Chapter 10

Actuation and Actuating Systems

10.1. ACTUATOR AND ACTUATION

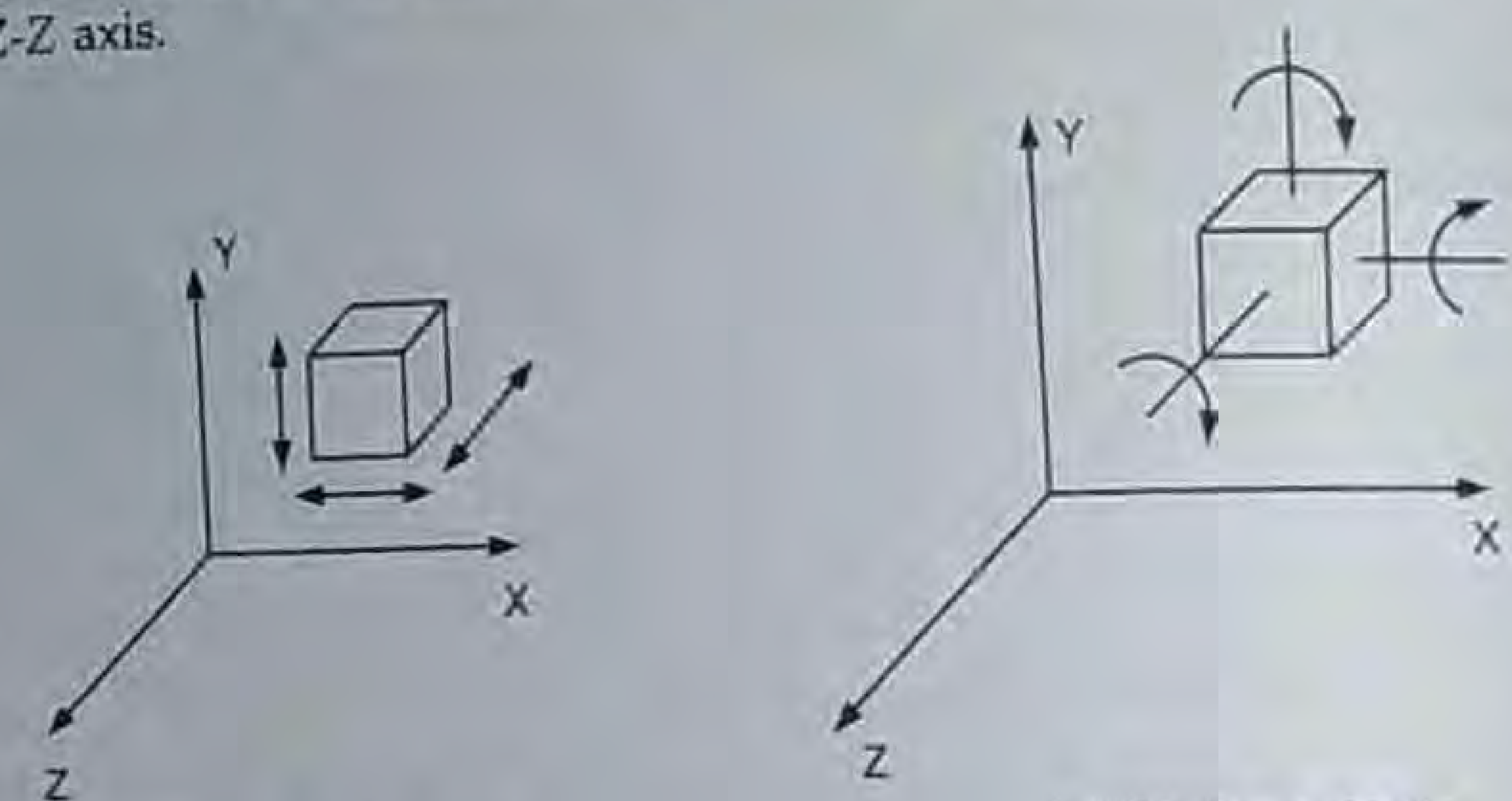
An **actuator** is an energy conversion device that makes somethings move or operate. Essentially it uses a form of power/energy to convert a control signal into mechanical motion, and the process of energy conversion to mechanical form is called **actuation**.

The actuators are encountered at house and at workplace. For example :

- automatic opening of door when a person enters a grocery shop.
- forward or backward movement of a car seat.

Actuators, are located in from electric door lock in automobiles to ailerons in air-crafts. Industrial plants use actuators to operate the valves, dampers and fluid couplings etc.

Actuators are classified by the type of motion and the power source. Linear actuators produce pull or push action and the linear motion may be along one or more of the three axes X-X axis, Y-Y axis and Z-Z axis.



(a) Translational motion (b) Rotational motion

Fig. 10.1 Types of motion

The rotational motion is the motion of a rigid body along one or more of the three axis along the *X-X* axis, *Y-Y* axis and *Z-Z* axis.

Depending upon the power source, actuators are categorized as :

- *Electromechanical actuators* : These actuators use AC, DC or stepper electric motors to convert electrical energy into mechanical work.
- *Fluid power actuators* : These actuators use fluid energy that is transmitted through a fluid under pressure. The fluid used is a liquid (water or oil) in a hydraulic actuator, and compressed air or some inert gas in a pneumatic actuator.

- *Active material based actuators* : The operation of these transducers is based on the fact that some materials like piezoelectric and magnetostrictive materials undergo some change in their character when subjected to some physical interaction. The piezoelectric material undergoes a dimensional change when voltage is applied and that dimensional change is utilized to produce actuators. The magnetostrictive material changes its shape when it is subjected to a magnetic field.

Though a new field, the active material based actuators have been successfully used in biomedical equipment, fluid control devices, aerospace and automotive monitoring systems, precision manufacturing and process monitoring equipment.

10.2 MECHANICAL ACTUATION SYSTEMS

Mechanical actuation systems are essentially the mechanisms formed by assembling a number of rigid bodies in such a way that the motion of one member causes the constrained and predictable motion of the other member. A modification in motion and its transfer from one location to another is achieved through application of specially designed rigid interfacing components and units. Such rigid bodies are called *mechanical components* and these are categorized as :

- passive components
- active components

The passive components do not transfer the mechanical power and include : nut-bolt, screws, springs and washers. The active components serve to transmit power in terms of force and torque, and motion regarding its speed and direction. Examples of active components are : kinematic chains, belt and chain drive, gears and gear trains, cams and followers, bearings, and ratchet-pawl mechanism etc.

The different applications of these active mechanical components are :

- conversion of the reciprocating/linear motion of the piston into rotational motion of the crankshaft in an internal combustion engine.
- transformation of the rotational motion of the cam into translatory motion of the follower in the cam-follower arrangement.
- change in magnitude and direction of speed by using gear train.
- transformation of motion in one direction into motion in another direction at right angle to it by using bevel gear.
- change of linear motion to circular motion and vice versa by the slider-crank mechanism.

The employment of these mechanisms within a mechanical system may also provide mechanical advantage, *i.e.*, amplify the force. The term *loading* associated with these active components refers to several factors such as :

- force, torque and speed of rotation.
- accuracy, precision and power consumption.

These units have been discussed in the following sections regarding their purpose, construction and operation.

10.3. KINEMATIC LINK, KINEMATIC PAIR, MECHANISM AND MACHINE

Each part of a machine which connects other parts having motion relative to it is called a kinematic link or simply a link. A link may also consist of a number of parts which are so connected that they form one unit and have no motion relative to each other.

With reference to the piston-cylinder arrangement of a reciprocating steam engine shown in Fig. 10.2.

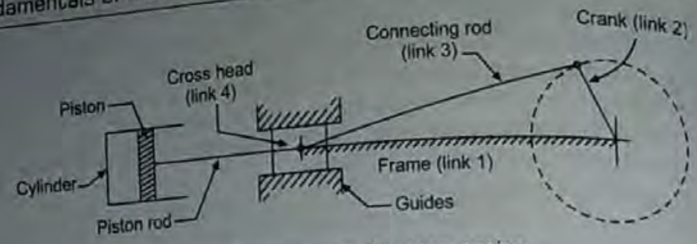


Fig. 10.2. Schematics of a steam engine

(*i*) Crank, connecting rod, cylinder and piston of steam engine constitute four links. Each element is a separate link moving relative to one another.

Link 1 is the fixed link and includes the engine frame and all other stationary parts like cylinder, and main bearings.

Link 2 includes crankshaft and flywheel, both having motion of rotation about the fixed axis.

Link 3 is the connecting rod having oscillatory motion.

Link 4 corresponds to the piston, piston rod and cross-head having reciprocating rectilinear translatory motion.

(*ii*) Piston, piston rod and cross head of a steam engine are rigidly fastened together and do not move relative to one another. They constitute one unit and are taken as one link.

(*iii*) Connecting rod, big and small end bearings, caps and bolts constitute one unit and are taken as one link.

The links are generally classified as:

(*i*) Rigid link: a link which exhibits no relative deformation between two parts when it is acted upon by a force system. The distance between any two particles remains constant, i.e., the size and the shape of the link do not change.

A link need not be a rigid body but it must be a resistant body. A body is said to be resistant if it is capable of transmitting the required motion with negligible deformation.

(*ii*) Flexible link: a link which gets only partly deformed while transmitting motion. However, a flexible link does not affect transmission of motion which is either a pull or a push in one direction only. The typical examples of flexible link are the belts and springs.

(*iii*) Fluid link: a link formed by having fluid in a container. The motion to other components is through the fluid by pressure or compression. Hydraulic brakes, jacks and presses represent fluid links.

10.3.1. Mechanism, machine and structure

Mechanism is an assemblage of number of bodies (usually rigid) assembled in such a way that motion of one causes constrained and predictable motion to the other. The function of a mechanism is to transmit and modify motion. Some examples of mechanism are type writers, watches, clocks and spring toys.

Machine is a mechanism or a combination of mechanisms which (apart from imparting definite motion to the parts) transmits and modifies the available energy into useful work. Steam engines, reciprocating compressors and pumps are the machines derived from slider crank mechanism.

Structure is a combination of a number of resistant links which are meant for carrying loads or are under the influence of forces having straining action. There is neither any relative motion between the links of a structure nor any useful energy is transmitted by it. Typical examples of a structure are the machine frame, bridge and the structure of a roof.

10.3.2. Kinematic pair and types of motion

A *kinematic pair* is a joint of two links or elements of a machine which are in contact and have relative motion between them.

With reference to the reciprocating engine, the kinematic pairs formed by different links are:

1. Crankshaft with bearings which are fixed
2. Crank with connecting rod
3. Connecting rod with piston
4. Piston with cylinder

Further, the relative motion between the two links of a kinematic pair has to be completely or successfully constrained to make the required pair.

(*a*) **Completely constrained motion:** Motion between the pair is limited to a definite direction, irrespective of the direction of the force applied. The constrained motion is completed by its own links.

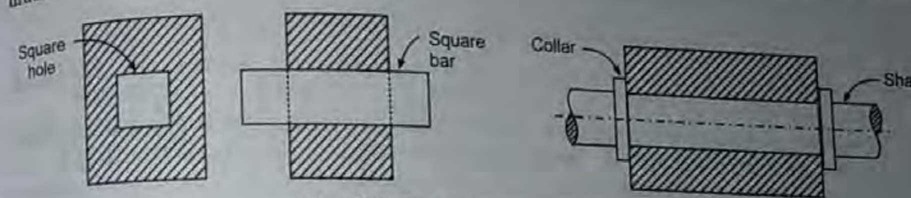


Fig. 10.3. Completely constrained motion

Examples of a completely constrained motion are

– rectangular bar in a rectangular hole
– shaft with collars at each end rotating in a round hole
– piston and a cylinder in a steam engine; here the piston has only a reciprocating motion relative to the cylinder irrespective of the direction of rotation of crank.

(*b*) **Incompletely constrained motion:** Motion between the pair is in more than one direction. There occurs a change in the direction of relative motion with change in the direction of impressed force.

A circular bar moving in a round hole provides incompletely constrained motion. Here the bar can reciprocate or rotate, and both these motions are independent of one another.

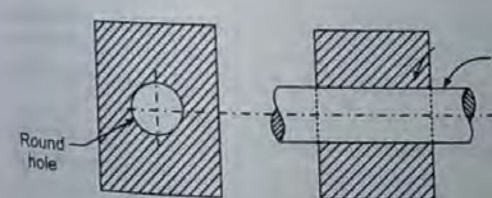


Fig. 10.4. Incompletely constrained

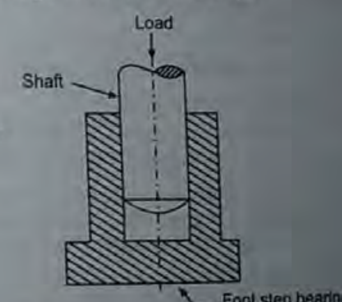


Fig. 10.5. Partially constrained motion
(Foot step bearing)

(c) **Partially or successfully constrained motion:** Motion between the elements forming the pair is not completed by itself but is done so by some other means.

The motion of a circular shaft in a foot step bearing is made completely constrained by applying adequate force as shown in Fig. 10.5.

Other examples of partially constrained motion are:

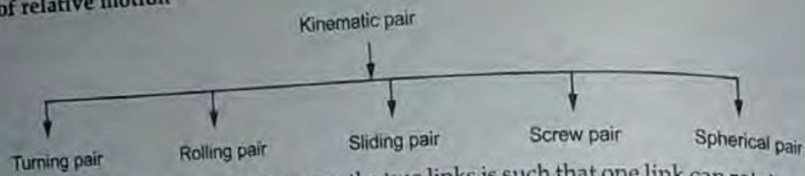
(i) rotor of a vertical turbine

(ii) IC engine valves which are kept on their seats by a spring.

10.3.3. Classification of a kinematic pair

A *kinematic pair* may be classified according to type of relative motion, type of contact and type of closure.

1. Type of relative motion



(i) *Turning pair:* The connection between the two links is such that one link can rotate or oscillate about the fixed axis of another link. The turning permits only one degree of freedom. This pair is also called a hinge, a pin joint or a revolute pair. Examples are:

- lathe spindle supported in head stock
- cycle wheels turning on its axles.
- shaft, with collars at both ends, fitted into a circular

(ii) *Rolling pair:* The connection between the two links in such way that one link can roll over another link, which is fixed.

Ball and roller bearings, wheels of locomotive and castor wheel of trolleys, etc., constitute the rolling pair. In a ball bearing, the ball and the shaft constitute one rolling pair whereas ball and bearing is another rolling pair.

(iii) *Slitting or prismatic pair:* The two elements of the pair are connected in such a way that one link can purely slide over another link. There is a relative motion of sliding only in one direction (along a line) and as such only one degree of freedom.

Examples of sliding pair are:

- piston and cylinder
- ram and its guides in a shaper
- tailstock on the lathe bed
- cross-head and its guides in a reciprocating steam engine

(iv) *Screw pair:* The connection between the links is such that one link can turn about another link by means of screw threads. Both axial sliding and rotational motions are involved. However, the sliding and rotational motions are related through helix angle, and the pair is said to have one degree of freedom. The rotating lead screw operating in nuts for accurate transmission of motion as in lathes, machine tools and measuring instruments form the screw pair.

(v) *Spherical or globul pair:* One of the elements is in the form of a sphere and it turns or swivels about the other element which is fixed. For a given position of spherical pair, the joint permits relative motion about three mutually perpendicular axes, and so has three degrees of freedom. Ball and socket joint, pen stand, and the attachment of a car mirror represent the spherical pairs.

(vi) *Cylindrical pair:* The relative motion in a cylindrical pair is a combination of rotation and translation parallel to the axis of rotation between the contacting elements. Apparently, the cylindrical pair has two degrees of freedom.

A shaft free to rotate in a bearing and also free to slide axially inside the bearing provides a cylindrical pair.

2. Type of contact

The links of a lower pair have surface contact while in motion and the relative motion is purely turning or sliding. Examples are:

- lathe spindle supported in headstock
- shaft revolving in a bearing
- straight line motion mechanisms
- Universal joint and automobile steering gear.

The links of a *higher pair* have point or line contact, and the relative motion is a combination of sliding and turning. All sliding, screw, spherical, cylindrical, revolute pairs fall in the category of lower pairs. Examples are:

- ball and roller bearings
- cam—follower
- belt, rope and chain drives
- meshing gear-teeth

Fig. 10.6. Lower and higher pair

With reference to Fig. 10.6 where link 1 is fixed:

- there is asurface contact between link 1 and link 2 and they constitute a lower pair.
- links 2 and 3 comprise a turning pair
- the pair of links 3 and 4 represent a higher pair due to a line contact between them.

3. Types of closure

A self-closed pair is formed when the elements constituting the pair are held together mechanically and only required kind of motion occurs. The lower pairs represent self-closed pairs.

There is no mechanical connection between the two links of the unclosed or force closed pair. The contact is maintained by an external force which may be either due to gravity or by spring action. Cam and follower are connected by forces exerted by spring and gravity and form an unclosed pair.

Fig. 10.7. Self-closed and force-closed pair

With reference to Fig. 10.7, we have:

- The links 1–2, 2 - 3 and 3 - 4 form a closed pair
- The links 4 - 1 is a force-closed pair

10.4. KINEMATIC CHAINS AND THEIR INVERSIONS

A kinematic chain is such a combination or coupling between kinematic pairs such that each link forms a part of two pairs and the relative motion between the links is completely or successfully constrained. The pairs are coupled in such a way that the last link joins the first link and a definite motion is transmitted.

Consider the following kinematic pairs formed by the links in the mechanism of a reciprocating engine:

- cylinder and piston
- piston and the connecting rod

- connecting rod and the crank
- crankshaft and the bearings

The total combination of all these pairs can be referred to as kinematic chain.

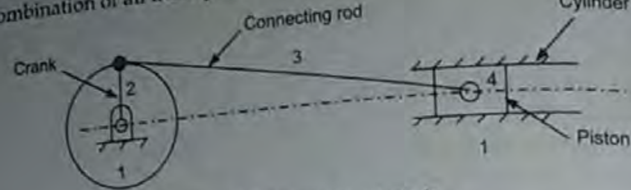


Fig. 10.8. Splider crank chain

The following correlations exist between the number of pairs (p), number of links (n or l) and the number of joints (j) forming a kinematic chain.

$$n = 2p = 4 \text{ and } j = \frac{3}{2}n - 2 \qquad ...(10.1)$$

This is on the assumption that each link forms two pairs with adjacent links. Moreover, these relations are essentially valid when the chain comprises only lower pair. Further, the following relation has been suggested by AW Klien to determine the nature of chain.

$$j = \frac{h}{2} = \frac{3}{2}n - 2... \qquad ...(10.2)$$

where j is the number of joints, h is the number of higher pairs and n is the number of links. When applying the above relations to a chain having ternary and quaternary joints

1 ternary joint = 2 binary joints

2 quaternary joint = 3 binary joints

- The chain is called a locked chain when the LHS is greater than RHS. The locked chain represents a frame or structure which is used in bridges and structures.
- The chain is called kinematic chain when LHS = RHS
- The chain is called unconstrained chain when LHS < RHS

Consider an arrangement of three links

AB, BC and CA with pin joints at A, B and C (Fig. 10.9). Here

$$n = 3;\ p = 3 \text{ and } j = 3$$

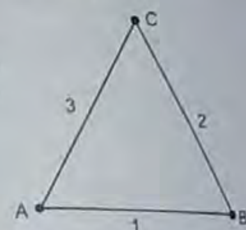


Fig. 10.9. Three-link mechanism

Applying the condition: $n = 2p - 4$

$$3 = 2 \times 3 - 4;\ 3 = 2, \text{ i.e., LHS > RHS}$$

Applying the condition: $j = \frac{3}{2}n - 2$

$$3 = \frac{3}{2} \times 3 - 2;\ 3 = 2.5, \text{ i.e., LHS > RHS}$$

The arrangement of three links does not satisfy the necessary conditions and obviously does not form a kinematic chain. Such type of chain where LHS > RHS is called ***locked chain***. This chain forms a rigid structure (no possibility of relative motion) and is used in trusses and bridges.

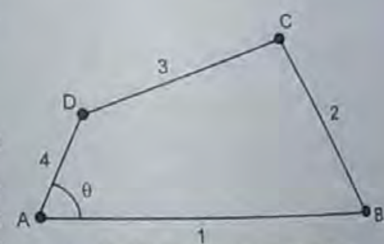


Fig. 10.10. Four-link mechanism

Consider an arrangement of four links AB, BC, CD and DA with joints at A, B, C and D (Fig. 10.10). Here

$$n = 4;\ p = 4 \text{ and } j = 4$$

Applying the conditions prescribed by equation 10.1, we get

$$n = 2p - 4;\ 4 = 2 \times 4 - 4 = 4, \text{ LHS = RHS}$$

and

$$j = \frac{3}{2}n - 2;\ 4 = \frac{3}{2} \times 4 - 2 = 4, \text{ LHS = RHS}$$

The arrangement of four links satisfies the necessary conditions and obviously forms a kinematic chain. Further, the position of a single link such as AD is sufficient to define the position of other links. The four bar arrangement is then called a kinematic chain of one degree freedom.

Fig. 10.11. Five-link mechanism

When the conditions of equations 10.1 and 10.2 are applied to an arrangement of five links (Fig. 10.11), it will be found that LHS < RHS. The equality condition is thus not satisfied and accordingly the five-bar arrangement does not form a kinematic chain.

Such an arrangement where LHS < RHS is called unconstrained chain, i.e., the relative motion is not completely constrained.

An arrangement of six links would satisfy the necessary conditions (LHS = RHS), and therefore forms a kinematic chain.

A kinematic chain having more than four links is called the compound kinematic chain.

The different types of joints as found in a chain are classified as :

(*i*) *Binary joint:* Two links are joined at the same connection. The chain shown in Fig. 10.10 has four links and four binary joints at A, B, C and D.

(*ii*) *Ternary joint:* Three links are joined at the same connection. The chain shown in Fig. 10.12 (*a*) has six links, three binary joints (A, B and D) and two ternary joints (C and E).

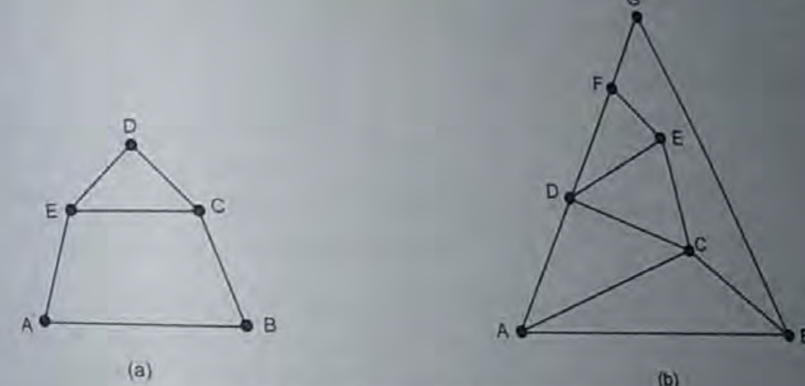


Fig. 10.12. Ternary and quaternary joints

(*iii*) *Quaternary joint:* Four links are joint at the same connection. The chain shown in Fig. 10.12 (*b*) has 11 links, one binary joint (G), four ternary joints (A, B, E and F) and two quaternary joints at (C and D).

EXAMPLE 10.1

Check the nature of chain and identify the links as binary, ternary or so with reference to the link arrangement as shown in Fig. 10.13.

Solution: The given assemblage of links has:

10 links, one binary joint (G) and six ternary joints (A, B, C, D, E and F). Since ternary joint equals 2 binary joints, the total number of binary joints in the system is

$$1 + 2 \times 6 = 13$$

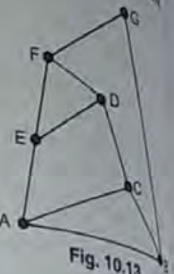


Fig. 10.13.

Applying the equation, $j = \frac{3}{2}n - 2$, we have

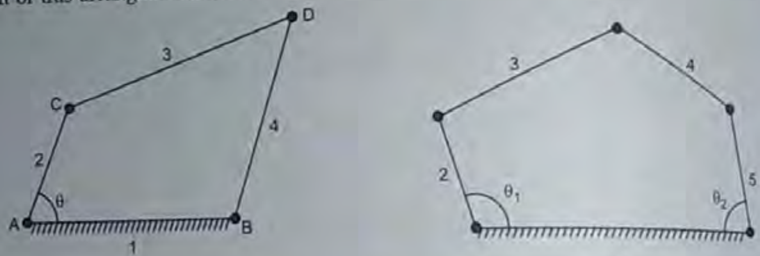
$$13 = \frac{3}{2} \times 10 - 2 = 13$$

Since left hand side is equal to right hand side, the given arrangement satisfies the necessary condition and obviously represents a kinematic or constrained chain.

Degree of freedom

The **degree of freedom** of a mechanism is defined as the number of inputs that need to be independently controlled to have a constrained motion of the other links, i.e., mechanism can be brought to serve a useful technological purpose.

With reference to Fig. 10.14 (*a*), the angle θ is sufficient to define the relative motion of all links and accordingly this mechanism has a single degree of freedom. The five-link mechanism depicted in 10.14 (*b*), needs both θ_1, an θ_2 to define the motion of all links and so the degree of freedom of this arrangement is two.



(a) (b)

Fig. 10.14. Concept of degree of freedom

The number of freedom of some of the systems most commonly involved in mechanisms and machines are given below:

(*i*) A rigid body has no restraints and has six degrees of freedom.

(*ii*) A circular shaft rotating in a hole and also having motion of translation parallel to axis of rotation has two degrees of freedom; angle turned and displacement.

(*iii*) A rectangular bar sliding in a rectangular hole has one degree of freedom; linear displacement only.

(*iv*) The resolute or turning pair has a single degree of freedom; this connection allows only a relative rotation between the elements 1 and 2.

(*v*) A ball and a socket joint has three degrees of freedom.

(*vi*) A screw pair has one degree of freedom; the relative motion between elements 1 and 2 is only rotational.

(*vii*) The position of the crank of a slider crank mechnism has one degree of freedom; position of crank is expressed by the angle through which it has turned.

Refer to Fig. 10.15 for the different mechanisms that would be obtained when one link of a four- bar kinematic chain is fixed at a time. Though all the four inversions look identical, yet their mobility changes when the proportions of various links are suitably altered.

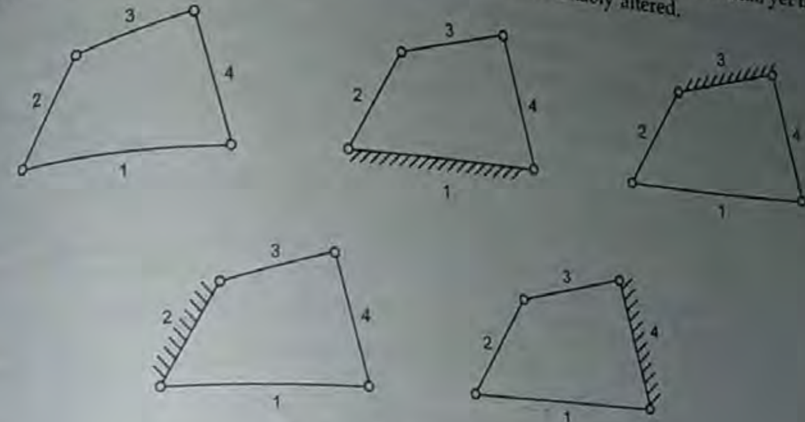


Fig. 10.15. Inversions of a four-bar chain

The main properties of inversion are :

1. The number of inversions that are possible is equal to the number of links in the parent kinematic chain.
2. The relative motion (displacement velocity and acceleration) between different links is a property of the parent kinematic chain and as such the relative motion between any two links would not change with inversion.
3. There may occur drastic change in the absolute motion of points on various links (measured with respect to the fixed link) from one inversion to the other.

A mechanism is formed by fixing one of the links of a chain and apparently different mechanisms result when different links of the same chain are chosen to become the fixed link. The process of choosing different links to become the fixed link is called kinematic inversion and the mechanisms obtained by fixing different links of a kinematic chain are called its *inversions*. Even though a four bar chain with single degree of freedom constitutes a kinematic chain, there are other chains also which provide inversions of practical utility.

A *quadratic cycle chain* is the simplest and basic kinematic chain consisting of four links which are connected in the form of quadrilateral by four-pin joints (turning pairs). The links form turning pairs and are capable of transmitting definite or completely constrained motion. Further, for continuous relative motion between the links, the sum of the lengths of the shortest and longest links is not to be greater than the sum of the lengths of the remaining two links.

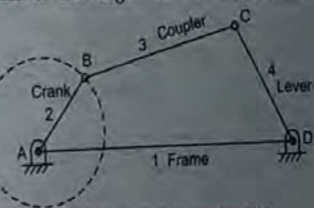


Fig. 10.16. Quadratic cycle chain

Any of the links, in particular the shortest link, which may be able to make a complete rotation is known as crank or driver. With reference to Fig. 10.16.

(*i*) The link *AD* (link 1) is fixed, and is referred to as *frame* of the mechanism.

(*ii*) The link *AB* (link 4) makes a complete rotation and is the *crank*.

(iii) The link BC (link 3) opposite to the fixed link is called the *coupler*.

(iv) The link BC (link 2) makes a partial rotation, *i.e.*, oscillates and is known as *lever* or *rocker* or *follower*.

The coupler connects the crank and lever.

An important inversion of quadratic cycle chain is the coupled wheel of a locomotive (double crank) meant for transmitting rotary motion of one wheel to another.

The system has two cranks AB and DC which are of equal radius and connect with their centre of wheels. The link AD is fixed and it maintains the centre distance between the wheel centres. The couples link BC transmits the motion from one wheel to the other wheel.

The mechanism couples the two driving wheels of a locomotive and servers to transmit rotary motion of one wheel to the other wheel.

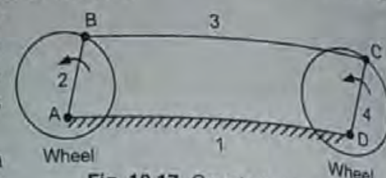


Fig. 10.17. Coupled wheels of a locomotive

The *slider crank chain* is another four bar linkage consisting of one sliding pair and three turning pairs, and it is meant for converting reciprocating motion of the piston of an IC engine into rotary motion of the crank.

With reference to Fig. 10.18, the four links are:

Link 1 (frame of the engine including cylinder and pivot) which is fixed; Link 2 (crank); Link 3 (connecting rod) and Link 4 (piston).

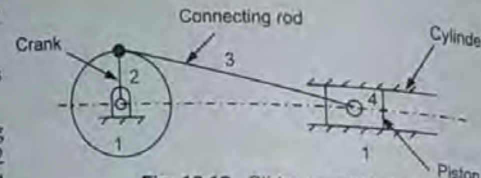


Fig. 10.18. Slider crank chain

The sliding pair is formed by cylinder and piston (links 1 and 4), and the three turning pairs are formed by frame and crank (links 1 and 2), crank and connecting rod (links 2 and 3) and connecting rod with piston (links 3 and 4).

When the piston reciprocates inside the cylinder, the connecting rod oscillates and the crank rotates.

The two common inversions of slider crank change are :

(i) *Oscillating cylinder engine* that converts reciprocating motion to rotary one.

The connecting rod (link 3) is fixed and is connected to piston rod (link 1) at point A. When the piston attached to the piston rod link reciprocates, the cylinder oscillates about a pin pivoted to the fixed linked at A and the crank rotates.

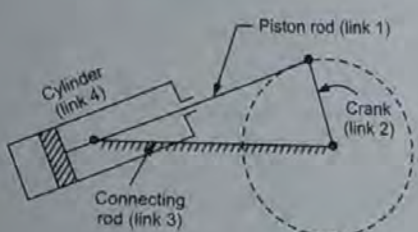


Fig. 10.19. Oscillating cylinder engine

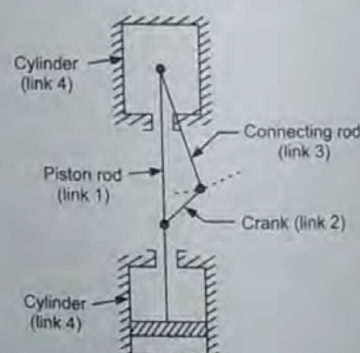


Fig. 10.20. Pendulum pump

(ii) *Pendulum pump* used for supplying feed water to boilers. The mechanism is obtained by fixing the cylinder of the sliding pair of the basic slider crank chain. When the crank (link 2) is made to rotate, the connecting rod (link 3) oscillates about a pin fitted to the stationary cylinder (link 4). The piston attached to the piston rod then reciprocates inside the cylinder.

10.5. BELT AND CHAIN DRIVE

The transmission of power in factories from one rotating shaft to another that lies at a considerable distance is achieved through belts and ropes. The shafts are fitted with pulleys, the belt is wrapped round the pulleys and its ends are connected to form an endless connector. The belts and the pulley remain in contact by frictional grip.

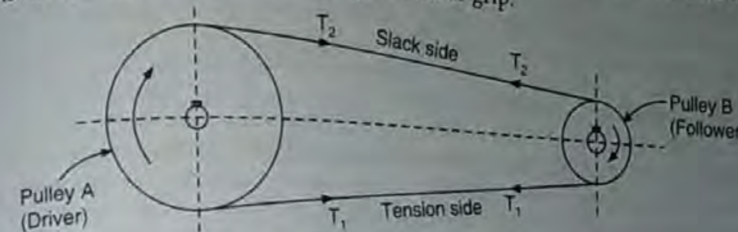


Fig. 10.21. Belt drive-open system

With reference to Fig. 10.21, the pulley A which is connected to the rotating shaft is called the *driver*. The pulley B that needs to be driven is termed as *follower*. When the driver rotates, it carries the belt because of friction that exists between the pulley and the belt. The frictional resistance develops all along the contact surfaces that makes the belt carry the follower which too starts rotating. The driving pulley pulls the belt from one side (called tension side) and delivers it to the other side (called slack side). The tension T_1 in the belt on the tension side is more than tension T_2 on the slack side.

Two parallel shafts may be connected by *open belt* (Fig. 10.21) or by *cross belt* (Fig. 10.22). In the open belt system, the rotation of both the pulleys is in the same direction. If a crossed belt system is used, the rotation of pulleys will be in the opposite direction.

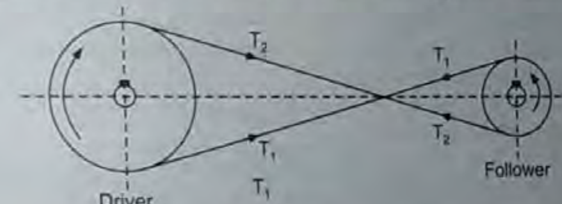


Fig. 10.22. Belt drive-cross system

The angle of contact in this system of drive is more and accordingly it can transmit more power than open belt drive system. However, the belt wears out fast at the places where crossing takes place in the crossed belt system. Further, for small centre distance, the belt is not fully utilized because of its larger slanted run off.

When a number of pulleys are used to transmit power from one shaft to another, a *compound* drive is used.

Fig. 10.23. Belt drive-compound system

With reference to Fig. 10.23, pulley 1 drives pulley 2. Since pulleys 2 and 3 are keyed to the same shaft, pulley 3 is also driven by pulley 1. Subsequently, pulley 3 drives pulley 4. The arrangement conforms to extended open system and all the pulleys rotate in the same direction.

10.5.1. Belt material and sections

A belt is a continuous band of flexible material having a rectangular, trapezoidal and round cross-section. Large variety of belts are available including those made of leather, fabric, rubber impregnated fabric and synthetics. Belts and ropes obtain their flexibility from the distortion of the material of which they are made.

Flat Belt

A flat belt is a belt with a narrow rectangular cross-section. The flat belts are easier to use and are subjected to minimum bending stress. The load carrying capacity of a flat belt depends on its width.

Material used for belt is generally leather of various types having ultimate tensile strength between 4.5 and 7 N per cm width. For heavy duty, two or three plies of leather are cemented and pressed one above the other. Such belts are called double or triple ply belts. Leather belts have the best pulling capacity, can be used both in dry and wet places at ordinary temperature, but are costly.

Belting is also made from plies of stitched canvas impregnated with rubber or balata gum and is correspondingly known as rubber belting or balata belting. **Rubber belts** give best results in damp condition as they do not absorb moisture as readily as leather. However, they are quite expensive and get ruined in the presence of oil and grease. **Balata belts** are acid and water proof and are used in heavily saturated steam laden atmosphere of dye house or where chemical fumes are likely to affect.

Steel belts are immune from stretching and slipping, remain unaffected by dampness or heat, and transmit more power per cm width.

The mid section of flat pulleys is provided a slight dwell to prevent the belt from running off the pulley. This is referred to as **crowning** of pulleys and may be rounded or tapered.

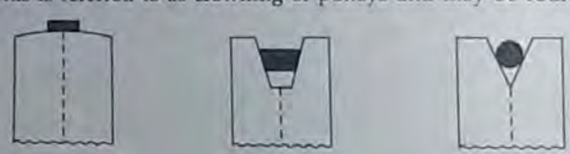


Fig. 10.24. Belt cross-section

V-belt

A V-belt is a belt of trapezoid section running on pulleys with grooves cut to match the belt. V-belts are usually made of cotton fabric, cords and rubber which is vulcanized in moulds of desired cross-section.

The salient features of V-belt are :

(i) The groove angle in the pulley for running the belt is between 40° and 60°. The belt rests on both sides of the pulley but does not touch the bottom of the groove. The grooved pulleys are generally made of cast iron.

(ii) The wedging action between the belt and sides of the groove increases the frictional grip and that reduces the chances of slipping ; there is no possibility of belt coming out of the groove. Due to reduced slipping, V-belts offer a more positive drive.

(iii) V-belts are used when the distance between the shafts is short, transmission members are large and large power is to be transmitted.
The ideal centre distance for V-belt drive is 1.25 to 1.5 times the diameter of the larger pulley.

(iv) Multiple V-belts can be employed for greater power outputs. Further in the multiple drive system, the machine does not come to stop even if one belt fails.

(v) V-belt drives run quietly at high speeds and are capable of absorbing high shock.

(vi) Larger reductions in speed are possible in a single drive by using V-belts over small pulleys.

(vii) There is better initial installation and replacement due to standardization of V-belts.

V-belts are available in five sections designated as *A,B,C,D* and *E*, and these are used in order of increasing loads. That is, section *A* is used for light loads only, and section *E* is used for heavy duty machines. The angle of V-belt for all sections is about 40-degree.

Round Belt

The round cross-section belts are employed when low power is to be transmitted such as in instruments, house hold appliances, table top machine tools and machinery of the clothing industry.

These belts are made of leather, canvas and rubber. Their diameter is usually within the range 4 mm to 8 mm, and the allowable ratio of the diameter of smaller pulley to the belt diameter is about 20.

10.5.2. Velocity ratio, slip and creep

The velocity ratio of belt drives is the ratio of the speed of driven pulley to that of driving pulley.

Let d_1, d_2 = diameters of driver and driven pulleys

ω_1, ω_2 = angular velocities of driver and driven pulleys

N_1, N_2 = rotational speeds of driver and driven pulleys; expressed in revolutions per second (rps)

Linear speed of driving pulley $= \omega_1 \times \dfrac{d_1}{2}$

Linear speed of driven pulley $= \omega_2 \times \dfrac{d_2}{2}$

Presuming that the belt is inelastic and there is sufficient friction to prevent any slip between the belt and pulleys, both the pulleys will have the same linear speed. That is

$$\omega_1 \times \frac{d_1}{2} = \omega_2 \times \frac{d_2}{2}$$

$$\therefore \quad \frac{\omega_2}{\omega_1} = \frac{d_1}{d_2} \quad \text{or} \quad \frac{2\pi N_2}{2\pi N_1} = \frac{d_1}{d_2}$$

That gives : $$\frac{N_2}{N_1} = \frac{d_1}{d_2} \qquad ...(10.3)$$

The ratio N_2/N_1 is measure of the velocity ratio of the rotating pulleys. Further, it is apparent from equation 10.3 that *speed of a pulley is inversely proportional to its diameter.*

If thickness t of the belt is taken into account, then

$$v = \omega_1 \times \frac{d_1 + t}{2} = \omega_2 \times \frac{d_2 + t}{2}$$

$$\frac{\omega_2}{\omega_1} = \frac{d_1 + t}{d_2 + t} \quad \text{or} \quad \frac{2\pi N_2}{2\pi N_1} = \frac{d_1 + t}{d_2 + t}$$

$\therefore$ Velocity ratio $$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t} \qquad ...(10.4)$$

Slip and its effect on velocity ratio : When the frictional grip between the belt and pulley becomes insufficient, there occurs some forward motion of the driver without carrying the belt with it. The relative motion between the pulley and belt is called slip. The difference between the linear speeds of the pulley rim and belt is the measure of slip.

Let S_1 = percentage slip between driver and the belt

S_2 = percentage slip between belt and the follower (driven pulley)

Linear velocity of driving pulley

$$v_1 = \omega_1 \times \frac{d_1}{2}$$

Due to slip between the driving pulley and the belt, the velocity of belt will decrease

$$\text{Velocity of belt} = v_1 - v_1 \times \frac{S_1}{100} = v_1\left(1 - \frac{S_1}{100}\right)$$

This will also be the velocity of belt as it passes over the driven pulley. As there is slip at the driven pulley also, the velocity of the follower pulley will become less.

Linear speed of the driven pulley

$$= v_1\left(1 - \frac{S_1}{100}\right) - v_1\left(1 - \frac{S_1}{100}\right) \times \frac{S_2}{100} = v_1\left(1 - \frac{S_1}{100}\right)\left(1 - \frac{S_2}{100}\right)$$

$$= v_1\left(1 - \frac{S_1 + S_2 + 0.01 S_1 S_2}{100}\right) = v_1\left(1 - \frac{S}{100}\right) = \omega_1 \times \frac{d_1}{2}\left(1 - \frac{S}{100}\right)$$

where $S = S_1 + S_2 + 0.01\, S_1 S_2$ is the percentage of total effective slip.

The linear speed of the driven pulley is also given by :

$$v_2 = \omega_2 \times \frac{d_2}{2}$$

$$\therefore \quad \omega_2 \times \frac{d_2}{2} = \omega_1 \times \frac{d_1}{2} \times \left(1 - \frac{S}{100}\right)$$

$$\frac{\omega_2}{\omega_1} = \frac{d_1}{d_2}\left(1 - \frac{S}{100}\right)$$

or

$$\frac{2\pi N_2}{2\pi N_1} = \frac{d_1}{d_2}\left(1 - \frac{S}{100}\right)$$

or

$\therefore$ Velocity ratio $$\frac{N_2}{N_1} = \frac{d_1}{d_2}\left(1 - \frac{S}{100}\right) \qquad ...(10.5)$$

It is apparent from equation 10.5 that the velocity ratio decreases due to slipping of belt. If thickness t of the belt is also taken into account, then

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t} \times \left(1 - \frac{S}{100}\right) \qquad ...(10.6)$$

Creep : When the belt passes from the slack side to the tight side a certain portion of the belt extends, and when the belt passes from the tight to slack side the belt contracts. Due to these changes in length, there is relative motion between the belt and pulley surfaces.

The relative motion is termed as **creep** of the belt. Like slip, creep also reduces the velocity of the belt drive system.

EXAMPLE 10.2.

An engine shaft running at 240 rpm is required to drive a machine shaft by means of a belt. The pulley on the engine shaft is 600 mm in diameter. Determine diameter of the pulley on the machine shaft if it is to run at 360 rpm under the following conditions :

(a) the belt thickness is negligible and there is no slip,

(b) the belt thickness is 5 mm and slip is neglected,

(c) the belt is 5 mm tick and a slip of 2 per cent is allowed between the belt and each pulley.

Solution : When the belt thickness is neglected and these is no slip, we have

(a) $$\frac{N_2}{N_1} = \frac{d_1}{d_2} \qquad \therefore \quad d_2 = d_1 \times \frac{N_1}{N_2} = 600 \times \frac{240}{360} = \mathbf{400\ mm}$$

(b) When belt thickness t = 5 mm is considered and slip is neglected, we have

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}$$

$$d_2 + t = (d_1 + t) \times \frac{N_1}{N_2} = (600 + 5) \times \frac{240}{360} = 403.3 \text{ mm}$$

$$\therefore \quad d_2 = 403.3 - 5 = \mathbf{398.3\ mm}$$

(c) When both belt thickness and slip are considered

$$\frac{N_2}{N_1} = \frac{d_1 + t}{d_2 + t}\left(1 - \frac{S}{100}\right)$$

Effective slip, $S = S_1 + S_2 - 0.01\ S_1\ S_2 = 2 + 2 - 0.01 \times 2 \times 2 = 3.96\%$

$$d_2 + t = (d_1 + t) \times \frac{N_1}{N_2} \times \left(1 - \frac{S}{100}\right) = (600 + 5) \times \frac{240}{360} \times \left(1 - \frac{3.96}{100}\right) = 387.23 \text{ mm}$$

$$\therefore \quad d_2 = 387.23 - 5 = \mathbf{382.2\ mm}$$

Power transmitted by the belt

$$= (T_1 - T_2) \times V = (2.1\ T_2 - T_2) \times 10.46 = 11.506\ T_2 \text{ Nm/s}$$

$$\therefore \quad 7.5 \times 10^3 = 11.506\ T_2$$

$$T_2 = 651.83 \text{ N} \quad \text{and}$$

$$T_1 = 651.83 \times 2.1 = 1368.85 \text{ N}$$

Hence necessary width of belt $= \dfrac{1368.85}{200} = \mathbf{6.84}$ **cm**

10.5.4. Chain drive

The velocity ratio in belt and rope drives may vary due to slip, momentary overloads or because of contact surface becoming slightly greasy. For constant velocity ratio positive drive with short distance between the drive and driven shafts, one would use the chain drive.

The chains are made of rigid links which are hinged together. This hinging provides flexibility needed for wrapping them around the driving and follower (driven) wheels. The wheels (called sprockets) have projecting teeth which fit into the corresponding recesses in the links of the chain. The wheel and the chain move together without slip and perfect velocity ratio is ensured.

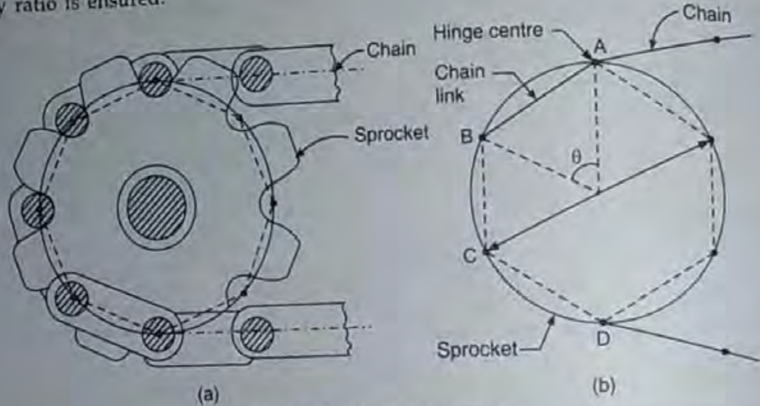


Fig. 10.27. Sprocket and chain

Chain drive is used where the distance between shaft centres is short such as in cycles, motor vehicles, agricultural machinery, road rollers, heavy earth moving machinery, etc.

The advantages of chain drive are :

- chain drive takes less space than a belt or rope drive,
- no slip takes place and that ensures perfect velocity ratio,
- more suitable for transmission of power when the distance between the shafts is less
- less load on the shaft
- high transmission efficiency
- capable of transmitting a good amount of power
- a single chain can transmit motion to several shafts.

However, the chain drive requires accurate mounting and careful maintenance, is relatively high in cost, and is quite prone to velocity fluctuations particularly when they are overstretched. Further, gradual stretching of chains necessitates removal of its links from time to time.

Classification of Chains

The chains are primarily classified into hoisting and hauling chains, conveyor chains, and power transmission chains.

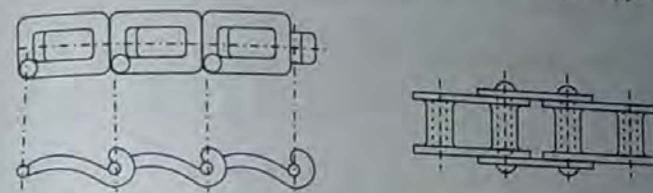
(a) Chain with oval links (b) Chain with square links

Fig. 10.28. Hoisting and hauling chains

- **Hoisting and hauling chains :** The links in such chains are of oval or square shape. The joint of oval links are welded, and the sprockets (wheels) used for such chains have receptacles to receive the links. These chains operate at low speeds such as in chain hoists or in anchors for marine works.

The chains with square links are used for hoists, cranes and dredgers. These chains have low manufacturing cost but are easily prone to kinks on overloading.

- **Conveyor chains :** The links in such chains are either of hook joint type or of closed joint type.



(a) Detachable or hook joint type chain (b) Closed joint type chain

Fig. 10.29. Conveyor chains

The conveyor chains are meant for continuous conveying and elevating material. These chains run at average slow speeds of 1.75 m/s and lack smooth running qualities.

- **Power transmission chains :** These chains are available in the block, roller and silent configuration. The link bearing surface for all these types is machined, hardened and ground. These chains are essentially built for high speed performance and have provision for efficient lubrication.

The ***Block chains*** belong to the earliest stages of development in power transmission and are being put to some use as conveyor chains operating at comparatively low speeds. There occurs rubbing action between the teeth and links when approaching or leaving the teeth of the sprocket. This leads to noisy operation.

The ***roller chain*** assembly essentially consists of :

(i) roller link plate and the pin link plate,

(ii) pins, bushes and rollers.

The roller is free to rotate on the bush which is secured in its hole. The pin passes through the hole and the roller is held by the roller link plates. The central pins are joined and held in position by the pin link plates provided on both sides. The outward lateral sliding of the pin link plates is prevented either by hammering the pin ends to rivet head shape or by using the split pins.

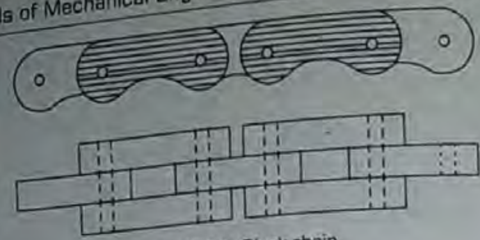


Fig. 10.30. Block chain

The salient features of roller chains are :

- strong and simple construction,
- quieter operation; there is only little noise due to impact of roller on the sprocket wheel teeth.
- provides good service even under severe working conditions,
- requires little lubrication.

However, the wear and stretching of the parts leads to elongation of chain. That results into unequal fitting of rollers into the cavities of the wheel and the load has to be carried by one or only a few teeth.

10.6. GEARS AND GEAR DRIVE

A toothed wheel or gear is essentially a wheel with teeth cut on its periphery.

Power or motion is transmitted from one shaft to another with gear drive when

- centre distances are relatively short
- speed of the shaft is low and the use of belt drive is not recommended
- positive drive is necessary, *i.e*, velocity ratio is fixed and known with certainty
- there is a need to step up or step down the speed
- high torque is to be transmitted
- precise timing is required.

The gear drive has a compact layout and it provides a highly efficient and reliable service. However, the operation tends to become noisy, the whole set is affected when one tooth gets damaged, and the manufacture of gears requires special tools and equipment.

10.6.1. Types of gears

The commonly used forms of toothed gearing are :

• **Spur gear :** It is a cylindrical gear whose tooth traces are straight lines parallel to the gear axis. Further, the tooth profile is identical from one side of the face to the other.

The spur gears are used for transmitting motion between two shafts whose axes are parallel and coplanar.

These gears have a high (96 - 98%) efficiency of power transmission and are free from any axial thrust during tooth engagement. Accordingly they have been successfully employed as sliding gears for speed change mechanism in gear boxes of lathe. However, compared to gears of other types, spur gears are more noisy in operation, wear out readily and develop backlash.

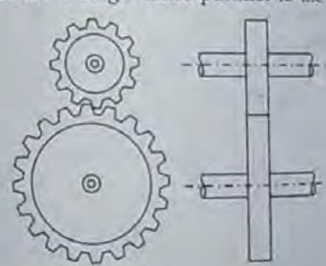


Fig. 10.31. Spur gear

• **Helical gear :** It is a cylindrical gear whose tooth traces are straight helices, teeth are inclined at an angle to the gear axis. The teeth are thus of helical or screw form. This ensures smooth action and accurate maintenance of velocity ratio. However, lateral thrust is set up due to teeth being inclined. This lateral thrust gets neutralized by using double helical gears called **Herringbone gears**. The double helical gears are the equivalent of two helical gears secured together; but they are manufactured as one piece. The teeth may be continuous or separated by a small gap. Further, the two helics are of opposite hand and meet at a common axis.

The helical gears are used in automobile gear boxes, and in steam and gas turbines for speed reduction. The herringbone gears are used in machinery where large power is transmitted at low speeds.

Fig. 10.32. Helical gear

• **Bevel gear :** The bevel gear wheels conform to the frusta of cones having a common vertex ; tooth traces are straight line generators of the cone.

The bevel gears are used to connect two shafts whose axes are coplanar but intersecting. When the shafts are at right angles and the wheels equal in size, the bevel gears are called *mitre gears*. When the bevel gears have their teeth inclined to the face of the bevel, they are known as *helical bevel gears*.

• **Spiral gear :** These are identical to helical gears with the difference that these gears have a point contact rather than a line contact. These gears are used when connection is to be made between intersecting and co-coplanar shafts.

Fig. 10.33. Bevel gear

• **Worm gear :** The system consists of a worm which is basically part of a screw. The warm meshes with the teeth on a gear wheel called worm wheel.

The worm gear is used for connecting two non-parallel, non-intersecting shafts which are usually at right angles. This gearing system is smooth and quiet in operation and provides a high gear ratio; rotational speed of the worm is quite high compared to that of wheel. Their use is recommended when high speed reduction (more than 10 : 1) is required.

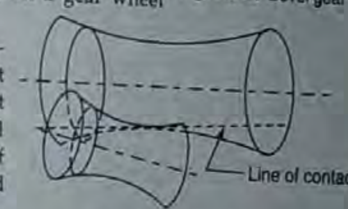


Fig. 10.34. Spiral gear

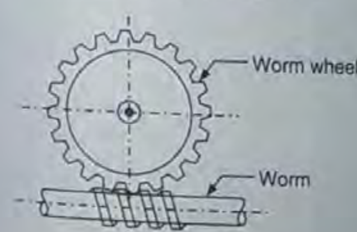


Fig. 10.35. Worm gear

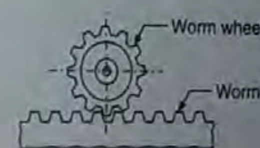


Fig. 10.36. Rack and pinion

• **Rack and pinion :** Rack is a straight line spur gear of infinite diameter. It meshes, both internally and externally, with a circular wheel called pinion. The arrangement finds application where linear motion is to be converted into rotary motion and vice-versa.

• **Internal and external gearing :** Two toothed wheels on parallel shafts may gear either externally or internally.

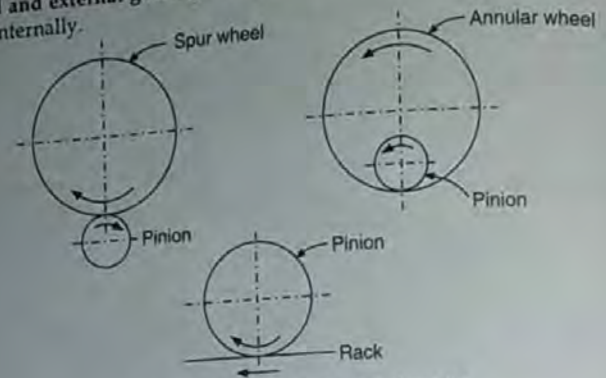


Fig. 10.37. Internal and external gearing

In external gearing, the motion of the two wheels is unlike, *i.e.*, in the opposite direction. The larger wheel of the gear system is known as **spur** and the smaller wheel as **pinion**.

In internal gearing, the motion of the two wheels is alike ; both rotate in the same direction. The external large wheel is called **annular wheel** and the small internal wheel as **pinion**.

The gears of shaft in the rack and pinion arrangement mesh externally and internally with the gears in a straight line. With the help of rack and pinion, it is possible to convert linear motion into rotary motion and vice-versa.

10.6.2. Gear terminology

Fig. 10.38 shows two gears 1 and 2 meshed against one another.

A tooth of the driving gear (centre O_1) meshes against a tooth of the driven gear (centre O_2) at point P. The point P is called the *pitch point*. The circles whose centres are O_1 and O_2, and radii O_1P and O_2P are called *pitch circles* of wheels 1 and 2 respectively. A pitch circle is essentially an imaginary circle which by pure rolling action gives the same motion as the actual gear.

Further, on the pitch circle of wheel 1, let A be a point on one tooth and B be the corresponding point on the adjacent tooth. The distance AB (measured along the circular arc) is called *circular pitch* of wheel

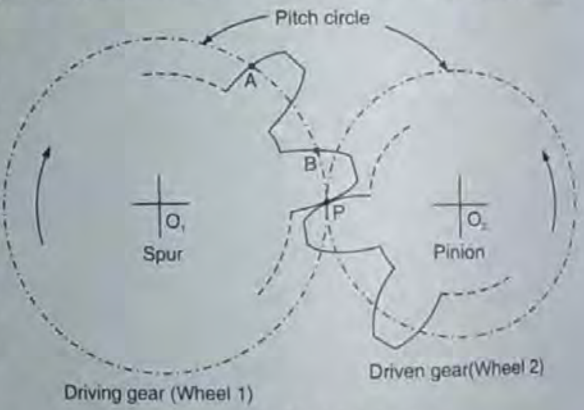


Fig. 10.38. External gearing

1. Circular pitch may then be defined as *"distance measured on the circumference of the pitch circle from a point of one tooth to the corresponding point on the next tooth"*

$$\text{Circular pitch } p_c = \frac{\text{circumference of pitch circle}}{\text{number of teeth}} = \frac{\pi D}{T}$$

where D is the pitch diameter and T is the number of teeth. ...(10.7)

Two gears will mesh correctly if they have the same circular pitch. Therefore

$$p_c = \frac{\pi D_1}{T_1} = \frac{\pi D_2}{T_1} \quad \text{or} \quad \frac{D_1}{T_1} = \frac{D_2}{T_2}$$

Thus the diameter of a wheel is proportional to the number of teeth on it.

Further, when the teeth of two or more gears mesh with one another, the linear speeds of the two gears will remain same.

$$\text{Linear speed of gear 1} = \pi\, D_1\, N_1$$

$$\text{Linear speed of gear 2} = \pi\, D_2\, N_2$$

where D_1 and D_2 are the diameters, and N_1 and N_2 are the revolutions made by gears 1 and 2 per unit time.

Equating the linear speeds of the two gears,

$$\pi\, D_1\, N_1 = \pi\, D_2\, N_2 \;, \quad \frac{N_2}{N_1} = \frac{D_1}{D_2}$$

Since the diameter of a wheel is proportional to the number of teeth on it, we get

$$\text{velocity ratio} = \frac{N_2}{N_1} = \frac{T_1}{T_2}$$

Diametral pitch : It represents the number of teeth on a wheel per unit of its diameter.

$$\text{Diametral pitch } p_d = \frac{\text{number of teeth}}{\text{pitch circle diameter}} = \frac{T}{D} \qquad ...(10.8)$$

From equations 10.7 and 10.8, we have

$$p_c \times p_d = \pi$$

Evidently if the circular pitches of two wheels are equal, their diametral pitches are also equal.

Module : Module represents the ratio of pitch circle diameter (in mm) to the number of teeth. Obviously, the module is the reciprocal of diametral pitch.

$$\text{Module} \quad m = \frac{D}{T} \qquad ...(10.9)$$

Refer Fig. 10.39 for some of the terms associated with profile of a gear tooth.

Addendum circle is the circle bounding the outer ends of the teeth and concentric with the pitch circle.

The radial distance between the pitch circle and addendum circle is called *addendum*.

Dedendum circle is the circle bounding the bottom of the tooth and concentric with the pitch circle.

The radial distance between the pitch circle and the dedendum circle is called *dedendum*.

The surface of tooth above the pitch surface is called the *face* of the tooth. *Flank* is the surface of tooth below the pitch surface. *Tooth space* is the width of tooth measured along the pitch circle. The length of arc between the sides of a gear tooth, measured on the pitch circle is called *thickness of tooth* or *circular thickness*.

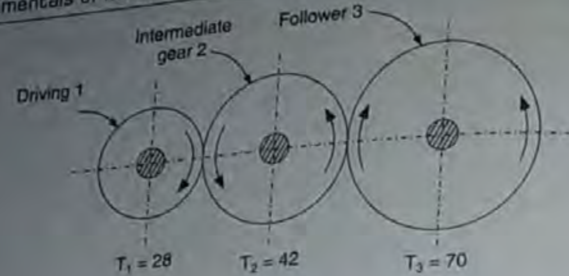


Fig. 10.45

Solution : Speed ratio $= \dfrac{\text{speed of driver}}{\text{speed of follower}} = \dfrac{\text{number of teeth on follower}}{\text{number of teeth on driver}}$

$= \dfrac{70}{28} = 2.5$

(ii) Speed ratio $= \dfrac{N_1}{N_3} = 2.5$ as calculated above

(ii) $\therefore$ Speed of follower $N_3 = \dfrac{N_1}{2.5} = \dfrac{1000}{2.5} = 400$ rpm

With odd number of idle (intermediate) gears, the direction of rotation of the follower is the same as that of driving wheel. Here the driving gear is rotating clockwise and as such the direction of rotation of the follower would also be clockwise.

EXAMPLE 10.16

A compound gear train consists of six gears and the number of teeth on the gears are as follows :

Gear	1	2	3	4	5	6
No. of teeth	35	80	40	125	45	115

The gears 2 and 3 are on the same shaft, and so are the gears 4 and 5 on another shaft. The gear 1 drives the gear 2, the gear 3 meshes with gear 4, and the gear 5 is in engagement with gear 6. Sketch the arrangement.

If the input shaft mounted on gear 1 turns 800 rev/min, determine the rotational speed of the output shaft mounted on gear wheel 6.

Solution : Refer Fig. 10.46 for the arrangement of the given compound gear train

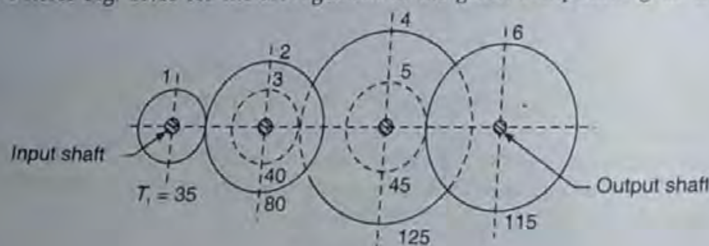


Fig. 10.46.

Driving gears are : 1, 3 and 5

Driven gears are : 2, 4 and 6.

For compound gear train,

speed or velocity ratio $= \dfrac{\text{speed of the first driver}}{\text{speed of the last driven}}$

$= \dfrac{\text{product of the number of teeth on drivens}}{\text{product of the number of teeth on drivers}}$

Thus, $\dfrac{N_1}{N_6} = \dfrac{T_2 \times T_4 \times T_6}{T_1 \times T_3 \times T_5} = \dfrac{80 \times 125 \times 115}{35 \times 40 \times 45} = 18.25$

$\therefore$ speed of the last gear (gear wheel 6) $N_6 = \dfrac{800}{18.25} = 43.83$ **rev/min**

EXAMPLE 10.17

A fixed gear having 200 teeth is in mesh with another gear having 50 teeth. The two gears are connected by an arm which makes one revolution about the centre of the bigger gear. Determine the number of turns made by the smaller gear.

Solution : Refer Fig. 7.43. The arrangement corresponds to epicyclic gear train.

The ratio of the speed of wheel B to that of arm C is given by

$$\frac{N_B}{N_C} = 1 + \frac{T_A}{T_B} = 1 + \frac{200}{50} = 4$$

$$N_B = 4\,N_C = 4 \times 1 = 4$$

10.7. CAMS AND FOLLOWERS

A cam-follower is a higher pair mechanism; its reciprocating or rotating element imparts a desired motion to another element. The desired motion may be reciprocating, rotating or oscillatory in nature. The driving element is called *cam* and the driven member is referred to as *follower*. The direct point contact between the cam and the follower is ensured by a spring.

Generally the cam is connected to a frame forming a turning pair, and the connection between the follower and the frame constitutes a sliding pair. As such the cam-follower mechanism is a three-link mechanism of the higher type. The three links are:

(i) cam which is the driving link and has a straight or curved surface.

(ii) follower which is the driven link; it gets its motion due to its contact with the surface of cam.

(iii) frame which supports the cam and guides the follower.

Cam-follower mechanisms are quite impact, easy to design and are used for generating complex coordinated movements. The common applications of cam-follower mechanism are found in

- clocks and watches
- IC engines for operating the valves
- automatic screw cutting machines
- printing machines and shoe making machines

Attention has been directed in this chapter to the study of various types of cams and followers, motion of follower with a given profile of cam and to draw the profile of cam with the given motion of the follower.

10.7.1. Classification of Cams and Followers

Cams are classified according to shape, follower movement and the manner of the constraint of the follower. However, the following two types are considered important:

(*a*) Radial cams in which the follower reciprocates or oscillates in a plane that is perpendicular to the axis of cam.

(*b*) Cylindrical cams in which the reciprocating or oscillatory movement of the cam is in a plane that lies parallel to the axis of cam.

The important shapes of the radial cams are:

- *Wedge cam* that has a specified contour. The reciprocating motion of the cam may impart reciprocating or an oscillatory motion to a knife edged follower.

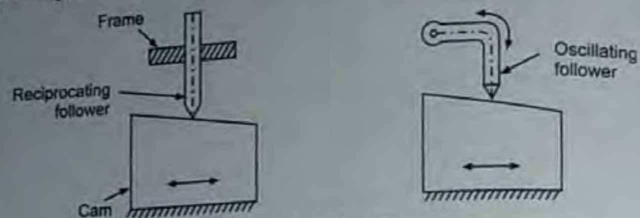


Fig. 10.47. Wedge cams

- *Tangent cam* that has straight flanks and circular nose. The roller follower used in conjunction with such a cam may get reciprocating or oscillating motion.

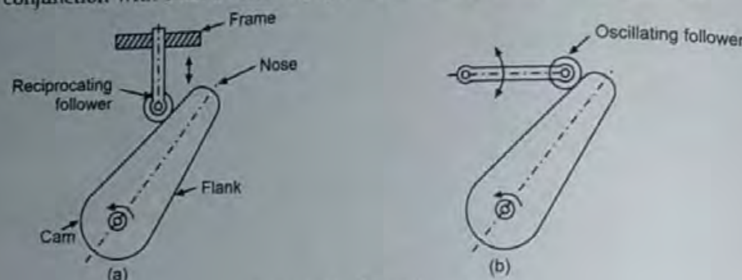


Fig. 10.48. Tangent cams

- *Circular cam* that has a circular flank and circular nose. Figure 10.49 shows a circular cam operating with a flat faced follower and an offset flat faced follower. In the offset arrangement, the vertical centre lines of the cam and the follower do not coincide.

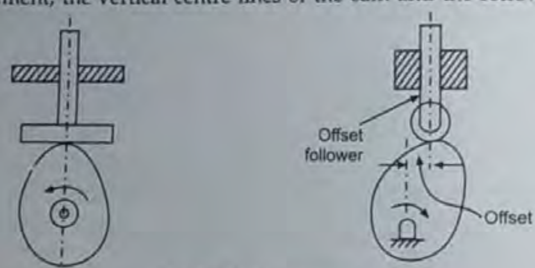


Fig. 10.49. Circular cams

The cyl... as drum or barrel cam has a circumferential contour cut on the surface of a cylinder which rotates about its axis. The reciprocating or oscillating motion of the follower is in a plane parallel to that of the axis of the cam.

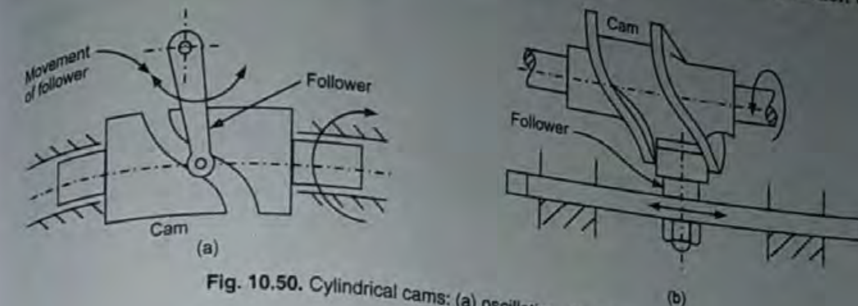


Fig. 10.50. Cylindrical cams: (a) oscillating, (b) reciprocating

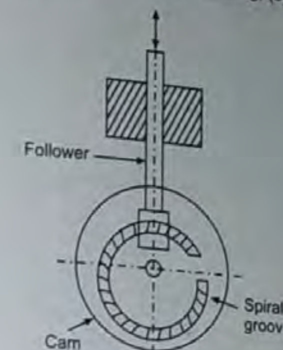


Fig. 10.51. Spiral cam

The *spiral* or *face* cam consists of a circular plate having a spiral groove cut into it. There is engagement between the teeth cut on the spiral groove and the pin gear follower.

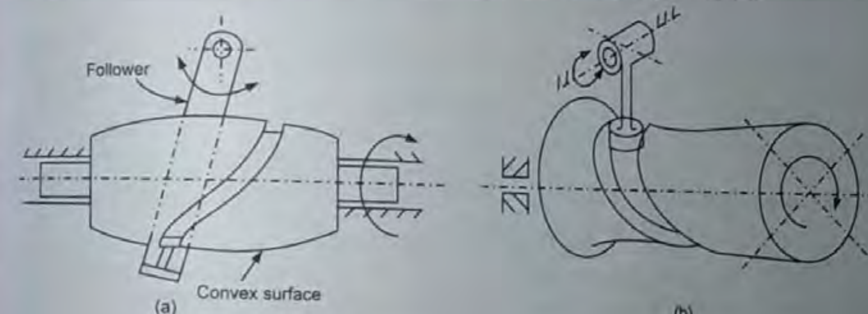


Fig. 10.52. Convex and concave globoidal cam

A globoidal cam is consistuted by a circumferential contour which is cut on a concave or convex surface. The motion of the follower in the globoidal cam is oscillatory in nature.

The follower in a cam-follower mechanism is classified either according to the type of motion of the follower or the nature of the surface in contact:

Based on the nature of surface in contact, the followers belong to the following four categories:

(*a*) **Knife-edge follower:** a follower that has a sharp pointed edge. There is a sliding motion between the contacting surfaces, i.e., the knife edge and the contacting surface. Though simple in construction, its utility is restricted due to considerable side thrust between the follower and guide and the high rate of wear.

(*b*) **Roller follower** a follower that consists of a cylindrical roller which rolls over the cam surface. Due to rolling motion between the contacting surfaces, there is much less wear as compared to that in a knife edge follower. A side thrust, however does exist between the follower and guide. Roller followers find favour for use in situations where more space is available such as in statioinary gas or oil engine.

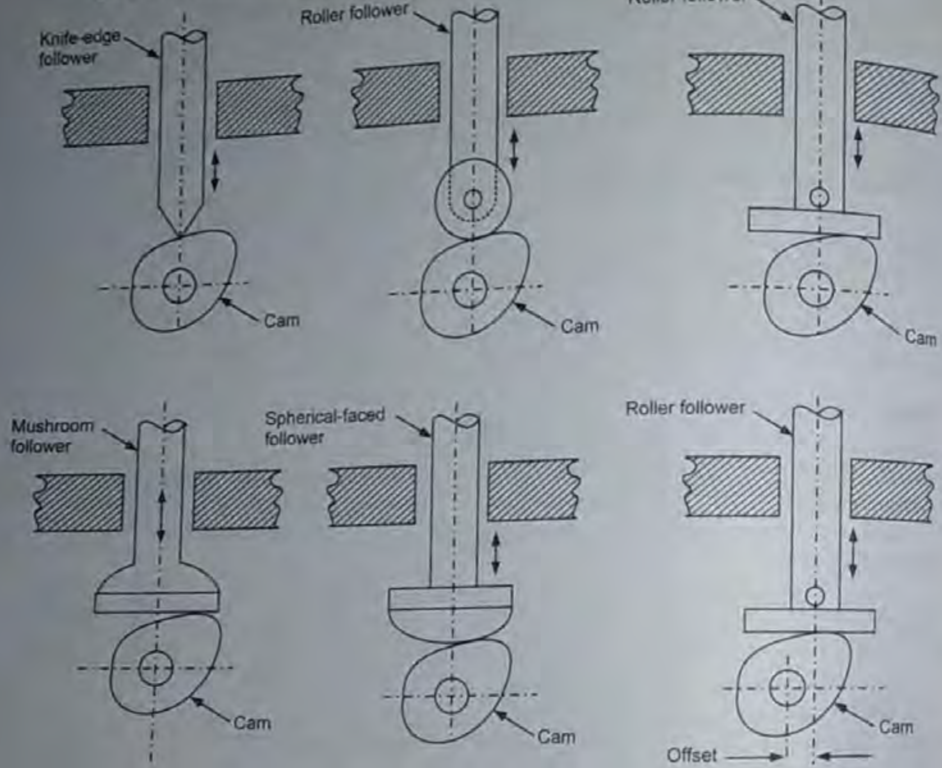


Fig. 10.53. Types of cam followers

(*c*) **Flat faced or mushroom follower:** a follower that has a flat contacting surface. Quite often, the flat end of the follower is machined to a spherical shaft and the resulting follower is called a spherical-faced follower.

With these ... less side thrust at the bearings and that implies reduced friction force and less changes of jamming in the bearings. These followers are generally used for operating the values of an automobile engine where space is limited.

Based on the path of morion axis of location of motion, followers are categorized as:

(*i*) **Radial followers:** the motion of the followers is along an axis that passes through the centre of the cam. The follower translates along a line passing through the axis of rotation of the cam (Fig. 10.54).

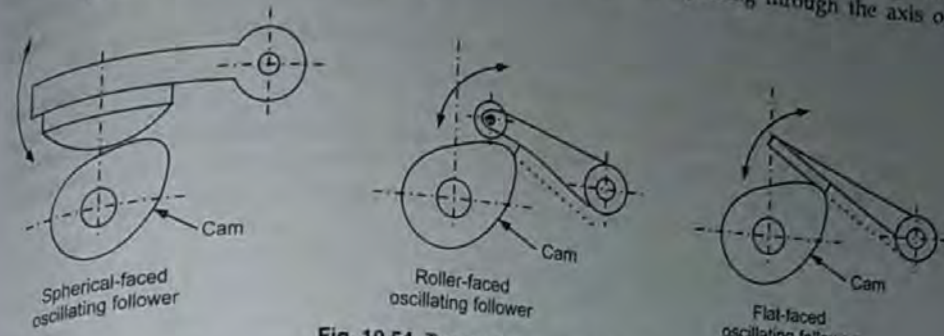


Fig. 10.54. Types of oscillating followers

(*ii*) **Off-set followers:** the motion of the follower is away from the centre of rotation of the cam.

Based on the nature of motion, there are the following two types of followers.

- *Reciprocating or translating followers:* the follower reciprocates in guides as the cam rotates.
- *Oscillating or rotating followers:* the follower oscillates about a hinge point as the cam rotates.

It would be appropriate to mention that the follower is always constrained to follow the cam, and objective is achieved by springs, gravity or hydraulic means.

10.8. BEARINGS

A ***bearing*** is a machine element which supports another moving machine element (*e.g.*, a rotating shaft) called *journal*. While carrying the load, the bearing allows relative motion between the contact surfaces of the members.

In ***radial*** or ***journal*** bearings, the main load is perpendicular to the axis of rotation of the moving element.

The portion of the shaft laying within the bearing is known as *journal*.

The plain journal or sleeve bearings are classified as

(*i*) *Full bearing*: The bearing completely envelops the journal.

(*ii*) *Partial bearing*: The enveloping angle is not 360° but is 120°. The friction in a partial bearing is less than that in full bearing but its applications are limited to only those situations where the load is always in one direction, *e.g.*, rail road car axles. The full bearing and partial bearings are also known as clearance bearings since the journal size is less than the bearing bore.

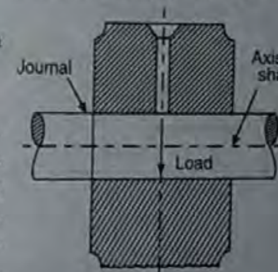


Fig. 10.55. Journal bearing

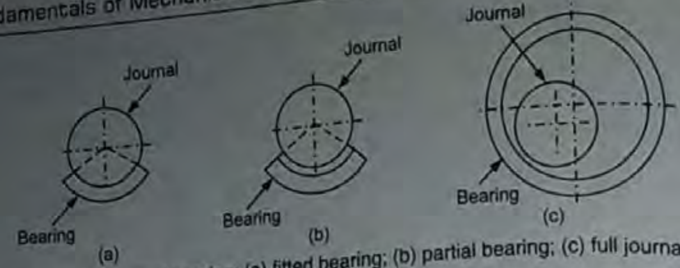


Fig. 10.56. Types of journal bearing: (a) fitted bearing; (b) partial bearing; (c) full journal bearing

(iii) *Fitted bearing*: A special case of partial bearing in which the sizes of the journal and bearing are equal and hence there is no clearance.

In the ***collar thrust*** bearings, the bearing pressure is parallel to the shaft axis and has end thrust.

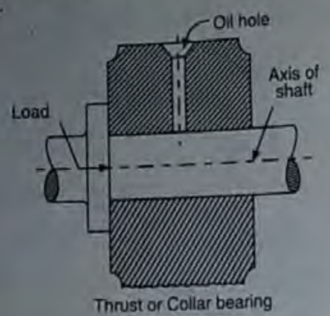


Fig. 10.57.

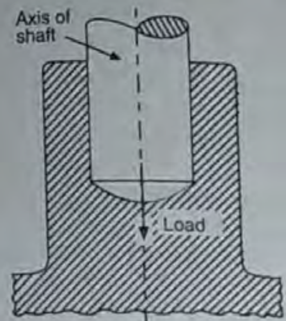


Fig. 10.58. Footstep or pivot bearing

These bearings provide horizontal load reactions in machine tools and marine drive shaft.

In the ***pivot*** or ***foot step*** bearing, the bearing pressure is exerted parallel to the shaft whose axis is vertical. The end of the shaft rests within the bearing body.

Such bearings carry the load in vertical steam turbines, water turbines, motors and pumps.

In ***ball bearings***, the rolling element is a spherical ball. The self-alignment ball bearing permits inclination of the inner race or shaft axis with relation to the axis of the outer race with 2-3 degree.

Refer Fig. 10.59 which shows a typical rolling bearing.

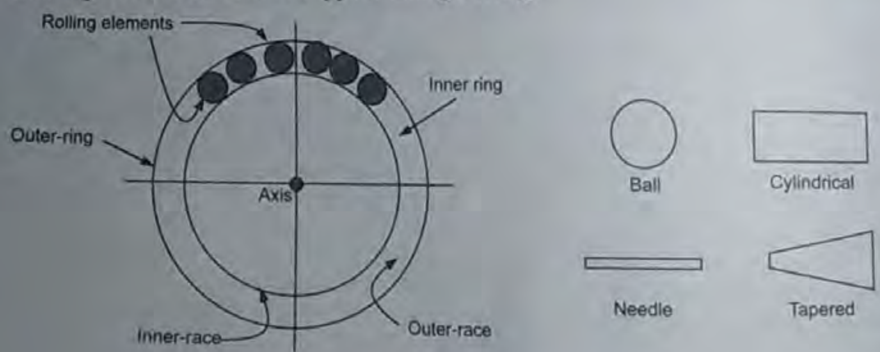


Fig. 10.59. Typical rolling bearing and rolling elements

The unit consists of an inner ring, an outer ring and the rolling elements (balls) placed at equal intervals in the open space between the two rings. The balls are usually made steel or ceramic material. The rings are of uniform stiffness throughout and their surfaces, called the raceways, are quite hard. Depending upon the geometrical configuration of the rolling elements, the rolling bearings are classified as :

(i) ball bearings
(ii) cylindrical roller bearings
(iii) needle-type roller bearings
(iv) tapered roller bearings.

Ball bearings can handle both radial and thrust loads but these loads have to be relatively small or moderate.

The needle and the taper roller bearings belong to the category of roller bearings in which the rolling element is a roller which may be cylindrical, conical, spherical or concave.

The sintered bearings are self lubricated bearings suitable for applications where regular maintenance is difficult.

In the roller contact bearings, the contact between the bearing elements is rolling instead of sliding as in plain bearings. Since the rolling friction is very less as compared to the sliding friction, such bearings are also known as antifriction bearings. The roller contact bearings have the disadvantage of low resistance to shock loading.

The ball bearings have been standardized into four classes and designated by a number consisting of atleast three digits

(i) 100 : extra light (ii) 200 : light
(iii) 300 : medium (iv) 400 : heavy

Additional digits are added to specify special features. The last three digits denote the series and the bore of the bearings. The last two digits from 04 onwards, when multiplied by 5, give the bore diameter in mm. Thus if a bearing is designated by number 308, it is a bearing of medium series with bore equal to 5 × 08, *i.e.*, 40 mm.

10.9. RATCHET AND PAWL

The ratchet and pawl mechanism is a mechanical gearing system used to transmit intermittent linear or rotary motion in one particular direction, and prevents motion in the opposite direction. The unit essentially comprises two components namely a ratchet and a pawl mounted on the base as shown in Fig. 10.60.

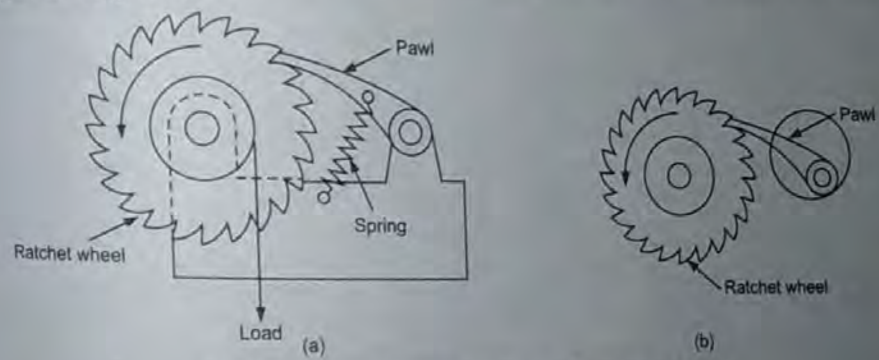


Fig. 10.60. (a) Weighing mechanism; (b) Ratchet and Pawl

The *ratchet* is a gearing component having star-shaped angled tooth around its outer periphery. The teeth are uniform but asymmetrical with each tooth having a moderate slope on one edge and a much steeper slope on the other edge. The ratchet receives an intermittent circular motion from the crank and serves to slow down the motion that happens in a jerky way. The pawl is a type of latch in the form of a teeth shaped solid part that is pivoted at one end and rests against the ratchet at the other end. The pawl is usually spring loaded and that ensures its automatic engagement with the ratchet.

When the ratchet rotates in one particular direction, the pawl rises and moves smoothly between the angled-teeth of the ratchet. Subsequently when the rotation of ratchet stops, the pawl rests and gets jammed against the depression between the gear teeth and that prevents any rotation of socket in the backward direction.

As the ratchet can stop backward motion only at discrete points, i.e., only at tooth boundaries, the ratchet does allow a limited amount of backward motion. This backward motion is limited is a maximum distance equal to the spacing between the teeth and is called backlash. In situations where backlash is required to be minimum, use is sometimes made of a toothless ratchet with a high friction surface. The pawl then bears against the surface at an angle; any backward motion causes the pawl to jam against the surface and that prevents any further backward motion.

The various applications of ratchet and pawl assembly are :

- spanners, wrenches and jacks
- free wheel mechanism of bicycles
- clocks, winches, turnstiles and typewriters.
- hoists and weight lifting machines.

In the weight lifting mechanism shown in Fig. 10.49 (*b*), a ratched wheel is fixed to a shaft and a drum around which a rope is wound. The winding of the rope is done by rotating the ratchet in anti-clockwise direction. When the rotation stops, the load on the rope tries to unwind the rope. This tendency is however prevented by the action of ratchet and pawl.

Ratchet and pawls are usually made of steel, stainless steel, cast iron, brass and other metallic materials. The product specifications for ratchet and pawl include the number of teeth, outside and bore diameter, face width and pitch.

10.10. HYDRAULIC AND PNEUMATIC ACTUATING SYSTEMS

Pascal law states that "*intensity of pressure is transmitted equally in all directions that a mass of fluid in a confined place.*" This characteristic property of the fluid forms the working principle of fluid power systems. With reference to Fig. 10.61, two cylinders of different cross-sectional arcs are interconnected at the bottom through a pipeline and are filled with some liquid (oil or water). The larger cylinder contains a raw of area A and a plunger of area a reciprocates inside the smaller cylinder. A force F_1 applied to plunger produces an intensity of pressure p_1 which is transmitted in all directions through the liquid. If the plunger and ram and at the some level and if their weights are neglected then pressure intensity $p_2 = \frac{F_1}{a}$ and $p_2 = \frac{F_2}{A}$ where F_2 is the upward force acting on the ram must equal p_1.

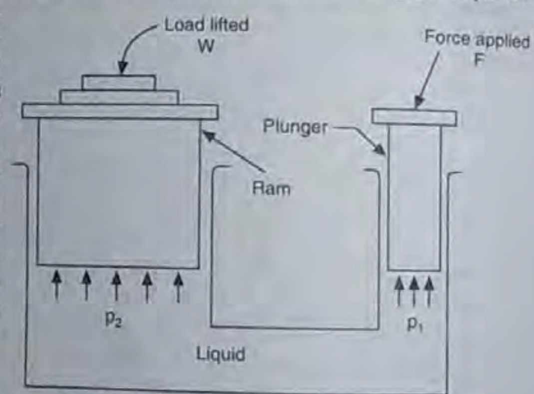


Fig. 10.61. Working principle of hydraulic press

Now $p_1 = \frac{F_1}{a}$ and $p_2 = \frac{F_2}{A}$ where F_2 is the upward force acting on the ram. Since $p_1 = p_2$, we have

$$\frac{F_1}{a} = \frac{F_2}{A} \; ; \quad F_2 = F_1\left(\frac{A}{a}\right)$$

The above expression indicates that by applying a small downward force F_1 on the plunger, a large upward force F_2 acts on the ram by suitably selecting ratio of the diameters of the two cylinders.

The fluid systems find applications in :

- hydraulic jack, hydraulic lift and hydraulic crane etc.
- measurement of process parameters and then using these parameters to act on necessary output.
- carrying out mechanical work and using the resulting motion (linear or rotating in cutting operations (sawing, turning, milling and drilling); feeding, sorting and packaging; damping, shifting and positioning.
- controlling of plant, process and equipment where a hydraulic or pneumatic system. The condition of the process is sensed and the information is fed to the controller for taking the appropriate corrective action.

The performance indices of a fluid system include :

- flow rate of fluid and chamber capacity
- pressure fatigue and bursting pressure
- cleaning, compressibility and viscosity of the fluid.

The process of pneumatic system is similar to that of water or oil, compressed air supplies the inlet power.

10.11. CONTROL VALVES: FUNCTIONS AND TYPE

A valve is a device that regulates, directs or controls the flow of fluid (liquids or gases) by opening, closing or partially obstructing the passage ways. The valves way he operated manually either by handle, lever, pedal or wheel. Modern control valves are automatically driven by changes in pressure, temperature or flow rate. These changes act on a diaphragm or piston which then activates the valve. The automatic operation of the valve is based on an external input (flow regulation to a changing set-point) that requires an actuator. The actuator operates depending on its input and the set-point and that allows the valve to be positioned accurately and allowing control over a variety of requirements.

Valves are found in virtually every industrial process and are used for :

- starting or stopping flow
- preventing back flow
- relieving and regulating pressure in a fluid

Valves are quite diverse and are generally classified by how they are actuated :

hydraulic, pneumatic, manual, solenoid and motor

Further, depending upon how they work/operate, the valves are categorized as :

(*i*) normally open or normally closed valves. These valves are acted upon by some forces that keep them either open or closed.

(*ii*) throttling valves that are intermittently opened or closed.

(*iii*) directional control valves which are used to redirect the flow. These diverter type valves are 2-way 3-way or 4-way valves.

- The 2-way valves are the shut off valves which have only the open and closed positions, and have two ports called the inlet and outlet port.
- The 3-way valves have three ports – an inlet port, the outlet port and the exhaust port. The operation is in two conditions : (*i*) exhaust closed and the inlet open to outlet and (*ii*) inlet blocked and the outlet connected to exhaust.
- The 4-way valve arrangement consists of two 3-way valves which are operated by an actuator. The inlet port and the two exhaust ports are joined internally and that provides four ports comprising the inlet port, two outlet ports and an exhaust port.

Further based upon construction, the control valves are categorized as : poppet valves (ball seat and disk seat type), spool valves (longitudinal slide and plate slide type) and rotary valves.

10.11.1 Pressure control, direction control and sequence valves

Pressure control valves are used virtually in every hydraulic system and they function to keep system pressure safely below a desired upper limit to maintain a set pressure in any part of the circuit. These valves are normally closed and have a restriction to produce the desired pressure.

The different types of pressure control valves are the relief, reducing, sequence, counter-balance and unloading type. The *relief valve* shown in Fig. 10.62 consists of a ball or poppet exposed to the system pressure on one side and opposed by a spring of pre-set force on the other side. The spring holds the ball tightly seated and that blocks the flow through the valve. When the force of system pressure rises and exceeds the spring force, the valve gets lifted from its seat and that lets the fluid out through the vent.

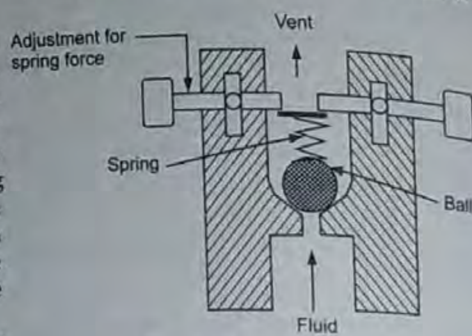


Fig. 10.62. Relief valve

A *sequence valve* functions to provide a path of flow alternate and sequential to the primary circuit. The operation of these valves is controlled mechanically or by pressure. The pressure operated sequence valve is normally closed poppet or spool valve that opens at an adjustable set pressure.

Fluid at inlet port of the valve will not pass to the secondary circuit or outlet port until the fluid pressure reaches the set pressure. When the set pressure is reached, the sequence valve directs the fluid to a second actuator or motor to do work in another part of the circuit.

Typically a sequential valve serves to operate multiple actuators and their sequence of operation.

The *direction control valves* (DCV's) are one of the most fundamental parts that are employed to start, stop and change the direction of fluid flow in a fluid control system. They allow the flow of working fluid (oil, gas) into different paths from one or more source.

The direction control valves are classified according to certain factors such as :

- number of ports or ways
- method of actuation : manual, mechanical, solenoid operated and pilot operated.
- shape of valving element which is mostly a ball, a sliding spool or a rotary spool.

The controlling of the passage of a fluid signal is done by generating, cancelling or redirecting signals and categorization of direction control valves is then done is signalling elements (input), processing elements, and power elements or final control elements.

The simplest direction control valve is a 2-way valve which either stops flow or allows flow.

The 3-way valve allows fluid flow to an actuator in one position and exhaust the fluid flow from it in other position.

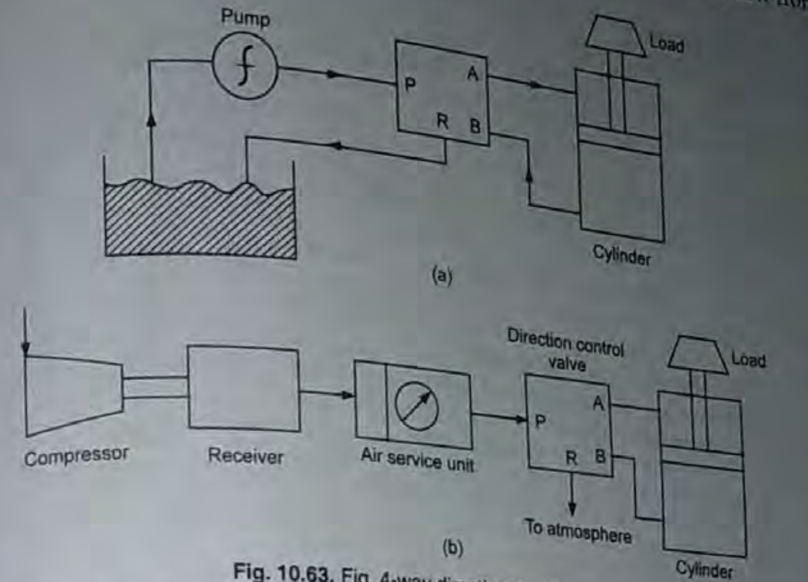


Fig. 10.63. Fig. 4-way directional valve

Refer to the 4-way directional valve which has four ports :

- Pressure port *P* in communication with a pump or compressor.
- Return or exhaust port *R* connected to the fluid reservoir or vented to atmosphere.
- Two output ports *A* and *B* connected to the actuating system (load).

The internal operation of the valve has been shown in Fig. 10.64 both for the extend (forward) and retract (backward) movement of the piston.

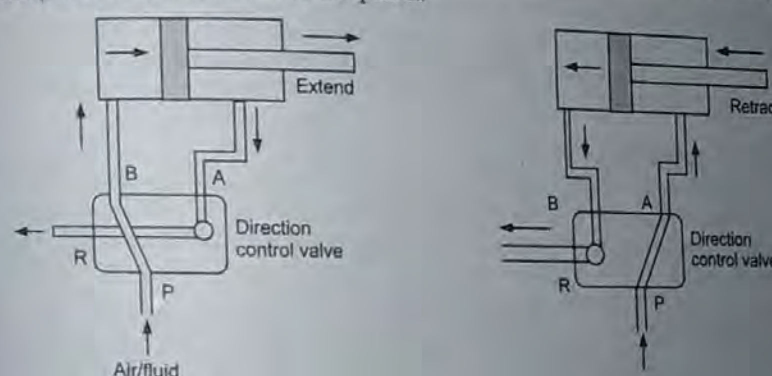


Fig. 10.64. Internal operation of the valve.

During extend movement, the pressure port *P* communicates with the outlet port *B* and that allows the liquid/air pressure to be directed to the cylinder from the left side. The exhaust of liquid/air from the right side of the cylinder then occurs to the liquid tank/atmosphere via exhaust port R through port A. During retract movement of piston, ports P and A deliver the fluid to right of piston and the ports B and R provide a path for the return to liquid sump/atmosphere.

Since the valve has four ports and two control positions, it is known as 4/2 valve.

10.12. ACCUMULATOR

An accumulator is a unit installed in a hydraulic system to store high pressure fluid during idle periods and later make it available to supplements pump flow and serve as a backup during power failure.

Three common types of accumulators used in a hydraulic system are of bladder, piston and diaphragm type. The bladder types find favour and Fig. 10.65 shows the constructional features of one such accumulator. There is a rubber member, called bladder which separates gas from the oil. The gas is filled inside the bladder and the oil surrounds the bladder. Initially the gas is pre-charged to around 80 to 90 percent of the system working pressure.

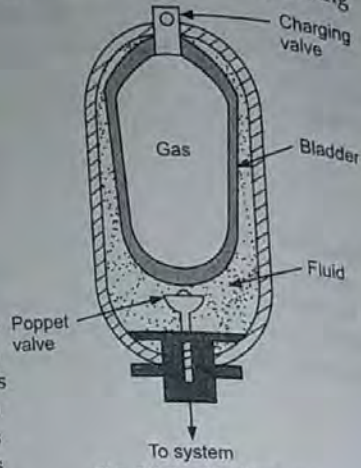


Fig. 10.65. Bladder type accumulator

This expands the bladder to fill most of the accumulator space with only a small amount of oil remaining inside. During operation, the hydraulic pump raises system pressure and forces fluid to enter the accumulator. The bladder moves and compresses the gas valve because the oil pressure exceeds the pre-charge pressure. The bladder movement stops when the system and pressures become equal. When there is creation in the system demand, the hydraulic pressure falls and the stored pressurized oil is released to the circuit.

An accumulator is essentially a energy storage device that enables the hydraulic system to cope with extremes of demand and yet maintain specified pressure in the system. Other advantages are :

- smooth out pulsations and provide shock cushioning
- supplement pump flow and allow the use of less powerful pump
- act as an emergency standby power source
- provide quick response to sudden enhanced demand
- compensate for any oil leakage, thermal expansion/contraction and hold the required pressure in the circuit.

10.13. AMPLIFICATION

The signal from the detector transducer is normally very weak, and it needs amplification to a certain level where it can be detected for display or record. Amplification is also needed to transmit the signal over some distance. The device used to increase or augment the weak signal is referred to as amplifier; it may operate on mechanical (levers, gears etc.), optical, pneumatic and hydraulic, or electrical and electronic principles.

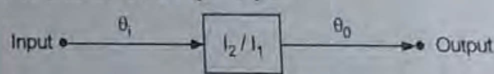


Fig. 10.66. Amplifier unit

The ratio of output signal (θ_0) to input signal (θ_1) for an amplifier is generally referred to as **gain, amplification or magnification**; the input to and output from the amplifier are related by the expression

$$\theta_0 = G\,\theta_1 \; ; \; G = \frac{\theta_0}{\theta_1}$$

where G is the gain or amplification. Since θ_0 and θ_1 are in the same unit, the gain is a dimensionless quantity.

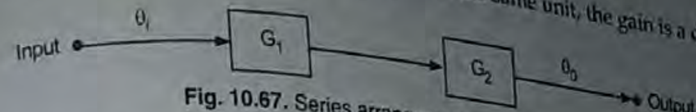


Fig. 10.67. Series arrangement of amplifiers

Quite often, two or more amplifiers are arranged in series/cascades to get greater amplification. Presuming that no loading occurs, the overall gain of the arrangement is given by the product of individual gains of the amplifying units, *i.e.*,

$$\frac{\theta_0}{\theta_1} = G_1\, G_2\, G_3 \ldots$$

EXAMPLE 10.18

A measuring system has two amplifiers arranged in series corresponding to an input of 10 units, the output from the system is 15 000 units. If the first amplifier has a gain of 75, determine the gain requirements of the second amplifier.

Solution : For a measuring system with two amplifiers arrangement in series,

$$\frac{\theta_0}{\theta_1} = G_1\, G_2$$

$$\frac{15000}{10} = 75 \times G_1 \; ; \; G_2 = \frac{15000}{10 \times 75} = 20$$

i.e., the second amplifier should has a gain of 20.

10.13.1. Mechanical amplifiers

Simple and compound levers

Figure 10.68. shows the arrangement and the associated block diagram for a simple lever supported on a pivot.

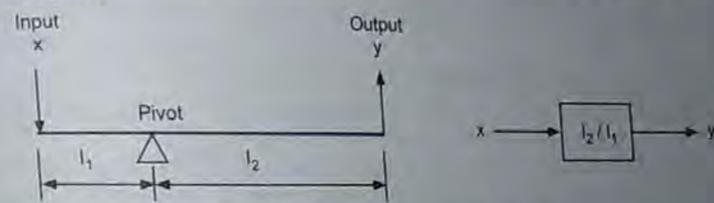


Fig. 10.68. Simple lever signal amplifier

For an input displacement x, the simple lever causes the output end to displace by an amount y ; x and y are related to each other by the expression

$$\frac{y}{x} = \frac{l_2}{l_1}$$

where l_1 is distance of input end from the pivot, and l_2 is distance of output end from the pivot. The ratio l_2/l_1 determines the displacement gain ; amplification of the lever can be varied depending on the relative distances of input and output ends from the pivot. With a simple lever, the input and out displacements are of opposite phase, *i.e.*, if the input x goes down the output y moves up.

A greater increase in amplification can be achieved by having a compound-lever system. The compound lever will have two or more levers linked together so that the output from one lever provides the input to the other.

For the lever arrangement shown in Fig. 10.69

$$\frac{y}{x} = \frac{l_2}{l_1} \quad \text{and} \quad \frac{z}{y} = \frac{l_4}{l_3}$$

Therefore, the overall gain or amplification is

$$\text{overall gain } G = \frac{z}{y} = \frac{l_2}{l_1} \times \frac{l_4}{l_3}$$

and
$$\text{output } z = \left(\frac{l_2}{l_1} \times \frac{l_4}{l_3}\right) \times x$$

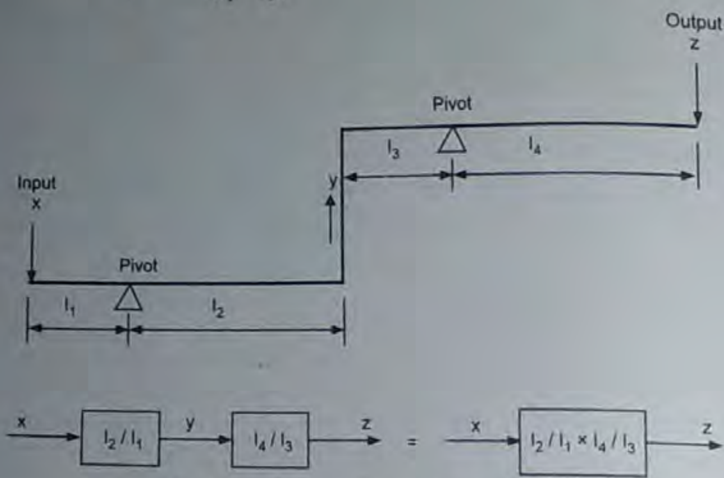


Fig. 10.69. Compound-lever signal amplifier

Apparently, there is no problem of phase reversal with the compound lever ; both the "d output y act along same directions.

EXAMPLE 10.19

Calculate the magnitude of the compound lever arrangement illustrated in Fig. 10.69. The length magnifying ratio of the levers are 2.5 : 1 and 4.5 : 1.

Solution: For the compound lever arrangement,

$$\frac{\theta_0}{\theta_1} = \frac{l_2}{l_1} \times \frac{l_4}{l_3}$$

$$= \frac{2.5}{1} \times \frac{4.5}{1}$$

Thus the compound lever has a displacement gain of 11.25.

10.13.2. Simple and compound gears

The simple and compound gear trains are used quite frequently to provide mechanical amplification of either angular displacement or rotary speed.

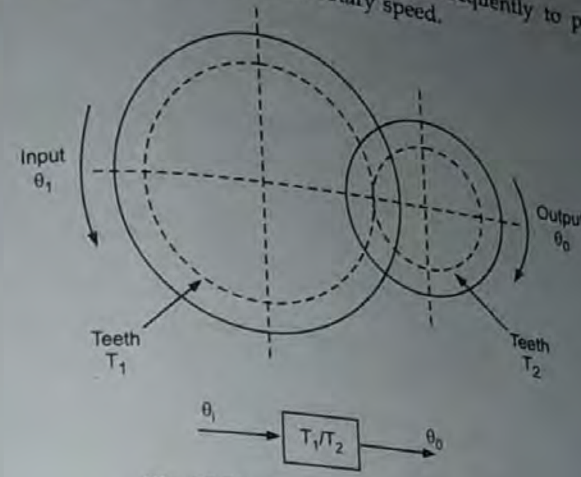


Fig. 10.70. Simple-gear signal amplifier

When the larger wheel (gear with teeth T_1) rotates with the input angular speed θi then each of its teeth fits into a corresponding space on the small wheel (gear with teeth T_2). That results into an angular speed θ_0 of the output gear ; θ_i and θ_0 are related to each other by the expression.

$$\frac{\theta_0}{\theta_1} = \frac{T_1}{T_2} = \frac{\text{number of the teeth on input gear wheel}}{\text{number of teeth on output gear wheel}}$$

The ratio T_1/T_2 determines the gain ; the amplification of a simple gear can be varied depending on the number of teeth on the input and output gear wheels. With a simple gear, the input and output displacements are of opposite phase, *i.e.*, if the input wheel rotates anti-clockwise the output wheel turns clock-wise.

A greater increase in amplification can be achieved by involving more gear wheels. In the compound gear train illustrated in Fig. 10.71, the input rotational signal θ_i is applied to wheel A which has T_1 signal teeth. The teeth on this wheel mesh with teeth on wheel B which has T_2 teeth. One rotation of wheel A would result into (T_1/T_2) rotations of wheel B as well as wheel C. This is due to the fact that gear wheels B and C are mounted on the same shaft. The gear wheel C has T_3 teeth and it drives the gear wheel D which has T_4 teeth.

Thus one rotation of wheel C would give (T_3/T_4) rotations to wheel D, and (T_1/T_2) rotations of wheel C would amount to $(T_1/T_2 \times T_3/T_4)$ rotations of wheel D. Thus overall magnification of gear would be teeth on wheel

$$\frac{\theta_0}{\theta_1} = \frac{T_1}{T_2} \times \frac{T_3}{T_4}$$

$$= \frac{\text{teeth on wheel } A}{\text{teeth on wheel } B} \times \frac{\text{teeth on wheel } C}{\text{teeth on wheel } D}$$

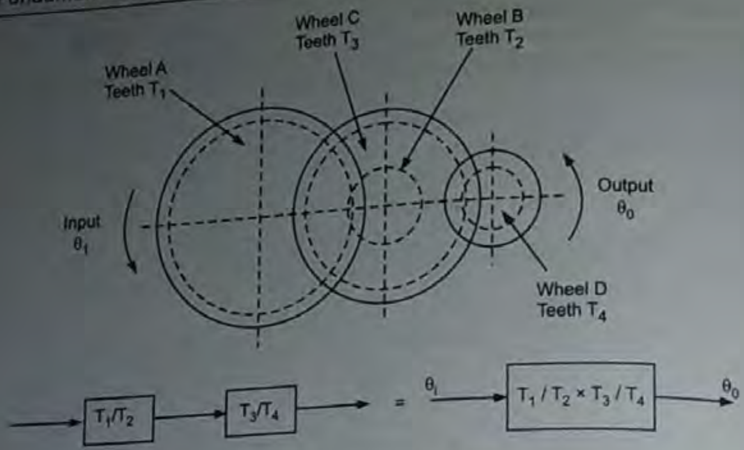


Fig. 10.71. Compound gear trains

A compound gear train gives, greater magnification with the additional advantage of no change in the direction of input signal. The gear trains are used for the magnification of displacement in the bourden tube pressure gauge and in the dial-test indicator where linear movement is translated into rotation by means of rack and pinion.

The mechanical amplification usually suffers from errors caused by :

- internal loading,
- friction at the mating parts,
- elastic deformation, and
- backlash.

EXAMPLE 10.20

Fig. 10.72 illustrates the gear arrangement for a particular dial test indicator. The plunger is a part of a rack and pinion mechanism so that movement of the plunger vertically upwards or

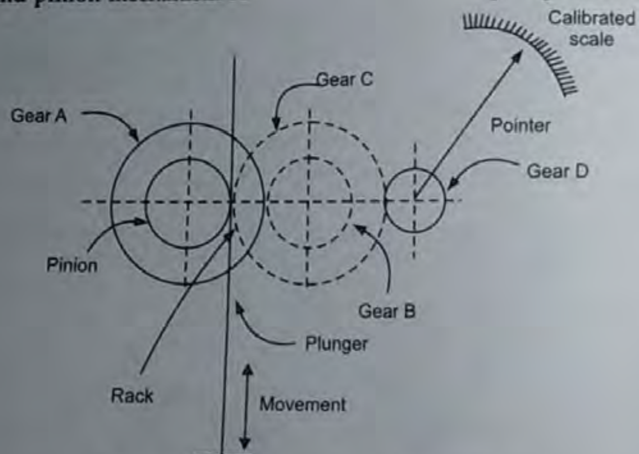


Fig. 10.72. Basic features of dial test indicator

downwards rotates the pinion. The rack has a tooth pitch of 5 teeth per cm and engages with a pinion having 15 teeth. The gears A and C have 30 teeth, and the gears B and D have 15 teeth respectively. Calculate the magnification and the number of revolutions of the pointer as the plunger moves 30 mm.

Solution: Corresponding to a plunger movement of 30 mm, the rack (5 teeth/cm) turns 15 teeth. Since there are 15 teeth on the pinion which engages with the rack, the pinion will rotate through one revolution.

Now, one revolution of pinion corresponds to

$$= 1 \text{ revolution of gear } A = 1 \times \frac{30}{15} \text{ revolutions of gear } B$$

$$= 1 \times \frac{30}{15} \text{ revolutions of gear } C$$

$$= 1 \times \frac{30}{15} \times \frac{30}{15} \text{ revolutions of gear } D$$

$$= 4 \text{ revolutions of gear } D$$

Since the pointer is attached to gear D, it will turn four revolutions as the plunger moves 30 mm.

10.13.3. Fluid amplifiers

Hydraulic amplifier: Refer 10.73.

When a small displacement x is applied to a piston operating inside a cylinder containing some liquid, there occurs a large displacement y of the liquid in the output tube which has a small diameter d. From volume balance of the liquid,

displacement of liquid in the cylinder = displacement of liquid in the output tube

$$\frac{\pi}{4} D^2 x = \frac{\pi}{4} d^2 y$$

Therefore the gain or amplification is

$$\text{gain } G = \frac{y}{x} = \left(\frac{D}{d}\right)^2$$

and

$$\text{output } y = \left(\frac{D}{d}\right)^2 x$$

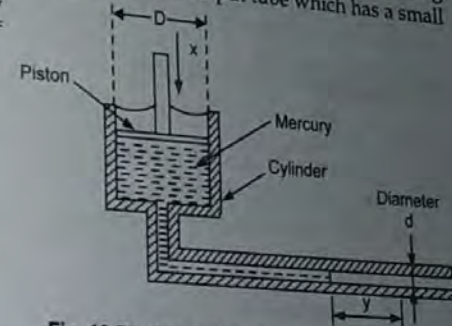


Fig. 10.73. Schematics of hydraulic amplification

This principle is employed in the mercury-in-glass thermometers and in the single column manometers.

EXAMPLE 10.21

In a fluid amplification system (Fig. 10.73), the amplification is to be 400. If the capillary tube is 1 mm diameter, calculate the required cylinder diameter.

Solution: Amplification $G = \dfrac{\text{movement of fluid meniscus in tube, } y}{\text{movement of pluger in cylinder, } x}$

From volume balance of liquid,

$$\frac{\pi}{4} D^2 x = \frac{\pi}{4} d^2 y \; ; \; \frac{y}{x} = \left(\frac{D}{d}\right)^2$$

$$G = \left(\frac{D}{d}\right)^2 \; ; \; 400 = \left(\frac{D}{d}\right)^2$$

$$\therefore \quad \frac{D}{d} = 20 \; ; \quad D = 20 \times 1 = 20 \text{ mm}$$

Thus the cylinder should have a diameter of 20 mm.

10.13.4. Pneumatic amplifier

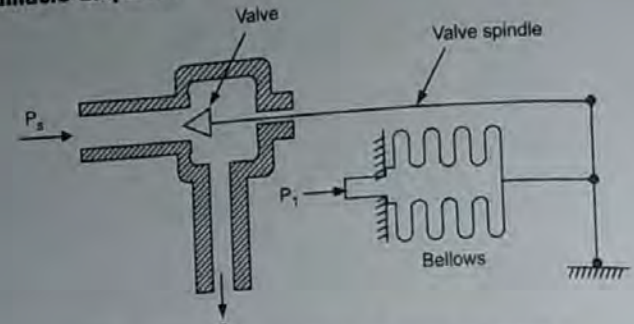


Fig. 10.74. Schematics of pneumatic amplification

When a pressure P_1 is applied to the bellows, the valve spindle moves towards right and that increases the gap between the valve and its seat. More of the high pressure air (P_s) rushes inwards and results into an increase in the output pressure P_2. Normally a linear relationship exists over a part of the range of movement of the valve : $P_2 = KP_1$, where K is a constant.

10.14. HYDRAULIC SYSTEMS

When the force required to affect a change in the flow variable is too large for self actuation by the fluid system, a separate force augmenting system is utilized. The force augmenting system may be hydraulic (using a liquid such as oil) or pneumatic (using a gas, particularly air).

Hydraulic actuators use a liquid control medium to provide an output signal which is a function of an input error signal. The schematics of a hydraulic actuating system are illustrated in Fig. 10.75; the major components being an error detector, an amplifier, a hydraulic control valve and an actuator.

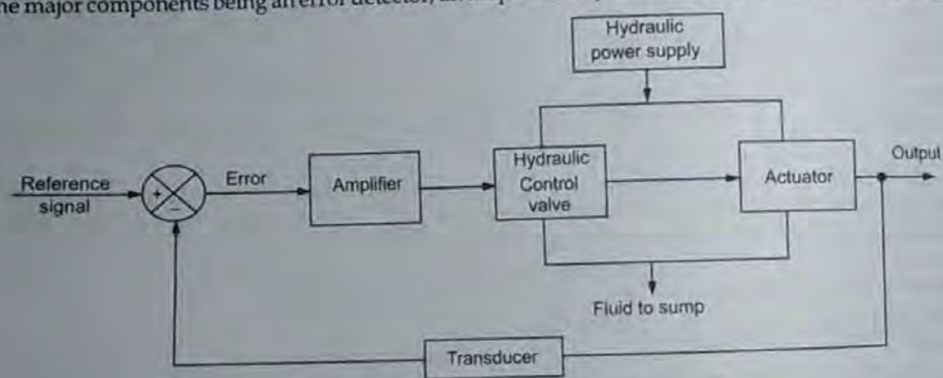


Fig. 10.75. Schematics of a hydraulic actuating system

There are two main types of high pressure hydraulic power supply systems :

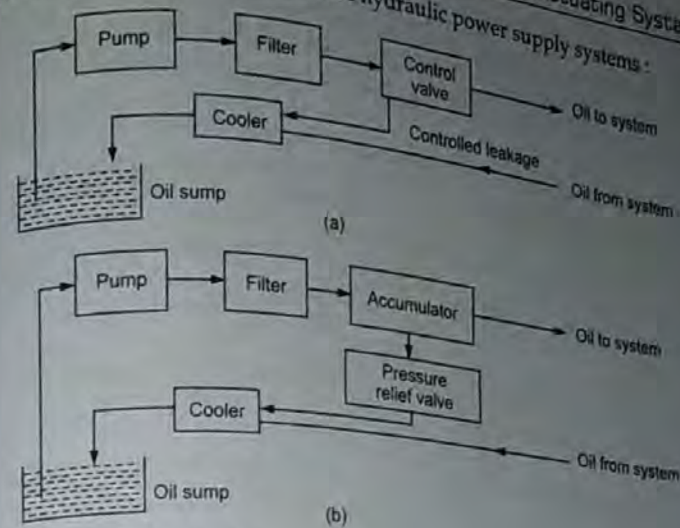


Fig. 10.76. Constant-flow and constant-pressure hydraulic actuator systems

(i) *Constant flow arrangement:* (Fig. 10.76 *a*) where a constant displacement type pump serves to draw oil from an oil sump and provides it at a constant flow rate. The high pressure oil is filtered and then led to a control valve with a controlled fluid leakage back to the sump. Under no load conditions ; the whole of the oil is returned to the sump. The arrangement works efficiently when full flow of oil is to be used for longer periods.

(ii) *Constant pressure arrangement :* (Fig. 10.76 *b*) where an accumulator is used which maintains oil at constant pressure, and also stores energy for peak power demands. At pressure values higher than the predetermined value, a pressure relief valve operates to bypass fluid between high pressure and return lines.

10.14.1. Hydraulic pump

There are three main types of pumps used in hydraulic systems viz., the gear pump, the vane pump and the piston pump.

Figure 10.77 shows the outline of a fixed displacement gear pump. The unit consists of two rotors which are mounted on separate parallel shafts. The oil trapped between the teeth of the suction port revolving gears and the metal casing is delivered from the suction port to the discharge port.

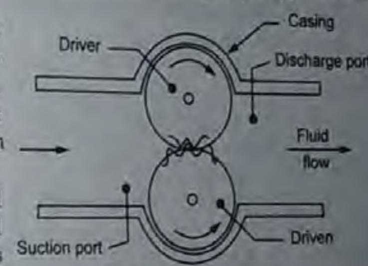


Fig. 10.77. Gear pump

To check leakage between the suction and discharge side of pump, the gear teeth are of special involute or cycloid construction. Meshing of gears serves both to transmit the drive and to maintain a seal between pressure and suction. Care is taken to ensure that the oil trapped between the successive lines of contact does not build up pressure. A change in flow direction can be affected by reversing the direction of the gear assembly.

The difficulties associated due to wear in gear pump are overcome in a vane pump which is shown in Fig. 10.78. The unit consists of a slotted rotor with a number of spring loaded radial vanes inserted in the slots. The rotor drum is mounted eccentrically in the casing and the vanes are allowed to slide into and out of rotor as its rotates. The space between the vanes is filled with the liquid at the suctions side and is carried to the delivery side. The quantity of liquid pumped and the direction of flow can be controlled by affecting a change in the degree of eccentricity.

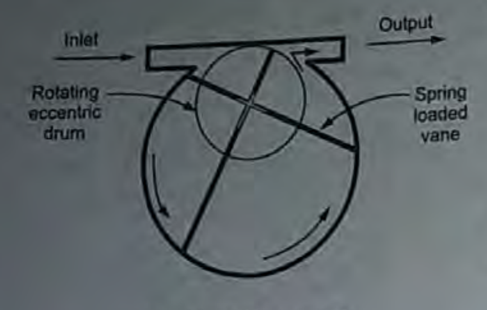


Fig. 10.78. Vane pump

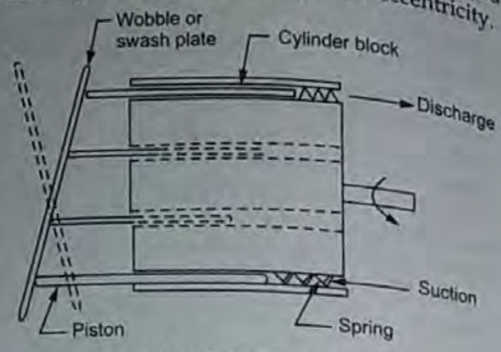


Fig. 10.79. Axial flow piston pump

Figure 10.79 depicts a simplified version of an axial flow piston pump. The unit consists essentially of a cylindrical block which is rotated along with its piston. This rotation causes the piston to move back and forth parallel to the shaft, and butt against the non-metal wobble or swash plate. The angle of wobble plate is either kept constant or is varied in accordance whether the pump is to have a fixed displacement or a variable displacement. No flow takes place when the wobble plate is perpendicular.

10.14.2. Hydraulic valves

These elements of a hydraulic actuating system function to regulate the flow of hydraulic fluid from the high pressure side to the actuator, *i.e.*, the hydraulic motor. There are three main types of valves ; the piston or spool type, the flapper and nozzle type, and the jet pipe valve. In all cases, the mechanical input motion can be controlled by manual operation, or by a limited motion electric motor or by hydraulic pilot method. The output results in a change of hydraulic pressure.

Piston or *spool valve* : The commonly used spool valve is constructed in either a three-way or a four-way valve arrangement as shown in Fig 10.80.

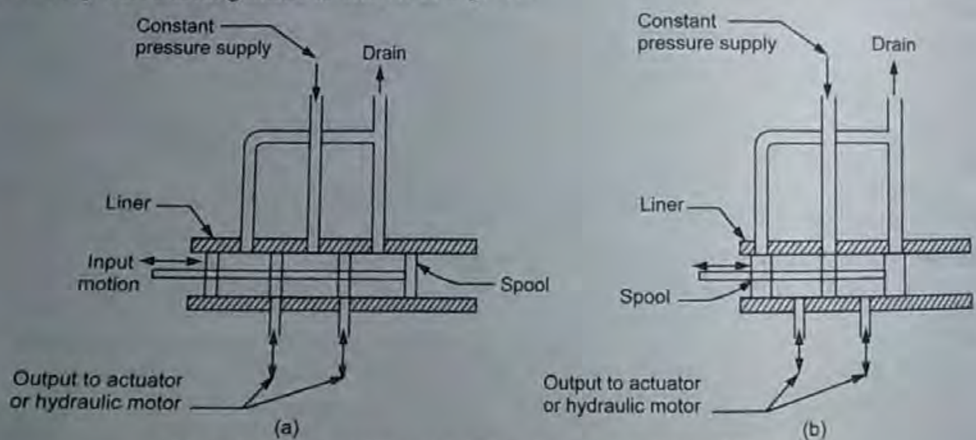


Fig. 10.80. Cylindrical spool valves : (a) 4-land, and (b) 3-land

When the spool is in the neutral position, the oil flow to the actuator is completely blocked. Displacement of the spool to right and left causes alternately pressure in one port to be higher than that in the other port. This is because when one of the pipelines to the actuator gets connected to the constant pressure supply, the other pipeline communicates with the drain. The differential pressure causes the hydraulic motor to rotate in a particular direction. The flow and, therefore, the motor speed are function of the spool-valve opening, affected somewhat by the load pressure.

Flapper nozzle valve : The operation of the unit is based on a variable leakage arrangement, which has the great virtues of simplicity and reliability. The unit incorporates two orifices in series, one of which is a fixed restriction and the other variable orifice consisting of a flapper and nozzle arrangement (Fig. 10.81). The nozzle restriction is changed as the flapper is positioned closer to or farther from the nozzle. The fluid at constant pressure passes through the restriction and a branch is led to the nozzle. When the flapper moves into a position that completely blocks the nozzle opening, there is very little leakage and the output pressure approaches that of the supply. With nozzle opening, there occurs an increase in pressure drop across the restriction, and consequently the output pressure diminishes. Thus the device produces a variable output pressure with flapper position.

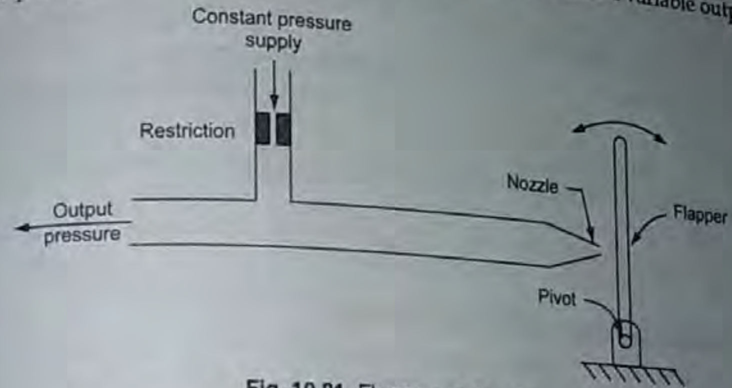


Fig. 10.81. Flapper nozzle valve

Jet-pipe valve : The device comprises a pivoted nozzle and two adjacent orifices. The nozzle converts the static pressure of the system into kinetic energy and then directs the high velocity jet of hydraulic fluid towards the orifices. During flow through orifices, the kinetic energy is reconverted to pressures ; the conversion being approximately 90% efficient at moderate supply pressure. This results from the act that friction can be reduced to minimum by making the space between the nozzle and orifices relatively large.

The flapper-nozzle and jet-pipe valve arrangements are frequently used as preamplifier to a piston valve.

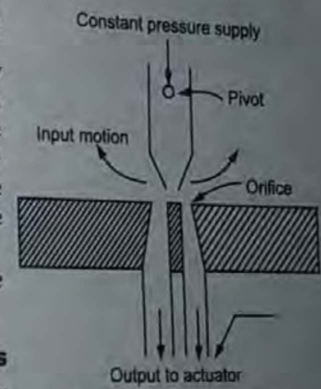


Fig. 10.82. Jet-pipe valve

10.14.3. Desirable characteristics of hydraulic fluids

- low level of impurities and high value of the bulk modulus
- sufficient film strength to prevent metal-to-metal contact between moving parts, i.e., to avoid wear of control valve, motor and pump elements
- adequate viscosity to give good seal at piston, glands and valves. At low viscosity, the moving parts would wear rapidly and there would be loss of fluid from the system. However

with highly viscous fluids, there would be excessive load to the moving parts and a considerable pressure drop along the feed lines

- good chemical stability, *i.e.*, resistance to formation of sludge and gum. The formation of emulsion during the churning of fluid as it passes through pumps, valves, piping and motors would reduce lubricity and often cause rusting
- freedom from acidity, a low pour point temperature, a high flash point, resistance to foaming, and anti-rust properties.

Petroleum based oils, with or without additives, are the most common of hydraulic fluids because of their superior lubricating and corrosive protecting features. However certain synthetic fluids like phosphate or silicate easter compounds, halogenated fluids, and silicon fluid are now being preferred because of their good fire resistance characteristics.

10.14.4. Advantages and limitations of hydraulic systems

- high response due to effectively incompressible nature of the liquid control medium
- high power gain due to readily conversion of liquids to high pressure or flows. The high energy liquid can be effectively piloted by the hydraulic controllers
- simplicity of the actuator system
- long life due to self-lubricating properties of the hydraulic liquids
- requirements of proper seals and connection so as to prevent the leakage of hydraulic fluid
- careful maintenance of the system to keep the fluid clean and pure
- stringent requirements for the hydraulic fluid to be fire-resistant, anti-corrosion and self-lubricating.

Their high power-to-weight ratio results in their finding a wide range of use in machine tools, speed governing systems and position control systems.

10.15. PNEUMATIC ACTUATORS SYSTEMS

Pneumatic controllers use air control medium to provide an output signal which is a function of an input error signal. The schematics of a pneumatic control system are illustrated in Fig. 10.83; the major components being an error detector, flapper-nozzle (controller mechanism), and an amplifier or pilot relay.

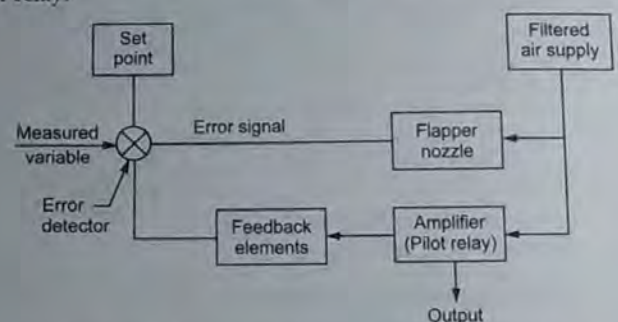


Fig. 10.83. Schematics of a pneumatic actuating system

The pneumatic actuating mechanisms are of two types : force balance and motion balance. In a force balance control (Fig. 10.84 *a*), the deviation of a diaphragm is proportional to the pressure difference (P_s - P_0); P_s and P_0 being the set-point and output signal pressures, respectively. In a motion balance controller (Fig. 10.84 *b*) the deviation signal is taken from a point on a mechanical linkage. A control on the deviation movement $(l_2 x - l_1 x)/(l_1 + l_2)$ can be affected by altering the values of l_1 and l_2.

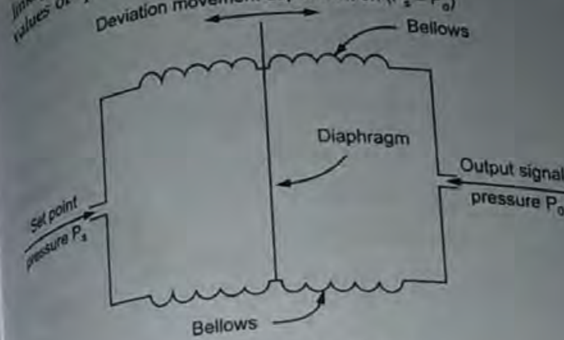


Fig. 10.84. (a) Force-balance actuator

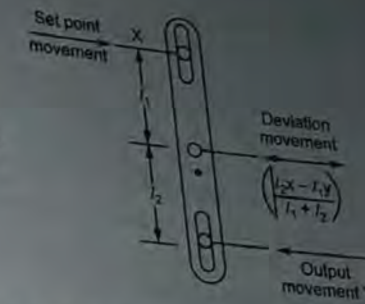


Fig. 10.84. (b) Motion-balance actuator

10.15.1. Pneumatic nozzle-flapper

A nozzle-flapper is a basic component of pneumatic and hydraulic measurements, control and transmission, and as a precision gauging equipment. Figure 10.85 shows a schematic diagram of a pneumatic nozzle-flapper amplifier.

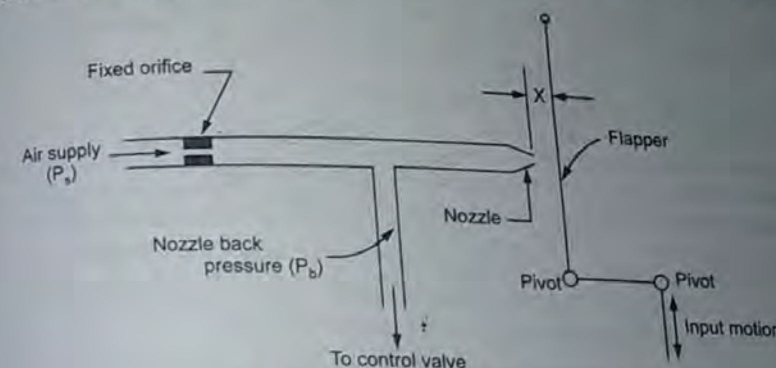


Fig. 10.85. Pneumatic nozzle flapper

The unit consists essentially of nozzle supplied with pressurized air (1.4 bar gauge) thro[ugh] an orifice restriction- The orifice diameter is approximately 0.25 mm and that of the nozzl[e] 0.675 mm. For proper functioning, the nozzle diameter must be larger than the orifice diam[eter]. Just in front of the nozzle, there is a flapper which is positioned by the input motion. As fla[pper] approaches the nozzle, there is an increase in resistance to the flow of air through the noz[zle] consequently the nozzle back pressure increases. If the nozzle is completely closed by the fla[pper] air cannot escape through the nozzle.

Consequently maximum air passes at the supply pressure P_s. When the nozzle-flapper di[stance] x is made wider by moving the flapper away from the nozzle, there is practically no rest[riction] and most of the air escapes to atmosphere. The nozzle back pressure takes on a minimum the lowest possible value being the atmospheric pressure P_a. Thus a movement x of the flapper causes proportional change in the nozzle back pressure P_b if the supply pressure P_s is kept constant. Figure 10.86 shows a typical curve relating the back pressure P_b to the nozzle flapper distance x. The slope of the curve at any point is called the nozzle sensitivity or gain. For the commonly used 0.25 mm diameter orifice and 0.625 mm nozzle combination, 0.3625 mm motion of the baffle causes a change in the nozzle back pressure greater than 0.56 bar in the central portion of the curve. The steep and almost linear part of the curve is utilized in actual operation of the nozzle-flapper unit.

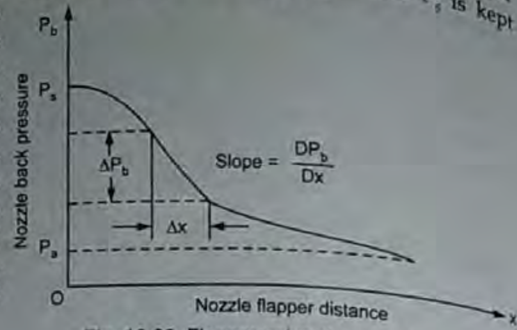


Fig. 10.86. Flapper-valve characteristic

10.15.2. Pneumatic relay

To increase gain and to have larger outputs required for the operation of large pneumatic actuators, a pneumatic relay is often used in conjunction with the nozzle-flapper unit. Figure 10.88 shows the outline of the most common from of a pneumatic relay widely used in process control. When the nozzle back pressure increases, the ball valve is forced towards the lower seat. The air supply is shut off and output pressure P_0 to the pneumatic valve drops. When the ball rests on its upper seat, the exhaust port is closed and the control pressure P_0 becomes equal to be supply pressure P_s. The control pressure can thus be varied from 0 bar to full supply pressure of 1.2 bar (gauge). The movement of the ball takes place due to change in the value of nozzle back pressure which results from displacement of flapper in front of the nozzle. The relay-system is of bleed type since at all positions of the ball except at the top-seat, the air continues to bleed into the atmosphere.

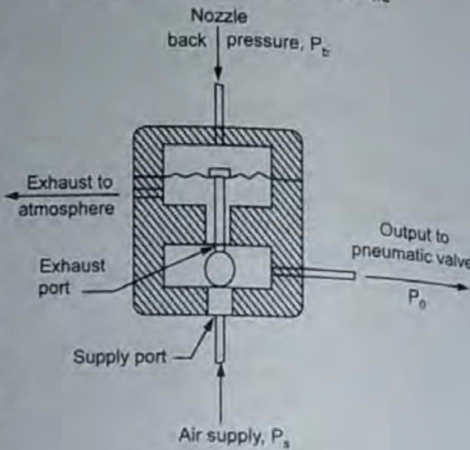


Fig. 10.87. Pneumatic relay unit

10.15.3 Single acting and double acting pneumatic actuators

In a pneumatic actuator, compressed air is the basic energy source. A reciprocating compressor sucks air from the atmosphere, compresses it to the requisite pressure and the compressed air is

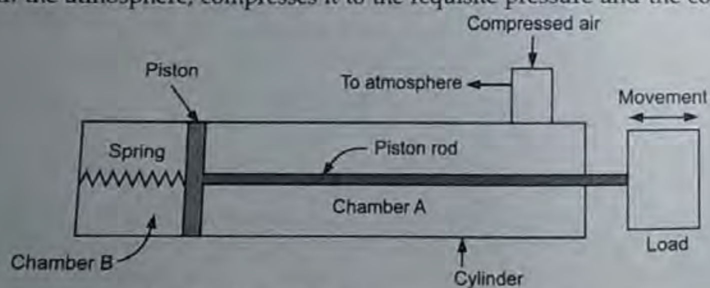


Fig. 10.88. Single acting pneumatic actuator

stored into a pressure vessel called the air receiver. This air is required to be dry and clean free from dirt and dust. For that a cooler is used to reduce the air temperature after compression and a separator is employed to remove any water vapour that may be present in the air.

In the single acting pneumatic actuator, spring force allows the piston to move linearly in one direction only into a cylinder which is a hollow chamber. The pressurized air from the receiver enters the cylinder into chamber A and overcomes the spring force. When the supply of compressed air is stopped and the air inside the chamber is allowed to escape to atmosphere, the piston travels towards right.

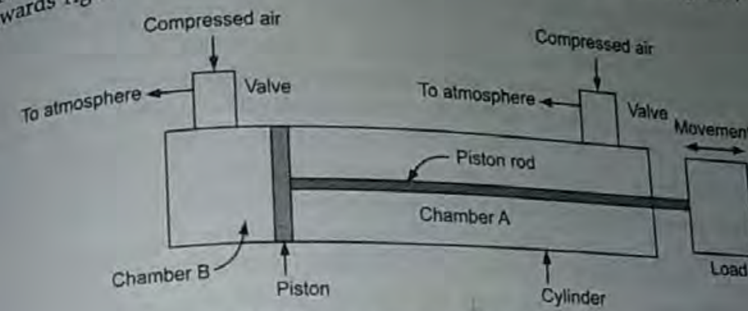


Fig. 10.89. Double acting pneumatic actuator

The double acting pneumatic arrangement has two ports (valves) for compressed air supply and its subsequent release to atmosphere. When pressurized air enters the chamber A, there is escape of air from the chamber B and the piston is pushed extreme on the left side of the cylinder. During next stroke, entry of pressurized air is chamber B, the escape of air to atmosphere is from chamber A and the piston movement is towards right.

The single and double acting linear pneumatic actuators have key advantages of:

- compact design and reliable operation
- rugged construction and suitable for harsh environments
- easy maintenance
- positioning accuracy and versality of use and application

These actuators have wide applications in industries where load needs to be lifted or lowered, pushed or pulled, and positioned. The main industries using linear actuators are:

- material handling and packaging
- food processing and pharmaceutical
- machine tool, automotive and defence

The *Rotary Actuators* have been designed and developed for the applications requiring a rotary motion or torque to control the speed and rotation of the attached equipment. In these actuators:

- there is transformation of pneumatic, hydraulic or electric energy to mechanical rotation
- the linear motion in one direction gives rise to a turning or angular movement through a pre-set (defined) angle.

The pneumatic rotary actuators utilize the pressure of compressed air to generate the oscillatory rotary motion. For operation, the force is applied at a distance away from the axis of rotation and that causes a turning moment. The two most common configurations of pneumatic rotary actuators are :

1. **Rotary vane actuator:** The vane actuator consists of a vane mounted on a central shaft/ spindle enclosed in a cylindrical chamber. A stream of pressurized air is made to impinge upon the vane and that push makes the spindle to turn. There is a port through which the air behind the vane is released to the atmosphere. When the vane has turned through a specfied angle, the air flow is reversed and the spindle rotates back to its original position. The stroke gets completed and subsequently the process is repeated.

The vane actuators are used for light loads.

2. **Rack and pinion actuator:** The rack and pinion actuator consists of a piston and rack that moves linearly and causes a pinion gear and output shaft to rotate. The rack is machined as a part of the piston rod of a double acting linear cylinder. The pinion gear meshes with the rack and turns the spindle as the piston moves when air pressure is applied to it. The spindle is at right angles to the piston and rotates clockwise, then anticlockwise as the linear cylinder completes its double action.

The rack and pinion actuators are used when more speed is required.

The key merits of pneumatic rotary actuators are:

zero backlash, high force output, high repeatability and positioning accuracy, less wear and maintenance and greater durabilit; ability to work effectively in hazardous environments.

These actuators are being successfully employed in the following industries for work with greater precision :

Robotics and CNC machines, Aerospace and flight simulation, Radar and monitoring systems, Medical industry.

10.15.4. Advantages and limitations of pneumatic systems

- simplicity of components and ease of maintenance
- explosion proof characteristics ; freedom from the hazards
- relatively high power amplification for operating the final control elements
- relatively inexpensive power system; abundance and free supply of air
- slow response of final control elements, and transmission lag
- operation difficult under freezing conditions.