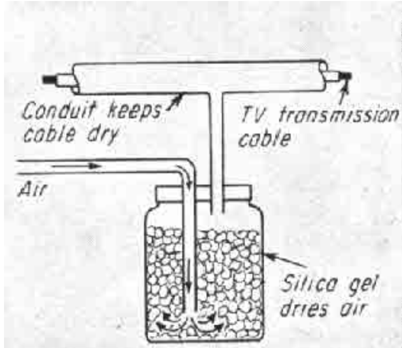
AGITATING LIQUIDS



AIR COOLING OR HEATING



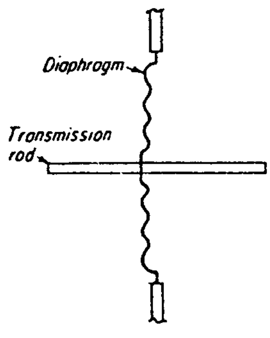
SUPPORTING COMBUSTION



PRESSURIZING CABLE

TEN WAYS TO USE METAL DIAPHRAGMS AND CAPSULES

Fig. 1



A metal diaphragm is usually corrugated (Fig. 1) or formed to some irregular profile. It can be used as a flexible seal for an actuating rod. The capsule (Fig. 2) is an assembly of two diaphragms sealed together at their outer edges, usually by soldering, brazing, or welding. Two or more capsules assembled together are known as a capsular element (Fig. 3). End fittings for the capsules vary according to their function; the "fixed end" is fixed to the equipment. The "free end" moves the related components and linkages. The nested capsule (Fig. 4) requires less space and can be designed to withstand large external overpressures without damage.

Fig. 2

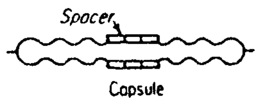


Fig. 3

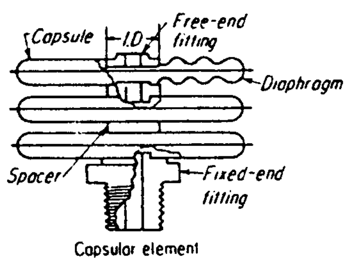


Fig. 4

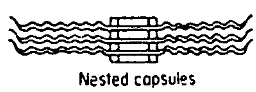
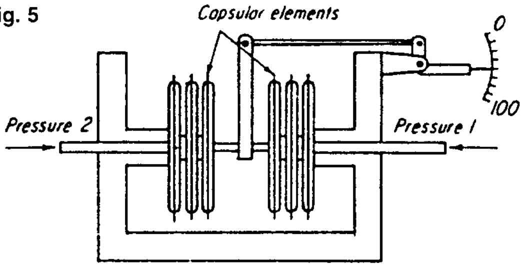
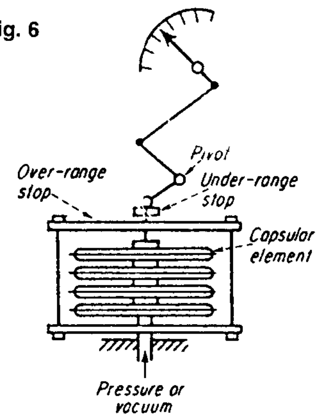


Fig. 5



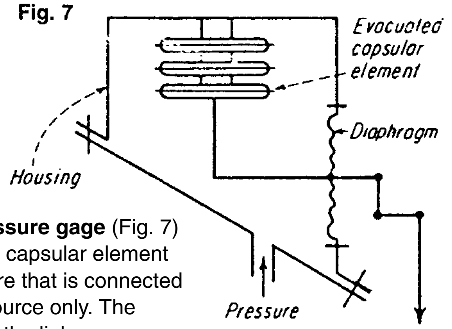
A differential pressure gage (Fig. 5) with opposing capsules can have either single or multicapsular elements. The multicapsular type gives greater movement to the indicator. Capsules give improved linearity over bellows for such applications as pressure-measuring devices. The force exerted by any capsule is equal to the total effective area of the diaphragms (about 40% actual area) multiplied by the pressure exerted on it. Safe pressure is the maximum pressure that can be applied to a diaphragm before hysteresis or set become objectionable.

Fig. 6



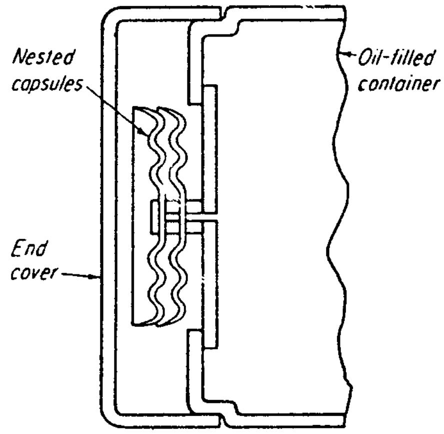
A pressure gage (Fig. 6) has a capsular element linked to a dial indicator by a three-bar linkage. Such a gage measures pressure or vacuum relative to prevailing atmospheric pressure. If greater angular motion of the indicator is required than can be obtained from the three-bar linkage, a quadrant and gear can be substituted.

Fig. 7



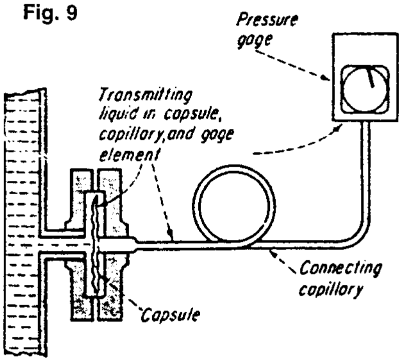
An absolute pressure gage (Fig. 7) has an evacuated capsular element inside an enclosure that is connected to the pressure source only. The diaphragm allows the linkage movement from the capsule to pass through a sealed chamber. This arrangement can also be used as a differential pressure gage by making a second pressure connection to the interior of the element.

Fig. 8



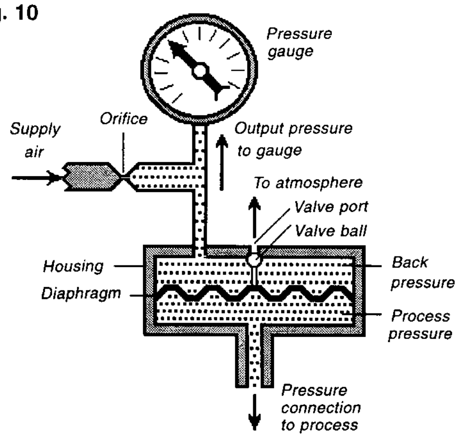
An expansion compensator (Fig. 8) for oil-filled systems takes up less space when the capsules are nested. In this application, one end of the capsule is open and connected to oil in the system; the other end is sealed. Capsule expansion prevents the internal oil pressure from increasing dangerously from thermal expansion. The capsule is protected by its end cover.

Fig. 9



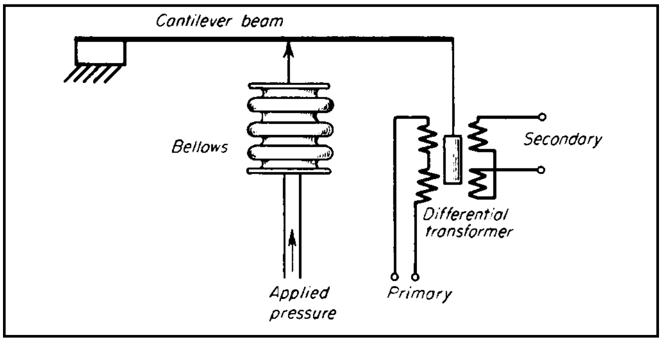
A capsule pressure-seal (Fig. 9) works like a thermometer system except that the bulb is replaced by a pressure-sensitive capsule. The capsule system is filled with a liquid such as silicone oil and is self-compensating for ambient and operating temperatures. When subjected to external pressure changes, the capsule expands or contracts until the internal system pressure just balances the external pressure surrounding the capsule.

Fig. 10

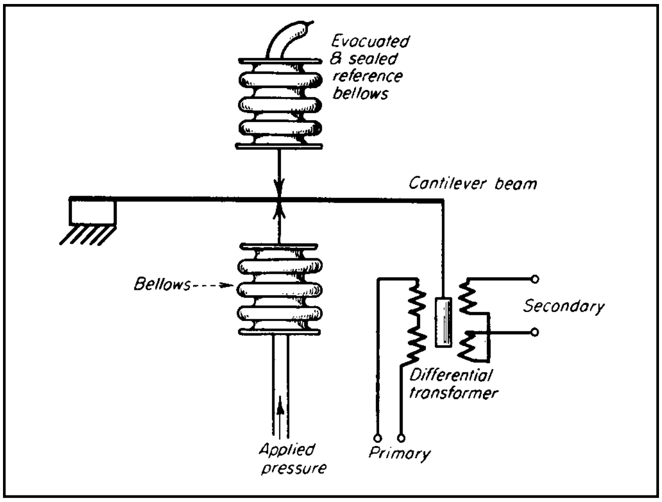


A force-balanced seal (Fig. 10) solves the problem, as in the seal of Fig. 9, for keeping corrosive, viscous or solids-bearing fluids out of the pressure gage. The air pressure on one side of a diaphragm is controlled so as to balance the other side of the diaphragm exactly. The pressure gage is connected to measure this balancing air pressure. The gage, therefore, reads an air pressure that is always exactly equal to the process pressure.

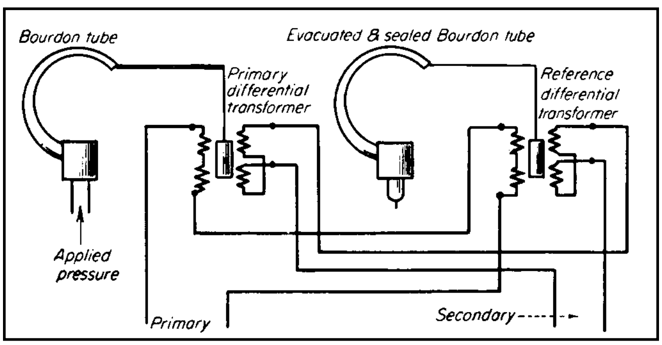
DIFFERENTIAL TRANSFORMER SENSING DEVICES



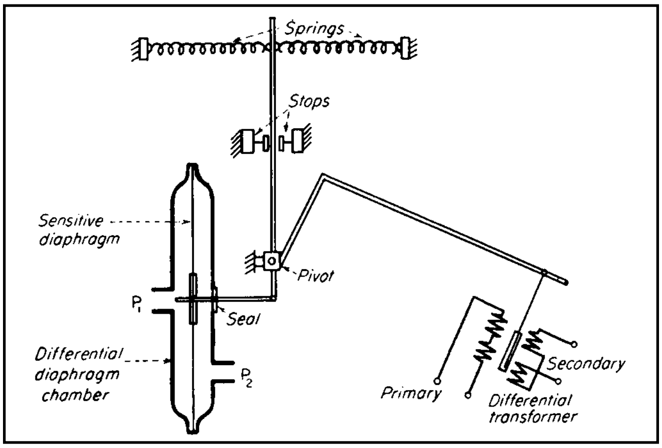
Gage pressure bellows transmitter. The bellows is connected to a cantilever beam with a needle bearing. The beam adopts a different position for every pressure; the transformer output varies with beam position. The bellows are available for ranges from 0–10 in. to 0–200 in. of water for pressure indication or control.



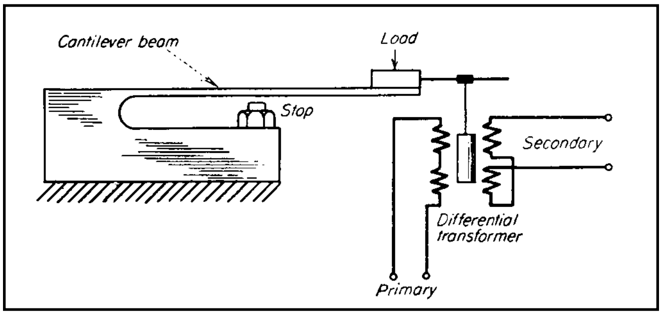
Absolute pressure bellows transmitter. This transmitter is similar to the differential diaphragm transmitter except for addition of a reference bellows which is evacuated and sealed. It can measure negative gage pressures with ranges from 0–50 mm to 0–30 in. of mercury. The reference bellows compensates for variations in atmospheric pressure.



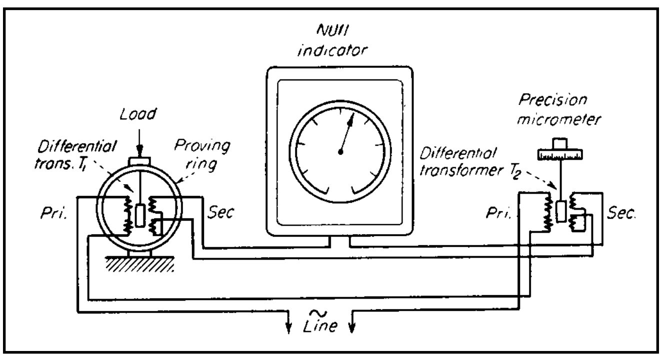
Absolute pressure Bourdon-tube transmitter. This device can indicate or control absolute pressures from 15 to 10,000 psi, depending on tube rating. The reference tube is evacuated and sealed, and compensates for variations in atmospheric pressure by changing the output of the reference differential transformer. The signal output consists of the algebraic sum of the outputs of both the primary and reference differential transformers.



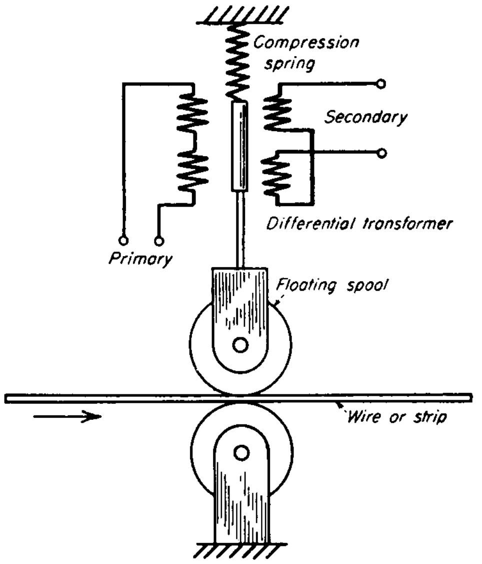
Differential diaphragm pressure transmitter. Differential pressures P_1 and P_2 act on the opposite sides of a sensitive diaphragm and move the diaphragm against the spring load. The diaphragm displacement, spring extension, and transformer core movement are proportional to the difference in pressure. The device can measure differentials as low as 0.005 in. of water. It can be installed as the primary element in a differential pressure flowmeter, or in a boiler windbox for a furnace-draft regulator.

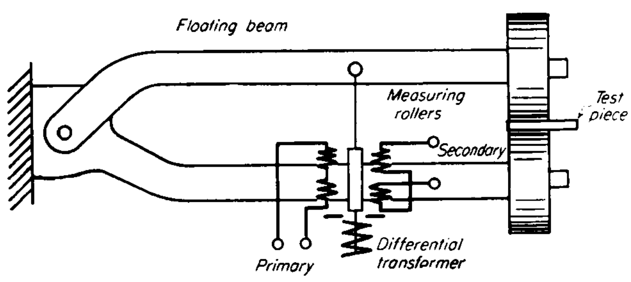


Cantilever load cell. The deflection of a cantilever beam and the displacement of a differential transformer core are proportional to the applied load. The stop prevents damage to the beam in the event of overload. Beams are available for ranges from 0–5 to 0–500 lb. And they can provide precise measurement of either tension or compression forces.

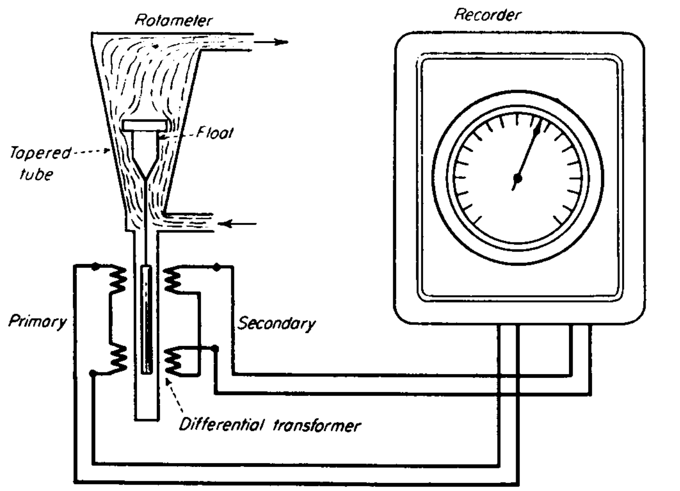


Proving ring. The core of the transmitting transformer, T_1, is fastened to the top of the proving ring, while the windings are stationary. The proving ring and transformer core deflect in proportion to the applied load. The signal output of the balancing transformer, T_2, opposes the output of T_1, so that at the balance point, the null point indicator reads zero. The core of the balancing transformer is actuated by a calibrated micrometer that indicates the proving ring deflection when the differential transformer outputs are equal and balanced.

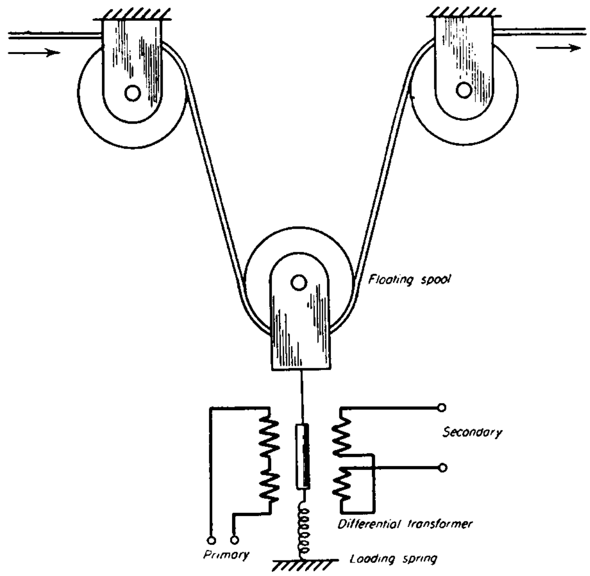




Gaging and callipering. The thickness of a moving wire or strip is gaged by the position of the floating spool and transformer core. If the core is at the null point for proper material thickness, the transformer output phase and magnitude indicate whether the material is too thick or thin and the amount of the error. The signal can be amplified to operate a controller, recorder, or indicator. The device at the right can function as a production caliper or as an accurate micrometer. If the transformer output is fed into a meter indicator with *go* and *no-go* bands, it becomes a convenient device for gaging items.



Flow meter. The flow area varies as the float rises or falls in the tapered tube. High flows cause the float to rise, and low flows cause it to drop. The differential transformer core follows the float travel and generates an AC signal which is fed into a square-root recorder. A servo can be equipped with a square root cam to read on a linear chart. The transformer output can also be amplified and used to actuate a flow regulating valve so that the flowmeter becomes the primary element in a flow controller. Normally meter accuracy is better than 2%, but its flow range is limited.



Tension control. The loading spring can be adjusted so that when the transformer core is at its null point, the proper tension is maintained in the wire. The amplified output of the transformer is transmitted to some kind of tension-controlling device which increases or reduces the tension in the wire, depending on the phase and magnitude of the applied differential transformer signal.

HIGH-SPEED COUNTERS

The electronic counter counts electrical pulses and gives a running display of accumulated pulses at any instant. Because the input is an electrical signal, a transducer is generally required to transform the nonelectrical signal into a usable input for the counter.

With a preset function on the counter, any number can be selected within the count capacity of the device. Once the counter reaches the preset number, it can open or close the relay to control some operation. The counter will either reset automatically or stop. A dual unit permits continuous control over two different count sequence operations. Two sets of predetermining switches are usually mounted on the front panel of the counter, but they can be mounted at a remote location. If two different numbers are programmed into the counter, it will alternately count the two selected numbers. Multiple presets are also available, but at higher cost.

In addition to performing two separate operations, a dual preset can control speeds, as shown in Fig. 1. In the metal shearing operation run at high speed, one preset switch can be used to slow the material down at a given distance before the second preset actuates the shearing. Then both switches automatically reset and start to measure again. The same presets could also be used. The same presets could also be used to alternately shear the material into two different lengths.

One form of measurement well adapted to high-speed counters is the measuring of continuous materials such as wire, rope, paper, textiles, or steel. Fig. 2 shows a coil-winding operation in which a counter stops the machine at a predetermined number of turns of wire.

A second application is shown in Fig. 3. Magazines are counted as they run off a press. A photoelectric pickup senses the alternate light and dark lines formed by the shadows of the folded edges of the magazines. At the predetermined number, a knife edge, actuated by the counter, separates the magazines into equal batches.

A third application is in machine-tool control. A preset counter can be paired with a transducer or pulse generator mounted on the feed mechanism. It could, for example, convert revolutions of screw feed, hence displacement, into pulses to be fed into the counter. A feed of 0.129 in. might represent a count of 129 to the counter.

When preset at that number, the counter could stop, advance, or reverse the feed mechanism.

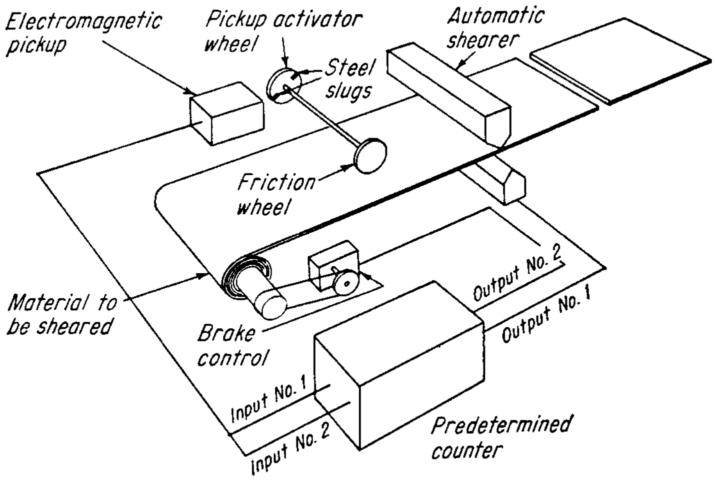


Fig. 1 A dual preset function on a high-speed counter controls the high-speed shearing operation. If the material is to be cut in 10-ft lengths and each pulse of the electromagnetic pickup represents 0.1 ft, the operator presets 100 into the first input channel. The second input is set to 90. When 90 pulses are counted, the second channel slows the material. Then when the counter reaches 100, the first channel actuates the shear. Both channels reset instantaneously and start the next cycle.

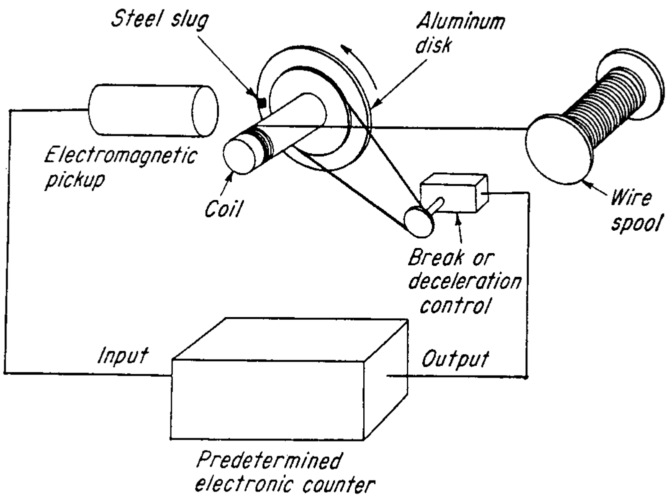


Fig. 2 A coil-winding machine with electronic counting for measuring length.

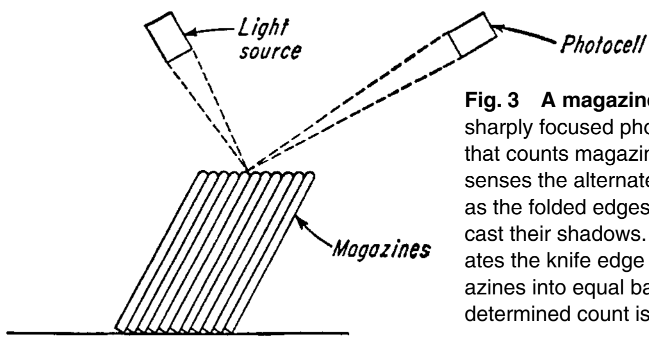


Fig. 3 A magazine counter has a sharply focused photoelectric pickup that counts magazines. The photocell senses the alternate light and dark lines as the folded edges of the magazines cast their shadows. The counter actuates the knife edge to separate the magazines into equal batches when the predetermined count is reached.

DESIGNING WITH PERMANENT MAGNETS

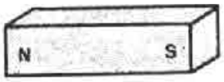


Fig. 1

Fig. 1 The simplest form of permanent magnet is a bar that has two poles which can have any cross section.

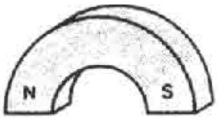
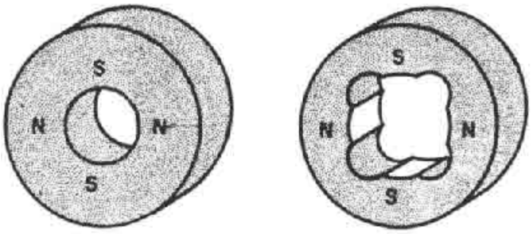


Fig. 2

Fig. 2 U or C-shaped permanent magnets are cast that way to bring both of the pole faces into the same plane.



(A) Fig. 3 (B)

Fig. 3 A cylindrical magnet can be magnetized with as many pairs of poles as desired on the outside diameter as in (A) or the inner diameter as in (B). Also, they can be made nonsalient (A) or salient (B).

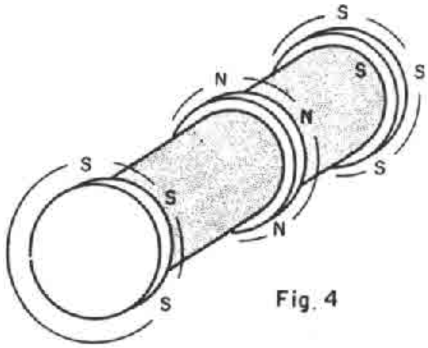


Fig. 4

Fig. 4 This magnetic roll for material separators is made from magnets and steel pole pieces that supply an equal magnetic field on 360º of the pole surface.

Fig. 5 Magnets for generators and other machines can be assembled from multiple magnets with laminated (A) or solid pole pieces (E), cast with inserts for pressing in shafts, nonsalient (B) or salient (C) or cast for assembly on shafts (D).

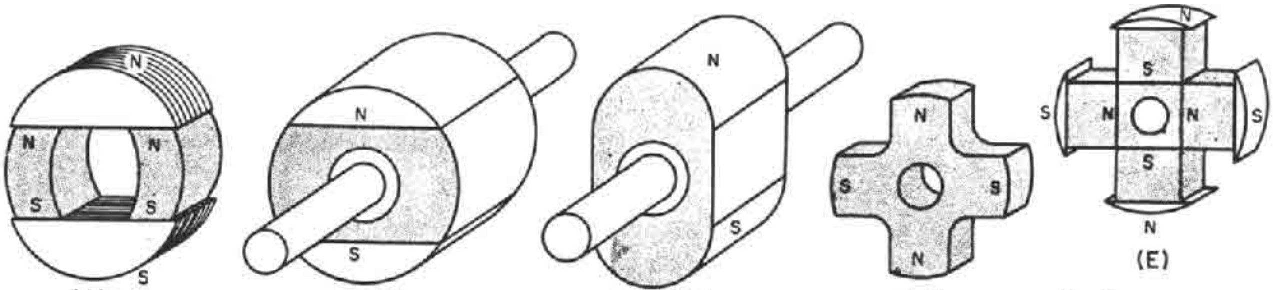


Fig. 5

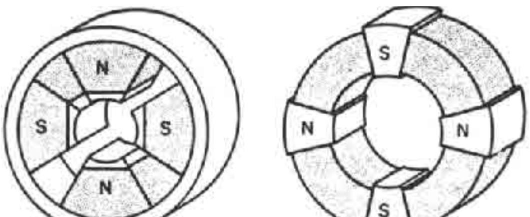
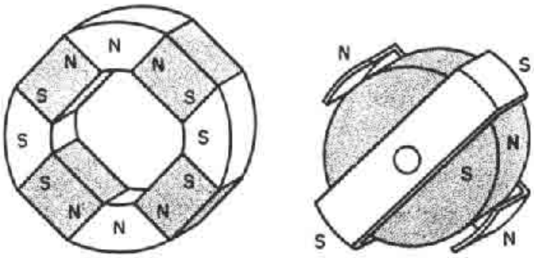


Fig. 6

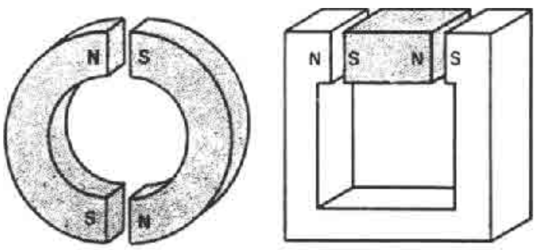
Fig. 6 Stator or internal pole assembles using steel pole faces and magnets are made in various ways depending on the mechanical space, magnetic and physical characteristics required.

Fig. 7 A four-pole magnet is shown with steel pole pieces, but it is possible to incorporate as many poles as are required (A) using bar magnets. In style (B) it is possible to obtain several poles by using one two-pole magnet.

Fig. 8 These permanent magnet assembles have double air gap. (A) has no steel poles, but (B) has them.



(A) Fig 7 (B)



(A) Fig. 8 (B)

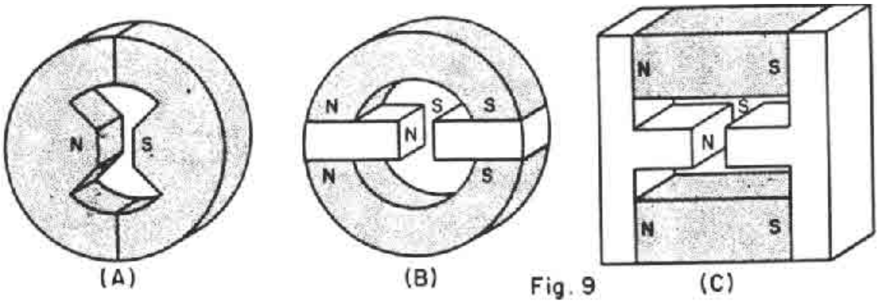
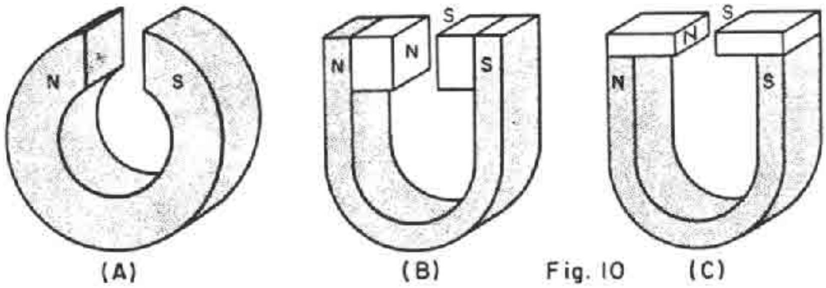


Fig. 9 Fixed air-gap magnets can be made from a multiplicity of magnets without steel pole pieces (A), or with pole pieces (B) and (C).

Fig. 10 Simple fixed air-gap magnets can be without pole pieces (A) or with pole pieces (B) and (C).



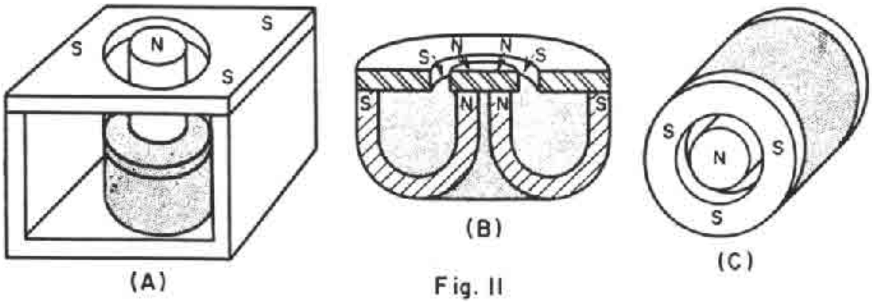


Fig. 11 Annular air gaps are useful in instruments and loudspeakers. The magnets can be (A) plug, (B) cylindrical, or (C) ring-type, depending upon the application.

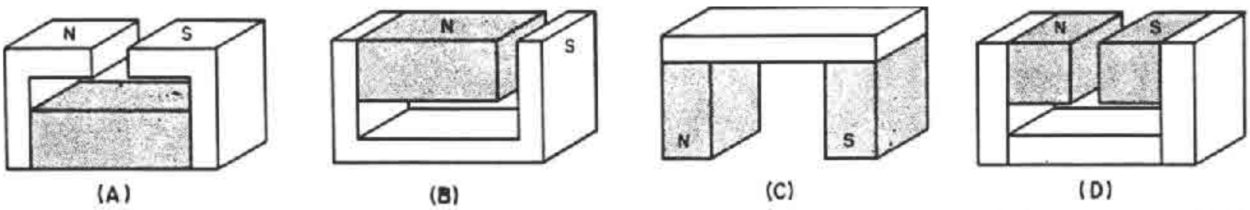
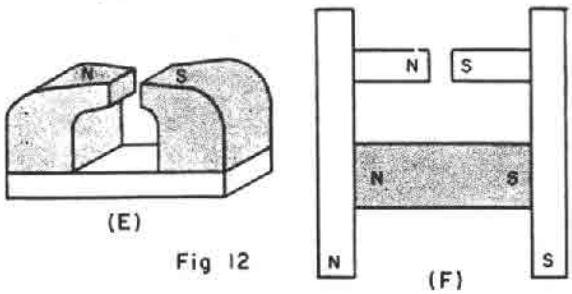


Fig. 12 Soft steel bars are used to complete the magnetic circuit, allowing maximum flux density through the air gap. The arrangement can use single magnets (A), (B), or double magnets (C), (D), (E), or a single magnet with a shunt (F).



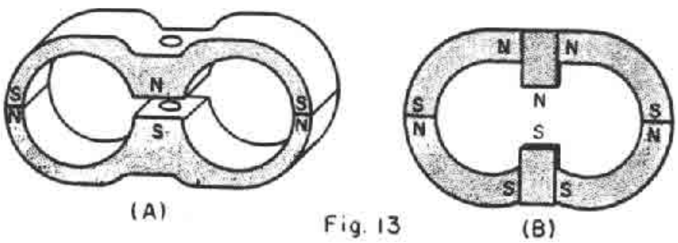


Fig. 13 Special magnets are cast for microwave power tubes—three-pole E style (A), and concentric-gap bowl type (B). Many other forms are possible.

PERMANENT MAGNET MECHANISMS

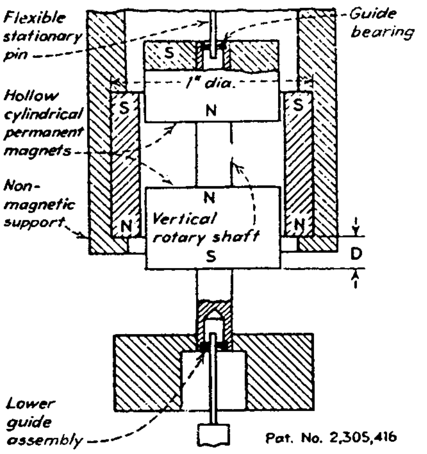


Fig. 1 A suspension.

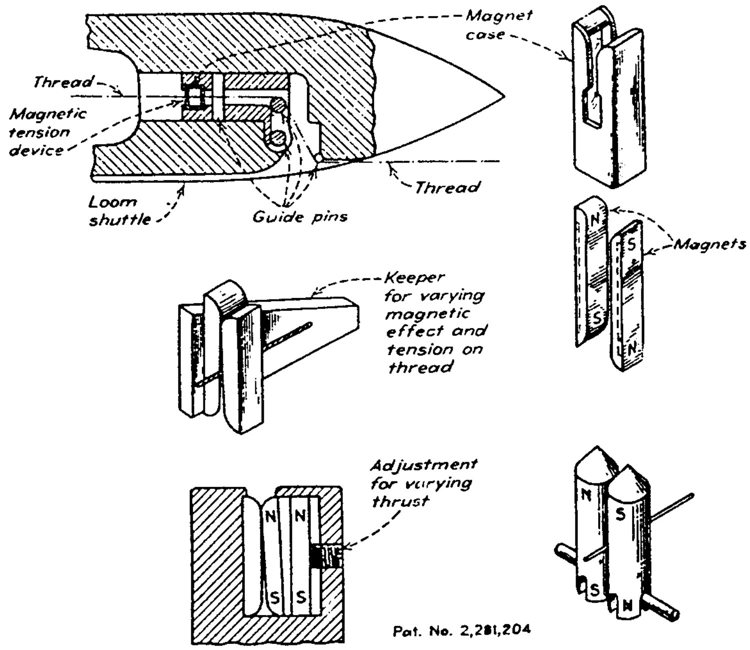


Fig. 2 Tension devices.

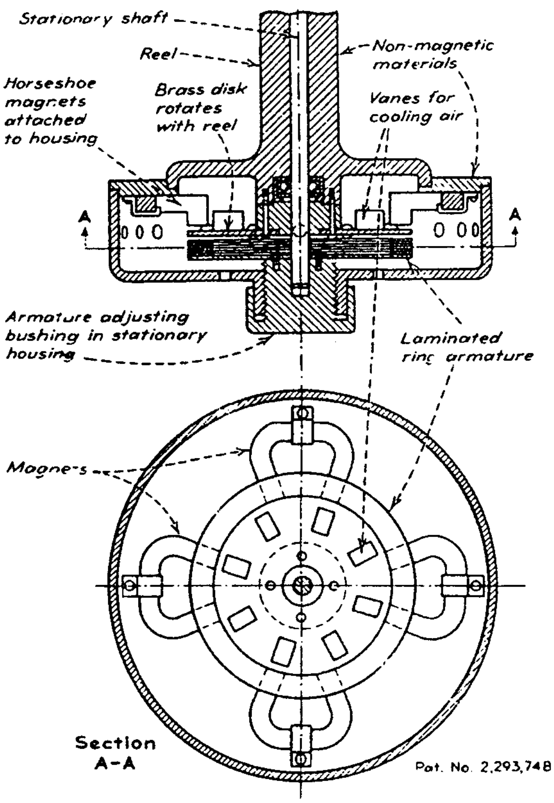


Fig. 3 A reel brake.

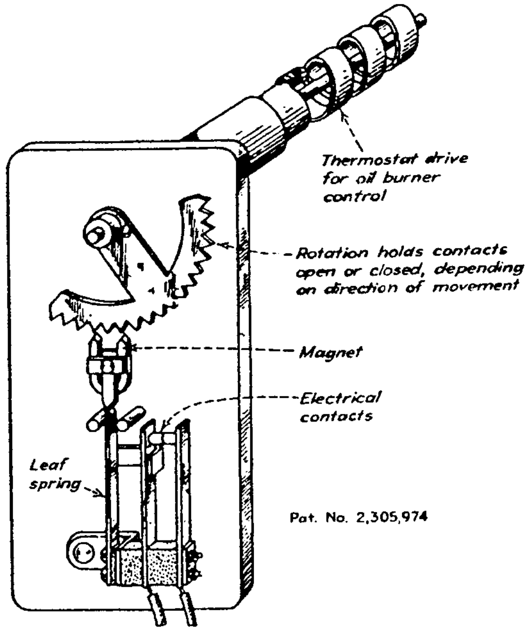


Fig. 4 An instrument coupling.

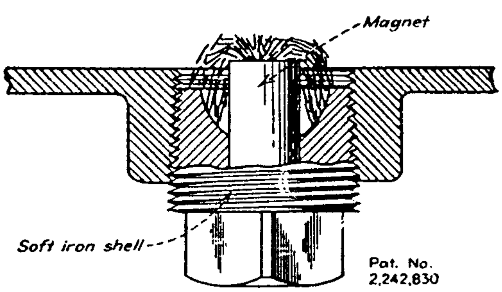


Fig. 5 A crankcase oil drain plug.

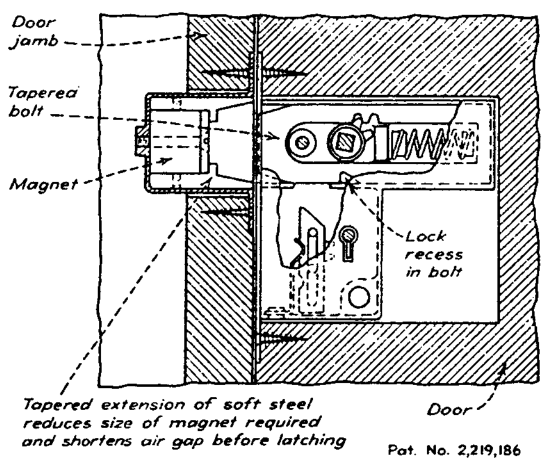


Fig. 6 A non-rattling door latch.

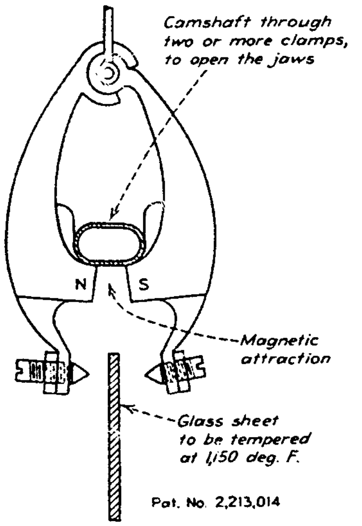


Fig. 7 A clamp.

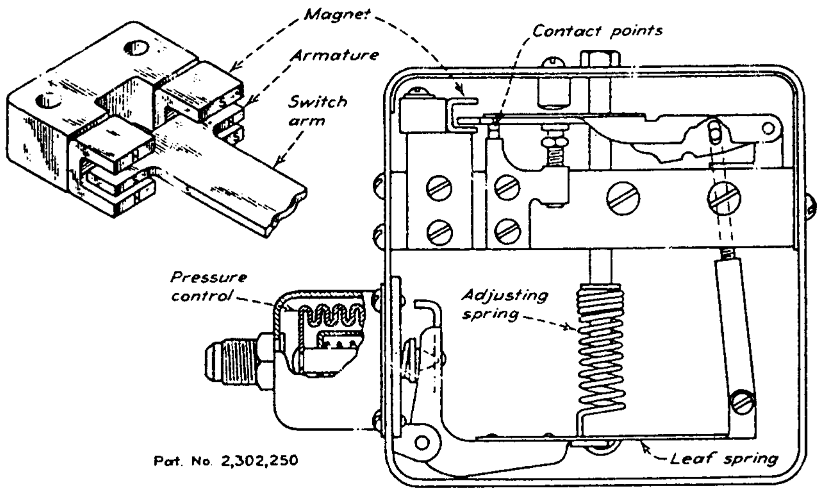


Fig. 8 A snap-action switch.

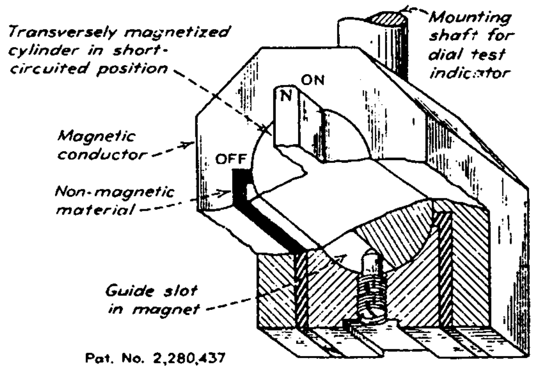


Fig. 9 An instrument holder.

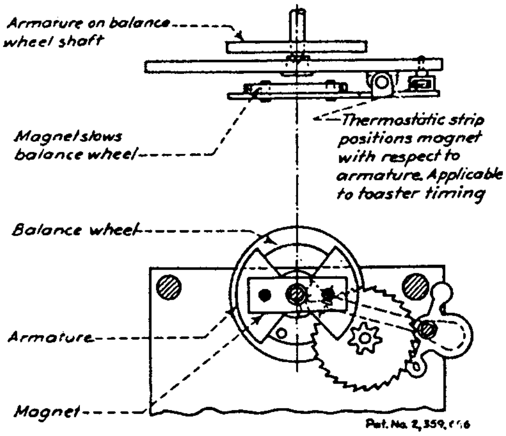


Fig. 10 An escape wheel.

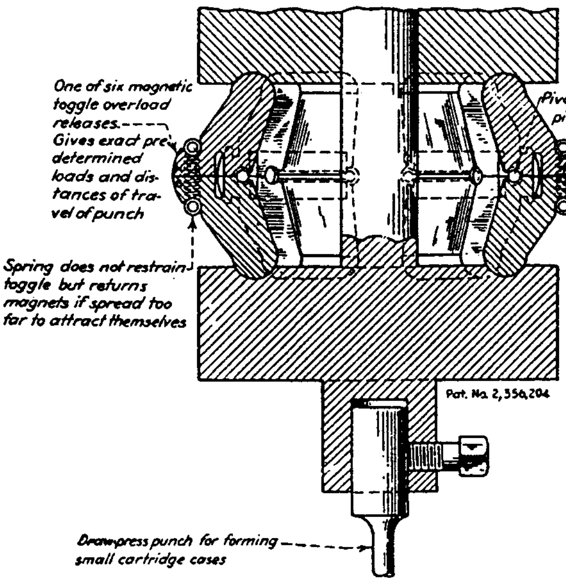


Fig. 11 A pressure release.

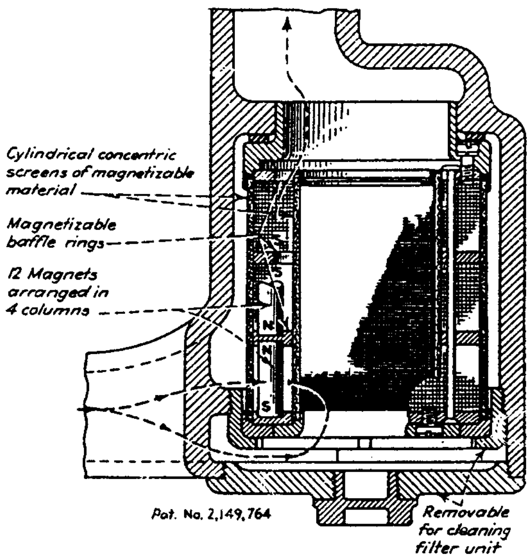
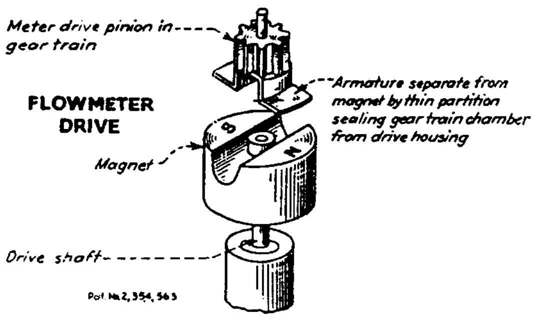


Fig. 12 A filter.



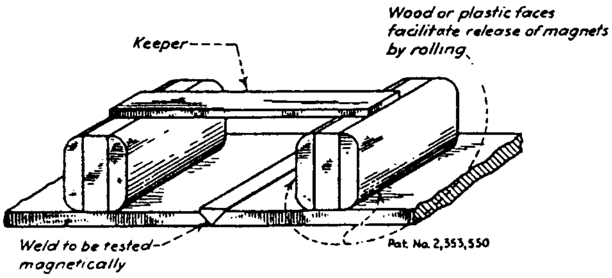


Fig. 13 **A weld tester.**

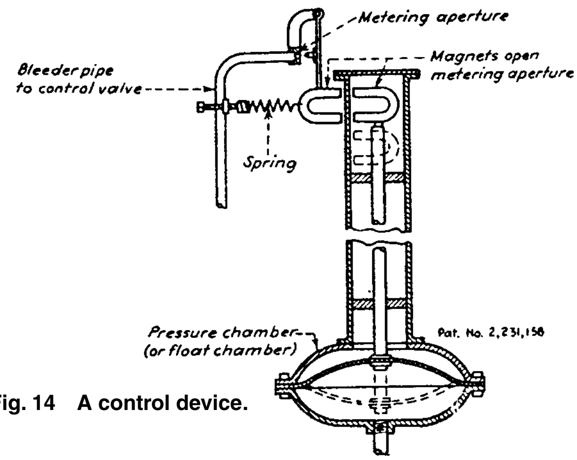


Fig. 14 **A control device.**

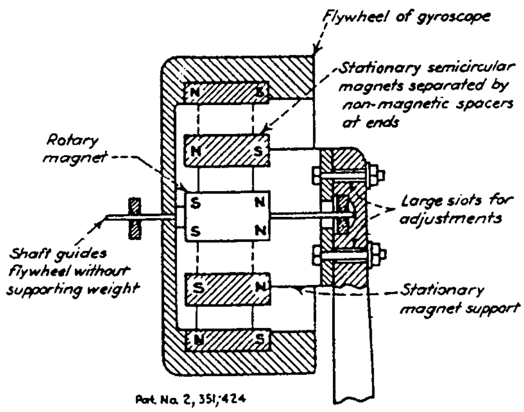


Fig. 15 **A horizontal-shaft suspension.**

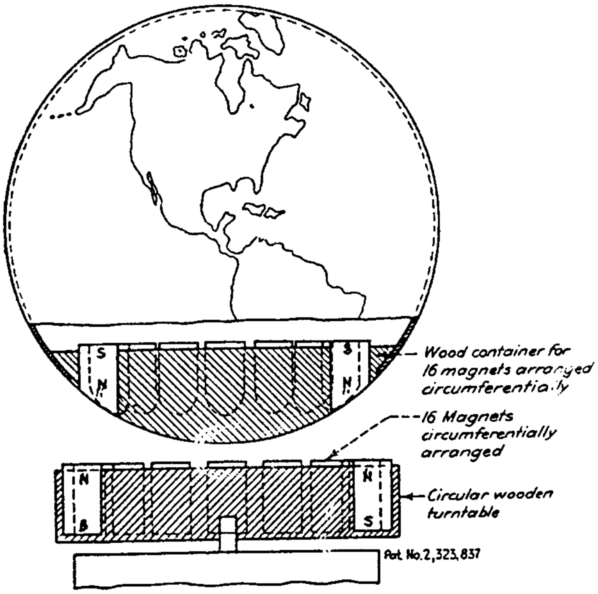
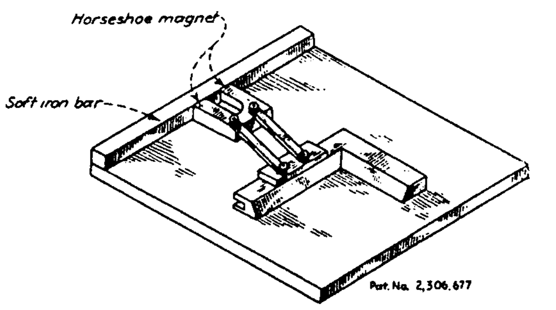


Fig. 16 **A floating advertising display.**



DRAFTING BOARD

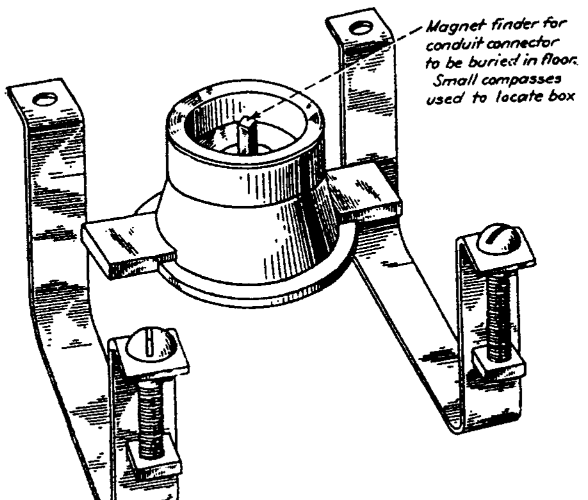


Fig. 17 **A finder.**

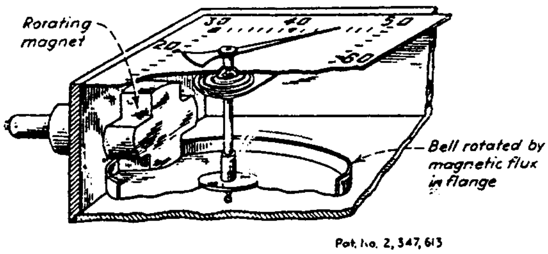


Fig. 18 **A tachometer.**

ELECTRICALLY DRIVEN HAMMER MECHANISMS

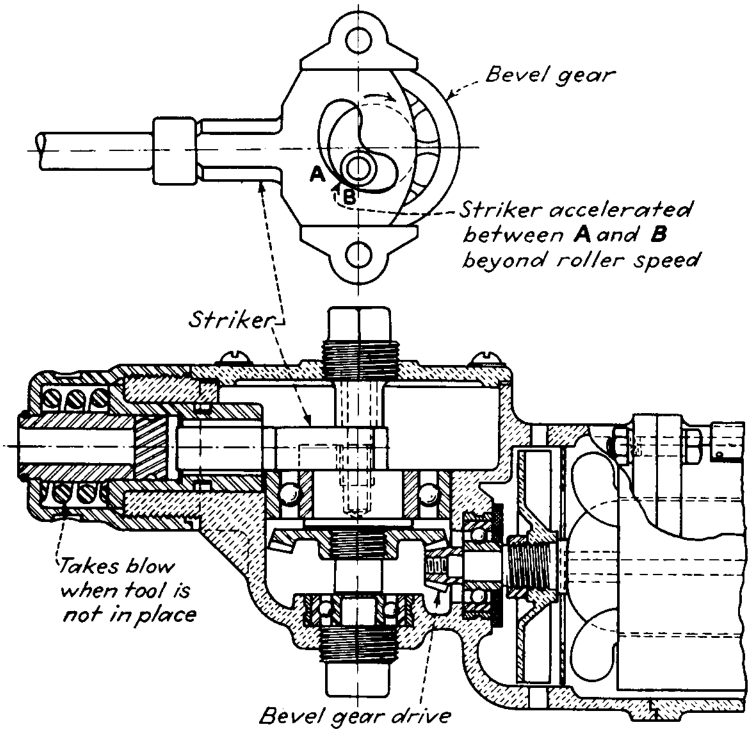


Fig. 1 A free-driving throw of the cam-slotted striker is produced by the eccentric stud roller during contact between points A and B of the slot. This accelerates the striker beyond the tangential speed of the roller for an instant before the striker is picked up for the return stroke.

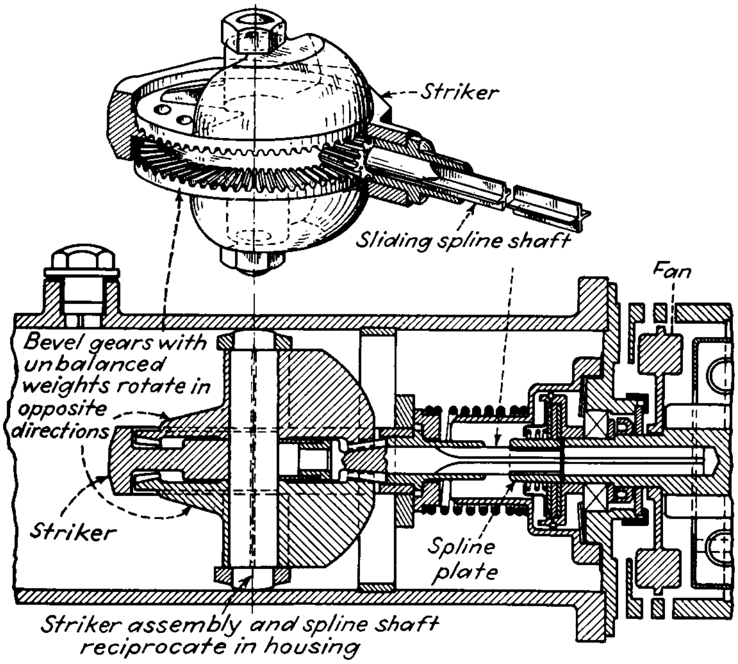


Fig. 2 The centrifugal force of two oppositely rotating weights throws the striker assembly of this hammer. The power connection is maintained by a sliding-splined shaft. The guide, not shown, prevents the rotation of the striker assembly.

The application of controlled impact forces can be as practical in specialized stationary machinery as in the portable electric hammers shown here. These mechanisms have been employed in vibrating concrete forms, nailing machines, and other special machinery. In portable hammers they are efficient in drilling, chiseling, digging, chipping, tamping,

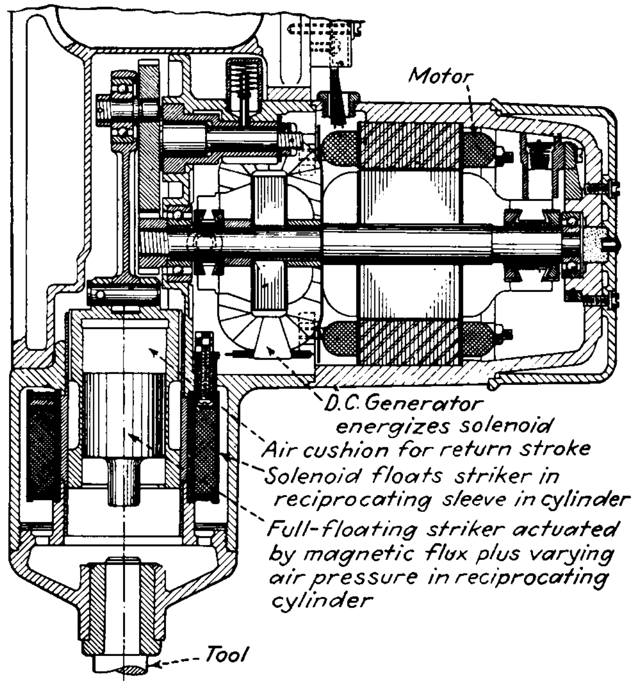


Fig. 3 The striker has no mechanical connection with the reciprocating drive in this hammer.

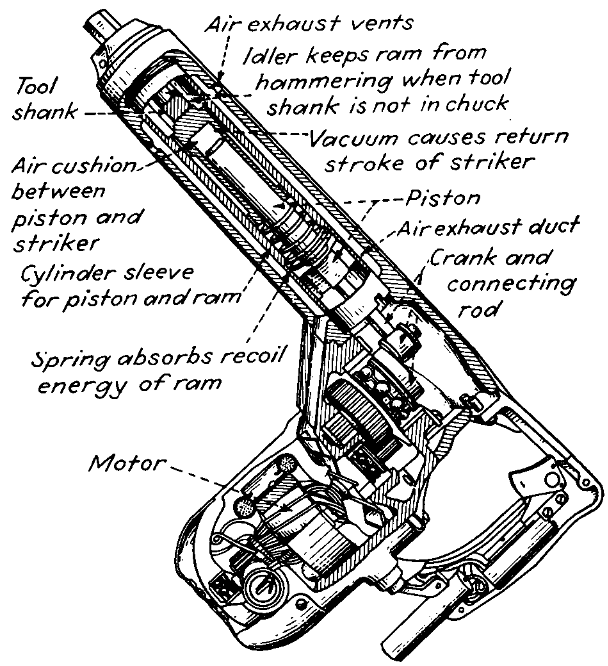


Fig. 4 This combination of mechanical, pneumatic, and spring action is included in this hammer.

riveting, and similar operations where quick, concentrated blows are required. The striker mechanisms illustrated are operated by springs, cams, magnetic force, air and vacuum chambers, and centrifugal force. The drawings show only the striking mechanisms.

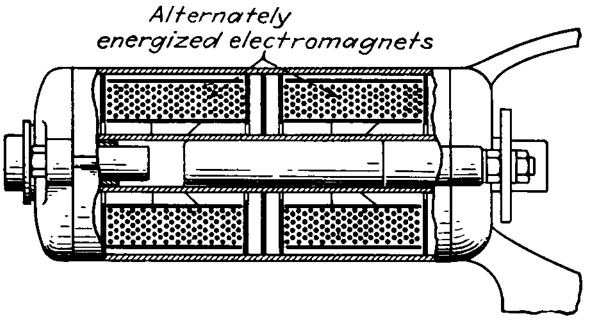


Fig. 5 Two electromagnets operate this hammer. The weight of the blows can be controlled by varying the electric current in the coils or timing the current reversals by an air-gap adjustment of the contacts.

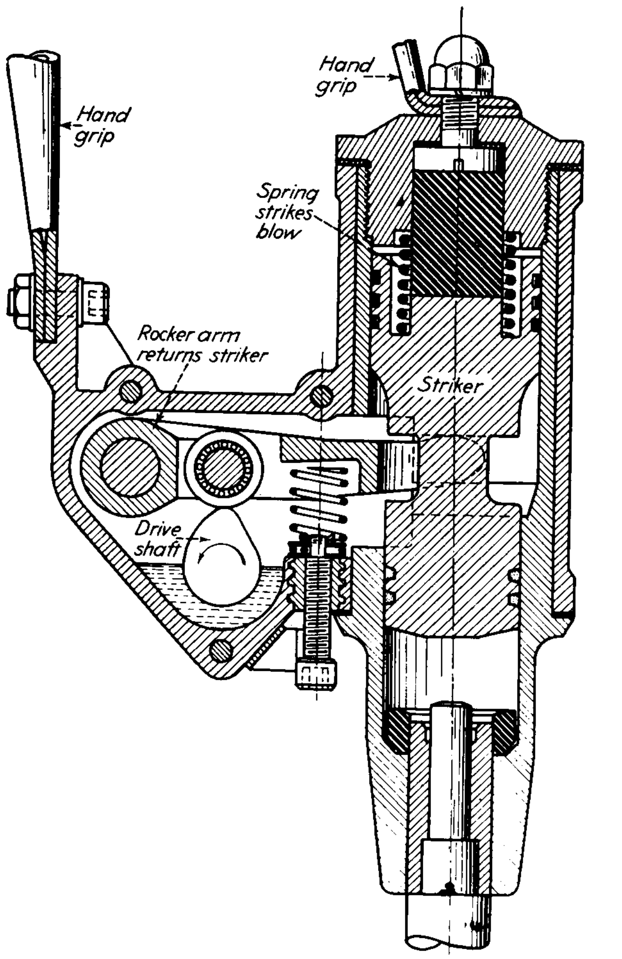


Fig. 6 This spring-operated hammer with a cam and rocker for the return stroke has a screw for adjusting the blow to be imparted.

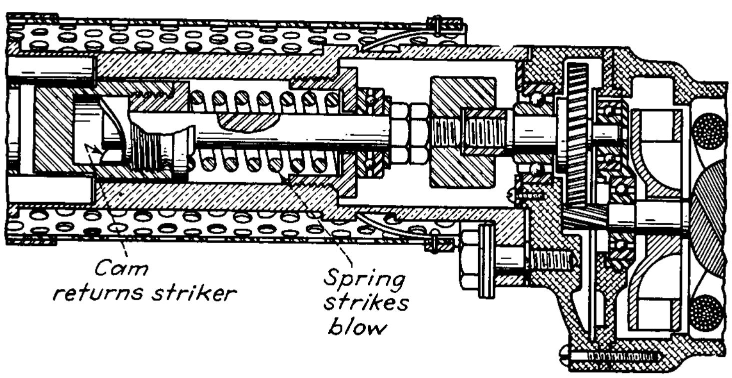


Fig. 7 This spring-operated hammer includes a shaft rotating in a female cam to return the striker.

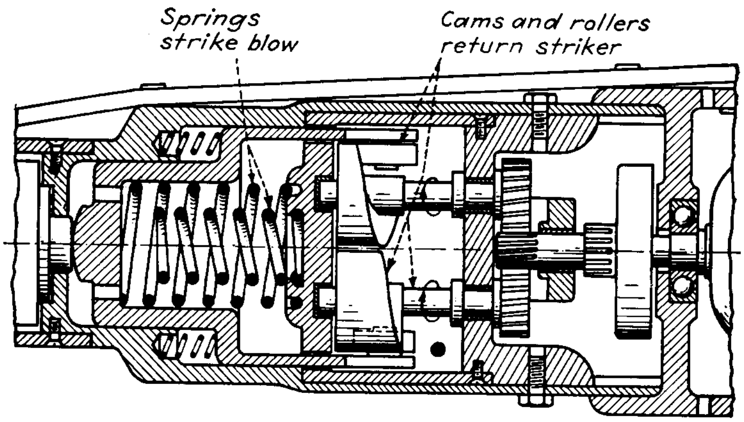


Fig. 8 This spring-operated hammer has two fixed rotating-barrel cams. They return the striker by means of two rollers on the opposite sides of the striker sleeve. Auxiliary springs prevent the striker from hitting the retaining cylinder. A means of rotating the tool, not shown, is also included in this hammer.

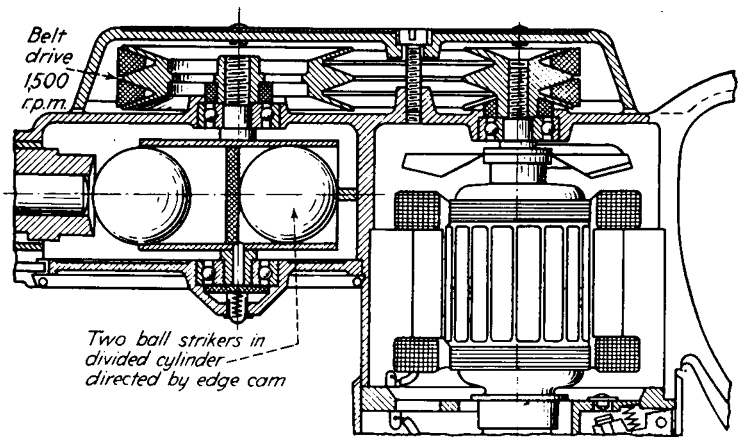


Fig. 9 Two steel balls rotated in a divided cylinder and steered by an edge cam develop centrifugal force to strike blows against the tool holder. The collar is held clear of the hammer by a compression spring when no tool is in the holder. A second spring cushions the blows when the motor is running, but the tool is not held against the work.

Sensitivity or change in deflection for a given temperature change depends upon the combination of meals selected and the dimensions of the bimetal element. Sensitivity increases with the square of the length and inversely with the thickness. The force developed for a given temperature change also depends on the type of bimetal. However, the allowable working load for the thermostatic strip increases with the width and the square of the thickness. Thus, the design of bimetal elements depends upon the relative importance of sensitivity and working load.

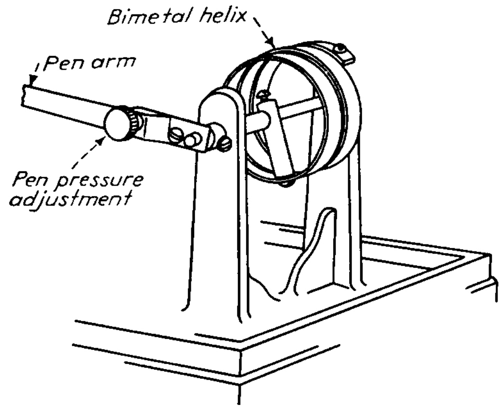


Fig. 1 This recording thermometer has a pen that is moved vertically across a revolving chart by a brass-invar bimetal element. To obtain sensitivity, the long movement of the pen requires a long strip of bimetal, which is coiled into a helix to save space. For accuracy, a relatively large cross section gives stiffness, although the large thickness requires increased length to obtain the desired sensitivity.

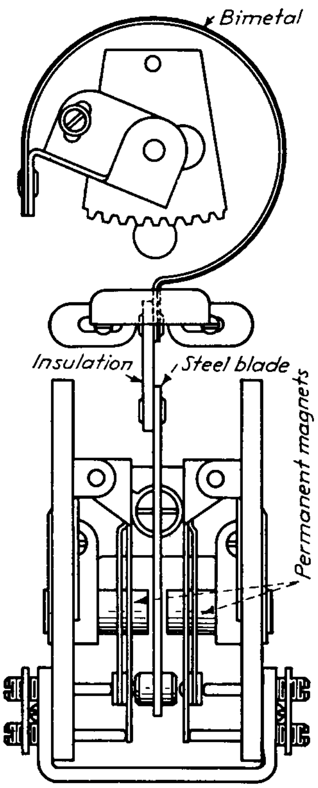


Fig. 2 Room temperatures in summer and winter are controlled over a wide range by a single, large-diameter coil of brass-invar in this thermometer. To prevent chattering, a small permanent magnet is mounted on each side of the steel contact blade. The magnetic attraction on the blade, which increases inversely with the square of the distance from the magnet, gives a snap action to the contacts.

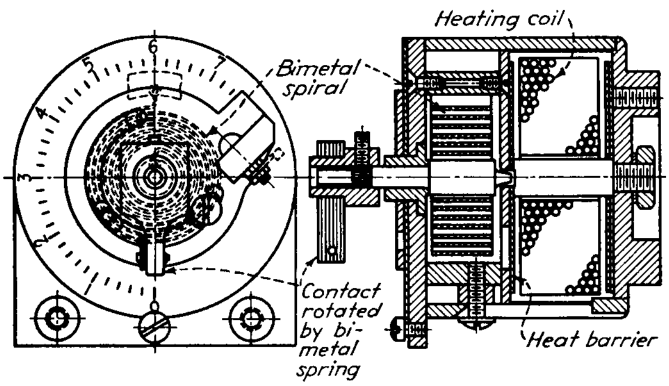


Fig. 3 This overload relay for large motors passes some of the motor current through a heating coil within the relay. Heat from the coil raises the temperature of a bimetal spiral which rotates a shaft carrying an electrical contact. To withstand the operating temperatures, it includes a heat-resistant bimetal spiral. It is coiled into the spiral form for compactness. Because of the large deflection needed, the spiral is long and thin, whereas the width is made large to provide the required contact pressure.

Heat barriers between the bimetal spiral and the heating coil make the temperature rise of the bimetal spiral closely follow the increase in temperature within the motor. Thus, momentary overloads do not cause sufficient heating to close the contacts. However, a continued overload will, in time, cause the bimetal spiral to rotate the contact arm around to the adjustable stationary contact, causing a relay to shut down the motor.

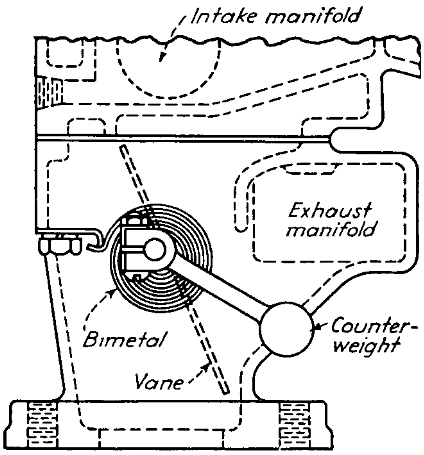


Fig. 4 Carburetor control. When the engine is cold, a vane in the exhaust passage to the "hot spot" is held open by a bimetal spring against the force of a small counterweight. When the thermostatic spiral is heated by the outside air or by the warm air stream from the radiator, the spring coils up and allows the weight to close the vane. Because high accuracy is not needed, a thin, flexible cross section with a long length provides the desired sensitivity.

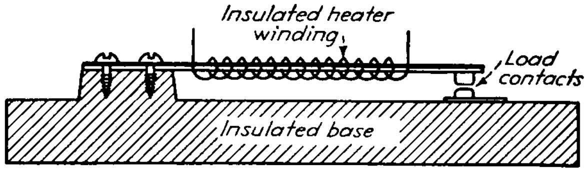


Fig. 5 Thermostatic relay. A constant current through an electrical heating coil around a straight bimetal strip gives a time-delay action. Because the temperature range is relatively large, high sensitivity is not necessary. Thus, a short, straight strip of bimetal is suitable. Because of its relatively heavy thickness, the strip is sufficiently stiff to close the contact firmly without chattering.

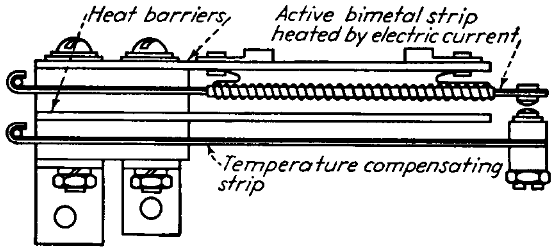


Fig. 6 The bimetal element in this time-delay relay protects mercury-vapor rectifiers. This relay closes the voltage circuit to the mercury tube only after the filament has had time to reach its normal operating temperature. To eliminate the effect of changes in room temperature on the length of the contact gap (and therefore the time interval) the stationary contact is carried by a second bimetal strip, similar to the heated element. Barriers of laminated plastic on both sides of the active bimetal strip shield the compensating strip and prevent air currents from affecting the heating rate. The relatively high temperature range allows the use of a straight, thick strip, but the addition of the compensating strip makes accurate timing possible with a short travel.

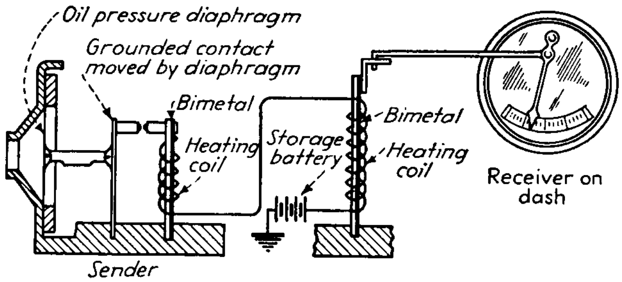


Fig. 7 Oil pressure, engine temperature, and gasoline level are indicated electrically on automobile dashboard instruments whose bimetal element is both the sender and receiver. A grounded contact at the sender completes an electric circuit through heaters around two similar bimetal strips. Because the same current flows around the two bimetal elements, their deflections are the same. But the sender element, when heated, will bend away from the grounded contact until the circuit is broken Upon cooling, the bimetal element again makes contact and the cycle continues. This allows the bimetal element to follow the movement of the grounded contact. For the oil-pressure gage, the grounded contact is attached to a diaphragm; for the temperature indicator, the contact is carried by another thermostatic bimetal strip; in the gasoline-level device, the contact is shifted by a cam on a shaft rotated by a float. Deflections on the receiving bimetal element are amplified through a linkage that operates a pointer over the scale of the receiving instrument. Because only small deflections are needed, the bimetal element is in the form of a short, stiff strip.

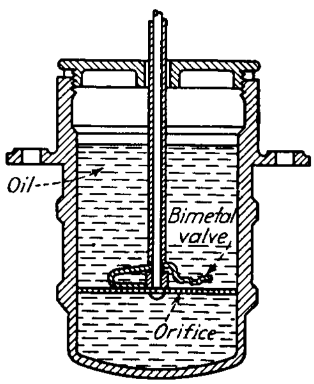


Fig. 8 Oil dashpots in heavy-capacity scales have a thermostatic control to compensate for changes in oil viscosity with temperature. A rectangular orifice in the plunger is covered by a swaged projection on the bimetal element. With a decrease in oil temperature, the oil viscosity increases, tending to increase the damping effect. But the bimetal element deflects upward, enlarging the orifice enough to keep the damping force constant. A wide bimetal strip provides sufficient stiffness so that the orifice will not be altered by the force of the flowing oil.

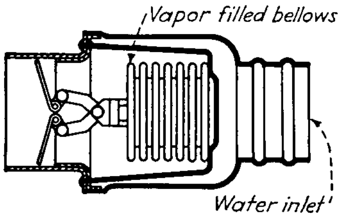


Fig. 9 Automobile cooling-water temperature is controlled by a self-contained bellows in the thermostat. As in the radiator air valve, the bellows itself is subjected to the temperature to be controlled. As the temperature of the water increases to about 140°F. the valve starts to open; at approximately 180°F., free flow is permitted. At intermediate temperatures, the valve opening is in proportion to the temperature.

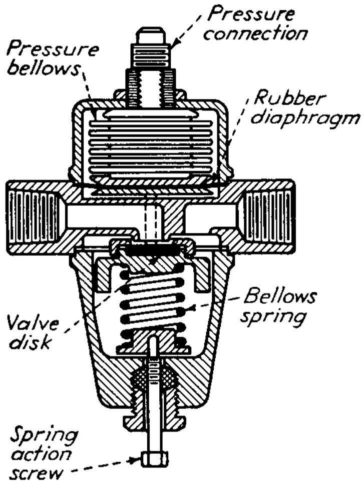


Fig. 10 A throttling circulating-water control valve for refrigeration plants has its valve opening vary with the pressure on the bellows. This valve controls the rate of flow of the cooling water through the condenser. A greater amount of water is required when the temperature, and therefore the pressure, increases. The pressure in the condenser is transmitted through a pipe to the valve bellows, thereby adjusting the flow of cooling water. The bronze bellows is protected from contact with the water by a rubber diaphragm.

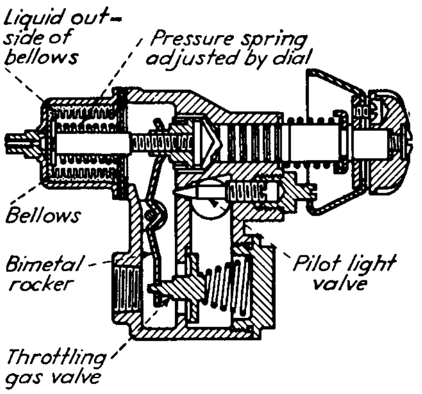


Fig. 11 An automatic gas-range control has a sealed thermostatic element consisting of a bulb, capillary tube, and bellows. As food is often placed near the bulb, a nontoxic liquid, chlorinated diphenyl, is in the liquid expansion system. The liquid is also non-flammable and has no corrosive effect on the phosphor-bronze bellows. By placing the liquid outside instead of inside the bellows, the working stresses are maximum at normal temperatures when the bellows bottoms on the cup. At elevated working temperatures, the expansion of the liquid compresses the bellows against the action of the extended spring. This, in turn, is adjusted by the knob. Changes in calibration caused by variations in ambient temperature are compensated by making the rocker arm of a bimetal suitable for high-temperature service.

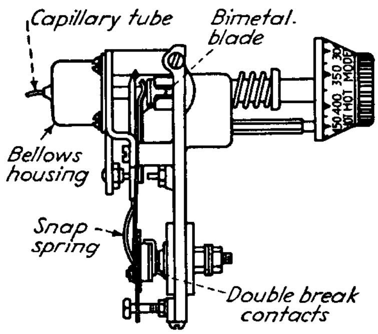


Fig. 12 For electric ranges, this thermostat has the same bellows unit as the gas-type control. But, instead of a throttling action, the thermostat opens and closes the electrical contacts with a snap action. To obtain sufficient force for the snap action, the control requires a temperature difference between *on* and *off* positions. For a control range from room temperature to 550°F., the differential in this instrument is ±10°F. With a smaller control range, the differential is proportionately less. The snap-action switch is made of beryllium copper, giving it high strength, better snap action, and longer life than is obtainable with phosphor bronze. Because of its corrosion resistance, the beryllium-copper blade requires no protective finish.

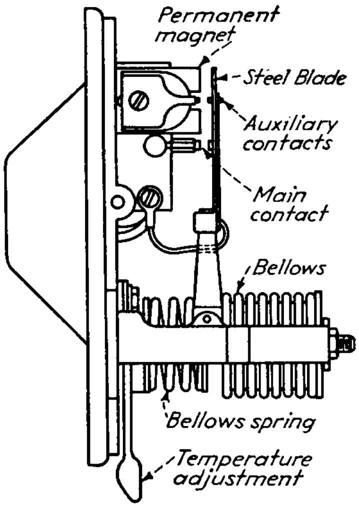


Fig. 13 For heavy-duty room-temperature controls, this thermostat has a bellows mechanism that develops a high force with small changes in temperature. The bellows is partly filled with liquid butane. At room temperatures this gas exhibits a large change in vapor pressure for small temperature differentials. Snap action of the electrical contact is obtained from a small permanent magnet that pulls the steel contact blade into firm contact when the bellows cools. Because of the firm contact, the device is rated at 20 A for noninductive loads. To avoid chattering or bounce under the impact delivered by the rapid magnetic closing action, small auxiliary contacts are carried on light spring blades. With the large force developed by the bellows, a temperature differential of only 2°F. is obtained.

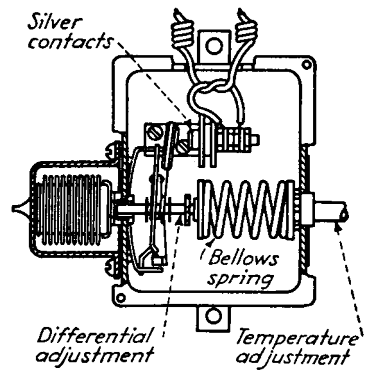
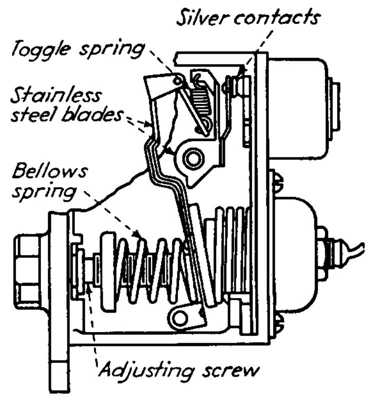


Fig. 14 Snap action in this refrigerator control is obtained from a bowed flat spring. The silver contacts carried on an extended end of the spring open or close rapidly when movement of the bellows actuates the spring. With this snap action, the contacts can control an alternating-current motor as large as 1½ hp without auxiliary relays. Temperature differential is adjusted by changing the spacing between two collars on the bellows shaft passing through the contact spring. For the temperatures needed to freeze ice, the bellows system is partly filled with butane.

Fig. 15 In this refrigerator control, the necessary snap action is obtained from a toggle spring supported from a long arm moved by the bellows. With this form of toggle action, the contact pressure is at a maximum at the instant the contacts start to open. Thermostatic action is obtained from a vapor-filled system. Sulfur dioxide is the fill for typical refrigerating service or methyl chloride where lower temperatures are required. To reduce friction, the bellows makes point-contact with the bellows cup. Operating temperature is adjusted by changing the initial compression in the bellows spring. For resistance to corrosion, levers and blades are made from stainless steel with bronze pin bearings.



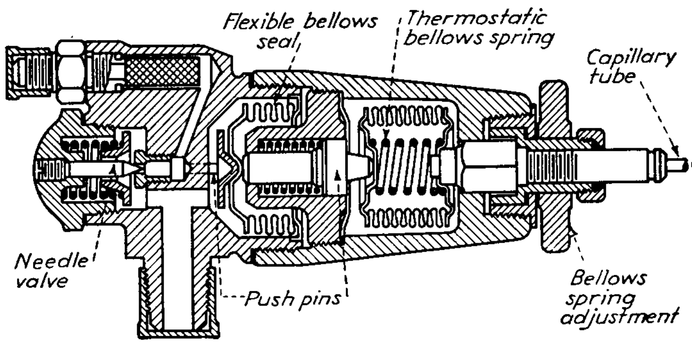


Fig. 16 Two bellows units in this thermostatic expansion valve control large refrigeration systems. A removable power bellows unit is operated by vapor pressure in a bulb attached to the evaporator output line. The second bellows serves as a flexible, gastight seal for the gas valve. A stainless-steel spring holds the valve closed until opened by pressure transmitted from the thermostatic bellows through a molded push pin.

TEMPERATURE-REGULATING MECHANISMS

Temperature regulators are either on-off or throttling. The characteristics of the process determine which should be used. Within each group, selection of a regulator is governed by the accuracy required, space limitations, simplicity, and cost.

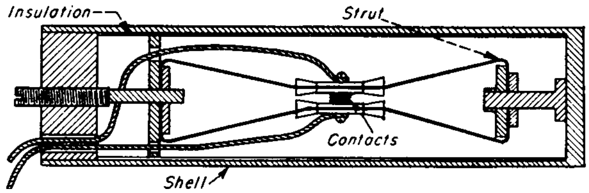


Fig. 1 A bimetallic sensor is simple, compact, and precise. Contacts mounted on low-expansion struts determine slow make-and-break action. A shell contracts or expands with temperature changes, opening or closing the electrical circuit that controls a heating or cooling unit. It is adjustable and resistant to shock and vibration. Its range is 100 to 1500°F, and it responds to a temperature changes of less than 0.5°F.

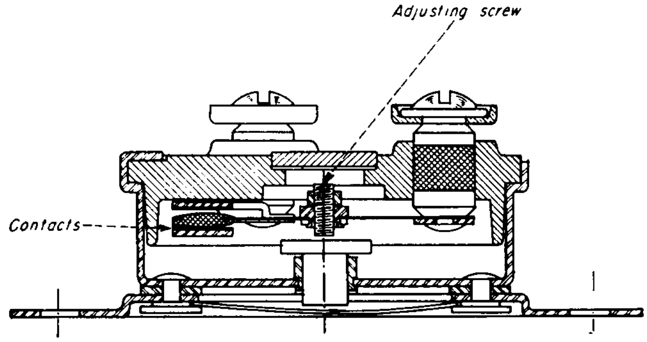


Fig. 2 This enclosed, disk-type, snap-action control has a fixed operating temperature. It is suitable for unit and space heaters, small hot water heaters, clothes dryers, and other applications requiring non-adjustable temperature control. It is useful where dirt, dust, oil, or corrosive atmospheres are present. It is available with various temperature differentials and with a manual reset. Depending on the model, its temperature setting range is from –10° to 550°F and its minimum differential can be 10, 20, 30, 40 or 50°F.

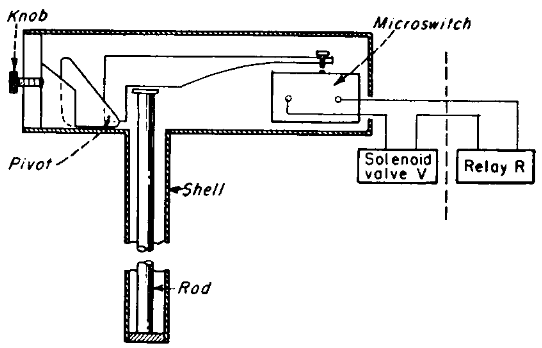


Fig. 3 This bimetallic unit has a rod with a low coefficient of expansion and a shell with a high coefficient of expansion. A microswitch gives snap action to the electrical control circuit. The current can be large enough to operate a solenoid valve or relay directly. The set point is adjusted by a knob which moves the pivot point of the lever. Its range is –20° to 175°F, and its accuracy is 0.25 to 0.50°.

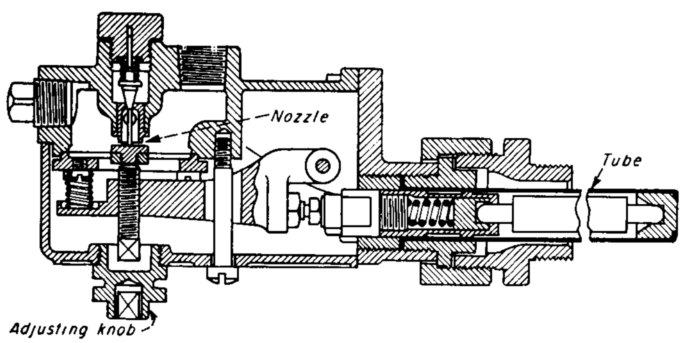


Fig. 4 This is a bimetallic-actuated, air-piloted control. The expansion of the rod causes an air signal (3 to 15 psi) to be transmitted to a heating or cooling pneumatic valve. The position of the pneumatic valve depends on the amount of air bled through the pilot valve of the control. This produces a throttling type of temperature control as contrasted to the on-off characteristic that is obtained with the three units described previously. Its range is 32 to 600°F, and its accuracy is ±1 to ±3°F, depending on the range.

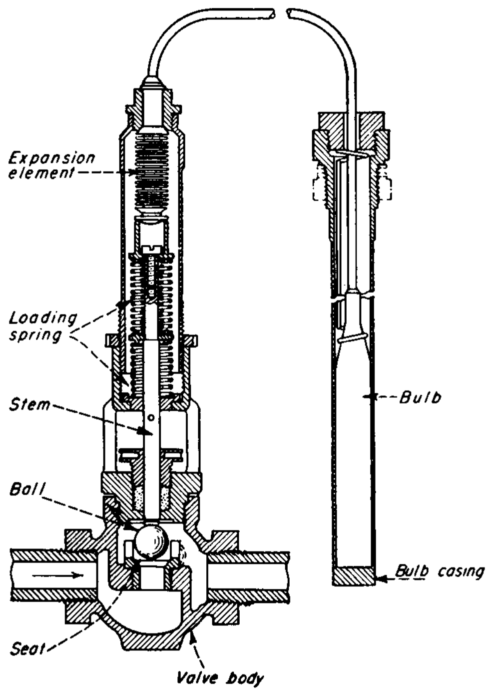


Fig. 5 This self-contained regulator is actuated by the expansion or contraction of liquid or gas in a temperature-sensitive bulb that is immersed in the medium being controlled. The signal is transmitted from the bulb to a sealed expansion element which opens or closes the ball valve. Its range is 20 to 270°F, and its accuracy is ±1°F. The maximum pressure rating is 100 psi for dead-end service and 200 psi for continuous flow.

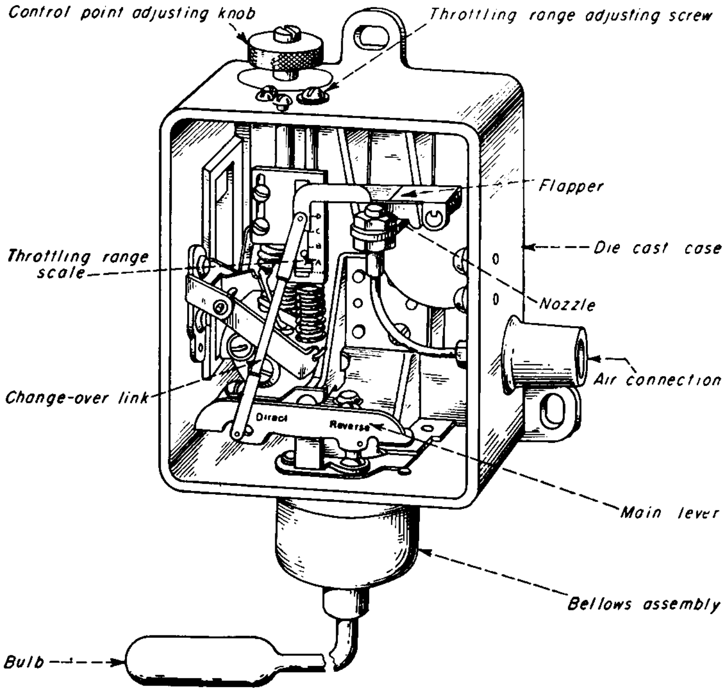


Fig. 6 This remote bulb, nonindicating regulator has a bellows assembly that operates a flapper. This allows air pressure in the control system to build up or bleed, depending on the position of the changeover link. The unit can be direct- or reverse-acting. A control knob adjusts the setting, and the throttling range adjustment determines the percentages of the control range in which full output pressure (3–15 psi) is obtained. Its range is 0 to 700°F, and its accuracy is about ±0.5% of full scale, depending on the way it is installed.

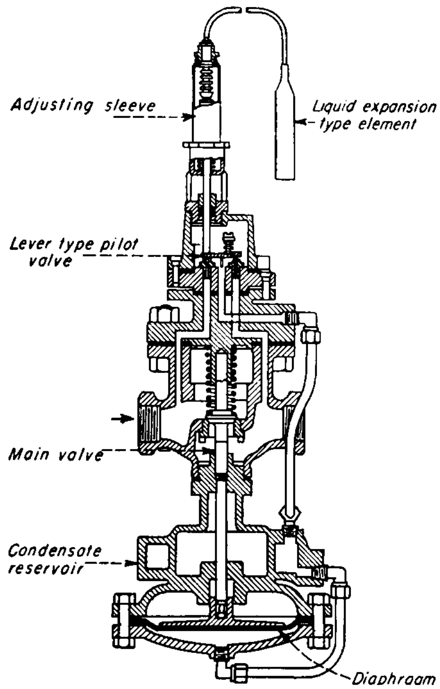


Fig. 7 This lever-type pilot valve is actuated by a temperature-sensitive bulb. The motion of the lever causes the water or steam being controlled to exert pressure on a diaphragm which opens or closes the main valve. Its temperature range is 20 to 270°F, and its accuracy is ±1 to 4°F. It is rated for 5 to 125 psi of steam and 5 to 175 psi of water.

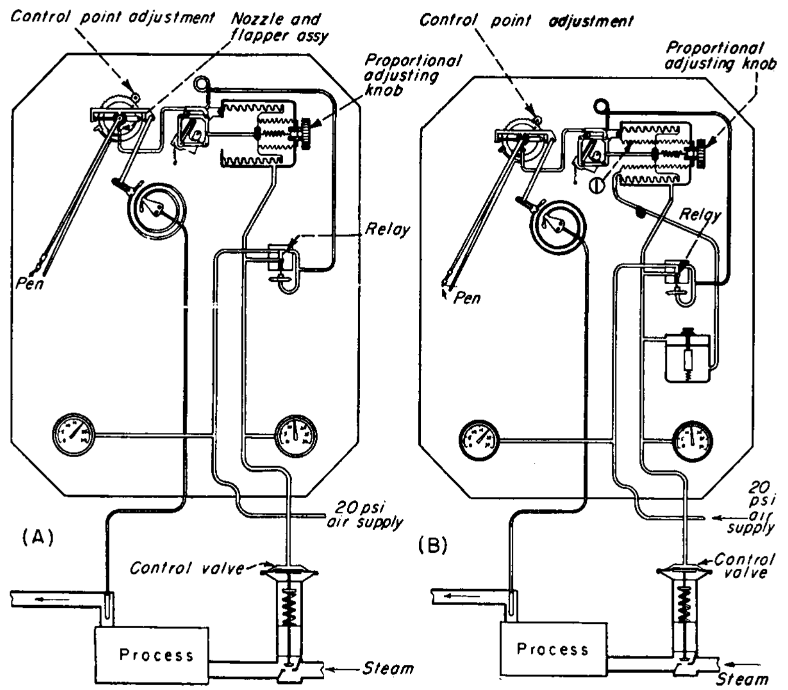


Fig. 8 These two recording and controlling instruments have adjustable proportional ranges. In both, air supply is divided by a relay valve. A small part goes through a nozzle and flapper assembly. The main part goes to the control valve. Unit B has an extra bellows for automatic resetting. It was designed for systems with continuously changing control points, and it can be used where both heating and cooling are required in one process. Both A and B are easily changed from direct to reverse acting. Its accuracy is 1% of its temperature range of –40 to 800°F.

PHOTOELECTRIC CONTROLS

Typical applications are presented for reducing production costs and increasing operator safeguards by precisely and automatically controlling the feed, transfer, or inspection of products from one process stage to another.

Fig. 1 Automatic weighing and filling. The task is to fill each box with an exact quantity of products, such as screws. An electric feeder vibrates parts through a chute and into a box on a small balance. The photoelectric control is mounted at the rear of the scale. The light beam is restricted to very small dimensions by an optical slit. The control is positioned so that the light is interrupted by a balanced cantilever arm attached to the scale when the proper box weight is reached. The photoelectric control then stops the flow of parts by deenergizing the feeder. Simultaneously, an indexing mechanism is activated to remove the filled box and replace it with an empty one. The completion of indexing reenergizes the feeder, which starts the flow of screws.

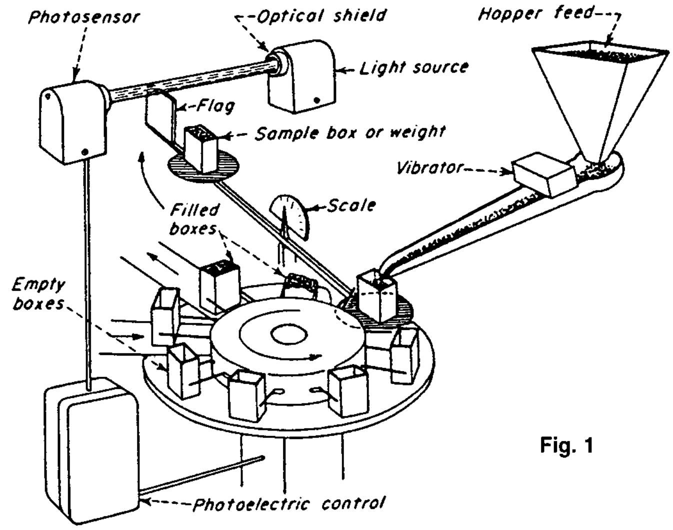


Fig. 1

Fig. 2 Operator safeguard. Most pressures are operated by a foot pedal that leaves the operator's hands free for loading and unloading. This creates a safety hazard. The use of mechanical gate systems reduces the speed of production. With photoelectric controls, a curtain of light is set up by a multiple series of photoelectric scanners and light sources. When a light beam is broken at any point by the operator's hand, the control energizes a locking mechanism that prevents the punch-press drive from being energized. A circuit or power failure causes the control to function as if the light beam were broken. In addition, the light beam frequently becomes the actuating control because the clutch is released as soon as the operator removes his hand from the die on the press table.

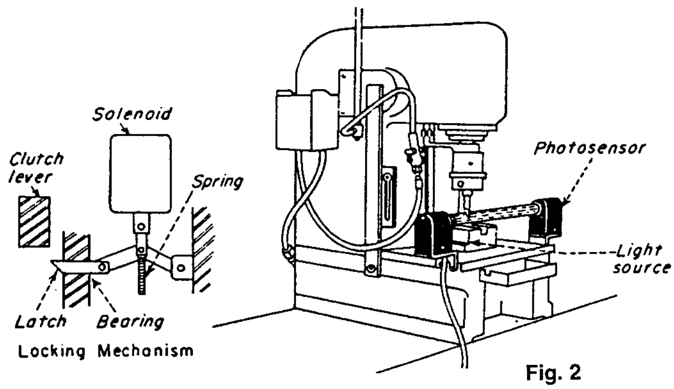


Fig. 2

Fig. 3 This apparatus sorts cartons of three different kinds of objects. Because the cartons containing objects differ widely in size, it is not feasible to sort by carton size and shape. A small strip of reflecting tape is put on the cartons by a packer during assembly. On one type of object, the strip is located along one edge of the bottom, and it extends almost to the middle. For the second type, the strip is located along the same edge, but from the middle to the opposite side. No tape is placed on the third type. Cartons are placed on the conveyor so that the tape is at right angles to the direction of travel. Photoelectric controls shown in A "see" the reflecting tape and operate a pusher-bar mechanism shown in B. This pushes the carton onto the proper distribution conveyor. Cartons without tape pass.

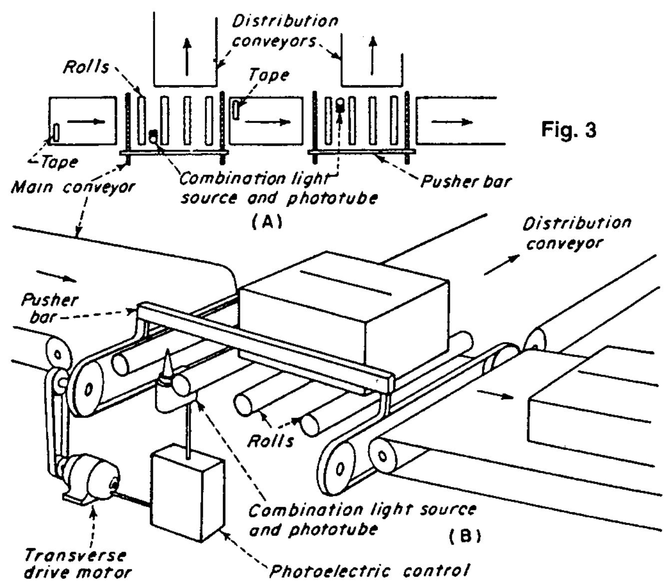


Fig. 3

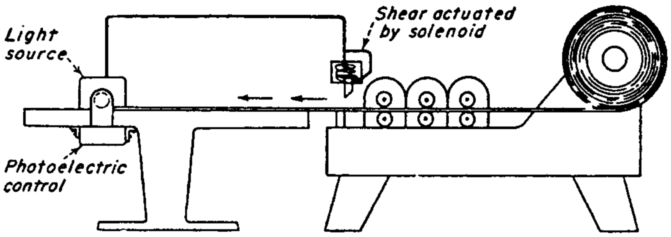


Fig. 4

Fig. 4 This cut-off machine has a photoelectric control for strip materials that lack sufficient mass to operate a mechanical limit switch satisfactorily. The forward end of the strip breaks the light beam, thus actuating the cut-off operation. The light source and the control are mounted on an adjustable stand at the end of the machine to vary the length of the finished stock.

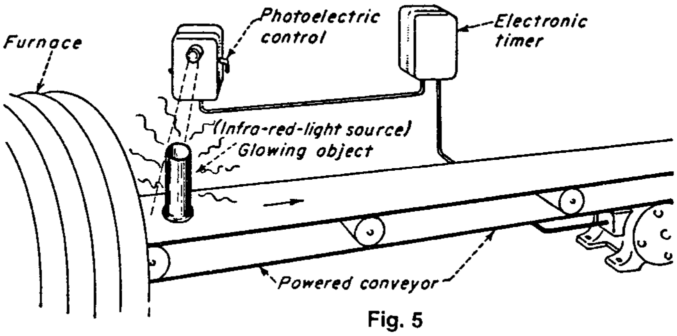


Fig. 5

Fig. 5 This heat-treating conveyor has an electronic timer paired with a photoelectric control to carry parts emerging from a furnace at 2300°F. The conveyor must operate only when a part is placed on it and only for the distance required to reach the next process stage. Parts are ejected onto the conveyor at varying rates. High temperatures caused failures when the mechanical switches were used. Glowing white-hot parts radiate infrared rays that actuate the photoelectric control as soon as a part comes in view. The control operates the conveyor that carries the parts away from the furnace and simultaneously starts the timer. The conveyor is kept running by a timer for the predetermined length of time required to position the part for the next operation.

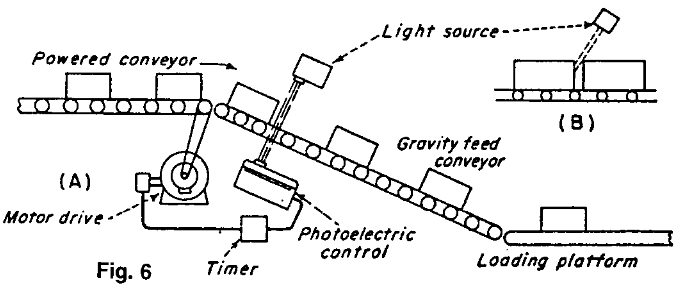


Fig. 6

Fig. 6 Jam detector. Cartons jamming on the conveyor cause losses in production and damage to cartons, products, and conveyors. Detection is accomplished with a photoelectric control that has a timer, as shown in (A). Each time a carton passes the light source, the control beam is broken. That starts the timing interval in the timer. The timing circuit is reset without relay action each time the beam is restored before the preset timing interval has elapsed. If a jam occurs, causing cartons to butt against each other, the light beam cannot reach the control. The timing circuit will then time-out, opening the load circuit. This stops the conveyor motor. By locating the light source at an angle with respect to the conveyor, as shown in (B), the power conveyor can be delayed if cartons are too close to each other but not butting each other.

Fig. 7

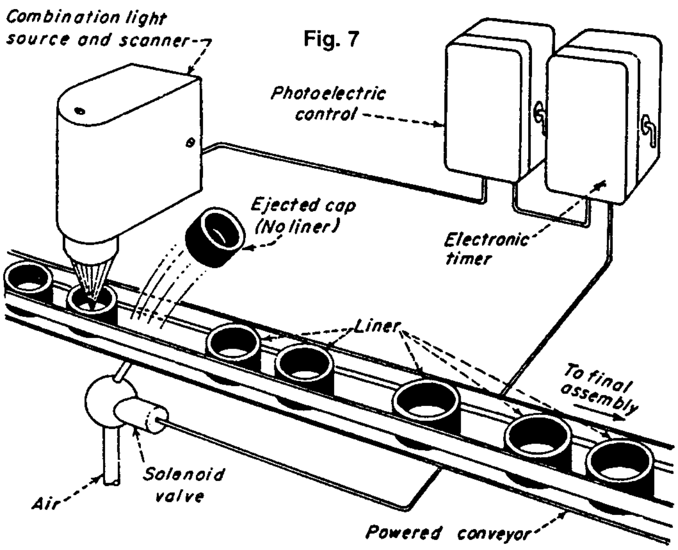
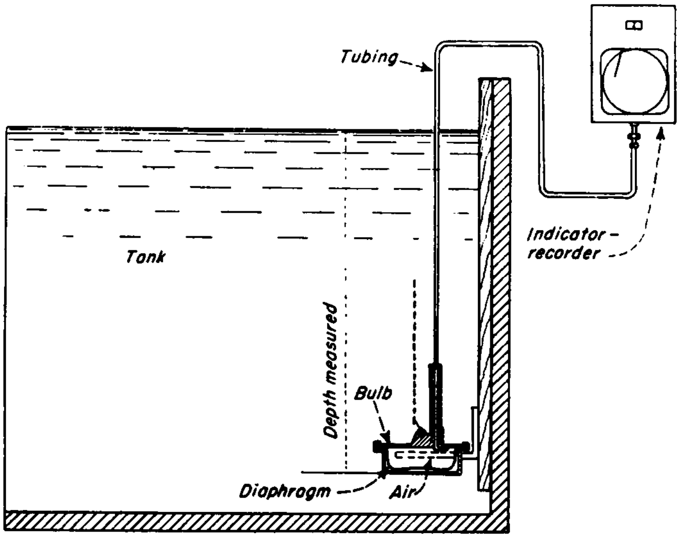


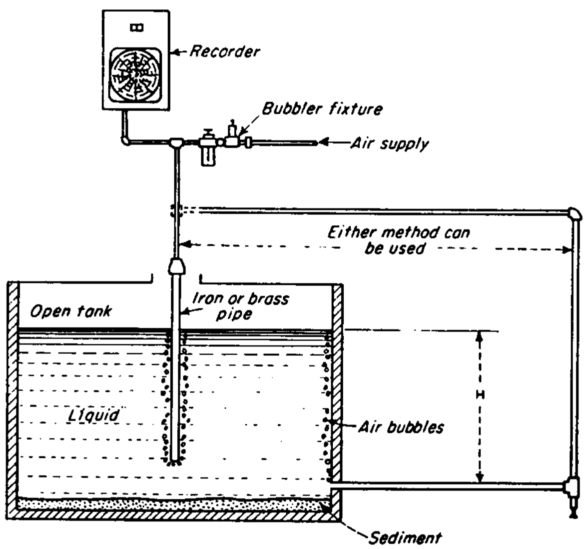
Fig. 7 Automatic inspection. As steel caps are conveyed to final assembly, they pass an intermediate stage where an assembler inserts an insulation liner into a cap. The inspection point for the missing liners has a reflection-type photoelectric scanner which incorporates both a light source and photosensor with a common lens system to recognize the difference in reflection between the dark liner and the light steel cap instantly. When it detects a cap without a liner, a relay operates an airjet ejector that is controlled by a solenoid valve. The start and duration of the air blast is accurately controlled by a timer so that no other caps are displaced.

LIQUID LEVEL INDICATORS AND CONTROLLERS

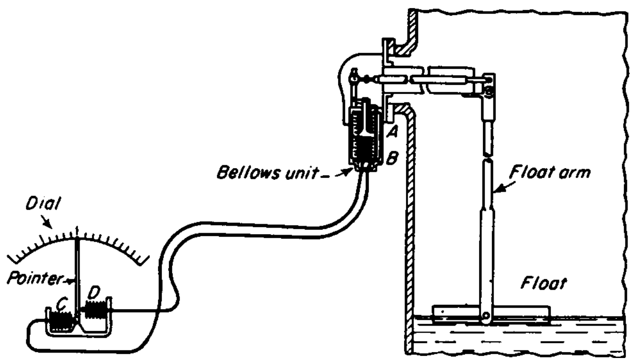
Thirteen different systems of operation are shown. Each one represents at least one commercial instrument. Some of them are available in several modified forms.



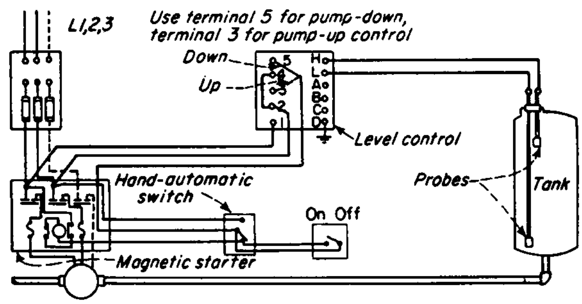
A diaphragm actuated indicator will work with any kind of liquid, whether it is flowing, turbulent, or carrying solid matter. A recorder can be mounted above or below the level of the tank or reservoir.



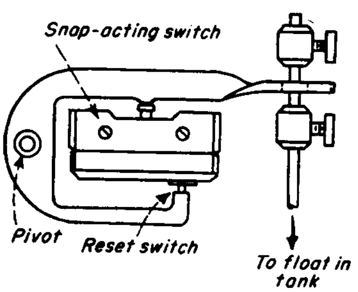
A bubbler-type recorder measures height *H.* It can be used with all kinds of liquids, including those carrying solids. A small amount of air is bled into a submerged pipe. A gage measures the air pressure that displaces the fluid.



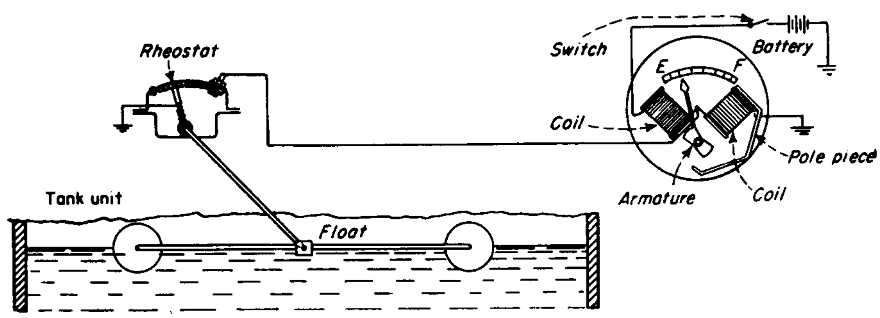
A bellows-actuated indicator. Two bellows and a connecting tubing are filled with incompressible fluid. A change in liquid level displaces the transmitting bellows and pointer.



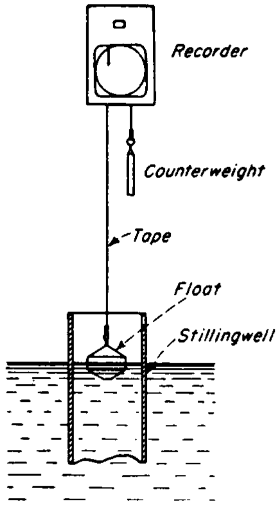
An electrical level controller. The positions of the probes determine the duration of pump operation. When a liquid touches the upper probe, a relay operates and the pump stops. Auxiliary contacts on the lower probe provide a relay-holding current until the liquid level drops below it.



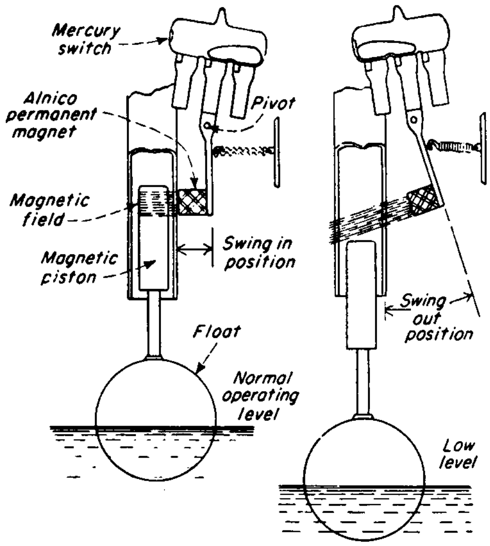
A float-switch controller. When liquid reaches a predetermined level, a float actuates a switch through a horseshoe-shaped arm. A switch can operate the valve or pump.



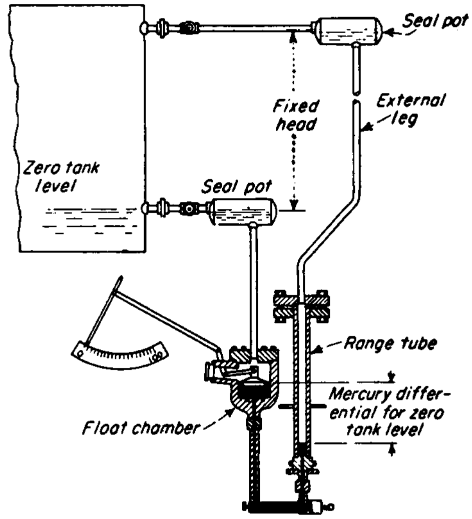
An automotive liquid-level indicator. The indicator and tank unit are connected by a single wire. As the liquid level in the tank increases, brush contact on the tank rheostat moves to the right, introducing an increasing amount of resistance into the circuit that grounds the F coil. The displacement of a needle from its empty mark is proportional to the amount of resistance introduced into this circuit.



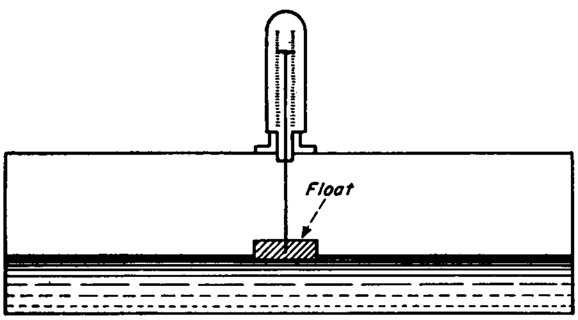
A float recorder. The pointer can be attached to a calibrated float tape to give an approximate instantaneous indication of fluid level.



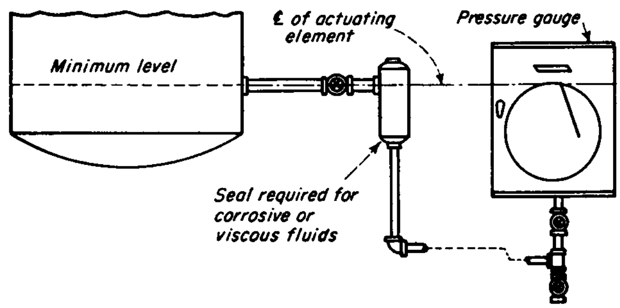
A magnetic liquid-level controller. When the liquid level is normal, the common-to-right leg circuit of the mercury switch is closed. When the liquid drops to a predetermined level, the magnetic piston is drawn below the magnetic field.



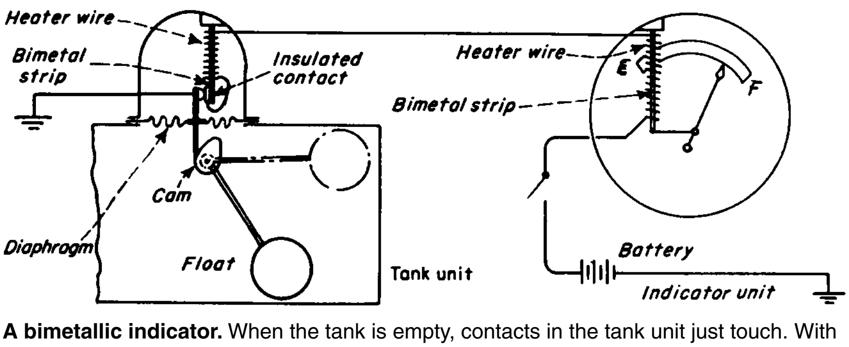
A differential pressure system. This system is applicable to liquids under pressure. The measuring element is a mercury manometer. A mechanical or electric meter body can be used. The seal pots protest the meter body.



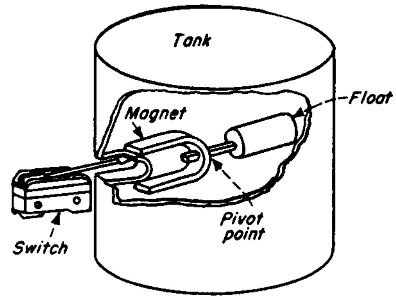
A direct-reading float gage. This inexpensive, direct-reading gage has a dial calibrated to the tank volume. A comparable gage, in terms of simplicity, has a needle connected through a right-angle arm to the float. As the liquid level drops, the float rotates the arm and the needle.



A pressure gage indicator for open vessels. The pressure of the liquid head is imposed directly upon the actuating element of the pressure gage. A center-line of the actuating element must coincide with the minimum level line if the gage is to read zero when the liquid reaches the minimum level.



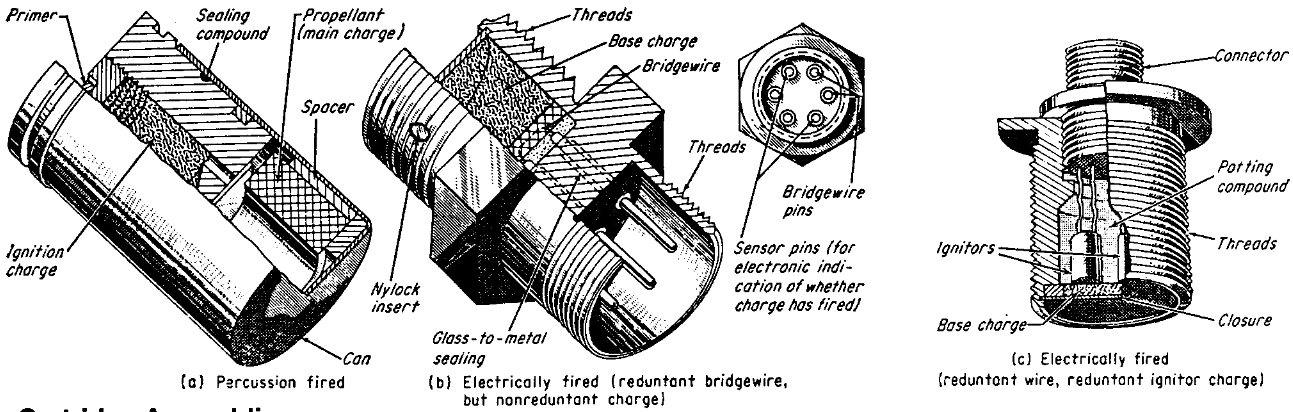
A bimetallic indicator. When the tank is empty, contacts in the tank unit just touch. With the switch closed, heaters cause both bimetallic strips to bend. This opens the contacts in the tank, and the bimetals cool, closing the circuit again. The cycle repeats about once per second. As the liquid level increases, the float forces the cam to bend the tank bimetal. This action is similar to that of the float gage, but the current and the needle displacement are increased.



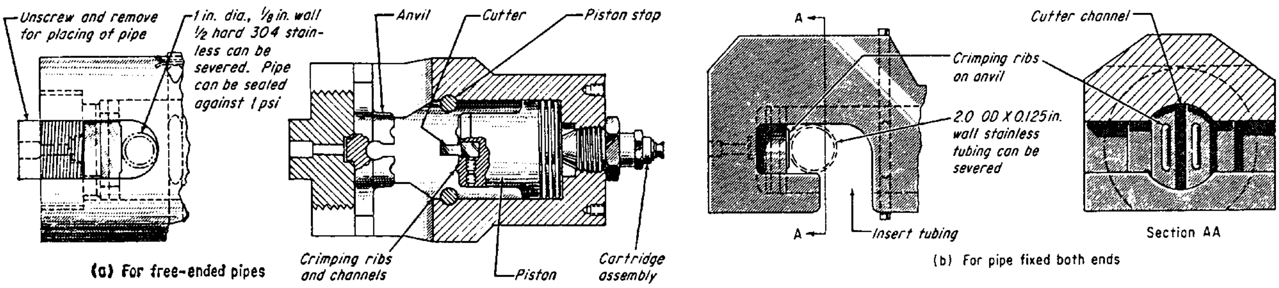
A switch-actuated level controller. This pump is actuated by the switch. The float pivots the magnet so that the upper pole attracts the switch contact. The tank wall serves as the other contact.

INSTANT MUSCLE WITH PYROTECHNIC POWER

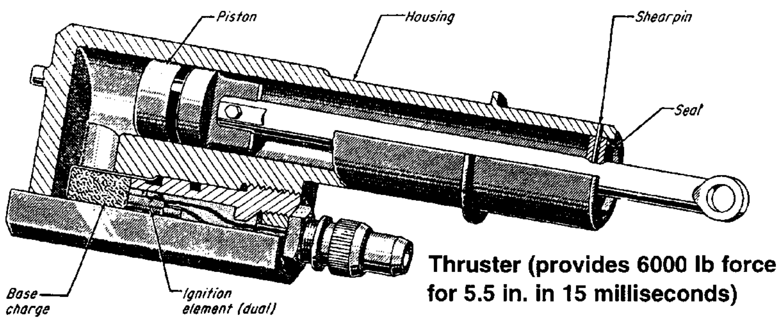
Cartridge-actuated devices generate a punch that cuts cable and pipe, shears bolts for fast release, and provides emergency thrust.



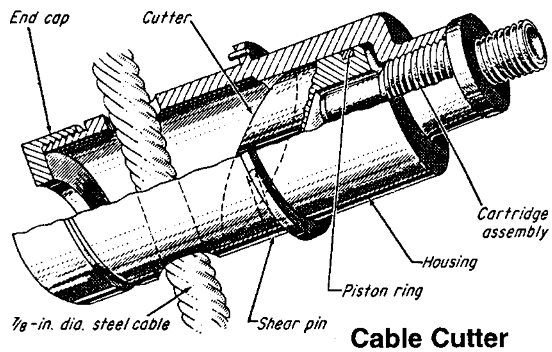
Cartridge Assemblies



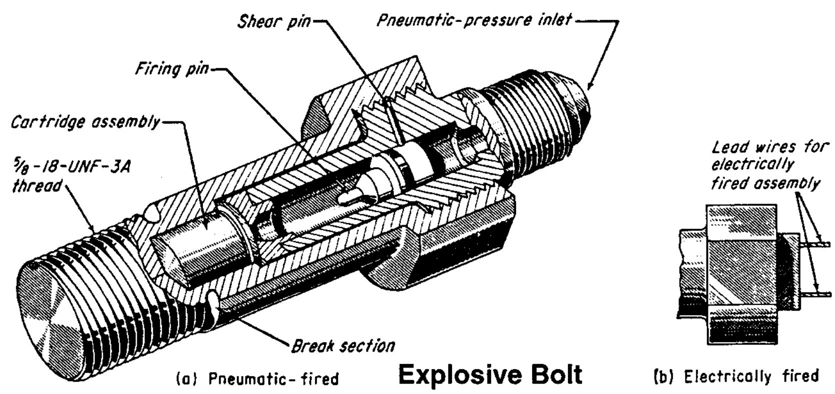
Pipe Cutters



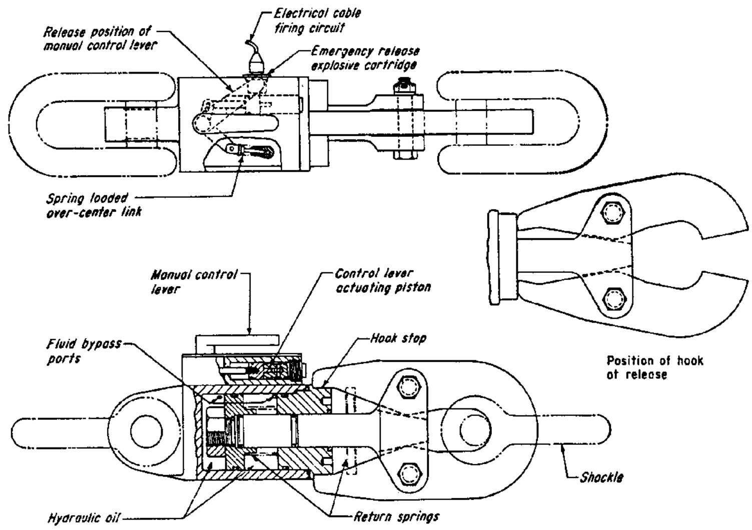
Thruster (provides 6000 lb force for 5.5 in. in 15 milliseconds)



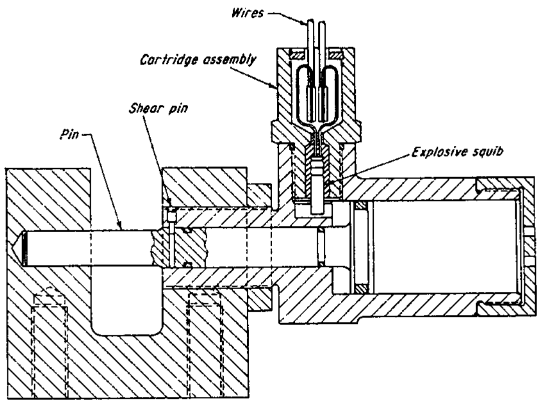
Cable Cutter



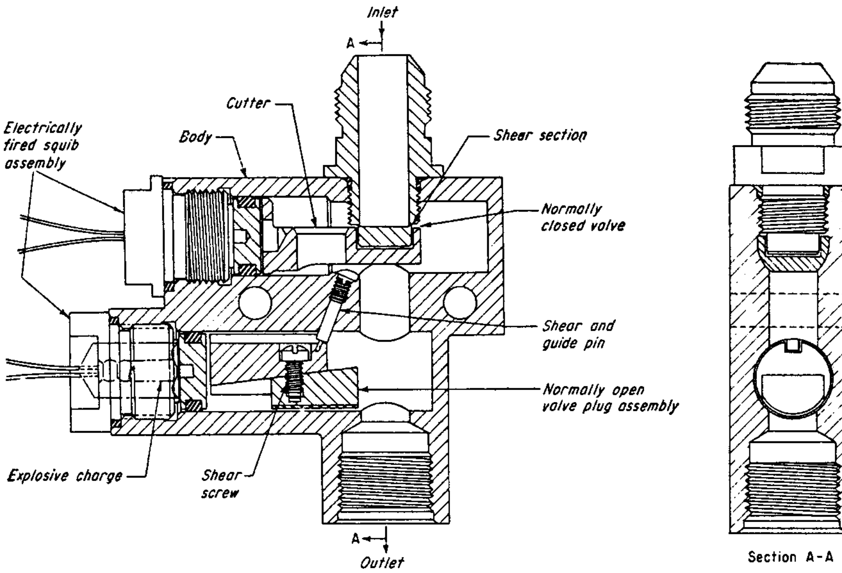
Explosive Bolt



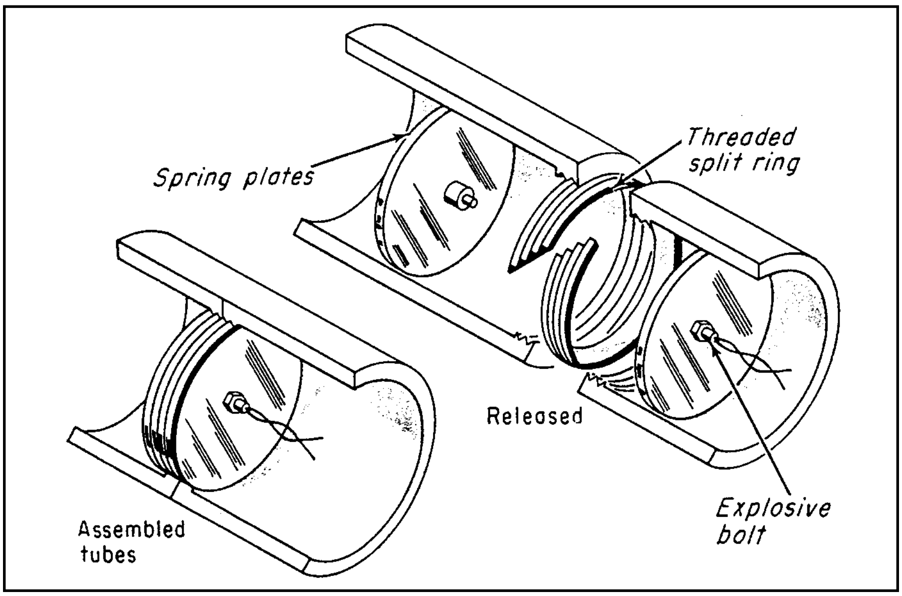
An emergency hook release lets the loads be jettisoned at any time. The hook is designed to release automatically if it is overloaded.



The pin retracts to release the load or clear a channel for free movement.



This dual valve is designed so that the flow will be started and stopped by the same unit. Firing one squib starts the flow; firing the other squib stops the flow.



A threaded split ring with a helical-spring response holds the ends of the tubes at a joint until the explosive bolt is fired; then it releases instantly.

Quick Disconnector

A tube joint can be separated almost instantaneously by remote control with an explosive bolt and a split threaded ring, in a design developed by James Mayo of NASA's Langley Research Center, Hampton, Virginia.

External threads of the ring mesh with the internal threads of the members that are joined—and they must be separated quickly. The ring has a built-in spring characteristic that will assume a helically wound shape and reduce to a smaller diameter when not laterally constrained. During assembly, it is held to its expanded size by two spring plates whose rims fit into internal grooves machined in the split ring. The plates are fastened together by an explosive bolt and nut.

Upon ignition of the explosive bolt, the plates fly apart form the axial spring tension of the ring. The ring then contracts to its normally smaller diameter, releasing the two structural members.

The tube joint can be made in any size and configuration. The retaining media need not be limited to V-type screw threads.

CHAPTER 12

FASTENING, LATCHING, CLAMPING, AND CHUCKING DEVICES

REMOTELY CONTROLLED LATCH

This simple mechanism engages and disengages parallel plates carrying couplings and connectors.

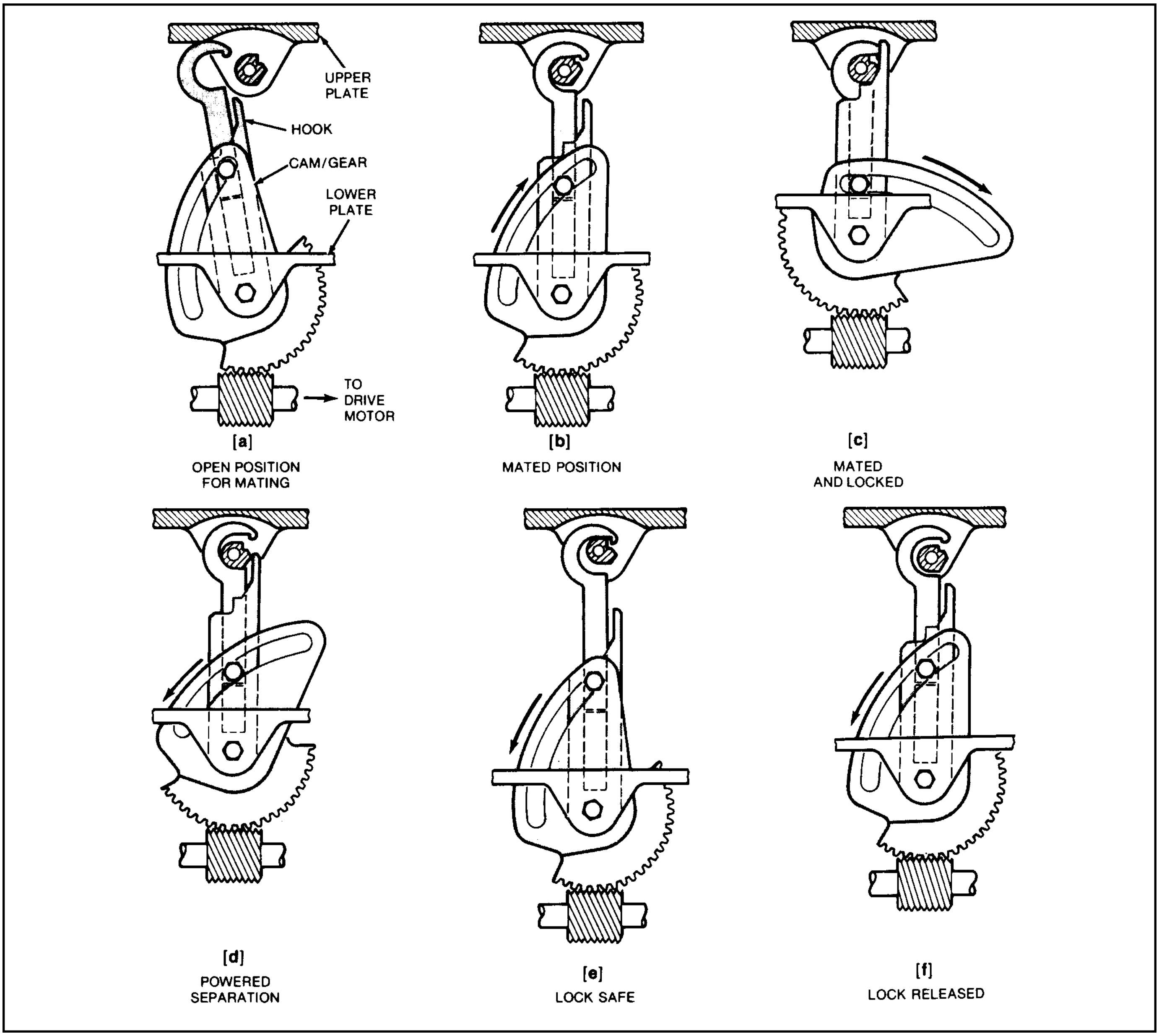


Fig. 1 The latch operation sequence is shown for locking in steps (a) through (c) and for unlocking in steps (d) through (f).

A new latch mates two parallel plates in one continuous motion (see Fig. 1). On the Space Shuttle, the latch connects (and disconnects) plates carrying 20 fluid couplings and electrical connectors. (The coupling/connector receptacles are one plate, and mating plugs are on the other plate). Designed to lock items in place for handling, storage, or processing under remote control, the mechanism also has a fail-safe feature: It does not allow the plates to separate completely unless both are supported. Thus, plates cannot fall apart and injure people or damage equipment.

The mechanism employs four cam/gear assemblies, one at each corner of the lower plate. The gears on each side of the plate face inward to balance the loading and help align the plates. Worm gears on the cam-gear assemblies are connected to a common drive motor.

Figure 1 illustrates the sequence of movements as a pair of plates is latched and unlatched. Initially, the hook is extended and tilted out. The two plates are brought together, and when they are 4.7 in. (11.9 cm) apart, the drive motor is started (a). The worm gear rotates the hook until it closes on a pin on the opposite plate (b). Further rotation of the worm gear shortens the hook extension and raises the lower plate (c). At that point, the couplings and connectors on the two plates are fully engaged and locked.

To disconnect the plates, the worm gear is turned in the opposite direction. This motion lowers the bottom plate and pulls the couplings apart (d). However, if the bottom plate is unsupported, the latch safety feature operates. The hook cannot clear the pin if the lower plate hangs freely (e). If the bottom plate is supported, the hook extension lifts the hook clear of the pin (f) so that the plates are completely separated.

This work was done by Clifford J. Barnett, Paul Castiglione, and Leo R. Coda of Rockwell International Corp. for **Johnson Space Center**.

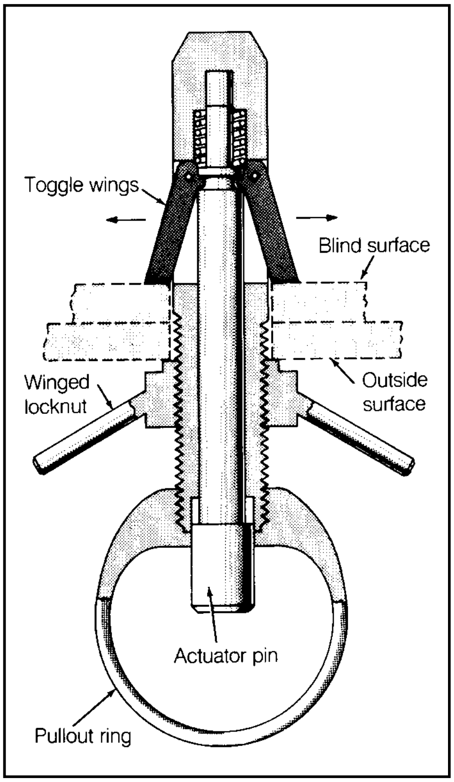
TOGGLE FASTENER INSERTS, LOCKS, AND RELEASES EASILY

A pin-type toggle fastener, invented by C.C. Kubokawa at NASA's Ames Research Center, can be used to fasten plates together, fasten things to walls or decks, or fasten units with surfaces of different curvatured, such as a concave shape to a convex surface.

With actuator pin. The cylindrical body of the fastener has a tapered end for easy entry into the hole; the head is threaded to receive a winged locknut and, if desired, a ring for pulling the fastener out again after release. Slots in the body hold two or more toggle wings that respond to an actuator pin. These wings are extended except when the spring-loaded pin is depressed.

For installation, the actuator pin is depressed, retracting the toggle wings. When the fastener is in place, the pin is released, and the unit is then tightened by screwing the locknut down firmly. This exerts a compressive force on the now-expanded toggle wings. For removal, the locknut is loosened and the pin is again depressed to retract the toggle wings. Meanwhile, the threaded outer end of the cylindrical body functions as a stud to which a suitable pull ring can be screwed to facilitate removal of the fastener.

This invention has been patented by NASA (U.S. Patent No. 3,534,650).



A fastener with controllable toggles can be inserted and locked from only one side.

GRAPPLE FREES LOADS AUTOMATICALLY

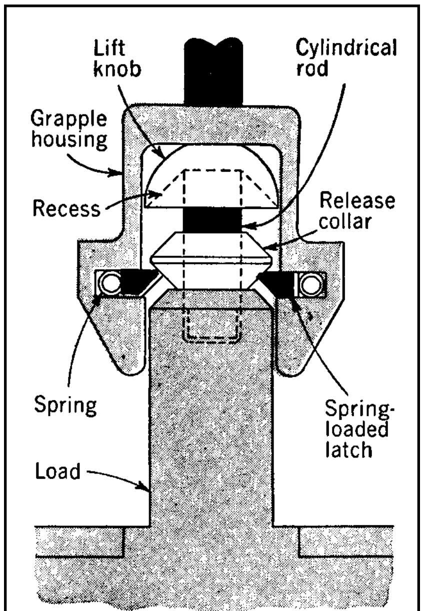
A simple grapple mechanism, designed at Argonne National Laboratory in Illinois, engages and releases loads from overhead cranes automatically. This self-releasing mechanism was developed to remove fuel rods from nuclear reactors. It can perform tasks where human intervention is hazardous or inefficient, such as lowering and releasing loads from helicopters.

The mechanism (see drawing) consists of two pieces: a lift knob secured to the load and a grapple member attached to the crane. The sliding latch-release collar under the lift knob is the design's key feature.

Spring magic. The grapple housing, which has a cylindrical inner surface, contains a machined groove fitted with a garter spring and three metal latches. When the grapple is lowered over the lift knob, these latches recede into the groove as their edges come into contact with the knob. After passing the knob, they spring forward again, locking the grapple to the knob. Now the load can be lifted.

When the load is lowered to the ground again, gravity pull or pressure from above forces the grapple housing down until the latches come into contact with a double cone-shaped release collar. The latches move back into the groove as they pass over the upper cone's surface and move forward again when they slide over the lower cone.

The grapple is then lifted so that the release collar moves up the cylindrical rod until it is housed in a recess in the lift knob. Because the collar can move no farther, the latches are forced by the upward pull to recede again into the groove—allowing the grapple to be lifted free.



A sliding release collar is a key feature of this automatic grapple.

QUICK-RELEASE LOCK PIN HAS A BALL DETENT

A novel quick-release locking pin has been developed that can be withdrawn to separate the linked members only when stresses on the joint are negligible.

The pin may be the answer to the increasing demand for locking pins and fasteners that will pull out quickly and easily when desired, yet will stay securely in place without chance of unintentional release.

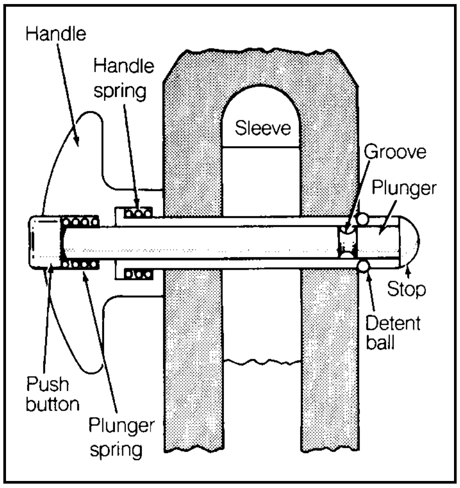
The key to this foolproof pin is a group of detent balls and a matching grooved. The ball must be in the groove whenever the pin is either installed or pulled out of the assembly. This is easy to do during installation, but during removal the load must be off the pin to get the balls to drop into the groove.

How it works. The locking pin was developed by T.E. Othman, E.P. Nelson, and L.J. Zmuda under contract to NASA's Marshall Space Flight Center. It consists of a forward-pointing sleeve with a spring-loaded sliding handle as its rear end, housing a sliding plunger that is pushed backward (to its locking position) by a spring within the handle.

To some extend the plunger can slide forward against the plunger spring, and the handle can slide backward against the handle spring. A groove near the front end of the plunger accommodates the detent balls when the plunger is pushed forward by the compression of its spring. When the plunger is released backward, the balls are forced outward into holes in the sleeve, preventing withdrawal of the pin.

To install the pin, the plunger is pressed forward so that the balls fall into their groove and the pin is pushed into the hole. When the plunger is released, the balls lock the sleeve against accidental withdrawal.

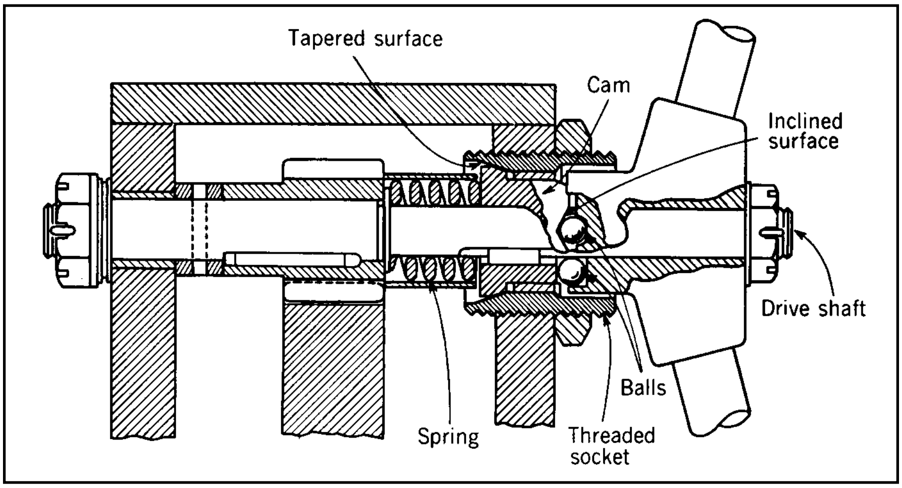
To withdraw the pin, the plunger is pressed forward to accommodate the locking balls, and at the same time the handle is pulled backward. If the loading on the pin is negligible, the pin is withdrawn from the joint; if it is considerable, the handle spring is compressed and the plunger is forced backward by the handle so the balls will return to their locking position.



A foolproof locking pin releases quickly when the stress on the joint is negligible.

The allowable amount of stress on the joint that will permit its removal can be varied by adjusting the pressure required for compressing the handle spring. If the stresses on the joint are too great or the pin to be withdrawn in the normal manner, hammering on the forward end of the plunger simply ensures that the plunger remains in its rearward position, with the locking balls preventing the withdrawal of the pin. A stop on its forward end prevents the plunger from being driven backward.

AUTOMATIC BRAKE LOCKS HOIST WHEN DRIVING TORQUE CEASES



When torque is removed, the cam is forced into the tapered surface for brake action.

A brake mechanism attached to a chain hoist is helping engineers lift and align equipment accurately by automatically locking it in position when the driving torque is removed from the hoist.

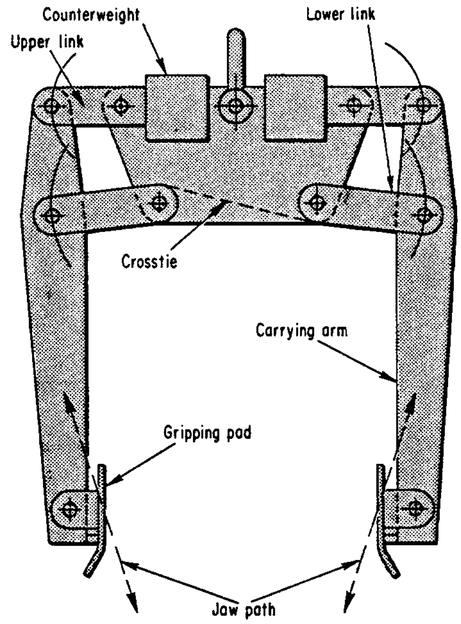
According to the designer, Joseph Pizzo, the brake could also be used on wheeled equipment operating on slopes, to act as an auxiliary brake system.

How it works. When torque is applied to the driveshaft (as shown in the figure), four steel balls try to move up the inclined surfaces of the cam. Although called a cam by the designer, it is really a concentric collar with a cam-like surface on one of its end faces. Because the balls are contained by four cups in the hub, the cam is forced to move forward axially to the left. Because the cam moves away from the tapered surface, the cam and the driveshaft that is keyed to it are now free to rotate.

If the torque is removed, a spring resting against the cam and the driveshaft gear forces the cam back into the tapered surface of the threaded socket for instant braking.

Although this brake mechanism (which can rotate in either direction) was designed for manual operation, the principle can be applied to powered systems.

LIFT-TONG MECHANISM FIRMLY GRIPS OBJECTS



Twin four-bar linkages are the key components in this long mechanism that can grip with a constant weight-to-grip force ratio any object that fits within its grip range. The long mechanism relies on a cross-tie between the two sets of linkages to produce equal and opposite linkage movement. The vertical links have extensions with grip pads mounted at their ends, while the horizontal links are so proportioned that their pads move in an inclined straight-line path. The weight of the load being lifted, therefore, wedges the pads against the load with a force that is proportional to the object's weight and independent of its size.

PERPENDICULAR-FORCE LATCH

The installation and removal of equipment modules are simplified.

A latching mechanism simultaneously applies force in two perpendicular directions to install or remove electronic-equipment modules. The mechanism (see Fig. 1) requires only the simple motion of a handle to push or pull an avionic module to insert or withdraw connectors on its rear face into or from spring-loaded mating connectors on a panel and to force the box downward onto or release the box from a mating cold plate that is part of the panel assembly. The concept is also adaptable to hydraulic, pneumatic, and mechanical systems. Mechanisms of this type can simplify the manual installation and removal of modular equipment where a technician's movement is restricted by protective clothing, as in hazardous environments, or where the installation and removal are to be performed by robots or remote manipulators.

Figure 2 sows an installation sequence. In step 1, the han-

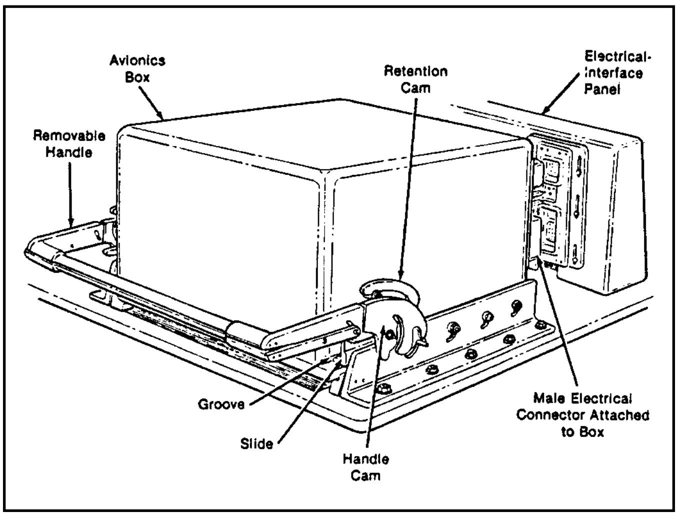


Fig. 1 An avionics box mates with electrical connectors in the rear and is locked in position on the cold plate when it is installed with the latching mechanism.

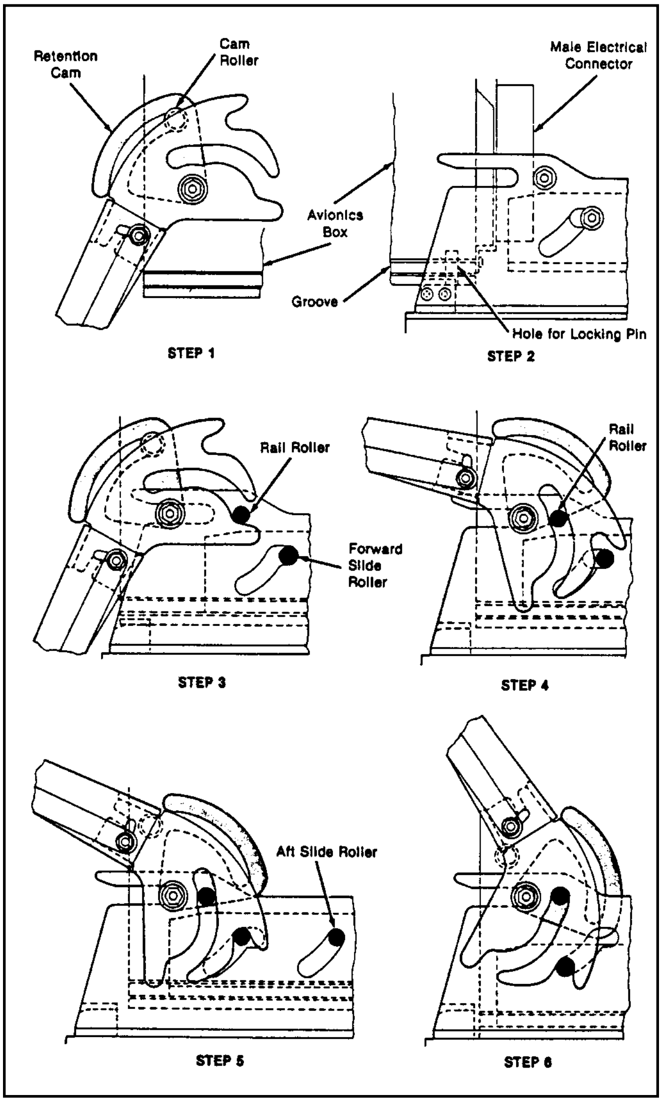


Fig. 2 This installation sequence shows the positions of the handle and retention cams as the box is moved rearward and downward.

dle has been installed on the handle cam and turned downward. In step 2, the technician or robot pushes the box rearward as slides attached to the rails enter grooves near the bottom of the box. In step 3, as the box continues to move to the rear, the handle cam automatically aligns with the slot in the rail and engages the rail roller.

In step 4, the handle is rotated upward 75º, forcing the box rearward to mate with the electrical connectors. In step 5, the handle is pushed upward an additional 15º, locking the handle cam and the slide. In step 6, the handle is rotated an additional 30º, forcing the box and the mating spring-loaded electrical connectors downward so that the box engages the locking pin and becomes clamped to the cold plate. The sequence for removal is identical except that the motions are reversed.

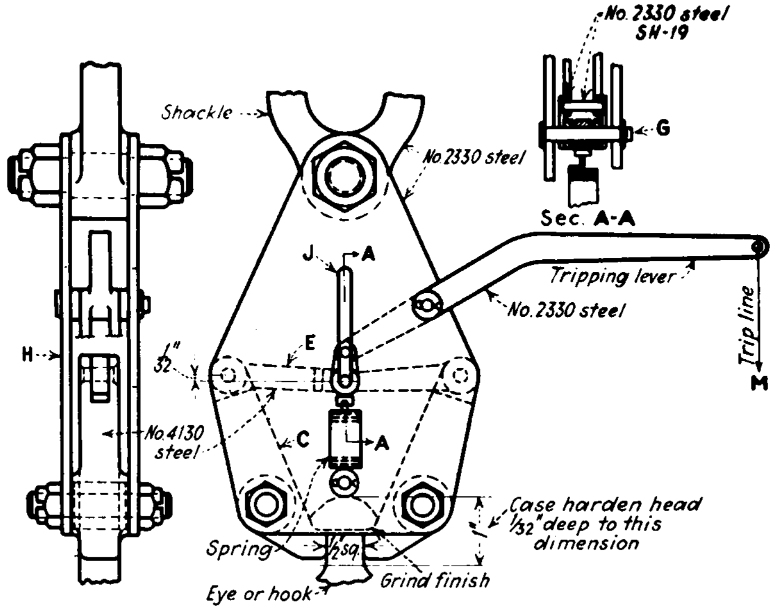
QUICK-RELEASE MECHANISMS

QUICK-RELEASE MECHANISM

Quick release mechanisms have many applications. Although the design shown here operates as a tripping device for a quick-release hook, the mechanical principles involved have many other applications. Fundamentally, it is a toggle-type mechanism with the characteristic that the greater the load the more effective the toggle.

The hook is suspended from the shackle, and the load or work is supported by the latch, which is machined to fit the fingers C. The fingers C are pivoted about a pin. Assembled to the fingers are the arms E, pinned at one end and joined at the other by the sliding pin G. Enclosing the entire unit are the side plates H, containing the slot J for guiding the pin G in a vertical movement when the hook is released. The helical spring returns the arms to the bottom position after they have been released.

To trip the hook, the tripping lever is pulled by the cable M until the arms E pass their horizontal center-line. The toggle effect is then broken, releasing the load.



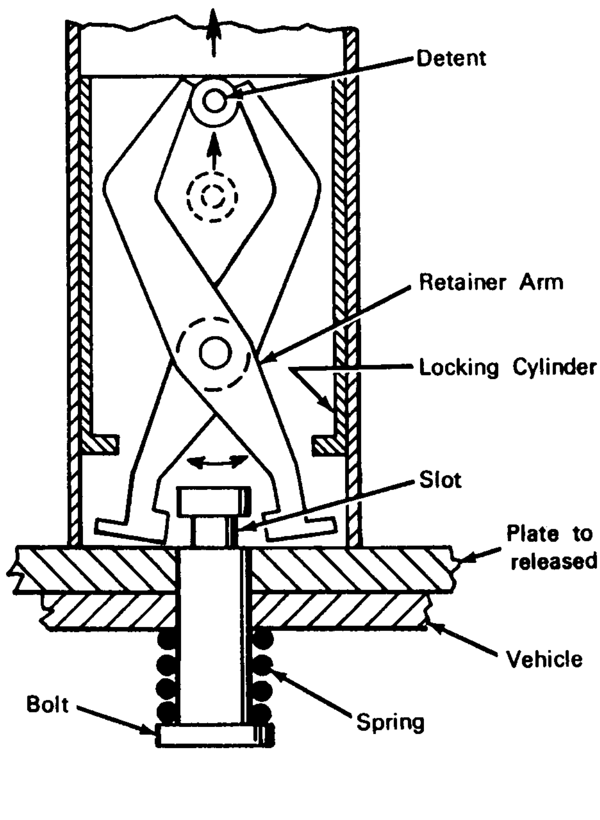
A simple quick-release toggle mechanism was designed for tripping a lifting hook.

POSITIVE LOCKING AND QUICK-RELEASE MECHANISM

The object here was to design a simple device that would hold two objects together securely and quickly release them on demand.

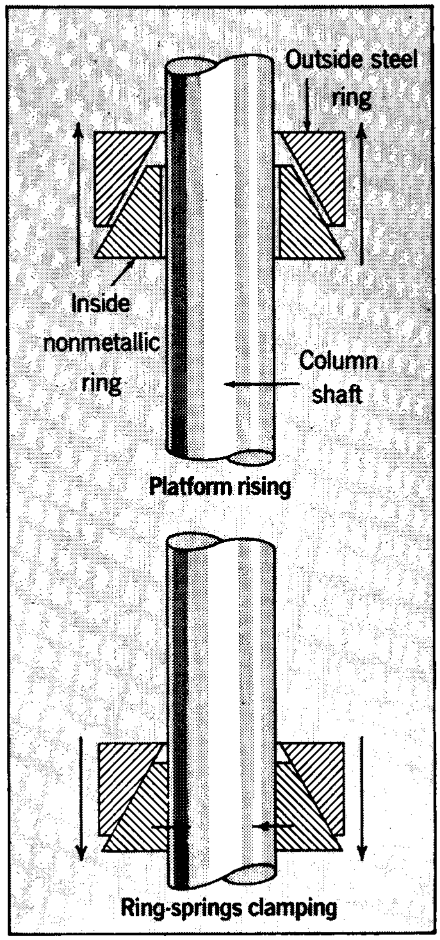
One object, such as a plate, is held to another object, such as a vehicle, by a spring-loaded slotted bolt, which is locked in position by two retainer arm. The retainer arms are constrained from movement by a locking cylinder. To release the plate, a detent is actuated to lift the locking cylinder and rotate the retainer arms free from contact with the slotted bolt head. As a result of this action, the spring-loaded bolt is ejected, and the plate is released from the vehicle.

The actuation of the slidable detent can be initiated by a squib, a fluid-pressure device, or a solenoid. The principle of this mechanism can be applied wherever a positive engagement that can be quickly released on demand is required. Some suggested applications for this mechanism are in coupling devices for load-carrying carts or trucks, hooks or pick-up attachments for cranes, and quick-release mechanisms for remotely controlled manipulators.



This quick-release mechanism is shown locking a vehicle and plate.

RING SPRINGS CLAMP PLATFORM ELEVATOR INTO POSITION



Ring springs unclamp the column as the platform is raised (upper). As soon as the jack power is removed (lower), the column is gripped by the inner ring.

A simple yet effective technique keeps a platform elevator locked safely in position without an external clamping force. The platform (see drawing) contains special ring assemblies that grip the four column-shafts with a strong force by the simple physical interaction of two tapered rings.

Thus, unlike conventional platform elevators, no outside power supply is required to hold the platform in position. Conventional jacking power is employed, however, in raising the platform from one position to another.

How the rings work. The ring assemblies are larger versions of the ring springs sometimes installed for shock absorption. In this version, the assembly is made up of an inner nonmetallic ring tapering upward and an outer steel ring tapered downward (see drawing).

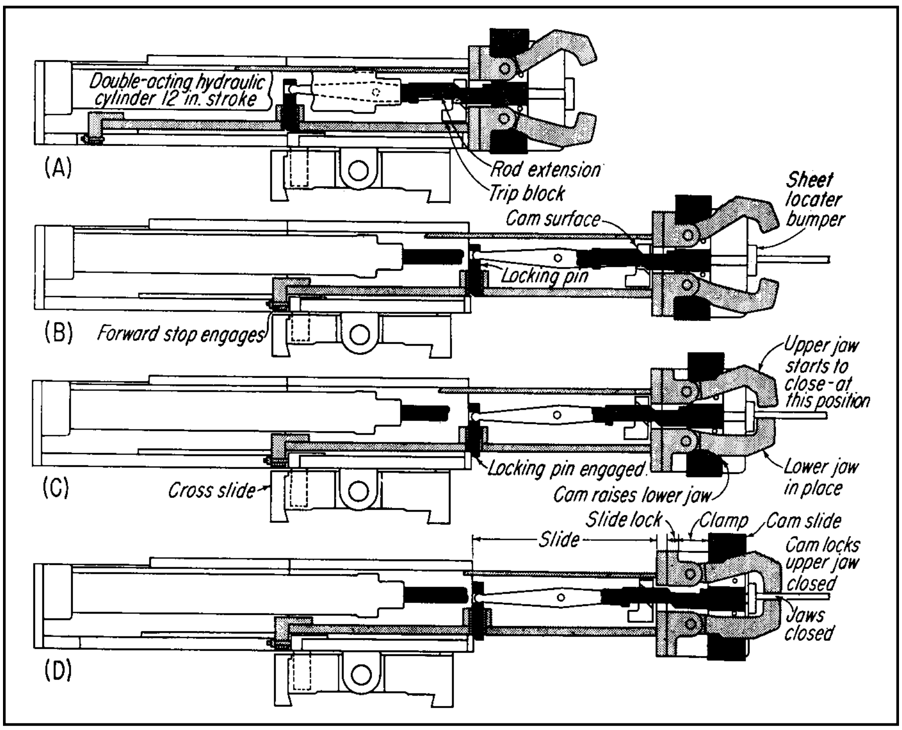
The outside ring is linked to the platform, and the inside ring is positioned against the circumference of the column shaft. When the platform is raised to the designed height, the jack force is removed, and the full weight of the platform bears downward on the outside ring with a force that, through a wedging action, is transferred into a horizontal inward force of the inside ring.

Thus, the column shaft is gripped tightly by the inside ring; the heavier the platform the larger the gripping force produced.

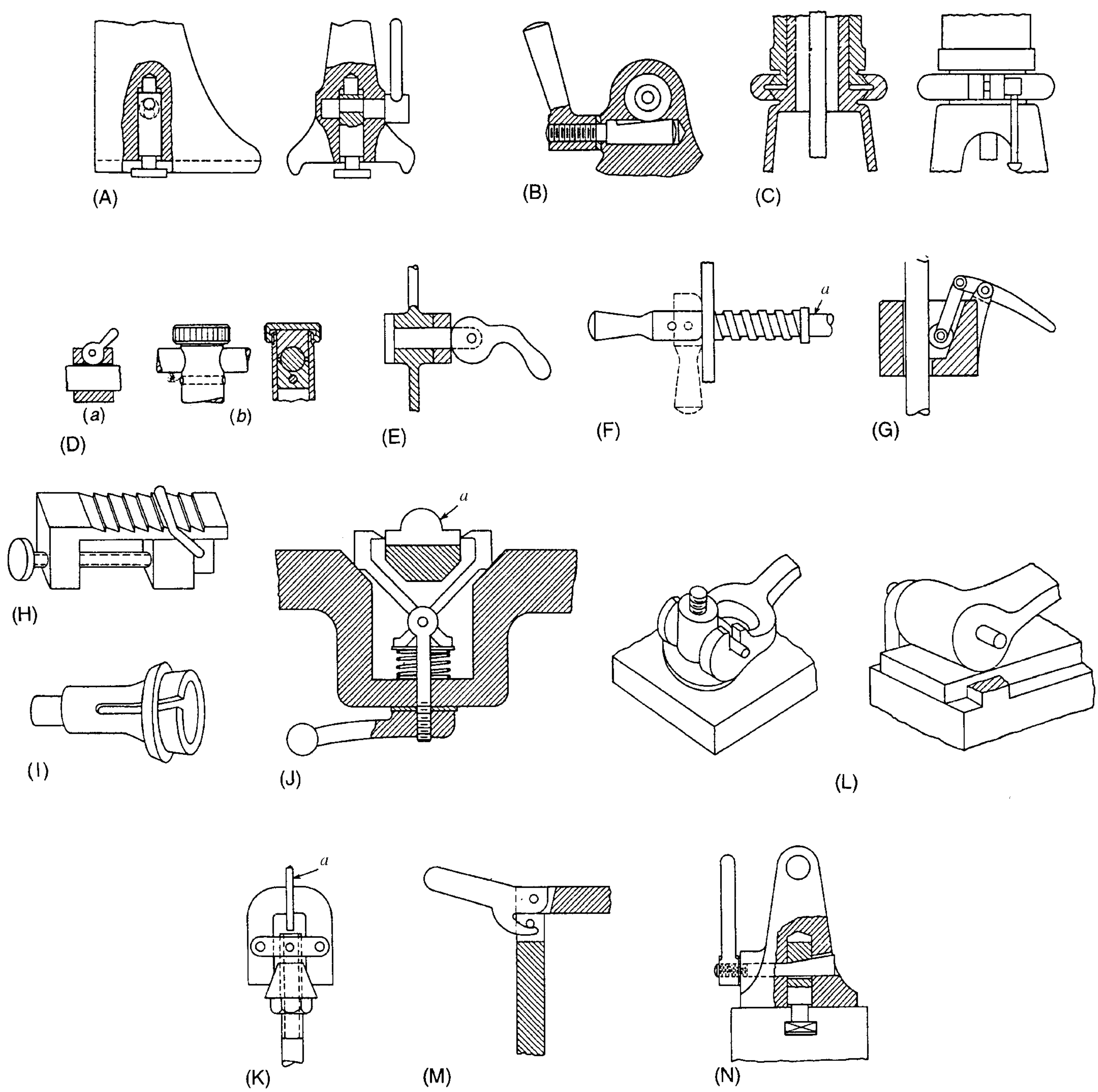
The advantage of the technique is that the shafts do not need notches or threads, and cost is reduced. Moreover, the shafts can be made of reinforced concrete.

CAMMED JAWS IN HYDRAULIC CYLINDER GRIP SHEETS

A single, double-acting hydraulic cylinder in each work holder clamps and unclamps the work and retracts or advances the jaws as required. With the piston rod fully withdrawn into the hydraulic cylinder (A), the jaws of the holder are retracted and open. When the control valve atop the work holder is actuated, the piston rod moves forward a total of 12 in. The first 10 in. of movement (B) brings the sheet-locater bumper into contact with the work. The cammed surface on the rod extension starts to move the trip block upward, and the locking pin starts to drop into position. The next ¾ in. of piston-rod travel (C) fully engages the work-holder locking pin and brings the lower jaw of the clamp up to the bottom of the work. The work holder slide is now locked between the forward stop and the locking pin. The last 1¼ in. of piston travel (D) clamps the workpiece between the jaws with a pressure of 2500 lbs. No adjustment for work thickness is necessary. A jaws-open limit switch clamps the work holder in position (C) for loading and unloading operations.

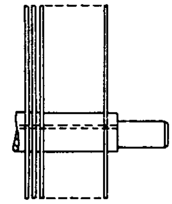


QUICK-ACTING CLAMPS FOR MACHINES AND FIXTURES

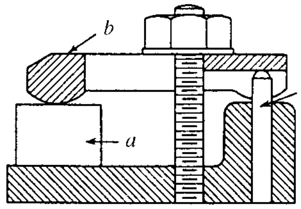


(A) An eccentric clamp. (B) A spindle-clamping bolt. (C) A method for clamping a hollow column to a structure. It permits quick rotary adjustment of the column. (D) (a) A cam catch for clamping a rod or rope. (b) A method for fastening a small cylindrical member to a structure with a thumb nut and clamp jaws. It permits quick longitudinal adjustment of a shaft in the structure. (E) A cam catch can lock a wheel or spindle. (F) A spring handle. Movement of the handle in the vertical or horizontal position provides movement at *a*. (G) A roller and inclined slot for locking a rod or rope. (H) A method for clamping a light member to a structure. The serrated edge on the structure permits the rapid accommodation of members with different thicknesses. (I) A spring taper holder with a sliding ring. (J) A special clamp for holding member *a*. (K) The cone, nut, and levers grip member *a*. The grip can have two or more jaws. With only two jaws, the device serves as a small vise. (L) Two different kinds of cam clamps. (M) A cam cover catch. Movement of the handle downward locks the cover tightly. (N) The sliding member is clamped to the slotted structure with a wedge bolt. This permits the rapid adjustment of a member on the structure.

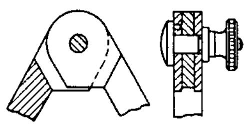
From *Handbook of Fastening and Joining of Metal Parts*, McGraw-Hill, Inc.



(A)



(B)



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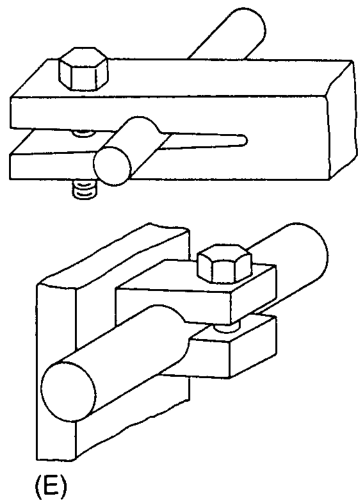
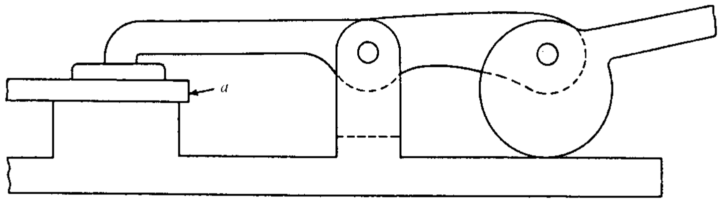
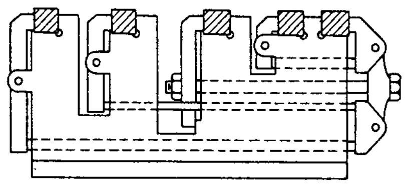


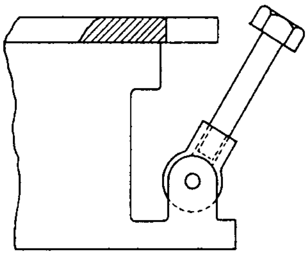
(E)



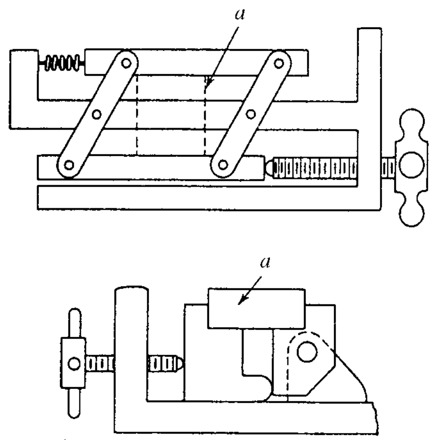
(D)



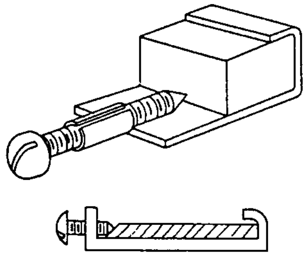
(G)



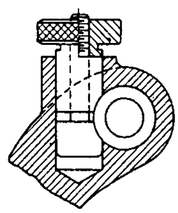
(H)



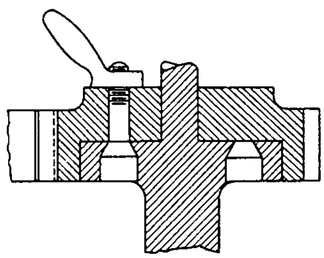
(F)



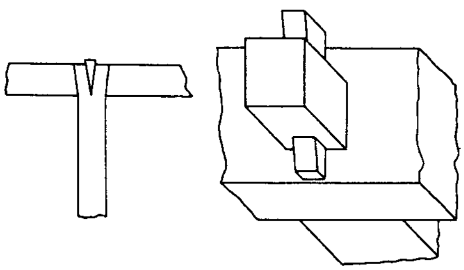
(I)



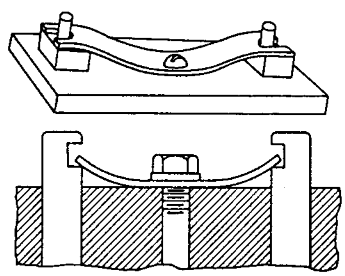
(J)



(K)



(L)

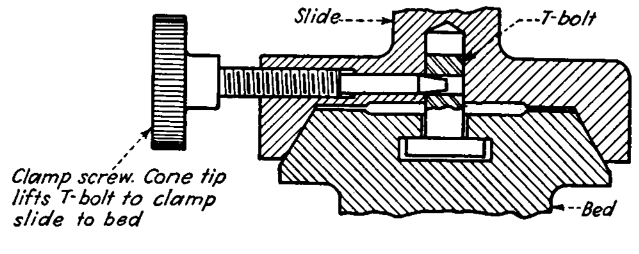


(M)

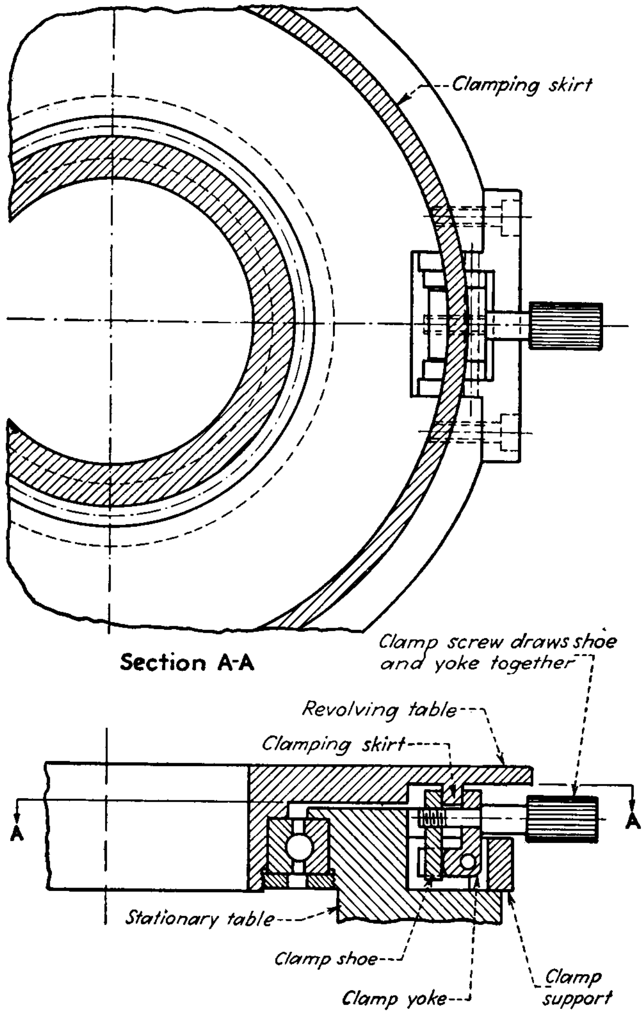
(A) A method for fastening capacitor plates to a structure with a circular wedge. Rotation of the plates in a clockwise direction locks the plates to the structure. (B) A method for clamping member *a* with a special clamp. Detail *b* pivots on pin *c.* (C) A method for clamping two movable parts so that they can be held in any angular position with a clamping screw. (D) A cam clamp for clamping member *a.* (E) Two methods for clamping a cylindrical member. (F) Two methods for clamping member *a* with a special clamp. (G) A special clamping device that permits the parallel clamping of five parts by the tightening of one bolt. (H) A method for securing a structure with a bolt and a movable detail that provides a quick method for fastening the cover. (I) A method for quickly securing, adjusting, or releasing the center member. (J) A method for securing a bushing in a structure with a clamp screw and thumb nut. (K) A method for securing an attachment to a structure with a bolt and hand lever used as a nut. (L) A method for fastening a member to a structure with a wedge. (M) Two methods for fastening two members to a structure with a spring and one screw. The members can be removed without loosening the screw.

FRICTION CLAMPING DEVICES

Many different devices for gaining mechanical advantage have been used in the design of friction clamps. These clamps can grip moderately large loads with comparatively small smooth surfaces, and the loads can be tightened or released with simple controls. The clamps illustrated here can be tightened or released with screws, levers, toggles, wedges, and combinations of them.

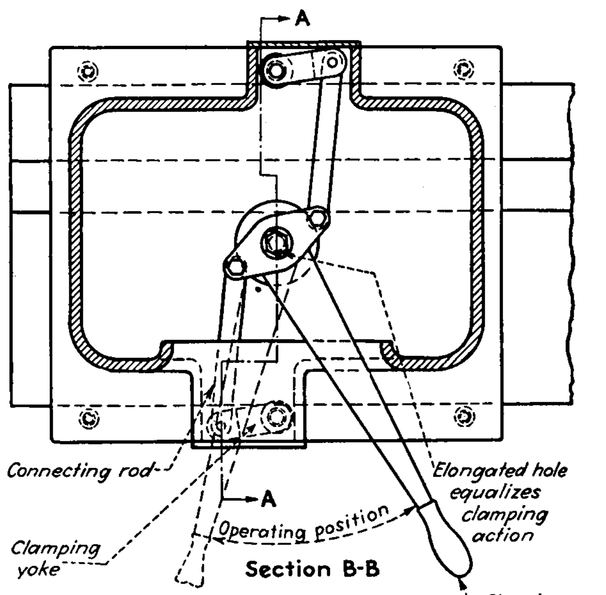


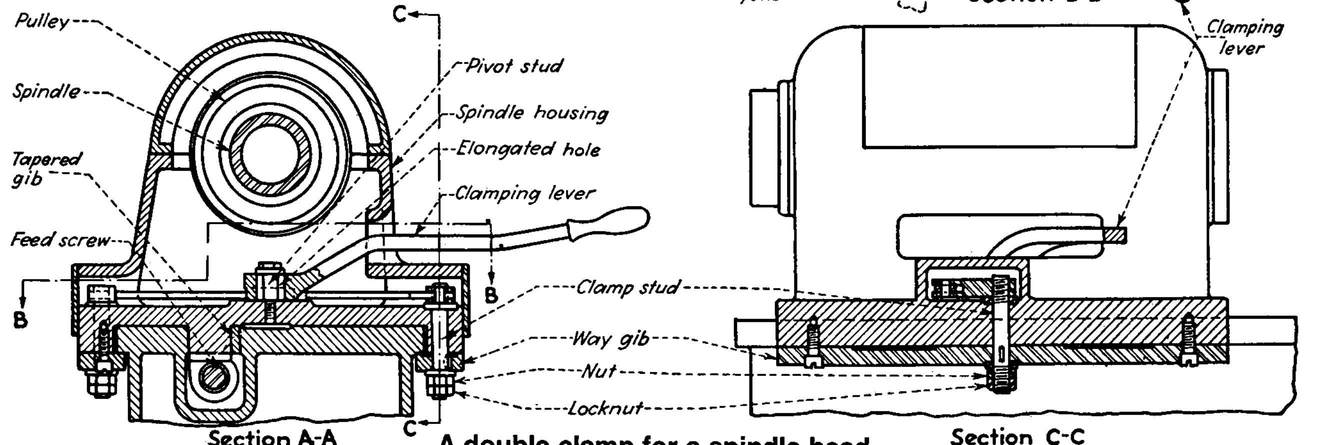
A slide clamp.



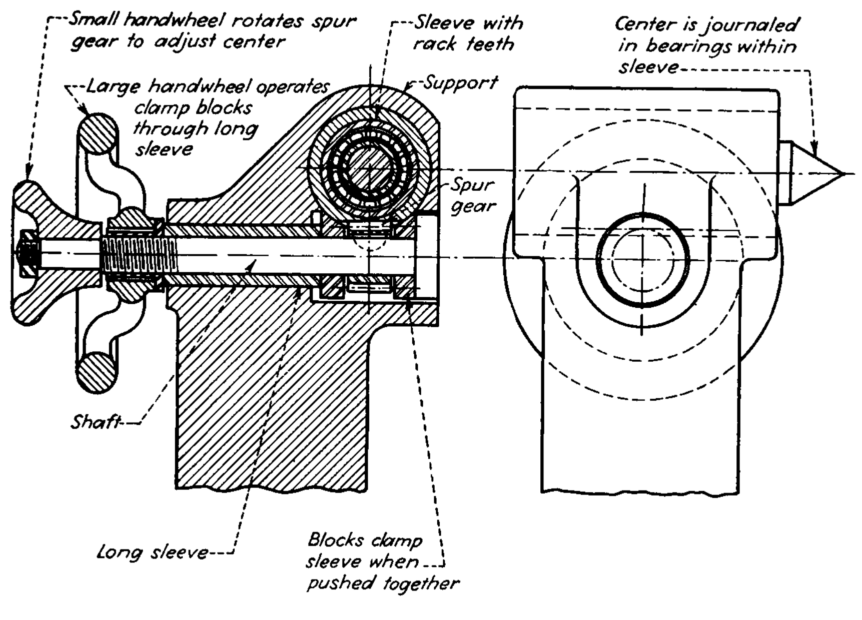
A turntable clamp.

The clamp assembly floats on a pin and does not disturb the table setting.

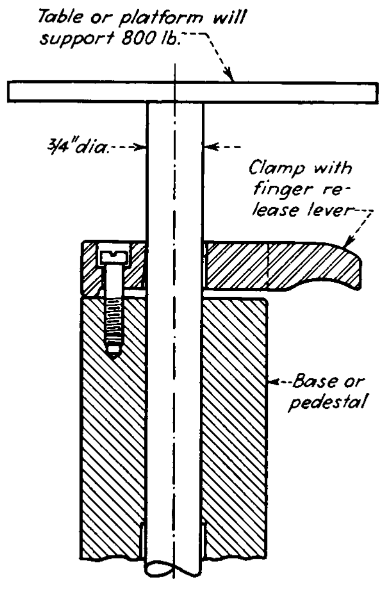




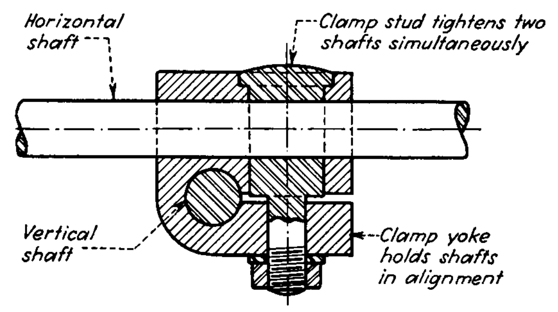
A double clamp for a spindle head.



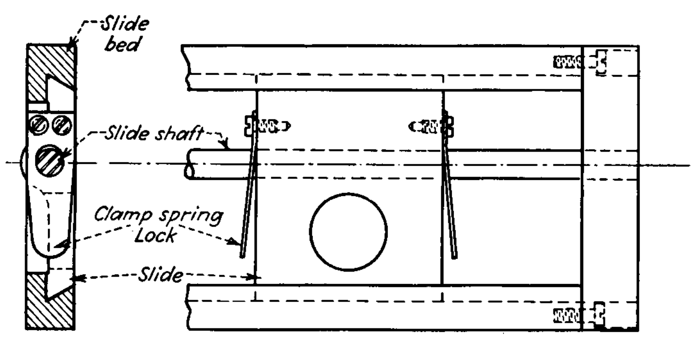
A center-support clamp.



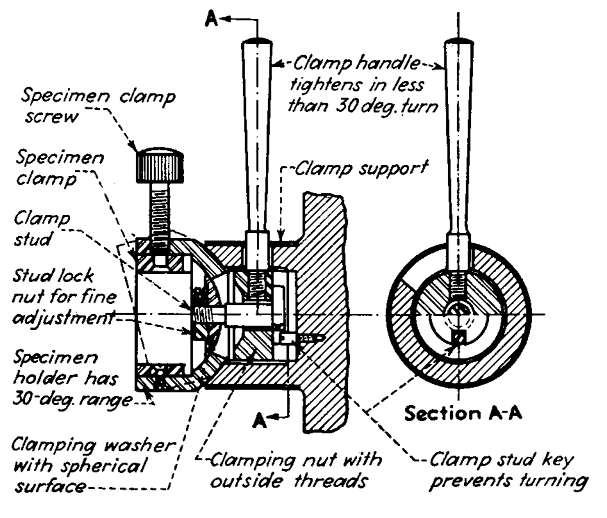
A pedestal clamp.



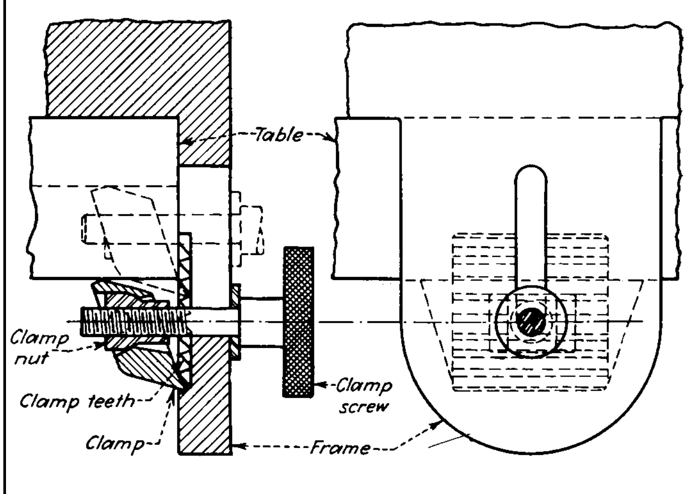
A right-angle clamp.



A slide clamp.



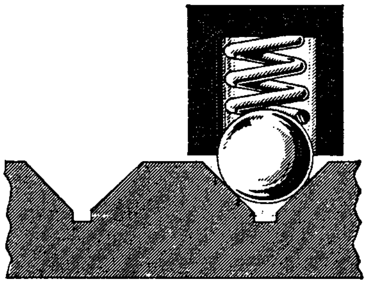
A specimen holder clamp.



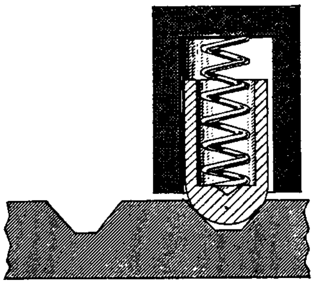
A table clamp.

DETENTS FOR STOPPING MECHANICAL MOVEMENTS

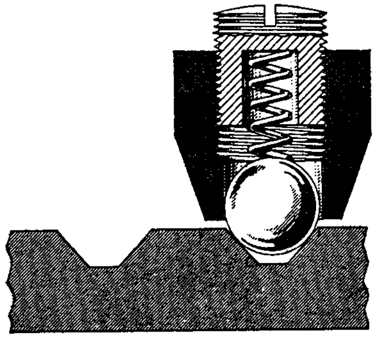
Some of the more robust and practical devices for stopping mechanical movements are illustrated here.



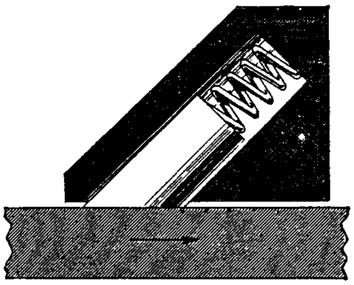
Fixed holding power is constant in both directions.



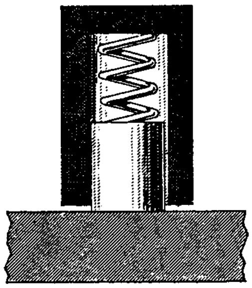
A domed plunger has long life.



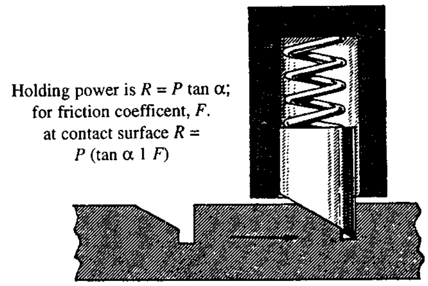
The screw provides adjustable holding.



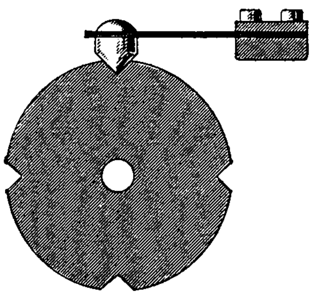
Wedge action locks the movement in the direction of the arrow.



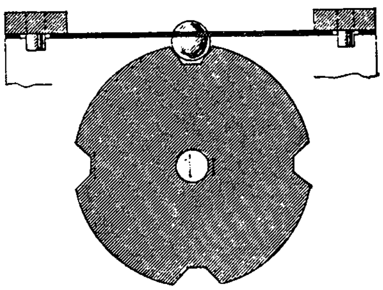
Friction results in holding force.



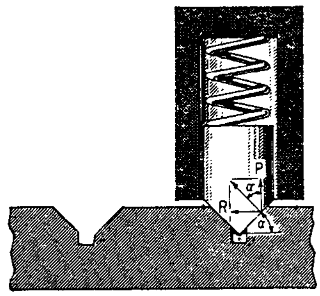
A notch shape dictates the direction of rod motion.



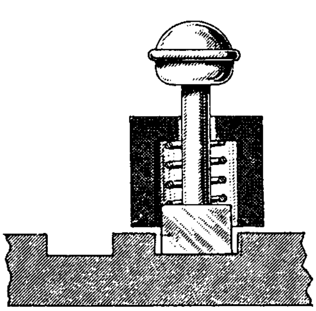
A leaf spring provides limited holding power.



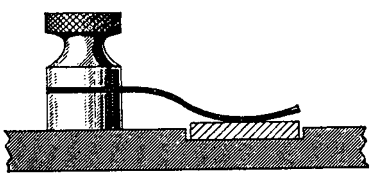
A leaf-spring detent can be removed quickly.



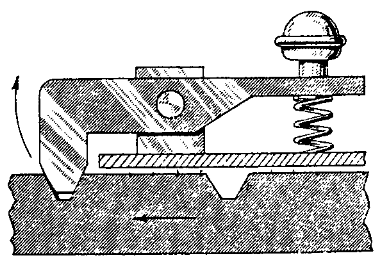
A conical or wedge-ended detent.



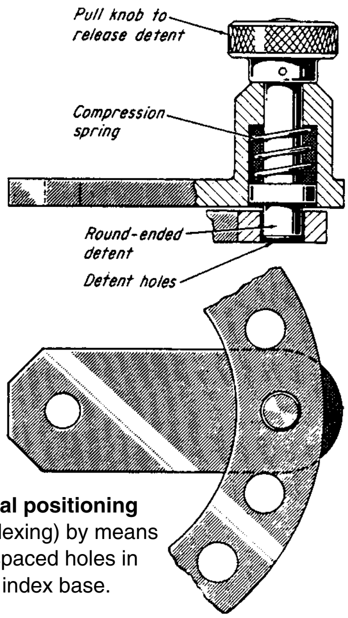
A positive detent has a manual release.



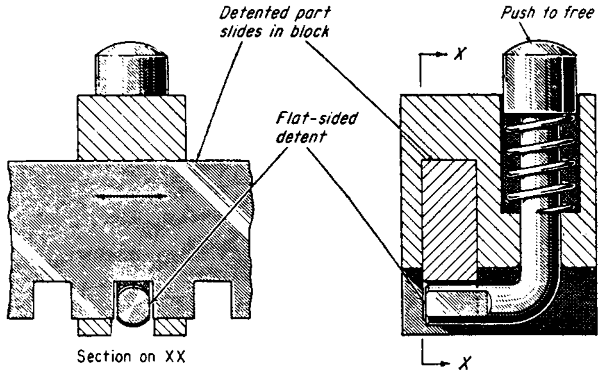
.**A leaf spring** for holding flat pieces.



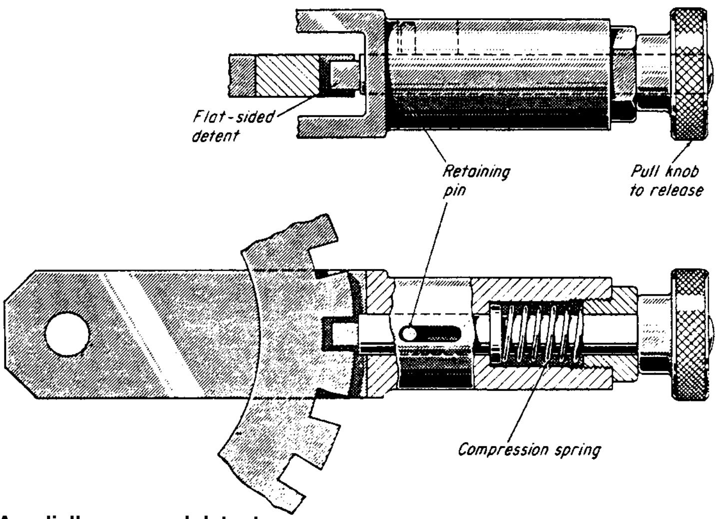
An automatic release occurs in one direction; manual release is needed in the other direction.



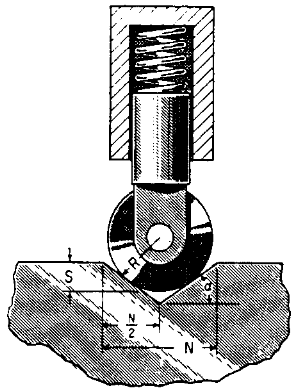
Axial positioning (indexing) by means of spaced holes in the index base.



A positive detent has a push-button release for straight rods.



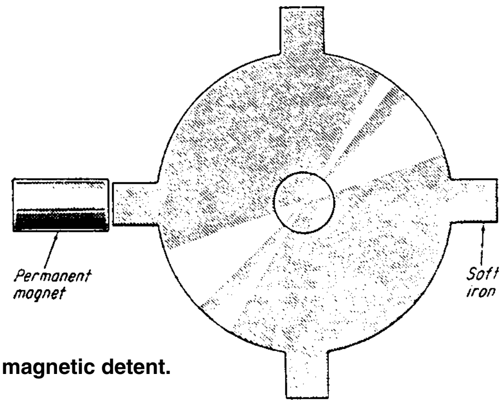
A radially arranged detent holds in slotted index base.



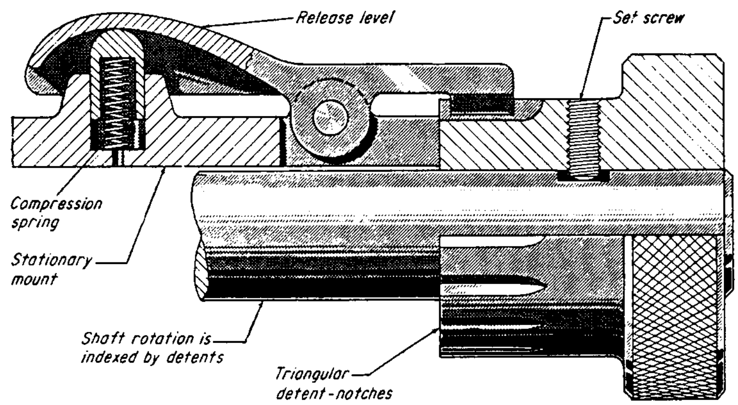
A roller detent positions itself in a notch.

$$\text{Rise, } S = \frac{N \tan a}{2} - R \times \frac{1 - \cos}{\cos a}$$

$$\text{Roller Radius, } R = \left(\frac{N \tan a}{2} - S\right)\left(\frac{\cos a}{1 - \cos a}\right)$$



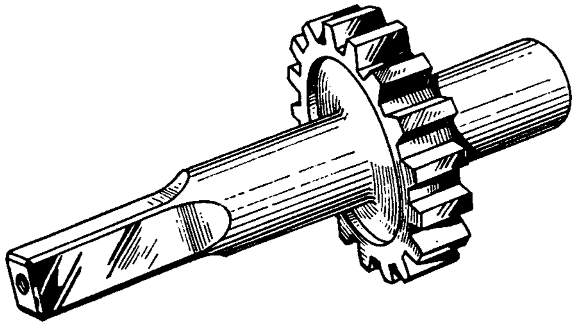
A magnetic detent.



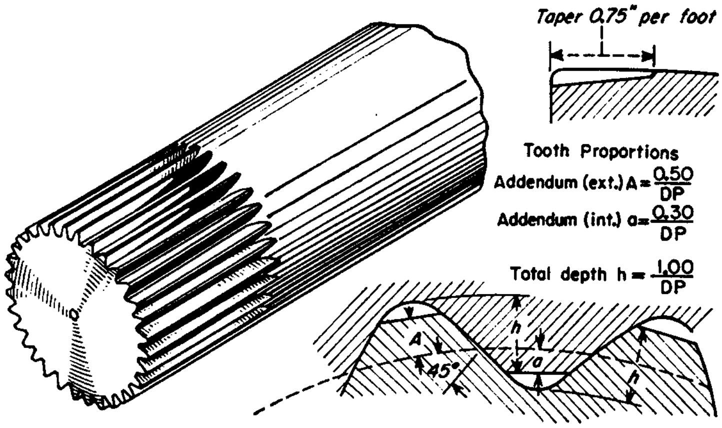
An axial detent for the positioning of the adjustment knob with a manual release.

TEN DIFFERENT SPLINED CONNECTIONS

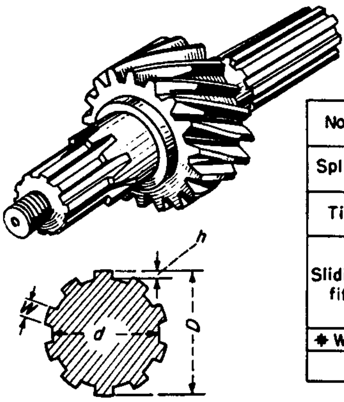
CYLINDRICAL SPLINES



1. SQUARE SPLINES make simple connections. They are used mainly for transmitting light loads, where accurate positioning is not critical. This spline is commonly used on machine tools; a cap screw is required to hold the enveloping member.



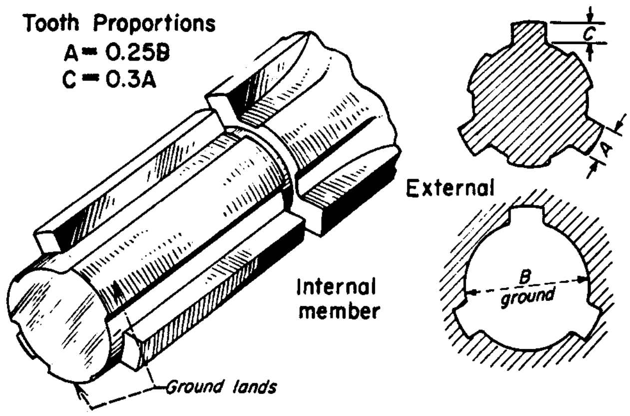
2. SERRATIONS of small size are used mostly for transmitting light loads. This shaft forced into a hole of softer material makes an inexpensive connection. Originally straight-sided and limited to small pitches, 45° serrations have been standardized (SAE) with large pitches up to 10 in. dia. For tight fits, the serrations are tapered.



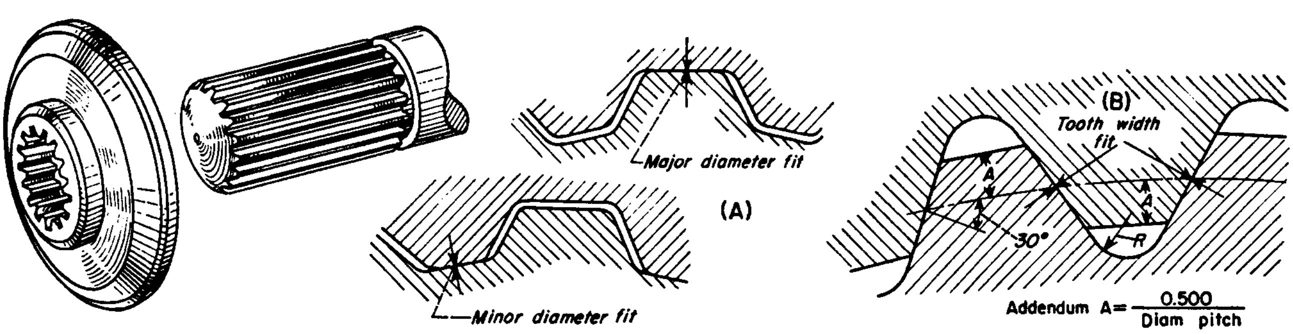
SAE STANDARD SPLINE PROPORTIONS

No. of splines			4	6	10 or 16
Spline width		W	0.241D	0.250D	0.156D*
Tight fit		h	0.075D	0.050D	0.045D
Sliding fit	Unloaded	h	0.125D	0.075D	0.070D
	loaded	h	——	0.100D	0.095D
* Width of 16-tooth spline is 0.098D					
Root dia, d = D − 2h					

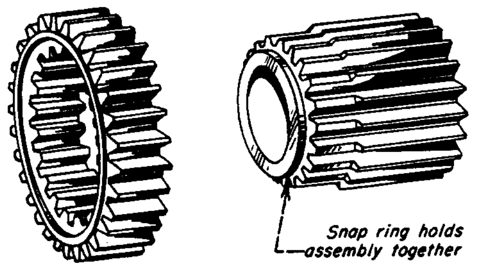
3. STRAIGHT-SIDED splines have been widely used in the automotive field. Such splines are often used for sliding members. The sharp corner at the root limits the torque capacity to pressures of approximately 1,000 psi on the spline projected area. For different applications, tooth height is altered, as shown in the table above.



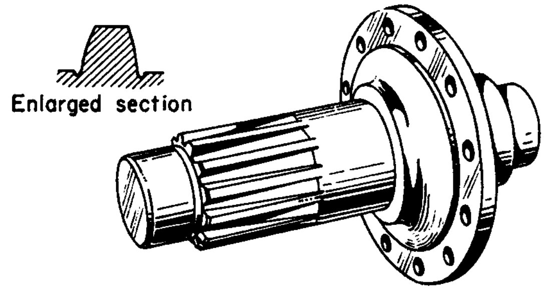
4. MACHINE-TOOL splines have wide gaps between splines to permit accurate cylindrical grinding of the lands—for precise positioning. Internal parts can be ground readily so that they will fit closely with the lands of the external member.



5. INVOLUTE-FORM splines are used where high loads are to be transmitted. Tooth proportions are based on a 30° stub tooth form. (A) Splined members can be positioned either by close fitting major or minor diameters. (B) Use of the tooth width or side positioning has the advantage of a full fillet radius at the roots. Splines can be parallel or helical. Contact stresses of 4,000 psi are used for accurate, hardened splines. The diametral pitch shown is the ratio of teeth to the pitch diameter.

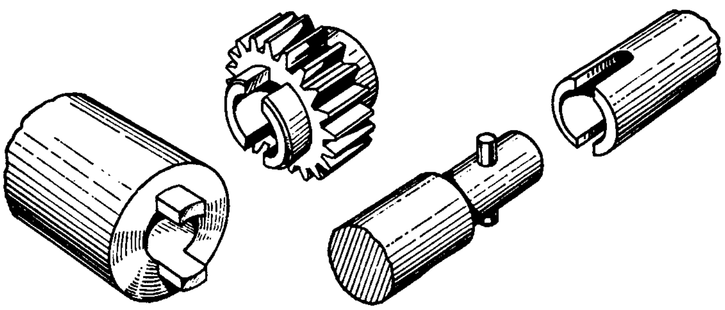


6. SPECIAL INVOLUTE splines are made by using gear tooth proportions. With full depth teeth, greater contact area is possible. A compound pinion is shown made by cropping the smaller pinion teeth and internally splining the larger pinion.

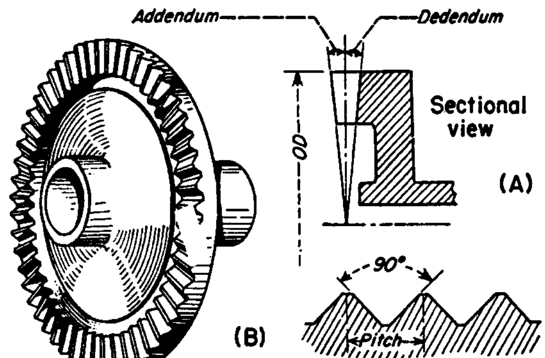


7. TAPER-ROOT splines are for drivers that require positive positioning. This method holds mating parts securely. With a 30º involute stub tooth, this type is stronger than parallel root splines and can be hobbed with a range of tapers.

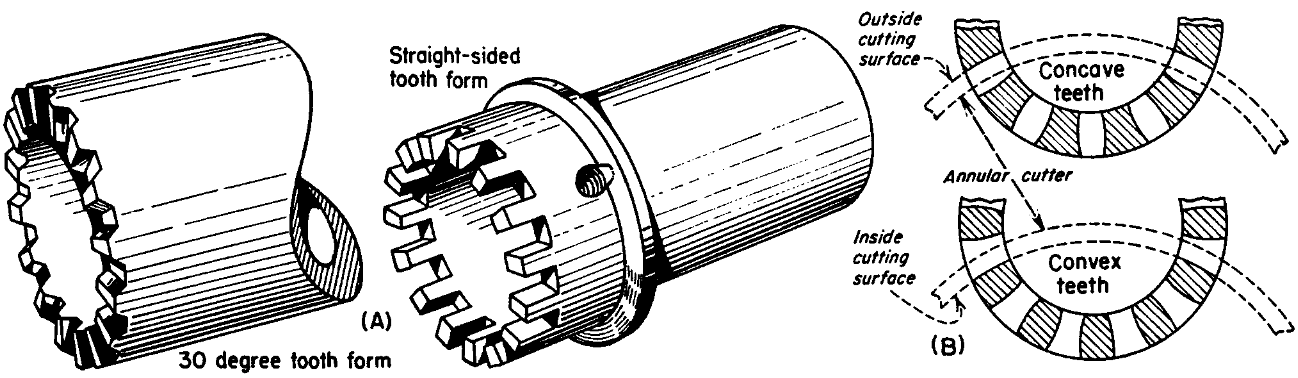
FACE SPLINES



8. MILLED SLOTS in hubs or shafts make inexpensive connections. This spline is limited to moderate loads and requires a locking device to maintain positive engagement. A pin and sleeve method is used for light torques and where accurate positioning is not required.



9. RADIAL SERRATIONS made by milling or shaping the teeth form simple connections. (A) Tooth proportions decrease radially. (B) Teeth can be straight-sided (castellated) or inclined; a 90º angle is common.



10. CURVIC COUPLING teeth are machined by a face-mill cutter. When hardened parts are used that require accurate positioning, the teeth can be ground. (A) This process produces teeth with uniform depth. They can be cut at any pressure angle, although 30º is most common. (B) Due to the cutting action, the shape of the teeth will be concave (hour-glass) on one member and convex on the other—the member with which it will be assembled.

FOURTEEN WAYS TO FASTEN HUBS TO SHAFTS

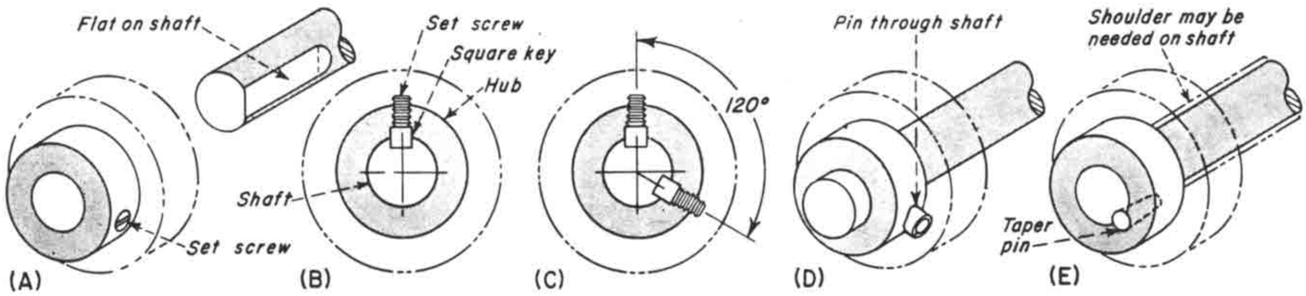


Fig. 1 A cup-point setscrew in hub (A) bears against a flat on a shaft. This fastening is suitable for fractional horsepower drives with low shock loads but is unsuitable when frequent removal and assembly are necessary. The key with setscrew (B) prevents shaft marring from frequent removal and assembly. It can withstand high shock loads. Two keys 120° apart (C) transmit extra heavy loads. Straight or tapered pin (D) prevents end play. For experimental setups an expanding pin is suitable yet easy to remove. Taper pin (E) parallel to shaft might require a shoulder on the shaft. It can be used when a gear or pulley has no hub.

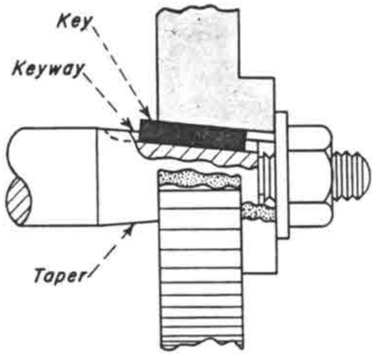


Fig. 2 A tapered shaft with a key and threaded end is a rigid concentric assembly. It is suitable for heavy-duty applications, yet it can be easily disassembled.

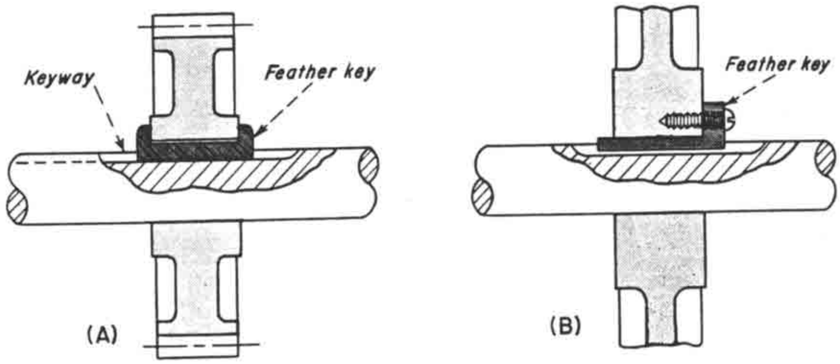


Fig. 3 A feather key (A) allows axial gear movement. A keyway must be milled to the end of the shaft. For a blind keyway (B) the hub and key must be drilled and tapped, but the design allows the gear to be mounted anywhere on the shaft with only a short keyway.

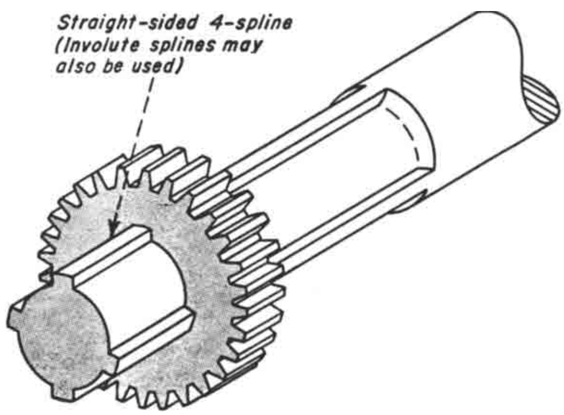


Fig. 4 Splined shafts are frequently used when a gear must slide. Square splines can be ground to close minor diameter gaps, but involute splines are self-centering and stronger. Non-sliding gears can be pinned to the shaft if it is provided with a hub.

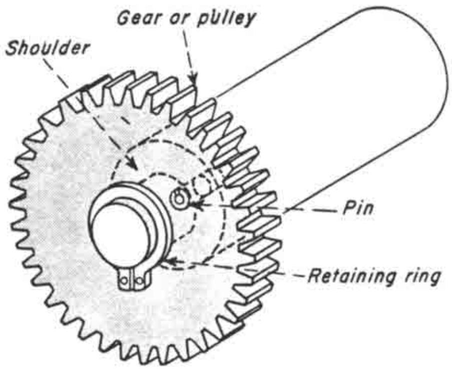


Fig. 5 A retaining ring allows quick gear removal in light-load applications. A shoulder on the shaft is necessary. A shear pin can secure the gear to the shaft if protection against an excessive load is required.

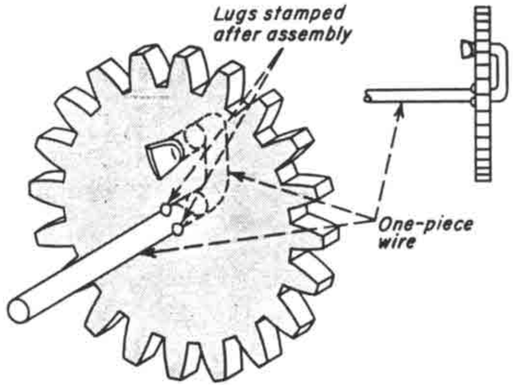


Fig. 6 A stamped gear and formed wire shaft can be used in light-duty application. Lugs stamped on both legs of the wire prevent disassembly. The bend radii of the shaft should be small enough to allow the gear to seat.

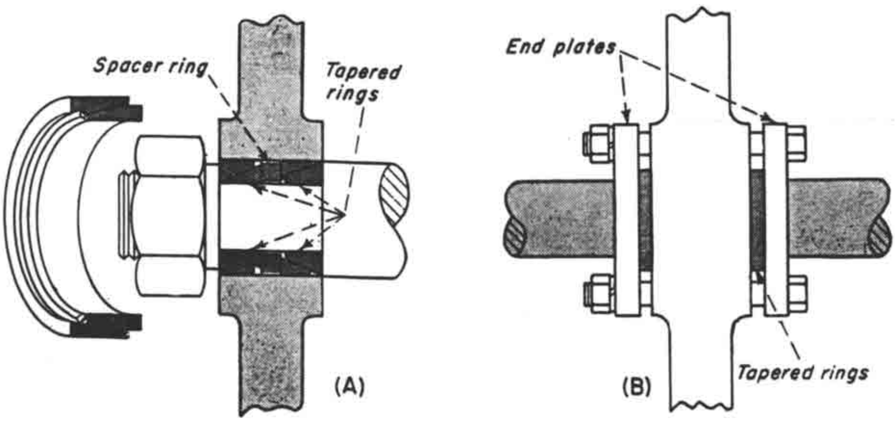


Fig. 7 Interlocking tapered rings hold the hub tightly to the shaft when the nut is tightened. Coarse machining of the hub and shaft does not affect concentricity as in pinned and keyed assemblies. A shoulder is required (A) for end-of-shaft mounting. End plates and four bolts (B) allow the hub to be mounted anywhere on the shaft.

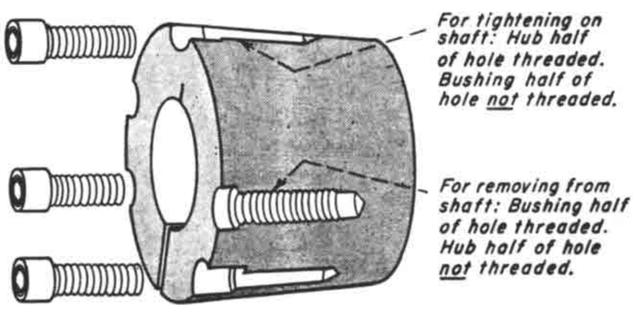


Fig. 8 This split bushing has a tapered outer diameter. Split holes in the bushing align with split holes in the hub. For tightening, the hub half of the hole is tapped, and the bushing half is un-tapped. A screw pulls the bushing into the hub as it is tightened, and it is removed by reversing the procedure.

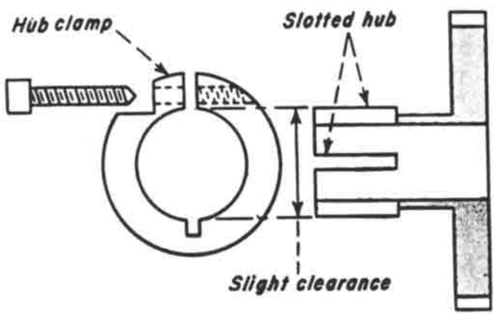


Fig. 9 The split hub of a stock precision gear is clamped onto a shaft with a separate hub clamp. Manufacturers list correctly dimensioned hubs and clamps so that they can be efficiently fastened to a precision-ground shaft.

CLAMPING DEVICES FOR ACCURATELY ALIGNING ADJUSTABLE PARTS

Methods for clamping parts that must be readily movable are as numerous and as varied as the requirements. In many instances, a clamp of any design is satisfactory, provided it has sufficient strength to hold the parts immovable when tightened. However, it is sometimes necessary that the movable part be clamped to maintain accurate alignment with some fixed part. Examples of these clamps are described and illustrated.

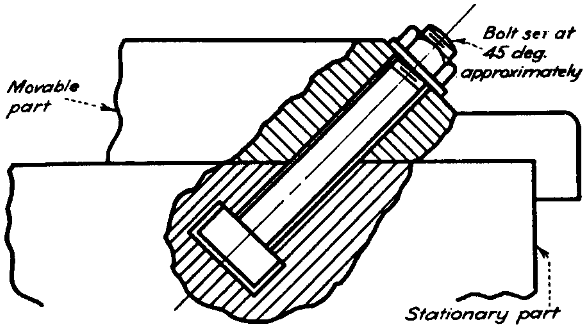


FIG. 1 When a nut is tightened, the flange on the edge of the movable part is drawn against the machined edge of the stationary part. This method is effective, but the removal of the clamped part can be difficult if it is heavy or unbalanced.

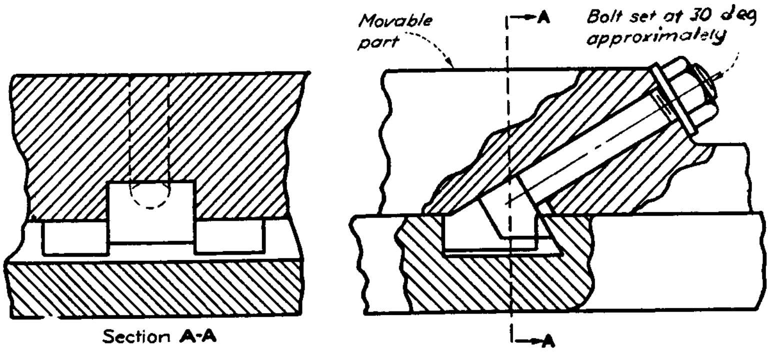


FIG. 2 The lower edged of the bolt head contacts the angular side of locating groove, causing the keys to be held tightly against the opposite side of the groove. This design permits easy removal of the clamped part, but it is effective only if the working pressure is directly downward or in a direction against the perpendicular side of the slot.

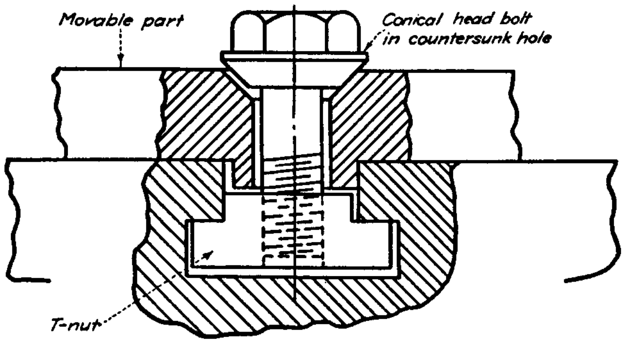


FIG. 3 The movable part is held against one side of the groove while the T-nut is forced against the other side. The removal of the screw permits easy removal of the clamped part. Heavy pressure toward the side of the key out of contact with the slot can permit slight movement due to the springing of the screw.

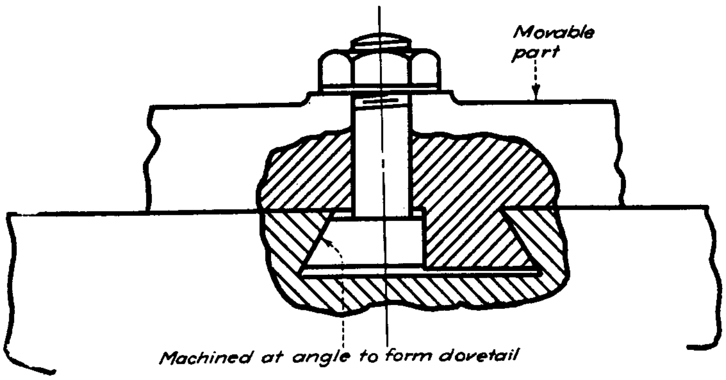


FIG. 4 One side of the bolt is machined at an angle to form a side of the dovetail, which tightens in the groove as the nut is drawn tight. The part must be slid the entire length of the slot for removal.

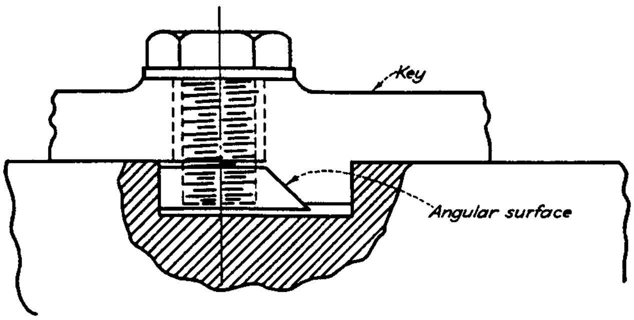


FIG. 5 The angular surface of the nut contacts the angular side of the key, and causes it to move outward against the side of the groove. This exerts a downward pull on the clamped part due to the friction of the nut against the side of a groove as the nut is drawn upward by the screw.

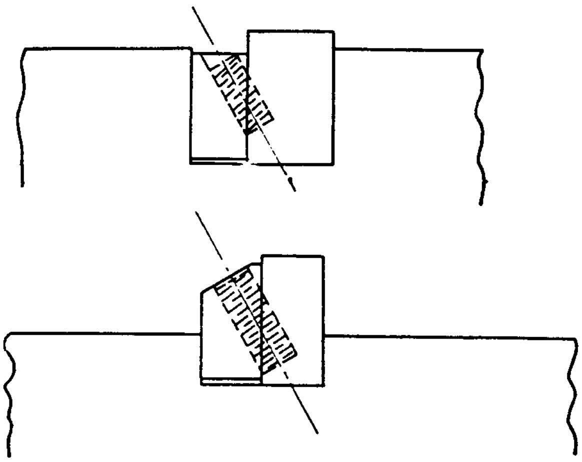


FIG. 6 and 7 These designs differ only in the depth of the grooves. They cannot withstand heavy pressure in an upward direction but have the advantage of being applicable to narrow grooves.

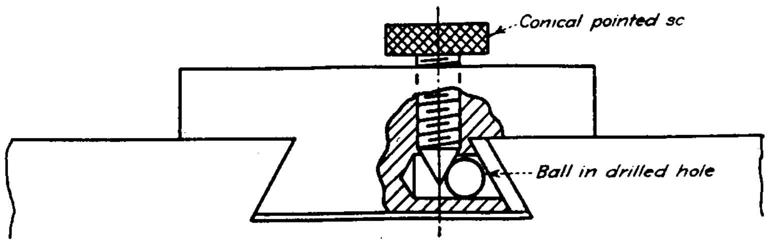


FIG. 8 Screw contact causes the ball to exert an outward pressure against the gib. The gib is loosely pinned to the movable part. This slide can be applied to broad surfaces where it would be impractical to apply adjusting screws through the stationary part.

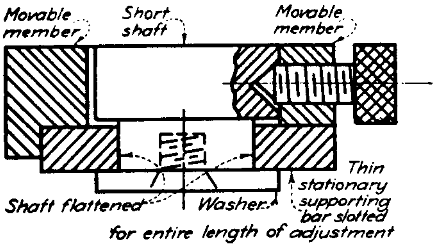


FIG. 9 The movable member is flanged on one side and carries a conical pointed screw on the other side. A short shaft passes through both members and carries a detent slightly out of alignment with the point of the screw. This shaft is flattened on opposite sides where it passes through the stationary member to prevent it from turning when the movable member is removed. A heavy washer is screwed to the under side of the shaft. When the knurled screw is turned inward, the shaft is drawn upward while the movable member is drawn downward and backward against the flange. The shaft is forced forward against the edge of the slot. The upper member can thus be moved and locked in any position. Withdrawing the point of the screw from the detent in the shaft permits the removal of the upper member.

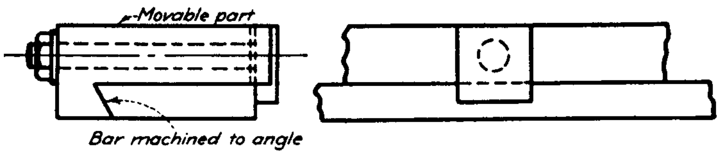


FIG. 10 One edge of a bar is machined at an angle which fits into mating surfaces on the movable part. When the bolt, which passes through the movable part, is drawn tight, the two parts are clamped firmly together.

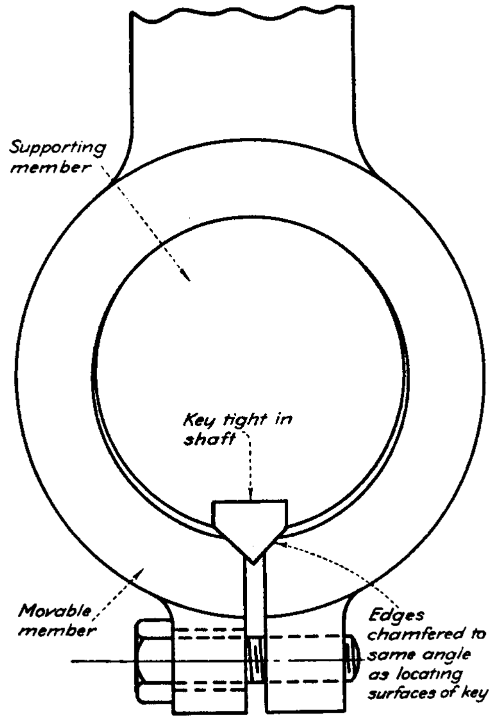
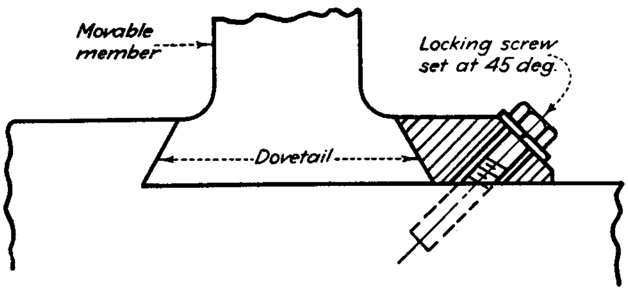


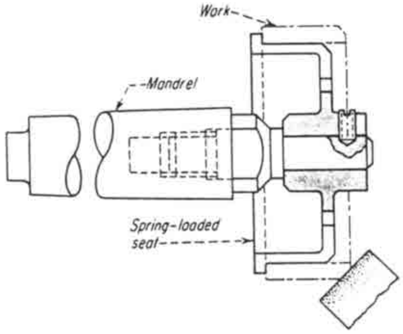
FIG. 11 As the screw is tightened, the chamfered edges of the cut tend to ride outward on the angular surfaces of the key. This draws the movable member tightly against the opposite side of the shaft.

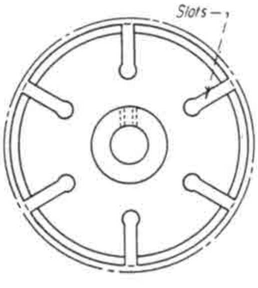
FIG. 12 As the screw is turned, it causes the movable side, which forms one side of the dovetail groove, to move until it clamps tightly on the movable member. The movable side should be as narrow as possible, because there is a tendency for this part to ride up on the angular surface of the clamped part.



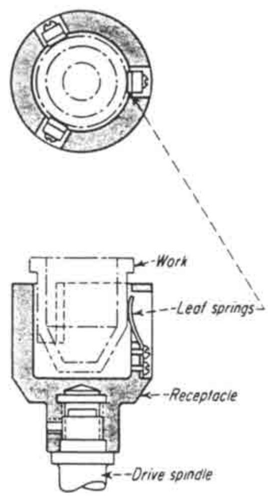
SPRING-LOADED CHUCKS AND HOLDING FIXTURES

Spring-loaded fixtures for holding work can be preferable to other fixtures. Their advantages are shorter setup time and quick workpiece change. Work distortion is reduced because the spring force can be easily and accurately adjusted.

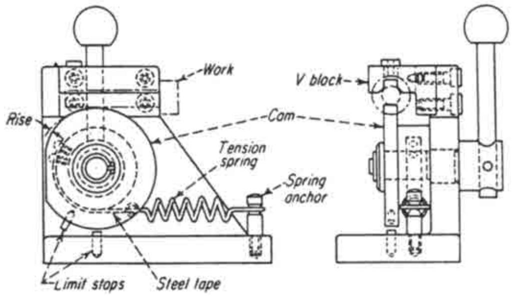




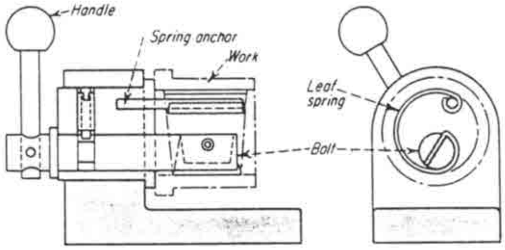
A spring-loaded nest has radial slots extending into its face. These ensure an even grip on the work, which is pushed over the rim. A slight lead on the rim makes mounting work easier. The principal application of this fixture is for ball-bearing race grinding where only light cutting forces are applied.



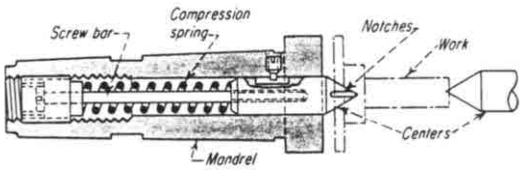
A cupped fixture has three leaf-springs equally spaced in a wall. The work, usually to be lacquered, is inserted into the cup during its rotation. Because the work is placed in the fixture by hand, the spindle is usually friction-driven for safety.



This spring clamp has a cam-and-tension spring that applies a clamping force. A tension spring activates the cam through a steel band. When the handle is released, the cam clamps the work against the V-bar. Two stop-pins limit travel when there is no work in thefixture.



A leaf-spring gripper is used mainly to hold work during assembly. One end of a flat coil-spring is anchored in the housing; the other end is held in a bolt. When the bolt is turned, the spring is tightened, and its outside diameter is decreased. After the work is slid over the spring, the bolt handle is released. The spring then presses against the work, holding it tight.

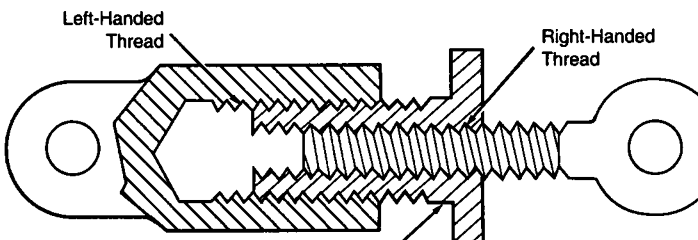


This lathe center is spring loaded and holds the work with spring pressure alone. Eight sharp-edged notches on the conical surface of the driving center bite into the work and drive it. Its spring tension is adjustable.

SHORT IN-LINE TURNBUCKLE

A short body is achieved without offset.

NASA's Jet Propulsion Laboratory, Pasadena, California



The three parts of the turnbuckle would fit together in a threaded, telescoping fashion.

A proposed turnbuckle would be shorter than conventional turnbuckles and could, therefore, fit in shorter spaces. Its ends would be coaxial. The design is unlike that of other short turnbuckles whose ends and the axes that pass through them are laterally offset.

The turnbuckle would consist of the following parts (see figure):

- An eye on a shank with internal left-handed threads,
- An eye on a shank with external right-handed threads, and
- A flanged collar with left-handed external threads to mate with the shank of the first-mentioned eye, and right-handed internal threads to mate with the shank of the second-mentioned eye. The flange would be knurled or hexagonal so that it could be turned by hand or wrench to adjust the overall length of the turnbuckle.

For fine adjustments of length, the collar could be made with only right-handed threads and different pitches inside and out. (Of course, the threads on the mating shanks of the eyes would be made to match the threads on the collar.) For example, with a right-handed external thread of 28 per in. (pitch ≈ 0.91 mm) and a right-handed internal thread of 32 per in. (pitch ≈ 0.79 mm) , one turn of the collar would change the length approximately 0.0045 in. (about 0.11 mm).

This work was done by Earl Collins and Malcolm MacMartin of Caltech for **NASA's Jet Propulsion Laboratory.**

ACTUATOR EXERTS TENSILE OR COMPRESSIVE AXIAL LOAD

A shearpin limits the load.
Marshall Space Flight Center, Alabama

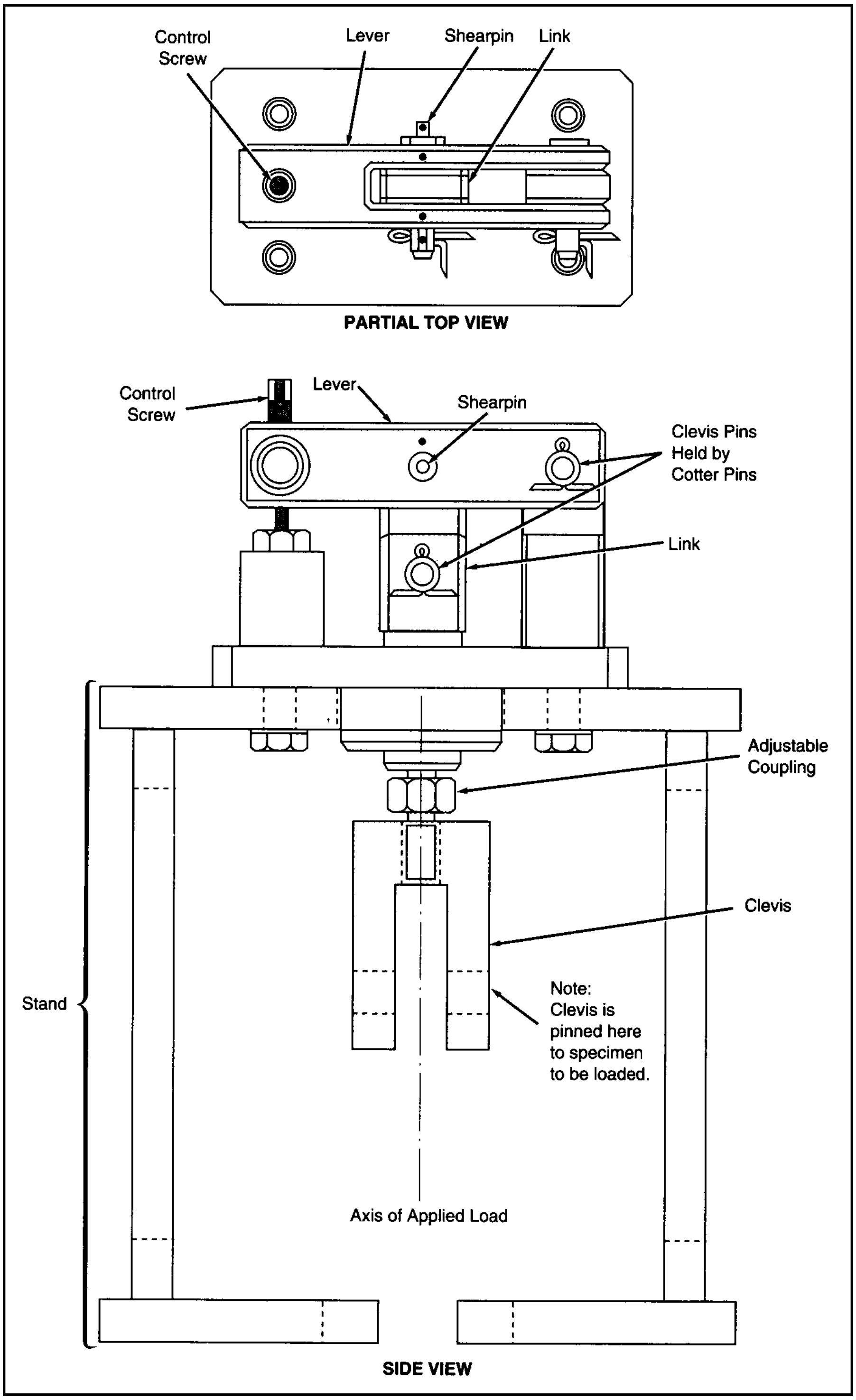
A compact, manually operated mechanical actuator applies a controlled, limited tensile or compressive axial force. The actuator is designed to apply loads to bearings during wear tests in a clean room. It is intended to replace a hydraulic actuator that is bulky and difficult to use, requires periodic maintenance, and poses the threat of leakage of hydraulic fluid, which can contaminate the clean room.

The actuator rests on a stand and imparts axial force to a part attached to a clevis inside or below the stand (see figure). A technician turns a control screw at one end of a lever. Depending on the direction of rotation of the control screw, its end of the lever is driven downward (for compression) or upward (for tension). The lever pivots about a clevis pin at the end opposite that of the control screw; this motion drives downward or upward a link attached through a shearpin at the middle of the lever. The link drives a coupling and, through it, the clevis attached to the part to be loaded.

The control screw has a fine thread so that a large adjustment of the screw produces a relatively small change in the applied force. With the help of a load cell that measures the applied load, the technician can control the load to within ±10 lb (45 N). An estimated input torque of only 40 to 50 lb·in. (4.5 to 5.6 N·m) is needed to apply the maximum allowable load of 2,550 lb (11.34 kN).

The shearpin at the middle of the lever breaks if a force greater than 2,800 ± 200 lb (12.45 ± 0.89 kN) is applied in tension or compressed, thus protecting the stressed part from overload. The shearpin is made of a maraging steel, selected because it fails more predictably and cleanly in shear than pins made from other alloys. Moreover, it is strong when machined to small pin diameters. Batches of pins are made from the same raw stock to ensure that all fail at or near the same load.

This work was done by John Nozzi and Cuyler H. Richards of Rockwell International Corp. for **Marshall Space Flight Center.**



This mechanical actuator applies an axial load to a test specimen inside or under its stand.

GRIPPING SYSTEM FOR MECHANICAL TESTING OF COMPOSITES

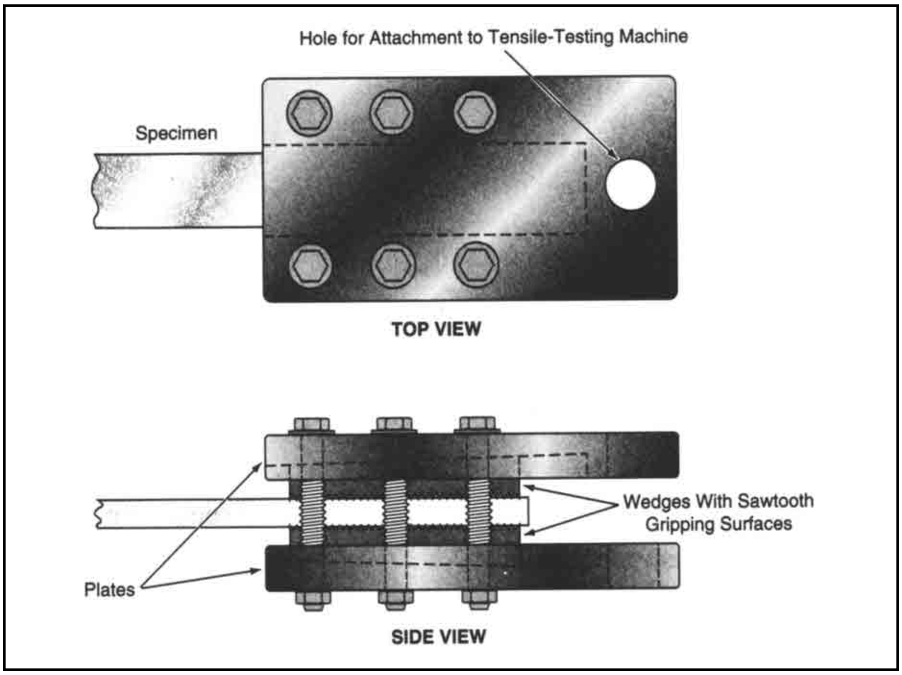
Specimens can be held without slippage, even at high temperatures.
Lewis Research Center, Cleveland, Ohio

An improved gripping system has been designed to hold the ends of a specimen of a composite material securely during a creep or tensile test. The grips function over a wide range of applied stress [3 to 100 kpsi (about 21 to about 690 Mpa)] and temperature [up to 1,800°F (about 980°C)].

Each grip includes a pair of wedges that have sharply corrugated (sawtooth-profile) gripping surfaces. The wedges are held between two plates that contain cavities which are sloped to accommodate the wedges (see figure). Two such grips—one for each end of the specimen—hold a specimen in a furnace which is connected to a tensile test machine for creep measurements.

In preparation for a test, the specimen is assembled with the grips in a fixture that maintains all parts in precise alignment: this step is necessary to ensure that the load applied during the test will coincide with the axis of the specimen. Unlike some older wedge grips, the specimen can be gripped in a delicate manner during assembly and alignment. While the assembled parts are still in the alignment fixture, hexagonal nuts and bolts on the grip can be tightened evenly with a torque wrench to 120 lb-in. (≈ 13.6 N-m).

During a test, the grips apply the required tensile stress to the specimen without slippage at high temperatures and, therefore, without loss of alignment.



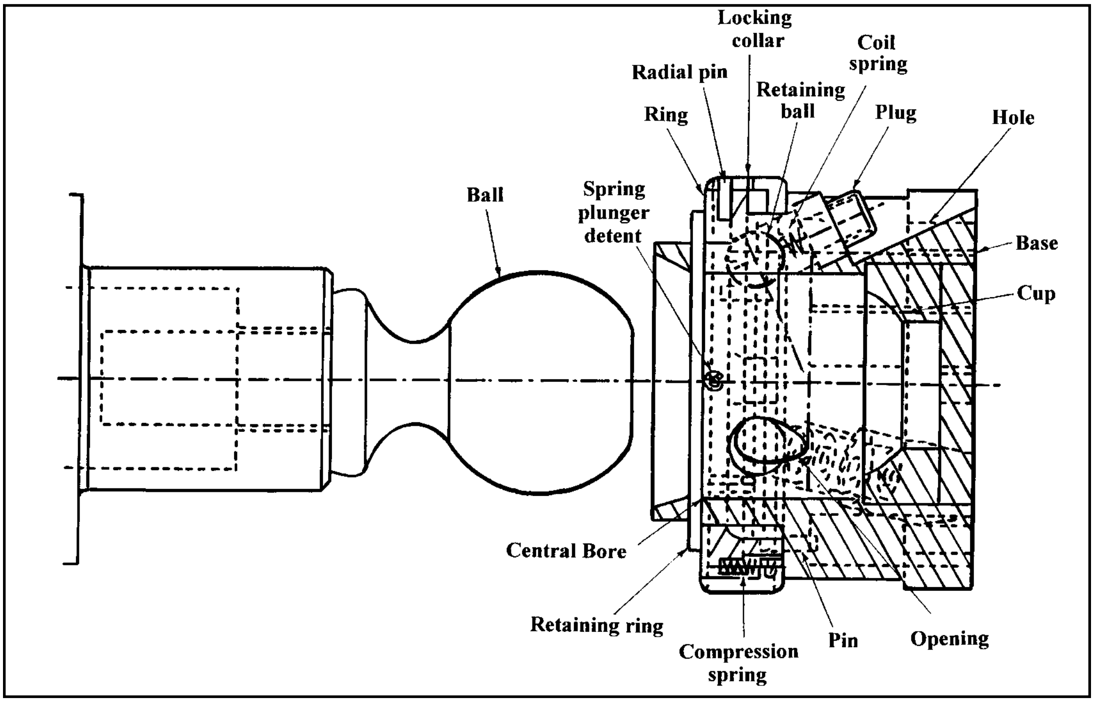
A pair of sawtooth wedges clamped between a pair of plates holds one end of a specimen. A mirror image of this grip is attached at the other end of the specimen. An alignment fixture (not shown) holds the grips and specimen during assembly.

In contrast, some older plate grips tended to sip at high temperatures when applied tensile stresses rose above 20 kpsi (≈ 140 Mpa), while older hydraulically actuated grips could not be allowed inside the testing furnaces, which introduced temperature gradients in the specimens.

This work was done by Rebecca A. MacKay and Michael V. Nathal of **Lewis Research Center.**

PASSIVE CAPTURE JOINT WITH THREE DEGREES OF FREEDOM

New joint allows quick connection between any two structural elements where rotation in all three axis is desired.
Marshall Space Flight Center, Alabama



The three-degrees-of-freedom capability of the **Passive Capture Joint** provides for quick connect and disconnect operations.

A new joint, proposed for use on an attachable debris shield for the International Space Station Service Module, has potential for commercial use in situations where hardware must be assembled and disassembled on a regular basis.

This joint can be useful in a variety of applications, including replacing the joints commonly used on trailer-hitch tongues and temporary structures, such as crane booms and rigging. Other uses for this joint include assembly of structures where simple rapid deployment is essential, such as in space, undersea, and in military structures.

This new joint allows for quick connection between any two structural elements where it is desirable to have rotation in all three axes. The joint can be fastened by moving the two halves into position. The joint is then connected by inserting the ball into the bore of the base. When the joint ball is fully inserted, the joint will lock with full strength. Release of this joint involves only a simple movement and rotation of one part. The joint can then be easily separated.

Most passive capture devices allow only axial rotation when fastened—if any movement is allowed at all. Manually- or power-actuated active joints require an additional action, or power and control signal, as well as a more complex mechanism.

The design for this new joint is relatively simple. It consists of two halves, a ball mounted on a stem (such as those on a common trailer-hitch ball) and a socket. The socket contains all the moving parts and is the important part of this invention. The socket also has a base, which contains a large central cylindrical bore ending in a spherical cup.

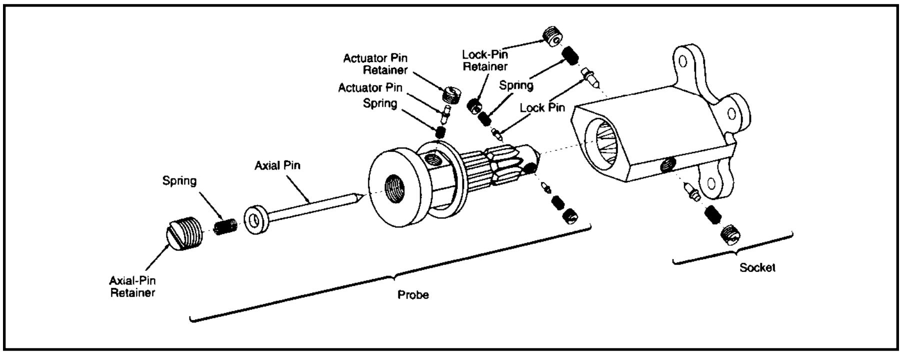
This work was done by Bruce Weddendorf and Richard A. Cloyd of the **Marshall Space Flight Center**.

PROBE-AND-SOCKET FASTENERS FOR ROBOTIC ASSEMBLY

Self-alignment and simplicity of actuation make this fastener amenable to robotic assembly.
Lyndon B. Johnson Space Center, Houston, Texas

A probe-and-socket fastening mechanism was designed to be operated by a robot. The mechanism is intended to enable a robot to set up a workstation in a hostile environment, for example. The workstation can then be used by an astronaut, aquanaut, or other human who could then spend minimum time in the environment. The human can concentrate on performing quality work rather than spending time setting up equipment, with consequent reduction of risk.

The mechanism (see figure) includes (1) a socket, which would be mounted on a structure at the worksite, and (2) a probe, which would be mounted on a piece of equipment to be attached to the structure at the socket. The probe-and-socket fastener is intended for use in conjunction with a fixed target that would aid in the placement of the end effector of the robot during grasping. There would also be a handle or handles on the structure. The robot would move the probe near the socket and depress the actuator pin in the probe. The inward motion of the actuator pin would cause rearward motion of the axial pin, thereby allowing two spring-loaded lockpins to retract into the probe. The robot would then begin to insert the probe into the socket.



Lockpins in the probe engage radial holes (not shown) in the socket. Depressing the actuator pin temporarily retracts the lockpins into the probe so that the probe can be inserted in the socket.

Tapered grooves in the socket mesh with tapered ridges on the probe, thereby aligning the fastener parts and preventing binding. When the probe bottoms out in the socket, the robot releases its grip on the actuator pin. The resulting forward motion of the axial pin pushes the lockpins of the probe outward into mating holes (not shown) in the socket. Also, when the probe bottoms in the socket, additional lockpins in the socket spring into detents located at about the midlength of the tapered ridges on the probe.

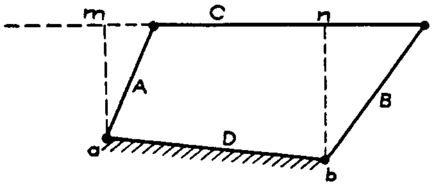
This work was done by Karen Nyberg of **Johnson Space Center.**

CHAPTER 13

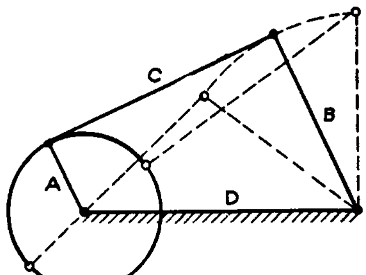
KEY EQUATIONS AND CHARTS FOR DESIGNING MECHANISMS

FOUR-BAR LINKAGES AND TYPICAL INDUSTRIAL APPLICATIONS

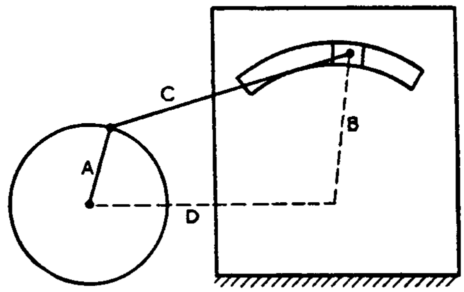
All mechanisms can be broken down into equivalent four-bar linkages. They can be considered to be the basic mechanism and are useful in many mechanical operations.



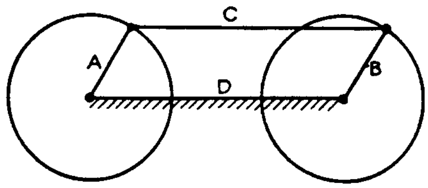
FOUR-BAR LINKAGES—Two cranks, a connecting rod and a line between the fixed centers of the cranks make up the basic four-bar linkage. Cranks can rotate if *A* is smaller than *B* or *C* or *D*. Link motion can be predicted.



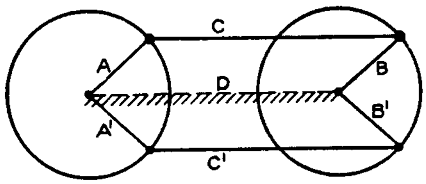
CRANK AND ROCKER—the following relations must hold for its operation: $A + B + C > D$; $A + D + B > C$; $A + C - B < D$, and $C - A + B > D$.



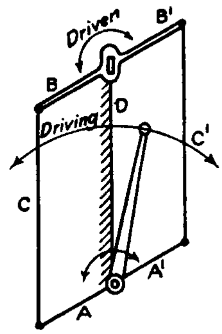
FOUR-BAR LINK WITH SLIDING MEMBER—One crank is replaced by a circular slot with an effective crank distance of *B*.



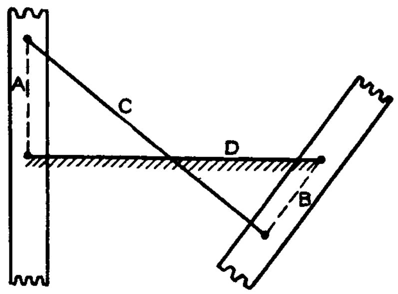
PARALLEL CRANK FOUR-BAR—Both cranks of the parallel crank four-bar linkage always turn at the same angular speed, but they have two positions where the crank cannot be effective.



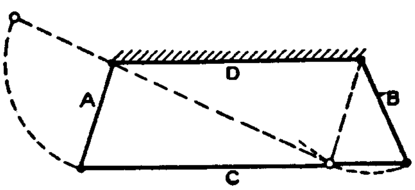
DOUBLE PARALLEL CRANK—This mechanism avoids a dead center position by having two sets of cranks at 90° advancement. The connecting rods are always parallel.



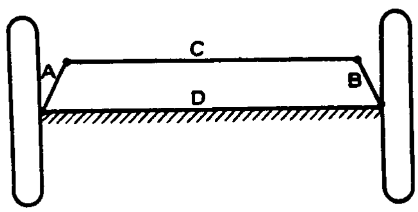
PARALLEL CRANK—Steam control linkage assures equal valve openings.



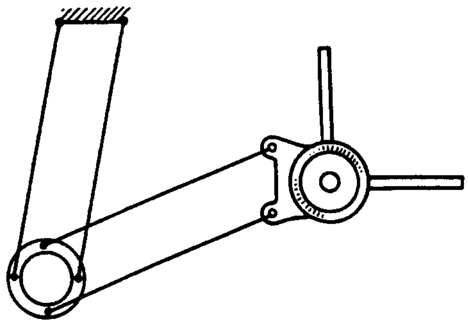
NON-PARALLEL EQUAL CRANK—The centrodes are formed as gears for passing dead center and they can replace ellipticals.



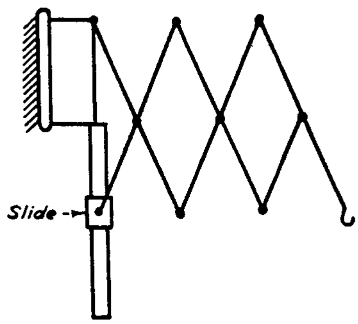
SLOW MOTION LINK—As crank *A* is rotated upward it imparts motion to crank *B*. When *A* reaches its dead center position, the angular velocity of crank *B* decreases to zero.



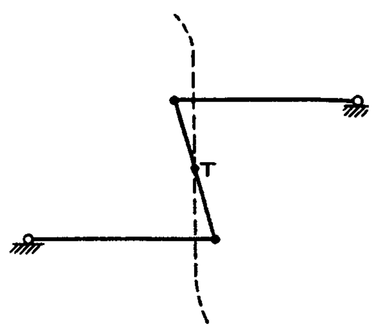
TRAPAZOIDAL LINKAGE—This linkage is not used for complete rotation but can be used for special control. The inside moves through a larger angle than the outside with normals intersecting on the extension of a rear axle in a car.



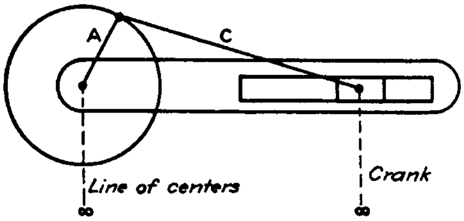
DOUBLE PARALLEL CRANK MECHANISM—This mechanism forms the basis for the universal drafting machine.



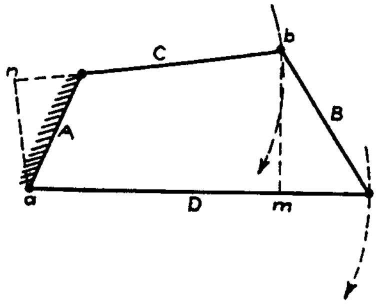
ISOSCELES DRAG LINKS—This "lazy-tong" device is made of several isosceles links; it is used as a movable lamp support.



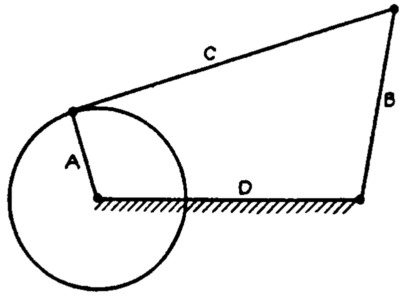
WATT'S STRAIGHT-LINE MECHANISM—Point T describes a line perpendicular to the parallel position of the cranks.



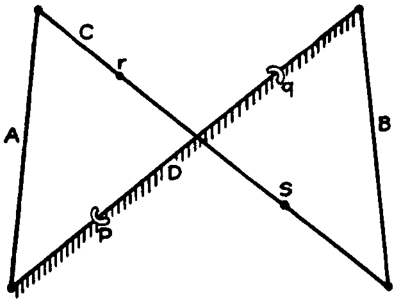
STRAIGHT SLIDING LINK—This is the form in which a slide is usually used to replace a link. The line of centers and the crank *B* are both of infinite length.



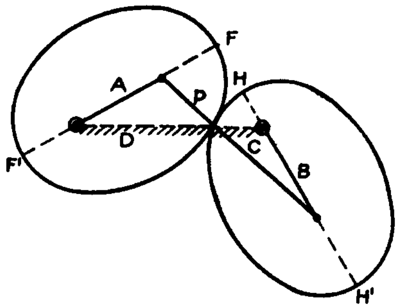
DRAG LINK—This linkage is used as the drive for slotter machines. For complete rotation: $B > A + D - C$ and $B < D + C - A$.



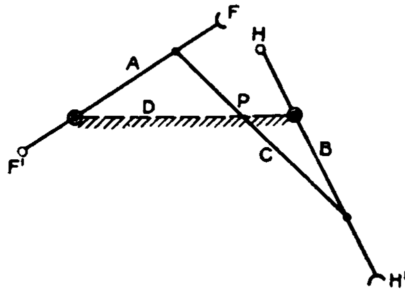
ROTATING CRANK MECHANISM—This linkage is frequently used to change a rotary motion to a swinging movement.



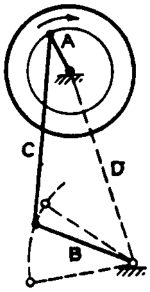
NON-PARALLEL EQUAL CRANK—If crank *A* has a uniform angular speed, *B* will vary.



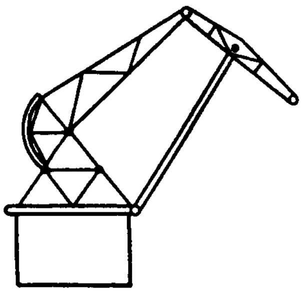
ELLIPTICAL GEARS—They produce the same motion as non-parallel equal cranks.



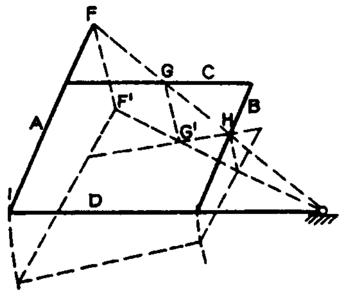
NON-PARALLEL EQUAL CRANK—It is the same as the first example given but with crossover points on its link ends.



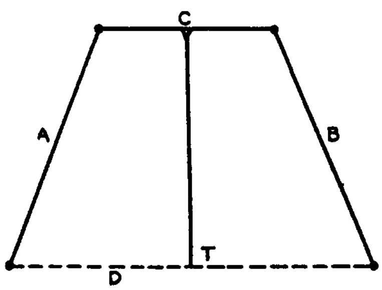
TREADLE DRIVE—This four-bar linkage is used in driving grinding wheels and sewing machines.



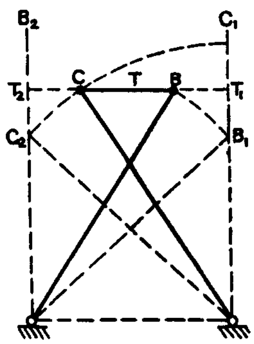
DOUBLE LEVER MECHANISM—This slewing crane can move a load in a horizontal direction by using the D-shaped portion of the top curve.



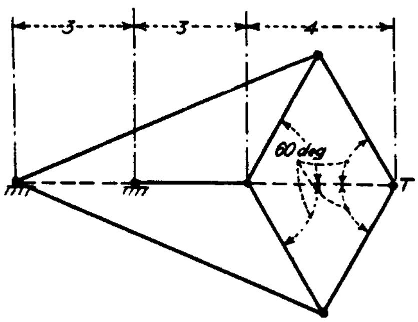
PANTOGRAPH—The pantograph is a parallelogram in which lines through *F*, *G* and *H* must always intersect at a common point.



ROBERT'S STRAIGHT-LINE MECHANISM—The lengths of cranks *A* and *B* should not be less than 0.6 *D*; *C* is one half of *D*.



TCHEBICHEFF'S—Links are made in proportion: $AB = CD = 20$, $AD = 16$, $BC = 8$.



PEAUCELLIER'S CELL—When proportioned as shown, the tracing point *T* forms a straight line perpendicular to the axis.

DESIGNING GEARED FIVE-BAR MECHANISMS

Geared five-bar mechanisms offer excellent force-transmission characteristics and can produce more complex output motions—including dwells—than conventional four-bar mechanisms.

It is often necessary to design a mechanism that will convert uniform input rotational motion into nonuniform output rotation or reciprocation. Mechanisms designed for such purposes are usually based on four-bar linkages. Those linkages produce a sinusoidal output that can be modified to yield a variety of motions.

Four-bar linkages have their limitations, however. Because they cannot produce dwells of useful duration, the designer might have to include a cam when a dwell is desired, and he might have to accept the inherent speed restrictions and vibration associated with cams. A further limitation of four-bar linkages is that only a few kinds have efficient force-transmission capabilities.

One way to increase the variety of output motions of a four-bar linkage, and obtain longer dwells and better force transmissions, is to add a link. The resulting five-bar linkage would become impractical, however, because it would then have only two degrees of freedom and would, consequently, require two inputs to control the output.

Simply constraining two adjacent links would not solve the problem. The five-bar chain would then function effectively only as a four-bar linkage. If, on the other hand, any two nonadjacent links are constrained so as to remove only one degree of freedom, the five-bar chain becomes a functionally useful mechanism.

Gearing provides solution. There are several ways to constrain two nonadjacent links in a five-bar chain. Some possibilities include the use of gears, slot-and-pin joints, or nonlinear band mechanisms. Of these three possibilities, gearing is the most attractive. Some practical gearing systems (Fig. 1) included paired external gears, planet gears revolving within an external ring gear, and planet gears driving slotted cranks.

In one successful system (Fig. 1A) each of the two external gears has a fixed crank that is connected to a crossbar by a rod. The system has been successful in high-speed machines where it transforms rotary motion into high-impact linear motion. The Stirling engine includes a similar system (Fig. 1B).

In a different system (Fig. 1C) a pin on a planet gear traces an epicyclic, three-lobe curve to drive an output crank back and forth with a long dwell at the

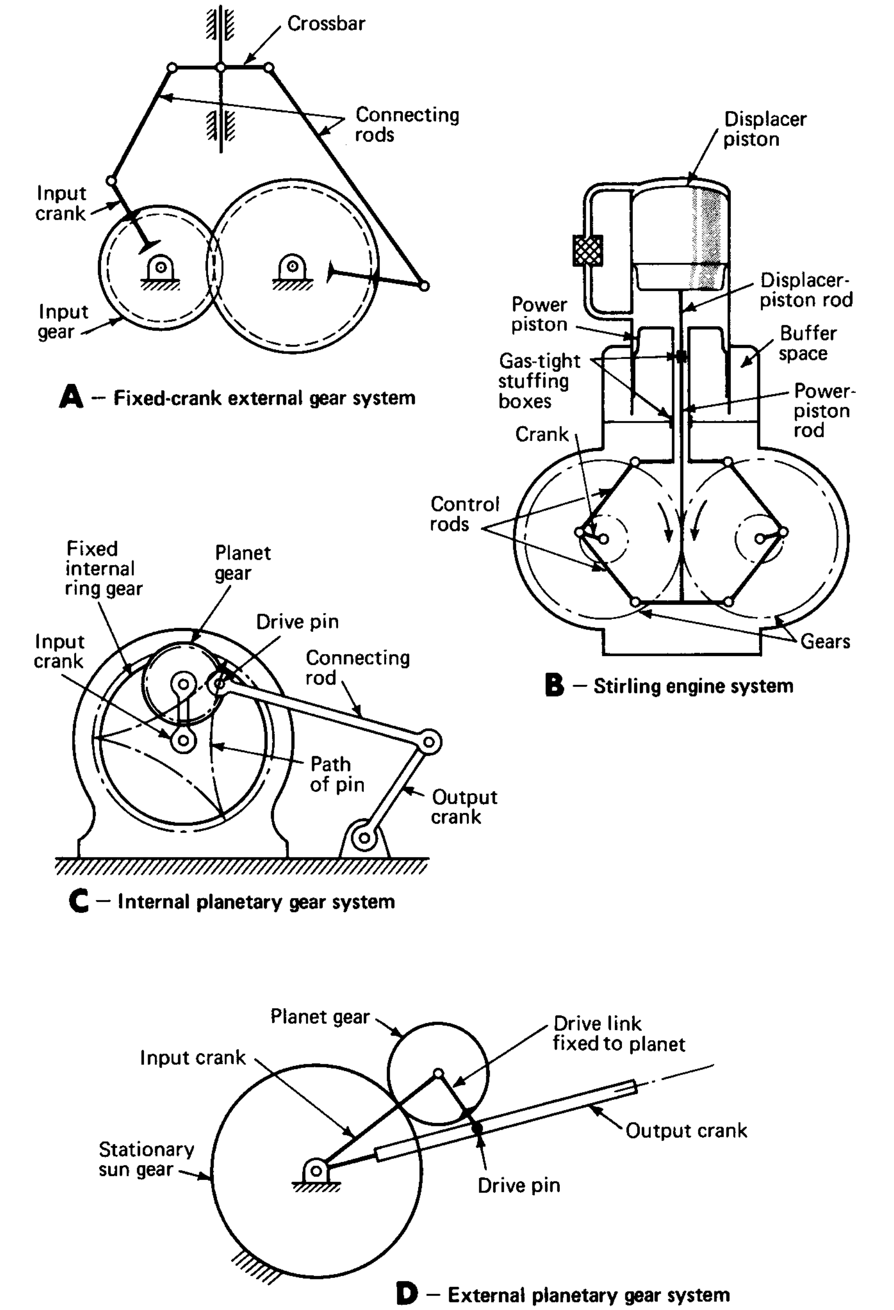


Fig. 1 Five-bar mechanism designs can be based on paired external gears or planetary gears. They convert simple input motions into complex outputs.

extreme right-hand position. A slotted output crank (Fig. 1D) will provide a similar output.

Two professors of mechanical engineering, Daniel H. Suchora of Youngstown State University, Youngstown, Ohio, and Michael Savage of the University of Akron, Akron, Ohio, studied a variation of this mechanism in detail.

Five kinematic inversions of this form (Fig. 2) were established by the two researchers. As an aid in distinguishing between the five, each type is named according to the link which acts as the fixed link. The study showed that the Type 5 mechanism would have the greatest practical value.

In the Type 5 mechanism (Fig. 3A), the gear that is stationary acts as a sun gear. The input shaft at Point E drives the input crank which, in turn, causes the planet gear to revolve around the sun gear. Link a_2, fixed to the planet, then drives the output crank, Link a_4, by means of the connecting link, Link a_3. At any input position, the third and fourth links can be assembled in either of two distinct positions or "phases" (Fig. 3B).

Variety of outputs. The different kinds of output motions that can be obtained from a Type 5 mechanism are based on the different epicyclic curves traced by link joint B. The variables that control the shape of a "B-curve" are the gear ratio GR ($GR = N_2/N_5$), the link ratio a_2/a_1 and the initial position of the gear set, defined by the initial positions of θ_1 and θ_2, designated as θ_{10} and θ_{20}, respectively.

Typical B-curve shapes (Fig. 4) include ovals, cusps, and loops. When the B-curve is oval (Fig. 4B) or semioval (Fig. 4C), the resulting B-curve is similar to the true-circle B-curve produced by a four-bar linkage. The resulting output motion of Link a_4 will be a sinusoidal type of oscillation, similar to that produced by a four-bar linkage.

When the B-curve is cusped (Fig. 4A), dwells are obtained. When the B-curve is looped (Figs. 4D and 4E), a double oscillation is obtained.

In the case of the cusped B-curve (Fig. 4A), dwells are obtained. When the B-curve is looped (Figs. 4D and 4E), a double oscillation is obtained.

In the case of the cusped B-curve (Fig. 4A), by selecting a_2 to be equal to the pitch radius of the planet gear r_2, link joint B becomes located at the pitch circle of the planet gear. The gear ratio in all the cases illustrated is unity ($GR = 1$).

Professors Suchora and Savage analyzed the different output motions produced by the geared five-bar mechanisms by plotting the angular position θ_4 of the output link a_4 of the output link a_4 against the angular position of the input link θ_1 for a variety of mechanism configurations (Fig. 5).

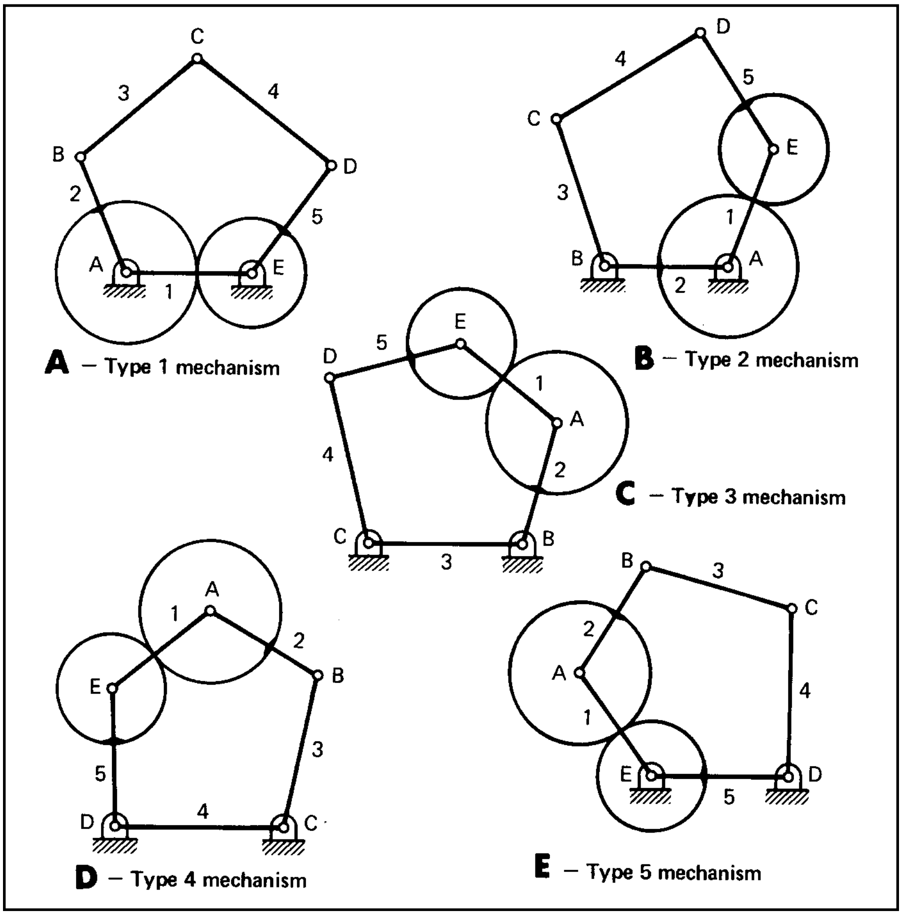


Fig. 2 Five types of geared five-bar mechanisms. A different link acts as the fixed link in each example. Type 5 might be the most useful for machine design.

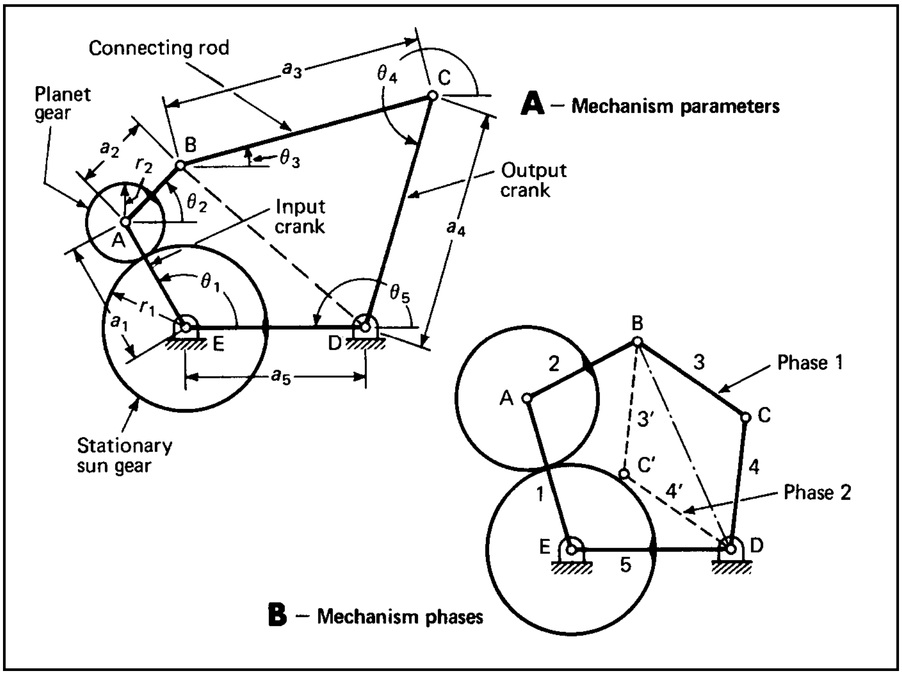


Fig. 3 A detailed design of a Type-5 mechanism. The input crank causes the planet gear to revolve around the sun gear, which is always stationary.

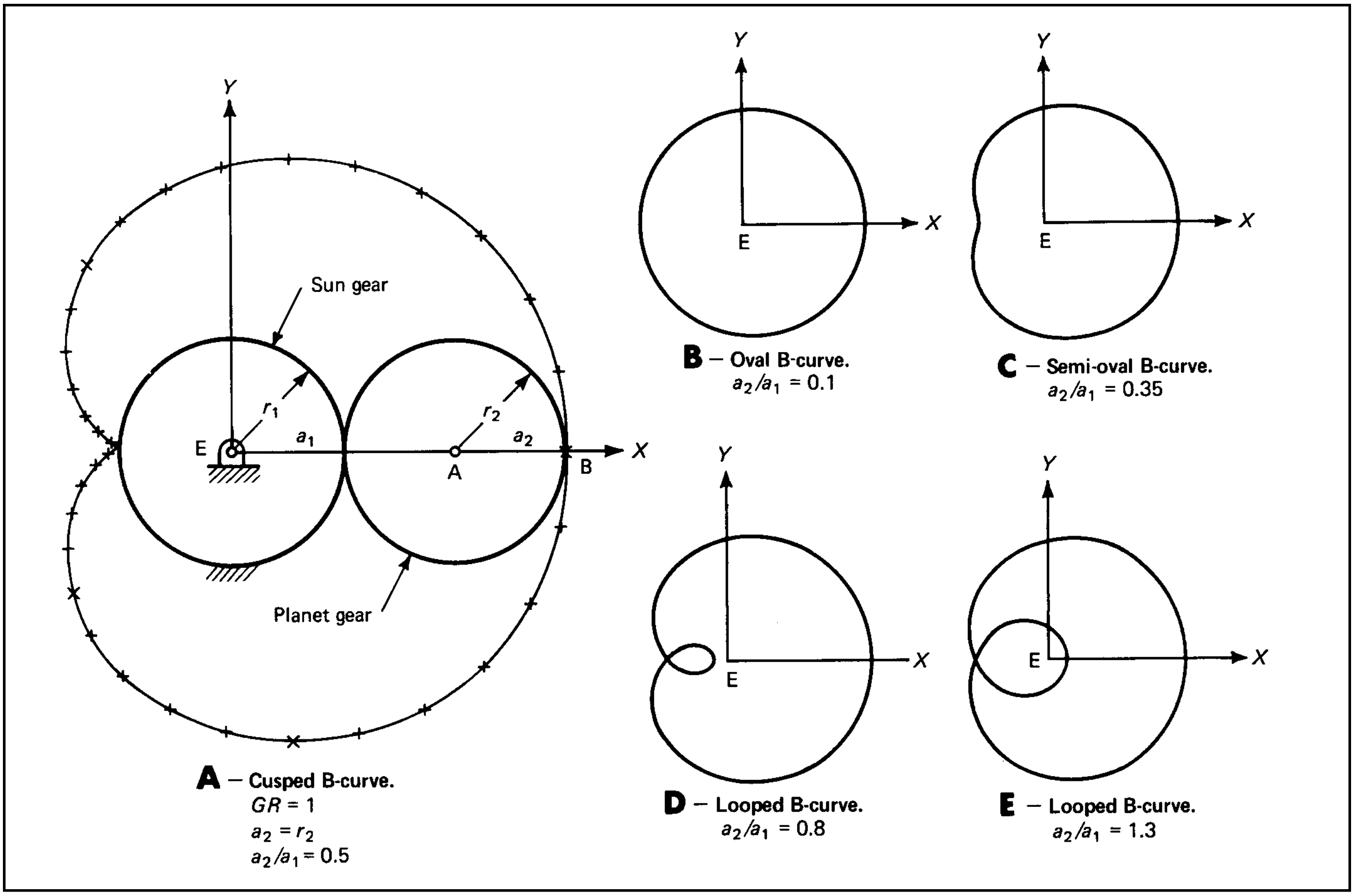


Fig. 4 Typical B-curve shapes obtained from various Type-5 geared five-bar mechanisms. The shape of the epicyclic curved is changed by the link ratio a_2/a_1 and other parameters, as described in the text.

Calculating displacement, velocity and acceleration

Displacement θ_4 can be found from the following equation:

$$\theta_4 = 2\tan^{-1}\left(\frac{I \pm \sqrt{I^2 + H^2 - J^2}}{H + J}\right)$$

where $H = a_1 \cos\theta_1 + a_2 \cos\theta_2 - a_5$; $I = a_1 \sin\theta_1 + a_2 \sin\theta_2$; and $J = 1/2a_4\ (a^2_3 - a^2_4 - H^2 - I^2)$; and where $\theta_2 = \theta_{20} + (1 + 1/GR)\ (\theta_1 - \theta_{10})$, where θ_{10} and θ_{20} are the initial values of the angles θ_1 and θ_2, respectively.

For layout purposes, once θ_4 is determined, θ_3 can be found from:

$$\theta_3 = \tan^{-1}\left(\frac{a_4 \sin\theta_4 + I}{a_4 \cos\theta_4 + H}\right)$$

To find velocities θ'_4 and θ'_3, use these equations:

$$\theta'_4 = \frac{a_1 \sin(\theta_3 - \theta_1) + a_2 \sin(\theta_3 - \theta_2)\ \theta'_2}{a_4 \sin(\theta_4 - \theta_3)}$$

$$\theta'_3 = \frac{a_1 \sin(\theta_1 - \theta_4) + a_2 \sin(\theta_2 - \theta_4)\ \theta'_2}{a_4 \sin(\theta_4 - \theta_3)}$$

where $\theta'_2 = (1 + 1/GR)$.

Use these equations to determine accelerations θ''_4 and θ''_3:

$$\theta''_4 = \frac{L}{a_3\ a_4 \sin(\theta_4 - \theta_3)}$$

$$\theta''_3 = \frac{K}{a_3\ a_4 \sin(\theta_4 - \theta_3)}$$

where $K = a_3a_4\cos(\theta_3 - \theta_4)\,\theta'^2_3 + a_4^2\theta'^2_4 + a_1a_2\cos(\theta_1 - \theta_4) + a_2a_4\cos(\theta_2 - \theta_4)\theta'^2_2$ and $L = a_3^2\theta'^2_3 - a_3a_1\cos(\theta_3 - \theta_4)\theta'^2_4 - a_1a_3\cos(\theta_3 - \theta_1) + a_2a_3\cos(\theta_3 - \theta_2)\theta'^2_2$.

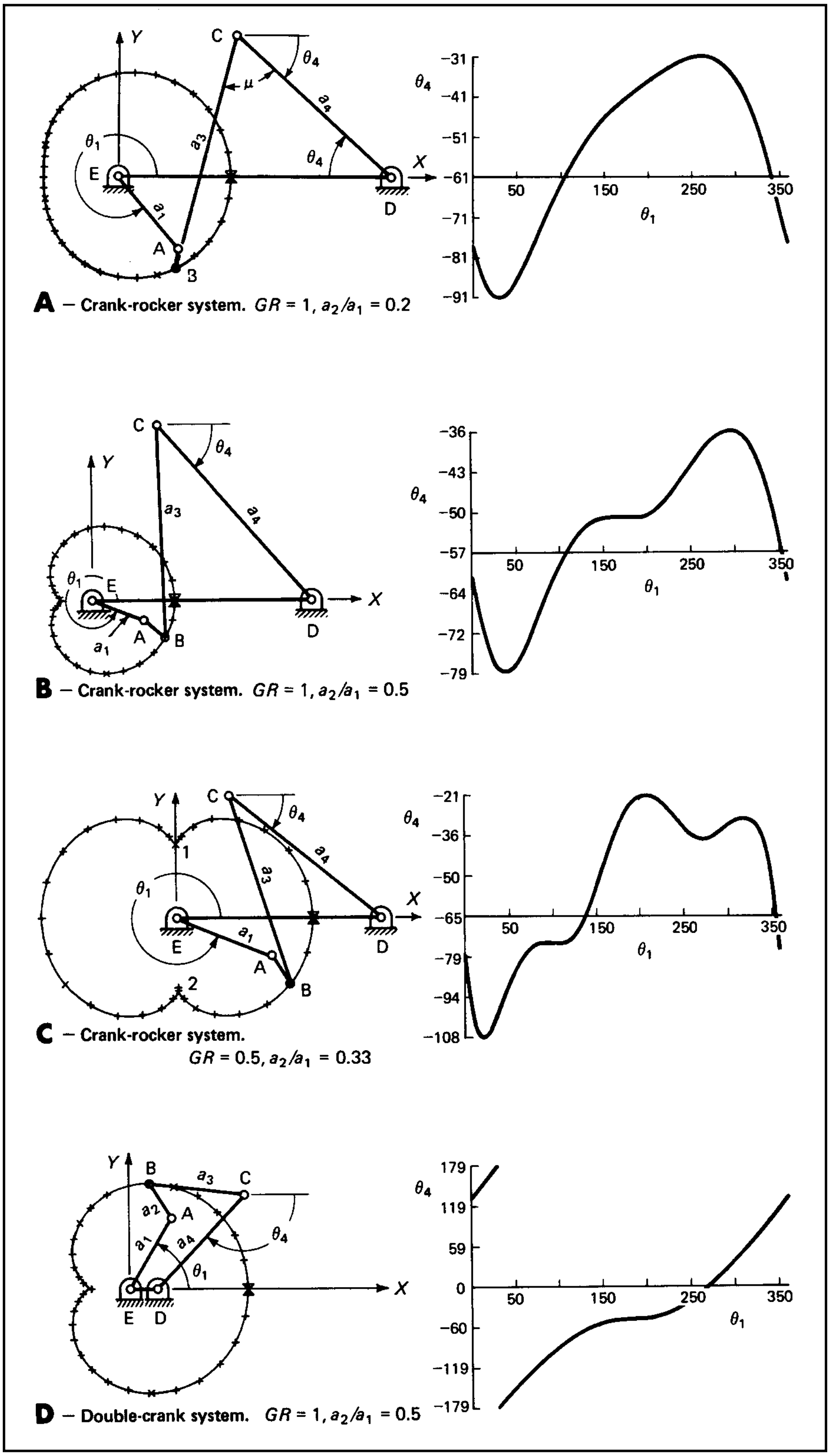


Fig. 5 A variety of output motions can be produced by varying the design of five-bar geared mechanisms. Dwells are obtainable with proper design. Force transmission is excellent. In these diagrams, the angular position of the output link is plotted against the angular position of the input link for various five-bar mechanism designs.

In three of the four cases illustrated, $GR = 1$, although the gear pairs are not shown. Thus, one input rotation generates the entire path of the B-curve. Each mechanism configuration produces a different output.

One configuration (Fig. 5A) produces an approximately sinusoidal reciprocating output motion that typically has better force-transmission capabilities than equivalent four-bar outputs. The transmission angle μ should be within 45 to 135° during the entire rotation for best results.

Another configuration (Fig. 5B) produces a horizontal or almost-horizontal portion of the output curve. The output link, link, a_4, is virtually stationary during this period of input rotation—from about 150 to 200° of input rotation θ_1 in the case illustrated. Dwells of longer duration can be designed.

By changing the gear ratio to 0.5 (Fig. 5C), a complex motion is obtained; two intermediate dwells occur at cusps 1 and 2 in the path of the B-curve. One dwell, from $\theta_1 = 80$ to 110°, is of good quality. The dwell from 240 to 330° is actually a small oscillation.

Dwell quality is affected by the location of Point D with respect to the cusp, and by the lengths of links a_3 and a_4. It is possible to design this form of mechanism so it will produce two usable dwells per rotation of input.

In a double-crank version of the geared five-bar mechanism (Fig. 5D), the output link makes full rotations. The output motion is approximately linear, with a usable intermediate dwell caused by the cusp in the path of the B-curve.

From this discussion, it's apparent that the Type 5 geared mechanism with $GR = 1$ offers many useful motions for machine designers. Professors Suchora and Savage have derived the necessary displacement, velocity, and acceleration equations (see the "Calculating displacement, velocity, and acceleration" box).

KINEMATICS OF INTERMITTENT MECHANISMS—THE EXTERNAL GENEVA WHEEL

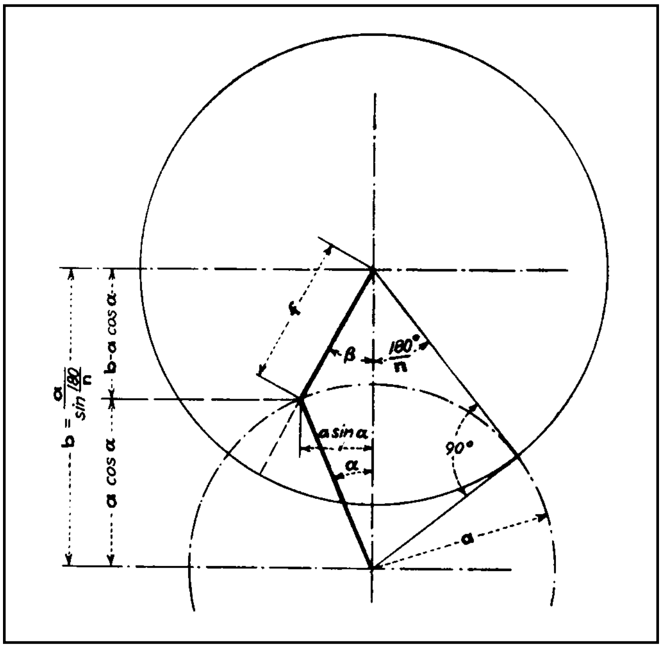


Fig. 1 A basic outline drawing for the external geneva wheel. The symbols are identified for application in the basic equations.

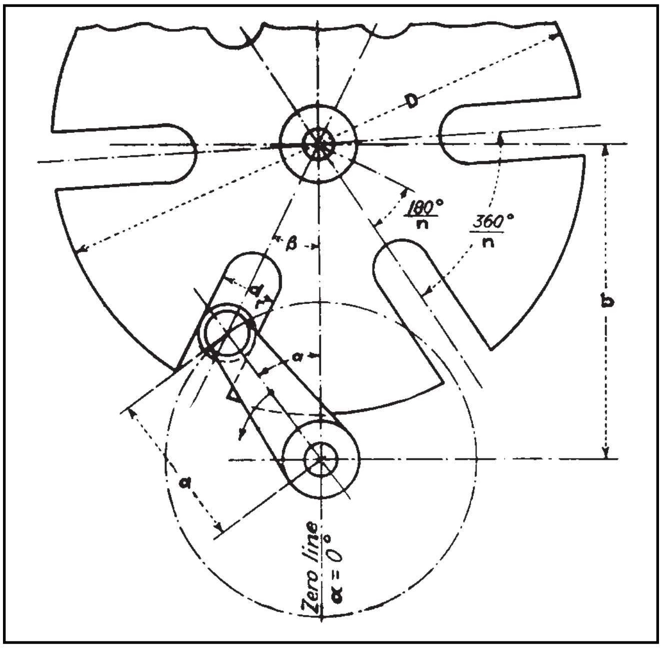


Fig. 2 A schematic drawing of a six-slot geneva wheel. Roller diameter, d_r, must be considered when determining D.

Table I—Notation and Formulas for the External Geneva Wheel

Assumed or given: a, n, d and p

a = crank radius of driving member

n = number of slots

d_r = roller diameter

p = constant velocity of driving crank in rpm

$$m = \frac{1}{\sin \frac{180}{n}}$$

b = center distance = am

$$D = \text{diameter of driven member} = 2\sqrt{\frac{d_r^2}{4} + a^2 \cot^2 \frac{180}{n}}$$

$$\omega = \text{constant angular velocity of driving crank} = \frac{p\pi}{30} \text{ radians per sec}$$

α = angular position of driving crank at any time

β = angular displacement of driven member corresponding to crank angle α

$$\cos\beta = \frac{m - \cos\alpha}{\sqrt{1 + m^2 - 2m\cos\alpha}}$$

$$\text{Angular Velocity of driven member} = \frac{d\beta}{dt} = \omega\left(\frac{m\cos\alpha - 1}{1 + m^2 - 2m\cos\alpha}\right)$$

$$\text{Angular Acceleration of driven member} = \frac{d^2\beta}{dt^2} = \omega^2\left(\frac{m\sin\alpha\,(1 - m^2)}{(1 + m^2 - 2m\cos\alpha)^2}\right)$$

Maximum Angular Acceleration occurs when $\cos\alpha =$

$$\sqrt{\left(\frac{1 + m^2}{4\,m}\right)^2 + 2} - \left(\frac{1 + m^2}{4\,m}\right)$$

Maximum Angular Velocity occurs at $\alpha = 0$ deg, and equals

$$\frac{\omega}{m - 1} \text{ radians per sec}$$

One of the most commonly applied mechanisms for producing intermittent rotary motion from a uniform input speed is the external geneva wheel.

The driven member, or star wheel, contains many slots into which the roller of the driving crank fits. The number of slots determines the ratio between dwell and motion period of the driven shaft. The lowest possible number of slots is three, while the highest number is theoretically unlimited. In practice, the three-slot geneva is seldom used because of the extremely high acceleration values encountered. Genevas with more than 18 slots are also infrequently used because they require wheels with comparatively large diameters.

In external genevas of any number of slots, the dwell period always exceeds the motion period. The opposite is true of the internal geneva. However, for the spherical geneva, both dwell and motion periods are 180°.

For the proper operation of the external geneva, the roller must enter the slot tangentially. In other words, the centerline of the slot and the line connecting the roller center and crank rotation center must form a right angle when the roller enters or leaves the slot.

The calculations given here are based on the conditions stated here.

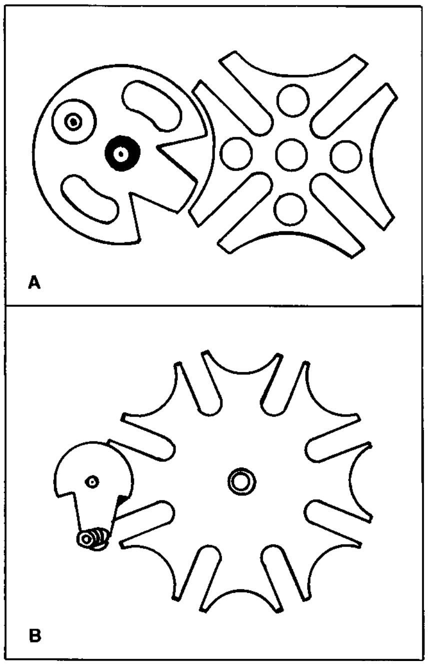


Fig. 3 A four-slot geneva (A) and an eight-slot geneva (B). Both have locking devices.

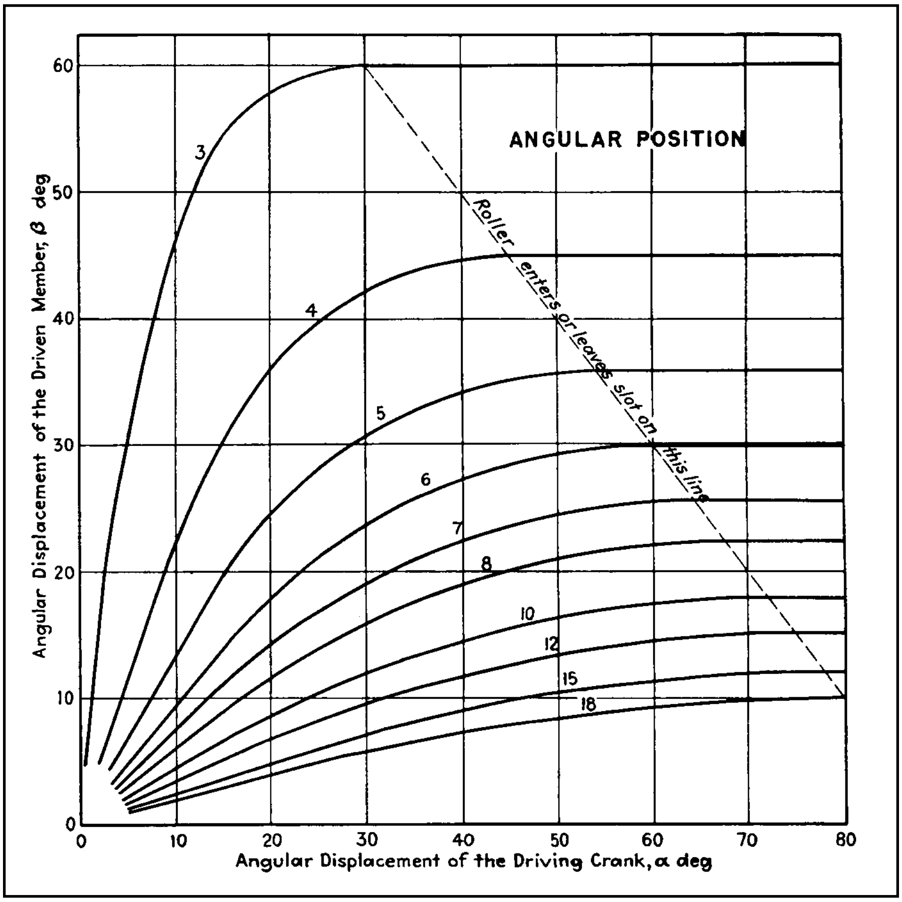


Fig. 4 Chart for determining the angular displacement of the driven member.

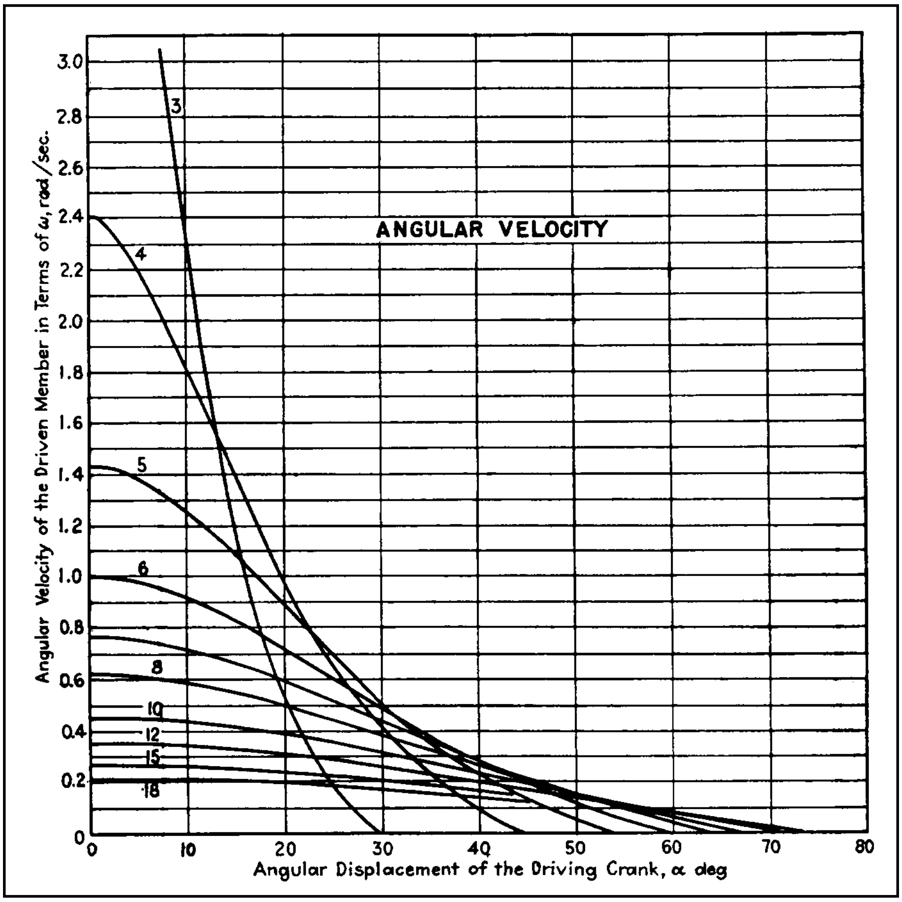


Fig. 5 Chart for determining the angular velocity of the driven member.

Consider an external geneva wheel, shown in Fig. 1, in which

n = number of slots

a = crank radius

From

Fig. 1, b = center distance = $\dfrac{a}{\sin\dfrac{180}{n}}$

Let $\dfrac{1}{\sin\dfrac{180}{n}} = m$

Then $b = am$

It will simplify the development of the equations of motion to designate the connecting line of the wheel and crank centers as the zero line. This is contrary to the practice of assigning the zero value of α, representing the angular position of the driving crank, to that position of the crank where the roller enters the slot.

Thus, from Fig. 1, the driven crank radius f at any angle is:

$$f = \sqrt{(am - a\cos\alpha)^2 + \alpha^2 \sin^2\alpha}$$

$$= \alpha\sqrt{1 + m^2 - 2m\cos\alpha} \qquad (1)$$

Table II—Principal Kinematic Data for External Geneva Wheel

No. of Slots	$\frac{360°}{n}$	Dwell period	Motion period	m and center-distance for $a = 1$	Maximum angular velocity of driven member, radians per sec. equals ω multiplied by values tabulated. Crank at 0° position	Angular acceleration of driven member when roller enters slot, radians² per sec², equals ω^2 multiplied by values tabulated.			Maximum angular Acceleration of driven member, radians² per sec², equals ω^2 multiplied by values tabulated		
						α	β	Multiplier	α	β	Multiplier
3	120°	300°	60°	1.155	6.458	30°	60°	1.729	4°	27° 58′	29.10
4	90°	270°	90°	1.414	2.407	45°	45°	1.000	11° 28′	25° 11′	5.314
5	72°	252°	108°	1.701	1.425	54°	36°	0.727	17° 31′	21° 53′	2.310
6	60°	240°	120°	2.000	1.000	60°	30°	0.577	22° 55′	19° 51′	1.349
7	51° 25′ 43″	231° 30′	128° 30′	2.305	0.766	64° 17′ 8″	25° 42′ 52″	0.481	27° 41′	18° 11′	0.928
8	45°	225°	135°	2.613	0.620	67° 30′	22° 30′	0.414	31° 38′	16° 32′	0.700
9	40°	220°	140°	2.924	0.520	70°	20°	0.364	35° 16′	15° 15′	0.559
10	36°	216°	144°	3.236	0.447	72°	18°	0.325	38° 30′	14° 16′	0.465
11	32° 43′ 38″	212° 45′	147° 15′	3.549	0.392	73° 38′ 11″	16° 21′ 49″	0.294	41° 22′	13° 16′	0.398
12	30°	210°	150°	3.864	0.349	75°	15°	0.268	44°	12° 26′	0.348
13	27° 41′ 32″	207° 45′	152° 15′	4.179	0.315	76° 9′ 14″	13° 50′ 46″	0.246	46° 23′	11° 44′	0.309
14	25° 42′ 52″	205° 45′	154° 15′	4.494	0.286	77° 8′ 34″	21° 51′ 26″	0.228	48° 32′	11° 3′	0.278
15	24°	204°	156°	4.810	0.263	78°	12°	0.213	50° 30′	10° 27′	0.253
16	22° 30′	202° 30′	157° 30′	5.126	0.242	78° 45′	11° 15′	0.199	52° 24′	9° 57′	0.232
17	21° 10′ 35″	201°	159°	5.442	0.225	79° 24′ 43″	10° 35′ 17″	0.187	53° 58′	9° 26′	0.215
18	20°	200°	160°	5.759	0.210	80°	10°	0.176	55° 30′	8° 59′	0.200

and the angular displacement β can be found from:

$$\cos\beta = \frac{m - \cos a}{\sqrt{1 + m^2 - 2m\cos\alpha}} \quad (2)$$

A six-slot geneva is shown schematically in Fig. 2. The outside diameter D of the wheel (when accounting for the effect of the roller diameter d) is found to be:

$$D = 2\sqrt{\frac{d_r^2}{4} + a^2\cot^2\frac{180}{n}} \quad (3)$$

Differentiating Eq. (2) and dividing by the differential of time, dt, the angular velocity of the driven member is:

$$\frac{d\beta}{dt} = \omega\left(\frac{m\cos\alpha - 1}{1 + m^2 - 2m\cos\alpha}\right) \quad (4)$$

where ω represents the constant angular velocity of the crank.

By differentiation of Eq. (4) the acceleration of the driven member is found to be:

$$\frac{d^2\beta}{dt^2} = \omega^2\left(\frac{m\sin\alpha(1 - m^2)}{(1 + m^2 - 2m\cos\alpha)^2}\right) \quad (5)$$

All notations and principal formulas are given in Table I for easy reference. Table II contains all the data of principal interest for external geneva wheels having from 3 to 18 slots. All other data can be read from the charts: Fig. 4 for angular position, Fig. 5 for angular velocity, and Fig. 6 for angular acceleration.

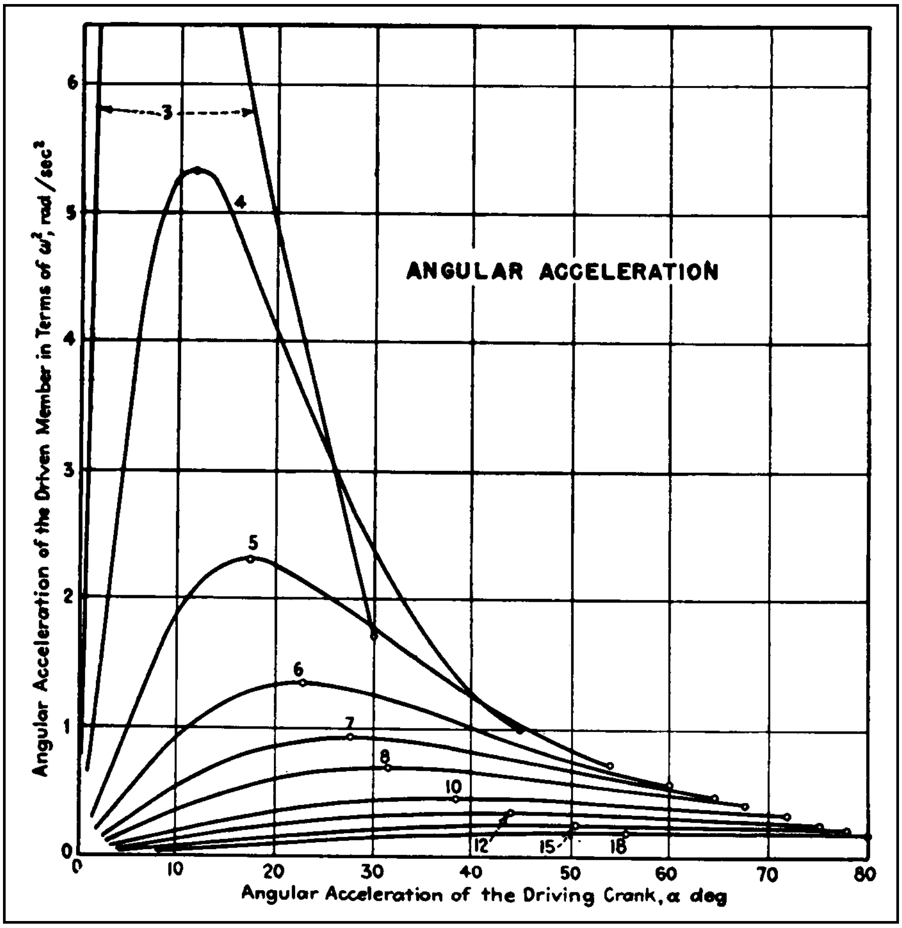


Fig. 6 Chart for determining the angular acceleration of the driven member.

KINEMATICS OF INTERMITTENT MECHANISMS—THE INTERNAL GENEVA WHEEL

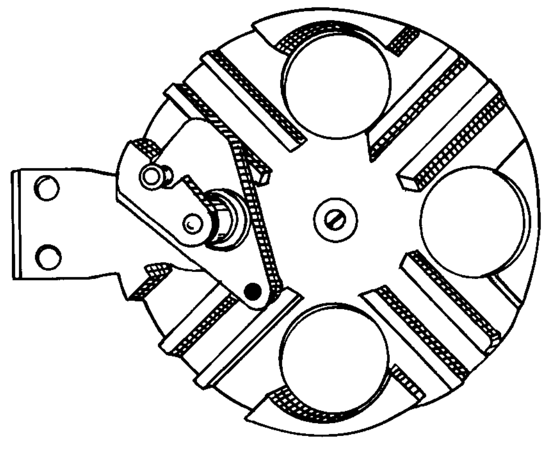


Fig. 1 A four-slot internal geneva wheel incorporating a locking mechanism. The basic sketch is shown in Fig. 3.

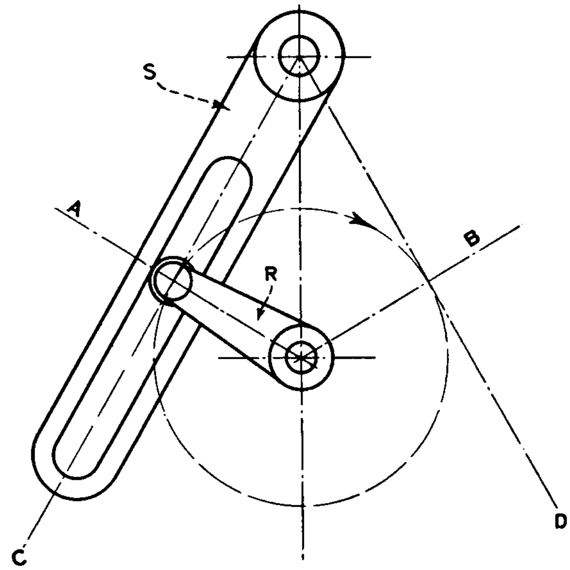


Fig. 2 Slot-crank motion from *A* to *B* represents external geneva action; from *B* to *A* represents internal geneva motion.

Where intermittent drives must provide dwell periods of more than 180°, the external geneva wheel design is satisfactory and is generally the standard device employed. But where the dwell period must be less than 180°, other intermittent drive mechanisms must be used. The internal geneva wheel is one way of obtaining this kind of motion.

The dwell period of all internal genevas is always smaller than 180°. Thus, more time is left for the star wheel to reach maximum velocity, and acceleration is lower. The highest value of angular acceleration occurs when the roller enters or leaves the slot. However, the acceleration occurs when the roller enters or leaves the slot. However, the acceleration curve does not reach a peak within the range of motion of the driven wheel. The geometrical maximum would occur in the continuation of the curve. But this continuation has no significance because the driven member will have entered the dwell phase associated with the high angular displacement of the driving member.

The geometrical maximum lies in the continuation of the curve, falling into the region representing the motion of the external geneva wheel. This can be seen by the following considerations of a crank and slot drive, drawn in Fig. 2.

When the roller crank *R* rotates, slot link *S* will perform an oscillating move-

Table I—Notation and Formulas for the Internal Geneva Wheel

Assumed or given: a, n, d and p

a = crank radius of driving member

n = number of slots

d = roller diameter

p = constant velocity of driving crank in rpm

$$m = \frac{1}{\sin \frac{180^\circ}{n}}$$

b = center distance = $a\,m$

$$D = \text{inside diameter of driven member} = 2\sqrt{\frac{d^2}{4} + a^2 \cot^2 \frac{180^\circ}{n}}$$

ω = constant angular velocity of driving crank in radians per sec =

$\frac{p\pi}{30}$ radians per sec

α = angular position of driving crank at any time

β = angular displacement of driven member corresponding to crank angle α

$$\cos\beta = \frac{m + \cos\alpha}{\sqrt{1 + m^2 + 2m\cos\alpha}}$$

$$\text{Angular velocity of driven member} = \frac{d\beta}{dt} = \omega\left(\frac{1 + m\cos\alpha}{1 + m^2 + 2m\cos\alpha}\right)$$

$$\text{Angular acceleration of driven member} = \frac{d^2\beta}{dt^2} = \omega^2\left[\frac{m\sin\alpha\,(1 - m^2)}{(1 + m^2 + 2m\cos\alpha)^2}\right]$$

Maximum angular velocity occurs at $\alpha = 0°$ and equals = $\frac{\omega}{1 + m}$ radians per sec

Maximum angular acceleration occurs when roller enters slot and equals =

$\frac{\omega^2}{\sqrt{m^2 - 1}}$ radians^2 per sec^2

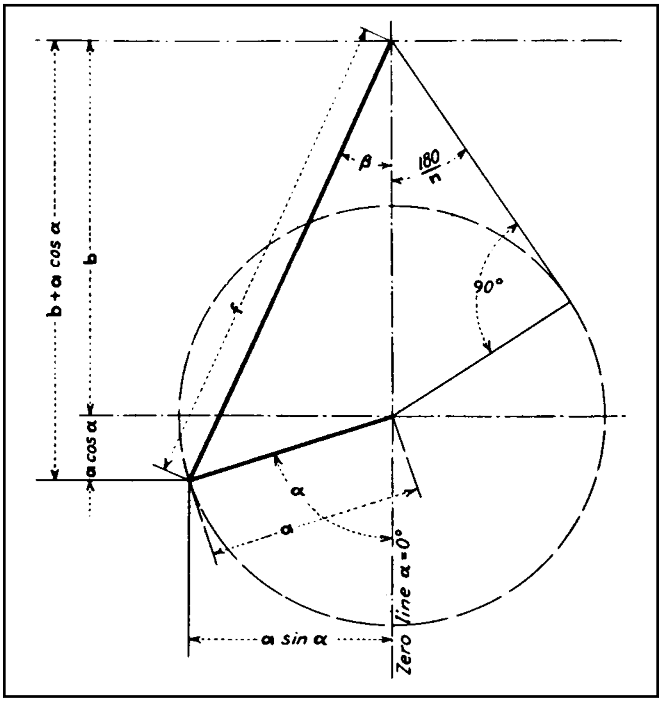


Fig. 3 A basic outline for developing the equations of the internal geneva wheel, based on the notations shown.

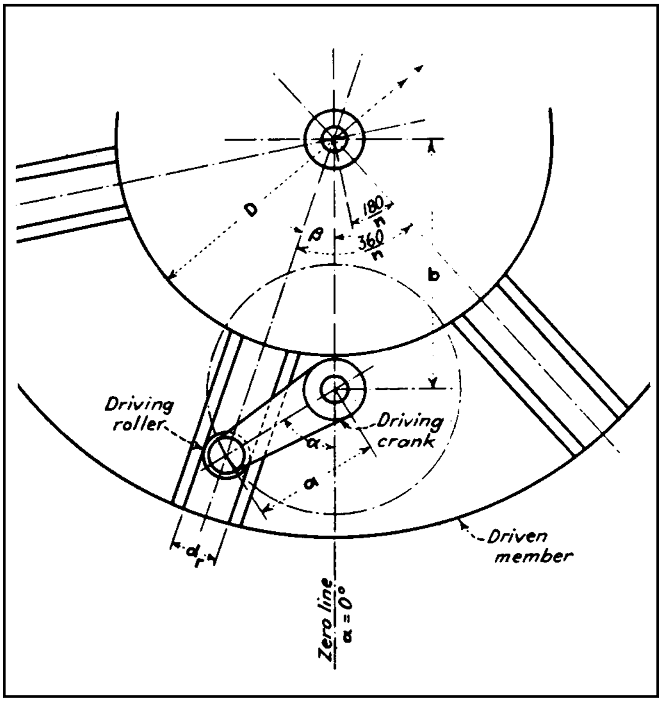


Fig. 4 A drawing of a six-slot internal geneva wheel. The symbols are identified, and the motion equations are given in Table I.

ment, for which the displacement, angular velocity, and acceleration can be given in continuous curves.

When the crank *R* rotates from *A* to *B*, then the slot link *S* will move from *C* to *D*, exactly reproducing all moving conditions of an external geneva of equal slot angle. When crank *R* continues its movement from *B* back to *A*, then the slot link *S* will move from *D* back to *C*, this time reproducing exactly (though in a mirror picture with the direction of motion being reversed) the moving conditions of an internal geneva.

Therefore, the characteristic curves of this motion contain both the external and internal geneva wheel conditions; the region of the external geneva lies between *A* and *B*, the region of the internal geneva lies between *B* and *A*.

The geometrical maxima of the acceleration curves lie only in the region between *A* and *B*, representing that portion of the curves which belongs to the external geneva.

The principal advantage of the internal geneva, other than its smooth operation, is it sharply defined dwell period. A disadvantage is the relatively large size of the driven member, which increases the force resisting acceleration. Another feature, which is sometimes a disadvantage, is the cantilever arrangement of the roller crank shaft. This shaft cannot be a through shaft because the crank must be fastened to the overhanging end of the input shaft.

To simplify the equations, the connecting line of the wheel and crank centers is taken as the zero line. The angular

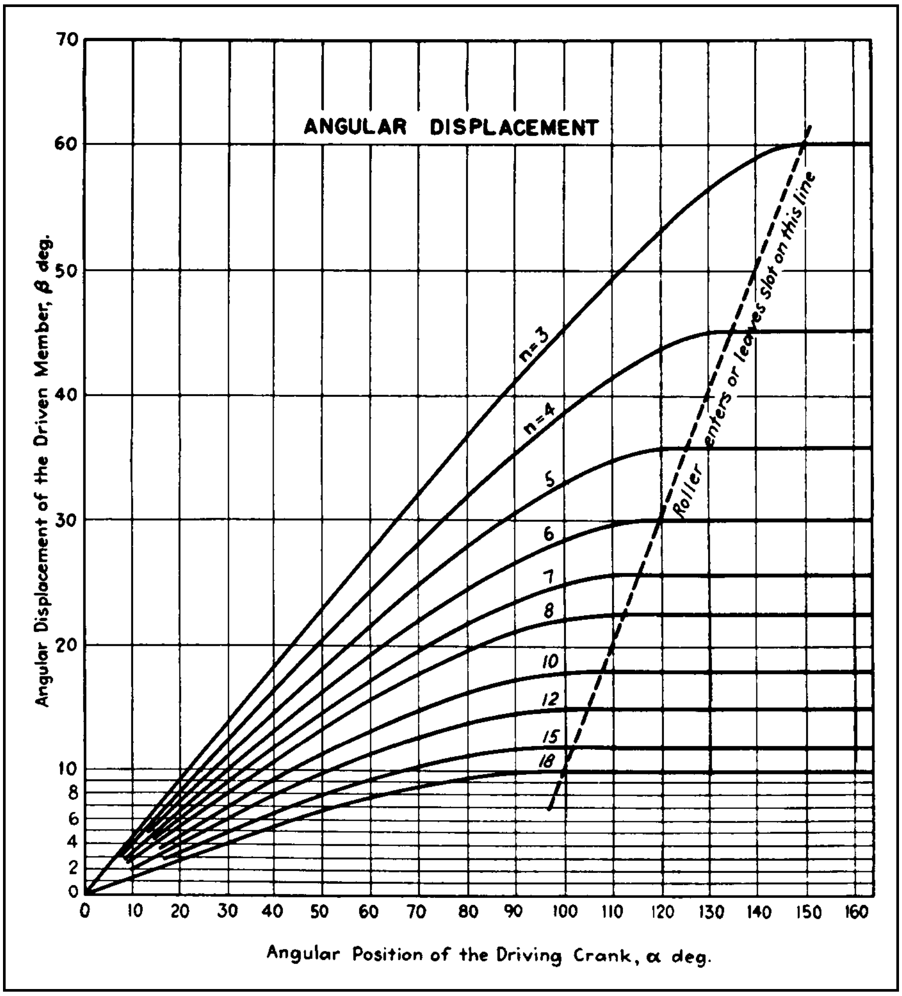


Fig. 5 Angular displacement of the driven member can be determined from this chart.

position of the driving crank α is zero when it is on this line. Then the following relations are developed, based on Fig. 3.

n = number of slots
a = crank radius
b = center distance = $\dfrac{a}{\sin\dfrac{180^\circ}{n}}$

Let

$$\frac{1}{\sin\dfrac{180^\circ}{n}} = m,$$

then; $b = am$

To find the angular displacement β of the driven member, the driven crank radius f is first calculated from:

$$f = \sqrt{a^2 \sin^2 \alpha + (am + a\cos\alpha)^2}$$
$$= \alpha\sqrt{1 + m^2 + 2m\cos\alpha} \qquad (1)$$

and because

$$\cos\beta = \frac{m + \cos\alpha}{f}$$

it follows:

$$\cos\beta = \frac{m + \cos\alpha}{\sqrt{1 + m^2 + 2m\cos\alpha}} \qquad (2)$$

From this formula, β, the angular displacement, can be calculated for any angle α, the angle of the mechanism's driving member.

The first derivative of Eq. (2) gives the angular velocity as:

$$\frac{d\beta}{dt} = \omega\left(\frac{1 + m\cos\alpha}{1 + m^2 + 2m\cos\alpha}\right) \qquad (3)$$

where ω designates the uniform speed of the driving crank shaft, namely:

$$\omega = \frac{p\pi}{30}$$

if p equals its number of revolutions per minute.

Differentiating Eq. (3) once more develops the equation for the angular acceleration:

$$\frac{d^2\beta}{dt^2} = \omega^2\left[\frac{m\sin\alpha(1 - m^2)}{(1 + m^2 + 2m\cos\alpha)^2}\right] \qquad (4)$$

The maximum angular velocity occurs, obviously, at $\alpha = 0^\circ$. Its value is found by substituting 0° for α in Eq. (3). It is:

$$\frac{d\beta}{dt_{\max}} = \frac{\omega}{1 + m} \qquad (5)$$

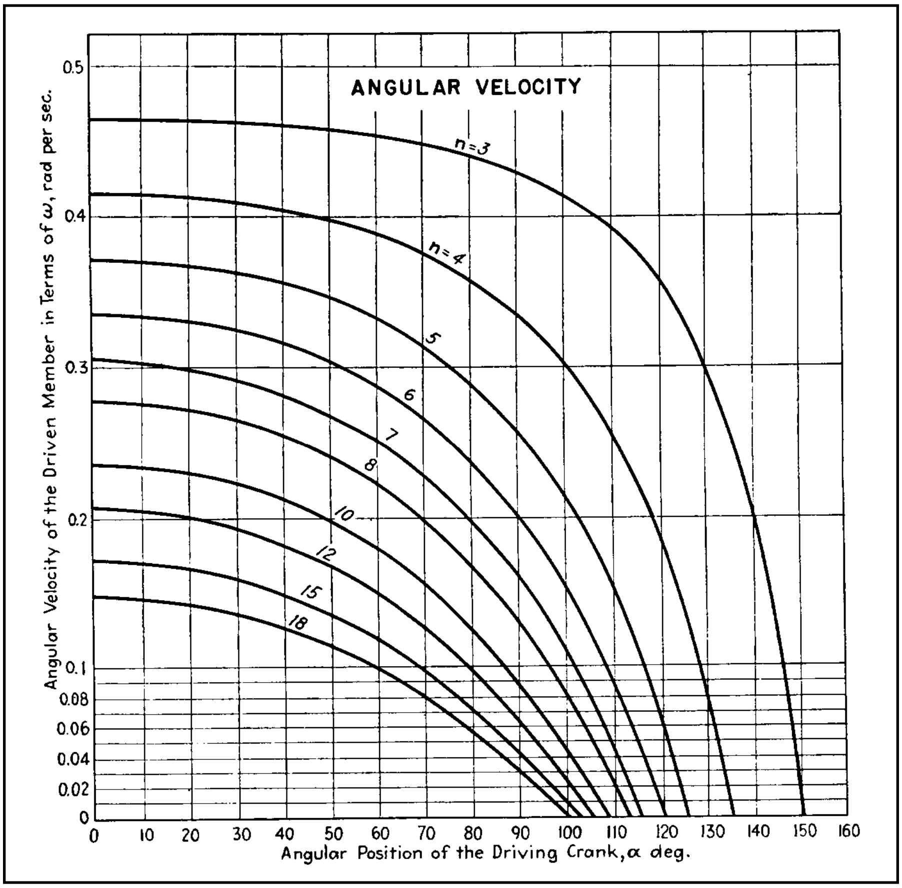


Fig. 6 Angular velocity of the driven member can be determined from this chart.

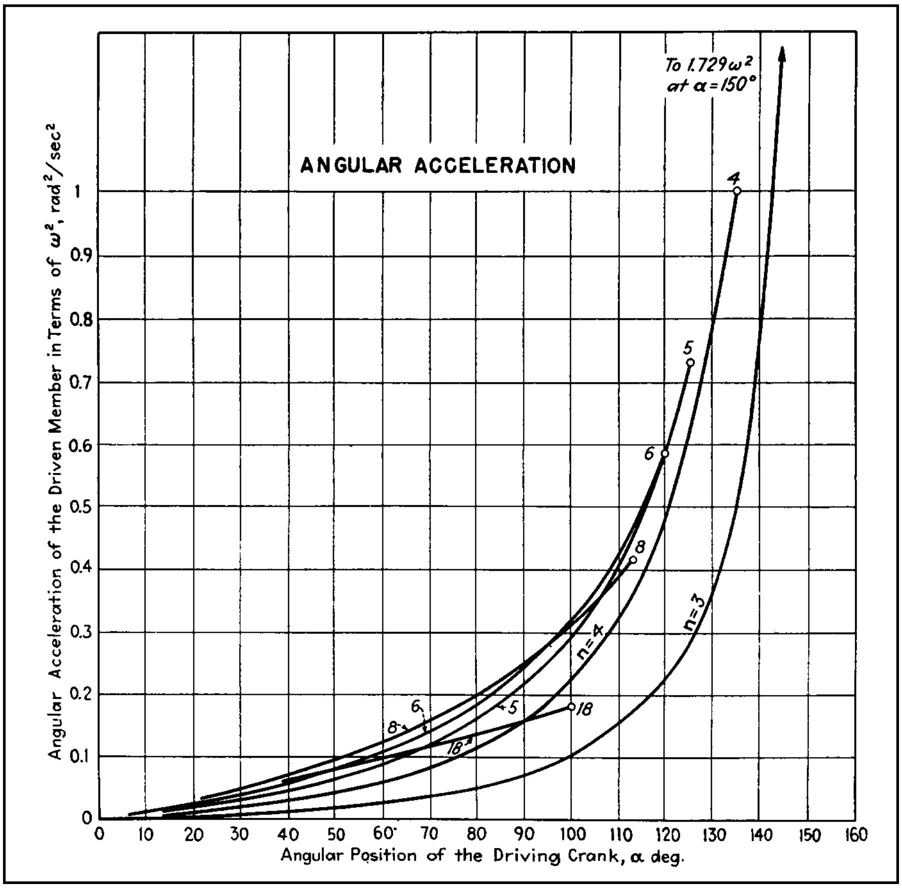


Fig. 7 Angular acceleration of the driven member can be determined from this chart.

Table II—Kinematic Data For the Internal Geneva Wheel

Number of slots, n	$\frac{360°}{n}$	Dwell period	Motion period	m and center-distance for $a = 1$	Maximum angular velocity of driven member equals ω radians per sec. multiplied by values tabulated. Both α and β in 0° position	Angular acceleration of driven member when roller enters slot equals ω^2 radians^2 per sec^2 multiplied by values tabulated: α	β	Multiplier
3	120°	60°	300°	1.155	0.464	150°	60°	1.729
4	90°	90°	270°	1.414	0.414	135°	45°	1.000
5	72°	108°	252°	1.701	0.370	126°	36°	0.727
6	60°	120°	240°	2.000	0.333	120°	30°	0.577
7	51° 25′ 43″	128° 30′	231° 30′	2.305	0.303	115° 42′ 52″	25° 42′ 52″	0.481
8	45°	135°	225°	2.613	0.277	112° 30′	22° 30′	0.414
9	40°	140°	220°	2.924	0.255	110°	20°	0.364
10	36°	144°	216°	3.236	0.236	108°	18°	0.325
11	32° 43′ 38″	147° 15′	212° 45′	3.549	0.220	106° 21′ 49″	16° 21′ 49″	0.294
12	30°	150°	210°	3.864	0.206	105°	15°	0.268

The highest value of the acceleration is found by substituting $180/n + 980$ for α in Eq. (4):

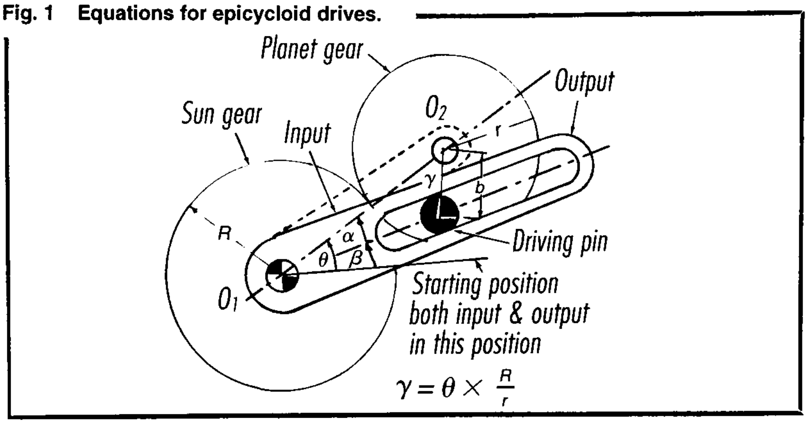
$$\frac{d^2\beta}{dt^2_{\max}} = \frac{\omega^2}{\sqrt{m^2 - 1}} \qquad (6)$$

A layout drawing for a six-slot internal geneva wheel is shown in Fig. 4. All the symbols in this drawing and throughout the text are compiled in Table I for easy reference.

Table II contains all the data of principal interest on the performance of internal geneva wheels that have from 3 to 18 slots. Other data can be read from the charts: Fig. 5 for angular position, Fig. 6 for angular velocity, and Fig. 7 for angular acceleration.

EQUATIONS FOR DESIGNING CYCLOID MECHANISMS

Fig. 1 Equations for epicycloid drives.



The equations for angular displacement, velocity, and acceleration for a basic epicyclic drive are given below.

Angular displacement

$$\tan\beta = \frac{(R+r)\sin\theta - b\sin(\theta+\gamma)}{(R+r)\cos\theta - b\cos(\theta+\gamma)} \qquad (1)$$

Angular velocity

$$V = \omega \frac{1 + \dfrac{b^2}{r(R+r)} - \left(\dfrac{2r+R}{r}\right)\left(\dfrac{b}{R+r}\right)\left(\cos\dfrac{R}{r}\theta\right)}{1 + \left(\dfrac{b}{R+r}\right)^2 - \left(\dfrac{2b}{R+r}\right)\left(\cos\dfrac{R}{r}\theta\right)} \qquad (2)$$

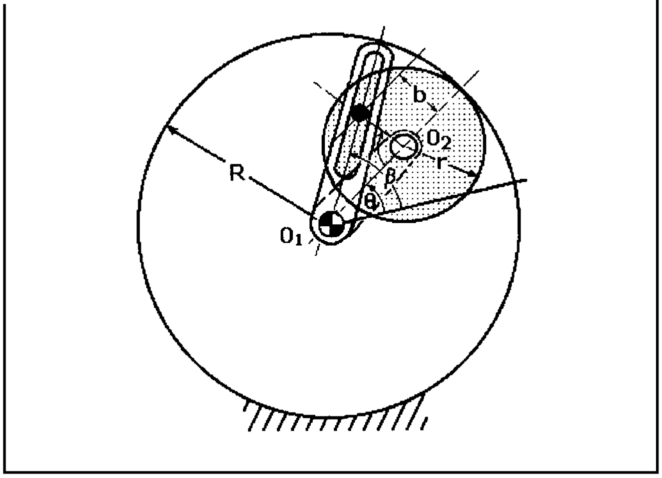
Angular acceleration

$$A = \omega^2 \frac{\left(1 - \dfrac{b^2}{(R+r)^2}\right)\left(\dfrac{R^2}{r^2}\right)\left(\dfrac{b}{R+r}\right)\left(\sin\dfrac{R}{r}\theta\right)}{\left[1 + \dfrac{b^2}{(R+r)^2} - \left(\dfrac{2b}{R+r}\right)\left(\cos\dfrac{R}{r}\theta\right)\right]^2} \qquad (3)$$

Symbols

- A = angular acceleration of output, degrees per second^2
- b = radius of driving pin from center of planet gear
- r = pitch radius of planet gear
- R = pitch radius of fixed sun gear
- V = angular velocity of output, degrees per second
- β = angular displacement of output, degree
- γ = $\theta R/r$
- θ = input displacement, degree
- ω = angular velocity of input, degrees per second

Fig. 2 Equations for hypocycloid drives.



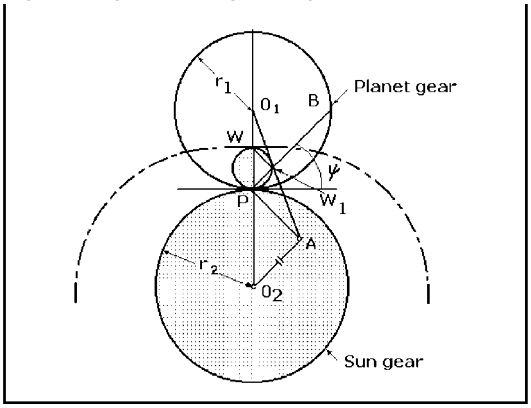
$$\tan\beta = \frac{\sin\theta - \left(\frac{b}{R-r}\right)\left(\sin\frac{R-r}{r}\theta\right)}{\cos\theta + \left(\frac{b}{R-r}\right)\left(\cos\frac{R-r}{r}\theta\right)} \qquad (4)$$

$$V = \omega\frac{1-\left(\frac{R-r}{r}\right)\left(\frac{b^2}{(R-r)^2}\right)+\left(\frac{2r-R}{r}\right)\left(\frac{b}{R-r}\right)\left(\cos\frac{R}{r}\theta\right)}{1+\frac{b^2}{(R-r)^2}+\left(\frac{2b}{R-r}\right)\left(\cos\frac{R}{r}\theta\right)} \qquad (5)$$

$$A = \omega^2\frac{\left(1-\frac{b^2}{(R-r)^2}\right)\left(\frac{b}{R-r}\right)\left(\frac{R^2}{r^2}\right)\left(\sin\frac{R}{r}\theta\right)}{\left[1+\frac{b^2}{(R-r)^2}+\left(\frac{2b}{R-r}\right)\left(\cos\frac{R}{r}\theta\right)\right]^2} \qquad (6)$$

DESCRIBING APPROXIMATE STRAIGHT LINES

Fig. 3 A gear rolling on a gear flattens curves.



It is frequently desirable to find points on the planet gear that will describe approximately straight lines for portions of the output curves. These points will yield dwell mechanisms. Construction is as follows (see drawing):

1. Draw an arbitrary line *PB*.
2. Draw its parallel O_2A.
3. Draw its perpendicular *PA* at *P*. Locate point *A*.
4. Draw O_1A. Locate W_1.
5. Draw perpendicular to PW_1 at W_1 to locate *W*.
6. Draw a circular with *PW* as the diameter.

All points on this circle describe curves with portions that are approximately straight. This circle is also called the **inflection circle** because all points describe curves that have a point of inflection at the position illustrated. (The curve passing through point *W* is shown.)

Fig. 4 A gear rolling on a rack describes vee curves.

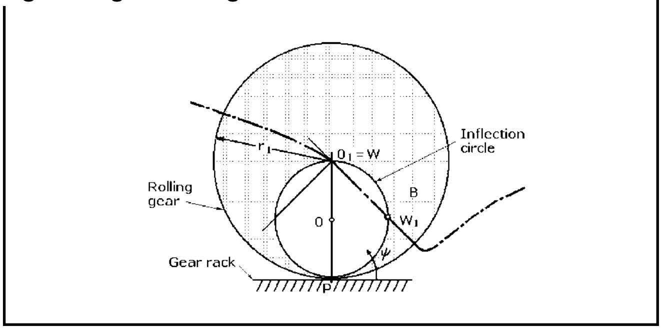
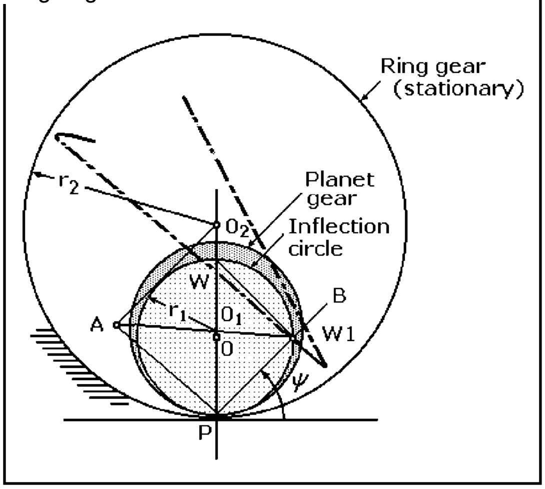


Fig. 5 A gear rolling inside a gear describes a zig-zag.



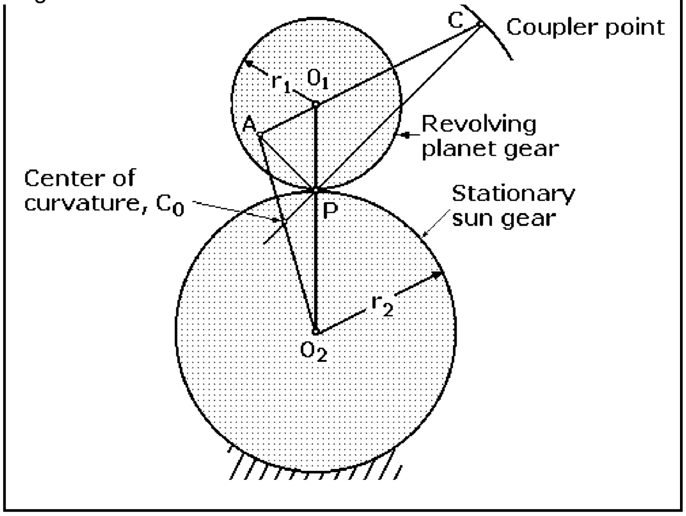
This is a special case. Draw a circle with a diameter half that of the gear (diameter O_1P). This is the inflection circle. Any point, such as point W_1, will describe a curve that is almost straight in the vicinity selected. Tangents to the curves will always pass through the center of the gear, O_1 (as shown).

To find the inflection circle for a gear rolling inside a gear:

1. Draw arbitrary line *PB* from the contact point *P*.
2. Draw its parallel O_2A, and its perpendicular, *PA*. Locate *A*.
3. Draw line AO_1 through the center of the rolling gear. Locate W_1.
4. Draw a perpendicular through W_1. Obtain *W*. Line *WP* is the diameter of the inflection circle. Point W_1, which is an arbitrary point on the circle, will trace a curve of repeated almost-straight lines, as shown.

DESIGNING FOR DWELLS

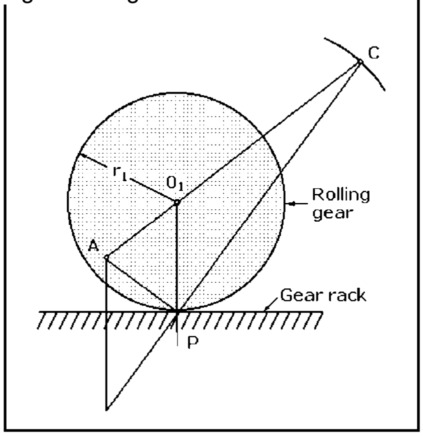
Fig. 6 The center of curvature: a gear rolling on gear.



By locating the centers of curvature at various points, one can determine the length of the rocking or reciprocating arm to provide long dwells.

1. Draw a line through points C and P.
2. Draw a line through points C and O_1.
3. Draw a perpendicular to CP at P. This locates point A.
4. Draw line AO_2, to locate C_0, the center of curvature.

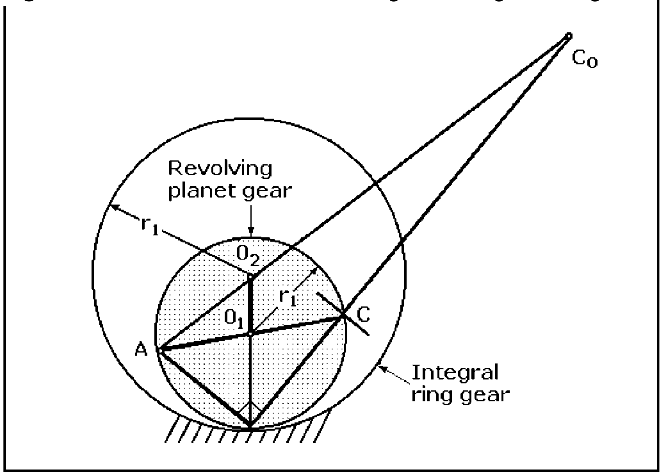
Fig. 7 The center of curvature: a gear rolling on a rack



Construction is similar to that of the previous case.

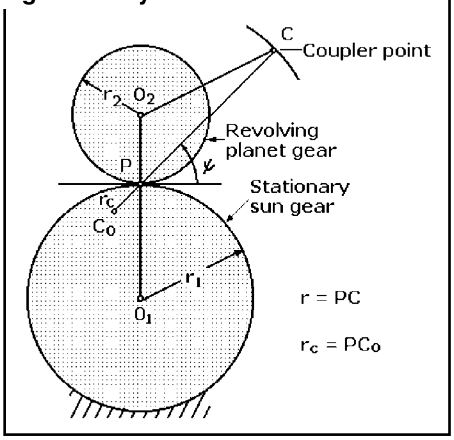
1. Draw an extension of line CP.
2. Draw a perpendicular at P to locate A.
3. Draw a perpendicular from A to the straight surface to locate C.

Fig. 8 The center of curvature: a gear rolling iside a gear.



1. Draw extensions of CP and CO_1.
2. Draw a perpendicular of PC at P to locate A.
3. Draw AO_2 to locate C_0.

Fig. 9 Analytical solutions.



The center of curvature of a gear rolling on an external gear can be computed directly from the Euler-Savary equation:

$$\left(\frac{1}{r}-\frac{1}{r_c}\right)\sin\psi = \text{constant} \qquad (7)$$

where angle ψ and r locate the position of C.

By applying this equation twice, specifically to point O_1 and O_2, which have their own centers of rotation, the following equation is obtained:

$$\left(\frac{1}{r_2}-\frac{1}{r_1}\right)\sin 90^\circ = \left(\frac{1}{r}+\frac{1}{r_c}\right)\sin\psi$$

or

$$\frac{1}{r_2}+\frac{1}{r_1} = \left(\frac{1}{r}+\frac{1}{r_c}\right)\sin\psi$$

This is the final design equation. All factors except r_c are known; hence, solving for r_c leads to the location of C_0.

For a gear rolling inside an internal gear, the Euler-Savary equation is:

$$\left(\frac{1}{r}+\frac{1}{r_c}\right)\sin\psi = \text{constant}$$

which leads to:

$$\frac{1}{r_2}-\frac{1}{r_1} = \left(\frac{1}{r}-\frac{1}{r_c}\right)\sin\psi$$

DESIGNING CRANK-AND-ROCKER LINKS WITH OPTIMUM FORCE TRANSMISSION

Four-bar linkages can be designed with a minimum of trial and error by a combination of tabular and iteration techniques.

The determination of optimum crank-and-rocker linkages has most effectively been performed on a computer because of the complexity of the equations and calculations involved. Thanks to the work done at Columbia University's Department of Mechanical and Nuclear Engineering, all you need now is a calculator and the computer-generated tables presented here. The computations were done by Mr. Meng-Sang Chew, at the university.

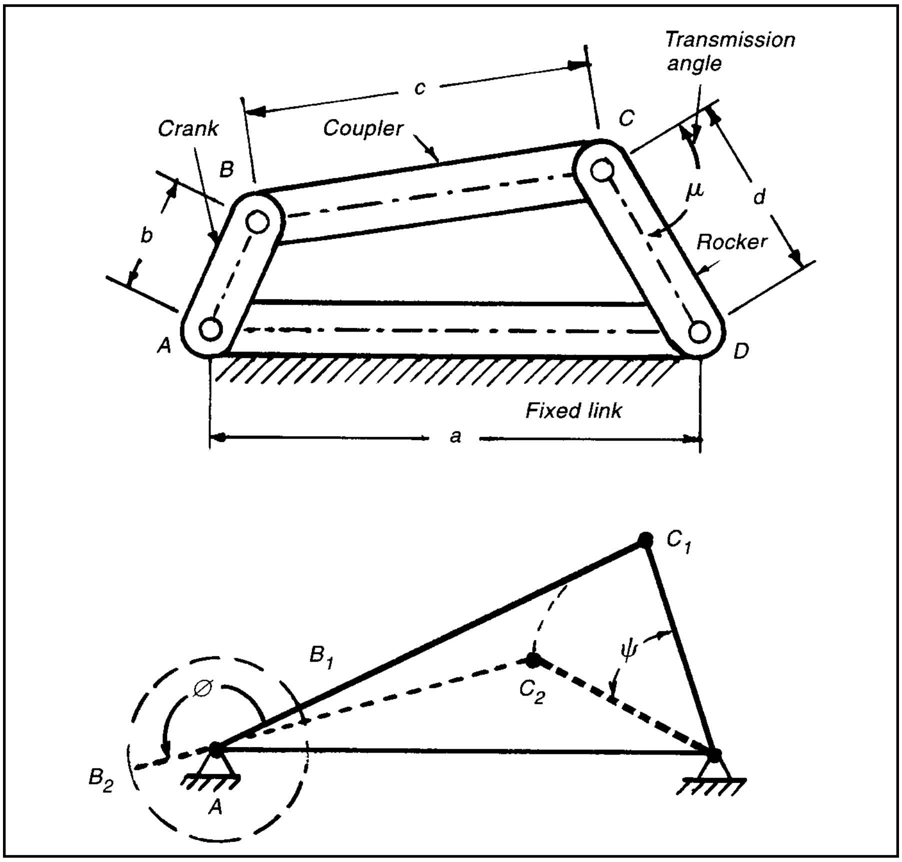
A crank-and-rocker linkage, *ABCD*, is shown in the first figure. The two extreme positions of the rocker are shown schematically in the second figure. Here ψ denotes the rocker swing angle and ϕ denotes the corresponding crank rotation, both measured counterclockwise from the extended dead-center position, AB_1C_1D.

The problem is to find the proportions of the crank-and-rocker linkage for a given rocker swing angle, ψ, a prescribed corresponding crank rotation, ϕ, and optimum force transmission. The latter is usually defined in terms of the transmission angle, m, the angle μ between coupler *BC* extended and rocker *CD*.

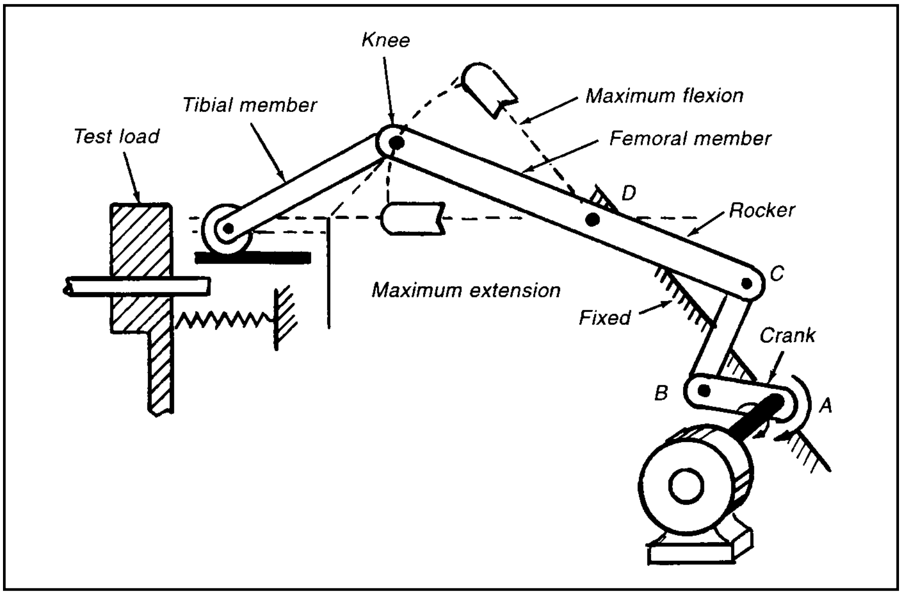
Considering static forces only, the closer the transmission angle is to 90º, the greater is the ratio of the driving component of the force exerted on the rocker to the component exerting bearing pressure on the rocker. The control of transmission-angle variation becomes especially important at high speeds and in heavy-duty applications.

How to find the optimum. The steps in the determination of crank-and-rocker proportions for a given rocker swing angle, corresponding crank rotation, and optimum transmission, are:

- Select (ψ, ϕ) within the following range:
  $0º < \psi < 180º$
  $(90º + 1/2\ \psi) < \phi < (270º + 1/2\ \psi)$
- Calculate: $t = \tan 1/2\ \phi$
  $u = \tan 1/2(\phi - \psi)$
  $v = \tan 1/2\ \psi$



The optimum solution for the classic four-bar crank-and-rocker mechanism problem can now be obtained with only the accompanying table and a calculator.



An example in this knee-joint tester designed and built by following the design and calculating procedures outlined in this article.

- Using the table, find the ratio λ_{opt} of coupler to crank length that minimizes the transmission-angle deviation from 90º. The most practical combinations of (ψ, ϕ) are included in the table. If the (ψ, ϕ) combination is not included, or if $\phi = 180º$, go to next steps (a,b,c):
- (a) If $\phi \neq 180º$ and (ψ, ϕ) fall outside the range given in the table, determine the arbitrary intermediate value Q from the equation:
  $Q^3 + 2Q^2 - t^2Q - (t^2/u^2)(1 + t^2) = 0$
  where $(1/u^2 < Q < t^2)$.

This is conveniently accomplished by numerical iteration:

Set $$Q_1 = \tfrac{1}{2}\left(t^2 + \frac{1}{u^2}\right)$$

Calculate Q_2, Q_3, . . . from the recursion equation:

$$Q_{i+1} = \frac{2Q_i^2(Q_i + 1) + (t^2/u^2)(1 + t^2)}{Q_i(3Q_i + 4) - t^2}$$

Iterate until the ratio $[(Q_{i+1} - Q_i)/Q_i]$ is sufficiently small, so that you obtain the desired number of significant figures. Then:

$$\lambda_{opt} = t^2/Q$$

(b) If $\phi \neq 180º$ and the determination of λ_{opt} requires interpolation between two entries in the table, let $Q_1 = t^2\lambda^2$, where λ corresponds to the nearest entry in the table, and continue as in (a) above to determine Q *and* λ_{opt}. Usually one or two iterations will suffice.

(c) $\phi = 180º$. In this case, $a^2 + b^2 = c^2 + d^2$; $\psi = 2\sin^{-1}(b/d)$; and the maximum deviation, Δ, of the transmission angle from 90º is equal to $\sin^{-1}(ab/cd)$.

- Determine linkage proportions as follows:

$$(a')^2 = \frac{u^2 + \lambda_{opt}^2}{1 + u^2}$$

$$(b')^2 = \frac{v^2}{1 + v^2}$$

$$(c')^2 = \frac{\lambda_{opt}^2 v^2}{1 + v^2}$$

$$(d')^2 = \frac{t^2 + \lambda_{opt}^2}{1 + t^2}$$

Then: $a = ka'$; $b = kb'$; $c = kc'$; $d = kd'$

where k is a scale factor, such that the length of any one link, usually the crank, is equal to a design value. The max devi-

Optimum values of lambda ratio for given ø and ψ

ø deg	ψ, deg 160	162	164	166	168	170	172	174	176	178
10	2.3532	2.4743	2.6166	2.7873	2.9978	3.2669	3.6284	4.1517	5.0119	6.8642
12	2.3298	2.4491	2.5891	2.7570	2.9636	3.2272	3.5804	4.0899	4.9224	6.6967
14	2.3064	2.4239	2.5617	2.7266	2.9293	3.1874	3.5324	4.0283	4.8342	6.5367
16	2.2831	2.3988	2.5344	2.6964	2.8953	3.1479	3.4848	3.9675	4.7482	6.3853
18	2.2600	2.3740	2.5073	2.6664	2.8615	3.1089	3.4380	3.9080	4.6650	6.2427
20	2.2372	2.3494	2.4805	2.6368	2.8282	3.0704	3.3920	3.8499	4.5848	6.1087
22	2.2145	2.3250	2.4540	2.6076	2.7954	3.0327	3.3470	3.7935	4.5077	5.9826
24	2.1922	2.3010	2.4279	2.5789	2.7631	2.9956	3.3030	3.7388	4.4338	5.8641
26	2.1701	2.2773	2.4022	2.5505	2.7314	2.9594	3.2602	3.6857	4.3628	5.7524
28	2.1483	2.2539	2.3768	2.5227	2.7004	2.9239	3.2185	3.6344	4.2948	5.6469
30	2.1268	2.2309	2.3519	2.4954	2.6699	2.8893	3.1779	3.5847	4.2295	5.5472
32	2.1056	2.2082	2.3273	2.4685	2.6401	2.8554	3.1384	3.5367	4.1668	5.4526
34	2.0846	2.1858	2.3032	2.4421	2.6108	2.8223	3.0999	3.4901	4.1066	5.3628
36	2.0640	2.1637	2.2794	2.4162	2.5821	2.7899	3.0624	3.4449	4.0486	5.2773
38	2.0436	2.1420	2.2560	2.3908	2.5540	2.7583	3.0259	3.4012	3.9927	5.1957
40	2.0234	2.1205	2.2330	2.3657	2.5264	2.7274	2.9903	3.3587	3.9388	5.1177
42	2.0035	2.0994	2.2103	2.3411	2.4994	2.6971	2.9556	3.3175	3.8868	5.0430
44	1.9839	2.0785	2.1879	2.3169	2.4728	2.6675	2.9217	3.2773	3.8364	4.9712
46	1.9644	2.0579	2.1659	2.2931	2.4468	2.6384	2.8886	3.2383	3.7877	4.9023
48	1.9452	2.0375	2.1441	2.2696	2.4211	2.6100	2.8563	3.2003	3.7404	4.8358
50	1.9262	2.0174	2.1227	2.2465	2.3959	2.5820	2.8246	3.1632	3.6945	4.7717

ation, Δ, of the transmission angle from 90º is:

$$\sin\Delta = \frac{\left|(a \pm b)^2 - c^2 - d^2\right|}{2cd}$$

$$0º \le \Delta \le 90º$$
$$+ \text{ sign if } \phi < 180º$$
$$- \text{ sign if } \phi > 180º$$

An actual example. A simulator for testing artificial knee joints, built by the Department of Orthopedic Surgery, Columbia University, under the direction of Dr. N. Eftekhar, is shown schematically. The drive includes an adjustable crank-and-rocker, ABCD. The rocker swing angle ranges from a maximum of about 48° to a minimum of about one-third of this value. The crank is 4 in. long and rotates at 150 rpm. The swing angle adjustment is obtained by changing the length of the crank.

Find the proportions of the linkage, assuming optimum-transmission proportions for the maximum rocker swing angle, as this represents the most severe condition. For smaller swing angles, the maximum transmission-angle deviation from 90º will be less.

Crank rotation corresponding to 48º rocker swing is selected at approximately 170º. Using the table, find $\lambda_{opt} = 2.6100$. This gives $a' = 1.5382$, $b' = 0.40674$, $c' = 1.0616$, and $d' = 1.0218$.

For a 4 in. crank, $k = 4/0.40674 = 9.8343$ and $a = 15.127$ in., $b = 4$ in., $c = 10.440$ in., and $d = 10.049$ in., which is very close to the proportions used. The maximum deviation of the transmission angle from 90º is 47.98º.

This procedure applies not only for the transmission optimization of crank-and-rocker linkages, but also for other crank-and-rocker design. For example, if only the rocker swing angle and the corresponding crank rotation are prescribed, the ratio of coupler to crank length is arbitrary, and the equations can be used with any value of λ_2 within the range $(1, u^2t^2)$. The ratio λ can then be tailored to suit a variety of design requirements, such as size, bearing reactions, transmission-angle control, or combinations of these requirements.

The method also was used to design dead-center linkages for aircraft landing-gear retraction systems, and it can be applied to any four-bar linkage designs that meet the requirements discussed here.

Optimum values of lambda ratio for given ø and ψ

ø deg	ψ, deg 182	184	186	188	190	192	194	196	198	200
10	7.2086	5.3403	4.4560	3.9112	3.5318	3.2478	3.0245	2.8428	2.6911	2.5616
12	7.0369	5.2692	4.4227	3.8969	3.5282	3.2507	3.0317	2.8528	2.7030	2.5748
14	6.8646	5.1881	4.3795	3.8739	3.5174	3.2478	3.0341	2.8589	2.7117	2.5855
16	6.6971	5.1013	4.3287	3.8435	3.5000	3.2392	3.0317	2.8610	2.7171	2.5934
18	6.5371	5.0121	4.2726	3.8071	3.4768	3.2252	3.0245	2.8589	2.7189	2.5982
20	6.3857	4.9226	4.2131	3.7663	3.4487	3.2065	3.0129	2.8528	2.7171	2.5998
22	6.2431	4.8344	4.1518	3.7221	3.4167	3.1837	2.9972	2.8428	2.7117	2.5982
24	6.1090	4.7484	4.0900	3.6759	3.3818	3.1575	2.9780	2.8293	2.7030	2.5934
26	5.9830	4.6652	4.0284	3.6284	3.3447	3.1286	2.9558	2.8127	2.6911	2.5855
28	5.8644	4.5849	3.9676	3.5804	3.3062	3.0976	2.9311	2.7833	2.6763	2.5748
30	5.7527	4.5079	3.9080	3.5324	3.2669	3.0652	2.9045	2.7718	2.6592	2.5616
32	5.6472	4.4339	3.8500	3.4849	3.2272	3.0318	2.8764	2.7484	2.6399	2.5461
34	5.5475	4.3630	3.7936	3.4380	3.1875	2.9979	2.8473	2.7236	2.6190	2.5287
36	5.4529	4.2949	3.7388	3.3920	3.1480	2.9636	2.8175	2.6977	2.5967	2.5097
38	5.3631	4.2296	3.6858	3.3470	3.1089	2.9294	2.7873	2.6711	2.5734	2.4894
40	5.2776	4.1669	3.6345	3.3031	3.0705	2.8953	2.7570	2.6440	2.5492	2.4680
42	5.1960	4.1067	3.5848	3.2602	3.0327	2.8615	2.7266	2.6166	2.5246	2.4459
44	5.1180	4.0487	3.5367	3.2185	2.9956	2.8282	2.6964	2.5891	2.4996	2.4232
46	5.0432	3.9928	3.4901	3.1779	2.9594	2.7954	2.6665	2.5617	2.4744	2.4001
48	4.9715	3.9389	3.4450	3.1384	2.9239	2.7631	2.6369	2.5344	2.4491	2.3767
50	4.9025	3.8869	3.4012	3.0999	2.8893	2.7314	2.6076	2.5073	2.4239	2.3533

DESIGN CURVES AND EQUATIONS FOR GEAR-SLIDER MECHANISMS

What is a gear-slider mechanism? It is little more than a crank-and-slider with two gears meshed in line with the crank (Fig. 1). But, because one of the gears (planet gear, 3) is prevented from rotating because it is attached to the connecting rod, the output is taken from the sun gear, not the slider. This produces a variety of cyclic output motions, depending on the proportions of the members.

In his investigation of the capabilities of the mechanism, Professor Preben Jensen of Bridgeport, Connecticut derived the equations defining its motion and acceleration characteristics. He then devised some variations of his own (Figs. 5 through 8). These, he believes, will outperform the parent type. Jensen illustrated how the output of one of the new mechanisms, Fig. 8, can come to dead stop during each cycle, or progressively oscillate to new positions around the clock. A machine designer, therefore, can obtain a wide variety of intermittent motions from the arrangement and, by combining two of these units, he can tailor the dwell period of the mechanism to fit the automatic feed requirements of a machine.

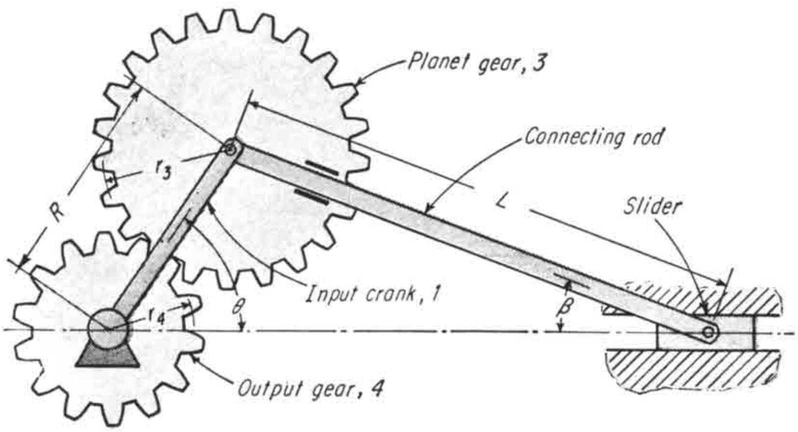


Fig. 1 A basic gear-slider mechanism. It differs from the better known three-gear drive because a slider restricts the motion of the planet gear. The output is taken from the gear, which is concentric with the input shaft, and not from the slider.

Symbols

L = Length of connecting rod, in.

r_3 = radius of gear fixed to connecting rod, in.

r_4 = radius of output gear, in.

R = length of crank, in.

α = angular acceleration of the input crank, rad/sec^2

β = connecting rod displacement, deg

γ = output rotation, deg

θ = input rotation, deg

θ_o = crank angle rotation during which the output gear reverses its motion, deg

ϕ = angle through which the output gear rotates back

ω = angular velocity of input crank, rad/sec

A single prime mark denotes angular velocity, rad/sec; double prime marks denote angular acceleration, rad/sec^2.

The Basic Form

The input motion is to crank 1, and the output motion is from gear 4. As the crank rotates, say counterclockwise, it causes planet gear 3 to oscillate while following a satellite path around gear 4. This imparts a varying output motion to gear 4, which rotates twice in the counterclockwise direction (when $r_3 = r_4$) for every revolution of the input.

Jensen's equations for angular displacement, velocity, and acceleration of gear 4, when driven at a speed of ω by crank 1, are as follows:

Angular Displacement

$$\gamma = \theta + \frac{r_3}{r_4}(\theta + \beta) \tag{1}$$

where β is computed from the following relationship (see the list of symbols in this article):

$$\sin\beta = \frac{R}{L}\sin\theta \tag{2}$$

Angular Velocity

$$\gamma' = \omega + \frac{r_3}{r_4}(\omega + \beta') \tag{3}$$

where

$$\frac{\beta'}{\omega} = \frac{R}{L}\frac{\cos\theta}{\left[1-\left(\frac{R}{L}\right)^2 \sin^2\theta\right]^{1/2}} \tag{4}$$

Angular Acceleration

$$\gamma'' = \alpha + \frac{r_3}{r_4}(\alpha + \beta'') \tag{5}$$

where

$$\frac{\beta''}{\omega^2} = \frac{R}{L}\frac{\sin\theta\left[\left(\frac{R}{L}\right)^2 - 1\right]}{\left[1-\left(\frac{R}{L}\right)^2 \sin^2\theta\right]^{3/2}} \tag{6}$$

For a constant angular velocity, Eq. 5 becomes

$$\gamma'' = \frac{r_3}{r_4}\beta'' \tag{7}$$

Design Charts

The equations were solved by Professor Jensen for various L/R ratios and positions of the crank angle θ to obtain the design charts in Figs. 2, 3, and 4. Thus, for a mechanism with

$L = 12$ in. $r_3 = 2.5$
$R = 4$ in. $r_4 = 1.5$
$\omega = 1000$ per second
= radians per second

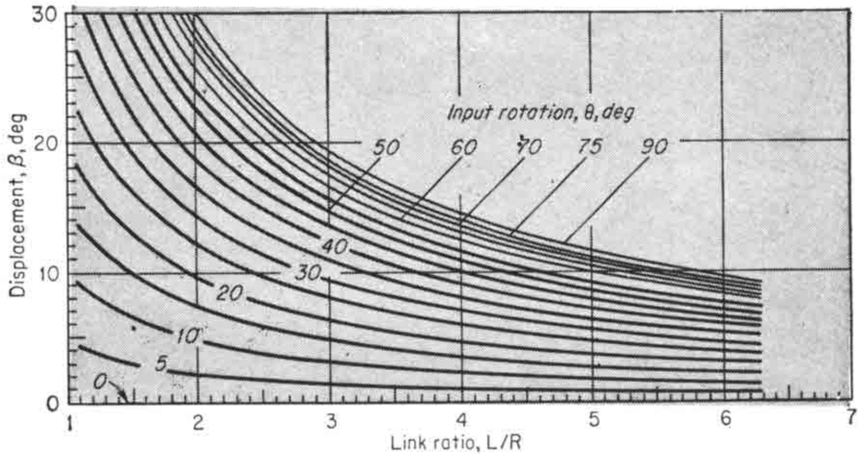


Fig. 2 Angular displacement diagram for the connecting rod.

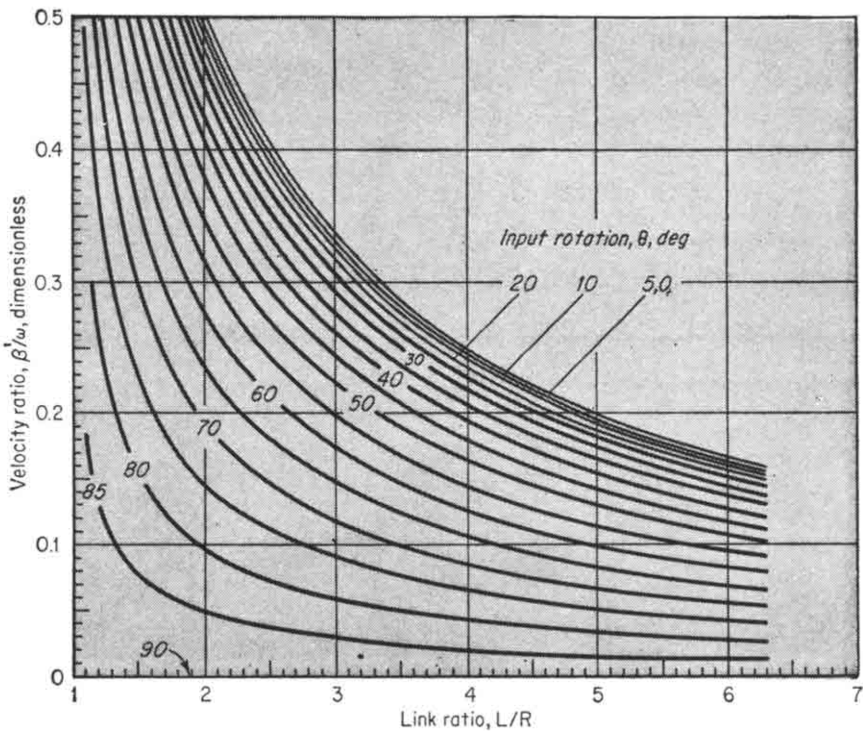


Fig. 3 Angular velocity curves for various crank angles.

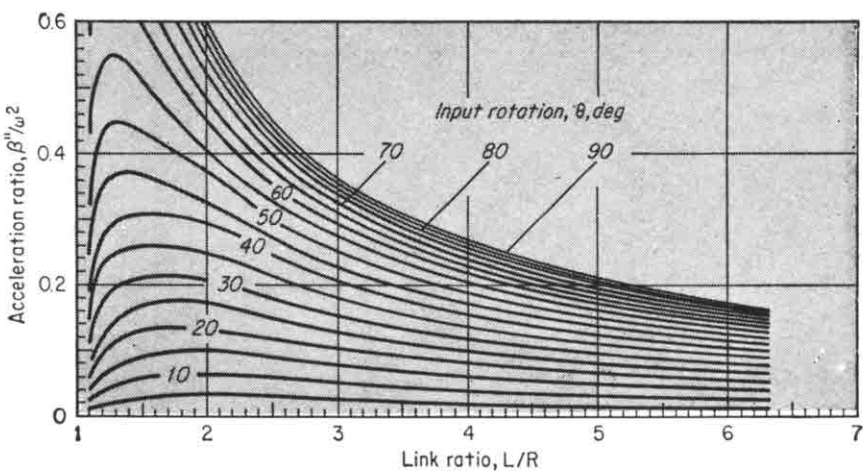


Fig. 4 Angular acceleration curves for various crank angles.

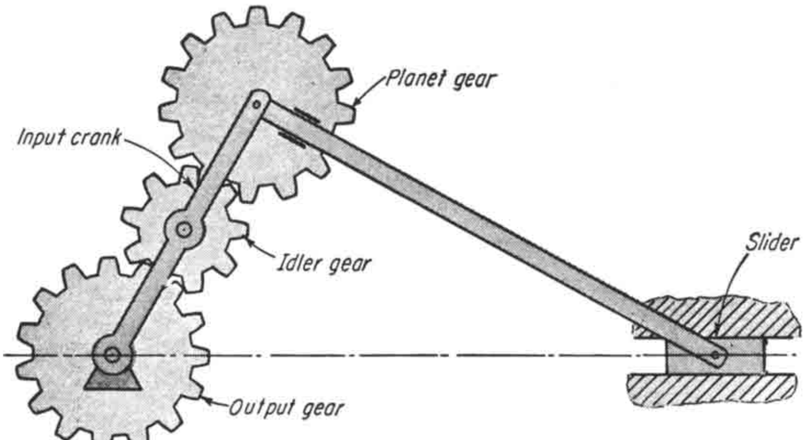


Fig. 5 Modified gear-slider mechanism.

the output velocity at crank angle $\theta = 60°$ can be computed as follows:

$$L/R = 12/4 = 3$$

From Fig. 3 $\beta'/\omega = 0.175$

$$\beta' = 0.175(1000)$$
$$= 175 \text{ radians per second}$$

From Eq. 3

$$\gamma = 2960 \text{ radians per second}$$

Three-Gear Variation

One interesting variation, shown in Fig. 5, is obtained by adding idler gear 5 to the drive. If gears 3 and 4 are then made equal in side, output gear 4 will then oscillate with exactly the same motion as connecting rod 2.

One use for this linkage, Jensen said, is in machinery where a sleeve is to ride concentrically over an input shaft, and yet must oscillate to provide a reciprocating motion. The shaft can drive the sleeve with this mechanism by making the sleeve part of the output gear.

Internal-Gear Variations

By replacing one of the external gears of Fig. 1 with an internal one, two mechanisms are obtained (Figs. 6 and 7) which have wider variable output abilities. But it is the mechanism in Fig. 7 that interested Jensen. This could be proportioned to give either a dwell or a progressive oscillation, that is, one in which the output rotates forward, say 360º, turns back to 30º, moves forward 30º, and then proceeds to repeat the cycle by moving forward again for 360º.

In this mechanism, the crank drives the large ring gear 3 which is fixed to the connecting rod 2. Output is from gear 4. Jensen derived the following equations:

Output Motion

$$\omega_4 = -\left(\frac{L - R - r_4}{L r_4}\right) R\omega_1 \qquad (8)$$

When $r_4 = L - R$, then $\omega_4 = 0$ from Eq. 8, and the mechanism is proportioned to give instantaneous dwell. To obtain a progressive oscillation, r_4 must

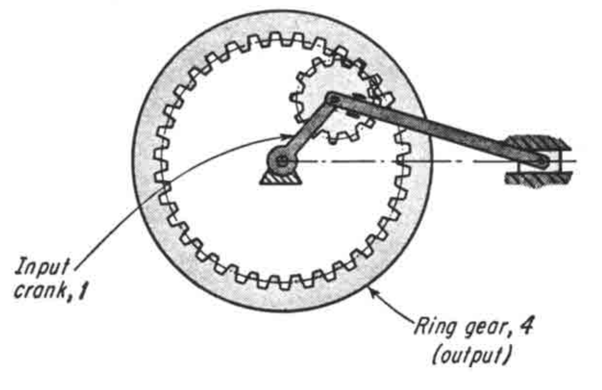


Fig. 6 A ring-gear and slider mechanism. The ring gear is the output and it replaces the center gear in Fig. 1.

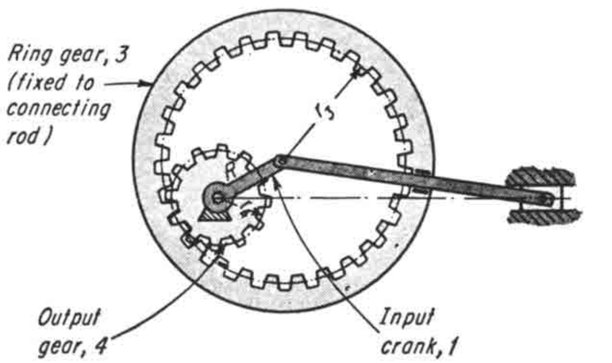


Fig. 7 A more practical ring-gear and slider arrangement. The output is now from the smaller gear.

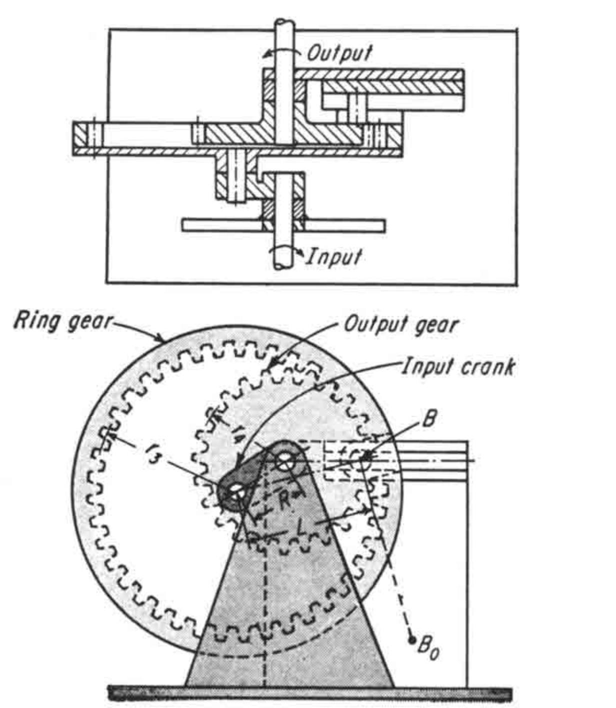


Fig. 8 Jensen's model of the ring-gear and slider mechanism shown in Fig. 7. A progressive oscillation motion is obtained by making r_4 greater than L-R.

be greater than $L - R$, as shown in Jensen's model (Fig. 8).

If gear 4 turns back and then starts moving forward again, there must be two positions where the motion of gear 4 is zero. Those two mechanisms are symmetrical with respect to A_0B. If θ_0 equals the crank-angle rotation (of input), during which the output gear reverses its motion, and ϕ equals the angle through which gear 4 rotates back, then

$$\cos\frac{\theta_0}{2} = \left[\frac{L^2 - R^2}{r_4(2R + r_4)}\right]^{1/2} \tag{9}$$

and

$$\gamma = \theta_0 - \frac{r_3}{r_4}(\theta_0 - \beta_0) \tag{10}$$

where

$$\sin\beta_0 = \frac{R}{L}\sin\frac{\theta_0}{2} \tag{11}$$

Chart for Proportioning

The chart in Fig. 9 helps proportion the mechanism of Fig. 8 to provide a specific kind of progressive oscillation. It is set up for R equals 1 in. For other values of R, convert the chart values for r_4 proportionally, as shown below.

For example, assume that the output gear, during each cycle, is to rotate back 9.2°. Thus ϕ = 9.2°. Also given is R = 0.75 in. and L = 1.5 in. Thus L/R = 2.

From the right side of the chart, go to the ϕ-curve for L = 2, then upward to the θ_0-curve for L = 2 in. Read θ_0 = 82° at the left ordinate.

Now return to the second intersection point and proceed upward to read on the abscissa scale for L = 2, a value of r_4 = 1.5. Since R = 0.75 in., and the chart is for R 1, convert r_4 as follows: r_4 = 0.75 (1.5) = 1.13 in.

Thus, if the mechanism is built with an output gear of radius r_4 = 1.13 in., then during 82° rotation of the crank, the output gear 4 will go back 9.2°. Of course, during the next 83°, gear 4 will have reversed back to its initial position—and then will keep going forward for the remaining 194° of the crank rotation.

Future Modifications

The mechanism in Fig. 8 is designed to permit changing the output motion easily from progressive oscillation to instantaneous dwell or nonuniform CW or CCW rotation. This is accomplished by shifting the position of the pin which acts as the sliding piece of the centric slider crank. It is also possible to use an eccentric slider crank, a four-bar linkage, or a sliding-block linkage as the basic mechanism.

Two mechanisms in series will give an output with either a prolonged dwell or two separate dwells. The angle between the separated dwells can be adjusted during its operation by interposing a gear differential so that the position of the output shaft of the first mechanism can be changed relative to the position of the input shaft of the second mechanism.

The mechanism can also be improved by introducing an additional link, B-B_0, to guide pin B along a circular arc instead of a linear track. This would result in a slight improvement in the performance of the mechanism.

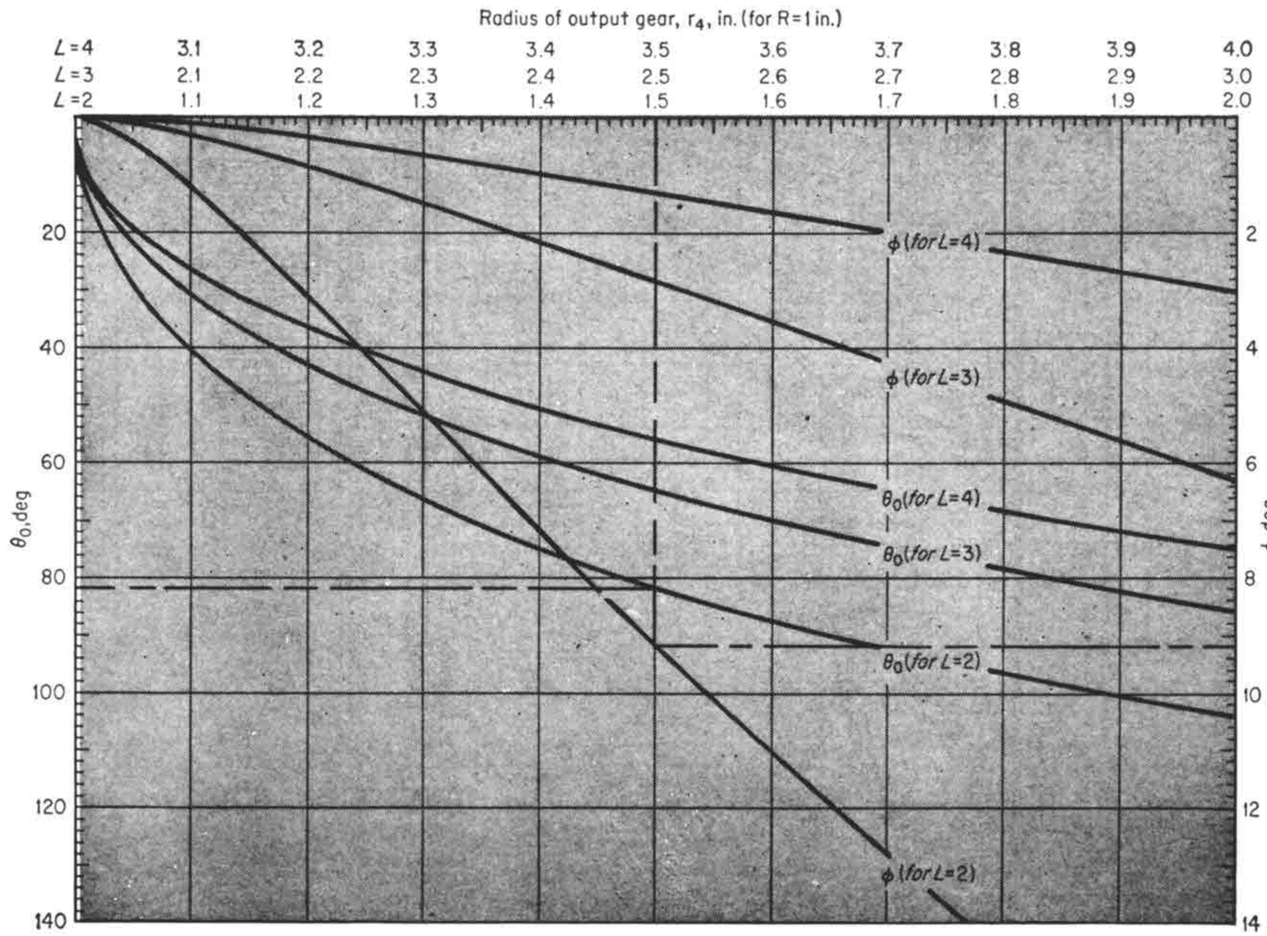


Fig. 9 A chart for proportioning a ring-gear and slider mechanism.

DESIGNING SNAP-ACTION TOGGLES

Theory, formulas, and design charts are presented for determining toggle dimensions to maximize snap-action.

Over-centering toggle mechanisms, as shown in Fig. 1, are widely used in mechanical and electrical switches, latch mechanisms and mechanical overload controls. These toggles also serve as: (1) detents (for holding other parts in selected position); (2) overload devices in mechanical linkages (they shift to the opposite position when sufficiently loaded); and (3) energy-storage devices.

Two applications, shown in Fig. 2, illustrate the snap-action of a toggle. As the toggle passes dead center, it is snapped ahead of the actuating force by the toggle spring. In most applications, the objective is to obtain maximum snap-action.

Snap-action is a function of the elongation per length of the toggle spring as it moves over dead center. Elongation at dead center is equal to:

$$J = K - H \tag{1}$$

The elongation e in percent of length is equal to:

$$e = (100)J/S \tag{2}$$

Because the resisting force of the spring increases with elongation but decreases with an increase in length, the ratio J/S should be as large as possible within the capacity of the spring for the best snap-action performance.

The ratio J/S as a function of angle θ can be derived as follows:

$$H = S - S\cos\varphi \tag{3}$$

and

$$K = A - A\cos\theta \tag{4}$$

Substituting Eqs. (3) and (4) into Eq. (1),

$$J = A(1-\cos\theta) - S(1-\cos\varphi) \tag{5}$$

or

$$J/S = (A/S)(1-\cos\theta) - (1-\cos\varphi) \tag{6}$$

The relationship between θ and ϕ is:

$$L = A\sin\theta = S\sin\varphi$$

or

$$\sin\varphi = (A/S)(\sin\theta) \tag{7}$$

By trigonometric identity,

$$\sin\theta = (1-\cos^2\theta)^{1/2} \tag{8}$$

Substituting Eq. (8) into Eq. (7) and squaring both sides,

$$\sin\varphi = (A/S)^2(1-\cos^2\theta) \tag{9}$$

By trigonometric identity,

$$\cos\varphi = (1-\sin^2\varphi)^{1/2} \tag{10}$$

Substituting Eq. (9) into Eq. (10),

$$\cos\varphi = [1-(A/S)^2 + (A/S)^2\cos^2\theta]^{1/2} \tag{11}$$

and Eq. (11) into Eq. (6),

$$J/S = (A/S)(1-\cos\theta) - 1 + [1-(A/S)(1-\cos\theta) - 1 - (A/S)^2\cos^{1/2} \tag{12}$$

Eq. (12) can be considered to have only three variables: (1) the spring elongation ratio J/S; (2) the toggle arm to spring length ratio, A/S; and (3) the toggle arm angle θ.

A series of curves are plotted from Eq. (12) showing the relationship between J/S and A/S for various angles of θ. The curves are illustrated in Fig. 3; for greater accuracy each chart has a different vertical scale.

Maximum Snap-Action

Maximum snap-action for a particular angle occurs when J/S is a maximum. This can be determined by setting the first derivative of Eq. (12) equal to zero and solving for A/S.

Differentiating Eq. (12),

$$\frac{d(J/S)}{d(A/S)} = 1-\cos\theta + \frac{[-2(A/S)+2(A/S)(\cos^2\theta)]}{2[1-(A/S)^2+(A/S)^2\cos^2\theta]^{1/2}} \tag{13}$$

Setting Eq. (13) Equal to zero and rearranging terms,

$$\frac{\cos\theta-1}{\cos^2\theta-1} = \frac{A/S}{(A/S)[(S/A)^2-1+\cos^2\theta]^{1/2}} \tag{14}$$

Cross-multiplying, squaring, and simplifying,

$$(S/A)^2 - 1 + \cos^2\theta = \cos^2\theta + 2\cos\theta + 1$$

Reducing,

$$(S/A)^2 = 2\cos\theta + 2$$

and finally simplifying to the following equation is A/S when J/S is a maximum:

$$A/S = [2(\cos\theta + 1)]^{-1/2} \tag{15}$$

The maximum value of J/S can be determined by substituting Eq. (15) into Eq. (12):

$$J/S_{\max} = \frac{1-\cos\theta}{[2(\cos\theta+1)]^{1/2}} - 1 + \left[1 - \frac{1}{2(\cos\theta+1)} + \frac{\cos^2\theta}{2(\cos\theta+1)}\right]^{1/2} \tag{16}$$

which is simplified into the following expression:

$$J/S_{\max} = \frac{2-[2(\cos\theta+1)]^{1/2}}{[2(\cos\theta+1)]^{1/2}} \tag{17}$$

The locus of points of J/S_{max} is a straight line function as shown in Fig. 3. It can be seen from Eq. (15) that the value of A/S at J/S_{max} varies from 0.500 when $\theta = 0$ to 0.707 when $\theta = 90°$. This relatively small range gives a quick rule-of-thumb to check if a mechanism has been designed close to the maximum snap-action point.

Elongation of the spring, Eq. (2), is based on the assumption that the spring is installed in its free length S with no initial elongation. For a spring with a free length E smaller than S, the total elongation in percent when extended to the dead center position is:

$$e = 100[(S/E)(1 + J/S) - 1] \tag{18}$$

The relationship between ϕ and θ at the point of maximum snap-action for any value of θ is:

$$\theta = 2\varphi \tag{19}$$

This can be proved by substituting Eqs. (9) and (11) into the trigonometric identity:

$$\cos 2\varphi = \cos^2\varphi - \sin^2\varphi \tag{20}$$

and comparing the resulting equation with one obtained by solving for $\cos\theta$ in Eq. (15). This relationship between the angles is another way to evaluate a toggle mechanism quickly.

Design Procedure

A toggle is usually designed to operate within certain space limitations. When the dimensions X and W, as shown in Fig. 4, are known, the angle θ resulting in maximum snap-action can be determined as follows:

$$A\sin\theta = S\sin\varphi = W/2 \tag{21}$$

Substituting Eq. (19) into Eq. (21),

$$A\sin\theta = S\sin(\theta/2) = W/2 \tag{22}$$

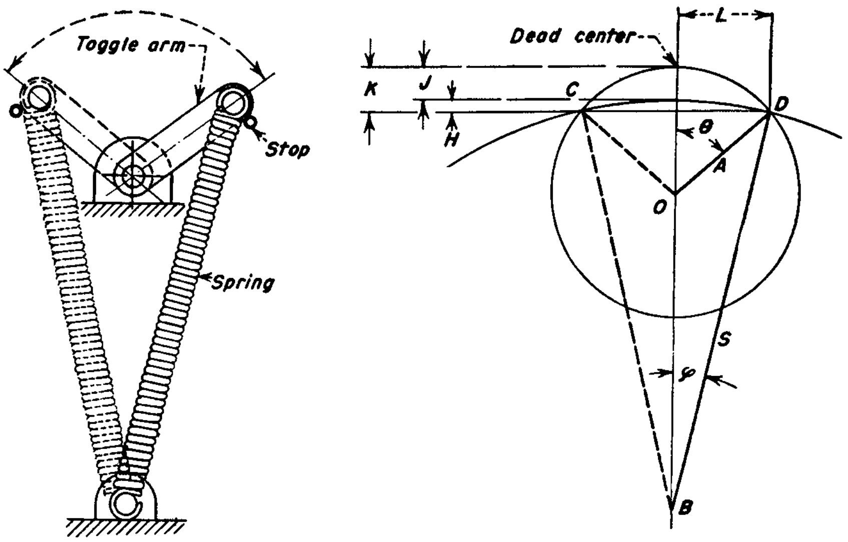


Fig. 1 The design analysis for a snap-action or over-centering toggle employing a link and spring. The mechanical view is shown at left, and the kinematic representation is at right.

Symbols

A = length of toggle arm

S = free length of toggle spring in detented positions

θ = angle swept by toggle arm moving from detented to dead center position

φ = angle swept by toggle spring in moving from detented to dead center position

CD = chordal distance between detent points

L = chordal distance between detent point and dead center

K = height of arc swept by toggle arm

O = pivot point of toggle arm

B = pivot point of toggle spring

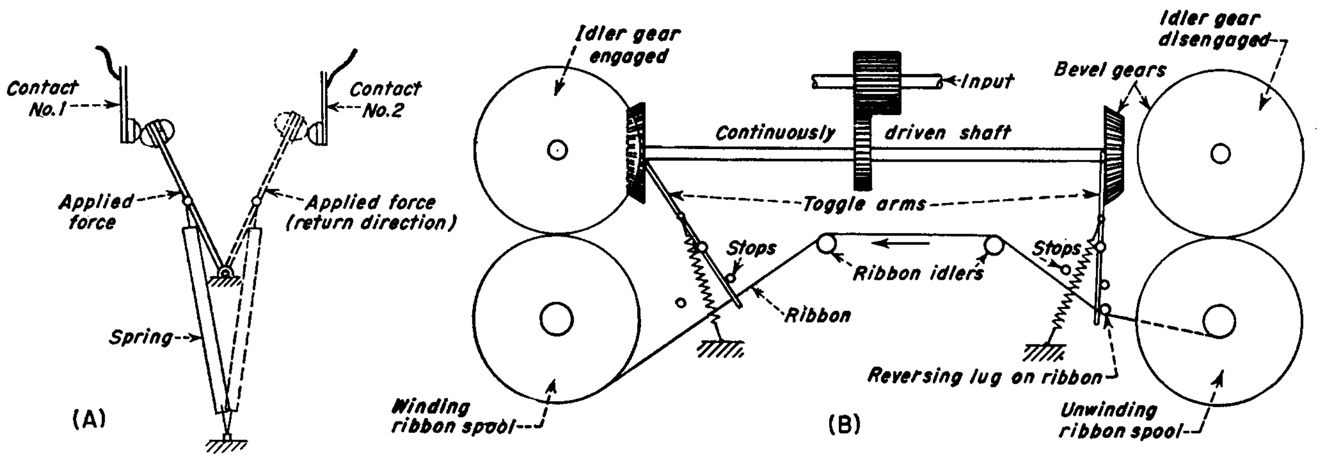


Fig. 2 Typical applications of toggles: (A) snap-action switches; (B) ribbon reversing mechanical for typewriters and calculators. The toggle in (B) is activated by a lug on the ribbon. As it passes dead center it is snapped ahead of the lug by the toggle spring, thus shifting the shaft and reversing the direction of the ribbon before next key is struck.

From Fig. 4:

$$X = S\cos(\theta/2) + A - A\cos\theta \quad (23)$$

Substituting Eq. (22) into Eq. (23),

$$X = \frac{W\cos(\theta/2)}{2\sin(\theta/2)} + \frac{W}{2\sin\theta} - \frac{W\cos\theta}{2\sin\theta} \quad (24)$$

Converting to half-angle functions and simplifying,

$$X = W/[2\sin(\theta/2)\cos(\theta/2)] \quad (25)$$

Using the trigonometric identity,

$$\sin\theta = 2\sin(\theta/2)\cos(\theta/2) \quad (26)$$

Eq. (25) becomes:

$$X = W/\sin\theta$$

or

$$\sin\theta = W/X \quad (27)$$

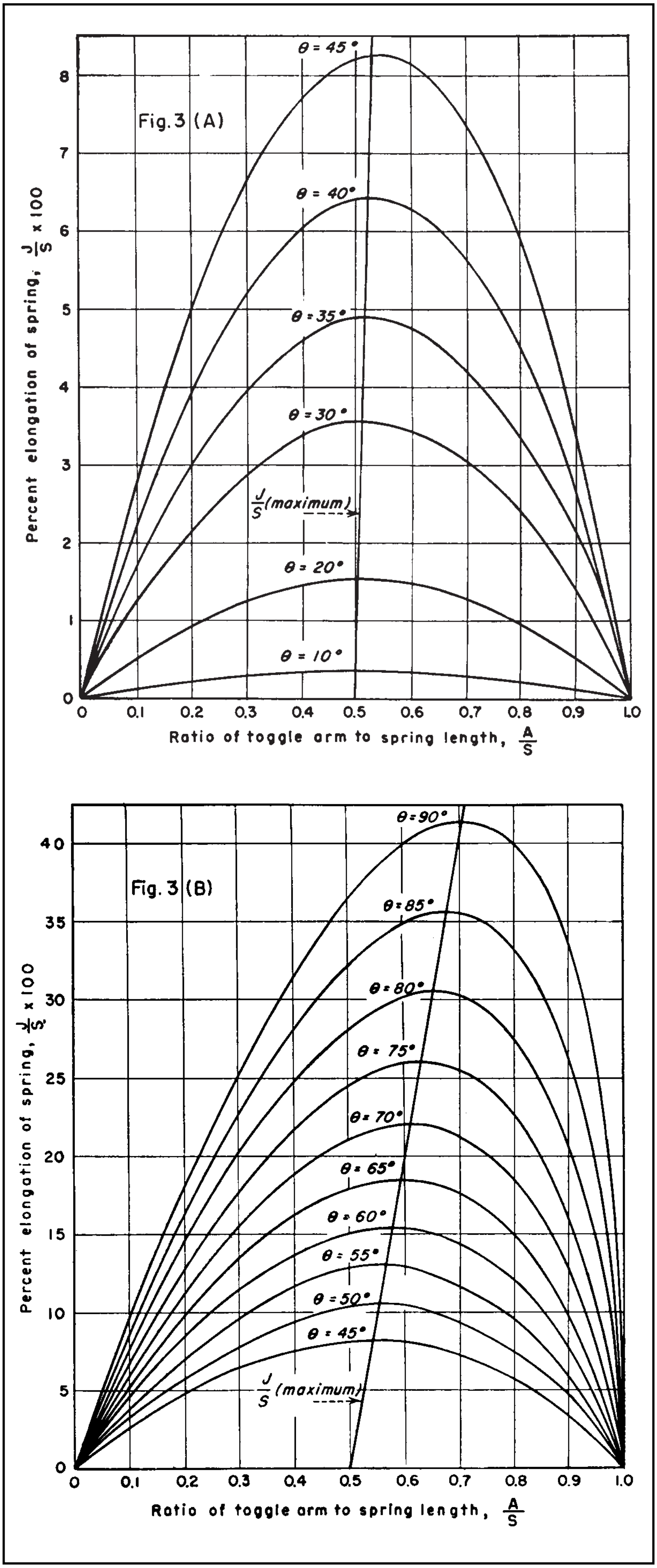
Solving for θ permits the ratios A/S and J/S to be determined from the charts in Fig. 3, when using the J/S (max) line. The values of S and A can then be obtained from Eq. (22).

It can be seen from Fig. 3 that $\theta = 90°$ results in maximum snap-action. Substitution of the sin of 90° in Eq. (27) results in $W = X$; in other words, the most efficient space configuration for a toggle is a square.

EDITOR'S NOTE—In addition to the toggles discussed here, the term "toggle" is applied to a mechanism containing two links that line up in a straight line at one point in their motion, giving a high mechanical advantage.

Toggles of this kind offer: (1) a high mechanical advantage, (2) a high velocity ratio, and (3) a variable mechanical advantage.

Fig. 3 Design charts for evaluating toggle arm and spring length for maximum spring elongation. Chart (B) is an extension of chart (A).



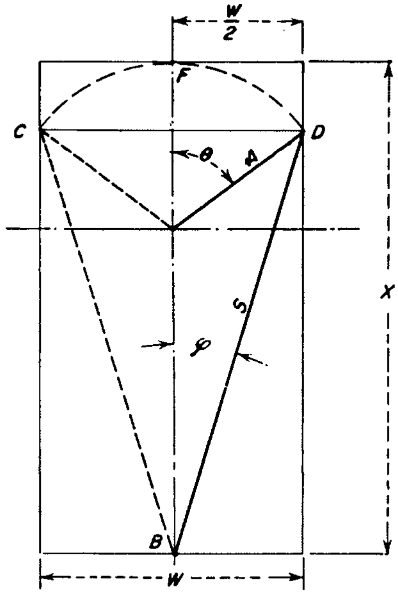


Fig. 4 Designing a toggle to lie within space boundaries *W* and *X*. It can be shown that for maximum snap-action, sin θ = *X/W*.

FEEDER MECHANISMS FOR ANGULAR MOTIONS

How to use four-bar linkages to generate continuous or intermittent angular motions required by feeder mechanisms

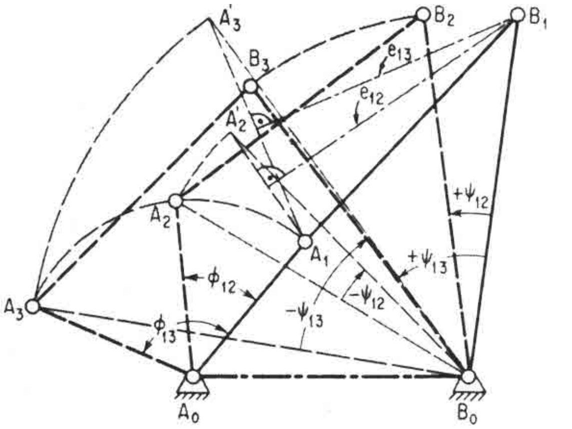


Fig. 1 Four-bar linkage synchronizes two angular movements, ϕ_{12} and ϕ_{13}, with ψ_{12}, and ψ_{14}.

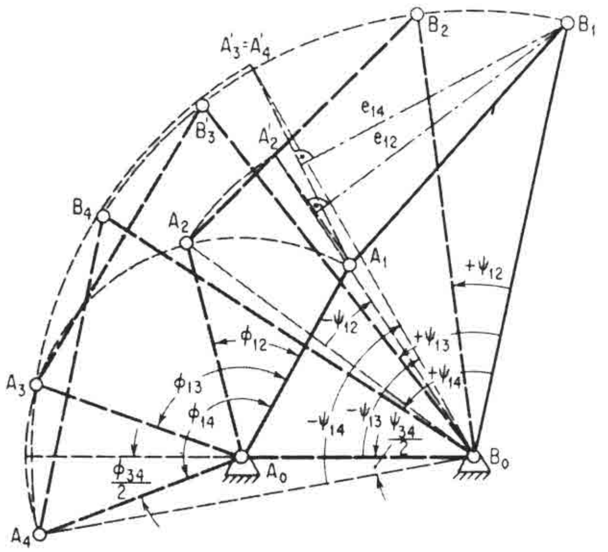


Fig. 2 Three angular positions, ϕ_{12}, ϕ_{13}, ϕ_{14}, are synchronized by four-bar linkage here with ψ_{12}, ψ_{13}, and ψ_{14}.

In putting feeder mechanisms to work, it is often necessary to synchronize two sets of angular motions. A four-bar linkage offers one way. For example, in Fig. 1 two angular motions, ϕ_{12} and ϕ_{13}, must be synchronized with two others, ψ_{12} and ψ_{13}, about the given pivot points A_0 and B_0 and the given crank length A_0A. This means that crank length B_0B. must be long enough so that the resulting four-bar linkage will coordinate angular motions ϕ_{12} and ϕ_{13} with ψ_{12} and ψ_{13}. The procedure is:

1. Obtain point A'_2 by revolving A_2 about B_0 through angle $-\psi_{12}$ but in the opposite direction.
2. Obtain point A'_3 similarly by revolving A_3 about B_0 through angle $-\psi_{13}$.
3. Draw lines $A_1A'_2$ and $A_1A'_3$ and the perpendicular bisectors of the lines which intersect at desired point B_1.
4. The quadrilateral $A_0A_1B_1B_0$ represents the four-bar linkage that will produce the required relationship between the angles ϕ_{12}, ϕ_{13}, and ψ_{12}, ψ_{13}.

Three angles with four relative positions can be synchronized in a similar way. Figure 2 shows how to synchronize angles ψ_{12}, ψ_{13}, ψ_{14} with corresponding angles ψ_{12}, ψ_{13}, and ψ_{14}, using freely chosen pivot points A_0 and B_0. In this case, crank length A_0A as well as B_0B is to be determined, and the procedure is:

1. Locate pivot points A_0 and B_0 on a line those bisects angle $A_3A_0A_4$, the length A_0B_0 being arbitrary.
2. Measure off ½ of angle $B_3B_0B_4$ and with this angle draw B_0A_4 which establishes crank length A_0A at intersection of A_0A_4. This also establishes points A_3, A_2 and A_1.
3. WithB_0 as center and B_0B_4 as radius mark off angles $-\psi_{14}$, $-\psi_{13}$, $-\psi_{12}$, the negative sign indicating they are in opposite sense to ψ_{14}, ψ_{13} and ψ_{12}. This establishes points A'_2, A'_3 and A'_4, but here A'_3 and A'_4 coincide because of symmetry of A_3 and A_4 about A_0B_0.
4. Draw lines $A_1A'_2$ and $A_1A'_4$, and the perpendicular bisectors of these lines, which intersect at the desired point B_1.
5. The quadrilateral $A_0A_1B_0B_1$ represents the four-bar linkage that will produce the required relationship between the angles ϕ_{12}, ϕ_{13}, ϕ_{14}, and ψ_{12}, ψ_{13}, ψ_{14}.

The illustrations show how these angles must be coordinated within the given space. In Fig. 3A, input angles of the crank must be coordinated with the output angles of the forked escapement. In Fig. 3B, input angles of the crank are coordinated with the output angles of the tilting hopper. In Fig. 3C, the input angles of the crank are coordinated with the output angles of the segment. In Fig. 3D, a box on a conveyor is tilted 90º by an output crank, which is actuated by an input crank through a coupler. Other mechanisms shown can also coordinate the input and output angles; some have dwell periods between the cycles, others give a linear output with dwell periods.

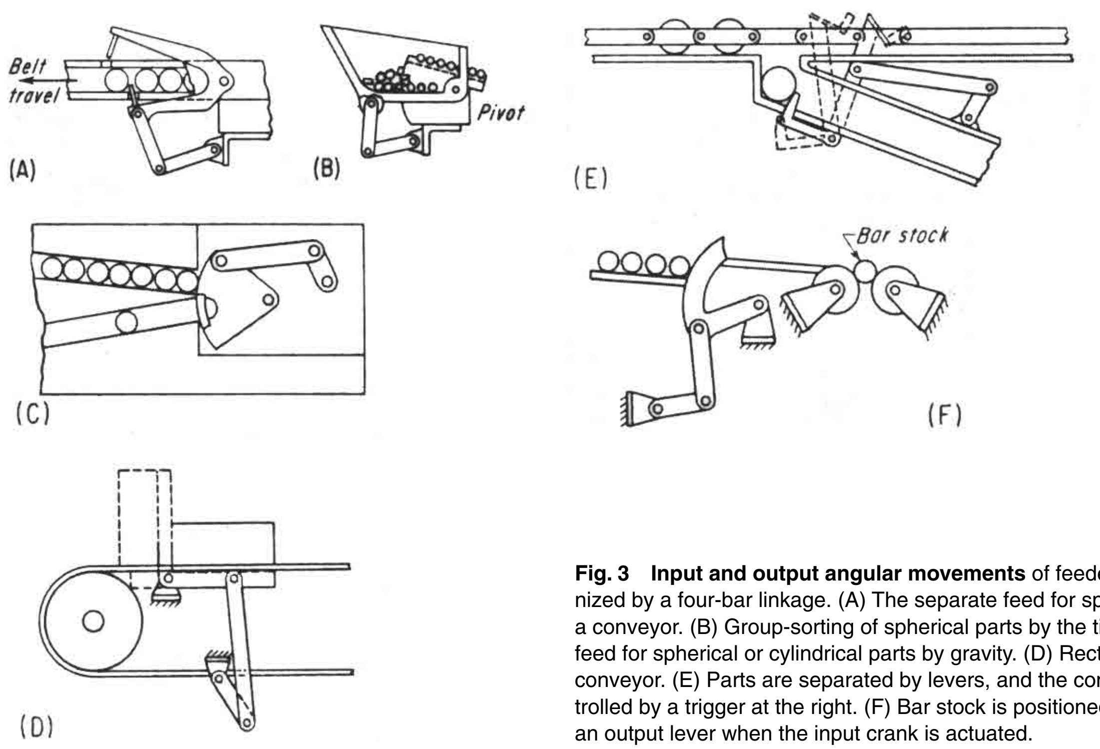


Fig. 3 Input and output angular movements of feeder mechanisms are synchronized by a four-bar linkage. (A) The separate feed for spherical or cylindrical parts on a conveyor. (B) Group-sorting of spherical parts by the tilting hopper. (C) A separate feed for spherical or cylindrical parts by gravity. (D) Rectangular parts are turned on a conveyor. (E) Parts are separated by levers, and the conveyor movement is controlled by a trigger at the right. (F) Bar stock is positioned by the angular oscillation of an output lever when the input crank is actuated.

FEEDER MECHANISMS FOR CURVILINEAR MOTIONS

Four-bar linkages can be combined into six, eight, or more linkages for the feeder mechanisms in cameras, automatic lathes, farm machinery, and torch-cutting machines.

When feeder mechanisms require complex curvilinear motions, it might be necessary to use compound linkages rather than four links. However, four-bar linkages can be synthesized to produce curvilinear motions of various degrees of complexity, and all possibilities for four-bar linkages should be considered before selecting more complex linkages.

For example, a camera film-advancing mechanism, Fig. 1, has a simple four-bar linage with a coupler point d, which generates a curvilinear and straight-line motion a resembling a D. Another more complex curvilinear motion, Fig. 2, is also generated by a coupler point E of a four-bar linkage, which controls an automatic profile cutter. Four-bar linkages can generate many different curvilinear motions, as in Fig. 3. Here the points of the coupler prongs, g_1, g_2, and g_3 on coupler b, and g_4 and g_5 on coupler e, are chosen so that their motions result in the desired progressive feeding of straw into a press.

A similar feeding and elevating device is shown in Fig. 4. The rotating device crank a moves coupler b and swinging lever c, which actuates the guiding arm f through the link e. The bar h carries the prone fingers g_1 through g_7. They generate coupler curves a_1 through a_7.

As another practical example, consider the torch-cutting machine in Fig. 5A designed to cut sheet metal along a curvilinear path a. Here the points A_0 and B_0 are fixed in the machine, and the lever A_0A_1 has an adjustable length to suit the different curvilinear paths a desired.

The length B_1B_1 is also fixed. The challenge is to find the length of the levers A_1B_1 and E_1B_1 in the four-bar linkage to give the desired path a, which is to be traced by the coupler point E on which the cutting torch is mounted.

The graphical solution for this problem, as shown in Fig. 5B, requires the selection of the points A_1 and E_4 so that the distances A_1E_1 to A_8E_8 are equal and the points E_1 to E_8 lie on the desired coupler curved a. In this case, only the points E_4 to E_8 represent the desired profile to be cut. The correct selection of points A_1 and E_1 depends upon making the following triangles congruent:

$$\Delta\, E_2A_2B_{01} = \Delta\, E_1A_1B_{02}$$
$$\Delta\, E_3A_3B_{01} = \Delta\, E_1A_1B_{03}$$
$$\Delta\, E_8A_8B_{01} = \Delta\, E_1A_1B_{08}$$

and so on until $E_8A_8B_{01} = E_1A_1B_{08}$. At the same time, all points A_1 to A_8 must lie on the arc having B_1 as center.

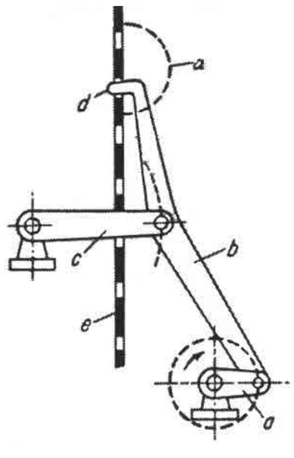


Fig. 1

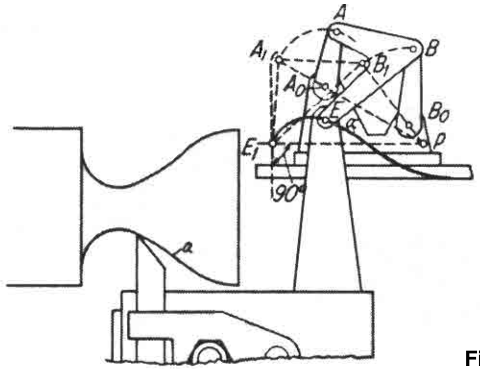


Fig. 2

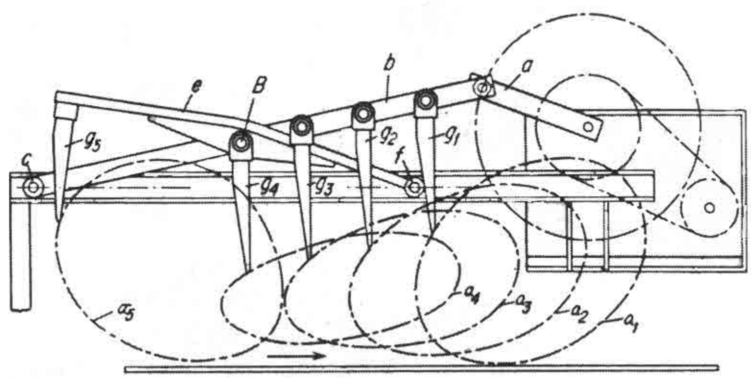
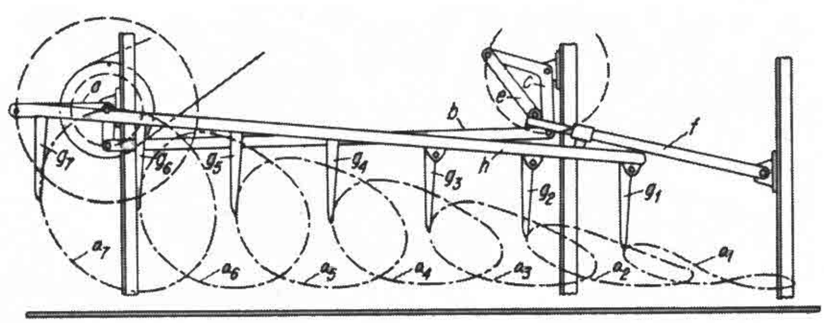


Fig. 3

Fig. 4



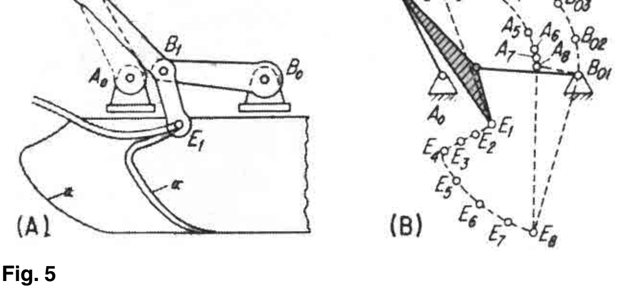


Fig. 5

Synthesis of an Eight-Bar Linkage

Design a linkage with eight precision points, as shown in Fig. 6. In this mechanism the curvilinear motion of one four-bar linkage is coordinated with the angular oscillation of a second four-bar linkage. The first four-bar linkage consists of AA_0BB_0 with coupler point E which generates γ with eight precision points E_1 through E_8 and drives a second four-bar linkage HH_0GG_0. Coupler point F generates curve δ with precision points F_1 through F_8. The coupler points F_2, F_4, F_6, F_8 are coincident because straight links GG_0 and GH are in line with one another in these coupler positions. This is what permits HH_0 to oscillate, despite the continuous motion of the coupler point F. The coupler points F_1 coincident with F_5, and F_3 coincident with F_7, have been chosen so that F_1 is the center of a circle k_1 and F_3 is the center of a circle k_3. These circles are tangent to coupler curve γ at E_1, and E_5, E_3, and E_7, and they indicate the limiting positions of the second four-bar linkage HH_0GG_0.

The limiting angular oscillation of HH_0, which is one of the requirements of this mechanism, is represented by positions H_0H_1 and H_0H_3. It oscillates four times for each revolution of the input crank AA_0, and the positions H_1 to H_8 correspond to input crank positions A_1 to A_8.

The synthesis of a compound linkage with dwell periods and coordinated intermittent motion is shown in Fig. 7. The four-bar linkage AA_0BB_0 generates an approximately triangular curve with coupler point E, which has six precision points E_1 through E_6. A linkage that will do this is not unusual and can be readily proportioned from known methods of four-bar linkage synthesis. However, the linkage incorporates dwell periods that produce coordinated intermittent motion with a second four-bar linkage FF_0HB_0. Here the tangent arcs k_{12}, k_{34} and k_{56} are drawn with EF as the radius from centers F_{12}, F_{34} and F_{56}.

These centers establish the circle with F_0 as the center and pivot point for the second four-bar linkage. Each tangent arc causes a dwell of the link FF_0, while AA_0 rotates continuously. Thus, the link FF_0, with three rest periods in one revolution, can produce intermittent curvilinear motion in the second four-bar linkage FF_0HB_0. In laying out the center, F_0 must be selected so that the angle EFF_0 deviates only slightly from 90º because this will minimize the required torque that is to be applied at E. The length of B_0H can be customized, and the rest periods at H_{34}, H_{12} and H_{56} will correspond to the crank angles ϕ_{34}, ϕ_{12} and ϕ_{56}.

A compound linkage can also produce a 360º oscillating motion with a dwell period, as in Fig. 8. The two four-bar linkages are AA_0BB_0 and BB_0FF_0, and the output coupler curve γ is traversed only through segment E_1, E_2. The oscillating motion is produced by lever HH_0, connected to the coupler point by EH. The fixed point H_0 is located within the loop of the coupler curve γ. The dwell occurs at point H_3, which is the center of circular arc k tangent to the coupler curve γ during the desired dwell period. In this example, the dwell is made to occur in the middle of the 360º oscillation. The coincident positions H_1 and H_2 indicate the limiting positions of the link HH_0, and they correspond to the positions E_1 and E_2 of the coupler point.

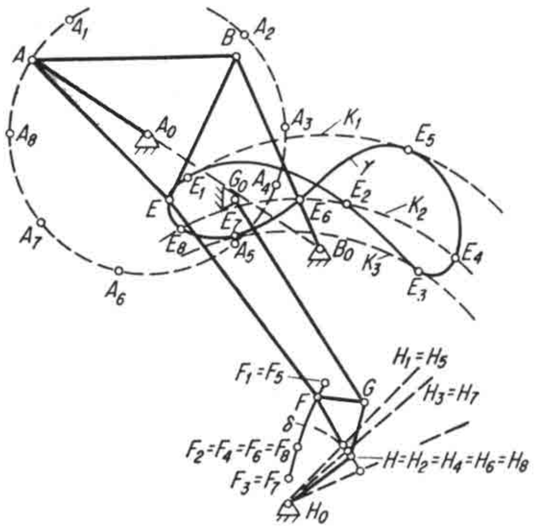


Fig. 6

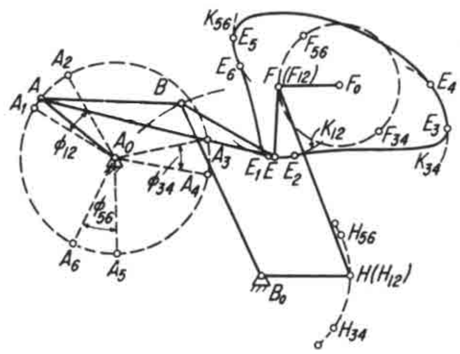


Fig. 7

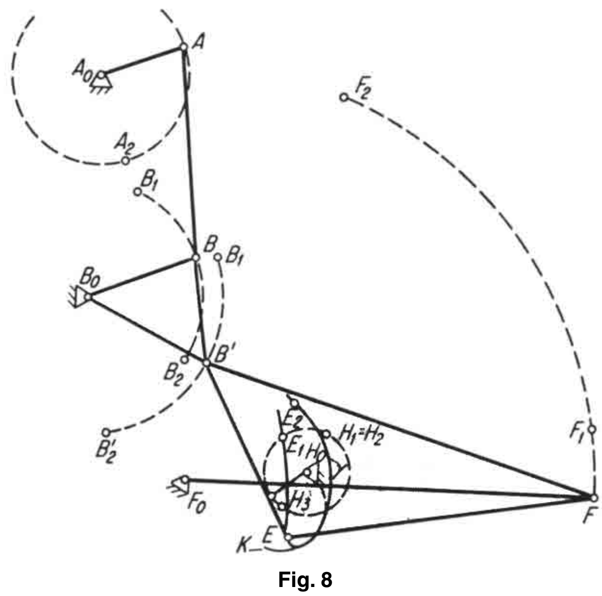


Fig. 8

ROBERTS' LAW HELPS TO FIND ALTERNATE FOUR-BAR LINKAGES

The three linkage examples

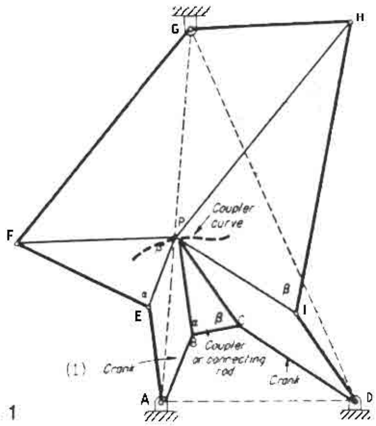


Fig. 1

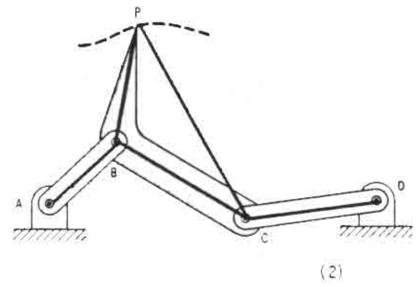


Fig. 2

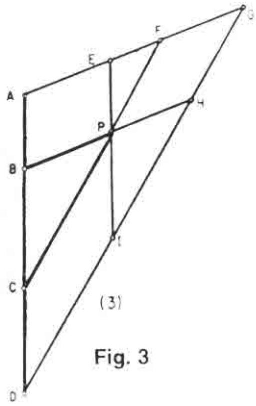


Fig. 3

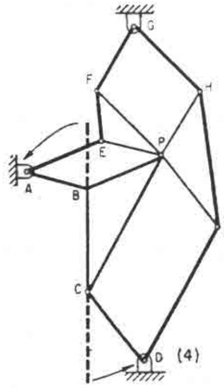


Fig. 4

Fig. 3

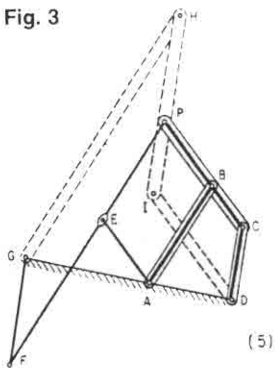


Fig. 1 The method of similar triangles.

Fig. 2, 3, 4 A step-by-step method.

Fig. 5 This special case shows the simplicity of applying Roberts' Law.

When a four-bar linkage has been designed or selected from a catalog to produce a desired coupler curve, it is often found that one of the pivot points is inconveniently located or that the transmission angles are not suitable. (A coupler curve is produced by a point on the connecting rod joining the two cranks of the four-bar linkage). According to *Roberts' Law* there are at least two other four-bar linkages that will generate the same coupler curve. One of these linkages might be more suitable for the application.

Robert's Law states that the two alternate linkages are related to the first by a series of similar triangles. This leads to graphical solutions; three examples are shown. The first involves similar triangles, the second is a more convenient step-by-step method, and the third illustrates the solution of a special case where the coupler point lies along the connecting rod.

Method of Similar Triangles

Four-bar linkage *ABCD* in Fig. 1 uses point *P*, which is actually an extension of the connecting rod *BC*, to produce desired curve. Point *E* is found by constructing *EP* parallel to *AB*, and *EA* parallel to *PB*. Then triangle *EFP* is constructed similar to triangle *BPC*. This calls for laying out angles a and β.

Point *H* is found in a similar way, and point *G* is located by drawing *GH* parallel to *FP* and *GF* parallel to *HP*.

The two alternate linkages to *ABCD* are *GFEA* and *GHID*. All use point *P* to produce the desired curve, and given any one of the three, the other two can be determined.

The Step-by-Step Method

With the similar-triangle method just described, slight errors in constructing the proper angles lead to large errors in link dimensions. The construction of angles can be avoided by laying off the link lengths along a straight line.

Thus, linkage *ABCD* in Fig. 2 is laid off as a straight line from *A* to *D*in Fig. 3. Included in the transfers is point *P*. Points *EFGHI* are quickly found by either extending the original lines or constructing parallel lines. Fig. 3, which now has all the correct dimensions of all the links, is placed under a sheet of tracing paper and, with the aid of a compass, links *AB* and *CD* are rotated (see Fig. 4) so that linkage *ABCD* is identical to that in Fig. 2. Links *PEF* and *PHI* are rotated parallel to *AB* and *CD*, respectively. Completion of the parallelogram gives the two alternate linkages, *AEFG* and *GHID*.

Special Case

It is not uncommon for the coupler point *P* to lie on a line through *BC*, as in Fig. 5. Links *EA*, *EP* and *ID* can be found quickly by constructing the appropriate parallel lines. Point *G* is located by using the proportion: $CB:BP = DA:AG$. Points *H* and *F* are then located by drawing lines parallel to *AB* and *CD*.

RATCHET LAYOUT ANALYZED

The ratchet wheel is widely used in machinery, mainly to transmit intermittent motion or to allow shaft rotation in one direction only. Ratchet-wheel teeth can be either on the perimeter of a disk or on the inner edge of a ring.

The pawl, which engages the ratchet teeth, is a beam pivoted at one end; the other end is shaped to fit the ratchet-tooth flank. Usually, a spring or counterweight maintains constant contact between wheel and pawl.

It is desirable, in most designs, to keep the spring force low. It should be just large enough to overcome the separation forces—inertia, weight, and pivot friction. Excess spring force should not be considered for engaging the pawl and holding it against the load.

To ensure that the pawl is automatically pulled in and kept in engagement independently of the spring, a properly drawn tooth flank is necessary.

The requirement for self-engagement is:

$$Pc + M > \mu Pb + P\sqrt{(1+\mu^2)}\mu_1 r_1$$

Neglecting weight and pivot friction:

$$Pc > \mu Pb$$

but $c/b = r/a = \tan\phi$, and because $\tan\phi$ is approximately equal to $\sin\phi$:

$$c/b = r/R$$

Substituting in term (1)

$$rR > \mu$$

For steel on steel, dry, $\mu = 0.15$ Therefore, using

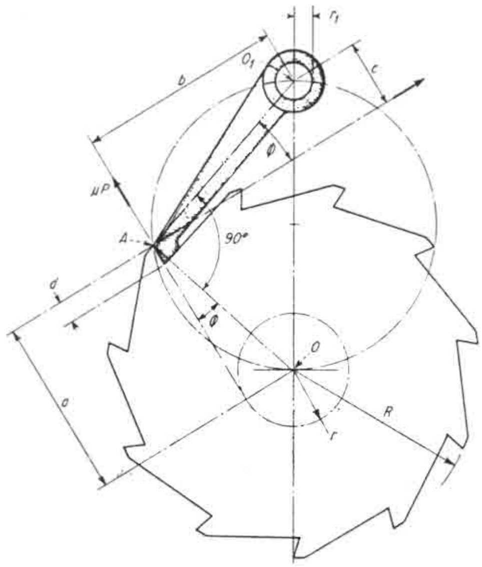
$$r/R = 0.20 \text{ to } 0.25$$

the margin of safety is large; the pawl will slide into engagement easily. For internal teeth with ϕ of 30º, c/b is tan 30º or 0.577, which is larger than μ, and the teeth are therefore self-engaging.

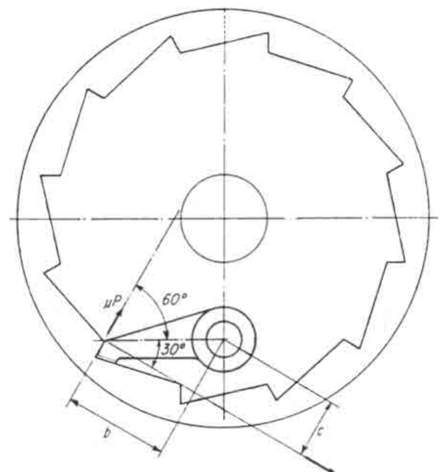
When laying out the ratchet wheel and pawl, locate points O, A and O_1 on the same circle. AO and AO_1 will then be perpendicular to one another; this will ensure that the smallest forces are acting on the system.

Ratchet and pawl dimensions are governed by design sizes and stress. If the tooth, and thus pitch, must be larger than required to be strong enough, a multiple pawl arrangement can be used. The pawls can be arranged so that one of them will engage the ratchet after a rotation of less than the pitch.

A fine feed can be obtained by placing many pawls side by side, with the corresponding ratchet wheels uniformly displaced and interconnected.



Pawl in compression has tooth pressure P and weight of pawl producing a moment that tends to engage pawl. Friction-force μP and pivot friction tend to oppose pawl engagement.



Inner teeth allow compact assembly.

a = moment arm of wheel torque

M = moment about O_1 caused by weight of pawl

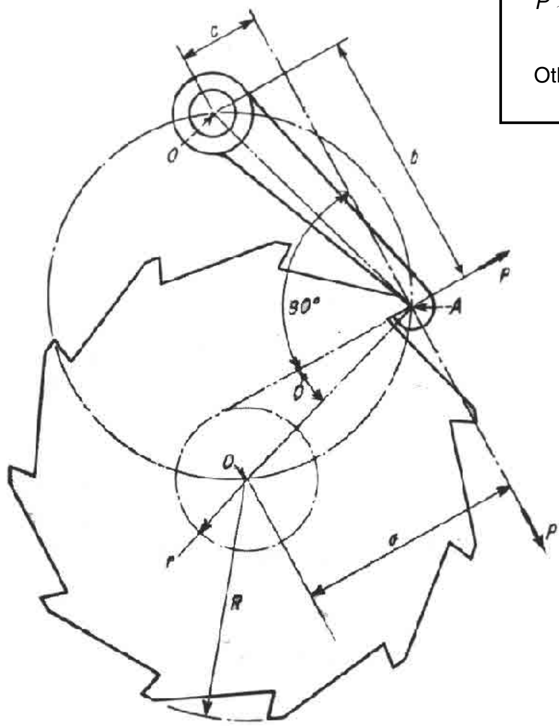
$O_1 - O_2$ = ratchet and pawl pivot centers respectively

P = tooth pressure = wheel torque/a

$P\sqrt{(1+\mu^2)}$ = load on pivot pin

μ, μ_1 = friction coefficients

Other symbols as defined in diagrams.

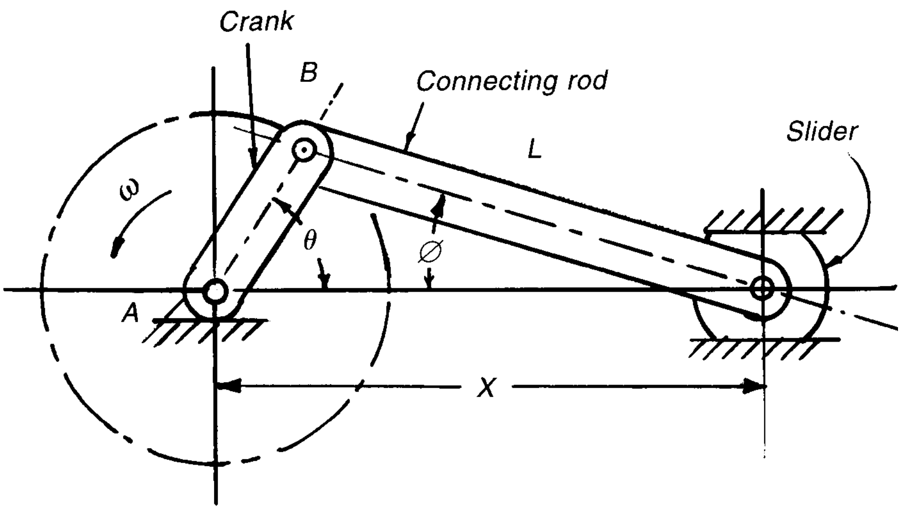


Pawl in tension has the same forces acting on the unit as other arrangements. The same layout principles apply.

SLIDER-CRANK MECHANISM

The slider crank, an efficient mechanism for changing reciprocating motion to rotary motion, is widely used in engines, pumps, automatic machinery, and machine tools.

The equations developed here for finding these factors are in a more simplified form than is generally found in text books.



SYMBOLS

L = length of connecting rod

R = crank length; radius of crank circle

x = distance from center of crankshaft A to wrist pin C

x' = slider velocity (linear velocity of point C)

x'' = slider acceleration

θ = crank angle measured from dead center (when slider is fully extended)

ϕ = angular position of connecting rod; $\phi = 0$ when $\theta = 0$

ϕ' = connecting-rod angular velocity = $d\phi/dt$

ϕ'' = connecting-rod angular acceleration = $d^2\phi/dl^2$

ω = constant crank angle velocity

Displacement of slider

$\boldsymbol{x = L} \cos \phi \boldsymbol{+ R} \cos \phi$

Also:

$$\cos\phi = \left[1 - \left(\frac{R}{L}\right)^2 \sin^2\theta\right]^{1/2}$$

Angular velocity of the connecting rod

$$\phi' = \omega\left[\frac{(R/L)\cos\theta}{[1-(R/L)^2\sin^2\theta]^{1/2}}\right]$$

Linear velocity of the piston

$$\frac{x'}{L} = -\omega\left[1+\frac{\phi'}{\omega}\right]\left(\frac{R}{L}\right)\sin\theta$$

Angular acceleration of the connecting rod

$$\phi'' = \frac{\omega^2(R/L)\sin\theta[(R/L)^2-1]}{[1-(R/L)^2\sin^2\theta]^{3/2}}$$

Slider acceleration

$$\frac{x''}{L} = -\omega^2\left(\frac{R}{L}\right)\left[\cos\theta+\frac{\phi''}{\omega^2}\sin\theta+\frac{\phi'}{\omega}\cos\theta\right]$$

CHAPTER 14
NEW DIRECTIONS IN MACHINE DESIGN

SOFTWARE IMPROVEMENTS EXPAND CAD CAPABILITIES

Computer Aided Design (CAD) is a computer-based technology that allows a designer to draw and label the engineering details of a product or project electronically on a computer screen while relegating drawing reproduction to a printer or X-Y plotter. It also permits designers in different locations to collaborate in the design process via a computer network and permits the drawing to be stored digitally in computer memory for ready reference. CAD has done for engineering graphics what the word processor did for writing. The introduction of CAD in the late 1960s changed the traditional method of drafting forever by relieving the designer of the tedious and time-consuming tasks of manual drawing from scratch, inking, and dimensioning on a conventional drawing board.

While CAD offers many benefits to designers or engineers never before possible, it does not relieve them of the requirement for extensive technical training and wide background knowledge of drawing standards and practice if professional work is to be accomplished. Moreover, in making the transition from the drawing board to the CAD workstation, the designer must spend the time and make the effort to master the complexities of the specific CAD software systems in use, particularly how to make the most effective use of the icons that appear on the screen.

The discovery of the principles of 3D isometric and perspective drawing in the Middle Ages resulted in a more realistic and accurate portrayal of objects than 2D drawings, and they conveyed at a glance more information about that object, but making a 3D drawing manually was then and is still more difficult and time-consuming, calling for a higher level of drawing skill. Another transition is required for the designer moving up from 2D to 3D drawing, contouring, and shading.

The D in CAD stands for design, but CAD in its present state is still essentially "computer-aided drawing" because the user, not the computer, must do the designing. Most commercial CAD programs permit lettering, callouts, and the entry of notes and parts lists, and some even offer the capability for calculating such physical properties as volume, weight, and center of gravity if the drawing meets certain baseline criteria. Meanwhile, CAD software developers are busy adding more automated features to their systems to move them closer to being true design programs and more user-friendly. For example, CAD techniques now available can perform analysis and simulation of the design as well as generate manufacturing instructions. These features are being integrated with the code for modeling the form and structure of the design.

In its early days, CAD required at least the computing power of a minicomputer and the available CAD software was largely application specific and limited in capability. CAD systems were neither practical nor affordable for most design offices and independent consultants. As custom software became more sophisticated and costly, even more powerful workstations were required to support them, raising the cost of entry into CAD even higher. Fortunately, with the rapid increases in the speed and power of microprocessors and memories, desktop personal computers rapidly began to close the gap with workstations even as their prices fell. Before long, high-end PCs become acceptable low-cost CAD platforms. When commercial CAD software producers addressed that market sector with lower-cost but highly effective software packages, their sales surged.

PCs that include high-speed microprocessors, Windows operating systems, and sufficient RAM and hard-drive capacity can now run software that rivals the most advanced custom Unix-based products of a few years ago. Now both 2D and 3D CAD software packages provide professional results when run on off-the-shelf personal computers. The many options available in commercial CAD software include

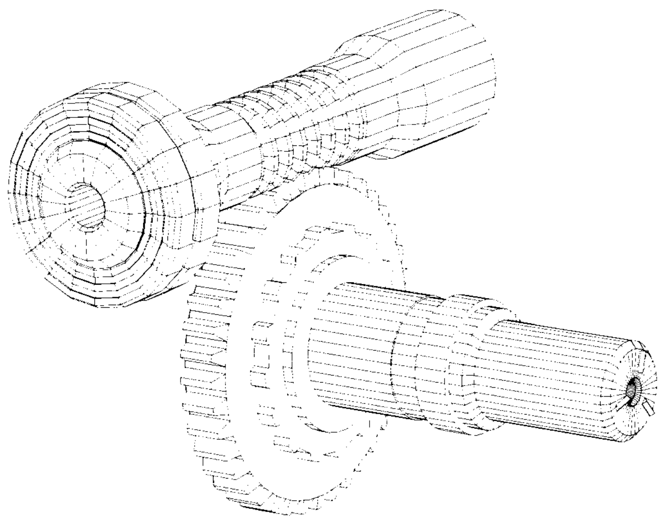
- 2D drafting
- 3D wireframe and surface modeling
- 3D solid modeling
- 3D feature-based solid modeling
- 3D hybrid surface and solid modeling

Two-Dimensional Drafting

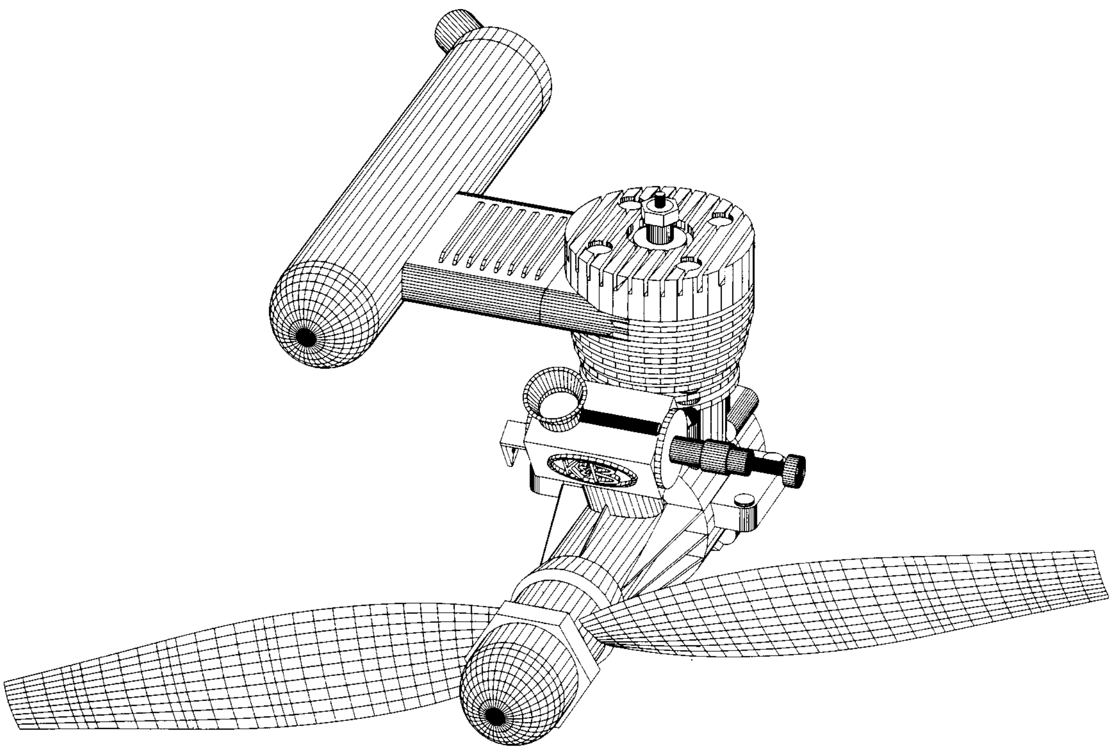
Two-dimensional drafting software for mechanical design is focused on drawing and dimensioning traditional engineering drawings. This CAD software was readily accepted by engineers, designers, and draftspersons with many years of experience. They felt comfortable with it because it automated their customary design changes, provided a way to make design changes quickly, and also permitted them to reuse their CAD data for new layouts.

A typical 2D CAD software package includes a complete library of geometric entities. It can also support curves, splines, and polylines as well as define hatching patterns and place hatching within complex boundaries. Other features include the ability to perform associative hatching and provide complete dimensioning. Some 2D packages can also generate bills of materials. 2D drawing and detailing software packages are based on ANSI, ISO, DIN, and JIS drafting standards.

In a 2D CAD drawing, an object must be described by multiple 2D views, generally three or more, to reveal profile and internal geometry from specific viewpoints. Each view of the object is created independently from other views. However, 2D views typically contain many visible and hidden lines, dimensions, and other detailing features. Unless careful checks of the finished drawing are made, mistakes in drawing or dimensioning intricate details can be overlooked. These can lead to costly problems downstream in the product design cycle. Also, when a change is



A three-dimensional "wireframe" drawing of two meshed gears made on a personal computer using software that cost less than $500. (*Courtesy of American Small Business Computers, Inc.*)



A three-dimensional "wireframe" drawing of a single-drawing model airplane engine showing the principal contours of both propeller and engine. This also was drawn on a personal computer using software that cost less than $500. (*Courtesy of American Small Business Computers, Inc.*)

made, each view must be individually updated. One way to avoid this problem (or lessen the probability that errors will go undetected) is to migrate upward to a 3D CAD system

Three-Dimensional Wireframe and Surface Modeling

A 3D drawing provides more visual impact than a 2D drawing because it portrays the subject more realistically and its value does not depend on the viewer's ability to read and interpret the multiple drawings in a 2D layout. Of more importance to the designer or engineer, the 3D presentation consolidates important information about a design, making it easier and faster to detect design flaws. Typically a 3D CAD model can be created with fewer steps than are required to produce a 2D CAD layout. Moreover, the data generated in producing a 3D model can be used to generate a 2D CAD layout, and this information can be preserved throughout the product design cycle. In addition, 3D models can be created in the orthographic or perspective modes and rotated to any position in 3D space.

The wireframe model, the simplest of the 3D presentations, is useful for most mechanical design work and might be all that is needed for many applications where 3D solid modeling is not required. It is the easiest 3D system to migrate to when making the transition from 2D to 3D drawing. A wireframe model is adequate for illustrating new concepts, and it can also be used to build on existing wireframe designs to create models of working assemblies.

Wireframe models can be quickly edited during the concept phase of the design without having to maintain complex solid-face relationships or parametric constraints. In wireframe modeling only edge information is stored, so data files can be significantly smaller than for other 3D modeling techniques. This can increase productivity and conserve available computer memory.

The unification of multiple 2D views into a single 3D view for modeling a complex machine design with many components permits the data for the entire machine to be stored and managed in a single wireframe file rather than many separate files. Also, model properties such as color, line style, and line width can be controlled independently to make component parts more visually distinctive.

The construction of a wireframe structure is the first step in the preparation of a 3D surface model. Many commercial CAD software packages include surface modeling with wireframe capability. The designer can then use available surface-modeling tools to apply a "skin" over the wire framework to convert it to a surface model whose exterior shape depends on the geometry of the wireframe.

One major advantage of surface modeling is its ability to provide the user with visual feedback. A wireframe model does not readily show any gaps, protrusions, and other defects. By making use of dynamic rotation features as well as shading, the designer is better able to evaluate the model. Accurate 2D views can also be generated from the surface model data for detailing purposes. Surface models can also be used to generate tool paths for numerically controlled (NC) machining. Computer-aided manufacturing (CAM) applications require accurate surface geometry for the manufacture of mechanical products.

Yet another application for surface modeling is its use in the preparation of photorealistic graphics of the end product. This capability is especially valued in consumer product design, where graphics stress the aesthetics of the model rather than its precision.

Some wireframe software also includes data translators, libraries of machine design elements and icons, and 2D drafting and detailing capability, which support design collaboration and compatibility among CAD, CAM, and computer-aided engineering (CAE) applications. Designers and engineers can store and use data accumulated during the design process. This data permits product manufacturers with compatible software to receive 2D and 3D wireframe data from other CAD systems.

Among the features being offered in commercial wireframe software are:

- Basic dimensioning, dual dimensioning, balloon notes, datums, and section lines.
- Automated geometric dimensioning and tolerancing (GD&T).
- Symbol creation, including those for weld and surface finish, with real-time edit or move capability and leaders.
- A library of symbols for sheet metal, welding, electrical piping, fluid power, and flow chart applications.

Data translators provide an effective and efficient means for transferring information from the source CAD design station to outside contract design offices, manufacturing plants, or engineering analysis consultants, job shops, and product development services. These include IGES, DXF, DWG, STL, CADL, and VRML.

Three-Dimensional Solid Modeling

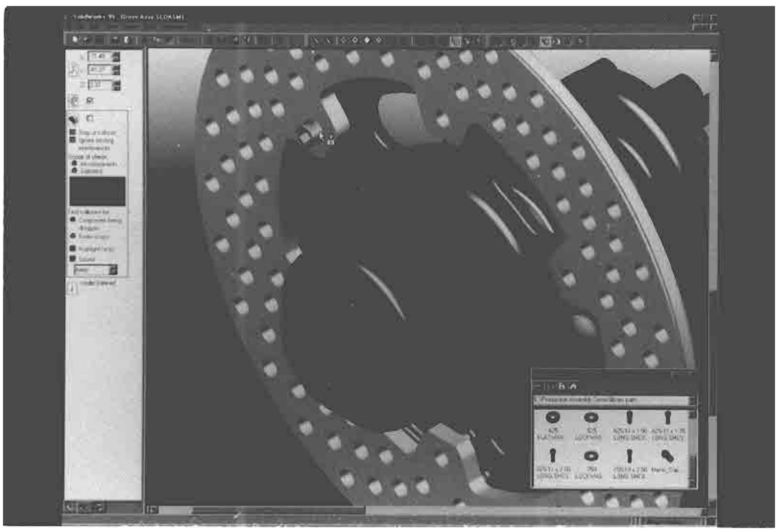
CAD solid-modeling programs can perform many more functions than simple 3D wireframe modelers. These programs are used to form models that are solid objects rather than simple 3D line drawings. Because these models are represented as solids, they are the source of data that permits the physical properties of the parts to be calculated.

Some solid-modeling software packages provide fundamental analysis features. With the assignment of density values for a variety of materials to the solid model, such vital statistics as strength and weight can be determined. Mass properties such as area, volume, moment of inertia, and center of gravity can be calculated for regularly and irregularly shaped parts. Finite element analysis software permits the designer to investigate stress, kinematics, and other factors useful in optimizing a part or component in an assembly. Also, solid models can provide the basic data needed for rapid prototyping using stereolithography, and can be useful in CAM software programs.

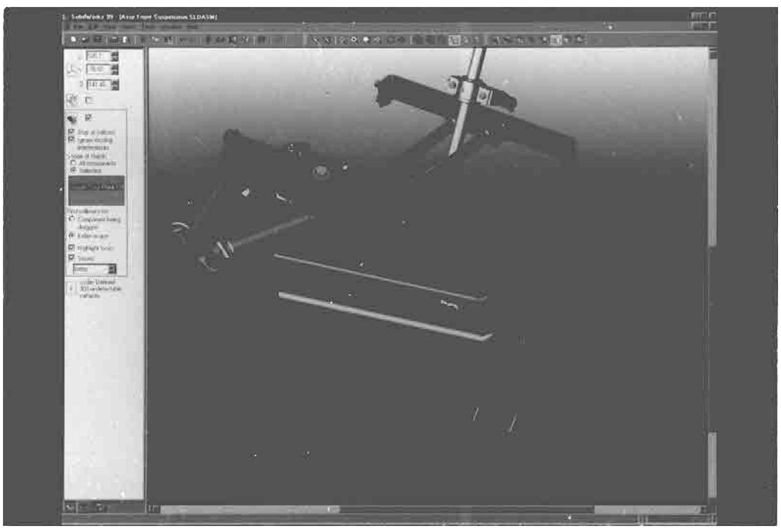
Most CAD solid-model software includes a library of primitive 3D shapes such as rectangular prisms, spheres, cylinders, and cones. Using Boolean operations for forming unions, subtractions, and intersections, these components can be added, subtracted, intersected, and sectioned to form complex 3D assemblies. Shading can be used to make the solid model easier for the viewers to comprehend. Precise 2D standard, isometric, and auxiliary views as well as cross sections can be extracted from the solid modeling data, and the cross sections can be cross-hatched.

Three-Dimensional Feature-Based Solid Modeling

3D feature-based solid modeling starts with one or more wireframe profiles. It creates a solid model by extruding, sweeping, revolving, or skinning these profiles. Boolean operations can



3D illustration of an indexing wheel drawn with 3D solid modeling software. *Courtesy of SolidWorks Corporation*



3D illustration of the ski suspension mechanism of a bobsled drawn with 3D modeling software. *Courtesy of SolidWorks Corporation* also be used on the profiles as well as the solids generated from these profiles. Solids can also be created by combining surfaces, including those with complex shapes. For example, this technique can be used to model streamlined shapes such as those of a ship's hull, racing-car's body, or aircraft.

3D feature-based solid modeling allows the designer to create such features as holes, fillets, chamfers, bosses, and pockets, and combine them with specific edges and faces of the model. If a design change causes the edges or faces to move, the features can be regenerated so that they move with the changes to keep their original relationships.

However, to use this system effectively, the designer must make the right dimensioning choices when developing these models, because if the features are not correctly referenced, they could end up the wrong location when the model is regenerated. For example, a feature that is positioned from the edge of an object rather than from its center might no longer be centered when the model is regenerated. The way to avoid this is to add constraints to the model that will keep the feature at the center of the face.

The key benefit of the parametric feature of solid modeling is that it provides a method for facilitating change. It imposes dimensional constraints on the model that permit the design to meet specific requirements for size and shape. This software permits the use of constraint equations that govern relationships between parameters. If some parameters remain constant or a specific parameter depends on the values of others, these relationships will be maintained throughout the design process. This form of modeling is useful if the design is restricted by space allowed for the end product or if its parts such as pipes or wiring must mate precisely with existing pipes or conduits.

Thus, in a parametric model, each entity, such as a line or arc in a wireframe, or fillet, is constrained by dimensional parameters. For example, in the model of a rectangular object, these parameters can control its geometric properties such as the length, width, and height. The parametric feature allows the designer to make changes as required to create the desired model. This software uses stored historical records that have recorded the steps in producing the model so that if the parameters of the model are changed, the software refers to the stored history and repeats the sequence of operations to create a new model for regeneration. Parametric modeling can also be used in trial-and-error operations to determine the optimum size of a component best suited for an application, either from an engineering or aesthetic viewpoint, simply by adjusting the parameters and regenerating a new model.

Parametric modeling features will also allow other methods of relating entities. Design features can, for example, be located at the origin of curves, at the end of lines or arcs, at vertices, or at the midpoints of lines and faces, and they can also be located at a specified distance or at the end of a vector from these points. When the model is regenerated, these relationships will be maintained. Some software systems also allow geometric constraints between features. These can mandate that the features be parallel, tangent, or perpendicular.

Some parametric modeling features of software combine freeform solid modeling, parametric solid modeling, surface modeling, and wireframe modeling to produce true hybrid models. Its features typically include hidden line removal, associative layouts, photorealistic rendering, attribute masking, and level management.

Three-Dimensional Hybrid Surface and Solid Modeling

Some modeling techniques are more efficient that others. For example, some are better for surfacing the more complex shapes as well as organic and freeform shapes. Consequently, commercial software producers offer 3D hybrid surface and solid-modeling suites that integrate 2D drafting and 3D wireframe with 3D surface and 3D solid modeling into a single CAD package. Included in these packages might also be software for photorealistic rendering and data translators to transport all types of data from the component parts of the package to other CAD or CAM software.

Glossary of Commonly Used CAD Terms

absolute coordinates: Distances measured from a fixed reference point, such as the origin, on the computer screen.

ANSI: An abbreviation for the American National Standards Institute.

associative dimensions: A method of dimensioning in CAD software that automatically updates dimension values when dimension size is changed.

Boolean modeling: A CAD 3D modeling technique that permits the user to add or subtract 3D shapes from one model to another.

Cartesian coordinates: A rectangular system for locating points in a drawing area in which the origin point is the 0,0 location and *X* represents length, *Y* width, and *Z* height. The surfaces between them can be designated as the *X–Z*, *X–Y*, and *Y–Z* planes.

composite drawing: A drawing containing multiple drawings in the form of CAD layers.

DXF: An abbreviation for Data Exchange Format, a standard format or translator for transferring data describing CAD drawings between different CAD programs.

FEM: An acronym for Finite Element Method for CAD structural design.

FTD: An abbreviation for File Transfer Protocol for upload and download of files to the Internet.

function: A task in a CAD program that can be completed by issuing a set of commands.

GD&T: An automated geometric, dimensioning, and tolerancing feature of CAD software.

GIS: An abbreviation for Geographic Information System.

IGES: An abbreviation for International Graphics Exchange Specification, a standard format or translator for transferring CAD data between different programs.

ISO: An abbreviation for International Standards Organization.

linear extrusion: A 3D technique that projects 2D into 3D shapes along a linear path.

MCAD: An abbreviation for mechanical CAD.

menu: A set of modeling functions or commands that are displayed on the computer screen. Options can be selected from the menu by a pointing device such as a mouse.

object snaps: A method for indicating point locations on existing drawings as references.

origin point: The 0,0 location in the coordinate system.

parametric modeling: CAD software that links the 3D drawing on the computer screen with data that sets dimensional and positional constraints.

polar coordinates: A coordinate system that locates points with an angle and radial distance from the origin, considered to be the center of a sphere.

polyline: A string of lines that can contain many connected line segments.

primitives: The basic elements of a graphics display such as points, lines, curves, polygons, and alphanumeric characters.

prototype drawing: A master drawing or template that includes preset computer defaults so that it can be reused in other applications.

radial extrusion: A 3D technique for projecting 2D into 3D shapes along a circular path.

spline: A flexible curve that can be drawn to connect a series of points in a smooth shape.

STL: An abbreviation for Solid Transfer Language, files created by a CAD system for use in rapid prototyping (RP).

tangent: A line in contact with the circumference of a circle that is at right angles to a line drawn between the contact point and the center of the circle.

NEW PROCESSES EXPAND CHOICES FOR RAPID PROTOTYPING

New concepts in rapid prototyping (RP) have made it possible to build many different kinds of 3D prototype models faster and cheaper than by traditional methods. The 3D models are fashioned automatically from such materials as plastic or paper, and they can be full size or scaled-down versions of larger objects. Rapid-prototyping techniques make use of computer programs derived from computer-aided design (CAD) drawings of the object. The completed models, like those made by machines and manual wood carving, make it easier for people to visualize a new or redesigned product. They can be passed around a conference table and will be especially valuable during discussions among product design team members, manufacturing managers, prospective suppliers, and customers.

At least nine different RP techniques are now available commercially, and others are still in the development stage. Rapid prototyping models can be made by the owners of proprietary equipment, or the work can be contracted out to various RP centers, some of which are owned by the RP equipment manufacturers. The selection of the most appropriate RP method for any given modeling application usually depends on the urgency of the design project, the relative costs of each RP process, and the anticipated time and cost savings RP will offer over conventional model-making practice. New and improved RP methods are being introduced regularly, so the RP field is in a state of change, expanding the range of designer choices.

Three-dimensional models can be made accurately enough by RP methods to evaluate the design process and eliminate interference fits or dimensioning errors before production tooling is ordered. If design flaws or omissions are discovered, changes can be made in the source CAD program and a replacement model can be produced quickly to verify that the corrections or improvements have been made. Finished models are useful in evaluations of the form, fit, and function of the product design and for organizing the necessary tooling, manufacturing, or even casting processes.

Most of the RP technologies are additive; that is, the model is made automatically by building up contoured laminations sequentially from materials such as photopolymers, extruded or beaded plastic, and even paper until they reach the desired height. These processes can be used to form internal cavities, overhangs, and complex convoluted geometries as well as simple planar or curved shapes. By contrast, a subtractive RP process involves milling the model from a block of soft material, typically plastic or aluminum, on a computer-controlled milling machine with commands from a CAD-derived program.

In the additive RP processes, photopolymer systems are based on successively depositing thin layers of a liquid resin, which are then solidified by exposure to a specific wavelengths of light. Thermoplastic systems are based on procedures for successively melting and fusing solid filaments or beads of wax or plastic in layers, which harden in the air to form the finished object. Some systems form layers by applying adhesives or binders to materials such as paper, plastic powder, or coated ceramic beads to bond them.

The first commercial RP process introduced was *stereolithography* in 1987, followed by a succession of others. Most of the commercial RP processes are now available in Europe and Japan as well as the United States. They have become multinational businesses through branch offices, affiliates, and franchises.

Each of the RP processes focuses on specific market segments, taking into account their requirements for model size, durability, fabrication speed, and finish in the light of anticipated economic benefits and cost. Some processes are not effective in making large models, and each process results in a model with a different finish. This introduces an economic tradeoff of higher price for smoother surfaces versus additional cost and labor of manual or machine finishing by sanding or polishing.

Rapid prototyping is now also seen as an integral part of the even larger but not well defined rapid tooling (RT) market. Concept modeling addresses the early stages of the design process, whereas RT concentrates on production tooling or mold making.

Some concept modeling equipment, also called 3D or office printers, are self-contained desktop or benchtop manufacturing units small enough and inexpensive enough to permit prototype fabrication to be done in an office environment. These units include provision for the containment or venting of any smoke or noxious chemical vapors that will be released during the model's fabrication.

Computer-Aided Design Preparation

The RP process begins when the object is drawn on the screen of a CAD workstation or personal computer to provide the digital data base. Then, in a post-design data processing step, computer software slices the object mathematically into a finite number of horizontal layers in generating an STL (Solid Transfer Language) file. The thickness of the "slices" can range from 0.0025 to 0.5 in. (0.06 to 13 mm) depending on the RP process selected. The STL file is then converted to a file that is compatible with the specific 3D "printer" or processor that will construct the model.

The digitized data then guides a laser, X-Y table, optics, or other apparatus that actually builds the model in a process comparable to building a high-rise building one story at a time. Slice thickness might have to be modified in some RP processes during model building to compensate for material shrinkage.

Prototyping Choices

All of the commercial RP methods depend on computers, but four of them depend on laser beams to cut or fuse each lamination, or provide enough heat to sinter or melt certain kinds of materials. The four processes that make use of lasers are Directed-Light Fabrication (DLF), Laminated-Object Manufacturing (LOM), Selective Laser Sintering (SLS), and Stereolithography (SL); the five processes that do not require lasers are Ballistic Particle Manufacturing (BPM), Direct-Shell Production Casting (DSPC), Fused-Deposition Modeling (FDM), Solid-Ground Curing (SGC), and 3D Printing (3DP).

Stereolithography (SL)

The stereolithographic (SL) process is performed on the equipment shown in Fig. 1. The movable platform on which the 3D model is formed is initially immersed in a vat of liquid photopolymer resin to a level just below its surface so that a thin layer of the resin covers it. The SL equipment is located in a sealed chamber to prevent the escape of fumes from the resin vat.

The resin changes from a liquid to a solid when exposed to the ultraviolet (UV) light from a low-power, highly focused laser. The UV laser beam is focused on an X-Y mirror in a computer-controlled beam-shaping and scanning system so that it draws the outline of the lowest cross-section layer of the object being built on the film of photopolymer resin.

After the first layer is completely traced, the laser is then directed to scan the traced areas of resin to solidify the model's first cross section. The laser beam can harden the layer down to a depth of 0.0025 to 0.0300 in. (0.06 to 0.8 mm). The laser beam scans at speeds up to 350 in./s (890 cm/s). The photopolymer not scanned by the laser beam remains a liquid. In general, the thinner the resin film (slice thickness), the higher the resolution or more refined the finish of the completed model. When model surface finish is important, layer thicknesses are set for 0.0050 in. (0.13 mm) or less.

The table is then submerged under computer control to the specified depth so that the next layer of liquid polymer flows over the first hardened layer. The tracing, hardening, and recoating steps are repeated, layer-by-layer, until the complete 3D model is built on the platform within the resin vat.

Because the photopolymer used in the SL process tends to curl or sag as it cures, models with overhangs or unsupported horizontal sections must be reinforced with supporting structures: walls, gussets, or columns. Without support, parts of the model can sag or break off before the polymer has fully set. Provision for forming these supports is included in the digitized fabrication data. Each scan of the laser forms support layers where necessary while forming the layers of the model.

When model fabrication is complete, it is raised from the polymer vat and resin is allowed to drain off; any excess can be removed manually from the model's surfaces. The SL process leaves the model only partially polymerized, with only about half of its fully cured strength. The model is then finally cured by exposing it to intense UV light in the enclosed chamber of post-curing apparatus (PCA). The UV completes the hardening or curing of the liquid polymer by linking its molecules in chainlike formations. As a final step, any supports that were required are removed, and the model's surfaces are sanded or polished. Polymers such as urethane acrylate resins can be milled, drilled, bored, and tapped, and their outer surfaces can be polished, painted, or coated with sprayed-on metal.

The liquid SL photopolymers are similar to the photosensitive UV-curable polymers used to form masks on semiconductor wafers for etching and plating features on integrated circuits. Resins can be formulated to solidify under either UV or visible light.

The SL process was the first to gain commercial acceptance, and it still accounts for the largest base of installed RP systems. 3D Systems of Valencia, California, is a company that manufactures stereolithography equipment for its proprietary SLA process. It offers the *ThermoJet Solid Object Printer*. The SLA process can build a model within a volume measuring 10 × 7.5 × 8 in. (25 × 19 × 20 cm). It also offers the SLA 7000 system, which can form objects within a volume of 20 × 20 × 23.62 in. (51 × 51 × 60 cm). Aaroflex, Inc. of Fairfax, Virginia, manufactures the Aacura 22 solid-state SL system and operates AIM, an RP manufacturing service.

Solid Ground Curing (SGC)

Solid ground curing (SGC) (or the "solider process") is a multistep in-line process that is diagrammed in Fig. 2. It begins when a photomask for the first layer of the 3D model is generated by the equipment shown at the far left. An electron gun writes a charge pattern of the photomask on a clear glass plate, and opaque toner is transferred electrostatically to the plate to form the photolithographic pattern in a xerographic process. The photomask is then moved to the exposure station, where it is aligned over a work platform and under a collimated UV lamp.

Model building begins when the work platform is moved to the right to a resin application station where a thin layer of photopolymer resin is applied to the top surface of the work platform and wiped to the desired thickness. The platform is then moved left to the exposure station, where the UV lamp is then turned on and a shutter is opened for a few seconds to expose the resin layer to the mask pattern. Because the UV light is so intense,

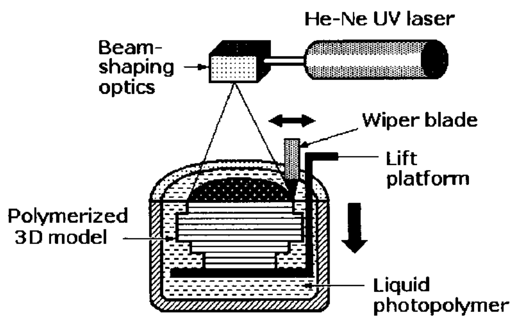


Fig. 1 Stereolithography (SL): A computer-controlled neon–helium ultraviolet light (UV)–emitting laser outlines each layer of a 3D model in a thin liquid film of UV-curable photopolymer on a platform submerged a vat of the resin. The laser then scans the outlined area to solidify the layer, or "slice." The platform is then lowered into the liquid to a depth equal to layer thickness, and the process is repeated for each layer until the 3D model is complete. Photopolymer not exposed to UV remains liquid. The model is them removed for finishing.

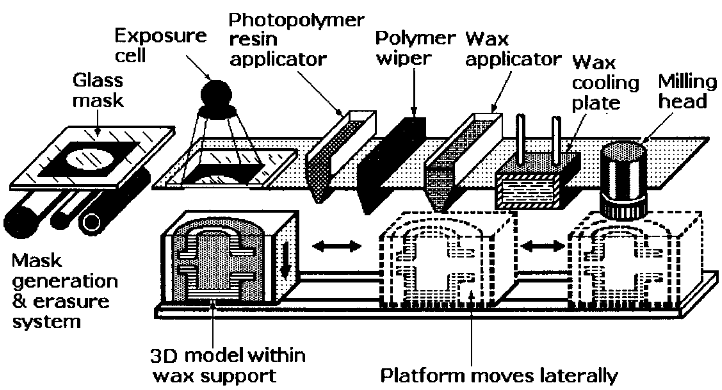


Fig. 2 Solid Ground Curing (SGC): First, a photomask is generated on a glass plate by a xerographic process. Liquid photopolymer is applied to the work platform to form a layer, and the platform is moved under the photomask and a strong UV source that defines and hardens the layer. The platform then moves to a station for excess polymer removal before wax is applied over the hardened layer to fill in margins and spaces. After the wax is cooled, excess polymer and wax are milled off to form the first "slice." The first photomask is erased, and a second mask is formed on the same glass plate. Masking and layer formation are repeated with the platform being lowered and moved back and forth under the stations until the 3D model is complete. The wax is then removed by heating or immersion in a hot water bath to release the prototype.

the layer is fully cured and no secondary curing is needed.

The platform is then moved back to the right to the wiper station, where all of resin that was not exposed to UV is removed and discarded. The platform then moves right again to the wax application station, where melted wax is applied and spread into the cavities left by the removal of the uncured resin. The wax is hardened at the next station by pressing it against a cooling plate. After that, the platform is moved right again to the milling station, where the resin and wax layer are milled to a precise thickness. The platform piece is then returned to the resin application station, where it is lowered a depth equal to the thickness of the next layer and more resin is applied.

Meanwhile, the opaque toner has been removed from the glass mask and a new mask for the next layer is generated on the same plate. The complete cycle is repeated, and this will continue until the 3D model encased in the wax matrix is completed. This matrix supports any overhangs or undercuts, so extra support structures are not needed.

After the prototype is removed from the process equipment, the wax is either melted away or dissolved in a washing chamber similar to a dishwasher. The surface of the 3D model is then sanded or polished by other methods.

The SGC process is similar to *drop on demand inkjet plotting*, a method that relies on a dual inkjet subsystem that travels on a precision X-Y drive carriage and deposits both thermoplastic and wax materials onto the build platform under CAD program control. The drive carriage also energizes a flatbed milling subsystem for obtaining the precise vertical height of each layer and the overall object by milling off the excess material.

Cubital America Inc., Troy, Michigan, offers the *Solider 4600/5600* equipment for building prototypes with the SGC process.

Selective Laser Sintering (SLS)

Selective laser sintering (SLS) is another RP process similar to stereolithography (SL). It creates 3D models from plastic, metal, or ceramic powders with heat generated by a carbon dioxide infrared (IR)–emitting laser, as shown in Fig. 3. The prototype is fabricated in a cylinder with a piston, which acts as a moving platform, and it is positioned next to a cylinder filled with preheated powder. A piston within the powder delivery system rises to eject powder, which is spread by a roller over the top of the build cylinder. Just before it is applied, the powder is heated further until its temperature is just below its melting point

When the laser beam scans the thin layer of powder under the control of the optical scanner system, it raises the temperature of the powder even further until it melts or sinters and flows together to form a solid layer in a pattern obtained from the CAD data.

As in other RP processes, the piston or supporting platform is lowered upon completion of each layer and the roller spreads the next layer of powder over the previously deposited layer. The process is repeated, with each layer being fused to the underlying layer, until the 3D prototype is completed.

The unsintered powder is brushed away and the part removed. No final curing is required, but because the objects are sintered they are porous. Wax, for example, can be applied to the inner and outer porous surfaces, and it can be smoothed by various manual or machine grinding or melting processes. No supports are required in SLS because overhangs and undercuts are supported by the compressed unfused powder within the build cylinder.

Many different powdered materials have been used in the SLS process, including polycarbonate, nylon, and investment casting wax. Polymer-coated metal powder is also being studied as an alternative. One advantage of the SLS process is that materials such as polycarbonate and nylon are strong and stable enough to permit the model to be used in limited functional and environmental testing. The prototypes can also serve as molds or patterns for casting parts.

SLS process equipment is enclosed in a nitrogen-filled chamber that is sealed and maintained at a temperature just below the melting point of the powder. The nitrogen prevents an explosion that could be caused by the rapid oxidation of the powder.

The SLS process was developed at the University of Texas at Austin, and it has been licensed by the DTM Corporation of Austin, Texas. The company makes a *Sinterstation 2500plus*. Another company participating in SLS is EOS GmbH of Germany.

Laminated-Object Manufacturing (LOM)

The Laminated-Object Manufacturing (LOM) process, diagrammed in Fig. 4, forms 3D models by cutting, stacking, and bonding successive layers of paper coated with heat-activated adhesive. The carbon-dioxide laser beam, directed by an optical system under CAD data control, cuts cross-sectional outlines of the prototype in the layers of paper, which are bonded to previous layers to become the prototype.

The paper that forms the bottom layer is unwound from a supply roll and pulled across the movable platform. The laser beam cuts the outline of each lamination and cross-hatches the waste material within and around the lamination to make it easier to remove after the prototype is completed. The outer waste material web from each lamination is continuously removed by a take-up roll. Finally, a heated roller applies pressure to bond the adhesive coating on each layer cut from the paper to the previous layer.

A new layer of paper is then pulled from a roll into position over the previous layer, and the cutting, cross hatching, web removal, and bonding procedure is repeated until the model is completed.

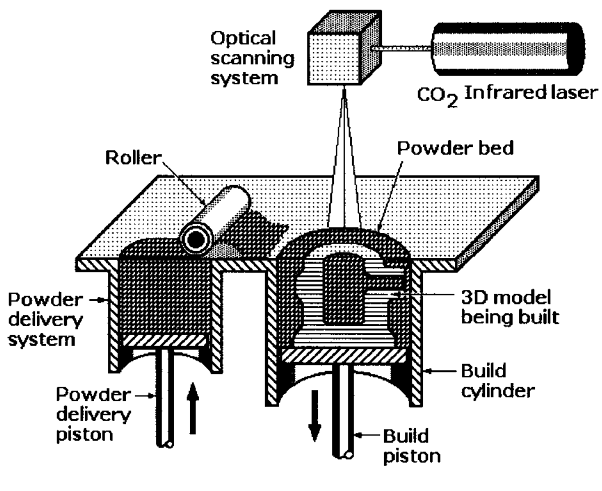


Fig. 3 Selective Laser Sintering (SLS): Loose plastic powder from a reservoir is distributed by roller over the surface of piston in a build cylinder positioned at a depth below the table equal to the thickness of a single layer. The powder layer is then scanned by a computer-controlled carbon dioxide infrared laser that defines the layer and melts the powder to solidify it. The cylinder is again lowered, more powder is added, and the process is repeated so that each new layer bonds to the previous one until the 3D model is completed. It is then removed and finished. All unbonded plastic powder can be reused.

When all the layers have been cut and bonded, the excess cross-hatched material in the form of stacked segments is removed to reveal the finished 3D model. The models made by the LOM have woodlike finishes that can be sanded or polished before being sealed and painted.

Using inexpensive, solid-sheet materials makes the 3D LOM models more resistant to deformity and less expensive to produce than models made by other processes, its developers say. These models can be used directly as patterns for investment and sand casting, and as forms for silicone molds. The objects made by LOM can be larger than those made by most other RP processes—up to 30 × 20 × 20 in. (75 × 50 × 50 cm).

The LOM process is limited by the ability of the laser to cut through the generally thicker lamination materials and the additional work that must be done to seal and finish the model's inner and outer surfaces. Moreover, the laser cutting process burns the paper, forming smoke that must be removed from the equipment and room where the LOM process is performed.

Helysys Corporation, Torrance, California, manufactures the LOM-2030H LOM equipment. Alternatives to paper including sheet plastic and ceramic and metal-powder-coated tapes have been developed.

Other companies offering equipment for building prototypes from paper laminations are the Schroff Development Corporation, Mission, Kansas, and CAM-LEM, Inc. Schroff manufactures the *JP System 5* to permit desktop rapid prototyping.

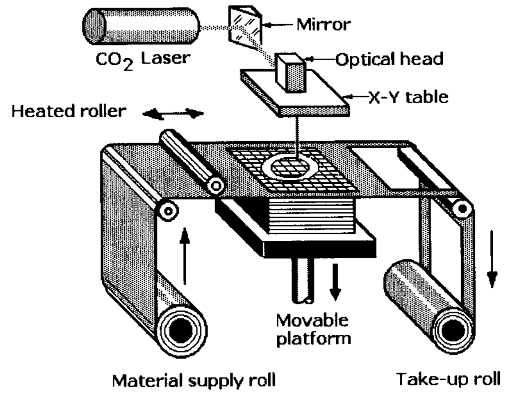


Fig. 4 Laminated Object Manufacturing (LOM): Adhesive-backed paper is fed across an elevator platform and a computer-controlled carbon dioxide infrared-emitting laser cuts the outline of a layer of the 3D model and cross-hatches the unused paper. As more paper is fed across the first layer, the laser cuts the outline and a heated roller bonds the adhesive of the second layer to the first layer. When all the layers have been cut and bonded, the cross-hatched material is removed to expose the finished model. The complete model can then be sealed and finished.

Fused Deposition Modeling (FDM)

The Fused Deposition Modeling (FDM) process, diagrammed in Fig. 5, forms prototypes from melted thermoplastic filament. This filament, with a diameter of 0.070 in. (1.78 mm), is fed into a temperature-controlled FDM extrusion head where it is heated to a semi-liquid state. It is then extruded and deposited in ultrathin, precise layers on a fixtureless platform under X-Y computer control. Successive laminations ranging in thickness from 0.002 to 0.030 in. (0.05 to 0.76 mm) with wall thicknesses of 0.010 to 0.125 in. (0.25 to 3.1 mm) adhere to each by thermal fusion to form the 3D model.

Structures needed to support overhanging or fragile structures in FDM modeling must be designed into the CAD data file and fabricated as part of the model. These supports can easily be removed in a later secondary operation.

All components of FDM systems are contained within temperature-controlled enclosures. Four different kinds of inert, nontoxic filament materials are being used in FDM: ABS polymer (acrylonitrile butadiene styrene), high-impact-strength ABS (ABSi), investment casting wax, and elastomer. These materials melt at temperatures between 180 and 220ºF (82 and 104ºC).

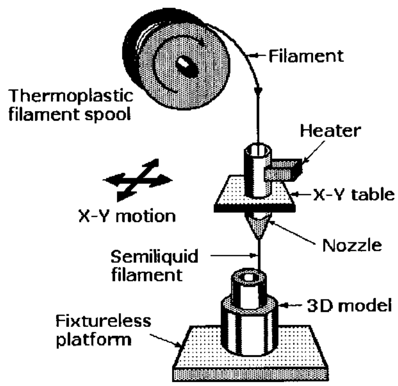


Fig. 5 Fused Deposition Modeling (FDM): Filaments of thermoplastic are unwound from a spool, passed through a heated extrusion nozzle mounted on a computer-controlled X-Y table, and deposited on the fixtureless platform. The 3D model is formed as the nozzle extruding the heated filament is moved over the platform. The hot filament bonds to the layer below it and hardens. This laserless process can be used to form thin-walled, contoured objects for use as concept models or molds for investment casting. The completed object is removed and smoothed to improve its finish.

FDM is a proprietary process developed by Stratasys, Eden Prairie, Minnesota. The company offers four different systems. Its *Genisys* benchtop 3D printer has a build volume as large as 8 × 8 × 8 in. (20 × 20 × 20 cm), and it prints models from square polyester wafers that are stacked in cassettes. The material is heated and extruded through a 0.01-in. (0.25-mm)–diameter hole at a controlled rate. The models are built on a metallic substrate that rests on a table. Stratasys also offers four systems that use spooled material. The *FDM2000*, another benchtop system, builds parts up to 10 in^3 (164 cm^3) while the *FDM3000*, a floor-standing system, builds parts up to 10 × 10 × 16 in. (26 × 26 × 41 cm).

Two other floor-standing systems are the *FDM 8000*, which builds models up to 18 × 18 × 24 in. (46 × 46 × 61 cm), and the *FDM Quantum* system, which builds models up to 24 × 20 × 24 in. (61 × 51 × 61 cm). All of these systems can be used in an office environment.

Stratasys offers two options for forming and removing supports: a breakaway support system and a water-soluble support system. The water-soluble supports are formed by a separate extrusion head, and they can be washed away after the model is complete.

Three-Dimensional Printing (3DP)

The Three-Dimensional Printing (3DP) or inkjet printing process, diagrammed in Fig. 6, is similar to Selective Laser Sintering (SLS) except that a multichannel inkjet head and liquid adhesive supply replaces the laser. The powder supply cylinder is filled with starch and cellulose powder, which is delivered to the work platform by elevating a delivery piston. A roller rolls a single layer of powder from the powder cylinder to the upper surface of a piston within a build cylinder. A multichannel inkjet head sprays a water-based liquid adhesive onto the surface of the powder to bond it in the shape of a horizontal layer of the model.

In successive steps, the build piston is lowered a distance equal to the thickness of one layer while the powder delivery piston pushes up fresh powder, which the roller spreads over the previous layer on the build piston. This process is repeated until the 3D model is complete. Any loose excess powder is brushed away, and wax is coated on the inner and outer surfaces of the model to improve its strength.

The 3DP process was developed at the Three-Dimensional Printing Laboratory at the Massachusetts Institute of Technology, and it has been licensed to several companies. One of those firms, the Z Corporation of Somerville, Massachusetts, uses the original MIT process to form 3D models. It also offers the *Z402* 3D modeler. Soligen Technologies has modified the 3DP process to make ceramic molds for investment casting. Other companies are using the process to manufacture implantable drugs, make metal tools, and manufacture ceramic filters.

Direct-Shell Production Casting (DSPC)

The Direct Shell Production Casting (DSPC) process, diagrammed in Fig. 7, is similar to the 3DP process except that it is focused on forming molds or shells rather than 3D models. Consequently, the actual 3D model or prototype must be produced by a later casting process. As in the 3DP process, DSPC begins with a CAD file of the desired prototype.

Two specialized kinds of equipment are needed for DSPC: a dedicated computer called a shell-design unit (SDU) and a shell- or mold-processing unit (SPU). The CAD file is loaded into the SDU to generate the data needed to define the mold. SDU software also modifies the original design dimensions in the CAD file to compensate for ceramic shrinkage. This software can also add fillets and delete such features as holes or keyways that must be machined after the prototype is cast.

The movable platform in DSPC is the piston within the build cylinder. It is lowered to a depth below the rim of the build cylinder equal to the thickness of each layer. Then a thin layer of fine aluminum oxide (alumina) powder is spread by roller over the platform, and a fine jet of colloidal silica is sprayed precisely onto the powder surface to bond it in the shape of a single mold layer. The piston is then lowered for the next layer and the complete process is repeated until all layers have been formed, completing the entire 3D shell. The excess powder is then removed, and the mold is fired to convert the bonded powder to monolithic ceramic.

After the mold has cooled, it is strong enough to withstand molten metal and can function like a conventional investment-casting mold. After the molten metal has cooled, the ceramic shell and any cores or gating are broken away from the prototype. The casting can then be finished by any of the methods usually used on metal castings.

DSPC is a proprietary process of Soligen Technologies, Northridge, California. The company also offers a custom mold manufacturing service.

Ballistic Particle Manufacturing (BPM)

There are several different names for the Ballistic Particle Manufacturing (BPM) process, diagrammed in Fig. 8.

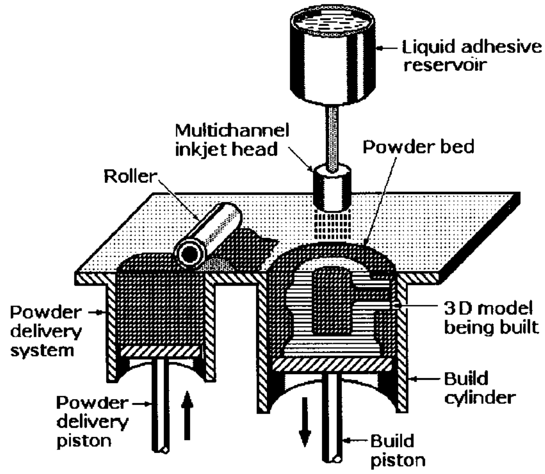


Fig. 6 Three-Dimensional Printing (3DP): Plastic powder from a reservoir is spread across a work surface by roller onto a piston of the build cylinder recessed below a table to a depth equal to one layer thickness in the 3DP process. Liquid adhesive is then sprayed on the powder to form the contours of the layer. The piston is lowered again, another layer of powder is applied, and more adhesive is sprayed, bonding that layer to the previous one. This procedure is repeated until the 3D model is complete. It is then removed and finished.

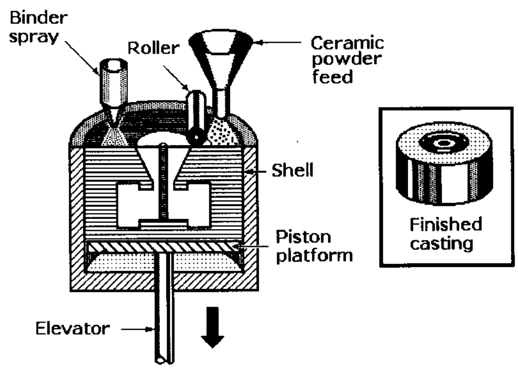


Fig. 7 Direct Shell Production Casting (DSPC): Ceramic molds rather than 3D models are made by DSPC in a layering process similar to other RP methods. Ceramic powder is spread by roller over the surface of a movable piston that is recessed to the depth of a single layer. Then a binder is sprayed on the ceramic powder under computer control. The next layer is bonded to the first by the binder. When all of the layers are complete, the bonded ceramic shell is removed and fired to form a durable mold suitable for use in metal casting. The mold can be used to cast a prototype. The DSPC process is considered to be an RP method because it can make molds faster and cheaper than conventional methods.

Variations of it are also called *inkjet methods*. The molten plastic used to form the model and the hot wax for supporting overhangs or indentations are kept in heated tanks above the build station and delivered to computer-controlled jet heads through thermally insulated tubing. The jet heads squirt tiny droplets of the materials on the work platform as it is moved by an X-Y table in the pattern needed to form each layer of the 3D object. The droplets are deposited only where directed, and they harden rapidly as they leave the jet heads. A milling cutter is passed over the layer to mill it to a uniform thickness. Particles that are removed by the cutter are vacuumed away and deposited in a collector.

Nozzle operation is monitored carefully by a separate fault-detection system. After each layer has been deposited, a stripe of each material is deposited on a narrow strip of paper for thickness measurement by optical detectors. If the layer meets specifications, the work platform is lowered a distance equal to the required layer thickness and the next layer is deposited. However, if a clot is detected in either nozzle, a jet cleaning cycle is initiated to clear it. Then the faulty layer is milled off and that layer is redeposited. After the 3D model is completed, the wax material is either melted from the object by radiant heat or dissolved away in a hot water wash.

The BPM system is capable of producing objects with fine finishes, but the process is slow. With this RP method, a slower process that yields a 3D model with a superior finish is traded off against faster processes that require later manual finishing.

The version of the BPM system shown in Fig. 8 is called *Drop on Demand Inkjet Plotting* by Sanders Prototype Inc, Merrimac, New Hampshire. It offers the *ModelMaker II* processing equipment, which produces 3D models with this method. AeroMet Corporation builds titanium parts directly from CAD renderings by fusing titanium powder with an 18-kW carbon dioxide laser, and 3D Systems of Valencia, California, produces a line of inkjet printers that feature multiple jets to speed up the modeling process.

Directed Light Fabrication (DLF)

The Directed Light Fabrication (DLF) process, diagrammed in Fig. 9, uses a neodymium YAG (Nd:YAG) laser to fuse powdered metals to build 3D models that are more durable than models made from paper or plastics. The metal powders can be finely milled 300 and 400 series stainless steel, tungsten, nickel aluminides, molybdenum disilicide, copper, and aluminum. The technique is also called *Direct-Metal Fusing*, *Laser Sintering,* and *Laser Engineered Net Shaping (LENS).*

The laser beam under X-Y computer control fuses the metal powder fed from a nozzle to form dense 3D objects whose dimensions are said to be within a few thousandths of an inch of the desired design tolerance.

DLF is an outgrowth of nuclear weapons research at the Los Alamos National Laboratory (LANL), Los Alamos, New Mexico, and it is still in the development stage. The laboratory has been experimenting with the laser fusing of ceramic powders to fabricate parts as an alternative to the use of metal powders. A system that would regulate and mix metal powder to modify the properties of the prototype is also being investigated.

Optomec Design Company, Albuquerque, New Mexico, has announced that direct fusing of metal powder by laser in its LENS process is being performed commercially. Protypes made by this method have proven to be durable and they have shown close dimensional tolerances.

Research and Development in RP

Many different RP techniques are still in the experimental stage and have not yet achieved commercial status. At the same time, practical commercial processes have been improved. Information about this research has been announced by the laboratories doing the work, and some of the research is described in patents. This discussion is limited to two techniques, SDM and Mold SDM, that have shown commercial promise.

Shape Deposition Manufacturing (SDM)

The Shape Deposition Manufacturing (SDM) process, developed at the SDM Laboratory of Carnegie Mellon University, Pittsburgh, Pennsylvania, produces functional metal prototypes directly from CAD data. This process, diagrammed in Fig. 10, forms successive layers of metal on a platform without masking, and is also called *solid free-form* (SFF) fabrication. It uses hard met-

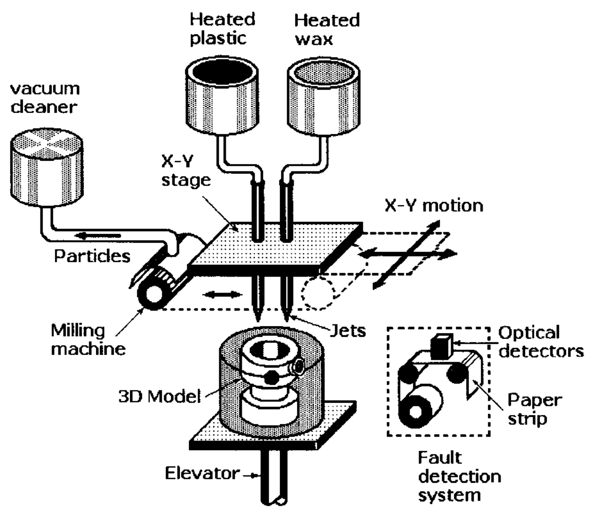


Fig. 8 Ballistic Particle Manufacturing (BPM): Heated plastic and wax are deposited on a movable work platform by a computer-controlled X-Y table to form each layer. After each layer is deposited, it is milled to a precise thickness. The platform is lowered and the next layer is applied. This procedure is repeated until the 3D model is completed. A fault detection system determines the quality and thickness of the wax and plastic layers and directs rework if a fault is found. The supporting wax is removed from the 3D model by heating or immersion in a hot liquid bath.

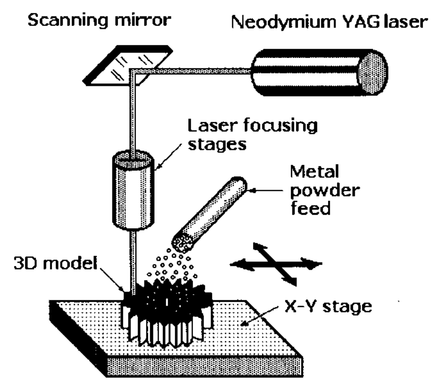


Fig. 9 Directed Light Fabrication (DLF): Fine metal powder is distributed on an X-Y work platform that is rotated under computer control beneath the beam of a neodymium YAG laser. The heat from the laser beam melts the metal powder to form thin layers of a 3D model or prototype. By repeating this process, the layers are built up and bonded to the previous layers to form more durable 3D objects than can be made from plastic. Powdered aluminum, copper, stainless steel, and other metals have been fused to make prototypes as well as practical tools or parts that are furnace-fired to increase their bond strength.

als to form more rugged prototypes that are then accurately machined under computer control during the process.

The first steps in manufacturing a part by SDM are to reorganize or destructure the CAD data into slices or layers of optimum thickness that will maintain the correct 3D contours of the outer surfaces of the part and then decide on the sequence for depositing the primary and supporting materials to build the object.

The primary metal for the first layer is deposited by a process called *microcasting* at the deposition station, Fig. 10(a). The work is then moved to a machining station (b), where a computer-controlled milling machine or grinder removes deposited metal to shape the first layer of the part. Next, the work is moved to a stress-relief station (c), where it is shot-peened to relieve stresses that have built up in the layer. The work is then transferred back to the deposition station (a) for simultaneous deposition of primary metal for the next layer and sacrificial support metal. The support material protects the part layers from the deposition steps that follow, stabilizes the layer for further machining operations, and provides a flat surface for milling the next layer. This SDM cycle is repeated until the part is finished, and then the sacrificial metal is etched away with acid. One combination of metals that has been successful in SDM is stainless steel for forming the prototype and copper for forming the support structure

The SDM Laboratory investigated many thermal techniques for depositing high-quality metals, including thermal spraying and plasma or laser welding, before it decided on microcasting, a compromise between these two techniques that provided better results than either technique by itself. The metal droplets in microcasting are large enough (1 to 3 mm in diameter) to retain their heat longer than the 50-μm droplets formed by conventional thermal spraying. The larger droplets remain molten and retain their heat long enough so that when they impact the metal surfaces they remelt them to form a strong metallurgical interlayer bond. This process overcame the low adhesion and low mechanical strength problems encountered with conventional thermal metal spraying. Weld-based deposition easily remelted the substrate material to form metallurgical bonds, but the larger amount of heat transferred tended to warp the substrate or delaminate it.

The SDM laboratory has produced custom-made functional mechanical parts and has embedded prefabricated mechanical parts, electronic components, electronic circuits, and sensors in the metal layers during the SDM process. It has also made custom tools such as injection molds with internal cooling pipes and metal heat sinks with embedded copper pipes for heat redistribution.

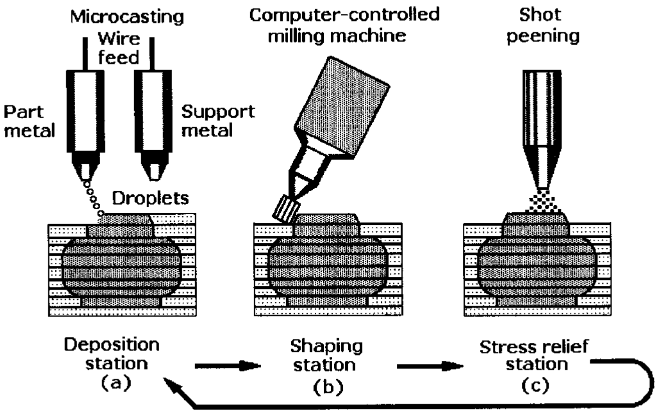


Fig. 10 Shape Deposition Manufacturing (SDM): Functional metal parts or tools can be formed in layers by repeating three basic steps repetitively until the part is completed. Hot metal droplets of both primary and sacrificial support material form layers by a thermal metal spraying technique (a). They retain their heat long enough to remelt the underlying metal on impact to form strong metallurgical interlayer bonds. Each layer is machined under computer control (b) and shot-peened (c) to relieve stress buildup before the work is returned for deposition of the next layer. The sacrificial metal supports any undercut features. When deposition of all layers is complete, the sacrificial metal is removed by acid etching to release the completed part.

Mold SDM

The Rapid Prototyping Laboratory at Stanford University, Palo Alto, California, has developed its own version of SDM, called Mold SDM, for building layered molds for casting ceramics and polymers. Mold SDM, as diagrammed in Fig. 11, uses wax to form the molds. The wax occupies the same position as the sacrificial support metal in SDM, and water-soluble photopolymer sacrificial support material occupies and supports the mold cavity. The photopolymer corresponds to the primary metal deposited to form the finished part in SDM. No machining is performed in this process.

The first step in the Mold SDM process begins with the decomposition of CAD mold data into layers of optimum thickness, which depends on the complexity and contours of the mold. The actual processing begins at Fig. 11(a), which shows the results of repetitive cycles of the deposition of wax for the mold and sacrificial photopolymer in each layer to occupy the mold cavity and support it. The polymer is hardened by an ultraviolet (UV) source. After the mold and support structures are built up, the work is moved to a station (b) where the photopolymer is removed by dissolving it in water. This exposes the wax mold cavity into which the final part material is cast. It can be any compatible castable material. For example, ceramic parts can be formed by pouring a gelcasting ceramic slurry into the wax mold (c) and then curing the slurry. The wax mold is then removed (d) by melting it, releasing the “green” ceramic part for furnace firing. In step (e), after firing, the vents and sprues are removed as the final step.

Mold SDM has been expanded into making parts from a variety of polymer materials, and it has also been used to make preassembled mechanisms, both in polymer and ceramic materials.

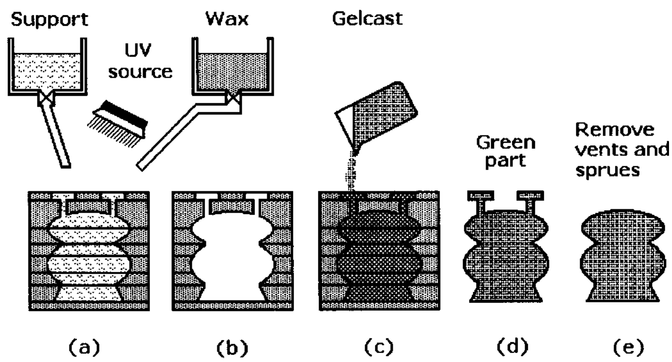


Fig. 11 Mold Shape Deposition Manufacturing (MSDM): Casting molds can be formed in successive layers: Wax for the mold and water-soluble photopolymer to support the cavity are deposited in a repetitive cycle to build the mold in layers whose thickness and number depend on the mold’s shape (a). UV energy solidifies the photopolymer. The photopolymer support material is removed by soaking it in hot water (b). Materials such as polymers and ceramics can be cast in the wax mold. For ceramic parts, a gelcasting ceramic slurry is poured into the mold to form green ceramic parts, which are then cured (c). The wax mold is then removed by heat or a hot liquid bath and the green ceramic part released (d). After furnace firing (e) any vents and sprues are removed.

MICROMACHINES OPEN A NEW FRONTIER FOR MACHINE DESIGN

A new technology for fabricating microminiature motors, valves, and transducers is a spinoff of the microcircuit fabrication technology that made microprocessors and semiconductor memories possible. This technology has opened the new field of microelectromechanical systems (MEMS) in machine design. These microscopic-scale machines require their own unique design rules, tools, processes, and materials.

These microminiature machines might not be familiar to traditional machine designers because their manufacture calls for photolithographic and chemical-etching processes rather than better-known casting, welding, milling, drilling, and lathe turning.

Nevertheless, even when made at a scale so small that they are best seen under an electronic microscope, the laws of physics, mechanics, electricity, and chemistry still apply to these micromachines. MEMS are moving machine and mechanism design down to dimensions measurable in atomic units. Until a few years ago, those dimensions were strictly the province of microbiologists, atomic physicists, and microcircuit designers.

Among the more remarkable examples of MEMS are a miniscule electric vehicle that can be parked on a pinhead, electric motors so small that they can easily fit inside the eye of a needle, and pumps and gear trains the size of grains of salt. Far from novelties that only demonstrate the feasibility of a technology, many are now being produced for automotive applications, and many more are being used in science and medicine.

The products now being routinely micromachined in quantity for the automotive industry are limited to microminiature pressures sensors, accelerometers, and fuel injectors. Nevertheless, research and development of microminiature actuators and motors for insertion into human arteries in order to perform certain kinds of delicate surgical procedures is now underway. In addition, other potential uses for them in biomedicine and electronics are now being tested.

The Microactuators

The rotary electrostatic motor shown in Fig. 1 is an outstanding example of a microactuator. The cutaway drawing shows a section view of a typical micromotor that is driven by static electricity. The experimental motors produced so far have diameters of 0.1 to 0.2 mm and they are about 4 to 6 μm high.

The rotor of a well-designed micromotor, driven with an excitation voltage of 30 to 40 V, can achieve speeds that exceed 10,000 rpm. Some of the tiny motors have operated continuously for 150 h. Motors of this kind have been made at the University of California at Berkeley and at the Massachusetts Institute of Technology (MIT).

The rotor shown in Fig. 1 has a “rising sun” geometry. It rests on bushings that minimize frictional contact with the base substrate, and it is free to rotate around a central hub. Slots separate individual commutator sections. Electrostatic forces are introduced by the inner surfaces of the stator and outer surfaces of the rotor, which form a rotating capacitor.

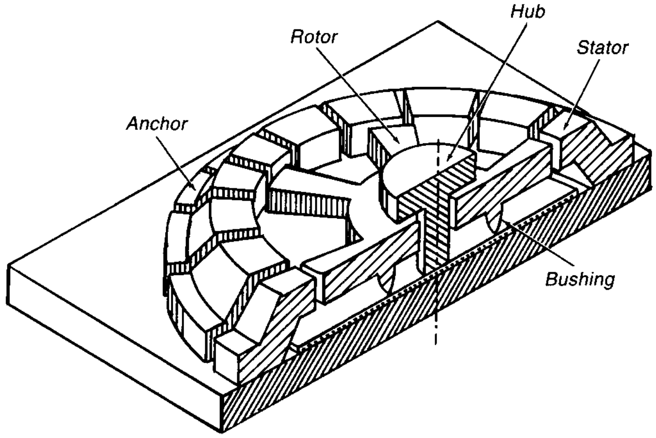


Fig. 1 A cross-section view of a typical micromotor that is driven electrostatically rather than electromagnetically.

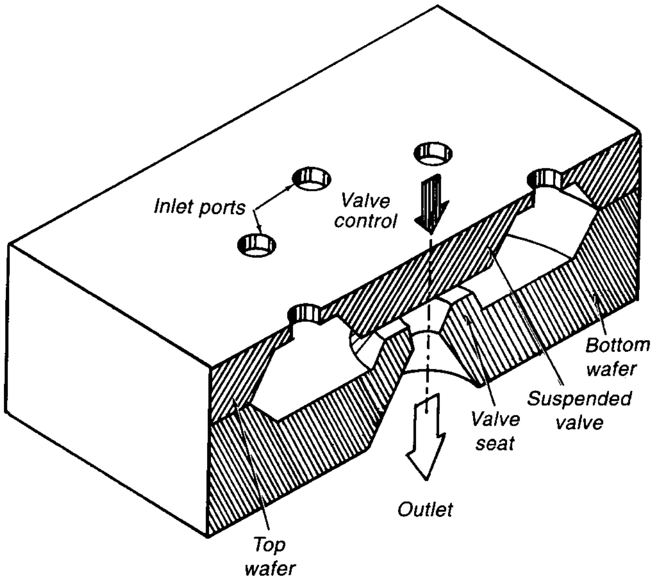


Fig. 2 A cutaway view of a typical microvalve. The diaphragm moves perpendicular to its base substrate. The diaphragms can be moved by an embedded piezoelectric film, by electrostatic forces, or by thermal expansion.

Micropumps and microvalves with deformable diaphragms are other forms of microactuator. Figure 2 is a cutaway view of a typical microvalve. The diaphragms of these devices flex in a direction that is perpendicular to their base substrates. The diaphragms can be moved by an embedded piezoelectric film, electrostatic forces, or thermal expansion. Applications are seen for the microminiature pumps and valves in biomedicine because they are orders of magnitude smaller than conventional biomedical pumps and valves.

Actuators have also been made in the form of vibrating microstructures with flexible suspensions. Figure 3 is a drawing of a linear resonator consisting of two identical folded beams of an interdigital electrostatic "comb" drive. The folded beams are supported by anchors grown on the semiconductor substrate, and the comb drive is supported by a pedestal grown on the same silicon substrate.

The folded beams are dimensioned to have a specific resonant frequency, and they are driven by electrostatic charges placed on the comb-drive digits, which act as capacitors. Both beam structures vibrate simultaneously but only in the X direction because lateral or Y-direction motion is constrained by the geometrics of the folded beams.

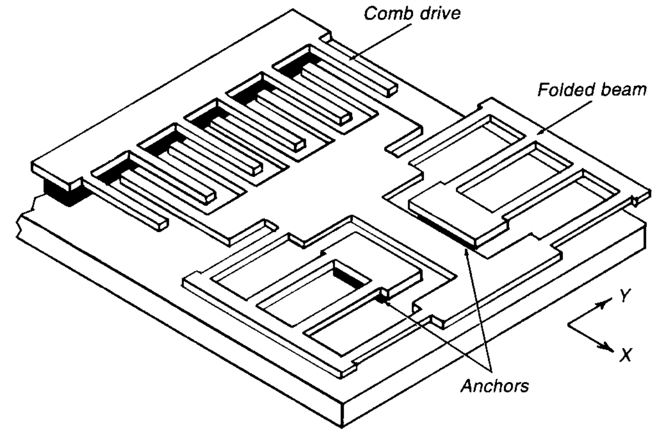


Fig. 3 This linear resonator consists of a pair of folded beams that are set in vibrational motion in the X direction by an electrostatically driven comb structure. Lateral or Y-direction motion is restrained by the geometry of the folded beams.

Materials

Currently, the most popular material for fabricating all of these micromachines is silicon, the material from which most microcircuit chips and discrete transistors are made. The silicon can be in either of several different forms. However, micromachines have also included parts made of aluminum and diamonds. The successful design and manufacture of billions of integrated circuits over the past 35 years have left an extensive database and body of knowledge about silicon—how to grow it, alter its structure chemically, mill it chemically, and bond slices of it together permanently.

Silicon is a very strong material with a modulus of elasticity that closely matches steel. Its lack of mechanical hysteresis makes it an almost perfect material for fabricating sensors and transducers. Silicon exceeds stainless steel in yield strength and aluminum in strength-to-weight ratio. It also exhibits high thermal conductivity and a low thermal expansion coefficient. Because silicon is sensitive to stress, strain, and temperature, silicon sensors can easily communicate with electronic circuitry for the transmission of electrical signals.

In the fabrication of micromachines, silicon is chemically etched into a wide variety of shapes rather than being machined with traditional cutting tools. Silicon, as well as such associated materials as polysilicon, silicon nitride, and aluminum, can be micromachined in batches into many different shapes and contours. In micromachining, mechanical structures are sculpted from a silicon wafer by selectively etching away sacrificial supporting layers or structures.

The etching process is complemented by such standard integrated circuit processes as photolithography for producing the required masks at the various stages of the process. Diffusion can alter the chemical makeup of the material by introducing "dopants." Epitaxy is a process for growing new material on the basic substrate, and deposition is a process for the "plating" of one type of material on another.

The bulk micromachining process is widely used for fabricating silicon accelerometers, but has also been applied to the fabrication of flow sensors, inkjet nozzles, microvalves, and motors. The etching process can be controlled by dispersing different doping materials within the silicon or by concentrating electrical current in specific regions.

Powering Micromachines

Micromachines can be actuated by the piezoelectric effect, thermal expansion, electrostatic force, or magnetic force. The choice of actuation method is influenced by the nature of the device and its specific application requirements. However, the microscopic dimensions of the devices generally rule out current-induced magnetic forces such as those that drive conventional electric motors and solenoids because those forces are too weak when scaled to the small sizes required to power the devices.

The high power consumption required to concentrate enough heat in a small local area to move parts by the thermal expansion of unlike materials is unacceptable for many applications. Therefore, the two most commonly used microactuation drive methods are the piezoelectric effect and electrostatic force.

Electrostatic Forces

Electrostatic forces are used to drive micromachines because, unlike magnetic forces, induced electrostatic forces can be scaled down favorably with size. Electrostatic force is induced by setting up equivalent parallel-plate capacitors between adjacent mechanical elements. There must be two conductive surfaces that act as opposing capacitor plates. The electrostatic force is directly proportional to the product of the square of the voltage across the plates and plate area, and it is inversely proportional to the square of the distance between the plates.

A surface-micromachined motor is shown in Fig. 1. The end surfaces of the rotor spokes and segmented inner walls of the insulated stator electrodes effectively form capacitors, which are separated by a small air gap. The rotor is the spoked wheel free to rotate around a central post. To drive the motor, the opposing insulated stator segments are energized in a rotating pattern and rotor spokes are attracted to the stator segments as they move into position near the stators to keep the rotor turning, making one revolution for many polarity changes in the stator elements. It can be seen that the spacing between the spokes and stator segments change with respect to time as the rotor turns. As a result, the electrostatic force varies with time or is a nonlinear function of the applied voltage.

Problems can arise if the rotor spokes are not uniform in radius dimensions, the bearing surfaces are not smooth, or the rotor does not rotate concentrically. Any of these mechanical defects could cause the rotor to stick in one position.

The handicap for surface micromachined motors is their small vertical dimensions, making it difficult for them to obtain large enough changes in capacitance when the rotor is in motion. Somewhat larger motors with thicker, taller stators and rotor segments have been made by LIGA techniques to overcome this drawback. (The abbreviation LIGA stands for the German words for lithography, electroplating, and molding-—Lithographie, Galvanoformung, Abformung.)

Another kind of electrostatic actuator, the electrostatic-comb drive shown in Fig. 3, was developed to maximize the capacitance effects in electrostatic micromachines by taking advantage of the classical parallel plate capacitor formula because only attractive forces can be generated.

$$E = \frac{CV^2}{2}$$

Where E is the energy stored,
C is the capacitance, and
V is the voltage across the capacitor.

Surface micromachined comb drives consist of many interdigitated fingers, as shown in Fig. 3. When a voltage is applied, an attractive force is developed between the fingers, which move together. The increase in capacitance is proportional to the number of fingers.

This means that large numbers of fingers are required to generate large forces. Because the direction of motion of the electrostatic comb drive is parallel with the length of the comb-finger electrodes, the effective plate area with respect to spacing between the plates remains constant.

Consequently, capacitance with respect to the direction of motion is linear, and the induced force in the X direction is directly proportional to the square of the voltage applied across the plates. Comb-drive structures have been driven to deflect by as much as one quarter of the comb finger length with DC voltages of 20 to 40 V.

A disadvantage of the comb drive is that if the lateral gaps between the fingers are not equal on the sides or if the fingers are not parallel, it is possible for the fingers to move at right angles to the intended direction of motion and adhere together until the voltage is turned off. They could remain stuck permanently.

Piezoelectric Films

Microminiature transducers have been made in the form of rigid beams and diaphragms with a core of polycrystalline zinc-oxide piezoelectric film. Figure 4 is a diagram of a beam with a central piezoelectric layer of insulated polycrystalline zinc oxide (ZnO) up to several micrometers thick. The layer is then insulated on both sides and sandwiched between two conductive electrodes to form the rigid structure.

When voltage is applied to the two electrodes, the piezoelectrically induced stress in the ZnO film causes the structure to deflect. The converse of the piezoelectric effect can be obtained in applications where it is desirable to convert the strain on the beam or diaphragm into electrical signals that are proportional to strain.

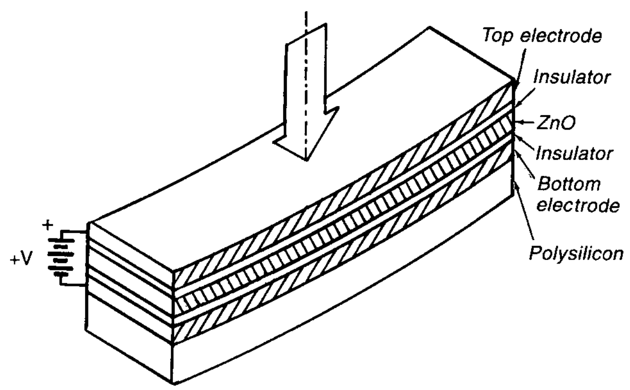


Fig. 4 A microminiature piezoelectric transducer is made as an insulated layer of polycrystalline zinc oxide (ZnO) sandwiched between two conductive electrodes to form a rigid bimetallic structure.

Bulk Micromachining

The development of micromachines, sensors, and actuators over the past decade has been driven by advancements in silicon microcircuits. Micromachining by the chemical etching of crystalline silicon wafers has been an important fabrication technique. Strong alkalines etch single-crystal silicon at a rate that depends on the crystal orientation, its dopant concentration, and an externally applied electric field.

The etching is controlled by photolithographic etch masks that are applied over silicon which has been coated with a photoresist. The photoresist can be chemically removed from those areas of the silicon that have been exposed to ultraviolet light through the transparent parts of the mask. When the silicon is exposed, it can be etched, plated, or diffused with dopants.

This bulk micromachining process has been combined with methods for fusion-bonding silicon substrates to form precise three-dimensional structures such as micropumps and microvalves. Two or more etched wafers can be bonded by pressing them together and annealing the structure, making the three-dimensional microstructure permanent. This approach permits internal or re-entrant cavities to be formed.

Surface Micromachining

Deposited thin films of such materials as polysilicon, silicon oxide (SiO_2), silicon nitride (Si_3N_4), and phosphosilicate glass (PSG) have been surface micromachined by both dry ionic and wet chemical etching to define those films. Freestanding structures have been formed by the removal of underlying sacrificial layers (typically of SiO_2 or PSG). The film is removed by a highly selective chemical etchant such as hydrofluoric acid after the structure layer, usually polysilicon, is deposited and patterned.

The electrostatic motor shown in Fig. 1 is made by a succession of deposition and masking steps in which alternate layers of permanent silicon material and sacrificial material are deposited until the structure of the motor—stator, rotor, and central hub—is completed. Then the sacrificial material is chemically removed, effectively sculpting the permanent silicon structure so that the rotor is free to move on the hub. At the same time, an electrostatic shield and interconnections are formed.

MULTILEVEL FABRICATION PERMITS MORE COMPLEX AND FUNCTIONAL MEMS

Researchers at Sandia National Laboratories, Albuquerque, New Mexico, have developed two surface micromachining processes for fabricating multilevel MEMS (microelectromechanical systems) from polysilicon that are more complex and functional than those made from two- and three-level processes. The processes are SUMMiT Technology, a four-level process in which one ground or electrical interconnect plane and three mechanical layers can be micromachined, and SUMMiT V Technology, a similar five-level process except that four mechanical layers can be micromachined. Sandia offers this technology under license agreement to qualified commercial IC producers.

According to Sandia researchers, polycrystalline silicon (also called polysilicon or poly) is an ideal material for making the microscopic mechanical systems. It is stronger than steel, with a strength of 2 to 3 GPa (assuming no surface flaws), whereas steel has a strength of 200 MPa to 1 GPa (depending on how it is processed). Also, polysilicon is extremely flexible, with a maximum strain before fracture of approximately 0.5%, and it does not readily fatigue.

Years of experience in working with polysilicon have been gained by commercial manufacturers of large-scale CMOS integrated circuits chips because it is used to form the gate structures of most CMOS transistors. Consequently, MEMS can be produced in large volumes at low cost in IC manufacturing facilities with standard production equipment and tools. The Sandia researchers report that because of these advantages, polysilicon surface micromachining is being pursued by many MEMS fabrication facilities.

The complexity of MEMS devices made from polysilicon is limited by the number of mechanical layers that can be deposited. For example, the simplest actuating comb drives can be made with one ground or electrical plane and one mechanical layer in a two-level process, but a three-level process with two mechanical layers permits micromachining mechanisms such as gears that rotate on hubs or movable optical mirror arrays. A four-level process such as SUMMiT permits mechanical linkages to be formed that connect actuator drives to gear trains. As a result, it is expected that entirely new kinds of complex and sophisticated micromachines will be fabricated with the five-level process.

According to the Sandia scientists, the primary difficulties encountered in forming the extra polysilicon layers for surface micromachining the more complex devices are residual film stress and device topography. The film stress can cause the mechanical layers to bow from the required flatness. This can cause the mechanism to function poorly or even prevent it from working. The scientists report that this has even been a problem in the fabrication of MEMS with only two mechanical layers.

To surmount the bowing problem, Sandia has developed a proprietary process for holding stress levels to values typically less than 5 MPa, thus permitting the successful fabrication and operation of two meshing gears, whose diameters are as large as 2000 μm.

The intricacies of device topography that make it difficult to pattern and etch successive polysilicon layers restrict the complexities of the devices that can be built successfully. Sandia has minimized that problem by developing a proprietary chemical-mechanical polishing (CMP) process called "planarizing" for forming truly flat top layers on the polysilicon. Because CMP is now so widely used in integrated circuit chip manufacture, it will allow MEMS to be batch fabricated by the SUMMiT processes using standard commercial IC fabrication equipment.

GALLERY OF MEMS ELECTRON-MICROSCOPE IMAGES

The Sandia National Laboratories, Albuquerque, New Mexico, have developed a wide range of microelectromechanical systems (MEMS). The scanning electron microscope (SEM) micrographs presented here show the range of these devices, and the captions describe their applications.

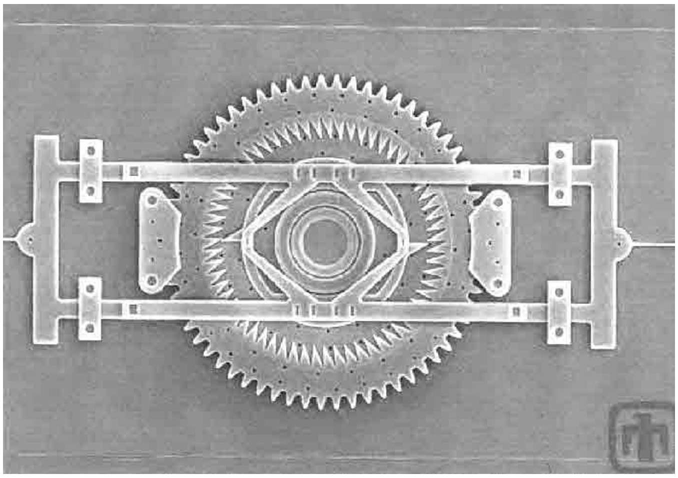


Fig. 1 Wedge Stepping Motor: This indexing motor can precisely index other MEMS components such as microgear trains. It can also position gears and index one gear tooth at a time at speeds of more than 200 teeth/s or less than 5 ms/step. An input of two simple input pulse signals will operate it. This motor can index gears in MEMS such as locking devices, counters, and odometers. It was built with Sandia's four-layer SUMMiT technology. Torque and indexing precision increase as the device is scaled up in size.

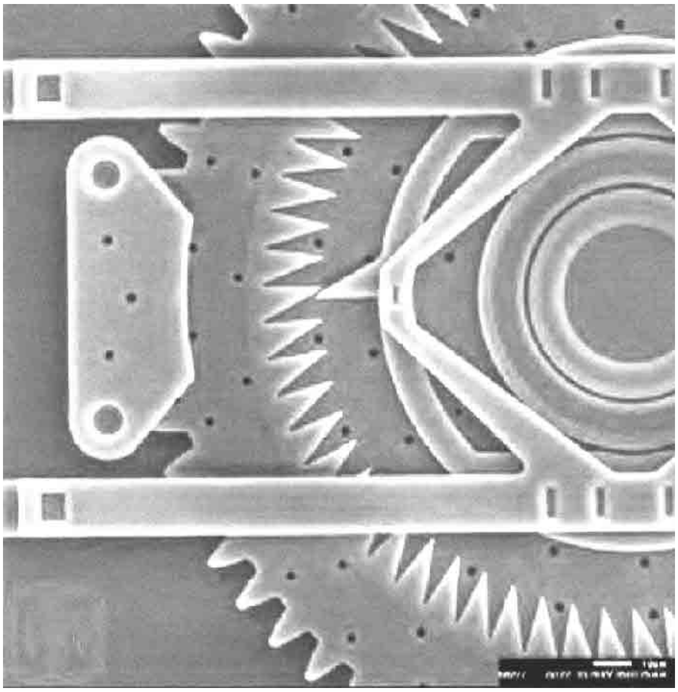


Fig. 2 Wedge Stepping Motor: A close-up view of one of the teeth of the indexing motor shown in Fig. 1.

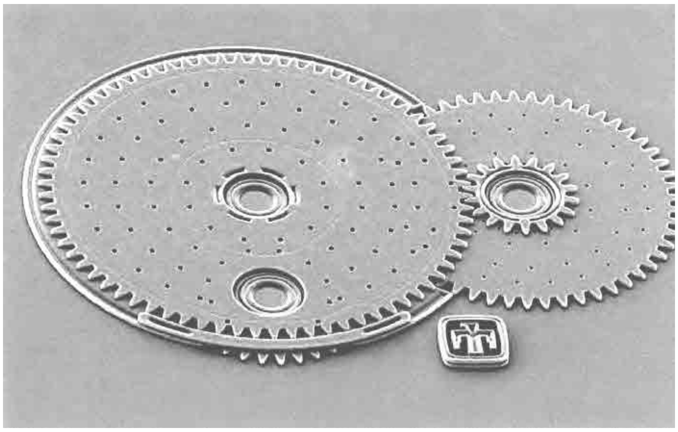


Fig. 3 Torque Converter: This modular transmission unit has an overall gear reduction ratio of 12 to 1. It consists of two multilevel gears, one with a gear reduction ratio of 3 to 1 and the other with a ratio of 4 to 1. A coupling gear within the unit permits cascading.

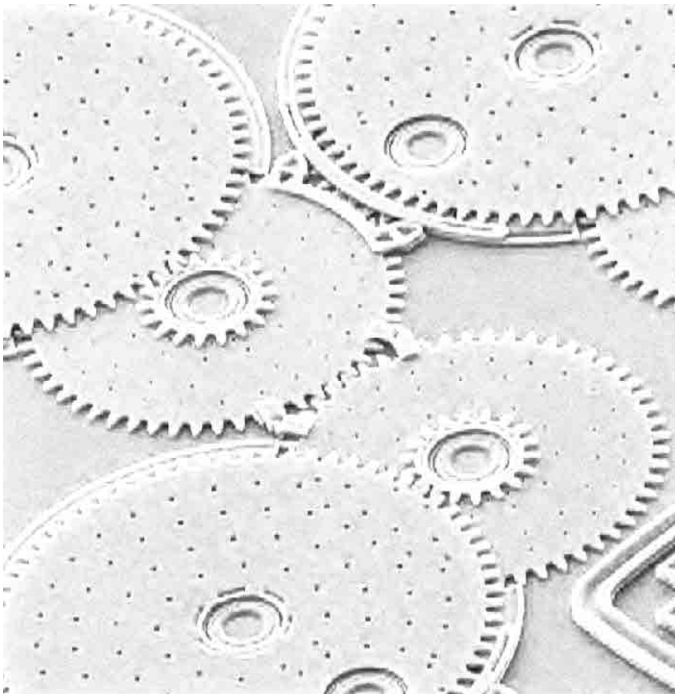


Fig. 4 Torque Converter: By cascading six stages of the modular 12-to-1 transmission units shown in Fig. 3, a 2,985,894-to-1 gear reduction ratio is obtained in a die area of less than 1 mm^2. The converter can step up or step down.

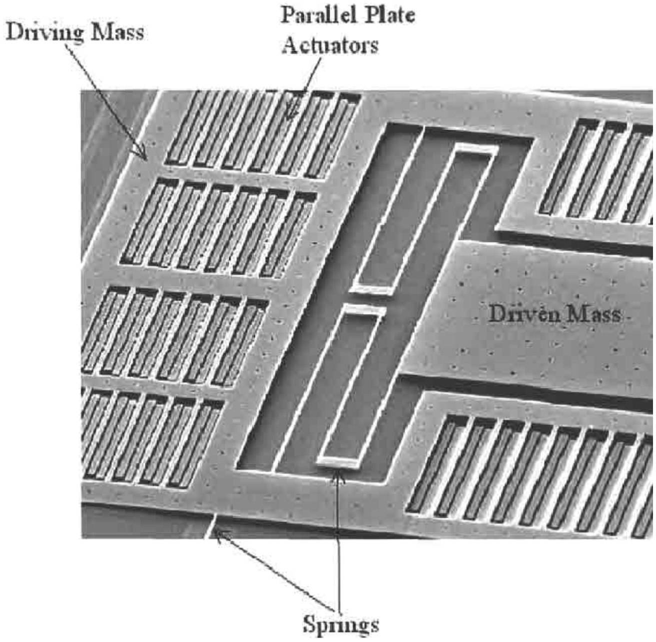


Fig. 5 Dual-Mass Oscillator: This oscillator uses parallel plate actuation and system dynamics to amplify motion. The 10-mm-long parallel plate actuators on the driving mass produce an amplified motion on the second mass when it is driven by a signal. The actuated mass remains nearly motionless, while the moving mass has an amplitude of approximately 4 μm when driven by a 4-V signal. It was designed to be part of a vibrating gyroscope.

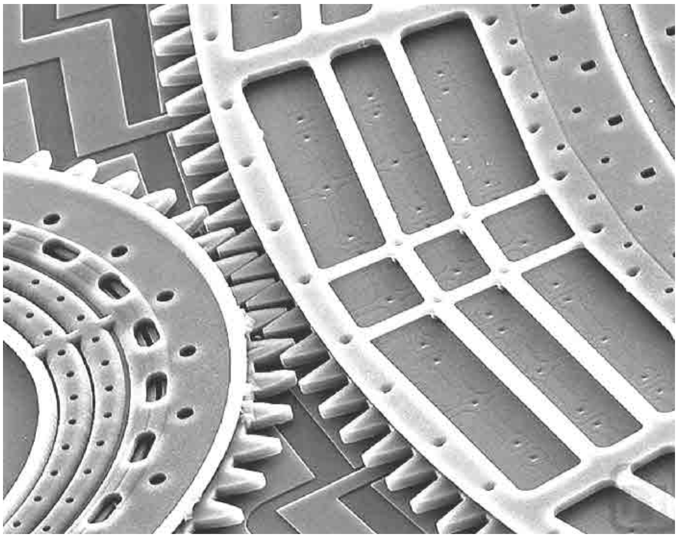


Fig. 6 Rotary Motor: This close-up shows part of a rotary motor that offers advantages over other MEMS actuators. Its operates on linear-comb drive principles, but the combs are bent in a circle to permit unlimited travel. The combs are embedded inside the rotor so that other micromachines can be powered directly from the rotor's perimeter. Built by Sandia's four-level SUMMiT technology, the motor is powered by a lower voltage and produces higher output torque than other MEMS actuators, but it still occupies a very small footprint. It can also operate as a stepper motor for precise positioning applications.

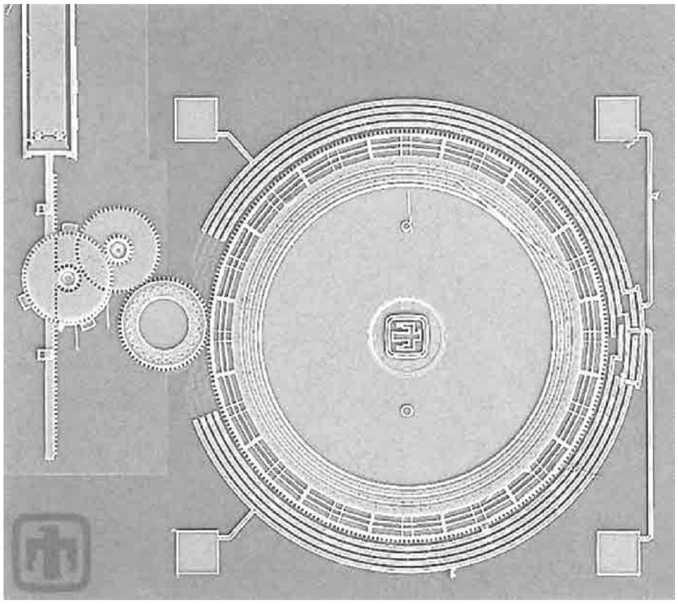


Fig. 7 Comb Drive Actuation: Two sets of comb-drive actuators (not shown) drive a set of linkages (upper right) to a set of rotary gears. The comb-drive actuators drive the linkages 90° out of phase with each other to rotate the small 19-tooth gear at rotational speeds in excess of 300,000 rpm. The operational lifetime of these small devices can exceed 8×10^9 revolutions. The smaller gear drives a larger 57-tooth (1.6-mm-diameter) gear that has been driven as fast as 4800 rpm.

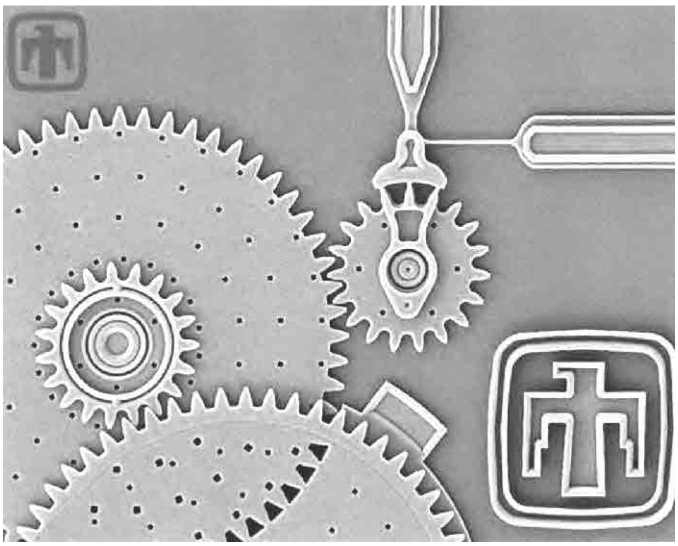


Fig. 8 Micro Transmission: This transmission has sets of small and large gears mounted on the same shaft so that they interlock with other sets of gears to transfer power while providing torque multiplication and speed reduction. Its output gear is coupled to a double-level gear train.

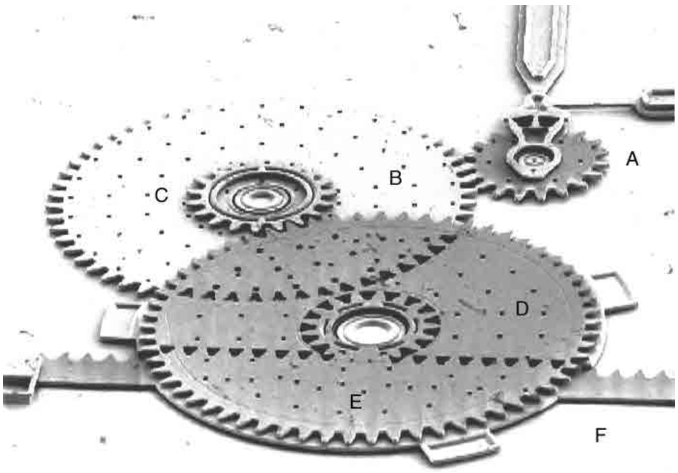


Fig. 9 Microtransmission and Gear Reduction Unit: This mechanism is the same as that in Fig. 8 except that it performs a gear-reduction function. The microengine pinion gear, labeled A in the figure, meshes directly with the large 57-tooth gear, labeled B. A smaller 19-tooth gear, C, is positioned on top of gear B and is linked to B's hub. Because the gears are joined, both make the same number of turns per minute. The small gear essentially transmits the power of the larger gear over a shorter distance to turn the larger 61-tooth gear D. Two of the gear pairs (B and C, D and E) provide 12 times the torque of the engine. A linear rack F, capable of driving an external load, has been added to the final 17-tooth output gear E to provide a speed reduction/torque multiplication ratio of 9.6 to 1.

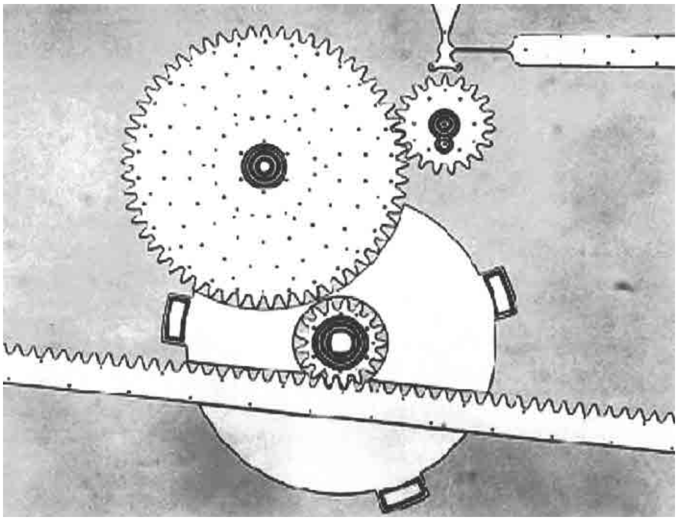


Fig. 10 Gear-Reduction Units: This micrograph shows the three lower-level gears (A, B, and E) as well as the rack (F) of the system shown in Fig. 9. The large flat area on the lower gear provides a planar surface for the fabrication of the large, upper-level 61-tooth gear (D).

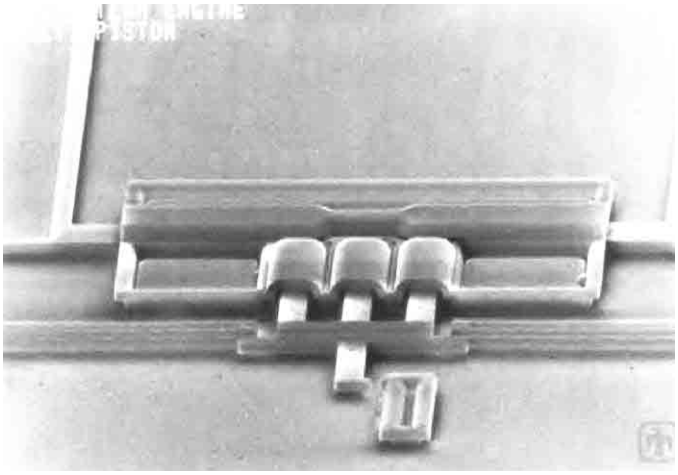


Fig. 11 Microsteam Engine: This is the world's smallest multipiston microsteam engine. Water inside the three compression cylinders is heated by electric current, and when it vaporizes, it pushes the pistons out. Capillary forces then retract the piston once current is removed.

MINIATURE MULTISPEED TRANSMISSIONS FOR SMALL MOTORS

Transmissions would be batch-fabricated using micromachining technologies.

NASA's Jet Propulsion Laboratory, Pasadena, California

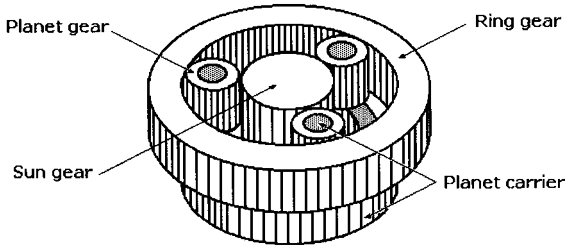


Fig. 1 Simple epicyclic gear train. Compound epicyclic gears in traditional automatic transmissions usually consist of simple epicyclics which are stacked one on top of the other along a radial axis.

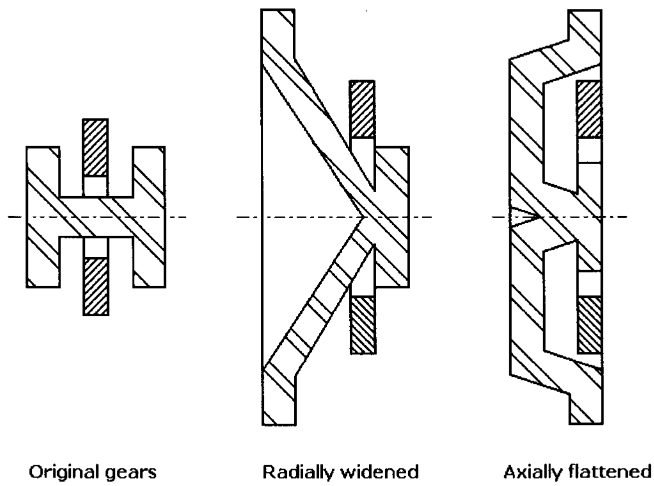


Fig. 2 Evolutionary stages in converting conventional gears to axially flattened gears.

A design has been developed for manufacturing multispeed transmissions that are small enough to be used with minimotors—electromagnetic motors with power ratings of less than 1 W. Like similar, larger systems, such as those in automobiles, the proposed mechanism could be used to satisfy a wider dynamic range than could be achieved with fixed-ratio gearing. However, whereas typical transmission components are machined individually and then assembled, this device would be made using silicon batch-fabrication techniques, similar to those used to manufacture integrated circuits and sensors.

Until now, only fixed-ratio gear trains have been available for minimotors, affording no opportunity to change gears in operation to optimize for varying external conditions, or varying speed, torque, and power requirements. This is because conventional multispeed gear-train geometries and actuation techniques do not lend themselves to cost-effective miniaturization. In recent years, the advent of microelectromechanical systems (MEMS) and of micromachining techniques for making small actuators and gears has created the potential for economical mass production of multispeed transmissions for minimotors. In addition, it should be possible to integrate these mechanisms with sensors, such as tachometers and load cells, as well as circuits, to create integrated silicon systems, which could perform closed-loop speed or torque control under a variety of conditions. In comparison with a conventional motor/transmission assembly, such a package would be smaller and lighter, contain fewer parts,

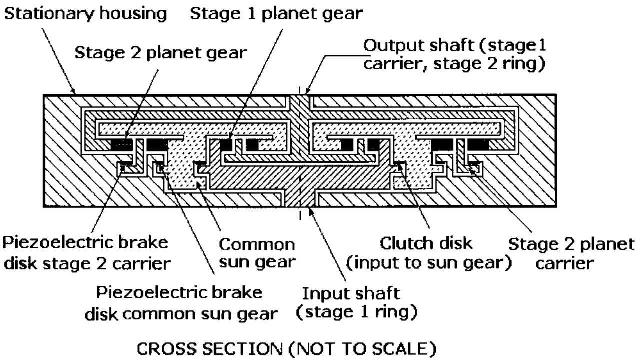


Fig. 3 This Miniature Transmission could be regarded as a flattened version of a conventional three-speed automatic transmission. The components would be fabricated by micromachining.

consume less power, and impose less of a computational burden on an external central processing unit (CPU).

Like conventional multispeed transmissions for larger motors, miniature multispeed transmissions would contain gears, clutches, and brakes. However, the designs would be more amenable to micromachining and batch fabrication. Gear stages would be nestled one inside the other (see figures 1, 2, and 3), rather than stacked one over the other, creating a more planar device. Actuators and the housing would be fabricated on separate layers. The complex mechanical linkages and bearings used to shift gears in conventional transmissions would not be practical at the small scales of interest here. Promising alternatives might include electrostatic-friction locks or piezoelectric actuators. For example, in the transmission depicted in the figure, piezoelectric clamps would serve as actuators in clutches and brakes.

The structures would be aligned and bonded, followed by a final etch to release the moving parts. The entire fabrication process can be automated, making it both precise and relatively inexpensive. The end product is a "gearbox on a chip," which can be "dropped" onto a compatible motor to make an integrated drive system.

This work was done by Indrani Chakraborty and Linda Miller of Caltech for **NASA's Jet Propulsion Laboratory**.

MEMS CHIPS BECOME INTEGRATED MICROCONTROL SYSTEMS

The successful integration of MEMS (microelectromechanical systems) on CMOS integrated circuit chips has made it possible to produce "smart" control systems whose size, weight, and power requirements are significantly lower than those for other control systems. MEMS development has previously produced microminiature motors, sensors, gear trains, valves, and other devices that easily fit on a silicon microchip, but difficulties in powering these devices has inhibited their practical applications.

MEMS surface micromachining technology is a spin-off of conventional silicon IC fabrication technology, but fundamental differences in processing steps prevented their successful integration. The objective was to put both the control circuitry and mechanical device on the same substrate. However, the results of recent development work showed that they could be successfully merged.

It has been possible for many years to integrate the transistors, resistors, capacitors, and other electronic components needed for drive, control, and signal processing circuits on a single CMOS silicon chip, and many different MEMS have been formed on separate silicon chips. However, the MEMS required external control and signal-processing circuitry. It was clear that the best way to upgrade MEMS from laboratory curiosities to practical mechanical devices was to integrate them with their control circuitry. The batch fabrication of the electrical and mechanical sections on the same chip would offer the same benefits as other large-scale ICs—increased reliability and performance. Component count could be reduced, wire-bonded connections between the sections could be eliminated, minimizing power-wasting parasitics, and standard IC packaging could replace multichip hybrid packages to reduce product cost.

MEMS sections are fabricated by multilevel polysilicon surface micromachining that permits the formation of such intricate mechanisms as linear comb-drive actuators coupled to gear trains. This technology has produced micromotors, microactuators, microlocks, microsensors, microtransmissions, and micromirrors.

Early attempts to integrate CMOS circuitry with MEMS by forming the electronic circuitry on the silicon wafer before the MEMS devices met with only limited success. The aluminum electrical interconnects required in the CMOS process could not withstand the long, high-temperature annealing cycles needed to relieve stresses built up in the polysilicon mechanical layers of the MEMS. Tungsten interconnects that could withstand those high temperatures were tried, but the performance of the CMOS circuitry was degraded when the heat altered the doping profiles in the transistor junctions.

When the MEMS were formed before the CMOS sections, the thermal problems were eliminated, but the annealing procedure tended to warp the previously flat silicon wafers. Irregularities in the flatness or planarity of the wafer distorted the many photolithographic images needed in the masking steps required in CMOS processing. Any errors in registration can lower attainable resolution and cause circuit malfunction or failure.

Experiments showed that interleaving CMOS and MEMS process steps in a compromise improved yield but limited both the complexity and performance of the resulting system. In other experiments materials such as stacked aluminum and silicon dioxide layers were substituted for polysilicon as the mechanical layers, but the results turned out to be disappointing.

Each of these approaches had some merit for specific applications, but they all resulted in low yields. The researchers persevered in their efforts until they developed a method for embedding the MEMS in a trench below the surface of the silicon wafer before fabricating the CMOS. This is the procedure that now permits the sections to be built reliably on a single silicon chip.

Sandia's IMEMS Technology

Sandia National Laboratories, Albuquerque, New Mexico, working with the University of California's Berkeley Sensor and Actuator Center (BSAC), developed the unique method for forming the micromechanical section first in a 12-μm-deep "trench"

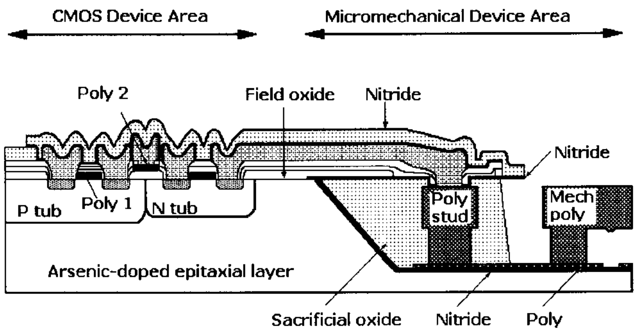


Fig. 1 A cross-section view of CMOS drive circuitry integrated on the same silicon chip with a microelectromechanical system.

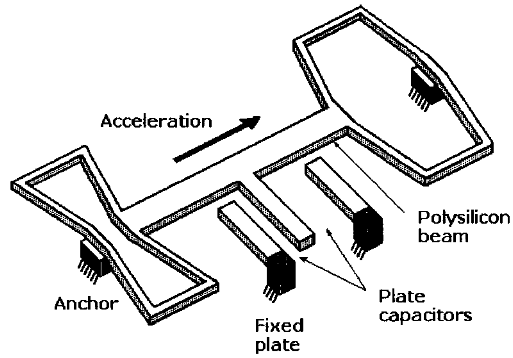


Fig. 2 A simplified view of the movement of a polysilicon beam in a surface-micromachined accelerometer moving in response to acceleration. The two fixed plates and one moving plate form a unit cell.

and backfilling that trench with sacrificial silicon dioxide before forming the electronic section. This technique, called Integrated MicroElectroMechanical Systems (IMEMS), overcame the wafer-warping problem. Figure 1 is cross-section view of both sections combined on a single chip.

The mechanical polysilicon devices are surface micromachined by methods similar to Sandia's SUMMiT process in the trench, using special photolithography methods. After the trench is filled with the silicon dioxide, the silicon wafer is annealed and that section is "planarized," or etched flat and flush with the rest of the wafer surface, by a process called chemical-mechanical polishing (CMP). After the CMOS section is complete, the sacrificial silicon dioxide in the trench is etched away, leaving the MEMS devices electrically interconnected with the adjacent CMOS circuitry.

Advantages of IMEMS

Sandia spokespersons say the IMEMS process is completely modular, meaning that the planarized wafers can be processed in any facility capable of processing CMOS, bipolar, and combinations of these processes. They add that modularity permits the mechanical devices and electronic circuitry to be optimized independently, making possible the development of high-performance microsystems.

Early Research and Development

Analog Devices Inc. (ADI) was one of the first companies to develop commercial surface-micromachined integrated-circuit accelerometers. ADI developed and marketed these accelerometer chips, demonstrating its capability and verifying commercial demand. Initially ADI built these devices by interleaving, combining, and customizing its internal manufacturing processes to produce the micromechanical devices with the same processes it used to produce monolithic electronic circuitry.

At the same time, researchers at BSAC developed the alternative process for replacing conventional aluminum interconnect layers with tungsten layers to enable the CMOS device to withstand the higher thermal stresses associated with subsequent micromechanical device processing. This process was later superseded by the joint BSAC–Sandia development of IMEMS.

Accelerometers

ADI offered the single-axis ADXL150 and dual-axis ADXL250, and Motorola Inc. offered the XMMAS40GWB. Both of ADI's integrated accelerometers are rated for ±5 *g* to ±50 *g*. They have been in high-volume production since 1993. The company is now licensed to use Sandia's integrated MEMS/CMOS technology. Motorola is now offering the MMA1201P and MMA2200W single-axis IC accelerometers rated for ±38 *g*.

These accelerometer chips differ in architecture and circuitry, but both work on the same principles. The surface micromachined sensor element is made by depositing polysilicon on a sacrificial oxide layer that is etched away, leaving the suspended sensor element. Figure 2 is a simplified view of the differential-capacitor sensor structure in an ADI accelerometer. It can be seen that two of the capacitor plates are fixed, and the center capacitor plate is on the polysilicon beam that deflects from its rest position in response to acceleration.

When the center plate deflects, its distance to one fixed plate increases while its distance to the other plate decreases. The change in distance is measured by the on-chip circuitry that converts it to a voltage proportional to acceleration. All of the circuitry, including a switched-capacitor filter needed to drive the sensor and convert the capacitance change to voltage, is on the chip. The only external component required is a decoupling capacitor.

Integrated-circuit accelerometers are now used primarily as airbag-deployment sensors in automobiles, but they are also finding many other applications. For example, they can be used to monitor and record vibration, control appliances, monitor the condition of mechanical bearings, and protect computer hard drives.

Three-Axis Inertial System

When the Defense Advanced Research Projects Agency (DARPA), an agency of the U.S. Department of Defense, initiated a program to develop a solid-state three-axis inertial measurement system, it found that the commercial IC accelerometers were not suitable components for the system it envisioned for two reasons: the accelerometers must be manually aligned and assembled, and this could result in unwanted variations in alignment, and the ICs lacked on-chip analog-to-digital converters (ADCs), so they could not meet DARPA's critical sensitivity specifications.

To overcome these limitations, BSAC designed a three-axis, force-balanced accelerometer system-on-a-chip for fabrication with Sandia's modular monolithic integration methods. It is said to exhibit an order of magnitude increase in sensitivity over the best commercially available single-axis integrated accelerometers. The Berkeley system also includes clock generation circuitry, a digital output, and photolithographic alignment of sense axes. Thus, the system provides full three-axis inertial measurement, and does not require the manual assembly and alignment of sense axes.

A combined *X*- and *Y*-axis rate gyro and a *Z*-axis rate gyro was also designed by researchers at BSAC. By using IMEMS

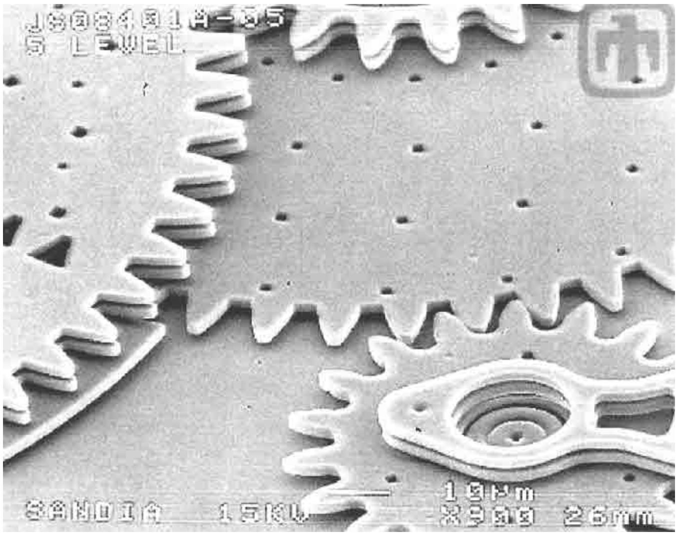


Fig. 3 This linear-rack gear reduction drive converts the rotational motion of a pinion gear to linear motion to drive a rack.
Courtesy of Sandia National Laboratories

technology, a full six-axis inertial measurement unit on a single chip was obtained. The 4- by 10-mm system is fabricated on the same silicon substrate as the three-axis accelerometer, and that chip will form the core of a future micro–navigation system. BSAC is teamed with ADI and Sandia Laboratories in this effort, with funds provided by DARPA's Microsystems Technology Office.

Micromechanical Actuators

Micromechanical actuators have not attained the popularity in commercial applications achieved by microminiature accelerometers, valves, and pressure sensors. The two principal drawbacks to their wider application have been their low torque characteristics and the difficulties encountered in coupling actuators to drive circuitry. Sandia has developed devices that can be made by its SUMMiT four-level polysilicon surface-micromachining process, such as the microengine pinion gear driving a 10 to 1 transmission shown in Fig. 3, to improve torque characteristics.

The SUMMiT process includes three mechanical layers of polysilicon in addition to a stationary level for grounding or electrical interconnection. These levels are separated by sacrificial silicon-dioxide layers. A total of eight mask levels are used in this process. An additional friction-reducing layer of silicon nitride is placed between the layers to form bearing surfaces.

If a drive comb, operating at a frequency of about 250,000 rpm, drives a 10-to-1 gear reduction unit, torque is traded off for speed. Torque is increased by a factor of 10 while speed is reduced to about 25,000 rpm. A second 10-to-1 gear reduction would increase torque by a factor of 100 while reducing speed to 2,500 rpm. That gear drives a rack and pinion slider that provides high-force linear motion. This gear train provides a speed-reduction/torque-multiplication ratio of 9.6 to 1.

LIGA: AN ALTERNATIVE METHOD FOR MAKING MICROMINIATURE PARTS

The Sandia National Laboratories, Livermore, California, is using a process called LIGA to form microminiature metal components as an alternative to producing them by the surface micromachining processes used to make microelectromechanical systems (MEMS). LIGA permits the fabrication of larger, thicker, and more durable components with greater height-to-width ratios. They can withstand high pressure and temperature excursions while providing more useful torques than polysilicon MEMS.

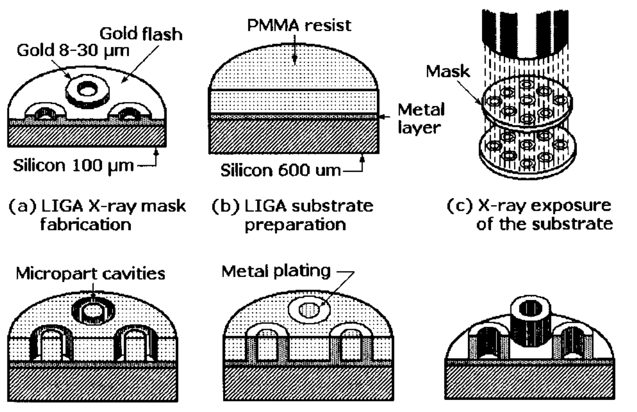


Fig. 1 Steps in fabricating microminiature parts by the LIGA process.

The acronym LIGA is derived from the German words for lithography, electroplating, and molding (Lithographie, Galvanoformung, and Abformung), a micromachining process originally developed at the Karlsruhe Nuclear Research Center in Karlsruhe, Germany, in the 1980s. Sandia Labs has produced a wide variety of LIGA microparts, including components for millimotors and miniature stepping motors. It has also made miniature accelerometers, robotic grippers, a heat exchanger, and a mass spectrometer. Sandia is carrying out an ongoing research and development program to improve the LIGA process and form practical microparts for various applications.

In the LIGA process, highly parallel X-rays from a synchrotron are focused through a mask containing thin 2D templates of the microparts to be formed. The X rays transfer the patterns to a substrate layered with PMMA (polymethylmethacrylate), a photoresist sensitive to X rays, on a metallized silicon or stainless-steel substrate. When the exposed layer of PMMA (better known as Plexiglas) is developed, the cavities left in the PMMA are the molds in which the microparts will be formed by electroplating. The thickness of the PMMA layer determines the large height-to-width ratio of the finished LIGA microparts. The resulting parts can be functional components or molds for replicating the parts in ceramic or plastic.

The highlights of the LIGA process as illustrated in Fig. 1 are

(a) An X-ray mask is prepared by a series of plating and lithographic steps. A metallized silicon wafer coated with photoresist is exposed to ultraviolet light through a preliminary mask containing the 2D patterns of the microparts to be produced. Development of the photoresist dissolves the resist from the plated surface of the wafer, forming the micropart pattern, which is plated in gold to a thickness of 8 to 30 µm. The remaining photoresist is then dissolved to finish the mask.

(b) Target substrate for forming microparts is prepared by solvent-bonding a layer of PMMA to a metallized-silicon or stainless-steel substrate.

(c) PMMA-coated substrate is then exposed to highly collimated parallel X rays from a synchrotron through the mask.

(d) PMMA is then chemically developed to dissolve the exposed areas down to the metallized substrate, etching deep cavities for forming microparts.

(e) Substrate is then electroplated to fill the cavities with metal, forming the microparts. The surface of the substrate is then lapped to finish the exposed surfaces of the microparts to the required height within ±5µm.

(f) Remaining PMMA is dissolved, exposing the 3D microparts, which can be separated from the metallized substrate or allowed to remain attached, depending on their application.

The penetrating power of the X rays from the synchrotron allows structures to be formed that have sharp, well-defined vertical surfaces or sidewalls. The minimum feature size is 20 µm, and microparts can be fabricated with thickness of 100 µm to 3 mm. The sidewall slope is about 1µm/mm. In addition to gold, microparts have been made from nickel, copper, nickel–iron, nickel–cobalt, and bronze.

An example of a miniature machine assembled from parts fabricated by LIGA is an electromagnetically actuated millimotor. With an 8-mm diameter and a height of 3 mm, it includes 20 LIGA parts as well as an EDM-machined permanent magnet and wound coils. The millimotor has run at speeds up to 1600 rpm, and it can provide torque in excess of 1 mN-m. Another example of a miniature machine built from LIGA parts is a size 5 stepper motor able to step in 1.8-deg increments. Both its rotor and stator were made from stacks of 50 laminations, each 1-mm thick.

According to Sandia researchers, the LIGA process is versatile enough to be an alternative to such precision machining methods as wire EDM for making miniature parts. The feature definition, radius, and sidewall texture produced by LIGA are said to be superior to those obtained by any precision metal cutting technique.

In an effort to form LIGA parts with higher aspect ratios, researchers at the University of Wisconsin in Madison teamed with the Brookhaven National Laboratory on Long Island to use the laboratory's 20,000-eV photon source to produce much higher levels of X-ray radiation than are used in other LIGA processes. The higher-energy X rays penetrate into the photoresist to depths of 1 cm or more, and they also pass more easily through the mask. This permitted the Wisconsin team to use thicker and stronger materials to make 4-in.-square masks rather than the standard 1- × 6-cm masks used in standard LIGA. Working with Honeywell, the team developed LIGA optical microswitches

The primary disadvantage to LIGA is its requirement for a synchrotron or other high-energy sources to image parallel X-rays on the PMMA covered substrate. In addition to their limited availability, these sources are expensive to build, install, and operate. Their use adds significantly to the cost of producing LIGA microstructures, especially for commercial applications.

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