Lecture 8

HYDRAULIC PUMPS [CONTINUED]

1.7 Vane Pumps

There are two types of vane pumps:

1. **Unbalanced vane pump:** Unbalanced vane pumps are of two varieties:

- Unbalanced vane pump with fixed delivery.
- Unbalanced vane pump with pressure-compensated variable delivery.

2. **Balanced vane pump.**

1.7.1 Unbalanced Vane Pump with Fixed Delivery

A simplified form of unbalanced vane pump with fixed delivery and its operation are shown in Figs. 1.12 and 1.13. The main components of the pump are the cam surface and the rotor. The rotor contains radial slots splined to drive shaft. The rotor rotates inside the cam ring. Each radial slot contains a vane, which is free to slide in or out of the slots due to centrifugal force. The vane is designed to mate with surface of the cam ring as the rotor turns. The cam ring axis is offset to the drive shaft axis. When the rotor rotates, the centrifugal force pushes the vanes out against the surface of the cam ring. The vanes divide the space between the rotor and the cam ring into a series of small chambers. During the first half of the rotor rotation, the volume of these chambers increases, thereby causing a reduction of pressure. This is the suction process, which causes the fluid to flow through the inlet port. During the second half of rotor rotation, the cam ring pushes the vanes back into the slots and the trapped volume is reduced. This positively ejects the trapped fluid through the outlet port. In this pump, all pump action takes place in the chambers located on one side of the rotor and shaft, and so the pump is of an unbalanced design. The delivery rate of the pump depends on the eccentricity of the rotor with respect to the cam ring.

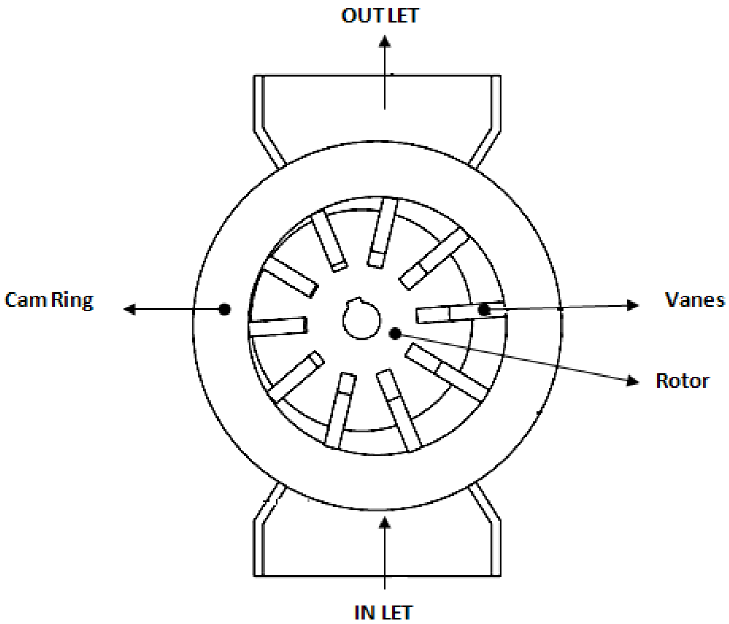


Figure 1.12 Simple vane pump

1.7.4 Pressure-Compensated Variable Displacement Vane Pump (an Unbalanced Vane Pump with Pressure-Compensated Variable Delivery)

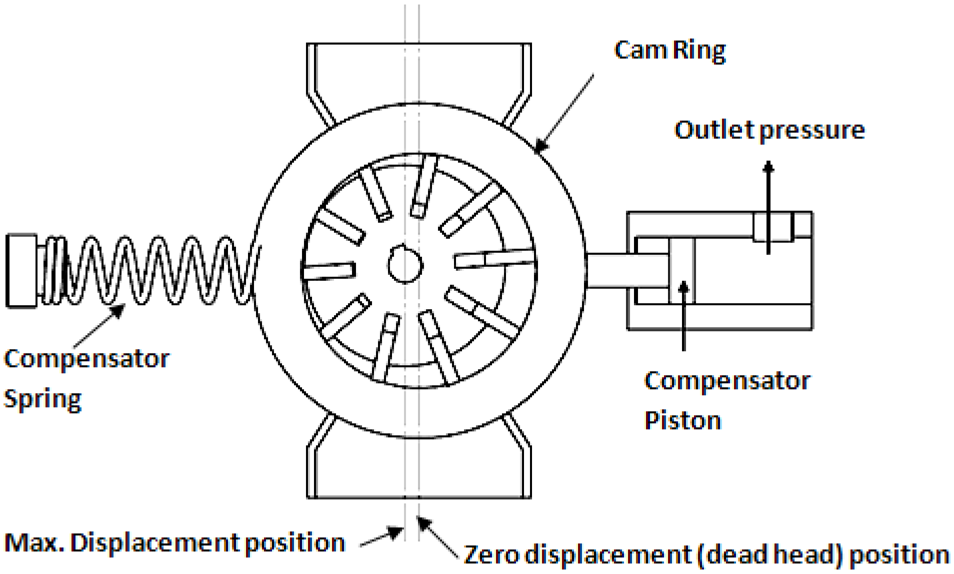


Figure 1.14 Operation of a variable displacement vane pump

Schematic diagram of variable displacement vane pump is shown in Fig.1.14.Variable displacement feature can be brought into vane pumps by varying eccentricity between the rotor and the cam ring. Here in this pump, the stator ring is held against a spring loaded piston.The system pressure acts directly through a hydraulic piston on the right side. This forces the cam ring against a spring-loaded piston on the left side. If the discharge pressure is large enough, it overcomes the compensated spring force and shifts the cam ring to the left.

This reduces the eccentricity and decreases the flow. If the pressure continues to increase, there is no eccentricity and pump flow becomes zero.

1.7.5 Balanced Vane Pump with Fixed Delivery

A balanced vane pump is a very versatile design that has found widespread use in both industrial and mobile applications. The basic design principle is shown in Fig. 1.15. The rotor and vanes are contained within a double eccentric cam ring and there are two inlet segments and two outlet segments during each revolution. This double pumping action not only gives a compact design, but also leads to another important advantage: although pressure forces acting on the rotor in the outlet area are high, the forces at the two outlet areas are equal and opposite, completely canceling each other. As a result, there are no net loads on shaft bearings. Consequently, the life of this type of pump in many applications has been exceptionally good. Operating times of 24000 h or more in industrial applications are widespread. In more severe conditions encountered in mobile vehicles, 5000–10000 h of trouble-free operation is frequently achieved.

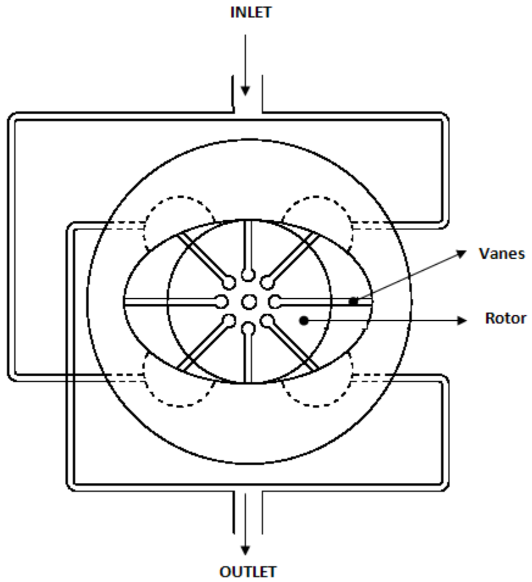


Figure 1.15 Operation of a balanced vane pump

1.7.2 Advantages and disadvantages of Vane Pumps

The advantages of vane pumps are as follows:

1. Vane pumps are self-priming, robust and supply constant delivery at a given speed.
2. They provide uniform discharge with negligible pulsations.
3. Their vanes are self-compensating for wear and vanes can be replaced easily.
4. These pumps do not require check valves.
5. They are light in weight and compact.
6. They can handle liquids containing vapors and gases.
7. Volumetric and overall efficiencies are high.
8. Discharge is less sensitive to changes in viscosity and pressure variations.

The disadvantages of vane pumps are as follows:

1. Relief valves are required to protect the pump in case of sudden closure of delivery.
2. They are not suitable for abrasive liquids.
3. They require good seals.
4. They require good filtration systems and foreign particle can severely damage pump.

Advantages and disadvantagesofbalancedvane pumps

The advantages of balanced vane pumps are as follows:

1. The balanced pump eliminates the bearing side loads and therefore high operating pressure can be used.
2.The service life is high compared to unbalanced type due to less wear and tear.

The disadvantages of balanced vane pumps are as follows:

1. They are fixed displacement pumps.
2. Design is more complicated.
3. Manufacturing cost is high compared to unbalanced type.

1.7.4 Expression for the Theoretical Discharge of Vane Pumps

Let D_C be the diameter ofa cam ring in m, $D_R$the diameter of rotor in m, L the width of rotor in m, e the eccentricity in m, V_D the pump volume displacement in m^3/rev and e_{max} the maximum possible eccentricity in m.

From geometry (Fig.1.13) the maximum possible eccentricity,

$$e_{max} = \frac{D_C - D_R}{2} \qquad (1.1)$$

Themaximum value of eccentricity produces the maximum volumetric displacement

$$V_{D(max)} = \frac{\pi}{4}(D_C^2 - D_R^2)L \qquad (1.2)$$

Using Equation (1.1), Equation (1.2) can be simplified as

$$V_{D(max)} = \frac{\pi}{4}(D_c - D_g)(D_c + D_g)L$$

<

$$V_{D(max)} = \frac{\pi}{4}(D_c + D_g)(2e_{max})L$$

The actual volumetric displacement occurs when $e_{max} = e$. Hence,

$$V_{D(max)} = \frac{\pi}{2}(D_C + D_R)e\,L\,\text{m}^3\text{/rev}$$

When the pump rotates at N rev/min (RPM),the quality of discharge by the vane pump is given by

$$Q_T = v_D \times N$$

Theoretical discharge

$$Q_T = \frac{\pi}{2}(D_C + D_R)e\,L \text{ m}^3/\text{min}$$

Example 1.6
A vane pump has a rotor diameter of 63.5 mm, a cam ring diameter of 88.9 mm and a vane width of 50.8 mm. What must be eccentricity for it to have a volumetric displacement of 115cm^3?

Solution: Volumetric displacement is

$$V_D = \pi\left(\frac{D_C + D_R}{2}\right)L\,e$$

where$D_C$is the diameter of the cam ring, D_R is the diameter of the rotor, e is the eccentricity and L is the width of the vane pump. So we have

$$115\times10^{-6} = \pi\times\frac{0.0889+0.0635}{2}\times e\times0.0508$$

Therefore eccentricity

$$e = 9.456\times10^{-3}\text{ m} = 9.456\text{ mm}$$

1.8Piston Pumps
Piston pumps are of the following two types:

1. **Axial piston pump:**These pumps are of two designs:

    - Bent-axis-type piston pump.
    - Swash-plate-type piston pump.

2. **Radial piston pump.**

1.8.1Bent-Axis-Type Piston Pump
Schematic diagram and detailed cut section of bent axis type piston pump is shown in Fig.1.16. It contains a cylinder block rotating with a drive shaft. However,thecenterline of the cylinder block is set at an offset angle relative to the centerline of the drive shaft.Thecylinder block contains a number of pistons arranged along a circle.The piston rods are connected to the drive shaft flange by a ball and socket joints. The pistons are forced in and out of their bores as the distance between the drive shaft flange and cylinder block changes. A universal link connects the cylinder block to the drive shaft to provide alignment and positive drive.
The volumetric displacement of the pump depends on the offset angle θ. No flow is produced when the cylinder blockis centerline.θcan vary from 0°to a maximum of about 30°.For a fixed displacement, units are usually provided with 23° or 30° offset angles.

1.8.2 Swash-Plate-Type Piston Pump
Schematic diagram of swash plate type piston pump is shown in Fig. 1.17. In this type, the cylinder block and drive shaft are located on the same centerline. The pistons are connected to a shoe plate that bears against an angled swash plate. As the cylinderrotates,the pistons reciprocatebecausethe piston shoesfollow the angled surface of the swash plate. The outlet

andinlet ports are located in the valve plate so that the pistons pass theinlet as they are being pulledout and pass the outlet as they are being forced back in. This type of pump can also be designed to have a variable displacementcapability. The maximumswash plate angle is limited to 17.5° by construction.

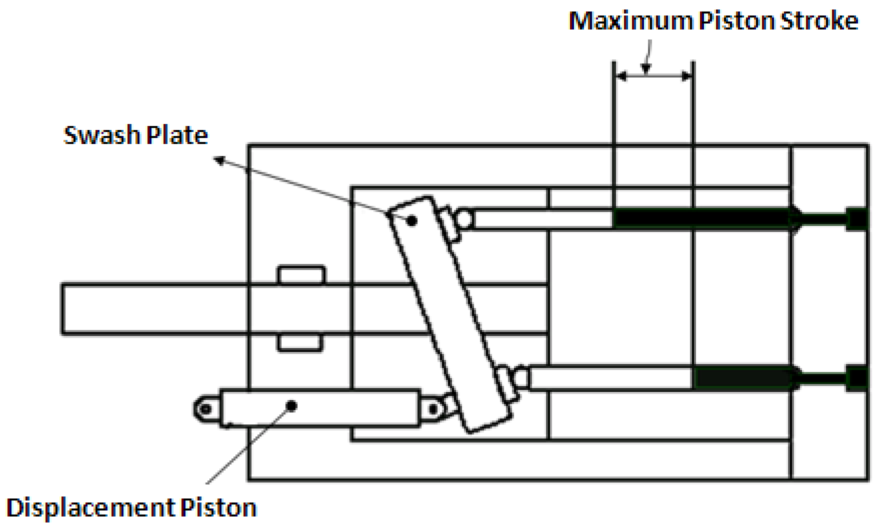
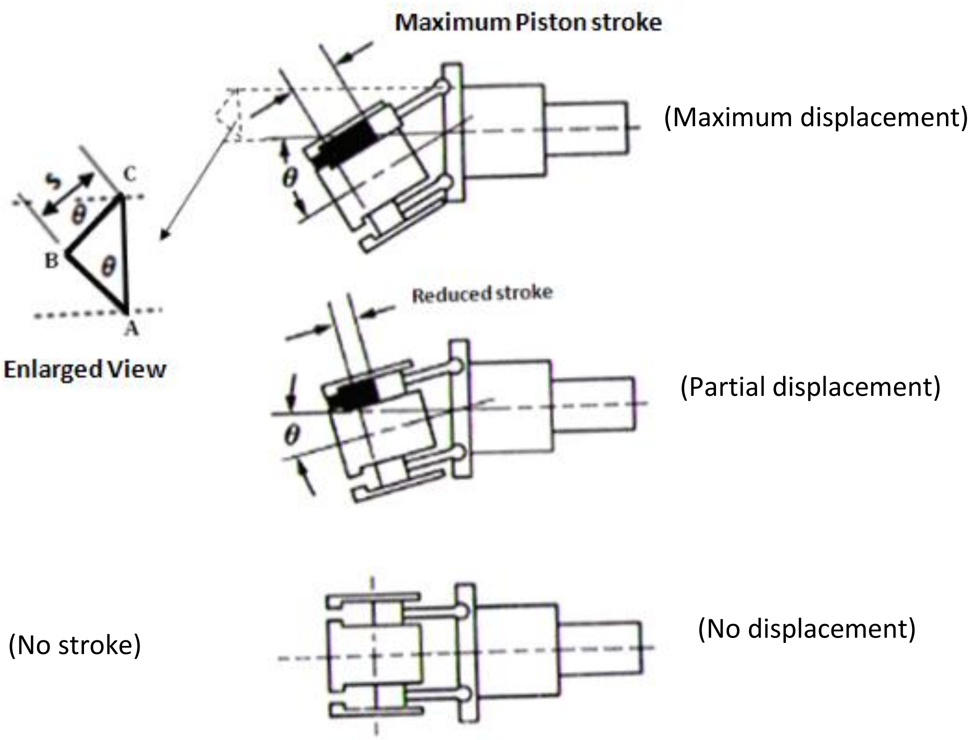


Figure 1.17 Operation of a swash-plate-type piston pump

1.8.4 Volumetric Displacement and Theoretical Flow Rate of an Axial Piston Pump

Figure 1.19(a) shows in and out position of the pistons of axial piston pump.Figure1.19(b) gives schematic diagram of stroke change with respect to offset angle.

Letθ be an offset angle, S the piston stroke in m, D the piston circle diameter, Y the number of pistons, A the piston area inm^2, N the piston speed in RPM and Q_T the theoretical flow rate in m^3/min.



(b)

Figure 1.19 Stroke changes with offset angle

From a right-angled triangle ABC [Fig. 1.19(b)]

$$\tan\theta = \frac{\mathrm{BC}}{\mathrm{AB}} = \frac{S}{D}$$

$$\Rightarrow S = D \times \tan\theta \qquad (1.3)$$

The displacement volume of one piston = ASm^3

Total displacement volume of Ynumber of pistons = YASm^3

$$V_{\mathrm{D}} = YAS \qquad (1.4)$$

From Eqs. (1.3) and (1.4), we have

$$V_{\mathrm{D}} = YAD\tan\theta \ \mathrm{m^3/rev} \qquad (1.5)$$

Theoretical flow rate is

$$Q_{\mathrm{T}} = DANY\tan\theta \ \mathrm{m^3/min}$$

Example 1.7

What is the theoretical flow rate from a fixed-displacement axial piston pump with a nine-bore cylinder operating at 2000 RPM? Each bore has a diameter of 15 mm and stroke is 20 mm.

Solution: Theoretical flow rate is given by

$$Q_{\mathrm{T}} = \text{ Volume} \times \text{ RPM} \times \text{ Number of pistons}$$

$$= \frac{\pi}{4} \times D^2 \times L \times N \times n$$

$$= \frac{\pi}{4} \times 0.015^2 \times 0.02 \times \frac{2000}{60} \times 9$$

$$= 10.6 \times 10^{-3} \mathrm{m^3/s}$$

$$= 1.06\ \text{LPS} = 63.6\ \text{LPM}$$

1.9Comparison of Hydraulic Pumps

Pump design with a wide range of operating characteristics are available. A designer must select carefully to achieve a circuit design that meets the functional objective while minimizing total cost which includes both ownership cost and operating cost over the life of component. Pump selection is important decision in circuit design. Designer must compare the various options available and then choose the optimum pump. Table 1.2 gives a typical comparison of all pumps.

The major factor in adopting a pump to a particular system is the system's overall needs. It would be wrong to use a pump with high delivery in a system that requires only a lowdelivery rate. On the contrary, using a pump that must produce at its peak continuously just to meet the minimum requirements of the system is equally wrong. Making either of these mistakes produces a poor system due to excessive initial pump costs or maintenance cost.

One should use a pump that is suited to the system, whether a gear pump which has fewer moving precision parts or a piston pump whichhas many parts fitted to close tolerance and is therefore more expensive.

Table 1.2

	Pressure (Bar)	**Discharge(LPM)**	**MaximumSpeed (RPM)**	**Overall Efficiency**
Gear pump	20–175	7–570	1800–7000	75–90
Vane pump	20–175	2–950	2000–4000	75–90
Axial piston pump	70–350	2–1700	600–6000	85–95
Radial piston pump	50–250	20–700	600–1800	80–92

1.10Pump Performance

The performance of a pump is a function of the precision of its manufacture. An ideal pump is one having zero clearance between all mating parts. Because this is not possible, working clearances should be as small as possible while maintaining proper oil films for lubrication between rubbing parts. The performance of a pump is determined by the following efficiencies:

1. Volumetric efficiency (η_v): It is the ratio of actual flow rate of the pump to the theoretical flow rate of the pump. This is expressed as follows:

$$\text{Volumetric efficiency } (\eta_\text{v}) = \frac{\text{Actual flow rate of the pump}}{\text{Theoretical flow rate of the pump}}$$

$$= \frac{Q_\text{A}}{Q_\text{T}}$$

Volumetric efficiency (η_v) indicates the amount of leakage that takes place within the pump.This is due to manufacture tolerances and flexing of the pump casing under designed pressure operating conditions.

For gear pumps, η_v = 80%–90%.

For vane pumps, η_v = 92%.

For piston pumps, η_v = 90%–98%.

2. Mechanical efficiency (η_m): Itis the ratio of the pump output power assuming no leakage to actual power delivered to the pump:

$$\text{Mechanical efficiency } (\eta_\text{m}) = \frac{\text{Pump output power assuming no leakages}}{\text{Actual power delivered to the pump}}$$

Mechanicalefficiency(η_m) indicates the amount of energy losses that occur for reasons other than leakage. This includes friction in bearings and between mating parts. This includes the energy losses due to fluid turbulence. Mechanical efficiencies are about 90%–95%. We also have the relation

$$\eta_\text{m} = \frac{p\,Q_\text{T}}{T_\text{A}\,N}$$

where$p$ is the pump discharge pressure in Pa or N/m^2, Q_T is the theoretical flow rate of the pump in m^3/s, T_A is the actual torque delivered to the pump in Nm and N is the speed of the pump in rad/s.

It (η_m) can also be computed in terms of torque as follows:

$$\eta_\text{m} = \frac{\text{Theoretical torque required to operate the pump}}{\text{Actual torque delivered to the pump}}$$

$$= \frac{T_\text{T}}{T_\text{A}}$$

The theoretical torque (T_T) required to operate the pump is the torque that would be required if there were no leakage.

The theoretical torque (T_T) is determined as follows

$$T_\text{T}(\text{N m}) = \frac{VD_\text{N}}{2\pi}\left(\text{m}^3 \times \frac{N}{\text{m}^2}\right) = \text{N m}$$

The actual torque (T_A) is determined as follows

Actual torque T_A (N m) $= \frac{P}{\omega}\left(\frac{\text{N m/s}}{\text{rad/s}}\right) = \text{N m}$

where $\omega = 2\pi N/60$. Here N is the speed in RPM.

3. Overall efficiency (η_o)**:** It is defined as the ratio of actual power delivered by the pump to actual power delivered to the pump

$$\text{Overall efficiency } (\eta_o) = \frac{\text{Actual power delivered by the pump}}{\text{Actual power delivered to the pump}}$$

Overall efficiency (η_o) considers all energy losses and can be represented mathematically as follows:

$$\text{Overall efficiency } (\eta_o) = \eta_v \eta_m$$

$$\Rightarrow \eta_o = \frac{Q_A}{Q_T} \times \frac{pQ_T}{T_A N}$$

Example 1.8

A gear pump has an outside diameter of 82.6 mm, inside diameter of 57.2 mm and a width of 25.4 mm. If the actual pump flow is 1800 RPM and the rated pressure is 0.00183 m^3/s, what is the volumetric efficiency?

Solution: We have

Outside diameter $D_o = 82.6$ mm

Inside diameter $D_i = 57.2$ mm

Width $d = 25.4$ mm

Speed of pump N = 1800 RPM

Actual flow rate = 0.00183 m^3/s

Theoretical flow rate

$$Q_T = \frac{\pi}{4} \times (D_o^2 - D_i^2) \times d \times \frac{N}{60}$$

$$= \frac{\pi}{4} \times (0.0826^2 - 0.0572^2) \times 0.0254 \times \frac{1800}{60}$$

$$= 2.125 \times 10^{-3}$$

Volumetric efficiency is

$$\eta_v = \frac{0.00183}{2.125 \times 10^{-3}} \times 100 = 86.11\%$$

Example 1.9

A pump having a volumetric efficiency of 96% delivers 29 LPM of oil at 1000 RPM. What is the volumetric displacement of the pump?

Solution:

Volumetric efficiency of the pump η_v = 96%
Discharge of the pump = 29 LPM
Speed of pump $N = 1000$ rpm
Now

$$\eta_v = \frac{\text{Actual flow rate of the pump}}{\text{Theoritical flow rate of the pump}} = \frac{Q_A}{Q_T}$$

$$\Rightarrow 0.96 = \frac{29}{Q_T}$$

$$\Rightarrow Q_T = 30.208 \text{ LPM}$$

Volumetric displacement

$$V_D = \frac{Q_T}{N} = \frac{30.208 \times 10^{-3} \times 60}{60 \times 1000}$$

$$= 30.208 \times 10^{-6} \text{ m}^3/\text{rev} = 0.0302 \text{ L}/\text{rev}$$

Example 1.10
A positive displacement pump has an overall efficiency of 88% and a volumetric efficiency of 92%. What is the mechanical efficiency?

Solution: Theoverall efficiency is

$$\eta_o = \eta_m \times \eta_v$$

$$\Rightarrow \eta_m = \frac{\eta_o}{\eta_v} = \frac{88}{92} \times 100 = 95.7\%$$

Example 1.11
Determine the overall efficiency of a pump driven by a 10 HP prime mover if the pump delivers fluid at 40 LPM at a pressure of 10 MPa.

Solution:

Output power = pQ

$$= 10 \times 10^6 \text{ N/m}^2 \times 40 \text{ L/min} \times \frac{\text{m}^3/\text{s}}{1000 \text{ L/s}} \times \frac{1 \text{ min}}{60 \text{ s}}$$

$$= 6670 \text{ W}$$

$$\text{Input power} = 10 \text{ HP} \times \frac{746 \text{ W}}{1 \text{ HP}} = 7460 \text{ W}$$

Now

$$\eta_o = \frac{\text{Pump output power}}{\text{Pump input power}}$$

$$= \frac{6670}{7460} = 0.894 = 89.4\%$$

Example 1.12

How much hydraulic power would a pump produce when operating at 140 bar and delivering 0.001 m^3/s of oil? What power rated electric motor would be selected to drive this pump if its overall efficiency is 85%?

Solution:

Operating pressure of the pump = 140 bar

Flow rate$Q$ = 0.001m^3/s. Now

$$\text{Power of pump} = \text{Pressure} \times \text{Flow rate}$$
$$= 140 \times 10^5 \times 0.001$$
$$= 14 \text{ kW}$$

Overall efficiency of pump η_o = 85%

Power to be supplied is

$$\frac{\text{Power of pump}}{\eta_o} = \frac{14\,\text{kW}}{0.85} = 16.47\,\text{kW}$$

Example 1.13

A pump has a displacement volume of 98.4 cm^3. It delivers 0.0152 m^3/s of oil at 1000 RPM and 70 bar. If the prime mover input torque is 124.3 Nm. What is the overall efficiency of pump? What is the theoretical torque required to operate the pump?

Solution:

Volumetric discharge $= 98.4\,\text{cm}^3$

Theoretical discharge is

$$Q_T = V_D \times \frac{N}{60} = 98.4 \times \frac{1000}{60} = 1.64 \times 10^{-3}\ \text{m}^3/\text{s}$$

Volumetric efficiency is

$$\eta_v = \frac{1.52 \times 10^{-3}}{1.64 \times 10^{-3}} \times 100 = 92.68\ \%$$

Overall efficiency is

$$\eta_o = \frac{Q_A \times \text{pressure}}{T \times \omega} = \frac{1.52 \times 10^{-3} \times 70 \times 10^5 \times 60}{124.3 \times 2 \times 1000 \times \pi} \times 100 = 81.74\%$$

The mechanical efficiency is

$$\eta_{\text{mechanical}} = \frac{\eta_{\text{overall}}}{\eta_{\text{volumetric}}} = \frac{81.74}{92.78} = 88.2$$

Now

$$\text{Theoretical torque} = \text{Actual torque} \times \eta_{\text{mechanical}} = 124.3 \times 0.882 = 109.6\ \text{Nm}$$

Note: Mechanical efficiency can also be calculated as

$$\eta_m = \frac{pQ_T}{T\omega}$$
$$= \frac{70 \times 10^5\ \text{N/m}^2 \times 0.00164\ \text{m}^3/\text{s}}{124.3\ (\text{N m}) \times \frac{1000}{60} \times 2\pi\ \text{rad/s}}$$
$$= 0.882 = 88.2\%$$