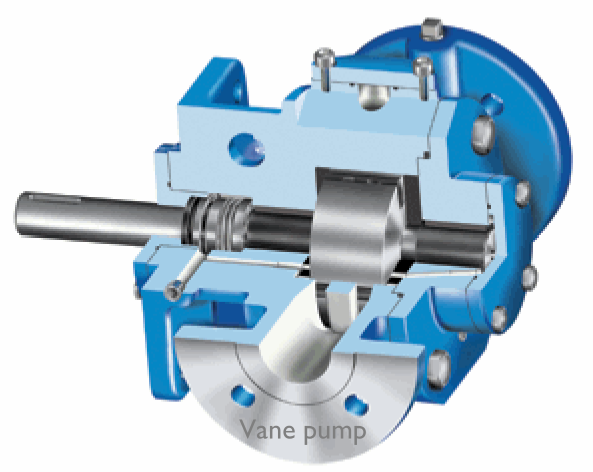
Lecture on

Positive Displacement Pump



by

Dr. Shibayan Sarkar

Department of Mechanical Engg

Indian Institute of Technology (ISM), Dhanbad

WHAT IS PUMP?

A hydrodynamic pump machine is a device which converts the mechanical energy held by a device into potential and kinetic energy in fluid.

Pumps enable a liquid to:

1. **Flow from a region or low pressure to one of high pressure.**
2. **Flow from a low level to a higher level.**
3. **Flow at a faster rate.**

Pump is the machine designed to move fluid and add energy to them.

There are two main categories of pump according to design and principle of operation:

- Rotodynamic pumps.
- Positive displacement pumps.

- **A rotodynamic or non- positive displacement pump imparts velocity energy to the fluid, which is converted to pressure energy upon exiting the pump casing.**
- **A positive displacement pump moves a fixed volume of fluid within the pump casing by applying a force to moveable boundaries containing the fluid volume.**

Based on **direction of fluid flow** relative to shaft

Radial (centrifugal pumps) Axial (boat propellers) Mixed

Single- vs **multi-stage**(diffuser type casing is used, pump impellers are connected in series)

Constant vs **variable speed** (to operate under different load efficiently)

CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

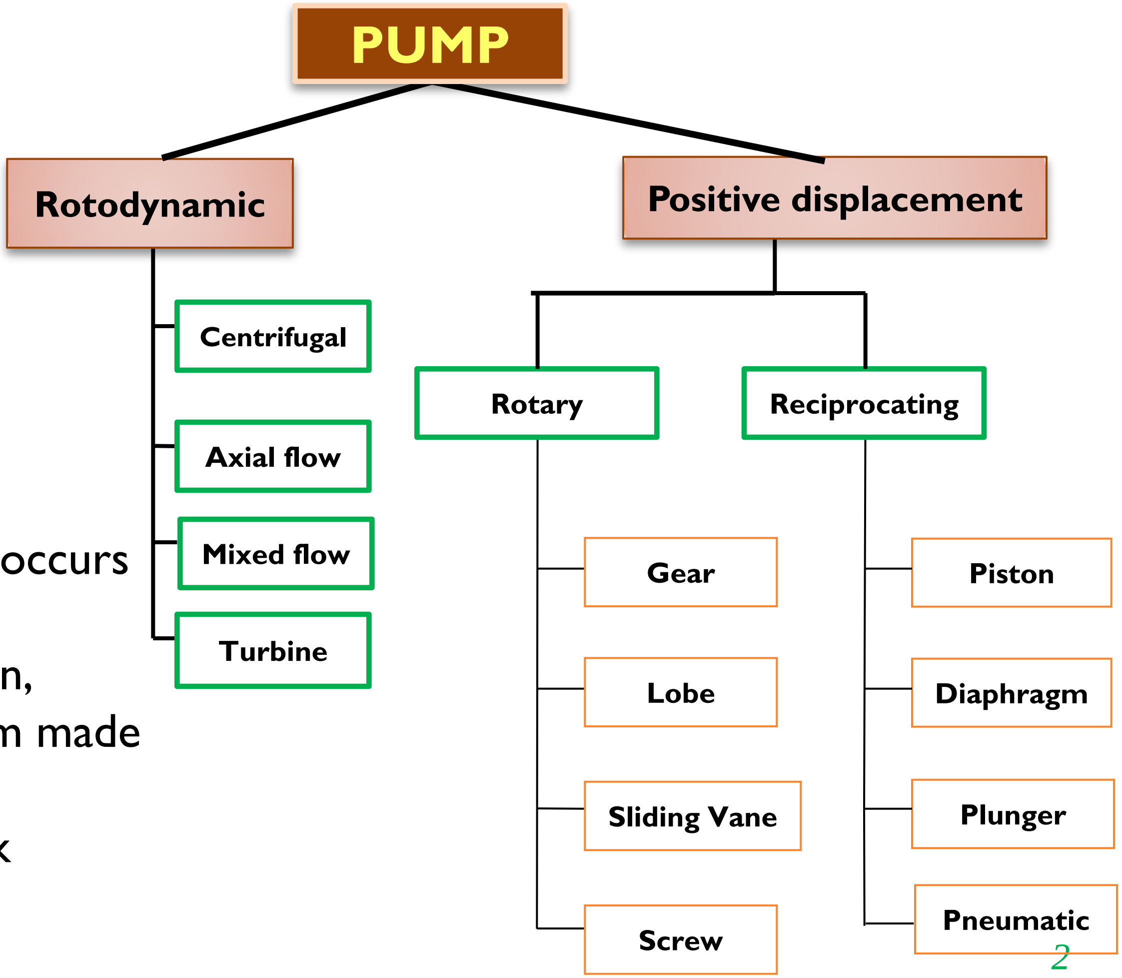
Rotary:- Rotating action occurs periodically.

Gear – comprises two gears in a housing with small radial end clearances. Used in lubrication system.

Lobe- Handles solids also. Used in paper and pulp industry.

Sliding Vane – comprises number of vanes.

Screw- three screw with housing is used with housing. Can create high pressure, uniform delivery, used to transfer lubricant.



Reciprocating:- Reciprocating action occurs periodically.

Piston – comprises a cylinder and piston,

Diaphram – comprises flexible diaphram made from rubber or rubberised febric

Plunger – comprises plunger, uses crank mechanism

Pneumatic – Handles compressed air.

CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

Gear Pump:

The crescent internal gear pump has an outer or rotor gear that is generally used to drive the inner or idler gear (Figure 1). The idler gear, which is smaller than the rotor gear, rotates on a stationary pin and operates inside the rotor gear. The gears create voids as they come out of mesh and liquid flows into the pump. As the gears come back into mesh, volumes are reduced and liquid is forced out the discharge port. Liquid can enter the expanding cavities through the rotor teeth or recessed areas on the head, alongside the teeth. The crescent is integral with the pump head and prevents liquids from flowing to the suction port from the discharge port.

External gear pumps are similar in pumping action to internal gear pumps in that two gears come into and out of mesh to produce flow (Figure 2). However, the external gear pump uses two identical gears rotating against each other. Each gear is supported by a shaft with bearings on both sides of each gear. Typically, all four bearings operate in the pumped liquid.

Because the gears are supported on both sides, external gear pumps are used for high pressure applications such as hydraulics.

Tighter internal clearances provide for a more reliable measure of liquid passing through a pump and for greater flow control. Because of this, external gear pumps are popular for precise transfer and metering applications involving polymers, fuels, and chemical additives

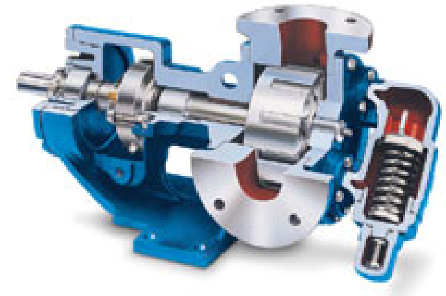


Fig 1

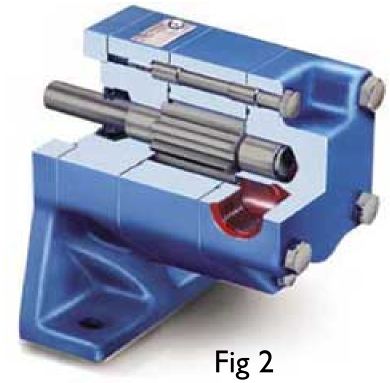
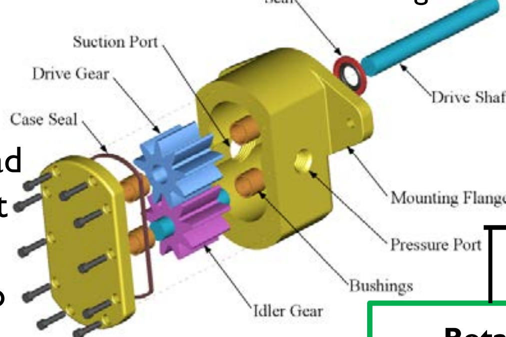
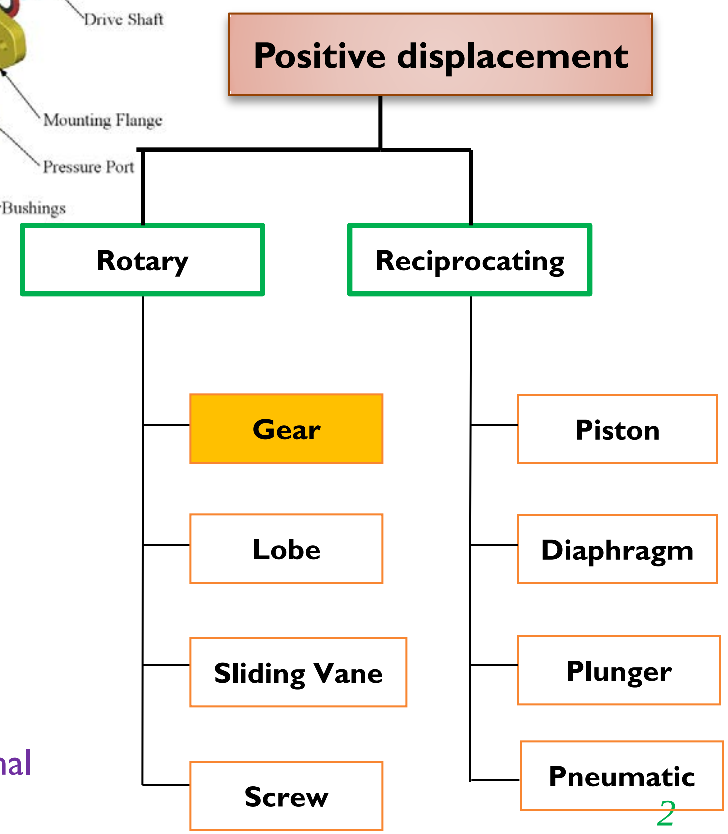


Fig 2



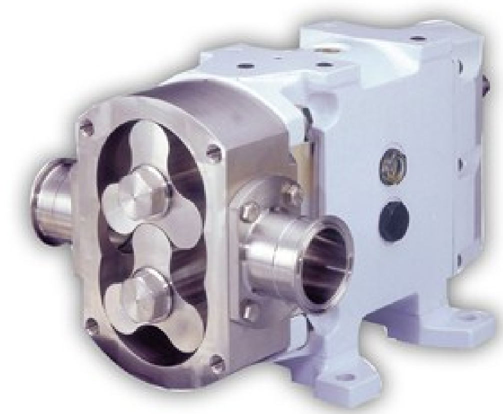


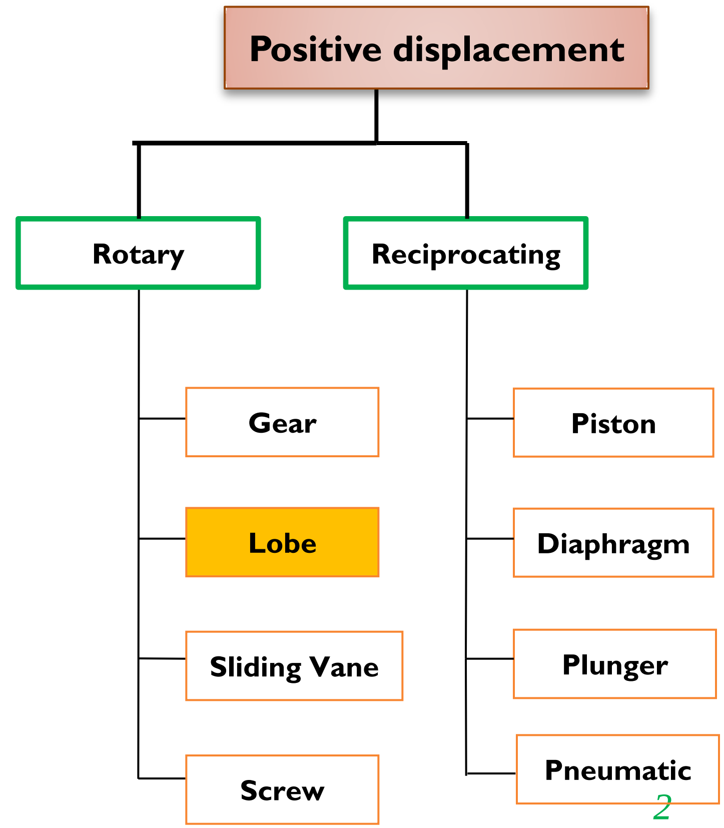
CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

Lobe Pump:

Lobe pumps are similar to external gear pumps in operation, except the pumping elements or lobes do not make contact. Lobe contact is prevented by external timing gears. Pump shaft support bearings are located in the timing gear case. Since the bearings are out of the pumped liquid, pressure is limited by bearing location and shaft deflection. There is not metal-to-metal contact and wear in abrasive applications is minimal. Use of multiple mechanical seals makes seal construction important.

Lobe pumps are frequently used in food applications, because they handle solids without damaging the pump.





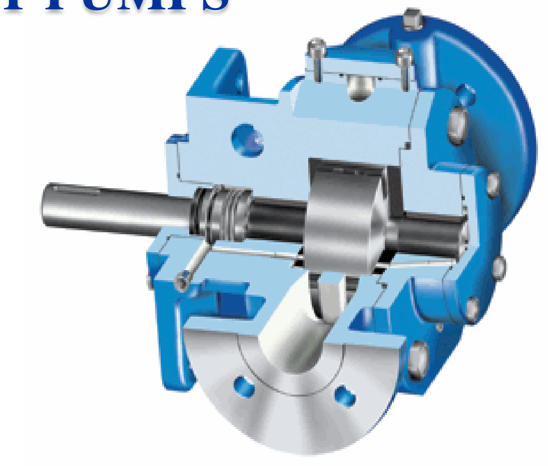
CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

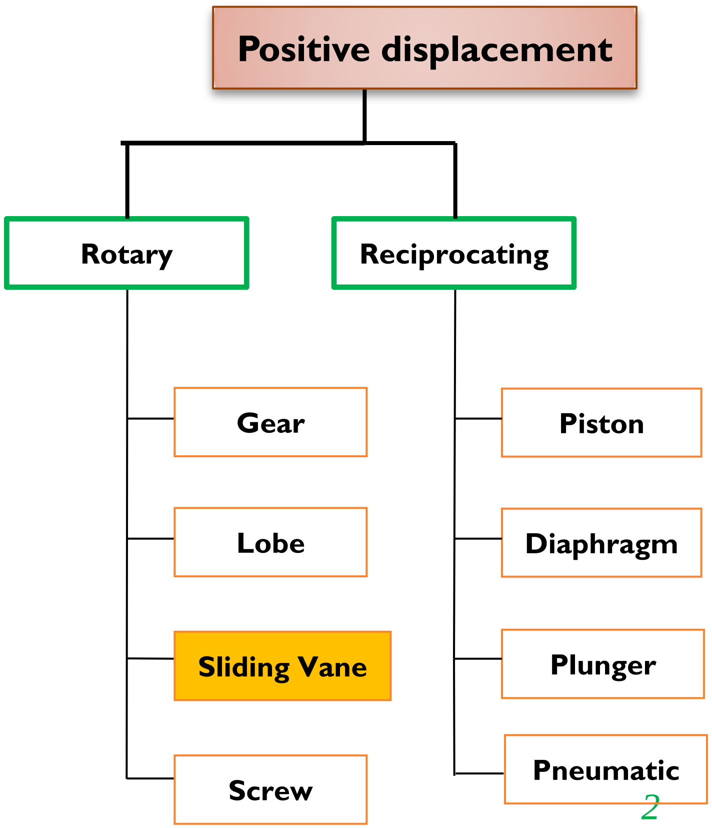
Sliding Vane Pump:

Sliding vane pumps operate quite differently from gear and lobe types. A rotor with radial slots, is positioned off-center in a housing bore. Vanes that fit closely in rotor slots slide in and out as the rotor turns. Vane action is aided by centrifugal force, hydraulic pressure, or pushrods. Pumping action is caused by the expanding and contracting volumes contained by the rotor, vanes, and housing.

Vanes are the main sealing element between the suction and discharge ports and are usually made of a nonmetallic composite material. Rotor bushings run in the pumped liquid or are isolated by seals.

The pumps work well with low-viscosity liquids that easily fill the cavities and provide good suction characteristics.

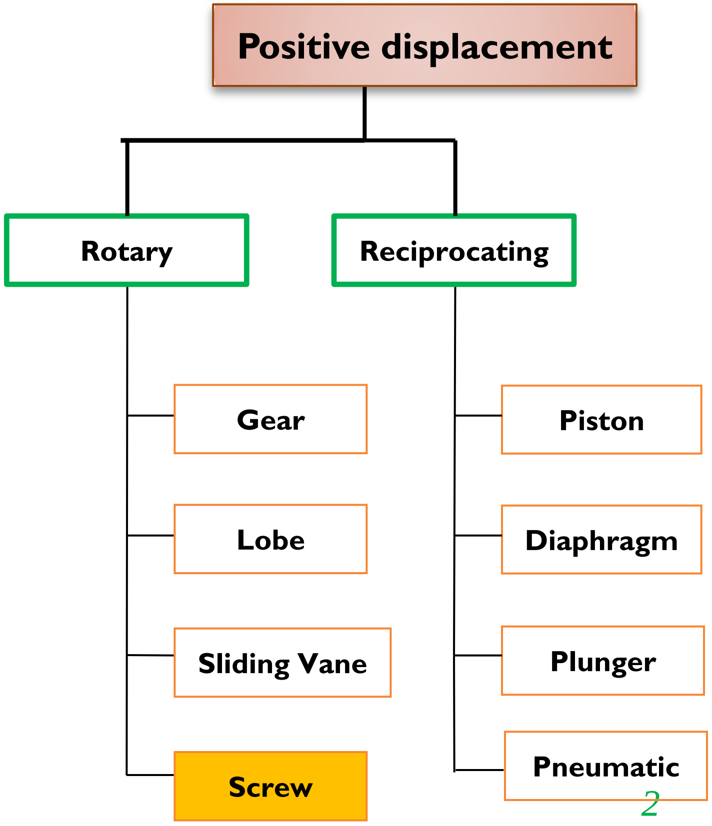




CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

Screw Pump:

Screw pumps are used for high flows at relatively low pressure and are used on board ships. It consist of two Archimedes screws that intermesh and are enclosed within the same chamber. Even though the efficiency is not high, the screw pumps have a low sound level. The major problem of screw pumps is that the hydraulic reaction force is transmitted in a direction that's axially opposed to the direction of the flow.



CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

Piston Pump:

Axial piston pumps use swash plate principle and is more compact in design (Fig 1). They use 'through-drive series mounted auxiliary rotating equipment based on their in-line design. Even though the pumps are easier and economical to design, they have a disadvantage that it is more sensitive to contamination. It is the most widely used variable displacement pump. The variable displacement pump by using various compensation techniques can continuously alter fluid discharge per revolution and system pressure based on load requirements, maximum pressure cut-off settings, horsepower/ratio control, and even fully electro proportional systems, requiring no other input than electrical signals. Thereby a huge amount of power can be saved by these pumps compare to other constant flow pumps in the systems.

Radial piston pumps are designed so that it can be used for high pressure and relatively small flows (Fig 2). The design is such that the plungers are connected to a floating ring. An eccentricity in the centre of rotation of the plungers is caused by this floating ring as they can be moved horizontally by a control lever. The amount of eccentricity can be controlled to vary the discharge. By shifting the eccentricity to the opposite side the suction and discharge can be totally reversed seamlessly. Therefore just like Swash plate pump, both quantity and direction can be varied in a radial piston pump.

Fig 1

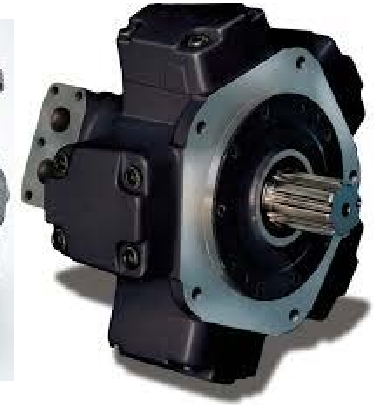
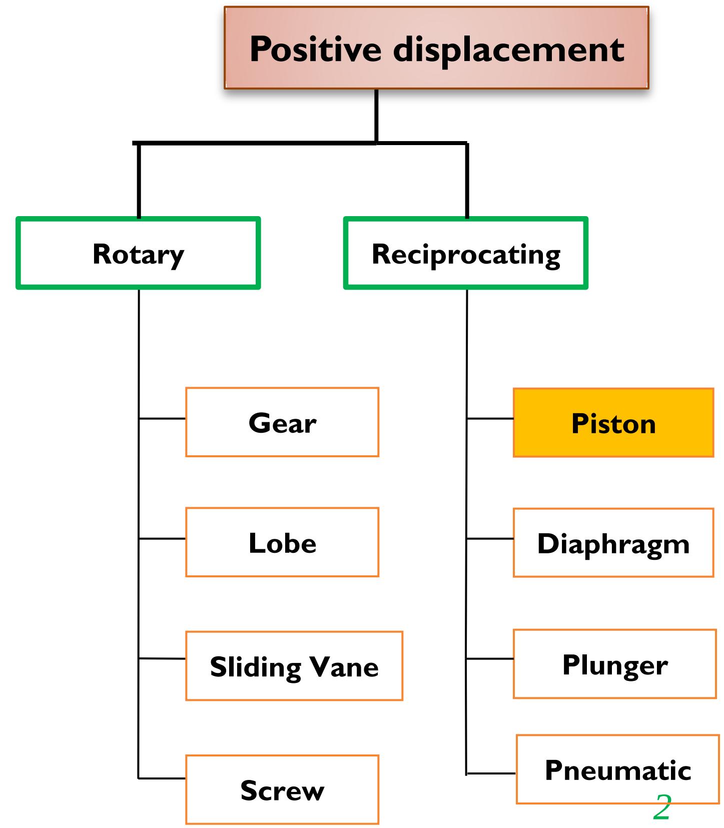


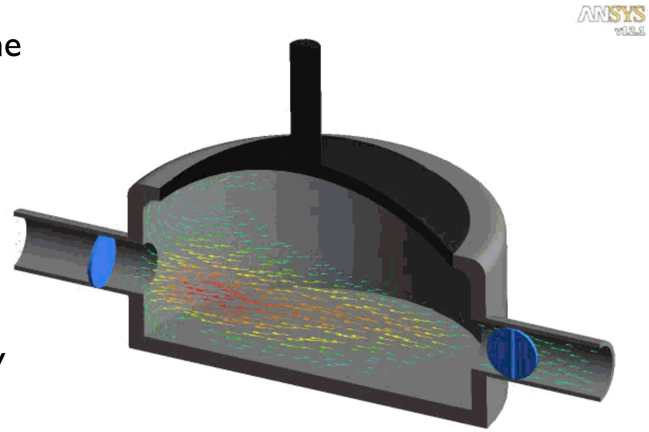
Fig 2

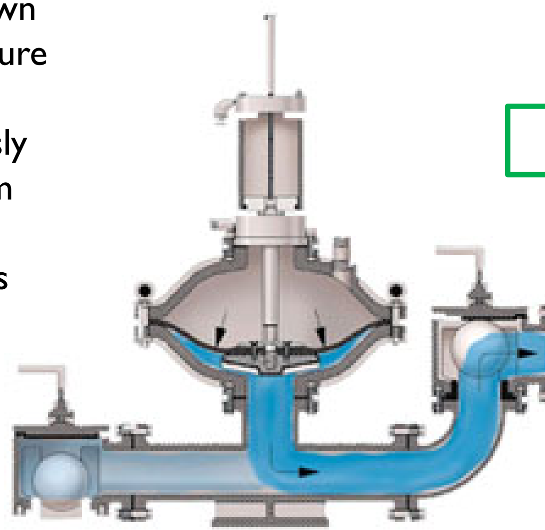


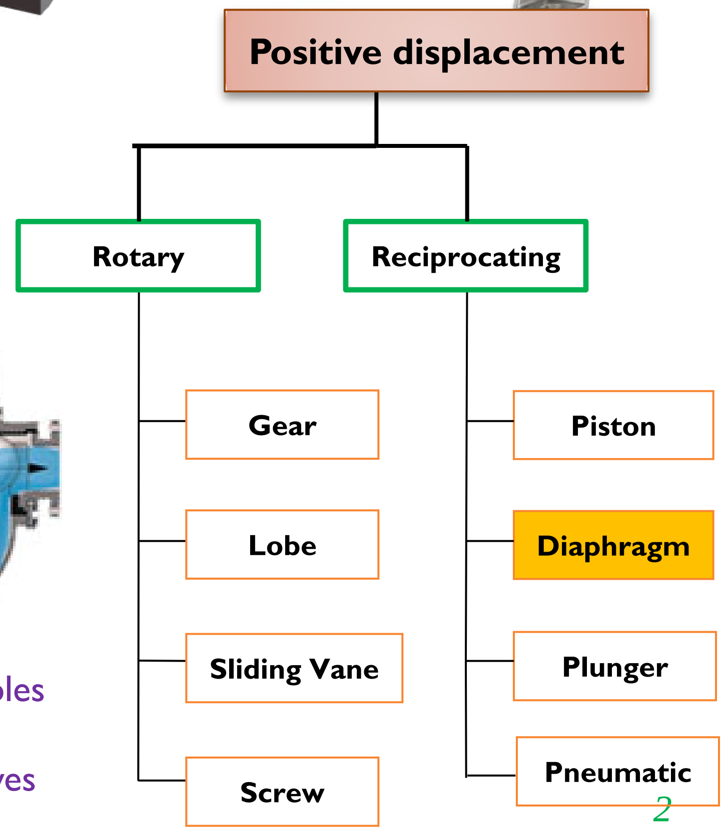
CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

Diaphragm Pump:

A diaphragm pump (also known as a Membrane pump, Air Operated Double Diaphragm Pump (AODD) or Pneumatic Diaphragm Pump) is a positive displacement pump that uses a combination of the reciprocating action of a rubber, thermoplastic or teflon diaphragm and suitable valves on either side of the diaphragm (check valve, butterfly valves, flap valves, or any other form of shut-off valves) to pump a fluid. When the volume of a chamber of either type of pump is increased (the diaphragm moving up), the pressure decreases, and fluid is drawn into the chamber. When the chamber pressure later increases from decreased volume (the diaphragm moving down), the fluid previously drawn in is forced out. Finally, the diaphragm moving up once again draws fluid into the chamber, completing the cycle. This action is similar to that of the cylinder in an internal combustion engine.







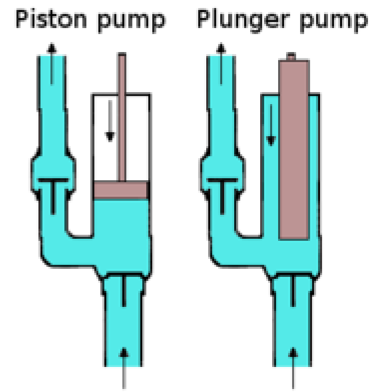
Diaphragm pumps will move nearly any kind of fluid. Some common examples of fluids pumped using a diaphragm pump are water, oil and acid.

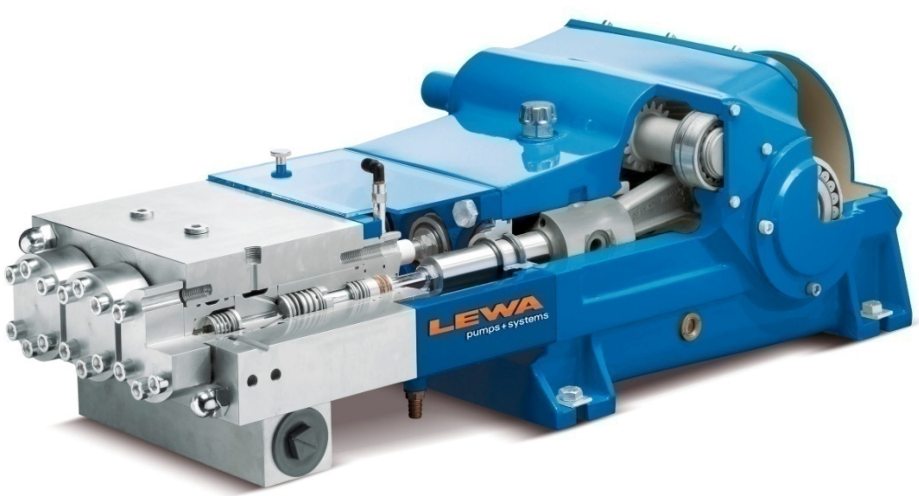
Diaphragm pumps are capable of pumping high viscosity fluids from adhesives and gear oils to hand lotions, surfactants and resins.

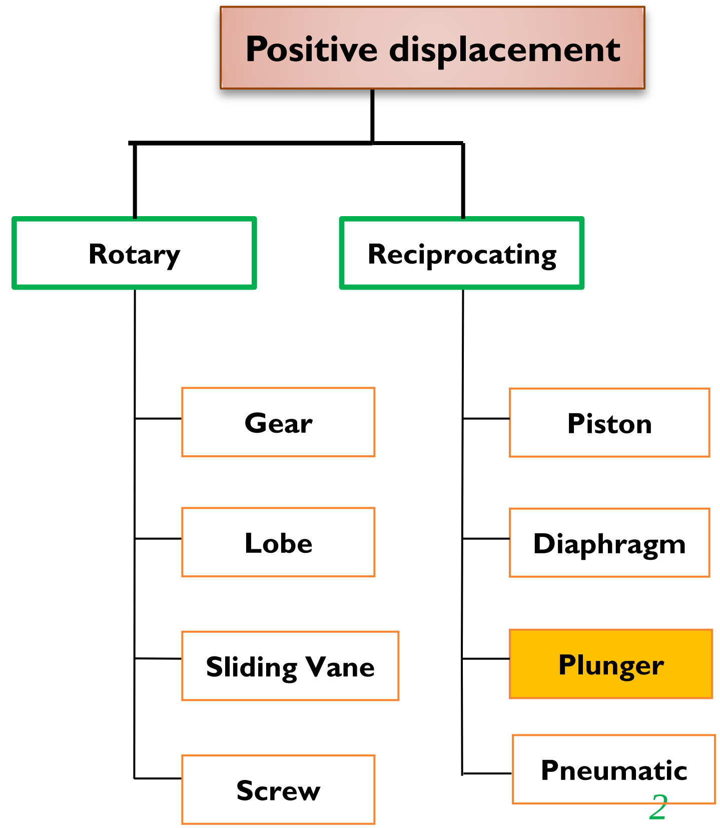
CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

Plunger Pump:

A plunger pump is a type of positive displacement pump where the high-pressure seal is stationary and a smooth cylindrical plunger slides through the seal. This makes them different from piston pumps and allows them to be used at higher pressures. This type of pump is often used to transfer municipal and industrial sewage. It is ideal to perform different offshore tasks also.







CHARACTERISTICS OF POSITIVE DISPLACEMENT PUMPS

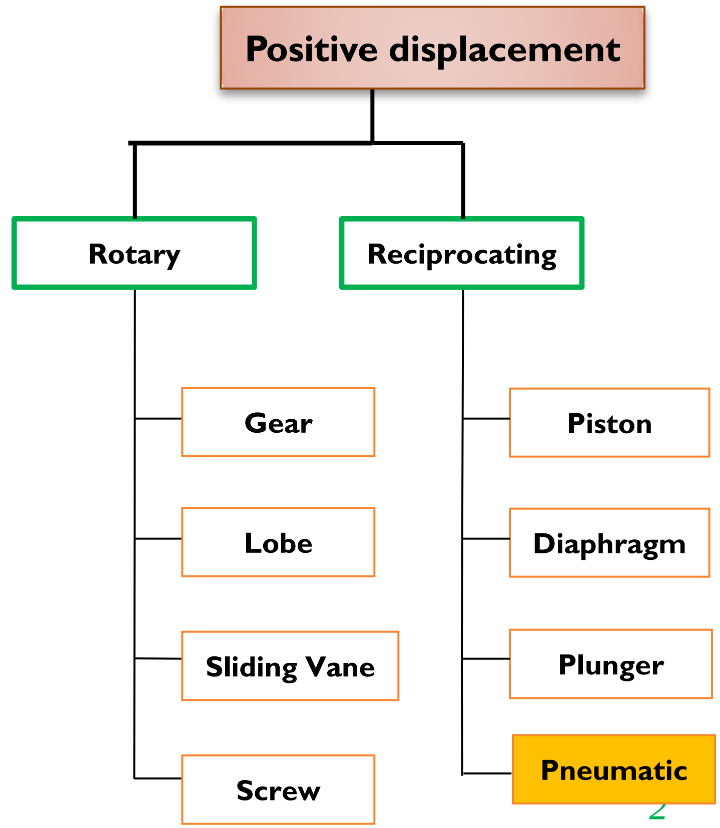
Pneumatic Pump:

Pneumatics are a branch of technology that uses the force of compressed gases to generate mechanical effects. Pneumatic pumps, in particular, use compressed air to create force that is used to move fluids through a piping system. Their system of operation is very similar to that of hydraulic pumps. Essentially, pneumatic pumps use air in the same way that hydraulic pumps use fluids. Both are capable of creating extremely amplified levels of pressure that can generate surprisingly large amounts of power.

These pumps use a double piston system, with one piston having a substantially larger diameter than the other. Between the two pistons are an airtight chamber filled with either liquid or another compressed gas. The compressed gas from the outside pushes on the larger-diameter piston, which in turn pushes against the gas or liquid in the intermediate chamber. Because there is no pressure lost between the larger piston and the smaller one, the smaller piston receives a highly amplified level of force, which can translate into powerful mechanical action.

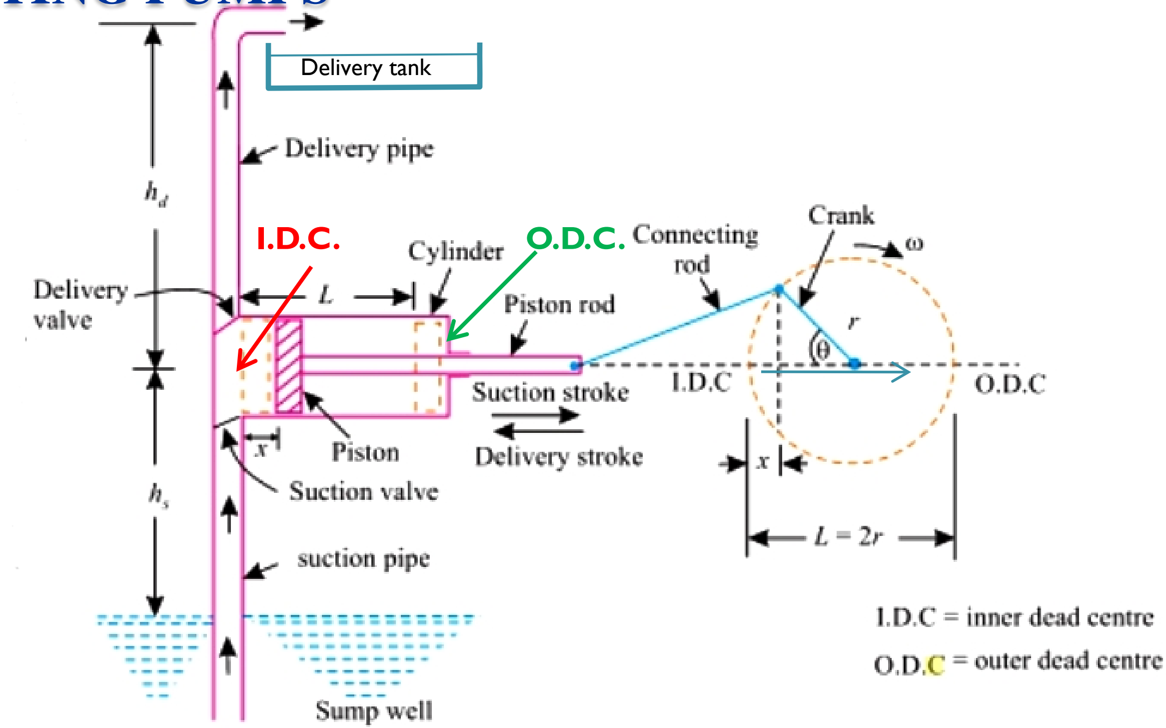
This pump is pneumatically powered, positive displacement, reciprocating injection pumps exhibit superior durability. Specifically designed for and proven in demanding oil and gas production applications, these pumps are used in many other industries, including pulp & paper, water treatment, fertilizer dosing, and food processing.

Pneumatic Grease Pump, which is a used for automatic feeding of oil or grease over the ball bearings, motor pulley and other moving components of vehicle.

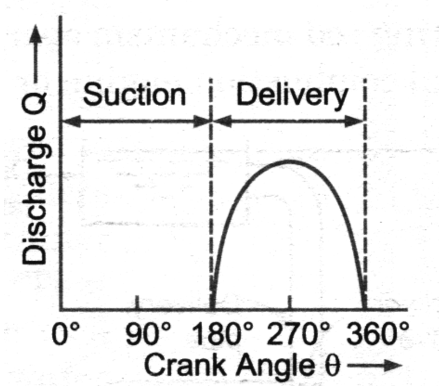


OPERATION OF RECIPROCATING PUMPS

- ✓ Piston or plunger, fitted with cylinder
- ✓ Suction and delivery pipe which are provided with non- return valve (suction and delivery valve)
- ✓ Crank and connecting rod mechanism operated by a power source which may be steam engine, I.C. Engine or motor.
- ✓ When crank turn clockwise, from Inner dead centre (I.D.C) to outer dead centre (O.D.C), rarefaction (reduction in density followed by partial vaccum, below P_{atm}) developed, thus it causes suction valve to open and liquid is forced from the sump to left side of the piston. When the crank is at O.D.C. Suction stroke is completed, left side is full of fluid.



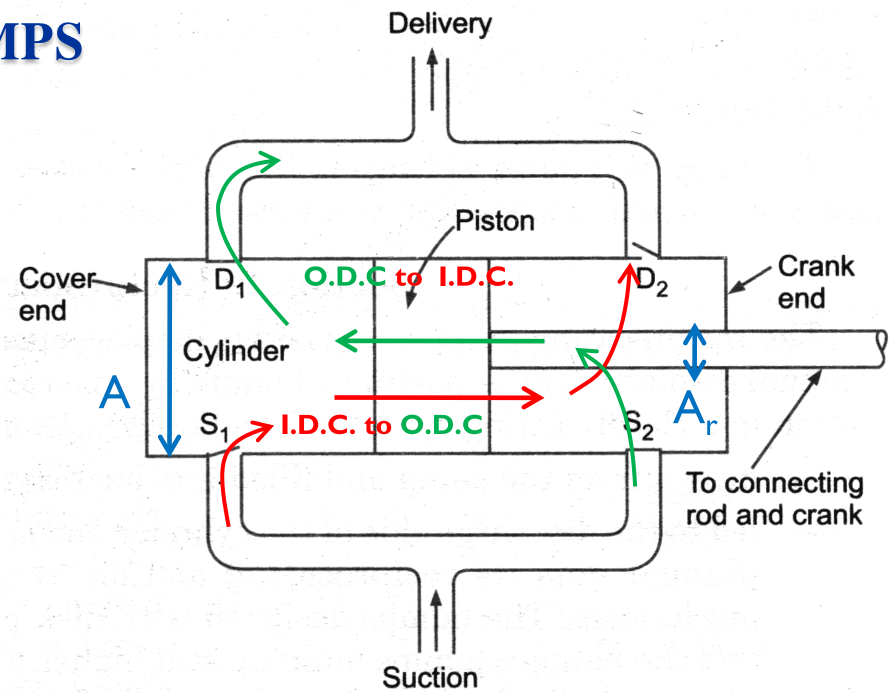
- ✓ When the crank turns from O.D.C to I.D.C. , high pressure developed in the cylinder, thus it causes suction valve to close and discharge valve to open.
- ✓ The delivery stroke is completed when crank occupy the I.D.C.

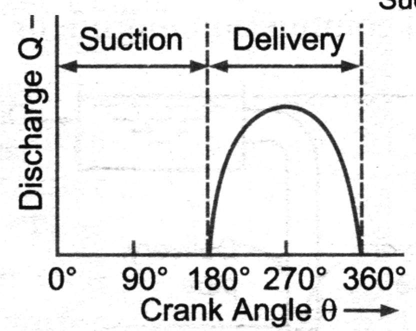


Sine curve

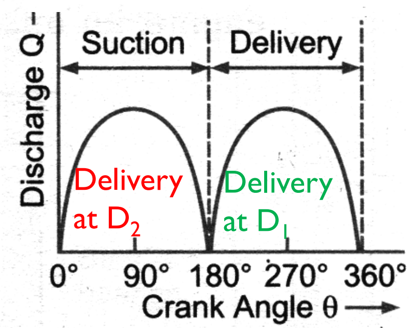
OPERATION OF RECIPROCATING PUMPS

- ✓ In a double acting pump, suction and delivery stroke occurs simultaneously.
- ✓ When crank turns from I.D.C to O.D.C, rarefaction occurs on left side of the piston, liquid is sucked in from the sump through the suction valve S_1. Simultaneously it build up pressure on the right side of the piston, so delivery valve D_2 opens and liquid passed out to discharge tank. It continue till crank reaches to O.D.C.
- ✓ When crank turns from O.D.C to I.D.C., suction occurs from S_2, and delivery occurs at D_1.

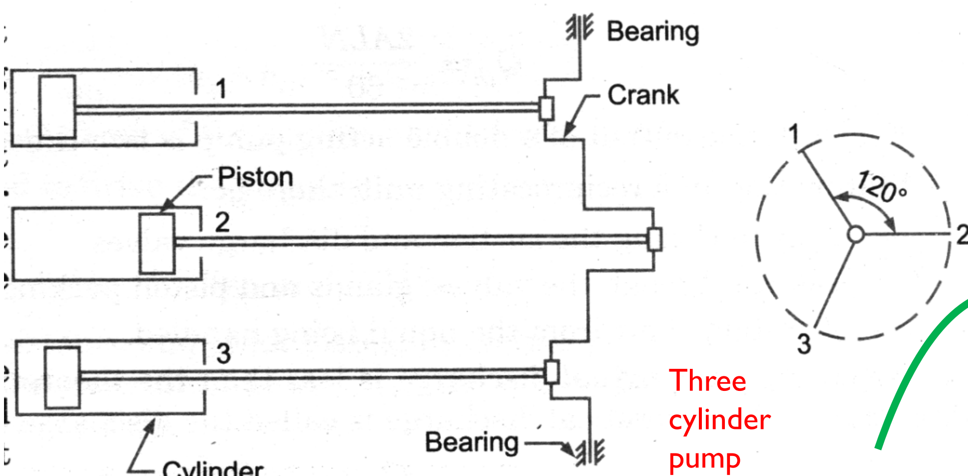




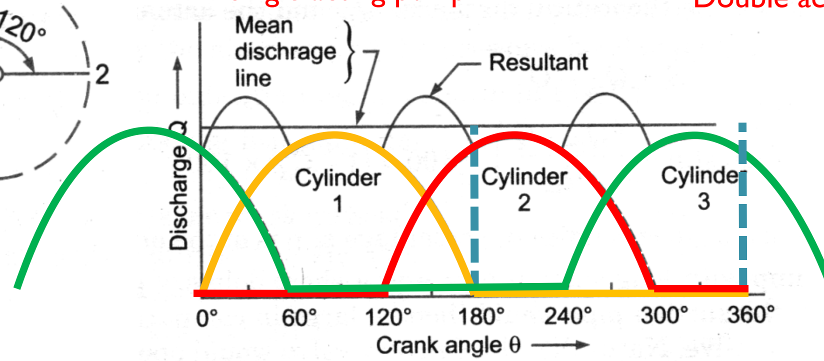
Single acting pump



Double acting pump



Three cylinder pump



Three cylinder pump

Discharge Coefficient, Volumetric Efficiency and Slip

A=Ap in next slides

A= cross section area of the piston, L = length of the stroke = 2r (crank radius), AL = Volume of the liquid sucked during suction stroke, Therefore theoretical discharge Q_{th} = ALN/60

For double acting pump

Where Ar = cross section area of the piston rod, $Q_{th} = \frac{ALN}{60} + \frac{(A - A_r)LN}{60}$

For double acting pump if Ar neglected $Q_{th} = \frac{2ALN}{60}$

Actually due to,

- Delay in closing the suction and delivery valve
- Leakage through valve, gland, piston
- Liberation of air from the liquid being handled

the actual discharge (Q) is less than Q_{th}

$C_d = \frac{Q}{Q_{th}}$ Value of C_d is known as coeff of discharge / volumetric efficiency η_v (85-98%)

$S = Q_{th} - Q$ S= Slip……Percentage slip is $\frac{(Q_{th} - Q)}{Q_{th}} \times 100$

Work and Power Input

Force on piston = $P_s A = \gamma h_s A$,where h_s=suction head, γh_s is vacuum pressure on the piston ***during suction***

Work done in suction side = $\gamma h_s AL$

Force on the piston =$\gamma h_d A$, where h_d=delivery head, γh_d is delivery pressure on the piston ***during delivery***

Work done in delivery side = $\gamma h_d AL$

Work done in a cycle = $\gamma AL(h_s + h_d)$, theoretical work per second = $\frac{\gamma ALN(h_s + h_d)}{60} = \gamma Q_{th} H$

Theoretical power required to drive the pump (P_{th}) $= \gamma Q_{th} H$

Actual Power = Theoretical power /η

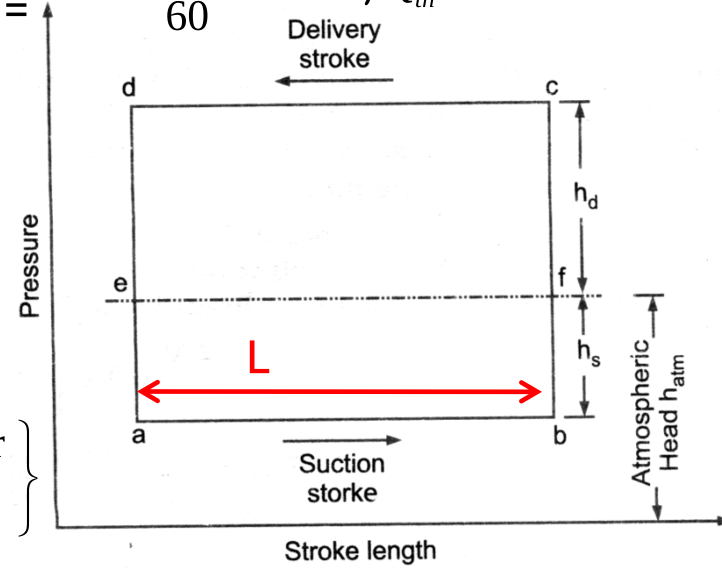
Theoretical Indicator Diagram

ef → atmospheric pressure head

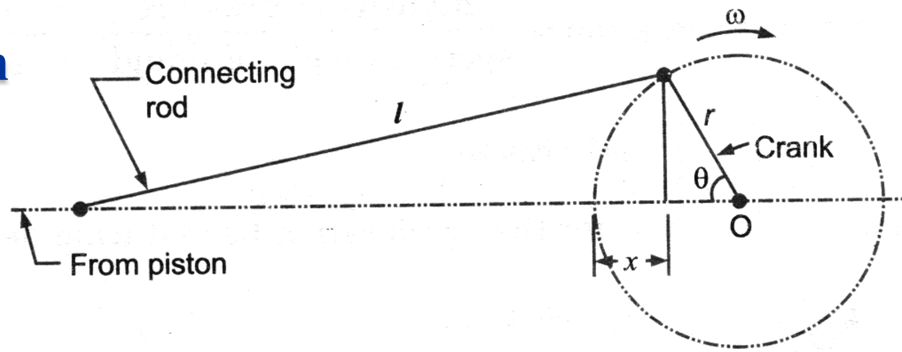
ab → pressure head in the cylinder during suction, less than h_{atm} or P_{atm} by h_s

cd → pressure head in the cylinder during delivery , more than h_{atm} or P_{atm} by h_d

Work input to drive the pump = $\frac{\gamma AN}{60} L(h_s + h_d) = \frac{\gamma AN}{60} \times \begin{Bmatrix} \text{indicator} \\ \text{plot area} \end{Bmatrix}$



Effect of Acceleration on Indicator Diagram



- ✓ Due to reciprocating action of piston, flow through single cylinder and single –acting pump is intermittent and uneven.
- ✓ It fluctuates from zero (at beginning of suction stroke) to a maximum value (at mid stroke) i.e. an acceleration head and back to zero (at end) i.e. Opposing effect.
- ✓ likewise, there will be an acceleration head at the start of delivery stroke and corresponding retardation head at termination.
- ✓ Thus reciprocating motion of the piston causes an acceleration during first half of each stroke, and retardation during late half of each stroke. That transmits a corresponding acceleration and retardation to liquid in suction and delivery pipes.
- ✓ If l/r is too large, motion of piston can be assumed to be simple harmonic motion.
- ✓ If crank turn through an angle θ (from IDC) during time t, with an angular velocity of ω rad/sec, then

$\theta = \omega t = \dfrac{2\pi N}{60} \times t$ The corresponding distance travelled by piston 'x' = $r - r(\cos\theta) = r(1 - \cos\omega t)$

The velocity of piston V_p = $\dfrac{dx}{dt} = \omega r \sin\omega t$ and acceleration a_p = $\dfrac{dVp}{dt} = \omega^2 r \cos\omega t$

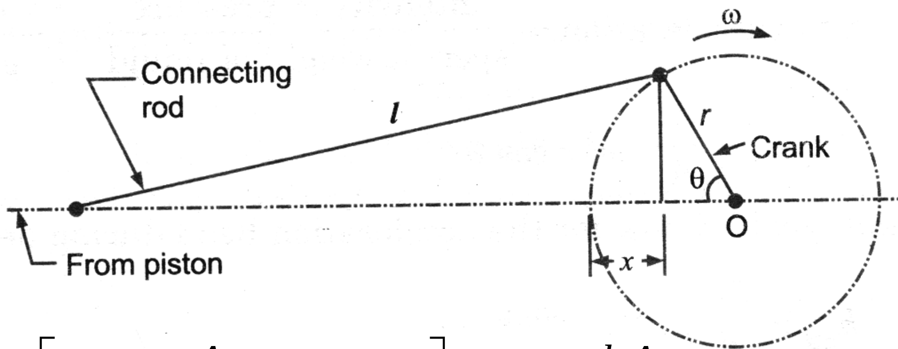
- ✓ During suction stroke, volume of liquid flowing from the suction pipe = volume of liquid in cylinder
- ✓ Therefore by continuity consideration ----

velocity of liquid in suction × area of suction pipe = Velocity of piston V_p × area of piston A_p

$$V_s \times A_s = V_p \times A_p \qquad V_s = V_p \times \frac{A_p}{A_s} = \frac{A_p}{A_s}\omega r \sin\omega t \qquad a_s = \frac{A_p}{A_s}\omega^2 r \cos\omega t$$

A=Ap in previous slides

Effect of Acceleration on Indicator Diagram



Mass of the liquid influenced by acceleration =

density× volume of liquid in suction pipe $= \rho A_s l_s = \dfrac{\gamma A_s l_s}{g}$

So, force required = mass × acceleration

$$= \frac{\gamma A_s l_s}{g} \times \frac{A_p}{A_s} \omega^2 r \cos \omega t$$

pressure intensity due to accln. in suction pipe= force / area = $\left[\dfrac{\gamma A_s l_s}{g} \times \dfrac{A_p}{A_s} \omega^2 r \cos \omega t\right] \times \dfrac{1}{A_s} = \dfrac{\gamma l_s A_p}{g A_s} \omega^2 r \cos \omega t$

$$= \frac{\gamma l_s A_p}{g A_s} \omega^2 r \cos \theta$$

So, **pressure head** due to acceleration in suction pipe (h_{as}) = Pr. Intensity / specific weight of fluid = $\boxed{\dfrac{l_s A_p}{g A_s} \omega^2 r \cos \theta}$

So, **pressure head** due to acceleration in delivery pipe (h_{ad}) = Pr. Intensity / specific weight of fluid = $\boxed{\dfrac{l_d A_p}{g A_d} \omega^2 r \cos \theta}$

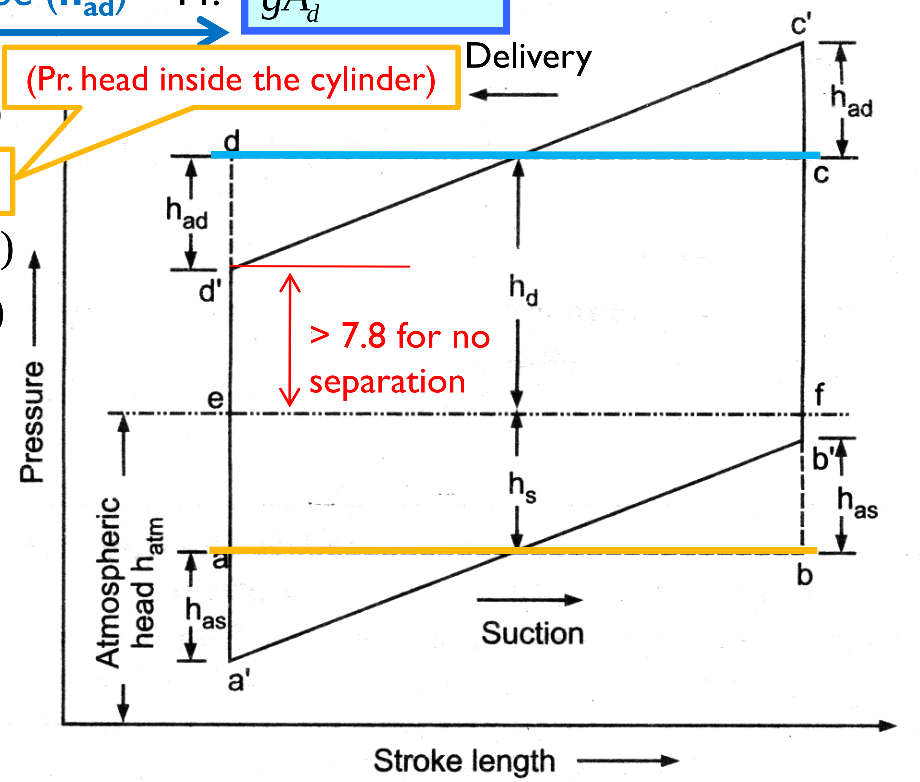
(Pr. head inside the cylinder)

Abs. Pr. Head at the beginning of suction $= h_{atm} - (h_s + h_{as})$

Abs. Pr. Head at the end of suction $= h_{atm} - \boxed{(h_s - h_{as})}$

Pr. Head at the beginning of delivery $= h_{atm} + (h_d + h_{ad})$

Pr. Head at the end of delivery $= h_{atm} + (h_d - h_{ds})$



Area of indicator diagram, i.e. Work done remains same. Inertia of fluid doesn't affect work, it affects variation of pressure.

To avoid flow separation absolute pressure head at delivery should not fall below vapour pressure.

$$h_{atm} + (h_d - h_{ds}) \geq h_{sep} = h_v$$
$$10.3 + (h_d - h_{ds}) \geq h_{sep} = 2.5$$
$$(h_d - h_{ds}) = 7.8$$

Effect of Friction on Indicator Diagram

Head Loss due to friction by Darcy Equation = $h_f = \dfrac{4flV^2}{2gd}$ $\quad V = \dfrac{A_p}{A}\omega r \sin\omega t \quad h_f = \dfrac{4fl}{2gd}\left(\dfrac{A_p}{A}\omega r \sin\omega t\right)^2$

Therefore variation of h_f and ɵ is parabolic

Beginning and end of suction, ɵ =0°,180°, sin ɵ =0, $h_f = 0$ Mid of suction, ɵ = 90°, sin ɵ =1 , $h_{fs} = \dfrac{4fl_s}{2gd_s}\left(\dfrac{A_p}{A_s}\omega r\right)^2$

Beginning and end of delivery, ɵ =180°,360°, sin ɵ =0 , $h_f = 0$ Mid of delivery, ɵ = 270°, sin ɵ =1, $h_{fd} = \dfrac{4fl_d}{2gd_d}\left(\dfrac{A_p}{A_d}\omega r\right)^2$

Area of indicator diagram during Suction

$$= h_s \times L + \boxed{\frac{2}{3}h_{fs} \times L} = \left(h_s + \frac{2}{3}h_{fs}\right) \times L$$

Area of parabolic segment

Area of indicator diagram during delivery

$$= \left(h_d + \frac{2}{3}h_{fd}\right) \times L$$

Total area of indicator diagram (ID)

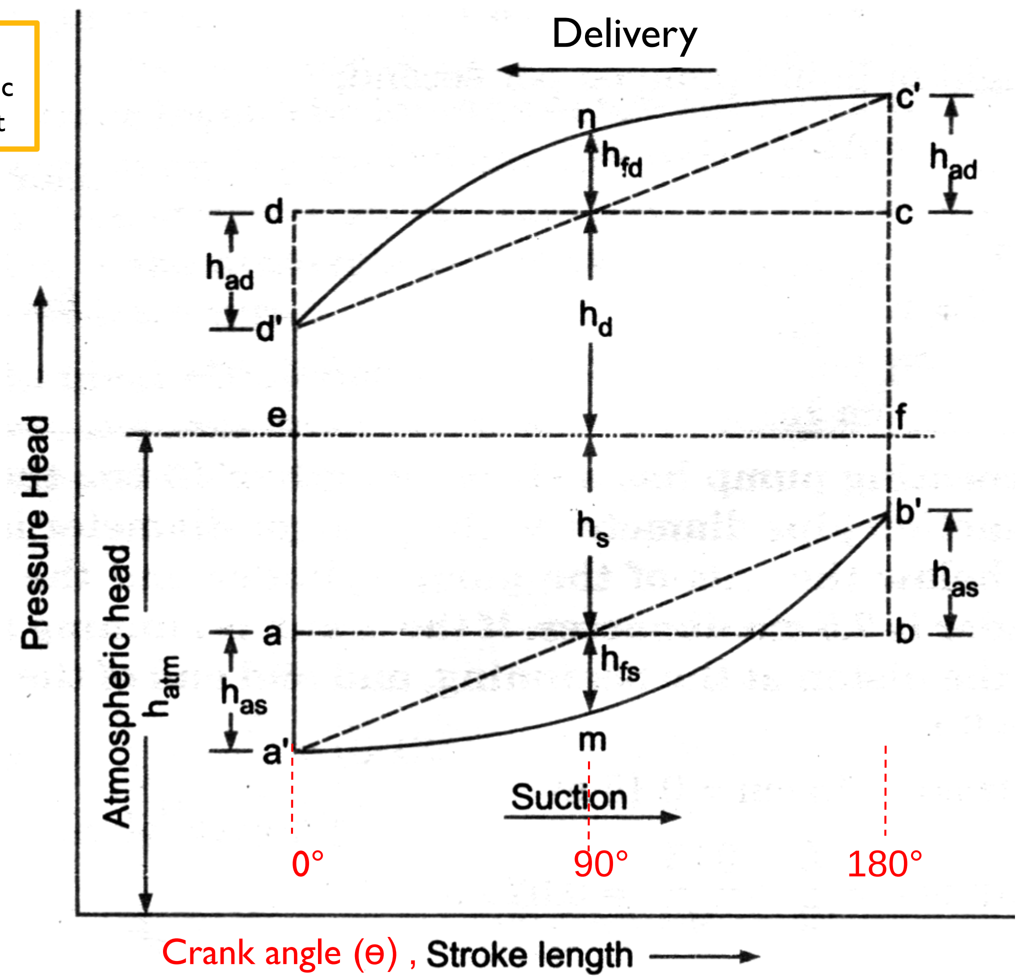
$$= \left(h_s + h_d + \frac{2}{3}h_{fs} + \frac{2}{3}h_{fd}\right) \times L$$

Average work done = $\dfrac{\gamma AN}{60}$ × area of ID

$$= \frac{\gamma ANL}{60}\left(h_s + h_d + \frac{2}{3}h_{fs} + \frac{2}{3}h_{fd}\right)$$

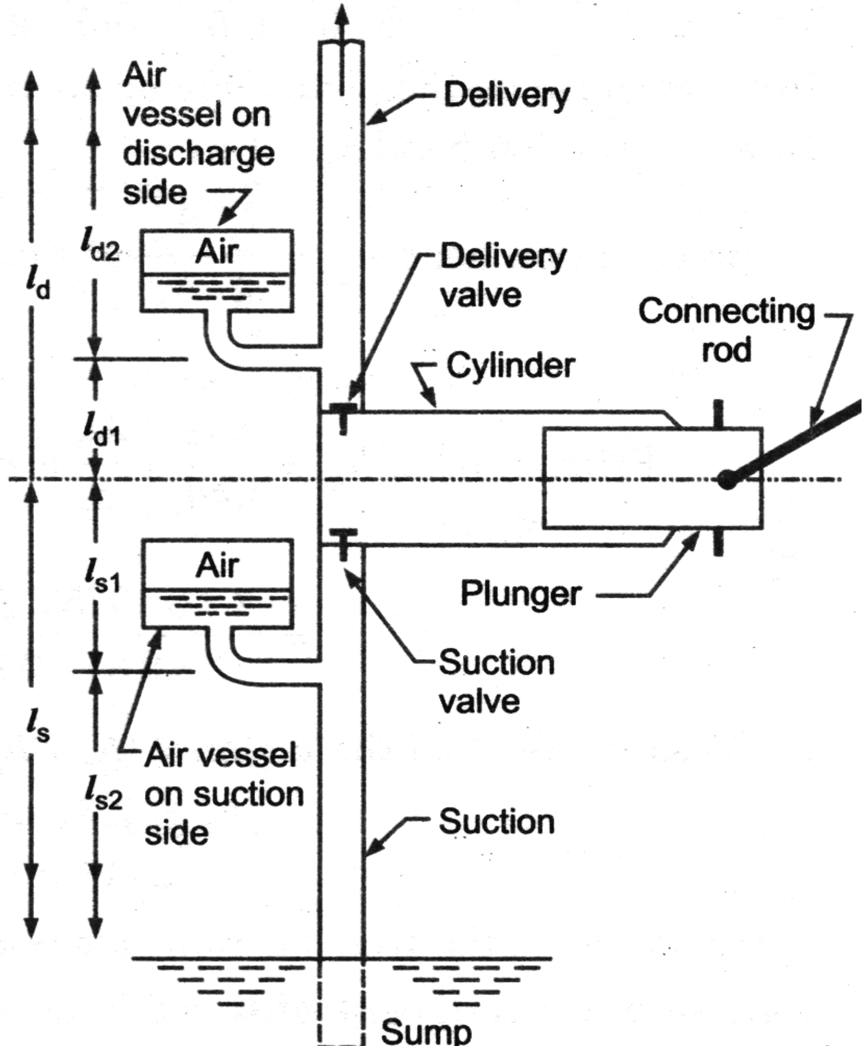
Weight of liquid pumped / sec for single acting pump $\quad W = \dfrac{\gamma ANL}{60}$

Weight of liquid pumped / sec for double acting pump $\quad W = 2\dfrac{\gamma ANL}{60}$



Effect of Air Vessel

The flow from a single cylinder and single acting reciprocating pump, working with simple harmonic motion, is of pulsating and intermittent in nature. For smooth running, continuous flow is desired during both suction and delivery stroke. This task is accomplished by air vessel on the suction and delivery side.



Without Air Vessel

Variation of friction loss h_f during a stroke is parabolic. Therefore area representing work done by pump per stroke against friction is 2/3×base×height .

Work done or power expended during a stroke against friction at θ= 90° / 270 ° (suction / delivery)

$$P_1 = \frac{\gamma AN}{60} \times L\left(\frac{2}{3}h_f\right) \text{ where } h_f = \frac{4fl}{2gd}\left(\frac{A_p}{A}\omega r\right)^2$$

With Air Vessel

When air vessels are employed, flow is assumed to be uniform and equal to mean flow velocity.

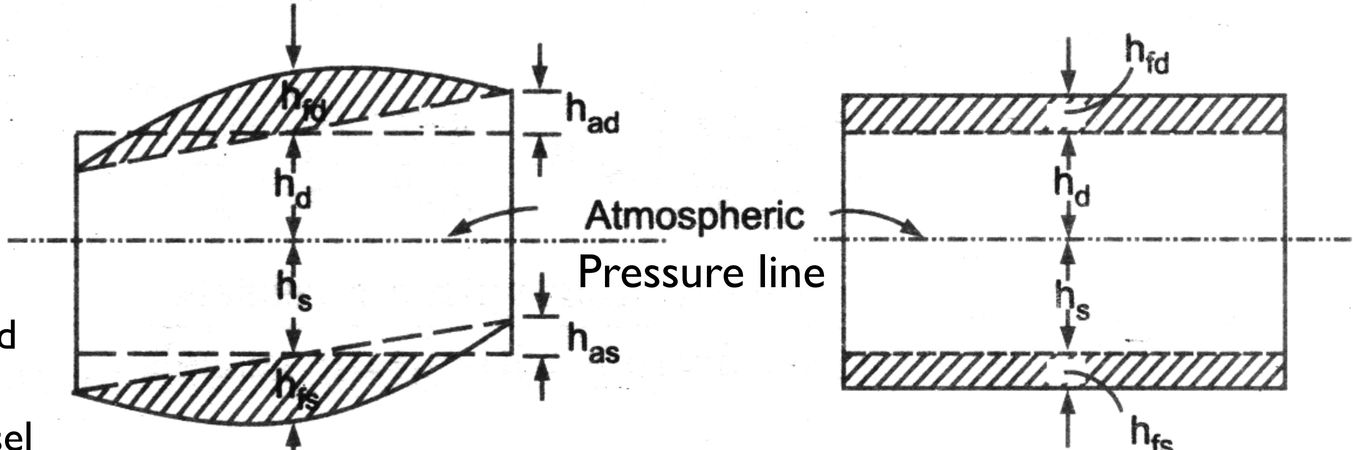
$$Q = A_p \times L \times \frac{N}{60} = A_p \times 2r \times \frac{\omega}{2\pi}$$

Therefore, flow velocity in the pipe $= A_p \times 2r \times \frac{\omega}{2\pi} \Big/ A$

Loss of head due to friction per stroke $h_f = \frac{4fl}{2gd}\left(\frac{A_p}{A} \times \frac{\omega r}{\pi}\right)^2$

Due to fitting of Air vessel, the head loss due to friction is independent of θ, and hence the indicator plot is a rectangle

Effect of Air Vessel



Indicator diagram without or with air vessel

Area of Indicator diagram

$$L\times\frac{4fl}{2gd}\left(\frac{A_p}{A}\times\frac{\omega r}{\pi}\right)^2$$

Work done or power expended to over come the friction in a single acting pump with air vessel

$$P_2=\frac{\gamma AN}{60}\times L\times\frac{4fl}{2gd}\left(\frac{A_p}{A}\times\frac{\omega r}{\pi}\right)^2$$

$$\frac{P_2}{P_1}=\frac{\dfrac{\gamma AN}{60}\times L\times\dfrac{4fl}{2gd}\left(\dfrac{A_p}{A}\times\dfrac{\omega r}{\pi}\right)^2}{\dfrac{\gamma AN}{60}\times L\times\left(\dfrac{2}{3}\times\dfrac{4fl}{2gd}\left(\dfrac{A_p}{A}\omega r\right)^2\right)}$$

$$P_2/P_1=3/2\pi^2$$

Percentage of Work saved in pipe friction by fitting air vessel $(P_1-P_2)/P_1=1-3/2\pi^2=0.848=84.8\ \%$

For double acting pump, Percentage of Work saved in pipe friction by fitting air vessel is 39.2 %

$$P_1=2\frac{\gamma AN}{60}\times L\times\left(\frac{2}{3}\times\frac{4fl}{2gd}\left(\frac{A_p}{A}\omega r\right)^2\right)$$

Therefore, flow velocity in the pipe $=2\times A_p\times 2r\times\frac{\omega}{2\pi}\Big/A$

$$P_2=2\frac{\gamma AN}{60}\times L\times\frac{4fl}{2gd}\left(\frac{2A_p}{A}\times\frac{\omega r}{\pi}\right)^2$$

$$(P_1-P_2)/P_1=1-(3/2)\times(4/\pi^2)=0.392=39.2\ \%$$