MOBILE HYDRAULICS MANUAL

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Foreword

Oil hydraulics today is an essential area of knowledge for anyone who has a technical interest in mobile machinery or any automotive-type vehicles. In farm tractors and implements, industrial trucks, earth-moving equipment, self-propelled vehicles of all kinds, we find applications of brute force with very precise control through hydraulic systems. Milady can park a two-ton automobile with only a slight effort at the steering wheel . . . the touch of a handle lifts several yards of dirt in the bucket of a loader . . . trenches are dug quickly without anyone lifting a shovel full of dirt . . . because the hidden giant hydraulics is there.

This manual has been prepared as an aid to basic training in mobile hydraulics. It is used as a text in the mobile training program at Sperry Vickers Hydraulics School. It also is available to others who are interested in this field of hydraulics but do not have the opportunity to attend the school; and it is designed to serve as a reference work for Sperry Vickers and customer personnel. In very simple language and with minimum use of mathematics, you are told the why's and how's of mobile hydraulics. The manual explains the fundamental principles of pressure and flow, describes the operation of basic hydraulic components, tells how these components are combined to do their many jobs, and explores the fundamental considerations of hydraulic equipment design, use and maintenance.

COLOR KEY

To make the manual easy to read and understand, it is liberally illustrated in color. All illustrations which require showing oil flow conditions or paths are prepared with industry-standardized color codes. Therefore, hydraulic lines and passages in the many diagrams are colored as follows:

Red	Operating Pressure
Blue	Exhaust
Green	Intake or Drain
Yellow	Metered Flow
Orange	Reduced Pressure or Pilot Pressure

Colors also are used to emphasize certain parts of diagrams. In this case, the colors will have no particular significance, and will not appear in the hydraulic lines.

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CHAPTER 1
Hydraulic Principles

Hydraulics, as we consider it in this manual, is the science of transmitting force and/or motion through the medium of a confined liquid. This is, in one way, a rather narrow scope . . . because in its broadest sense, hydraulics encompasses any study of fluids in motion. As such, it has its foundations from many thousands of years ago in ancient water works and irrigation systems. The name "hydraulics" comes from the Greek "hydros", which means "water".

Just prior to the Christian era, the mathematician Archimedes invented a device that would pump water. The Archimedes screw, a hollow tube formed into a screw, is still in use today in Europe for drainage systems. Near the same time as Archimedes, Hero of Alexandria actually built a turbine for harnessing the power of a moving liquid. However, the water wheel, a form of turbine, probably dates back as much as five thousand years in China and Egypt.

Yet the transmission of power through pressure in a confined fluid has been limited to comparatively recent times. Prior to the 15th century, when Leonardo da Vinci was the mental giant of Europe, the very concept of pressure was virtually unknown . . . the study of hydraulics was limited to the principles of flow. Even da Vinci, though he suggested several designs for hydraulic machinery, failed to develop a clear understanding of pressure.

More than a hundred years later, the Italian Evangelista Torricelli observed the principle of the mercury barometer and related it to the weight of the atmosphere. Building upon Torricelli's findings, the French scientist Blaise Pascal discovered the principle of hydraulic leverage known as Pascal's Law. It is from this Law that the entire science of pressure hydraulics has developed in a few hundred years. The first practical use of pressure hydraulics was in 1795, when Joseph Bramah developed the original hydraulic press . . . using water as the hydraulic medium, and using the principle of Pascal to achieve a huge mechanical advantage or force multiplication.

HYDRODYNAMICS VS. HYDROSTATICS

Today, there are many thousands of pressure-operated machines and they are so distinct from earlier devices we must divide hydraulics into two sciences – hydrodynamics and hydrostatics. Hydrodynamics can be called the science of moving liquids; hydrostatics, the science of liquids under pressure. A water wheel or turbine (Figure 1) represents a hydrodynamic device. Energy is transmitted by the impact of a moving fluid against blades or vanes. In other words, we are using the kinetic energy, or energy of motion, that the liquid contains.

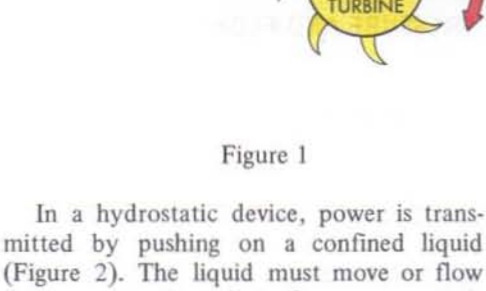


Figure 1

In a hydrostatic device, power is transmitted by pushing on a confined liquid (Figure 2). The liquid must move or flow to cause motion; but the movement is incidental to the force output. A transfer of energy takes place because a quantity of liquid is subject to pressure.

Most of the hydraulic machines in use today operate hydrostatically; that is, through pressure. Their study should technically be referred to as hydrostatics or pressure hydraulics. But the new science has so far dwarfed the old, that the name "Hydraulics" has stuck to the new.

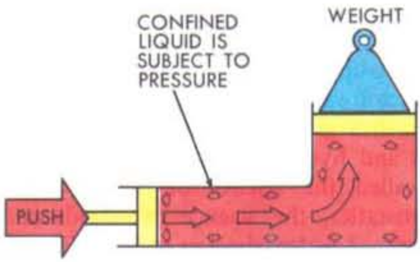


Figure 2

As we've said, this manual is limited to the pressure hydraulics branch . . . and particularly to the application of hydrostatic equipment for mobile machinery of all kinds. We will not "fly in the face of" tradition in these pages . . . we will use the terms "hydraulic" and "hydraulics" as is customary in the industry.

PRESSURE AND FLOW

In studying the basic principles of hydraulics, we will be concerned with forces, energy transfer, work and power. We will relate these to the two fundamental conditions or phenomena that we encounter in a hydraulic system. They are pressure and flow.

Pressure and flow, of course, must be inter-related in considering work, energy and power. On the other hand, each has its own particular job to do.

- o Pressure is responsible for pushing or exerting a force or torque.
- o Flow is responsible for making something move; for causing motion.

Because these jobs are often confused by the uninitiated, try to keep them distinct as we consider separately . . . and then together . . . the phenomena of pressure and flow.

PRESSURE PROVIDES THE PUSH [FUNDAMENTALS OF PRESSURE]

WHAT IS PRESSURE?

To the engineer, pressure is a term used to define how much force is exerted against a specific area. The technical definition of pressure, in fact, is force per unit area.

AN EXAMPLE OF PRESSURE

One example of pressure is the tendency to expand (or resistance to compression) that is present in a fluid which is being squeezed. A fluid, by definition, is any liquid or any gas (vapor).

The air that fills your automobile tires is a gas, and obeys the laws of fluids. When you inflate a tire, you are squeezing in more air than the tire would like to hold. The air inside the tire resists this squeezing by pushing outward on the casing of the tire. The outward push of the air is pressure.

Air, of course, like all gases, is highly compressible. That is, you can squeeze it into a smaller volume, or you can squeeze more air into the same space. As you squeeze additional air into a tire, it takes a greater force, and the pressure within the tire increases.

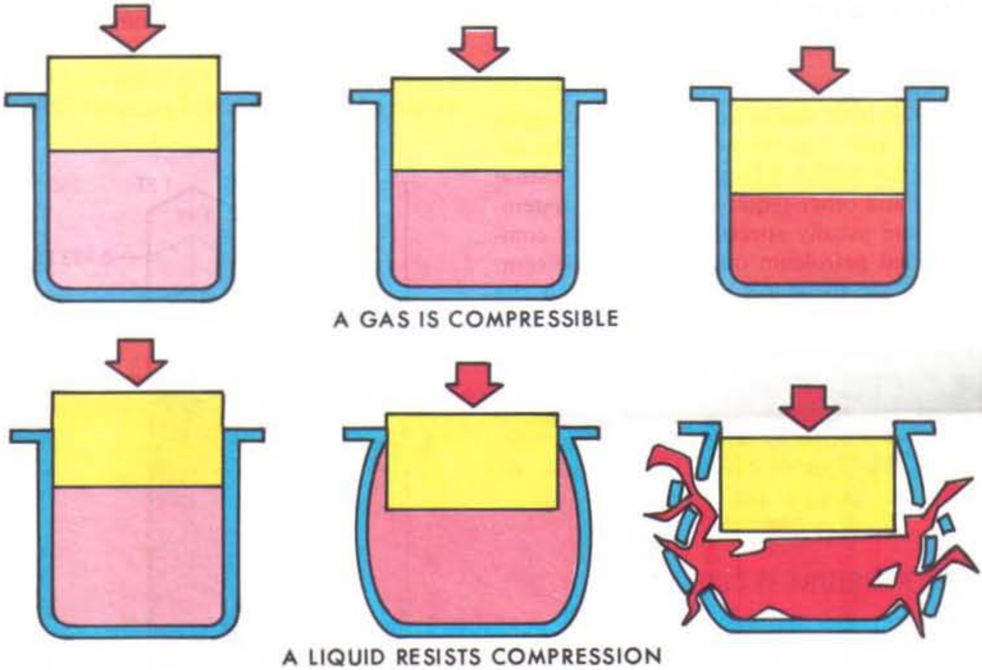


Figure 3

PRESSURE IS EQUAL THROUGHOUT

You know, too, that the outward push of the air in a tire is uniform throughout. That is, all the inner surface of the tire is subject to the same amount of pressure. If it weren't, the tire would be pushed into odd shapes because of its elasticity.

Equal pressure throughout the area of confinement is a characteristic of any pressurized fluid, whether gas or liquid. The difference is that liquids are only very slightly compressible.

PRESSURE IN A CONFINED LIQUID

If you have ever tried to force a stopper into a bottle that was completely full of water, you have experienced the near-incompressibility of a liquid. Each time you tried to push the stopper in, it sprang back immediately when you let it go. And if you were so bold as to hit the stopper with a hammer, chances are you broke the bottle.

When a confined liquid is pushed on, there is a pressure build-up (Figure 3). The pressure is still transmitted equally throughout the container. The bottle that breaks from excess pressure may break anywhere; or in several places at once.

This behavior of a fluid is what makes it possible to transmit a push through pipes, around corners, up and down, and so on. In hydraulic systems, we use a liquid, because its near-incompressibility makes the action instantaneous, so long as the system is primed or full of liquid.

ANOTHER DEFINITION OF FLUID

We have defined a fluid as any liquid or gas. You have also heard the term "hydraulic fluid" and you know that all hydraulic fluids are liquids. (To satisfy the lubrication needs and other requirements in the system, they are usually specially refined and compounded petroleum oils.) Because the term "fluid" is so widely used to refer to the hydraulic liquid, we will use it that way in this manual. So when we refer to "the fluid" in a system, you will understand that we mean the hydraulic liquid being used to transmit force and motion. When we refer to "a fluid" in describing natural laws, we mean any liquid or gas.

HOW PRESSURE IS CREATED

It is fundamental that pressure can be created by squeezing or pushing on a confined fluid only if there is a resistance to flow. There are two ways to push on a fluid . . . either by the action of some sort of mechanical pump or by the force exerted by the weight of the fluid itself.

THE WEIGHT OF A FLUID

You know that a diver cannot go to great depths in the ocean because of the tremendous pressure. This pressure is due entirely to the force exerted by the weight of the water above the diver; and it increases in proportion to the depth. Knowing the force exerted by the weight of a cubic foot of water, we can calculate the pressure at any depth exactly.

Suppose, as shown in Figure 4, we isolate a column of water one-foot (0.3048 m) square and ten feet (3.048 m) high. We want to determine what the pressure is at the bottom of this column. Since the force exerted by the weight of a cubic foot ($0.028\ m^3$) of water is 62.4 pounds (277.6 N) and we have 10 cubic feet ($0.283\ m^3$) in this example, the total force exerted by the weight of the water is 624 pounds (2776 N).

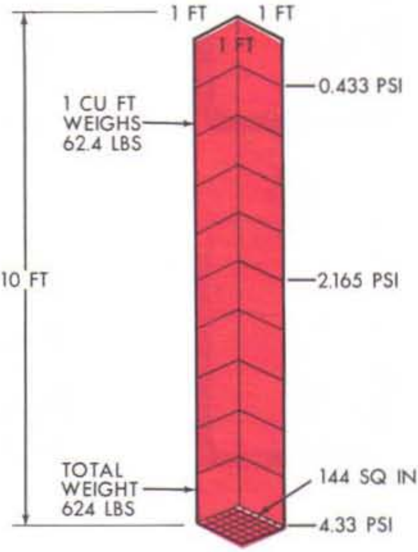


Figure 4

At the bottom, this force is distributed over 144 square inches ($0.093\ m^2$) (the equivalent of one square foot). Each single square inch ($0.000645\ m^2$) of the bottom is subject to 1/144 of the total force, or 4.33 pounds (19.28 N). We say, then, that the pressure at this depth is 4.33 pounds per square inch ($19.28\ N/0.000645\ m^2 = 29891\ N/m^2 = 0.299$ bar).

Pounds-per-square-inch (bar) is the common unit of pressure. We abbreviate it psi (bar). It tells us how many pounds (1×10^5 N) are exerted on a unit area of one square inch (one square metre).

OTHER WAYS OF CREATING PRESSURE

We could just as easily create an equal pressure of 4.33 psi (0.299 bar) in a liquid as shown in Figure 5. If we trap the liquid under a piston which has an area of 10 square inches (0.00645 m^2), and place a weight on the piston so that it pushes down with 43.3 pounds (192.8 N) of force, we again get a pressure of 4.33 psi (0.299 bar).

$$\frac{43.3 \text{ pounds}}{10 \text{ square in.}} = \frac{4.33 \text{ pounds}}{\text{sq. in.}} \text{ or } 4.33 \text{ psi}$$

$$\left(\frac{192.8 \text{ N}}{0.00645 \text{ m}^2} = \frac{29891 \text{ N}}{\text{m}^2} = 0.299 \text{ bar}\right)$$

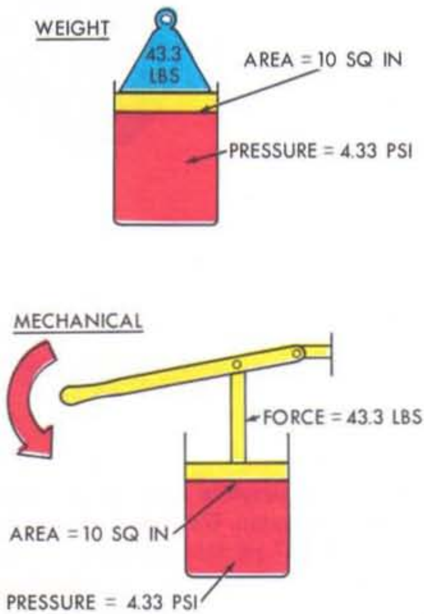


Figure 5

FORCE IS ANY PUSH OR PULL

Of course, it is not necessary to push downward with force exerted by the weight to create pressure in a fluid. It is only necessary to apply any kind of force. We define force as any push or pull. Weight is only one kind of force – the force of gravity on an object. We could just as easily turn the container in Figure 5 on its side and push on the piston with a spring or a strong right arm, or a crankshaft driven by an engine. In any case, we measure the applied force in pounds (Newtons). And pressure is created in the liquid in proportion to the force.

A HEAD OF LIQUID

You have likely heard the expression "work up a head of steam". This, of course, refers to building up pressure in a steam boiler.

Before Pascal clarified the concept of pressure, "head" was the only way pressure could be measured. Torricelli proved, for instance, that if you punch a hole in the bottom of a tank, the water runs out faster when the tank is full, and slows down as the level lowers.

We know now that this occurs because pressure at the bottom is highest when the tank is full, but decreases as the water level goes down. Torricelli knew only that there was a difference in "head" or in the height of the column of water. Head thus was a measure of pressure, but it could be expressed only as "feet (metres) of water".

Today, we still define a head as the vertical distance between two levels in a fluid. But now we can just as easily express it in psi (bar), as in feet (metres) of water.

In Figure 4, the head between the top and bottom of the water is 10 feet (3.048 m). A 10-foot (3.048 m) head of water is equivalent to 4.33 psi (0.299 bar). Each foot (0.3048 m) of water thus is equivalent to 0.433 psi (0.0299 bar). A five foot (1.524 m) head would be 2.165 psi (0.149 bar) and so on.

Oil, being slightly lighter than water, creates slightly less pressure from the force exerted by its weight. A foot (0.3048 m) of oil is equivalent to approximately 0.4 psi (0.028 bar); ten feet (3.048 m) to 4.0 psi (0.28 bar); and so on. (It should be noted that, in a hydraulic system, the relatively minor pressure difference due to elevation of the fluid is usually ignored . . . except in the case of the pump inlet condition.)

When we express pressure as feet (metre) of head, we must specify the liquid to be able to convert feet (metre) of head to psi (bar).

The term "head" has come to mean any pressure condition regardless how it is created. It is not incorrect to use the terms "pressure" and "head" interchangeably.

ATMOSPHERIC PRESSURE

In today's space age, everyone is familiar with the fact that the earth has an atmosphere of air extending some 50 miles (80.5 km) up; and we know that the air exerts a force because of its weight. Thus, the air above us must create a head of pressure on the earth's surface . . . and it does. We call this pressure created by the earth's atmosphere, appropriately enough, atmospheric pressure.

The force exerted by the weight of a column of air one square inch (0.000645 m^2) in cross-section and the height of the atmosphere (Figure 6), is 14.7 pounds (65.4 N) at sea level. Thus, everything on the face of the earth is subject to 14.7 psi (101300 N/m^2 = 1.01 bar) in normal conditions. Atmospheric pressure, then, is normally 14.7 psi (1.01 bar) absolute (psia). (In the mountains, atmospheric pressure is less because the head of air is less. Below sea level, the pressure is higher because the head is greater.)

Any pressure condition less than atmospheric pressure is referred to as a vacuum or partial vacuum.

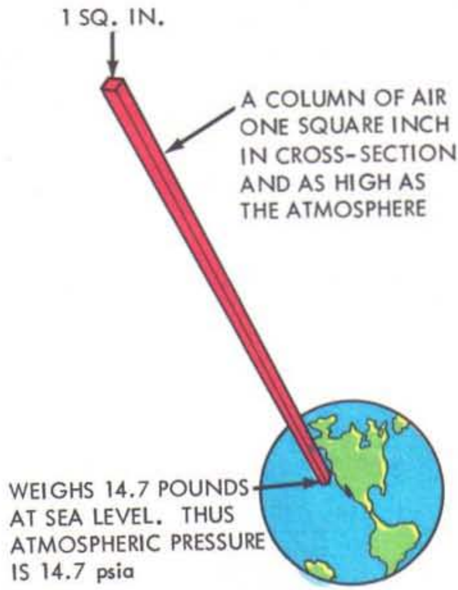


Figure 6

ABSOLUTE AND GAUGE PRESSURE

Absolute pressure is a scale with its zero point at the complete absence of pressure, or a perfect vacuum. We refer to atmospheric pressure as 14.7 psi absolute (psia) (1.01 bar absolute) to distinguish it from gauge pressure. Gauge pressure ignores atmospheric pressure. A pressure gauge reads zero when exposed to the atmosphere.

Thus: Gauge Pressure Plus 14.7 (1.01 bar) = Absolute Pressure

Absolute Pressure Minus 14.7 (1.01 bar) = Gauge Pressure

Gauge pressure may be abbreviated psig; but usually is abbreviated simply psi. We always designate absolute pressure as psia.

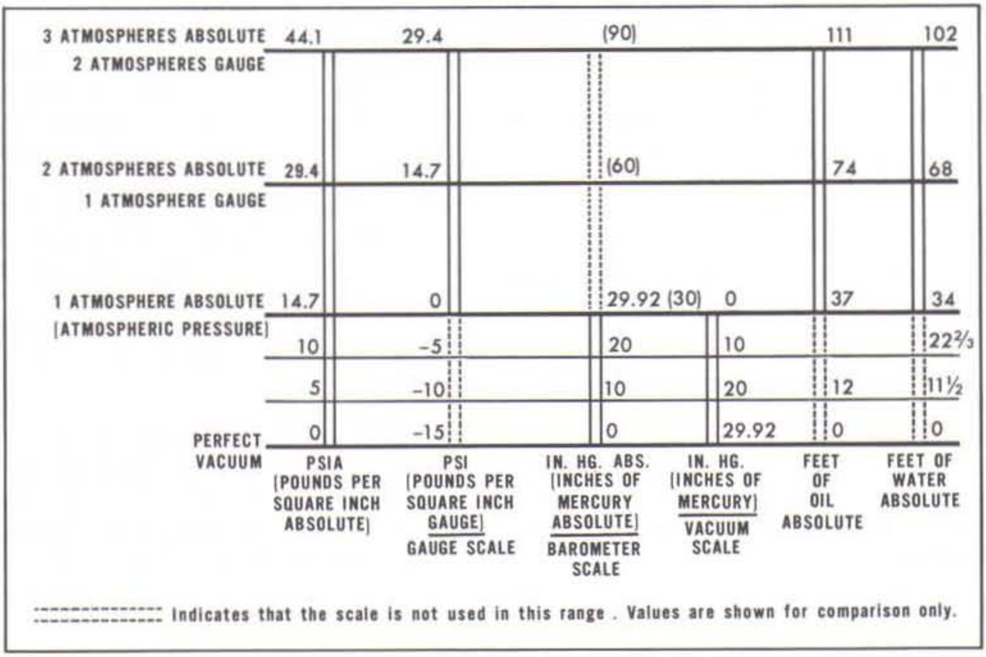


Figure 6A. Pressure and Vacuum Scales

PRESSURE AND VACUUM SCALES

We have mentioned psi (bar), feet (m) of water and feet (m) of oil as different ways of measuring pressure. We have two other common pressure units – the atmosphere and the inch of Mercury (in. Hg) (Figure 6A).

One atmosphere is simply the equivalent of atmospheric pressure. It can be either absolute or gauge. Thus 14.7 psia (1.01 bar absolute) is one atmosphere; 29.4 psia (2.02 bar absolute) is two atmospheres; 44.1 psia (3.03 bar absolute) is three atmospheres; and so on.

Inches of Mercury is taken from Torricelli's Mercury barometer (Figure 7). He found that when a tube full of Mercury is inverted in a pan of the liquid, the column falls until atmospheric pressure on the surface is balanced against the vacuum above

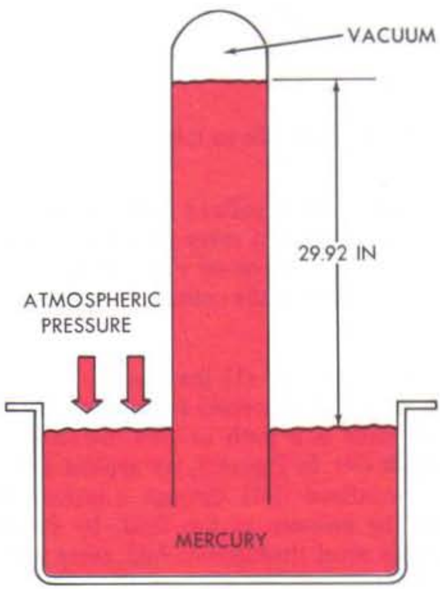


Figure 7

the liquid in the tube. In a normal atmosphere, the column is supported 29.92 inches high by a vacuum. The barometer scale, inverted, has become the standard scale for measuring vacuum. Thus, atmospheric pressure is zero vacuum and a vacuum is 29.92 in. Hg (1.01 bar vacuum).

Notice by comparing scales that one psi is equivalent to about two in. Hg.

An interesting comparison of the weights of water and Mercury is evident when comparing their scales. An inch of Mercury is equal to more than a foot of water. A perfect vacuum will support a column of water 34 feet (10.4 m) at atmospheric pressure.

Now that we have examined the phenomenon of pressure and how it is measured, we can go on to see how it behaves in a hydraulic circuit.

PASCAL'S LAW

Pascal's Law tells us this:

Pressure on a confined fluid is transmitted undiminished in every direction, and acts with equal force on equal areas, and at right angles to the container walls.

We know now (1) that pressure is force per unit area, expressed as psi (bar) and (2) that force is a push or pull, measured in pounds (N). In Figure 5, we applied a force to a confined fluid through a piston. The resulting pressure in the fluid, by Pascal's Law, is equal throughout. And, every square inch (m^2) of the container wall is subject to an equal force because of the pressure.

THE HYDRAULIC LEVER

The apparatus Pascal used to develop his law probably consisted of two cylinders of different diameters connected as shown in Figure 8, with a liquid trapped between them. He might as well have called this apparatus the hydraulic lever, since it proved that leverage can be gained hydraulically as well as mechanically. Pascal found that a small force on a small piston will balance a larger force on a larger piston . . . provided that the piston areas are in proportion to the forces.

Thus in Figure 8, a two-pound (8.9 N) force on a one square inch (0.000645 m^2) piston balances a 100 pound (444.8 N) force on a 50 square inch (0.03225 m^2) piston. (We are ignoring the weights of the pistons themselves.)

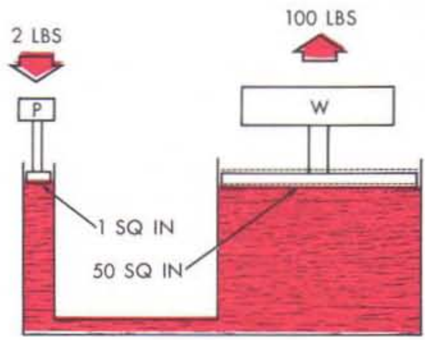


Figure 8

If we suppose that the force exerted on the small piston is the pressure source, the pressure would be the force divided by the piston area.

$$\text{Pressure} = \frac{2 \text{ pounds}}{1 \text{ sq. in.}} = 2\,\frac{\text{lbs}}{\text{sq. in.}} = \frac{2 \text{ psi}}{(0.14 \text{ bar})}$$

$$(\text{Pressure} = \frac{8.9 \text{ N}}{0.000645 \text{ m}^2} = 13792 \text{ N/m}^2 = 0.14 \text{ bar})$$

The resulting force on the large piston is equal to this pressure multiplied by the piston area. Thus:

$$\text{Force} = 2\,\frac{\text{pounds}}{\text{sq. in.}} \times 50 \text{ sq. in.} = 100 \text{ lbs}$$

$$(\text{Force} = 13792 \text{ N/m}^2 \times 0.03225 \text{ m}^2 = 444.8 \text{ N})$$

We have multiplied force 50 times in this example; in other words, obtained leverage or mechanical advantage of 50 to 1.

Compare this to the mechanical lever (Figure 9). Here also, force is multiplied from 2 pounds (8.9 N) to 100 pounds (444.8 N) by placing the two pound (8.9 N) force 50 feet (15.2 m) from the fulcrum and the 100 pound (444.8 N) force only one foot (0.305 m) from the fulcrum. It's just like putting a big boy and a little boy on a teeter-totter. The little boy needs a longer end of the board to balance the big boy.

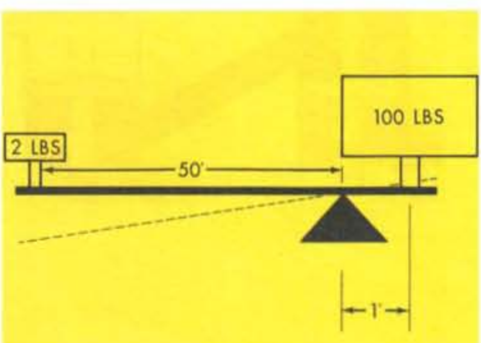


Figure 9

PRESSURE AND FORCE RELATIONSHIPS

Our example of hydraulic force multiplication has given us two important relationships from Pascal's Law. We can express these relationships as equations to solve simple problems of pressure and force.

First, pressure is equal to force divided by area:

$$P = \frac{F}{A}$$

Second, the force on any area is equal to the area multiplied by the pressure on the area:

$$F = P \times A$$

When using these relationships, always use these units:

F (Force) - - - Pounds (N)
P (Pressure) - - - PSI ($N/m^2 \times 10^{-5}$= bar)
A (Area) - - - Square Inches (m^2)

If any other units are given, convert them to these units before solving the problem. For instance, convert ounces to pounds; square feet to square inches.

Let's have a look at illustrations of these relationships.

RAISING A LOAD

A very common application of the first relationship is determining the pressure required to raise a load on a fork lift truck. If the total load requires a force of 7000 pounds (31138 N) and the lift cylinder has a 7 square inch (0.0045 m^2) piston, the pressure required is 1000 psi (6894857 N/m^2 = 68.9 bar).

$$P = \frac{F}{A} = \frac{7000 \text{ pounds}}{7 \text{ sq. in.}} = 1000 \text{ psi}$$

$$(P = \frac{F}{A} = \frac{31138 \text{ N}}{0.0045 \text{ m}^2} = 68.9 \times 10^5 \text{ N/m}^2 = 68.9 \text{ bar})$$

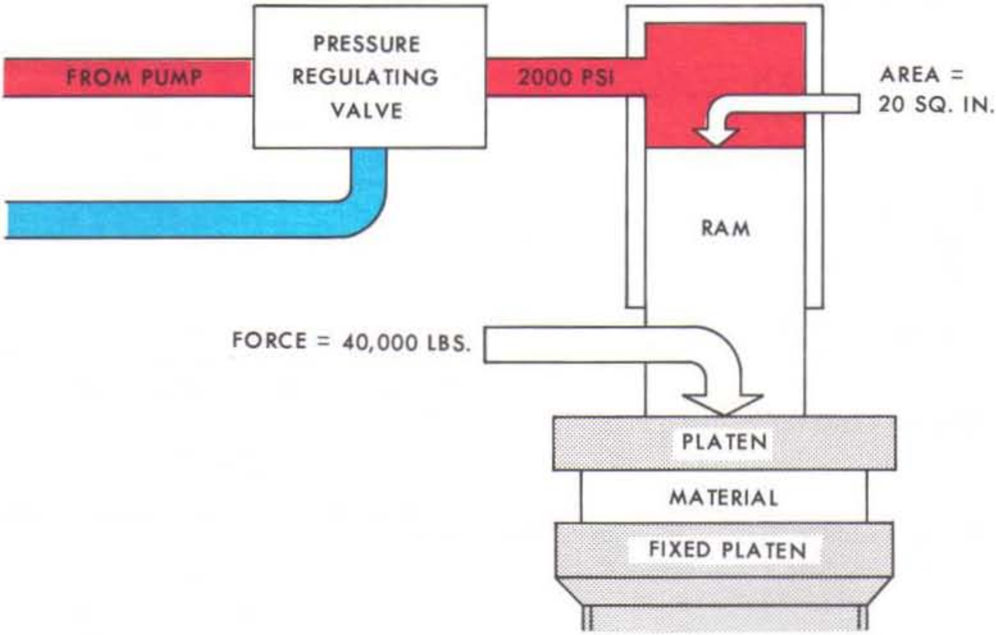


Figure 10

HYDRAULIC PRESS

To illustrate the second relationship, let's look at a simplified press (Figure 10). Pressure is regulated at 2000 psi (138 bar) against a ram area of 20 square inches (0.0129 m^2). The force output is 20 tons (1.78 N x 10^5).

$$F = P \times A = 2000 \text{ psi} \times 20 \text{ sq. in.} = 40{,}000 \text{ pounds} = 20 \text{ tons}$$

$$(F = P \times A = 13800000 \text{ N/m}^2 \times 0.0129 \text{ m}^2 = 1.78 \text{ N} \times 10^5)$$

BACK-PRESSURE

If two hydraulic cylinders are connected to operate in series (Figure 11), the pressure required to move the second cylinder is effective against the first cylinder as a back-pressure. If each cylinder requires 500 psi (34.5 bar) separately to raise its load, the

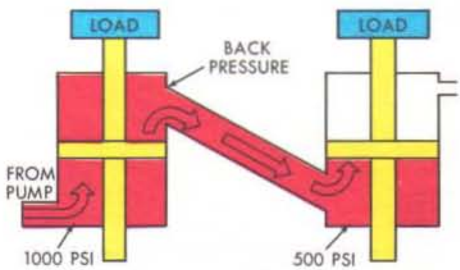


Figure 11

500 psi (34.5 bar) of the second cylinder adds to the load of the first cylinder.

The piston areas as shown are equal, so the first cylinder would have to operate at 1000 psi (69 bar); 500 psi (34.5 bar) to lift its load and 500 psi (34.5 bar) to overcome back-pressure.

Series operation is not common. We've used it here to illustrate a principle; namely, that pressures add up in series. Anything that creates a back pressure on the device that moves the load, in effect, adds to the load, and increases the pressure requirement of the system.

PRESSURE IN PARALLEL CONNECTION

When several loads are connected in parallel (Figure 12) the oil takes the path of least resistance. Since cylinder A requires the least pressure, it will move first. Furthermore, pressure won't build up beyond the needs of A until it has reached its travel limit. Then pressure will rise just high enough to move cylinder B. Finally when B is at its limit, pressure will rise to move cylinder C.

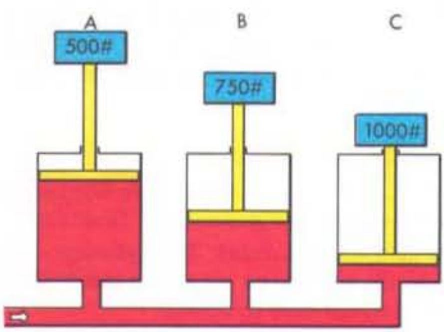


Figure 12

Parallel operation is possible with mobile valving which can meter a portion of the pump flow to each load.

FLOW MAKES IT GO

WHAT IS FLOW?

Flow is much easier for us to visualize than pressure, because we see it every time we turn on a water faucet. It is the movement of the hydraulic fluid caused by a difference in pressure at two points.

In our kitchen sink, for instance, we have atmospheric pressure. The city water works has built up a pressure or head in our pipes. When we open the tap, the pressure difference forces the water out.

In a hydraulic system, flow is usually produced by the action of a hydraulic pump . . . a device used to continuously push on the hydraulic fluid.

VELOCITY AND FLOW RATE

We have two ways of measuring flow . . . velocity and flow rate.

Velocity of the fluid is the average speed of its particles past a given point. It is usually measured in feet-per-second (fps) [metres-per-second (m/s)].

Velocity is an important consideration in sizing the hydraulic lines that carry the fluid between components.

Flow rate is the measure of how much volume of the liquid passes a point in a given time. It is usually measured in gallons per minute (gpm) [litres-per-minute (L/min)]. Flow rate determines the speed at which the load moves, and therefore is important to the consideration of power.

To understand the distinction between the two, suppose that we are pumping a constant rate of one gallon-per-minute (3.79 L/min) through two chambers of different diameter (Figure 13). Each chamber holds one gallon (3.79 L), so each will be emptied and refilled once each minute.

Chamber A, however, is twice as long as chamber B. Oil traveling through chamber A must move at a velocity of two feet per minute (0.6096 m/s). Oil traveling through chamber B moves only half as fast . . . one foot per minute (0.3048 m/s).

If A and B are sections of pipe, it is obvious that a constant flow rate (gpm) (L/min) will result in a lower velocity when the diameter increases or a higher velocity when the diameter decreases. In fact, the velocity of oil in a hydraulic line is inversely proportional to the cross-sectional area (or to the diameter squared). Thus, flow velocity in a two inch (50.8 mm) diameter line will be only 1/4 as fast as the same flow rate (gpm) in a one-inch (25.4 mm) diameter pipe. A low velocity is desirable to reduce friction and turbulence in the hydraulic fluid.

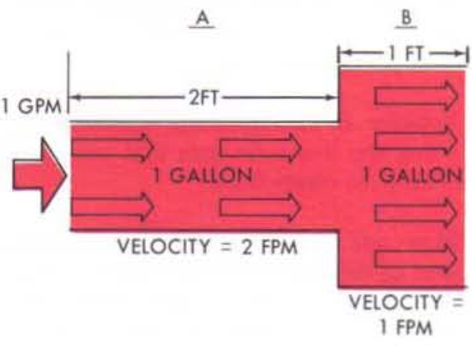


Figure 13

FLOW RATE AND SPEED

We can very easily relate flow rate (gpm) (L/min) to the speed at which the load moves, if we consider the cylinder volume we must fill and the distance the cylinder piston travels (Figure 14). The volume of the cylinder is simply the length of the stroke multiplied by the piston area. The piston area can be found by squaring the diameter and multiplying by 0.7854.

Volume (Cubic Inches (m^3) = Area (Square Inches) (m^2) x Length (Inches) (m)

Area = 0.7854 x Diameter Squared

This will give you the volume in cubic inches (m^3). A gallon is 231 cubic inches. To convert cubic inches to gallons:

$$\text{Gallons} = \frac{\text{Cubic Inches}}{231}$$

$$\left(\text{Litres} = \frac{\text{Cubic Metres}}{100000}\right)$$

Suppose we've determined that cylinder A is 2 feet (0.6096 m) long and holds one gallon (3.785 L). Cylinder B also holds one gallon (3.785 L), but is only one foot (0.3048 m) long. If we pump one gallon (3.785 L) per minute into each, both pistons will move their full travel in one minute. However, A must move twice as fast because it has twice as far to go in the same amount of time.

So we see that a small diameter cylinder moves faster with an equal flow rate into it. Now suppose we increase the flow rate to two gpm (7.57 L/min). The cylinders then would fill in half the time. The speed of the pistons would have to double.

We now have two ways of increasing the speed at which the load moves; decrease the size of the cylinder or increase the gpm

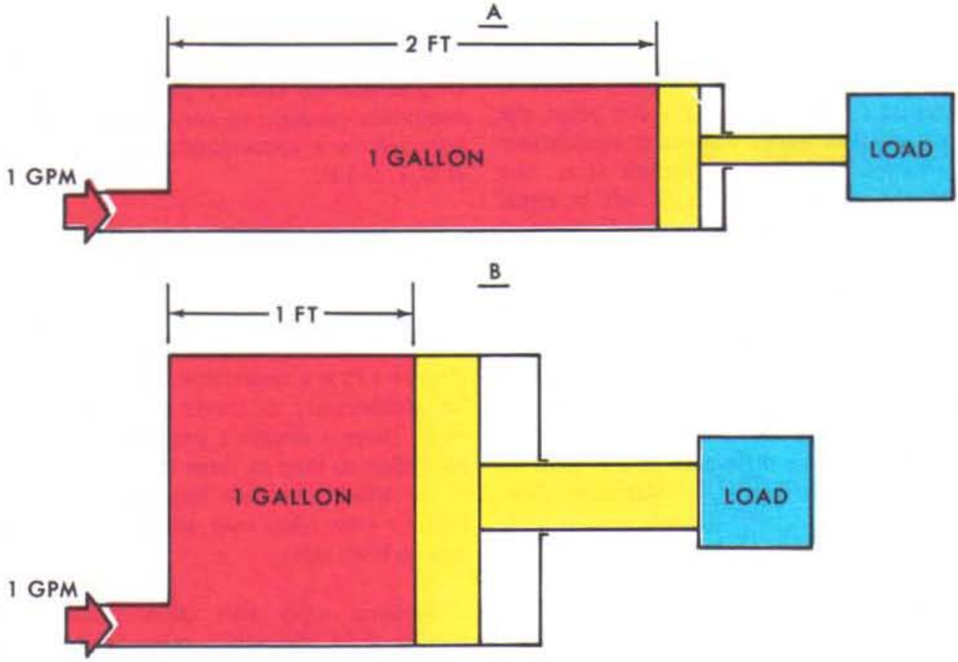


Figure 14

(L/min) flow to the cylinder. Conversely, we slow the load down by reducing the flow or increasing the size (diameter) of the cylinder.

The speed of a cylinder, then, must be proportional to flow and inversely proportional to the piston area (or the diameter squared).

FLOW AND PRESSURE DROP

A basic rule of hydraulics is that wherever there is flow, there must be a pressure difference or pressure drop. Conversely, where there is a difference in pressure, there must be either flow or at least a difference in the level of the liquid.

When a liquid is not subject to a pressure difference, it simply seeks a level, as you see in Figure 15, view A.

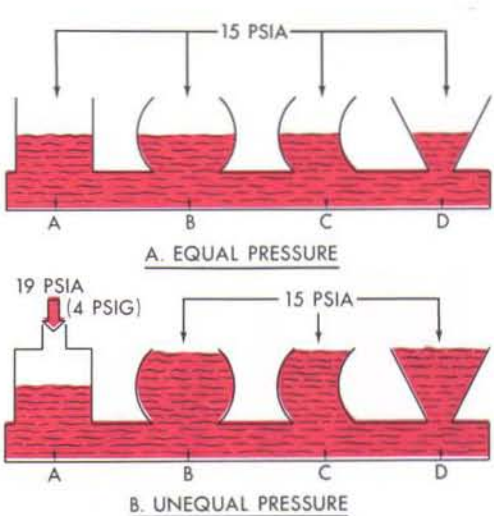


Figure 15

Everywhere in the containers, the liquid is subject only to atmospheric pressure; therefore it does not move. If pressure is increased or decreased at any one point, the liquid will flow until a balance or equilibrium is reached. In the equilibrium state, the measured difference in height will be equal to the head that the difference in pressure would create. For instance, if the liquid is oil, a pressure difference of 4 psi (0.28 bar) is equivalent to a difference in height of 10 feet (Figure 15, view B). A foot of oil, remember, is equivalent to 0.4 psi (0.028 bar).

If the pressure difference is too great to create an equilibrium, continuous flow results.

The pressure difference when a liquid is flowing is used to overcome friction and to lift the fluid where necessary. When a liquid is flowing, the pressure is always highest upstream and lowest downstream. That is why we refer to the difference as "drop".

FRICTION AND PRESSURE DROP

Pressure drop due to friction is shown as a difference in head in Figure 16. Since there is no resistance to flow at point B (we call this condition "free flow"), the pressure there is zero.

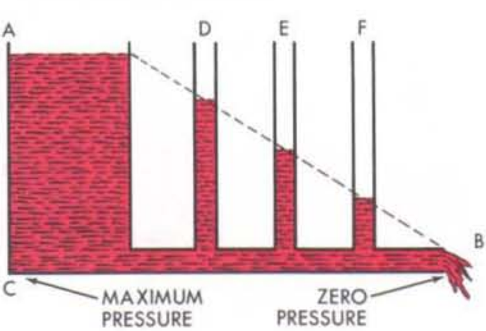


Figure 16

Pressure at point C is maximum because of the head at A. As the liquid flows from C to B, friction causes a pressure drop from maximum pressure to zero pressure. This is reflected in a succeedingly decreased head at D, E and F.

FLOW THROUGH AN ORIFICE

Pressure drop occurs to a greater degree when the flow is restricted. An orifice (Figure 17) is a restriction often placed in a line deliberately to create a pressure difference. There is always a pressure drop across an orifice so long as there is flow. However, if we block the flow beyond the orifice, Pascal's Law takes over and pressure equalizes on both sides.

Pressure drop also takes place when passing fluid through a valve or line. The smaller the valve passage or line the greater

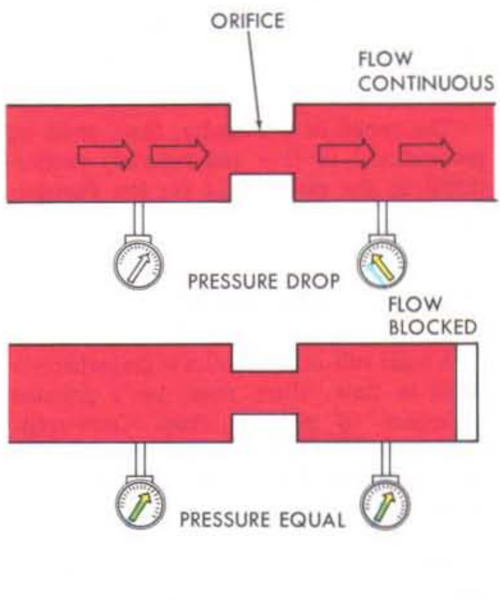


Figure 17

the pressure drop. In effect, the restrictive area acts as an orifice.

The energy lost due to pressure drop is converted to heat.

LAMINAR AND TURBULENT FLOW

A streamlined or laminar flow (Figure 18) is desirable to keep friction minimized. Abrupt changes in section, sharp turns and too high velocity all induce turbulent flow. Then, instead of moving in smooth, parallel paths, the fluid particles develop cross currents. The result is a significant increase in friction and pressure drop.

WORK AND ENERGY

Earlier in this chapter, we developed the concepts of force and pressure primarily as measures of effort. Work, on the other hand, is a measure of accomplishment. It requires motion to make a force do work. Therefore, to do work in a hydraulic system, we must have flow.

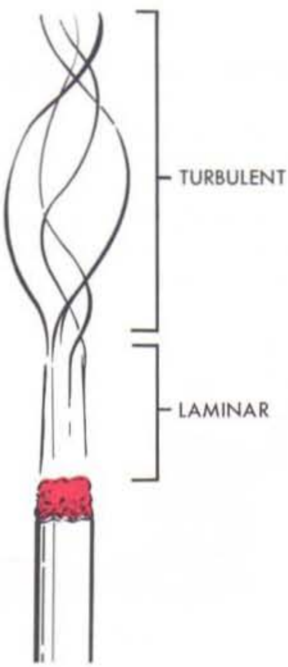


Figure 18

We could best define work as exerting a force over a definite distance. Thus, if we raise a 1000 pound (4448 N) load 4 feet (1.22 m), we do 4000 foot pounds (5427 J) of work. Work is a measure of force multiplied by distance. We usually express it in foot pounds or inch pounds (Joules).

Work (ft. lbs) =
Force (lbs) x Distance (ft)

(Work (Joules) = Force (Newtons) x
Distance (Metres))

Energy is the capacity to do work, and is expressed in the same units as work. We are familiar with several forms of energy. The thousand-pound weight just mentioned, when it is raised, has potential energy. It is capable of doing work when it is lowered.

A body in motion has kinetic energy, capable of doing work. A lump of coal contains heat energy; a battery, electrical energy; a steam boiler, pressure energy.

CONSERVATION OF ENERGY

Before the smashing of the atom, we were told that energy could neither be created nor destroyed. This is the law of conservation of energy.

In a hydraulic system, we do not destroy any energy. We transfer energy from one point to another and from one form to another. Energy that we lose to friction is transformed into heat; but is not lost to the universe. It is simply wasted energy.

ENERGY TRANSFER IN THE HYDRAULIC LEVER

Let's have another look now at Pascal's hydraulic lever and see how energy is transferred there.

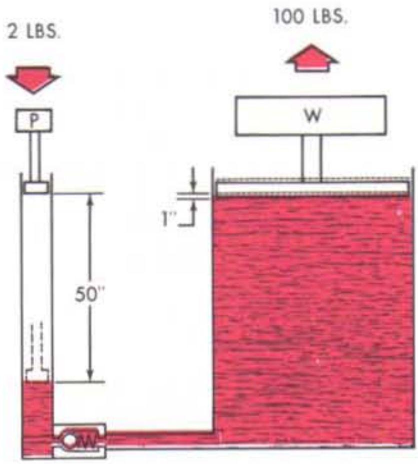


Figure 19

In Figure 19, we have just slightly upset the balance so that the small piston is forced down and pushes the large piston up. For simplicity's sake, let's leave the numbers the same and ignore the effects of friction.

At the small piston, we have moved two pounds (8.9 N) of force downward for 50 inches (1.27 m). In doing so, we gave up 100 inch pounds (11.3 J) of potential energy. We changed that 100 inch pounds (11.3 J) to pressure energy in 50 cubic inches (820 mL) of liquid, the displaced volume.

The 50 cubic inches (820 mL) of liquid moved the 50-square-inch (0.032 m^2) piston up one inch (25.4 mm). Thus, the 100 pound (445 N) weight received an increase of 100 inch pounds (11.3 J) of potential energy. At each piston, then, we did 100 inch pounds (11.3 J) of work. Thus, energy was transferred without loss from the two pound (8.9 N) force to the 100 pound (445 N) force. In the mechanical lever (Figure 9) we also transferred energy from one weight to the other without loss. Again, the energy transferred in both places is equal to the force multiplied by the distance.

KINDS OF ENERGY IN HYDRAULIC SYSTEM

The purpose of a hydraulic system is to transfer mechanical energy from one place to another through the medium of pressure energy. Mechanical energy driving the hydraulic pump is converted to pressure energy and kinetic energy in the fluid. This is reconverted to mechanical energy to move a load. Friction along the way causes some losses in the form of heat energy.

The prime source of the energy may be heat in a motor fuel or electrical energy in a battery or from power lines.

BERNOULLI'S PRINCIPLE

Bernoulli's principle tells us that the sums of the pressure and kinetic energy at various points in a system must be constant, if flow is constant. When a fluid flows through areas of different diameters (Figure 20), there must be corresponding changes in

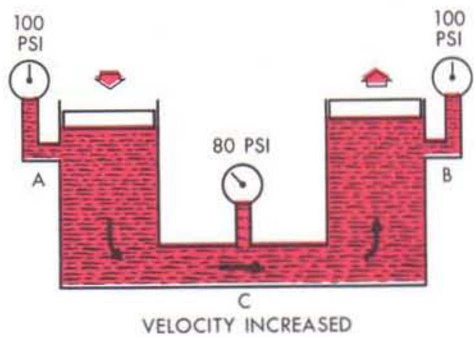


Figure 20

velocity. At the left, the section is large so velocity is low. In the center, velocity must be increased because the area is smaller. Again at the right, the area increases to the original size and velocity again decreases.

Bernoulli proved that the pressure component at C must be less than at A and B because velocity is greater. An increase in velocity at C means an increase in kinetic energy. Since energy cannot be created, kinetic energy can increase only if the static component, pressure, decreases. At B, the extra kinetic energy has been converted back to pressure. If there is no frictional loss, the pressure at B is equal to the pressure at A.

Figure 21 shows us the combined effects of friction and velocity changes. As in Figure 16, pressure drops from maximum at C to zero at B. At D, velocity is increased, so the pressure head decreases. At E, the head increases as most of the kinetic energy is given up to pressure energy because velocity is decreased. Again at F, the head drops as velocity increases.

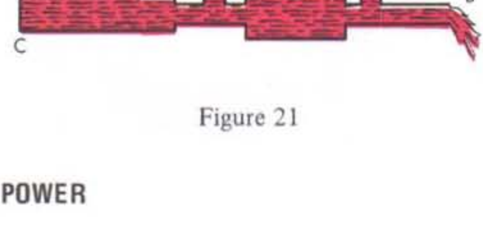


Figure 21

POWER

Now let's conclude the fundamentals of flow with a word about power. Power is the rate of doing work or the rate of energy transfer.

To visualize power, think about climbing a flight of stairs. If you walk up, it's pretty easy. But if you run up, you're liable to get to the top out of breath. You did the same amount of work either way . . . but when you ran up, you did it at a faster rate, which required more power.

The standard unit of power is the horsepower . . . devised by James Watt to relate the ability of his steam engine to the pulling power of a horse. By experimenting with weights, pulleys and horses, Watt decided that a horse could comfortably do 550 foot pounds of work a second, or 33,000 foot pounds per minute, hour after hour. This value has since been designated as one horsepower (hp). Power, then, is force multiplied by distance divided by time:

$$P\ (\text{Power}) = \frac{F\ (\text{Force}) \times D\ (\text{Distance})}{T\ (\text{Time})}$$

1 hp = 33,000 foot pounds/minute or
550 foot pounds/second or
746 watts (electrical power) or
42.4 British Thermal Units of
Heat/minute

In SI (metric) units:

(1 kW = 1000 J/sec = 1000 N x m/sec)

The power used in a hydraulic system can be computed if we know the flow rate and the pressure:

$$\text{hp} = \frac{\text{gpm} \times \text{psi}}{1714}$$

$$\left(\text{kW} = \frac{\text{L/min} \times \text{bar}}{600}\right)$$

or

$$\text{hp} = \text{gpm} \times \text{psi} \times .000583$$

$$(\text{kW} = \text{L/min} \times \text{bar} \times 0.00166)$$

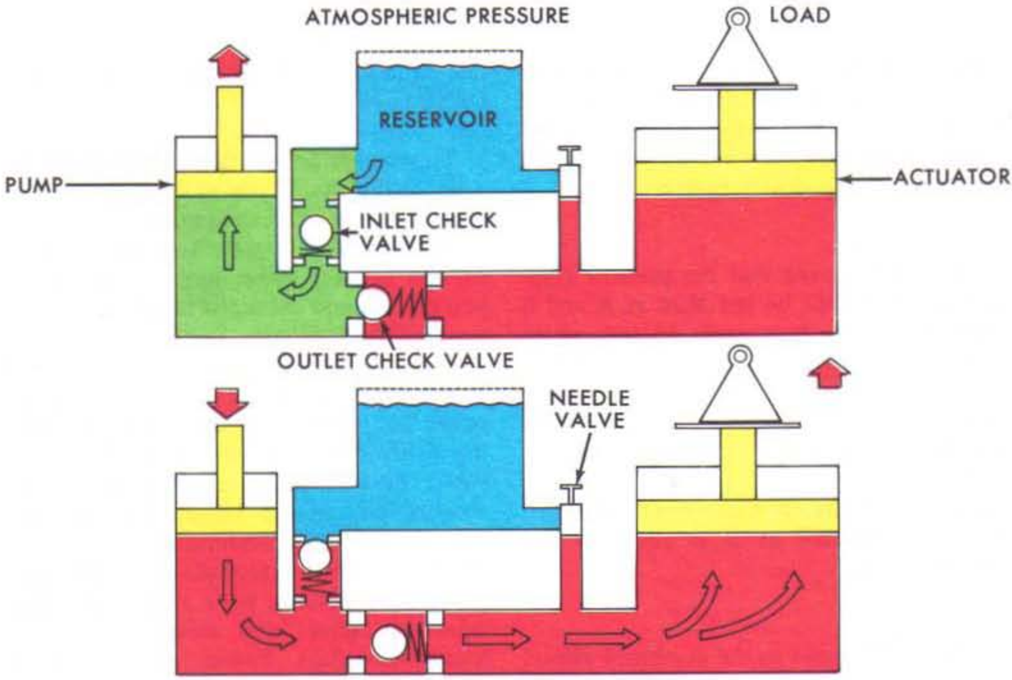


Figure 22

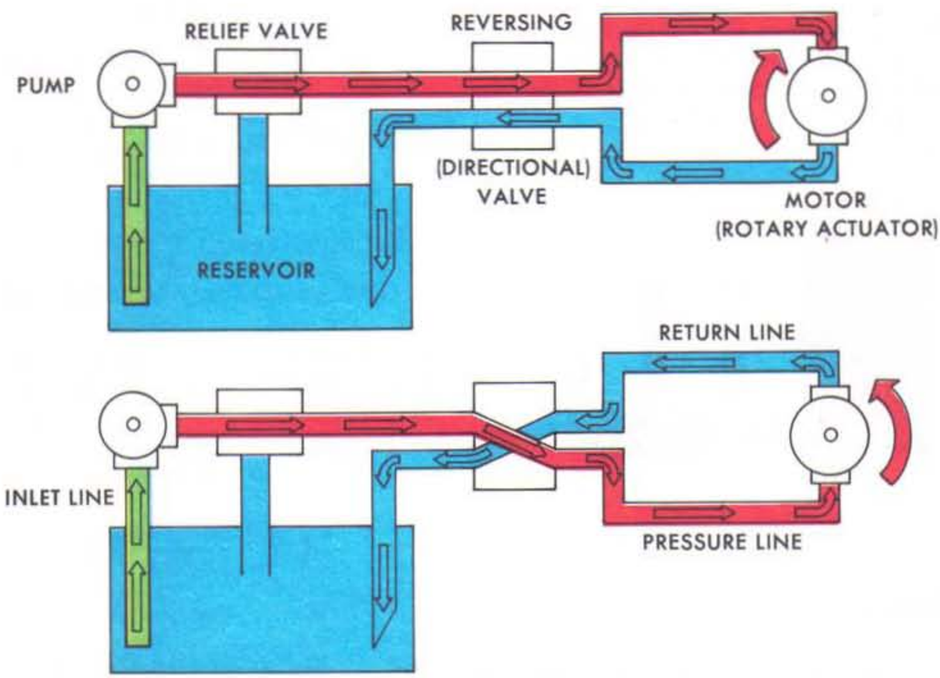


Figure 23

We can see from this relationship that an increase in either pressure or flow will increase horsepower. Also, if pressure or flow decreases, horsepower decreases.

HYDRAULIC SYSTEM COMPONENTS

In the following chapters, you will be studying the components that go into mobile hydraulic circuits, and how they are put together to do their jobs. Almost from the start of Chapter 2, we'll be forced to talk of the component's interrelationships. Therefore, we must give them basic attention now.

We have already mentioned that a pump is required to push the fluid. We have implied, too, that a cylinder is the output of the system. Besides a pump and cylinder, we also require valves to control the fluid flow; a reservoir to store fluid and supply it to the pump; connecting lines; and various hydraulic accessories.

Let's look at two very simple systems now, and see how the basic components are classified.

THE HYDRAULIC JACK

If Figure 22 resembles Pascal's hydraulic lever, it is intentional. In this system, we have added a reservoir and a system of valves to permit stroking the small cylinder or pump continuously and raising the large piston or actuator a notch with each stroke. In the top view, we see the intake stroke. The outlet check valve is closed by pressure under the load and the inlet check valve opens to allow liquid from the reservoir to fill the pumping chamber.

In the bottom view, the pump is stroked downward. The inlet check valve is closed by pressure and the outlet valve opens. Another "slug" of liquid is pumped under the large piston to raise it.

To lower the load, we open a third valve, a needle valve, which opens the area under the large piston to the reservoir. The load then pushes the piston down and forces the liquid into the reservoir.

This is the basic circuit for a hydraulic jack.

MOTOR-REVERSING SYSTEM

In Figure 23, we have an entirely different kind of system. Here, a power-driven pump operates a reversible rotary motor. A reversing valve directs fluid to either side of the motor and back to the reservoir. A relief valve protects the system against excess pressure, and can bypass pump output to the reservoir if pressure rises too high.

PUMP CLASSIFICATIONS

The pump in Figure 22 is called a reciprocating pump. Most pumps are of the rotary type, as in Figure 23, and are driven by engines or electric motors. Rotary pumps can be constant displacement; meaning they deliver the same amount of fluid every stroke, revolution or cycle. Flow rate varies in proportion to drive speed. Or they can be variable displacement pumps, which can have their delivery rates changed by external controls while drive speed remains constant.

ACTUATOR CLASSIFICATIONS

The actuator is the system's output component. It converts pressure energy to mechanical energy.

A cylinder or ram is a linear actuator. Its outputs are force and straight line motion. A motor is a rotary actuator. Its outputs are torque and rotating motion.

The large piston actuator in Figure 22 is a single-acting cylinder. This means it is operated hydraulically in one direction only and returned by another means . . . in this case, gravity. A double-acting cylinder operates hydraulically in both directions.

The motor in Figure 23 is a reversible motor. Other motors are uni-directional or non-reversible. They can only rotate in one direction.

VALVE CLASSIFICATIONS

We will study three classes of valves in Chapter 5. They are:

(1) Directional control valves,
(2) Pressure control valves and
(3) Flow or volume control valves.

Directional Valves

Directional control valves tell (direct) the oil where to go by opening and closing passages. The check valves in Figure 22 are directional valves, and the reversing valve in Figure 23 is a directional valve. The check valves are called one-way valves, because they permit only one flow path. The reversing valve is a four-way valve, because it has four flow paths.

Pressure Control Valves

A pressure control valve is used to limit pressure or to control or regulate pressure in the system. The relief valve in Figure 23 is a pressure control valve . . . it limits the pressure that can be developed in the circuit. Other types of pressure controls are brake valves, sequence valves, pressure reducing valves and back-pressure or counterbalance valves.

Flow Control Valves

Flow control valves regulate flow to control the speed of an actuator. The needle valve in Figure 22 has a flow control function. It restricts flow so that the load can't come down too fast.

CLASSIFICATION OF LINES

The lines which connect the components of our systems are classified according to their functions. The principal kinds of lines are:

$$\text{Working Lines: } \begin{cases} \text{Pressure Lines} \\ \text{Inlet Lines} \\ \text{Return Lines} \end{cases}$$

$$\text{Non-working Lines: } \begin{cases} \text{Drain Lines} \\ \text{Pilot Lines} \end{cases}$$

Working Lines

Working lines are lines which carry the mainstream of fluid in the system; that is, the fluid involved in energy transfer. Starting at the reservoir, we have an inlet line which carries the fluid to the pump inlet (Figure 23). From the pump to the actuator, is the pressure line, which carries the same fluid under pressure to do work. After the pressure energy in the fluid is given up at the actuator, the exhaust fluid is re-routed to the reservoir through the exhaust or return line.

Non-Working Lines

Non-working lines are auxiliary lines which do not carry the main stream of flow. A drain line is used to carry leakage oil or exhaust pilot fluid back to the reservoir. A pilot line carries fluid that is used to control the operation of a component.

CIRCUIT DIAGRAMS

We have used some very simple schematic diagrams in this chapter to illustrate hydraulics principles and the operation of components. There are many kinds of schematics that can be used to show the operation of components and circuits. We will use several types for instruction in this manual. However, outside of instructional materials we almost exclusively use a type of shorthand known as graphical diagrams. Each component and line has a graphical symbol, which is a simple geometric form. Graphical diagrams do not attempt to show how the parts are constructed; only their functions and their connections.

Notice that the graphical diagram for the reversible motor circuit (Figure 24) does show all the port connections, lines and flow paths that are contained in Figure 23. The differences are that in the graphical diagram the parts are easier to draw and the shorthand is universal; anyone trained in hydraulics can understand the system.

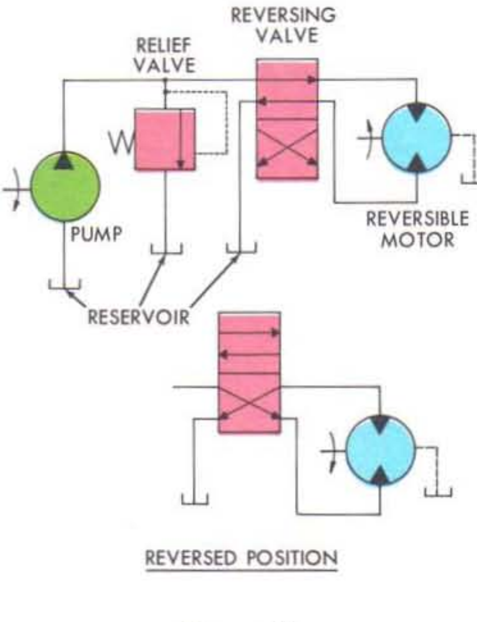


Figure 24

We will study this "shorthand" system in detail in Chapter 6.

ADVANTAGES OF HYDRAULIC SYSTEMS

Now that we have studied basic principles and have an idea how hydraulics work, let's close this chapter with a brief survey of the advantages of hydraulics over other methods of power transmission.

Design is simpler. In most cases, a few pre-engineered components will replace complicated mechanical linkages.

Flexibility. Hydraulic components can be located with considerable flexibility. Pipes and hoses in place of mechanical elements virtually eliminate location problems.

Smoothness. Hydraulic systems are smooth and quiet in operation. Vibration is kept to a minimum.

Control. Control of a wide range of speeds and forces is easily possible.

Cost. High efficiency with minimum friction loss keeps the cost of a power transmission at a minimum.

Overload Protection. Automatic valves guard the system against a breakdown from overloading.

Of course, as the saying goes, no one is perfect. The disadvantages of hydraulics are in the precision parts that are exposed to unsympathetic climates and dirty atmospheres. Maintenance to protect against rust, corrosion, dirt, oil deterioration and other adverse environment is doubly important on mobile machinery.

CHAPTER 2
Reservoirs and Accessories

In this chapter, we'll begin studying the hydraulic system by considering the components that store and condition the fluid. The oil reservoir (or sump, or tank, as it is often called) usually functions both as a storehouse and a fluid conditioner. Filters, strainers and magnetic plugs condition the fluid by removing harmful impurities that could clog passages and damage parts. Heat exchangers or coolers often are used to keep the oil temperature within safe limits and prevent deterioration of the oil. Accumulators, though technically energy storage devices, function as fluid storehouses also, and will be discussed in this chapter.

THE MANY FUNCTIONS OF THE RESERVOIR

The oil reservoir can easily become the biggest challenge when designing a mobile hydraulic system. The other parts of the system – pumps, valves, actuators, lines, fittings – all come pre-engineered and tested. The reservoir alone must be custom designed for the particular vehicle . . . and size, shape and location are nearly always problems.

It is easy enough to design an ideal reservoir if you have unlimited space, don't have to worry about weight and can choose your location. Designers of machinery that will be bolted to a floor, in fact, can buy complete reservoir assemblies with every desirable feature built in. But the mobile equipment designer may have to fit his reservoir into the tubular arms of a front end loader or tuck it into a minimal space in an engine compartment or contour it to fit under a driver's seat. The reservoir, then, may not be able to condition as well as store the oil, so other hydraulic accessories will be required.

RESERVOIR DESIGN

A properly constructed reservoir is more than just a tank to hold the oil until the pump demands fluid. Whenever practical, it should also be capable of:

- o Dissipating heat from the oil.
- o Separating air from the oil.
- o Settling out contamination in the oil.

Let us look at some of the design features of a typical reservoir (Figure 25).

Shape

Ideally, a reservoir should be high and narrow, rather than shallow and broad. The oil level should be as high as possible above the opening to the pump inlet line. This prevents a vortex or whirlpool effect. Anytime you see a whirlpool at the inlet line opening, the system is possibly taking in air. Aerated oil can't do a proper job of transmitting power because the air is compressible. Further, aerated oil has a tendency to break down and lose its lubricating ability.

Size

For a long time, there has been a "thumb rule" that the reservoir should be sized two to three times the pump output per minute. By this rule, which works well for stationary machinery, a 10-gpm (38 L/min) system would call for a 20- or 30 gal. (76 or 114 litre) reservoir.

This seldom happens on a mobile machine. You're more likely to find a 20- or 30-gallon (76 or 114 L) tank used with a 100-gpm (379 L/min) system. Many 10-gpm (38 L/min) systems are operating with two or three gallon (7.6 or 11.4 L) tanks. This is possible because mobile systems operate intermittently rather than constantly, and

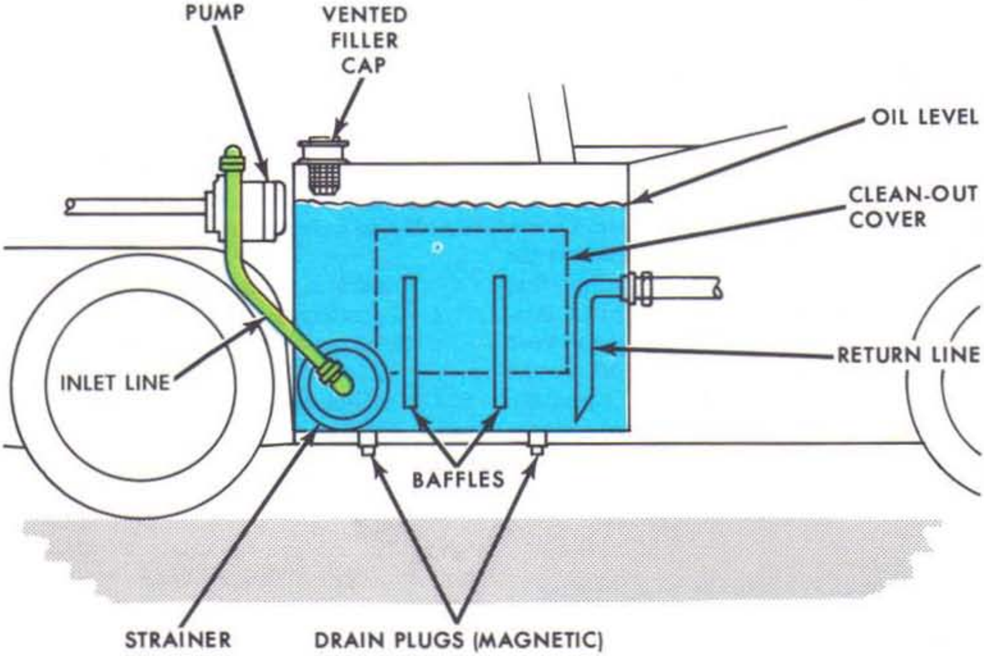


Figure 25

because other accessories are available. The largest reservoirs on mobile equipment are found on road machinery. They may be 40 or 50 gallon (151 to 189 L) capacity to handle more than 200 gpm (757 L/min).

A large size tank is highly desirable for cooling. Large surface areas exposed to the outside air transfer heat from the oil. A large tank also, by reducing recirculation, helps to settle out contamination and separate air.

The reservoir must be sized so that there is a reserve of oil with all the cylinders in the system fully extended. The reserve must be high enough to prevent a vortex at the inlet line opening. Also, there must be enough space to hold all the oil when the cylinders are retracted . . . with some space to spare to allow for expansion when the oil is hot.

Baffles

A baffle plate is desirable to separate the inlet line from the return line. This will cause the return oil to circulate around the outer wall for cooling before it can get to the pump again. The baffle plate should be about 2/3 the height of the oil level. The lower corners are diagonally cut to allow circulation. The cuts must be larger in area than the inlet line cross-section. Otherwise there might be an unequal level of oil between the return and inlet side.

Baffling also prevents the oil from sloshing around when the machine is moving. Many large reservoirs are cross-baffled to provide cooling and prevent sloshing.

LOCATION

A good percentage of mobile equipment reservoirs are located above the pumps. This creates the desirable condition of a flooded pump inlet. A flooded inlet reduces the possibility of pump cavitation, a condition where all the available space is not filled, and which often results in erosion of metal parts. Flooding the inlet also reduces the vortex tendency at the inlet pipe opening.

Location also will obviously affect heat dissipation. Ideally, all the tank walls should be exposed to the outside air. Heat moves from a hot substance to a cold substance . . . and heat transfer is greatest when there is a large temperature difference. Locating the reservoir in a hot engine compartment, for instance, obviously will not permit it to dissipate heat most effectively. The reservoirs built into front end loader arms are very effective in transferring heat.

Vented or Pressurized

Most reservoirs are vented to the atmosphere. A vent opening is provided to allow air to leave or enter the space above the oil as the level of the oil goes up or down. This maintains a constant atmospheric pressure above the oil.

The reservoir filler cap often is used as the vent. It is protected with a micronic filter element to keep dirt from entering with the air.

Some reservoirs are pressurized rather than vented. A typical pressurized reservoir will use a simple pressure control valve in place of a vent. The valve automatically lets filtered air into the tank but prevents its release unless the pressure reaches a preset level. Pressurization takes place when the oil and air in the tank expand from heat. Some reservoirs are pressurized by the air compressor on the vehicle.

Maintenance Features

Maintenance procedures always include draining and cleaning the reservoir periodically. The design should make this beneficial activity as easy to do as possible. Otherwise it might not get done.

A dished bottom fitted with a drain plug at the lowest point is always desirable. It helps, too, if the plug fitting is flush with the inside of the tank to allow full drainage.

On large tanks, access plates may be bolted on the ends for easy removal and servicing.

A sight gauge or dipstick should be provided to make it easy to check the oil level. If this is not done, the oil level often is not checked. Then, if a leak occurs, the pump can be starved and damaged from loss of lubrication.

A strainer or strainers on the pump inlet line may not require maintenance frequently. However, a filter in the return line will require changing the element at regular intervals. Thus, it is best not to locate the return line filter inside the reservoir.

When the reservoir is pressurized by compressed air, moisture can become a maintenance problem. There is always some water in the air. A water trap should be provided to remove moisture; and it should be placed where it can be inspected every day.

Connecting The Lines

The pump inlet and tank return lines should be attached by flanges or by welded heavy-duty couplings. Standard couplings are usually not suitable because they spread when welded.

If the inlet line is connected at the bottom, the coupling should extend well above the bottom inside the tank. That way, residual dirt won't get in the inlet line when the tank or strainer is cleaned.

The return line should discharge near the tank bottom; always below the oil level. The pipe is usually cut at a 45° angle and the flow aimed away from the inlet line to improve circulation and cooling.

Fabrication

Reservoirs range in construction from small steel stampings to large cast or fabricated units. The large tanks should ideally be sand blasted after all the welding is completed; then flushed and steam cleaned. This will remove welding scale and scale left from hot rolling the steel. The inner surface then should be sealed with a paint compatible with the hydraulic fluid. Non-bleeding red engine enamel is suitable for petroleum oil, and seals in any residual dirt not removed by flushing and steam cleaning.

POWER STEERING RESERVOIRS

At the beginning of this chapter you read that mobile reservoirs must be custom-designed. The exception to this is the power steering reservoir. All Vickers power steering components are pre-engineered to operate together. Power steering pumps are furnished with integral reservoirs (Figure 26) or without. The integral reservoirs mount either on top of the pump or surround the pump. Where there is not space to mount a pump with an integral reservoir, the OEM may design his own reservoir, or use Vickers TM1 reservoir assembly (Figure 27). The TM1 reservoir is also suitable for other low-volume applications.

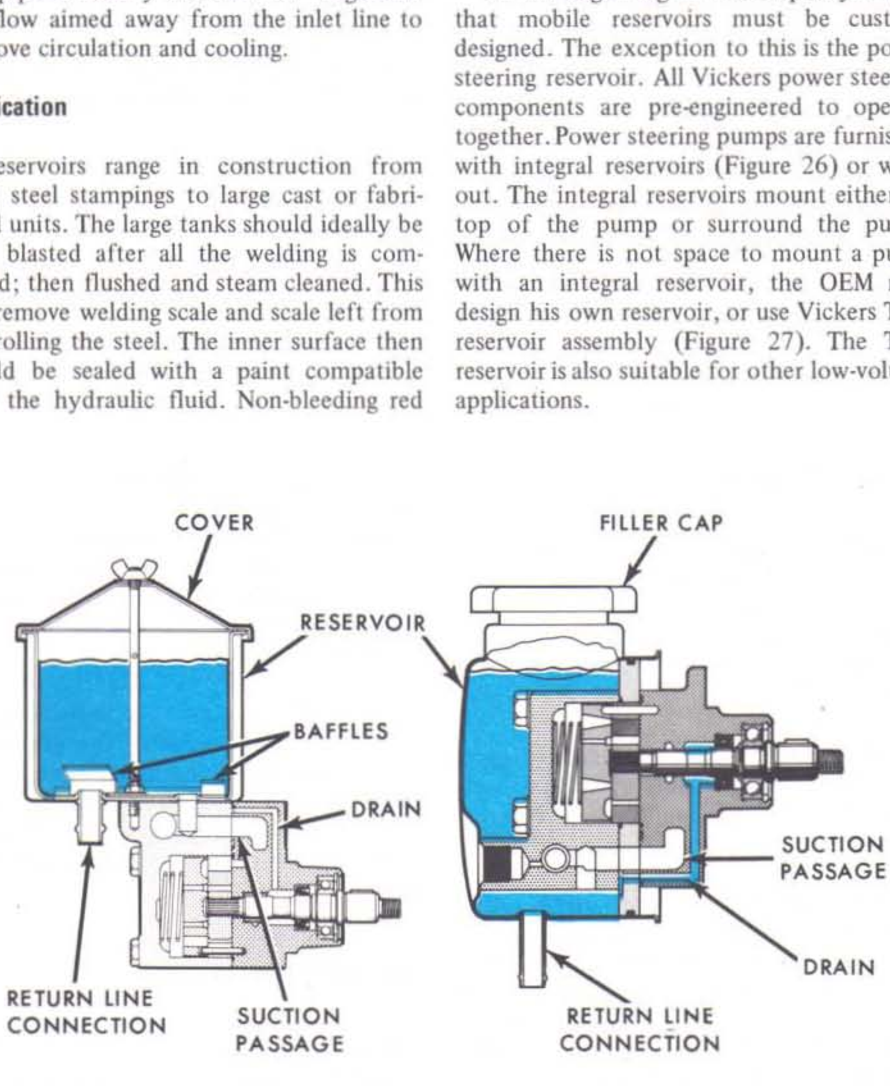


Figure 26

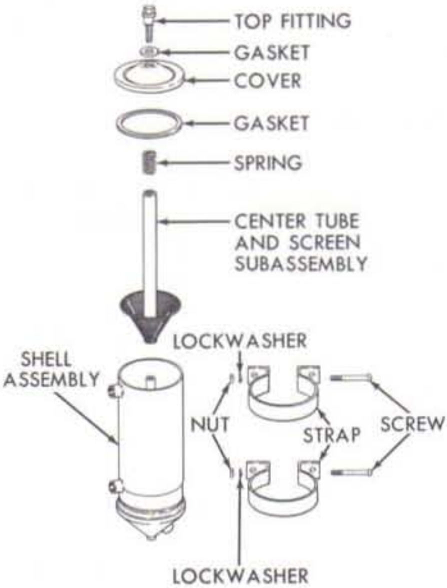


Figure 27

KEEPING THE OIL CLEAN

Contamination in hydraulic fluid has been given a lot of attention by the industry . . . and is in for a whole lot more attention. Research and testing of the effects of dirt, water, wear products, oil deterioration products and other contaminants in the oil are important factors in today's high performance hydraulic equipment. It has been found that contaminated fluid is the major cause of hydraulic system failure.

WHY CLEANLINESS?

The demand for ever greater performance from smaller packages has greatly increased the need for keeping the oil clean. When we were operating at low speeds, low temperatures, and low pressures, cleanliness was a virtue. Today it is a necessity, which affects the life of the components and the oil; and the operation of the system. Briefly, here are some of the factors involved: Wear occurs in all hydraulic systems. If dirt particles remain suspended in the oil, they act just like a grinding compound and speed up wear. Other foreign particles, particularly metal, have the same effect.

The parts of hydraulic components move together on thin films of oil. The higher the pressure, the thinner the film; therefore the less contamination tolerant the system is.

The performance of hydraulic components is affected by contamination. Sticking and sluggish action can occur; small controlling passages may become plugged. Dirt can prevent valves from seating, resulting in leakage and loss of control.

The hydraulic oil itself is affected by contamination. Water has a tendency to separate certain inhibitors from high-performance hydraulic oil, reducing its usable life. Other contaminants seem to have a catalyst or "helping hand" effect on oil oxidation. And, it has been demonstrated that fine particle contamination actually reduces the safe operating temperature. Extremely clean fluids can operate as much as 25^{o} to 50^{o} hotter than contaminated fluids, without oxidation.

Hydraulic oil is kept clean by the use of magnetic plugs, strainers and filters.

MAGNETIC PLUGS

Magnetic plugs are useful for removing iron or steel particles from the oil. We install them in the reservoir where they will attract the particles from the fluid. Naturally, they should be accessible for cleaning. Figure 28 illustrates how magnetic plugs can look after being in use for several months.

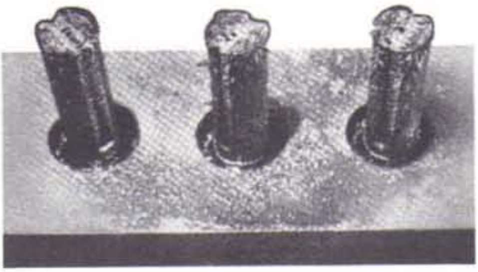


Figure 28

THE DIFFERENCE BETWEEN A FILTER AND A STRAINER

Because they both do the same kind of job, filters and strainers have been traditionally called filters. A few years back, the Joint Industry Conference (JIC), an organization of several industrial groups interested in hydraulics, attempted to standardize definitions. From this organization came definitions of filters and strainers according to their construction. Thus, a strainer was defined as a "device for the removal of solids from a fluid wherein the resistance to motion of such solids is in a straight line." A filter was "a device for the removal of solids from a fluid wherein the resistance to motion of such solids is in a tortuous path."

Without commenting on these definitions, let's just note that the National Fluid Power Association, composed of the manufacturers of hydraulic and pneumatic equipment, has lately given us new definitions:

<u>Strainer</u>: A coarse filter.

<u>Filter</u>: A device whose primary function is the retention, by some porous medium, of insoluble contaminants from a fluid.

The porous medium is the screening or filtering material that allows oil to flow through it but traps solid particles.

STRAINERS

Strainers are constructed of fine mesh wire screens, or of screening elements consisting of specially processed wire of varying thickness wrapped around metal frames. They do not provide as fine a screening action as filters, but they offer less resistance to flow and are used in pump inlet lines where pressure drop must be kept to a minimum.

Figure 29 illustrates a typical strainer and three of the many possible arrangements for using them in an inlet line. If one strainer is not big enough to supply the demands of the pump, two or more can be used in parallel as shown. Since strainers require periodic cleaning, they need only be hand tightened, so long as the fittings are submerged. Any exposed inlet line fittings, of course, must be air tight.

FILTERS

There are many kinds and sizes of filters, using many different means of screening out solid particles. In general, we can say that a filter will have a body or base with port connections (Figure 30), a cover of some kind and a filtering element that is removable for easy cleaning or replacement. Most often, the element is designed for replacement.

Filters differ in micron rating, flow conditions, type of element and element material, and location in the circuit.

Filtering Materials

There are three general classes of filter materials – mechanical, absorbent (inactive) and adsorbent (active).

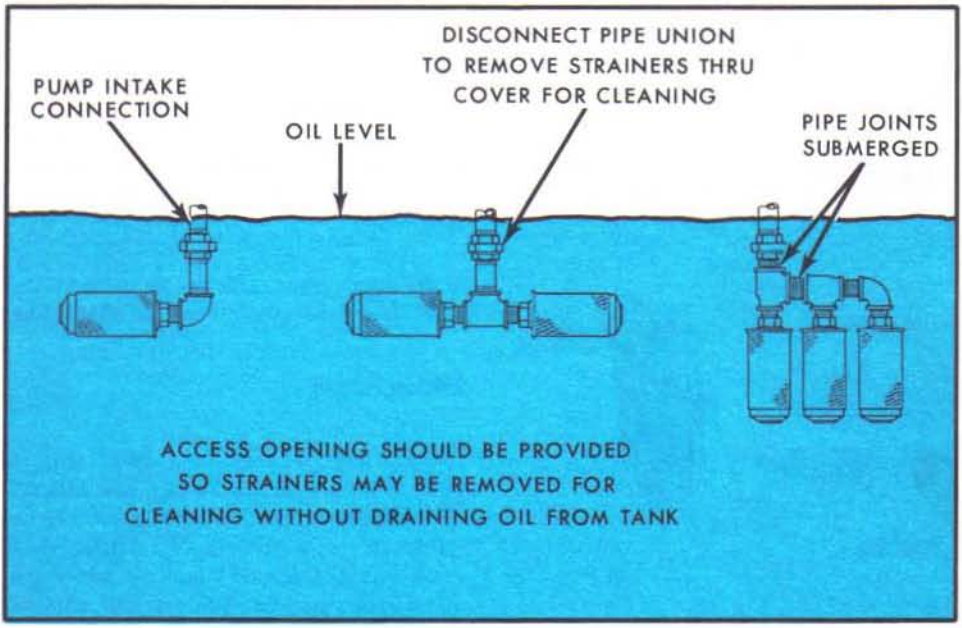


Figure 29

Mechanical filters contain closely woven metal screens or discs. They generally remove only fairly coarse, insoluble particles.

Absorbent (inactive) filters have material such as cotton, wood pulp, yarn, cloth or resin-impregnated paper. They remove much smaller particles, and some remove water and water-soluble contaminants. The elements often are treated to make them sticky-fingered; that is, to give them an affinity for the contaminants found in hydraulic oil.

Adsorbent or active filtering materials such as charcoal and Fuller's earth are not recommended for hydraulic systems. They remove particles by adsorption as well as mechanically and often remove the additives compounded into hydraulic oil for wear protection.

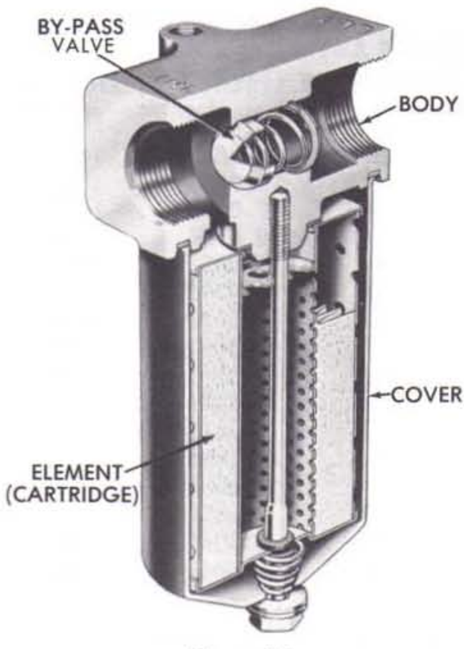


Figure 30

Types of Elements

There also are three basic types of filter elements; surface type, edge type, and depth type.

A surface type element is made of closely woven fabric or treated paper. Oil flows through the pores of the filter material and contaminants are stopped. The element in Figure 30 is a surface type filter.

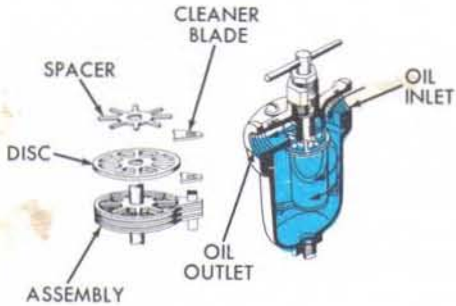


Figure 31

In the edge type filter (Figure 31), the oil flows through the spaces between paper or metal discs. The fineness of the filtration is determined by how close the discs are together.

A depth type element is composed of thick layers of cotton, felt or other fibers.

Micron Rating

The size of particle a filter will trap determines its micron rating. One micron is 39 millionths of an inch (0.000039) (one millionth of a meter). To visualize how small this is, think of a grain of salt which is 70 microns in size. The smallest particle a sharp eye can see is about 40 microns.

US and ASTM Std. Sieve No.	ACTUAL OPENING Inches	Microns
10	0.0787	2000
12	0.0661	1680
14	0.0555	1410
16	0.0469	1190
18	0.0394	1000
20	0.0331	840
25	0.0280	710
30	0.0232	590
35	0.0197	500
40	0.0165	420
45	0.0138	350
50	0.0117	297
60	0.0098	250
70	0.0083	210
80	0.0070	177
100	0.0059	149
120	0.0049	125
140	0.0041	105

US and ATM Std. Sieve No.	ACTUAL OPENING Inches	Microns
170	0.0035	88
200	0.0029	74
	0.0026	65
230	0.0024	62
270	0.0021	53
	0.0020	50
325	0.0017	44
	0.0016	40
400	0.00142	36
	0.00118	30
550	0.00099	25
625	0.00079	20
	0.00059	15
1,250	0.000394	10
1,750	0.000315	8
2,500	0.000197	5
5,000	0.000099	2.5
12,000	0.0000394	1

Figure 32

In today's hydraulic systems, we often recommend filtering to 10 microns or less. A screen used to filter this fine would have to be 1250 mesh; a one-micron screen would be 12,000 mesh (Figure 32).

There are two ways of expressing a filter's micron rating. Absolute micron rating tells us the size of the actual largest pore in the filter. A filter with a 10-micron absolute rating would not pass anything larger than 10 microns. However, a filter usually stops many particles much smaller than its absolute rating . . . particularly as contaminants begin to collect on the element. Therefore, we also have a nominal rating. For example, if a filter rated absolutely at 25 microns traps most particles down to 10 microns, we would rate it nominally at 10 microns. Absolute rating is critical only in highly sophisticated systems where it is essential not to pass any particle above a certain size.

Location

Filters in most mobile equipment circuits are located in the tank return line (Figure 33). There they trap wear products and other contaminants before the oil returns to the reservoir. This location also permits using a low-pressure type filter.

There is a pressure drop across a filter. With fine filtration, it may be 25 psi (1.72 bar) or more. This will induce a back pressure on the actuator.

Coarser filters with very low pressure drop may be used with CAUTION in the inlet line. A fine filter cannot be used there since it would starve the pump.

Flow Conditions

When we say that a filter is full-flow, we mean that all the fluid entering the filter passes through the element. In a partial-flow or proportional flow filter, some of the fluid goes through the filter and some bypasses it.

OFM FILTERS

Vickers OFM Series Filters (Figure 34) are full-flow at lower flow rates with a bypass valve to prevent a dirty element from stopping flow. As flow rate approaches rated maximum, the bypass valve partially opens bypassing part of the flow. Full flow filtration at any flow rate can be achieved by using large filters or connecting more filters in parallel.

Flow, as you can see, is outside-in; that is, from around the element to the center, then to the outlet. (This is typical of most filters.) The by-pass valve is spring loaded to stay closed unless pressure reaches a predetermined value. Then the valve will open to route part or all of the flow directly through the filter body.

The filter element or cartridge can be replaced by removing its cover.

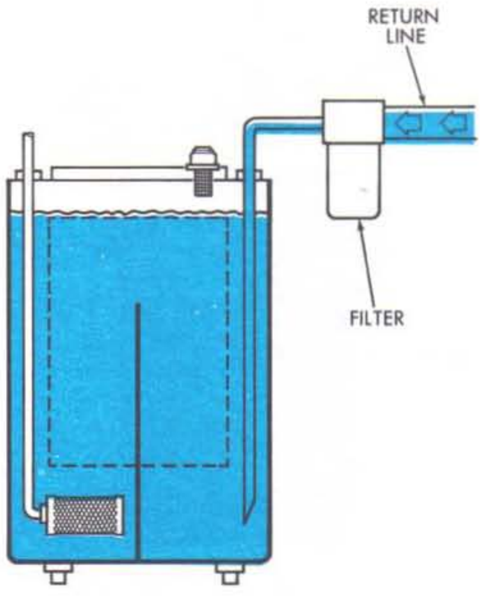


Figure 33

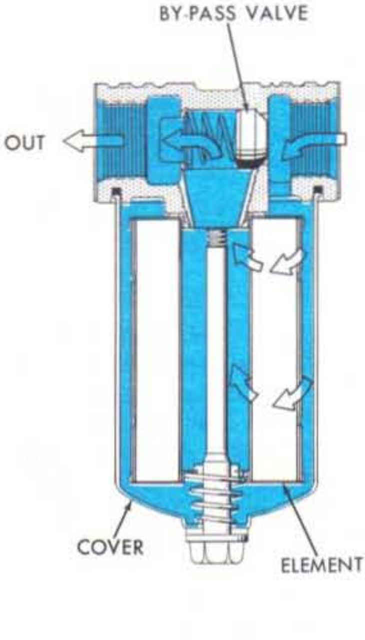


Figure 34

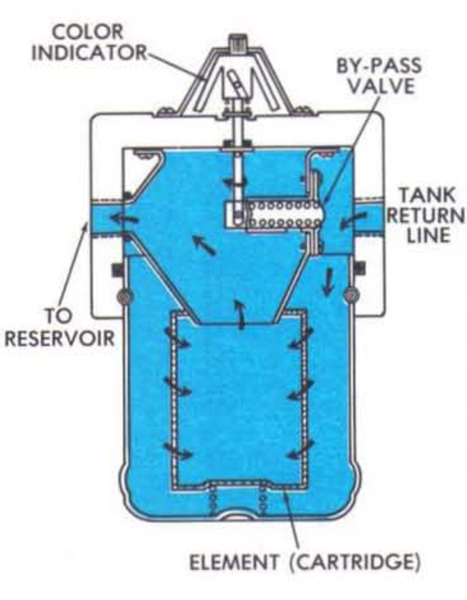


Figure 35

OF*21 FILTERS

These filters (Figure 35) feature a rotary type color indicator which is controlled by the bypass valve. The indicator shows the condition of the element through transparent windows in a protective hood. Green indicates a closed bypass and clean element. Yellow warns that pressure drop is increasing because the element is loading. Red indicates the bypass has opened and the element should be replaced. The bypass valve allows fluid to bypass the element when pressure drop across the element exceeds 25 psi (1.72 bar).

A "memory" condition can be obtained by removing the hood and indicator, rotating the indicator 180°, and reassembling. The indicator then functions in one direction only. It rotates to the maximum opening of the bypass valve and remains in that position until manually returned by rotating a knob atop the hood. This feature allows determination of element condition when machinery is shut down.

KEEPING IT COOL

We have seen how it would be ideal for the reservoir to maintain a low operating temperature in the fluid . . . and that this is often not practicable in mobile machinery. An external oil cooler is a necessity in many mobile circuits.

If the oil is permitted to get too hot, it thins and loses some of its lubricating ability. Also, excessive heat speeds oxidation of the oil and can lead to all sorts of messy conditions, such as corrosion, sludge and varnish formation.

With today's hydraulic oils, we can operate in the 180° F (82° C) range, while a few years ago, 120° F (49° C) was the highest recommended temperature. Whenever the system temperature exceeds 160° F (71° C) a cooler is probably a good investment in oil life.

AIR COOLERS

An air cooler can be a length of tubing fitted with a number of fins for maximum heat transfer. The oil is routed through the tube and gives up its heat to the surrounding air. In vehicles with internal combustion engines, it is often practical to use a radiator bottom tank for cooling the hydraulic oil.

WATER TYPE COOLERS

A water type cooler (Figure 36) generally will have a greater cooling capacity than an air cooler. In this design, water is circulated around the cooling element and the hydraulic oil is pumped through the element. Heat in the oil is transferred to the cooler water.

The use of water coolers on mobile equipment is limited. Cool water just isn't that available.

Figure 36

ACCUMULATORS

We learned in Chapter 1 that the speed at which we can move a load is proportional to the fluid flow to the actuator. In other words, if we supply a cylinder with 10 gpm (37.85 L/min) of oil, it will move twice as fast as with five gpm (18.92 L/min). When a system is operated intermittently, it is often possible to charge another kind of oil storehouse with a low gpm (L/min) pump and then to release the oil at a higher flow rate for fast action when required.

WHAT IT DOES

An accumulator is a device that stores or accumulates oil under pressure to operate an actuator by itself or to supplement pump delivery. It can also be used to absorb and cushion shocks or surges in a system; or to operate a system requiring very smooth action.

Some kinds of accumulators are weight-loaded cylinders, spring-loaded cylinders, and gas charged accumulators.

WEIGHTED ACCUMULATOR

Though a weighted accumulator is too big and heavy to use on a mobile system, it still deserves a brief mention. A weighted accumulator is the only type that maintains a uniform pressure regardless of the volume of oil it contains. It consists of a piston fitted in a cylinder with provision for applying weights to the top of the piston (Figure 37). Oil pumped under the piston then is subject to pressure proportional to the weight. Since the load on the piston doesn't change, pressure is constant throughout the piston stroke.

SPRING LOADED ACCUMULATOR

If we substitute a spring for the weights (Figure 38), we will still subject oil pumped under the piston to pressure. Now, though, we'll have a variable load on the piston. The force of a spring is equal to the spring rate multiplied by the distance the spring is compressed:

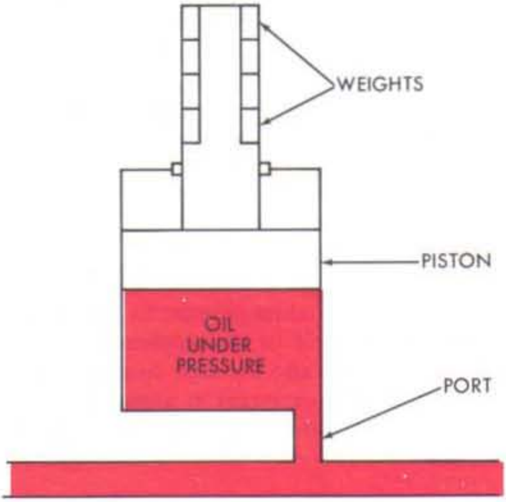


Figure 37

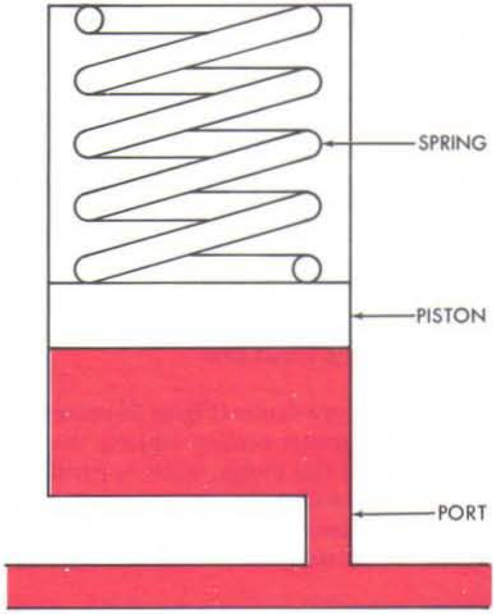


Figure 38

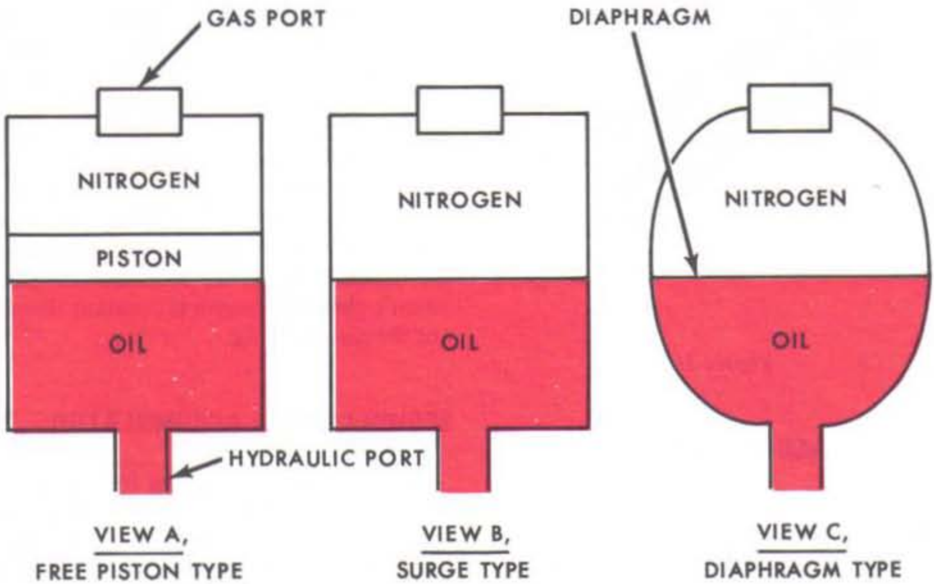


Figure 39

$$\text{Spring Force} = \text{Rate}\ \frac{\text{pounds}}{\text{inch}} \times \text{Distance (inches)}$$

As the piston travels up, the spring force increases and pressure rises proportionately.

GAS CHARGED ACCUMULATOR

Now suppose that we replace the spring with a charge of gas under pressure as in the free-piston accumulator (Figure 39, view A). The gas most often used is nitrogen. The gas charge maintains a load on the piston, resulting in pressure on the fluid under the piston.

It is possible even to eliminate the piston as in the surge-type accumulator (Figure 39, view B). The gas then will maintain pressure directly on the oil. In this design, the accumulator must always be mounted vertically and must always have some quantity of oil left in it. Otherwise the gas would leak out the oil port.

A third kind of gas-charged accumulator is the diaphragm type (Figure 39, view C). It is spherical in shape. A diaphragm made of synthetic rubber separates the gas and oil chambers.

All gas charged accumulators obey Boyle's Law. Boyle's Law, in simple terms, tells us that pressure in a gas increases with compression. Thus, as we add more liquid and compress the gas charge more, pressure rises. So again, we do not have a constant pressure accumulator.

BLADDER ACCUMULATORS

Vickers bladder type accumulators are another design of gas charged accumulator (Figure 40). The synthetic rubber bladder is molded to an air valve in the top of the high-pressure shell. The poppet-type valve in the oil port is normally held open by a spring to allow the accumulator to fill and discharge. If all the liquid is discharged, the bladder forces the valve closed. This prevents the bladder from extruding into the port.

Accumulators are rated by volume capacity in cubic inches of oil and by pressure.

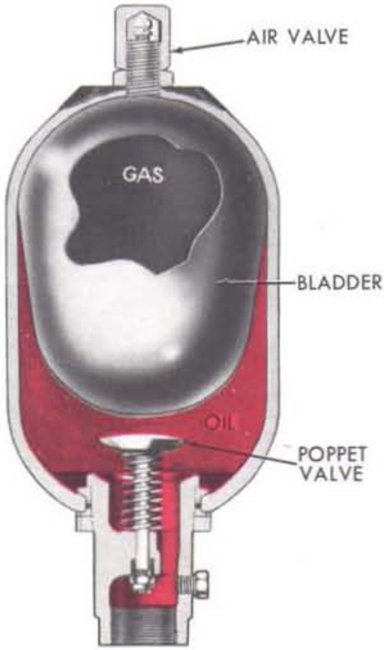


Figure 40

CHAPTER 3
Principles of Pump Operation

The pump is probably the most misunderstood component in hydraulics, and can be among the most complex. In fact, if the components in a hydraulic system ever became characters in a detective story, the pump would certainly be the villain. Since it is the heart of the system, and none of the other actuators can function if it isn't operating, the pump often becomes suspect anytime something goes wrong.

It is fairly safe to say that the pump has the greatest opportunity to fail. It may be started up at the beginning of a work day and run continuously until the end. It is the first component exposed to any contamination that finds its way into the reservoir. Because we're packing so much performance into small units, the pump is more susceptible to damage from improper operation or maintenance. Yet with proper selection and operation, and with good maintenance practices, the pump can become a most reliable component.

With experience and good maintenance, one can predict when a pump will need rebuilding. If the rebuilding is part of the maintenance program, there are virtually no actual failures.

In this chapter, we will strive for a good understanding of this most important component in the system. We will see what the pump does and what it doesn't do; how pumps are rated and selected; how various kinds of pumps work; and what the operating characteristics of the various designs are. Recommendations on pump start-up are in Appendix B.

PURPOSE OF THE PUMP

Fundamentally, the pump's purpose is to push on the hydraulic fluid and create flow. We say that the pump converts mechanical energy from the prime mover (engine or electric motor) into pressure energy in the fluid. The hydraulic energy is used then to operate an actuator, often with very precise control.

It is common practice to refer to the pump as the source of pressure in the hydraulic system. However, to assume that any loss of pressure must be caused by the pump is not necessarily true.

The pump does create pressure in that it applies the push that causes flow. But to create pressure, there must be a resistance to flow. Further, if the resistance is a load on an actuator, only enough pressure is created to handle the load. Pressure can be lost through any alternate flow path that offers less resistance to flow. So a leak in another component, a valve for instance, is a more likely cause of pressure loss than a defective pump.

Remember that the pump is there to cause flow. Where the flow goes depends on the other parts of the system.

WHAT MAKES A PUMP?

The essentials of any hydraulic pump (Figure 41) are:

- An inlet port which is supplied fluid from a reservoir or other source.
- An outlet port connected to the pressure line.
- Pumping chamber(s) to carry the fluid from the inlet to the outlet port.
- A mechanical means for activating the pumping chamber(s).

In most rotary hydraulic pumps, the design is such that the pumping chambers increase in size at the inlet, thereby creating

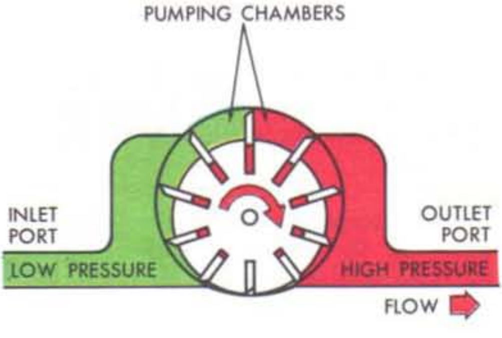


Figure 41

a partial vacuum. The chambers then decrease in size at the outlet to push the fluid into the system. The vacuum at the inlet is used to create a pressure difference so that fluid will flow from the reservoir to the pump. However, in many systems, the inlet is charged or supercharged; that is, a positive pressure rather than a vacuum is created by a pressurized reservoir, a head of fluid above the inlet, or even a low pressure charging pump.

There are many different basic types of pumps used in pressure hydraulic systems. We will examine some of these after we consider how pumps are classified and rated.

PUMP CLASSIFICATIONS AND WHAT THEY MEAN

POSITIVE AND NON-POSITIVE DISPLACEMENT

Our first division of pump classifications tells us whether the pump inlet is sealed from the outlet.

- If the inlet and outlet are connected hydraulically so that the fluid can recirculate in the pump when pressure builds up, the pump is non-positive displacement.
- If the inlet is sealed from the outlet, the pump will deliver fluid anytime the inlet is kept supplied and the pump is driven. Such a pump is classified as positive delivery or positive displacement, and requires a relief valve to protect it from pressure overloads.

Centrifugal Pumps are Non-Positive

Most non-positive displacement pumps operate by centrifugal force (Figure 42). Fluid is supplied to a rotating impeller at or near its center. The impeller blades or vanes start the fluid revolving and centrifugal force accelerates the fluid to the outer interior of the housing.

The housing is spiral or volute in shape, and the moving fluid follows the increasing diameter to the outlet. There is always clearance between the impeller and housing, so that the inlet and outlet are connected hydraulically.

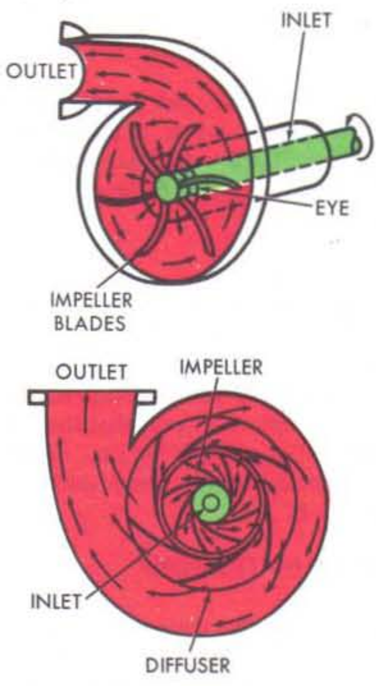


Figure 42

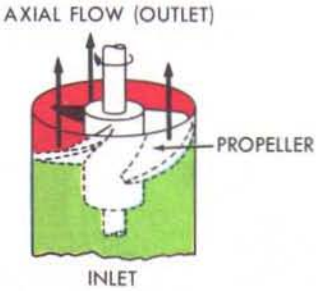


Figure 43

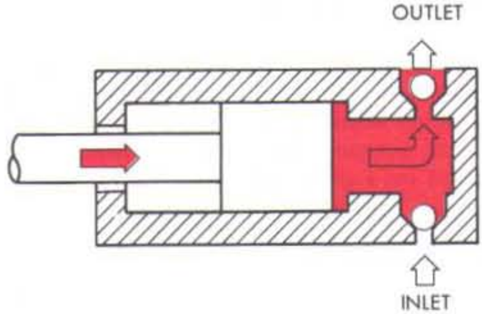


Figure 44

A propeller pump (Figure 43) also is a non-positive displacement pump. It is different from the centrifugal pump in that the propeller blades sweep the fluid axially through the pump rather than radially. It operates just like a venting fan enclosed in a tube, except that it moves liquid instead of air.

Positive Displacement Pumps

The single-piston type pump we illustrated in the hydraulic jack circuit (Figure 44) is a positive displacement pump. As soon as the inlet check valve is forced against its seat on the pumping stroke, the inlet is sealed. Except for leakage, fluid can only flow to the outlet regardless of the pressure. This is true of all positive displacement pumps whether reciprocating or rotary.

Characteristics

There are three contrasting characteristics in the operation of positive and non-positive displacement pumps:

- A non-positive displacement pump provides a smooth, continuous flow. A positive displacement pump has a pulse with each stroke or each time a pumping chamber opens to the outlet port.
- The delivery of a non-positive pump is reduced by pressure. A high enough outlet pressure can actually stop any output . . . the liquid simply recirculates inside the pump. In a positive displacement pump, pressure affects the output only to the extent that it increases internal leakage.
- A non-positive displacement pump, with its inlets and outlets connected hydraulically, cannot create a vacuum sufficient for self-priming; it must be started with the inlet line full of liquid and free of air. Positive displacement pumps often are self-priming when started properly (see Appendix B).

These are general characteristics, and also are reasons that non-positive displacement pumps are seldom used in mobile hydraulic systems. Occasionally you may find one used as a charging pump or replenishing pump. Or you may find several connected in series by feeding the output of one into the inlet of the next, and so on. This arrangement, which permits developing flow against high pressure, is not practical for mobile equipment.

We will consider other characteristics of positive displacement pumps when we examine the various designs.

PRESSURE RATING

One of the most important ratings a manufacturer must assign a pump is its pressure rating. Pressure rating tells us how much pressure the pump can safely withstand for a given time without damage to its parts. This in turn determines how much load the system can handle.

It's important to observe a pump's pressure rating. It is up to the relief valve to protect the pump from overload damage. If the operator gives the relief valve adjustment "a little twist" to get another half-yard of dirt in the bucket . . . and in so doing exceeds the recommended pressure limit . . . he risks accelerated wear, or even broken parts. Even though no apparent damage is done, a good general rule is that as pressure goes up, pump service life goes down.

FLOW RATINGS

Pressure is one of the two most important pump ratings . . . the other is flow. We can express the flow rating in terms of gallons per minute (gpm) (L/min) delivery or displacement ($in.^3$/rev) (mL/rev).

Displacement

Displacement is the volume which is "swept" by a pump in one revolution or cycle. For example, if there is no internal leakage the displacement of a rotary pump is equal to the volume of fluid discharged in one revolution. The usual way to express displacement is in cubic inches (millilitre) per revolution for a rotary pump or cubic inches (millilitre) per cycle for a reciprocating pump.

The actual volume of fluid discharged by a pump per revolution or cycle is equal to the displacement minus the volume lost due to internal leakage. With positive displacement pumps the internal leakage is low so that the volume discharged per revolution is approximately equal to displacement

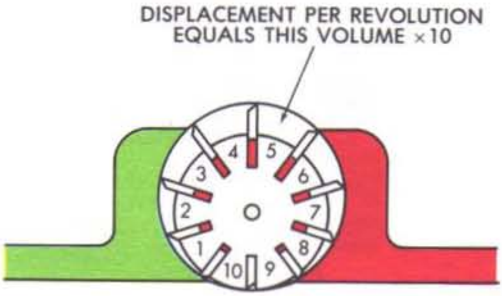


Figure 45

(Figure 45). In fact, the lower the operating pressure, the lower will be internal leakage so volume discharged per revolution approaches the pump displacement at zero outlet pressure.

Gpm Delivery

While the term displacement is most often used in discussions of pump size, the term volume also is common. Volume, of course, refers to an amount of fluid . . . but volume also is used informally to refer to flow rate, or delivery in gallons per minute (gpm) (litres per minute (L/min)).

Remembering that there are 231 cubic inches in a gallon, we can easily convert displacement to gpm if we know the pump drive speed in revolutions per minute (rpm).

$$\text{Delivery (gpm)} = \frac{\frac{\text{Disp. (cu. in.)}}{\text{Rev}} \times \text{rpm}}{231}$$

$$\text{Delivery (L/min)} = \frac{\frac{\text{Disp. (mL)}}{\text{Rev}} \times \text{rpm}}{1000}$$

Thus, a pump with a displacement of 2 cu. in./rev. (32.8 mL/rev.) driven at 1200 rpm has a delivery of about 10.4 gpm (39.4 L/min):

$$\text{Delivery} = \frac{2 \times 1200}{231} = \text{approx. } 10.4 \text{ gpm}$$

$$(\text{Delivery} = \frac{32.8 \times 1200}{1000} = \text{approx. } 39.4 \text{ L/min})$$

We can see easily from this that increasing either the pump displacement or the drive speed will increase the pump delivery. Thus, when the bucket or lift fork isn't moving fast enough to suit the operator, he simply speeds up his engine and drives the pump faster. Pump delivery then increases and the actuator moves the load faster.

Gpm Rating Conditions

Since it is very common to drive the pump at varying speeds on mobile equipment, we must have a standard speed to establish the nominal gpm rating of the pump. In fact, we have a set of standard rating conditions for hydraulic pumps. They are:

CONDITIONS	VANE PUMP	PISTON PUMP
Drive Speed	1200 rpm	1800 rpm
Outlet Pressure	100 psi (6.9 bar)	100 psi (6.9 bar)
Inlet Pressure	0 psi (0 bar)	0 psi (0 bar)

If we say that a pump is a 15-gpm (56.8 L/min) pump, we mean that it delivers 15 gpm (56.8 L/min) with these conditions only. To find the delivery at other drive speeds, we use this relationship:

$$\text{Delivery at rpm} = \frac{\text{Rated gpm (L/min)} \times \text{rpm}}{\text{Rated Speed (rpm)}}$$

At 600 rpm, then, our 15 gpm (56.8 L/min) vane pump would deliver only 7½ gpm (28.4 L/min); but at 3600 rpm, it would deliver 45 gpm (170.4 L/min).

- At 600 rpm: Delivery =

$$\frac{15 \times 600}{1200} = 7.5 \text{ gpm}$$

$$\left(\frac{56.8 \times 600}{1200} = 28.4 \text{ L/min}\right)$$

- At 3600 rpm: Delivery =

$$\frac{15 \times 3600}{1200} = 45 \text{ gpm}$$

$$\left(\frac{56.8 \times 3600}{1200} = 170.4 \text{ L/min}\right)$$

This gives you an idea of the extreme flexibility we have in mobile equipment, even with a constant displacement pump.

FIXED AND VARIABLE DISPLACEMENT

We add considerably to the pump's flexibility when we are able to vary its displacement.

Many types of pumps are built in both fixed (or constant) displacement and variable (or adjustable) displacement versions.

- In a fixed displacement pump, the gpm (L/min) output can be changed only by varying the drive speed.

- In a variable displacement pump, there is provision for changing the size of the pumping chamber(s). The gpm (L/min) delivery then can be changed by moving the displacement control or changing the drive speed; or both.

To generalize, we use fixed displacement pumps in open center systems. These are systems where the pump output has a free-flow path back to the reservoir in the neutral condition of the circuit. Variable displacement pumps can be used in closed center systems. These are systems where the pump continues to operate against a load in the neutral condition. These are generalizations, however, and there are exceptions.

We have theorized that a pump delivers oil equal to its displacement each cycle of revolution. Actually, the true output is less than the displacement because of slip.

Slip is the leakage of oil from the pressure outlet to a low-pressure area or back to the inlet. It is often referred to simply as internal leakage. A passage used to permit leakage oil to return to the inlet or to the reservoir is known as a drain passage.

Some slip is designed into all pumps for lubrication. More slip will occur as a pump begins to wear.

Slip also increases with pressure. Remember that oil flow through a given orifice size depends on the pressure drop. An internal leakage path is the same as an orifice – so if pressure increases, there will be more flow through the leakage path and proportionately less flow from the outlet port. We refer to any increase in slip as a loss of efficiency.

VOLUMETRIC EFFICIENCY

Volumetric efficiency is the ratio between actual pump delivery at a given pressure and rated pump delivery.

$$\text{Efficiency} = \frac{\text{Actual Delivery}}{\text{Rated Delivery}}$$

It is usually expressed as a percentage.

If our 15 gpm pump actually delivers 15 gpm (56.8 L/min) at 100 psi (6.9 bar) but only 12 gpm (45.4 L/min) at 2000 psi (138 bar), we say it is 100% efficient at 100 psi (6.9 bar), and 80% efficient at 2000 psi (138 bar).

$$\text{Efficiency} = \frac{12}{15} = \frac{4}{5} = 0.8 = 80\%$$

$$\text{Efficiency} = \frac{45.4}{56.8} = 0.8 = 80\%$$

Overall pump efficiency can be expressed as output power divided by input power. Power of course is proportional to delivery (gpm) (L/min), so this includes volumetric efficiency as well as mechanical efficiency.

POWER RATING

In Chapter 1, we introduced this relationship between power, pressure and flow rate:

$$\text{hp} = \text{gpm} \times \text{psi} \times 0.000583$$

$$(\text{kW} = \text{L/min} \times \text{bar} \times 0.00167)$$

You'll remember that we related this formula to the power used in a hydraulic circuit to move a load. The power required to drive the pump naturally would be higher because efficiency is lost both in the pump and through friction and leakage in the system.

We can apply the power formula to the complete system, less the pump, by measuring the flow and pressure at the pump outlet. Then, if the pump efficiency is known, the total power needed to drive the pump can be computed.

For instance, 12 gpm (45.4 L/min) flowing at 2000 psi (138 bar) would require 14 horsepower (10.4 kW).

$$\text{hp} = 12 \text{ gpm} \times 2000 \text{ psi} \times 0.000583 = 14 \text{ hp}$$

$$(\text{kW} = 45.4 \text{ L/min} \times 138 \text{ bar} \times 0.00167 = 10.4 \text{ kW})$$

Input Power

If the pump is 80% efficient, the power needed to drive it is 17.5 hp (13 kW).

$$\text{Input hp} = \frac{\text{Output hp} \quad 14\ (10.4\text{kw})}{\text{Efficiency } 0.80} = \frac{17.5 \text{ hp}}{(13 \text{ kW})}$$

Often it is convenient to approximate input power to the pump as follows:

$$\text{Input hp} = \text{gpm} \times \text{psi} \times 0.0007$$

$$\text{Input kW} = \text{L/min} \times \text{bar} \times 0.002$$

This relationship assumes an overall efficiency of about 83 percent . . . a good average for most commonly-used mobile pumps at rated conditions.

PUMP NOISE

We have mentioned that the output of a positive displacement pump is pulsating . . . that there is a pulse at the outlet port each time a pumping chamber arrives to unload. These pulses result in the generation of sound pulses at the pumping frequency and at harmonics (multiples) of the pumping frequency.

However, the noise we hear is not just the sound coming directly from the pump. It includes the vibration and fluid pulsations produced by the pump as well. Pumps are compact, and because of their relatively small size, they are poor radiators of noise – particularly at the lower frequencies. Reservoirs, motors and piping, being larger, are better radiators so that pump-induced pulsations can cause them to radiate audible noise which actually exceeds that coming from the pump itself. To really control noise, therefore, the designer must carefully consider all three forms of pump noise – structureborne, airborne, and fluid-borne noise.

Pump Noise Frequencies

The largest noise energy component is at the pumping frequency, which is the shaft frequency times the number of pumping elements (vanes, pistons, gear teeth, etc.). Significant noise amplitudes also occur at many multiples of up to 15 or 20 times this frequency. Generally speaking, the amplitudes of these components tend to decrease as frequency increases. However, many pumps are too small to efficiently radiate the energy at the pumping frequency and the first few harmonics (multiples). Therefore, the direct sound of small pumps may peak at, say, the 6th harmonic while that of large pumps may peak at the 3rd.

Vibration and fluid pulse noise energy of such units is still highest at the pumping frequency, however. When this energy excites a surface that is larger than the pump (e.g., the side of the reservoir) the sound component at the pumping frequency may predominate. Unbalance in the pump, its drive motor, or the couplings joining them produce noise energy at shaft frequency, that is, shaft speed in rpm divided by 60 to give Hertz or cycles per second. Misalignments in this power train produce components at twice and four times this frequency.

PUMP TYPE, OPERATING CONDITIONS, AND DUTY CYCLE DETERMINE NOISE

When a designer observes that there are gear, vane, piston, screw and gerotor type pumps on the market, he expects to find one type that is inherently quieter than all the others. If this were true, all hydraulic component manufacturers would offer this type. Actually, low noise levels are the result of development effort; a well-developed pump of any type will be quieter than undeveloped pumps of any other type when operating under comparable conditions.

The proper selection of pump operating conditions provides another noise control opportunity. Pump speed has a strong effect on noise while pressure and pump size (displacement) have about equal, but smaller, effects. Since these three factors determine power, they provide a basis for a tradeoff for noise. To achieve the lowest noise levels, the designer should use the lowest practical speed (1000 or 1200 rpm where electric motors are used, a reducer gear for engine drives), and select the most advantageous combination of size and pressure to provide the needed power.

Poor inlet conditions which allow aeration or cause cavitation are also causes of pump noise and in addition will cause damage to the pump. Later paragraphs discuss the causes of these two conditions.

TYPES OF POSITIVE DISPLACEMENT PUMPS

Now that you are acquainted with pumps in general, we can go on to examine the various kinds of positive displacement pumps used in hydraulic systems. As you learned in earlier pages, there are both rotary and reciprocating pumps classified as positive displacement.

RECIPROCATING PUMPS

The operation of a simple reciprocating pump (Figure 44) has already been illustrated in this chapter and in Chapter 1. The pump illustrated delivers oil only when stroked in one direction. There also are designs that permit pumping on both strokes.

However, reciprocating pumps are not easily adaptable to a rotary power source. Therefore, you won't find them on power-driven mobile machinery. All of the pumps we are about to look at are the rotary type.

VANE TYPE ROTARY PUMPS

In the vane type pump (Figure 46), a slotted rotor driven by a driveshaft rotates between closely fitted side plates, and inside of an elliptical or circle shaped ring. Polished, hardened vanes slide in and out of the rotor slots and follow the ring contour by centrifugal force. Pumping chambers are formed between succeeding vanes, carrying oil from the inlet to the outlet. A partial

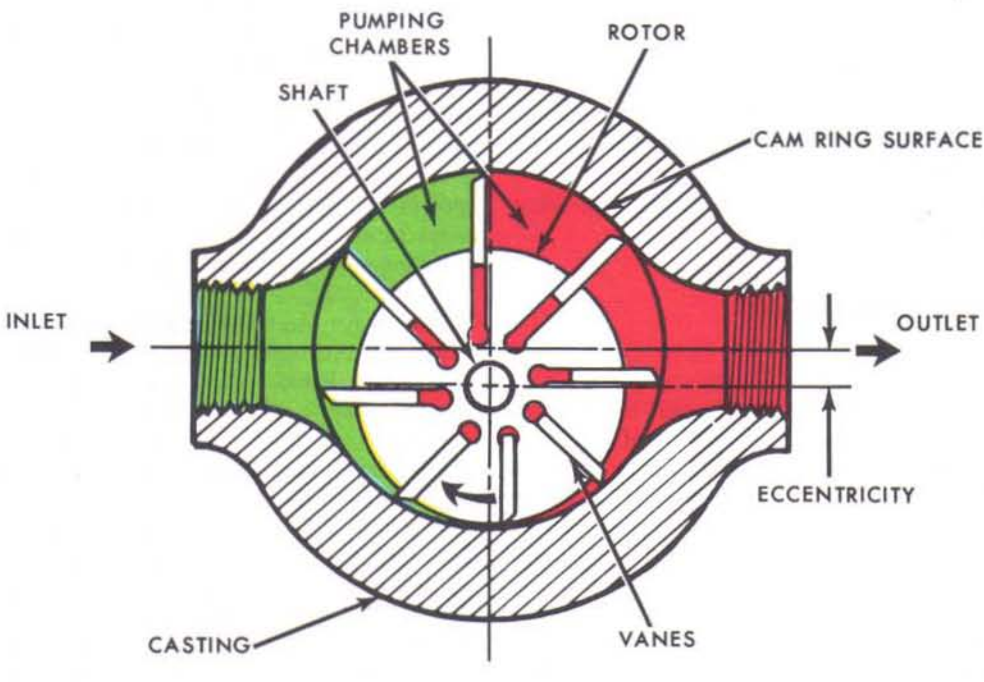


Figure 46

vacuum is created at the inlet as the space between vanes opens. The oil is squeezed out at the outlet as the pumping chamber size decreases.

The normal wear points in a vane pump are the vane tips and the ring surface, so the vanes and ring are specially hardened and ground. The vane pump is the only design that has automatic wear compensation built in. As wear occurs, the vanes simply slide farther out of the rotor slots and continue to follow the ring contour. Thus efficiency remains high throughout the life of the pump.

Unbalanced Vane Pump

In the unbalanced design vane pump (Figure 46), the cam ring shape is a true circle which is on a different centerline from the rotor. The pump displacement depends on how far the rotor and ring are eccentric. The advantage of the true circle ring is that a control can be applied to vary the eccentricity and thus vary the displacement. The main disadvantage is that an unbalanced pressure at the outlet is effective against a small area of the rotor edge and thus imposes side loads on the shaft. Thus there is a limit on the pump size unless very large bearings and heavy supports are used. This pump is primarily used on machine tool applications.

Balanced Vane Pump

The balanced design vane pump (Figure 47) uses a stationary, elliptical, cam ring and has two sets of ports internally. A pumping chamber is formed between any two vanes twice in each revolution.

The two inlets are 180 degrees apart, as are the two outlets. Thus, back pressures against the edges of the rotor cancel each

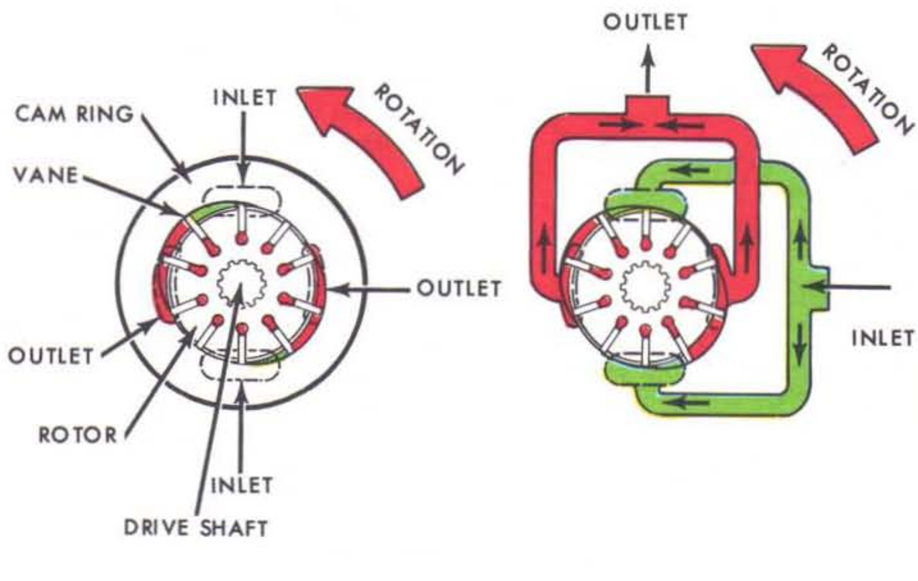


Figure 47

other. Large displacements can be obtained with a relatively small package size.

Design improvements in recent years that permit operation at high speeds and high pressures have made this type pump the most universally accepted in the mobile equipment field.

Vane Pump Displacement

You can readily see that the displacement of a vane type pump depends on the width of the ring and rotor and the "throw" of the cam ring (Figure 48).

Interchangeable rings are designed to permit converting a basic pump to several displacements. Usually it is only necessary to change the ring, but when you run out of "throw," some models have wider cartridges to permit a further increase in output.

Balanced design vane pumps all are fixed displacement. The unbalanced design can be built in either fixed or variable displacement.

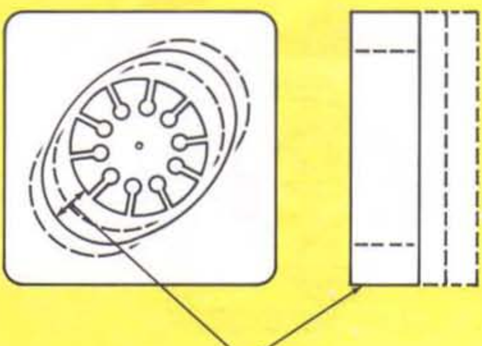


Figure 48

VANE PUMP CHARACTERISTICS

In general, vane pumps have good efficiency and durability if operated in a clean system and with the right kind of fluid. They cover the low to medium-high pressure, capacity and speed ranges. Package size in relation to output is small.

VICKERS PUMPS USED IN MOBILE APPLICATIONS

With that general look at pump designs completed, let's now go on to some specific designs of vane and piston pumps. In so doing, we will take a closer look at some refinements of operation and construction.

V10 AND V20 SERIES VANE PUMPS

The V10 and V20 series pumps (Figure 49) are fixed displacement, hydraulically balanced units ranging in size from 0.20 to 2.59 cubic inches per revolution (3.3 to 42.4 mL/rev.). At the SAE rating of 1200 rpm and 100 psi, flow rate is approximately 1 gpm (3.7854 L/min) for the smallest displacement unit and 13 gpm (49.2 L/min) for the largest unit.

The inlet port is in the pump body, which also supports the shaft bearing (Figure 49). The outlet port is in the pump cover. The cam ring is sandwiched between the body and cover. A machined surface on the body serves as one side plate for the pumping unit or cartridge. A ported pressure plate is fitted in the cover to serve as the other side plate. O-ring seals separate high pressure areas from the inlet, and a passage through the body returns leakage oil to the pump inlet.

Pressure Plate Operation

In operation, the pressure plate (Figure 50) is held against the ring and rotor by a

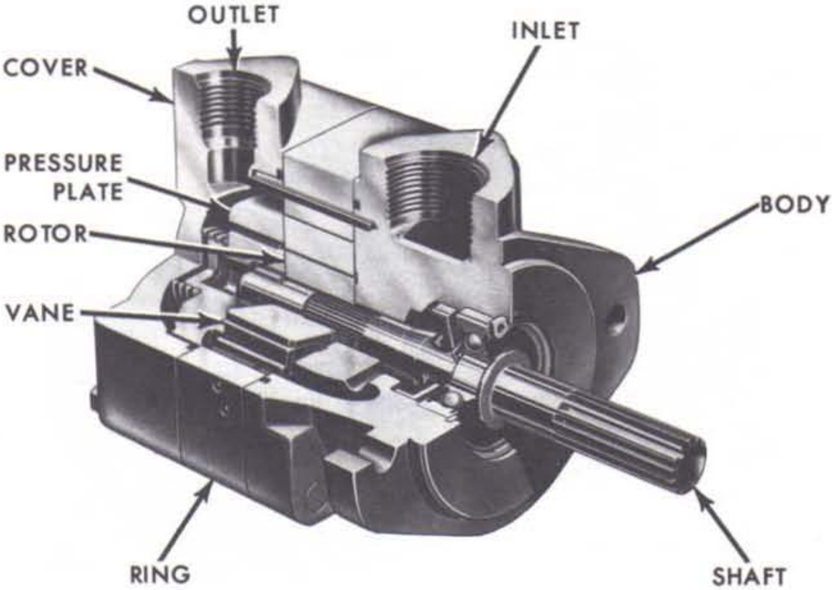


Figure 49

spring until pressure builds up in the system. Then the system pressure against the plate holds it against the cartridge. The plate is designed so that the forces against it offset the inherent tendency inside the cartridge to deflect it outward and so there is a tight seal against the side of the cartridge to provide the proper running clearances.

A second function of the pressure plate is to direct oil pressure through ports to feed under the vanes. This pressure extends the vanes radially so that they will maintain contact with the ring during pump operation.

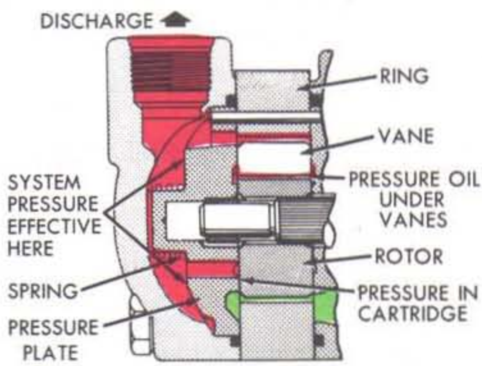


Figure 50

DOUBLE PUMPS

The operating principles of the double pumps (Figure 51) are identical to single

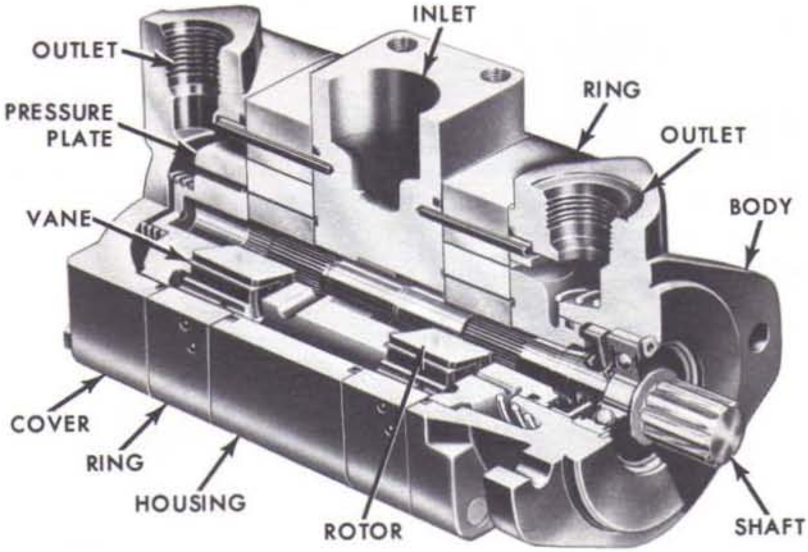


Figure 51

pumps. A large center housing serves as the inlet and the low-pressure side plate for both cartridges. The pressure plates are enclosed in the pump body and cover which contain the outlet ports. The shaft is supported by bearings in the body and inlet housing.

HIGH PERFORMANCE SERIES PUMPS

The high performance series pumps also are balanced vane pumps. They are built in four basic package sizes of single pumps. Delivery rates range from 12 to 109 gpm (45.4 to 412.6 L/min) at 1200 rpm and pressure ratings are 2000 to 3000 psi (138 to 207 bar) depending on model series. As with the V20 pumps, there are various gpm ratings in each basic size. Several series of double pumps also are built.

Cartridge Construction

The complete pumping cartridge in high performance pumps (Figure 52) is removable and replaceable as a unit. In fact, replacement cartridges from the factory are completely assembled and tested for quick installation without removing the pump from the vehicle. The support plates are bolted together with the side plates, ring, rotor and vanes sandwiched in between them.

Intra-Vane Design

The intra-vane design of high performance pumps (Figure 53) varies the outward force on the vanes due to system pressure so there is less force in the inlet or low pressure quadrants, where system pressure is not opposing centrifugal force. Each vane has a

Figure 52

small insert fitted into it at the bottom, with space between them for oil under pressure to exert an upward force on the vane. System pressure is constantly applied to the area between the vane and insert through porting in the side plate. The larger area below the vane will be subject to whatever pressure the top of the vane is subject to.

This is because of drilled holes in the rotor that permit pressure at the top of the vane to be sensed at the large area below it. Thus, in the pressure quadrants, system pressure is applied over the entire vane area, but in inlet quadrants, only over the small area between the vane and insert.

Construction and Assembly

In the single pump (Figure 54) the inlet port is in the cover and the outlet port is in

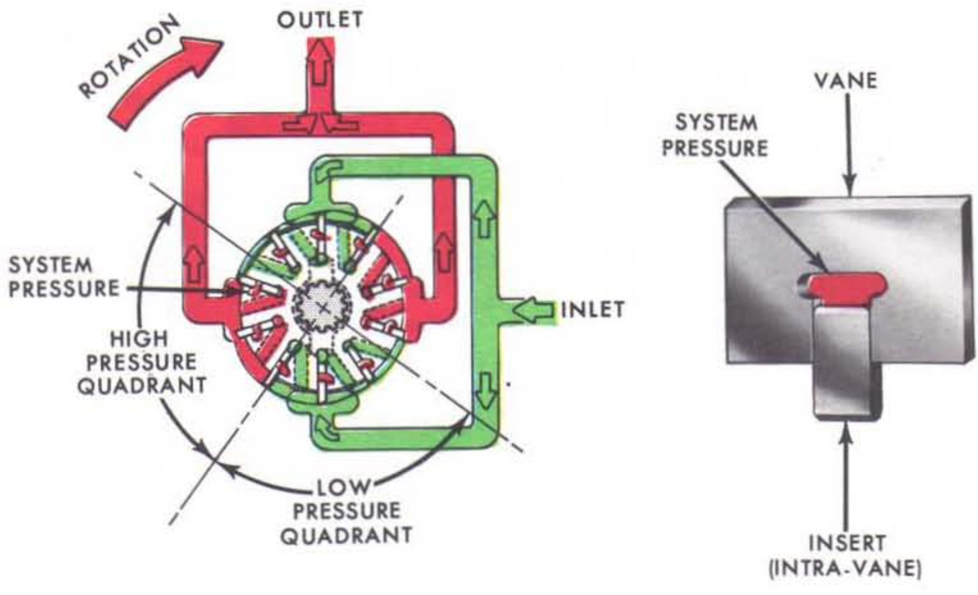


Figure 53

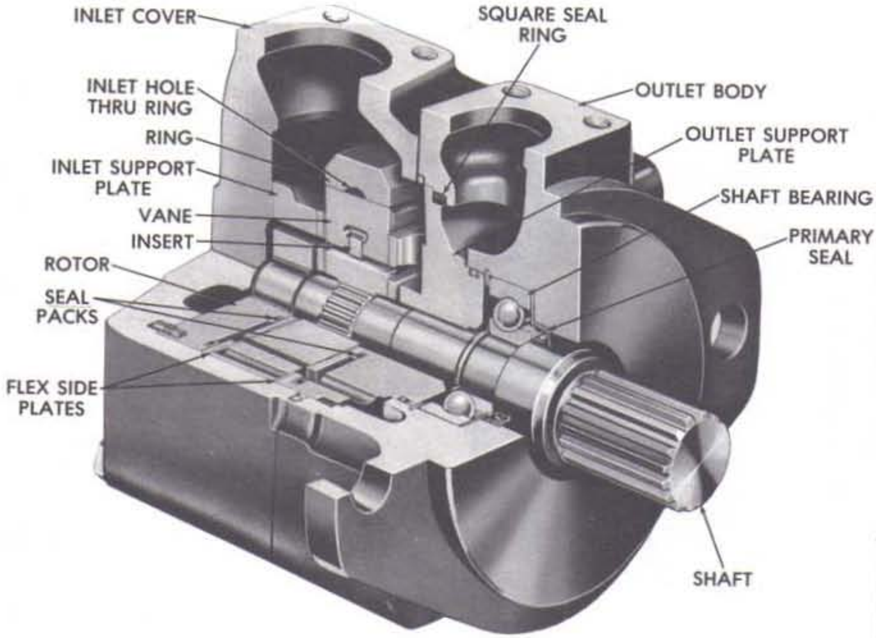


Figure 54

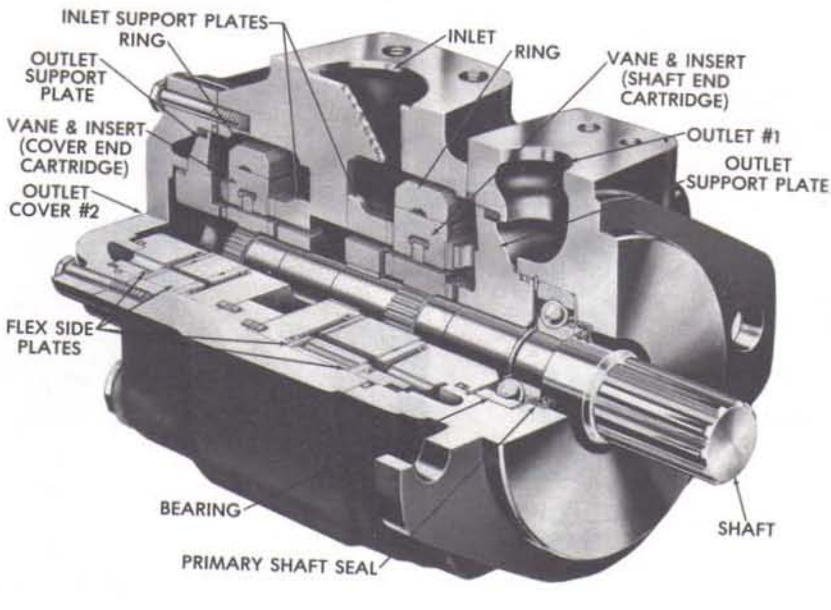


Figure 55

the body. The body supports the shaft ball bearing and seal. An "O" ring seal and a square cut teflon ring separate the high- and low-pressure cavities.

Double pumps (Figure 55) use the same bodies as single pumps. There is a bushing in the inlet support plate of the shaft-end cartridge to support the longer shaft needed to drive two rotors. Displacement can be changed by changing cartridges or rings.

Port Positions

The V10, V20, and high performance design pumps are built so the relative position of the ports can be changed easily. This is usually accomplished simply by removing the four cover screws and turning the cover.

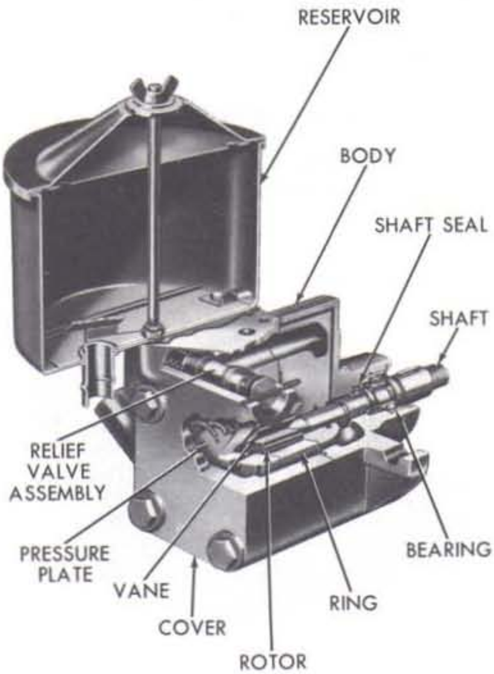


Figure 56

POWER STEERING PUMPS

Vickers power steering pumps (Figure 56) also are the balanced vane design. The pumping cartridge is sandwiched between the pressure plate in the cover and the body which forms the low-pressure side plate. The driveshaft is supported by a bearing. Power steering pumps, and some V10-V20 pumps include a built-in flow control and relief valve. We'll take a closer look at that feature in Chapter 7.

GEAR TYPE PUMPS

There are several kinds of pumps that fall in the broad classification of gear pumps. External gear pumps have the largest application in power transmission, but internal gear pumps also are used – particularly for automatic transmissions and power steering in automobiles. Lobe pumps are essentially external gear pumps with higher displacement.

EXTERNAL GEAR PUMPS

An external gear pump (Figure 57) consists essentially of two meshed gears in a closely fitted housing with inlet and outlet ports opposite each other. One gear is driven by the power source and in turning

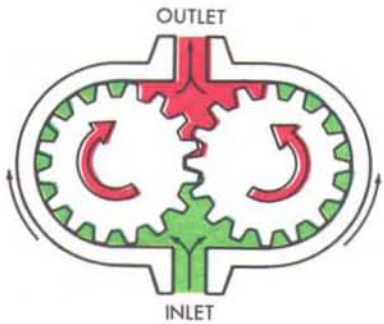


Figure 57

drives the other. As the gear teeth separate and travel past the inlet a partial vacuum is formed. Oil entering the inlet is carried to the outlet in pumping chambers formed between the gear teeth and housing. As the gear teeth mesh at the outlet there is no place for the oil to go but out.

Gear Design

Different tooth forms (Figure 58) can be used with gear pumps depending on the requirements. A spur gear pump is the easiest to make and is reversible. Helical gears also are reversible, can be run at higher speeds and have large displacements. Herringbone gears eliminate the side thrusts inherent in helical gears and decrease pulsations, but they are not reversible.

G*0 SERIES GEAR PUMP

The Vickers gear pumps (Figure 59) for mobile applications feature a three-piece housing design. Cast iron in the front and rear covers provides added strength and quieter operation; aluminum in the center section enables the gear teeth to wipe their own path during break-in for efficient sealing. An integral, one-piece shaft and gear combination provides high strength and permits use of a large, high capacity bearing.

INTERNAL GEAR PUMP

The internal gear pump (Figure 60) consists of an inner gear keyed to the drive shaft, a larger external gear, a crescent seal and a closely fitted housing. The two gears are not concentric, so as they rotate pumping chambers open up between them at the inlet and close off at the outlet. The crescent seals the inlet port from the outlet and both gears carry oil past it.

LOBE (ROTOR) PUMPS

A lobe or rotor pump (Figure 61) works the same way as an external gear pump except that external idler gears are needed to synchronize the lobes. You can easily see that displacement is higher than in a gear pump, but so is the opportunity to lose efficiency as the result of wear. High displacement lobe pumps are generally limited to moving large volumes of liquid. However, some of these pumps have more lobes, hence smaller displacement, and are used in low pressure systems.

PISTON-TYPE ROTARY PUMPS

A piston-type pump could properly be classified a rotary-reciprocating pump. In

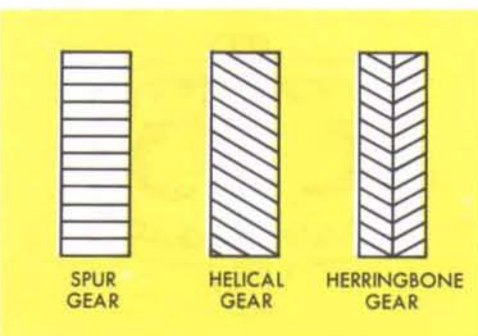


Figure 58

Figure 59

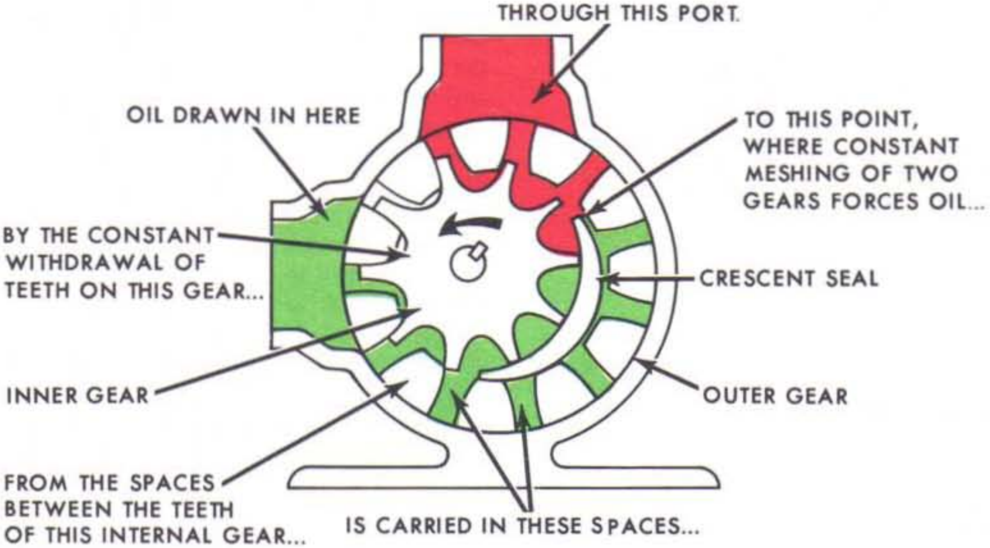


Figure 60

most piston pumps, several pistons (usually seven or nine) reciprocate in rotating cylinder barrels. The pumps are constructed so that the pistons retract while passing the inlet port to create a vacuum and permit oil to flow into the pumping chambers. They then extend at the outlet to push the oil into the system.

The two general classifications of piston pumps are radial and axial.

RADIAL PISTON PUMP

In a radial piston pump (Figure 62), the pistons are arranged like wheel spokes in a short cylindrical block. This cylinder block is rotated by the driveshaft inside a circular ring. The block turns on a stationary pintle that contains the inlet and outlet ports.

As the cylinder block turns, centrifugal force and supercharge pressure force the pistons outward and they follow the circular ring. The ring centerline is offset from the cylinder block centerline. The amount of eccentricity between the two determines the piston stroke and therefore the pump displacement. Controls can be applied to change the ring location and thereby vary the pump delivery from zero to maximum.

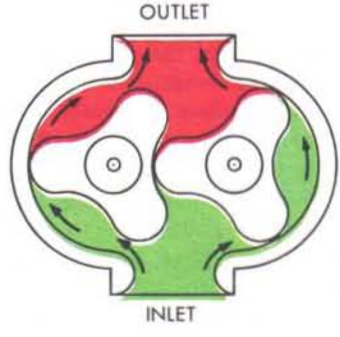


Figure 61

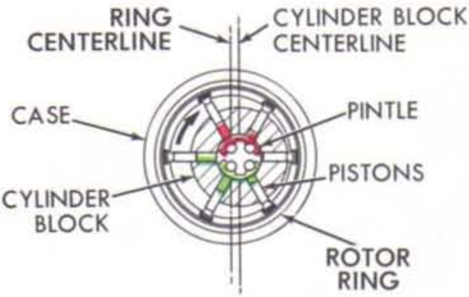


Figure 62

AXIAL PISTON PUMPS

In axial piston pumps, the pistons stroke axially, or in the same direction as the cylinder block centerline. Axial piston pumps may be the inline design or the angle design.

SWASH PLATE TYPE IN-LINE PISTON PUMPS

In an in-line piston pump (Figure 63, top), the shaft and cylinder block are on the same centerline. Reciprocation of the pistons is caused by a swash plate that the pistons run against as the cylinder block rotates. The driveshaft turns the cylinder block which carries the pistons around the shaft. The piston shoes slide against the swash plate and are held against it by the shoe plate. The angle of the swash plate causes the cylinders to reciprocate in their bores. At the point where a piston begins to retract, the opening in the end of the bore slides over the inlet slot in the valve plate and oil enters the bore through somewhat less than half a revolution. Then there is a solid area in the valve plate as the piston becomes fully retracted. As the piston begins to extend the opening in the cylinder barrel moves over the outlet slot and oil is forced out the pressure port.

In-Line Pump Displacement

The displacement of the pump depends on the bore and stroke of the piston, and the number of pistons. The swash plate angle (Figure 63, bottom) determines the stroke, which can be varied by changing the angle. In the fixed angle unit, the swash plate is stationary in the housing. In the variable unit, it is mounted on a yoke which can turn on pintles. Various controls can be attached to the pintles to vary pump delivery from zero to maximum. With certain controls, the direction of flow can be reversed by swinging the yoke past center. In the center position, of course, the swash plate is perpendicular to the cylinders and there is no piston reciprocation; therefore no oil is pumped.

IN-LINE PISTON PUMPS

A typical fixed displacement in-line pump is shown in Figure 64. The major parts are the housing, a bearing-supported driveshaft, a rotating group, a shaft seal, and the valve plate. The valve plate contains the inlet and outlet ports and functions as the back cover.

In the "rotating group" are the cylinder block which is splined to the driveshaft; a splined spherical washer; a spring; nine pistons, with "shoes"; the swash plate and the shoe plate. When the group is assembled, the spring forces the cylinder block against the valve plate and the spherical washer against the shoe plate. The piston shoes thus are held positively against the swash plate, insuring that the pistons will reciprocate as the cylinder turns. The swash plate, of course, is stationary in the fixed displacement design.

A variable displacement in-line pump is shown in Figure 65. Operation is the same as the fixed angle, except that the swash plate is mounted on a pivoted yoke. The yoke can be "swung" to change the plate

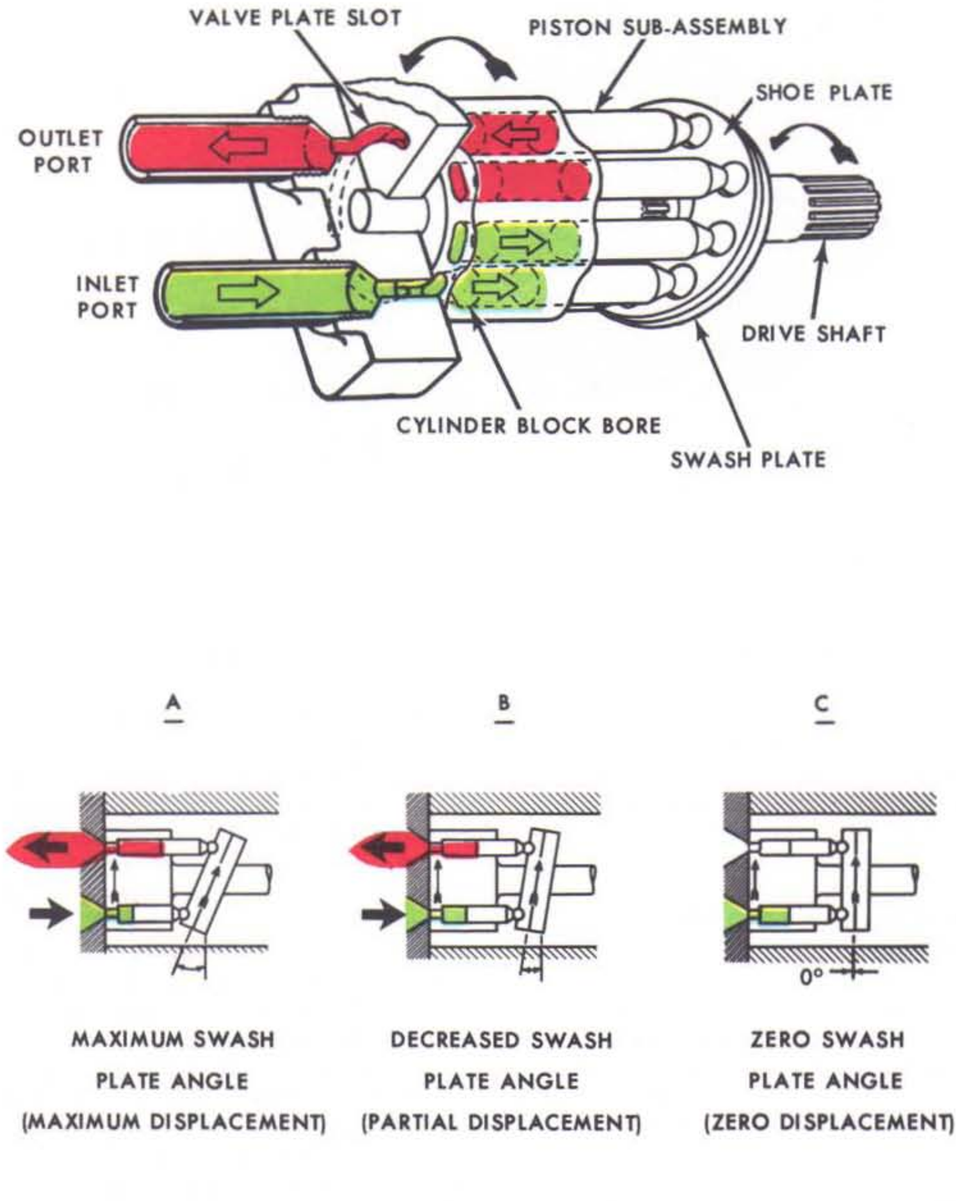

VALVE PLATE SLOT
PISTON SUB-ASSEMBLY
SHOE PLATE
OUTLET PORT
INLET PORT
DRIVE SHAFT
CYLINDER BLOCK BORE
SWASH PLATE
A
B
C
0°
MAXIMUM SWASH PLATE ANGLE (MAXIMUM DISPLACEMENT)
DECREASED SWASH PLATE ANGLE (PARTIAL DISPLACEMENT)
ZERO SWASH PLATE ANGLE (ZERO DISPLACEMENT)

Figure 63

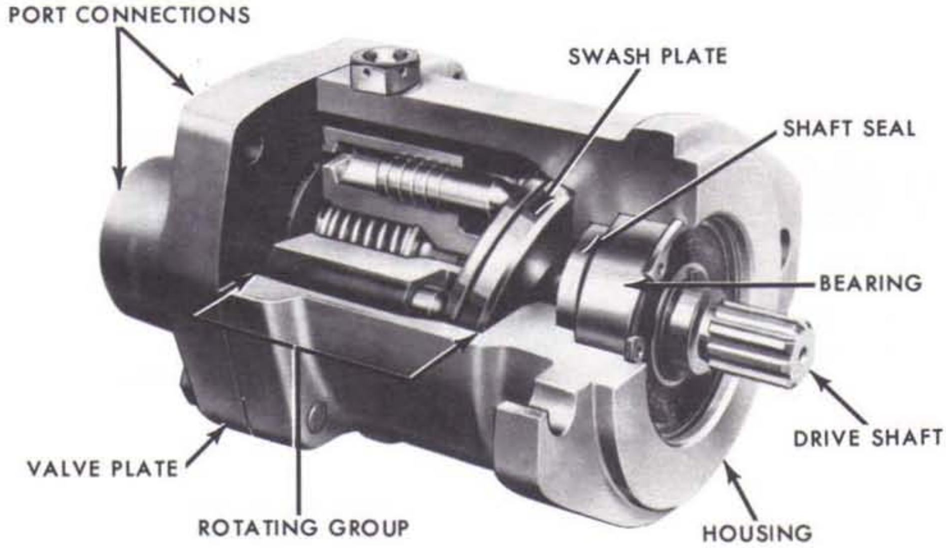


Figure 64

angle and thus change the pump displacement.

The yoke can be positioned manually with a screw or lever, or by a servo or compensator control. The pump in Figure 65 has a compensator control.

PRESSURE COMPENSATOR OPERATION

The pressure compensator control positions the yoke automatically to limit output pressure. It consists of a valve balanced between a spring and system pressure; and a spring-loaded yoke-actuating piston which is controlled by the valve.

The yoke return spring initially holds the yoke to a full delivery position. Pressure from the discharge of the pump is continually applied through passage "A" to the end of the compensator valve. The adjustment spring acting on the opposite end of the compensator valve opposes the fluid pressure in passage "A". When pressure in passage "A" is sufficient to overcome the load of the adjustment spring, the compensator valve moves, allowing fluid to enter the yoke actuating piston. The fluid then forces the yoke actuating piston to move the yoke and decrease the stroke of the pistons in the cylinder block. Thus the delivery of the pump is reduced. If the pressure in passage "A" (outlet pressure) decreases, the adjustment spring will move the compensator spool, closing off passage "A" and permitting fluid in the yoke actuating piston to drain through passage "B" to the housing. The yoke return spring then moves the yoke back toward its maximum delivery position.

The pump compensator control thus reduces pump output to only the volume

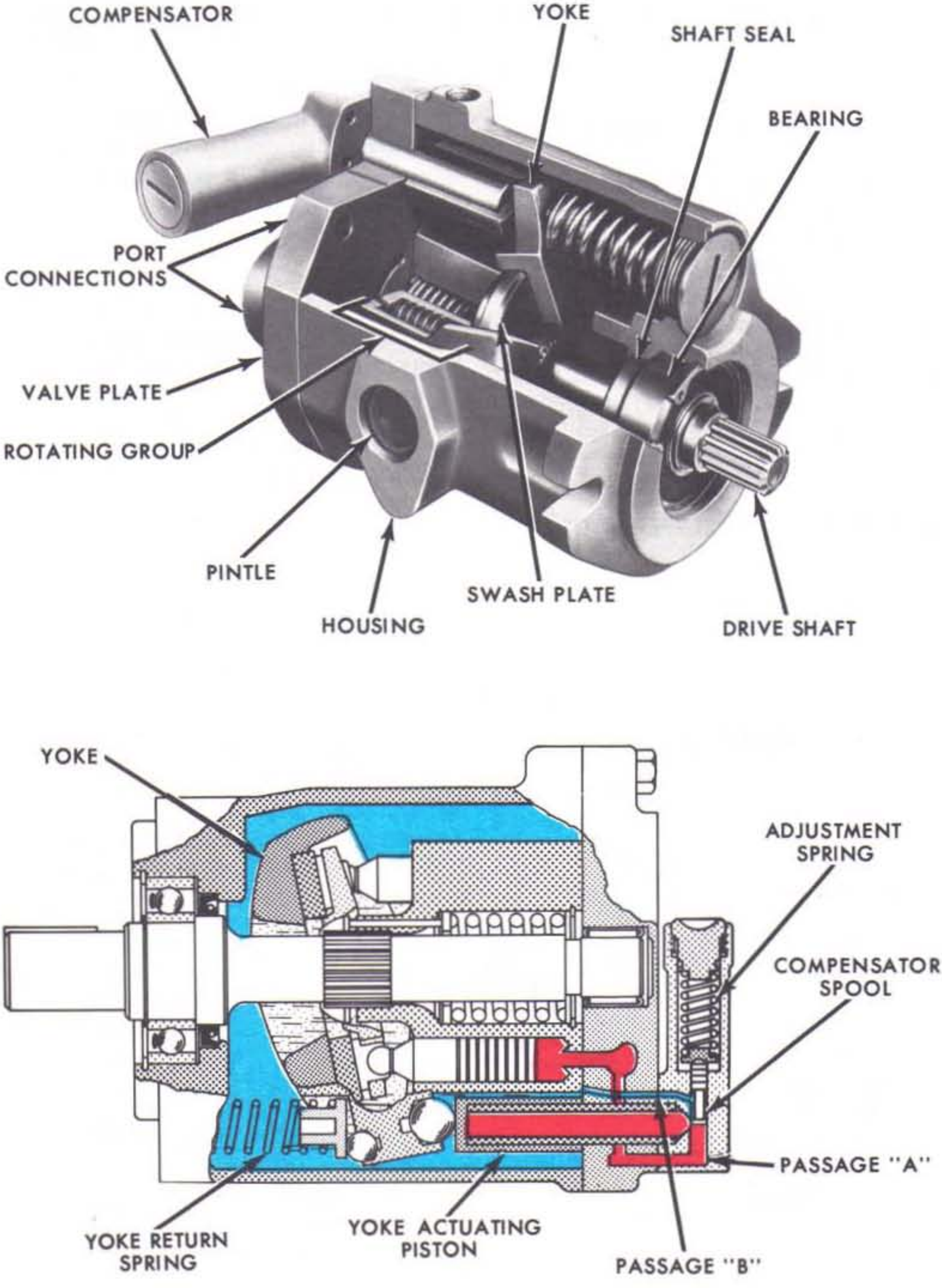

COMPENSATOR
YOKE
SHAFT SEAL
BEARING
PORT
CONNECTIONS
VALVE PLATE
ROTATING GROUP
PINTLE
HOUSING
SWASH PLATE
DRIVE SHAFT
YOKE
ADJUSTMENT
SPRING
COMPENSATOR
SPOOL
PASSAGE "A"
YOKE RETURN
SPRING
YOKE ACTUATING
PISTON
PASSAGE "B"

Figure 65

required to maintain a pre-set maximum pressure. Maximum delivery occurs when pressure is less than the compensator setting.

WOBBLE PLATE IN-LINE PUMP

There is a variation of the in-line piston pump known as a wobble-plate pump. In this design, the cylinder barrel doesn't turn; the plate turns instead. Of course, being canted, the plate "wobbles" as it turns . . . and the wobbling pushes the pistons in and out of the pumping chambers in the stationary cylinder barrel. In a wobble plate pump, separate inlet and outlet check valves are required for each piston, since the pistons do not move past the ports.

BENT AXIS (ANGLE) TYPE PUMP

In the angle or bent-axis type piston pump (Figure 66) the piston rods are attached by ball joints to the driveshaft flange. A universal link keys the cylinder block to the shaft so that they rotate together but at an offset angle. The cylinder barrel turns against a slotted valve plate to which the ports connect. Otherwise pumping action is the same as the in-line pump.

Angle Pump Displacement

The angle of offset determines the displacement of this pump, just as swash plate angle determines an in-line pump's displacement. In fixed delivery pumps the angle is constant. In variable models, a yoke mounted on pintles swings the cylinder block to vary displacement. Flow direction can be reversed also with appropriate controls.

PISTON PUMP CHARACTERISTICS

Piston pumps range from low to very high capacity. Pressures range to 5000 psi

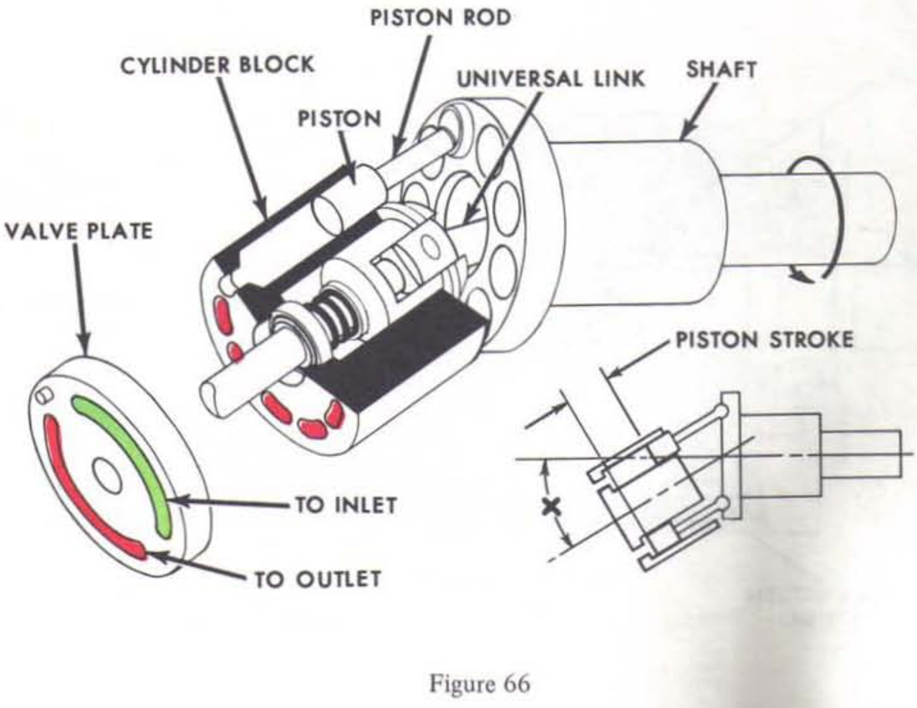


Figure 66

(345 bar) and drive speeds are medium to high. Efficiency is high and the pumps generally have excellent durability.

The pressure compensated pumps save horsepower and reduce heat generation because the flow is minimum at maximum pressure as compared to a fixed displacement pump where the flow goes over the relief valve when pressure is maximum.

Piston pumps can have a variety of controls in large displacement models, and they have wide usage in hydrostatic drives.

PUMP OPERATION RECOMMENDATIONS

Some of the harmful effects of overloading were mentioned when we discussed pressure rating. Among the dangers involved is the risk of excess torque on the driveshaft. We can define torque roughly as a circular force, and it depends on the pump displacement and the pressure in the system. An increase in either pressure or pump displacement will increase the torque on the shaft if the other remains constant. You'll often find that in a given package size, the higher gpm pump will have a lower pressure rating than the lower gpm pump. Sometimes a field conversion to get more speed out of the actuator will cause the pump to be overloaded. So remember that there are times when you have to sacrifice pressure to increase speed and vice-versa. If the sacrifice can't be made, it's best to go to a larger pump than risk an early failure.

AVOID EXCESS SPEED

Running the pump at too high a speed causes loss of lubrication. This can cause early failure. If the needed delivery requires a higher drive speed than the pump is rated at, a higher displacement pump should be used.

Excess speed also runs a risk of damage from cavitation.

AVOID CAVITATION

Cavitation is a condition where the available fluid doesn't fill the existing space. It often occurs in a pump inlet when conditions are not right to supply enough oil to keep the inlet filled. The resulting bubbles implode as they are exposed to system pressure at the outlet of the pump. Besides excessive speed, the reasons for this condition can be too much restriction in the inlet line, or the reservoir oil level too far below the inlet, or too high an oil viscosity.

Remember in Chapter 1 we figured that the weight of oil is equivalent to 0.4 psi (0.028 bar) per foot (0.3048 m) of height. We stated that a ten foot (3.048 m) head of oil (reservoir level ten feet (3.048 m) above the pump inlet) would charge the pump inlet to 4 psi (0.28 bar). It is equally true that if the oil level is ten feet (3.048 m) below the pump inlet, it will take a pressure difference of 4 psi (0.28 bar) just to raise the oil. This doesn't consider friction and restrictions in the line and in any inlet filters or strainers.

PUMP INLET VACUUM

The maximum allowed vacuum at the inlet is 5 in. Hg. for most pumps. Ideally there should be no vacuum or even a slight positive pressure at the inlet; otherwise cavitation can occur. Cavitation causes erosion of the metal within the pump and speeds deterioration of the hydraulic oil. A badly cavitating pump makes a very distinctive noise as the bubbles implode under pressure. Unfortunately, the noise often doesn't start until about ten inches of vacuum, but the damage is being done whether you can hear it or not. The only positive way to be sure a pump isn't cavitating is to check the inlet with a vacuum gage. Cavitation is prevented by keeping the inlet clean and free of obstructions, by using as large and short an inlet line as possible with minimum bends, and by operating within rated drive speeds.

CHARGED INLET

Another way to avoid cavitation is to pressurize the pump inlet. The easiest way to charge the inlet is by locating the reservoir above the pump. Where this is not possible and good inlet conditions can't be created otherwise, a pressurized reservoir should be used. Or, an auxiliary pump can be used to maintain a supply of oil to the inlet at low pressure. A centrifugal pump can be used for this purpose, or a positive displacement pump with a pressure relief valve set to maintain the desired charging pressure.

DIAGNOSING TROUBLES

As we said at the beginning of this chapter many troubles in the system are wrongly blamed on the pump. A lot of pumps are sent back to the factory on warranty claims that have nothing wrong with them or that failed because of improper operation. So it's important to understand how the system works – where the oil goes and what happens to it – to diagnose troubles. Here are some of the problems you may encounter and possible reasons for them:

No Pressure: Remember that a pump doesn't put out pressure but flow.

Pressure is caused by resistance to flow. Low pressure means the fluid is meeting little resistance. If the load does not move the oil has probably found an easier path back to the reservoir through leakage. But remember you must leak the full pump delivery to get any loss in pressure.

A pump usually will not lose its efficiency all at once, but gradually. So there will be a gradual slowing down of the actuator speed as the pump wears. If the loss is sudden, and the pump isn't making a furious racket, best chances are the leak is somewhere else.

Slow Operation: This can be caused by a worn pump or by partial leakage of the oil somewhere else in the system. There will not be a corresponding drop in pressure if the load moves at all. Therefore, horsepower is still being used and is being converted into heat at the leakage point. You can often find this point by feeling the components for unusual heat.

No Delivery: If you know for certain that no oil is being pumped it can be because the pump is not assembled correctly, is being driven in the wrong direction, hasn't been primed or has a broken driveshaft. The reasons for no prime are usually improper start-up, inlet restrictions or low oil level in the reservoir.

Noise: Any unusual noise is reason to shut a pump down immediately. Find the trouble before a lot of damage is done. Cavitation noise is caused by a restriction in the inlet line, a dirty inlet filter or too high a drive speed. You already know what cavitation does to a pump.

Air in the system also causes noise. Air will severely damage a pump because there won't be enough lubrication. This can occur from low oil in the reservoir, a loose connection in the inlet, a leaking shaft seal, or no oil in the pump before starting.

Finally, noise can be caused by worn or damaged parts. Continued operation like this will spread harmful particles through the system causing more damage.

CHAPTER 4
Principles of Actuator Operation

We move now from the input of the hydraulic system to the output. In this chapter, we will study actuators, the devices that receive hydraulic energy and convert it to mechanical force and motion. Though we have begun studying components with the storehouse and the energy source, the output actuator is really the beginning point in designing a circuit. Unless we have a bucket to lift, a blade to lower, a wheel to turn, or some sort of load to be actuated, we have no need for a hydraulic system. The designer's first considerations then are selecting the device that will produce the needed motion and thrust. When that is done, he can work backward to power and control the system.

LINEAR OR ROTARY

As we noted in Chapter 1, actuators are either linear or rotary. A linear actuator (cylinder or ram) can give us force and motion outputs in a straight line. A rotary actuator or motor produces torque and rotating motion.

LINEAR ACTUATORS

CYLINDERS ARE LINEAR ACTUATORS

There are many names for linear actuators in the hydraulics industry. They are known variously as cylinders, rams, reciprocating motors, linear motors, and by probably several other names. Since you may sometime hear all these terms used, it is well to know they exist and what they mean. But in this manual, let's stick to the two most common terms, cylinder and ram. And here is the way we will use them:

- Cylinder – any hydraulic actuator constructed of a piston or plunger operating in a cylindrical housing by the action of liquid under pressure.
- Ram – a single-acting, plunger type cylinder; or, the plunger in this type of cylinder.

In other words, we will use the word "cylinder" to mean linear actuator, and the word "ram" to refer to a particular kind of cylinder.

PARTS OF A CYLINDER

Before we go on to further classify cylinders, let's identify their basic parts (Figure 67). The cylinder or barrel or housing of course is a tube in which the plunger or piston operates. In the ram-type cylinder, the plunger or ram actuates the

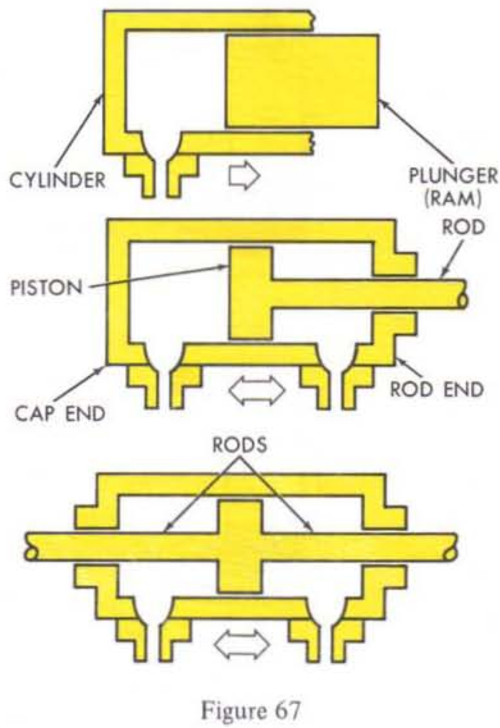


Figure 67

load directly. In the piston cylinder, a rod is connected to the piston to actuate the load.

The end of the cylinder from which the rod or plunger protrudes is called, appropriately, the rod end. The end opposite is called the cap end. The hydraulic connections are called the cap end port and the rod end port.

CLASSIFICATION OF CYLINDERS

We classify cylinders as single-acting or double-acting. And we further classify double-acting cylinders as differential or non-differential.

Single-Acting

A single-acting cylinder (Figure 68) has only one port and is operated hydraulically in one direction only. When oil is pumped into the port, it pushes on the plunger or piston, and the plunger or rod extends. To return or retract the cylinder, we must release the oil to the reservoir. The rod or plunger returns because of the weight of the load, or from some mechanical force such as a spring.

In mobile equipment, flow to and from a single-acting cylinder is controlled by a reversing directional valve of the single acting type.

Double-Acting

In a double-acting cylinder, we let the oil do the job in both directions. The cylinder must have ports at both the cap end and the rod end (Figure 69).

Pumping oil into the cap end moves the piston to extend the rod. At the same time,

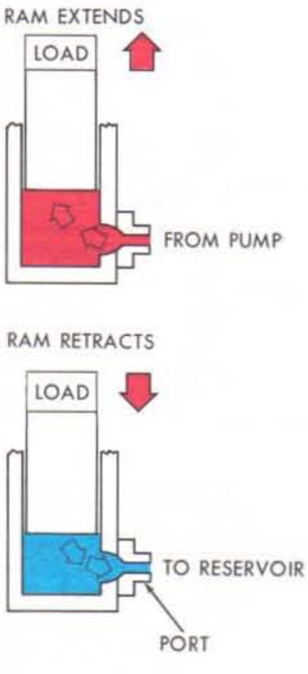


Figure 68

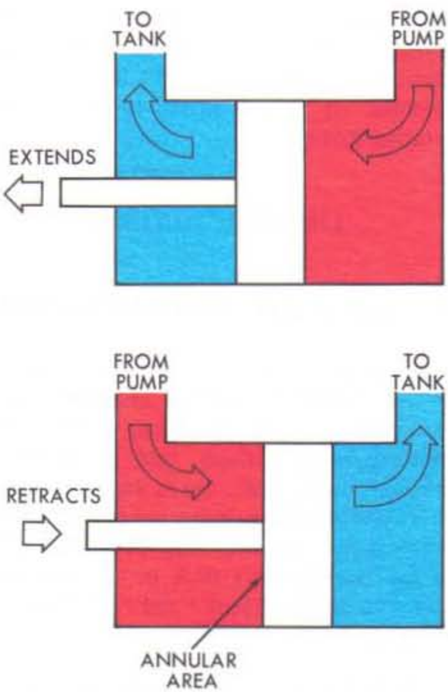


Figure 69

any oil in the rod end is pushed out and must be returned to the reservoir. To retract the rod, flow is reversed. Oil from the pump goes into the rod end and the cap end port is connected to allow return flow. The direction of flow to and from a double-acting cylinder can be controlled by a directional valve, or by actuating the control of a reversible flow pump.

Differential Cylinder

The double-acting cylinder in Figure 69 is called a differential cylinder, because the areas on the piston are not equal. On the cap end, the full piston area is available. At the rod end, we must subtract the area of the rod, so that only the annular area is available to develop force. The space that the rod takes up also reduces the volume of oil the rod end will hold.

Thus, we can make two general rules about a differential cylinder:

- With an equal gpm delivery to either end, the cylinder will move faster when retracting because of the reduced volume capacity.
- With equal pressure at either end, the cylinder can exert more force when extending because of the greater piston area. In fact, if we apply an equal pressure to both ports at the same time, the cylinder will extend because of the higher resulting force on the cap end.

A common ratio of piston area to annular area is 6-to-5 with a standard size piston rod. With a heavy-duty rod, the ratio may be as much as 1½-to-1 or 2-to-1.

Non-Differential Cylinder

A non-differential cylinder (Figure 70) has a piston rod extending from each end. It has equal thrust and speed either way, provided that pressure and flow are unchanged.

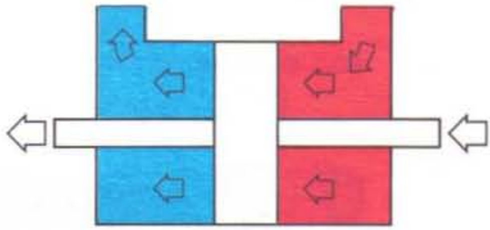


Figure 70

Few, if any, non-differential cylinders are used on mobile equipment.

CYLINDER CONSTRUCTION

A cylinder is constructed basically of a barrel or tube; a piston and rod (or ram); two end caps; and suitable oil seals.

The barrel is usually seamless steel tubing, or cast, and the interior is finished very true and smooth. The steel piston rod is highly polished and usually hard-chrome-plated to resist pitting and scoring. It is supported in the end cap by a bushing or polished surface.

The cylinder's ports are built into the end caps. End caps can be screwed on to the tubes, welded (Figure 71), or attached by tie bolts (Figure 72) or bolted flanges.

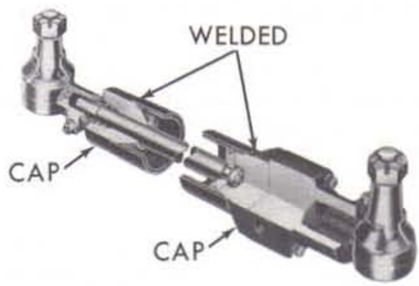


Figure 71

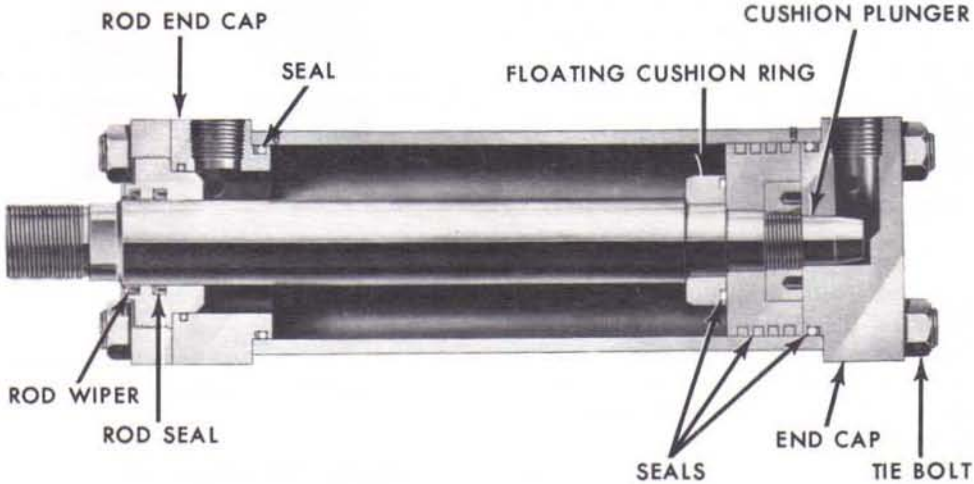


Figure 72

If the cylinder barrel is cast, the cap end may be part of the barrel. Mounting provisions often are made in the end caps, including flanges for stationary mounting or clevises for swinging mounts.

Seals and wipers are installed in the rod end cap to keep the rod clean and to prevent external leakage around the rod. Other points where seals are used are at the end cap, joints and between the piston and barrel. Depending on how the rod is attached to the piston, a seal may be required there also. Internal leakage past the piston is undesirable. It wastes energy and will permit the piston to drift under load.

Cylinder Cushions

A design feature of some cylinders is a cushion (Figure 73) which decelerates the piston smoothly at the end of its stroke. A tapered plunger or cushion ring on the rod enters a counterbore in the end cap and cuts off oil flow out of the cylinder. A small orifice in the end cap then controls flow out for the remaining short distance of piston travel. An adjustable valve is provided to increase or decrease the orifice size to control the deceleration rate.

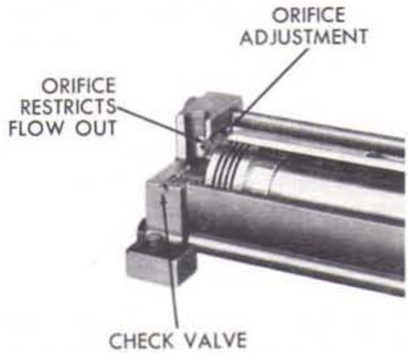


Figure 73

On the return stroke, we don't want to wait for the oil to squeeze through the orifice to start the piston up, so we also provide a free-flow check valve. The check valve blocks flow out of the cylinder but passes it in freely.

CYLINDER MOUNTING LEVERAGE

We saw in Chapter 1 how mechanical advantage or leverage is obtained either by design of the hydraulic system or the mechanical linkage.

A hydraulic cylinder can be attached to the load through linkage to multiply force by sacrificing distance, or to gain distance by reducing force.

Some typical leverage mountings are shown in Figure 74 along with their mechanical advantages and the actual forces exerted by the cylinder. Mechanical advantage (MA) is shown as the amount that force is multiplied.

Many cylinder mountings on mobile equipment are far more complex than these, particularly where the cylinder swings to various angles during operation. Such force vs. geometry relationships are outside the scope of this manual. The engineer, of course, is able to easily compute actual cylinder forces in these many conditions.

CYLINDER RATINGS

Cylinders are rated by size and pressure capacity. Size includes the bore diameter, the rod diameter and the stroke length. Bore diameters range from less than an inch to several feet; however, mobile cylinders are seldom larger than a few inches. Stroke lengths of several feet are not uncommon.

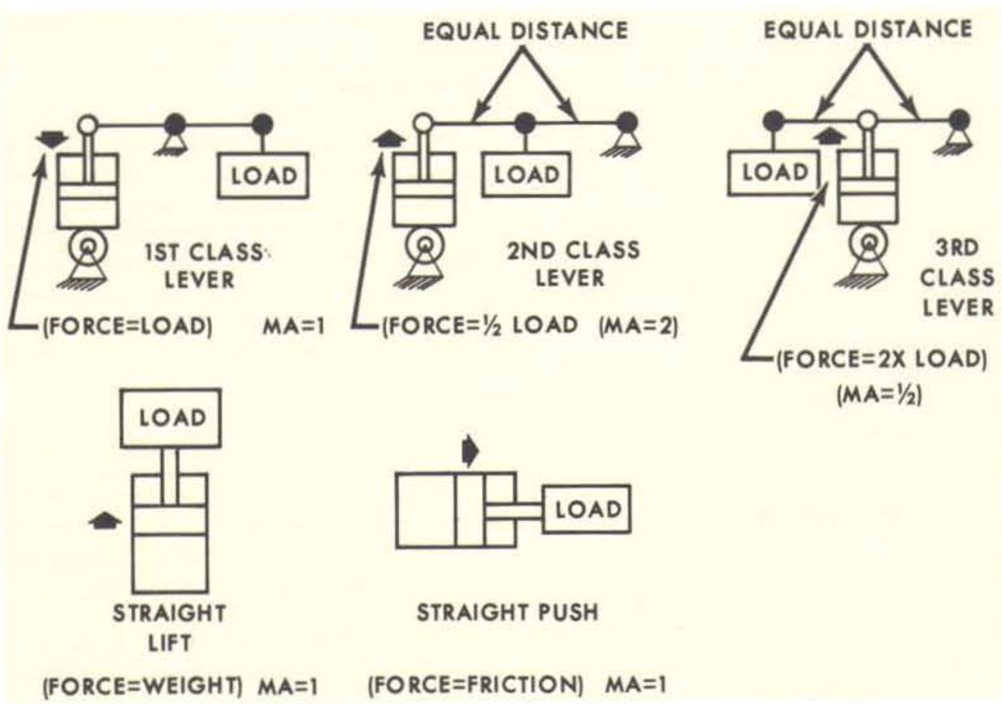


Figure 74

The length is limited by the tendency of a long cylinder to bend.

FORCE OUTPUT

You saw in Chapter 1 that the force a cylinder can develop depends on the pressure it can withstand and the area of the piston.

If we know the bore diameter, we can compute the cap end area like this:

$$A\ (\text{area}) = \frac{3.1416 \times D^2}{4}$$

D in this case is the diameter. To compute the rod-end annular area, we can use the same formula after first subtracting the rod diameter squared.

$$A_a = \frac{3.1416 \times (D_c{}^2 - D_r{}^2)}{4}$$

For example, if the bore is 36 square inches and the rod is 4 square inches, the rod-end annular area would be 32 square inches. Having the area and the pressure rating, then, we compute force like this:

$$F\ (\text{force}) = P\ (\text{pressure}) \times A\ (\text{area})$$

As we've said, force is in pounds, pressure in psi and area in square inches.

We can increase the force capability of a cylinder by increasing either pressure or size (piston area).

PRESSURE DEVELOPED

The force formula tells us the available force from the cylinder at a specified pressure. But we know that the actual pressure developed in moving a load is equal to the force of the load divided by the area on which the oil acts.

$$P = \frac{F}{A}$$

For instance, a pressure of 500 psi (34.5 bar) will be generated in lifting a 3500 pound (15569 N) load on a seven-square-inch (0.0045 m^2) piston.

$$P = \frac{F}{A} = \frac{3500\#}{7\ \text{sq. in.}} = \frac{500\#}{\text{sq. in.}} = 500\ \text{psi}$$

$$\left(P = \frac{F}{A} = \frac{15569\ \text{N}}{.0045\ \text{m}^2} = 34.5\ \text{N/m}^2 \times 10^5 = 34.5\ \text{bar}\right)$$

Any increase in load would raise the operating pressure. A lighter load, of course, would require less pressure.

As before, our units of measurement are the psi (bar), pound (N), and square inch (m^2).

We have neglected the effects of friction in these formulas. The force required to overcome friction adds to the force of the load.

Nomographic Charts

To simplify these calculations when only an approximation is required, nomagraphic charts are available in many hydraulics handbooks. A partial nomographic chart for pressure, force and cylinder diameter is shown in Figure 75. By laying a straight edge across any two known values, you can find the unknown where the line crosses its scale.

SPEED OF A CYLINDER

A cylinder's speed is independent of the load or the pressure. It depends on the volume of the space to be filled and the gpm (L/min) delivery to the cylinder

Volume (Figure 76) is equal to the area (0.7854 D^2) in square inches (mm^2) multiplied by the length in inches (mm). Thus, a cylinder 36 inches (914 mm) long with a

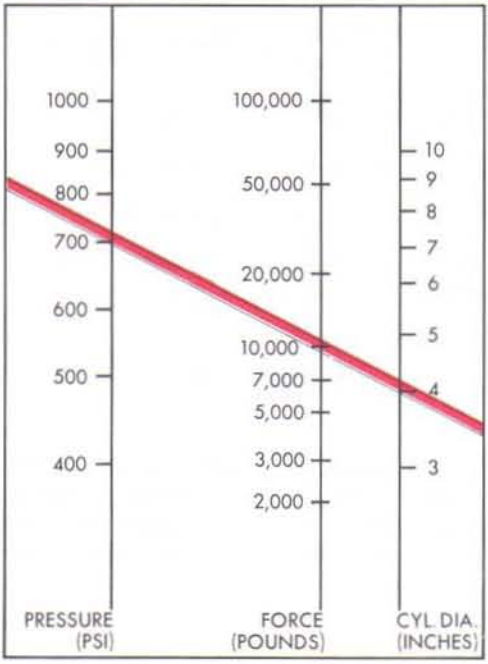


Figure 75

piston area of 7 square inches (4516 mm^2) is 252 cubic inches (4127 $mm^3 \times 10^3$ = 4127 mL) or slightly over a gallon (3.7854L) (a gallon, remember, is 231 cubic inches) (3785 mL):

$$\begin{aligned} \text{Volume} &= \text{Area x Length} \\ &= \text{7 sq. in. x 36 inches} \\ &= (4516\ mm^2 \times 914\ mm) \\ &= \text{252 cu. in.} \\ &= (4127\ mm^3 \times 10^3 = 4127\ mL) \end{aligned}$$

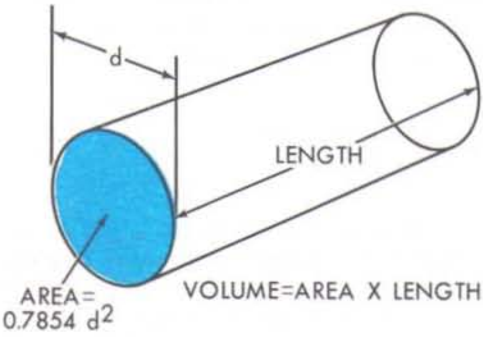


Figure 76

If we pump one gpm (3.785 L/min) of oil into this cylinder, the piston will travel its 36 inches (914 mm) length in just over a minute. The speed then would be approximately 36 inches (914 mm) per minute. If the delivery is doubled, the cylinder fills in half the time; therefore the speed is doubled to 72 inches (1828 mm) per minute. Thus, as flow rate increases, speed increases.

We can also increase the speed of a cylinder by decreasing the size of the cylinder. However, making the cylinder smaller will raise the operating pressure for a given load.

Finding Flow Required For a Given Speed

In general, to find the gpm (L/min) required for a given cylinder speed:

(1) Find the travel required in one minute.

(2) Find the volume required for that much travel.

(3) Convert the volume to gallons (litres)

$$\text{Required gpm} = \frac{\text{cubic inches per minute}}{231}$$

$$\left(\text{Required L/min} = \frac{\text{mL per minute}}{1000}\right)$$

Let's suppose that we have a cylinder with a four-inch (101.6 mm) diameter and want to know what gpm (L/min) delivery we must force into it to extend it the full length of 30 inches (762 mm) in fifteen seconds.

First, let's get the speed of the cylinder into inches (mm) per minute so that we can work with cubic inches (mL) and gpm (L/min). If the cylinder travels 30 inches (762 mm) in 15 seconds, it must travel 30 ÷ 15 or 2 inches (762 ÷ 15 or 50.8 mm) in one second.

$$\frac{30\text{ inches}}{15\text{ sec.}} = \frac{2\text{ inches}}{\text{sec.}}$$

$$\left(\frac{762\text{ mm}}{15\text{ s}} = \frac{50.8\text{ mm}}{\text{s}}\right)$$

Knowing the velocity in inches-per-second, we're only one step from delivery in gallons-per-minute (L/min). All we have to do is find out how many gallons (litres) would be required in 60 seconds.

$$\frac{\text{in.}^3}{\text{min.}} = \text{Area x Velocity x} \frac{60 \text{ sec}}{\text{min}}$$

$$= 0.7854 \text{ D}^2 \text{ x} \frac{2 \text{ in.}}{\text{sec}} \text{ x} \frac{60 \text{ sec}}{\text{min}}$$

$$= 0.7854 \text{ x 4 in. x 4 in. x 2 in. x} \frac{60}{\text{min}}$$

$$= \frac{1500 \text{ cu. in.}}{\text{min}} \text{(approximately)}$$

$$\left(= 0.7854 \text{ D}^2 \text{ x} \frac{50.8 \text{ mm}}{\text{s}} \text{ x} \frac{60 \text{ s}}{\text{min}}\right)$$

$$\left(= 0.7854 \text{ x 101.6 mm x 101.6 mm x 50.8 mm x} \frac{60}{\text{min}}\right)$$

$$\left(= \frac{24700 \text{ mm}^3 \text{ x } 10^3}{\text{min}} = \frac{24700 \text{ mL}}{\text{min}} \text{approx}\right)$$

Now we know that it will take 1500 cu. in./minute (24,700 mL/min) to do the job. We convert that to gpm (L/min) by dividing by 231 (1000):

$$\frac{1500 \text{ cu. in.}}{231} = 6.5 \text{ gpm approximately}$$

$$\frac{24{,}700}{1000} = 24.7 \text{ L/min approximately}$$

POWER

There are two ways of finding the power a cylinder uses:

If the force, distance and time involved are known, find the total power and convert to horsepower (kilowatts):

$$\text{Power} = \frac{\text{Force (lbs) x Distance (feet)}}{\text{Time (min.)}}$$

$$\left(\text{W} = \frac{\text{N x m}}{\text{Time (s)}}\right)$$

$$\text{HP} = \frac{\text{Power (ft. lbs/min)}}{33000} \text{ or } \frac{\text{ft. lb./sec.}}{550}$$

If the flow and pressure are known:

$$\text{HP} = \text{gpm x psi x } 0.000583$$

$$(\text{kW} = \text{L/min x bar x } .00167)$$

These are the same general relationships we brought out in Chapter 1.

MOTORS

In construction, rotary hydraulic motors closely resemble pumps. Some pumps, in fact, can be operated as motors without any change at all and others require only minor modifications.

MOTOR OPERATION

We could say that a motor is simply a pump that is being pushed instead of doing the pushing. Oil is pumped into one port and causes the shaft to turn. The same hydraulic chambers carry the oil to the other port and discharge it to return to the tank or the pump inlet.

The principal types of motors, then, are the same as pumps . . . vane, piston and gear. They can be uni-directional or reversible. Most Vickers motors designed for mobile equipment are reversible. There is one model series that is uni-directional.

MOTOR RATINGS

The principal ratings of a motor are pressure, displacement and torque. Displacement tells us how much flow is required for a specified drive speed. Torque and pressure ratings tell us how much load the motor can handle.

Displacement

Displacement of a hydraulic motor is expressed in cubic inches (millilitre) per revolution . . . the same as pump displacement. A motor's displacement is the amount of oil we must pump into it to turn it one revolution. Most motors are fixed displacement, but there are many variable displacement piston motors in use, particularly in hydrostatic drives.

Required Delivery for Given Speed

If we know the displacement and the desired speed of a motor, we find the delivery requirements like this:

$$\text{Gpm} = \frac{\text{Speed (rpm)} \times \text{Displacement (cu in./rev)}}{231}$$

$$\left(\text{L/min} = \frac{\text{Speed (rpm)} \times \text{mL/rev}}{1000}\right)$$

For instance, a motor with displacement of 2.31 cubic inches (37.9 mL) per revolution, to run at 1000 rpm, would require a supply of 10 gpm (37.9 L/min).

$$\text{Gpm} = \frac{1000 \text{ rpm} \times 2.31 \text{ cu in./rev.}}{231} = 10$$

$$\left(\text{L/min} = \frac{1000 \text{ rpm} \times 37.9 \text{ mL/rev}}{1000} = 37.9 \text{ L/min}\right)$$

Drive Speed

If we know the displacement and the gpm (L/min) delivery that is supplied, we can calculate the drive speed in revolutions per minute:

$$\text{Rpm} = \frac{\text{gpm} \times 231}{\text{Displacement (cu. in./rev.)}}$$

$$\left(\text{Rpm} = \frac{\text{L/min} \times 1000}{\text{Displacement (mL/rev.)}}\right)$$

From this, we see that increasing the displacement of a motor reduces its speed; decreasing the displacement increases speed. The other way to increase speed is to increase the delivery (gpm) (L/min).

What is Torque?

Torque, by definition is a turning or twisting effort . . . or we might say, a rotary thrust. Motor torque is usually measured in pound inches (Nm) or pound feet. A torque wrench is simply a wrench with a scale on it to indicate how much twist the wrench is exerting.

The torque a motor develops depends on the load and the radial distance from the center of the motor shaft. For instance, the 5 pound (22.2 N) force exerted by the weight in Figure 77, view A, is effective five inches (0.127 m) from the shaft center. The torque on the motor shaft is equal to the pulley radius multiplied by the weight or 25 pound inches. In view B, the pulley is smaller so that torque is less. The small pulley imposes less torque on the motor, but the large pulley raises or lowers the weight faster is the motor speed doesn't change.

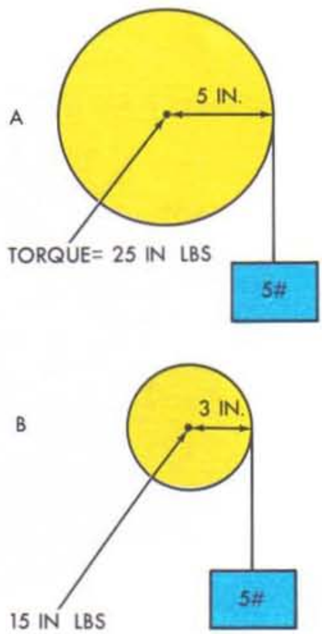


Figure 77

Torque Rating

The torque rating of a motor is expressed as a torque rate in pound inches (Nm) per 100 psi (100 bar) of pressure. Thus, a 25 pound-inch/100 psi (41 Nm/100 bar) motor requires 100 psi (100 bar) of operating pressure with a 25 pound inch (41 Nm) load. With a 50 pound inch load (82 Nm), the pressure required would be doubled or 200 psi (200 bar); with a 75 pound inch (123 Nm) load, it would be 300 psi (300 bar); and so on.

In general, the working pressure of a motor is:

$$\text{Working Pressue} = \frac{\text{Torque Load x 100}}{\text{Torque Rate}}$$

The maximum torque the motor can handle depends on its maximum rated pressure and its torque rating:

$$\text{Max. Torque} = \frac{\text{Torque Rate x max. pressure}}{100}$$

So if our 25-pound-inch (41 Nm) motor can be operated at 2000 psi (138 bar), it will develop 500 pound inches (56.5 Nm) of torque:

$$\text{Torque} = \frac{\text{25 in. lb x 2000 psi}}{\text{100 psi}} = \text{500 lb in.}$$

$$(\text{T} = \frac{\text{41 Nm x 138 bar}}{\text{100 bar}} = \text{56.5 Nm})$$

A general formula for torque in a hydraulic motor is this:

$$\text{Torque (lb. in.)} = \frac{\text{Pressure x Displacement (cu. in./rev.)}}{2\pi}$$

$$\left(\text{Nm} = \frac{\text{bar x mL/rev.}}{20\pi}\right)$$

From this we can see that torque increases whenever either the pressure or the displacement increases. However, with a larger displacement, motor speed is reduced in proportion to the gain in torque.

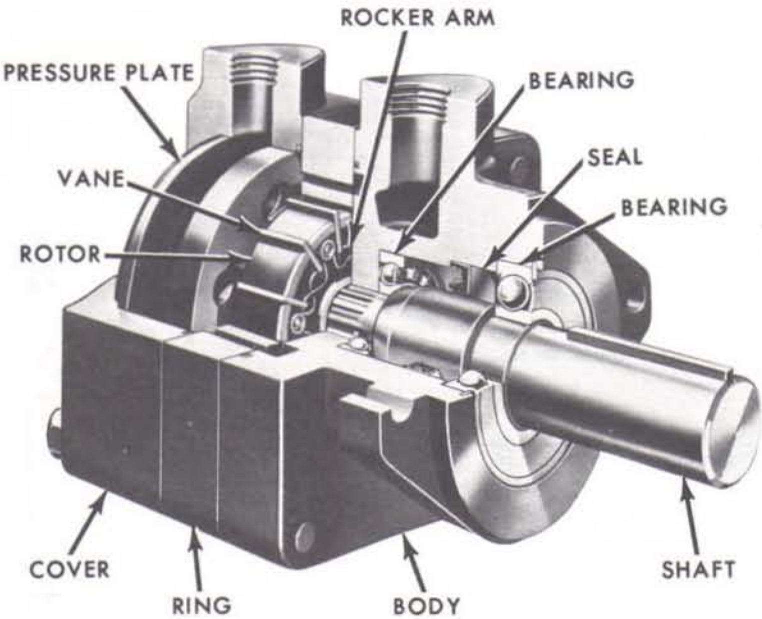


Figure 78

Torque and Power

There are two general relationships between torque and power for any rotating device, and they apply equally well to a hydraulic motor.

$$\text{Torque (lb. in.)} = \frac{63025 \times \text{hp}}{\text{rpm}}$$

$$\left(\text{Nm} = \frac{\text{kW} \times 9550}{\text{rpm}}\right)$$

$$\text{Hp} = \frac{\text{Torque (lb. in.)} \times \text{rpm}}{63025}$$

$$\left(\text{kW} = \frac{\text{Nm} \times \text{rpm}}{9550}\right)$$

Also, the hydraulic power formula can be used if we know the pressure and flow rate:

$$\text{Hp} = \text{gpm} \times \text{psi} \times .000583$$

$$(\text{kW} = \text{L/min} \times \text{bar} \times .00167)$$

"SQUARE" DESIGN VANE MOTOR

"Square" design vane motors (Figure 78) look very much like their pump counterparts. They have springs to initially hold the vanes out against the cam ring in the absence of centrifugal force.

They are balanced hydraulically to prevent the rotor from side-loading the shaft. The shaft is supported by two ball bearings.

Balanced Vane Motor Operation

Torque is developed by a pressure difference as oil from the pump is forced through the motor. We can see this easiest by looking at the pressure differential on a single vane as it passes the inlet port (Figure 79). On the side open to the inlet port the vane is subject to system pressure. The opposite side of the vane is subject to the much lower exhaust pressure. The difference in

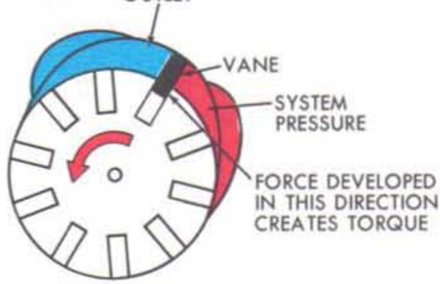


Figure 79

pressure exerts a force on the vane that is, in effect, tangential to the rotor. Just as the weight in Figure 77 caused a torque on the pulley shaft, this tangential force causes torque on the motor shaft.

This pressure difference is effective across vanes 3 and 9 in Figure 80. The other vanes,

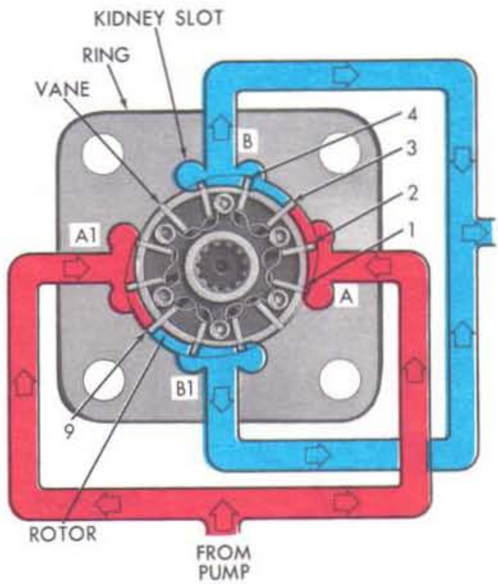


Figure 80

as shown, are subject to essentially equal force on both sides. Each will take its turn in developing torque as the rotor turns.

We are looking at the flow condition for counterclockwise rotation as viewed from the cover end. The body port is the inlet and the cover port the outlet. If flow is reversed, the rotation becomes clockwise.

Rocker Arms

In the vane type pump, you'll remember that the vanes are pushed out against the cam ring by centrifugal force when the pump is started up. When we start a motor, centrifugal force isn't available to do this job. We must find another way to get the vanes out, or the oil will just flow through without developing any torque.

This design motor used steel wire rocker arms (Figure 81) to hold the vanes against the cam ring. The arms pivot on pins attached to the rotor. The ends of each arm support two vanes which are 90 degrees apart.

When the vane at one end of the arm(A) is being pushed into its slot by the cam ring, the other (B) is being given room to slide out. The extension of vane A is always accompanied by retraction of vane B and vice-versa. That way, though the rocker arm exerts a little spring force on the vanes, it doesn't flex appreciably.

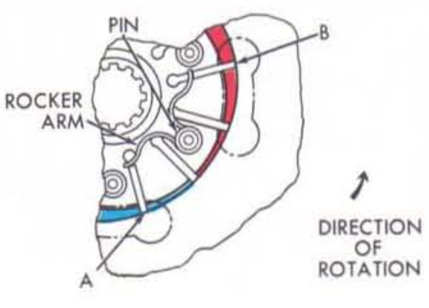


Figure 81

Pressure Plate Functions

The pressure plate in this motor has the same functions as in the pump. It seals the side of the rotor and ring against internal leakage; and it feeds system pressure under the vanes to hold them out against the ring. This is very simply done in the pump, because the pressure plate is in the cover and always under system pressure.

In the reversible motor, though, the cover port is sometimes the return or low pressure port (Figure 82), so the pressure plate design is considerably different. Notice that the pressure chamber (A in Figure 82) is sealed from the cover port. The cover port is open to an annular passage around the plate, and this passage opens to the shuttle valve. The body port also is connected to the shuttle valve.

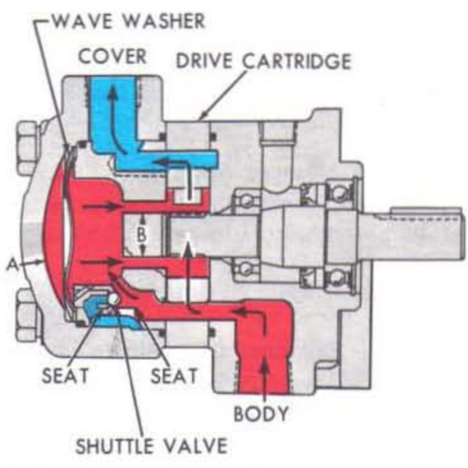


Figure 82

As shown, the body port is under pressure. This pressure forces the shuttle valve to the left and seals off the connection from the low-pressure port. System pressure is routed into chamber A.

If the flow is reversed, the cover port is pressurized and the shuttle valve is forced to the right, blocking the connection from the body port. Again, system pressure is directed into Chamber A, but this time from the other port.

Pressure in Chamber A holds the pressure plate against the ring and rotor. It also is effective under the edges of the vanes through passage B.

"S2" Pressure Plate Modification

A special modification of this pressure plate (Figure 83) permits operating the motor without vane rocker arms or shuttle valves. A "pressure drop" check valve is placed in the pressure line ahead of the reversing directional valve. This check valve creates a back-pressure that is always 30 psi (2 bar) higher than the operating pressure of the motor. The "30-plus" (2 bar) pressure then is routed to Chamber A by an external connection. There, the pressure holds the vanes extended and pushes the pressure plate against the cam ring and rotor anytime the pump is running.

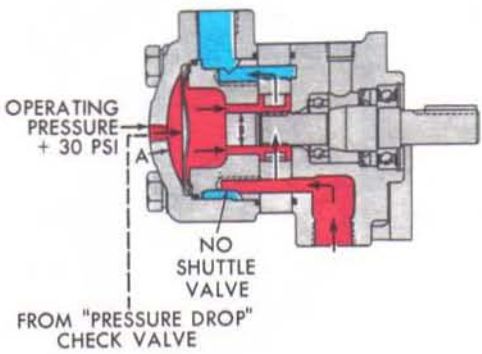


Figure 83

M2U UNI-DIRECTIONAL MOTORS

Uni-directional motors (Figure 84) are similar in design to the "square" motors just described. However, because the direction of flow doesn't have to be reversed, shuttle valves are not used in the pressure plate. The cover port is always the pressure port and the motor is internally drained.

Torque is developed as in the other square motors (Figure 85). The vanes are extended by pressure as in the S2 design reversible motors. But in this design, the check valve which creates the pressure difference is built into the motor cover (Figure 84). Uni-directional motors are built in torque capacities from 18 to 35 inch pounds (29.5 - 57.3 Nm) per 100 psi (100 bar).

HIGH PERFORMANCE VANE MOTORS

Another design of balanced type vane motor is the high performance motor (Figure 86). It is a brother to the high-performance vane pump, and shares the feature of a removable cartridge. The motor design is simplified by making both end plates act as pressure plates, and by holding the vanes out with springs. Shuttle valves and rocker arms aren't needed.

Torque is developed just as in the "square" motor (Figure 87). The pressure areas on the dual pressure plates are shown in Figure 88.

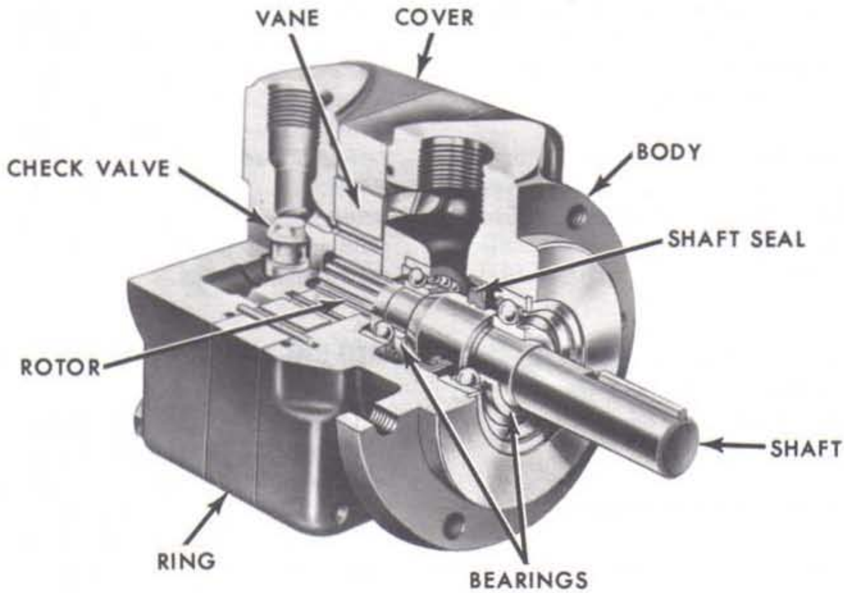


Figure 84

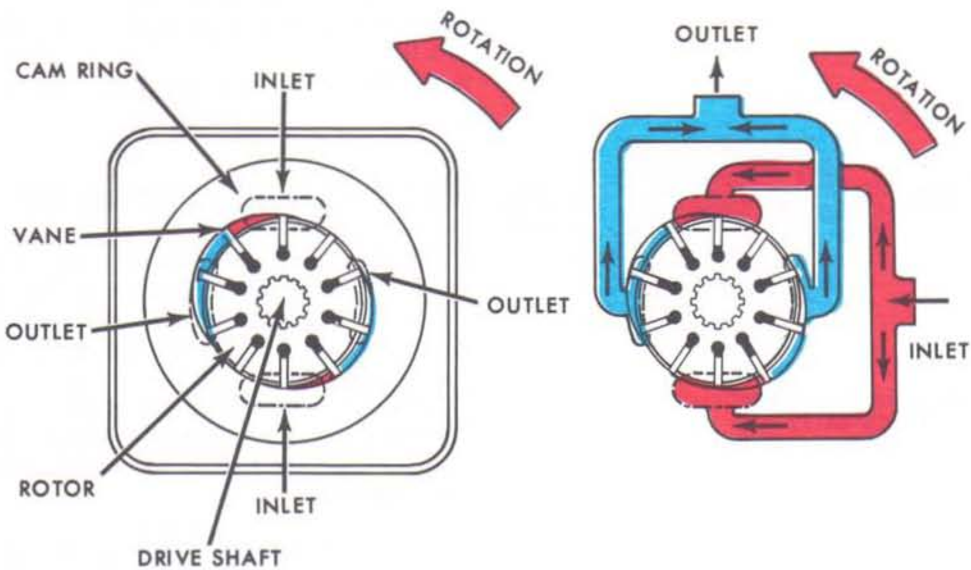


Figure 85

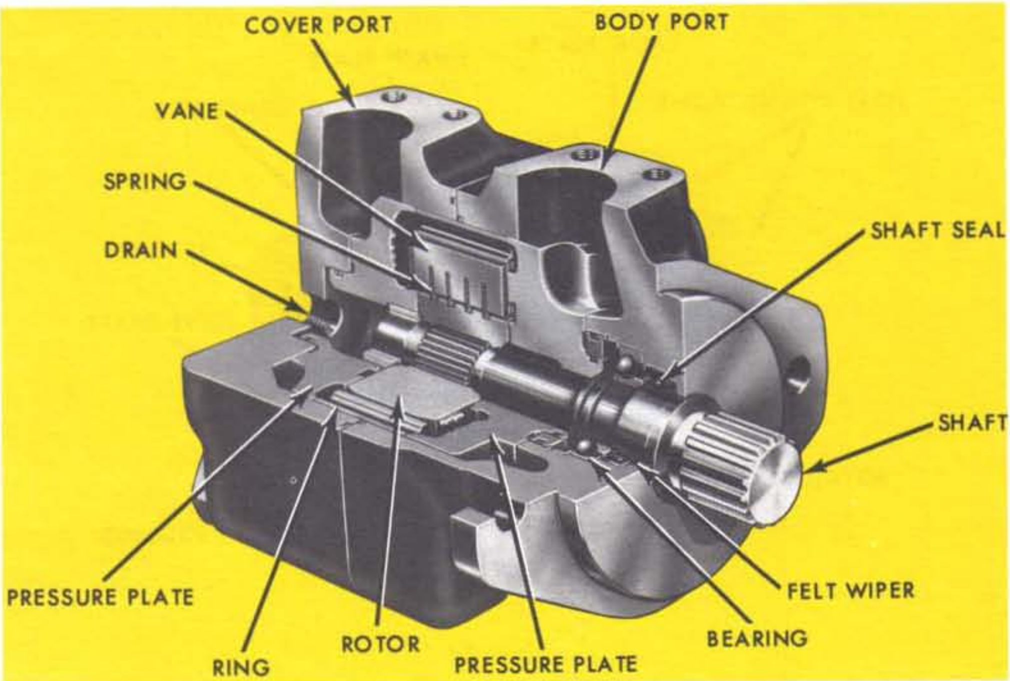


Figure 86

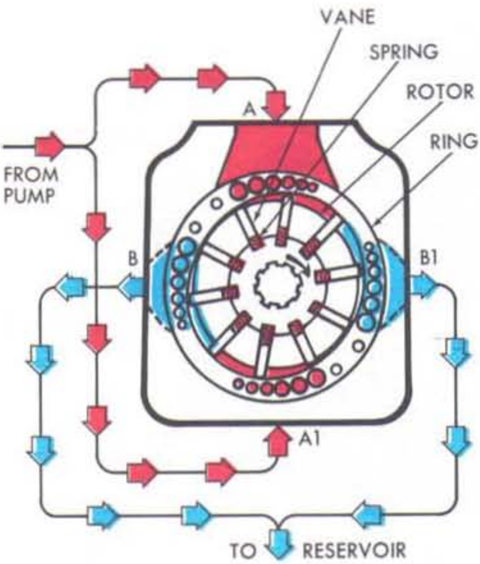


Figure 87

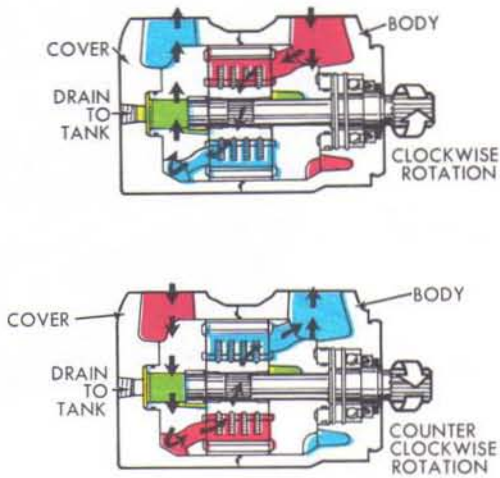


Figure 88

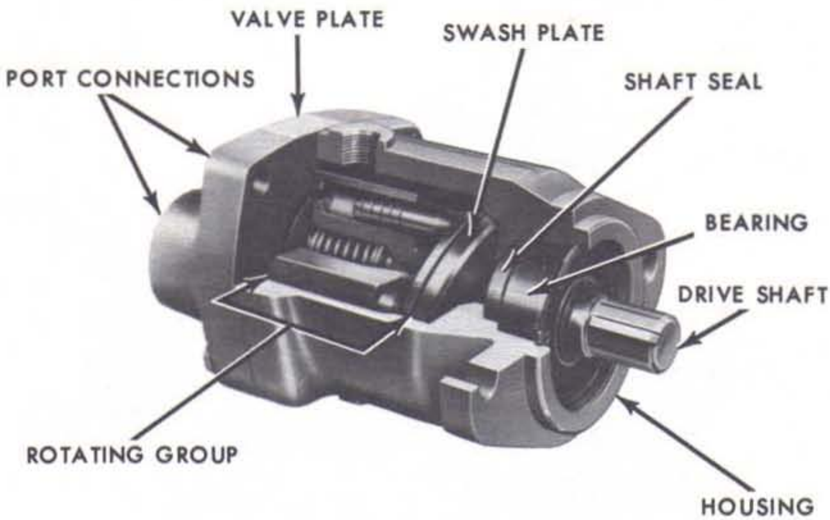


Figure 89

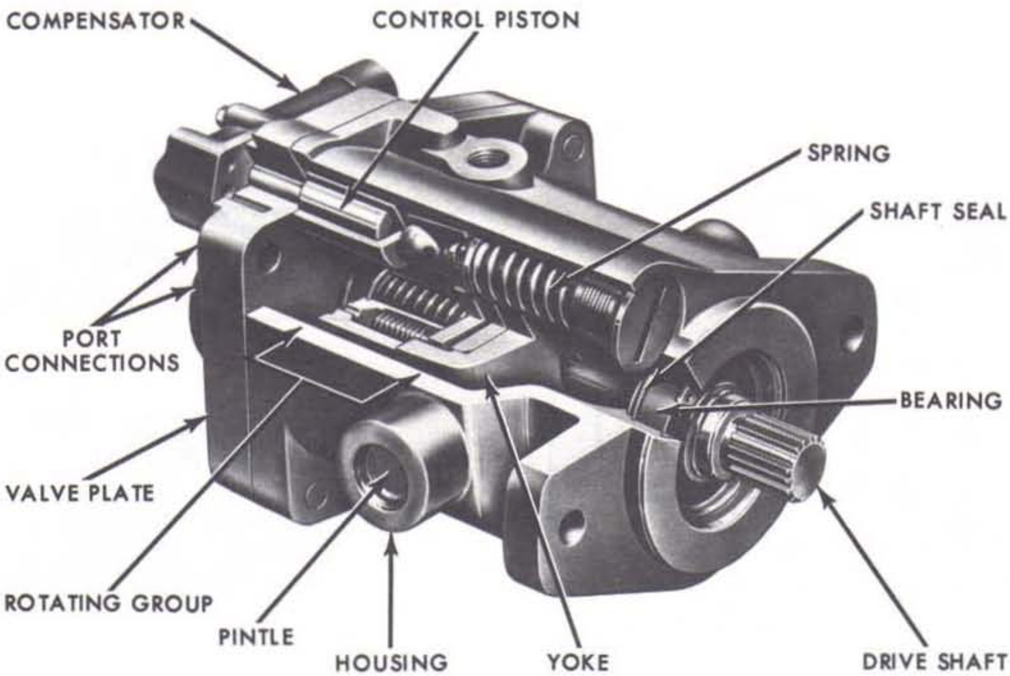


Figure 90

EXTERNALLY DRAINED

Most reversible hydraulic motors require external drain lines to carry off leakage oil. You can see why this is so by looking at the internal leakage paths in Figure 88.

When the body port is under pressure, oil could leak past the seal on the pressure plate hub into the chamber surrounding the shaft. If a relief weren't provided, this chamber could fill up and eventually become pressurized. The result would be a blown shaft seal. A similar leakage path is at the hub of the cover-end pressure plate. A passage along the shaft connects into the cover, where an external line is connected to carry leakage oil back to the tank.

Some pumps also are externally drained. However, we don't usually switch ports back and forth on the pump, so it is usually possible to drain it internally. In internally drained components, drain passages lead to a port that is always at low pressure.

INLINE PISTON-TYPE MOTORS

The in-line design piston motors (Figures 89 and 90) are virtually identical to the pumps. They are built in both fixed- and variable-displacement models in several sizes. Torque is developed by the pressure drop through the motor; the pressure exerting a force on the ends of the pistons which is translated into shaft rotation. Shaft rotation of most models can be reversed at will by reversing the direction of flow.

Operation

Oil from the pump is forced into the cylinder bores through the motor inlet port (Figure 91). The force on the pistons at this point pushes them against the swash

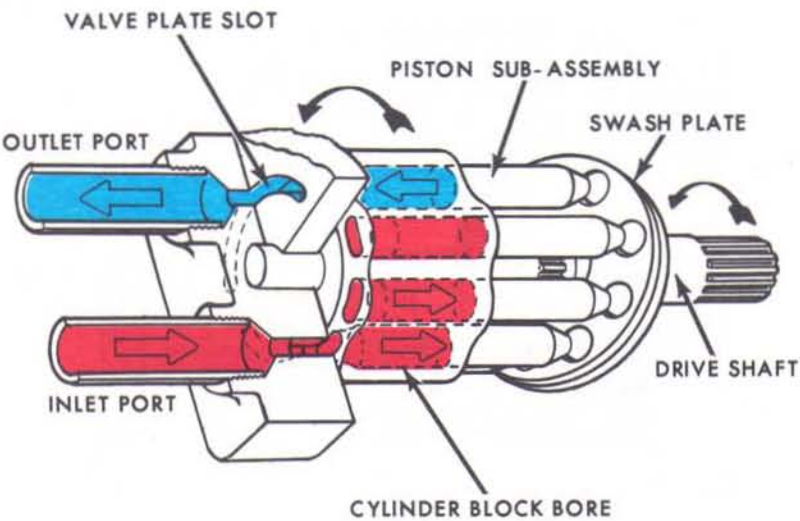


Figure 91

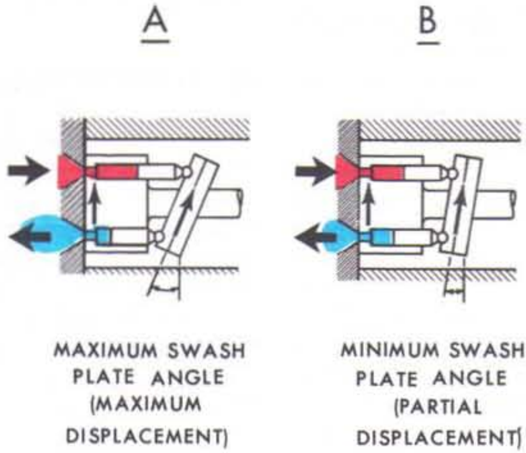


Figure 92

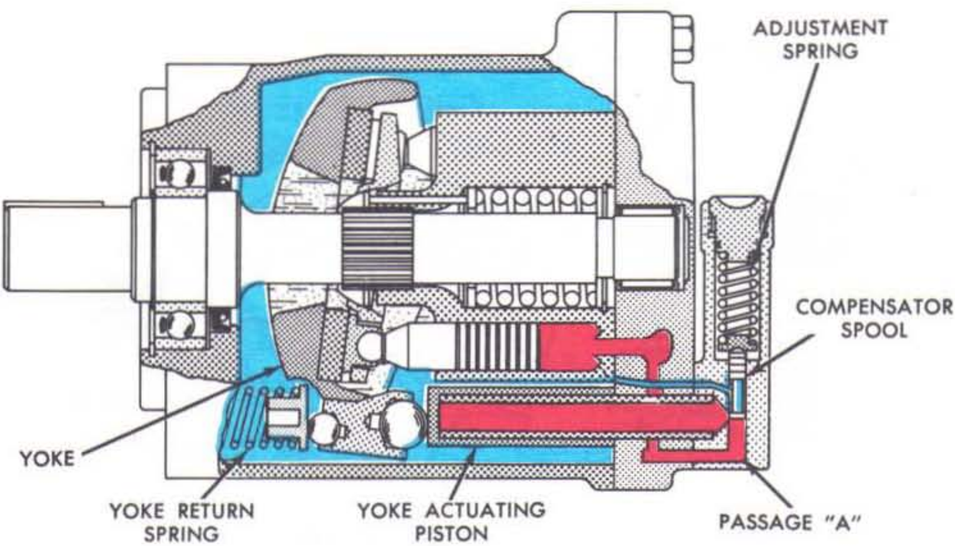


Figure 93

plate. They can move only by sliding along the swash plate to a point further away from the cylinder barrel. In so doing, they cause the cylinder barrel to rotate. The cylinder barrel is splined to the shaft, so the shaft must turn, too.

Inline Motor Displacement

The motor's displacement depends on the angle of the swash plate (Figure 92). At maximum angle, the displacement is maximum because the pistons travel maximum length.

When the angle is reduced, piston travel shortens, reducing displacement. If the flow remains constant, the motor then runs faster, but torque is decreased. Torque is greatest at maximum displacement because the component of piston force parallel to the swash plate is greatest.

Variable Displacement Models

In the variable displacement models, the swash plate is mounted in a yoke that swings on "pintles" to change the plate angle. The yoke can be moved mechanically or by a compensator control.

Compensator Control

The compensator control (Figure 93) operates by balancing system pressure against a spring loaded valve spool. The valve controls an actuating piston which swings the yoke in response to changes in pressure. The yoke is in the minimum-angle position when pressure is lower than the adjusted setting of the valve spring.

If the load increases, pressure also increases. When the pressure reaches the spring setting, it lifts the valve spool against the spring force. Passage "A" is opened to the yoke actuating piston. The piston moves and forces the yoke to a greater angle . . . thus increasing displacement. Motor speed is reduced but more torque is available to handle the load.

The compensator control, then, regulates motor displacement for maximum performance under all load conditions up to the relief valve setting.

BENT AXIS PISTON MOTORS

Angle-type or bent axis piston motors also are nearly identical to the pumps. They are built in both fixed-displacement and variable-displacement versions (Figure 94) in several sizes. Variable displacement motors can be controlled mechanically or by pressure compensation.

Operation of the motor is nearly identical to in-line motors, except that the thrust of the pistons is against the driveshaft flange. The parallel component of the thrust causes the flange to turn. Torque is maximum at maximum displacement; speed is minimum.

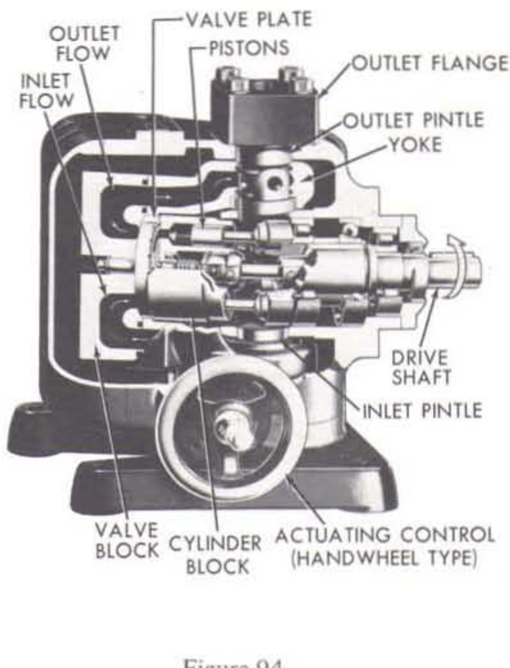


Figure 94

CHAPTER 5
Principles of Valve Operation

Valves are used in our hydraulic systems to control the operation of the actuators. Very often, in fact, we find the valves referred to as the "control", particularly where a number of them are built into a single assembly.

The valves assert their authority in the circuit by regulating pressure; by creating special pressure conditions; by deciding how much oil will flow in portions of the circuit; and by telling the oil where to go.

We group hydraulic valves into three general categories: pressure controls, flow controls and directional controls. Some valves, however, have multiple functions that fall into more than one of these categories.

Valves are rated by their size, pressure capabilities and pressure drop vs. flow. They are usually named for their functions, but may be named for their construction as well.

In construction, they vary from a simple ball and seat to a many-element, spool-type valve with a jet-pipe pilot stage and electrical control. Fortunately, our general classifications permit us to begin with the simplest valves and build up the complex designs. Let us do just that, beginning with pressure control valves.

PRESSURE CONTROL VALVES

A pressure control valve may have the job of (1) limiting or otherwise regulating pressure; (2) creating a particular pressure condition required for control; or (3) causing operations of actuators to occur in a specific order.

Pressure Controls Are Balanced Valves

All pure pressure control valves operate in a condition approaching hydraulic balance. Usually the balance is very simple: pressure is effective on one side or end of a ball, poppet or spool; and is opposed by a spring. In operation, the valve takes a position where the hydraulic pressure exactly balances the spring force.

INFINITE POSITIONING

Since spring force varies with compression distance, and since pressure also can vary, a pressure control valve is said to be infinite positioning. In other words, it can take a position anywhere between two finite flow conditions . . . passing a large volume of flow to a small volume, or passing no flow at all.

Normally-Open or Normally-Closed

Most pressure control valves are classified as normally-closed. This means that flow to the valve inlet port is blocked from the outlet port until pressure becomes high enough to cause "unbalanced" operation.

In a normally-open valve, there is free flow through the valve until it begins to operate in balance. Then flow is partially restricted or cut off.

Pressure Override

Pressure override is a characteristic of normally-closed pressure controls when they are operating in balance. Because a compression spring's force increases as its height is reduced, the pressure when the valve first "cracks" is less than when it is passing a large volume or "full-flow". The difference between the full-flow and cracking pressure is called override.

When we refer to a pressure control valve's "setting" in this chapter, we'll mean its cracking pressure. It will be understood that the actual operating pressure can be higher if the valve is handling a large volume of flow.

CHECK VALVES

A check valve can be either a pressure control or a directional control, or both. To use it as a pressure control, we "load" it with a spring and connect it into a line in series . . . so that it creates a pressure drop or back-pressure.

Often a check valve is nothing more than a ball and seat placed between two ports (Figure 95). As a directional control, it has a free-flow and a no-flow direction. Flow through the seat will push the ball away and permit free flow. Flow in the other direction pushes the ball against the seat, and pressure build-up forces it to seal the passage so flow is blocked.

The valve spring may be very light if it is used only to return the ball to its seat when flow stops. In that case, pressure drop through the valve will probably be no more than 5-10 psi (0.34 to 0.69 bar). When the valve is used to create a back-pressure, a heavier spring is used. Pressure at the inlet balances against the spring force to produce a significant pressure difference, depending on the spring rate.

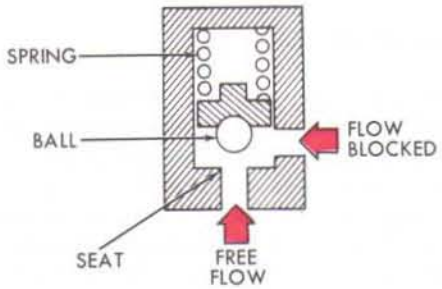


Figure 95

IN-LINE CHECK VALVES

In-line check valves (Figure 96) are designed for straight-through flow with piping connections in line. A cone-shaped poppet is normally spring-loaded against a seat machined in the valve body. These valves are built in three- to 50-gpm (11.4 L/min to 189 L/min) sizes and with cracking pressures of five psi (0.34 bar) to 65 psi (4.48 bar).

RIGHT-ANGLE CHECK VALVES

A 90-degree turn inside the valve body is responsible for the name given the valve you see in Figure 97. This is heavier-duty check

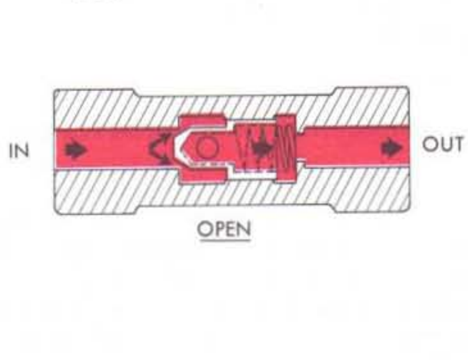


Figure 96

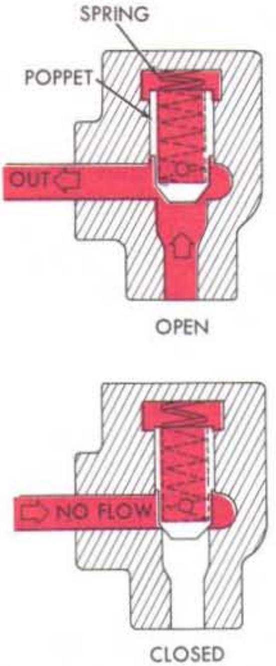


Figure 97

valve with a steel poppet and a hardened seat pressed into the cast body. It is built in three- to 320-gpm (11.4 to 1211 L/min) sizes with five-psi (0.34 bar) or 50-psi (3.45 bar) cracking pressure.

Some Are Back-Mounted

The valve pictured in Figure 97 has threaded connections. Up to 50 gpm (189 L/min), they also are built for back-mounting. Back-mounted valves have all their port connections on a single face for mounting against a port plate or sub-plate. Pipe connections are made to the plate.

Most back-mounted valves today have the ports sealed against the mounting plate by individual seal rings. But early designs used gaskets; hence they are often referred to as "gasket-mounted" valves.

Flange Connections Too!

From 90- to 320-gpm (341 to 1211 L/min) sizes of right-angle check valves (and most other valves too) have their connections made with packed, gasket-sealed or O-ring-sealed flanges.

RELIEF VALVES

A relief valve is required in any hydraulic circuit that uses a positive displacement pump to protect the system against excessive pressure. If the actuator is stalled, or simply travels as far as it can go, there must be an alternate flow path for the pump's output. Otherwise pressure will instantly rise until either something breaks or the prime mover stalls.

The relief valve is connected between the pump outlet (pressure line) and tank. It is normally closed. We set it to open at a pressure somewhat higher than the load requirement and divert the pump delivery to tank when this pressure is reached.

Of course, if the relief valve were set lower than the load pressure requirement, it would provide a path of "least resistance". Then the oil would take a short cut home instead of moving the load.

A relief valve also can be used to limit the torque or force output of an actuator, as in the hydraulic press or a hydrostatic transmission.

SIMPLE RELIEF VALVE

Relief valves are classified as simple or compound.

A simple relief valve (Figure 98) may be little more complicated in construction than a check valve. A spring-loaded ball or poppet seals against a seat to prevent flow from the

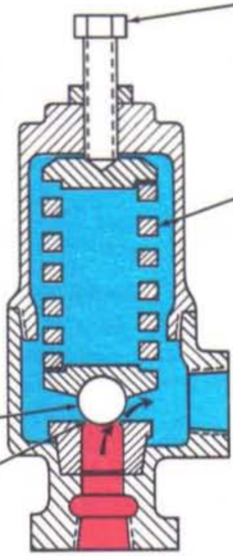


Figure 98

inlet (pressure) port to the outlet (reservoir) port. An adjusting screw can be turned in or out to adjust the spring load . . . which in turn adjusts the valve's cracking pressure.

Pressure override is often a problem with a simple relief valve. When handling flow from a fair-size pump, the valve may override several hundred psi . . . not only wasting power, but overloading circuit components. Another disadvantage of this type valve is a tendency to chatter when it is "relieving". These disadvantages are overcome to a great extent in the two-stage or compound relief valve.

COMPOUND RELIEF VALVE

A compound relief valve is designed with a small pilot valve to limit pressure and a larger valve controlled by the pilot valve to divert the large volume of flow. Its override is low and nearly constant over a wide range of flow rates.

"RM" TYPE RELIEF VALVE

"RM" series valves (Figure 99) are compound relief valves with the pilot stage built into a valve spool. The pilot stage is a spring-loaded poppet. The spring in the pilot stage controls the cracking pressure; the larger spring pushing against the spool determines the maximum override.

When the valve first fills through the pressure port, drilled passages carry the oil through an orifice to the spring end of the valve spool. An opening in this end of the spool leads to the head of the pilot stage.

Normally Closed. When the passages are filled, any pressure less than the valve setting, by Pascal's Law, will equalize . . . from the pressure port, to the large spring chamber, and to the head of the pilot stage poppet (View A). We then have equal pressure on both ends of the valve spool. The only effective force on the spool is the large spring. It holds the spool to the left, or in its normally closed position.

Relief Operation. If pressure builds up high enough to force the pilot stage poppet off its seat, we begin to get pilot flow (View B). Oil flows from the pressure port, through the orifice, into the spool, past the pilot poppet and through a drilled hole to the tank port.

Pilot flow causes a pressure drop across the orifice, so that pressure is no longer equal on both ends of the spool. At about a 40 psi (2.8 bar) pressure difference, pressure at the inlet side overcomes the large spring. The entire spool then is pushed to the right and cracks the pressure port to the tank port.

The spool assumes a position where it is balanced between system pressure on the

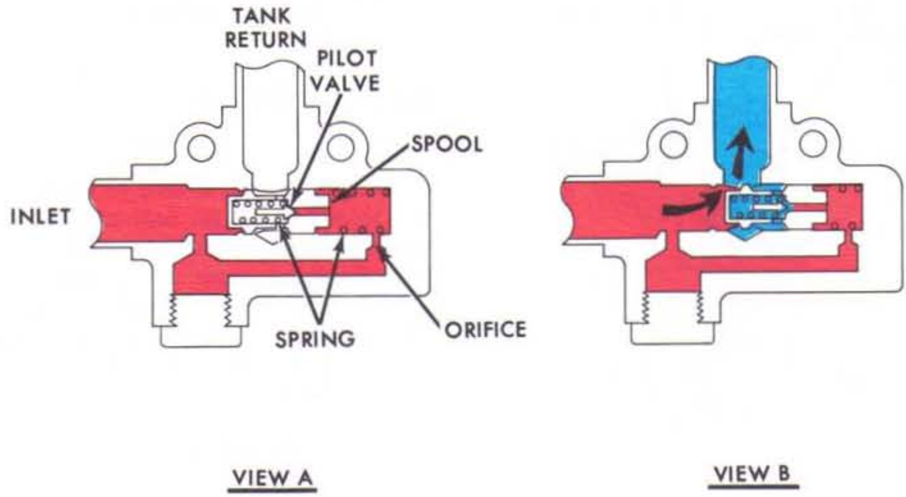


Figure 99

left and pilot stage pressure plus the large spring's force on the right. It throttles pump delivery to the tank while maintaining pressure in the system.

When the system pressure drops, the pilot stage closes and pilot flow stops. With no flow across the orifice, pressure on the spool ends again equalizes. The spring then moves the spool back to its closed position.

Since the large spring is very light, its override is negligible. Override in the pilot stage also is slight, because it handles only a small part of the total flow, thus providing low override.

RM valves are pressure preset at the factory. Interchangeable valve spool assemblies are available with different pressure settings up to 2500 psi (172 bar). If an external pressure adjustment or higher capacity is required, the balanced piston type valve is used.

BALANCED PISTON RELIEF VALVE

The balanced piston relief valve shown in Figure 100 operates the same as the RM valve. The pilot stage is in a separate cover bolted onto the main body. The main body contains a piston which handles the large volume of flow. An orifice is drilled in the piston to balance pressure on both sides when there is no pilot flow.

Pilot flow at the valve setting is through the piston orifice, past the pilot poppet and through the hollow center of the piston to the tank port. A pressure differential of about 20 psi (1.4 bar) across the piston will overcome its spring and open the pressure port to the tank port.

A Dynamic Skirt

The skirt on the bottom of the piston is a hydrodynamic helper when pressure drops

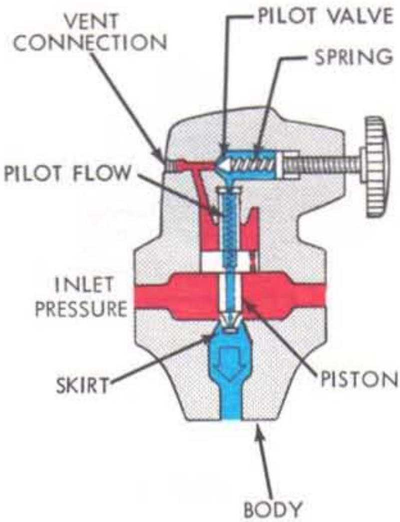


Figure 100

off. Flow through the tank port strikes the top of the skirt and makes the piston close faster.

Venting

A vent connection on this valve allows us to use it to "unload" the pump with an external control. If the vent connection is opened to atmosphere, it provides a flow path that bypasses the pilot stage. Furthermore, a vented or no-pressure condition above the piston will require only 20 psi (1.4 bar) below it to open the tank port. Thus, when the valve is vented, the pump operates under a 20 psi (1.4 bar) load only.

The vent connection also can be used for remote control of pressure by connecting a second pilot stage to it. Thus the operator can select one of two pressures by operating another valve.

Balanced piston valves range to 320 gpm (1211 L/min) flow rates and to 5000 psi (345 bar) pressure capabilities.

SEQUENCE VALVES

A sequence valve is used to transfer flow to a secondary system only after an action has taken place in a primary system. It is a normally-closed valve that opens to the secondary system only when a preset pressure is reached in the primary system. Primary system pressure is maintained after the valve "sequences".

Sequence Valve Principle

Figure 101 is a schematic representation of a simple sequence valve. The piston is loaded by an adjustable spring, which sets the sequencing pressure. Until that pressure is reached, the piston blocks flow from the primary to the secondary port (View A).

When pressure under the piston rises to the spring setting (View B), the piston is pushed up and the valve operates in a balanced condition. Pressure is maintained in the primary system and flow is throttled to the secondary system.

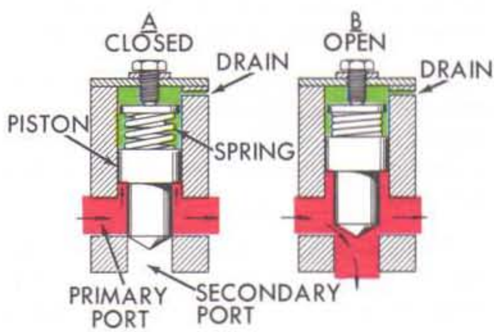


Figure 101

Balanced Piston Sequence Valves

Sequence valves also are built in two-stage designs for higher capacities and pressure ratings (Figure 102). Sequencing to the secondary port occurs when the pilot flow causes a 20 psi (1.4 bar) differential across the piston.

Notice that the path of pilot flow is through an external drain connection in the sequence valve. We can't send it through the center of the piston (as in the relief valve) because the outlet port is pressurized rather than connected to tank.

All sequence valves are externally drained.

"X" and "Y" Types

There are two modifications of this design valve for different pressure requirements in the primary system. In the "Y" type, the piston stem is hollow. Secondary system pressure is effective on the small area above the piston stem through the center passage. This balances the same pressure below the piston so that it doesn't affect the throttling action.

The "X" type piston has a solid center and the top of the stem is open to the drain passage. Pressure in the secondary system acts on the bottom of the piston and forces it wide open. In effect, the piston is vented as soon as sequencing starts. Pressure in both systems equalizes at the secondary system pressure.

"R" AND "RC" TYPE VALVES

"R" and "RC" type valves are simple spool-type pressure controls that can be used for several functions depending on how they are assembled and connected.

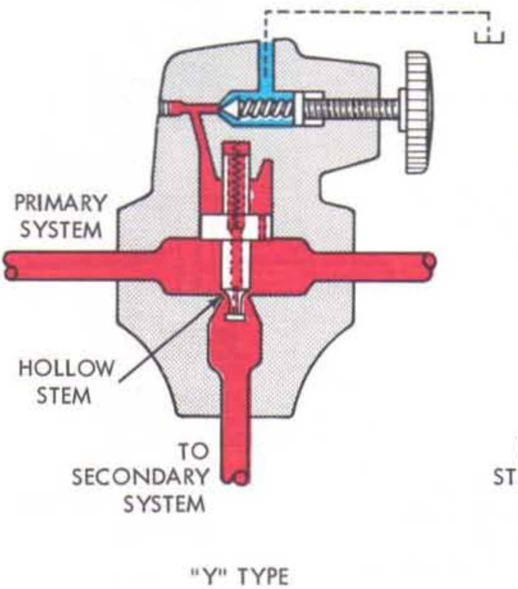


Figure 102

The "R" type valve (Figure 103) has a cylindrical valve spool fitted into a bore in the valve body. It is closed by two end caps . . . one containing an adjusting screw for the valve spring; the other a free piston with a control pressure passage terminating at one end. The other end of the piston butts against the valve spool. Pressure under the piston thus is balanced against the valve spring. The piston, having a very small area compared to the end of the valve spool, permits using a relatively light spring to control the valve.

The primary or inlet port of the valve is blocked from the secondary port in the normally-closed position. Oil is throttled to the secondary port when pressure under the piston reaches the valve setting and pushes the spool up against spring force.

RELIEF VALVE OPERATION

The relief valve function is shown in Figure 103. The primary port is connected to the pressure line and the secondary port to tank. The piston-end cover is assembled to connect the piston chamber to the pressure port internally. The valve relieves to the secondary port at the preset pressure.

Internally Drained. You can see that with high pressure in the piston chamber and primary port, there will be two internal leakage paths. One will be past the piston and under the spool. Since we don't want to build pressure under the spool, a drilled passage carries this leakage to the spring chamber. The spring end cover is assembled to drain this leakage to the secondary port. The other leakage path is past the spool and to the secondary port, which always is at tank pressure.

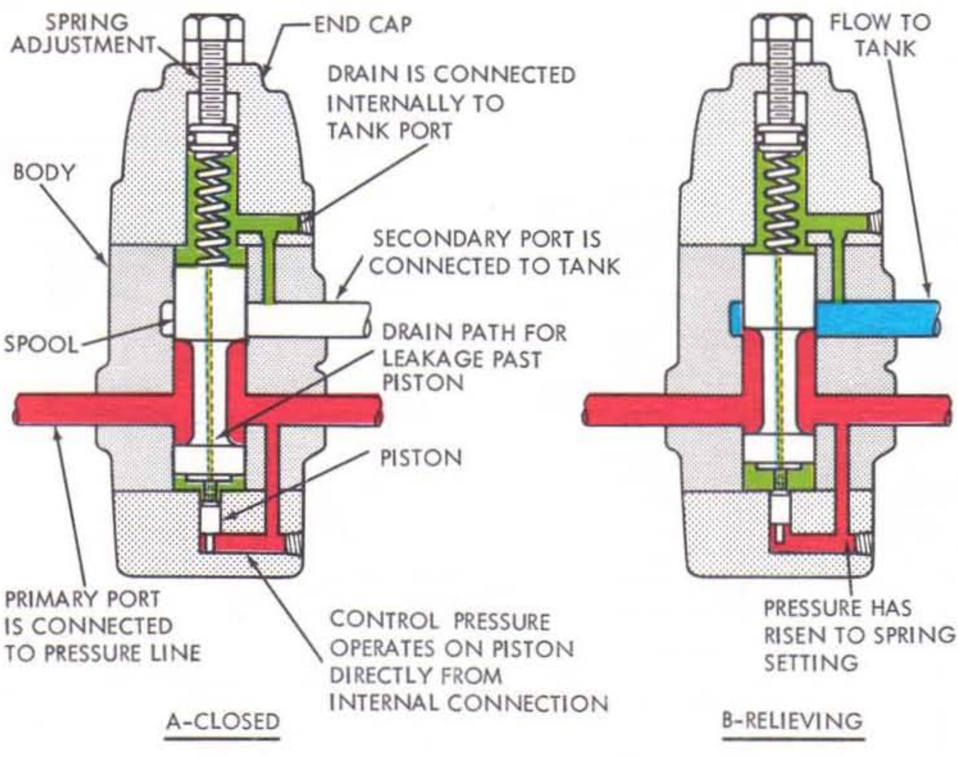


Figure 103

SEQUENCE VALVE OPERATION

To use the "R" valve as a sequence valve, we connect the primary port to the primary system and the secondary port to the secondary system (Figure 104). The valve is operated internally by primary system pressure. However, the spring-end cap is assembled to block the internal drain passage. An external drain must be supplied because the secondary port is under pressure when the valve sequences.

In this design, secondary system pressure can back up and force the valve wide open if it is higher than the valve setting. The primary system, then, would be subject to secondary system pressure.

Sequence and Check Valve. In many sequencing hook-ups, the same line that carries oil from the sequence valve to the secondary cylinder has to carry it back to tank when the cylinder is reversed. The sequence valve then being in its normally-closed position, we must find a way to by-pass it. A check valve can be piped around the sequence valve, but we'd probably use an "RC" sequence valve which has the check valve already built in (Figure 105). The check valve is closed when flow is going to the cylinder but opens to permit free flow from the secondary to the primary port on the return stroke. We call this check valve a return flow by-pass.

"RC" TYPE COUNTERBALANCE VALVE

The "RC" valve can also be used as a counterbalance valve (Figure 106). In this application, the valve is internally operated and internally drained. The primary port is connected to the lower port of a vertical cylinder and the secondary port to the reversing valve. The purpose is to create a

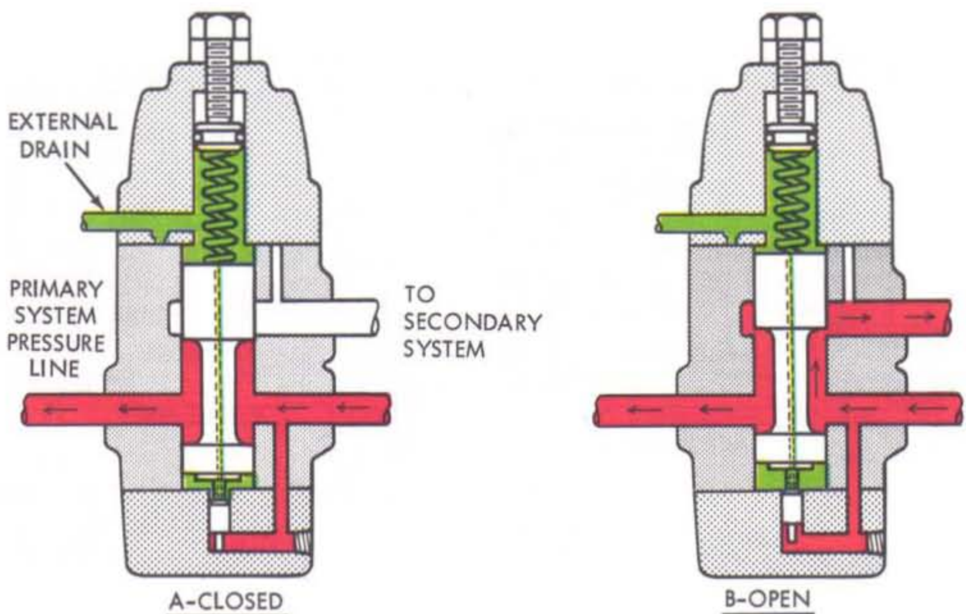


Figure 104

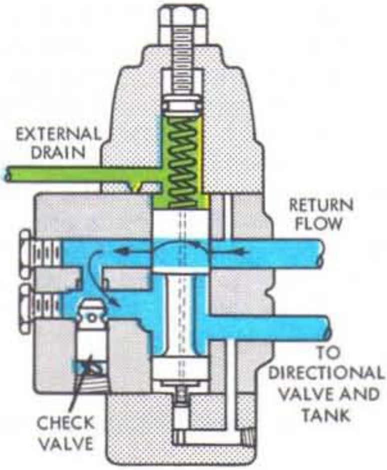


Figure 105

back pressure under the cylinder piston so that the pump will determine the rate of descent rather than gravity.

The valve is set at a pressure just higher than the load can generate from its weight. Thus, when the pump flow is diverted elsewhere, the return of oil from the cylinder is blocked and the load remains stationary.

When pump delivery is directed to the top of the cylinder, it forces the piston down (View A). Return flow pressure must build up to the counterbalance valve setting. This back pressure is maintained throughout the downward stroke preventing the load from falling out of control.

The check valve allows free flow to the cylinder when the reversing valve is shifted to raise the load.

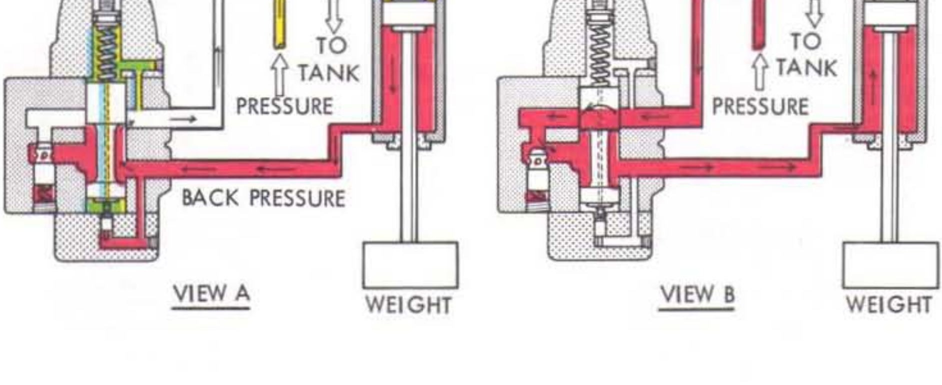


Figure 106

"RC" TYPE BRAKE VALVE

The brake valve application (Figure 107) is similar to counterbalancing. A brake valve is used in a hydraulic motor circuit to create a back-pressure for control during operation and to stop the motor when the circuit is in neutral.

Control is effected by two pressure areas with a differential of 8-to-1. The small piston is connected internally to pressure at the primary port. An external connection from the pressure line ports operating pressure under the valve spool, which has an area eight times the piston area.

In View A, the load is being accelerated from a stop. Motor torque is highest during acceleration, so pressure is maximum. With operating pressure under the large spool, the brake valve is forced wide open and exhaust flow from the motor is unrestricted. After the motor gets up to speed, the valve adjusts to create a back pressure if the motor tries to overrun the pump delivery. Any attempt to overrun would cause an instantaneous pressure drop at the large area under the spool. Pressure in the exhaust line, then, working under the small piston, would operate the valve like a counterbalance valve until pump delivery caught up.

View B shows the operation in neutral. The pump is unloaded through the directional valve and the motor is being driven by its load's inertia. Back pressure created by the valve spring balanced against pressure under the small piston decelerates the motor.

The internal check valve permits reverse free flow to turn the motor in the opposite direction.

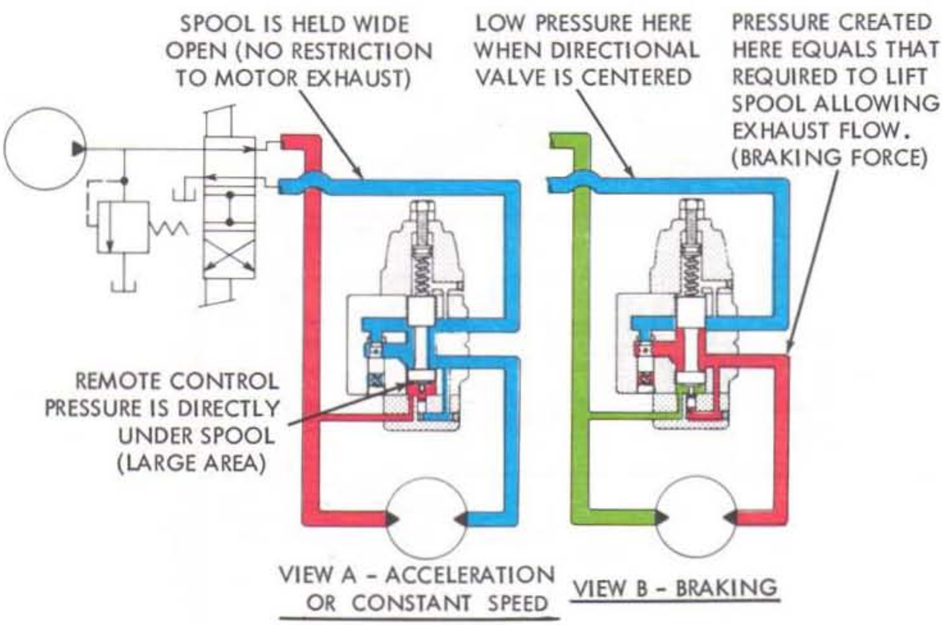


Figure 107

UNLOADING VALVE

Generally speaking, an unloading valve is a directional valve. It operates in one of two definite positions, open or closed. Its purpose, however, is to unload the pump, that is, to divert pump flow directly to the reservoir in response to an external pressure signal. So it might better be thought of as a pressure control.

As you see in Figure 108, the connection is the same as a relief valve. The difference is that the unloading valve is not operated internally, nor in balance. An external control connection is made by reversing the piston-end cover. Pressure from a remote source forces the valve wide open to route pump delivery to tank.

TILT CONTROL VALVE

A tilt control valve (Figure 108A) operates as a counterbalance valve on a tilt

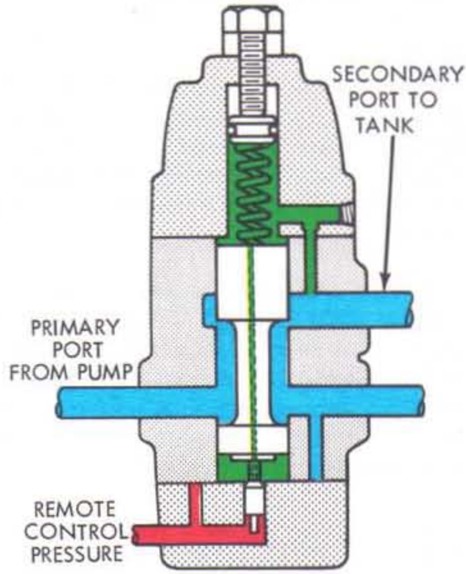


Figure 108

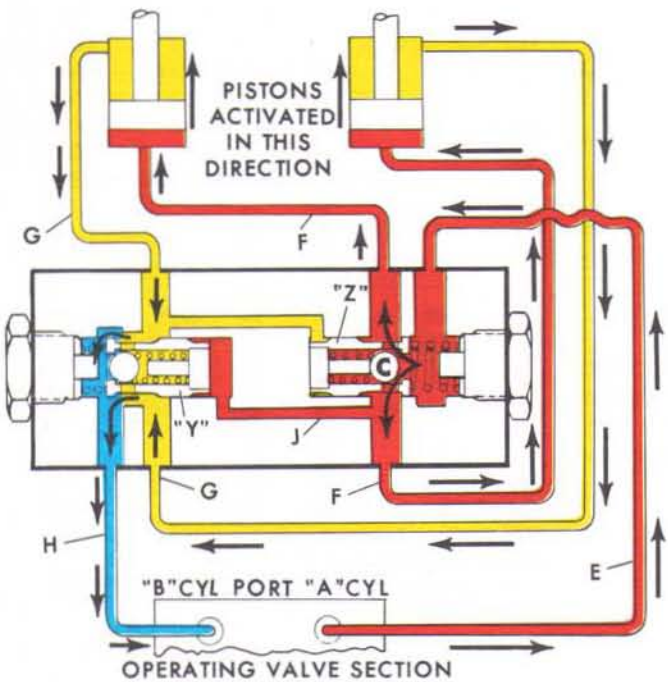


Figure 108A

cylinder (or cylinders), which might require control in either direction of movement. The two counterbalance valves (Y and Z) operate in balance between a spring and inlet pressure (passage J). Free flow check valves (C) are built into each counterbalance valve to permit flow in without restriction.

In the operating direction shown, oil enters the valve through line E and passes freely over the check valve to extend the cylinders. Return flow (passage G) is restricted by the opposite valve if required. With a positive pressure in passages F and J, the return is unrestricted. If the cylinder tries to run away, the pressure drops. The spring moves the valve to restrict return flow.

Operation is the same for retracting the cylinders, but flow is reversed and the opposite counterbalance valve provides control.

PRESSURE REDUCING VALVES

A pressure reducing valve is a normally open valve used to limit pressure in a branch circuit to something less than the pressure source. We use it to "get around" the law of least resistance.

For example, a high volume pressure-compensated pump might be operating a main system at 2000 psi (138 bar) and we want to tap into this system to move a load that only takes 500 psi (34 bar). The pressure reducing valve will pass only enough flow to maintain our 500 psi (34 bar) in the branch without losing pressure in the main system.

Operating Principle

Figure 109 shows the operating principle of a simple pressure reducing valve. Flow to the inlet passes right through to the outlet if the pressure is less than the valve setting. The pressure is sensed under the valve spool through an internal connection.

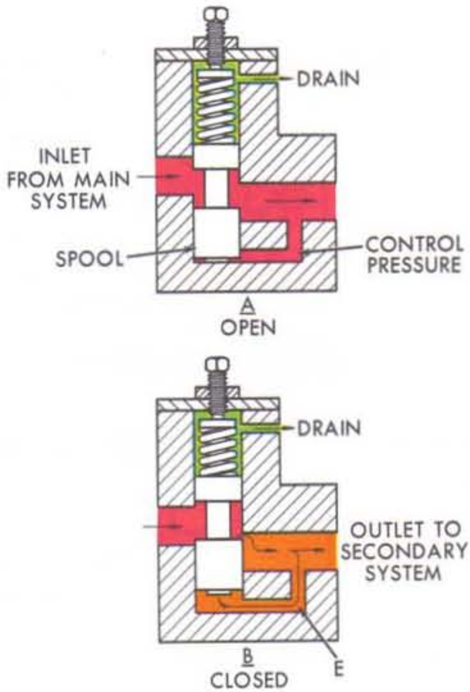


Figure 109

If the pressure at the outlet tries to increase above the valve setting, the spool is forced up (View B). This partially cuts off the flow from the main system. The spool assumes a position of balance between the valve's outlet pressure and the spring force. It varies the opening then to maintain the pressure setting at the outlet.

An external drain is required because both valve ports are subject to pressure.

Pilot Operated Design

The pilot-operated design of pressure reducing valve (Figure 110) is built in 8-gpm (30 L/min) to 125-gpm (473 L/min) sizes with operating pressure adjustable to 2850 psi (197 bar). The operation is essentially the same as the simple valve, except that the valve spool is controlled by a pilot valve. The operating pressure is set by adjusting the load on the pilot valve spring.

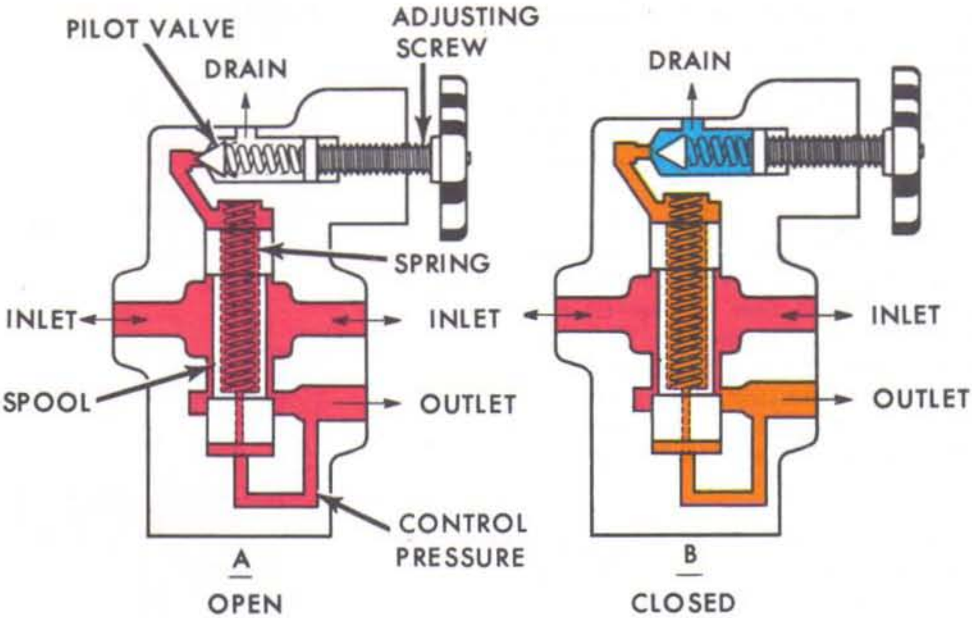


Figure 110

Below the valve operating pressure, the valve spool is held open by a light spring. The internal control pressure is equalized on opposite ends of the spool through an orifice in the spool.

When the valve setting is reached, the pilot valve cracks. Pilot oil flows through the orifice and the center of the spool, past the pilot poppet and out the drain connection. The pressure difference across the orifice lets the spool move up against spring force. It assumes a throttling position where outlet pressure at the bottom balances the combination of reduced pressure and spring force at the top.

Pressure Reducing and Check Valve

A modification of this design includes a reverse by-pass check valve (Figure 111). The check valve permits reverse free flow from the outlet back to the inlet at pressures

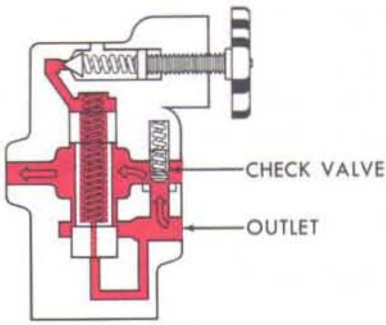


Figure 111

above the valve setting. There is no pressure reducing action in the free-flow direction.

UNLOADING RELIEF VALVES

An unloading relief valve (Figure 112) is a valve that has two functions. Used in an accumulator-charging circuit, it limits maximum pressure while the accumulator is being charged and unloads the pump when the accumulator reaches the pressure desired.

The valve is essentially a balanced-piston relief valve with a check valve built in to prevent the accumulator from discharging backwards through the valve. A pressure-operated plunger is built into the pilot stage to mechanically vent the relief valve when a preset pressure is reached. The relief valve remains vented until the accumulator pressure drops to about 85% of the valve setting. Then the valve closes and pump output again is delivered to the accumulator.

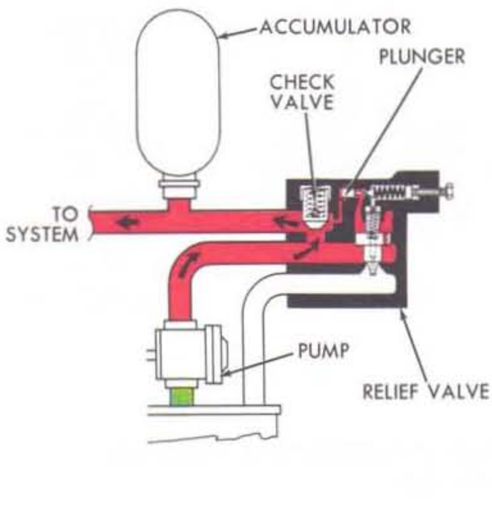


Figure 112

FLOW CONTROL VALVES

A flow control valve is used to control the actuator speed by metering the flow. Metering means "measuring" or regulating the flow rate to or from the actuator.

There are three ways to meter the fluid to control speed. We call them meter-in, meter-out and bleed-off.

Meter-In

To meter in (Figure 113), the flow control valve is placed in series between the pump and the actuator. It controls the amount of fluid that goes to the actuator. Whatever the pump delivers in excess of the measured flow is forced over the relief valve. This method is used in systems where the load continually resists pump delivery . . . for instance, raising a vertical cylinder.

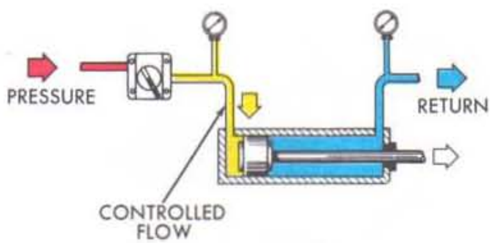


Figure 113

Meter-Out

If the load could tend to "run away" from the pump delivery, a meter-out connection (Figure 114) is preferable for flow control. The valve is placed between the actuator and the reservoir to control flow away from the actuator. As with the meter-in method, extra output from the pump goes over the relief valve.

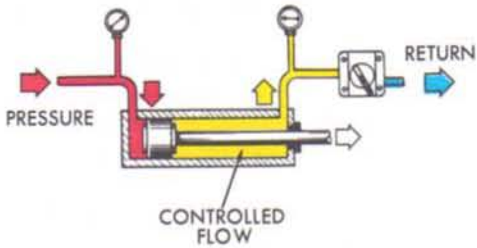


Figure 114

Bleed-Off

When the control of flow can be a little less accurate, the valve is connected to bleed off (Figure 115). It is placed between the pump outlet and the reservoir and meters the diverted flow rather than the working flow. The metered flow returns to tank at essentially the load pressure rather than relief valve pressure. The difference may be as much as 30-35 percent.

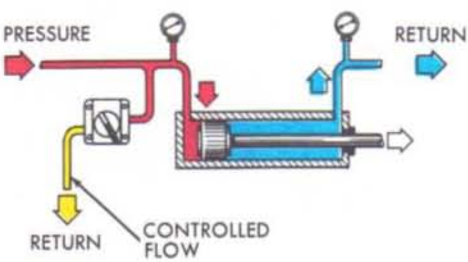


Figure 115

CLASSIFICATIONS

Flow control valves are rated according to capacity and operating pressure. They are classified as adjustable or non-adjustable, and may or may not be temperature- and pressure-compensated.

AN ORIFICE IS A FLOW CONTROL

An orifice, a simple fixed restriction, can function as a flow control valve. If it is placed in a line so that it controls speed by causing flow to be diverted or slowed, it is a flow control. Automation machinery uses many flow control valves that are nothing but fixed-size orifices.

NEEDLE AND GLOBE VALVES

A needle valve or globe valve (Figure 116) is an adjustable flow control valve. Turning a handle or knob or set screw adjusts the size of an opening to regulate flow. Control is fairly accurate so long as the load doesn't change.

If the load changes, we know that causes pressure to change. Any variation in pressure drop across the orifice will cause a variation in flow through the valve. For accurate control with varying loads, the flow control valve must be pressure compensated.

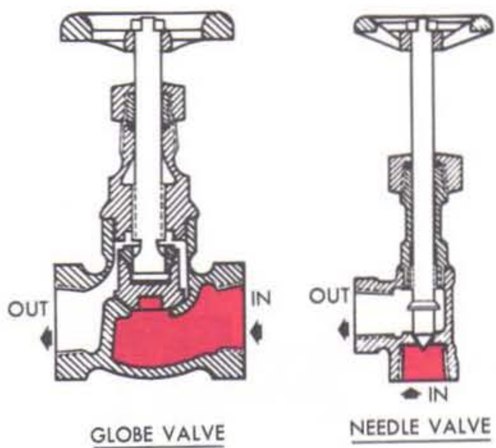
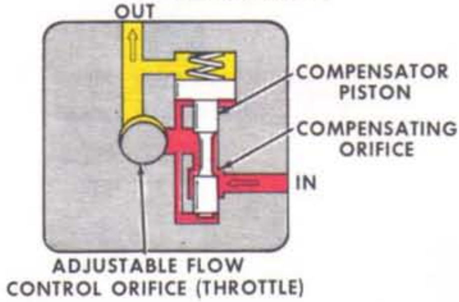


Figure 116

PRESSURE COMPENSATED FLOW CONTROL

A typical restrictor pressure-compensated flow control valve (Figure 117) has a flow control orifice that is adjustable to control

A– NON TEMPERATURE COMPENSATED



B – TEMPERATURE COMPENSATED

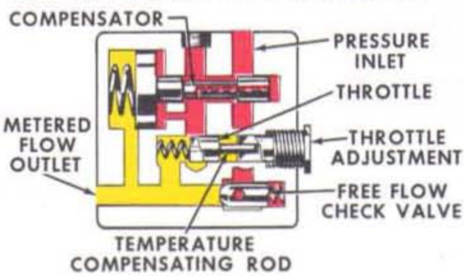


Figure 117

flow rate, and a compensator piston whose job is to maintain a constant pressure drop across the controlling orifice.

The compensator piston actually functions as a balanced valve. Pressure just upstream from the orifice is effective against two areas of the piston. Pressure beyond the orifice is effective against an equivalent area on the downstream side of the piston. This pressure downstream will be less than upstream because of flow across the orifice. So it gets the assistance of a 20 psi (1.4 bar) spring to balance the piston.

When flow starts, the compensator piston automatically takes a position of balance . . . maintaining a pressure difference of 20 psi (1.4 bar) across the orifice. A land on the compensator piston throttles just enough flow from the inlet port to maintain the constant pressure drop. The constant pressure drop results in constant flow.

Temperature Compensation

A temperature compensated design of this valve uses the expansion of metal from heat to make the orifice size smaller as the oil warms up. This prevents the flow rate from increasing when the oil is thinner and consequently flows easier.

FLOW CONTROL AND RELIEF VALVE

Another double-duty valve is the "FM" series flow control and relief valve (Figure 118). Similar to the "RM" relief valve, this design incorporates a fixed orifice for flow control. It functions both as a compound relief valve and a by-pass type pressure compensated flow control. We use it in applications where a constant actuator speed is required in the face of varying pump output . . . for example, in power steering.

Flow conditions are illustrated in Figure 118. In View A, the pump is being driven very slowly. Its output is less than our controlled flow rate, therefore, there is little pressure drop across throttle orifice 1. Pressure difference on the ends of the valve spool is not great enough to overcome the spring which holds it closed.

In View B, the valve spool is acting as a compensating piston. Pump output now is more than the controlled flow rate. Pressure upstream is being balanced against load pressure assisted by the light spring. The spool assumes a balance condition to maintain a 40 psi (2.8 bar) difference across orifice 1. Excess pump delivery is diverted to the tank.

View C shows the valve relieving. Pressure has reached the valve setting and we now have pilot flow to produce a pressure drop across orifice 2. Pump delivery is returned to tank at relief valve pressure.

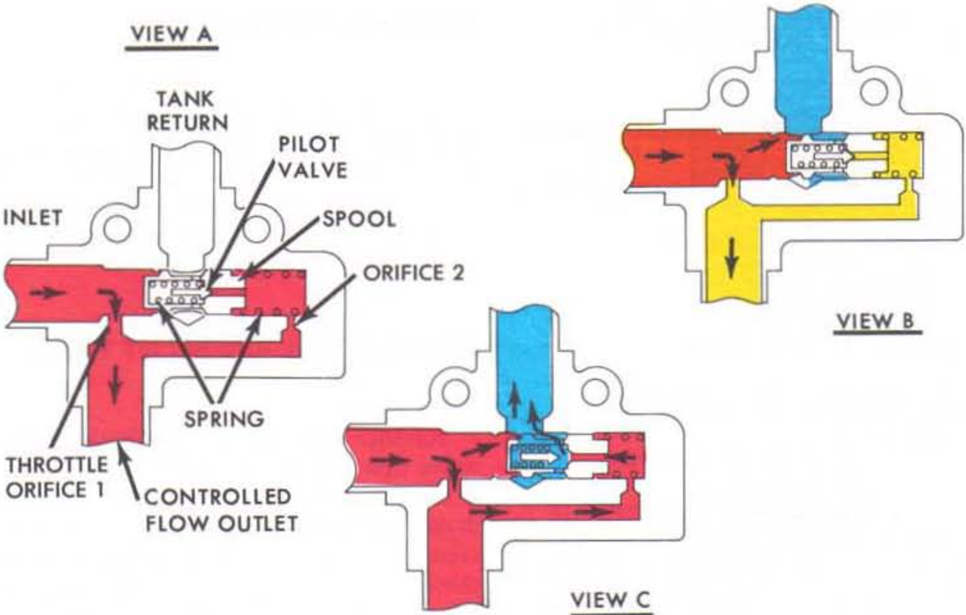


Figure 118

Besides the FM series valves, this flow control-relief valve arrangement is an integral part of all Vickers power steering pumps.

DIRECTIONAL CONTROL VALVES

Properly speaking, a directional valve is any valve which controls the direction of flow. But aside from check valves, which we've already described, most directional valves are the reversing or four-way type. By four-way, we mean the valve has four possible flow paths. Common practice in the industry is to apply the term directional valve to the reversing four-way valve. When we say a directional valve, we mean any valve that controls flow paths. When we say the directional valve, we're talking about a reversing valve.

Four Flow Paths

It is characteristic of reversing directional valves to have at least two finite positions, with two possible flow paths in each extreme position. The valve must have four ports: P-pump (or pressure), T-tank, and actuator ports A and B (Figure 119). In one extreme position, the valve has the pump port connected to A and the tank port to B. In the opposite position, flow is reversed: pump to B and A to tank.

Center Position is Neutral

If the valve has a center position, it is a neutral position; that is, the pump is either unloaded to tank (open center) or blocked from the other ports (closed center).

Where a neutral position is required, springs or detents are incorporated in the design to hold the valve "centered".

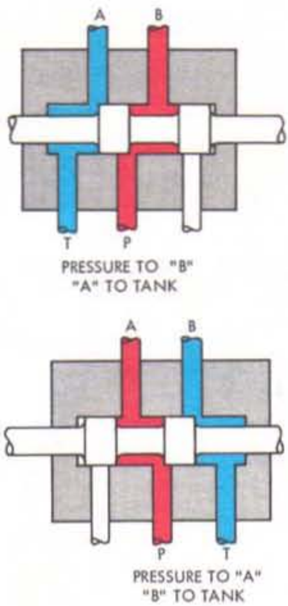


Figure 119

Finite or Infinite Positioning

Directional valves are usually considered finite positioning, having the three positions mentioned above.

In mobile applications where valves are operated manually they can be positioned infinitely between neutral and the two extremes to control both the direction and the rate of flow.

Controls

Directional valves can be shifted by many different controls . . . in fact, by anything that can move the valve spool. Handle (manual) controls and servo controls can be used for infinite positioning. For finite positioning (Figure 120) the valve can be shifted by pressure from another directional valve (pilot operation); mechanically by a cam or through linkage; by a push-type solenoid; or by any finite positioning control.

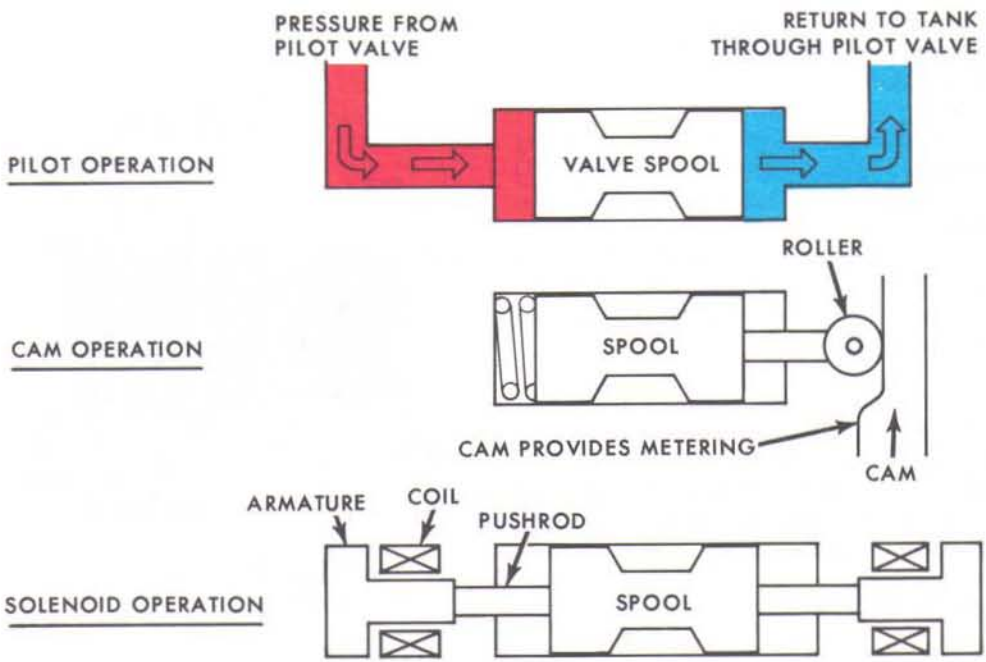


Figure 120

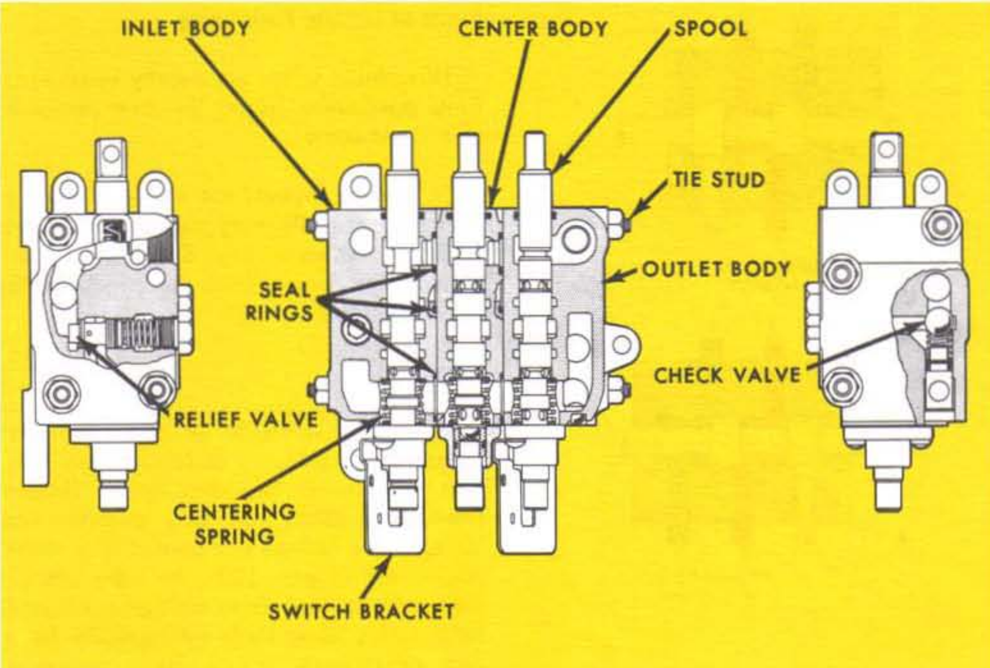


Figure 121

MULTIPLE-UNIT DIRECTIONAL VALVES

Multiple-unit directional valves, or simply mobile directional valve sections (Figure 121) are designed to be mounted in valve banks. Passages in the valves are interconnecting and sealed between sections. Tie studs of varying lengths hold the banks of valves together. Mounting lugs are provided in the inlet and outlet sections.

Each valve section contains a reversing valve spool and has two actuator or cylinder ports in its body. The pressure port is in the inlet section. Inlet sections contain relief valves to limit operating pressure. The tank port is in the outlet section.

Internal Passages

Internal passages through the valve bank (Figure 122) are a pressure passage, tank

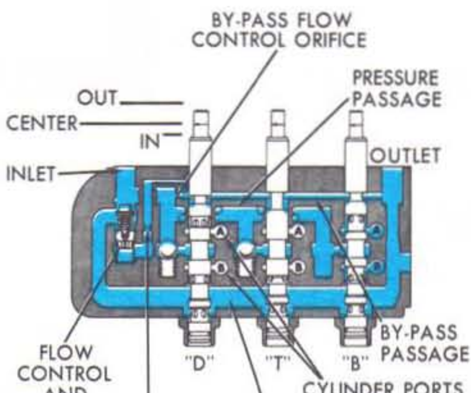


Figure 122

passage and by-pass passage. The pressure passage begins at the inlet port on the high-pressure side of the relief valve. In each valve section, it connects through a check valve to a chamber between the metering lands on the valve spool. Its purpose is to carry pump flow which can be directed to either of the actuator ports.

The by-pass passage is parallel to the pressure passage. It carries pump delivery to the outlet or tank port when the valve spools are in neutral. When a spool is shifted, a valve land cuts off or partially blocks this passage depending upon how far it is shifted. As the by-pass closes, the pressure port opens to a cylinder port. This combined action permits metering of the flow to the actuator.

A tank passage is provided to return exhaust flow from the actuators to the outlet port.

CM11 VALVE OPERATION

The smaller size of mobile directional valves is designated as the CM11 series. It is available with a double acting cylinder spool (D), orificed double-acting spools (A, A3, A4, A6 and A8), a double-acting meter spool (B), two types of single acting spools (T and W) and a float spool (C). The inlet relief valve shown in Figure 122 is the partial flow by-pass design, which limits flow through the by-pass passage in neutral for improved pressure drop characteristics. Earlier valves used a simple relief valve. Optional features include electric switch actuators, spool detents and narrow by-pass spools.

Neutral Operation

These valves are operated manually . . . either by handles mounted on the valves themselves or by linkage to remote controls. In Figure 122 you see the condition of a three-section valve when the controls are in neutral. Centering springs (not shown) at the spool ends are holding them in the center positions, so that the by-pass passage is open. The pressure passage is blocked between spool lands at each valve section. Pump delivery to the inlet is routed through the by-pass and tank passages to the outlet. The outlet being connected to the tank return line, the system pressure is tank pressure plus any back pressure created by drop in the return line and the valve.

The cylinder ports, except for the motor spool (B), are also blocked in neutral. Thus, a cylinder connected to the ports would be held from moving by a hydrostatic lock . . . that is, the incompressibility of the oil trapped in the lines.

Double-Acting "D" Spool

For a simple operation of reversing a cylinder, we use the "D" spool (Figure 123).

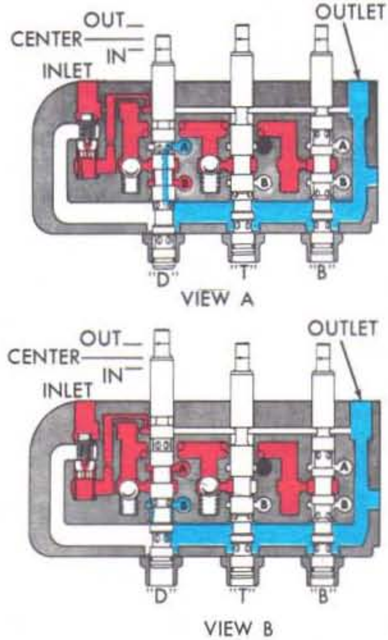


Figure 123

(The "D" refers to the spool designation in the valve model number.) When we actuate the control to shift the spool in (View A), the pressure passage is connected to the B port. The A port is connected to the tank return through a passage drilled in the valve spool.

To reverse the cylinder, the spool is shifted out (View B). Now the pressure passage is connected to port A and port B is open to tank return.

Metering Operation

We are able to build pressure in the pressure passage only when a spool is shifted. Notice in both views of Figure 123 that the spool has the by-pass passage blocked. All of the pump delivery is being directed to the cylinder ports.

Mobile directional valves are used to meter flow to the cylinder ports by "feathering" or shifting the valve less than fully off-center. Shifting the valve only slightly "cracks" the pressure port to the cylinder port while only partially blocking the by-pass. Thus, we are able to divide the flow . . . sending partial pump delivery to the cylinder and by-passing the rest to tank. The valves are infinite positioning between center and wide-open, and they therefore function both as directional valves and flow controls. The standard by-pass is designed to handle the normal flow rates of the valve. Narrow by-pass spools are used for better metering when the flow rate is low (small pump).

Two At A Time

A significant feature of these valves is that a skilled operator can use two of them at once. They operate in parallel.

If two valves are shifted wide-open, the oil will, of course, take the easiest route, and probably only one load will move.

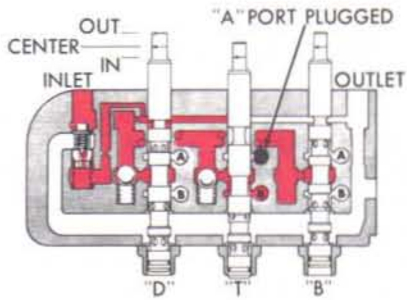
But if the operator is "feathering" the valves, pressure is backed up from each spool and two or more spools can be operated moving two or more loads simultaneously.

Single-Acting "T" Spool

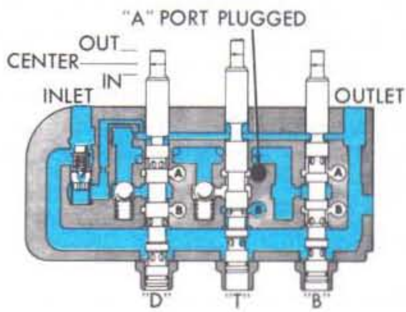
A "T" spool as illustrated in Figure 124 is used to reverse a single-acting cylinder. It directs flow to and from port B only. Port A is simply plugged.

Shifting the spool in (View A) blocks the by-pass and opens the pressure passage to port B. Oil is routed to raise the cylinder. The cylinder is lowered (View B) by shifting the spool out. If it is only cracked from center, it will allow the load to lower slowly; if it is wide-open, the load lowers quickly.

Since the A port in this section is not used, there's no need to block the by-pass when the valve is shifted out. The pump



VIEW A



VIEW B

Figure 124

thus remains unloaded while lowering the load. . .unless another valve is being operated at the same time.

Single-Acting "W" Spool

Our single-acting cylinder can be operated from the A port by using a "W" spool (Figure 125). This spool is simply the reverse of the "T" spool. In the out position, oil is directed to port A; shifting the spool in returns the oil to tank.

With both the "T" and "W" spools, oil is trapped under the cylinder piston in neutral.

Double-Acting "B" Spool

A "B" spool has A and B ports open to tank in neutral and is called a "motor" spool.

A hydraulic motor turning a load at any appreciable speed will try to keep turning when the valve is centered because of the load's inertia. If the load inertia is allowed to drive the motor, the motor acts as a positive displacement pump. Using a "D" spool to reverse the motor would block its outlet in neutral and could result in extremely high pressure and shock. Instead, we use a spool that has the actuator ports open to tank in neutral (Figure 126). (The motor can then be decelerated by a brake valve if necessary (Figure 107).)

In the shifted positions the "B" spool operates exactly as the "D" spool.

Double-Acting Spools With Return Restrictions

In some applications, we have the possibility of the load running away from the pump. It was noted earlier that when we use a flow control on this type application, we connect it to meter out. Our mobile directional valves, though, are designed to meter in. Therefore, we also have special spools with return restrictions to maintain back-pressure on the load. The restrictions, in effect, operate as counterbalance valves and still permit us to meter in.

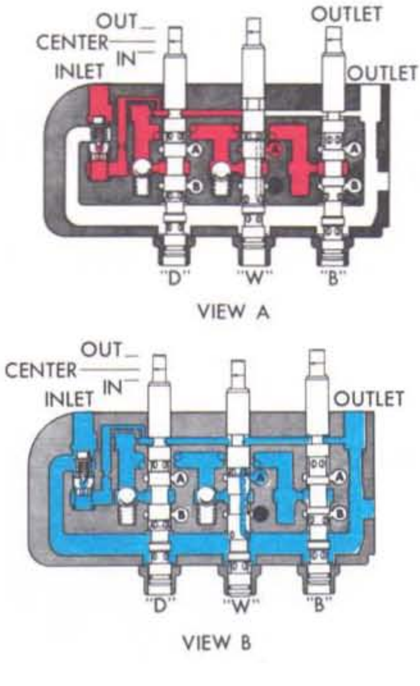


Figure 125

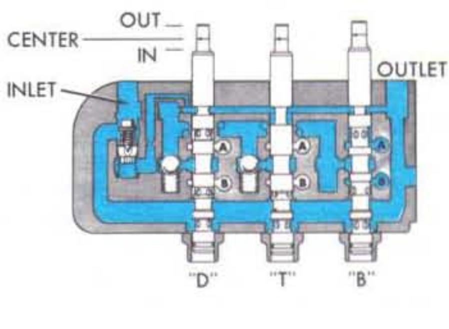


Figure 126

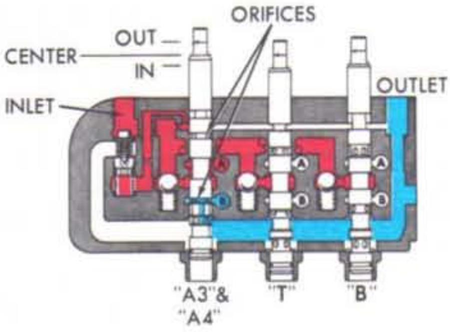


Figure 127

"A3" and "A4" spools (Figure 127) have smaller holes drilled into the spool tank return passage. The holes actually are fixed orifices which restrict return flow to the tank passage. The "A3" and "A4" type differ only in orifice size.

The "A" spool (Figure 128) has a variable orifice . . . an internal spring-loaded spool that operates as a counterbalance valve.

Inlet (operating) pressure on the internal spool is balanced against the spring. When there is a positive inlet pressure, the internal spool permits unrestricted return flow as shown. If the load tries to run away, inlet pressure drops and the spring moves the internal spool up to restrict return flow.

Other than the return flow restrictions, "A", "A3", "A4", "A6" and "A8" spools operate as "D" spools.

Double Acting "C" (Float) Spool

A "C" spool is a double-acting spool with an open or "float" position. It is used in applications where the operator may want the cylinder hydrostatically locked at times and free to float at other times.

For instance, a bulldozer blade would be positioned and locked in forward opera-

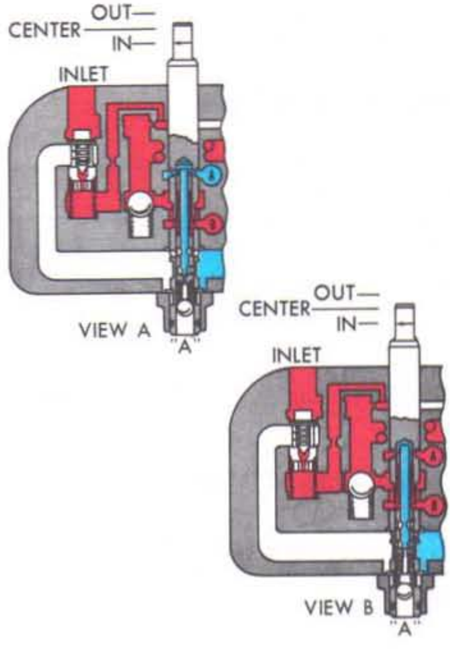


Figure 128

tion. But for back-leveling, it might be desirable to let it "float" or move up and down over irregularities. The "C" spool gives the operator this option.

As you see in Figure 129, the "C" spool has four positions. It is detented in center

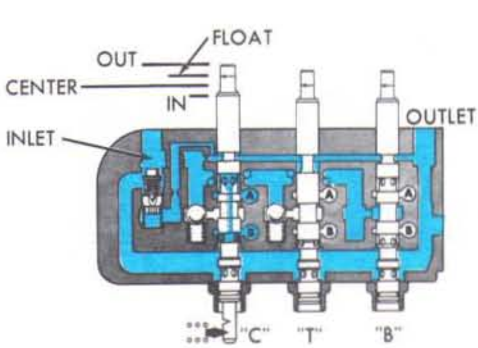


Figure 129

and float. In float, the cylinder ports are both connected to the tank passage; in center, they both are blocked. Operation otherwise is the same as the "D" spool.

Pressure Passage Check Valves

We've seen that these valves can be positioned so that the pressure passage is cracked to a cylinder port with the by-pass only partially blocked. Since the by-pass and pressure passages are hydraulically the same, then, a weighted load could force oil under the cylinder piston to return back through the by-pass when the valve is cracked.

The check valves in the pressure passage prevent return flow through a cracked valve until the by-pass closes enough to match the load-generated pressure. They are used in all operating sections, except with the motor ("B") spool.

Flow Control and Relief Valve

A relief valve is always placed in the inlet body and limits maximum operating pressure of any of the operating sections. In early designs, a simple relief valve was used, and functioned only as a pressure limiter.

In the latest design, we provide a partial flow by-pass system, where the compound relief valve also functions as a pressure-compensated by-pass type flow control (Figure 130). An orifice at the entrance to the by-pass limits by-pass flow to about seven gpm. The rest of the flow is diverted to the tank passage. In older designs, all pump delivery had to go through the by-pass. This often resulted in very high pressure drop when the pump was driven at high speed.

The partial flow by-pass system actually operates identically to the FM flow control and relief valve. The only differences are that it is built into the CM11 inlet body

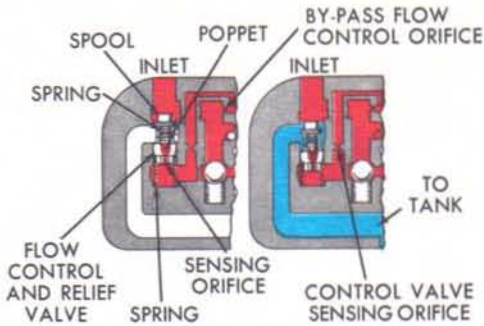


Figure 130

and controls flow through the by-pass rather than to the load. This operation is illustrated in Figure 122. When a spool is shifted to block the by-pass, the flow control becomes inoperative and the valve operates as a compound relief valve (Figure 130). We have described this operation in detail earlier in the chapter.

Tandem Operation

It is possible to connect two banks of valves in tandem to operate them from the same pumping source (Figure 131). To do so, we install a plug in the first section to separate its by-pass and tank passages. The tank passage then is connected to the reservoir through an alternate discharge port. The by-pass is connected in series to the inlet of the next valve bank. The pressure line connects both banks in parallel.

Either valve bank can now be operated separately, or both can be operated together. If neither is operating, flow is divided at the flow control valve in the first bank. Part of it goes through both by-passes to tank; the rest through the tank passage and alternate discharge port of the first bank.

With both pressure inlets in parallel, the relief valve in the first bank protects both systems. So in the second bank, a plug

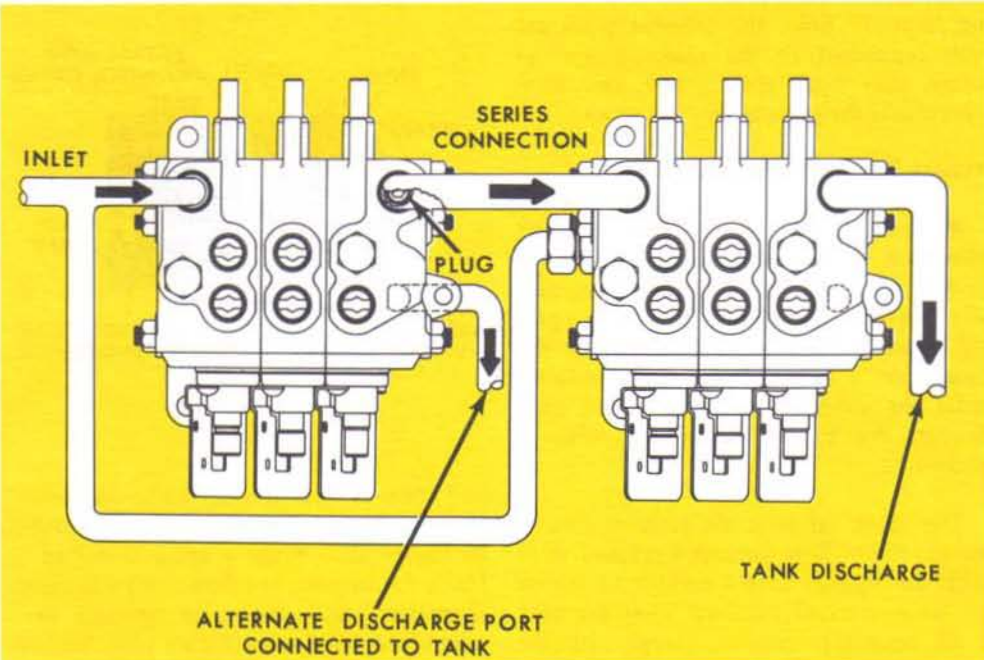


Figure 131

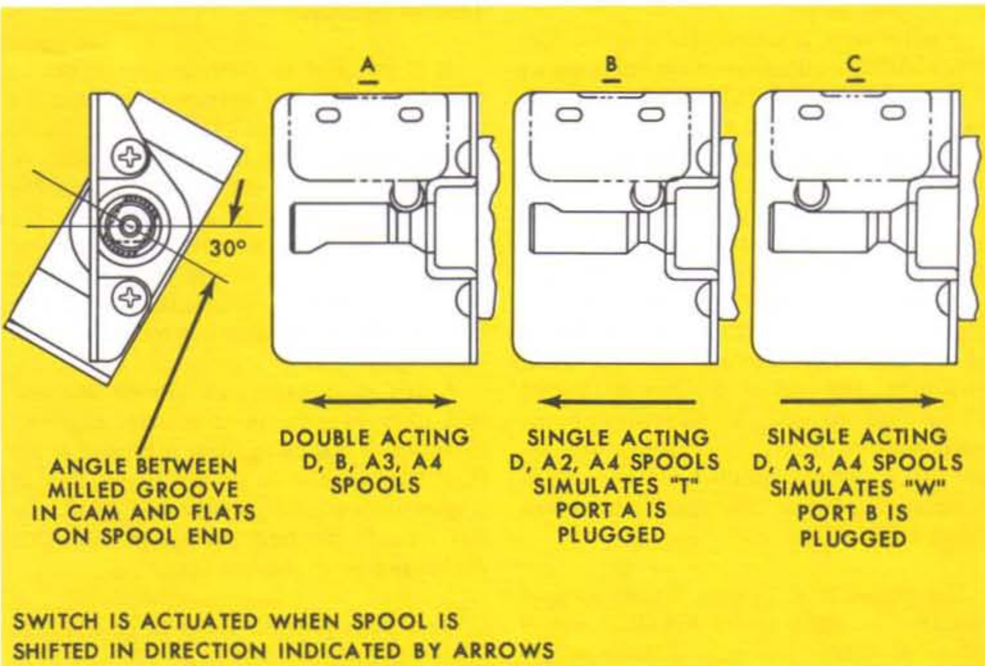


Figure 132

replaces the relief valve spool. It is used just to seal the pressure inlet from the tank passage.

Electric Switch Sections

An electric motor-driven pump uses a certain amount of power whenever it is running . . . unloaded or not. On many machines, particularly battery-powered lift trucks, the pump is not run constantly so that power can be conserved. An electric switch starts the pump drive motor only when the pump is needed.

Often the electric switch is mounted on the end of the valve body and actuated by a cam extension on the spool (Figure 132). In the center position, the switch is open and the pump stops. When the spool is moved to a position that demands pump output, the switch closes to start the pump.

Single-acting ("T" and "W") spools are replaced with double-acting ("D", "B", "A3" or "A4") spools when electric switches are used. To simulate the "T" spool, we change the switch cam position (View B) so that the pump starts only when the spool is moved out. For reverse single action (View C), the switch is turned over and closes only when the spool is shifted in.

Of course, with a double-acting spool, we do block the by-pass passage when the spool is shifted to lower the cylinder. But you can't put a load on a pump that isn't running, so this doesn't create any problems.

Spool Detents

Spool detents are used when the operator may want to shift the valve to an extreme position and hold it from being returned to center when he lets go of the handle. Figure 133 shows how this is done. A special end cap supports a spring-loaded plunger that engages in notches in a spool extension. There are two detent arrangements: "in",

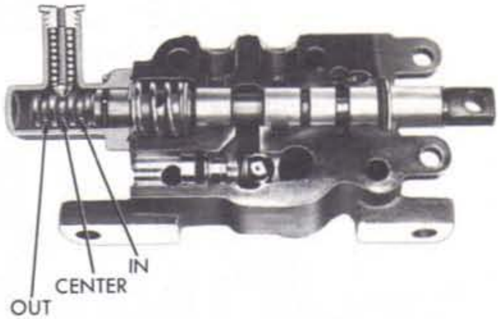


Figure 133

"center" and "out" position for all except float spools; and center and float only for float spools.

CM2-CM3 SERIES VALVES

CM2 and CM3 valves (Figure 134) are larger sized mobile directional valves. Differing slightly in construction, the larger valves have multiple tank passages and use poppet check valves instead of balls. But their operating principles are the same as the valves we just described.

Operating Spools

Four operating spools are available: a double-acting motor spool ("B"); a double-acting float spool ("C"); double-acting cylinder spool ("D") and single-acting ("T" or "W"). Their flow paths are shown in Figures 135, 136, 137, and 138. Except for locations of passages, the operation is identical to CM11 valves.

Flow Control and Relief Valve

The latest designs of these valves also use the partial flow by-pass system with a combined flow control and relief valve (Figure 139). A special tank passage is supplied for by-passed flow. Otherwise

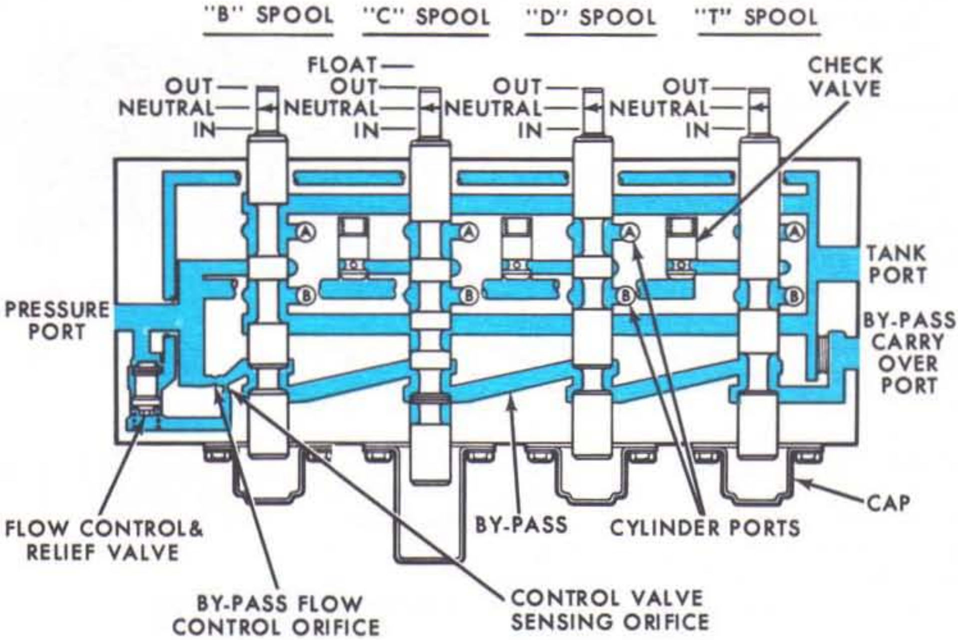


Figure 134

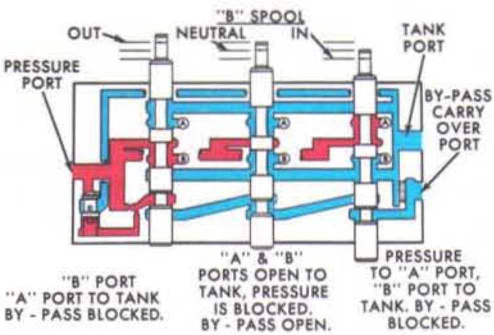


Figure 135

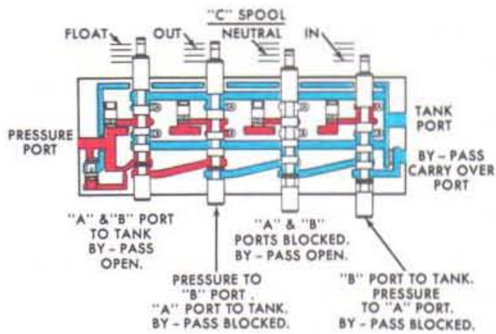


Figure 136

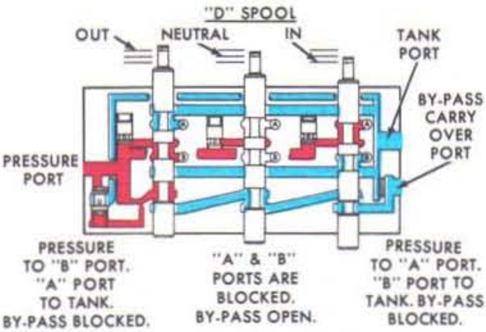


Figure 137

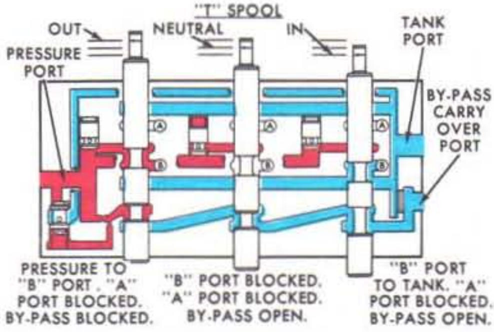


Figure 138

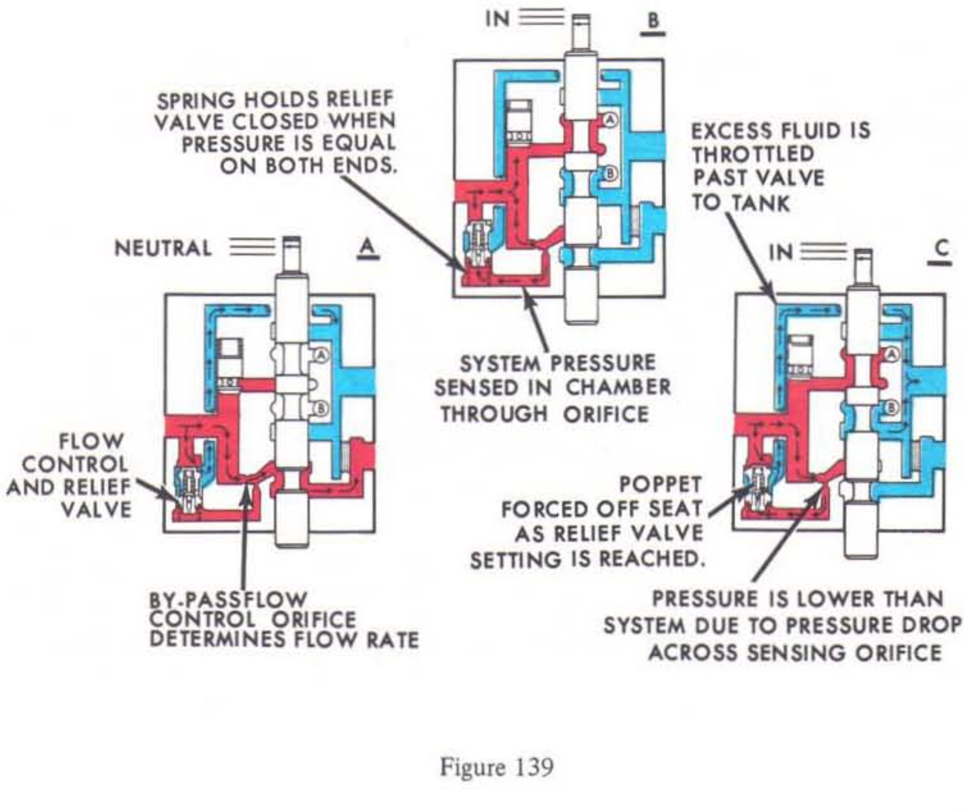


Figure 139

there's no difference in operation from the CM11 design.

Tandem Operation

A standard tandem hook-up of two valve banks (Figure 140) operates the same as the hook-up in Figure 131. Both banks can be operated at once; or either bank separately.

In some cases, a pure series connection may be desirable to give the first bank priority over the second. For that type operation, a special outlet section is used in the first bank and the parallel connection isn't used. The second bank can be operated only on by-pass oil from the first bank. If a spool in the first bank has the by-pass closed, the second bank is out of service.

Series Center Section

A special center section, called a "U" section, is available to "criss-cross" the ports of two adjoining motor valve (B) sections. Sandwiched between the two valves, the "U" section ports the outlet of the first section to the inlet of the second. The motors then operate in series.

This arrangement allows both motors to run at full speed because each can receive full pump delivery instead of sharing it with the other. But of course, in series operation, the pressure developed will be the sum of the pressure drop across the two sections.

Pre-Fill Valve Option

Another option on these larger valves is a prefill or anti-cavitation check valve. This

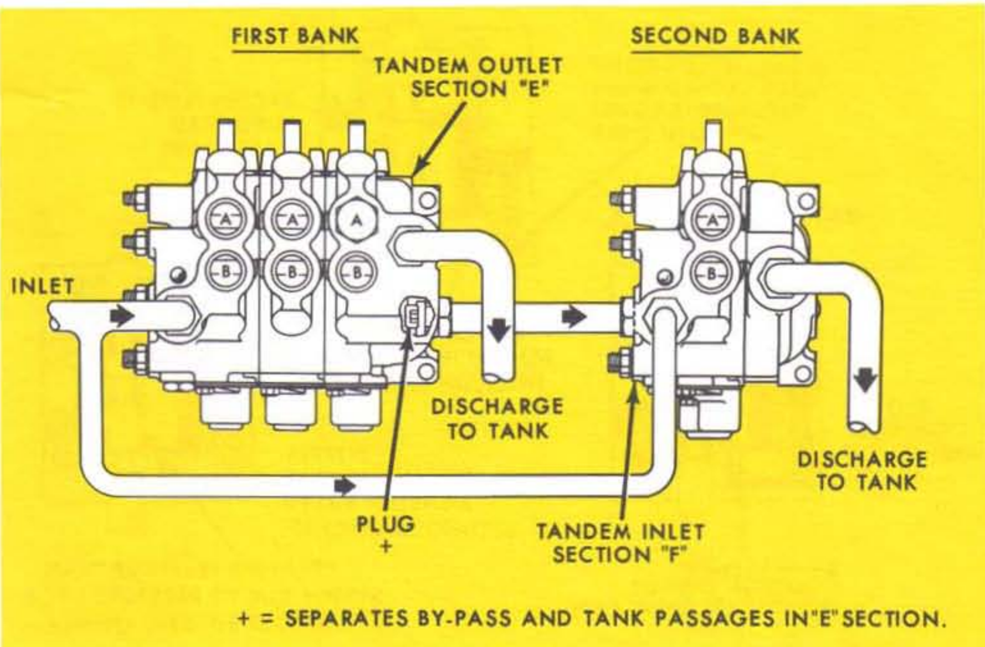


Figure 140

valve can be installed on the cylinder ports where required to prevent cavitation if a load is running away from the pump. The prefill valve admits fluid from the tank passage back to the cylinder port to prevent the overrunning load from "pulling a vacuum" on the pressure side.

The prefill valve also is available with a cylinder port relief valve, which operates for the particular port only. This relief valve can be set higher than the system relief valve in the inlet body. It might be required with a bulldozer cylinder, for example. Pushing several tons of dirt against a hydrostatic lock could generate an excessive pressure in the cylinder. With the valve in neutral, this pressure would not be felt back at the main relief valve.

CMD SERIES DIRECTIONAL CONTROL VALVES

The CMD Series valves (Figure 141) are called Enblock Valves because all of the component parts are assembled in a one piece body. They are balanced, spring-centered, sliding spool units designed for nominal flows to 250 gpm (946 L/min) and pressures to 3000 psig (207 bar). The valves have open center circuitry allowing free

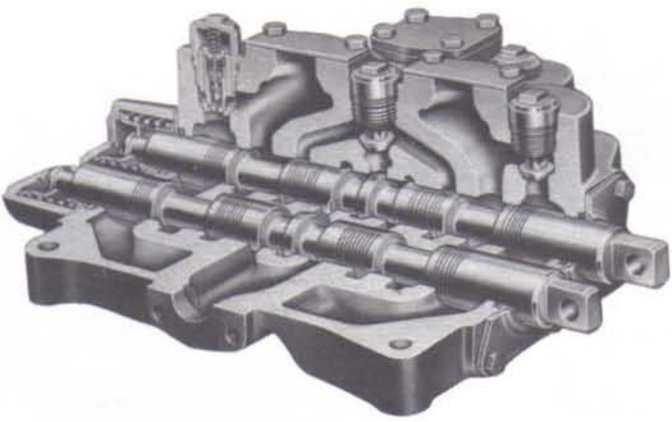


Figure 141

flow from inlet to outlet in the neutral position. Depending on size, they may be pilot operated or manually actuated with or without a power assist actuator. The actuator (Figure 142) provides for remote operation of the valve and greatly reduces operators' effort. Spool and associated mechanisms are housed in a one-piece valve body. Integral load drop check valves are standard. Cylinder port relief valves and anti-cavitation check valves can be included if required. Built in system relief valves are specially designed to assure stable operation and minimize noise. Detent release mechanisms are available for most valves. They are capable of holding the spool in position at maximum rated flow and pressure. Detent release assembly options are of manual, hydraulic, electrical or pneumatic types.

Load Drop Check Valve

The timing of the directional valve spool is such that the pressure passage to the cylinder port opens before the bypass passage is closed. This timing allows pressure to build up gradually within the valve to match load demand. The load drop check prevents reverse flow and supports the load. When the system pressure matches the load pressure, the check valve opens, porting the fluid to the actuator.

System Relief and Cylinder Port Valves

The system relief valve (Figure 143) is of the pre-set design, although an adjustable model is available. Its function is to prevent overloading of the main hydraulic circuit.

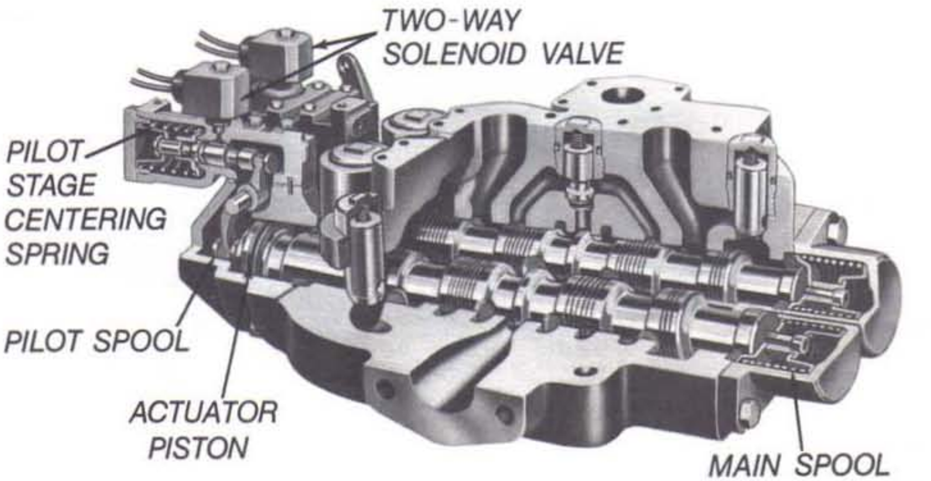


Figure 142

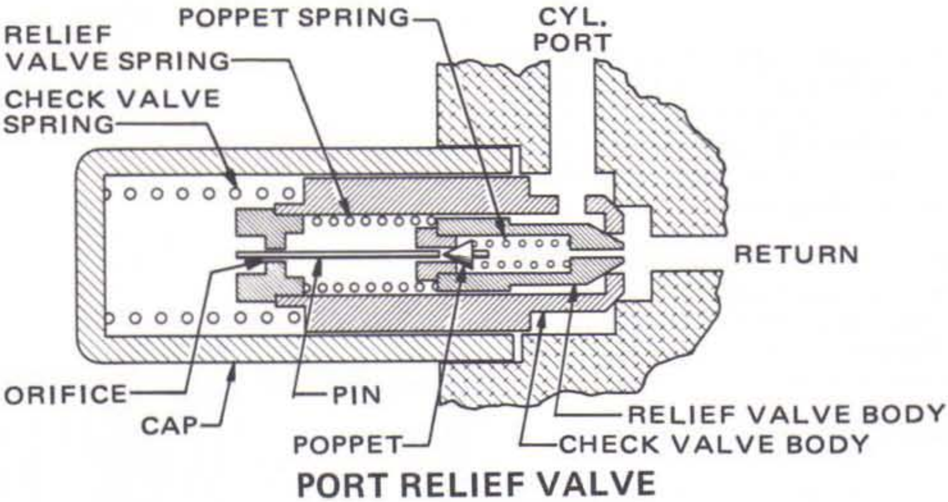


Figure 143

Port relief valves prevent excessive pressure build-up on a particular cylinder or motor if required for sudden stops or braking. Anti-cavitation check valves permit flow from the tank into the cylinder ports when actuator speed exceeds flow capability of the pump. An example would be the dropping of the bucket on a front end loader. Anti-cavitation check valves can be used singly or in combination with cylinder port relief valves (Figure 144).

SERVO VALVES

In the glossary of this manual, you will find the term "servo" defined as:

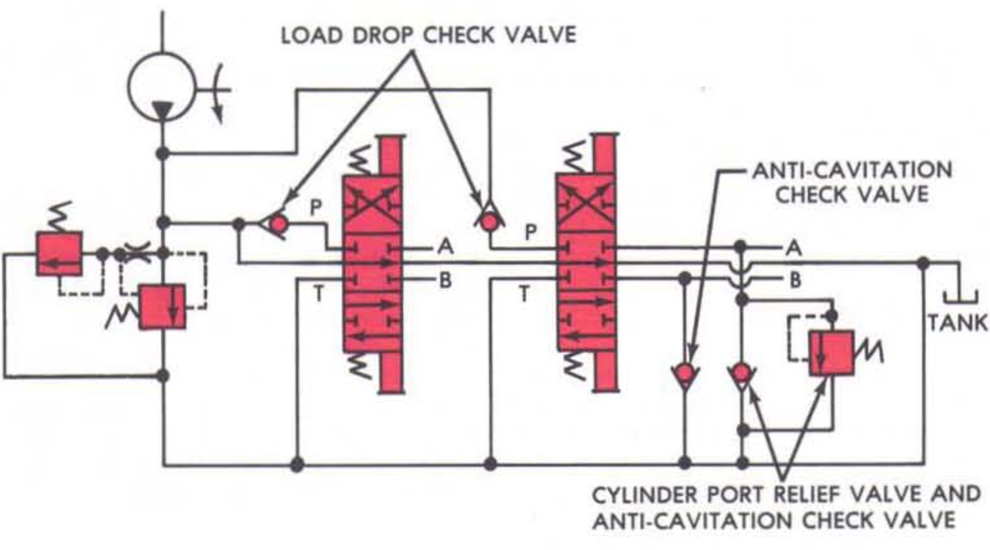


Figure 144

A mechanism subjected to the action of a controlling device so that it will operate as if it were directly actuated by the controlling device; but capable of supplying power output many times that of the controlling device . . .

This is a general definition relating to any kind of servomechanism. In hydraulics, we encounter two very distinct kinds of servomechanisms: mechanical and electrical (or electro-hydraulic).

Follow Valve (Mechanical Servo)

A follow valve is a mechanically-operated servo valve. In a mechanical hydraulic servo (Figure 145), we use a follow valve with a cylinder to provide a hydraulic boost to move a load. In most instances, we connect the valve body to the load and our control to the valve spool. Then, when we move the spool, it directs oil to a cylinder to move the load. But the valve body, being connected to the load, follows the spool. So when the spool stops, the valve body catches up to it. Their relative positions become neutral to stop flow to the cylinder.

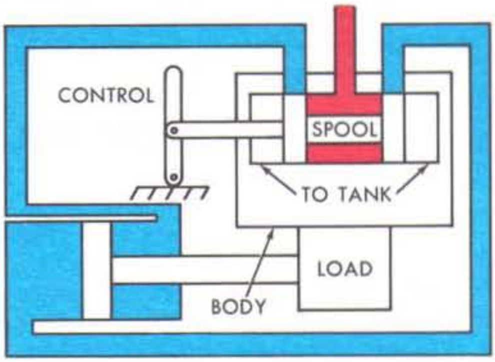


Figure 145

Thus, the load moves only when the control moves and stops when the control stops. The only purpose of the servomechanism is to provide a hydraulic boost where manpower isn't enough to do the job. We often use mechanical servos to actuate the displacement control of a variable piston pump or motor. A power steering system also is a mechanical servo.

Electrical Servo Valves

A mechanical servo is a positioning device only. It moves the load to a certain place and stops it. Electrical servo valves can be used for either positioning or velocity controls, or both. An electro-hydraulic servo valve is controlled by a voltage signal which actuates the valve spool, or a pilot valve which in turn actuates the valve spool. For velocity control, we make the valve infinite positioning in response to the magnitude of the control signal.

An electrical servo valve has no mechanical connection to the load. Rather, some device such as a tachometer-generator or a potentiometer is actuated as the load moves. This device produces a voltage that tells the valve where the load is or how fast it is moving. The signal that comes from the load is called the feedback, and is compared to the control signal. If there is a discrepancy in the system position or velocity, an error signal develops to actuate the servo valve spool thereby causing a correction.

CHAPTER 6
Mobile Circuits and Circuit Diagrams

This chapter has two purposes: to show you how to find your way around graphical circuit diagrams; and to acquaint you with some of the applications of the components to mobile equipment.

We'll begin by seeing how the various components and lines are diagrammed and how to follow flow in a graphical circuit. Then we will analyze the hydraulic circuits for several different kinds of mobile vehicles . . . describing in detail how various operations are accomplished.

CIRCUIT DIAGRAMS

Accurate diagrams of hydraulic circuits are essential to the designer, to the people who must build the machine, and to the man who must repair it. The diagram shows how the components will interact. It shows the manufacturing engineer and the assembler how to connect the components. It shows the field technician how the system works . . . what each component should be doing and where the oil should be going . . . so that he can diagnose and repair the system. Whichever of these categories your interest in hydraulics falls into, you'll certainly want to be able to "read the blueprints" . . . to interpret the diagrams and be able to analyze and classify circuits from their diagrams.

NOTE

A fluid power diagram is a complete drawing including description, sequence of operations, notes, component list, etc.

KINDS OF DIAGRAMS

There are basically four kinds of circuit diagrams you'll encounter in studying hydraulics. You'll also find many combinations of the four.

A block diagram indicates the presence of components with lines between the blocks to show connections and/or interaction. We have used partial block diagrams in this handbook in combination circuits where we were not concerned with a detailed study of the "blocked" sections. In fact, we will continue to use blocks in this chapter to indicate the presence of components we aren't ready to discuss.

Most of the diagrams we've used so far, though, have been cutaway diagrams. Cutaway diagrams are ideally suited to instruction because they show the internal construction of the components as well as the flow paths. By using colors, shades or various patterns in the lines and passages, we are able to show many different conditions of flow and pressure. Cutaway diagrams, of course, are expensive to produce and take considerably longer because of their complexity.

Pictorial diagrams are designed to show the piping arrangement of a circuit. The components are seen externally, usually in a close reproduction of their actual shapes and related sizes. The pictorial diagram in Figure 146 also shows the locations of the components.

Graphical diagrams, the "shorthand" system of the industry, are usually preferred for design and trouble shooting. A graphical diagram is made up of simple geometric symbols for the components and their controls and connections.

Figure 147 is a graphical diagram for part of the circuit in Figure 146. Comparing the two, notice that the graphical diagram doesn't show anything about the construction or relative locations of the components. Its purpose is to show functions, port connections and flow paths.

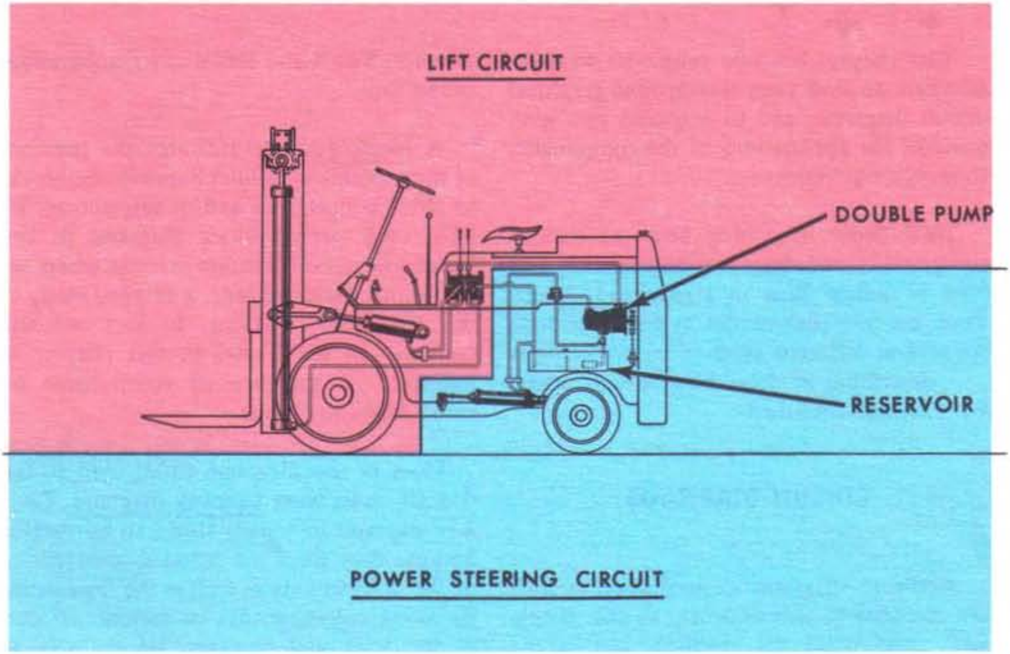


Figure 146

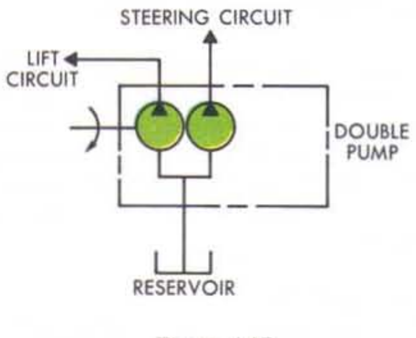


Figure 147

We will cover a similar lift truck circuit later in the chapter and power steering systems in Chapter 7. But first we'll discuss the component symbols one at a time and see how they build up to make a simple circuit.

THREE SYSTEMS OF SYMBOLS *

We are using the new set of A.N.S.I. graphical symbols in this manual. You may encounter circuits using the old A.S.A. or J.I.C. symbols. There are many differences. The new A.N.S.I. symbols are designed to eliminate the use of letters, so they are capable of crossing language barriers and can promote a universal understanding of fluid power systems.

There is enough similarity between the two systems, though, that if you understand the A.N.S.I. system, you'll be able to

* A.N.S.I. = American National Standards Institute
A.S.A. = American Standards Association
REF. ASA Y32. 10-1958 & ASA Y14. 17-1959
J.I.C. = Joint Industry Conference

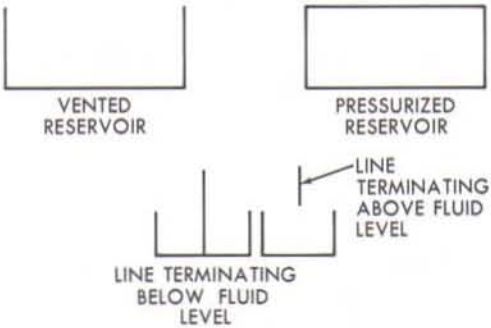


Figure 148

interpret A.S.A. or J.I.C. symbols too. A.N.S.I. symbols, as they apply to mobile hydraulic equipment, are tabulated for reference later in this chapter.

USING A.N.S.I. GRAPHICAL SYMBOLS

THE RESERVOIR

A rectangle (Figure 148), with the long side horizontal, is the symbol for a reservoir. It is open at the top if the reservoir is vented to atmosphere. If the reservoir is pressurized, the top is closed.

Lines connected to the reservoir usually are drawn from the top, regardless of where the actual connection is. If the line terminates below the fluid level, it is drawn all the way to the bottom of the symbol.

A line connected to the bottom of the reservoir may be drawn from the bottom of the symbol . . . if the bottom connection is essential to the system's operation. For instance, when the pump inlet must be charged or flooded by a positive head of oil above the inlet port, we would position the reservoir symbol above the pump symbol and draw the suction line out the bottom of the symbol.

Every reservoir has at least two hydraulic lines connected to it; some have many more. And often the components that are connected to the reservoir are spread all over the diagram, making it inconvenient to draw all the return or drain lines to one symbol. It is customary then to draw individual reservoir symbols wherever convenience indicates it. The reservoir, though, is usually the only component pictured more than once.

LINES ARE LINES

A hydraulic pipe, tube, hose or other conductor that carries the liquid between components is drawn as a single line (Figure 149). A working line (inlet, pressure or return) is drawn as a solid line. Pilot or control lines are broken into long dashes; drain lines for leakage oil are broken into short dashes. A flexible line is drawn as an arc between two dots and is always represented by a solid line.

Crossing or Joining?

The shortest distance between two components that are connected is a straight line . . . and it is desirable to draw it that way to avoid following a line all over the diagram just to get back near where you started. So we do cross lines that aren't connected to each other when it is necessary.

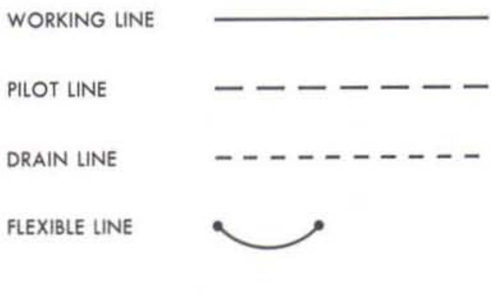


Figure 149

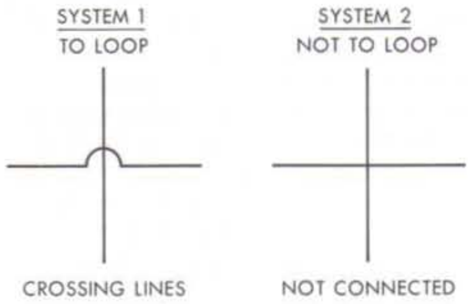


Figure 150

To show that two crossed lines are not connected, we put a short loop (Figure 150) in one of the lines at the intersection. However, some people simply let the lines cross.

A connection between two crossing lines (Figure 151) must be designated by placing a dot at the crossing, if loops are used to designate crossings. The dot is omitted if no loops are used for crossings, but all joining lines must be shown as tees. Cross connections are not permitted in this no dot system. Only one system or the other shall be used throughout a diagram.

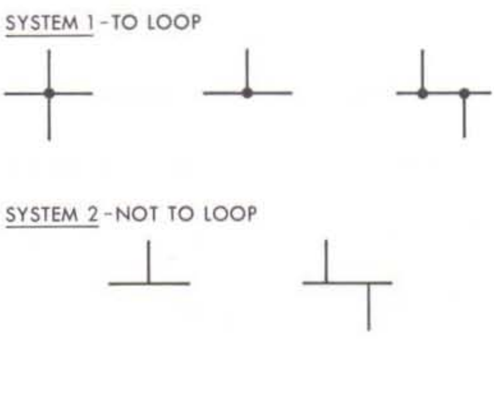


Figure 151

NOTE

For maximum clarity of circuits, the "loop" and "dot" system is recommended.

PUMP SYMBOLS

Would you believe that pump symbols are even easier than reservoir symbols? The basic symbol is a circle with a black triangle pointing outward (Figure 152).

There are probably a score or more basic designs of pumps, but they all have the same function, and one basic symbol is all we need to depict that function.

The black triangle will be used with many symbols to indicate that they are either receivers or sources of energy. It points out from a source; in to a receiver.

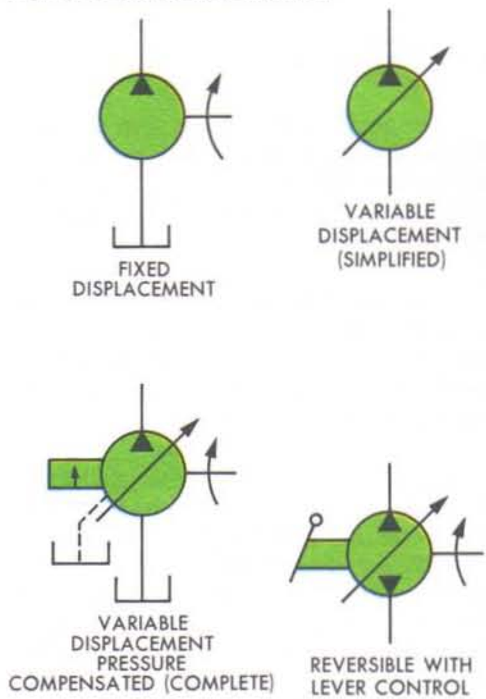


Figure 152

The pressure line from the pump is drawn from the tip of the triangle; the inlet line is drawn opposite it. Thus, the triangle also indicates the direction of flow. If a pump is reversible, it will have two triangles . . . one pointing out of each port.

Port connections to the pump (or any other component with the exception of the reservoir) are at the points where the lines touch the symbols.

A variable (or adjustable) component is designated by drawing an arrow through it at 45 degrees.

A line with short dashes extending from the pump symbol to the reservoir indicates that leakage oil within the pump is drained externally.

Optional Symbols

Occasionally it may be desirable to show the prime mover and the direction of rotation (Figure 153). If the prime mover is an electric motor, it appears as a circle with an "M" in the center. A heat engine (gasoline or diesel) is shown as two squares; one inside the other. A curved arrow crossing a line from the pump symbol indicates the direction of shaft rotation where required.

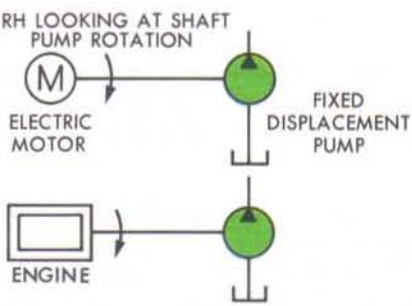


Figure 153

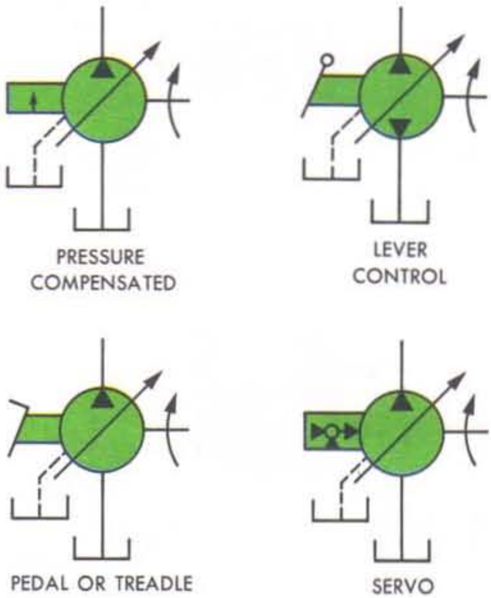


Figure 154

Displacement Controls

A displacement control for a pump (or motor) is drawn beside the symbol (Figure 154). As you can see, the control symbol sometimes has a resemblance to the control; for instance, the lever has a knob.

The pressure compensator symbol is a small arrow parallel to short side of symbol. This symbol is used with any pressure compensated component, and may adjoin the symbol or be placed right on it.

MOTOR SYMBOLS

Motor symbols also are circles with black triangles (Figure 155); but the triangles point in to show that the motor is a receiver of pressure energy. One triangle is used in a non-reversible motor symbol; two are used for a reversible motor.

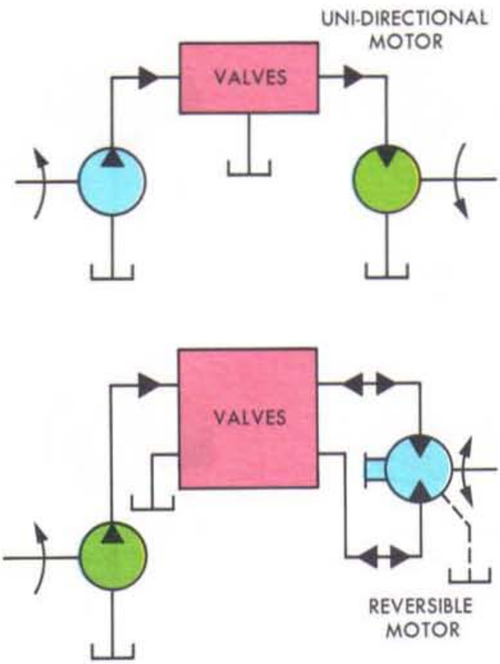


Figure 155

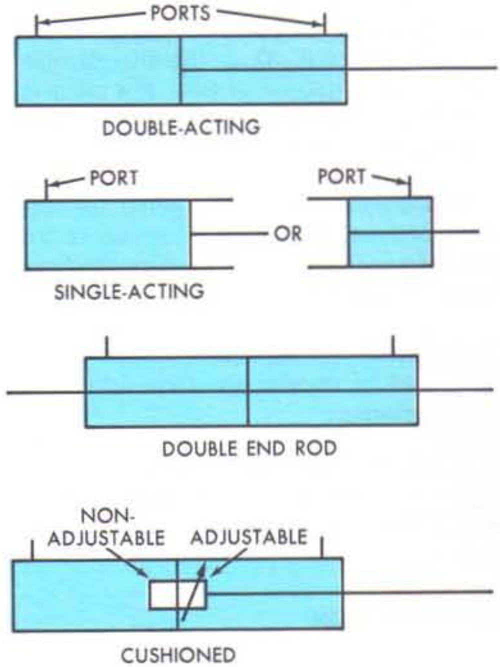


Figure 156

The direction of flow is easily evident with a single triangle . . . it is the way the triangle points. In the reversible motor, we must refer to the pump and valve symbols to trace the flow direction. The arrows outside the lines show the flow direction . . . always away from the pump's pressure port and into the motor port that is connected to the pressure line. The opposite port then must be discharging back to the tank.

Control symbols and rotation direction indicators used with pump symbols also apply to motors.

CYLINDER SYMBOLS

A cylinder symbol (Figure 156) is a simple rectangle representing the barrel with a T-shaped representation of a piston and rod. The symbol can be drawn in any position.

If the cylinder is single-acting, there is only one hydraulic line drawn to the symbol. Also, the end of the symbol opposite the port is left open.

A double-acting cylinder symbol has both ends closed and has two lines meeting the symbol at the port connections. A double end rod cylinder has a "rod" line extending from each end.

Cylinder cushions are drawn as smaller rectangles against the piston line. If the cushion has an adjustable orifice, the slanted arrow is drawn across the symbol.

Flow to and from a cylinder must be traced by observing which lines it is connected to. There is no provision in the symbol for flow direction. This is really not a problem, though. We're about to see that valve symbols are copiously decorated with arrows indicating the direction of flow.

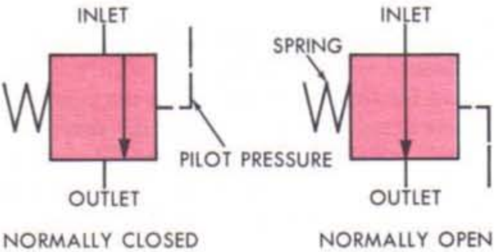


Figure 157

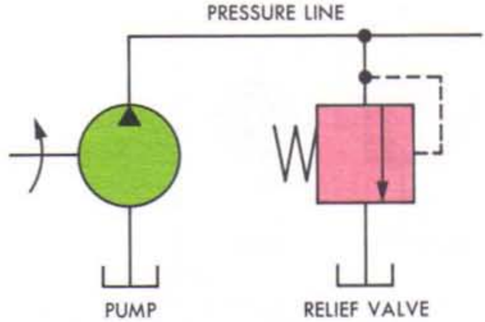


Figure 158

PRESSURE CONTROL SYMBOLS

A pressure control valve, you'll recall, is infinitely positioned between two flow conditions. Its basic symbol is a square (Figure 157) with external port connections and an arrow inside to show the direction of flow. Usually this type valve operates by balancing pressure against a spring, so we show a spring at one side of the symbol and a pilot pressure line at the other.

Normally Closed

A normally-closed valve, such as a relief or sequence valve, is shown with the arrow offset from the ports toward the pilot pressure line. This indicates that the spring holds the valve closed until it is overcome by pressure. We mentally visualize the arrow moving over to complete the flow path from inlet to outlet when pressure rises to the valve setting.

The actual function of the valve is shown by its connection into the circuit diagram.

Normally Open

When the arrow connects the two ports, we know that the valve is normally open. It closes only when pressure overcomes the spring force.

Relief Valve

We diagram a relief valve (Figure 158) with a normally-closed symbol connected between the pressure line and tank. The flow direction arrow points away from the pressure line port and toward the tank port. This shows very graphically how a relief valve operates. When pressure in the system overcomes the valve spring, flow is from the pressure port to the tank port.

We don't attempt to show whether this is a simple or compound relief valve. All that's important is to show its function in the circuit.

Sequence Valve

The same symbol is used for a sequence valve (Figure 159). This time, though, the inlet port is connected to a primary cylinder line; the outlet port to the secondary cylinder line. Pilot pressure from the primary cylinder line sequences the flow to the outlet port when it reaches the setting of the valve.

Since the sequence valve is externally drained, we have added a drain connection to the symbol at the drain's location in the valve.

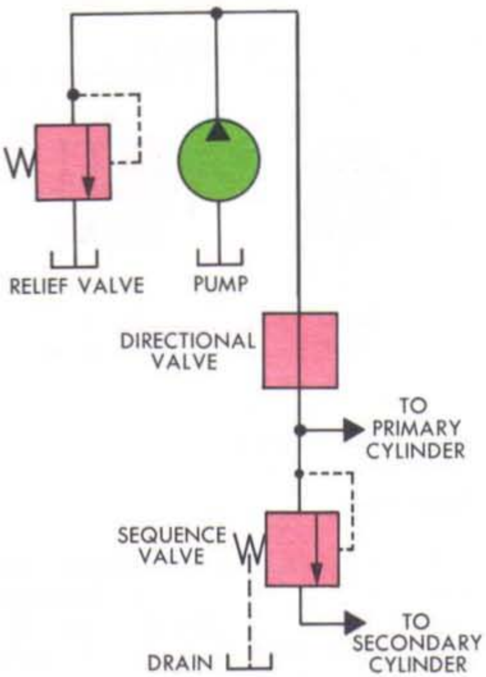


Figure 159

Sequence and Check Valve

Remember that with this connection a sequence valve must be used with a check valve for free return flow when the cylinders are reversed. Figure 160 shows the simplified check valve symbol and its parallel connection. As you are looking at it, free flow is to the up . . . away from the "V" which represents a seat.

In the top view, we see the check valve as a separate unit. When the check valve is built into the sequence valve, we enclose both valves with a box called an enclosure.

An enclosure is used to show the limits of a component or an assembly containing more than one component. It is an alternate long-and-short dash line. External ports are assumed to be on the enclosure line and indicate connections to components.

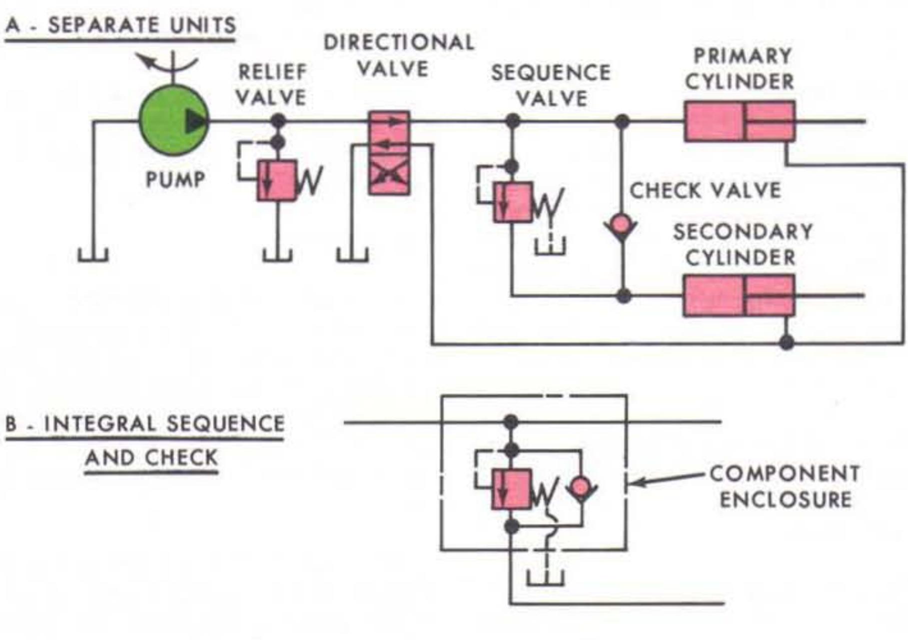


Figure 160

Counterbalance Valve

A counterbalance valve is a normally closed pressure control with an integral check valve. For a directly controlled valve, we use the same symbol (Figure 161) with the primary port connected to the bottom port of the cylinder and the secondary port to the directional valve. The drain connection isn't shown, because the valve is internally drained. If valve body has two primary ports, a complete symbol should show one of them plugged.

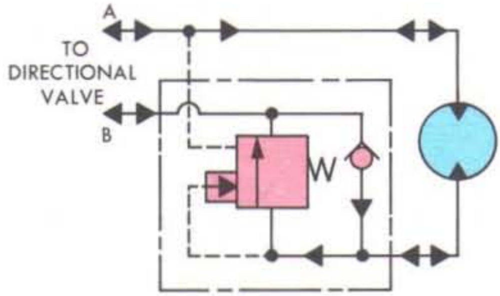


Figure 162

Relief (Brake) Valve

A relief valve with auxiliary remote control connection can be used as a brake valve when connected between the motor outlet and the directional valve (Figure 162). It looks just like the counterbalance valve diagram, except that it has two pilot control connections. A low pressure in line "A" will open the valve to permit free flow from the motor through the valve to "B", but higher braking pressure will be required from the motor to open the valve internally if driving pressure "A" is removed.

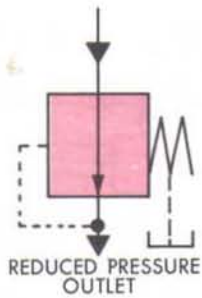


Figure 163

Pressure Reducing Valve

The normally-open pressure reducing valve is diagrammed in Figure 163. Outlet pressure is shown opposing the spring to modulate or shut off flow when the valve setting is reached.

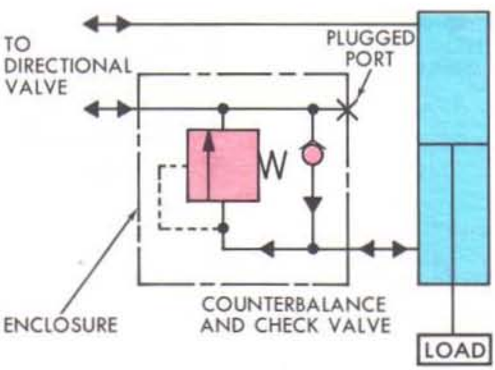


Figure 161

FLOW CONTROL SYMBOLS

The basic flow control valve symbol (Figure 164) is a simple representation of a restriction. If the valve is adjustable, the slanted arrow is drawn across the symbol.

A complete by-pass type pressure compensated flow control with built in relief valve operation is diagrammed in Figure 165. This then is the symbol for the "FM" Series valve. One limitation of the bypass type flow control is that it can only be used to meter fluid into an actuator.

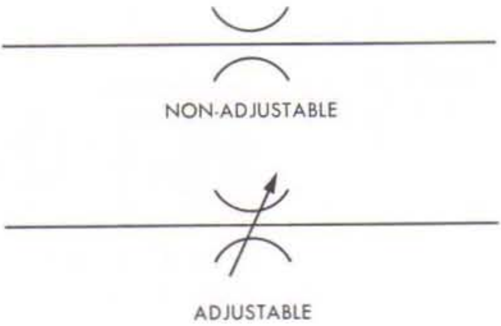


Figure 164

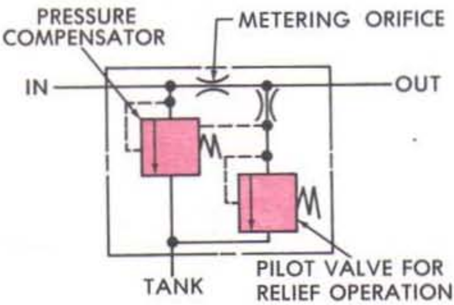


Figure 165

Restrictor type flow controls can be applied to meter-out, meter-in and bleed-off circuits and are shown in Figures 166, 167 and 168 respectively. The perpendicular arrow indicates pressure compensation.

DIRECTIONAL CONTROL SYMBOLS

A directional control valve symbol uses a multiple envelope system that has a separate rectangle for each position. All the port connections are made to the envelope that shows the neutral condition of the valve. Within each envelope are arrows showing the flow paths when the valve is shifted to that position.

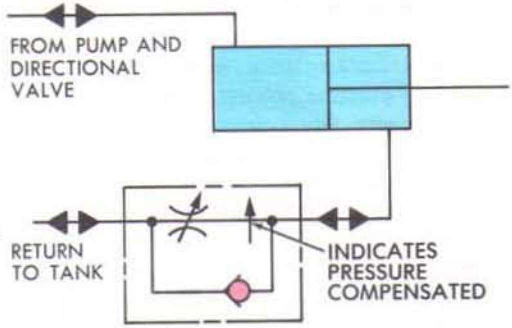


Figure 166

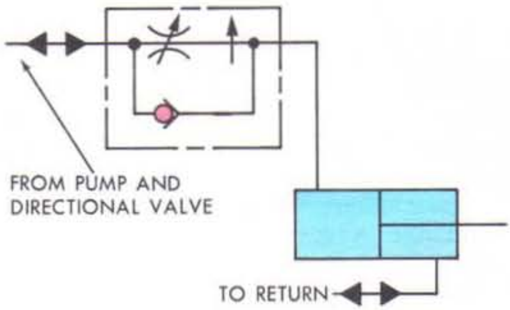


Figure 167

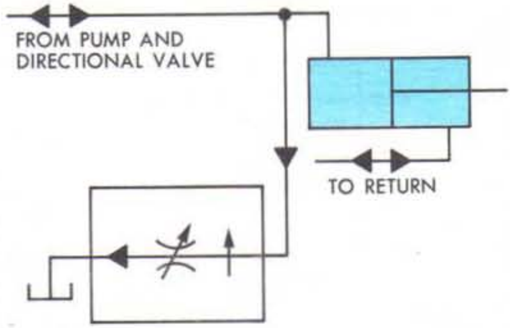


Figure 168

One Way Valve

You have already seen the simplified symbol for a check valve. Compare it with the composite symbol (Figure 169) and decide for yourself which will get the most use. However, the multiple envelope system does provide a simple way of showing function when applied valve has several flow paths.

Unloading Valve

An unloading valve symbol (Figure 170) is shown with one envelope. In the normal closed position, flow is shown blocked inside the valve. The arrow is displaced toward the opposite side of the envelope to show that the spring controls this position.

External pilot pressure is indicated against the bottom envelope to show that this is the flow condition when the pilot pressure takes over. When pressure is great enough to overcome spring force, the flow path arrow connects the pump outlet to the reservoir.

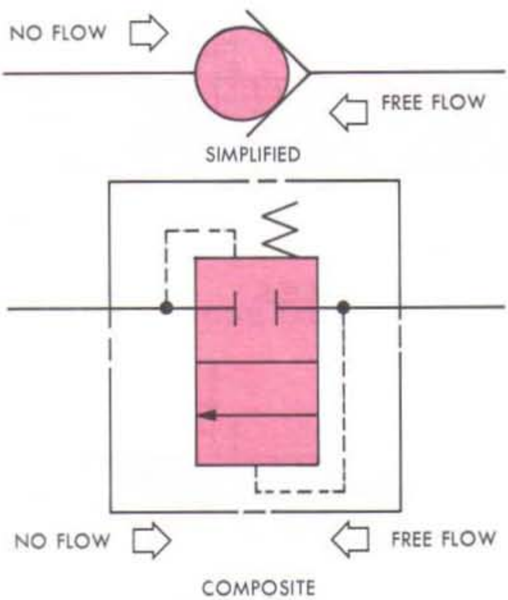


Figure 169

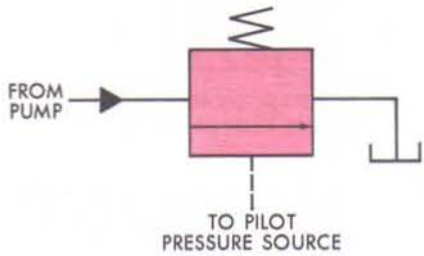


Figure 170

Four-Way Valve

An ordinary four-way valve has two envelopes if it is a two-position valve (Figure 171) or three envelopes if it has a center position. The actuating control symbols are placed at ends of the envelopes. The extreme envelopes show the flow conditions when their adjacent controls are actuated.

The manual, lever, pedal, and mechanical control symbols are used as appropriate with directional valves. Spring symbols, pilot lines, solenoid symbols and internal-pilot black triangles also are used as appropriate.

Mobile Directional Valves

The symbol for a mobile directional valve section (Figure 172) resembles a four-way valve symbol, but it has added connections and flow paths to represent the by-pass passage, There is a separate envelope for each finite position and connections are shown to the center or neutral position. A manual lever control with centering springs is shown at each end in View B.

Complete symbols for B, C, and T spools are shown in Figure 172, views B, C, and D

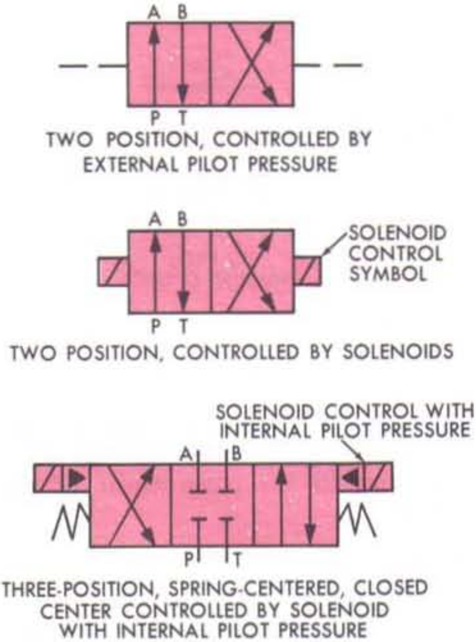


Figure 171

respectively. These illustrations show only the spools. A complete mobile valve bank would also show relief valves and internal connections within an enclosure.

ACCESSORIES

Fluid conditioners are represented as squares (Figure 173) that are turned 45 degrees and have the port connections to the corners. A dotted line at right angles to port connections tells us the conditioner is a filter or strainer. A cooler symbol has a solid line at right angle to fluid line with energy triangles (indicating heat this time) pointing out.

An accumulator (Figure 174) appears as an oval and may have added inside details to indicate spring load, gas charge, or other features. The divider line indicates that there is a separator between the gas or spring and the oil.

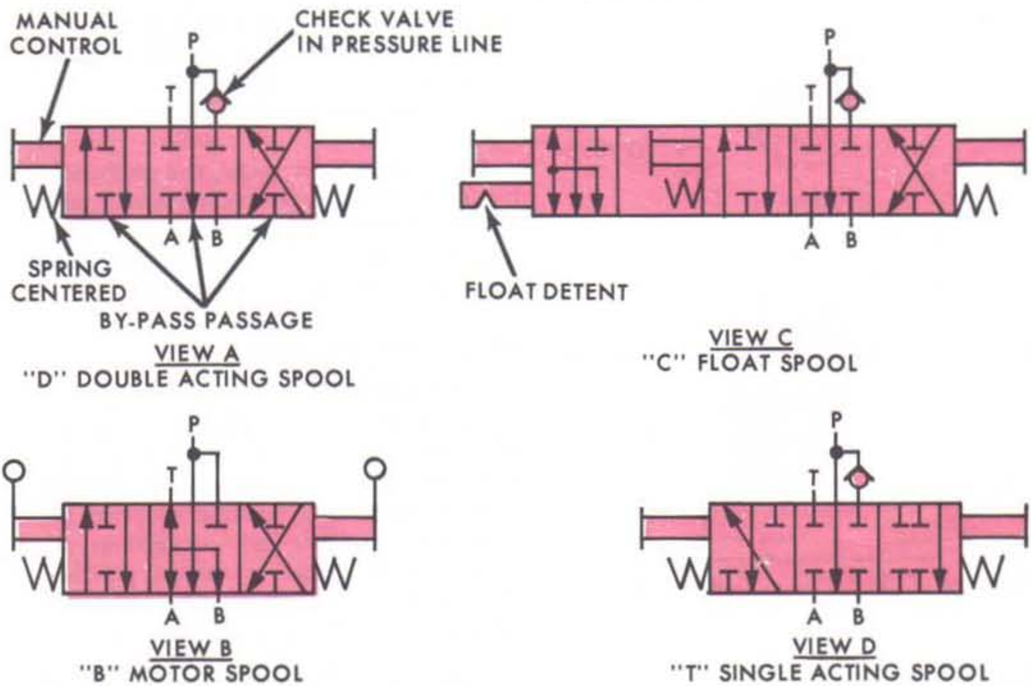


Figure 172

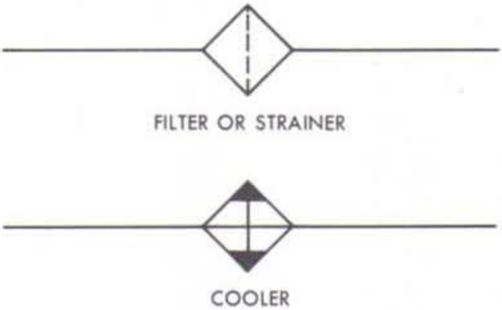


Figure 173

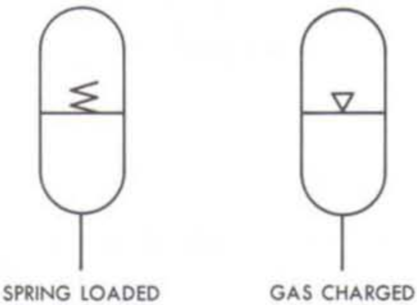


Figure 174

SYMBOLS TABULATED

In these pages, we have reviewed the major graphical symbols and how they are used. We couldn't attempt to cover every possible symbol and combination; that would take several books the size of this one.

For your reference, all the basic hydraulic symbols are tabulated at the end of this chapter. The new A.N.S.I. symbols are used throughout the complete manual.

TYPICAL MOBILE CIRCUITS

Now, having seen the way basic symbols of lines and components are drawn, we can see how they are put together in typical mobile circuits. We'll begin with the lift truck circuit that you saw pictorially earlier in this chapter.

LIFT TRUCK CIRCUIT

The circuit shown in Figure 175 is the lift portion of the hydraulic system shown in Figure 147. The steering portion is covered in Chapter 7.

In this circuit we have two cylinders . . . a single-acting lift cylinder and a double-acting tilt cylinder. The lift cylinder, of course, moves the lifting fork up and down. The tilt cylinder tilts the mast back and forth to support or dump the load.

A two-section multiple unit directional valve controls the cylinder's operation. The first valve, used for the tilt cylinder, has a double-acting "D" spool. It operates the tilt cylinder hydraulically in either direction. The outer envelopes show the typical four flow paths for reversing the cylinder. The second valve has a single-acting "T" spool to operate the lift cylinder. This cylinder is returned by gravity; hence the valve symbol shows only one open flow path in each shifted position and the by-pass unloads the pump.

The pump is driven by the lift truck engine and supplies the lift circuit from the large volume end. Notice we have placed an enclosure around the two pump symbols to show that both pumping units are contained in a single assembly. Similarly, the two directional valves and the relief valve built into the inlet section are surrounded by an enclosure. So we know that these valves all are in a single assembly.

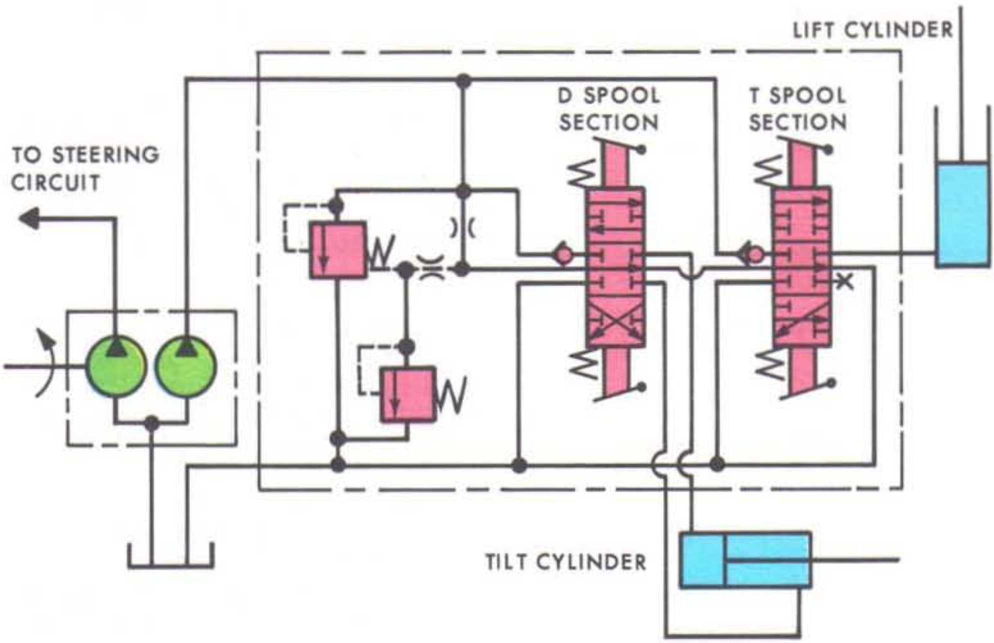


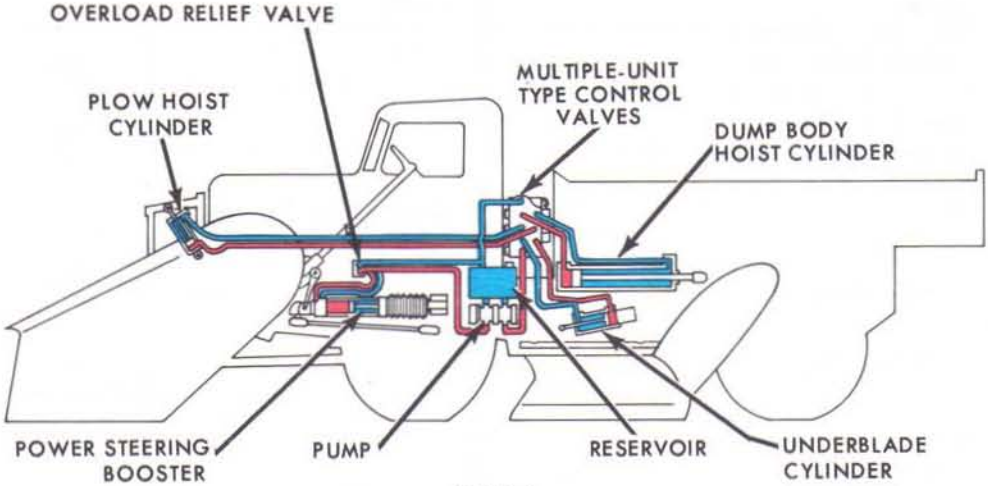
Figure 175

As is common practice, we have shown the circuit in neutral; that is, with the valves centered. For operating conditions, we imagine the outer envelopes on the valve symbols to shift over to align with ports at the center envelopes. The arrows in the envelopes then show the flow paths from the pressure inlet to the cylinders and/or the return flow to tank when the cylinders are in operation.

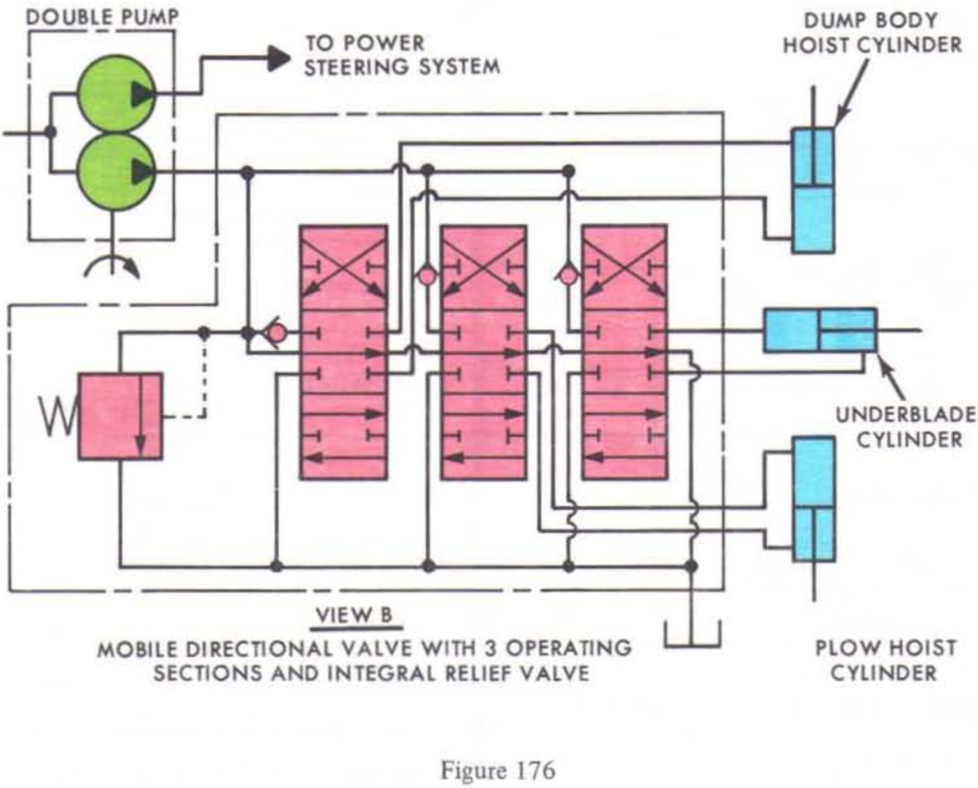
ROAD PATROL TRUCK HYDRAULIC SYSTEM

The simplicity and versatility of hydraulics are evident in examining another hydraulic circuit (Figure 176). A typical road patrol truck requires three double-acting cylinders to operate its blades and dump body. They are a plow hoist cylinder for the front plow, an underblade cylinder and a dump body hoist cylinder. Notice in the pictorial diagram (View A) that this truck also has a power steering system operated from the other half of the double pump. In the schematic (View B), we have omitted the steering system.

The three cylinders are operated through a three-spool mobile directional valve fed from the large volume end of the double pump. Comparing this schematic with Figure 175, you can see many similarities. In fact, the only apparent differences are that we have an additional cylinder and valve section and all our valves are double-acting. Though all the components of the two circuits are probably of different sizes, their functions are nearly identical.



VIEW A
MOTOR TRUCK HYDRAULIC SYSTEM

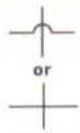
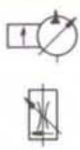
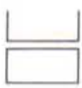


VIEW B
MOBILE DIRECTIONAL VALVE WITH 3 OPERATING SECTIONS AND INTEGRAL RELIEF VALVE

Figure 176

A.N.S.I. SYMBOLS FOR SPERRY VICKERS EQUIPMENT

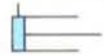
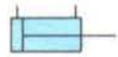
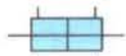
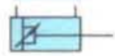
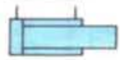
Lines

LINE, WORKING (MAIN)	
LINE, PILOT (FOR CONTROL)	
LINE, LIQUID DRAIN	
FLOW, DIRECTION OF — HYDRAULIC PNEUMATIC	
LINES CROSSING	
LINES JOINING	
LINE WITH FIXED RESTRICTION	
LINE, FLEXIBLE	
STATION, TESTING, MEASUREMENT OR POWER TAKE-OFF	
VARIABLE COMPONENT (RUN ARROW THROUGH SYMBOL AT 45°)	
PRESSURE COMPENSATED UNITS (ARROW PARALLEL TO SHORT SIDE OF SYMBOL)	
TEMPERATURE CAUSE OR EFFECT	
RESERVOIR — VENTED PRESSURIZED	
LINE, TO RESERVOIR ABOVE FLUID LEVEL BELOW FLUID LEVEL	 
VENTED MANIFOLD	

Pumps

HYDRAULIC PUMP FIXED DISPLACEMENT	
VARIABLE DISPLACEMENT	

Motors and Cylinders

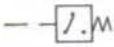
HYDRAULIC MOTOR FIXED DISPLACEMENT	
VARIABLE DISPLACEMENT	
CYLINDER, SINGLE ACTING	
CYLINDER, DOUBLE ACTING SINGLE END ROD	
DOUBLE END ROD	
ADJUSTABLE CUSHION ADVANCE ONLY	
DIFFERENTIAL PISTON	

Miscellaneous Units

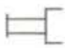
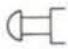
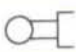
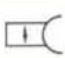
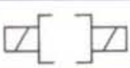
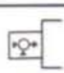
ELECTRIC MOTOR	
ACCUMULATOR, SPRING LOADED	
ACCUMULATOR, GAS CHARGED	
HEATER	
COOLER	
TEMPERATURE CONTROLLER	

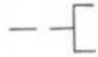
A.N.S.I. SYMBOLS FOR SPERRY VICKERS EQUIPMENT

Miscellaneous Units (cont.)

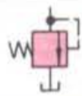
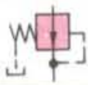
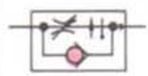
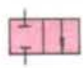
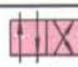
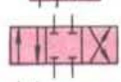
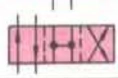
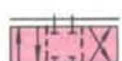
FILTER, STRAINER	
PRESSURE SWITCH	
PRESSURE INDICATOR	
TEMPERATURE INDICATOR	
COMPONENT ENCLOSURE	
DIRECTION OF SHAFT ROTATION (ASSUME ARROW ON NEAR SIDE OF SHAFT)	

Methods of Operation

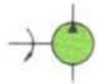
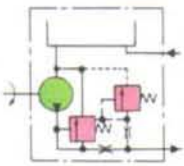
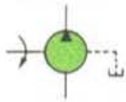
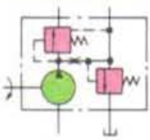
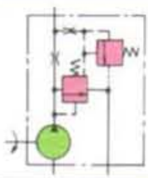
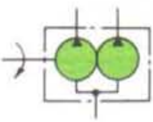
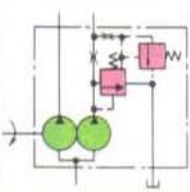
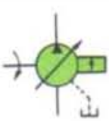
SPRING	
MANUAL	
PUSH BUTTON	
PUSH-PULL LEVER	
PEDAL OR TREADLE	
MECHANICAL	
DETENT	
PRESSURE COMPENSATED	
SOLENOID, SINGLE WINDING	
SERVO MOTOR	

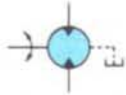
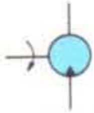
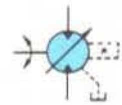
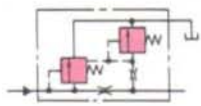
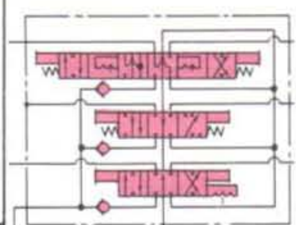
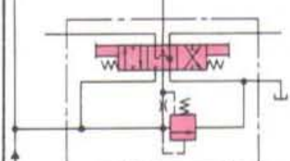
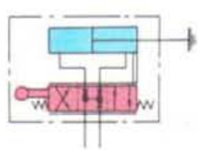
PILOT PRESSURE REMOTE SUPPLY	
INTERNAL SUPPLY	

Valves

CHECK	
ON-OFF (MANUAL SHUT-OFF)	
PRESSURE RELIEF	
PRESSURE REDUCING	
FLOW CONTROL, ADJUSTABLE—NON-COMPENSATED	
FLOW CONTROL, ADJUSTABLE (TEMPERATURE AND PRESSURE COMPENSATED)	
TWO POSITION TWO CONNECTION	
TWO POSITION THREE CONNECTION	
TWO POSITION FOUR CONNECTION THREE POSITION FOUR CONNECTION TWO POSITION IN TRANSITION	  
VALVES CAPABLE OF INFINITE POSITIONING (HORIZONTAL BARS INDICATE INFINITE POSITIONING ABILITY)	

A.N.S.I. SYMBOLS FOR SPERRY VICKERS EQUIPMENT

SINGLE, vane & gear type pump TYPICAL SERIES V100, 200 V10, V20 25VQ, 35VQ, 45VQ, 50VQ G20	
SINGLE, Power Steering Pump, with integral flow control and relief valves TYPICAL SERIES VTM27-**-**-**-07-R*-12 VTM28-**-**-**-07-R*-12 VTM42-**-**-**-11-R*-12	
SINGLE, piston type pump with drain TYPICAL SERIES M-PFB5, 10, 15, 20, 29, 45	
SINGLE Pump, with integral priority valve TYPICAL SERIES V20P	
SINGLE PUMP, with integral flow control valve TYPICAL SERIES V20 F	
DOUBLE PUMP, TYPICAL SERIES V2010, V2020 V2200 252*VQ, 352*VQ, 452*VQ	
DOUBLE PUMP, with integral flow control TYPICAL SERIES V2020F V2200, 252*VQ, 352*VQ, 452*VQ	
PUMP WITH PRESSURE COMPENSATOR CONTROL TYPICAL SERIES M-PVB5, 6, 10, 15, 20, 29, 45, 90	

FIXED DISPLACEMENT MOTOR, DUAL DIRECTIONAL TYPICAL SERIES M2-200 25M, 35M, 45M, 50M M-MFB-5, 6, 10, 15, 20, 29, 45	
FIXED DISPLACEMENT MOTOR, UNI-DIRECTIONAL TYPICAL SERIES M2U, M3U	
VARIABLE DISPLACEMENT MOTOR, DUAL DIRECTIONAL TYPICAL SERIES M-MVB5, 10	
FLOW CONTROL AND OVERLOAD RELIEF VALVE (Non-adjustable) TYPICAL SERIES FM3	
MULTIPLE UNIT VALVE CONSTRUCTION TYPICAL SERIES CM*NO*-FDITCL	
MULTIPLE UNIT VALVE CONSTRUCTION TYPICAL SERIES CM*NO*R**BE	
STEERING BOOSTER TYPICAL SERIES S20	

CHAPTER 7
Power Steering

Power steering is as old as the horse-drawn carriage or buggy with its swinging front axle. The early stagecoach driver didn't steer his coach by his own muscle power. Instead, a light tug on the reins was transmitted to a mammoth source of power . . . a team of horses. The horses steered the vehicle in response to the driver's signal. And that is the principle of today's hydraulic power steering. The driver, with a slight effort, controls the "horses" in his engine to obtain a greatly reduced steering effort.

When dobbin was first put out to pasture in favor of the engine, power steering went with him. Early automobiles and trucks were steered by human muscle power through a tiller handle . . . and the axles had to swing to the same angle as the tillers. But the vehicles were light, tires were small in cross-section and road speeds that seemed foolhardy then were really pretty slow. So the steering effort was easily within the limits of physical ability.

STEERING GEARS

As automotive vehicles developed they became heavier, and consequently harder to steer. A fellow named Dunlop, in an effort to make it easier for his son to win bicycle races, invented the pneumatic tire which soon after became standard equipment on cars and trucks. While greatly improving the ride, the rubber tire was not at all conducive to steering ease . . . particularly as it matched the growth in size of the vehicle and presented ever-increasing contact areas to the road surface. And so a simple lever from the tiller or steering wheel to the steered wheels was no longer good enough. More force had to be made available for steering.

The direct steering linkage was first replaced with a steering gear. A steering gear is simply an arrangement of gears in a box that multiplies the input torque from the steering wheel to a much greater torque at the steering shaft (Figure 177). The steering shaft, through the Pitman arm (or steering shaft arm), transmits this increased torque through steering linkage to the steering arms which turn the wheels.

There are many designs of manual steering gears, and many arrangements of steering linkage. But the basic system of a steering wheel, steering gear and linkage to the steered wheels is almost universal . . . even with power-steered vehicles.

HIGH STEERING RATIOS

The disadvantage of the steering gear-linkage system is that we always lose distance

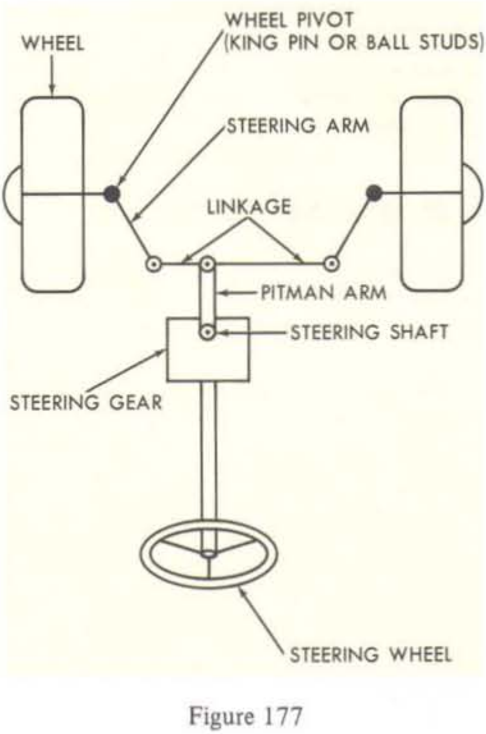


Figure 177

when we multiply force. So if, for instance, we want four times as much steering torque at the road wheel as at the steering wheel, then the steering wheel must turn four times as far. Putting it another way, we must turn the steering wheel four degrees for every one degree of road wheel turn. We express this situation as 4-to-1 steering ratio.

Steering ratios have actually been as high as 40-to-1! The driver had to turn the steering wheel 400 degrees, or more than a full revolution just to turn the front wheels 10 degrees! This is potentially dangerous, because the driver may not be able to turn the steering wheel fast enough to control the vehicle.

In 1925 Harry F. Vickers, founder of what is now the Sperry Vickers Division of Sperry Rand Corp., developed some of the first practical power steering applications for commercial vehicles. Power steering systems have been operated by compressed air and through electrical power. The systems that Vickers developed used oil hydraulic power . . . and almost without exception, today's power steering systems are hydraulic.

POWER STEERING ADVANTAGES

Power steering has many benefits to the vehicle operator; and in the case of a commercial vehicle, to the owner.

Steering ratios can be greatly reduced by power steering, so that the driver has the best possible control of his vehicle. Steering effort is at a minimum . . . the days when steering a heavy truck required a 100-pound (445 N) pull at the steering wheel are hopefully gone forever. Reduced driver fatigue results in longer runs and safer operation. And, load carrying ability of trucks is greatly increased, because now the steering axle can be loaded as well as the other axles.

Today, we find power steering almost universal on full-size passenger cars. The trucking industry, though slower to take advantage of power steering, is being influenced by passenger car sales. Earth-moving, construction machinery and materials handling equipment manufacturers have adopted power steering on most of their vehicles . . . as have the manufacturers of motor coaches.

WHAT IS POWER STEERING?

Hydraulic power steering is essentially the incorporation of a hydraulic assist into a basic manual steering system.

POWER BOOSTER

The hydraulic boost may be applied to the steering linkage (Figure 178) or within the steering gear itself. It is basically a mechanically-operated hydraulic servo. Movement of the steering wheel actuates the steering valve, which directs fluid under pressure to actuate a piston. The piston is mechanically connected to the steering linkage and provides the power boost. Movement of the linkage is transmitted to the steering valve body, which "follows" the valve spool. Hydraulic boost is applied then only when the steering wheel is being moved.

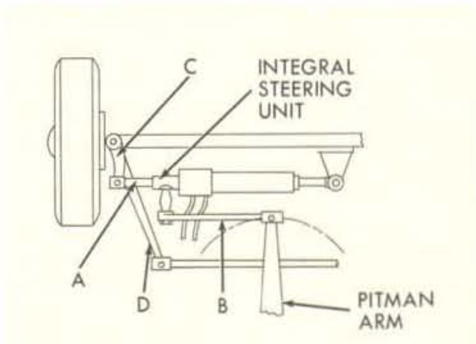


Figure 178

In the case of a hydraulic system failure, the steering reverts to manual operation.

FULL OR PART-TIME POWER STEERING

A few years ago, a great deal of advertising "hay" was made over "full-time" power steering . . . and anyone who remembers this will be curious about the distinction between full-time and part-time power.

Most power steering systems can operate either way. With the power booster incorporated into a conventional steering system, the wheels will always be steered by power if the steering valve is actuated. However, if the valve is not actuated, the system steers manually and the hydraulic components just go along for the ride.

Part-Time

Whether the valve is actuated depends on the steering effort required and the stiffness of the valve's centering springs. Suppose that the valve has fairly heavy centering springs. With a light steering load, such as a gentle, banked turn at cruising speed, the steering effort would be less than the spring force. Then, we'd just steer manually . . . pushing right through the spring. For parking, though, more effort is required. The spring is compressed, the servo valve spool moves relative to its body, and we have power boost. That is part-time power steering.

With part-time power steering, the force of the centering springs gives the driver the "feel" of the road at the steering wheel.

Full-Time

In a full-time power steering system, the valve is installed without centering springs. Thus any movement of the steering wheel results in hydraulic boost being applied.

If road feel is required in a full-time system, it must be built in through a hydraulic "reaction" device which, in effect, resists the turning of the steering wheel in proportion to effort at the wheels.

We will discuss power steering hydraulic circuits after a look at several of the most common types of systems.

POWER STEERING SYSTEMS

INTEGRAL POWER STEERING GEAR

An integral power steering gear system (Figure 179) has the hydraulic boost system built into the mechanical steering gear. The steering valve is actuated by movement of the steering shaft, and controls the operation of the power cylinder. Thrust from the power cylinder is transmitted directly to the steering shaft. Road shock transmitted back from the wheels is taken up in the steering gear.

This design is quite suitable for passenger cars and light trucks. It has the advantage of very simplified plumbing. The only external lines required are a pressure line from the pump and a return line to the reservoir.

This gear is not suitable for large vehicles, because it would have to be too bulky. It also is more difficult to service. A passenger car can be tied up all day with little inconvenience but serviceability is essential in a truck system where every hour out of service costs the owner money.

SEMI-INTEGRAL POWER STEERING GEAR

The semi-integral power steering gear system (Figure 180) is sometimes called a valve-on-gear system. The steering valve is

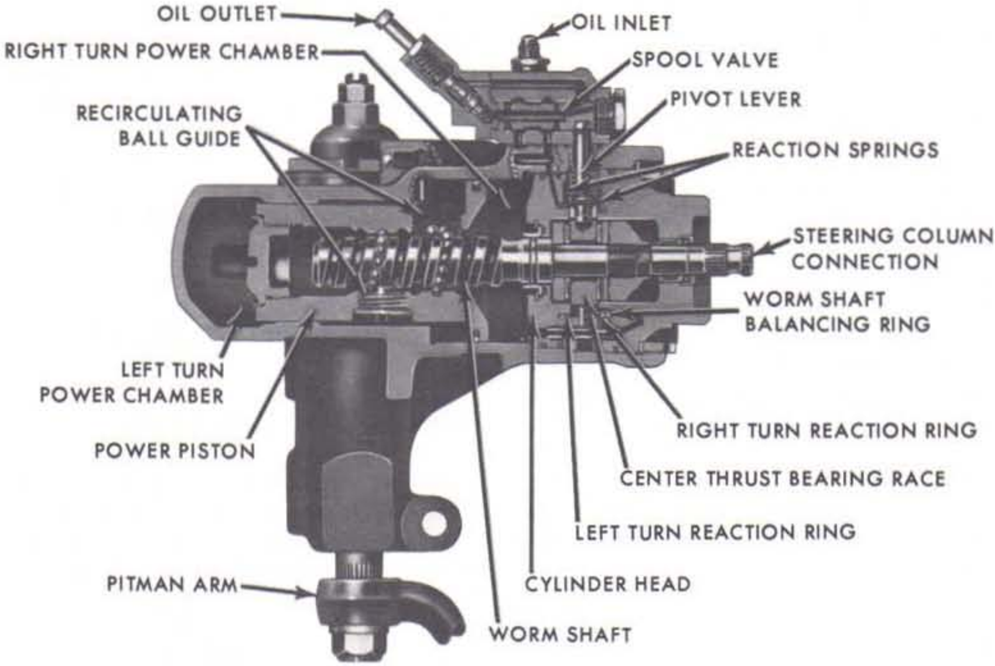


Figure 179

Courtesy Ross Gear Div., TRW

Figure 180

built into the steering gear. The power cylinder is attached to the vehicle frame and to the linkage. Road shock and thrust are absorbed in the frame.

This type of steering gear is light and easily adaptable to many different vehicles, but the plumbing gets a little complicated. An extra set of hoses is needed between the steering gear and the power cylinder. Also, steering valve repairs require removing and dismantling the steering gear.

INTEGRAL LINKAGE POWER SYSTEM

The integral linkage power steering system (Figure 178) has both the valve and the power cylinder built into an integral power steering unit (or booster) mounted between the linkage and the frame. In this design, the steering valve is actuated through a drag link (B) by the steering gear Pitman arm. The Pitman arm, of course, responds to steering wheel movement. Power cylinder thrust is applied to the linkage at the steering arm (C).

The Ackerman linkage is shown in Figure 178. The booster actuates the steering arm (C) of the left wheel, and the right wheel is steered through a cross steering arm (D) and a tie rod. The booster is anchored at the cylinder rod end, so that the complete assembly moves with the steering linkage. The valve thus is centered as the linkage catches up with Pitman arm movement.

Plumbing is simplified in this system . . . only pressure and tank lines are required to the booster. Other advantages are easy adaptability of the system to various steering linkages, absorption of shock and thrust in the frame rather than in the steering gear, and ease of service.

Where this system isn't adaptable to a vehicle design, a "split" or remote type linkage system may be used.

REMOTE LINKAGE SYSTEM

Figure 181 illustrates a typical linkage system using a remotely mounted valve. The valve is connected between the Pitman arm and a secondary steering arm (C). The vehicle actually could be steered mechanically through this connection. The boost is applied from the power cylinder, which is mounted between the frame and the primary steering arm.

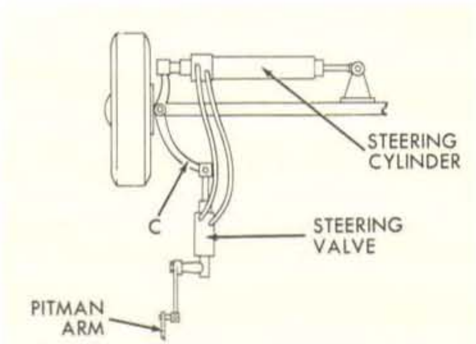


Figure 181

In any remote system, there must be a linkage provided to "feed back" steering movement to the valve. The secondary steering arm (C) provides the feedback in this instance.

COMBINED INTEGRAL-REMOTE LINKAGE SYSTEM

A combination of integral and remote operation of a linkage system might be used on a heavy vehicle where space isn't available for a power cylinder large enough to handle the steering load. Sometimes, too, it is desirable to divide the force, thus avoiding distortion of the linkage.

In Figure 182 we have a swinging "beam" type axle as used on some very large off-highway trucks. An integral steering unit (valve and cylinder combined) is installed at one side and a separate power cylinder at the

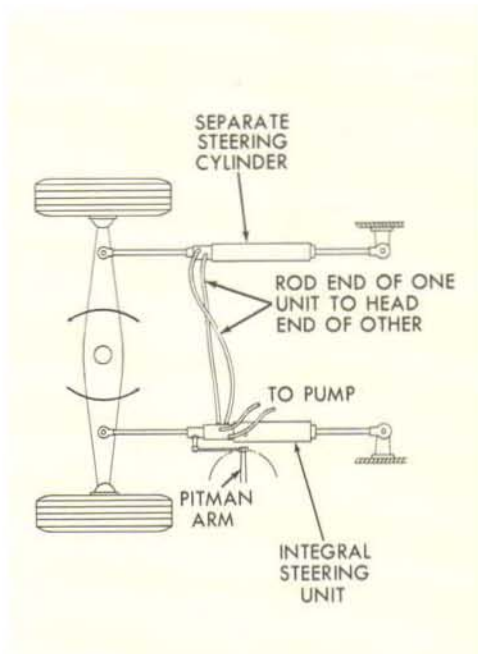


Figure 182

other side. The steering valve in the integral unit has an extra set of ports to control the remote cylinder also. The hydraulic connections are made so that one cylinder extends while the other retracts.

The valve again is actuated by the drag link from the Pitman arm. Feedback to the valve is through movement of the integral steering unit.

REMOTE DUAL SYSTEM

Another system using dual power cylinders is shown in Figure 183. Manually operated, this also would be an Ackerman system, with the drag link from the Pitman arm steering the left wheel and a tie rod connection to the right wheel.

The steering cylinders are anchored between the wheels and operated by a single, remote valve. The valve is connected between the Pitman arm and left-wheel steering arm. The steering arm provides the feedback of linkage movement to the valve body.

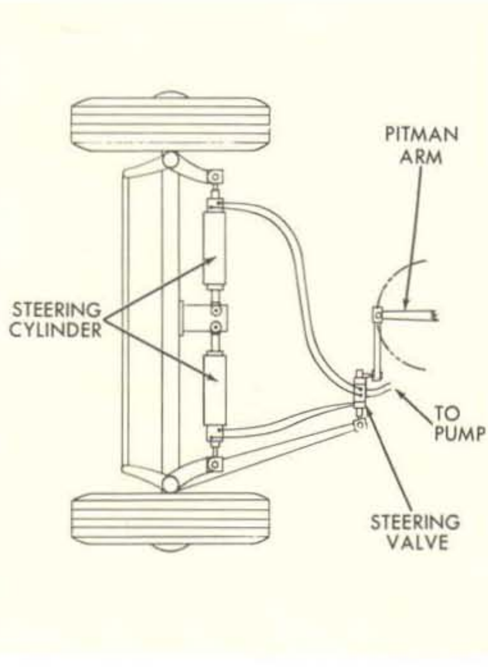


Figure 183

This system is used on many vehicles that are steered by the rear wheels.

As we mentioned before, there are many arrangements of steering linkage; and therefore many arrangements of power steering. We have seen the four basic arrangements of linkage systems. Now we can go on to study the hydraulic circuits.

POWER STEERING CIRCUITS

CIRCUIT COMPONENTS

Reading the descriptions of the kinds of power steering systems gave you some idea of what hydraulic components must go into them. For the record, let's list them all and what each must do before we look at a few circuit variations.

Manual Steering Gear

The manual steering gear transmits motion of the steering wheel to the Pitman arm and provides some reduction of steering wheel movement. We could actually eliminate the steering gear with power steering. A shaft from the steering wheel could actuate the steering valve directly. There are two reasons we don't do this:

First, in the event of a hydraulic system failure, the steering gear ratio reduces the heavy manual steering effort. Second, automotive vehicle drivers just aren't ready for a 1-to-1 ratio steering system. One automobile manufacturer, in fact, was criticized for a low-ratio power steering system until the public became used to it. There is initially a tendency to over-steer with a low ratio gear.

Power Steering Pump

A power steering pump is usually a vane-type pump or some similar construction of positive displacement. It is driven by the vehicle engine, usually through a pulley and V-belt or other type indirect coupling. However, some power steering pumps are designed for direct coupling to the generator.

Special power steering pumps (Chapter 3, Figure 57) are built with integral reservoirs and flow control relief valves required for power steering operation. When the vehicle also has another hydraulic system or systems, it is common to equip it with a double pump (Figure 184). The low volume (cover) end of the pump supplies the steering system; the shaft end cartridge supplies the other operating systems.

Relief Valve

A relief valve is required in the pressure line to protect the pump from overloads. In a Vickers power steering pump, the relief valve is integral with the flow control valve. It can also be built into the steering valve.

Flow Control Valve

The flow control valve maintains a constant rate of flow to operate the power cylinder(s). Without this valve, variations in engine speed would affect the sensitivity of the steering unit by causing variations in pump flow. For safety, it's best to have the unit respond with exactly the same sensitivity at all speeds.

You'll remember from Chapter 5 that the FM series valves are combined flow

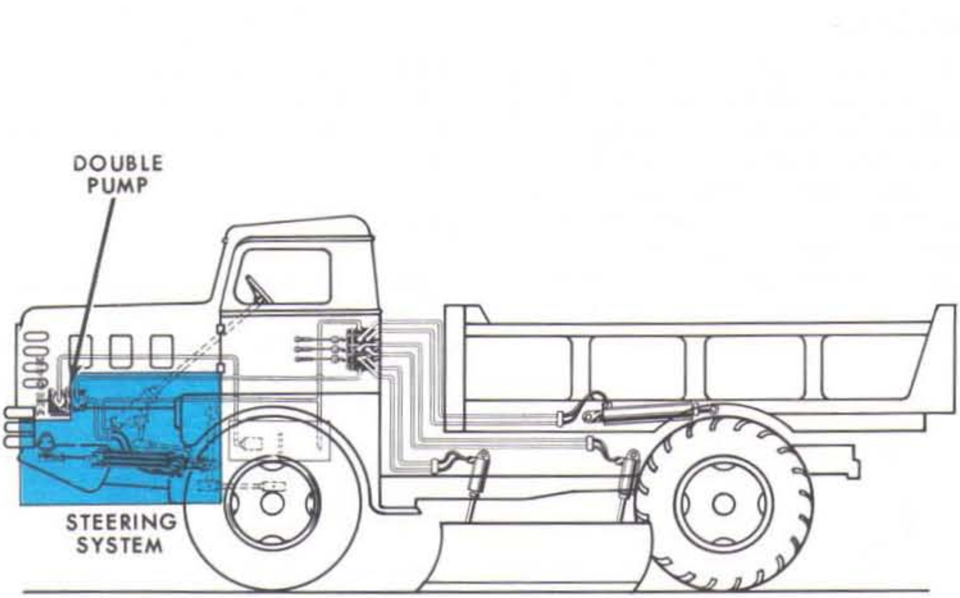


Figure 184

controls and relief valves designed for power steering systems. The flow control and relief valve is built into Vickers power steering pumps, and is optional in other vane pumps that may be used for power steering.

Steering Valve

The steering valve is a four-way valve that functions as a positioning servo valve. It must direct fluid to either end of the power cylinder. Most steering valves are the open-center type. When the valve is in neutral, oil from the pump is recirculated freely through the valve back to the reservoir.

Vickers steering valves are designed for incorporation into integral steering units or for separate mounting in remote linkage installations.

Power Cylinder

The power cylinder is double-acting. It is a differential cylinder, so that steering response may be slightly different in left or right turns with a controlled flow rate. The differential is slight, however, and unnoticeable to most people. When two cylinders are used, oil is pumped simultaneously to the rod end of one and the head end of the other, so the differential is cancelled.

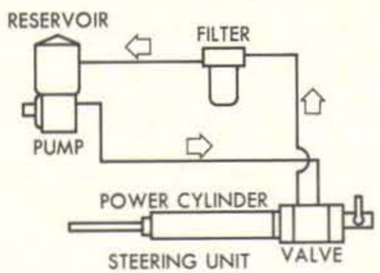


Figure 185

Oil Filter

A 10-micron (nominal) or smaller filter is recommended in power steering systems to prevent damage to the pump and steering valve from metal particles and dirt. The filter is preferably installed in the return line (Figure 185) . . . and should have a by-pass valve to prevent blocking flow if the element is clogged.

Air Breather Filter

Power steering systems operate with vented reservoirs. The "breather" or vent in the reservoir should be equipped with a 3-micron filter element.

Oil Reservoir

The reservoir must hold all the oil required by system during operation, plus a sufficient level to avoid a vortex at the suction line. It should be capable of dissipating heat generated in the steering system. Power steering reservoirs were described in Chapter 2.

Hydraulic Lines

Working hydraulic lines for the most part are flexible hoses, since the steering components move during operation. Long lines may be partly flexible hoses and partly tubing where flexibility isn't required.

INTEGRAL LINKAGE STEERING UNIT CIRCUIT

The hydraulic circuit for an integral unit system can be as simple as shown in the pictorial diagram in Figure 186. Controlled flow originates in the pump and is routed through the pressure line to the steering valve. The valve directs the flow to the power cylinder and returns it through the return line to the tank.

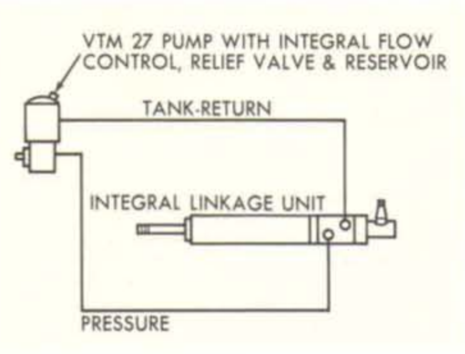


Figure 186

S20 Steering Unit

In Figure 187, we see a cutaway view of a typical S20 series steering unit. It consists of a power cylinder bolted to a steering valve. The rod end (anchor) ball stud mounts the unit to the vehicle frame. The cap end ball stud is connected to the steering linkage (Figure 182). The center or control ball stud is connected to the Pitman arm drag link to actuate the steering valve. A single centering spring is mounted between the ball stud and valve, and is flanked by centering washers. This arrangement gives the driver road feel in both directions.

Oil Flow

Oil flow through the steering unit is shown in Figure 188. Notice that the valve has two external ports and two internal ports. The inlet port is connected to the pressure line from the pump. The outlet port is the tank return. The upper internal port in the diagram connects between the coaxial tubes of the cylinder to the cylinder rod end. The lower internal port connects to the cap end of the cylinder.

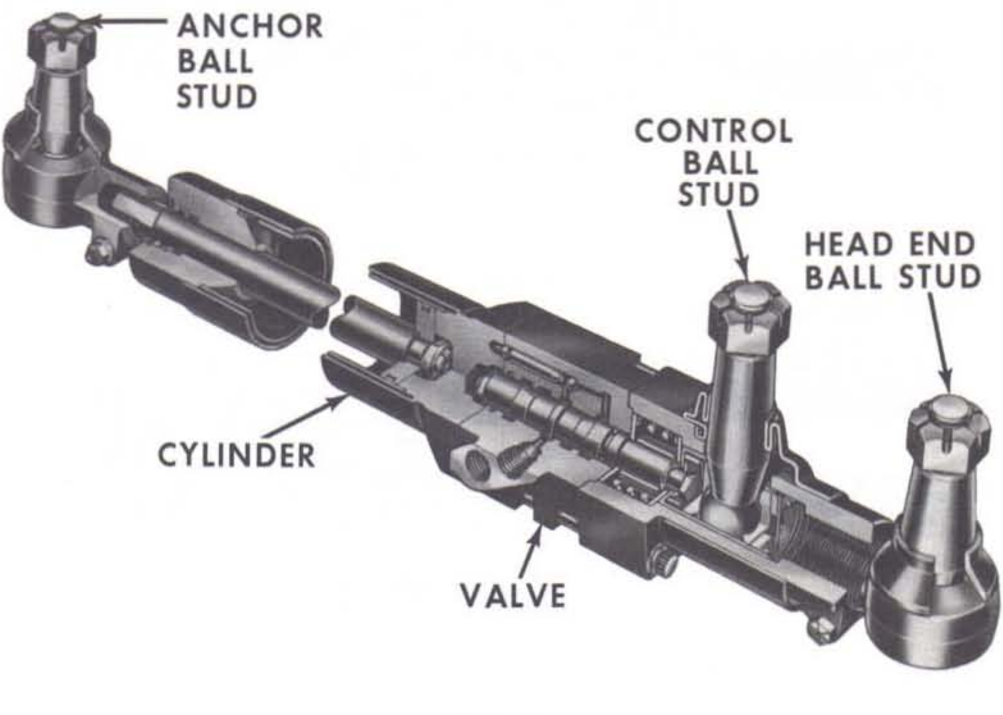


Figure 187

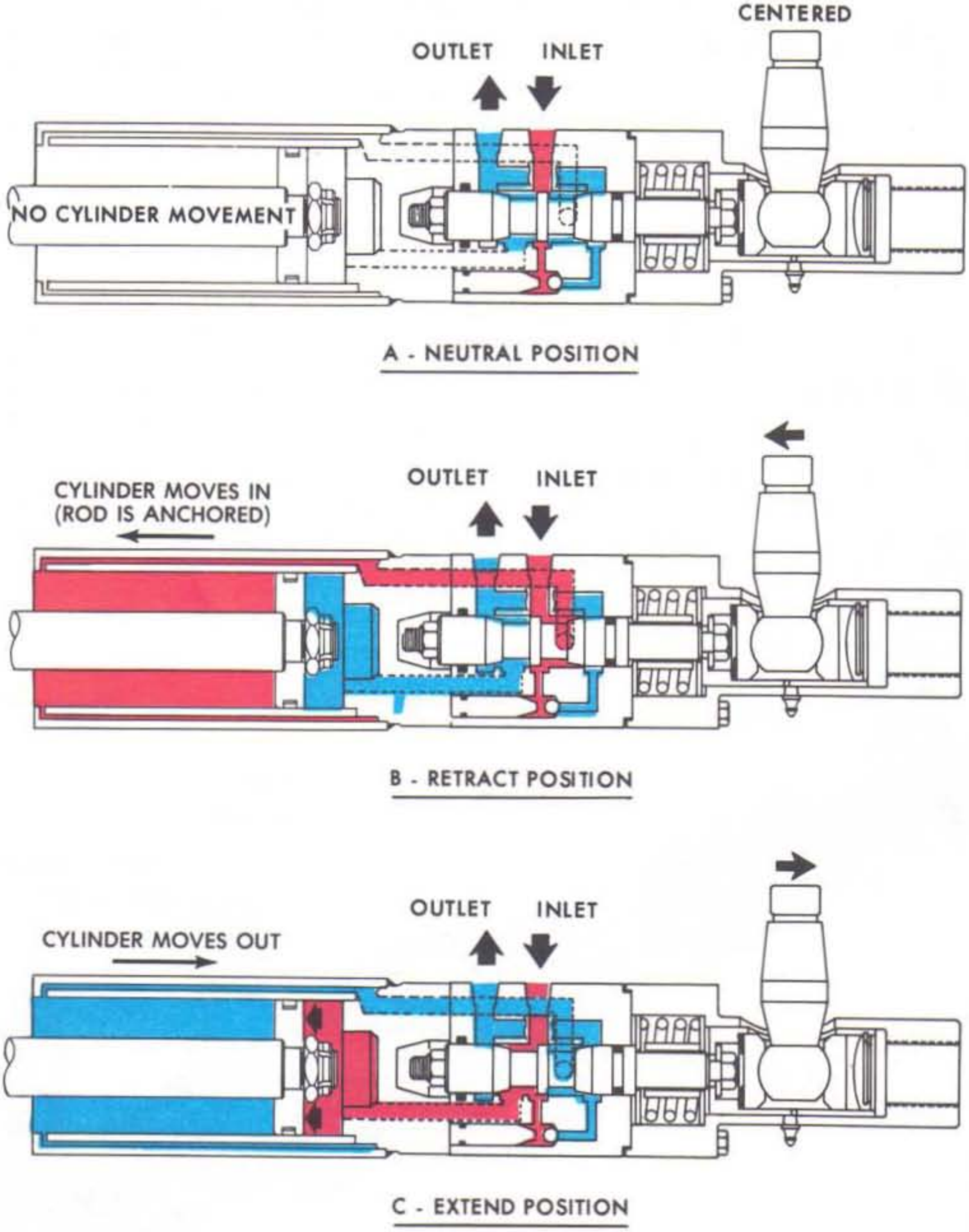

CENTERED
OUTLET
INLET
NO CYLINDER MOVEMENT
A - NEUTRAL POSITION
CYLINDER MOVES IN
(ROD IS ANCHORED)
OUTLET
INLET
B - RETRACT POSITION
CYLINDER MOVES OUT
OUTLET
INLET
C - EXTEND POSITION

Figure 188

In View A is the neutral flow condition. There is no relative movement between the spool and valve body; in other words, the spool is centered. Oil from the pump is directed back out to the tank.

View B shows the control ball stud actuated to retract the cylinder. The spool, as we see it, has been pushed to the left. Oil from the pump is directed to the rod end of the cylinder. Since the rod is anchored, pressure pushes against the rod packing to move the entire steering unit to the left. At the same time, oil pushed out of the cap end is returned to the tank.

Flow continues as shown until the control ball stud stops. The valve body then immediately catches up with the spool and the flow condition again is as in View A.

When the ball stud moves to the right (View C), flow reverses. Pump delivery is routed to the cap end of the cylinder and oil pushed out of the rod end is returned to tank. Pressure in the cylinder pushes on the cap end and moves the steering unit to the right to follow the control ball stud.

In either direction of operation, relative movement between the spool and valve body is slight . . . only enough to open the ends of the cylinder to pressure and return.

Check Valve

The small check ball in the valve body is normally held seated by pressure at the valve inlet (pressure port). If there is a hydraulic failure or a loss of power, pressure drops and lets this valve unseat. Then, oil can circulate freely between the cap and rod ends of the cylinder. This avoids a hydrostatic lock and permits manual steering. The control ball stud then simply moves the whole steering unit, except for the anchored rod. The steering unit thus acts as a drag link to steer the vehicle.

Relief Valve

An optional relief valve (Figure 189) can be incorporated in the steering valve if the flow control and relief valve is not used. This is actually a compound relief valve as described in Chapter 5. It doubles as a check valve in case of power loss.

Ball Stud Mounting

As shown in Figure 190, the control valve ball stud can be mounted in any of four positions relative to the port connections in the valve.

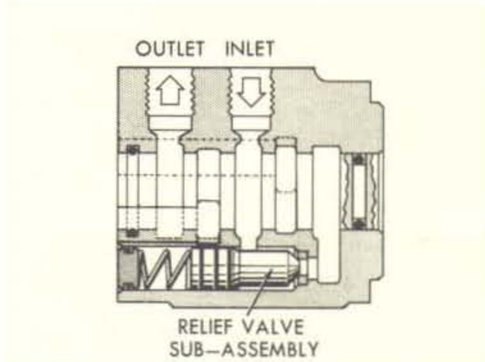


Figure 189

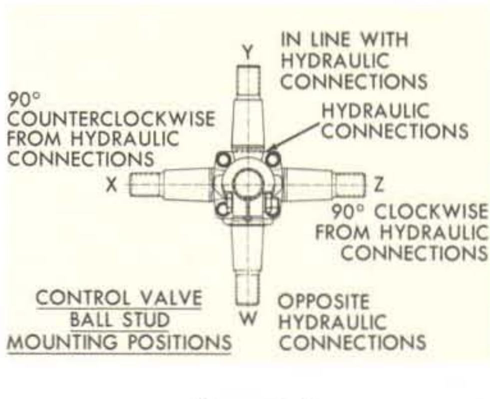


Figure 190

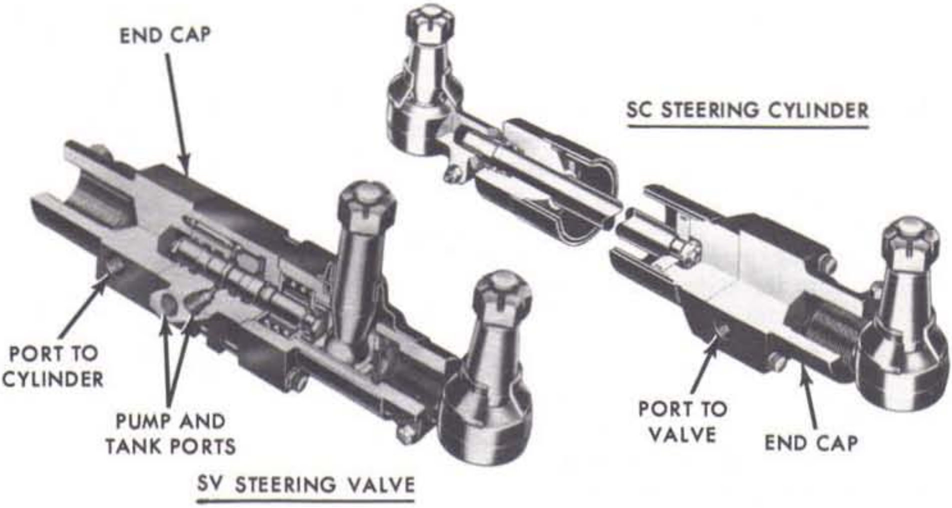


Figure 191

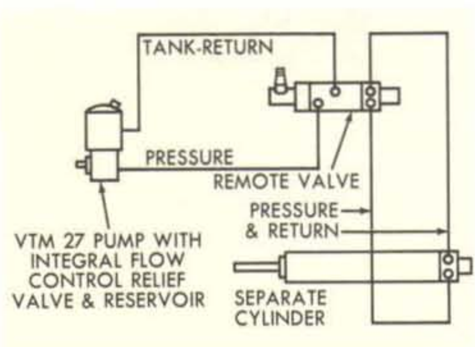


Figure 192

REMOTE LINKAGE SYSTEM CIRCUIT

In a remote installation, the valve and cylinder are mounted separately. As shown in Figure 191, each is equipped with an end cap threaded to accommodate a mounting ball stud. The end caps also contain ports to make the hydraulic connections between the valve and cylinder. Otherwise, the cylinder and valve are the same construction as in the integral steering unit.

Figure 192 shows the hydraulic connections. Except for the external lines from the valve to the cylinder, oil flow is the same as shown in Figure 188.

Auxiliary Side Ports

When two cylinders are operated from the same valve, a special body is available which incorporates auxiliary ports on the side. Figure 193 shows the hydraulic connections when a separate cylinder is used together with an integral steering unit. . .that is, a combination integral-remote circuit. Connections to the remote circuit are shown from the valve's side ports.

In Figure 194, you see a dual-remote circuit. This circuit can use the special

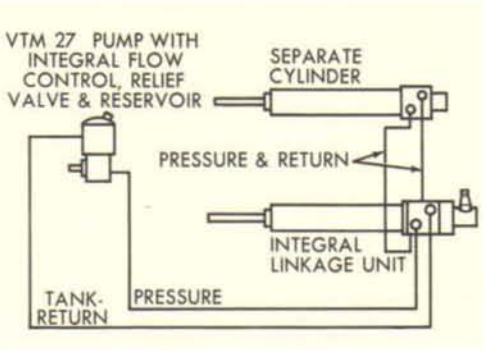


Figure 193

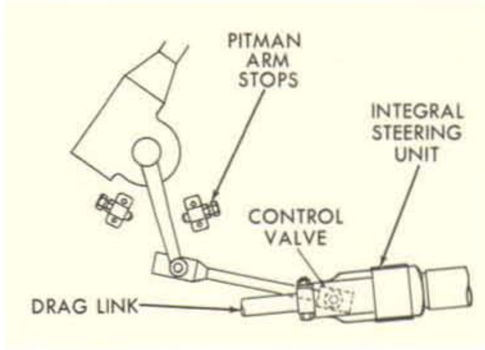


Figure 195

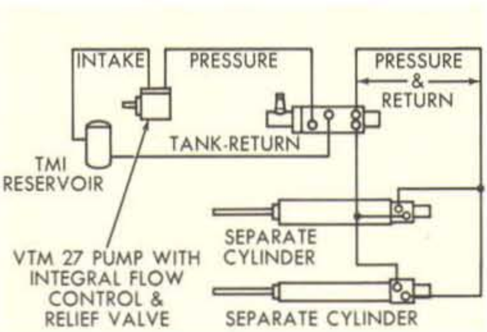


Figure 194

side-ported body, but as shown the second remote cylinder can be "teed" into the lines to the opposite cylinder. Notice that the port connections to the cylinders are opposite, so that one will retract as the other extends.

Pitman Arm Stops

Pitman arm stops (Figure 195) are used in many power steering systems to protect against overloading and overheating when the steering wheel is "hard over".

Very obviously, when the wheels turn to steer the vehicle, they can turn only so far. Either stops are provided at the wheels, or the steering linkage can travel only so far before something interferes.

Without Pitman arm stops, the steering valve continues to supply some boost when the wheels are turned as far as possible and the driver is still pulling the steering wheel. But the only place the oil can go is over the relief valve. This generates excessive heat and can burn the pump out in a very short time.

The Pitman arm stops are adjusted so that the Pitman arm is blocked just before the wheels must stop. This gives the steering valve room to center. Thus the pump is unloaded through the open-center valve when the steering wheel is "hard-over".

Pitman arm stops can and should be incorporated with any system that has a separate steering valve. They cannot be incorporated in valve-on-gear or integral power gear systems.

CHAPTER 8
Hydrostatic Drives

In the broadest sense of definition, any pressure hydraulic circuit has to be considered a hydrostatic drive or hydrostatic transmission. The definition is inherent in the function of the system, which is to transmit power from one point to another . . . or to transmit "drive" from an input to an output member of a vehicle.

We usually see the terms hydrostatic transmission and hydrostatic drive (we'll use them interchangeably) defined as "any hydraulic drive in which a positive displacement pump and motor transfer rotary power by means of fluid under pressure."

This definition narrows our scope considerably. We are immediately eliminating all drives that involve reciprocating pumps and linear actuators. We now are talking only about rotary inputs and outputs.

TRACTION DRIVES

In mobile hydraulics, our interest in hydrostatic drives is even narrower. We are accustomed to labeling as hydrostatic drives principally those systems used to operate the driving or traction-producing wheels of a vehicle.

Thus, we must immediately divide hydrostatic drives into traction drives and non-traction drives. A traction drive is a drive used to propel a wheeled vehicle. A non-traction drive is used for some other function on the machine . . . winching, for example.

The purpose of this chapter is to acquaint you with the operating principles of all hydrostatic drives, but with special emphasis on traction drives.

WHAT IS A TRANSMISSION?

A transmission is any device that is capable of matching the torque and speed of the input (engine or electric motor) to the torque and speed requirements of the output member which drives the load. This is very often accomplished through a gear box or mechanical transmission. When we replace the gear box with a hydraulic pump at the input and a hydraulic motor at the output, we have a hydrostatic transmission.

ADVANTAGES OF HYDROSTATIC DRIVES

The art of hydrostatic drives has been well known and well-explored for more than 75 years. Hydrostatic transmissions have been used extensively in machinery, mobile, marine and aircraft applications since the turn of the century. But until recent years, they were slow to be adopted on mass-produced vehicles. Economical, light-weight, compact components suited to these applications just weren't available. Now that the components have been developed specifically with these drives in mind, we can expect to see many manufacturers incorporating hydrostatic transmissions in the next few years. The advantages are many:

- With a hydrostatic drive, we can have infinitely variable regulation of output speed and torque. Control can be accomplished easily and accurately.
- Sixty-five to ninety percent of maximum torque is available for start-up or "breakaway".
- The drive is capable of smooth acceleration without the "steps" we have in a gear box.

- Low inertia of the rotating parts allows rapid starting, stopping and reversing . . . and with outstanding smoothness, accuracy and precision.

- The power source (engine) can be located anywhere in the machine without worrying about complicated driveshaft and driving axle arrangements.

- The hydraulic components are reliable and long-lived.

- With the latest components, weight and size are less in relation to the power transmitted.

DRIVE COMPONENTS

Piston, gear and vane-type pumps and motors have been used in hydrostatic drives. Piston units are most common, because they are readily adapted to variable displacement designs. Also, piston units can be designed for higher working pressures with good efficiency. However, high performance vane type motors also are found in many hydrostatic drives where step type, rather than infinitely variable transmission is acceptable.

OPERATING CONTROLS

The maximum torque of a hydrostatic drive is limited by the pressure setting of a relief valve placed in the circuit between the pump and motor. As we'll see in the next part of this chapter, additional controls include flow and directional control valves and controls of pump and/or motor displacement.

CLASSIFICATIONS AND CHARACTERISTICS

Let's now turn our attention to the various ways hydrostatic drives are classified . . . and at the same time see the basic ways they are controlled. Then, we can examine some typical transmission circuits.

We classify hydrostatic drives as:

- Narrow, medium or wide in torque range.
- Integral or split.
- Open or closed circuit.
- Constant or variable torque and/or horsepower.

TORQUE RANGE

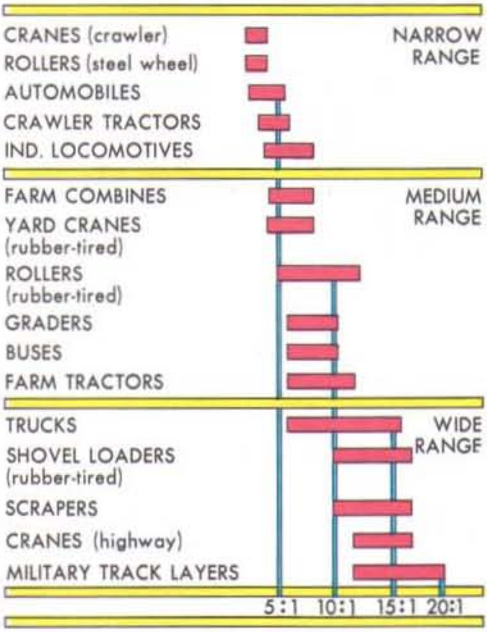
Torque range is actually a classification of the minimum and maximum torque requirements of the output member. We define it as the ratio between the torque required for maximum pull and the torque available at maximum speed. This involves calculations far beyond the scope of this manual, but we can illustrate torque range in simple terms by going to an automobile drive train and assuming some gear ratios.

Suppose that the lowest transmission gear has a 5-to-1 ratio and the high gear has a 1-to-1 ratio. The ratio between the two ratios is 5-to-1, and we could call this the torque range.

A vehicle's torque range depends on how much engine torque is available, what the tractive effort or force at the edges of the wheels is and the maximum speed required. It is relatively low in today's automobile because of high-powered engines and low tractive efforts. But vehicles that are lower powered and have very high tractive efforts have much higher torque ranges.

The Fluid Power Handbook classifies torque ranges like this:

Narrow – Up to 5-to-1
Medium – 5-to-1 to 10-to-1
Wide – Above 10-to-1



From FLUID POWER HANDBOOK
Courtesy Industrial Publishing Co.

Figure 196

Figure 196 shows typical torque ranges for the drives of several kinds of vehicles.

A hydrostatic drive can operate in a range of torques only if the pump or the motor, or both, are variable displacement. The torque range of the drive is always the range of ratios between the motor and pump displacement. When the displacements are equal, there is direct 1-to-1 drive. When the motor displacement is greater, there is reduction in speed and multiplication of torque.

Though in theory, we have an infinite torque range, practical considerations may limit us considerably. Motors particularly are seldom operated at less than 1/3 to 1/5 maximum displacement.

So varying the motor displacement may only give us a narrow torque range. Since pressure is also a function of torque, the range can be extended by varying the operating pressure. Thus a variable delivery pump can provide a large volume of oil at low pressure or a small volume at high pressure with the same horsepower input.

INTEGRAL AND SPLIT HYDROSTATIC TRANSMISSIONS

Suppose now that we want to take the transmission out of the automobile drive train and substitute a hydrostatic transmission. Since the transmission input and outputs are in a straight line, we could substitute an integral drive. . . that is, a pump and motor contained in a single assembly with their drive shafts in line with each other (Figure 197).

The other possibility is to substitute the hydrostatic drive for the transmission and driving axle; that is, to couple the pump to the engine and the motor(s) to the driving wheel(s) (Figure 198). This is called a split installation, because the motor is mounted remotely from the pump. If we use one pump and two motors, we have a dual split drive. If a separate pump drives each motor, it is a twin split drive.

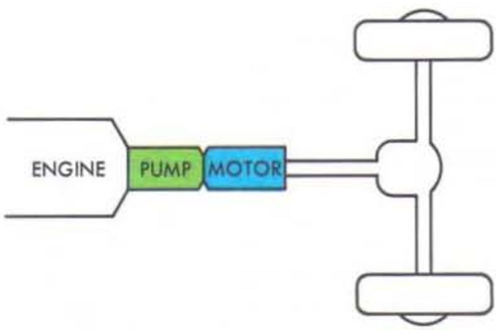


Figure 197

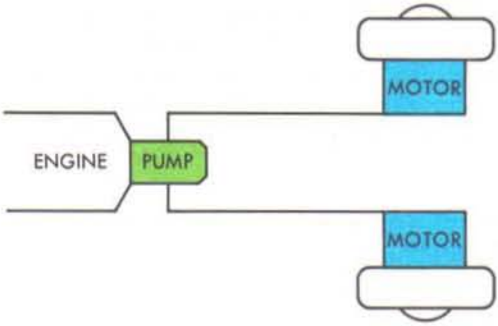


Figure 198

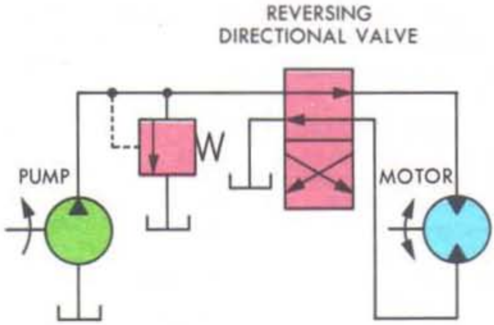


Figure 199

OPEN CIRCUIT

The difference between an open- and closed-circuit drive is in where the oil goes after it leaves the motor. If it is returned directly to the pump inlet we have a closed circuit or closed loop. If it returns to the reservoir first, we have an open circuit.

In open circuit drives, we may use fixed-displacement pumps and motors. The important considerations in these drives are the engine characteristics; not the pump and motor performance. We can vary the drive speed of the motor by varying the speed of the engine or by using a flow control valve.

Constant Speed Open Circuit Drive

Figure 199 shows a simple open circuit drive with the only controls a directional valve for reversing and a relief valve to prevent overload damage. If the pump and motor have the same displacement, this drive is nothing more than a reversible fluid driveshaft. Discounting losses to friction and slip, the output speed and torque will be equal to the input speed and torque. Torque can be multiplied, but only at a constant ratio. The ratio will be equal to the motor displacement divided by the pump displacement. Of course we'd have a proportionate reduction in output speed.

Variable Speed Open Circuit Drives

We can vary the speed of the motor in an open center drive by using a flow control valve. Any of the three connections you studied in Chapter 5 can be used . . . bleed-off, meter-in and meter-out.

Bleed-Off Drive Control. In the bleed-off arrangement (Figure 200), the control is an adjustable flow control valve connected between the pressure line and the tank. Flow from the pump to the motor is not restricted; therefore, we don't lose torque through the drive to an artifically-induced pressure drop. Oil that is bled off doesn't get to the motor, so the motor speed varies inversely with bleed-off flow.

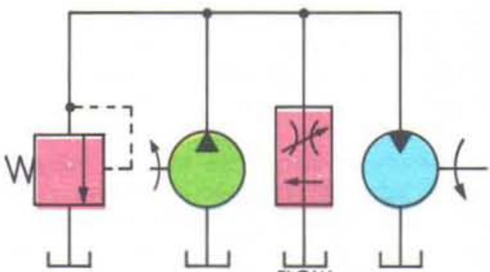


Figure 200

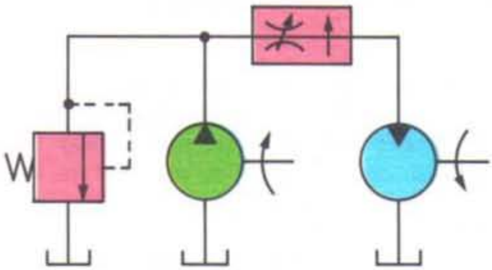


Figure 201

There are two definite advantages to this arrangement. (1) Pressure adjusts to the load, which reduces the torque effort required from the engine, and (2) throttling losses are in the branch circuit and are at the operating pressure rather than the relief valve setting.

Meter-In Drive Control. In the meter-in drive control (Figure 201), more exact control can be achieved, but we are placing a pressure drop between the pump and the motor. Thus, there must be a loss of power proportional to the percentage of pressure drop. This loss can be reduced by using a flow control and relief valve combination, which, you'll recall is designed only for meter-in circuits.

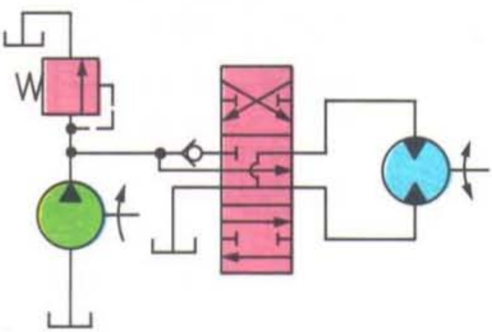


Figure 202

Meter-in drive control also can be accomplished with an infinite-positioning mobile directional valve (Figure 202). The open center motor spool is used.

Meter-Out Drive Control. The meter-out drive control, with the flow control valve in the return line (Figure 203) has the advantage of controlling overhauling loads. Again, though, placing a pressure drop in series with the motor creates a loss.

Double-Pump Open Circuit Drive

It is possible to create a three-speed hydrostatic drive by using a double pump and two directional valves (Figure 204). The two pumping cartridges have different

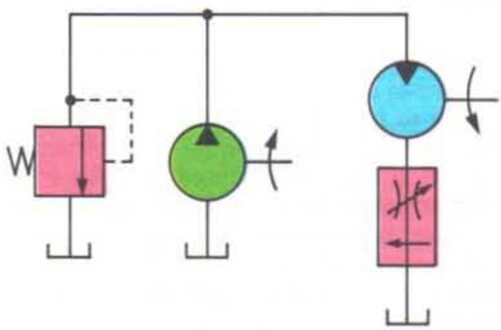


Figure 203

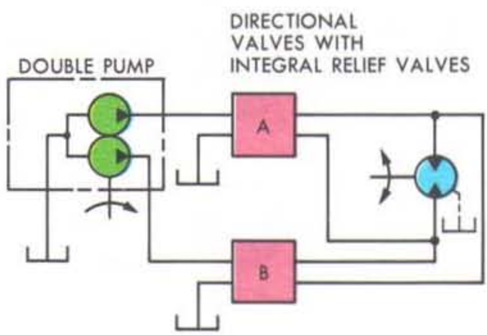


Figure 204

outputs. With valve A shifted, the motor receives flow from one cartridge. Shifting valve B feeds the motor from the other cartridge for a different speed. Maximum speed occurs with both valves shifted; the motor then receives the combined delivery of the two cartridges.

In this design, of course, we don't necessarily have stepless or infinitely variable control. In this respect, the double-pump drive is comparable to a clutch and three-ratio gear box. However, we can make the control stepless by using the infinitely-positioning type of mobile directional valves.

This drive circuit has been used on hydraulically operated rototillers.

CLOSED CIRCUIT DRIVES

In a closed circuit drive (Figure 205), the pump and motor are connected inlet-to-outlet to form a closed "loop" for oil flow. There are no flow control or directional valves. If the output speed is to be controlled, either the pump or motor must be variable displacement. Direction of rotation of output can be changed by using an over-center variable displacement pump. Pump and motor must be sized to meet torque, speed and horsepower transmission requirements.

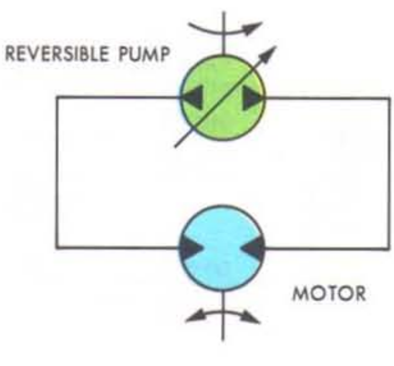


Figure 205

Relief and Replenishing Network

We don't get away without using valves in a closed circuit because we still need a relief valve and we must have a way to replenish the circuit with oil; that is, to make up for any oil lost to leakage and insure that the pump inlet is always supplied. The valves that are required form the relief and replenishing network (Figure 206). The system works like this:

- Replenishing fluid is supplied between the replenishing check valves C and D at charge pressure. The reservoir may be pressurized. If not, we must use a separate charging pump to supply replenishing oil. The replenishing check valves direct this low pressure fluid to whichever side of the pump is operating as the inlet.

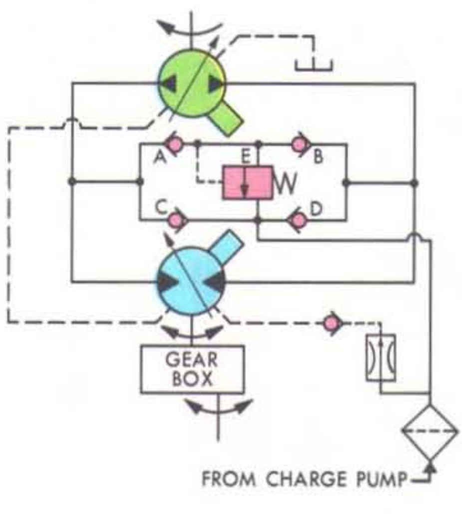


Figure 206

- The relief valve (E) is connected between the cross-line check valves A and B and can divert pump output through replenishing check valve to pump inlet line if excess pressure is encountered.

- In Figure 206, let's assume that the main stream of flow is going clockwise. The left side of the loop therefore is under low pressure and the right side is at system operating pressure. Check valve B will open to connect the high pressure side to the relief valve inlet. Check valve A is held closed by system pressure. Check valve C is open to replenish the left-hand branch of the loop and check valve D is held closed.

- If operation is reversed, the left side of the loop is the pressure side. The right side is replenished through check valve D and the relief valve is connected to the pressure side through valve A. Valves B and C are held closed by pressure.

CHARACTERISTICS OF CLOSED CIRCUIT DRIVES

We have said that the characteristics of the closed loop drive depends on the pump and motor characteristics. These vary with the displacement controls. An "over-center" control on the pump lets us reverse the direction of the drive. A pump displacement control regulates the rate of oil flow and therefore the speed of the motor. A motor displacement control regulates the motor's speed and torque.

Let's look at the possible combinations, assuming that the pump is driven at a constant speed (rpm).

Fixed Displacement Pump and Motor

If neither the pump or motor has a displacement control, we again have a simple hydraulic driveshaft. With equal displacements, output speed is equal to input speed and output torque equals input torque. If the displacements are unequal, the torque and speed change in proportion to the displacements.

Variable Displacement Pump, Fixed Displacement Motor

The combination of a variable displacement pump and a fixed displacement motor is called a constant torque drive. With the fixed displacement motor, the operating pressure is always proportional to the motor torque as determined by the load. The term "constant torque" means that torque and pressure are always proportional, regardless of the speed. Speed, of course, depends on the pump delivery.

This drive is reversible with an over-center control on the pump. It is suitable for narrow and medium torque range applications without an additional transmission, or for wider ranges with the addition of a two-speed gear box. Most hydrostatic traction drives use this combination.

Fixed Displacement Pump, Variable Displacement Motor

Constant horsepower is the name given a drive which combines a fixed displacement pump with a variable displacement motor. If we assume that the pressure doesn't change, neither does the horsepower. You'll remember that we had exactly that situation in a pressure compensated, variable motor in Chapter 4. The motor was "compensated" for a pre-set pressure, and torque increases simply resulted in proportionate speed decreases. The horsepower thus was always constant.

Variable Displacement Pump and Motor

When both the pump and motor have variable displacement, the full speed and torque characteristics of both the constant torque and constant horsepower drives are combined. In this combination, we can effectively increase the range of the transmission.

For example, let's take a pump with a minimum operating displacement of 2 cu. in./min. (32.77 mL/min.) and a maximum of 20 cu. in./min. (327.7 mL/min.). We'll operate it with a motor four times as large, and a maximum displacement of 80 cu. in./min. (1311 mL/min.). Let's assume the motor operates at a minimum of 20 cu. in./min. (327.7mL/min.).

With the pump at minimum displacement and the motor at maximum, our torque multiplication is:

$$\frac{\text{Motor Disp.}}{\text{Pump Disp.}} = \frac{80}{2} = 40\text{-to-}1$$

With the pump at maximum output and the motor at minimum displacement:

$$\frac{\text{Motor Disp.}}{\text{Pump Disp.}} = \frac{20}{20} = 1\text{-to-}1$$

Thus, we can transmit anywhere from direct drive to a 40-to-1 reduction. In other words, we have a 40-to-1 torque range. We have multiplied the pump's 10-to-1 range by the 4-to-1 range motor.

DISPLACEMENT CONTROLS

The displacement of the pump or motor can be controlled by a pressure compensator, a manual control or a mechanical servo control.

We described compensator controls in detail in Chapter 3 and Chapter 4, for pumps and motors respectively. A manual control is simply a handle of some sort linked to the pump or motor's yoke to change the angle of the swash plate in in-line units or the angle between the cylinder barrel and driveshaft in bent axis units.

Servo Control

A mechanical servo control (Figure 207) simply applies a hydraulic boost to the effort on the handle. It is nothing more than a follow valve operating inside a power piston which doubles as a valve body. The piston is connected to the yoke which swings the swash plate to varying angles. The valve spool is integral with the control-actuating stem.

Oil under pressure is supplied to the control from a passage in the pump head. When the stem is moved in or out, the spool ports the oil under pressure to one end of the piston and relieves the other end into the case, which is drained to the tank. The piston moves until it catches up to the valve, and thus actuates the swash plate yoke.

A similar servo control is used to swing the yoke on bent axis piston pumps or motors.

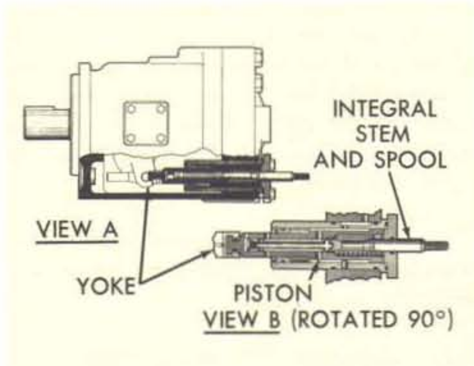


Figure 207

EFFICIENCY OF HYDROSTATIC DRIVES

As the saying goes, nobody's perfect. And we do have some power losses in hydrostatic drives. We can divide these power losses into two groups: (1) loss of speed and (2) loss of torque.

Speed Losses

Speed is lost through slip (internal leakage). Speed loss increases as pressure increases. High pressure at low displacement makes speed losses large in proportion to total speed. However, as displacement increases, the losses remain fairly constant. Remembering that efficiency is a percentage we could say that speed losses result in lowest efficiency at low speed but have less affect on efficiency at high speed. Unit volumetric (or speed) efficiency alone may be 95% or better at high speed and pressure.

Torque Losses

Torque losses tend the other way. We always lose some torque to friction and to pressure drop through the lines and components. Increased speed and increased pressure both contribute to higher torque losses. Calculations of power losses are best left to the design engineers.

HYDROSTATIC DRIVE CIRCUITS

Let's close this chapter with quick looks at some traction drive circuits. We'll first look at some basic combinations involving more than one pump or motor . . . then at some complete circuits.

TWIN DRIVE

A twin drive (Figure 208) consists of two pumps of equal displacement (or a double pump) driven by the same engine and a

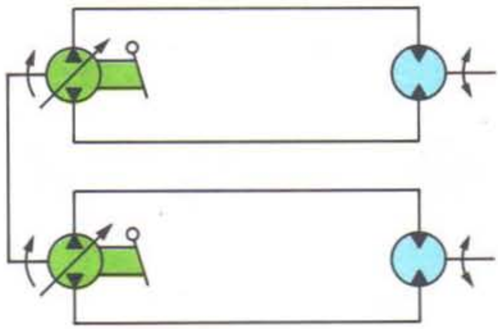


Figure 208

separate motor driven by each pump. The pumps are variable displacement and the motors are fixed.

Many tracked vehicles use this drive. One loop serves each track. Actuating the pump controls for varying displacements steers the vehicle by running one track faster than the other. Pivot turns also are possible by stopping one track entirely or backing one track at the same speed as the other track goes forward.

Another application of this basic drive is to front end loaders and windrowers with two wheel drive and caster wheel. Each motor drives the wheels on one side of the vehicle. Turns are made the same way as a tracked vehicle.

PARALLEL DUAL DRIVE

A parallel dual drive (Figure 209) has two fixed motors driven in parallel from the same variable pump. Since the flow must divide, the maximum output speed is only half the input if the pump and motors are equal in displacement. Using a smaller displacement pump further decreases maximum output speed. Also, one motor can spin-out at twice designed speed, if no special overspeed controls are provided.

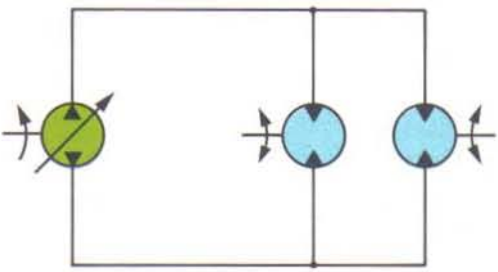


Figure 209

SERIES-PARALLEL DUAL DRIVE

With a special control valve, the dual drive can be made a series-parallel drive (Figure 210). This arrangement doubles the normal range. In one valve position, the operation is the same as a parallel drive. But the valve can be shifted to operate the motors in series. In series, the flow is not split, so the motors run twice as fast, but the pressure required is also doubled. Shifting gears from series to parallel is not stepless, and the pump displacement control must be reset during the shift.

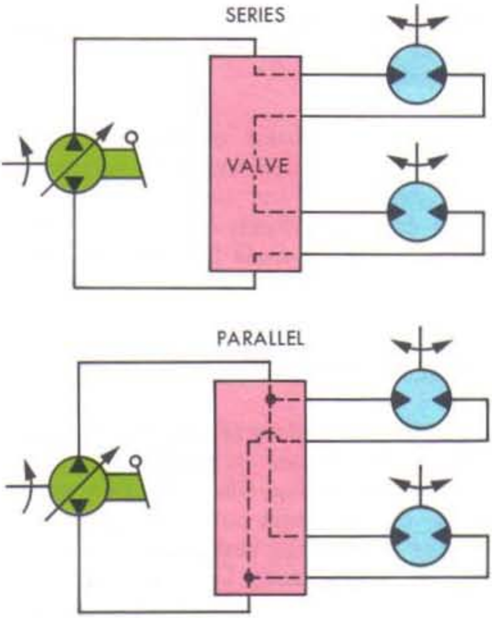


Figure 210

GARDEN TRACTOR DRIVE

A typical garden tractor drive (Figure 211) uses a variable displacement inline piston pump with a fixed displacement inline piston motor of the same size in an integral installation. The pump is coupled directly to the engine which is mounted at the rear of the vehicle. The motor drives the rear axle through a reduction gear.

The circuit (Figure 212) is a conventional closed loop with cross-line check valves to direct pressure to the main relief valve and replenishing check valves to open the replenishing pump outlet to the low pressure side of the loop. The replenishing pump is positive displacement. It has its own relief valve, which limits the maximum replenishing pressure.

This system on a garden tractor makes it extremely simple to operate. With the

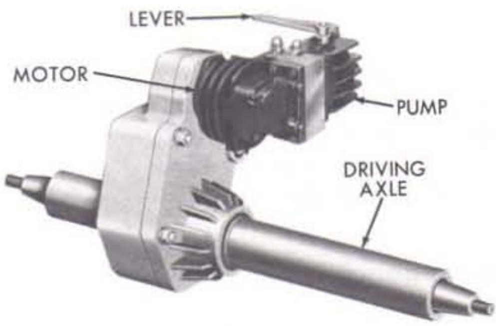


Figure 211

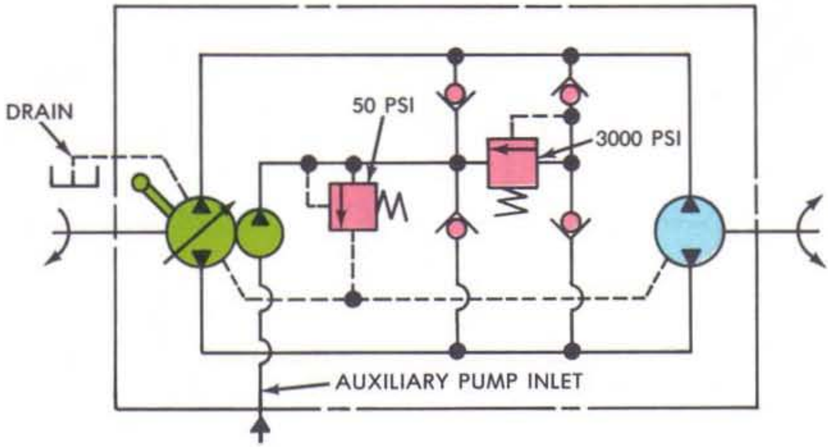


Figure 212

engine governed to run at a constant speed, the operator is concerned with only one control. This manual control actuates the pump yoke thereby controlling speed, direction, acceleration and deceleration.

LOG SKIDDER DRIVE

Earlier in this chapter, we said that although the constant horsepower drive gives the same basic reaction as the torque converter in an automatic transmission, few mobile traction drives use this combination of fixed pump and variable motor.

The circuit in Figure 213 is one of the few that does. It was designed to provide a log skidder with a drive capable of increasing torque with a heavier load and increasing speed when the pull decreases.

This circuit is a twin drive, with each drive in a separate open circuit. Mobile directional valves are used to operate the drive. Three other circuits (not shown) are operated from the same double pump that supplies the drive motors (Dozer, Harp and Steering Circuits).

The motors are pressure-compensated, in-line piston units; the pump is a high-performance balanced vane unit of slightly smaller displacement. Each circuit drives a pair of wheels through a three-speed transmission-and-axle assembly (trans-axle). Including the hydrostatic drive and a 9-to-1 reduction at the wheels the overall reduction ratios are:

Low gear:	193:1
Second gear:	91:1
High gear:	43:1

Gearing for these ratios, of course, would not be the problem. The advantages of this drive are elimination of a long driveshaft

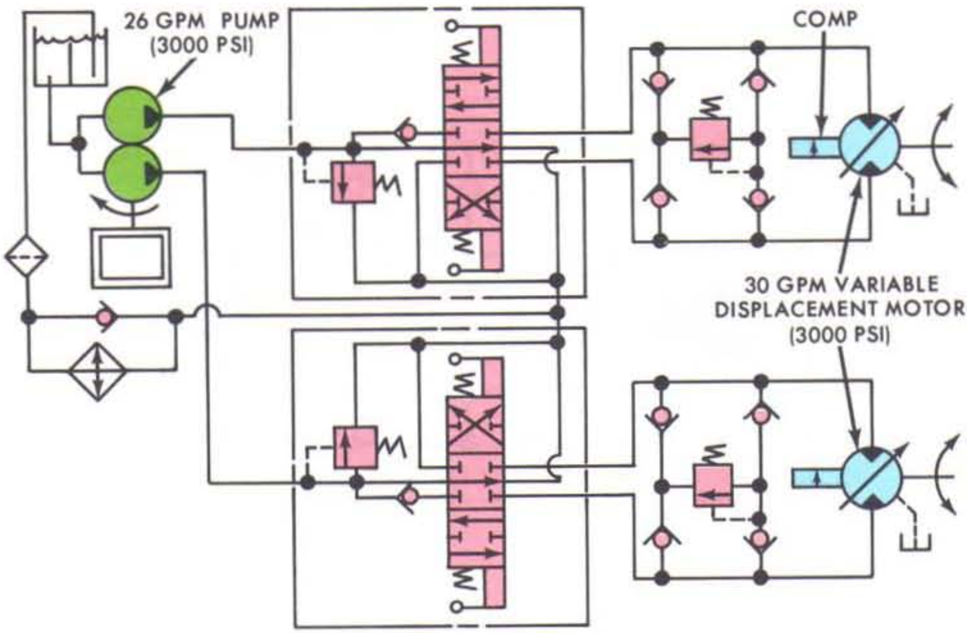


Figure 213

from the engine to the driving trans-axle, the constant horsepower feature, and the throttle-response characteristics which closely match the automatic transmission in an automobile.

PACKAGED DRIVES

Vickers right angle transmission units (Figure 214) combine in-line piston pump and an in-line piston motor in a single package. In the drive shown, both units are variable displacement . . . giving this transmission the characteristics of variable-torque and variable horsepower. The pump can be stroked over-center for reversing.

Besides these basic drive components, the assembly contains a vane type replenishing pump and all the necessary control valves.

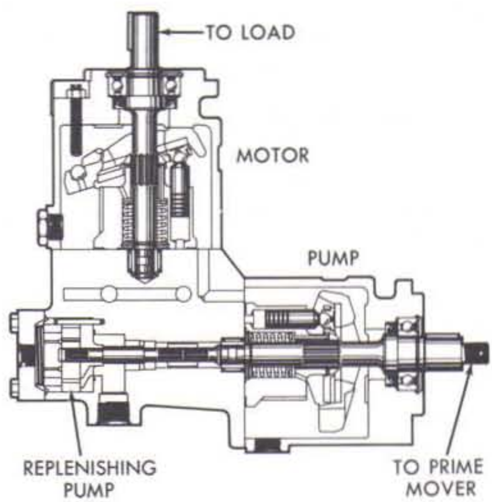


Figure 214

A typical package drive circuit diagram is shown in Figure 215. Notice that the replenishing pump, driven from the same shaft as the piston pump, has its output directed to an external port. Below that port is an IN port to return replenishing oil to the closed loop. Between these ports, we can place a filter and, if required, an oil cooler. We also can use the replenishing pump output for supplying a second transmission in a twin installation or even for operating another system when the main drive is in neutral. The 25 psi (1.7 bar) relief valve directs excess replenishing oil back to the reservoir.

To use the drive, we merely couple the pump shaft to a prime mover and the motor shaft to the output, and make the replenishing pump connections. The hydrostatic transmission we described for the garden tractor is one of this general series of packaged drives.

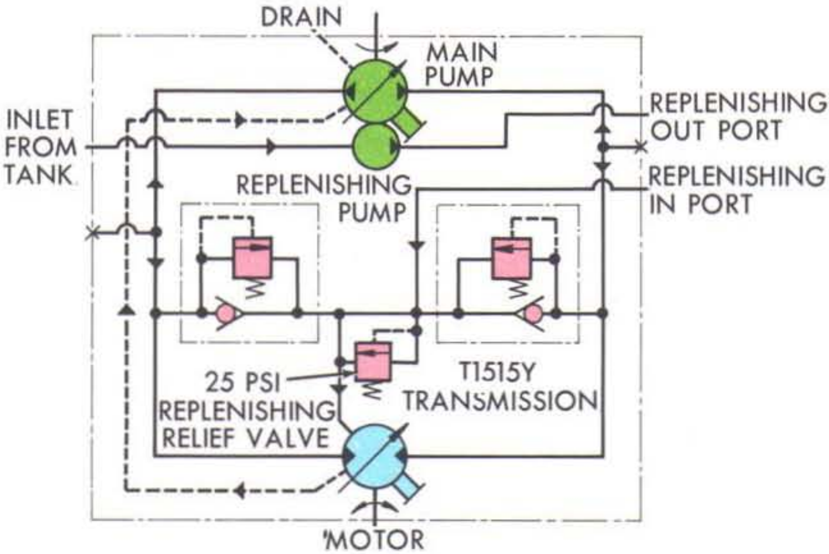


Figure 215

CHAPTER 9

Hydraulic Tubing, Piping, Hose and Fittings

There are various kinds of connections used to pass hydraulic fluid from its source and put it where the action is. Just as a garden hose takes water from a faucet and passes it through its length to water the lawn, so does the plumbing in a hydraulic system allow fluid to pass from the reservoir to the work. Most of us are somewhat familiar with the piping, tubing and flexible type connections used in a mobile hydraulic system. However, the real importance of this plumbing tends to be neglected. As a matter of fact, the plumbing is every bit as important as any component in the system. Without its being properly installed, clean, stable and correctly sized, the entire system is subject to extra-expensive operation and the possibility of complete system failure. In other words, no hydraulic system is any better than its weakest link.

This chapter concerns itself with the classification, selection and installation of hydraulic tubing, piping, flexible hose and the various fittings used in normal hydraulic systems.

ABOUT TUBING

Generally speaking, tubing can be used in a system requiring no more than one inch (25.4 mm) diameter plumbing nor pressures exceeding 6000 psi (414 bar). The tubing is flared (discussed later in the chapter) and fitted with threaded compression fittings.

There are two types of tubing used for hydraulic lines: seamless and electric welded. Both are suitable for hydraulic application. Seamless tubing is produced by cold-drawing pierced or hot extruded billets from a dead-soft steel. Welded tubing is made from a cold-rolled strip of steel which is shaped into a tube, welded and drawn. Seamless tubing is made in larger sizes than tubing which is electric welded.

When practicable, tubing is highly desirable because it is easily bent, and so requires fewer pieces and fewer fittings. Unlike pipe, tubing can be cut, flared and fitted in the field. In general, tubing makes a neater, less costly, lower-maintenance type system with fewer flow restrictions and less chance of leakage.

SIZING TUBING

The type tubing used is determined by the flow, type of fluid, fluid velocity and system pressures. Nominal dimensions of tubing are given as fractions in inches or as "dash numbers". Both are in reference to the O.D. (outside diameter) of the tube. The dash number represents the O.D. of the tube in 1/16th of an inch (1.59 mm). In other words, a 5/8th inch (15.9 mm) tube would have a dash number of -10. This would indicate an O.D. of 10/16th or 5/8th of an inch (15.9 mm).

Tubing sizes increase as follows:

- 1/16 inch (1.59 mm) – increments from 1/8 inch (3.2 mm) tube through 3/8 inch (9.5 mm) tube.
- 1/8 inch (3.2 mm) – increments from 1/2 inch (12.7 mm) tube through 1 inch (25.4 mm) tube.
- 1/4 inch (6.4 mm) – increments from 1 inch (25.4 mm) tube and beyond.

TUBING WALL THICKNESS

The system pressure determines the thickness of the various tubing walls. Proper thickness and pressures are as shown in Tables 1 and 2.

TABLE 1 – Pressures From 0-1000 psi (0-69 bar)

Flow Rate		Valve Size		Tubing Outside Diameter		*Tubing Wall Thickness	
Gpm	lpm	in.	mm	in.	mm	in.	mm
1	3.79	1/8	3.18	1/4	6.35	0.035	0.89
1.5	5.68	1/8	3.18	5/16	7.94	0.035	0.89
3	11.4	1/4	6.35	3/8	9.53	0.035	0.89
6	22.7	3/8	9.53	1/2	12.7	0.042	1.07
10	37.9	1/2	12.7	5/8	15.88	0.049	1.24
20	75.7	3/4	19.1	7/8	22.23	0.072	1.84
34	128.7	1	25.4	1-1/4	31.6	0.109	2.77
58	219.6	1-1/4	31.6	1-1/2	38.1	0.120	3.05

TABLE 2 – Pressures From 1000-2500 psi (69-172 bar)

Flow Rate		Valve Size		Tubing Outside Diameter		*Tubing Wall Thickness	
gpm	lpm	in.	mm	in.	mm	in.	mm
2.5	9.46	1/4	6.35	3/8	9.53	0.058	1.47
6	22.7	3/8	9.53	3/4	19.1	0.095	2.41
10	37.9	1/2	12.7	1	25.4	0.148	3.76
18	68.14	1	25.4	1-1/4	31.6	0.180	4.57
42	155.2	1-1/4	31.6	1-1/2	38.1	0.220	5.59

*1010 Steel

Tubing above 1/2 inch (12.7 mm) O.D. is usually installed with either flange fittings with metal or pressure seals or with welded joints (Figure 216). If joints are welded, they should be stress relieved.

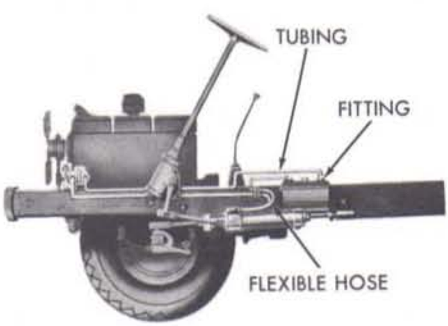
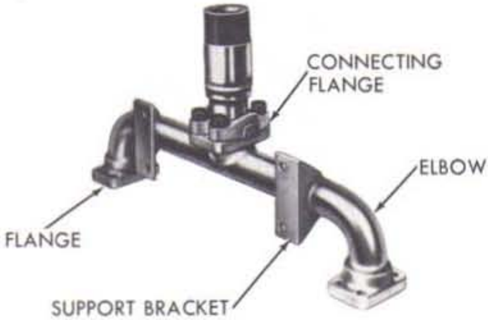


Figure 216

ABOUT PIPING

Piping threaded with screwed fittings can be used with diameters up to 1-1/4 inch (31.8 mm) and pressures of up to 1000 psi (69 bar). In instances where pressures will exceed 1000 psi (69 bar), and required diameters are over 1-1/4 inch (31.8 mm), piping with welded, flanged connections and socket-welded fittings (Figure 217) are used. Pipe size is specified by nominal I.D. (inside diameter) dimensions. The thread remains the same for any given pipe size regardless of wall thickness.

Pipe is used economically in larger size hydraulic systems where large flow is carried. It is particularly suited for long, permanent straight lines. Pipe is taper-threaded on its O.D. into a tapped hole or fitting. However,



Courtesy Anchor Coupling Co., Inc.

Figure 217

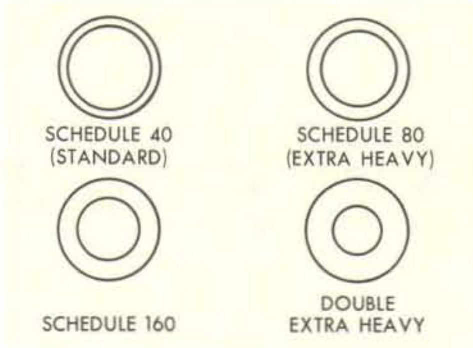


Figure 218

it cannot be bent. Instead, fittings are used wherever a joint is required. This results in additional cost and increased chance for leakage.

SIZING PIPING

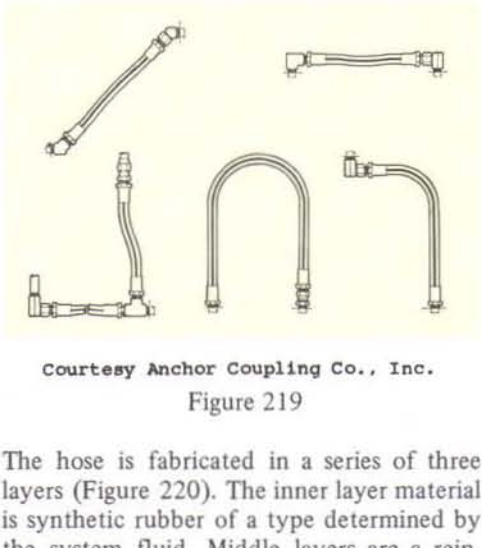
Wall thickness ratings have been adopted which define the thickness of a given pipe wall by "schedule". There are three schedules or wall thickness commonly used in hydraulic systems (Figure 218). They are as follows:

- Schedule 40 – Standard Pipe
- Schedule 80 – Extra Heavy Pipe
- Schedule 160 – Between Schedule 80 and Double Extra Heavy Pipe

Nominal size, schedule, system pressures, anticipated surge pressures and required flow must be known to insure the proper size pipe for use in a hydraulic system.

ABOUT FLEXIBLE HOSE

Flexible hose (Figure 219) is used where the hydraulic line is to be subjected to movement . . . such as the hydraulic lines from a farm tractor to a hydraulically controlled, tractor-drawn farm implement.

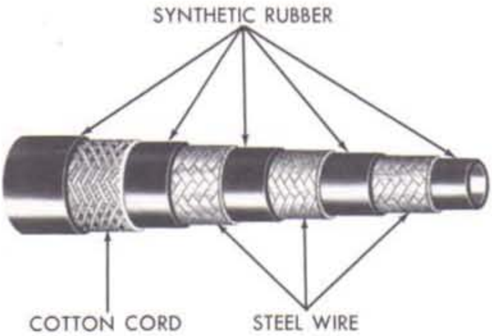


Courtesy Anchor Coupling Co., Inc.

Figure 219

The hose is fabricated in a series of three layers (Figure 220). The inner layer material is synthetic rubber of a type determined by the system fluid. Middle layers are a reinforcement of either fabric or rubber for low pressure hose; or for higher pressures, wire braid is substituted for the fabric braid. Hose can be used with either one or two wire braids depending on pressure of the system. The outer layer of the hose is a protective cover.

Hose is a common choice because it can simplify the plumbing in a hydraulic system. When properly installed it is a real benefit in absorbing shock, it offers installation freedom and helps avoid mechanical bending of fittings between end connections.



Courtesy Anchor Coupling Co., Inc.

Figure 220

Hose size is specified by I.D., O.D. and dash number.

FITTINGS

Fittings (Figure 221) should be fabricated or forged steel, not cast. They may be threaded or flanged for use with piping; or they may be flanged or of a compression type for use with the tubing. Compression fittings are either flared or bite-type (flareless). Since pipe threaded connections are subject to leakage, they are avoided whenever possible in most modern equipment. If, however, a pipe thread fitting must be used, the threads should be cut clean and smooth and assembled with a protective compound to help seal and protect the threads from corrosion.

Flanged fittings may be either threaded or welded to the pipe ends. Flanges are often provided with gaskets of a softer material to make a good seal.

PIPING RECOMMENDATIONS

KEEP YOUR PLUMBING CLEAN

You wouldn't pour clean water into a dirty glass and drink it. Neither can we

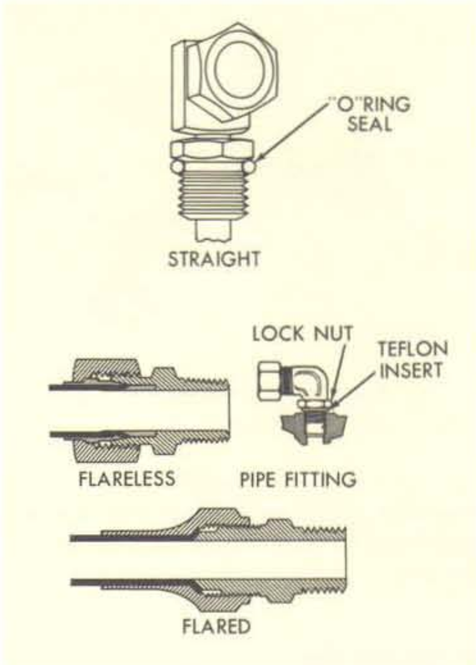


Figure 221

expect clean oil to do an adequate job if it has to go through dirty plumbing before reaching the work. Dirty oil can be the result of dirty plumbing and dirty oil causes a majority of hydraulic ailments.

Sand blasting, de-greasing and pickling are recommended for thoroughly cleaning lines prior to installation. Information of these processes is available from your Vickers application engineer or from manufacturers and distributors of commercial cleaning equipment.

SUPPORT YOUR PLUMBING

Proper support of long lines is necessary to keep vibration to a minimum. For ease in assembly, we try to keep the clamps or connecting devices away from the fittings where possible. (Exceptions to this practice might occur with high pressure lines where connections can be brazed for additional security or where welded construction is used.)

INLET (SUCTION) LINES

Inlet (suction) lines should be as short and have as large an inside diameter as possible. Where inlet lines are long, it is wiser to adapt to a larger capacity line than the pump inlet openings call for. Inlet lines should never be reduced smaller than the port size in the pump. There should be as few bends and fittings in the line as possible. Also, high pressure fittings should not be used for inlets. If you can't get the oil in – you can't get it out. Inlet connections should always be tight; loose connections allow air to enter the system.

RETURN LINES

Use of high pressure hose in return lines should be avoided. They can add to the pressure drop through the system.

Return lines when loose can draw air into the system. They should always empty below the oil level. Where long return lines are necessary, hose or tubing larger than the ports in the hydraulic units should be used. They should never be reduced in size. A minimum amount of bends and fittings should be used.

PRESSURE LINES

All pressure lines where piping is used should have forged steel fittings according to the operating pressure. Pressure lines carry ratings of 2000 psi (138 bar), 3000 psi (207 bar), and 5000 psi (345 bar).

PIPING AND FITTINGS

The proper choice of piping and fittings is very important. Tubing is more leak free than piping.

Malleable iron fittings are suitable for inlet, return, and drain lines only. Galvanized pipe or fittings have no place in a hydraulic system, except to connect cooling water to heat exchangers. The zinc has adverse effects on some types of oil additives and could flake off and cause unit failure. Use of copper tubing in hydraulic systems should also be avoided. Vibration is an inherent feature on most mobile applications. Copper tends to harden and crack at the flares.

HOSE INSTALLATION

When installing a hose, we always allow enough slack to avoid kinking the hose . . . remembering that only the hose is flexible, not the fittings. A taut run of hose will not allow movement with pressure surges – slack in the line compensates for this and relieves strain. The hose should not be twisted during installation or while in operation as it will weaken the hose and loosen the connections. A neater installation can usually be had by using extra fittings to cut down unusually long loops in a line. Hoses should be clamped adequately to prevent rubbing and to insure they will not entangle with moving parts. Where hoses are subject to chafing, they should be run through protective neoprene hose.

CHAPTER 10
Leakage and Sealing

For nine chapters now, we have been talking about moving hydraulic oil from one place to another to control motion and force. It would be an ideal situation if the oil always followed these prescribed paths without finding other places to go along the way . . . but this ideal is seldom achieved. Every step of the way, the design engineer must consider the possibility of (or the necessity for) some leakage. Sealing methods and techniques must occupy a substantial space in his bag of tricks. Sealing, in fact, is a science in itself . . . or at least it is an industry . . . for there are many companies that do nothing but design and produce parts and materials capable of sealing fluids.

In this chapter, we will consider briefly the desirable and undesirable aspects of leakage; the ways in which oil is sealed in a hydraulic system; typical seal materials we use; and ways of preventing undesirable leakage.

LEAKAGE

In Chapter 8 on Hydrostatic Drives, we stated that when a pump drives a hydraulic motor of equal displacement, the motor will turn at the same speed as the pump. This of course assumed that all the oil entering the pump was delivered to the motor and was effective in making the motor turn. But we know that we never achieve 100 percent volumetric efficiency in the system, because we do have a certain amount of leakage. Some of it is planned or designed-in leakage; some may not be planned. Any leakage, planned or not, reduces efficiency and causes power loss. It is something we live with in hydraulic systems . . . a price we pay for the many benefits we gain over other methods of transmitting power.

INTERNAL LEAKAGE

Internal leakage must be built into hydraulic components to provide lubrication for valve spools, shafts, pistons, bearings, pumping mechanisms and other moving parts. Also, in some of our hydraulic valves and in pump and motor compensator controls, leakage paths are built in to provide precise control and to avoid "hunting" or oscillation of spools and pistons. Oil is not lost in internal leakage; it always finds its way back to the reservoir through return lines or through specially provided drain passages.

However, too much internal leakage will definitely slow down our actuator. The loss in power is accompanied by heat generated at the leakage path. In some instances, excess leakage in a valve can permit a cylinder to drift or even cause it to creep when the valve is (supposedly) in neutral. In the case of flow or pressure control valves, leakage can often reduce effective control or even cause control to be lost.

Normal wear increases internal leakage by providing bigger flow paths for leakage oil. An oil that is low in viscosity leaks more readily than a heavy oil. Therefore viscosity and viscosity index of the oil (see Appendix A) are important considerations in providing or preventing internal leakage.

Internal leakage also increases with pressure, just as a higher pressure causes a greater flow through an orifice. Operating at a pressure above recommendations adds the danger of excess internal leakage and heat generation to the other possible harmful effects.

A "blown" or ruptured internal seal can open a large enough leakage path to divert all of the pump delivery. When this happens, everything can stop . . . everything, that is, except the flow of oil and generation of heat at the leakage point.

EXTERNAL LEAKAGE

At the time of this writing, no one has yet found anything useful about external leakage. External leakage combines the unfortunate aspects of internal leakage with a few more of its own. It creates a housekeeping problem; it can be hazardous, expensive and unsightly. In short, it is not to be recommended.

Incorrect installation and poor maintenance are the prime causes of external leakage. Joints may leak because they weren't put together properly, or because shock and vibration in the lines shook them loose. Adequate support of the lines helps to prevent this.

The components themselves seldom leak if they are assembled and installed correctly. However, failure to connect drain lines, excessive pressures, or contamination can cause seals to be blown or damaged and result in external leakage from the components.

SEALING

Sealing, in its broadest sense, is anything we do to keep the hydraulic oil from flowing between certain passages; to hold pressure; and to keep foreign material from getting into the hydraulic passages. When we wish to completely prevent leakage, we use a positive sealing method. When we say that a sealing method is non-positive, we mean that it allows some leakage for lubrication.

In most of our hydraulic components, non-positive sealing is usually accomplished by fitting the parts closely together. The strength of the film of oil that the parts slide against provides an effective seal. For a positive seal, however, we must provide an actual sealing part or material. In general references, we'll apply the term "seal" to any gasket, packing, seal ring or other part designed specifically for sealing.

Sealing applications are usually classified as either static or dynamic, depending on whether the parts being sealed move in relation to one another.

STATIC SEALS

A static seal is placed between parts that do not move in relation to each other. Mounting gaskets and seals are of course static, as are seals used in making connections between components. Some typical static seals in flanged connections are shown in Figure 222. Pipe thread seals, seal rings

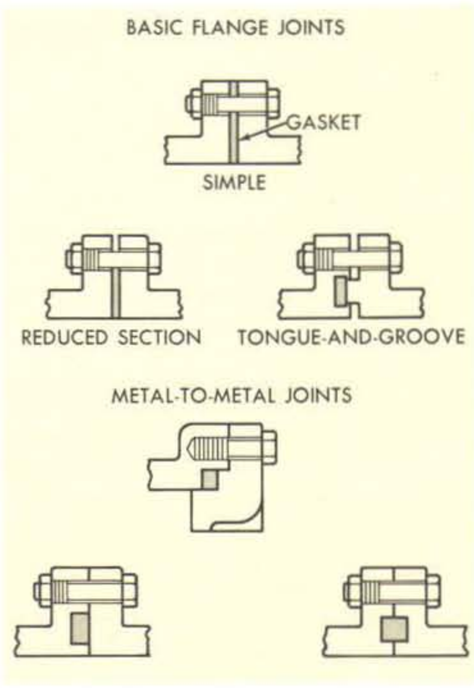


Figure 222

used with tube fittings, valve end cap seals and many other seals on non-moving parts are classified as static seals.

DYNAMIC SEALS

In a dynamic sealing application, there is either reciprocating or rotary motion between the two parts being sealed; for example, the piston-to-barrel seal in a hydraulic cylinder and the driveshaft seal in a pump or motor. Dynamic seals are many and varied, and require highly specialized knowledge to design.

O-RING SEALS

O-rings (Figure 223) are used in both static and dynamic applications. In fact, the O-ring has nearly made the flat gasket a museum piece on modern hydraulic equipment.

An O-ring is a positive seal. In installation, it is squeezed at the top and bottom in its groove and against the mating part. It is capable of sealing very high pressure. Pressure forces the seal against the side of its groove . . . in effect, packs it into a corner . . . and a positive seal results on three sides.

RECIPROCATING PARTS

Dynamic applications of O-rings are usually limited to reciprocating parts which have relatively short motion. For instance, we might seal the ends of a valve spool this way, but probably would not use it for a cylinder piston-to-barrel seal.

BACK-UP RINGS

A back-up ring (Figure 224) may be used with an O-ring to prevent its being extruded into the space between the mating parts. A combination of high pressure and clearance between the parts often calls for a back-up ring. Back-up rings are usually made of stiff nylon or teflon.

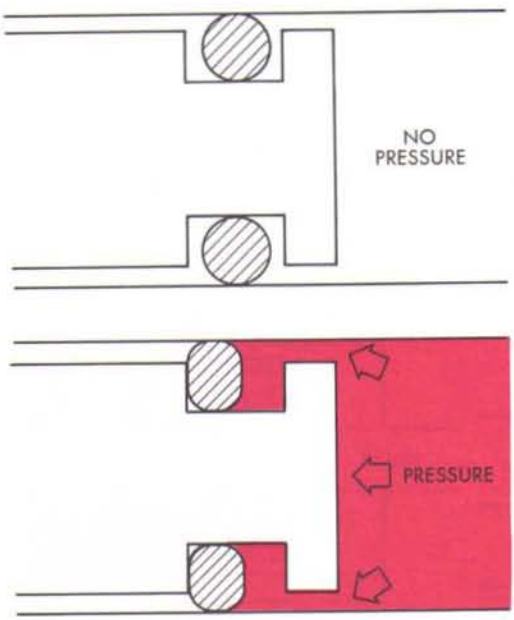


Figure 223

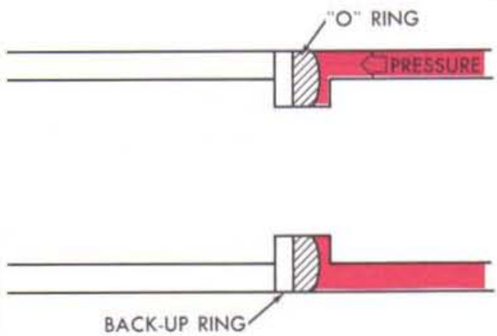


Figure 224

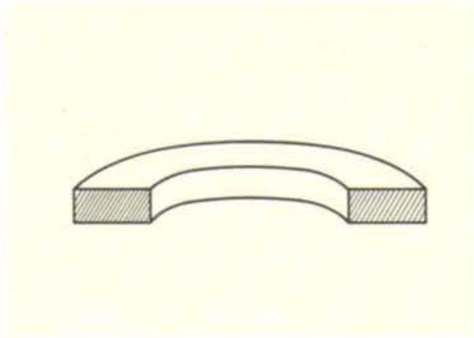


Figure 225

LATHE-CUT O-RINGS

A lathe-cut seal (Figure 225) is an O-ring that is square in cross-section rather than round. Lathe-cut rings are actually cut from extruded tubes, while round-section O-rings must be individually molded. In many static applications, round- and square-section seals are interchangeable if made from the same material.

T-RING SEAL

A T-ring seal (Figure 226) is named for the shape of its cross-section. These seals are reinforced with back-up rings on each side. They are used in reciprocating dynamic applications; particularly on cylinder pistons and around piston rods.

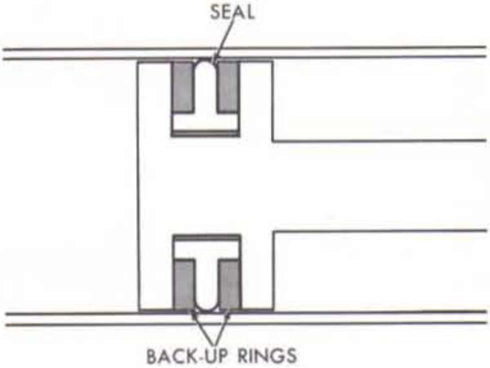


Figure 226

LIP SEAL

A lip seal (Figure 227) is a dynamic seal, used principally on rotating shafts. There are probably more of these used as shaft seals than all other kinds put together.

The sealing lip provides a positive seal against low pressure. The lip is installed toward the pressure source. Pressure against the lip "balloons" it out to aid sealing. Very high pressure, however, can get past this kind of seal, because it doesn't have the back-up support that an O-ring has.

DOUBLE LIP SEALS

We often find double lip seals used on the shafts of reversible pumps or motors. Reversing the unit can give us alternating pressure and vacuum conditions in the chamber adjacent to the seal. A double lip seal then prevents oil from getting out or air and dirt from getting in.

CUP SEALS

Cup seals (Figure 228) are in very common use on hydraulic cylinder pistons. They also are positive seals, and seal much the same way as a lip seal. Notice though that the cup seal is backed-up, so that it can handle very high pressure.

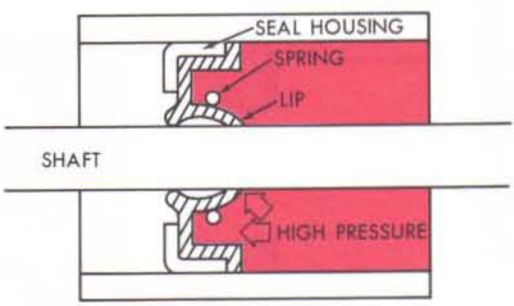


Figure 227

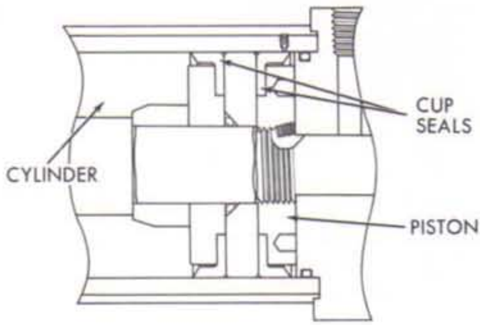


Figure 228

PISTON RINGS

Piston rings (Figure 229) are very similar to the piston rings in an automobile engine . . . and we use them the same way in hydraulic systems . . . to seal pressure at the end of a reciprocating piston. They are particularly useful where we must keep friction at a minimum in a hydraulic cylinder; they offer less resistance to movement than cup seals. Piston rings also are used in many complex components and systems, such as automatic transmissions, to seal fluid passages leading from hollow rotating shafts.

A piston ring is great for high pressure, but is not necessarily a positive seal. It

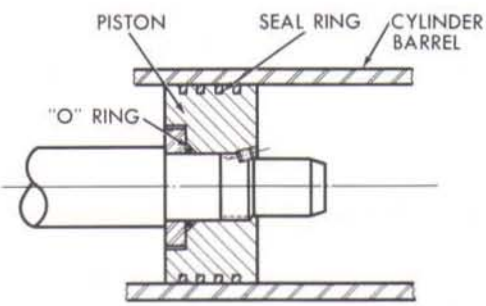


Figure 229

becomes more positive when placed side-by-side with several more of the same. However, piston rings often are designed to allow some leakage for lubrication.

PACKINGS

Packings (Figure 230) can be either static or dynamic. They have been and are used as rotating shaft seals, as reciprocating piston rod seals and as gaskets in many static applications. In static applications particularly, packings are largely being replaced by seal rings and other more effective kinds of seals. (Names stick in this industry, as we've commented before. Many of the new seals are still referred to as packings.)

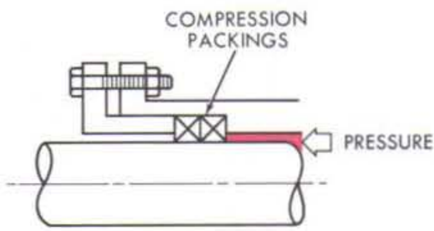


Figure 230

A packing is simply some type of twisted or woven fiber or soft metal strands "packed" between the two parts being sealed. A packing gland is a part used to support and back-up the packing.

Compression packings are usually placed in coils or layers in a bore and compressed by tightening a flanged member. Molded packings (Figure 231) are molded into a precise cross-sectional form, such as a U or V. Several such packings can be used together, with a back-up that is spring loaded to compensate for wear.

The L-shaped packing in Figure 231 (bottom) is pressure-loaded to increase its sealing ability as pressure rises.

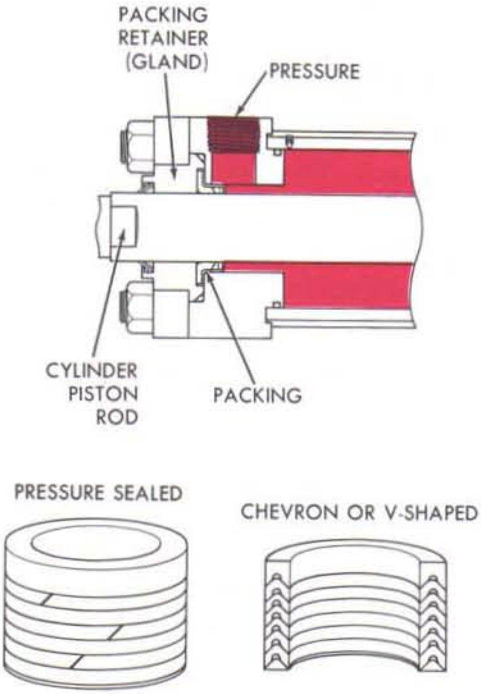


Figure 231

FACE SEALS

In a face seal (Figure 232) two smooth, flat elements run together to seal a rotating shaft. Usually one element is metallic and the other is non-metallic. The two seal members are attached respectively to the shaft and the body, so that one face is stationary and the other turns against it. A spring often loads one of the members to take up wear.

This type seal is expensive because of its multiple parts and the necessity for extremely flat surfaces on the faces. It is used primarily where there is a condition of high speed, pressure, and temperature.

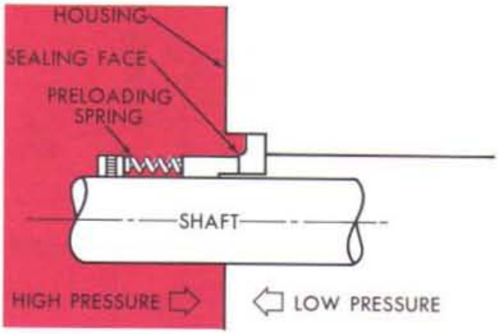


Figure 232

SEAL MATERIALS

The earliest sealing materials for hydraulic components were principally leather, cork and impregnated fibers. This may surprise some readers who have seen only modern equipment proudly displaying its O-rings, T-rings and lip seals. But these seals were made possible only through the development of synthetic rubber during World War II. Natural rubber has never been happy around petroleum. Its tendency to swell and deteriorate in an oily atmosphere made it unacceptable as a hydraulic oil sealing material.

Synthetic rubbers or elastomers are capable of many compositions . . . they can practically be tailor-made to suit the conditions under which they must seal. Most of the sealing materials you see in today's hydraulic system are made from one of these synthetic materials . . . Nitrile (Buna N), Silicone, Neoprene, Teflon and Butyl.

LEATHER SEALS

Leather, however, is still a good sealing material, and hasn't been completely replaced by the elastomers. It is low in cost and is very tough. It resists abrasion and has the ability to hold lubricating fluids in its fibers. Impregnation with synthetic rubber improves leather's sealing ability and reduces its friction. We can expect to see cup seals and lip seals made of leather for a long time yet.

Leather's disadvantages are that it tends to squeal when it's dry and that it can't stand high temperatures. Many leather seals are not recommended in conditions above 165°F (74°C). The absolute limit seems to be around 200°F (93°C).

NITRILE SEALS (BUNA-N)

Nitrile is the seal material that made the automatic transmission possible. It is most frequently used for hydraulic seals today. It is a comparatively tough material with excellent wearability . . . and economical. It is easily varied in composition to be compatible with petroleum oils and can easily be molded into the many seal shapes.

The temperature range Buna N can be used in without difficulty is from -40°F (-40°C) to 230°F (110°C). It probably has the best resistance to swelling and softening at moderately high temperatures of any sealing material.

SILICONE SEALS

Silicone was the second elastomer to become popular as a sealing material. Silicone seals have a much wider temperature range than Buna N . . . from -60°F (-51°C) to 400°F (204°C). Unlike Buna N, silicone cannot be used for reciprocating seals because it isn't tough. It tears, elongates and abrades fairly easily. Many lip-type shaft seals made from silicone are used in extreme temperature applications. There are also silicone O-rings for static applications.

Silicone has a tendency to swell since it absorbs a fair volume of oil while running hot. This is an advantage if the swelling isn't objectionable, because the seal can run dry for a longer time at start-up.

NEOPRENE

We should also mention neoprene which was one of the earliest elastomers and is also used in hydraulic seals. At very low temperature, neoprene is compatible with petroleum oil. Above 150°F (66°C), it has a habit of cooking or vulcanizing and becomes less useful.

TEFLON AND NYLON

Teflon and nylon technically are plastics rather than elastomers. The chemist calls them fluoro-elastomers, which means they combine the new decay-fighting ingredient, fluorine, with a synthetic rubber. They have exceptionally high heat resistance (witness the new teflon-coated frying pans and nylon spatulas) . . . and have prices in proportion. Both nylon and teflon are used for back-up rings and as sealing materials in special applications.

PREVENTIVE LEAKAGE

Three general factors enter into the prevention of leakage: designing to minimize the possibility, controlling operating conditions and proper installation. Let's close this final chapter by briefly exploring these factors.

ANTI-LEAKAGE DESIGNS

Straight thread, flange and gasket mounting (back mounting) are contributing greatly to decreasing external leakage . . . especially with the latest seals. Most of these connections now use O-rings which have far less tendency to leak than gaskets or tapered pipe threads. Back-mounted valves are also being sealed by O-rings rather than gaskets. The piping connections to the mounting plates are permanent . . . and a touch of teflon tape helps them considerably to avoid leaking.

Manifold mounting further decreases the possibility of leakage. A manifold is a mounting plate that has several interconnecting passages between valves; thereby eliminating much piping. If there is a place for a valve manifold on the machine, complicated circuits can be "piped" with no more external connections than a pressure line, return line and lines to the actuator.

OPERATING CONDITIONS

Controlling operating conditions can be very important to seal life. A shaft seal or piston rod seal exposed to the atmosphere will have its life shortened considerably if the atmosphere contains moisture, salt, dirt or any other abrasive contaminant. If it is possible to protect the seal from an undesirable atmosphere, it is worth the trouble.

CHEMISTRY IS INVOLVED

Chemical compatibility with the fluid we're sealing is an important consideration. Few mobile machines use fluid other than petroleum oil, but there's always the exception that requires a fire-resistant fluid. Some of these fluids will attack some elastomer seals and disintegrate them in short order. Anytime you switch to a fire-resistant fluid, it's a good idea to have the supplier check its compatibility with your seals . . . and change the seals if necessary. Your Vickers application engineer can help you in this regard.

OIL AND OIL MAINTENANCE

High quality hydraulic oil, filtration and regular oil changes seem to add significantly to seal life. Good maintenance prevents deposits of impurities and circulation of ingredients that aren't friendly to dynamic seals.

But beware of oil additives that are marketed as "do-it-yourself" projects. They may attack your seals and interfere with some of the desirable properties of the oil (see Appendix A). Additives should never be used without the approval of the equipment and oil suppliers.

TEMPERATURE

Most sealing materials have temperature limits . . . and will harden, soften or swell if the systems run above those limits for very long. They also have minimum temperatures and become too brittle to seal if they become too cold.

PRESSURE

We have noted that lip seals aren't designed for excessive pressure. High pressure chambers are always separated from shaft seals by low pressure chambers . . . and internal leakage into these low pressure chambers is always drained either internally or through a separate drain passage. If you ever forget to connect a necessary external drain passage on a pump or motor, you can be sure a leak will soon call the oversight to your attention.

Operators should always try to keep their loads within recommended limits to prevent leakage caused by excessive pressures. It's nice to get an extra half-yard of dirt in the bucket, but if you blow a seal in the cylinder, a lot of yards won't get moved while the leak is being found and repaired.

LUBRICATION

Lubrication can be critical to oil seal life in dynamic applications. The synthetics particularly don't absorb a lot of oil and so they have to get wet in a hurry before being subjected to rubbing. Leather and fiber packings are better in this respect because they absorb oil. However, manufacturers of leather seals often recommend soaking them in oil overnight before installation. No seal should ever be installed dry. Always coat them in clean hydraulic oil before installation.

INSTALLATION

Careful attention to piping and tubing installation recommendations (see Chapter 9 and Appendix B) will promote long life of external seals. Vibration or stresses resulting from incorrect installation can shake connections loose and provide you an extra supply of puddles. Care is also essential in assembly of units to avoid pinching, cocking or otherwise incorrectly installing the seals. When special tools are recommended by the manufacturer for installing seals, they should definitely be used.

APPENDIX A
Hydraulic Fluid Recommendations

The oil in a hydraulic system serves as the power transmission medium. It is also the system's lubricant and coolant. Selection of the proper oil is a requirement for satisfactory system performance and life.

TWO IMPORTANT FACTORS IN SELECTING AN OIL ARE:

1. Antiwear Additives – The oil selected must contain the necessary additives to insure high antiwear characteristics.

2. Viscosity – The oil selected must have proper viscosity to maintain an adequate lubricating film at system operating temperature.

SUITABLE TYPES OF OIL ARE:

1. Crankcase Oil meeting performance classification, letter designations, SC, SD or SE of SAE J183. Note that one oil may meet one or more of these classifications.

2. Antiwear Type Hydraulic Oil – There is no common designation for oils of this type. However, they are produced by all major oil suppliers and provide the antiwear qualities of the above designated crankcase oils.

3. Certain Other Types Of Petroleum Oils are suitable for Mobile hydraulic service if they meet the following provisions:

(A) Contain the type and content of antiwear additives found in the above designated crankcase oils or have passed pump tests similar to those used in developing the antiwear type hydraulic oils.

(B) Meet the viscosity recommendations shown in the following table.

(C) Have sufficient chemical stability for Mobile hydraulic system service.

The following table shows oil-viscosity recommendations for use with Vickers equipment in Mobile hydraulic systems:

Hydraulic System Operating Temperature Range (Min.* to Max.)	SAE Viscosity Designation
***	5W
-10°F to 130°F (-23°C to 54°C)	5W-20 5W-30
0°F to 180°F (-18°C to 83°C)	10W
0°F to 210°F (-18°C to 99°C)	10W-30**
50°F to 210°F (10°C to 99°C)	20-20W

*Ambient Start-Up Temperature
**See Paragraph on Viscosity Index
***See Paragraph on Arctic Conditions

OPERATING TEMPERATURES

The temperatures shown in the table are cold start-up to maximum operating. Suitable start-up procedures must be followed to insure adequate lubrication during system warm-up.

ARCTIC CONDITIONS

Arctic conditions represent a specialized field when extensive use is made of heating equipment before starting. If necessary, this and judicious use of the following recommendations may be used:

1. SAE 5W or SAE 5W-20 oil, in line with the viscosity guidelines shown in the table.

2. Oils especially developed for use in arctic conditions such as synthetic hydrocarbons, esters, or mixtures of the two.

3. Dilution of SAE 10W oil with maximum of 20% kerosene or low temperature diesel fuel is permissible. However, dilution of the special oils (see 2 above) should not be attempted unless the supplier and Vickers concur. The addition of the dilutant will not necessarily improve the cold cranking and may have an adverse affect on the performance of the oils in (2) above.

During cold start-up, avoid high speed operation of hydraulic system components until the system is warmed up to provide adequate lubrication.

Operating temperature should be closely monitored to avoid exceeding a temperature of 130°F (54°C) with any of these light weight or diluted oils.

OTHER FACTORS IN SELECTING AN OIL ARE:

1. Viscosity – Viscosity is the measure of fluidity. In addition to dynamic lubricating properties, oil must have sufficient body to provide adequate sealing effect between working parts of pumps, valves, cylinders and motors, but not enough to cause pump cavitation or sluggish valve action. Optimum operating viscosity of the oil should be between 16 cSt (80 SSU) and 40 cSt (180 SSU).

2. Viscosity Index – Viscosity index reflects the way viscosity changes with temperature. The smaller the viscosity change, the higher the viscosity index. The viscosity index of hydraulic system oil should not be less than 90. Multiple viscosity oils, such as SAE 10W-30, incorporate additives to improve viscosity index (polymer thickened). Oils of this type generally exhibit both temporary and permanent decrease in viscosity due to the oil shear encountered in the operating hydraulic system. The actual viscosity can, therefore, be far less in the operating hydraulic system than what is shown in normal oil data. Accordingly, when such oils are selected, it is desirable to use those with high shear stability to insure that viscosity remains within recommended limits.

3. Additives – Research has developed a number of additive agents which materially improve various characteristics of oil for hydraulic systems. These additives are selected to reduce wear, increase chemical stability, inhibit corrosion and depress the pour point. The most desirable oils for hydraulic service contain higher amounts of antiwear compounding.

4. Chemical Stability – Oxidative and thermal stability are essential characteristics of oils for Mobile Hydraulic systems. The combination of base stocks and additives should be stable during the expected lifetime of the oil when exposed to the environment of these systems.

SPECIAL REQUIREMENTS

Where special considerations indicate a need to depart from the recommended oils or operating conditions, see your Vickers representative.

CLEANLINESS

Thorough precautions should always be observed to insure that the hydraulic system is clean:

1. Clean (flush) entire system to remove paint, metal chips, welding shot, etc.

2. Filter each change of oil to prevent introduction of contaminant into the system.

3. Provide continuous oil filtration to remove sludge and products of wear and

corrosion generated during the life of the system.

4. Provide continuous protection of system from entry of airborne contamination, by sealing the system and/or by proper filtration of the air.

5. During usage, proper oil filling and servicing of filters, breathers, reservoirs, etc., cannot be over-emphasized.

6. Aeration – thorough precautions should be taken, by proper system and reservoir design, to insure that the aeration of the oil will be kept to a minimum.

APPENDIX B

Starting and Operating Practices

GOOD ASSEMBLY PRACTICES

1. The most important practice to observe in assembling hydraulic systems is cleanliness. Serious damage can result quickly from foreign material in the system.

2. Always seal all reservoir openings after cleaning the reservoir. Periodic cleaning and oil changes should be part of every maintenance schedule.

3. When a hydraulic system is opened, cap or plug all ports to keep out dirt and moisture-laden air. Keep them plugged except when repairing or connecting a unit.

4. Keep all mineral spirits in safety containers.

5. Use air hoses to clean fittings.

6. Examine pipe fittings and hoses and tube to be certain there are no scale, nicks, burrs, or dirt present. Hoses and tubes should be capped when stored.

7. Ream pipe and tubing ends to prevent swaged-over material from restricting flow or causing turbulence.

8. Never use high-pressure fittings on inlet lines, since they are smaller in inside diameter and may restrict flow.

9. No snagging or welding should be done in areas where hydraulic systems are open in any way.

10. Don't use teflon tape or pipe compound on straight threads.

11. When you use flexible couplings on pump and motor shafts:

    a. Align the coupling halves as closely as possible . . . always within .020 inch (5.08 mm).

    b. Allow 1/32 to 1/16 inch (0.79 to 1.59 mm) clearance between the coupling halves; or follow the manufacturer's recommendation for clearance.

    c. Never drive couplings onto shafts. They should always be slip fit or shrunk fit by using hot oil for heating prior to assembly.

12. Use grease liberally on splines at assembly time to increase life.

13. When double universal joints are used for coupling, there should be angularity in only one direction.

14. For assembling parts of components, a coat of clean hydraulic oil aids initial lubrication until the system is well primed. Petroleum jelly or grease are oil-soluble and can be used to stick parts together if desired.

15. Before installing a "vee" drive belt, be certain the pump or motor is built for indirect coupling. Line up both pulleys as closely as possible. Minimize overhang; that is, install the pulleys inward on the shafts as far as possible without interference with body faces. This practice will greatly increase bearing life.

INITIAL START-UP PROCEDURES FOR VANE PUMPS AND MOTORS

The earliest design of vane pumps, the "round" pumps, were designed to start under load. Tags on these pumps caution the user to start them against pressure.

The vane pumps and motors discussed in Chapter 3 are designed for no-load starting. It is important that they be started with the outlet vented to purge air from the system. Otherwise the pump might not prime and will be damaged by loss of lubrication.

Never start these vane pumps against:

- A closed valve
- A charged accumulator
- A closed loop with the hydraulic motor

Mobile directional valves are usually the by-pass type, so the pump can be started simply by centering the valve spools. But, if fluid cannot be circulated at low pressure, there should be a small valve in the pressure line, or a fitting in the line should be cracked for start-up. Leave the outlet vented until a clear stream of fluid comes out. Automatic air bleed can be accomplished by installation of an air bleed valve which opens to bleed air but closes as oil starts to flow.

MOUNTING AND START-UP PROCEDURES FOR IN-LINE PISTON PUMPS

INSTALLATION

1. Get the best possible inlet condition. Preferably, "flood" the inlet with the reservoir mounted above the pump. Don't restrict the inlet line anywhere. Make the line large enough so the fluid velocity is no more than 3 to 4 feet-per-second (0.9 to 1.2 metres per second).

2. Install a drain line to empty below the fluid level in the reservoir. The pump housing must be full of fluid at all times during operation. If the line isn't submerged, the fluid can drain out and cause damage.

3. Loop the drain line to prevent siphoning or draining out when the pump is shut down.

4. Take care in mounting that the pilot diameter fits properly in the mating section of the prime mover. Never force it in.

5. Tighten the flange mounting screws carefully to prevent misalignment. Align shafts very carefully.

6. For indirect drives, observe the side load limits specified on installation drawings.

7. If reservoir is vented type, check that the reservoir air breather is clean and large enough to keep the reservoir vented.

8. Check for absolute cleanliness in the system. Provide 25 micron filtration to keep it clean. Be sure contaminants are not introduced into system when checking oil level or when adding oil to system.

9. See that inlet and return line fittings are tight so that air isn't drawn into the system.

10. Be certain the oil is at the proper level and of the recommended quality (see Appendix A).

11. Check for the correct direction of rotation.

START-UP

1. Start a variable displacement pump at half or more of its maximum displacement.

2. Be sure to start up under no-load conditions (see Vane Pump Start-Up). Automatic air bleed valves are a definite advantage with piston pumps as with vane pumps.

3. Never operate at inlet vacuum or at a housing pressure greater than recommended on installation print.

4. Be absolutely certain the housing is full of oil.

5. Check that the pump primes within one minute of operation. If it doesn't, recheck the oil level in the reservoir.

6. Bleed the outlet (pressure) line; slowly extend and retract all cylinders; then bleed the line again. Repeat as necessary to purge all trapped air from the system. If the air is not all expelled, check the inlet lines and fittings for leaks.

7. Let the pump run at low speed as long as possible while bleeding and leak checking.

8. Do not remove a compensator control adjustment plug while the pump is in operation.

HYDROSTATIC TRANSMISSION START-UP PROCEDURE

1. When the engine starts, operate it at idle speed until the hydraulic oil has had a chance to warm. The pump case must be warmed to the touch. If there's time, allow the lines to warm up too.

2. With the engine running, just fast enough to prevent stalling, shift the control lever to move the vehicle slowly about 100 yards (91 m). Slow movement lets the oil from the motor warm before it returns to the pump.

3. Try to avoid putting the drive to work until the oil temperature is 100 degrees Fahrenheit (38^{o} C.).

This procedure shouldn't take over five or ten minutes, and will avoid damage from starting up too quickly on cold mornings.

APPENDIX C

Glossary of Mobile Hydraulics Terms

ABSOLUTE – A measure having as its zero point or base the complete absence of the entity being measured.

ABSOLUTE PRESSURE – A pressure scale with the zero point at a perfect vacuum.

ACCUMULATOR – A container in which fluid is stored under pressure as a source of fluid power.

ACTUATOR – A device for converting hydraulic energy into mechanical energy. A motor or cylinder.

AERATION – Air in the hydraulic fluid. Excessive aeration causes the fluid to appear milky and components to operate erratically because of the compressibility of the air trapped in the fluid.

AMPLITUDE OF SOUND – The loudness of a sound.

ANNULAR AREA – A ring shaped area – often refers to the net effective area of the rod side of a cylinder piston, i.e., the piston area minus the cross-sectional area of the rod.

ATMOSPHERE (ONE) – A pressure measure equal to 14.7 psi (1.01 bar).

ATMOSPHERIC PRESSURE – Pressure on all objects in the atmosphere because of the weight of the surrounding air. At sea level, about 14.7 psi absolute (1.01 bar).

BACK CONNECTED – A condition where pipe connections are on normally unexposed surfaces of hydraulic equipment. (Gasket mounted units are back connected.)

BACK PRESSURE – A pressure in series. Usually refers to pressure existing on the discharge side of a load. It adds to the pressure required to move the load.

BAFFLE – A device, usually a plate, installed in a reservoir to separate the pump inlet from return lines.

BLEED-OFF – To divert a specific controllable portion of pump delivery directly to reservoir.

BREATHER – A device which permits air to move in and out of a container or component to maintain atmospheric pressure.

BY-PASS – A secondary passage for fluid flow.

CARTRIDGE

1. The replaceable element of a fluid filter.

2. The pumping unit from a vane pump, composed of the rotor, ring, vanes and one or both side plates.

CAVITATION – A localized gaseous condition within a liquid stream which occurs where the pressure is reduced to a low value.

CHAMBER – A compartment within a hydraulic unit. May contain elements to aid in operation or control of a unit. Examples: Spring chamber, drain chamber, etc.

CHANNEL – A fluid passage, the length of which is large with respect to its cross-sectional dimension.

CHARGE (supercharge)

1. To replenish a hydraulic system above atmospheric pressure.

2. To fill an accumulator with fluid under pressure (see pre-charge pressure).

CHARGE PRESSURE – The pressure at which replenishing fluid is forced into the hydraulic system (above atmospheric pressure).

CHECK VALVE – A valve which permits flow of fluid in one direction only.

CIRCUIT – The complete path of flow in a hydraulic system including the flow-generating device.

CLOSED CENTER – The condition where pump output is not unloaded to sump in the center or neutral operating condition.

CLOSED CIRCUIT – A piping arrangement in which pump delivery, after passing through other hydraulic components, by-passes the reservoir and returns directly to pump inlet.

COMPENSATOR CONTROL – A displacement control for variable pumps and motors which alters displacement in response to pressure changes in the system as related to its adjusted pressure setting.

COMPONENT – A single hydraulic unit.

COMPRESSIBILITY – The change in volume of a unit volume of a fluid when it is subjected to a unit change in pressure.

CONTROL – A device used to regulate the function of a unit (see Hydraulic Control, Manual Control, Mechanical Control, and Compensator Control).

COOLER – A heat exchanger used to remove heat from the hydraulic fluid.

COUNTERBALANCE VALVE – A valve which maintains resistance to flow in one direction but permits free flow in the other. Usually connected to the outlet of a vertical double-acting cylinder to support weight or prevent uncontrolled falling or dropping.

CRACKING PRESSURE – The pressure at which a pressure actuated valve begins to pass fluid.

CUSHION – A device sometimes built into the ends of a hydraulic cylinder which restricts the flow of fluid at the outlet port, thereby arresting the motion of the piston rod.

CYLINDER – A device which converts fluid power into linear mechanical force and motion. It usually consists of a movable element such as a piston and piston rod, plunger rod, plunger or ram, operating within a cylindrical bore.

DELIVERY – The volume of fluid discharged by a pump in a given time, usually expressed in gallons per minute (gpm).

DE-VENT – To close the vent connection of a pressure control valve permitting the valve to function at its adjusted pressure setting.

DIFFERENTIAL CURRENT – The algebraic summation of the current in the torque motor; measured in MA (milliamperes).

DIFFERENTIAL CYLINDER – Any cylinder in which the two opposed piston areas are not equal.

DIRECTIONAL VALVE – A valve which selectively directs or prevents fluid flow to desired channels.

DISPLACEMENT – The quantity of fluid which can pass through a pump, motor or cylinder in a single revolution or stroke.

DOUBLE ACTING CYLINDER – A cylinder in which fluid force can be applied in either direction.

DRAIN – A passage in, or a line from, a hydraulic component which returns leakage fluid independently to reservoir or to a vented manifold.

EFFICIENCY – The ratio of output to input. Volumetric efficiency of a pump is the actual output in gpm divided by the theoretical or design output. The overall efficiency of a hydraulic system is the output power divided by the input power. Efficiency is usually expressed as a percent.

ELECTRO-HYDRAULIC SERVO VALVE – A directional type valve which receives a variable or controlled electrical signal and which controls or meters hydraulic flow.

ENERGY – The ability or capacity to do work. Measured in units of work.

FEEDBACK (or feedback signal) – The output signal from a feedback element.

FILTER – A device whose primary function is the retention by a porous media of insoluble contaminants from a fluid.

FLOODED – A condition where the pump inlet is charged by placing the reservoir oil level above the pump inlet port.

FLOW CONTROL VALVE – A valve which controls the rate of oil flow.

FLOW RATE – The volume, mass, or weight of a fluid passing through any conductor per unit of time.

FLUID

1. A liquid or gas.

2. A liquid that is specially compounded for use as a power-transmitting medium in a hydraulic system.

FOLLOW VALVE – A control valve which ports oil to an actuator so the resulting output motion is proportional to the input motion to the valve.

FORCE – Any push or pull measured in units of weight. In hydraulics, total force is expressed by the product P (force per unit area) and the area of the surface on which the pressure acts. $F = P \times A$.

FOUR-WAY VALVE – A directional valve having four flow paths.

FREQUENCY – The number of times an action occurs in a unit of time. Frequency is the basis of all sound. A pump or motor's basic frequency is equal to its speed in revolutions per second multiplied by the number of pumping chambers.

FRONT CONNECTED – A condition wherein piping connections are on normally exposed surfaces of hydraulic components.

FULL FLOW – In a filter, the condition where all the fluid must pass through the filter element or medium.

GAUGE PRESSURE – A pressure scale which ignores atmospheric pressure. Its zero point is 14.7 psi absolute (1.01 bar).

HEAD – The height of a column or body of fluid above a given point expressed in linear units. Head is often used to indicate gage pressure. Pressure is equal to the height times the density of the fluid.

HEAT – The form of energy that has the capacity to create warmth or to increase the temperature of a substance. Any energy that is wasted or used to overcome friction is converted to heat. Heat is measured in calories or British Thermal Units (BTU's). One BTU is the amount of heat required to raise the temperature of one pound of water one degree Fahrenheit.

HEAT EXCHANGER – A device which transfers heat through a conducting wall from one fluid to another.

HORSEPOWER – (HP) – The power required to lift 550 pounds one foot in one second or 33,000 pounds one foot in one minute. A horsepower is equal to 746 watts or to 42.4 British Thermal Units per minute.

HYDRAULIC BALANCE – A condition of equal opposed hydraulic forces acting on a part in a hydraulic component.

HYDRAULIC CONTROL – A control which is actuated by hydraulically induced forces.

HYDRAULICS – Engineering science pertaining to liquid pressure and flow.

HYDRODYNAMICS – The science dealing with liquids in motion and particularly their kinetic energies.

HYDROSTATICS – The science of liquid pressure.

KINETIC ENERGY – Energy that a substance or body has by virtue of its mass (weight) and velocity.

LAMINAR (FLOW) – A condition where the fluid particles move in continuous parallel paths. Streamline flow.

LEVERAGE – A gain in output force over input force by sacrificing the distance moved. Mechanical advantage or force multiplication.

LIFT – The height a body or column of fluid is raised; for instance from the reservoir to the pump inlet. Lift is sometimes used to express a negative pressure or vacuum. The opposite of head.

LINE – A tube, pipe or hose which acts as a conductor of hydraulic fluid.

LINEAR ACTUATOR – A device for converting hydraulic energy into linear motion – a cylinder or ram.

MANIFOLD – A fluid conductor which provides multiple connection ports.

MANUAL CONTROL – A control actuated by the operator, regardless of the means of actuation. Example: Lever or foot pedal control for directional valves.

MANUAL OVERRIDE – A means of manually actuating an automatically-controlled device.

MECHANICAL CONTROL – Any control actuated by linkages, gears, screws, cams or other mechanical elements.

METER – To regulate the amount or rate of fluid flow.

METER-IN – To regulate the amount of fluid flow into an actuator or system.

METER-OUT – To regulate the flow of the discharge fluid from an actuator or system.

MICRON (MICROMETRE) – One-millionth of a meter or about .00004 inch.

MICRON RATING – The size of the particles a filter will remove.

MOTOR – A rotary motion device which changes hydraulic energy into mechanical energy; a rotary actuator.

OPEN CENTER – A condition where pump delivery recirculates freely to sump in the center or neutral position.

ORIFICE – A restriction, the length of which is small in respect to its cross-sectional dimensions.

PASSAGE – A machined or cored fluid conducting path which lies within or passes through a component.

PILOT PRESSURE – Auxiliary pressure used to actuate or control hydraulic components.

PILOT VALVE – An auxiliary valve used to control the operation of another valve. The controlling stage of a 2-stage valve.

PISTON – A cylindrically shaped part which fits within a cylinder and transmits or receives motion by means of a connecting rod.

PLUNGER – A cylindrically shaped part which has only one diameter and is used to transmit thrust. A ram.

POPPET – That part of certain valves which prevents flow when it closes against a seat.

PORT – An internal or external terminus of a passage in a component.

POSITIVE DISPLACEMENT – A characteristic of a pump or motor which has the inlet positively sealed from the outlet so that fluid cannot recirculate in the component.

POTENTIOMETER – A control element in the servo system which measures and controls electrical potential.

POWER – Work per unit of time. Measured in horsepower (hp) or watts.

PRECHARGE PRESSURE – The pressure of compressed gas in an accumulator prior to the admission of liquid.

PRESSURE – Force per unit area; usually expressed in pounds per square inch (psi).

PRESSURE DROP – The reduction in pressure between two points in a line or passage due to the energy required to maintain flow; may be deliberate.

PRESSURE LINE – The line carrying the fluid from the pump outlet to the pressurized port of the actuator.

PRESSURE OVERRIDE – The difference between the cracking pressure of a valve and the pressure reached when the valve is passing full flow.

PRESSURE PLATE – A side plate in a vane pump or motor cartridge on the pressure port side.

PRESSURE REDUCING VALVE – A pressure control valve whose primary function is to limit the outlet pressure.

PROPORTIONAL FLOW – In a filter, the condition where part of the flow passes through the filter element in proportion to pressure drop.

PUMP – A device which converts mechanical force and motion into hydraulic fluid power.

RAM – A single-acting cylinder with a single diameter plunger rather than a piston and rod. The plunger in a ram-type cylinder.

RECIPROCATION – Back-and-forth straight line motion or oscillation.

RELIEF VALVE – A pressure operated valve which by-passes pump delivery to the reservoir, limiting system pressure to a predetermined maximum value.

REPLENISH – To add fluid to maintain a full hydraulic system.

RESERVOIR – A container for storage of liquid in a fluid power system.

RESTRICTION – A reduced cross-sectional area in a line or passage which produces a pressure drop.

RETURN LINE – A line used to carry exhaust fluid from the actuator back to sump.

REVERSING VALVE – A four-way directional valve used to reverse a double-acting cylinder or reversible motor.

ROTARY ACTUATOR – A device for converting hydraulic energy into rotary motion – a hydraulic motor.

SEQUENCE

1. The order of a series of operations or movements.

2. To divert flow to accomplish a subsequent operation or movement.

SEQUENCE VALVE – A pressure operated valve which diverts flow to a secondary actuator while holding pressure on the primary actuator at a predetermined minimum value after the primary actuator completes its travel.

SERVO MECHANISM (servo) – A mechanism subjected to the action of a controlling device which will operate as if it were directly actuated by the controlling device, but capable of supplying power output many times that of the controlling device, this power being derived from an external and independent source.

SERVO VALVE

1. A valve which controls the direction and quantity of fluid flow in proportion to an input signal.

2. A follow valve.

SIGNAL – A command or indication of a desired position or velocity.

SINGLE ACTING CYLINDER – A cylinder in which hydraulic energy can product thrust or motion in only one direction. (Can be spring or gravity returned.)

SLIP – Internal leakage of hydraulic fluid.

SPOOL – A term loosely applied to almost any moving cylindrically shaped part of a hydraulic component which moves to direct flow through the component.

STRAINER – A course filter.

STREAMLINE FLOW – (See laminar flow.)

STROKE

1. The length of travel of a piston or plunger.

2. To change the displacement of a variable displacement pump or motor.

SUB-PLATE – An auxiliary mounting for a hydraulic component providing a means of connecting piping to the component.

SUCTION LINE – The hydraulic line connecting the pump inlet port to the reservoir or sump.

SUMP – A reservoir.

SUPERCHARGE – (See charge.)

SURGE – A momentary rise of pressure in a circuit.

SWASH PLATE – A stationary canted plate in an axial type piston pump which causes the pistons to reciprocate as the cylinder barrel rotates.

TACHOMETER – (AC) (DC) – A device which generates an AC or DC signal proportional to the speed at which it is rotated and the polarity of which is dependent on the direction of rotation of the rotor.

TANK – The reservoir or sump.

THROTTLE – To permit passing of a restricted flow. May control flow rate or create a deliberate pressure drop.

TORQUE – A rotary thrust. The turning effort of a fluid motor usually expressed in inch pounds.

TORQUE CONVERTER – A rotary fluid coupling that is capable of multiplying torque.

TORQUE MOTOR – An electromagnetic device consisting of coils and the proper magnetic circuit to provide actuation of a spring-restrained armature, either rotary or translatory.

TURBULENT FLOW (TURBULENCE) – A condition where the fluid particles move in random paths rather than in continuous parallel paths.

TURBINE – A rotary device that is actuated by the impact of a moving fluid against glades or vanes.

TWO-WAY VALVE – A directional control valve with two flow paths.

UNLOAD – To release flow (usually directly to the reservoir), to prevent pressure being imposed on the system or portion of the system.

UNLOADING VALVE – A valve which by-passes flow to tank when a set pressure is maintained on its pilot port.

VACUUM – The absence of pressure. A perfect vacuum is the total absence of pressure; a partial vacuum is some condition less than atmospheric pressure. Measured in inches of Mercury (in. Hg.).

VALVE – A device which controls fluid flow direction, pressure, or flow rate.

VELOCITY

1. The speed of flow through a hydraulic line. Expressed in feet per second (fps) or inches per second (ips).

2. The speed of a rotating component measured in revolutions per minute (rpm).

VENT

1. To permit opening of a pressure control valve by opening its pilot port (vent connection) to atmospheric pressure.

2. An air breathing device on a fluid reservoir.

VISCOSITY – A measure of the internal friction or the resistance of a fluid to flow.

VISCOSITY INDEX – A measure of the viscosity-temperature characteristics of a fluid as referred to that of two arbitrary reference fluids.

VOLUME

1. The size of a space or chamber in cubic units.

2. Loosely applied to the output of a pump in gallons per minute (gpm).

WOBBLE PLATE – A rotating canted plate in an axial type piston pump which pushes the pistons into their bores as it "wobbles".

WORK – Exerting a force through a definite distance. Work is measured in units of force multiplied by distance; for example, pound-feet.

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