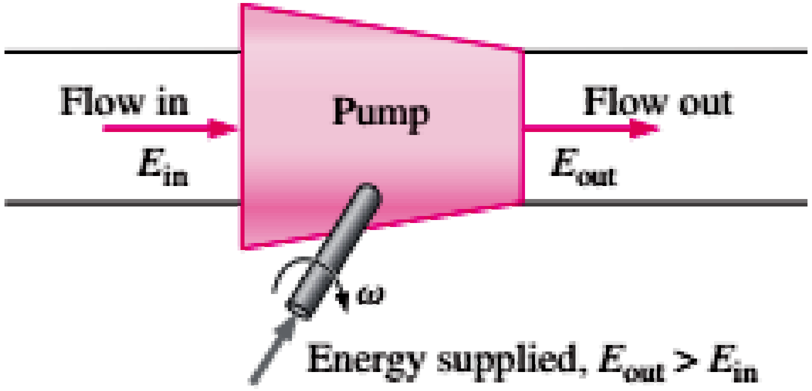
PUMPS

The word ***pump*** is a general term for any fluid machine that ***adds* energy to a fluid**. Some authors call pumps **energy absorbing devices** ***since energy is supplied to them***, and they transfer most of that energy to the fluid, usually ***via a rotating shaft*** (Fig.*a*). ***The increase in fluid energy is usually felt as an increase in the pressure of the fluid***. The pumps are generally used for lifting liquids from a lower level to a higher level.



(*a*)

	Fan	Blower	Compressor
ΔP	Low	Medium	High
$\dot{V}$	High	Medium	Low

Fluid ***machines that move liquids*** are called **pumps,** but there are several other names for ***machines that move gases***.

A **fan** is a ***gas pump*** with relatively ***low pressure rise (a few inches of water) and high flow rate***. **Examples** include ceiling fans, house fans, and propellers.

A **blower** is a ***gas pump*** with relatively ***moderate to high pressure rise (up to 1 atm) and moderate to high flow rate***. **Examples** include centrifugal blowers and squirrel cage blowers in automobile ventilation systems, furnaces, and leaf blowers.

A **compressor** is a ***gas pump*** designed to deliver a ***very high pressure (above 1 atm) rise, typically at low to moderate flow rates***. **Examples** include air compressors that run pneumatic tools and inflate tires at automobile service stations, and refrigerant compressors used in heat pumps, refrigerators, and air conditioners.

Classification of Pumps

There are two basic types of pumps:

Positive-displacement (PDPs) and

Rotodynamic or momentum-change pumps.

Positive-displacement ***pump forces the fluid along by volume changes***. A cavity opens, and Energy transfer to the fluid is accomplished by movement of the boundary of the closed volume, causing the volume to expand or contract, thereby sucking fluid in or squeezing fluid out, respectively.

- Pump
  - Positive displacement pump
    - Reciprocating pump
      - Piston pump
        - Single acting
        - Double acting etc.
      - Plunger pump
        - Single acting
        - Double acting etc.
      - Bucket pump

      (Also two throw, three throw pump etc).
    - Rotary pump (positive displacement with circular motion)
      - Constant delivery pump
        - Gear pump
        - Screw pump
        - Vane pump
        - Radial piston pump
      - Variable delivery pump
        - Rotary piston pump
        - Radial piston pump
  - Rotodynamic or dynamic pressure pump
    - Centrifugal pump (Radial type)
      - Volute type pump
        - Single stage pump
        - Multi-stage pump
      - Turbine or diffusion type pump
        - Single stage pump
        - Multistage pump or Deepwell pump (deepwell pump is (always vertical)
    - Axial or propeller pump
    - Semi-axial or mixed flow pump

A brief **classification of PDP** designs is as follows:

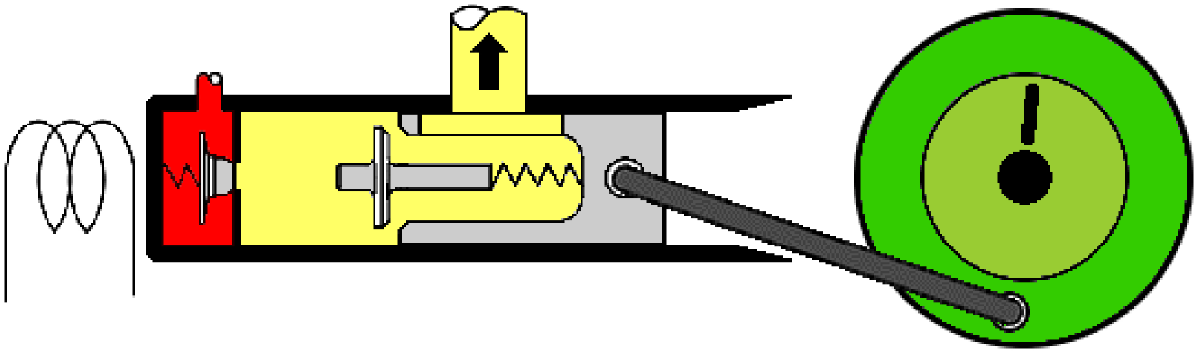
A. Reciprocating

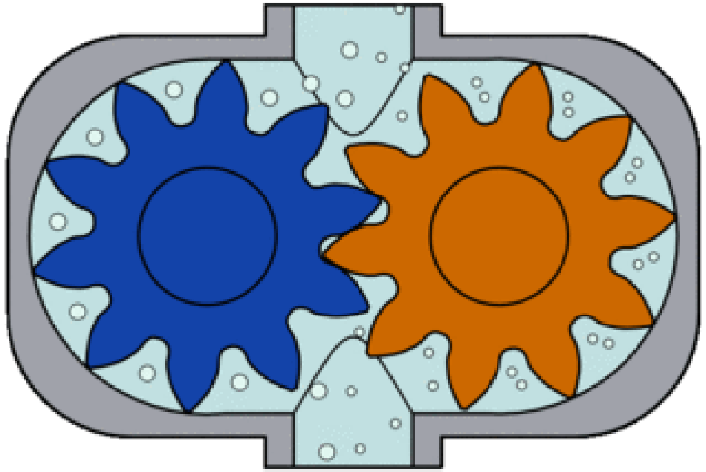
1. Piston or plunger
   a. Single acting
   b. Double acting
2. Diaphragm

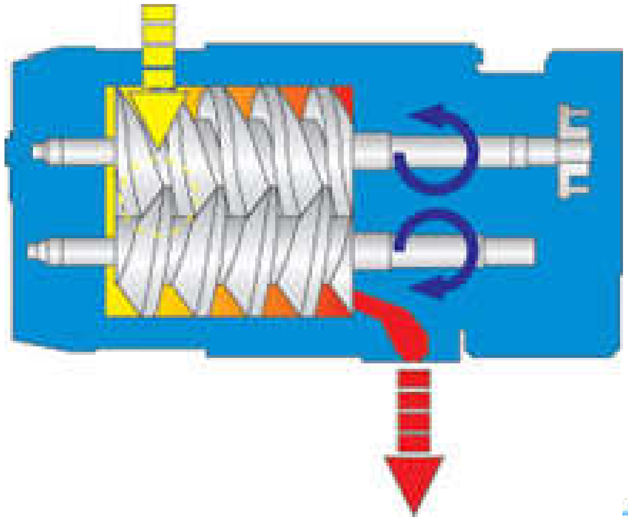
B. Rotary

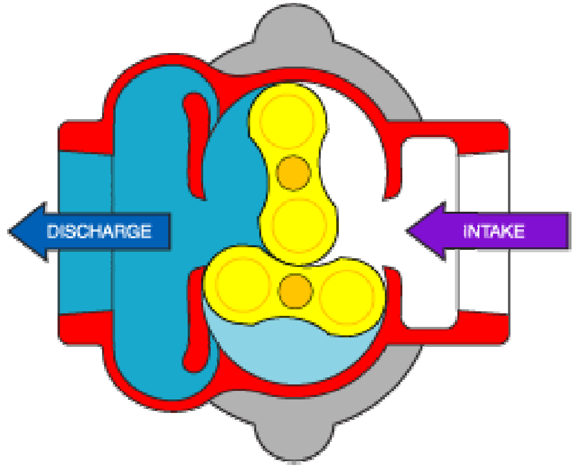
1. **Single rotor**
   a. Sliding vane
   b. Flexible tube or lining
   c. Screw
   d. Peristaltic (wave contraction)
2. **Multiple rotors**
   a. Gear
   b. Lobe
   c. Screw
   d. Circumferential piston

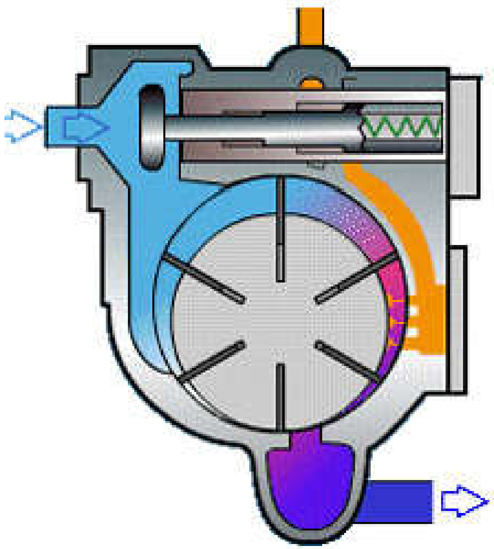
The **mammalian heart** is a good example of positive displacement pump. **All PDPs deliver a pulsating or periodic flow** as the cavity

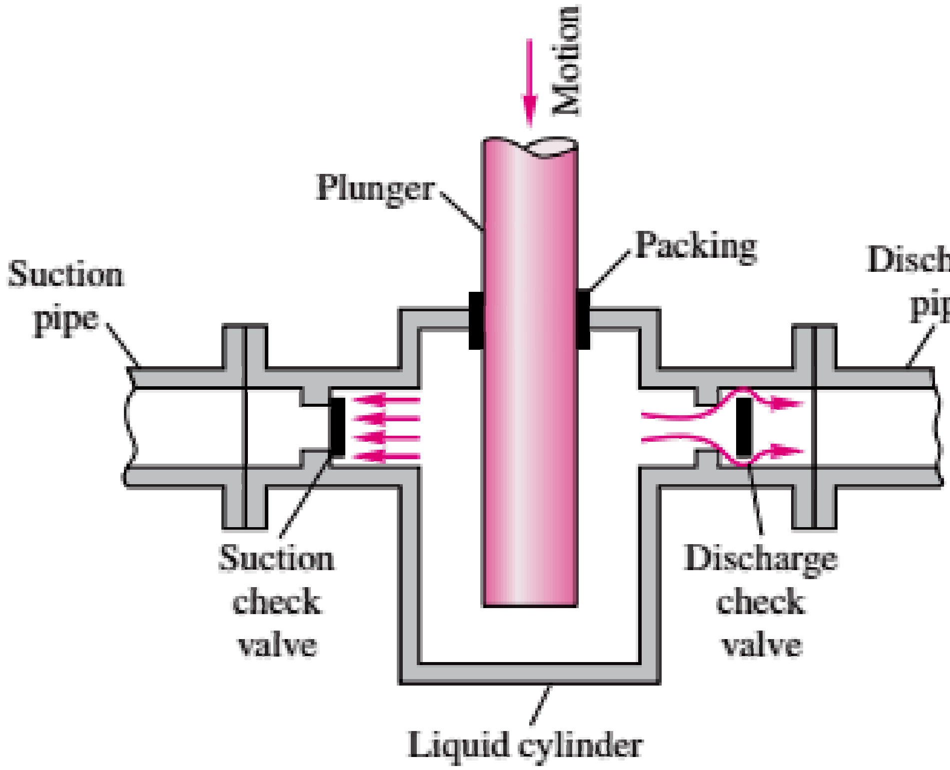




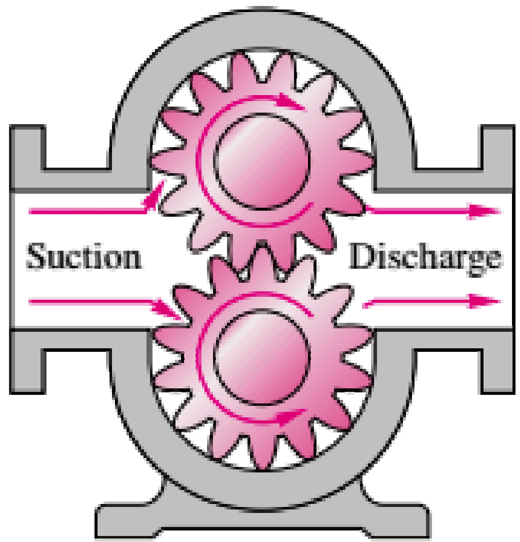



DISCHARGE
INTAKE

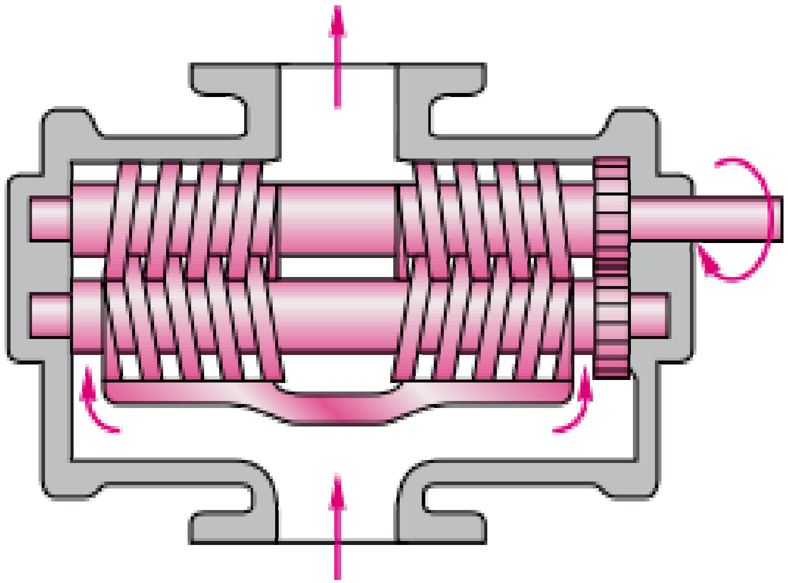




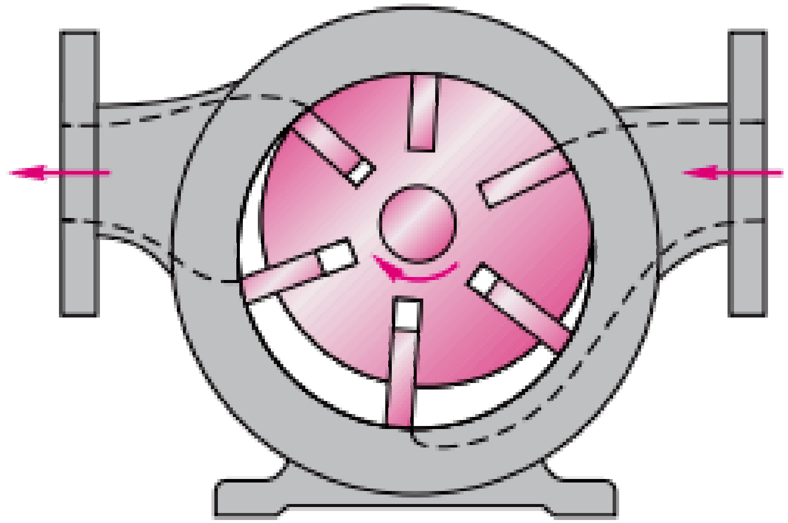
(*a*) reciprocating piston or plunger,



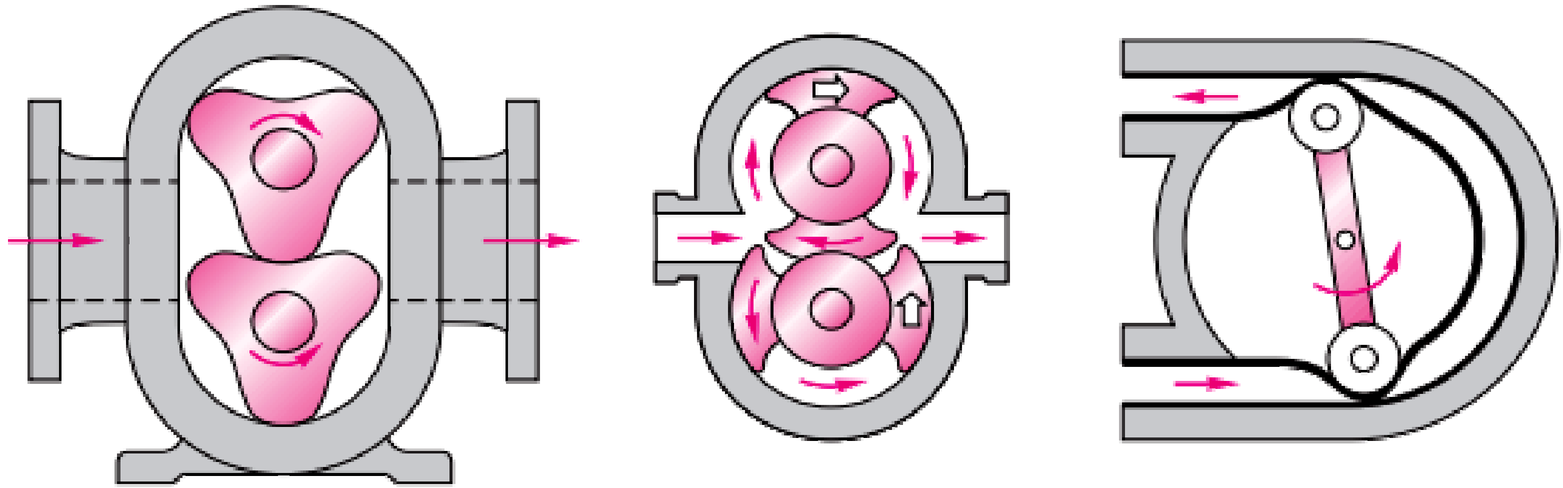
(*b*) external gear pump



(*c*) double-screw pump,



(*d*) sliding vane



(*e*) three-lobe pump, (*f*) double circumferential piston, (*g*) flexible-tube squeegee

Fig. Schematic design of positive-displacement pumps.

volume opens, traps, and squeezes the fluid. ***Their great advantage is the delivery of any fluid regardless of its viscosity***. Since PDPs compress mechanically against a cavity filled with liquid, **a common feature is that they develop immense pressures if the outlet is shut down for any reason**. Sturdy construction is required, and complete shutoff would cause damage if pressure relief valves were not used.

Dynamic pumps simply ***add momentum to the fluid by means of fast-moving blades or vanes or certain special designs***. ***There is no closed volume***: ***The fluid increases momentum while moving through open passages and then converts its high velocity to a pressure increase by exiting into a diffuser section***.

Dynamic pumps can be classified as follows:

A. Rotary

1. Centrifugal or radial exit flow
2. Axial flow (propeller pumps)
3. Mixed flow (between radial and axial)

B. Special designs

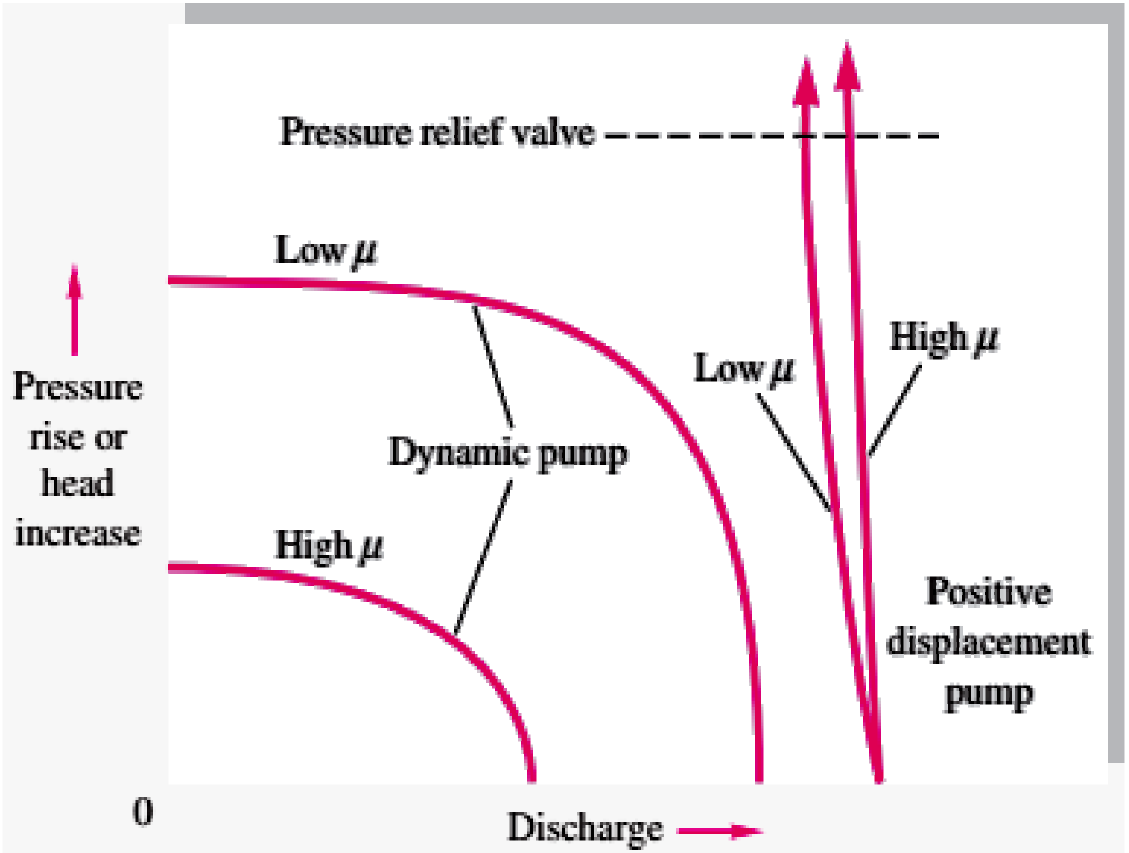
1. Jet pump or ejector
2. Electromagnetic pumps for liquid metals
3. Fluid-actuated: gas lift or hydraulic ram

Dynamic pumps generally provide ***a higher flow rate than PDPs*** and ***a much steadier discharge*** ***but are ineffective in handling high-viscosity liquids***. Dynamic pumps also generally need ***priming***; if they are filled with gas, they cannot suck up a liquid from below into their inlet. **The PDP**, on the other hand, ***is self-priming for most applications.***

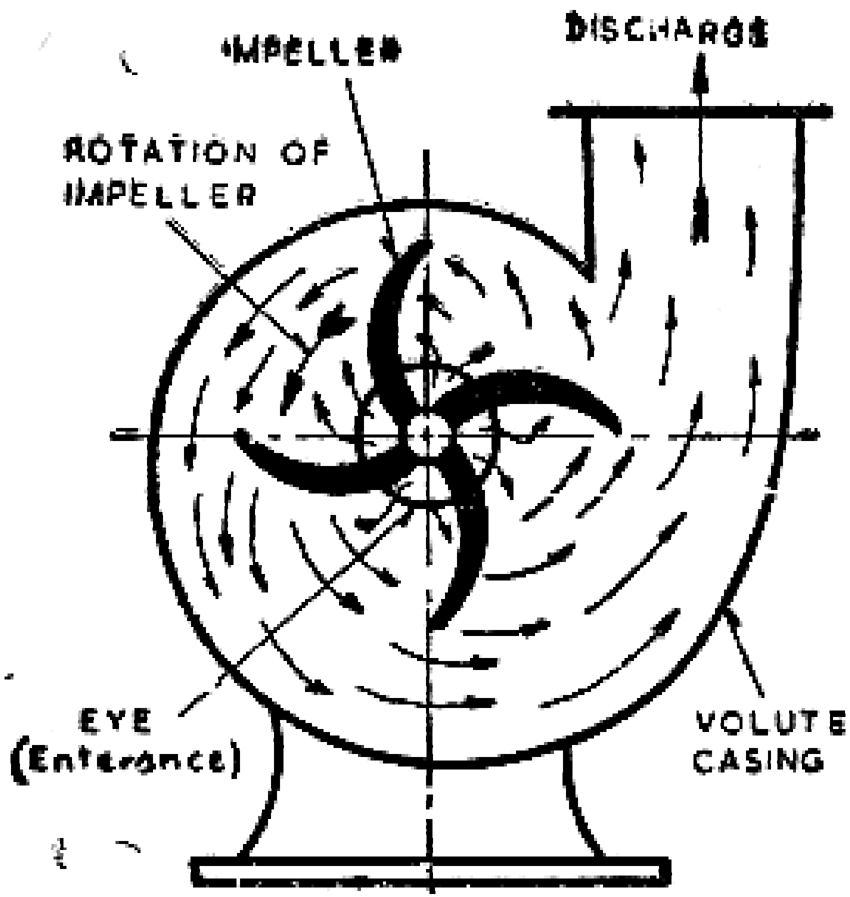
- **A dynamic pump** can ***provide very high*** **flow rates** (up to 300,000 gal/min) but usually ***with moderate pressure rises*** (a few atmospheres). The rate of flow depends not only upon the size and speed of the pump, but also upon the resistance the liquid encounters in the conduit. **In contrast, PDP can operate** up to ***very high pressures (300 atm) but typically produces low flow rates (100 gal/min).***

The relative *performance* (Δp versus Q) is quite different for the two types of pump, as shown in Fig. **At constant shaft rotation speed**, ***the PDP produces nearly constant flow rate and virtually unlimited pressure rise***, with little effect of viscosity. ***The flow rate of a PDP cannot be varied except by changing the displacement or the speed***. The reliable constant-speed discharge from PDPs has led to their wide use in metering flows.

The dynamic pump, by contrast in Fig., ***provides a continuous constant-speed variation of performance, from near-maximum Δp at zero flow (shutoff conditions) to zero Δp at maximum flow rate***. High-viscosity fluids sharply degrade the performance of a dynamic pump.

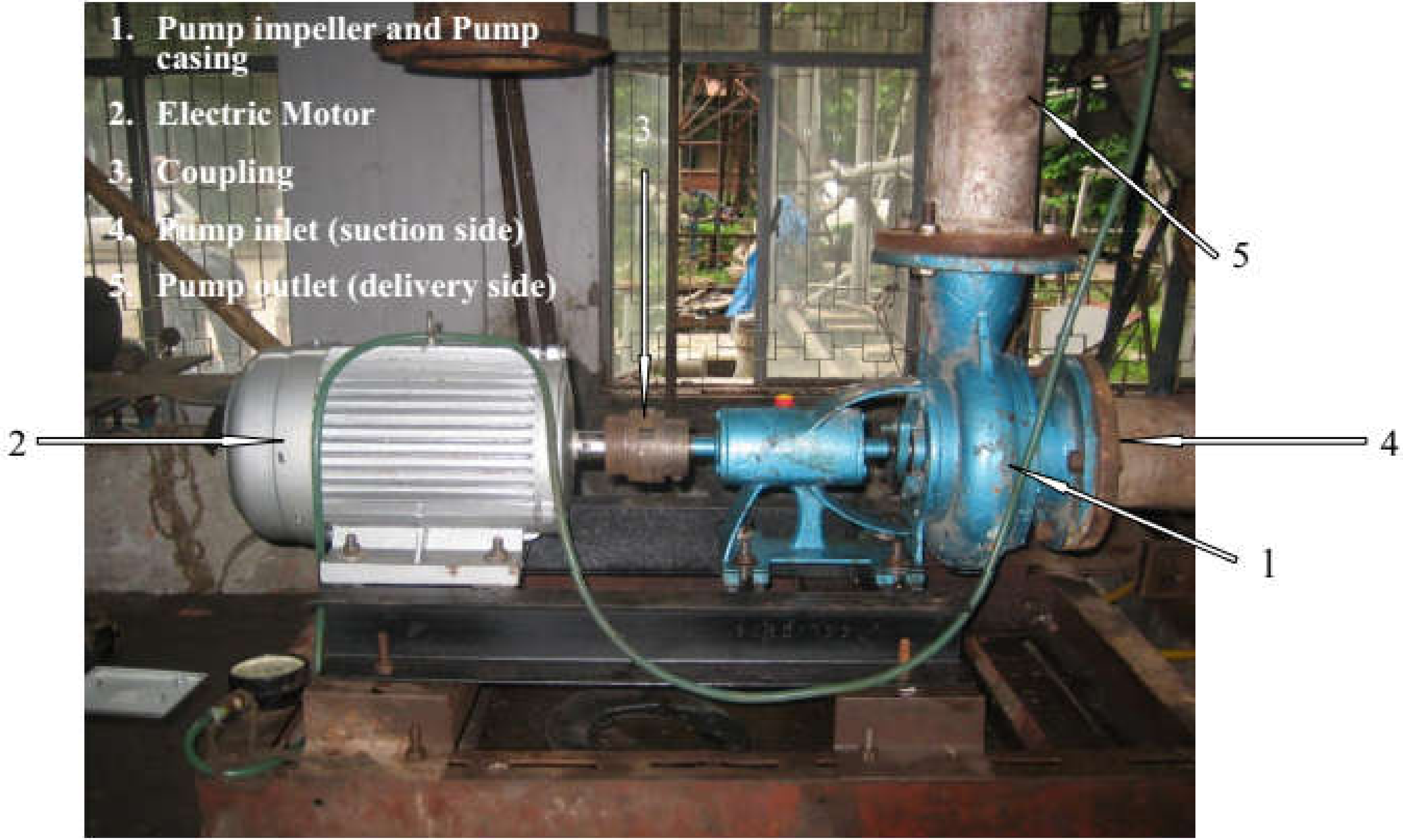


Centrifugal Pumps



The centrifugal pump is a contrivance to raise liquids from a lower to a higher level by creating the required pressure with the help of centrifugal action. In general it can be defined as a machine which increases the pressure energy of a fluid, as a pump may not be used to lift water at all, but just to boost the pressure in a pipe line. The pump is driven by power from an external source; usually an **electric motor**. Whirling motion is imparted to the liquid by means of backward curved blades mounted on a wheel known as **the impeller**. Liquid enters the impeller at its centre technically known as the **eye of the pump** and discharges into the casing surrounding the impeller. ***The pressure head developed by centrifugal action is entirely due to the velocity imparted to the liquid by the rotating impeller*** and not due to any displacement or impact.

Identification of different major parts of a centrifugal pump:



Principle and Operation

The first step in the operation of a **pump is priming** that is, the suction pipe and casing are filled with water **so that no airpocket is left**. Now ***the revolution of the pump impeller inside a casing full of water produces a forced vortex which is responsible for imparting a centrifugal head to the water***. ***Rotation of impeller effects a reduction of pressure at the centre. This causes the water in the suction pipe to rush into the eye***. The speed of the pump should be high enough to produce centrifugal head sufficient to initiate discharge against the delivery head.

Mechanical action of the pump is to impart a velocity to the water. A water particle with a given velocity will rise to the same vertical height through which any particle should fall freely under gravity in order to attain the same velocity starting from rest. The required relation therefore is,

$$v = \sqrt{2gH} \qquad \text{or} \qquad H = \frac{v^2}{2g}$$

Thus if the outlet velocity of water in a pump is *v*, the pump can theoretically deliver against a head of $\boldsymbol{v^2/2g}$.

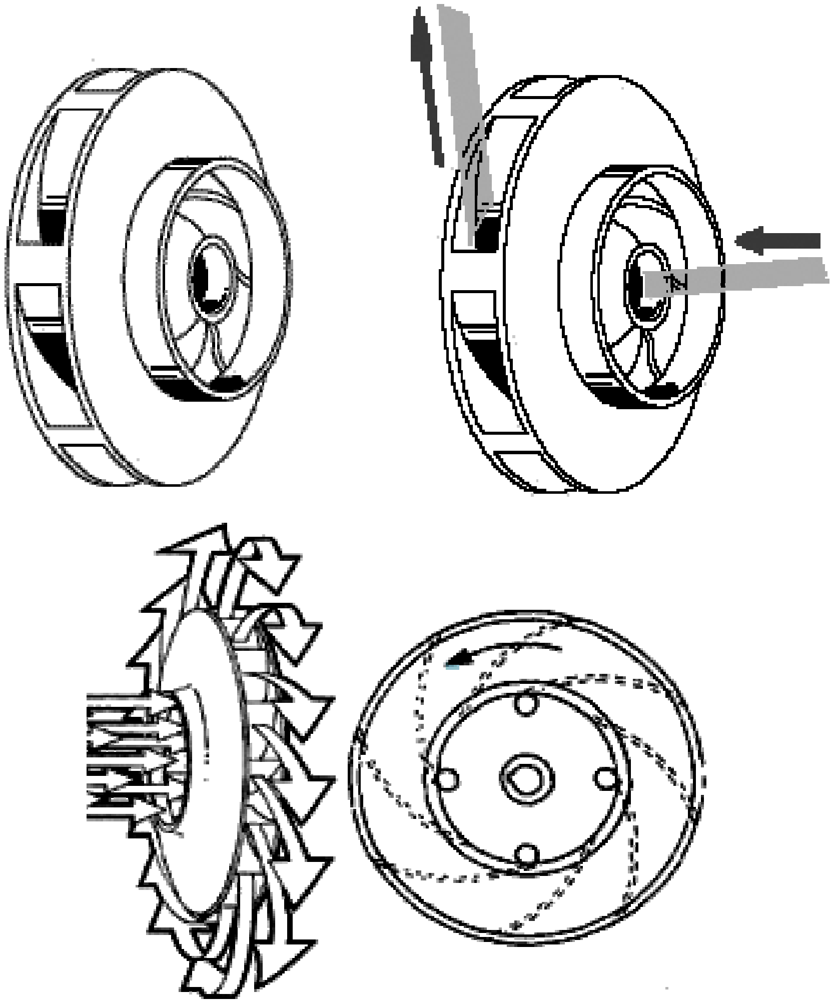
Component Parts of a Centrifugal Pump

The **main components** of centrifugal pumps are

(1) the impeller,
(2) the casing,
(3) the drive shaft with gland and packing.
(4) Suction Pipe
(5) Delivery pipe

(1) Impeller

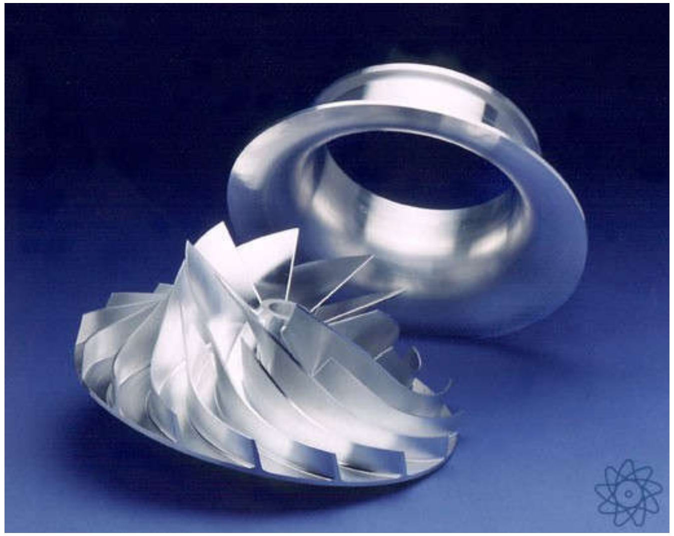
The wheel (or disc) fitted with a series of curved vanes (or blades) mounted perpendicularly on its surface is known as an impeller. The impeller is mounted on a shaft. The shaft is usually coupled to an electric motor.



The impellers are usually **of 3 types :**

(a) Shrouded or closed impeller

Fig. (a) shows a shrouded impeller. ***The vanes are completely closed by the plates on both sides***. ***This type of impeller has a very high efficiency*** as it provides a smooth passage for the liquid. This type of impeller is used when the ***liquid to be pumped is relatively free from debris*** so that the passage is not choked.



This type is meant to handle non-viscous liquid such as ordinary water, hot water, hot oil and chemicals like acids etc. Material of the impeller should be selected according to the chemical properties of liquid used.

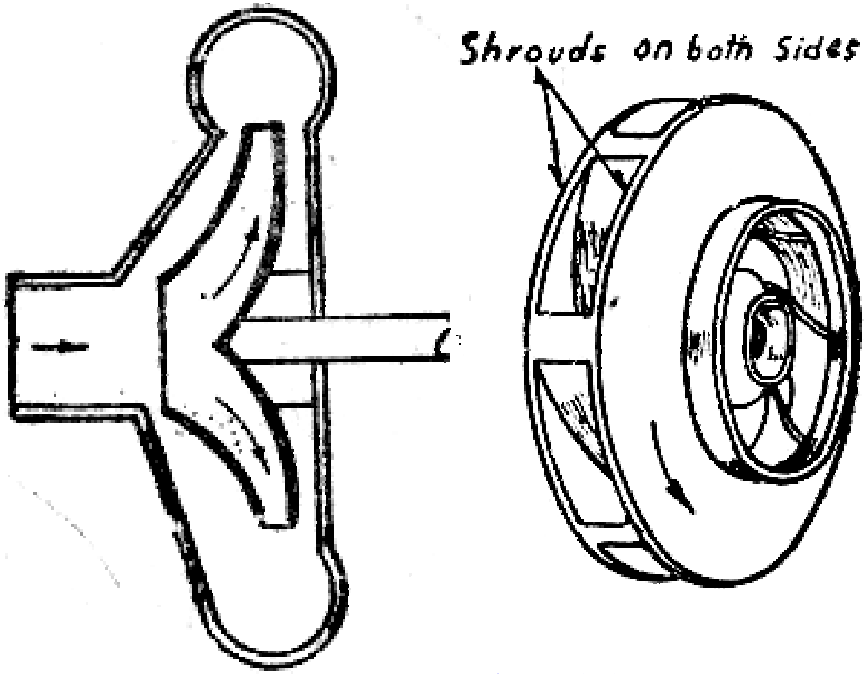
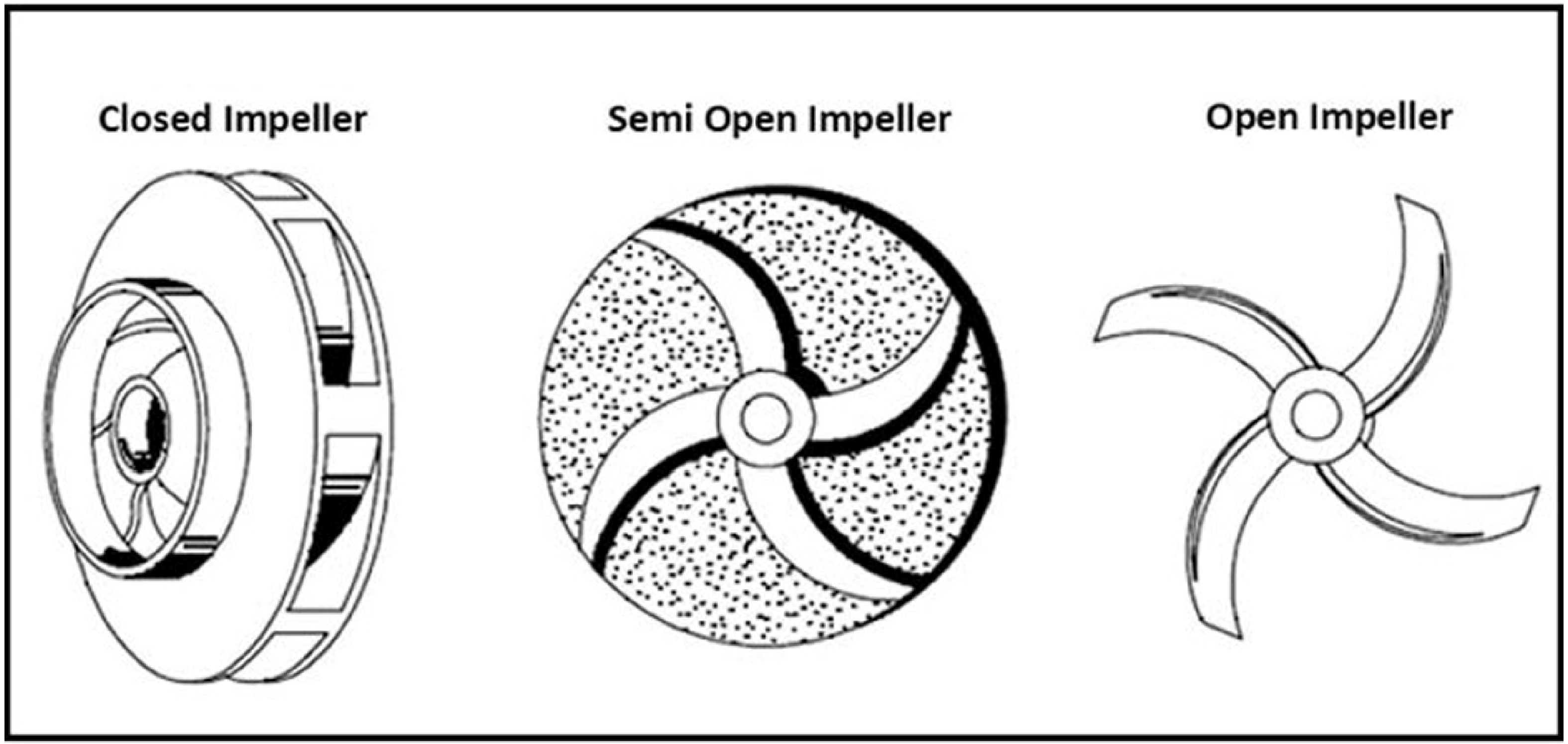


Fig. (a)


Closed Impeller
Semi Open Impeller
Open Impeller

(b) Semi-open impeller:

Fig. (b) shows a semi-open impeller. In this type of impeller, ***there is a plate only on one side; it has only the base plate, there is no crown plate***. This type of impeller is used ***when the liquid contains a small amount of debris***.

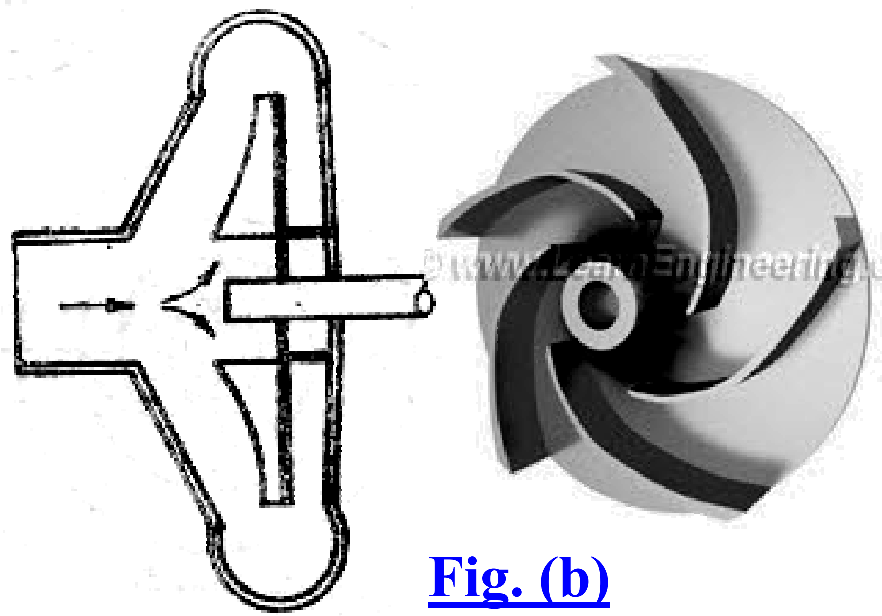


Fig. (b)

This pump ***is used for viscous liquids*** such as sewage water, paper pulp, sugar molasses etc. ***In order to minimize the changes or impeller getting clogged, the number of vanes is reduced and their height is increased***.

(c) Open impeller:

Fig. (c) shows an open impeller. ***The vanes are open on both sides. They have neither the crown plate nor the base plate***. This type of impeller is used ***when the liquid contains a large amount of debris***.

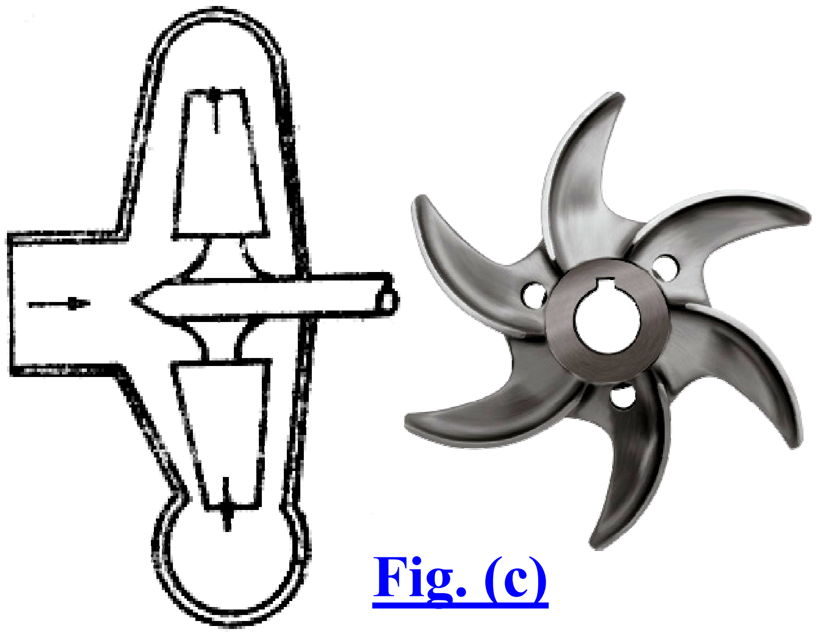


Fig. (c)

Such pumps ***are used in dredgers and elsewhere for handing mixtures of water, sand, pebbles and clay***, ***in which the solid contents may be as high as one part in four***. The impeller has very rough duty to perform. Its life depends upon the material handled, may be 40 or 50 hours or in some cases from 500 to 1,000 hours.

(2) Casing

The casing of a centrifugal pump is similar to the casing of a reaction turbine. It is an air tight chamber covering the impeller. **Pump casing should be so designed as to minimize the loss of kinetic head through eddy formation etc**. **Efficiency of the pump largely depends on the type of casing**. Depending upon the type of the casing the centrifugal pumps are classified as

- **(a) Volute pump**,
- **(b) Volute pump with a vortex chamber**,
- **(c) Diffuser pump**,

(a) Volute pump:

In a volute pump, the casing is of spiral form. The cross sectional area of the moving stream gradually increases from the tongue towards the delivery pipe. Thus the kinetic energy is converted into the pressure energy. ***The cross-sectional area at any point is therefore proportional to the quantity of water flowing across that section and therefore the mean velocity remains constant; the streamlines may be assumed to be continuous***. Thus the losses of kinetic head which would occur if simply a circular casing were employed are avoided.

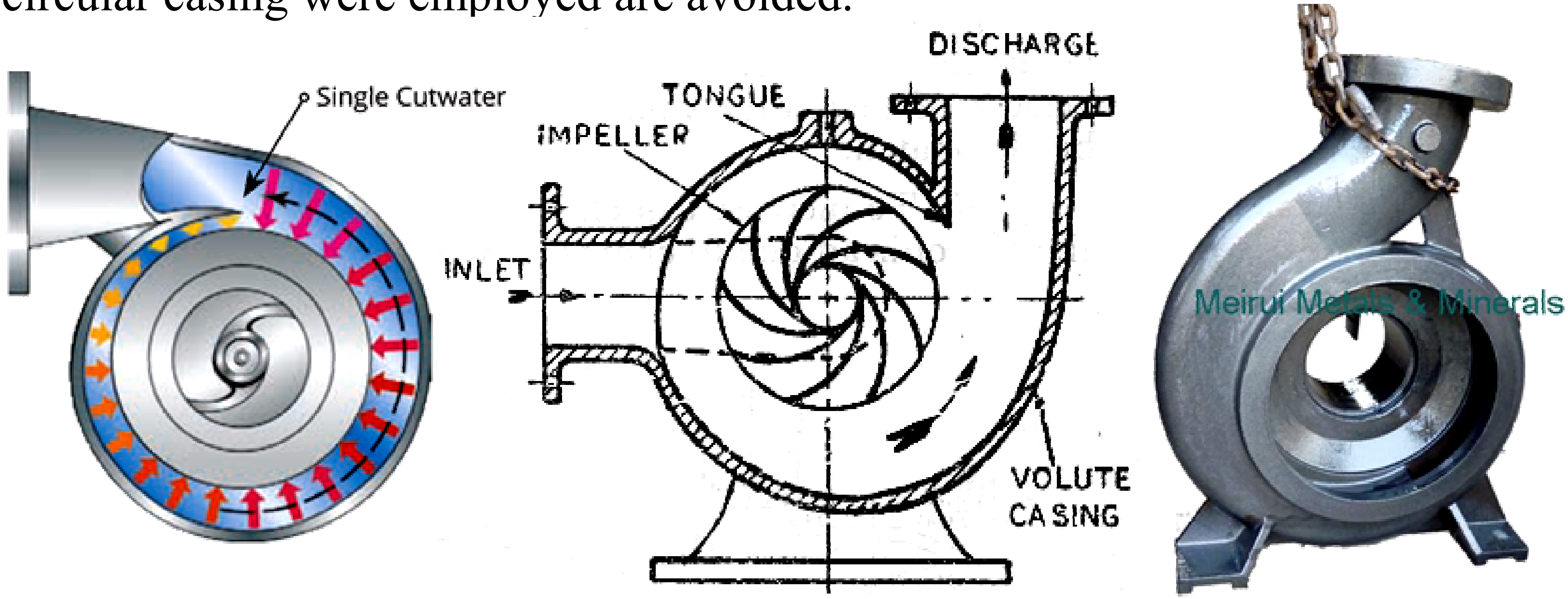


Fig.(a) Volute pump.

(b) **Volute pump with a vortex chamber**:

Prof. Thomson improved the volute pump by introducing a vortex chamber.

The vortex chamber, which is a circular chamber, is provided between the impeller and the volute chamber. ***A free vortex is formed and as the water moves radially away from the centre***, ***velocity of whirl decreases thus building up pressure at the cost of velocity.*** ***The vortex chamber converts some of the kinetic energy into the pressure energy***. The volute chamber further increases the pressure energy. Thus a volute pump fitted with a vortex chamber is more efficient than a simple volute pump.

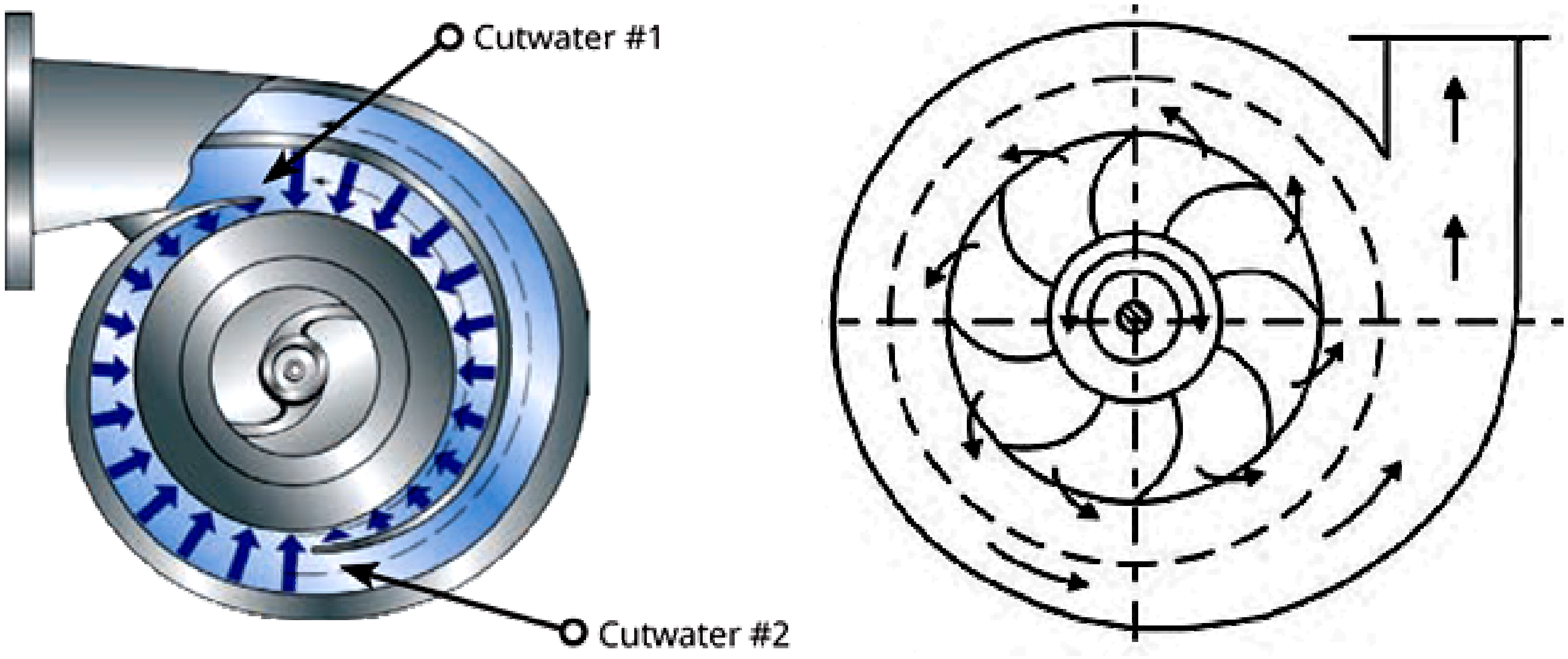
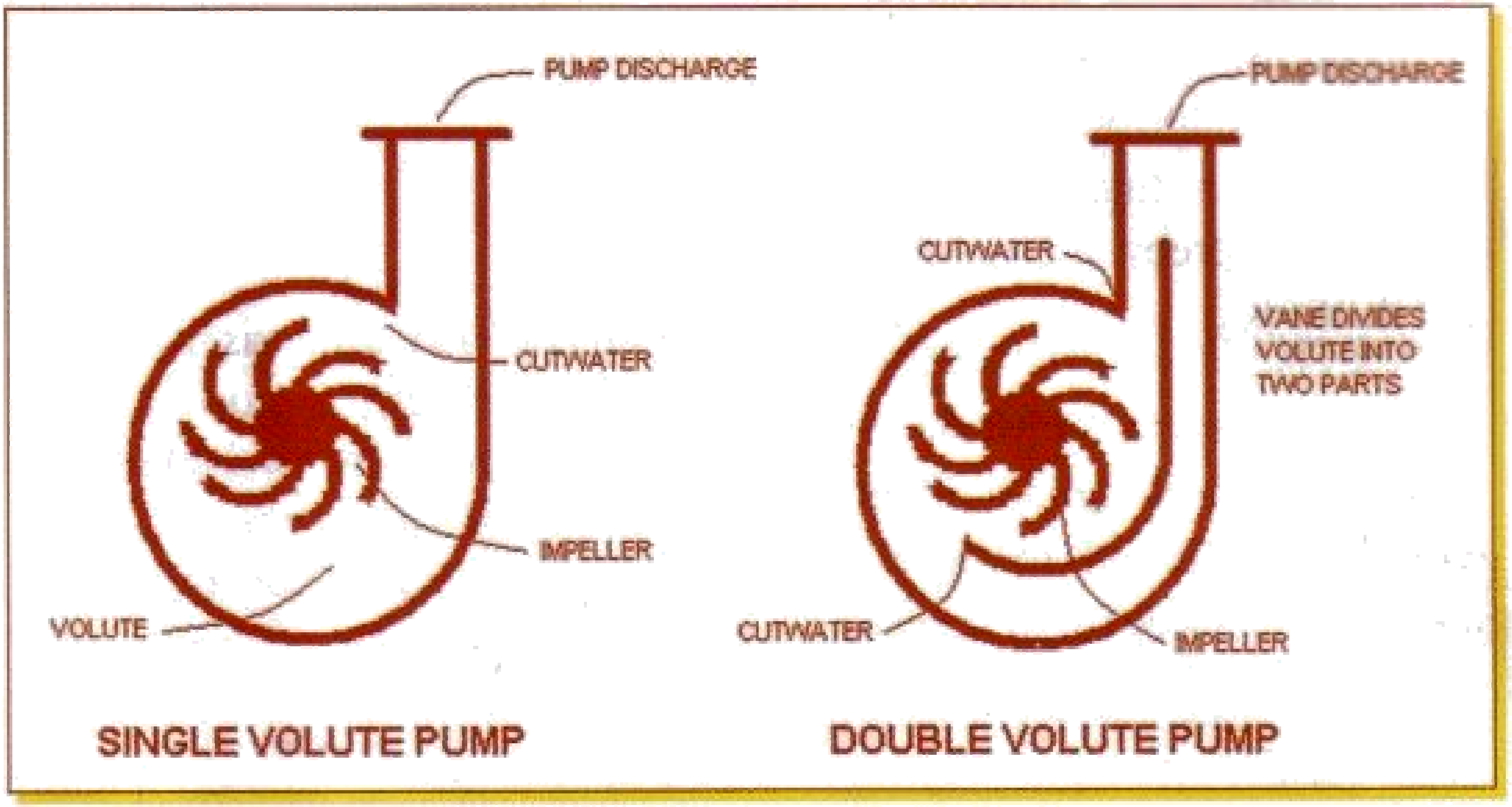


Fig.(b) Volute pump with a vortex chamber


PUMP DISCHARGE
CUTWATER
IMPELLER
VOLUTE
SINGLE VOLUTE PUMP
PUMP DISCHARGE
CUTWATER
VANE DIVIDES VOLUTE INTO TWO PARTS
CUTWATER
IMPELLER
DOUBLE VOLUTE PUMP

(c) Diffuser or turbine pump :

In this type of pump, the casing consists of guide vanes like a reaction turbine. The liquid leaving the impeller passes through the passage between guide vanes. The passage has a gradually increasing area. The velocity of flow increases and the kinetic energy is converted into pressure energy. ***Angle of guide vanes at the entrance should coincide with the direction of absolute velocity of water at impeller outlet***. This type of pump has the following two drawbacks:

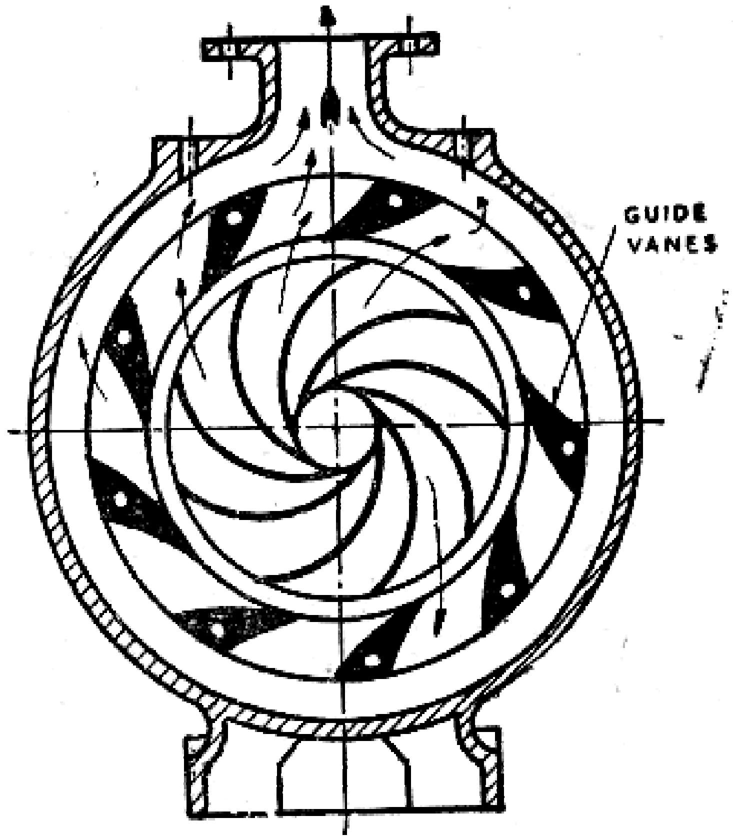


Fig.(c) Diffuser pump

(i) ***They give the maximum efficiency only at one discharge for which the guide vanes are designed***. At other discharges, the efficiency is low.

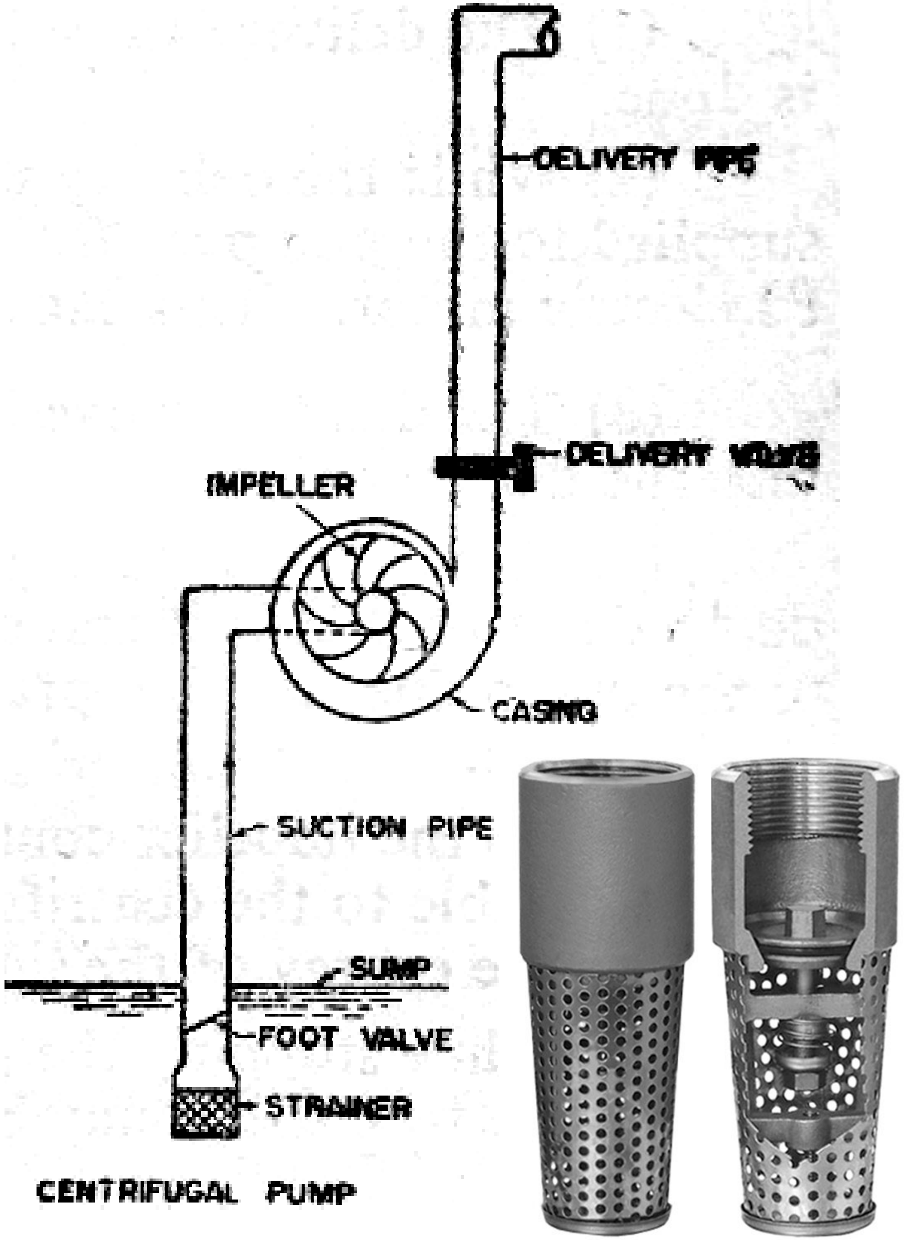
(ii) They are costlier than volute pump.

This arrangement is employed in all multi-stage pumps. Single stage pumps, however, have less expensive volute casing.

Diffusion Pumps may be either horizontal or vertical shaft type. The vertical type occupies very little space and suitable for installation in deep wells. It is often called a deepwell pump. They may also be used in narrow wells and mines etc.

(4) Suction Pipe:

The suction pipe connects the sump with the pump inlet. ***The lower end of the pipe is fitted with a non-return foot valve***. The foot valve does not permit the liquid to drain out of the suction pipe when the pump is not working. This also helps in priming. **A strainer** is usually provided at the lower end so that only relatively clear liquid enters the suction pipe. The upper end of the suction pipe is connected to the center of the impeller known as the **eye of the pump**.



CENTRIFUGAL PUMP

(5) Delivery pipe:

The delivery pipe connects outlet of the casing to the delivery reservoir. A valve is provided on the delivery pipe to regulate the supply of water. The valve should be close to the outlet of the pump,

Priming of a Centrifugal Pump

Before starting a pump, its impeller and suction pipe have to be filled with water in order to remove any air, gas or vapour from the waterways of the pump. **If a centrifugal pump is not primed before starting**, ***air pockets inside the impeller may give rise to vortices and cause discontinuity of flow***. ***The wearing rings may rub and seize causing serious damage if the pump is allowed to run dry***. It is also essential that packing be lubricated by liquid leaking past it.

Priming is done by pouring water through a funnel, displaced air being allowed to escape through air vents. The **air-vent valve** provided in the pump casing is opened when priming is done. The priming is continued till all the air from the suction pipe, impeller, and the casing has been removed. It may be noted that had there been no foot valve in the suction pipe, the entire liquid poured into the priming funnel would have gone to the sump, and the priming would not have been complete. When a pump is being primed or stopped, **the delivery valve should be kept closed**.

Large pumps are primed by evacuating the casing and the suction pipe by a **vacuum pump or by an ejector**. The air is thus drawn up the suction pipe from the sump and the pump is filled with liquid. Sometimes, a special priming reservoir containing the liquid is provided on the suction pipe. By directing the flow from this reservoir, it is possible to prime the pump.

Necessity of priming is the main disadvantage of a centrifugal pump, To overcome this difficulty, the following methods are employed in practice:

(a) the pump is installed below the suction water level,

(b) the pump is equipped with one of the priming or self-priming devices.

Working of a Cenfrifugal Pump

The following procedure is adopted when starting the pump :

(1) The delivery valve is closed and the priming of the pump is done.

(2) While the delivery valve is still-closed, the external energy is supplied to the pump shaft. It is usually done by starting the coupled electric

motor. This causes an increase in the impeller pressure.

(3) The delivery valve is then opened. The liquid starts flowing into the delivery Pipe.

(4) A partial vacuum is created at the eye of the centrifugal pump due to the centrifugal action. The liquid rushes from the sump to the pump due to the pressure difference at the two ends of the suction pipe.

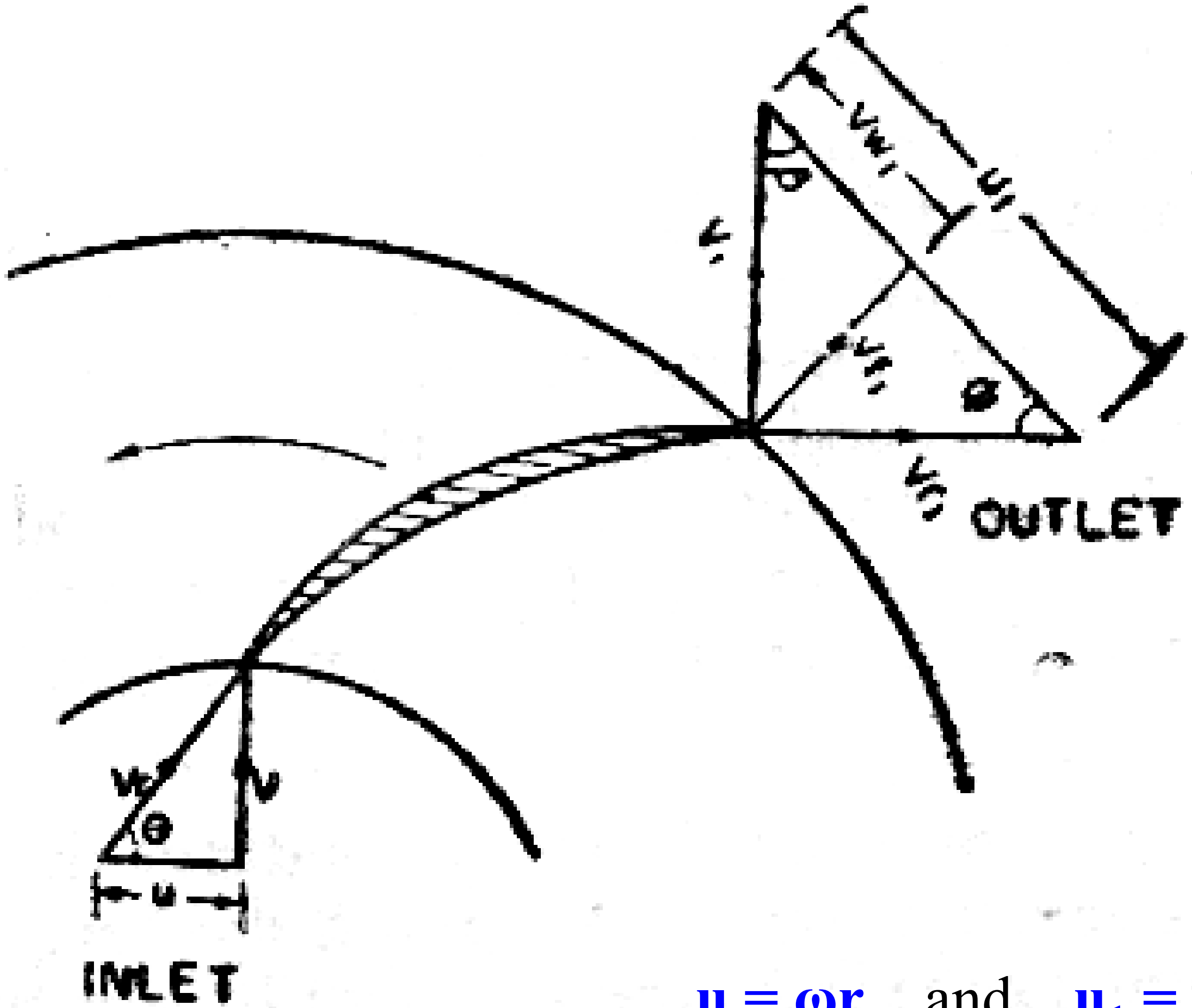
(5) As the impeller continues to run, more and more liquid is made available to the centrifugal pump at the eye. The impeller increases the energy of the liquid and delivers it to the reservoir.

(6) While stopping the pump, the delivery valve should be first closed, otherwise there may be some backflow from the reservoir.

Expression for the Work Done on the Impeller

Fig. shows one vane of the impeller. The liquid enters the impeller at its centre and leaves at its outer periphery. Let us consider the case, **when the liquid enters the impeller radially, that is the absolute velocity of the liquid at the inlet is radial.**

Let ω, be the angular velocity of the impeller and r and r_1 be the radii at inlet and outlet respectively.



$$\mathbf{u = \omega r} \quad \text{and} \quad \mathbf{u_1 = \omega r_1}$$

While ***passing through the impeller, the velocity of whirl changes and there is a change of moment of momentum***.

Torque on the impeller = Rate of change of moment of momentum

Moment of momentum at inlet = 0

Moment of momentum at outlet = $M(V_{w1} r_1)$

Therefore, Torque $= M(V_{w1} r_1)$

Work done per second = Torque × Angular velocity

$$= M(V_{w1} r_1) \times \omega$$

$$= M(V_{w1} u_1)$$

Work done per second per kg of liquid $= V_{w1} u_1$

Work done per second per unit weight $= V_{w1} u_1 / g$

The above equation has been developed on the assumption of the radial flow of inlet. **If the flow is not radial, the expression for work done may be written as**

Work done per second $= M(V_{w1} u_1 - V_w u)$

Work done per second per kg of liquid $= (V_{w1} u_1 - V_w u)$

Work done per second per unit weight $= (V_{w1} u_1 - V_w u)/g$

This is also known as the **Euler momentum equation for centrifugal pumps.**

By substituting the values of V_{w1} and V_w from the velocity triangles, the equation may be transformed as given below:

Work done per second per kg mass $= \left(\frac{V_1^2 - V^2}{2} + \frac{u_1^2 - u^2}{2} + \frac{V_r^2 - V_{r1}^2}{2}\right)$

This is sometimes called **the fundamental equation of the centrifugal pump.**

Work done per second per unit weight $= \left(\frac{V_1^2 - V^2}{2g} + \frac{u_1^2 - u^2}{2g} + \frac{V_r^2 - V_{r1}^2}{2g}\right)$

Head Developed, Efficiency, and power Required

(a) Static Head:

It is the difference of elevation between the liquid surface in the sump and that in the reservoir to which the liquid is delivered by the pump. Thus

$$\mathbf{H_s = h_s + h_d}$$

where h_s is the suction head and h_d is the delivery head.

(b) Manometric Head:

The manometric head (H_m) is **the head developed in the pump**. It is equal to the energy given to the liquid by the impeller minus the losses in the pump. It is convenient to consider energy per unit weight. Thus

$$\mathbf{H_m = V_{w1} u_1/g - (h_{Li} + h_{Lc})}$$

where h_{Li} is the head loss in impeller and h_{Lc} is the head loss in the casing. Obviously, **the manometric head is greater than the static head**. The two heads are related by the expression

$$H_m = H_s + \text{losses in the pipe} + v_d^2/2g$$
$$= (h_s + h_d) + h_{fs} + h_{fd} + v_d^2/2g$$

where h_{fs} and h_{fd} are frictional losses in the suction and delivery pipes and v_d is the velocity in the delivery pipe.

(1) Mechanical Efficiency:

Because of mechanical losses, the power supplied to the shaft (P) is greater than the power actually delivered to the impeller.

$$P = \text{Impeller power} + \text{Mechanical losses}$$
$$= (W/g)(V_{w1}u_1) + \text{Mechanical losses}$$

The ratio of the impeller power to the shaft power is called the Mechanical efficiency (η_m). Thus

$$\boxed{\eta_m = \frac{\left(\frac{W}{g}\right)(V_{w1}\,u_1)}{P}}$$

(2) Manometric Efficiency:

The power delivered by the pump is less than the impeller power because of hydraulic losses in the pump.

Power delivered by the pump is $= WH_m$

It is also **known as Water power**. Thus

Water Power = Impeller power – Power lost in the pump

$$WH_m = (W/g)(V_{w1}u_1) - W(h_{Li} + h_{Lc})$$

The ratio of water power to the impeller power is known as the manometric efficiency (η_{mano}).

$$\boxed{\eta_{mano} = \frac{WH_m}{\left(W/g\right)(V_{w1}u_1)} = \frac{H_m}{V_{w1}u_1/g}}$$

(3) Overall Efficiency (η_o):

It is the ratio of the water power to shaft power. Thus

$$\boxed{\eta_o = \frac{WHP}{SHP} = \frac{WH_m}{P}}$$

$$= WH_m \times \frac{1}{W(V_{w1}u_1/g)} \times \frac{W(V_{w1}u_1/g)}{P}$$

$$\boxed{\eta_o = \eta_{mano} \times \eta_m}$$

Therefore, **the overall efficiency is the product of the manometric efficiency and the mechanical efficiency**.

(4) Volumetric efficiency (η_v):

It is the ratio of the discharge from the pump (*Q*) **to the discharge flowing through the impeller** ($Q + \Delta Q$). ***The difference between the two discharges is because of leakage***. Thus

$$\boxed{\eta_v = \frac{Q}{Q + \Delta Q}}$$

The overall efficiency can be written accounting volumetric efficiency as

$$\boxed{\eta_o = \eta_{mano} \times \eta_m \times \eta_v}$$

The volumetric efficiency is slightly less than one. **Unless otherwise mentioned, the volumetric efficiency will be taken as unity**.

There is an another power used sometimes known as the static power. **The static power is given by WH_s**, where h_s is the total static head. ***Because of frictional losses in the pipes, the static power is less than the water power***. Thus

Static power = Water power - Losses in pipes

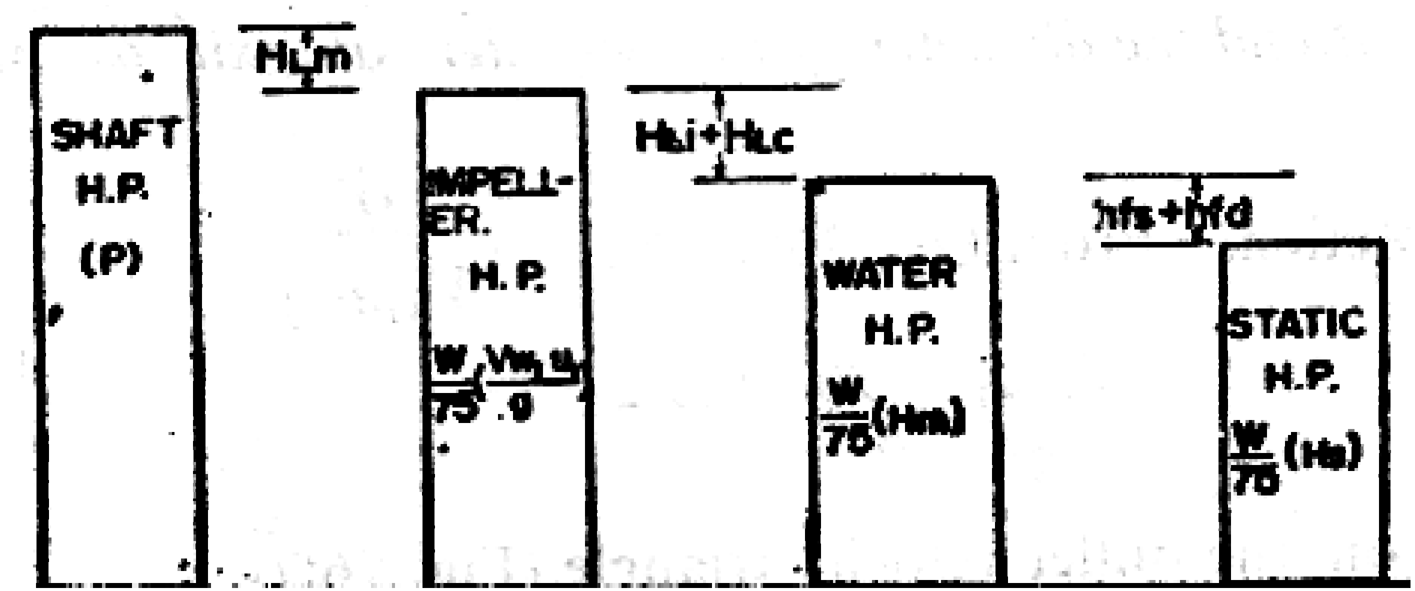
$$WH_s = WH_m - W(h_{fs} + h_{fd} + v^2_d/2g)$$

Various losses and powers are shown diagrammatically in Fig. The powers indicated are horse powers in this figure.

H_{Lm} = Mechanical Losses

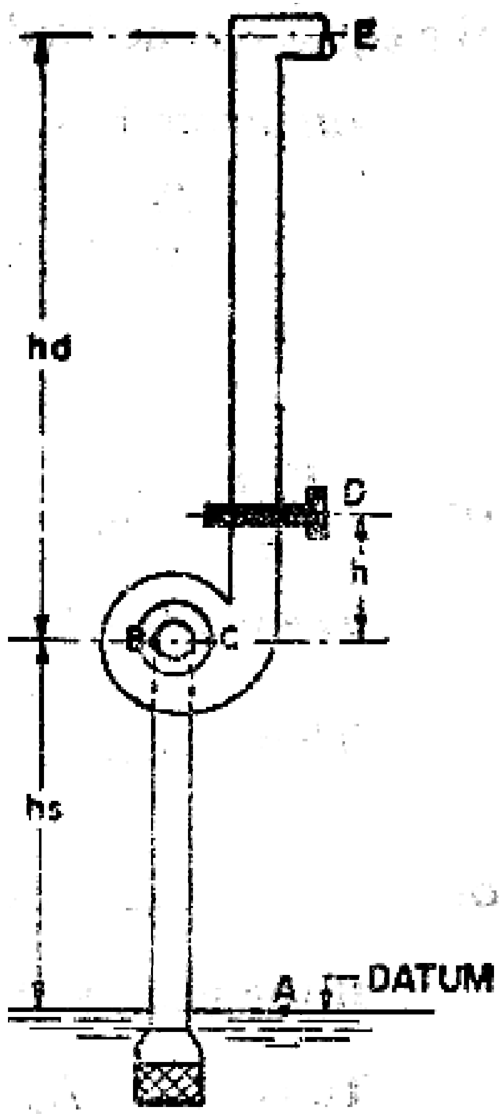
$H_{Li} + H_{Lc}$ = Losses in Pump

$h_{fs} + h_{fd}$ = Losses in Pipes



Pressure Changes in Centrifugal Pumps

Fig. shows a centrifugal pump installation. Let us consider, the point ***A at the sump surface***, ***point B at the inlet to the impeller***, ***point C at the outlet of the impeller***, ***point D at the delivery valve*** and ***point E at the delivery reservoir.***



(i) Applying Bernoulli's equation to points A and B, with datum at the liquid surface in the sump.

$$\frac{P_A}{\gamma} + \frac{V_A^2}{2g} + Z_A = \frac{P_B}{\gamma} + \frac{V_B^2}{2g} + Z_B + h_{fs}$$

$$0 = \frac{P_s}{\gamma} + \frac{V_s^2}{2g} + H_s + h_{fs}$$

where P_s = pressure at point B, V_s = velocity in suction pipe,
H_s = suction lift, h_{fs} = head loss due to friction in suction pipe.

(ii) Applying Bernoulli's equation to points B and C, <u>with datum at the centre of the pump,</u>

$$\frac{P_B}{\gamma}+\frac{V_B^2}{2g}+Z_B+\frac{V_{w1}u_1}{g}=\frac{P_c}{\gamma}+\frac{V_c^2}{2g}+Z_c+h_{Li}$$

$$\frac{P_s}{\gamma}+\frac{V_s^2}{2g}+\frac{V_{w1}u_1}{g}(\text{work done by impeller})=\frac{P_1}{\gamma}+\frac{V_1^2}{2g}+h_{Li}$$

where P_c = pressure at the outlet of impeller,

V_c = absolute velocity at the outlet of impeller

h_{Li} = head loss in the impeller

And $Z_B = Z_c$ because point B and C are at same level

(iii) Applying Bernoulli's theorem to the points C and D, <u>with datum at the centre of the pump,</u>

$$\frac{P_c}{\gamma}+\frac{V_c^2}{2g}+Z_c=\frac{P_D}{\gamma}+\frac{V_D^2}{2g}+Z_D+h_{Lc}$$

$$\frac{P_1}{\gamma}+\frac{V_1^2}{2g}=\frac{P_D}{\gamma}+\frac{V_D^2}{2g}+h+h_{Lc}$$

where P_D = pressure at D (delivery valve),

V_c = velocity at D

h_{Lc} = head loss in the casing

And $h = Z_D - Z_c$ = height of point D above the pump centre

(iv) Applying Bernoulli's equation to points D and E, with datum at pump centre,

$$\frac{P_D}{\gamma} + \frac{V_D^2}{2g} + h = \frac{P_E}{\gamma} + \frac{V_E^2}{2g} + h_d + h_{fd}$$

$$\frac{P_D}{\gamma} = h_d + h_{fd} - h$$

where P_E = pressure at E (delivery reservoir) = 0 (atm pressure),

V_E = velocity at E = V_D

h_{fd} = head loss due to friction in delivery pipe

And

The head **($h_d + h_{fd} + v^2{}_d/2g$) is known as delivery head.**

Minimum Starting Speed

The **centrifugal head created in the pump should be equal to the manometric head** given by,

$$H_m = h_s + h_d + h_{fs} + h_{fd} + V_d^2/2g$$

There will be no flow of liquid until the speed of the pump is such that the required centrifugal head is developed. The centrifugal head developed is given by

$$\boxed{H = \frac{u_1^2}{2g} - \frac{u^2}{2g}}$$

For the minimum starting speed, $\quad H \geq H_m$

Or, $$\frac{u_1^2 - u^2}{2g} = H_m \qquad \text{------ (a)}$$

Again, we know, $H_m = \eta_{mano}\left(\dfrac{V_{w1}\, u_1}{g}\right)$

and $u_1 = \pi D_1 N/60$ and $u = \pi D N/60$. Thus equ. (a) becomes

$$\frac{(\pi N^2)}{2g(60)^2}\,(D_1^2 - D^2) = \frac{\eta_{mano}}{g}\left(V_{w1} \times \frac{\pi D_1 N}{60}\right)$$

$$N = \frac{120 \times \eta_{mano} \times V_{w1}}{\pi} \left(\frac{D_1}{D_1^2 - D^2} \right)$$

This is the **minimum starting speed**.

Problem:

A centrifugal pump impeller has an external diameter of 45cm and discharge area of 0.11 metre^2. The vanes are bent backwards so that the direction of the relative velocity at the outlet makes an angle of 145° with the tangent to the outer periphery, drawn in the direction of the impeller rotation. The diameters of the suction and delivery pipes are 30cm and 23cm respectively. Pressure gauges are at points on the suction and delivery pipes close to the pump, and each gauge 1.50 m above the level in the supply sump showed gauge pressure heads of 3.7 meters below and 19 meters above atmosphere head respectively, when the pump was delivering 200 litres per second of water at 800 rpm. It requires 97 horses power to drive the pump. Find

(i) The loss of head in the suction pipe

(ii) The manometric efficiency

(iii) The overall efficiency

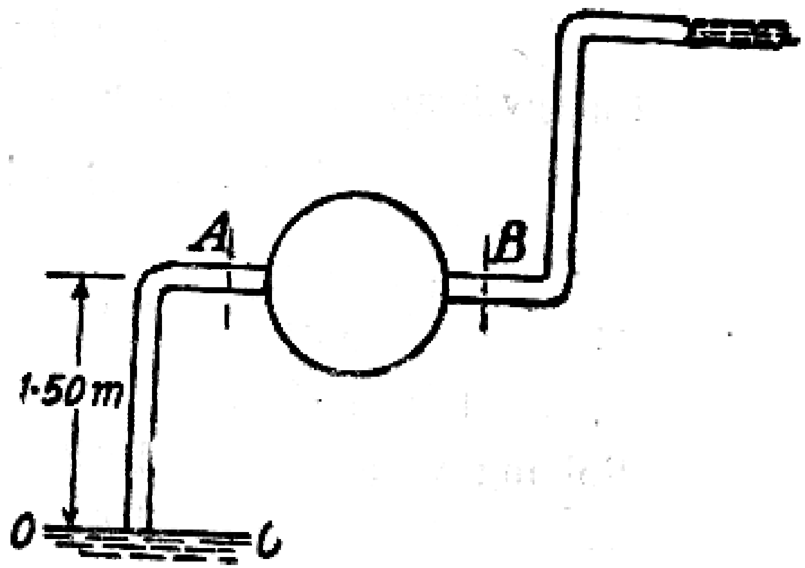
Solution:

$D_1 = 45$ cm, $A_1 = 0.11 m^2 = \pi k D_1 b_1$, $\phi = 180° - 145° = 35°$, $d_s = 30$ cm, $d_d = 23$ cm, $h_s = 1.5$ m, $P_A/\gamma = -3.7m$, $P_B/\gamma = +19m$, $Q = 200$ l/s $= 0.2\ m^3/s$, $N = 800$ rpm, Horse power of the prime mover = 97 hp.

Let, A and B be points in the suction and delivery pipe close to the pump.

Velocity in the suction Pipe,

$$V_s = Q / A_s = 0.2 / (\pi d_s^2/4)$$
$$= 0.2 / (\pi 0.3^2/4)$$
$$= 2.83 \text{ m/s}$$



Velocity in the delivery Pipe,

$$V_d = Q / A_d = 0.2 / (\pi d_d^2/4)$$
$$= 0.2 / (\pi 0.23^2/4)$$
$$= 4.81 \text{ m/s}$$

Let, h_{fs} = loss of head in the suction pipe.

Applying Bernoulli's equation to sump water level and A,

$$\frac{P_w}{\gamma} + \frac{V_w^2}{2g} + Z_w = \frac{P_A}{\gamma} + \frac{V_A^2}{2g} + Z_A + h_{fs}$$

$$0+0+0=-3.7+\frac{2.83^2}{2g}+1.5+h_{fs}$$

h_{fs} = 1.79m = loss of head in the suction pipe

Manometric head (H_m) = Energy head at B - Energy head at A

$$=\left[\frac{P_B}{\gamma}+\frac{V_B^2}{2g}+Z_B\right]-\left[\frac{P_A}{\gamma}+\frac{V_A^2}{2g}+Z_A\right]$$

$$=\left[19+\frac{4.81^2}{2g}+1.5\right]-\left[-3.7+\frac{2.83^2}{2g}+1.5\right]=23.47\ m$$

Velocity of flow at outlet, $V_{f1} = Q / (\pi k D_1 b_1)$ = 0.2 / 0.11 = 1.82 m/s

Peripheral velocity at outlet, $u_1 = \pi D_1 N/60$ = (π × 0.45 × 800)/60
= 18.85 m/s

Velocity of whirl at outlet, $V_{w1} = u_1 - V_{f1} \cot\phi$ = 18.85 – 1.82 cot 35°
= 16.25 m/s

Energy head generated by the impeller = $V_{w1}u_1/g$ = (16.25 ×18.85)/9.81
= 31.22 m

Manometric efficiency (η_{mano}): $$\eta_{mano} = \frac{H_m}{\left[V_{w1}u_1/g\right]}$$

$$= (23.47 / 31.22) \times 100$$

$$= 75.18\ \%$$

Overall efficiency (η_o): $$\eta_o = \frac{WHP}{SHP} = \frac{WH_m}{P} = \frac{\gamma Q H_m}{P}$$

$$= \frac{1000 \times 0.2 \times 23.47}{75 \times 97} \times 100 = 64.52\ \%$$

Performance of Centrifugal Pumps

A pump is designed to work under design speed, discharge and head. When the pump runs at conditions different from the design condition, its performance is quite different. In order to predict the behavior of the pump under varying conditions of speed, discharge and head, tests are usually performed. The results of these tests are plotted in the form of characteristic curves. The following types of curves are usually plotted for the centrifugal pump:

(1) Main Characteristic Curve:

The centrifugal pump ***is operated*** ***at constant speed and the discharge is varied by means of the delivery valve***. Then The manometric head (H_m) and shaft power (P) are measured for each discharge and overall efficiency (η_o) is calculated. The curves are plotted between Q & H_m , Q & P and Q and η_o for that speed. The same procedure is repeated by running the pump at another speed. The curves so obtained are the main Characteristic curves.

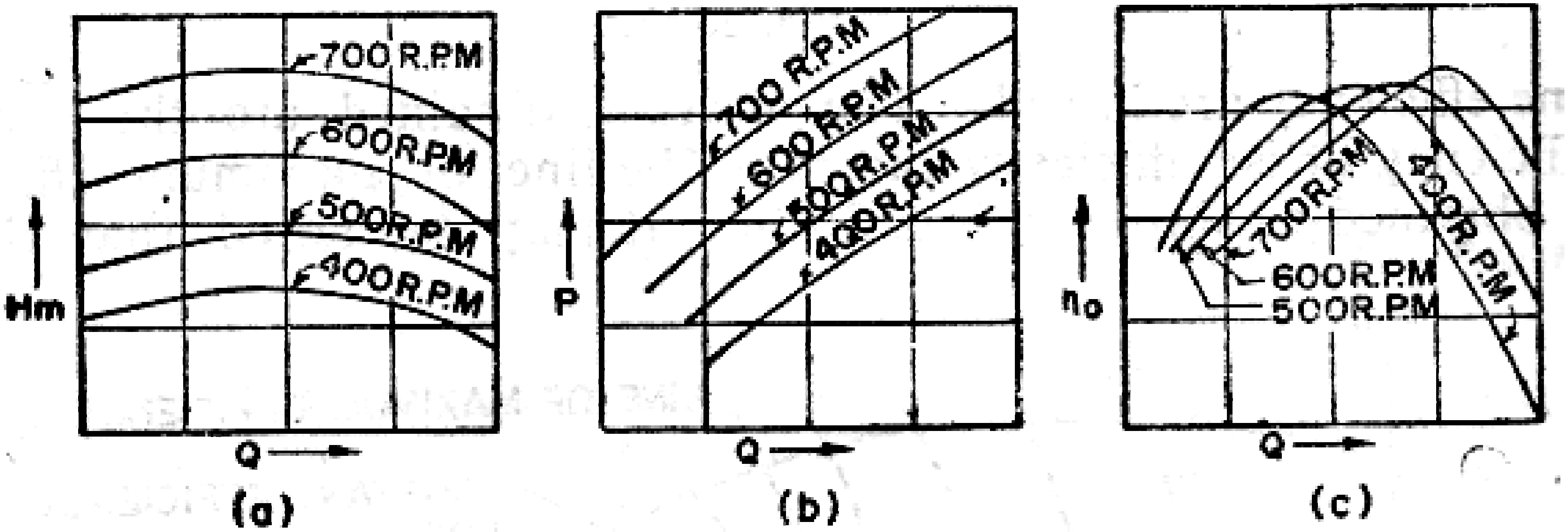
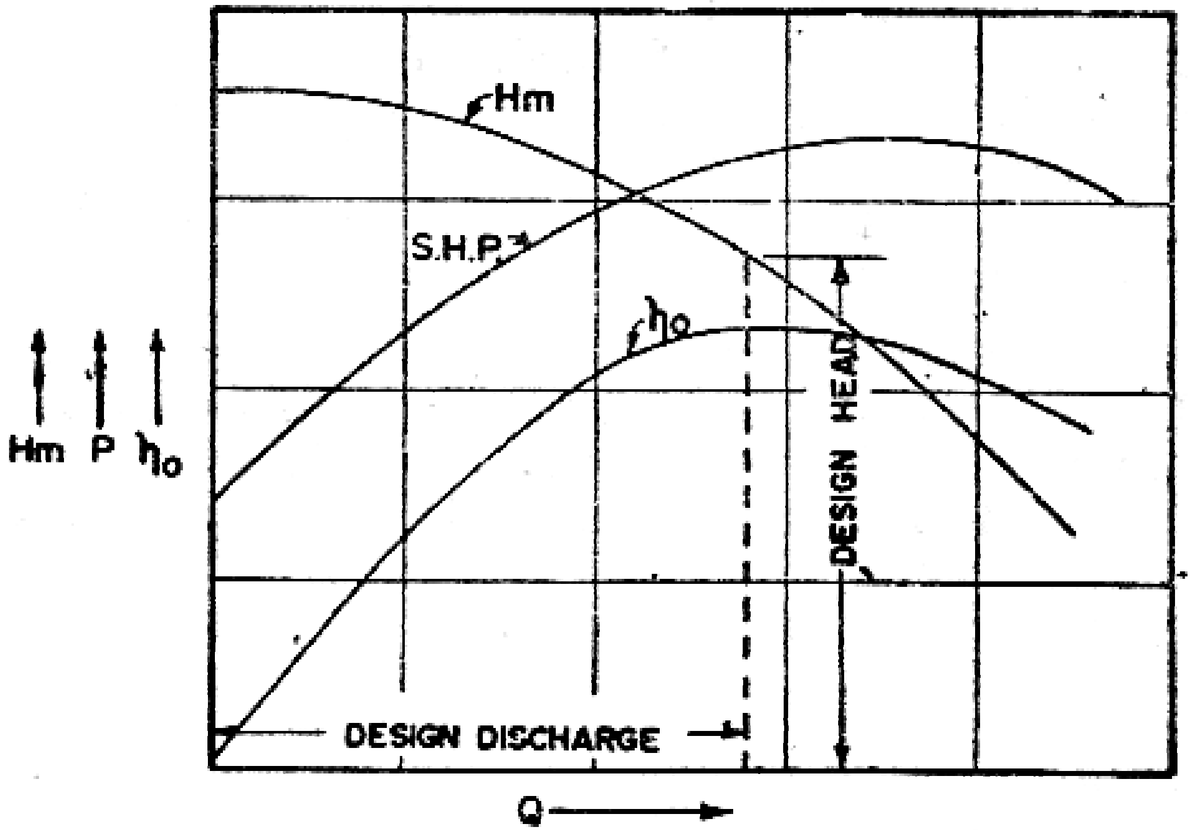


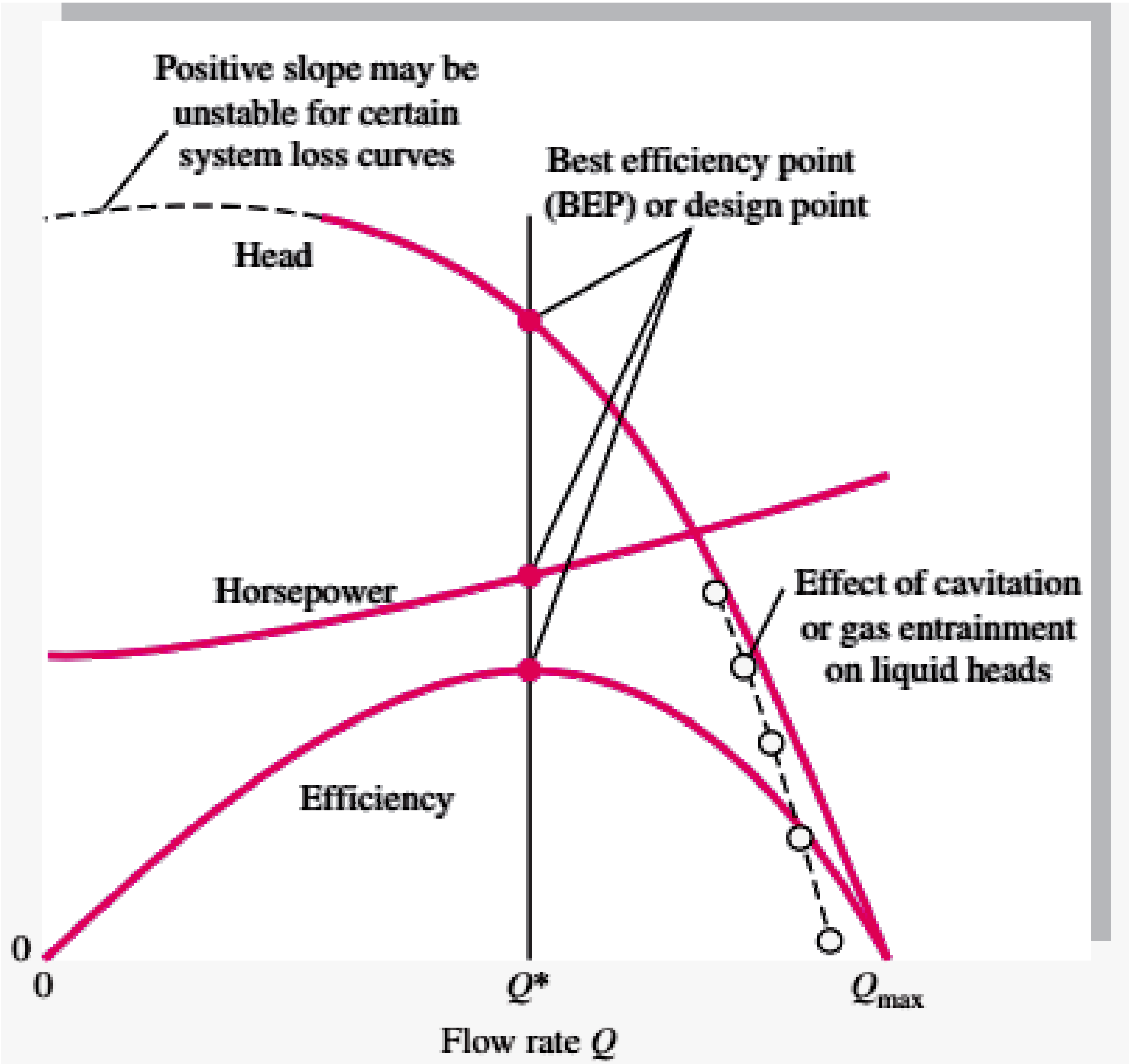
Fig: Main Characteristic Curves

(2) Operating characteristic curves:

The operating characteristic curves are obtained **by running the pump** **at the design speed of the driving motor**. In fact, the operating characteristic curve and the main characteristic curve are identical when the pump is run at the design speed. The design discharge and head are obtained from the corresponding curve, where the efficiency is maximum shown by dotted line in fig.



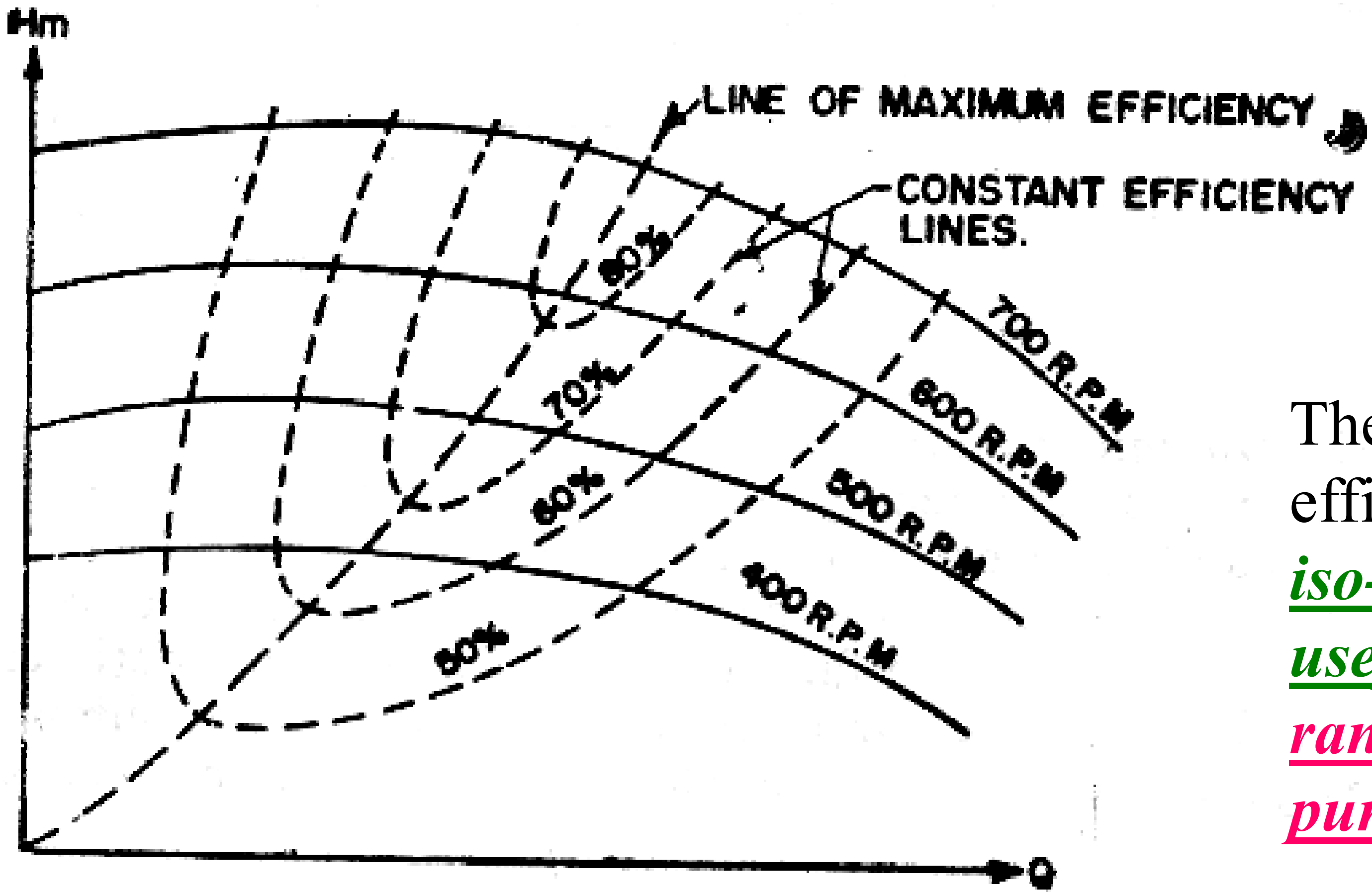
OPERATING CHARACTERISTIC CURVES



Positive slope may be unstable for certain system loss curves
Best efficiency point (BEP) or design point
Head
Horsepower
Effect of cavitation or gas entrainment on liquid heads
Efficiency
0
0
Q^*
Q_{max}
Flow rate Q

(3) Constant efficiency curves:

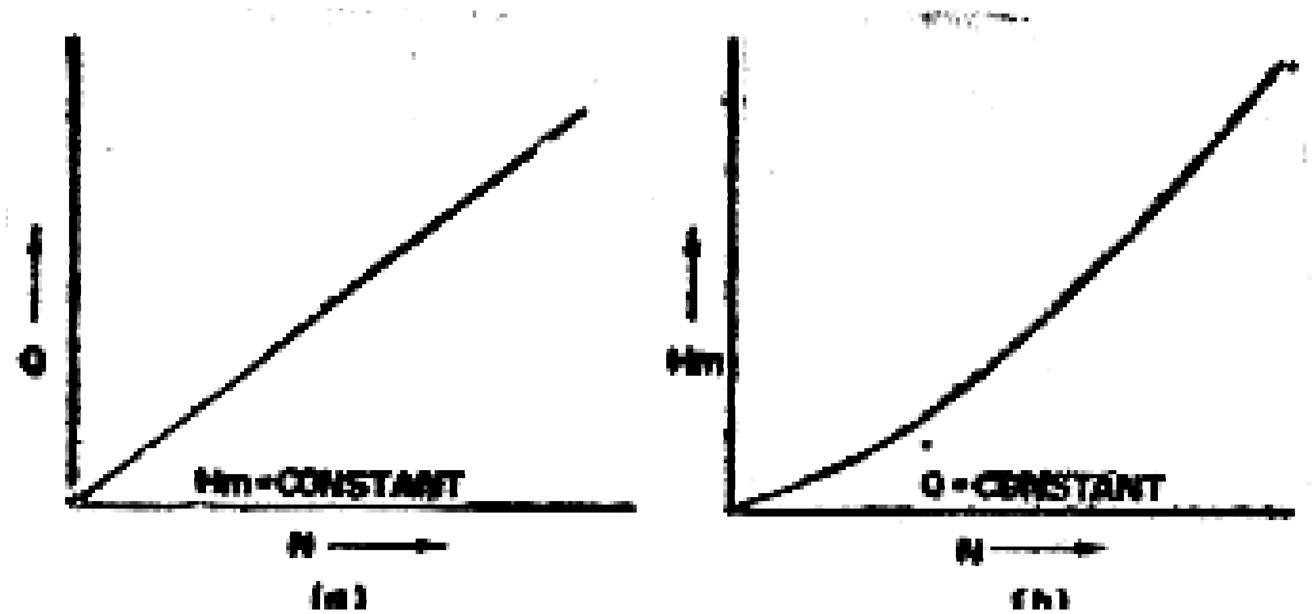
The constant efficiency curves are **obtained from the main characteristic curves**. For a given efficiency, the values of discharge are obtained. These points are projected on the H_m versus Q curve for that speed. Likewise, for another value of efficiency and speed, the points are obtained and projected. The points corresponding to one efficiency are joined. The curves so obtained are the constant efficiency or **iso-efficiency curves**.



The line of maximum efficiency is obtained. ***The iso-efficiency curves are useful in determining the range of operation of a pump at a given efficiency***.

(4) Constant head and constant discharge curves:

If the pump has a variable speed, the plots between Q and N, and that between H_m & N are obtained. In the first case, the manometric head is kept constant and in the second case, the discharge is kept constant shown in the two curves.



Net Positive Suction Head (NPSH):

The net positive suction head (NPSH) is defined as the net head in metres of liquid that is required to make the liquid flow through the suction pipe from the sump to the impeller. Obviously, **it is equal to the barometric head minus the sum of static suction head, vapour pressure head, friction head loss and the velocity head in the suction pipe**.

$$\boxed{\mathrm{NPSH} = \frac{p_{atom}}{\gamma} - \left(\frac{p_v}{\gamma} + h_s + h_{fs} + \frac{V_s^2}{2g}\right)}$$

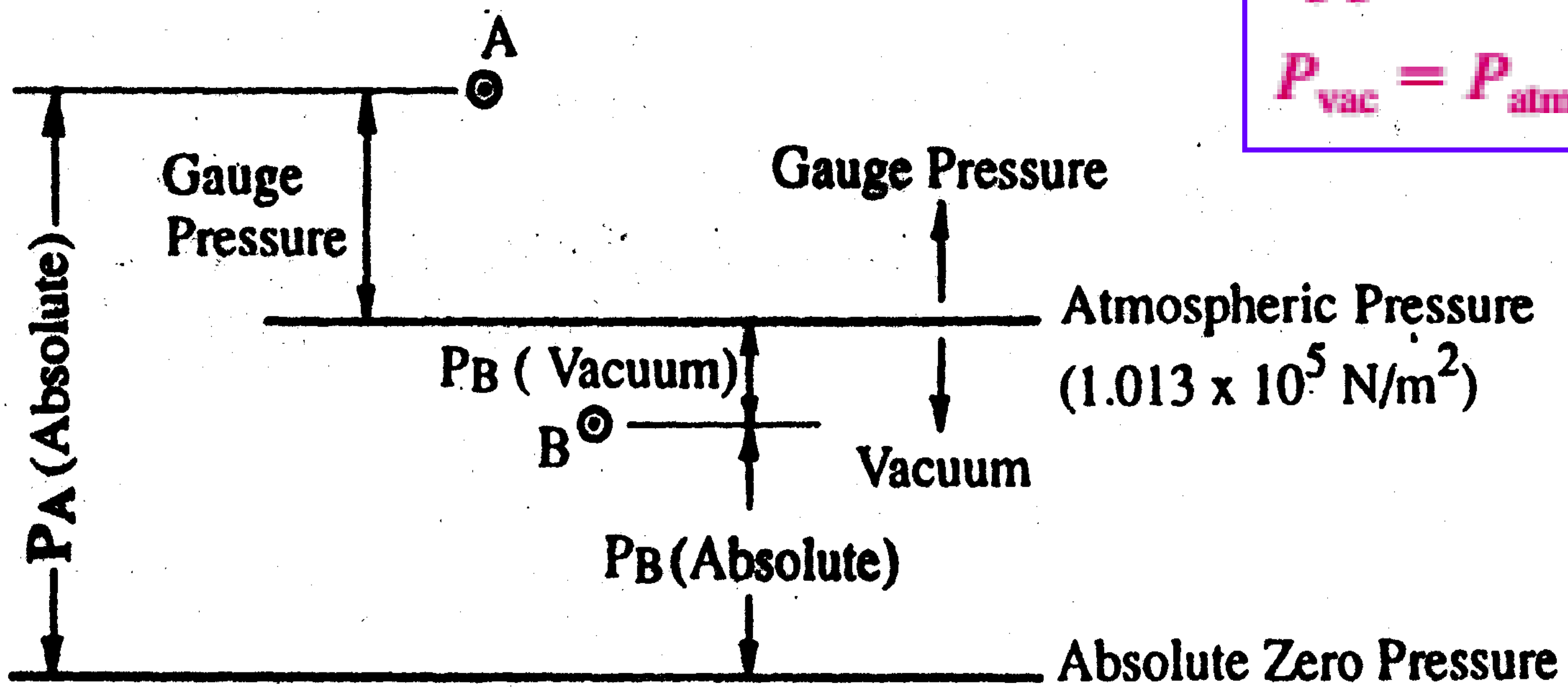
The above equation is applicable when the sump is at lower level than the pump axis. If it is at higher level, the sign of '$\mathbf{h_s}$' will be negative. Moreover. ***it has been assumed that the sump is open to atmosphere***. **If it is not so, the term P_{atm}/γ will be accordingly changed**.

It is also defined as the gauge reading in metres of liquid, taken on the suction side referred to the pump axis, minus the sum of the vapour pressure in the metres of liquid corresponding to the temperature and altitude of the place and the velocity head at that point. Thus

Net Positive Suction Head or NPSH for pumps can be defined as the difference between liquid pressure at pump suction and liquid vapor pressure, expressed in terms of height of liquid column.

$$P_{gage} = P_{abs} - P_{atm}$$

$$P_{vac} = P_{atm} - P_{abs}$$



It may be assumed that the separation occurs at an absolute pressure of 24.525 kN/m^2 (1.03 kgf/cm^2) .

$$\text{NPSH} = \text{Suction gauge reading in metres of liquid} - \left(\frac{p_v}{\gamma} + \frac{V_s^2}{2g} \right)$$

Net Positive-suction Head (NPSH), ***is the head required at the pump inlet to keep the liquid from cavitating or boiling***. It is defined as **the available total suction head at the pump inlet above the head corresponding to the vapour pressure at that temperature**. The pump inlet or suction side is the low-pressure point where cavitation will first occur. The NPSH is defined as

$$NPSH = \frac{P_s}{\gamma} + \frac{V_s^2}{2g} - \frac{P_v}{\gamma}$$

$$NPSH = \frac{P_a}{\gamma} - \frac{P_v}{\gamma} - Z - h_{fs}$$

Thoma cavitation parameter is defined by **NPSH = σH$_m$**

$$\sigma = \frac{(NPSH)}{H} = \frac{(P_A/\gamma) - (P_v/\gamma) - Z - h_{fs}}{H}$$

At cavitation conditions,

$$\sigma = \sigma_c \quad \text{and} \quad \frac{P_s}{\gamma} = \frac{P_v}{\gamma}$$

$$\sigma_c = \frac{(P_a/\gamma) - (P_v/\gamma) - Z - h_{fs}}{H}$$

One correlation for **critical cavitation parameter** for pumps is given as

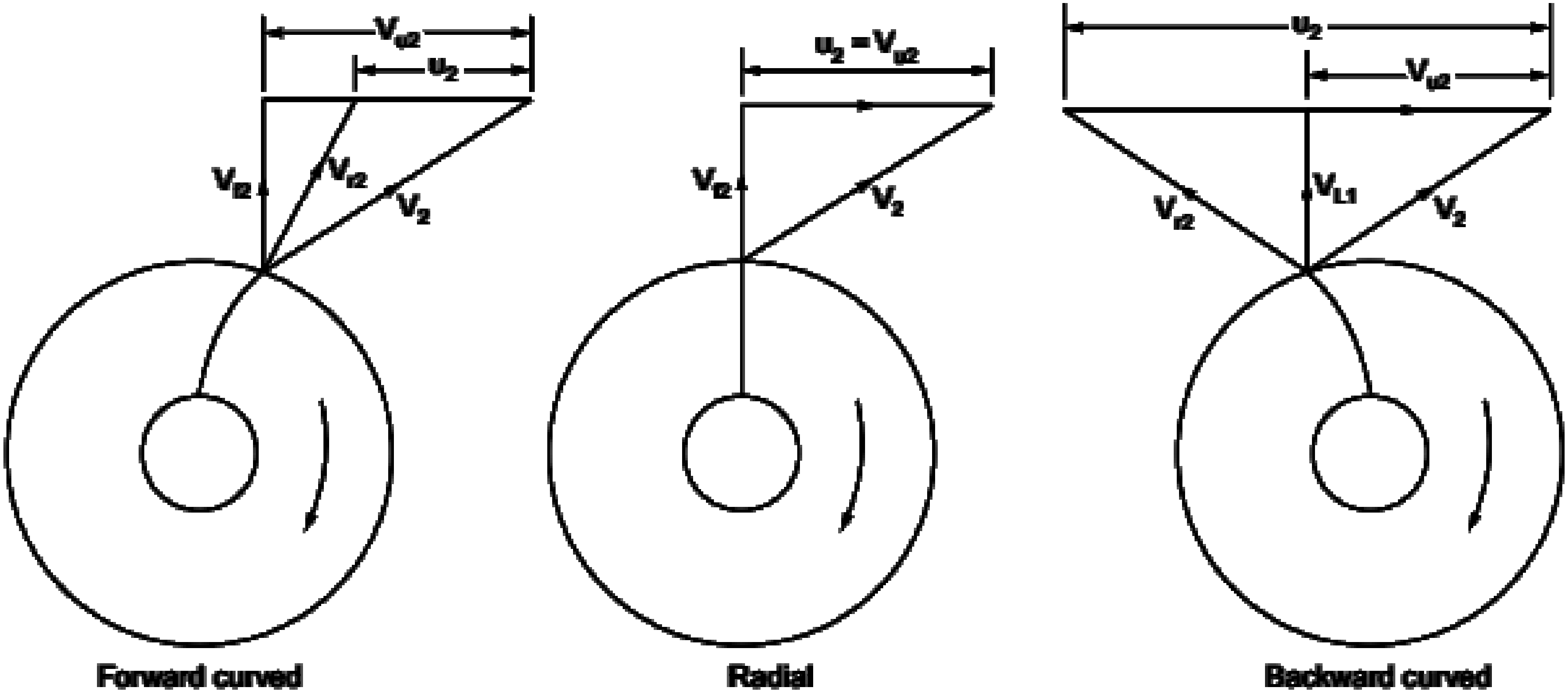
$$\boxed{\sigma_c = \left(\frac{n_s}{175}\right)^{4/3}}$$

The minimum NPSH depends upon the pump used design, its speed and the discharge. Its value is given by the manufacturer. In order to avoid cavitation, the available NPSH should be equal to or greater than the minimum NPSH. NPSH also depends upon the size of the suction pipe. If the diameter of the suction pipe is different from that quoted by the manufacturer, the minimum NPSH should be accordingly modified.

Effect of Outlet Blade Angle

There are **three possible orientation of the blade at the outlet**. These are: ***forward curved***, ***radial*** and ***backward curved*** arrangements. The velocity triangles for the three arrangements are shown in Fig. In the case of **forward curved blading** $V_{w1} > u_1$ and V_1 **is larger** comparatively. In the case of **radial blades**, $V_{w1} = u_1$. In the case of **backward curved blading**,

$V_{w1} < u_{1}$.



Theoretical Water horsepower is $P_w = T\omega = M(V_{w1}u_1 - V_w u)$

$$= \rho Q(V_{w1}u_1 - V_w u)$$

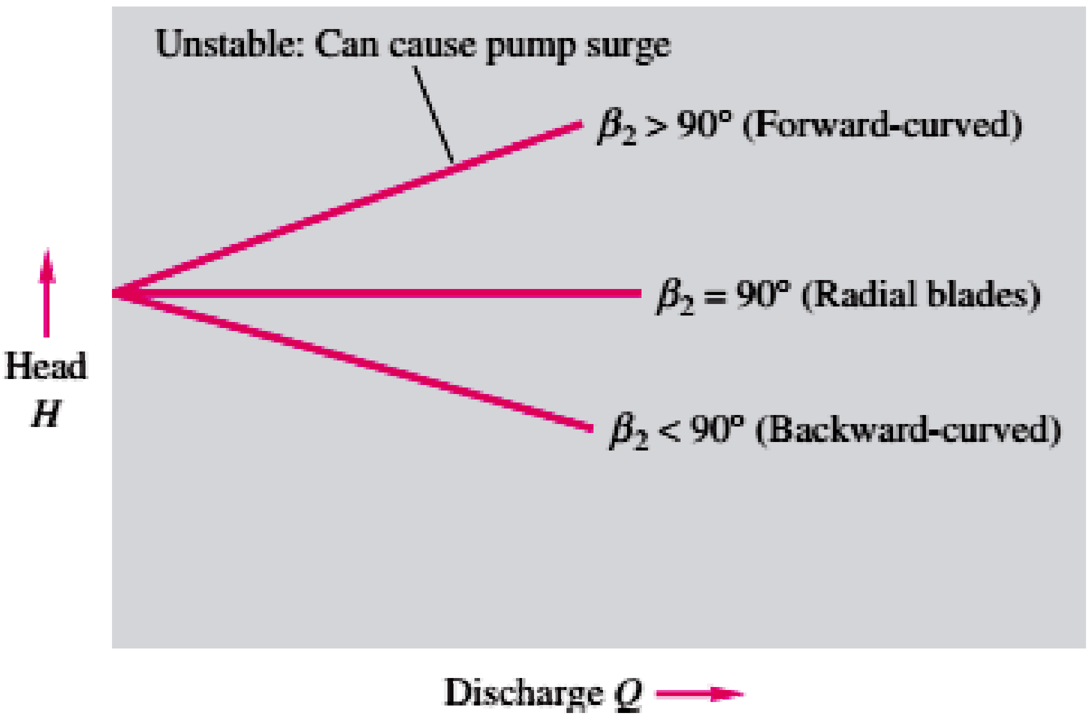
$$H = P_w / \rho g Q = (V_{w1}u_1 - V_w u)/g$$

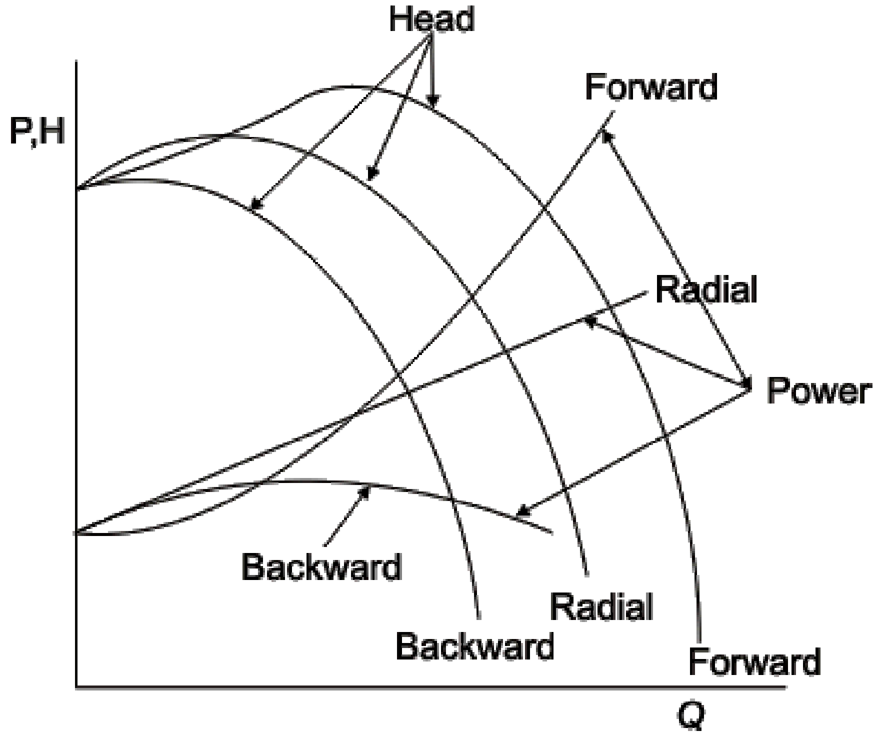
For radial flow at inlet, $V_w u = 0$

$V_{w1} = u_1 - V_{f1}\cot\phi$ **and** $V_{f1} = Q / (\pi k D_1 b_1)$

$$H = V_{w1}u_1/g = [(u_1 - V_{f1}\cot\phi)\, u_1] / g = u_1^2/g - [(u_1 \cot\phi)Q]/(\pi k D_1 b_1 g)$$

The head varies linearly with discharge Q, having a shutoff value u^2_1/g, where u_1 is the exit blade-tip speed. The slope is negative if **ϕ < 90°** (backward-curved blades) and positive for **ϕ > 90°** (forward-curved blades). This effect is shown in Fig. and is accurate only at low flow rates.





<u>The positive slope condition in above Fig. can be</u> ***<u>unstable and can cause pump surge, an oscillatory condition</u>*** where the pump "hunts" for the proper operating point. ***<u>Surge may cause only rough operation in a liquid pump, but it can be a major problem in gas compressor operation</u>***. For this reason a backward-curved or radial blade design is generally preferred.