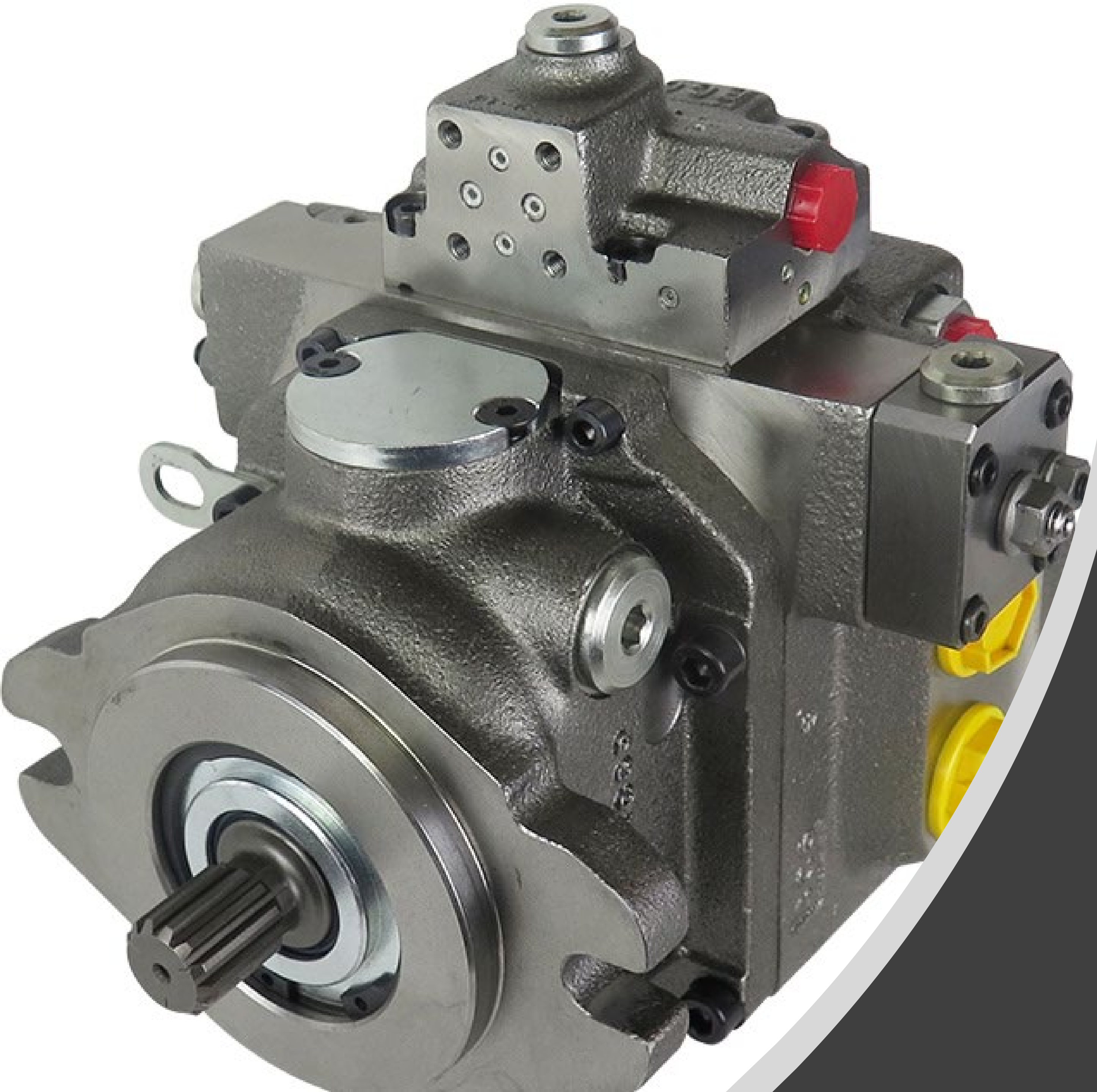


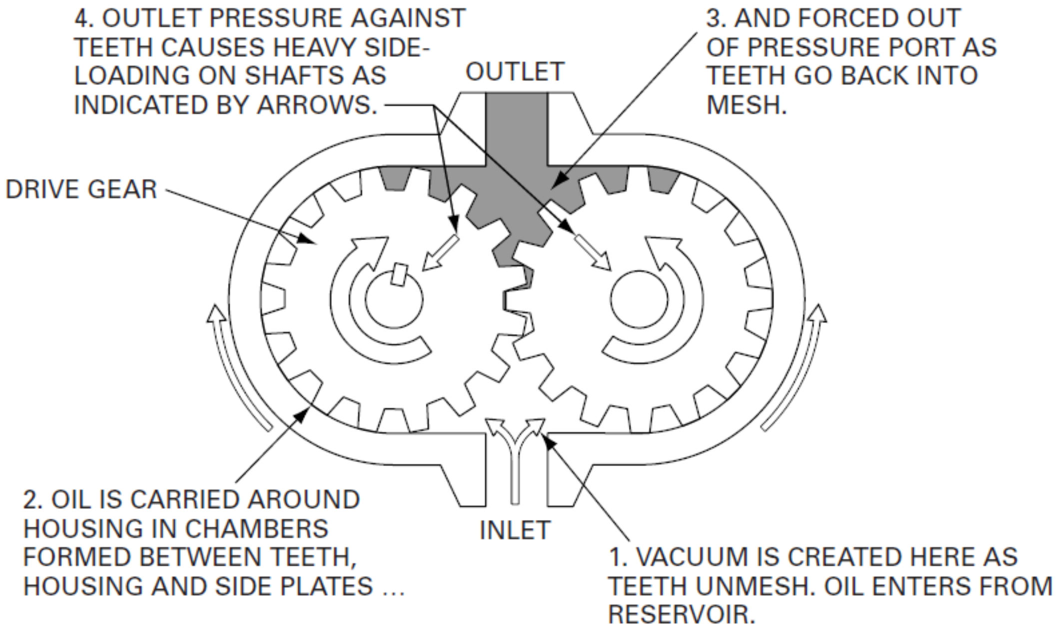
Pneumatics and hydraulics

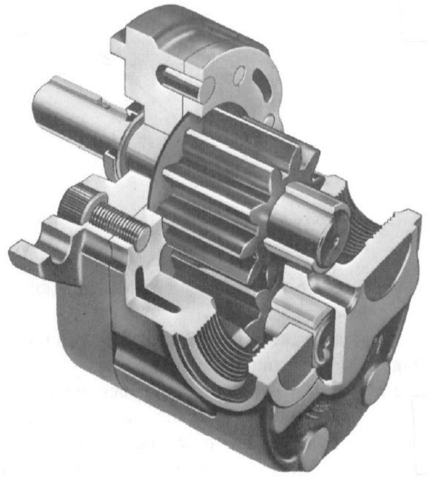
Hydraulic Pumps part 2

Dr. Ahmad Al-Mahasneh

External gear pump operation

- One of the gears is connected to a drive shaft connected to the prime mover.
- The second gear is driven as it meshes with the driver gear. Oil chambers are formed between the gear teeth, the pump housing, and the side wear plates.
- The suction side is where teeth come out of mesh, and it is here that the volume expands, bringing about a reduction in pressure to below atmospheric pressure.
- Fluid is pushed into this void by atmospheric pressure because the oil supply tank is vented to the atmosphere.
- The discharge side is where teeth go into mesh, and it is here that the volume decreases between mating teeth.
- Since the pump has a positive internal seal against leakage, the oil is positively ejected into the outlet port.





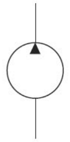
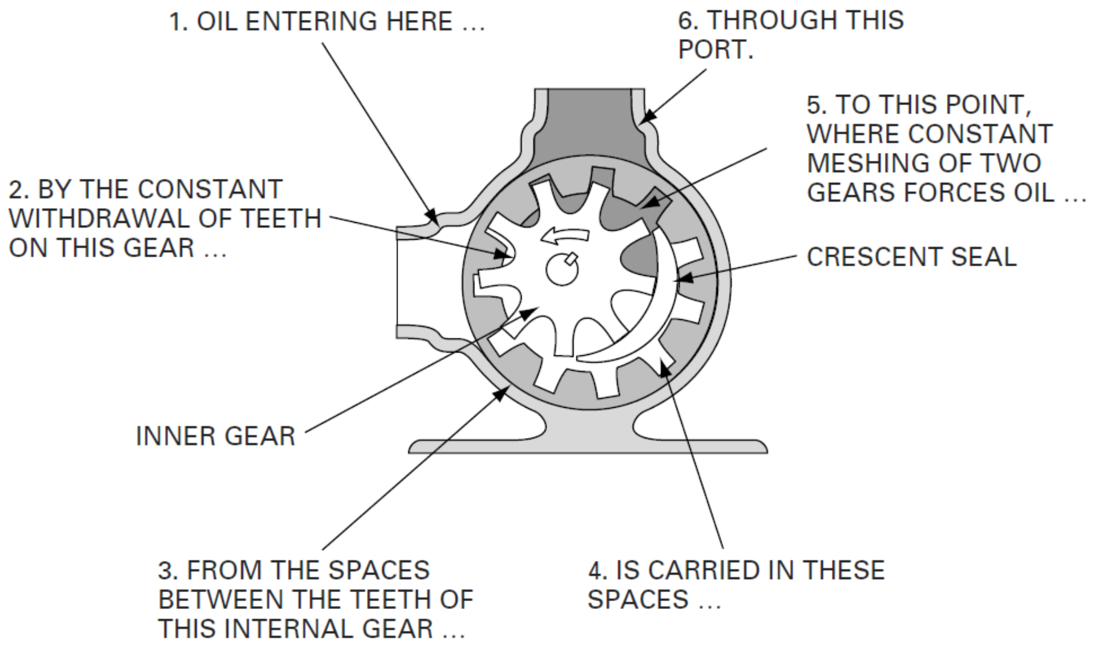
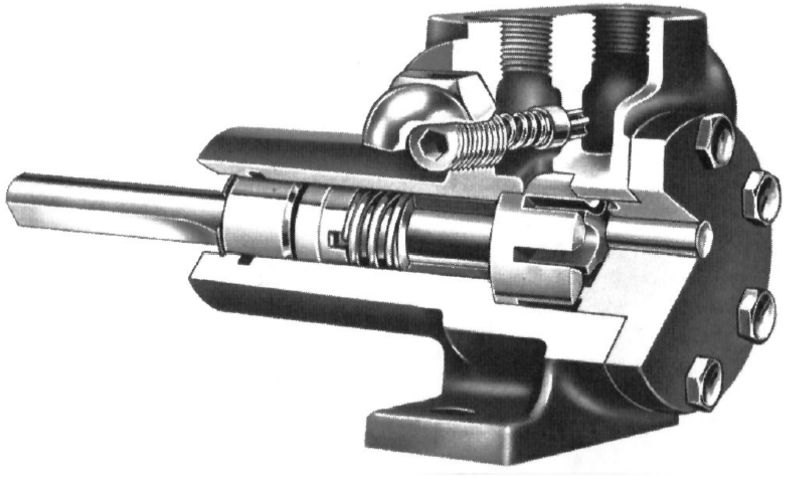


Figure 5-9. Detailed features of an external gear pump. *(Courtesy of Webster Electric Company, Inc., subsidiary of STA-RITE Industries, Inc., Racine, Wisconsin.)*

Internal Gear Pump

- This design consists of an internal gear, a regular spur gear, a crescent-shaped seal, and an external housing.
- As power is applied to either gear, the motion of the gears draws fluid from the reservoir and forces it around both sides of the crescent seal, which acts as a seal between the suction and discharge ports.
- When the teeth mesh on the side opposite to the crescent seal, the fluid is forced to enter the discharge port of the pump.

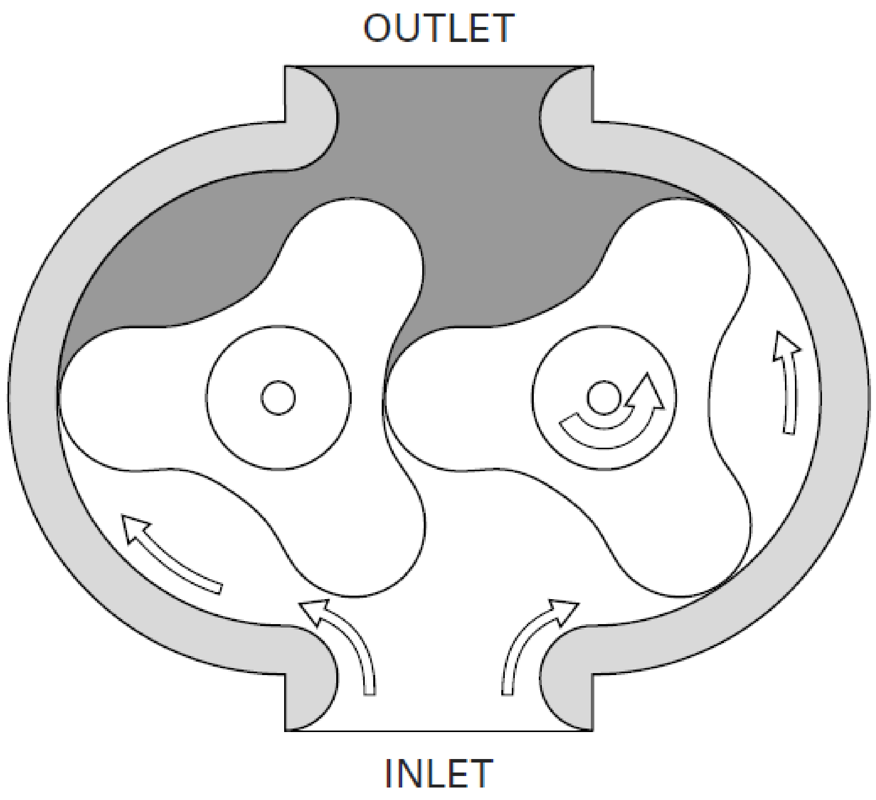


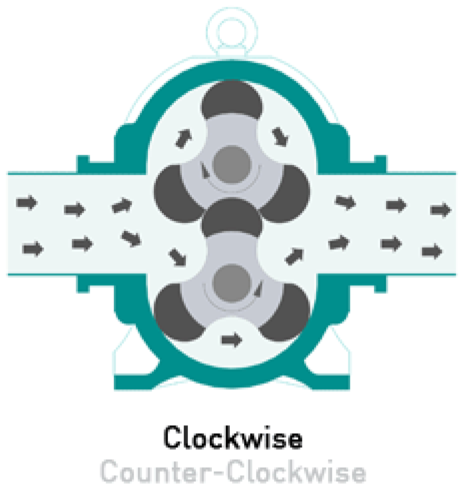


Internal Gear Pump

Lobe pump

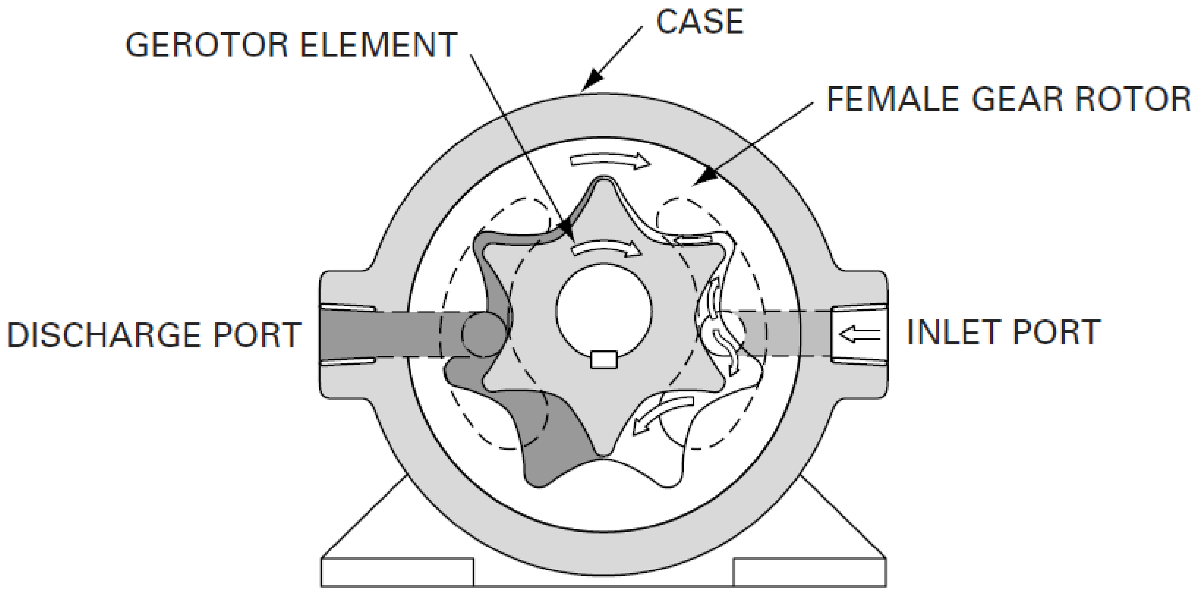
- This pump operates in a fashion similar to the external gear pump. But unlike the external gear pump, both lobes are driven externally so that they do not actually contact each other.
- Thus, they are quieter than other types of gear pumps.
- Due to the smaller number of mating elements, the lobe pump output will have a somewhat greater amount of pulsation, although its volumetric displacement is generally greater than that for other types of gear pumps.

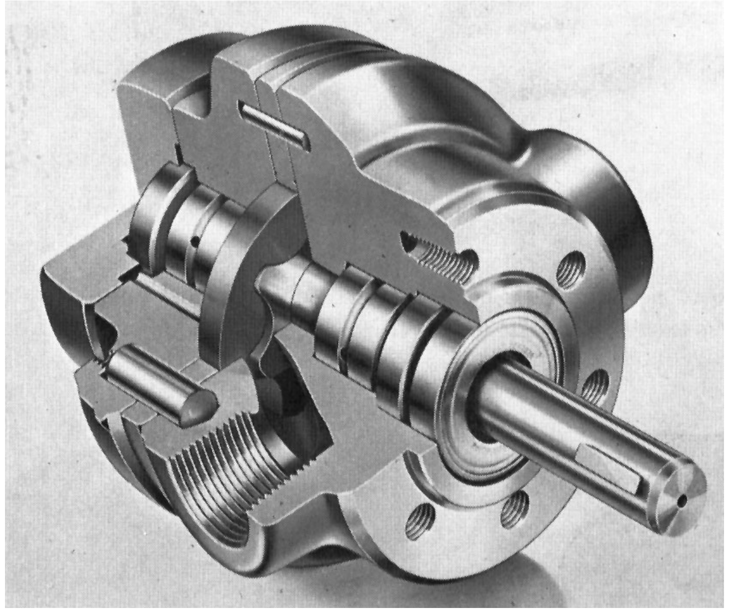




Gerotor Pump

- The Gerotor pump operates very much like the internal gear pump.
- The inner gear rotor (Gerotor element) is power-driven and draws the outer gear rotor around as they mesh together.
- This forms inlet and discharge pumping chambers between the rotor lobes.
- The tips of the inner and outer rotors make contact to seal the pumping chambers from each other.
- The inner gear has one tooth less than the outer gear, and the volumetric displacement is determined by the space formed by the extra tooth in the outer rotor.

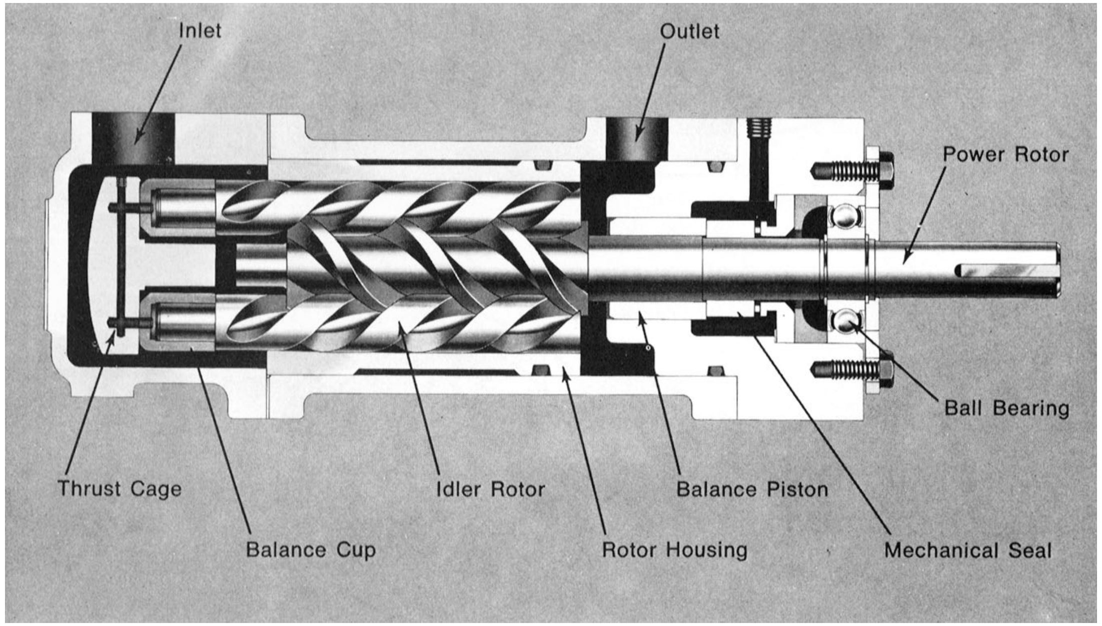


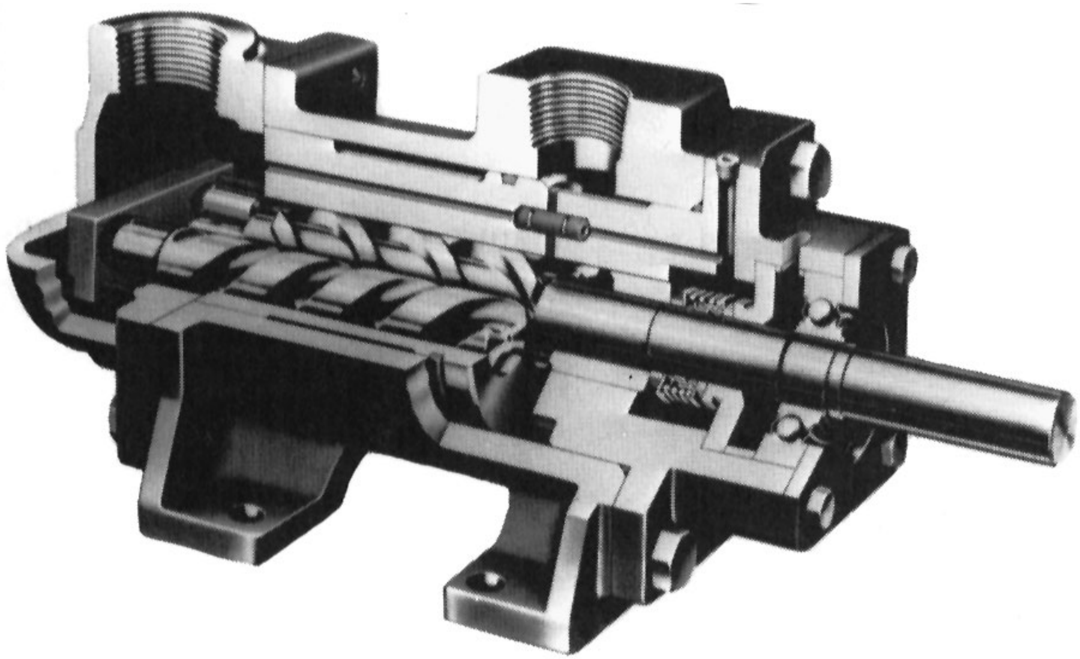


Gerotor Pump

Screw Pump

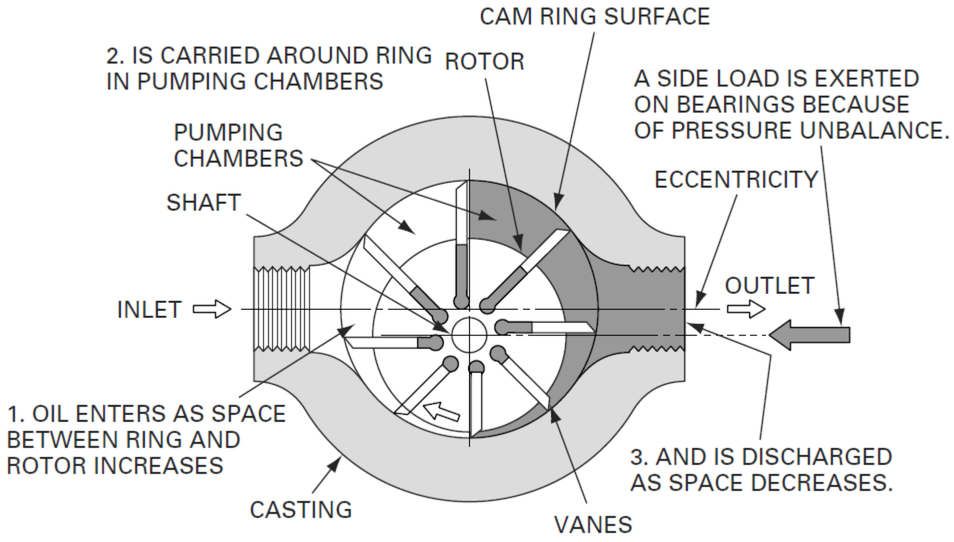
- The screw pump is an axial flow positive displacement unit.
- Three precision ground screws, meshing within a closefitting housing, deliver nonpulsating flow quietly and efficiently.
- The two symmetrically opposed idler rotors act as rotating seals, confining the fluid in a succession of closures or stages.
- The idler rotors are in rolling contact with the central power rotor and are free to float in their respective housing bores on a hydrodynamic oil film.
- There are no radial bending loads.
- Axial hydraulic forces on the rotor set are balanced, eliminating any need for thrust bearings.
- It is rated at 500 psi and can deliver up to 123 gpm.
- High-pressure designs are available for 3500-psi operation with output flow rates up to 88 gpm.





VANE PUMPS

- The rotor, which contains radial slots, is splined to the drive shaft and rotates inside a cam ring.
- Each slot contains a vane designed to mate with the surface of the cam ring as the rotor turns.
- Centrifugal force keeps the vanes out against the surface of the cam ring.
- During one-half revolution of rotor rotation, the volume increases between the rotor and cam ring.
- The resulting volume expansion causes a reduction of pressure.
- This is the suction process, which causes fluid to flow through the inlet port and fill the void.
- As the rotor rotates through the second half revolution, the surface of the cam ring pushes the vanes back into their slots, and the trapped volume is reduced.
- This positively ejects the trapped fluid through the discharge port.



Analysis of Volumetric Displacement

From geometry, we can find the maximum possible eccentricity:

$$e_{max} = \frac{D_C - D_R}{2}$$

This maximum value of eccentricity produces a maximum volumetric displacement:

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

Noting that we have the difference between two squared terms yields

$$V_{D_{max}} = \frac{\pi}{4}(D_C + D_R)(D_C - D_R)L$$

Substituting the expression for e_{max} yields

$$V_{D_{max}} = \frac{\pi}{4}(D_C + D_R)(2e_{max})L$$

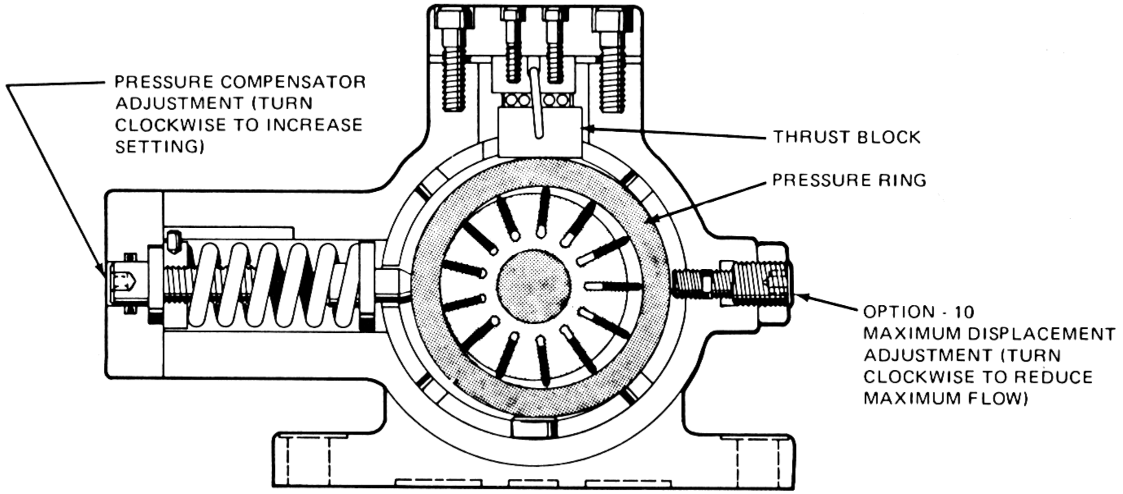
The actual volumetric displacement occurs when $e_{max} = e$:

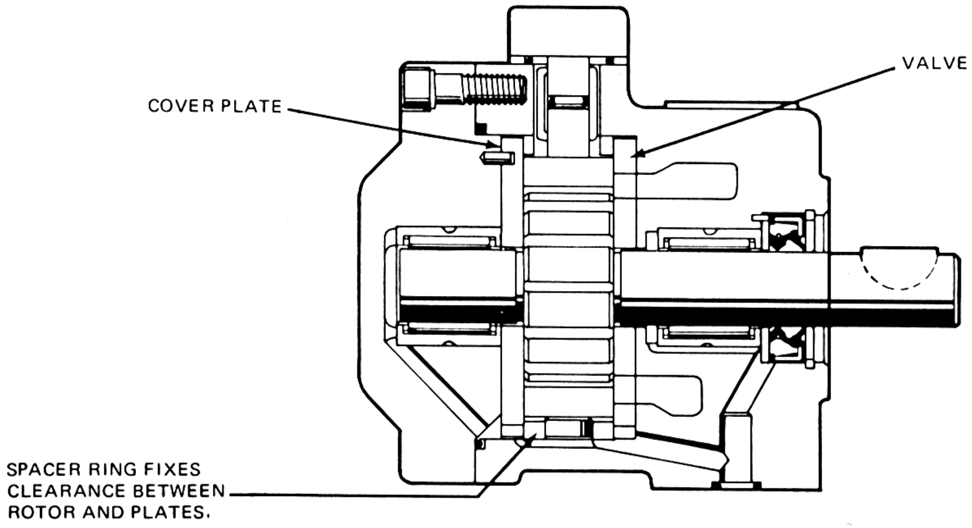
$$V_D = \frac{\pi}{2}(D_C + D_R)eL \qquad \textbf{(5-4)}$$

D_C = diameter of cam ring (in, m)
D_R = diameter of rotor (in, m)
L = width of rotor (in, m)
V_D = pump volumetric displacement (in^3, m^3)
e = eccentricity (in, m)
e_{max} = maximum possible eccentricity (in, m)
V_{Dmax} = maximum possible volumetric displacement (in^3, m^3)

Variable displacement vane pump

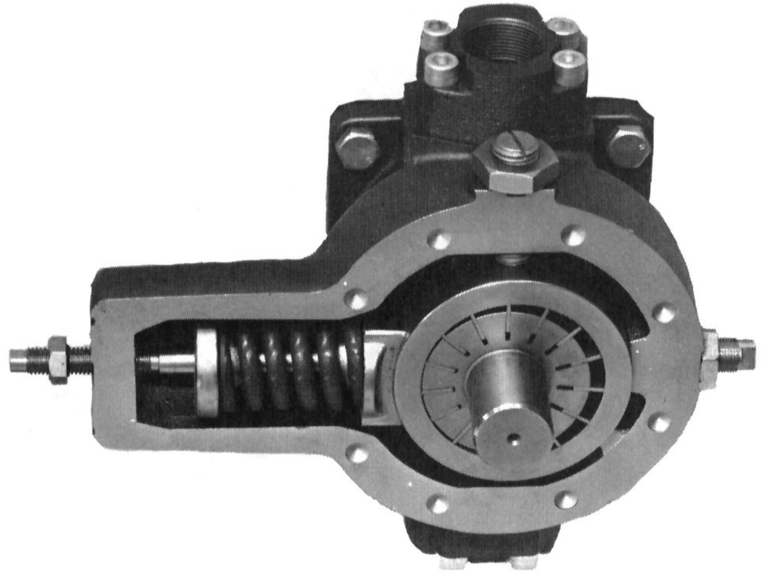
- Some vane pumps have provisions for mechanically varying the eccentricity.
- Such a design is called a *variable displacement pump.*
- A handwheel or a pressure compensator can be used to move the cam ring to change the eccentricity.
- The direction of flow through the pump can be reversed by movement of the cam ring on either side of center.

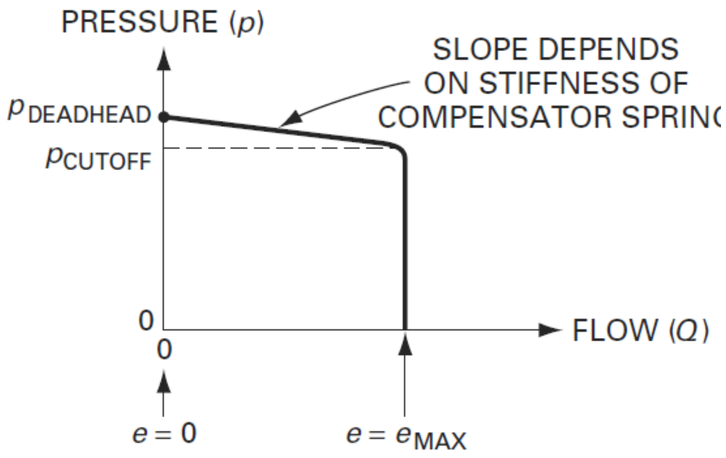




Pressure-Compensated Vane Pump

- System pressure acts directly on the cam ring via a hydraulic piston on the right.
- This forces the cam ring against the compensator spring-loaded piston on the left side of the cam ring.
- If the discharge pressure is large enough, it overcomes the compensator spring force and shifts the cam ring to the left.
- This reduces the eccentricity, which is maximum when discharge pressure is zero.
- As the discharge pressure continues to increase, zero eccentricity is finally achieved, and the pump flow becomes zero.
- Such a pump basically has its own protection against excessive pressure buildup.
- When the pressure reaches a value called p_{cutoff}, the compensator spring force equals the hydraulic piston force.
- As the pressure continues to increase, the compensator spring is compressed until zero eccentricity is achieved.
- The maximum pressure achieved is called $p_{deadhead}$, at which point the pump is protected because it produces no more flow.
- As a result, there is no power wasted and fluid heating is reduced.





Pressure versus flow for pressure compensated vane pump.

Balanced Vane Pump

- A balanced vane pump is one that has two intake and two outlet ports diametrically opposite each other. Thus, pressure ports are opposite each other, and a complete hydraulic balance is achieved.
- One disadvantage of a balanced vane pump is that it cannot be designed as a variable displacement unit.
- Instead of having a circular cam ring, a balanced design vane pump has an elliptical housing, which forms two separate pumping chambers on opposite sides of the rotor.
- This eliminates the bearing side loads and thus permits higher operating pressures.

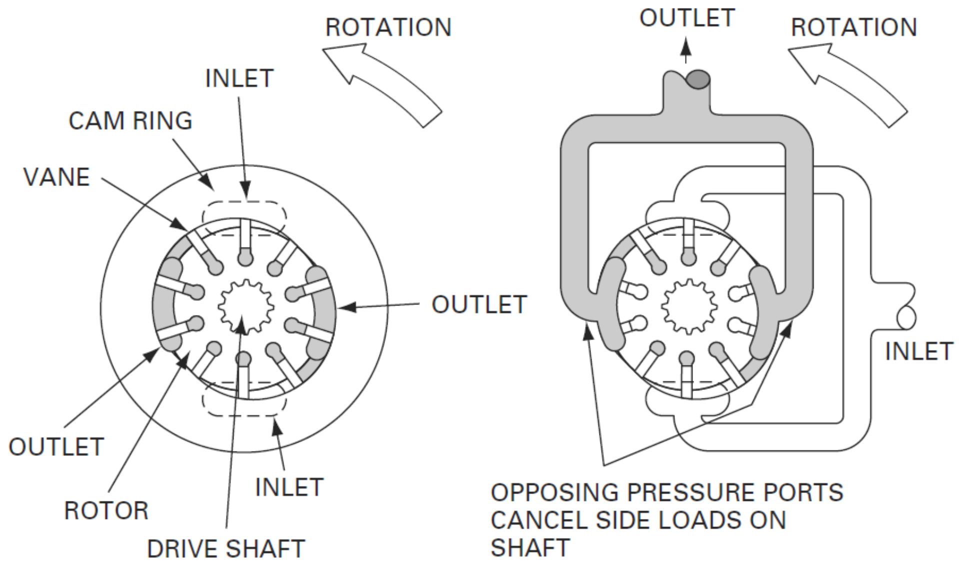
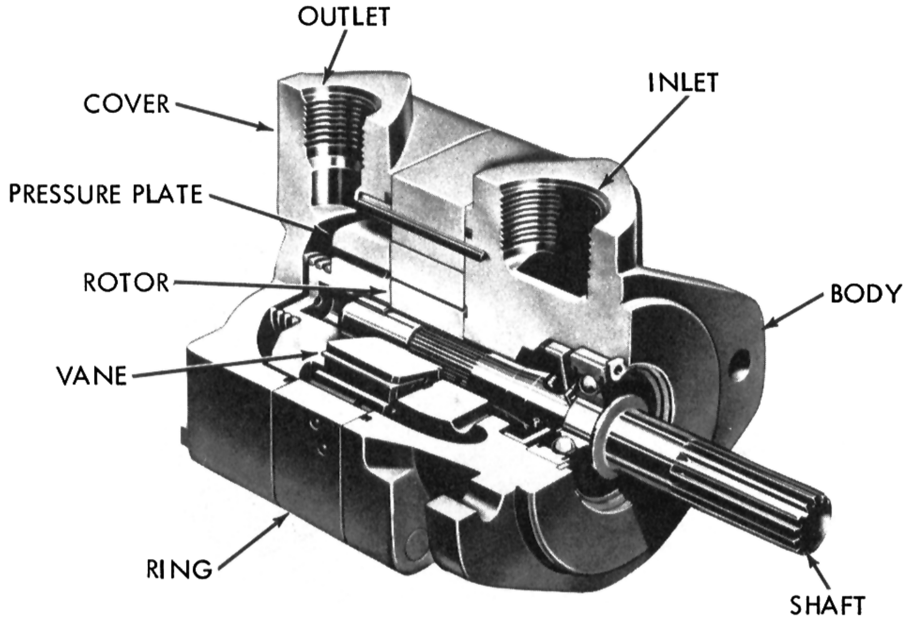


Figure 5-21. Balanced vane pump principles. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*



Examples

EXAMPLE 5-3

A vane pump is to have a volumetric displacement of 5 in³. It has a rotor diameter of 2 in, a cam ring diameter of 3 in, and a vane width of 2 in. What must be the eccentricity?

Solution Use Eq. (5-4):

$$e = \frac{2V_D}{\pi(D_C + D_R)L} = \frac{(2)(5)}{\pi(2 + 3)(2)} = 0.318 \text{ in}$$

Examples

EXAMPLE 5-4

A vane pump has a rotor diameter of 50 mm, a cam ring diameter of 75 mm, and a vane width of 50 mm. If the eccentricity is 8 mm, determine the volumetric displacement.

Solution Substituting values into Eq. (5-4) yields

$$V_D = \frac{\pi}{2}(0.050 + 0.075)(0.008)(0.050) = 0.0000785 \text{ m}^3$$

Since 1 L = 0.001 m³, $V_D = 0.0785$ L.

Examples

EXAMPLE 5-5

A fixed displacement vane pump delivers 1000 psi oil to an extending hydraulic cylinder at 20 gpm. When the cylinder is fully extended, oil leaks past its piston at a rate of 0.7 gpm. The pressure relief valve setting is 1200 psi. If a pressure-compensated vane pump were used it would reduce pump flow from 20 gpm to 0.7 gpm when the cylinder is fully extended to provide the leakage flow at the pressure relief valve setting of 1200 psi. How much hydraulic horsepower would be saved by using the pressure-compensated pump?

Solution The fixed displacement pump produces 20 gpm at 1200 psi when the cylinder is fully extended (0.7 gpm leakage flow through the cylinder and 19.3 gpm through the relief valve). Thus, we have

$$\text{hydraulic HP lost} = \frac{pQ}{1714} = \frac{1200 \times 20}{1714} = 14.0 \text{ hp}$$

Examples

A pressure-compensated pump would produce only 0.7 gpm at 1200 psi when the cylinder is fully extended. For this case we have

$$\text{hydraulic HP lost} = \frac{pQ}{1714} = \frac{1200 \times 0.7}{1714} = 0.49 \text{ hp}$$

Hence, the hydraulic horsepower saved = 14.0 − 0.49 = 13.51 hp. This horsepower savings occurs only while the cylinder is fully extended because either pump would deliver 1000 psi oil at 20 gpm while the cylinder is extending.

PISTON PUMPS

- A piston pump works on the principle that a reciprocating piston can draw in fluid when it retracts in a cylinder bore and discharge it when it extends.
- The basic question is how to mechanize a series of reciprocating pistons.
- There are two basic types of piston pumps.
- One is the axial design, having pistons that are parallel to the axis of the cylinder block.
- Axial piston pumps can be either of the bent axis configuration or of the swash plate design.
- The second type of piston pump is the radial design, which has pistons arranged radially in a cylinder block.

Axial Piston Pump (Bent-Axis Design)

- Axial piston pump (bent-axis type) that contains a cylinder block rotating with the drive shaft. However, the centerline of the cylinder block is set at an offset angle relative to the centerline of the drive shaft.
- The cylinder block contains a number of pistons arranged along a circle.
- The piston rods are connected to the drive shaft flange by 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 block to the drive shaft to provide alignment and positive drive.

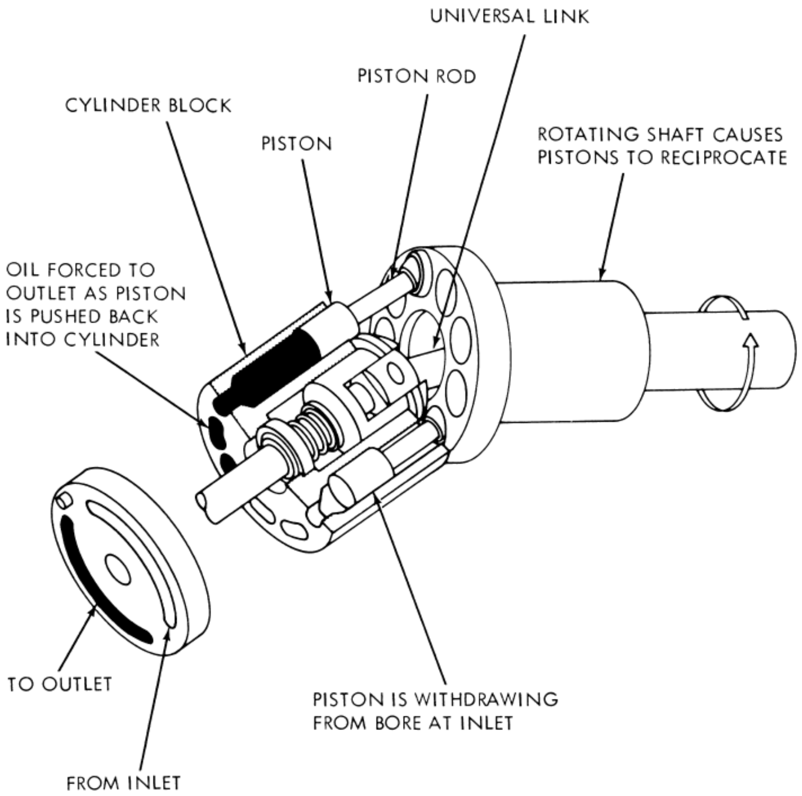
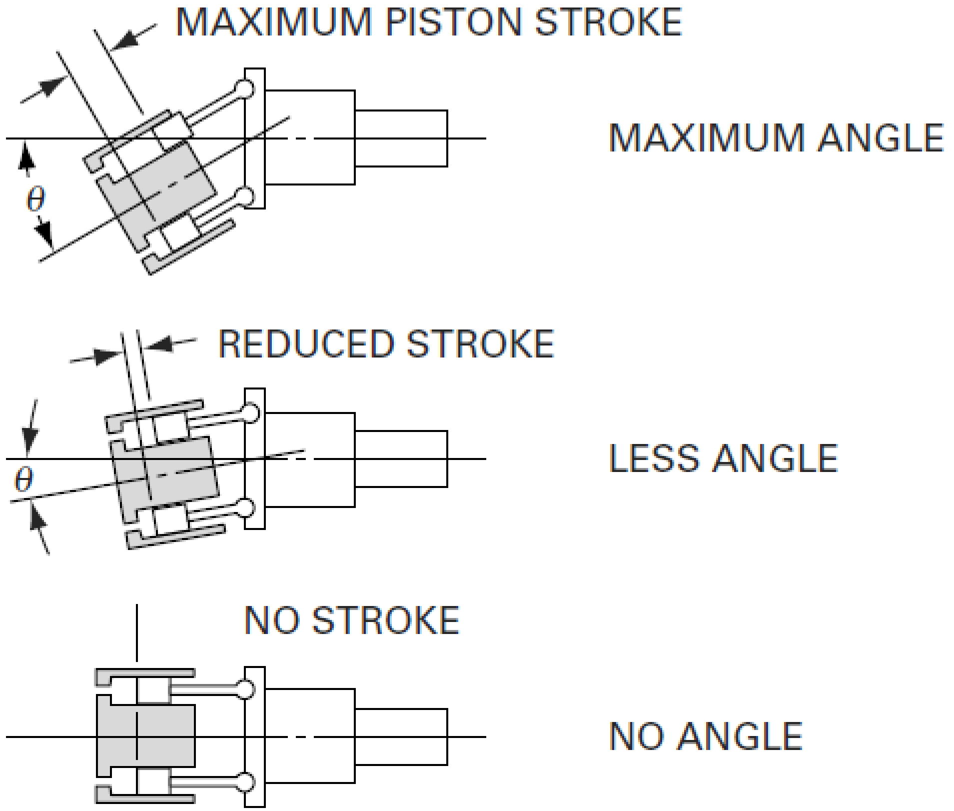


Figure 5-23. Axial piston pump (bent-axis type). *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

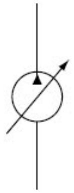
Axial Piston Pump (Bent-Axis Design)

- The volumetric displacement of the pump varies with the offset angle θ.
- No flow is produced when the cylinder block centerline is parallel to the drive shaft centerline.
- θ can vary from 0° to a maximum of about 30°. Fixed displacement units are usually provided with 23° or 30° offset angles.
- Variable displacement units are available with a yoke and some external control to change the offset angle such as a stroking cylinder.
- Some designs have controls that move the yoke over the center position to reverse the direction of flow through the pump



Axial Piston Pump (Bent-Axis Design)

- Figure 5-25 is a cutaway of a variable displacement piston pump in which an external handwheel can be turned to establish the desired offset angle.
- Also shown is the hydraulic symbol used to represent variable displacement pumps in hydraulic circuits.



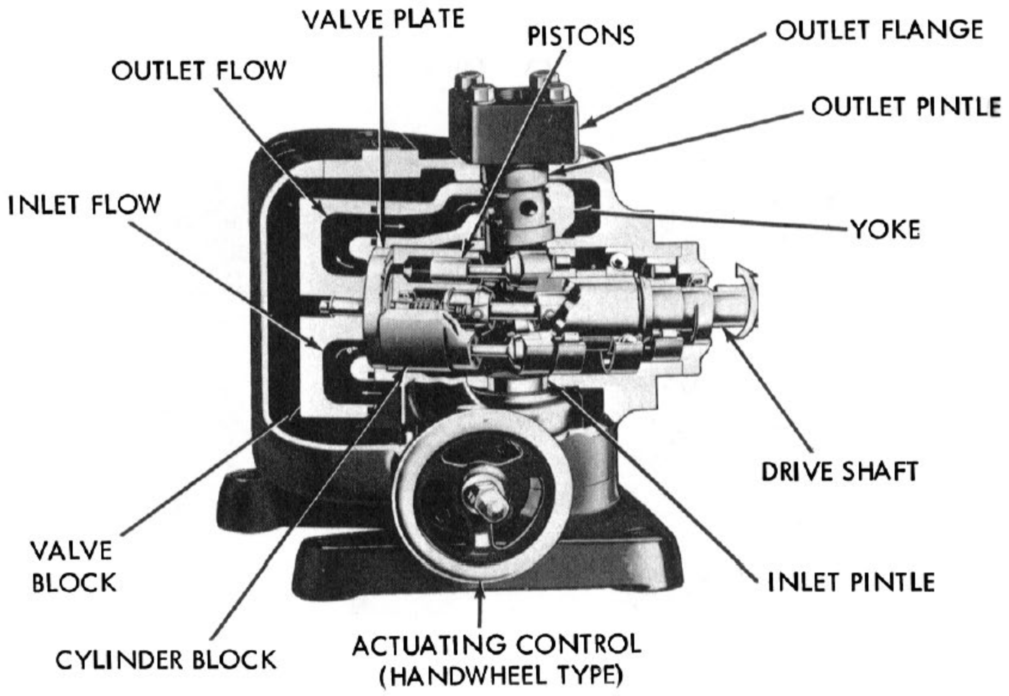


Figure 5-25. Variable displacement piston pump with handwheel. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Volumetric Displacement and Theoretical Flow Rate

The following nomenclature and analysis are applicable to an axial piston pump:

θ = offset angle (°)
S = piston stroke (in, m)
D = piston circle diameter (in, m)
Y = number of pistons
A = piston area (in^2, m^2)
N = pump speed (rpm)
Q_T = theoretical flow rate (gpm, m^3/min)

From trigonometry we have

$$\tan(\theta) = \frac{S}{D}$$

or

$$S = D \tan(\theta)$$

Volumetric Displacement and Theoretical Flow Rate

The total displacement volume equals the number of pistons multiplied by the displacement volume per piston:

$$V_D = YAS$$

Substituting, we have

$$V_D = YAD \tan(\theta) \qquad \textbf{(5-5)}$$

From Eqs. (5-2) and (5-5) we obtain a relationship for the theoretical flow rate using English units.

$$Q_T(\text{gpm}) = \frac{DANY \tan(\theta)}{231} \qquad \textbf{(5-6)}$$

Similarly, using Eqs. (5-2M) and (5-5), we obtain a relationship for the theoretical flow rate in metric units.

$$Q_T(\text{m}^3/\text{min}) = DANY \tan(\theta) \qquad \textbf{(5-6M)}$$

Volumetric Displacement and Theoretical Flow Rate

EXAMPLE 5-6

Find the offset angle for an axial piston pump that delivers 16 gpm at 3000 rpm. The pump has nine $\frac{1}{2}$-in-diameter pistons arranged on a 5-in-diameter piston circle. The volumetric efficiency is 95%.

Solution From Eq. (5-3) we calculate the theoretical flow rate:

$$Q_T = \frac{Q_A}{\eta_v} = \frac{16 \text{ gpm}}{0.95} = 16.8 \text{ gpm}$$

Using Eq. (5-6) yields

$$\tan(\theta) = \frac{231 Q_T}{DANY} = \frac{231 \times 16.8}{5[\pi/4(1/2)^2] \times 3000 \times 9} = 0.146$$

$$\theta = 8.3^\circ$$

Volumetric Displacement and Theoretical Flow Rate

EXAMPLE 5-7

Find the flow rate in units of L/s that an axial piston pump delivers at 1000 rpm. The pump has nine 15-mm-diameter pistons arranged on a 125-mm-diameter piston circle. The offset angle is set at 10° and the volumetric efficiency is 94%.

Solution Substituting directly into Eq. (5-6M) yields

$$Q_T(\text{m}^3/\text{min}) = D(\text{m}) \times A(\text{m}^2) \times N(\text{rev/min}) \times Y \tan(\theta)$$

$$= 0.125 \left(\frac{\pi}{4} \times 0.015^2\right) \times 1000 \times 9 \times \tan 10^\circ = 0.0351 \text{ m}^3/\text{min}$$

From Eq. (5-3) we calculate the actual flow rate:

$$Q_A = Q_T \eta_v = 0.0351 \text{ m}^3/\text{min} \times 0.94 = 0.0330 \text{ m}^3/\text{min}$$

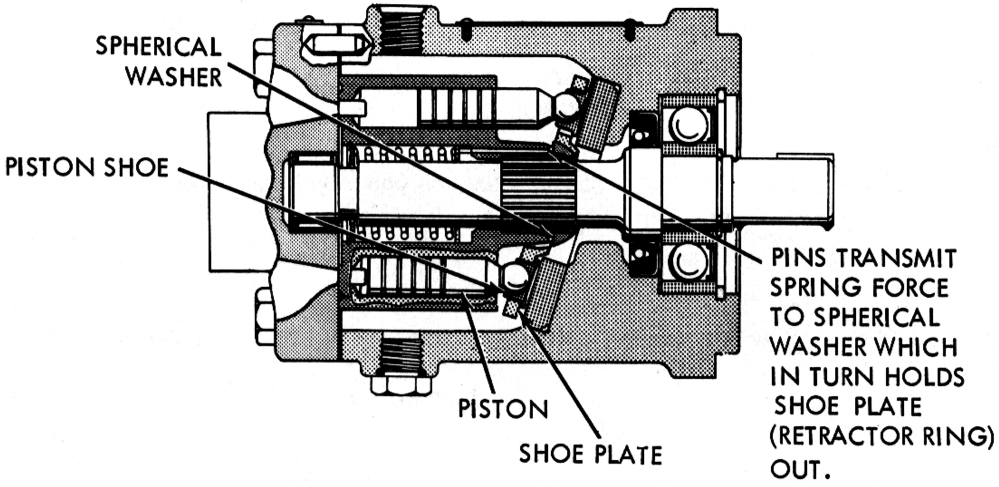
To convert to flow rate in units of L/s, we perform the following manipulation of units:

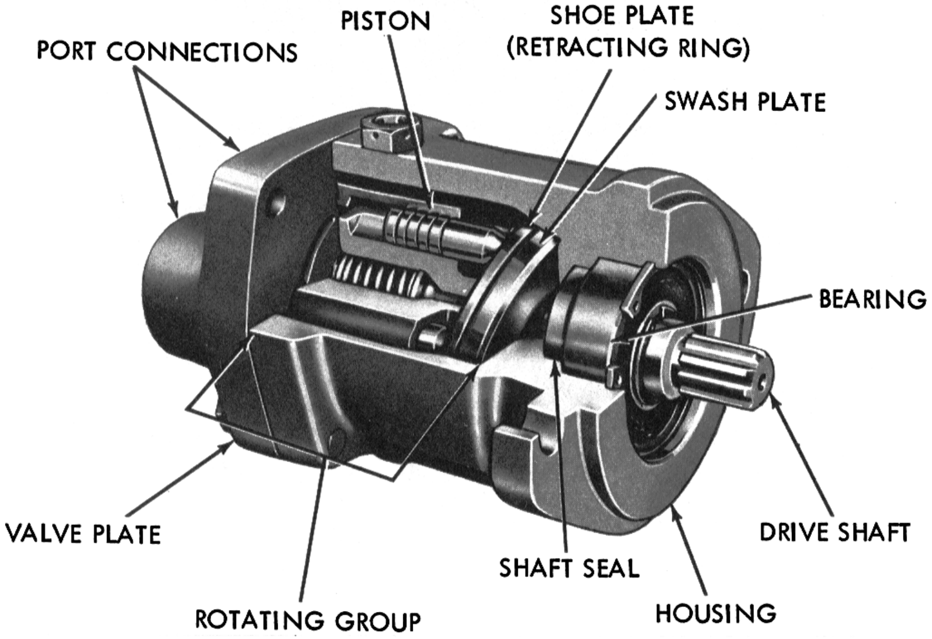
$$Q_A(\text{L/s}) = Q_A(\text{m}^3/\text{min}) \times \frac{1 \text{ min}}{60 \text{ s}} \times \frac{1 \text{ L}}{0.001 \text{ m}^3}$$

$$= 0.0330 \times \frac{1}{60} \times \frac{1}{0.001} = 0.550 \text{ L/s}$$

In-Line Piston Pump (Swash Plate Design)

- In this type, the cylinder block and drive shaft are located on the same centerline.
- The pistons are connected to a shoe plate, which bears against an angled swash plate.
- As the cylinder rotates, the pistons reciprocate because the piston shoes follow the angled surface of the swash plate.
- The outlet and inlet ports are located in the valve plate so that the pistons pass the inlet as they are being pulled out and pass the outlet as they are being forced back in.





In-Line Piston Pump (Swash Plate Design)

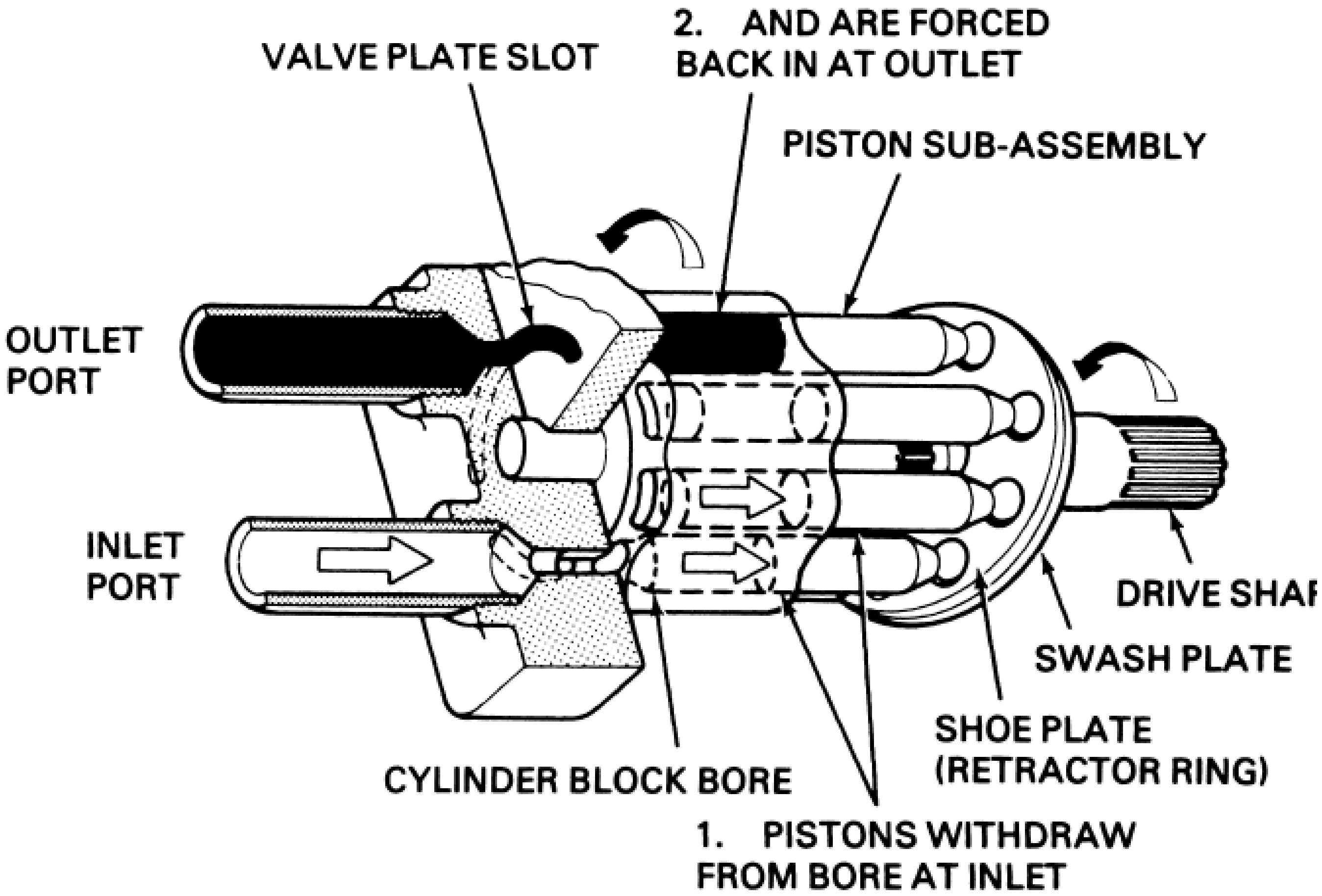
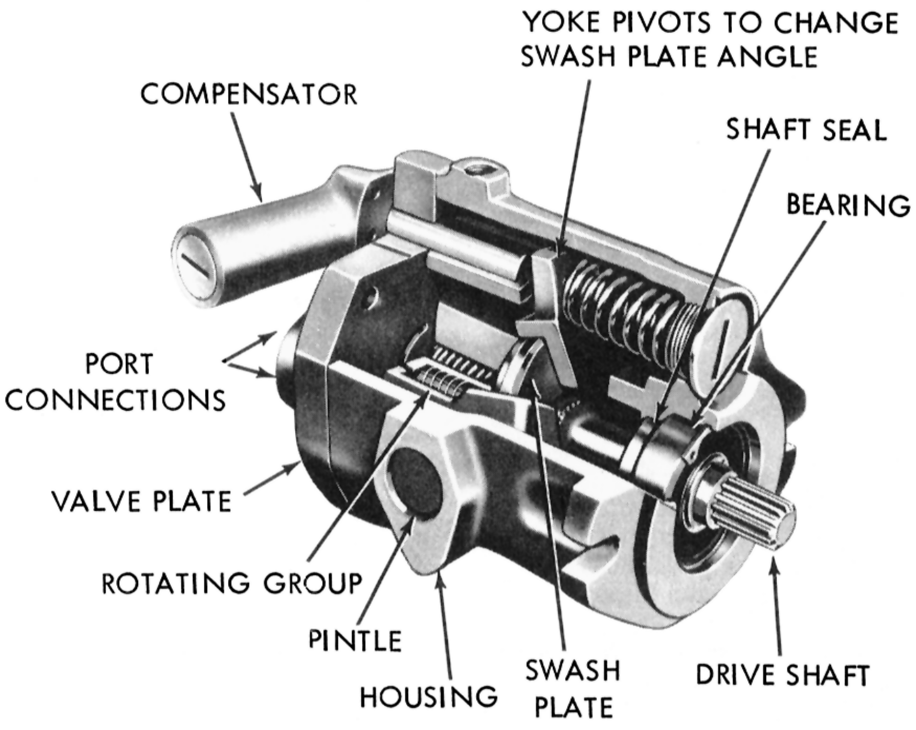


Figure 5-27. Swash plate causes pistons to reciprocate. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

In-Line Piston Pump (Swash Plate Design)

- This type of pump can also be designed to have variable displacement capability.
- In such a design, the swash plate is mounted in a movable yoke, as depicted in Figure 5-28.The swash plate angle can be changed by pivoting the yoke on pintles (see Figure 5-29 for the effect of swash plate angle on piston stroke).
- Positioning of the yoke can be accomplished by manual operation, servo control, or a compensator control, as shown in Figure 5-28. The maximum swash plate angle is limited to 17.5 ° by construction.



In-Line Piston Pump (Swash Plate Design)

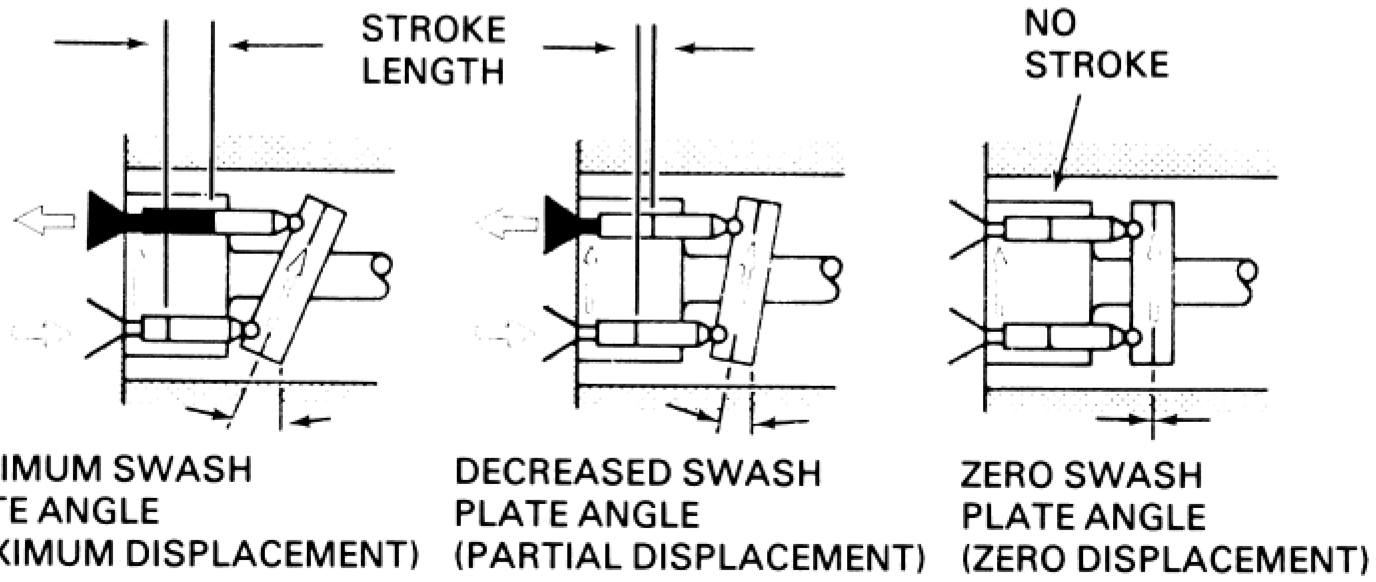


Figure 5-29. Variation in pump displacement. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

Radial Piston Pump

- This design consists of a pintle to direct fluid in and out of the cylinders, a cylinder barrel with pistons, and a rotor containing a reaction ring.
- The pistons remain in constant contact with the reaction ring due to centrifugal force and back pressure on the pistons.
- For pumping action, the reaction ring is moved eccentrically with respect to the pintle or shaft axis. As the cylinder barrel rotates, the pistons on one side travel outward.
- This draws in fluid as each cylinder passes the suction ports of the pintle.
- When a piston passes the point of maximum eccentricity, it is forced inward by the reaction ring. This forces the fluid to enter the discharge port of the pintle.
- In some models, the displacement can be varied by moving the reaction ring to change the piston stroke.
- This pump is available in three sizes (2.40-, 3.00-, and 4.00-in3 volumetric displacements) and weighs approximately 60 lb.
- Variable displacement is accomplished by hydraulic rather than mechanical means and is responsive to discharge line pressure.

Radial Piston Pump

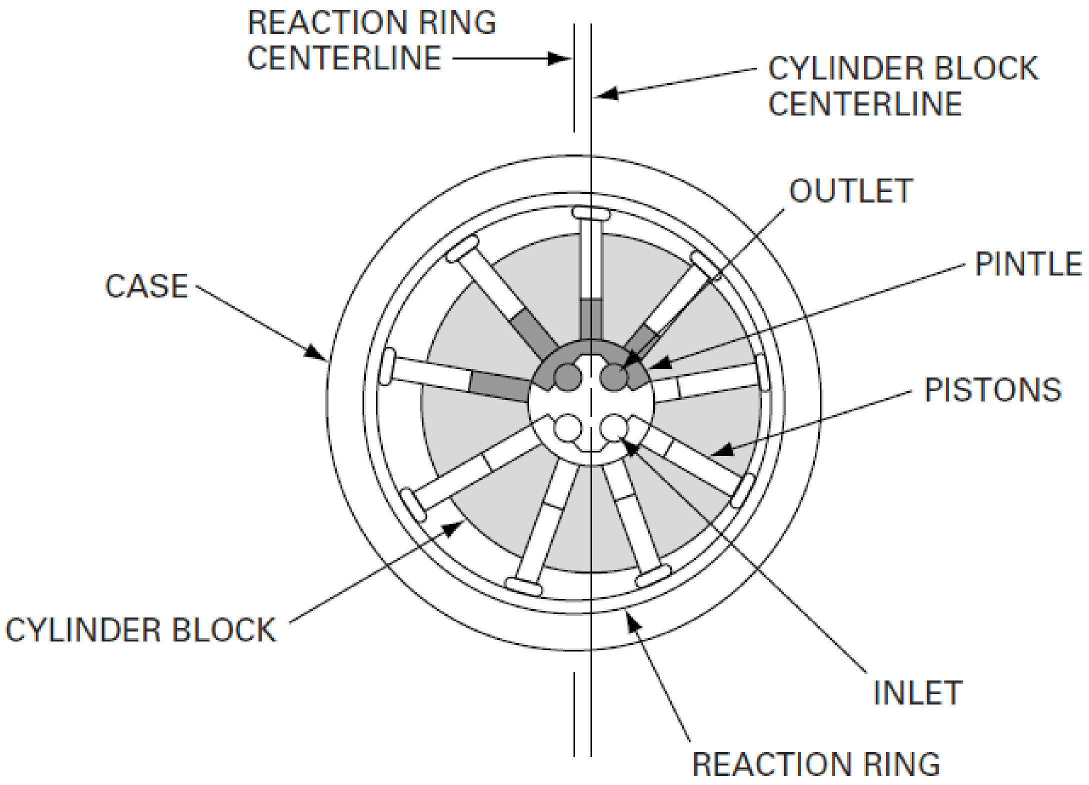
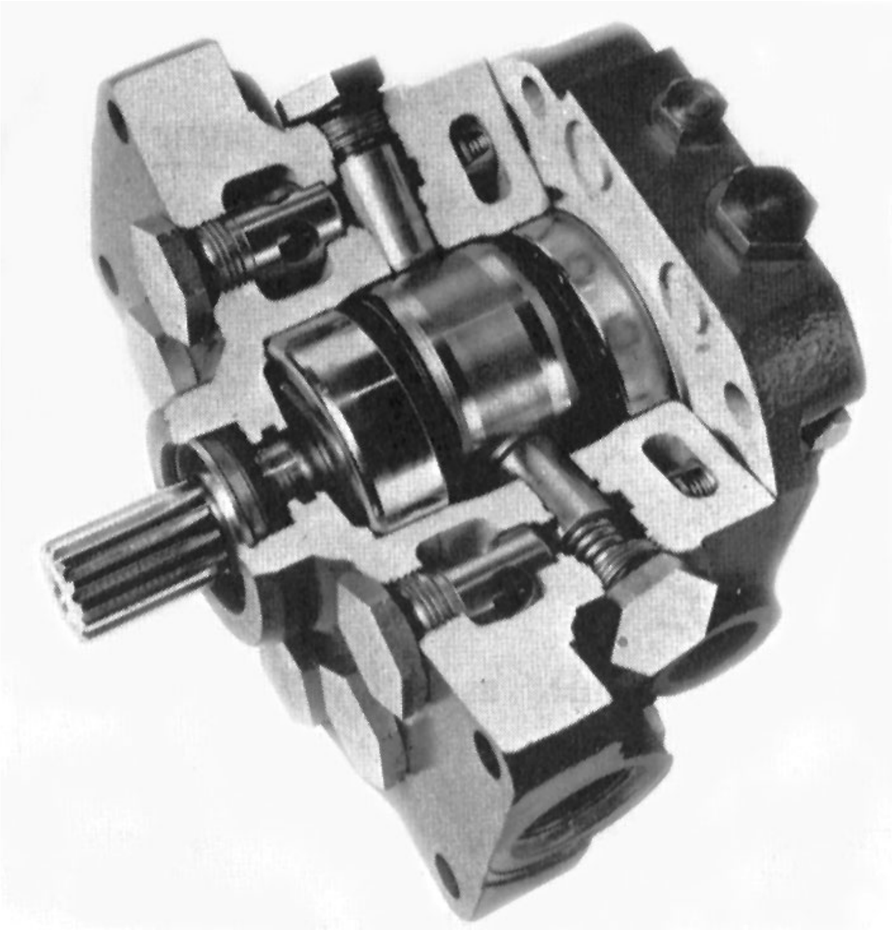


Figure 5-30. Operation of a radial piston pump. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*



PUMP PERFORMANCE

- The performance of a pump is primarily a function of the precision of its manufacture.
- Components must be made to close tolerances, which must be maintained while the pump is operating under design conditions.
- The maintenance of close tolerances is accomplished by designs that have mechanical integrity and balanced pressures.
- Theoretically the ideal pump would be one having zero clearance between all mating parts.
- Although this is not feasible, working clearances should be as small as possible while maintaining proper oil films for lubrication between rubbing parts.

Pump Efficiencies

- Pump manufacturers run tests to determine performance data for their various types of pumps.
- The overall efficiency of a pump can be computed by comparing the hydraulic power output of the pump to the mechanical input power supplied by the prime mover.
- Overall efficiency can be broken into two distinct components called volumetric efficiency and mechanical efficiency.
- These three efficiencies are discussed on the following pages.

Pump Efficiencies

1. Volumetric efficiency (η_v). *Volumetric efficiency* indicates the amount of leakage that takes place within the pump. This involves considerations such as manufacturing tolerances and flexing of the pump casing under design pressure operating conditions:

$$\eta_v = \frac{\text{actual flow-rate produced by pump}}{\text{theoretical flow-rate pump should produce}} = \frac{Q_A}{Q_T} \qquad \textbf{(5-7)}$$

Volumetric efficiencies typically run from 80% to 90% for gear pumps, 82% to 92% for vane pumps, and 90% to 98% for piston pumps. Note that when substituting efficiency values into equations, decimal fraction values should be used instead of percentage values. For example, an efficiency value of 90% would be represented by a value of 0.90.

Pump Efficiencies

2. Mechanical efficiency (η_m). *Mechanical efficiency* indicates the amount of energy losses that occur for reasons other than leakage. This includes friction in bearings and between other mating parts. It also includes energy losses due to fluid turbulence. Mechanical efficiencies typically run from 90% to 95%.

$$\eta_m = \frac{\text{pump output power assuming no leakage}}{\text{actual power delivered to pump}}$$

Using English units and horsepower for power yields

$$\eta_m = \frac{pQ_T/1714}{T_A N/63{,}000} \qquad \textbf{(5-8)}$$

Pump Efficiencies

In metric units, using watts for power,

$$\eta_m = \frac{pQ_T}{T_A N} \tag{5-8M}$$

The parameters of Eqs. (5-8) and (5-8M) are defined as follows in conjunction with Figure 5-32.

p = pump discharge pressure (psi, Pa)
Q_T = pump theoretical flow rate (gpm, m³/s)
T_A = actual torque delivered to pump (in · lb, N · m)
N = pump speed (rpm, rad/s)

Mechanical efficiency can also be computed in terms of torques:

$$\eta_m = \frac{\text{theoretical torque required to operate pump}}{\text{actual torque delivered to pump}} = \frac{T_T}{T_A} \tag{5-9}$$

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

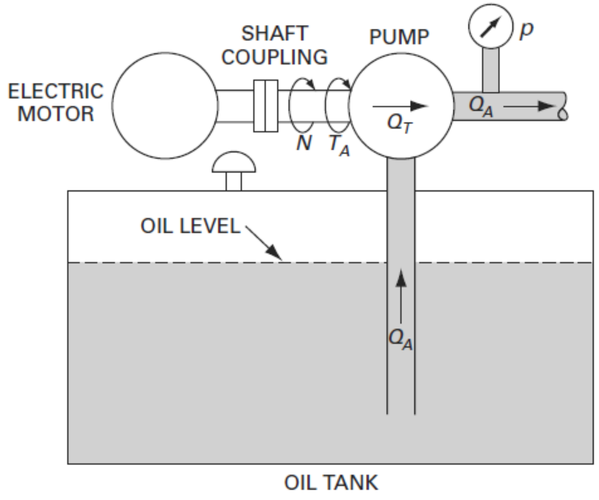


Figure 5-32. Terms involving pump mechanical efficiency.

Pump Efficiencies

Equations for evaluating the theoretical torque and the actual torque are as follows:

Theoretical Torque

$$T_T(\text{in} \cdot \text{lb}) = \frac{V_D(\text{in}^3) \times p(\text{psi})}{2\pi} \qquad \textbf{(5-10)}$$

or

$$T_T(\text{N} \cdot \text{m}) = \frac{V_D(\text{m}^3) \times p(\text{Pa})}{2\pi} \qquad \textbf{(5-10M)}$$

Actual Torque

$$T_A = \frac{\text{actual horsepower delivered to pump} \times 63{,}000}{N(\text{rpm})} \qquad \textbf{(5-11)}$$

or

$$T_A = \frac{\text{actual power delivered to pump (W)}}{N(\text{rad/s})} \qquad \textbf{(5-11M)}$$

where

$$N(\text{rad/s}) = \frac{2\pi}{60} N\,(\text{rpm})$$

Pump Efficiencies

3. Overall efficiency (η_o). The overall efficiency considers all energy losses and hence is defined as follows:

$$\text{overall efficiency} = \frac{\text{actual power deliveed by pump}}{\text{actual power delivered to pump}} \tag{5-12}$$

The overall efficiency can also be represented mathematically as follows:

$$\eta_o = \eta_v \times \eta_m \tag{5-13}$$

Substituting Eq. (5-7) and (5-8) into Eq. (5-13), we have (for English units):

$$\eta_o = \eta_v \times \eta_m = \frac{Q_A}{Q_T} \times \frac{pQ_T/1714}{T_A N/63{,}000}$$

Canceling like terms yields the desired result showing the equivalency of Eq. (5-12) and Eq. (5-13).

$$\eta_o = \frac{pQ_A/1714}{T_A N/63{,}000} = \frac{\text{actual horsepower delivered by pump}}{\text{actual horsepower delivered to pump}} \tag{5-14}$$

Repeating this substitution for metric units using Eqs. (5-7), (5-8M), and (5-13) yields:

$$\eta_o = \frac{pQ_A}{T_A N} = \frac{\text{actual power delivered by pump}}{\text{actual power delivered to pump}} \tag{5-14M}$$

Note that the actual power delivered to a pump from a prime mover via a rotating shaft is called *brake power* and the actual power delivered by a pump to the fluid is called *hydraulic power*.

Examples

EXAMPLE 5-8

A pump has a displacement volume of 5 in³. It delivers 20 gpm at 1000 rpm and 1000 psi. If the prime mover input torque is 900 in · lb,

a. What is the overall efficiency of the pump?

b. What is the theoretical torque required to operate the pump?

Solution

a. Use Eq. (5-2) to find the theoretical flow rate:

$$Q_T = \frac{V_D N}{231} = \frac{(5)(1000)}{231} = 21.6 \text{ gpm}$$

Next, solve for the volumetric efficiency:

$$\eta_v = \frac{Q_A}{Q_T} = \frac{20}{21.6} = 0.926 = 92.6\%$$

Then solve for the mechanical efficiency:

$$\eta_m = \frac{pQ_T/1714}{T_A N/63{,}000} = \frac{[(1000)(21.6)]/1714}{[(900)(1000)]/63{,}000} = 0.881 = 88.1\%$$

Finally, we solve for the overall efficiency:

$$\eta_o = \eta_v \eta_m = 0.926 \times 0.881 = 0.816 = 81.6\%$$

b. Using Eq. (5-9) to solve for the theoretical torque we have

$$T_T = T_A \eta_m = 900 \times 0.881 = 793 \text{ in} \cdot \text{lb}$$

Thus, due to mechanical losses within the pump, 900 in · lb of torque are required to drive the pump instead of 793 in · lb.

Pump Performance Curves

Pump manufacturers specify pump performance characteristics in the form of graphs. Test data are obtained initially in tabular form and then put in graphical form for better visual interpretation. Figure 5-33 represents typical performance curves obtained for a 6-in^3 variable displacement pump operating at full displacement. The upper graph gives curves of overall and volumetric efficiencies as a function of pump speed (rpm) for pressure levels of 3000 and 5000 psi. The lower graph gives curves of pump input horsepower (hp) and pump output flow (gpm) as a function of pump speed for the same two pressure levels.

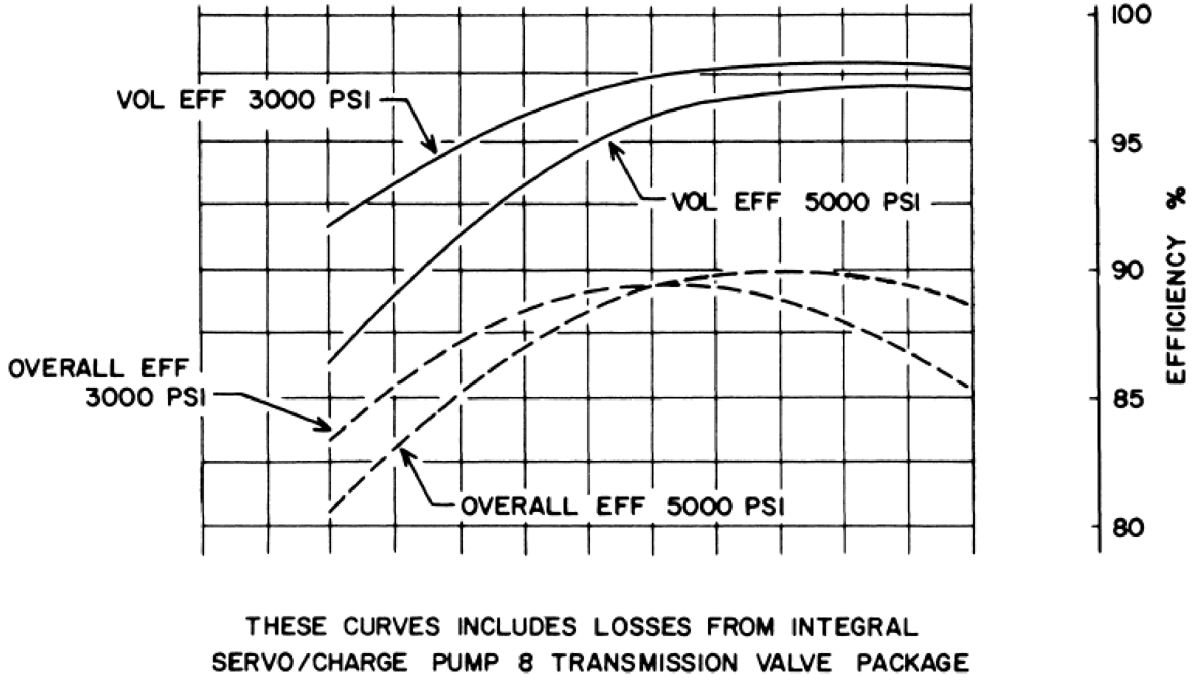
Performance curves for the radial piston pump of Figure 5-31 are presented in Figure 5-34. Recall that this pump comes in three different sizes:

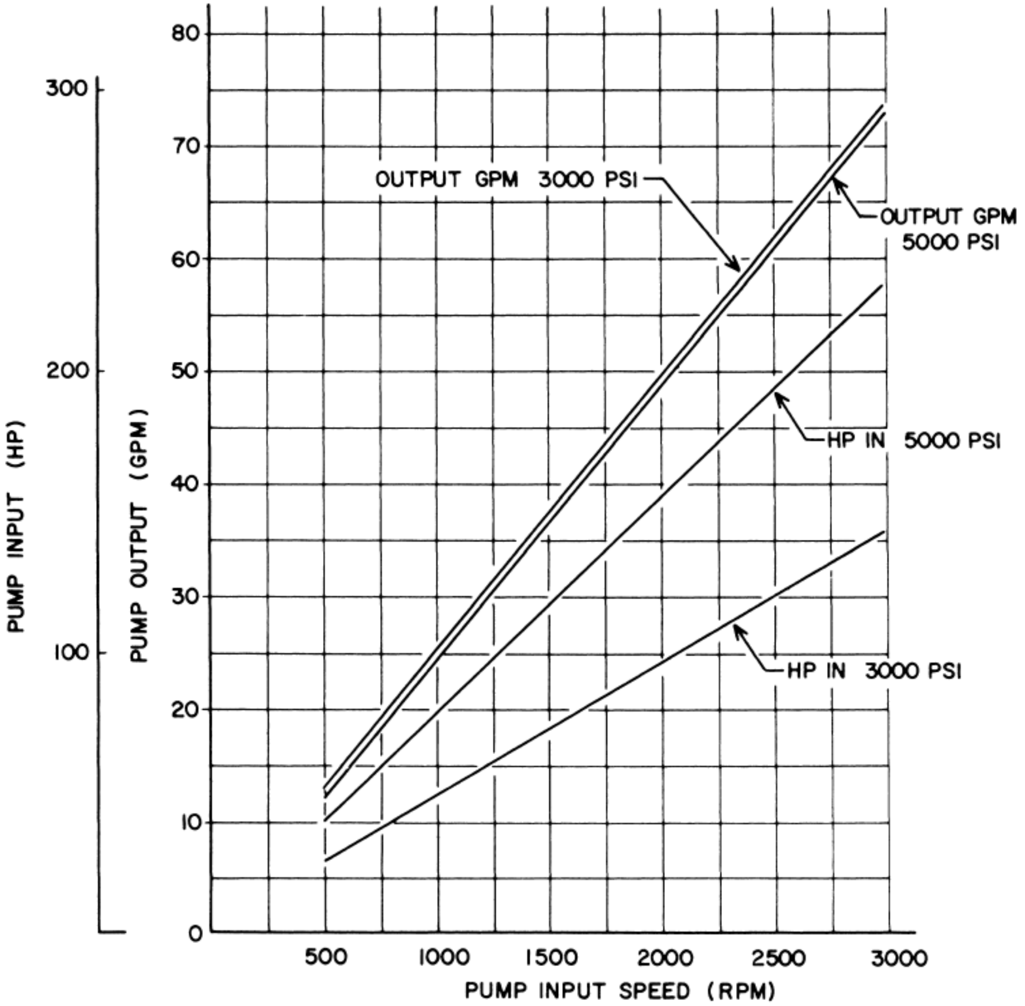
PR24: 2.40-in^3 displacement

PR30: 3.00-in^3 displacement

PR40: 4.00-in^3 displacement

Pump Performance Curves





Performance curves for 6-in3 variable displacement piston pump.

Pump Performance Curves

- Thus, there are three curves on two of the graphs. Observe the linear relationship between discharge flow (gpm) and pump speed (rpm).
- Also note that the discharge flow of these pumps is nearly constant over a broad pressure range.
- Discharge flow is infinitely variable between the point of inflection on the constant-discharge portion of the curve and zero flow.
- The volumetric and overall efficiency curves are based on a 2000-psi pump pressure.

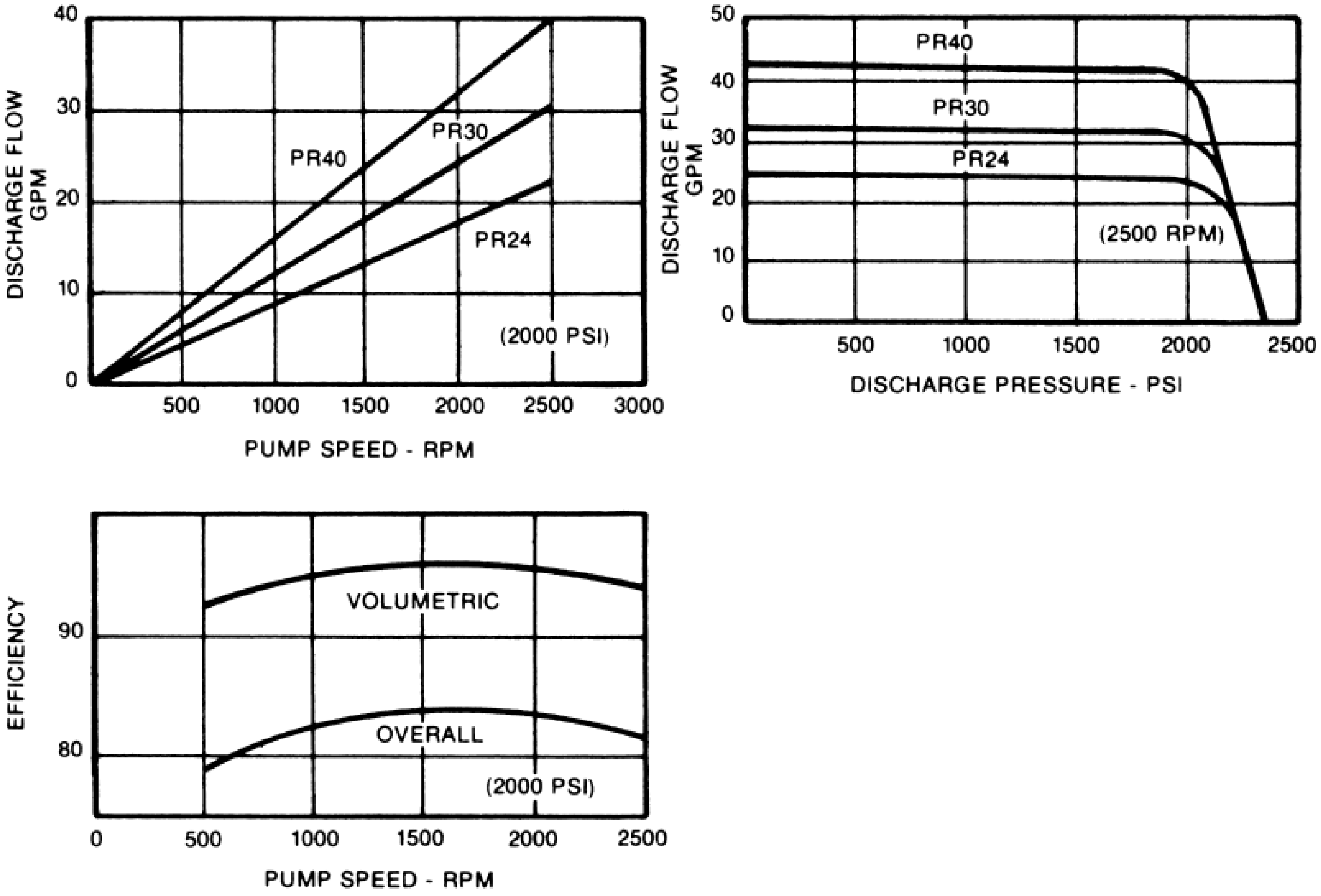


Figure 5-34. Performance curves of radial piston pumps. *(Courtesy of Deere & Co., Moline, Illinois.)*

Pump Performance Comparison Factors

- In general, Gear pumps are the least expensive but also provide the lowest level of performance.
- In addition, gear pump efficiency is rapidly reduced by wear, which contributes to high maintenance costs.
- The volumetric efficiency of gear pumps is greatly affected by the following leakage losses, which can rapidly accelerate due to wear:

1. Leakage around the outer periphery of the gears
2. Leakage across the faces of the gears
3. Leakage at the points where the gear teeth make contact

- Gear pumps are simple in design and compact in size. Therefore, they are the most common type of pump used in fluid power systems.
- The greatest number of applications of gear pumps are in the mobile equipment and machine tool fields.

Pump Performance Comparison Factors

- Vane pump efficiencies and costs fall between those of gear and piston pumps.
- Vane pumps have good efficiencies and last for a reasonably long time. However, continued satisfactory performance necessitates clean oil with good lubricity.
- Excessive shaft speeds can cause operating problems. Leakage losses in vane pumps occur across the faces of the rotor and between the bronze wear plates and the pressure ring.
- Piston pumps are the most expensive and provide the highest level of overall performance. They can be driven at high speeds (up to 5000 rpm) to provide a high horsepower-to-weight ratio.
- They produce essentially a nonpulsating flow and can operate at the highest pressure levels.
- Due to very close-fitting pistons, they have the highest efficiencies. Since no side loads occur to the pistons, the pump life expectancy is at least several years.
- However, because of their complex design, piston pumps cannot normally be repaired in the field.

Pump Performance Comparison Factors

PUMP TYPE	PRESSURE RATING (PSI)	SPEED RATING (RPM)	OVERALL EFFICIENCY (PERCENT)	HP PER LB RATIO	FLOW CAPACITY (GPM)	COST (DOLLARS PER HP)
EXTERNAL GEAR	2000–3000	1200–2500	80–90	2	1–150	4–8
INTERNAL GEAR	500–2000	1200–2500	70–85	2	1–200	4–8
VANE	1000–2000	1200–1800	80–95	2	1–80	6–30
AXIAL PISTON	2000–12,000	1200–3000	90–98	4	1–200	6–50
RADIAL PISTON	3000–12,000	1200–1800	85–95	3	1–200	5–35

Figure 5-35. Comparison of various performance factors for pumps.

Pump selection

- Pumps are selected by taking into account a number of considerations for a complete hydraulic
- system involving a particular application. Among these considerations are flow-rate requirements (gpm), operating speed (rpm), pressure rating (psi), performance, reliability, maintenance, cost, and noise.
- The selection of a pump typically entails the following sequence of operations:
- 1. Select the actuator (hydraulic cylinder or motor) that is appropriate based on the loads
- encountered.
- 2. Determine the flow-rate requirements. This involves the calculation of the flow rate necessary
- to drive the actuator to move the load through a specified distance within a given time limit.
- 3. Select the system pressure. This ties in with the actuator size and the magnitude of the
- resistive force produced by the external load on the system. Also involved here is the total amount
- of power to be delivered by the pump.
- 4. Determine the pump speed and select the prime mover. This, together with the flow-rate
- calculation, determines the pump size (volumetric displacement).
- 5. Select the pump type based on the application (gear, vane, or piston pump
- and fixed or variable displacement).

Pump selection

- 6. Select the reservoir and associated plumbing, including piping, valving, filters and strainers,
- and other miscellaneous components such as accumulators.
- 7. Consider factors such as noise levels, horsepower loss, need for a heat exchanger due to generated heat, pump wear, and scheduled maintenance service to provide a desired life of the total system.
- 8. Calculate the overall cost of the system.
- Normally the sequence of operation is repeated several times with different sizes and types of components.
- After the procedure is repeated for several alternative systems, the best overall system is selected for the given application. This process is called optimization.
- It means determining the ultimate selection of a combination of system components to produce the most efficient overall system at minimum cost commensurate with the requirements of a particular application.

PUMP PERFORMANCE RATINGS IN METRIC UNITS

Performance data for hydraulic pumps are measured and specified in metric units as well as English units. Figure 5-39 shows actual performance data curves for a variable displacement, pressure-compensated vane pump operating at 1200 rpm. The curves give values of flow rate (gpm), efficiency, and power (hp and kW) versus output pressure (psi and bars). This particular pump (see Figure 5-39) can operate at speeds between 1000 and 1800 rpm, is rated at 2540 psi (175 bars), and has a nominal displacement volume of 1.22 in^3 (20 cm^3 or 0.02 L). Although the curves give flow rates in gpm, metric flow rates of liters per minute (Lpm) are frequently specified.

PUMP PERFORMANCE RATINGS IN METRIC UNITS

EXAMPLE 5-9

A pump has a displacement volume of 100 cm³. It delivers 0.0015 m³/s at 1000 rpm and 70 bars. If the prime mover input torque is 120 N · m,

a. What is the overall efficiency of the pump?

b. What is the theoretical torque required to operate the pump?

Solution

a. Using Eq. (5-2M), where the volumetric displacement is

$$V_D = 100\ \text{cm}^3/\text{rev} \times \left(\frac{1\ \text{m}}{100\ \text{cm}}\right)^3 = 0.000100\ \text{m}^3/\text{rev}$$

we have

$$Q_T = V_D N = (0.000100\ \text{m}^3/\text{rev})\left(\frac{1000}{60}\ \text{rev/s}\right) = 0.00167\ \text{m}^3/\text{s}$$

Next, solve for the volumetric efficiency:

$$\eta_v = \frac{Q_A}{Q_T} = \frac{0.0015}{0.00167} = 0.898 = 89.8\%$$

PUMP PERFORMANCE RATINGS IN METRIC UNITS

Then solve for the mechanical efficiency:

$$\eta_m = \frac{pQ_T}{T_A N} = \frac{(70 \times 10^5\ \mathrm{N/m^2})(0.00167\ \mathrm{m^3/s})}{(120\ \mathrm{N \cdot m})\left(1000 \times \frac{2\pi}{60}\ \mathrm{rad/s}\right)}$$

$$\eta_m = \frac{11{,}690\ \mathrm{N \cdot m/s}}{12{,}570\ \mathrm{N \cdot m/s}} = 0.930 = 93.0\%$$

Note that the product $T_A N$ gives power in units of N · m/s (W) where torque (T_A) has units of N · m and shaft speed has units of rad/s. Finally, we solve for the overall efficiency:

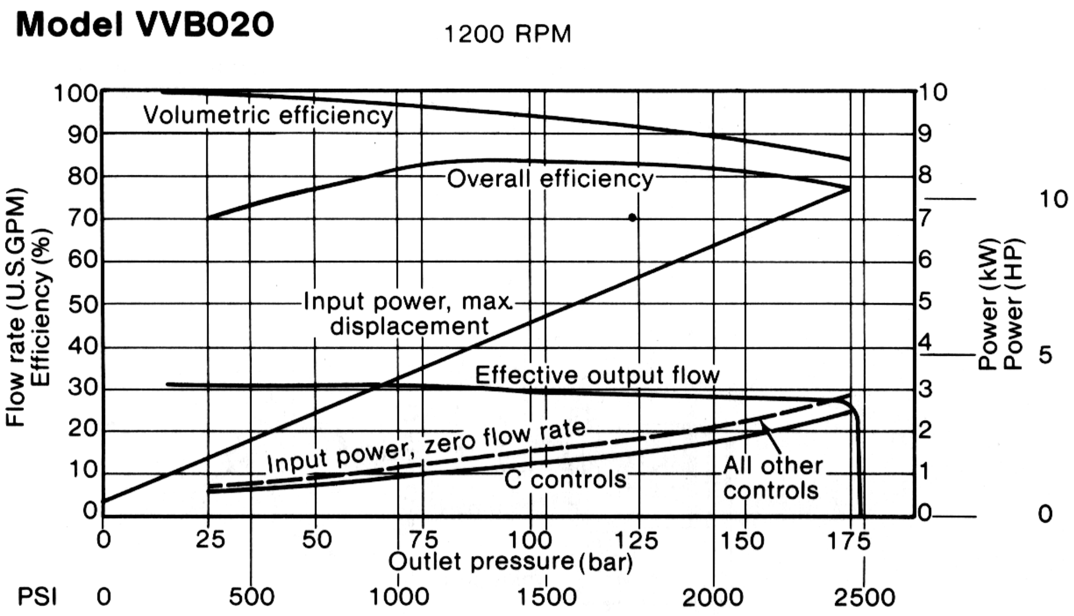
$$\eta_o = \eta_v \eta_m = 0.898 \times 0.930 = 0.835 = 83.5\%$$

b.

$$T_T = T_A \eta_m = (120)(0.93) = 112\ \mathrm{N \cdot m}$$

Thus, due to mechanical losses within the pump, 120 N · m of torque are required to drive the pump instead of 112 N · m.

PUMP PERFORMANCE RATINGS IN METRIC UNITS



PUMP PERFORMANCE RATINGS IN METRIC UNITS

EXAMPLE 5-10

The pump in Example 5-9 is driven by an electric motor having an overall efficiency of 85%. The hydraulic system operates 12 hours per day for 250 days per year. The cost of electricity is \$0.11 per kilowatt hour. Determine

a. The yearly cost of electricity to operate the hydraulic system

b. The amount of the yearly cost of electricity that is due to the inefficiencies of the electric motor and pump

Solution

a. First, we calculate the mechanical input power the electric motor delivers to the pump. Per Eq. (3-37), we have

$$\text{Pump input power (kW)} = \frac{T_A(\text{N}\cdot\text{m}) \times N(\text{rpm})}{9550} = \frac{120 \times 1000}{9550} = 12.6 \text{ kW}$$

Next, we calculate the electrical input power the electric motor receive:

$$\text{Electric motor input power} = \frac{\text{Electric motor output power}}{\text{Electric motor overall efficiency}}$$

Since the electric motor output power equals the pump input power we have

$$\text{Electric motor input power} = \frac{12.6 \text{ kW}}{0.85} = 14.8 \text{ kW}$$

Finally, we determine the yearly cost of electricity:

$$\begin{aligned}\text{yearly cost} &= \text{power rate} \times \text{time per year} \times \text{unit cost of electricity}\\ &= 14.8 \text{ kW} \times 12\,\frac{\text{hr}}{\text{day}} \times 250\,\frac{\text{days}}{\text{year}} \times \frac{\$0.11}{\text{kw hr}} = \$4884/\text{yr}\end{aligned}$$

PUMP PERFORMANCE RATINGS IN METRIC UNITS

b. The total kW loss equals the kW loss due to the electric motor plus the kW loss due to the pump. Thus, we have

$$\text{Total kW loss} = (1 - 0.85) \times 14.8 + (1 - 0.835) \times 12.6$$
$$= 2.22 + 2.08 = 4.30 \text{ kW}$$

$$\text{Yearly cost due to inefficiencies} = \frac{4.3}{14.8} \times \$4884/\text{yr} = \$1419/\text{yr}$$

Since 4.3/14.8 = 0.29, we conclude that 29% of the total cost of electricity is due to the inefficiencies of the electric motor and pump. This also means that only 71% of the electrical power entering the electric motor is transferred into hydraulic power at the pump outlet port.

END