

ME 221

ELEMENTS OF FLUID MECHANICS & MACHINERY

RECIPROCATING PUMP

LECTURE 6

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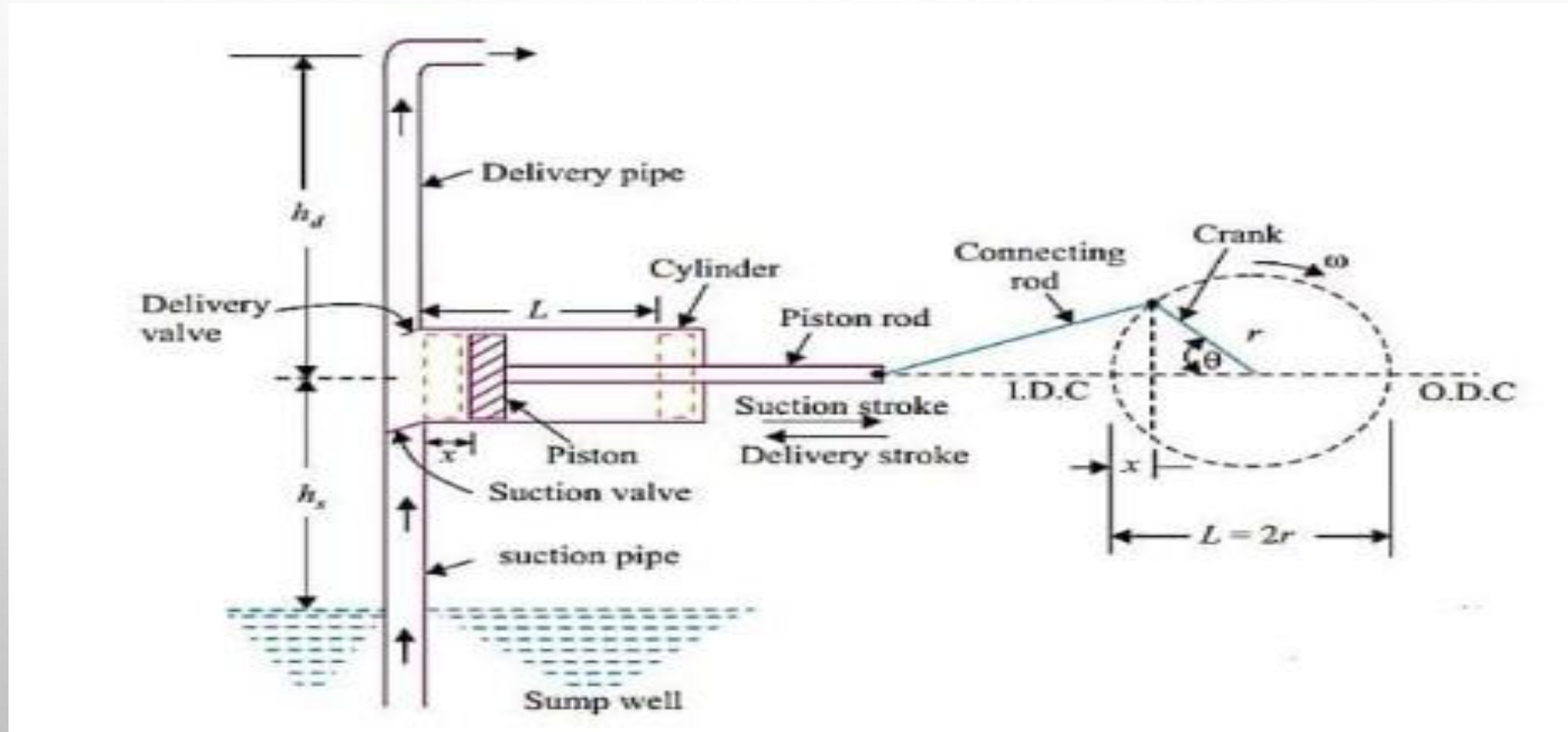
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INTRODUCTION

- ❖ Reciprocating pumps are **positive displacement** pump, i.e. initially, a small quantity of liquid is taken into a chamber and is physically displaced and forced out with pressure by a moving mechanical elements.
- ❖ The reciprocating pumps are **used** where relatively **small quantity** (fixed) of liquid per unit time is to be transferred but at relatively higher **pressure**.
- ❖ For **industrial** purposes, they have become **obsolete** due to their **high initial** and **maintenance costs** as compared to centrifugal pumps.
- ❖ These are also useful where **high heads** are required with **small discharge**, as **oil drilling** operations.

MAIN COMPONENTS



MAIN COMPONENTS

The **main components** of a reciprocating pumps are as follows:

1. Suction pipe and suction valve
2. Delivery pipe and delivery valve
3. Cylinder
4. Piston and piston rod
5. Strainer and Air vessel

MAIN COMPONENTS

- ❖ Through **suction valve**, liquid can only be admitted into the cylinder and through the **delivery valve**, liquid can only be discharged into the delivery pipe
- ❖ At the cylinder ends of these pipes, **non-return** valves are provided. A non-return valve allows the liquid to pass in only one direction
- ❖ The cylinder is connected to the sump by a suction pipe and to the delivery tank by a delivery pipe
- ❖ A reciprocation pumps consists of a **plunger** or a **piston** that moves forward and backward inside a cylinder with the help of a connecting rod and a crank. The crank is rotated by an external source of power

MAIN COMPONENTS

- ❖ **Strainer** is provided at the end of suction pipe to prevent the entrance of solids from water source into the cylinder
- ❖ **Air vessels** are connected to both suction and delivery pipes to eliminate the frictional head and to give uniform discharge rate

WORKING PRINCIPLE

- ❖ When the **power source** is connected to crank, the crank will start rotating and connecting rod also displaced along with crank
- ❖ The piston connected to the connecting rod will move in linear direction. If crank moves **outwards** then the piston moves towards its right and create **vacuum** in the cylinder
- ❖ This **vacuum** causes suction valve to open and liquid from the source is forcibly sucked by the suction pipe into the cylinder
- ❖ When the crank moves **inwards** or towards the cylinder, the piston will move towards its left and compresses the liquid in the cylinder

WORKING PRINCIPLE

- ❖ Now, the pressure makes the delivery valve to open and liquid will discharge through delivery pipe
- ❖ When piston reaches its extreme left position whole liquid present in the cylinder is delivered through delivery valve
- ❖ Then again the crank rotate outwards and piston moves right to create suction and the whole process is repeated
- ❖ Generally the above process can be observed in a **single acting** reciprocating pump where there is only one delivery stroke per one revolution of crank. But when it comes to **double acting** reciprocating pump, there will be two delivery strokes per one revolution of crank

CLASSIFICATION

❖ According to use of piston sides

➤ **Single acting Reciprocating Pump:**

If there is only **one suction** and **one delivery pipe** and the liquid is **filled only on one side** of the piston, it is called a single-acting reciprocating pump.

➤ **Double acting Reciprocating Pump:**

A double-acting reciprocating pump has **two suction** and **two delivery pipes**, liquid is receiving on **both sides of the piston** in the cylinder and is delivered into the respective delivery pipes.

CLASSIFICATION

❖ According to number of cylinder

➤ Single cylinder pump

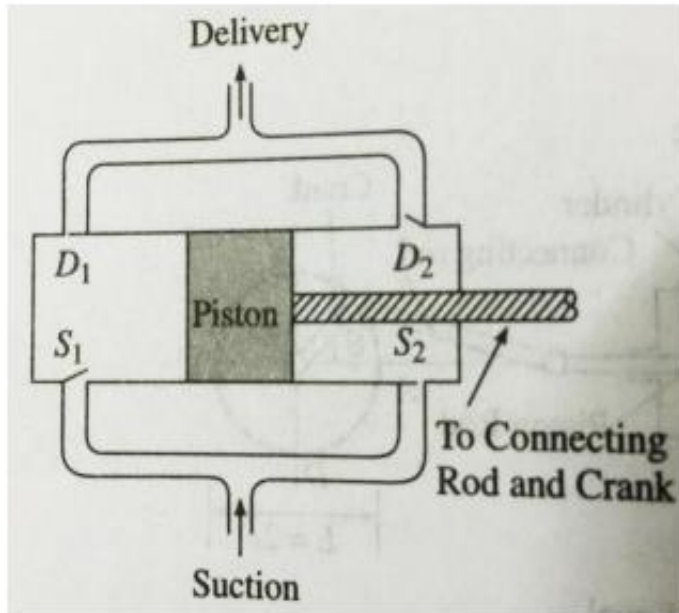
A single-cylinder pump can be either single or double acting

➤ Double cylinder pump (or two throw pump)

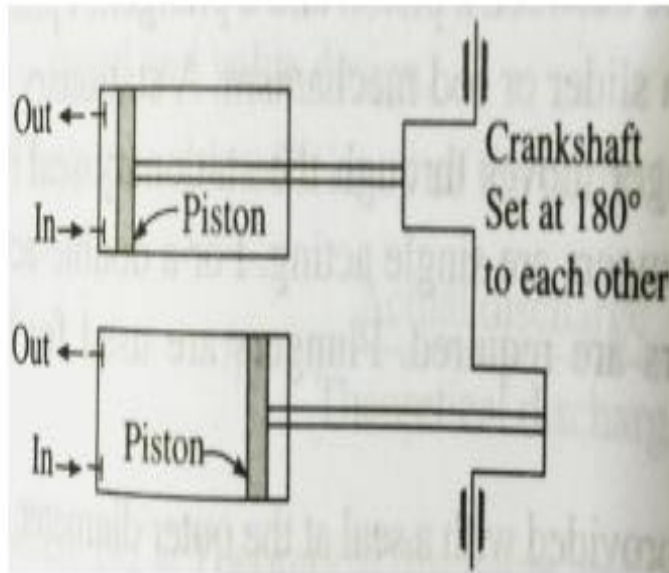
A double cylinder or two throw pump consist of two cylinders connected to the same shaft.

➤ Triple cylinder pump (three throw pump)

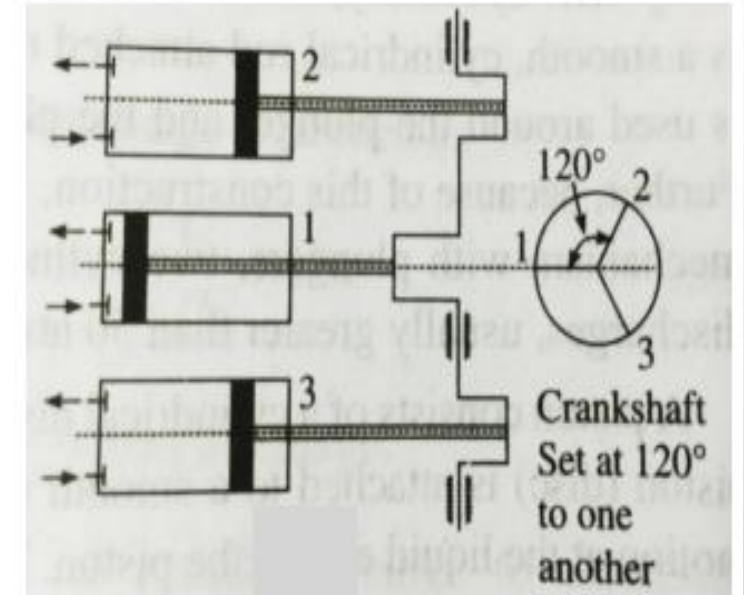
A triple-cylinder pump or three throw pump has three cylinders, the cranks of which are set at 120° to one another. Each cylinder is provided with its own suction pipe, delivery pipe and piston.



Double Acting Reciprocating Pump



Double Cylinder Reciprocating Pump



Triple Cylinder Reciprocating Pump

There can be **four-cylinder** and **five cylinder** pumps also, the cranks of which are arranged accordingly.

DISCHARGE THROUGH A RECIPROCATING PUMP

Let A = cross sectional area of cylinder

r = crank radius

N = rpm of the crank

L = stroke length ($2r$)

Discharge through pump per second = Area x stroke length x rpm/60

$$Q_{th} = \frac{ALN}{60}$$

This will be the discharge when the pump is **single acting**.

DISCHARGE THROUGH A RECIPROCATING PUMP

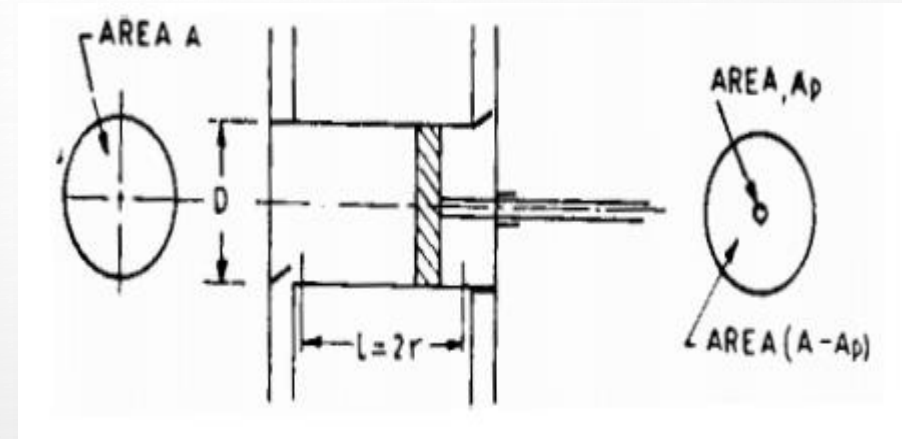
Discharge in case of **double acting** pump

Discharge/Second,

$$Q_{th} = \frac{ALN}{60} + \frac{(A-A_p)LN}{60}$$

$$Q_{th} = \frac{(2A - A_p)LN}{60}$$

where, A_p = area of cross section of piston rod



DISCHARGE THROUGH A RECIPROCATING PUMP

However, if **area** of the **piston rod** is **neglected**

$$\text{Discharge/Second} = \frac{2ALN}{60}$$

- ❖ Thus discharge of a double-acting reciprocating pump is **twice** than that of a single-acting pump.
- ❖ Owing to **leakage** losses and **time delay** in closing the valves, actual discharge Q_a is usually lesser than the theoretical discharge Q_{th} .

SLIP

Slip of a pump is defined as the **difference** between the theoretical discharge and actual discharge of the pump. The actual discharge of a pump is less than the theoretical discharge due to leakage.

Mathematically, $\text{slip} = Q_{th} - Q_a$

Slip can also be expressed in terms of % and given by

$$\begin{aligned}\% \text{slip} &= \frac{Q_{th} - Q_a}{Q_{th}} \times 100 \\ &= \left(1 - \frac{Q_a}{Q_{th}}\right) \times 100 = (1 - C_d) \times 100\end{aligned}$$

SLIP

Here C_d is known as **co-efficient of discharge** and is defined as the ratio of the actual discharge to the theoretical discharge.

$$C_d = \frac{Q_a}{Q_{th}}$$

Value of C_d when expressed in percentage is known as **volumetric efficiency** of the pump. Its value ranges between **95-98 %**. Percentage **slip** is of the order of **2%** for pumps in good conditions.

NEGATIVE SLIP

It is not always that the actual discharge is lesser than the theoretical discharge. In case of a reciprocating pump with **long suction pipe, short delivery pipe** and **running at high speed**, inertia force in the suction pipe becomes large as compared to the pressure force on the outside of delivery valve. This opens the delivery valve even before the piston has completed its suction stroke. Thus some of the water is pushed into the delivery pipe before the delivery stroke is actually commenced. This way the actual discharge becomes more than the theoretical discharge.

NEGATIVE SLIP

Thus co-efficient of discharge increases from one and the slip becomes negative.

Negative slip **occurs** when delivery pipe is short, suction pipe is long and pump is running at high speed.

POWER INPUT AND OUTPUT

Consider a **single acting** reciprocating pump.

Let

$h_s =$ Suction head or difference in level between centre line of cylinder and the sump.

$h_d =$ Delivery head or difference in between centre line of cylinder and the outlet of delivery pipe.

$$H_{st} = \text{Total static head} = h_s + h_d$$

Theoretical **work done** by the pump per second (Hydraulic power output)

$$= Q_{th} \gamma H_{st} = \frac{ALN}{60} \times \gamma \times (h_s + h_d)$$

POWER INPUT AND OUTPUT

Theoretical **Power input** to the pump = Theoretical **work done** by the pump per second = $\frac{ALN}{60} \times \gamma \times (h_s + h_d)$

However, due to the **leakage** and **frictional** losses, actual power input will be more than the theoretical power.

Let η = Efficiency of the pump.

Then **actual power** input to the pump = $\frac{1}{\eta} \frac{ALN}{60} \times \gamma \times (h_s + h_d)$

Problem 1:

A single-acting reciprocating pump discharge $0.018 \frac{m^3}{s}$ of water per second when running at 60 rpm. Stroke length is 500 mm and the diameter of the piston is 220 mm. If the total lift is 15 m, determine:

- a) Theoretical discharge of the pump
- b) Slip and percentage slip of the pump
- c) Co-efficient of discharge
- d) Theoretical Power required for running the pump

Problem 2:

A three-throw reciprocating pump is delivering $0.1 \frac{m^3}{s}$ of water against a head of 100 m. Diameter and stroke length of the cylinder are 250 mm and 500 mm respectively. Friction losses amount to 1 m in the suction pipe and 16 m in the delivery pipe. If the velocity of water in the delivery pipe is 1.4 m/s, pump efficiency 90% and slip 2%, determine the pump speed and the input power supplied.

COMPARISON OF CENTRIFUGAL AND RECIPROCATING PUMP

Sl. No	Reciprocating Pumps	Centrifugal pumps
1	Low flow rates and high pressures up to about 700 atmospheres.	High flow rates at moderate pressures of a few atmospheres.
2	At constant speed, constant flow rate at widely different outlet pressures.	At constant speed, variable discharge depending on outlet pressure.
3	Nil or very limited effect of viscosity. Eminently suitable for high-viscosity fluids.	Viscosity greatly affects the performance
4	Flow-rate control only through control of stroke length or speed.	Flow can be throttled and hence valve-type control is adopted.

COMPARISON OF CENTRIFUGAL AND RECIPROCATING PUMP

Sl. No	Reciprocating Pumps	Centrifugal pumps
5	Efficiency is high at around 90% and remains constant over the entire operating range.	Efficiency around 80% and varies in the operating range.
6	Niche and specialty applications	General and very vast range of applications.
7	Low-speed operation.	High-speed operation.
8	In handling shear sensitive liquids, the pump induced shear is very little.	Considerable shear during pumping—a disadvantage in handling shear sensitive liquids.

SEPARATION OF A RECIPROCATING PUMP

For the **satisfactory** working of a reciprocating pump, the **pressure inside** the **cylinder** at any instant **must not** be **less** than the **vapor pressure** of the liquid.

If the absolute pressure inside the cylinder (i.e., absolute pressure on the piston or plunger) is less than or equal to vapor pressure of the liquid, then **separation (cavitation)** will occur.

i.e. to avoid separation, $H_{cyl}(absolute) \geq H_{vapor}$

SEPARATION OF A RECIPROCATING PUMP

There are **two** situations when separation can happen. One is at the **beginning of the suction stroke** and another is at the **end of the delivery stroke**.

Maximum speed of the reciprocating pump is determined based on the above mentioned condition, i.e. pressure inside the cylinder during the suction and delivery stroke should not fall below vapor pressure of the following fluid in the suction and delivery pipe.

MAXIMUM POSSIBLE SPEED OF THE PUMP

The **absolute pressure** at the cylinder or on the piston at any instant during the suction stroke,

$$\begin{aligned} H_{cyl}(absolute) \dot{:} suction} &= H_{atm} - H_s - H_{as} - H_{fs} \\ &= H_{atm} - H_s - \left(\frac{l_s}{g} \frac{A}{a_s} \omega^2 r \cos\theta \right) - f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \sin\theta \right)^2 \end{aligned}$$

It can be proved that the H_{cyl} is **minimum** when $\theta=0$, i.e. at the **beginning** of the suction stroke. So **cavitation** can occur at the beginning of the suction stroke.

MAXIMUM POSSIBLE SPEED OF THE PUMP

Taking water vapor pressure at standard atmospheric condition as **2.5 m** of water absolute, it can be observed that to **avoid separation** (cavitation) during **suction**, $H_{cyl}(absolute) \text{ :}_{suction} \geq 2.5$, i.e. $H_{atm} - H_s - \left(\frac{l_s}{g} \frac{A}{a_s} \omega^2 r \right) \geq 2.5(abs)$

This equation shows that as **ω increases** the absolute cylinder **pressure decreases**, so, ω cannot be increased more than a maximum value. Thus, equating the above equation and substituting $\omega = \frac{2\pi N}{60}$ the maximum speed of the pump can be determined.

MAXIMUM POSSIBLE SPEED OF THE PUMP

$$H_{atm} - H_s - \frac{l_s}{g} \frac{A}{a_s} \left(\frac{2\pi N_{max}}{60} \right)^2 r = 2.5m \text{ (abs)}$$

$$-H_s - \frac{l_s}{g} \frac{A}{a_s} \left(\frac{2\pi N_{max}}{60} \right)^2 r = -7.8m \text{ (gauge)}$$

MINIMUM ABSOLUTE PRESSURE IN THE DELIVERY STROKE

The absolute pressure at the cylinder or on the piston at any instant during the **delivery stroke**,

$$\begin{aligned} H_{cyl}(absolute) \therefore_{delivery} &= H_{atm} + H_d + H_{ad} + H_{fd} \\ &= H_{atm} + H_d + \left(\frac{l_d}{g} \frac{A}{a_d} \omega^2 r \cos\theta \right) + f \frac{l_d}{d_d} \frac{1}{2g} \left(\frac{A}{a_d} \omega r \sin\theta \right)^2 \end{aligned}$$

It can be proved that the H_{cyl} is **minimum** when $\theta = 180$, i.e., at the **end** of the **delivery stroke**.

$$H_{cyl}(absolute) \therefore_{delivery} = H_{atm} + H_d - \left(\frac{l_d}{g} \frac{A}{a_d} \omega^2 r \cos\theta \right)$$

MINIMUM ABSOLUTE PRESSURE IN THE DELIVERY STROKE

- ❖ The **friction** in pipes does not affect the minimum absolute pressure.
- ❖ If **velocity head** in the pipes is appreciable, it can be included as follows:

$$H_{cyl}(absolute) \text{ :}_{suction} = H_{atm} - H_s - H_{as} - H_{fs} - \frac{v_s^2}{2g}$$

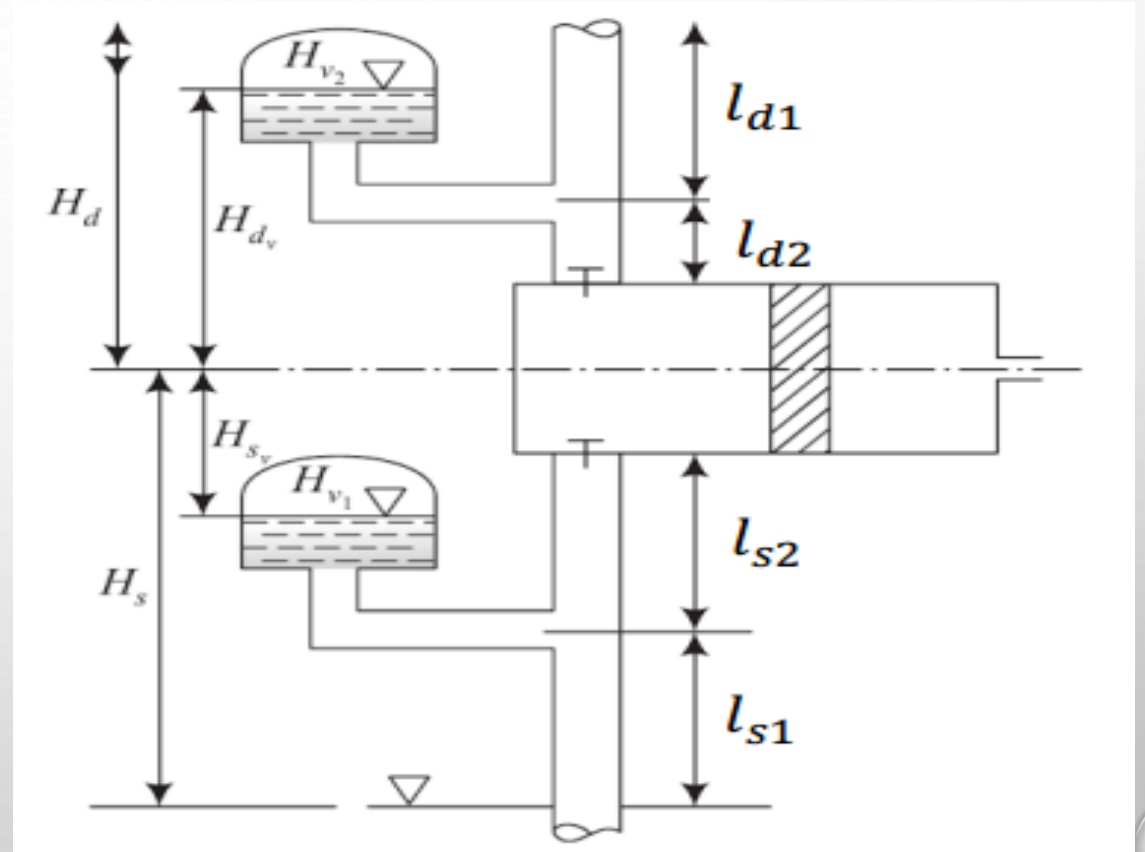
$$H_{cyl}(absolute) \text{ :}_{delivery} = H_{atm} + H_d + H_{ad} + H_{fd} + \frac{v_d^2}{2g}$$

AIR VESSEL

- ❖ Air vessel is a **closed chamber** containing **compressed air** at the **top** portion and **liquid** at the **bottom** portion of the chamber. At the **base** there is a opening through which the liquid may **flow into** the vessel or **flow out** of the vessel.
- ❖ Air and liquid are separated by a **flexible** diaphragm which is movable as per difference of pressure between two fluids.
- ❖ Usually one air vessel is connected to suction side and one to the delivery pipe. The vessels are connected as **close** to the pump as possible.

EFFECT OF AIR VESSELS

Air vessel acts like a **flywheel**. The compressed air at the top **contracts** or **expands** to absorb most of the pressure fluctuations. The friction head loss becomes **rectangular** instead of **parabolic**. There is no acceleration head in the pipes beyond the air vessels. The acceleration head remains **confined** into a shorter length between the **pump** and the **air vessels** i.e., between l_{s2} and l_{d2} .



EFFECT OF AIR VESSELS

- ❖ The air vessels **smooth out** the flow in the suction and delivery pipes and the flow is continuous beyond the air vessels. **Fluctuations** remain confined between the air vessels and the pump.
- ❖ By fitting air vessel as **close** to the **pump** as possible, the length of the pipe in which acceleration head occurs is reduced. This reduces acceleration head and the pump can be run at a much higher speed without any danger of separation.
- ❖ As the acceleration head and frictional head are considerably reduced, the **work done** is also reduced, hence, the **power input** is also reduced.

EFFECT OF AIR VESSELS

❖ Delivery Pipe:

Let l_{d1} be the length of the delivery pipe beyond the air vessel and l_{d2} be the length between the cylinder and the air vessel.

$$\text{Acceleration Head, } H_{ad2} = \frac{l_{d2}}{g} \left(\frac{A}{a_d} \right) (\omega^2 r \cos\theta)$$

$$\text{Friction Head, } H_{fd2} = f \frac{l_{d2}}{d_d} \frac{1}{2g} \left(\frac{A}{a_d} \omega r \sin\theta \right)^2$$

$$\text{Head lost beyond air vessel, } H_{fd1} = f \frac{l_{d1}}{d_d} \frac{1}{2g} v_d^2$$

EFFECT OF AIR VESSELS

Average velocity in the delivery pipe, $v_d = \frac{Q}{a_d} = \left[\frac{ALN}{60} \right] \times \frac{1}{a_d}$ (Single acting pump)

Total pressure head (abs):

$$\begin{aligned} H_{cyl}(absolute) \dot{:}delivery} &= H_{atm} + H_d + H_{ad2} + H_{fd2} + H_{fd1} + \frac{v_d^2}{2g} \\ &= H_{atm} + H_d + \frac{l_{d2}}{g} \left(\frac{A}{a_d} \right) (\omega^2 r \cos \theta) + f \frac{l_{d2}}{d_d} \frac{1}{2g} \left(\frac{A}{a_d} \omega r \sin \theta \right)^2 + f \frac{l_{d1}}{d_d} \frac{1}{2g} v_d^2 + \frac{v_d^2}{2g} \end{aligned}$$

EFFECT OF AIR VESSELS

❖ Suction Pipe:

Let l_{s1} be the length of the suction pipe beyond the air vessel and l_{s2} be the length between the cylinder and the air vessel.

$$\text{Acceleration Head, } H_{as2} = \frac{l_{s2}}{g} \left(\frac{A}{a_s} \right) (\omega^2 r \cos\theta)$$

$$\text{Friction Head, } H_{fs2} = f \frac{l_{s2}}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \sin\theta \right)^2$$

$$\text{Head lost beyond air vessel, } H_{fs1} = f \frac{l_{s1}}{d_s} \frac{1}{2g} v_s^2$$

EFFECT OF AIR VESSELS

Average velocity in the delivery pipe, $v_s = \frac{Q}{a_s} = \left[\frac{ALN}{60} \right] \times \frac{1}{a_s}$ (Single acting pump)

Total pressure head (abs):

$$\begin{aligned} H_{cyl}(absolute) :_{suction} &= H_{atm} - H_s - H_{as2} - H_{fs2} - H_{fs1} - \frac{v_s^2}{2g} \\ &= H_{atm} - H_s - \frac{l_{s2}}{g} \left(\frac{A}{a_s} \right) (\omega^2 r \cos \theta) - f \frac{l_{s2}}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \sin \theta \right)^2 - f \frac{l_{s1}}{d_s} \frac{1}{2g} v_s^2 - \frac{v_s^2}{2g} \end{aligned}$$

EFFECT OF AIR VESSELS

Work done and Power required for Pumps fitted with Air Vessels:

Work done in one revolution of the crank (single acting),

$$\begin{aligned} &= \gamma Q \left[\left(H_s + H_{as2} + \frac{2}{3} H_{fs2} + H_{fs1} + \frac{v_s^2}{2g} \right) + \left(H_d + H_{ad2} + \frac{2}{3} H_{fd2} + H_{fd1} + \frac{v_d^2}{2g} \right) \right] \\ &= \gamma AL \left[\left(H_s + H_{as2} + \frac{2}{3} H_{fs2} + H_{fs1} + \frac{v_s^2}{2g} \right) + \left(H_d + H_{ad2} + \frac{2}{3} H_{fd2} + H_{fd1} + \frac{v_d^2}{2g} \right) \right] \\ &= \gamma A(2r) \left[\left(H_s + H_{as2} + \frac{2}{3} H_{fs2} + H_{fs1} + \frac{v_s^2}{2g} \right) + \left(H_d + H_{ad2} + \frac{2}{3} H_{fd2} + H_{fd1} + \frac{v_d^2}{2g} \right) \right] \end{aligned}$$

EFFECT OF AIR VESSELS

$$= 2\gamma Ar \left[\left(H_s + H_{as2} + \frac{2}{3}H_{fs2} + H_{fs1} + \frac{v_s^2}{2g} \right) + \left(H_d + H_{ad2} + \frac{2}{3}H_{fd2} + H_{fd1} + \frac{v_d^2}{2g} \right) \right]$$

Work done **per second** (single acting), i.e., Power (hydraulic) developed

$$= \frac{\gamma ALN}{60} \left[\left(H_s + H_{as2} + \frac{2}{3}H_{fs2} + H_{fs1} + \frac{v_s^2}{2g} \right) + \left(H_d + H_{ad2} + \frac{2}{3}H_{fd2} + H_{fd1} + \frac{v_d^2}{2g} \right) \right]$$
$$= \frac{2\gamma ArN}{60} \left[\left(H_s + H_{as2} + \frac{2}{3}H_{fs2} + H_{fs1} + \frac{v_s^2}{2g} \right) + \left(H_d + H_{ad2} + \frac{2}{3}H_{fd2} + H_{fd1} + \frac{v_d^2}{2g} \right) \right]$$

EFFECT OF AIR VESSELS

Here $H_{fs2} = f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \right)^2$ and $H_{fd2} = f \frac{l_d}{d_d} \frac{1}{2g} \left(\frac{A}{a_d} \omega r \right)^2$

$$H_{fs1} = f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \frac{\omega r}{\pi} \right)^2 \quad \text{and} \quad H_{fd1} = f \frac{l_d}{d_d} \frac{1}{2g} \left(\frac{A}{a_d} \frac{\omega r}{\pi} \right)^2$$

Work done and Power required for Pumps fitted with Air Vessels:

Neglecting the velocity head and the friction head in the smaller pipe

Sections (l_{s2} and l_{d2}), we have,

EFFECT OF AIR VESSELS

Work done in one revolution of the crank (single acting), [when acceleration head loss is very negligible]

$$= \gamma Q [(H_s + H_{fs1}) + (H_d + H_{fd1})]$$

Work done per second (single acting), i.e., Power (hydraulic) developed

$$= \frac{2\gamma ArN}{60} [(H_s + H_{fs1}) + (H_d + H_{fd1})]$$

EFFECT OF AIR VESSELS

Work done when air vessel is NOT fitted:

Work done **against friction** during the suction stroke,

$$P_1 = 2\gamma Ar \left(\frac{2}{3} H_{fs2} \right)$$

$$P_1 = 2\gamma Ar \left[\frac{2}{3} f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \right)^2 \right]$$

$$P_1 = \frac{4\gamma Ar}{3} \left[f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \right)^2 \right]$$

EFFECT OF AIR VESSELS

Work done when air vessel is fitted:

$$P_2 = 2\gamma Ar(H_{fs1}) = 2\gamma Ar \left(f \frac{l_s}{d_s} \frac{1}{2g} v_s^2 \right)$$

$$v_s = \frac{Q}{a_s} = \left[\frac{ALN}{60} \right] \times \frac{1}{a_s} = \frac{A}{a_s} \frac{\omega}{2\pi} (2r) = \frac{A}{a_s} \frac{\omega r}{\pi}$$

$$P_2 = 2\gamma Ar \left[f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \frac{\omega r}{\pi} \right)^2 \right]$$

EFFECT OF AIR VESSELS

Work saved in percentage for single acting pump:

$$\begin{aligned} & \frac{P_1 - P_2}{P_1} \times 100\% \\ &= \frac{2\gamma Ar \left[f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \right)^2 \right] \left(\frac{2}{3} - \frac{1}{\pi^2} \right)}{\frac{4\gamma Ar}{3} \left[f \frac{l_s}{d_s} \frac{1}{2g} \left(\frac{A}{a_s} \omega r \right)^2 \right]} \times 100\% \\ &= \frac{3}{2} \times \left(\frac{2}{3} - \frac{1}{\pi^2} \right) \times 100\% = 84.8\% \end{aligned}$$

EFFECT OF AIR VESSELS

Work saved in percentage For double acting pump:

$$Q = \frac{2ALN}{60} = 2 \frac{A \omega r}{a_s \pi}$$
$$\frac{P_1 - P_2}{P_1} \times 100\%$$
$$= \frac{3}{2} \times \left(\frac{2}{3} - \frac{2}{\pi^2} \right) \times 100\% = 39.2\%$$

Problem 3:

A single acting reciprocating pump has the following characteristics:

Piston diameter = 100 mm; Stroke length = 300 mm.

Suction head = 4 m; Diameter of suction pipe = 75 mm;

Suction pipe length = 4 m;

Atmospheric pressure = 10 m water (abs)

Cavitation pressure = 2.5 m water (abs)

Determine the **maximum speed** at which the pump can be run without cavitation.

Assume Frictional losses = 1 m.

Problem 4:

A single acting reciprocating pump has a piston of diameter 200 mm and a stroke of 300 mm. It draws water from a sump 3.5 m below the pump through a pipe of 5.5 m long. The separation pressure head is 2.5 m water abs, and atmospheric pressure is 10.3 m abs. When the pump runs at 60 rpm, find the **minimum diameter** of suction pipe for no separation. Assume simple harmonic motion of the piston.

Problem 5:

A single acting reciprocating pump has 125 mm diameter cylinder with a stroke of 500 mm. The length and diameter of the suction pipe are 5.2 m and 100 mm respectively. The suction lift is 3.25 m and the delivery lift is 12 m. The pump speed is 45 rpm. If an air vessel is fitted very close to the cylinder in the delivery side, calculate **power required** to pump water. Assume the frictional head in the delivery pipe to be 0.15 m and the velocity heads in the pipes can be neglected. Take, $\eta_{pump} = 0.9$ and $f = 0.02$.

Problem 6:

A single acting reciprocating pump has an air vessel in the delivery side fitted very close to the cylinder. The cylinder has a diameter of 300 mm and a stroke length of 450 mm. The delivery pipe is 40 m long and has a diameter of 200 mm. The speed of the pump is 60 rpm. Determine the **power saved** by the **air vessel** in overcoming friction in the delivery pipe. Take friction factor, $f = 0.03$.