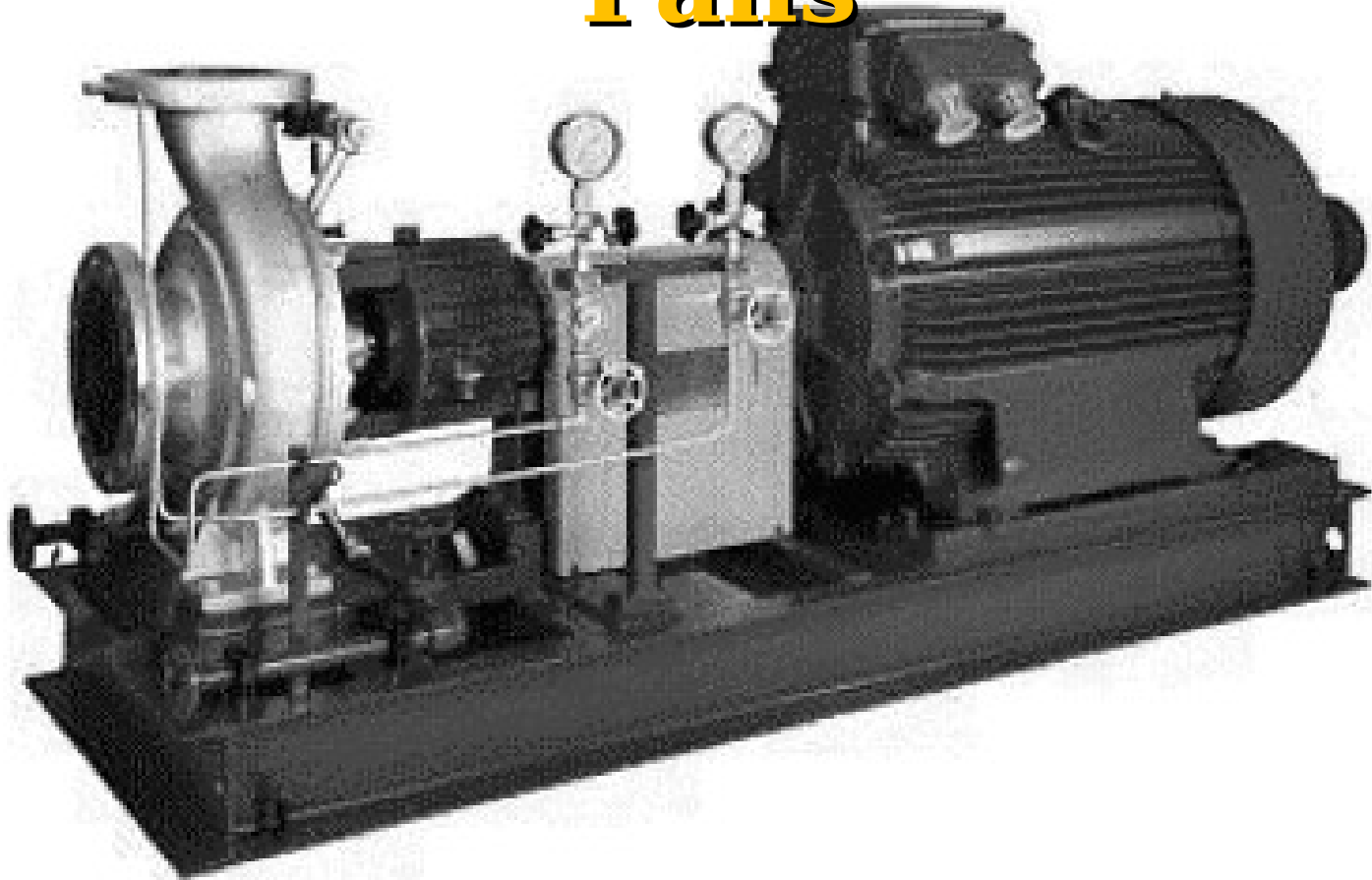


Chapter 2

Centrifugal Pumps and Fans



Pump assembly

- Pump is used to raise the pressure of a liquid (water, chemicals, petroleum,...)
- If a turbomachine is used to raise the pressure of a gas it is called a **Compressor** or a Blower or a Fan, depending upon the pressure developed.
- The classification of work absorbing machines is as shown in the figure below.

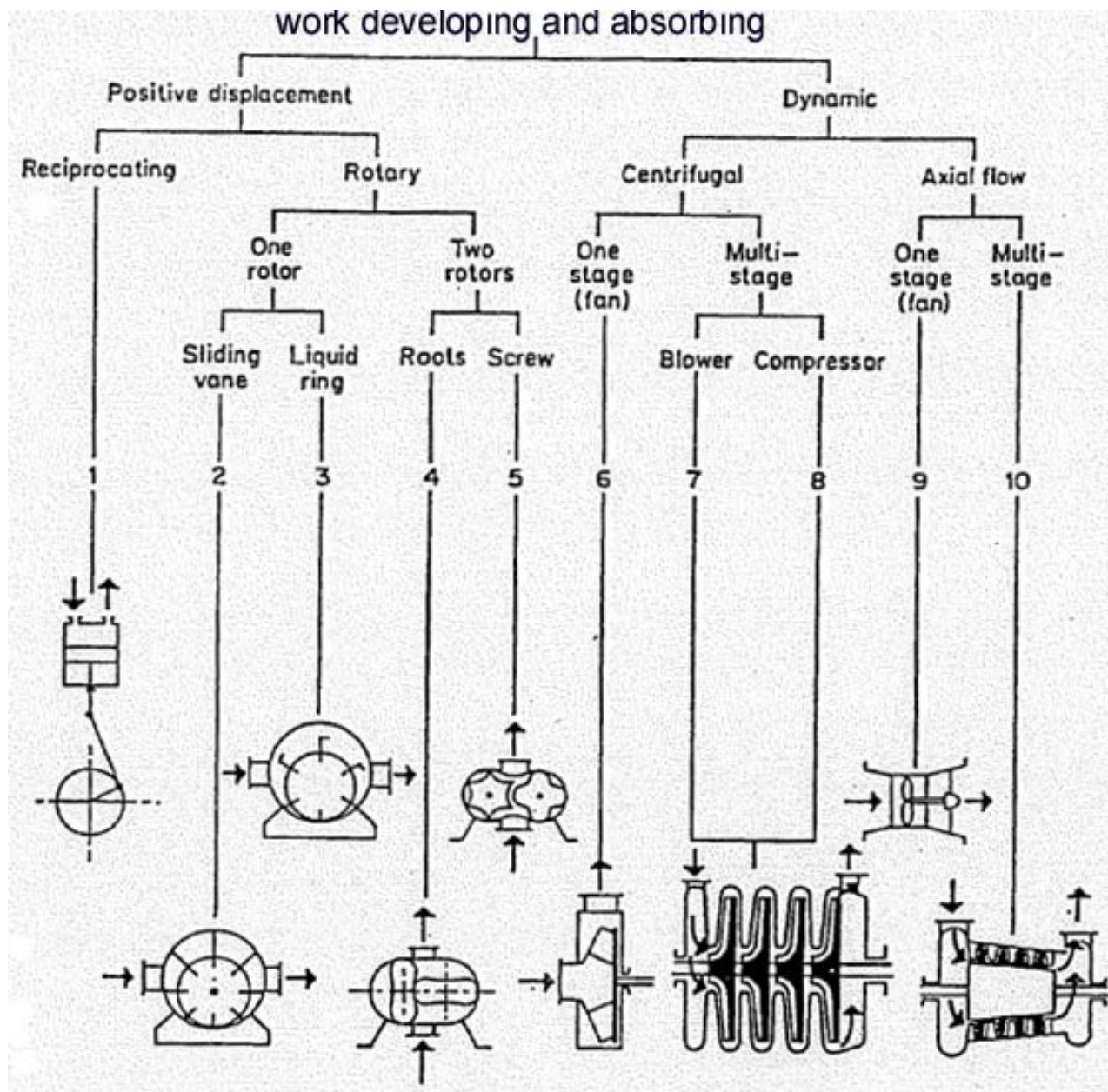
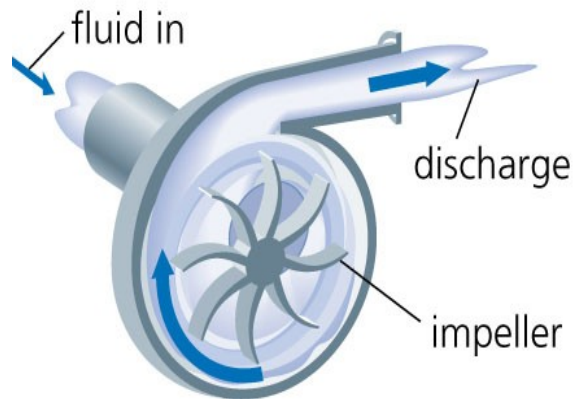
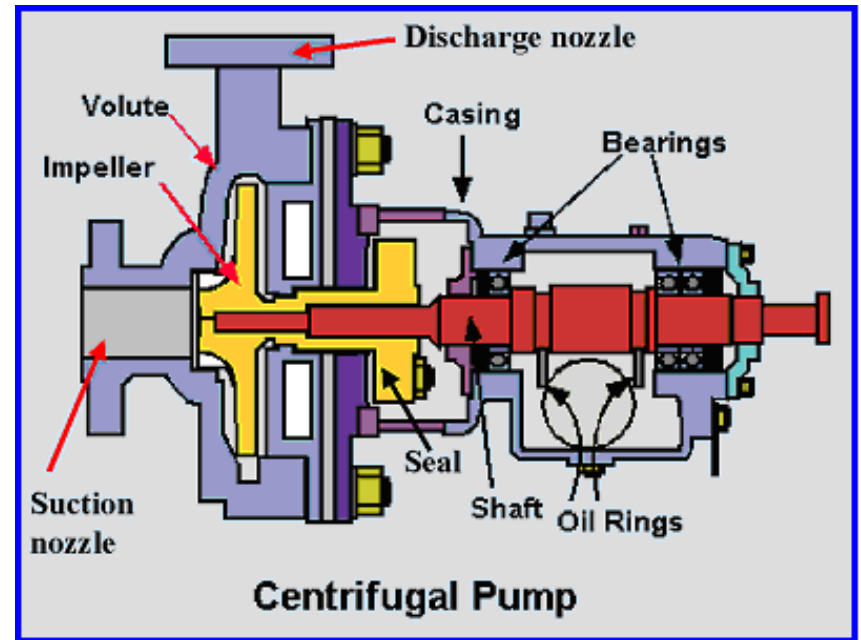
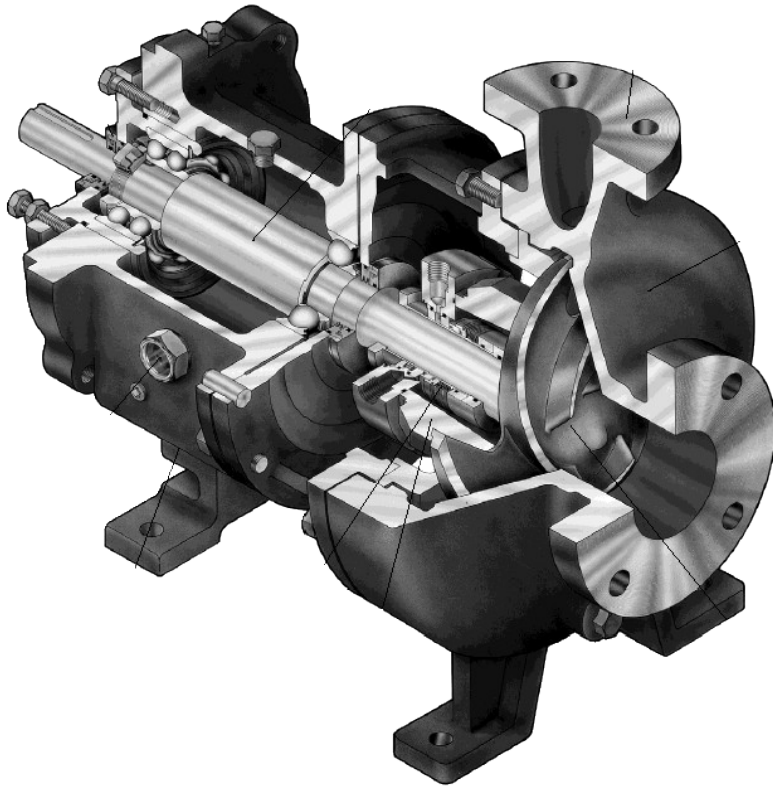


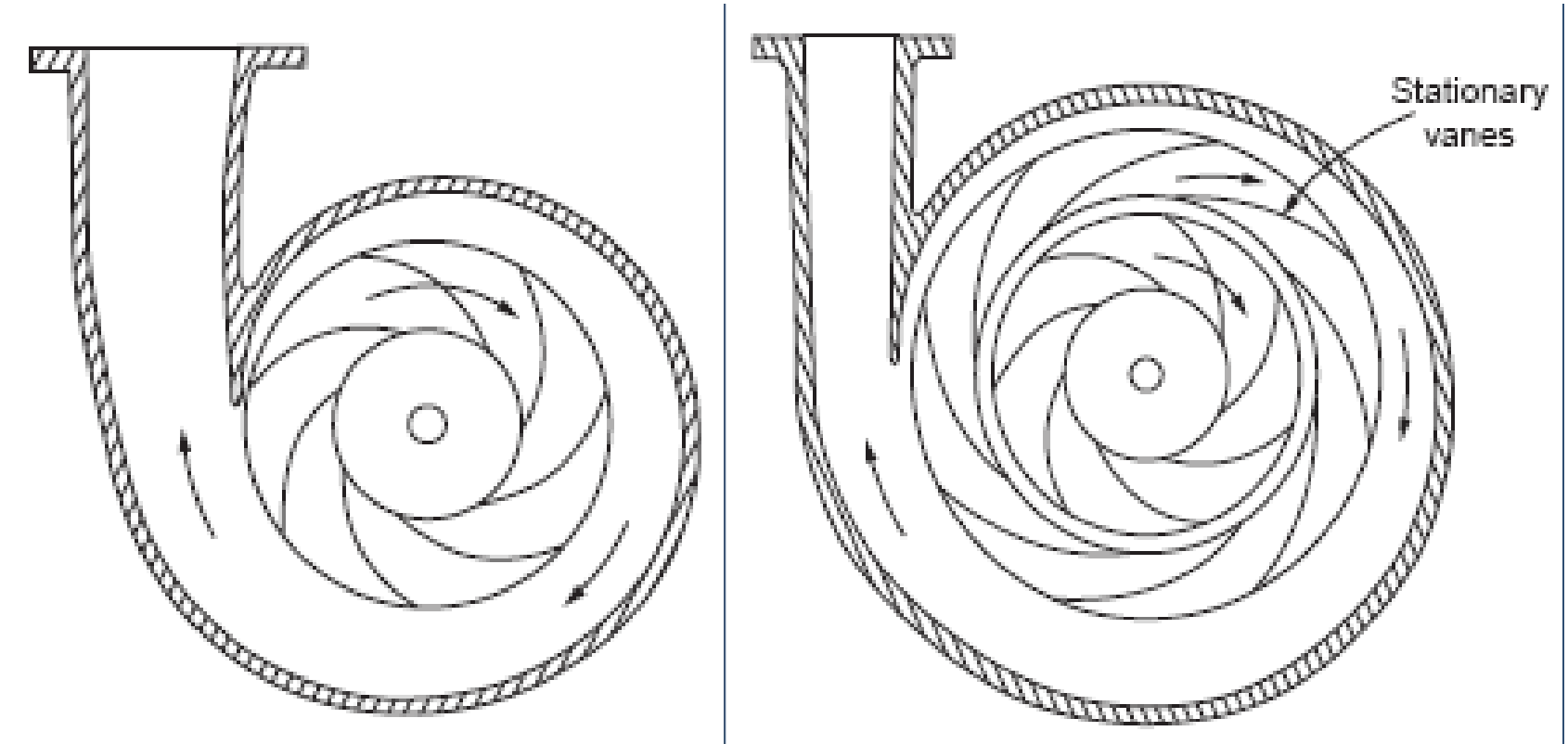
Fig. 2-1 Classification of work absorbing machines

Centrifugal Pumps



Classification of Centrifugal Pumps

1.By the type of energy conversion,



Volute casing

Volute casing with guide blades

2. By the number of stages,

- Single stage
- Multi-stage

3. By impeller type, 4. By the axis of rotation,

- | | |
|-------------|--------------|
| • open | • Horizontal |
| • Semi-open | • Vertical |
| • Closed, | |

5. By the method of drive,

- Direct (rigid or flexible couplings)
- Indirect (gear drive, belt drive)

Centrifugal Pumps

- The centrifugal pump is a **roto-dynamic type** pump and it works on the principle of forced vortex flow.
- When a **certain mass of fluid** is rotated by an ***external torque***, the rise in pressure head of liquid takes place. The rise in velocity head (H_v) at any point of the rotating liquid is proportional to the square of tangential velocity (U) of the liquid at that point.

$$\text{The velocity head } (H_v) = \frac{U^2}{2g} = \frac{\omega^2 r^2}{2g}$$

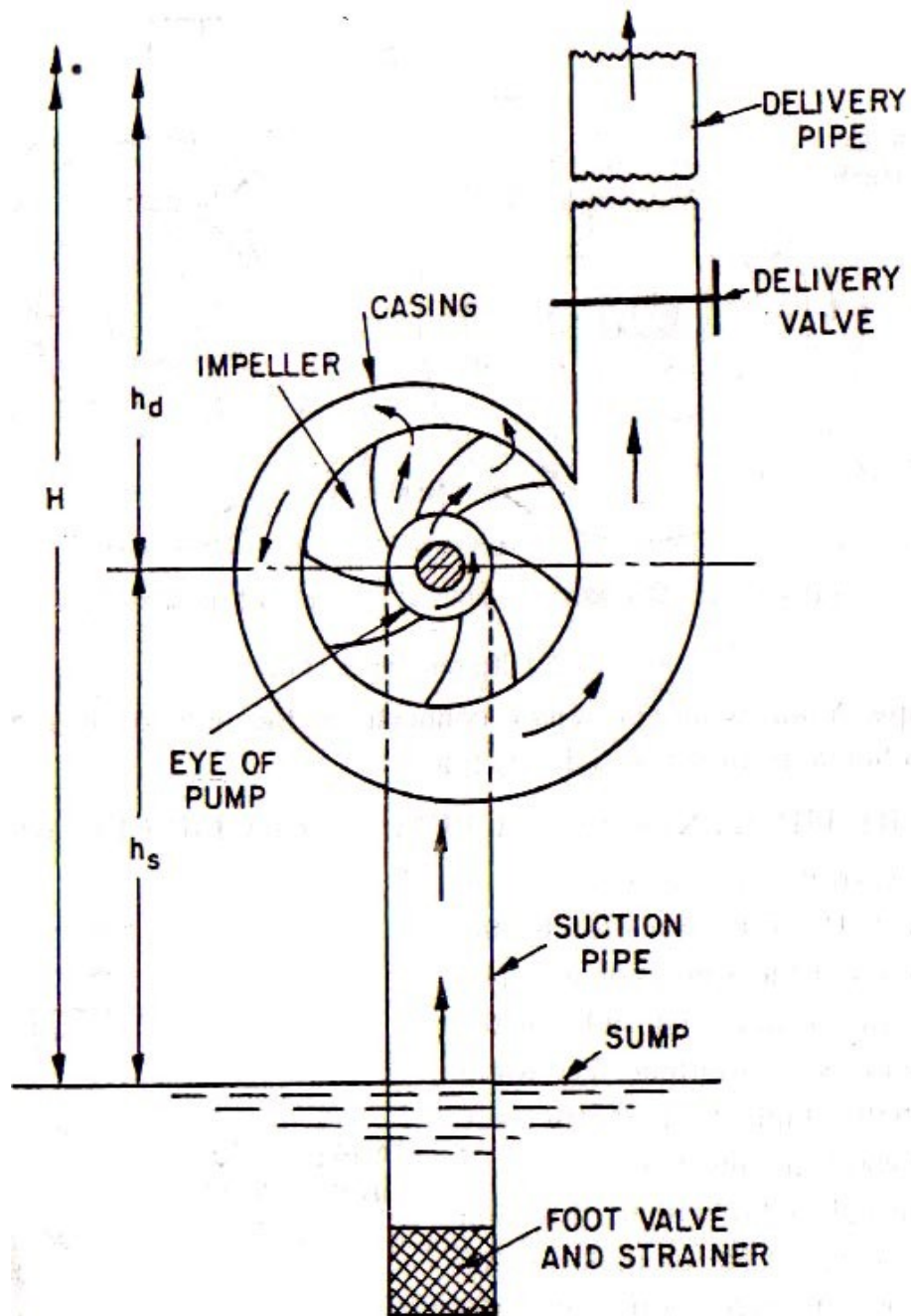
- At the outlet of the impeller where radius is more, the rise in velocity head (H_v) is more. The so obtained velocity head is then converted into pressure head (H_p) as the liquid passes through a gradually increasing cross-sectional area of the casing (volute).

Main component parts of a Centrifugal Pump

1. Impeller

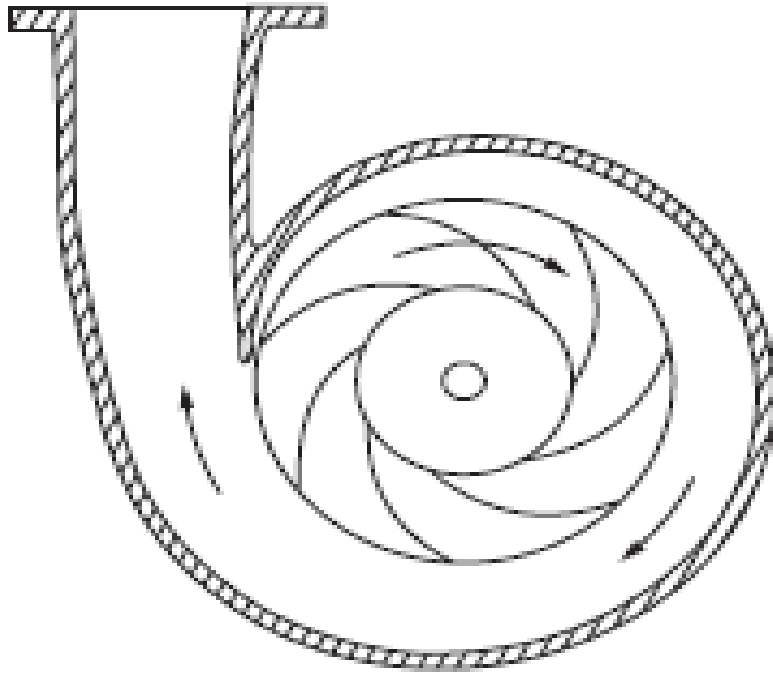
2. Casing

3. Shaft

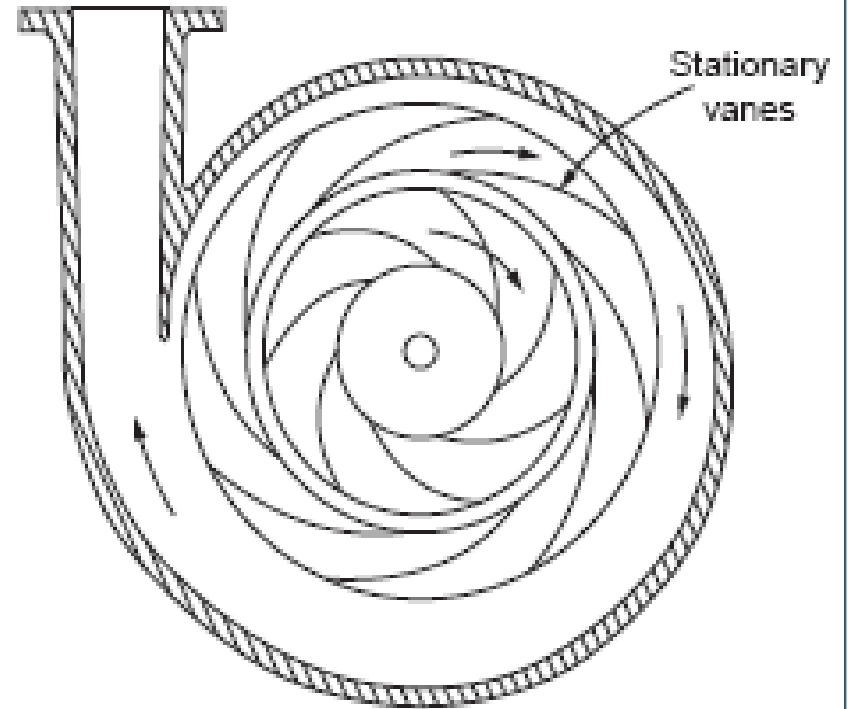


1.Impeller:- it is a rotating part of a centrifugal pump. It consists of a series of backward curved vanes and is mounted on a shaft which is connected to the shaft of a prime mover (Electric motor, IC engine, ...).

2.Casing:- The casing (volute) of a centrifugal pump is similar to the casing of a reaction turbine. It is an air-tight passage where liquid surrounds the impeller and is designed in such a way that the velocity head (H_v) or kinetic energy of the water discharged at the outlet of the impeller is converted into pressure head (H_p) or pressure energy before the water leaves the casing and enters the



Volute casing



Volute casing with
guide blades

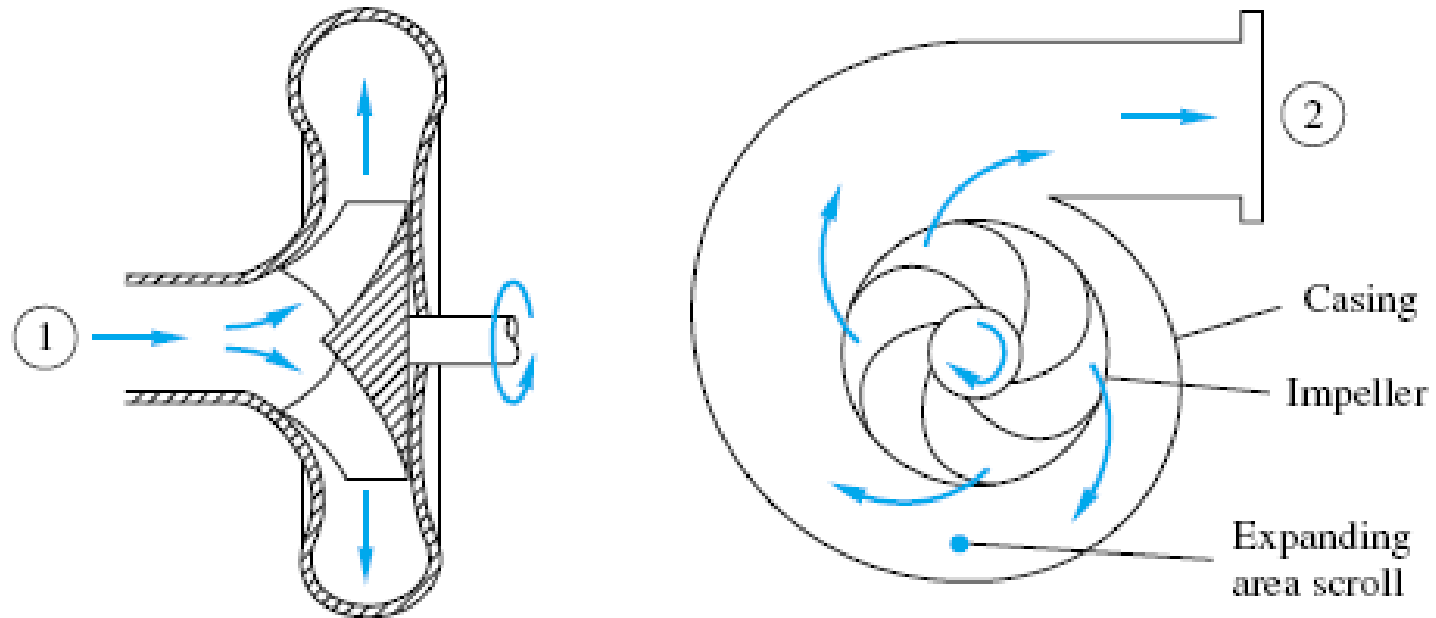
3. **Suction pipe with a Foot-valve and a Strainer**:- A pipe whose one end is connected to the inlet of the pump and other dips into water in a sump is known as a suction pipe. A foot valve which is a **non-return valve** or **one-way valve** is fitted at the lower end of the suction pipe.

4. **Delivery pipe**:- a pipe whose one end is connected to the outlet of the pump and other end delivers the water at a required height is known as delivery pipe.

Principle of operation

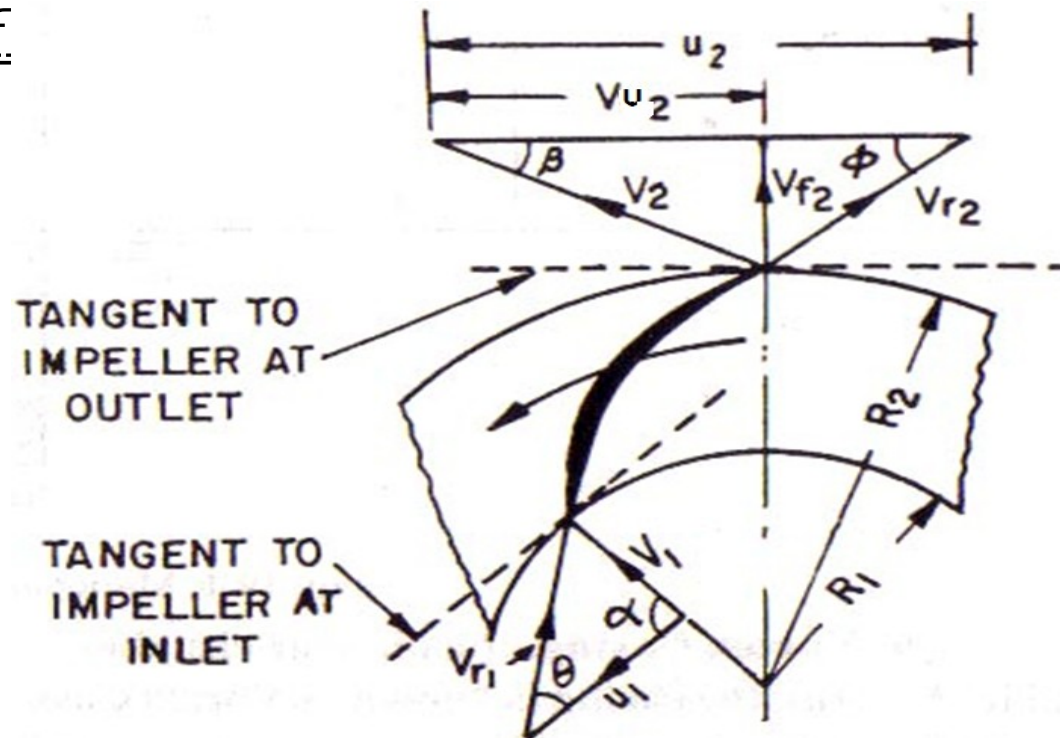
➤ As shown in the figure, **fluid enters axially** through the eye of the casing and is caught up in the impeller blades, where it is whirled tangentially and **radially outward** until it leaves the impeller. In doing so, the fluid velocity head (H_v) keeps on increasing.

- When the fluid passes through the gradually increasing cross-sectional area (casing or volute), the velocity head (H_v) keeps on converting into pressures head (H_p).
- As a result, a high pressure liquid is obtained in



Energy transfer by the by the impeller to the fluid (water)

- In the case of centrifugal pumps, work is done by the impeller on the fluid (water). The expression for work done by the impeller on the water is obtained by drawing
- In the design of such pumps, it is better to consider a radial f or best efficiency



- Tangential velocities at the inlet (U_1) and at the outlet (U_2) is given by, $U_1 = \frac{\pi D_1 N}{60}$ and $U_2 = \frac{\pi D_2 N}{60}$

- A centrifugal pump is the reverse of a radially inward flow reaction turbine and the power (P_t) imparted by the water on the runner from **Euler's Equation** is given by,

$$P_t = \dot{m}(V_{u1} \times U_1 - V_{u2} \times U_2)$$

- But, for pumps it given by,
- $$P_t = \dot{m}(V_{u2} \times U_2 - V_{u1} \times U_1)$$

➤ The Euler's turbomachine equations are

Shaft torque: $T_{\text{shaft}} = \rho Q(r_2 V_{t2} - r_1 V_{t1})$

Water horsepower: $P_w = \omega \cdot T_{\text{shaft}} = \rho Q(u_2 V_{t2} - u_1 V_{t1})$

Pump head: $H = P_w / \rho g Q = (u_2 V_{t2} - u_1 V_{t1}) / g$

- If the flow is radial at inlet, $V_{1t} = 0$.

Hence, $P_t = \dot{m}(V_{u2} \times U_2)$

Flow rate is given by,

$$Q = (\pi D_1 - nt) b_1 V_{1r} = (\pi D_2 - nt) b_2 V_{2r}$$

where n is number of impellers on the runner

t is thickness of each impeller

Pressure developed by the impeller

The head against which a centrifugal pump has to work is called manometric head (H_m) and is given by:

$$1. H_m = \text{Pump total outlet head} - \text{Pump total inlet head}$$

$$= \left(\frac{p_o}{\rho g} + \frac{V_o^2}{2g} + Z_o \right) - \left(\frac{p_i}{\rho g} + \frac{V_i^2}{2g} + Z_i \right)$$

$$= \left[\left(\frac{P_o - P_i}{\rho g} \right) + \left(\frac{V_o^2 - V_i^2}{2g} \right) + (Z_o - Z_i) \right] \approx \frac{P_o - P_i}{\rho g}$$

$$2. H_m = H_s + \sum H_{\text{losses}} \quad \text{where } H_s = h_d + h_s \text{ is the static head}$$

$$= H_s + h_{li} + h_{fi} + h_{lo} + h_{fo}$$

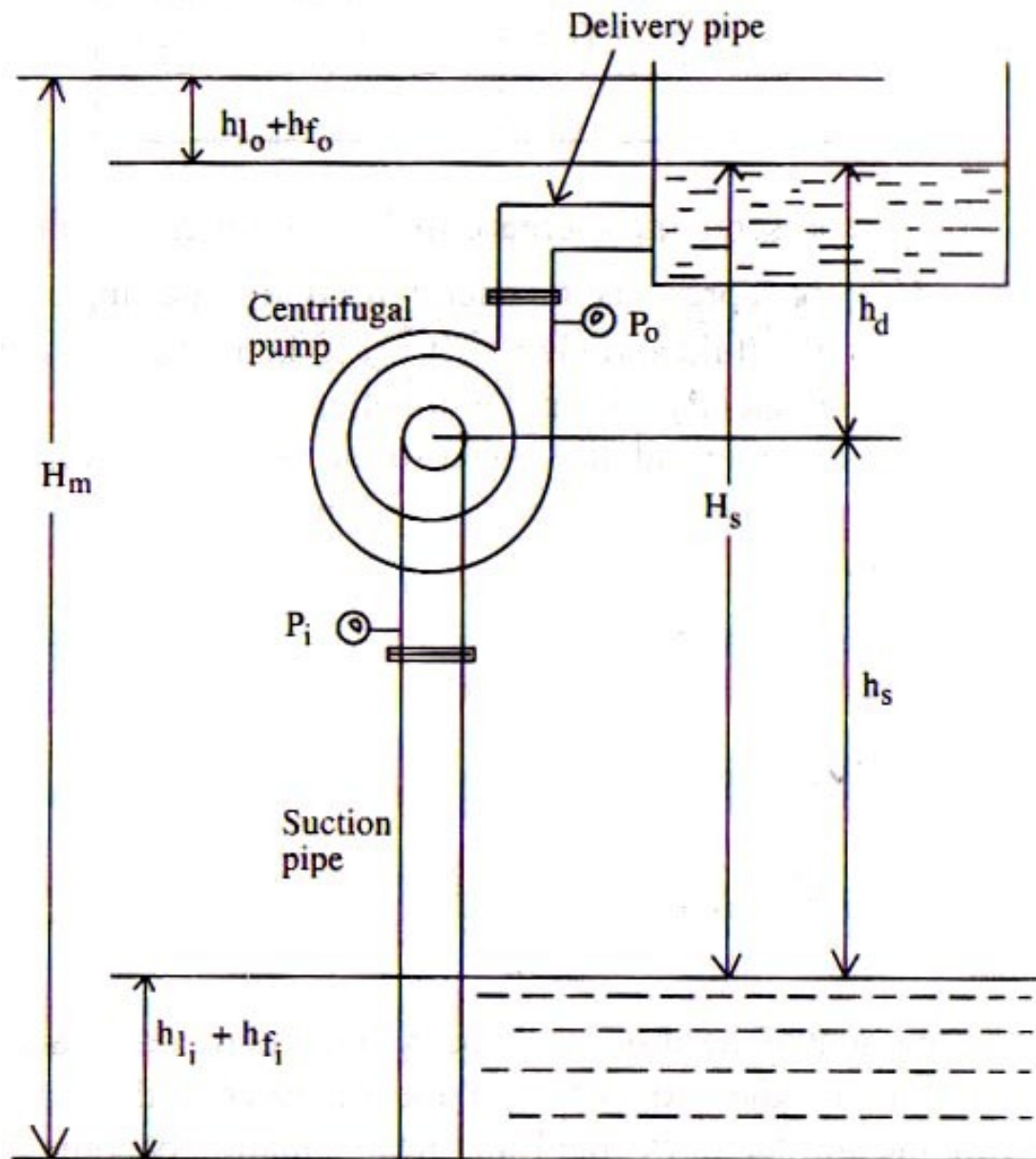
where h_l and h_f are major and minor losses.

$$3. H_m = \text{Euler head} - \text{Losses in the impeller and casing}$$

$$= H_E - H_L$$

$$= \frac{V_{2t} U_2}{g} - H_L \quad \text{if inlet flow velocity is radial}$$

Major and Minor Losses.



Pump Losses

➤ The shaft power (P_s) supplied to the pump by the prime mover is not the same as the energy received by the liquid. This difference is mainly due to the following losses:

1. **Impeller friction losses** :- are losses due to friction between the fluid and the impeller surfaces.
2. **Shock losses**:- are losses due to the mismatch of the fluid and metal angles at the inlet side of the impeller.
3. **Recirculation losses**:- the pressure difference in the impeller eye and tip can cause a recirculation of a small volume of liquid, thus reducing the energy to be transferred.
4. **Leakage** :- is due to the loss of liquid in-between the impeller and the pump.
5. **Volute friction losses**:- this loss is as a result of a mismatch of the velocity leaving the impeller and

Due to losses, shaft power supplied by prime mover is not the same as the energy received by the liquid.

□ Pipe losses

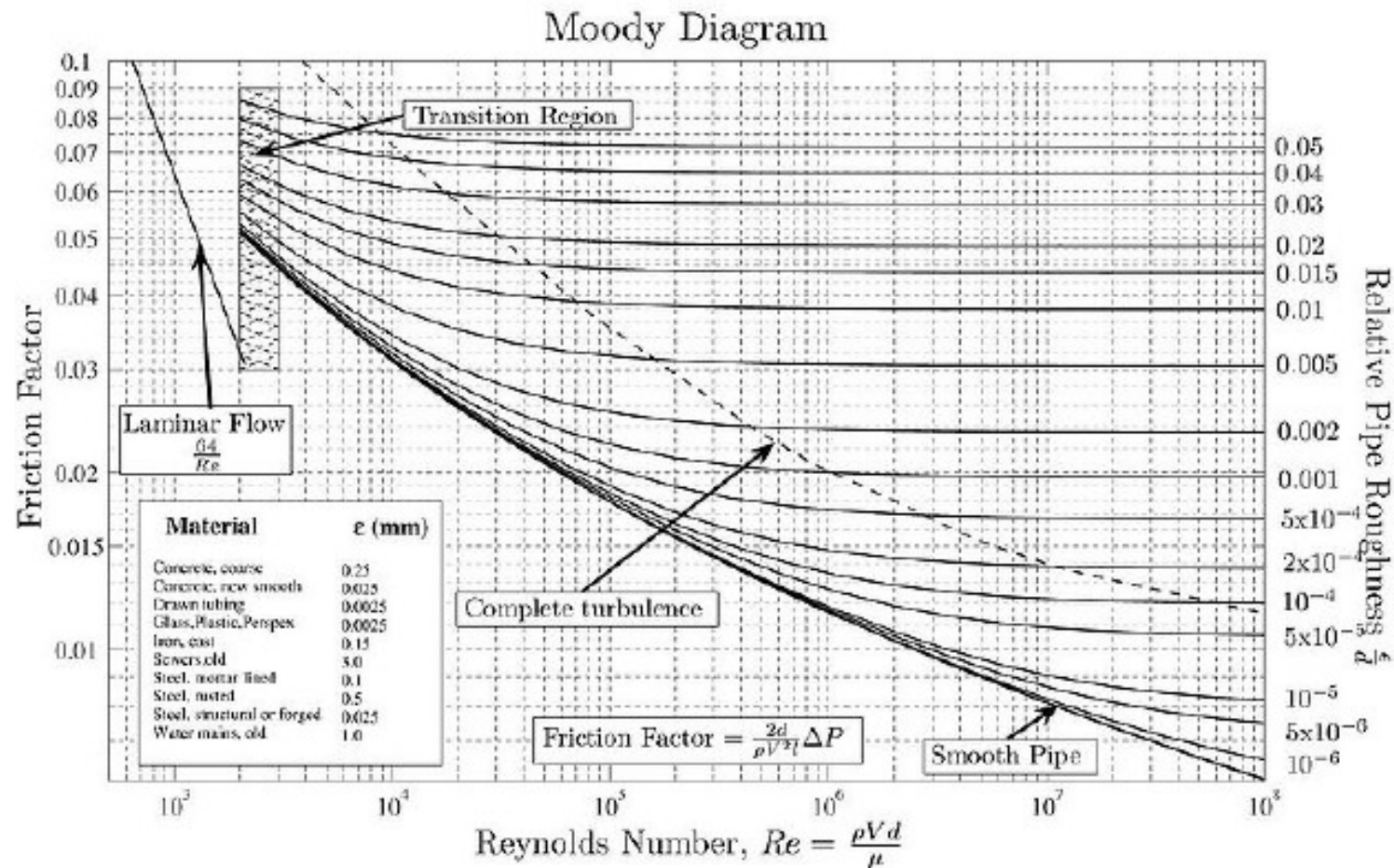
- ❖ *Friction loss*: due to b/n rotating face of the impeller & the liquid

Give $h_f = f \frac{l v^2}{D 2g}$, most known Darcy -Weisbach equation

f , is friction factor which can be read from the Moody diagram for different pipe relative roughness (e/d) and Reynolds number for turbulent flow. For laminar flow an approximate value can be found by using $f = 64/R_e$

- ❖ *Elbow loss*
- ❖ *Cross section change loss*

Moody diagram for flow in pipes



Pump Efficiencies

1. Mechanical efficiency (η_m) 3. Volumetric efficiency (η_v)

$$\eta_m = \frac{\text{Power supplied to the fluid by the impeller}}{\text{Shaft power}}$$

$$= \frac{P_i}{P_s} = \frac{\dot{m}(V_{U2}U_2)}{P_s}$$

$$\eta_v = \frac{\text{Flow rate at the pump outlet}}{\text{Flow rate through the impeller}}$$

$$= \frac{\dot{Q}}{\dot{Q} + \Delta \dot{Q}}$$

2. Manometric efficiency (η_H)

$$\eta_H = \frac{\text{Actual head developed by the pump}}{\text{Theoretical head developed by the impeller}}$$

$$= \frac{H_m}{H_E} = \frac{H_m}{\left(\frac{V_{U2}U_2}{g} \right)}$$

4. Overall efficiency (η_o)

$$\eta_o = \frac{\text{Output fluid or liquid Power (} P_f \text{)}}{\text{Shaft power (} P_s \text{)}}$$

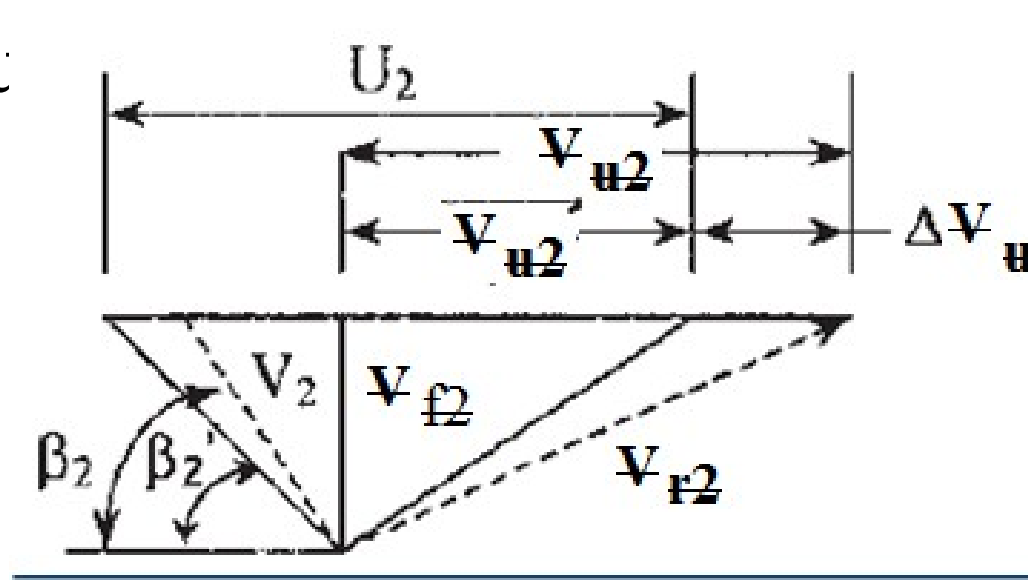
$$= \frac{\rho g \dot{Q} H_m}{T \omega} = \eta_m \eta_H \eta_v \rightarrow P_s = T \omega$$

Slip factor (σ_s)

- In the actual flow through the impeller, the pressure at the leading edge is higher than the trailing edge of the impeller, that means the velocity is lower at the leading edge and higher at the trailing edge.
- This leads to a non-uniform velocity distribution. As a result, the fluid is discharged from the impeller at an angle relative to the impeller less than the vane angle.
- The result is a reduction in the exit whirl or tangential velocity from C_{w2} to $C_{w2'}$. This is illustrated in the figure below.

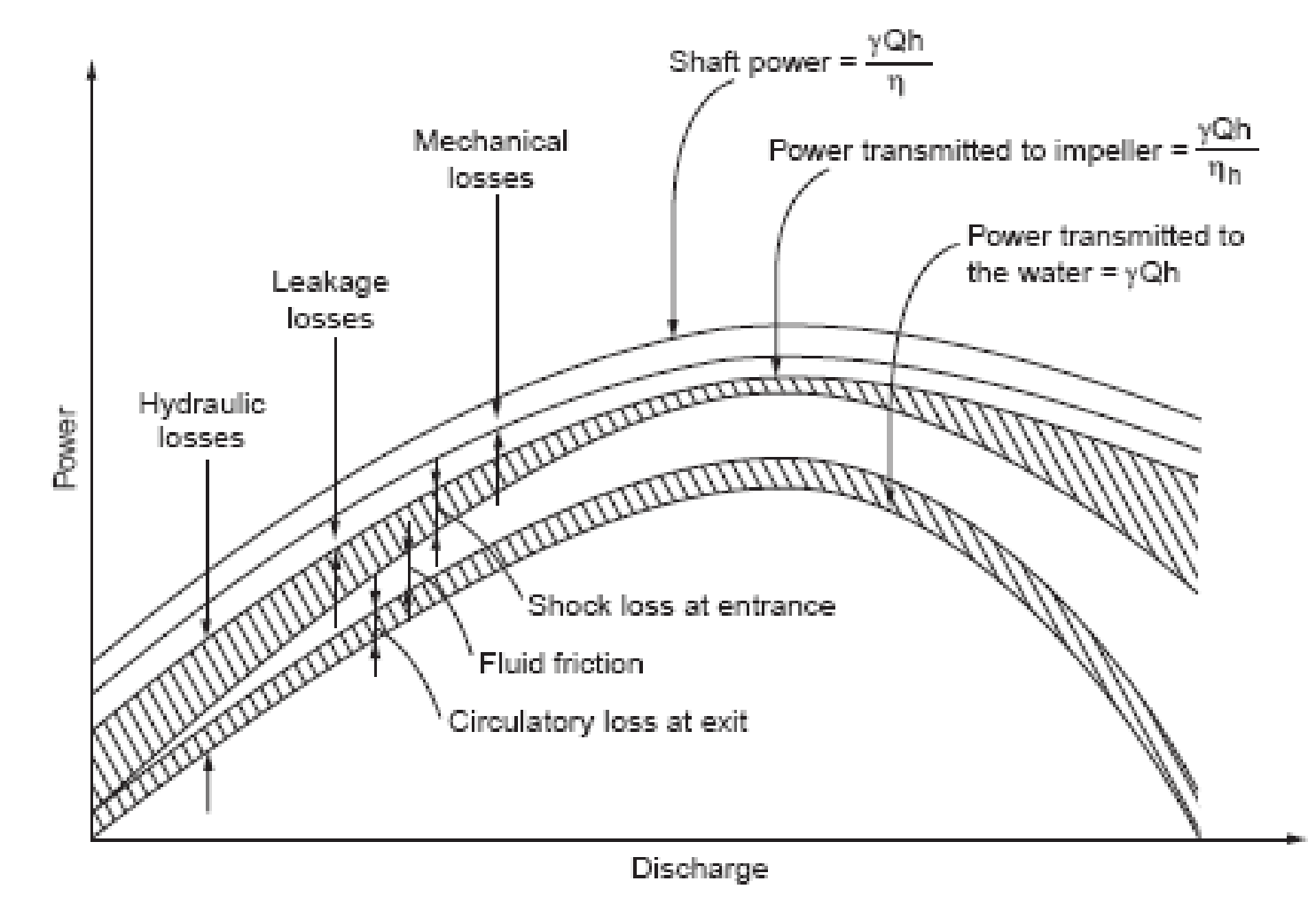
Euler's eqn with slip $\Rightarrow \frac{P_t}{\dot{m}} = \sigma_s V_{u2} U_2 - V_{ut} U_1$

- The dotted lines represent the **velocity diagram without slip**. The angle β_2 (ϕ_2) is the vane angle. **The solid line represents the velocity diagram with slip**. The angle $\beta_2' < \beta_2$. It may be seen that V_{U2}' / V_{U2} .
- The ratio of V_{U2}' / V_{U2} is known as slip factor (σ_s).
- For pu factor



90), the slip

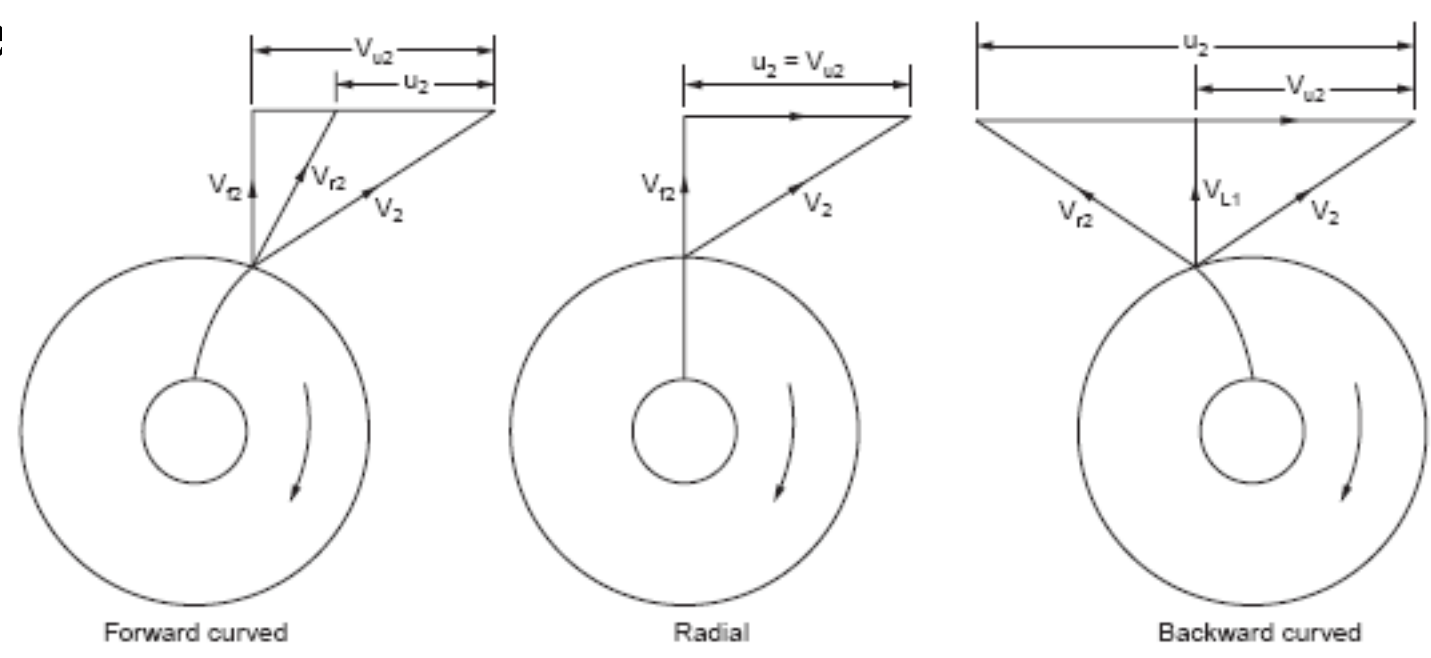
Various losses in a Pump



**Losses in a
Pump**

Effect of outlet blade angle

- There are three possible orientation of the blade at the outlet: forward curved, radial and backward curved arrangements. The velocity triangles for the three arrangements are as shown in the figure below



- In case of forward curved, $V_{u2} > U_2$ and V_2 is larger comparatively. In the case of radial, V_{u2} is equal to

Cavitation in Pumps

- *Cavitation* is the formation of vapors bubbles of the working fluid, in the region where local absolute static pressure becomes less than the vapor pressure of the liquid at some temperature.
- And when the fluid moves in a high pr region these bubbles collapse giving a pr as high as 3500atm.
- Doing so the enormous burst out pressure is instantaneously changed to the high velocity of the fluid element enough to pit the metal surface.
- In turbomachines there are different occasions in which low pressure regions may form where the pressure can even down below the vapor pressure of the liquid.

❑ Regions of Cavitation

- Suction region (Impeller eye in pumps)
- Regions of adverse pressure gradient
- Regions near air leakage (causes air cavitation)

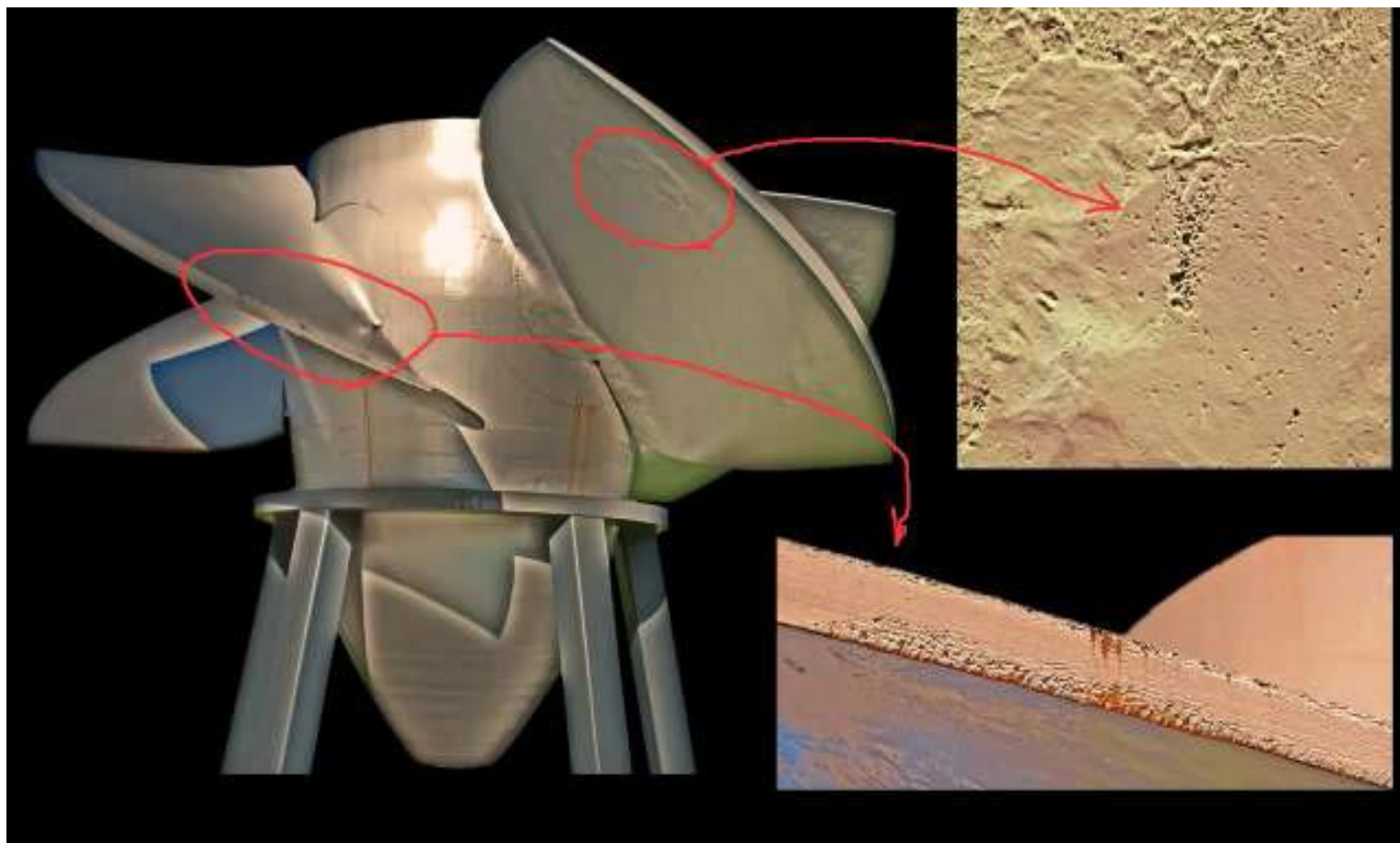
❑ Causes that pronounce cavitation in pumps

- Suction side losses, Leakage and High impeller speed

❑ To reduce the possible occurrence of cavitation

- Pr of the fluid should be $>$ vapor pr at any point, example absolute pr head should be $> 2.5\text{m}$ for water
- Use cavitation resistance materials like Al, bronze and stainless steel
- Reducing negative suction head
- Reducing losses in the suction side
- Working at low temperature

- ❑ Consequences of cavitation
 - Low efficiency pump operation
 - Pitting and erosion of metal surface and premature pump failure
 - Misalignment of shaft and bearing as a result of vibration
- ❑ Symptoms of cavitation in pumps
 - High noise
 - Vibration
 - Erratic pump operation (power fluctuation)



Net Positive Suction Head (NPSH)

- Is the head required at the pump inlet (low pr. Point where cavitation occur) to keep the liquid from cavitation or boiling.
- It is also defined as a measure of energy available on the suction side of the pump.
- It is the height of the pump axis from the water reservoir w/h can be permitted for installation.
- NPSH is the pressure head difference between the impeller eye absolute pressure and the vapor pressure of the

$$NPSH = \frac{P_e - P_v}{\rho g}$$

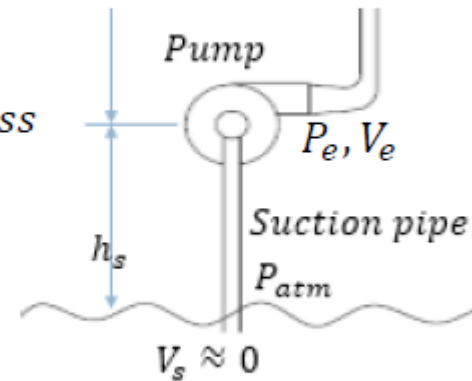
- Applying Bernoulli's equation from pipe suction end to impeller eye

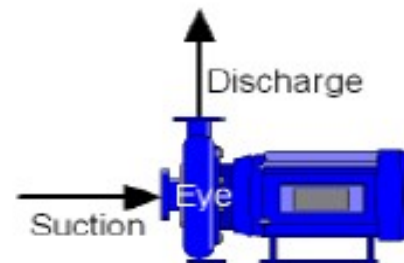
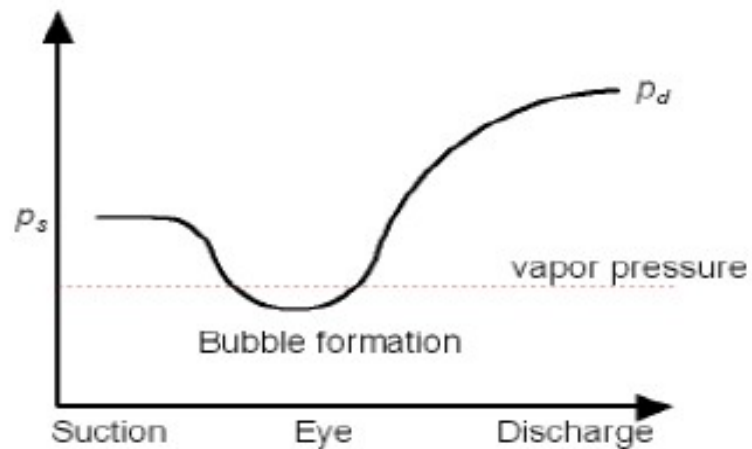
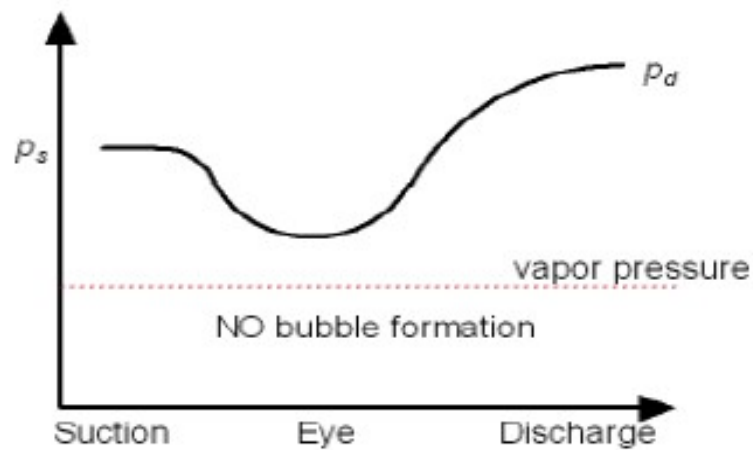
$$\frac{P_e - P_{atm}}{\rho g} + \frac{V_e^2 - V_s^2}{2g} + h_s + h_{loss} = 0$$

$$\frac{P_e}{\rho g} = \frac{P_{atm}}{\rho g} - \frac{V_e^2}{2g} - h_s - h_{loss}$$

$$NPSH = \frac{P_e - P_v}{\rho g}$$

$$= \frac{P_{atm} - P_v}{\rho g} - \frac{V_e^2}{2g} - h_s - h_{loss}$$





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NPSHr and NPSHa

❑ ***NPSH required***

is the NPSH head that is set by the pump manufacturer in which the pump should operate above this value to avoid potential cause of cavitation. It is obtained by a thorough experimental investigation.

❑ ***NPSH available***

is the NPSH value of the actual working condition.

- The same pump may face different systems and working conditions so that NPSHa varies. Pump users should always make sure that $NPSHa > NPSHr$.

Pump priming

- Centrifugal pumps are not capable of pumping air.
- It is because the density of water is almost 1000 times greater than that of air. It means when some air is trapped between the water and the pump suction, the pump needs to develop a head 1000 times in order to create the same suction effect to the liquid which is impossible.
- Therefore, it is inevitable to remove the air. Pump priming is a manual way of removing air by filling water inside the impeller casing which displaces the air.
- Suction pipe, casing of pump, portion of delivery pipe should be filled with liq. to be pumped.
- Pumps which can automatically prime themselves are called **self – priming pumps**.

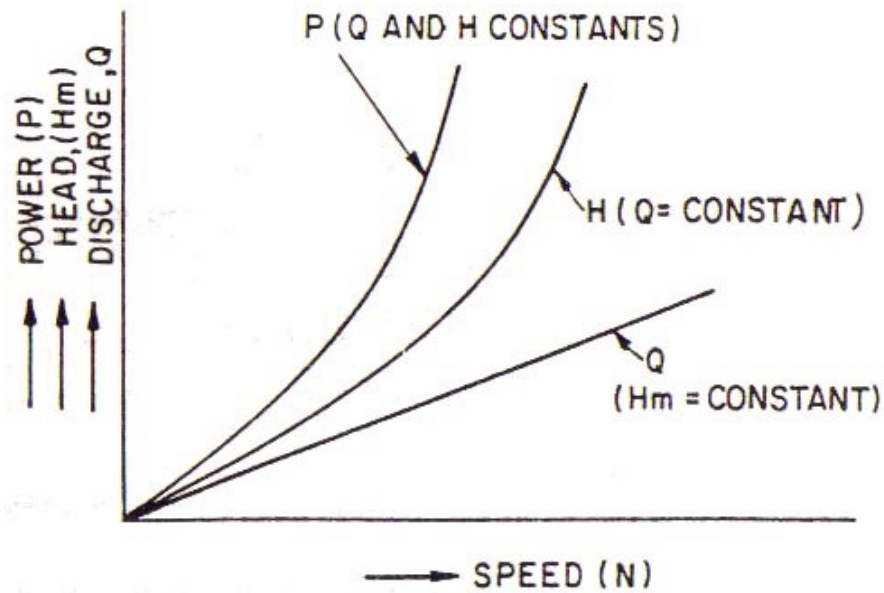
Characteristic Curves of a Centrifugal pump

- It is important to identify the difference between *pump* and *system* and the corresponding characteristic curves.
- When we say pump we are considering the process between the *pump suction eye* to the *pump outlet*.
- The system is the whole piping system both in the suction and delivery sides.
- Therefore, pump performance and system performance are different but needs to be compromised for best overall performance.
- Curves plotted at $N=\text{constant}$, Q independent variable, and H , P and efficiency are common dependent variables.
- Most of the performance curves in pumps are given by discharge Vs head, break horse power and efficiency.
- Such curves for pump and system are plotted in a

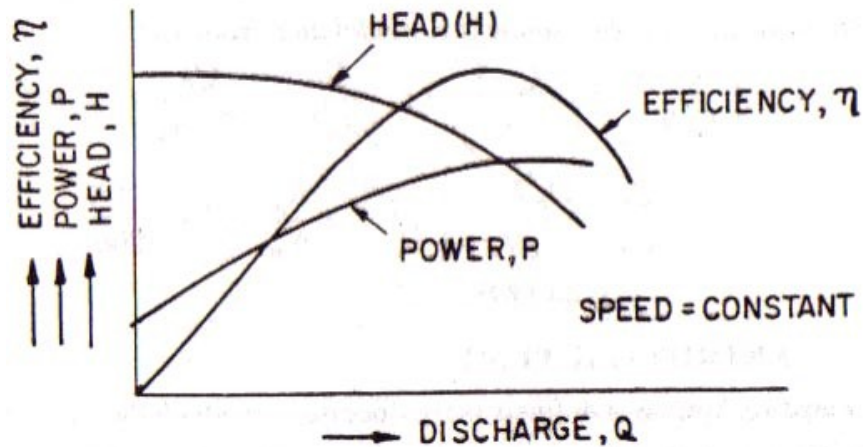
- **Shut off head point** is the point at which the pump develops the highest head but no discharge.
- **Run out point** is the point at which the pump delivers maximum discharge with zero head. The theoretical run out and shut off head points can be estimated from

$$H_P = \frac{U_2^2}{g} - \frac{U_2 \cot \beta_2}{\pi D_2 b_2 g} Q$$

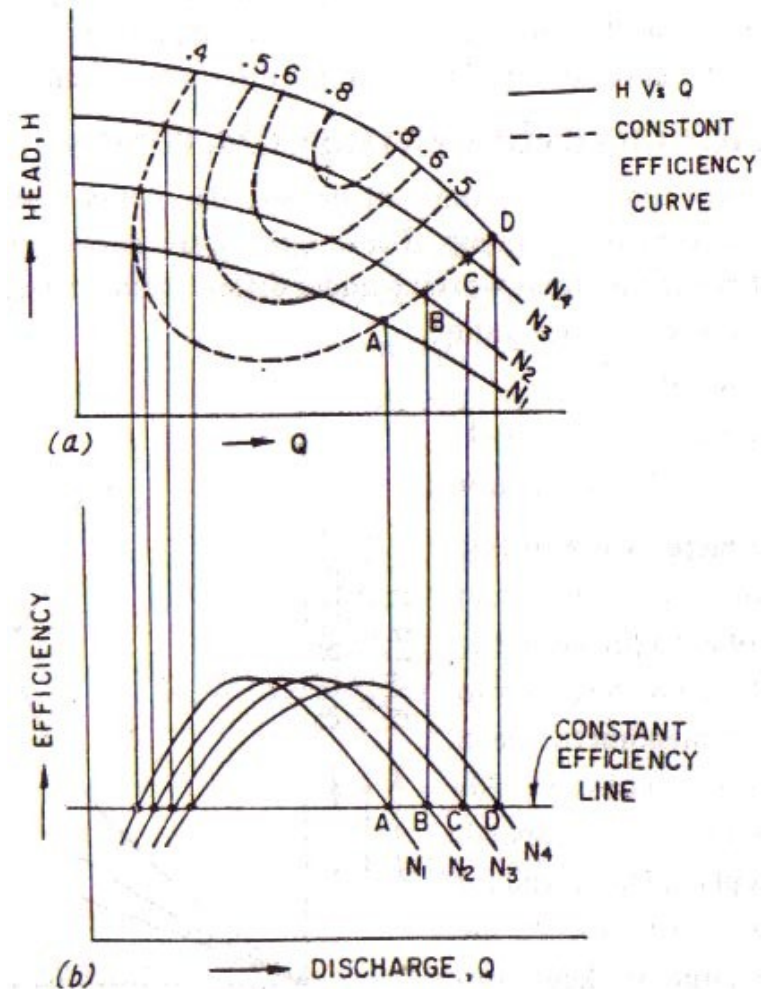
- The actual Characteristics curves of a Centrifugal pump are curves with the help of which an exact behavior and performance of the pump under different working conditions can be determined. These curves are plotted using data obtained from the test conducted in the laboratory.
- The important curves to be plotted are:
 - ✓ Main Characteristic Curves,
 - ✓ Operating Characteristic Curves,
 - ✓ Constant Efficiency Curves.



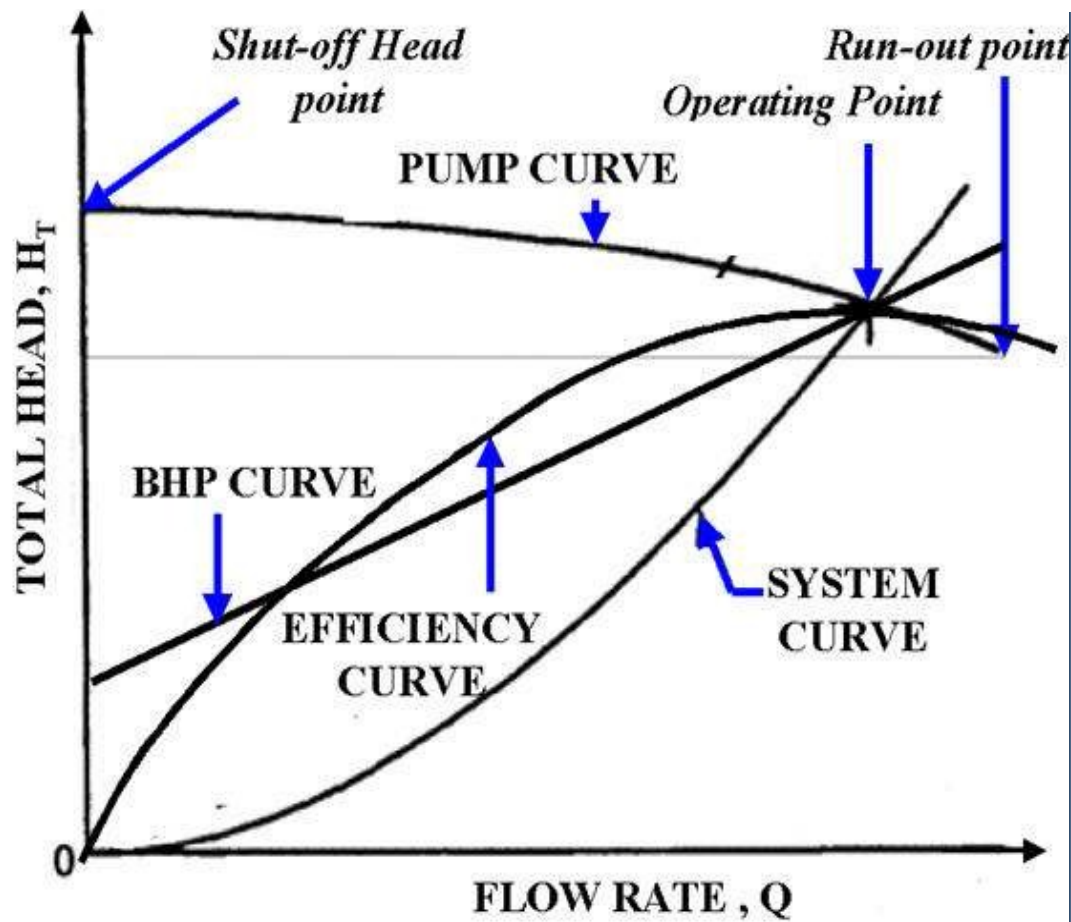
Main Characteristic Curves



Operating Characteristic Curves



Constant Efficiency Curves



Efficiency is max
at about
60% Q_{max}

Multi -Staging in Pumps

- Multistage pumps are employed to produce a high head or a high discharge.

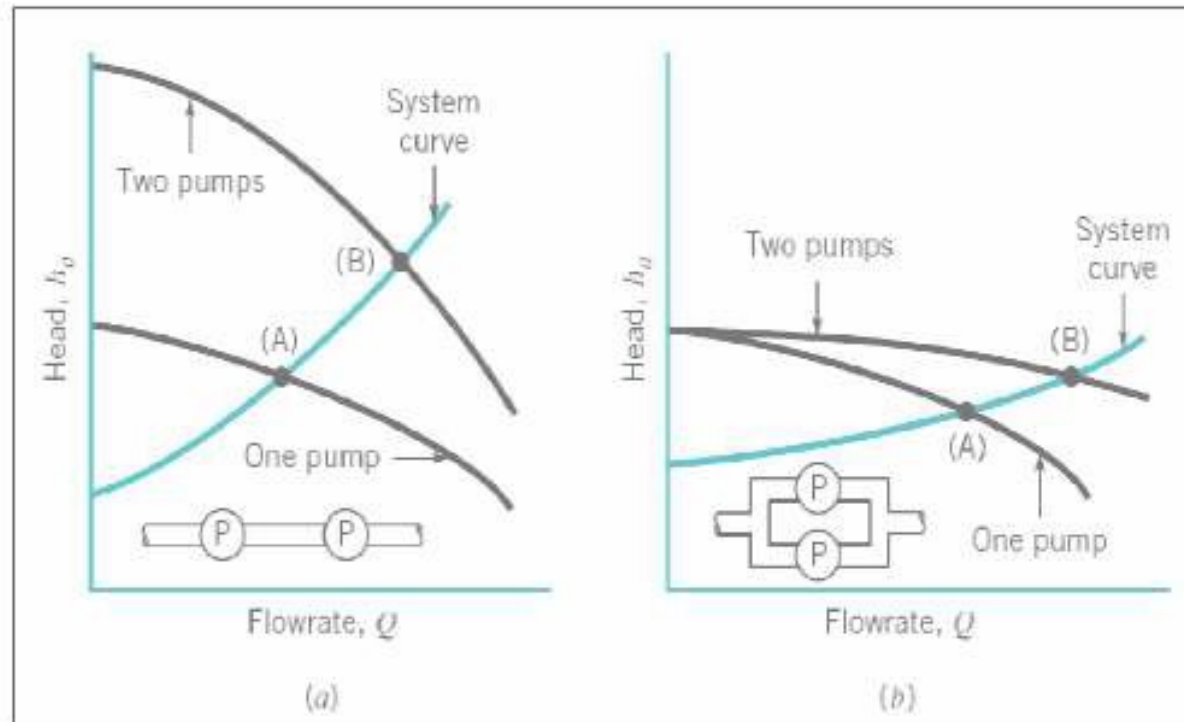
□ **Pumps in series:** When we need to increase the head we should connect pumps in series. The volume flow rate remains constant since similar pumps are connected usually.

A single pump failure results in system failure.

□ **Pumps in parallel:** This is applied when more discharge is needed. The head remains constant when similar pumps are connected.

- But the point where the multi - staged pump curve intersects the system curve should be the duty point.

Performance curves for multi - staged



Effect of operating pumps in (a) series and (b) in parallel.

Advantages of Multi staging

- A higher suction lift is possible because of lower specific speed of individual impeller.
- The stresses induced in the impeller are less than that in a single stage pump.
- The loss due to friction in a multi stage pump is less than that in equivalent single stage pump.
- The slip leakage in multistage pump is small.
- Number of stages can be chosen so that the speed of the pump suits to the speed of the motor.

Centrifugal Fans

- ❑ Rotating impeller increases air velocity
- ❑ Air speed is converted to pressure
- ❑ High pressures for harsh conditions
 - High temperatures
 - Moist/dirty air streams
 - Material handling
- ❑ Categorized by blade shapes
 - Radial
 - Forward curved
 - Backward inclined

Centrifugal Fan Impeller Types

Forward-curved fans are used in clean environments and operate at lower temperatures. Well suited for low tip speed and high air flow at lower

Radial fans have high static pressures (up to 1400 mm WC) and can handle heavily contaminated airstreams. Well suited for high temperatures and medium blade tip speeds.

Backward-inclined fans are more efficient than forward curved fans. Also known as "non-overloading" because changes in static pressure do not overload the motor

Type	Characteristics	Typical applications
Radial	High pressure, medium flow, efficiency close to tube axial fans, power increases continuously	Various industrial applications, suitable for dust laden, moist air/gases
Forward curved blades	Medium pressure, high flow, dip in pressure curve, efficiency higher than radial fans, power rises continuously	Low pressure HVAC, packaged units, suitable for clean and dust laden air/gases
Backward curved blades	High pressure, high flow, high efficiency, power reduces as flow increases beyond the point of highest efficiency	HVAC, various industrial applications forced draft fans etc
Airfoil	Same as backward curve	Same as backward

END

Example 1

A centrifugal pump impeller runs at 1400 rpm, and

vanes angle at exit is 25° . The impeller has an external diameter of 0.4 m and an internal diameter of 0.2 m. Assuming a constant radial flow through the impeller at 2.6 m/s, calculate (1) the angle made by the absolute velocity of water at exit with the tangent, (2) the inlet vane angle, and (3) the work done per kg of water

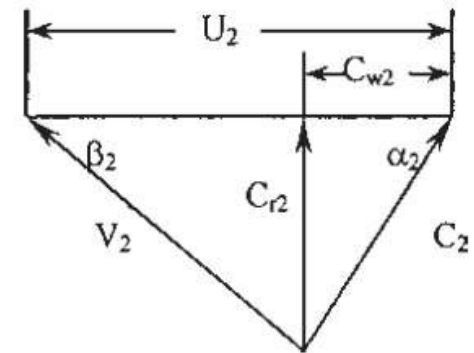
Solution:

1. Impeller tip speed is given by

$$U_2 = \frac{\pi D_2 N}{60} = \frac{\pi(0.4)(1400)}{60} = 29.33 \text{ m/s}$$

Whirl velocity at impeller tip

$$C_{w2} = U_2 - \frac{C_{r2}}{\tan \beta_2} = 29.33 - \frac{2.6}{\tan 25^\circ} = 23.75 \text{ m/s}$$



Now, from the velocity triangle at impeller tip,

$$\tan \alpha_2 = \frac{C_{r2}}{C_{w2}} = \frac{2.6}{23.75} = 0.1095$$

Therefore, $\alpha_2 = 6.25^\circ$.

2. Impeller velocity at inlet

$$U_1 = \frac{\pi D_1 N}{60} = \frac{\pi(0.2)1400}{60} = 14.67 \text{ m/s}$$

$$\tan \beta_1 = \frac{C_{r1}}{U_1} = \frac{2.6}{14.67} = 0.177$$

Therefore, $\beta_1 = 10.05^\circ$.

3. Work done per kg of water is given by

$$C_{w2}U_2 = 23.75(29.33) = 696.59 \text{ Nm} = 696.59 \text{ J.}$$

Example 2

A centrifugal pump runs at a tip speed of 12 m/s and a flow velocity of 1.5 m/s. The impeller diameter is 1.2 m and delivers 3.8 m³/min of water. The outlet blade angle is 28° to the tangent at the impeller periphery. Assuming that the fluid enters in the radial direction and zero slip, calculate the **torque delivered** by the impeller.

§ $H = E = (U_2 C_{w2} - U_1 C_{w1})/g$. Since $C_{w1} = 0$, as there is no inlet whirl component, head H is given by

$$H = \frac{U_2 C_{w2}}{g} = \frac{U_2}{g} \left(U_2 - \frac{1.5}{\tan 28^\circ} \right) = \frac{12}{9.81} \left(12 - \frac{1.5}{\tan 28^\circ} \right)$$
$$= 11.23 \text{ m}$$

$$\text{Power delivered} = \rho g Q H \frac{J}{s} = \frac{1000(9.81)(3.8)(11.23)}{60(1000)}$$
$$= 6.98 \text{ kW}$$

$$\text{Torque delivered} = \text{Power/angular velocity} = 6980(0.6)/12 = 349 \text{ Nm.}$$