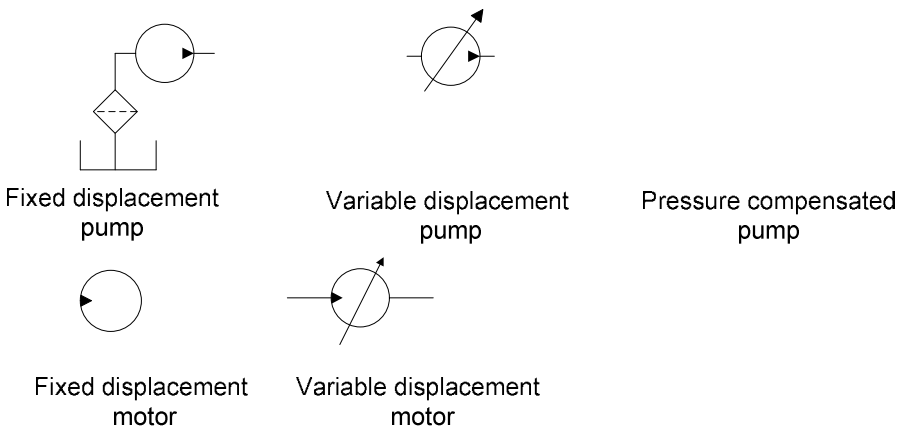
Chapter 4. Pumps and Motors

4.1 General:

Pump/motor Symbols



4.1.1 Pumps/motors

- Pumps are units that transfer mechanical (electrical) power into fluid power
- Motors transfer fluid power into mechanical power (translational or rotary)
- Pumps can be driven by electric drives, internal combustion engine, turbine drive, PTO's (Power take offs) etc.
- It is essential that the rpm of the input drive be compatible with the rated rpm of the pump. The use of gears for high speed main drives is often employed. Also belts and chains are used which can also acts as gears to reduce or increase speed.
- As a quick review, if N_1 is the number of gears in gear 1 and N_2, in gear 2, the gear ratio is $\frac{N_1}{N_2}$.
- If electric drives are used, the most common outputs speeds are:
  860 - 1140 - 1750 - 3420 rpm
  (900 - 1200 - 1800 - 3600 rpm) Nominal ratings

- Most industrial drives are 1140 or 1750 rpm, 21 MPa, 80 lpm
- aircraft and missile pumps (typical):
  - 11,000 -14,000 rpm at 40 – 50 lpm and 21 MPa

4.1.2 Pump/Motor Ratings

- Displacement: l/rev (in^3/rev)
- Flow Rate l/min (gpm(US))
- Pressure (MPa), (Bars), (psi)
- power(PQ)

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4.2 Pump/motor types

- This section introduces the various types of pumps and motors that are available for hydraulic systems.
- Since pumps and motors are similar in many cases, just one shall be discussed.
- There are two main categories of pumps and motors - hydrodynamic and positive displacement

4.2.1 Hydrodynamic

- Two basic types exist - centrifugal and turbine pumps.

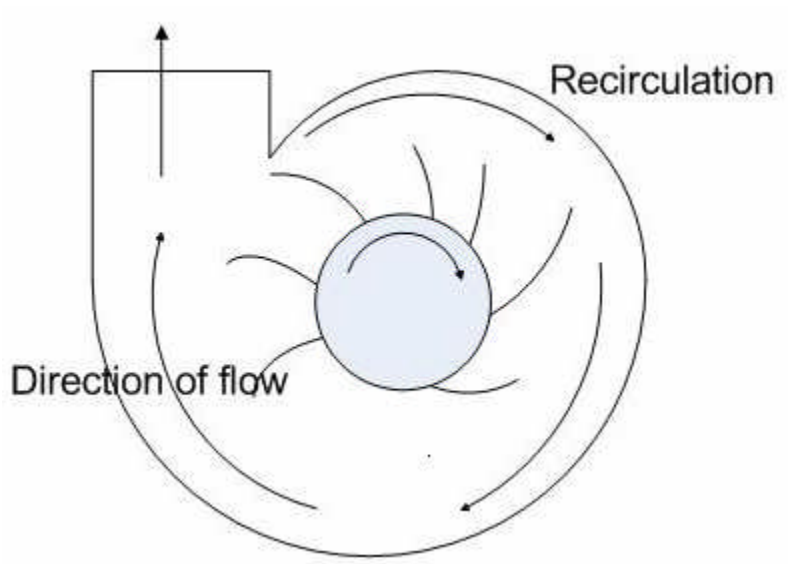

Figure 4.1.1 Centrifugal (non – positive displacement) pump

USE:

Used as auxiliary functions in hydraulic systems. (Primarily for circulating fluid through cooling and cleaning devices and supercharging larger piston or other rotary pumps.)

Used in most applications where the only resistance encountered is that created by the weight of the fluid itself and friction.

LIMITATION:

Because there is no positive seal between inlet and outlet, the pump can be completely blocked with the pump running. Therefore, their flow capabilities depend on the load resistance.

4.2.2 Positive Displacement

- A positive displacement pump delivers a fixed amount of fluid per revolution.
- Two types: Fixed Displacement - capacity (outlet flow) is fixed
  Variable Displacement - flow capacity can be varied

POSITIVE DISPLACEMENT PUMPS DO NOT GENERATE PRESSURE. THEY GENERATE FLOW TO A CIRCUIT. THE NATURE OF THE CIRCUIT LOAD DETERMINES THE PUMP PRESSURE.

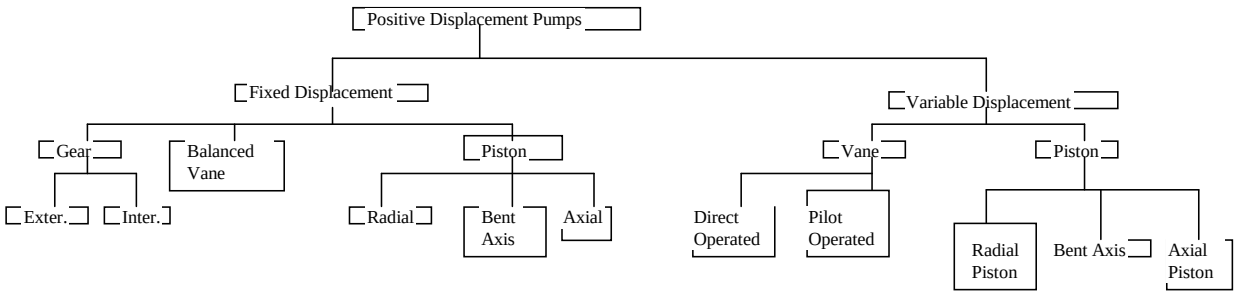

Figure 4.2.1 Pump classification

4.2.3 Gear Pumps and Motors

4.2.3.1 External Gear Pumps and Motors. (Figure 4.2.2)
Most gear motors are uni-directional unless special units are employed

OPERATION:
- Spur gears trap fluid between gear teeth and casing. In a pump capacity, fluid is removed from suction cavity reducing pressure. Fluid is drawn from a reservoir to replace it.
- When used as a motor, an external drain which is connected to the bearings for lubrication is needed.
- Torque is produced through pressure on the surface of gear teeth.
- These are not balanced units and hence can have reduced mechanical efficiency under high pressures.

LIMITATIONS:
- Usually restricted to pressures of 13.8 MPa (2000 psi) and 2400 rpm. But can be operated up to 31 MPa (4500 psi) with special bearings.
- Spur gear types can be noisy.

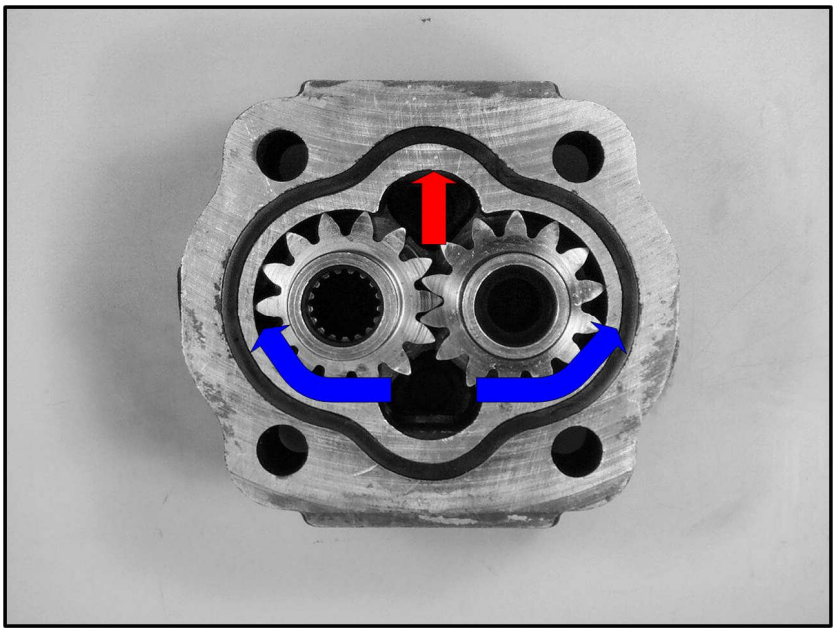

Figure 4.2.2 External Gear Pump

ADVANTAGES & DISADVANTAGES (Also applies to internal and gerotor types.)

- Simplicity of operation and they have a high dirt tolerance.
- Offset by a lower efficiency.
- Most popular in low pressure range.
- Display good life expectancy.
- Low in cost.
- Are fixed displacement by nature.

4.2.3.2 Internal Gear Pumps/motors (Figure 4.2.3)

- Gears unmesh and remesh at a relatively low speed (smoother flow)
- Usually a lower pressure range unit.

OPERATION:
- With reference to Figure 4.2.3, the drive gear is attached to shaft. (pump operation) Withdrawal of the drive gear from the internal rotor gear on the suction sides creates a partial vacuum drawing fluid. Intermeshing of gears at outlet discharges fluid to system.
- Characteristics similar to external gear pumps.
- Used where gear shaft must pass through pump such as automatic transmissions.

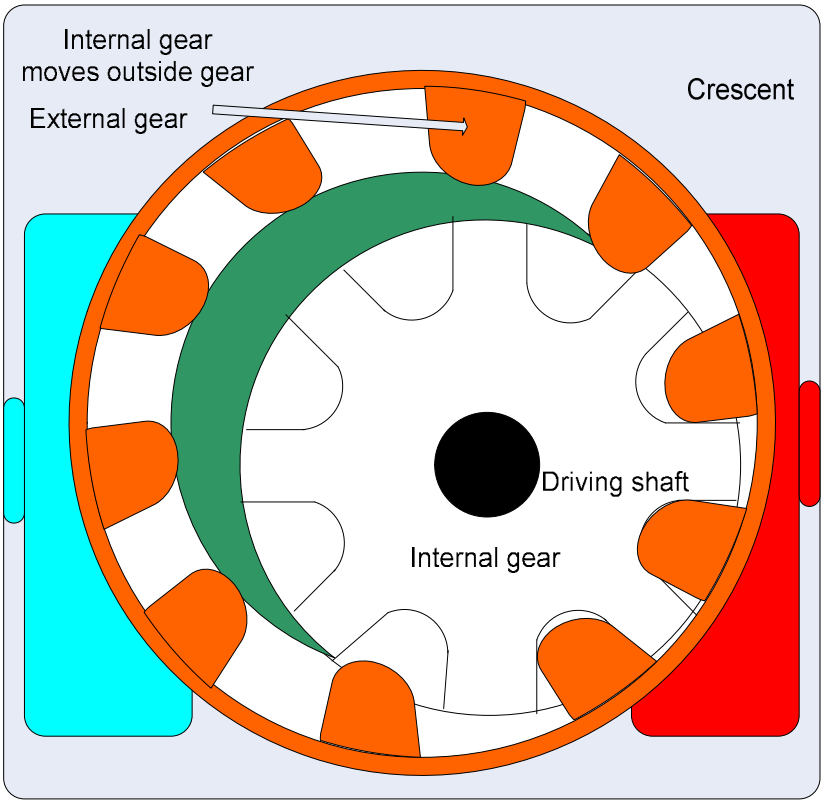

Figure 4.2.3 Internal Gear Pump

4.2.3.3 Gerotor

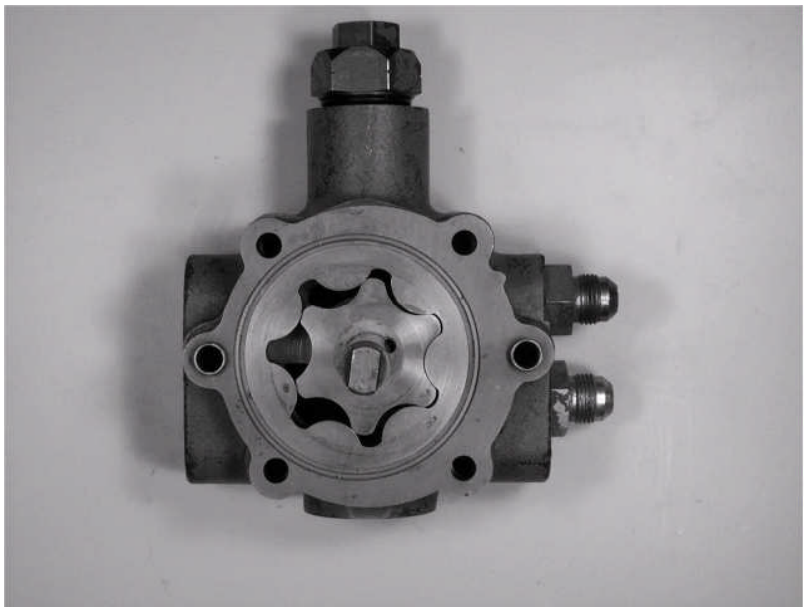

Figure 4.2.4 Gerotor

OPERATION: - With reference to Figure 4.2.4, the inner rotor is driven and drives the outer rotor in the mesh. The tips of the inner rotor contact the outer rotor to seal the chambers from each other.
- Used for presses, injection molding machines, etc.
- Two pumps cascaded in series can increase pressure. Pressure usually limited to 13.8 MPa (2000) psi.
- Pump capacity is determined by vol. of missing tooth x number of driving teeth = total vol./rev.

4.2.3.4 Problems with gear pumps

Leakage can occur in gear pumps/motor and can occur
- cross ports (outlet to inlet)
- around the outside (minor)
- in the middle (major)
- along the sides

To minimize, use bronze wear plates

4.2.3.5 Screw Pumps

- Flow is generated by the action of meshing screws.
- Two screw pump - two parallel rotors with intermeshing threads rotating in a closely machined housing.
- Flow is axial.
- No flow pulsations as delivery is continuous.
- Very quiet operating pump.

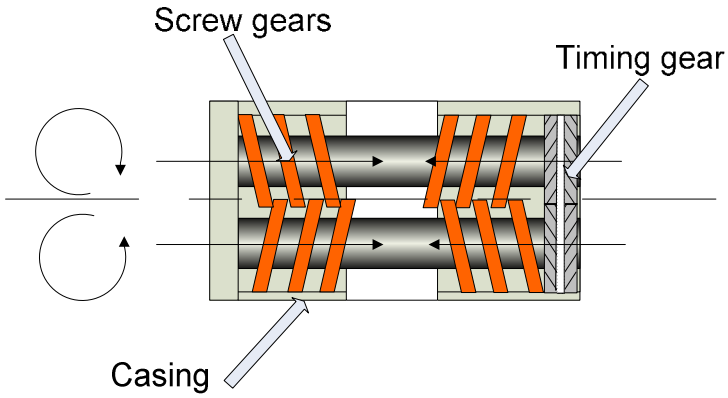

Figure 4.2.5 Screw Pump

ADVANTAGES & DISADVANTAGES

- Quiet and smooth flow.
- Excellent pressure and flow ranges.
- Used in submarines (noise is critical).

4.2.4 Sliding Vane Pumps & Motors

In sliding vane pumps and motors, the pumping action is created by the changing volume between sliding vanes. Such a pump is shown schematically in Figure 4.2.5. As the rotor on the shaft rotates, the vanes slide in and out of the vane guides. For the pump shown, the volume trapped between the two vanes either decreases (discharging fluid) or increases (drawing in fluid). The vanes are held out against the stator (ring) by centrifugal action and by ported pressurized fluid at the bottom of the vanes as illustrated. Other high performance types of pumps/motors use "rocker springs" to keep the vanes in contact with the outer casing.

This unit is unbalanced. If one looks at Figure 4.2.5, a net force acts on the shaft bearings because one side of the rotor is at high pressure and the other at low pressure. This unbalance can reduce the mechanical efficiency .

This unit as shown is also fixed displacement.

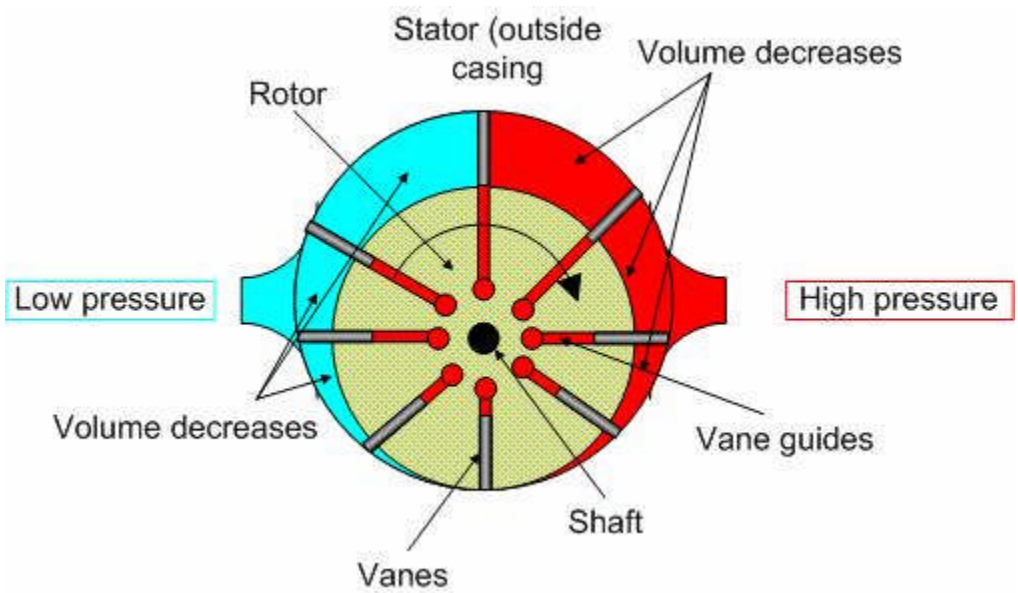

Figure 4.2.5 Schematic of a Vane Pump (Unbalanced and fixed displacement)

An example of a balanced vane pump is shown in Figure 4.2.6. There are two discharge and inlet ports now. Thus the differential force is counterbalanced by an opposite force which then acts to balance the net force on the bearings. Thus mechanical efficiency is improved.

Note also the different size porting holes. The reason for this is to cerate a more uniform discharge of flow at different locations of the vane chambers. For the point where the vane chamber is small, the fluid velocity can be very high. Thus the discharge hole at that location is made larger to compensate.

It is just barley visible in Figure 4.2.6 but on two kidney ports on the valve plate (right hand side), there are small little grooves on each end. The purpose of these grooves to prevent pressure spikes from occurring in the small transition regions between the high pressure and low pressure ports. If fluid is trapped in a small volume with no where to go between the two regions, pressure can rise dramatically; hence groves bleed off the fluid to the low pressure side effectively reducing these spikes but not substantially increasing leakage between the two ports.

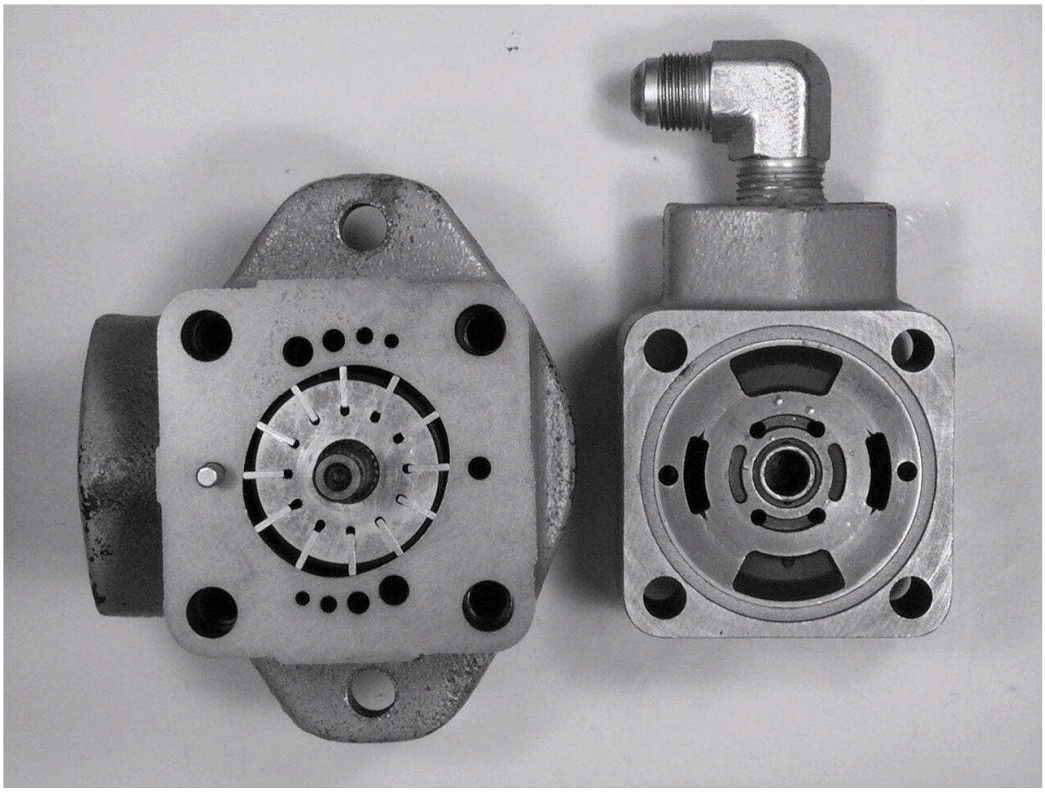

Figure 4.2.6. Vane pump (Balanced)

LIMITATIONS of VANE PUMPS/MOTORS:

Pressures can be limited to 17 MPa (2500 psi) for a single stage (or unbalanced) pump or motor. Speed of rotation is limited to less than 2500 rpm due to centrifugal forces.

ADVANTAGES & DISADVANTAGES:

Variable displacement vane pumps are commercially available.
The units are reliable, efficient, and easy to maintain.
Ring and surface wear is compensated by vanes moving out of their slots.
Note: Vane pumps can suffer from cavitation problems due to "shooting" of the vane to the ring during the transition from low pressure to high pressure.

4.2.4.1 Other considerations

- Pressure plates (seals rotor to body) are used to seal the vanes at their sides
- Rocker arms - force vanes against cam ring. The use of pressurized fluid at the bottom may cause excessive forces on the vanes at high pressures.
- Shuttle valve arrangement makes it reversible
- Variable displacement possible by varying ring "throw" or eccentricity (to be discussed).

4.2.4.2 High performance vane motors

- Ring, rotor, vane, and side plates as an assembly is removable as a cartridge
- Vanes held out against the ring by coil springs

4.2.4.3 Combination high pressure Vane Pumps (Figure 4.2.7)

A special type of pump can be created by using one vane pump as the pre-charger to a second pump. This can step up pressure considerably. Both units are driven at the same speed. But if one unit is just slightly out of sink with the other, then cavitation or excessive pressure in the intermediate line can. This is compensated for by the controller shown in Figure 4.2.7. It is left as an exercise to the reader to reason how the controller works.

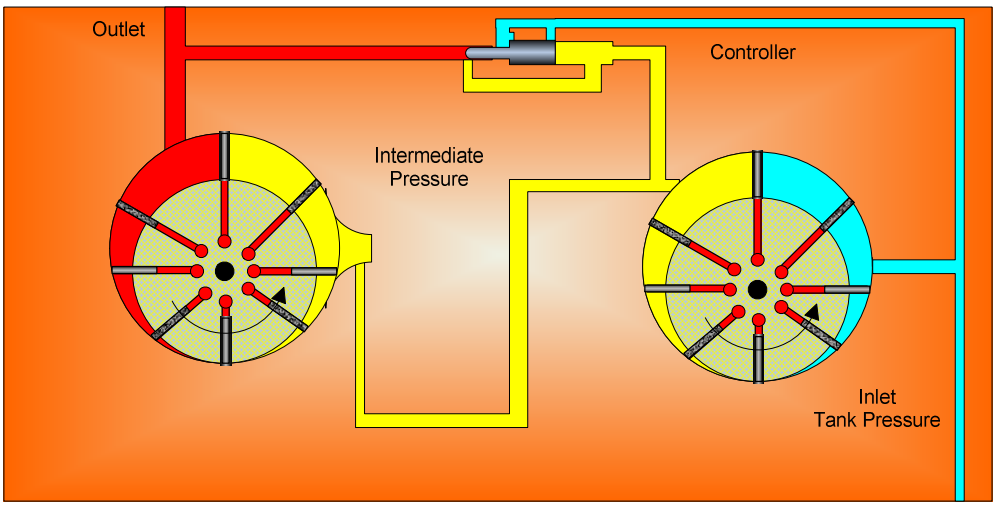

Figure 4.2.7 Combination pump

4.2.4.4 Variable displacement pumps (Vane) (Pressure Compensated) Figure 4.2.8

- This is accomplished by varying the "ring throw" (eccentricity).
- This is acheived by making the cam ring housing adjustable from maximum eccentricity (Figure 4.2.8(a)) to a concentric position. Fig 4.2.8(b)

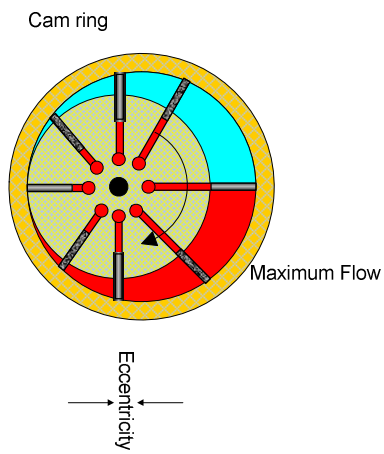

Figure 4.2.8 (a) Maximum eccentricity

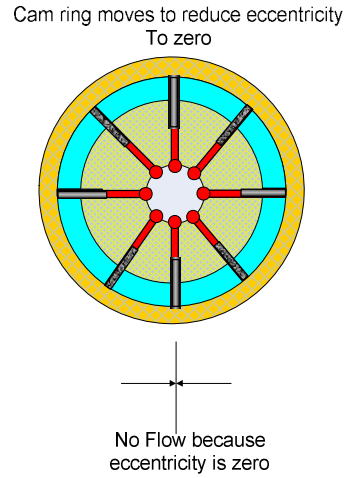

Figure 4.2.8 (b) Minimum throw

At this point it is useful to introduce the idea of pressure compensation Consider Figure 4.2.9.

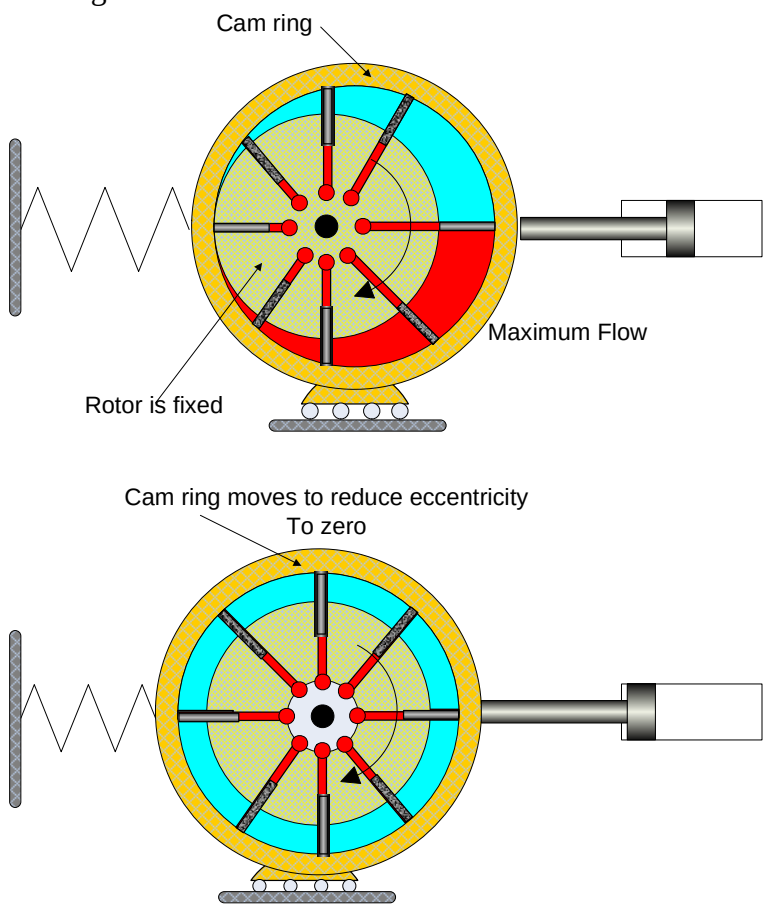

Figure 4.2.9 Pressure compensation

- A pressure compensated pump is one that changes its volume output with changes in system pressure.

  In the top figure, there is no pressure on the blank side of the piston, hence no force. The spring on the left hand side forces the can ring and the piston the right as shown in the top figure. The pump is now at maximum eccentricity and hence flow is maximum. Pressure builds up on the blank end of the piston due to the load, but flow is still maximum until the spring pretension is just overcome by pressure forces on the piston (labeled as the **cut-off pressure** in Figure 4.2.10). At this point, the cam ring shifts to the left, reducing eccentricity and hence reducing flow. Eccentricity is reduced until the rotor is at zero eccentricity (this is called the deadhead pressure – see Figure 4.2.10).

  It is very important to realize that a pressure force greater than the sum of spring pretension and the spring constant (times compression displacement of the spring) must exist to reduce eccentricity. This is partially why the slope of the pressure compensated pump between the cut off and deadhead pressuare is at some slope. If this pressure is reduced below the spring pretension, then the spring pushes the can ring back to its original position and flow is now maximum. In essence in this particular region, the pump acts as a fixed displacement region.

  The pump flow characteristics are shown in Figure 4.2.10. The red line shows an actual curve which is not "flat" as the ideal case because of leakage in the system.

  The main purpose of the pressure compensator is to limit the maximum pressure in the circuit and to become a demand flow system. This will be discussed at great length in later chapters.

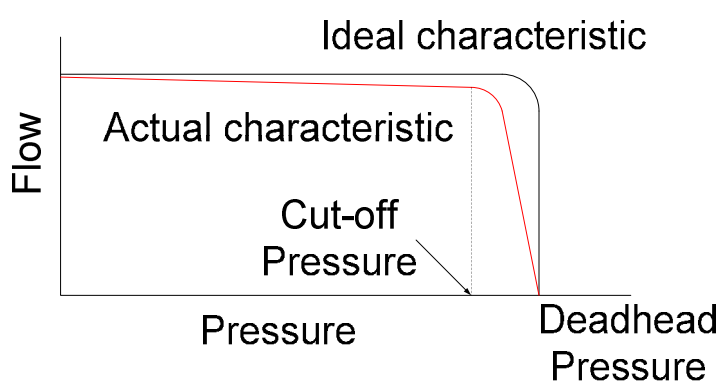

Figure 4.2.10 Characteristics of a Pressure Compensated pump. (The reduction in the actual flow vs. pressure from the ideal is due to leakage (an indicator of volumetric efficiency)

4.2.5 Reciprocating Pumps & Motors

Types: Radial piston, axial piston

BASIC PRINCIPLES:

- A piston reciprocating in a bore will draw in fluid as it is retracted and expel it on the forward stroke (pump) and vice-versa for a motor.
- Radial piston type arranged radially in a cylinder block.
- Axial piston type arranged parallel to each other and to cylinder block. (Can also be subdivided into in line (swash plate or wobble plate) and bent axis types.)

ADVANTAGES & DISADVANTAGES:

- Units provide the highest degree of sophistication found in pumps and motors
- Highest volumetric efficiency (97%)
- Higher pressure ratings (69 MPa (10,000 psi) and higher)
- High operating speeds (12,000 rpm)
- Since very efficient, less power is converted to heat, and hence smaller sizes and weight are possible

4.2.5.1 Radial Piston Pumps & Motors (Figure 4.2.11)

OPERATION: Cylinder block rotates on a stationary pintle and inside a circular reaction ring or rotor. As rotor block rotates, centrifugal forces, charging pressure, or some form of mechanical action causes the pistons to follow the inner surface of the ring which is offset from the centerline of cylinder block. Reciprocating action of the pistons discharge fluid or draw fluid into the appropriate posting chambers isolated by the pintle. The displacement can be varied by moving the reaction ring (slide block) to increase or decrease piston travel.

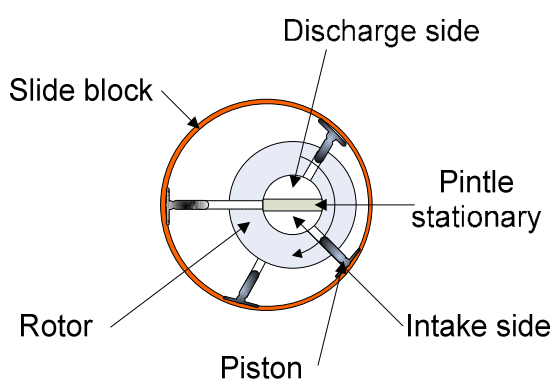

Figure 4.2.11 Pintle valve pump

4.2.5.2 Radial Piston Pumps & Motors

-Similar to the pintle valve arrangement in which the pistons move radially about some axis.

- These pumps can operate at very high pressures and produce very large flows
- Have a volumetric efficiency of ≅ 93% and an overall efficiency of 85-87%
- Displacement of the pump/motor = single piston area times its stroke times the number of pistons.

- Pumps with small displacement best suited for high pressure applications
- Start up priming may be necessary (noise a factor)

4.2.5.3 In-Line Axial Displacement Pumps & Motors

An in line axial displacement pump is one in which the pistons are in line with the rotating shaft. Figure 4.2.12 shows the various parts of such a pump or motor. Because it is a very common type of unit, we shall spend some time looking at each part.

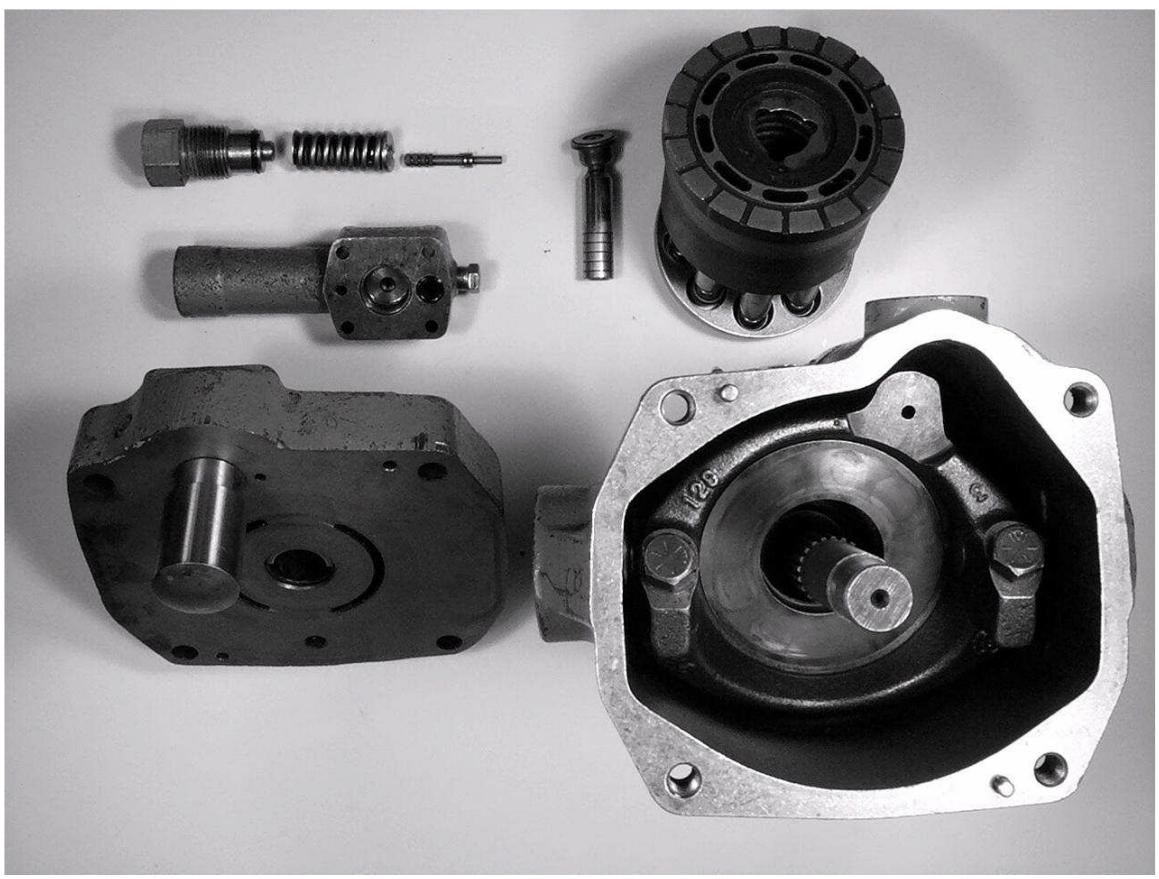

Figure 4.2.12(a) Axial in-line piston pump

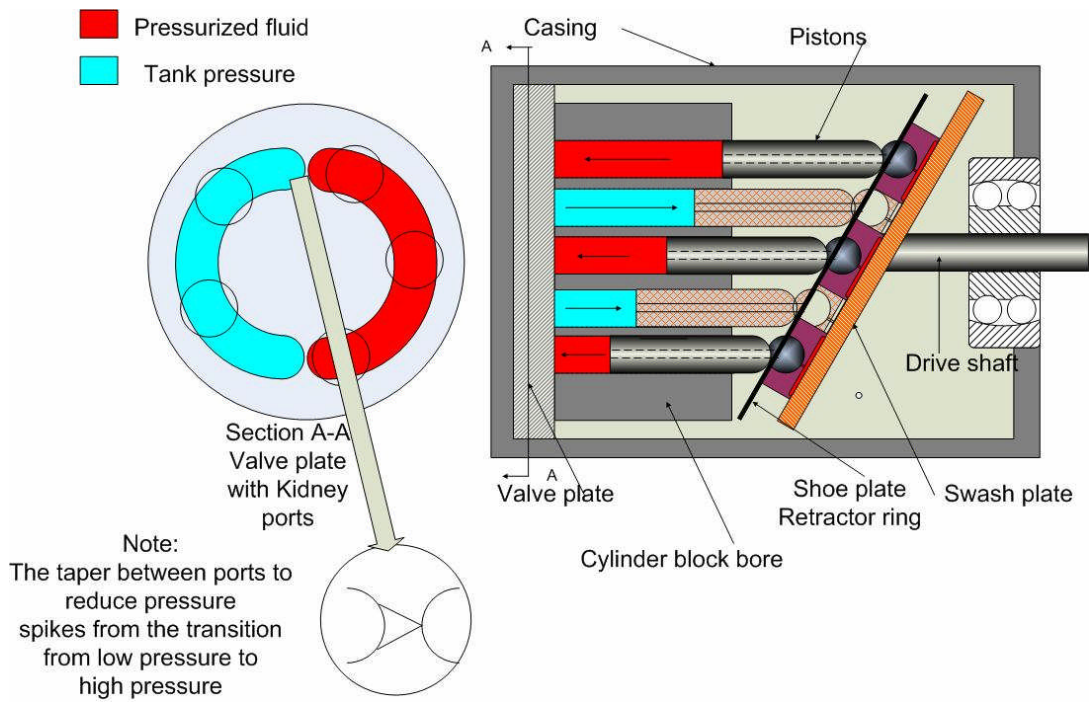

Figure 4.2.12(b) Axial in-line piston pump (Schematic but not exactly the same as shown in part(a))

- These pumps have high flow and pressure capabilities
- All pistons are in line with the shaft
- Usually there are an odd number of pistons (which have been shown to minimize pressure fluctuations)

<u>OPERATION</u>:

Pumps have a cylinder block with its pistons which is rotated on a shaft in such a way that the pistons are driven back and forth in their cylinders in a direction parallel to the shaft. In other designs, the cylinder and barrel remain stationary but the drive plate rotates.

- Figure 4.2.13 shows a typical piston and Figure 4.2.14 shows a side view of the piston held against the "swash plate" via a retaining ring. The retaining ring is also shown in Figure 4.2.15. The piston is attached through a swivel ball at its end, to a piston slipper. This slipper slides on a smooth surface (called a swash plate) as the barrel and hence pistons rotate about the shaft.
- Note the circular grooves around the piston base (Figure 2.4.13). These groves appear here to help distribute the pressure evenly around the piston and hence to keep it centered. It prevents "hydraulic lock" which can cause the pistons to become so de-centered that they literally fuse to the chambers in which they slide in (almost like a spot weld at the point of contact).

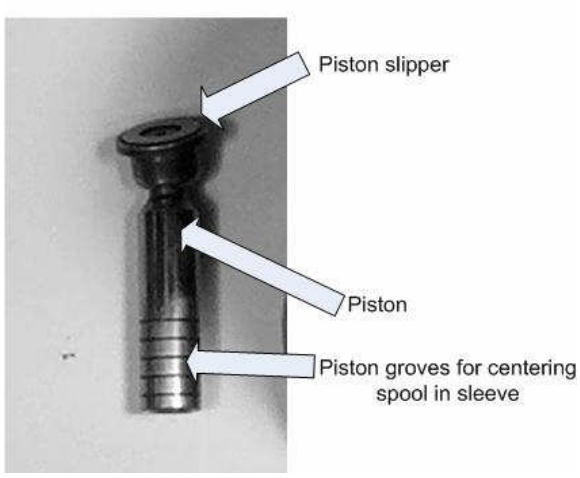

Figure 4.2.13 Typical piston

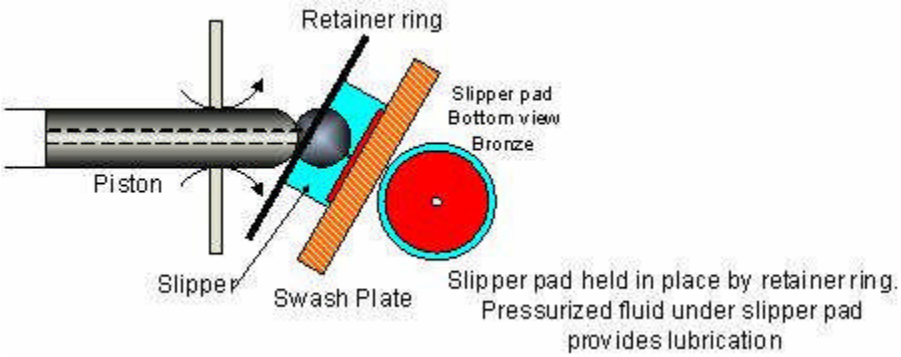

Figure 4.2.14 Side view of piston, slipper, retaining ring and swash plate

- Figure 4.2.15 shows the "barrel" in which the piston reciprocates. The barrel rotates around the middle shaft.
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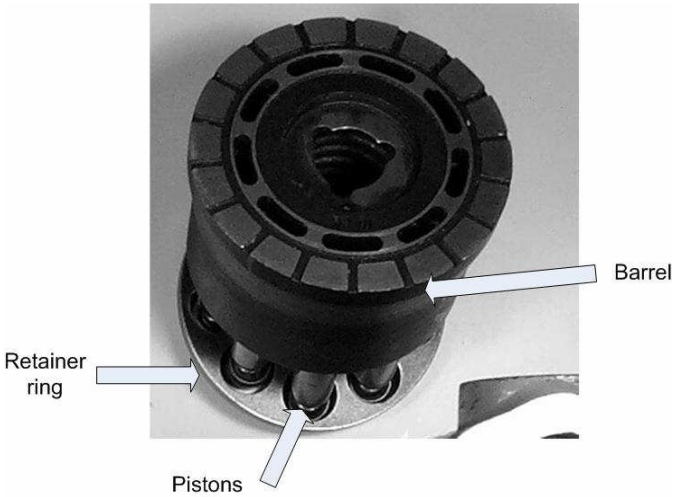


Figure 4.2.15 Barrel of an inline pump/motor

Figure 4.2.16 shows an end assembly which houses the swash plate on which the slippers slide. The swash plate itself is located on a “yoke” which is, in turn” is attached to a shaft called a pintle. The yoke pivots about the pintle which means that the angle the swash plate makes with respect to the barrel can be varied.

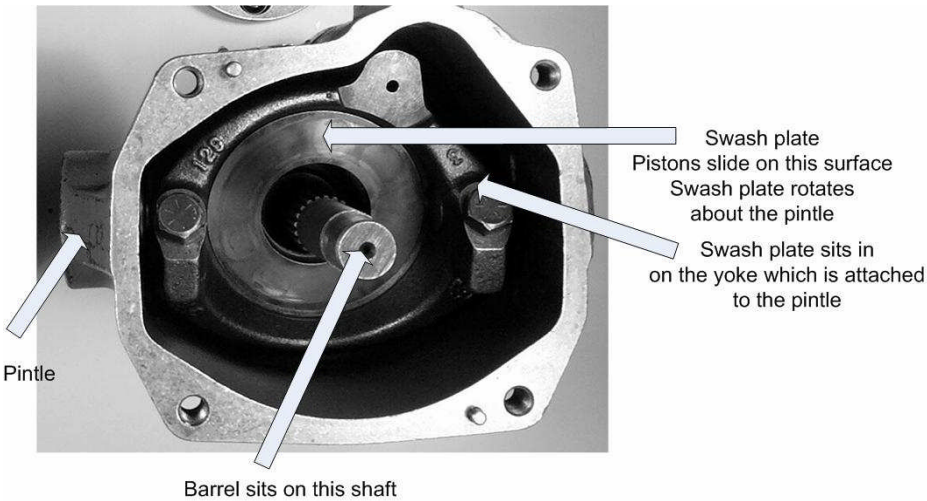


Figure 4.2.16 End assembly showing swash plate, yoke and pintle.

A side view of the pistons, barrel, retaining ring yoke and pintle is shown in Figure 4.2.17.

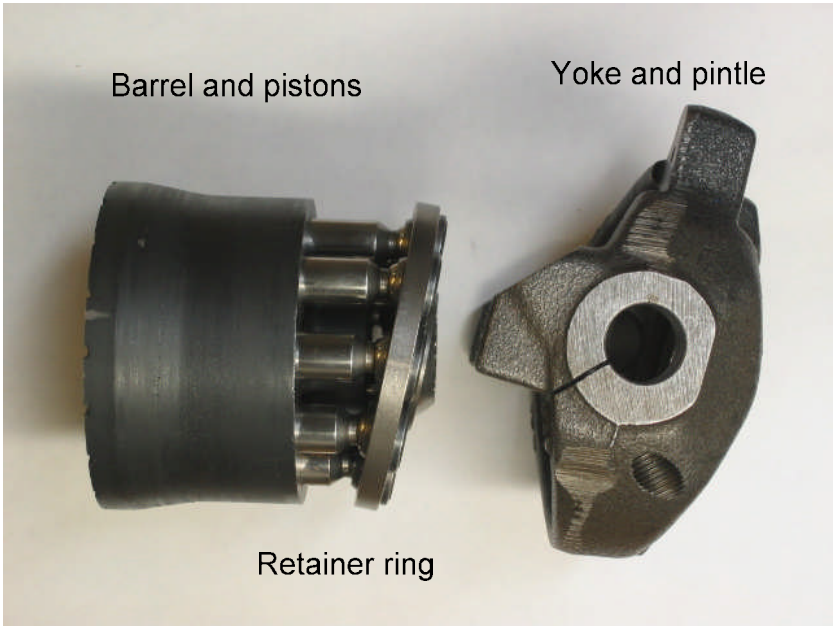


Figure 4.2.17 Side view of barely and yoke

- Unlike other pumps, piston pumps/motors have a case drain
- Fluid in case drain is essential for lubrication of bearings etc.
- Case drains can withstand some pressure, but usually not higher than ½ to 1 bar.

4.2.5.4 Valve end plate

The fluid which exits the barrel chambers is directed to the outside lines via a valve plate or end plate. For the pump illustrated in this work, the valve plate and end plate are essentially one of the same. Consider Figure 4.2.18. Consider only the centre area. (The unit that sticks out is called a control piston and shall be discussed later). The barrel rotates about the shaft which fits through the shaft hole and bearings. The kidney shape holes are called kidney ports and as the pistons travel towards (away) the plate and rotate within the barrel, the fluid is discharged (drawn in) through one of the ports. Because there are nine pistons in this unit, the kidney ports are exposed to four of them at any one time. Note again the groves at one ends of the kidney port.

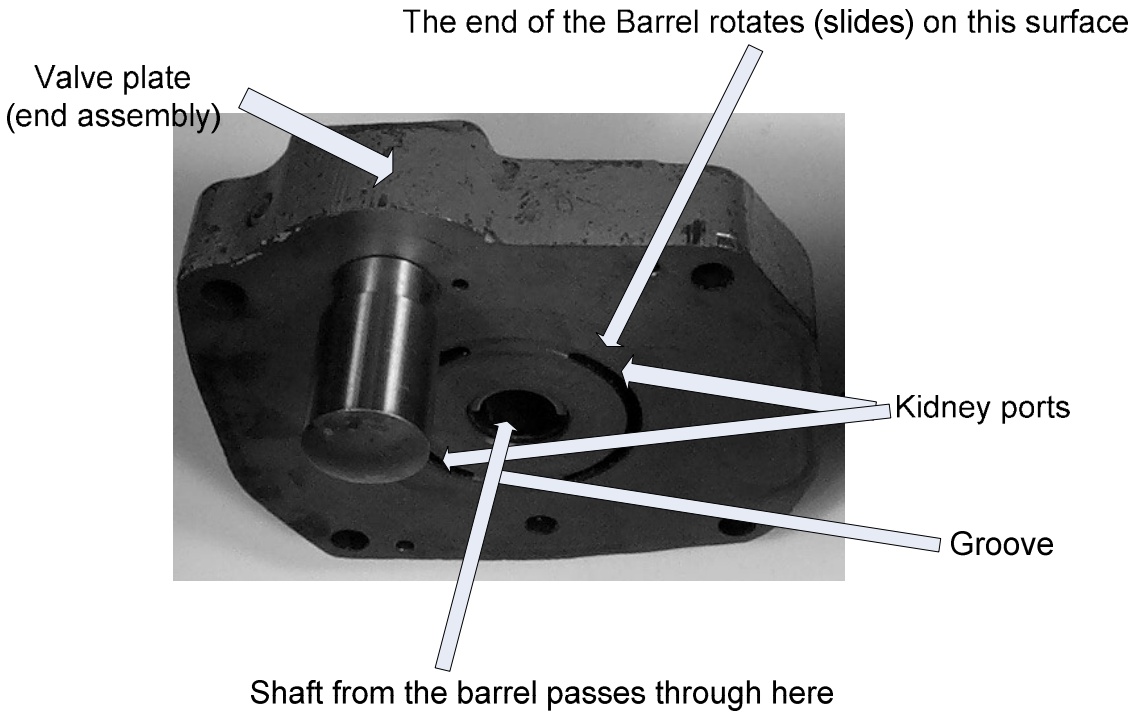
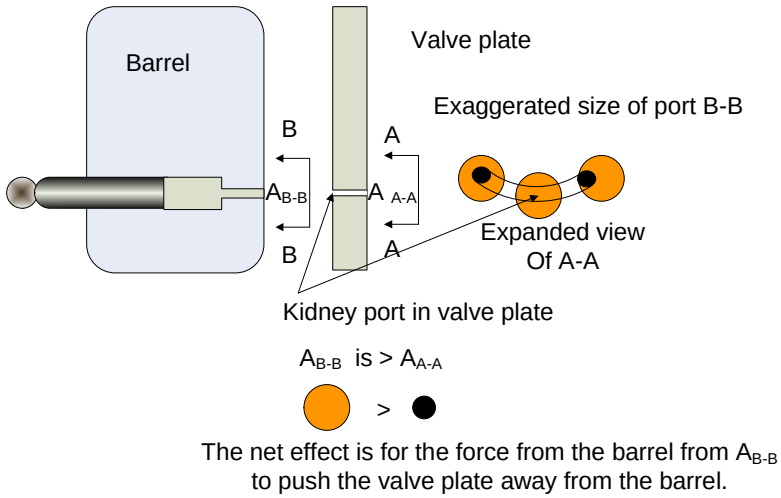


Figure 4.2.18 Valve plate (End assembly)

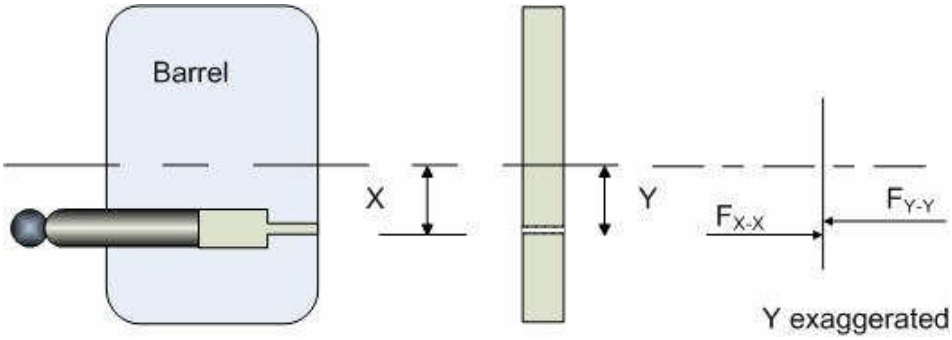
Consider Figures 4.2 19 (a), (b) and (c). This represents a side view of the barrel and valve plate. As the pistons travel towards the end plate and as the barrel in which they are houses rotates about the shaft, a separation torque exists which tries to move the valve plate and barrel apart. This is a consequence of the piston diameter being just slightly larger than the diameter of the kidney port.

Pressure which acts on $A_{B\text{-}B}$ is larger that on kidney port area $A_{A\text{-}A}$. Further, the Centre line of the $A_{B\text{-}B}$ does not exactly line up with the Centre line of $A_{A\text{-}A}$. Thus a torque on the barrel and the valve plate exists which try to separate them as illustrated in Figure 4.2.19(c).



The net effect is for the force from the barrel from A_{B-B} to push the valve plate away from the barrel.

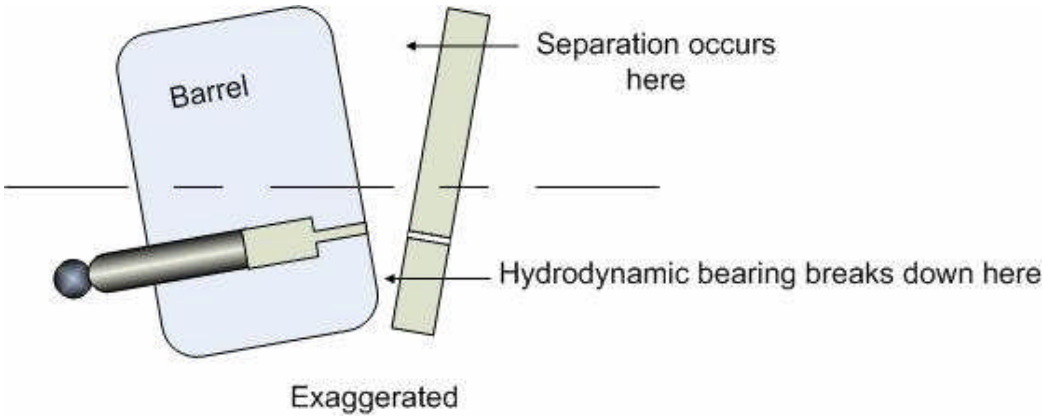
(a) Side view



(b) Nomenclature

Let x = effective point at which the pressure on the barrel acts
Let y = effective point at which the pressure from the kidney port acts

If $F_p \cdot x >> F_u y$, then twisting of the barrel occurs.



(c) Separation occurs

Figure 4.2.19 Bending torques on the barrel and valve plate

4.2.5.5 Variable displacement Axial-Piston Pumps & Motors

Consider again the axial piston in line pump of Figure 4.2.20

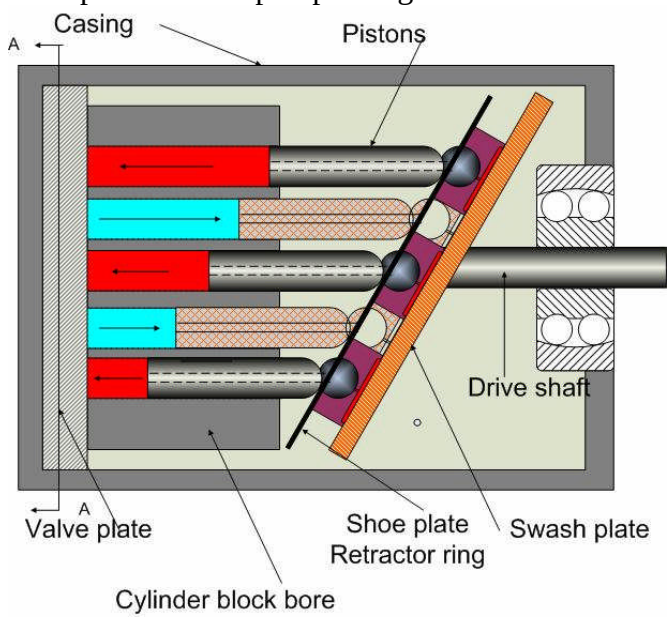


Figure 4.2.20 Axial piston pump

- The swash plate is installed in a movable yoke (not shown in Figure 4.2.20). "Pivoting" the yoke on pintles via the yoke change the swash plate angle to increase or decrease the piston stroke as illustrated in Figures 4.2.21 and 4.2.22 (where the yoke is shown)

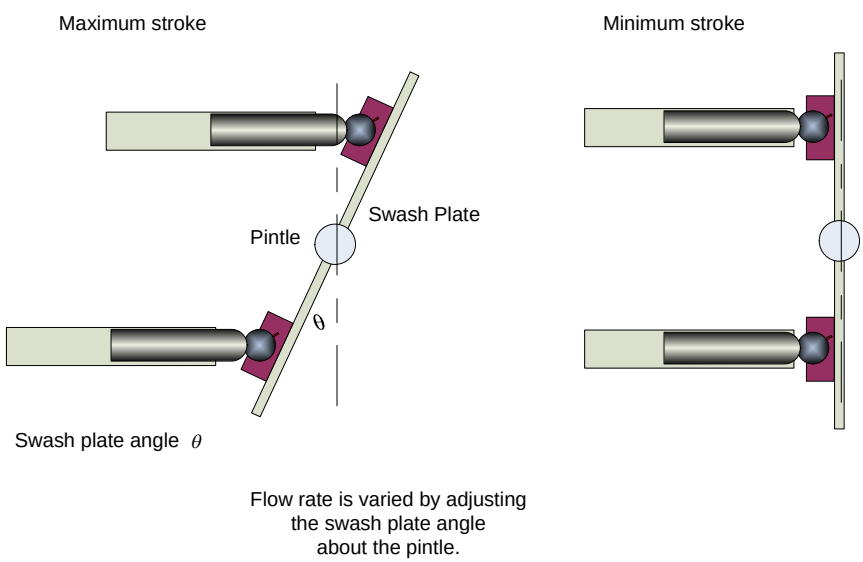


Figure 4.2.21 Variable Swashplate angle.

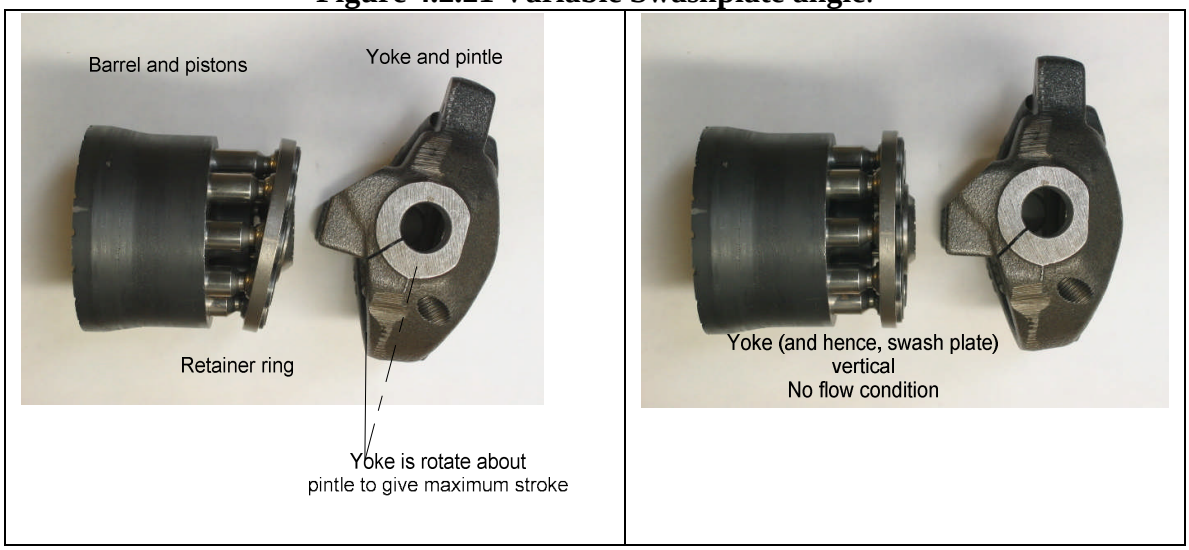


Figure 4.2.22 Side view of barrel, yoke and pistons.

4.2.5.6 Pressure compensated piston pump (in line)

Pressure compensation can also be applied to piston pumps. The essential components of the compensator are shown in Figure 4.2.23

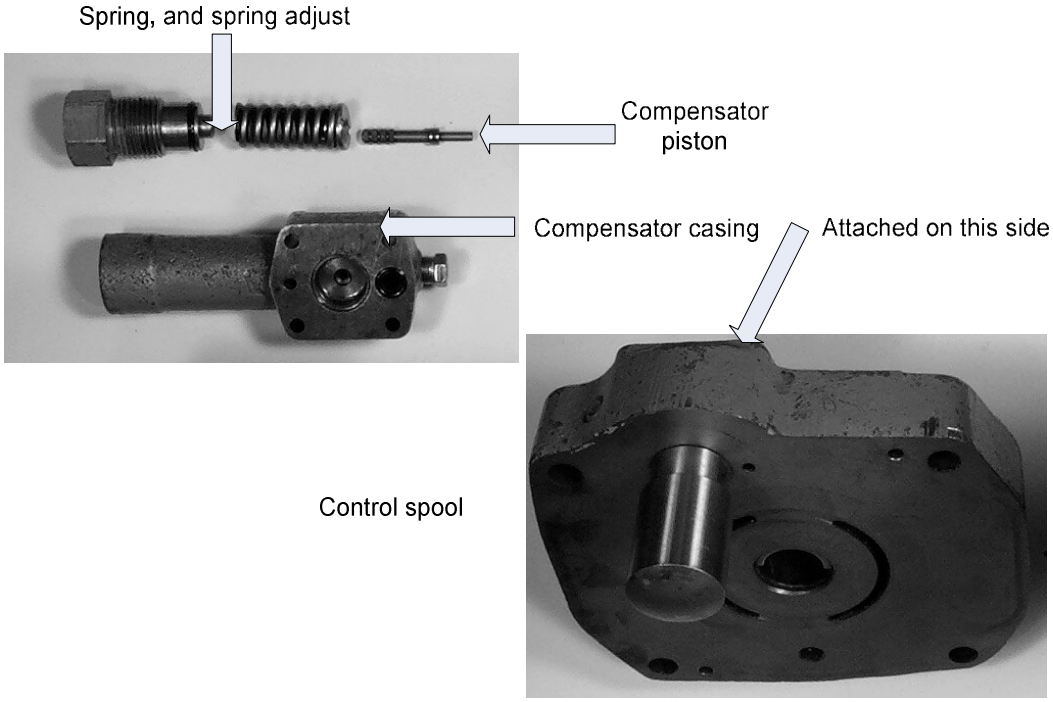


Figure 4.2.23 Components of a pressure compensator.

Consider Figure 4.2.23 and the schematic of its assembly in Figure 4.2.24

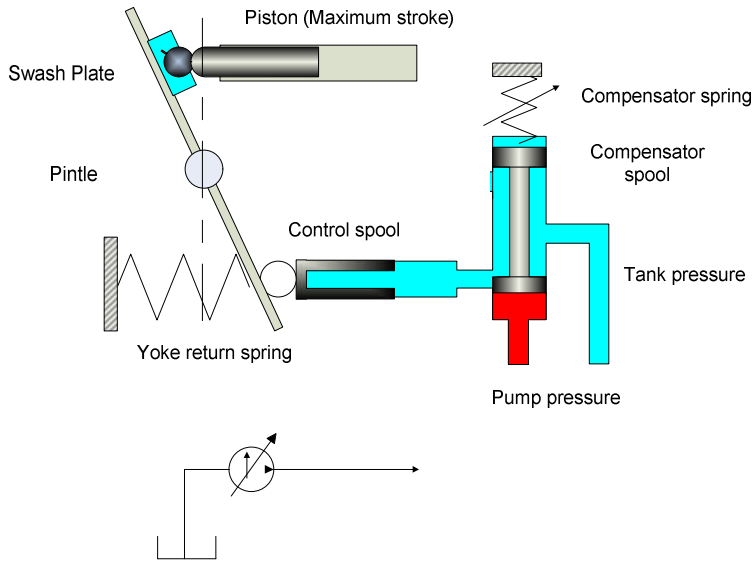


Figure 4.2.24 Schematic of a pressure compensator

For pump pressures less than $P_{set\ point}$ (which is set by the compensator spring manually), the compensator spool is located as shown in Figure 4.2.24. The control spool is at tank pressure and the yoke spring forces the control spool to its uttermost right position (and hence the swash plate is rotated to its maximum angle as shown. The pump acts as a fixed displacement pump for all pressures less than $P_{set\ point}$. When the system pressure builds up to the $P_{set\ point}$ (of the compensator spring), the spool moves upwards as illustrated in Figure 4.2.25. High pressure fluid is now ported to the control piston which pushes the swash plate and yoke assembly about the pintle to their vertical position. At this point, the swash plate angle is zero and minimum flow to the circuit exists.

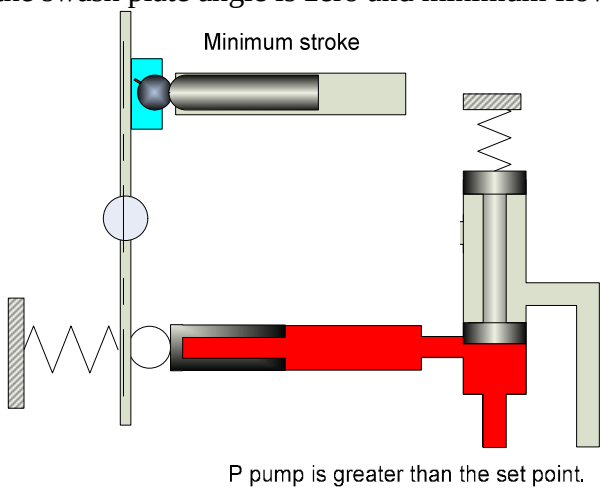
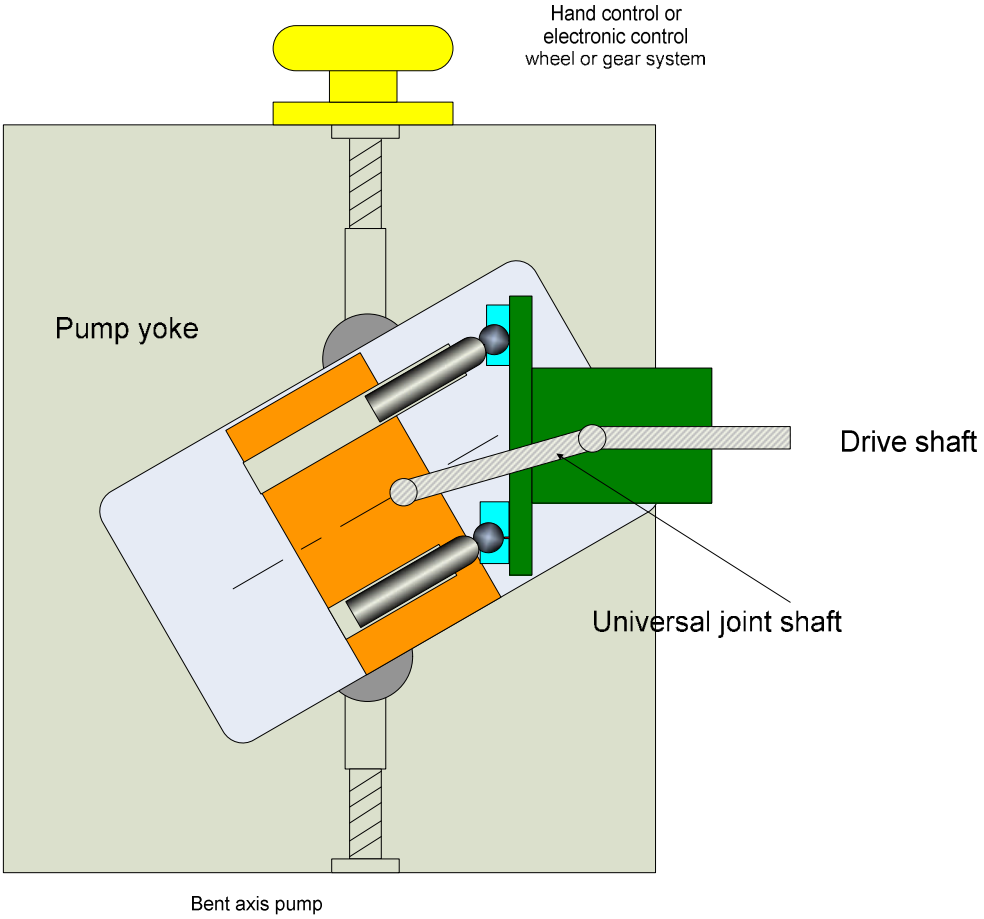


Figure 2.4.25 Minimum swash plate angle.

4.2.5.7 Bent Axis fixed displacement pumps (Figure 4.2.26)

- Bent axis pumps/motors have their piston axis at an angle with respect to the shaft
- Good for higher flows, pressures and speeds
- Transmission of torques (forces) better suited
- The operation is the same as the axial displacement pump/motor except torque is transmitted to the main shaft via a universal joint shaft
- The piston barrel rotates inside a yoke. If the yoke is rotated as shown in Figure 4.2.27, then a variable stroke is possible.
- The cylinder block (barrel) turns with the drive shaft but at an offset angle.



4.2.26 Bent Axis variable displacement pumps and motors

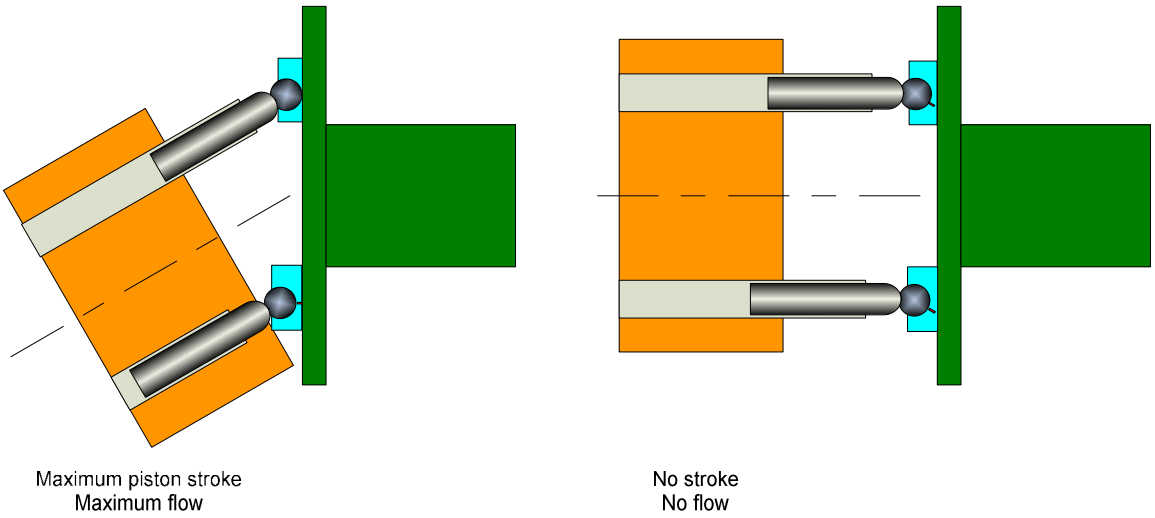


Figure 4.2.27 Variable displacement (stroke) bent axis pump/motor

4.3 Pump and Motor Analysis

In this section, the basic characteristics and describing equations are discussed. These equations and definitions will provide the basis for pump (motor) selection and for component and systems analysis.

4.3.1 Definitions:

(a) Displacement (D_m, D_p) is the amount of fluid that a motor (pump) will accept (deliver) in turning one revolution - i.e., capacity of one chamber x no. of chambers. in^3/rev ; cm^3/rev; m^3/rev

(b) Torque Rating of a Motor - Torque generated/690kPa (100 psi) of pressure

e.g. If a load is 50 Nm at an operating pressure of 13.8MPa (13800kPa) the rating of the motor would be

$$\text{T.R.} = \frac{50 * 690}{13800} = 2.5 \text{ Nm/690kPa}$$

(c) Ideal Pump Flow (Delivery) (U.S. gpm)

Pump: $Q_p = D_p \dot{\theta}_p$

Motor: $Q_m = D_m \dot{\theta}_m$

As a specials case, in the English system, displacement is often given in ($\frac{in^3}{rev}$). If the speed is specified in rpm, the flow in US gals per minute (gpm) is given by:

$$Q(\text{gpm US}) = \frac{\text{Displacement}}{231} \left(\frac{in^3}{rev}\right) . \text{ Speed (rpm)}$$

$$= \frac{D_p \dot{\theta}_p}{231} \text{gpmUS}$$

(d) Pump Rating - Max operating pressure capacity and their output (lpm or gpm) at a given speed.

(e) <u>Pressure Rating</u> - Pressure at which reasonable life expectancy can be obtained under specified operating conditions - if operation conditions exceed this rating, excessive wear may occur.

(f) <u>Pump (Motor) Characteristics</u> - A graphical display of pump delivery, volumetric efficiency, mechanical efficiency, input horsepower, etc., as function of outlet (load) pressure.

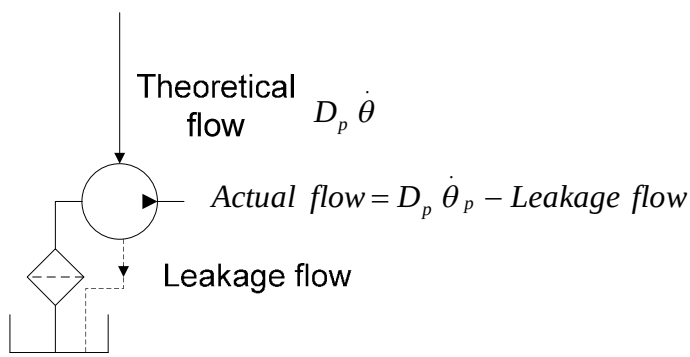
(g) <u>Volumetric Efficiency</u>

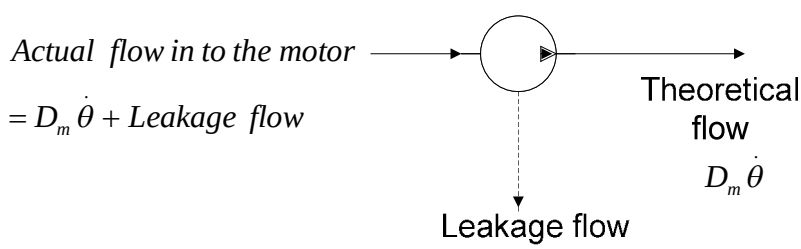
$$\eta_{vp} = \frac{\text{Actual Flow}}{\text{Ideal Flow}} \quad \text{(Pump) :}$$

Actual flow is less than ideal because fluid is lost due to leakage in the seals and piston clearances etc.

$$\eta_{vm} = \frac{\text{Ideal Flow}}{\text{Actual Flow}} \quad \text{(Motor) :}$$

Actual flow is greater than ideal because to make the ideal flow in a motor, some "extra fluid " must be added to compensate for leakage.





(h) Mechanical Efficiency

$$\eta_{mp} = \frac{\text{Ideal Torque Required}}{\text{Actual Torque Required}} \quad \text{(Pump)}$$

Actual Torque required = Ideal Torque + Losses

$$\eta_{mm} = \frac{\text{Actual Torque Output}}{\text{Ideal Torque Output}} \quad \text{(Motor)}$$

Ideal Torque = Actual Torque + Losses

(i) Overall Efficiency

$$\eta_{op} = \frac{Power\ out}{Power\ in} \quad \text{for both pumps and motors}$$

= Volumetric efficiency * mechanical efficiency (to be shown presently)

(j) Hydraulic Power

Hydraulic power to a **motor** is defined as the product of the differential pressure across the motor times the average motor flow $(P_m \cdot Q_m)$.
Hydraulic power from a **pump** is defined as the product of the output port pressure (inlet is a tank pressure and is neglected) times the average motor flow $(P_p \cdot Q_p)$

Mechanical power of a motor is the product of actual torque and the shaft speed (in rad/sec) $\dot{\theta}_m T_L$ where T_L actual torque.

4.3.2 Ideal pump/motor analysis (Merritt)

The objective of the next two sections is to develop basic equations for the efficiency (mechanical and volumetric) first for an ideal pump/motor and then for a "practical" pump/motor. These sections are EXTREMELY important in gaining insight into why pump and motor characteristics appear in the manufactures' specification, the way they do.

For the ideal case, there are no losses, (100% efficient). Although pumps and motors are not 100% efficient, this analysis is useful for initial pump and motor considerations. Quite often we use these equations first to understand what circuit or component efficiencies might be and then modify them to reflect the real situation.

Consider an ideal **motor**.	Consider an ideal **pump**
Since we assume the motor to be 100% efficient, $H.P_{out} = H.P_{in}$ $H.P_{in}$ = hydraulic H.P = $P_m \cdot Q_m$ *(4.1)* $H.P_{out}$ = mechanical H.P = $T_m \cdot \dot{\theta}_m$ *(4.2)* (T_m in Nm or in lb_f, $\dot{\theta}_m$ in rad/min). From equations 4.1 and 4.2, $P_m \cdot Q_m = T_m \dot{\theta}_m$ From which $T_m = \dfrac{P_m \dot{\theta}_m}{Q_m}$ (4.3) *Now,* $Q_m = D_m \cdot \dot{\theta}_m$, consequently equation 4.3 becomes $T_m = P_m \cdot D_m$ Units T_m Nm in lb, P_m N/m^2 l b/in^2 D_m m^3/rad in^3/rad	Since we assume the motor to be 100% efficient, $H.P_{out} = H.P_{in}$ For a pump, the input torque would be $T_p = \dfrac{P_p Q_p}{D_p}$ $T_p = P_p \cdot D_p$

∴ Only one motor or pump parameter need be specified to define the torque capabilities of an ideal motor - that is D_m - motor displacement or D_p – pump displacement Motor sizes are designated by the ideal theoretical displacement.

Pump sizes are usually designated by the flow obtained at a certain shaft speed.

4.3.3 Practical pump/motor analysis

To demonstrate how losses affect the performance of pumps and motors, a particular example shall be considered - that of a piston pump shown below. The analysis for a motor is essentially the same and so we will not repeat it here.
This analysis will be very important in gaining insight into the characteristics of a motor (pump). It is not an academic exercise as we hope you will soon appreciate.

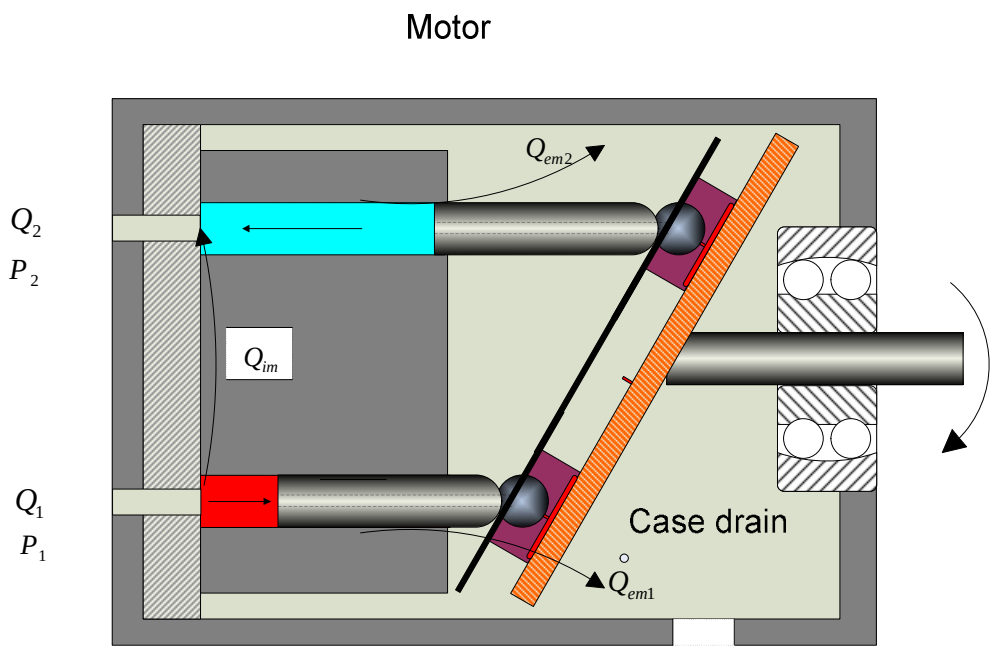


Figure 4.3.1 Schematic of a piston motor and nomenclature

Let	C_{im}	=	internal or cross-port leakage coefficient	m^3/(secPa)	in^3/sec/psi
	P_m	=	differential pressure across the motor	Pa	psi
	C_{em}	=	external leakage coefficient	m^3/(secPa)	in^3/sec/psi
	P_i	=	pressure in chamber	Pa	psi

In this example, the pistons and piston cylinder rotate about the stationary cam plate. Fluid porting is accomplished by the valving plate.

OBJECT: Determine the volumetric, mechanical and overall efficiency of a practical motor.

ASSUMPTIONS: Clearances between all parts are small such that leakage flow is laminar. This means that $Q \propto P_m$

- Drain case pressure negligible,

- Neglect compressibility
- P_2 is assumed at tank pressure and is small compared to the motor inlet or pump outlet value.

4.3.3.1 Flow Considerations

For flow into chambers the continuity equations gives:

$$Q_1 - Q_{im} - Q_{eml} = D_m \dot{\theta}_m \qquad 4.4$$

$$D_m \dot{\theta}_m + Q_{im} - Q_{em2} = Q_2 \qquad 4.5$$

Defining $Q_m = \dfrac{Q_1 + Q_2}{2}$, as the average flow,

$$\frac{Q_1 + Q_2}{2} = \frac{1}{2} \left[2 D_m \dot{\theta}_m + 2 Q_{im} - Q_{em2} + Q_{em1} \right] \qquad 4.6$$

Now since leakage flows are assumed laminar,

$Q_{im} = C_{im} (P_1 - P_2)$ = leakage flow internal $\approx C_{im} P_1$ if P_2 is assumed small

$Q_{em1} = C_{em1} P_1$ = leakage flow external

$Q_{em2} = C_{em2} P_2$ = leakage flow external ≈ 0 if P_2 is assumed small

$\therefore$ Equation. 4.6 becomes

$$Q_m = D_m \dot{\theta}_m + C_{im} (P_1) + \frac{C_{em1} P_1}{2} \qquad P_2 \text{ neglected}$$

Also, since $\dfrac{C_{em1}}{2}$ is a constant, we can rewrite this as

$$Q_m = D_m \dot{\theta}_m + C_{im} P_1 + C'_{em} P_1 \qquad \text{where} \qquad C'_{em} = \frac{C_{em1}}{2}$$

or

$$Q_m = D_m \dot{\theta} + (C_{im} + C'_{em})\, P_1$$

For the more general case in which P_2 is not small nor negligible, and assuming $C_{em1} \approx C_{em2}$,

$$Q_m = D_m \dot{\theta} + (C_{im} + C'_{em})\,(P_1 - P_2) \qquad \text{General case}$$

Now defining the load pressure P_m as P_1 - P_2

$$Q_m = D_m \dot{\theta}_m + (C_{im} + C'_{em})\, P_m \qquad \text{General case} \qquad 4.7$$

$$Q_m = D_m \dot{\theta} + (C_{im} + C'_{em})\, P_1 \qquad P_2 \text{ neglected}$$

This, then, is the "general" flow equation into a motor which reflect internal and external leakage.

Before proceeding, we have a few more things to consider. A great deal of research on the leakage coefficients has been carried out.

The term $(C_{im} + C'_{em})\, P_m$ is called SLIP FLOW, Q_s. This term is used quite often and it represents the amount of fluid that is not converted to shaft rotation.

Since the slip flow is laminar, it has been shown to be proportional to the motor (pump) displacement and inversely proportional to the viscosity or

$$(C_{im} + C'_{em}) \approx \frac{C_s D_m}{\mu}$$

C_s is defined as **the leakage slip coefficient** and is determined experimentally. Thus

$$Q_s = \frac{C_s D_m}{\mu} P_1 \text{ or ,} \qquad Q_s = \frac{C_s D_m}{\mu}(P_m) \text{ for the general case}$$

$$Q_m = D_m \dot{\theta}_m + \frac{C_s D_m}{\mu} P_m \qquad \text{General case}$$

$$Q_m = D_m \dot{\theta} + \frac{C_s D_m}{\mu} P_1 \qquad P_2 \text{ neglected}$$

4.3.3.2 Torque Considerations

Consider the torques which act on a motor.

(a) Ideal Torque = $D_m (P_1 - P_2) = T_m$ However this torque is reduced by a viscous torque, friction torques, and seal friction torques.

(b) Viscous Damping Torque

- Arises because a torque is required to shear fluids in small clearance. Viscous Damping Torque is a function of the velocity:

$$\therefore T_d = \beta_m \dot{\theta}_m = C_d D_m \mu \dot{\theta}_m \qquad 4.8$$

where:	β_m = viscous friction coefficient	N-m sec	in lb_f sec.
	C_d = damping coefficient		
	μ = absolute viscosity	N sec/m^2	lb_f sec/in^2

This states that the viscous friction coefficient β_m is a function of the motor displacement, D_m, the fluid viscosity, μ , and a damping constant which is determined experimentally. This expression has been derived and verified experimentally (Merritt) and is a very convenient way to express the viscous friction for our purposes.

(c) Friction Torque

- Friction forces arise as a result of movement of the piston in its bore, loaded bearings, etc. An opposing friction torque exists which is proportional to the motor displacement D_m, and the pressures at the inlet and outlet ports (Merritt).

$$T_f = C_f D_m (P_1 + P_2) \qquad 4.9$$

In many cases, P_2 is at tank pressure and so we can rewrite this as

$$T_f = C_f (P_1) D_m \qquad 4.9(a)$$

where C_f is a dimensionless coefficient relating the friction - torque parameter to the ideal torque.

This torque is sometimes called Coulomb friction for a fixed pressure

- A large value of C_f indicates metal to metal contact, therefore serious wear.

(d) <u>Breakaway Torque</u> T_c

- A constant value due seal friction, etc. In fact this is called stiction and when combined with the friction torque (Coulomb), the friction characteristics are often referred to as slip-stick.

<u>Net Torque to the Load</u>

The net torque available to the load is equal to the ideal torque minus friction type torques for a motor,

The net torques required to drive the pump is the sum of the ideal toque and friction types torques.

T_{Net} *Applied to Load*	$= D_m (P_1 - P_2) - C_d \mu D_m \dot{\theta}_m - C_f D_m (P_1 + P_2) - T_c$	Motor
T_{Net} *Applied to Load*	$= D_m (P_1) - C_d \mu D_m \dot{\theta}_m - C_f D_m (P_1) - T_c$	Motor P_2 neglected
T_{Net} *To Drive Pump*	$= D_p (P_1 - P_2) + C_d \mu D_p \dot{\theta}_p + C_f D_p (P_1 + P_2) + T_c$	Pump
T_{Net} *To Drive Pump*	$= D_p (P_1) + C_d \mu D_p \dot{\theta}_p + C_f D_p (P_1) + T_c$	Pump P_2 neglected

4.10

4.3.4 <u>Volumetric Efficiency</u>

Our objective was to determine the efficiencies of the motor (pump). If we use the general relationship, we have a problem in that we see the term ($P_1 + P_2$) in the torque equations. In our flow general equation we have $P_1 - P_2$. Thus to make this simple, we shall consider the case where $P_2 = 0$. There is only a slight loss of generality because many applications have P_2 at tank pressure. We are looking only at trends here as these equations are far from being exact, anyways. So lets go for it: let $P_2 = 0$. Our two equations for a motor are

$$Q_m = D_m \dot{\theta} + \frac{C_s \; D_m}{\mu} P_1 \qquad\qquad T_{Net} = D_m (P_1) - C_d \mu D_m \dot{\theta}_m - C_f D_m (P_1) - T_c$$

From this information, the volumetric and mechanical efficiencies can be calculated.

Recall $\eta_{vm} = \dfrac{\text{ideal flow}}{\text{actual flow to motor}} = \dfrac{D_m \dot{\theta}_m}{Q_m}$

$$= \frac{D_m \dot{\theta}_m}{D_m \dot{\theta}_m + C_s \dfrac{P_1 D_m}{\mu}} = = \frac{1}{1 + \dfrac{C_s P_1}{\mu \dot{\theta}_m}}$$

For a motor

$$\eta_{vm} = \frac{1}{1 + \dfrac{C_s P_1}{\mu \dot{\theta}_m}} \qquad 4.16$$

Similarly for a pump

$$\eta_{vp} = \frac{1}{1 - \dfrac{C_s P_1}{\mu \dot{\theta}_m}} \qquad 4.17$$

4.3.5 Mechanical Efficiency

Recall $\eta_{tm} = \dfrac{\text{actual torque}}{\text{ideal torque}} = \dfrac{D_m P_1 - C_d \mu D_m \dot{\theta}_m - C_f D_m P_1 - T_c}{D_m P_1}$

For a motor:

$$\eta_{tm} = 1 - C_d \mu \frac{\dot{\theta}_m}{P_1} - C_f - \frac{T_c}{D_m P_1} \qquad 4.18$$

Similarly, for a pump

$$\eta_{tp} = \frac{1}{1 + C_d \frac{\mu \dot{\theta}_p}{P_1} + C_f + \frac{T_c}{D_m P_1}} \qquad 4.19$$

4.3.6 Overall efficiency

The overall efficiency η_{om} or η_{op} is defined as the product of the mechanical efficiency times the volumetric efficiency. (See problem at the end of this section). Therefore

For a motor

$$\eta_{om} = \eta_{vm}\, \eta_{tm} = \frac{1 - \frac{C_d \mu \dot{\theta}_m}{P_1} - C_f - \frac{T_c}{D_m P_1}}{1 + \frac{C_s P_1}{\mu \dot{\theta}_m}} \qquad 4.20$$

Similarly for a pump

$$\eta_{op} = \eta_{vp}\, \eta_{tp} = \frac{1 - \frac{C_s P_1}{\mu \dot{\theta}_p}}{1 + \frac{C_d \mu \dot{\theta}_p}{P_1} + C_f + \frac{T_c}{D_m P_1}} \qquad 4.21$$

(1) Generally, T_c is small when compared to the other terms and can be neglected. In fact T_c occurs only at low velocities and so when the motor or pump is in operation, it is zero, So this is not a limiting assumption

2) The quantity $\frac{\mu \dot{\theta}_m}{P_1}$ is dimensionless. This is very important because it facilitates interpretation.

(3) If we can measure C_s, C_d, C_f, then the static performance with $P_2 = 0$ can be defined as a function of $\frac{\mu \dot{\theta}_m}{P_1}$

Typical Efficiency Curves for a motor (pump) are illustrated in Figure 4.3.2:

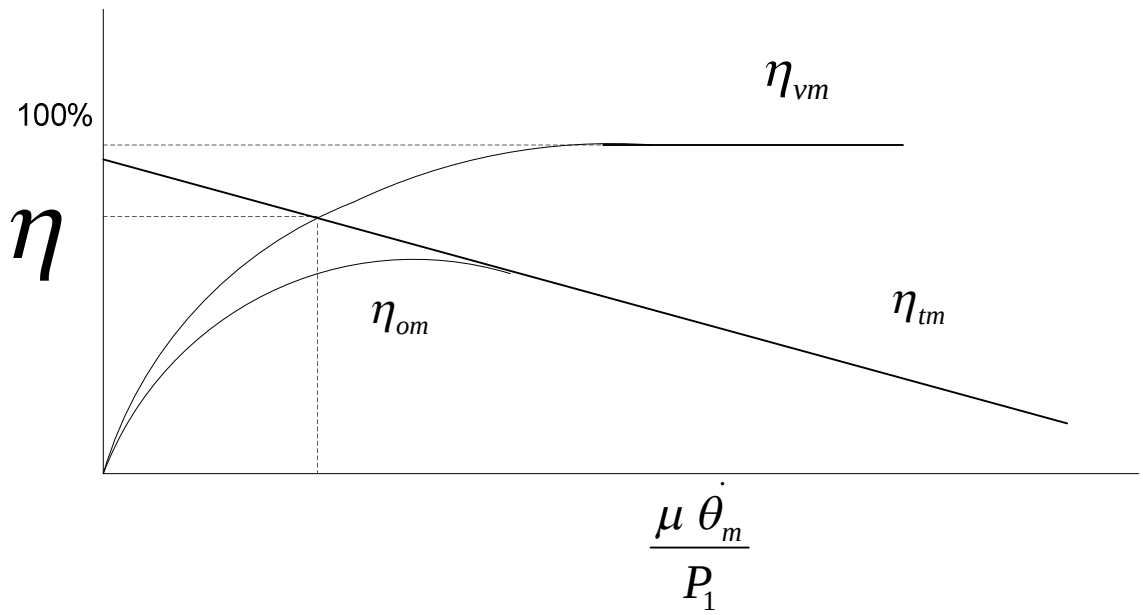


Figure 4.3.2 Efficiency Curves

Now, if the speed or angular velocity of shaft is fixed, and P_1 varied, the efficiencies vary: At low P_1, the volumetric efficiency is large, the mechanical efficiency is small, hence, the overall efficiency is small. At high P_1, η_{Vm} is low, η_{Tm} large, and once again η_{om} is small.

The significance of this curve is immense. Yes, it is an approximation but in reality, manufactures' curves do follow these trends. It also says that there is an "optimal" range that any pump or motor should operate in and that we cannot expect the same performance if conditions vary widely.

4.4 Analysis of a variable displacement pump and motor

4.4.1 Assumptions:

(1) Volumetric & mechanical efficiencies = 100%
(2) No leakage in circuit
(3) Ignore dynamic effects

Define

$\dot{\theta}_{.p}$ = pump speed (rpm)
D_p = pump displacement

$\dot{\theta}_{.m}$ = motor speed (rpm)
D_m = motor displacement

Circuit

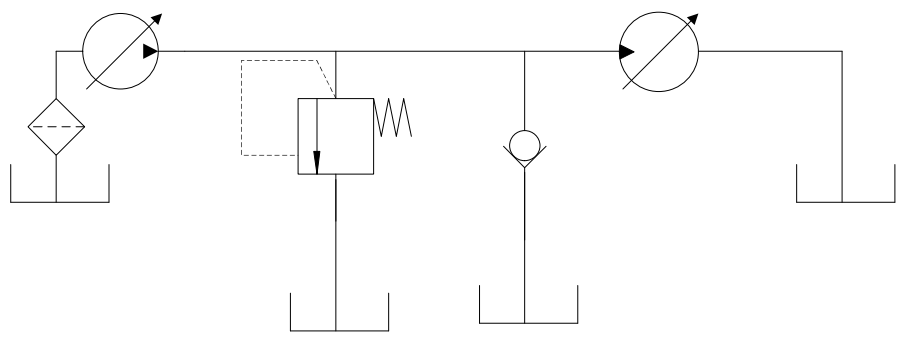


Figure 4.4.1 Basic circuit

Basic Equations:

Pump:

Q_p = pump flow = $D_p \dot{\theta}_p$

Motor

$Q_m = D_m \dot{\theta}_m$

By assumption 2,

$Q_p = Q_m$

$$D_p \dot{\theta}_p = D_m \dot{\theta}_m$$

$$\therefore \text{ Motor speed} = \dot{\theta}_m = \left(\frac{D_p}{D_m}\right) \dot{\theta}_p \qquad (4.22)$$

Torque on the hydraulic motor shaft =

$$T_m = \frac{D_m \Delta P_m}{2\pi} \qquad \text{(Over } 2\pi \text{ because } D_m \text{ is define per revolution)} \qquad (4.23)$$

Output power from the motor =

$$HP_m = Q_m \Delta P_m = D_m \dot{\theta}_m \Delta P_m = D_p \dot{\theta}_p \Delta P_p \qquad (4.24)$$

<u>Case (a)</u> Let us that the input speed $\dot{\theta}_p$ is fixed and we shall vary D_p (variable displacement pump)

Our output characteristics (using eq's 4.22,4.23 & 4.24) are

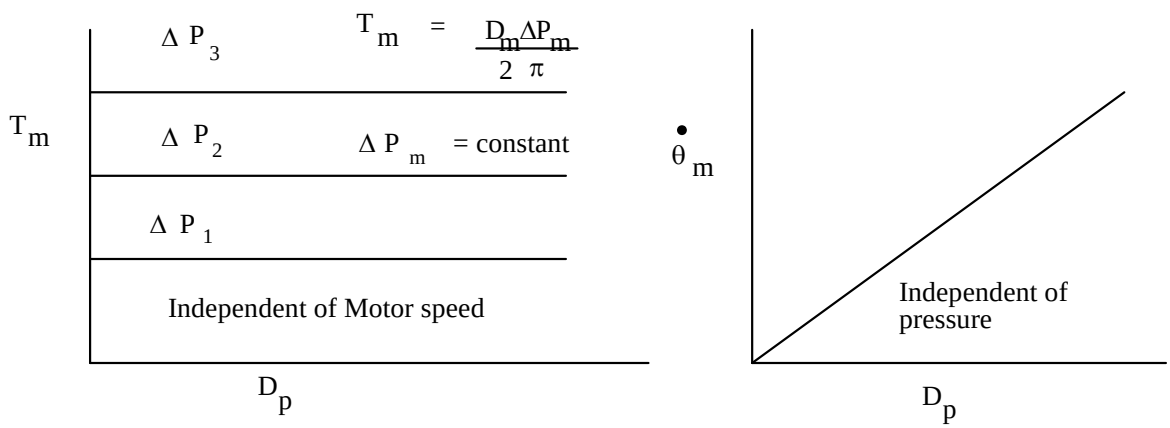


Figure 4.4.2 Characteristics: input speed $\dot{\theta}_p$ fixed, D_p varied

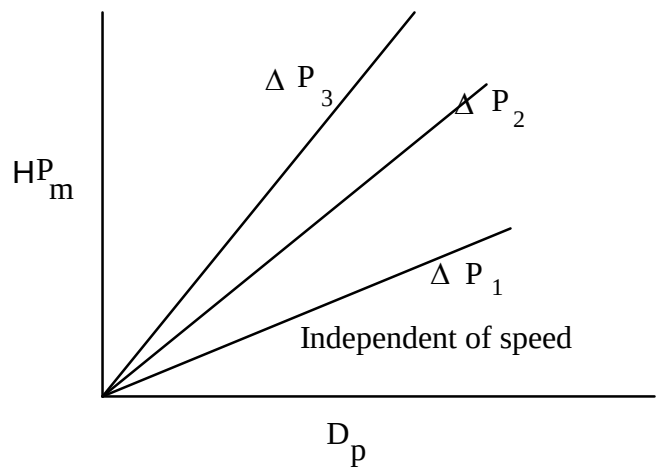


Figure 4.4.3 HP Characteristics: input speed $\dot{\theta}_p$ fixed, D_p varied

When D_m is fixed and D_p varies, this is often called a **"CONSTANT TORQUE TRANSMISSION"**

- This is true for both open and closed circuits (T.B.A.)

<u>Case (b)</u> Fixed displacement pump, fixed pump speed, <u>variable</u> motor displacement

Our output characteristics become (using eq's 1, 2 & 3)

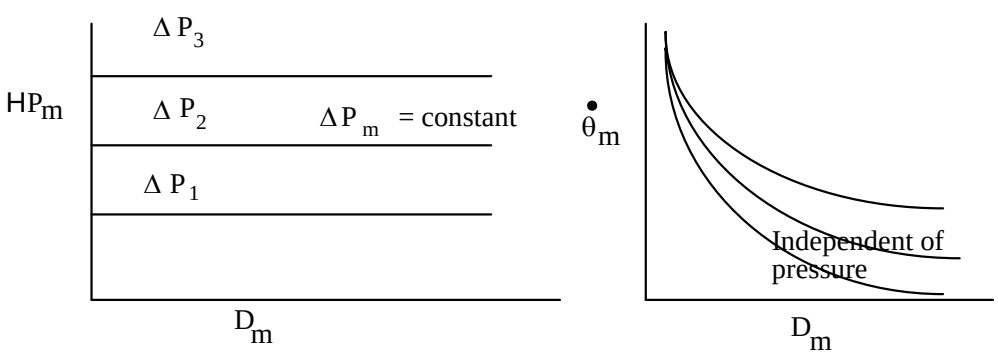


Figure 4.4.4 Characteristics: input speed $\dot{\theta}_p$ fixed, D_m varied

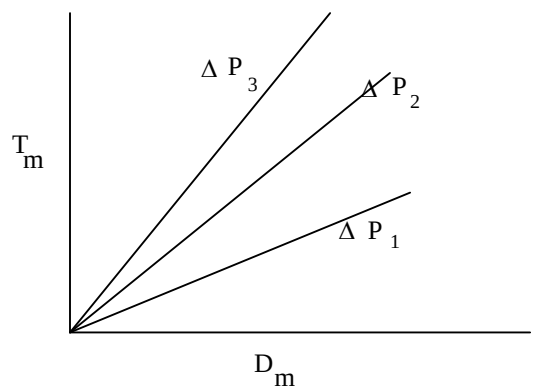


Figure 4.4.5 Torque characteristics: input speed $\dot{\theta}_p$ fixed, D_m varied

When D_p, $\dot{\theta}_p$ are fixed and D_m varies, this is often called a **"CONSTANT POWER TRANSMISSION"**