PUMP

A mechanical device to increase the pressure energy of liquid. It is mostly used to raise fluid from lower to higher level. This is achieved by creating a low pressure at the suction side and ahigh pressure at the discharge side of the pump.

Classification

- **PUMPS**
  - POSITIVE DISPLACEMENT PUMPS
    - RECIPROCATING PUMPS
      - PISTON PUMPS
      - PLUNGER PUMPS
      - DIAPHRAGM PUMPS
    - ROTARY PUMPS
      - GEAR PUMPS
      - LOBE PUMPS
      - SCREW PUMPS
      - CAM PUMPS
      - VANE PUMPS
  - CENTRIFUGAL PUMPS

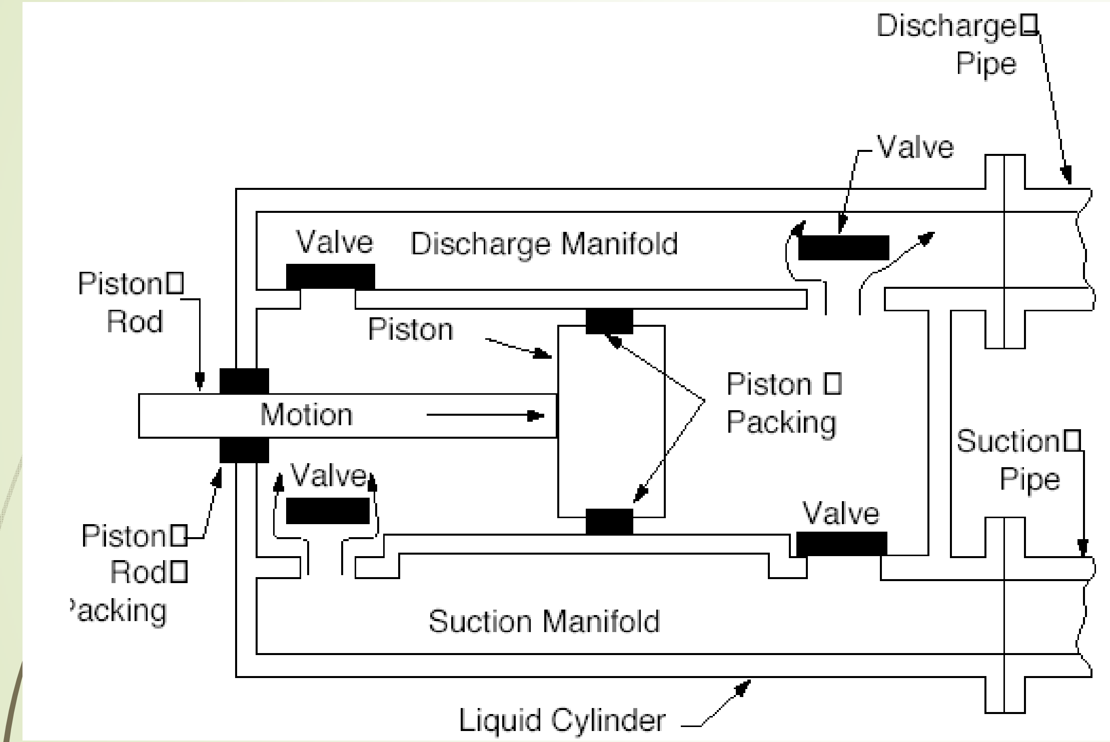
RECIPROCATING PUMPS

Based on two stroke principles:

- √ **High pressure, high efficiency**
- √ **Self-priming**
- √ **Small quantity, vibration, physical dimension, uneven flow**

Used mainly for handling slurries in plant processes and pipeline applications

RECIPROCATING PUMPS



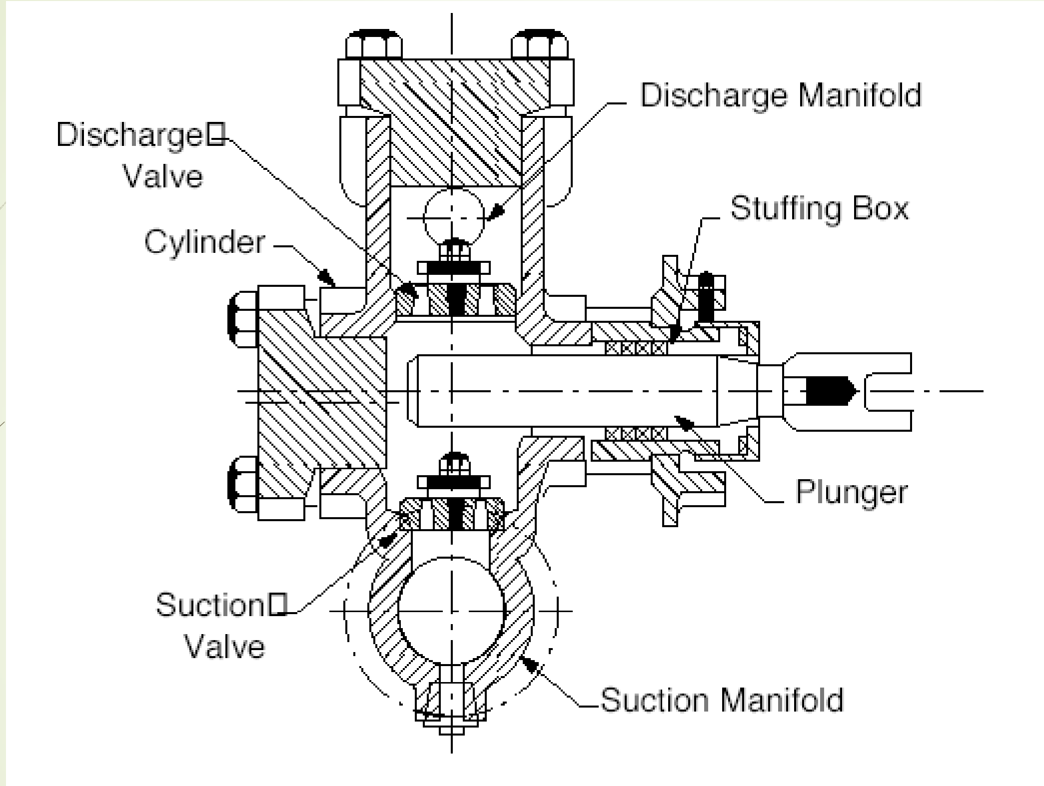
Reciprocating Piston Pump

Based on two stroke principles:

- √ High pressure, high efficiency
- √ Self-priming
- √ Small quantity, vibration, physical dimension, uneven flow

Used mainly for handling slurries in plant processes and pipeline applications

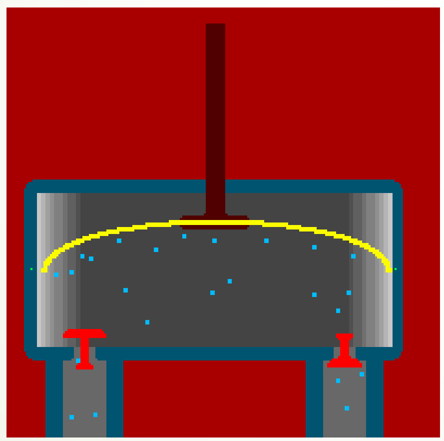
Plunger Pump

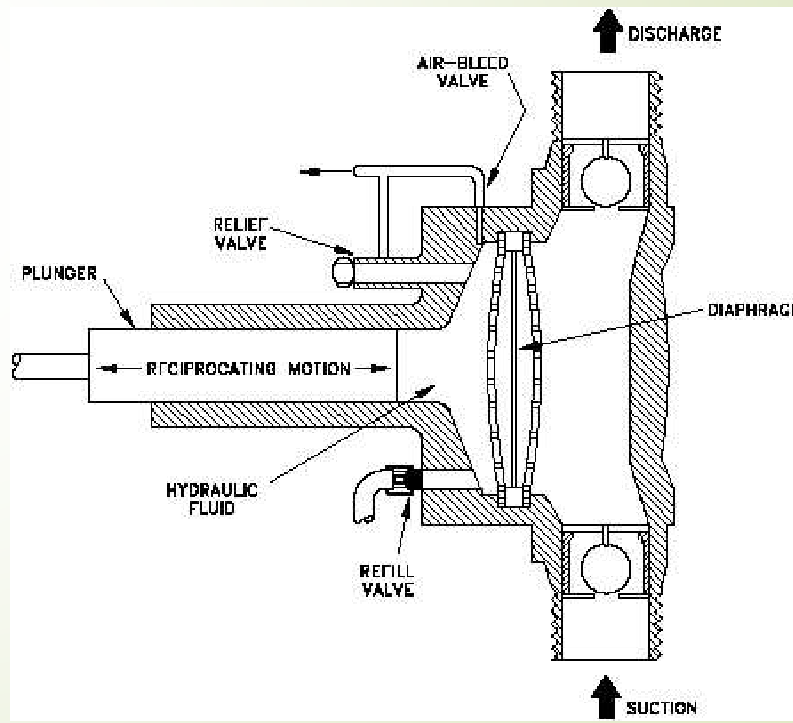


- Two ball check valves on each side
- Low pressure on the upward part,
  high pressure on the downward part

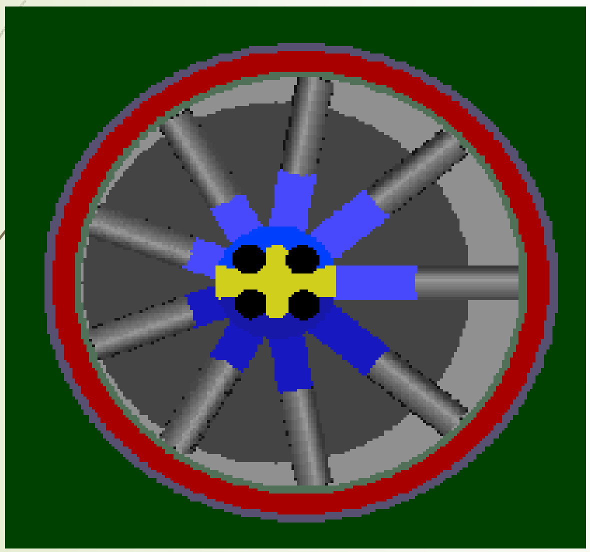
Diaphragm Pump

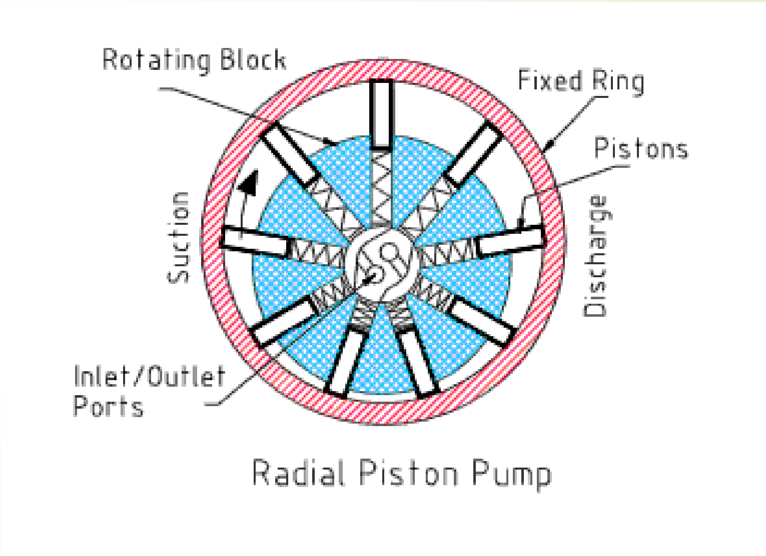
- Rod is moved to push and pull the diaphragm.
- Can be used to make artificial hearts.





Radial Piston Pump





Rotary Positive Displacement pumps

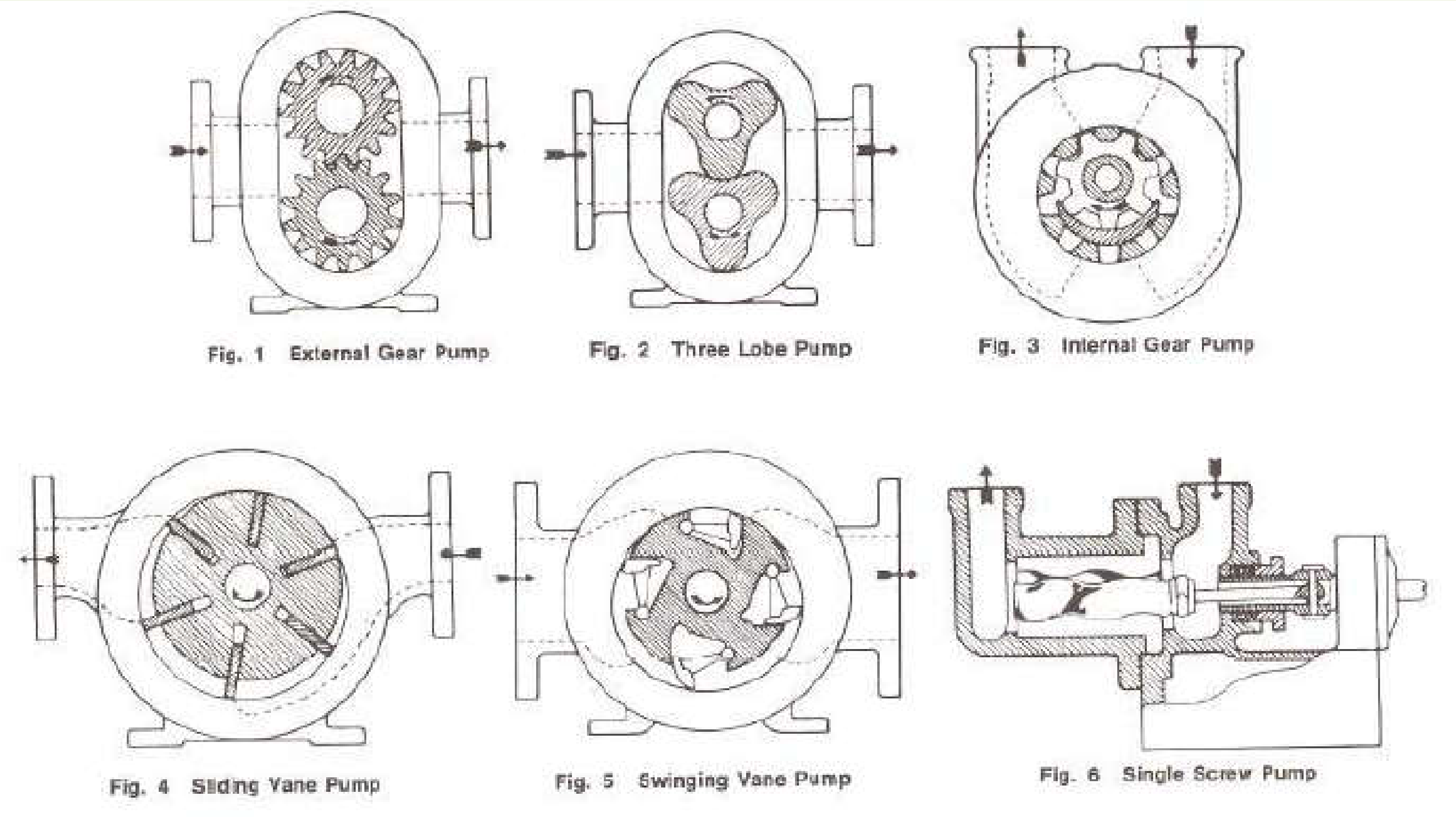
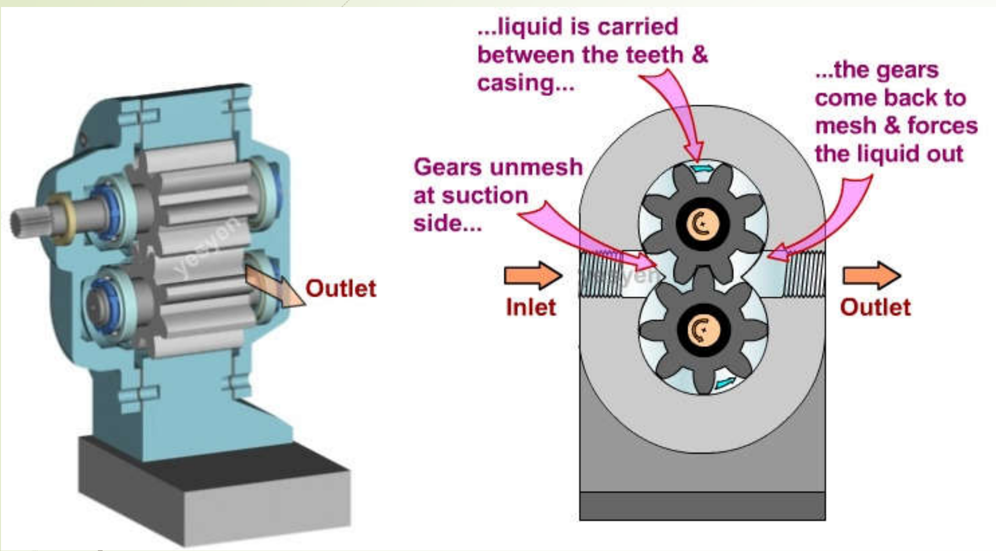
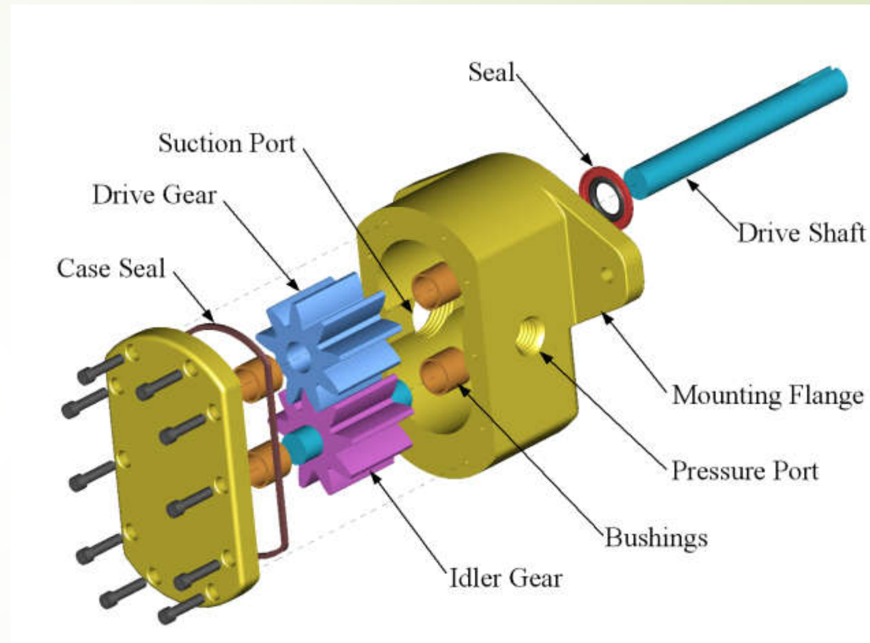


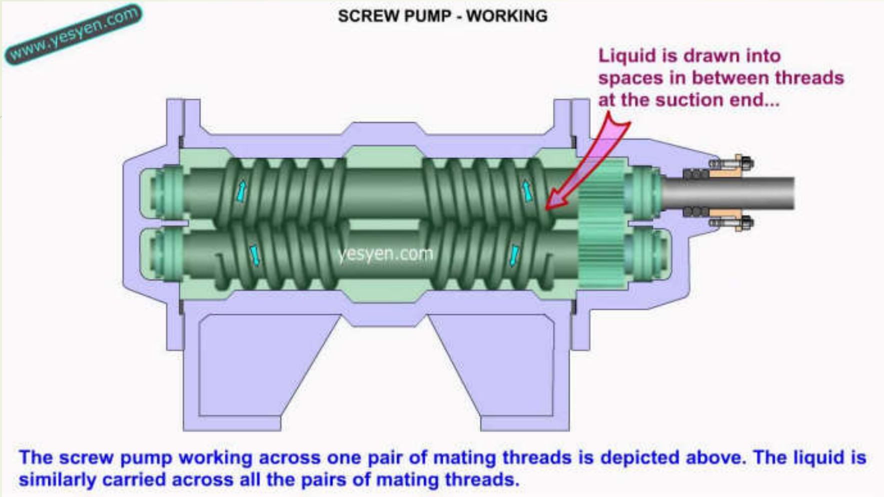
Fig. 1 External Gear Pump
Fig. 2 Three Lobe Pump
Fig. 3 Internal Gear Pump
Fig. 4 Sliding Vane Pump
Fig. 5 Swinging Vane Pump
Fig. 6 Single Screw Pump

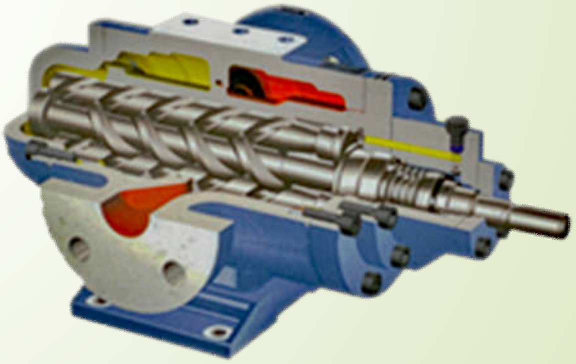
Gear Pump


Outlet
...liquid is carried between the teeth & casing...
Gears unmesh at suction side...
...the gears come back to mesh & forces the liquid out
Inlet
Outlet


Seal
Suction Port
Drive Gear
Case Seal
Drive Shaft
Mounting Flange
Pressure Port
Bushings
Idler Gear

Screw Pump

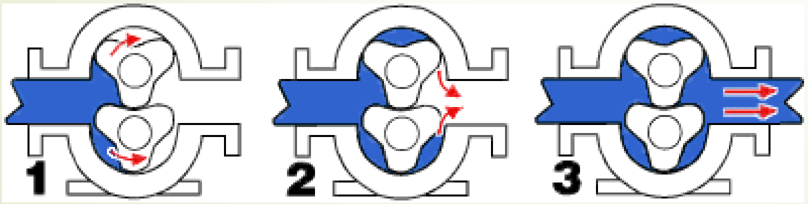




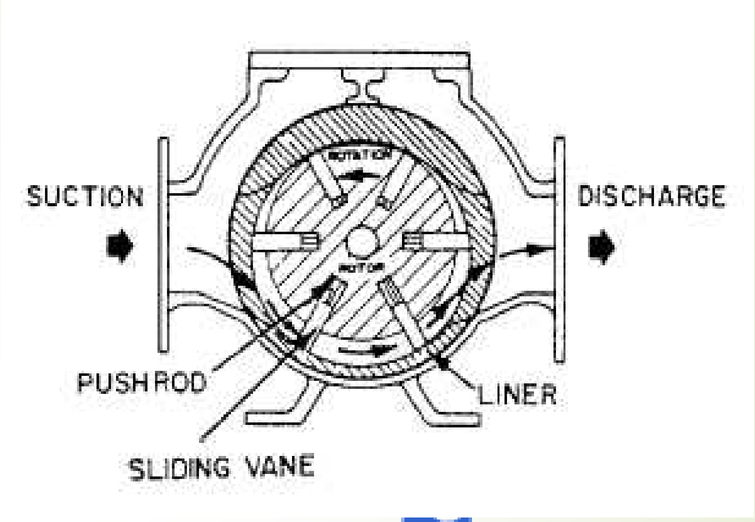
- **Screw pumps carry fluid in the spaces between the screw threads.**
- **The fluid is displaced axially as the screws mesh.**

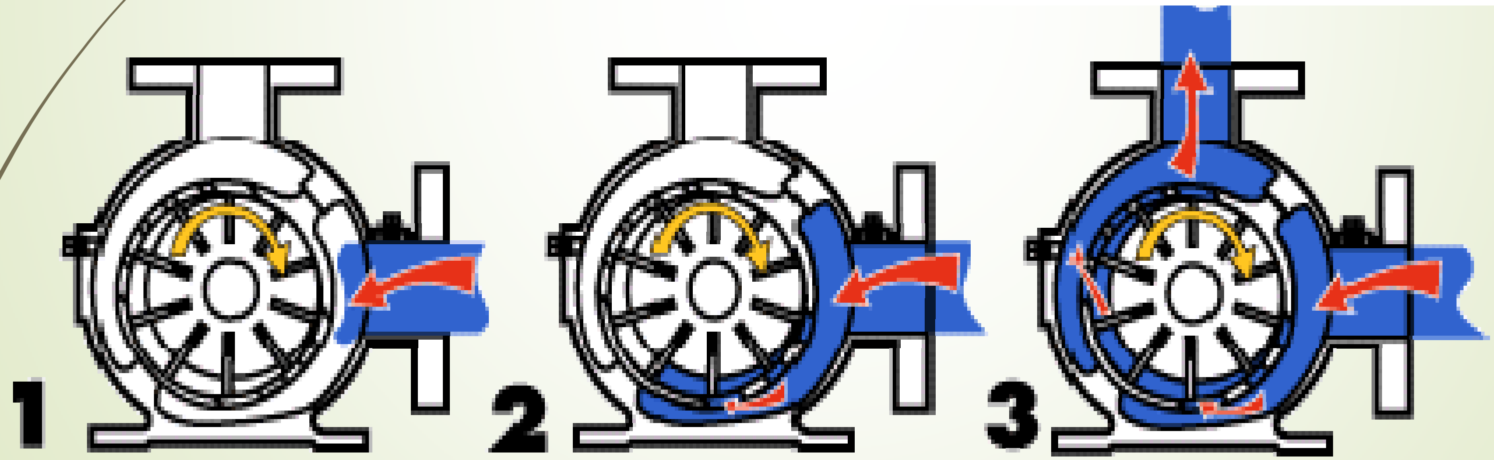
Lobe Pump

- **Fluid is carried between the rotor teeth and the pumping chamber**
- **The rotor surfaces create continuous sealing**
- **Rotors include bi-wing, tri-lobe, and multi-lobe configurations**



Vane Pump





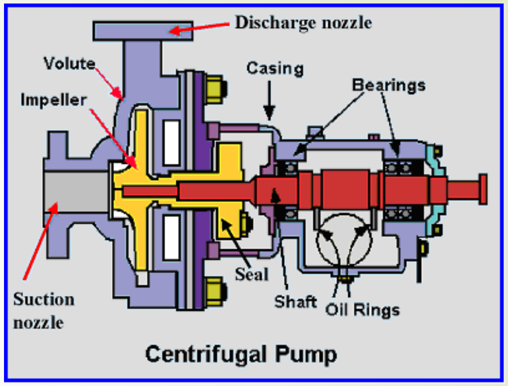
CENTRIFUGAL PUMP

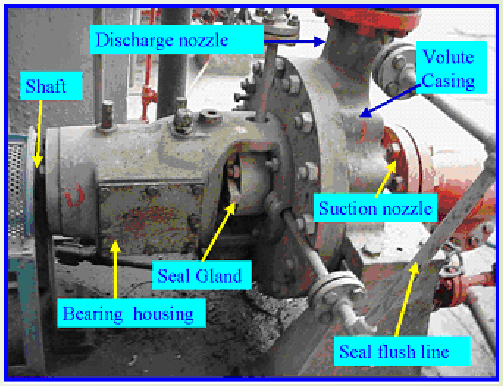
- Convert the mechanical energy into hydraulic energy by centrifugal force on the liquid
- Constitute the most common type of pumping machinery
- Used to move liquids through a piping system
- Has two main components:

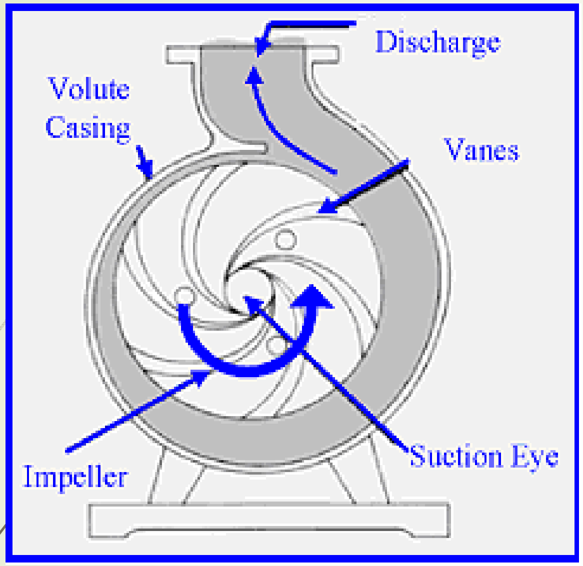
1. Stationary componets, casing, casing cover and bearings

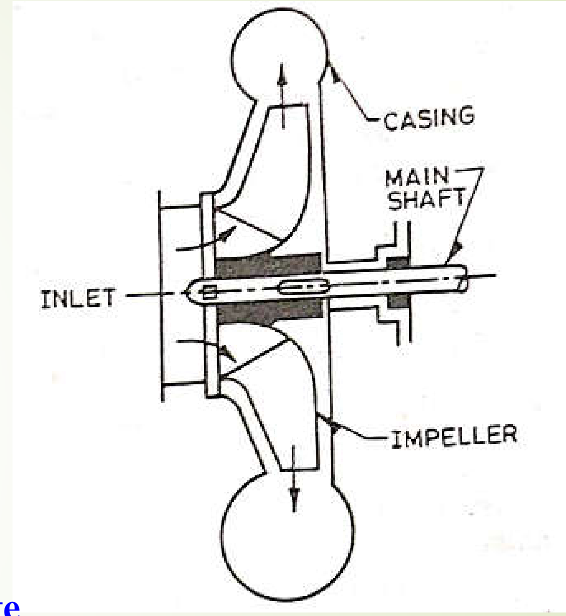
2. Rotating components, impeller and shaft

Classified into three categories ; Radial Flow, Mixed Flow, Axial Flow









- Simplest piece of equipment in any process plant
- Energy changes occur by virtue of impeller and volute
- Liquid is fed into the pump at the center of a rotating impeller and thrown outward by centrifugal force
- The conversion of kinetic energy into pressure energy supplies the pressure difference between the suction side and delivery side of the pump

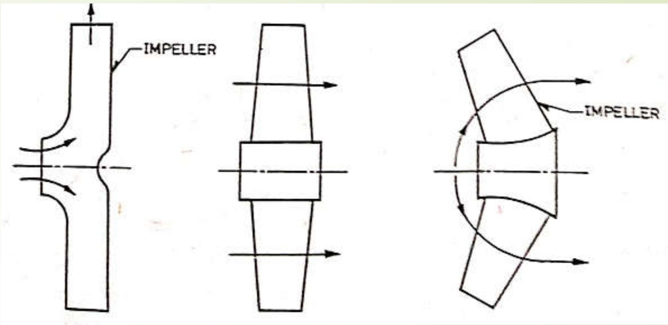
Centrifugal Pump

Advantages

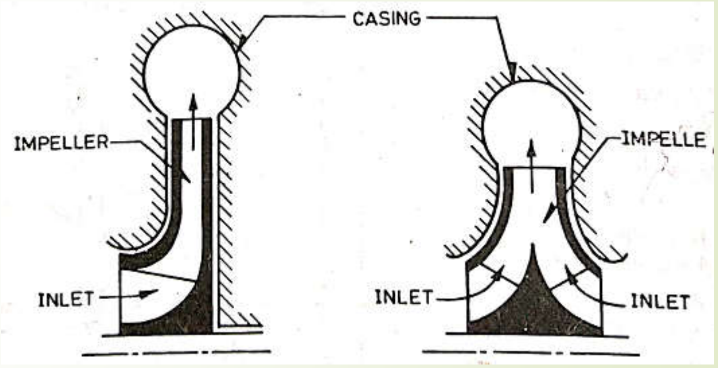
- Simple in construction and cheap
- Handle liquid with large amounts of solids
- No metal to metal fits
- No valves involved in pump operation
- Maintenance costs are lower

Disadvantages

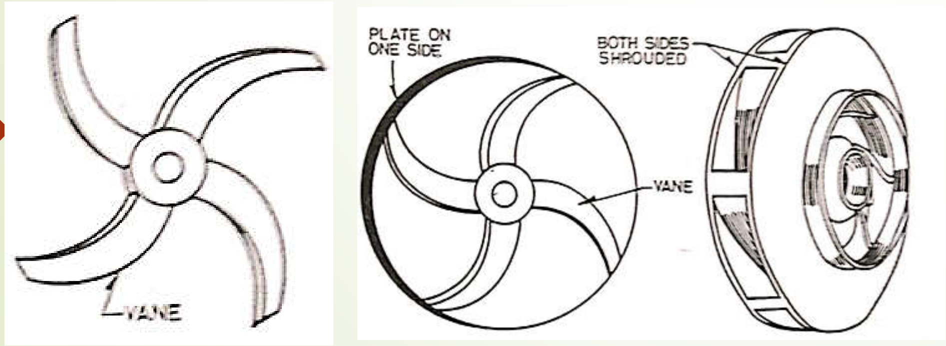
- Cannot handle highly viscous fluids efficiently
- Cannot be operated at high heads
- Maximum efficiency holds over a narrow range of conditions



Radial, axial and mixed flow impellers



Single and double suction impeller



Open, Semi-open and enclosed impellers

Specific speed

Specific speed is the dimensionless parameter for comparison of pump. It is the speed of a geometrically similar pump producing unit head and deliver unit quantity of fluid

$$N_S = \frac{N\sqrt{Q}}{(gH)^{3/4}}$$

More often dimensional specific speed is used in practise

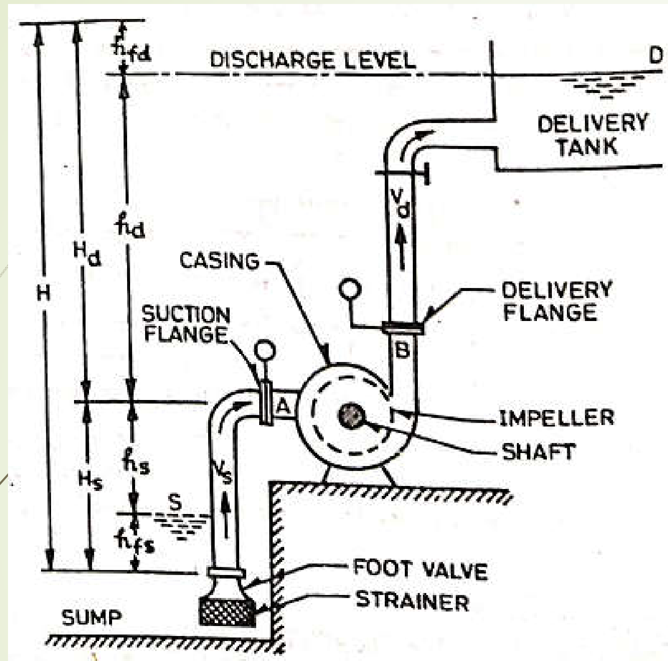
$$N_S = \frac{N\sqrt{Q}}{H^{3/4}}$$

Q= flow rate (m^3/s) , N = rotor speed (RPM)and H= head developed (m)

Specific speed classification of pumps

Flow direction	speed	Dimensional specific speed	Non Dimensional specific speed
Radial	Low	10 – 30	1.8 – 5.4
	Medium	30 – 50	5.4 – 9.0
	High	50 – 80	9.0 – 14.0
Mixed flow		80 – 160	14 – 29
Axial flow		100 – 450	18 – 81

The best efficiency is obtained for the various types of pumps in this range of specific speeds indicated



Typical Installation of a centrifugal pump showing change in pressure

$$\text{Manometric head } H_m = \left(\frac{p_d}{w} + \frac{V_d^2}{2g} + h\right) - \left(\frac{p_s}{w} + \frac{V_s^2}{2g}\right)$$

γ and W both represents specific weight, ρg

Suction Head $H_s = h_i + h_{fs} + h_s + \frac{V_s^2}{2g}$

h_i = loss in suction inlet pipe entry

h_{fs} = loss due to friction at suction pipe.

V_s = flow velocity at suction pipe.

$(h_i + h_{fs} + h_s)$ is measured by installing a vacuum gauge at pump suction. quite adjacent to pump.

$$\therefore \quad H_s = \frac{P_s}{\gamma} + \frac{V_s^2}{2g}$$

Delivery Head $H_d = h_{fd} + h_d + \frac{V_d^2}{2g}$

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

V_d = flow velocity at delivery pipe.

$(h_{fd} + h_d)$ is measured by a gauge at delivery pipe adjacent to the pump.

∴ Total external head against which a pump has to work

$$H = H_s + H_d - \frac{V_s^2}{2g} = (h_i + h_{fs} + h_s) + \left(h_{fd} + h_d + \frac{V_d^2}{2g}\right)$$

Often $\frac{V_d^2}{2g}$ is very less, compared to other terms and can be neglected.

Power required to drive the pump $= \frac{\gamma Q H}{1000}$ **kW**

$\gamma = \rho . g$ Q = flow rate (m^3/s)   H = head in m

ρ = density (kg/m3)

Cavitation and NPSH

Cavitation is the formation and subsequent collapse of vapour bubbles in a flowing liquid and is often responsible for significant damage of impellers of pumps. The formation of vapour bubbles in the pumping fluid will occur when the when the fluid pressure drops below its vapour pressure.

Net positive suction head (NPSH) represents a combination of following heads :

$NPSH$ = (absolute pressure at inlet to pump) – (vapour pressure of liquid being pumped) + (velocity head in suction pipe)

$$NPSH = \frac{p_s}{w} - \frac{p_v}{w} + \frac{V_s^2}{2g} = \left(\frac{p_a}{w} - h_s - h_{fs} - \frac{V_s^2}{2g}\right) - \frac{p_v}{w} + \frac{V_s^2}{2g}$$

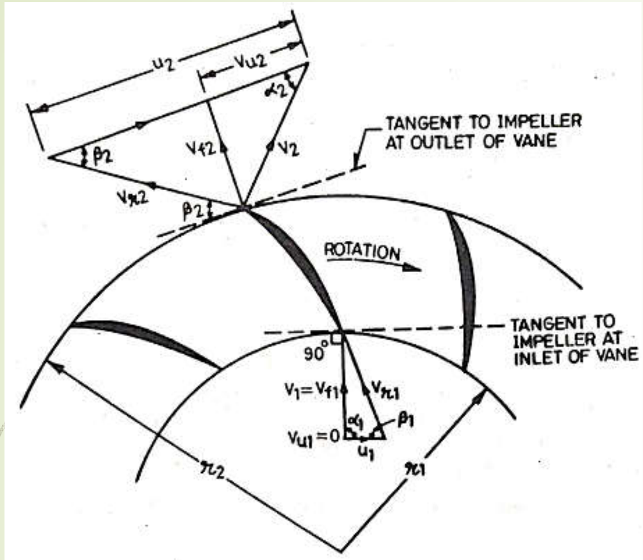
where p_a denotes the atmospheric pressure on the surface of liquid in the suction well. Simplification gives :

$$NPSH = \left(\frac{p_a}{w} - \frac{p_v}{w} - h_s - h_{fs}\right)$$

NPSH should be such that the fluid does not boil under reduced pressure.

The above equation indicates the NPSH available. If NPSH available is less than NPSH required then cavitation will occur.

Velocity vector diagram and work done for a centrifugal pump



V = absolute velocity of fluid ; u = blade or peripheral velocity ; V_r = relative fluid velocity ; V_f = flow component of absolute velocity ; V_u = whirl or tangential component of absolute velocity.

Further suffix 1 and 2 represent the conditions at inlet and outlet of the impeller.

Rate of change of angular momentum

$$= m\,(V_{u2}\,r_2 - V_{u1}\,r_1)$$

$$= \rho\,Q\,(V_{u2}\,r_2 - V_{u1}\,r_1) = \frac{wQ}{g}\,(V_{u2}\,r_2 - V_{u1}\,r_1)$$

Q = Liquid flow rate; *w*= specific weight, ρg
r_1 & r_2 impeller radius at inlet and outlet respectively

$$\text{Torque} = \frac{wQ}{g}\,(V_{u2}\,r_2 - V_{u1}\,r_1)$$

Now, energy transfer = torque × rotational speed in radian/sec

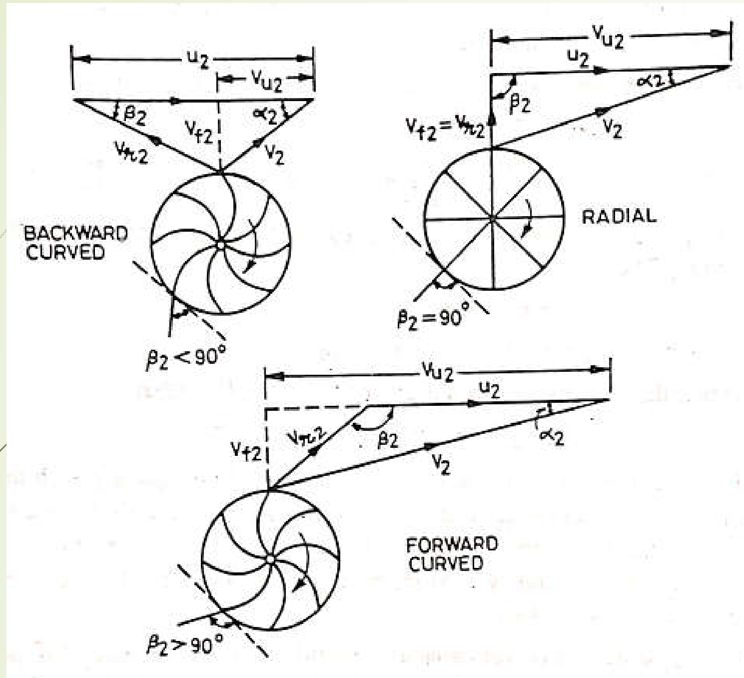
$$E = \frac{wQ}{g}\,(V_{u2}\,r_2 - V_{u1}\,r_1) \times \omega = \frac{wQ}{g}\,(V_{u2}\,u_2 - V_{u1}\,u_1)$$

Energy transfer per unit weight is referred as Euler Head (H_e)

$$H_e = \frac{E}{wQ} = \frac{(V_{u2}\,u_2 - V_{u1}\,u_1)}{g}$$

For axial or radial fluid entry (no whirl component), term $V_{u1}u_1$ vanishes and the Euler equation takes the form

$$H_e = \frac{V_{u2}\,u_2}{g}$$



Influence of vane exit angle on head capacity and power capacity relationship

Backward curved, radial and forward curved vanes

- **Backward curved :** Outlet tip of blade curves in a direction opposite to that of motion, and the angle between the blade tip and the angle tangent to rotor at exit is acute ($\beta_2 < 90^o$).
- **Radial :** Liquid leaves the vane with relative velocity in a radial direction and angle $\beta_2 = 90^o$.
- **Forward curved :** Outlet tip of blade curves in the direction of motion, and the angle between the blade tip and the tangent to rotor at exit is obtuse ($\beta_2 > 90^o$).

Head -Capacity and Head- Power Relationship

$$\text{Euler head } H_e = \frac{V_{u2} u_2}{g} = \frac{u_2}{g}\left[u_2 - V_{f2} \cot \beta_2 \right] = \frac{u_2}{g}\left[u_2 - \frac{Q}{A_2} \cot \beta_2 \right]$$

When u_2, β_2 and A_2 fixed $H_e = K_1 - K_2 Q$

For backward curved vanes $\beta_2 < 90^0$ and $\cot \beta_2$ is positive. Consequently with increase in mass flow rate the Euler head falls ; the head capacity characteristics has a negative slope. For radial vanes $\beta_2 = 90^0$ and $\cot \beta_2 = 0$. Thus the head remains constant with variation in flow rate. For forward curved vanes $\beta_2 > 90^0$ and $\cot \beta_2$ is negative. Obviously with increase in mass flow rate the Euler head rises : the head capacity characteristic has a positive slope.

Proceeding further, the power developed is given by $P = wQH$, so that

$$p \propto QH \quad ; \quad P = A' Q - B' Q^2$$

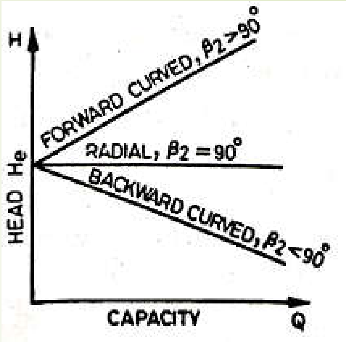
When u_2, A_2 and Q is fixed $H_e = k_1 - k_2 \cot\beta_2 . Q$

where k_1 and k_2 are constants and β_2 is the outlet blade angle. cot β_2 becomes negative for forward curved blading. So head increases with flow rate. For radial blading cost $\beta_2 = 0$, and hence the head is constant with flow rate. In the case of backward curved blading, the head decreases with flow rate.

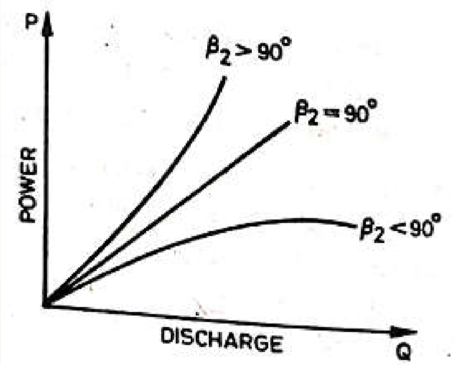
The rising characteristics of the forward curved blading leads to increase of power input with increase of Q. The power curve is not self limiting and damage to motor is possible. The forward curved blading is rarely used.

The backward curved blading leads to self limiting power characteristics and reduced losses in the exit kinetic energy.

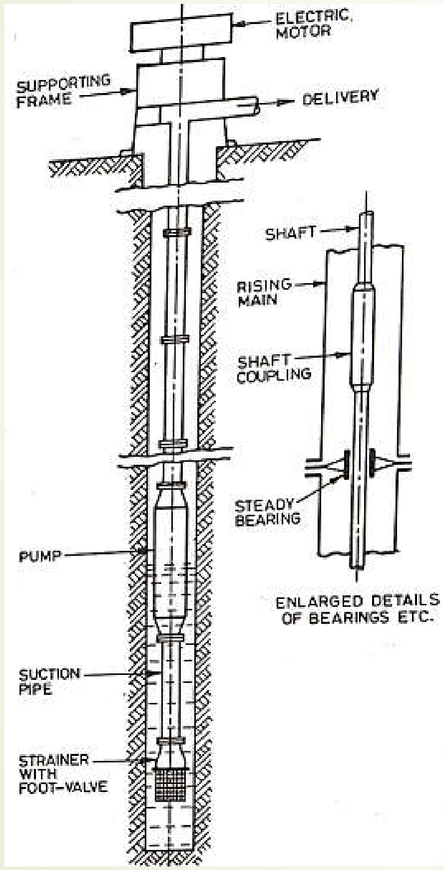
So the backward curved blading is almost universally used. The radial blading also leads to rising power characteristics and it is used only in small sizes.



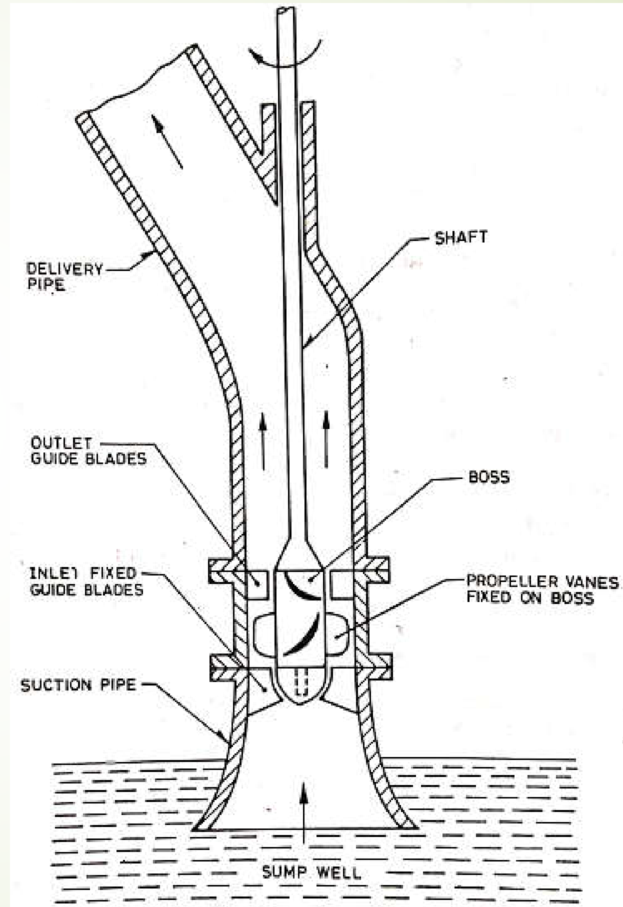
Influence of vane exit angle on head capacity relationship



Influence of vane exit angle on Power capacity relationship

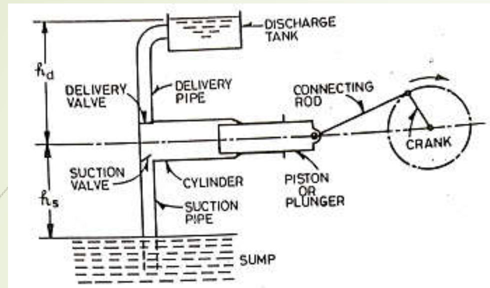


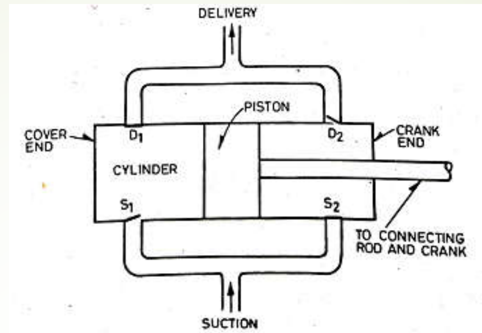
Bore Hole Pump



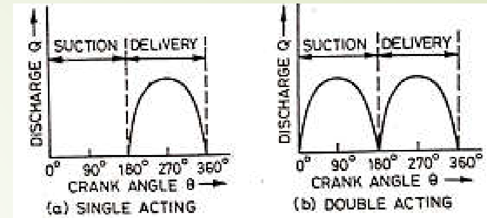
Axial Flow Pump

Operation of reciprocating pump

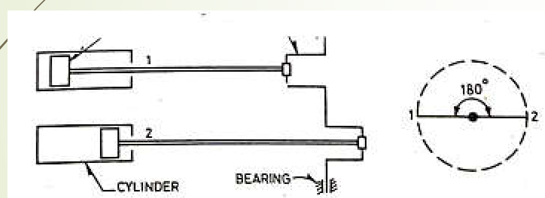




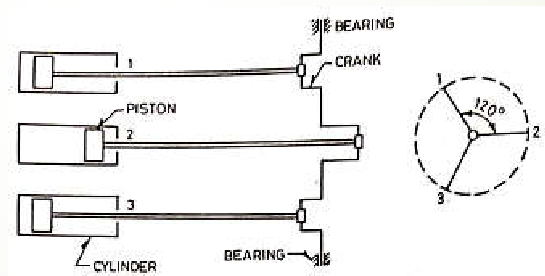
Single & Double acting reciprocating pump



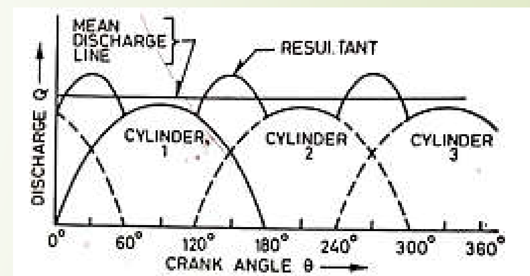
Discharge-crank angle diagram for Single & Double acting reciprocating pump



Schematics of a double cylinder reciprocating pump



Schematic of a three throw pump



Performance of a three throw pump

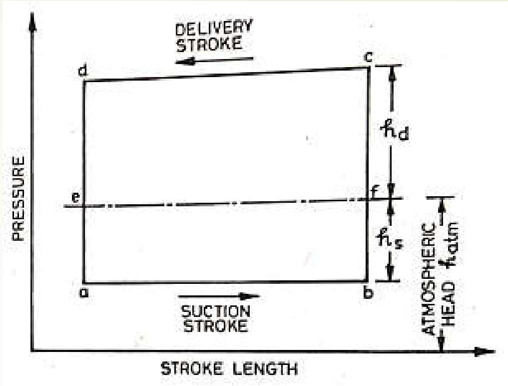
Work and Power output

Suction side; force on piston= wh_sA**; work done =** wh_sAL
***Where*, A= area of piston, L= stroke,** h_s**= suction head**

Delivery side; force on piston= wh_dA**; work done =** wh_dAL
***Where*, A= area of piston, L= stroke,** h_d **= delivery head**

Total work done = $w(h_s + h_d)AL$
Theoretical Power required to drive the pump
$= w(h_s + h_d)AL.N/60 = wQ_{th}H$ **;** ***where*, N= RPM**

Actual Power =Theoretical Power / Efficiency of the pump



Theoretical Indicator diagram