Pumps

Pump Theory

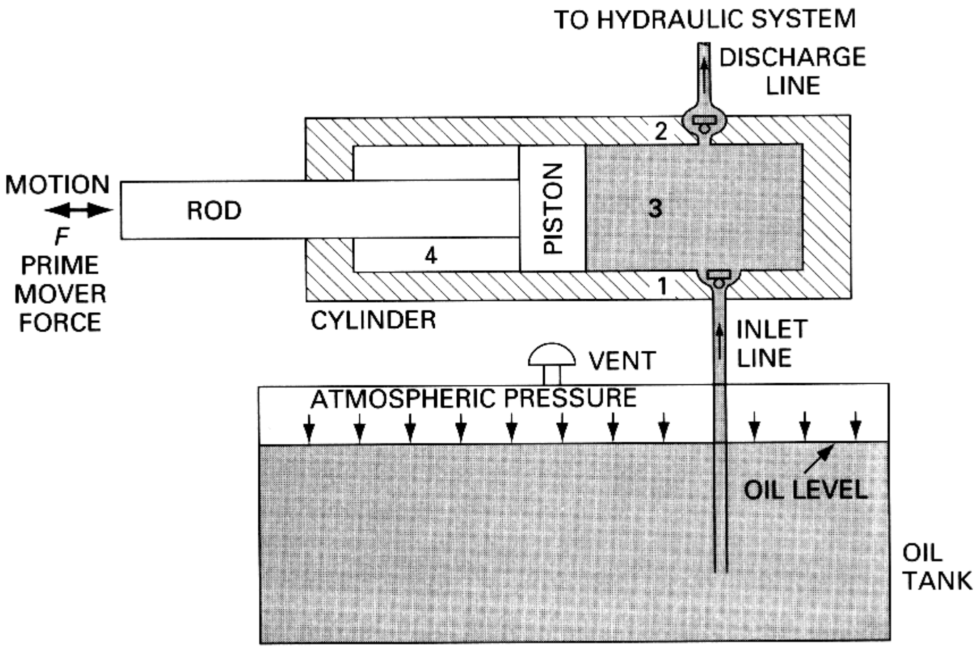
A Pump, which is the Heart of a Hydraulic System, converts Mechanical Energy into Hydraulic Energy. The Mechanical Energy is delivered to the Pump via a Prime Mover such as an Electric Motor. Due to Mechanical Action, the Pump Creates a Partial Vacuum at its Inlet. This permits Atmospheric Pressure to Force the Fluid through the inlet line and into the Pump. The Pump then pushes the Fluid into the Hydraulic System.

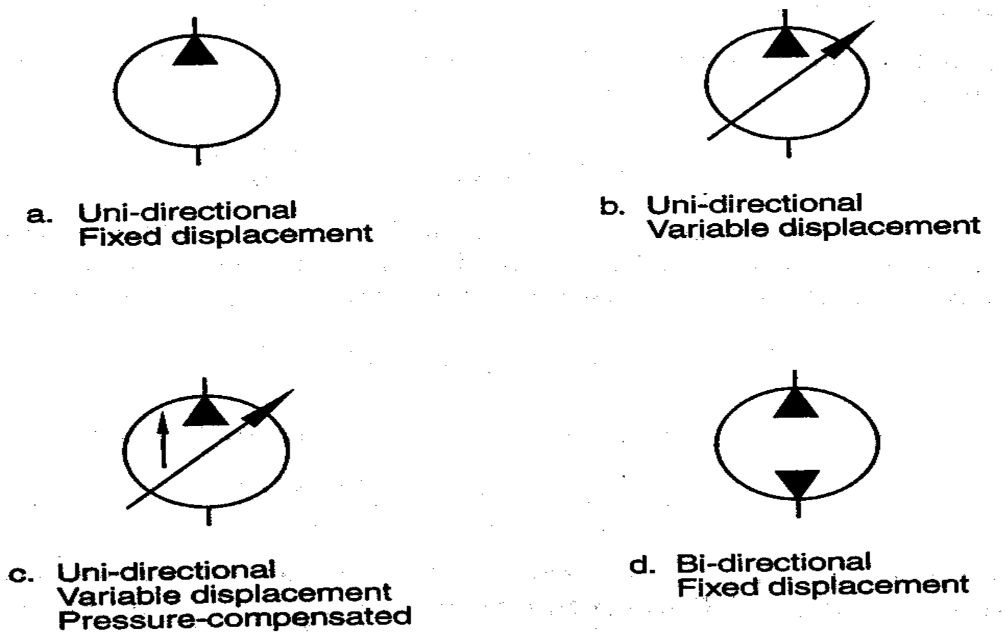
Pumps operate on the Principle whereby a Partial Vacuum is created at the Pump Inlet due to the Internal Operation of the Pump. This allows Atmospheric Pressure to push the Fluid out of the Oil Tank (Reservoir) and into the Pump Intake. The Pump then mechanically pushes the Fluid out the Discharge Line.

For Example

Note that this Pump contains Two Ball Check Valves, which are described as follows:

Check Valve 1 is connected to the Pump Inlet Line and allows fluid to enter the pump only at this location. Check Valve 2 is connected to the Pump discharge Line and allows fluid to leave the pump only at this location. As the Piston is pulled to the Left, A Partial Vacuum is generated in Cavity 3, because the close tolerance between the Piston and Cylinder (or the use of Piston Ring Seals) prevents Air inside Cavity 4 from traveling into Cavity 3. This Flow of Air, if allowed to occur, would Destroy the Vacuum. This Vacuum holds the Ball of Check Valve 2 against its seat (lower position) and allows Atmospheric Pressure to push Fluid from the Reservoir into the Pump via Check Valve 1. This Inlet Flow occurs because the Force of the Fluid pushes the ball of Check Valve 1 Off its Seat. When the Piston is pushed to the Right, the fluid movement closes Inlet Valve 1 and Opens Outlet Valve 2. The Quantity of Fluid, displaced by the piston, is forcibly ejected Out the Discharge Line leading to the Hydraulic System. The Volume of Oil displaced by the Piston during the Discharge Stroke is called the Displacement Volume of the Pump.





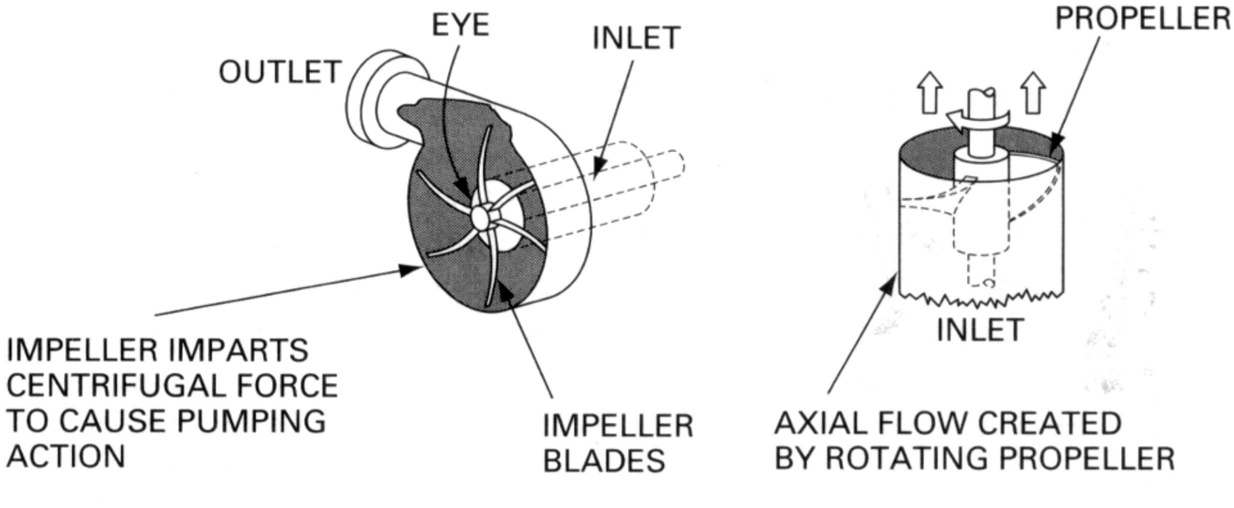
Pump Classification

1. Dynamic (Non-Positive Displacement) Pumps
2. Positive Displacement Pumps

1. Dynamic (Non-Positive Displacement) Pumps

This type is generally used for: Low-Pressure, High-Volume Flow Applications because they are Not capable of withstanding High Pressures. They are of Little Use in the Fluid Power Field. This type of Pump is primarily used for transporting fluids from One Location to Another. The Two Most Common Types of Dynamic Pumps are: The Centrifugal (Impeller) and The Axial flow Propeller pumps.

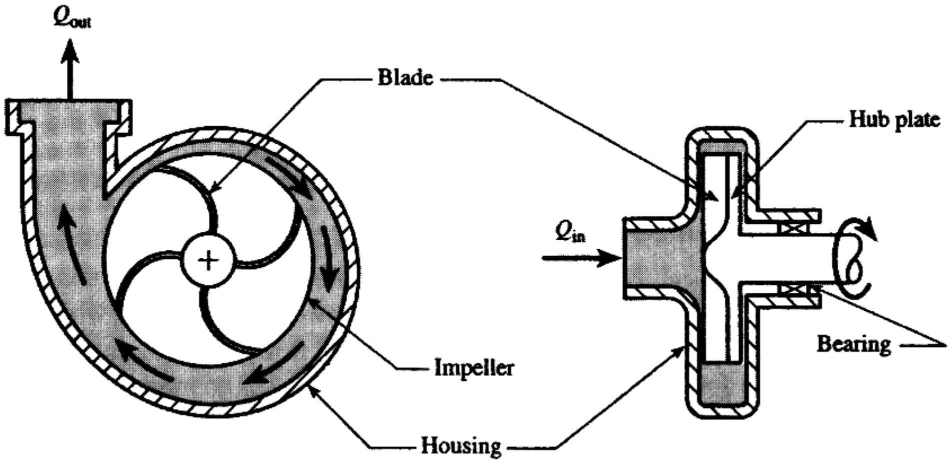
Although these Pumps provide Smooth Continuous Flow, their Flow Output is reduced as circuit Resistance is increased and thus are Rarely Used in Fluid Power Systems. In Dynamic Pumps there is a great deal of Clearance between the Rotating Impeller or Propeller and the Stationary Housing. As the Resistance of the External System starts to increase, some of the Fluid Slips Back into the Clearance Spaces, causing a Reduction in the Discharge flow-rate. This Slippage is due to the Fluid Follows the Path of Least Resistance. When the Resistance of the External System becomes Infinitely Large (For Example, a valve is closed in the outlet line) Pump will produce No Flow.



CENTRIFUGAL (IMPELLER) TYPE AXIAL (PROPELLER) TYPE

Dynamic Pumps are not Self-Priming unlike Positive Pumps. If the fluid is being pumped from a

Reservoir located Below the Pump, priming is required. Priming is the Pre-Filling of the Pump Housing and Inlet pipe with Fluid so that the Pump can initially draw in the fluid and pump it efficiently.

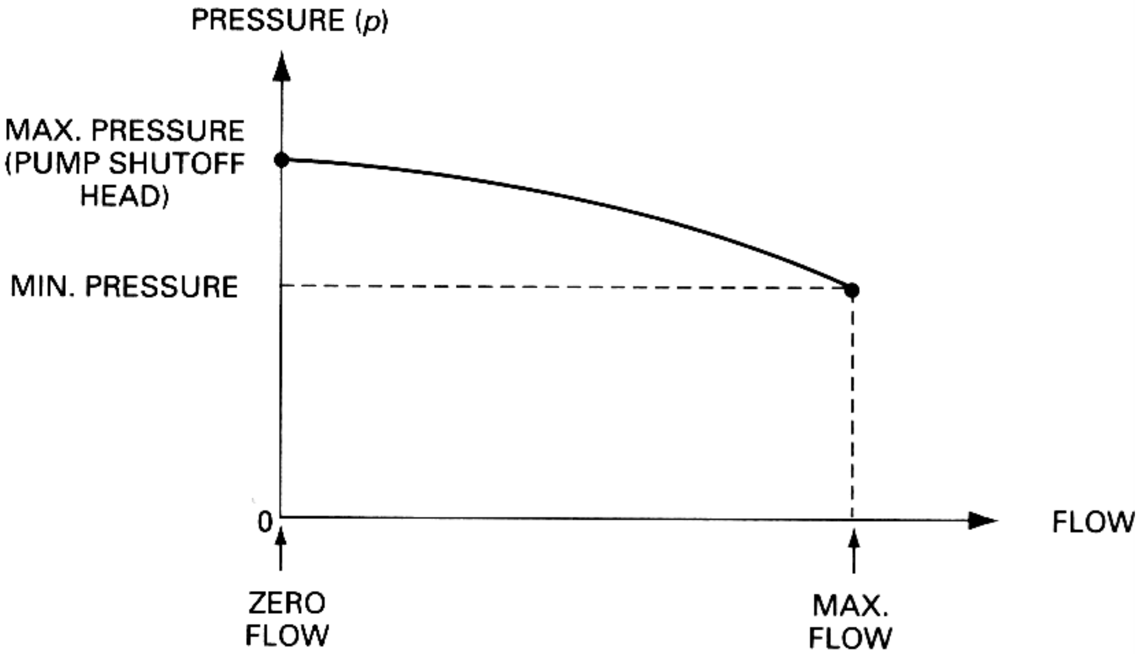


(a) Construction features

The Construction features of a Centrifugal Pump; the Most commonly used Type of Dynamic Pump.

The Operation of a Centrifugal Pump is as follows. The Fluid enters at the center of the impeller and is picked up by the Rotating Impeller. As the Fluid rotates with the Impeller, the Centrifugal Force causes the Fluid to move radially outward. This causes the fluid to flow through the outlet discharge port of the housing.

One of the interesting Characteristics of a Centrifugal Pump is its behavior when there is No Demand for Fluid. In this case, there is No Need for a Pressure Relief Valve to Prevent Pump Damage. The Tips of the Impeller Blades merely slosh through the Fluid, and the Rotational Speed maintains a Fluid Pressure corresponding to the Centrifugal Force established. Although Dynamic Pumps provide Smooth continuous Flow, their Output Flow rate is reduced as Resistance to flow is increased.



(b) Pressure versus flow curve

This figure is shown for Centrifugal Pumps, where Pump Pressure is plotted versus Pump Flow. The Maximum Pressure is called the Shutoff Head because an external circuit valve is closed which shuts off the flow. As the External Resistance decreases due to the Valve being opened, the Flow increases at the expense of reduced Pressure.

2. Positive Displacement Pumps

This type is universally used for Fluid Power Systems. It ejects a Fixed Amount of Fluid into the Hydraulic System per revolution of pump shaft rotation. Such a Pump is capable of Overcoming: the Pressure resulting from the Mechanical Loads on the System as well as the Resistance to Flow due to Friction.

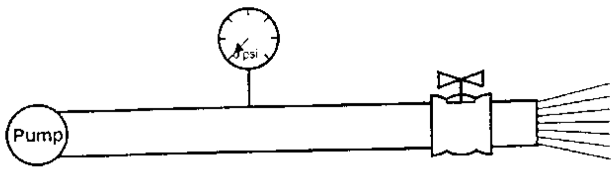
Positive Pumps have the following Advantages over Non-Positive Pumps:

a. High-Pressure Capability (up to 12,000 psi)
b. Small, Compact Size
c. High Volumetric Efficiency
d. Small Changes in Efficiency throughout the design pressure range
e. Great Flexibility of Performance

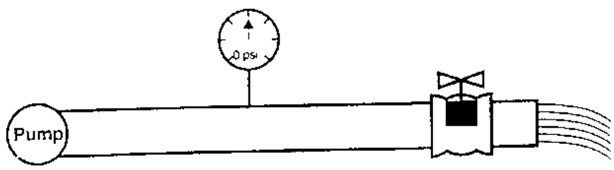
(can operate over a wide range of pressure requirements and speed ranges)

There are Three Main Types of Positive Displacement Pumps: Gear, Vane, and Piston. Many Variations exist in the Design of each of these Main Types of Pumps. For Example, Vane and Piston Pumps can be of either Fixed or Variable Displacement.

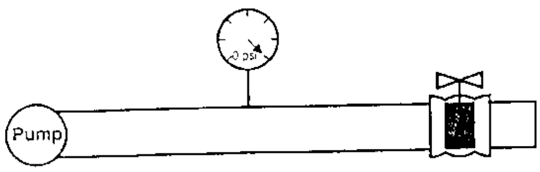
A Fixed Displacement Pump is one in which the amount of Fluid ejected per revolution (displacement) cannot be Varied. In a Variable Displacement Pump, the Displacement can be varied by changing the physical relationships of various pump elements. This Change in pump Displacement produces a Change in Pump Flow Output even though Pump Speed remains Constant. It should be understood that Pumps Do Not Pump Pressure, instead they produce Fluid Flow. The Resistance to this Flow, produced by the Hydraulic System, is What determines the Pressure.



a. Valve is wide open.



b. Valve is partially closed.



c. Valve is completely closed.

Pressure is the result of Resistance to Flow

For Example,

If a Positive Pump has its Discharge Line opens to the Atmosphere, there will be Flow, but there will be No discharge Pressure above Atmospheric because there is essentially no Resistance to Flow. If the Discharge Line is blocked, then we have Theoretically Infinite Resistance to Flow. Hence, there is No Place for the Fluid to go. The Pressure will therefore Rise until some Component Breaks unless Pressure Relief is provided. This is the Reason a Pressure Relief

Valve is needed when a Positive Displacement Pump is used. When the Pressure reaches a Set Value, the Relief Valve will open to allow Flow Back to the oil Tank. A Pressure Relief Valve determines the Maximum Pressure Level that the System will Experience regardless of the magnitude of the load Resistance.

Some Pumps are made with: Variable Displacement and Pressure Compensation Capability. Such Pumps are designed so that as System Pressure Builds up they produce Less Flow. Finally at some Predetermined Maximum Pressure Level, the Flow Output goes to Zero due to Zero Displacement. This prevents any additional Pressure Buildup. Pressure Relief Valves are not needed when Pressure-Compensated Pumps are used. The Hydraulic Power developed by Pumps is converted back into Mechanical Power by Hydraulic Cylinders and Motors, which produce the useful Work Output. A variable Displacement, Pressure-Compensated, Axial-Piston Pump is used to provide Optimum Performance in both Backhoe and Loader Operations. The Backhoe portion of the machine performs operations such as digging a Trench. The Front Loader portion can then be used to Load a Dump Truck with the earth removed from the Trench Dug by the Backhoe. The Pump delivers the Desired Flow to the Hydraulic Cylinders at the Required Pressure to Fulfill implement Demands. At an Operating Speed of 2200 rpm, the Pump produces a Maximum Flow of 43 gpm (163 Lpm) at a System Pressure of 3300 psi (22,700 kPa).

Pump Output Flow, Neglecting changes in the Small Internal Leakage, is Constant and Not Dependent on System Pressure. This makes them particularly well suited for Fluid Power Systems. Positive Displacement Pumps must be protected against Overpressure If the Resistance to flow becomes Very Large. This can happen If a Valve is completely closed and there is No physical Place for the Fluid to Go. A Pressure Relief Valve is used to protect the pump against Overpressure by Diverting Pump Flow Back to the Hydraulic Tank, where the Fluid is stored for system use.
Positive Displacement Pumps can be classified by the Type of Motion of Internal Elements.

The Motion may be either Rotary or Reciprocating. There are essentially Three Basic Types:

1. Gear Pumps (Fixed Displacement Only by Geometrical Necessity)
   a. External Gear Pumps
   b. Internal Gear Pumps
   c. Lobe Pumps
   d. Gerotor Pump
   e. Screw Pumps

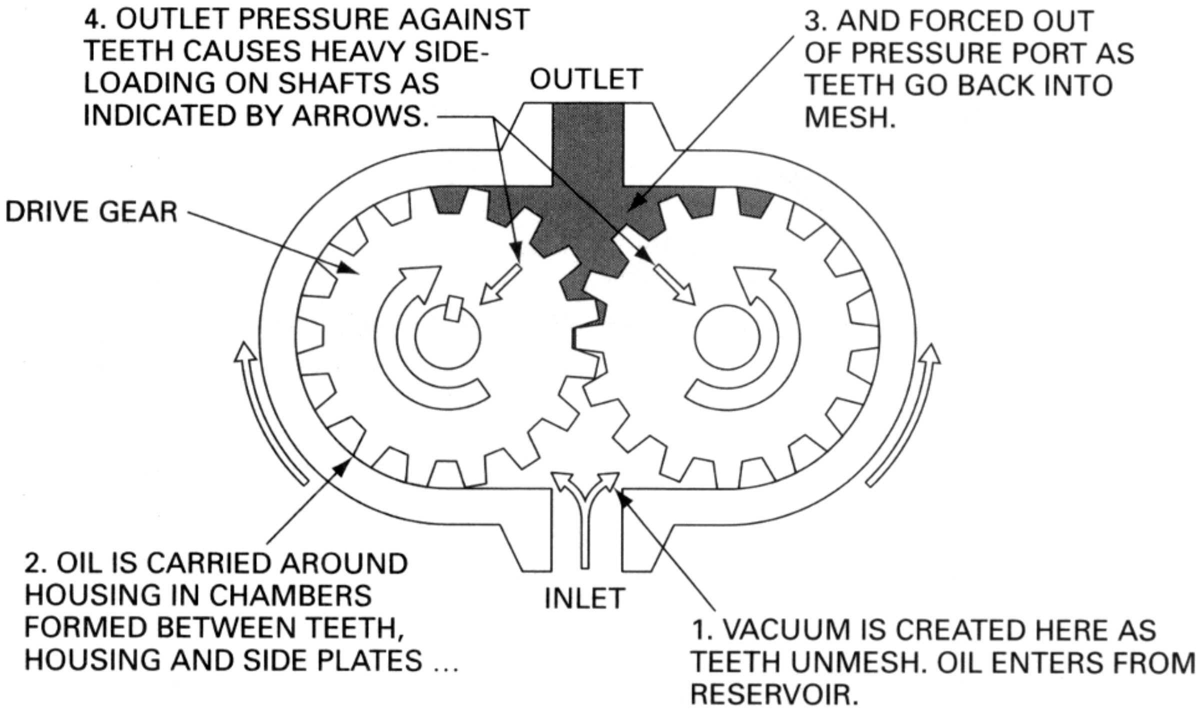
2. Vane Pumps
   a. Unbalanced Vane Pumps (Fixed or Variable Displacement)
   b. Balanced Vane Pumps (Fixed Displacement Only)

3. Piston Pumps (Fixed or Variable Displacement)
   a. Axial Design (Bent Axis or Swash Plate).
   b. Radial Design

The Unbalanced Vane Pump have Pressure Compensation Capability, which Automatically Protects the Pump against Overpressure.

Gear Pumps

External Gear Pump



External Gear Pump develops Flow by Carrying Fluid between the Teeth of Two Meshing Gears. 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.

Volumetric Displacement and Theoretical Flow Rate

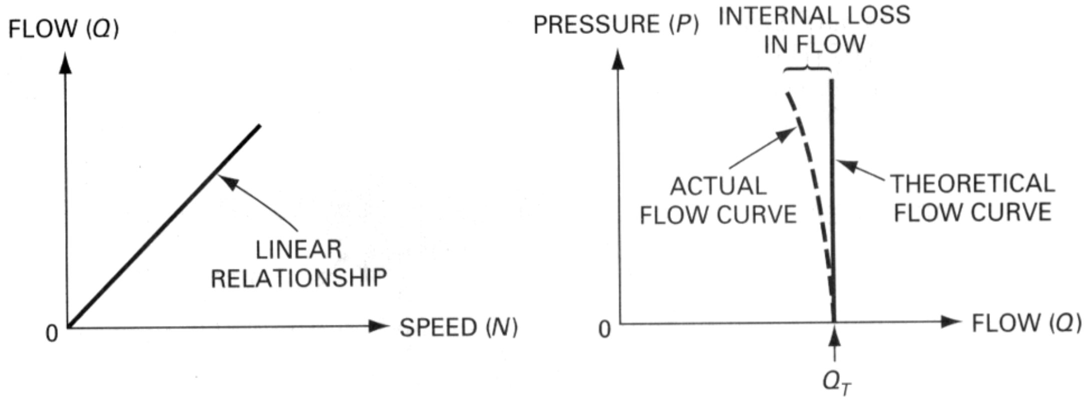
D_0 = Outside Diameter of Gear Teeth (in, m)
D_i = Inside Diameter of Gear Teeth (in, m)
L = Width of Gear Teeth (in, m)
V_D = Displacement Volume of Pump (in3/rev, m3/rev)
N = rpm of Pump
Q_T = Theoretical Pump Flow-Rate

$$V_D = \frac{\pi}{4}(D_o^2 - D_i^2)L$$

$$Q_T\,(\text{in}^3/\text{min}) = V_D\,(\text{in}^3/\text{rev}) \times N\,(\text{rev/min})$$

$$Q_T(\text{gpm}) = \frac{V_D(\text{in}^3/\text{rev}) \times N\,(\text{rev/min})}{231}$$

$$Q_T\,(\text{m}^3/\text{min}) = V_D\,(\text{m}^3/\text{rev}) \times N\,(\text{rev/min})$$



(a) FLOW VERSUS SPEED CURVE

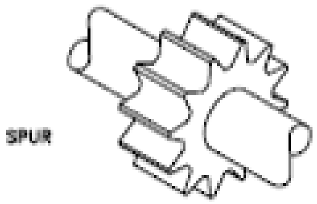
(b) FLOW VERSUS PRESSURE CURVE AT CONSTANT PUMP SPEED

<u>Volumetric Efficiency</u>

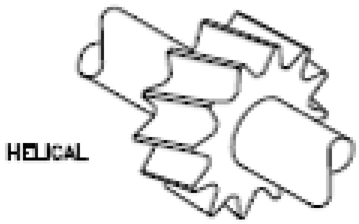
There must be a Small Clearance (about 0.001 in) between the Teeth Tip and Pump Housing. As a Result, Some of the Oil at the Discharge Port can Leak directly Back toward the Suction Port. This Means that the Actual Flow-rate Q_A is less than the Theoretical Flow rate Q_T, which is based on Volumetric Displacement and Pump Speed. This Internal Leakage, called Pump *Slippage*, is identified by The Term Volumetric Efficiency η_v, which equals about 90% for Positive Displacement Pumps, Operating at Design Pressure:

$$\eta_v = \frac{Q_A}{Q_T} \tag{3}$$

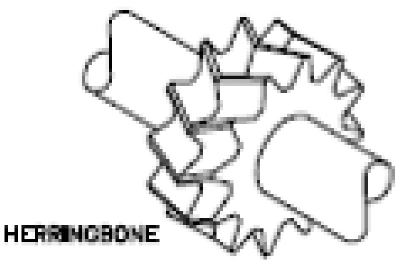
The Higher the Discharge Pressure, the Lower the Volumetric Efficiency because Internal Leakage Increases with Pressure. Pump Manufacturers usually specify Volumetric Efficiency at the Pump Rated Pressure. The *Rated Pressure* of a Positive Displacement Pump is that Pressure Below which No Mechanical Damage due to Overpressure will occur to the Pump and the Result will be a Long Reliable Service Life. Too High a Pressure not only produces Excessive Leakage but also can Damage a Pump by Distorting the Casing and Overloading the Shaft Bearings. This brings to mind once again the Need for Overpressure Protection. High Pressures occur when a High Resistance to Flow is encountered, such as a Large Actuator Load or a Closed Valve in the Pump Outlet Line.



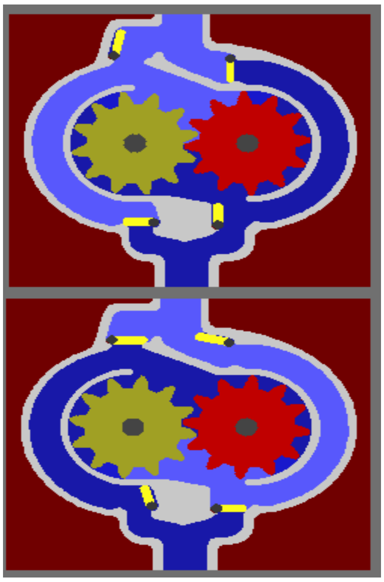
External Gear Pump uses Spur Gears, Teeth are Parallel to the Axis of the Gear which are Noisy at Relatively High Speeds.



To Reduce Noise and provide Smoother Operation, Helical Gears (Teeth Inclined at a small angle to the Axis of the Gear) are sometimes used. Helical Gear Pumps are limited to Low-Pressure Applications (< 200 psi) because they develop Excessive End Thrust due to the action of the Helical Gears.



Herringbone Gear Pumps eliminate this Thrust Action and thus can be used to develop Much Higher Pressures (up to 3000 psi). Herringbone Gears consist of Two Rows of Helical teeth cut into one Gear. One of the Rows of each Gear is Right-Handed and the other is Left-Handed to cancel out the Axial Thrust Force. Herringbone Gear Pumps operate as smoothly as Helical Gear Pumps and provide Greater Flow Rates with Much Less Pulsating Action.

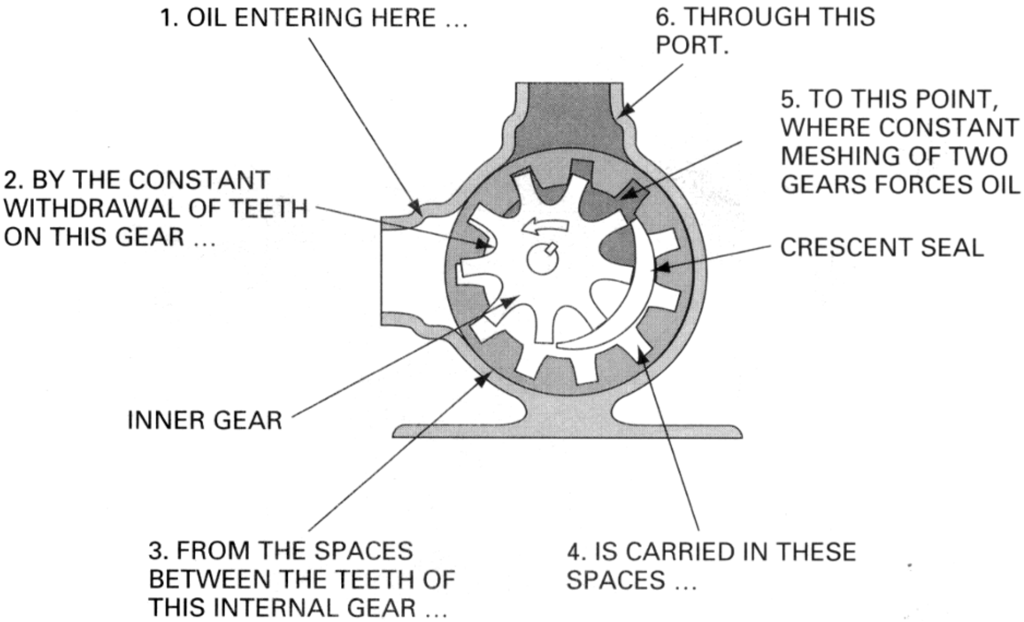


Reversing gear pump

This type of rotary gear pump moves liquid in the same direction regardless of the direction the gears turn. The valves (yellow) are forced open and closed by the difference in pressure on the input and output sides. Fluid in the higher-pressure output side is shown in light blue while fluid in the lower-pressure input side is dark blue. Notice that when the green gear starts moving clockwise, the fluid in the outer circular tubes changes pressure but stops flowing, and fluid flows more or less straight up. When the red gear is moving clockwise, fluid travels in a sideways 'S' shape through the outer circular tubes.

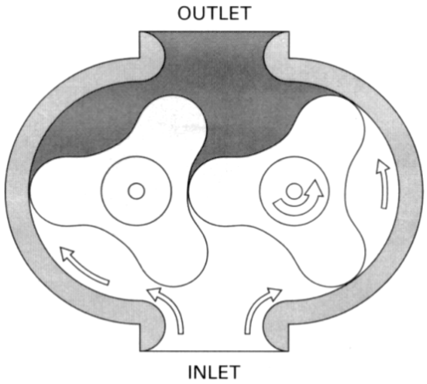
Internal Gear Pump

The Internal Gear Pump 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.

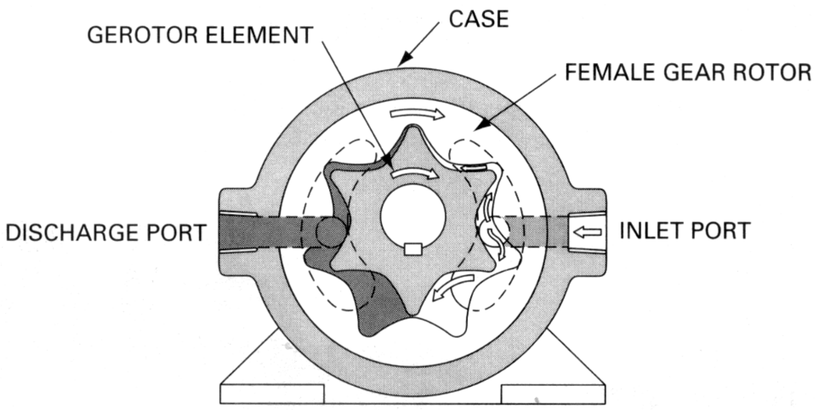


Lobe Pump

Also in the General Family of Gear Pumps is the Lobe Pump, which 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 Greater Amount of Pulsation, although its Volumetric Displacement is generally Greater than that for Other Types of Gear Pumps.

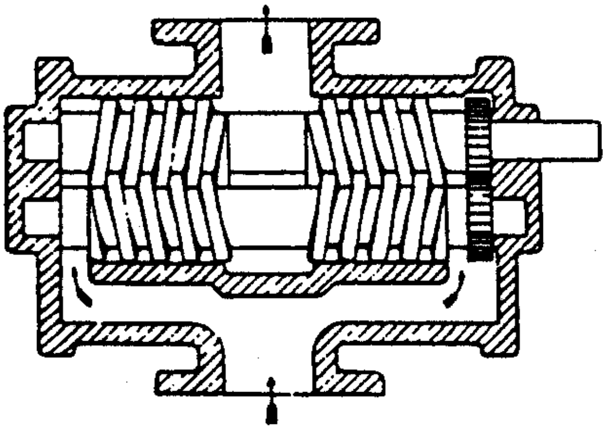


Gerotor Pump



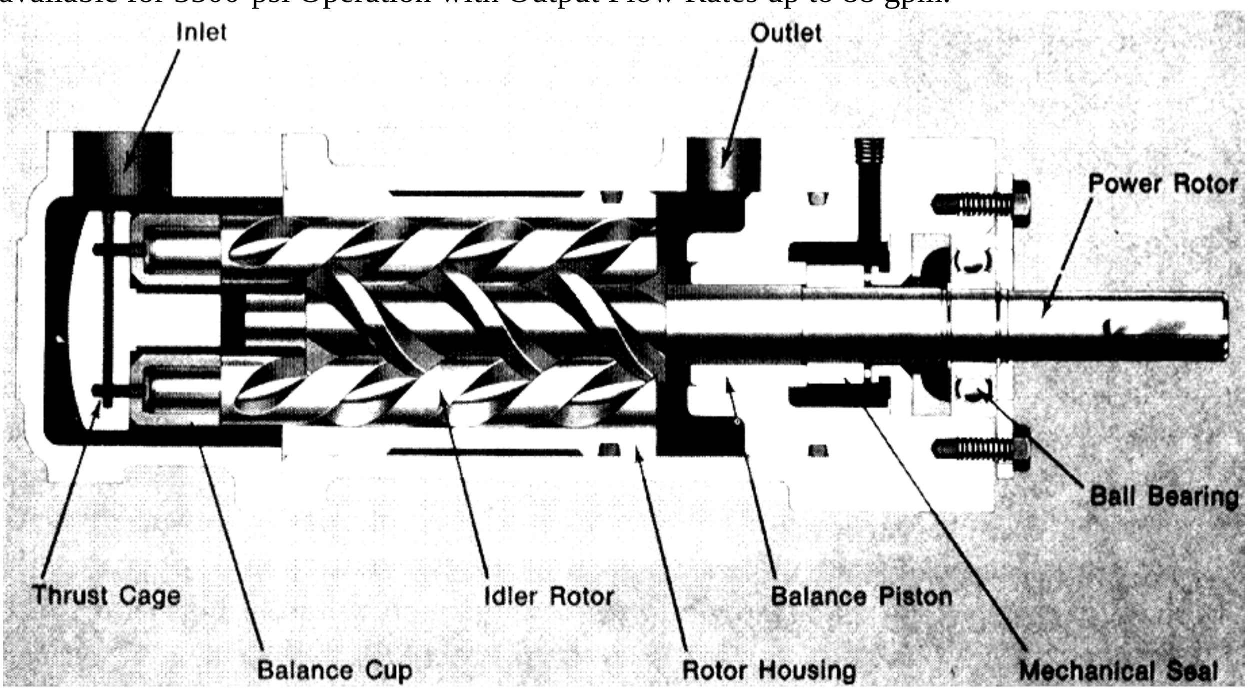
The Gerotor Pump operates Very Much Like the Internal Gear Pump. The Inner Gear Rotor (Gerotor Element) is Power-Driven and draws the Outer Gear Rotor around as They Mesh Together. This forms Inlet and Discharge Pumping Chambers between the Rotor Lobes. The Tips of the Inner and Outer Rotors make Contact 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. This is a Simple Type of Pump since there are Only Two Moving parts.

Screw Pump



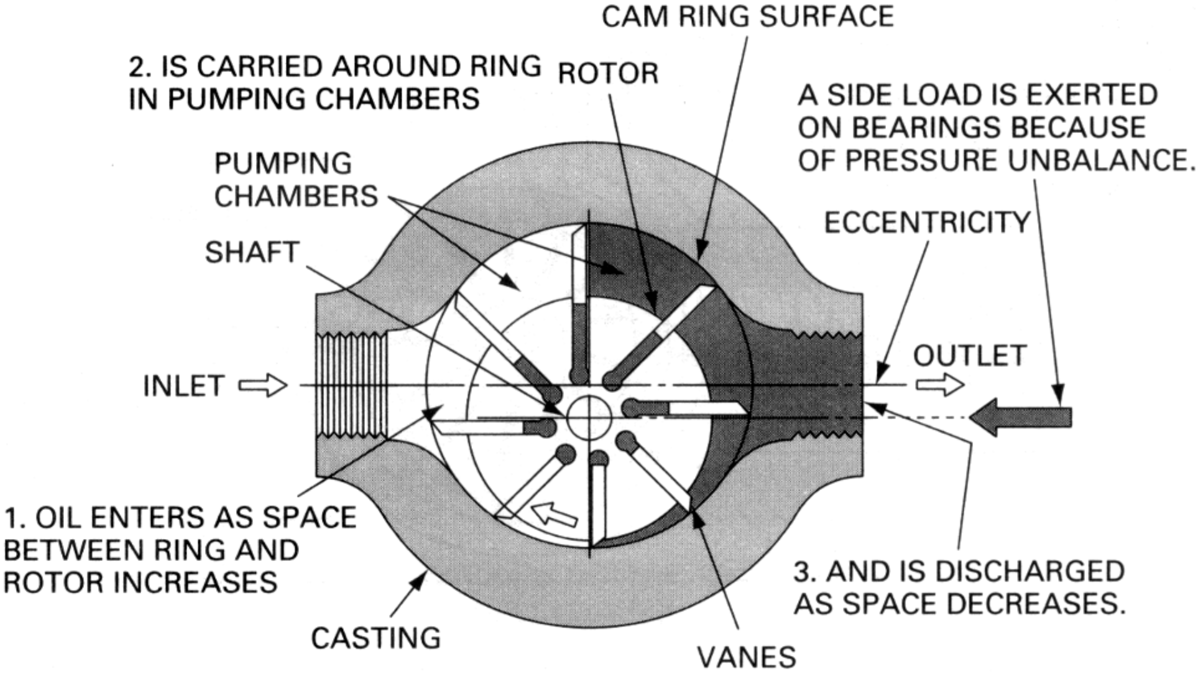
Two–Screw Pump

The Screw Pump is an Axial Flow Positive Displacement Unit. Three Precision Ground Screws, Meshing within a Close-Fitting Housing, Deliver Non-pulsating 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

Careful Observation will reveal that there is an Eccentricity between the Centerline of the Rotor and the Centerline of the Cam Ring. If the Eccentricity is Zero, there will be No Flow.
The Following Analysis and Nomenclature is Applicable to the Vane Pump:
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)

From Geometry, we can find the Maximum possible Eccentricity:

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

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

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

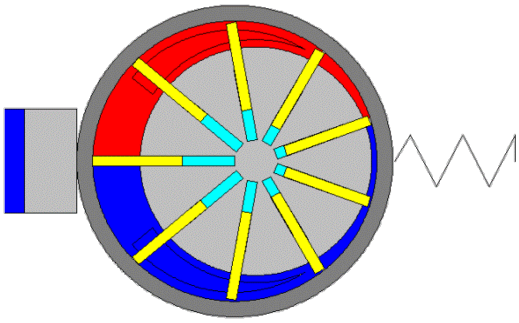
Substituting the expression for e_{max} yields

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

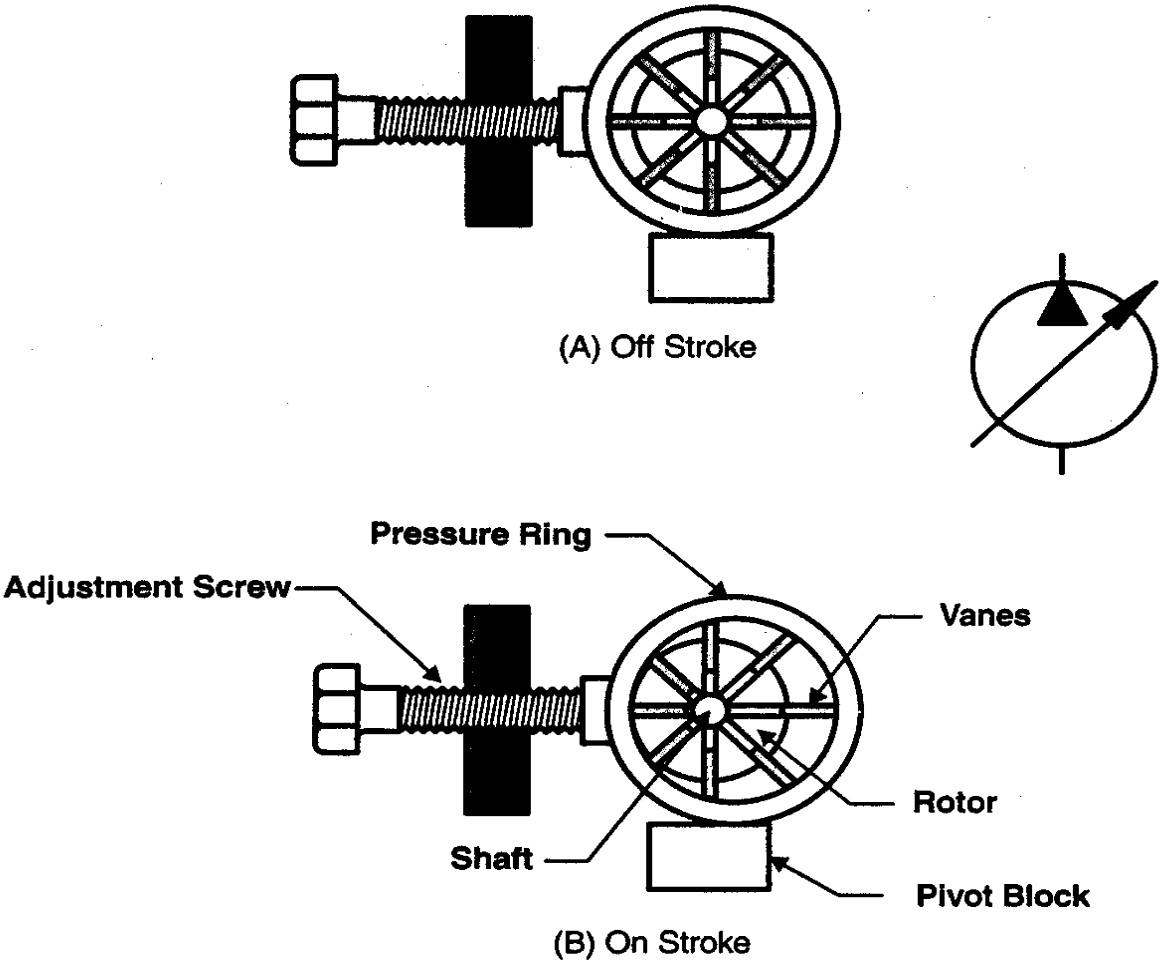
The Actual Volumetric Displacement occurs when e_{max} = e:

$$V_D = \frac{\pi}{2}(D_C + D_R)eL$$

Some Vane Pumps have provisions for Mechanically Varying the Eccentricity. Such a Design is called a *Variable Displacement Pump*. A Hand wheel 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.

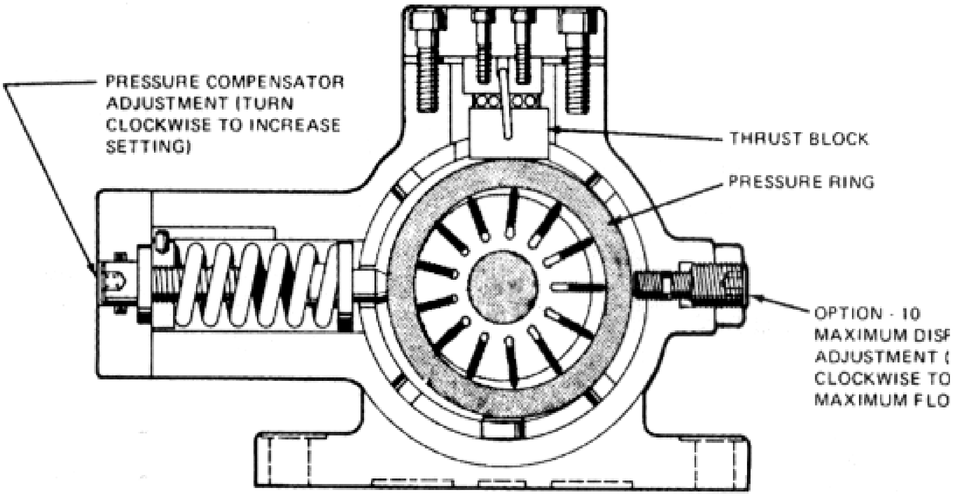


Variable Displacement Vane Pump



The Flow of a Variable Displacement Pump can be varied by Using the Adjustment Screw

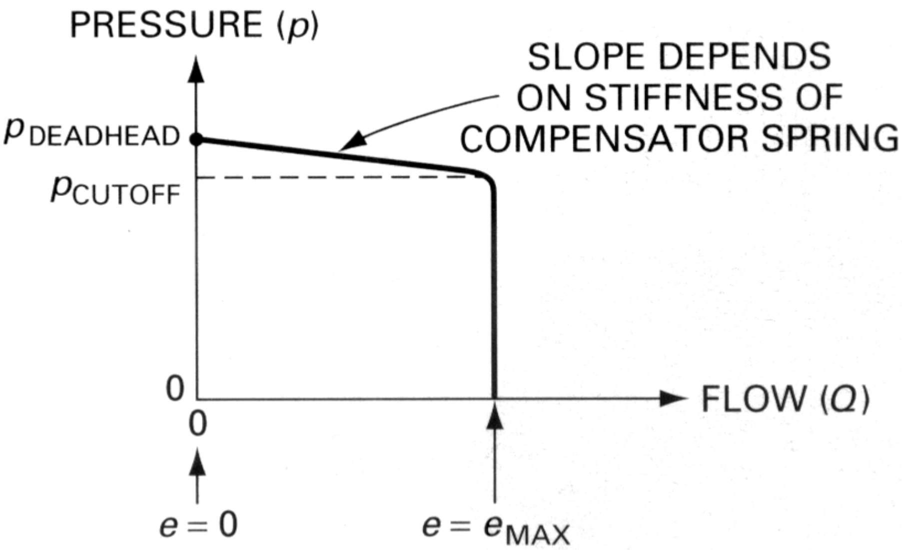
Pressure-Compensated Vane Pump



Variable Displacement Pressure-Compensated Vane Pump

The Design we see is a Pressure-Compensated one in which System Pressure acts directly on the Cam Ring via A Hydraulic Piston on the Right Side. 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.

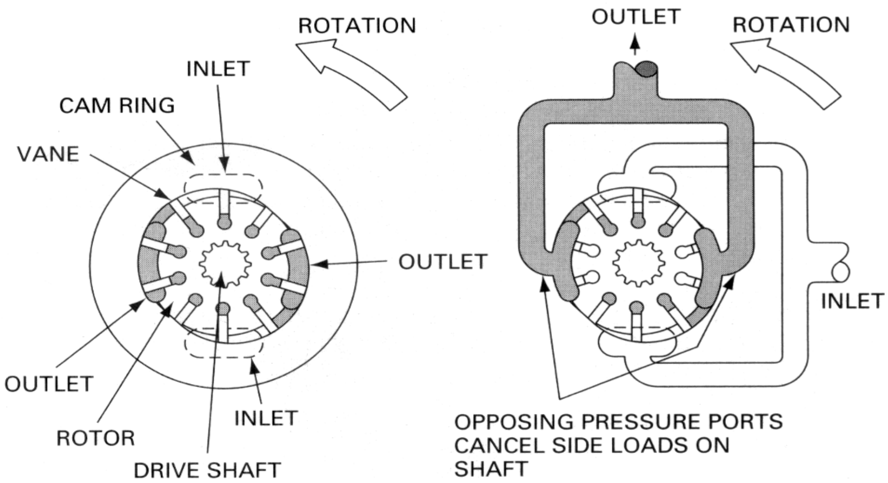


Pressure versus Flow for Pressure-Compensated Vane Pump

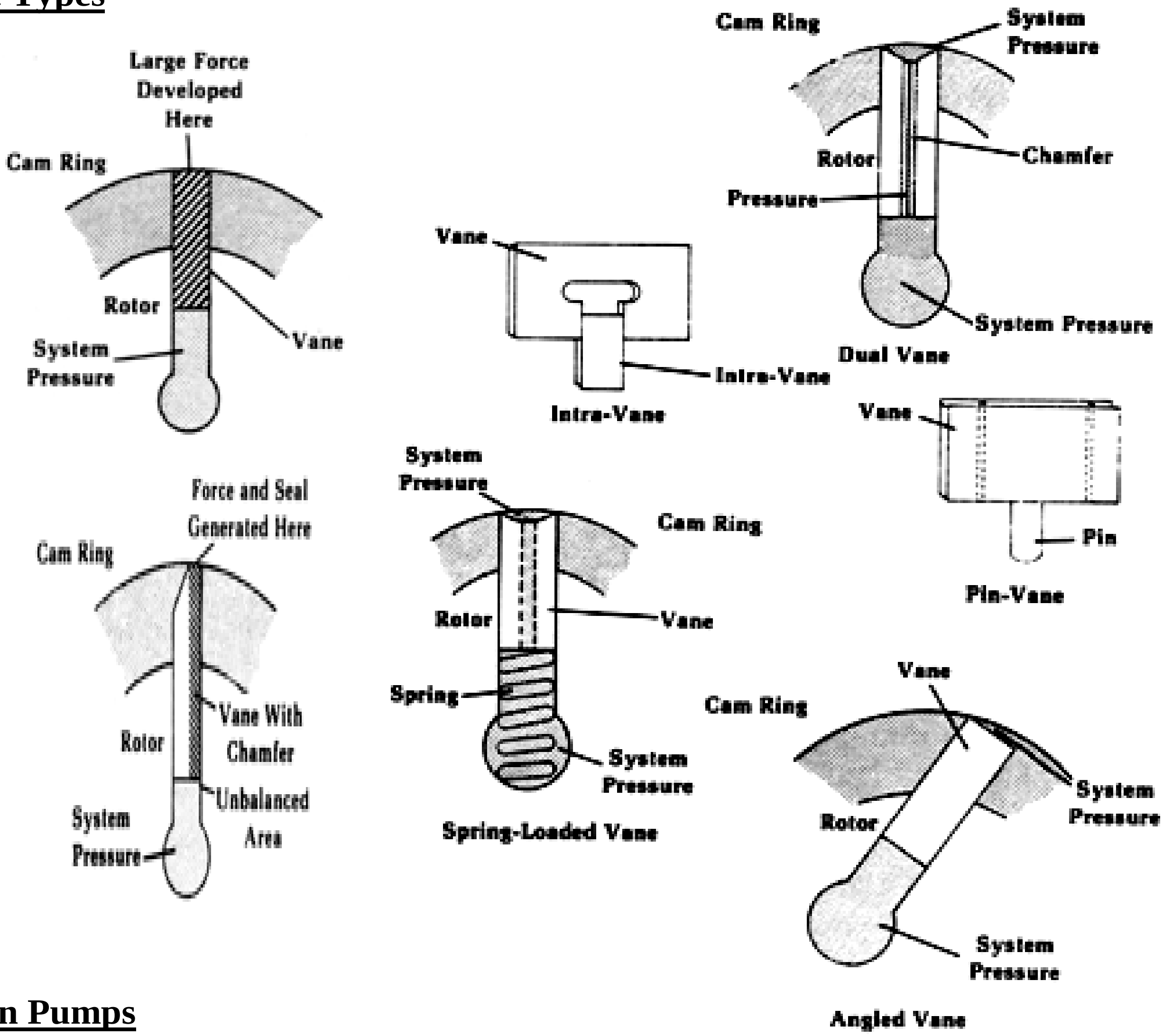
The Internal Configuration of an Actual Pressure-Compensated Vane Pump design contains a Cam Ring that rotates slightly during Use, thereby distributing wear over the Entire Inner circumference of the Ring. Note that a Side Load is exerted on the bearings of the Vane Pump because of Pressure unbalance. This Same Undesirable Side Load exists for the Gear Pump. Such Pumps are Hydraulically Unbalanced.

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.



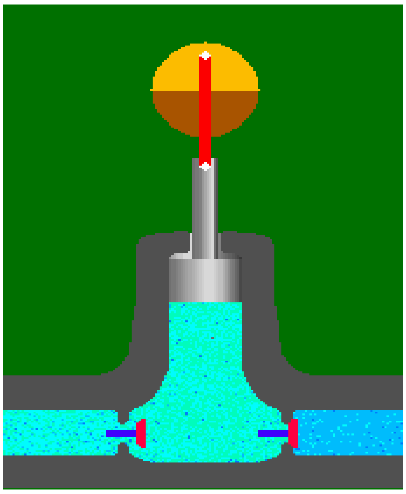
Vane Types

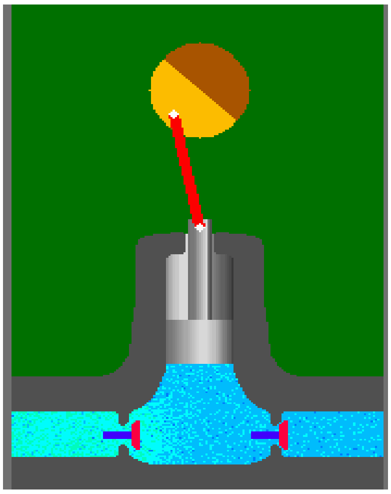


Piston Pumps

Introduction

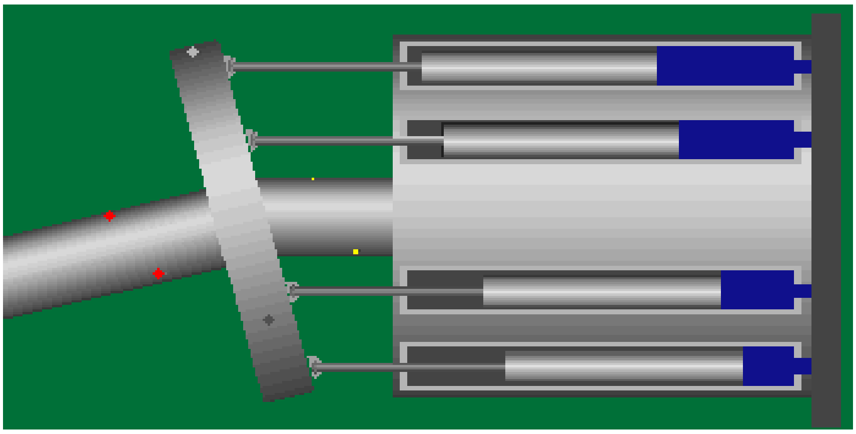
A Piston Pump works on the Principle that a Reciprocating Piston can draw in Fluid when it retracts in a Cylinder Bore and Discharge it when it extends. The basic question is How to Mechanize a Series of Reciprocating Pistons. There are Two Basic Types of Piston Pumps. One is the Axial Design, having 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.



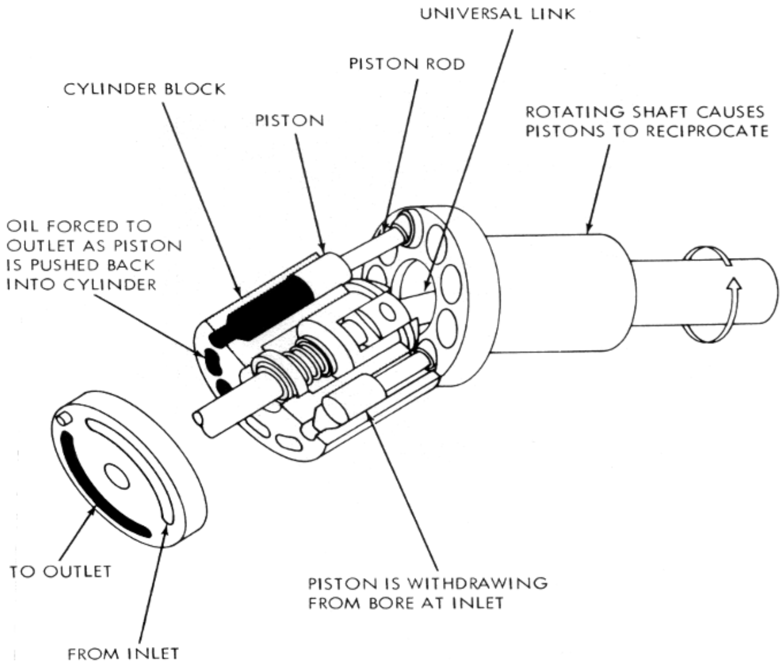


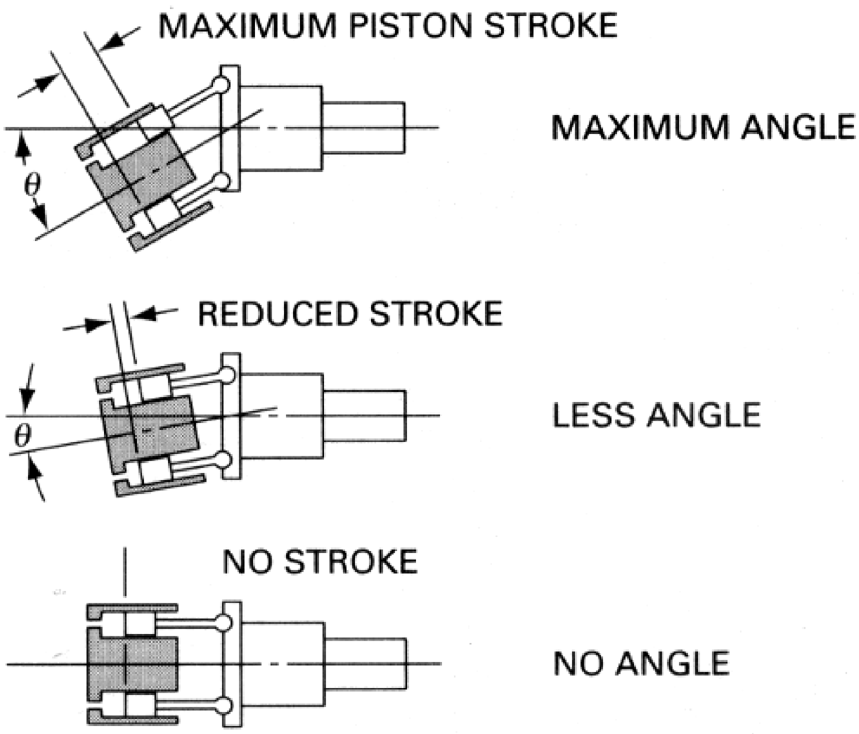
Axial Piston Pump Bent-Axis Design

It contains a Cylinder Block Rotating with the Drive Shaft. 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.
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. Some Designs have Controls that Move the Yoke over the Center Position to Reverse the Direction of Flow through the Pump.



Bent-Axis Design Axial Piston Pump





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)

QT = Theoretical Flow-Rate (gpm, m^3/min)

From Trigonometry we have

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

The Total Displacement Volume equals the Number of Pistons Multiplied by the Displacement Volume per Piston:

$$V_D = YAS$$

Substituting we have $V_D = YAD \tan(\theta)$

Then we obtain a Relationship for the Theoretical Flow-Rate using English Units.

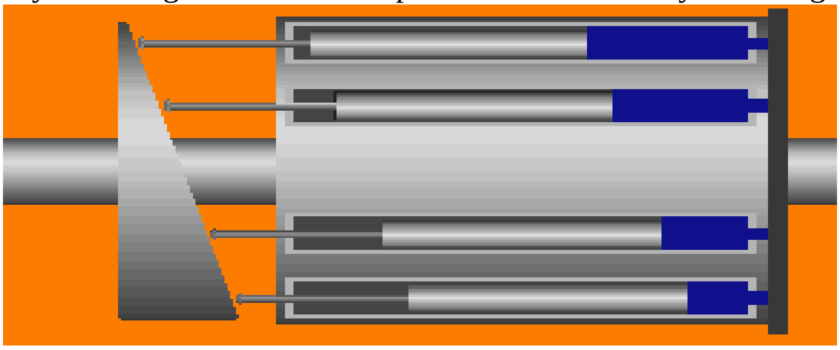
$$Q_T(\text{gpm}) = \frac{DANY \tan(\theta)}{231}$$

Relationship for the Theoretical Flow-Rate in Metric units.

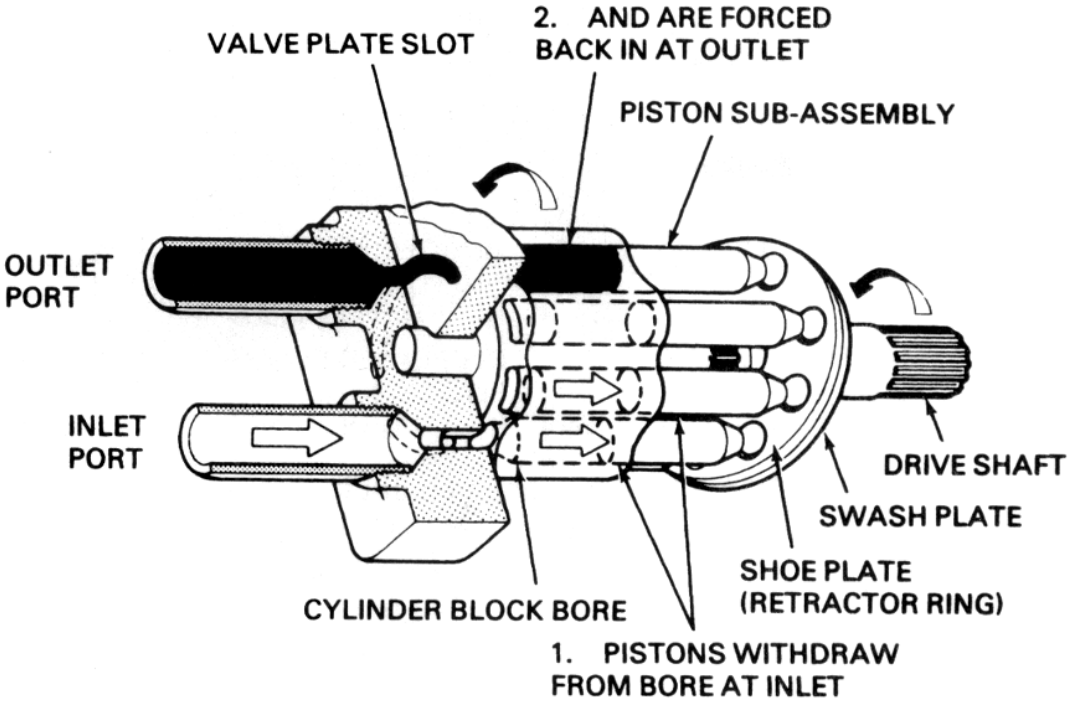
$$Q_T(\text{m}^3/\text{min}) = DANY \tan(\theta)$$

Swash Plate Design Axial Piston Pump

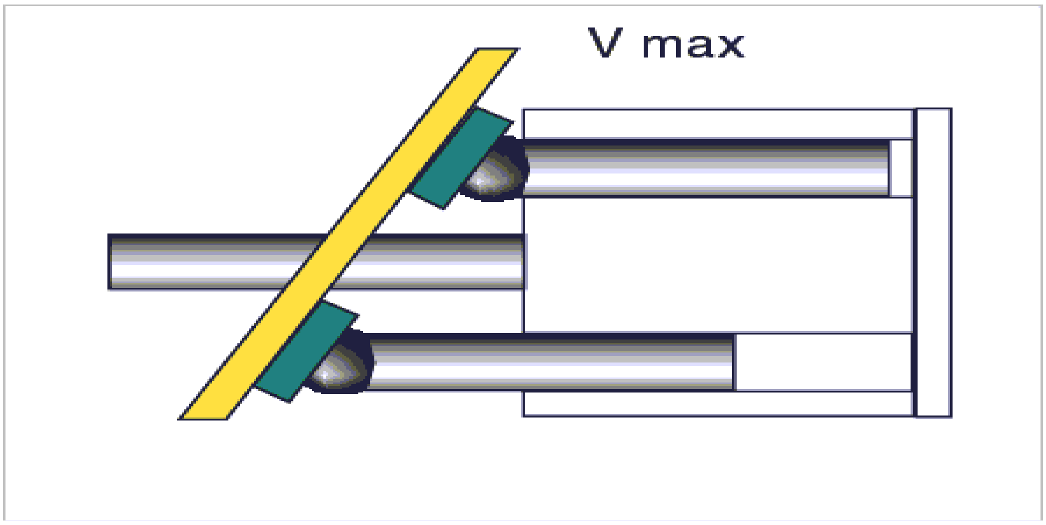
In This Type, the Cylinder Block and Drive Shaft are located on the Same Centerline. The Pistons are connected to a Shoe Plate which bears against an Angled Swash Plate. As the Cylinder Rotates, the Pistons reciprocate because the Piston Shoes follow the Angled Surface of the Swash Plate. The Outlet 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.



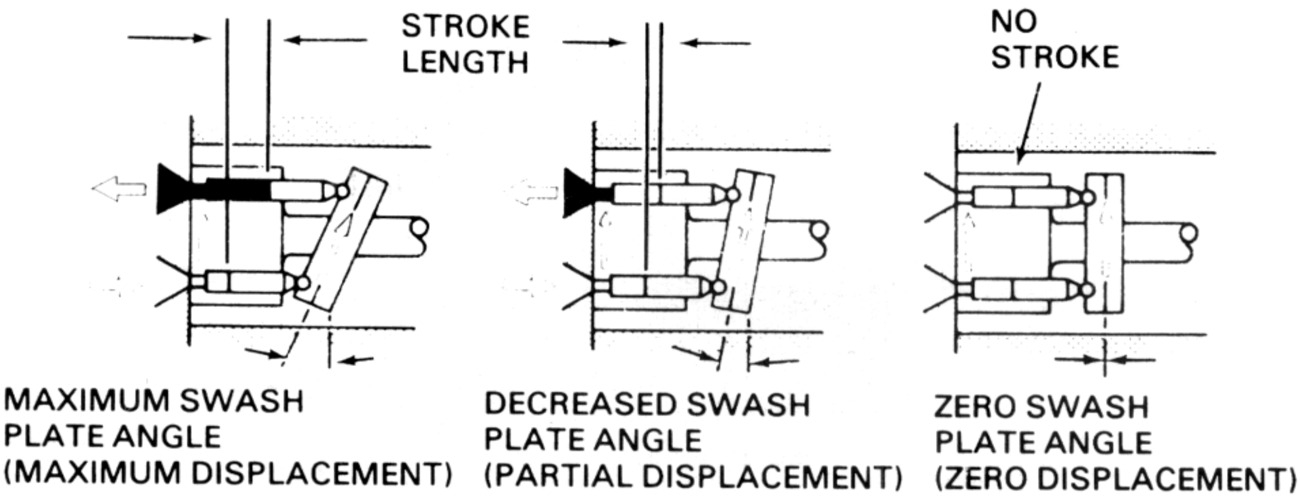
Swash Plate Design Axial Piston Pump



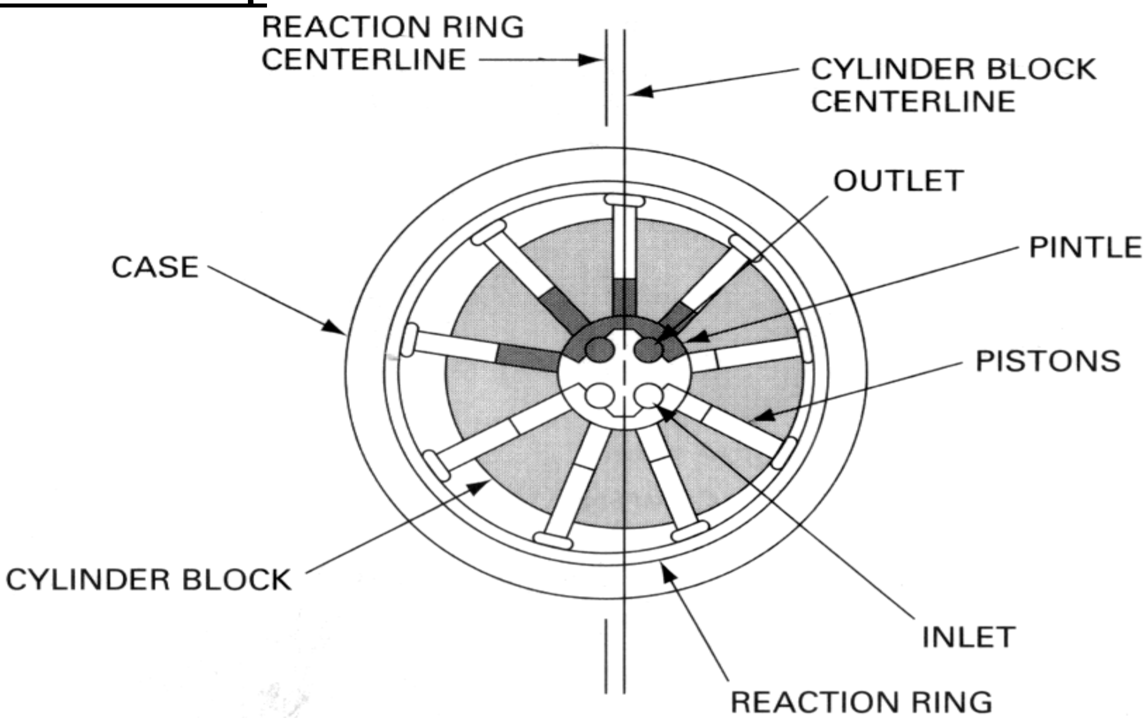
The Swash Plate Design Inline Piston Pump can also be designed to have Variable displacement capability. In such a Design, the Swash Plate is mounted in a Movable Yoke. The Swash Plate Angle can be changed by pivoting the Yoke on Pintles. Positioning of the Yoke can be accomplished by: Manual Operation, Servo Control, or a Compensator Control. The Maximum Swash Plate Angle is Limited to 17½° by Construction.



Variable Displacement Version of In-Line Piston Pump



Radial Piston Pump



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. An Actual Radial Piston Pump has Variable Displacement, Pressure-Compensated Discharge. This Pump is Available in 3 Sizes (2.40, 3.00, and 4.00 in^3 Volumetric Displacements) and Weighs approximately 60 Ib. Variable Displacement is accomplished by Hydraulic rather than Mechanical Means and is responsive to Discharge Line Pressure.

Pump Performance

Introduction

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 Below.

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:

$$\eta_v = \frac{\text{actual flow-rate produced by pump}}{\text{theoretical flow-rate pump should produce}} = \frac{Q_A}{Q_T}$$

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 % Values. For Example, An Efficiency Value of 90% would be represented by A Value of 0.90.

2. Mechanical Efficiency (η_m): Mechanical Efficiency indicates the Amount of Energy Losses that occurs 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

$$\eta_m = \frac{pQ_T/1714}{T_A N/63{,}000}$$

In Metric Units, using Watts for Power,

$$\eta_m = \frac{pQ_T}{T_A N}$$

P = Pump Discharge Pressure (psi, Pa)
QT = Pump Theoretical Flow-rate (gpm, rrr^3/s)
TA = Actual Torque delivered to pump (in • Ib, N • m)
N = Pump Speed (rpm, rad/s)
Mechanical Efficiency can also be Computed in terms of Torques:

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

Note that the Theoretical Torque required to Operate a Pump (TT) is the Torque that would be Required If there were No Leakage. Equations for evaluating the Theoretical Torque and the Actual Torque are

Theoretical Torque

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

Or

$$T_T(\text{N} \cdot \text{m}) = \frac{V_D(\text{m}^3) \times p(\text{Pa})}{2\pi}$$

Actual Torque

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

or

$$T_A = \frac{\text{actual power delivered to pump (W)}}{N\ (\text{rad/s})}$$

where

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

3. Overall Efficiency (η_0)

The Overall Efficiency considers All Energy Losses and hence is defined as follows:

$$\text{overall efficiency} = \frac{\text{actual power delivered by pump}}{\text{actual power delivered to pump}}$$

The Overall Efficiency can also be Represented Mathematically as follows:

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

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

Canceling Like Terms yields the desired result showing the Equivalency of Eq.

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

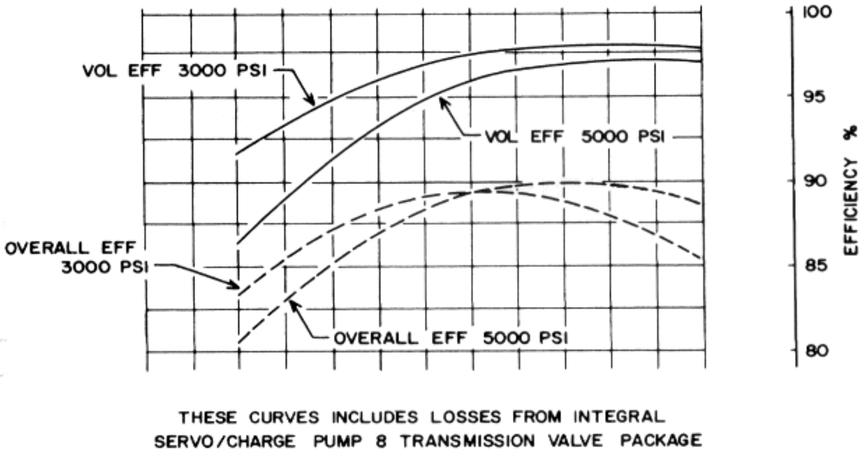
Repeating this substitution for Metric Units yields:

$$\eta_o = \frac{pQ_A}{T_A N} = \frac{\text{actual power delivered by pump}}{\text{actual power delivered to pump}}$$

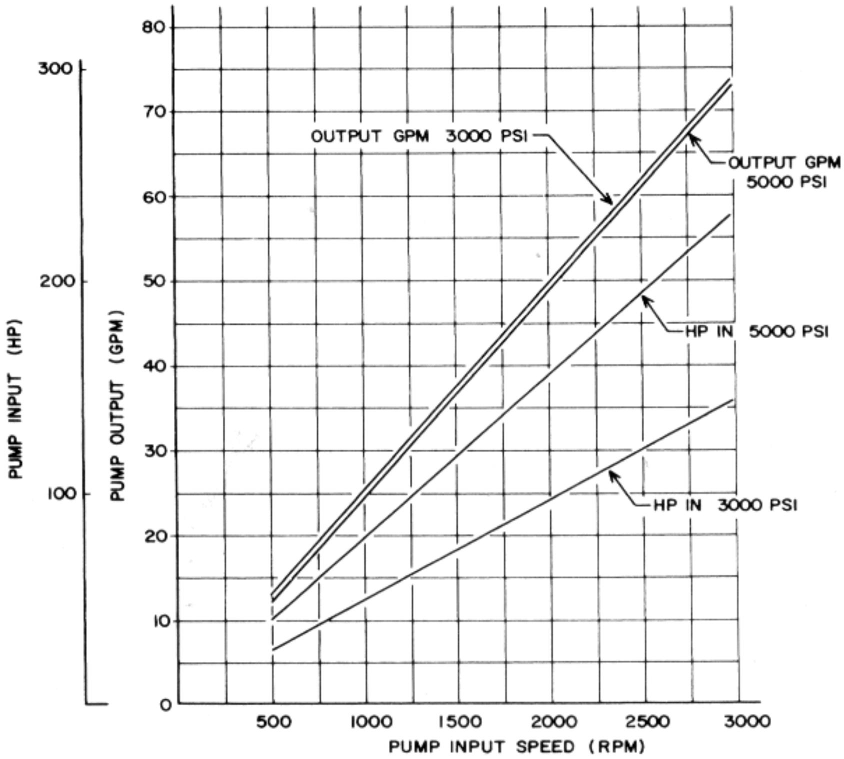
Note that: The Actual Power delivered to a pump from a prime mover via a Rotating Shaft is called Brake Power and the Actual Power delivered by a Pump to the Fluid is called hydraulic Power.

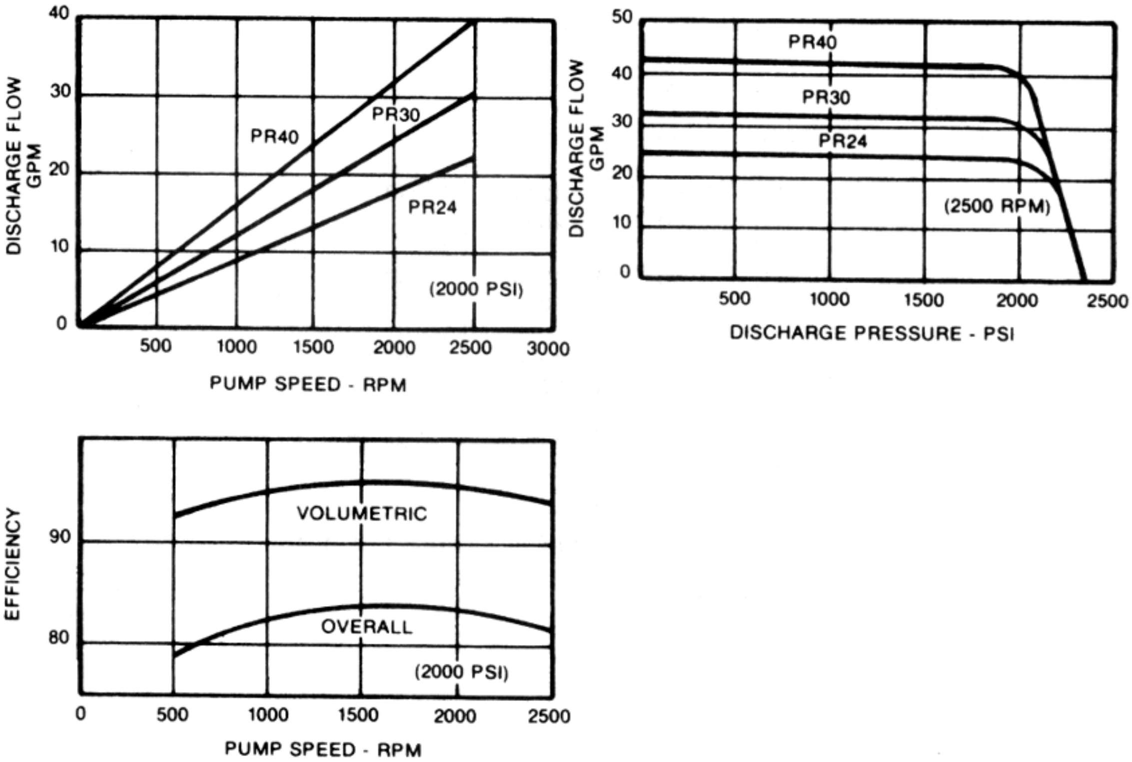
Pump Performance Curves

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.



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 <u>the Radial Piston Pump</u>

This pump comes in 3 Different Sizes:
PR24: 2.40-in^3 displacement
PR30: 3.00-in^3 displacement
PR40: 4.00-in^3 displacement

Thus, there are Three Curves on two of the Graphs. Observe the Linear Relationship between Discharge Flow (gpm) and Pump Speed (rpm). <u>Also, Note that</u> 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.

<u>**Pump Performance Comparison Factors**</u>

PUMP TYPE	PRESSURE RATING (PSI)	SPEED RATING (RPM)	OVERALL EFFICIENCY (PER CENT)	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 compares various performance factors for hydraulic pumps.

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: <u>Leakage around the Outer Periphery of the Gears</u>. <u>Leakage across the Faces of the Gears</u>. <u>Leakage at the Points where the Gear Teeth make Contact</u>. Gear Pumps are Simple in Design and Compact in Size. Gear Pumps are the Most Common Type of Pump Used in Fluid Power Systems. The Greatest Number of Applications of Gear Pumps is in the Mobile Equipment and Machine Tool Fields. 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 Power-to-Weight Ratio. They produce essentially a Non-pulsating 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.

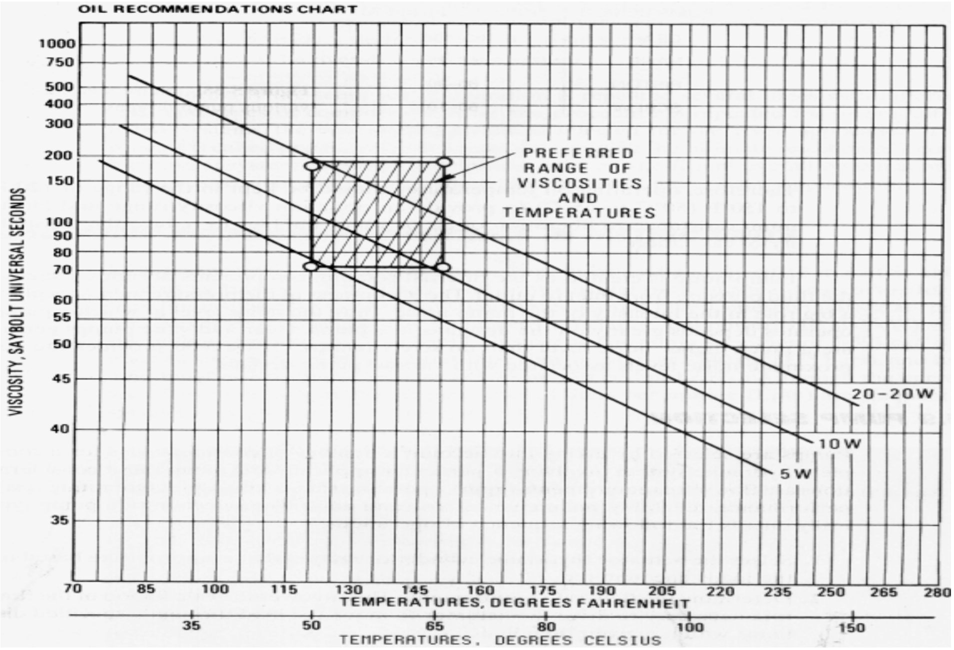
Pump Cavitation

Another Noise Problem, called pump cavitation, can occur due to Entrained Air Bubbles in the Hydraulic Fluid or Vaporization of the Hydraulic Fluid. This Occurs When Pump Suction Lift is excessive and the Pump Inlet Pressure Falls below the Vapor Pressure of the Fluid (usually about 5-psi suction). As a Result, Air or Vapor Bubbles, which Form in the Low-Pressure Inlet Region of the Pump, are Collapsed When they Reach the High-Pressure Discharge Region. This produces High Fluid Velocity and Impact Forces, which can erode the Metallic Components and Shorten Pump Life.

The Following Rules will Control or Eliminate Cavitation of a Pump by keeping the Suction Pressure above the Saturation Pressure of the Fluid:

1. Keep Suction Line Velocities Below 4 ft/s (1.2 m/s).
2. Keep Pump Inlet Lines as Short as Possible.
3. Minimize the Number of Fittings in the Inlet Line.
4. Mount the Pump as Close as possible to the Reservoir.
5. Use Low-Pressure Drop Inlet Filters or Strainers. Use Indicating-Type Filters and Strainers so that they can be Replaced at Proper Intervals as they Become Dirty.
6. Use the Proper Oil as Recommended by the Pump Manufacturer.

The next figure shows the Preferred Range of Viscosities and Temperatures for Optimum Pump Operation. The Importance of Temperature Control lies in the fact that Increased Temperatures tend to accelerate the Liberation of Air or Vapor Bubbles. Therefore, Operating Oil Temperatures should be kept in the Range of 120^{o}F to 150°F (50°C to 65oC) to provide an Optimum Viscosity Range and Maximum Resistance to Liberation of Air or Vapor Bubbles to Reduce the Possibility of Cavitation. Pump Noise is created as the Internal Rotating Components abruptly Increase the Fluid Pressure from Inlet to Outlet. The Abruptness of the Pressure Increases plays a Big Role in the Intensity of the Pump Noise. The Noise Level at which a Pump Operates Depends Greatly on the Design of the Pump



Preferred Range of Oil Viscosities and Temperatures.

Gear and Vane Pumps generate a Much Higher Noise Level than do Screw Pumps. Next table provides the Approximate Noise Levels associated with Various Pump Designs

PUMP DESIGN	NOISE LEVEL (dB-A)
GEAR	80–100
VANE	65–85
PISTON	60–80
SCREW	50–70

Noise Levels for Various Pump Designs

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

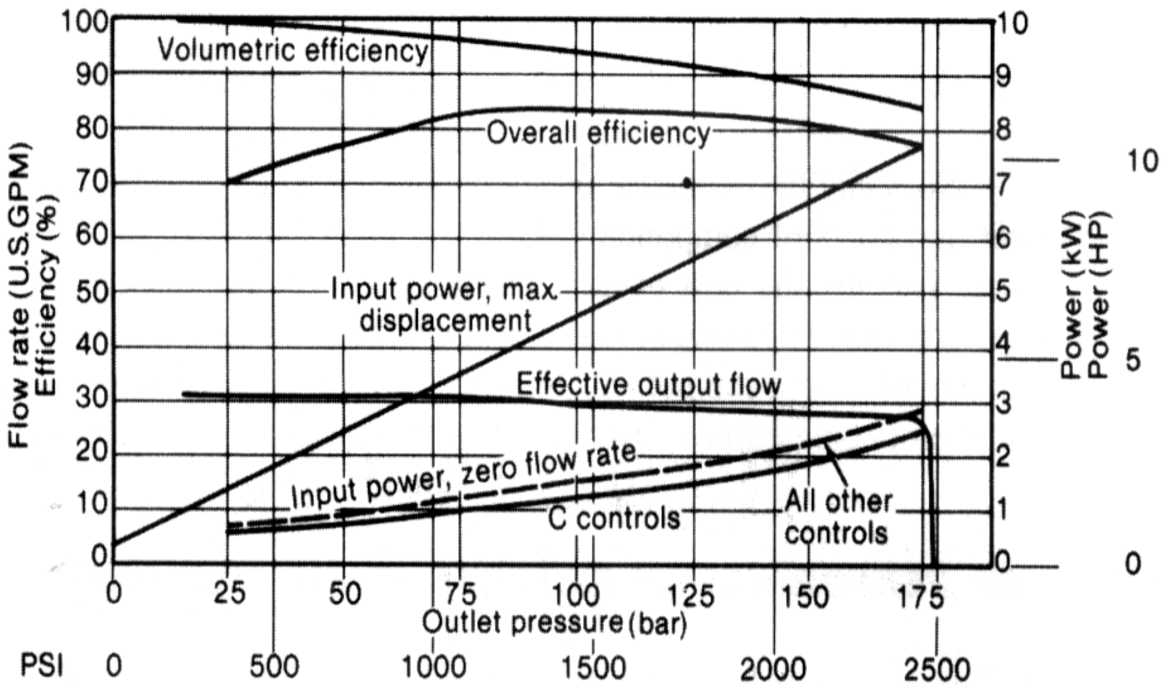
4- Determine the Pump Speed and Select the Prime Mover. This, together with the Flow-

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).
6- Select the Reservoir and Associated Plumbing, including piping, valving, Fillers and Strainers, 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 Rating in Metric Units

Performance Data for Hydraulic Pumps are Measured and Specified in Metric Units as well as English Units. Next figure 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 Pump (Figure) 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 in3 (20 cm3 or 0.02L). Although the Curves give Flow Rates in gpm, Metric Flow rates of Liters per minute (Lpm) are Frequently Specified.



English/Metric Performance Curves for Variable Displacement Pressure-Compensated Vane Pump at 1200 rpm