The significance of fluid mechanics can be well judged by citing just one example of automobile drive where suspension is provided by pneumatic tyres, road shocks are reduced by hydraulic shock absorbers, gasoline is pumped through tubes and later atomized, air resistance creates a drag on the vehicle as whole and the confidence that hydraulic brakes would operate when the vehicle is made to stop.

Undoubtedly, a study of the science of fluid mechanics is a must for an engineer so that he can understand the basic principles of fluid behaviour and apply the same to flow situations encountered in engineering and physical problems.

5.4 FLUID PROPERTIES

Every fluid has certain characteristics by means of which its physical condition may be described. Such characteristics are called *properties* of the fluid. Before an analyst of fluid flow problems can venture to formulate the physical principles governing the flow situation, he has to be thoroughly familiar with the physical properties of fluids. Towards that end, this section seeks to provide basic insight into the fluid properties and their behaviour.

5.4.1 Specific weight, mass density and specific gravity

(*a*) *Specific weight* (w) of a fluid is its weight per unit volume.

$$w = \frac{W}{V} \qquad \text{... (5.1)}$$

where W is the weight of the fluid having volume V. The weight of a body is the force with which the body is attracted to the centre of the earth. It is the product of its mass and the local gravitational acceleration, *i.e.*, $W = mg$. The value of g at sea level is 9.807 m/s² approximately. Since weight is expressed in newton, the unit of measurement of specific weight is N/m³. In terms of fundamental units, the dimensional formula of specific weight is $\left[\frac{F}{L^3}\right]$ or $\left[\frac{M}{L^2T^2}\right]$.

For pure water under standard atmospheric pressure of 760 mm of mercury at mean sea level and a temperature of 4°C, the specific weight is 9810 N/m³. For sea water, the specific weight equals 10 000 - 10 105 N/m³. The increased value of specific weight of water is due to the presence of dissolved salts and suspended matter. The specific weight of petroleum and petroleum products varies from 6 350 - 8 350 N/m³ and that of mercury at 0°C is 13 420 N/m³. Air has a specific weight of 11.9 N/m³ at 15°C temperature and at standard atmospheric pressure. The specific weight of a fluid changes from one place to another depending upon changes in the gravitational acceleration.

(*b*) ***Density*** (ρ-pronounced rho) is a measure of the amount of fluid contained in a given volume and is defined as the mass per unit volume.

$$\rho = \frac{m}{V} \qquad \text{... (5.2)}$$

where m is the mass of fluid having volume V. Fluid mass is a measure of the ability of a fluid particle to resist acceleration and is approximately independent of its location on the earth's surface. The units of density correspond to those of mass and volume. The dimensional formula of density in fundamental units is $\left[\frac{M}{L^3}\right]$ or $\left[\frac{FT^2}{L^4}\right]$ and the corresponding units are kg/m³ or N s²/m⁴.

The density of a fluid diminishes with rise of temperature except for water which has a maximum value at 4°C. The mass density of water at 15.5°C is 1000 kg/m³, and for air at 20°C and at atmospheric pressure the mass density is 1.24 kg/m³.

Relations 5.1 and 5.2 are valid only when the fluid medium fills the given volume completely without any blank space, *i.e.*, the fluid is a continuum. For a non-homogeneous fluid, these relations give average specific weight and density. To determine the absolute values of w and ρ at any point, the volume is regarded as tending to zero and the limit of the corresponding ratio is calculated.

$$w = \underset{V \to 0}{Lt} \frac{m}{V} = \frac{dW}{dV}$$

$$\rho = \underset{V \to 0}{Lt} \frac{m}{V} = \frac{dm}{dV}$$

The weight W and the mass m of a fluid are related to each other by the expression $W = mg$. Dividing this expression throughout by volume V of the fluid, we obtain :

$$\frac{W}{V} = \frac{m}{V} g \quad \text{or} \quad w = \rho g \qquad \text{... (5.3)}$$

Equations 5.3 reveals that specific weight w changes with location depending upon gravitational pull.

(*c*) ***Specific gravity*** (s) refers to the ratio of specific weight (or mass density) of a fluid to the specific weight (or mass density) of a standard fluid. For liquids the standard fluid is water at 4°C, and for gases the standard fluid is taken either air at 0°C or hydrogen at the same temperature. Specific gravity is dimensionless and has no units.

A statement that the specific gravity of mercury is 13.6 implies that its weight (or mass) is 13.6 times that of same volume of water. In other words, mercury is 13.6 times heavier than water.

(*d*) ***Specific volume*** (v) represents the volume per unit mass of fluid; specific volume is the inverse of the mass density.

$$v = \frac{V}{m} \; ; v = \frac{1}{\rho} \qquad \text{... (5.4)}$$

The concept of specific volume is found to be practically more useful in the study of flow of compressible fluids, *i.e.*, gases.

EXAMPLE 5.1

2 litre of petrol weighs 14 N. Calculate the specific weight, mass density, specific volume and specific gravity of petrol with respect to water.

Solution : 2 litre = 2×10^{-3} m³

Specific weight is a measure of the weight per unit volume :

$$\therefore \quad \text{Specific weight } w = \frac{14}{2 \times 10^{-3}} = \mathbf{7000\ N/m^3}$$

Mass density is related to specific volume by the relation,

$$w = \rho g$$

$$\text{Mass density } \rho = \frac{w}{g} = \frac{7000}{9.81} = \mathbf{713.56\ kg/m^3}$$

Specific volume v is the inverse of mass density

$$v = \frac{1}{\rho} = \frac{1}{713.56} = \mathbf{1.4 \times 10^{-3}\ m^3/kg}$$

$$\text{Specific gravity } s = \frac{\text{density of oil}}{\text{density of water}} = \frac{713.56}{1000} = \mathbf{0.7136}$$

EXAMPLE 5.2

If specific gravity of a liquid is 0.80, make calculations for its mass density, specific volume and specific weight (weight density).

Solution : Specific gravity = $\frac{\text{mass density of liquid}}{\text{mass density of water}}$

$\therefore$ Mass density of liquid $\rho = 0.80 \times 1000 = \mathbf{800\ kg/m^3}$

$$\text{Specific volume } v = \frac{1}{\rho} = \frac{1}{800} = \mathbf{1.25 \times 10^{-3}\ m^3/kg}$$

Specific weight (weight density) $w = \rho\, g = 800 \times 9.81 = \mathbf{7848\ N/m^3}$

5.4.2 Viscosity

Viscosity is a property of the fluid by which it offers resistance to shear or angular deformation. Experimental evidence indicates that when any fluid flows over a solid surface the velocity is not uniform at any cross-section; it is zero (no slip) at the solid surface and progressively approaches the free stream velocity in the fluid layers far away from the solid surface. This aspect of the velocity profile (a curve connecting the tips of velocity vectors) indicates the existence of some resistance to flow due to friction between a fluid layer and the solid surface, and between adjacent layers of fluid itself. Again the velocity gradient (the spatial rate of change of velocity du/dy) is large at the solid surface and gradually diminishes to zero with distance from the wall. Evidently the resistance between the fluid and surface is greater when compared to that between the fluid layers themselves.

The resistance to flow because of internal friction is called ***viscous resistance***, and the property which enables the fluid to offer resistance to relative motion between adjacent layers is called the ***viscosity*** of fluid. Viscosity is thus a measure of resistance to relative translational motion of adjacent layers of fluid. This property is manifested by all the real fluids, and it distinguishes them from ideal or non-viscous fluids. Molasses, tar and glycerine are examples of highly viscous liquids ; the intermolecular force of attraction between their molecules is very large and consequently they cannot be easily poured or stirred. Fluids like water, air and petrol have a very small viscosity; they flow much more easily and rapidly and are called thin fluids.

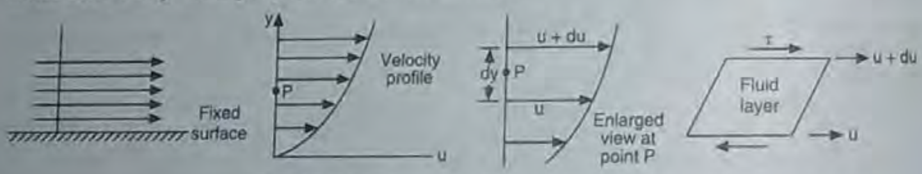


Fig. 5.3. Velocity profile and viscosity concept

Newton's Law of Viscosity

Consider two adjacent layers at an infinitesimal distance dy apart and moving with velocity u and $(u + du)$, respectively. The upper layer moving with velocity $(u + du)$ drags the lower layer along with it by exerting a force F. However, the lower layer tries to retard or restrict the motion of upper layer by exerting a force equal and opposite to F. These two equal and opposite forces induce a shear or viscous resistance τ (pronounced tau) given by F/A where A is the contact area between the two layers. Experimental measurements have shown that the shear stress is proportional to the spatial rate of velocity normal to the flow :

$$\tau \propto \frac{du}{dy};\ \tau = \mu\,\frac{du}{dy} \qquad \text{... (5.5)}$$

The term $\frac{du}{dy}$ is more usually called the velocity gradient at right angles to the direction of velocity itself. The proportionality constant μ (pronounced mew) in equation 5.5 is a function of the fluid involved and is called the ***coefficient of viscosity, absolute viscosity*** or ***dynamic viscosity***. Equation 5.5 was first suggested by Newton and is referred to as the ***Newton's viscosity equation*** or ***Newton's law of viscosity***.

The following observations help to appreciate the interaction between viscosity and velocity distribution :

- Maximum shear stresses occur where the velocity gradient is the largest, and the shear stresses disappear where the velocity gradient is zero.
- Velocity gradient at the solid boundary has a finite value. The velocity profile cannot be asymptotic to the boundary because that would imply an infinite velocity gradient and, in turn, an infinite shear stress.
- Velocity gradient becomes less steep (du/dy becomes small) with distance from the boundary. Consequently maximum value of shear stress occurs at the boundary and it progressively decreases with distance from the boundary.

Deformation of fluid elements can be prescribed in terms of the angle of shear strain $d\theta$. Figure 5.4 indicates a thin sheet of fluid element ABCD placed between two plates distance dy apart. The length and the width of the plates are much larger than the thickness dy so that the edge effects can be neglected. When force F is applied to the upper plate, it causes it to move at a small speed du relative to the bottom plate. Velocity gradient sets up a shear stress $\tau = F/A$ which makes the fluid element distort to position AB'C'D after a short time interval dt.

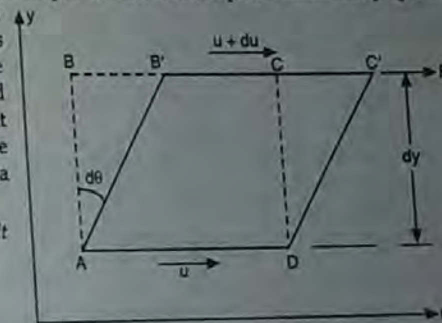


Fig. 5.4 Shear stress and time rate of shear strain

Distance BB' = CC' = speed × time = $du \times dt$

For small angular displacement $d\theta$, BB'

$= dy \times d\theta$

$\therefore \quad du \times dt = dy \times d\theta$;

$$\frac{du}{dy} = \frac{d\theta}{dt}$$

Invoke Newton's law of viscosity, *i.e.*, express the shear stress in terms of velocity gradient.

$$\tau = \mu\frac{du}{dy};\ \tau = \mu\frac{d\theta}{dt} \qquad \text{... (5.6)}$$

Apparently the shear stress in fluids is dependent on the rate of fluid deformation $\frac{d\theta}{dt}$. This characteristic serves to distinguish a solid from a fluid. Whereas the shear stress in a solid material is generally proportional to shear strain; the shear stress in a viscous fluid is proportional to time rate of strain.

Dimensional Formula and Units of Viscosity

The units of viscosity can be worked out from Newton's equation of viscosity; $\tau = \mu\ du/dy$. Solving for the viscosity μ and inserting dimensions F, L, T for force, length and time :

$$\mu = \frac{\tau}{du/dy} = \left[\frac{F}{L^2}\right] \div \left[\frac{L}{T} \times \frac{1}{L}\right] = \left[\frac{FT}{L^2}\right]$$

When the force dimension is expressed in terms of mass, $F = \left[\frac{ML}{T^2}\right]$, the dimensions for viscosity in terms of mass, length and time become $\left[\frac{M}{LT}\right]$.

When appropriate units are inserted for force, length and time, the dynamic viscosity will have the units :

$$\mu = \frac{\tau}{du/dy} = \frac{\text{N/m}^2}{\left(\frac{\text{m}}{\text{s}} \times \frac{1}{\text{m}}\right)} = \frac{\text{N s}}{\text{m}^2} = \text{Pa s}$$

Sometimes, the coefficient of dynamic viscosity μ is distinguished by poise (P)

$$1 \text{ poise} = \frac{1 \text{gm}}{\text{cm sec}} = \frac{1 \text{ dyne sec}}{\text{cm}^2}$$

$$= \frac{10^{-5}}{(10^{-2})^2} \frac{\text{N s}}{\text{m}^2} = \frac{0.1 \text{N s}}{\text{m}^2} = 0.1 \text{ Pa s}$$

A poise turns out to be a relatively large unit, hence the unit centipoise (cP) is generally used : 1 cP = 0.01 P. Typical values of viscosity for water and air at 20°C and at standard atmospheric pressure are :

$$\mu \text{ water} = 1.0 \text{ cP} = 10^{-3} \text{ N s/m}^2$$

$$\mu \text{ air} = 0.0181 \text{ cP} = 0.0181 \times 10^{-3} \text{ N s/m}^2$$

i.e., water is nearly 55 times as viscous as air.

Specific viscosity is the ratio of the viscosity of fluid to the viscosity of water at 20°C. Since water has a viscosity of 1 cP at 20°C, the viscosity of any fluid expressed in centipoise units would be a measure of the viscosity of that relative to water.

Kinematic Viscosity

The ratio between the dynamic viscosity and density is defined as ***kinematic viscosity*** of fluid and is denoted by ν (pronounced new) :

$$\text{Kinematic viscosity} = \frac{\text{dynamic viscosity}}{\text{mass density}}; \nu = \frac{\mu}{\rho} \qquad \text{... (5.7)}$$

The dimensional formula for kinematic viscosity is :

$$\nu = \left[\frac{M}{LT}\right] \div \left[\frac{M}{L^3}\right] = \left[\frac{L^2}{T}\right]$$

The kinematic viscosity does not involve force; its only dimensions being length and time as in kinematics of fluid flow. Typical units of ν are m^2/s or cm^2/s, the latter being referred to as stoke (St) to perpetuate the name of the English physical Sir George Stokes. A centistoke (c St) is one-hundredth of a stoke : 1 c St = 0.01 St. Typical values of kinematics viscosity at 20°C and at standard atmospheric pressure are :

$$\nu \text{ water} = 1.0 \text{ c St} = 1 \times 10^{-6} \text{ m}^2/\text{s}$$

$$\nu \text{ air} = 15.0 \text{ c St} = 15 \times 10^{-6} \text{ m}^2/\text{s}$$

i.e., the kinematic viscosity of air is about 15 times greater than the corresponding value of water.

EXAMPLE 5.3

(i) A lubricating oil of viscosity μ undergoes steady shear between a fixed lower plate and an upper plate moving at speed *V*. The clearance between the plates is *h*. Show that a linear velocity profile results if the fluid does not slip at either plate.

(ii) Two horizontal plates are placed 1.25 cm apart, the space between them being filled with oil of viscosity 14 poise. Compute the shear stress in the oil if the upper plate is moved with a velocity of 2.5 m/s.

Solution : The shear stress τ is constant throughout the fluid for the given geometry and motion and, therefore, from Newton's law of viscosity

$$\frac{du}{dy} = \frac{\tau}{\mu} = \text{constant}$$

$$\text{or} \qquad u = a + by$$

The constants *a* and *b* are evaluated from the no slip conditions at the upper and lower plates.

$$u = 0 \text{ at} = y = 0 \text{ ; } 0 = a$$

$$u = V \text{ at } y = h \text{ ; } V = a + bh$$

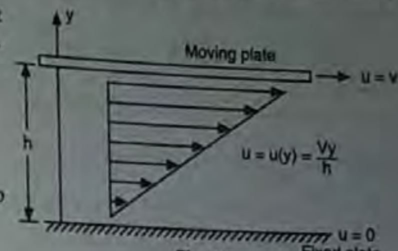


Fig. 5.5

Hence $a = 0$ and $b = V/h$. The velocity profile between the plates is than given by $u = \frac{Vy}{h}$ and is linear as indicated in Fig. 5.6.

(ii) Viscous shear stress is given by the Newton's law of viscosity

$$\tau = \mu \frac{du}{dy}$$

Given $\mu = 14 \text{ poise} = 1.4 \text{ N s/m}^2$

$du = 2.5$ m/s and $dy = 1.25 \times 10^{-2}$ m

$$\therefore \qquad \tau = 1.4 \times \frac{2.5}{1.24 \times 10^{-2}} = 280 \text{ N/m}^2 = \mathbf{280\ Pa}$$

Although oil is very viscous, this is a modest shear stress; about 360 times less than atmospheric pressure.

EXAMPLE 5.4

The clearance space between a shaft and a concentric sleeve has been filled with a Newtonian fluid. The sleeve attains a speed of 60 cm/s when a force of 500 N is applied to it parallel to the shaft. What force is needed if it is desired to move the sleeve with a speed of 300 cm/s ?

Solution : For a Newtonian fluid, $\tau = \mu \frac{du}{dy}$. Since the space between the shaft and the sleeve is very small, *i.e.*, the oil film is thin, it can be presumed that $\frac{du}{dy} = \frac{u}{t}$ where *u* is the sleeve speed and *t* is the oil film thickness. Further

$$\text{Shear stress } \tau = \frac{\text{force}}{\text{area}} = \frac{F}{A}$$

$$\therefore \qquad \frac{F}{A} = \mu \frac{u}{t} \quad \text{or} \quad F = A \mu \frac{u}{t}$$

A, μ and t are constant and therefore $F \propto u$ and accordingly $\frac{F_1}{u_1} = \frac{F_2}{u_2}$

Inserting the appropriate values,

$$\frac{500}{60} = \frac{F_2}{300}\ ;\ F_2 = 2500 \text{ N}$$

EXAMPLE 5.5

Two horizontal flat plates are placed 0.15 mm apart and the space between them is filled with an oil of viscosity 1 poise. The upper plate of area 1.5 m² is required to move with a speed of 0.5 m/s relative to the lower plate. Determine the necessary force and power required to maintain this speed.

Solution : Viscous shear stress $\tau = \mu \frac{du}{dy}$

Given $\mu = 1$ poise $= 0.1$ N s/m²; $du = 0.5$ m/s

$dy = 0.15$ mm $= 0.15 \times 10^{-3}$ m

$$\therefore \quad \text{Shear stress } \tau = \frac{0.1 \times 0.5}{0.15 \times 10^{-3}} = 333.3 \text{ N/m}^2$$

(*i*) Shear resistance or force,

F = shear stress × area

$= 333.3 \times 1.5 = $ **500 N**

(*ii*) Power required to move the upper plate at a speed of 0.5 m/s.

$= Fu = (500 \times 0.5)$ Nm/s $= 250$ W $=$ **0.25 kW**

EXAMPLE 5.6

A dash pot 10 cm diameter and 12.5 cm long slides vertically down in a 10.05 cm diameter cylinder. The oil filling the annular space has a viscosity of 0.80 poise. Find the speed with which the piston slides down if load on the piston is 10 N.

Solution : Since the space between the dash pot and the cylinder is very small, *i.e.*, the oil film is thin, we can presume that $\frac{du}{dy} = \frac{u}{t}$ where u is the piston speed and t is the oil film thickness.

$$\text{Shear stress } \tau = \mu \frac{du}{dy} = \mu \frac{u}{t}$$

Shear or viscous force = shear stress × area $= \mu \frac{u}{t} (2\pi rl)$

Given : $r = \frac{10}{2} = 5$ cm $= 0.05$ m

$u = 0.8$ poise $= 0.08$ N/sm²

$t = \frac{10.05 - 10}{2} = 0.025$ cm $= 0.00025$ m

Viscous force equals the load of 10 N

$$\therefore \quad 10 = 0.08 \times \frac{u}{0.00025} \times (2\pi \times 0.05 \times 0.125)$$

Hence piston speed $u =$ **0.796 m/s**

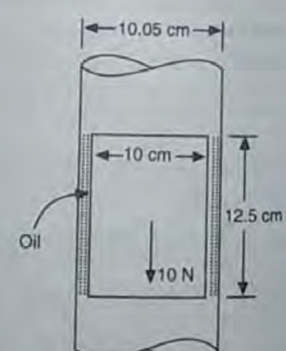


Fig. 5.6

EXAMPLE 5.7

A cylinder of diameter 15 cm and weight 90 N slides a distance of 12.5 cm in a lubricated pipe. The clearance between the cylinder and pipe is 2.5 × 10⁻³ cm. The cylinder is noted to decelerate at a rate of 0.6 m/s² when the speed is 6 m/s. Calculate the viscosity of the oil used for lubricating the pipe.

Solution : Viscous shear stress $\tau = \mu \frac{du}{dy} = \mu \frac{u}{t}$

Viscous resistance or force = shear stress × area

$$= \mu \frac{u}{t} \times \pi dl = \frac{\mu \times 6}{2.5 \times 10^{-5}} \times \pi (0.15)(0.125) = 14130\ \mu \text{ N}$$

Invoking Newton's second law : ΣF = mass × acceleration

$$\therefore \quad 90 - 14130\ \mu = \frac{90}{9.81}(-0.6)$$

$$90 - 14130\ \mu = -5.5$$

$$\therefore \quad \mu = \frac{95.5}{14130} = 6.76 \times 10^{-3} \text{ N s/m}^2$$

EXAMPLE 5.8

Find the kinematic viscosity of a liquid in stokes whose specific gravity is 0.95 and dynamic viscosity is 0.012 poise.

Solution : $\mu = 0.012$ poise $= 0.012 \times 0.1 = 1.2 \times 10^{-3}$ N s/m²

Mass density of liquid = specific gravity × mass density of water

$\rho = 0.95 \times 1000 = 950$ kg/m³

$$\therefore \text{ Kinematic viscosity } \nu = \frac{\mu}{\rho} = \frac{1.2 \times 10^{-3}}{950} = 1.263 \times 10^{-6} \text{ m}^2/\text{s}$$

$= 1.263 \times 10^{-2}$ cm²/s $=$ **1.263 × 10⁻² stokes**

EXAMPLE 5.9

A hydraulic lift used for lifting automobiles has 20 cm diameter ram which slides in a 20.016 cm diameter cylinder. The annular space between the cylinder and ram is filled with an oil of kinematic viscosity 3.5 stokes and relative density 0.85. If the travel of 3.2 m long ram has a uniform rate of 15 cm/s, estimate the frictional resistance experienced by the ram.

Solution : Kinematic viscosity $\nu = 3.5$ stokes $= 3.5$ cm²/s $= 3.5 \times 10^{-4}$ m²/s

Mass density $\rho = 0.85 \times 1000 = 850$ kg/m³

$\therefore$ Dynamic viscosity $\mu = \rho\nu$

$= 850 \times (3.5 \times 10^{-4}) = 0.2975$ N s/m²

$$\text{Thickness of oil film} = \frac{(20.016 - 20)}{2} \times 10^{-2} = 0.00008 \text{ m}$$

$$\text{Shear stress } \tau = \mu \frac{du}{dy} = \mu \frac{V}{t} = \frac{0.2975 \times 0.15}{0.00008 \text{ m}} = 557.81 \text{ N/m}^2$$

Frictional resistance = shear stress × area

$= 557.81 \times (\pi \times 0.20 \times 3.2) = 1121$ N $=$ **1.12 kN**

EXAMPLE 5.10

Two square flat plates with each side 60 cm are spaced 12.5 mm apart. The lower plate is stationary and the upper plate requires a force of 100 N to keep it moving with a velocity of 2.5 m/s. The oil film between the

ension effects

$$= (4.72 - 4.29) \times 10^{-3}$$
$$= 0.43 \times 10^{-3} \text{ m} = \mathbf{0.43 \text{ mm}}$$

5.4.4 Newtonian and non-newtonian fluids

Distinction between Newtonian and non-Newtonian fluids can be readily illustrated when the velocity gradient du/dy is plotted against the viscous shear stress τ.

Fluids for which the viscosity is independent of velocity gradient are called Newtonian fluids. For these fluids the plot between shear stress and velocity gradient is a straight line passing through the origin. Slope of the line equals the coefficient of viscosity, $\mu = \tau/(du/dy)$. Fluids represented by curves (*a*) and (*b*) are Newtonian fluids; fluid represented by line (*a*) is more viscous than that represented by line (*b*). Fluids like air, water, kerosene and thin lubricating oils are essentially Newtonian in chapter under normal working conditions.

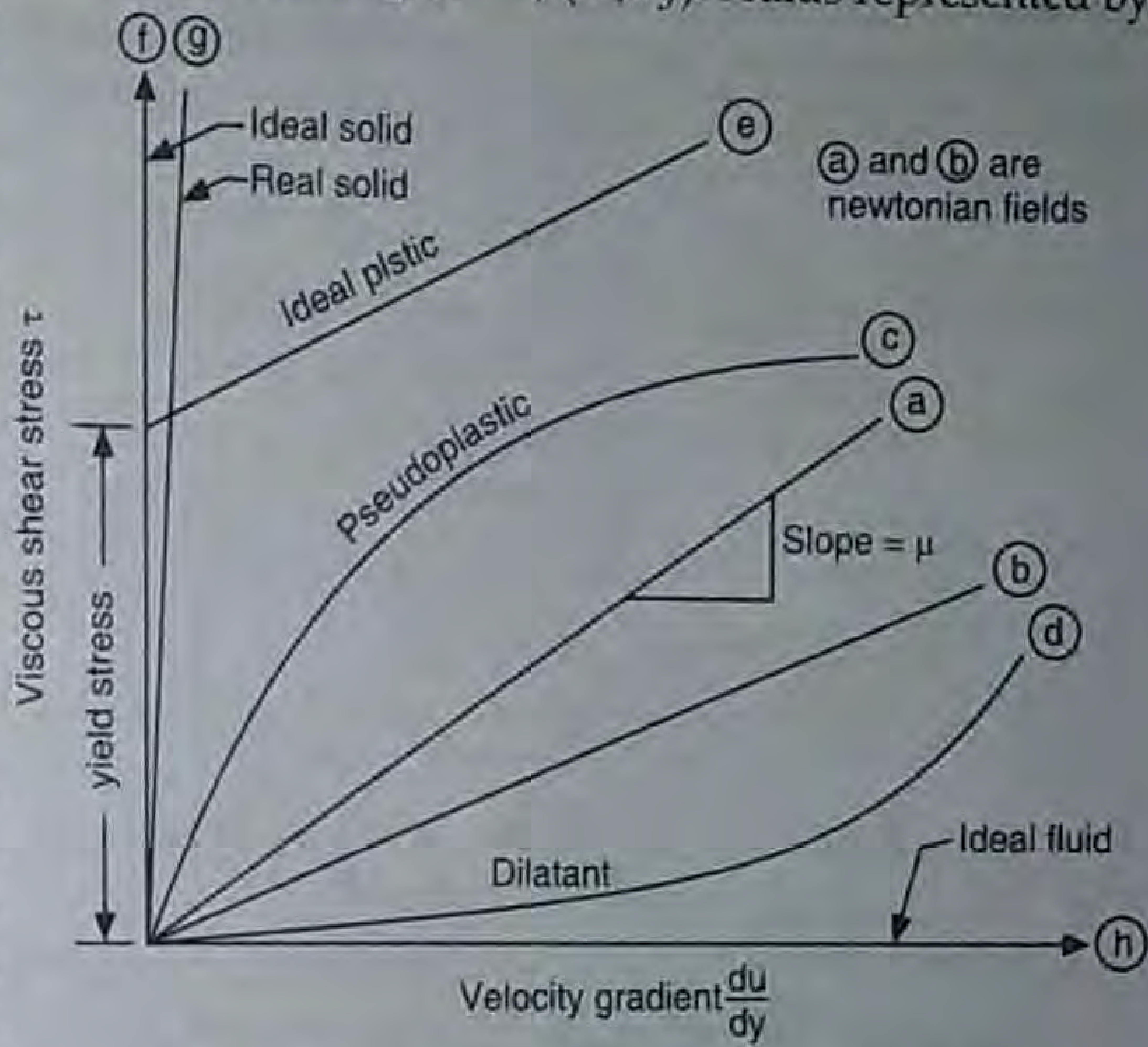


Fig. 5.12. Variation of shear stress with velocity gradient (time rate of deformation)

Fluids such as human blood, thick lubricating oils and certain suspensions for which the viscosity coefficient depends upon velocity gradient are referred to as non-Newtonian fluids. The viscous behaviour of a non-Newtonian fluid may be prescribed by the power law equation $\tau = k\,(du/dy)^n$ where k is a consistency index and n is a flow behaviour index. For a Newtonian fluid, the consistency index k becomes the dynamic viscosity coefficient μ and and the flow behaviour index n assumes a unity value.

Fluids for which the flow behaviour index n is less than unity are called ***pseudo-plastic***. Viscosity coefficient is smaller at greater rates of velocity gradient and the curve becomes flatter as the shear rate (*i.e.*, velocity gradient) increases (curve *c*). Examples of pseudo-plastic fluids are the milk, blood, clay and liquid cement. Fluids for which the index n is greater than unity are called ***dilatant***. Viscosity coefficient is more at greater rates of viscosity and the flow curve steeps with increasing shear rate (curve *d*). Concentrated solution of sugar and aqueous suspension of rice starch are examples of dilatant fluids.

An *ideal plastic* substance indicates no deformation when stressed upto a certain point (yield stress) and beyond that it behaves like a Newtonian fluid and hence is represented by line (e). For certain substances, there is finite deformation for a given load, *i.e.*, rate of deformation is zero. These materials plot as ordinate (curve *f*) and are called ***elastic materials*** or ***ideal solids***. Actual solids deform slightly when subjected to shear stress of larger magnitude and hence plot as a straight line almost vertical (*g*). A fluid for which shear stress is zero (even if there is velocity gradient) is the ***ideal fluid*** and it plots as abscissa (*h*). Fluids which show an apparent increase in viscosity with time are called ***thixotropic***. Conversely if the apparent viscosity decreases with time, the fluid is called *rheopectic*.

5.5 PRESSURE AND ITS RELATIONSHIP WITH HEIGHT

5.5.1. Pressure

A fluid element or mass is essentially acted upon by two categories of forces ; body forces and surface forces. ***Body forces*** on fluids element are caused by agencies such as gravitational, electric or magnetic fields. The magnitude of these forces is proportional to the mass of the fluid. ***Surface forces*** represent the action of the surrounding fluid on the element under consideration through direct contact. These forces are due to surface stresses like pressure (normal force) and shear (tangential force). In fluids at rest, there is no relative motion between the layers of the fluid. The velocity gradient is zero and hence there is no shear in the fluid. Consequently there is no tangential component of force and hence for a stationary fluid, the force exerted is normal to the surface of the containing vessel. This normal surface force is called the pressure force. The mathematical definition of ***intensity of pressure*** (or simply pressure), in the absence of shearing stress, is

$$p = \frac{dF}{dA}$$

where dF represents the resultant force acting normal to an infinitesimal area dA. If the total force F acts uniformly over the entire area A, then $p = F/A$. Pressure has the dimensions of $\left[FL^{-2}\right]$ and is usually expressed in N/m² (pascal), bar or atmosphere.

$$1 \text{ bar} = 10^5 \text{ N/m}^2 = 100 \text{ kPa}$$

$$1 \text{ atm} = 101.3 \text{ kPa}$$

5.5.2 Pascal's law

An important and unique property of hydrostatic pressure is reflected in Pascal's law, which states that :

Intensity of pressure at a point in a fluid at rest is same in all directions

Consider a small wedge shaped element of stationary fluid and assume that the element has a unit depth perpendicular to the plane of the paper (Fig. 5.13). The element is acted upon by the normal pressure forces and the vertical forces due to weight. Let p_x, p_y, and p_θ be the pressure intensities on the faces AB, BC and AC respectively. Then :

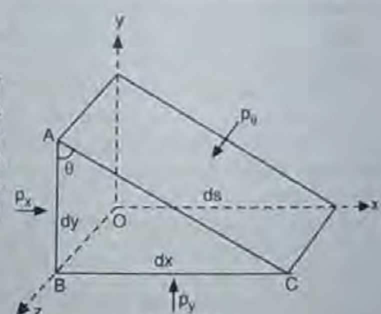


Fig. 5.13

Force on face AB $= p_x \times$ area of face AB

$= p_x (dy \times 1) = p_x\, dy$

Likewise, Force on face BC $= p_y\, dx$

Force on face AC $= p_\theta\, ds$

The weight of fluid element is,

= (area of triangular element × depth) × specific weight

$$= \left(\frac{1}{2} dx\, dy \times 1\right) \times w = \frac{1}{2}\, w\, dx\, dy$$

and it acts through the centre of gravity. Since the fluid element is in equilibrium, the forces in the horizontal and vertical directions must balance.

Resolving the forces in x-direction,

$$p_x\, dy = p_\theta\, ds \sin\theta$$

From Fig. 5.13 : $\quad dy = ds \sin\theta$

$$\therefore \quad p_x\, dy = p_\theta\, dy\, ;\; p_x = p_\theta$$

Resolving the forces in y-direction,

$$p_y\, dx = \frac{1}{2}\, w\, dx\, dy + p_\theta\, ds \cos\theta$$

Let the size of the elemental system approach smaller and smaller dimensions; then the gravitation force (weight) which diminishes as the product of two dimensions (dx and dy) can be neglected in comparison with the pressure forces for which the diminishing effect is proportional to be reduction in single dimension (dx). Thus in the limit

$$p_y\, dx = p_\theta\, ds \cos\theta$$

From Fig. 5.13 : $\quad dx = ds \cos\theta$

$$\therefore \quad p_y\, dx = p_\theta\, dx\, ;\; p_y = p_\theta$$

From equations 5.2 and 5.3, we have

$$p_x = p_y = p_\theta \qquad \ldots(5.9)$$

This result is independent of the angle θ and, therefore, it follows that pressure acts equally in all directions in a stationary fluid. Pressure at a point has only one value regardless of the orientation of the area upon which it is determined. Independence of direction implies that pressure is a scalar quantity.

5.5.3 Hydrostatic law

Rate of increase of pressure in a vertical direction is equal to weight density (specific weight) of the fluid.

The fundamental equation relating pressure, density and vertical distance can be established by considering the equilibrium of an imaginary cylindrical element in a body of fluid at rest. The cylindrical element is of cross-sectional area dA and height dy.

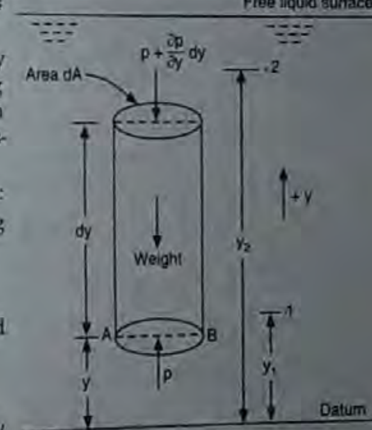


Fig. 5.14 Pressure-density-height relationship

The pressure forces acting on the fluid element are :

(*i*) Pressure force on bottom face AB $= p\, dA$ acting in the upward direction.

(*ii*) Pressure force on top face CD

$= \left(p + \frac{\partial p}{\partial y} dy\right) dA$ acting in the downward direction.

(*iii*) Weight of fluid element = specific weight × volume $= w\, dA\, dy$

(*iv*) Pressure forces on surface AC and BD are equal and opposite and hence cancel out.

∴ Distance through which load is raised on the ram side

= (distance moved by the ram in one stroke) × number of strokes/minute

= 0.833 × 100 = **83.3 cm/minute**

(c) Work done = load × distance moved

$= (20 \times 10^3) \times 0.833 = 16660$ Nm/min

∴ Power required to operate the plunger

$$= \frac{16660}{60} = 277.67 \text{ Nm/s} = \mathbf{277.67\ W}$$

5.6. EQUATIONS OF MOTION

5.6.1. Flow rate and continuity equation

Consider flow of an ideal fluid through a stream tube (Fig. 5.20). Since no flow takes place across the streamlines, the fluid must enter and leave the tube only at the end sections. At the inlet section 1-1, the flow characteristics are : tube cross-sectional area A, average fluid density ρ and the mean flow velocity V. The corresponding parameters at the exit section 2-2 are $(A + dA)$, $(\rho + d\rho)$ and $(V + dV)$.

The mass flow entering the stream tube at section 1-1 during time interval dt is given by $(AV \rho dt)$. During the same time interval a quantity of mass flow $(A + dA)(V + dV)(\rho + d\rho)\, dt$ out flows from the exist section 2-2. The fluid mass that accumulates between the two sections is then given by :

$$dm = AV\rho\, dt - (A + dA)(V + dV)(\rho + d\rho)\, dt$$

Simplification yields,

$$\frac{dm}{dt} = -(AV\, d\rho + V\rho\, dA + A\rho\, dV)$$

For steady flow $\frac{dm}{dt} = 0$ and therefore,

$$AV\, d\rho + V\rho dA + A\rho\, dV = 0$$

Dividing throughout by ρAV, one obtains :

$$\frac{d\rho}{\rho} + \frac{dA}{A} + \frac{dV}{V} = 0$$

or

$$d(\rho AV) = 0$$

$$\rho AV = \text{constant} \qquad ...(5.14)$$

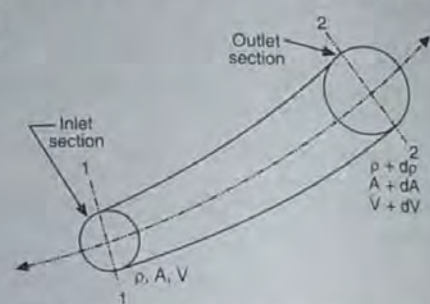


Fig. 5.20. Steady flow through a stream tube

Evidently the mass of fluid per unit time passing through any section of a stream tube is constant.

For an incompressible fluid, the mass density ρ is constant and therefore :

$$AV = \text{constant}$$

i.e.,

$$A_1 V_1 = A_2 V_2 \qquad ...(5.15)$$

Equation 5.15 represents the continuity equation for the steady incompressible flow through an elementary stream tube. The continuity equation states that in a varying duct, the average velocity may change along the direction of flow but the product (area × velocity) remains constant. Further the mean velocities are inversely proportional to the cross-sectional areas of the flow passage, *i.e.,* $\frac{V_1}{V_2} = \frac{A_2}{A_1}$.

Continuity relation $Q = A_1 V_1 = A_2 V_2$ works with the restrictions : constant density, no friction, steady flow, parallel stream tubes at each sections 1 and 2 and no branching of stream tubes.

The product AV is the quantity (volume measure) of fluid which passes a reference point per unit time and is called the *flow rate* or discharge. *Mass flow* rate is the quantity (mass measure) ρAV of fluid which passes per unit time; the units are expressed in kg/s. *Weight flow* rate is the quantity (weight measure) $w AV$ of fluid which passage the reference point per unit time; value is expressed in N/s.

For a two-dimensional flow, the stream tube can be considered to be a unit dimension in the z-direction. Then the area along the stream tube is numerical equal to the spacing t of the streamlines. Continuity equation for a stream tube in a two-dimensional, steady flow or incompressible fluid may then be prescribed as ;

$$V_1 t_1 = V_2 t_2 = \text{constant} \qquad ...(5.16)$$

Apparently the mean velocity is inversely proportional to the spacing of the streamlines.

EXAMPLE 5.30

From a flow net diagram, it was found that the distances between two consecutive streamlines at two successive sections are 1 cm and 0.6 cm respectively. If the velocity at the first section is 1 m/s, find the velocity at the other section. Also find the discharge between the two streamlines.

Solution : For a two-dimensional flow

$$V_1 t_1 = V_2 t_2 = \text{constant}$$

Given : $V_1 = 1$ m/s ; $t_1 = 1$ cm $= 0.01$ m ; $t_2 = 0.6$ cm $= 0.006$ m

$$\therefore \quad \text{Velocity } V_2 = \frac{V_1 t_1}{t_2} = \frac{1 \times 0.01}{0.006} = \mathbf{1.67\ m/s}$$

Further, discharge $= V_1 t_1 = 1 \times 0.01 = \mathbf{0.01\ m^2/s}$ **unit depth**

EXAMPLE 5.31

Water is flowing through a pipe of 0.5 m diameter with an average velocity of 1 m/s. What is the rate of discharge of water ? The same flow then passes through another section where the diameter is 1 m. What is the average flow velocity at this section ?

Solution : Discharge $Q = \text{area} \times \text{velocity} = \frac{\pi}{4}(0.5)^2 \times 1 = \mathbf{0.196\ m^3/s}$

Let V_2 be the velocity at the section where diameter is 1 m. From continuity considerations :

$$Q = A_1 V_1 = A_2 V_2$$

$$V_2 = \frac{A_1}{A_2} V_1 = \frac{\frac{\pi}{4}(0.5)^2}{\frac{\pi}{4}(1)^2} \times 1 = \mathbf{0.25\ m/s}$$

EXAMPLE 5.32

A pipe AB branches into two pipes C and D as shown in Fig. 5.21. The pipe has diameter of 45 cm at A, 30 cm at B, 20 cm at C and 15 cm at D. Determine the discharge at A if the velocity at A is 2 m/s. Also determine the velocities at B and D, if the velocity at C is 4 m/s.

Solution : The quantity of liquid passing through section A is :

$$Q_A = A_A \times V_A = \frac{\pi}{4}(0.45)^2 \times 2 = 0.318 \text{ m}^3/\text{s}$$

From continuity considerations :

$$Q = A_A \times V_A = A_B \times V_B ;$$

compressors) device whilst a turbomachine is always a rotary machine. Further in a positive displacement machine, an interaction between the moving part and the fluid involves a change in the volume and/or displacement of the fluid. When the fluid volume increases, there is a transfer of energy from the fluid to the mechanical system. Conversely when the fluid volume diminishes, energy is transferred to the fluid system.

Compared to the positive displacement machine, a turbomachine unit has the advantages of few balancing problems due to the absence of reciprocating and rubbing parts, exceptionally low consumption of lubricating oil, no conversion loss of power from rectilinear motion into rotary motion, and high reliability.

6.1. HYDRAULIC TURBINES

Hydraulic turbines are required to transform fluid energy into usable mechanical energy as efficiently as possible. Further depending on the site, the available fluid energy may vary in its quantum of potential and kinetic energy. Accordingly a suitable type of turbine needs to be selected to perform the required job.

Depending upon the basic operating principle, hydraulic turbines are categorised into impulse and reaction turbines depending on whether the pressure head available is fully or partially converted into kinetic energy in the nozzle.

- *Impulse turbine* wherein the available hydraulic energy is first converted into kinetic energy by means of an efficient nozzle. The high velocity jet issuing from the nozzle then strikes a series of suitably shaped buckets fixed around the rim of a wheel (Fig. 6.1). The buckets change the direction of jet without changing its pressure. The resulting change in momentum sets buckets and wheel into rotary motion and thus mechanical energy is made available at the turbine shaft. The fluid jet leaves the runner with a reduced energy. An impulse turbine operates under atmospheric pressure : there is no change of static pressure across the turbine runner and the unit is often referred to as a free jet turbine. Important impulse turbines are: Pelton wheel, Turgo-impulse wheel, Girad turbine, Banki turbine and Jonval turbine etc.; Pelton wheel is predominantly used at present.

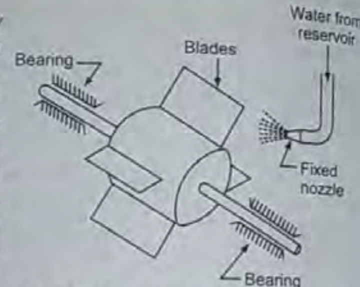


Fig. 6.1. Principle of an impulse turbine

- *Reaction turbine* wherein a part of the total available hydraulic energy is transformed into kinetic energy before the water is taken to the turbine runner. A substantial part remains in the form of pressure energy. Subsequently both the velocity and pressure change simultaneously as water glides along the turbine runner. The flow from inlet to outlet of the turbine is under pressure and, therefore, blades of a reaction turbine are closed passages sealed from atmospheric conditions.

Fig. 6.2 illustrates the working principle of a reaction turbine in which water from the reservoir is taken to the hollow disc through a hollow shaft. The disc has four radial openings, through tubes which are shaped as nozzles. When the water escapes 4through these tubes its pressure energy decreases and there is increase in kinetic energy relative to the rotating disc. The resulting reaction force sets the disc in rotation. The disc and shaft rotate in a direction opposite to the direction of water jet.

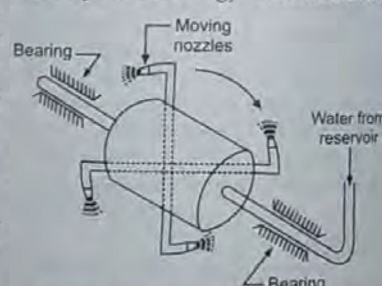


Fig. 6.2. Principle of a reaction turbine

Important reaction turbines are : Fourneyron, Thomson, Francis, Kaplan and Propellor turbines; Francis and Kaplan turbines are widely used at present.

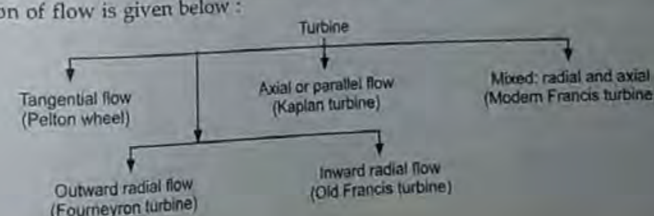
The following table lists salient points of difference between the impulse and reaction turbines with regard to their operation and application.

Table 6.1. Impulse versus Reaction Turbines

Impulse Turbine	Reaction Turbine
• All the available energy of the fluid is converted into kinetic energy by an efficient nozzle that forms a free jet.	Only a portion of the fluid energy is transformed into kinetic energy before the fluid enters the turbine runner.
• The jet is unconfined and at atmospheric pressure throughout the action of water on the runner, and during its subsequent flow to the tail race.	Water enters the runner with an excess pressure, and then both the velocity and pressure change as water passes through the runner.
• Blades are only in action when they are in front of the nozzle.	Blades are in action all the time.
• Water may be allowed to enter a part or whole of the wheel circumference.	Water is admitted over the circumference of the wheel.
• The wheel does not run full and air has free access to the buckets.	Water completely fills the vane passages throughout the operation of the turbine.
• Casing has no hydraulic function to perform; it only serves to prevent splashing and to guide the water to the tail race.	Pressure at inlet to the turbine is much higher than the pressure at outlet; unit has to be sealed from atmospheric conditions and, therefore, casing is absolutely essential.
• Unit is installed above the tail race.	Unit is kept entirely submerged in water below the tail race.
• Flow regulation is possible without loss.	Flow regulation is always accompanied by loss.
• When water glides over the moving blades, its relative velocity either remains constant or reduces slightly due to friction.	Since there is continuous drop in pressure during flow through the blade passages, the relative velocity does increase.

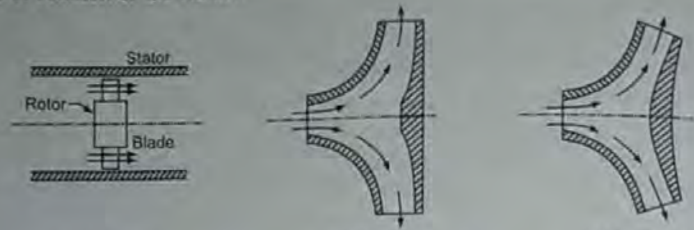
In addition to the concept of impulse and reaction, hydraulic turbines may be further classified into various kinds according to:

(*i*) *Direction of water flow through runner* : Classification of turbines based on consideration of direction of flow is given below :



Flow path in different types of runners has been illustrated in Fig. 6.3.

- Pelton wheel is the tangential flow turbine; here the centreline of jet is tangential to the path of rotation of the runner.



(a) Axial flow (b) Radial flow (c) Mixed flow

Fig. 6.3. Flow path in different types of rotors

- Propellor and Kaplan turbines are axial flow turbines; here water enters and leaves the runner along a direction parallel to the axis of the shaft.
- Radial flow turbines wherein the fluid passes through the runner in a plane practically perpendicular to the axis of rotation; water flows radially through the turbine. Further the flow of water may be radially inward or radially outwards. In a radially inward flow turbine water enters at the outer periphery, glides over the moving blades and then flows radially inwards towards the centre of runner. The old Francis turbine and the Thomson turbine are the inward flow turbines. In a radially outward flow turbine water enters at the inner periphery, glides over the blades and then moves radially outwards/towards the outer periphery of the runner. Fourneyrón turbine is an example of outward radial flow turbine.
- Mixed flow turbines where water enters the runner at the outer periphery in the radial direction and leaves it at the centre in the direction parallel to the axis of rotation of the runner. Modern Francis turbine is a mixed flow machine.

(ii) Available head and discharge:

- High head turbines which operate under high head (above 250 m) and require relatively small rates of flow. Pelton wheel is a high head turbine.
- Medium head turbines which operate under medium heads (60 m to 250 m) and require medium flow rates. Modern Francis turbine belongs to this category.
- Low head turbines which operate under heads upto 30 m and require very large volumetric rates of flow. Units of axial flow turbine (Propeller and Kaplan) are examples of low head turbine.

(iii) Specific speed:

Refers to the speed of a geometrically similar turbine (*i.e.*, a turbine identical in shape, blade angles and gate openings etc.) which would develop unit power when working under a unit head. The turbine specific speed is prescribed by the relation $N_s = N\sqrt{P}/H^{5/4}$ where P is the power in kW, H is the net available head in m and N is the speed is rpm. Specific speed is a characteristic index which serves to identify the types of hydraulic turbine.

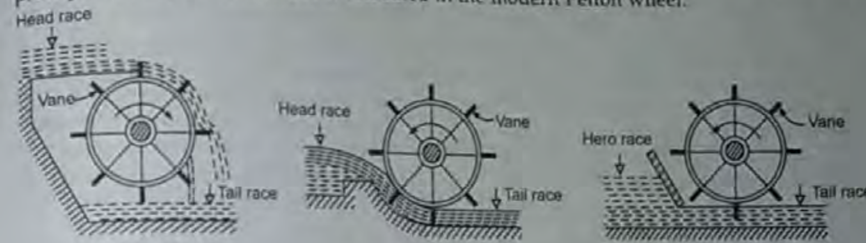
- For Pelton wheel : N_s = 9 – 17 for a slow runner
  = 17 – 25 for a normal runner
  = 25 – 30 for a fast runner = 40 for a double jet

- Francis turbine : N_s = 50 – 100 for a slow runner
  = 100 – 150 for a normal run
  = 150 – 250 for a fast runner
- Kaplan turbine N_s = 250 – 850

(iv) Disposition of shaft : Impulse turbines have usually a horizontal shaft and vertical runner arrangement. Reaction turbines may be either of vertical or horizontal shaft type.

6.2. PELTON TURBINE

The oldest form of water turbine is the water wheel. The natural head (difference in water level) of a stream is utilized to drive it. In its conventional form the water wheel is made of wood and is provided with buckets or vanes round the periphery. The water thrusts against these, causing the wheel to rotate. The latter drives the millstones and sometimes other machinery. In the case of an *overshot wheel* the water pours onto the buckets from above. If the water thrusts against the vanes on the underside of water wheel, it is called an *undershot wheel*. The principle of the old water wheel is embodied in the modern Pelton wheel.



(a) Overshot wheel (b) Breast wheel (c) Undershot wheel

Fig. 6.4. Water wheels

A Pelton wheel is a free-jet impulse turbine named after the American engineer Lesser Pelton (1829-1908) who contributed much to its development. It is simple, robust and the only hydraulic turbine which operates efficiently and is invariably used for heads in excess of 450 m. Smooth running and good performance are other common features of this unit.

Component Parts : Construction and Operation

(i) Penstock : It is a large sized conduit which conveys water from the high level reservoir to the turbine. Depending upon low head or high head installations, a penstock may be made of wood, concrete, or steel. Further the penstock may be of any length depending upon distance between the reservoir and power house. For the regulation of water flow from the reservoir to the turbine, the penstock is provided with control valves. Again screens called trashracks are provided at inlet of the penstock to prevent the debris from entering into it.

Fig. 6.5. Elements of a Pelton wheel

(ii) Spear and nozzle : At its downstream end, the penstock is fitted with an efficient nozzle that converts the whole of hydraulic energy into a high speed jet. To regulate the water flow through the nozzle and to obtain a good jet of water at all loads, a spear or

needle is so arranged that it can move forward or backward thereby decreasing or increasing the annular area of the nozzle flow passage. The movement of the spear is controlled either manually by a hand wheel (in case of very small units) or automatically by a governing mechanism (in case of almost all the bigger units).

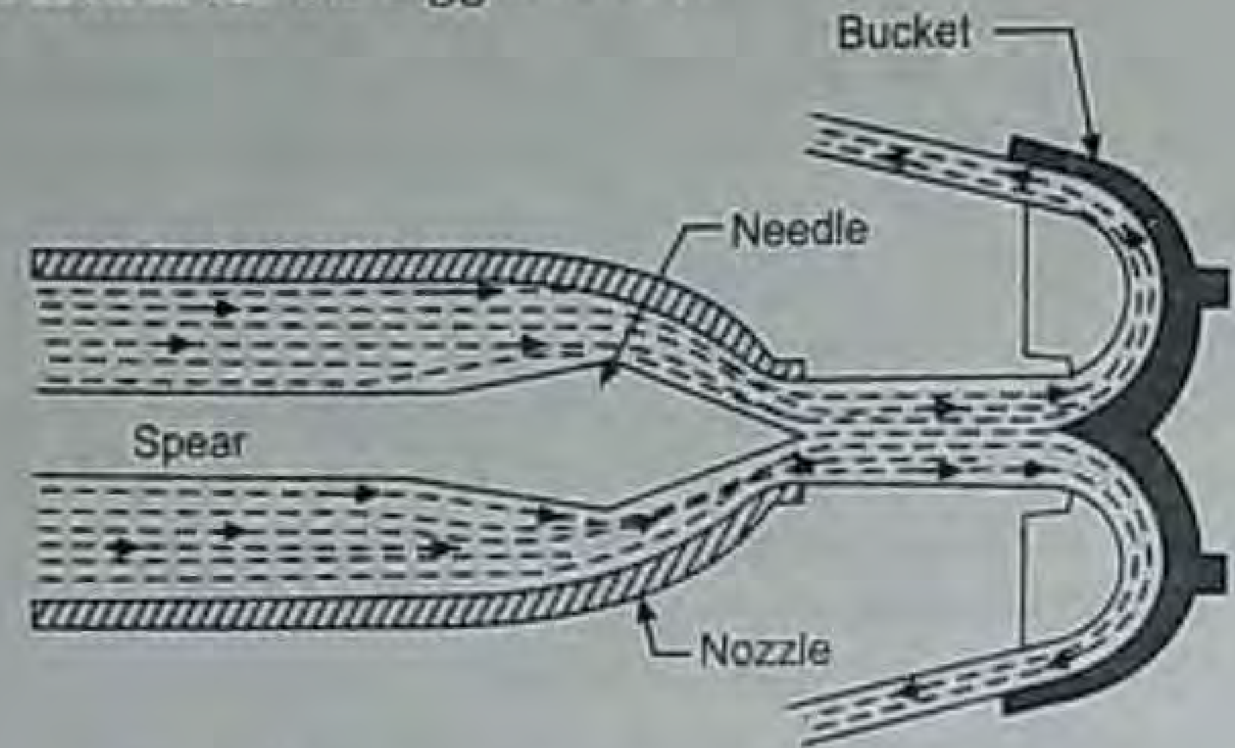


Fig. 6.6. Spear and bucket of a Pelton wheel

(iii) Runner with buckets : The turbine rotor, called the runner, is a circular disk carrying a number (seldom less than 15) of cup-shaped buckets which are arranged equidistantly around the periphery of the disk. The runner is generally mounted on a horizontal shaft supported in small thrust bearings, and the buckets are either cast integrally with the disk or fastened separately. The bolt fastening facilitates easy replacement of buckets when necessary. For low heads the buckets are made of cast iron, but for higher heads they are made of bronze, cast steel, or stainless steel. Further, the inner surface of the buckets is polished to reduce frictional resistance to the water jet.

Each bucket has a ridge or splitter which distributes the striking jet equally into two halves of the hemispherical bucket. Again there is a cut (notch) in the outer rim of each bucket; this notch is provided to make the jet face the bucket only when it has come into proper position with respect to the jet. This position occurs when face of the bucket and axis of the jet are approximately at 90 degree to each other. Maximum driving force will be exerted on the disk when the jet gets deflected through 180-degree, *i.e.*, when the bucket is exactly hemispherical. However, in practice, the angular deflection of jet in the bucket is limited to about 165-170 degree. This is to ensure that the water jet whilst leaving one bucket does not strike the back of the succeeding bucket. This avoids the splashing of water and unnecessary interference which could impair the overall efficiency of the turbine.

Since the two hemispherical cups are joined together and water is directed at the junction, the side thrusts produced by the fluid in each half balance each other. The arrangement has thus the advantage that bearings supporting the wheel shaft are not subjected to any axial or end thrust.

(iv) Casing : Out flow from the runner buckets is in the form of a strong splash which scatters in all directions. To prevent this and to guide the water to the tail race, a casing is provided all around the runner. The casing also acts as a safeguard against accidents. Evidently the casing has no hydraulic function to perform. A baffle is arranged in the casing to prevent the discharged water being carried along the runner direction.

(v) Governing mechanism : Speed to the turbine runner is required to be maintained constant so that the electric generator coupled directly to the turbine shaft runs at constant speed under varying load conditions. The task is accomplished by a governing mechanism that automatically regulates the quantity of water flowing through the runner in accordance with any variations in the load.

6.3. FRANCIS TURBINE

Francis turbine is an inward flow reaction turbine which was designed and developed by the American engineer James B. Francis (1815-92). In the earlier stages of its development, Francis turbine had a purely radial flow runner; the flow passing through the runner had velocity component only in a plane normal to the axis of the runner. The modern Francis turbine is, however, a mixed flow unit in which the water enters the runner radially at its outer periphery and leaves axially at its centre. This arrangement provides a large discharge area with the prescribed diameter of the runner. Francis turbine with its full peripheral admission enjoys a great superiority and is well adopted in the hydroelectric power plants where large quantity of water is available at low and medium heads.

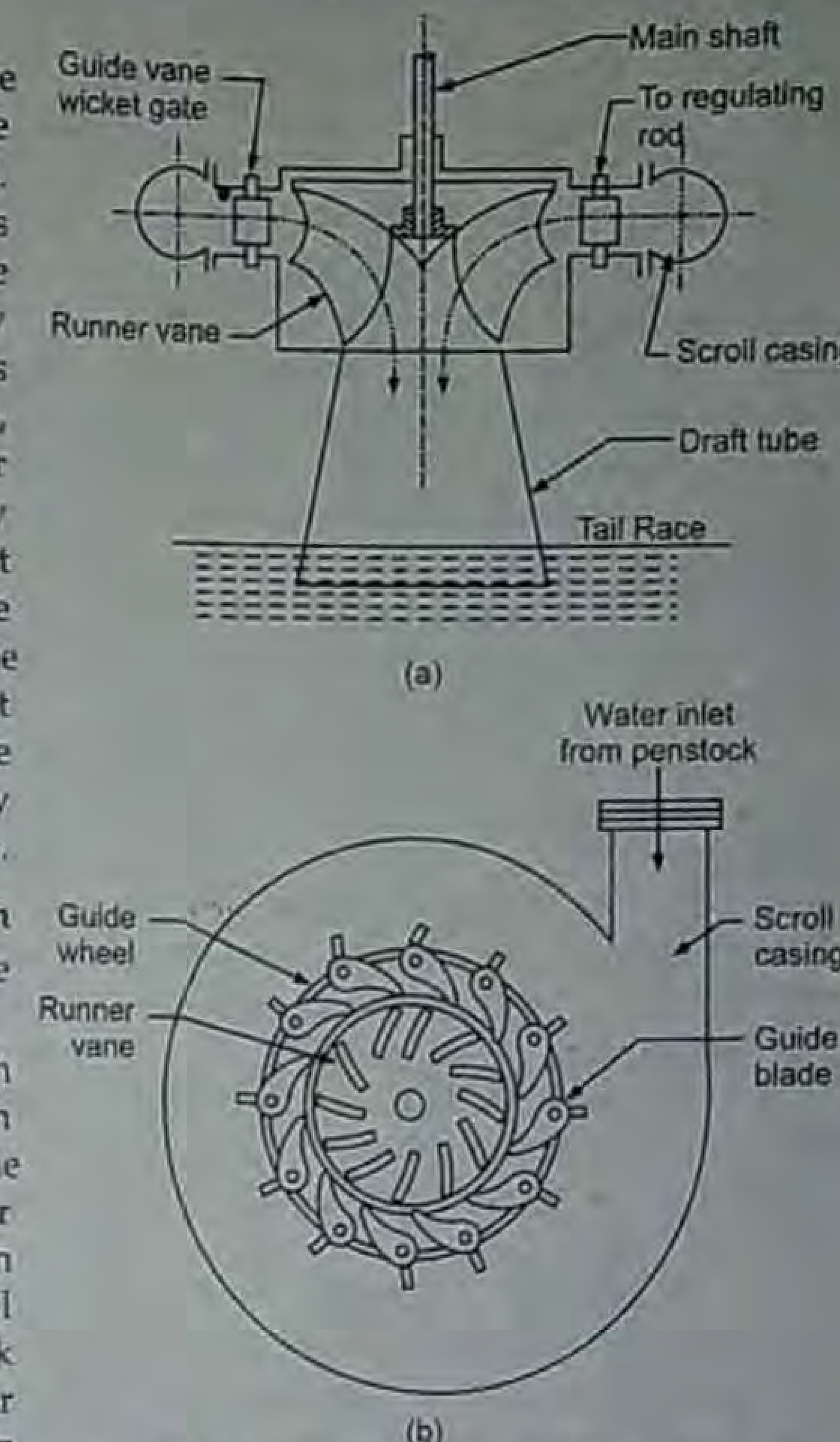


Fig. 6.7. Elements of a Francis turbine

Component parts: Construction and operation

The main features of the Francis turbine are illustrated schematically in Fig. 6.7.

(i) Penstock : It is a large sized conduit which conveys water from the upstream of the dam to the turbine runner. Because of the large volume of water flow, size of the penstock required for a Francis turbine is larger than that of a Pelton wheel. The penstock is invariably made of steel and is embedded inside the dam. Trashrack are provided at inlet of the penstock in order to obstruct the entry of debris and other foreign matter.

(ii) Scroll casing : Penstock is connected to and feeds water directly into an annular channel surrounding the turbine runner. The channel is spiral in its layout and is known as the spiral or scroll casing. Casing constitutes a closed passage whose cross-section area gradually decreases along the flow direction ; area is maximum at inlet and nearly zero at exit. The decrease in area is in proportion to the decreasing volume of water to be handled and that ensures that the velocity of water is constant along its path. After entry into the casing, the water starts distributing itself into the guide blades which are arranged inside the casing. At the turn of 360-degrees, the entire water has passed to the guide blades.

The casing is made of cast steel, plate steel, and concerte depending upon the pressure/head to which the casing is subjected. Further, in the case of bigger units, stay vanes are usually provided inside the casing to support it and to direct the water from the casing to the guide vanes.

(iii) Guide vanes or wicket gates : A series of airfoil shaped vanes, called the guide vanes or wicket gates, are arranged inside the casing to form a number of flow passages between the casing and the runner blades. The guide vanes direct the water onto the runner at an angle appropriate to the design. They direct the flow just as the nozzle of the Pelton wheel. The configuration and arrangement of the guide vanes is such that the energy of water is not consumed by eddies and other undesirable flow phenomenon causing energy losses.

Guide vanes are fixed in position, *i.e.*, they do not rotate with the rotating runner. However, they can swing around their own axes and that helps to bring about a change in the flow area between two consecutive runner blades. This provides a degree of adaptability to the quantity of water to be admitted to the runner in the wake of load variations. Motion is given to the guide vanes either by means of a hand wheel or automatically by a governor.

(iv) Guide wheel and governing mechanism : The governing mechanism changes the position of guide blades to affect a variation in the water flow rate in the wake of changing load conditions on the turbine. The system consists of a centrifugal governing mechanism, linkages, servomotor with its oil pressure governor and the guide wheel. When the load changes, the governing mechanism rotates all the guide blades about their axes through the same angle so that the water flow rate to the runner and its direction essentially remain the same at all the passages between any two consecutive guide vanes. The penstock pipe feeding the turbine is often fitted with a relief valve, also known as the pressure regulator. When the guide vanes are suddenly closed, the relief valve opens and diverts the water directly to tail race. The simultaneous operation of guide vanes and relief valve is termed as double regulation.

(v) Runner and runner blades : Runner of the Francis turbine is a rotor which has passages formed between crown and shroud in one direction and two consecutive blades on the other. These passages take water in at the outer periphery in the radially inward direction and discharge it in a direction parallel to the axis of rotor. The driving force on the runner is both due to impulse (deviation in the direction of flow) and reaction (change in pressure and velocity energy) effects.

The number of runner blades usually varies between 16 to 24. With small units the runner is made of cast iron whilst the bigger units have runner essentially made of stainless steel or a non-ferrous metal like bronze when the water is chemically impure and there is danger of corrosion. The runner is keyed to the shaft which may be of vertical or horizontal disposition; mostly vertical.

(vi) Draft tube : After passing through the runner, the water is discharged to the tail race through a gradually expanding tube called the draft tube. The free end of the draft tube is submerged deep into the tail race. Evidently then the entire water passage from the head race to the tail race is totally closed; does not communicate with the surrounding atmospheric pressure.

Because of its gradually increasing cross-section, the discharge velocity from the turbine runner is not all wasted; it is partly converted into a useful pressure head and the water discharges at a relatively low velocity to the tail water.

6.4. PROPELLER AND KAPLAN TURBINES

The propeller turbine is a reaction turbine which is particularly suited for low head (upto 30m) and high flow rate installations *i.e.*, at barrages in rivers. The unit is like the propeller of a ship operating in reverse. The ship propeller rotates, thrusts the water away behind it and thus causes the ship to move forward. In a propeller turbine, the water flows through the propeller and sets it in motion. Water enter the turbine laterally, gets deflected by the guide vanes and then flows through the propeller. For this reason, these machines are referred to as axial flow units.

Component Parts : Construction and Operation

The main features of a propeller turbine are illustrated schematically in Fig. 6.8 and 6.9. Except the runner, all other parts such as the scroll casing, stay ring, guide mechanism (arrangement of guide vanes) and the draft tube of a propeller turbine are similar to those of a Francis turbine. Between the guide vanes and the runner, the water turns through right angle and

subsequently flows parallel to the shaft. This purely axial flow arrangement provides the largest flow area; even at larger flow rates the flow velocities are not too large.

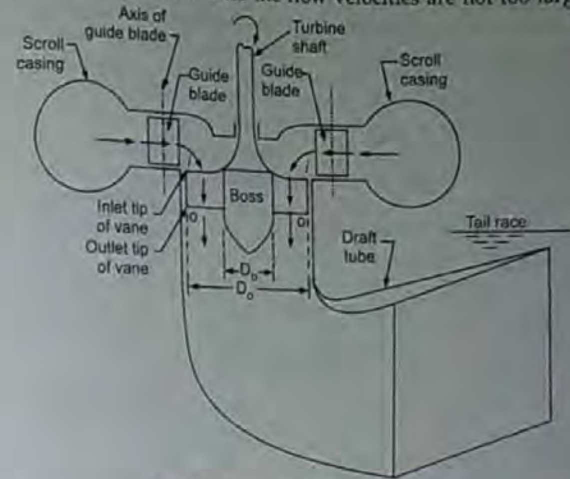


Fig. 6.8. Elements of a Kaplan turbine

The runner is in the form of a boss which is nothing but extension of bottom end of the shaft into a bigger diameter. On the periphery of the boss are mounted equidistantly 3 to 6 vanes made of stainless steel. Thus compared to the Francis turbine which has 16 to 24 number of blades, a propeller turbine with only 3 to 6 vanes will have less contact surface with water and as such a low value of frictional resistance. Further more, the runner blades are directly attached to the hub and this feature eliminates the frictional losses which are caused by the bend provided in a Francis turbine.

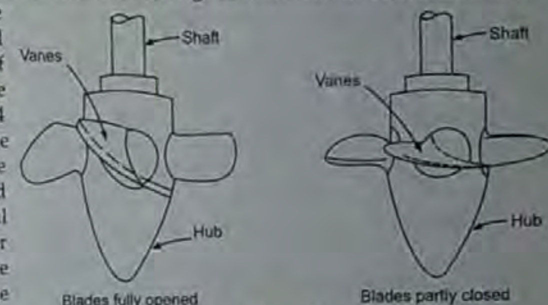


Fig. 6.9. Kaplan turbine runner

The fixed blade propeller turbine is installed only at the sites where the head and load are constant. At part load, the power efficiency curve of such a unit is very much peaked, *i.e.*, a poor performance is indicated. This problem of poor efficiency at part load was successfully solved by the Australian engineer Victor Kaplan who introduced the concept of adjusting the runner vanes in the face of changing load conditions on the turbine. Hence the name *variable pitch propeller turbine* is often given to the Kaplan turbine. With proper adjustment of blades during its running, the Kaplan turbine is capable of giving a high efficiency for a wide range of load conditions. The pitch of the runner blades is automatically adjusted by the governor through the action of a servomotor.

The Kaplan turbine has double regulation which comprises the movement of guide vanes and rotation of runners blades (Fig. 6.10). The mechanism employs two servomotors; one controls the guide vanes and the second operates on the runner vanes. The governing is done by the governors (servomotors) from the inside of the hollow shaft of the turbine runner and the movement of piston is employed to twist the blades through suitable linkages. The double regulation ensures a balanced and most satisfactory relationship between the relative positions of the guide and working vanes. Both the servomotors are synchronised ; they are actuated simultaneously and a high efficiency is maintained at all loads.

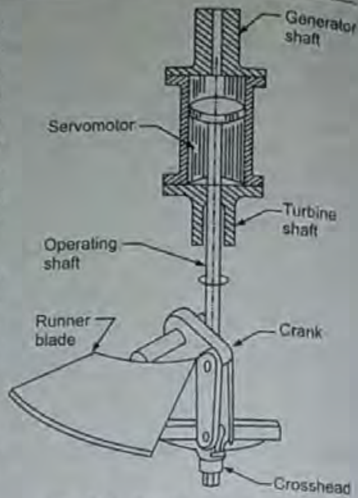


Fig. 6.10. Kaplan runner blade mechanism

Kaplan turbines are capable of taking overloads from 15 to 20 percent and give a very high efficiency at all the gate openings while working at full load and part load conditions (speed and the head remain constant). The velocity plots do change with the flow rate variations caused by changes in load on the turbine. However, the blade angles also get simultaneously adjusted and as such under all working conditions, water enters and flows through the runner blades without shock. As such the eddy losses, which are inevitable in the Francis and the fixed-blade propeller turbine, get entirely eliminated in the Kaplan turbine.

Quite often the electric generator coupled to the Kaplan turbine is enclosed and works inside a straight passage having the shape of a bulb. The water tight bulb is submerged directly into the stream of water, and the bends at inlet to casing, draft tube etc., which are responsible for the loss of head are dispensed with. The unit then needs less installation space with a consequent reduction in excavation and other civil engineering works. These turbines are referred to as *bulb or tubular* turbines and the power stations using such turbines are called *under water power stations.* The tubular turbines have become very popular and are invariable employed for very low head (as low as 4 m) installations such as those in tidal power plants and in rivers at very modest falls.

The salient points of difference between the Francis and Kaplan turbines are enumerated below :

Francis turbine	Kaplan turbine
1. Radially inward or mixed flow turbine.	Purely axial flow turbine.
2. Horizontal or vertical disposition of shaft.	Only vertical shaft disposition.
3. Runner vanes are not adjustable.	Runner vanes are adjustable.
4. Large number of vanes; 16 to 24 blades.	Small number of vanes; 3 to 8 blades.
5. Large resistance needs to be overcome owing to more vanes and greater area of contact with water.	Less resistance as there are fewer vanes and less wetted area.
6. Medium head turbine (60 m to 250 m) and works under medium flow rate.	Low head turbine (upto 30 m) and requires very large volumetric flow rates.
7. Specific speed ranges from 50-250.	Specific duty ranges from 250-850.
8. Ordinary governor is sufficient for speed control as the servomotor is of larger size.	Heavy duty governor is essential for speed control due to smaller sizes of the servomotors.

6.5. HYDRAULIC PUMPS

Man's economic and technical progress through the ages might be measured in terms of development from the primitive pumping devices operated either by man or animal to the positive displacement and dynamic pumps he fashions today. Typical progress in the development of pumps is found in municipal water works, power plants, agriculture, transport and many other utility services and industries.

A *pump* has been defined differently by different investigators ; the different definitions are, however, all similar and nearly equivalent.

- a device which raises or transfers liquids at the expense of power input
- a machine designed to elevate, deliver and move various liquids
- a unit that transfers the mechanical energy of a motor or an engine into potential and kinetic energy of a liquid.

By their action, the pumps require that energy must be expended and as such they belong to the category of power absorbing machines. Further, since the temperature gradients are minimal, pumps are the non-thermal machines. The expended energy enables the pump to overcome the hydraulic resistance and make the fluid rise through a geodetic elevation.

6.6. PUMP CLASSIFICATION AND SELECTION CRITERION

According to design and principal of operation, pumps may be placed in one of the two general categories :

(*a*) dynamic pumps, and

(*b*) positive displacement pumps

These two categories are further subdivided as depicted below :

Pumps
- Dynamic pressure pumps
  - Centrifugal
  - Turbine
  - Propeller
  - Jet
- Positive pressure pumps
  - Reciprocating
    - Piston plunger
    - Diaphragm
  - Rotary
    - Gear
    - Lobe
    - Vane
    - Screw
    - Rotary plunger

Essential data for the selection of a pump includes :

- pressure and capacity of the liquid being handled
- properties such as viscosity, temperature, corrosiveness and grittiness etc., of the flowing liquid
- initial and maintenance cost
- pump duty, *i.e.,* whether the pump is to transfer the liquid or to meter it also
- availability of space, size and position of locating the pump
- speed of rotation and power required
- standardisation with respect to the types and makes of pumps already available at the site
- scale up problems.

Each pump has its own operating characteristics that limit its practical applications. For example in a centrifugal pump, a small change in pressure differential causes a relatively large change in flow. A positive displacement pump, on the other hand, delivers an almost constant quantity regardless of pressure fluctuations. Thus if finite pressure differences are known to exit in a particular application, then the demand of a constant supply of liquid would be met by installing a positive displacement type of pump. Likewise a centrifugal pump would be the obvious choice if in a particular pump application it is necessary to maintain a constant head/pressure on the mains despite fluctuations in capacity/discharge.

Centrifugal pumps have high output and high efficiency. Their simple design and convenient operation has resulted in their wide spread use. In general, it is always advantageous to go for a centrifugal pump unless :

(i) viscosity of the liquid is greater than 1000 centipoise

(ii) low capacity and high heads are in demand

(iii) percentage volume of dissolved gases is greater than 5%

Reciprocating pumps are best in the field of high pressure and moderate capacity pumping. Rotary positive displacement pumps are employed in oil conduits, hydraulic devices etc.

6.7. PUMP APPLICATIONS

A pump adds to the pressure, existing on a liquid, and increment sufficient to do the required service. This service may be increasing the pressure, imparting kinetic energy, lifting and circulating, exhausting or extracting liquids etc. Some notable applications of pump installation are in the fields of :

- agriculture and irrigation works
- municipal water works and drainage system (sewrage disposal)
- fire protection systems
- condensing water, condensate, boiler feed, sump drain and such other services in a steam power plant
- hydraulic control systems
- circulation of water in compressor and diesel engine cooling systems
- oil pumping
- transfer of raw materials, materials in manufacture and the finished products in industry.

6.8. CENTRIFUGAL PUMPS

Centrifugal pumps belong to the category of dynamic pressure pumps wherein the pumping of liquids or generation of head is affected by rotary motion of one or more rotating wheels called the impellers.

A centrifugal pump consists essentially of the following elements:

(i) *Rotating element* consisting of shaft and a vaned rotor called impeller. The vanes are curved, cylindrical or have more complex surfaces. The unit has a finite number of vanes; the number is selected to assure motion of the liquid in the desired direction and varies with diameter of the impeller eye and the radial depth of the vanes. The number usually ranges between six and twelve.

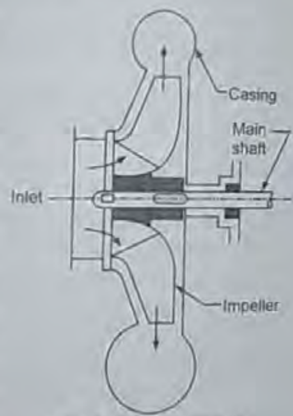


Fig. 6.11. Elements of centrifugal pumps

The impeller is mounted on a shaft coupled to the driving unit which may be an internal combustion engine or an electric motor. By virtue of force interaction between the vanes and the liquid, the mechanical energy of the driver is transformed into the energy of flow.

(ii) *Stationary element* consisting of casing, stuffing box and bearings. The casing is an airtight chamber surrounding the pump impeller; it collects liquid from the impeller and leads it away under high pressure to the delivery side. Packings, labyrinths and glands are needed to reduce the shaft leakage, both internal and external.

(iii) *Suction pipe, strainer and foot valve* : Suction pipe connects the centre (eye) of the impeller to the sump from which the liquid is to be lifted. The pipe is laid airtight so that there is no possibility of formation of air pockets.

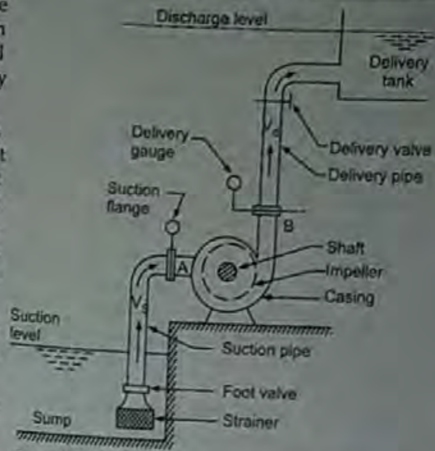


Fig. 6.12. Typical installation of a centrifugal pump

Suction pipe is provided with a strainer at its lower end so as to prevent the entry of solid particles, debries etc. into the pump. These foreign materials, if carried into the pump would adversely affect its performance. The foot valve is a one-way valve located above the strainer into the suction pipe. It serves to fill the pump with liquid before it is started, and prevents back flow when the pump is stopped.

(iv) *Delivery pipe and delivery valve* : Delivery pipe leads the liquid from the pump outlet to the point of use. A regulating valve provided just near the pump outlet serves to control the flow of liquid into the delivery pipe.

Working

The pump is initially primed wherein the suction pipe, casing and portion of the delivery pipe upto the delivery valve are completely filled with the liquid to be pumped. Rapid motion imparted to impeller then builds up centrifugal force which throws the liquid towards the impeller periphery. This causes pressure gradient in the suction pipe, *i.e.* a partial vacuum exists at the impeller eye while the liquid in the sump is at atmospheric pressure. Consequently liquid from the sump is sucked in towards the impeller eye. When the liquid passes through the impeller, it receives energy and that results in the growth of both pressure and velocity. The casing collects the liquid from the impeller and guides it to the delivery pipe. Since the casing increases in cross-sectional area towards the delivery, kinetic head represented by the high discharge velocity is partially transferred into pressure head before the liquid leaves the pump. The process is continuous as long as motion is given to the impeller and there is supply of liquid to draw upon.

6.9. CLASSIFICATION OF CENTRIFUGAL PUMPS

Based on their utility, design and constructional features, centrifugal pumps can be classified with respect to the following characteristics :

Shape and Type of Casing

Liquid leaving the impeller has an appreciable high velocity. This necessitates some arrangement to bring about the desired conversion of kinetic energy to pressure energy before the liquid reaches the discharge end of the pump. This conversion has to be accomplished with minimum energy loss. The task is accomplished by providing a casing around the pump impeller.

In general, there are three casing arrangements, and the pump is named after the casing arrangement it uses.

- *Volute* or *spiral casing* wherein cross-section of the moving stream gradually increases from the tongue towards the discharge pipe. This increase in area results in a gradual decrease in velocity with corresponding increase in the pressure. Most of the single stage pumps are built with volute casings. However, the volute casing has greater eddy losses and hence lower overall efficiency.

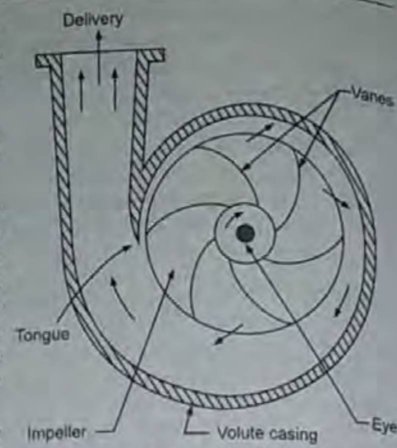


Fig. 6.13. Pump with volute casing

- *Vortex or whirlpool chamber* wherein an annular space is provided between the volute and impeller. This arrangement arrests the formation of eddies and gives an improved performance.
- *Volute casing with guide blades* wherein fixed guide blades are provided around the impeller periphery. When the liquid flows through diverging passages formed between the guide vanes, conversion of dynamic into static head occurs. Liquid leaving the vanes is then collected in a volute chamber where further diffusion occurs before the liquid is discharged to the delivery pipe. Pumps fitted with guide vanes are called "diffuser pumps" or "turbine pumps" as distinct from "volute pumps". Diffuser type casing is adopted when the pump impellers are to be connected in series, *i.e.*, in multistage deep well pumps. Machines with diffuser blades have rather maximum efficiency, but are less satisfactory when a wide range of operating conditions is required. This may be attributed to the losses caused by the change of blade incidence angle with flow rate.

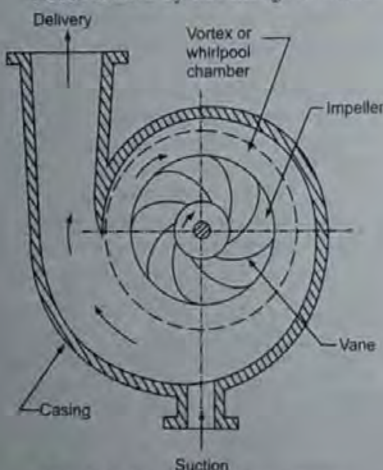


Fig. 6.14. Pump with vortex chamber

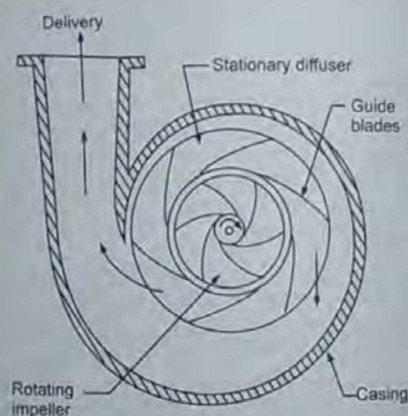


Fig. 6.15. Pump with guide vanes

With respect to mechanical construction of casing, we have :

(*i*) Integral casing pumps: pumps equipped with a casing made in a single piece,
(*ii*) Horizontally split casing pumps : pumps equipped with a casing split on the horizontal centre line,
(*iii*) Vertically split casing pumps : pumps equipped with a casing split on the vertical centre line,
(*iv*) Diagonally split casing pumps ; pumps equipped with a casing split diagonally,
(*v*) Segmented casing pumps ; pumps equipped with a casing made up of segments. These may either be of the band type for multi purpose pumps or of the bowl type for turbine pumps.

Closed, Semi-closed and Open Impellers

In the closed or shrouded impellers the vanes are covered with shrouds (sideplates) on both sides. The back shroud is mounted into the shaft and the front shroud is coupled to the former by the vanes. The arrangement provides a smooth passage for the liquid ; wear is reduced to minimum. This ensures full capacity operation with high efficiency for a prolonged running period. This type is however meant to pump only clear liquids of low viscosity ; liquids may be ordinary water, hot water and acids.

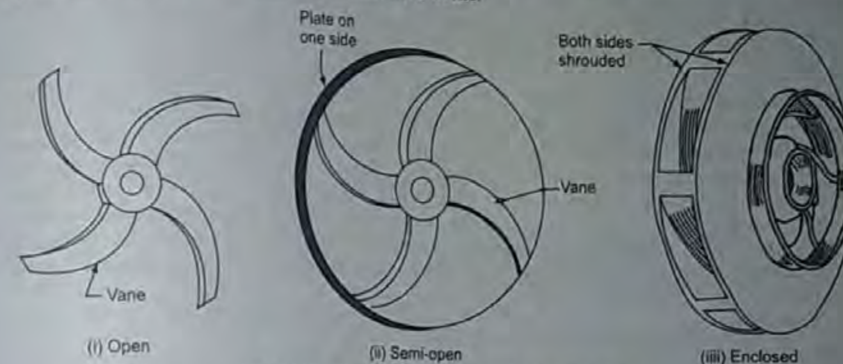


Fig. 6.16. Closed, semi-closed and open impellers

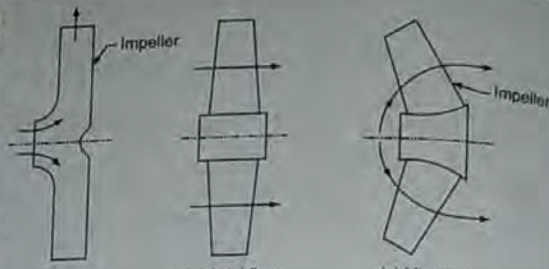
The semi-open impeller has a plate (shroud) only on the back side. The design is adapted to industrial pump problems which require a rugged pump to handle liquids containing fibrous material such as paper pulp, sugar molasses, and sewage water etc.

In an open impeller, no shroud or plate is provided on either side. That is the vanes are open on both sides. Such pumps are used where the pump has a very rough duty to perform *i.e.*, to handle abrasive liquids such as a mixture of water sand, pebbles and clay. Presence of these foreign materials is liable to clog between the impeller and stationary side plates of a closed or semi-closed type impeller.

Axial, Radial and Mixed Flow Impellers

In the axial flow pumps, the head is developed by the propelling or lift action of the vanes on the liquid which enters the impeller axially and discharges axially. The action is similar to the generation of lift by the wings of aeroplane. Axial flow pumps have a very large discharge and are best suited for irrigation purposes.

In radial flow impellers, the head is developed by the action of centrifugal force upon the liquid which enters the impeller axially at the centre and flows radially to the periphery. Flow through a mixed flow impeller is a combination of axial and radial flows. The head is developed partly by the action of centrifugal force and partly by axial propulsion as a result of which the fluid entering the impeller axially at the centre is discharged in an angular direction. Mixed flow impellers resemble the shape of screw and are sometimes called screw impellers. Centrifugal pumps with mixed flow impellers are best suited for irrigation purposes where large quantities of water at low head are required.



(a) Radial (b) Axial flow (c) Mixed flow

Fig. 6.17. Axial, radial and mixed flow impellers

Shape and Number of Vanes

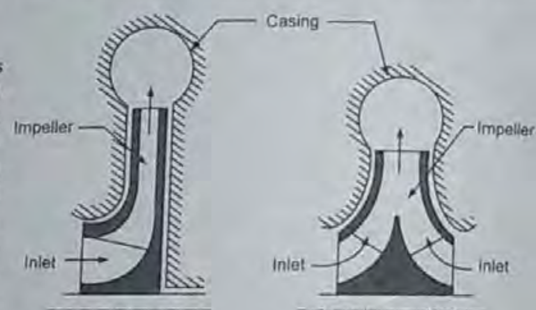
Impeller of a centrifugal pump has a finite number of vanes ; usually from 6 to 12. These vanes may he curved, cylindrical or of more complex surfaces.

Working Head and Number of Stages

Based upon the range of working head, centrifugal pumps are called low head (upto to 15 m), medium head (15 m to 40 m) and high head (over 40 m) pumps. Maximum head built up by a single a stage centrifugal pump seldom exceeds 40 m of water. Greater heads are achieved by having pumps with several stages; the number of stages is indicated by the number of impellers in series. In a multistage pump, the liquid discharging from one impeller and its volute enters the eye of the succeeding impeller, and so forth, there by increasing the head. The total head added by a multistage pump equals the sum of the heads built-up by each impeller.

Single Suction and Double Suction

With respect to how the liquid enters the impeller, the pumps may be with one-sided suction (admission) or with two-sided suction. The one-side feed arrangement has the liquid entering through one side of the impeller. In two-sided feed the liquid enters on both sides thereby increasing the discharge of the pump. Further this arrangement eliminates the axial thrust. A single sided impeller would, however, experience an axial thrust towards the inlet end.



(a) Single suction (b) Double suction

Fig. 6.18. Single suction and double suction pumps

Specific Speed

Specific speed is a term used for classifying pumps on the basis of their performance and dimensional proportions regardless of their actual size or the speed at which they operate. It is defined as the speed of an imaginary pump geometrically similar in every respect to the actual

pump and capable of delivering unit quantity against a unit head. Mathematically, the specific speed N_s for a pump is given by,

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

where N = pump speed in rev/min; Q = discharge in m^3/s of a single suction impeller, and H = head per stage in meters.

Representative values of specific speed for different type of pump impellers are tabulated below :

Pump	Speed	Specific speed
Radial flow	Slow Medium High	10-30 30 -35 50-80
Mixed flow		80-160
Axial flow		100-450

Shaft Position

Most of the centrifugal pumps are of horizontal shaft disposition. However, to affect economy in space, the pumps may be designed with vertical shafts as is done for deep well and mine pumps.

Classification of pumps can also be made on the basis of :

- type of the liquid to be handled such as water, solids in suspension and viscous liquids
- application such as irrigation, boiler feed and condensate circulation
- power used such as I.C. engine or electric motor

6.10. HYDRAULIC SYSTEMS

There exist numerous hydraulic systems and devices in which force and energy are transmitted through an incompressible fluid, generally an oil. Notable examples are the hydraulic accumulators, intensifiers, lifts and cranes, fluid coupling and torque converters.

Figure 6.19 illustrates one typical layout of a hydraulically operated machinery (press, crane or lift) incorporating a pump (power source to provide hydraulic energy), an accumulator and an intensifier unit. The operation of these devices is essentially based on the principles of hydrostatics and hydro-kinetics.

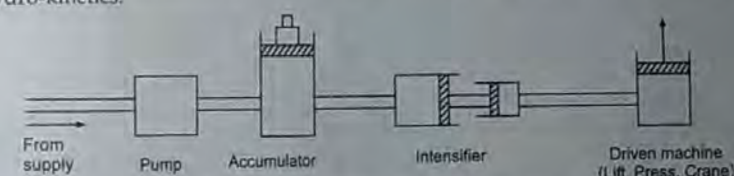


Fig. 6.19. Typical layout of a hydraulic device

6.10.1. Hydraulic Accumulator

Function : To store the energy of fluid under pressure and make this energy readily available as a quick secondary source of power to fluid machines such as presses, lifts and cranes. This function is analogous to that of an electric storage battery, and the flywheel of a reciprocating engine. An

6.10.3. Hydraulic Lift

Function : To lift or bring down load and passengers from one floor to another in a multi-storeyed building.

Construction and operation : A hydraulic lift consists essentially of a ram and cylinder arrangement with a cage or platform fitted to the top end of ram. When fluid under pressure is forced into the cylinder, the ram gets a push vertically upwards. The platform carries loads or passengers and moves between the guides. At requisite height, it can be made to stay in level with each floor so that the goods/passengers can be transferred. In these direct acting lifts, stroke of the ram is equal to the lift of the cage.

In the suspended hydraulic lifts (Fig. 6.24) motion of the platform or cage is obtained by the cylinder and ram arrangement of a hydraulic jigger. Modern lifts are generally of suspended type and these have lifting speeds of 150 m/min or even more.

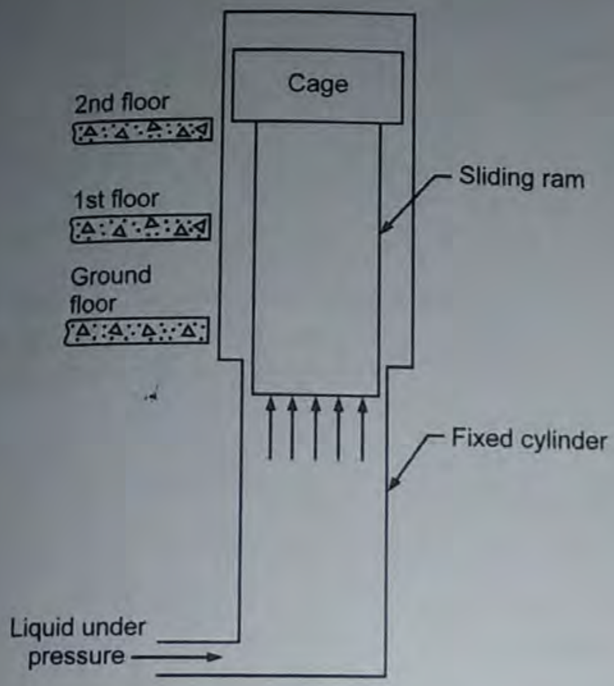


Fig. 6.23. Direct acting hydraulic lift

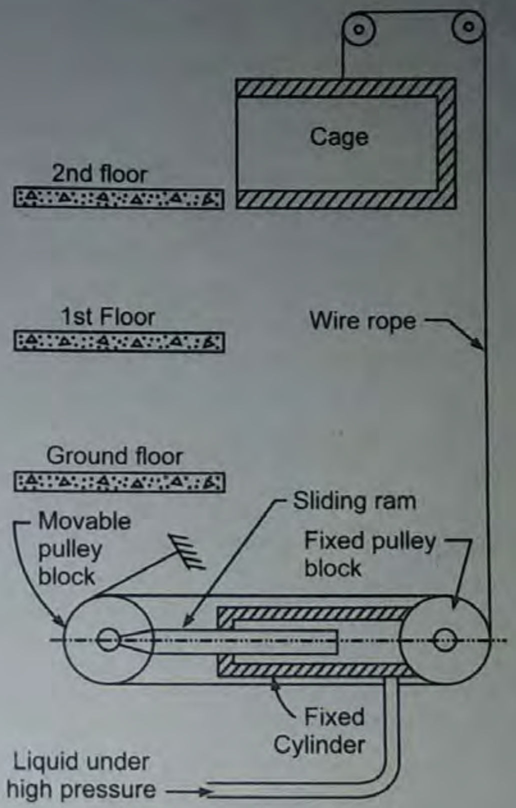


Fig. 6.24. Suspended hydraulic lift

Hydraulic lifts have, in general, been superseded by the electric lifts. Hydraulic lifts then find applications as stand by units to electric lifts or in places where there is danger due to fire or explosion.

REVIEW QUESTIONS

A. Conceptual and conventional questions

1. How hydraulic turbines are classified ?
2. Describe, with sketch, the construction and working of
   (*a*) Pelton wheel ; (*b*) Francis turbine ; (*c*) Kaplan turbine
3. Sketch the Pelton turbine/Francis turbine/Kaplon turbine. Name the various components and state their function.
4. Distinguish between :
   (*a*) Impulse and reaction turbine
   (*b*) Kaplan and propeller turbine
   (*c*) Inward and outward flow reaction turbine.
5. Draw the schematic arrangement of a centrifugal pump installation and state the function of different components.
6. Explain with a neat sketch the construction, operation and utility of the following hydraulic devices :
   (*a*) simple and differential accumulator
   (*b*) hydraulic intensifier
   (*c*) hydraulic lift

▶ 20.1 INTRODUCTION

In the last chapter, we have defined the pumps as the hydraulic machines which convert the mechanical energy into hydraulic energy which is mainly in the form of pressure energy. If the mechanical energy is converted into hydraulic energy, by means of centrifugal force acting on the liquid, the pump is known as centrifugal pump. But if the mechanical energy is converted into hydraulic energy (or pressure energy) by sucking the liquid into a cylinder in which a piston is reciprocating (moving backwards and forwards), which exerts the thrust on the liquid and increases its hydraulic energy (pressure energy), the pump is known as reciprocating pump.

▶ 20.2 MAIN PARTS OF A RECIPROCATING PUMP

The following are the main parts of a reciprocating pump as shown in Fig. 20.1.

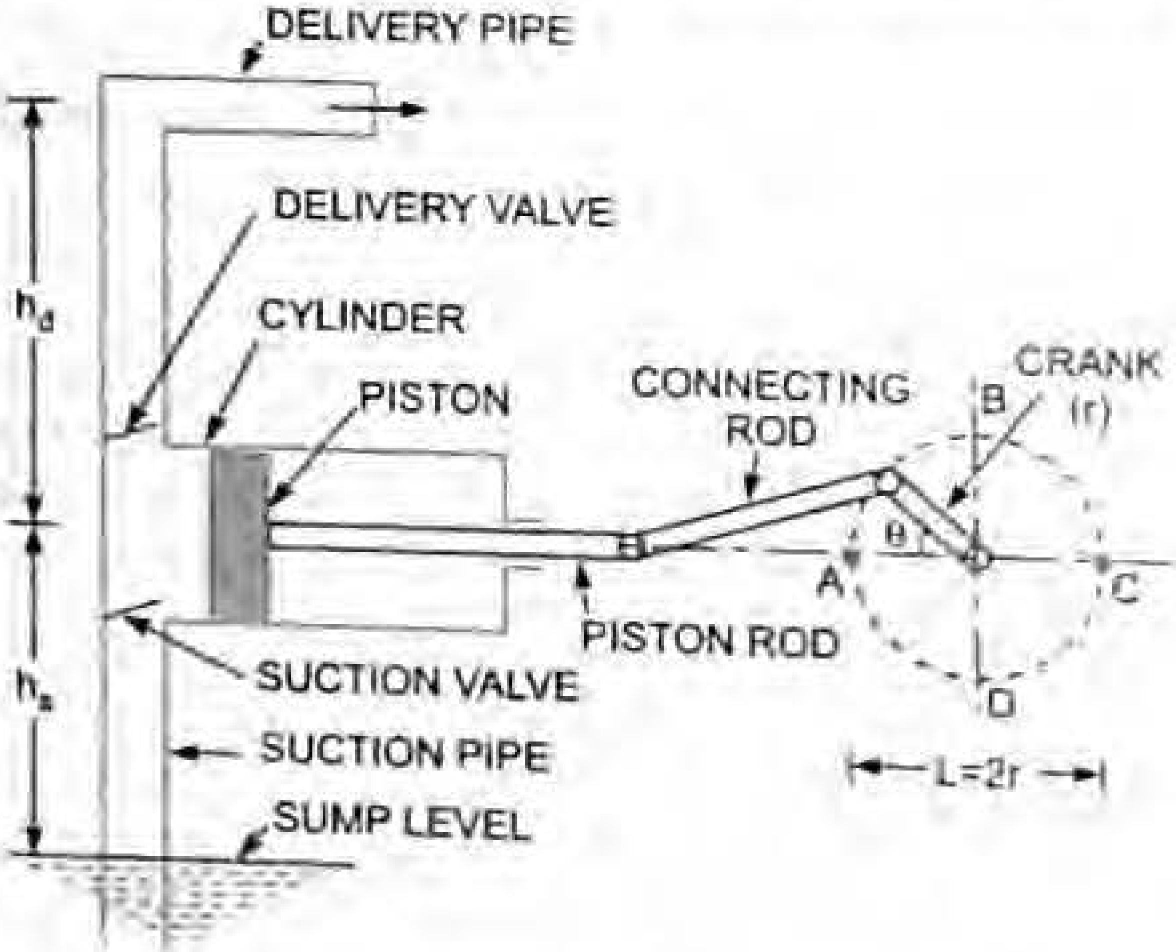


Fig. 20.1 *Main parts of a reciprocating pump.*

1. A cylinder with a piston, piston rod, connecting rod and a crank,
2. Suction pipe,
3. Delivery pipe,
4. Suction valve, and
5. Delivery valve.

▶ 20.3 WORKING OF A RECIPROCATING PUMP

Fig. 20.1 shows a single acting reciprocating pump, which consists of a piston which moves forwards and backwards in a close fitting cylinder. The movement of the piston is obtained by connecting the piston rod to crank by means of a connecting rod. The crank is rotated by means of an electric motor. Suction and delivery pipes with suction valve and delivery valve are connected to the cylinder. The suction and delivery valves are