Pumps

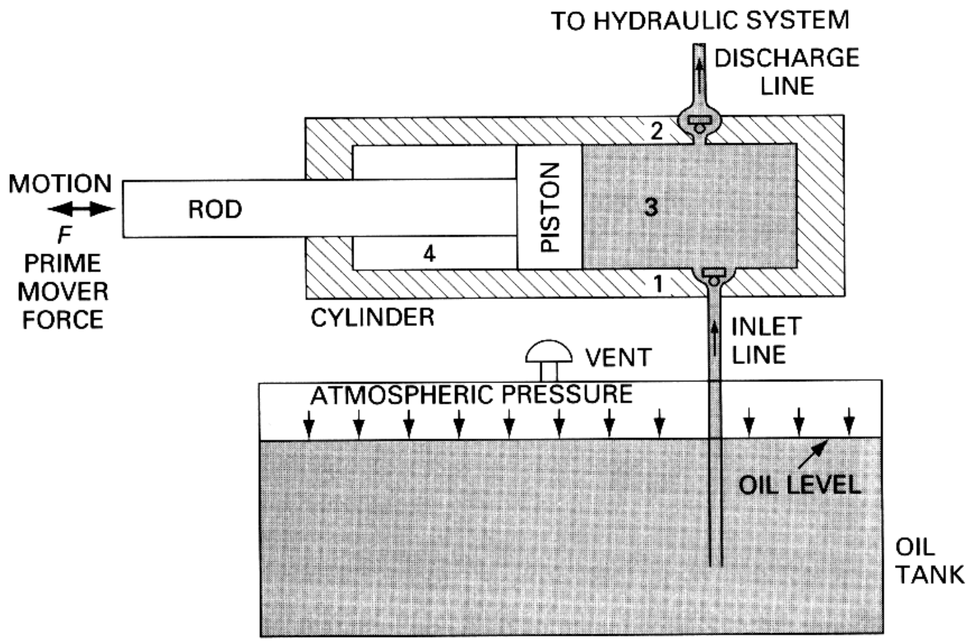
Pump Theory

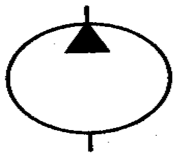
A Pump, which is the Heart of a Hydraulic System, converts Mechanical Energy into Hydraulic Energy. The Mechanical Energy is delivered to the Pump via a Prime Mover such as an Electric Motor. Due to Mechanical Action, the Pump Creates a Partial Vacuum at its Inlet. This permits Atmospheric Pressure to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

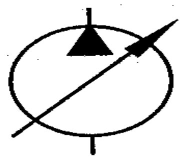
For Example

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.

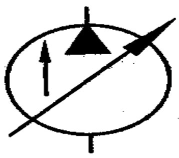




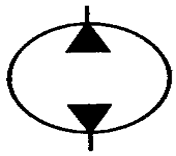
a. Uni-directional
Fixed displacement



b. Uni-directional
Variable displacement



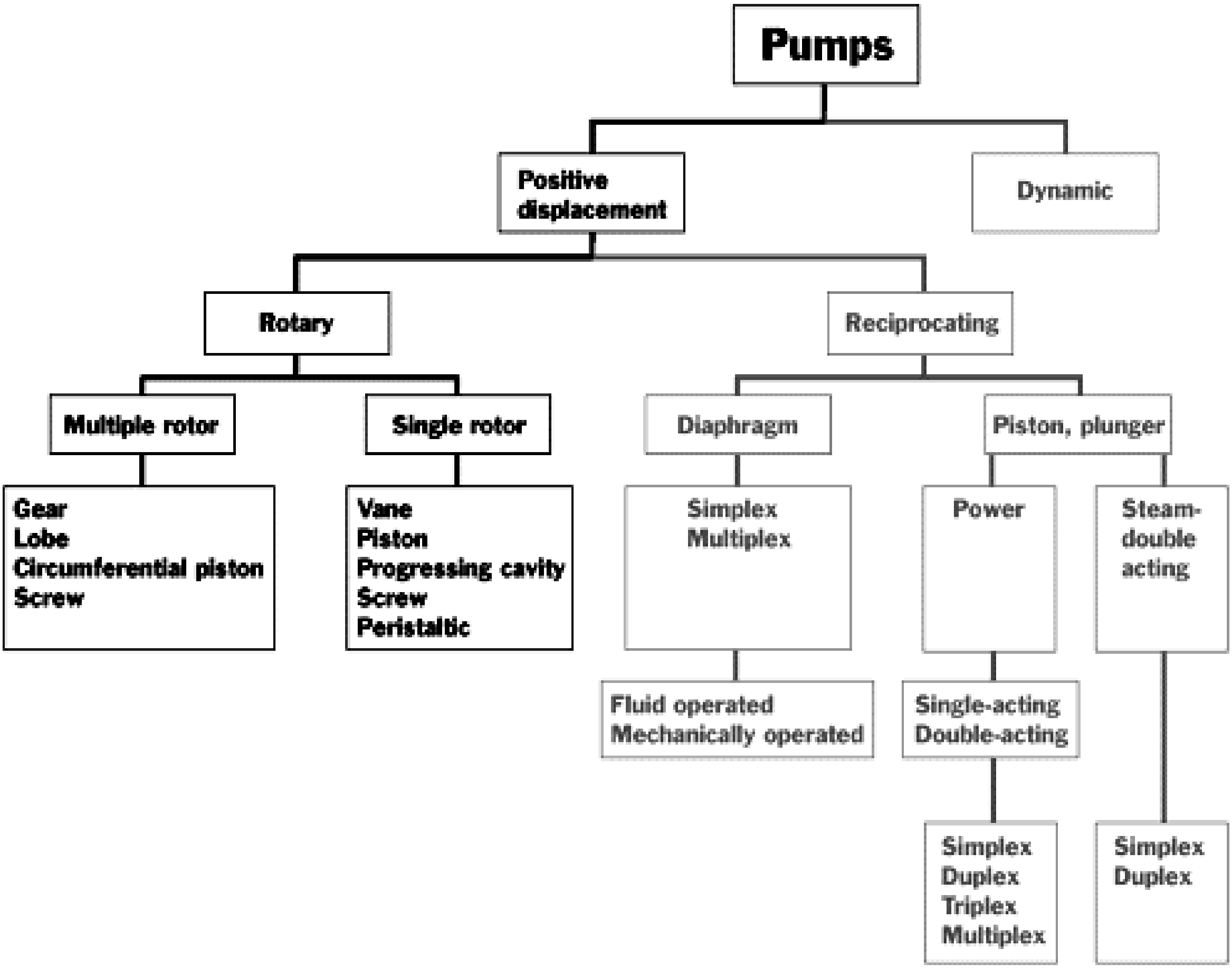
c. Uni-directional
Variable displacement
Pressure-compensated



d. Bi-directional
Fixed displacement

Pump Classification

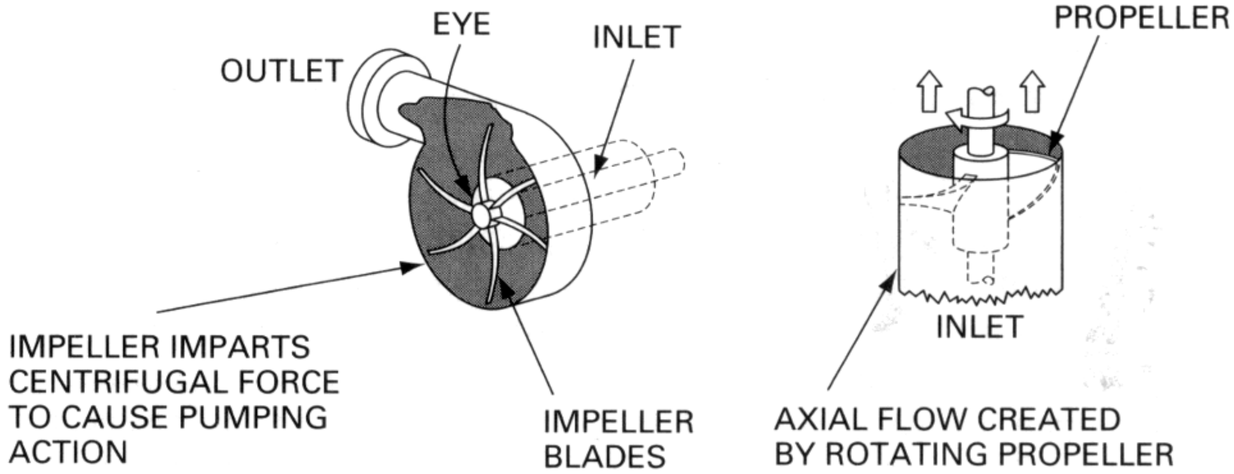
1. Dynamic (Non-Positive Displacement) Pumps
2. Positive Displacement Pumps



1. Dynamic (Non-Positive Displacement) Pumps

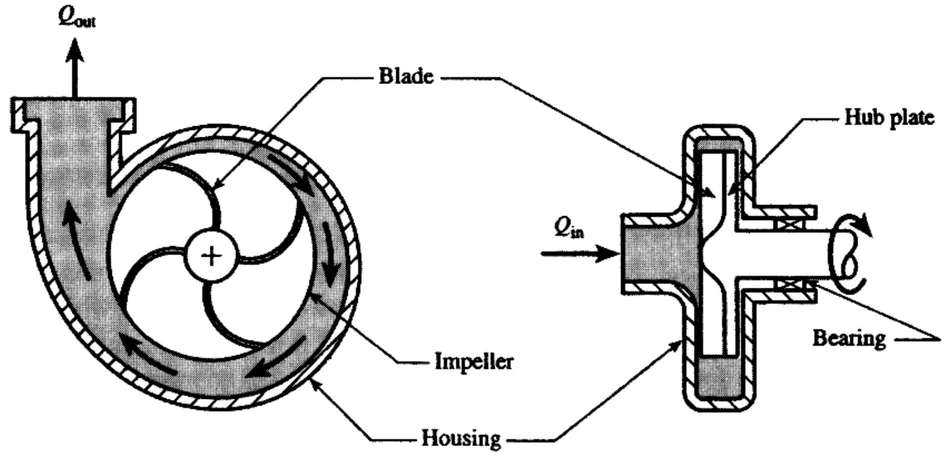
This type is generally used for: Low-Pressure, High-Volume Flow Applications because they are Not capable of withstanding High Pressures. They are of Little Use in the Fluid Power Field. This type of Pump is primarily used for transporting fluids from One Location to Another. The Two Most Common Types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

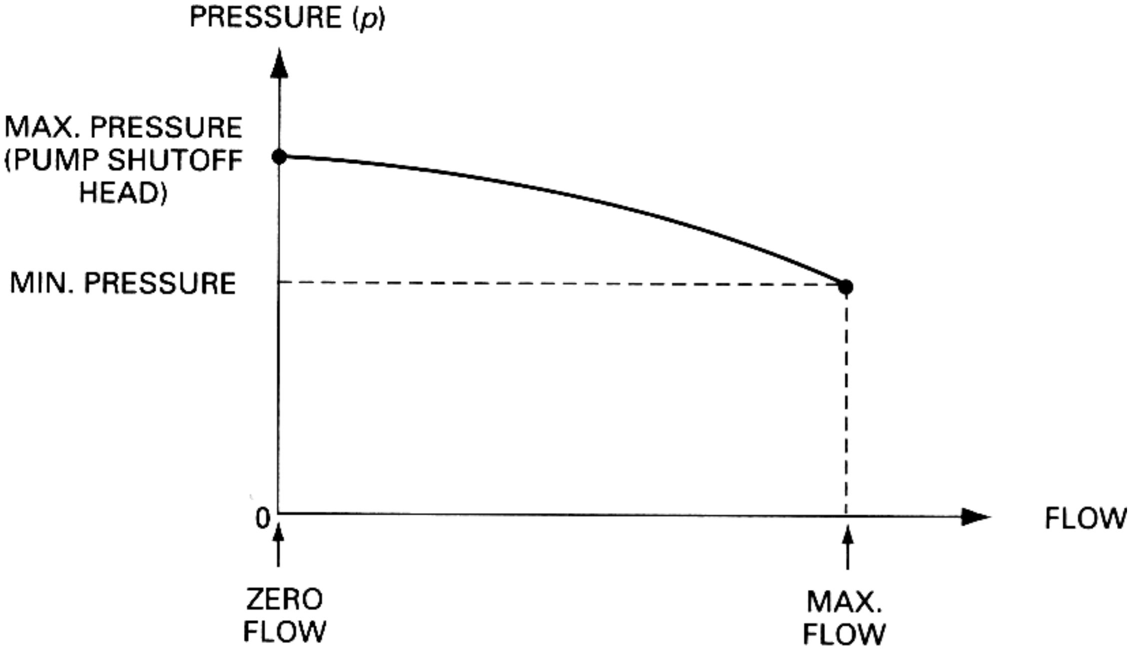


(a) Construction features

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:

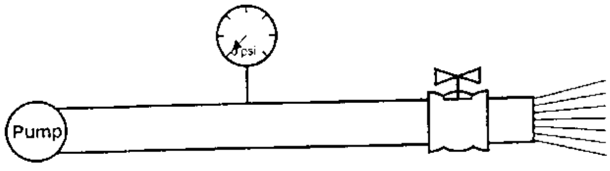
a. High-Pressure Capability (up to 12,000 psi)
b. Small, Compact Size
c. High Volumetric Efficiency
d. Small Changes in Efficiency throughout the design pressure range
e.Great Flexibility of Performance

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

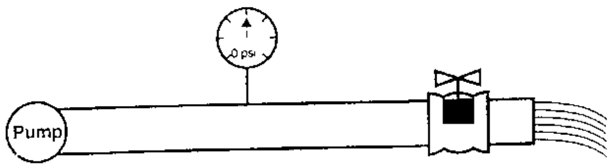
There are Three Main Types of Positive Displacement Pumps: Gear, Vane, and Piston. Many

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.

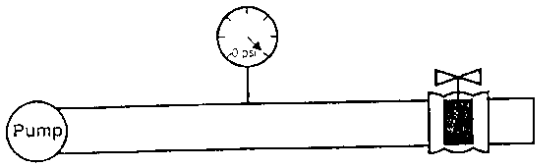
A Fixed Displacement Pump is one in which the amount of Fluid ejected per revolution (displacement) cannot be Varied. In a Variable Displacement Pump, the Displacement can be varied by changing the physical relationships of various pump elements. This Change in pump Displacement produces a Change in 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 Over pressure 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

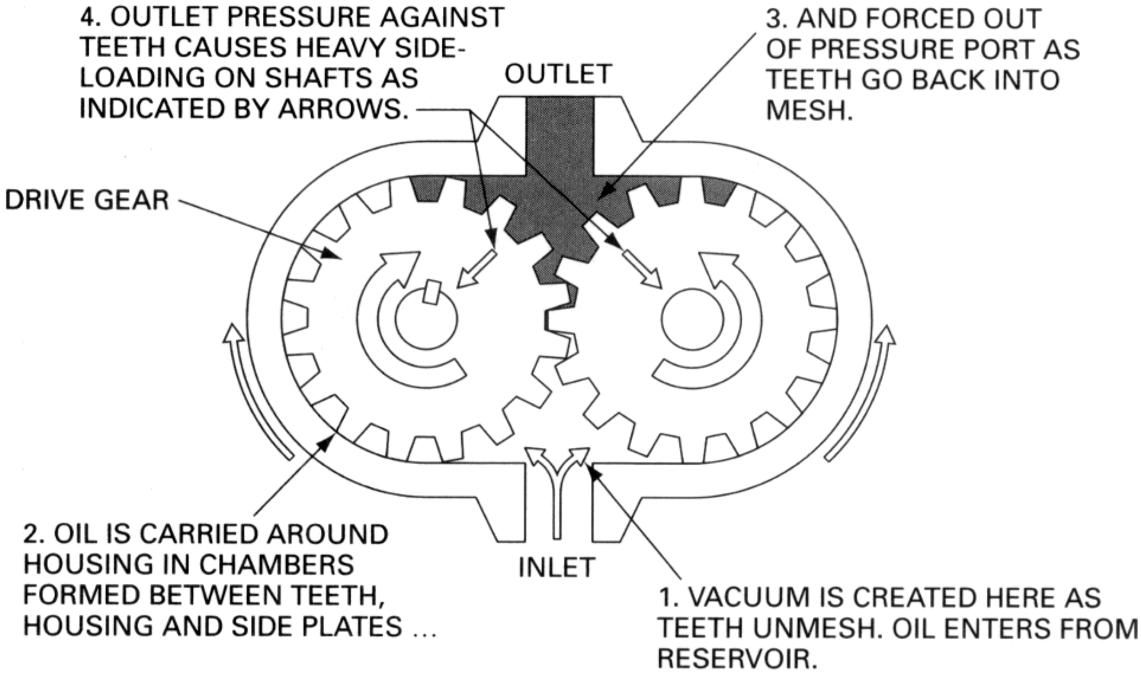
The Unbalanced Vane Pump have Pressure Compensation Capability, which Automatically Protects the Pump against Overpressure.

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.



<u>Volumetric Displacement and Theoretical Flow Rate</u>

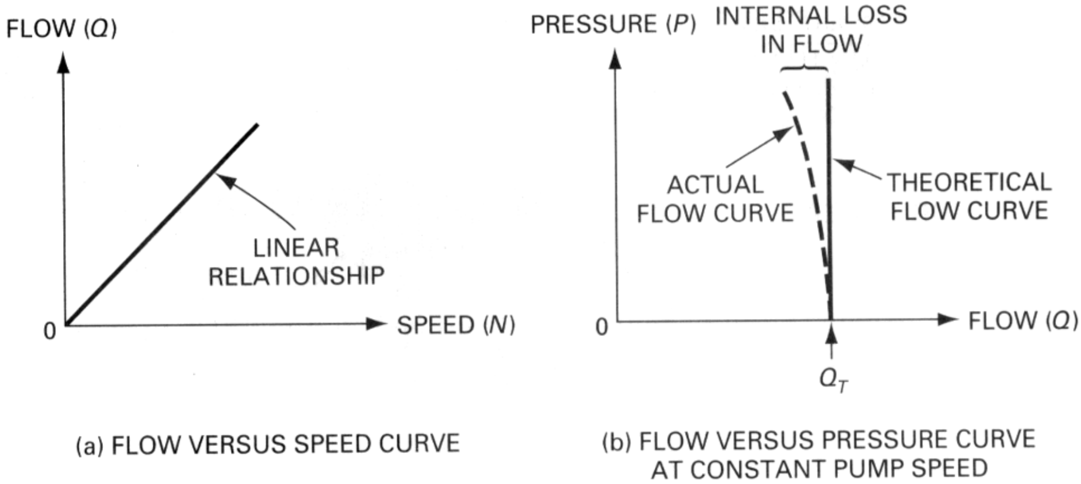
D_0 = Outside Diameter of Gear Teeth (in, m)
D_i = Inside Diameter of Gear Teeth (in, m)
L = Width of Gear Teeth (in, m)
V_D = Displacement Volume of Pump (in3/rev, m3/rev)
N = rpm of Pump
Q_T = Theoretical Pump Flow-Rate

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

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

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

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



Volumetric Efficiency

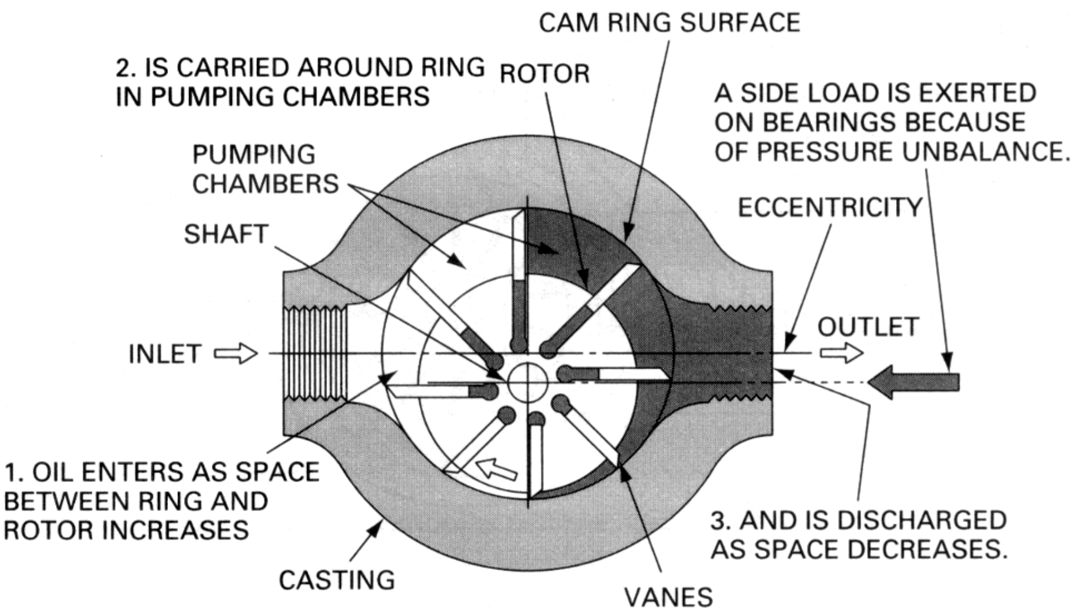
There must be a Small Clearance (about 0.001 in) between the Teeth Tip and Pump Housing. As a Result, Some of the Oil at the Discharge Port can Leak directly Back toward the Suction Port. This Means that the Actual Flow-rate Q_A is less than the Theoretical Flow rate Q_T, which is based on Volumetric 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:

$$\eta_v = \frac{Q_A}{Q_T} \qquad (3)$$

The Higher the Discharge Pressure is, 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 Loador a Closed Valve in the Pump Outlet Line.

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

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

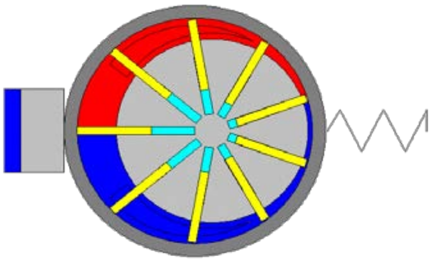
Substituting the expression for e_{max} yields

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

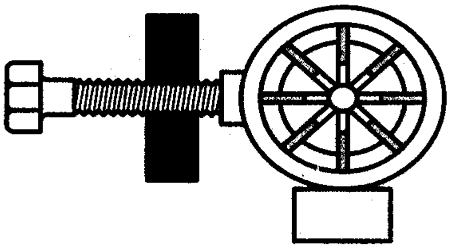
The Actual Volumetric Displacement occurs when e_{max} = e:

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

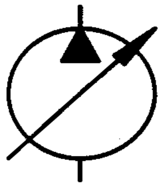
Some Vane Pumps have provisions for Mechanically Varying the Eccentricity. Such a Design is called a *Variable Displacement Pump.* A Hand wheel or a Pressure Compensator can be used to move the Cam Ring to Change the Eccentricity. The Direction of Flow through the Pump can be reversed by Movement of the Cam Ring on Either Side of Center.

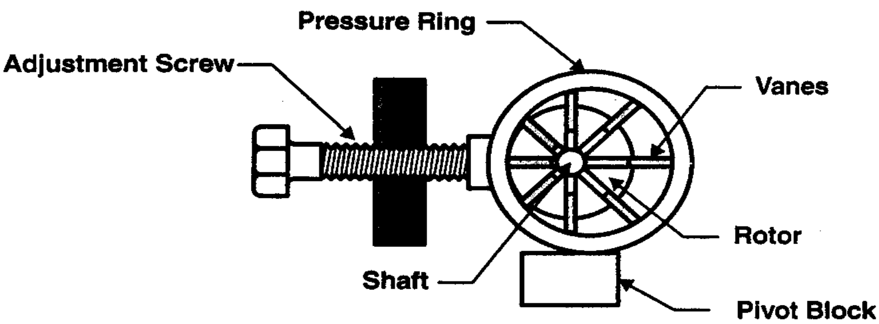


Variable Displacement Vane Pump



(A) Off Stroke





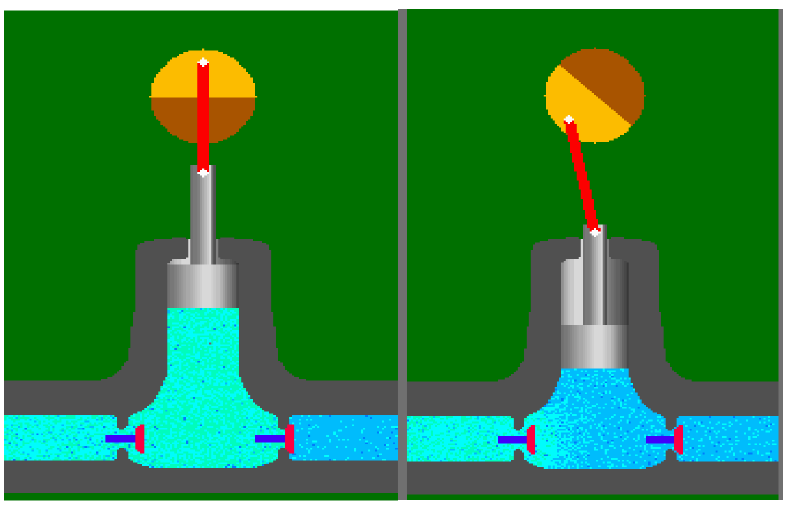
(B) On Stroke

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

Piston Pumps

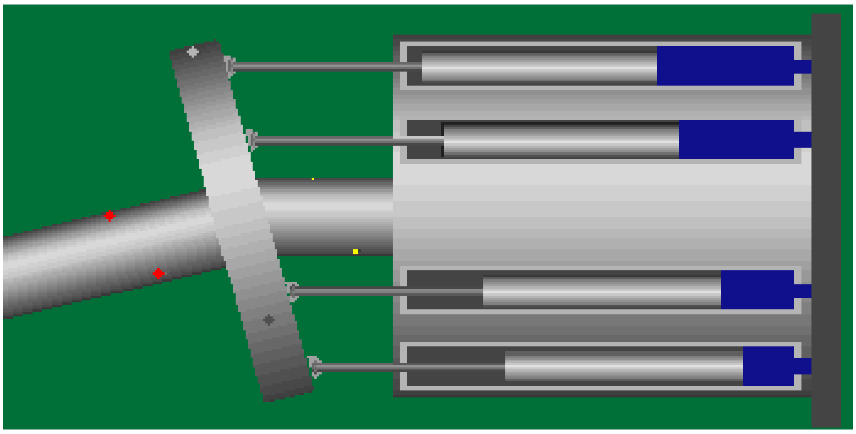
Introduction

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.



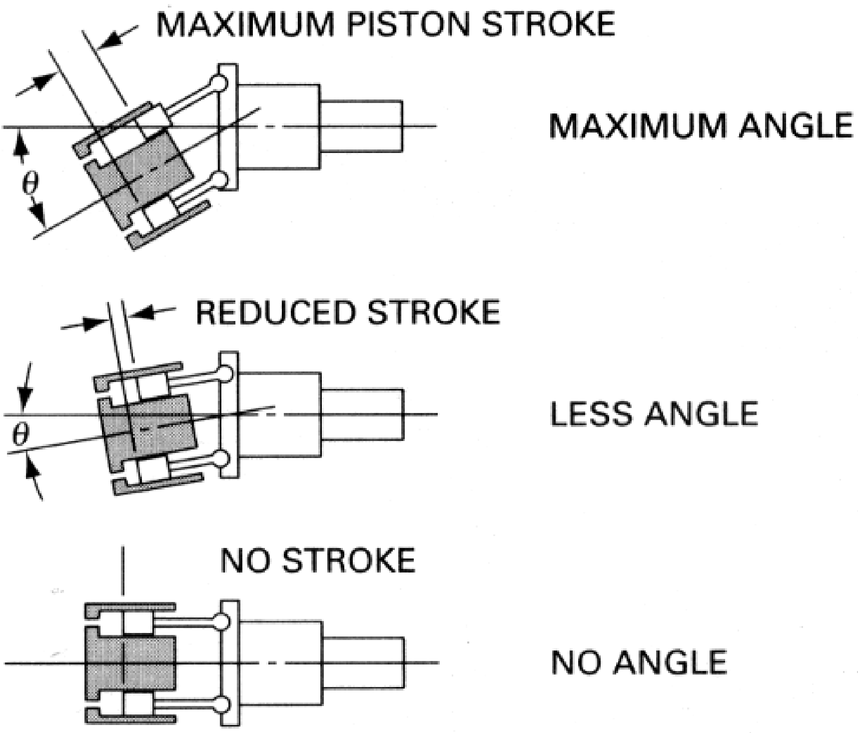
Axial Piston Pump Bent-Axis Design

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.



Bent-Axis Design Axial Piston Pump

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.



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

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

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.

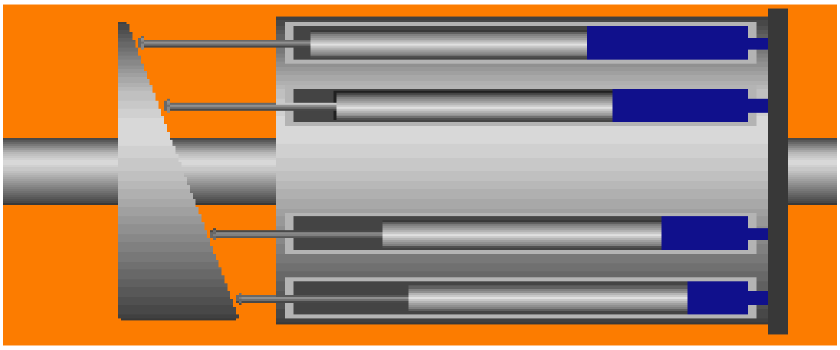
$$Q_T(\text{gpm}) = \frac{DANY\tan(\theta)}{231}$$

Relationship for the Theoretical Flow-Rate in Metric units.

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

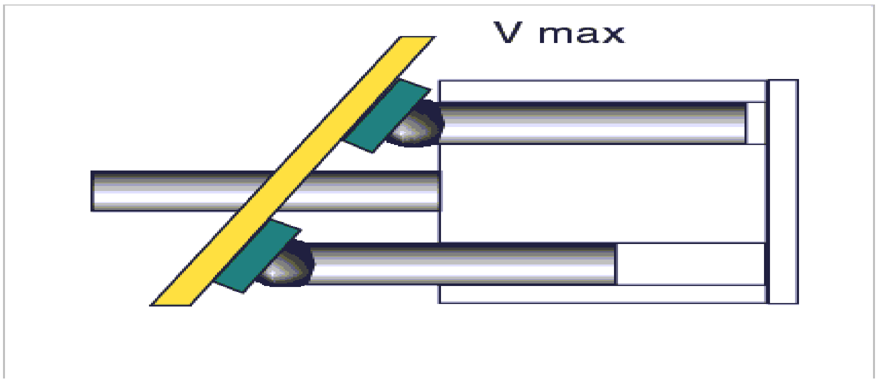
Swash Plate Design Axial Piston Pump

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.

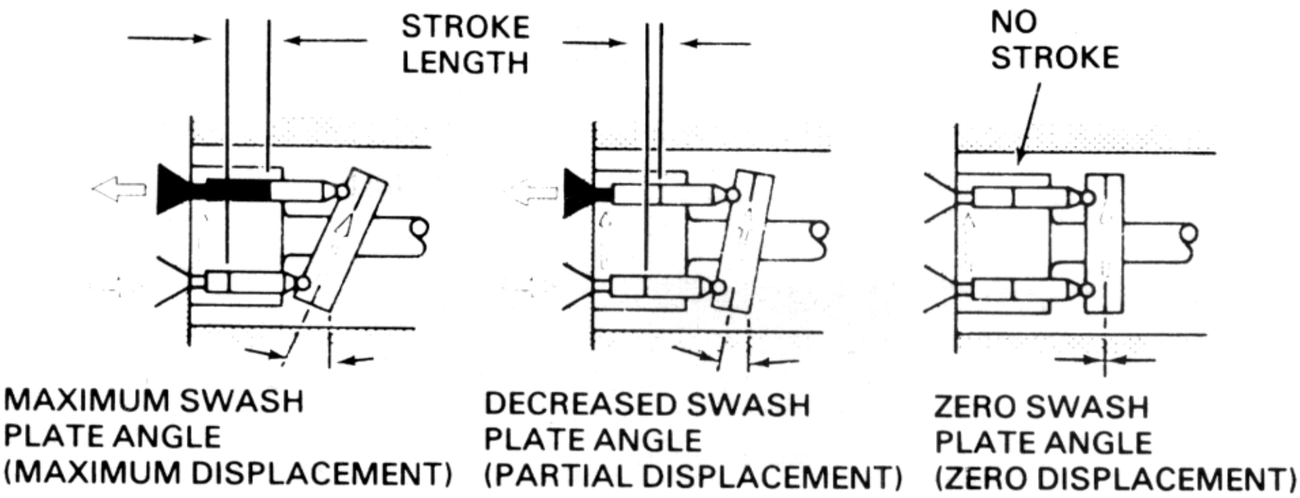


Swash Plate Design Axial Piston Pump

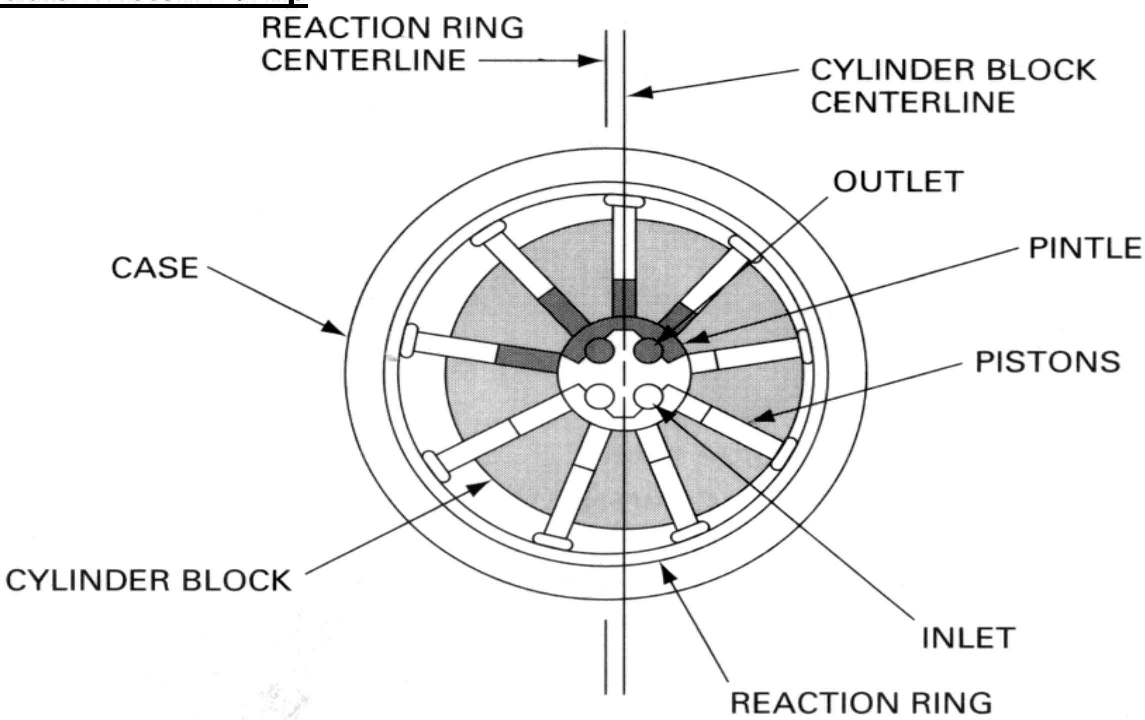
The Swash Plate Design Inline Piston Pump can also be designed to have Variable displacement capability. In such a Design, the Swash Plate is mounted in a Movable Yoke. The Swash Plate Angle can be changed by pivoting the Yoke on Pintles. Positioning of the Yoke can be accomplished by: Manual Operation, 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 henceis 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.