Written by: Bud Trinkel, Certified Fluid Power Engineer
Edited by Mary Gannon and Richard Schneider, Hydraulics & Pneumatics magazine.

PREFACE

Fluid Power Basics starts with background information about simple air and hydraulic circuits, principles of fluid power operation and physical laws governing fluid power. Subsequent chapters cover different types of hydraulic fluids, fluid rating, operating parameters, and how to apply them. Next, a discussion on plumbing of fluid power systems covers tubing, pipe, and hose installations. A short section on vacuum and its applications is followed by basic circuit information. Coverage then shifts to discussing different components that make up a complete hydraulic or pneumatic system: reservoirs, filters, pumps, flow meters, gauges, and relief valves.

A detailed discussion of directional control valves covers check and prefill valves; decompression circuits; sliding plate, spool and poppet designs; in-line and sub-plate mounted valves, as well as screw-in and slip-in cartridge valves. One chapter is dedicated to an explanation of proportional and servovalves.

Subsequent chapters cover all types of flow controls and their use in a circuit. Next are pressure controls except relief and unloading valves. This chapter includes sequence, counterbalance, and reducing valves. Shuttle valves, quick exhaust valves, and other special-purpose valves are explained. There is a chapter on accumulators that shows and explains how the different types work and common applications.

The book also covers all types of actuators, including cylinders, rams, motors, and rotary actuators. Application of these components in different circuits gives a general overall view of how they are used.

Circuit diagrams are intended to show the function of the components and do not necessarily show all the components to make a safe and reliable system. Drawing practices and symbols according to ISO standards have been used when possible.

FOREWORD

I began my fluid power profession as a salesman of cylinders and valves. I had used the same cylinders as a designer of plastic injection molds, diecast dies, and related tooling. In my ten years in design, I never had to think about what it took to make the cylinders operate. The people who built the fixtures or ran the tooling took care of getting the valves and hooking them to the cylinders. Not until I started my sales career did I realize the tool shop depended on fluid power salesmen to tell them what they needed. It may surprise you to know that salesman design more than 90% of all fluid power circuits in the United States.

The first two cylinder air circuits I designed took several hours and some of these didn't even work. Fortunately, the company I started with was committed to training, so design time decreased and working circuits increased. In a few years air logic controls and a full hydraulic pump and valve line brought more challenges and more knowledge about fluid power.

I quickly found teachers of fluid power were in demand at schools with adult evening classes. Many people worked with fluid power but had little or no training in how it worked. I can testify to the fact that teachers always learn more than their students. I often think about some of the early classes and questions I could not answer.

This material is the text for a basic fluid power course I have been teaching for several years. It is dedicated to people who bought fluid power products even if the circuit didn't work the first time. It is also dedicated to many students who asked questions I'd never thought of, and to students who came up with ideas I might have passed over because I was sure it could not be done.

Bud Trinkel, CFPE
Fluid Power Consultant

****Editor's Note: The fluid power industry lost an icon in 2009 when Bud Trinkel passed away, but we are glad to keep his work alive with his technical training manuals published as eBooks. Please see his obituary below, or read a tribute to him from H&P Editor Alan Hitchcox here.*
Edgar W. "Bud" Trinkel Jr., of Evansville, Ind., died suddenly on August 12. He was best known in the fluid power industry for his years spent working as a hydraulic pneumatic specialist in sales and later for starting his own consulting business, Hydra-Pneu Consulting. He wrote several books on fluid power, including Fluid Power Basics, Fluid Power Circuits Explained, and others to aid his training endeavors. They became so well-accepted that he began producing them as stand-alone books. As president of Hydra-Pneu, Trinkel designed fluid power circuits, provided training, and performed troubleshooting for industrial clients. Prior to founding Hydra-Pneu Consulting in 1984 as a part-time fluid power consulting firm — which became a full-time endeavor in 1988 — Trinkel worked as a technical sales and service representative for a fluid power distributor. Prior to that he served as a sales and service representative for Miller Fluid Power for 14 years. Earlier in his career he worked as an industrial designer in the plastics industry. A veteran of the United States Air Force, he is survived by his wife of 56 years, Sharon; son, Charles (Michelle); daughter, Julie Woodson (Russ); and six grandchildren

Any media (liquid or gas) that flows naturally or can be forced to flow could be used to transmit energy in a fluid power system. The earliest fluid used was water hence the name hydraulics was applied to systems using liquids. In modern terminology, hydraulics implies a circuit using mineral oil. ***Figure 1-1*** *shows a basic power unit for a hydraulic system. (Note that water is making something of a comeback in the late '90s; and some fluid power systems today even operate on seawater.) The other common fluid in fluid power circuits is compressed air. As indicated in* ***Figure 1-2****, atmospheric air -- compressed 7 to 10 times -- is readily available and flows easily through pipes, tubes, or hoses to transmit energy to do work. Other gasses, such as nitrogen or argon, could be used but they are expensive to produce and process.*

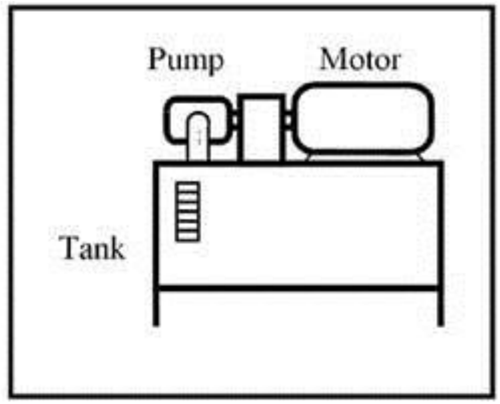


Fig. 1-1: Basic hydraulic power unit.

Of the three main methods of transmitting energy mechanical, electrical, and fluid fluid power is least understood by industry in general. In most plants there are few persons with direct responsibility for fluid power circuit design or maintenance. Often, general mechanics maintain fluid power circuits that originally were designed by a fluid-power-distributor salesperson. In most facilities, the responsibility for fluid power systems is part of the mechanical engineers' job description. The problem is that mechanical engineers normally receive little if any fluid power training at college, so they are ill equipped to carry out this duty. With a modest amount of fluid power training and more than enough work to handle, the engineer often depends on a fluid power distributor's expertise. To get an order, the distributor salesperson is happy to design the circuit and often assists in installation and startup. This arrangement works reasonably well, but as other technologies advance, fluid power is being turned down on many machine functions. There is always a tendency to use the equipment most understood by those involved.

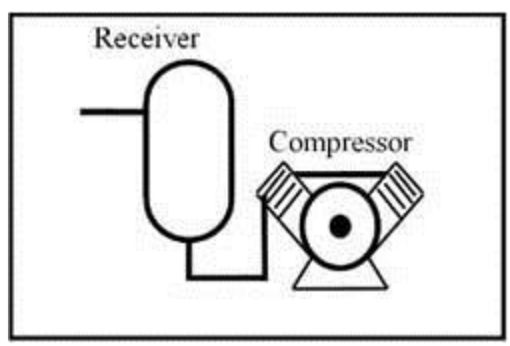


Fig. 1-2: Basic pneumatic power arrangement.

Fluid power cylinders and motors are compact and have high energy potential. They fit in small spaces and do not clutter the machine. These devices can be stalled for extended time periods, are instantly reversible, have infinitely variable speed, and often replace mechanical linkages at a much lower cost. With good circuit design, the power source, valves, and actuators will run with little maintenance for extended times. The main disadvantages are lack of understanding of the equipment and poor circuit design, which can result in overheating and leaks. Overheating occurs when the machine uses less energy than the power unit provides. (Overheating usually is easy to design out of a circuit.)

Controlling leaks is a matter of using straight-thread O-ring fittings to make tubing connections or hose and SAE flange fittings with larger pipe sizes. Designing the circuit for minimal shock and cool operation also reduces leaks.

A general rule to use in choosing between hydraulics or pneumatics for cylinders is: if the specified force requires an air cylinder bore of 4 or 5 in. or larger, choose hydraulics. Most pneumatic circuits are under 3 hp because the efficiency of air compression is low. A system that requires 10 hp for hydraulics would use approximately 30 to 50 air-compressor horsepower. Air circuits are less expensive to build because a separate prime mover is not required, but operating costs are much higher and can quickly offset low component expenses. Situations where a 20-in. bore air cylinder could be economical would be if it cycled only a few times a day or was used to hold tension and never cycled. Both air and hydraulic circuits are capable of operating in hazardous areas when used with air logic controls or explosion-proof electric controls. With certain precautions, cylinders and motors of both types can operate in high-humidity atmospheres . . . or even under water.

When using fluid power around food or medical supplies, it is best to pipe the air exhausts outside the clean area and to use a vegetable-based fluid for hydraulic circuits.

Some applications need the rigidity of liquids so it might seem necessary to use hydraulics in these cases even with low power needs. For these systems, use a combination of air for the power source and oil as the working fluid to cut cost and still have lunge-free control with options for accurate stopping and holding as well. Air-oil tank systems, tandem cylinder systems, cylinders with integral controls, and intensifiers are a few of the available components.

The reason fluids can transmit energy when contained is best stated by a man from the 17th century named Blaise Pascal. Pascal's Law is one of the basic laws of fluid power. This law says: ***Pressure in a confined body of fluid acts equally in all directions and at right angles to the containing surfaces.*** *Another way of saying this is: If I poke a hole in a pressurized container or line, I will get PSO. PSO stands for pressure squirting out and puncturing a pressurized liquid line will get you wet.* ***Figure 1-3*** *shows how this law works in a cylinder application. Oil from a pump flows into a cylinder that is lifting a load. The resistance of the load causes pressure to build inside the cylinder until the load starts moving. While the load is in motion, pressure in the entire circuit stays nearly constant. The pressurized oil is trying to get out of the pump, pipe, and cylinder, but these mechanisms are strong enough to contain the fluid. When pressure against the piston area becomes high enough to overcome the load resistance, the oil forces the load to move upward. Understanding Pascal's Law makes it easy to see how all hydraulic and pneumatic circuits function.*

Notice two important things in this example. First, the pump did not make pressure; it only produced flow. Pumps never make pressure. They only give flow. Resistance to pump flow causes pressure. This is one of the basic principles of fluid power that is of prime importance to troubleshooting hydraulic circuits. Suppose a machine with the pump running shows almost 0 psi on its pressure gauge. Does this mean the pump is bad? Without a flow meter at the pump outlet, mechanics might change the pump, because many of them think pumps make pressure. The problem with this circuit could simply be an open valve that allows all pump flow to go directly to tank. Because the pump outlet flow sees no resistance, a pressure gauge shows little or no pressure. With a flow meter installed, it would be obvious that the pump was all right and other causes such as an open path to tank must be found and corrected.

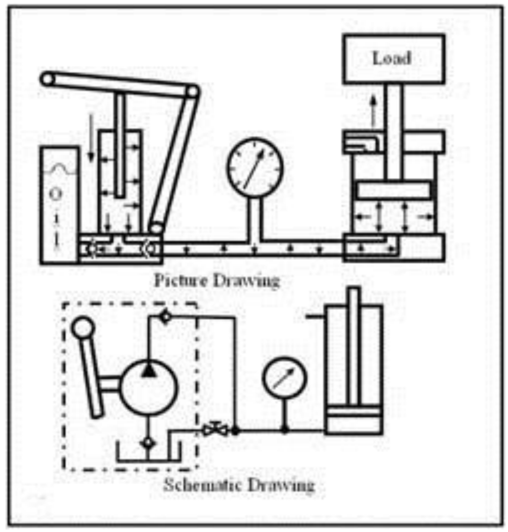


Fig. 1-3: How Pascals Law affects a cylinder

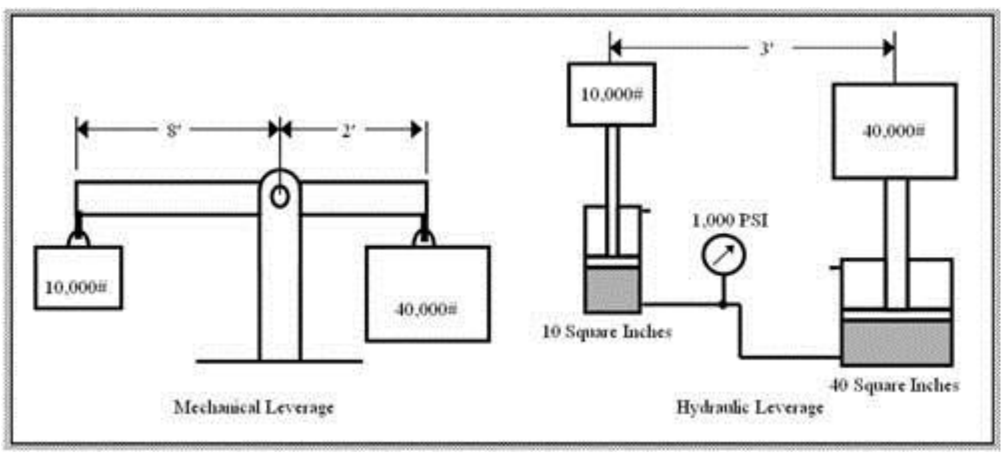


Fig. 1-4: Comparison of mechanical and hydraulic leverage

Another area that shows the effect of Pascal's law is a comparison of hydraulic and mechanical leverage. ***Figure 1-4*** *shows how both of these systems work. In either case, a large force is offset by a much smaller force due to the difference in lever-arm length or piston area.*

Notice that hydraulic leverage is not restricted to a certain distance, height, or physical location like mechanical leverage is. This is a decided advantage for many mechanisms because most designs using fluid power take less space and are not restricted by position considerations. A cylinder, rotary actuator, or fluid motor with almost limitless force or torque can directly push or rotate the machine member. These actions only require flow lines to and from the actuator *and feedback devices to indicate position. The main advantage of linkage actuation is precision positioning and the ability to control without feedback.*

At first look, it may appear that mechanical or hydraulic leverage is capable of saving energy. For example: 40,000 lb is held in place by 10,000 lb in ***Figure 1-4****. However, notice that the ratio of the lever arms and the piston areas is 4:1. This means by adding extra force say to the 10,000-lb side, it lowers and the 40,000-lb side rises. When the 10,000-lb weight moves down a distance of 10 in., the 40,000-lb weight only moves up 2.5 in.*

Work is the measure of a force traversing through a distance. ***(Work = Force X Distance.)****. Work usually is expressed in foot-pounds and, as the formula states, it is the product of force in pounds times distance in feet. When a cylinder lifts a 20,000-lb load a distance of 10 ft, the cylinder performs 200,000 ft-lb of work. This action could happen in three seconds, three minutes, or three hours without changing the amount of work.*

When work is done in a certain time, it is called power. ***{Power = (Force X Distance) / Time.}*** *A common measure of power is horsepower - a term taken from early days when most persons could relate to a horse's strength. This allowed the average person to evaluate to new means of power, such as the steam engine. Power is the rate of doing work. One horsepower is defined as the weight in pounds (force) a horse could lift one foot (distance) in one second (time). For the average horse this turned out to be 550 lbs. one foot in one second. Changing the time to 60 seconds (one minute), it is normally stated as 33,000 ft-lb per minute.*

No consideration for compressibility is necessary in most hydraulic circuits because oil can only be compressed a very small amount. Normally, liquids are considered to be incompressible, but almost all hydraulic systems have some air trapped in them. The air bubbles are so small even persons with good eyesight cannot see them, but these bubbles allow for compressibility of approximately 0.5% per 1000 psi. Applications where this small amount of compressibility does have an adverse effect include: single-stroke air-oil intensifiers; systems that operate at very high cycle rates; servo systems that maintain close-tolerance positioning or pressures; and circuits that contain large volumes of fluid. In this book, when presenting circuits where compressibility is a factor, it will be pointed out along with ways to reduce or allow for it.

Another situation that makes it appear there is more compressibility than stated previously is if pipes, hoses, and cylinder tubes expand when pressurized. This requires more fluid volume to build pressure and perform the desired work. In addition, when cylinders push against a load, the machine members resisting this force may stretch, again making it necessary for more fluid to enter the cylinder before the cycle can finish.

As anyone knows, gasses are very compressible. Some applications use this feature. In most fluid power circuits, compressibility is not advantageous; in many, it is a disadvantage. This means it is best to eliminate any trapped air in a hydraulic circuit to allow faster cycle times and to make the system more rigid.

Boyle's Law

Boyle's Law for gasses states: ***It is the principle that, for relatively low pressures, the absolute pressure of an ideal gas kept at constant temperature varies inversely with the volume of the gas.*** *In down-home language this means if a ten cubic foot volume of atmospheric air is squeezed into a one cubic foot container, pressure increases ten times. (10 X 14.7 psia = 147 psia.) Notice that pressure is stated as psia.*

Normally, pressure gauges read in psi (with no additional letter). Commonly called gauge pressure, psi disregards the earth's atmospheric pressure of 14.7 psia, because it has no effect either negative or positive on a fluid power circuit. The a on the end of psia stands for absolute, and would be shown on a gauge with a pointer that never goes to zero unless it is measuring vacuum. Another type of gauge that shows both negative and positive pressures would have a pointer with an inches-of-mercury (in. Hg) scale below zero and a psig scale above zero. Both of these gauges could read pressure or vacuum. (They are always found in a refrigeration repairperson's tool kit. Refrigeration units have both vacuum and pressure in different sections of the system at the same time.) ***Figure 1-5*** *pictures a typical psig gauge and one type of psia gauge.*

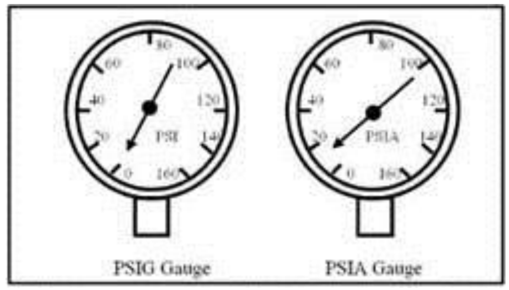


Fig. 1-5: Measurement of gauge and absolute pressure

In the example above, when ten cubic feet of air was squeezed into a one cubic-foot space, both pressures were given in psia. To see what gauge pressure (psig) would be, subtract one atmosphere from the 147-psia reading. (147 psia 14.7 psia = 132.3 psig.) To calculate the amount of compression of air in a system, always use absolute pressure, or psia, not psig. For example: the cylinder in ***Figure 1-6*** *contains eight cubic feet of air at 70 psig. To what will pressure increase when an external force pushes the piston back until the space behind the piston is two cubic foot? It is obvious the pressure will rise four times. At first it might look easy to take 70 psig X 4 = 280 psig, but this answer is wrong. For the correct answer, gauge pressure must be changed to absolute pressure. In this case by adding one atmosphere to the 70-psig reading. (70 psig + 14.7 psia = 84.7 psia.) Now multiply the 84.7-psia pressure by 4 to see what the absolute pressure is when the cylinder stops at one cubic foot volume. (84.7 X 4 = 338.8 psia.) Finally, to return to gauge pressure, subtract one atmosphere from the absolute pressure. (338.8 psia 14.7 psia = 324.1 psig.) Notice that the correct pressure is 44.1 psig higher than when gauge pressure is the multiplier.*

Temperature was not considered in both preceding cases, but notice that the law says kept at constant temperature. Compressing a gas always increases its temperature because the heat in the larger volume is now packed into a smaller space. The next law says that increasing temperature increases pressure if the gas cannot expand. This means the pressures given are measured after the gas temperature returns to what it was originally.

Gauges today read in psi and bar. Bar is a metric or SI unit for pressure and is equal to approximately the barometer reading or one atmosphere. One atmosphere is actually 14.696 psi but the SI unit for bar is 14.5 psi.

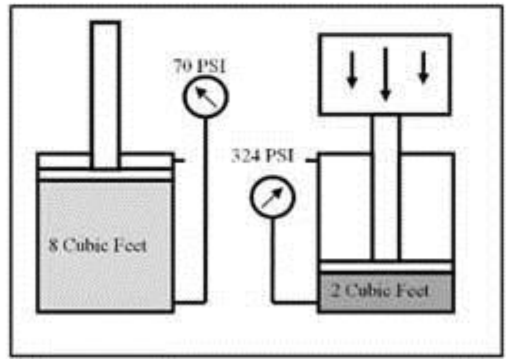


Fig. 1-6: Pressure change as air is compressed

Charles' Law

Heating a gas or liquid causes it to expand. Continuing to heat a liquid will result in it changing to the gaseous state and perhaps spontaneous combustion. If the gas or liquid cannot expand because it is confined, pressure in the contained area increases. This is stated in Charles' Law as: The volume of a fixed mass of gas varies directly with absolute temperature, provided the pressure remains constant. Because fluid power systems have some areas in which fluid is trapped, it is possible that heating this confined fluid could result in part damage or an explosion. If a circuit must operate in a hot atmosphere, provide over pressure protection such as a relief valve or a heat- or pressure-sensitive rupture device. Never heat or weld on any fluid power components without proper preparation of the unit.

Static head pressure

The weight of a fluid in a container exerts pressure on the containing vessel's sides and bottom. This is called static head pressure. It is caused by earth's gravitational pull. A good example of head pressure is a community water system. ***Figure 1-7*** *shows a water tower with a topmost water level of 80 feet. A cubic inch of water weighs 0.0361 pounds. Therefore a one square-inch column of water will exert a force of 0.0361 psi for every inch of elevation. This works out to .433 psi per foot of elevation. For the water tower in* ***Figure 1-7****, the pressure at the base would be: 80 ft X 0.433 psi/ft = 34.6 psi. This pressure is always available, even when no pumps are running. Of course, if the water level drops, static head pressure also will drop.*

The specific gravity of hydraulic oil is approximately 0.9, so multiplying water's 0.433 psi per foot by 0.9 shows oil exerts 0.39 psi per foot of elevation. Usually this fraction is rounded to 0.4 for simplicity. If the water tower were filled to 80 ft with oil, it would exert a pressure of 32 psi at ground level. Other fluids would develop a higher or lower static pressure according to their specific gravities.

This pressure is only realized at ground level at the tower. Outlets at other levels would be higher or lower according to their distance below the fluid surface.

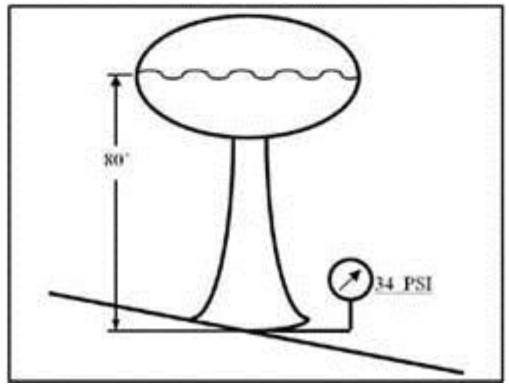


Fig. 1-7: Pressure measurement for water tower

Tanks seen on most water towers simply store volume. Pressure does not drop rapidly or require frequent pump starts to maintain the fluid level. The size or shape of the tank does not affect pressure at the base. Pressure at the base of a straight 80-ft pipe would be the same, but useful volume before pressure drop would change drastically. Always remember: it is not the physical size of a body of fluid that determines pressure but how deep it is.

Head pressure can have an adverse effect on a hydraulic system because many pumps are installed above the fluid level. This means the pump must first create enough vacuum to raise the fluid and then create even higher vacuum to accelerate and move it. Therefore there is a limit to how far a pump can be located above the oil level. Most pumps specify a maximum suction pressure of 3 psi. At 4- to 5-psi suction pressure, pumps start to cavitate . . . causing internal damage. At 6- to 7-psi vacuum, cavitation damage is severe and noise levels increase noticeably. (The effects of cavitation are covered fully in Chapter 8, Fluid power pumps and accessory items.) Axial- or in-line-piston pumps are especially vulnerable to high inlet vacuum damage and should be set up below the fluid level to produce a positive head pressure.

Many modern hydraulic systems place the pump next to the reservoir so the fluid level is always above the pump inlet. With this type of installation the pump always has oil at startup and has a positive head pressure at its inlet. A better arrangement puts the tank above the pump to take advantage of even greater head pressure. Everything possible should be done to keep pressure drop low in the pump inlet line because the highest possible pressure drop allowable is one atmosphere (14.7 psi at sea level).

*The earth's atmosphere the air we breathe exerts a force of 14.7 psi at sea level on an average day. This pressure covers the whole earth's surface, but at elevations higher than sea level, it is reduced by approximately 0.5 psi per 1000 feet. This pressure of earth's atmosphere is the source of the power of vacuum. The highest possible vacuum reading at any location is the weight of the air above it at that time. A reading of maximum vacuum available is given during the local weather forecast as the barometer reading. Divide the barometer reading by two to get the approximate atmospheric pressure in psi. This force could be directly measured if it were possible to isolate a one square-inch column of air one atmosphere tall at a sea level location. Because this is not possible, the method used to measure vacuum is demonstrated in **Figure 1-8**.*

Submerge a clear tube with one closed end in a container of mercury and allow it to fill completely. (The tube must be more than 30-in. long for this example to work when mercury is the liquid.) After the mercury displaces all the air in the tube, carefully raise the tube's closed end, keeping the open end submerged so the mercury can't run out and be replaced by air. When the tube is positioned vertically, the liquid mercury level will lower to give the atmospheric pressure reading in inches of mercury (29.92-in. H_g *at sea level). The mercury level will fluctuate from this point as high and low-pressure weather systems move past. If the tube had been 100-in. tall, the mercury level would still have dropped to whatever the atmospheric pressure was at its location. The reason the mercury does not all flow out is that atmospheric pressure holds it in.*

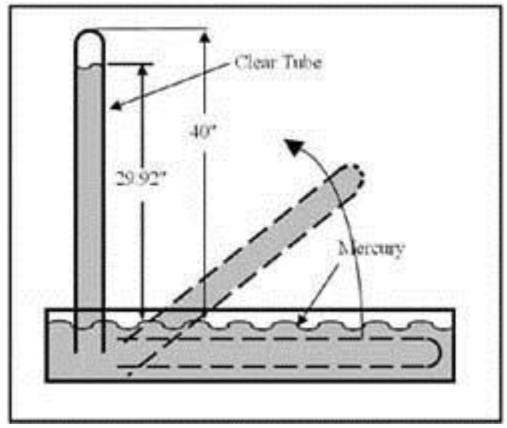


Fig. 1-8: Vacuum measurement with mercury

This barometer could have been built using another liquid but the tube would have to be longer because most other liquids have a much lower specific gravity than mercury's 13.546. Water, with a specific gravity of 1.0, would require a closed-end tube at least 33.8 ft long, while oil, with a specific gravity of approximately 0.9, would have to be even longer.

Vacuum pumps can be similar in design to air compressors. There are reciprocating-piston, diaphragm, rotary-screw, and lobed-rotor designs. (See air compressor types in Chapter 8, Fluid power pumps and accessory items.) Imagine hooking the inlet of an air compressor to a receiver tank and leaving the outlet open to atmosphere. As the pump runs, it evacuates air from the receiver and causes a negative pressure in it.

Vacuum pumps are an added expense and normally are only found in facilities that use a constant supply of negative pressure to operate machines or make products.

Vacuum generators that use plant compressed air as a power source are also available. These components have no moving parts but use plant air flowing through a venturi to produce a small supply of negative pressure. ***Figure 1-9*** *shows a simplified cutaway view of a venturi-type vacuum generator. The device consists of body A with compressed-air inlet B that passes air flow through venturi nozzle C. The air exhausts at a higher velocity to atmosphere through orifice D. As air at increasing velocity flows past opening E near the venturi nozzle, it creates a negative pressure and draws in atmospheric air through port F. Port F can connect to any external device that needs a vacuum source. A vacuum gauge at port F shows negative pressure when compressed air is supplied to port B.*

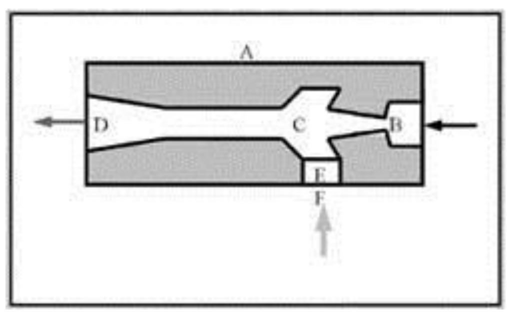


Fig. 1-9: Cross-sectional view of venturi vacuum generator

Vacuum generators are inexpensive, but can be costly to operate. For every 4 cfm of air supply required to power them, they use approximately one compressor horsepower. For this reason, venturi-type vacuum generators usually are installed with a control valve to turn them on only when needed.

Vacuum is limited to one atmosphere maximum at any location, and standard vacuum pumps only reach about 85% (approximately 12 psi) of this on average. As a result, vacuum is not powerful enough to do much work unless it acts on a large area.

Many industrial vacuum applications have to do with handling parts. Large-area suction cups can lift a large heavy part with ease, as illustrated in ***Figure 1-10****. When the lift rises, negative pressure (vacuum) inside the suction cups causes atmospheric pressure on the opposite side of the part to push it up.*

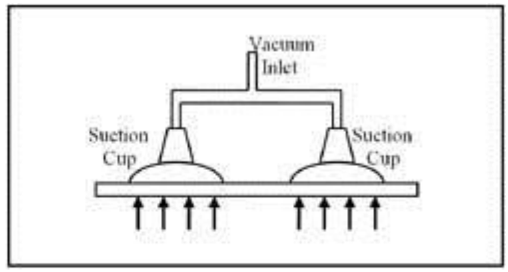


Fig. 1-10: Simplified representation of lifting with vacuum

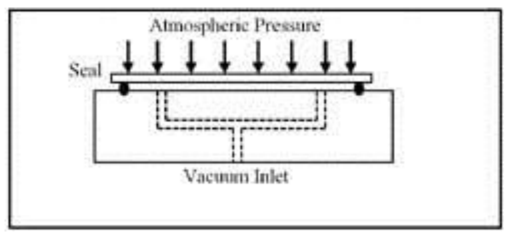


Fig. 1-11: Simplified representation of work holding with vacuum

Industries such as glass and wood manufacturing use vacuum to hold work pieces during machining or other operations, as shown in ***Figure 1-11****. The pieces are held firmly in place as the negative pressure under them causes atmospheric pressure to push against them. A resilient seal laid in a groove in the fixture keeps atmospheric air from entering the cavity beneath the part. This groove can be cut to match the contour of the part. In machining operations, the seals can isolate interior cutouts, allowing them to be removed while firmly holding the rest of the piece.*

*Heated plastic sheet can be vacuum-formed to make some products at a much lower cost than other types of plastic forming, as suggested in **Figure 1-12**. Forming heated plastic sheet in a cavity or over a shape is quick and positive. When atmospheric pressure tries to fill the negative-pressure area under the softened sheet, the sheet is forced into the desired shape. Large parts such as pickup-truck bed liners are formed by this method*

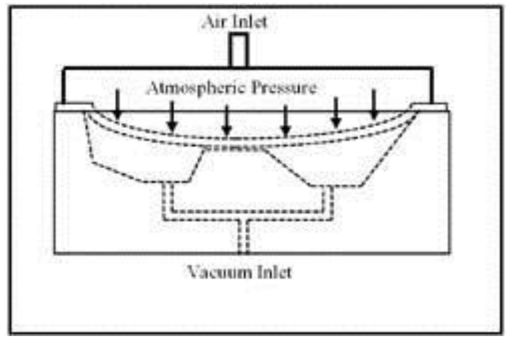


Fig. 1-12: Simplified representation of plastic-sheet forming with vacuum

For long service life, safety reasons, and reliable operation of hydraulic circuits, it is very important to use the correct fluid for the application. The most common fluid is based on mineral oil, but some systems require fire resistance because of their proximity to a heat source or other fire hazard. (Water is also making its return to some hydraulic systems because it is inexpensive, fireproof, and does not harm the environment.

Transmit energy.

*The main purpose of the fluid in any system is to **transmit energy**. Electric, internal combustion, steam powered, or other prime movers drive a pump that sends oil through lines to valves that control actuators. The fluid in these lines must transmit the prime movers energy to the actuator so it can perform work. The fluid must flow easily to reduce power losses and make the circuit respond quickly.*

Lubricate.

*In most hydraulic systems, the fluid must have good **lubrication** qualities. Pumps, motors, and cylinders need ample lubrication to make them efficient and extend their service life. Mineral oils with anti-wear additives work well and are available from most suppliers. Some fluids may need special considerations in component design to overcome their lack of lubricity.*

Seal.

*Fluid thickness can be important also because one of its requirements is for **sealing**. Almost all pumps and many valves have metal to metal sealing fits that have minimal clearance but can leak at elevated pressures. Thin watery fluid can flow through these clearances, reducing efficiency and eroding the mating surfaces. Thicker fluids keep leakage to a minimum and efficiency high.*

*There are several areas that apply to specifying fluids for a hydraulic circuit. **Viscosity** is the measure of the fluids thickness. Hydraulic oils thickness is specified by a SUS or SSU designation, similar to the SAE designation used for automotive fluids. SUS stands for Saybolt Universal Seconds (or as some put it, Saybolt Seconds Universal). It is a measuring system set up by a man named Saybolt. Simply stated, the system takes a sample of fluid, heats it to 100° F, and them measures how much fluid passes through a specific orifice in a certain number of seconds.*

Viscosity is most important as it applies to pumps. Most manufacturers specify viscosity limits for their pumps and it is best to stay within the limits they suggest. The prime reason for specifying a maximum viscosity is that pressure drop in the pump suction line typically is low and if the oil is too thick, the pump will be damaged due to cavitation. A pump can move fluid of any viscosity if the inlet is amply supplied. On the other end, if fluids are too thin, pump bypass wastes energy and generates extra heat. All other components in the circuit could operate on any viscosity fluid because they only use what is fed to them. However, thicker fluids waste energy because they are hard to move. Thin fluids waste energy because they allow too much bypass.

***Viscosity index (or VI)** is a measure of viscosity change from one temperature to another. It is common knowledge that heating any oil makes it thinner. A normal industrial hydraulic circuit runs at temperatures between 100° and 130° F. Cold starts could be as low as 40° to 50° F.*

Using an oil with a low VI number might start well but wind up with excessive leakage and wear or cause cavitation damage at startup and run well at temperature. Most industrial hydraulic oils run in the 90- to 105-VI range and are satisfactory for most applications.

Pour point *is the lowest temperature at which a fluid still flows. It should be at least lower than the lowest temperature to which the system will be exposed so the pump can always have some lubrication. Consider installing a reservoir heater and a circulation loop on circuits that start or operate below 60° F.*

Refined mineral oil does not have enough lubricating qualities to meet the needs of modern day hydraulic systems. Several ***lubricity additives*** *to enhance that property are added to mineral oil as a specific manufacturers package. These additives are formulated to work together and should not be mixed with others additives because some components may be incompatible.*

Refined mineral oil also is very much affected by temperature change. In its raw state it not only has low lubricity but also would thin out noticeably with only a small increase in temperature. ***Viscosity modifiers*** *enhance the oils ability to remain at a workable viscosity through a broad temperature range.*

There are several causes of hydraulic oil oxidation. These include contamination, air, and heat. The interaction of these outside influences cause sludge and acids to form. Oxidation inhibitors slow or stop the fluids degradation and allow it to perform as intended.

Wear inhibitors *are additives that bond with metal parts inside a hydraulic system and leave a thin film that reduces metal-to-metal contact. When these additives are working, they extend part life by reducing wear.*

In most hydraulic systems, fast and turbulent fluid flow can lead to foaming. ***Anti-foaming*** *agents make the fluid less likely to form bubbles and allow those that do form to dissipate more rapidly.*

Moisture in the air can condense in a hydraulic reservoir and mix with the fluid. Rust inhibitors negate the effect of this unwanted water and protect the surfaces of the systems metal components. All of these additives are necessary to extend system life and improve reliability.

Overheating *the fluid can counteract the additives and decrease system efficiency.* ***Overheating*** *also thins the oil and reduces efficiency because of internal bypassing. Clearances in pump and valve spools let fluid pass as pressure increases, causing more heating until the fluid breaks down. External leaks through fittings and seals also increase as fluid temperatures rise. Another problem caused by* ***overheating*** *is a breakdown of some seal materials. Most rubber compounds are cured by controlled heat over a specific period of time. Continued heating inside the hydraulic system over long periods keeps the curing process going until the seals lose their resiliency and their ability to seal. It is best if hydraulic oil never exceeds 130° F for any extended period. Installing heat exchangers is the most common cure for* ***overheating*** *but designing heat out of a circuit is the better way.*

Cold oil is not a problem as far as the oil is concerned but cooling does increase viscosity. When viscosity gets too high, it can cause a pump to cavitate and damage itself internally. Thermostatically controlled reservoir heaters easily eliminate this problem in most cases.

Fire-resistant fluids
Certain applications must operate near a heat source with elevated temperatures or even open flames or electrical heating units. Mineral oil is very flammable. It not only catches fire easily but will continue to burn even after removing the heat source. This fire hazard situation can be eliminated by several different choices of fluids. These fluids are not fireproof, only fire-resistant, which means they will burn if heated past a certain temperature but they will not continue to burn after removing them from the heat source.

*Generally, the **fire-resistant fluids** do not have the same specifications as mineral oil-based fluids. Pumps often must be down rated because the fluids lubricity or specific gravity is different and would shorten the pumps service life drastically at elevated pressures or high rotary speeds. Some **fire-resistant fluids** are not compatible with standard seal materials so seals must be changed. Always check with the pump manufacturer and fluid supplier before using or changing to a **fire-resistant fluid**.*

Water
*Originally, hydraulic circuits used **water** to transmit energy (hence the word hydraulics). The main problem with water-filled circuits was either low-pressure operation or very expensive pumps and valves to operate with this low viscosity fluid above 500 to 600 psi. When huge oil deposits were discovered, mineral oil replaced water because of its additional benefits. Water made a brief comeback during an oil shortage crisis but quickly succumbed when oil flowed freely again.*

In the late 90s, water again made inroads into oil-hydraulic systems. Several companies have developed reliable pumps and valves for water that operate at 1500 to 2000 psi. There are still limitations (such as freezing) to using water, but in certain applications it has many benefits. One big advantage is that there are fewer environmental problems during operation or in disposing of the fluid. Price also is a factor because water costs so little and is readily available almost anywhere.

Some suppliers are making equipment that operates on seawater to eliminate possible contamination of the earths potable water sources. These systems operate at elevated pressures without performance loss.

High water-content fluids
*Some types of manufacturing still use water as a base and add some soluble oil for lubrication. This type of fluid is known as **high water-content fluid (or HWCF)**. The common mixture is 95% water and 5% soluble oil. This mixture takes care of most of the lubricity problems but does not address low viscosity concerns. Therefore, systems using **HWCF** still need expensive pumps and valves to make them efficient and extend their life.*

*Rolling mills and other applications with molten metals are one area where **HWCF** is prevalent. Often the soluble oil is the same compound used for coolant in the metal-rolling process. This eliminates concerns about cross-contamination of fluids and the problems it can cause.*

Water-in-oil emulsions

*Some systems use around 40% water for fire resistance and 60% oil for lubrication and viscosity considerations. Again, these are not common fluids because they require special oil and continuous maintenance to keep them mixed well and their ratio within limits. Most manufacturers do not want the problems associated with **water-in-oil emulsions** so their use is very limited.*

Water glycol

*A very common **fire-resistant** fluid is **water glycol**. This fluid uses water for fire resistance and a product like ethylene glycol (permanent anti-freeze) for lubricity, along with thickeners to enhance viscosity. Ethylene glycol will burn, but the energy it takes to vaporize the water present quickly quells the fire once it leaves the heat source. This means a fire would not spread to other parts of the plant. Always remember fire-resistant not fireproof.*

Water glycol fluids are heavier than mineral oil and do not have its lubricating qualities, so most pump manufacturers specify reduced rpm and lower operating pressures for water glycol. In addition, the water in this fluid can evaporate, especially at elevated temperatures, so it must be tested regularly for the correct mixture.

*Cost is also a consideration. **Water glycol** is more expensive than oil and requires most of the same considerations when disposing of it.*

*Always check with the pump manufacturer before specifying **water glycol** fluid to see what changes are necessary to run the pump with this fluid. Seal compatibility is usually not a problem, but always check each manufacturers specifications before implementing this fluid. In addition, it is imperative to completely flush a system of any other fluids before refilling with **water glycol**.*

Synthetics

*The other main **fire-resistant** fluids are **synthetic** types. They are made from mineral oil, but have been processed and contain additives to obtain a much higher flash point. It takes more heat to start them burning but there is not enough volatile materials in them to sustain burning. These fluids may catch fire from a pot of hot metal but quickly self-extinguish after leaving the heat source.*

***Synthetic** fluids retain most of the qualities of the mineral oil from which they are derived, so most hydraulic components specify no operating restrictions. However, most of these fluids are not compatible with common seal materials so seal specification changes are usually necessary. Special consideration must be given to handling of **synthetics** because they can cause skin irritation and other health hazards. Also most **synthetic** fluids require protective epoxy paint for all components in contact with them.*

Of all the fluids discussed, synthetics are the most expensive. They can cost up to five times more than mineral oil.

No matter which fluid is chosen, design the circuit to work in a reasonable temperature range; install good filters and maintain them; and check the fluids regularly to see if they are within specification limits.

A good operating temperature range is between 70° and 130° F with the optimum being around 110° F. A rule of thumb would be: warm enough to feel hot to the touch but cool enough to hold tightly for an extended period. Overheating hydraulic fluids is second only to contamination when it comes to reasons for fluid failure.

Continuous filtration of any hydraulic system is necessary for long component life. Fluids seldom wear out but they can become so contaminated that the parts they drive can fail. (The filter section of this book offers some good recommendations on keeping fluid clean.)

Even with the best of care, any hydraulic fluid should be checked at least twice a year. Systems located in dirty atmospheres may need to be checked more often to see if a pattern exists that requires special consideration. Pay close attention to the sampling process and packaging procedures recommended by the test facility that will process the sample. Expect a report on the level of contamination plus an analysis of the additive contents, water content, ferrous and non-ferrous material amounts, and any other problem areas the test facility finds. Use this information to know when to change fluids and to check for abnormal part wear problems.

New oil or other fluids from the supplier are not necessarily clean. The fluids are shipped in drums or by bulk, and there is no way of knowing how clean these containers are. Some suppliers offer filtered oil with a guaranteed contamination level at added cost. Otherwise, about the lowest level of contamination from most manufacturers is 25 microns.

Anytime a system needs new fluid, it is best to use a transfer unit, ***Figure 2-1****, with a 10-micron or finer filter in the loop. Another way of filtering new or refill fluid is with a filter permanently attached to the reservoir,* ***Figure 2-2****. In this arrangement, the breather or other possible fill points should be made inaccessible.*

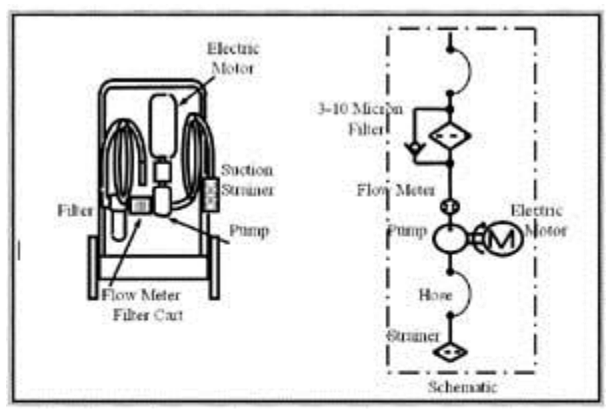


Fig. 2-1. Filter cart (used to transfer hydraulic fluids) and its circuit schematic diagram

*The filter cart shown in **Figure 2-1** can also be used to filter any hydraulic unit in the plant. Instead of this filter unit sitting idle except when filling systems, set it up at a machines power unit for a timed run. Place the suction hose in one end of the reservoir and the return hose in the opposite end. This adds a continual filtration loop to any machine even when the machines main pump is shut off. Run the cart until the fluid is clean and then move is to another power unit. Repeating this process on a regular schedule can assist the hydraulic units filters and add extra life to the fluid and the hydraulic components. This process may also show a pattern on machines that have a contamination problem.*

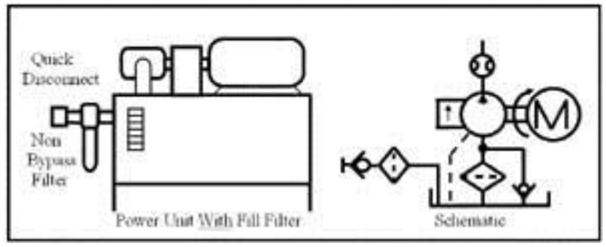


Fig. 2-2 Hydraulic power unit and circuit diagram of its filter arrangement

Hydraulic fluids should be stored in a clean dry atmosphere. Keep all containers closed tightly and reinstall covers on any partially used drums.

Never mix fluids in any hydraulic system. Make sure all containers are clearly marked and segregated so fluids will not be mixed with one another. Mixing fluids can result in damage to components and some combinations are very difficult to clean up. Be especially careful when mineral oils and synthetic or water-glycol fluids are used in different parts of the same plant.

Fluids are the lifeblood of any hydraulic system and should be given the utmost care.

Poor plumbing practices can permanently cripple a fluid power circuit even if it was designed with the best engineering practices and assembled with the most up-to-date components. Undersized lines, elbows instead of bends, incorrect component placement, and long piping runs are a few of the items that strangle fluid flow.

Other problems, such as using tapered pipe threads or lines with thin walls, can make a circuit a maintenance nightmare that requires daily attention. Fortunately, there are numerous publications that assist in specifying correct line size and conductor thickness to give low pressure drop and safe working-pressure limits.

Because pneumatic circuits are less complicated and operate at lower pressures, they are not as vulnerable to plumbing problems. One very important aspect that often is overlooked is the length and size of lines between the valves and actuators. Piping between the valve and actuator should be as short as possible and of the minimum diameter to carry the required flow. The reason for this is that all the air in the pipes between the actuator and valve is wasted every cycle. These runs must be filled to make the device move but the air it takes to fill them does no work. During each cycle, air in the actuator lines exhausts to atmosphere without helping cycle time or force. For this reason, always mount the valve close to the actuator ports.

Another aspect of plumbing a pneumatic system is the in-plant pipe installation procedure. To get the required amount of compressed air to the point of usage requires some planning -- or the site may be starved at times.

CFM	50'	100'	150'	200'	250'	300'	350'	400'	450'	500'	550'	600'	650'	700'	HP
10	¾"	¾"	1"	1"	1"	1"	1"	1"	1"	1"	1"	1"	1 ¼"	1 ¼"	2 ½
25	1"	1"	1 ¼"	1 ¼"	1 ¼"	1 ¼'	1 ¼"	1 ½'	1 ½"	1 ½"	1 ½"	1 ½'	1 ½"	1 ½"	6
50	1 ¼"	1 ½"	1 ½"	1 ½"	2"	2"	2"	2"	2"	2"	2"	2'	2"	2"	12
100	1 ½"	2"	2"	2"	2 ½"	2 ½"	2 ½"	2 ½"	2 ½"	2 ½"	3"	3"	3"	3"	25
200	2"	2 ½'	2 ½"	3"	3"	3"	3"	3"	3 ½"	3 ½"	3 ½"	3 ½"	3 ½"	3 ½"	50
300	2 ½"	3"	3"	3 ½"	3 ½"	3 ½"	3 ½"	3 ½"	4"	4"	4"	4"	4"	4"	75
400	3'	3"	3 ½"	3 ½"	4"	4"	4"	4"	4"	5"	5"	5"	5"	5"	100
600	3 ½"	3 ½"	4"	4"	5"	5"	5"	5"	5"	5"	5"	5"	6"	6"	150
800	3 ½"	4"	5"	5"	5"	5"	6"	6"	6"	6"	6"	6"	6"	6"	200

Fig. 3-1. Pipe size selection chart (in feet) for plant-air systems

Pipe materials and size: Air systems are normally plumbed with Schedule 40 black iron pipe. (Galvanized pipe is not recommended because some galvanizing material may flake off and get into moving parts.) Several other available plumbing materials could be used for air piping because pressure is relatively low. Some mechanics recommend plastic pipe, but be aware a few synthetic compressor lubricants attack plastic and cause it to lose strength. This type of damage weakens the plastic until it can burst, sending shards of plastic flying everywhere in the plant. Never use any piping material not specifically designated by code.

*To help select pipe size, the chart in **Figure 3-1** shows flow (in cfm) down the left-hand side, length of run (in feet) across the top, and minimum Schedule 40 pipe size in the body at the intersection of these two.*

This chart is based on a 1-psi pressure drop for the run lengths given. The right-hand column shows approximate compressor horsepower for the flow figures on the left. Using larger than specified pipe is of little help in reducing pressure drop, but provides more storage volume to handle short brief-flow needs. This chart does not consider fittings and valves, but they also must be considered when figuring the length of a run. Add 5 to 7 feet of pipe length for each fitting or valve -- to be on the safe side.

Not having enough air to run the equipment is expensive, so never try to save a few cents at installation by skimping on pipe size. One or two pipe sizes over minimum add little to cost up front, but can make a big difference later. It is less expensive to run oversize pipe initially than to have to add a line later.

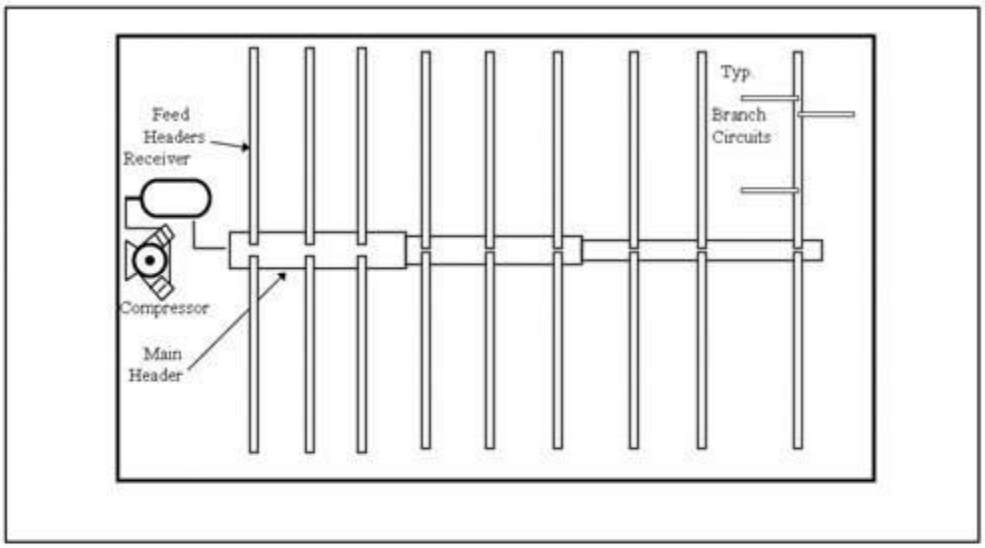


Fig. 3-2. Typical grid system layout for plant air

*There are three basic compressed-air piping layouts that meet the requirements of most industrial plants. Some facilities may have two or more of these systems to handle special needs. In general, smaller plants use a modified **grid system**, especially when the facility is growing. A **unit distribution system** offers flexibility, but can be expensive up front. A **loop system** is best suited to new construction; it provides extra storage capacity and dual supply for short bursts of high flow.*

***Figure 3-2** shows a typical grid-system layout using a centrally located air compressor. All air from the receiver goes to a large header pipe that runs down the center of the plant or department. Branch lines from the header go to separate areas where working drops come down* *to specific machines. With preplanning for future working drops, this arrangement is very flexible.*

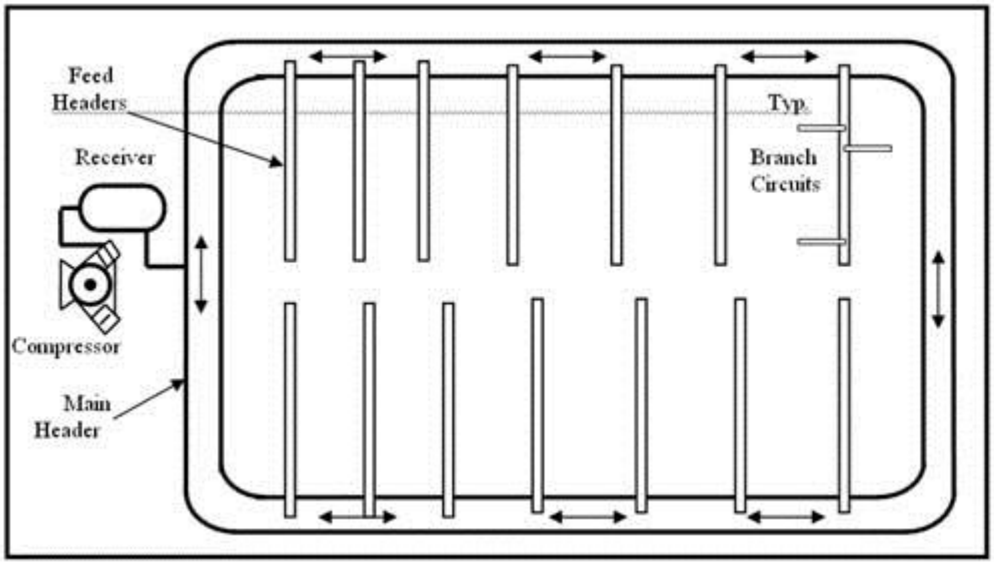


Fig. 3-3. Typical loop-piping system layout for plant air

Figure 3-3 *shows a typical loop piping system for compressed air. Again, the compressor and receiver are at a central location. The oversized loop around the periphery of the plant -- or department -- adds storage and allows flow with low pressure drop. It also allows for short bursts of high-volume flow to any section because flow in the loop is bi-directional. (Another way to get short high-volume flows with any of these piping systems is to install extra receiver tanks at or near areas that need such flow.)*

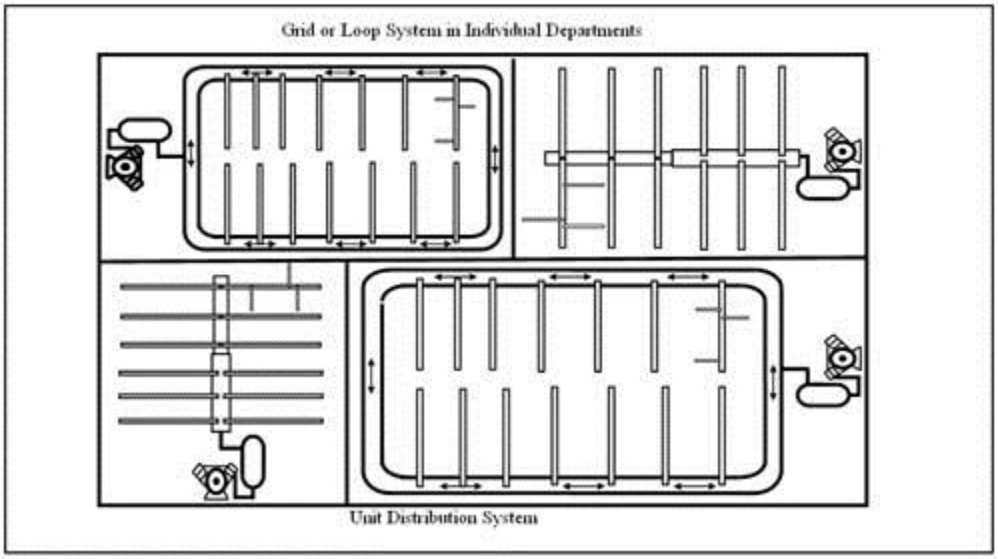


Fig. 3-4. Typical unit distribution layout for plant air

Figure 3-4 *illustrates a unit distribution layout that works well in plants that run departments on different days or shifts -- or plants that started out small and added compressors as business grew. It is the most expensive configuration of the three for a new installation, so is not often used there. One advantage of the multiple compressors is that there is always backup air available for critical operations should a single compressor fail. The disadvantage . . . besides higher price . . . is that some compressors might be neglected by maintenance personnel because they are spread throughout the facility.*

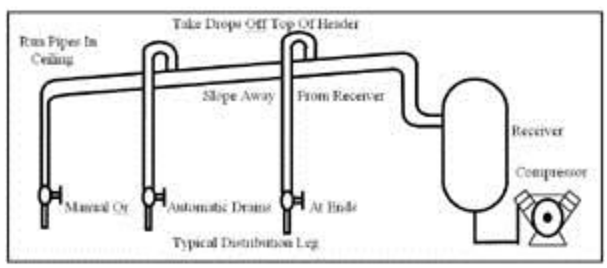


Fig. 3-5. Side view of typical compressed-air header and drop arrangement

Figure 3-5 *shows a typical pipe run layout for optimum performance from a compressed air system. Strict attention to the details shown here assures a smooth-operating and trouble-free air system.*

Pneumatic-machine plumbing

Machines can be plumbed with any of the materials recommended for plant piping. However, because piping at the machine is usually much smaller, polyethylene, nylon, or vinyl tubing with push-to-connect fittings will work very well. Such tubing and fittings come in a variety of sizes (and colors) and require only a few tools to install. The push-to-connect fittings also release easily for troubleshooting checks or rework.

Pipe materials and sizes

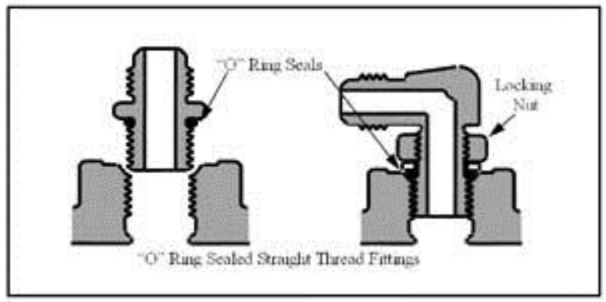


Fig. 3-6. O-ring sealed straight-thread fittings

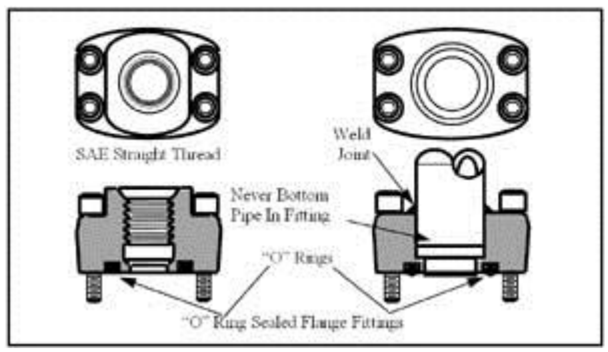


Fig. 3-7. Two types of O-ring sealed flange fittings

Though many hydraulic circuits are plumbed using black-iron pipe with tapered pipe threads, this is not the recommended way. It is nearly impossible to maintain leak-free operation of a 1000- to 3000-psi hydraulic system for any period of time with tapered pipe threads. Even if pipe-connection compound is used, expansion and shock soon loosen the taper interference and fluid weeps through the resulting openings. Another problem with tapered pipe threads occurs on circuits that must be routinely dismantled. Every time a tapered pipe thread is unscrewed, it must be tightened past where it was originally to get a good seal at reassembly. This can only happen so often before the pipe and/or valve must be replaced.

*The recommended plumbing material is steel tubing with straight-thread O-ring fittings up to 2-in. OD, **Figure 3-6**. In sizes larger than 2 in., use steel pipe with welded SAE O-ring-sealed flange fittings on each end, **Figure 3-7**. For flexible connections, reinforced rubber hose is most common; however, some prefer sealed steel swivel joints.*

A good reason for using steel tubing is that it is easily formed to allow for direction changes. Instead of installing fittings that can cause turbulence, use a tube bender to make sweeping turns that eliminate most of the pressure drop associated with elbows. This produces less pressure drop and less heat. Tubing is designated by its outside diameter (OD). As wall thickness increases, inside diameter (ID) decreases. (Black-iron pipe is measured by its nominal ID, but also has a decrease in ID with increase in wall thickness.)

Hydraulic hose

There are places on many machines where rigid pipe or tubing cannot be used because of their inflexibility. Rigid lines can cause problems at cylinders with pivot mountings, pumps on noise-isolation mounts, or connections between separate units. Hose avoids these problems.

However, wholesale use of hose in place of rigid lines it is not generally recommended. Hose is expensive, must be replaced on a regular basis, and flexes and stretches under pressure surges. This flexing produces extra volume and adds to cycle time. It is never recommended to use hose in a servo circuit (although there are times it can't be avoided). Servo circuits are for actuators that need precise control and flexing of hose lines can cause these valves to respond slowly and then go into high frequency oscillations.

Hose is specified by its ID and, unlike pipe and tube, this dimension does not change. Thicker walls for higher pressures make the outside diameter (OD) of hose greater. Pressure is specified in working and burst values (similar to pipe). Working pressure should always be equal to or higher than maximum system operating pressure. Flow rates of hose are slightly higher than pipe and about the same as steel tubing due to hose's smooth bore. However, many of the end connectors for hose are restrictive because they always go inside the inner liner. These fittings are only short restrictions, but can raise pressure drop noticeably in some cases.

Several factors influence hose service life and each one is controllable by some up-front fact finding and planning. First: never go under the manufacturers recommendation for minimum bend radius. Bending hose always causes stress but flexibility is the main reason for using it. Standard hose construction entails wire- or fiber-braided material laid down when the product is straight. Bending these braided materials puts extra stress on the outside of the bend and bunches up those on the inside. Add the constant expansion and contraction from pressure fluctuations during operation and it is easy to see the adverse effect.

Second: don't use hose above its rated working pressure. While maximum pressure might be set at or below the hose rating, higher shock pressures could be damaging during every cycle. Make sure the pressure rating of the hose on all machines is at or above operating pressure -- and design out system shock to protect the hose and other hydraulic components.

Third: avoid operating at temperatures above the rating of the hose. Most hose manufacturers make hoses in different temperature ratings. Of course, the higher the temperature rating, the more expensive the hose is, but it is false economy to use the wrong hose to save a few pennies.

Fourth: don't install hose where it must twist during each cycle or make it operate in a twist because of poor tightening procedures. Always hold the hose straight while tightening a connection. Either case stresses the hose and causes premature failure and its accompanying extra expense.

Hose distributors know of these pitfalls and can help with installation suggestions, as well as troubleshooting hose problems. The causes of hose problems are usually quite evident to someone who works regularly with hose, even when all he or she sees is the damaged part.

Sizing hydraulic lines

Fluid flow is measured in feet per second (fps), so the type of conduit is irrelevant. Many books have charts that relate gpm to fps for all standard piping systems. Use these charts to pick out the correct size fluid carrier for the required flow.

Pump inlet line (suction line)

Fluid velocity should not exceed 2 to 4 fps. The reason for this recommendation is that the highest possible pressure drop in the pump inlet line is one atmosphere. Actually, no type of hydraulic pump can even come close to this, so most inlet lines never see much more than 3- to 4-psi vacuum. Using velocity higher than 2 to 4 fps dramatically increases pressure -- causing cavitation and pump damage. It is best to use a suction line equal to or larger than the size of the pump inlet being plumbed. There are circumstances when a smaller suction line is satisfactory, but only do this when absolutely necessary and with the supplier's approval.

The suction line should be full size; as straight as possible; have no or the minimum number of fittings; never include a standard pipe union; and be completely sealed. Using hose in place of pipe or tube can overcome many possible suction-line problems. Hose is a viable alternative and is quite satisfactory if certain precautions are addressed. Always use hose designed and specified for suction (vacuum) use. Hose normally used for pressure may be rated at 3000 psi but is not suited for suction lines. The reason for this is pressure hose uses an inner lining like a tube in a tube-type tire. The outer layers are strong but they are porous and would leak high-pressure fluid except for the inner tube. High-pressure hose as a suction line sees constant negative pressure trying to collapse the inner tube. After some time, it is possible for the inner liner to be drawn in, restricting flow and causing pump cavitation. This phenomenon may not happen immediately, but usually does cause problems in time.

Return lines

Fluid velocity in return lines should be held between 10 and 15 fps. The pump can push oil returning to tank, but any backpressure in these lines must be overcome by extra pressure at the pump outlet. To maintain a high-efficiency circuit, it is important to keep pressure drop in all *lines as low as possible. All energy used to push oil through the lines is wasted and converts to heat.*

Working-pressure lines

Medium pressure lines (up to 2000 psi) should not exceed 15 to 20 fps. Flow in systems that operate above 2000 psi can go as high as 30 fps. Unlike air systems, there is usually excess pressure capacity in hydraulic circuits when actuators are in motion. Typically, high pressure only comes into play when the actuators near the end of stroke. In an effort to keep line and valve sizes small, it is common practice to use these higher velocities -- but keep in mind this practice wastes energy.

Several fluid power handbooks are available with excellent charts showing tubing and pipe in all different wall thickness, along with flow in gallons per minute (gpm) for all standard sizes. Remember each fitting or valve in the circuit has its own pressure-drop adders and they must be taken into consideration as part of the overall pressure-loss picture.

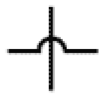
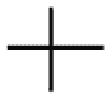
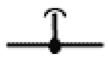
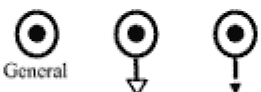
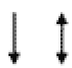
A family of graphic symbols has been developed to represent fluid power components and systems on schematic drawings. In the United States, the American National Standards Institute (ANSI) is responsible for symbol information. The Institute controls the make-up of symbols and makes changes or additions as required. The International Standards Organization (ISO) has the same control over symbols used internationally. Both systems have almost the same format (especially since ANSI changed its symbols in 1966 to eliminate all written information).

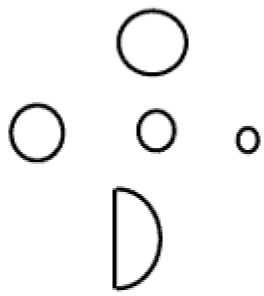
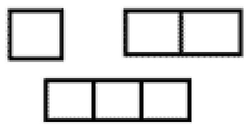
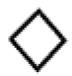
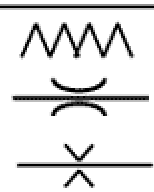
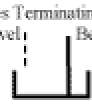
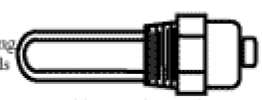
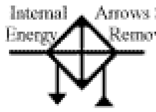
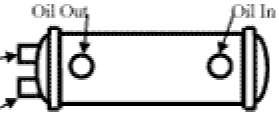
Standard symbols allow fluid power schematic diagrams to be read and understood by persons in many different countries, even when they don't speak the same language. Either symbol set (ANSI or ISO) may be -- and is -- used in the United States. However, many companies today use the ISO symbols as their standard for work with foreign suppliers and customers.

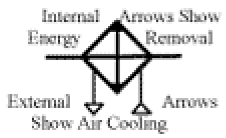
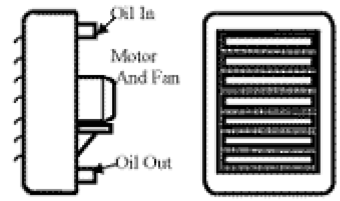
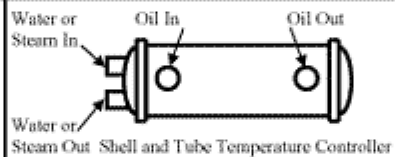
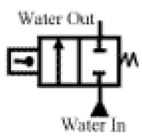
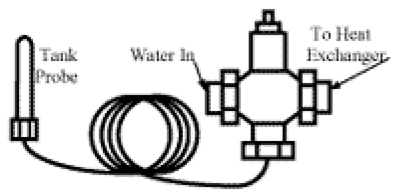
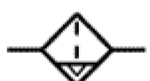
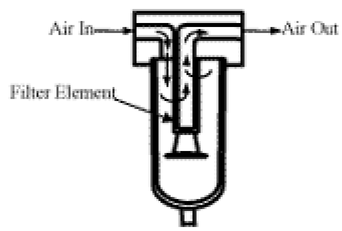
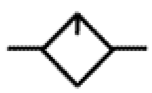
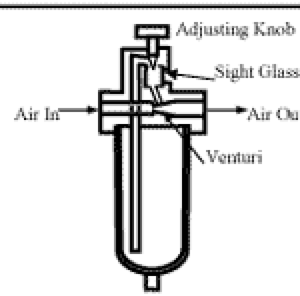
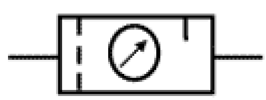
The following pages go through all standard ISO symbol information as it applies to hydraulic and pneumatic schematics. There are still many plants that modify the standards to suit some individual's taste. This widespread practice may be confusing to novices. Symbols have been developed to represent most of the available fluid power components. However, some parts must be made up of combinations of different symbols to show how they function. Other times there is no standard symbol and one must be made up. In such cases, look first in the supplier's catalog for the symbol they show. If the supplier did not make a symbol, the only other option is design one for the new part. Try to design the new symbol using standard practices shown here.

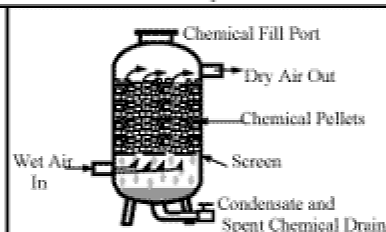
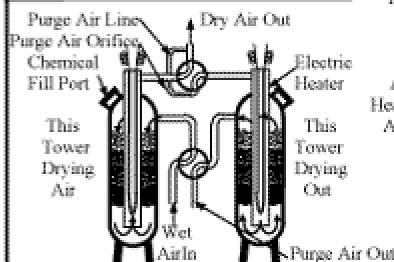
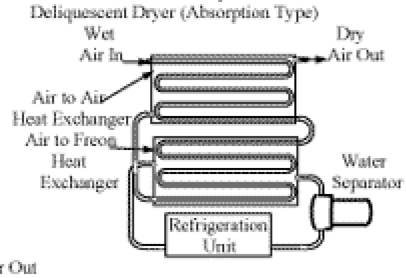
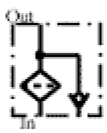
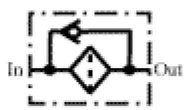
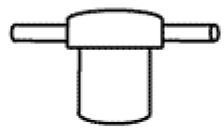
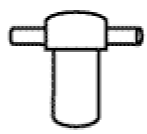
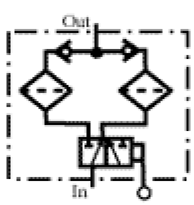
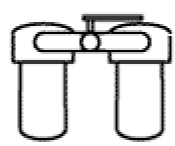
As the phrase fluid power implies, these symbols cover both hydraulic and pneumatic components. Any exceptions are noted.

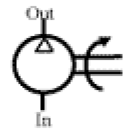
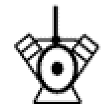
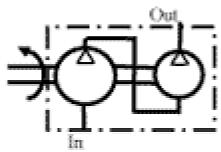
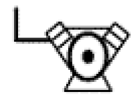
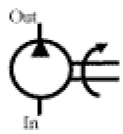
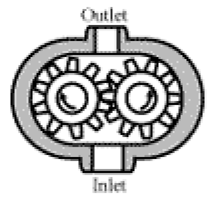
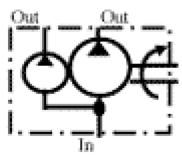
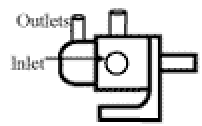
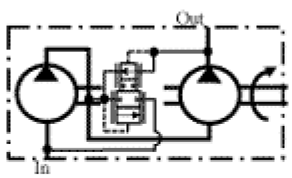
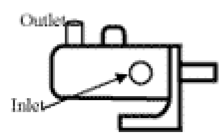
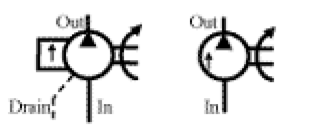
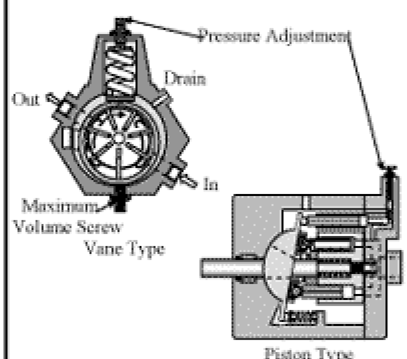
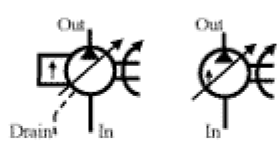
ISO Designation	Symbol	Picture Representation
Basic Information		
Lines		
Continuous		Represents a working fluid line. This fluid comes from a prime mover and goes to the actuator to perform work. May be a 1/8" plastic air line or any size pipe or tube in a hydraulic system.
Non-Flowing	Black	
Pump Flow	Red	
Tank Flow	Blue	
Suction Flow	Green	
Metered Flow	Yellow	COLOR CODING FOR OVERHEADS
Reduced Pressure	Orange	
Intensified Fluid	Purple	
Long Dashes	Orange	Represents pilot lines that supply a small amount fluid to another valve or device making it operate. The length of these dashes should be at least ten times their thickness .
Short Dashes	Green	Represents drain lines for hydraulic circuits. Many hydraulic valves have internal leakage that can get trapped and cause a malfunction .A drain line is a small line giving trapped fluid a free flow path to tank. The length of these dashes is five times their thickness.
Double Lines		Represents a mechanical connection Between components, A pump motor shaft, feedback connections between valves and actuators, etc. The outside dimension of these lines should be at least five times the line hickness.
Center Line		Represents an enclosure outline that indicates the parts inside it are a unit. This unit may be a casting with the parts machined in it or it might be a several components assembly.
Electric Line		Denotes a line carrying electrical power or signal.

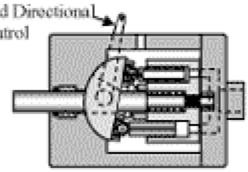
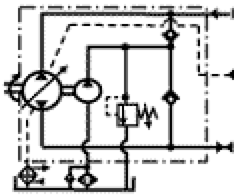
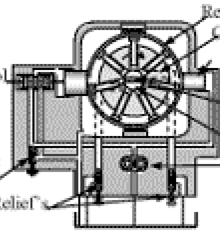
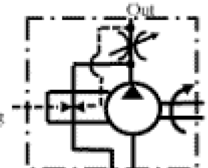
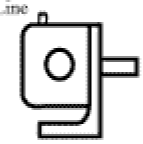
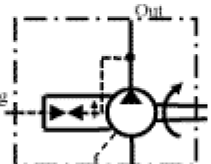
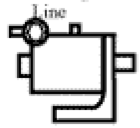
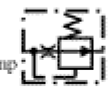
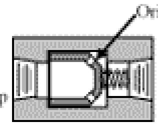
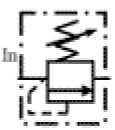
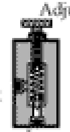
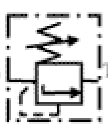
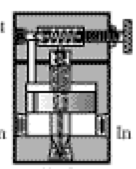
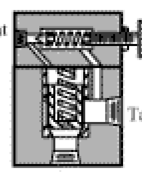
ISO Designation	Symbol	Picture Representation
Flexible Line		Shows hose or other flexible line. Could be whole length or only a portion of the conduit.
Pipe Junction		Denotes a tee or cross where lines connect to one another. The connecting Dot is always used with the Jumper below. Preferred way.
Pipe Junction		Denotes pipes connecting at a tee or cross Optional way.
		Denotes pipes crossing. Preferred way
Pipes Crossing		Denotes pipes crossing. Optional way
Plug		
Air Bleed		Shows a connection for bleeding trapped air
Pressure Takeoff		A connection for taking power from a line or for pressure testing. Shown plugged.
		With takeoff line connected .
Energy Triangle	 Pneumatic Hydraulic	Shows direction of flow and type of fluid.
Power Source	 General Pneumatic Hydraulic	Denotes a power source from another part of the schematic or a another source
Arrows		Indicates direction of rotation of a pump or motor shaft, valve actuator or other actuator.
		Indicates direction of movement of a component.
		Arrows used for flow direction in valves. Arrows with a perpendicular line opposite the arrow head Arrows with a perpendicular line at the head end indicates the path stays connected to its outlet when it moves
Sloping Arrow		A sloping arrow through a pump, valve, spring, cushion plunger, solenoid or other device indicates it is adjustable or variable.

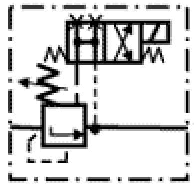
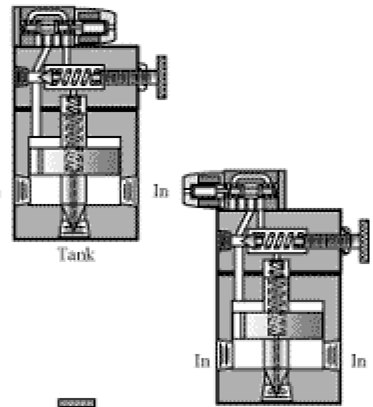
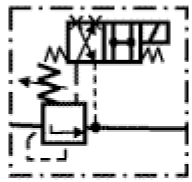
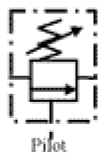
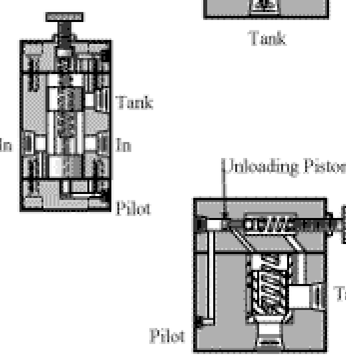
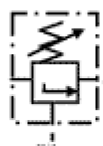
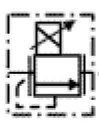
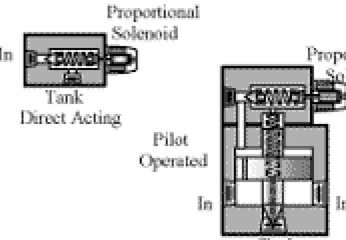
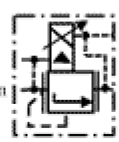
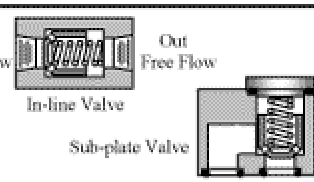
ISO Designation	Symbol	Picture Representation
Circles and Semi-circles		Represents a rotary device such as a pump or motor. Denotes the device is capable of continuous rotation in one or both directions. Circles of different sizes are the basis of unit of measurement instruments such as gauges. Also represents mechanical rollers on cam operated valves. Semi-circles denote rotary output devices that are not capable of continuous rotation. These actuators only oscillate through some arc.
Squares and Rectangular Boxes (Envelopes)		Square or rectangular boxes or envelopes are the basic unit of pressure and directional control valves. Single boxes denote pressure controls while multiple boxes show directional controls. The valve operator box is also a rectangle
Diamond Shaped Boxes		Diamond shaped boxes indicate a fluid conditioning device like a filter, lubricator, or heat exchanger.
Miscellaneous Symbols		A sawtooth line represents a spring. Back to back semi-circles on a line show a standard orifice that is affected by viscosity. Back to back "V" on a line denotes an orifice That is not affected by viscosity.
Reservoirs Chapter 6 Reservoir Open To Atmosphere Pressurized Reservoir	Lines Terminating Above Oil Level Below Oil Level  	 For systems that operate in contaminated locations or when using pressure to force fluid flow.
Heaters Chapter 6 Tank Heater		 Electric Heating Unit
Heat Exchangers (Coolers) Chapter 6 Water Cooled Heat exchanger	 External Show Liquid Cooling	 Shell and Tube Heat Exchanger

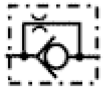
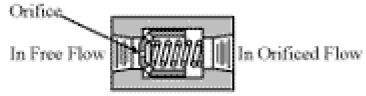
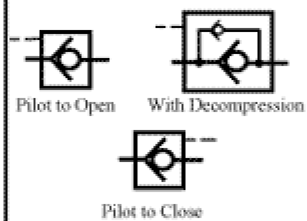
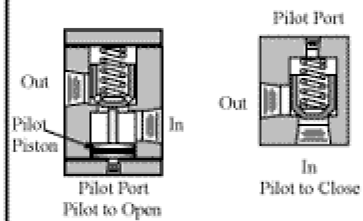
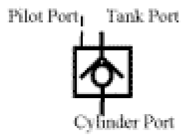
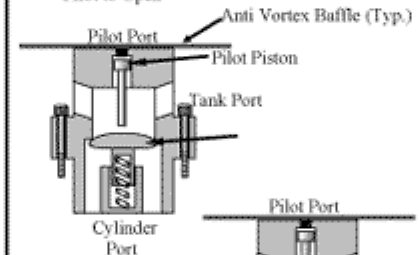
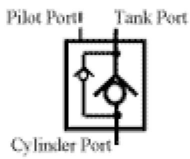
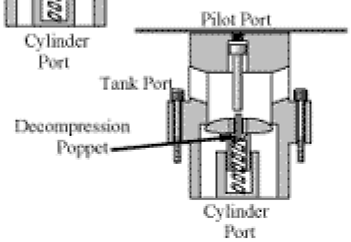
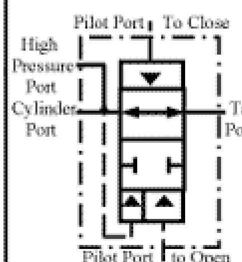
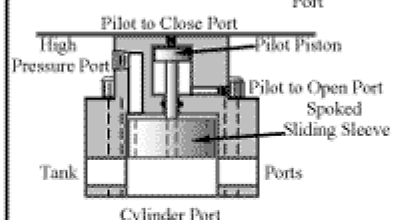
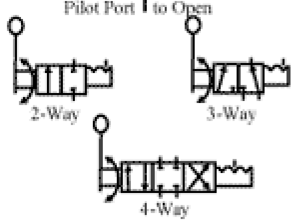
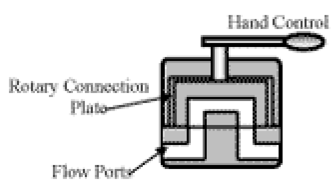
ISO Designation	Symbol	Picture Representation
Heat Exchangers (Coolers) Chapter 6 Air Cooled Heat Exchanger	 	 Fan Cooled Radiator Type Heat Exchanger
Temperature Controller Chapter 6	 	 Shell and Tube Temperature Controller
Temperature Controlled Water Valve Chapter 6	 	 Temperature Controlled Water Valve
Air Line Filter Chapter 7	 With Manual Drain  With Automatic Drain	
Air Line Lubricator Chapter 7		
Air Line Filter, , Regulator and Lubricator Chapter 7	 Simplified Symbol	

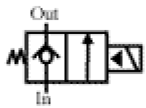
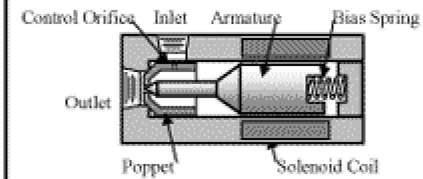
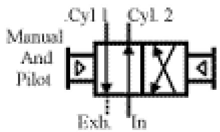
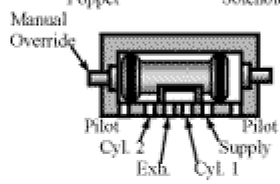
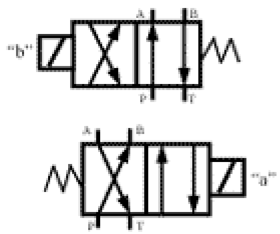
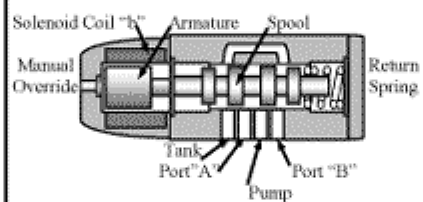
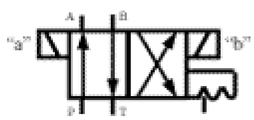
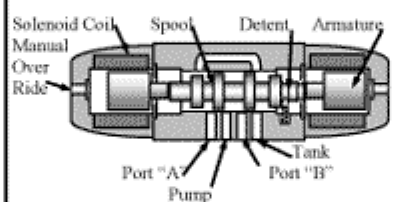
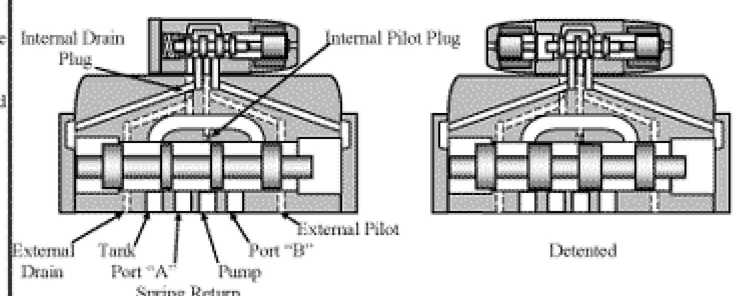
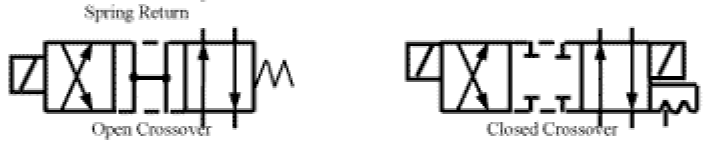
ISO Designation	Symbol	Picture Representation
Air Dryer **Chapter 7**		 Deliquescent Dryer (Absorption Type)
	 Desiccant Dryer (Adsorption Type)	 Refrigeration Dryer (Condensing Type)
Hydraulic Filters **Chapter 7** In Tank Hydraulic Suction Strainer With Bypass		 In Tank Type Suction Strainer
In-Line Return Filter With Bypass		
In-Line Pressure Filter Without Bypass		
Dual Pressure Filter For Changing Elements While Circuit is Running		

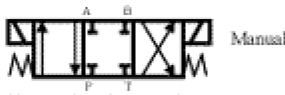
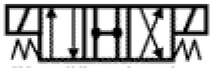
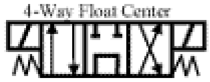
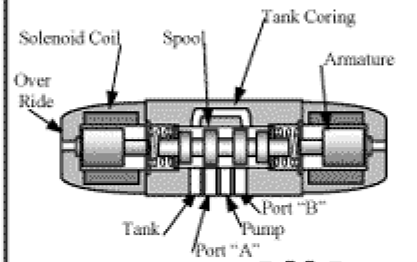
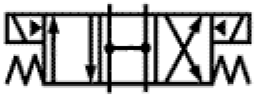
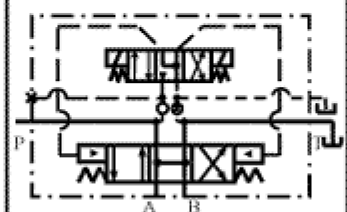
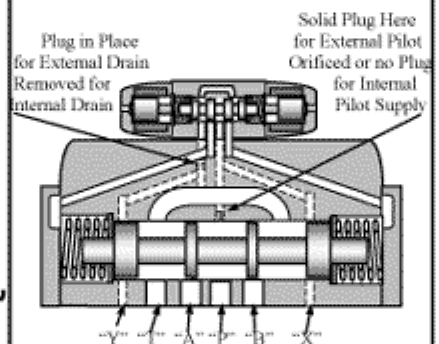
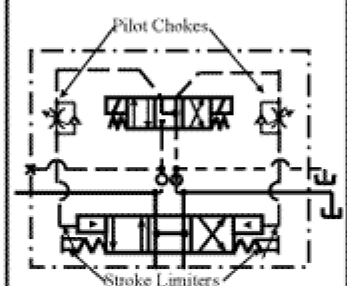
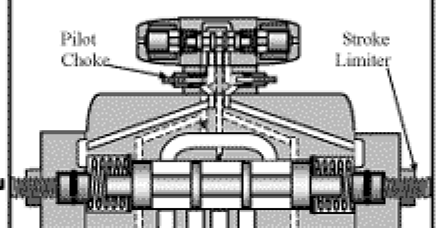
ISO Designation	Symbol	Picture Representation
Pumps **Chapter 8** Air Pump Single Stage (Compressor)		
Air Pump Two Stage (Compressor)		
Hydraulic Pump Fixed Volume Single Flow Single Stage		
Hydraulic Pump Fixed Volume Double Flow Single Stage		
Hydraulic Pump Fixed Volume Single Flow Two Stage		
Hydraulic Pump Pressure Compensated Single Flow Single Stage	Complete Symbol Simplified Symbol	Vane Type Piston Type
Hydraulic Pump Pressure Compensated Variable Volume Single Flow Single Stage		

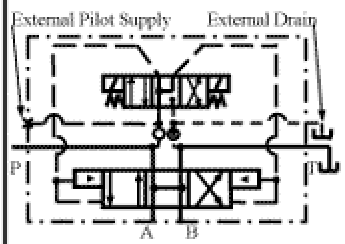
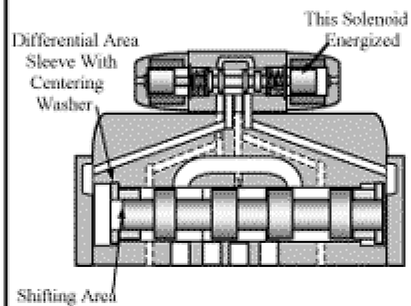
ISO Designation	Symbol	Picture Representation
Pumps (Cont'd.) Hydraulic Pump Variable Volume Bi-directional Flow Single Stage		
Hydraulic Pump Variable Volume Bi-directional Single Stage With Charge Pump For Closed Loop Circuits		
Hydraulic Pump Fixed Volume Single Flow Single Stage With Load Sensing		
Hydraulic Pump Pressure Compensated Single Flow Single Stage With Load Sensing		
Air Bleed Valve for closed center circuits to let trapped air escape from pump outlet		
Relief and Unloading Valves Chapter 9 Relief Valve Direct Acting Adjustable		
Relief Valve Pilot Operated Adjustable With Vent Port	 Typical Symbol  Showing Vent Port	 Balanced Piston Design  Poppet Design

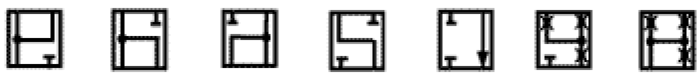
ISO Designation	Symbol	Picture Representation
Relief and Unloading Valves (Cont'd.) Solenoid Operated Relief Valve Normally Open		
Solenoid Operated Relief Valve Normally Closed		
Unloading Valve External Pilot Operated Direct Acting		
Unloading Valve External Pilot Operated Pilot Operated Also Used as an Accumulator Circuit Pump Unloading Valve Through Unloading Spool		
Proportional Relief Valve Direct Acting		
Proportional Relief Valve Pilot Operated		
Directional Control Valves Chapter 10 Check Valves Inline, Sub-plate Mounted, Screw In Cartridge	Plain Check Function As a Back Pressure Valve	

ISO Designation	Symbol	Picture Representation
Directional Control Valves (Cont'd.) Orificed Check Valve		
Pilot Operated Check Valves		
Pre-fill Valves Without Decompression Feature		
With Decompression Feature		
Sliding Sleeve Pre-fill Valve		
Rotary Shear Valve		

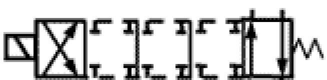
ISO Designation	Symbol	Picture Representation
Directional Control Valves (Cont'd.) Poppet Type Valve		
Sliding Plate Valve		
Spool Type Valve 4-Way 2 Position Valve Spring Return Direct Acting Solenoid		
4-Way 2 Position Valve Detented Direct acting Solenoid		
4-Way 2 Position Valve Spring Return and Detented Solenoid Pilot Operated		
2 Position Valve Typical Transition or Crossover Condition		

ISO Designation	Symbol	Picture Representation
Directional Control Valves (Cont'd.) Spool Type Valves Direct Solenoid Operated, Three Position, Spring Centered Directional Control Valves	 4-Way All Port Blocked Center  4-Way All Ports Open Center  4-Way Float Center  4-Way Tandem Center **Four Standard Center Conditions**	  All Ports Blocked Spool  All Ports Open Spool  Float Center Spool  Tandem Center Spool
Solenoid Pilot Operated Three Position, Spring Centered Directional Control Valve	 Simplified Symbol  Complete Symbol	
Solenoid Pilot Operated, Three Position, Spring Centered Directional Control Valves With Pilot Chokes and Main Spool Stroke Limiting Screws		

ISO Designation	Symbol	Picture Representation
Directional Control Valves (Cont'd.) Solenoid Pilot Operated, Three Position, Hydraulic Centered, Directional Control Valve	 Complete Symbol Internal Pilot Supply, External Drain	 Valve Shown in Shifted Condition



Other Available Center Conditions From Most Manufacturers

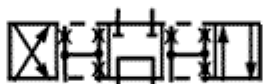


Typical Crossover or Transition Conditions

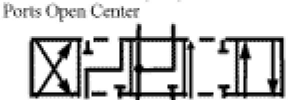
4-Way 2 Position Single Solenoid
All Ports Blocked Center

4-Way 3 Position Double Solenoid Spring Centered
All Ports Open Center

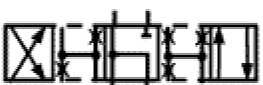
Float Center



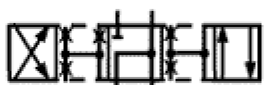
Tandem Center



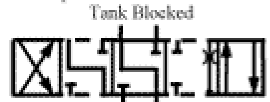
Pump "A" and "B" Connected
Tank Blocked



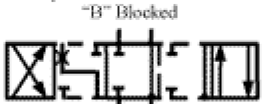
Pump "A" and Tank Connected
"B" Blocked



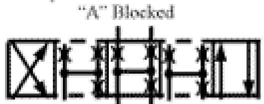
Pump "B" and Tank Connected
"A" Blocked



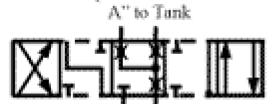
Pump and "B" Blocked
"A" to Tank



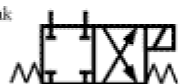
Pump and "A" Blocked
"B" to Tank



Open Center Orificed Flow



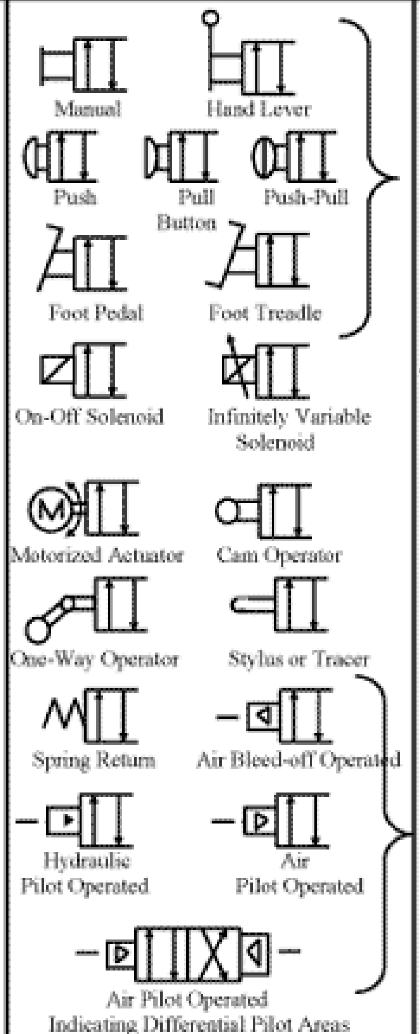
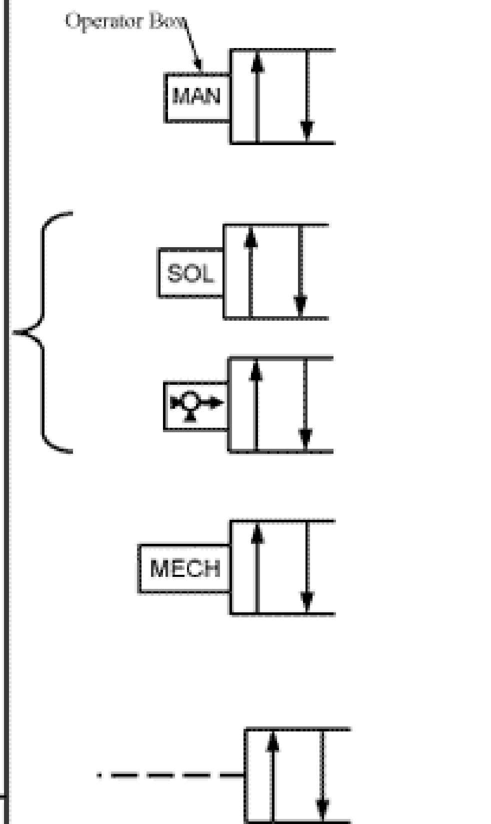
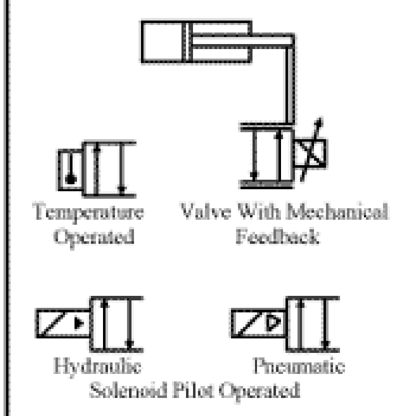
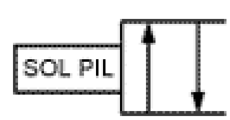
Float Center Orificed Flow



Single Solenoid Spring Centered
At Rest in Center Condition Shift to Flow Path

Single Solenoid Spring Return
At Rest in Flow Path, Shift to a Center Condition

.

ISO Designation	Symbol	Pre 1966 ANSI Symbol
Directional Control Valves (Cont'd.) Valve Operators		
Valve Operators		

ISO Designation	Symbol	Picture Representation

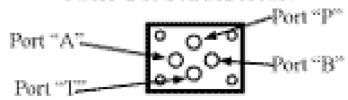
Directional Control Valves (Cont'd.)

NFPA AND ISO STANDARD INTERFACE LAYOUTS FOR HYDRAULIC DIRECTIONAL CONTROL VALVES

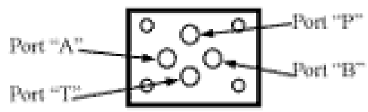
NFPA	ISO	*CETOP*	NG	PORT DIA	NOMINAL FLOW
DO2	02	*2*	4	.177	5 GPM
D03	03	*3*	6	.295	10 GPM
D05	05	5	10	.440	20 GPM
D05H	05			.440	25 GPM
D07	07	7	16	.690	30 GPM
D08	08	*8*	25	.984	60 GPM
D10	10	*10*	32	1.250	100 GPM

PORT CONFIGURATION

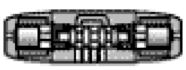
DO2

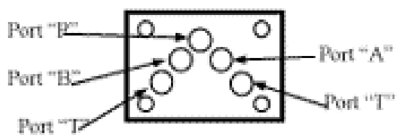


D03



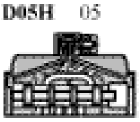
D05

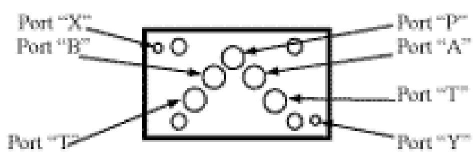




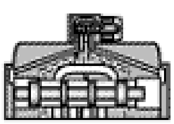
All Views Shown Looking Into Ports On The Valve

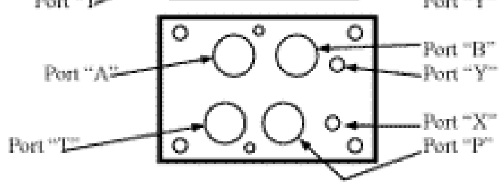
D05H



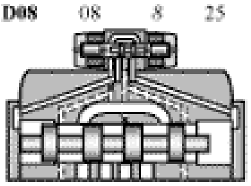


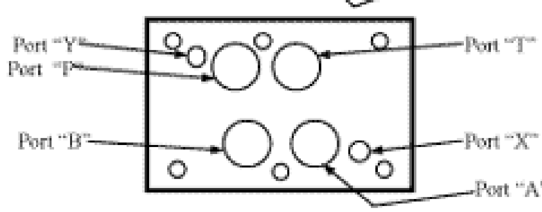
D07



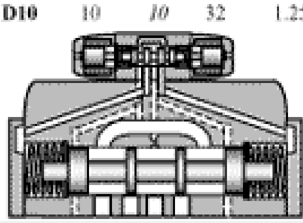


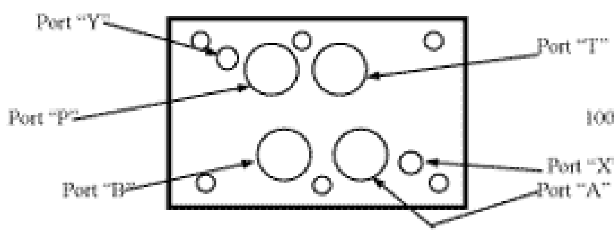
D08





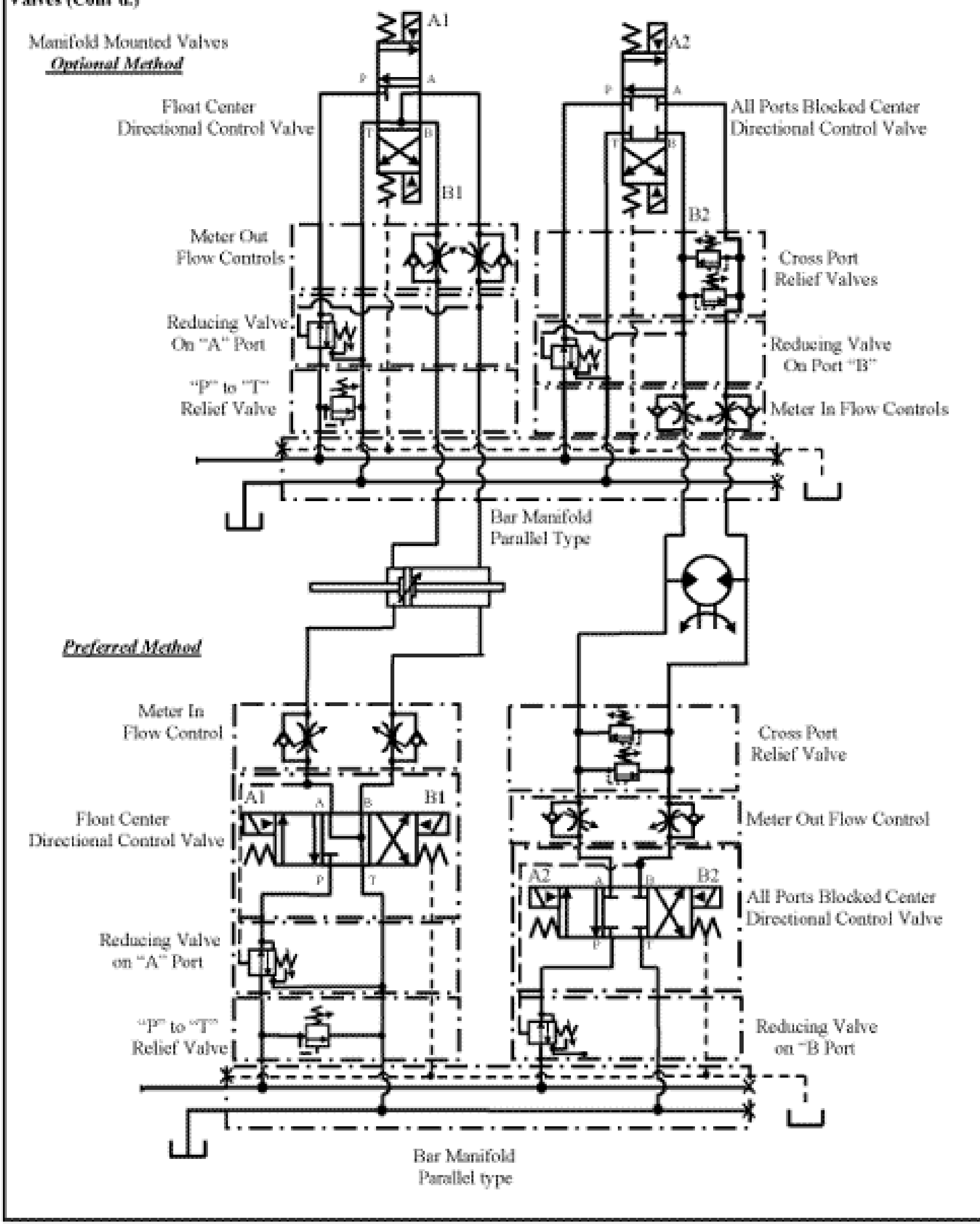
D10





ISO Designation	Symbol	Picture Representation

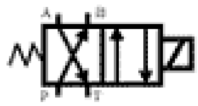
Directional Control Valves (Cont'd.)

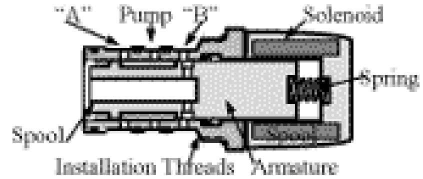


ISO Designation	Symbol	Picture Representation

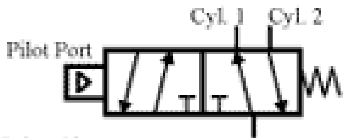
Directional Control Valves (Cont'd.)

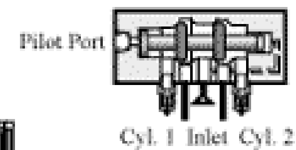
Screw-in Cartridge Valves, All the same functions as In-line and Sub-Plate mounted Valves.



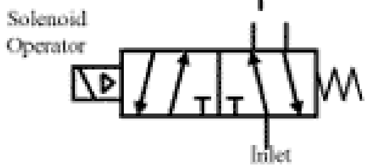


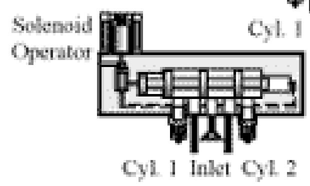
5-Way Air Pilot Operated, Spring Return Air Valve, Poppet Type



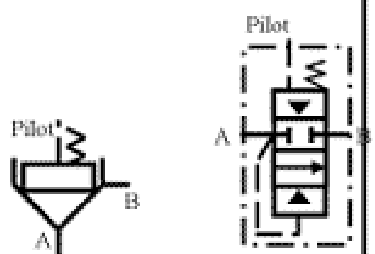


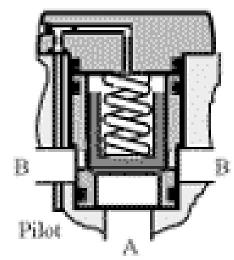
5-Way Solenoid Air Pilot Operated, Spring Return Air Valve, Spool Type



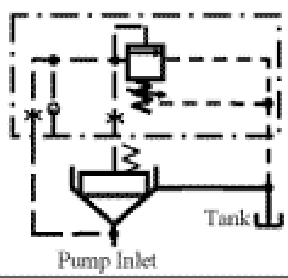


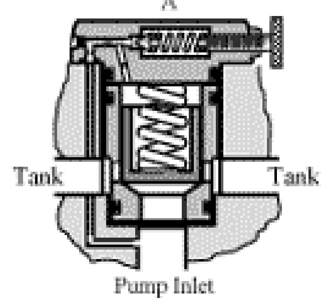
1:1 Poppet Ratio used for Pressure Control

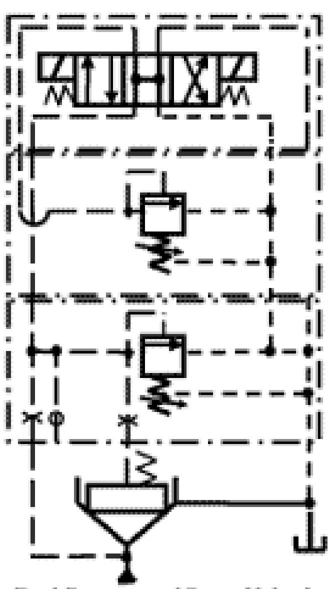




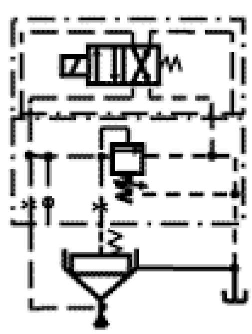
1:1 Poppet Ratio as a Relief Valve



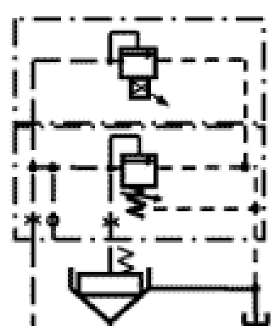




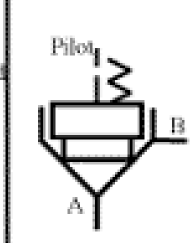
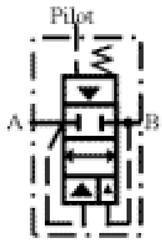
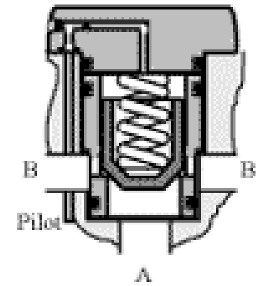
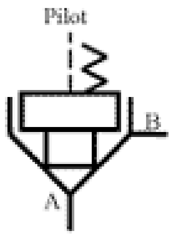
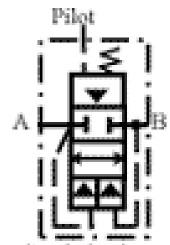
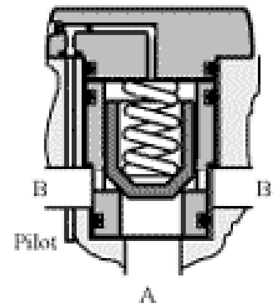
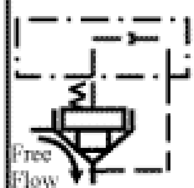
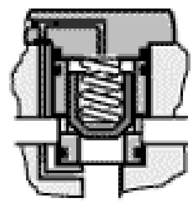
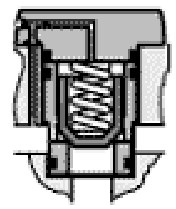
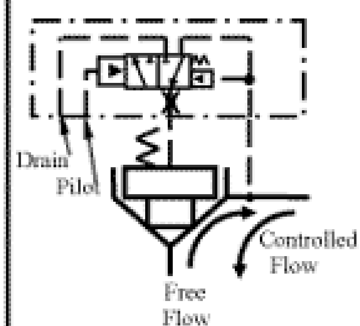
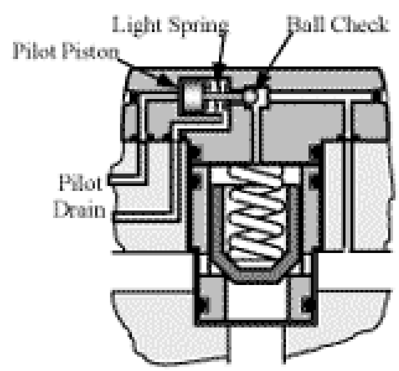
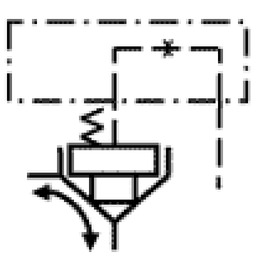
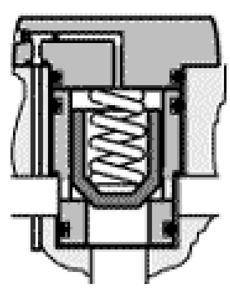
Dual-Pressure and Pump Unload and Relief Valve

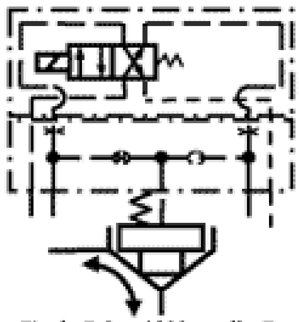
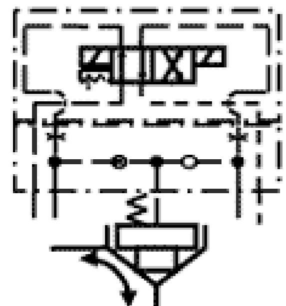
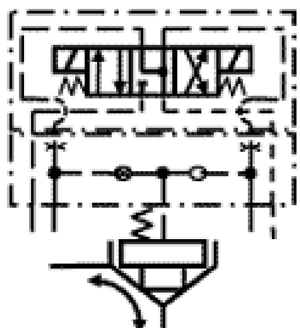
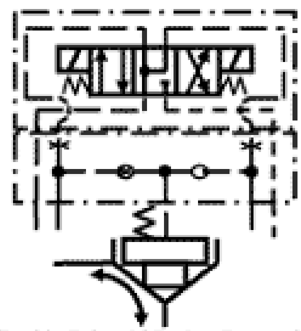
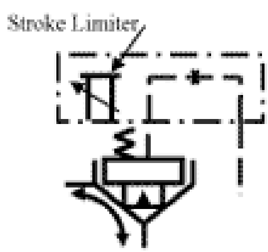
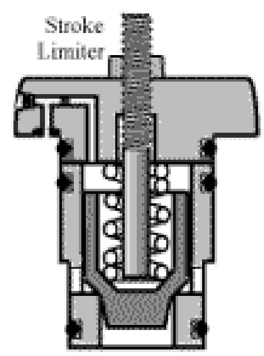
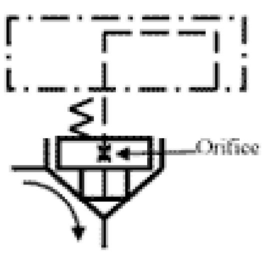
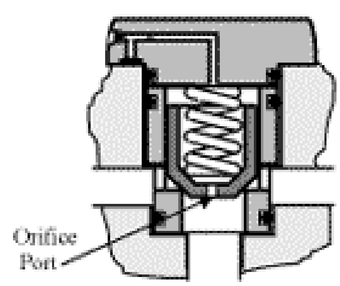


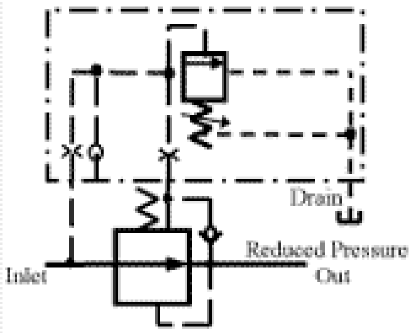
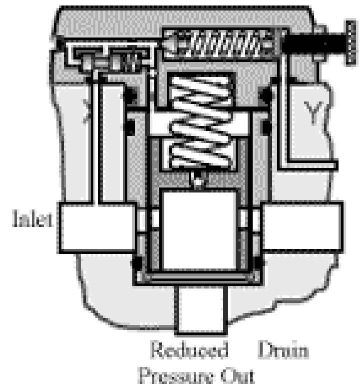
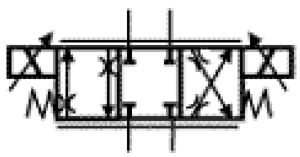
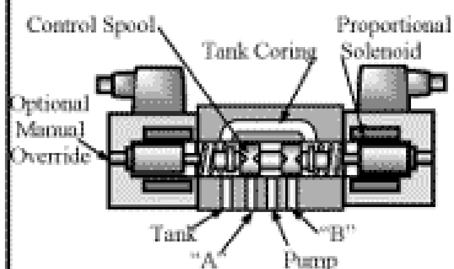
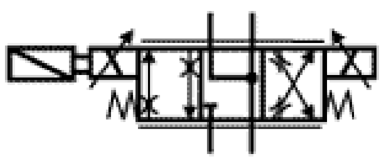
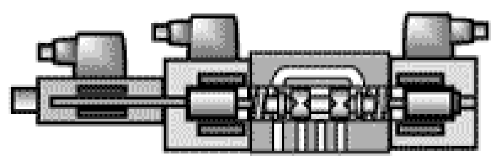
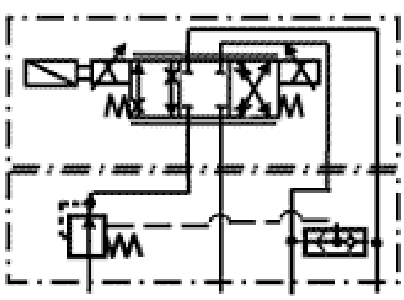
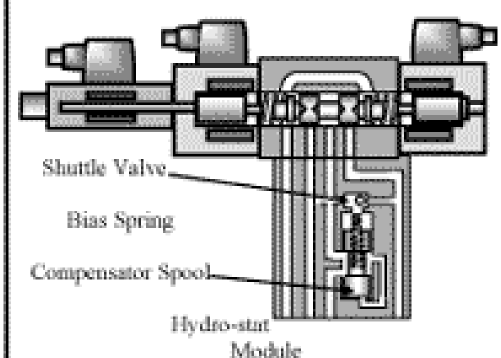
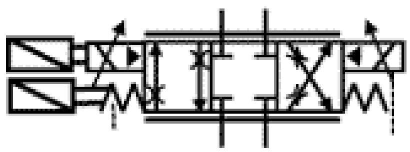
Normally open Solenoid Operated Relief Valve

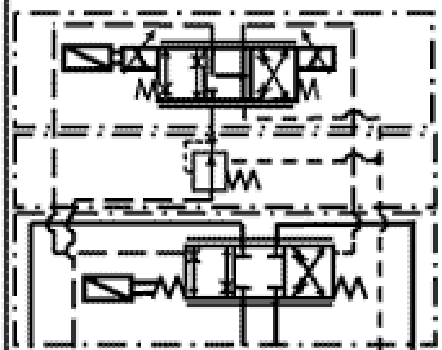
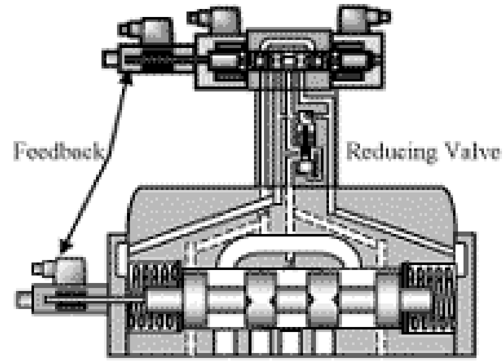
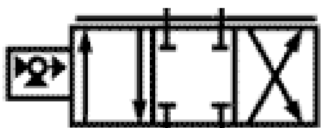
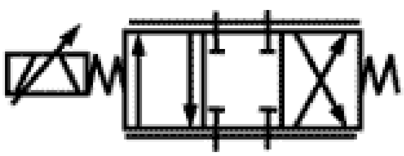
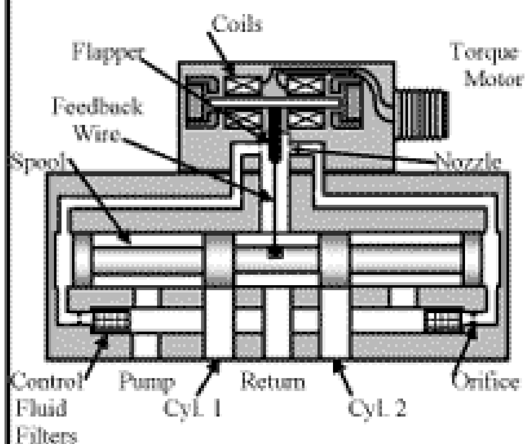
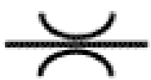
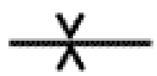
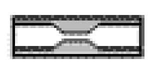
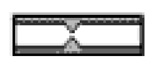
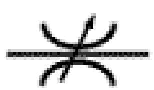
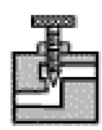
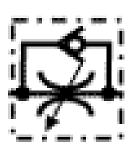
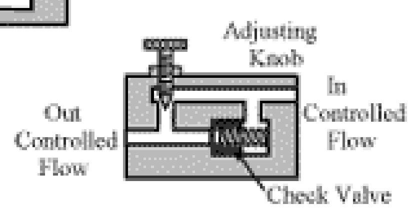
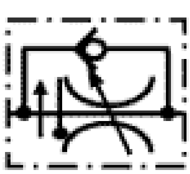
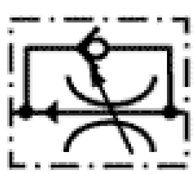
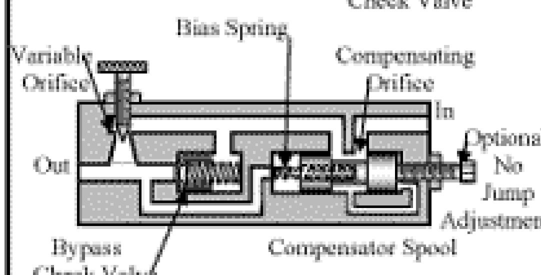


Proportional Relief Valve

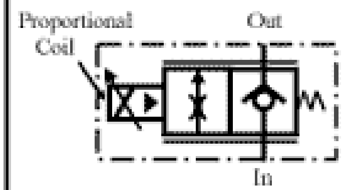
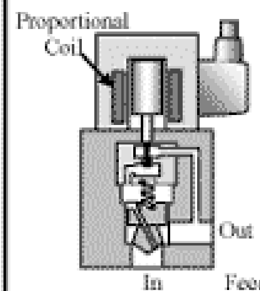
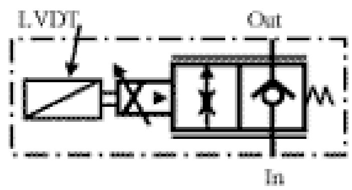
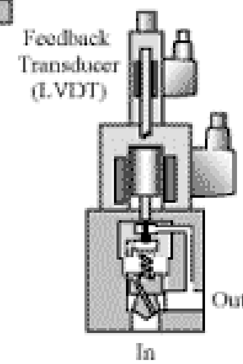
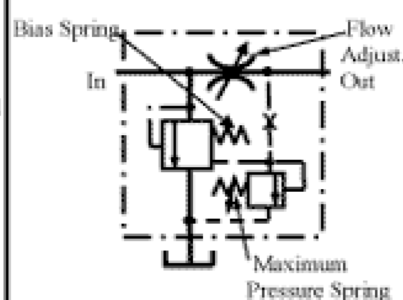
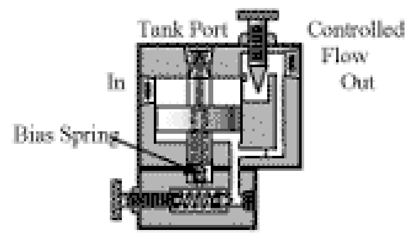
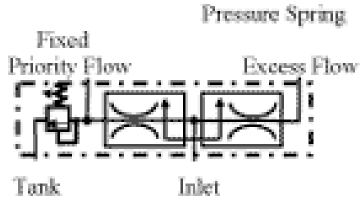
ISO Designation	Symbol	Picture Representation
Slip-in Cartridge Valves (Cont'd.) 1:1.1 Poppet Ratio Used for Pressure Control or Directional Control	 Frequently Used Symbol  Symbol using ISO Rules	
1:2 Poppet Ratio Used For Directional Control	 Frequently Used Symbol  Symbol using ISO Rules	
1:2 Poppet Ratio as a Check Valve With Plain Cover and Internal Connection	 	 
1:2 Pilot Ratio as a Pilot Operated Check Valve With Pilot Operated Check Valve Cover		
1:2 Pilot Ratio as a Directional Control Valve With Plain Cover and External Connection		

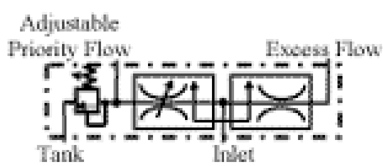
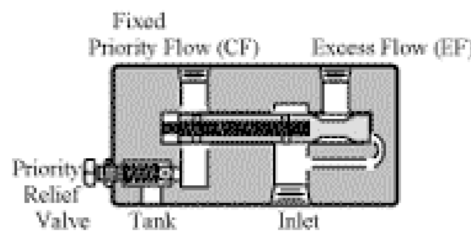
ISO Designation	Symbol	Picture Representation
Slip-in Cartridge Valves (Cont'd.) 1:2 Pilot Ratio as:	 Single Solenoid Normally Open	 Double Solenoid Detented Normally Open
	 Double Solenoid Spring Centered Normally Open	 Double Solenoid Spring Centered Normally Closed
Slip-in Cartridge Valve With Stroke Limiter and Deceleration Skirts		
Slip-in Cartridge valve With Orifice port As a Check Valve		

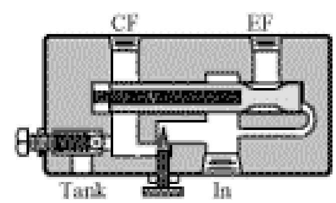
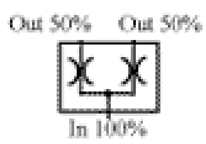
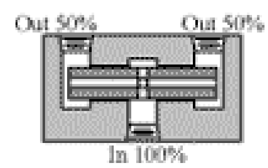
ISO Designation	Symbol	Picture Representation
Slip-in Cartridge Valves (Cont'd.) Slip-in Cartridge Valve as a Reducing Valve		
Infinitely Variable Directional Valves Chapter 12 Proportional Controls Direct Solenoid Operated Without Spool Position Feedback		
Direct Solenoid Operated With Spool Position Feedback		
Direct Solenoid Operated With Spool Position Feedback and Pressure Compensating Module (Hydro-Stat)		
Simplified Symbol for a Solenoid Pilot Operated Valve With Spool Position Feedback		SEE NEXT PAGE PROPORTIONAL VALVE COMPLETE SYMBOL

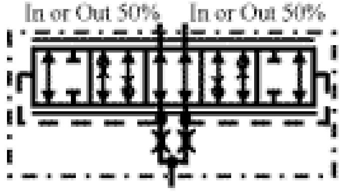
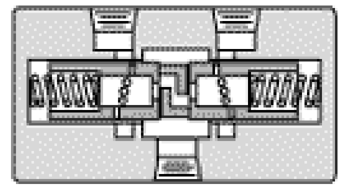
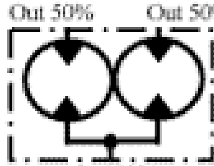
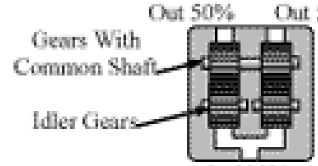
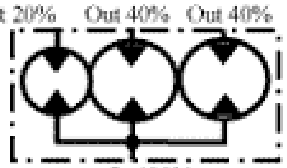
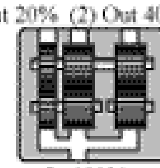
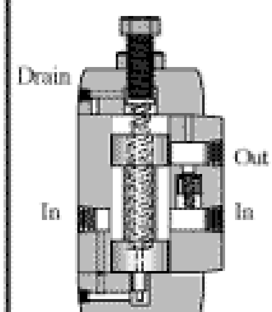
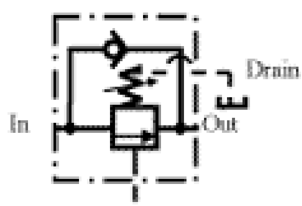
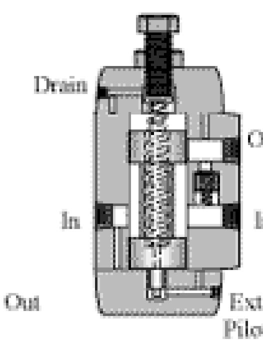
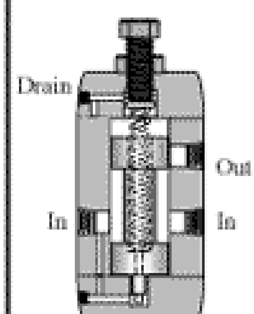
ISO Designation	Symbol	Picture Representation
Infinitely Variable Directional Valves (Cont'd.) Complete Symbol for a Solenoid Pilot Operated Valve With Spool Position Feedback		
Servo Valves	 ANSI Symbol  ISO Symbol	
Flow Control Valves Chapter 13		
Fixed Orifice	  Affected by Un-affected by Viscosity	  Standard Orifice Knife Edge Orifice
Needle Valve		
Non-Compensated Flow Control Valve		
Pressure and Temperature Compensated Flow Control	  ANSI Symbol ISO Symbol	

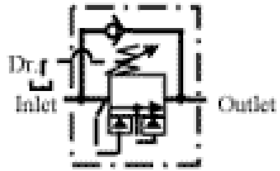
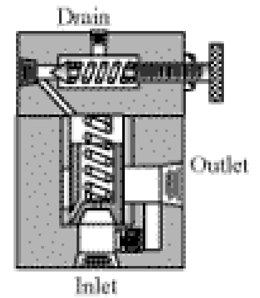
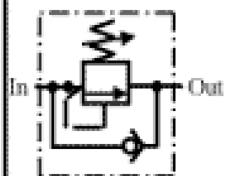
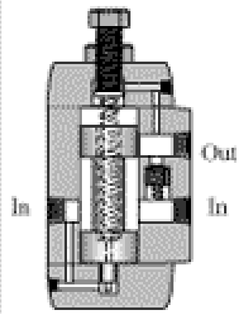
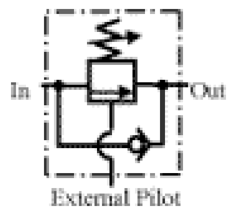
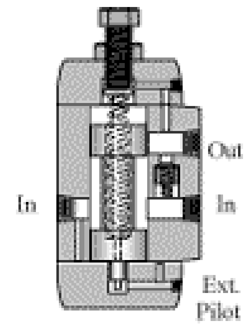
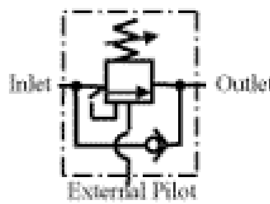
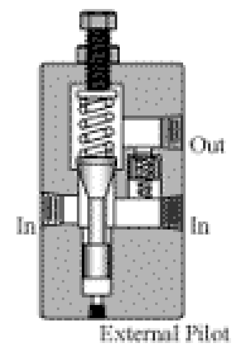
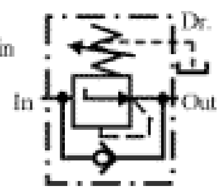
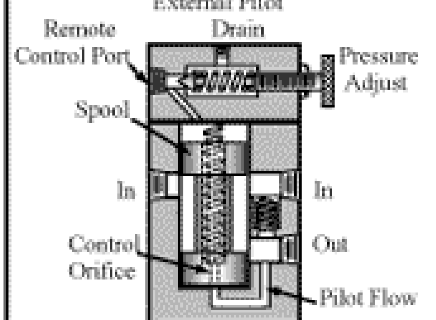
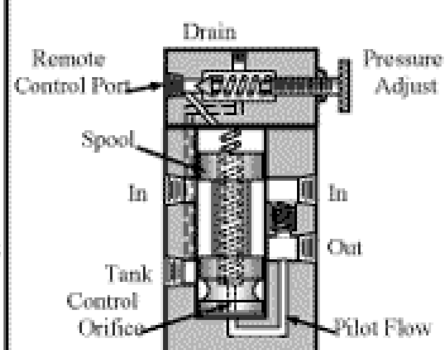
<table>
<tr><th>ISO Designation</th><th>Symbol</th><th>Picture Representation</th></tr>
<tr><td>Flow Control Valves (Cont'd.)

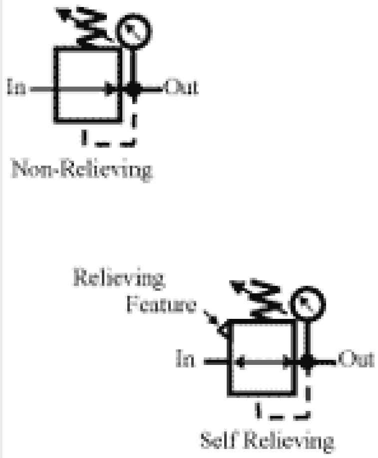
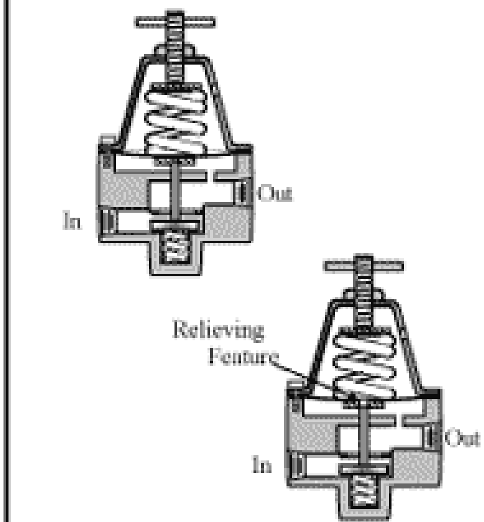
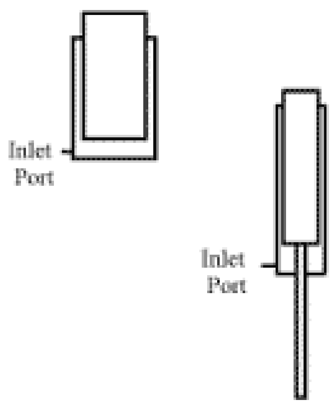
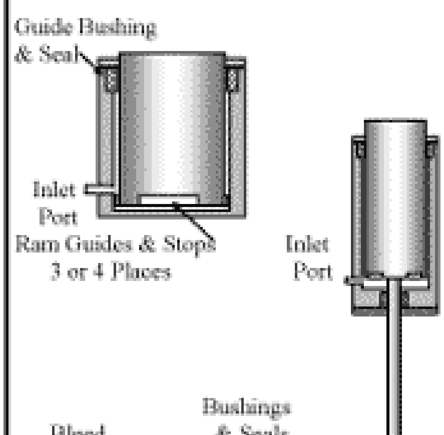
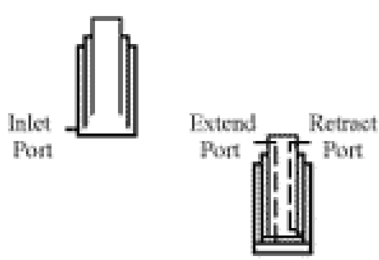
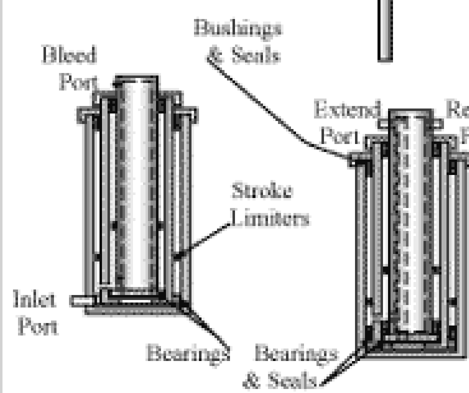
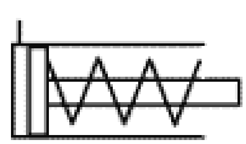
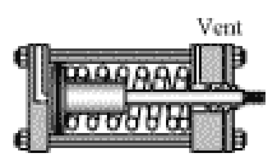
Proportional Flow Control Valves Without Feedback</td><td>
</td><td>
</td></tr>
<tr><td>With Feedback</td><td>
</td><td>
</td></tr>
<tr><td>Three Port Flow Control With Built-in Relief Valve.
Also Called a Bypass Flow Control.
Bypass Flow Cannot be Used to do Work</td><td>
</td><td>
</td></tr>
<tr><td>Priority Type Three Port Flow Control
Bypass Flow can be Used to do Work</td><td>

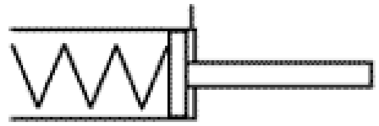
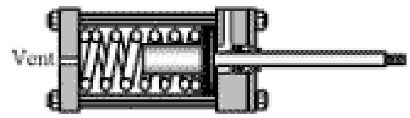
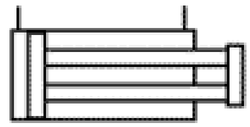
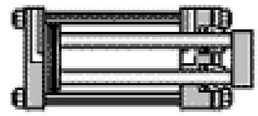
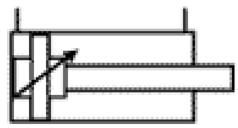
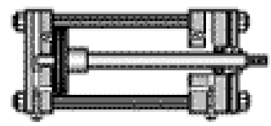
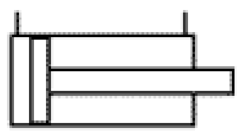
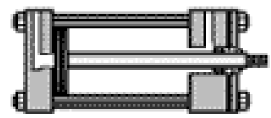
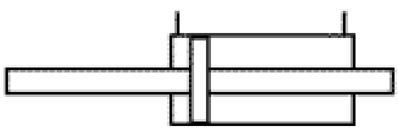
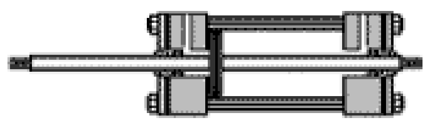
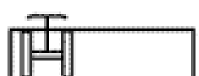
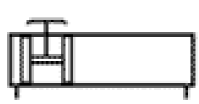
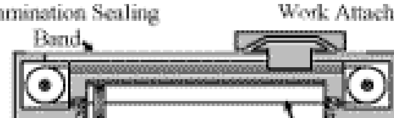

</td><td>

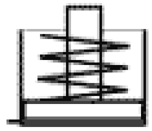
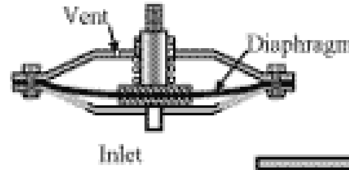
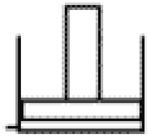
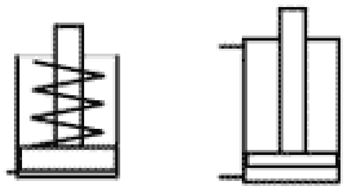
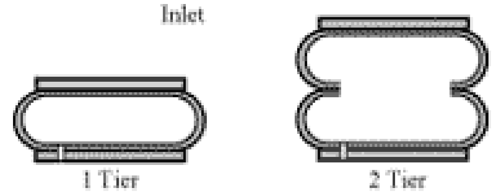
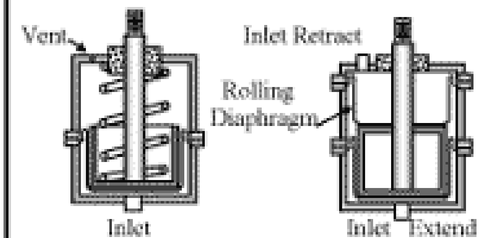
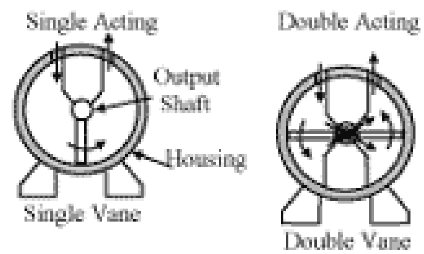
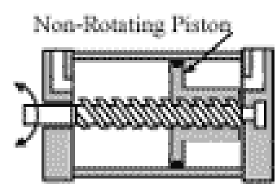
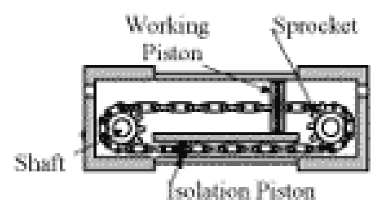
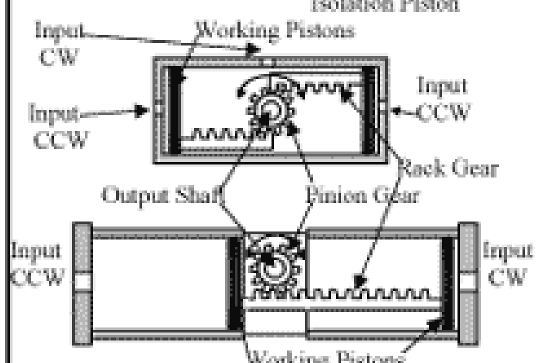

</td></tr>
<tr><td>Flow Dividers
Spool Type
One Direction Only</td><td>
</td><td>
</td></tr>
</table>

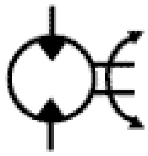
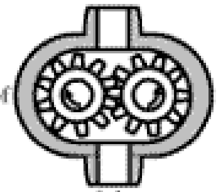
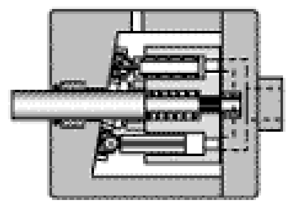
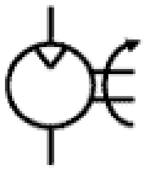
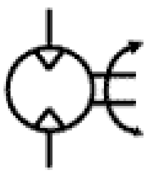
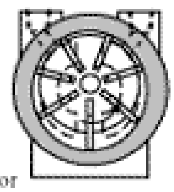
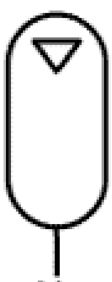
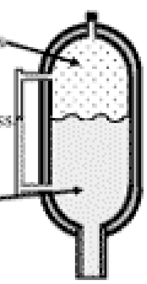
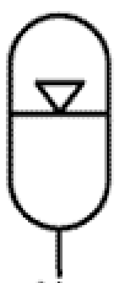
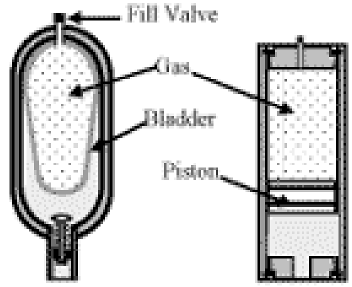
ISO Designation	Symbol	Picture Representation
Flow Control Valves (Cont'd.) Flow Dividers Spool Type Bi-directional (Functional Symbol)		
Flow Dividers Motor Type		
		
Pressure Control Valves Chapter 14 (Except Relief and Unloading Valves) Sequence Valve Internal Pilot		
External Pilot		
Internal Pilot Without Check Valve		

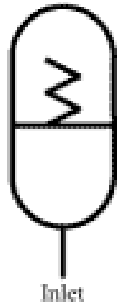
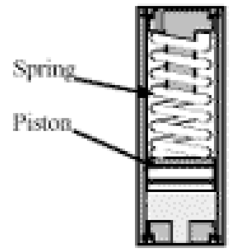
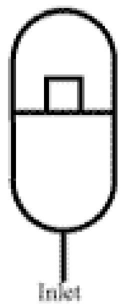
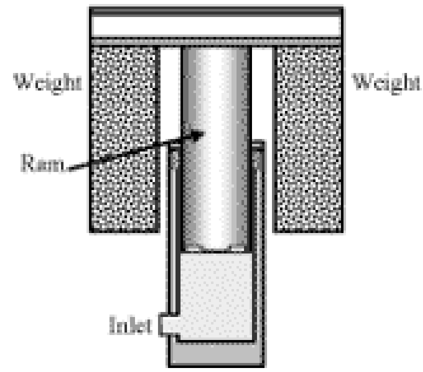
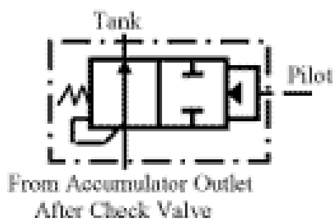
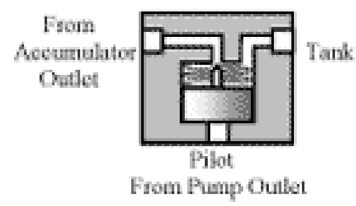
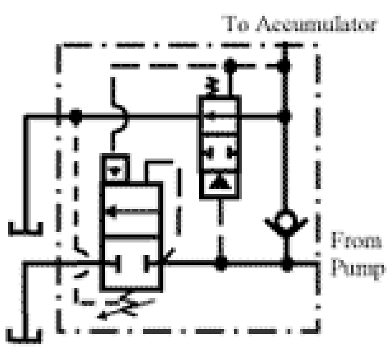
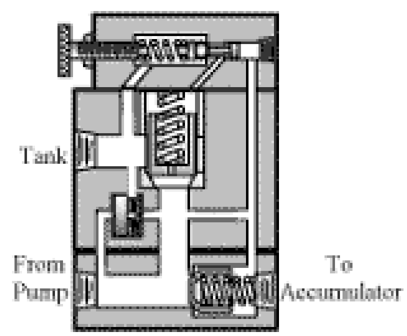
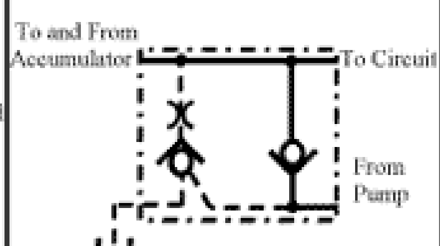
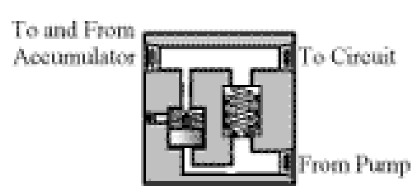
ISO Designation	Symbol	Picture Representation
Pressure Control Valves (Except Relief and Unloading Valves) (Cont'd.)		
Kick Down Sequence Valve		
Counter Balance Valve		
Internal Pilot Internal Drain		
External Pilot Internal Drain		
Internal and External Pilot, Internal Drain		
Reducing Valves		
Reducing Only With and Without Bypass Check	 	
Reducing, Relieving Without Bypass Check		
With Bypass Check With External Drain	 	

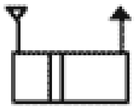
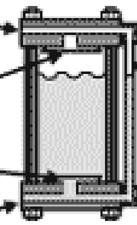
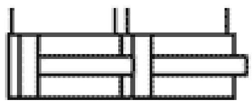
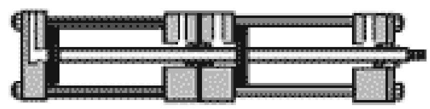
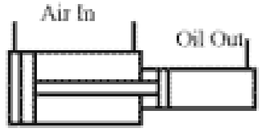
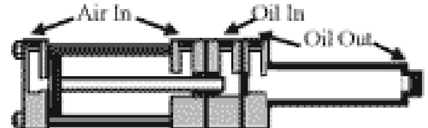
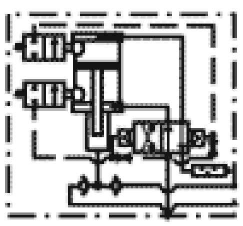
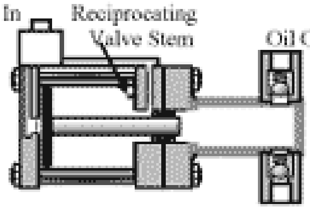
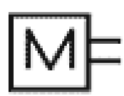
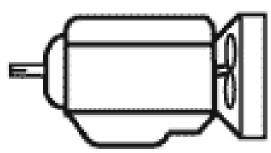
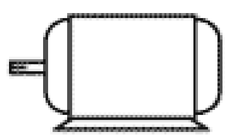
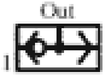
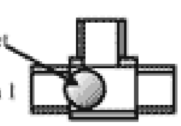
ISO Designation	Symbol	Picture Representation
Pressure Control Valves (Except Relief and Unloading Valves) (Cont'd.) Air Line Regulator (Reducing Valve)		
Actuators Chapter 15 Single Acting Rams Push Type The Inlet Port Can be Anyplace on the Body The Only Seals are Where the Rams Exit The Body. Pull Type May Have a Cover Over the Top Ram to Protect Personnel		
Single Acting Telescoping Double Acting Telescoping		
Single Acting Spring Return		

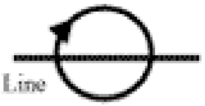
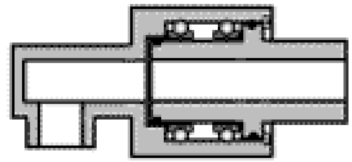
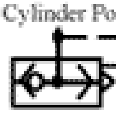
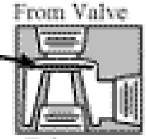
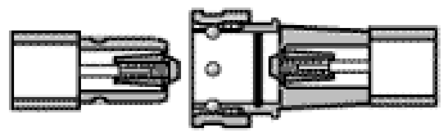
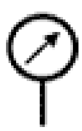
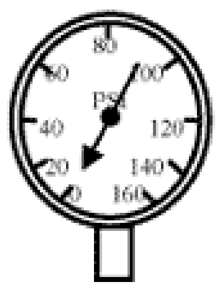
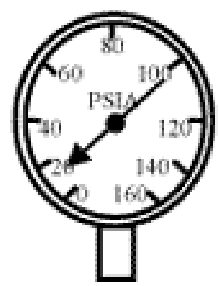
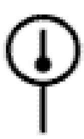
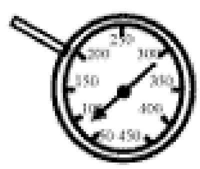
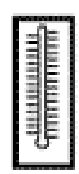
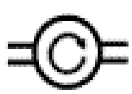
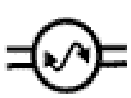
ISO Designation	Symbol	Picture Representation
Actuators (Cont'd.)		
Single Acting Spring Extend		
Non-rotating Rod Cylinder (Twin Rod)	 Symbol From Manufacturers Catalog	
Double Acting Cylinder With Cushions Both Ends. Arrow through Cushion Blocks Indicate They Are Adjustable. May Have Cushions Either End, Neither End or Both Ends..		
Double Acting Cylinder Non-Cushioned		
Double Acting, Double Rod End Cylinder Non-Cushioned		
Cable Cylinder	 Proposed Symbol	
Rodless Cylinder (Slot Type)	 Symbol From Manufacturers Catalog	
Rodless Cylinder (Band Type)		

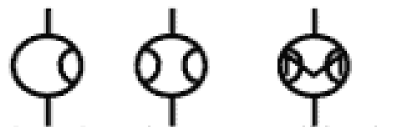
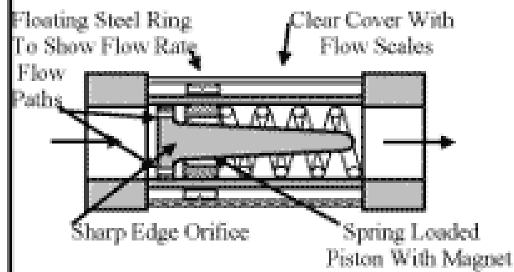
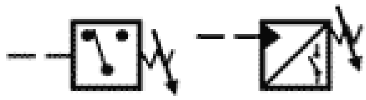
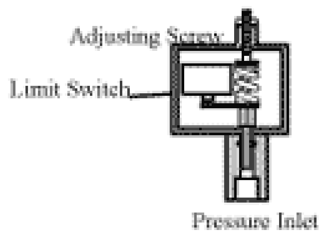
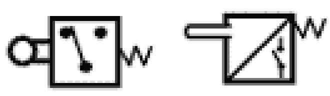
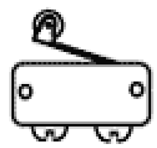
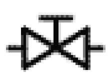
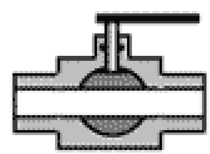
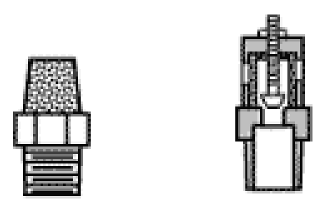
ISO Designation	Symbol	Picture Representation
Actuators (Cont'd.) Diaphragm Type		
Bellows Type	 	 
Rotary Actuators Vane Type	 Pneumatic	
Helical Screw		
Chain & Sprocket	 Hydraulic	
Rack & Pinion		

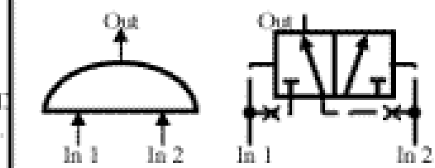
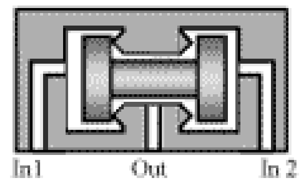
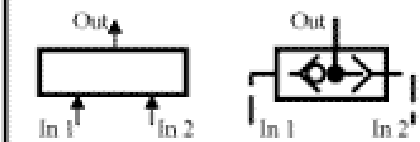
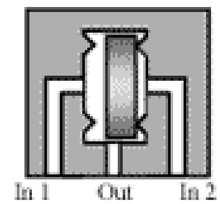
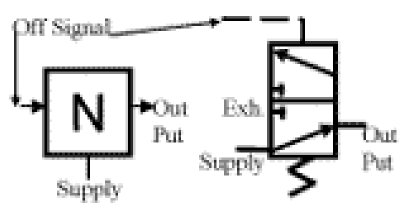
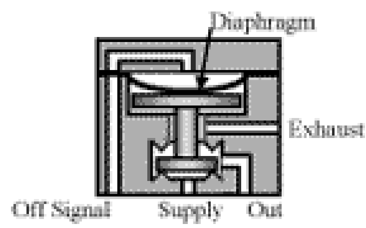
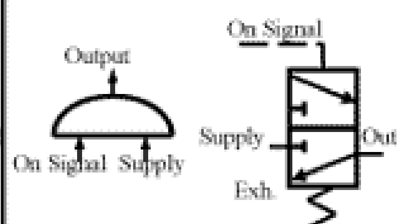
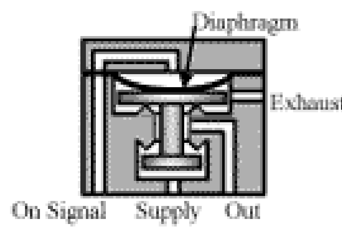
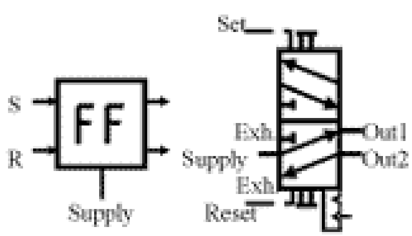
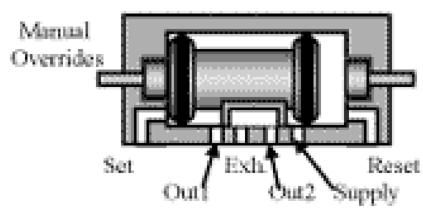
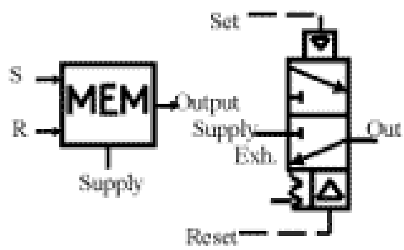
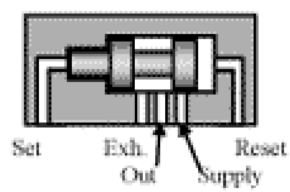
ISO Designation	Symbol	Picture Representation
Actuators (Cont'd.) Hydraulic Motors Fixed Displacement Bi-directional		Outlet Porting For This Direction of Rotation  Inlet Gear Motor
Variable Volume Bi-directional		Drain  Piston Motor
Air Motors Uni-directional		
Bi-directional		 Vane Motor
Accumulators Chapter 16 Gas Charged Non -separated	 Inlet	 Inlet
Gas Charged Bladder or Piston Type	 Inlet	 Inlet Inlet

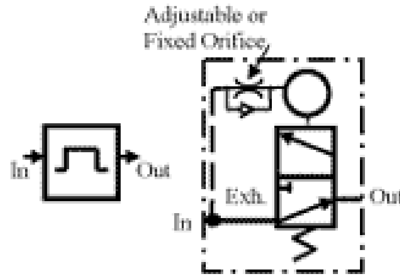
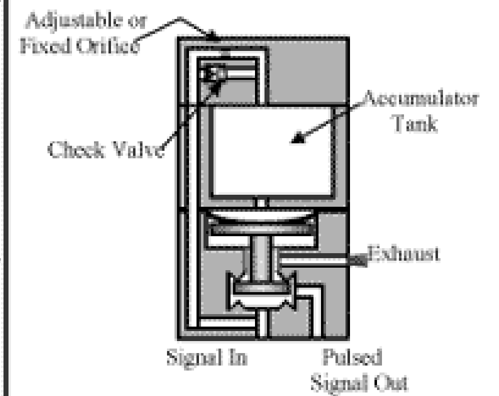
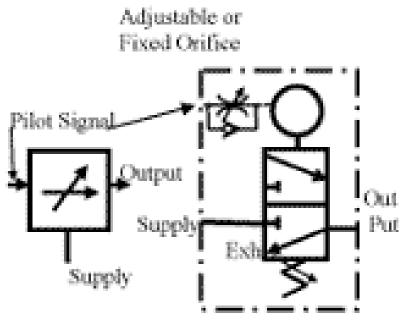
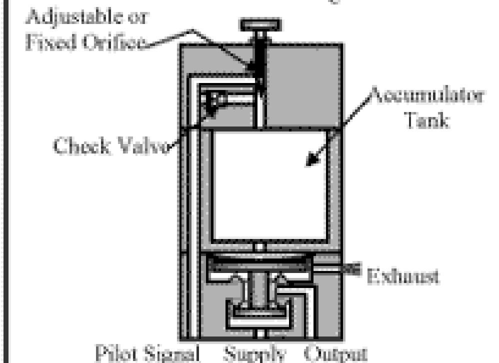
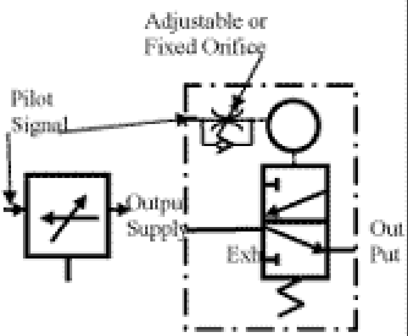
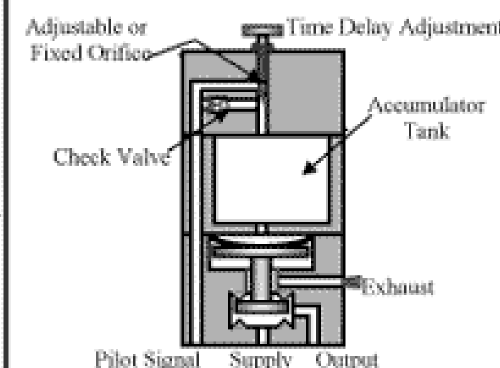
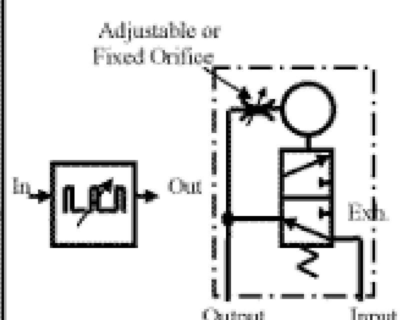
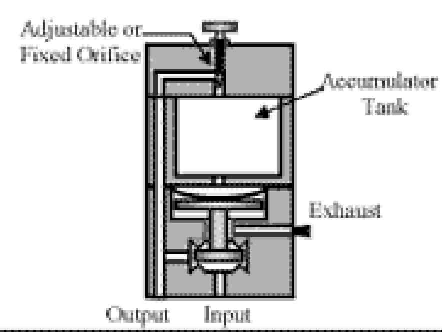
ISO Designation	Symbol	Picture Representation
Accumulators (Cont'd.) Spring Loaded		
Weight Loaded		
Accumulator Dump Valve (High Ratio)		
Accumulator Unloading Relief Valve and Dump Valve For Fixed Volume Pumps		
Accumulator Dump and Isolation Check Valve For Pressure Compensated Pumps		

ISO Designation	Symbol	Picture Representation
Air-Oil & Intensifiers Chapter 17		
Air-Oil Tank Changing a Pneumatic Pressure into an Equal Liquid Pressure		
Tandem Cylinder		
Intensifier or Booster	 Single Shot Intensifier	 Three Head Intensifier
Air-Oil Intensifier Reciprocating Type Self Contained		
Miscellaneous Fluid Power Items Chapter 18		
Heat Motor		
Electric Motor		
Shuttle Valve		

ISO Designation	Symbol	Picture Representation
Miscellaneous Fluid Power Items (Cont'd.)		
Rotating Union	Working Flow Line 	
Quick Exhaust	To Cylinder Port From Valve  Exhaust	From Valve Shut-off Wafer  To Cylinder Port Exhaust
Quick Disconnects Without Shut-offs		 Plug Socket
Shown Dis-connected		
Shown Connected		
Shown Without Shut-offs		
Pressure Gauges		 Gauge Pressure  Absolute Pressure
Temperature Gauges		 Probe Gauge  Surface Gauge Often Part of Tank Sight Gauge
Tachometer Measurement of Rotation in RPM		
Torque Meter Measurement of Torque		

ISO Designation	Symbol	Picture Representation
Miscellaneous Fluid Power Items (Cont'd.)		
Liquid or Gas Flow Meter	 Flow Only Flow Meter Totalizing Flow	
Pressure Switch	 ANSI ISO	
Limit Switch	 Roller Plunger	
Shut-off Valve		
Muffler (Silencer)	 Muffler With Speed Control	
Machine Member To Show Contact by a Valve or Other Device With a Machine Part		 Shows Valve is in Shifted Condition at Rest

ISO Designation	Symbol	Picture Representation
Air Logic Chapter 19	**ANSI B93-38 1976** **ISO Graphic Symbol**	
"AND" Element A Signal at "In 1" AND "In 2" Gives an Output. Output Always Lowest Of Input Signals	 	
"OR" Element A Signal at "In 1" OR "In 2" Gives an Output. Output Always Highest Input Signals	 	
"NOT" Element 3-Way Normally Open "Supply" Goes to Output Until an "OFF Signal" is Received.	 	
"YES" Element 3-Way Normally Closed Similar to "AND" But Can Choose "Output" Signal	 	
"FLIP-FLOP" Element 4-Way Double Pilot Operated Supplies and Exhaust's Either of Two Outputs	 	
"MEMORY" Element 3-Way Normally Closed Supplies or Exhaust's a Single Port. Can be "Reset" While "Set" Signal is on.	 	

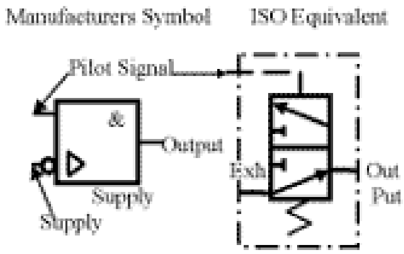
ISO Designation	Symbol	Picture Representation
Air Logic (Cont'd.)	**ANSI B93-38 1976** **ISO Graphic Symbol**	
"DIFFERENTIATOR" Element 3-Way Normally Open (Also Called a "ONE SHOT" Element or an ("IMPULSE GENERATOR") A Fixed Signal in Gives a Fixed or Adjustable Timed Signal Out		
"ON DELAY TIMER" Element 3-Way Normally Closed An. "Output" is Given After a "Pilot Signal" Has Been on for a Fixed or Adjustable Time Delay.		
"OFF DELAY TIMER" Element 3-Way Normally Open The "Output" is Lost After a "Pilot Signal" Has Been on for a Fixed or Adjustable Time Delay		
"FREQUENCY GENERATOR" Element 3-Way Normally Open An "Input" Signal Gives a Fixed or Adjustable Output for One Time Period, Then it Goes Off for Two Time Periods.		

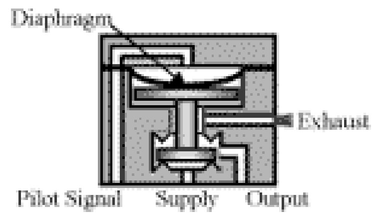
ISO Designation	Symbol	Picture Representation

Air Logic (Cont'd.)

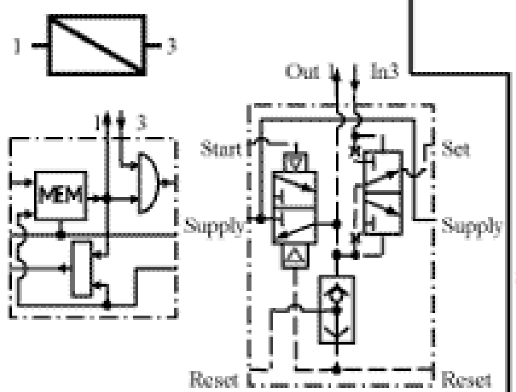
"PRESSURE DECAY SENSOR" Element
3-Way Normally Open
Same as a "NOT" except it Takes a lower "Pilot Signal" to Shift it.

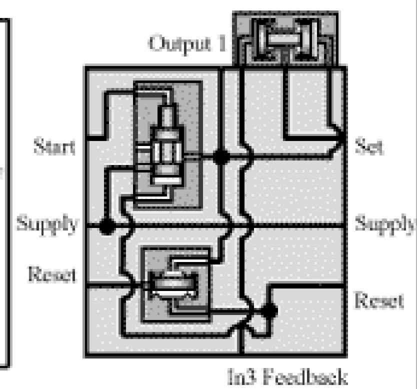
Used for Indicating end of Stroke of Cylinders.



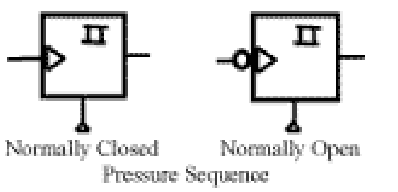


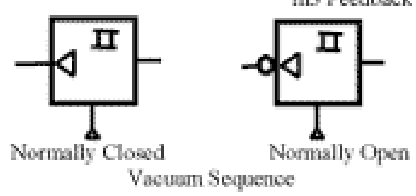
"SEQUENCER" Element
A Combination of an "AND", "OR" and a "MEMORY" Element in a Common Housing to Simplify Circuit Design. Used in Series To Make a Complete Operating System.



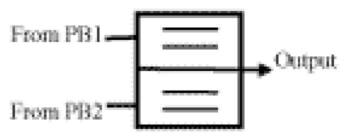


"SEQUENCE" Element
Gives or Drops a Signal at a pre-determined Adjustable Pressure.



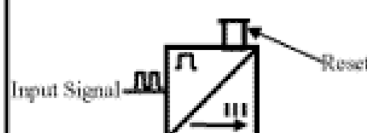


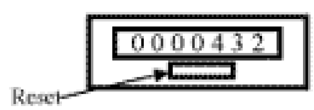
"ANTI TIEDOWN" Element
Requires Two Inputs to Get an Output



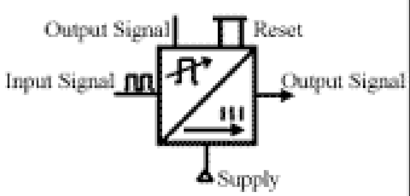
Counters:

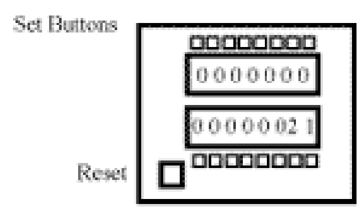
Counts Number of Cycles



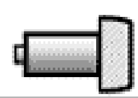


Counts Down From Pre-selected Number Then Gives an Output





Pressure Indicator



Absolute viscosity *- the ratio of shear stress to shear rate. It is a fluid's internal resistance to flow. The common unit of absolute viscosity is the poise. Absolute viscosity divided by fluid density equals kinematic viscosity.*

Absorption *- The physical mechanism by which one substance attracts and takes up another substance (liquid, gas, or vapor) into its interior.*

Accelerator *- A substance that hastens the vulcanization of an elastomer, causing it to take place in a shorter time period or at a lower temperature.*

Accumulator *-A container in which fluid is stored under pressure as a source of fluid power.*

Accumulator, hydro-pneumatic bladder *- A hydro-pneumatic accumulator in which the liquid and gas are separated by an elastic bag or bladder.*

Actuator, pneumatic/hydraulic *- A device in which power is transferred from one pressurized medium (pneumatic) to another (hydraulic) without intensification.*

Additive - *A chemical added to a fluid to impart new properties or to enhance those that already exist.*

Adiabatic compression *occurs when no heat is transferred to or from the air during compression.*

Adsorption *- The physical mechanism by which one substance attracts another substance (either solid, liquid, gas, or vapor) to its surface and through molecular forces causes the incident substance to adhere to that surface.*

Aftercooler *- A device that cools a gas after it has been compressed.*

Aftercooling *- The cooling of air after it has been compressed to lower its temperature and precipitate condensed vapors.*

Afterfilter *- A filter that follows the compressed air dryer, usually for the protection of downstream equipment from desiccant dust.*

Air bleeder *- A device for removal of air.*

Air breather *- A device permitting air movement between atmosphere and the component in which it is installed.*

Air motor *- A device that converts pneumatic fluid power into mechanical torque and motion. It usually provides rotary mechanical motion.*

Air, compressed (pressure) *- Air at any pressure greater than atmospheric pressure.*

Air, dried *- Air with moisture content lower than the maximum allowable for a given application.*

Air, free *- Air at ambient temperature, pressure, relative humidity, and density.*

Air, saturated *- Air at 100% relative humidity, with a dew point equal to temperature.*

Air, standard *- Air at a temperature of 68.8°F, a pressure of 14.70 pounds per square inch absolute, and a relative humidity of 36% (0.0750 pounds per cubic foot). In gas industries the temperature of "standard air" is usually given as 60.8°F. producing tiny bubbles that expand explosively at the pump outlet, causing metal erosion and eventual pump destruction.*

Air *- A gas mixture consisting of nitrogen, oxygen, argon, carbon dioxide, hydrogen, small quantities of neon, helium and other gases.*

Amplification, power *- The ratio between the output signal variations and the corresponding input (control) power variation (for analog devices only).*

Amplification, pressure *- The ratio between outlet pressure and inlet (control) pressure.*

Amplification *- The ratio between the output signal variations and the control signal variations (for analog devices only)*

Analog *- Of or pertaining to the general class of fluidic devices or circuits whose output varies as a continuous function of its input.*

AND device *- A control device that has its output in the logical 1 state if and only if all the control signals assume the logical 1 state.*

Aniline point *- The lowest temperature at which a liquid is completely miscible with an equal volume of freshly distilled aniline (ASTM Designation D611-64).*

Anti-foam agent *- One of two types of additives used to reduce foaming in petroleum products: silicone oil to break up large surface bubbles, and various kinds of polymers that decrease the number of small bubbles entrained in the oils.*

Asperities *- Microscopic projections on metal surfaces resulting from normal surface-finishing processes. Interference between opposing asperities in sliding or rolling applications is a source of friction, and can lead to metal welding and scoring. Ideally, the lubricating film between two moving surfaces should be thicker than the combined height of the opposing asperities.*

Bactericide *- An additive included in the formulations of water-mixed fluids to inhibit the growth of bacteria.*

Bernoulli's Law *- If no work is done on or by a flowing frictionless liquid, its energy due to pressure and velocity remains constant at all points along the streamline.*

Beta Ratio *- A rating applied to filters (it is also known as the "Filtration Ratio." It is a measure of the particle-capture efficiency of a filter element.*
The ISO 4572 Multipass Test Procedure passes fluid through a test circuit to check for contaminant retention. A measured amount of contaminant is injected upstream of the filter and laser particle counters record particles in to particles out across the filter. When 100,000 particles are measured upstream of a 10(filter and 10,000 downstream it would have a Beta Ratio of 10. (B10(=100,000/10,000=10)

A Beta Ratio number is of no use alone but it is required to find the filter's efficiency rating. Efficiency of a filter element is what counts when comparing one filter to another. The higher the efficiency, the fewer contaminants get by. Efficiency coupled with the volume of contaminant retention can make a more expensive filter cost less due to its longer useful life.

Efficiency is figured by the formula:

- *Efficiency = (1 1/Beta) 100*
- *Efficiency10 = (1 1/10) 100*

Bleeding *- Migration to the surface of plasticizers, waxes, or similar materials to form a film or beads.*

Boundary lubrication *- A form of lubrication between two rubbing surfaces without development of a full-fluid lubricating film. Boundary lubrication can be made more effective by including additives in the lubricating oil that provide a stronger oil film, thus preventing excessive friction and possible scoring. There are varying degrees of boundary lubrication, depending on the severity of service.*

Boyle's Law *- The absolute pressure of a fixed mass of gas varies inversely as the volume, provided the temperature remains constant.*

Breakout *- The force necessary to inaugurate sliding. Expressed in the same terms as friction. An excessive breakout value indicates the development of adhesion.*

Breathing capacity *- A measure of flow rate through an air breather.*

Bulk modulus *- The measure of a fluid's resistance to compressibility. It is the reciprocal of compressibility.*

Cavitation *- A localized gaseous condition within a liquid stream that occurs where the pressure is reduced to the liquid's vapor pressure, often as a result of a solid body, such as a propeller or piston, moving through the liquid. Also, the pitting or wearing away of a solid surface as a result of low fluid levels that draw air into the system.*

Charles' Law *- The volume of a fixed mass of gas varies directly with absolute temperature, provided the pressure remains constant.*

Circuit, meter-in *- A speed-control circuit in which the control is achieved by regulating the supply flow to the actuator.*

Circuit, meter-out *- A speed-control circuit in which the control is achieved by regulating the exhaust flow from the actuator.*

Circuit, open *- A circuit in which return fluid is directed to the reservoir before reciprocation.*

Circuit, regenerative *- A circuit in which pressurized fluid discharged from a component is returned to the system to reduce input power requirements.*

Circuit, sequence *- A circuit that establishes the order in which two or more phases of a circuit occur.*

Circuit *- An arrangement of interconnected components and parts.*

Cold flexibility *- Flexibility following exposure to a predetermined time.*

Cold flow *- Continued deformation under stress.*

Compatibility, seal *- Ability of an elastomer to resist the action of a fluid on its dimensional and mechanical properties.*

Compressibility *- The change in volume of a unit volume of a fluid when subjected to a unit change in pressure.*

Compression efficiency *- The ratio of the theoretical work required (in a given process) to the actual work required to be done to compress and de-liver the air. Expressed as a percentage. Compression efficiency accounts for fluid-friction losses, leakage, and thermodynamic variations from the theoretical process.*

Compression modulus *- The ratio of the compressive stress to the resulting compressive strain (the latter expressed as a fraction of the original height or thickness in the direction of the force). Compression modulus may be either static or dynamic.*

Compression ratio *- The ratio of the absolute discharge pressure to the absolute intake pressure.*

Compression set *- The amount by which a rubber specimen fails to return to original shape after release of the compressive load.*

Compressor *- A device that converts mechanical force and motion into pneumatic fluid power.*

Condensation *- The process of changing a vapor into a liquid condensate by the extraction of heat.*

Conditioner, air *- An assembly comprising a filter, a pressure-reducing valve with gauge, and a lubricator, intended to deliver compressed air in suitable condition for its application.*

Conductor *- A component whose primary function is to contain and direct fluid.*

Contaminant *- Any material or substance that is unwanted or adversely affects the fluid power system or components, or both.*

Control *- A device used to regulate the function of a component or system.*

Controller *- A device that senses a change of fluid state and automatically makes adjustments to maintain the state of the fluid between predetermined limits, e.g., pressures, temperatures, etc.*

Copolymer *- A polymer consisting of two different monomers that are chemically combined.*

Corrosion inhibitor *- An additive that protects wetted metal surfaces from chemical attack by water or other contaminants. Polar compounds wet the metal surface preferentially, protecting it with a film of oil. Other compounds may absorb water by incorporating it in a water-in-oil emulsion so that only the oil touches the metal surface. Another type of corrosion inhibitor combines chemically with the metal to present a non-reactive surface.*

Creep *- The progressive relaxation of a given rubber material while it is under stress. This relaxation eventually results in permanent deformation or "set."*

Cushion *- A device that provides controlled resistance to motion.*

Cylinder *- A device that converts fluid power into linear mechanical force and motion. It usually consists of a movable element such as a piston and piston rod, plunger or ram, operating within a cylindrical bore.*

Cylinder, adjustable stroke *- A cylinder equipped with adjustable stops at one or both ends to limit piston travel.*

Cylinder, area, piston, effective *- The area upon which fluid pressure acts to provide a mechanical force.*

Cylinder, area, piston rod *- The cross-sectional area of the piston rod.*

Cylinder, bore *- The internal diameter of the cylinder body.*

Cylinder cap *- An end closure for a cylinder that completely covers the bore area.*

Cylinder capacity *- The volume of a theoretically incompressible fluid that would be displaced by the piston during a complete stroke. (For double-acting cylinders it must be given for both directions of stroke.)*

Cylinder capacity, extending *- The volume required for one full extension of a cylinder.*

Cylinder capacity, retracting *- Volume (annular) absorbed by one full retraction of the cylinder oscillation of an output shaft.*

Cylinder, cushioned *- A cylinder with a piston-assembly deceleration device at one or both ends of the stroke.*

Cylinder, differential *- A double-acting cylinder in which the ratio of the area of the bore to the annular area between the bore and the piston rod is significant in circuit function.*

Cylinder, double-acting *- A cylinder in which fluid force can be applied to the moveable element (piston) in either direction.*

Cylinder, double rod *- A cylinder with a single piston and a piston rod extending from each end.*

Cylinder, dual stroke *- A cylinder combination that provides two working strokes.*

Cylinder, duplex *- A unit comprised of two cylinders with independent control, mechanically connected on a common axis to provide three or four positions depending on the method of application.*

Cylinder force, theoretical *- The pressure multiplied by the effective piston area (ignoring friction). For double-acting cylinders, the value must be given for both directions of stroke.*

Cylinder, piston-type *- A cylinder in which the piston has a greater cross-sectional area than the piston rod.*

Cylinder, plunger (ram) *- A cylinder in which the piston has the same cross-sectional area as the piston rod.*

Cylinder, rotating *- A cylinder in which the piston and piston rod, plunger or ram, is permitted to rotate with reference to the cylinder housing.*

Cylinder, single-acting *- A cylinder in which the fluid force can be applied to the movable element in only one direction.*

Cylinder, tandem *- An arrangement of at least two pistons on the same rod moving in separate chambers on the same cylinder body allowing the compounding of force on the piston rod.*

Cylinder, telescoping *- A cylinder with two or more stages or extensions, achieved by hollow piston rods sliding one within the other (may be single- or double-acting).*

Cylinder, tie rod *- A cylinder with head and cap end closures that are secured by tie rods.*

***Cylinder, rotary actuator** - A cylinder that translates piston reciprocation into oscillation of an output shaft.*

***Darcy's Formula** - A formula used to determine the pressure drop due to flow friction through a conduit.*

***Deliquescent** -A special hygroscopic compound with absorptive properties that can separate moisture.*

***Demulsibility** - The ability of an oil to separate from water.*

***Density** - The weight of a given volume of air, usually expressed in lb/ft3 at standard temperature and pressure.*

***Desiccant** - A material that tends to remove moisture from compressed air.*

***Dew point** - The temperature at which vapors in a gas condense. For practical purposes, it must be referred to a stated pressure.*

***Dewaxing** - Removal of paraffin wax from lubricating oils to improve low-temperature properties, especially to lower the cloud point and pour point.*

***Digital** - Of or pertaining to the general class of fluidic devices or circuits whose output varies in discrete steps (i.e., pulses or "on-off" characteristics).*

***Displacement** - The net volume swept by the moving parts of a compressor per unit of time. This term applies only to positive-displacement compressors.*

***Displacement, volumetric** - The volume absorbed or displaced per stroke of a cylinder or per cycle of a pump or motor.*

***Dissolved air** - Air that is dispersed at a molecular level in hydraulic fluid to form a single phase.*

***Dissolved water** - Water that is dispersed at a molecular level in hydraulic fluid to form a single phase.*

***Dither** - A low-amplitude, relatively high-frequency periodic electrical signal, some-times superimposed on a servovalve input to improve system resolution. Dither is expressed by the dither frequency (Hz) and the peak-to-peak dither current amplitude.*

***Droop** - The deviation between no flow secondary pressure and secondary pressure at a given flow.*

***Dryer, compressed air** - A device for reducing the moisture content of the working compressed air.*

Durometer *- 1. An instrument for measuring the hardness of rubber. It measures the resistance to the penetration of an indenter point into the surface of rubber. 2. Numerical scale of rubber hardness.*

EP additive *- A lubricant additive that prevents sliding metal surfaces from seizing under conditions of extreme pressure (EP). At the high local temperatures associated with metal-to-metal contact, an EP additive com-bines chemically with the metal to form a surface film that prevents the welding of opposing asperities, and the consequent scoring that is destructive to sliding surfaces under high loads. Reactive compounds of sulfur, chlorine, or phosphorus are used to form these inorganic films.*

Efficiency *- The ratio of output to the corresponding input.*

Elasticity *- The property of a material that tends to return to its original shape after deformation.*

Elastomer *- Any synthetic or natural material with resilience or memory sufficient to return to its original shape after distortion.*

Elongation *- Generally means "ultimate elongation" or percent increase in original length of a specimen when it breaks.*

Emulsion, oil-in-water *- A dispersion of oil in a continuous phase of water.*

Emulsion, water-in-oil *- A dispersion of water in a continuous phase of oil.*

Emulsifier *- An additive that promotes formation of a stable mixture . . . or emulsion . . . of oil and water.*

Emulsion *- A homogeneous dispersion of two immiscible liquids, generally of a milky or cloudy appearance.*

Entrained air *- A mechanical mixture of air bubbles having a tendency to separate from the liquid phase.*

Expectancy, life *- The predicted working period during which a component or system will maintain a specified level of performance under specified conditions. Sometimes expressed in statistical terms as a probability.*

Filter *- 1. A device whose primary function is the removal by porous media of insoluble contaminants from a liquid or a gas. 2. Chemically inert, finely divided material added to the elastomer to aid in processing and improve physical properties.*

Filter, strainer *- A coarse hydraulic filter, usually of woven wire construction. This may be in the form of a complete filter or just an element.*

Filter, bypass (reserve) *- A filter which provides an alternate unfiltered flow path around the filter element when a preset differential pressure is reached.*

Filter, spin-on *- A filter with spin-on element sealed in its own pressure housing for independent mounting to the filter.*

Filtration ratio (bm) *- The ratio of the number of particles greater than a given size (b) in the influent fluid to the number of particles greater than the same size (m) in the effluent fluid.*

Filter, strainer *- A coarse hydraulic filter usually of woven wire construction. This may be in the form of a complete filter or just an element.*

Fire-resistant fluid *- Hydraulic oil used especially in high-temperature or hazardous applications. Three common types of fire-resistant fluids are: (1) water-petroleum oil emulsions, in which the water prevents burning of the petroleum constituent; (2) water-glycol fluids; and (3) non-aqueous fluids of low volatility, such as phosphate esters, silicones, polyolesters, and halogenated hydrocarbon-type fluids.*

Fitting *- A connector or closure for fluid power lines and passages.*

Fitting, compression *- A fitting that seals and grips by manual adjustable deformation.*

Fitting, flange *- A fitting that utilizes a radially extending collar for sealing and connection.*

Fitting, flared *- A fitting that seals and grips by a pre-formed flare at the end of the tube.*

Fitting, flareless *- A fitting that seals and grips by means other than a flare.*

Flash point *- The temperature to which a liquid must be heated under specified conditions of the test method to give off sufficient vapor to form a mixture with air that can be ignited momentarily by a flame.*

Flip flop *- A digital component or circuit with two stable states and sufficient hysteresis so that it has "memory." Its state is changed with a control pulse; a continuous control signal is not necessary for it to remain in a given state.*

Flow characteristic curve *- The change in regulated (secondary) pressure occurring as a result of a change in the rate of air flow over the operating range of the regulator.*

Flow rate *- The volume, mass, or weight of a fluid passing through any conductor per unit of time.*

Flow, laminar (streamline) *- A flow situation in which fluid moves in parallel lamination or layers.*

Flow, output *- The flow rate discharged at the outlet port.*

Flow, turbulent - *A flow situation in which the fluid particles move in a random fluctuating manner.*

Flow - *Movement of fluid generated by pressure differences.*

Fluid capacity - *The liquid volume coincident with the "high" mark of the level indicator.*

Fluid friction - *Friction due to the viscosity of fluids.*

Fluid logic - *A branch of fluid power associated with digital signal sensing and information processing, using components with or without moving parts.*

Fluid miscibility - *Capacity of fluids to be mixed in any ratio without separation into phases.*

Fluid power system - *A system that transmits and controls power through use of a pressurized fluid within an enclosed circuit.*

Fluid power - *Energy transmitted and controlled through use of a pressurized fluid.*

Fluid stability - *The resistance of a fluid to permanent changes in properties.*

Fluid stability, oxidation - *The resistance of a fluid to permanent changes caused by chemical reaction with oxygen.*

Fluid, anti-corrosive - *A fluid containing metal corrosion inhibitors.*

Fluid, aqueous - *A fluid that contains water as a major constituent besides the organic material. The fire-resistance properties are derived from the water content.*

Fluid, fire-resistant - *A fluid that is difficult to ignite and shows little tendency to propagate flame.*

Fluid, hydraulic - *A fluid suitable for use in a hydraulic system.*

Fluid, Newtonian - *Fluid having a viscosity that is always independent of the rate of shear.*

Fluid, pneumatic - *A fluid suitable for use in pneumatic systems . . . usually air.*

Fluid, rust protection - *Capacity of a fluid to prevent the formation of rust under specified conditions.*

Fluid - *A liquid, gas, or combination thereof.*

Force motor - *A type of electromechanical transducer having linear motion used in the input stages of servovalves.*

Free air *- Any compressible gas, air, or vapor trapped within a hydraulic system that does not condense or dissolve to form a part of the system fluid.*

Free water *- Water droplets or globules in the system fluid that tend to accumulate at the bottom or top of the system fluid depending on the fluid's specific gravity.*

Frequency response *- The changes, under steady-state conditions, in the output variable that are caused by a sinusoidal input variable.*

Full-fluid-film lubrication *- The presence of a continuous lubricating film sufficient to completely separate two surfaces (as distinct from boundary-layer lubrication). Full-fluid-film lubrication is normally hydrodynamic lubrication, whereby the oil adheres to the moving part and is drawn into the area between the sliding surfaces, where it forms a pressure or hydrodynamic wedge.*

Gauge damper (snubber) *- A device employing a fixed or variable restrictor inserted in the pipeline leading to a pressure gauge. It prevents damage to the gauge mechanism caused by rapid fluctuations of fluid pressure.*

Gauge protector *- A device inserted in the pipeline to a pressure gauge and arranged to isolate the pressure gauge from the fluid pressure if this pressure exceeds a predetermined limit. The device can usually be adjusted to suit the range of the pressure gauge.*

Gauge, bourdon tube *- A pressure gauge in which the sensing element is a curved tube that tends to straighten out when subjected to internal fluid pressure.*

Gauge, diaphragm *- A gauge in which the sensing element is relatively thin and its inner portion is free to deflect with respect to its fixed periphery.*

Gauge, instrument *- An instrument or device for measuring, indicating, or comparing a physical characteristic.*

Gauge, pressure *- A gauge that indicates the pressure in the system to which it is connected.*

Head *- The height of a column or body of fluid above a given point expressed in linear units. Head is often used to indicate gauge pressure. Pressure is equal to the height times the density of the fluid.*

Head, cylinder *- The cylinder end closure that covers the differential area between the bore area and the piston rod area.*

Head, friction *- The pressure required to overcome the friction at the interior surface of a conductor and between fluid particles in motion. It varies with flow, size, type and condition of conductors and fittings, and the fluid characteristics.*

Head, pressure *- The pressure due to the height of a column or body of fluid.*

Head, static *- The height of a column or body of fluid above a given point.*

Heat exchanger *- A device that transfers heat through a conducting wall from one fluid to another. (Typically to cool a system.)*

Heater *- A device that transfers heat through a conducting wall from one fluid to another. (Typically to warm up a system.)*

Hose, wire-braided *- Hose consisting of a flexible material reinforced with woven wire braid.*

Hose *- A flexible line or conductor whose nominal size is its inside diameter.*

Hydraulic amplifier *- A fluid device that enables one or more inputs to control a source of fluid power and thus is capable of delivering at its output an enlarged reproduction of the essential characteristics of the input. Hydraulic amplifiers may utilize sliding spools, nozzle-flappers, jet pipes, etc.*

Hydraulic fluid *- The fluid that serves as the power transmission medium in a hydraulic system. The most commonly used fluids are petroleum and synthetic oils, oil-water emulsions, and water-glycol mixtures. The principal requirements of a premium hydraulic fluid are: proper viscosity, high viscosity index, anti-wear protection (if needed), good oxidation stability, adequate pour point, good demulsibility, rust inhibition, resistance to foaming, and compatibility with seal materials. Anti-wear oils are frequently used in compact, high-pressure, and high-capacity pumps that require extra lubrication protection.*

Hydraulic motor *- A device that converts hydraulic fluid power into mechanical force and motion. It usually provides rotary mechanical motion.*

Hydraulic motor efficiency, hydro-mechanical *- The ratio of the effective torque to the derived torque.*

Hydraulic motor efficiency, overall *- The ratio of the output power to the effective power.*

Hydraulic motor efficiency, volumetric *- The ratio of the derived output flow to the effective input flow.*

Hydraulic motor, fixed displacement *- A hydraulic motor in which the displacement per unit of output motion cannot be varied.*

Hydraulic motor, flow, input *- The flow rate crossing the transverse plane of the inlet port.*

Hydraulic motor, gear, external *- A motor having two or more external gears.*

Hydraulic motor, gear, internal *- A motor with an internal gear in engagement with one or more external gears.*

Hydraulic motor, gear - *A motor in which two or more gears act in arrangement as working members.*

Hydraulic motor, vane - *A motor in which the fluid under pressure acting on a set of radial vanes causes rotation of an internal member.*

Hydraulic stepping motor - *A hydraulic motor that follows the commands of a stepped input signal to achieve positional accuracy.*

Hydraulics - *The engineering science that pertains to liquid pressure and flow.*

Hydrodynamics - *The engineering science that governs the movement of liquids and the forces opposing that movement.*

Hydrokinetics - *The engineering science that pertains to the energy of liquid flow and pressure.*

Hydro-pneumatics - *The combination of hydraulic and pneumatic fluid power.*

Hydrostatic transmission - *The combination of one or more hydraulic pumps and motors to form a unit.*

Hydrostatics - *The engineering science that pertains to the energy of liquids at rest.*

Ideal gases - *Gasses that follow the perfect gas laws without deviation. There are no real ideal gases, but they provide a common starting point for calculations and corrections.*

Immiscible - *Incapable of being mixed without separation of phases. Water and petroleum oil are immiscible under most conditions, although they can be made miscible with the addition of a proper emulsifier.*

Indicator, differential pressure - *An indicator that signals a difference in pressure between two points in a fluid power system.*

Inhibitor - *Any substance that, when present in very small proportions, slows, prevents or modifies chemical reactions such as corrosion or oxidation.*

Intensification, ratio of - *The ratio of the secondary pressure to the primary pressure or of the primary flow rate to the secondary flow rate.*

Intensifier, double-acting - *A unit that magnifies the secondary fluid pressure regardless of the direction of flow of the primary fluid.*

Intensifier, single-acting - *A unit that only magnifies the fluid pressure in one direction of flow of the primary fluid.*

***Intensifier, single shot** - An intensifier in which the continuous application of primary fluid at the inlet port can only give a limited volume of secondary fluid.*

***Intensifier** - A device that converts low-pressure fluid power into higher-pressure fluid power.*

***Inter-cooling** - The process of cooling air between stages of compression to liquefy condensed vapors and save power by reducing the temperature of air entering the next stage.*

***Isothermal compression** - A compression arrangement in which the temperature of the air remains constant during compression.*

***Joint** - A line-positioning connector.*

***Joint, rotary** - A joint connecting lines that have relative operational rotation.*

***Kinematic viscosity** - The absolute viscosity of a fluid divided by its density at the same temperature of measurement. It is the measure of a fluid's resistance to flow under gravity.*

***Leakage rate** - The rate at which a gas or liquid passes through a barrier. Total leakage rate includes the amounts that diffuse or permeate through the material of the barrier as well as the amount that escapes around it.*

***Line, return** - A conductor (pipe) to return the working fluid to the reservoir.*

***Line, working** - A line that conducts fluid power.*

***Line** - A tube, pipe, or hose for conducting fluid.*

***Lubricator** - A device that adds controlled or metered amounts of lubricants into a pneumatic system.*

***Lubricity** - The ability of an oil or grease to lubricate (also called film strength).*

***Magnetic plug** - A plug that attracts and holds ferromagnetic particles.*

***Manifold** - A conductor that provides multiple connection ports.*

***Maximum inlet pressure** - The maximum rated gauge pressure applied to the inlet port of the regulator.*

***Mechanical efficiency** - The ratio of the indicated horsepower to the actual shaft horsepower.*

***Memory** - The tendency of a material to return to its original shape after deformation.*

***Miscible** - Capable of being mixed in any concentration without separation of phases; e.g., water and ethyl alcohol are miscible.*

Modulus of elasticity *- One of the several measurements of stiffness or resistance to deformation. Often incorrectly used to indicate specifically static tension modulus.*

Modulus *- Tensile stress at a specified elongation. (Usually 100% elongation for elastomers.)*

Moving parts logic *- The technology of achieving logic control by means of fluid devices having moving parts.*

Muffler *- A device for reducing gas flow noise. Noise is decreased by backpressure control of gas expansion.*

Newt *- A unit of kinematic viscosity in the English system. It is expressed in square inches per second (see Stokes).*

Newtonian fluid *- A fluid, such as a straight mineral oil, whose viscosity does not change with rate of flow.*

Non-Newtonian fluid *- A fluid, such as a grease or a polymer containing oil (e.g. multi-grade oil), in which shear stress is not proportional to shear rate.*

NOR device *- A control device that has its output in the logical 1 state if and only if all the control signals assume the logical 0 state.*

NOT device *- A control device that has its output in the logical 1 state if and only if the control signal assumes the logical 0 state. (The NOT device is a single-input NOR device.)*

Oil swell *- The change in volume of a rubber article due to absorption of oil or other fluid.*

OR device *- A control device that has its output in the logical 0 state if and only if all the control signals assume the logical 0 state.*

Out-gassing *- A vacuum phenomenon wherein a substance spontaneously releases volatile constituents in the form of vapors or gases. In rubber compounds, these constituents may include water vapor, plasticizers, air, inhibitors, etc.*

Output stage *- The final stage of hydraulic amplifications used in a servovalve.*

Oxidation inhibitor *- A substance added in small quantities to a petroleum product to increase its oxidation resistance, thereby lengthening its service or storage life. Also called an anti-oxidant.*

Ozone resistance *- A material's ability to withstand the deteriorating effect of ozone (which generally causes cracking).*

Packing *- A sealing device consisting of bulk deformable material of one or more mating deformable elements, reshaped by manually adjustable compression to obtain and maintain effectiveness. Packing usually uses axial compression to obtain radial sealing.*

Pascal's Law *- A pressure applied to a confined fluid at rest is transmitted with equal intensity throughout the fluid.*

Permanent set *- The deformation remaining after a specimen has been stressed in tension for a definite period and released for a definite period.*

Permeability *- The rate at which a liquid or gas under pressure passes through a solid material by diffusion and solution. In rubber terminology, it is the rate of gas flow expressed in atmospheric cubic centimeters per second through an elastomeric material one centimeter square and one centimeter thick (atm cm3/cm2 ∞cm∞sec)*

Petroleum fluid *- A fluid composed of petroleum oil that may contain additives and/or inhibitors.*

Pipe *- A conductor whose outside diameter is standardized for threading. Pipe is available in standard, extra-strong, or double extra-strong wall thickness.*

Piston rod *- The element that transmits mechanical force and motion from the piston.*

Plasticizer *- A substance -- usually a heavy liquid -- added to an elastomer to decrease stiffness, improve low-temperature properties, and improve processing.*

Pneumatics *- The engineering science pertaining to gaseous pressure and flow.*

Poise *- The standard unit of dynamic viscosity in the centimeter-gram-second (CGS) system. It is the ratio of the shearing stress to the shear rate of fluid and is expressed in millipascal sec. (equals 1 centipoise).*

Polar compound *- A chemical compound whose molecules exhibit electrically positive characteristics at one extremity and negative characteristics at the other. Polar compounds are used as additives in many petroleum products.*

Polymer *- A material formed by the joining together of many (poly) individual units (mer) of one or more monomers. Synonymous with elastomers.*

Polytropic compression *- A process that occurs when heat is transferred to or from air at a precise rate during compression so that PVn is constant.*

Port *- A terminus of a passage in a component to which conductors can be connected.*

Port, differential pressure *- A port that provides a passage to the upstream and downstream sides of a component.*

***Post cure** - The second step in the vulcanization process for the more exotic elastomers. It provides stabilization of parts and drives off decomposition products resulting from the vulcanization process.*

***Pour point** - The lowest temperature at which an oil or distillate fuel is observed to flow when cooled under conditions prescribed by test method ATM D 97. The pour point is 3°C (5°F) above the temperature at which the oil in a test vessel shows no movement.*

***Power unit** - A combination of pump, pump drive, reservoir, controls and conditioning components to supply hydraulic power to a system.*

***Pressure, absolute** - The pressure above zero absolute, i.e., the sum of atmospheric and gauge pressure. In vacuum related work it is usually expressed in millimeters of mercury (mm-Hg).*

***Pressure, atmospheric** - Pressure exerted by the atmosphere at any specific location. (Sea level pressure is approximately 14.7 pounds per square inch absolute. 1 bar = 14.5 PSI).*

***Pressure, back** - The pressure encountered on the return side of a system.*

***Pressure, break loose (breakout)** - The minimum pressure that initiates movement.*

***Pressure, burst** - The pressure that causes failure of and consequential loss of fluid through the product envelope.*

***Pressure, charge** - The pressure at which replenishing fluid is formed into a fluid power system.*

***Pressure, control range** - The permissible limits between which system pressure may be set.*

***Pressure, cracking** - The pressure at which a pressure-operated valve begins to pass fluid.*

***Pressure, differential (pressure drop**) - The difference in pressure between any two points of a system or a component.*

***Pressure, gauge** - The pressure differential above or below ambient atmospheric pressure.*

***Pressure, induced** - The pressure generated by an externally applied force.*

***Pressure, inlet** - The pressure at the apparatus inlet port.*

***Pressure, intensified** - The outlet pressure in a fluid power cylinder required to slow the piston rod extending under regulated pressure introduced at the cap end.*

***Pressure, maximum inlet** - The maximum rated gauge pressure applied to the inlet.*

***Pressure, nominal** - A pressure value assigned to a component or system for the purpose of convenient designation.*

Pressure, outlet *- Pressure at the apparatus outlet port.*

Pressure, override *- The difference between the cracking pressure of a valve and the pressure reached when the valve is passing its rated flow.*

Pressure, peak *- The maximum pressure encountered in the operation of a component.*

Pressure, pilot *- The pressure in the pilot circuit.*

Pressure, precharge *- The pressure of compressed gas in an accumulator prior to the admission of a liquid.*

Pressure, proof *- The non-destructive test pressure -- in excess of the maximum rated operating pressure -- that causes no permanent deformation, excessive external leakage, or other resulting malfunction.*

Pressure, rated *- The qualified operating pressure that is recommended for a component or system by the manufacturer.*

Pressure, shock *- The pressure existing in a wave moving at sonic velocity.*

Pressure, static *- The pressure in a fluid at rest.*

Pressure, surge *- The pressure resulting from surge conditions.*

Pressure, system *- The pressure that overcomes the total resistance in a system. It includes all losses as well as useful work.*

Pressure *- Force per unit area, usually expressed in pounds per square inch or bar.*

Pump, fixed-displacement *- A hydraulic pump in which the volume displaced per cycle cannot be varied.*

Pump, gear, external *- A pump with two or more external gears.*

Pump, gear, internal *- A pump with an internal gear in engagement with one or more external gears.*

Pump, gear *- A pump in which two or more gears act in engagement as pumping members.*

Pump, hydraulic *- A device that converts mechanical force and motion into hydraulic fluid power.*

Pump, multiple-stage *- Two or more hydraulic pumps connected in series.*

***Pump, piston, axial** - A pump having several pistons with mutually parallel axes that are arranged around and parallel to a common axis.*

***Pump, piston, inline** - A pump having several pistons with mutually parallel axes arranged on a common plane.*

***Pump, piston, radial** - A pump having several pistons arranged to operate radially.*

***Pump, piston** - A pump in which the fluid volume is displaced by one or more reciprocating pistons.*

***Pump, screw** - A hydraulic pump having one or more screws rotating in a housing.*

***Pump, vane, balanced** - A pump in which the transverse forces on the rotor are balanced.*

***Pump, vane, unbalanced** - A pump in which the transverse forces on the rotor are not balanced.*

***Pump, vane** - A hydraulic pump having multiple radial vanes within a supporting rotor.*

***Pump, variable-displacement** - A hydraulic pump in which the volume displaced per cycle can be varied.*

***Pump-motor** - A unit that functions either as a pump or as a rotary motor.*

***Pump** - A device that converts mechanical torque and motion into hydraulic fluid power.*

***Pumping or surge** - The reversal of flow within a dynamic compressor. It takes place when insufficient pressure is generated to maintain flow.*

***Quick-disconnect coupling** - A component that can quickly join or separate a fluid line without the use of tools or special devices.*

***Refrigerated dryer** - A device that separates moisture by lowering the air temperature by means of refrigeration compressor and heat exchanger.*

***Regenerative dryer** - A dryer that restores its own capacity to separate moisture -- without replacing the drying compound.*

***Regenerative circuit** - see Circuit, regenerative.*

***Regulator, air line pressure** - A regulator that transforms a fluctuating air pressure supply to provide a constant lower pressure output.*

***Reinforcing agent** - A material dispersed in an elastomer to improve compression, shear, or other stress properties.*

Reservoir (tank) *- A container for storage of liquid in a fluid power system.*

Reservoir, hydraulic *- A reservoir for storing and conditioning a liquid in a hydraulic system.*

Reservoir, pressure-sealed *- A sealed reservoir for storage of fluids under pressure.*

Resilient *- Capable of returning to original size and shape after deformation.*

Reyn *- The standard unit of absolute viscosity in the English system. It is expressed in pound-seconds per square inch.*

Reynolds Number *- A numerical ratio of the dynamic forces of mass flow to the shear stress due to viscosity. Flow usually changes from laminar to turbulent between Reynolds Numbers 2000 and 4000.*

Ring, O- *- A ring that has a round cross-section.*

Ring, piston *- A piston sealing ring. It is usually one of a series and often is split to facilitate expansion or contraction.*

Ring, scraper *- A ring that removes material by a scraping action.*

Rotation *- The direction of rotation is always quoted as viewed looking at the shaft end. In dubious cases, provide a sketch.*

Seal, cup *- A sealing device with a radial base integral with an axial cylindrical projection at its outer diameter.*

Seal, dynamic *- A sealing device used between parts that have relative motion.*

Seal, elastomer *- A material having rubber-like properties; i.e., having the capacity for large deformation, with rapid and substantially complete recovery on release from the deforming force.*

Seal, rod (shaft) *- A sealing device that seals the periphery of a piston rod.*

Seal, static (gasket) *- A sealing device used between parts that have no relative motion.*

Sensor *- A device that detects and transmits changes in external conditions.*

Separator *- A device whose primary function is to isolate contaminants by physical properties other than size. (Separators remove gas from liquid media or remove liquid from gaseous media).*

Servovalve *- A valve that modulates output as a function of an input command.*

Servovalve, electrohydraulic - *A servo-valve that is capable of continuously controlling hydraulic output as a function of an electrical input.*

Servovalve, electrohydraulic, flow-control - *An electrohydraulic servovalve whose primary function is control of output flow.*

Servovalve hysteresis - *The difference in the servovalve input currents required to produce the same output during a single cycle of valve input current when cycled at a rate below that at which dynamic effects are important.*

Servovalve null leakage - *The total internal leakage from the valve in the null position.*

Servovalve, pressure-control - *A hydraulic servo valve whose primary function is the control of output pressure.*

Shear rate - *The rate at which adjacent layers of fluid move with respect to each other, usually expressed as reciprocal seconds.*

Shear stress - *The frictional force overcome in sliding one layer of fluid along another, as in any fluid flow. The shear stress of petroleum oil -- or other Newtonian fluid -- at a given temperature varies directly with shear rate (velocity). The ratio between shear stress and shear rate is constant; this ratio is termed viscosity.*

Shrinkage - *The decreased volume of a seal, usually caused by extraction of soluble constituents from fluids, followed by air drying.*

Silencer - *A device for reducing gas flow noise. Noise is decreased by tuned resonant control of gas expansion.*

Snubber - *See gauge damper.*

Solenoid, digital - *An electrically energized device that generates on-off signals.*

Solenoid, proportional - *An electrical device that reacts proportionally to the strength of electrical signals.*

Sorption - *The term used to denote the combination of absorption and adsorption processes in the same substance.*

Specific gravity, liquid - *The ratio of the weight of a given volume of liquid to the weight of an equal volume of water.*

Squeeze - *The cross-sectional diametrical compression of an O-ring between the surface of the groove bottom and the surface of the other mating metal part in the gland assembly.*

Stage - *A hydraulic amplifier used in a servovalve. Servovalves may be single-stage, two-stage, three-stage, etc.*

Standard - *A document, or an object for physical comparison, for defining product characteristics, products, or processes; prepared by the consensus of a properly constituted group of those substantially affected and having the qualifications to prepare the standard for voluntary use.*

Stokes - *The standard unit of kinematic viscosity in the CGS system. It is expressed in square centimeters per second; 1 centistokes equals 0.01 stokes.*

Strainer - *See filter, strainer.*

Surface tension - *The surface force of a liquid in contact with a fluid by which it tends to assume a spherical form and to present the least possible surface. It is expressed in pounds per foot or dynes per centimeter.*

Surfactant - *A surface-active agent that reduces interfacial tension of a liquid. A surfactant used in petroleum oil may increase the oil's affinity for metals and other material.*

Surge - *A transient rise of pressure or flow.*

Swell - *The increased volume of specimen caused by immersion in a fluid (usually a liquid).*

Switch, float - *An electric switch that is responsive to liquid level.*

Switch, flow - *An electric switch operated by fluid flow.*

Switch, pressure-differential - *An electric switch operated by a difference in pressure.*

Switch, pressure - *An electric switch operated by fluid pressure.*

Standard pressure and temperature - *Generally defined as 68° F and 14.70 psia with a relative humidity of 36%.*

Synthetic fluid, silicate ester - *A fluid compound of organic silicates. It may contain additives.*

Synthetic fluid - *Fluid other than mineral that has been artificially compounded for use in a fluid power system.*

Temperature, ambient - *The temperature of the environment in which an apparatus is working.*

Tensile strength - *The force in pounds per square inch required to cause the rupture of a specimen of a rubber material.*

Terpolymer - *A polymer consisting of three different monomers chemically combined.*

***Tie rod** - An axial external element that traverses the length of the cylinder. It is pre-stressed at assembly to hold the ends of the cylinder against the tubing. (Tie rod extensions can serve as mounting devices.)*

***Torque motor** - A type of electro-mechanical transducer having rotary motion used in the input stages of servovalves.*

***Torque** - Rotary force transmitted by the driving shaft of a motor or pump.*

***Torr** - A unit of pressure equal to 1/760 of an atmosphere.*

***Torricelli's Theorem** - The liquid velocity at an outlet discharging into the free atmosphere is proportional to the square root of the head.*

***Transducer, flow** - A device that converts fluid flow to an electrical signal.*

***Transducer, pressure** - A device that converts fluid pressure to an electrical signal.*

***Trunnion** - A cylinder-mounting device consisting of a pair of opposite projecting cylindrical pivots. The cylindrical pivot pins are at right angle or normal to the piston rod centerline to permit the cylinder to swing in a plane.*

***Tube** - A conductor whose size is its outside diameter. Tube is available in varied wall thickness and material.*

***Vacuum** - Pressure less than ambient atmospheric pressure.*

***Vacuum pump** - A device that uses mechanical force and motion to evacuate gas from a connected chamber to create sub-atmospheric pressure.*

***Valve** - A device that controls fluid flow direction, pressure, or flow rate.*

***Valve actuator** - The valve part(s) through which force is applied to move or position flow-directing elements.*

***Valve, air** - A valve for controlling air flow direction, pressure, or flow rate.*

***Valve, cartridge** - A valve with working parts contained in a cylindrical body. The cylindrical body must be inserted into a housing for use. Ports through the body cooperate with ports in the containing housing.*

***Valve, directional-control** - A valve whose primary function is to direct or prevent flow through selected passages.*

***Valve, directional-control, 3-way** - A directional control valve whose primary function is to pressurize and exhaust a port.*

Valve, directional-control, 4-way *- A directional control valve whose primary function is to pressurize and exhaust two ports.*

Valve, directional-control, check *- A directional control valve that permits flow of fluid in only one direction.*

Valve, directly operated *- A valve in which the controlling forces acting on the element directly influence the movement of the control elements.*

Valve, electrohydraulic, proportional *- A valve that responds proportionally to input signals.*

Valve, flow-control (flow-metering) *- A valve whose primary function is to control flow rate.*

Valve, flow-control, bypass *- A pressure-compensated flow-control valve that regulates the working flow, diverting surplus fluid to the reservoir or to a second service.*

Valve, flow-control, deceleration *- A flow control valve that gradually reduces flow rate to provide deceleration.*

Valve, flow-control, pressure-compensated *- A flow control valve that controls the rate of flow independent of pressure fluctuations.*

Valve, flow-dividing, pressure-compensated *- A flow-dividing valve that divides flow at a constant ratio regardless of difference in resistance of the branches.*

Valve, flow-dividing *- A valve that divides the flow from a single source into two or more branches.*

Valve, hydraulic *- A valve for controlling liquid flow direction, pressure, or flow rate.*

Valve, needle *- A flow-control valve in which the adjustable control element is a tapered needle. Its usual purpose is to accurately control the rate of flow.*

Valve, pilot-operated (indirect) *- A valve in which a relatively small flow through an integral vent line relief (pilot) controls the movement of the main element.*

Valve, pilot *- A valve applied to operate another valve or control.*

Vapor pressure *- The pressure of a confined vapor in equilibrium with its liquid at a specified temperature; thus, a measure of a liquid's volatility.*

Viscosity *- The measurement of a fluid's resistance to flow. The common metric unit of absolute viscosity is the poise, which is defined as the force in dynes required to move a surface one square centimeter in area past a parallel surface at a speed of one centimeter per second, with the surfaces separated by a fluid film one centimeter thick. In addition to kinematic viscosity, there are other methods for determining viscosity, including, Saybolt Universal viscosity, Saybolt*

Furol viscosity, Engier viscosity, and Redwood viscosity. Because viscosity varies inversely with temperature, its value is meaningless until the temperature at which it is determined is reported.

Viscosity index (V.I.) *- An empirical, unit-less number indicating the effect of temperature changes on the kinematic viscosity of an oil. Liquids change viscosity with temperature, becoming less viscous when heated. The higher the V.I. of an oil, the lower its tendency to change viscosity with temperature.*

Volume change *- A change in the volume of a seal as a result of immersion in a fluid -- expressed as a percentage of the original volume.*

Volumetric efficiency *- The ratio of the actual volume of air admitted (at a specified temperature and pressure), to the full piston-displacement volume- obviously for reciprocating compressors only. Volume (annular) absorbed by one full retraction of the cylinder.*

Vulcanization *- A thermo-setting reaction involving the use of heat and pressure, resulting in greatly increased strength and elasticity of rubber-like materials.*

Two types of fluid power circuits

Most fluid power circuits use compressed air or hydraulic fluid as their operating media. While these systems are the same in many aspects, they can have very different characteristics in certain ways.

For example: remote outdoor applications may use dry nitrogen gas in place of compressed air to eliminate freezing problems. Readily available nitrogen gas is not hazardous to the atmosphere or humans. Because nitrogen is usually supplied in gas cylinders at high pressure, it has a very low dew point at normal system pressure. The gas may be different but the system's operating characteristics are the same.

Hydraulic systems may use a variety of fluids -- ranging from water (with or without additives) to high-temperature fire-resistant types. Again the fluid is different but the operating characteristics change little.

Pneumatic systems

Most pneumatic circuits run at low power -- usually around 2 to 3 horsepower. Two main advantages of air-operated circuits are their low initial cost and design simplicity. Because air systems operate at relatively low pressure, the components can be made of relatively inexpensive material -- often by mass production processes such as plastic injection molding, or zinc or aluminum die-casting. Either process cuts secondary machining operations and cost.

First cost of an air circuit may be less than a hydraulic circuit but operating cost can be five to ten times higher. Compressing atmospheric air to a nominal working pressure requires a lot of horsepower. Air motors are one of the most costly components to operate. It takes approximately one horsepower to compress 4 cfm of atmospheric air to 100 psi. A 1-hp air motor can take up to 60 cfm to operate, so the 1-hp air motor requires (60/4) or 15 compressor horsepower when it runs. Fortunately, an air motor does not have to run continuously but can be cycled as often as needed.

Air-driven machines are usually quieter than their hydraulic counterparts. This is mainly because the power source (the air compressor) is installed remotely from the machine in an enclosure that helps contain its noise.

Because air is compressible, an air-driven actuator cannot hold a load rigidly in place like a hydraulic actuator does. An air-driven device can use a combination of air for power and oil as the driving medium to overcome this problem, but the combination adds cost to the circuit. *(Chapter 17 has information on air-oil circuits.)*

Air-operated systems are always cleaner than hydraulic systems because atmospheric air is the force transmitter. Leaks in an air circuit do not cause housekeeping problems, but they are very expensive. It takes approximately 5 compressor horsepower to supply air to a standard hand-held blow-off nozzle and maintain 100 psi. Several data books have charts showing cfm loss *through different size orifices at varying pressures. Such charts give an idea of the energy losses due to leaks or bypassing.*

Hydraulic systems

A hydraulic system circulates the same fluid repeatedly from a fixed reservoir that is part of the prime mover. The fluid is an almost non-compressible liquid, so the actuators it drives can be controlled to very accurate positions, speeds, or forces. Most hydraulic systems use mineral oil for the operating media but other fluids such as water, ethylene glycol, or synthetic types are not uncommon. Hydraulic systems usually have a dedicated power unit for each machine. Rubber-molding plants depart from this scheme. They usually have a central power unit with pipes running to and from the presses out in the plant. Because these presses require no flow during their long closing times, a single large pump can operate several of them. These hydraulic systems operate more like a compressed-air installation because the power source is in one location.

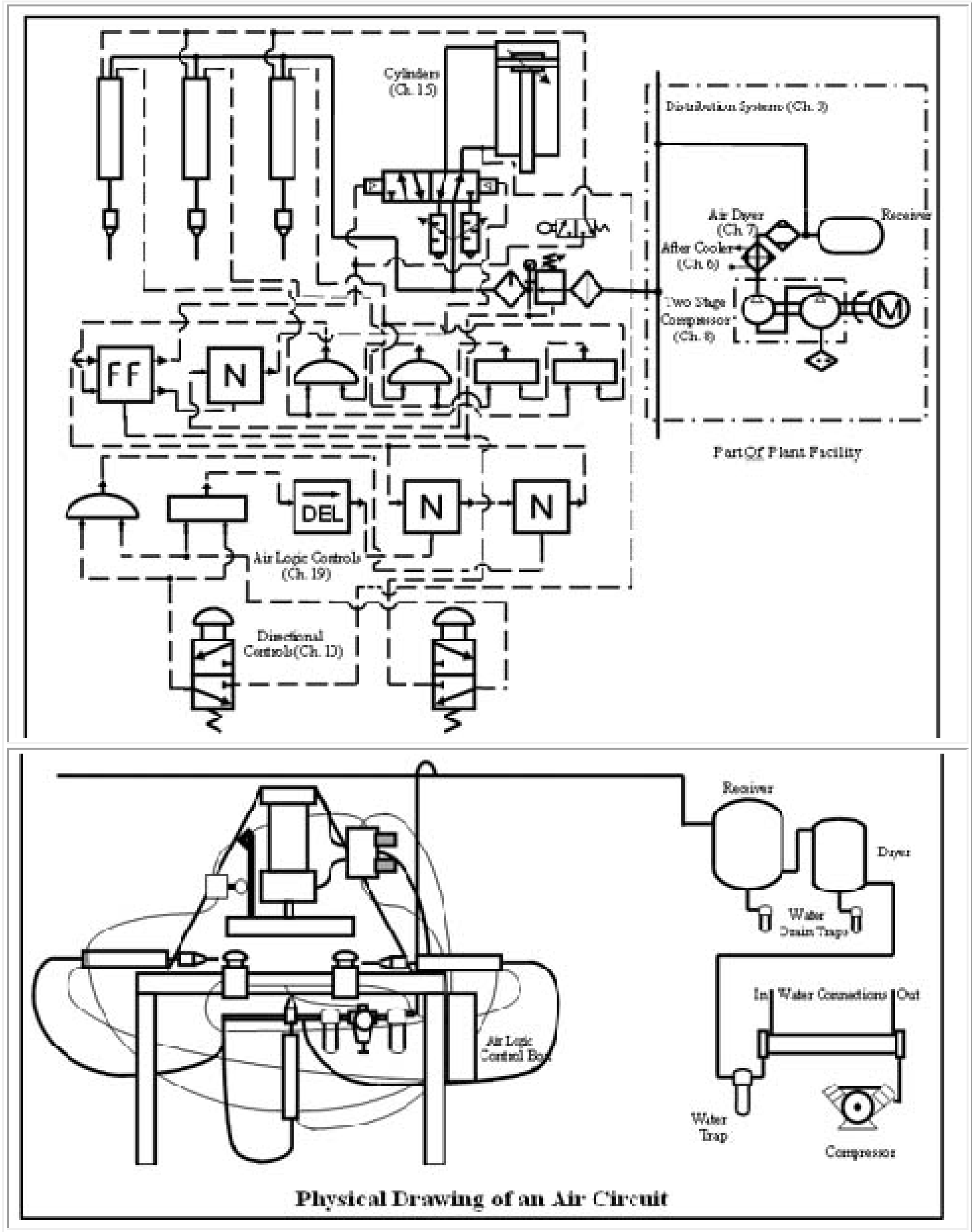
A few other manufacturers are setting up central power units when the plant has numerous machines that use hydraulics. Some advantages of this arrangement are: greatly reduced noise levels at the machine, the availability of backup pumps to take over if a working pump fails, less total horsepower and flow, and increased uptime of all machines.

Another advantage hydraulic-powered machines have over pneumatic ones is that they operate at higher pressure -- typically 1500 to 2500 psi. Higher pressures generate high force from smaller actuators, which means less clutter at the work area.

The main disadvantage of hydraulics is increased first cost because a power unit is part of the machine. If the machine life is longer than two years, the higher initial cost is often offset by lower operating cost due to the much higher efficiency of hydraulics. Another problem area often cited for hydraulics is housekeeping. Leaks caused by poor plumbing practices and lack of pipe supports can be profuse. This can be exaggerated by overheated low-viscosity fluid that results from poor circuit design. With proper plumbing procedures, correct materials, and preventive maintenance, hydraulic leaks can be virtually eliminated.

Another disadvantage could be that hydraulic systems are usually more complex and require maintenance personnel with higher skills. Many companies do not have fluid power engineers or maintenance personnel to handle hydraulic problems.

5-1. Schematic drawing of a hydraulic circuit, and physical drawing of the components in the circuit.



Typical pneumatic circuit

Figure 5-1 *includes a pictorial representation and a schematic drawing of a typical pneumatic circuit. It also has a pictorial and schematic representation of a typical compressor installation to drive the circuit (and other pneumatic machines). Seldom, if ever, is the compressor part of a pneumatic schematic. Power for a typical pneumatic circuit comes from a central compressor facility with plumbing to carry pressurized air through the plant. Pneumatic drops are similar to electrical outlets and are available at many locations.*

Why schematic drawings?

Schematic drawings make it possible to show circuit functions when using components from different manufacturers. A 4-way valve or other component from one supplier may bear little physical resemblance to one from other suppliers. Using actual cutaway views of valves to show how a machine operates would be fine for one circuit using a single supplier's valves. However, another machine with different parts would have a completely different-looking drawing. A person trying to work on these different machines would have to know each brand's ins and outs . . . and how they affect operations. This means designing and troubleshooting every circuit would require special and different knowledge. Using schematic symbols requires learning only one set of information for any component.

Schematic symbols also give more information than a picture of the part. It may almost impossible to tell if a 4-way valve is 3-position by looking at a pictorial representation. On the other hand, its symbol makes all features immediately clear. Another advantage is that by using ISO symbols the drawing can be read by persons from different countries. Any notes or the material list may be unreadable because of language differences, but anyone trained in symbology can follow and understand circuit function.

Parts of a typical pneumatic system

*The schematic in **Figure 5-1** starts at the filter, regulator, and lubricator (FRL) combination that is connected to the plant-air supply. FRL units are important because they assure a clean, lubricated supply of air at a constant pressure. It's important to keep these units supplied, drained, and set correctly to keep the circuit operating smoothly and efficiently.*

The filter is first in line to remove contamination and condensed water. It should be drained regularly or fitted with an automatic drain. The regulator should be set at the lowest pressure that will produce good parts at the cycle rate specified. The lubricator should be adjusted to allow oil to enter the air stream at a reasonable rate. In poorly maintained plants, the filter may be completely full of contaminants, the regulator is screwed all the way in, and the lubricator is completely empty.

Air-logic controls

*Air-operated miniature valves called air-logic controls control the circuit in **Figure 5-1**. Air-logic controls run on shop air and are actuated by air palm buttons and limit valves to start and continue a cycle.*

This circuit has an OSHA safe anti tie-down dual palm button start control. The two palm buttons must be operated at almost the same time or the cylinder will not extend. Tying down one palm button renders the circuit inoperative until it is released. The rest of the logic circuit causes the drills to extend and keeps the clamp cylinder down until they have all retracted and stopped. This circuit also has an anti-repeat feature, which means the cycle only operates once, even if the operator continues to hold the palm buttons down. Safety features such as these are easy to implement.

Directional-control valves

A 5-way, double-pilot-operated directional control valve operates the cylinder. This valve extends and retracts the cylinder according to signals from the air logic controls in the cabinet. Movement also requires inputs from the palm buttons to make sure the operator is safely clear of the cylinder before it operates. This directional control valve has speed-control mufflers in its exhaust port to control cylinder speed in both directions. These devices also reduce noise from exhausting air.

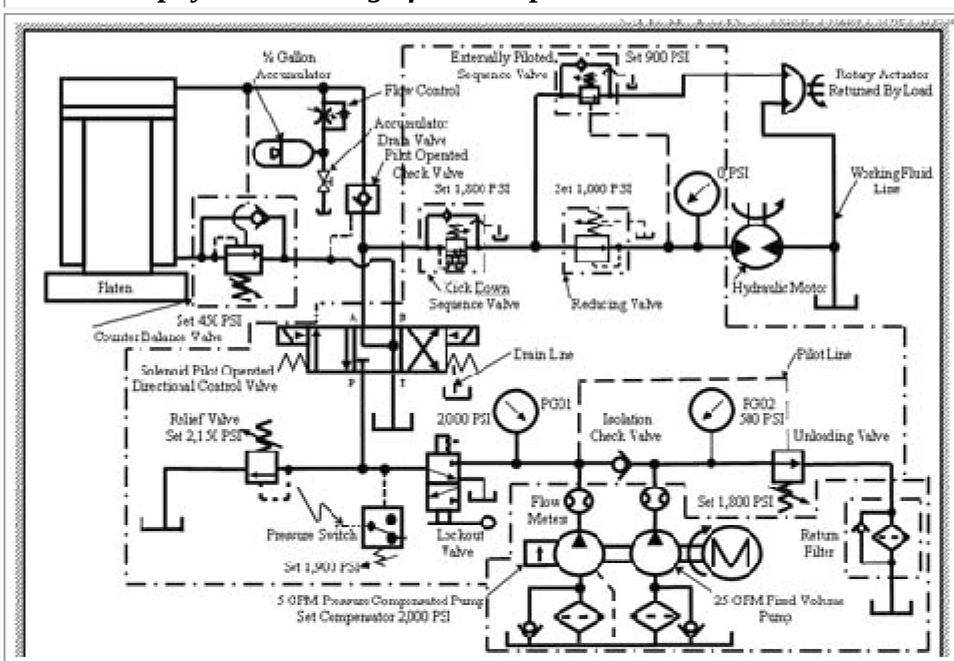
A limit valve at the extend stroke of the cylinder makes sure it has reached the part before the drills start. A limit valve monitors position but it cannot tell if the cylinder has reached full clamping force. In most applications when the cylinder is close enough to make the limit valve, it will be at or near clamping force before the next operation gets to the work. In some applications it might be necessary to add a pressure sequence valve to make sure the cylinder reaches a certain pressure before the cycle continues.

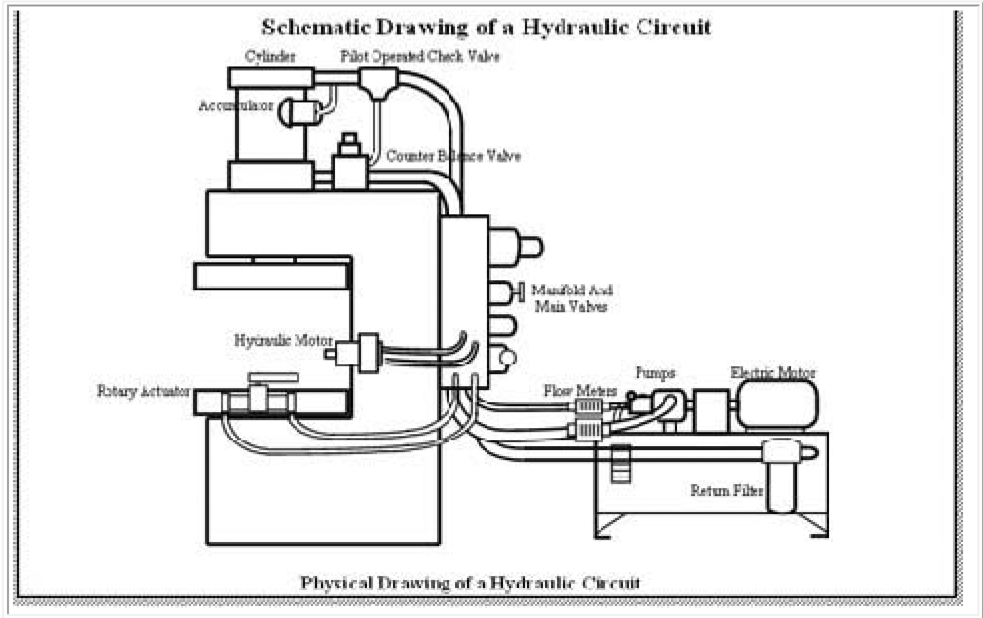
Air drills

Rotary output devices such as air motors with built-in cycling valves and rotary actuators that make only a fraction of a turn are available to perform many functions. Because compressed air is the driving force, these devices are explosion-proof and can operate in dirty or wet atmospheres without the problems posed by electrical equipment. Carefully applied air-operated devices can be an improvement in many situations.

These and other air-operated components are explained and applied in the following chapters.

5-2. Schematic drawing of an air circuit with air-logic controls, and physical drawing of the components in the circuit.





Typical hydraulic circuit

Figure 5-2 *provides a pictorial representation and a schematic drawing of a typical hydraulic circuit. Notice that the hydraulic power unit is dedicated to this machine. Unlike pneumatic circuits, most hydraulic systems have a power unit that only operates one machine. (As mentioned before, some new installations are using a central hydraulic power source with piping throughout the plant to carry pressurized and return fluid.)*

Why a schematic drawing?

Schematic drawings make it possible to show circuit functions when using components from different manufacturers. A 4-way valve or other part from a different supplier may bear little resemblance to one from other suppliers. Using actual cutaways of a valve to show how a machine operates would be fine for one circuit using one supplier's valves. Nevertheless, another machine with different parts would have a completely different looking drawing. A person trying to work on these different machines would have to know each brand and how they affect operations. This means designing and trouble shooting every circuit would require special different knowledge. Using schematic symbols requires learning only one set of information for any component.

Schematic symbols also give more information than a picture of the part. It may be hard to impossible to tell if a 4-way valve is 3-position by looking at a pictorial representation while its symbol makes all features immediately clear. Another feature is by using ISO symbols the drawing can be read by persons from different languages. Any notes or the material list may be in a language foreign to you but following and understanding circuit function should not be a problem.

Parts of a typical hydraulic schematic

A good starting point for any hydraulic schematic is at the power unit. The power unit consists of the reservoir, pump or pumps, electric motor, coupling and coupling guard, and entry and exit piping, with flow meters and return filter. It also might include relief valves, unloading valves, pressure filters, off-line filtration circuits, and control valves. The power unit must be able to cycle all functions in the allotted time at a pressure high enough to do the work intended. A well-designed circuit will run efficiently with little to no wasted energy that generates heat. It will run many years with minimum maintenance if its filters are well maintained and it is not overheated.

When items such as pressure gauges and flow meters are installed, it is easy to troubleshoot any system malfunction quickly and accurately. Flow meters always show pump flow (or lack thereof) and eliminate premature pump replacement. They can indicate impending pump failure well in advance of system failure. Also quick-disconnect plug-in type ports at strategic locations make it easy to check pressure at any point.

Directional control valves

*The circuit in **Figure 5-2** has only one directional control valve to extend and retract the main cylinder. Pressure-control valves make the hydraulic motor and rotary actuator operate in sequence after the cylinder extends and builds a preset pressure. (This is not the best way to control actuators, but it is shown here to demonstrate the use of different valves.)*

An isolation check valve between the pumps keeps the high-pressure pump from going to tank when the low-volume pump unloads. A pilot-operated check valve in the line to the cap end of the main cylinder traps fluid in the cylinder while the motor and rotary actuator operate.

Pressure-control valves

A pressure-relief valve at the pumps automatically protects the system from overpressure. An unloading valve dumps the high-volume pump to tank after reaching a preset pressure. A kick-down sequence pressure-control valve forces all oil to the cylinder until it reaches a preset pressure. After reaching this pressure, the valve opens and sends all pump flow to the hydraulic motor first. A sequence valve upstream from the rotary actuator keeps it from moving until the hydraulic motor stalls against its load. A pressure-reducing valve ahead of the hydraulic motor allows the operator to set maximum torque by adjusting pressure to the motor inlet. (All of these controls are covered in the text of this manual.)

Another pressure-control valve -- called a counterbalance valve -- located in the rod end line of the main cylinder keeps it from running away when the directional control valve shifts. The counterbalance valve is adjusted to a pressure that keeps the cylinder from extending, even when weight on its rod could cause this to happen.

Accumulators

Because hydraulic oil is almost non-compressible, a gas-charged accumulator allows for storage of a volume of fluid to perform work. The expandable gas in the accumulator pushes the oil out when external pressure tries to drop. The accumulator in this circuit makes up for leakage in the *cylinder cap-end circuit while pump flow runs the hydraulic motor and rotary actuator. Use care when specifying and using accumulators because they can be a safety issue.*

These and other hydraulic components are explained and applied in the following chapters.

Parallel and series circuits

There are parallel and series type circuits in fluid power systems. Pneumatic and hydraulic circuits may be parallel type, while only hydraulic circuits are series type. However, in industrial applications, more than 95% of hydraulic circuits are the parallel type. All pneumatic circuits are parallel design because air is compressible it is not practical to use it in series circuits.

In parallel circuits, fluid can be directed to all actuators simultaneously. Hydraulic parallel circuits usually consist of one pump feeding multiple directional valves that operate actuators one at a time or several in unison.

Figure 5-3 *shows a typical pneumatic parallel system schematic. All actuators in this circuit can operate at the same time and are capable of full force and speed if they have ample supply. The filter, regulator, and lubricator combination must be sized to handle maximum flow of all actuators in motion at the same time, When the air supply is insufficient, the cylinder with the least resistance will move first.*

5-3. Schematic drawing of three cylinders in a typical pneumatic parallel circuit.

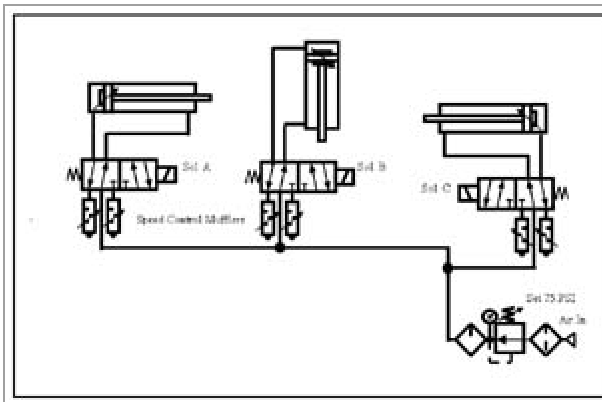
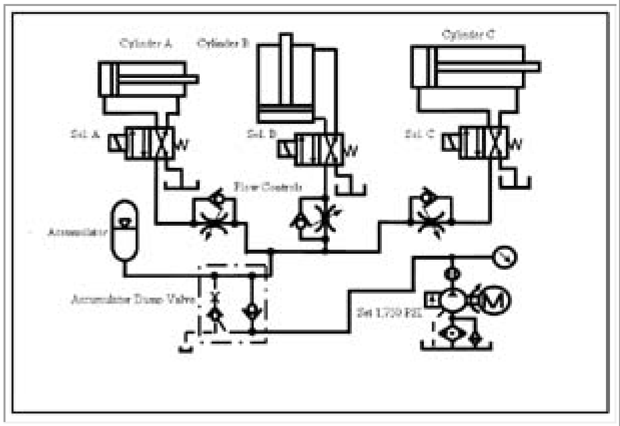


Figure 5-4 *shows a typical hydraulic parallel system schematic. Any actuator in this circuit can move at any time and is capable of full force and speed when the pump produces sufficient flow. Parallel circuits that have actuators that move at the same time must include flow controls to keep all flow from going to the path of least resistance.*

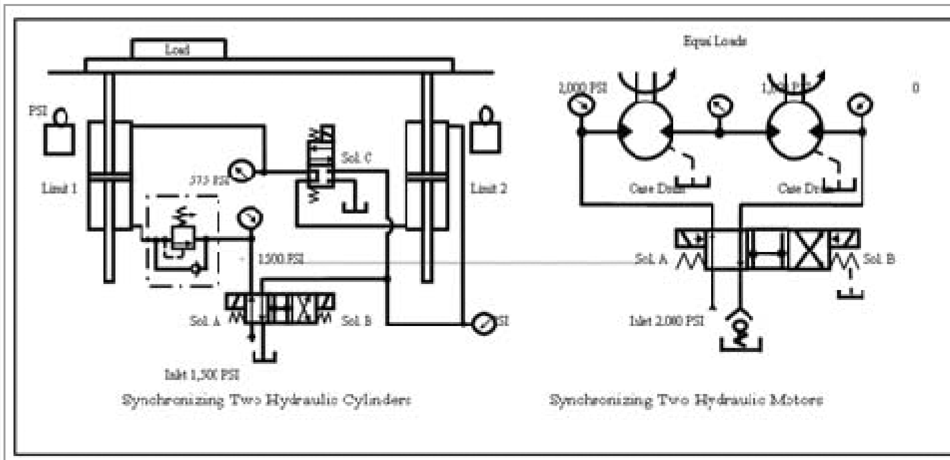
5-4. Schematic drawing of three cylinders in a typical hydraulic parallel circuit.



*Flow controls are usually required to keep single cylinder movement from over speeding. The circuit in **Figure 5-4** shows a meter-in flow control at each directional control valve's inlet to control speed in both directions. Placing flow controls at the cylinder ports would allow separate speeds for extension and retraction.*

***Figure 5-5** illustrates cylinders or hydraulic motors in typical series circuits. These synchronizing circuits are the most common use for actuators in series. The schematic drawing at left shows how to control two or more cylinders so they move simultaneously at the same rate. Oil is fed to the cylinder on the left and it starts to extend. Oil trapped in its opposite end transfers to the right cylinder, causing it to extend at the same time and rate. Oil from the right cylinder goes to tank. The platen moves and stays level regardless of load placement. Notice that this circuit uses double-rod end cylinders so the volumes in both ends are the same. (Other variations of this circuit are shown in the chapter on cylinders, which also explains synchronizing circuits in detail.)*

5-5. Schematic drawings of two synchronizing hydraulic circuits.



*The hydraulic motor circuit on the right in **Figure 5-5** shows a simple way to run two or more motors at the same speed. Fluid to the first motor flows into the inlet of the second motor to turn it at the same time and speed. Except for internal leakage in the motors, they will run at exactly the same rpm. As many as ten motors can operate in series -- based on their loads and speeds.*

Hydraulics vs. pneumatics

Pressurized fluids act in a certain manner in most situations. However, there are instances where a gas-type fluid does not perform as its liquid counterpart does. As mentioned earlier in this chapter, a pneumatic actuator is incapable of holding a position against increasing external forces because the air can be compressed more. Other situations such as flow-control circuits, return-line backpressure, energy-transfer considerations, and more are covered and explained in the text.

Conventions used in this manual

All schematic symbols and drawings are in accordance with the International Standards Organization (ISO) format. These symbols and representative parts are laid out in Chapter 4 either in whole or in part. Some symbols are made up of several standard parts and are not shown in their entirety in Chapter 4.

When a symbol is not shown it is good practice to use the symbol shown in the suppliers catalog. If no symbol is given there then use standard symbol parts to make a representation of the new item.

As in all cases of drawings using schematic symbols, the circuit designer may use his or her experience or opinion to interpret some parts. This usually does not make the schematic harder to read, just different. If a part representation is not clear, refer to the material list and check the supplier's catalog for an explanation of the valve's function.

Color coding

To better understand how a part or circuit works, consider using color coding for the lines and components. Color coding is instituted by the instructor, designer, or engineer and is according to his or her interpretation, so it might not be consistent in each case. Most training manuals and manufacturers use the following color code.

- ***Red**: Working fluid flow lines, usually from the pump to a device. This line is always solid. It can represent plastic tubing as small as 5/32-in. OD for air or any size pipe or tubing for hydraulics.*
- ***Blue**: Return lines from valves and other devices for hydraulic circuits. This line always is solid, and can represent any size pipe or tubing.*
- ***Yellow**: Metered or flow-controlled fluid that is at a reduced speed in relation to the same line without a restriction. This line could be solid or a series of long dashes if pilot flow must be metered.*

- ***Orange**: A reduced-pressure line, such as a pilot-pressure line or one carrying accumulator precharge gas. This line could be a solid after a reducing valve or a long-dashed line for pilot flow.*
- ***Green**: Pump inlet lines (suction lines) or drain lines. These lines would be solid for the pump inlet and a series of short dashes for drains. Two types of lines with the same color are not confusing -- even when in close proximity to each other.*
- ***Purple or indigo**: These colors usually indicate working fluid that has been pressure-intensified by area differences or load-induced conditions. These pressures are usually greater than the setting of the main relief valve or reducing valve that feeds the circuit.*
- *Lines without color are considered non-working or to have no flow at present.*

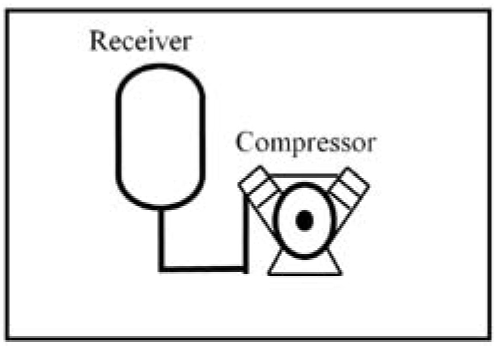
This color-coding technique is used with this manual and can be seen in Chapter 4.

Hydraulic reservoirs

Fluid power reservoirs

Fluid power systems require air or a liquid fluid to transmit energy. Pneumatic systems use the atmosphere -- the air we breathe -- as the source or reservoir for their fluid. A compressor takes in atmospheric air at 14.7 psia, compresses it to between 90 and 125 psig, and then stores it in a receiver tank. A receiver tank is similar to a hydraulic system's accumulator. A receiver tank, Figure 6-1, stores energy for future use similar to a hydraulic accumulator. This is possible because air is a gas and thus is compressible. A receiver tank is a pressure vessel and is constructed to pressure vessel standards. At the end of the work cycle the air is simply returned to the atmosphere.

6-1. Simple pneumatic power unit.



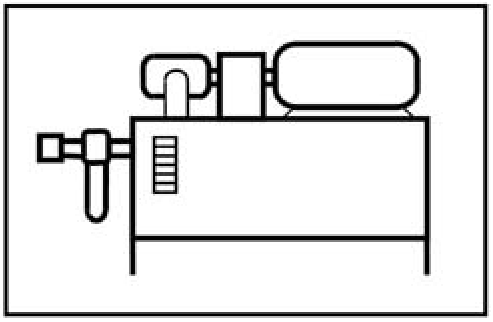
Hydraulic reservoirs

Hydraulic systems, on the other hand, need a finite amount of liquid fluid that must be stored and reused continually as the circuit works. Therefore, part of any hydraulic circuit is a storage reservoir or tank. This tank may be part of the machine framework or a separate stand-alone unit. In either case, reservoir design and implementation is very important. The efficiency of a well-designed hydraulic circuit can be greatly reduced by poor tank design. A hydraulic reservoir does much more than just provide a place to put fluid. A well-designed reservoir also dissipates heat, allows time for contamination to drop out of the fluid, and allows air bubbles to come to the surface and dissipate. It may give a positive pressure to the pump inlet and makes a convenient mounting place for the pump and its motor, and valves.

Some standard reservoir layouts

Pump on top. Figure 6-2 *shows this common reservoir/pump layout -- used by many suppliers. The flat top surface of a standard reservoir makes a perfect place to mount the pump and motor.*

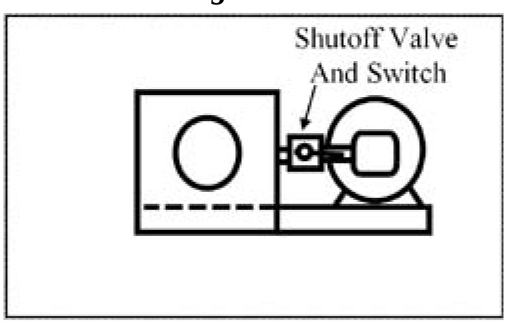
6-2. Pump and motor mounted on top of tank.



The main disadvantage to this configuration is that the pump must create enough vacuum to raise and accelerate the fluid into the pump inlet. For most pumps, this is not a big problem, but it is not the best situation for any of them. Axial or in-line piston pump life can be adversely affected by medium to high vacuum at its inlet when using this layout. The piping in this configuration must be sealed, should be as short as possible, and have few or no bends.

Pump alongside tank. Figure 6-3 *shows another design that is satisfactory for any type pump. (Many suppliers prefer this layout.) This arrangement is sometime called a flooded suction, because the pump inlet always is filled with fluid.*

6-3. Pump and motor mounted alongside tank.

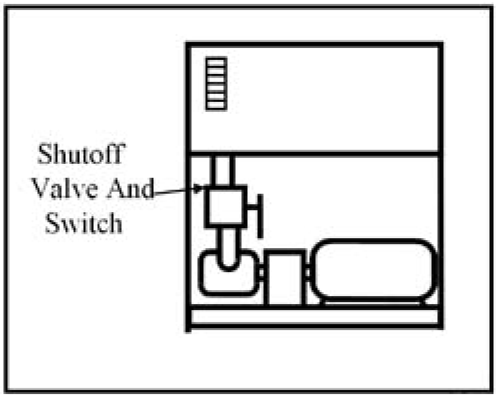


Although the pump inlet always has fluid, there will be some vacuum in the inlet line when the pump is running. A pump with its inlet below fluid level no longer has to raise the fluid, but it does have to accelerate and move it. However, this design is far better than the pump on top and can extend the service life of any type pump.

Notice the shutoff valve in the inlet line. This valve allows maintenance work to be done on the pump without draining the tank. Some precautions: install a free-flowing valve (such as a quarter-turn ball type) and use a valve with a limit switch to indicate full open. Wire this limit switch in parallel with the pump motor starter, so the pump cannot start until the shutoff valve is open.

6-4. Tank located above pump and

motor.

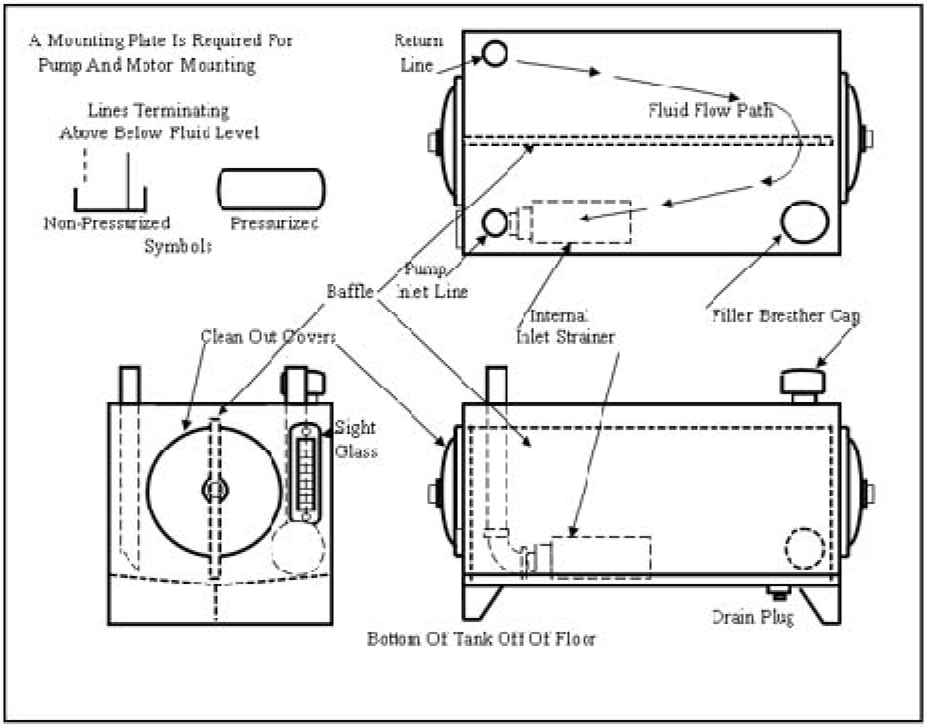


Pump under tank. Figure 6-4 *shows the very best pump/tank layout. This design puts the pump below the reservoir to take advantage of static head pressure. As explained in Chapter 1, there is pressure at the bottom of any column of fluid (about .4 psi per foot of elevation). With the tank above, the pump not only has fluid at its inlet all the time, but this fluid also could be at 2- to 4-psi positive pressure. (Note that this arrangement can be difficult to work on without ample headroom for the mechanic.) The same shutoff valve precaution goes for this layout as mentioned for the pump-alongside design.*

Tank functions

The main reason the reservoir exists is to store fluid. The accepted rule for sizing a tank is: the tank volume should be two to four times the pump flow in gpm. This is only a general rule. Some circuits may require more volume, while less fluid may be adequate for other circuits. A 25-gpm pump would work well with a 50- to 75-gallon reservoir for most circuits. With this general rule, the returned fluid theoretically will have two to three minutes in the tank before it circulates again. As ***Figure 6-5*** *shows, a baffle separates the return line from the pump inlet line, forcing the fluid to take the longest possible path through the reservoir before returning to the pump inlet. This arrangement also mixes the fluid well and provides more time to drop contaminates and de-aerate. In addition, the fluid spends more time in contact with the outer walls of the reservoir to dissipate heat.*

6-5. Standard features of non-pressurized reservoir designed for pump to be mounted on top.



When a circuit has single-acting cylinders or cylinders with large rods, the volume of fluid returned on the extend stroke is greatly reduced – or even non-existent. In these cases, the tank must be larger than the general rule states to keep fluid level from falling below the pump inlet line.

Another situation where a tank may need to be larger is if the circuit has accumulators. Accumulators need fluid to fill them at start up and space into which to discharge this fluid at shut down. An undersized reservoir may not have enough fluid to keep the pump inlet covered at all times.

*Another case for making the tank larger than the general rule is to add cooling capacity. All of the tank's exterior walls can radiate heat to the atmosphere, so the larger the tank the greater the heat dissipation. Use the formula in **Figure 6-6** to figure tank-cooling capacity. An example problem is shown later in this chapter. Several data books include formulas and charts showing tank-cooling capacity. These also can be used in conjunction with this manual.*

6-6. Formula for estimating how much heat a tank of a given size can dissipate.

$$HP = 0.001 \times (T_{Fmax} - T_{Amax}) \times A$$

HP = maximum power this size tank can dissipate — hp
T_{Fmax} = maximum allowable fluid temperature — °F
T_{Amax} = maximum ambient air temperature — °F
A = tank area in contact with fluid — ft^2

Heat dissipation is the main reason for having the tank bottom off the floor and why it is important not to stop free air flow around the tank. It is not good practice to enclose a power unit to reduce noise.

Tank components

The filler-breather cap should include a filter media to block contaminants as the fluid level lowers and rises during a cycle. If the cap is used for filling, it should have a filter screen in its neck to keep large particles out. It is best to pre-filter any fluid entering the tank . . . either by a filter cart transfer unit or by a filter fill unit (as shown in ***Chapter 2, Figure 2-2****.)*

Remove the drain plug to empty the tank when the fluid needs be changed. At the same time, the clean-out covers should be removed to provide access to clean out all residue, rust, and flaking paint that may have accumulated in the tank (and doesn't flow out with the fluid). If this is not done, the new fluid gets dirty immediately — defeating the purpose of the fluid change. In the design in Figure 6-5, the clean-out covers and internal baffle are assembled together, with some brackets to keep the baffle upright. Rubber gaskets seal the clean-out covers to prevent leaks.

If the system is badly contaminated, it is wise to flush all pipes and actuators while changing the tank fluid. This can be done satisfactorily by disconnecting the return line and placing its end in a drum, then cycling the machine. Do not over-fill the drum during this operation or it may rupture and spill fluid.

Sight glasses make it easy to visually check fluid level. Calibrated sight gauges provide even more accuracy. If the sight glass or gauge is difficult to see or is damaged, find another way to check the fluid level.

Many sight gauges include a fluid-temperature gauge. Tanks that feel hot to the touch may actually be within operating range. The temperature gauge gives a more specific indication. On older systems where the temperature gauge may have stopped working, it's best to check fluid temperature with some other method.

Pump connections

The inlet line to the pump should be at the same end of the reservoir as the return line, with the baffle between them forcing returning fluid to travel to the opposite end of the tank and back. The inlet must be below the fluid level and may include a strainer. If the inlet line is just a straight piece of pipe installed vertically, it is best to cut the line's open end at 45° so it is *impossible to butt it against the floor of the tank, which could block flow. Outside the tank, this line should lead as directly to the pump as possible, with no unnecessary bends or connections. Never use a pipe union in the inlet line; unions are almost impossible to seal against air leaks. Even a minute leak in the inlet line can cause pump cavitation and all of its problems.*

*The return line should be located in the same end of the tank as the inlet line and on the opposite side of the baffle (as shown in **Figure 6-5**). The return line must terminate below fluid level to reduce turbulence and aeration. The open end of the return line also should be cut at 45° to eliminate the chance of stopping flow if it gets pushed to the bottom. Another good practice is to point the opening toward the side wall to get the most heat-transfer surface contact possible.*

When a hydraulic reservoir is part of the machine base or body, it may not be possible to incorporate some of the features discussed in this chapter. However, keep in mind the different functions mentioned to try to eliminate ongoing problems.

Non-pressurized and pressurized reservoirs

Reservoirs are seldom pressurized because that feature is not required under most circumstances. One reason for using a pressurized reservoir is to provide the positive inlet pressure required by some pumps -- usually in line piston types. Another reason is to force fluid into a cylinder through an undersized prefill valve. Both of these reasons may require pressures between 5 and 25 psi and could not use a conventional rectangular reservoir design.

Another reason for pressurizing a tank is to keep out contaminates. If the reservoir always has a positive pressure in it there is no way for atmospheric air with its contaminants to enter. Pressure for this application is very low – 0.1 to 1.0 psi -- and may be all right even in a rectangular design tank.

*A pressurized reservoir would be built like any pressure vessel, but with the baffling and other features shown in **Figure 6-5**. Note however that the reservoir pictured in **Figure 6-5** is non-pressurized. The symbol for this type tank is shown at the left. The symbol also indicates how lines that terminate above and below fluid level are shown. If a drain line comes from an area that might have suction part of the time, it might not be best to terminate it below fluid level. If this type line terminated below fluid level, it could suck oil into the unit and possibly cause sluggish action. The drain line from the case drain of a pressure-compensated piston pump and an air bleed valve should terminate below fluid level at all times. This keeps air from being sucked in and causing problems.*

Heat in hydraulic systems

All heat in a hydraulic circuit comes from wasted energy. Any horsepower put into the circuit that does not do useful work wastes energy.

Any circuit has inefficiencies that can be up to 15% of input power. This is bypass fluid in pumps, valves or other components and pressure drop through these components and the flow lines. These losses can be reduced, but never completely eliminated in a typical hydraulic circuit.

Some ways to reduce inefficiency losses is to correctly size piping and valves, keep working pressure at or only slightly higher than required for all operations, and never allow fluid to relieve to tank. Flow controls also generate heat because they restrict flow. Reducing valves, counterbalance valves, and sequence valves also waste energy... especially if they are not set correctly. A pressure sequence or a counterbalance valve will do its job even when set too high, but will waste more energy at an unnecessarily high setting.

Each lost horsepower generates 2,545 BTU/hour of heat. To put this in perspective, 10 hp would heat a three-bedroom home when the outside air temperature is 30°F. Thus it is obvious what that much heat would do to the temperature of a 20-gallon tank of hydraulic fluid.

In a hydraulic circuit, you must calculate wasted horsepower to determine heat generation. In a highly efficient circuit this figure could be low enough to use the reservoir's cooling capacity to keep maximum operating temperature below 130°F. If heat generation is slightly higher than a standard tank will handle, it may be best to oversize the tank rather than adding a heat exchanger. An oversize tank is less expensive than a heat exchanger; and avoids the cost of installing water lines.

It is easy to figure heat generation by figuring horsepower input and subtracting horsepower output. With the gauge at a fixed-volume pump's outlet reading 150 psi and a gauge at the working cylinder reading 125 psi, there is a 25-psi pressure difference between energy in and work out. To figure horsepower loss, multiply (0.000583)(gpm)(psi). For this example, assume a 40-gpm pump. Then, lost horsepower = (0.000583)(40)(25) = .58. To determine actual heat loss, this figure must be divided by the percentage of the cycle that it occurs. If this figure is from the cylinder extending and the extend time was four seconds during a total cycle of 12 seconds, then figure 1/3 of the .58 hp or .19 hp as waste. Do this operation for each actuator in both directions of travel to determine total wasted horsepower. After all actuator losses are calculated, add them together to determine total wasted horsepower. (Note that when this horsepower total is less than the answer from the formula in ***Figure 6-6****, no heat exchanger is required.)*

The example just cited would be a straightforward circuit without any flow controls or other added restrictions. With a flow control in the circuit, pressure drop would be much higher and energy waste would increase drastically. Most circuits using a pressure-compensated pump would have flow controls so the pump would be at compensator setting while the actuator would be at whatever pressure is required. A circuit without flow controls or other restrictive plumbing usually has low energy losses. This type system may get by without a heat exchanger when ambient temperature stays below 110°F.

Tank cooling capacity

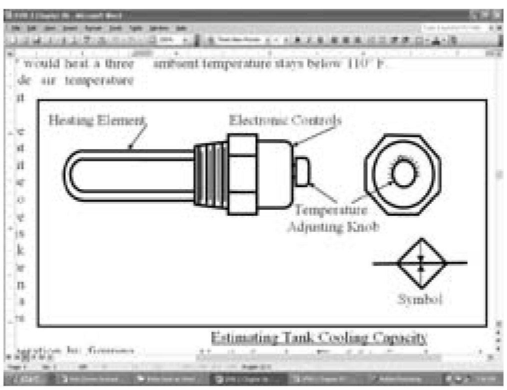
Use the formula in ***Figure 6-6*** *to estimate how much heat a given tank can dissipate. This formula assumes the tank is open on all sides with free air movement around it. (Remember, pipes, cylinders, and valves also have surface areas that can dissipate heat but those areas are usually not included when calculating cooling capacity.) For a 100-gal tank and a 30°F. temperature difference, this amounts to about 1.4 hp. A single cylinder circuit wasting only .7 hp* *would not require a heat exchanger on a 100°F. day while maintaining a maximum fluid temperature of 130°F.*

Heating and cooling devices

Tank heaters: Most industrial hydraulic units operate in a warm indoor atmosphere so low temperature is not a problem. For circuits that see temperatures below 65° to 70°F., some sort of fluid heater is recommended. The most common tank heater is an electric-powered immersion type. These units consist of resistive wire in a steel housing with a mounting option. They often have an integral thermostatic control. The heating element on most units contacts the fluid directly, similar to a hot water heating system. Some immersion heaters in a sealed housing heat the air in the enclosure. The air then transfers heat to the fluid. This heater-in-tube arrangement allows the heating unit to be changed without draining the reservoir.

Figure 6-7 *shows an electric immersion heater and its ISO symbol for schematic drawings. (This symbol may be replaced by some sort of pictorial rendition of the heating unit on many schematics.)*

6-7. Typical electric-powered in-tank heater.



Electric heating units should not have concentrated heat like those used to heat water. Oil viscosity at low temperature is thick enough to retard movement. This could allow fluid next to the heating rods to overheat and possibly breakdown. The usual recommendation is for the heating rods not to have over 8 to 10 watts per square inch density. This limit may require multiple heating rods to meet the heat requirement of some systems.

Another way to electrically heat a tank is with a mat that has heating elements similar to those in an electric blanket. This mat is attached to the outside of the bottom of the reservoir and adds heat during low temperature conditions. This type heater requires no ports in the tank for insertion. It also evenly heats the fluid even during times of low or no fluid circulation.

Heat can be introduced through a heat exchanger by using hot water or steam in place of the cooling water. The exchanger becomes a temperature controller when it also uses cooling water *to take away heat at other times.* ***Figure 6-10*** *depicts the symbol for this type heat exchanger. (****Chapter 7*** *shows an alternate way to add heat while filtering system fluid.) Temperature controllers are not a common option in most climates because the majority of industrial applications operate in controlled environments.*

The formula for estimating how many kilowatts are needed to heat a certain size tank from an expected minimum ambient to a nominal working temperature of 50° to 70°F. is shown in Figure 6-8. Use it to size a heater when the tank is exposed to low temperatures.

6-8. Formula for estimating heater capacity to increase fluid to a minimum temperature.

$$kW = \frac{C_T \times (T_{Fd} - T_{Amin})}{800 \times t}$$

kW = power required to heat fluid — kW
C_T = capacity of tank — gal
T_{Fd} = desired fluid temperature — °F
T_{Amin} = minimum ambient air temperature — °F
A = tank area in contact with fluid — ft^2
t = allowable time — hr

Note: To minimize the size of heating elements, allow 1 to 3 hr for heating in this worst-case scenario.

To only maintain a desired temperature, reduce power (kW) by $\frac{1}{2}$ to $\frac{2}{3}$ based on time (t).

Heat exchangers

Cooling hydraulic systems is necessary more often than heating them due to wasted energy from inefficiency and/or poor circuit design. A well-designed circuit eliminates most heat generation and may not need a heat exchanger. Use the same method to estimate how much heat a system generates as was used for the previous tank-cooling example.

When figuring wasted horsepower, see if there is any way to reduce or eliminate it so it does not have to be paid for twice. It costs money to produce the unused heat and it is expensive to get rid of it after it enters the system. Heat exchangers are expensive, the water that through them is not free, and maintenance of this cooling system can run high.

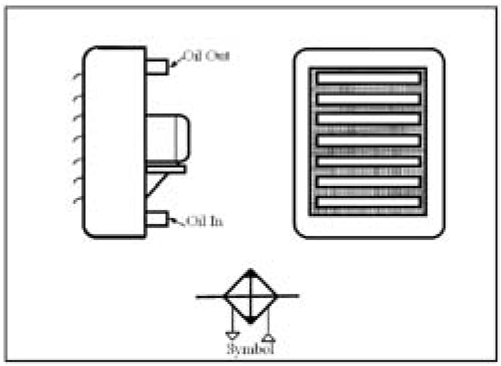
Items such as flow controls, sequence valves, reducing valves, and undersized directional control valves can add heat to any circuit. Are these items absolutely necessary? Can they be replaced with another valve or part that does the same thing with less pressure drop? Anytime these questions can be answered with a yes, the circuit is not ready to build.

Air-cooled heat exchangers

After calculating wasted horsepower, review heat exchanger manufacturers' catalog to pick out a unit that will dissipate that amount of energy. Most catalogs include charts for given size heat *exchangers that show the amount of horsepower and/or BTU they can remove at different flows, oil temperatures, and ambient air temperatures.*

Figure 6-9 *shows a typical air-cooled heat exchanger that may be used in place of a water-cooled unit in some applications. Air-cooled heat exchangers are not as efficient as water-cooled heat exchangers, but they require only an electrical outside hookup. They work well in cool atmospheres or when the amount of heat to be removed is low. Note that airborne contaminants such as heavy dust or water and coolant vapors can quickly reduce an air-cooled heat exchangers low efficiency to almost nothing. Some manufacturers offer a filter pack to take out airborne contamination before it clogs the heat exchanger's radiator fins and tubes.*

6-9. Typical radiator-and-fan oil cooler.



On circuits with pressure-compensated pumps, a small air-cooled heat exchanger often is used to cool case drain flow. On this type system, most of the heat generation is from internal leakage and control oil that flows to tank through the pump case drain. One type heat exchanger -- called a coupling cooler -- is a finned tube formed into a circle and wrapped around a blower that is driven by the motor turning the pump. A similar arrangement uses a small flat radiator attached to the intake end of the fan-cooled electric motor that drives the pump. Both units are low-flow, low-backpressure devices and dissipate only a small amount of heat.

Some systems use a water-cooled heat exchanger in the summer and an air-cooled one in the winter. This arrangement eliminates plant heating in summer weather and saves heating expense in the winter.

Water-cooled heat exchangers

As before, manufacturers' catalogs will assist you in picking out a unit to dissipate the amount of wasted heat energy. For water-cooled heat exchangers, catalogs ask for information such as how much and what temperature water is available, how many horsepower or BTU of energy must be dissipated, what is the fluid flow in gpm, and how many passes will the water makes to get through the body. The more passes -- up to four maximum usually -- the greater the heat dissipation per gallon of water flow. Charts that use this information make it easy to pick the correct size heat exchanger.

6-10. Two popular water-cooled heat exchanger designs.

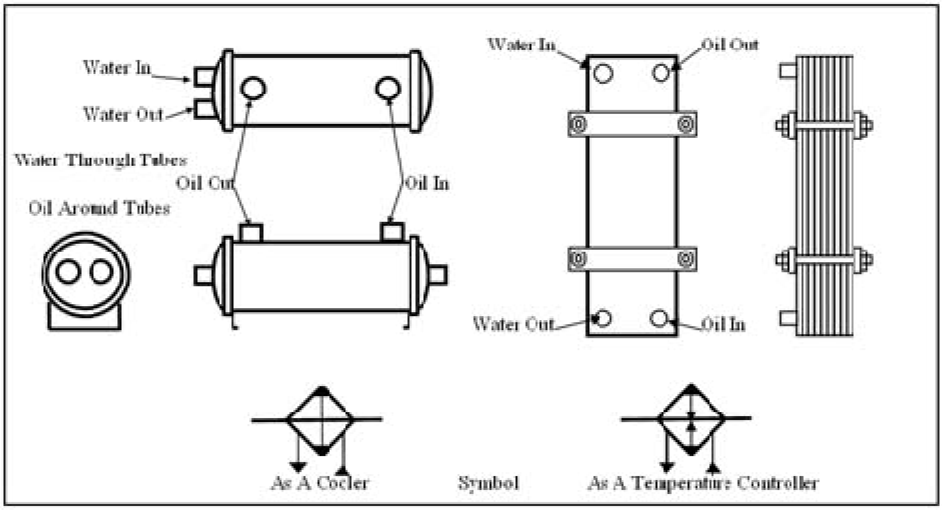
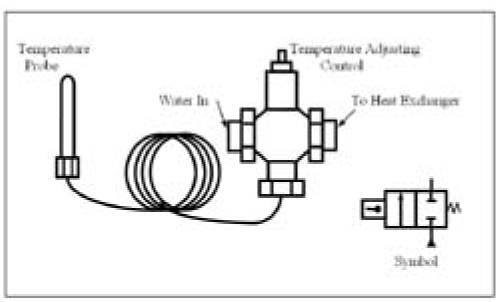


Figure 6-10 *shows two types of water-cooled heat exchangers commonly used for hydraulic systems. The shell-and-tube design is the most common one at present, but the plate-and-frame or brazed-plate type are coming along because they are much smaller and easier to maintain. It is important to use clean water in either type unit to keep from building up insulating deposits or corroding the tubes until they leak water into the oil. Treated water from a cooling tower works best.*

6-11. Temperature-controlled water valve.



Either type heat exchanger should have a thermostatic control to turn on the water or fan only when the fluid temperature rises above its normal operating range. Without a thermostatic control, fluid could be too cold and thick, while wasting the energy to operate the heat exchanger. ***Figure 6-11*** *shows one type of water-control valve that requires no electrical hookup. Heat-sensitive liquid in a thermometer-type probe in the tank expands and opens a water valve upon reaching a preset temperature. The temperature is fully adjustable to meet any requirement and it operates in all types of fluid. Another option is an electrically operated temperature sensor that controls a solenoid-operated water valve. This installation requires an electrical hookup but is able to maintain a fluid-temperature range at any desired setting.*

Fluid power filters

Moving parts in fluid power systems are subject to wear from contamination. Neither atmospheric air nor hydraulic fluids are clean enough as supplied to avoid this and they both *become more contaminated with use. Therefore all fluid power systems require filters to remove contamination and thereby increase component life.*

Pneumatic filters and lubricators

A pneumatic filter should be the first component at the inlet of most air circuits. This unit usually is one part of a combination of components that filters the air, regulates its pressure, and adds lubricants for moving parts in the circuit. The air filter and lubricator are covered in this section. (An air line regulator performs the same function as a hydraulic reducing valve and is covered in that section.)

Air from the compressor contains dust from the ambient atmosphere, condensed water, and rust and oil sludge that bypass the compressor rings. These by-products of compressing and transmitting air must be removed to keep moving parts of the machine working properly. Most filters clean the air and separate condensed water from it before the air enters the circuit.

7-1. Cross-section of typical air filter, with ISO symbols.

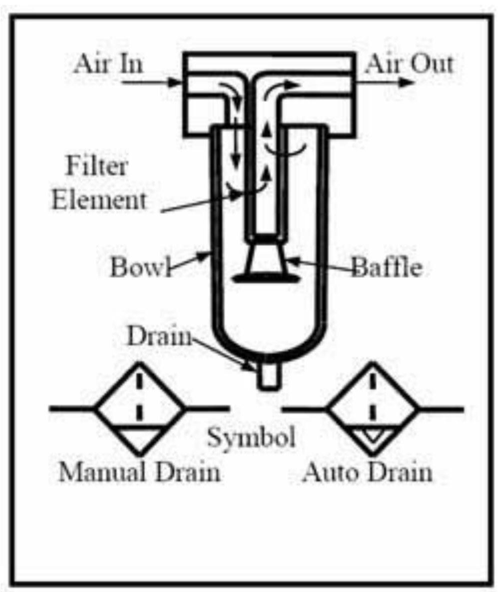
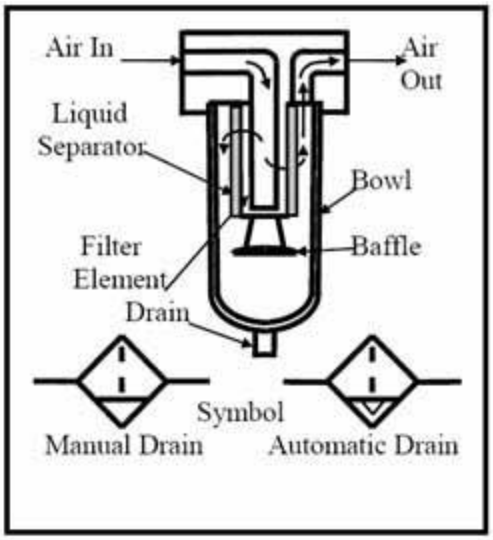


Figure 7-1 *shows a simple air filter (and the ISO symbols that represent it). These units usually require little attention if the compressor has an air dryer at its outlet. Air enters at the left and is channeled into the bowl with a downward circular motion. The centrifugal force of this swirling action slings water droplets outward. They collect and fall to the bottom of the bowl below the baffle into a quiet zone for draining either manually or automatically. The air then flows through a porous filter element to the outlet. These units typically remove particles of 40-micron (40-µ) size or larger but they also are available for particles as small as 5 µ if required.*

7-2. Cross-section of coalescing air filter, with ISO symbols.



*The coalescing filter shown in **Figure 7-2** removes water and oil vapors as well as condensed moisture from an air line. To accomplish this, coalescing filters are not only designed differently but they have reverse flow in relation to a standard filter. These filters will remove particles as small as 0.3 to 0.6 µ. They use a coarse mat of very fine fibers that are small enough to catch and hold these very fine particles. As the vapor collects into droplets, they are channeled to the quiet zone and drained.*

Coalescing filters must be applied according to manufacturers' specification to keep collected liquid from being re-entrained into the air stream. Also, always make sure the flow direction is according to information on the filter housing. Several companies make these high-efficiency air filters though they are not often applied to every day circuits.

7-3. Cross-section of typical air-line lubricator, with ISO symbol.

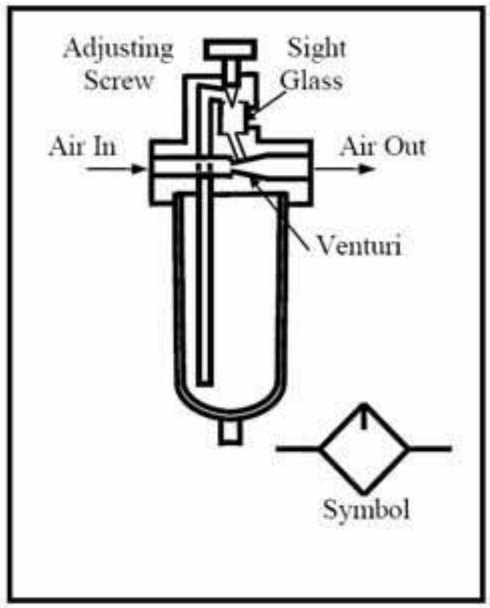


Figure 7-3 *shows a cutaway view of an air-line lubricator. After the combination unit filters and regulates air pressure, some downstream system components may require a small amount of lubrication. (For example: air motors are one item that needs a constant supply of oil to extend their life and maintain torque.) Some cylinders are pre-lubed and most valves require little if any lubrication, so keep oil supply to these units at minimum. A general rule: a ½-pint bowl of oil in a lubricator should last three weeks to a month in most situations.*

When air passes through the lubricator's venturi section, pressure drop across it gives a negative pressure in the area below the adjustable orifice. Vacuum in this area draws oil from the bowl as fast as the adjustable orifice will allow. These droplets then mix with the air as it passes through. This arrangement means that oil flows only when there is air flow and only as fast as the adjusting screw allows.

Air line lubricators are designed to send a mist of oil to the parts in the downstream circuit. However, the physical size of some circuits makes it impossible for the mist to stay in suspension long enough to reach some parts. In this case -- and for some air motor applications -- it is necessary to inject oil at the components inlet. There are electric and air-driven lubrication units to meet the needs of these applications.

Compressed-air dryers

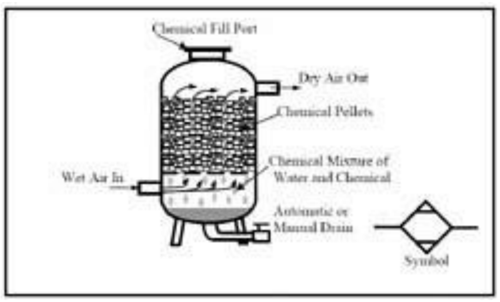
Why are air dryers necessary? All air compressors take in atmospheric and compress it eight or ten times to a pressure between 110 and 125 psig. All the moisture vapor and heat in the atmospheric air is also compressed and concentrated, so air at the compressor outlet is hot and wet. The temperature of the air at the compressor outlet can be as high as 350ºF. The air also can be saturated with water vapor. As this air cools in the receiver and plant piping, the water vapor in it condenses into water droplets.

A 25-hp compressor running at 75% capacity pumps 18 gallons of water into a plant air system on a day with average ambient humidity. An aftercooler condenses and removes about 11 gallons of this liquid, but that still leaves 7 gallons to collect in low spots, retard valve movement, damage production parts, and cause problems in general.

Types of air dryers

Figure 7-4 *shows a cutaway view of (and the symbol for) a deliquescent air dryer. Wet air enters the dryer (which is a pressure vessel), passes up through a bed of hygroscopic chemicals, and flows on to the outlet. The chemicals (often a form of sodium) absorb moisture from the air as it passes through the bed. As they dry the air, the chemicals break down into a slurry of water and chemical drops that falls to the bottom of the tank. A manual or automatic drain keeps the water mixture from rising too high and mixing with inlet air flow.*

7-4. Absorbent-type deliquescent air dryer.



A typical deliquescent dryer removes moisture to a dew point of about 40ºF. Air that has passed through this dryer must be cooled below 40ºF before any more moisture will condense. This dew point is satisfactory for most plants -- even during winter cold. However, the hotter the air passing through the chemicals, the less the amount of moisture the chemicals will collect. Thus it is important to keep the incoming air at or below 100ºF. This usually requires an upstream aftercooler to lower the temperature of the air being delivered from the compressor. Air at higher temperatures at the inlet results in higher dew points at the outlet.

Deliquescent dryers are the least expensive of the three types mentioned in this section, but they might cause problems in some installations. There is always a chance of the chemicals or their vapors being picked up by the air stream and sent into the pneumatic system. These chemicals are corrosive and can damage internal parts. Also chemicals must be replenished on a regular basis. This means shutting down the compressor or bypassing the dryer when chemicals get low. Finally, slurry must be removed.

7-5. Condensing-type refrigeration dryer.

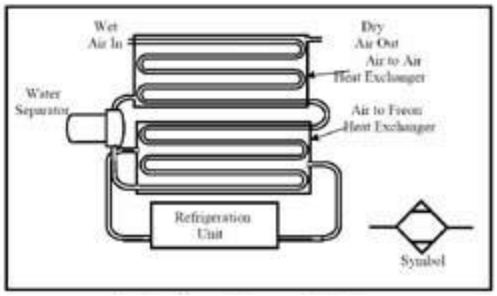


Figure 7-5 *shows a cutaway view of a refrigeration-type dryer. This unit condenses water vapor by cooling compressed air to lower its dew point. There are no chemicals to replace or break down, so the system can run as long as required. A refrigeration dryer is more complex and more expensive to purchase than chemical dryers, but has a lower operating cost. The main operating cost is electricity. Maintenance cost is minimal and life expectancy is high, so this type dryer can be cost effective over the long haul.*

To keep operating cost down these systems usually have an upstream water-cooled aftercooler to cool compressor air and take out the bulk of the moisture. Normally a maximum incoming temperature of 100ºF is recommended. When a higher inbound temperature is present, oversize the refrigerated dryer to handle the extra energy removal. (Of course, this adds extra cost to the purchase of the unit, as well as increasing operating cost over the life of the system.)

Wet air enters the unit through an internal air-to-air heat exchanger. This unit is piped to pre-cool the hot inbound air and re-warm the cold dry air before it exits the dryer to enter the plant piping. This arrangement saves energy by removing some of the heat in the incoming air. It also keeps the plant piping from condensing water vapor from ambient air on its cold exterior surfaces and dripping water on the production floor.

The pre-cooled air then passes through a Freon-to-air heat exchanger that reduces its temperature to approximately 35ºF. This procedure condenses more water vapor to achieve a 35ºF dew. If plant temperature stays above 35ºF (as it does in most plants), there will be no more condensation inside the air piping. The condensed water drains from the dryer through a water separator.

The main potential problem with refrigerated dryers is they will freeze up if temperature is set too low or if the amount of air passing through them is low and intermittent.

Another approach to refrigerated drying is drying the air before it enters the compressor. One company offers chillers that take atmospheric air down to –40ºF. and feed it to the compressor. After it is compressed to 100 psi, the air has a pressure dew point of approximately 35ºF. Because the compressor takes in air that is denser and at such a low temperature, the heat of compression is negligible. Also, most of the airborne contaminants are removed during the cooling, condensing, and freezing process. These inlet air dryers use dual refrigeration units. While one is drying input air, the wasted heat from the drying unit defrosts the other.

7-6. Adsorbing-type desiccant air

dryer.

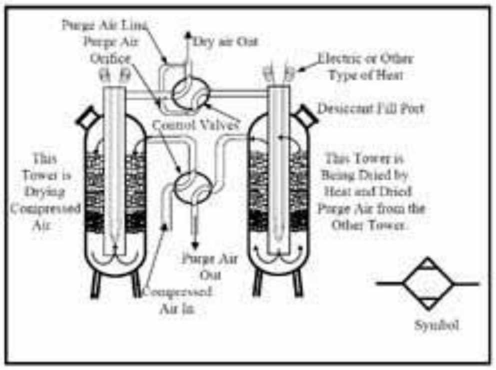


Figure 7-6 *pictures a twin-tank desiccant dryer that uses a hygroscopic material (such as silica gel or activated alumina) that collects water vapor but is not broken down by it. This type dryer is called an adsorber because it collects water vapor but once the moisture is removed by heat or other methods, the chemical is ready to work again. This desiccant dryer may achieve dew points of –40ºF or lower, so the air can be used in most outdoor circuits without fear of freeze ups.*

Electric heaters and purge air from the opposite tank handle the drying process. Other methods are steam heat, dried purge air only, and desiccant replacement.

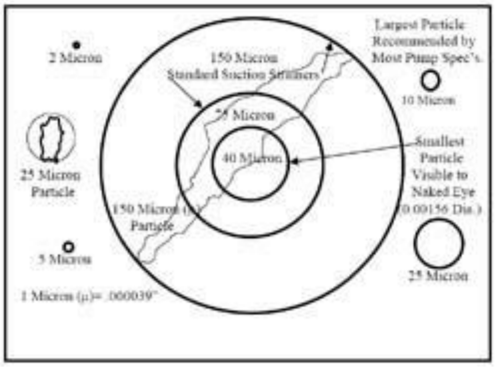
As wet compressed air enters the control valve, it is channeled to one of the desiccant tanks. (The control valves are usually set up to shift automatically, triggered by a signal from a device that monitors the output air's dew point.) Wet air is forced through the desiccant material to take out water vapor, then sent on to the plant. Some of the dried air is diverted through an orifice to the spent tank, where it is heated and then passed through the wet desiccant in the other tank. Water again vaporizes and exhausts to atmosphere. This drying process continues at the rate necessary to maintain the required dew point.

Desiccant dryers are the most expensive type to operate and maintain. In particular, they are subject to failure if the incoming air contains carry-over compressor oil. The oil can coat the desiccant and make it incapable of collecting moisture. The main expense of operating these dryers is the energy used to dry out the idle tank of desiccant.

Hydraulic filters

In ***Figure 7-7****, a schematic drawing of a hydraulic circuit shows filters in the standard locations, with typical filtration ratings listed next to them. Note that most circuits would not have all of these filters, but every circuit should have adequate filtration to protect the pump, valves, and actuators from contamination.*

7-7. Relative size of contamination particles at 500X magnification.

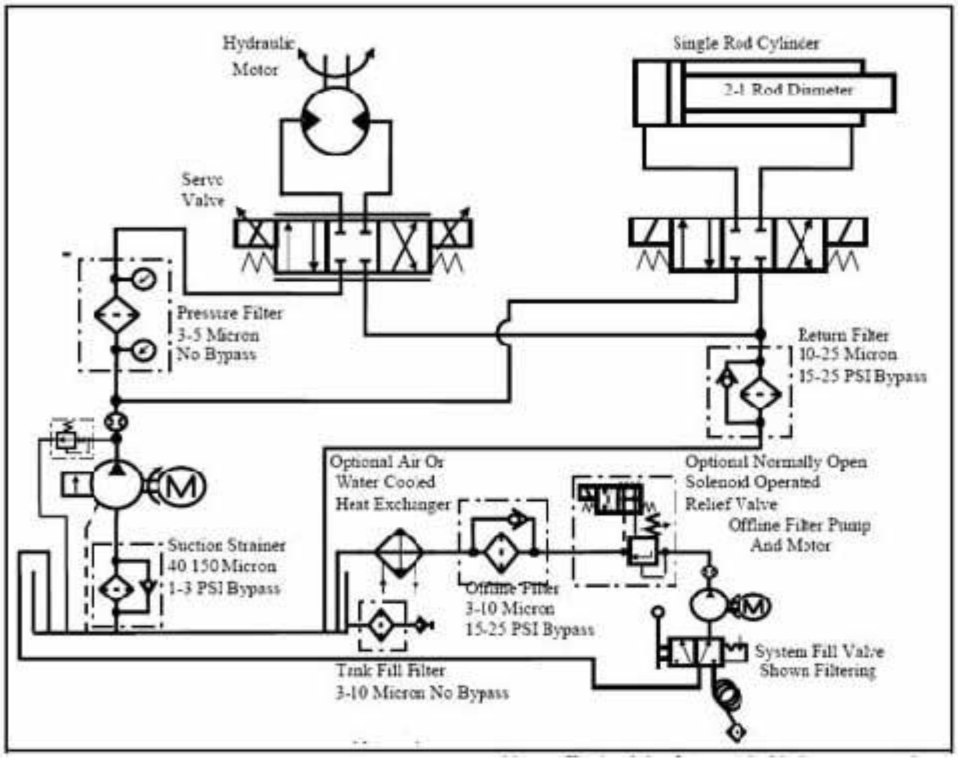


There are several sources of contamination in and around hydraulic units. Normal component wear, contamination in new oil, sloppy filling practices, poor plumbing installation, and dirt carried in on piston rods are the main ones. Some of these areas are simple to address ahead of time, while others can only be handled by filtration.

New oil from the supplier is not as clean as most hydraulic circuits require. At best about 25-µ cleanliness is all most suppliers will offer. One reason: the drums that new oil comes in probably were used before. Although they were cleaned, they are not contaminant-free. In addition, a standard drum pump draws fluid from the bottom of the drum where all the residue has settled.

Figure 7-8 *shows a manually operated 3-way valve installed in the suction line of an offline filter pump. When this valve is shifted, all new fluid has to go through the offline filter before it enters the tank.*

7-8. Schematic drawing of hydraulic circuit showing typical locations for filters.



*Another item in **Figure 7-8** is a tank-fill filter with no bypass, which filters all new oil entering the tank. Another option is to use a purchased filter cart to fill the tank through its normal fill port.*

Whatever the method, it is important to keep filling practices from introducing contamination. While the fill port offers one of the simplest places to address contamination problems, it often is overlooked.

Another way that contamination enters a hydraulic circuit – even before startup -- is through poor plumbing practices. All pipe, tube, and hoses should be inspected for contamination before installation and cleaned if necessary. It's good practice to seal clean conductors with caps until installation time. Use care when cutting and preparing pipe or tube ends to make sure no metal chips or filings stay in the conduits. If the system includes servovalves, flush it with filtered oil through flushing covers for the time recommended by the valve manufacturer before startup. Use every possible precaution on a new or replacement plumbing system to make startup go smoothly.

In a running circuit, one of the most common sources of ingressed contamination is the cylinder piston rods. Every time a piston rod extends, it is damp with system oil. In a dusty atmosphere this oil-dampened piston rod attracts and holds fine particles. Many of these are dragged back into the cylinder and washed off. Most cylinders have a rod wiper to help keep out contamination but this wiper only catches large pieces. Everything smaller passes by. This type contamination must be filtered out continuously to protect system components. Some suggest flexible boots or bellows over the rod end of the cylinder to eliminate rod contamination ingression. A flexible *bellows does a good job unless it gets a tear or other type hole. At this point it actually sucks in ambient air with its contamination and holds it closely to the rod.*

A similar situation takes place at the tank breather. During each cycle, the fluid level in the reservoir changes -- either drawing dirty atmospheric air in or discharging air through the breather. The filler breather should be capable of trapping the inbound contaminants in this air flow.

The other main cause of contamination is normal component wear. Dynamic pumps, motors, and cylinders have constantly rubbing metal-to-metal contact areas. Even with good lubrication, small eroded particles get into the fluid. This contamination must be constantly captured by filters to eliminate a damaging buildup of residue.

Condensed water is another form of contaminant that should be addressed. Water in hydraulic fluid can cause corrosion, break down fluid additives, and make viscosity vary. Water vapor often enters the system through the breather and condenses to liquid form when the tank cools. Using a breather with a hygroscopic media can eliminate most water contamination. Most of these hygroscopic breathers must be changed when they become saturated. They often use silica gel that is blue tinted when dry and turns a pink tint as it gets wet, signaling that a change is needed.

Cleanliness levels

The circles in ***Figure 7-7*** *indicate relative sizes of contamination particles. Not all contamination is nice round marble like pieces as the two overlaid examples show. Particle size is measured by enclosing it in a circle until it touches at two or more places. The 150-micron (150-μ) particle may only be 20 to 25 μ thick but is considered 150 μ because of its length.*

Most pump manufacturers specify at least 10-μ clean fluid to protect their pumps from premature failure.

ISO has set up cleanliness level standards for hydraulic and lubrication fluids. The system most used for hydraulics is based on the ISO 4406 standard. This standard covers the number of particles of a given size that can be present in a fluid sample in three different micron ranges. It is designated ISO Code XXX/XXX/XXX, where the numbers relate to the minimum and maximum number of particles of a given size that can be present in a 100 milliliter sample. The first number indicates how many particles of 2-μ size can be present; the second number is for 5-μ particles; and the third number is for 15-μ particles. It might be written ISO Code 18/16/13. The numerals always descend in value from left to right. This code would mean that there could be between 1300 and 2500 particles of 2-μ size, 320 to 640 particles of 5-μ size, and 40 to 80 particles of 25 μ.

ISO 4406 Chart		
Range Number	Number of Particles per ml	
	More Than	Up to and Incl.
24	80,000	160,000
23	40,000	80,000
22	20,000	40,000
21	10,000	20,000
20	5,000	10,000
19	2,500	5,000
18	1,300	2,500
17	640	1,300
16	320	640
15	160	320
14	80	160
13	40	80
12	20	40
11	10	20
10	5	10
9	2.5	5
8	1.3	2.5
7	.64	1.3
6	.32	.64

***Figure 7-7** indicates that, even with good eyesight, people cannot see a particle smaller than 40µ. Thus it is impossible to look at a sample of fluid and determine whether it is clean. Of course, it is possible to tell it is contaminated when large particles are plainly visible in the sample.*

*The **ISO 4406 chart** shows the range number and the number of particles that it represents. From this chart it is easy to set up or pick out any ISO Code cleanliness level.*

Some typical fluid cleanliness level ISO Code's are shown in the following chart:

Beta ratios

*Another rating applied to hydraulic filters is the Beta ratio (also known as the Filtration ratio). It is a measure of the particle-capture efficiency of a filter element. The ISO 4572 Multipass Test passes fluid through the circuit shown in **Figure 7-7** to check for contaminant retention. A measured amount of contaminant is injected upstream of the filter. Laser particle counters record the number of particles into and out of the filter. When 100,000 particles are measured upstream of a 10-µ filter and 10,000 downstream, it would have a Beta ratio of 10 (100,000/10,000 = 10).*

Component ISO chart

COMPONENT	ISO Code
Servo Control Valves	16/14/11
Proportional Valves	17/15/12
Vane and Piston Pumps	
And Motors Directional and	18/16/13
Pressure Control Valves	18/16/13
Gear Pumps and Motors	18/17/14
Flow Control Valves and Cylinders	20/18/15
New Unused Oil	20/18/15

A Beta Ratio number is of no use alone, but it is required to find the filter's efficiency rating. Efficiency of a filter element is what counts when comparing one filter to another. The higher the efficiency, the fewer contaminants will pass through it. Efficiency coupled with the volume of contaminant retention can make a more expensive filter cost less due to its longer useful life.

Efficiency is calculated by the formula:

$Efficiency_{10} = (1–1/10)\ X\ 100$
$Efficiency_{10} = (0.9)\ X\ 100$
$Efficiency_{10} = 90\%$

Always make sure the filter meets or exceeds the desired cleanliness level of the system it is protecting.

Filter locations

***Figure 7-8** shows most of the locations where filters might be found in any hydraulic circuit. Note that all of these filters are seldom found in a single circuit but some circuits might have filters in two places. Two other types of filtration -- off-line and filter fill -- also are shown in **Figure 7-8**. Any of these filters could be the dual, change-on-the-run type when required. Dual filters are more expensive but can reduce downtime.*

Suction strainers

*A suction strainer (as shown at the lower left in **Figure 7-8**) often is found on the pump inlet line. Strainer is a common term for filters with openings of 75 m1 or larger. Strainers on the pump inlet line protect the pump from large, damaging contaminant particles that could cause catastrophic failure. These particles might be startup debris left in the tank and piping or large contamination introduced to the system from external sources or from internal part failure.*

Pumps without supercharged inlets can only tolerate a portion of one atmosphere pressure drop without affecting inlet flow. With this low pressure drop, (14.7 psi maximum at sea level on an average day), a restriction such as a low-micron filter can cause the pump to cavitate.

*Cavitation causes pump failure faster than dirty oil; so avoid it in every situation. (See the write up on pump inlet conditions and cavitation in **Chapter 8**.)*

Suction strainers are available with 75- to 150-µ openings. Some manufacturers have inlet filters with ratings as low as 25µ. A low-micron element needs large filtering surfaces to keep pressure drop low. When a pump is force fed by another pump -- sometimes called a supercharging pump -- a low-micron rated element can be used. The supercharging pump forces fluid through a very fine filter to the working pump, thus keeping it from cavitating.

A suction strainer or filter should have a bypass relief valve. Set the bypass to open at a pressure of 1 to 3 psi if the strainer clogs. The reasoning behind this is that the pump will run many hours on contaminated oil, but will fail in a short time with little or no oil. Suction strainers may be located inside or outside the reservoir. Internal strainers are less expensive, but their condition is more difficult to monitor. External strainers are easy to service and often have an indicator to show when the filter starts bypassing. The indicator can be as simple as a vacuum gauge or it might be a vacuum-operated electrical output to a warning light, alarm, or shut-down controller.

Many older circuits have nothing but a suction strainer for filtration. Retrofitting these systems with off-line or kidney-loop filters (discussed later) is advisable.

Return-line filters

*Another common location for filters is in the return line. (**Figure 7-8** indicates this location at the right center.). A return-line filter keeps most contamination that is caused by part wear or ingestion from getting into the tank, this protecting the whole system. Return-line filter protection ratings typically range from 3 to 25 µ. Obviously, you should select a return-line filter rated for the desired system-cleanliness level or less.*

Return-line filters should have integral bypass check valves. If the filter becomes loaded, return oil needs an open flow path to tank until it is convenient to change the filter. Without a bypass, the filter element probably will collapse, or the element housing or seal may rupture.

Typical bypass checks require 10 to 50 psi to open. The bypass pressure should be high enough to stop fluid from going around the filter during normal conditions, but low enough to avoid damaging filter element and its housing seal.

Some designers size return-line filters just large enough to handle the pump's rated flow. This can cause problems, especially if cylinders in the circuit have oversized rods, or if one cylinder must return one or more other cylinders. For example, if a cylinder has a 2:1 rod diameter, flow to tank while the cylinder is retracting is double the pump flow. Sizing the filter just for pump flow in this case allows contaminated oil to bypass the filter, and may damage the housing or seals. Paper filters can collapse, have holes blown through the element, stop filtering, and never indicate they need replaced. On pleated elements, the pleats can collapse, giving a "loaded element" indication prematurely.

Even with a correctly sized return-line filter, flow through it changes constantly. Steady flow through the element gives the most efficient filtering. If a filter passes constant flow, the bypass valve will not open until the filter fills with contaminants. This means only clean fluid leaves the filter. Visual and electrical indicators are available to show when a return filter is bypassing.

Pressure filters

*Another location for filters is in the pressure line (as shown at the left middle of **Figure 7-8**). These filters are mandatory in systems using servovalves. Servovalves have low contamination tolerance. They have small internal orifices, very close tolerance fits, and must shift rapidly at low pilot pressure differential. A servovalve can stop functioning in as little as two minutes when supplied with oil that is clean enough for a typical hydraulic system. Even when a 3-µ return-line filter is in place, contamination generated by the pump is enough to shut down a servovalve in a short time. (Note that fluid for a proportional-valve circuit often requires the same cleanliness level as a servovalve circuit to maintain fast response and consistent operation.)*

Actually, pressure-line filters would be an added advantage for any hydraulic circuit, but high initial and replacement cost limits their use. Pressure-line filter housings must be strong enough to withstand full system pressure. When there is a high pressure drop across the filter, the element must not collapse. These requirements make filter housings and elements much more expensive than other type filters. Pressure-line filters usually have elements with 1- to 5-µ openings. The pressure-line filter should have an absolute rating, or have a Beta Ratio of 50 or higher. A pressure-line filter should not have a bypass. If the filter element clogs, it is better to stop flow to a servovalve than to contaminate it. Visual and electrical clogging indicators are available for most pressure-line filters. They warn of potential clogging so that elements can be replaced well before production speed is affected.

Off-line filtration

*Off-line filtration systems -- sometimes called kidney loops or bypass filters -- consist of a separate pump, motor, and filter circuit that takes oil from the reservoir and re-circulates it. The system pumps oil from one end of the tank, passes it through a filter, and returns it to the opposite end of the tank. **Figure 7-8** shows an off-line filtration system at the lower right. This arrangement is a good way to provide high-micron continuous filtration. The systems are easy to retrofit to existing hydraulic circuits and offer an excellent way for new installations to get high cleanliness levels.*

Off-line filter circuits are usually rated in the 3- to 10-µ range and should sized to filter the volume of fluid in the reservoir every 1 to 3 hours minimum. This low, constant flow rate makes the filter very efficient, never opens the bypass, never causes media channeling, and never blows holes in the element.

When the filter indicator shows a clogged element in the off-line system, the main hydraulic circuit can continue to operate during filter change. Conversely, this off-line filter system can continue to run when the main hydraulic circuit is off overnight or weekends.

System-fill filters

New oil is not as clean as most hydraulic systems require so it should be filtered before use. To do this, introduce new oil to the tank through a pair of shut off valves, or a 3-way ball valve (as shown in ***Figure7-8*** *at the lower right) in the suction line of the off-line filter pump. Rotate the 3-way ball valve 180 degrees, hooking the off-line pump's suction to a flexible hose from the oil drum or fluid container. This setup filters all oil from the fluid container as it fills the reservoir.*

Another way to make sure all fluid is filtered before use is through the use of a tank-fill filter (as shown in ***Figure 7-8*** *at the lower middle). Here a low-micron pressure filter is installed in the tank wall and provides the only way to fill it. The filling process can only introduce clean fluid to the reservoir.*

Additions to a filter loop

An off-line filter circuit also can provide heating or cooling functions. ***Figure 7-8*** *shows a bypass circuit with a normally open solenoid relief valve, a high-horsepower motor, a temperature switch, a heat exchanger, and a temperature-controlled water valve or switch. These additions can effectively control temperature while filtering the fluid. To only filter the oil, leave the water or fan turned off and the solenoid relief valve de-energized (or open).*

If oil temperature drops, a temperature switch energizes the solenoid on the relief valve, and pressure rises. All electric motor horsepower converts to heat until the temperature switch indicates correct oil temperature. Unlike an immersion-type electric tank heater, the fluid is being circulated, so there are no hot spots.

For every electric horsepower, there will be 2544 Btu/hour heating capacity. After figuring the Btu/hour to heat or maintain minimum temperature, divide by 2544 to calculate the horsepower needed. (The formula for calculating tank heating appeared in ***Chapter 6****.)*

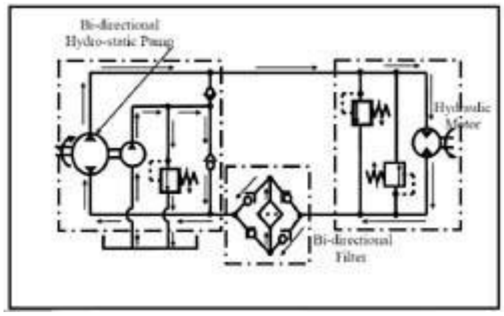
If the tank fluid temperature goes over a preset limit, a temperature-controlled water valve opens to send water through a heat exchanger or a temperature switch turns on the fan of an air-cooled heat exchanger. All filtered flow is cooled when the temperature-control device indicates elevated temperatures.

When installed in an off-line filtration loop, the heat exchanger receives constant flow, so it needs no bypass valve. Also, the heat exchanger sees flow even when the system uses a pressure-compensated pump.

Bidirectional pressure filters

The only difference between a bidirectional pressure filter, ***Figure 7-9****, and a standard pressure filter is the four check valves in the housing. These check valves cause oil flow to pass through filter element in the same direction regardless of the direction fluid enters the housing. Another name for a bidirectional filter is last chance filter. It is installed in a working line to an actuator* *so it must withstand maximum system pressure. Bidirectional filters with 3- to 10-μ ratings are adequate for most circuits.*

7-9. Schematic drawing of hydrostatic drive circuit with bi-directional filter.



Closed-loop hydrostatic transmission circuits are one place to use bidirectional filters. The oil can stay in the loop between the pump and motor for long periods. Any contamination in this closed loop continues to cause damage, even with ample filtering of oil in the tank.

Hydraulic filters

Contaminated fluid causes most hydraulic system failures. Oil in a reservoir may look clean to the naked eye, but silt contamination particles too small to see can still wreck pumps, cause valves to stick, and erode cylinder bores. In many facilities, components may take the blame for problems in error, when contaminated fluid is the culprit. It is amazing that some plants will change pumps every six months (believing that is normal component life), when they could add a proper filtration system and get many times longer pump service life.

A well-filtered hydraulic system should not have particles in the fluid larger than 10 microns. (A micron is 0.000039 inches.) A contamination particle that measures 0.001 in. across is 25 microns. The smallest dirt particle that is visible to the naked eye is 40 microns. Simply looking at an oil sample is not a good way to tell if the filters are cleaning the fluid.

Nominal or absolute are common terms found in hydraulic filter micron ratings. A filter with a nominal rating takes out most of the particles that measure the same size or larger than the stated micron size. A filter with an absolute rating takes out all particles the same size or larger than the rated micron size. A newer filter-rating system called the beta ratio is replacing the old nominal and absolute designations.

The beta ratio indicates what size particles the filter removes, followed by the ratio of the number of this size particle in the fluid upstream from the filter, divided by number of particles that size in the fluid downstream from the filter. For example: a filter rating of beta 5 = 90 indicates the filter will remove 90 of every 100 particles of 5 micron or larger size from the fluid passing through it. The efficiency of this filter would be 98.9% -- or 100 - (100/90).

Most hydraulic filters employ a closely controlled paper fiber mat or a woven wire mesh element to trap particles. While woven wire is more expensive than paper, the ability to manufacture it with more precisely sized fluid flow openings makes it a better choice. Also, woven wire elements can withstand higher pressure drops without collapsing.

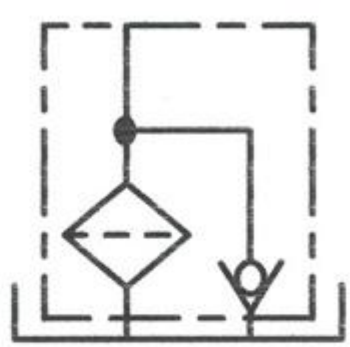


Fig. 9-1. Suction filter (or strainer) with bypass check valve.

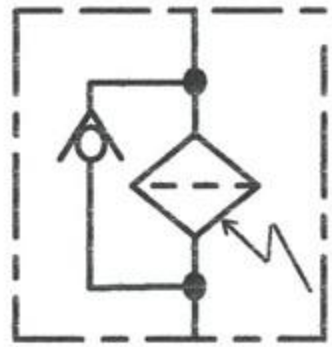


Fig. 9-2. Return-line filter with a bypass check valve (and electrical clogging indicator).

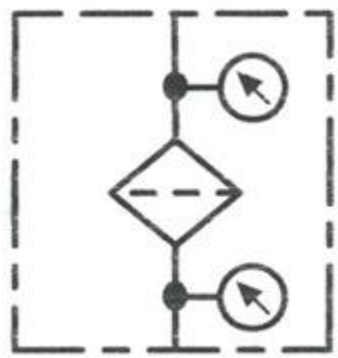


Fig. 9-3. Pressure filter (without bypass) that has pressure gauges to indicate pressure drop across the filter.

Figures 9-1, 9-2, and 9-3 show the symbols used in circuit diagrams for the common filter types. The hydraulic circuit diagrammed in Figure 9-4 has these filters in typical locations.

Suction strainers

Figure 9-4 shows a hydraulic circuit with filters in standard locations. Strainer is a common name for filters with openings of 75 microns or larger. Suction strainers usually are installed in the pump inlet line to protect the pump from large, damaging contamination particles that can cause catastrophic failure. The suction strainer also protects the pump from ingesting any start-up debris left in the tank and piping. In addition, the suction strainer traps large contamination particles introduced to the system from external sources or resulting from internal part failure.

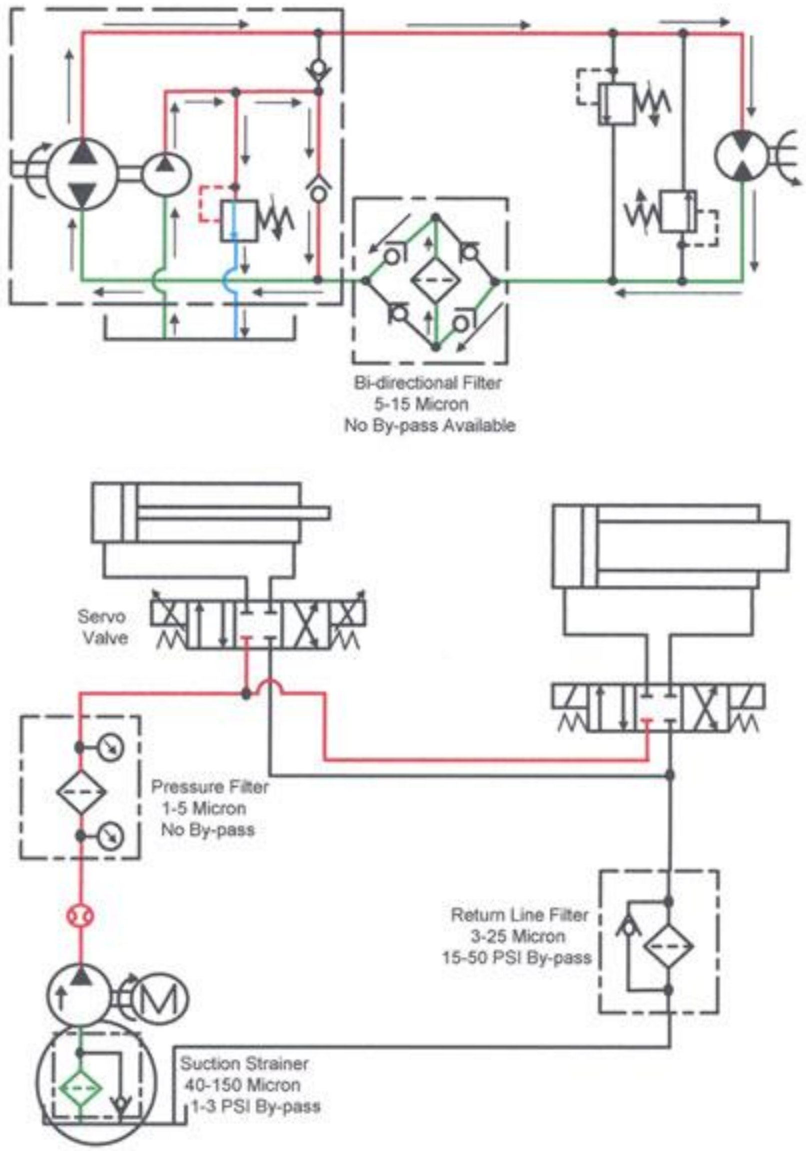


Fig. 9-4. Typical filter locations – with micron ranges and bypass settings.

Don't use filters with low-micron ratings in suction lines because pumps without supercharged inlets can only tolerate a portion of one atmosphere pressure drop without affecting inlet flow. With this low-pressure drop (14.7 psi maximum, at sea level on an average day), a restriction such as a low-micron filter can cause the pump to cavitate. Cavitation will cause pump failure almost as fast as dirty oil, so avoid it in every situation.

Suction strainers are normally available with openings ranging from 75 to 150 microns. Some manufacturers offer inlet filters with ratings as low as 25 microns. These low-micron elements have large filtering surfaces.

If the pump is force-fed by another pump (sometimes called a supercharge pump), use of a low-micron rated element is possible. The supercharging pump will force fluid through a very fine filter to the working pump without cavitation.

A suction strainer or filter should have a bypass relief valve. Set the bypass to open at a pressure of 1 to 3 psi when the strainer becomes clogged. The reasoning behind this is that the pump will run many hours on contaminated oil, but will fail in a few minutes with little or no oil.

Suction strainers can be located inside or outside the reservoir. Internal strainers are less expensive, but their condition is more difficult to monitor. External strainers are easy to service and often include an indicator to show when the filter starts bypassing. The indicator can be as simple as a vacuum gauge or it might be a vacuum-operated electrical output to a warning light or controller.

Many older circuits have nothing but a suction strainer for filtration. Retrofitting these systems with the off-line or kidney filters discussed later in this chapter is advisable.

Return-line filters

Another common location for filters is in the return line. (See the circled item in Figure 9-5.) The return-line filter keeps most contamination caused by part wear from getting into the tank. These filters are offered with ratings ranging from 3 to 25 microns. A common level of filtration is 10 microns. Obviously, if the desired system cleanliness is 10 microns, use a filter of 10 microns or less.

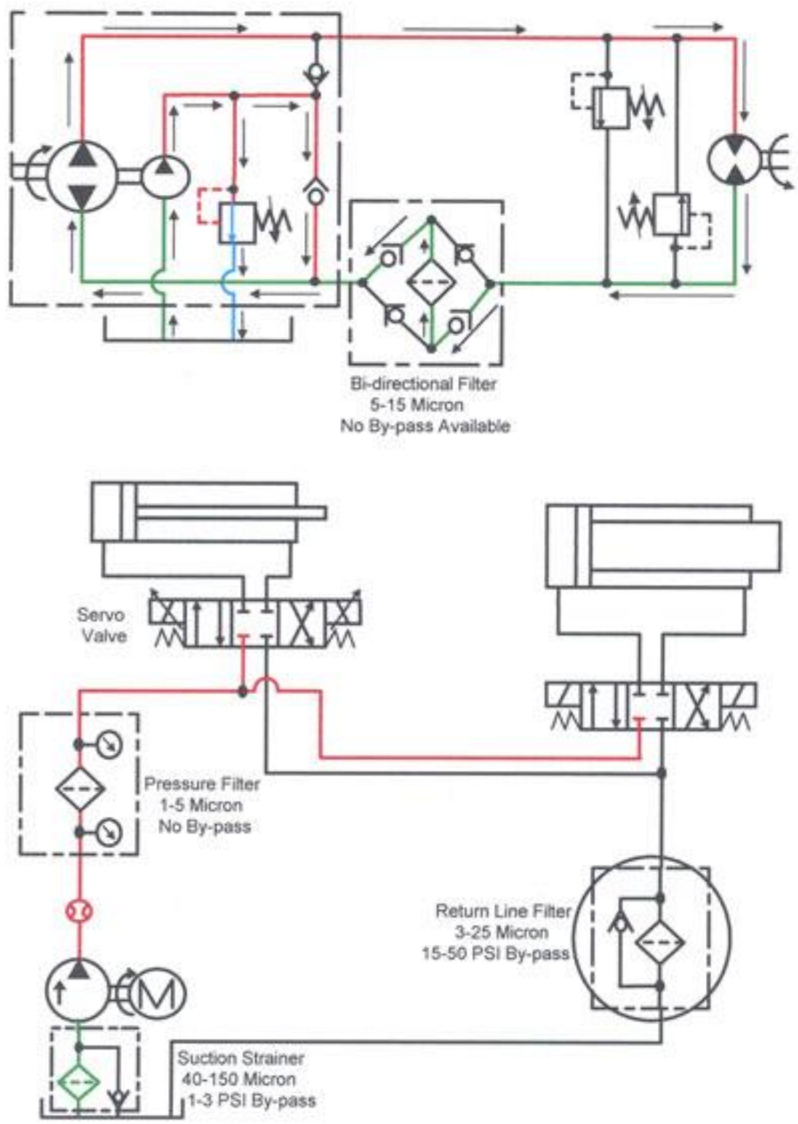


Fig. 9-5. Typical filter locations – with micron ranges and bypass settings.

Return-line filters should have integral bypass check valves. If the filter becomes loaded, return oil needs a flow path to tank until it is convenient to change the filter. Without a bypass, the filter element may collapse, or the element housing or seal may rupture. The bypass check valve usually requires 10 to 50 psi to open. The bypass pressure should be high enough to stop fluid from going around the filter except under unusual conditions, but low enough to keep the filter element and housing seal from being damaged.

Sizing return-line filters just to handle pump flow is a common practice. However, sizing the filter to pump flow can cause problems if cylinders in the circuit have oversized rods, or if one cylinder returns one or more other cylinders. For example, if a cylinder has a 2:1 rod diameter, flow to tank while the cylinder is retracting will be twice pump flow. Sizing the filter just for *pump flow will allow contaminated oil to bypass at least -- and may damage the housing or seals. Paper filters can collapse, have holes blown through the element, stop filtering, and never indicate they need to be replaced. With pleated elements, the pleats can collapse, giving a premature "loaded element" indication.*

Even with a correctly sized return-line filter, the flow through it changes constantly. A steady flow through the element gives the most efficient filtration. If a filter passes constant flow, the bypass valve will not open until the filter fills with contaminants. This means only clean fluid leaves the filter.

Visual and electrical indicators also are available to show when the return-line filter is bypassing.

Pressure line filters

Servo directional control valves normally require pressure-line filters because these valves have low contamination tolerance, as shown in the lower circle on Figure 9-6. These valves have small internal orifices, very close-tolerance fits, and must shift rapidly at low pilot-pressure differential. A servovalve can stop functioning in as little as two minutes with oil from a typical hydraulic system. Any servovalve circuit operates best with a pressure filter.

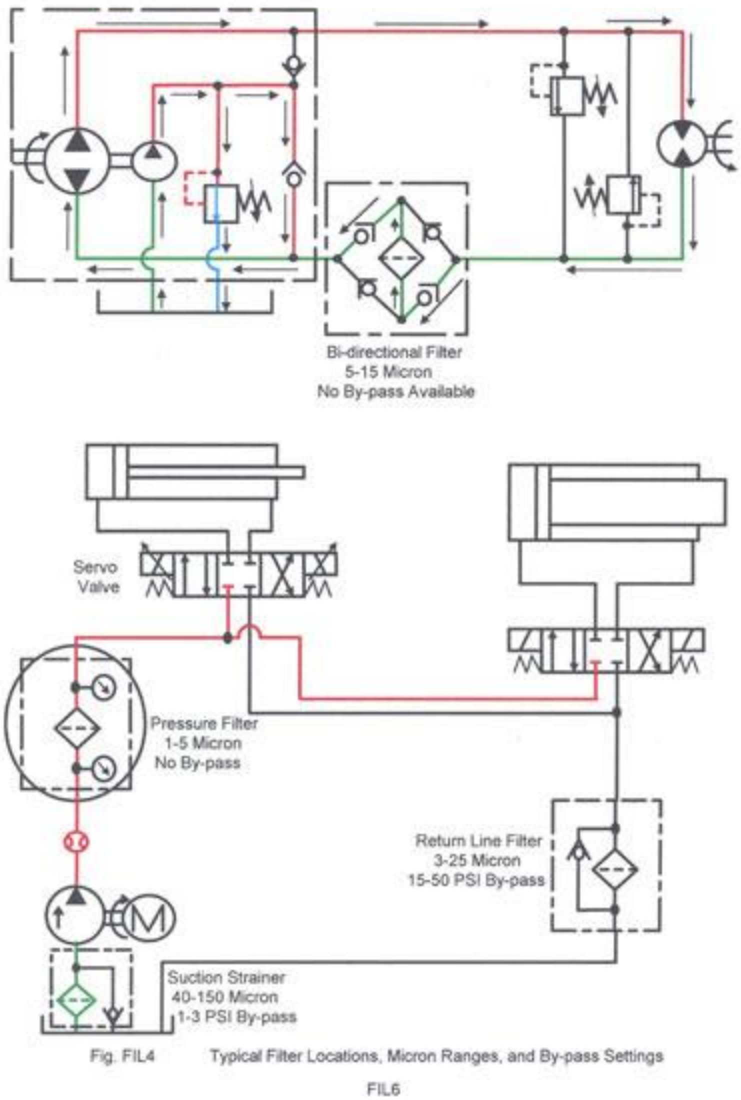


Fig. 9-6. Hydrostatic transmission circuit with bi-directional filter.

Even when a 3-micron return-line filter is installed, contamination generated by the pump is enough to shut down a servovalve in a short time. To solve this problem, place a pressure filter in the line between the pump and the servovalve. To eliminate servovalve contamination in circuits with long lines, place a pressure filter at each valve inlet.

Pressure-line filters normally have elements with 1- to 5-micron openings. The pressure filter should be of the absolute-rated type, or have a beta ratio of 50 or higher.

A pressure-line filter should not have a bypass. If the filter element clogs, it is better to stop flow to servovalves than to contaminate them. Indicators on the filters warn of clogging to allow the *elements to be changed before production speed is affected. Visual and electrical clogging indicators also are available for most pressure-line filters.*

Pressure-line filter housings must be strong enough to withstand full system pressure. When there is a high pressure drop across the filter, the element must not collapse. These requirements make pressure-line filter housings and elements more expensive than return-line filters. High cost is the main reason for not using pressure-line filters on all systems.

Figure 9-6 pictures a bi-directional pressure-line filter. Another name for this bi-directional filter is "last-chance filter." Because it is in the working line to an actuator, this filter has to withstand maximum system pressure. The only difference between a bi-directional filter and a standard pressure filter is the four check valves in the housing. The four check valves cause oil flow to pass through filter element in the same direction regardless of the direction that the fluid enters the housing. A bi-directional filter will normally have a 3- to 10-micron rating for most circuits. Pipe a bypass check externally when required.

Closed-loop hydrostatic transmission circuits are one place to use bi-directional filters. Note that the oil between the pump and motor can stay in the loop for long periods. Any contamination in this closed loop will continue to cause damage, even after changing oil in the tank.

Off-line filtration

The top image in Figure 9-7 shows an off-line filtration circuit. This is an easy circuit to retrofit to existing hydraulic systems. Also, it is an excellent circuit for new systems where high cleanliness levels are needed. Sometimes called kidney filters or bypass filters, off-line filtration systems consist of a separate pump, motor, and filter that re-circulates oil in the reservoir. Oil from one end of the tank passes through the filter and returns to the opposite end of the tank.

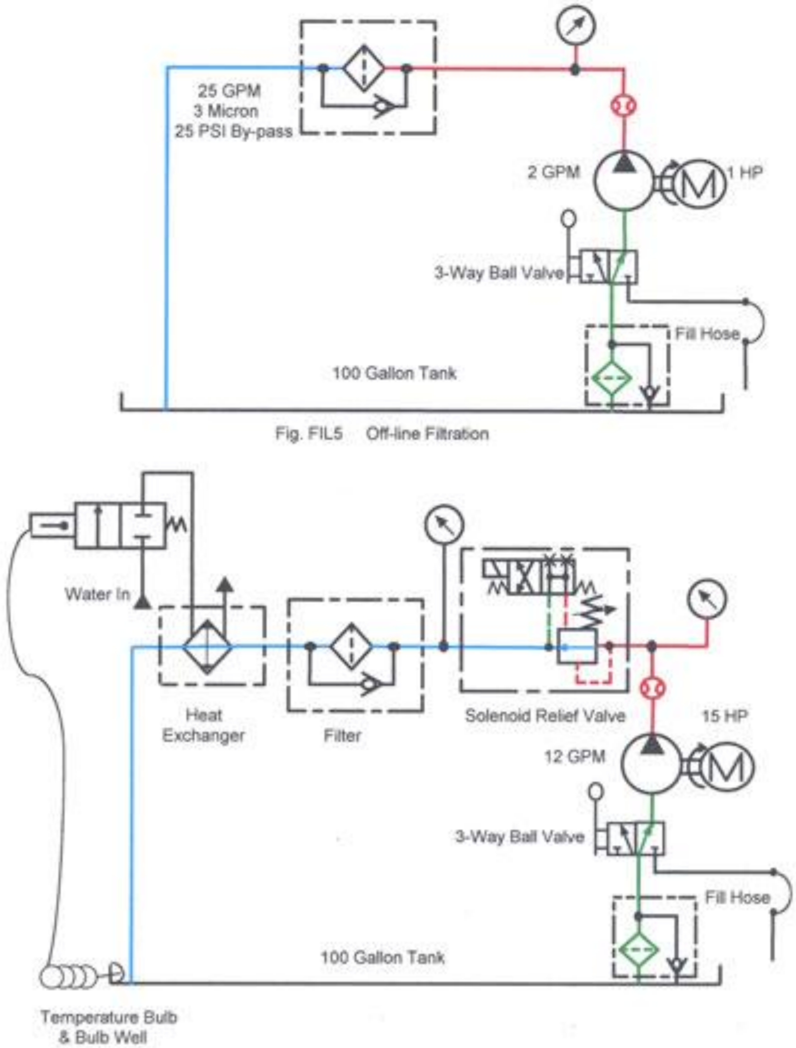


Fig. 9-7. Off-line filtration arrangement (top image) and off-line filtration circuit with heating or cooling capacity (bottom image)

The filter in the off-line circuit should be rated in the 3- to 10-micron range. The circuit should be set up to filter the volume of fluid in the reservoir every 1 to 3 hours minimum. This low, constant flow rate makes the filter very efficient, never opens the bypass, never causes channeling, and never blows holes in the element.

When the off-line filter indicator shows a clogged element, the main hydraulic circuit can continue to run during filter change. Also, this type filter system can operate while the main hydraulic circuit is shut off over nights or weekends.

Always filter new oil before use since it not as clean as most hydraulic systems require. Put new oil into the tank through a pair of shut off valves, or a 3-way ball valve in the suction line (as diagrammed in the bottom of Figure 9-7). Rotate the 3-way ball valve 180 degrees, hooking the *pump suction to a flexible hose in an oil drum or fluid container. This set-up filters all oil from the fluid container before it enters the reservoir.*

Any circuit with a servovalve still requires a pressure filter downstream from the pump. Also, according to the working conditions, a return-line filter may be helpful to take out system-generated particles before the fluid goes back to tank.

A heating or cooling loop is another function sometimes performed in an-off line filter circuit. Figure 9-8 shows a bypass circuit with a normally open solenoid relief valve, a high-horsepower motor, a temperature switch, a heat exchanger, and a temperature-controlled water valve. These additions can effectively control temperature while filtering the fluid. (To only filter the oil, leave the water turned off and the solenoid relief valve open.) If oil temperature drops too low, the temperature switch will energize the solenoid on the relief valve and pressure will rise. All electric-motor horsepower converts into heat until the temperature switch indicates the correct oil temperature. Unlike an immersion-type electric tank heater, the fluid is being circulated, so there are no hot spots. For every electric horsepower, there will be 2544 Btu/hour heating capacity. After figuring the Btu/hour to heat or maintain minimum temperature, divide by 2544 to calculate the horsepower needed.

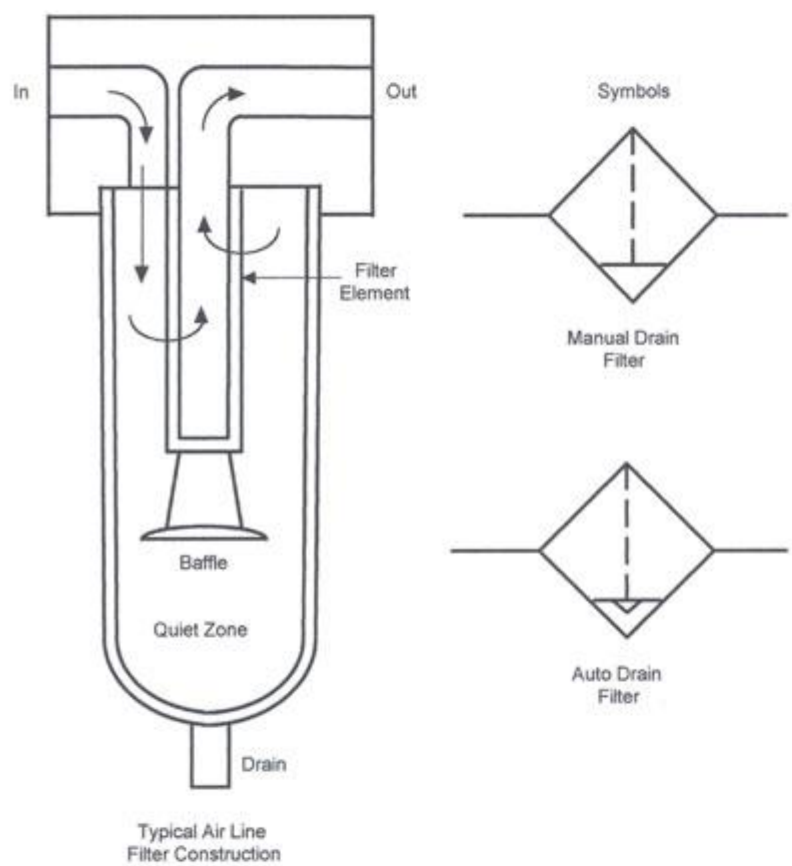


Fig. 9-8. Cross-section of typical air-line filter, left, with symbols that show drains at right.

If the tank fluid overheats, the temperature-controlled water valve will open, sending water through the heat exchanger. All filtered flow cools while the temperature valve indicates elevated temperatures. The heat exchanger always passes constant flow, so a bypass valve around it is unnecessary. Also, the heat exchanger passes flow even when a pressure-compensated pump in the circuit is holding pressure without flow.

Air-line filters

Air-line filters trap debris in air lines to protect downstream valves and cylinders. They also capture condensed water in the air stream. Most air-line filters have a manual drain to get rid of the trapped water. Several manufacturers offer an automatic drain at added cost.

The filter media in air-line filters consists of compressed fibers, ceramics, or sintered metal. A standard air filter removes particles 40 microns or larger. Most manufacturers also can supply filters with ratings as low as 5 microns when required.

Air entering the filter, Figure 9-8, flows along the walls of the bowl to swirl out condensed water – which drops to the bottom of the bowl. The air then passes through the filter media, and on to the regulator and lubricator. A baffle separates the lower part of the bowl, making a quiet zone for trapped water so it won't be picked up again.

To get even better air quality, coalescing filters are available. Coalescing filters remove up to 99.9% of oil aerosols, as well as particles down to 0.3 microns. These filters are desirable in instrument air and paint spraying applications, or any other circuit that requires very clean air. The basic design of a coalescing filter is the same as a standard filter. The main difference in is the filter element. The filter element is for one-time use and is quite expensive. Most suppliers recommend standard filters upstream to remove larger particles and liquids, thus extending the service life of the expensive coalescing element.

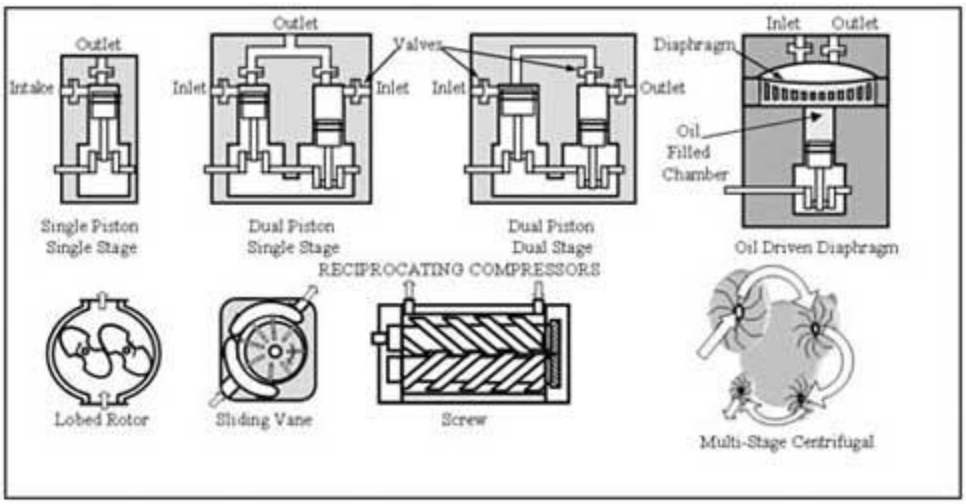
Air filters usually do not come with a bypass check valve. When the filter becomes clogged, flow restriction increases until air flow finally stops. Pressure drops on gauges at the inlet and outlet show when to change the filter element.

Fluid power pumps

A fluid power system's prime mover is a pump or compressor that converts electricity or some form of heat energy into hydraulic or pneumatic energy. These devices can be rotary or reciprocating, single or multiple stage, and fixed or variable volume. They may move a variety of fluids and come in many different designs. Some pump designs offer unique features that make them especially suitable for a particular application.

Figure 8-1 *shows several types of compressors in simplified cutaway form. These cutaways represent many standard designs used in industrial applications. They are not complete representations but simply show general working principles.*

Fig. 8-1. Several designs of rotary air compressors



Reciprocating-piston air compressors

The single-piston/single-stage, dual-piston/single-stage, and dual-piston/dual-stage compressors illustrated in ***Figure 8-1*** *are typical designs for piston-type air pumps. Compressors of these designs may be rated as low as horsepower or as high as 1000 or more horsepower. The smaller sizes are air cooled while larger ones are water cooled.*

Single-stage compressors normally operate at 125 psi or less and produce approximately 4 scfm (standard cubic feet per minute) of flow at 100 psi. (One scfm is 1 ft^3 of gas at 68°F, 14.69 psia, and a relative humidity of 36%.

Diaphragm air compressors keep lubricating fluids out of the air or gas they are compressing. This arrangement often makes the air suitable for breathing and it can be used in applications where contamination from compressor oil cannot be tolerated. The cutaway view in ***Figure 8-1*** *shows an oil-driven diaphragm compressor that is capable of very high pressure. As the oil piston extends, it forces oil against the diaphragm to compress the gas. On the retract stroke, pressure inside the diaphragm plus vacuum returns the bladder to pick up more atmospheric air.*

Piston-type reciprocating compressors below a 15- to 25-hp range usually start and stop at preset low and high pressure settings. Larger reciprocating compressors typically continue to run after pressure reaches the preset maximum, but they then stop compressing by holding their inlet valves open. This arrangement is called unloading. It saves wear on the electric motor because the motor only has to start one time.

Rotary compressors

Rotary compressors employ lobed rotors, vanes, screws, or impellers to draw in ambient air and compress it. ***Figure 8-1*** *also shows these devices. While these types of air pump are more compact and produce less vibration, they have lower efficiency than other types. All these designs (except the multi-stage centrifugal compressor) are limited to a maximum of 150 to 200 horsepower.*

Rotary compressors run continuously and are capable of no flow to full flow at any time. An inlet-restricting valve closes or opens in response to pressure changes. Many rotary compressor installations do not require a receiver tank, due to their ability to change flow in relation to demand.

Pneumatic pump efficiency

Using atmospheric air as a means to transmit energy is very inefficient. A 1-hp air motor requires between 7 and 15 compressor hp while it runs. A hydraulic motor that produces the same output would only need 1½ to 2 hp input.

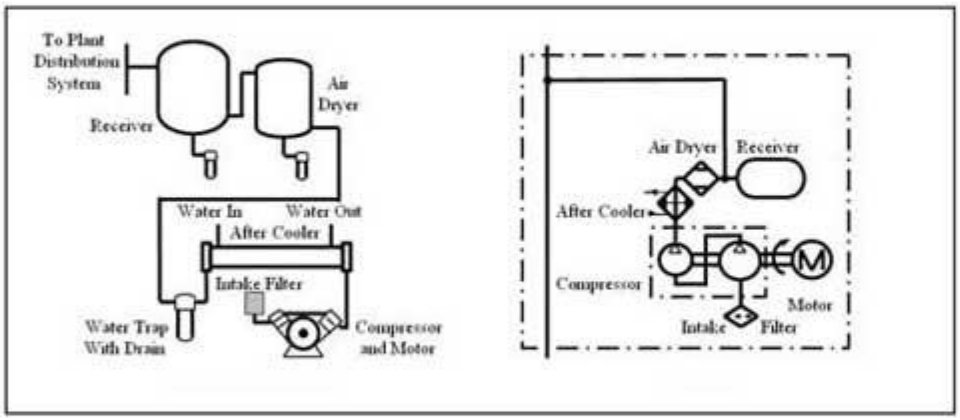
Air cylinders are more efficient than air motors, but still require three to four times more prime mover energy than their hydraulic counterparts. The general rule of thumb is: use hydraulic cylinders when an air-cylinder circuit would require a 4- or 5-in. or larger bores to produce the necessary force. This is especially important when the cylinders must operate at high cycle rates. Up-front cost of the hydraulic system is more, but operating cost savings soon pay for the added expense.

On the other hand, a 20-in. bore air cylinder used to maintain tension on a conveyor belt (with minimal cycling) would be a very efficient system.

Complete air compressor installation

Figure 8-2 *combines the schematic diagram and picture representation of a typical air compressor installation. (The compressor could be a reciprocating or rotary type.) The aftercooler may not be required on installations under 50 hp, and it could be air-cooled instead of water-cooled. An air dryer is necessary in certain applications, but is often left out due to added cost. As noted earlier, a receiver tank might be eliminated with a rotary compressor is there never is a demand for short bursts of high-volume air. Water traps with drains are required on all systems because a compressor takes in a lot of water with the ambient air. (Even with an air dryer there is always the time when the dryer needs service but the system cannot be* *shut down. A trap will help during these times.) Other components, such as isolation or bypass valves for the aftercooler and air dryer, often are part of the circuit.*

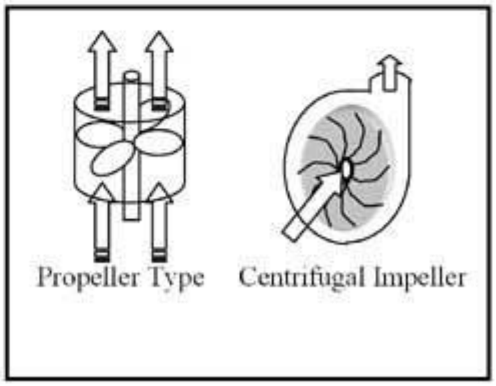
Fig. 8-2. Pictorial (at left) and schematic representations of typical air compressor installation



Hydraulic pumps

Most hydraulic pumps are positive-displacement devices. Pumps with positive sealing parts -- whether rotary or reciprocating -- move fluid every time they operate. This means that if the pump is turning, it produces flow. (Conversely; blocking flow stops the pump's rotation mechanically.) Positive-displacement pumps have higher efficiencies than their non-positive-displacement counterparts, such as impeller or centrifugal designs. ***Figure 8-3*** *illustrates some non-positive-displacement designs that could be used to run hydraulic circuits. Because these pumps only run at 50 to 75% efficiency, they are not used in high-pressure circuits. They are frequently found in systems with high-water-content fluids (HWCF), such as 95% water and 5% soluble oil, because these pumps require little or no lubrication. Also, these systems usually operate at or below 400 psi.*

Fig. 8-3. Two types of non-positive-displacement pumps



Some positive-displacement pumps are paired with centrifugal pumps to pressurize their inlets to keep them from cavitating. Or, when a positive-displacement pump is run at higher rpm than specified, the inlet may not be large enough to let in enough fluid at atmospheric pressure. In this case a non-positive-displacement pump can force fluid into the undersized inlet and eliminate cavitation.

A non-positive-displacement pump does not require a relief valve in many installations. There is enough slippage in most designs to allow for stopping flow while not over pressuring the circuit. However, if the pump operates at no flow for more than two or three minutes, simple bypass circuit to move fluid for cooling purposes should be added. The bypass circuit could be a small relief valve, a manual petcock, or a normally closed solenoid valve operated by a timer or pressure switch.

The propeller design is the least efficient of these pumps because there is a direct path from inlet to outlet through the blades. The minimum rpm of this type pump is high due to this open path. The centrifugal-impeller design operates at much closer tolerance so it slips less fluid while operating.

Fixed-displacement pumps

Fixed-displacement pumps are found most commonly in circuits with a single actuator. This allows the pump to be unloaded at little or no pressure when not performing work. A multiple-actuator circuit, where only one device moves at a time, can also be practical for fixed-displacement pumps if the actuators use about the same volume of fluid. This means total pump flow is either doing work at load pressure or is being sent to tank at very low pressure.

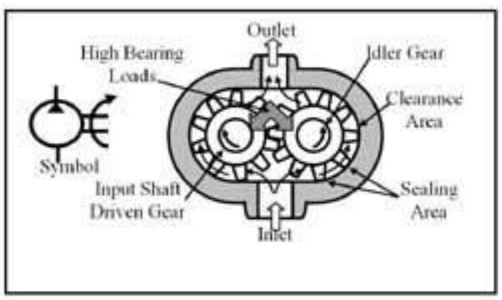
Avoid using meter-in or meter-out flow controls with fixed-volume pumps because a flow restriction increases pressure and the increase sends fluid to tank at the relief valve setting. This produces excess heat and all the problems associated with it. One way to use fixed-displacement pumps with multiple-actuator circuits is to include an accumulator with an unloading and dump valve. With this circuit, the pump is only on pressure when fluid is required. The accumulator accepts excess pump flow and provides working flow when the pump is unloaded. ***Figure 8-12*** *shows a fixed-volume pump with an accumulator.*

Fixed-displacement pumps are usually less expensive and more contamination tolerant than pressure-compensated pump. Note: this does not mean they should be run with dirty fluid or that cheaper is really less expensive. It only means they fill the bill in many applications where cost is a factor.

Gear-on-gear fixed-displacement pumps

One of the oldest hydraulic pumps is the gear-on-gear design shown in ***Figure 8-4****. As the driven gear turns, the idler gear turns in the opposite direction. At first, air trapped between the teeth and housing is moved to the outlet and forced out by the meshing teeth in the center. This starting action creates a negative pressure (vacuum) at the inlet. Atmospheric pressure then pushes oil into the pump. Now hydraulic fluid flows around the teeth and out to the circuit. Because the sealing action -- between the gear teeth and the housing, and where the teeth mesh -- has minimum clearance, when fluid is blocked, the gears stop turning.*

Fig. 8-4. Gear-on-gear positive-displacement pump



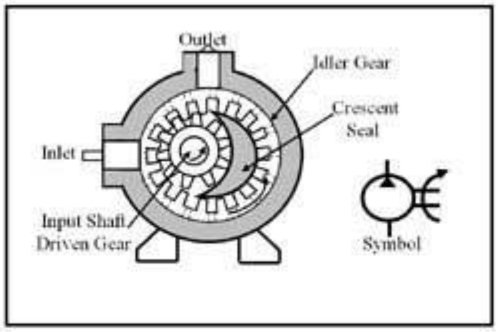
A standard gear pump is unbalanced because there is high pressure on one side and low pressure or vacuum on the other side of the gears. This causes high bearing loads and shortened service life at pressures above 1500 psi. Some newer designs reduce this unbalance by clearing the housing (or clearance area) and only having a short sealing area. This greatly reduces bearing forces so that pressures up to 4000 psi continuous are commonplace today. However, even with this new design there is no compensation for gear or housing wear.

Gear-on-gear pumps can have more than one pumping section within a common housing. This allows for different flows or pressures to some circuits for speed and force changes.

Internal-gear fixed-displacement pumps

Figure 8-5 *shows a cutaway view and the symbol for an internal-gear pump. The standard design is unbalanced and has no way to compensate for tooth or housing wear. Most pumps of this type are limited to 1000 psi or less. They are often used as transfer or supercharging pumps at low pressure due to their less efficient design. (There is a German-designed internal-gear pump that has a wear-compensating feature and a special bearing arrangement that allows it to operate continuously at up to 5000 psi and with more than 95% overall efficiency throughout its* *life.) Standard gear pumps start out at 85 to 90% efficiency when new. As the gears and housing wear, their efficiency deteriorates until they no longer supply enough fluid to maintain cycle time.*

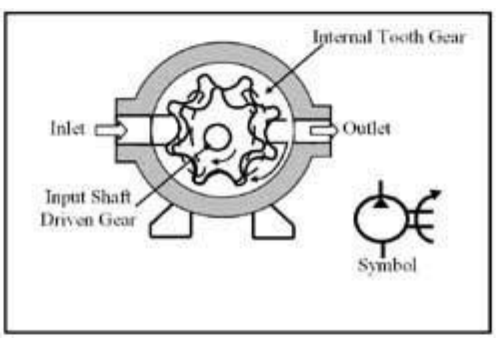
Fig. 8-5. Internal-gear positive-displacement pump



Gerotor fixed-displacement pumps

The newest design of a gear pump is called a gerotor (combining the words generated and rotor). A cutaway view and symbol is shown in ***Figure 8-6****. This pump design is not common in the marketplace. At present there are only one or two manufacturers that offer this type. On the other hand, as a fluid motor it is one of the most common designs and is offered by more than 15 different companies.*

Fig. 8-6. Gerotor-type positive-displacement pump



A gerotor pump uses a driven gear of, say, seven teeth inside an internal-tooth gear with eight teeth. The driven gear rotates inside the internal tooth gear and they both turn in the same direction. Because of the machined shapes, the driven gear always makes contact with the internal tooth gear at different points as they rotate. As the example in ***Figure 8-6*** *shows, this allows cavities to open and close as the gears turn.*

In the example, as the driven gear turns clockwise, the internal tooth gear turns the same direction, but at one tooth per revolution slower speed. This action causes cavities to form on the left hand that start reducing pressure in this area. This reduced pressure (vacuum) allows higher atmospheric pressure to push fluid into the pump and fill the forming cavities. Kidney-shaped cavities in this sector, on both sides of the teeth, accept fluid to fill them for 180° around the inlet side. As the gears continue to turn, the cavities formed on the left side start closing on the right hand side. This forces fluid through the kidney-shaped openings and to the outlet port.

Like other gear pumps, gerotor pumps are unbalanced and have no way to compensate when clearances become worn. Although a new gerotor pump starts out at 85 to 90% efficiency, it deteriorates as it runs and constantly loses volume.

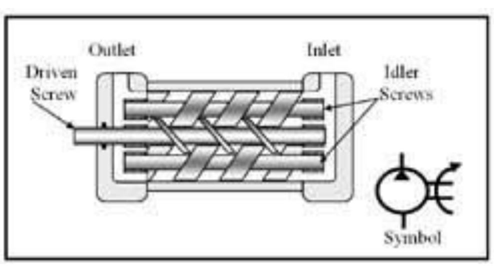
Gerotor pumps also can have more than one pumping section in a common housing, again allowing for different flows or pressures to some circuits for speed and force changes.

Another point on gear pumps: their output flow cannot be varied -- except by changing them physically or running them at a different speed. The next two types of pumps are capable of changing volume while running the same speed. These pumps can also reduce flow on a pressure build-up signal and almost eliminate the need for a relief valve.

Multi-screw fixed-displacement pumps

*The pump in **Figure 8-7** is similar to a gear pump but uses helical gears or screws to move the fluid. The driven screw is in close fit mesh with the idler screws and all gears have minimum clearance in the housing. As the driven screw turns, the idler screws also turn and the cavities between the screws move toward the outlet. This action forms a vacuum at the inlet. Atmospheric pressure then pushes fluid into the cavities and the fluid moves to the outlet. This pump has very smooth flow -- without the pulses produced by the other positive-displacement pumps in this manual. Flow from the outlet is smooth and continuous. However, screw pumps are not highly efficient. There is a lot of bypass in the original design and as the screws and housing wear, bypass increases. This design pump often is used to supercharge other pumps, as a filter pump, or a transfer pump at low pressure.*

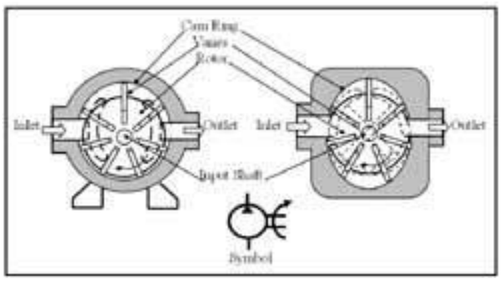
Fig. 8-7. Multiple-screw pump



Vane-type fixed-displacement pumps

*The most common pump for industrial applications is the vane design shown in **Figure 8-8**. The left-hand cutaway view illustrates the original unbalanced design. Today, most vane pumps are of the balanced design shown on the right. Balanced vane pumps operate at higher pressures and have long bearing life. All vane designs compensate for wear, so their efficiency stays in the 90 to 95% range throughout their service lives. Vane pumps are efficient, quiet, and inexpensive. They have great longevity when supplied with clean fluid.*

Fig. 8.8 Two designs of vane pumps



As a prime mover turns the rotor, centrifugal force slings the vanes outward. (Most manufacturers recommend a minimum speed of 600 rpm to make the vanes extend.) Now, as the vanes follow the off-center cam ring, a chamber is formed between the cam ring and the rotor. This chamber gets larger as the vanes extend, creating a negative pressure (vacuum) at the inlet port. Atmospheric pressure then forces fluid into these enlarging voids and fluid starts to move. As a vane passes the highest point on the cam ring, it is forced back into its slot and the chambers between the vanes decrease. As a chamber size decreases, fluid is forced out through the kidney-shaped openings to the outlet. Even though vane tips wear, they still touch the cam ring, so efficiency is not affected for a long time.

The other leakage and wear point is at the sides of the gears or rotors of these pumps. Most modern vane pumps have pressure-loaded floating plates that are hydraulically forced against the turning members. Hydraulic pressure tries to push the plates away from the gears or rotors in a certain area, but a slightly larger area on the opposite side of the plates pushes back under the same pressure. This keeps the side areas sealed without applying excess force against the turning members. (Some inexpensive low-pressure pumps may not have floating side plates but depend instead on manufacturing tolerances to control leakage.)

*Note that the unbalanced vane pump in **Figure 8-8** has pressure on one side of the rotor and vacuum on the other side. This pump has to have large bearings or operate at lower pressures. The balanced-design pump pictured on the right has pressure on opposite sides of the rotor. As a result, the bearing load is the same at 0 psi, 2000 psi, or any pressure at which the pump runs. The balanced design also produces twice the flow for the same overall package size.*

Vane pumps are available with two or three pumps in one housing to give more flow or different rates of flow to satisfy the needs of some circuit designs. These pumps have a common inlet and separate outlets as required.

Typical circuits for fixed-volume pumps

Figure 8-9 *shows a circuit using a fixed-volume pump in a simple, single-cylinder circuit. A tandem-center directional control valve routes all pump flow to tank at low pressure when the cylinder is idle. When the cylinder cycles, pressure never goes higher than necessary to do the work at hand, so energy waste is minimal. With an efficient pump, this circuit operates all day without a heat exchanger and fluid temperature never increases more than 10° or 15°F above ambient.*

Fig. 8-9. Schematic diagram of open-center circuit with fixed-volume pump supplying single cylinder

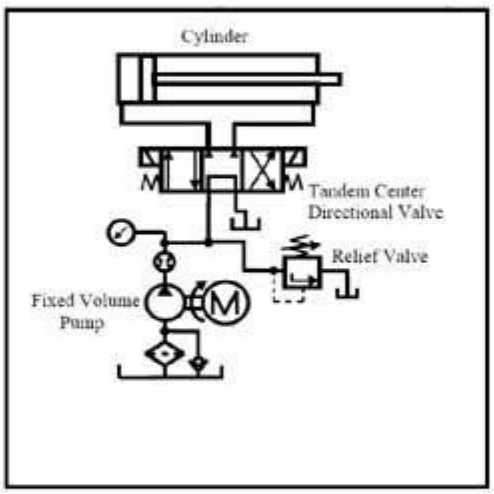
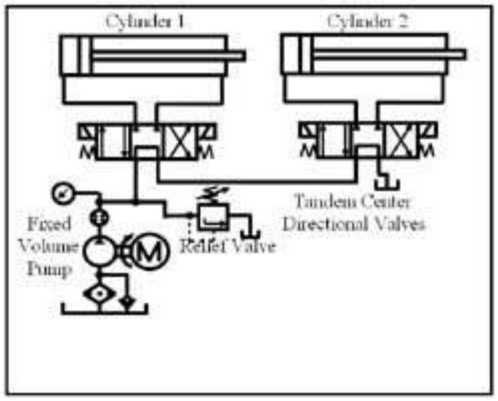


Figure 8-10 *shows a multiple-cylinder circuit supplied by a fixed-volume pump. Here, the tandem-center valves are connected in series, so all pump flow can go to tank when the actuators are idle. This circuit works best when the actuators do not move simultaneously. When two or more actuators move at the same time, the pressure to make the cylinders move is additive and may exceed the relief valve setting. Also, downstream actuators only get fluid from the actuators upstream from them. As a result, stroke lengths may be limited.*

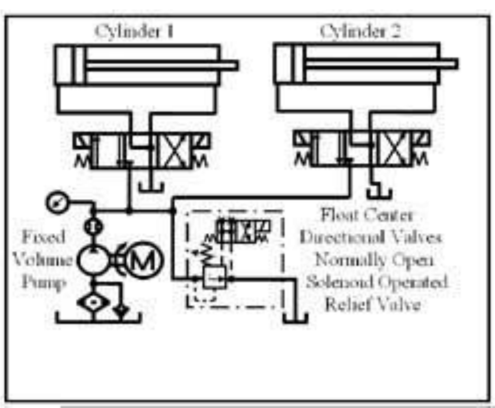
Fig. 8-10. Schematic diagram of open-center circuit with fixed-volume pump supplying multiple cylinders



Use caution when selecting directional valves for this circuit. Pay particular attention to pressure-drop charts because pressure drop is additive for each valve. This circuit could start up with a 200-psi drop at idle. With more valves in series, pressure drop at idle and running can cause sluggish operation and generate heat. Also, choose valves that are able to operate at tank line pressure. Every upstream valve sees pressure at pump and tank ports while a downstream actuator is working.

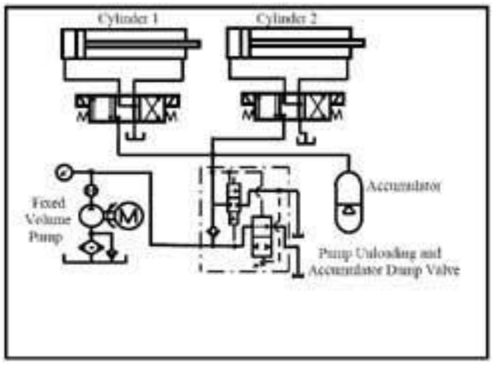
***Figure 8-11** shows a multiple-cylinder circuit that uses a normally open solenoid-operated relief valve to unload the pump when the actuators are idle. Anytime an actuator cycles, a solenoid on its directional control valve and the solenoid on the normally open solenoid-operated relief must be energized at the same time. This circuit often requires flow controls -- and may need a heat exchanger to get rid of wasted energy.*

Fig. 8-11. Schematic diagram of closed-center circuit with relief valve and fixed-volume pump supplying multiple cylinders



*The circuit in **Figure 8-12** has a fixed-volume pump with an accumulator to store energy and allow the pump to unload when no fluid is required to do work. It is similar to a pressure-compensated pump circuit because there is only pump flow at pressure when the circuit calls for* *it. The pump-unloading-and-accumulator-dump valve sends pump flow to the circuit until pressure reaches its set level. After reaching set pressure, the valve opens fully and dumps all pump flow to tank at minimum pressure. When circuit pressure drops about 10 to 15%, this valve closes and again directs pump flow to the circuit. (A normally open solenoid-operated relief valve controlled by a pressure switch could be used in place of the pump-unloading-and-accumulator-dump valve.)*

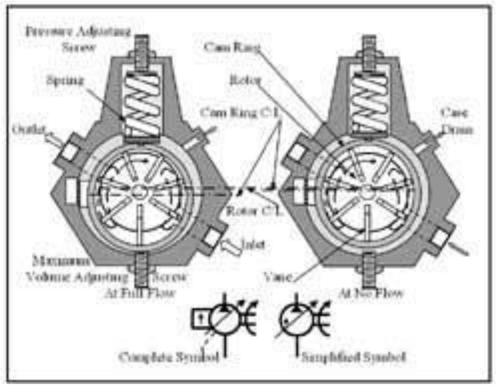
Fig. 8-12. Schematic diagram of closed-center circuit with pump-unloading and accumulator-dump valve, and fixed-volume pump supplying multiple cylinders



Pressure-compensated, variable-volume vane pumps

Figure 8-13 *shows cutaway views and symbols for a pressure-compensated vane pump. Vane pumps are one type of fixed-volume pump that can be made to function as variable volume and/or pressure compensated. The pumping action is the same as the fixed-volume, unbalanced vane pump previously discussed. The difference is that the cam ring is not fixed but can move in relation to the rotor. An adjustable force spring holds the cam ring in its offset position until enough pressure builds inside it to push against the spring and drive it toward center. As the cam ring moves closer to center, output flow decreases until it finally stops. The cam ring never makes it all the way to center because some flow is always needed to make up for internal bypass.*

Fig. 8-13. Cross-sectional views of vane pump at full flow and at no flow



Internal leakage in fixed-volume pumps passes into the case and back into the inlet flow. Internal leakage in variable-volume pumps also passes into the case but has no passageway to return to the inlet line. All internal leakage must be drained from the case directly to tank through a full-flow drain line. This case-drain line should exit from the highest point on the pump so the case stays full of fluid at all times. Always fill the case of a newly installed pump to make sure it has lubrication at startup. Also, make sure the case-drain line terminates below fluid level in the tank so it cannot suck air.

Some pressure-compensated pumps have a maximum-volume adjusting screw to prevent the cam ring from going to full stroke. This feature makes it possible to adjust the maximum flow when pressure is below the compensator setting. The feature could be used to limit maximum horsepower when only a small portion of a higher flow pump is required. (In most circuits this feature has no use because flow is usually controlled by flow controls or actuator size.)

Two symbols can indicate pressure-compensated pumps schematically. The complete symbol on the left shows all the functions, while the simplified symbol on the right omits the case drain and shows the compensating arrow inside the pump circle. Because most schematic drawings now are done on CAD systems that automatically produce the complete symbol, the simplified symbol seldom appears today.

Pressure-compensated pumps normally do not need a relief valve to protect the system from over pressure. However, many circuits with pressure-compensated pumps use a relief valve just in case the pump hangs on flow. When a relief valve, for whatever reason, is used with a pressure-compensated pump, it is imperative that it be set 100 to 150 psi higher than the pump compensator. If the relief valve is set lower than the compensator, the circuit will operate as a fixed-volume setup and quickly overheat the fluid. If the relief valve is set at the same pressure as the compensator, it is possible that the relief valve will start to dump at the same time the compensator starts to reduce flow. Then the pressure drop lets the relief valve shut and the compensator asks for more flow. These oscillations can continue until the pump fatigues and fails.

Setting the relief valve and compensator is a four-step operation.

1. Set the relief valve at maximum pressure.
2. Set the pump compensator at a pressure that is 200 to 300 psi higher than final system pressure.
3. Set the relief valve 100 to 150 higher than the final compensator setting.
4. Set the pump compensator at system pressure.

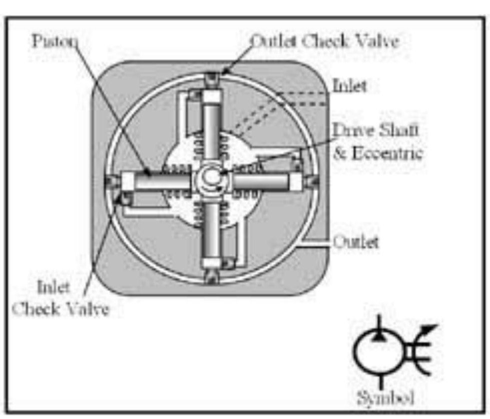
The other reason often stated for using a relief valve in a pressure-compensated pump circuit is because of pressure spikes. When a pressure-compensated pump has to instantaneously shift from full flow to no flow, fluid leaving the pump while it is shifting to center has no place to go. Because pressure is resistance to flow and resistance is a maximum at this point, pressure can climb very high. These full-flow to no-flow spikes can easily go as high as five to seven times the pump compensator setting (depending on the pump volume). Adding a relief valve to this scenario can reduce the spikes because a relief valve will respond much faster than a pressure-compensated pump. However, a pilot-operated relief valve still has some response time and will often spike two to three times its setting before opening fully.

A better way to protect the pump and circuit is to install a small accumulator at the pump outlet and pre-charge it to approximately 80% of set pressure. Now when the pump must react rapidly, the accumulator provides a place for excess fluid to go. An accumulator also helps actuator response time at cycle start because there is a ready supply of fluid even though the pump is at no flow.

Piston-type, fixed-displacement pumps

There are two types of piston pumps in use today. The oldest design is the radial-piston type. Radial-piston pumps come in two different configurations. The one shown in ***Figure 8-14*** *is sometime called a check valve or eccentric pump. The design in* ***Figure 8-15*** *is what usually comes to mind when radial pumps are mentioned.*

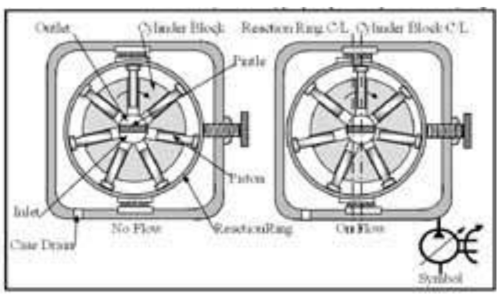
Fig. 8-14. Cross-sectional view of radial-piston pump (check valve or eccentric type)



*The cutaway in **Figure 8-14** shows how the pistons move fluid when the eccentric turns and strokes them forward, while springs return them. Check valves at the piston ends allow flow from the inlet chamber and exit flow to the outlet port.*

Many of these type pumps are capable of very high pressures -- up to and exceeding 10,000 psi. At the same time they usually flow low volume -- below 6 gpm. They are highly efficient pumps, with unidirectional flow. In fact cw or ccw shaft rotation produces the same flow rate and direction. (An eccentric pump can be made pressure compensated and/or variable volume by restricting inlet flow or pressurizing the area under the pistons to keep the springs from fully extending them.)

Fig. 8-15. Two cross-sectional views of variable-displacement radial-piston pump



Variable-displacement radial-piston pumps

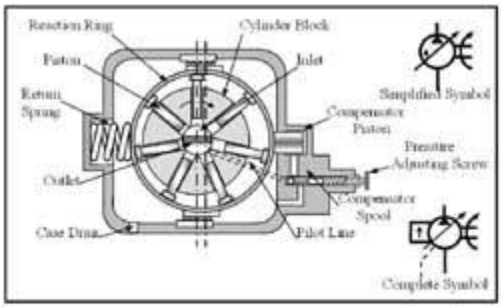
***Figure 8-15** shows a cutaway view of a basic radial-piston pump that can function as fixed volume, variable volume, pressure compensated, and bidirectional flow, or a combination of these functions. The pump in **Figure 8-15** is variable volume only. As a fixed-volume pump it would have the reaction ring offset as shown in the right hand cutaway view, with no method of changing that condition. (This is one configuration that will probably never be used with this design pump.)*

As the cylinder block and pistons rotate, centrifugal force pushes the pistons against the reaction ring. When the pump is in the on-flow condition (as in the right-hand cutaway view), the pistons are moving out of their bores in the lower half of the picture and forming a vacuum. Fluid is forced into the inlet and fills these voids. As the pistons pass left center, they stop extending and begin to be pushed back into their bores. During the top half of their travel, the pistons force the trapped fluid through the outlet to the circuit. Moving the reaction ring's centerline closer to the cylinder block's centerline reduces flow

Pressure-compensated, radial-piston pumps

*The radial-piston pump in **Figure 8-16** is pressure compensated. This pump produces flow when the outlet pressure falls below the level set by the pressure-adjusting screw. When pressure in the pilot line increases enough to compress the compensator spool's spring, pilot flow is connected* *to the compensator piston, and its drain to the case is blocked. Pilot flow to the compensator piston forces the reaction ring to move against the return spring and reduce outlet flow. The reaction ring never reaches center because the circuit, pilot control, and internal leakage must be overcome to hold pressure.*

Fig. 8-16. Cross-sectional view of pressure-compensated radial-piston pump, with symbols



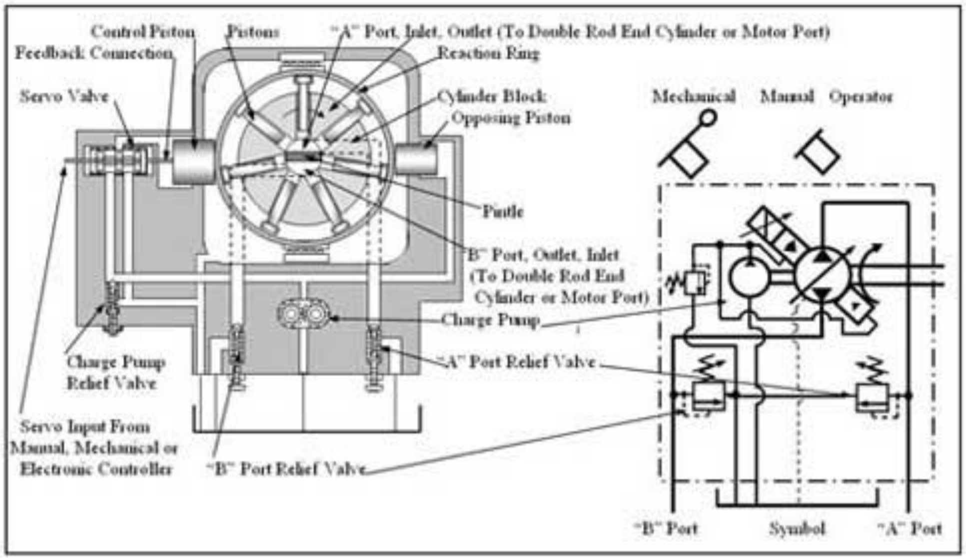
*Two symbols can be used to show pressure-compensated pumps schematically. The complete symbol at the lower right of **Figure 8-16** shows all the functions, while the simplified symbol above it omits the case drain and places the compensating arrow inside the pump circle. Again, because most schematic drawings are done on CAD systems now, the simplified symbol is seldom used.*

A radial-piston pump can also produce bi-directional flow. It can take in or force out fluid from either port while turning the same direction. This design pump is used in closed-loop circuits where all outlet flow goes to an actuator and return flow from the actuator goes back to the pump inlet. A common circuit of this type is a hydrostatic drive. Fluid from a bi-directional pump goes to a bi-directional motor to give infinitely variable output speed and force in either direction of rotation without requiring a directional control valve.

Bi-directional, radial-piston pumps

*The pump in **Figure 8-17** has a small opposing piston that pushes continuously against a larger control piston on the opposite side of the reaction ring. The control piston can be pressurized or exhausted by a 3-way servovalve, thus infinitely varying the reaction ring position to either side of center. Input signals to the servovalve can come from manual, mechanical, or electronic controllers. A common circuit produces four manually variable flows and directions, using four single-solenoid directional control valves.*

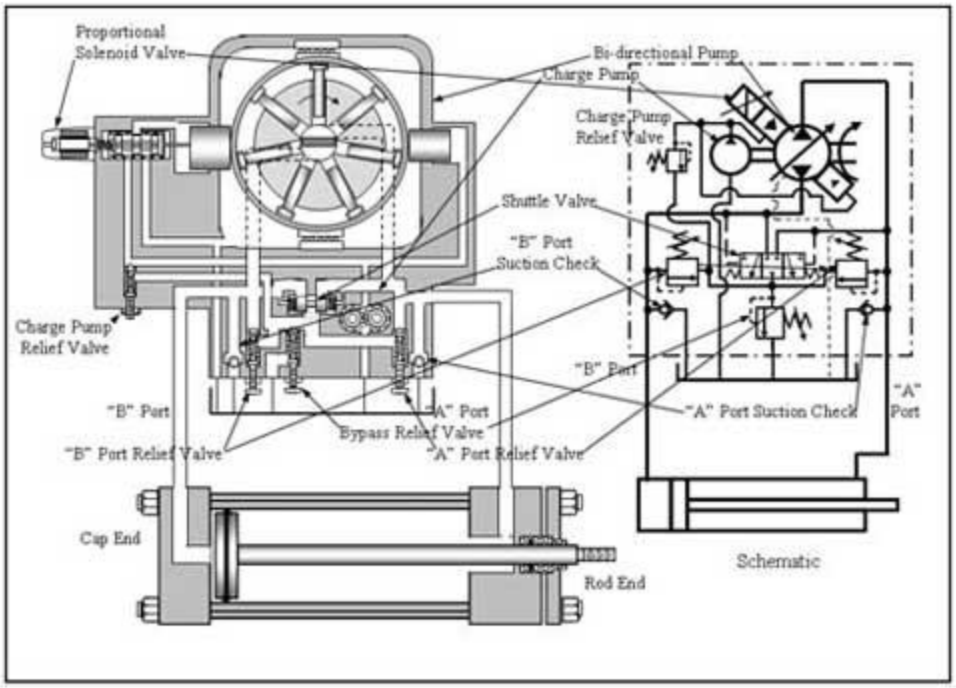
Fig. 8-17. Radial-piston pump used in bi-directional flow circuit



A charge pump, driven off the main pump shaft, supplies pilot oil to maintain pressure on the opposing piston. It also supplies oil to the mechanical-feedback servovalve that pressurizes or exhausts the control piston. The charge and pilot circuits usually run at 250 to 400 psi. Notice that the "A" and "B" ports are only connected to the actuator -- not to tank -- when using a hydraulic motor or double rod-end cylinder. (The pump must have added tank ports to operate a single rod-end cylinder circuit.)

***Figure 8-18** shows a cutaway view and schematic drawing of a bi-directional pump driving a single rod-end cylinder. Because there is less volume in the rod end of a single rod-end cylinder, flow to and from that end is less in relation to the cap end. This poses a problem when using a closed-loop circuit.*

Fig. 8-18. Cross-sectional view and schematic diagram of closed-loop circuit with bi-directional pump supplying single rod-end cylinder

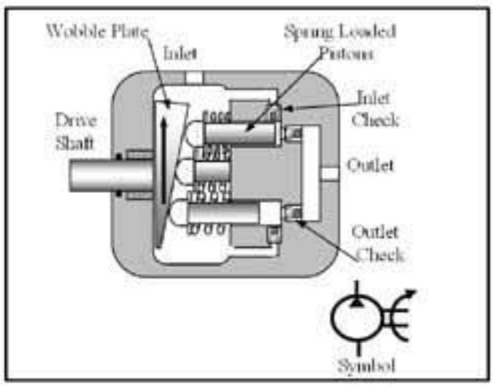


The pump cutaway and schematic show how adding suction check valves, a shuttle valve, and a bypass relief valve allow the pump to bypass excess flow from the cap end and take in added flow for the rod end. This is a common circuit for this type pump. With this circuit, cylinder speed is infinitely variable and direction change requires no directional control valve. Direction change is very smooth because flow must go to zero in one direction before it can reverse. Because of this, the actuator rapidly and smoothly decelerates to a stop condition. When flow reverses, it increases steadily to full flow in the opposite direction without system shock.

Wobble-plate piston pump

*The wobble-plate piston pump design shown in **Figure 8-19** is one type of inline or axial-piston pump. As the wobble plate turns, the spring-loaded pistons reciprocate -- drawing in fluid as they spring return and discharging it as they are forced to extend. Direction of rotation is not important for this pump because flow is the same when it turns either way.*

Fig. 8-19. Cross-sectional view of wobble-plate pump

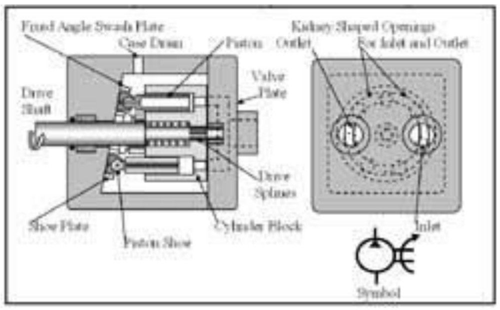


Many pumps of this design operate at very high pressure and can flow high volume as well. Another feature is the ability to isolate the outlet of one or more pistons to give more than one flow volume to a circuit. This allows a single pump to function like other double or triple pumps in hi-lo circuits or to operate different actuators at various flows and pressures. This design pump can also be made variable volume and/or pressure compensated. Some designs use a restricted inlet to accomplish both functions because the spring-loaded pistons will not fill as far if their inlet is restricted.

Inline or axial-piston, fixed-volume pumps

***Figure 8-20** shows a more common design for piston pumps. This design is seldom used as a fixed-volume pump because it can be made pressure compensated -- which many circuits require. This design can be fixed-volume, variable volume, pressure compensated and bi-directional flow, the same as the radial-piston design. The main reasons for its popularity are its compact design and its lower price. A radial-piston pump of the same flow will normally cost four to six times as much as the inline design and be three to four times larger physically.*

Fig. 8-20. Cross-section view and symbol for fixed-volume inline or axial-piston pump



*An inline piston pump like the one in **Figure 8-20** is similar in design to the wobble-plate pump. The main difference is in the way the pistons move and stroke. An inline pump uses a fixed-angle swashplate instead of a wobble plate. The pistons are not spring loaded but are held against the swashplate by piston shoes and a shoe plate. The pistons are pulled out of and pushed into their bores mechanically.*

The pistons are fitted in the cylinder block, which is splined to the drive shaft, and they turn along with the shoes and the shoe plate. As the pistons slide down the swashplate, they are pulled out of their bores and create a vacuum at the inlet port. Atmospheric pressure forces fluid to fill the piston bores until the pistons reach the bottom of the swashplate angle. Fluid enters through the kidney-shaped openings half way around one revolution. As the cylinder block continues to turn, the pistons are forced back into their bores and fluid discharges through the outlet. A kidney-shaped opening on the other half of the valve plate allows fluid to flow until the pistons are fully returned. Inline pumps always have an odd number of pistons, so one never can be directly across from another at the transition from being pulled out to being pushed in.

Inline piston pumps require a case drain to send bypass and/or control oil back to tank. The drain line should be unrestricted at all times and should terminate below the fluid level in the tank. If the drain line terminates above fluid level, the pump housing can be vacuumed dry, causing damage to the pump.

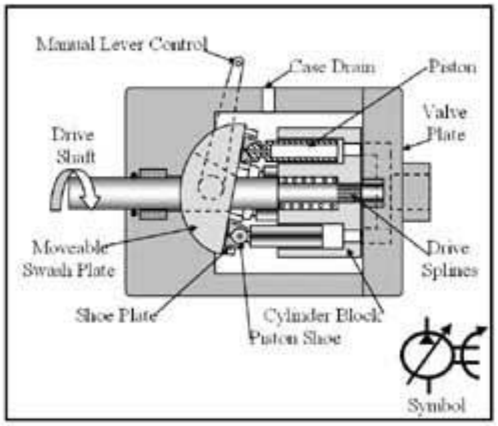
It is good practice to install a flow meter in the drain line. The flow meter indicates when to change out the pump before it loses efficiency or is worn beyond repair. A flow meter with an integral limit switch can be set to give a warning when case drain flow goes above a specified volume. Usually a pump should be changed when case flow is greater than 7 to 10% of maximum rated flow.

Inline piston pump efficiency runs in the 95 to 98% range. They, are very versatile, have many control options, and would work well on any type circuit. They are more expensive than gear and vane pumps so they lose out when price is the deciding factor.

Variable-volume inline or axial piston pumps

*Most inline pumps have some way to change the angle of the swashplate. This makes the pump capable of variable volume, pressure compensation, and bi-directional flow. **Figure 8-21** shows a variable-volume setup with a manual control. Low-flow pumps (those under 20 gpm) can use* *manual controls. Higher-flow pumps need hydraulically powered pistons to move against the higher forces in the pump.*

Fig. 8-21. Cross-section view and symbol for variable-volume inline or axial-piston pump



The basic operation of this pump is the same as a fixed-volume inline piston pump. The difference here is the angle of the swashplate can be changed manually to allow longer or shorter piston strokes for more or less volume while the pump turns at the same speed. This feature can conserve energy when an actuator needs variable speeds. It replaces a flow control that limits flow and either sends excess fluid across a relief valve or forces a pressure-compensated pump to go to high pressure and reduced flow. Other controls include manual servo, manual handwheel, and electronic servo, to name a few.

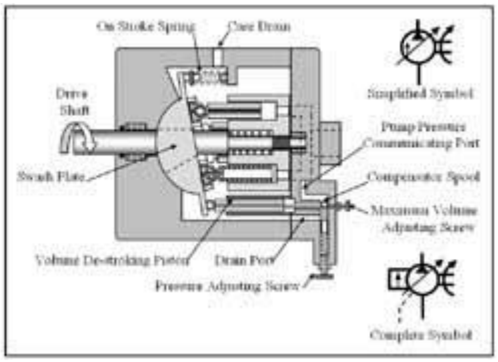
If this pump is in an open-loop circuit, make sure the control cannot go past center -- or no flow -- condition. If the lever is moved left of perpendicular, flow reverses and the pump tries to take fluid from the circuit and send it to tank. Very soon the pump will run dry and be damaged due to lack of lubrication. (Later in this text, a bi-directional pump circuit is shown with all the necessary additions to make the pump work properly in a bi-directional mode.)

*Notice that the symbol in **Figure 8-21** duplicates the standard pump symbol with a sloping arrow added to it. This indicates a pump with variable or adjustable flow.*

Pressure-compensated inline or axial piston pumps

*The pressure-compensated pump shown in **Figure 8-22** can change outlet flow when pressure tries to go above a predetermined setting. This design pump only has outlet flow when there is a pressure drop due to circuit demand. (Most manufacturers offer an option to limit maximum flow when pressure drops to add versatility to the circuit.) The maximum-volume adjusting screw keeps the volume-destroking piston from retracting all the way even when pressure drops.*

Fig. 8-22. Cross-section view and symbol for pressure-compensated *inline or axial-piston pump*



Pump operation is the same as previously explained for fixed-volume inline or axial piston pumps. The difference is that this design has a moveable swashplate that is held on stroke by the on-stroke spring. These pumps always produce full flow when pressure is below the compensator setting.

When this pump's outlet flow meets resistance, pressure builds in the pump-pressure communicating port. This pressure pushes against the spring-loaded compensator spool. In its normal position, this spool allows fluid behind the volume-destroking piston to go to tank through the case drain. When pressure is high enough to force the compensator spool against its spring, the spool allows fluid to flow into the chamber behind the volume-destroking piston while it blocks flow to tank. Enough fluid enters the chamber behind the volume-destroking piston to push the swashplate against its spring and start destroking the pump. The swashplate moves to a position to stroke the pistons just enough to makeup for the bypass and control fluid used by itself and any fluid used in the circuit. This could be any amount of flow -- even zero. Because of this, the pump never sends fluid to tank across a high-pressure relief valve, so heat generation is minimal. Pump stroke varies anytime fluid is required -- from maximum to minimum depending on circuit use.

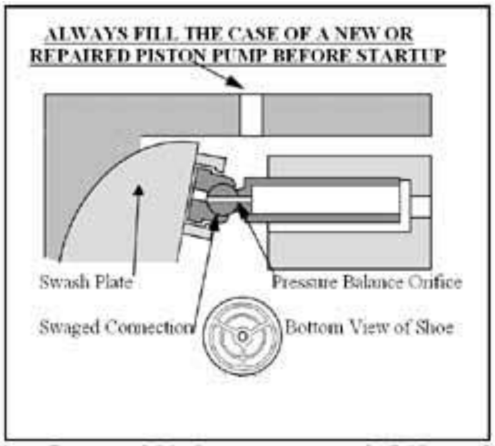
Two symbols can be used to show pressure-compensated pumps schematically. The complete symbol on the left shows all the functions, while the simplified symbol on the right omits the case drain and puts the compensating arrow inside the pump circle. Because most schematic drawings are done on CAD systems now, the simplified symbol is seldom used.

The inline pump design is subject two common problems:

1. *Operating the pumps at high vacuum inlet can quickly deteriorate the swaged connection between the piston and shoe (see **Figure 8-23**). When this joint is subjected to extra pulling and then pushing 12 to 1800 times per minute, it wears and comes apart quickly. When it does come apart, it wrecks the swashplate surface and the rest of the piston shoes. Most manufacturers recommend 1 psi or less vacuum at the inlet, and indicate longer life if the pump is supercharged by another pump at 5- to 30-psi inlet pressure.*
2. *The shoe has hollowed out areas on its face that receive oil through an orifice in the piston as it forces fluid out. This bypass oil lubricates the shoe and causes it to float a few micrometers off the swashplate. This happens because the shoe's area is greater than the*

area of the opposing piston. Because there is no metal-to-metal contact between these parts, the pump has long life expectancy. If contamination stops flow of pressurized fluid, the shoes will contact the swashplate at high force while there is minimum lubrication. The pump fails shortly thereafter. When possible, feed the pump with at least tank head pressure by mounting it alongside or under the tank. Also, keep fluid cleanliness level at least that specified by the pump manufacturer.

Fig. 8-23. Potential problem areas within inline-piston pumps

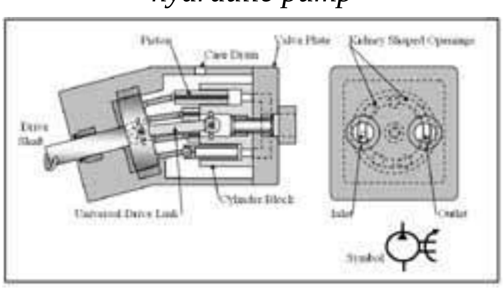


*As noted in **Figure 8-23**, always fill the case of a new or repaired piston pump with fluid before startup. The pump needs lubrication and will have very little until bypass fills the case. Also, a filled case will seal clearances and make it easier for the pump to prime.*

Fixed-volume bent-axis pumps

*Another type piston pump is the bent-axis design shown in **Figure 8-24**. Like the radial piston pumps previously discussed, this is a very expensive pump and it is physically large when all its optional features are installed. Therefore, this design pump is not a common sight in industry.*

Fig. 8-24. Cross-section view and symbol for fixed-volume bent-axis hydraulic pump



Its main advantage over an inline piston pump is that it holds up much better when the inlet sees high vacuum. The piston connections are stronger and are not prone to separating from the drive.

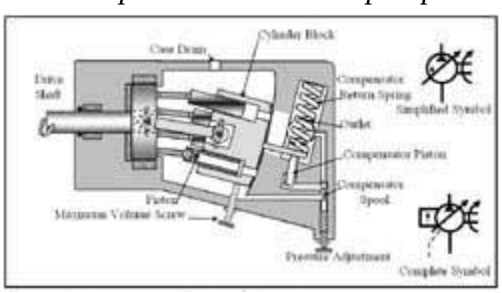
This type pump is manufactured in fixed-volume, variable-volume, and pressure-compensated models, as well as with bi-directional flow and combinations these functions. It has an efficiency range from 95 to 98% and gives long service life when supplied with clean fluid. Most manufacturers make this pump in low- to high-volume sizes. Most are capable of 4000 psi and more.

*The cutaway view of a fixed-volume bent-axis pump in **Figure 8-24** shows that as the drive shaft turns, the cylinder block also turns at the same rate through the universal drive link. Because the cylinder block is at an angle to the drive shaft, the pistons reciprocate in their bores. The pistons draw in fluid during one half of each revolution and discharge fluid during the other half. Kidney-shaped openings in the valve plate direct the fluid in and out of the piston bores. Because the housing is a single piece, the angle and volume is fixed for a given rpm.*

Variable-volume, pressure-compensated bent-axis pumps

*The cutaway view in **Figure 8-25** shows a bent-axis pump that is capable of variable volume as well as pressure compensation. This means the pump output can be varied by a manual control or it can automatically change as pressure increases to a predetermined setting. (This cutaway represents only one way such a pump might be built.) The operates in the same way as the pump in **Figure 8-24**, but the angle of the cylinder block can vary to reduce flow on a pressure demand. Also, the maximum-volume screw can limit the maximum angle of the cylinder block to establish maximum flow. This is an option on many manufacturers' designs.*

Fig. 8-25. Cross-section view and symbol for variable-volume pressure-compensated bent-axis pump

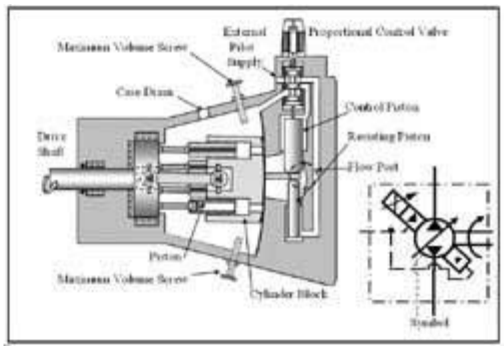


As pressure at the outlet builds to the setting of the pressure adjustment, the compensator spool is pushed back. The spool forces the compensator piston to push the cylinder block to a lesser angle. When pressure reaches the preset level, the cylinder block stays in any position required to maintain the flow needed at the preset pressure.

Two symbols can be used to show pressure-compensated pumps schematically. The complete symbol on the left shows all the functions, while the simplified symbol on the right omits the case *drain and shows the compensating arrow inside the pump circle. Because most schematic drawings are done on CAD systems now, the simplified symbol is seldom used.*

Figure 8-26 *shows the symbol and a cutaway view of a bi-directional, bent-axis pump for closed-loop circuits. This pump operates in the same manner as the previously described bent-axis pumps, but is capable of drawing in and discharging fluid from either port while turning the same direction. This design needs an external pilot supply because it has no integral pilot pump.*

Fig. 8-26. Cross-section view and symbol for bi-directional bent-axis pump

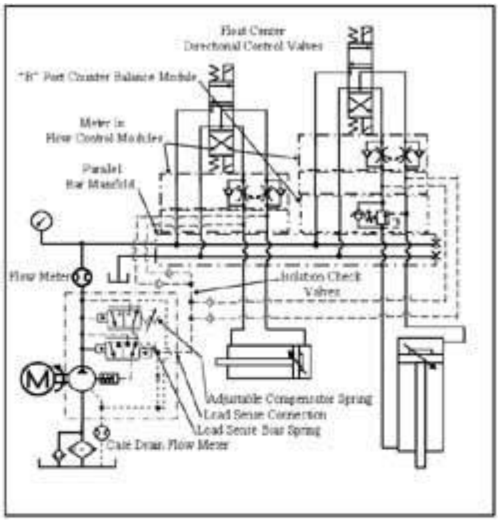


The cutaway shows optional features such as: maximum-volume screws in both flow directions and a proportional-control valve for infinitely variable flow from either port. Manual, mechanical, and solenoid controls also are available. A control piston that is offset by a resisting piston with a smaller diameter moves the cylinder block. The 3-way servovalve ports fluid to or exhausts fluid from the larger piston to position the cylinder block. This pump design is not readily available currently, but there still are many of them operating in the field.

Load-sensing function

All pumps that can be pressure compensated can also be made load sensing. Load sensing is a control technique that keeps the pump compensator from holding full pressure until an actuator stalls. Normally a pressure-compensated pump circuit operates at full compensator pressure setting unless an actuator is using all the pump flow. While an actuator is using all pump flow, pressure is whatever it takes to move the load. This is an ideal setup because all energy -- except for component inefficiencies -- is being used to do work. There is no wasted energy except for inefficiencies and very little heat is generated. A load-sensing circuit uses a feedback signal from the actuator that keeps pump pressure at 100 to 300 psi above the load. Some load-sensing pumps have a fixed differential while others are adjustable. When no actuator is moving, system pressure is at the load-sensing setting of 100 to 300 psi instead of the compensator setting. Energy savings is the main advantage of a load-sensing function, but it also makes a non-compensated flow control perform like it is pressure compensated.

Fig. 8-27. Schematic diagram of pressure-compensated closed-center *load-sensing circuit*



*The schematic drawing in **Figure 8-27** is a typical load-sensing circuit with two actuators. Notice that the sensing lines from the actuator flow lines to the pump compensator. A load-sensing pump must be able to read any load it is powering so that ample pressure can be maintained. Also notice that the load-sensing lines go through check valves to isolate the flow lines from each other.*

All flow controls in a load-sensing circuit must be meter-in type so pressure at the actuator is always high enough to move the load. In the case of the vertical cylinder, a counterbalance valve keeps it from running away while extending. Notice that the load-sensing line from the rod end of the vertical cylinder is connected between the counterbalance valve and the directional control valve so it does not see a load when the circuit is at rest.

Because the pump is pressure compensated, the directional control valve's pump port is blocked in center position. This circuit uses a bar manifold with modular meter-in flow controls and a modular "B" port counterbalance valve sandwiched under float-center directional control valves for piping convenience and leak prevention. The lines connected to the "A" and "B" ports below the meter-in flow controls go to isolation check valves, then on to the load-sensing connection.

With the circuit at rest as shown, the load-sensing connection sees little or no pressure because the actuator ports are connected to tank. At this point, circuit pressure is equal to the load-sensing bias spring, regardless of the setting of the adjustable compensator spring. At this low pressure, the circuit consumes very little horsepower and generates little heat. The pump's internal parts are subject to low stress, which makes them last longer and maintain high efficiency

When a cylinder cycles, the load-sensing connection sees whatever pressure it takes to move it. Pump outlet pressure rises to load pressure plus load-sensing bias-spring force. When both cylinders operate simultaneously, the load-sensing connection receives pressure from the highest load through the isolation check valves. Pump pressure is always that needed to move the *highest load plus a value added by the load-sensing bias spring. (Some load-sensing bias springs are adjustable within a narrow range.)*

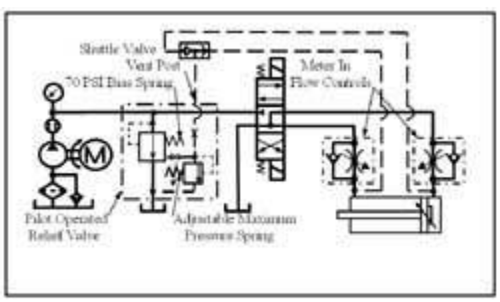
When load-sensing valves have low or no bypass flow, use shuttle valves in place of the isolation check valves. Shuttle valves will not trap backflow when a directional control valve shifts to center position. Check the chosen pump to see if this feature is standard, must be specified, or is not available.

Load-sensing, fixed-volume pumps

The oldest load-sensing circuits for fixed-volume pumps are like those diagrammed in ***Figures 8-9 and 8-10****. The pumps in these circuits never operate at a higher pressure than work resistance and never send fluid across the relief valve unless there is a malfunction in the hydraulics or control circuit.*

Figure 8-28 *diagrams a simple load-sensing pump circuit using standard valves. A pilot-operated relief valve with a 70-psi spring dumps pump flow to tank at 70 psi when the vent port is at 0 psi. A shuttle valve receives pressure feedback from the actuator and signals the pilot-operated relief valve's vent port with the actual working pressure. As the actuator moves at a reduced speed, pump pressure stays 70 psi above actual load pressure, so excess flow that goes to tank wastes less energy.*

Fig. 8-28. Schematic diagram of fixed-volume pump in closed-center load-sensing circuit



Several manufacturers offer fixed-volume pumps with integral load-sensing valves. Hookup is simple for these pumps, and in some designs, bias pressure can be adjusted.

This setup is not as efficient as a pressure-compensated pump with load sensing, but it always provides an advantage in fixed-volume pump circuits. The results are best when maximum system pressure is high and actuator's extension and retraction speeds are low.

Horsepower- and/or torque-limiting pumps

Horsepower or torque limiting is another control technique that only works on pumps that are capable of variable volume. Its main application is in the mobile-equipment field, where most hydraulic circuits are powered by gas or diesel engines. These engines usually must move the *machine as well. To maximize actuator speed and force while minimizing horsepower drain, all actuators can be fast at low loads but still able to move heavy loads without pulling excess horsepower. Each manufacturer that supplies this setup may have a different way of doing it.*

Fig. 8-29. Schematic diagram of horsepower- or torque-limiting pump control

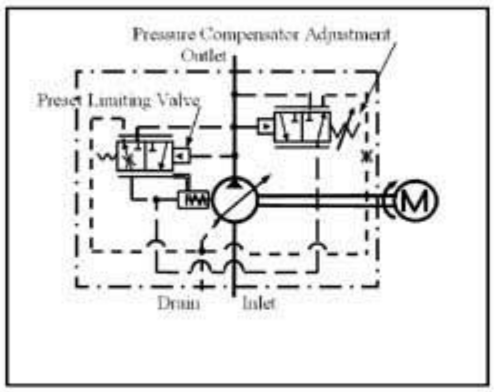
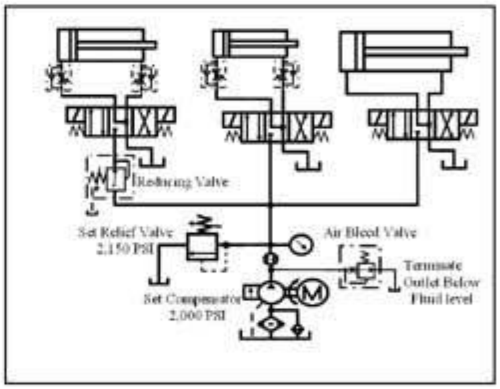


Figure 8-29 *shows the schematic diagram of a circuit with a horsepower-limiting pump. A pressure-compensator adjustment still controls maximum output pressure but the preset limiting valve can reduce flow as pressure increases. Reducing flow as pressure increases keeps horsepower or torque from exceeding a preset limit. The horsepower/torque limiter is preset for a given pressure and flow. This system could be useful in an accumulator circuit to allow higher flow as pressure decreases while limiting horsepower draw as pressure climbs.*

Typical circuit for pressure-compensated pumps

Most pressure-compensated pumps use a closed-center circuit such as the one in ***Figure 8-30****. These circuits could have load sensing or other controls. They usually include multiple actuators. Closed-center circuits typically operate at maximum system pressure and output flow matches the circuit requirement. Flow controls keep actuators at operating speed because maximum flow may make them move too rapidly. Flow controls also make it possible for more than one actuator to move simultaneously without affecting their stroke times. Note that flow controls also increase heat generation because the moving or work force may not require full system pressure. Also, some actuators may require pressure-reducing valves to lower the maximum force so that it doesn't cause damage.*

Fig. 8-30. Schematic diagram of typical pressure-compensated pump circuit



Normally, pressure-compensated pumps do not need relief valves to protect their systems from overpressure. However, many circuits with pressure-compensated pumps include a relief valve just in case the pump hangs on flow. When a relief valve, for whatever reason, is used on a pressure-compensated pump, it is imperative that the relief valve is set 100 to 150 psi higher than the pump compensator. If the relief valve is set lower than the compensator, the circuit will operate as a fixed-volume setup and quickly overheat the fluid. If the relief valve is set at the same pressure as the compensator, the relief valve can start to dump as the compensator starts to reduce flow. Then pressure drop lets the relief valve shut and the compensator ask for more flow. This oscillating action can continue until the pump fatigues and fails.

Setting the relief valve and compensator is a four-step operation:

1. *Set the relief valve at maximum pressure.*
2. *Set the pump compensator at a pressure 200 to 300 psi higher than the final relief valve pressure.*
3. *Set the relief valve 100 to 150 psi higher than the final compensator setting.*
4. *Set the pump compensator at system pressure.*

Another reason often stated for using a relief valve in a pressure-compensated pump circuit is because of pressure spikes. When a pressure-compensated pump has to instantaneously shift from full flow to no flow, fluid leaving the pump while it is shifting to center has no place to go. Because pressure is resistance to flow and resistance is maximum at this point, pressure can climb very high. These full-flow-to-no-flow spikes can easily go up to five to seven times the pump compensator setting, depending on the pump volume.

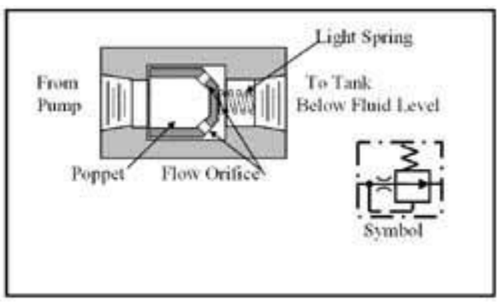
Adding a relief valve to this scenario can reduce the spikes because a relief valve will respond much faster than a pressure-compensated pump. However, a pilot-operated relief valve still has some response time and will often spike two to three times its setting before opening fully.

A better way to protect the pump and circuit is to install a small accumulator at the pump outlet, pre-charged to approximately 80% of set pressure. Now, when the pump must react quickly, the excess fluid can go into the accumulator with very little pressure spike. An accumulator also helps actuator response time at cycle start because there is a ready supply of fluid even though the pump is at no flow.

Another consideration is pump priming when a pressure-compensated pump is mounted above fluid level. When a system first starts (and sometimes when it has not been operated recently), the inlet line holds no oil above tank level. Atmospheric air in this line above fluid level must be evacuated before atmospheric pressure can push fluid in. Because most pressure-compensated pumps operate against a closed-center circuit, there is no place for this trapped air to go. Hydraulic pumps may easily move 100 gpm of fluid at 3000 psi, but they are very poor air movers. At startup, the pump never primes and could be damaged from lack of lubrication -- especially if the case has not been filled. Usually the outlet line is opened at a union or some other fitting and the pump primes as soon as the trapped air can leave.

*A better approach is to install the air-bleed valve shown in **Figure 8-31**. This valve is not required in most cases when the pump is along side or below the tank because filling the tank should also fill the inlet line. It is also seldom required with a fixed-volume pump in an open-center circuit because the pump outlet has a direct path to tank. However, when priming is a problem and there are no inlet line leaks or restrictions, then the air-bleeds valve may be required. The circuit in **Figure 8-30** shows the correct location and piping for this valve.*

Fig. 8-31. Cross-section view and symbol for typical air-bleed valve



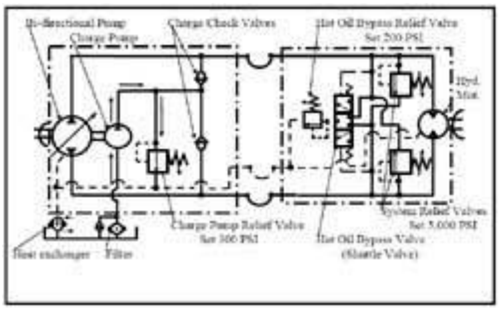
*The cutaway view in **Figure 8-31** shows the internal configuration of a typical air-bleed valve. The poppet in this valve normally is held open by a light spring, so trapped air can flow easily through its flow orifices to tank. When the pump primes and oil tries to flow through these orifices, pressure builds and the poppet closes. The poppet stays closed as long as the pump is running.*

Always pipe the air bleed valve as close to the pump outlet as possible. Any oil in the line must be pushed out before trapped air can be exhausted, so the closer the better. Always terminate the air bleed valve's outlet below fluid level. If it terminates above fluid level, air can pass through the valve and let oil in the pump return to tank.

Closed-loop circuits

*The circuit in **Figure 8-32** is a typical hydrostatic-transmission setup. It uses a variable volume, bi-directional pump to drive a hydraulic motor at infinitely variable speed. Hydrostatic drives are normally used to drive vehicles but can be used in industrial applications where smooth acceleration, deceleration, and reversing are required. These circuits usually incorporate an inline or axial-piston pump coupled to a variety of hydraulic motors. As a closed-loop circuit, all* pump flow goes to the motor and all motor flow returns to the pump. With 100% efficient parts, the circuit could run with the same oil its whole life. In the real world however, the hydraulic motor and bi-directional pump have internal bypass so a fixed-volume charge pump is placed in the circuit to make up for leaks. The charge pump can also supply fluid to control circuits and accessory devices.

Fig. 8-32. Schematic diagram of typical hydrostatic drive circuit



The charge pump inlet draws fluid from a reservoir through a low-micron filter and sends it to the inlets of the charge check valves. When the hydraulic motor is not turning, any fluid not used by the closed loop goes through the charge-pump relief valve, then back to tank through the pump case and a heat exchanger. When the hydraulic motor is turning, all charge flow goes to the low-pressure side of the loop through one of the charge check valves, the hot-oil bypass valve, and the hot-oil bypass relief valve at a lower pressure. This action makes sure the closed loop receives cooled, filtered oil that can carry away heat and contamination. It also sends cool, clean oil through the motor and pump case to flush contamination and dissipate heat.

Small hydrostatic pumps can be controlled manually, hydraulically, or electro hydraulically. Larger systems cannot be controlled manually, due to the high force required to move the swashplate.

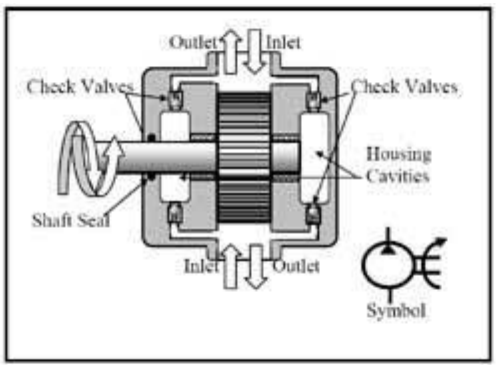
The system-relief valves protect the hydraulic motor and bi-directional pump from excess pressure when the motor is powered. When the pump center's motor outlet flow is blocked, the motor may be driven by external forces and cannot stop immediately. At this time, the system-relief valves allow fluid from the motor -- now acting like a pump -- to bypass at high pressure to the opposite motor port. This allows the motor to stop smoothly even when an operator tries to stop it abruptly. (Other options to protect the circuit from bypassing through the system-relief valves during deceleration are available from most suppliers.)

Bi-rotational pumps

Unirotational pumps can only move fluid when rotating in one direction. These pumps usually have a larger inlet port in relation to the outlet port size. They are limited as to inlet-outlet function because internal bypass is always ported to the housing on the inlet side. This means all internal bypass goes to the case and then back to the pump inlet. Because all pumps have a shaft sticking out of the housing, there must be a seal to stop fluid leak when the pump is at rest and *vacuum leak when it is running. A unirotational pump could move fluid when turning either way but the shaft seal would blow above 25- to 50-psi outlet pressure in reverse flow.*

Bi-rotational pumps can move fluid while turning in either direction of rotation if they are piped correctly. Both ports on these pumps are usually sized as inlets. ***Figure 8-33*** *shows a cutaway view of a bi-rotational pump with internal check valves that allow bypass to go to the inlet side of the pump. Bi-rotational pumps are mainly used on mobile equipment where the prime mover cannot easily change direction of rotation. This means right- and left-hand rotation pumps would have to be kept to satisfy different pieces of equipment.*

Fig. 8-33. Cross-sectional view and symbol for bi-directional pump



In industrial applications where a 3-phase motor's direction of rotation can be easily changed, pump rotation direction is not important. There is only one instance where a left-hand rotation pump must be specified. This is the case where a double-shafted electric motor drives a pump at both shafts. One of the pumps in this application must be setup for left-hand rotation.

Pump horsepower

Two formulas often used to figure hydraulic pump horsepower are:
hp = (psi)(gpm)/1714 -- (to calculate pure horsepower), and
hp = (psi)(gpm)/1714(actual pump efficiency).
Normally efficiency is assumed to be 85% because most new industrial pumps are at or above this figure.

The first formula is for a known pump volume -- figured from its cubic inches/revolution times the number of revolutions per minute. Say this displacement at 1200 rpm came to 12 gpm, but a flow meter at the pump outlet only shows 10.6 gpm at 1000 psi. The pump is still moving 12 gpm as far as its horsepower requirement is concerned, but the speed of the driven device will be that produced by 10.6 gpm. The 12-gpm pump was picked because the actual flow required was at least 10 gpm.

The second formula is applied when pump efficiency is known and horsepower is being figured for the actual 10-gpm requirement. Now pump efficiency must be considered because theoretical *flow is greater than 10 gpm, and the electric motor must be able to pump the extra fluid even though it does not get to the actuator.*

These two formulas can be simplified to:
hp = 0.000583 (gpm)(psi) -- (for pure horsepower), and
hp = 0.0007 (gpm)(psi) -- (for an 85% efficiency pump).
A common rule of thumb is: 1 gpm at 1500 psi = 1 hp.

Most suppliers' catalogs show the horsepower required to drive a given pump at different pressures. These figures are usually conservative so designers can use them with confidence. Also, most electric motors can operate continuously at 110% of nameplate rating (and up to 140% for short bursts). Remember too that the only time a fixed-volume pump will be at full flow and full pressure is when the device it is driving has stalled. A pressure-compensated pump draws the highest horsepower just before it starts reducing flow slightly below its pressure setting. That event usually is not of long duration.

Many formula-data books have horsepower charts that make picking an electric motor simple. These charts are usually based on the 85% efficiency formula.

Cavitation

Next to contamination, cavitation causes more pump damage than anything else. Cavitation occurs when a pump needs 10.8 gpm at its inlet, but only gets 10.5 gpm. The missing 3 gpm winds up as voids or vacuum bubbles that implode when they go from suction to pressure. The implosions are rapid and damaging to adjacent surfaces when outlet pressure is high. They can take a pump out of service in hours. When outlet pressure is low -- under 200 psi -- there is still some noise and damage but it is minimal.

Some mobile equipment shuts the inlet to their pumps when the equipment travels, only allowing 1or 2 % of pump flow. This small volume goes through an open-center circuit at less that 15 psi, so implosions are not a problem. Another advantage is fuel savings. Because pump flow is so low, horsepower drain is much less.

Cavitation comes from several situations that are easy to rectify:

- *Long suction lines with many turns.*
- *Undersize suction lines.*
- *The pump mounted too far above the fluid.*
- *Fluid viscosity too high (either wrong viscosity or low temperature).*
- *A collapsed suction hose.*
- *Turning the pump faster than the manufacturer recommends.*
- *A clogged inlet strainer*
- *A blocked air breather (especially in circuits with oversize rods or single-acting cylinders).*

Any of the above could be eliminated immediately with a supercharging pump. This is a separate pump operating at low pressure (usually under 30 psi), that forces fluid to the system pump inlet.

Most of these conditions also can be eliminated by good design practices:

- *Locate the pump close to the tank -- preferably alongside or under it.*
- *Never use a suction line smaller than the pump inlet port.*
- *Use the fluid recommended by the pump supplier, and install tank heaters if the system will be exposed to temperatures below 65°F.*
- *Never use pressure hose for suction lines. The lining of a pressure hose is not firmly attached to its body and can collapse under vacuum. Use hose specifically designed for suction service.*
- *When a pump must turn faster than recommended, install a supercharging pump or elevate the tank to provide head pressure. Make sure a vacuum gauge at the pump inlet never goes above 1.5 to 3 psi.*
- *Use a good filtering system – rated at least 10 µ -- so the suction strainer cannot block flow. Consider the suction strainer as insurance against startup contamination large enough to wreck a pump instantaneously.*

Another situation that occurs in suction lines is air leaks. Air leaks are not cavitation but make the same noises and damage as vacuum cavitation. The only way to tell the difference in these situations is to look at the oil in the tank. If the oil is foamy from aeration, there is an air leak in the circuit. If the fluid is clear or almost clear of bubbles, there is a vacuum cavitation problem.

Air leak problems can come from poor piping practices. It is best to never use a standard pipe union in the inlet line. It is practically impossible to seal a standard union against an air inlet leak. Plumb the inlet line with as few fittings as possible and make sure any joints are sealed. If a plumbing connection is suspect, apply some of the system fluid to each joint to see if the noise stops. This type air leak problem usually shows up at system start. It seldom happens to a running circuit.

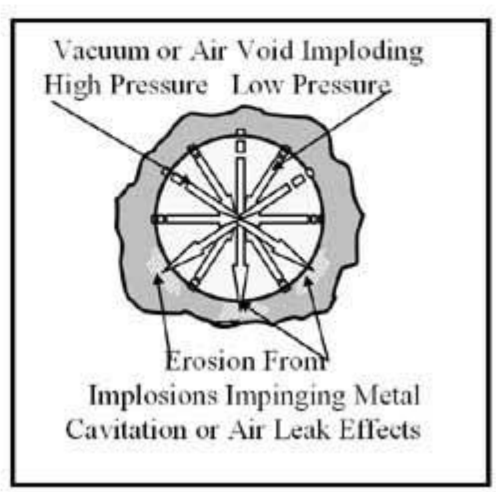
On systems that have been running for some time, a good place to look for air leaks is at the pump shaft seal. Fixed-volume pumps have their drive shaft sticking out of the housing and inside the housing is suction vacuum. When a shaft seal wears or is damaged from heat, it may let atmospheric air in before it lets oil leak out. The oil application test works here also, but can be messy because of shaft rotation speed.

The suction line is the most important line on the hydraulic circuit. Fluid can be pushed through pressure lines but a suction line only has one atmosphere (approximately 14.7 psi at sea level) with which to work. Most pumps are slightly damaged above 3 psi. At 4 psi and higher, cavitation noise is evident and pump damage escalates. Higher vacuum accelerates the damage.

What causes cavitation damage?

Erosion is the result of cavitation implosions as fluid passes from the inlet side of a pump to the outlet side. ***Figure 8-34*** *shows how the change from vacuum to pressure makes the vacuum or* *air voids collapse or implode. At low pressure, these voids merely close up and no damage is done. At high pressure, the fluid does not stop when the void is full but continues at high velocity through the void and impinges metal surfaces to the point of getting into the metal's pores. Pressure drops as the next pulse of fluid approaches and high-pressure fluid in the metal pores rushes back out. During this part of the cycle, some very small particles of metal are dislodged and a cavity starts to form. Because this high- to low-pressure cycle can happen more than 200 times per second on a 12-vane pump at 1200 rpm, it is easy to understand how a pump can be physically damaged so quickly. These implosions are in the area where metal against metal is the only sealing action between vacuum and system pressure, so once the metal erodes, pump efficiency decreases because fluid bypasses through the damaged area.*

Fig. 8-34. Representation of erosion from implosions impinging on metal caused by cavitation or air leaks

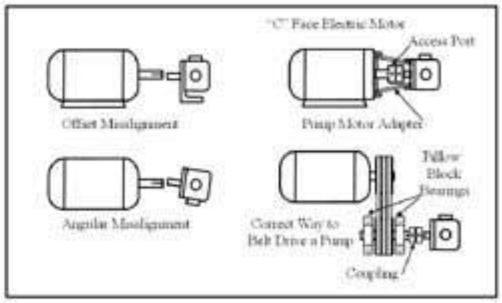


Pump-motor alignment

Most hydraulic pumps have light bearings while the electric motors by which they are driven have heavy-duty bearings. This makes it extremely important that the alignment of the pump and motor shaft be near perfect. Angular or offset misalignment always results in pump bearing failure, followed by internal failure soon after startup. Shaft couplings can take care of minor inconsistencies in shaft alignment, but they wear out very soon when not properly applied.

Figure 8-35 *illustrates examples of misalignment. When the pump and motor are mounted separately, they must be aligned as nearly perfect as possible. Straight edges, dial indicators, and lasers give accuracy ranging from low tech to high tech, but they are only part of the answer. The pump and motor must sit on a rigid base and must be held down with ample force so they do not slip around during operation. The best alignment job possible can be rendered useless by inadequate mounting hardware.*

Fig. 8-35. Correct way to belt-drive a pump



A simple way to overcome alignment problems is to use the pump-motor adapter shown in ***Figure 8-35****. The pump-motor adapter is attached to a "C" face electric motor that has a flat-machined face and pilot protrusion. This face and pilot are perpendicular to the shaft and concentric to very close tolerances. A matching pilot and face are machined on the pump. The pump-motor adapter has matching machined faces and pilot bores. It is purchased for a particular motor and pump, so it is the right length for the shafts specified and matches the motor and pump mounting flanges. When this assembly is bolted together all parts align perfectly.*

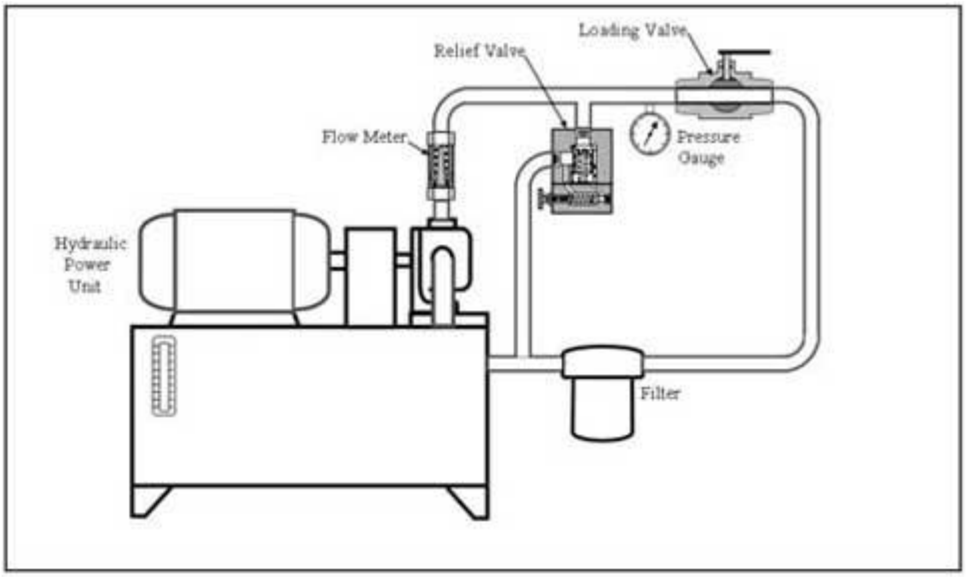
The shaft coupling then is slipped together and its setscrews tightened through the access port provided. The motor can be mounted on almost any surface without a chance of misalignment, and the pump can be changed without alignment problems anytime or place. Always use a coupling guard with an open coupling arrangement. Install the access-port cover before operating the pump when using the pump-motor adapter setup.

Figure 8-35 *also shows the correct way to drive a pump with a belt. Light bearings on the pump cannot stand the side loads from belts so the pump fails very soon. Use pillow-block bearings to take the side load and couple the pump to the bearing guided shaft. This arrangement gives long service in applications where belts must be used.*

Testing a pump

Figure 8-36 *shows a typical setup for testing a pump that is suspect, has been out of service, or has been rebuilt. The flow meter could be an added device if the unit does not have one. It could be part of a test stand setup but is a necessary item when checking pump efficiency. The loading valve could be a ball valve as shown in the figure or another type valve as long as it can take the maximum pressure it will see. The relief valve must be in place and set for maximum rated pressure or operating pressure as needed. A pressure gauge is required to indicate system pressure. The filter should be part of a standard hydraulic power unit, but would usually be an off-line setup on a test stand.*

Fig. 8-36. Test set-up for repaired pumps



To test a pump, lower the relief valve pressure setting to minimum. Then start the electric motor and check for flow on the flow meter. The meter should read at or very near catalog rating with all flow going directly to tank.

If the meter shows the pump producing ample flow, start closing the loading valve and watch the pressure gauge as it climbs. The reading should be low because the relief valve is set low. When the loading valve is closed completely, reset the relief valve to test pressure and observe the flow meter. Flow will drop somewhat, depending on the type of pump being tested. Most manufacturers publish rated flow at pressure in their literature. If the flow meter reads at or near cataloged rated flow, the pump is ready to put in service. If not, the pump should be checked or rebuilt to bring it up to specification.

Other pumps

Chapter 18 covers air- and hydraulic-driven intensifiers or boosters, which technically are pumps. These units usually are associated with air-oil systems. That is why their descriptions are in Chapter 18.

Air-to-hydraulic intensifiers are 100% efficient in the hydraulic end and are pressure compensated. They usually produce low volume so they are not normally used as a system's prime mover. Their main advantage is they can hold pressure for long periods without generating heat or consuming energy. (Check out Chapter 17 to learn more about this unique pumping system.)

Pumps

Figures 15-1 through 15-5 show the schematic symbols for several fixed-displacement pumps. Use fixed-displacement pumps in simple, one- or two-cylinder circuits that never stop under pressure. Also use them for single-speed motor circuits, or circuits where several cylinders operate simultaneously but never stop and hold at full pressure. Fixed-displacement pumps always move a set volume of fluid at a pressure between that dictated by resistance and the maximum relief valve setting. Blocking the outlet of a fixed-displacement pump sends excess flow through the relief valve to tank. When fluid goes across the relief valve at pressure, all of the input energy generates heat.

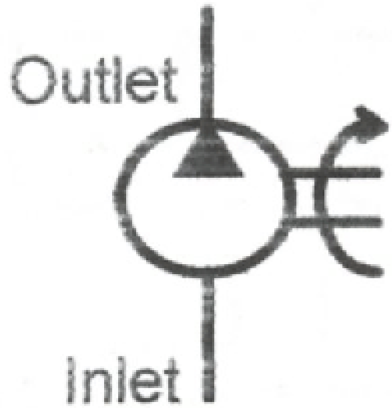


Fig. 15-1. Symbol for single pump

Fixed-displacement pumps may be gear, gerotor, vane, or piston types. The most common are gear and vane. They are relatively inexpensive, very reliable, and generate little heat when used correctly.

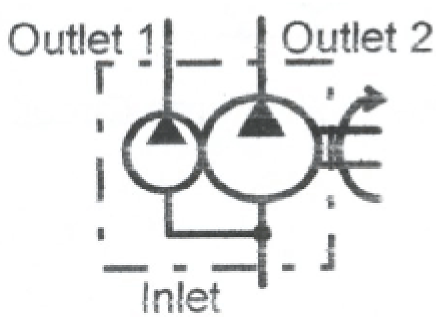


Fig. 15-2. Symbol for double pump

Gear and vane pumps come in a wide variety of configurations. Figures 15-1 through 15-3 show one or more pumps in a single housing. The pumps may share a common inlet or have multiple *inlets. Most combination pumps have separate outlets for use in different sub-circuits. The flow from each pump in the combination may be the same or different.*

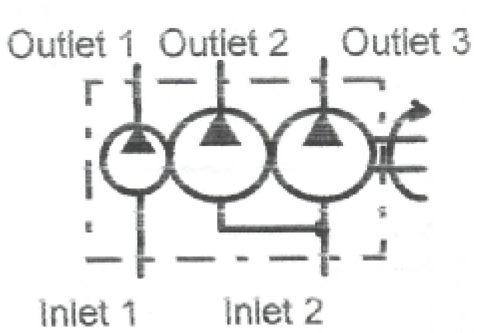


Fig. 15-3. Symbol for triple pump

Figure 15-4 shows the symbol for a self-contained double pump for a high-low circuit. Flow from both pumps moves an actuator to and from the work at low pressure. The high-volume pump unloads through an integral unloading valve at work contact. This leaves all motor horsepower to drive the low-volume/high-pressure pump. This circuit usually consumes less horsepower without sacrificing cycle time. The packaged pump represented here is compact and inexpensive, but any double pump with the correct valves can supply a high-low circuit.

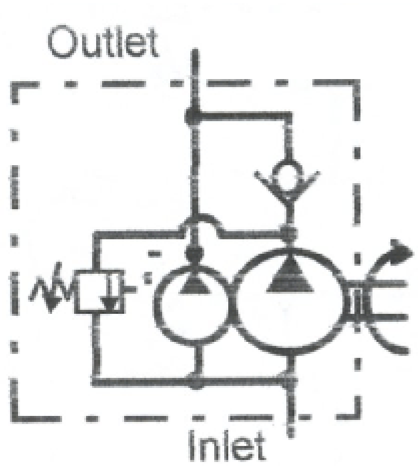


Fig. 15-4. Symbol for High-low pump

Many manufacturers produce thru-drive pumps like the one shown in Figure 15-5. A double-shafted electric motor normally operates both pumps. With a thru-drive pump, a second pump is bolted to and driven by the shaft of the first pump. When connecting more than two pumps, consider some possible problems: will the shaft of the first pump handle the torque of additional pumps; will additional pumps result in too much overhung load from too many pumps.

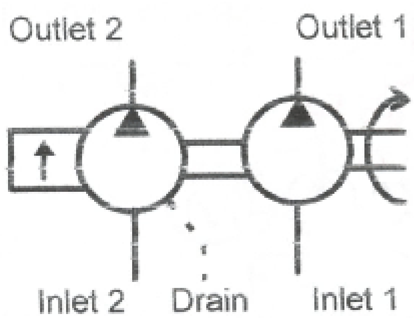


Fig. 15-5. Symbol for thru-drive pump

Fixed-displacement pump circuits

Figure 15-6 shows a schematic circuit for a fixed-displacement pump operating a single cylinder. At rest, the pump unloads through a tandem-center valve at minimum pressure. When the cylinder extends, pressure is whatever it takes to stroke the cylinder. When the cylinder contacts the work, pressure increases to whatever it takes to perform the work. As the cylinder retracts, pressure is whatever it takes to return the cylinder and load. At no time does the relief valve dump oil to tank. Therefore, this circuit operates with little heat and should not require a heat exchanger when using high-efficiency parts.

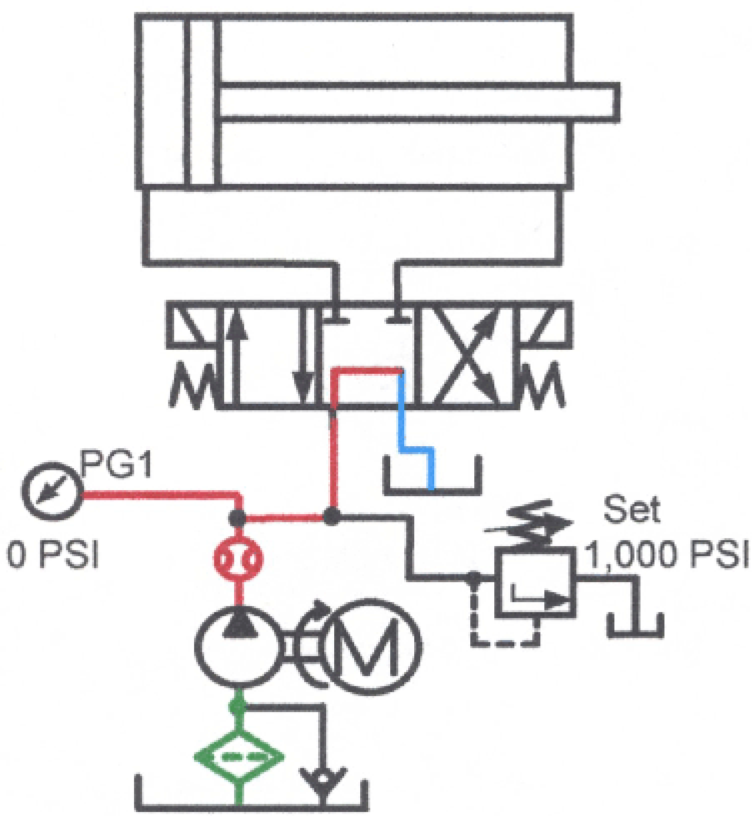


Fig. 15-6. Typical fixed-displacement pump circuit

Figure 15-7 depicts one way to use fixed-displacement pumps in a multiple-cylinder circuit. Each of the three cylinders in this example has a separate pump, relief valve, and directional valve. The actuators move at the desired speed and force because each pump's flow and relief valve settings match their cylinder's work requirement. Because there are no flow controls, the relief valves never dump excess fluid, allowing all input energy to do useful work. Heat should not be a problem in this circuit.

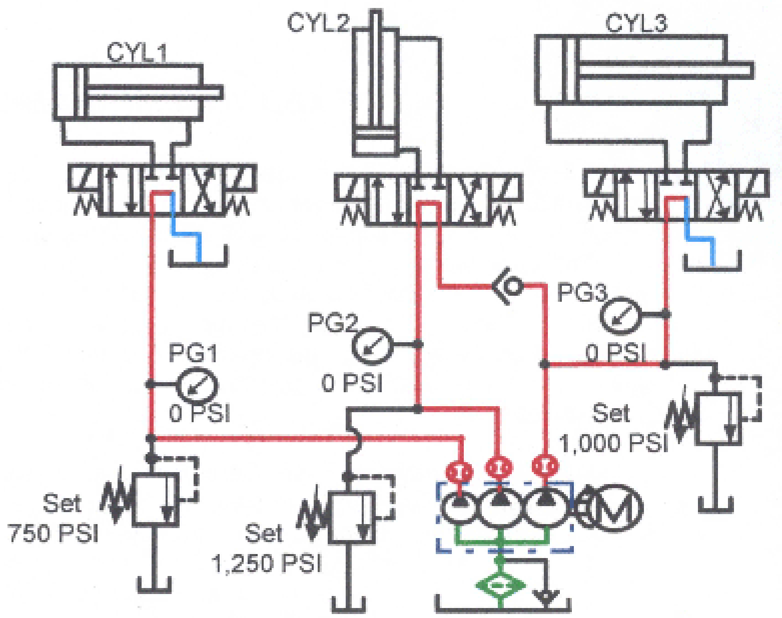


Fig. 15-7. Circuit with three fixed-displacement pumps supplying three actuators

Two pumps supply CYL3 to stroke it rapidly. This circuit works best when CYL2 does not cycle at the same time CYL3 does.

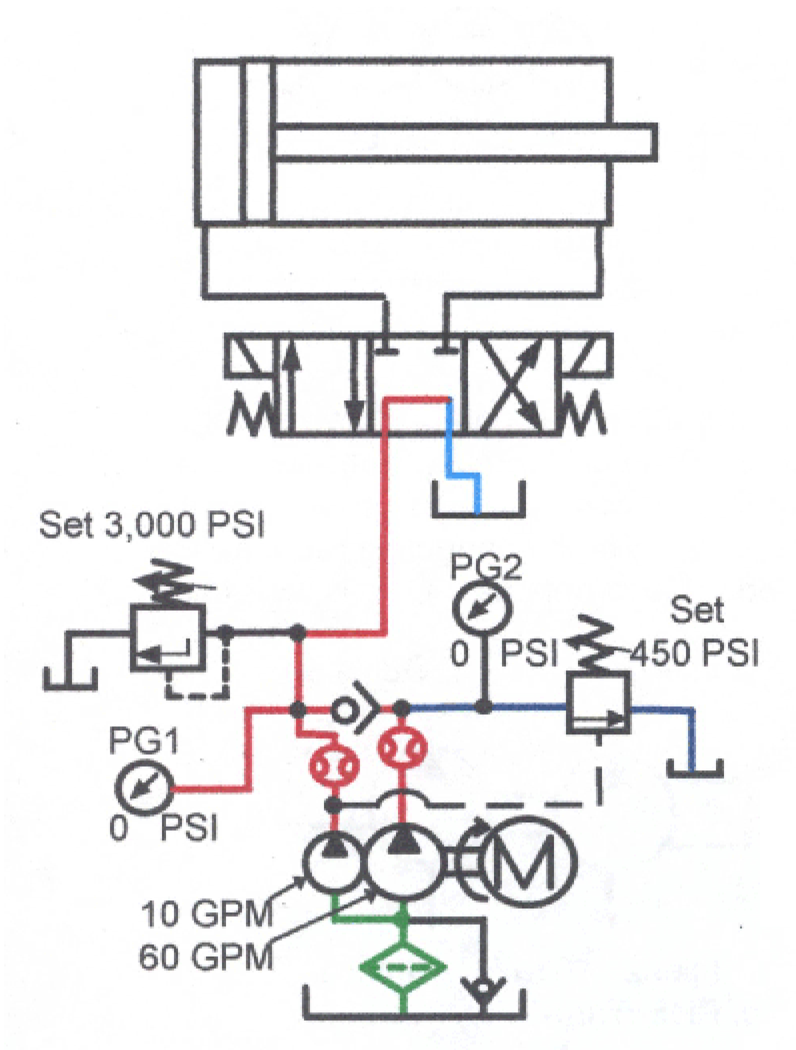


Fig. 15-8. Typical fixed-displacement high-low circuit

It takes time to design efficient circuits, but the results pay off in future savings. The high-low circuit in Figure 15-8 — which cycles a large fast-stroking cylinder — saves on both first cost and operating cost. If a single 60-gpm pump operating at 3000 psi were used, a 120-hp motor would be required. By substituting a double pump with 60- and 10-gpm sections, the motor size can be reduced without sacrificing cycle time. The big difference occurs because moving the cylinder at say 450 and 500 psi, only requires 20.4 hp. When the cylinder meets resistance and pressure builds to about 500 psi or higher, the 60-gpm pump section unloads at no pressure while the 10-gpm pump does the work. The 10-gpm pump at 3000 psi requires 17.5 hp. Although work speed is slower, travel time is faster. With a little figuring, it's easy to save money on the electric motor and controls up front, and reduce energy cost for the life of the machine.

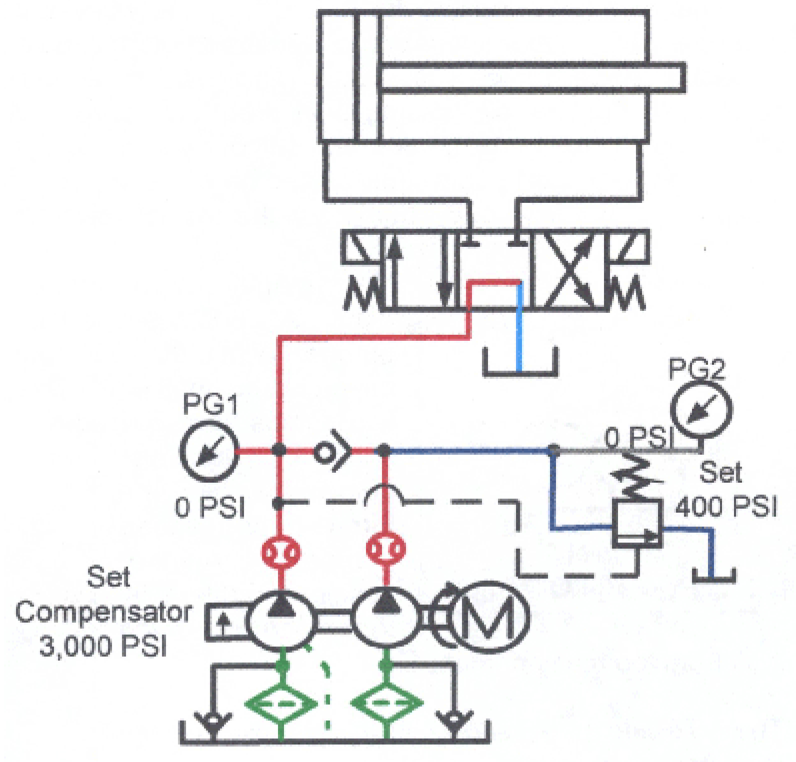


Fig. 15-9. High-low pump circuit to operate clamping cylinder

In Figure 15-9, a fixed high-displacement thru-drive pump, coupled with a low-displacement, pressure-compensated pump, creates a different kind of high-low circuit. This circuit provides fast travel and then maintains clamping pressure for extended periods with little heat generation. The circuit operation is the same as Figure 15-8. It requires no special electric controls because the unloading valve automatically dumps the high-displacement pump at any pressure above 400 psi. The low-displacement' pressure-compensated pump reduces energy cost and heating. This pump arrangement takes the place of a large pressure-compensated pump in certain applications.

Pressure-compensated and variable displacement pumps

One way to keep from generating heat while maintaining pressure is to use pressure-compensated pumps. Flow from pressure-compensated pumps drops to almost nothing when they reach compensator pressure. Reduced flow cuts horsepower consumption and keeps the system from overheating. Be advised: pressure-compensated pumps are more expensive than fixed-displacement pumps and usually are less contamination tolerant. Also, pressure-compensated pumps only come in vane or piston design. Other pump designs are not capable of variable-displacement while turning the same rotational speed.

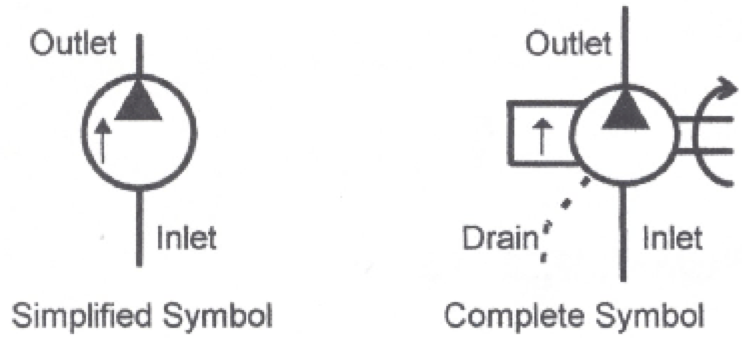


Fig. 15-10. Symbols for pressure-compensated pumps

Figure 15-10 shows symbols for a pressure-compensated pump. The arrow inside the circle, parallel to the flow path, indicates pressure compensation. The complete symbol shows all the operating features. The simplified symbol leaves out some details (such as the case drain) and assumes that the person reading the schematic diagram knows their necessity.

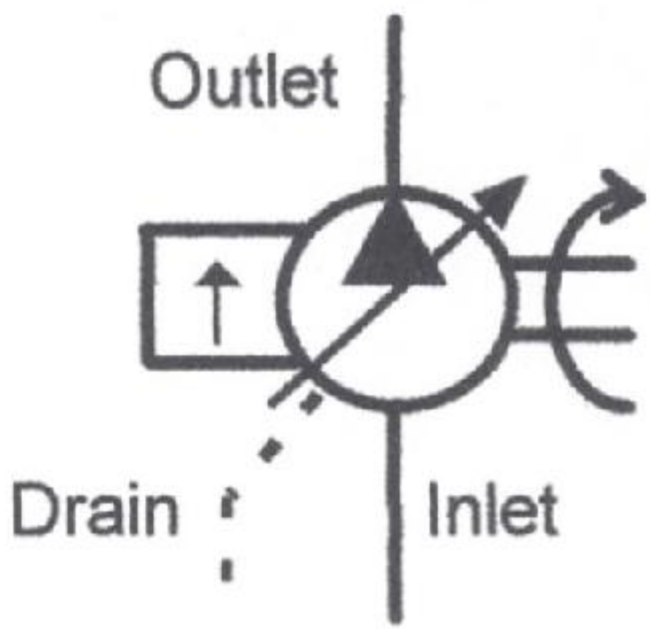


Fig. 15-11. Symbol for pressure-compensated, variable-displacement pump

Many pressure-compensated pumps include a method to adjust maximum flow, making the pump more versatile. The symbol in Figure 15-11 indicates a pressure-compensated, variable-displacement pump. The angled arrow through the pump symbol designates variable or adjustable flow. Pressure-compensated pump flow automatically decreases when pressure *increases, but the sloping arrow indicates a variable maximum output volume as well. A variable-displacement pump can eliminate the need for flow controls in some circuits.*

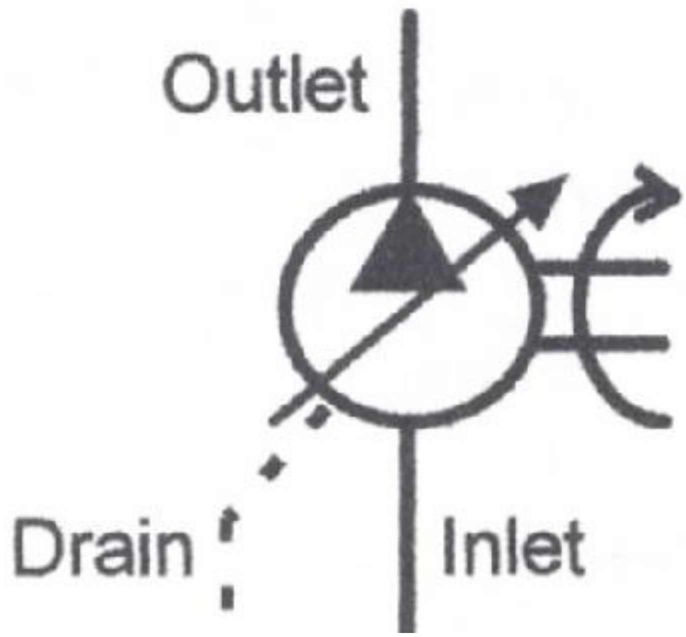


Fig. 15-12. Symbol for variable-displacement pump

The variable-displacement pump in Figure 15-12 is not pressure-compensated. Use this type of pump to change the speed of an actuator without wasting energy. Controlling speed this way produces less heat. Control of variable-displacement pumps may be manual, hydraulic, or electrical with servo or proportional valves.

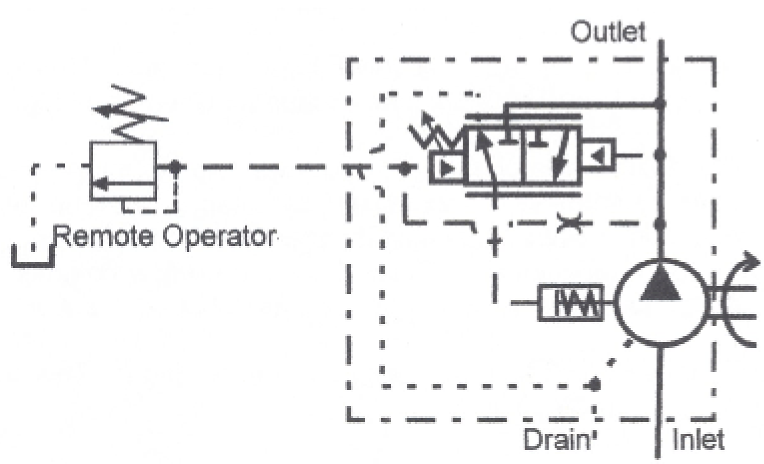


Fig. 15-13. Circuit for pressure-compensated pump with remote operator

Figure 15-13 shows the symbol for a pressure-compensated pump with a remote operator to adjust maximum pressure. Set the pump compensator for minimum pressure and adjust system pressure at a remote location. This schematic shows a manually adjustable remote relief valve installed near the operator for easy access.

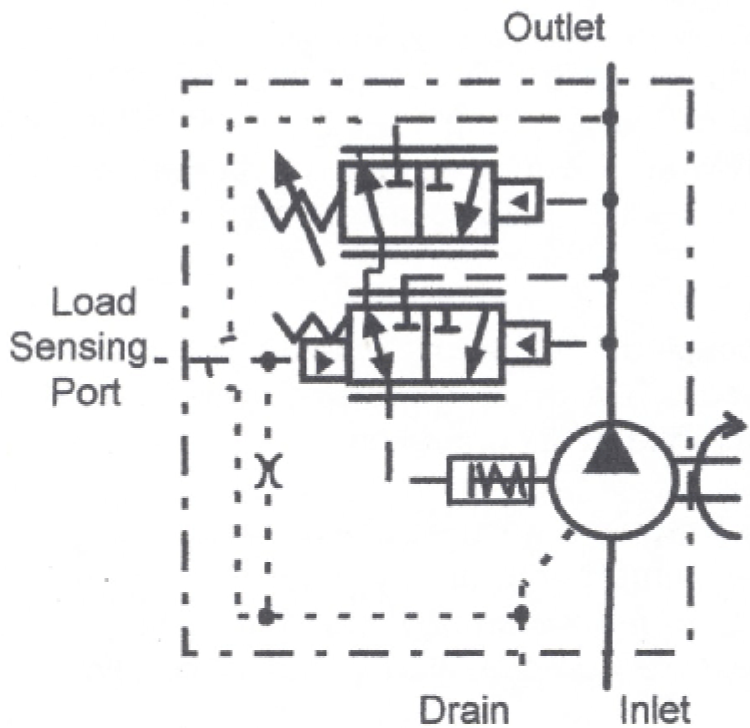


Fig. 15-14. Load-sensing, pressure-compensated pump

A load-sensing feature can be added to pressure-compensated pumps. Figure 15-14 shows a pump symbol for such a combination. An extra port in the pump samples pressure in the flow lines to the actuator. Sensing actual working pressure causes the pump to compensate to flow demand at pressures 100 to 150 psi higher than working pressure. Load sensing is only advantageous in circuits using less than maximum pump flow. In these circuits load-sensing pumps are more efficient — wasting less energy by and reducing oil heating.

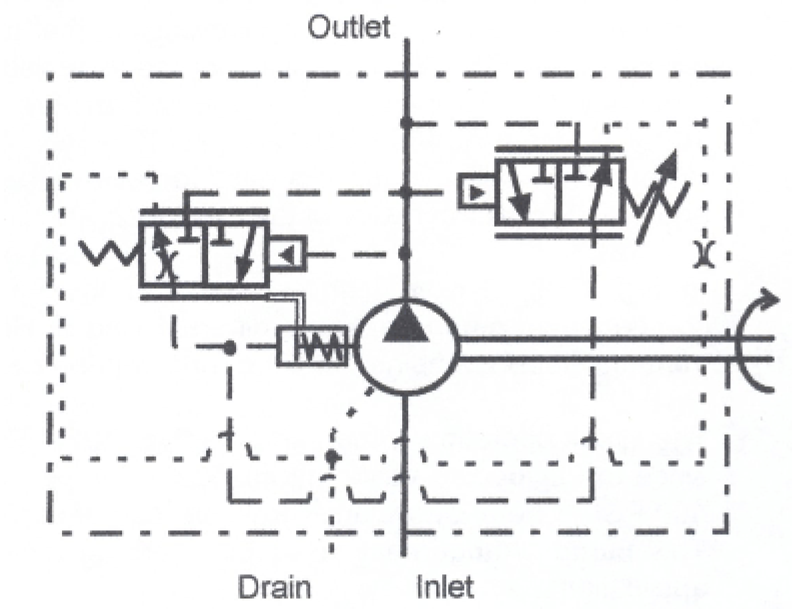


Fig. 15-15. Pressure-compensated pump with horsepower-limiting feature

---

The pump in Figure 15-15 is pressure-compensated with horsepower limiting added. When maximum required pump horsepower can exceed the power of the prime mover, use a horsepower limiter. Horsepower limiting allows the use of a smaller gas or diesel engine with a high volume pump on off road equipment.

Set the compensator on a horsepower-limiting pump for maximum system pressure at compensated flow. As pressure increases at high flow, the horsepower needed could exceed that available. A horsepower limiter reduces pump displacement at a predetermined pressure. Reducing pump displacement as pressure climbs lowers the horsepower requirement to that available. With this system a 20-hp motor can drive a 60-gpm pump to 5000 psi at reduced flow.

Pressure-compensated, variable-displacement pump circuits

To control the speed of an actuator while generating little or no heat, try the circuit in Figure 15-16. A variable-displacement pump controls cylinder speed fairly accurately while using minimum power. As the cylinder strokes, pressure in the system is only what it takes to move the load. All pump flow goes to the cylinder so the only wasted energy is from component inefficiency. A setup like this operates continuously without a heat exchanger. The oil temperature may rise 15 to 25 degrees above ambient temperature only when cycle rates exceed ten or more per minute.

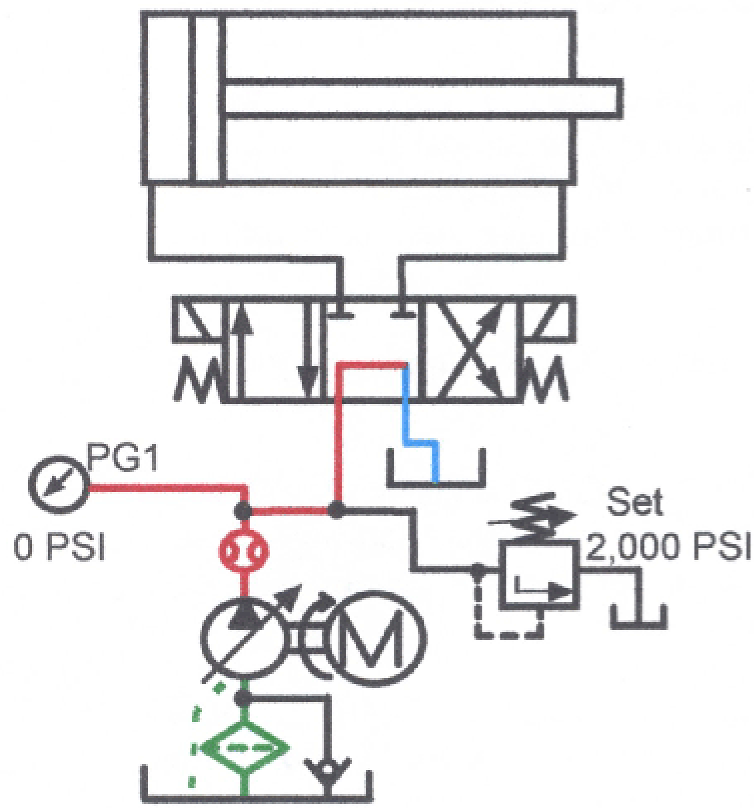


Fig. 15-16. Variable-displacement pump circuit for speed control

---

The circuit in Figure 15-17 is a typical pressure-compensated pump setup. This circuit allows multiple cylinders to run separately or together. When the cylinders cycle simultaneously, add flow controls to restrict the actuator that meets the least resistance.

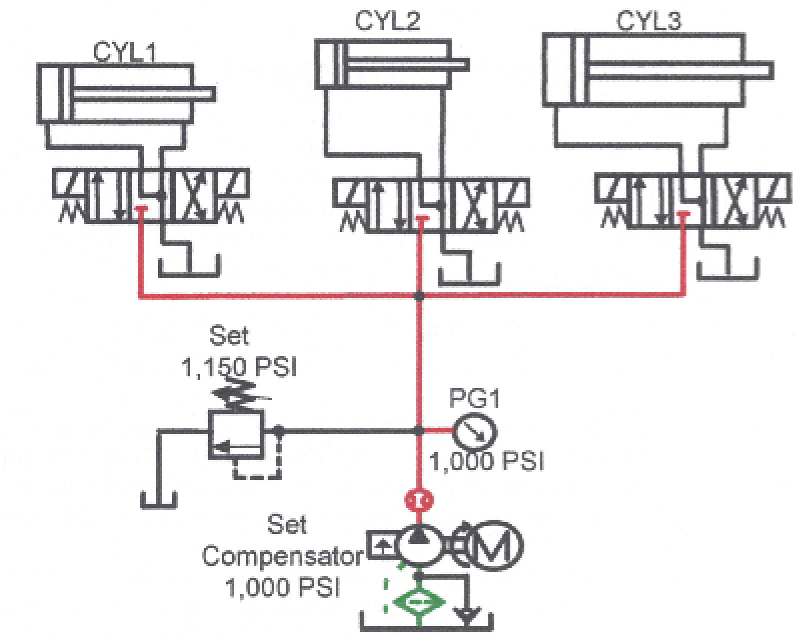


Fig. 15-17. Typical pressure-compensated pump circuit

Oil heating can be a problem in pressure-compensated pump circuits. With the pump set at high pressure and/or if flow controls are used in the circuit, energy losses produce excess heat. The efficiency of the directional valves also is a factor. Because the system maintains maximum pressure most of the time, leakage at valve spools adds extra heat.

Pressure-compensated pumps often fail prematurely with high actuator cycle rate. High cycle rates work the compensating mechanism rapidly and resulting pressure spikes can cause part failure. A small accumulator at the pump outlet smoothes the compensator shift cycle, reducing pressure spikes and extending component life.

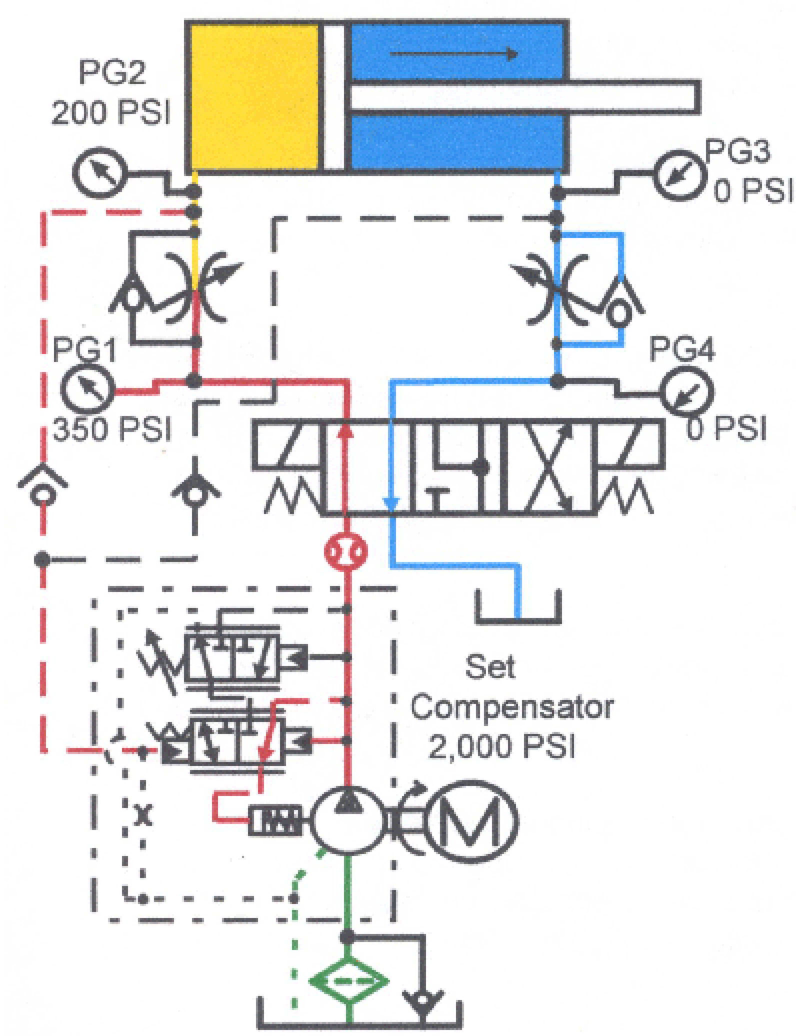


Fig. 15-18. Load-sensing pump circuit

One way to overcome a heating problem is with the pump in Figure 15-18. When the cylinder cycles, this load-sensing, pressure-compensated pump never allows system pressure to go more than 150 to 200 psi above load demand. The pump constantly senses the load and compensates at that pressure plus the load-sensing spring rate. Load sensing usually eliminates the need for a heat exchanger — even on a system with flow controls.

Run a sensing line from each port in a multiple-actuator circuit. The different feedback lines meet at the pump's load-sensing port with a check valve to isolate them from each other. The pump always sees the highest load in the circuit and sets output pressure accordingly. A meter-in flow control circuit is the only way to control the actuator. With over-running loads, use a counterbalance valve to keep the actuator from running away.

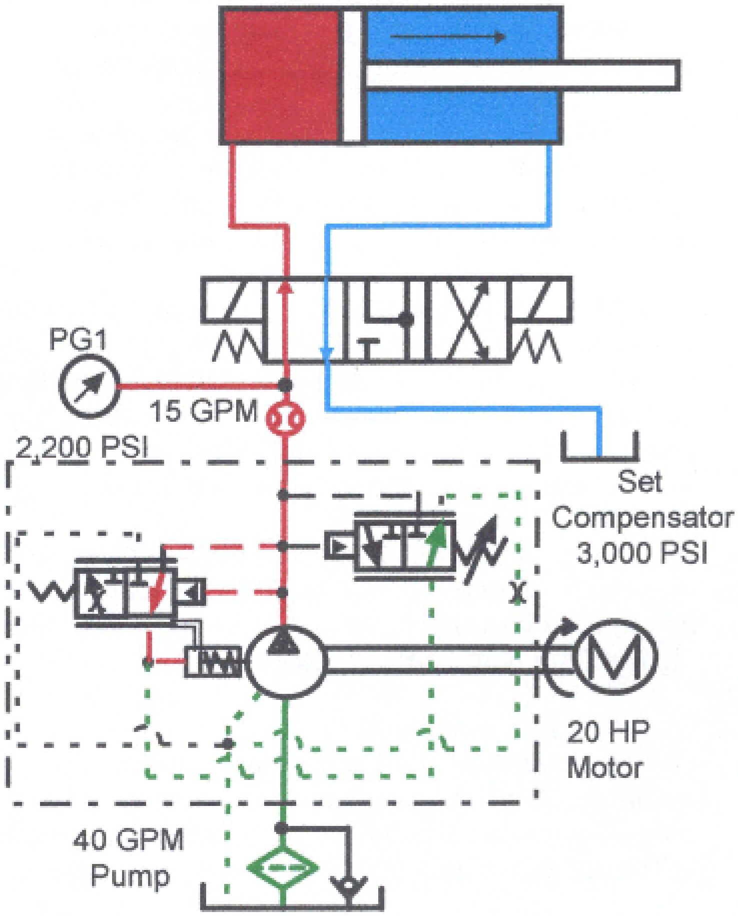


Fig. 15-19. Horsepower-limiting pump circuit

When driving the pump with an engine — or to save energy with a smaller electric motor — use the horsepower-limiting circuit in Figure 15-19. This circuit changes pump displacement whenever the horsepower required is greater than that called for by the compensator spring setting. The horsepower compensator can be factory set or field adjustable. When system pressure reaches the setting of the pump compensator, output goes to no flow like any pressure-compensated pump.

Bi-directional pumps
Axial- and radial-piston pumps can output fluid from either port while rotating in one direction. Closed-loop circuits take advantage of this feature of piston pumps. A closed-loop pump circuit sends fluid to an actuator while fluid from the same device comes back to the pump's inlet.

(Do not confuse bi-directional pumps with bi-rotational pumps. Bi-rotational pumps can flow out of either port, but only when rotation reverses. A bi-rotational pump has one port hooked to tank and the other port piped to the circuit. Most bi-rotational pumps operate hydraulic circuits on off-road equipment because rotation of the pump-driving shaft is different from one piece of equipment to another.)

Normally, bi-directional pumps do not have a port piped to tank. Both ports hook directly to the cylinder or motor ports. Many bi-directional circuits operate hydraulic motors, because they accept and return nearly the same amount of fluid. The most common closed-loop circuit is the hydrostatic drive — often used on off-road equipment.

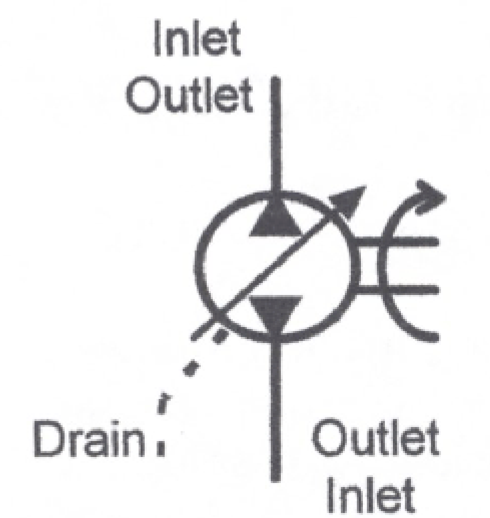


Fig. 15-20. Symbol for bi-directional pump

Figure 15-20 shows the schematic symbol for a bi-directional pump. Notice that there are two energy triangles to show that fluid flows out of both ports. The pump only outputs from one port at a time while the opposite port is inlet. With one port hooked to tank and the other port piped to a circuit, the pump serves as a variable-displacement uni-directional pump. Flow direction of a bi-directional pump hooked up this way depends on the position of the stroking control. By changing the position of the stroking control, either port can serve as the inlet or outlet.

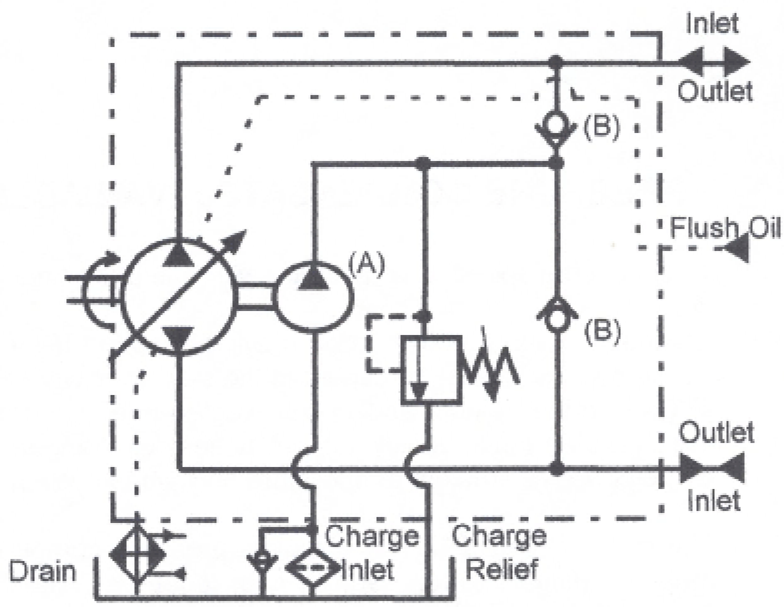


Fig. 15-21. Bi-directional pump in closed-loop hydrostatic circuit

---

Figure 15-21 shows a hydrostatic transmission — a common bi-directional pump circuit. Small fixed-displacement pump A (called a charge pump) makes up for leakage in the main pump and motor while the circuit operates. Check valves B protect the charge pump and only allow oil into the return side of the closed loop. Charge relief valve C dumps excess charge flow to tank at 150 to 300 psi. Charge pump flow generates heat in hydrostatic systems. Many hydrostatic systems use charge pump fluid to operate pump controls and/or auxiliary circuits.

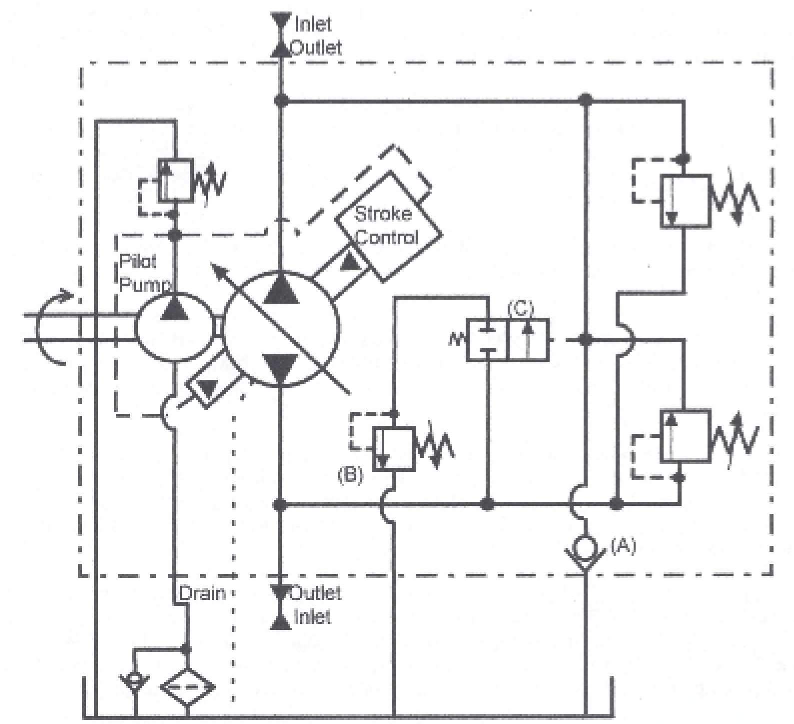


Fig. 15-22. Bi-directional closed-loop pump circuit

When return flow does not equal output flow, use the bi-directional pump schematic shown in Figure 15-22. With a single-rod end cylinder attached to a bi-directional pump, the volume of fluid going to the cap end when the cylinder extends is greater than the flow returning to the pump from the rod end. When cylinder direction changes, the opposite is true. Without a way to overcome flow inequalities, a bi-directional pump powering a single-rod cylinder would not work.

For single-rod cylinders, add check valve A, low-pressure relief valve B, and NC pilot-operated 2-way valve C to the closed-loop circuit. Check valve A allows the pump to take oil from the tank when the cylinder extends. Relief valve B and 2-way valve C provide a path for excess oil to go to tank when the cylinder retracts.

Often large cylinders operating at high pressure and speed use bi-directional pumps with unequal flow capabilities. This circuit is very efficient and virtually eliminates hydraulic shock

Bi-directional pump circuits

By controlling the volume of flow and its direction from a bi-directional pump, a hydraulic motor can be made to turn in either direction at infinitely variable speeds. A closed-loop circuit wastes very little energy. There is minimum shock when starting or changing direction because the pump starts from and passes through no-flow as it cycles. The hydraulic motor decelerates smoothly when pump flow goes to zero, slowing whatever load it drives.

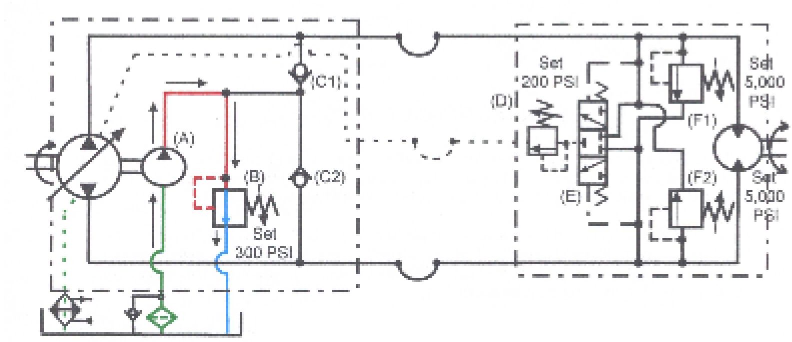


Fig. 15-23. Closed-loop bi-directional hydrostatic drive; at rest with pump running

---

Figure 15-23 shows the parts of a simple hydrostatic transmission that uses this type of circuit. It consists of a variable-displacement bi-directional pump piped to a fixed-displacement bi-directional hydraulic motor in a closed loop. Charge pump A, driven off the bi-directional pump, takes oil from the tank and feeds it through check valves C1 and C2 to keep the closed loop filled. Excess oil from the charge pump discharges across relief valve B to tank. Shuttle valve E and relief valve D send charge flow into the low-pressure side of the closed loop when the hydraulic motor runs. This happens because the set pressure of relief valve D is approximately 100 psi lower than that of relief valve B. Continuous infusion of cool filtered oil protects the closed loop from overheating and contamination.

Cross-port relief valves F1 and F2 protect the pump and motor from excess pressure. When pressure in the closed loop exceeds the relief valve setting, oil bypasses to the opposite line. However, because system volume is small, the flow through the bypass builds heat rapidly. This heat can damage components, hoses, and seals. Most hydrostatic circuits now use valves to destroke the pump at a slightly lower pressure than the cross-port relief valve setting. This destroking valve eliminates pump flow heating but does not help when a driven hydraulic motor acts as a pump.

(Replacing the closed loop circuit with a 4-way directional valve and a fixed-displacement pump with flow controls to vary speed could also operate the hydraulic motor in either direction. This simplified circuit costs about one fifth as much as a hydrostatic transmission. However, the costs of system shock, oil heating, and machine damage caused by the cheaper system far outweigh the original cost savings.)

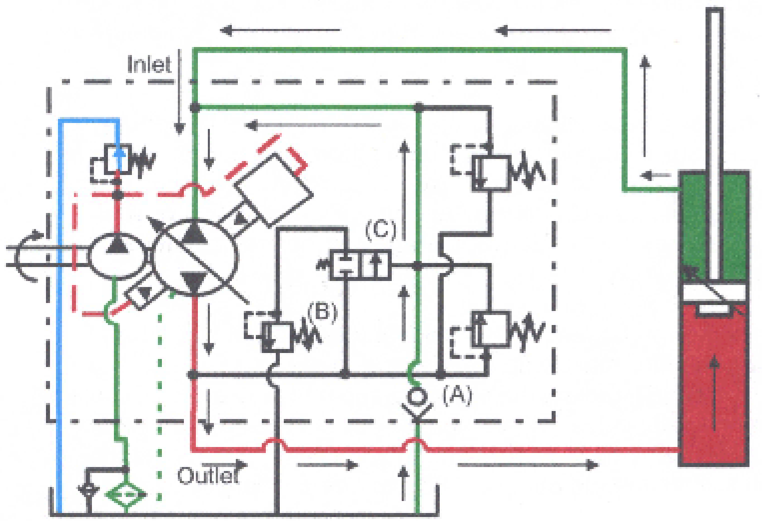


Fig. 15-24. Bi-directional closed-loop pump extending single-rod cylinder

---

Using a closed-loop pump with a single-rod cylinder requires additional valving in the pump. Figures 15-24 and 15-25 show a single-acting cylinder circuit operated from a bi-directional closed-open-loop pump. The term closed-open-loop indicates that the pump is bi-directional, but one port is connected to tank through check valve A. This keeps the pump from starving when the cylinder extends. Also, low-pressure relief valve B and NC 2-way valve C provide a path to tank for excess flow from the cylinder cap end when it retracts.

When the cylinder extends as in Figure 15-24, flow from the rod end of the cylinder cannot fill the pump. Because the pump needs more oil than the cylinder supplies, check valve A opens to allow oil from the tank to enter the pump. (Note that large cylinder rods increase the need for flow from the charge pump and tank.)

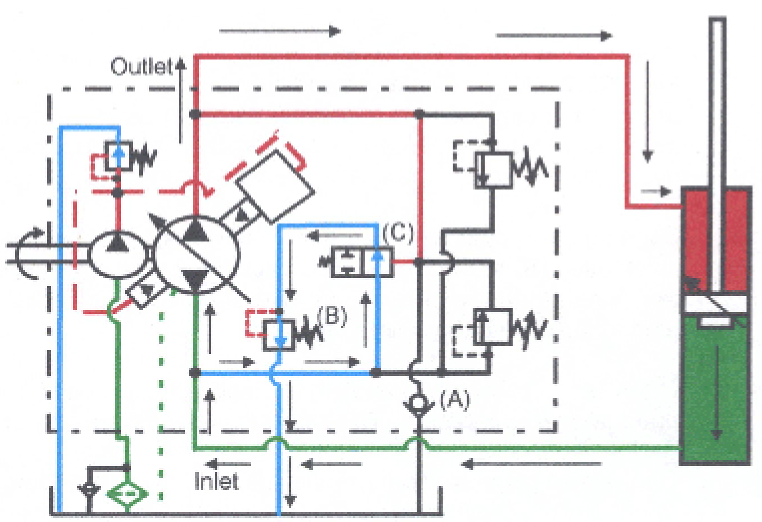


Fig. 15-25. Bi-directional closed-loop pump retracting single-

rod cylinder

When the cylinder retracts as in Figure 15-25, oil from the cap end of the cylinder is more than the pump needs. During this part of the cycle, pilot pressure opens NC 2-way valve C to allow excess cylinder flow to pass through low-pressure relief valve B to tank. (Again, the larger the cylinder rod, the greater the volume of oil ported to tank.)

The main reason for using bi-directional pumps is the very smooth control of the actuator they provide. Bi-directional pumps completely control starting, stopping, and reversing of the largest high-speed actuators. This practically eliminates system shock and greatly extends machine life.

Pressure-control valves

Several types of pressure-control valves are found in fluid power circuits. Some keep the whole system from excess pressure while others only protect a portion of the system. Others allow flow to an isolated circuit after reaching a preset pressure. Some bypass fluid at low or no pressure when activated.

This chapter only covers relief valves and unloading valves because they are closely associated with hydraulic pumps. The other pressure-control valves are part of the control circuit and will be dealt with after directional control valves.

Why relief valves?

All fixed-volume pump circuits require a relief valve to protect the system from excess pressure. Fixed-volume pumps must move fluid when they turn. When a pump is unloading through an open-center circuit or actuators are in motion, fluid movement is not a problem. It is when the actuators stall with the directional valve still shifted that a relief valve is essential.

Pressure compensated pump circuits could run successfully without relief valves because they only move fluid when pressure drops below their compensator setting. (Most designers still use a relief valve in these circuits for reasons explained later.)

In either case, a relief valve is similar to a fuse in an electrical system. When circuit amperage stays below the fuse amperage, all is well. When circuit amperage tries to exceed fuse amperage, the fuse blows and disables the circuit. Both devices protect the system from excess pressure by keeping it below a preset level.

The difference is that when an electrical fuse blows it must be reset or replaced by maintenance personnel before the machine can cycle again. This requirement alerts the electricians to a possible problem and usually causes them to look for the reason before restarting the machine. Without the protection of a fuse, the electrical circuit would finally overheat and start a fire.

In a hydraulic circuit, a relief valve opens and bypasses fluid when pressure exceeds its setting. The valve then closes again when pressure falls. This means a relief valve can bypass fluid anytime . . . or all the time . . . without intervention by maintenance. (It also means the system can run hot even with a heat exchanger installed.)

Many fixed-volume pump circuits depend on this bypassing capability during the cycle, and some even bypass fluid during idle time. A well-designed circuit never bypasses fluid unless there is a malfunction, such as a limit switch not closing or an operator overriding the controls. This eliminates most overheating problems and saves energy.

Relief valve operation

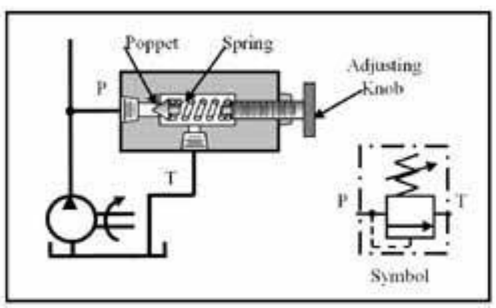
There are two different designs of relief valves in use: direct acting and pilot operated. Both types have advantages and work better in certain applications.
Some terms relating to relief valves and their function are:

- *Overshoot*: The actual pressure reading when a relief valve first opens to bypass fluid. (It can be up to twice the actual pressure setting.)
- *Hysteresis*: The difference in pressure between when a relief valve starts letting some flow pass (cracking pressure) and when full flow is passing.
- *Stability*: The fluctuation of pressure as a relief valve is bypassing at set pressure.
- *Reseat pressure*: The pressure at which a relief valve closes after it has been bypassing.
- *Pressure override*: The difference in the pressure reading from the time a relief valve first opens (cracking pressure) until it is passing all pump flow to tank.

Direct-acting relief valves

Figure 9-1 *shows a cutaway view and the symbol for a direct-acting relief valve. The valve has a poppet that is pressed against its seat by an adjustable spring. An adjusting knob can be change the force on the spring to raise or lower maximum pressure. The poppet remains seated while pump flow goes to the circuit and pressure is lower than the relief valve setting. If pressure tries to go above spring setting, the poppet is forced off the seat just enough to pass excess pump flow to tank.*

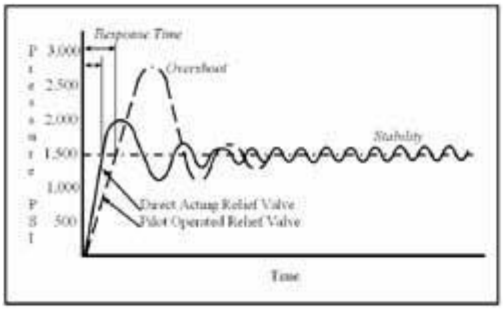
Fig. 9-1. Cutaway drawing and symbol for
direct-acting relief valve.



The symbol shows a single box with a flow arrow offset from the inlet P and outlet T flow lines. The dashed pilot line from the inlet line to the bottom of the box indicates inlet pressure can push against the flow arrow. On the opposite side of the box is a spring with a sloping arrow through it to show an opposing force on the flow arrow. When pressure at port P builds enough to overcome spring pressure, it forces the flow arrow up until there is a path from P to T. Although there is no pilot passage in the actual valve, the function is implied and thus is part of the symbol.

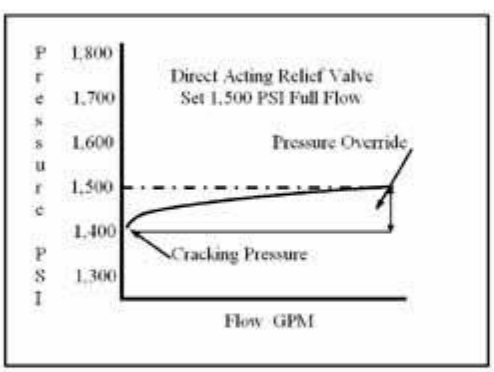
The main advantage of direct-acting relief valves over pilot operated relief valves is that they respond very rapidly to pressure buildup. Any relief valve does not know there is a problem until pressure is very near or at its setting. Then it must open to relieve excess flow as quickly as possible to keep pressure overshoot low. Because there is only one moving part in a direct-acting relief valve, it can open rapidly, thus minimizing pressure spikes. ***Figure 9-2*** *shows typical performance graphs from direct-acting and pilot-operated relief valves. Notice the difference in response time and pressure spikes as the valves open to send excess flow to tank.*

Fig. 9-2. Typical performance plots for direct-acting and pilot-operated relief valves



The main disadvantage of direct-acting relief valves is that they open partially at about 150 psi below set pressure. Because the poppet is in direct contact with the spring that sets maximum pressure, when the poppet opens it forces the spring back and increases pressure. The amount depends on the spring's length and stiffness. The plot in ***Figure 9-3*** *shows the flow/pressure relationship of a typical direct-acting relief valve. With a direct-acting relief valve setting of 1500 psi at 10 gpm, it is very possible that some fluid will start to pass when pressure is as low as 1350 to 1400 psi. Continued pressure increase allows more flow until all pump flow goes to tank at 1500 psi. If work is still being performed at 1450 psi, it will be at a reduced speed because some flow is going to tank. When this valve is set at 1500-psi cracking pressure, no flow will bypass until pressure reaches that level, but final pressure would be as high as 1650 psi. (Pilot-operated relief valves . . . discussed next . . . do not start to open until pressure is within 25 to 50 psi of their settings.)*

Fig. 9-3. Plot of flow-pressure relationship of a typical direct acting relief valve.



Direct-acting relief valves often are quite noisy due to the high velocity of the fluid bypass and the instability inherent in their design.

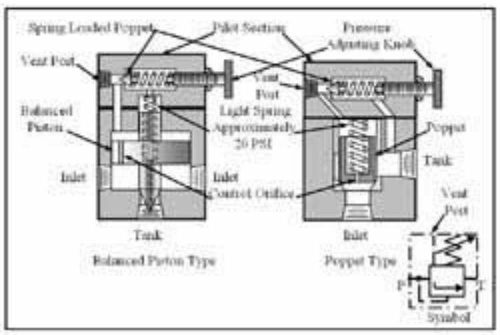
Direct-acting relief valves are not normally used on industrial hydraulic systems, except for those with flows under 3 gpm, and as pilot control devices. Most industrial designs use long springs that gain little force per compression increment to keep pressure override low.

When a direct-acting relief valve is specified as preset, non-adjustable, always specify whether the valve is to be set for cracking pressure or full flow. If full flow is desired, a flow must be specified also.

Pilot-operated relief valves

Figure 9-4 *shows cutaway views of two common types of pilot-operated relief valves. There are many variations of these designs but the function and symbol are the same. The pilot section on each valve is a low-flow direct-acting relief valve that sets maximum system pressure. Because the valve is small and passes very little flow, it has less than 50-psi pressure override as it operates.*

Fig. 9-4. Cutaway view and symbol for two common types of pilot-operated relief valves.



The control orifice in the balanced piston or poppet usually has a diameter around 0.040 in. This size gives good relief-flow stability and is not prone to becoming blocked with contamination. If the orifice is plugged, the balanced piston or poppet will open at approximately 20 psi and dump all pump flow to tank.

A flow path from the outlet of the control orifice . . . on top of the balanced piston or poppet . . . leads up to the pilot section, which contains a spring-loaded poppet. Adjusting the tension on the spring-loaded poppet sets the pressure in the circuit. Fluid used by the pilot section returns to tank through the tank port. The balanced-piston type has a hole through it that lets control fluid flow to tank. The vent port in the pilot section is normally plugged. (Removing the plug allows this valve to perform other functions.)

Many inline-mounted valves have two inlet ports as a piping convenience. Pump flow comes in one inlet and exits through the opposite one. This eliminates the need for a tee in the pump line plumbing.

How a pilot-operated relief valve works

Pump flow enters the inlet port and flows to the circuit and through the control orifice to the top side of the balanced piston or poppet. It also travels up to the pilot section's spring-loaded poppet, where it is blocked. When pressure is too low to unseat the spring-loaded poppet, pressure is the same on either side of the balanced piston or poppet. Because hydraulic forces are equal on both sides of the balanced piston or poppet, the light spring holds them in their normally closed position. This condition continues until pressure reaches approximately 25 to 50 psi below the pressure set at the relief valve pressure-adjusting knob.

For example, if pressure was set at 1000 psi, at around 950 psi the spring-loaded poppet in the pilot section will crack open and allow a small amount of fluid to pass to tank. At this point the amount of fluid passing the spring-loaded poppet can easily flow through the control orifice so pump flow to tank is blocked. As pressure continues to increase, it finally forces the spring-loaded poppet in the pilot section to open far enough so that flow through it is greater than flow through the control orifice. When flow through the spring-loaded poppet is more than flow through the control orifice, pressure on top of the balanced piston or poppet decreases. When the pressure imbalance is great enough, the balanced piston or poppet moves toward the decreased pressure and opens a flow path to tank. Flow to tank is just enough to bypass any excess fluid the system is not using. As a relief function, this valve never opens more than enough to bypass excess flow.

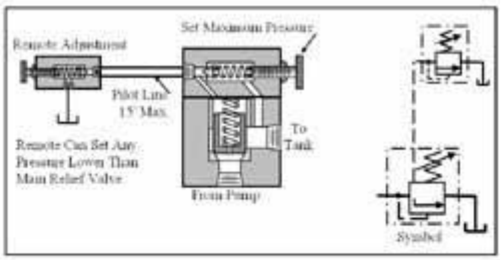
When system pressure decreases, the spring-loaded poppet in the pilot section reseats. Fluid trapped on top of the balanced piston or poppet forces it to close and block pump flow to tank.

A pilot-operated relief valve allows all pump flow to go to the actuators almost to its final setting. This means the valve can operate at a lower maximum pressure and it will not slow actuator speed when forces increase.

Remote pilot operation

Another capability of pilot-operated relief valves is that they can be operated remotely. **Figure 9-5** shows the vent port connected to a direct-acting relief valve at a remote location for easy pressure adjustment. Because a relief valve is normally mounted at or very near the pump outlet, it can be difficult to reach. When it is necessary to change pressures on a regular basis, the setup in **Figure 9-5** works well. The vent port of the pilot-operated relief valve is connected to a direct-acting relief valve at a distance of 15 ft maximum. The pilot-operated relief valve is set for maximum pressure and the remote adjustment can set at any pressure lower than this maximum.

Fig. 9-5. Pilot-operated relief valve connected for remote control.

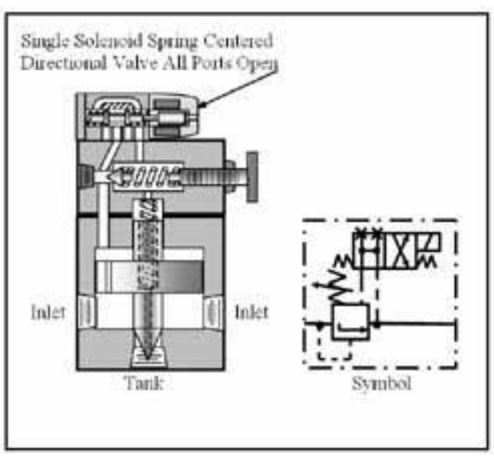


Using a 4-way directional control valve and three remote adjustments could allow electrical selection of three different pressures. Using more directional controls and more remote adjustments could give multiple pressure selections electrically.

Solenoid-operated relief valves

Figure 9-6 *shows how a directional control valve attached to the pilot section and piped to the vent port and tank can bypass or block flow from the control orifice. Bypassing the control-orifice fluid allows pump flow to unload to tank at about 20 psi. Blocking control-orifice flow forces fluid to the circuit at pressures up to relief valve setting. This is one way to keep a fixed-volume pump from overheating the fluid when it is not performing work. (See* ***Chapter 8, Figure 8-11*** *for a circuit that uses a normally open solenoid-operated relief valve to unload a fixed-volume pump in a multiple cylinder circuit.)*

Fig. 9-6. Normally open solenoid-operated relief valve.

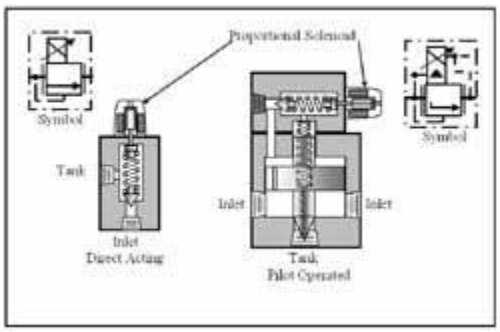


Solenoid-operated relief valves can be purchased in normally open mode (as shown), normally closed mode, and double-solenoid dual- or tri-pressure setups. (See ***Chapter 4*** *for symbols.) A solenoid-operated relief valve also can be used as a 2-way normally open or normally closed directional valve in high-flow circuits.*

Proportional-solenoid relief valves

*The relief valves in **Figure 9-7** are electronically adjusted by using a proportional solenoid instead of an adjusting knob. A proportional solenoid produces increased force with increased voltage. These solenoids usually operate at 0 to.10 V on DC current. They can produce infinitely variable force. The direct-acting type is for low (below 3 gpm) flow. It also can serve in the pilot section of high-flow pilot-operated valves. Operation of a proportional relief valve is the same as for manually controlled valves. The difference is how the force on the control poppet is generated.*

Fig. 9-7. Relief valves operated by proportional solenoid.



Unloading valves

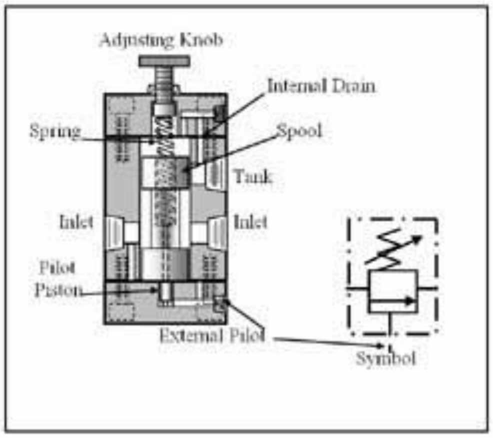
Unloading valves are pressure-control devices that are used to dump excess fluid to tank at little or no pressure. A common application is in hi-lo pump circuits where two pumps move an actuator at high speed and low pressure, the circuit then shifts to a single pump providing high pressure to perform work.

Another application is sending excess flow from the cap end of an oversize-rod cylinder to tank as the cylinder retracts. This makes it possible to use a smaller, less-expensive directional control valve, while keeping pressure drop low.

Direct-acting unloading valves

*The cutaway view in **Figure 9-8** shows the construction of a direct-acting unloading valve. The valve consists of a spool held in the closed position by a spring. The spool blocks flow from the inlet to the tank port under normal conditions. When high-pressure fluid from the pump enters at the external-pilot port, it exerts force against the pilot piston. (The small-diameter pilot piston allows the use of a long, low-force spring.) When system pressure increases to the spring setting, fluid bypasses to tank (as a relief valve would function). When pressure goes above the spring setting, the spool opens fully to dump excess fluid to tank at little or no pressure. (The example circuit in **Figure 9-10** illustrates this function.)*

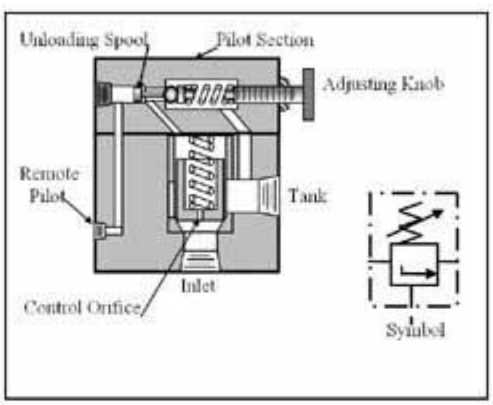
Fig. 9-8. Direct-acting unloading valve



Pilot-operated unloading valve

*The cutaway view in **Figure 9-9** shows a pilot-operated unloading valve. A pilot-operated unloading valve has less pressure override than its direct-acting counterpart, so it will not dump part of the flow prematurely. It also will go from no flow to maximum flow quickly, thus using all the flow from the high-volume pump flow for a longer period, and rapidly dropping horsepower draw from the high-volume pump.*

Fig. 9-9. Pilot-operated unloading valve.



*(This valve design is also used as an unloading relief valve in accumulator circuits. **Chapter 16** on Accumulators will have a circuit using this valve.)*

A pilot-operated unloading relief valve is the same as a pilot-operated relief valve with the addition of an unloading spool. Without the unloading spool, this valve would function just like any pilot-operated relief valve. Pressure buildup in the pilot section would open some flow to tank and unbalance the poppet, allowing it to open and relieve excess pump flow.

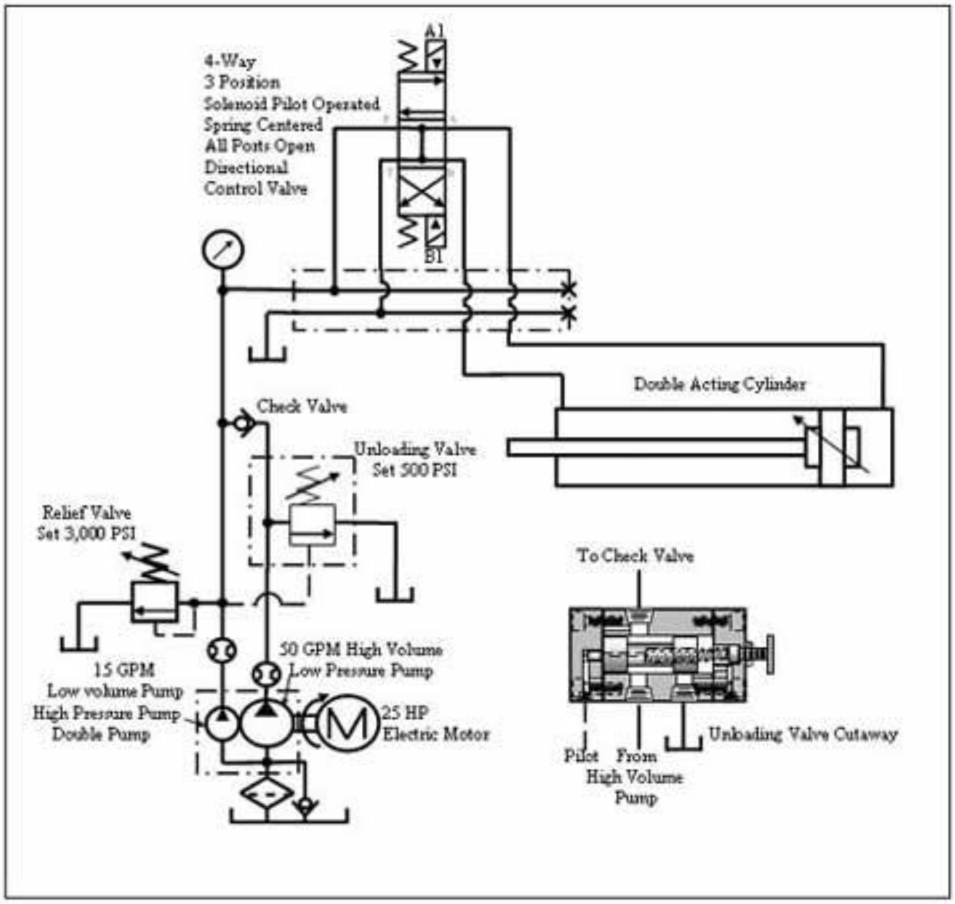
In a pilot-operated unloading valve, the unloading spool receives a signal through the remote-pilot port when pressure in the working circuit goes above its setting. At the same time, pressure on the spring-loaded ball in the pilot section starts to open it. Pressure drop on the front side of the unloading spool lowers back force and pilot pressure from the high-pressure circuit forces the spring-loaded ball completely off its seat. Now there is more flow going to tank than the control orifice can keep up with. The main poppet opens at approximately 20 psi. Now, all high-volume pump flow can go to tank at little or no pressure drop and all horsepower can go to the low volume pump to do the work. When pressure falls approximately 15% below the pressure set in the pilot section, the spring-loaded ball closes and pushes the unloading spool back for the next cycle.

An unloading valve requires no electric signals. This eliminates the need for extra persons when troubleshooting. These valves are very reliable and seldom require maintenance, adjustment or replacement.

Hi-lo pump circuit

*Often a cylinder needs very little force to stroke to and from the work -- and only a short high-force stroke to perform the work. When this is the case, the hi-lo circuit in **Figure 9-10** works well and costs less.*

Fig. 9-10. Typical hi-lo circuit using two pumps.



For example: if a single-pump circuit needs 60 gpm to make the required cycle time and 3000 psi to perform the operation, the circuit would require a 110-hp electric motor to drive it. (60 X 3000 X 0.000583 = 104 hp)

*The circuit in **Figure 9-10** is a typical hi-lo pump circuit that consumes less horsepower while maintaining fast cycle times. It uses a 25-hp motor and supporting equipment for less expense up front, as well as during its useful life. The motor drives a 50-gpm low-pressure pump and a 15-gpm high-pressure pump -- for a total of 65 gpm. The extra flow is required to maintain cycle time because the work stroke is slower. The tank, valves, and line sizes are still rated for 65-gpm flow and 3000 psi, but the electric motor and controls are much smaller.*

*As shown in **Figure 9-10**, the hi-lo circuit also has a relief valve, an unloading valve, and a check valve. The relief valve protects the low-volume/high-pressure pump from pressure above 3000 psi. The unloading valve is set at 500 psi to divert flow from the high-volume/low-pressure pump to tank when system pressure climbs above this setting. A check valve after the high-volume/low-pressure pump isolates system pressure from the unloading valve circuit while performing work at maximum pressure.*

A 4-way, 3-position, solenoid pilot-operated, spring-centered, all-ports-open directional control valve sends all pump flow to tank while the system is idle. This power unit and valve arrangement send a double-acting cylinder through a fast-approach, high-force work stroke and fast return – driven by a 25-hp electric motor. The unloading valve cutaway view shows the pipe connections to this in-line mounted valve.

Energizing solenoid A1 on the directional valve sends flow from both pumps to the cap end of the double-acting cylinder. The cylinder advances rapidly at low pressure until it contacts work. At this point, contact pressure builds quickly and when it passes 500 psi, the unloading valve is forced open. Now, all high-volume pump flow is diverted to tank at very low pressure (and horsepower). Up to this point, the highest horsepower draw would be: (65 gpm)(500 psi)(0.000583) = 19 hp.

With the high-volume pump unloaded, there is plenty of horsepower to raise the high-pressure pump to the 3000-psi pressure required to do the work. The work requires (15 gpm)(3000 psi)(0.000583) = 26 hp. This is well within the capability of the 25-hp motor specified.

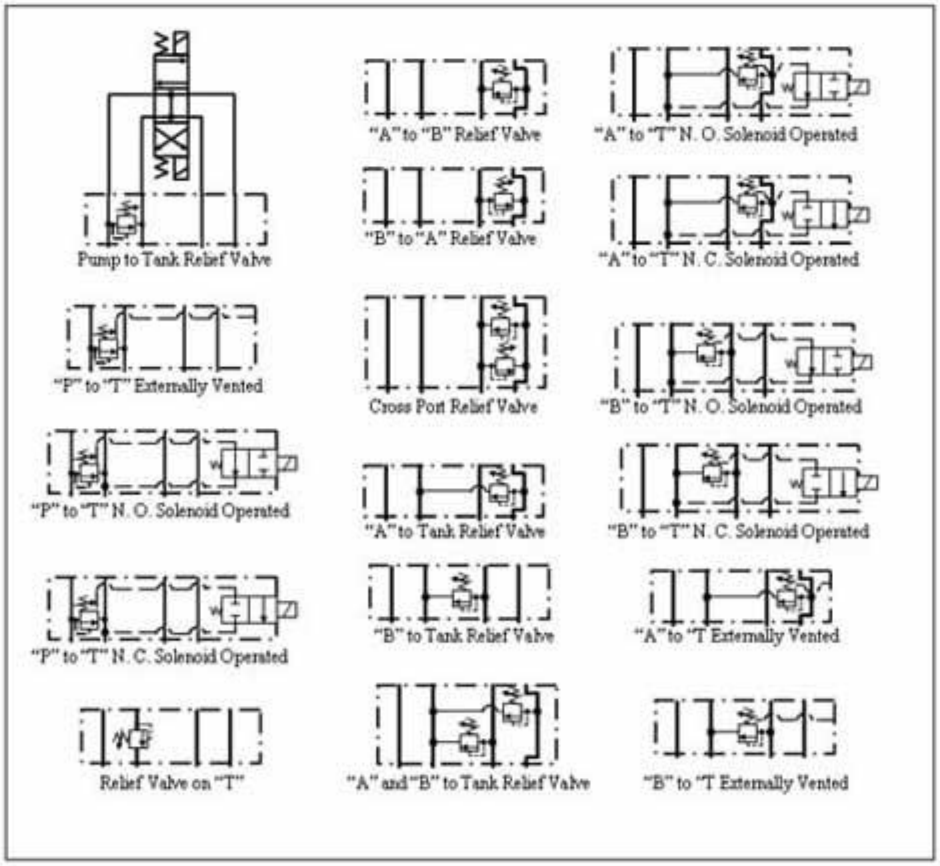
A hi-lo circuit makes it possible to replace a high-horsepower motor and its control components with a much smaller less-expensive setup.

Other applications for relief valves

Relief valves are used in circuits to protect components from excess pressure due to heat or external forces where pressure buildup in a blocked flow circuit could damage an actuator or be a safety hazard.

In hydraulic motor circuits, relief valves can eliminate shock when the motor must be decelerated quickly. In this function, fluid is ported from the high-pressure outlet port of the motor to the low-pressure inlet port, while holding ample backpressure to stop the motor without damage.

Fig. 9-11. Symbols for modular relief valves. (Note that these symbols do not show X and Y ports for solenoid pilot-operated valves.)



Most relief valve functions are available as modular or sandwich valves that mount between the directional control valve and sub-plate. ***Figure 9-11*** *shows most of the common configurations presently offered by fluid power suppliers. These modules are usually available in all valve sizes up to D08 (3/4 in.) ports.*

Always use a relief valve with fixed-displacement hydraulic pumps. Pressure-compensated pump circuits also may use a relief valve for certain applications.

Think of a relief valve in a hydraulic system as a fuse or circuit breaker in an electric circuit. An electric circuit never blows a fuse unless it overloads. When an electric circuit overloads, it is inoperable until reset. Usually the person responsible for resetting the fuse looks for the reason it blew and fixes the problem before restarting the machine. Many hydraulic circuits allow the relief valve to dump some or all pump flow to tank all or part of the time. The extra power to produce that unused flow is expensive. Also, heat generation from excess flow requires larger heat exchangers that are expensive to buy and operate.

Protecting the pump and the system from excess pressure is the only valid function for a relief valve. At no time should the relief valve be used to pass excess pressure fluid to tank. When excess pump flow goes to tank, it generates heat. The relief valve in a well-designed hydraulic circuit never relieves oil to tank — unless there is a circuit or control malfunction.

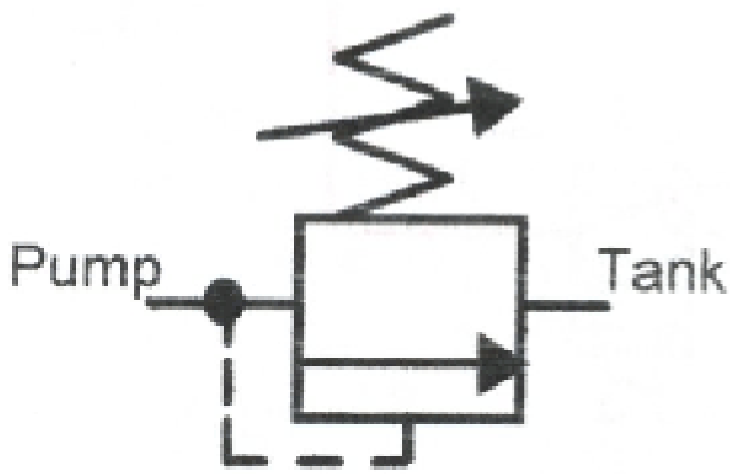


Fig. 18-1. Direct-acting relief valve.

Figure 18-1 pictures the symbol for a direct-acting relief valve. A direct-acting relief valve responds quickly when pressure tries to go above the valve's setting. It can be use it on circuits with pressure-compensated pumps to reduce pressure spikes. On a hydraulic circuit with a fixed-displacement pump, a direct-acting relief valve opens partially early and thus wastes energy. When the system must operate near maximum pressure without any fluid bypass, use a pilot-operated relief valve.

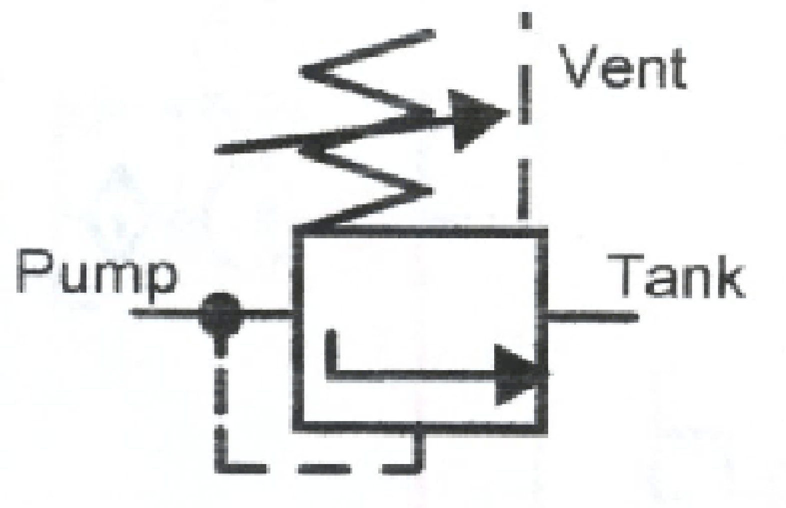


Fig 18-2. Simplified symbol for a pilot-operated relief valve.

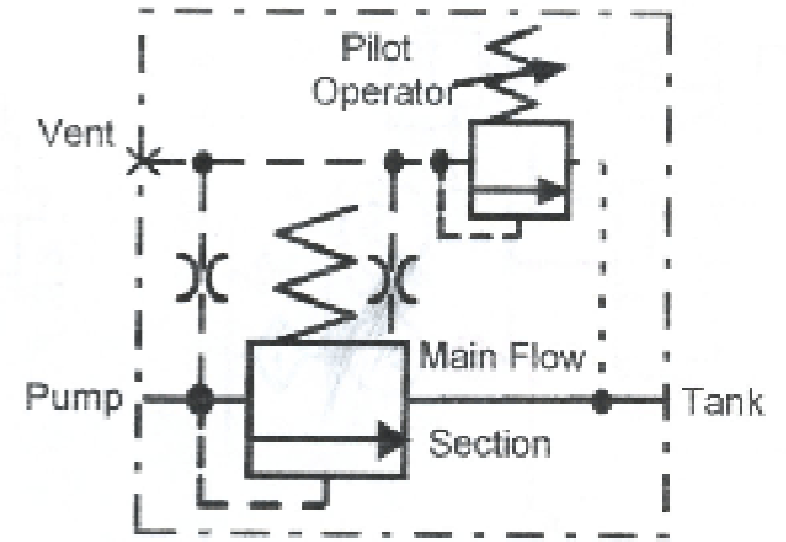


Fig. 18-3. Complete symbol for a pilot-operated relief valve.

Figures 18-2 and -3 show the simple and complete symbols for a pilot-operated (or compound) relief valve. This type relief valve has two sections. The pilot operator on top is a small, poppet-type direct-acting relief valve. The main flow section of the valve is a poppet- or piston-type, normally closed 2-way valve. Through internal porting, a small direct-acting relief poppet controls a large poppet or piston. A pilot-operated relief valve responds more slowly, but does not even partially open until system pressure reaches approximately 95% of set pressure. Pilot-operated relief valves are suitable for remote operation, they open to unload pumps at pressures below 50 psi, and they act as large 2-way valves in some circuits.

Examples of relief-valve circuits
Always locate the relief valve as close as possible to the outlet of a fixed-displacement pump. A

pilot-operated relief works best because it does not pass any fluid until system pressure is very near the valve's set pressure.

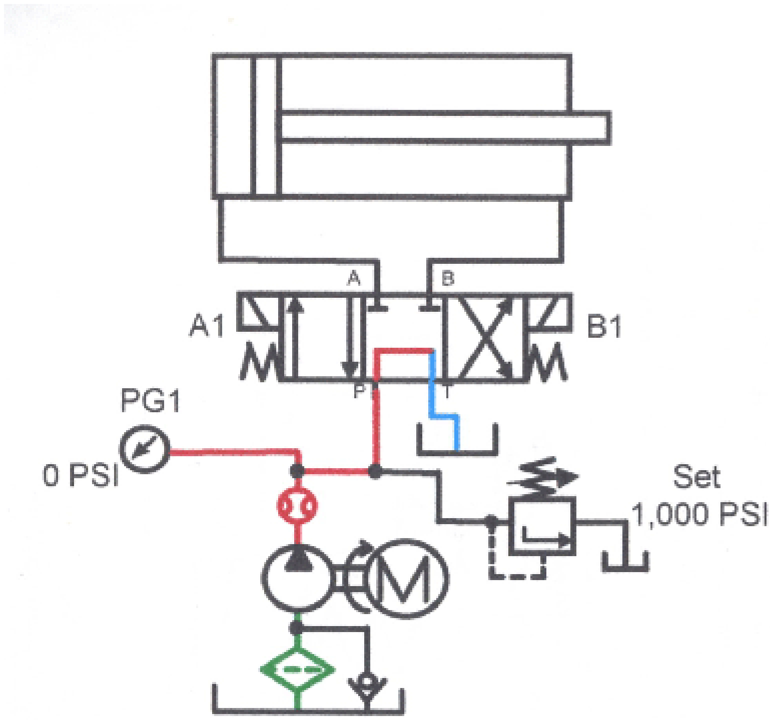


Fig. 18-4. Fixed-displacement pump circuit with relief valve.

Figure 18-4 shows a typical fixed-displacement pump circuit. Except in the event of a control circuit malfunction or if it is used to hold the cylinder at pressure, the relief valve never opens. Heat generation is minimal and the circuit usually can run without a heat exchanger.

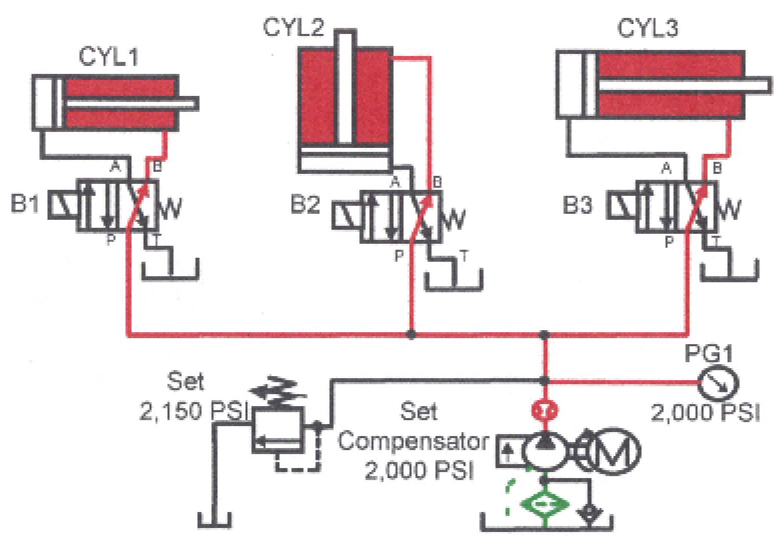


Fig. 18-4. Fixed-displacement pump circuit with relief valve.

Figure 18-5 shows a pressure-compensated pump with a direct-acting relief valve to protect it against overpressure. Pressure spikes often occur in pressure-compensated pump circuits with high flow or fast cycling. When the pump must compensate rapidly or often from full flow to no flow, the resulting overpressure drastically shortens pump life.

In Figure 18-5, the pump would be at low pressure and full flow when cylinder <I>CYL3 </I> extends rapidly. When the cylinder stops, fluid requirement is zero, but pump flow is still 40 gpm. As pressure builds, the pump finally starts compensating at about 1900 or 1950 psi. It is still producing 40 gpm — with no place for the oil to go. Without a relief valve in the circuit, system pressure spikes during each cycle can reach four to ten times the compensator setting. Pressure spikes damage the pump and piping after a few hours of operation. The faster the cycle, the more quickly shock damage from pressure spikes causes problems.

A relief valve, installed in Figure 18-5, reduces pressure spikes to protect the system. When the pump shifts to no flow, excess flow goes to tank through the relief valve. When the pump reaches compensator pressure, the relief valve closes. (For another and better way to reduce pressure spikes and protect a pressure-compensated pump from rapid cycling, see Chapter 1, Figures 17-19.)

Set the relief valve in a pressure-compensated pump circuit at 150 to 200 psi higher than the pump compensator. With relief pressure below compensator setting, pump flow goes to tank and makes heat. With relief pressure set at compensator pressure, the relief valve starts dumping when the pump starts compensating. When the relief valve passes fluid, the pump sees a pressure drop, and starts flowing again. The resultant pressure drop allows the relief valve to close and the dump/flow cycle starts again. After a few hours of this erratic operation, the pump fails.

Adding a solenoid valve to the vent port of a pilot-operated relief valve makes an effective unloading valve. Figure 18-6 shows a fixed-displacement pump supplying three cylinders. There is no power to the solenoid on the relief valve with the cylinders idle, so pump flow goes to tank at low pressure. Energizing a solenoid on the relief valve and one cylinder's directional valve causes an action. Energizing both solenoids at the same time sends pump flow to the cylinder until reaching maximum relief pressure. A solenoid relief valve always has a slight delay before blocking flow to tank after energizing the solenoid. The delay is in milliseconds so it usually is only noticeable on very fast cycles.

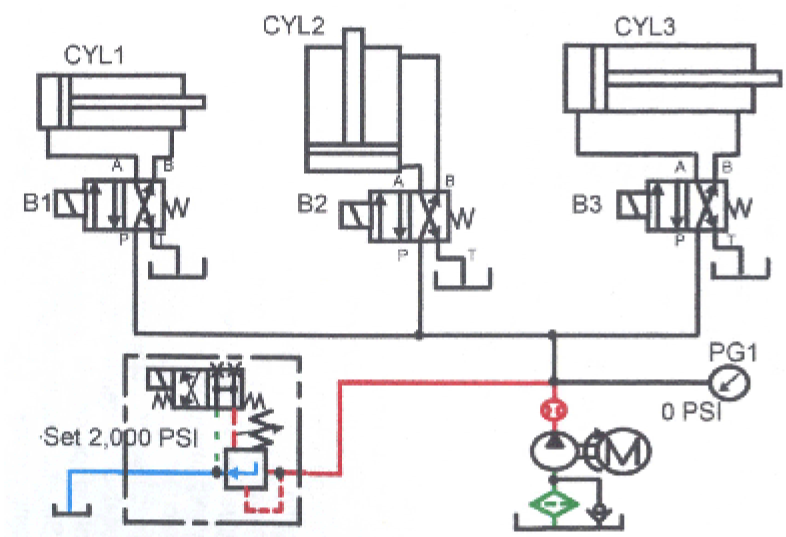


Fig. 18-6. Fixed-displacement pump unloading circuit using a normally open solenoid relief valve.

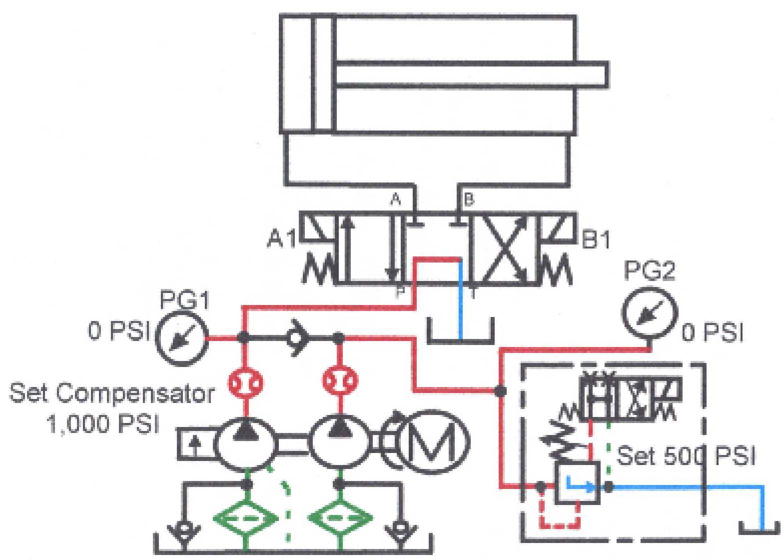


Fig. 18-7. Hi-lo pump circuit in which a solenoid relief valve unloads the high-volume pump.

The circuit in Figure 18-7 uses a solenoid-operated relief valve to unload the high-volume pump in a hi-lo circuit. Instead of waiting for pressure to build before the high-volume pump dumps to tank, the solenoid relief dumps oil on demand. The demand signal could come from a pressure switch, a limit switch, or an electric eye that senses cylinder position (then slows it before it contacts the work).

A relief valve that decelerates an actuator

Figures 18-8 through 18-14 show normally closed, solenoid-operated relief valve B used to *rapidly extend, then decelerate a free-falling cylinder. Deceleration takes place when the cylinder makes a limit switch that deenergizes the solenoid on relief valve B . Relief pressure should be set 150 to 200 psi higher than the pressure required to raise the cylinder. Any higher relief pressure shortens the deceleration stroke and increases shock.*

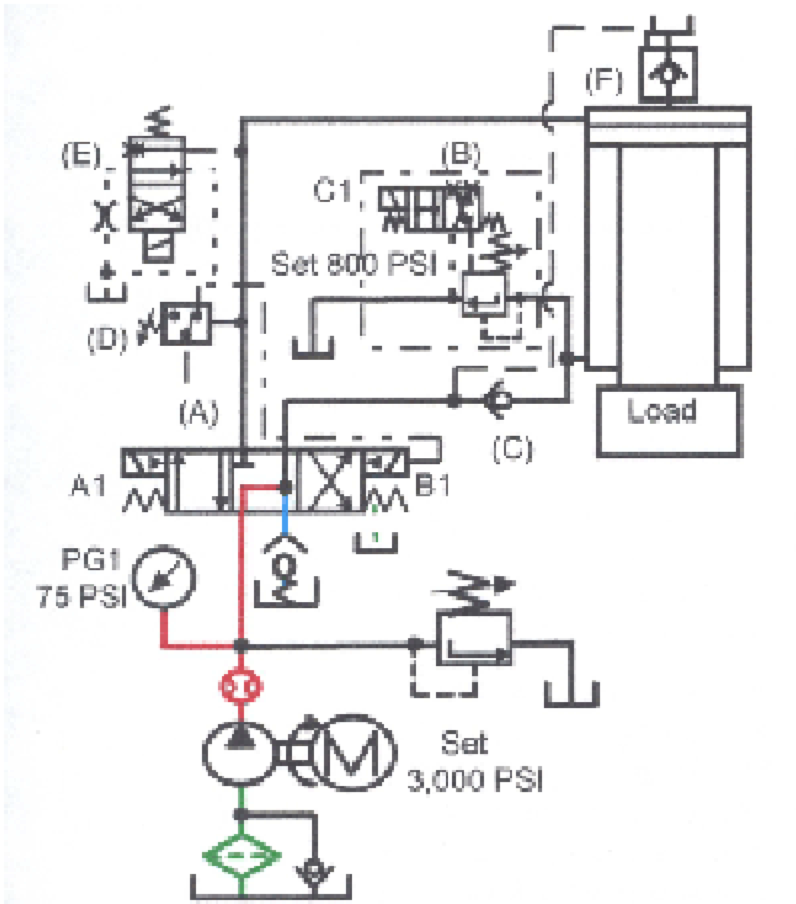


Fig. 18-8. Press circuit with NC solenoid relief valve for fast forward and deceleration to work. Shown at rest with pump running.

Figure 18-8 shows a cylinder with its rod port piped to tank through normally closed solenoid-operated relief valve B . Prefill valve F allows the cap end of the cylinder to fill during rapid advance. (See Chapter 7 for an explanation of the prefill valve's function.) Check valve C at the rod port keeps cylinder flow from going to tank through directional valve A.

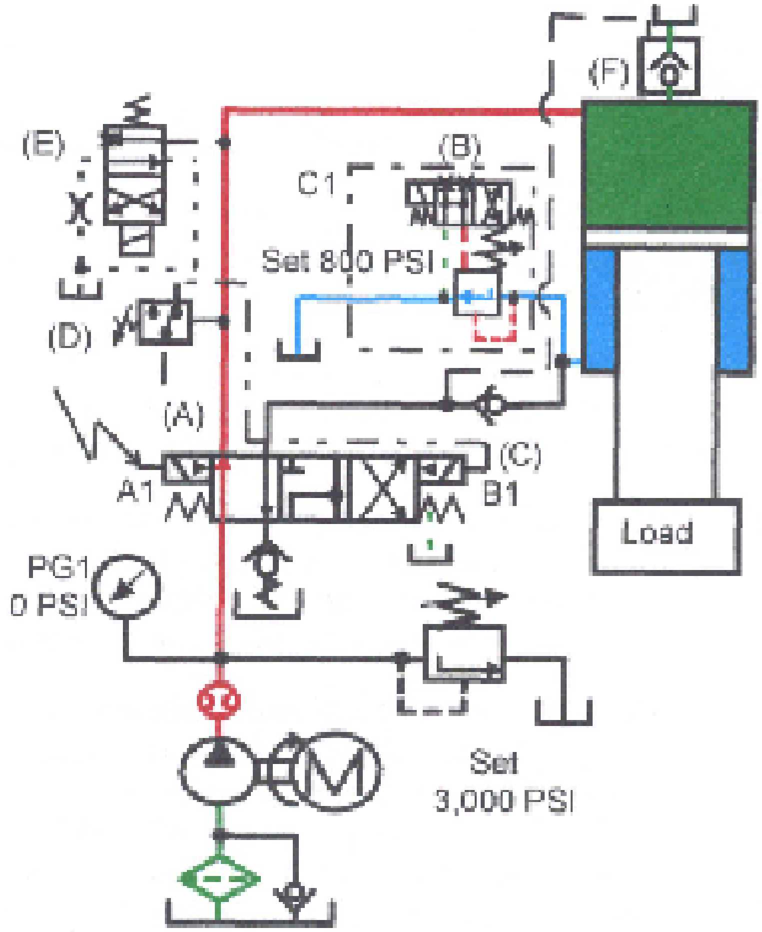


Fig. 18-9. Press circuit with NC solenoid relief valve for fast forward and deceleration to work. Shown with cylinder stroking fast forward.

To extend the cylinder, energize solenoid A1 on directional valve A to pass oil to the cylinder's cap end, as in Figure 18-9. Also energize solenoid C1 on relief valve B , venting it to tank and allowing the cylinder to fall freely. As the cylinder falls, the cap end fills from the pump and from tank directly through prefill valve F.

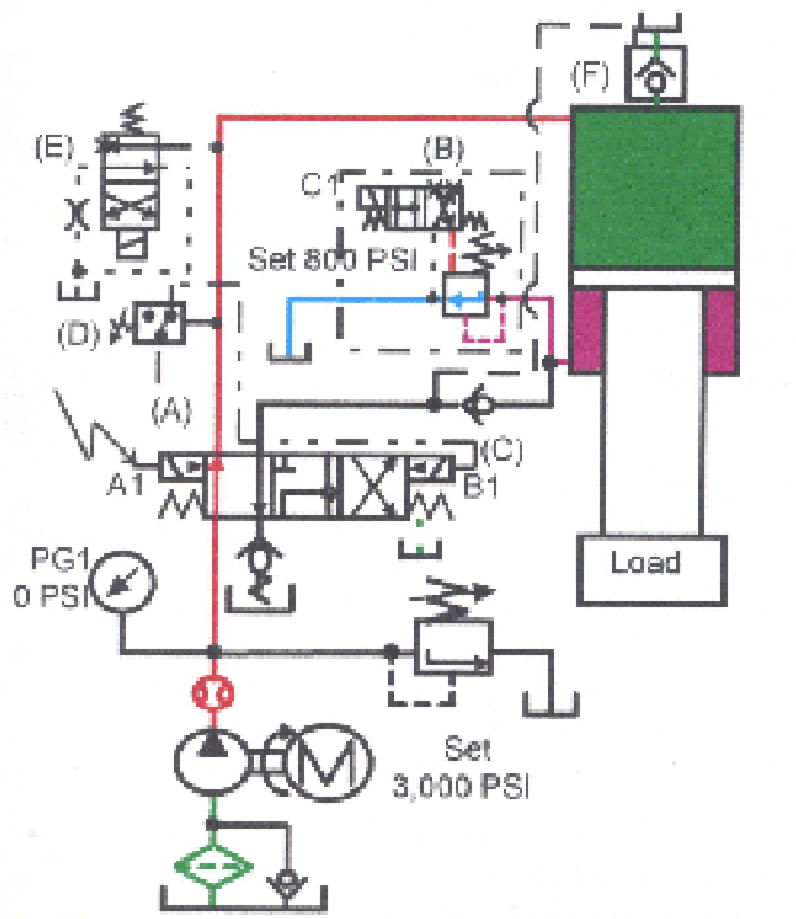


Fig. 18-10. Press circuit with NC solenoid relief valve for fast forward and deceleration to work. Shown with cylinder decelerating.

As the cylinder extends, high flow leaving the cylinder's rod end goes to tank. Just before the rod contacts the work, a limit switch deenergizes solenoid C1 on relief valve B , Figure 18-10. As valve B tries to close, pressure increases in the cylinder's rod end, keeping the valve partially open. Backpressure from relief valve B quickly and smoothly slows cylinder descent. The cylinder continues to slow while the relief valve shuts. The cylinder does not completely stop because the pump forces it to extend after free fall.

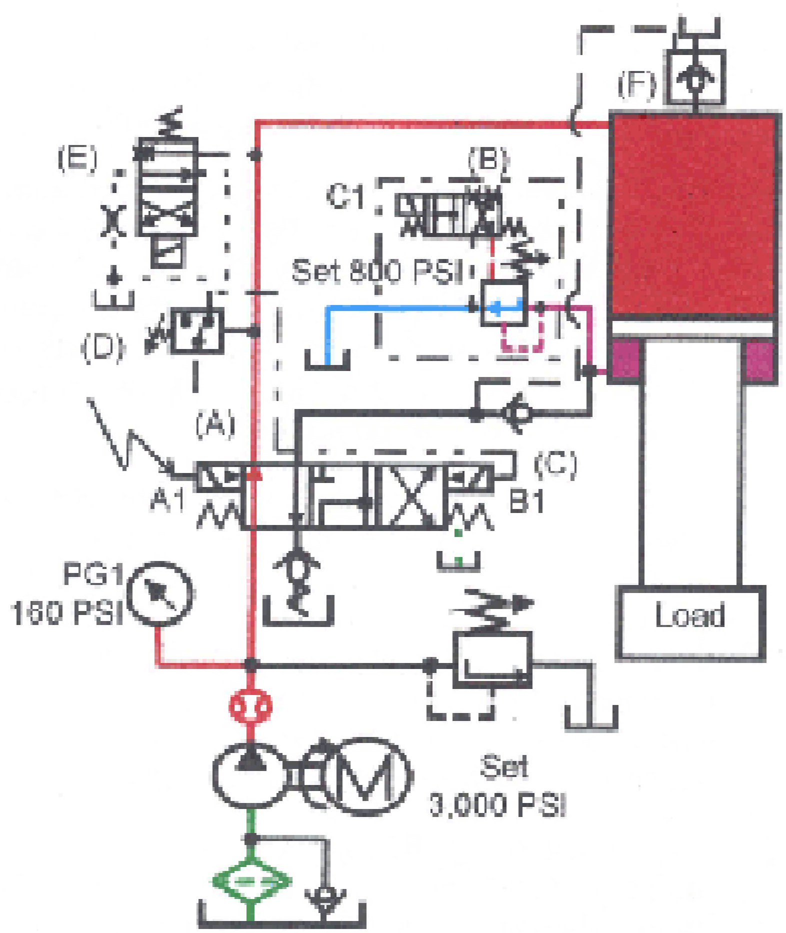


Fig. 18-11. Press circuit with NC solenoid relief valve for fast forward and deceleration to work. Shown with cylinder approaching work.

After deceleration, relief valve B acts as a counterbalance valve, as in Figure 18-11, so the load cannot run away. The cylinder extends at pressing speed to the work. This part of the stroke should be as short as possible to save time. Prefill valve F closes as the cylinder decelerates and allows pressure to build in the cap end. The slowdown is smooth and controlled — without shock or bouncing. This circuit decelerates the cylinder when commanded by an electrical signal at any point in its stroke.

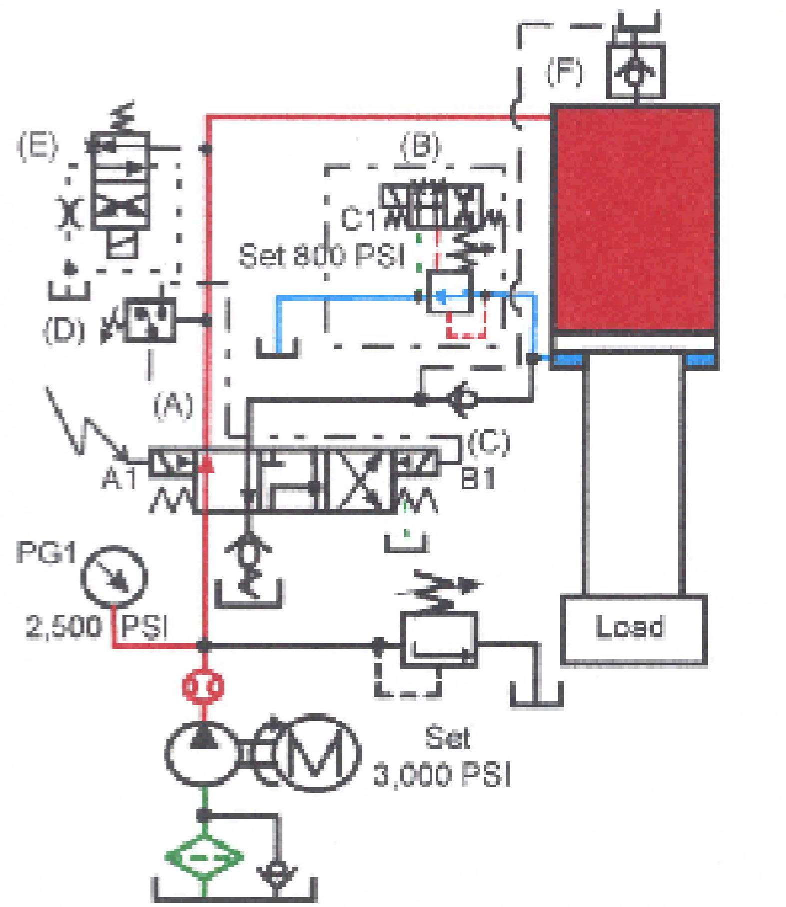


Fig. 18-12. Press circuit with NC solenoid relief valve for fast forward and deceleration to work. Shown with cylinder pressing.

Figure 18-12 shows the circuit while the cylinder is pressing. When the cylinder contacts the work, energize solenoid C1 on relief valve B again. Energizing the solenoid on the relief valve lets oil from the cylinder's rod end flow to tank at minimal pressure. This allows the weight of the platen and tooling to add to the pressing force because they are no longer counterbalanced. Pressure increases in the cylinder's cap end to perform the work.

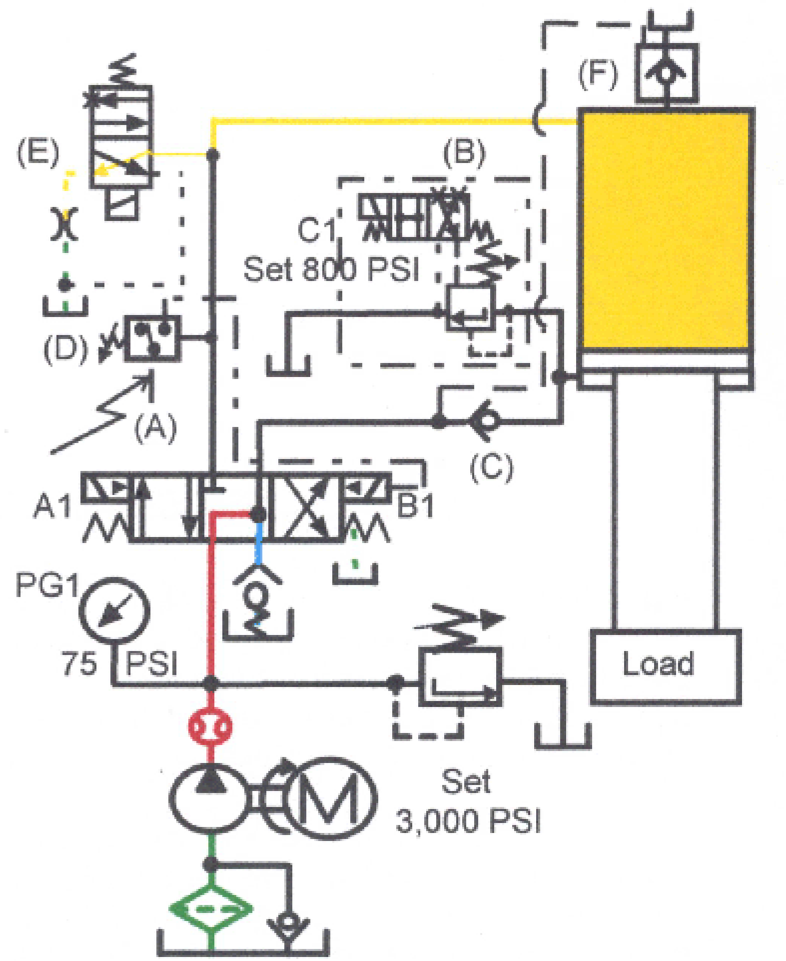


Fig. 18-13 Press circuit with NC solenoid relief valve for fast forward and deceleration to work. Shown with cylinder decompressing.

Deenergizing solenoid A1 on directional valve A lets it center and decompress the cylinder, Figure 18-13 shows directional valve A centered, blocking the cylinder's cap-end port and unloading the pump. At the same time, a signal to single-solenoid valve E in the cap-end line shifts it open. Trapped pressurized oil in the cylinder's cap end flows to tank through an orifice, thus lowering pressure without shock. Pressure switch D indicates when pressure is low enough to shift valve A to retract the cylinder. (See Chapter 7 for an explanation of a decompression circuit. A decompression circuit keeps the cylinder from rapidly losing pressure and shocking the system.)

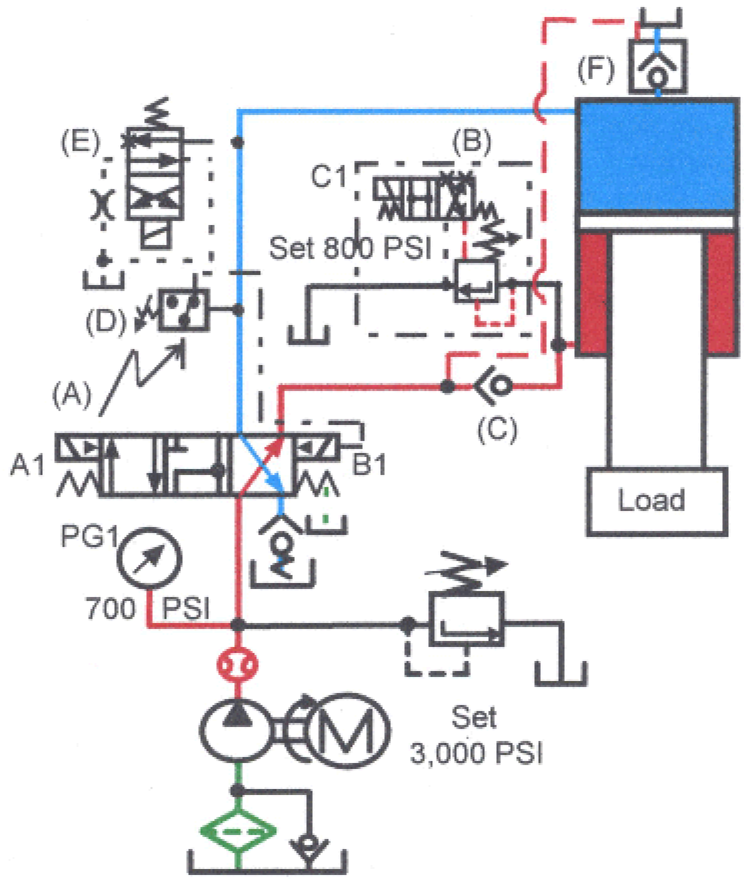


Fig. 18-14. Press circuit with NC solenoid relief valve for fast forward and deceleration to work. Shown with cylinder retracting.

To retract the cylinder, energize solenoid B1 on directional valve A to send oil to the cylinder's rod end, as in Figure 18-14. Oil from the pump starts to retract the cylinder. Pilot oil opens prefill valve F to tank. Oil from the cylinder's cap end flows to tank through the prefill valve and the main directional control valve. The cylinder retracts rapidly at low pressure.

Using solenoid relief valves as 2-way valves

High-flow (above 50 gpm) 2-way valves are not readily available for hydraulic circuits. To get around this problem, use a solenoid relief valve. Several circuits shown here are in use in many hydraulic applications.

For flows above 150 to 200 gpm, use slip-in cartridge valves (as explained in Chapter 4). Slip-in cartridge valves use simple directional controls to operate large poppets that can handle flows in excess of 600 gpm.

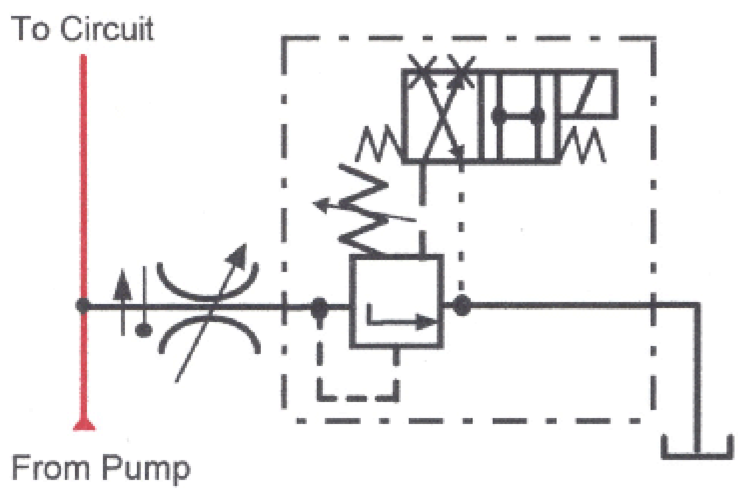


Fig. 18-15. NC solenoid relief valve used for high-flow bleed-off flow-control circuit. Shown at rest with pump running.

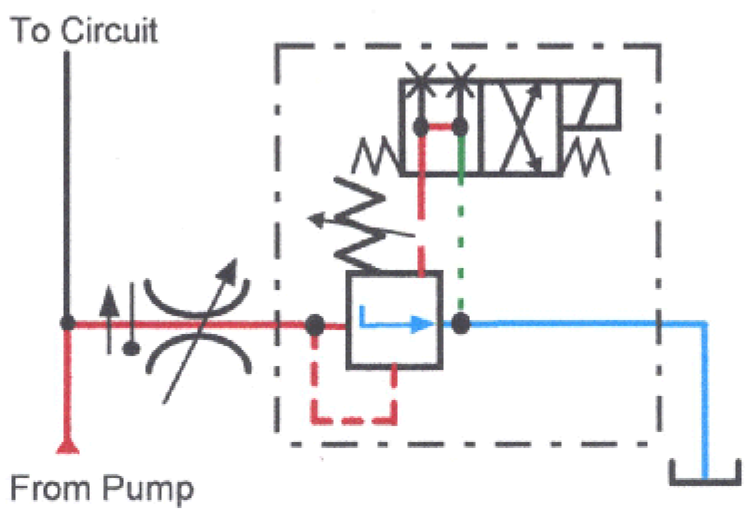


Fig. 18-16. NO solenoid relief valve used for high-flow bleed-off flow-control circuit. Shown at rest with pump running.

Figures 18-15 through 18-16 show the schematic diagram for a high-flow, on-off, bleed-off flow control circuit. Set the solenoid relief valve higher than system pressure so it never passes fluid unless vented. The normally closed bleed-off circuit shown in Figure 18-15 passes fluid when the solenoid is energized. Figure 18-16 shows a normally open solenoid-operated relief valve. With this valve, energizing the solenoid stops flow.

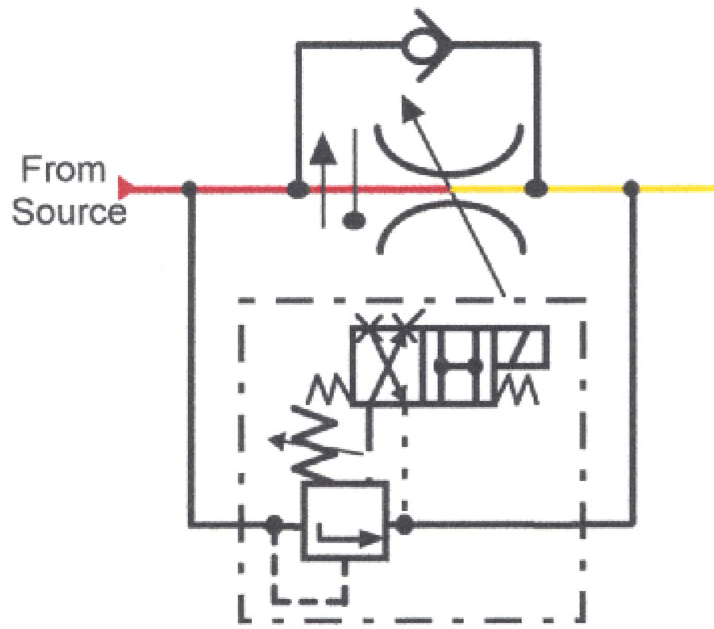


Fig. 18-17. NC solenoid relief valve used for high-flow bleed-off flow-control circuit. Shown at rest with pump running.

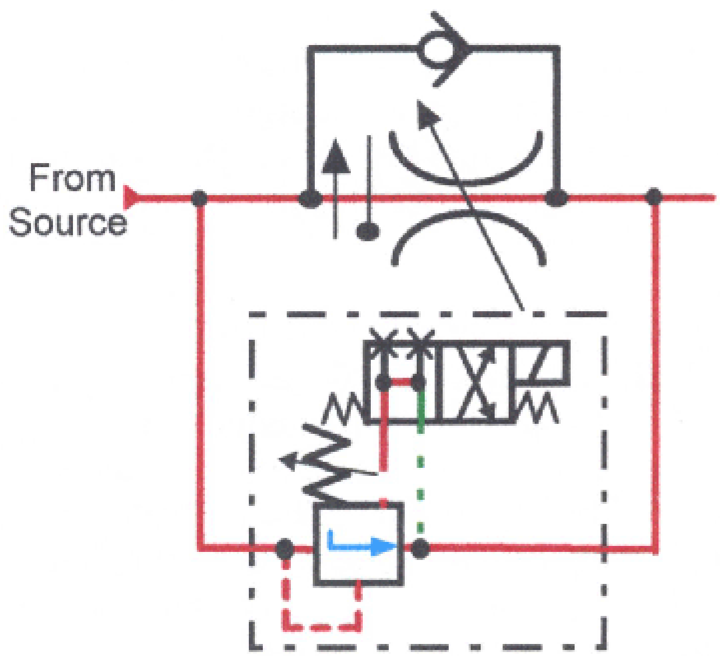
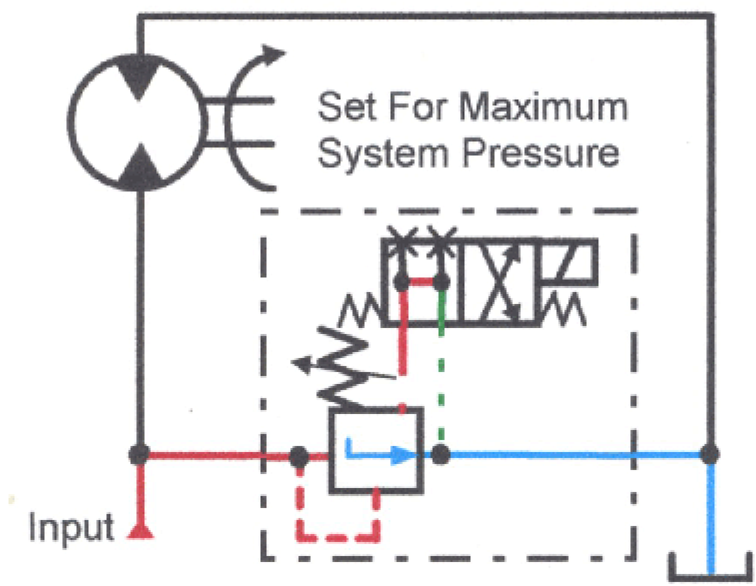


Fig. 18-18. NO solenoid relief valve used for high-flow bleed-off flow-control circuit. Shown at rest with pump running.

The circuits in Figures 18-17 and 18-18 show solenoid-operated relief valves bypassing large flow controls in a 2-speed circuit. Because there is backpressure downstream from the solenoid relief valve in this type circuit, use a valve with an external drain. (Backpressure at the outlet of a relief valve causes it to shut when internally drained.) Externally draining the relief valve eliminates backpressure at the vent port so it stays open when bypassing.

Solenoid reliefs used as shut-off valves do not cause as much shock as spool-type valves because relief valves cushion as they close.



ig. 18-19 NO solenoid relief valve for starting and stopping a large hydraulic motor. Shown at rest with pump running.

In Figure 18-19, a normally open solenoid-operated relief valve protects a large hydraulic motor from overpressure, and also starts and stops it for a single-rotation application. Energizing the solenoid on the relief valve blocks its vent, causing it to close. Closing is smooth because pressure builds to relief setting quickly, providing some fluid a path to tank while the motor comes up to speed. When the motor reaches full speed, the relief valve closes completely. The motor then continues at full speed at whatever pressure it takes to keep it rotating.

Deenergizing the solenoid on the solenoid-operated relief valve connects pump flow to tank and the hydraulic motor coasts to a stop. A brake valve ***(Chapter 12)*** *would stop the motor quickly and smoothly if required.*

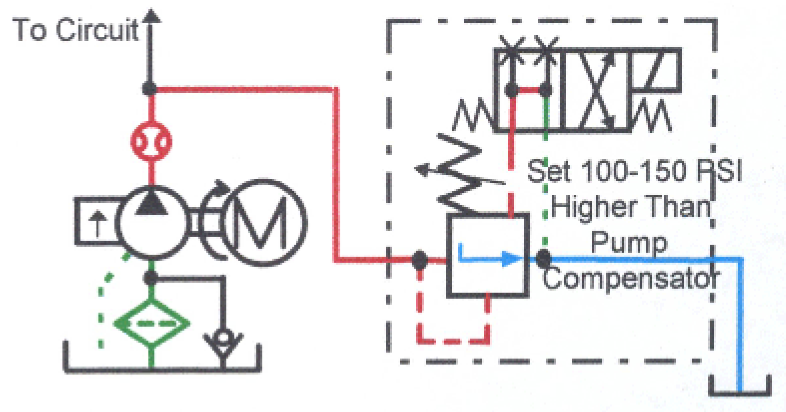


Fig. 18-20. NO solenoid relief valve for starting a pressure-compensated pump at no load. Shown with pump just starting.

Figure 18-20 shows a normally open solenoid-operated relief valve that allows a large pressure-compensated pump to start at no load. A normally open solenoid-operated relief valve lets flow from the pressure-compensated pump go to tank until the electric drive motor is up to speed. A time delay or a flow meter with a flow switch energizes the solenoid on the relief valve to load the circuit. Deenergizing the normally open solenoid-operated relief valve unloads the pump any time to reduce power consumption, heat buildup, and noise.

Controlling a pilot-operated relief valve remotely

The system relief valve is normally located near the pump outlet on a typical hydraulic unit. The hydraulic unit could be at a distance from the operator, or cramped conditions could make the relief valve hard to get near. If an application's relief pressure must change often, add a remote relief valve control to a pilot-operated relief valve for convenience.

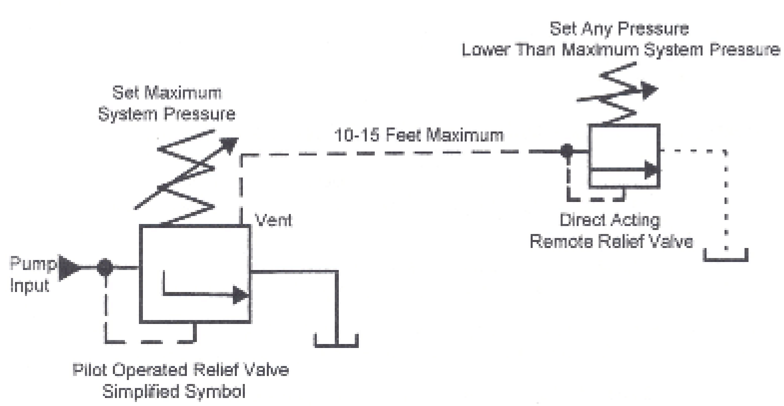


Fig. 18-21. Simplified schematic of remote-operated relief valve.

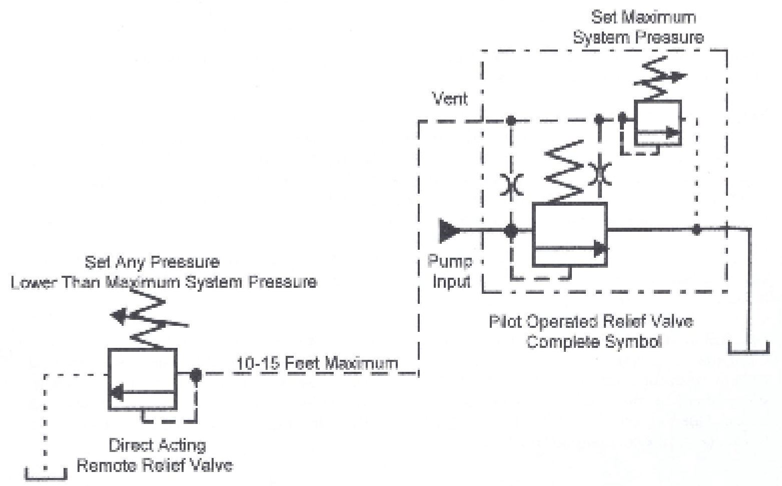


Fig. 18-22. Complete symbol for a remotely controlled pilot-operated relief valve.

Figures 18-21 and 18-22 show a symbol for a remote relief valve setup. Figure 18-21 shows the simplified symbol; Figure 18-22 shows the complete symbol. All pilot-operated relief valves have a vent port. The vent port tees into the pilot line that connects system pressure to the direct-acting relief valve's pilot section. The vent port tees in after the control orifice. With the vent blocked, the relief valve functions normally. With the vent open to atmosphere, the relief valve opens at the pressure of the internal main poppet or piston spring. This is usually between 20 and 70 psi. Figure 18-21 shows a small, direct-acting relief valve piped to the vent port of a pilot-operated relief valve. The small direct-acting relief functions the same as the pilot-valve section on the main relief. An operator can use the remote relief to adjust main system pressure from any convenient location within 10 to15 feet of the system relief valve.

To adjust a circuit with a remote relief valve, use the following procedure. First, set the main relief at minimum pressure and the remote relief at maximum pressure. Start the pump and check for obvious leaks and incorrect plumbing. Pressure is low during this part of the procedure. Next, slowly raise the main relief to maximum system pressure and lock it. Now, use the remote relief to set pressure to any setting less than the main relief. The operator can only adjust pressure to a level lower than the main relief setting. This is an important safety factor because it eliminates damage or injury from excess pressure caused by an inexperienced operator.

Most manufacturers recommend that the remote valve be located a maximum of 10 to 15 feet from the main relief. The greater the distance between the remote and main relief, the longer the response time of the main relief. An increase in response time allows higher pressure override, causing pressure spikes. Pressure spikes may cause premature pump, piping, or valve failure.

With a solenoid or manual valve to select more than one remote relief, it is easy to select multiple preset pressures.

Multi-pressure relief valves

Pilot-operated relief valves have a vent port. In Figures 18-21 and 18-22, the vent port is piped to a single, remote direct-acting relief for adjusting pressure remotely. Figures 18-23 through 18-25 show the vent port connected to directional valves and multiple remote reliefs. These circuits allow changes from maximum pressure to several preset or infinitely variable limits during a cycle.

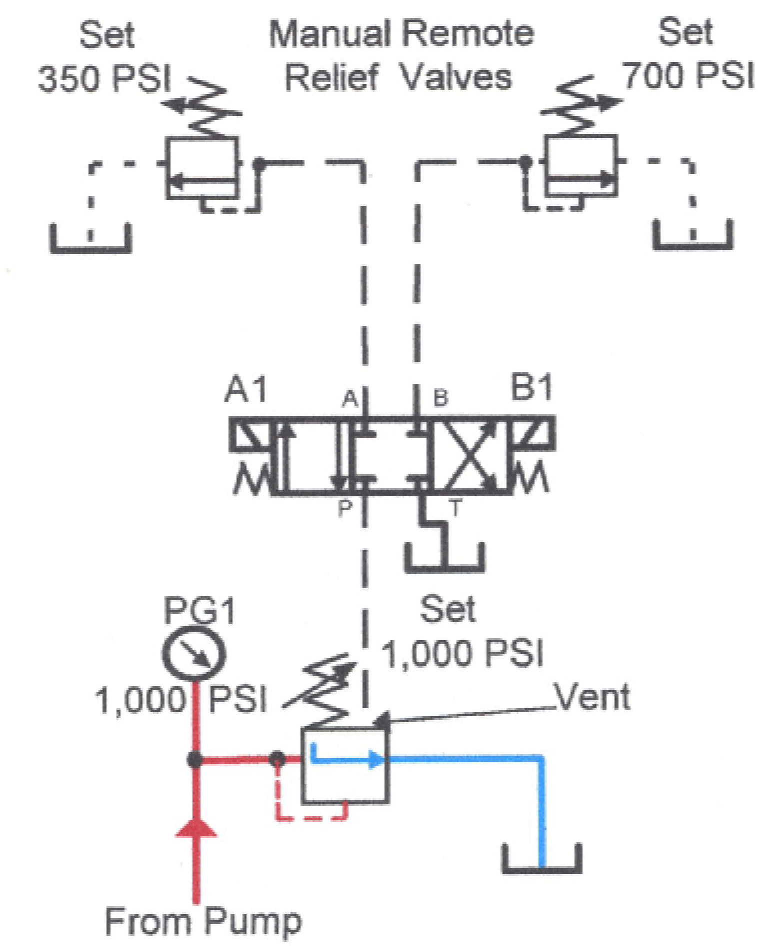


Fig. 18-23. Using the vent port for three different solenoid-selected system pressures. Shown with no solenoids energized.

Figure 18-23 shows a pilot-operated relief valve with the vent port connected to a 3-position directional control valve. With the directional valve centered, it blocks the vent port on the relief to keep system pressure at the setting of the main relief. An open-center directional valve would vent the main relief, lowering pressure to a 20- to 70-psi range.

Some manufacturers offer a relief valve with the remote pilot heads and solenoid valve built into the valve body. This eliminates external piping but is less flexible than piping the vent of a standard pilot-operated relief valve to standard directional valves.

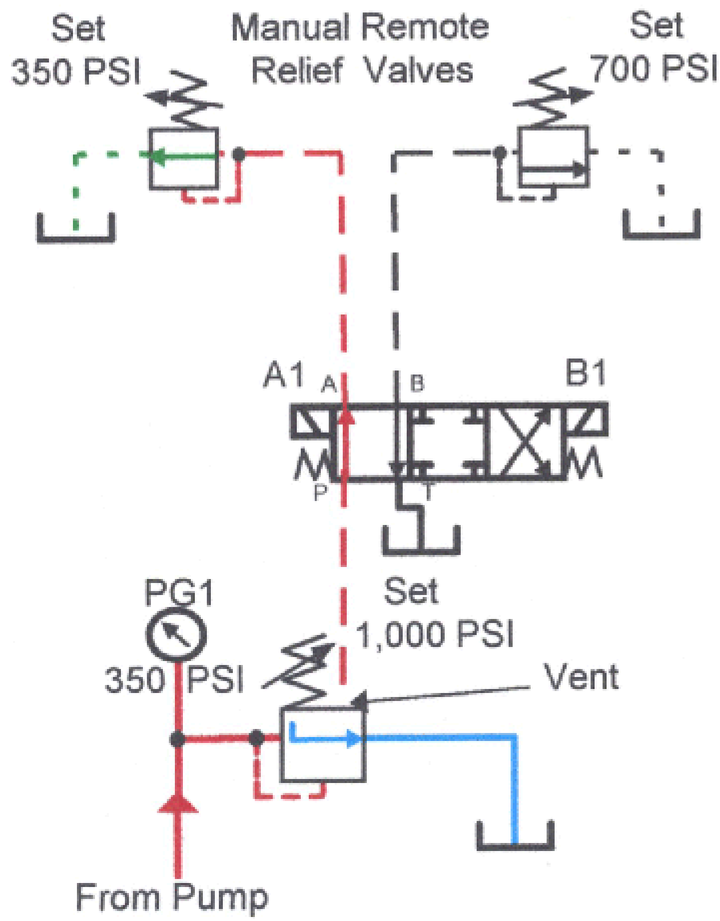


Fig. 18-24. Using the vent port for three different solenoid-selected system pressures. Shown with solenoid A1 energized.

In Figure 18-24, solenoid A1 is energized. This connects the vent to the left remote direct-acting relief, dropping system pressure to a maximum of 350 psi. Energizing solenoid A1 keeps pressure from going above the setting of the left direct-acting relief. The main relief valve always limits maximum system pressure.

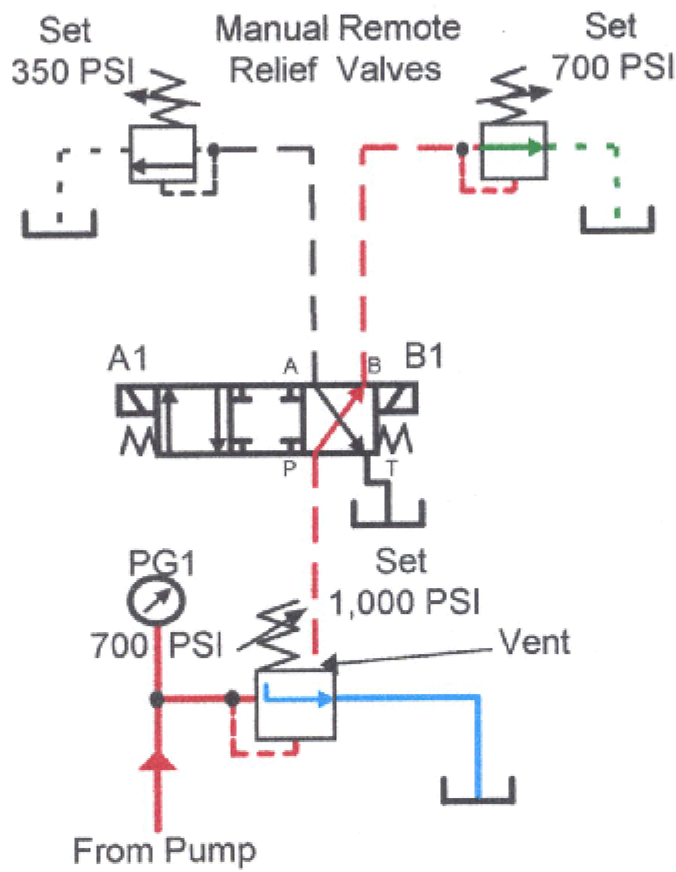
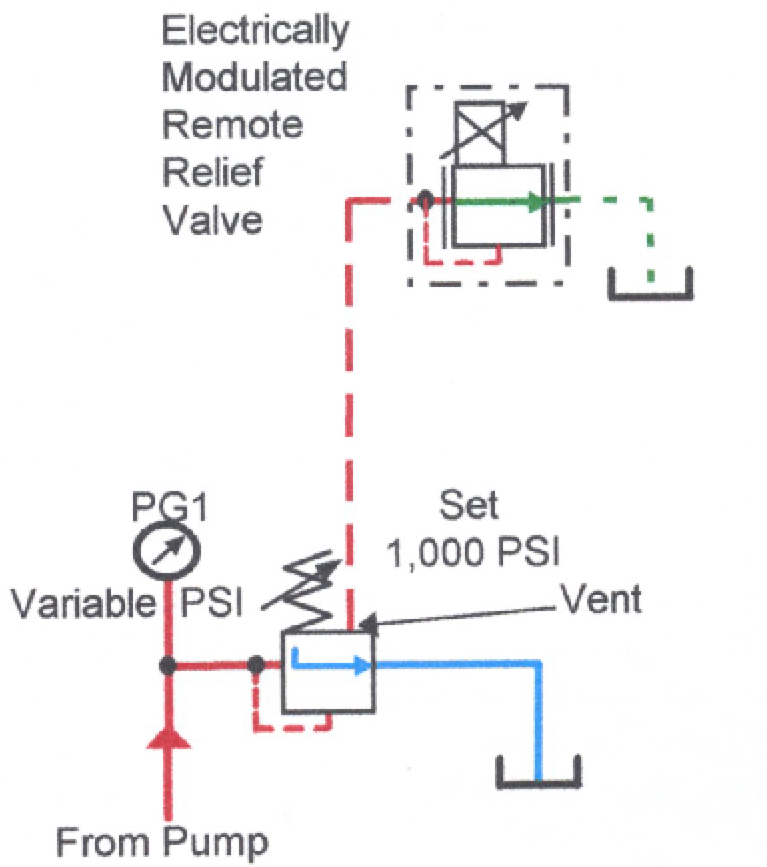


Fig. 18-25 Using the vent port for three different solenoid-selected system pressures. Shown with solenoid B1 energized.

Figure 18-25 shows solenoid A1 energized, allowing system pressure to go to 700 psi. In this condition the right remote direct-acting relief controls system pressure. Set remote reliefs at any pressure lower than the main relief valve.



18-26. Using the vent port with a servo pressure controller for infinitely variable pressure.

Figure 18-26 shows a pilot-operated relief controlled by an infinitely variable proportional or servovalve. Using a variable-flow valve to control a pilot-operated relief valve gives infinitely variable pressure. The control signal may come from a rheostat, a programmable controller, or a computer.

Purchase an infinitely variable relief valve as an assembled unit or pipe one remotely. In each case, the pilot head relief controls maximum pressure, while the servo or proportional valve only sets a lower pressure.

Unloading relief valves
An accumulator circuit using a fixed-displacement pump must have some way to unload the pump after reaching maximum pressure. A normally open solenoid-operated relief valve controlled by a pressure switch is one way to unload a pump. ***Chapter 1 shows this circuit and explains its operation.***

Some accumulator circuits use a special type valve called an unloading relief valve. This relief valve eliminates the need for electrical, high and low pressure switches and a solenoid-operated dump valve to unload the pump. Only a few manufacturers make an unloading relief valve. Two of these operate at preset pressure differentials and may not be suitable for some accumulator circuits. One manufacture makes an unloading relief valve with adjustable pressure differentials.

Several companies make a relief unload-and-dump valve combination with other features. Operation is the same as an unloading relief valve but includes a back-flow check valve and an accumulator dump valve in the same body. ***See Chapter 1, Figure 44, for an explanation of this accumulator unload and dump valve.***

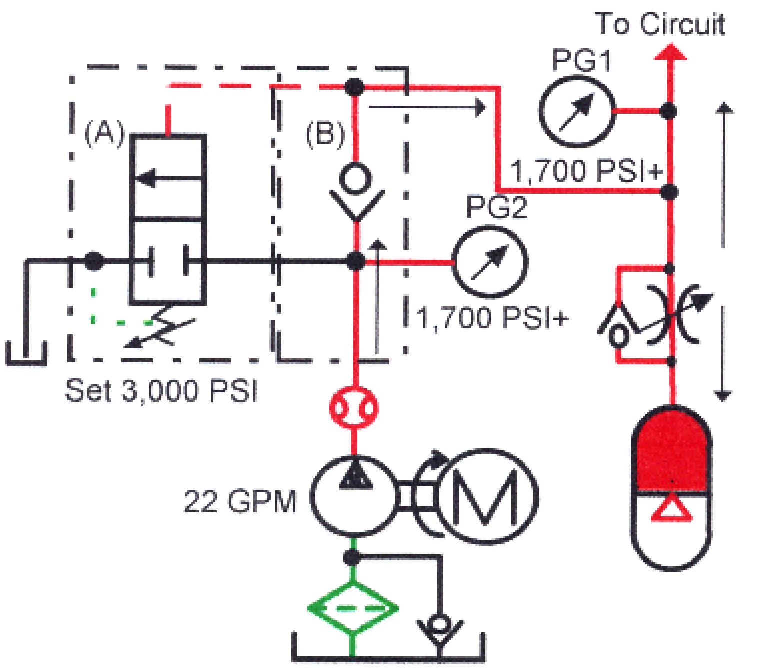


Fig. 18-27. Unloading relief valve in an accumulator circuit.
Shown with pump just turned on.

Figures 18-27 through 18-30 schematically depict an unloading relief valve in an accumulator circuit. Figure 18-27 shows the circuit after the pump starts. Normally closed relief valve A forces fluid to the accumulator and the circuit. Pressure increases as fast as the pump fills the accumulator. When the accumulator and circuit reach set pressure of 3000 psi, pilot pressure opens relief valve A and unloads the pump to tank.

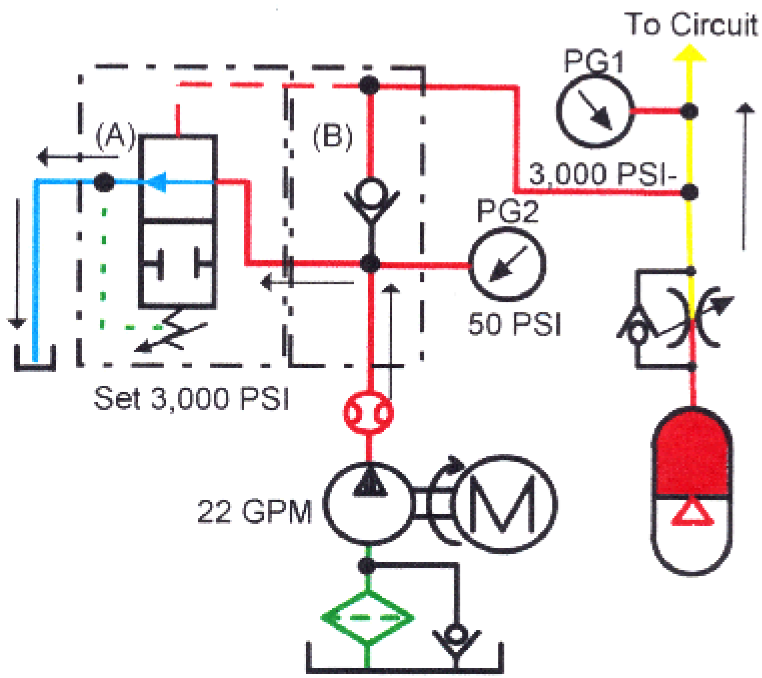


Fig. 18-28. Unloading relief valve in an accumulator circuit.
Shown with system running at pressure.

In Figure 18-28, the accumulator is at pressure and the pump is unloading. The relief valve is fully open or vented because the control piston pushes the pilot control piston off its seat. Without a control piston, relief valves relieve excess pump flow at set pressure, generating a lot of heat. This unloading relief valve has a preset differential of 15% between unloading and reloading the pump.

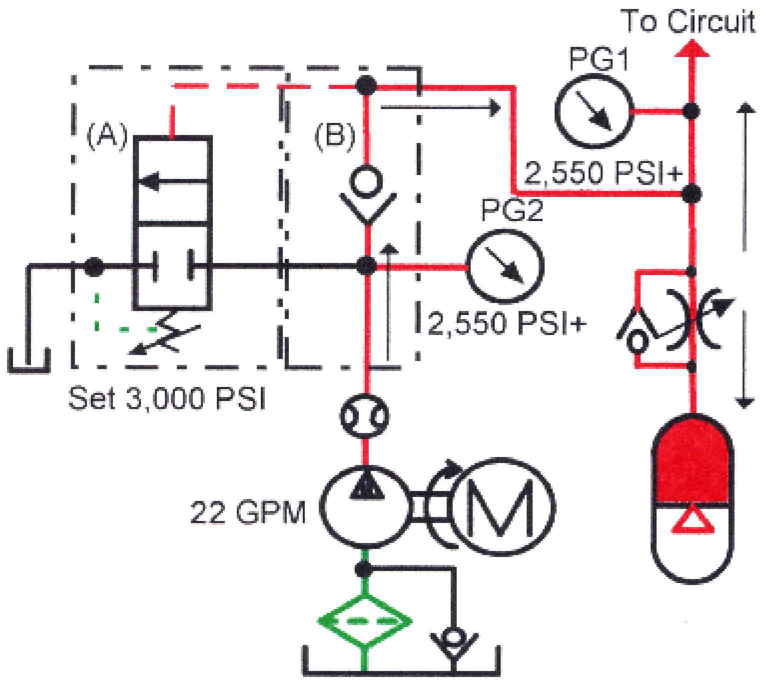


Fig. 18-29. Unloading relief valve in an accumulator circuit. Shown with pump loading again after 15% pressure drop.

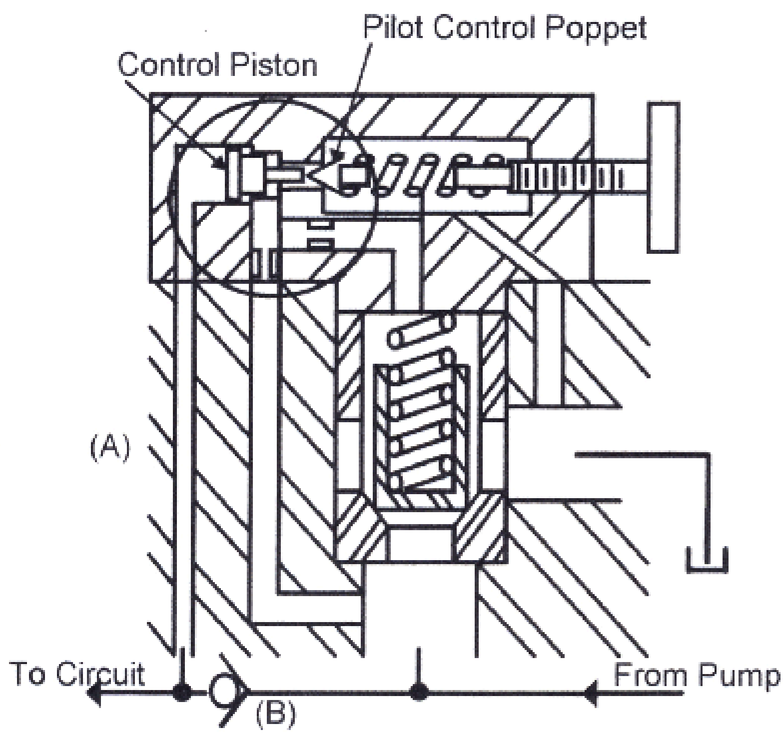


Fig. 18-30. Cutaway view of unloading relief valve.

When system pressure drops to approximately 2550 psi, as in Figure 18-29, its spring force reseats the pilot control poppet again. This forces pump flow into the circuit. This action repeats as long as the pump runs. With a tight circuit and the machine not cycling, the pump unloads approximately 80% of the time.

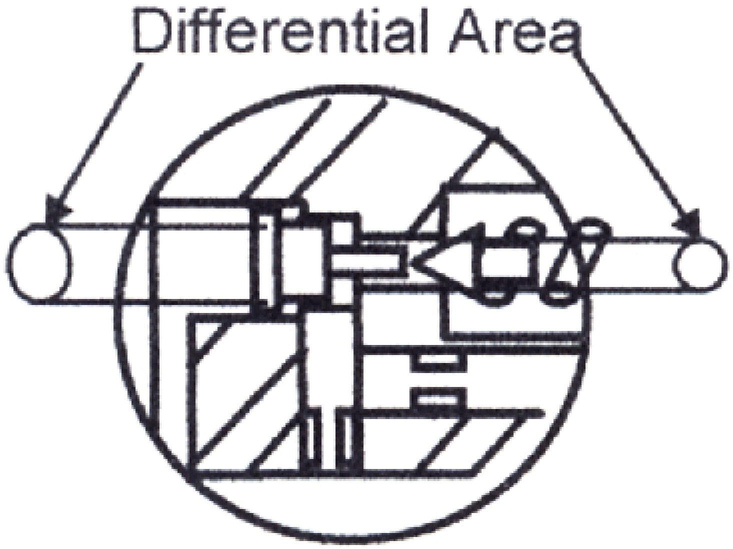


Fig. 18-31. Control piston before reaching set pressure.

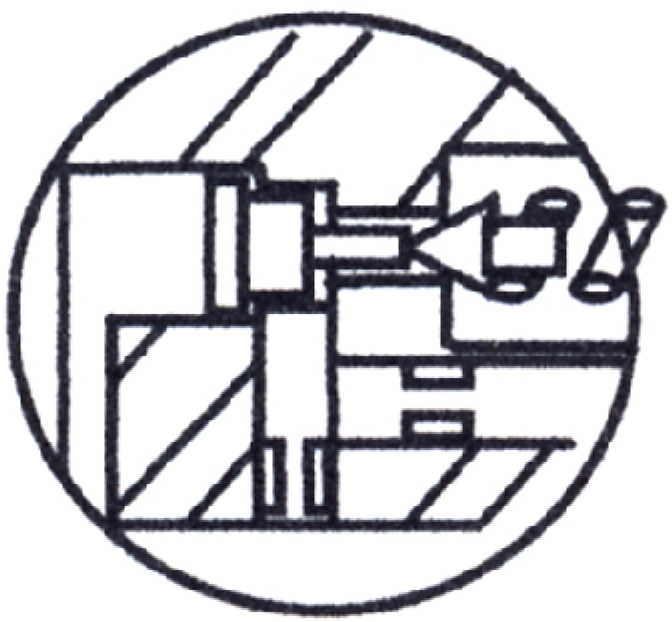


Fig. 18-32. Control piston just at set pressure.

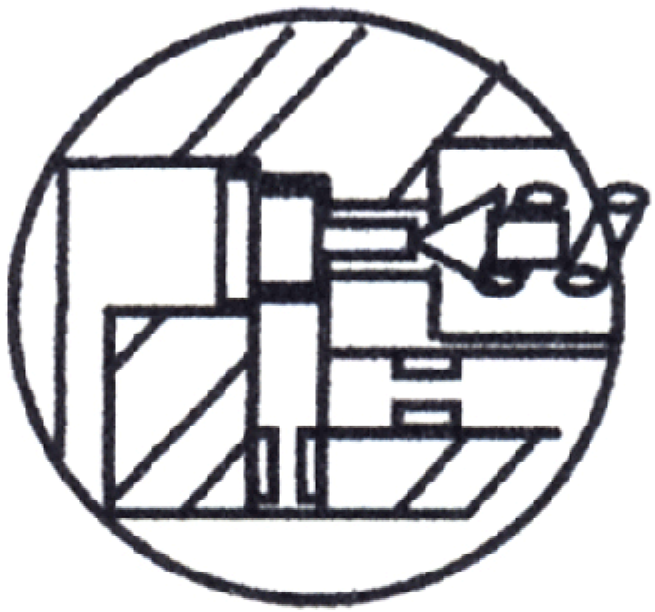


Fig. 18-33. Control piston while pump is unloading.

Figure 18-31 shows a cutaway view of an unloading relief valve. It is similar to a standard relief valve but has an extra control piston in its head. There is an approximate 15% difference in the area of the control piston and the pilot-control poppet seat. As pressure builds, it pushes against both sides of the control piston and against the pilot-control poppet. Nothing moves until pressure starts to force the pilot-control poppet off its seat, as in Figure 18-32. Pressure drop in front of the control piston lets it move and force the pilot-control poppet completely off its seat, Figure 18-33. Forcing the pilot-control poppet off its seat unloads the pump at 20 to 70 psi. The pilot-control poppet stays open until system pressure drops approximately 15%, then closes to force pump flow into the circuit again. When pressure rises to maximum, the pilot-control poppet is pushed off its seat and unloads the pump. This action continues anytime the pump runs.

Directional control valves are the most widely used . . . and the least understood . . . valves in fluid power circuits. Many people are confused by the schematic symbol representations, and have difficulty understanding the terms ways, positions, and operators. Learning to read schematic drawings is similar to learning a foreign language. To the trained eye, a symbol speaks volumes even when no words are present. This chapter attempts to take away some of the confusion and apparent magic of fluid power schematic drawings, and thus help make designing and maintaining fluid power systems easier.
Directional control valves can only perform three functions:

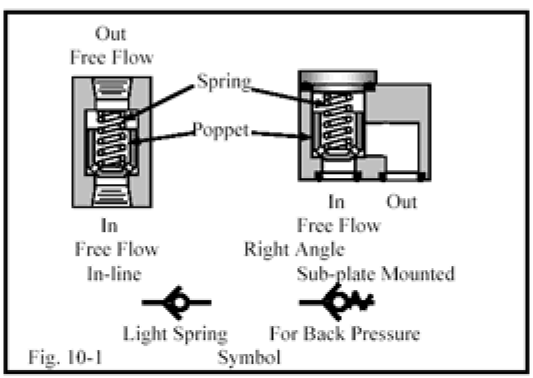
- *stop or block fluid flow*
- *allow fluid flow, and*
- *change direction of fluid flow.*

This seems to simplify a seemingly complex subject, but remember, many valves may combine these functions. This makes them a little more complicated, but still not rocket-science material.

Check valves

*At first glance, the valve type shown in **Figure 10-1** does not appear to be a directional control valve. However, check valves do allow flow in one direction and block flow the opposite direction. Use a check valve in any line where back flow cannot be tolerated. Also pilot-operated check valves (discussed in the next section) can be shifted by an external source to allow reverse flow or stop free flow.*

Fig. 10-1. Cross-sectional views and symbols for two types of check valves



The cross-sectional views show the standard poppet design used in most check valves. As in most early designs, the symbol still pictures a ball on a seat. Ball check valves work well until they are disassembled for repair or when troubleshooting. As these valves operate, they wear a groove where the ball contacts the seat. If this wear groove is not reinstalled exactly where it was, the valve is no longer leak free. On the other hand, a guided poppet always goes back in the same relationship to the seat and seals easily after reassembly.

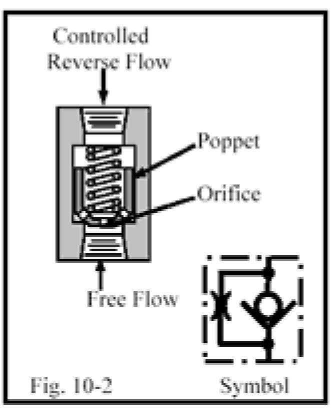
It is easy to understand the function of a check valve. Fluid entering opposite a spring pushes against the poppet and spring to move it out of the way. The inline valve has holes around the angled seat face above the body seat to allow flow to pass. The right-angle design pushes the poppet out of the way and fluid flows by with little restriction.

Check valves are almost trouble-free devices. Seldom is one the cause of a problem. Potential problems can be minimized further if the check valves are: right-angle types, screw-in cartridges, or subplate mounted. Note that an inline check valve's plumbing must be disassembled before the valve can be checked.

Check valves also can control pressure. Almost all check valves use a spring to return the poppet. In most valves, this spring has very light force, because any spring force results in an energy loss and heat. The light springs from most suppliers require about 5 psi to move the poppet against them (some go as low as 1 psi). Some large check valves, when they are mounted vertically, may require no spring because the weight of the poppet causes it to fall onto its seat.

Strong springs give extra resistance to flow so a check valve could replace a relief valve when low-pressure bypass is required. Many manufacturers have check valves with springs that require as much as 125 psi to push their poppets back. These valves work for low-pressure circuits such as a bypass around a low-pressure filter or heat exchanger, or to maintain minimum pilot pressure for pilot-operated directional control valves. When the spring functions as a backpressure or relief valve, the symbol usually shows the spring as part of the symbol.

Fig. 10-2. In-line check valve with orifice drilled through poppet

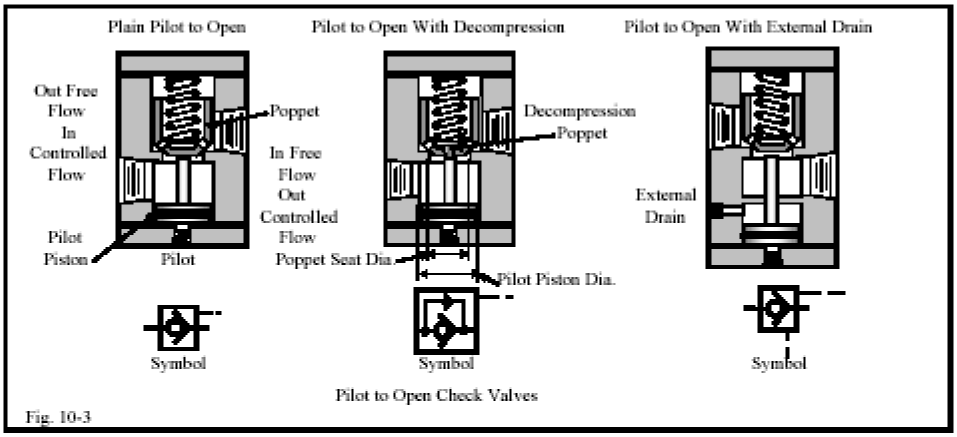


Another lesser-known use for check valves is as a fixed-orifice flow-control function. ***Figure 10-2*** *shows an inline check valve with an orifice drilled through the poppet. The orifice allows free flow in one direction and measured flow the opposite way. The orifice is non-adjustable, so this component is tamper proof. The only way to change actuator speed is to physically change the orifice size. This orificed check valve could protect an actuator that might run away if a line broke or a valve malfunctioned. It will not affect speed in the opposite direction. For this application it should be flange fitted or hard piped directly to the actuator port.*

Pilot-operated check valves

*The check valves in **Figure 10-3** operate like standard check valves, but can permit reverse flow when required. They are called pilot-to-open check valves because they are normally closed but can be opened for reverse flow by a signal from an external pilot supply.*

Fig. 10-3. Three types of pilot-to-open check valves with symbols



*The first cutaway view of a pilot-to-open check valve in **Figure 10-3** is a standard design using a pilot piston with a stem to unseat the check valve poppet for reverse flow. The pilot piston has an area three to four times that of the poppet seat. This produces enough force to open the poppet against backpressure. Some pilot-operated check valves have area ratios up to 100:1, allowing a very low pilot pressure to open the valve against high backpressure.*

*The second valve in **Figure 10-3** shows a pilot-to-open with decompression function. It has a small, inner decompression poppet that allows low pilot pressure to open a small flow passage to reduce backpressure. After releasing high backpressure, the pilot piston can easily open the main poppet for full flow to tank. (This arrangement does not work when the high backpressure is load-induced or generated by other continuous forces.)*

The third valve, pilot-operated with external drain, isolates the stem side of the pilot piston from the in free-flow port backpressure that would resist pilot pressure trying to open the poppet. Notice that in the other two cutaway views, any pressure in the in free-flow port pushes against the pilot piston stem side and resists pilot pressure's attempt to open the poppet. Backpressure could be from a downstream flow control or counterbalance valve in some circuits.

The external-drain port also can be used to make the pilot piston return when using the valve for a pilot-operated 2-way function.

Fig. 10-4. Typical circuit incorporating pilot-operated check valve

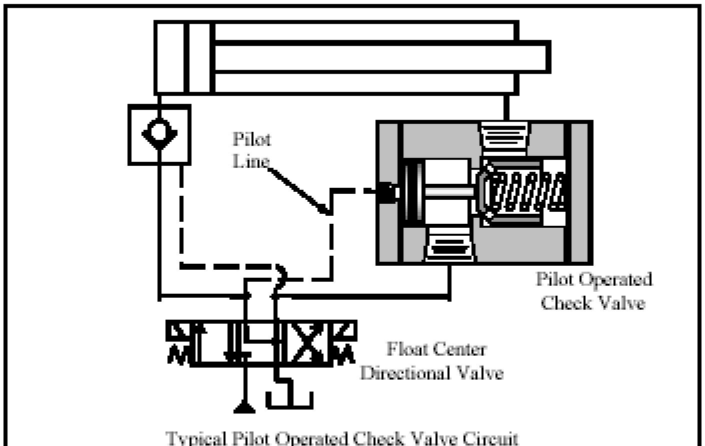


Fig. 10-4

*The circuit in **Figure 10-4** shows a typical application for pilot-operated check valves. Spool-type directional control valves cannot keep a cylinder from moving from a mid-stroke position for any length of time. All spool valves allow some bypass, so a cylinder with an outside force working against it slowly moves out of position when stopped. Installing pilot-operated check valves in the cylinder lines and connecting the directional valve's A and B ports to tank in center position assures that the cylinder will stay where it stops (unless the piston seals leak).*

Fig. 10-5. Circuit with pilot-operated check valve that prevents load from running away if pilot pressure is lost

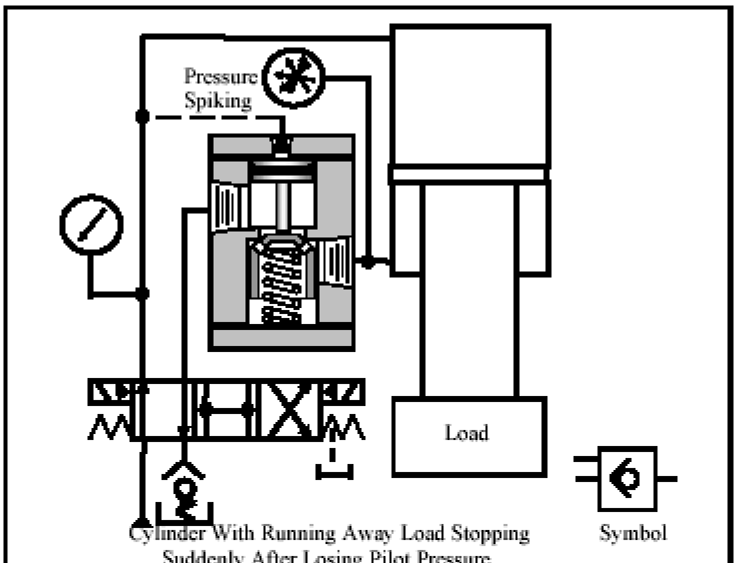


Fig. 10-5

*The circuit in **Figure 10-5** shows a pilot-operated check valve holding a load on the rod end of a vertically mounted cylinder. Pilot-operated check valves can hold potential runaway loads in place without creep, but this circuit usually has problems on the extend stroke. This is because a pilot-operated check valve opens the rod end of the cylinder to tank, letting it run away. When the cylinder moves faster than the pump can fill it, pressure in the cap end and pilot pressure to* *the pilot-operated check valve's pilot port drops and the valve closes quickly. This can generate high-pressure spikes that may cause pipe and part damage. Almost immediately, pressure to the pilot-operated check valve's pilot port builds again and the runaway/stop scenario repeats until the cylinder meets resistance or something fails. The best valve to control runaway loads is the counterbalance valve explained in Chapter 14.*

Fig. 10-6. Circuit with vertically mounted cylinder that is unable to extend

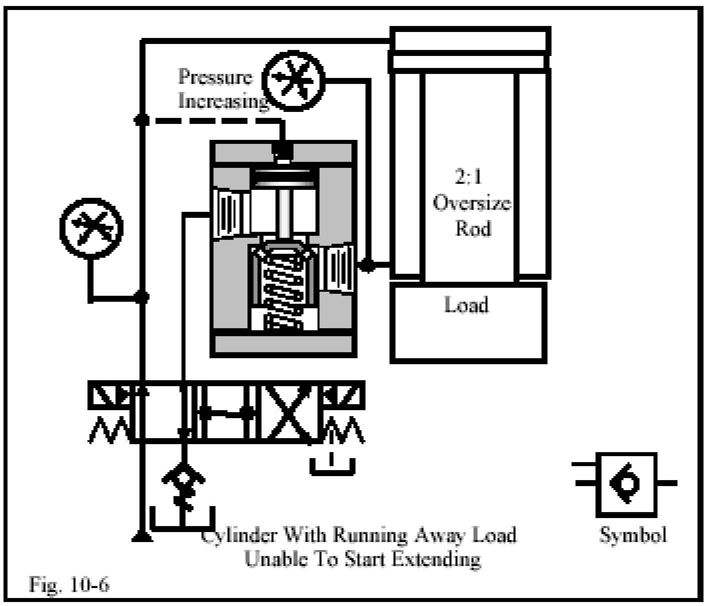
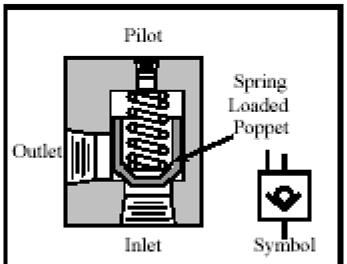


***Figure 10-6** illustrates another problem with using a pilot-operated check valve to hold back a runaway load: a pilot-operated check valve may not open when signaled to let a cylinder with an oversize rod and heavy load extend. When the directional valve shifts to extend the cylinder, load-induced pressure can hold the pilot-operated check valve poppet closed. It may take 300 to 400 psi to force the poppet open, even with its 3:1 or 4:1 area difference. Pressure builds at the pilot port, but at the same time it increases in the cylinder cap end. With a 2:1 rod-differential cylinder, it can add 600 to 800 psi to the load-induced pressure. The additional downward force causes pilot pressure to increase, which causes more downward force, which causes more pilot pressure -- until the circuit reaches maximum pressure. At that point, the relief valve bypasses or the pump compensator kicks in to stop flow. The cylinder simply cannot start to extend . . . and even if it could, the action would be erratic, as in **Figure 10-5**.*

Pilot-to-close check valves

There is also a pilot-to-close check valve, but it is seldom used. It is rarely necessary to have a valve that always stops flow in one direction and also is capable of stopping it the opposite direction.

Fig. 10-7. Pilot-to-close check valve and symbol



Pilot to Close Check Valve
Fig. 10-7

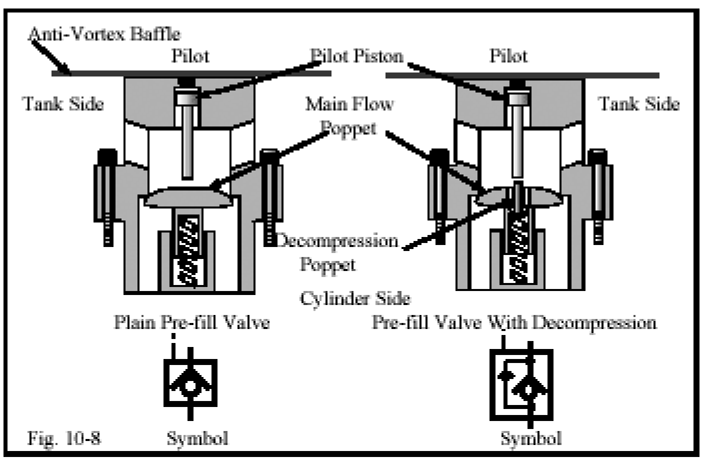
*Notice in the cutaway view in **Figure 10-7** that the spring-loaded poppet does not have communicating holes through it to the spring chamber. Flow passes freely from inlet to outlet until a pilot signal is fed to the pilot port. Because the pilot port side of the main-flow poppet has more area than the inlet side, this valve can be closed against free flow.*

Poppet-type pre-fill valves

Pre-fill valves operate similarly to pilot-operated check valves, but they are usually much larger. Some pre-fill valves can handle flows in excess of 6000 gpm at pressure drops of less than 4 to 8 psi. Their normal function is to fill and exhaust a large bore cylinder as it travels to and from contact with the work piece. Large, high-tonnage presses -- both vertical and horizontal -- use pre-fill valves to reduce pump size while maintaining cycle time.

The cutaway view and symbol in ***Figure 10-8*** *show the construction of a typical poppet-type pre-fill valve. A large main-flow poppet seals the path between the tank and the cylinder ports. As the piston advances, vacuum in the void behind it allows atmospheric pressure to push the main-flow poppet open so fluid from the tank can fill this void. On the retraction stroke, a signal to the pilot piston pushes the main-flow poppet open so fluid can return to tank. While a pilot-operated check valve's pilot piston is larger than the poppet it opens, the main-flow poppet in a pre-fill is much larger in diameter than the pilot piston. Thus it is impossible to open the main-flow poppet against high backpressure. This keeps decompression shock from damaging pipes and components.*

Fig. 10-8. Two models of pre-fill valves with symbols



Decompression shock occurs when large volumes of fluid at high pressure are released suddenly. Because all hydraulic oil has some entrained air (bubbles so small they cannot be seen without magnification), there is a 0.5 to 1% compressibility that must be dealt with when using large-bore cylinders. On top of fluid compressibility, the cylinder tube may stretch diametrically and longitudinally. In addition, the framework that is resisting the tonnage produced also can stretch. Summing all these factors, a 50-in. bore cylinder with a 72-in. stroke can contain more than 25 gal of extra fluid at 3000 psi. If this trapped fluid suddenly has a large open path to atmosphere, its velocity at first release is such that it can break fittings, blow hoses, straighten tubes or pipe bends with relative ease. Releasing this same trapped fluid in a controlled manner over a few seconds dissipates the excess energy and no damage is seen.

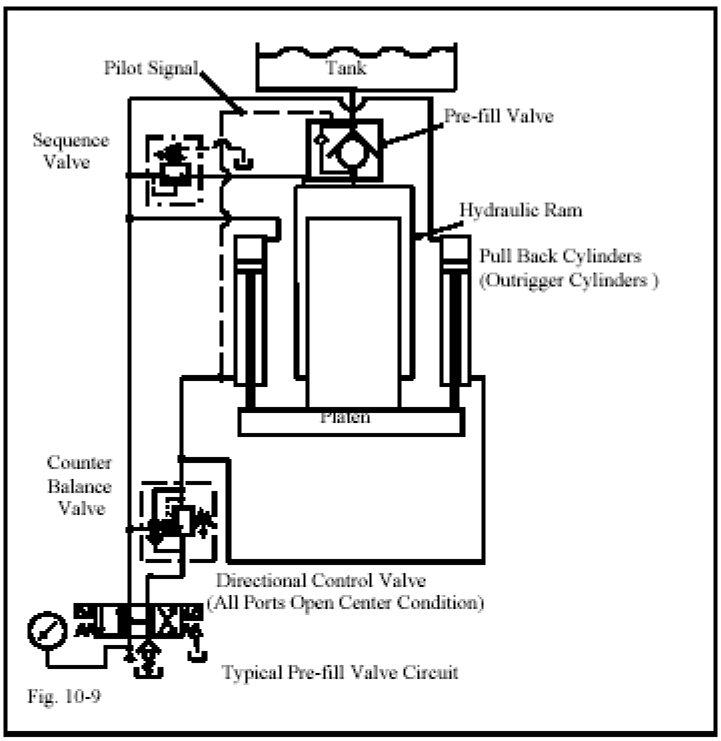
The plain pre-fill valve might be used on smaller cylinders or circuits that have other means for decompressing. The pre-fill valve with decompression has a small poppet in the large poppet *that is easy to open at high pressure but will not allow the high flow that causes decompression shock. This decompression poppet usually has a means to adjust how fast the cylinder decompresses.*

Another pre-fill valve design is the sleeve type that must be externally shifted open and closed. Both designs give the same results even though their operation is different. (See Chapter 4 for a cutaway view and symbol of a sleeve type pre-fill valve.)

Typical decompression circuit

*The circuit in **Figure 10-9** operates a vertical single-acting hydraulic ram press with pullback cylinders for the retraction stroke. The press has a poppet-type pre-fill and gets a fast stroke from only filling the pullback cylinders during the approach stroke. A sequence valve keeps pump flow from going to the ram until pressure reaches a preset level.*

Fig. 10-9. Typical vertical ram circuit with pre-fill valve



During the approach part of the stroke, atmospheric pressure pushes fluid into the large-bore ram through the pre-fill valve because there is vacuum behind the extending ram. When it contacts the work, the ram stops and the pre-fill valve closes. Pressure starts to rise and when it is high enough to open the sequence valve, pump flow goes to the pullback cylinders and the ram. Extension speed slows and tonnage increases to do the work required.

A signal that the work is complete shifts the directional control valve to send pump flow to the rod ends of the pullback cylinders and to the pilot signal of the pre-fill valve. The pre-fill valve's *pilot piston moves forward and contacts the decompression poppet. This lets trapped fluid flow out at a controlled rate. Pressure in the ram drops quickly and smoothly. When pressure is low enough, the pilot piston opens the main poppet to let fluid from the ram return to tank. When the ram loses pressure, the pullback cylinders can raise the platen and push fluid from the ram back to tank.*

General directional control valve terminology

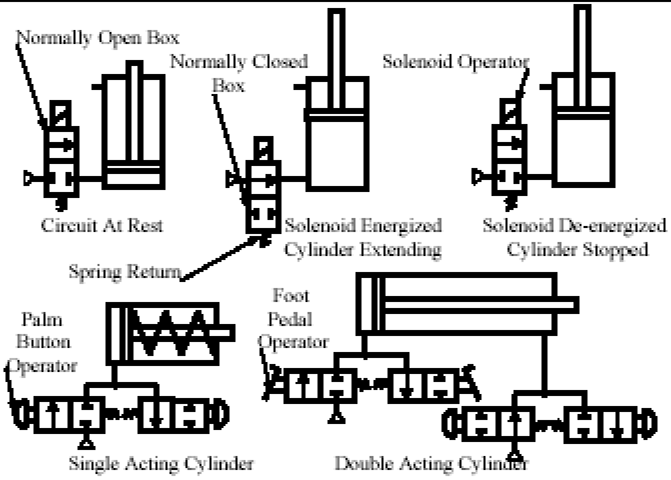
Directional control valves are specified generally by the number of ports or ways (lines attached to the symbol's box) and the number of positions (boxes or envelopes in the symbol) they have. Other information about them includes whether they are normally closed (not passing fluid), normally open (passing fluid), how they are operated (solenoid, manual, or spring) and other features such as manual overrides, drain ports, pilot ports, etc.
Some general rules for drawing symbols are:

- *only draw flow lines to one box of the symbol*
- *always see that flow paths and direction of flow in each box is compatible*
- *on 4-way hydraulic valves, pipe the A port to the cap end of the cylinder and the B port to the rod end*
- *draw all symbols in their at-rest position. Show valves that are held actuated by a machine member in their shifted condition, and*
- *provide information such as pressure settings, flow rates, orifice sizes, horsepower and rpm where applicable.*

(According to this method of specifying, check valves and pre-fill valves would be 2-way valves because they have two ports. However, because these valves are basically single function and have infinitely variable flow paths, their symbols and terminology do not follow general directional control valve rules.)

***Figure 10-10** shows the symbol for a 2-way directional control valve and how it could function in a circuit. Notice the symbol has two boxes (or envelopes) to indicate two positions. Each position is a flow path. The box with flow lines coming to it is the normal or at-rest position of the valve. The normal or at-rest position is usually at the spring end of a spring-return valve as seen in the figure.*

Fig. 10-10. Circuits in which 2-position, 2-way valves operate cylinders



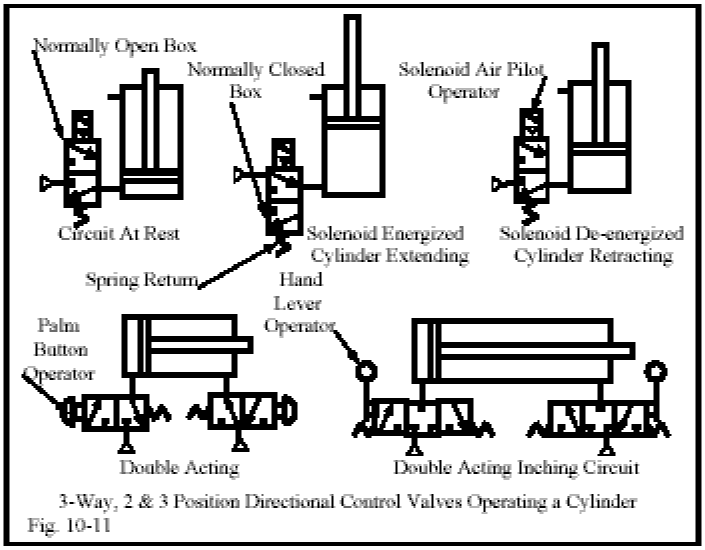
2-Way, 2-Position Directional Control Valves Operating a Cylinder
Fig. 10-10

*The circuit at rest in **Figure 10-10** illustrates how a schematic drawing shows the component symbols for the system builder or troubleshooter. Valves, actuators, flow paths and line connections are all shown according to the ANSI or ISO graphic symbols that were explained in Chapter 4. To understand how the circuit operates, a person must be able to read the symbols and know how they represent a piece of hardware. The valve in this circuit is 2-way, 2-position, direct solenoid-operated, spring return, normally closed. The diagrams to the right of the circuit at rest show how the directional control valve shifts to its second position and ports fluid to the cylinder. In the real world, this is done in a person's imagination . . . and can be confusing when several valves are working simultaneously. In the diagram it is easy to see that with the solenoid energized, the normally open box moves in line with the input flow and sends fluid to the cylinder. The arrow in the normally open box shows flow from inlet to cylinder port, causing the piston to extend. If the solenoid is de-energized, the spring returns the valve to the circuit at rest condition and the cylinder stops in its last position.*

*Two-way valves cannot have more than two positions because they can only stop or allow fluid flow. It is easy to see that a 2-way directional control valve will not operate a single-acting cylinder. These valves are only good for operations that require an on-off supply. As shown in the bottom half of **Figure 10-10**, two 2-way valves are needed to control a single-acting cylinder. A double-acting cylinder needs four 2-way valves to control it. There are both normally closed and normally open valves in these circuits.*

***Figure 10-11** shows how 3-way valves can replace 2-way valves and make a machine simpler. This circuit at rest has a cylinder powered by a 3-way, 2-position, solenoid pilot-operated, spring-return, normally closed directional control valve. Because this valve has a flow path from the pressure port to the cylinder port and from the cylinder port to atmosphere, it can control a single-acting cylinder. The diagrams to the right show that when the solenoid is energized, the cylinder extends under power. The next schematic diagram shows the cylinder retracting from external forces with the solenoid de-energized.*

Fig. 10-11. Circuits in which 2- and 3-position, 3-way valves operate cylinders



Two 3-way valves are needed to power a double-acting cylinder as shown in ***Figure 10-11****. The double-acting palm button activates this circuit. The valve on the cap end is normally closed and the valve on the head end is normally open. This is a simple anti-tie down circuit, but is not OSHA safe because one palm button can be depressed before the second one and the cylinder will move. OSHA requires that both buttons be operated concurrently to make the cylinder extend. It does meet the anti-tie down requirement because the cylinder will not retract until both palm buttons are released.*

The double-acting inching circuit in ***Figure 10-11*** *uses two 3-way, 3-position, spring-centered valves to make it possible to stop the cylinder at any point in its stroke. A 3-way valve can have a third position to perform another function. The pictured center condition has all ports blocked, which stops flow at all ports. This is the center condition normally found on a 3-way valve.*

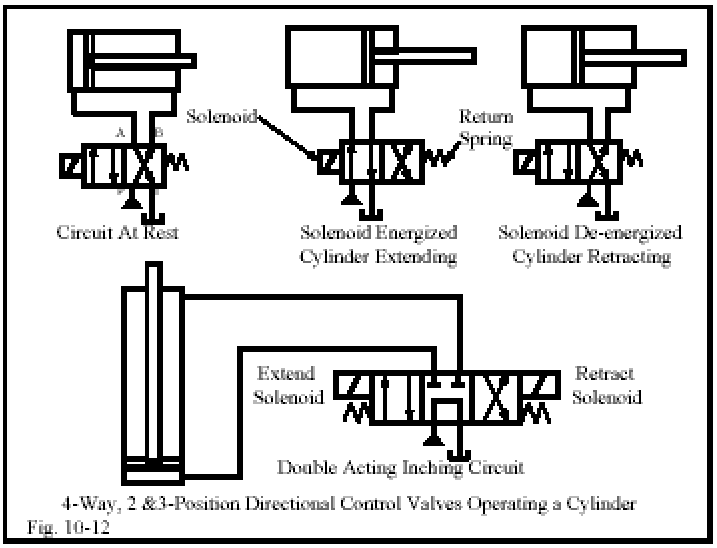
Note: pneumatic inching circuits cannot stop and hold a load consistently. Any change in speed, load, or pressure can produce a different stopping position. About plus or minus one inch would be the best position accuracy an air cylinder would achieve, unless it is moving very slowly. Air leaks in the plumbing or valves also interfere with trying to stop and hold an intermediate position. Leaks may let one end of the cylinder bleed off and allow air from the opposite end to expand and move the cylinder out of position.

Using two 3-way valves attached directly to each cylinder port will save air. By eliminating all piping between the valve and the actuator, less air is consumed during each cycle. The air savings per cycle may not be great, but it can add up on fast-cycling equipment with multiple cylinders.

A 3-way valve can be used as a 2-way function when an on-off condition is needed.

*The 4-way valve in **Figure 10-12** makes it possible to operate a double-acting cylinder with a single valve. The four ports on hydraulic valves are marked P for pump, T for tank, and A and B for cylinder or outlet ports. Most valve manufacturers follow this universal marking system. Most air valves are configured as 5-way functions with two exhaust ports. This works well for air valves because atmosphere is the tank. Return piping is not required.*

Fig. 10-12. Circuits in which 2- and 3-position, 4-way valves operate cylinders



*The circuit at rest in **Figure 10-12** shows a 4-way, 2-position, direct solenoid-operated, spring-return directional control valve. In at-rest condition, pump flow holds the cylinder in the retracted position while the cap end is ported to tank.*

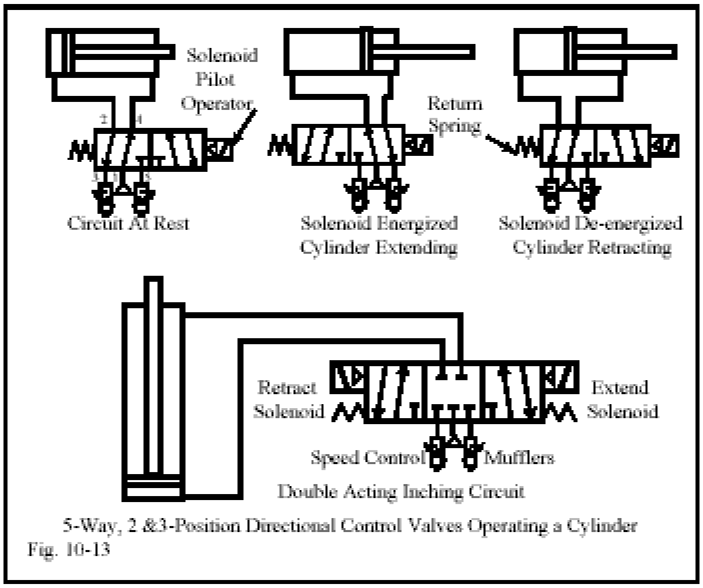
In the solenoid-energized, cylinder-extending condition, pump flow connects to the cylinder cap end while the head end is connected to tank. The cylinder is extending under power at this time. In the solenoid-de-energized, cylinder-retracting condition, the valve returns to normal and the cylinder retracts under power.

*A single 4-way directional control valve can power an actuator in both directions. At the bottom of **Figure 10-12**, a 4-way, 3-position, double direct solenoid-operated, spring-centered, tandem-center directional control valve powers a double-acting cylinder in the vertical position with its rod up. As shown in the at-rest condition, pump flow goes to tank and the cylinder ports are blocked. Energizing the extend solenoid sends pump flow to the cylinder cap end to make it extend. Energizing the retract solenoid sends pump flow to the cylinder head end, making it retract. With both solenoids de-energized, the cylinder stops and holds position for some time. Because most directional control valves use a metal-to-metal fit spool, there is some bypass, so the cylinder might drift when it has external forces acting on it. Note: this double-acting inching circuit may need a counterbalance valve to stop it from running away as it retracts.*

Some manufacturers offer 4-way valves in special 4-position configurations. The fourth position is often a regeneration path to move the cylinder more rapidly at reduced force.

*The 5-way valve in **Figure 10-13** is found most often in pneumatic circuits. Although most hydraulic valve designs are 5-ported, the tank ports are connected internally by cored passages so only one external tank connection is needed. Air valves exhaust to atmosphere so having two exhaust ports is not a problem.*

Fig. 10-13. Circuits in which 2- and 3-position, 5-way valves operate cylinders



Notice the speed-control mufflers in these circuits. They reduce exhaust noise and act as meter-out flow controls. A 5-ported valve, especially if it's spool type, can offer advantages when piping certain pneumatic circuits.

When this circuit is at rest, air pressure ported to the cylinder's head end holds the cylinder in its retracted position. Meanwhile the cap end is exhausted to atmosphere. With the solenoid energized, cylinder-extending air is ported to the cylinder cap end while the head end exhausts. With the solenoid de-energized, the return spring shifts the valve back to normal and the cylinder retracts under power.

A 5-way spool-type valve also can be piped with dual inlets at different pressures -- to conserve energy, to smooth stroke times and speed, or to cause a cylinder to stroke at high speed. (See Chapter 13 on Flow Controls and Chapter 17 on Quick-Exhaust Valves for circuits to do these.)

*At the bottom of **Figure 10-13**, the double-acting inching circuit uses a 3-position, 5-way valve with all ports blocked in center condition to cycle a cylinder. Within reasonable limits, the cylinder can be stopped and held for short periods. (See the note on 3-way valves from **Figure 10-11** on the reasons for poor results in pneumatic inching circuits.)*

Another center condition for a 5-way air valve is pressure blocked and cylinder ports open to atmosphere. This center condition can be used for mid-stroke stopping of a horizontally mounted cylinder.

Both 4- and 5-way valves can replace 2- and 3-way valves by plugging or not using certain ports to produce the desired function. This can save money in inventory and time when troubleshooting. Only one spare valve of a given size takes care of many problems on the floor.

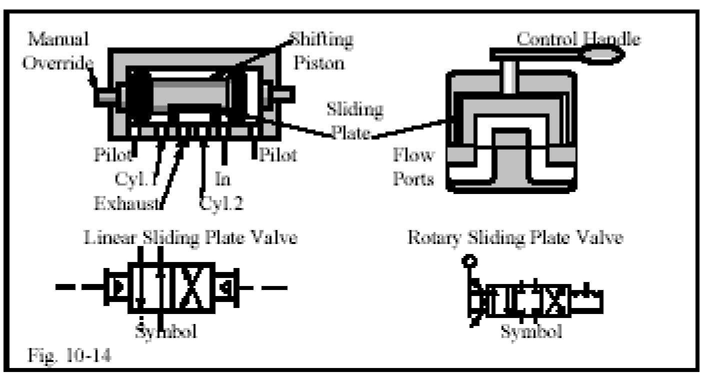
Types of directional control valves

Directional control valve designs generally fall into three categories:

- sliding-plate valves
- poppet valves, and
- spool valves.

Sliding-plate valves use linear or rotary action to open and close ports to change flow paths. **Figure 10-14** has a cutaway representation of each type. A linear sliding-plate valve usually is pilot- or solenoid pilot-operated to generate enough force to reliably move the lap-fitted linear sliding plate. As the hollow linear sliding plate passes over openings in the body, fluid is channeled to a working port or to exhaust through the hollowed out cavity or through the body. (Linear sliding-plate valves are used only in pneumatic circuits.)

Fig. 10-14. Two types of sliding-plate valves, with their symbols

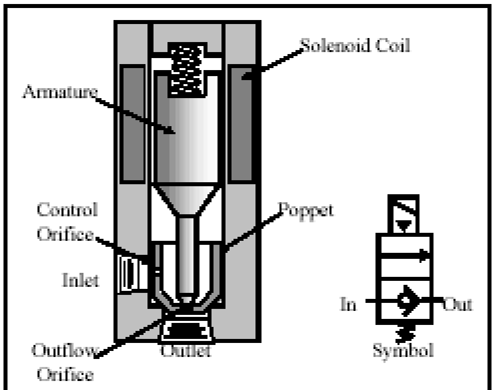


Rotary sliding-plate valves are often manually operated. Some manufacturers offer a pneumatic or electrically powered rotary actuator for automatic operation as well. These valves are used in pneumatic and hydraulic circuits as control and/or isolation valves -- when high shifting speed is not needed. The seal between the rotary sliding-plate and the body can be lap-fitted or may have spring- and pressure-loaded seals to eliminate leakage.

Poppet-type directional control valves

*Poppet-type directional control valves are similar to pilot-to-close check valves. The cutaway view in **Figure 10-15** shows the construction of a hydraulic 2-way, normally closed, poppet-type directional control valve. Fluid at the inlet port passes through the control orifice to the backside of the poppet. The tip of the spring-loaded armature closes off the outflow orifice to trap fluid behind the poppet. As the symbol shows, the valve is a check valve that stops flow from inlet to outlet in the normal condition. This design will not stop flow from outlet to inlet, although flow in this direction may be at a reduced rate. If using this valve for reverse free-flow in its normally closed condition, make sure to choose one with free-flow capability.*

Fig. 10-15. Poppet-type, 2-way hydraulic directional control valve



2-Way Poppet Type Hydraulic Directional Control Valve
Fig. 10-15

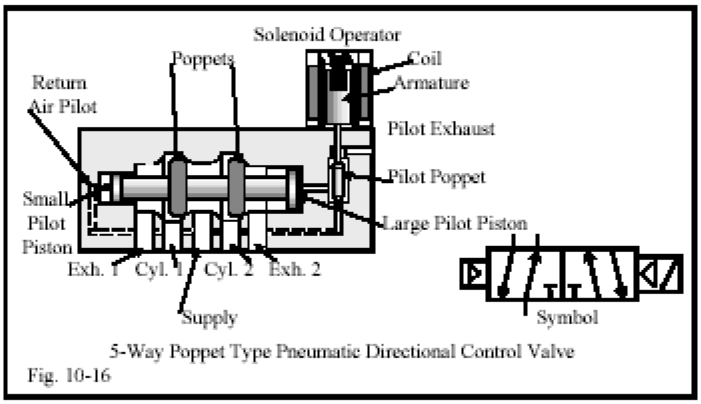
Energizing the solenoid coil creates a magnetic field that raises the armature to open the outflow orifice. This orifice is larger than the control orifice, so the greater flow through it causes a pressure drop behind the poppet. Now, inlet pressure pushing on the poppet's annulus area outside the seat diameter unseats it to allow fluid flow to the outlet. De-energizing the solenoid coil lets spring force reseat the armature tip to again trap fluid behind the poppet and close it.

Unlike spool valves whose lands overlap, poppet valves open a flow path to outlet immediately. This means response time of whatever the valve controls is very fast. Also, when a spool valve shifts open it goes to the end of its stroke regardless of the amount of flow. On the other hand, a poppet only opens as much as the flow going through it needs. This means the poppet has less distance to move to stop flow, so again its response is faster.

Chapter 11 covers poppet-type slip-in cartridge valves used for directional control. These valves have the same characteristics as just explained here and they work well in circuits that require *fast response. Chapter 12 covers infinitely variable spool valves that also offer very fast response.*

*The 4-way poppet valve in **Figure 10-16** is a typical design for pneumatic service. Poppet design valves are very tolerant of contamination and many plants use them for this reason. They are also very responsive and provide a positive seal when their poppets seat. (Many poppet valves are built with resilient materials on the poppets where they contact the seats.*

Fig. 10-16. Poppet-type, 5-way pneumatic directional control valve



*One drawback to this design is that air is free to go any direction as the poppets shift from one flow path to the other. In valve terminology this is called open crossover (and can be helpful with hydraulic valves as explained later). The cutaway view in **Figure 10-16** shows how flow can go to both cylinder ports and to both exhausts as the poppets move to the opposite seat.*

Another possible problem with poppet valves is that they usually only operate in one manner. When you purchase a 2-way, normally closed poppet valve, it cannot be changed to normally open. The port marked In is always the supply line. Air piped to the Out or Cyl port usually blows through the valve with little resistance. Spool-type valves (discussed next) overcome these problems in most cases.

*The poppet valve in **Figure 10-16** shifts to its second condition when the coil of the solenoid operator receives an electrical signal and pulls the armature up. This action lets supply air into the large pilot piston to move the poppets to the second valve position. Even though the small pilot piston has supply air against it all the time, it has less force. De-energizing the solenoid operator exhausts the large pilot piston and the poppets return to the normal position.*

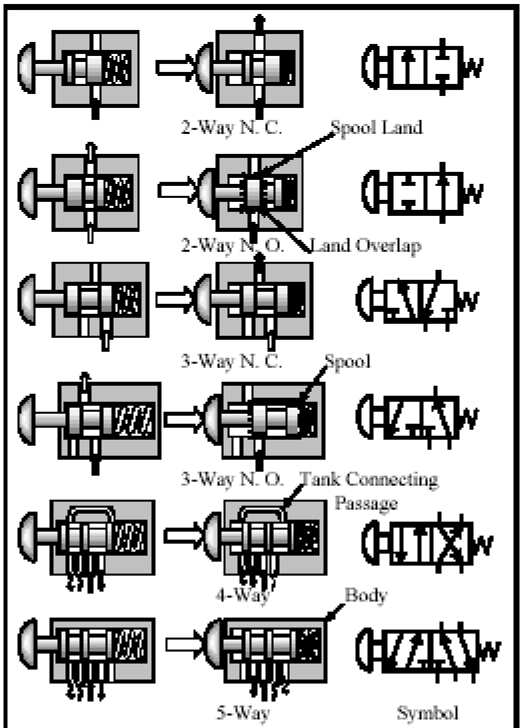
This is a very reliable design because there are no springs to rust, weaken, or break. Usually the area ratio is 2:1; so shifting force is equal in both directions. Some manufacturers also use a spring in the return end to keep the poppets in place when there is no air supply. Valves with this *type of shifting arrangement usually require a minimum pressure of 25 to 40 psi or an external pilot supply at least that high.*

Spool-type directional control valves

*For circuits with flows less than 100 gpm, the most common hydraulic directional control valves use a spool-like internal member to direct flow. (Many air valves also use a spool, due to the advantages offered by this design.) The cutaway views in **Figure 10-17** show some simplified spool arrangements and terms associated with this valve. Notice that counting the number of ports that carry working fluid on the cutaway or symbol gives the number of ways the valve has. A 2-ported valve is a 2-way valve and a 5-ported valve is a 5-way valve.*

*All valves in **Figure 10-17** are two position as shown by two boxes in the symbol. As stated before, a 2-way valve can have only two positions because it can only stop or allow flow. All other valves are able to have three positions, while 4-way valves can have four positions in special cases. A 5-way valve is a special case mainly used in pneumatic applications where an extra exhaust port is not a problem. Notice that a 4-way valve has five ports but its tank ports are internally connected to eliminate an extra port in the body. This is important in hydraulic valves because it reduces piping and potential leak points.*

Fig. 10-17. Views of a variety of valve spool configurations, with their symbols. (All have palm-button operators.)



Different Spool Valve Configurations
Fig. 10-17

Spool valve advantages

The main advantage of spool valves is that fluid entering the valve from any working port does not affect spool movement. The poppet in a poppet valve can have pressure on one side and only a light spring on the other. This can result in premature movement of the poppet when pressure enters a port. In a spool valve, pressure always is applied to two equal opposing areas or the edge of a land. Thus pressure forces that could move the spool are cancelled or non-existent. This means that a spool valve can be shifted manually, electrically, mechanically, pneumatically, or hydraulically with the same force regardless of the operating pressure. Low-force solenoids can be used because the most they need to overcome is mechanical friction and light springs.

Spool valve disadvantages

Many spool valves are designed with metal-to-metal sliding fits. As a result, some fluid may bypass these seals. If this happens, an actuator may not hold its position if outside forces are applied. It also means wasted energy and resulting heat. (Many pneumatic valves use some sort of resilient seal in the body and/or on the spool to eliminate air leaks.) To reduce bypass, spool valves have land overlap, so as they start shifting to open a flow path, there is a delay before fluid starts flowing. The delay only lasts for milliseconds and does not cause a problem -- unless the cycle is very fast and/or there are several valve shifts per cycle.

Another time delay occurs when a spool shifts to the end of its stroke. There is often more movement than required for the flow needed. When the spool shifts back to center or to the *opposite flow path, it consumes more time to travel the extra distance. This slows the cycle, especially when several valves are involved. Stroke limiters that control maximum spool movement can eliminate this delay, but are seldom seen in actual practice. The common fix for these problems is to speed up traverse time by installing a larger pump. However, faster actuator movement can add shock and heat due to higher energy input.*

Hydraulic 4-way spool valves

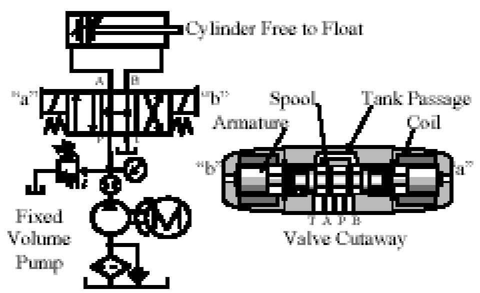
Most manufacturers of hydraulic valves only build a basic 4-way function. When they offer a 2-way function, it is usually a 4-way valve with a different spool and the unused ports plugged or piped to tank. Any 4-way valve can perform 2- or 3-way functions in a normally closed or normally open configuration by using the right ports and plugging or draining unused ones.

Hydraulic 4-way valves usually come in 2- or 3-position configurations. They may be 2-position, single-solenoid, spring-return; double-solenoid, detented; or 3-position, double-solenoid, spring-centered. Some manufacturers offer 4-position valves with a float or regeneration center position for special circuits, but they are rare.

The majority of hydraulic circuits use a 4-way, 3-position directional valve even though it complicates the electrical circuit. One reason may be to provide the ability to stop an actuator in mid cycle -- either for manufacturing or setup functions. Other reasons are to port pump flow to tank while the machine is idling or to let external forces move an actuator.

***Figures 10-18** through 10-21 show typical circuits in schematic form with valve cutaways for the four commonly used center conditions in hydraulic 4-way directional control valves. (Symbols for other spool center conditions are shown in Chapter 4.) Each center condition offers a flow path to meet a specific circuit need that should be obvious when reading a schematic. Note that these typical circuits are not the only way to apply these valves.*

Fig. 10-18. Typical hydraulic circuit for a 4-way valve with all-ports-open center condition



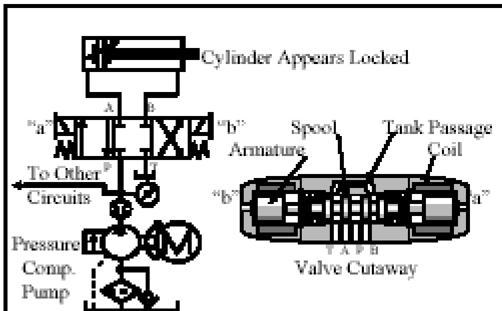
Typical Circuit For A 4-Way Directional Control Valve
"All Ports Open" Center Condition

Fig. 10-18

*The circuit in **Figure 10-18** uses an all-ports-open center-condition valve that allows flow to and from all ports. Notice how the spool lands are too narrow to block the fluid ports. This means fluid is free to go to other ports while the spool is centered. The symbol for the valve plainly shows this open-center condition. A circuit with this type of valve normally has a fixed-volume pump. The open center lets all pump flow return to tank at little or no backpressure. This saves energy and reduces heat to the point that a heat exchanger is not necessary on most circuits.*

*The cylinder in **Figure 10-18** is free to float when acted on by outside forces. Otherwise it sits still. Normally this circuit only has one valve and actuator. Other valves and actuators trying to use this pump would not receive fluid due to the free path to tank.*

Fig. 10-19a. Typical hydraulic circuit for a 4-way valve with all-ports-closed center condition

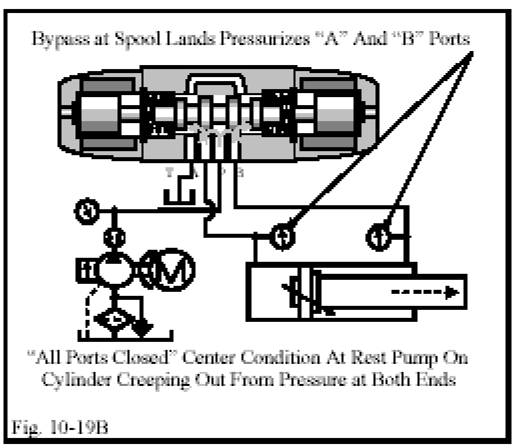


Typical Circuit For A 4-Way Directional Control Valve
"All Ports Closed" Center Condition
Fig. 10-19A

*The valve in **Figure 10-19a** has an all-ports-closed center-condition that blocks pump flow. This valve appears to be able to stop and hold a cylinder in place. Notice how the spool lands are wide enough to completely cover the A and B ports. This blocks flow to and from them, and also stops flow at the P port. This circuit normally has a pressure-compensated pump. System* *pressure is at the pump compensator setting until all pump flow is going to the actuators at their working pressures. The pump in a closed-center circuit can supply other circuits one at a time or simultaneously with low to medium energy loss -- even when operating at less than maximum flow.*

Because all metal-to-metal fit valves have some spool bypass, a closed-center valve will not stop and hold a single-rod cylinder except for a short period. ***Figure 10-19b*** *shows how bypass fluid at the spool lands leaks directly into the A and B ports and pressurizes both ends of the cylinder at roughly half system pressure. Equal pressure at both ends of a single-rod cylinder always causes it to extend due to unequal forces on unequal areas. The cylinder will not move rapidly because some fluid must go to tank across another leak path. This cylinder action is called regeneration, and will be explained under cylinders in Chapter 15.*

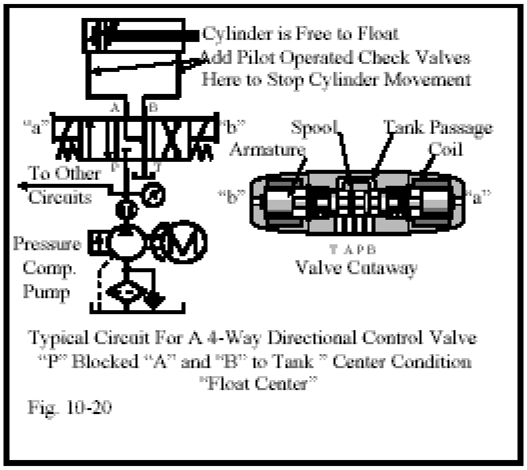
Fig. 10-19b. With all-ports-closed valve center condition, bypass flow will extend cylinder



In a new circuit, bypassing fluid may not affect the cycle, but it can cause problems later on. Also, cylinders with small rods and/or heavy loads may not have enough force to move -- especially when machine fits are new and tight. The actual force in this regeneration mode is calculated by multiplying the rod area by pressure at the cylinder cap end.

The circuit in ***Figure 10-20*** *shows a float-center valve. The P port to the pump is blocked, and ports A, B, and T are interconnected so that both cylinder ports are open to tank. Notice that the spool lands are wide enough to block the P port but still allow flow to or from A and B ports flow to or from each other or tank. A pressure-compensated pump normally supplies a circuit with this valve. System pressure is the pump compensator setting until all pump flow is going to the actuators at their working pressures. The pump in a float-center circuit can supply other circuits one at a time or simultaneously with low to medium energy loss, even when operating at less than maximum flow.*

Fig. 10-20. Typical hydraulic circuit for a 4-way valve with float center condition

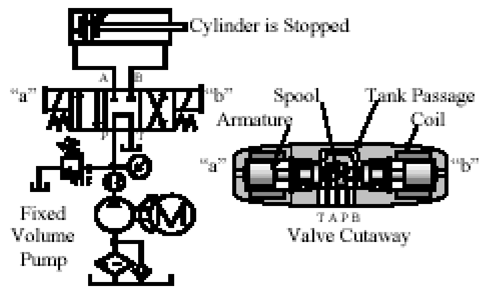


When a cylinder in a multi-actuator circuit must be positively locked in place, select a valve with a float-center condition, and add pilot-operated check valves or a counterbalance valve. (Adding these blocking valves to an all-ports-closed-center directional control valve often does not prevent cylinder movement.)

With a float-center valve a single-rod cylinder may extend at any speed when the circuit has high tank backpressure. High tank backpressure goes to both ends of the cylinder and causes it to extend if the load is low and/or the cylinder has an oversize rod. To overcome this situation, install a check valve at the T port to stop back flow from the tank line.

On horizontally mounted cylinders, use pilot-operated check valves in the cylinder lines to positively lock the cylinder from moving due to external forces. On vertically mounted cylinders, use a counterbalance valve to hold the load and keep it from running away while cycling. (Counterbalance valves are explained in Chapter 14.)

Fig. 10-21. Typical hydraulic circuit for a 4-way valve with tandem center condition



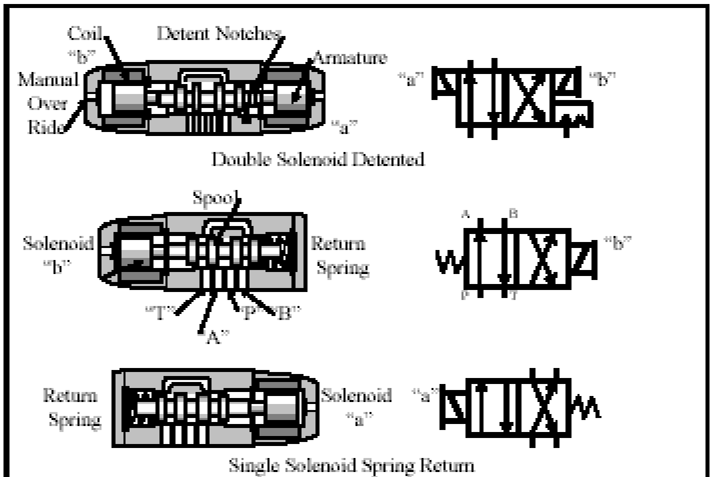
Typical Circuit For A 4-Way Directional Control Valve
"A" and "B" Blocked "P" to "T" Center Condition
"Tandem Center"

Fig. 10-21

*The circuit in **Figure 10-21** uses a valve center condition with the A and B ports blocked, and the P connected to T. This valve lets pump flow go to tank and blocks both cylinder ports. This configuration is often referred to as a tandem-center valve. Notice that the spool lands are wide enough to block the A, B, and P ports – the same as an all-ports-closed valve. However, this valve has a hollow spool and cross-drilled ports at P and both ends at T. The drilled passages provide a path for all pump flow to go directly to tank in the center position. Because the drilled passages also introduce extra backpressure, most suppliers' catalogs show a lower nominal flow or higher pressure drop for tandem-center valves.*

Circuits with tandem-center valves normally have fixed-volume pumps. The pump-to-tank path lets all flow return to tank at little or no backpressure. This saves energy and reduces heat to the point that a heat exchanger is unnecessary in most circuits. Be aware of the reduced flow or higher backpressure when specifying or using tandem-center valves. A circuit may look good on paper, but can run hot because of wasted energy. This is especially true when using tandem-center valves in series. The backpressure for each valve is additive. A three-valve circuit can easily require more than 300 psi to unload the pump.

Fig. 10-22. Views of double-solenoid detented and single-solenoid, spring-return valves, with symbols



Two Position Single Solenoid Spring Return and Detented 4-Way Directional Control Valve

Fig. 10-22

Figure 10-22 *presents symbols and cutaway views for standard 2-position valves. Notice that the double-solenoid detented valve has the same outward appearance as a 3-position valve. The only way to tell the difference in these two configurations is to know the part number designation or to probe the manual overrides. A 3-position valve shifts easily and follows the probe as it is withdrawn. A detented valve shifts hard when breaking out of the detent notches but stays shifted when the probe is withdrawn.*

Most 2-position valves are found in circuits with pressure-compensated pumps because they block flow when an actuator stalls. When actuators need pressure continuously, a 2-position valve simplifies the electrical control circuit. A momentary signal shifts a double-solenoid detented valve to its other position and the valve stays there until it receives the opposite signal. A single-solenoid spring-return valve must have a maintained electrical signal to stay shifted. This solenoid setup causes all actuators to return to home position at the loss of control circuit power or after an emergency-stop signal.

Single-solenoid spring-return or double-solenoid detented valves are commonly used in air circuits because pressure is always available and blocking flow does not cause overheating.

Solenoid pilot-operated valves

All the foregoing valves are operated directly by a solenoid plunger pushing against a spool. All D02-and D03-, and most D05-size valves are direct operated. This arrangement works well for small, low-flow valves, but is not a good setup for high-flow valves with large spools or poppets. Most direct solenoid-operated valves are rated at 20 gpm or less nominal flow. (Nominal flow is usually considered as the maximum flow a valve can pass at 35- to 50-psi pressure drop.) Most manufacturers' literature shows flow and pressure-drop information for all valve sizes and flow paths. Keeping pressure drop low saves energy, reduces shock, makes for a quieter system, and minimizes leakage potential.

For systems with flows higher than 10 to 20 gpm, use solenoid pilot-operated valves with high flow capacity. They offer low pressure drop by incorporating small solenoid-operated valves that hydraulically shift large, high-flow spools. These valves look different physically and also have additional ports. The valves require a minimum pressure of 50 to 75 psi to shift the working spool, especially on spring-return or spring-centered models. A D02, D03, or D05 directly operated, 3-position valve that is spring returned, detented, or has a float center is the control choice. (See Chapter 4 for NFPA, ISO, CETOP, and NG valve size designations, relative physical size, port diameters, port configurations, and nominal flow ratings. For actual flow information and dimensions, always check the supplier's catalog.)

Fig. 10-23. Single-solenoid, 2-position, pilot-operated, spring-return 4-way directional control valve

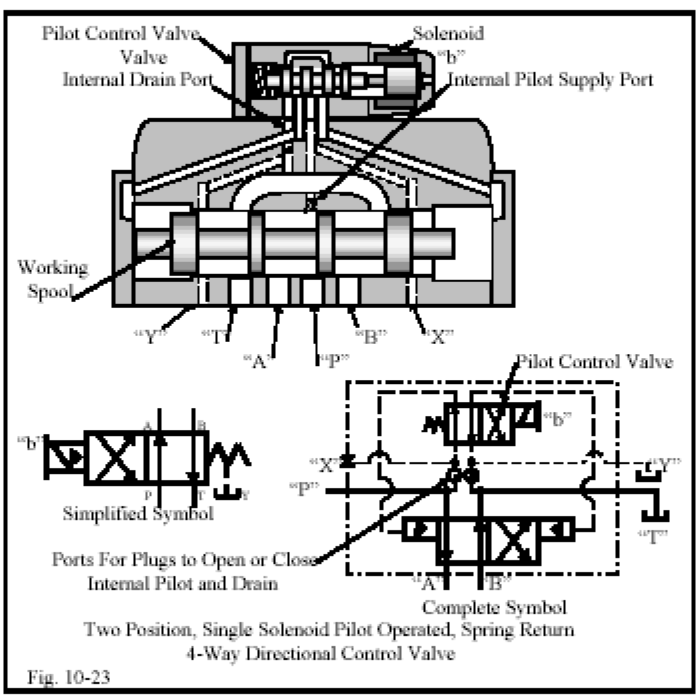


***Figure 10-23** shows a cutaway view and the symbol for a single-solenoid, spring-return, solenoid pilot-operated directional valve. In the normal condition, a D03 single-solenoid spring-*

return, pilot-control valve receives its pilot oil from the X port or the internal pilot-supply port, and sends it to the right end of the working spool. When the pump runs, pilot pressure holds the working spool in place to open flow paths from P to A and B to T. Because system fluid controls the working spool's position, a minimum of 50- to 75-psi pressure must be available at all times. As system pressure rises, pilot pressure also rises but this has little affect on spool shifting time because of flow restrictions and the distance the spool must travel.

*The simplified symbol typically is found on schematic drawings because internal details usually are not important. Notice that the symbol in **Figure 10-23** has a solenoid slash and an energy triangle in the operator box to indicate solenoid pilot operation. The arrows in the symbol show the flow configuration of the working spool because it makes the actuator move. The only indication of the solenoid-operated valve is the solenoid slash mark in the operator box. The pilot control valve must have the right configuration to make the working spool operate, but otherwise is unimportant. In the case of this 2-position, spring-return valve, the pilot-control valve must also be a 2-position, spring-return model.*

The complete symbol includes directional controls, internal porting, and internal connections that are plugged or open. (Later in this text, other features of the complete symbol will be shown and explained.) The enclosure outline around the two valves indicates they are a single piece of hardware. Lines passing through the enclosure outline are external connections and must be plugged if not connected.

Fig. 10-24. Double-solenoid, 2-position, pilot-operated, detented 4-way directional control valve

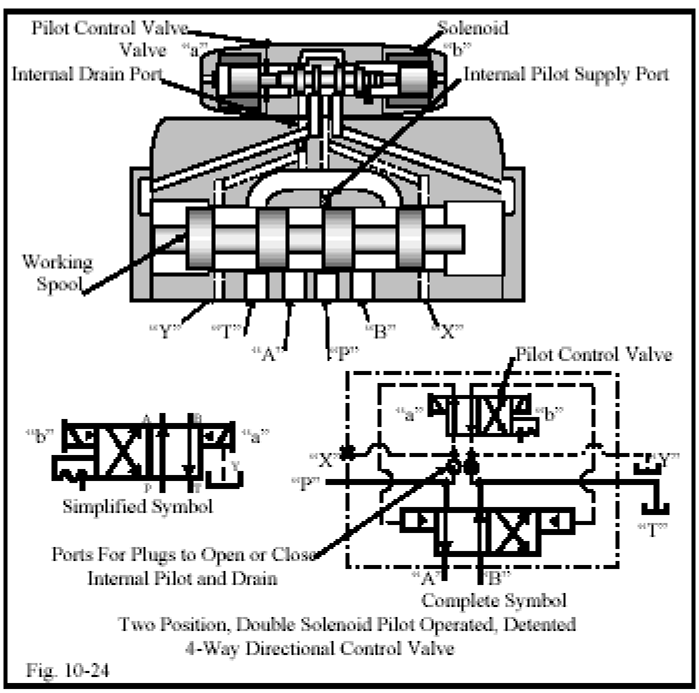


***Figure 10-24** has a cutaway view and symbol for a detented, solenoid pilot-operated directional control valve. The only difference between this valve and symbol and those in **Figure 10-23** is*

that the pilot control is a double-solenoid, detented valve instead of having a return spring. A spring-return valve can be converted to a detented valve merely by changing the pilot control. It is also easy to replace this double-solenoid, detented valve with a double-solenoid, 3-position, spring-centered valve that could allow the working spool to float when both solenoids are de-energized.

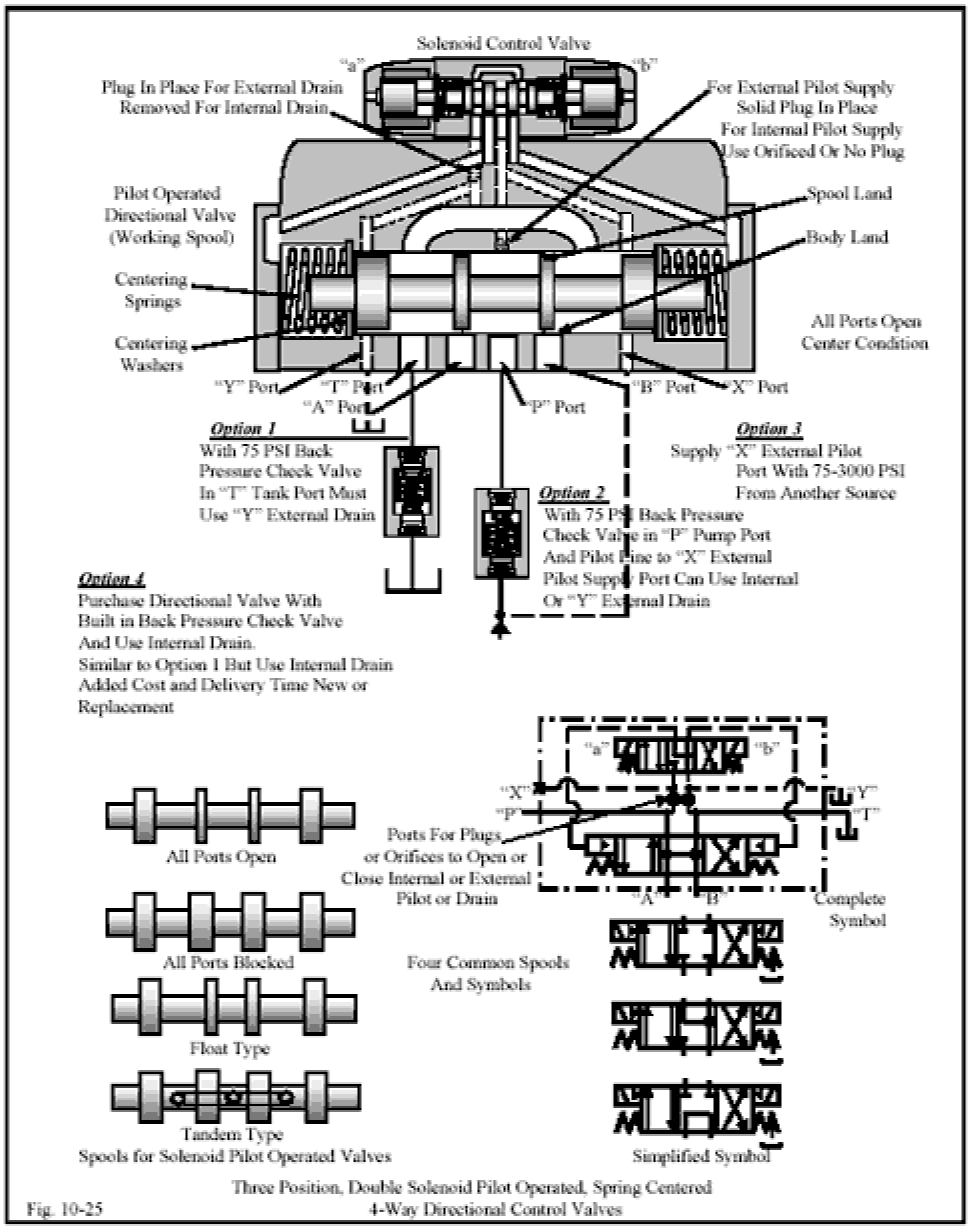
These first two types of solenoid pilot-operated valves must always be mounted so the working spool is horizontal. This keeps the working spool in place when there is no pilot pressure. The working spool could drift when mounted vertically because there are no springs to maintain its position. It is good practice to mount all spool valves horizontally to keep spools in place, especially during shutdown.

The Y (external drain) and X (external pilot supply) ports must be used in some circuits but their use is optional and up to the designer in others. A general rule is to drain pilot oil through the Y port when backpressure at T is (or can be) equal to or higher than pressure at P or X. Backpressure at T is seen at the tank port of the pilot-control valve. It resists pressure from the P or X port that tries to move the spool.

The X port is seldom used except where a constant pilot pressure is necessary. Another possible circuit is where a backpressure check valve in the T port can cause cylinder regeneration when the spool shifts through the crossover or transition position.

The solenoid pilot-operated 3-position, spring-centered valve in ***Figure 10-25*** *is the most common configuration for this type of hydraulic directional control valve. The center condition is often used to unload a fixed-volume pump, stop an actuator or allow it to float, and hold a cylinder at mid stroke while installing or removing tooling. More than 80% of hydraulic directional control valves are 3-position spring-centered for the above reasons and others. All directional valves have a spool that is capable of shifting to a center condition, but 2-position valves never stop in center position when working properly.*

Fig. 10-25. Double-solenoid, 3-position, pilot-operated, spring-centered, 4-way directional control valve



Three Position, Double Solenoid Pilot Operated, Spring Centered
4-Way Directional Control Valves
Fig. 10-25

The cutaway view in ***Figure 10-25*** *represents a solenoid pilot-operated valve with an all-ports-open center condition. These larger, high-flow valves operate in the same way and perform the same functions as the direct solenoid-operated valves discussed earlier. (Refer to* ***Figure's. 10-18*** *through* ***10-20*** *for typical spring-centered circuits.)*

The main difference in the circuits for solenoid pilot-operated valves is that the models with a pump-to-tank center condition need some method to maintain a continuous minimum pilot pressure to the pilot-control valve. ***Figure 10-25*** *shows the standard methods for maintaining such minimum pilot pressure.*

These valves may also need a way to provide a free path for pilot oil to flow to tank (port Y) under some conditions. Adding or removing plugs – or installing an orificed plug -- can stop or allow flow to the internal-drain port or from the internal pilot-supply port.

*The cutaway in **Figure 10-25** shows springs and centering washers at both ends holding the working spool in its center position. The centering washers prevent a stronger spring from pushing the spool past center in case one spring is weaker. The spool in the cutaway view and the complete symbol represent an all-ports-open center condition. Other spools and their simplified symbols are shown below the cutaway.*

The solenoid-control valve is a double-solenoid, spring-centered model with a float-center spool. All 3-position, solenoid pilot-operated, directional control valves must have a float-center spool to work properly. Any other spool center condition (except one with T blocked, and A, B, and P connected) will either not let the working spool center or will generate excessive heat. Always make sure the solenoid-control valve is 3 position and has a float-center spool when the pilot-operated directional valve is 3 position.

When the a solenoid of the solenoid-control valve is energized, it sends pilot oil to the right end of the working spool. This shifts the spool to the left. Pump flow at the P port is directed to the A port, and flow from the B port goes to tank through the T port. It is easy to follow flow paths on the complete symbol to see how any manufacturer's valve functions, even when its construction is different. Energizing the b solenoid shifts the working spool to the right, producing the opposite flow condition and reversing flow to the actuator.

*With the valve in **Figure 10-25** at rest, it is obvious pressure would be low. When the valve and lines are sized for low pressure drop, pressure would probably be below 50 psi. This pressure would not generate enough force to shift the working spool against the centering springs, so no fluid can flow to or from the actuator. To overcome low or no pilot pressure the following options are available:*

Option 1: Use a 75-psi backpressure check valve out of the T port, use an orificed plug in the internal pilot-supply port, and drain the pilot-control valve externally through the Y (external-drain) port. When the pump runs, the backpressure check valve maintains at least 75-psi pilot pressure, so the working spool can shift when signaled. When the circuit operates, pilot pressure may go higher, but it never drops below 75 psi. With an open-center valve, this option can make the cylinder regenerate if the cylinder has low resistance and/or an oversized rod. If this situation is suspect, use Option 2.

Option 2: Install the 75-psi backpressure check valve in the pump line and route a pilot signal from upstream of it to the X (external pilot-supply) port. This makes it possible to internally drain the pilot-control valve. An external drain could still be used, but would not be required. This option keeps pressure off a cylinder that could regenerate with the open-center valve in its center position. It also eliminates the possibility of regeneration when a tandem-center valve moves through an open crossover.

Option 3: Supply the X port with 75- to 3000-psi pilot pressure from another source. The external supply does not need to provide high flow. A constant pressure might be desirable to keep spool shifting and cycle time consistent.

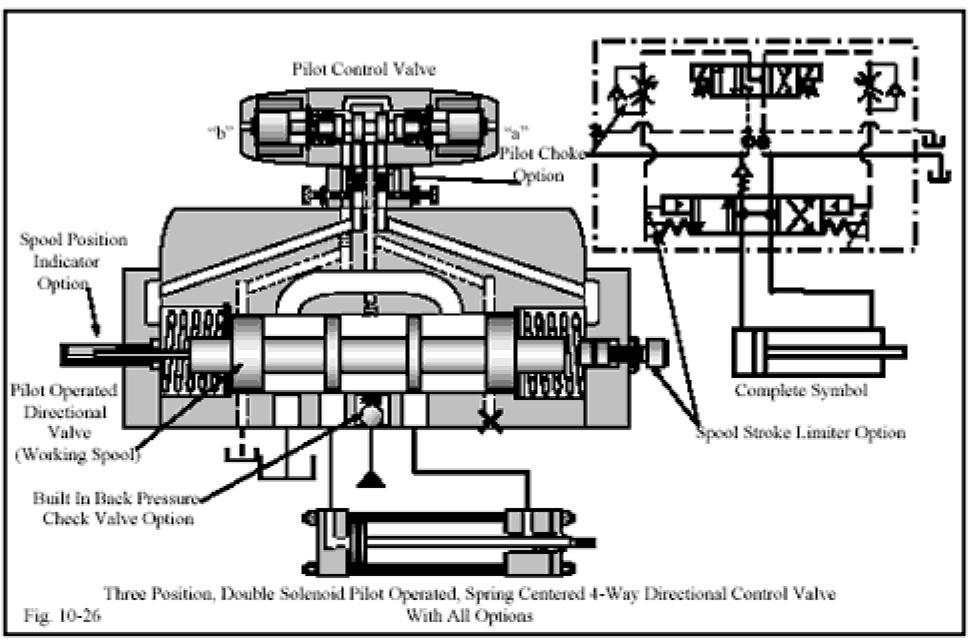
Check the supplier's catalog information to see what pilot pressure should be for a specific valve size and operating pressure. These figures change for different spools and are directly affected by tank line backpressure when using internal drains.

Option 4: Most suppliers offer an internal backpressure check valve option that operates almost the same as Option 1. The difference is that there is no pressure in the T tank line so an internal drain may be used. The main difficulty with this option is that this valve is not standard. Delivery probably is long and could affect machine downtime if the valve needs to be replaced.

Other solenoid pilot-operated valve options

*The solenoid pilot-operated valve in **Figure 10-26** shows some other options that manufacturers offer for special needs. All of these options may not be available from all suppliers, so check with the distributor before specifying a brand.*

Fig. 10-26. Double-solenoid, 3-position, pilot-operated, spring-centered, 4-way directional control valve with all options



The spool-position indicator option helps to troubleshoot a circuit. When the pilot-control valve shifts electrically or manually but the actuator will not move, one reason could be that the working spool did not shift. On a valve without the indicator, the only positive way to know if this spool can move is to take the valve apart and check it.

The spool stroke limiter option imitates a simple flow control for different actuator speeds in both directions. This option limits spool travel, which restricts flow to or from the actuator -- similar to a flow control. This option should only be used where speed can fluctuate as pressure and force change.

The pilot-choke option installs a modular meter-out flow control between the pilot-control valve and the working spool to slow spool movement. Slowing the spool can give an actuator smoother acceleration and deceleration, thus reducing shock. The idea is great, but note that slower spool movement may increase cycle times beyond limits.

One thing that causes cycle time to increase is the fact that all solenoid-operated spool valve lands overlap the body lands. This overlap means the spool has to move some distance before fluid flow starts. When the spool moves slowly enough to give good control, the shift time out of overlap can be 0.5 to 1.0 second or greater. After the spool clears overlap, the actuator can accelerate very smoothly, but the extra time often cannot be tolerated.

Another addition to cycle time comes when the spool shifts to the end of its stroke. A spool can continue to move to the end of its stroke even though a partial stroke is passing all available flow. When reversing actuator motion or decelerating before the end of stroke, the spool may be shifted 1/16 in. or more past available flow. When the spool starts slowly moving to center, the actuator continues at full speed until the spool moves far enough to start restricting flow. From this point on, deceleration is very smooth, but time has been lost. Also remember: the speed at which the spool goes to center is the rate for accelerating the actuator in the opposite direction. This means that adjusting for acceleration both ways also affects deceleration in both ways. The spool stroke-limiter option can eliminate the time loss here, but will not help return shifting speed.

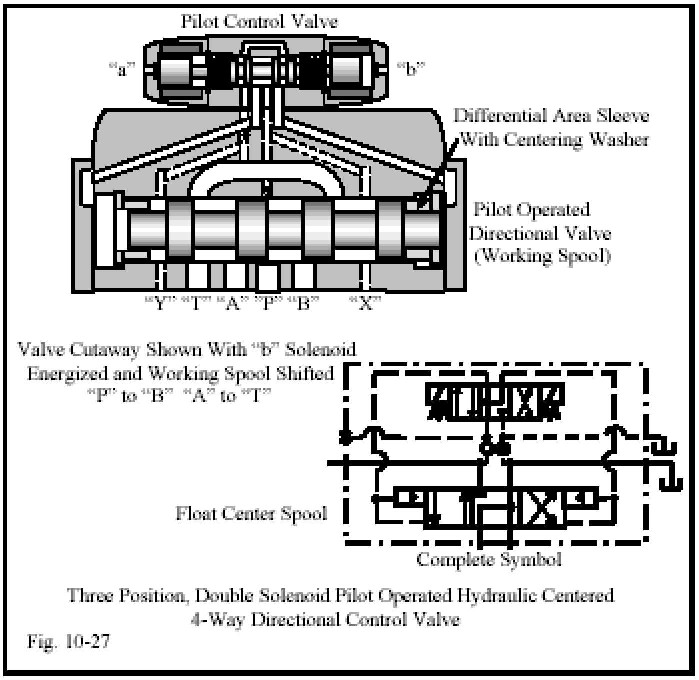
Another possible problem with the pilot-choke module is that it often has a flow rating of only 4 to 8 gpm. Adjusting that small amount of flow is very difficult, if not impossible. This problem limits the usefulness of the pilot-choke option. At present, the need for acceleration, deceleration, and flow variation can be handled better by proportional valves, which will be discussed in Chapter 12.

*The integral backpressure check valve option was discussed as Option 4 on **Figure 10-12**.*

Hydraulically centered valves

*The cutaway view in **Figure 10-27** represents a solenoid pilot-operated directional control valve that is hydraulically centered. A few designers prefer hydraulic centering to spring centering. The reasons given are: spring force changes over time, springs may break, response is slower with springs, and springs are relatively weak. Hydraulic centering has none of these faults, but is still specified on less than 2% of all hydraulic circuits. Part of the reason is lack of knowledge of many designers and users.*

Fig. 10-27. Double-solenoid, 3-position, pilot-operated, hydraulically centered, 4-way directional *control valve*



Notice that in the complete symbol, the pilot-control valve has port T to tank blocked in the center condition, with ports P, A, and B connected. With the pump running, the pilot-control valve sends pilot oil to both ends of the working spool, centering it. The working spool can center because the differential-area sleeves with centering washers can only move until they contact the valve body. With pressure at both ends, these items give a difference in area that causes the working spool to move until it centers. Other than the way the working spool centers, this design valve works the same as other solenoid pilot-operated directional valves.

Valve operators

Figure 10-28 *shows all the operators for directional control valves. Prior to 1966, the operator box on the symbol had letter abbreviations for the method of operating the valve written in them. (See Chapter 4, where old operator symbols are shown across from present day symbols.).*

Fig. 10-28. Symbols for valve operators

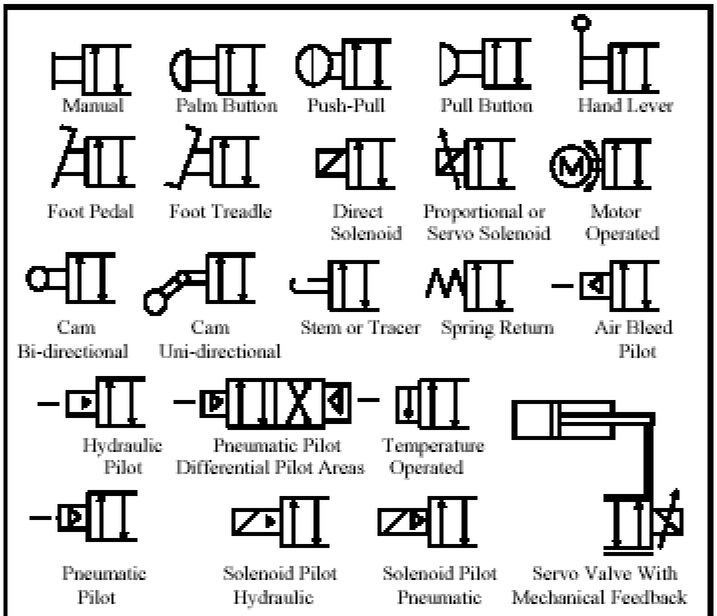


Fig. 10-28 Valve Operators

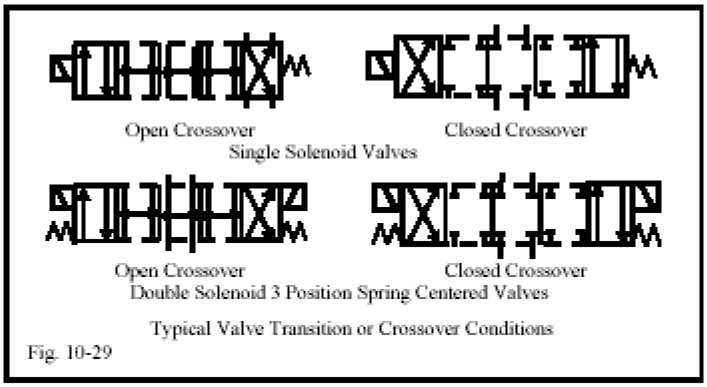
When ISO standards were adopted, all writing was eliminated from the picture-like symbols. The abbreviation MAN, for a manual operator, changed to extended lines at the operator box outer end or to a stick drawing of a palm button, hand lever, foot pedal or treadle.

With picture-like drawings there was no language barrier when schematic diagrams went from one country to another. ANSI adopted the new standards (with a few exceptions) and the fluid power industry changed soon thereafter. Old machines with pre-1966 schematics can confuse newcomers, but the drawing usually can be deciphered with a little effort.

Spool-valve transition conditions

When a spool valve shifts from one flow condition to another, it can pass through different flow conditions than those indicated by straight or crossing arrows in the symbol. These transition or crossover conditions are unimportant in most circuits, but can cause shock, drifting, or cylinder regeneration under certain circumstances. ***Figure 10-29*** *shows typical transition conditions for open- and closed-center, 2- and 3-position valves. For transition conditions of other valve center types, see Chapter 4.*

Fig. 10-29. Symbols for typical valve transition or crossover conditions



An example of how an open transition or crossover can eliminate shock is the case of a fast-moving actuator that must reverse direction before the end of its stroke. With a closed-crossover valve in this circuit, all flow to and from the actuator is blocked for a brief period as the spool shifts from extend to retract. The pump side of the valve always has a relief valve (or a pressure-compensated pump) to protect it from over pressure. However, the actuator side has no protection of any kind. When flow from the extending actuator is blocked, pressure can build to levels well above the pressure ratings of pipes, seals, and hardware. This pressure spike is very brief, but it happens every cycle -- and soon shows up as cracked fittings, blown hoses, leaking seals, or broken parts. Changing to an open-crossover spool in this application would connect the impending spike to the relief valve or pressure-compensated pump. There is still a pressure spike but its intensity is now below the damage level of the components.

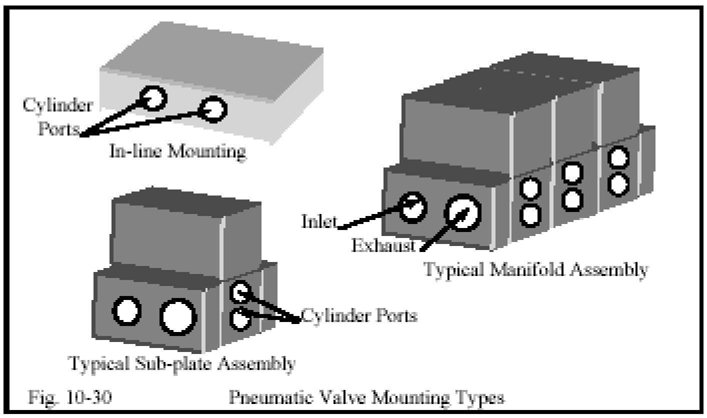
An example of where a closed transition spool helps is when a vertical cylinder with a heavy platen must be reversed in mid stroke. With an open transition, the cylinder will continue its forward travel after the valve receives the reverse signal. With a closed transition, the cylinder will stop almost immediately and start reversing shortly thereafter. Also, when the platen must retract from a stop position, such as during set up, it can drop before being powered up with an open-crossover valve in the circuit.

Crossover problems usually show up when a 3-position valve shifts to or from center condition. Cylinders may move in the opposite direction or move when signaled to stop for no apparent *reason. Most suppliers show crossover flow in their catalog so check it out if the problem is an unusual movement.*

Mounting pneumatic valves

*Pneumatic valves can be line-, subplate-, or manifold-mounted. Inline or bar-type manifolds make it convenient to stack valves with a common inlet and/or exhaust. Pneumatic circuits seldom, if ever, require custom manifolds as hydraulic circuits do. The graphics in **Figure 10-30** depict some pneumatic-valve mounting styles. In-line mounting types have the whole valve assembly in a body with ports out the sides. This style is usually less expensive, but is more trouble if it has to be replaced.*

Fig. 10-30. Mounting arrangements for pneumatic valves



The typical pneumatic subplate assembly is often a subplate with end covers bolted to it. Fasteners hold the parts together and molded seals eliminate leaks. Valves with seals mount to the subplate and all piping connects to it. Some manufacturers have wiring troughs in the subplate and use plug-in connectors on solenoid-operated valves.

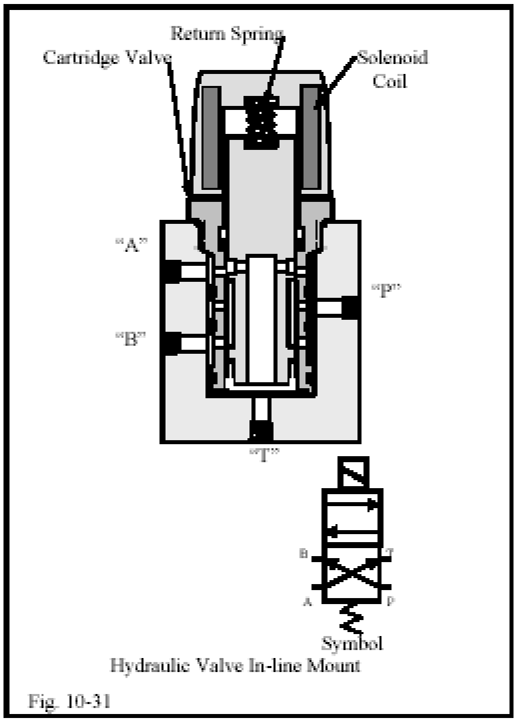
The typical pneumatic manifold assembly consists of two or more subplates connected to make a valve stack with a common inlet and exhaust. These assemblies eliminate many connections and make valve installation replacement easy. These units also are available with wiring troughs and plug-in valves for solenoid operation. Most air valves use unique mountings and port arrangements that are not inter-changeable. However, there is an ISO standard subplate mounting that several companies offer. The valves match each other's mounting patterns but otherwise do not have interchangeable parts. This assembly is physically large and thus more expensive, but it makes it easy to combine valves from several suppliers.

Mounting hydraulic valves

*In-line and subplate-mounted hydraulic valves are common. However, most in-line valves are screw-in cartridge type with aluminum or steel bodies. **Figure 10-31** shows an example of an in-*

line cartridge valve. (This valve also could be screwed into the custom manifolds discussed later.)

Fig. 10-31. Cross-sectional view of in-line mounted hydraulic valve



Screw-in cartridge valves perform all directional and flow control, relief, sequence, counterbalance and reducing functions -- the same as in-line and subplate valves. Only their physical makeup is different. They normally handle flows less than 40 gpm, but some manufacturers offer sizes with up to 120-gpm capacity.

There are worldwide interface standards for subplate-mounted hydraulic directional control valves. The information in ***Figure 10-32*** *shows port and bolt locations and relative sizes for all standard sizes. The figure also lists the numbering systems for U. S. National Fluid Power Association (NFPA), worldwide International Standards Organization (ISO), European Committee for Oilhydraulics and Pneumatic Control (CETOP), and the NG part of the German DIN Standard, which relates only to port size in metrics. (Actually, the NG port size can be for any type valve.)*

Fig. 10-32. NFPA and ISO standard interface layouts for hydraulic directional control valves

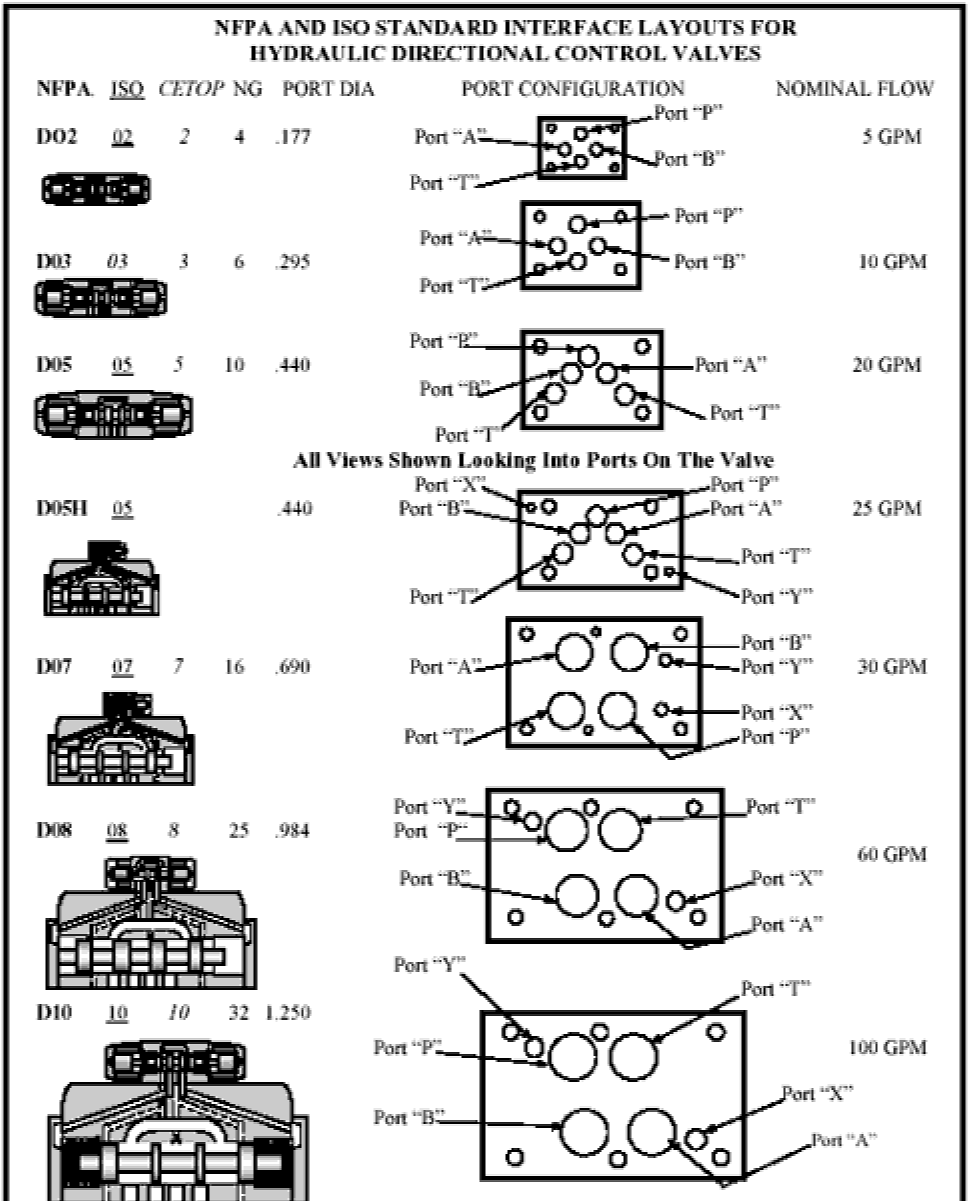
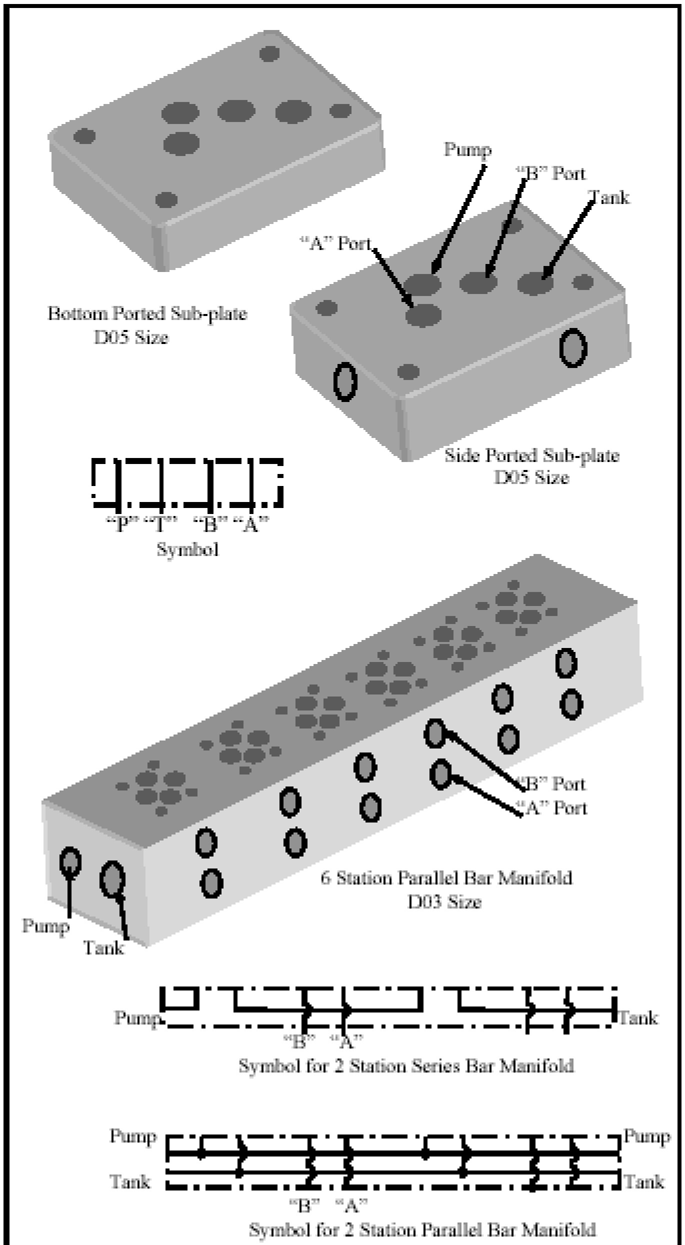


Fig. 10-32

Interface standards cover both size and location for ports and bolts. A directional valve from any country or manufacturer using this standard is interchangeable with all other valves of the same size. The only difference should be whether the bolts have SAE or metric threads.

Figure 10-33 *depicts typical subplate and bar-manifold mounting arrangements for subplate-mounted directional control valves. Subplates mount a single valve and are used in simple single-cylinder applications. They come in bottom- and side-ported models for piping convenience. Some side-ported models have bottom ports as well.*

Fig. 10-33. Subplates and bar manifold for mounting



Hydraulic Valve Sub-plates and Bar Manifolds

Fig. 10-33

Subplates can be used for multi-cylinder circuits but require a lot of pipe connections that can restrict flow and may be potential leakage points. Many multi-cylinder circuits work well with the bar manifold shown in ***Figure 10-33****.*

Subplates are available for all the valves listed in ***Figure 10-34****. Porting for larger valves usually involves SAE flanges on the valve body. When they have subplate mounts, a special subplate must be made or they are mounted on a custom manifold.*

Bar manifolds come with series and parallel porting related to pump and tank connections. Series manifolds usually are limited to three stations or three valves maximum, while parallel manifolds can have as many as 16 stations.

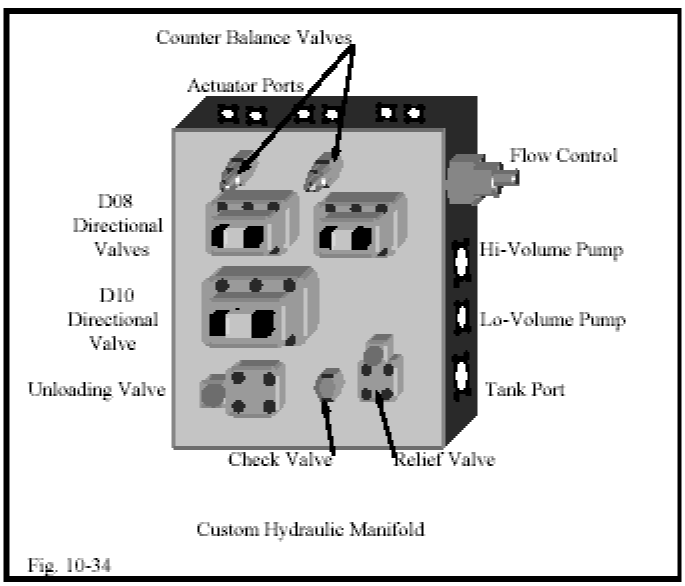
When circuits are not too complex, bar manifolds and modular accessory valves can eliminate most pipe connections and put everything in one location. Symbols for these modular valves are shown for most of the types available. They are always at the end of a section for a particular accessory valve.

Bar manifolds are only offered in sizes D02, D03, D05, D05H, D07 and D08. Size D10 valves use subplates or specially made bar manifolds, or are mounted on a custom manifold.

Custom manifolds

Figure 10-34 *depicts a custom manifold that can eliminate many plumbing connections and make valve replacement easy. Such manifolds are usually more expensive than simple plumbing, but can save many times the extra first cost by minimizing fluid loss through leakage and the cost of cleanup.*

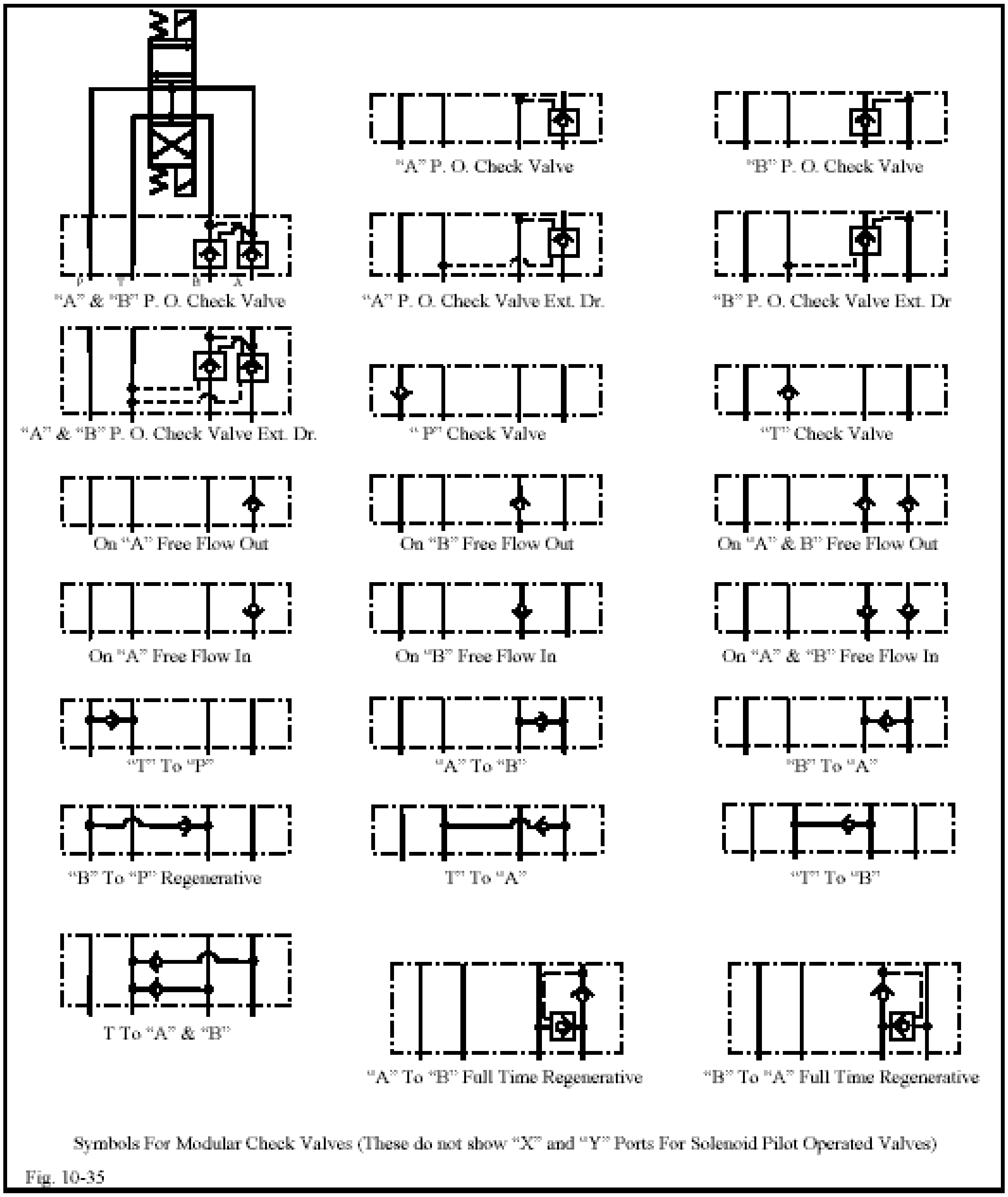
Fig. 10-34. Custom hydraulic manifold



A number of companies specialize in designing and building custom manifolds. All they require is a schematic of the circuit with the valves to be included inside chain lines; plus other information such as preferred pipe connection type, connection locations and size, which face or faces to leave clear and manifold material. If there is a preferred mounting arrangement, it should be noted along with mounting bolt locations.

Remember: manifolds are difficult to modify, so they should only be applied to proven working circuits. If in doubt about the viability of a new design, work out the bugs in standard piping before buying a manifold.

Fig. 10-35. Symbols for most common modular check valves



Most check valve functions are available as modular or sandwich valves that mount between the directional control valve and subplate. ***Figure 10-35*** *shows most of the common configurations presently offered by fluid power suppliers. These modules are usually available in all valve port sizes up to D08 (3/4 in.).*

Directional Control Valves

Directional control valves perform only three functions:

- stop fluid flow
- allow fluid flow, and
- change direction of fluid flow.

These three functions usually operate in combination.

The simplest directional control valve is the 2-way valve. A 2-way valve stops flow or allows flow. A water faucet is a good example of a 2-way valve. A water faucet allows flow or stops flow by manual control.

A single-acting cylinder needs supply to and exhaust from its port to operate. This requires a 3-way valve. A 3-way valve allows fluid flow to an actuator in one position and exhausts the fluid from it in the other position. Some 3-way valves have a third position that blocks flow at all ports.

A double-acting actuator requires a 4-way valve. A 4-way valve pressurizes and exhausts two ports interdependently. A 3-position, 4-way valve stops an actuator or allows it to float. The 4-way function is a common type of directional control valve for both air and hydraulic circuits. A 3-position, 4-way valve is more common in hydraulic circuits.

The 5-way valve is found most frequently in air circuits. A 5-way valve performs the same function as a 4-way valve. The only difference is an extra tank or exhaust port. (Some suppliers call their 5-way valves, "5-ported 4-ways.") All spool valves are five ported, but hydraulic valves have internally connected exhaust ports going to a common outlet. Because oil must return to tank, it is convenient to connect the dual tank ports to a single return port. For air valves, atmosphere is the tank, so exhaust piping is usually unimportant. Using two exhaust ports makes the valve smaller and less expensive. As will be explained later, dual exhausts used for speed-control mufflers or as dual-pressure inlets make this configuration versatile.

Following are schematic symbols for commonly used directional control valves.

2-way directional control valves

A 2-way directional valve has two ports normally called inlet and outlet. When the inlet is blocked in the at-rest condition, as shown in Figure 8-1, it is referred to as "normally closed" (NC). The at-rest box or the normal condition is the one with the flow lines going to and from it.

The boxes or enclosures represent the valve's positions. In Figure 8-1, the active box shows blocked ports, or a closed condition, while the upper box shows a flow path. When an operator shifts the valve, it is the same as sliding the upper box down to take the place of the lower box. In the shifted condition there is flow from inlet to outlet. Releasing the palm button in Figure 8-1 *allows the valve spring to return to the normal stop flow condition. A 2-way valve makes a blow-off device or runs a fluid motor in one direction. By itself, a 2-way valve cannot cycle even a single acting cylinder.*

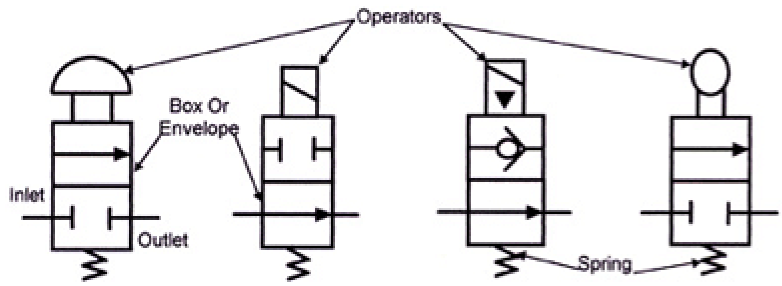


Figure 8-2 shows a "normally open" (NO) 2-way directional valve. Energizing the solenoid on this valve stops fluid flow.

Valve operators come in different types. Figure 8-3 shows a solenoid pilot operator using solenoid-controlled pressure from the inlet port to move the working directional spool. Figure 8-4 shows a cam-operated valve. A moving machine member usually operates this type valve.

3-way directional control valves
A 3-way valve has three working ports. These ports are: inlet, outlet, and exhaust (or tank). A 3-way valve not only supplies fluid to an actuator, but allows fluid to return from it as well. Figures 8-5 through 8-10 show schematic symbols for 3-way directional control valves.

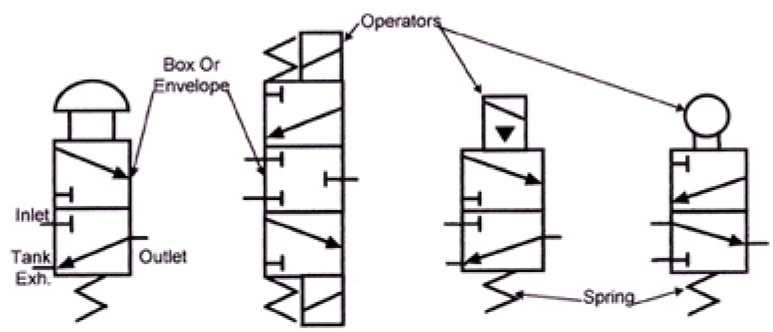


Figure 8-6 depicts an all-ports-blocked, 3-way, 3-position valve. A valve of this type connected to a single-acting, weight- or spring-returned cylinder could extend, retract, or stop at any place in the stroke.

Some 3-way valves select fluid flow paths as in Figure 8-9. Use a spool-type valve for this operation. Another flow condition is the diverter valve shown in Figure 8-10. A diverter valve sends fluid to either of two paths.

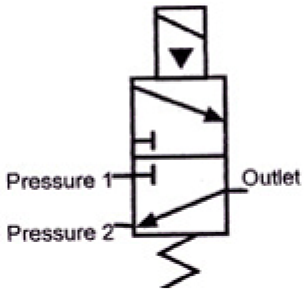


Figure 8-9. Solenoid pilot-operated 3-way selector valve.

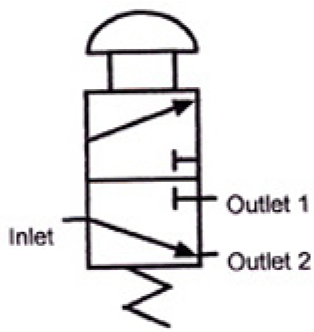


Figure 8-10. Palm-button-operated 3-way diverter valve.

4-way directional control valves

Figures 8-11 to 8-15 show different configurations available in 4-way directional control valves. They range from the simple, two-position, single, direct solenoid, spring-return valve shown in Figure 8-11, to the more complex three-position, double solenoid, pilot-operated, spring-centered, external-pilot supply, external drain valve shown in Figure 8-15.

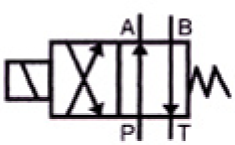


Figure 8-11. 4-way, 2-position direct solenoid-operated

spring return.

Lines to the boxes show flow to and from the valve, while lines with arrows in the boxes show direction of flow. The number of boxes tells how many positions the valve has.

INSERT FIG 8-12

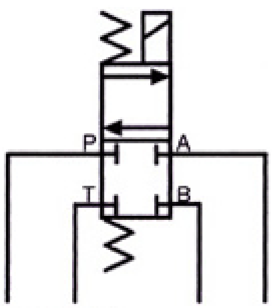


Figure 8-12. 4-way, 2-position direct solenoid-operated spring centered.

Figure 8-12 shows a single solenoid, spring-centered valve. This valve has a third position but there is no operator for it. Use this spring-centered, single solenoid valve in control circuits for special functions. In the past, to get this configuration, you only had to wire one solenoid of a double-solenoid, three-position valve.

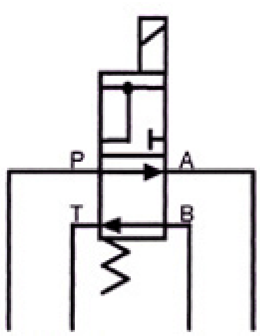


Figure 8-13. 4-way, 2-position direct solenoid-operated

spring return.

Figure 8-13 shows another unusual 4-way configuration. This valve shifts from an actuator moving flow path to center condition for certain special circuits.

5-way directional control valves

Figures 8-16 through 8-20 show symbols of some 5-way air valves. Most spool-type air valves come in a 5-way configuration. Because air usually exhausts to atmosphere, the extra exhaust port is no problem.

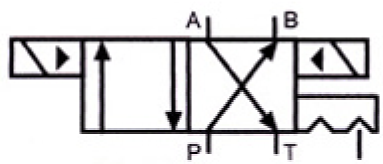


Figure 8-14. 4-way, 2-position solenoid, pilot operated detented, line-mounted.

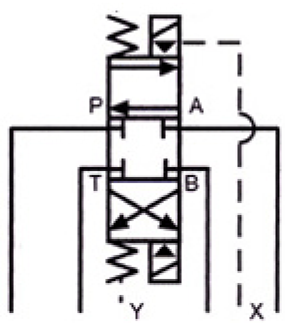


Figure 8-15. 4-way, 3-position solenoid, pilot operated centered, manifold-mounted.

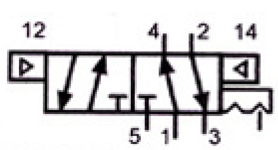


Figure 8-16. Line-mounted, air-pilot-operated, 2-position, detented 5-way valve.

Many valves use the two exhaust ports for speed control mufflers. Mufflers not only make the exhaust quieter, but throttle the exhaust, which in turn controls cylinder speed in a meter-out circuit.

Another example later in this section shows dual exhaust ports piped with different pressures to save air. Also use dual inlet piping to make an air cylinder operate quickly and smoothly. (See Figures 8-48 through 8-55.)

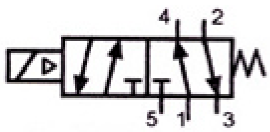


Figure 8-17.Line-mounted, solenoid pilot-operated, 2-position,spring returned 5-way valve.

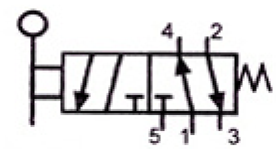


Figure 8-18. Line-mounted, hand lever-operated, 2-position, spring-returned 5-way valve.

Most air cylinders stroke from one extreme to the other. A two position, single solenoid, spring return valve is sufficient for this operation. About 90% of air circuits use this type of valve. To stop an air cylinder in mid-stroke, use the 3-position valve shown in Figures 8-19 through 8-21.

It is difficult — if not impossible — to accurately stop an air cylinder any place other than at end of the stroke. When the cylinder moves slowly, a repeatable mid stroke position of plus or minus an inch might be possible. The problem is, if the load on the cylinder changes or there is any slight leak in the piping or seals, it will not hold position once it stops.

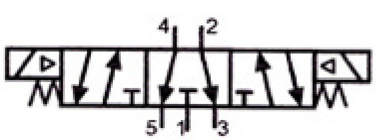


Figure 8-19. 5-way, 3 position, spring-centered solenoid, pilot-

operated, cylinder ports open-center condition, line mounted.

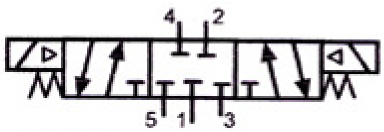


Figure 8-20. 5-way, 3 position, spring-centered solenoid, pilot-operated, all ports blocked center condition, line mounted.

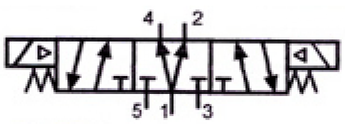


Figure 8-21. 5-way, 3 position, spring-centered pressure to cylinder ports, exhausts blocked center condition, solenoid-pilot operated, line mounted.

Three-position valves come in several styles, including: cylinder ports open as seen in Figure 8-19; all ports blocked as seen in Figure 8-20; and pressure to cylinder ports as seen in Figure 8-21.

Using 2-way valves

Figures 8-22, 8-23, and 8-24 show some uses for 2-way directional control valves.

One use is the blow-off function shown in ***Figure 8-22****. A 2-way valve in* ***Figure 8-23*** *operates a one-direction motor with an open exhaust in the motor housing. The circuit in* ***Figure 8-24*** *works well for electrically unloading a pump for easy start up and/or reduced heat generation*

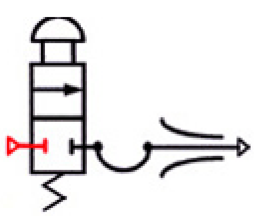


Figure 8-22. Blow-

off.

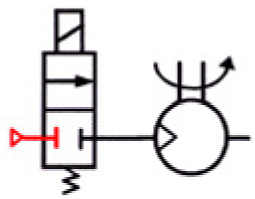


Figure 8-23. Running a one-direction fluid motor.

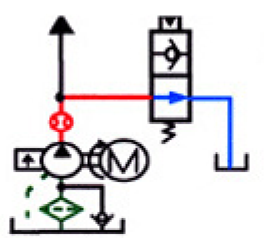


Figure 8-24. Unloading a pump.

Figure 8-25 *shows a weight-returned, single-acting cylinder powered by a 2-way in the ****at rest**** condition. At first sight it looks as if this circuit might work. Shifting the 2-way valve, or* ***extending****, sends fluid to the cylinder cap end and it extends. The problem comes when the 2-way returns to normal at the* ***end of cycle****. Instead of the cylinder retracting after the solenoid de-energizes, it stays in the extended position. The cylinder would only return if the valve, cylinder seals, or pipe connections leak.*

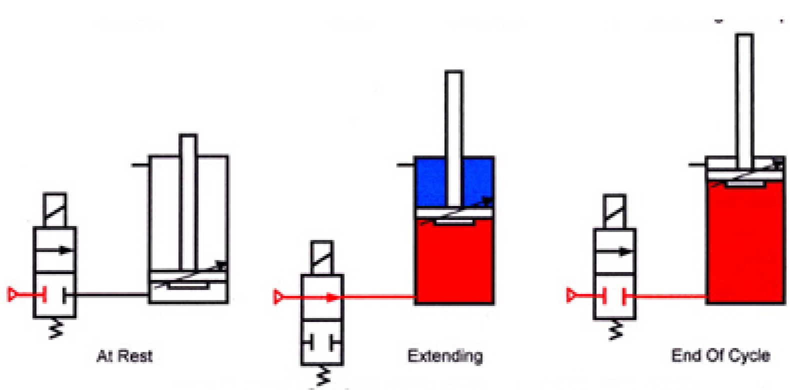


Figure 8-25. Using one 2-way valve to operate a single-acting

cylinder.

Figure 8-26 *shows a circuit that operates a single-acting cylinder with 2-way valves. One (NO) and one (NC) 2-way directional valve piped to the cap end cylinder port allows fluid to enter and exhaust from it. Actuating both operators simultaneously extends the cylinder. According to valve size and inlet air flow, the cylinder might not extend if just energizing the (NC) valve. If the cylinder extends with only one valve actuated, it would be slow and waste a lot of air.*

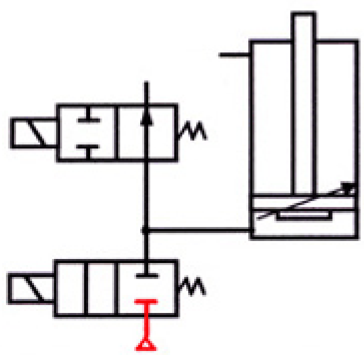


Figure 8-26. Operating a single-acting cylinder with two 2-way valves.

Figure 8-27 *shows four 2-way valves piped to operate a double-acting cylinder. A pair of 2-way valves at each cylinder port gives a power stroke in both directions. Energize and de-energize all four valves simultaneously to cycle the cylinder and keep from wasting fluid.*

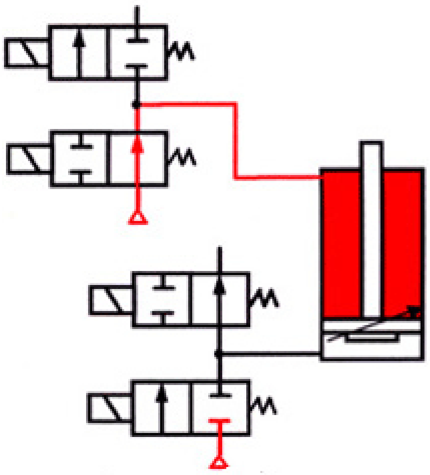


Figure 8-27. Operating a single-acting cylinder with four 2-way valves.

Four 2-way valves may seem to be a complex and expensive way to operate a cylinder. However, in the past few years, poppet type slip-in cartridge valves have been operating large bore hydraulic cylinders this way. See chapter four on Cartridge Valves for the advantages of these valves in high flow circuits.

Using 3-way valves

Figure 8-28 shows a 3-way valve, used to select *Pr. 1* or *Pr. 2*. Use a spool type directional control valve in this type of circuit. Spool valves normally take pressure at any port without malfunction. Poppet design valves normally take pressure at the inlet port only.

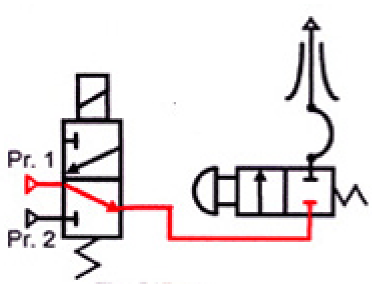


Figure 8-28. Pressure selector.

Since the example selector valve is solenoid pilot-operated, it is important to determine which port has the higher pressure. Most solenoid pilot-operated valves take air from the normal inlet port to operate the pilot section. If both inlet pressures are too low to operate the valve, plumb an external pilot supply from the main air system.

When it is necessary to lock out one of two circuits while the other one operates, the hookup in Figure 8-29 works well. While circuit one has fluid going to it, working on circuit two is no problem. Use a spool type valve here also. Poppet valves usually only take pressure at one port.

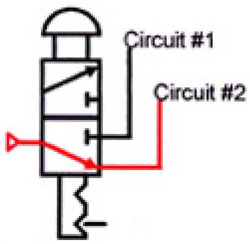


Figure 8-29. Fluid diverter.

The most common limit valve is a miniature 3-way like the one shown in Figure 8-30. This particular example is (NC). Contact with a machine member opens it. Except for bleeder type control circuits, a limit valve requires at least a 3-way function. Once this normally closed valve shifts, it passes a signal on to continue the cycle. In normal condition, fluid in the control circuit exhausts through the exhaust port.

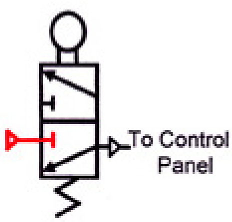


Figure 8-30. NC limit valve.

Figure 8-31 shows a single-acting cylinder with a 3-way valve powering it. Energizing the solenoid, or extending, allows flow to move to the cylinder port and it extends. Deenergizing the solenoid or retracting, lets the valve shift to home position, and the cylinder retracts from outside forces. The exhaust port on a 3-way valve lets fluid in the cylinder escape to atmosphere.

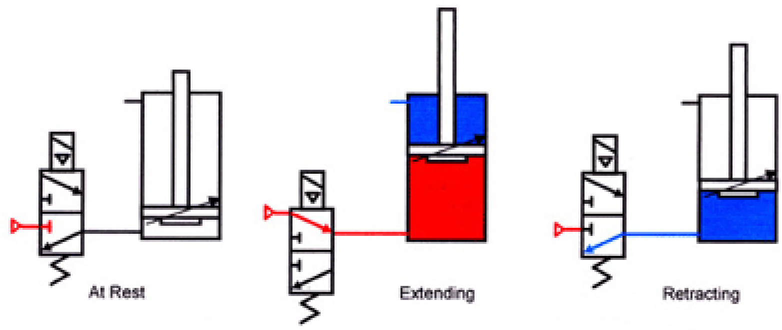


Figure 8-31. Operating a single-acting cylinder with one 3-way valve.

To operate a double-acting cylinder with 3-way valves, use the hookup shown in Figure 8-32. With a 3-way directional valve at both ports, both extend and retract strokes of a double-acting cylinder have force.

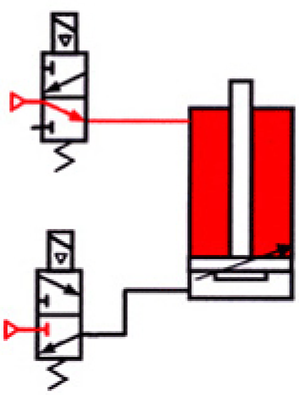


Figure 8-32. Operating a double-acting cylinder with two 3-way valves.

Some manufacturers use dual 3-way valves to conserve air. Piping between the valve and cylinder ports wastes air. Every time a cylinder cycles, the lines to both ports fill and exhaust. The longer the valve-to-cylinder lines are, the greater the air waste. Mounting air valves directly to the cylinder ports minimizes air waste. The higher cycle rate results in greater savings.

Lowering pressure at the rod end port is another way to save air with dual 3-way valves mounted directly to the cylinder port. As discussed before, reducing air pressure at the cylinder *uses less compressor horsepower. Usually, force required to return a cylinder is minimal, so lower pressure at the rod port saves energy.*

Speed-control mufflers in the direct-mounted 3-way valves independently control the extend and retract speed of the cylinder. This saves piping time and the cost of flow control valves.

***Figure 8-33** shows an air cylinder inching circuit**.** It is possible to inch an air circuit if accuracy and repeatability are not important. An inching circuit's repeatability is usually not closer than ±1 in. if travel speed is slow. Faster travel speeds give less control.*

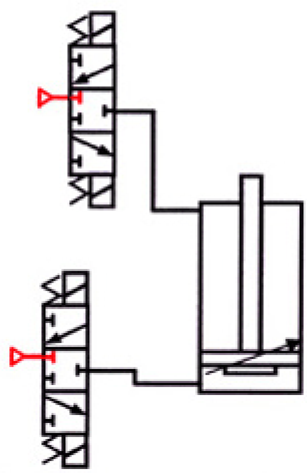


Figure 8-33. Inching circuit for a double-acting cylinder with two 3-way spring-centered valves.

A 3-way valve can replace a 2-way valve. To duplicate the 2-way function, block the exhaust port of the 3-way valve. Blocking the exhaust of a 3-way is usually not necessary for most 2-way applications. Using 3-way valves in place of 2-way valves reduces inventory cost and saves time.

Using 4-way valves

See Figures 8-34 to 8-36 for some uncommon uses of 4-way directional control valves. Using directional controls in ways other than normal is a common practice. Make sure the valve is capable of pressure in all ports before applying it to some of these circuits. If the valve is solenoid pilot-operated, where does pilot supply come from? Also check with the manufacturer if there is any doubt about the valve's performance in an unusual application.

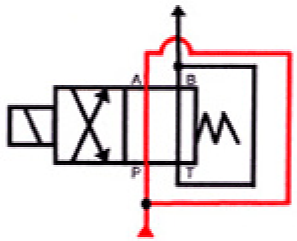


Figure 8-34. Double flow capacity.

To make a high flow 2-way valve from a 4-way valve try the circuit shown in Figure 8-34. Connect pump flow to the normal inlet port and its outlet port, then connect the other outlet port to the normal tank port and on to the system. In the at-rest condition there is no flow through the valve. When the valve shifts, flow is fromP through B to system and from A through T to system. A valve rated at 10 gpm is now good for 20 gpm with little or no increase in pressure drop. Make sure the valve is capable of backpressure at the tank port.

This piping arrangement comes in handy in hydraulic circuits, since most manufacturers do not offer a 2-way valve. Also, a lot of 2-way hydraulic valves only stop flow in one direction, so they are useless in a bi-directional flow line.

For a full time regeneration circuit, pipe the 4-way as shown in Figure 8-35. Read Chapter 17 for a full explanation of this regeneration circuit.

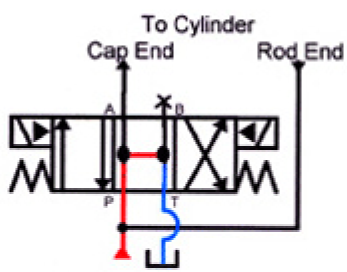


Figure 8-35. Full time regeneration.

Figure 8-36 shows how to pressurize both ends of the cylinder when a 4-way valve centers. When a cylinder retracts to pick up another part, it often has to go too far to make sure it is behind the part. Low backpressure from the check valve makes the cylinder creep forward at low power so the cylinder is in contact with a part before the next cycle starts.

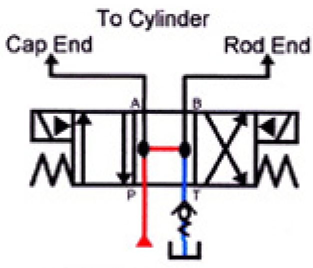


Figure 8-36. Low-pressure cylinder extend.

Figure 8-37 shows the normal hookup of a 4-way directional valve. A double-acting cylinder only needs one 4-way directional valve to extend and retract it. The three sequences show a 4-way valve in action. Add flow controls or a counterbalance valve to complete the circuit when there is weight on the rod. Note the port hookup is A to cap and B to rod. Using this port connection arrangement consistently makes it is easy to wire the circuit because the electrician knows A solenoid extends the cylinder while B solenoid retracts it. Maintenance persons always know which manual override to push during trouble shooting or setup.

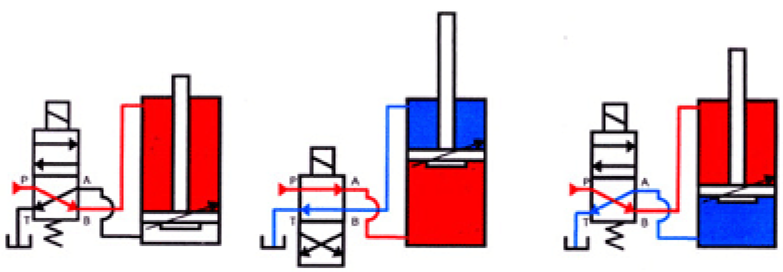


Figure 8-37. Operating a double-acting cylinder with one 4-way valve.

Most hydraulic directional control valves are 3-position. Valve center conditions perform different functions in relation to the actuator and pump.

An all-ports open center condition directional valve unloads the pump and allows the actuator to float as shown in Figure 8-38. This reduces heat build up and allows opposing forces to move the cylinder without building backpressure.

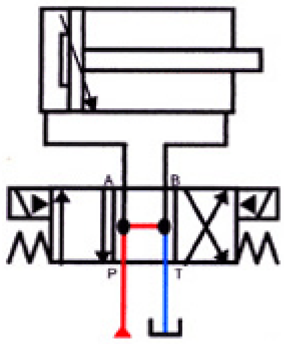


Figure 8-38. Inching circuit with pump unloaded, cylinder floating.

To block the cylinder while unloading the pump, use the center condition shown in Figure 8-39. Most hydraulic valves are a metal-to-metal fit spool design, so do not depend on the cylinder setting dead still with a tandem center spool. If there are outside forces on the cylinder, it will creep when the valve centers.

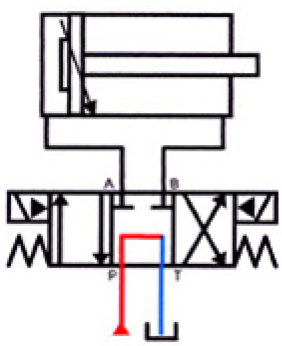


Figure 8-39. Inching circuit with pump unloaded, cylinder blocked.

If the cylinder needs to float while blocking pump flow, use the center condition shown in Figure 8-40.

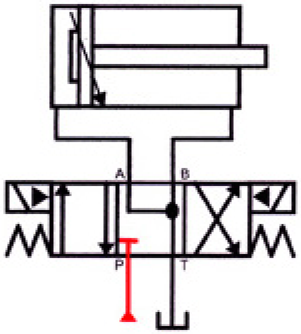


Figure 8-40. Inching circuit with pump blocked, cylinder floating.

Figures 8-41 to 8-46 show several commonly used 4-way hydraulic valve center conditions. The first four account for about 90% of all 3-position hydraulic valves in use.

The center condition of a 3-position valve can unload a pump, open actuator ports to tank for free movement, block actuator ports to stop movement, give regeneration, or work in combinations of these functions.

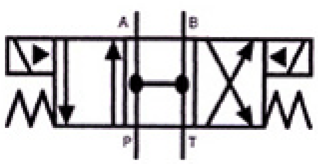


Figure 8-41. All ports open, center condition.

Figure 8-41 shows an all-ports-open center condition valve. The open center condition unloads the pump and allows the actuator to coast to a stop or float. In the crossover or transition condition it causes very little shock. Fixed volume pumps use this center condition.

The all-ports-blocked center condition valve of Figure 8-42 appears to block the cylinder ports. In actual use, leakage oil across the spool lands pressurizes A and B ports, possibly causing a single rod cylinder to extend. This is not a good choice for stopping and holding a cylinder as the symbol seems to indicate. To positively stop a cylinder, use a valve with the cylinder ports *hooked to tank, and pilot-operated check valves in the cylinder line or lines. (See the section on "Check Valves as Directional Valves.")*

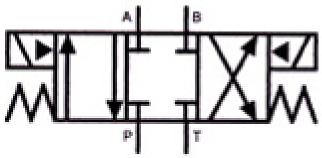


Figure 8-42. Ports blocked, center condition.

The float center valve of Figure 8-43 allows the actuator to float while blocking pump flow. Pump output is available for other valves and actuators with this center condition. It also works well for pilot-operated check valve locking circuits or with counterbalance valves. This is the normal center condition for the solenoid valve on a solenoid pilot-operated, spring-centered directional valve.

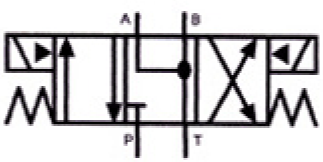


Figure 8-43. Float center condition.

Figure 8-44 shows a tandem center valve. A tandem center valve lets the pump unload while blocking the cylinder ports. The cylinder sits still unless there is an outside force trying to move it. Any metal-to-metal fit spool valve never fully blocks flow. With external forces working on the cylinder, it may slowly creep with the valve centered. This is another common center condition for fixed volume pumps.

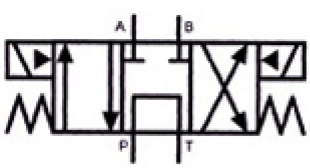


Figure 8-44. Tandem center condition.

The regeneration center position of the valve in Figure 8-45 pressurizes and connects both ports of a cylinder to each other. Connecting pressure oil to both cylinder ports and to each other regenerates it forward when the valve centers. This valve is the pilot operator for hydraulically centered directional valves or normally closed slip in cartridge valves.

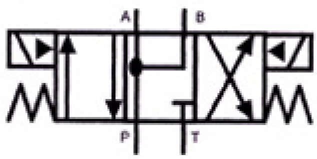


Figure 8-45. Regeneration center condition.

To unload the pump while blocking the cylinder from moving, use the valve shown in Figure 8-46. However, the metal-to-metal fit spool will not lock the cylinder when there are external forces.

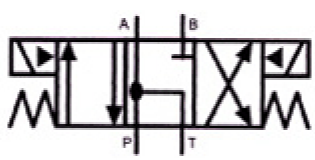


Figure 8-46. Pump unload, port B blocked, center condition.

Figures 8-47 to 8-48 show what is commonly referred to as the "crossover" or "transition" condition of a spool. In some actuator applications it is important to know what the valve port flow conditions are as it shifts. As shown in these figures, dashed lined boxes show crossover condition. Normally discussions about crossover conditions cover "open" or "closed" types; in reality, the crossover condition may be a combination of these and may be different on either side of center. Open crossover stops shock while the spool shifts, while a closed crossover reduces actuator override travel. If the crossover condition is important to the circuit or machine function, show it on the schematic drawing.

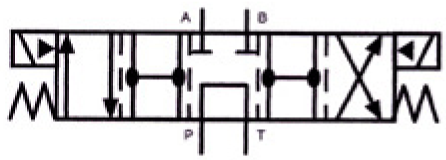


Figure 8-47. Open crossover or transition condition.

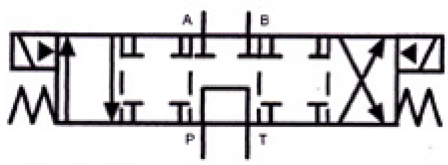


Figure 8-48. Closed crossover or transition condition.

Figure 8-49 shows an all ports blocked center condition, solenoid pilot-operated valve, as a simplified and complete symbol. On most schematics, the simplified symbol is sufficient. The solenoid slash and energy triangle in the operator box show the valve has a solenoid operated valve piloting a pilot-operated valve. The boxes show the function of the main or working spool that controls the actuator. On valves with other hardware added (here, pilot chokes and stroke limiters), it is better to show the complete symbol. Both symbols in Figure 8-49 represent the same valve. The complete symbol gives more information about the valve function and helps with troubleshooting and valve replacement.

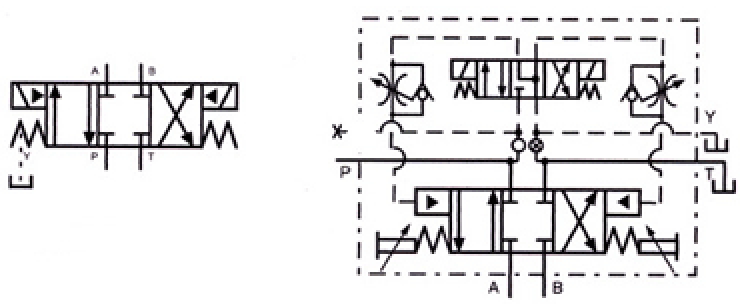


Figure 8-49. Solenoid pilot-operated valve with pilot chokes and stroke limiters. Internal pilot supply (X) and external drain (Y).

Using 5-way air valves

The 5-way selector valve and shuttle valve in Figure 8-50 works where a 3-way selector may not. The 3-way selector does fine when going from low to high pressure, but if there is no air usage to allow expansion, it is almost impossible to go from high to low pressure.

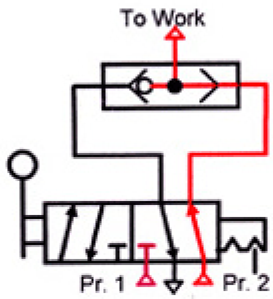


Figure 8-50. Pressure selector.

The 5-way and shuttle valve arrangement gives an exhaust path for high-pressure air when shifting to low pressure. After the air exhausts to the lower pressure, PR.1, the shuttle shifts and low pressure holds in the system.

Figure 8-51 shows a pair of 5-way valves piped to act like a three way light switch. Either valve moves the cylinder to its opposite position when activated.

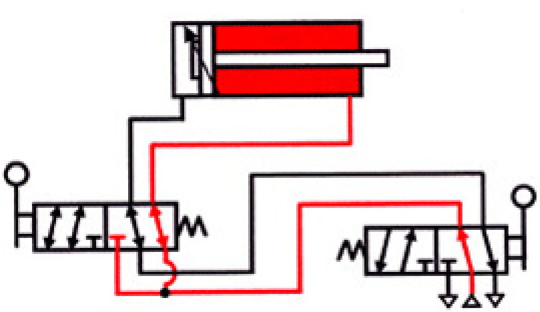


Figure 8-51. Operating an actuator from two locations.

Figure 8-52 shows the normal hookup of a 5-way valve. Normally, input air goes to the center port of the side with three ports. A lot of air valve manufacturers call this #1 port. In the at rest *condition, air flows from #1 to #4 port and on to the cylinder rod end, while #2 port exhausts the cylinder cap end through #3 port.*

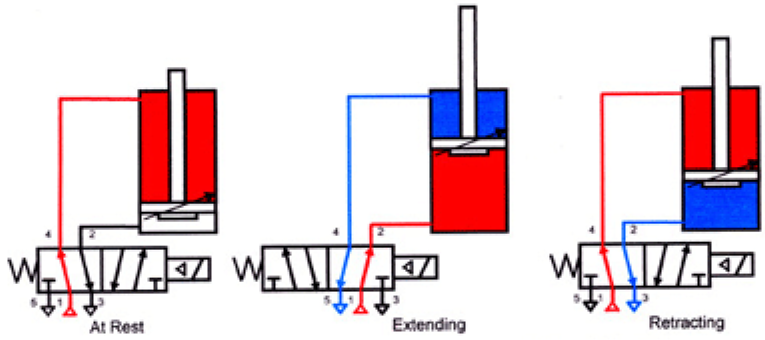


Figure 8-52. Operating a double-acting cylinder with one 5-way valve.

After shifting the valve, or extending, air flows from #1 port through #2 port to the cylinder cap end. Flow from the cylinder rod end goes to #4 port and exhausts through #5 port. The exhaust ports often have speed control mufflers to reduce noise and control the amount of exhaust flow. Speed control mufflers give individual meter-out speed control in each direction of travel.

Deenergizing the solenoid, or retracting, lets the valve spring return to its normal condition causing the cylinder to retract.

In Figure 8-53, the 5-way has a dual inlet instead of dual exhaust. Use a spool type valve for this hookup, since it takes pressure at any port without malfunction.

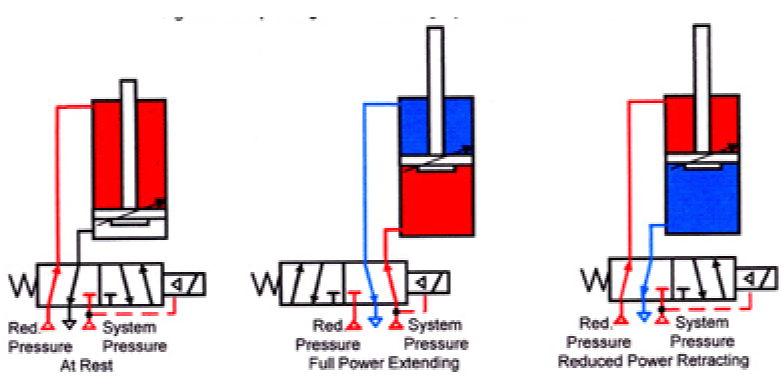


Figure 8-53. Air-saving circuit using a 5-way valve.

On most air circuits the cylinder does little or no work on the retract stroke. Putting low pressure on the rod side of the cylinder uses less compressor air without affecting the operation. This air savings results in lower operating cost and leaves more air to run other actuators. Install flow controls in the lines to the cylinder ports for individual speed control.

If the valve is solenoid pilot-operated, the supply to the pilot valve usually comes from port #1. This means, with a dual pressure inlet, pilot supply must come from some other source. On the circuit in Figure 8-53 a pilot line from system pressure goes directly to the pilot valve. System pressure goes into the external pilot supply port and a plug shuts off the internal pilot port. Changing the pilot line in the field with assistance from the supplier's catalog is quite easy.

Figures 8-54 to 8-61 show another reason for using dual pressure inlets. They depict air cylinder movement with conventional hookup. The cylinder pauses before raising and drops rapidly when starting to retract.

Dual-pressure 5-way valves for air cylinder actuation

A vertical, up-acting air cylinder, with a heavy load, gives sluggish and jerky operation when valved conventionally. Figure 8-54 shows a conventional 5-way valve hook up on a cylinder raising a 600-lb load. This figure shows weight, cap and head end areas, and pressures at both cylinder ports.

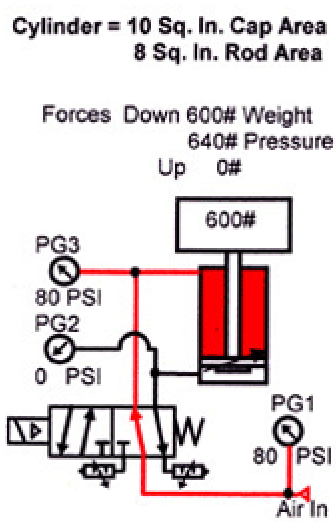


Figure 8-54. Cylinder at rest.

When the directional valve shifts, as seen in Figure 8-55, there is a pause before the cylinder extends. The weight-to-cylinder force ratio and the rate of cylinder travel speed control the length of pause. The heavier the weight and the slower the cylinder speed, the longer the pause. The delay could be three to four seconds in extreme cases.

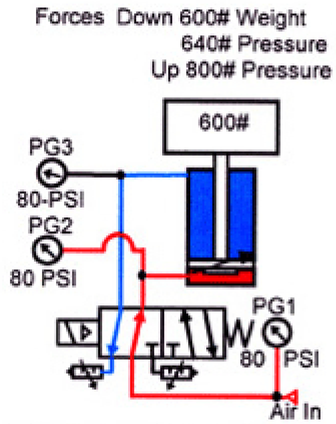


Figure 8-55. Valve just shifted, cylinder pauses.

The pause comes from weight pushing down along with force from air pressure on the cylinder rod end. At the moment the valve shifts to extend the cylinder, down forces are up to 1240-lb while up force is only 800 lb. As long as down forces exceed up force, the cylinder will not move. The slower the air exhausts, the longer it takes to get enough differential pressure across the cylinder piston to move it. The speed of exhausting air controls how fast the cylinder moves once it starts.

When pressure in the head end of the cylinder reaches about 15 psi, as shown in Figure 8-56, the cylinder starts to move. It moves up smoothly and steadily as long as the load remains constant.

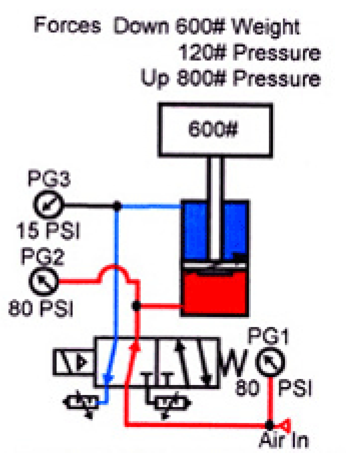


Figure 8-56. Cylinder starts to move after rod-end pressure drops.

When the valve shifts to retract the fully extended cylinder, there is another problem. Figure 8-57 shows the cylinder at rest at the top. Up force is 800 lb from air pressure on the cap end, and down force is 600 lb from the weight.

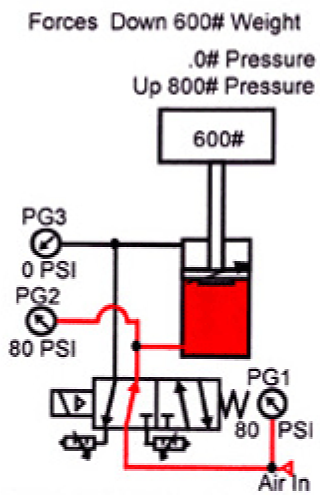


Figure 8-57. Cylinder travels to end of stroke.

When the directional valve returns to normal, as shown in Figure 8-58, down force quickly changes to 1240 lb. Now the load drops rapidly until air pressure in the cap compresses to approximately 120 psi. It takes about 120 psi on the 10-in.2 area to slow the cylinder's rapid retraction.

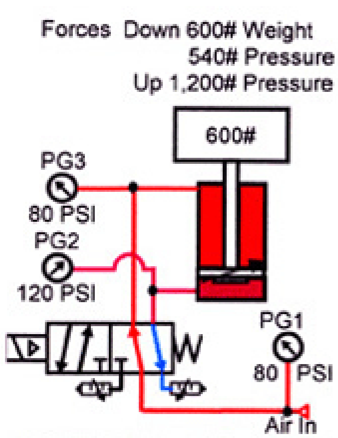


Figure 8-58. Valve shifted

to retract cylinder, which drops rapidly.

Both pauses that occur when extending and retracting are eliminated by using the dual-inlet feature of a 5-way valve.

With a dual inlet pressure circuit shown in Figure 8-59, the cap end port has 80 psi while the rod end port is only 15 psi. This sets a pressure differential across the piston before the valve shifts.

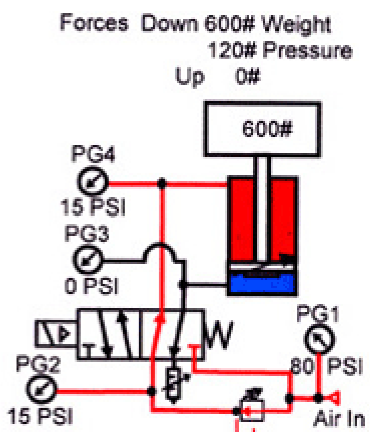


Figure 8-59. Dual-pressure valve at rest.

When the valve shifts, as seen in Figure 8-60, down force is 720 lb and up force is 800 lb. The cylinder starts to move almost immediately and continues moving smoothly to the end.

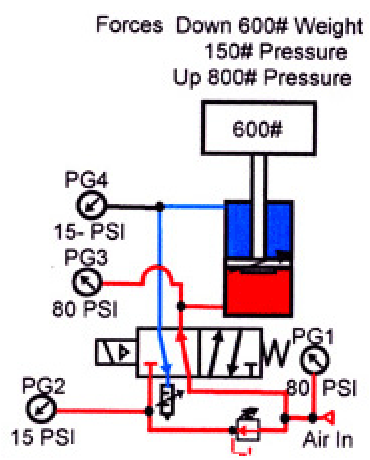


Figure 8-60. Valve shifts, cylinder starts moving quickly.

In Figure 8-61 the valve shifts and the cylinder retracts. With the head end regulator set at 15 psi, down force from air pressure and the load is almost offset by up force. The load lowers smoothly and safely without lunging or bouncing, as fast as cap end air exhausts. In figure 8-59 to 8-61, the cylinder strokes smoothly and quickly in both directions with dual-pressure valve.

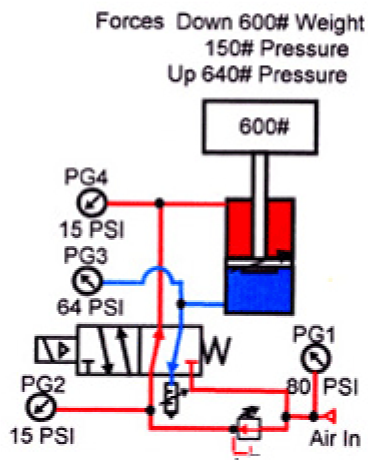


Figure 8-61. Valve shifts to normal, cylinder moves with no lunge.

Check valves as directional valves

Normally a check valve is not thought of as a directional control valve, but it does stop flow in one direction and allow flow in the opposite direction. These are two of the three actions a directional control valve can perform. An inline check valve stops any chance of reverse flow and is useful and/or necessary in many applications. Figure 8-62 shows the symbol for a plain check valve.

Figure 8-62. Plain

check valve.

Another application for a check valve is a relief function, which can be seen in Figure 8-63. Heat exchangers, filters, and low-pressure transfer pumps often need a low-pressure bypass or relief valve. A check valve with a 25-125 psi spring makes an inexpensive, non-adjustable, flow path for excess fluid. It protects low-pressure devices in case of through flow blockage. Pilot operated directional valves commonly use a check valve in the tank or pump line to maintain at least 50-75 psi pilot pressure during pump unload. Some manufacturers make a check valve with an adjustable spring, for pressures up to 200 psi or more.

Figure 8-63. Backpressure check valve

Some check valves have a removable threaded plug in them that may be drilled to allow controlled flow in the reverse direction. The symbol in Figure 8-64 shows how to represent this in a symbol. A common use for a drilled check valve is as a fixed, tamper proof, flow control valve. Fluid free flows in one direction, but has controlled flow in the opposite direction. The only way to change flow is to change the orifice size. This flow control valve is not pressure compensated.

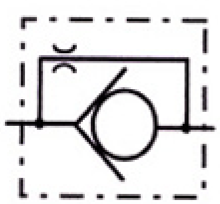


Figure 8-64. Check valve with orifice plug.

Many of the circuits in this manual show standard check valves in use. Hi-L pump circuits, reverse free flow bypass for flow controls, sequence valves or counterbalance valves, and multi-pump isolation, to name a few. Figure 8-65 shows some other applications for check valves.

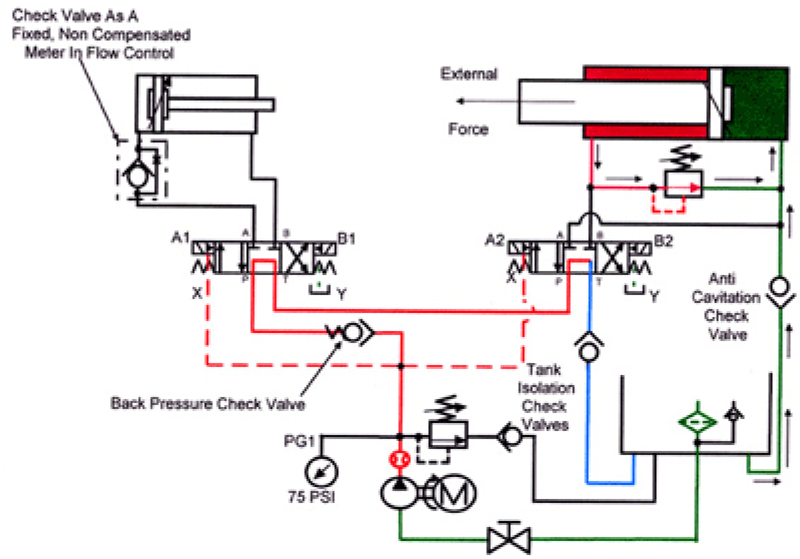


Figure 8-65. Check valves in different circuit applications.

When the tank is higher than the pump or directional valves, always install some means to block flow lines for maintenance. If the valves are not blocked, the tank must be drained when changing a hydraulic component. Shut-off valves are the only option for lines that flow out of the tank to a pump or other fluid using device. To avoid running the pump dry, its shutoff should have a limit switch indicating full open before the electrical control circuit will allow the pump to start. All return lines though, can have a check valve piped as shown in Figure 8-65. A check valve with a low-pressure spring, called an tank isolation check valve, on each return line allows free flow to tank, while blocking flow out of it. A check valve in the tank lines makes shut off automatic and eliminates chances of blowing a filter or wrecking a valve at startup.

The backpressure check valve in the pump line maintains a minimum pilot pressure while the pump unloads. Here it is in the line feeding the directional valves, other times it is in the tank line. In either case it provides pilot pressure to shift the directional valves when a new cycle starts.

The circuit in Figure 8-65 also shows an anti-cavitation check valve for the cylinder with a relief valve to protect it from over pressure. An external force can pull against the trapped oil in the cylinder and cause damage or failure without relief protection. When outside forces move the cylinder, fluid from the rod end goes to the cap end, but is not enough to fill it. If a void in the *cap of the cylinder is no problem then an anti-cavitation check valve is unnecessary. However, this void can cause erratic action when the cylinder cycles again, so install an anti-cavitation check valve. The anti-cavitation check valve has a very low-pressure spring, which requires 1-3 psi to open, so it allows tank oil to fill any vacuum void that might form. The anti-cavitation check valve has no effect during any other part of the cycle.*

Pilot-operated check valves

There are some circuits that need the positive shut off of a check valve but in which reverse flow is also necessary. The following images show symbols of pilot-operated check valves that allow reverse flow. Figure 8-66 shows the symbol for a standard pilot to open check valve. Figure 8-67 shows a pilot-operated check with a decompression feature. The symbol in Figure 8-68 shows a pilot-operated check valve with an external drain for the pilot piston. Each of these pilot-operated check valves allow reverse flow, but two of them have added features to overcome certain circuit conditions.

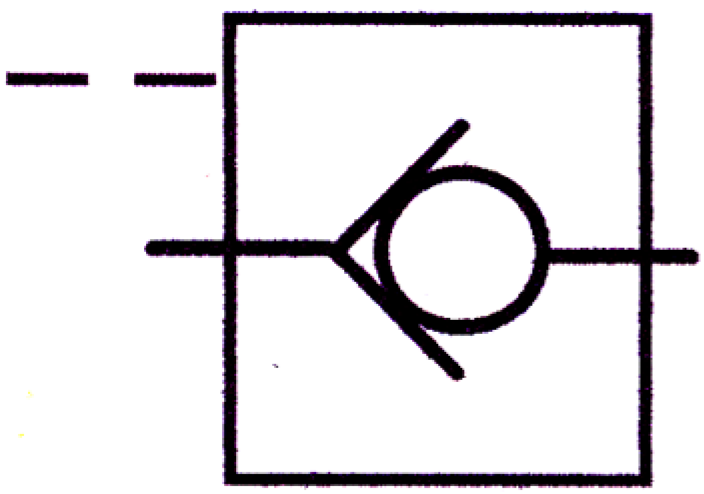


Figure 8-66. Pilot-operated check valve.

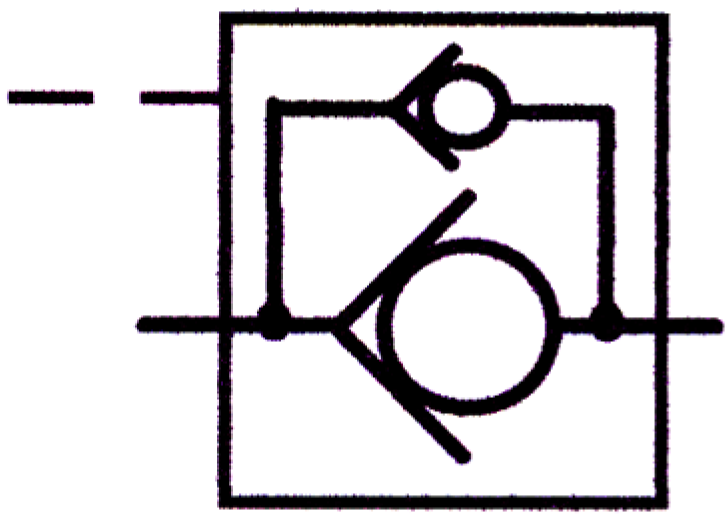


Figure 8-67. Pilot-operated check valve with decompression poppet.

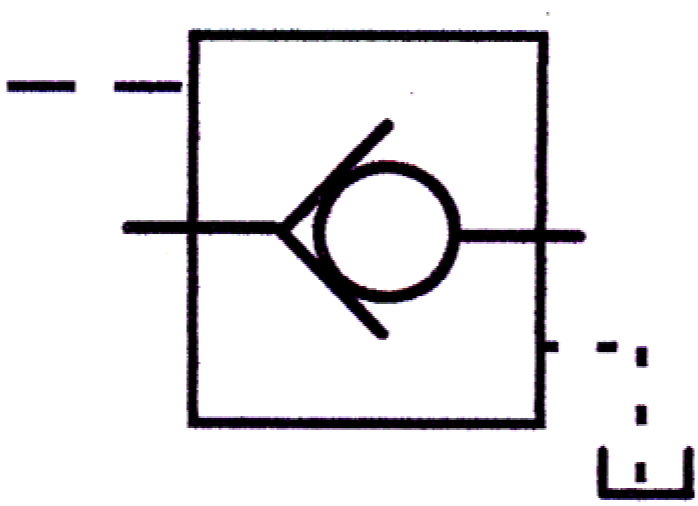


Figure 8-68. Pilot-operated check valve with external drain.

To hold a cylinder stationary, it must have resilient continuous non-leaking seals, no plumbing leaks, and a non-leaking valve. Metal-to-metal fit spool valves will not hold a cylinder for any length of time. As shown in Figure 8-69, a blocked center valve can actually cause a cylinder to creep forward. Vertically mounted cylinders with down acting loads always creep when using a metal-to-metal fit spool valve. Hydraulic motors always have internal leakage so the circuits *shown here will not hold them stationary. Figures 8-70, 8-71, and 8-72 show a typical pilot-operated check valve circuit that prevents cylinder creep.*

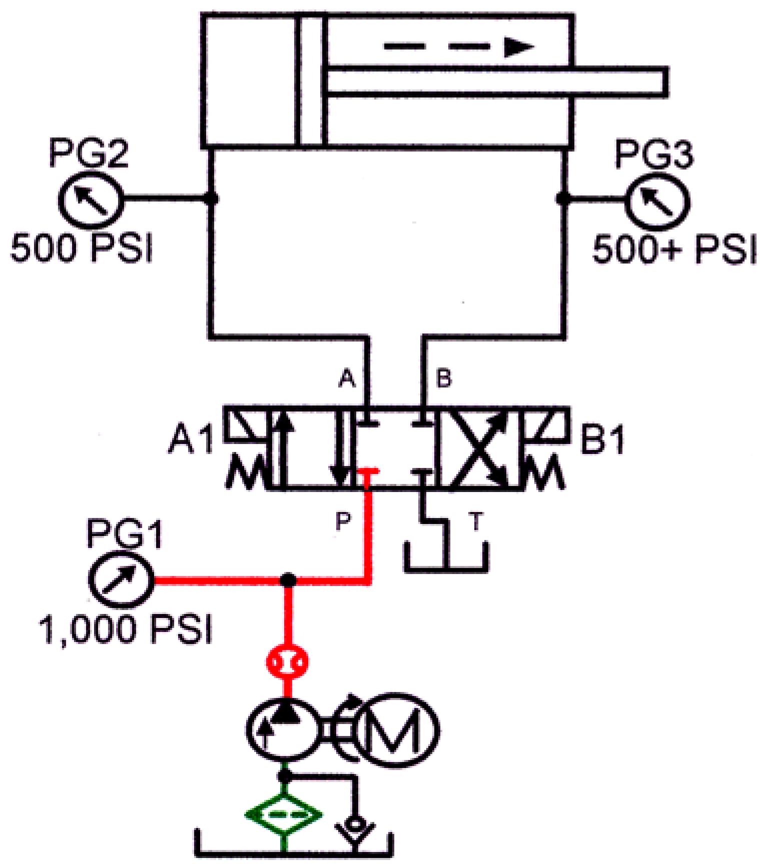


Figure 8-69. Blocked center directional valve, cylinder creeping forward.

The circuit in Figure 8-70 shows a horizontally mounted, non-leaking cylinder, positively locked in place any time the directional centers. When using an on-off type solenoid valve, a fast moving cylinder stops abruptly when the directional valve centers. Use a proportional valve with ramp timers to decelerate the actuator and eliminate shock damage.

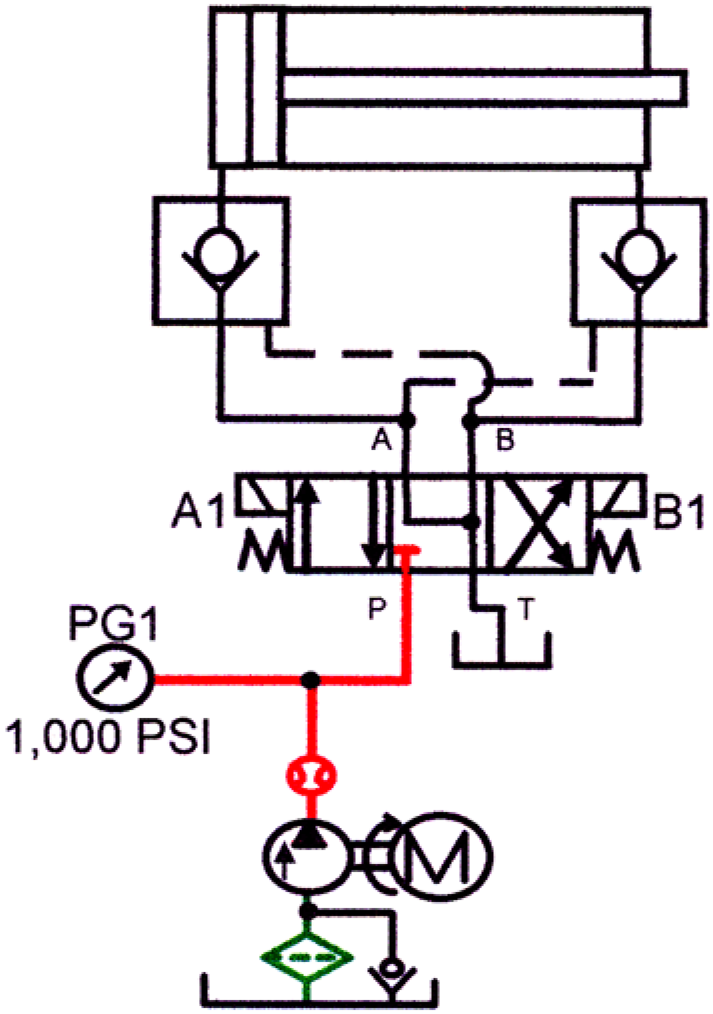


Figure 8-70. Pilot-operated check circuit at rest with pump running.

Notice the directional valve has A and B ports open to tank in the center condition. This center condition allows pilot pressure to drop and the pilot-operated check valves to close. Using a directional valve with blocked A and B ports in center condition, may keep the pilot-operated check valves open and allow cylinder creep. If it is only necessary to keep the cylinder from moving in one direction, one pilot-operated check valve will suffice.

When solenoid A1 on the directional valve shifts, as seen in Figure 8-71, the cylinder extends. Pump flow to the cylinder cap end builds pressure in the pilot line to the rod end of the pilot-operated check valve, causing it to fully open. The pilot-operated check valve in the line to the cap end opens by pump flow like any check valve. Energizing and holding a directional valve *solenoid causes the cylinder to move. Pilot operated check valves positively lock the cylinder but are invisible to the electric control circuit.*

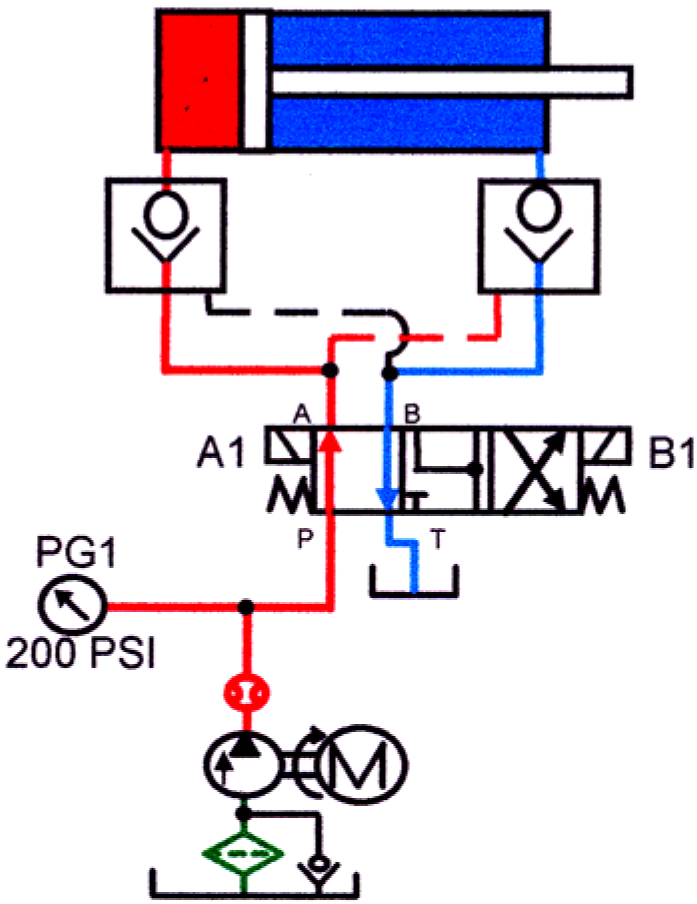


Figure 8-71. Pilot-operated check circuit with cylinder extending.

When solenoid B on the directional valve shifts, as seen in Figure 8-72, the cylinder retracts. Pump flow to the cylinder rod end builds pressure in the pilot line to the cap end of the pilot-operated check valve, causing it to fully open. The pilot-operated check valve in the line to the rod end opens by pump flow like any check valve. Energizing and holding a directional valve solenoid causes the cylinder to move.

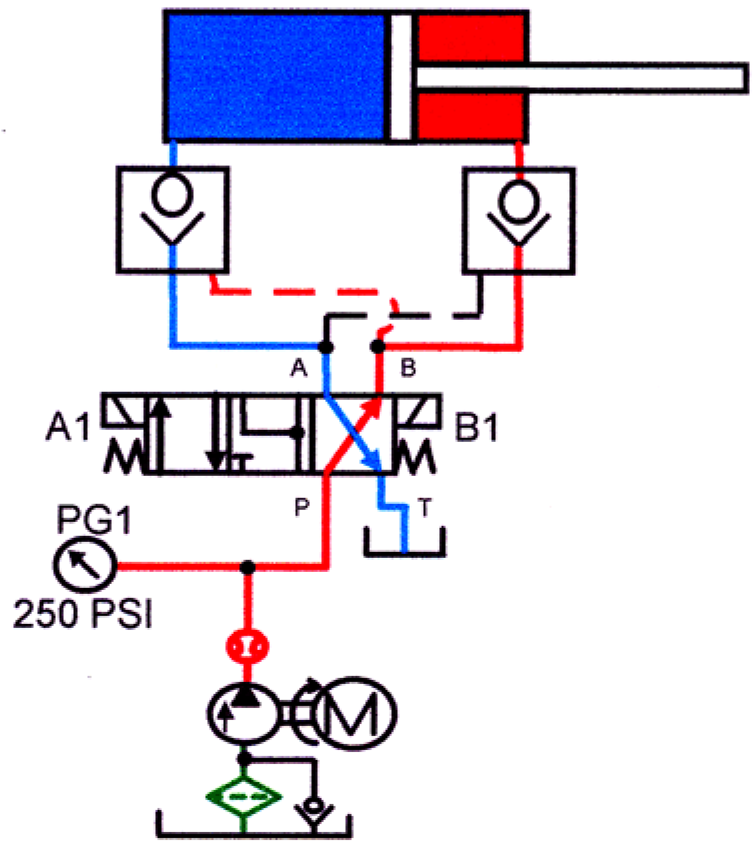


Figure 8-72. Pilot-operated check circuit with cylinder retracting.

The following will describe how pilot-operated check valves can cause problems in some applications.

Pilot-operated check valves

Figure 8-73 shows how using a pilot-operated check valve to keep a heavy platen from drifting can cause problems.

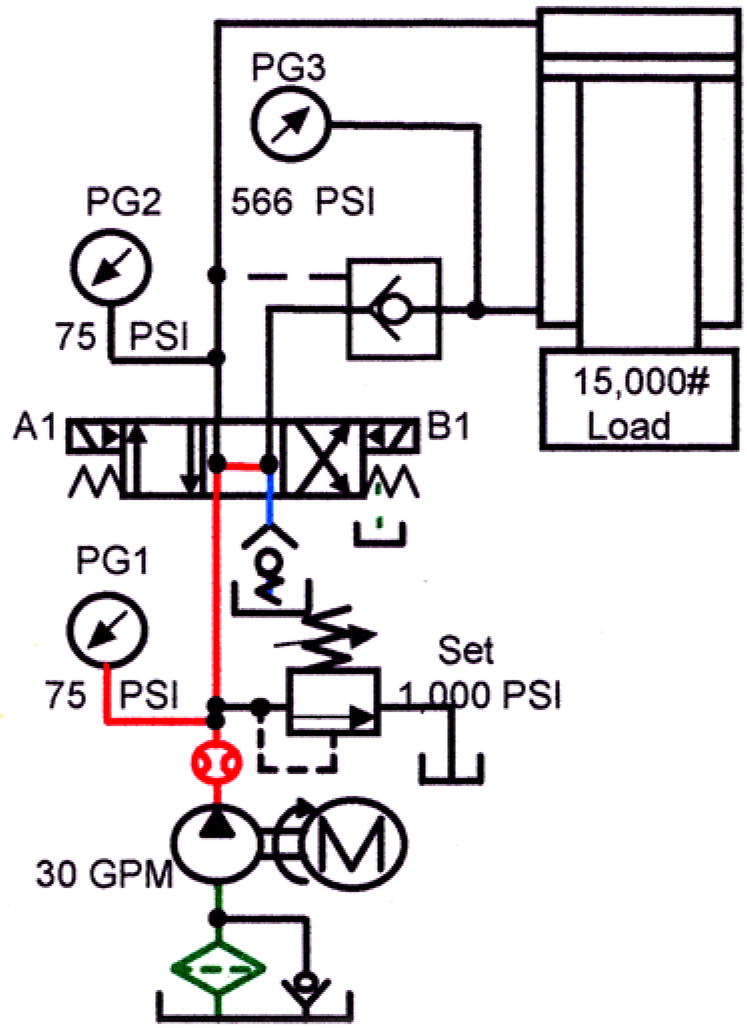


Figure 8-73. Pilot-operated check valve on running away load, at rest, pump running.

When a cylinder has a load, trying to extend it causes load-induced pressure. In the example cited, a 15,000-lb platen pulling against a 26.51 square inch rod end area gives a 566 psi load-induced pressure. This load-induced pressure holds against the poppet in the pilot-operated check valve, forcing it closed. The pilot piston must have sufficient pressure to open the poppet with 566 psi pushing against it. The pilot piston on most pilot-operated check valves has an area that is three to four times that of the poppet. This means it will take approximately 141-188 psi at the cap end cylinder port to open the poppet for reverse flow.

*When the directional valve shifts, starting the cylinder forward, as shown in Figure 8-74***,** *pressure in the cap end cylinder port starts climbing to 150 psi. At about 150 psi the poppet in* *the pilot-operated check valve opens and allows oil from the cylinder rod end a free flow path to tank. The cylinder immediately runs away, pressure in cylinder cap port drops, the pilot-operated check valve closes fast and hard, and the cylinder stops abruptly. When the pilot-operated check valve closes, pressure at the cap end cylinder port again builds to 150 psi, opening the check valve, and the process starts again. A cylinder with these conditions falls and stops all the way to the work unless it meets enough resistance to keep it from running away.*

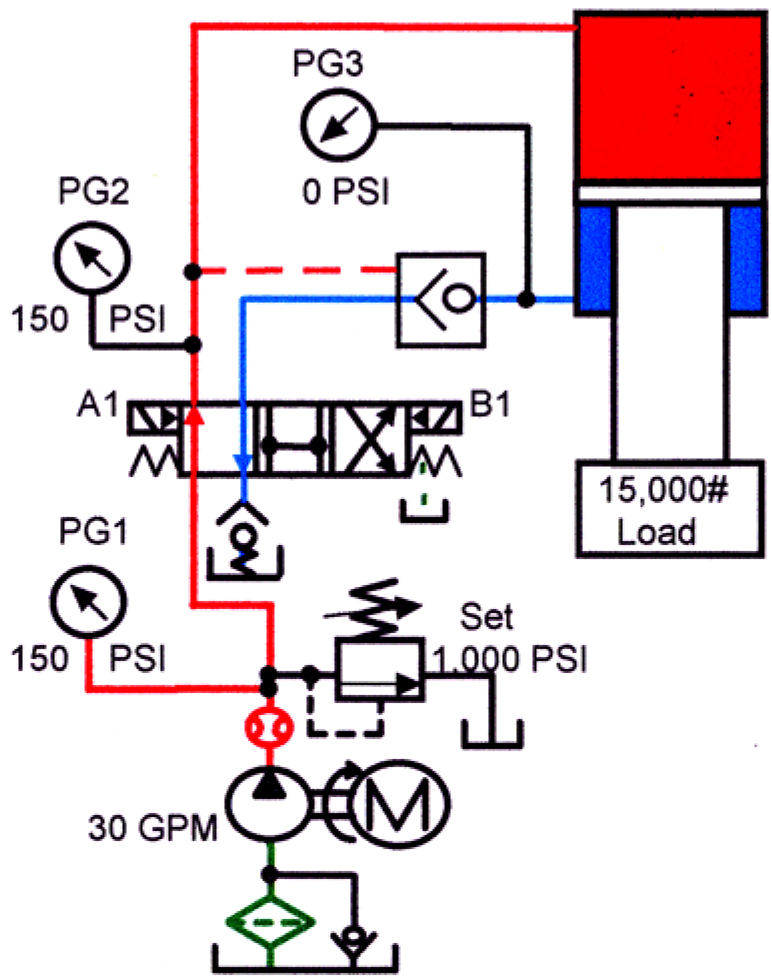


Figure 8-74. Pilot-operated check valve on running away load, cylinder extending, free fall.

With this circuit, system shock very quickly damages piping, cylinders, and valves.

Adding a flow control between the cylinder and pilot-operated check valve is one way to keep it from running away. However, the restriction could cause fluid heating and slow cycling, and would need frequent adjustment to maintain optimum control.

Placing a flow control after the pilot-operated check valve causes backpressure against its pilot piston and could keep it from opening at all. With the flow control after the pilot-operated check valve, use one with an external drain. When there is much backpressure on the outlet of a pilot-operated check valve, it is best to use one with an external drain.

It is best to control the cylinder shown here with a counterbalance valve. See chapter five for the different types of counterbalance circuits.

Even with some spool type counterbalance valves, the cylinder still drifts. Adding an externally drained pilot-operated check valve between the counterbalance valve and the cylinder holds it stationary. The counterbalance valve keeps the cylinder from running away no matter the flow variations, while the pilot-operated check valve holds it stationary when stopped.

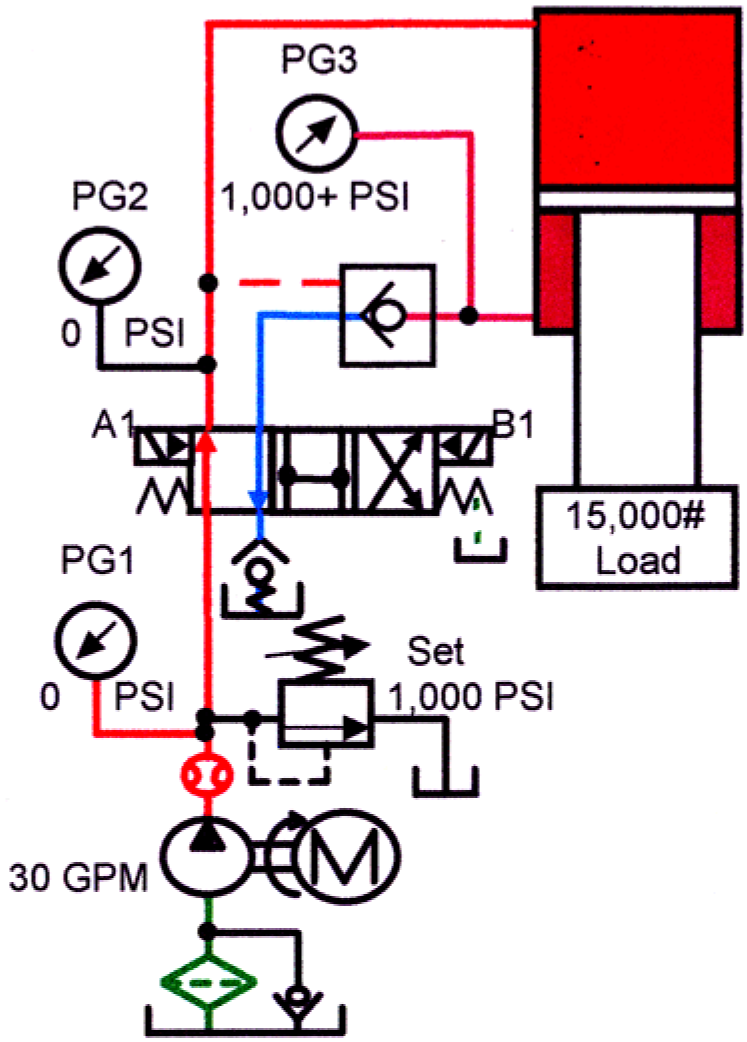


Figure 8-75. Pilot-operated check valve on running away load, cylinder stopping on closed P.O. check.

A pilot-operated check valve with the decompression feature would not help in this circuit.

Figures 8-76 and 8-78 show another possible problem using a pilot-operated check valve to keep a vertical down-acting cylinder from drifting. The cylinder in this example has a heavy weight pulling against the rod side. A load induced pressure of 1508 psi plus 142 psi from pilot pressure acts against the poppet in the pilot-operated check valve. This requires a high pilot pressure to open the pilot-operated check valve.

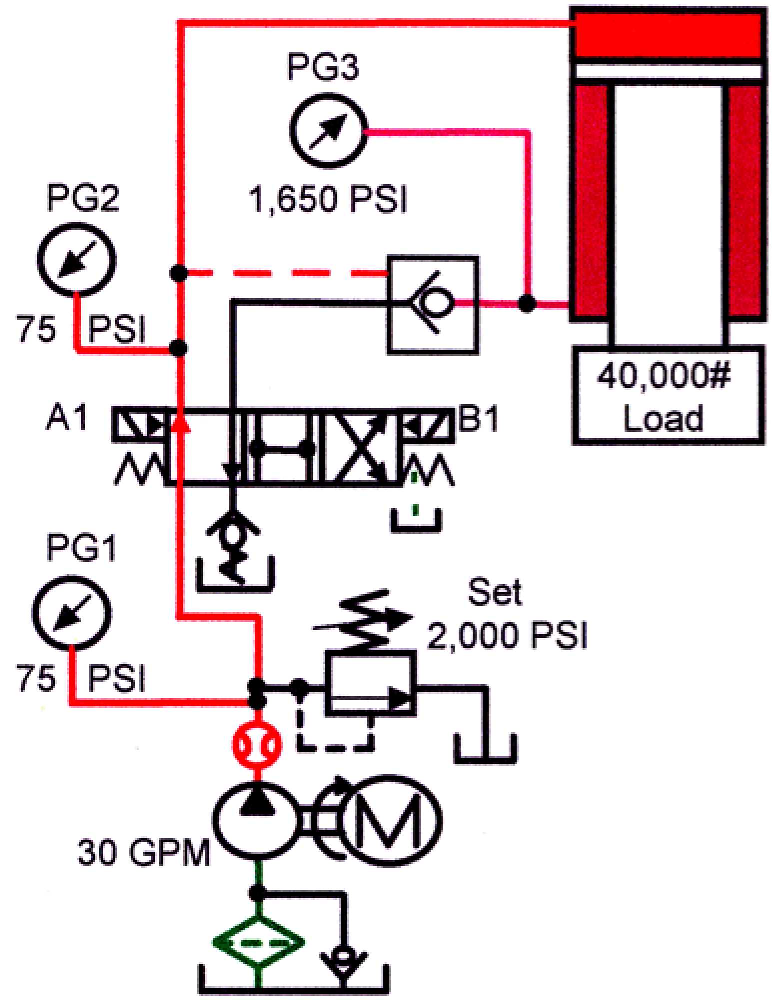


Figure 8-76. Pilot-operated check valve on running away load, cylinder just starting to extend.

It requires approximately 500 psi pilot pressure to open the pilot-operated check valve with 1650 psi against the poppet. As pilot pressure builds to open the poppet, it also pushes against the full piston area of the cylinder. This cylinder has nearly twice the area on the cap side as the rod side, so every 100 psi on the cap side gives about 200 psi on the rod side. As pilot pressure builds to the 500 psi required, pressure against the poppet in the pilot-operated check valve increases at twice the rate. Figure 8-77 shows the start of this condition.

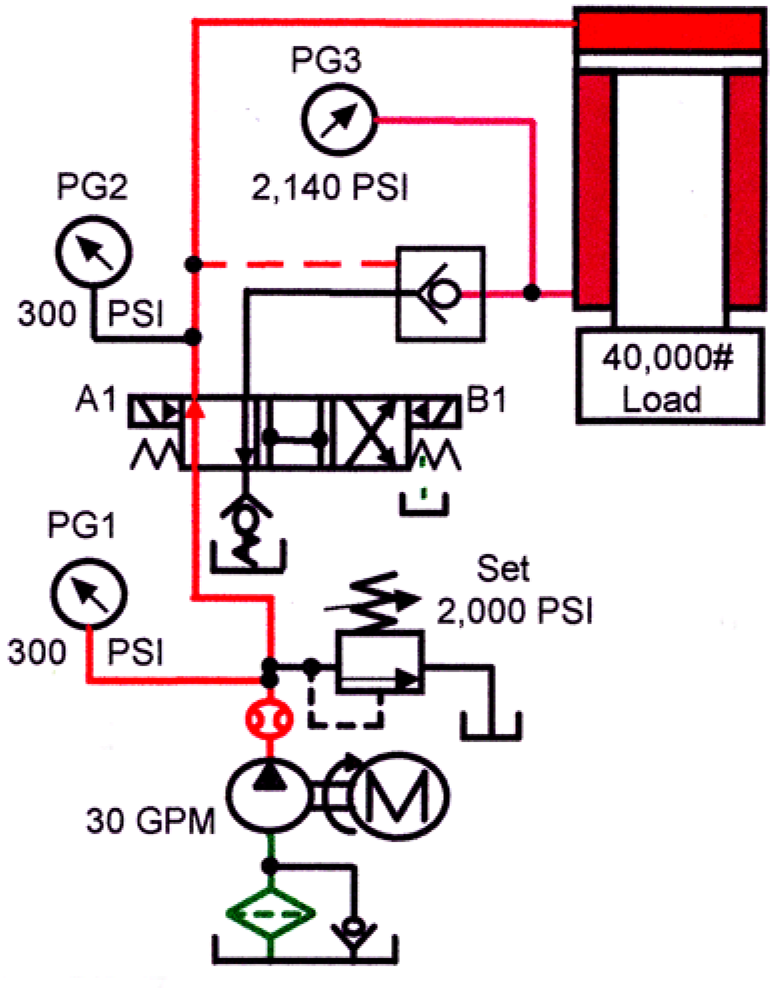


Figure 8-77. Pilot-operated check valve on running away load, cylinder still trying to extend.

In Figure 8-77, the cylinder rod end pressure is at 300 psi, which adds 570 psi to the 1508 psi load-induced pressure. The extra hydraulic pressure pushes harder against the pilot-operated check valve poppet, making pilot pressure increase even more.

As pilot pressure increases, down force and rod end pressure escalates also. In Figure 8-78, ***,*** *rod end pressure is at 3565 psi because pilot pressure continues to climb. In the situation shown here, it is obvious the relief valve will open before reaching a pilot pressure high enough to open the pilot-operated check valve. Even if pilot pressure could go high enough to open the pilot-operated check valve, the cylinder runs away and stops.*

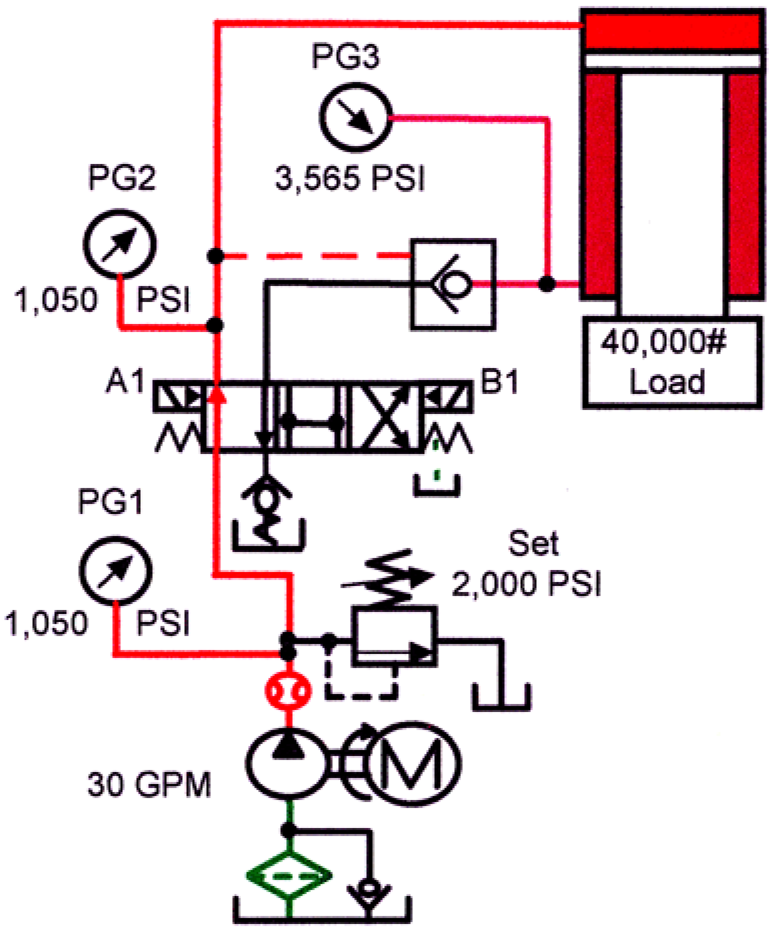


Figure 8-78. Pilot-operated check valve on running away load, cylinder still trying to extend.

A pilot-operated check valve with a decompression poppet would not help in this situation. Flow from the small decompression poppet is not enough to handle cylinder flow. The cylinder would extend with a decompression poppet, but at a very slow rate.

It is best to control the cylinder in this example with a counterbalance valve. See chapter five for the different types of counterbalance circuits.

Even with some spool type counterbalance valves, the cylinder still drifts. Adding an externally drained pilot-operated check valve between the counterbalance valve and the cylinder will hold it stationary. The counterbalance valve keeps the cylinder from running away no matter the flow variations, while the pilot-operated check valve holds it stationary when stopped.

Shown are circuits that require a pilot-operated check valve to have external drain and/or decompression capabilities.

A standard pilot-operated check valve circuit usually has minimum backpressure at the reverse flow outlet port. If there is a restriction causing high backpressure in the reverse flow outlet port, a standard valve may not open when applying pilot pressure. The reason this might happen is the pilot piston sees backpressure from the reverse flow outlet port. If the pilot-operated check valve poppet has load induced pressure holding it shut, plus reverse flow outlet port backpressure opposing the pilot piston, there is not enough pilot piston force to open the check poppet.

If the reverse flow outlet port backpressure cannot be eliminated, then specify a pilot-operated check valve with an external drain. Pipe the external drain to a low or no pressure line going to tank. With an external drain pilot-operated check valve, the pilot piston usually opens the check poppet to allow reverse flow.

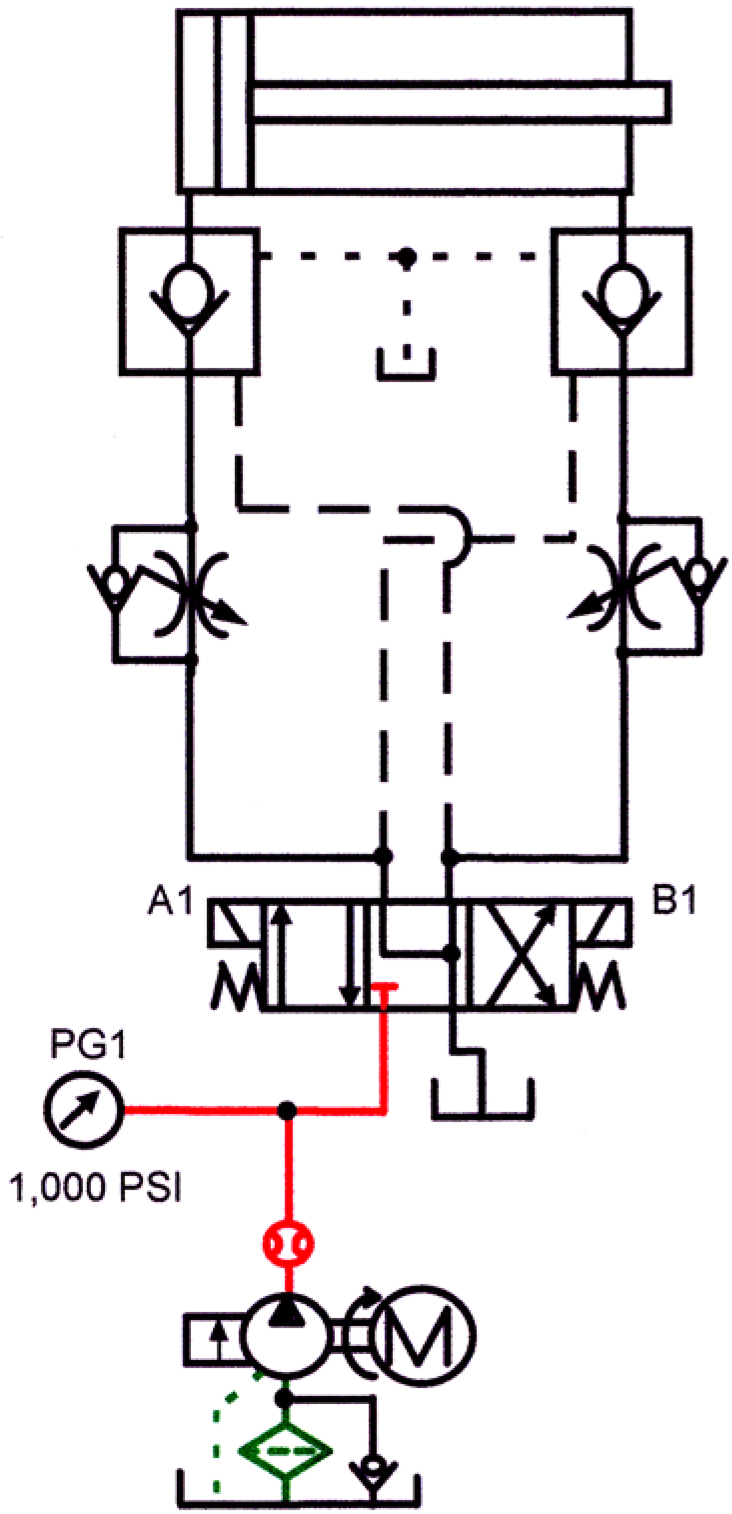


Figure 8-79. Pilot-operated check valve circuit with external drain function at rest, pump running.

The schematic drawing in Figure 8-79 shows a cylinder with pilot-operated check valves at each port and meter out flow controls downstream of the reverse flow outlet port. If this circuit did not have externally drained pilot-operated check valves, the cylinder would operate in jerks or not at all when the directional valve shifts. Backpressure from the flow controls can push the pilot piston closed and stop the cylinder, then pressure would drop and it would start again. This oscillating movement would continue until the cylinder competes its stroke. With externally drained pilot-operated check valves, the cylinder is easy to control at any speed.

Placing the flow controls in Figure 8-79 between the cylinder ports and the pilot-operated check valve eliminates backpressure. This move eliminates the need for externally drained pilot-operated check valves.

In Figure 8-80, a running away load had a drifting problem with only the counterbalance valve installed. Adding a pilot-operated check valve in front of the counterbalance valve stopped cylinder drifting. Using a decompression poppet made it easy to open the main check poppet against the high load induced pressure. The decompression poppet releases trapped fluid in the piping between the pilot-operated check valve and the counterbalance valve allowing the main check poppet to open.

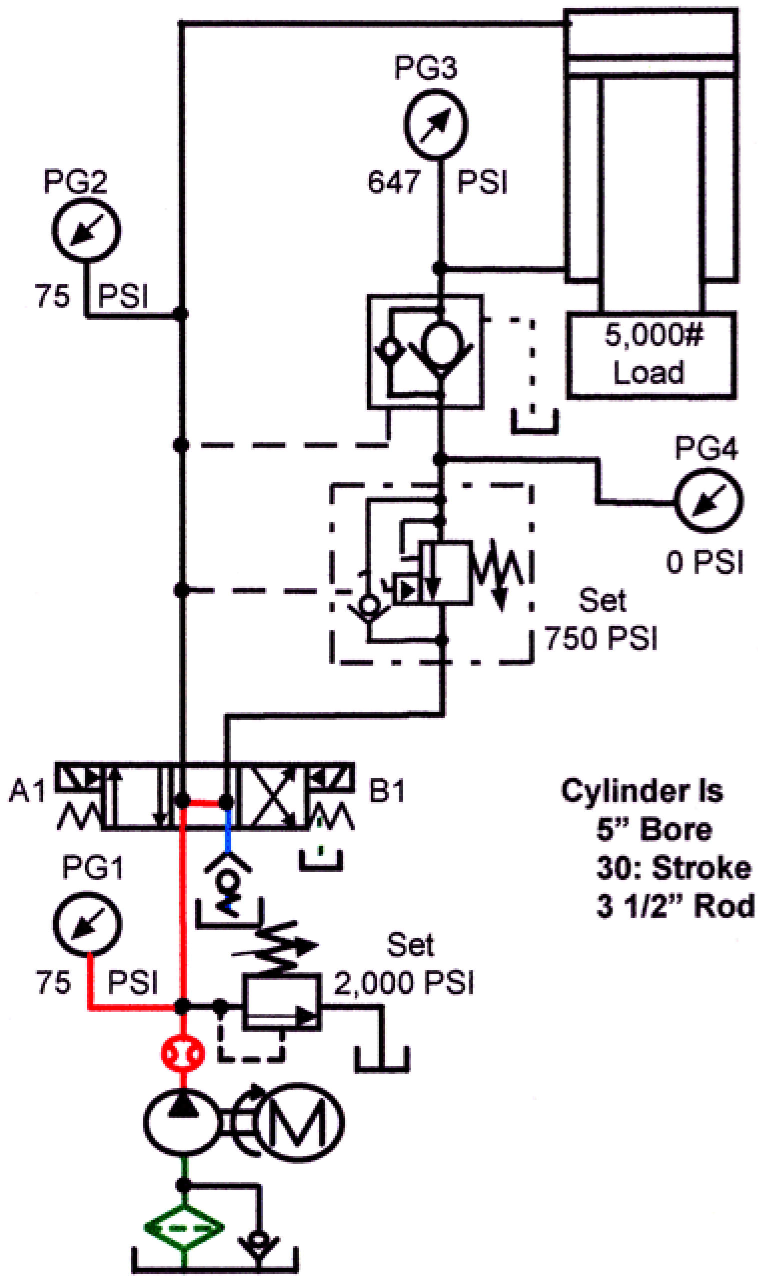


Figure 8-80. Pilot-operated check on running away load with external drain and decompression poppet with P.O. check for no leak holding, counterbalance valve for smooth control of the extend stroke at rest, with pump running.

Notice the pipe between the pilot-operated check valve and the counterbalance valve is at zero psi while the cylinder is held retracted. This pressure would have been about 1200 psi while the *cylinder was retracting, but quickly drops to zero when the directional valve centers. The reason for this pressure drop is leakage past the counterbalance valve spool, which is the reason for adding the pilot-operated check valve.*

If the pilot-operated check valve did not have an external drain, backpressure from the counterbalance valve can force it shut when the cylinder starts moving. The external drain and decompression features are both necessary in this holding circuit.

Placing the pilot-operated check valve in the line after the counterbalance valve would require neither an external drain nor decompression feature. However, the reason for installing the pilot-operated check valve was to stop drifting. With the pilot-operated check valve after the counterbalance valve, the counterbalance valve must have an external drain. An external drain indicates there is internal leakage, so the drift problem may decrease -- but would not go away.

Cartridge valves

The term cartridge valves commonly refers to pressure, directional, and flow control valves that screw into a threaded cavity. These valves are mostly rated for low flows - 40 gpm or less, although some manufacturers have units that will flow more than 100 gpm. Compact screw-in cartridges help build inexpensive circuits that are reliable and easy to maintain. Screw-in cartridges are most often part of a drilled manifold but can be purchased in individual bodies. The performance and function of screw-in cartridge valves is similar to the in-line or sub-plate-mounted valves discussed in Chapter 10.

Slip-in cartridge valves (sometime referred to as logic valves) are different because, except for pressure controls, they are simply 2-way, bi-directional, pilot-to-close check valves. Most circuits using slip-in cartridge valves flow at least 60 gpm and go as high as 3000 gpm. (Several companies make screw-in logic valves in sizes as small as 15 gpm for use when the special features of logic valves are required at lower flows.) Slip-in cartridges also are compact, have low pressure drop, and operate at pressures up to 5000 psi. Slip-in cartridges can function as pressure, flow, and directional valves.

The symbols that illustrate this chapter are the preferred type as first used by the manufacturers. Chapter 4 shows cartridge valve symbols that follow the using ISO rules.

Why use slip-in cartridge valves?

The main reason for using slip-in cartridge valves in high-flow circuits is economy. Large spool valves are available with high flow capacity but few are manufactured, making them expensive with long delivery times. A better choice is a slip-in cartridge valve in a manifold body, piloted by D03- or D05-size directional control valves. (There is at least one supplier of valves that uses logic elements instead of spools that bolt directly to D08- and D10-size sub-plates.)

One important feature of slip-in cartridge valves is that there is almost zero bypass through the A port. Leakage for this port is the same as any good check valve. Flow past the B port is very low because the leak path is long and has a close fit. So when applying this type of valve, always use the A port for the connection that must block flow completely.

Another feature is fast response on opening. Because there is no overlap, flow is almost instantaneous after the valve receives a start signal. Even when controlling poppet-opening speed, there is no lag in flow response that increases cycle time. Response also is fast on closing because the poppet only opens far enough to pass the flow going through it. This means it does not have to move any extra distance to start to restrict flow and shut it off.

Logic valve circuits are also very versatile when set up with multiple control valves. Many center or crossover conditions can be duplicated by which control valves are open and how they are signaled during a cycle. This means that when designers want to try different valve configurations, they only have to change control valves or control circuitry. This versatility can be applied anytime with the right design parameters up front.

How do slip-in cartridge valves work?

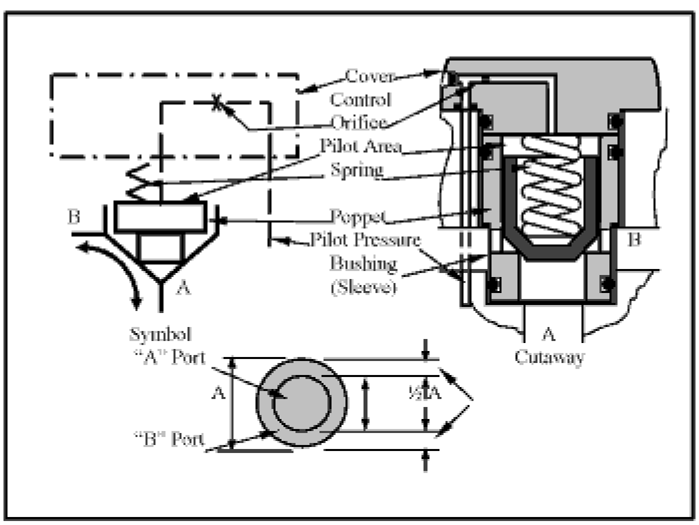
As directional control valves, the most common slip-in cartridge valve has the 1:2 poppet-area ratio shown in ***Figure 11-1****. This is a pilot-to-close check valve with the pilot area and the areas at the A and Bports all equal. There are no communicating holes through the poppet to allow fluid from the A or B ports to get behind it. Fluid entering the A or B port pushes the poppet open, so flow can go either way, restricted only by the light spring that holds the poppet in place during shutdown. Spring force choices from most suppliers are usually 25, 50, or 75 psi.*

It is easy to see that flow through the valve in either direction can be blocked by pressure on the pilot area. Such pressure must be equal to or greater than the pressure at the A or B port. When there is equal pressure at the A and B ports, pressure on the pilot area must be equal to or greater than that which is trying to push the poppet open.

From the foregoing description it should be obvious that when the pump is off and pilot pressure is gone, any load-induced pressure will push the poppets open and running-away loads will run away. (Note that this does not happen with spool valves.) Circuit design is different with slip-in cartridge valves and some of the pitfalls will be discussed later.

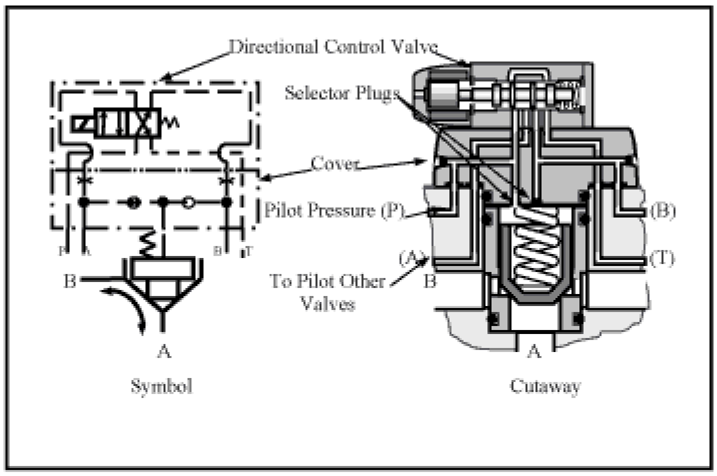
Slip-in cartridge valves are held in place by a cover that also contains passages for pilot oil. In addition, the covers may have an interface for directional or pressure control functions. Covers can also have control orifice inserts to retard poppet movement for better control. The plain cover shown in ***Figure 11-1*** *would receive a signal from another slip-in cartridge valve with a solenoid-operated valve interface. (A plain cover may serve as a check valve as shown in* ***Figure 11-4****.)*

Fig. 11-1. 1:2 poppet-type slip-in cartridge valve



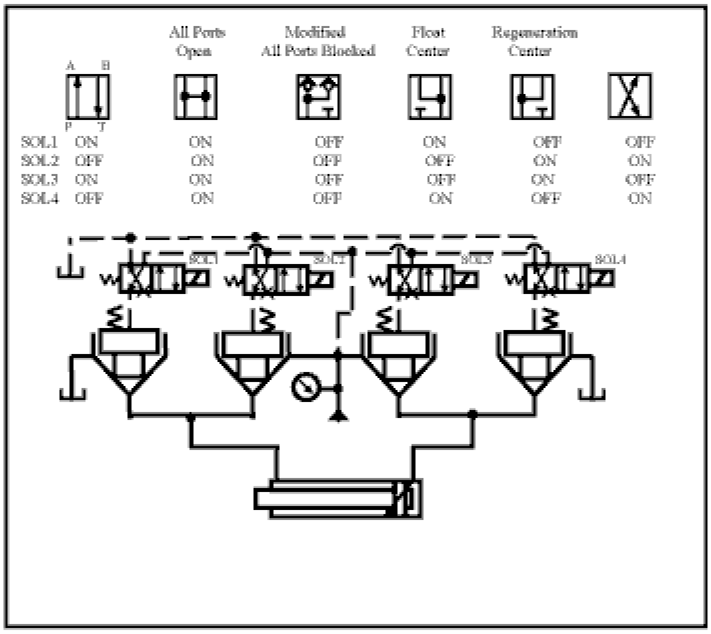
The symbol and cutaway view in ***Figure 11-2*** *is for a 1:2 slip-in cartridge valve with a single-solenoid directional control valve supplying pilot pressure to the spring side of the poppet. Pilot* *pressure would be the same as system pressure when the pump is running, so this valve would be normally closed - even with pressure at both ports. Pilot pressure could be blocked and the spring side of the poppet could be open to tank by changing the selector plug location or by using a directional control valve with a P-to-A, B-to-T at-rest condition. The P and T ports connect to pressure and tank in the manifold block through the cover. The A and B ports also go to the manifold block to be connected to other valves as required.*

Fig. 11-2. 1:2 poppet-type cartridge valve with directional control valve interface



*It would take four separate spool-type directional control valves to do what four of the slip-in cartridge valves shown in **Figure 11-2** can accomplish. All that is required to change a cartridge valve circuit is to use different selector plug locations or different directional control valves and/or a change in the electric control circuit. **Figure 11-3** shows the equivalent spool valve conditions that are possible with different solenoids energized or de-energized.*

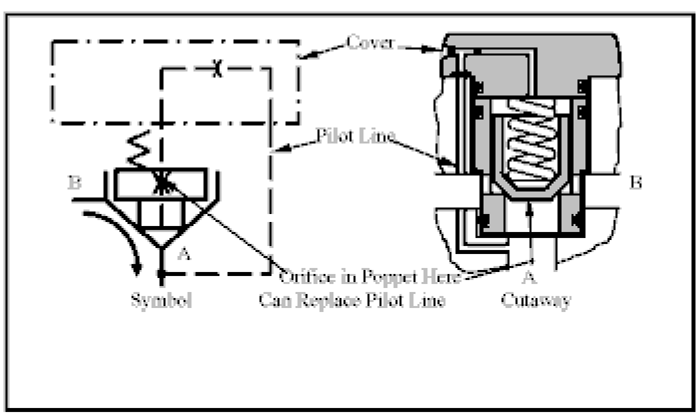
Fig. 11-3. 1:2 poppet-type cartridge valves with separate directional control valves -- showing different spool configurations possible



The directional control valve operator could also be a 2-position detented or a 3-position spring-centered valve according to circuit needs. (Chapter 4 presents symbols of other control valve configurations.)

The slip-in cartridge valve in ***Figure 11-4*** *has a drilled pilot line that intersects the A port of the cartridge. Fluid is free to flow from port B to port A, but is blocked when it tries to reverse. Changing the pilot line to communicate with the B port can change this check valve to flow freely from port A to port B.*

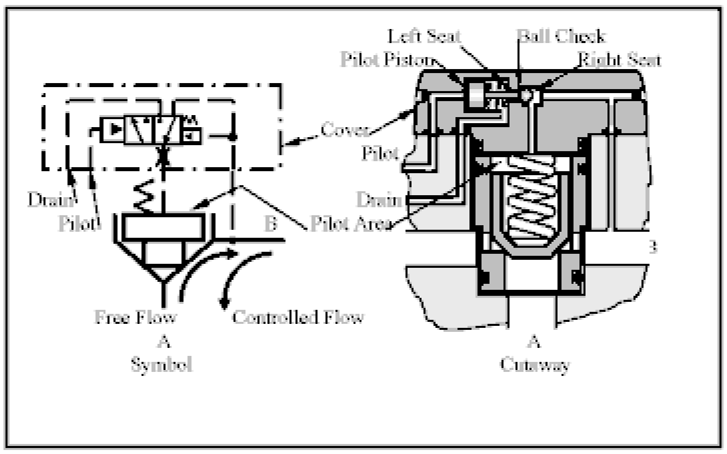
Fig. 11-4. 1:2 poppet-type cartridge valve as a check valve
(with free flow from *B* to *A*, and checked flow from *A* to *B*)



A single-function check valve can be made with an available orificed poppet. The orifice always communicates with the A port so fluid is free to flow from B to A, but blocked from A to B. The *orifice is drilled through the poppet as indicated by the phrase: Orifice in Poppet Here Can Replace Pilot Line.*

An orificed poppet can serve other functions. For example, it can be used as a blocking valve controlled by a 2-way pilot valve. The symbol and cutaway view in ***Figure 11-5*** *show a slip-in cartridge valve set up as a pilot-operated check valve. A special cover with an integral pilot-operated 3-way valve either delivers fluid to or exhausts fluid from the pilot area. The pilot signal comes from other sources -- such as the circuits shown in Chapter 10.*

Fig. 11-5. 1:2 poppet-type cartridge valve as a pilot-operated check valve (with free flow from *B* to *A*, and controlled flow from *A* to *B*)



Fluid entering port A is always free to flow out of port B after its pressure overcomes the spring holding the poppet down. Flow from B to A is blocked until the valve receives a pilot signal. Without such external intervention, a pilot-operated check valve works like any other check valve.

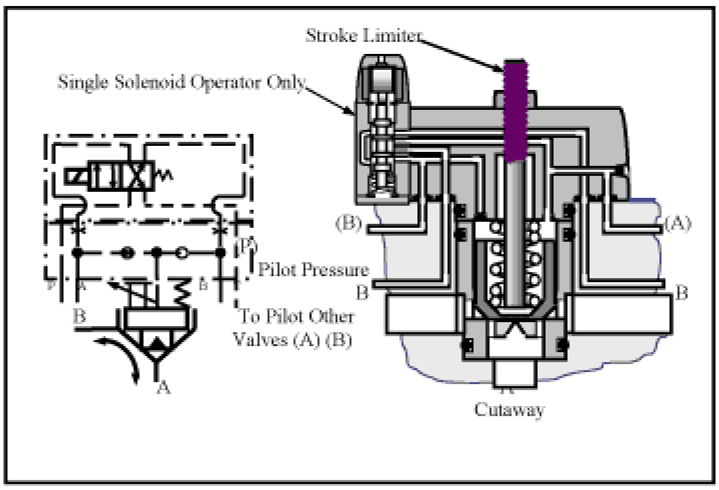
The cover on a pilot-operated check valve has a ball check held on the left seat by a light spring. The ball check traps fluid from the B port, forcing it to hold the poppet down on its full diameter. When there is no pilot signal, there is no flow from port B to port A. The symbol plainly shows this function with the 2-position, 3-way, spring-and-pressure shifted valve held in a position to allow fluid from the B port to hold the poppet closed.

The pilot piston has an area that is three to four times that of the ball check. Thus, a pilot signal that is one quarter to one third the pressure holding the ball check on the left seat will shift it to the right seat to block fluid from the B port. At the same time, fluid trapped behind the poppet is free to go to tank through the drain port. At this point, fluid can freely flow from the B port to the A port with almost no restriction.

A slip-in cartridge valve operating as a pilot-operated check valve has all the circuit problems explained in Chapter 10. The only difference is its physical size and flow capacity.

Another feature that is available on slip-in cartridge valves is the poppet stroke limiter shown in ***Figure 11-6****. A stroke limiter is simply an adjustable screw inside the poppet that can limit the poppet's travel. The stroke limiter can be used as a non-compensated flow control or a maximum flow limiter on running away loads in certain applications. It works without problems as a meter-in device, but can cause unexpected regeneration as a meter-out device on the rod end of a cylinder with load-induced pressure and/or an oversized rod.*

Fig. 11-6. 1:2 poppet-type cartridge valve with stroke limiter and directional control valve interface

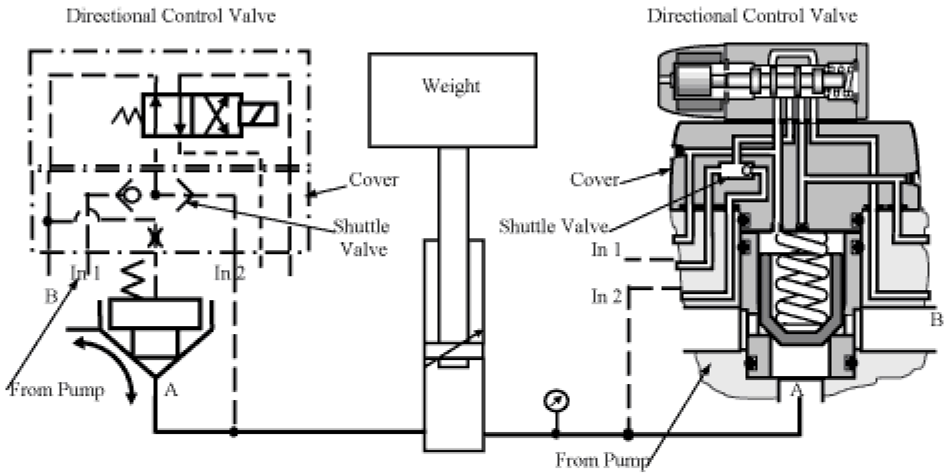


Most slip-in cartridge valves with stroke limiters have poppets with extended noses. These may be tapered or have vee notches cut in them. This type poppet gives a better profile for flow reduction as the poppet moves toward shutoff. For best control of flow, the cartridge should be sized so the poppet travel is maximum.

The valve in ***Figure 11-6*** *has a solenoid-operated directional control valve interface. The stroke limiter also can come in a plain-cover model.*

Because slip-in cartridge valves are pilot-to-close check valves, they must have pilot pressure to stay closed. A vertical cylinder holding a load, as in ***Figure 11-7****, will not stay up when the pump is shut off without some means of maintaining pilot pressure. Slip-in cartridge valves are available with a shuttle-valve function so pilot pressure can be taken from more than one source. In the case of the loaded vertical cylinder, the second source is load-induced pressure.*

Fig. 11-7. 1:2 slip-In cartridge valve with dual-pilot source pump (pump not running)



The cutaway and symbol in ***Figure 11-7*** *show how a shuttle valve works. Its simplest form contains a free-floating ball that can seal ports to the right or left to block flow from the opposite side. A pilot signal from the right or left always exits from the top -- never from the opposite port. If a shuttle valve receives two signals, it will always pass the highest one. Different pressures on equal areas always move the blocking device toward the lower pressure.*

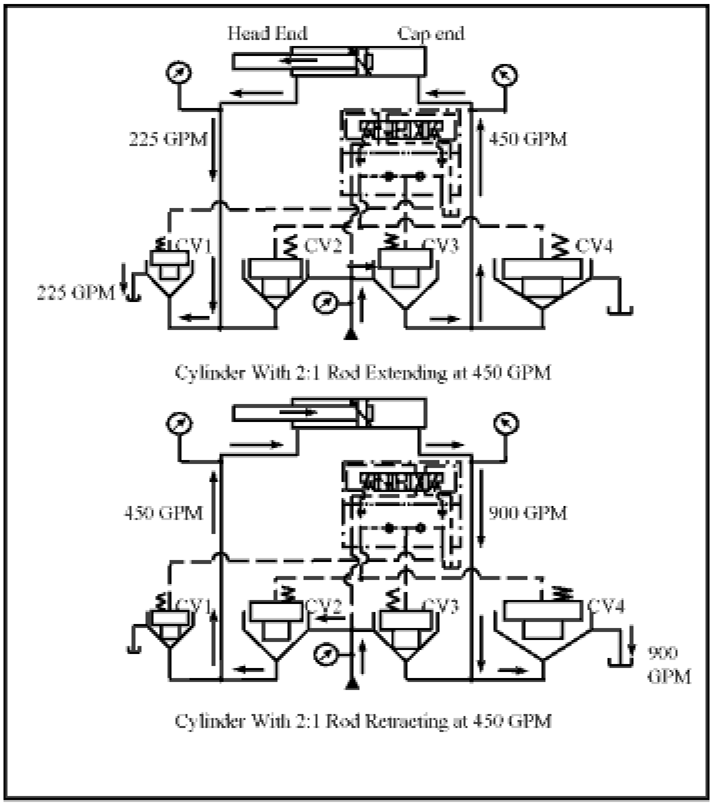
Figure 11-7 *shows a slip-in cartridge valve cover with a built-in shuttle valve that can accept fluid from two sources and send it to the pilot area of the poppet. The cover may have an interface for a directional valve (as shown), or it can be a plain type.*

Because pump pressure is normally higher than load-induced pressure, pilot pressure to the poppet would be from the pump at port In 1. The cylinder's load in ***Figure 11-7*** *would drop when the pump was shut down if not for the load-induced pressure going to port In 2. At pump shutdown, the shuttle ball shifts to the left (as shown) and the load-induced pilot pressure holds the poppet shut and the cylinder stationery.*

Pilot pressure could come from any source, but in the case of a loaded cylinder the most reliable place is the cylinder itself. This pilot pressure would not be suitable for other functions because the cylinder may be in a position where load-induced pressure does not exist.

The circuit in ***Figure11-8*** *shows why it is less expensive to use slip-in cartridge valves to power an actuator requiring high flow. This circuit uses a cylinder with a 2:1 rod-area differential. Assume the application calls for 450 gpm. With the cylinder extending with 450 gpm entering the cap end, only 225 gpm exits from the head end.*

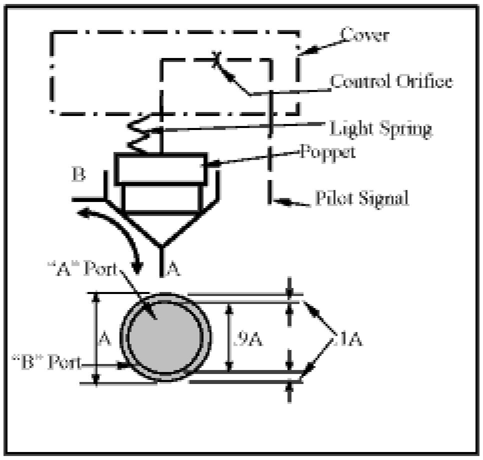
Fig. 11-8. Typical circuit for 1:2 slip-In cartridge valve



Conversely, while retracting at 450 gpm, 900 gpm exits from the cap end. Without special circuit design, a spool valve to cycle this cylinder would have to be capable of handling 900 gpm -- the high flow from the cap end. A spool valve with this flow capacity would be very large and expensive.

*Note that the circuit in **Figure 11-8** incorporates three different sizes of slip-in cartridge valves. Small cartridge CV at the cylinder head end handles the 225-gpm tank flow, medium-size cartridges CV2 and CV3 handle the 450 -gpm pump flow to both ends, and large cartridge CV4 at the cap end returns 900 gpm to tank. Each slip-in cartridge valve is sized to handle the flow it sees at a 50- to 75-psi pressure drop. These four standard cartridges, the manifold to contain them, and the directional control valve or valves to control them would cost less than half what a 900-gpm spool-type directional control valve would cost.*

Fig. 11-9. 1:1 poppet-type slip-in
cartridge valve



*A 1:2 area ratio is the most common slip-in cartridge valve design and fits more than 90% of all circuits. To meet some special requirements, there is also a 1:1.1 area ratio poppet shown in **Figure 11-9**. With this cartridge valve, area at the A port is 90% and area at the B port is 10% of poppet area. With these area ratios, fluid entering the A port flows at a much lower pressure drop than fluid entering the B port. Another way of saying this is it takes just 10% as much pressure to flow from A to B as it does to flow from B to A.*

Cartridge valves to control pressure

For pressure-control cartridge valves, the poppet has an area ratio of 1:1. This means flow can only go from the A port to the B port. A 1:1 area ratio makes the valve stable when controlling flow at pressure. Notice that the poppet is straight sided and sits on a tapered seat.

*The symbol and cutaway view in **Figure 11-10** illustrate a relief valve, but the design would function as a sequence or counterbalance valve as well. A relief valve never needs to have reverse flow, while a sequence or counterbalance valve usually requires a reverse-flow check valve piped around it. Also, as a sequence valve, it would always need a separate drain line to tank. These valves and an unloading valve are shown and discussed next.*

Fig. 11-10. 1:1 poppet-type slip-in cartridge valve as a
pilot-operated relief valve

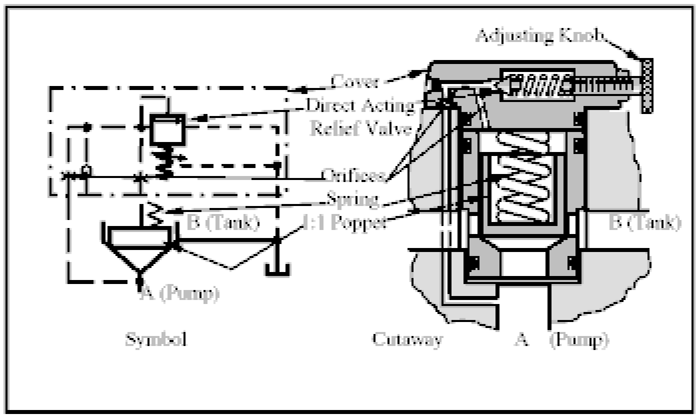
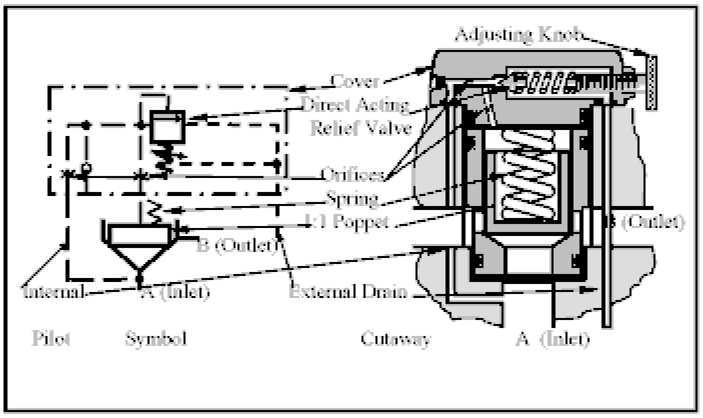


Figure 11-10 *shows a direct-acting relief valve symbol in the cover with pilot lines and orifices connecting it to a 1:1 area-ratio poppet. The A port is always the inlet, and as a relief valve, the B port is always the outlet to tank. The operation of a slip-in cartridge valve as a relief valve is identical to the poppet-type relief valve discussed in Chapter 9. The main attraction of slip-in cartridge valves is that they come in larger sizes for high-flow applications. Most relief valves for everyday circuits handle less that 200 gpm. Cartridge relief valves can handle flows in excess of 1500 gpm.*

Slip-in cartridge relief valves can be set up with all the features of the pilot-operated relief valves that were shown in Chapter 7. Solenoid venting and multiple pressure selection work in the same manner except for higher flow capabilities. Look at the symbols of the different types of arrangements for slip-in cartridge relief valves in Chapter 4.

Figure 11-11 *shows an internally piloted, externally drained sequence valve. This is the same valve illustrated in* ***Figure 11-10****, except the drain fluid from the direct-acting relief valve section must be ported directly to tank. If fluid is drained to the B port, backpressure at that port would add to the spring set pressure. In some cases, the internal pilot is changed to an external pilot as the circuit dictates. As mentioned previously, a sequence valve circuit often needs reverse flow so a cartridge-type check valve would be piped around it to provide free flow in reverse.*

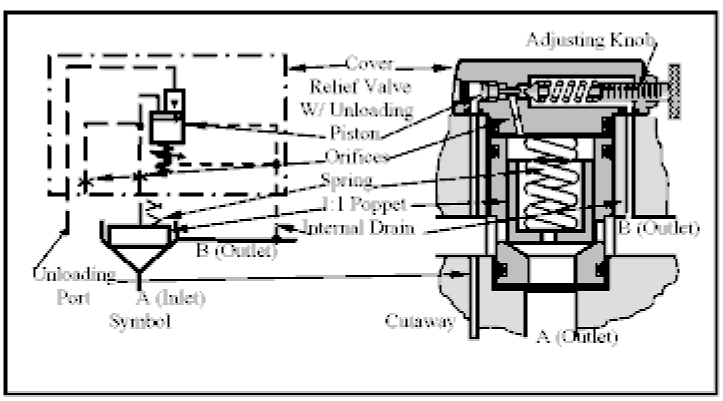
Fig. 11-11. 1:1 poppet-type slip-in cartridge valve as an internally piloted sequence valve or counterbalance valve



*The valve pictured in **Figure 11-11** also can perform as a counterbalance valve. Counterbalance valves can be internally or externally piloted but they usually are internally drained. Chapter 14 discusses sequence and counterbalance valves in detail and shows why pilot and drain functions are not always the same. Without exception, counterbalance valve circuits must have reverse flow so they always need a bypass check valve.*

*To add an unloading valve function to a cartridge valve, a special cover is required. The symbol and cutaway view in **Figure 11-12** show the setup for an unloading valve that could be used for a hi-lo pump circuit. Chapter 9 provides a complete explanation of unloading valves.*

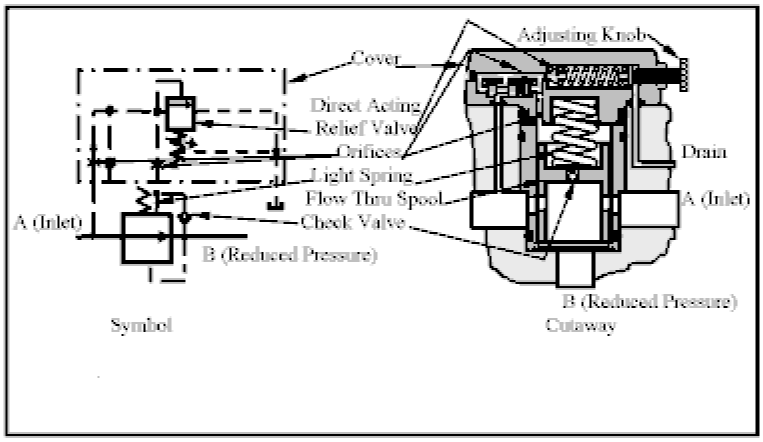
Fig. 11-12. 1:1 poppet-type slip-in cartridge valve as an unloading valve



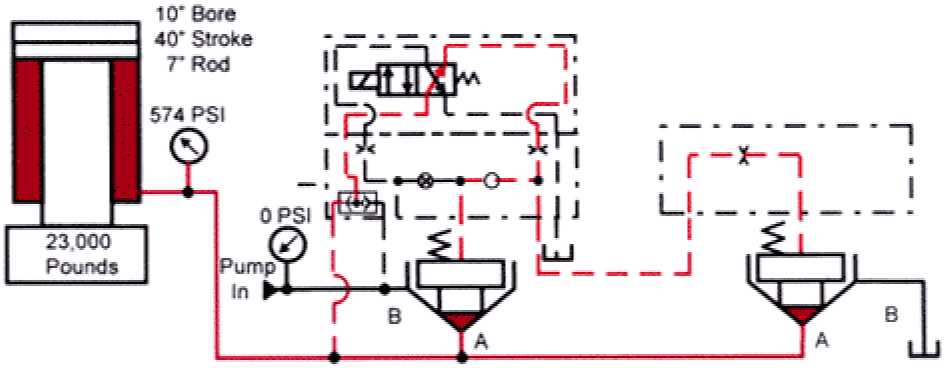
An unloading valve is similar to -- and functions just like -- a relief valve if the unloading port were not connected. Pressure would build enough to push the ball against the adjustable spring; then all excess pump flow would go to tank at set pressure. The addition of the unloading piston makes it possible to move the ball back far enough so that all pressure on top of the 1:1 poppet drops off. The valve then opens pump flow to tank at 30 to 50 psi. Pressure to the unloading port usually comes from a high-pressure pump. This keeps a high-volume pump unloaded during the work stroke.

The last of the pressure-control valves is the reducing valve. Unlike the other four pressure controls, a reducing valve is normally open instead of normally closed. The valve also is the one slip-in cartridge valve design that does not use a poppet to block fluid. Slip-in cartridge reducing valves are similar to in-line and subplate-mounted valves except that they have higher flow capabilities.

Fig. 11-13. Slip-in cartridge valve as a reducing valve



The symbol and cutaway view in ***Figure11-13*** *show a common construction for pilot-operated slip-in cartridge reducing valves. Notice the symbol for the cartridge is the standard ANSI-ISO drawing for any reducing valve. A cover containing a direct-acting relief valve sets the pressure. As fluid enters port A, it is free to go directly out port B . When outlet flow meets resistance, pressure starts to build and goes to the inlet of the direct-acting relief valve. The flow path is directly from ports A and B through the check valve. When pressure reaches the direct-acting relief valve setting, fluid starts to flow slowly through the valve. Flow increases as pressure continues to increase. At set pressure, flow through the direct-acting relief valve exceeds flow through the orifices. At that point, pressure on top of the flow-through spool drops. When this pressure drops enough, the spool raises and restricts outlet flow at set pressure. Flow never shuts off completely because flow from the reduced pressure is always going to tank through the direct-acting relief valve. When pressure at the outlet drops, the direct-acting relief valve closes and forces the passage through the spool to reopen.*


10" Bore
40" Stroke
7" Rod
574 PSI
23,000
Pounds
0 PSI
Pump
In
B
A
A
B

Slip-in cartridge directional valves (continued)

The symbol and cutaway for a slip-in cartridge valve in Figure 4-22 include a stroke-adjusting screw that limits poppet travel. Restricting flow by limiting poppet movement controls the actuator's maximum speed. The filled triangle in the poppet symbol shows the skirted or modified poppet that allows smooth flow change as it shifts.

The cutaway in Figure 4-23 shows one design of a slip-in cartridge with a stroke limiter. The cartridge function is identical to any 1:2-ratio poppet except for the limited movement. Restricting the poppet movement makes the cartridge function as a flow control as well as a directional valve.

Figure 4-23. Slip-in cartridge directional valve with plain cover, stroke limiter, and dampening function

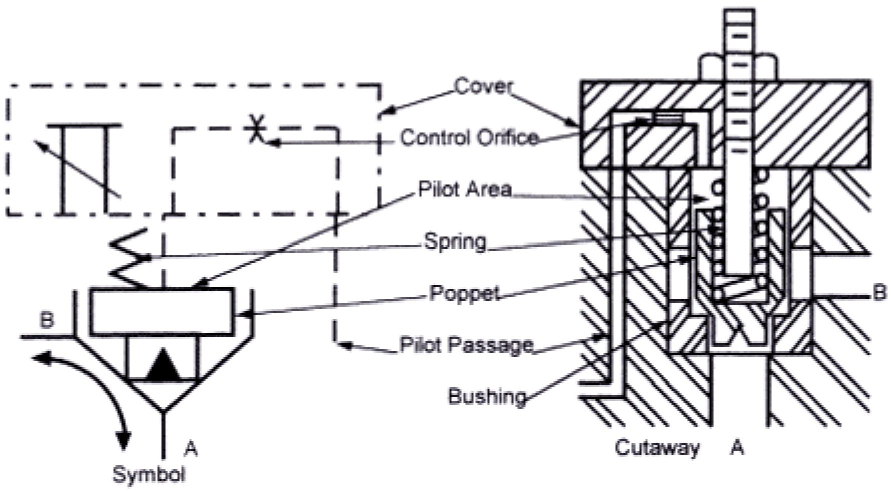


Figure 4-24 shows the symbol for an adjustable-stroke cartridge valve with a directional control valve cover. This particular valve only comes in a single-solenoid configuration as shown. Also, it cannot pilot other cartridge valves in the manifold. (Figures 4-27 and 4-28 show an adjustable stroke-cartridge in a circuit. These examples also show a problem that can occur when using a stroke limiter as a flow control in a meter-out circuit.)

Figure 4-24. Slip-in cartridge directional valve with single-solenoid operator (N.C.)

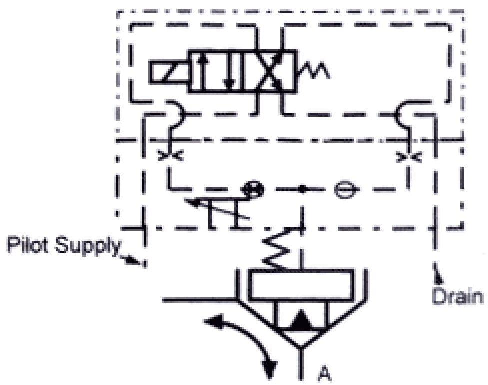
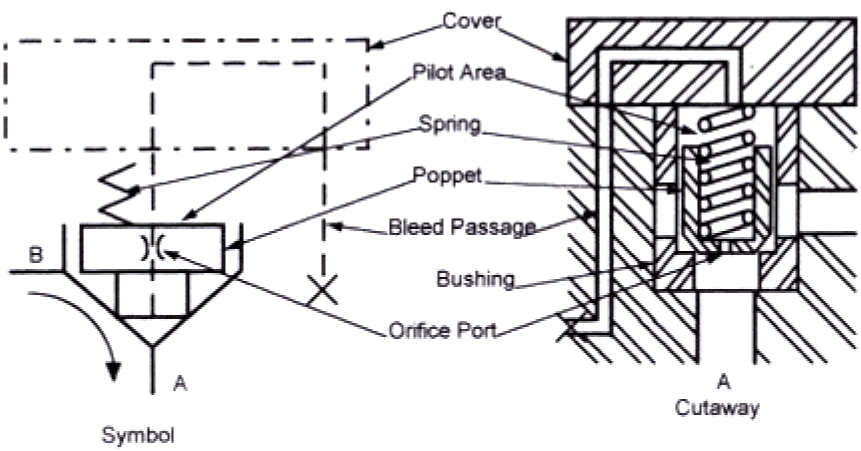


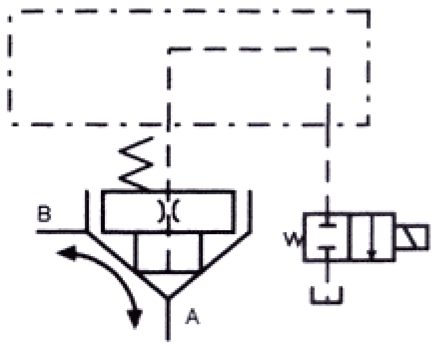
Figure 4-25 shows the symbol and cutaway for an internal-poppet, orifice-type cartridge valve. The internal poppet orifice supplies pilot oil from the A port only. Standard orifices that meet most needs are available. The internal pilot supply cartridge valve provides a check-valve function without drilling pilot passages in the manifold. As a check valve, it always allows free flow from the B port to the A port and blocks flow from the A port to the B port.

Figure 4-25. Slip-in cartridge directional valve with orifice port in poppet for pilot supply



The 2-way cartridge shut-off valve in Figure 4-26 is for high flow systems. This 2-way shut-off might allow pump flow to a circuit as shown in the schematic. Also use a 2-way shut-off to let fluid flow from a large cylinder to tank for rapid advance. Using a normally open solenoid valve in place of the normally closed one shown allows flow through the cartridge valve until the solenoid is energized.

Figure 4-26. Slip-in cartridge directional valve with NC orifice port in poppet for pilot supply

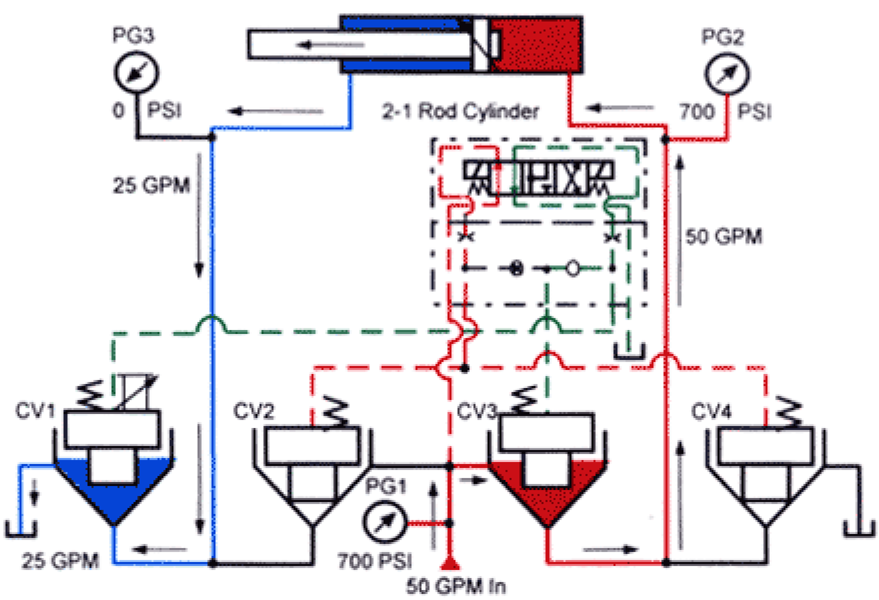


One advantage of the internal-pilot-supply-type cartridge valve is that it is not necessary to keep the pump running to have pilot pressure. This can eliminate a shuttle valve when an over-running load tries to move the cylinder.

The internally piloted slip-in cartridge always controls flow from the A port to the B port. Fluid is free to flow from the B port to the A port because pilot supply comes only from the A port.

When using a stroke-adjusted poppet to meter-out flow from a cylinder with an oversize piston rod, look out for the problem that appears in Figure 4-27. This circuit pictures a horizontal cylinder with a 2:1 rod that needs a meter-out flow control. This is good circuit design for spool-type valves, but when using an adjustable-stroke slip-in cartridge valve, it can cause trouble. This circuit can actually increase the cylinder speed when making an adjustment to slow it.

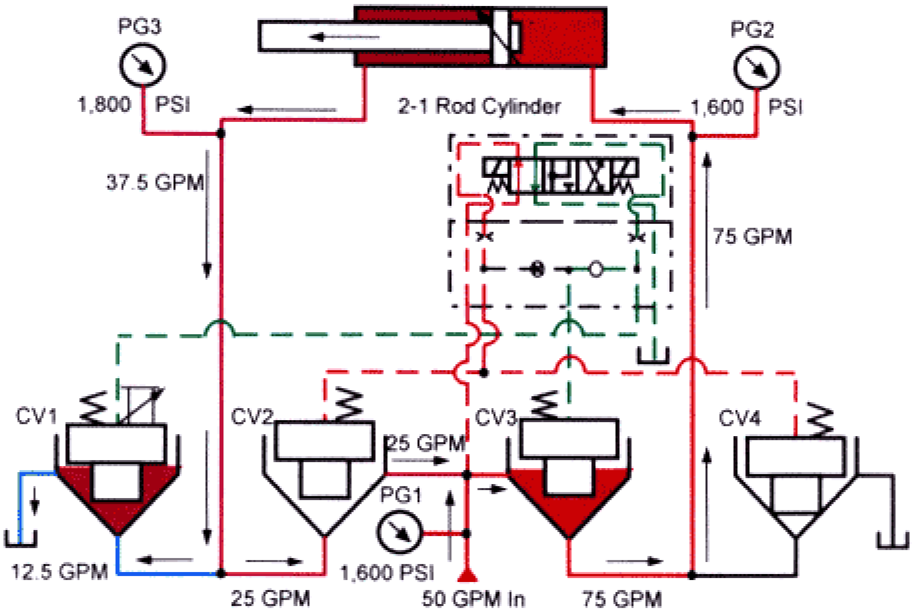
Figure 4-27. Slip-in cartridge valves with adjustable-stroke poppet at CV1 (extending with poppet on CV1 full open)



The circuit in Figure 4-27 shows the valves shifted to extend the cylinder. Flow from the pump is passing through CV3 to the cylinder's cap end. Oil from the cylinder's rod end is flowing to tank freely through CV1 because the stroke adjuster is fully open. Pressure gauge PG1 shows a system pressure of 700 psi. Gauge PG2 in the cylinder's cap line reads 700 psi, and PG3 at the cylinder's rod line reads 0 psi. The 700-psi reading is from the load's resistance (the cylinder is moving with no flow restriction). Pilot pressure is always the same as system pressure. Flow to the cylinder's cap end is 50 gpm and flow from the rod end to tank is 25 gpm.

With the stroke limiter screwed in to restrict tank flow to 12.5 gpm, the conditions shown in Figure 4-28 will prevail. In a normal meter-out circuit with a flow control and a spool-type directional valve, the cylinder speed slows and system pressure increases.

Figure 4-28. Slip-in cartridge valves with adjustable stroke poppet at CV1 (extending with poppet on CV1 set at 12.5 gpm)



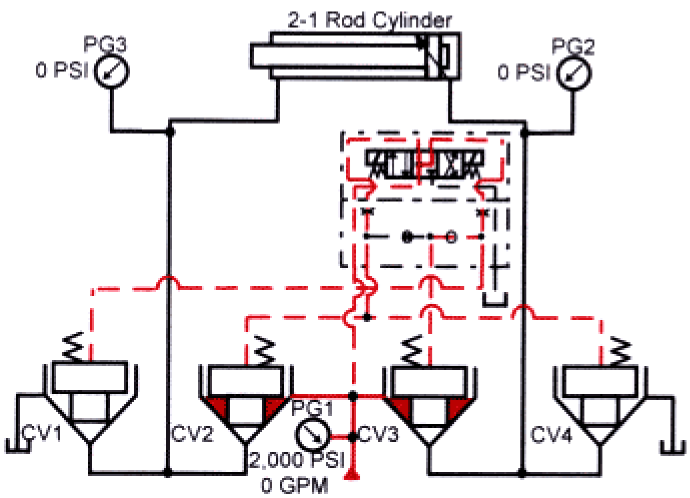
With a cartridge-valve circuit, however, restricting flow from the cylinder's rod end increases system pressure. Gauges PG1 and PG2 register approximately 1600 psi -- or a little more than twice the non-restricted flow pressure. This is because the load now is being moved by pressure on half the piston area in a regeneration circuit. Gauge PG3, at the cylinder's rod end, climbs to approximately 1800 psi due to area-ratio intensification.

This intensified pressure acts on the half A port area at CV2, while half the B port area sees system pressure. Pilot pressure on the full pilot area of CV2 is 1600 psi, plus a spring force of, say, 75 psi. If the full pilot area is one square inch, the poppet has a closing force of 1675 lb. The 800-lb opening force on the poppet is generated by 1600 psi on half the area. The opening force on the other half area of the poppet is 900 pounds (1800 psi X 1/2 sq. in.), making the total opening force 1700 lb. With 1700-lb opening force and 1675-lb closing force, the poppet opens to allow rod-end oil to regenerate to the cap end. Instead of the cylinder slowing to half speed, it *moves 150% faster due to regeneration. The more the flow from CV1 to tank decreases, the faster the cylinder extends. Note that restricting flow at CV3 as a meter-in flow-control circuit would allow infinite control of cylinder speed. Another option would be to use a shuttle cover and take pilot pressure from the pump or the cylinder's rod end. As pressure intensified at the rod end, pilot pressure to CV2 would increase also.*

Slip-in cartridge directional valves compared to spool-type 4-way directional valves

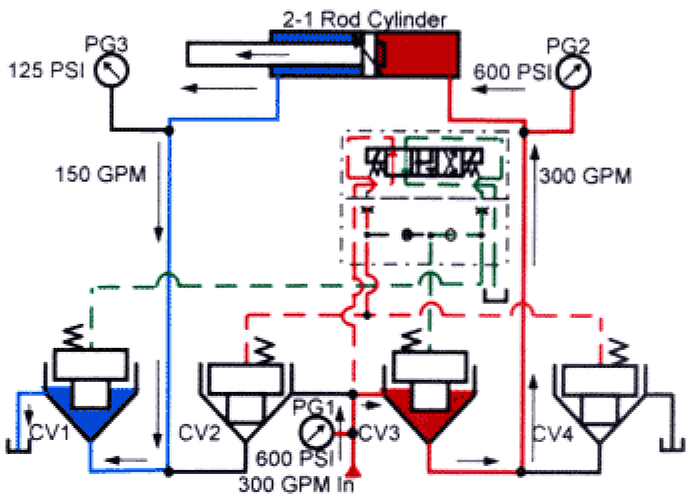
Figure 4-29 pictures a circuit with a 300-gpm pump powering a large-bore, 2:1 rod-diameter cylinder. This is the type of circuit that uses an important feature of slip-in cartridge valves. Flow from the rod end is only 150 gpm as the cylinder extends, but while retracting, flow from the cap end is 600 gpm. A conventional 4-way valve to operate the cylinder in Figure 4-32 must be capable of 600-gpm flow. A 4-way valve with this capacity is large and expensive. Its delivery may involve a long lead time.

Figure 4-29. Slip-in cartridge valves for high-flow circuits (at rest, pump running)



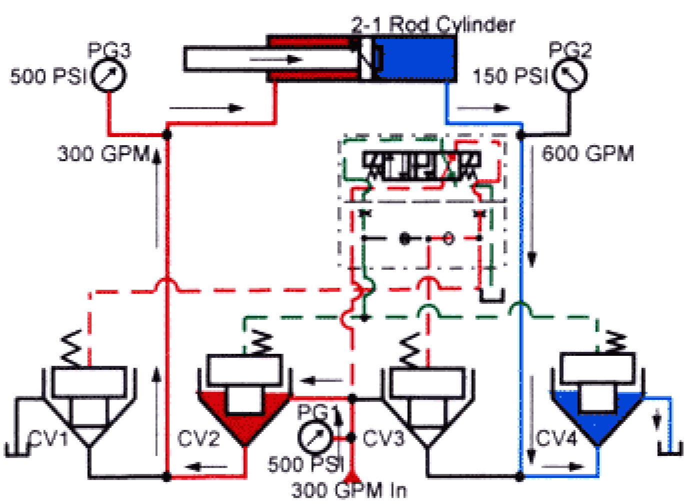
Four slip-in cartridge valves can duplicate the function of the 600 gpm 4-way valve. This may sound expensive and inefficient, but with a circuit such as the one in Figure 4-29, it actually is more efficient, less expensive, and saves space.

Figure 4-30. Slip-in cartridge valves for high-flow circuits (cylinder extending)



The cylinder is extending in Figure 4-30, with 300 gpm going to the cap end through CV3. Simultaneously, the rod end of the cylinder is discharging 150 gpm to tank through CV1. With this difference in flow, it costs less and saves space to use cartridges of different sizes. Size the cartridges for nominal pressure drop at their maximum flow.

Figure 4-31. Slip-in cartridge valves for high-flow circuits (cylinder retracting)

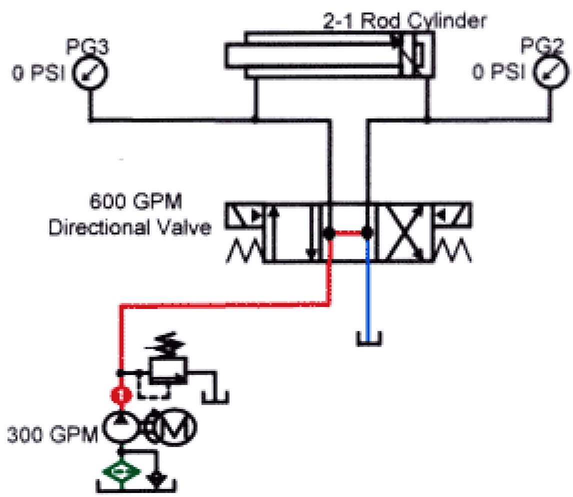


The cylinder is retracting in Figure 4-31. Flow to the rod end is 300 gpm while flow from the cap end is 600 gpm. For this higher cap-end flow, use a larger cartridge to minimize backpressure. This circuit will have three different-sized cartridges to carry the flow required during each phase of the cycle. Size CV1 for 150 gpm, CV4 for 600 gpm, and CV2 and CV3 for 300 gpm. When using a regeneration circuit, size CV3 for 600-gpm flow also.

The small amount of space taken by the cartridges, plus the lower cost and better availability of the parts make this system superior to one with a spool-type 4-way valve for high flows.

Another advantage of slip-in-type cartridge valves is their short response time. Cartridge poppets do not have land overlaps like spool valves have. Without land overlap there is flow when pilot pressure drops. Also, when the poppet opens, it only moves far enough to allow system flow to pass. When applying pilot pressure again, the poppet closes quickly without the extra travel often seen in spool valves. A spool valve, without stroke limiters, shifts full stroke. This full shifting may be far enough to pass several times the flow required. Then, when the spool starts returning to center, there is extra spool travel just to get back to controlling flow. This does not sound like much but faster response of cartridge valves can shorten cycle time and increase production.

Figure 4-32. Conventional 4-way directional valve for high-flow circuit (at rest, pump running)

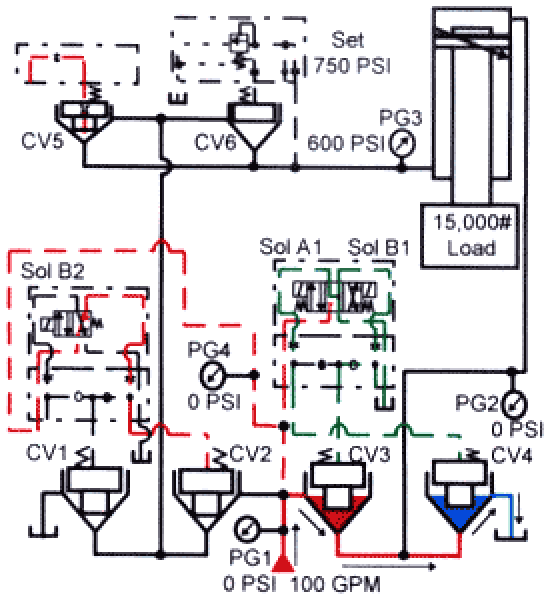


The circuit in Figures 4-29 to 4-32 uses only one pilot control valve. This limits the versatility of the slip-in cartridges. Multiple pilot control valves, shown in the following circuits, make the use of cartridge valves even more attractive.

Slip-in cartridge directional valves on running-away loads

Figures 4-33 to 4-36 show a vertically mounted (rod down) cylinder holding a heavy platen and tooling. This cylinder will run away if oil discharges to tank uncontrolled.

Figure 4-33. Slip-in cartridge valve circuit for running-away load (at rest, pump running)



A pressure-control cover (that makes CV6 a counterbalance valve) prevents rapid flow to tank. Cartridges CV1 through CV4 control cylinder flow and direction, and solenoid-operated directional valves shift their positions. Cartridge check valve CV5 bypasses normally closed counterbalance valve CV6 to retract the cylinder. With all solenoids deenergized, the pump unloads to tank through cartridge valves CV3 and CV4. Counterbalance valve CV6 keeps the load from falling at this time.

Figure 4-34. Slip-in cartridge valve circuit for running-away load (cylinder extending, regeneration)

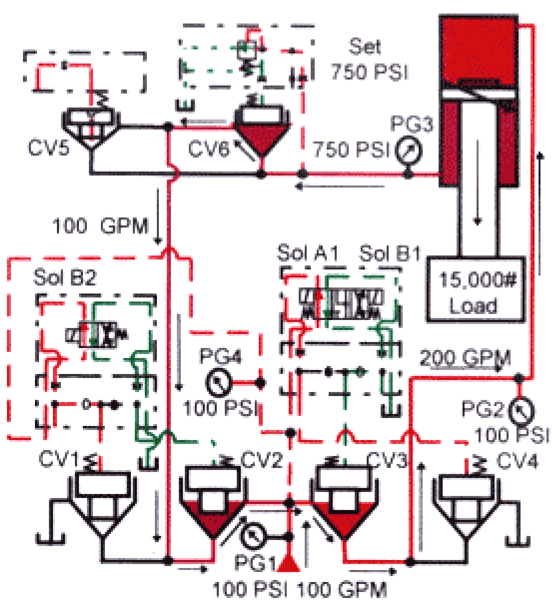
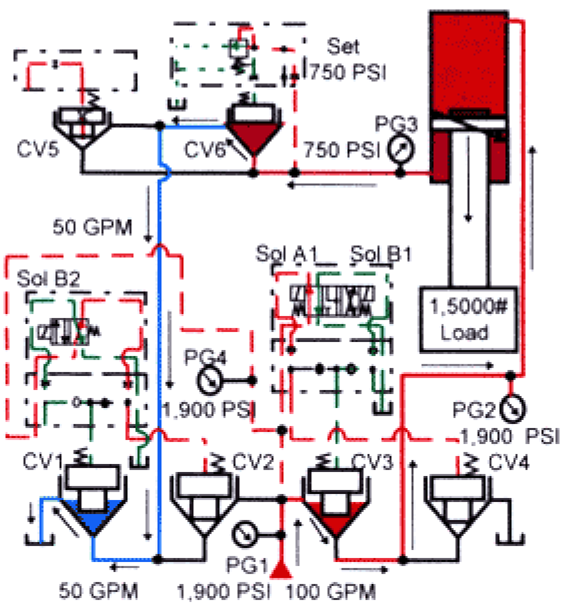


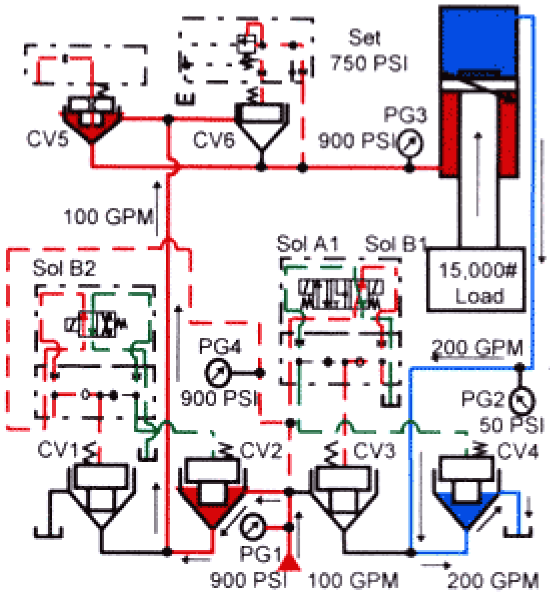
Figure 4-34 shows valve positions and likely pressures as the cylinder is regenerating forward. Energizing solenoids A1 and B2 closes CV4 and opens CV3, porting pump flow to the cylinder. This action also opens CV2 to allow cylinder's rod end flow to go through CV3, combine with pump flow, and regenerate the cylinder forward. Counterbalance valve CV6 keeps the forward motion of the cylinder from going faster than the pump and regeneration volume as indicated by the 750 psi seen on gauge PG3. The cylinder is extending rapidly at low or no force.

Figure 4-35. Slip-in cartridge valve circuit for running-away load (cylinder extending at full tonnage)



As the cylinder extends, it makes a limit switch to deenergize solenoid B2. When solenoid B2 drops out, Figure 4-35, pilot pressure closes CV2, while CV1 opens to tank. The cylinder slows to about half the regeneration speed, but is now able to generate full force. Counterbalance CV6 still keeps the cylinder from free-falling as it approaches the work. When the cylinder starts to form a part, pressure increases to whatever it takes to do the work. The cylinder continues extending until it finishes the work stroke.

Figure 4-36. Slip-in cartridge valve circuit for running-away load (cylinder retracting)



To retract the cylinder, the valve conditions shown in Figure 4-36 prevail. Energizing solenoids B1 and B2 allows CV2 and CV4 to open, and closes CV1 and CV3. Pump flow now goes to the cylinder's rod end through cartridge check valve CV5. That bypasses normally closed counterbalance valve CV6. Oil from the cylinder's cap end goes to tank through cartridge valve CV4.

Anytime all solenoids are deenergized, the cylinder stops and holds position. Counterbalance valve CV6 holds the cylinder in place as long as its pressure setting is greater than the load-induced pressure in the cylinder's rod end.

When sizing the counterbalance valve, be sure to consider the cylinder's static pressure. Slip-in cartridge valves have high flow capacity at nominal pressure drops. When available pressure drop is high, flow can increase to a point that the counterbalance valve's response is too slow to stop the cylinder quickly.

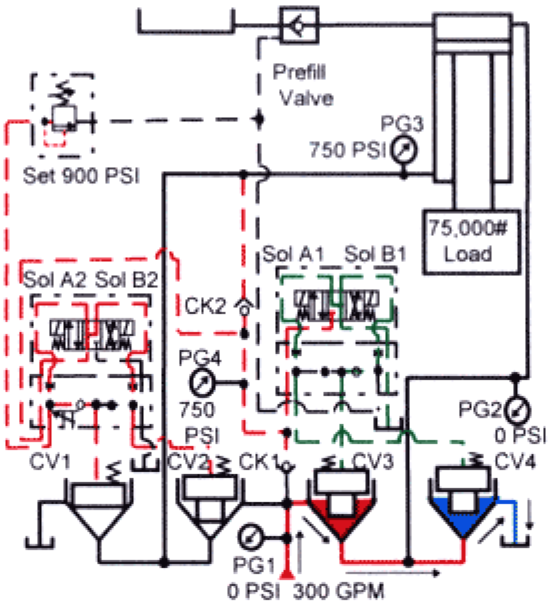
Slip-in cartridge directional valves with prefill valves

Figures 4-37 to 4-40 shows cartridge valves controlling a 50-in. cylinder with a 48.75-in. piston rod.

In Figure 4-37 the circuit is at rest with all solenoid valves deenergized. The cylinder maintains its position because load-induced pressure on the CV1 and CV2 pilot areas holds them closed. Pilot pressure reaches the pilot valves through CK1, while CK2 blocks flow to tank. When system pressure is higher than pressure in the cylinder's rod end, CK2 lets this higher pressure into the pilot circuit. The pump unloads to tank through CV3 and CV4.

Figure 4-37. Slip-in cartridge valve for vertically mounted cylinder with prefill valve

(at rest, pump running)

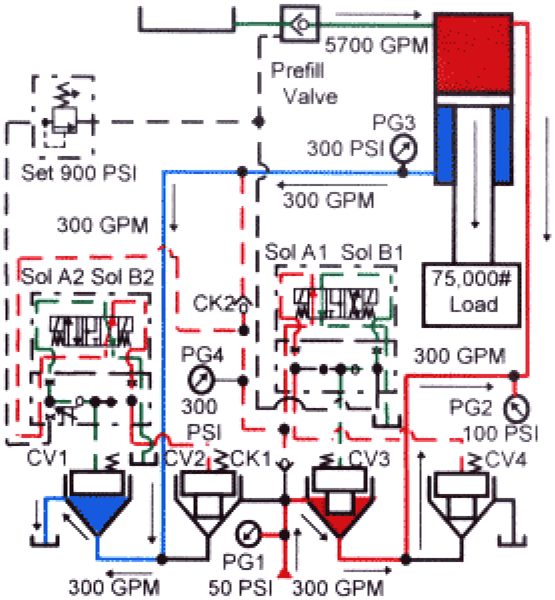


The size of CV1 is important, because controlling flow through it sets the cylinder free-fall speed. Size the valve for the pressure drop generated by load-induced pressure. A stroke limiter in CV1 actually sets maximum cylinder extension speed.

Energizing solenoid A1 shifts CV4 closed to block tank flow and leaves CV3 open to send pump flow to the cylinder's cap end. Figure 4-38 shows the cylinder in a controlled free fall. Energizing solenoid B2 lets CV1 open, while holding CV2 closed. Oil from the cylinder's rod end now has a path to tank through CV1.

A prefill valve lets oil from the tank into the cylinder's cap end. The cylinder will advance as fast as the stroke limiter on CV1 allows. Free-fall speed can be in excess of 15 in./sec.

Figure 4-38. Slip-in cartridge valve for vertically mounted cylinder with prefill valve (cylinder extending, controlled fall)



As the cylinder extends in free fall, it contacts a limit switch that de-energizes solenoid B2, Figure 4-39. CV2 remains closed and CV1 tries to close. As CV1 is closing, backpressure on the cylinder's rod end will build to 900 psi and the pressure control will keep CV1 from fully closing. Because CV1 is restricting flow at 900 psi, the cylinder decelerates and tries to stop. While the cylinder is slowing, decreased vacuum in the cap end lets the pre-fill valve close. After the prefill closes, pump flow forces the cylinder to keep moving and rod-end pressure keeps the pressure control on CV1 open. Deceleration is smooth and rapid. The cylinder continues extending toward the work at the slower pump rate.

Figure 4-39. Slip-in cartridge valve for vertically mounted cylinder with prefill valve (cylinder approaching work, decelerating)

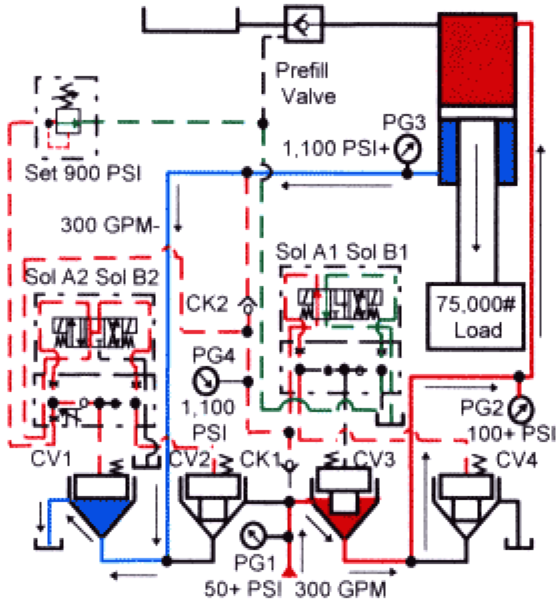
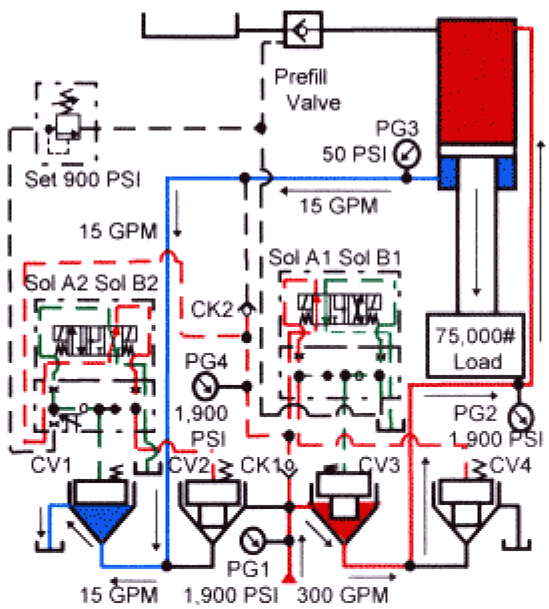
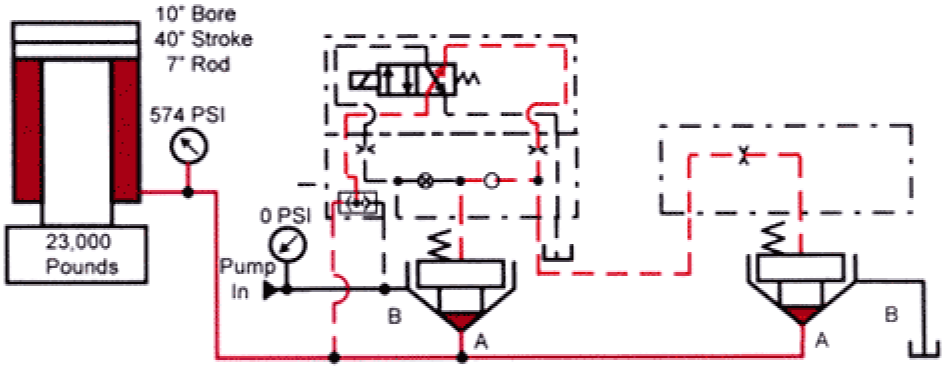


Figure 4-40 shows the cylinder at work. Solenoid A1 is still energized and solenoid B2 has been energized again. (Solenoid B2 could be reenergized by a limit switch or by a pressure switch when the cylinder contacts the work.) Energizing solenoid B2 lets CV1 open fully, taking away the 900-psi backpressure that decreases tonnage. The cylinder extends at the force required to do the work (up to relief pressure setting).

Figure 4-40. Slip-in cartridge valve for vertically mounted cylinder with prefill valve (cylinder pressing)



To retract the cylinder, energize solenoids A2 and B1. Solenoid B2 closes CV3, pilots the pre-fill valve open, and opens CV4 to tank. Solenoid A2 closes CV1 and opens CV2, sending pump flow to the cylinder's rod end. The cylinder retracts rapidly with most of the cap-end flow going to tank through the pre-fill valve. In case of power failure or emergency stop, the cylinder stays where it is, or if it is moving, it decelerates, stops, and holds its position.


10" Bore
40" Stroke
7" Rod
574 PSI
23,000
Pounds
0 PSI
Pump
In
B
A
A
B

Slip-in cartridge directional valves (continued)

The symbol and cutaway for a slip-in cartridge valve in Figure 4-22 include a stroke-adjusting screw that limits poppet travel. Restricting flow by limiting poppet movement controls the actuator's maximum speed. The filled triangle in the poppet symbol shows the skirted or modified poppet that allows smooth flow change as it shifts.

The cutaway in Figure 4-23 shows one design of a slip-in cartridge with a stroke limiter. The cartridge function is identical to any 1:2-ratio poppet except for the limited movement. Restricting the poppet movement makes the cartridge function as a flow control as well as a directional valve.

Figure 4-23. Slip-in cartridge directional valve with plain cover, stroke limiter, and dampening function

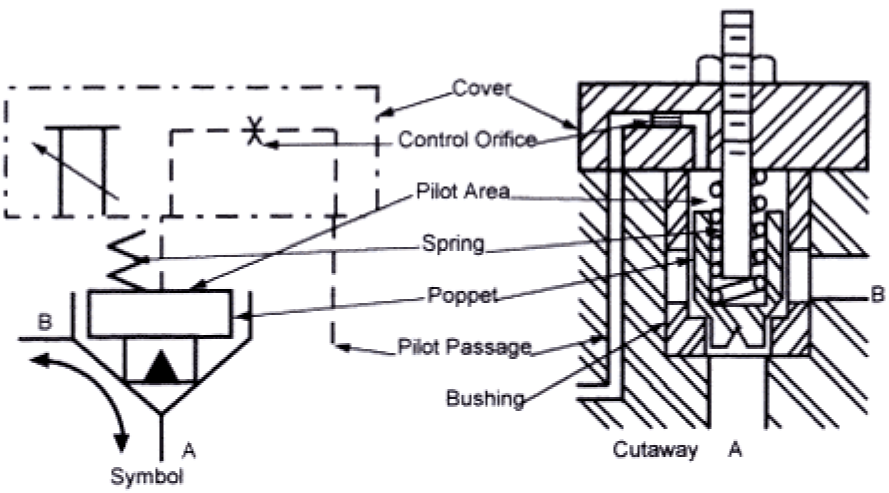


Figure 4-24 shows the symbol for an adjustable-stroke cartridge valve with a directional control valve cover. This particular valve only comes in a single-solenoid configuration as shown. Also, it cannot pilot other cartridge valves in the manifold. (Figures 4-27 and 4-28 show an adjustable stroke-cartridge in a circuit. These examples also show a problem that can occur when using a stroke limiter as a flow control in a meter-out circuit.)

Figure 4-24. Slip-in cartridge directional valve with single-solenoid operator (N.C.)

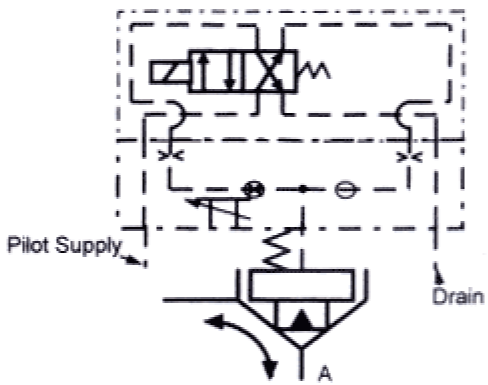
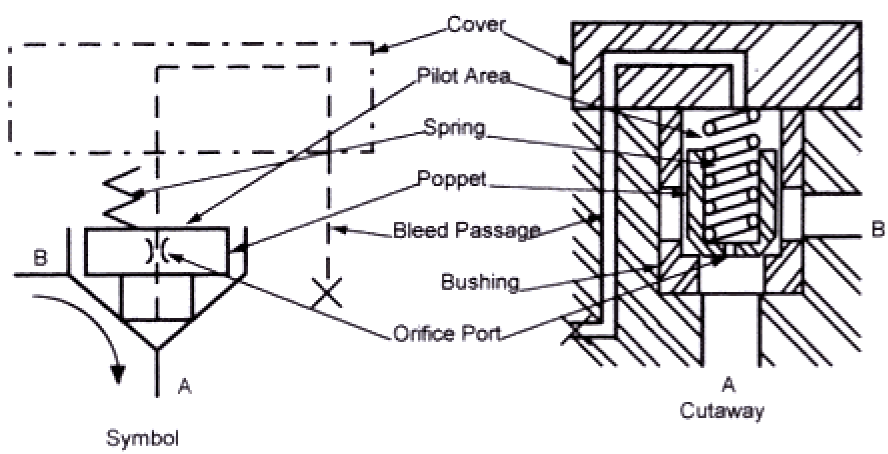


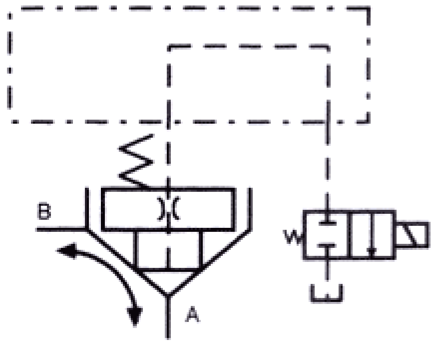
Figure 4-25 shows the symbol and cutaway for an internal-poppet, orifice-type cartridge valve. The internal poppet orifice supplies pilot oil from the A port only. Standard orifices that meet most needs are available. The internal pilot supply cartridge valve provides a check-valve function without drilling pilot passages in the manifold. As a check valve, it always allows free flow from the B port to the A port and blocks flow from the A port to the B port.

Figure 4-25. Slip-in cartridge directional valve with orifice port in poppet for pilot supply



The 2-way cartridge shut-off valve in Figure 4-26 is for high flow systems. This 2-way shut-off might allow pump flow to a circuit as shown in the schematic. Also use a 2-way shut-off to let fluid flow from a large cylinder to tank for rapid advance. Using a normally open solenoid valve in place of the normally closed one shown allows flow through the cartridge valve until the solenoid is energized.

Figure 4-26. Slip-in cartridge directional valve with NC orifice port in poppet for pilot supply

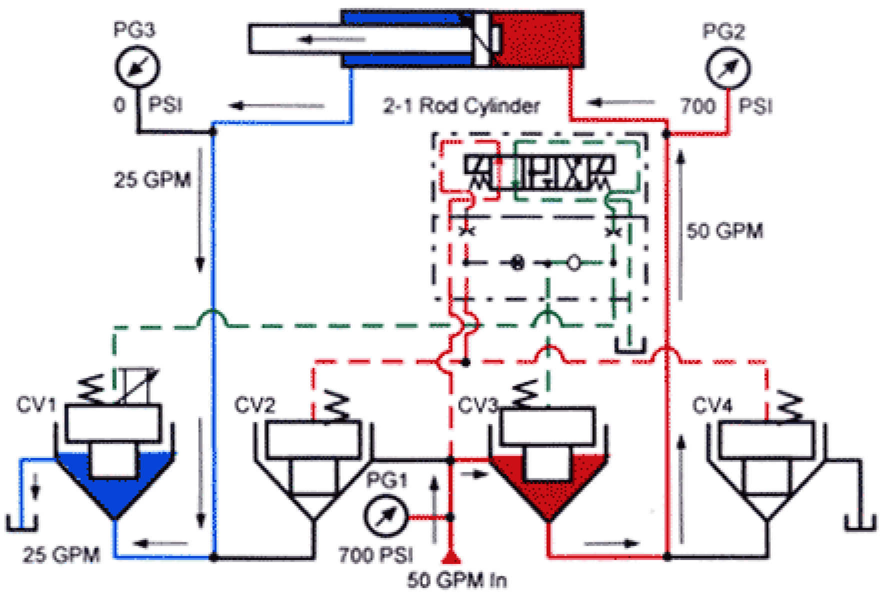


One advantage of the internal-pilot-supply-type cartridge valve is that it is not necessary to keep the pump running to have pilot pressure. This can eliminate a shuttle valve when an over-running load tries to move the cylinder.

The internally piloted slip-in cartridge always controls flow from the A port to the B port. Fluid is free to flow from the B port to the A port because pilot supply comes only from the A port.

When using a stroke-adjusted poppet to meter-out flow from a cylinder with an oversize piston rod, look out for the problem that appears in Figure 4-27. This circuit pictures a horizontal cylinder with a 2:1 rod that needs a meter-out flow control. This is good circuit design for spool-type valves, but when using an adjustable-stroke slip-in cartridge valve, it can cause trouble. This circuit can actually increase the cylinder speed when making an adjustment to slow it.

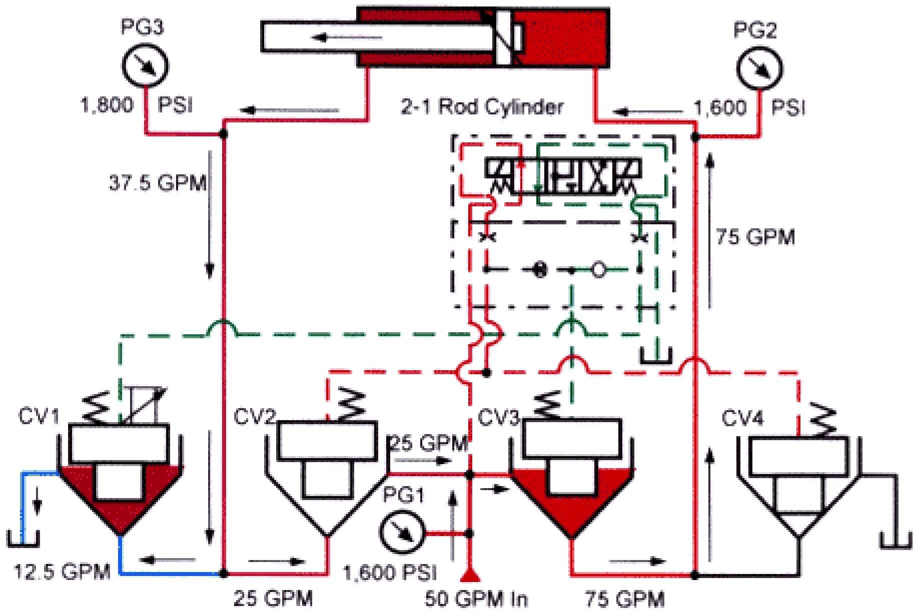
Figure 4-27. Slip-in cartridge valves with adjustable-stroke poppet at CV1
(extending with poppet on CV1 full open)



The circuit in Figure 4-27 shows the valves shifted to extend the cylinder. Flow from the pump is passing through CV3 to the cylinder's cap end. Oil from the cylinder's rod end is flowing to tank freely through CV1 because the stroke adjuster is fully open. Pressure gauge PG1 shows a system pressure of 700 psi. Gauge PG2 in the cylinder's cap line reads 700 psi, and PG3 at the cylinder's rod line reads 0 psi. The 700-psi reading is from the load's resistance (the cylinder is moving with no flow restriction). Pilot pressure is always the same as system pressure. Flow to the cylinder's cap end is 50 gpm and flow from the rod end to tank is 25 gpm.

With the stroke limiter screwed in to restrict tank flow to 12.5 gpm, the conditions shown in Figure 4-28 will prevail. In a normal meter-out circuit with a flow control and a spool-type directional valve, the cylinder speed slows and system pressure increases.

Figure 4-28. Slip-in cartridge valves with adjustable stroke poppet at CV1
(extending with poppet on CV1 set at 12.5 gpm)



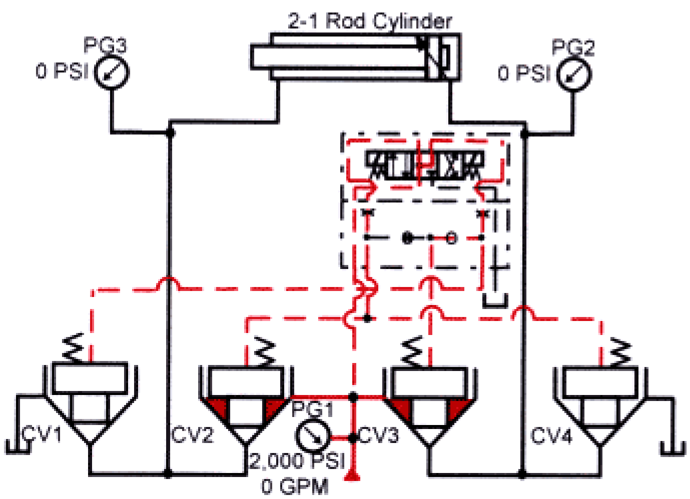
With a cartridge-valve circuit, however, restricting flow from the cylinder's rod end increases system pressure. Gauges PG1 and PG2 register approximately 1600 psi -- or a little more than twice the non-restricted flow pressure. This is because the load now is being moved by pressure on half the piston area in a regeneration circuit. Gauge PG3, at the cylinder's rod end, climbs to approximately 1800 psi due to area-ratio intensification.

This intensified pressure acts on the half A port area at CV2, while half the B port area sees system pressure. Pilot pressure on the full pilot area of CV2 is 1600 psi, plus a spring force of, say, 75 psi. If the full pilot area is one square inch, the poppet has a closing force of 1675 lb. The 800-lb opening force on the poppet is generated by 1600 psi on half the area. The opening force on the other half area of the poppet is 900 pounds (1800 psi X 1/2 sq. in.), making the total opening force 1700 lb. With 1700-lb opening force and 1675-lb closing force, the poppet opens to allow rod-end oil to regenerate to the cap end. Instead of the cylinder slowing to half speed, it moves 150% faster due to regeneration. The more the flow from CV1 to tank decreases, the faster the cylinder extends. Note that restricting flow at CV3 as a meter-in flow-control circuit would allow infinite control of cylinder speed. Another option would be to use a shuttle cover and take pilot pressure from the pump or the cylinder's rod end. As pressure intensified at the rod end, pilot pressure to CV2 would increase also.

Slip-in cartridge directional valves compared to spool-type 4-way directional valves

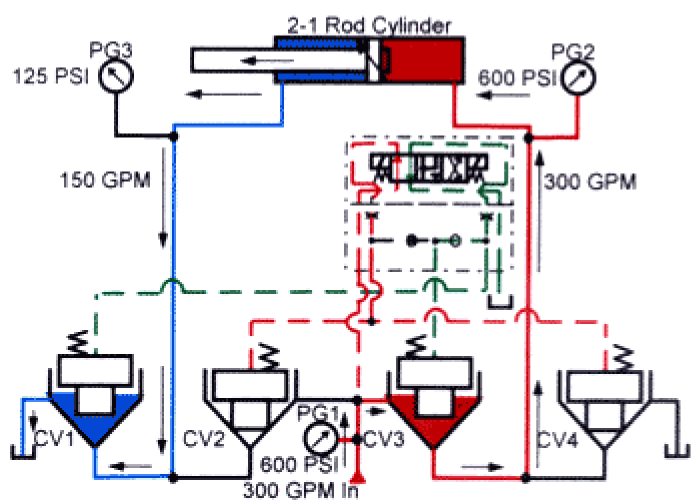
Figure 4-29 pictures a circuit with a 300-gpm pump powering a large-bore, 2:1 rod-diameter cylinder. This is the type of circuit that uses an important feature of slip-in cartridge valves. Flow from the rod end is only 150 gpm as the cylinder extends, but while retracting, flow from the cap end is 600 gpm. A conventional 4-way valve to operate the cylinder in Figure 4-32 must *be capable of 600-gpm flow. A 4-way valve with this capacity is large and expensive. Its delivery may involve a long lead time.*

Figure 4-29. Slip-in cartridge valves for high-flow circuits (at rest, pump running)



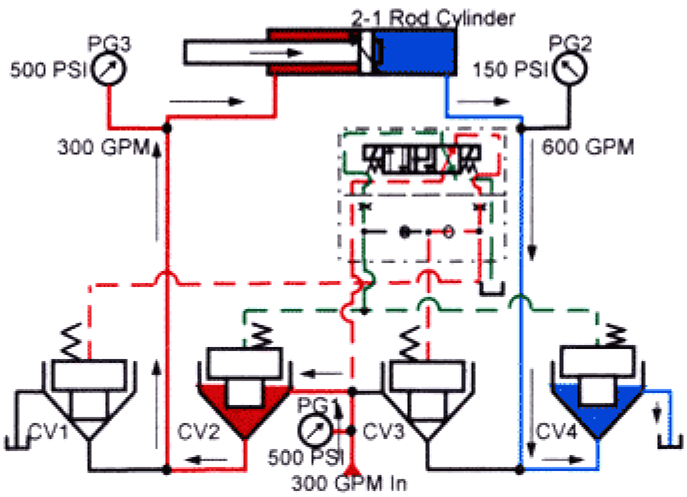
Four slip-in cartridge valves can duplicate the function of the 600 gpm 4-way valve. This may sound expensive and inefficient, but with a circuit such as the one in Figure 4-29, it actually is more efficient, less expensive, and saves space.

Figure 4-30. Slip-in cartridge valves for high-flow circuits (cylinder extending)



The cylinder is extending in Figure 4-30, with 300 gpm going to the cap end through CV3. Simultaneously, the rod end of the cylinder is discharging 150 gpm to tank through CV1. With *this difference in flow, it costs less and saves space to use cartridges of different sizes. Size the cartridges for nominal pressure drop at their maximum flow.*

Figure 4-31. Slip-in cartridge valves for high-flow circuits (cylinder retracting)

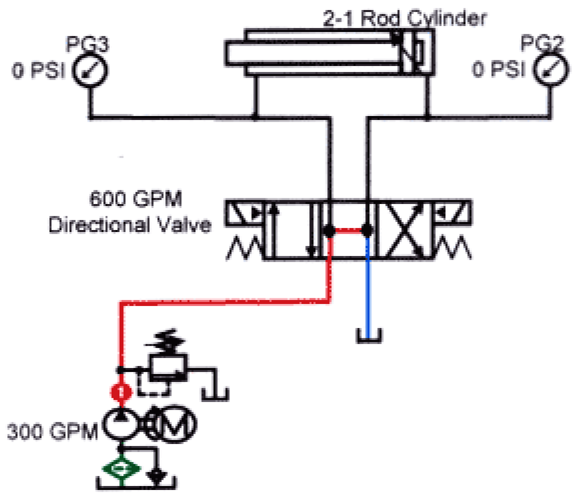


The cylinder is retracting in Figure 4-31. Flow to the rod end is 300 gpm while flow from the cap end is 600 gpm. For this higher cap-end flow, use a larger cartridge to minimize backpressure. This circuit will have three different-sized cartridges to carry the flow required during each phase of the cycle. Size CV1 for 150 gpm, CV4 for 600 gpm, and CV2 and CV3 for 300 gpm. When using a regeneration circuit, size CV3 for 600-gpm flow also.

The small amount of space taken by the cartridges, plus the lower cost and better availability of the parts make this system superior to one with a spool-type 4-way valve for high flows.

Another advantage of slip-in-type cartridge valves is their short response time. Cartridge poppets do not have land overlaps like spool valves have. Without land overlap there is flow when pilot pressure drops. Also, when the poppet opens, it only moves far enough to allow system flow to pass. When applying pilot pressure again, the poppet closes quickly without the extra travel often seen in spool valves. A spool valve, without stroke limiters, shifts full stroke. This full shifting may be far enough to pass several times the flow required. Then, when the spool starts returning to center, there is extra spool travel just to get back to controlling flow. This does not sound like much but faster response of cartridge valves can shorten cycle time and increase production.

Figure 4-32. Conventional 4-way directional valve for high-flow circuit (at rest, pump running)

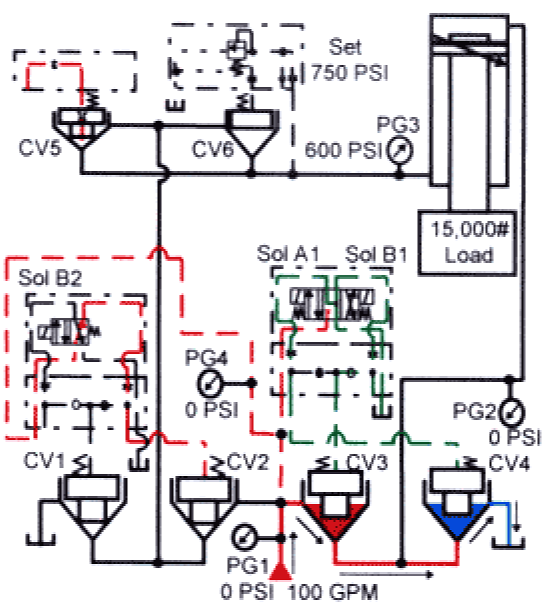


The circuit in Figures 4-29 to 4-32 uses only one pilot control valve. This limits the versatility of the slip-in cartridges. Multiple pilot control valves, shown in the following circuits, make the use of cartridge valves even more attractive.

Slip-in cartridge directional valves on running-away loads

Figures 4-33 to 4-36 show a vertically mounted (rod down) cylinder holding a heavy platen and tooling. This cylinder will run away if oil discharges to tank uncontrolled.

Figure 4-33. Slip-in cartridge valve circuit for running-away load (at rest, pump running)



A pressure-control cover (that makes CV6 a counterbalance valve) prevents rapid flow to tank. Cartridges CV1 through CV4 control cylinder flow and direction, and solenoid-operated directional valves shift their positions. Cartridge check valve CV5 bypasses normally closed counterbalance valve CV6 to retract the cylinder. With all solenoids deenergized, the pump unloads to tank through cartridge valves CV3 and CV4. Counterbalance valve CV6 keeps the load from falling at this time.

Figure 4-34. Slip-in cartridge valve circuit for running-away load (cylinder extending, regeneration)

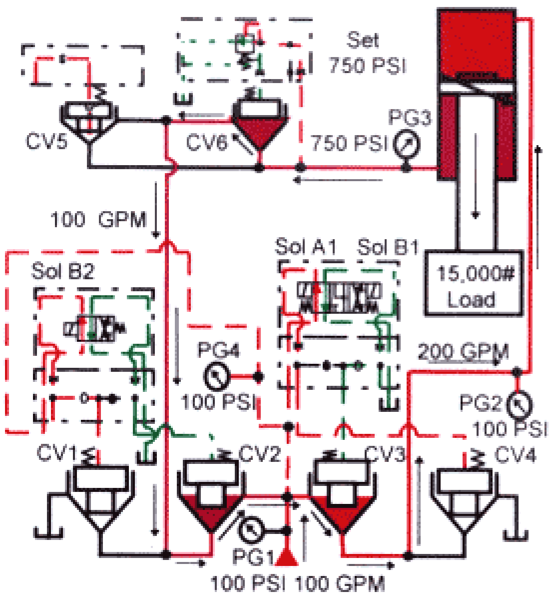
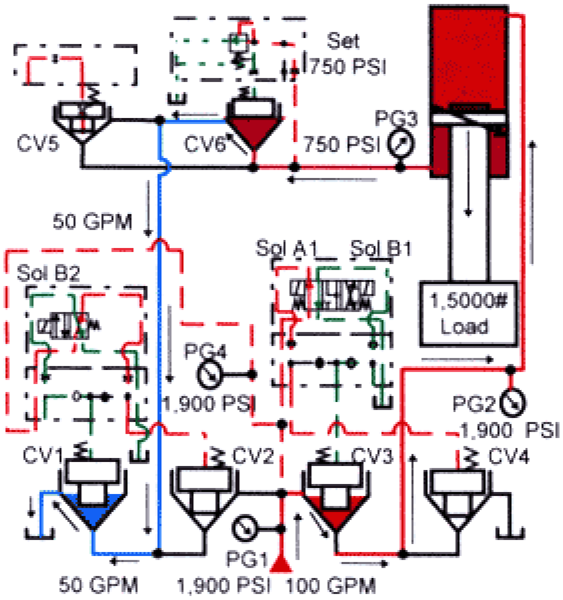


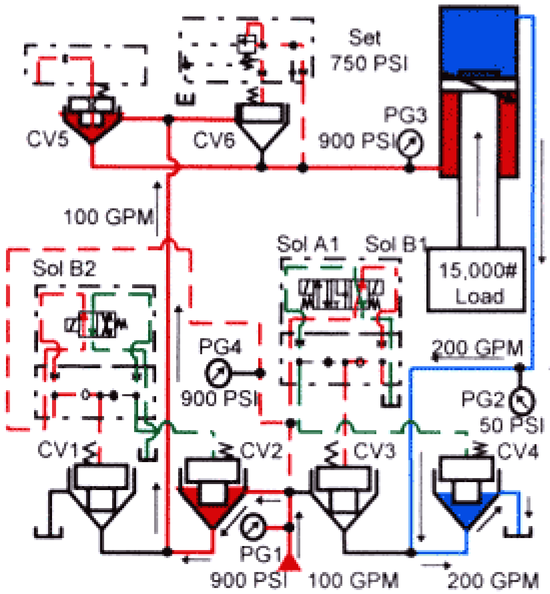
Figure 4-34 shows valve positions and likely pressures as the cylinder is regenerating forward. Energizing solenoids A1 and B2 closes CV4 and opens CV3, porting pump flow to the cylinder. This action also opens CV2 to allow cylinder's rod end flow to go through CV3, combine with pump flow, and regenerate the cylinder forward. Counterbalance valve CV6 keeps the forward motion of the cylinder from going faster than the pump and regeneration volume as indicated by the 750 psi seen on gauge PG3. The cylinder is extending rapidly at low or no force.

Figure 4-35. Slip-in cartridge valve circuit for running-away load (cylinder extending at full tonnage)



As the cylinder extends, it makes a limit switch to deenergize solenoid B2. When solenoid B2 drops out, Figure 4-35, pilot pressure closes CV2, while CV1 opens to tank. The cylinder slows to about half the regeneration speed, but is now able to generate full force. Counterbalance CV6 still keeps the cylinder from free-falling as it approaches the work. When the cylinder starts to form a part, pressure increases to whatever it takes to do the work. The cylinder continues extending until it finishes the work stroke.

Figure 4-36. Slip-in cartridge valve circuit for running-away load (cylinder retracting)



To retract the cylinder, the valve conditions shown in Figure 4-36 prevail. Energizing solenoids B1 and B2 allows CV2 and CV4 to open, and closes CV1 and CV3. Pump flow now goes to the cylinder's rod end through cartridge check valve CV5. That bypasses normally closed counterbalance valve CV6. Oil from the cylinder's cap end goes to tank through cartridge valve CV4.

Anytime all solenoids are deenergized, the cylinder stops and holds position. Counterbalance valve CV6 holds the cylinder in place as long as its pressure setting is greater than the load-induced pressure in the cylinder's rod end.

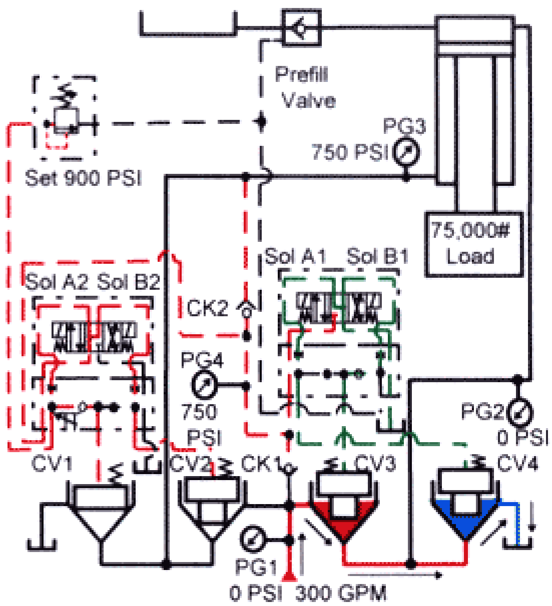
When sizing the counterbalance valve, be sure to consider the cylinder's static pressure. Slip-in cartridge valves have high flow capacity at nominal pressure drops. When available pressure drop is high, flow can increase to a point that the counterbalance valve's response is too slow to stop the cylinder quickly.

Slip-in cartridge directional valves with prefill valves

Figures 4-37 to 4-40 shows cartridge valves controlling a 50-in. cylinder with a 48.75-in. piston rod.

In Figure 4-37 the circuit is at rest with all solenoid valves deenergized. The cylinder maintains its position because load-induced pressure on the CV1 and CV2 pilot areas holds them closed. Pilot pressure reaches the pilot valves through CK1, while CK2 blocks flow to tank. When system pressure is higher than pressure in the cylinder's rod end, CK2 lets this higher pressure into the pilot circuit. The pump unloads to tank through CV3 and CV4.

Figure 4-37. Slip-in cartridge valve for vertically mounted cylinder with prefill valve (at rest, pump running)

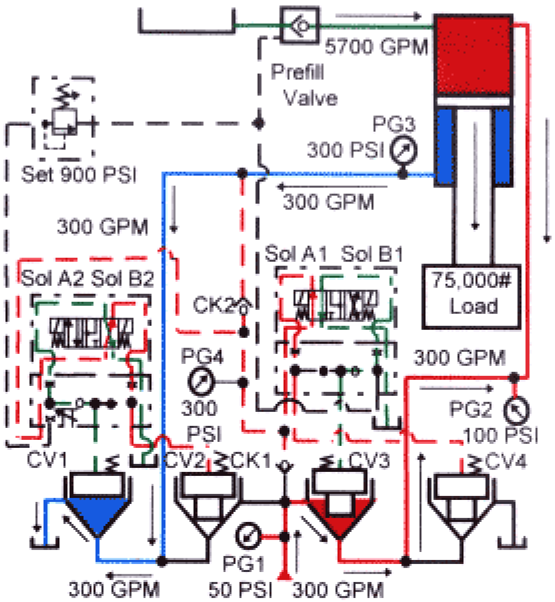


The size of CV1 is important, because controlling flow through it sets the cylinder free-fall speed. Size the valve for the pressure drop generated by load-induced pressure. A stroke limiter in CV1 actually sets maximum cylinder extension speed.

Energizing solenoid A1 shifts CV4 closed to block tank flow and leaves CV3 open to send pump flow to the cylinder's cap end. Figure 4-38 shows the cylinder in a controlled free fall. Energizing solenoid B2 lets CV1 open, while holding CV2 closed. Oil from the cylinder's rod end now has a path to tank through CV1.

A prefill valve lets oil from the tank into the cylinder's cap end. The cylinder will advance as fast as the stroke limiter on CV1 allows. Free-fall speed can be in excess of 15 in./sec.

Figure 4-38. Slip-in cartridge valve for vertically mounted cylinder with prefill valve (cylinder extending, controlled fall)



As the cylinder extends in free fall, it contacts a limit switch that de-energizes solenoid B2, Figure 4-39. CV2 remains closed and CV1 tries to close. As CV1 is closing, backpressure on the cylinder's rod end will build to 900 psi and the pressure control will keep CV1 from fully closing. Because CV1 is restricting flow at 900 psi, the cylinder decelerates and tries to stop. While the cylinder is slowing, decreased vacuum in the cap end lets the pre-fill valve close. After the prefill closes, pump flow forces the cylinder to keep moving and rod-end pressure keeps the pressure control on CV1 open. Deceleration is smooth and rapid. The cylinder continues extending toward the work at the slower pump rate.

Figure 4-39. Slip-in cartridge valve for vertically mounted cylinder with prefill valve (cylinder approaching work, decelerating)

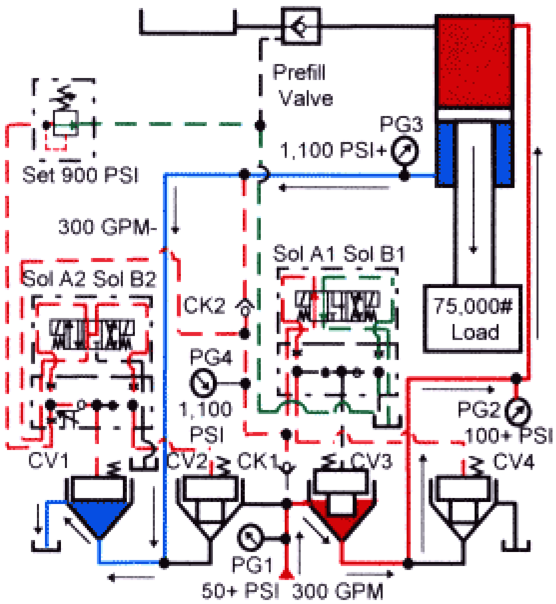
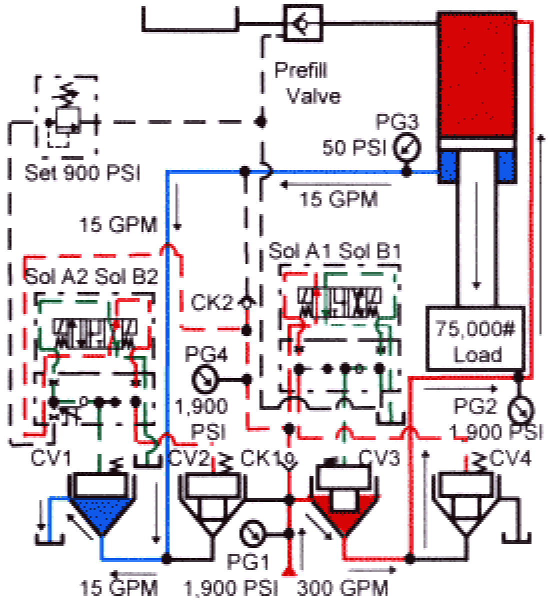


Figure 4-40 shows the cylinder at work. Solenoid A1 is still energized and solenoid B2 has been energized again. (Solenoid B2 could be reenergized by a limit switch or by a pressure switch when the cylinder contacts the work.) Energizing solenoid B2 lets CV1 open fully, taking away the 900-psi backpressure that decreases tonnage. The cylinder extends at the force required to do the work (up to relief pressure setting).

Figure 4-40. Slip-in cartridge valve for vertically mounted cylinder with prefill valve (cylinder pressing)



To retract the cylinder, energize solenoids A2 and B1. Solenoid B2 closes CV3, pilots the pre-fill valve open, and opens CV4 to tank. Solenoid A2 closes CV1 and opens CV2, sending pump flow to the cylinder's rod end. The cylinder retracts rapidly with most of the cap-end flow going to tank through the pre-fill valve. In case of power failure or emergency stop, the cylinder stays where it is, or if it is moving, it decelerates, stops, and holds its position.

Proportional and servovalves

Infinitely variable directional control valves

The directional control valves discussed so far in this series have all been configured to either pass full flow or completely block flow. The only way to decrease flow through these valves is by adding flow controls or by mechanically limiting movement of an internal part.

The first infinitely variable valve available was the servovalve. Internal flow-modifying parts could be moved to any position at any rate, so output from any port could be varied at will. (Some call these valves infinitely variable 4-way flow controls.) The main problem with servovalves was (and still is) that they require very clean fluid to keep them operating effectively. Fluid from a standard well-maintained hydraulic circuit contains enough contamination to cause most servovalves to fail in a matter of minutes or only last a few hours at best. This meant that the original servovalves were tried and removed from many machines that needed precise control but not at the perceived cost of cleaning up the hydraulic oil.

Why use infinitely variable valves?

Some actuators must move at a precise speed, stop at a close-tolerance position, or produce a very accurate force to perform the work for which they were designed. With the proper input signals and feedback devices, proportional or servovalves can make an actuator perform any or all these functions flawlessly.

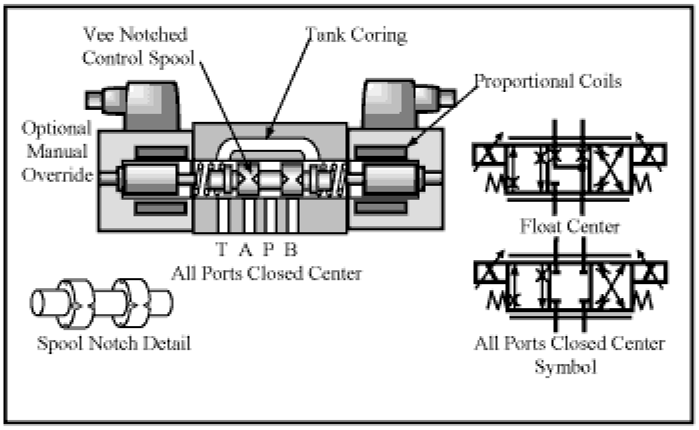
Rolling mills turn out sheet consistently to a tolerance of ±0.0005 in. at sheet speeds of 2000 to 5000 feet per minute. Hydraulic cylinders controlled by servovalves maintain the proper force and position the rolls precisely from feedback signals sent by sensors that measure metal thickness, cylinder force, and position. Airline pilots train in simulators moved by hydraulic cylinders so precisely that the pilots get the feel of landing gear raising and locking in position. Even entertainment rides use servovalves to make passengers think they are in 20-ft waves when they are actually in an enclosed articulated room in a shopping mall.

For less precise movement, there are proportional valves that mimic the output of servovalves but respond more slowly. They are less expensive than servovalves and more contamination tolerant, so they have replaced cam valves and other mechanical devices used to get smooth motions.

Hydraulic proportional directional control valves

The symbol and cutaway view in ***Figure 12-1*** *represent a direct-acting proportional valve that handles flows as high as 10 to 30 gpm in D03- or D05-size valve interfaces. Proportional valves use the same interface standards as NFPA and ISO directional valves so they can be installed in a circuit without having to change the piping.*

Fig. 12-1. Direct-acting proportional valve



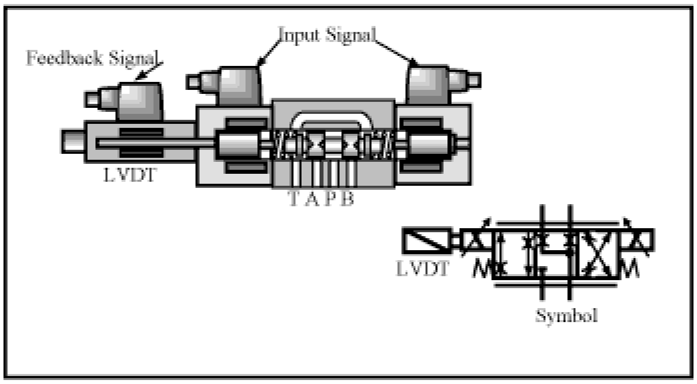
Physically, proportional valves appear the same as their on/off solenoid counterparts. The big difference is in the way their solenoid coils perform. Proportional coils operate on DC current and produce varying force with varying voltage. The symbol shows the solenoid slash in the operator box with a sloping arrow through the slash. This indicates the solenoid has variable force that moves the spool more or less as voltage increases and falls. The other indication on the symbol that shows the spool is infinitely variable is the parallel lines down both sides of the boxes. Proportional valves operate similarly to manual valves, but they use electronics instead of hand power.

To eliminate flow lag from spool overlap, most manufacturers cut vee notches or use some similar method that allows some flow to pass as soon as the spool moves. Vee notches also give smooth flow buildup until the spool moves through the land overlap.

*Proportional valves only have two center configurations (as shown by the symbols in **Figure 12-1**). This means that pressure-compensated pumps with accumulators normally power circuits with proportional valves. The circuits are pressurized at all times to produce fast response from an actuator when motion is called for. (A pressure-compensated circuit also wastes the least amount of energy when throttling flow.)*

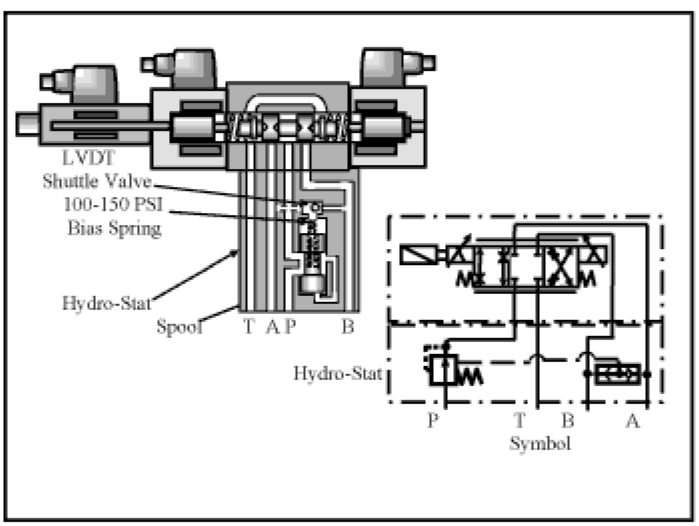
*The valve in **Figure 12-1** depends on a certain voltage to move the spool a certain distance to pass a certain flow. This works reasonably well, but is not accurate over a broad range of pressures, flows, and temperatures. Most valves of this design are used to smoothly accelerate and/or decelerate an actuator. The spool is electronically controlled to shift over a period of time to increase flow at a controlled rate. Spool-shift speed can be controlled electronically as it opens and closes to give smooth acceleration and deceleration. Spool-shift distance can also be limited electronically to set a maximum speed when required.*

Fig. 12-2. Direct-acting proportional directional control valve with spool-position feedback transducer



To give better spool control, a linear variable-displacement transducer (LVDT) is added to the basic valve. The cutaway view and symbol in ***Figure 12-2*** *represent a direct-acting valve with an LVDT. Feedback from the LVDT tells the electronic controller the spool's position and makes sure it goes to the same place when it receives the same signal. With this arrangement, the spool always shifts to an exact location and opens the same size orifice so it can pass the same flow when pressure drop and viscosity stay the same. Control of flow is more accurate with an LVDT but pressure drop does not stay constant and viscosity often changes throughout the day so speed variations are still apparent. Such flow variations caused by system pressure fluctuation can almost be eliminated by the addition of a hydrostat module in port P as shown in* ***Figure 12-3****.*

Fig. 12-3. Direct-acting proportional directional control valve with LVDT and pressure-compensating hydrostat module

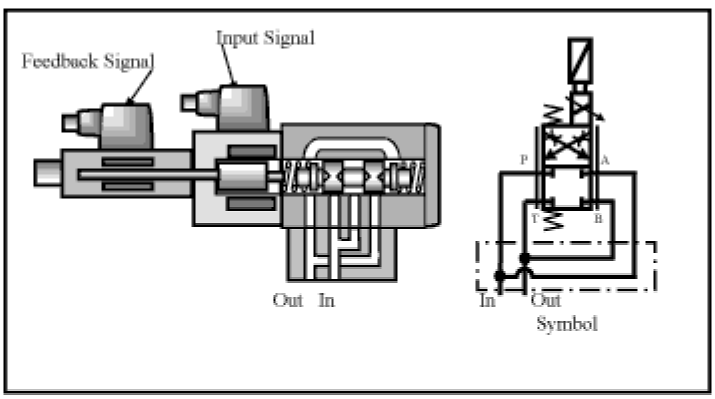


A hydrostat is simply a pressure-reducing valve set to hold downstream pressure in a 100- to 150-psi range. However, a hydrostat has a pilot line from a shuttle valve that reads downstream *pressure at ports A and B, then feeds it back to the bias-spring end of the spool that controls the 100 to 150 psi. The function of a hydrostat is to maintain a constant pressure drop across the spool orifice so flow stays constant regardless of changes in system pressure. (For a thorough understanding of how pressure compensation works, see the pressure-compensated flow control valve section in Chapter 13.)*

With the addition of a hydrostat, actuator speed is controlled as accurately as possible without a closed-loop electronic circuit that reads speed and modifies spool position. Closed-loop electronic circuits are used with proportional valve systems, but they only give nominal control. When accurate control is required, use servovalves with closed-loop electronics.

*The symbol and cutaway view in **Figure 12-4** is for a proportional valve that only controls flow. Such valves are commonly called throttle valves because they are not pressure compensated unless a hydrostat module is added.*

Fig. 12-4. Direct-acting proportional throttle valve with spool-position feedback transducer

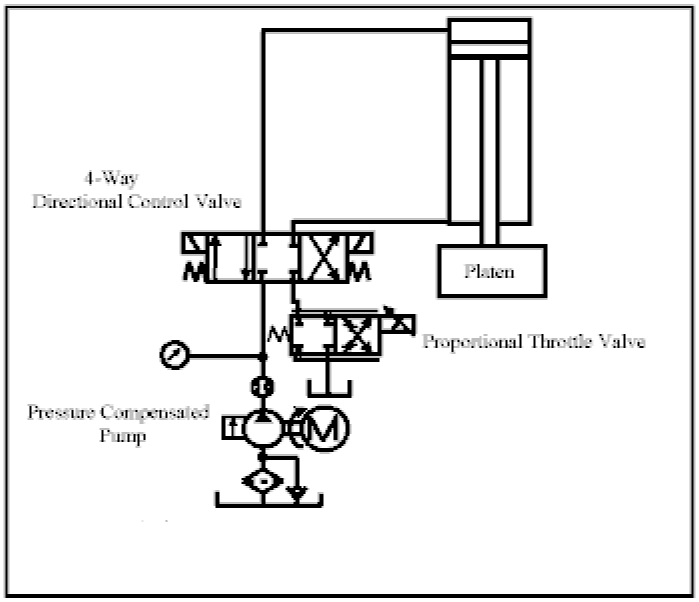


Basic operation is identical to the proportional control valves just discussed. The only difference is they have a single solenoid and may be piped with dual flow (as shown). Dual-flow piping allows a given size valve to pass twice the volume at the same pressure drop. It can be used with any 4-way directional control valve with one precaution: the valve must be capable of handling maximum system pressure in its tank port. Many wet-armature valves will not operate at full rated pressure at their tank port. Check the supplier's catalog to see what maximum tank line pressure is allowed. Air-gap solenoid valves and solenoid pilot-operated valves with external drains normally allow full rated pressure in the tank port.

*The circuit in **Figure 12-5** shows a possible use for a proportional throttle valve. The vertical-down acting cylinder with a platen needs speed, acceleration, and deceleration control. This could be done with a 4-way proportional valve, but the circuit uses an inline or screw-in cartridge valve that is not directly replaceable. Adding a proportional throttle valve to the tank line of the present 4-way circuit can give the required control without extensive piping changes.*

*The circuit is shown using a single flow path for low volume. A dual flow path setup (like the one in **Figure 12-4**) would allow as much as twice the flow. As stated earlier, make sure the 4-way directional control valve can accept tank-line backpressure without damaging it or causing a malfunction.*

Fig. 12-5. Typical circuit using a direct-acting proportional throttle valve with spool-position feedback transducers



*If the cylinder had a resistive load, the proportional throttle valve could be placed in the pump line of the 4-way as a meter-in flow-control circuit. A meter-in circuit would not damage the directional control valve or cause it to malfunction. The circuit in **Figure 12-5** could have a counterbalance valve in the rod-end line to make it resistive. (See Chapter 14 for counterbalance valve operation and applications.)*

The platen would start to move at a controlled rate when the 4-way valve shifts and the proportional throttle valve is signaled to open slowly. The shift time for the throttle valve determines the acceleration time, while shift travel distance determines maximum cylinder speed. Cylinder speed would be infinitely variable to match any production need.

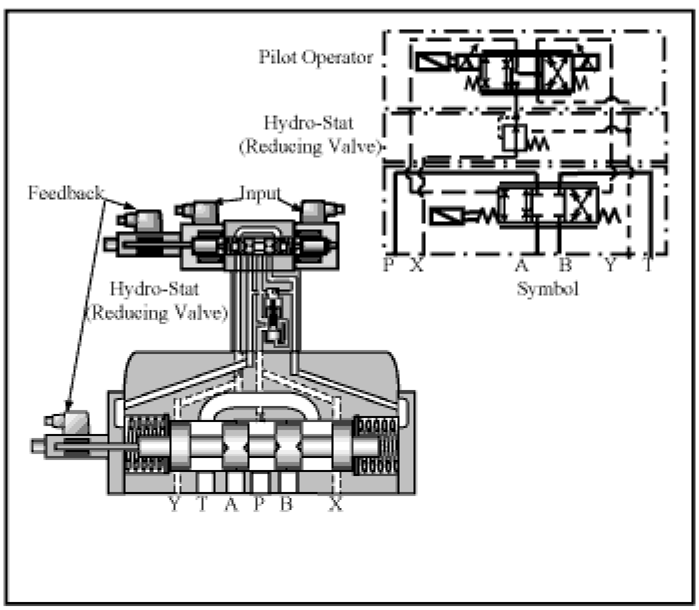
Near the end of the stroke, a slowdown limit switch would signal the proportional throttle valve to start shifting back to its closed position. The proportional throttle valve would close at a controlled rate and flow from the cylinder would be retarded smoothly. When the cylinder slows sufficiently, it contacts the end-of-stroke limit switch and the 4-way directional control valve shifts to center to stop it.

*The circuit in **Figure 12-5** would only hold position if some retract signal was applied when stopped. This is due to internal leakage by the spool of the 4-way directional control valve and* *the proportional throttle valve. Another option would be to use a float-center directional control valve and a counterbalance valve.*

Proportional valves for flows higher than 25-to-30 gpm use solenoid pilot-operation similar to conventional directional control valves. A small pilot-operated valve receives a signal and then sends hydraulic oil to proportionally move a larger control spool that controls actuator movement.

The cutaway view and symbol in ***Figure 12-6*** *depict a typical solenoid-pilot valve arrangement. A reducing valve module, between the pilot operator and the pilot-operated valve, keeps maximum pilot pressure below 200 psi. A proportional Input to one of the coils on the pilot operator directs flow to the spool of the pilot-operated valve and shifts it against a spring. As pressure against the spool increases, it shifts farther and sends more flow to the actuator. Feedback signals from both spools tell the electronic controls that the command has been carried out. A vee-notched spool allows flow to increase at a smooth rate so actuator speed is consistent throughout the speed range.*

Fig. 12-6. Solenoid pilot-operated proportional directional control valve with spool-position feedback transducers



Ports for internal or external pilot X or drain Y provide options for these control lines to meet a particular requirement.

The complete symbol is shown in ***Figure 12-6****. (For the simplified symbol, see Chapter 4.)*

Flows up to 200 gpm are common for a D10-size proportional valve. For higher flows, use slip-in cartridge valves (discussed in Chapter 11) with proportional operators.

Hydraulic servo directional control valves

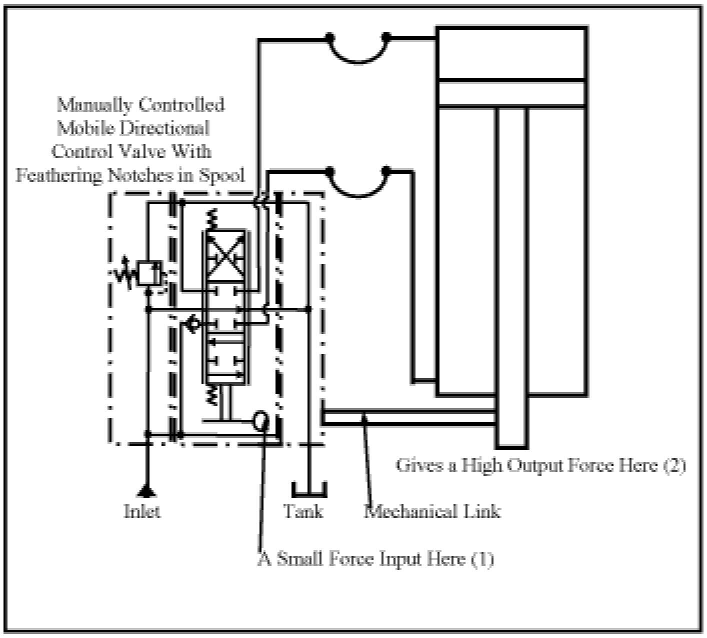
Proportional control valves are infinitely variable but they are neither highly responsive nor capable of handling minute flow changes rapidly and accurately. On the other hand, servovalves easily meet both of these requirements . . . but at a cost. They are more expensive than proportional valves, they require super-clean fluid, and they need extra electronics to exploit their full capabilities.

The three common servovalve types are flapper, jet pipe, and mechanical. Each design has advantages as far as operation accuracy, leakage, contamination tolerance, and price. They range in flow capacity from less than 1 gpm to more than 1000 gpm. Most manufacturers make valves that operate at 3000 psi, but some offer valves at 5000 psi.

The main difference between proportional and servovalve circuit design is that servo systems have a method of feedback that assures that the actuator is doing what the controller tells it to do. A super-simple form of servo control would be a backhoe operator moving manual valves to cause a bucket to move toward him at a given rate. Feedback from the operator's eyes would tell his hands when and how far to move the levers to give more or less flow to maintain the action he wants. Other familiar mechanical feedback examples are hydraulic driven power steering and hydraulic power brakes on a vehicle.

*The circuit in **Figure 12-7** is an example of a working circuit with mechanical feedback that controls a hydraulic press. The operator needs to have a feel for the motion of a platen as it cuts through some tubing. Originally this was done with an arbor press, but it was hard for the operator to keep up with production due to the physical exertion. Now all the operator has to do physically is overcome the spring force of the manually controlled mobile directional valve to make the platen move. As the platen moves, the directional valve body also moves, so the operator has to keep moving the lever to advance or retract the platen. Notice the mechanical link between the valve body and the cylinder rod that moves the valve body at the same rate the cylinder rod moves. The operator now has a hydraulic force multiplier that gives some feel to what he is doing.*

Fig. 12-7. Simple mechanical servo system for force multiplication



The reason for using a mobile-type valve is because those valves have less spool overlap and the spool has notches cut in it. The notches pass a small flow almost immediately when the spool moves. That flow increases in proportion to spool movement.

Most industrial applications use feedback from electronic linear, rotary, or force transducers. A transducer is a device that produces an electrical signal in direct relation to a position, force, or speed.

Linear potentiometers work for short strokes (12 in. or less). Longer strokes require a device such as a Temposonics transducer. In either case, these devices feed a precise position or speed indication back to an electronic controller.

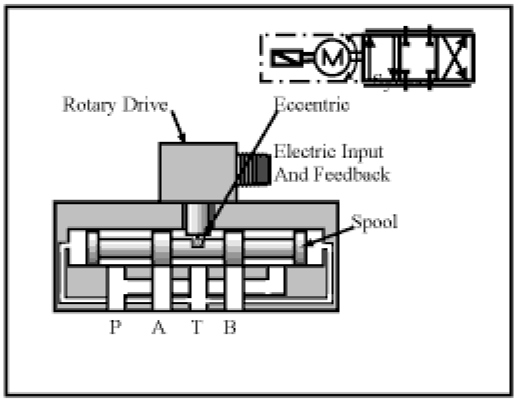
For rotary motion, an encoder or similar device that produces multiple pulses per revolution sends a signal about rpm or angle of rotation to the controller.

When information about force is required, a load cell sends the data to the controller.

With these very accurate feedback devices and a fast-response servovalve, an actuator's position, speed, and/or force can be repeatedly established within an extremely close range. Electronics provides the accuracy while hydraulics provides the force via a super-responsive servovalve.

*The cutaway view and symbol in **Figure 12-8** show a less-responsive but more contamination-tolerant servovalve. There are other mechanical ways of driving the spool. The valve in **Figure 12-9** uses a rotary drive and an eccentric to move the spool left or right to an infinite number of positions. Because the drive is quite strong and there are no orifices to clog, this valve can operate with fluid that meets ISO Code 4406 20/16/13.*

Fig. 12-8. Rotary-drive servovalve



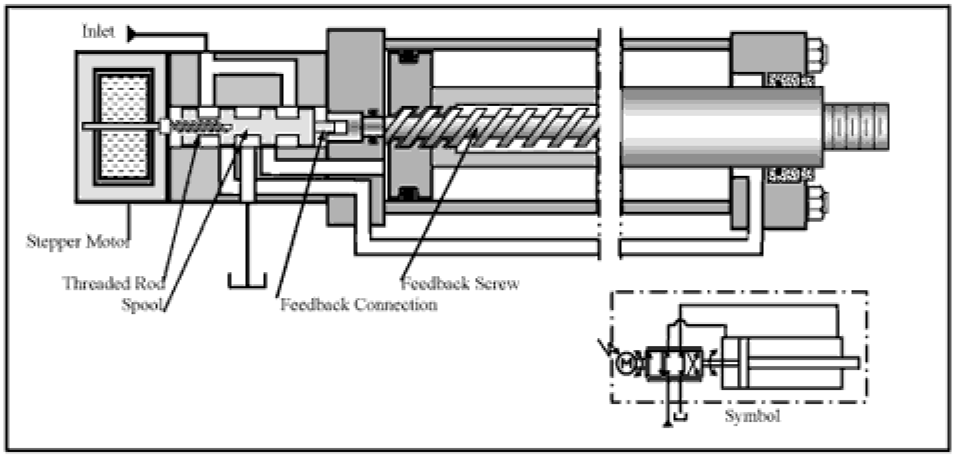
Notice the difference in design between spools in proportional valves and servovalves. Most proportional valves use spools with overlap and some sort of notches that pass flow while moving out of overlap. A servovalve has no overlap or underlap of the spool lands to the body lands. (One manufacturer calls it "Critical lap" because all points blocking fluid cannot move without passing flow.) This spool design makes the valve very responsive (as well as very expensive and prone to above average bypass). Servovalve spools and bodies always come in matched sets because of their close fit and four points of land-to-land match.

*The rotary-drive eccentric valve pictured in **Figure 12-8** has fast, controllable spool movement from a rotary drive that incorporates a feedback loop. When the drive receives a signal to move the spool to pass a certain flow, a position feedback output sends a signal back to the controller when the motion is complete. There is still feedback from the actuator that what was commanded is happening, so spool position can be changed via the electronic feedback and controller as necessary.*

Several factors determine when a given input will not produce the desired actuator output. The main factor is actuator load. As load changes, input force must change -- by allowing more or less fluid into the circuit. Fluid viscosity also has an effect, so the flow path must be reduced as viscosity lowers and enlarged as viscosity rises. Then there is system pressure. As pressure fluctuates, flow across the spool orifice changes. The higher the pressure drop, the greater the flow. Because a servovalve circuit has feedback from the actuator, it can adjust flow or pressure to match system changes continuously.

***Figure 12-9** shows another mechanically driven servovalve. This setup works for cylinders and hydraulic motors, but must be directly attached to the actuator (as shown) or driven by it with a toothed belt and pulleys. This is a very contamination-tolerant valve arrangement because the stepper motor is quite strong and there are no small orifices to clog. The spool has no overlap or underlap so any movement immediately initiates fluid flow to and from the cylinder. The piston and rod cannot rotate so the feedback screw turns as the piston extends or retracts.*

Fig. 12-9. Stepper-motor-driven servovalve with mechanical feedback

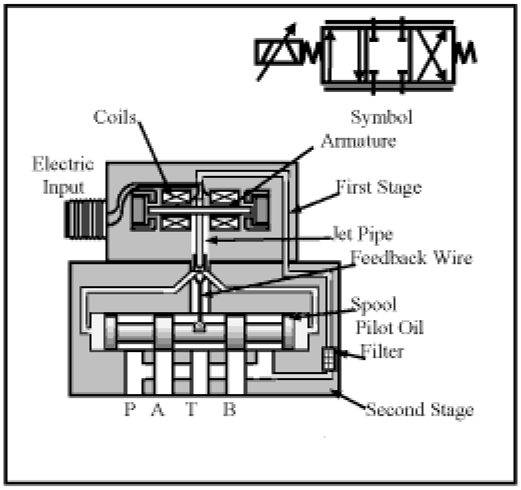


As the stepper motor turns, a threaded rod inside the threaded spool moves the spool to direct fluid to extend or retract the piston. When the cylinder piston moves, the feedback screw turns the spool back on the threaded rod to counteract the stepper motor shift. When the stepper motor turns, the piston moves at a speed proportional to the stepper motor's rpm. When the stepper motor stops, the piston catches up and stops also. Manufacturers claim a tolerance of ±0.001 in. repeatability. If an external force tries to push or pull the piston out of place, the feedback screw shifts the spool and fluid starts resisting movement.

From the foregoing explanation, it is easy to see how this valve – when attached to a hydraulic motor -- would give an exact number of turns and repeatedly cause the motor to stop at exactly the same place. This will happens even if the hydraulic motor has internal leakage. The only time the actuator gets out of place is when it cannot overcome the load and stalls. The stepper motor continues to receive pulses, but it also stalls when the spool shifts all the way. The stepper motor received a signal that should have placed the actuator at a certain distance but the actuator did not get there because of insufficient force. When the valve reverses, it starts its motion from the wrong point and will overshoot home position as it returns. A limit switch at the home position can alert the controller that there is a problem when the actuator overshoots -- and prevent the machine from producing scrap.

*The jet-pipe servovalve pictured in **Figure 12-10** also tolerates contamination due to a control orifice that is large enough to pass large particles. Pilot oil is tapped off the system fluid inlet, sent through a coarse filter, and on to the jet pipe that terminates in an orifice. The orifice outlet is centered over the inlet of two passages that terminate at each end of a critical-lap spool. Flow into these passages puts equal pressure on both ends of the spool as the feedback wire holds it centered. Current signals to the coils cause the armature to rotate and shift more of the output of the jet pipe to one passageway than the other. Pressure increases on one end of the spool and decreases on the other end. As the spool shifts, it starts to pass flow to the actuator at a rate set by the input electrical signal.*

Fig. 12-10. Jet-pipe servovalve



When the jet pipe shifts to the left, the spool moves to the right. At the same time, the feedback wire also moves to the right, pulling the jet pipe nozzle back to center and stopping spool movement. A given input to the coils electromechanically shifts the armature that moves the jet pipe. This moves the spool hydraulically and forces the jet pipe back to center mechanically through the feedback wire.

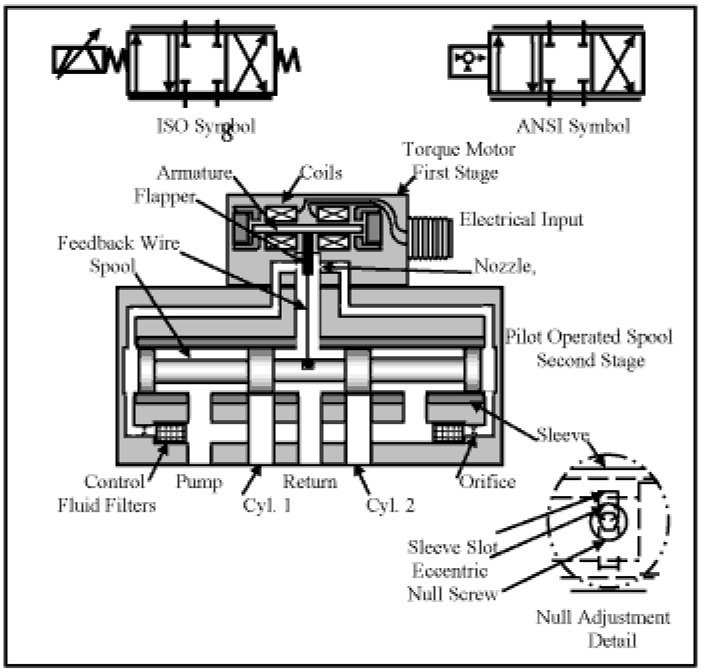
A measured electrical input to a servovalve produces a fixed flow output, similar to a proportional valve. This control alone does not give much better control than a proportional circuit even though the valve is more responsive. There is still no compensation for viscosity or pressure changes that can cause the actuator's speed to fluctuate. To overcome this problem, some sort of electronic feedback from the actuator is necessary. The feedback signal through an electronic circuit board modifies the signal to the servovalve to make the actuator perform as planned. Actually, the electronics do the work as long as the valve can respond quickly enough to keep everything working at the correct rate. This means the spool must be free enough to move easily without excessive bypass. Anytime the spool moves, it should pass flow to and from the actuator.

The valve in ***Figure 12-10*** *is considered a 2-stage valve. The first stage is electronic and receives an electronic input signal, while the second stage is fluid powered by a hydraulic signal.*

The jet-pipe servovalve depends on clean oil for long trouble-free operation -- not as clean as the requirement for the flapper-valve design discussed next, but clean enough to prevent the jet-pipe nozzle from clogging. It is obvious that once nozzle flow is retarded enough or stops, the valve loses all ability to control flow to the actuator.

The most responsive and accurate servovalve design is the flapper valve, shown in ***Figure 12-11****. This design is the least tolerant of contamination because it depends on very small orifices for fast response with minimal wasted energy. It is called a flapper valve because the element that* *holds equal pressure on both ends of the spool at rest reminds one of a flapping device. It is a 2-stage valve with an electronically controlled torque motor as the first stage and a pilot-operated spool as the second stage. As in all servovalves, the spool has no overlap or underlap that would make it sluggish or bypass a lot of fluid unnecessarily.*

Fig. 12-11. Flapper-design servovalve



Fluid from the pump inlet is tapped off through rather-coarse filter elements, passes through orifices past both ends of the spool, goes on to nozzles, and out to the return line. The orifice diameters are slightly larger than the nozzle diameters, so there is a pressure buildup at both ends of the spool. A feedback wire attached to the flapper terminates in a ball end that sits in a very close-fit slot in the spool. A sleeve around the spool can be moved left or right by a null adjustment to align the spool and body lands perfectly when the valve is first installed. (Usually null adjustment is only required at startup of the valve.)

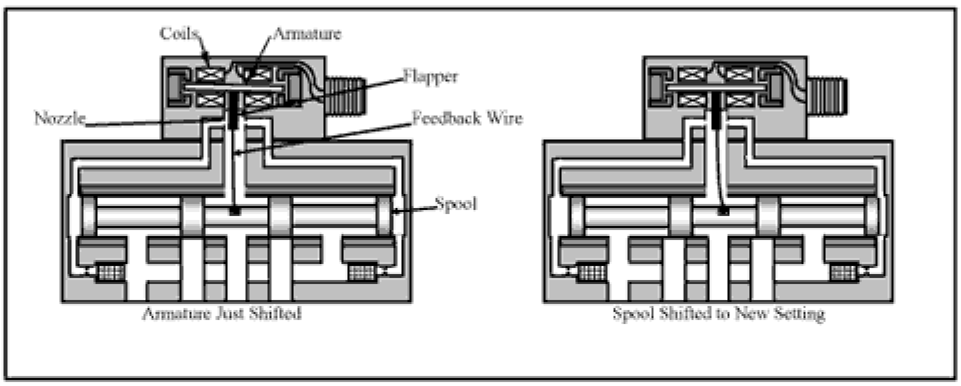
The null adjustment usually is a hexagonal wrench fitting attached to an eccentric pin located in the sleeve slot. With the null adjustment centered, turning it one round moves the sleeve from center to full right, back to center to full left, and back to center. If the valve cannot be nulled within one rotation of the null adjustment, replace it and send it in for repair. This usually indicates a clogged orifice or nozzle controlling one end of the spool.

Unplug the electrical supply to the valve before setting null. Start the pump and watch for actuator movement. If the actuator moves, loosen the null lock screw and carefully turn it. Observe whether the actuator slows or picks up speed. A nulled valve stops actuator movement because the forces on both sides are equal. High-flow 3-stage valves cannot be nulled to the *point of stopping an actuator due to the piloted spool slipping by the stop-flow position as the pilot operator is adjusted. When null is set, lock the null screw and reattach the electrical plug.*

Turning the null screw with the electric plug detached is one way of moving an actuator manually. This might be done to prove the valve is working properly and the problem is electrical.

When the torque-motor coils receive a current signal, the armature rotates clockwise or counter-clockwise and pushes the flapper closer to one nozzle and farther away from the opposite one. This allows pressure to increase at one end of the spool and decrease at the other. The spool then starts to move away from the higher pressure. If the armature turns clockwise, pressure builds on the left end of the spool and it moves to the right, as shown in the left cutaway view of ***Figure 12-12****.*

Fig. 12-12. Flapper-type servovalve shifting from an electrical input signal

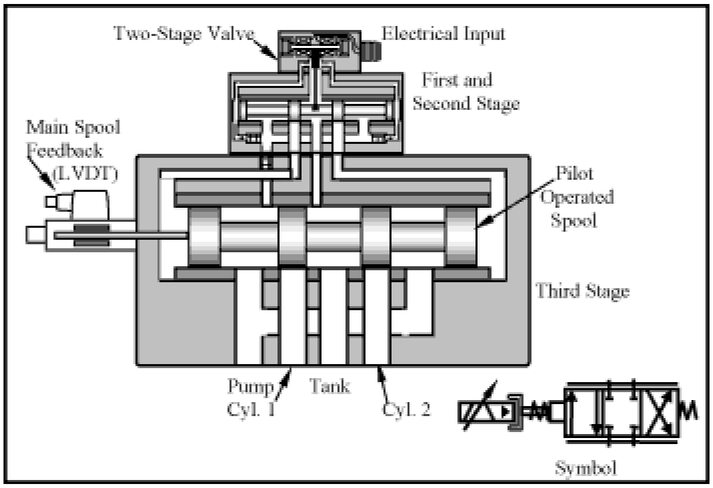


As the spool moves to the right, it also drives the feedback wire to the right. The feedback wire is strong enough to overcome armature force and pull the flapper back to center. After the flapper centers, pressure is equal on both ends of the spool and it stops. More current to the coils causes more rotation and additional spool shift until the feedback wire again centers the flapper.

From the foregoing explanation, it is obvious why this valve needs clean oil. If an orifice or nozzle clogs, the spool shifts all the way to one end and the actuator moves until it runs into a resistance it can't overcome. Also, the spool must start shifting at a very low pressure drop across it to keep response high. Contaminated fluid can cause sticking and require high differential shifting pressure that makes spool movement erratic.

For flows above 60 to 80 gpm, a 3-stage servovalve is required. It consists of a small 2-stage pilot operating a large pilot-operated spool, as depicted in ***Figure 12-13****. The 2-stage valve operates as just explained, but its output goes to move a pilot-operated spool in the third stage to a precise position to control high flow to large actuators.*

Fig. 12-13. Three-stage flapper-type servovalve



An LVDT signals the electronic control circuit that the pilot-operated spool is where it was signaled to go. After receiving that position signal, the 2-stage valve shifts to no flow or whatever flow it takes to keep the pilot-operated spool in place.

A 3-stage valve also depends on feedback signals from the actuator to modify the input signal when the action is not in compliance with the command. This makes 3-stage valves very accurate controllers of large cylinders.

Pneumatic proportional and servo directional control valves

Since the late '80s, several companies have been controlling air cylinders with open- and closed-loop proportional or servovalve circuits. The difference between the air valves in these circuits is how fast they respond. Most proportional valves have a sealed spool that controls direction and flow so the valves tend to hang up and jump. Pneumatic servovalves often have spools with metal-to-metal fits that float on bypass air.

*The valve shown in **Figure 12-8** is sold as a proportional or servo directional control valve for hydraulic or air circuits. Controlling the amount of air and which direction it goes is not a problem but the compressibility of air creates some giant hurdles to overcome.*

Proportional valves usually control only acceleration, deceleration and/or speed because these circuits do not include feedback transducers. It is very easy to get smooth acceleration and deceleration with high speed in between without other controls or shock absorbers to stop the load mechanically.

Adding feedback transducers to a proportional air circuit can provide servo-like control for light loads -- such as those found in pick-and-place applications. However, a proportional valve is usually not responsive enough for exacting part placement or speed control. For very accurate *control, a servovalve with feedback transducers can give close-tolerance positioning (with light loads), repeatable velocity control, and very accurate holding force.*

Pneumatic proportional and servovalves are not a replacement for electromechanical or servo hydraulics, but they have price advantages over both systems. When the loads are light and cost is a factor, they are worth a look.

General information for hydraulic infinitely variable valves

- The symbol for proportional and servovalves shows a 4-way, 3-position function and the valve can move to each of the positions. However, the parallel lines along the sides of the symbol indicate the valve does not have to shift all the way all at once. These valves can shift into straight or crossed arrows in any proportion from 0 to 100%. They are infinitely variable and can pass any flow desired.
- servovalves are always 3-position, all-ports-blocked center condition, as shown by the symbols in **Figures 12-8** through 12-11.
- always size proportional or servovalves for high pressure drop. Proportional valves should have 200- to 500-psi pressure drop at full flow. Most servovalve manufacturers rate their valves at 1000-psi pressure drop at full flow. This means the valve may look physically small for a given flow in relation to conventional valves. It also means most servo and proportional valve circuits require a heat exchanger to deal with excess wasted energy.
- always mount the valve as close as possible to the actuator ports. Any piping between the valve and the actuator holds extra fluid that can make the system softer and less responsive. This is especially important on air-powered circuits.
- never use hose between the valve and the actuator. If isolation is necessary, mount the valve on the moving part and use flexible lines for supply and return.
- use in-line pressure filters at the supply to each servovalve or bank of valves to protect the valve from contamination in the pump and piping.

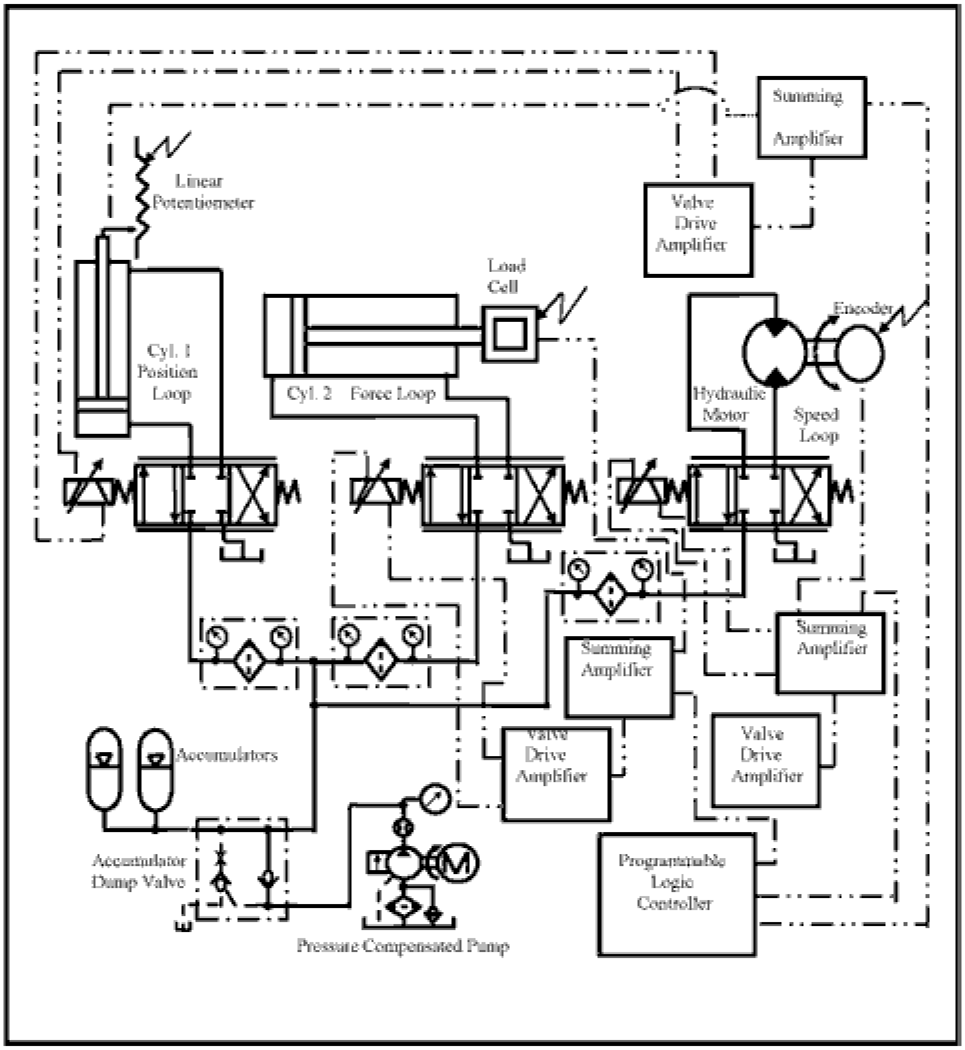
Specific Information for pneumatic infinitely variable valves

- use air at the highest pressure possible that does not exceed component or plumbing limits. This is usually as high as 250 psi.
- size the valve to flow just enough to produce the maximum desired actuator speed. Oversize valves produce erratic control because a small spool movement gives more flow than required.
- use an actuator with the largest area practicable so the load moves with a low pressure difference. Note: the larger the cylinder, the more air it consumes so operating cost escalates.
- pneumatic servo circuits do not work well when outside forces push against the actuator. The actuator tries to resist, but force buildup is slow in comparison to electromechanical or electrohydraulic systems.

Typical servo circuits

Figure 12-14 *shows schematic drawings of three typical servo circuits. In the figure, each type circuit controls a different actuator, but any actuator could have more than one type of control.*

Fig. 12-14. Servovalve and closed-loop electronic circuit for accurate position, force, and speed control



The typical power unit for a servo system is a pressure-compensated pump with an accumulator or accumulators. A servovalve must always have a ready supply of fluid because no matter how fast it reacts, without an immediate supply of fluid the system will be sluggish. The pressure-compensated pump may be at full pressure when no actuators are moving, but its flow is zero. Adding accumulators assures that there is no wait for the pump to come on stroke before the actuator receives flow. (A full discussion of how accumulators work and how they are applied is in Chapter 16.)

Again, pressure filters in the lines to the servovalves make sure they receive clean oil. One filter could be sufficient for multiple valves when the valves are close to each other. The reason for pressure filters in the valve lines is the pump constantly produces contamination particles that will shut down flapper-type servovalves.

Cylinder 1 is in a position loop. It can be placed at a precise location repeatedly within ±0.0005 in. A programmable logic controller (PLC) sends a signal to the summing amplifier's control card. The signal passes on to the valve-driver card and then to the valve coils. This signal shifts the servovalve to start cylinder movement at a set rate. The linear potentiometer sends position feedback to the summing amplifier and modifies valve position to find and maintain a certain position. Often, position control is paired with speed control to accelerate the actuator to a certain speed, then decelerate and stop it at the desired position.

Cylinder 2 is in a force loop. A certain size cylinder operating at a given pressure produces a given force. This force can be calculated by multiplying area times pressure, but the result is not exact. Friction from seals and between external machine members can reduce this force by a few pounds or more on an operating machine. When an exact force calculation is required, a servovalve-controlled cylinder that has a load cell for feedback can keep forces within 1/2% with ease. The summing amplifier sends a signal from the PLC and feedback from the load cell modifies the valve position to exactly match the input signal to generate the desired force.

The hydraulic motor is in a speed loop that maintains the motor's rpm when the fluid viscosity, pressure, or load changes. A rotary device called an encoder constantly sends rpm information back to the summing amplifier to open or close the servovalve as needed. Just as a cylinder does, a hydraulic motor will slow when the load increases, when fluid gets thinner due to temperature increases, or when system pressure fluctuates as other actuators move. If the encoder sends a reduced-rpm signal back, the servovalve opens to let more fluid in. If the hydraulic motor tries to speed up, the servovalve closes enough to maintain the set speed.

Other infinitely variable valve applications

Proportional and servovalves can be applied to variable-volume pump controls to accurately set flow from the pump in relation to feedback from an actuator. Most pressure-compensated pumps available today have this feature as an option.

An example might be a closed-loop hydrostatic pump and motor that must stay at a constant speed. Feedback from an encoder signals the pump to increase or decrease flow as the motor overspeeds or underspeeds.

Proportional coils make excellent infinitely variable flow controls that can be controlled electrically. They can act as simple throttle valves or perform as full-blown pressure-compensated types that adjust to pressure fluctuations.

Common circuits for proportional flow controls include feed-speed changes to deal with load fluctuations; hydraulic motor rpm changes to control product backup; or maintaining constant speed as loads vary. (Pressure-compensated proportional flow control valves are covered in Chapter 13.)

Proportional coils also make excellent infinitely variable pressure-control valves. Pressure settings from remote locations or a PLC at anytime from minimum to maximum almost instantaneously. Most of these pressure controls are for relief and reducing functions but could be used on any of the pressure controls. (Proportional pressure-relief valves were covered in Chapter 9.)

Proportional Control Valve Circuits

The spool in a standard 2-position-solenoid-operated valve shifts all the way to its new position at high speed. (Hence the nickname: bang-bang solenoid.) This rapid, full shift can cause an actuator to jump or lunge on start up and produce excessive shock when stopping. Pressure spikes and shock are noisy, may cause machine damage, and can adversely affect piping, causing leaks.

A soft-shift solenoid with hydraulically dampened spool movement slows the rate of shift and reduces shock in some applications. However, many machines need a variable shifting rate to match changing power and work requirements. Soft-shift solenoids with variable flow controls offer more range and give better control for some circuits.

Other options include valves with specially designed flow controls and spool-stroke adjusters set for a specific machine function. This type of variable valve works on some machines, but requires many precision adjustments to attain the wanted actuator control.

Variable-volume, bi-directional pumps in closed-loop circuits give very smooth action, but are limited to operating a single actuator. For extremely accurate control, a servovalve with actuator feedback is the ultimate motion controller. In between a servo circuit and the other controls mentioned above are proportional valves.

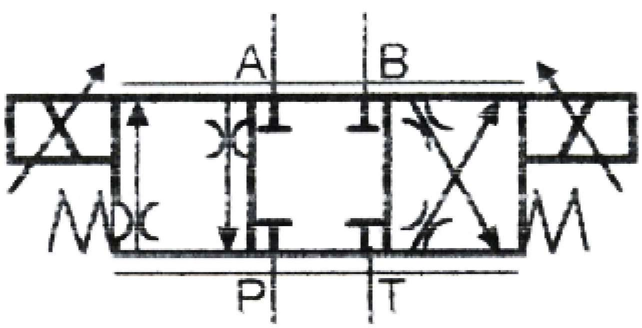


Figure 14-1. Direct-solenoid-operated proportional valve.

Proportional valves are well suited for circuits that need to vary either flow or pressure to reduce lunge and shock. The solenoids on these valves shift the spool more or less, According to the voltage applied to proportional solenoids, they can change the speed at which the spool shifts or the distance that it travels. Because the spool in a proportional valve does not shift all the way, all at once, the valves can control the acceleration and deceleration of an actuator. Usually, varying shifting time of the spool controls acceleration and deceleration. Varying voltage to the coil limits spool travel to control the maximum speed of an actuator. A computer, *a PC, a programmable logic controller, or even a simple rheostat can produce the variable electric signal.*

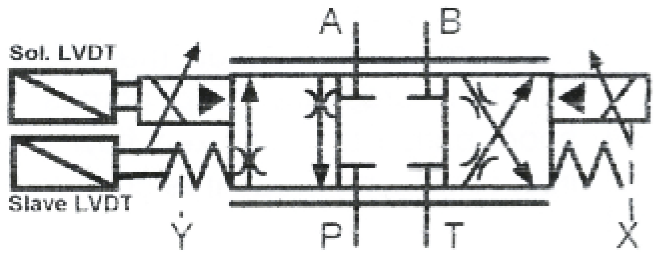


Figure 14-2. Simplified symbol for solenoid pilot-operated proportional valve with LVDT.

If flow is low (less than 20 to 25 gpm), use a direct solenoid-operated proportional valve, such as shown in Figure 14-1. Direct-operated valves are smaller and less expensive than solenoid-pilot valves. However, solenoid-pilot proportional valves can handle higher flows — some in excess of 200 gpm. Figure 14-2 shows the simplified symbol for a solenoid pilot-operated proportional valve. Figure 14-3 shows the complete symbol for the same valve. The complete symbol includes details of the control and slave valves, the reducing valve in the pilot circuit, and the routing of the pilot lines.

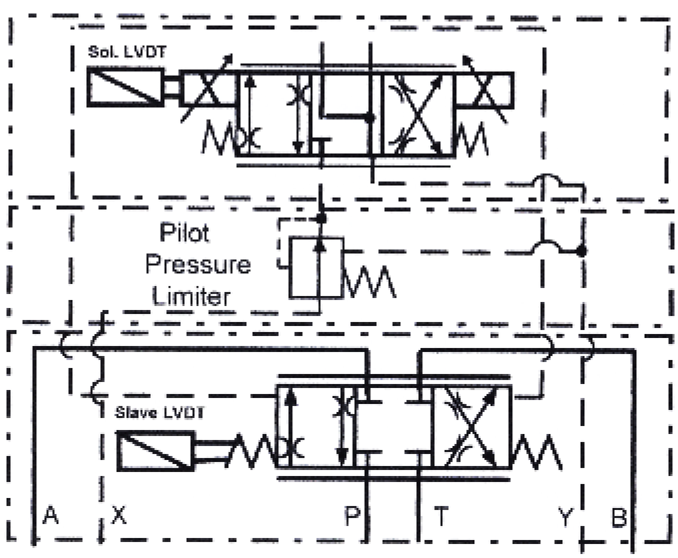


Figure 14-3. Complete symbol for solenoid pilot-operated proportional valve with LVDT.

A simple proportional valve depends on solenoid force working against a spring to position the spool. Because flow, pressure, temperature, and fluid cleanliness change constantly, a given input voltage may not always produce the same spool position. To resolve spool position accuracy, use a linear variable differential transformer (LVDT), such as shown in Figures 14-2 through 14-5. An LVDT electronically compares the input signal with spool position and modifies voltage to give the same spool position regardless of system changes. An LVDT adds cost to the valve and the electronics, but is usually necessary in all but simple acceleration/deceleration circuits.

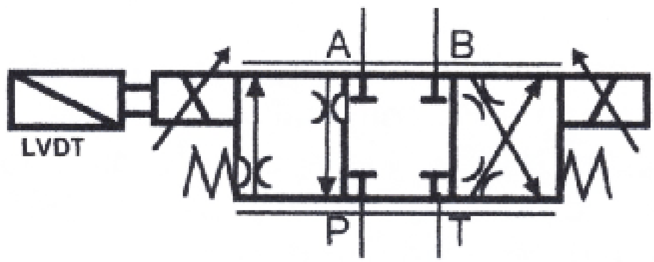


Figure 14-4. Direct solenoid-operated proportional valve with LVDT.

An LVDT does not control repeatability of flow through the valve because flow is a function of pressure drop and fluid viscosity as well as orifice size. Changes in pressure or fluid thickness will modify actuator speed. To reduce speed change, add a feedback signal from the actuator (similar to a servovalve circuit). Actuator feedback will help but is still not extremely accurate because most proportional valves do not respond quickly enough to overcome sudden system changes.

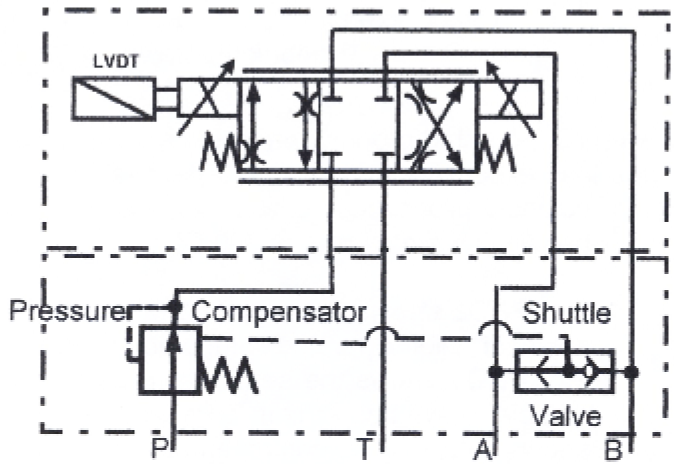


Figure 14-5. Direct solenoid-operated proportional valve with LVDT and pressure compensator.

In Figure 14-5, the pressure-compensating valve in the inlet line reduces flow fluctuations due to system pressure changes. The pressure compensator maintains a constant pressure drop across the spool orifice to keep flow constant when inlet or working pressures change. The pressure compensator is a reducing valve that has a fixed spring setting (say 150 psi). A shuttle valve provides pressure feedback from each cylinder port to the reducing valve's remote-control port. As pressure in a working port changes, it modifies reducing-valve pressure to maintain a constant 150-psi drop across the proportional valve's spool.

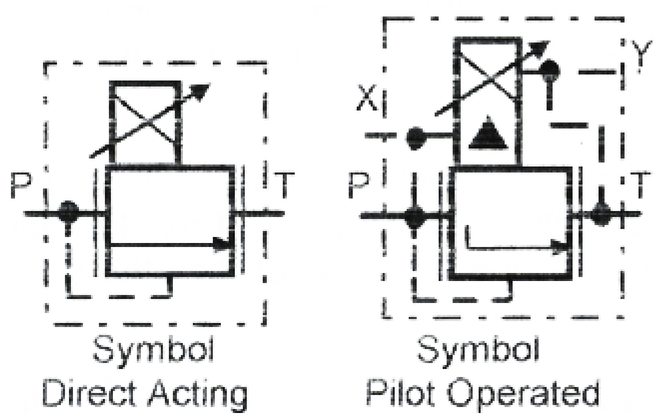


Figure 14-6. Proportional relief valves.

Using electrical signals to a proportional solenoid to vary the force against a poppet or orifice allows infinitely variable control of pressure. Figures 14-6 and 14-7 show symbols for infinitely *variable pressure-relief and reducing valves. Use a PC or PLC to produce the variable signal to change pressure any time the machine sequence requires it.*

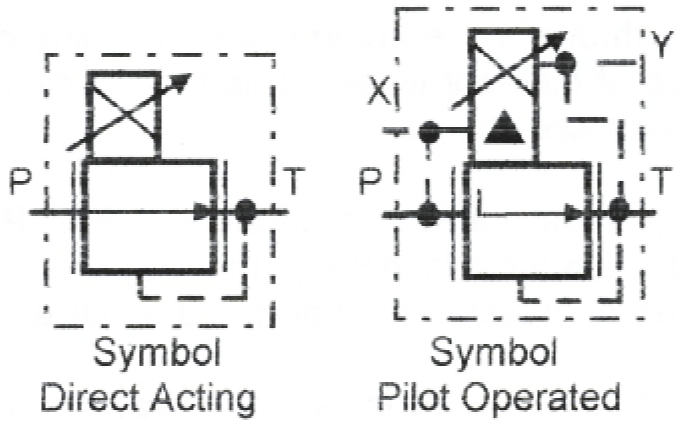


Figure 14-7. Proportional reducing valves.

Remote control of a pressure-compensated pump with a proportional pressure valve makes these pumps more versatile also.

Figure 14-8 depicts the symbol for a slip-in cartridge relief valve. When flows go above 150 to 200 gpm, use a slip-in cartridge relief valve with a direct-acting proportional relief pilot. These cartridge valves come in stand-alone bodies or as part of a special high-flow manifold.

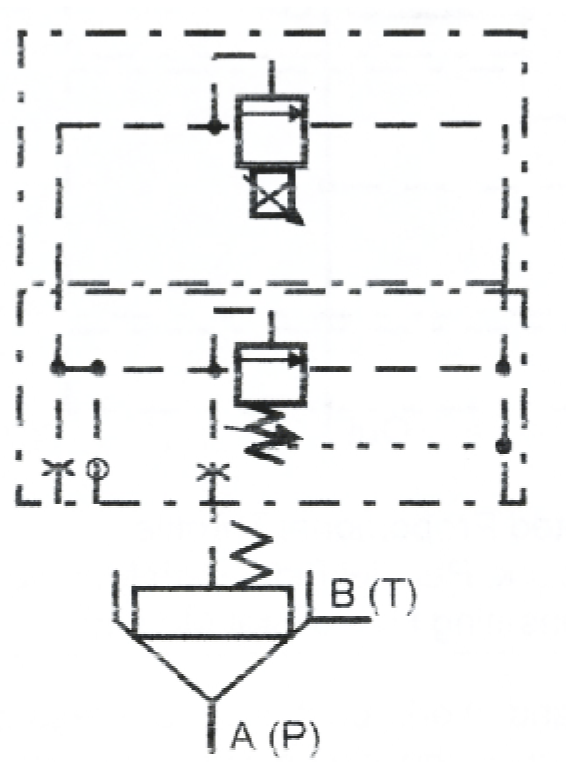


Figure 14-8. Slip-in cartridge relief valve – proportionally operated for infinitely variable pressure.

Proportional directional control valves are more tolerant of contamination and cost less than the servovalves that they often replace. When a circuit does not require extreme accuracy or flow repeatability, the savings in first cost, plus a less-expensive filtration requirement, make proportional valves a good choice.

One reason a servosystem is more accurate is the electronic feedback signal from the actuator. The feedback signal modifies the servovalve's spool position to put the actuator in an exact place, or produce the speed or force that the controller requires. A proportional valve may have feedback control, but the response time of the valve is too slow to get the precise control that a servovalve circuit provides.

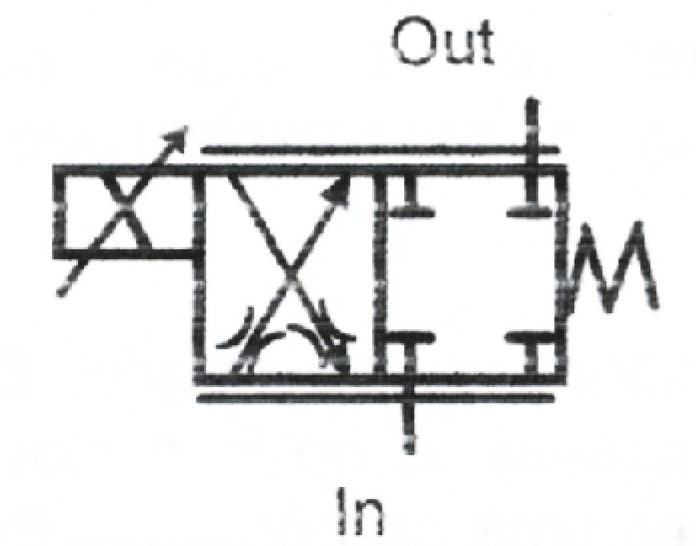


Figure 14-9. Direct solenoid-operated proportional throttle valve without LVDT.

Figure 14-9 depicts a proportional valve used for a throttle function. This valve is an infinitely variable, electrically controlled flow control. As coil voltage increases, the spool shifts farther to increase flow. The symbol in Figure 14-9 shows the valve piped for a single flow path. Dual flow paths shown in Figure 14-10 give twice the flow at the same pressure drop in either flow path. Use the throttle valve shown in Figure 14-10 to control flow in a bleed-off or bypass circuit, or to control flow to or from a conventional solenoid valve.

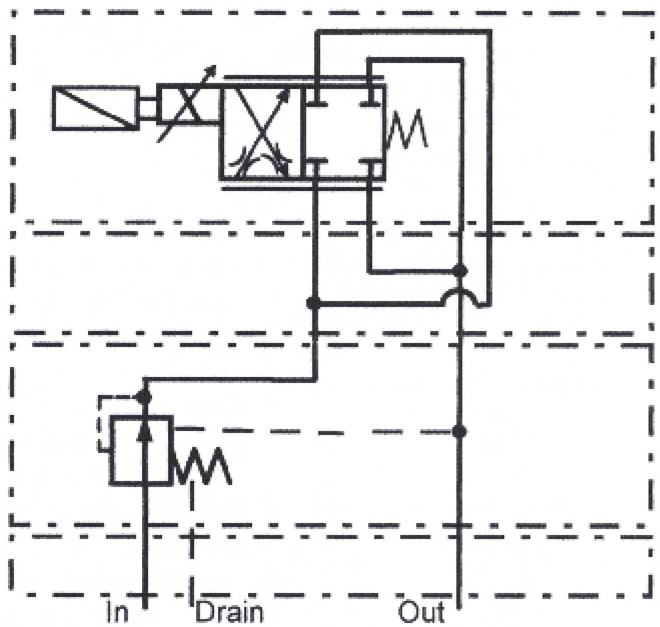


Figure 14-10. Direct solenoid-operated proportional throttle valve with LVDT feedback, parallel flow-path module, and pressure-compensating hydrostat module.

The throttle function varies flow to an actuator that needs frequent or constant adjustment. Also use a throttle valve and a conventional directional valve to give smooth acceleration and deceleration of a cylinder to eliminate shock.

A throttle valve in the tank line of a conventional solenoid valve controls actuator speed in a meter-out configuration. The actuator cannot run away with a throttle valve at this location. Make sure the directional valve can withstand any backpressure in the tank line that is greater than the circuit produces.

One throttle valve in the main pump line can vary the speed to one actuator or several that cycle at different times. This type of circuit is less expensive but requires a more-complex electrical control circuit.

The throttle valve configuration in Figure 14-10 gives infinitely variable flow. Adding the hydro stat module to the pump line keeps the pressure drop across the orifices constant. With a constant pressure drop, flow does not fluctuate. Because the 4-way valve never sees reverse flow, both flow paths can supply the circuit. Either flow path has a nominal pressure drop at a specified flow. This arrangement gives twice rated flow without excess pressure drop or heat.

The parallel flow path module comes with all flow paths internally drilled and sized to keep pressure drop to a minimum. This module is available in D03 and D05 sizes for flows up to approximately 50 gpm.

Use proportional control valves to reduce shock and give a finer degree of control to circuits that do not require extreme position accuracy, or repeatable speed and force.

Proportional valves restrict flow to and from an actuator. They work best with a pressure-compensated pump in a closed-center circuit. An accumulator in the circuit enhances cycle response time and protects the pump from pressure spikes. Systems that use proportional valves usually require a heat exchanger because energy waste is higher with this type circuit.

The following sections describe a few more circuits — with some pointers for using proportional valves in several applications. Always remember to size the valves for maximum flow and pressure drop to get optimum response and repeatability from the circuit.

Circuits with proportional throttle valves

The circuits in Figures 14-11 and 14-12 control acceleration and deceleration of an actuator. Electronic signals to these circuits also can vary the speed of the actuators infinitely.

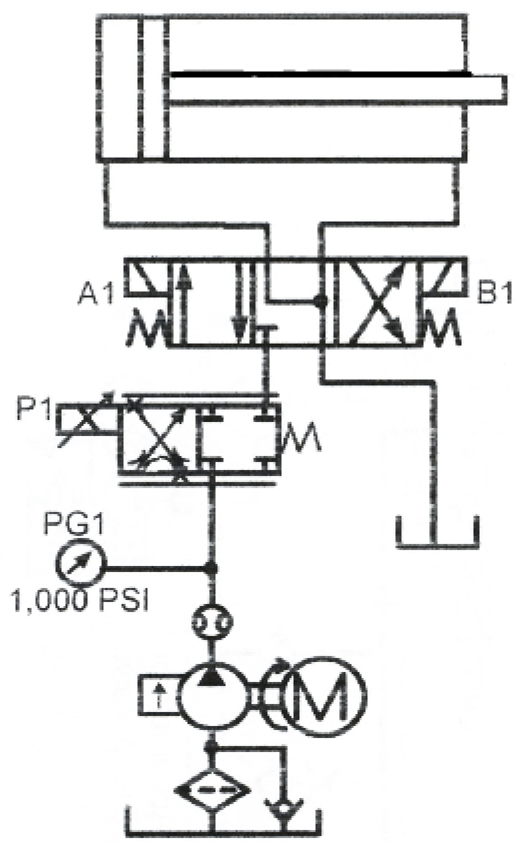


Figure 14-11. Proportional throttle valve in a meter-in circuit on the pump line — for smooth acceleration, deceleration, and speed control.

A proportional throttle valve in the pump line of Figure 14-11 controls flow to a standard solenoid valve. This circuit is good for resistive loads only because it meters fluid to the cylinder. To reduce energy waste, use a load-sensing pump and sense the line between the proportional valve and the directional valve. Load sensing lets the system operate at lower pressures during most of the cycle. Load sensing also makes the circuit pressure compensated.

The proportional throttle valve in Figure 14-12 meters flow out of the tank line of a standard solenoid valve. This circuit is good for over-running loads because it meters fluid from the cylinder. CAUTION: The directional valve may see pressure as high as twice the pump compensator setting. Make sure this pressure does not exceed its tank line rating. Allowing the throttle valve to shift abruptly in this meter-out circuit could result in detrimental shock. Use a proportional control card with adjustable ramps for this application.

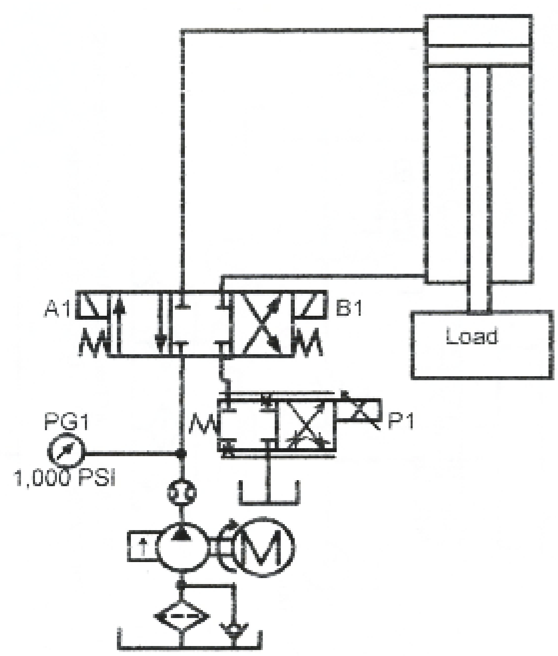


Figure 14-12. Proportional throttle valve in a meter-out circuit on the tank line — for smooth acceleration, deceleration, and speed control.

If the cylinder must set without creep, use a counterbalance valve. A throttle valve has internal leakage and may not be able to prevent cylinder drift. A counterbalance valve in this circuit must have an external drain. Backpressure at the counterbalance valve outlet modifies the pressure setting of an internally drained valve. (See Chapter 5 for a full explanation of counterbalance circuits.)

Typical conventional valve circuit with resistive load

A horizontally mounted cylinder typically requires force at all times to stroke. This cylinder configuration is known as a resistive-load application. Heavy loads at fast operating speeds usually require a means of acceleration and deceleration for smooth operation. One way to control acceleration in these circuits is to shift a standard open-center solenoid valve to extend the cylinder and let excess pump flow relieve to tank during acceleration. A small pressure spike and some heat generation take place during this part of the cycle, but otherwise cylinder start up is smooth. The schematic diagram in Figure 14-13 shows a double pump in a hi-lo circuit that operates this way. Figure 14-14 shows the circuit with a closed-center valve and a pressure-compensated pump. This arrangement eliminates some of the pressure spikes and reduces heat generation, but is more expensive.

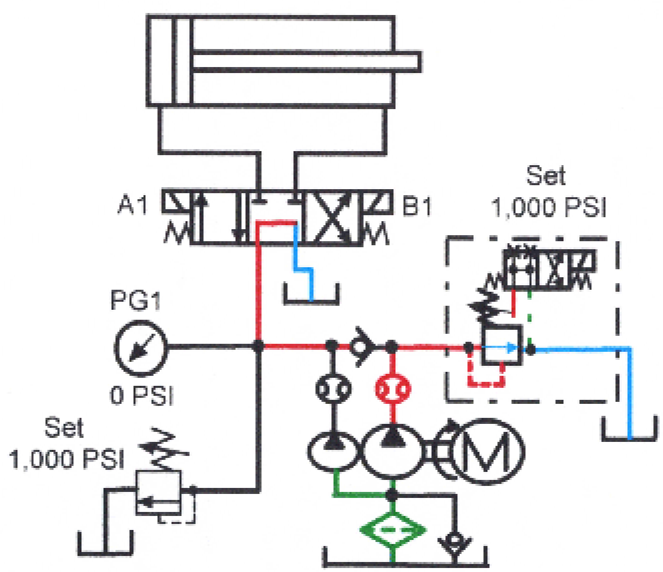


Figure 14-13. Typical hi-lo pump circuit that accelerates and decelerates actuator smoothly.

When the cylinder approaches the end of its stroke, a limit switch unloads the high-volume pump of the hi-lo circuit, decelerating the cylinder as quickly as friction on the machine members allows. When the cylinder slows to the speed of the low-volume pump, it continues to the end of stroke at a velocity low enough to eliminate most of the shock. (In this application, a cylinder with standard cushions will eliminate virtually all shock.)

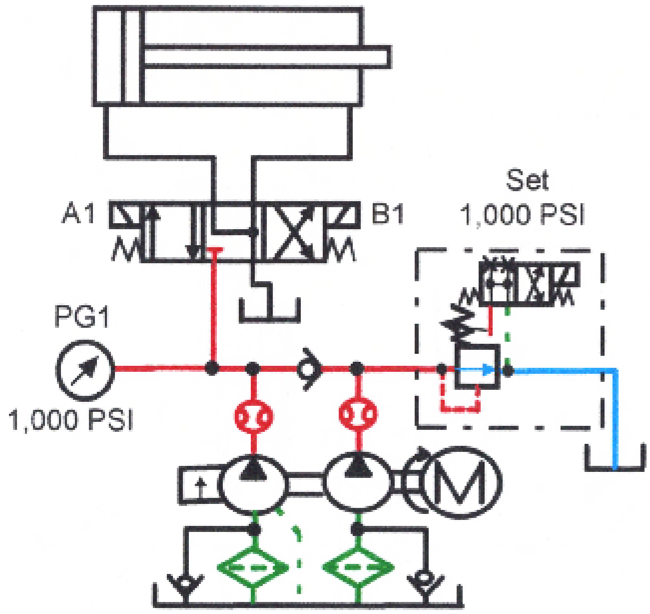


Figure 14-14. Pressure compensator and fixed-volume pump circuit that accelerates and decelerates actuator smoothly.

Figure 14-15 shows another shock-free deceleration circuit. Here a pressure-compensated bleed-off flow control dumps excess flow from a single fixed-volume or pressure-compensated pump. Deceleration is still as fast as the friction of the machine dictates. Secondary speed is adjustable to meet any requirement.

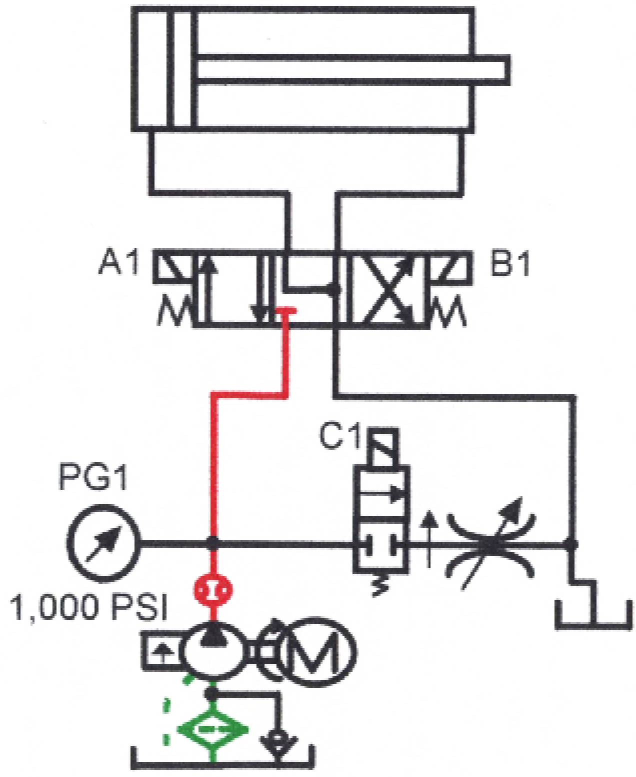


Figure 14-15. Pressure-compensated pump and flow control circuit that accelerates and decelerates actuator smoothly.

Another option for decelerating a load is to specify a cylinder with longer than standard cushions that have a tapered flow cutoff. Always specify load, pressure, and speed when ordering tapered cushions. Tapered cushions are very effective for machines that have fixed working parameters. If the load constantly changes, tapered cushions are only effective over a narrow range of the change.

Proportional valves in resistive-load circuits

The circuit in Figure 14-16 arranges a proportional valve and a pressure-compensated pump to cover all the situations in the previous section. Acceleration and deceleration are fully adjustable through a broad range with this circuit. When the load, speed, or pressure changes, it is easy to change the control parameters to match the new situation. Normally an electronic dashpot changes shifting speed of the spool between zero and five seconds. To make up for changes in fluid viscosity, pressure, or load, decelerate to a minimum creep speed and finally close the valve completely via the end-of-stroke limit switch.

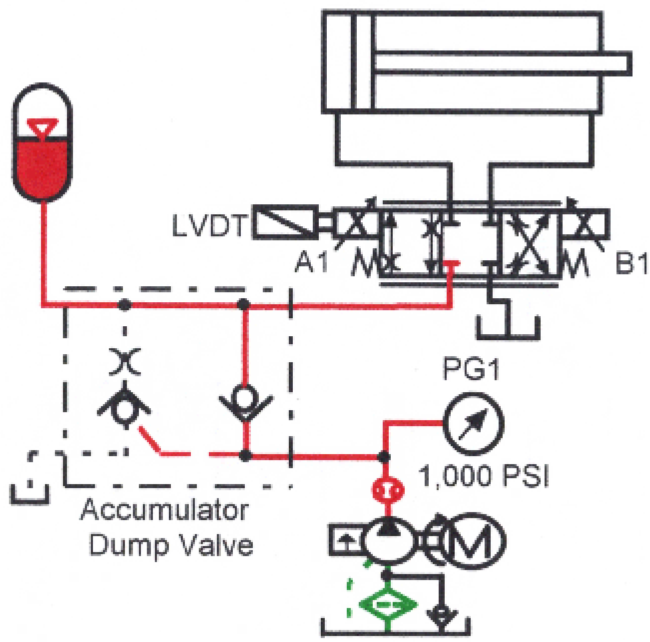


Figure 14-16. Pressure-compensated pump and proportional valve circuit that accelerates and decelerates actuator smoothly.

Proportional valves for running-way loads

Loaded cylinders that are vertically mounted usually run away or over-run pump flow in one direction. When an on/off directional valve shifts, the cylinder free falls. Free fall is a safety hazard that can cause tool or machine damage.

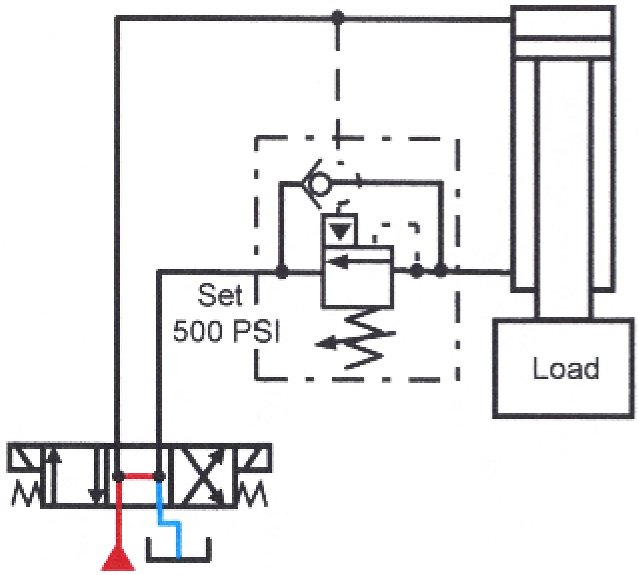


Figure 14-17. Typical circuit for a counterbalance valve with internal and external pilots to control an over-running load.

The counterbalance valve in Figure 14-17 controls an over-running cylinder. The valve allows flow from the run-away end of the cylinder as fast as the pump supplies the opposite end. When the cylinder strokes in the opposite direction, the load is resistive. Control acceleration and deceleration with any of the resistive-load circuits in the previous section when using a counterbalance valve.

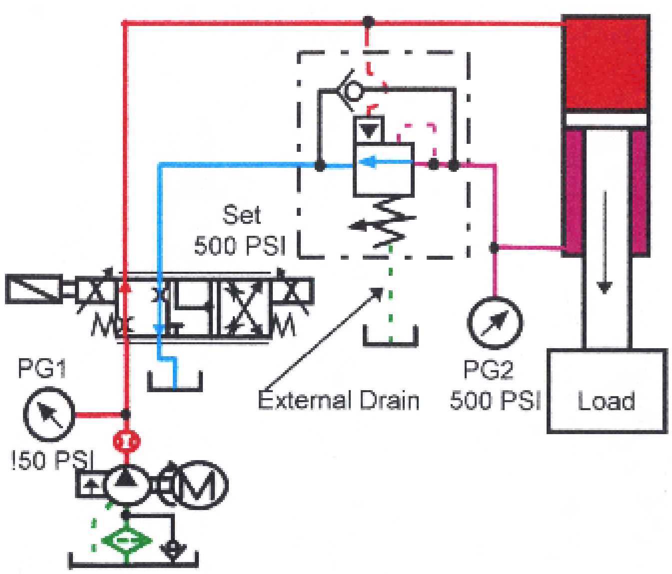


Figure 14-18. Proportional valve and externally drained counterbalance valve controlling a running-away cylinder while it is extending.

Proportional directional valves control inlet and outlet flow so that there is pressure at both ends of an actuator when it moves. A counterbalance valve often needs an external drain when used with a proportional directional valve. Without an external drain, pressure at the outlet of the counterbalance valve adds to the spring setting that keeps the valve from opening. Notice that the circuit in Figure 14-18 shows the external drain line on the counterbalance valve. With this circuit the cylinder stops smoothly when the proportional directional valve centers rapidly, as in an emergency stop.

Proportional directional valves control running-away loads because most spool designs control flow to and from the actuator. If the actuator is a hydraulic motor or a double rod-end cylinder, volume at the inlet and outlet is the same. As the proportional valve shifts to move the actuator, restricted flow from the opposite side controls acceleration, deceleration, and maximum speed.

However, the majority of cylinders have a single rod, making the volume leaving the rod end less than what enters the cap end. The volume difference is almost 50% when using a 2:1 rod cylinder. In these cylinders, the rod area equals half the piston area. (Some manufacturers offer proportional valves with spools that only allow approximately half flow through the rod port. These valves work well with a 2:1 rod cylinder.)

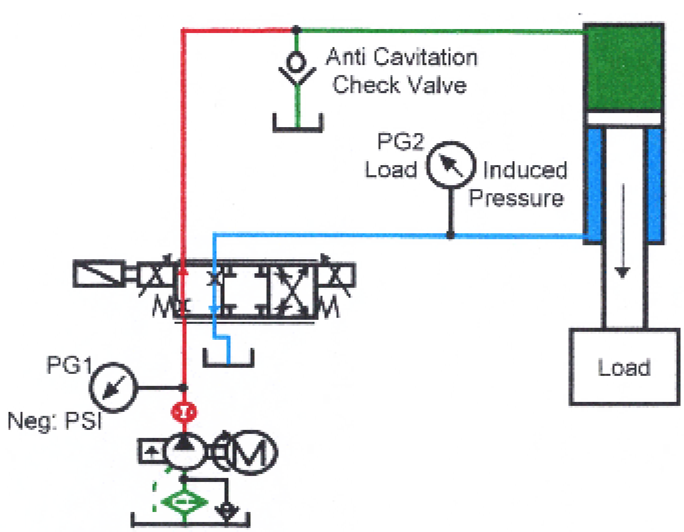


Figure 14-19. Proportional valve with anti-cavitation check valve controlling a running-away cylinder while it is extending.

Two problems can occur when using a standard spool-type proportional valve with single-rod-end cylinders and running-away loads. Figure 14-19 shows the cylinder running away from the pump, causing cavitation in the cylinder's cap end. The cylinder runs away because the proportional directional valve's meter-out function lets out more oil than it allows in at the cap end. Because the cap end does not stay full, it will pause when it meets a load while the pump fills the cap-end void. When a cylinder runs ahead of the pump, use an anti-cavitation check valve to allow fluid from the tank into the cylinder's cap end. This circuit works for applications with the over-running load at the cylinder's rod end.

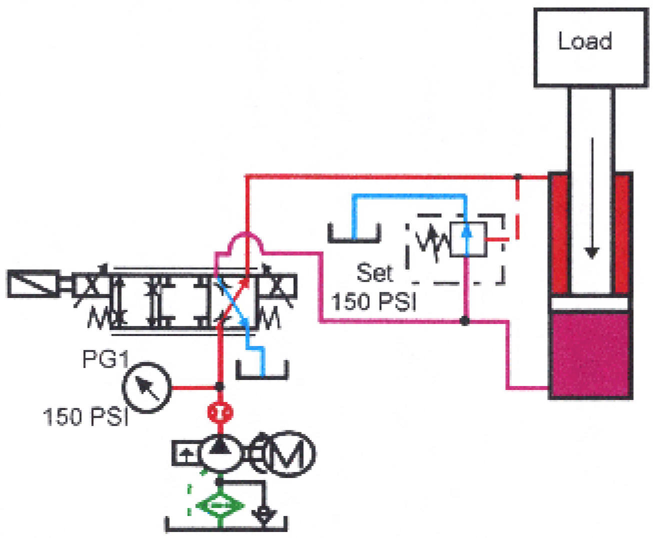


Figure 14-20. Proportional valve with externally piloted pressure-control valve controlling a running-away cylinder while it is extending.

With an over-running load at the cap end of the cylinder, the pump tries to force the cylinder to move faster than fluid can leave it. The excess fluid retards the cylinder's motion. The circuit works, but the pump wastes energy because it is at full pressure unnecessarily. The circuit in Figure 14-20 shows an external pilot-operated pressure control valve teed into the cap-end line to provide a path for excess fluid to flow directly to tank. Giving the extra oil a second path reduces rod-end pressure and wasted energy. In effect, this is a meter-in circuit for a running-away load.

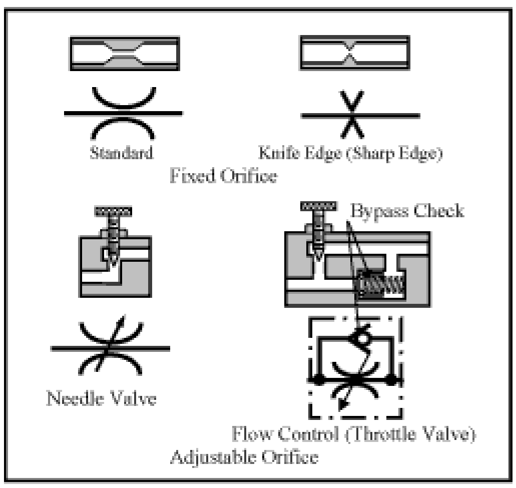
Speed control of hydraulic and pneumatic actuators

In some applications, there are times when it is necessary to vary the speed of an actuator. One method of controlling an actuator's speed is by using a variable-volume pump. This works well for a circuit with a single actuator or in multi-actuator circuits where only one actuator moves at a time. However, most circuits that need actuator-speed control have multiple actuators and some of them operate simultaneously. For most circuits, a variable orifice called a needle valve or flow control is common. Fixed orifices may be used in some cases.

Non-compensated flow control valves

Figure 13-1 *shows non-compensated flow devices in symbol and cutaway form. At the top are non-compensated fixed-orifice in-line flow controls for tamper-proof applications. These can be purchased as in-line valves or they could be a drilled plug or insert located in a pipe fitting or valve port.*

Fig. 13-1. Non-compensated flow devices

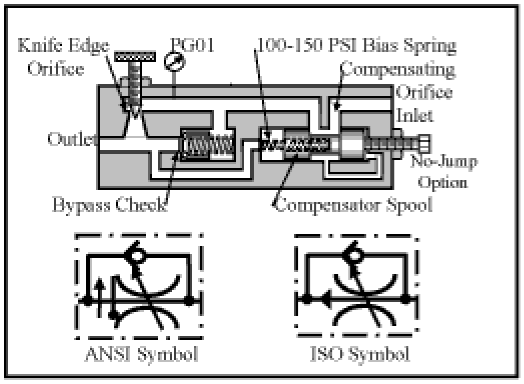


Flow through standard orifices is affected by viscosity changes in the fluid, while flow through knife-edge (or sharp-edge) orifices changes very little when fluid viscosity changes from thin to thick. A knife-edge orifice is the style used on most valves that are designated as temperature compensated. (A classic example of a non-compensated fixed orifice with a bypass check is the orificed check valve shown in ***Figure 10-2****.)*

Pressure-compensated flow control valves

The pressure-compensated flow control cutaway view and symbols depicted in ***Figure 13-2*** *are the component used with actuators that must move at a constant rate. A non-compensated flow control passes more or less fluid as pressure raises and lowers. This is because more fluid can pass through a certain size orifice when pressure drop across the orifice increases.*

Fig. 13-2. Pressure- and temperature-compensated flow control



The needle valve section of a pressure-compensated flow control is the same as any flow control. The difference is the addition of a compensator spool that can move to restrict Inlet flow at the compensating orifice. The compensator spool is held open by a 100- to 150-psi bias spring that sets pressure drop across the knife-edge orifice.

Flow from the inlet goes through the compensating orifice, past the compensator spool, and out through the knife-edge orifice. A drilled passage ports Inlet fluid to the right end of the compensator spool, which forces the spool to the left when pressure tries to go above 100 to 150 psi at gauge PG01. After pressure reaches or goes above 100 to 150 psi, the compensator spool moves to the left and restricts flow to the knife-edge orifice flow control. Pressure at gauge PG01 never goes above 100 to 150 psi (plus any backpressure at the outlet). Pressure at the outlet is ported to the bias-spring chamber and increases the spring force. The compensator spool assures that pressure drop across the knife-edge orifice flow control stays at a constant 100 to 150 psi. With a constant pressure drop, flow stays the same regardless of inlet or outlet fluctuations.

Pressure-compensated flow controls are four to eight times more expensive than standard controls so they should only be applied to actuators that must move consistently.

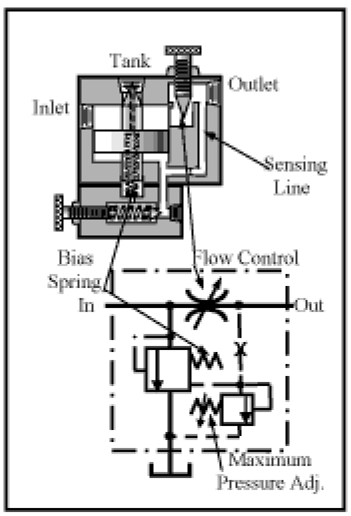
The no-jump option is an adjusting screw that holds the compensator spool within a few tenths of an inch of its operating position. This is an especially important option when the valve is oversize for the present flow setting. A compensator spool without a stroke limiter may close and open violently until it stabilizes and sets pressure drop for the orifice. During this time the actuator also moves erratically.

The two symbols represent the American National Standards Institute (ANSI) and the International Standards Organization (ISO) way of indicating that the valve is pressure compensated. The arrow indicating pressure compensation is easier to distinguish in the ANSI symbol -- especially when the schematic drawing has been reduced to fit into a machine's documentation book.

Three-port flow control valve

Three-port flow controls are mainly used in fixed-volume pump circuits to save energy. (See the load-sensing pump circuit explained in Chapter 8.) If 20 gpm of fluid enters the Inlet and the flow control is set at 12 gpm, 8 gpm goes to tank as wasted energy. With a conventional relief valve setup, pressure between the pump and flow control would be maximum. With the 3-port flow control, pressure in this portion of the circuit is whatever it takes to move the actuator plus bias-spring force. (Bias-spring force is usually 70 to 125 lb.) An outlet pressure of 200 psi gives a pressure of 270 psi between the pump and the flow control. All fluid going to tank is discharged at 270 psi, not 2000 psi. This takes place because the sensing line sends feedback to the pressure-control side of the relief valve, allowing it to open at load pressure plus bias-spring force. Pressure between the pump and flow control constantly changes with load variations. When the load requires more than the maximum-pressure adjustment setting, the relief valve opens and sends all pump flow to tank at maximum pressure.

Fig. 13-3. Three-port flow control

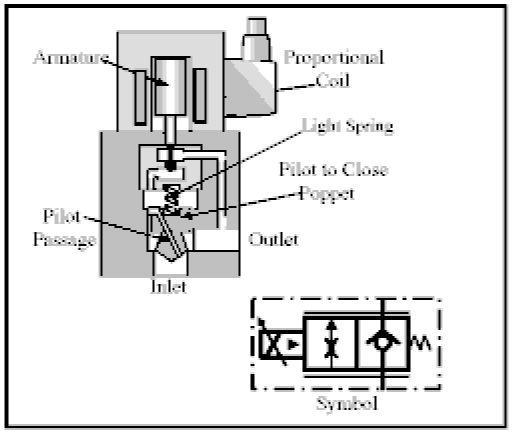


A 3-port flow control is only effective with one actuator -- or one actuator at a time. It would not be useful on a pressure-compensated pump circuit because a load-sensing circuit for this type pump would save even more energy. (See Chapter 8 for a load-sensing circuit with a pressure-compensated pump.)

Proportional flow control valves

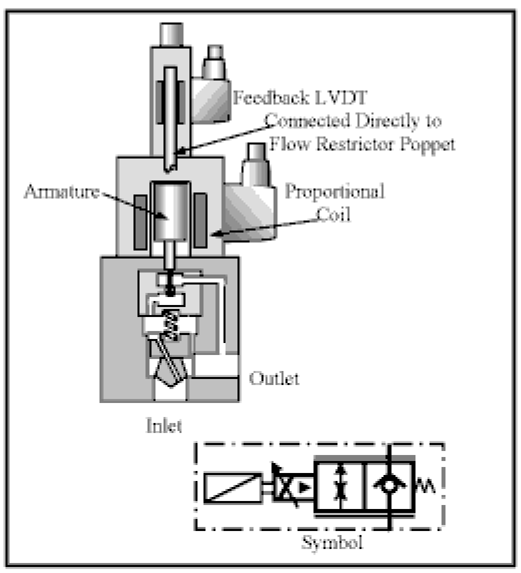
Figures 13-4 and 13-5 *show cutaways and symbols for proportional flow control valves that can electronically remotely control flow through a PLC or other controller. There are many different designs of valves and controllers that control pneumatic or hydraulic fluid. The design in* ***Figure 13-4*** *uses a modified 2-way pilot-to-close poppet with a drilled pilot passage to send inlet fluid behind it. A light spring holds the poppet closed when there is no pressurized fluid at the Inlet.*

Fig. 13-4. Proportional flow control valve
without feedback



The armature controls a small normally closed poppet and shifts the signaled amount to let fluid behind the pilot-to-close poppet leave faster than the pilot passage can supply it. This causes a pressure imbalance that lets the pilot-to-close poppet open enough to give the correct fluid flow. The flow rate is infinitely variable and can be controlled from a variety of inputs.

Fig. 13-5. Proportional flow control valve
with feedback

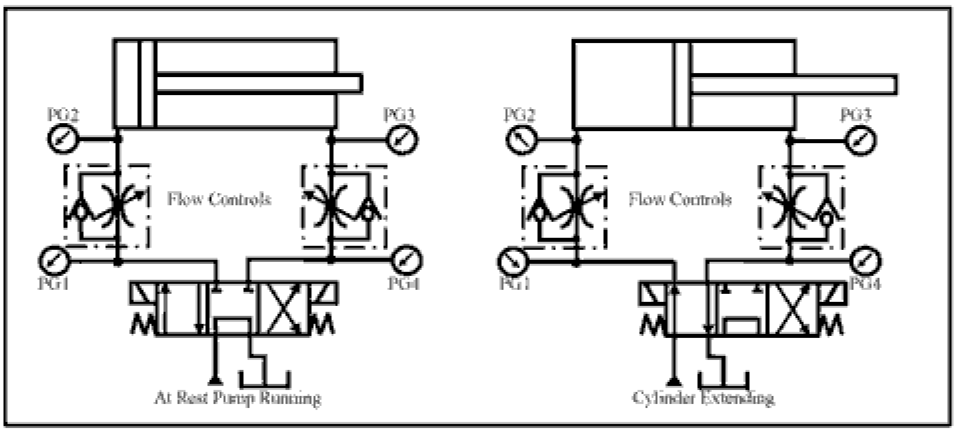


*The valve in **Figure 13-4** opens from a given signal but may not always repeat a set flow from the same input. The feedback LVDT added to the valve in **Figure 13-5** assures that the pilot-to-close poppet always shifts the same amount so it has the same size flow opening. However, pressure or viscosity changes still affect actual flow, so a hydrostat is necessary when exact flow* *repeatability is required. Many manufacturers make valves with a built-in hydrostat for pressure compensation.*

Meter-in flow control circuits

Figure 13-6 *provides a schematic drawing of a meter-In flow control circuit restricting fluid as it enters an actuator port. Meter-in circuits work well with hydraulic fluids, but can give erratic action with air. Note that the cylinder is horizontally mounted, which makes it a resistive load. Meter-in flow controls only work on resistive loads because a running-away load can move the actuator faster than the circuit can fill it with fluid.*

Fig. 13-6. Meter-in flow control circuit



The left-hand circuit in Figure 13-6 is shown at rest with the pump running. Notice that the check valves in the flow controls force fluid through the orifices as it enters the cylinder and lets fluid bypass them as it leaves.

The right-hand circuit depicts conditions as the cylinder extends. The directional control valve shifts to straight arrows and pump flow passes through the left-hand flow control to the cylinder cap end at a controlled rate. Fluid leaving the cylinder rod end flows to tank without restriction. The cylinder extends at a reduced speed (in a hydraulic circuit) until it meets a resistance it can't overcome or it bottoms out. With the non-compensated valve shown, speed can vary as pressure fluctuates or viscosity changes.

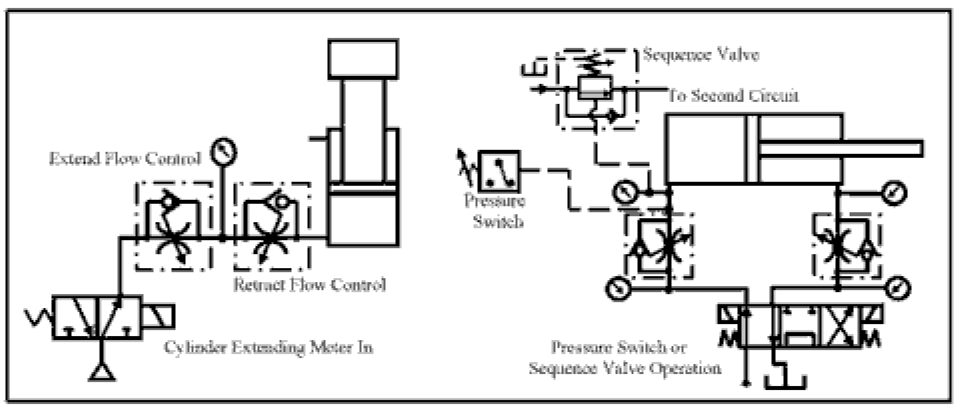
While the cylinder is in motion, pressure at PG1 reads the setting of the relief valve or pump compensator. The pressure at PG2 reads whatever it takes to move the load at any point in the cycle. Pressures at PG3 and PG4 only read tank-line backpressure as the cylinder extends.

It is obvious that if the cylinder had an external force pulling on it, it would extend rapidly. Because fluid enters the cap end at a reduced flow rate, a vacuum void would form there until the pump had time to fill it.

Meter-in flow controls can have a problem in pneumatic circuits. When fluid is directed to the cylinder cap end, pressure at PG1 immediately rises to the regulator setting. However, pressure at PG2 starts at zero and increases slowly. Until pressure at PG2 rises enough to generate breakaway force, the cylinder does not move. At breakaway pressure, the cylinder extends quickly and expanding air may cause it to lunge. Often, the lunge forward moves the piston ahead of the incoming air and pressure drops back below the breakaway level so the piston stops. Pressure starts to build again and the lunge/stop scenario continues to the end of stroke. The meter-out circuit discussed next is always the best choice to control air cylinders.

*The circuits in **Figure 13-7** show applications where a meter-in circuit is the only choice for both pneumatics and hydraulics. On the left in **Figure 13-7**, a single-acting pneumatic cylinder is mounted with the rod vertically up. The only way to control extension speed is via a meter-in flow control. When retraction speed must be controlled as well, a meter-out flow control also is necessary.*

Fig. 13-7. Circuits where meter-in flow control is required



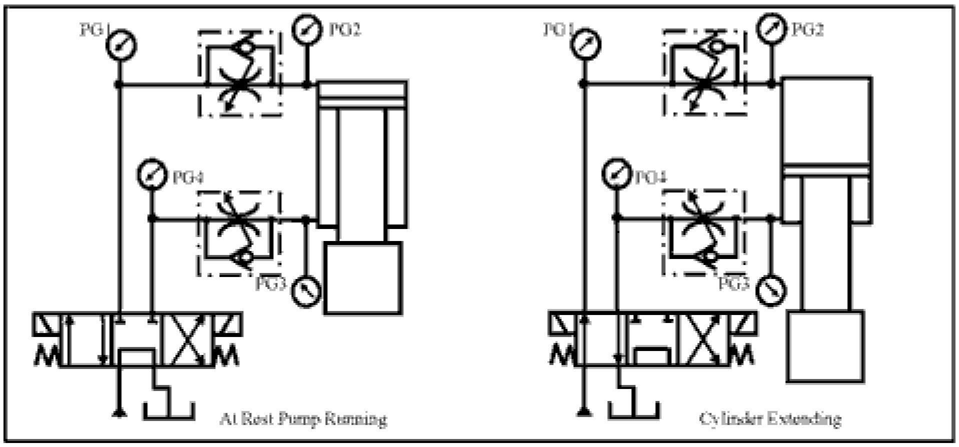
*The cylinder pictured on the right in **Figure 13-7** is extending to perform an operation prior to retracting or starting the cycle of another actuator. A signal to continue the cycle can come from a pressure switch or a sequence valve. Either of these devices can be set to give an output at any pressure. Usually they are set 50 to 150 psi below system operating pressure for hydraulics, or 5 to 15 psi lower for air. The reason for meter-in flow control is that pressure between the flow control and the cylinder normally stays low until the cylinder contacts the workpiece. At work contact, the resulting pressure buildup switches these pressure-actuated devices and starts the next sequence. Always remember: a pressure switch or sequence valve does not directly indicate that the actuator has reached a physical position. They only indicate that pressure has reached a predetermined setting . . . not why it has.*

Other circuits that require meter-in flow controls are the load-sensing pump circuits in Chapter 8.

Meter-out flow control circuits

Figure 13-8 *shows a schematic drawing of a meter-out flow control circuit that restricts fluid as it leaves an actuator port. Meter-out circuits work well with both hydraulic and pneumatic actuators. Cylinder-mounting attitude is not important because outlet flow is restricted and an actuator cannot run away. Meter-out flow controls work on resistive loads or running away loads because the actuator can never move faster than the fluid leaving it allows.*

Fig. 13-8. Meter-out flow control circuit



The left-hand circuit in ***Figure 13-8*** *is shown at rest with the pump running. Notice how the check valves in the flow controls allow fluid to bypass the orifices and freely enter the cylinder. As fluid leaves the cylinder, it is forced through the orifices at a set rate. The only gauge showing pressure is PG3 because the load on the cylinder rod is inducing pressure at the valve's blocked port.*

The right-hand circuit shows conditions when the cylinder is extending. The directional control valve shifts to straight arrows and pump flow bypasses the upper flow control to go to the cylinder cap end. Fluid leaving the cylinder rod end is held back before it goes to tank -- even with an external load trying to move it. The cylinder extends at a reduced speed in both hydraulic and pneumatic circuits until it meets a resistance it can't overcome or it bottoms out. With the non-compensated valve shown, speed can vary as pressure fluctuates or viscosity changes in a hydraulic system. (There are no pressure-compensated flow controls for pneumatic circuits.)

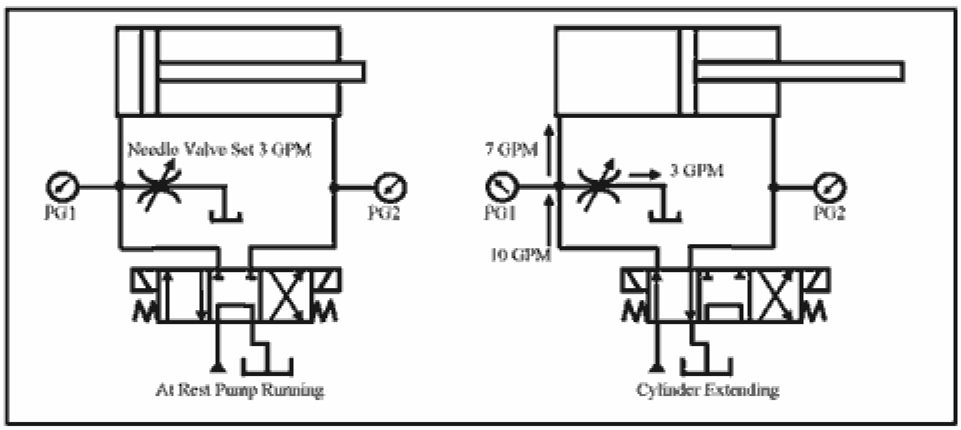
While the cylinder is in motion, gauges PG1 and PG2 read the relief valve or pump compensator setting. Gauge PG4 reads tank backpressure. Gauge PG3 reads load-induced pressure plus the pressure from cap-area-to-rod-area intensification. This intensified pressure could be 1.2 to 2 times the cap-end pressure, or higher, depending on the rod size.

Meter-out flow controls work equally well in pneumatic circuits when the load is constant. Changing loads can cause the actuator to stop and/or lunge under certain circumstances. (For a *more extensive coverage of flow control circuits and situations that can arise with them, see our second e-book entitled "Fluid Power Circuits Explained," which will be launched on hydraulicspneumatics.com in the coming months.*

Bleed-off flow control circuits

Bleed-off flow control circuits are found only in hydraulic systems and normally only in those with fixed-volume pumps. There is little or no advantage to using this type flow control with pressure-compensated pumps. ***Figure 13-9*** *shows a bleed-off circuit at rest with the pump running. A needle valve's inlet is teed into a line going to the cylinder and its outlet is connected to tank. The circuit only works with one actuator moving at a time because all pump flow goes to the presently operating function. Like a meter-in circuit, it only works with resistive loads because it controls fluid into the actuator. The main plus for this type speed control is it saves energy while using a fixed-volume pump with low-pressure travel forces.*

Fig. 13-9. Bleed-off flow control circuit



When the directional valve in ***Figure 13-9*** *shifts, all pump flow passes through it and toward the actuator. On the way to the actuator, part of the flow is bled off to tank, so the actuator does not reach full speed. Pressure at PG1 only rises to whatever it takes to move the actuator and its load, so excess flow goes to tank at low pressure. (When using a fixed-volume pump and a meter-in or meter-out circuit, excess flow also goes to tank, but at relief valve pressure.) Many circuits only perform work at the end of stroke so this flow control system saves energy while the actuator moves to and from the work position, yet still gives good speed control.*

Some words of caution:

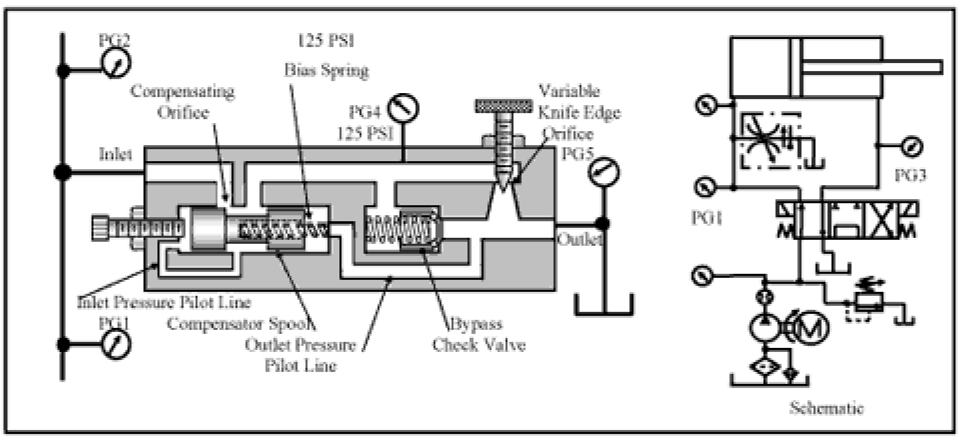
- Pressure in the actuator during traverse time must be higher than the pressure in the path to tank, so fluid will flow to tank.
- Because pressure may change during traverse time (especially when the actuator contacts the workpiece), use a pressure-compensated needle valve so flow to tank remains constant.

- Even with a pressure-compensated needle valve, actuator speed will be inconsistent. Pump and/or actuator efficiency allows bypass that directly affects flow to the actuator not bleed-off to tank.

Pressure-compensated flow control valve applications

*When pressure drop across an orifice changes, flow through the orifice also changes. As pressure drop increases, flow increases, and as pressure drop decreases, flow decreases. Because of this fact, if pressure drop across an orifice were constant, regardless of upstream and downstream pressure fluctuations, then flow through it would stay the same. A pressure-compensated flow control valve (such as the one shown in **Figure 13-2**) automatically maintains a constant pressure drop across the orifice. There is a short discussion on pressure-compensated flow control valves on page 13-1, but a valve in cutaway form is applied to a bleed-off circuit in **Figure 13-10**.*

Fig. 13-10. Bleed-off flow control circuit with positive-displacement pump and pressure-compensated flow control valve



In the bleed-off circuit, fluid from the directional control valve is sent to the cylinder to start it extending. Because the circuit has a fixed-volume pump and needs speed control, a bleed-off flow control is used to save energy. Instead of controlling flow to or from the actuator, excess flow is bled to tank across a pressure-compensated flow control at whatever pressure it takes to move the fluid. A meter-in or meter-out flow control circuit would send excess flow to tank across the relief valve at maximum pressure – wasting a lot more energy.

The reason for using a pressure-compensated flow control is that pressure will fluctuate as the actuator moves toward the workpiece and the flow to tank from a non-compensated flow control would change continuously. As a result, actuator speed could vary considerably while it moves. With a pressure-compensated flow control, flow to tank is constant, but actuator speed could still change due to pump efficiency as pressure increases or decreases. Any speed change from pump efficiency is present but practically imperceptible.

*In the **Figure 10-13** circuit, a 10-gpm pump sends 7 gpm to the cylinder and 3 gpm to tank. Fluid entering the pressure-compensated flow control passes by the compensator spool and flows on to the variable knife-edge orifice, which is set at 3 gpm. The variable knife-edge orifice restricts flow and creates backpressure in the incoming fluid. When backpressure reaches (and attempts to exceed) 125 psi, fluid in the inlet-pressure pilot line forces the compensator spool to the right. This restricts flow at the compensating orifice. After the compensator spool settles in at its 125-psi bias-spring setting, pressure at PG3 reaches 125 psi and stays there. This means that pressure drop across the variable knife-edge orifice is 125 psi. As the cylinder continues to move and pressure at PG1 and PG2 increases or decreases, pressure at PG4 stays at 125 psi and flow is constant. The cylinder moves at the same speed whether pressure is at or above 125 psi, and as much as 125 psi below the maximum pressure setting.*

Fig. 13-11. Meter-in flow control circuit with pressure-compensated pump and pressure-compensated flow control valve

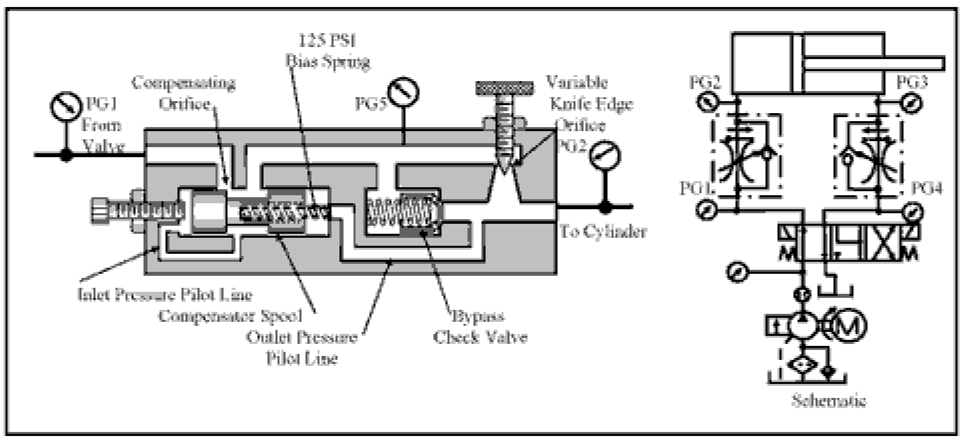


***Figure 13-11** shows a pressure-compensated flow control in a meter-in circuit. Fluid from the valve enters the flow control and is restricted. Backpressure from restricted flow goes through the inlet-pressure pilot line and shifts the compensator spool to the right, restricting flow to the variable knife-edge orifice. Backpressure from cylinder resistance acts on the right end of the compensator spool through the outlet-pressure pilot line and adds to the 125-psi bias-spring force. This action and interaction always keeps pressure 125 psi higher at PG5 than at PG2. A constant pressure drop across the orifice maintains a constant flow to the cylinder.*

Fig. 13-12. Meter-out flow control circuit with pressure-compensated pump and pressure-compensated flow control valve

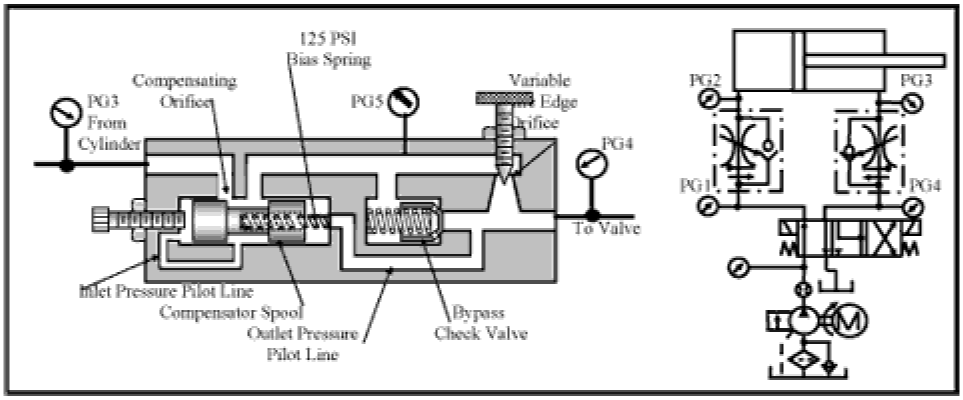


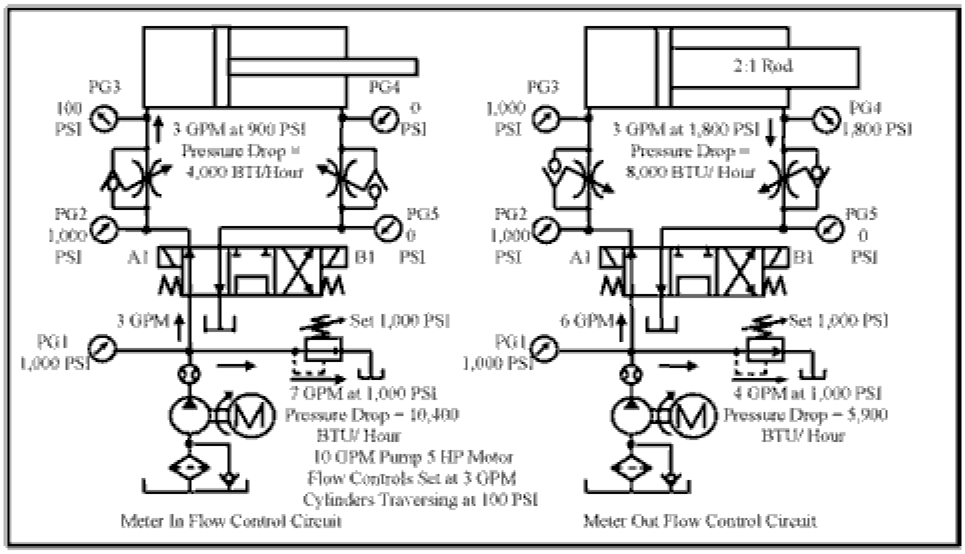
Figure 13-12 *shows a pressure-compensated flow control in a meter-out circuit. Fluid from the cylinder rod end enters the pressure-compensated flow control and is restricted at the variable knife-edge orifice. Backpressure through the inlet-pressure pilot line shifts the compensator spool to the right and restricts flow to the variable knife-edge orifice. Pressure at PG5 settles in at 125 psi and flow stays the same across the variable knife-edge orifice. Any backpressure from tank flow adds to the 125-psi bias-spring force and increases pressure at PG5 so it always stays 125 psi above PG4.*

Pressure-compensated flow control valves are as much as five times more expensive than non-compensated models, so they should not be specified when accurate flow control is not required.

Changes in fluid viscosity also cause flow fluctuations. Thick fluid flows more slowly than thin fluid. A flow control valve without temperature compensation allows varying flow from cool oil at startup to oil running at normal or high temperature. The most common fix for viscosity variations is to use a knife-edge orifice. Knife-edge orifices have no flats to slow fluid flow, so they produce little change in flow between thick and thin fluids. Other devices to obtain constant flow with viscosity variations are available, but they can be complex and may cause malfunctions.

A flow control in a hydraulic circuit always generates heat. Some pump and flow control combinations produce a lot more heat and should be avoided if possible. The following examples show different pump and flow control combinations and suggest how much heat can be expected.

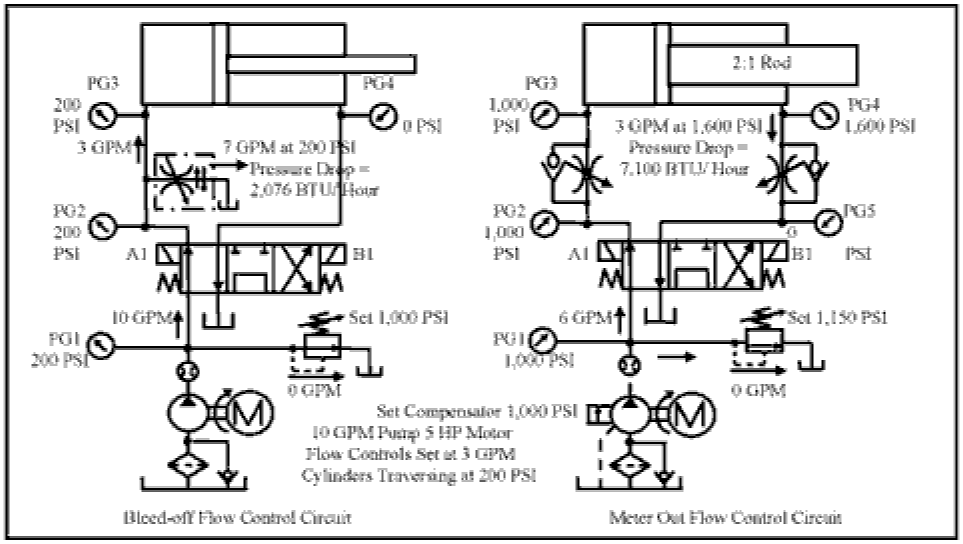
Fig. 13-13. Heat generation in fixed-volume pump circuits with meter-in and meter-out flow controls



The fixed-volume pump and meter-in or meter-out flow control combination in ***Figure 13-13*** *is the worst-case situation. The example shows a cylinder stroking to the workpiece with flow controls set at 3 gpm. A 10-gpm pump driven by a 5-hp electric motor powers the circuit. Because it only takes 100 psi to move the cylinder while traversing, a lot of heat-generating energy is wasted. This example is somewhat exaggerated, but is not at all unheard of. Note the example only shows energy wasted on the extension stroke. With a reduced-speed retraction stroke, heat generation could almost double the figures shown.*

The main generator of heat is the excess pump flow going across the relief valve at 1000 psi. The two circuits in ***Figure 13-14*** *show how to eliminate such wasted energy with a different flow control circuit or a different pump. While the energy wasted across the flow control valve is much less at these low flows, it still adds heat to a system. Also, the amount of pressure drop may be lower than indicated here because some actuators require more pressure to move them to and from the workpiece. Energy loss across a flow control cannot be eliminated. The amount of loss depends on pressure drop and flow rate across the orifice.*

Fig. 13-14. Two flow control circuits that reduce heat generation

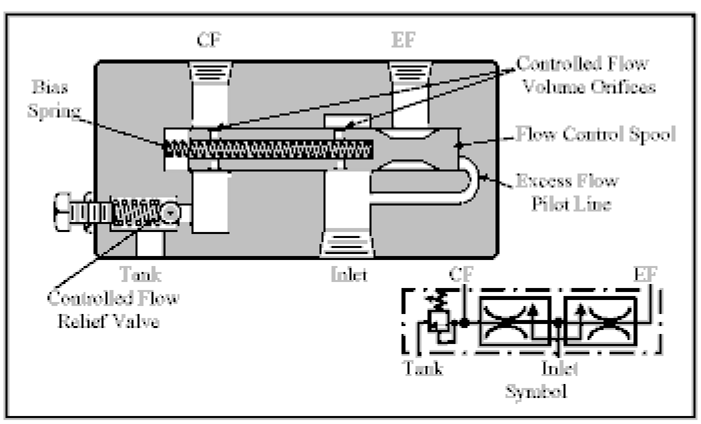


*The circuits in **Figure 13-14** show a fixed-volume pump with a bleed-off circuit and a pressure-compensated pump with a meter-in circuit. Both of these combinations save a lot of energy (although not as much as the load-sensing circuit that was shown in **Figure 8-27**). This type of flow control circuit wastes the least energy possible when using flow controls for speed control.*

Fluid flow dividers

*The flow divider in **Figure 13-15** is called a priority flow divider because it splits pump flow into a fixed controlled-flow (CF) outlet and sends excess fluid out an excess flow (EF) port. Volume orifices (drilled as specified by the purchaser) preset fluid flow out of the CF port. EF flow is any flow the pump produces over and above the controlled flow. This type flow divider is often used on vehicle power steering, where an engine-driven pump's output may vary as rpm changes or as its flow is used for other functions. A priority flow divider assures that the power steering always has ample fluid at any engine speed or when other functions are active.*

Fig. 13-15. Priority flow divider with relief valve in priority leg



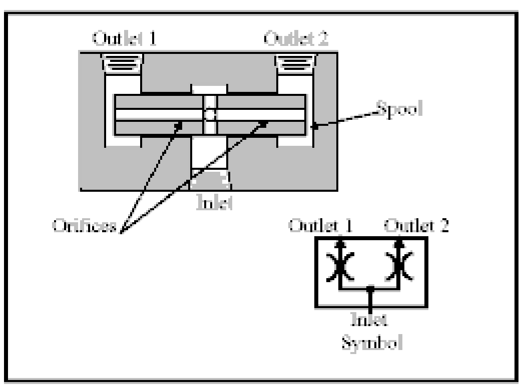
As fluid enters the valve, the path of least resistance leads through the controlled-flow-volume orifices and out port CF. If pump flow is more than the volume orifices can pass, pressure builds on the right end of the flow-control spool through the excess-flow pilot line. When pressure rises enough to overcome the bias spring and any backpressure from the steering circuit, the flow-control spool moves to the left, just enough to let excess flow exit through port EF. Excess flow changes as pump flow varies, but flow to port CF takes priority. A relief valve in port CF can be set for any pressure and has no affect on pressure at port EF. The controlled-flow relief valve is required even when maximum pressure is the same for both outlets.

Notice that controlled flow is pressure compensated. As pressure builds at port CF, it pushes back against the excess-flow pilot-pressure pilot to maintain a constant pressure drop across the volume orifices.

Priority flow dividers are also manufactured with adjustable flow for the priority port and without a relief valve for circuits that already have one. (The symbol shown is borrowed from a manufacturer's catalog because there is no standard symbol in ANSI or ISO literature.)

The flow divider in ***Figure 13-16*** *is a spool-type divider that splits flow at any predetermined rate according to the sizes of the drilled orifices. It is usually set up with identical orifice sizes for a 50-50 split. This particular design does not allow reverse flow, so bypass check valves are required when flow must return the same way it entered.*

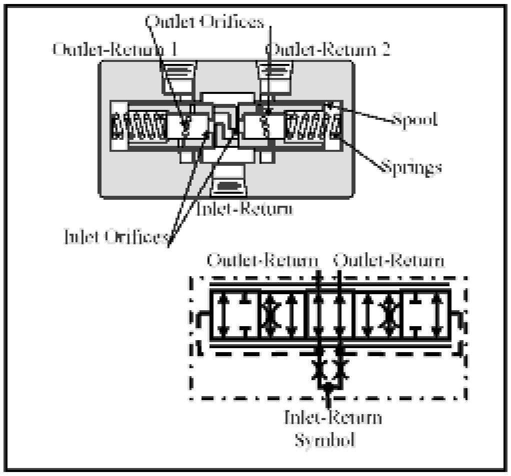
Fig. 13-16. Spool-type flow divider for 50-50 split



Fluid entering the Inlet port goes left and right through orifices, then out outlets 1 and 2. When either outlet encounters more backpressure than the other does, the high-pressure side forces the spool towards the low-pressure side until pressures on both sides equalize. Equal pressure drop across both orifices produces equal flow. (Most manufacturers specify flow equality at ±5%.) Pressure differences at the two outlets should be low because Inlet pressure always equals the highest outlet pressure -- which means pressure drop across the low-pressure outlet wastes energy.

Spool-type flow dividers only split flow. When more than two outlets are required, dividers must be used in series. A 50-50 split divider flowing into two more 50-50 dividers gives four equal outlets. A 66-33 divider into a 50-50 divider gives three equal outlets. The flow divider/combiner in ***Figure 13-17*** *equalizes flow in both directions. It can be used with double-acting actuators to synchronize speed in both directions of travel. The spool in this divider is made in two sections with a connecting link that allows the sections to move together in the closed condition (as shown) for combining, or be spread by Inlet pressure when they are dividing. Springs at both ends of the spool keep the sections together when pressure equalizes or is not present. Inlet orifices set nominal flow, while outlet orifices control flow to or from an actuator.*

Fig. 13-17. Spool-type flow
divider/combiner with 50-50 split



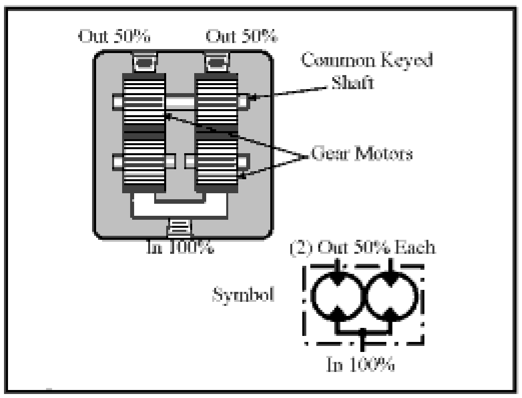
Flow to the inlet-return port goes through the inlet orifices to split into two equal parts. Pressure drop across the orifices causes the split spool to separate so the outlet orifices are working at the outer edge of the outlet-return ports. When unequal pressures on its ends shift the spool, flow is retarded to the low-pressure outlet port to keep it from receiving too much fluid. When the actuator reverses, flow into the outlet-return ports goes through the outlet orifices and on through the inlet orifices, causing the spool sections to come together. Now the outlet orifices control return flow on the inner edge of the outlet-return ports. They will retard flow from any actuator port that is trying to run ahead.

Motor-type flow dividers

A motor flow divider is constructed from two or more hydraulic motors -- in a common housing -- with a common shaft running through one set of gears on all motor sets. There is a common Inlet to all motors and separate outlets. The motors are usually gear-on-gear or gerotor design. Flow split is commonly 50-50 but many outlet flow combinations are possible by changing gear or gerotor widths.

*The cutaway view and symbol in **Figure 13-18** pictures a 2-outlet 50-50 split gear-motor-type flow divider. (There is no ISO or ANSI symbol for a motor flow divider so the one shown in the figure is from a supplier's catalog.) One gear from each motor set is keyed to the common shaft, so both motors must turn at the same rate. If one motor stalls, they both stop because of the common-shaft arrangement. Due to internal clearances in the motor elements, there is some bypass flow that does not turn the motors. As a result, the outlet flows are not always exactly equal . . . especially at high outlet-pressure differences.*

Fig. 13-18. Motor-type flow divider with 50-50 split

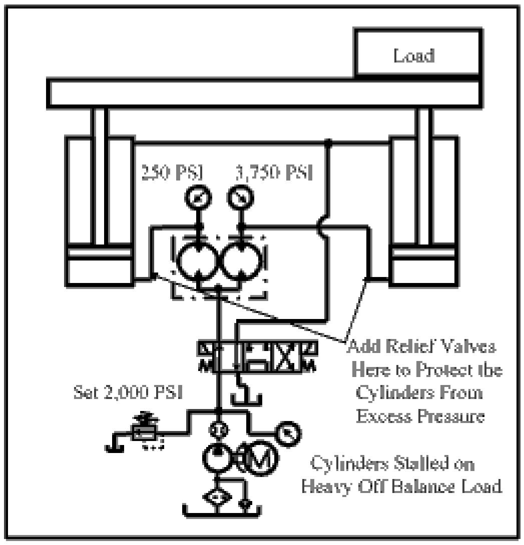


*From **Figure 13-18**, it should be obvious that this flow divider does not have a priority side like a spool-type flow divider does. Thus, when Inlet flow changes, it is always split equally. The main advantage of motor-type over spool-type flow dividers is there is less wasted energy when the outlets are not at or near the same pressure. If pressure at the right outlet was 1500 psi and pressure at the left outlet was 300 psi, pressure at the inlet would be 900 psi. Pressure at the inlet is always the average of the sum of the outlets.*

This feature can be an asset or a problem. If one outlet meets resistance while the other is flowing to tank, an inlet pressure of 2000 psi can result in the pressurized outlet intensifying to 4000 psi. If pressure that high cannot be tolerated, a relief valve must be installed at the outlets. On the other hand, intensification can allow a 1000-psi system to produce 2000 psi to perform work -- similar to a hi-lo pump circuit. Note that while pressure doubles, flow is halved through the high-pressure outlet.

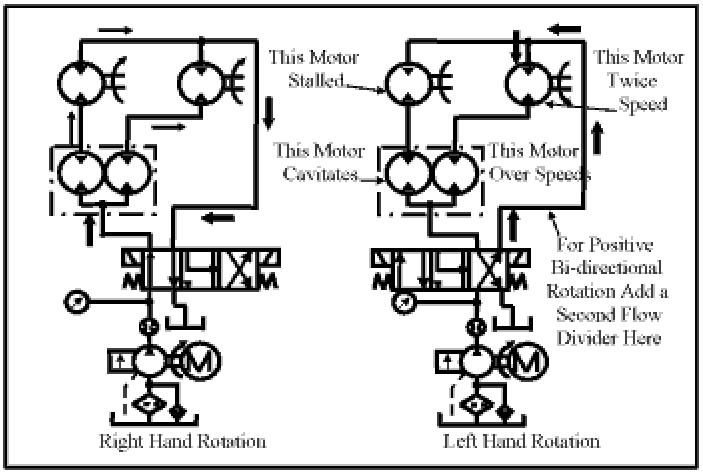
*Looking at **Figure 13-18**, it appears the motor flow divider is also a combiner. This is partially true. The circuit in **Figure 13-19** shows a motor flow divider synchronizing two hydraulic motors. As the motors turn in right-hand rotation, they stay almost perfectly synchronized. Pressure to each motor may vary but flow from each flow-divider outlet remains near constant. If the directional control valve shifts to turn the motors in left-hand rotation, the flow divider may get equal flow and the hydraulic motors may stay synchronized. However, if one hydraulic motor meets more resistance than it can overcome and stalls, all pump flow goes to the running* *hydraulic motor. The second motor then turns twice as fast. During this scenario, one flow-divider motor overspeeds while the opposite one cavitates. The only way to make sure both hydraulic motors stay synchronized in both directions of rotation is to install motor flow dividers at both valve ports.*

Fig 13-19. Synchronizing circuit for 50-50 flow divider



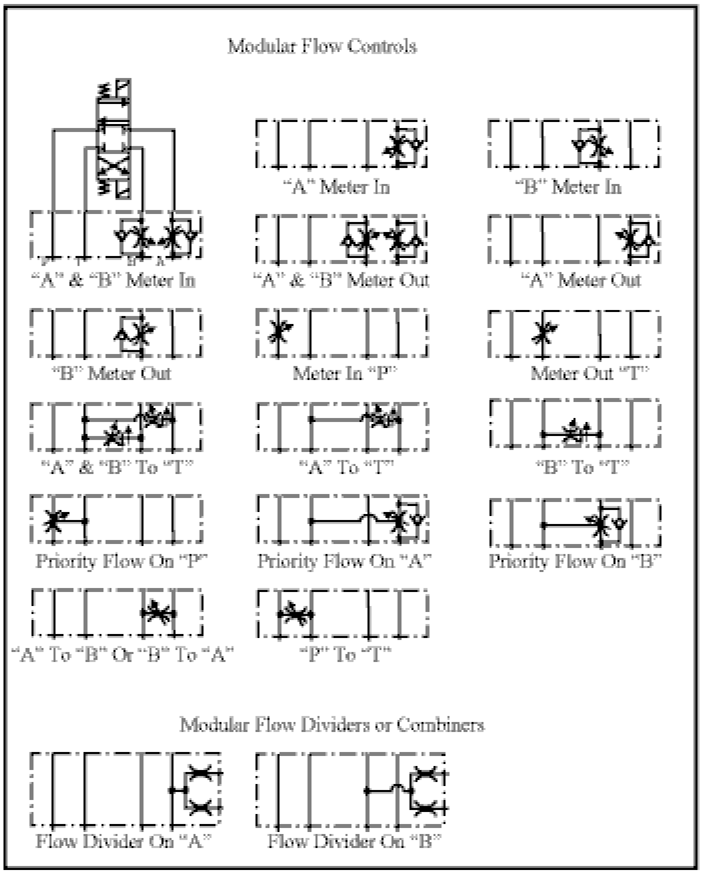
Spool and motor flow dividers work reasonably well to synchronize circuits with hydraulic motors and cylinders. However, because both devices do not divide flow perfectly, the actuators they control will not stay perfectly synchronized. A high-pressure difference at the divider's outlets is the worst problem; it can allow a 5 to 10% lag in actuator position. This means that synchronizing circuits using flow dividers often require some type of re-synchronizing valving to realign the actuators more exactly when they stop at home position. (Due to internal bypass, actuators with short cycles may re-synchronize themselves because the error is small.)

Fig. 13-20. Motor-type flow-divider circuit with 50-50 split



*Another design consideration is the intensification of pressure at the outlets of a motor flow divider. The circuit in **Figure 13-20** has two cylinders that are synchronized by a motor flow divider. Because this circuit operates at 2000 psi, it is possible that pressure at one cylinder could reach as much as 4000 psi due to intensification. Intensification occurs when one cylinder is lightly loaded or has no load and the other one is loaded heavily. In **Figure 13-19**, the load is shifted to one side of the platen -- making the right-hand cylinder do all the work. Inlet pressure is at 2000 psi and the cylinders are stalled. Pressure at the lightly loaded left-hand cylinder is 250 psi, so pressure at the right-hand cylinder is 3750 psi. The intensification is due to energy transfer through the motors in the flow divider. Because inlet pressure for both motors is 2000 psi, the unused 1750 psi from the left side is transmitted through the common shaft and drives the opposite motor to 3750 psi. (For other flow-divider circuits. see the author's book, "Fluid Power Circuits Explained," available through the same outlet for this manual.)*

Fig. 13-21. Symbols for modular flow controls and flow dividers



Most flow control functions are available as modular or sandwich valves that mount between directional control valves and a subplate. ***Figure 13-21*** *shows most of the common configurations presently offered by fluid power suppliers. Although the symbols show non-compensated flow controls, most configurations also are available with pressure-compensated flow controls. Where a needle valve is shown, a flow control with bypass may actually be installed. This is not a problem because there is never a reason for flow reversal.* ***Figure 13-21*** *also shows two modular flow dividers that are available from one supplier. These modules are usually available in all valve sizes up to D08 (3/4-in. ports).*

Flow Control Circuits

To control the speed of an actuator, most designers use flow controls. Air circuits normally need controlled flow because the plant air compressor is greatly oversized for almost any given circuit. Hydraulic circuits usually have a dedicated power source sized to meet the cycle time so flow restrictors are unnecessary.

Flow controls always generate some heat in hydraulic circuits, so consider some other method of controlling actuator speed where possible. The circuit examples in this chapter explain the types of flow-control systems and how to apply them.

Figures 10-1 and 10-2 show symbols for fixed orifices, rudimentary components that will control flow. A fixed orifice can be a simple restriction in a line or a factory-preset control with pressure compensation and a bypass. Their low cost and the fact that they are tamper-proof are two main reasons for using fixed orifices.

Fig. 10-1: Fixed orifice.

Fig. 10-2: Pressure- and temperature-compensated fixed orifice.

Use the needle valve shown in Figure 10-3 when control of fluid flow in both directions is necessary. Add the check valve arrangement shown in Figure 10-4 when a needle valve needs pressure compensation in both directions. These check valves, sometime referred to as bridge rectifiers, force fluid to flow through the needle valve in the same direction regardless of actuator movement. (Remember, pressure compensation only works in one direction of flow.)

Fig. 10-3: Needle valve.

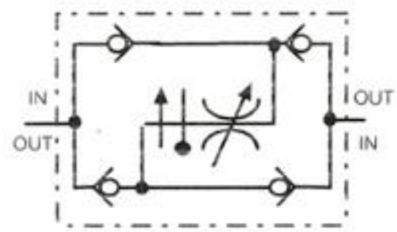


Fig. 10-4: Pressure- and temperature-compensated needle valve.

When talking about flow-control hardware, some manufacturers use different terminology. Normally the term flow control refers to an adjustable needle valve with an integral bypass, as pictured in Figure 10-5. This type of flow control meters flow in one direction and allows free flow in the opposite direction. However, some companies identify the flow control in Figure 10-5 as a throttle valve. These companies say a flow control must have a bypass and be pressure-compensated as shown in Figure 10-6.

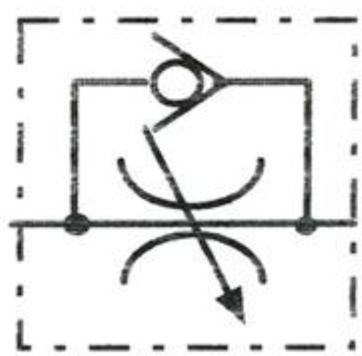


Fig. 10-5: Flow control with

bypass (or throttle valve).

When a hydraulic actuator needs accurate speed control, use a pressure-compensated flow control. System pressure fluctuations or load changes will cause actuator velocity to change. Regardless of the cause of the pressure differences, flow across the orifice will change unless the flow control is pressure compensated. Only use a pressure-compensated valve when very accurate speed control is needed because its cost is as much as six times that of a non-compensated valve.

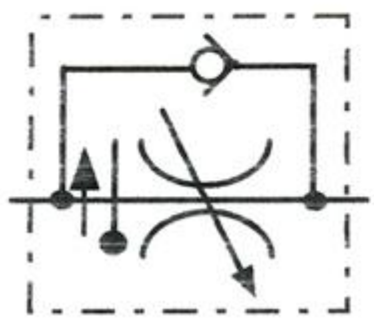


Fig. 10-6: Pressure- and temperature-compensated flow control.

Types of flow-control circuits

There are three types of flow control circuits from which to choose. They are: meter-in, meter-out, and bleed-off (or bypass). Air and hydraulic systems use meter-in and meter-out circuits, while only hydraulic circuits use bleed-off types. Each control has certain advantages in particular situations.

Figure 10-7 shows a meter-in flow-control circuit for a cylinder. Notice that a bypass check valve forces fluid through an adjustable orifice just before it enters the actuator. Figure 10-8 shows the circuit while the cylinder is extending – with the pressures and flows indicated. With a meter-in circuit, fluid enters the actuator at a controlled rate. If the actuator has a resistive load, movement will be smooth and steady. This is because hydraulic fluid is almost incompressible.

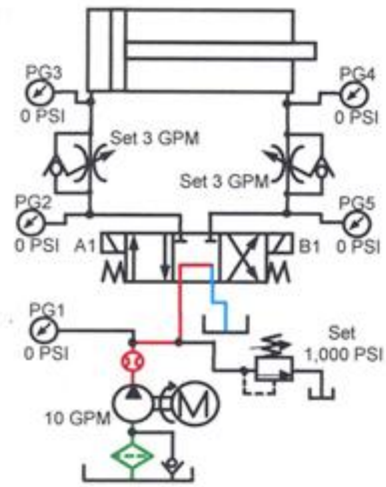


Fig. 10-7: Meter-in flow-control circuit – at rest.

In pneumatic systems, cylinder movement may be jerky because air is compressible. As air flows into a cylinder, as depicted in Figure 10-9, pressure increases slowly until it generates the breakaway force needed to start the load moving. Because the subsequent force needed to keep the load moving is always less than the breakaway force, the air in the cylinder actually expands. The expanding air increases the cylinder speed, causing it to lunge forward. The piston moves faster than the incoming air can fill the cylinder, pressure drops to less than it takes to keep the cylinder moving and it stops. Then pressure starts to build again to overcome breakaway force and the process repeats. This lunging movement can continue to the end of the stroke. A meter-out circuit is the best control to avoid air-cylinder lunging.

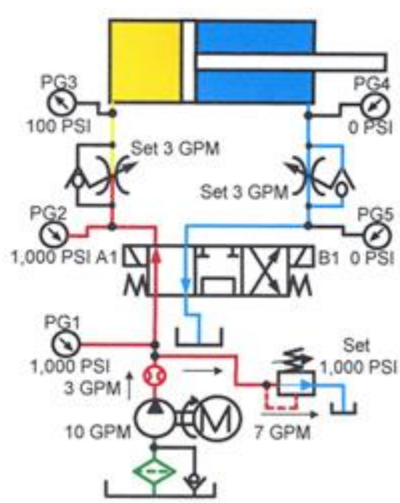


Fig. 10-8: Meter-in flow-

control circuit – with cylinder extending.

Figure 10-7 shows the components in a meter-in flow-control circuit. Notice that a bypass check valve forces fluid through an adjustable orifice just before it enters the actuator.

Figure 10-8 shows an extending hydraulic cylinder and indicates the pressures and flows in various parts of the circuit. With a meter-in circuit, fluid enters the actuator at a controlled rate. If the actuator has a resistive load, movement will be smooth and steady with a hydraulic circuit. This is because oil is almost non-compressible.

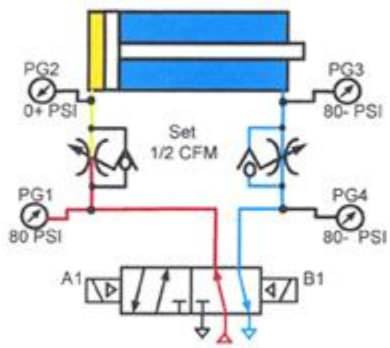


Fig. 10-9: Pneumatic meter-in flow-control circuit – with cylinder extending

---

In the case of an air system, pressure builds slowly and cylinder movement may be jerky. This jerky movement comes from compressibility of the air. As air enters the cylinder, Figure 10-9, pressure builds slowly until it generates the breakaway force to start the piston moving. Because moving force is always less than breakaway force, air in the cylinder expands. The expanding air speeds up cylinder movement, causing it to lunge forward. This increased speed moves the piston faster than the incoming air can fill the space behind it, so pressure drops to less than it takes to keep it moving and the cylinder stops. After the cylinder stops, pressure starts to build again to develop breakaway force and the process repeats. This lunging movement can continue to the end of the stroke. A meter-out circuit is the best control for an air cylinder.

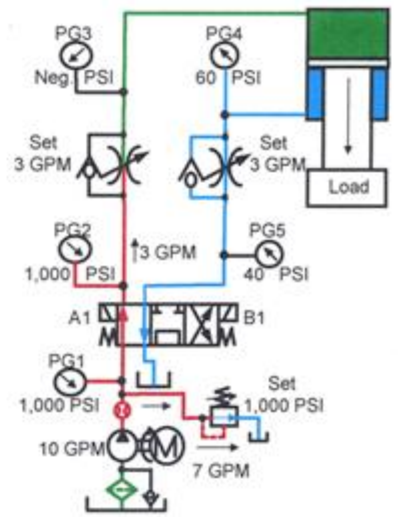


Fig. 10-10: Meter-in flow-control circuit for overrunning load – with cylinder extending.

If the actuator has an overrunning load, such as in Figure 10-10, a meter-in flow control will not work. When the directional valve shifts, the vertical load on the cylinder rod makes it extend. Because fluid cannot enter the cylinder's cap end fast enough, a vacuum void forms there. The cylinder then free falls, regardless of the setting of the meter-in flow adjustment. The pump will continue to supply metered fluid to the cap end of the cylinder and will eventually fill the vacuum void. After the vacuum void fills, the cylinder can produce full force.

3-speed meter-in circuit

The schematic diagram in Figure 10-11 shows a 3-speed, meter-in flow-control circuit using modular valves. Energizing different combinations of solenoids changes cylinder speed at will. To get additional speeds, add more tandem-center directional valves and flow-control modules like station DV01. The limiting factor would be pressure drop through the valves' tandem centers. Using a bar manifold and modular valves eliminates many fittings and possible leak sources. As in all meter-in circuits, the pressure-compensated pump shown here generates less heat than a fixed-volume pump.

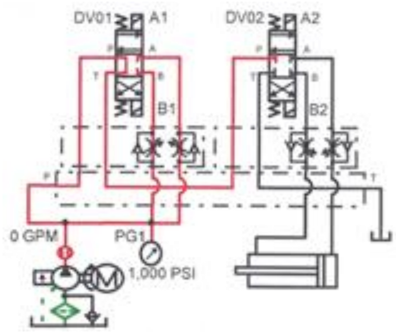


Fig. 10-11: Three -speed meter-in flow-control circuit using modular valves mounted on bar manifold – at rest with pump running.

To extend the cylinder at fast speed, shift the valves as shown in Figure 10-12. Energizing solenoid A2 on directional valve DV02 sends fluid through the meter-in flow-control module directly under it to the cylinder's cap end. This condition is always set for the fastest extension speed. Solenoid A2 stays energized for all extension speeds.

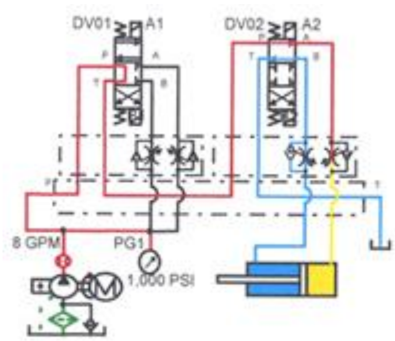


Fig. 10-12: Three -speed meter-in flow-control circuit using modular valves mounted on bar manifold – extending at fast speed.

Energizing solenoid *B1* on directional valve *DV01*, Figure 10-13, sends pump flow through the right-hand flow control in the module underneath it. This will produce a slower speed -- here called *middle speed*.

Either solenoid *A1* or *B1* could produce middle speed, making the opposite solenoid produce *slow speed*. As with fast speed, the cylinder speed is variable, but never faster than fast speed.

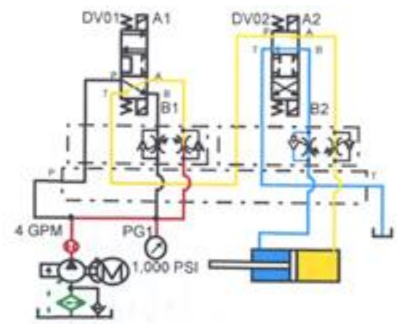


Fig. 10-13: Three -speed meter-in flow-control circuit using modular valves mounted on bar manifold – extending at middle speed.

By actuating solenoid A1 on directional valve DV01, fluid passes through the left-hand flow control in the module underneath it. This will produce a different speed here called slow speed.

The cylinder can retract rapidly or at any of the same slower-flow settings as above. By energizing solenoid B2of directional valve DV02, flow will pass through the opposite meter-in flow control. This means fast-speed retracting can be different from the extending speed. The middle and slow speeds will be at the same flow rate as extension. Cylinder speed during these reduced flows will be somewhat faster due to the decreased rod-end area.

A simple manifold can give multiple speeds inexpensively, while eliminating potential plumbing leaks.

Note: Select a valve for DV01 that can withstand tank-line backpressure.

Speed changes with this meter-in circuit will be smooth because the cylinder can coast while slowing down. (It also is possible that the cylinder could cavitate when slowing down, so an anti-cavitation check valve may be needed.)

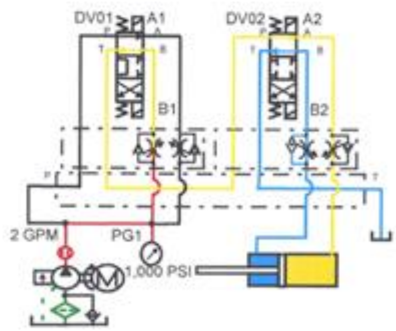


Fig. 10-14: Three -speed meter-in flow-control circuit using modular valves mounted on bar manifold – extending at slow speed.

Meter-in flow control of a running-away load

Figures 10-15 and 10-17 show a running-away load controlled by a meter-in circuit and counterbalance valves. The meter-in flow control works exactly as explained previously, while a counterbalance valve makes cylinder movement resistive. (See Chapter 5 for an explanation of counterbalance valves.) Figure 10-16 pictures a bleed-off flow control circuit that gives the same results as a meter-in circuit -- without most of the heat generation.

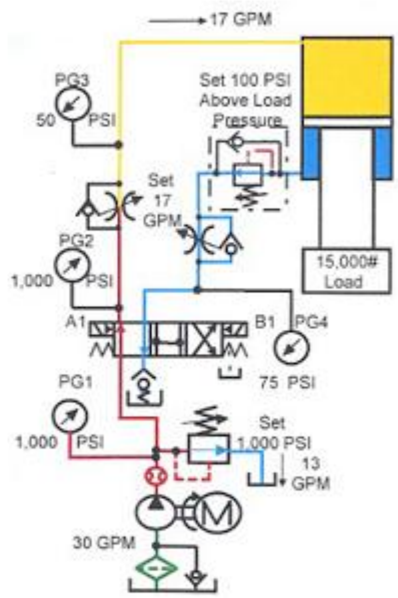


Fig. 10-15: Meter-in flow-

control circuit for vertical cylinder extending with over-running load. Counterbalance valve prevents load from falling.

A meter-in flow control circuit for an over-running load is not the normal design but it may be necessary when the circuit has a pressure switch or a sequence valve.

In any meter-in circuit with a fixed-volume pump, the wasted energy will heat the fluid. In the circuit in Figure 10-15, almost 95% of the power used by the system becomes heat. In this circuit, fluid from the pump enters the cylinder as fast as the meter-in flow control allows. A counterbalance valve at the rod-end port keeps the cylinder from running away as it extends.

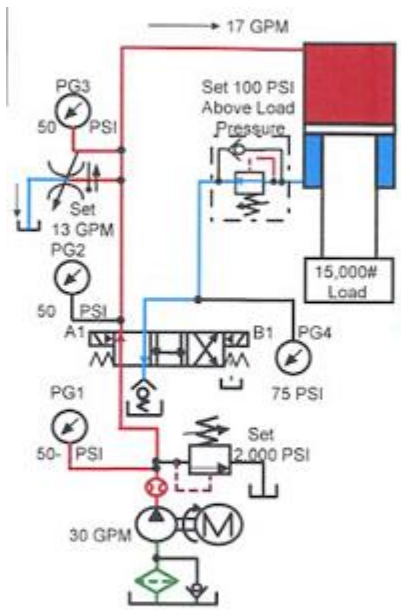


Fig. 10-16: Bleed-off or bypass flow-control circuit for vertical cylinder extending with overrunning load. Counterbalance valve prevents load from falling.

.

To save energy while using a fixed-volume pump, the circuit in Figure 10-16 works well. A bleed-off or bypass flow control greatly reduces the amount of wasted energy. With a bleed-off circuit, excess pump flow goes to tank at the pressure required to move the cylinder. In the circuit in Figure 10-16, pressure would be approximately 50 psi as the cylinder extends. The extension-stroke speed is still infinitely variable, while pressure in the cylinder cap end line never goes higher than that caused by load resistance.

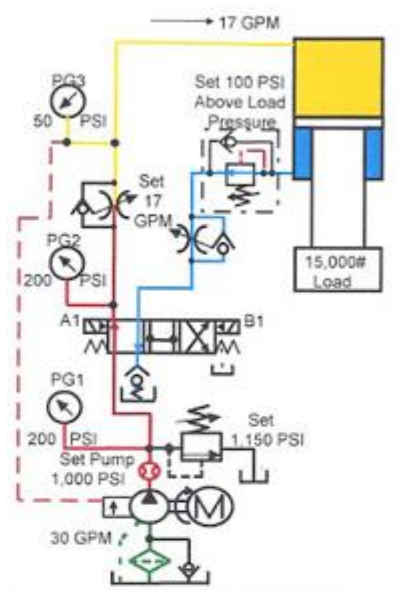


Fig. 10-17: Meter-in flow-control circuit with load-sensing pressure-compensated pump and vertical cylinder extending with over-running load. Counterbalance valve prevents load from falling.

Figure 10-17 shows another way to reduce energy loss and heat generation – using as load-sensing, pressure-compensated pump in conjunction with a meter-in flow-control circuit. A sensing line, teed into the cylinder cap-end line after the meter-in flow control, transmits pressure information to the pump. With a load-sensing, pressure-compensated pump, pressure at the pump outlet stays 150 to 200 psi higher than the load until it tries to go above the compensator's pressure setting. The only energy loss here is the 150- to 200-psi pressure drop across the flow control at the volume set. (Heat generation within a load-sensing pump circuit is explained later in this Chapter.)

When meter-in circuits are necessary

In some cases a meter-in circuit is the only way to control the speed of an actuator -- even for pneumatic devices. Figures 10-18 through 10-21 show several instances requiring meter-in circuits.

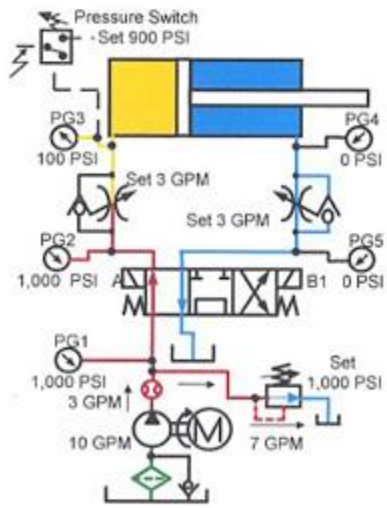


Fig. 10-18: Meter-in flow-control circuit with pressure switch for end-of-stroke indication.

Many machine circuits use pressure switches to indicate when an actuator meets resistance. If pressure in the actuator builds prematurely, the machine cycle gets out of phase. With the meter-in circuit shown in Figure 10-18, pressure in the cylinder will be just enough to move the cylinder and its attachments until it reaches the load. With a meter-out circuit, pressure in the cylinder cap end would build as soon as the directional valve shifts, tripping the pressure switch long before the cylinder contacts the load.

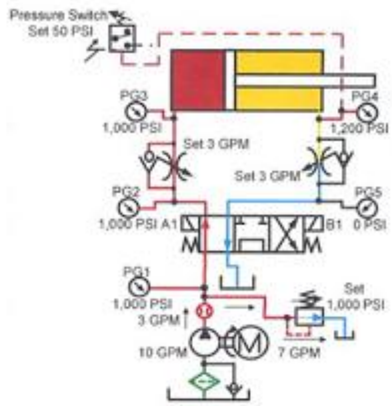


Fig. 10-19: Meter-out flow-control circuit with pressure switch for end-of-stroke indication.

However, it is possible to use a pressure switch with a meter-out flow control circuit. In Figure 10-19, notice that the pressure switch is on the cap-end line to the cylinder. It passes a signal when the cylinder cannot extend farther. Notice also that the pressure switch setting is very low (50 psi). While the cylinder is moving, oil flowing from the cylinder head-end port remains pressurized by the meter-out flow control. When the cylinder contacts the load, pressure in the head-end port drops, actuating the pressure switch and sending a signal. (Use a normally open, 3-way, pilot operated, spring-return directional valve in place of the pressure switch to produce an air or hydraulic pilot signal.)

The pressure-decaying circuit in Figure 10-19 works well in pneumatic circuits because metering air flow out provides good control. Several companies furnish air logic elements designed specifically for this type of circuit.

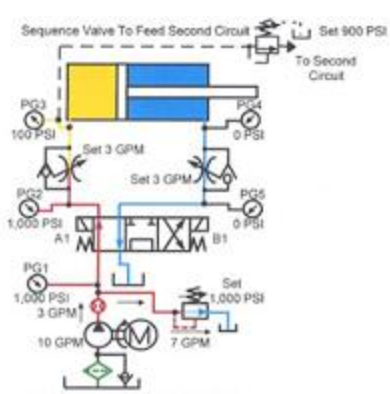


Fig. 10-20. Meter-in flow-

control circuit with sequence valve for end-of-stroke indication.

Sequence valves often are used to start a second actuator after a cylinder meets resistance and builds tonnage. With the meter-in flow control shown in Figure 10-20, pressure in the cylinder cap end increases when -- but not before -- the cylinder contacts the work. Pressure at the sequence valve's inlet stays lower than its spring setting while the actuator is moving. (With a meter-out circuit, pressure in the cylinder's cap end would go to system pressure when the directional valve shifts. Because of this, flow to the secondary circuit would take place prematurely.

The vertical single-acting, weight-return cylinder shown in Figure 10-21 needs meter-in control as it extends. This will be the case even if it is a pneumatic cylinder where meter-out control works best. For a different retraction speed, use a second meter-out flow control (as shown in Figure 10-21).

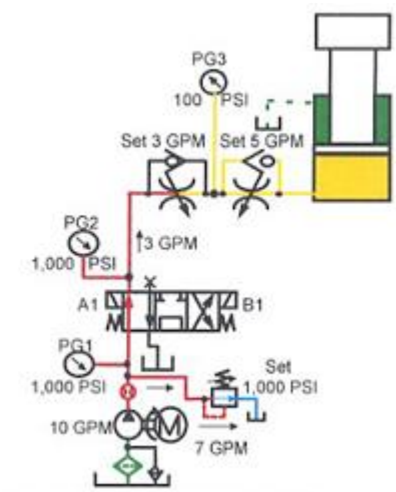


Fig. 10-21. Meter-in flow-control circuit for a single-acting cylinder extending.

Note: A pressure switch or a sequence valve will only indicate that pressure has reached a certain level. What caused the pressure build-up is unimportant. If the actuator positively has to be at a certain position before the next function starts, do not use a pressure-sensing device.

Always use a limit switch. (Use a pressure switch or sequence valve in series with the limit switch if cylinder position and force are both important.)

Action of a meter-in air circuit with a varying load

When using a meter-in circuit on an air cylinder with a variable load, movement will not be consistent. Depending on the required range of forces, movement may be smooth, the cylinder may over-speed, or it may even stop. In Figure 10-22, the cylinder is moving smoothly at a pressure difference of 30 psi. (It takes 30 psi in the bore size being used to generate the force to move the load.) If the load remains constant, the cylinder can -- and probably will -- advance smoothly.

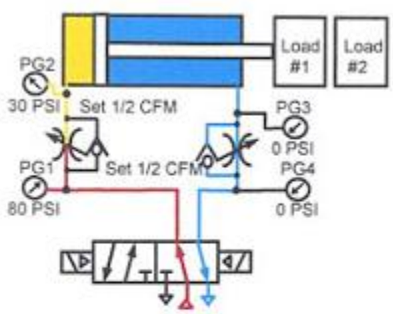


Fig. 10-22. Meter-in pneumatic flow-control circuit for loaded cylinder extending slowly and smoothly toward a second load.

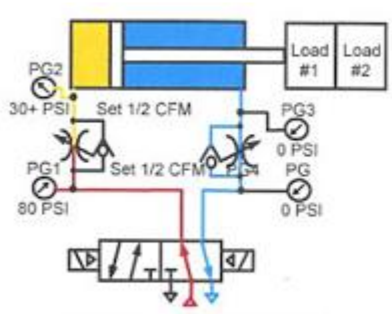


Fig. 10-23. Meter-in pneumatic flow-control circuit as cylinder contacts second equal load. Cylinder stops

while cap-end pressure builds
to produce force required.

When the load doubles, as in Figure 10-23, 30 psi is not enough to keep the cylinder moving. At this point the cylinder will stop until pressure in the cap end reaches 60 psi. (The meter-in flow setting determines how long this takes.)

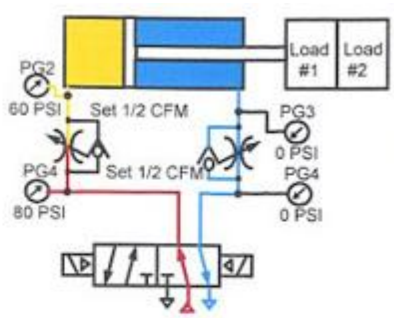


Fig. 10-24. Meter-in pneumatic flow-control circuit moving both loads after cap-end pressure reaches load requirement.

Once pressure in the cylinder cap end reaches 60 psi, Figure 10-24, the cylinder starts moving again. If the higher load stays constant, movement is steady.

When the second load is reduced, as diagrammed in Figure 10-25, 60 psi in the cap end is more pressure than needed. This high pressure will cause the cylinder to lunge forward and, as a result, pressure in the cap end will start to decay. The amount of lunge is in direct proportion to the total volume of air in the cylinder's cap end and the piping leading to it. Next, as Figure 10-26 shows, once decompression reaches 30 psi, the cylinder slows to its original speed.

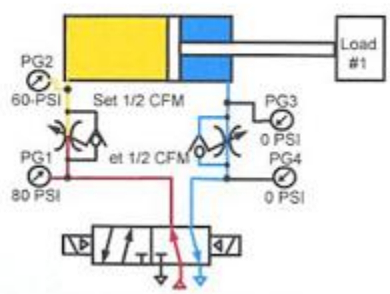


Fig. 10-25. Meter-in pneumatic flow-control circuit as second equal load drops off. Cylinder lunges forward as pressure in cap end decreases.

If this stop/lunge/over-speed problem is intolerable and air is the required power source, add some method of oil control to the circuit. (See Chapter 3 on air-oil circuits.)

Meter-out flow controls

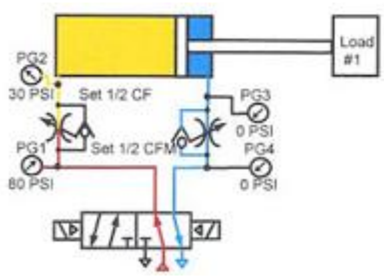


Fig. 10-26. Meter-in pneumatic flow-control circuit as cylinder again moves forward slowly and smoothly after pressure decreases to that required to move the single load.

Meter-out controls restrict fluid leaving the cylinder to retard the cylinder's movement. This type of flow-control circuit works for any type of load -- and works best with air-operated devices. Figure 10-27 shows a meter-out flow-control circuit in the at rest condition.

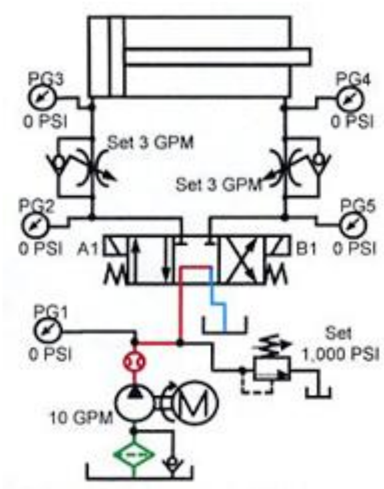


Fig. 10-27. Meter-out flow-control circuit – at rest with pump running.

In Figure 10-28, the directional valve has shifted and the cylinder starts to extend. Fluid in the cap end of the cylinder is at system pressure and the relief valve is dumping excess pump flow to tank. Pressure at the head end of the cylinder will be at system pressure or higher according to the rod size and force required to move the load. The action of meter-out flow controls is smooth and steady in hydraulic circuits.

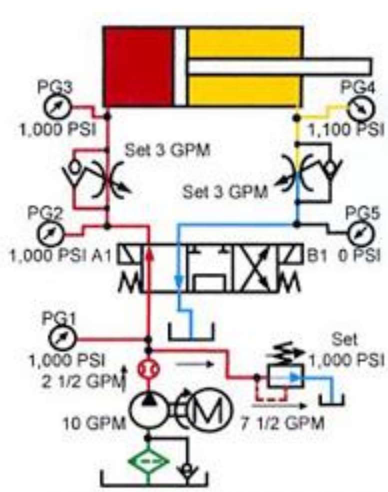


Fig. 10-28. Meter-out flow-control circuit – with cylinder extending.

Figure 10-29 shows the pressure pattern of an air cylinder while it is extending. By restricting flow out of the cylinder, the action will be smooth when the load remains constant. (Figures 10-35 through 10-39 show the action of an air cylinder that is moving a changing load.)

In Figure 10-30, meter-out flow controls are controlling the load on a down-acting vertical cylinder. This over-running load moves steadily because fluid flow leaving the cylinder is restricted. The meter-out circuit keeps the load from running away, but depending on the load and the rod size, there could be excessive pressure in the cylinder's head end. Notice that the rod-end pressure is 3000 psi when extending. This is because the rod is oversize (2:1) and the load is heavy. At a relief valve setting of 3000 psi, this head end pressure could be as high as 7000 psi.

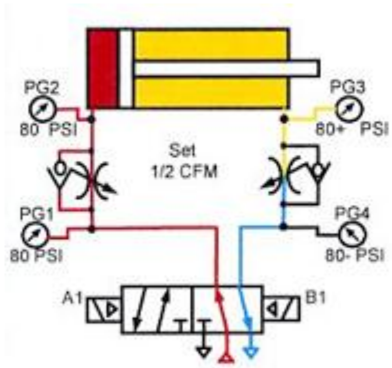


Fig. 10-29. Pneumatic meter-out flow-control circuit – with cylinder extending.

When using a meter-out system with a running-away load, check load-induced pressure and hydraulic-force-induced pressure at the rod end. This pressure can be much higher than components' rating -- even when the relief valve setting is well below maximum rated pressure.

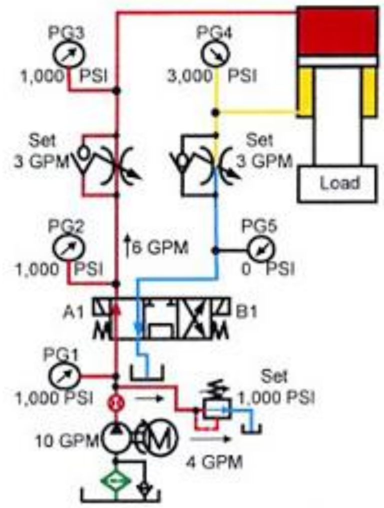


Fig. 10-30. Meter-out flow-control circuit – cylinder extending with over-running load.

Three-speed meter-out circuit

The schematic diagram in Figure 10-31 shows a 3-speed meter-out flow-control circuit using modular valves. Energizing different combinations of solenoids changes speeds at will. To get additional speeds, add more tandem-center directional valves and flow-control modules like station DV02. The limiting factor would be pressure drop through the valves' tandem centers. Using a bar manifold and modular valves eliminates many fittings and possible leak sources. As in all meter-out circuits, the pressure-compensated pump shown here generates less heat than a fixed-volume pump.

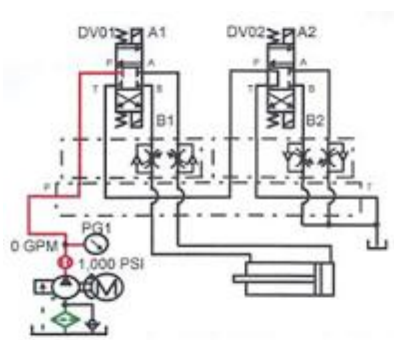


Fig. 10-31. Three-speed

meter-out flow-control circuit using modular valves on bar manifold – at rest with pump running.

Figure 10-32 shows the cylinder extending at fast speed. Solenoid A1 of directional valveDV01 shifts and fluid from the cylinder passes through the meter-out flow-control module directly under it. This flow path will always produce the fastest speed.

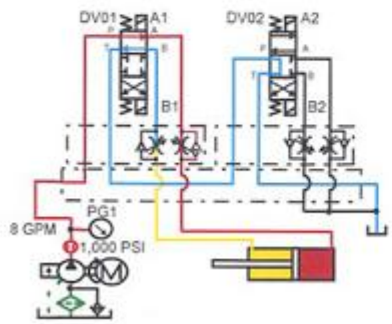


Fig. 10-32. Three -speed meter-out flow-control circuit using modular valves on bar manifold – cylinder extending at fast speed.

By energizing solenoid B2 on directional valve DV02, as in Figure 10-33, return flow from the cylinder passes through the left flow control in the module underneath it. This will produce a slower speed -- here called middle speed. Either solenoid A2 or B2 could be assigned to middle speed (making the opposite solenoid produce slow speed). As with fast speed, the rate of cylinder movement is variable, except it can never be faster than fast speed.

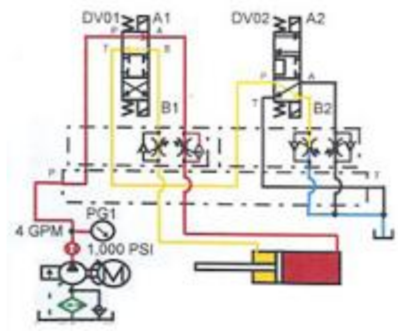


Fig. 10-33. Three -speed meter-out flow-control circuit using modular valves on bar manifold -- cylinder extending at middle speed.

By actuating solenoid A2 on directional valve DV02, Figure 10-34, oil from the cylinder passes through the right flow control in the modular valve underneath it. This will be a different speed --here called slow speed.

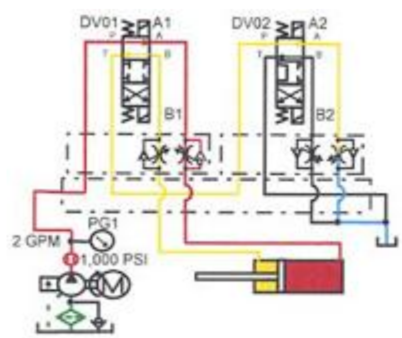


Fig. 10-34. Three -speed meter-out flow-control circuit using modular valves on bar manifold -- cylinder extending at slow speed.

The cylinder can retract at fast speed or at any of the same slower flow settings as above. By energizing solenoid B2 of directional valve DV01, flow will pass through the opposite meter-out flow control. This means fast speed can be different when the cylinder extends. The middle and *slow speeds will be at the same flow rate as extend. Cylinder speed during these reduced flows will be somewhat slower due to increased area on the cap end.*

When changing from a faster to a slower speed, the action will be abrupt. This is because the meter-out circuits control the flow exiting the cylinder.

Note: A simple manifold can set multiple speeds inexpensively while eliminating plumbing leaks. Also, be sure to use a valve for DV01 that can withstand tank-line backpressure.

Meter-out pneumatic circuit with a variable load

When using a meter-out circuit on an air cylinder with a changing load, movement will not be consistent. According to the amount of force change required, movement can range from smooth, to stop, to lunging. (Note that in Figures 10-35 through 10-39 there is no allowance made for rod differential.)

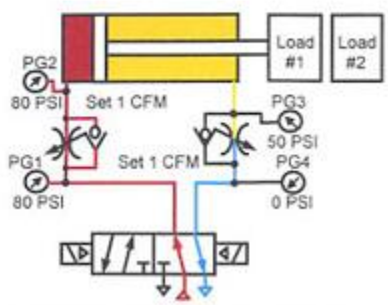


Fig. 10-35. Pneumatic meter-out flow-control circuit – with loaded cylinder extending slowly and smoothly toward second equal load.

In Figure 10-35, a loaded air cylinder is stroking smoothly at a pressure difference of 30 psi between its cap and head ends. (At the bore size for this example, it takes a 30-psi difference to generate enough force to move the load). If the load remains constant, the cylinder usually will advance smoothly.

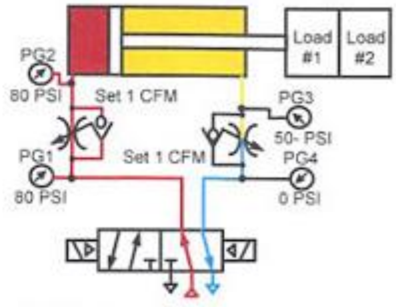


Fig. 10-36. Pneumatic meter-out flow-control circuit – as cylinder contacts second load and stops while rod-end pressure drops.

But when the load is doubled, as shown in Figure 10-36, a difference of 30 psi across the piston is not enough to keep the cylinder moving; the cylinder stops while pressure in the rod end decreases to about 20 psi. The speed of the pressure decay is in direct proportion to how quickly air discharges through the flow control. Once pressure in the cylinder rod end reaches 20 psi, Figure 10-37, the cylinder will start moving again. If the additional load stays constant, movement is smooth and steady again.

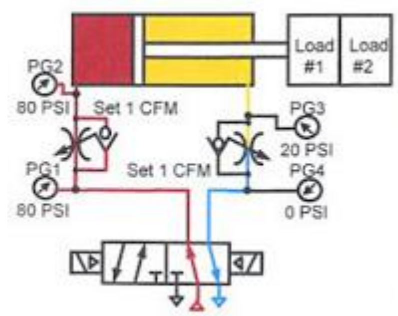


Fig. 10-37. Pneumatic meter-out flow-control circuit – with cylinder moving both loads after rod-end pressure has dropped.

When the second load is removed, as in Figure 10-38, 20 psi in the rod end is less pressure than needed to hold the piston back. At this time the cylinder will lunge forward until the pressure in *the rod end increases to about 50 psi. The amount of lunge is in direct relation to the volume of air in the cylinder rod end and the piping to it. As Figure 10-39 shows, once air in the rod end again compresses to about 50 psi, the cylinder returns to normal speed.*

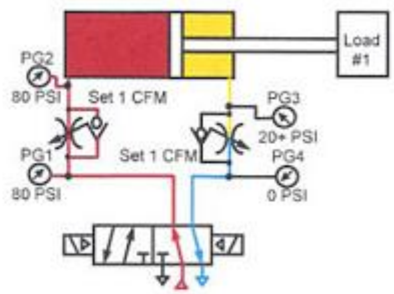


Fig. 10-38. Pneumatic meter-out flow-control circuit – cylinder lunging forward after second load is removed. Pressure in rod end is increasing to level that will hold cylinder back.

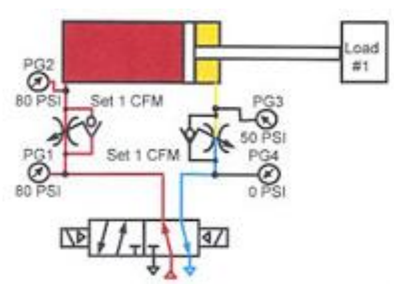


Fig. 10-39. Pneumatic meter-out flow-control circuit – after return to original conditions, cylinder extends slowly and smoothly with single load.

If this stop-and-lunge problem is intolerable and air is the power source, add a method of oil control to the circuit. (See Chapter 3 on air-oil circuits for ways to overcome the problem.)

Bleed-off or bypass flow controls

Figure 10-40 shows a bleed-off or bypass flow-control circuit in the at-rest condition. This type of flow control circuit bleeds off excess fluid to tank. A bleed-off circuit works best in hydraulic circuits using fixed-volume pumps. And a bleed-off circuit only works with multiple actuators if they operate one at a time.

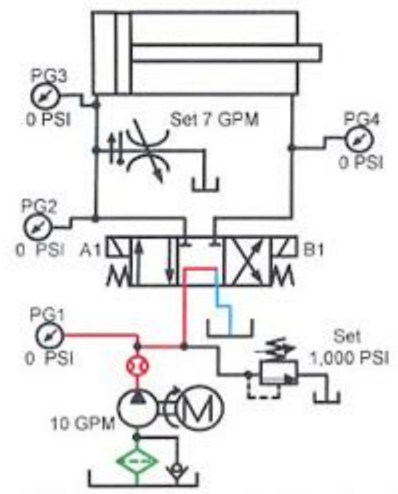


Fig. 10.40. Bleed-off or bypass flow-control circuit -- at rest with pump running.

When oil passes to tank through a pressure-compensated flow control, cylinder movement will slow while system pressure only climbs high enough to move the load. (While this arrangement wastes energy, the amount is minimal.)

The cylinder in Figure 10-41 is extending at 3 gpm while 7 gpm passes to tank through the pressure-compensated needle valve. Because the resistance of the cylinder and load is only 100 psi, the energy needed is very low. The 3 gpm flowing to the cylinder generates no heat because it is doing useful work. The 7 gpm going to tank at a 100-psi pressure drop is the only wasted energy. If the cylinder were to contact a load that requires 300 psi then the whole system would climb to 300 psi. Energy losses would increase, but would still be much less than those in a meter-in or meter-out system.

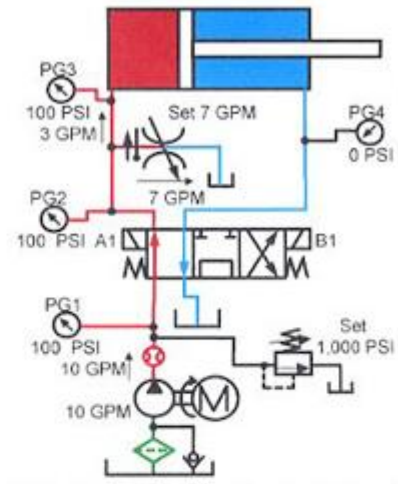


Fig. 10.41. Bleed-off or bypass flow-control circuit – with cylinder extending

---

Even with a pressure-compensated flow control valve, cylinder speed will change slightly as pressure increases. This is because pump flow decreases slightly when pressure increases. The flow control still passes 7 gpm. (All pump inefficiency losses reduce cylinder speed.) When the cylinder bottoms out, pressure increases until the relief valve opens. At this time all input energy generates heat. Note that this only happens when the cylinder must maintain force while stalled.

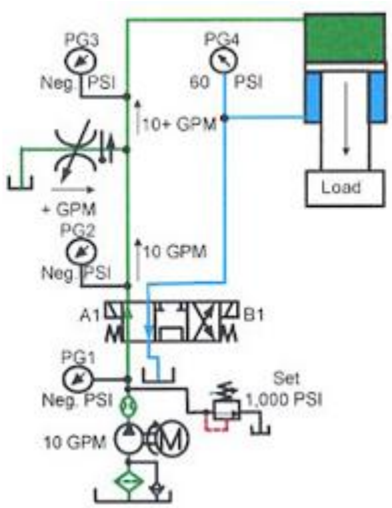


Fig. 10.42. Bleed-off or bypass flow-control circuit – vertical cylinder extending with over-

running load.

As with meter-in circuits, bleed-off circuits do not work with a running-away load. Figure 10-42 shows the cylinder extending rapidly and fully when the directional valve shifts. (When using a bleed-off circuit with a running-away load, use the counterbalance circuit shown in Figure 10-16. With a counterbalance valve creating resistance, oil entering the cylinder sets its speed as with any resistive load.)

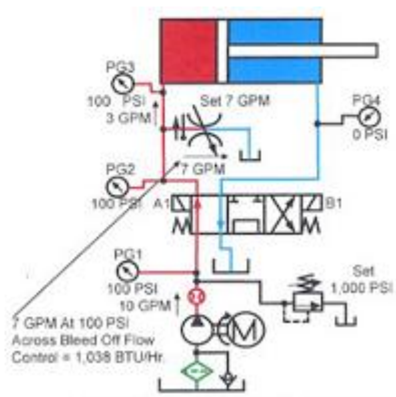


Fig. 10.43. Bleed-off or bypass flow-control circuit – with energy losses identified.

Heat generation can be minimal if the cylinder force is low while extending and retracting. Figure 10-43 shows only 1038 BTU/hr loss -- even with 7 gpm being bled to tank. Normally, use a bleed-off circuit where only a small amount of fluid goes to tank to fine tune actuator speed.

A bleed-off circuit requires a pressure-compensated flow control to keep the cylinder from slowing as pressure increases. Without a pressure-compensated flow control, cylinder speed slows as load increases and speeds up when load decreases. With a bleed-off flow-control circuit, turn the flow control clockwise to increase speed and counter-clockwise to decrease speed.

Three-speed bleed-off circuit

Figure 10-44 shows a 3-speed bleed-off circuit, controlled by energizing different solenoids. To get additional speeds, add more blocked-center directional valves and flow modules like DV01. The limiting factor would be the number of stations on the manifold. (Using a bar manifold and *modular valves eliminates many fittings and possible leaks.) Use this circuit with fixed-volume hydraulic pumps only.*

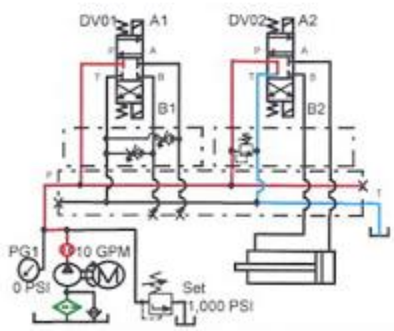


Fig. 10.44. Three-speed bleed-off flow-control circuit using modular valves on bar manifold – at rest with pump running.

To extend the cylinder at fast speed, as in Figure 10-45, shift solenoid A2 on directional valve DV02. All pump output flows to the cylinder to produce fast speed. This condition always is the highest speed. Adding a bleed-off modular flow control under DV02 will allow adjustment of fast speed. Use of a primary adjustment at DV02 would slow both other speeds.

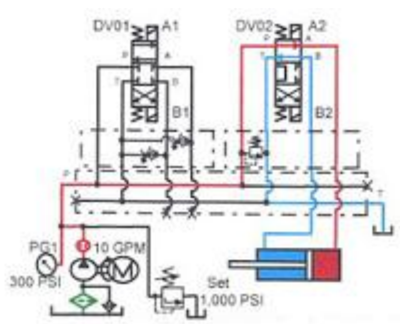


Fig. 10.45. Three-speed bleed-off flow-control circuit using modular valves on bar manifold – cylinder extending at full speed.

Energizing solenoid B1 on directional valve DV01, as in Figure 10-46, directs a portion of pump flow to tank through the module underneath it. The cylinder speed varies, but it can only be slower than fast speed. This is middle speed. Either solenoid A1 or B1 could produce middle speed while the opposite solenoid would produce slow speed.

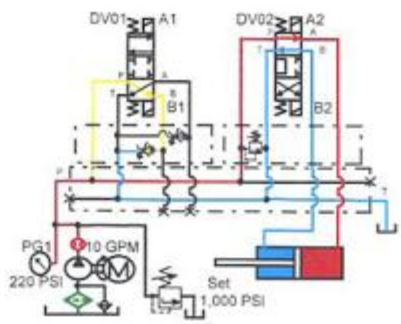


Fig. 10.46. Three-speed bleed-off flow-control circuit using modular valves on bar manifold -- cylinder extending at middle speed.

Actuating solenoid A1 on directional valve DV01, as in Figure 10-47, bleeds oil to tank through the other flow control, resulting in slow speed.

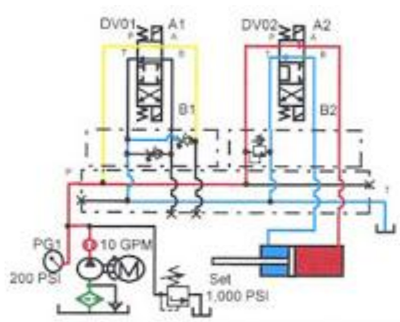


Fig. 10.47. Three-speed bleed-off flow-control circuit using modular valves on bar manifold -- cylinder extending at slow speed.

The cylinder can retract at fast speed or at any of the same slower flow settings as above. Energizing solenoid B2 on directional valve DV02 sends pump flow through the directional valve directly to the cylinder for fast speed. The middle and slow speeds give the same flow rate as extend. Cylinder speed during these reduced flows is somewhat faster due to the decreased area on the rod end.

Note: Speed change with a bleed-off circuit is very smooth. The cylinder decelerates smoothly, although a slight jerk or shock may be evident when changing from slow to a faster speed.

Different locations for flow controls

Figures 10-48 through 10-51 suggest other possible locations for flow-control devices in circuits. Figure 10-48 shows a meter-in needle valve in the pump line. At first, it appears this arrangement results in equal speed in both directions of travel. While it does slow the stroke in both directions, the speeds are not identical with a single-rod cylinder. As the cylinder advances, metered oil flows to the cap-end area (4.91-in.2 for a 2.50-in. bore). With a 1.375-in. diameter rod, the head-end area would only be 3.43 in.2 Thus, with the same flow entering the different areas, retraction speed would be faster.

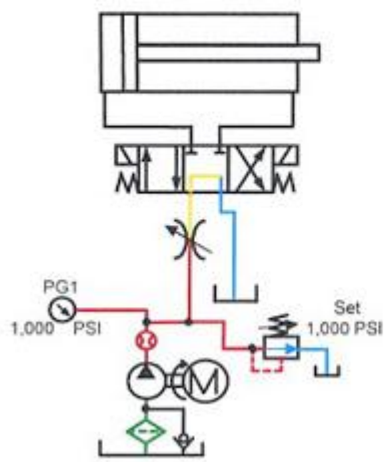


Fig. 10.48. Meter-in flow-control circuit with needle valve in pump line.

Note that all of the same precautions given previously for meter-in circuits apply to this single meter-in needle valve as well. In addition, some solenoid valves cannot stand backpressure on the tank line, so use caution when applying this circuit.

The circuit in Figure 10-49 locates a single meter-out needle valve in the tank line. Again extension and retraction speeds appear to be identical. But as with the tank-line meter-in circuit, piston area differences when extending and retracting result in different speeds. The single meter-out flow-control circuit is faster on the extension stroke with a single-rod cylinder.

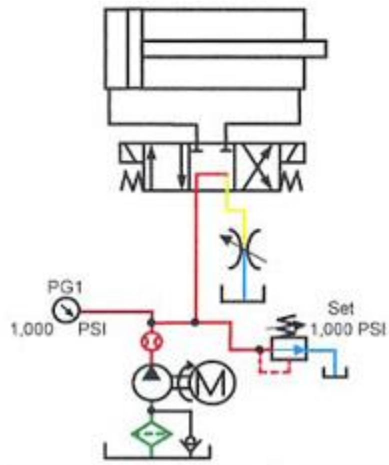


Fig. 10.49. Meter-in flow-control circuit with needle valve in tank line.

The pump-line bleed-off circuit diagrammed in Figure 10-50 is identical in action to the meter-in circuit in Figure10-48. If a fixed-volume pump supplied both circuits, the bleed-off circuit would generate less heat.

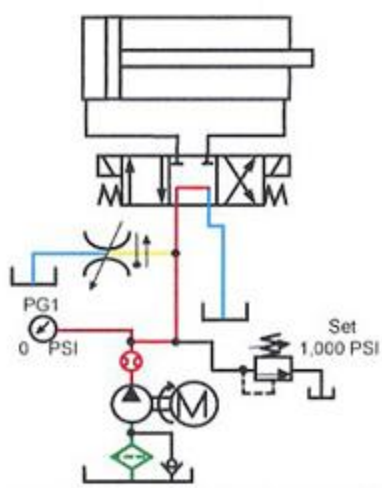


Fig. 10-50. Bleed-off flow control circuit with pressure-

compensated needle valve teed into pump line.

To get identical speed in both directions of cylinder travel, use the circuit in Figure 10-51. Locating a needle valve in the rod- or cap-end line controls the same cylinder area extending and retracting. This circuit meters in flow during one direction of travel and meters flow out in the opposite direction.

Single-rod cylinders have different areas on the two faces of the piston so pressure drop across the needle valve is different when extending and retracting. The circuit in Figure 10-51 has a higher pressure drop across the needle valve orifice when the cylinder is extending. The higher pressure drop has no effect on the pressure-compensated needle valve in this circuit due to the rectifier check valve shown. The rectifier check valve allows oil to flow through the needle valve in the same direction when the cylinder extends or retracts. Cylinder speed will stay the same with this setup, even though pressure drop differs during extension and retraction.

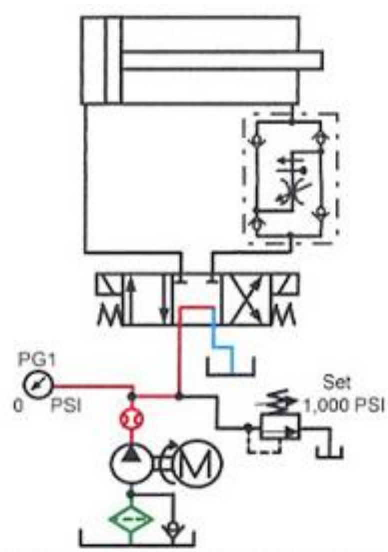


Fig. 10-51. Meter-in/meter-out flow-control circuit with pressure-compensated needle valve and rectifier on rod-end cylinder port.

Heat generation in hydraulic flow-control circuits

Many of the circuit examples in this chapter use fixed-volume pumps. Fixed-volume pumps always generate heat when used with any flow-control circuit. Also, flow controls always generate some heat regardless of the type pump used. Figures 10-52 through 10-55 show meter-in flow-control circuits and explain how different type pumps affect heat generation.

Checking horsepower loss is one way to determine the amount of heat generated by a circuit. To figure horsepower in a hydraulic system use the formula:
(horsepower) = 0.000583 (pressure in psi) (flow in gpm).
Multiply the horsepower by 2545 to calculate the amount of heat produced in British Thermal Units per hour.

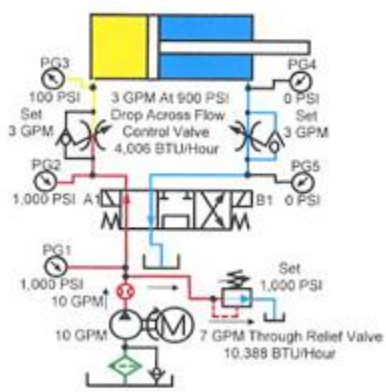


Fig. 10-52. Meter-in flow-control circuit with fixed-volume pump – cylinder extending.

In the circuit in Figure 10-52, 7 gpm flows across the relief valve at 1000 psi. When this oil reaches system pressure, it flows to tank without doing useful work. If you multiply 0.000583 (7 gpm)(1000 psi)(2545 BTU/hr), the heat loss comes to 10,386 BTU/hr. Add to this the heat loss due to the 3 gpm at 900-psi pressure drop passing through the flow control (4006 BTU/hr). Thus, when the cylinder is moving under the conditions shown in Figure 10-52, more than 14,000 BTU/hr of energy turns into heat. The maximum heat that the unit could produce is 0.000583(10 gpm)(1000 psi)(2545 BTU/hr) or 14,837 BTU/hr. Therefore, approximately 97% of the system's energy is wasted heating the fluid.

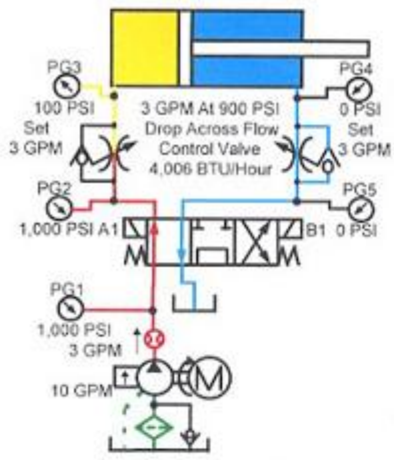


Fig. 10-53. Meter-in flow-control circuit with pressure-compensated pump – cylinder extending.

The pressure-compensated pump in Figure 10-53 produces only the flow needed, so there is no oil going over a relief valve. The only heat loss in this circuit is the 900 psi across the flow-control valve. There are 10,000-BTU/hr fewer entering the system just by changing the type of pump.

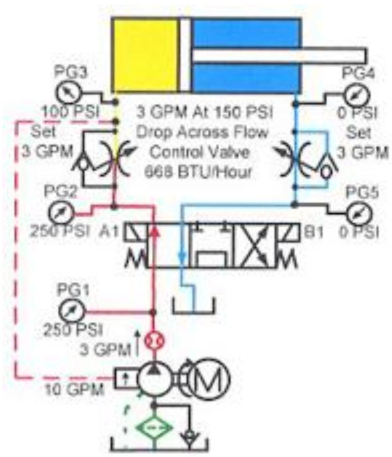


Fig. 10-54. Meter-in flow-control circuit with load-sensing, pressure-compensated pump – cylinder extending.

To cut even more heat from the system, the circuit in Figure 10-54 uses a load-sensing pressure-compensated pump. This pump has a sensing line that monitors the pressure required to move the cylinder, then sets the compensator 150 to 200 psi higher. With a 100-psi cylinder requirement, the pump would operate at approximately 250 psi. This low pressure drop across the meter-in flow control generates a heat loss of just 668 BTU/hr.

To sense the load at both ends of the cylinder, or if there is more than one cylinder to control, the sensing lines come back to the pump through check valves. These check valves allow the pump to see the system's highest pressure requirement and set the pump pressure 150 to 200 psi above it.

When using a load-sensing pump, always use a meter-in flow-control circuit. A meter-out circuit shows pressure at both ends of the cylinder all the time it is moving. With pressure at the load-sensing port, the pump would go to compensator setting and stay there. (For more information on load-sensing pumps, look in Chapter15.)

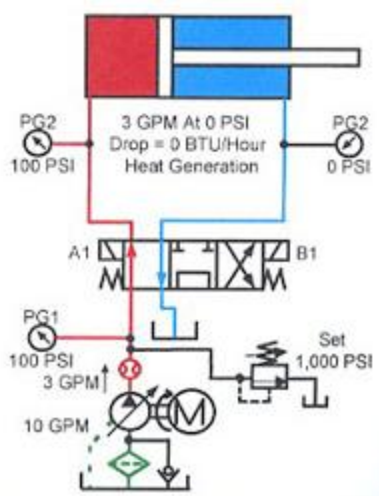


Fig. 10-55. Meter-in flow-control circuit with variable-volume pump – cylinder extending.

Heat generation for the circuit in Figure 10-55 is the lowest possible when variable speed is necessary. In this circuit, a variable-volume pump set for 3-gpm flow replaces the flow controls. Because the cylinder uses all the flow produced, system pressure only goes to the 100 psi required to move the cylinder. There is no excess horsepower, so no heat is generated. (In actual practice, pump inefficiency, pressure drop in the lines, and friction between parts generate some *heat. These losses may cause the system temperature to rise 5 to 15° above ambient temperature.)*

When practical, a variable-volume pump provides the best way to control actuator speed. Variable-volume pumps may require some electronic controls if there is more than one cylinder, but lower operating cost for the life of the machine quickly offsets this one-time first cost.

Motor-type flow-divider speed control

There are some ways to use a fixed-volume pump and motor-type flow dividers to change speeds with minimal heat generation as shown in Figures 10-56 through 10-63. These circuits will only give fixed preset speeds without changing hardware.

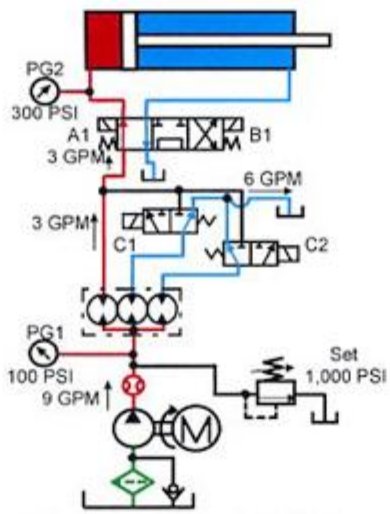


Fig. 10-56. Meter-in flow-control circuit with motor-type flow divider (to minimize heat) – cylinder extending at slow speed.

Figure 10-56 shows a 3-speed flow-control circuit using a motor-type flow divider. Here the cylinder is extending at slow speed. With the circuit set up as shown, it defaults to slow speed. Notice that there are no flow controls. To split pump flow evenly and reduce energy loss, use a motor-type flow divider at its outlet. Each outlet of the flow divider will put out about 3 gpm.

In Figure 10-56 the cylinder is receiving 3 gpm of oil and requires a pressure of 300 psi to move. Notice the pump pressure reads 100 psi. This will happen because the flow divider is taking in 9 gpm, but using only 3 gpm to do work. The other two 3-gpm flows go back to tank at 0 psi. While *it appears these other flows waste energy, they are actually transferring their energy through the motor flow divider to the left-hand motor. The left-hand motor becomes a pump with a 100-psi inlet and two motors driving it to 300 psi. In flow-divider circuits, the average of the sum of the outlets always will be the inlet pressure. In this case: (300 psi) + (0 psi) + (0 psi) = (300 psi)/3 = (100 psi) With this system, speed slows but energy loss is only the inefficiency of the parts used.*

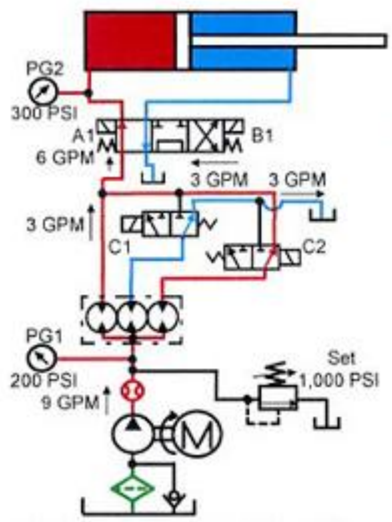


Fig. 10-57. Meter-in flow-control circuit with motor-type flow divider (to minimize heat) – cylinder extending at middle speed.

To get mid speed, the directional valves shift as shown in Figure 10-57. By energizing solenoid C2 on the right-hand 3-way valve, an extra 3 gpm goes to the cylinder to produce mid speed. Notice that the pump pressure goes to 200 psi as the cylinder speed doubles. Still there is only hardware inefficiency to waste energy, so the system runs cool.

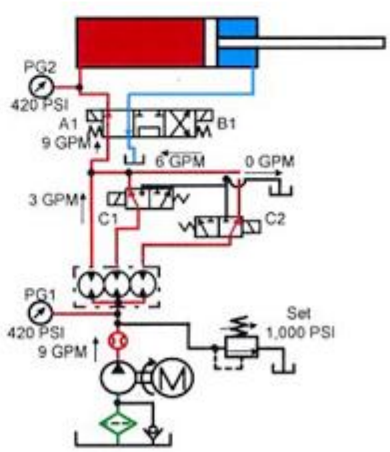


Fig. 10-58. Meter-in flow-control circuit with motor-type flow divider (to minimize heat) – cylinder extending at fast speed.

To make the cylinder extend at fast speed, shift the directional valves as shown in Figure 10-58. By energizing solenoids C1 and C2, both 3-way valves shift to send all pump flow to the cylinder. Because the cylinder is at fast speed, pump and cylinder pressure are the same.

To retract the cylinder at fast speed, shift solenoids B1, C1, and C2 as shown in Figure 10-59. Energizing one or more solenoids in the retract mode gives different speeds that are nearly the same as when extending. (If the flow divider had more and/or unequal size motors, selection of a combination of speeds by selecting different flow outputs is possible.) Notice that this circuit is tamperproof. To change the preset speeds, the flow divider and/or pump must be changed.

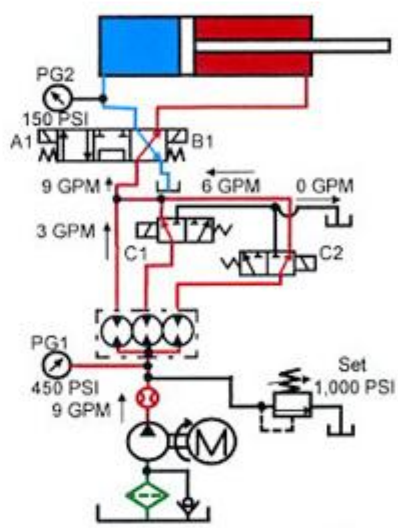


Fig. 10-59. Meter-in flow-control circuit with motor-type flow divider (to minimize heat) – cylinder retracting at fast speed.

Any flow-divider circuit will intensify pressure. If the cylinder in Figure 10-56 stalled, the pressure would continue to climb. When it reached the relief valve setting, pressure at the cylinder would be 3000 psi. A second pressure-relief valve installed between the flow divider and the pump port of the main directional valve could be set at a safer pressure in case the cylinder stalls.

Another motor-type flow-divider speed control

Figures 10-60 through 10-63 show a different type of motor flow-divider circuit for variable speed. This circuit uses a smaller pump, electric motor, and tank to give the same speed but less force at high speed. Notice there is a 3-gpm pump feeding one section of the flow divider. As this section of the flow divider turns, the other two sections turn and pump fluid directly from the tank. In Figure 10-60 the two right-hand sections of the flow divider are only circulating oil. All pump flow is going to the cylinder, which is operating at slow speed. In this condition, the cylinder is capable of generating its highest tonnage. Notice the cylinder requires 300 psi to move it and the pump is showing 300 psi.

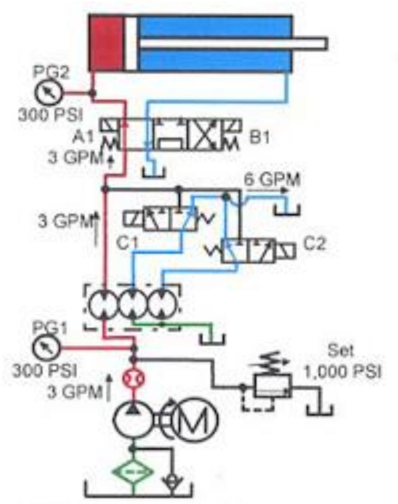


Fig. 10-60. Meter-in flow-control circuit with motor-type flow divider (to minimize heat) – cylinder extending at slow speed.

The cylinder speeds up when solenoid C2 on the left-hand 3-way valve is energized, as in Figure 10-61. Now, one flow-divider section sends its oil to the cylinder along with pump flow. The cylinder goes to mid speed and pump pressure climbs to 600 psi.

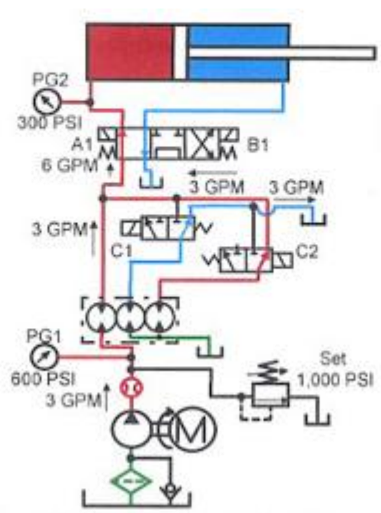


Fig. 10-61. Meter-in flow-control circuit with motor-type flow divider (to minimize heat)

– cylinder extending at middle speed.

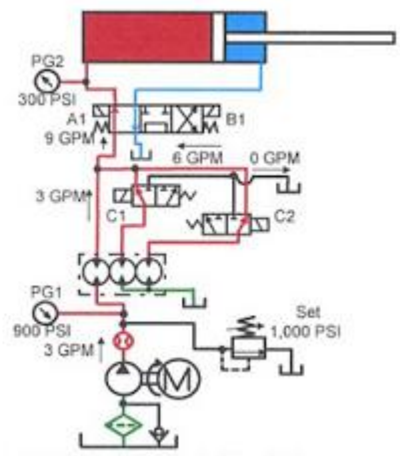


Fig. 10-62. Meter-in flow-control circuit with motor-type flow divider (to minimize heat) – cylinder extending at fast speed.

To get full speed from the cylinder, energize solenoid C1 on the right hand 3-way valve, as shown in Figure10-63. Now all three sections of the flow divider feed the cylinder. The cylinder strokes at fast speed and pump pressure climbs to 900 psi.

If the pressure required to move the cylinder to the load is low, this system works well. There is enough flow to move rapidly at low pressure and enough pressure at low flow to do the work.

Note: Standard gear-motor flow dividers are noisy. In the two circuits just discussed, the flow divider runs continuously. The high noise level may be detrimental in some locations.

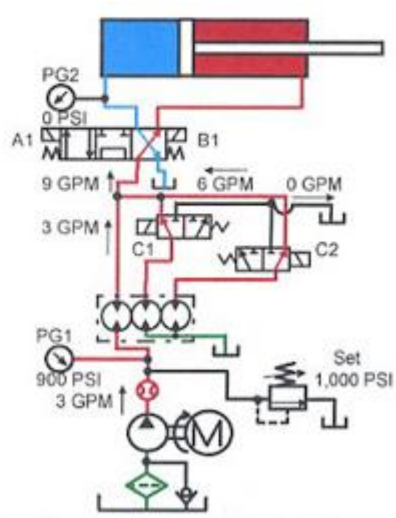


Fig. 10-63. Meter-in flow-control circuit with motor-type flow divider (to minimize heat) – cylinder retracting at fast speed.

Controlling speed of hydraulic motors

Figures 10-64 through 10-67 show schematic diagrams of flow-control circuits for hydraulic motors. The circuits in Figures 10-64 and 10-66 are the best for this purpose. In both of these circuits the motor only has pressure on one port. This arrangement provides the motors' internal leakage with a path to tank at minimal backpressure. High backpressure in a motor can cause wear on the motor shaft seal and leakage. Motors with external case drains eliminate high case backpressure at both ports, but motor drain flow increases.

For a running-away load, the schematic diagram shown in Figure 10-65 is best. When using a meter-out circuit, specify a high-pressure shaft seal due to backpressure at the motor outlet port. A case drain to let internal leakage go directly to tank can take the place of the high-pressure shaft seal, and is preferred in most cases.

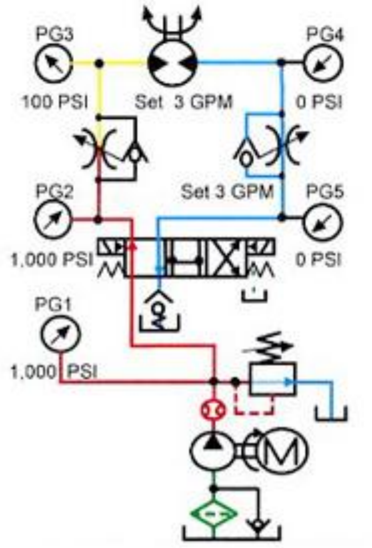


Fig. 10-64. Meter-in flow-control circuit for hydraulic motor.

The bleed-off circuit in Figure 10-66 will generate the least heat of the first three. The only problem is that the motor will slow a little as pressure increases -- due to pump and motor inefficiency. The pressure-compensated flow control will maintain a constant bypass of fluid as pressure climbs. However, pump output will decrease and motor bypass will increase, causing the motor to slow. Use high-efficiency components to keep this speed change minimal.

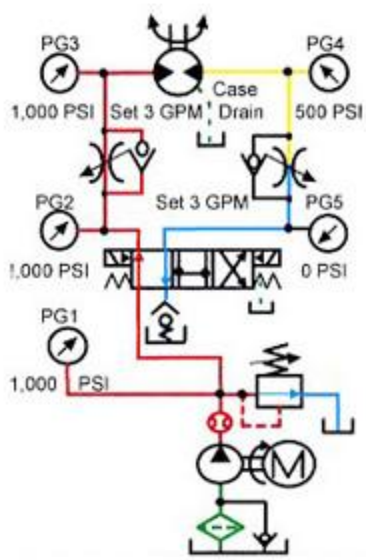


Fig. 10-65. Meter-in flow-control circuit for hydraulic motor. (Requires high-pressure seals and case drain, as shown.)

The circuit in Figure 10-67 is the most efficient of the four in this section. With a variable-volume pump, pressure is always just what the load requires, backpressure is negligible, and heat generation is minimal. To obtain very accurate control, use a servo-controlled pump with electronic feedback from the motor.

Any of the flow-control circuits for cylinders works well with motors. The main difference is dealing with leakage. Most cylinders have dynamic seals that do not leak. Also, there are no areas to trap internal leakage in a cylinder.

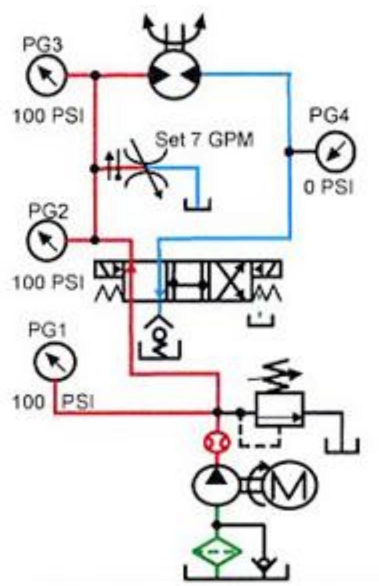


Fig. 10-66. Bleed-off or bypass circuit for hydraulic motor.

Three-port flow control

Figures 10-68 through 10-70 diagram how to control the output of a fixed-volume pump with a 3-port flow control. A 3-port flow-control circuit operates the same as a bleed-off flow control. The main difference is that a 3-port flow control does not need an external relief. The flow *control's integral relief valve protects the pump and makes the flow control pressure compensated.*

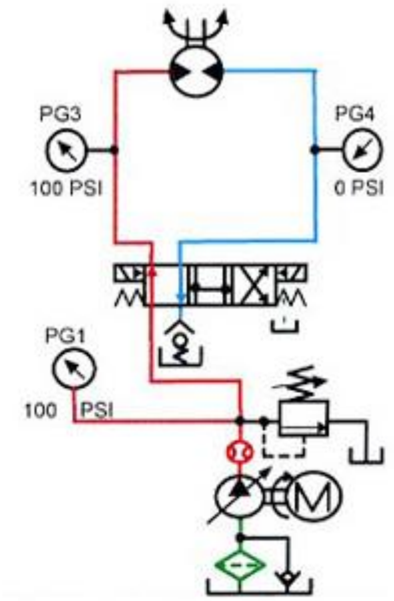


Fig. 10-67. Meter-in flow-control circuit for variable-volume pump supplying hydraulic motor.

A 3-port flow control used with a running away load requires a counterbalance valve (as was shown in Figure 10-15). Normal use is with a single-actuator circuit because it controls maximum pump pressure and flow, and affects all actuators in a circuit.

Figure 10-68 shows a typical 3-port flow-control circuit at rest with the pump running. Flow from the pump is 10 gpm. The needle valve is set for 5 gpm. The bias spring on the main relief valve is about 25 to 70 psi (according to the manufacturer). In the at rest condition, 5 gpm flows to tank through the directional valve and 5 gpm flows to tank through the main relief valve (at a pressure drop of approximately 70 psi). The normally closed pilot operator on the relief valve stays closed until system pressure reaches its setting.

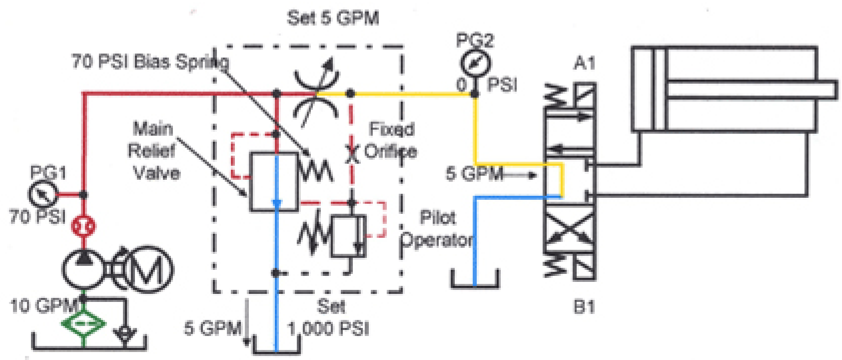


Fig. 10-68: Three-port flow-control circuit at rest.

When the cylinder starts to move, as in Figure 10-69, its resistance will increase pressure at the outlet of the flow control. Pressure build-up sensed through the fixed orifice acts on the main relief valve with the 70-psi spring. Any resistance from the actuator -- plus the 70 psi of the bias spring -- will increase pressure at the pump. The constant 70-psi pressure drop will maintain an accurate and constant flow to the cylinder while it moves. This 3-port flow control generates very little heat when the pressure requirement is low. It works the same as a bleed-off flow control except it has a built in relief valve.

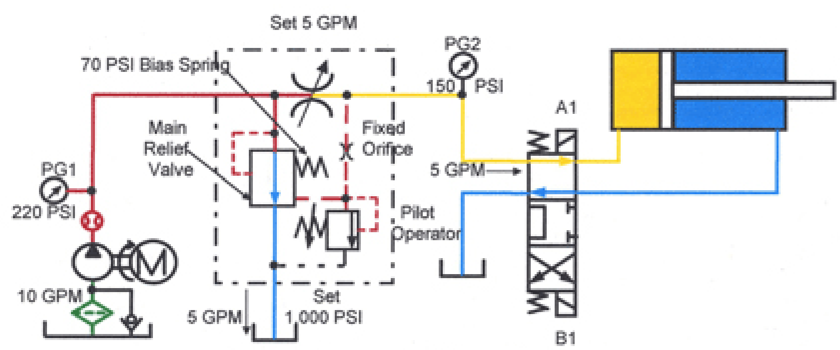


Fig. 10-69: Three-port flow-control circuit with cylinder extending.

When the cylinder reaches the load, as in Figure 10-70, pressure increases. As pressure at the cylinder rises, the main relief valve senses it and maintains a 70-psi differential right up to the setting of the pilot operator. When cylinder pressure reaches 1000 psi, all pump flow will go to tank across the main relief valve. The pilot-operator section of the 3-port flow control sets maximum system pressure.

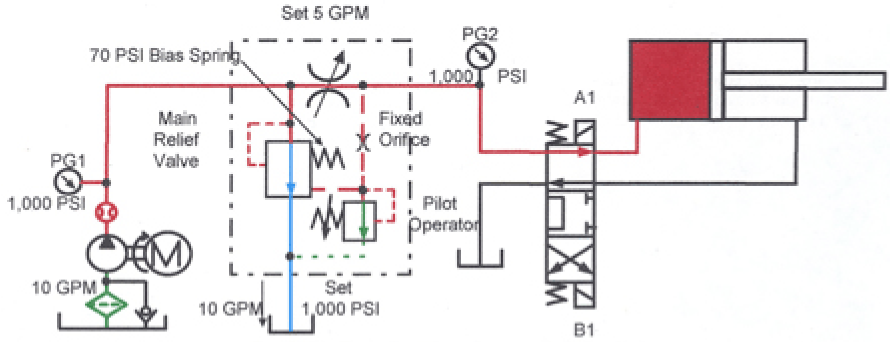


Fig. 10-70: Three-port flow-control circuit with cylinder stalled.

The 3-port flow control is similar to the priority flow divider that will be explained in Chapter 11. The main difference between these two valves is that tank flow always goes directly to tank on a 3-port flow control. Any backpressure in the tank line will add to the maximum system pressure. A priority flow divider can use the excess flow to operate other valves and actuators. Relief valves in the lines leaving the priority flow divider protect the circuits from overpressure

Flow Divider Circuits

When it is necessary to split a single hydraulic line into two or more identical flow paths, a tee or several tees can be the first solution. However, if the resistance in all the branches is not identical, flow can vary greatly in each path. Adding flow controls at the tee outlets makes it possible to change resistance and equalize flow in each branch, but as the machine operates, work resistance changes often require constant flow modifications. A device called a flow divider splits flow and compensates for pressure differences in most cases. A flow divider can split flow equally, unequally, and into more than two paths. One design maintains a constant flow for one outlet and directs any excess flow to a second outlet.

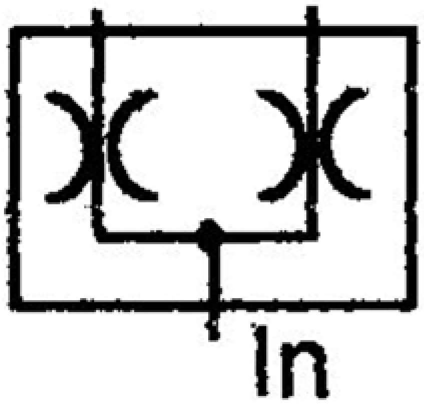


Figure 11-1. ISO symbol for flow divider.

Figure 11-1 pictures the ISO symbol for a flow-dividing valve. While the ISO symbol shows the function of the valve, it does not indicate which design it is. Fluid entering the flow divider splits and passes to both outlets equally. Figure 11-2 shows the symbol for a spool-type flow-divider and gives a better indication of the valve's operation. Note that a spool-type flow divider will not allow reverse flow. When using a spool-type flow divider to synchronize cylinders, add check valves to pass reverse flow. However, when the cylinders reverse, there is no synchronization with a spool-type flow divider.

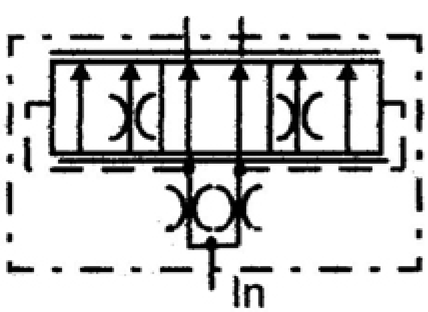


Figure 11-2. Spool-type flow divider.

Figure 11-3 shows a divider/combiner that synchronizes actuators in both directions of travel. It splits pump flow to the actuators and also assures that equal reverse flow returns from both cylinder ports.

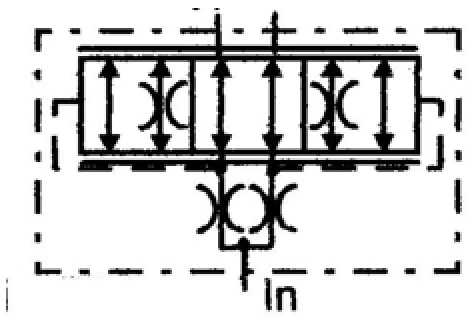


Figure 11-3. Spool-type flow divider and combiner.

Figure 11-4 pictures a flow divider with bypass relief valves that allow a lagging cylinder to complete its stroke. Reverse-flow check valves allow free flow around the divider spool while the actuator returns.

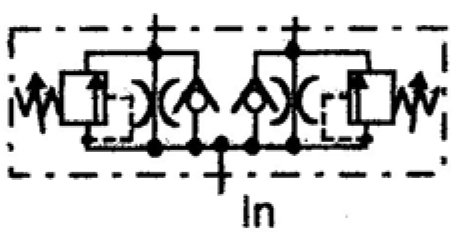


Figure 11-4. Spool-type flow divider with bypass reliefs and check valves.

Figures 11-5 and 11-6 show a priority flow-divider symbol. Port CF (controlled flow) of this flow divider always has the same flow when the pump is producing that flow or more. Excess pump flow goes through port EF (excess flow) to tank — or to another circuit.

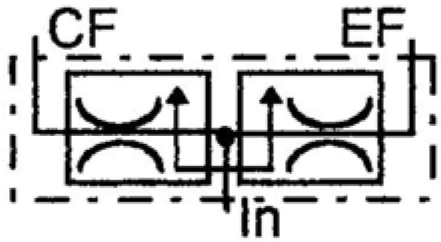


Figure 11-5. Fixed-flow spool-type priority flow divider.

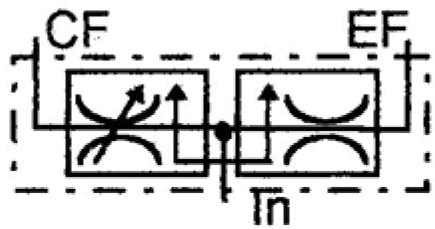


Figure 11-6. Variable spool-type priority flow divider.

Figures 11-7 and 11-8 show motor-type flow-divider symbols (as drawn by the manufacturers). This type flow divider is more efficient in most circuits. Motor-type flow dividers also work well in flow- and/or pressure-intensification circuits. They are available with multiple outlet ports and/or unequal flows.

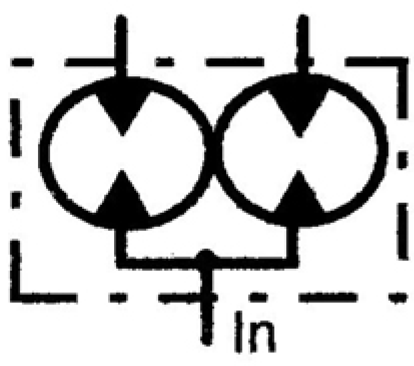


Figure 11-7. Motor-type flow divider.

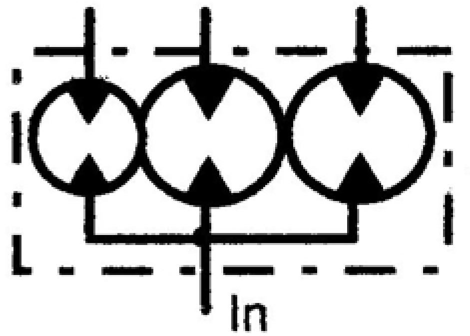


Figure 11-8. Unequal, triple-outlet motor-type flow divider.

Spool-type flow dividers

Spool-type flow dividers split flow through pressure-compensated fixed orifices. The pressure-compensation feature ensures near-equal flow through the orifices — even when inlet and/or outlet pressures fluctuate.

Spool-type flow dividers can split flow equally or unequally, according to the orifice sizes. Always use spool-type flow dividers at or near their rated flow. Because most designs use fixed orifices, equality of flow is poor when used below their rated flow. If flow exceeds the rating of the valve, high pressure drop causes poor performance and fluid heating.

The dividing accuracy of spool-type flow dividers can be as close as ±5%, depending on the pressure difference at the outlet ports.

Figure 11-9 shows a spool-type flow divider splitting pump flow equally. With this circuit, flow to each directional valve is nearly equal, even with one cylinder working at high pressure while the other cylinder is at low pressure or stopped by a centered valve.

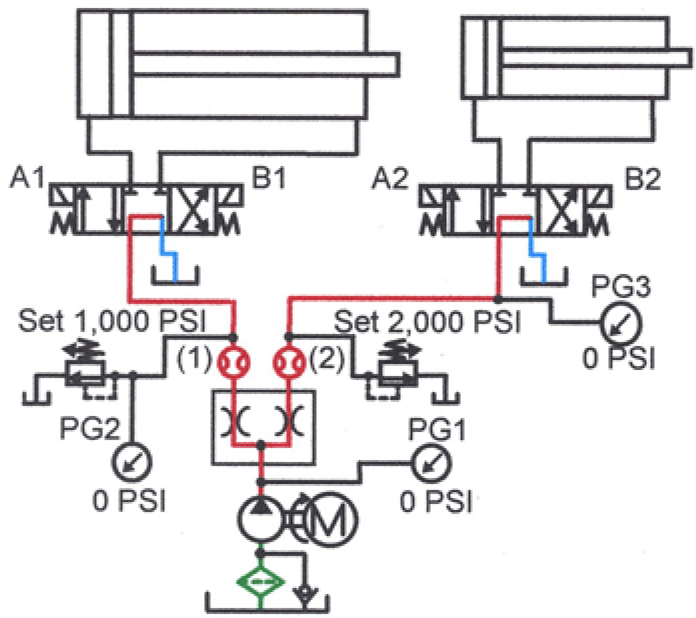


Figure 11-9. Spool-type flow divider piped to split pump flow. (Shown at rest with pump running.)

In Figure 11-10, fluid from port 1 flows to tank through the directional valve while fluid from port 2 drives a cylinder. Pressure at port 1 is 0 psi while pressure at port 2 is 1500 psi. Under these conditions, pressure at the flow divider inlet also is 1500 psi. Pressure at the inlet of a spool-type flow divider is always equal to the highest-pressure outlet. This condition generates a lot of heat because pressurized oil leaving port 1 is not doing work. It is best to use a spool-type flow divider in circuits where both outlet ports are at or near the same pressure. The higher the pressure variation, the greater the energy wasted as heat with spool-type flow dividers. When outlet pressures continuously vary by more than 300 to 500 psi, it is best to use a motor-type flow divider.

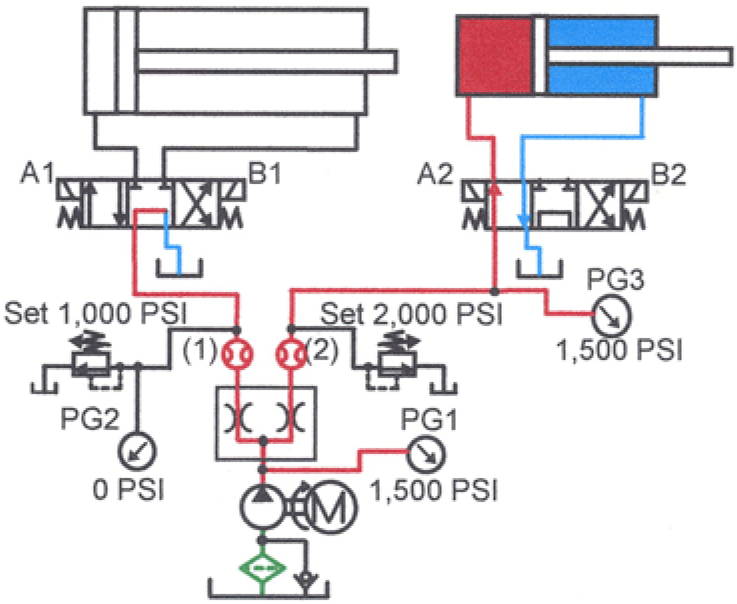


Figure 11-10. Spool-type flow divider piped to split pump flow. (Shown with right-hand cylinder extending.)

When splitting flow into more than two paths, add another spool-type flow divider to each outlet of the first divider. Figure 11-11 shows a synchronizing circuit for four unidirectional hydraulic motors. Flow split equally by the first spool-type flow divider goes to two more spool-type flow dividers. The second pair of spool-type flow dividers split the half flow from the first spool-type flow divider, and sends equal flow to the four motors.

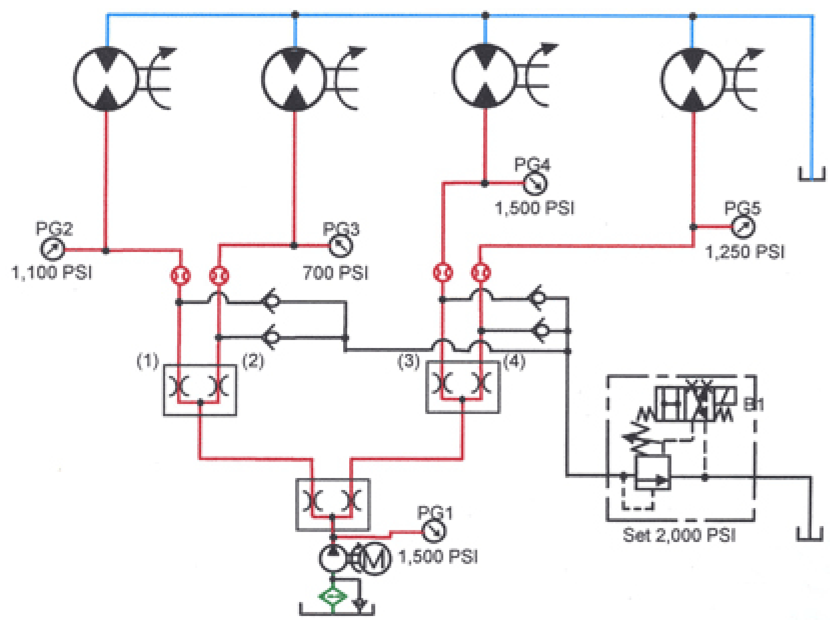


Figure 11-11. Spool-type flow divider piped to split pump flow into four equal parts (Shown with pump running.)

When using spool-type flow dividers for equal flow, the total number of dividers must be an odd number. If used in any even combination, flow will not be equal from all outlets -- unless the first divider has unequal flow from its outlets.

To get three equal outputs with spool-type flow dividers use one with unequal outputs, say 33.3% and 66.7%. Send flow from the 33.3% side to power the first actuator. Send flow from the 66.7% side to an equal-flow divider. Flows from the equal flow divider outlets is now 33.3% of total pump flow, so all three outputs are the same.

Notice that these circuits cannot handle reverse flow. Reverse flow through a spool-type flow divider will lock up one actuator when return pressure differs at the outlet ports.

Also notice that each outlet of a flow divider can have a different pressure. Figure 11-9 shows outlet 1 with a relief valve set at 1500 psi, and outlet 2 set at 2000 psi. (If both cylinders operate at the same pressure, substitute a single relief valve at the pump.) However, if both cylinders are moving and one of them stalls at 2000 psi, both cylinders will stop. The relief valve arrangement in Figure 11-11 allows any motor needing more than 2000 psi to stop while all other motors continue turning.

Spool-type flow divider/combiners

Spool-type flow dividers only allow flow in one direction. From the symbol in Figure 11-2, it is plain that reverse flow would lock up one of the cylinders. The cylinder that needs less resistance actually gets more. In a circuit where flow must go both ways, use a check valve to bypass the flow divider.

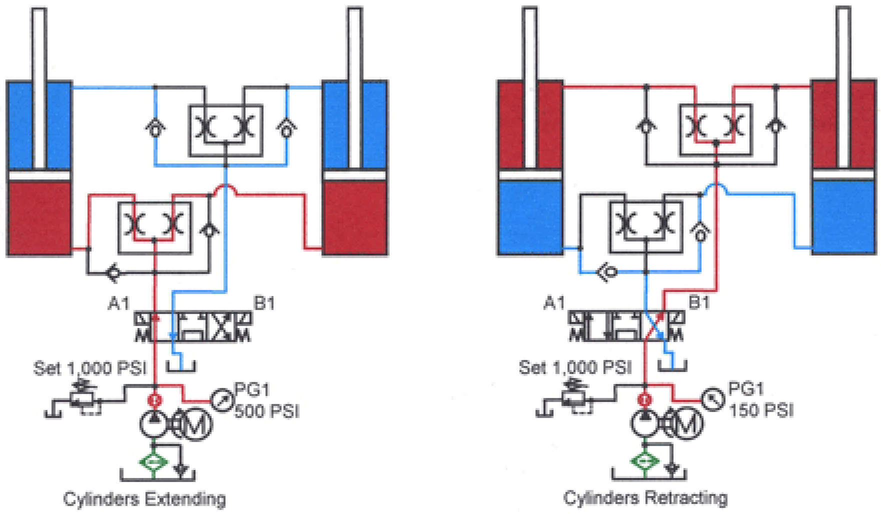


Figure 11-12. Spool-type flow divider arranged to synchronize two cylinders.

Figure 11-12 shows spool-type flow dividers in a circuit that synchronizes two cylinders. As the cylinders extend, the flow divider splits the flow and cylinder speed is nearly the same. When the cylinders retract, bypass check valves allow fluid to go around the divider. There is no synchronization from the cap-end flow divider at this time. A second flow divider with bypass check valves on the rod-end ports (as shown) is necessary for identical movement while retracting. As depicted in Figure 11-4, some flow dividers come with integral bypass check valves. Integral bypass check valves save piping time, have fewer leaks, and are more compact.

Because flow dividers are not 100% accurate, one of the cylinders may lag. Because there is internal leakage past the spool, any flow divider will let the lagging cylinder continue its travel. Because of the bypass leakage, the speed of the lagging cylinder while it is going to the end of its stroke is very slow. Integral relief valves (as shown in Figure 11-4) allow the lagging cylinder to catch up quickly. Set these relief valves between 50 and 150 psi. Once the pressure difference across the valve reaches this pressure range, fluid bypasses the restricted spool to quickly re-phase the cylinders.

*In Figure 11-13, a single flow divider/combiner synchronizes cylinders in both directions of travel***.** *Here a flow divider/combiner replaces the flow divider and check valves in Figure 11-12. Because there is no ANSI symbol for the flow divider/combiner, add bi-directional arrows to the one-way flow-divider symbol. This more-detailed symbol helps to clarify the valve's action. Bi-directional arrows show the divider/combiner function. These detailed symbols come from manufacturers' catalogs and represent their interpretation of their valve's function.*

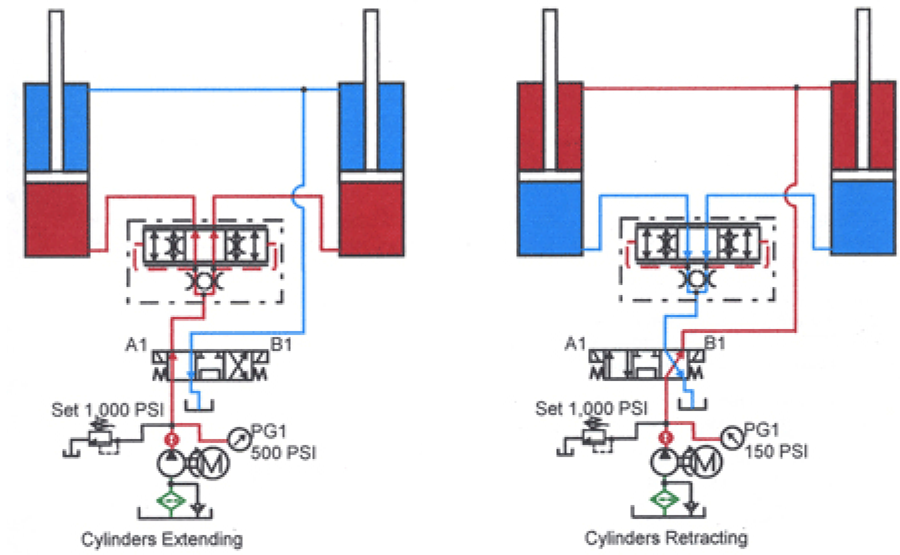


Figure 11-13. Spool-type flow divider/combiner arranged to synchronize two cylinders.

As the cylinders extend, the divider/combiner splits the flow to keep cylinder speeds nearly the same. When the cylinders retract, the divider/combiner shifts internally and equalizes return flow also.

A flow divider/combiner wastes energy the same as a standard flow divider. In essence these devices are infinitely variable pressure-compensated flow control pairs. Any flow control will cause heat because it is a restriction.

Flow dividers or flow divider/combiners are not designed to control running-away loads. For the circuits in Figures 11-12 and 11-13, a counterbalance valve in the line between the directional valve and the flow divider may be necessary if the loads can run away.

Spool-type priority flow dividers

Figure 11-14 shows a typical spool-type priority flow divider circuit. A priority flow divider maintains constant flow from the controlled flow (CF) port. Any additional flow passes out the *excess flow (EF) port. The non-standard symbol in the Figure is one typically found in manufacturers' catalogs. The controlled flow may be fixed or adjustable, according to the circuit needs. The excess flow may be sent to tank or to another circuit as required. (When there is pressure at the excess flow port, make sure the valve design can handle it.)*

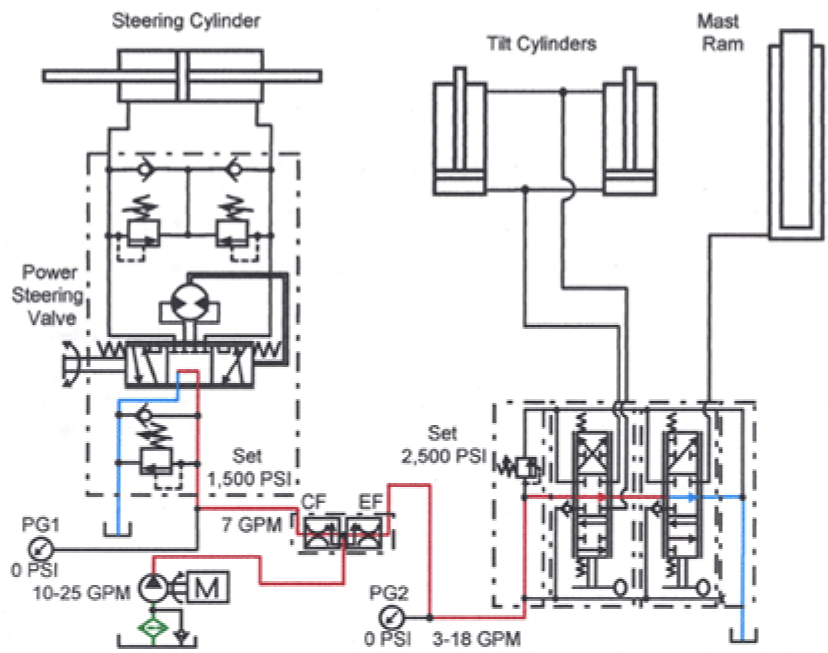


Figure 11-14. Typical lift-truck circuit using spool-type priority flow divider.

Some priority flow dividers are more like 3-port flow controls and cannot stand backpressure at the EF port. Use these flow dividers for bleed-off flow controlling only. With a bleed-off type priority flow divider, pressure at EF port causes flow at the CF port to fluctuate.

In Figure 11-14, a fixed-orifice priority flow divider is used on a vehicle with power steering and hydraulic actuators. This is the standard circuit for a forklift truck using a fixed-volume pump. The power-steering circuit needs 7 gpm and pump flow at idle is a minimum of 10 gpm. The actuators need as much as 15 gpm for maximum speed.

When the vehicle is operating, the power steering circuit will always have at least 7 gpm. When the mast or tilt cylinders need fluid, excess pump flow operates them. Because there is little excess flow at idle, the mast and tilt cylinder's speeds are slow at this time.

The circuit in Figures 11-15 and 11-16 controls the speed of a hydraulic cylinder powered by a fixed-volume pump. The adjustable controlled-flow port of the priority flow divider connects to the cylinder valve, with the excess-flow port piped to tank. This arrangement controls cylinder *speed and keeps heat build up low because the pressure in this circuit is only slightly higher than the cylinder needs.*

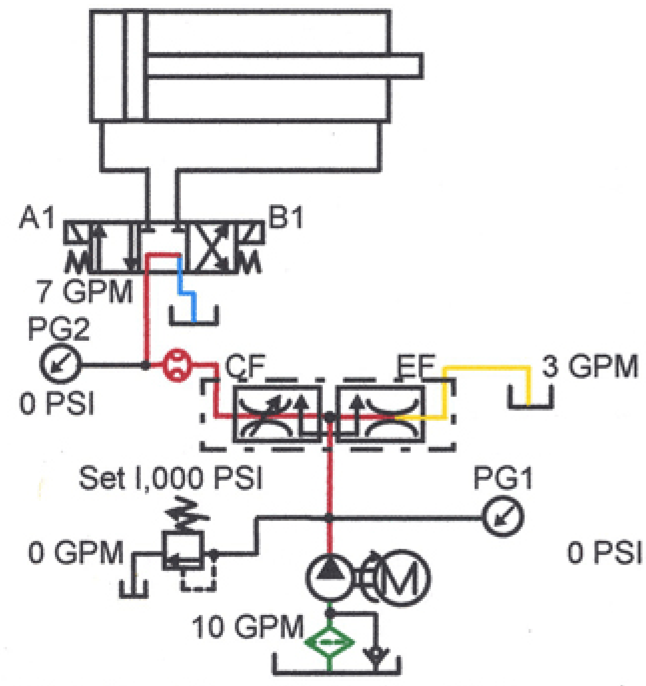


Figure 11-15. Spool-type priority flow divider arranged to bleed-off excess flow to tank. (Shown with pump running.)

Most priority flow dividers are pressure compensating so the priority flow remains constant even when pressure changes occur. As long as there is enough pump output, the controlled flow is constant. Excess flow changes as pump volume varies.

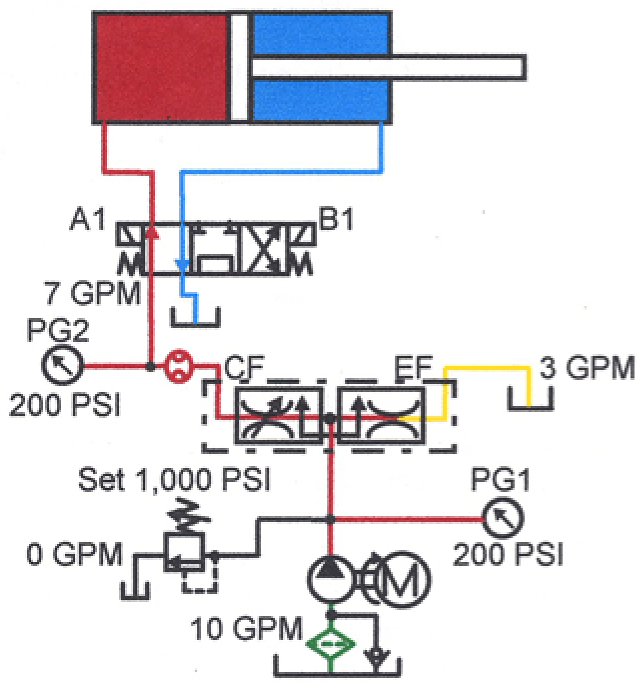


Figure 11-16. Spool-type priority flow divider arranged to bleed-off excess flow to tank. (Shown with cylinder extending.)

A priority flow divider wastes energy just like any spool-type divider. The inlet pressure to the divider is the same as the highest outlet pressure. When either outlet port is pressurized, the port with little or no pressure is wasting energy and generating heat.

Motor-type flow dividers

Motor-type flow dividers consist of two or more hydraulic motors in a common housing. All the motors share a common shaft, so they all turn at the same speed. All motors have a common inlet but separate outlets. If the motors have the same displacement, the output from each motor is nearly equal. (Some motor-type flow dividers use motors with different displacements, so each section's output differs.) The big advantage of a motor-type flow divider over a spool-type flow divider is energy transfer between sections. A spool-type flow divider's inlet pressure is always equal to the highest outlet pressure. This means heat generation from the lower or 0 pressure outlets, because pressurized fluid goes to tank without doing any work.

In contrast, a motor-type flow divider's inlet pressure is the average of the sum of the outlet pressures. Because there is a mechanical link between sections, excess energy transfer via this *link greatly reduces heat generation. Because hydraulic motors are not 100% efficient, there still is some energy loss and heat generation in any motor-type flow divider.*

Another advantage of motor-type flow dividers is their outlet options. A spool-type flow divider has only two outlets; a motor-type flow divider can have many outlets — in even or odd numbers. Most manufacturers catalog units with 6 to 8 outlets, but also will custom-build dividers to suit.

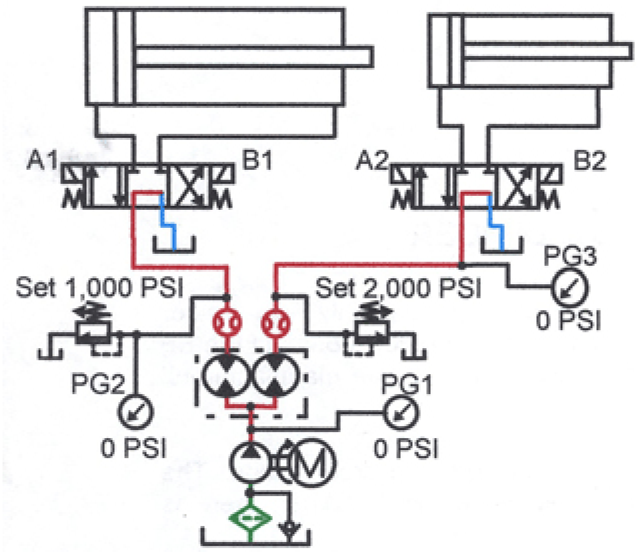


Figure 11-17. Motor-type flow divider piped to split pump flow. (Shown at rest with pump running.)

Figure 11-17 shows a motor-type flow divider splitting flow from a fixed-volume pump to separate actuators. With the cylinders at rest, all flow goes to tank through the tandem-center valves with minimal energy loss. To stroke the cylinder on the right, shift its directional valve as in Figure 11-18. Flow from the right-hand section of the motor-type flow divider sends half of the pump's flow to the right-hand cylinder at 1500 psi. The other half of the pump's flow goes to tank through the left valve at 0 pressure. Notice that pump pressure is approximately 750 psi instead of 1500 psi as in Figure 11-10. Pump pressure is low because most of the energy in the flow divider outlet going to tank mechanically transfers from the idling motor to the working motor. Whether one or both cylinders do work, energy going in is always equal to energy needed plus inefficiencies.

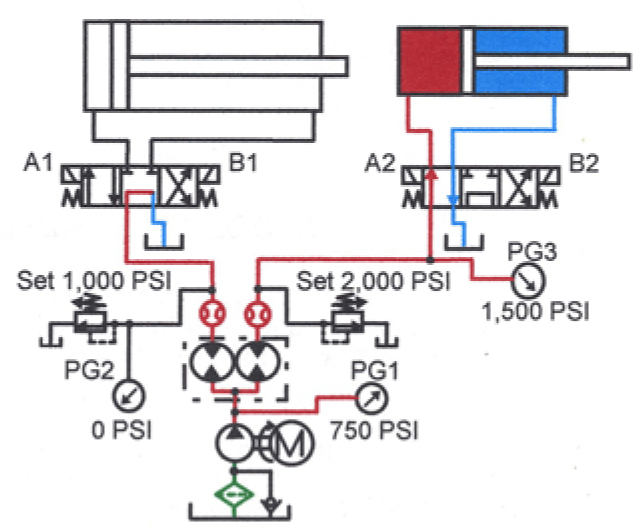


Figure 11-18. Motor-type flow divider piped to split pump flow. (Shown with right-hand cylinder extending.)

The 4-outlet motor-type flow divider in Figure 11-19 supplies four hydraulic motors. Because each motor has a different load, pressure at the motor inlets is not the same. To figure the approximate inlet pressure to the flow divider, add the outlet pressures and divide by the number of outlets. (1100 psi + 700 psi + 1250 psi + 1500 psi = 4550 psi. Divide by four outlets and 1138 psi is the pressure at the pump outlet). The 1138-psi figure is approximate due to losses in piping and the motors of the flow divider.

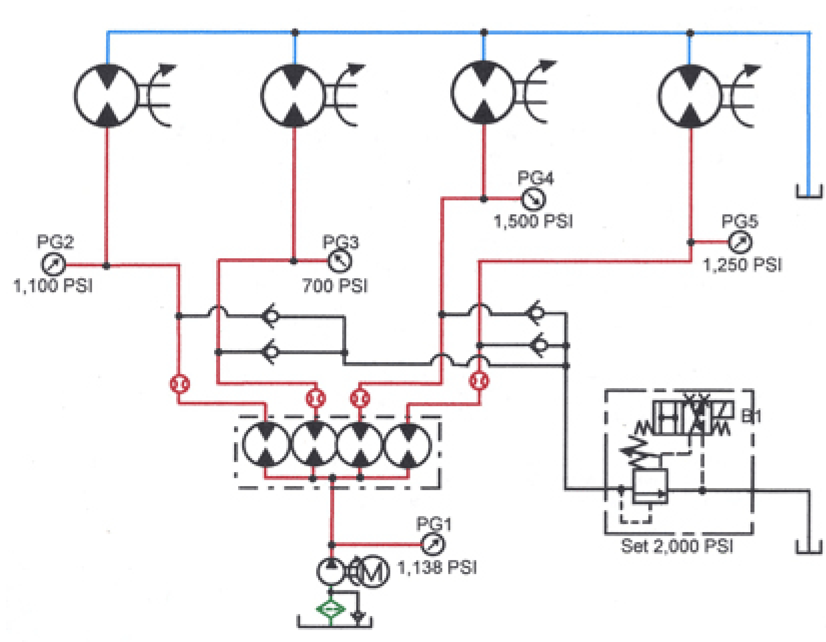


Figure 11-19. Motor-type flow divider piped to split pump flow into four equal parts. (Shown with pump running).

Notice the relief valve at the flow divider outlets. Because a motor-type flow divider also acts as an intensifier (See Figures 11-45 through 11-48), it is necessary to limit the pressure at each outlet. If each motor needs a different pressure, use separate relief valves at each flow divider outlet. In Figure 11-19, a set of check valves and a single relief valve sets the same pressure for each motor — and protects them from overpressure. Because the relief valve is solenoid-operated it also starts and stops all motors simultaneously.

Motor-type flow divider synchronizing two cylinders

Motor-type flow dividers work well for synchronizing actuators. Figure 11-20 shows two cylinders synchronized by a double equal-outlet, motor-type flow divider. Install the flow divider between the valve and the cylinder cap-end ports as shown. This arrangement synchronizes the extension stroke of the cylinders and provides some control for the retraction stroke (See Figures 11-22 and 11-23). Use a second flow divider at the rod-end ports for precise control on the retraction stroke when required.

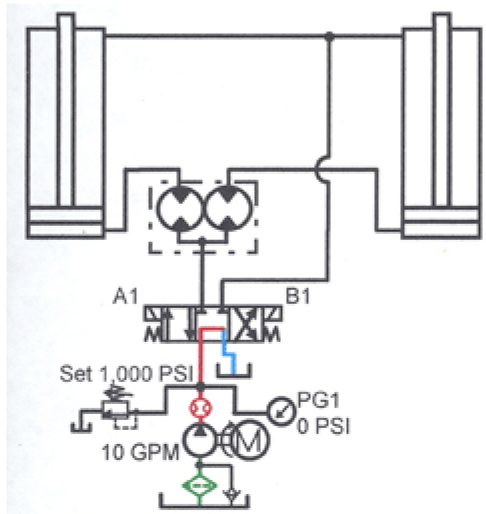


Figure 11-20. Motor-type flow divider piped to synchronize two cylinders. (Shown at rest with pump running.)

As the cylinders extend, as in Figure 11-21, the flow divider splits pump flow, causing the actuators to extend at the same time. If the cylinders' loads require different pressures, the flow divider still sends almost equal flow to each port. A motor-type flow divider has some internal bypass, causing the section with the higher outlet pressure to pass less than half flow. Therefore, use motor-type flow dividers for circuits needing only nominal synchronization. With any type of hydraulically controlled synchronization, always take the cylinders to a fixed position at one or both ends of the stroke.

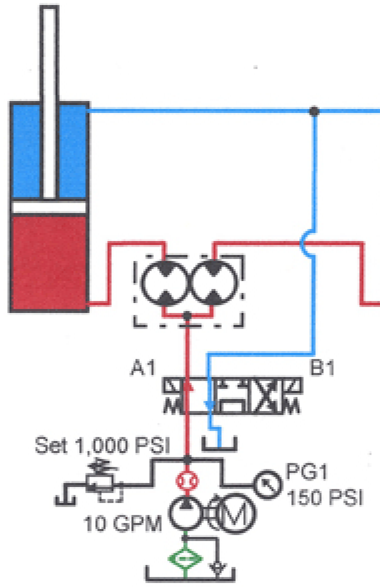


Figure 11-21. Motor-type flow divider piped to synchronize two cylinders. (Shown with cylinders extending.)

Also, if pressure intensification above any of the system's component ratings is possible, put a relief valve at the flow divider outlets. Several manufacturers supply their flow dividers with integral bypass relief valves. Set these reliefs for a safe pressure differential so intensification will not damage the cylinder. When a bypass relief valve starts relieving, the cylinder on that side stops while the opposite cylinder's speed doubles. (If integral relief valves are not available, install external reliefs when there is a chance for actuator damage from high pressures.)

Some manufacturers pipe the integral relief valve's outlet to tank instead of back to the flow divider inlet. This type of relief valve circuit dumps fluid to tank at a pressure low enough to keep from damaging the actuator. Using a relief valve with its outlet piped to tank causes one actuator to stop and allows the other one to continue at the same speed.

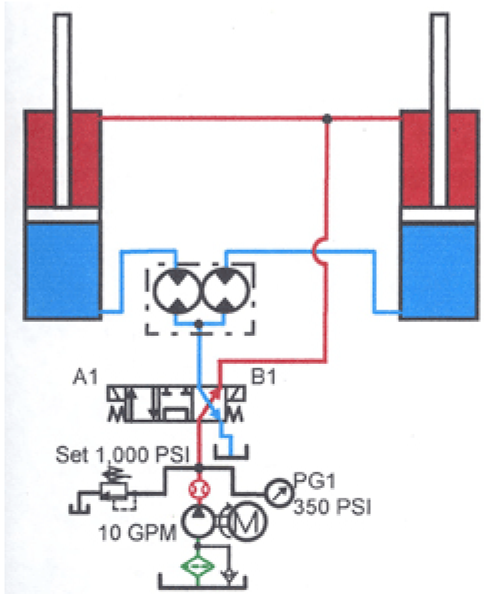


Figure 11-22. Motor-type flow divider piped to synchronize two cylinders. (Shown with cylinders retracting.)

Figure 11-22 shows how the cylinders retract under normal conditions. Flow from the pump goes to both rod-end ports and the cylinders retract together. The flow divider combines the oil from the cap-end ports and synchronization continues. However, if one cylinder binds on the retract stroke, the cylinder with less drag will run away.

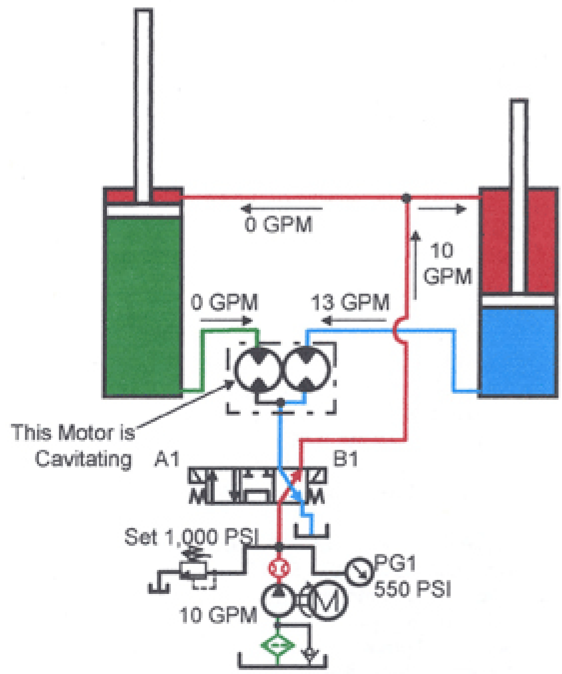


Figure 11-23. Motor-type flow divider piped to synchronize two cylinders. (Left-hand cylinder shown binding.)

Figure 11-23 depicts what happens when a cylinder binds. All flow from the pump goes to the right-hand cylinder — retracting it at double speed. The right-hand motor of the flow divider turns rapidly due to the high flow. The left-hand motor of the flow divider also turns rapidly, but no oil passes through it. The left-hand motor cavitates due to this lack of fluid. After the right-hand cylinder bottoms out, pressure buildup may cause the left-hand cylinder to retract. As the left-hand cylinder retracts, the right-hand motor of the flow divider cavitates.

If the cylinders in a circuit have different return-force requirements, or are subject to binding, add a second motor-type flow divider at the cylinders' rod-end ports. The second flow divider assures that the cylinders are synchronized on their retraction strokes also. (See Chapter 22, covering Synchronizing Circuits, for other ways to make actuators move at the same rate.)

Motor-type flow divider in a priority circuit

Using a motor-type flow divider in a priority circuit like the one shown in Figure 11-16 will give unsatisfactory results. A spool-type priority flow divider sends a constant flow to one outlet as long as the pump produces at least that much flow. When pump flow increases, priority flow *stays the same while the other outlet's flow starts or increases. Flow from the priority outlet stays constant through the entire pump range.*

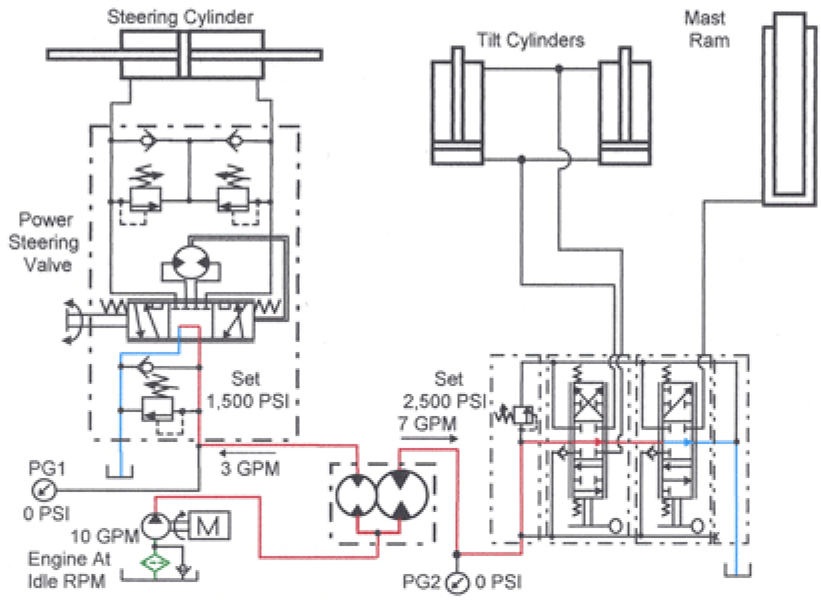


Figure 11-24. Motor-type priority flow divider in lift-truck circuit. (Shown with engine idling.)

Figures 11-24 and 11-25 show what happens when using a motor-type flow divider in place of a spool-type priority flow divider. With the engine at idle speed, 3 gpm flows to the power steering and 7 gpm to the cylinder circuit. This circuit works well at idle — if 3 gpm is enough for the power steering. Figure 11-25 indicates what happens when engine rpm and flow increase. As pump flow increases, both the power steering and the cylinder circuits receive more fluid in the same ratio. This overspeeds the power steering while robbing oil from the cylinder circuit. (There would be little or no heat generation from this circuit, but the end result is less than satisfactory.) Motor-type flow dividers with unequal flow outlets are available in various combinations and multiple flow paths. However, the flow from each outlet changes proportionately as the inlet flow changes. This feature makes them hard to adapt to the engine-driven pumps on much off-road equipment.

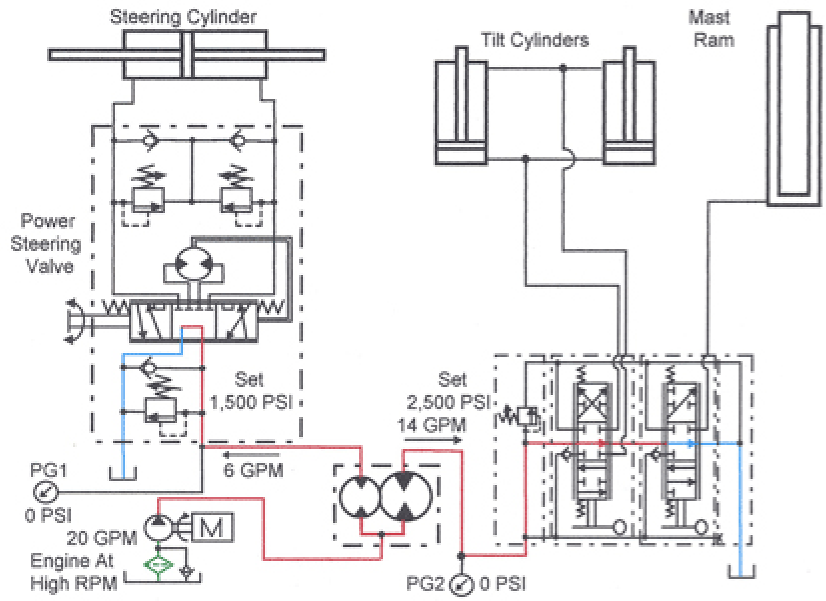


Figure 11-25. Motor-type priority flow divider in lift-truck circuit. (Shown with engine speed increased.)

Motor-type flow divider speed control

There are ways to use a fixed-volume pump and motor-type flow dividers to change speeds with minimal heat generation. Figures 11-26 through 11-33 depict some of these. These circuits only give fixed preset speeds without changing hardware.

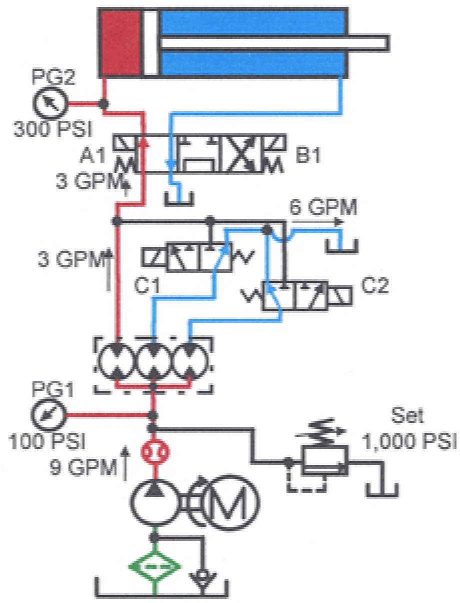


Figure 11-26. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder extending at slow speed).

Figure 11-26 shows a 3-speed flow control circuit using a motor-type flow divider. Here the cylinder is extending slow speed. With the circuit set up as shown, it defaults to slow speed. Notice there are no flow controls. To split pump flow evenly and reduce energy loss, use a motor-type flow divider at its outlet. Each outlet of the flow divider will put out about 3 gpm.

In Figure 11-26 the cylinder is getting 3 gpm of oil and requires a pressure of 300 psi to move. Note that the pump pressure is only 100 psi. This happens because the flow divider is taking in 9 gpm and using 3 gpm to do work. The other two 3-gpm flows are going back to tank at 0 psi. While it appears these two 3-gpm flows waste energy, they are actually transferring their energy through the common to the left-hand motor. The left-hand motor becomes a pump with a 100-psi inlet and two motors driving it to 300 psi. As always in flow-divider circuits, the average of the sum of the outlets will be the inlet pressure. (300 psi + 0 psi + 0 psi = 300 psi; divide by 3 to get 100 psi.) With this system, cylinder speed slows, but the only energy loss is the inefficiency of the components used.

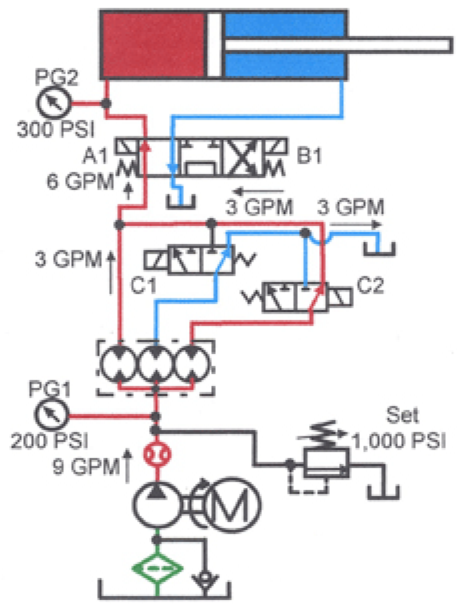


Figure 11-27. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder extending at medium speed.)

To get mid speed, the directional valves shift as indicated in Figure 11-27. By energizing solenoid C2 on the right-hand 3-way valve, an extra 3 gpm goes to the cylinder to give mid speed. Note that the pump pressure rises to 200 psi as the cylinder speed doubles. There still is only hardware inefficiency to waste energy, so the system runs cool.

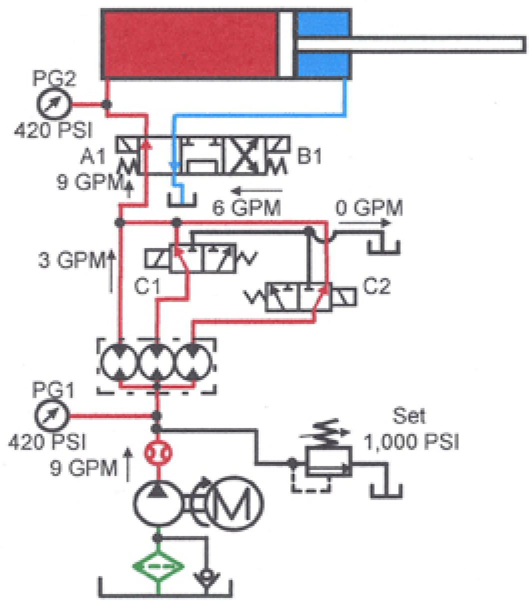


Figure 11-28. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder extending at fast speed.)

To make the cylinder stroke at fast speed, shift the directional valves as shown in Figure 11-28. By energizing solenoids C1 and C2, both 3-way valves shift to send all pump flow to the cylinder. While the cylinder is in fast speed mode, pump and cylinder pressure are the same.

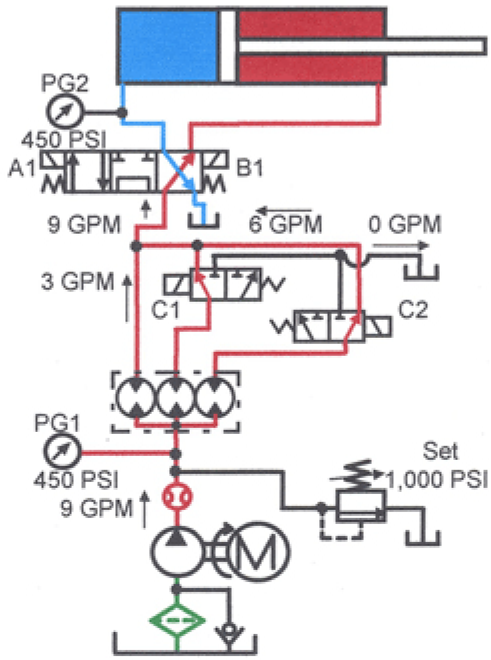


Figure 11-29. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder retracting at fast speed.)

To retract the cylinder at fast speed, shift solenoid B1 along with C1 and C2, as shown in Figure 11-29. Energizing one or more solenoids in the retract mode gives different speeds that are nearly the same as when extending.

If the flow divider had more and/or unequal-size motors, selection of a combination of speeds by selecting different flow outputs is possible.

This circuit is tamper-proof. To change the preset speeds, the flow divider and/or pump must be changed.

***Note**: Any flow-divider circuit will intensify pressure. In Figure 11-26, if the cylinder stalled, the pressure would continue to rise. When the pump reached the relief valve setting, pressure at the cylinder would be 3000 psi. Use a second pressure-relief valve between the flow divider and the pump port of the cylinder directional valve to set a safe pressure in case of cylinder stall.*

Speed control with motor-type flow dividers

Figures 11-30 through 11-33 show a different type of motor-type flow-divider circuit for variable speed. This circuit uses a smaller pump, electric motor, and tank to give the same speed but less *high-speed force. Notice there is a 3-gpm pump supplying one section of the flow divider. As the fed section of the flow divider turns, the other two sections also turn and pump fluid directly from the tank. Thus, in Figure 11-30, the two right-hand sections of the flow divider are only circulating oil. All pump flow is going to the cylinder, which is operating in slow-speed mode. In this condition, the cylinder is capable of generating its highest tonnage. Notice that the cylinder requires 300 psi to move it and the pump is showing 300 psi.*

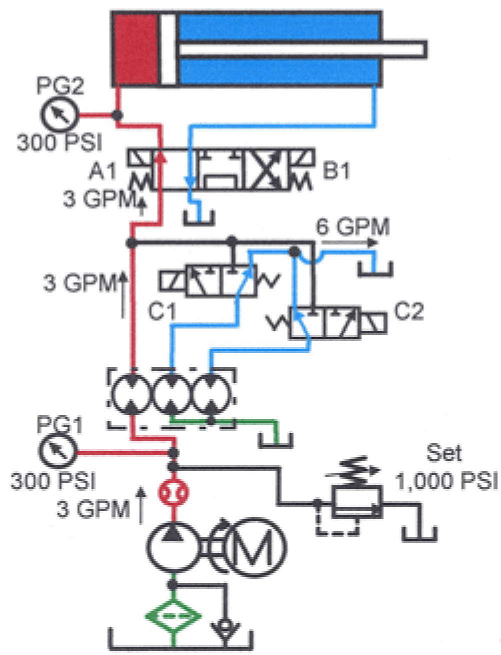


Figure 11-30. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder extending at slow speed.)

The cylinder speeds up when solenoid C2 on the left-hand 3-way valve is energized as in Figure 11-31. Now, one flow divider section sends its oil to the cylinder along with pump flow. The cylinder goes to mid-speed mode and pump pressure climbs to 600 psi.

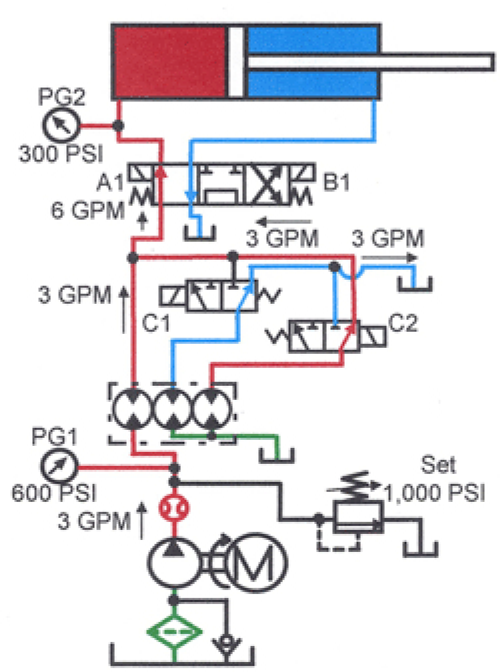


Figure 11-31. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder extending at medium speed.)

To get full speed from the cylinder, solenoid C1 on the right-hand 3-way valve is energized as shown in Figure 11-32. Now all three sections of the flow divider feed the cylinder. The cylinder is at fast-speed mode and pump pressure climbs to 900 psi.

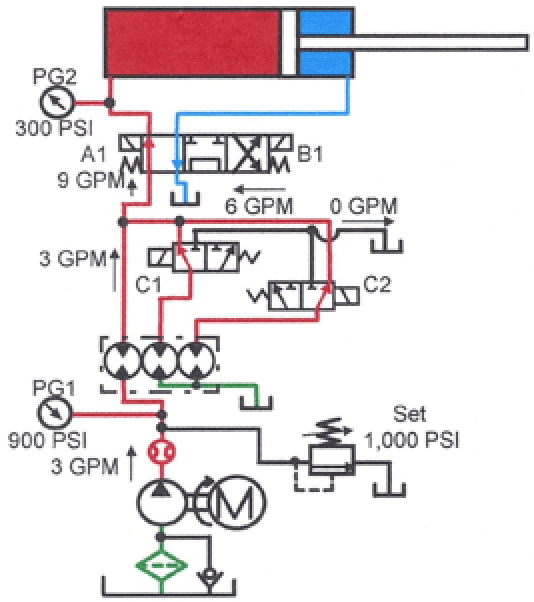


Figure 11-32. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder extending at fast speed.)

If the pressure required to move the cylinder to the work is relatively low, this circuit works well. There is enough flow to move rapidly at low pressure, and enough pressure at low flow to do the work.

Note*: The gears in standard motor-type flow dividers are noisy. In the above two systems, the flow divider turns continuously. The noise level may be unacceptable in low-noise areas.*

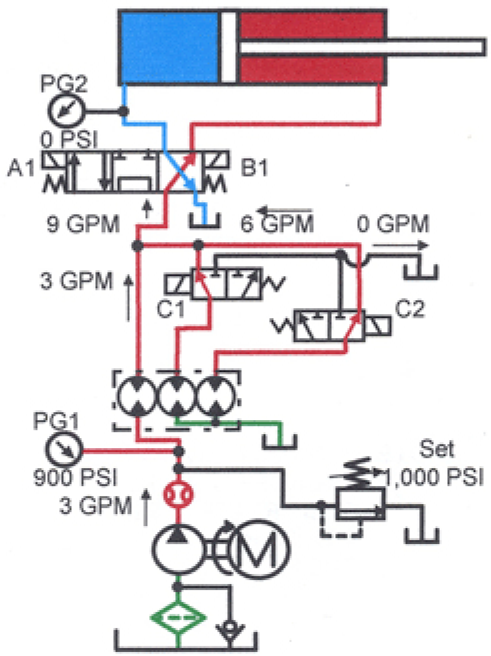


Figure 11-33. Meter-in flow-control circuit with motor-type flow divider to minimize heat generation. (Shown with cylinder retracting at fast speed.)

Motor-type flow divider in full-time regeneration circuit

Figures 11-34 through 11-44 picture a unique regeneration circuit using a motor-type flow divider. Normally flow-divider circuits use the split flow to synchronize actuator movement. This circuit uses a flow divider to intensify flow for regeneration. This circuit works best on cylinders with small rods; and gives exactly twice speed on double-rod cylinders and hydraulic motors.

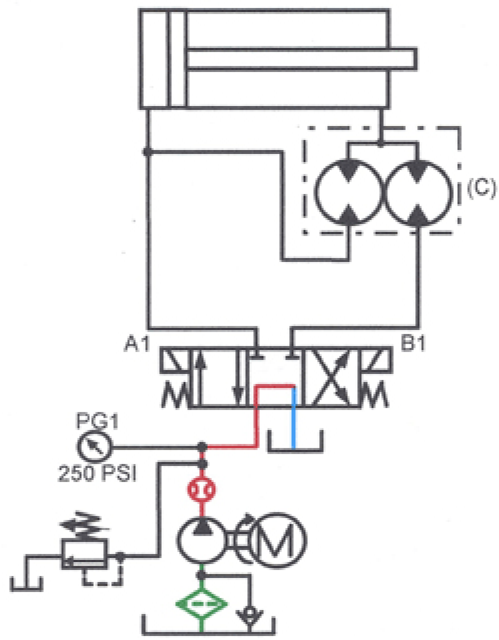


Figure 11-34. Full-time regeneration circuit using a motor-type flow divider. (Shown at rest with pump running.)

Figure 11-34 shows the circuit in the at-rest condition. Equal-outlet motor-type flow divider C is piped between the cylinder rod-end port and the directional valve. The flow divider's normal inlet port connects to the cylinder; one outlet connects to the directional valve; and the other outlet is teed into the cylinder cap-end line.

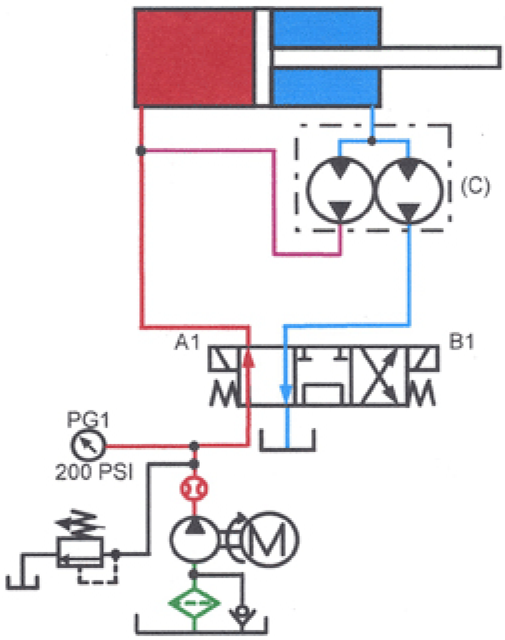


Figure 11-35. Full-time regeneration circuit using a motor-type flow divider. (Shown with cylinder extending under regeneration.)

Figure 11-35 depicts solenoid A1energized so that flow from the pump goes past the teed-in flow divider line to the cylinder cap end. As the cylinder extends, oil from the rod end enters the flow divider. The flow divider splits this oil. Half goes to tank at 0 pressure and half goes to the cylinder cap-end tee at pressure high enough to mix it with pump flow. As the cylinder starts to extend, speed quickly increases to almost twice the original speed. Maximum cylinder speed directly relates to the rod size: the larger the rod, the slower the speed. With a double rod-end cylinder, speed exactly doubles. As with any regeneration circuit, speed increases but force decreases.

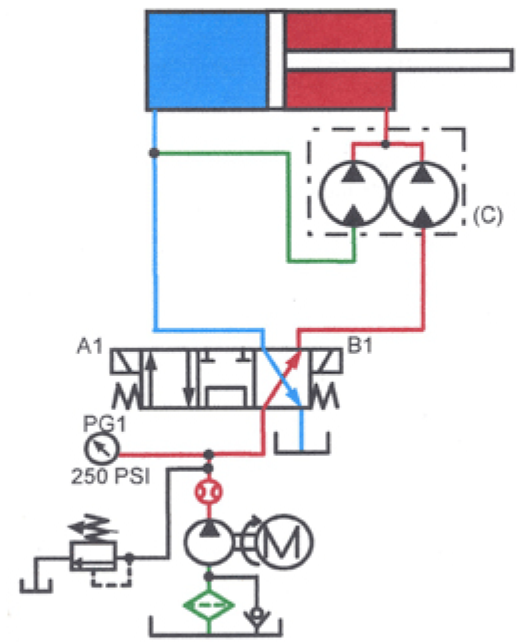


Figure 11-36. Full-time regeneration circuit using a motor-type flow divider. (Shown with cylinder retracting.)

Figure 11-36 shows the cylinder retracting. Energizing solenoid B1 of the 4-way directional valve sends pump flow to one outlet of the flow divider. Both of the motors in the flow divider turn at the rate of flow from the pump. During this part of the cycle, the motor that has its inlet teed into the cap-end line becomes a pump. Pump flow plus the same flow from the second motor makes the cylinder retract twice as fast as a conventional circuit. (However, cylinder thrust is only half that of a conventional circuit.) This flow-divider regeneration circuit doubles the cylinder speed without making the pump work harder. Size the pump, valve, tank, and piping up to the regeneration circuit according to pump flow. The only high flows are at or very near the cylinder.

Using a motor with a higher displacement on the left-hand side of the flow divider increases speed even more. The limit is reached when pressure to run the cylinder at the faster rate exceeds the relief valve setting. When using unmatched motors, make sure the line from the cylinder cap-end to the motor will handle the higher suction flow.

Motor-type flow-divider regeneration circuit – pressure-activated to full thrust

When it is necessary to get out of regeneration and into full thrust, add other valving to the motor-type flow-divider regeneration circuit.

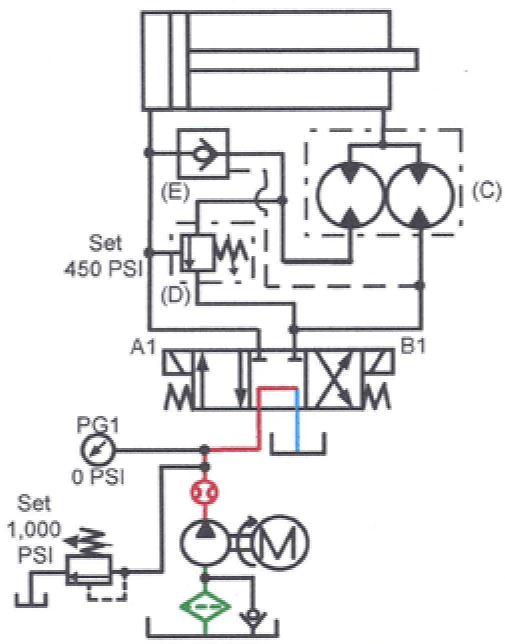


Figure 11-37. Regeneration circuit with motor-type flow divider that can be pressure activated to full thrust. (Shown at rest with pump running.)

The regeneration circuit shown at rest in Figure 11-37 can be pressure activated to produce full thrust. Equal flow divider C is piped between the 4-way directional valve and the cylinder. The normal inlet port connects to the cylinder; one outlet connects to the directional valve; and the other outlet passes flow freely through pilot-operated check valve E to a tee in the cylinder cap-end line. Pilot-operated check valve E gets its pilot signal from the cylinder rod-end line before the flow divider port. Teed into the line between the flow divider and check E is the inlet to sequence valve D. Sequence valve D's outlet tees into the cylinder rod-end line. Sequence valve D is internally drained and gets its external pilot signal from the cylinder cap-end line.

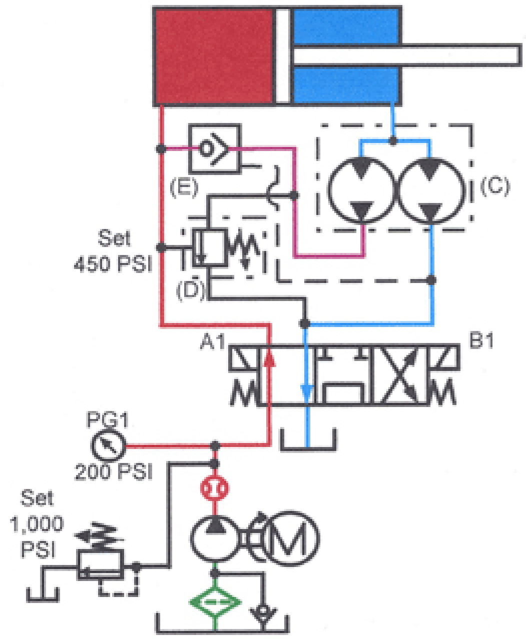


Figure 11-38. Regeneration circuit with motor-type flow divider that can be pressure activated to full thrust. (Shown with cylinder extending under regeneration.)

In Figure 11-38, solenoid A1 is energized so flow from the pump goes past the tee in the flow divider line to the cylinder cap-end. As the cylinder extends, oil from the rod end enters the flow divider. This oil splits; half goes to tank at no pressure, and half free-flows through pilot-operated check valve E to the cylinder cap-end tee at a pressure high enough to mix with pump flow. When the cylinder starts to extend, its speed quickly increases to almost twice speed. Maximum cylinder speed directly relates to the rod size. The larger the rod, the slower the speed. For a double rod-end cylinder, speed exactly doubles. As with any regeneration circuit, speed increases but force decreases.

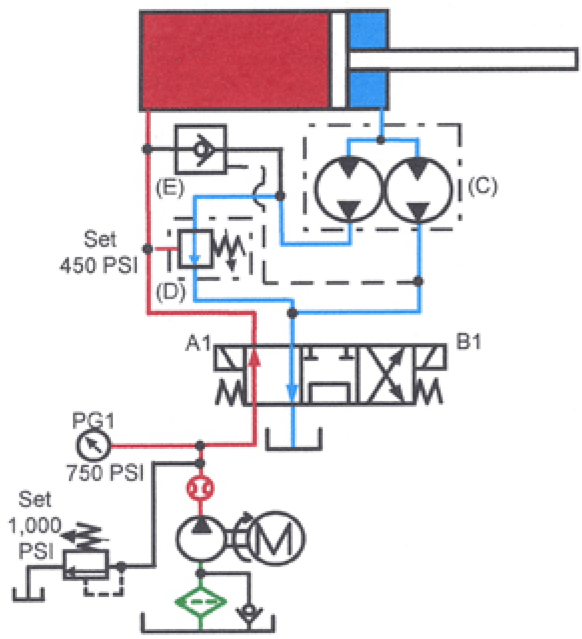


Figure 11-39. Regeneration circuit with motor-type flow divider that can be pressure activated to full thrust. (Shown with cylinder extending at full power.)

When the cylinder meets resistance, pressure increases. When the cylinder butts against the work, as in Figure 11-39, pressure build up in the cap-end line pilots sequence valve D open. When sequence valve D opens, oil from both sides of the flow divider returns to tank at no pressure. At the same time, pilot-operated check valve E closes to keep the pump from relieving to tank. With the rod end of the cylinder hooked to tank and the pump feeding the cap end, the cylinder produces full thrust.

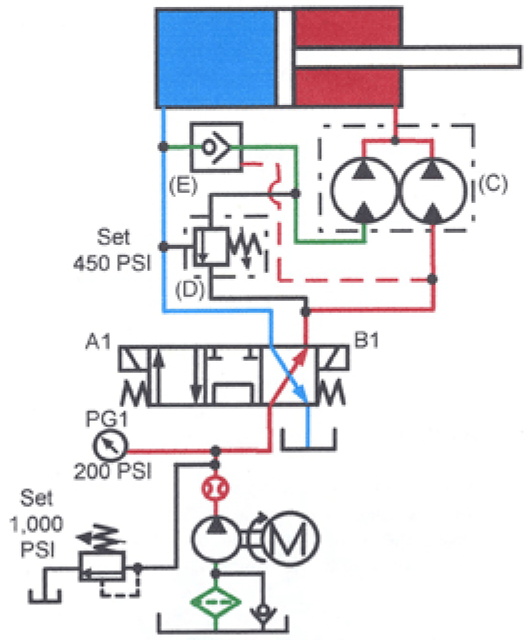


Figure 11-40. Regeneration circuit with motor-type flow divider that can be pressure activated to full thrust. (Shown with cylinder retracting.)

In Figure 11-40, the cylinder is retracting. By energizing solenoid B1, the 4-way directional valve sends pump flow to one outlet of the flow divider. Both of the motors in the flow divider turn at the rate of flow from the pump. During this part of the cycle, the motor — with its inlet teed into the cap-end line — acts as a pump. Pilot pressure from the cylinder rod-end port opens pilot-operated check valve E to allow this flow. Pump flow, plus the same flow from the second motor, makes the cylinder retract twice as fast as a conventional circuit. Again, however, cylinder thrust is only half that of a conventional circuit.

Motor-type flow divider regeneration circuit — solenoid-activated to full thrust

Other valving added to the motor-type flow-divider regeneration circuit produces an arrangement that can be switched to full thrust by activating a solenoid. Shown with the circuit at rest in Figure 11-41, equal-output flow divider C is piped between the 4-way directional valve and the cylinder. The flow divider's normal inlet port connects to the cylinder; one outlet connects to the directional valve; and the other outlet passes flow freely through pilot-operated check valve E to a tee in the cylinder cap-end line. Check valve E receives its pilot signal from the cylinder rod-end line before the flow divider port. Teed into the line between the flow divider *and the check valve is the inlet to normally closed directional-control valve D. Valve D's outlet tees into the cylinder rod-end line. Valve D is direct solenoid operated and thus does not need a pilot supply.*

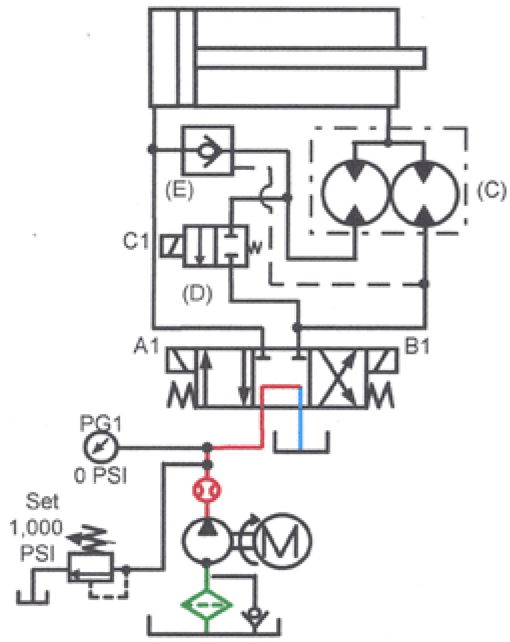


Figure 11-41. Regeneration circuit with motor-type flow divider that can be solenoid activated to full thrust. (Shown at rest with pump running.)

When solenoid A1 of the main directional valve is energized — as in Figure 11-42 — flow from the pump goes past the flow divider line to the cylinder. As the cylinder extends, oil from its rod end enters the flow divider. The flow divider splits this oil; half of it goes to tank at no pressure, and the other half of it passes freely through pilot-operated check valve E to the cylinder cap end at a pressure high enough to mix it with pump flow. As the cylinder moves, its speed almost doubles. The amount of speed increase is directly related to the size of the piston rod. The larger the rod, the slower the speed. A double rod-end cylinder would go exactly twice as fast. As with any regeneration circuit, the speed increases but force decreases.

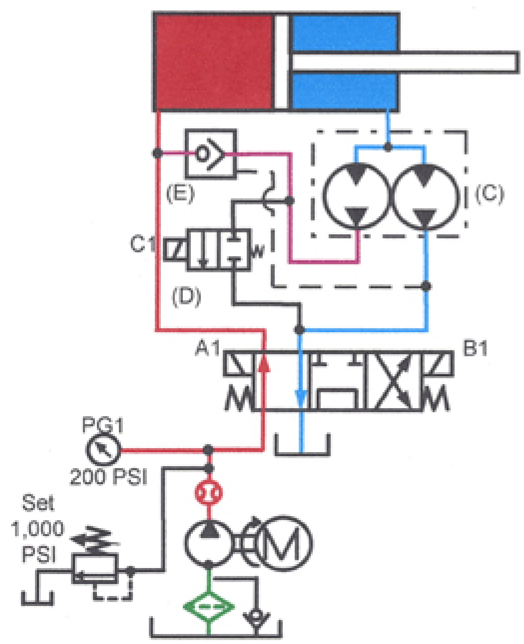


Figure 11-42. Regeneration circuit with motor-type flow divider that can be solenoid activated to full thrust. (Shown with cylinder extending under regeneration.)

When the cylinder rod trips a limit switch, as in Figure 11-43, the switch sends an electrical signal to solenoid-operated 2-way directional valve D, causing it to open. When valve D opens, oil from both sides of the flow divider returns to tank at no pressure. The cylinder slows before it contacts the work with this arrangement. At the same time, pilot-operated check valve E closes to prevent pump flow from bypassing to tank also. With the rod end of the cylinder connected to tank and the pump feeding the cap end, the cylinder generates full thrust.

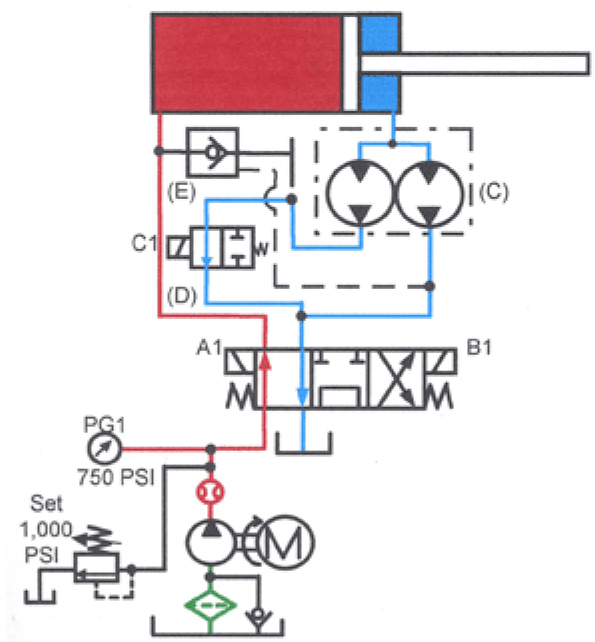


Figure 11-43. Regeneration circuit with motor-type flow divider that can be solenoid activated to full thrust. (Shown with cylinder extending at full power.)

In Figure 11-44, the cylinder is retracting. Energized solenoid B1 shifts the 4-way directional valve to send pump flow to one outlet of the flow divider. Both of the motors in the flow divider turn at the rate of flow from the pump. During this part of the cycle, the motor — with its inlet teed into the cap-end line — acts as a pump. Pilot pressure from the cylinder rod-end port opens check valve E to pass this flow. Pump flow, plus the same flow from the second motor, makes the cylinder retract twice as fast as a conventional circuit. However, cylinder thrust is only half that of a conventional circuit.

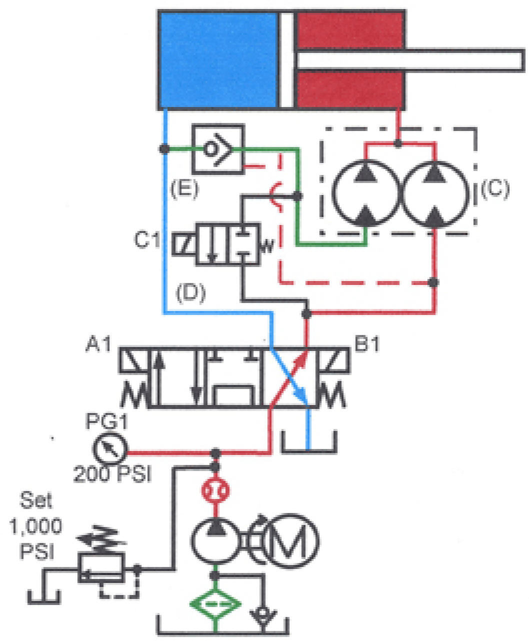


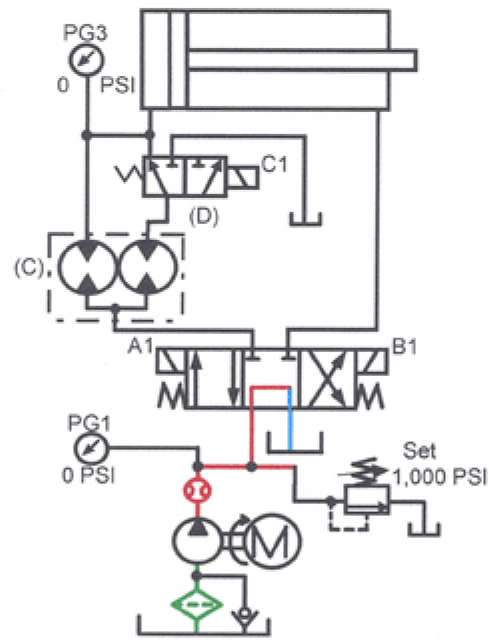
Figure 11-44. Regeneration circuit with motor-type flow divider that can be solenoid activated to full thrust. (Shown with cylinder retracting.)

Motor-type flow divider as an intensifier

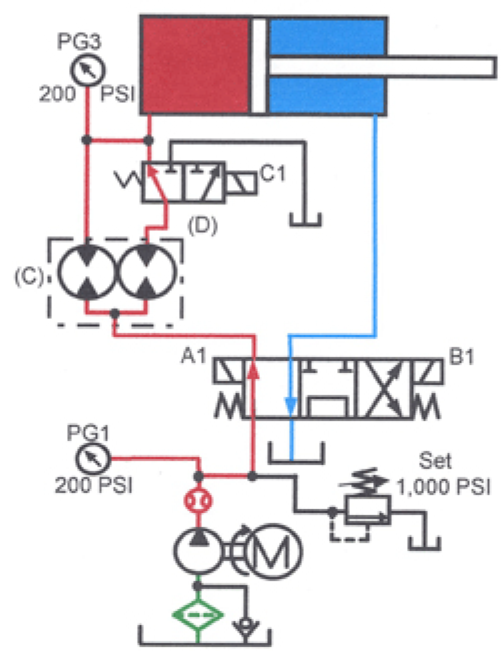
As noted earlier, a motor-type flow divider intensifies pressure at one outlet when the other outlet is at a lower or 0 pressure. In the case of a 2-outlet motor-type flow divider with equal displacements, if the inlet pressure is 1000 psi, one outlet can be at 2000 psi while the other outlet is at 0 psi. While pressure doubles, flow from the intensified outlet is one half that at the inlet. The energy from the outlet motor with 0 pressure transfers to the other motor via the common shaft, thus intensifying the pressure.

With more than one section going to tank — say a 4-outlet divider with three outlets to tank — intensification would quadruple the pressure. While the intensified fluid reaches that pressure, volume is only one-fourth of inlet flow.

Using motor-type flow dividers with unequal sections is another way to get high intensification. If the motor in one section discharges 3 gpm to tank and the other section discharges 1 gpm. Intensification is still 4:1.

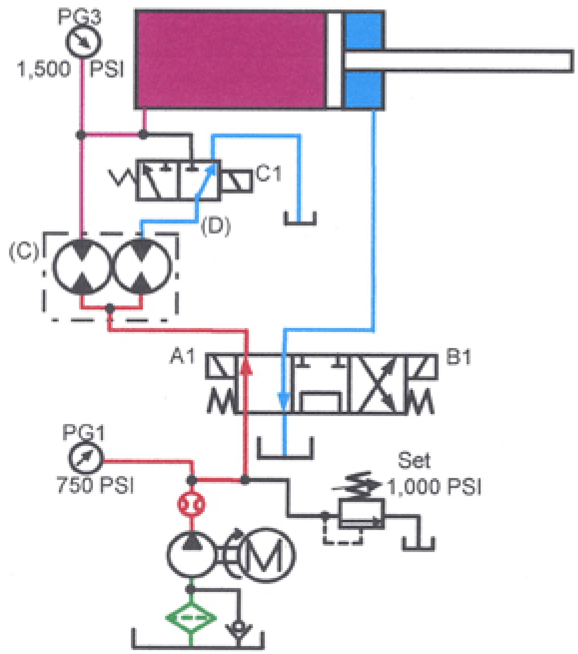


Figures 11-45 through 11-48 depict how to use this feature of motor-type flow dividers in a circuit. This circuit has equal-outlet flow divider C and 3-way directional valve D in the cylinder cap-end line. In the at-rest condition, shown in Figure 11-45, both outlets of the flow divider connect to the cap-end port.



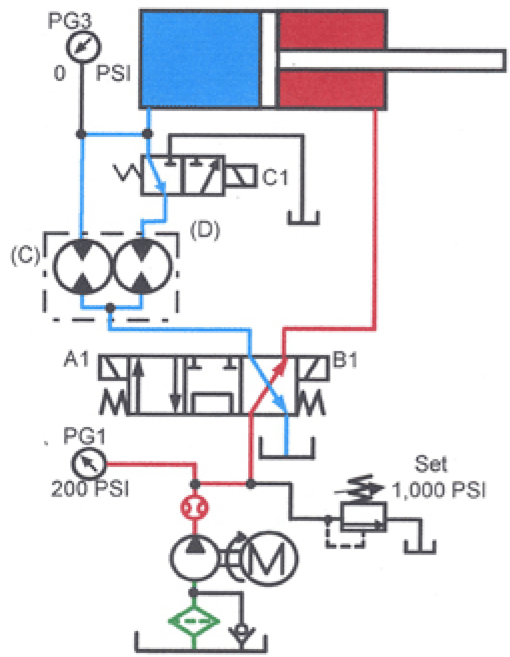
In Figure 11-46, the cylinder is extending at full speed and low thrust. Energizing solenoid A1 shifts the directional valve to pass oil through one side of the flow divider and the 3-way valve to the rod-end port. Fluid from the other side of the flow divider goes directly to the cylinder rod-

end port. The pump and valves must be sized to handle enough flow for the speed required during the fast-forward portion of the cycle. (Normally, motor horsepower is low for a cylinder moving a light load.)



When the cylinder contacts the limit switch, as in Figure 11-47, the switch energizes solenoid C1 on the 3-way valve. The valve shifts and oil from one section of the motor-type flow divider then goes to tank. Pressure doubles while cylinder speed drops to half what it was before solenoid C1 was energized.

This circuit works best on actuators that are not required to stall. Using this setup for a fast advance followed by a clamping operation might result in excess heat because internal leakage in the flow divider while the cylinder is holding.



Energizing solenoid B1, as in Figure 11-48, makes the cylinder retract. Oil from the cap-end port goes through both sections of the flow divider and back to tank through the directional valve.

When using a motor-type flow divider as an intensifier, make sure it is capable of operating at the elevated pressure. The pressure rating of an inexpensive gear-motor-type flow divider may be only 2000-psi intermittent and 1500-psi continuous. On the other hand, some gerotor-type flow dividers handle as much as 4500-psi intermittent and 3000-psi continuous — at a higher price.

Pressure controls (other than relief and unloading valves)

There are some parts of fluid power circuits that need pressure control. (Chapter 9 covered relief and unloading valves that control pressure in pump circuits.) Other types of pressure controls include sequence valves, counterbalance valves, and reducing valves. Though the internal works (and the symbols) are similar, these three pressure controls perform entirely different functions. Sequence valves and counterbalance valves are normally closed -- like relief valves and unloading valves -- but they usually allow bi-directional flow, so they need a bypass check valve in their bodies. Sequence valves always have an external drain connected directly to tank. Counterbalance valves are internally drained, except when used in some regeneration circuits.

Reducing valves are normally open and respond to outlet pressure to keep outlet flow from going above their set pressure. They also can have a bypass check valve. Reducing valves always have an external drain connected directly to tank. Any backpressure in this drain line adds to the valve's spring setting.

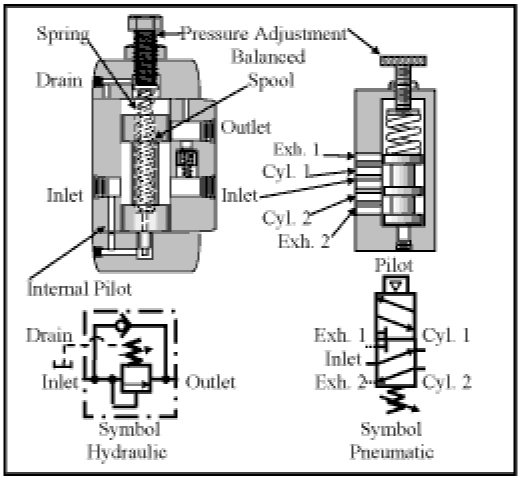
Relief valves, unloading valves, sequence valves, counterbalance valves, and reducing valves are the most difficult to discern on a schematic drawing because their symbols are so similar. Take extra care when diagnosing a problem to make sure these valves are correctly identified and their function understood.

Sequence valves

There are times when two or more actuators, operating in a parallel circuit, must move in sequence. The only positive way to do this is with separate directional control valves and limit switches or limit valves. This setup assures the first actuator has reached a specific location before the next operation commences. If there is no safety concern or possibility of product damage if the first actuator does not complete its cycle before the second starts, a sequence valve can be a simple way to control the actuators' actions.

*The symbols and cutaways in **Figure 14-1** are for hydraulic and pneumatic sequence valves. The main difference between these valves is that most hydraulic sequence valves are single purpose and must be used in series with a directional control valve, while many air sequence valves are pilot-operated directional control valves with an adjustable spring return. In either case, a preset pressure must be reached before the valves allow fluid to pass or change flow paths. Many manufacturers offer a direct-acting internally piloted hydraulic sequence valve like the design shown in **Figure 14-1**. This valve can be changed to external pilot in the field if required.*

Figure 14-1. Hydraulic and pneumatic sequence valves



Several manufacturers offer pilot-operated sequence valves also. Pilot-operated sequence valves stay closed to within 50 psi or less of their set pressure. Direct-acting sequence valves may partially open at pressures that are 100 to150 psi below set pressure -- and thus allow premature actuator creep.

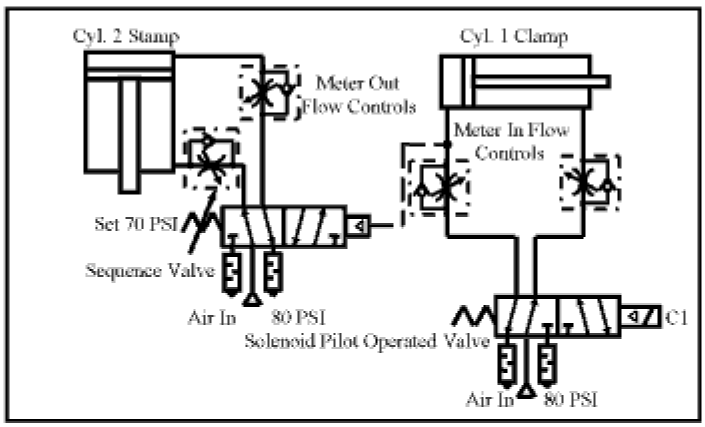
A balanced spool -- held in place by an adjustable-force spring -- blocks fluid at the hydraulic sequence valve's inlet. When pressure at the inlet reaches the spring setting, pressure in the internal pilot line pushes the spool up to allow enough flow to the outlet to keep pressure from going higher. Pressure at the inlet never drops below set pressure when there is flow to the outlet. When outlet pressure exceeds set pressure, the valve opens fully and pressure at both ports equalizes. Notice that the drain port hooked to tank must be at no pressure or constant pressure because any pressure in this line adds to spring setting. (Remember that a sequence valve must always have an external drain.)

A bypass check valve allows reverse flow when the valve is used in a line with bi-directional flow. In some applications a sequence valve may be externally piloted from another operation. Most valves can be converted in the field. (The designer should always change the part number to reflect the conversion.)

Pneumatic sequence valves typically are 5-way directional control valves with adjustable springs to set their shifting pressure. They are used to start a second operation after the preceding one finishes. Some older machines have one solenoid valve to start the cycle and several sequence valves to extend and retract all other actuators. Some precautions: • A sequence valve shifts on a pressure build-up and may start a second operation prematurely if an actuator stalls or is stopped for any reason. If personnel safety or product damage can occur due to an incomplete stroke, don't use sequence valves. Instead, use limit switches or limit valves and directional control valves for each operation sequence. • When flow controls are required they must be meter-in types. Take the signal to the sequence valve from the line downstream from the *flow control because pressure at this point will be whatever is required to move the actuator and its load.*

*The circuit in **Figure 14-2** is typical for air-powered machines. Cyl. 1 extends to clamp a part when an electrical input signal shifts the solenoid pilot-operated valve. As Cyl. 1 extends, pressure beyond the meter-in flow control at its cap end becomes as high as necessary to move the cylinder and its load. With the sequence valve set to shift at 70 psi, Cyl. 2 should not move until Cyl. 1 has extended and securely clamped the part. If the clamp does not make a full stroke for any reason, the Cyl. 2 extending prematurely will not damage the part or be unsafe. When the clamp is at 70 psi or higher, the sequence valve shifts to extend Cyl. 2. Both cylinders can return simultaneously without causing any problems.*

Figure 14-2. Typical pneumatic sequence valve circuit

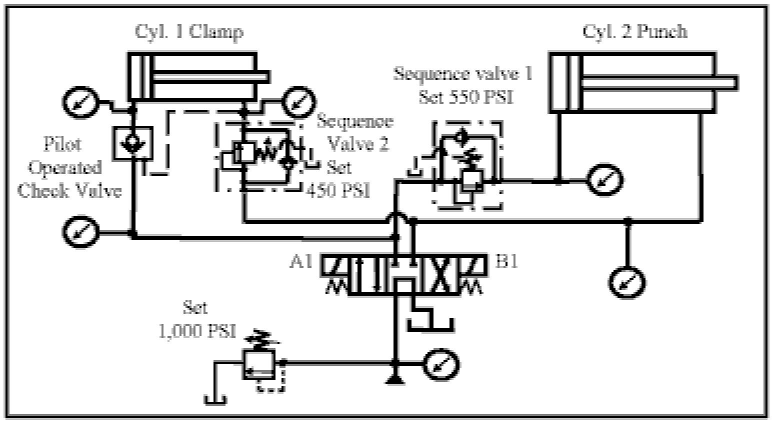


One great feature of a sequence-operated circuit is it does not matter how far the first cylinder must move before the next operation takes place. Thick or thin parts are clamped at the same force before the next operation starts because pressure must build to the same level to trigger the next sequence.

Cyl. 2 has meter-out flow controls to retard its movement and hold pressure on Cyl. 1 during the stamping operation. De-energizing the solenoid pilot-operated valve allows both cylinders to return home at the same time.

*The hydraulic sequence circuit in **Figure 14-3** is typical for a machine that must clamp and hold pressure while a second operation takes place. Sequence valve 1 is set at 550 psi; pressure at clamp Cyl. 1 must be at least 550 psi before punch Cyl. 2 can extend. While punch Cyl. 2 is extending, pressure in the circuit never drops below 550 psi. If the punching operation requires more than 550 psi, the pressure in the whole circuit increases -- up to the relief valve setting.*

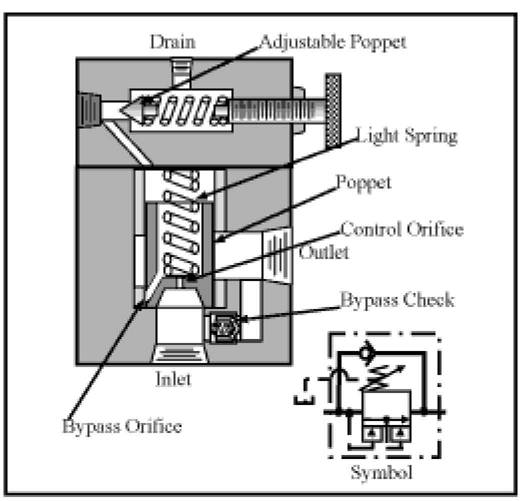
Figure 14-3. Typical hydraulic sequence valve circuit



Sequence valve 2 (set at 450 psi) keeps Cyl. 1 from getting a retract signal until Cyl. 2 has returned and pressure increases. A pilot-operated check valve maintains clamp force while the punch cylinder retracts. The signal to open the pilot-operated check valve comes from the line between Sequence Valve 2 and Cyl. 1, so there is no signal until Cyl. 2 fully retracts. (This circuit is not safe if pressure buildup comes from some source other than clamp contact or the end of stroke so that the punch cylinder operates prematurely.)

Sequence valves often generate a great deal of heat because the first actuator to move takes higher pressure than the subsequent actuators. This means there is usually a high pressure drop across a sequence valve that results in wasted energy. In some circuits, a kick-down sequence valve can reduce the energy loss. The cutaway view and symbol in ***Figure 14-4*** *show the inner workings of a kick-down sequence valve to explain how it controls opening pressure and then unloads it.*

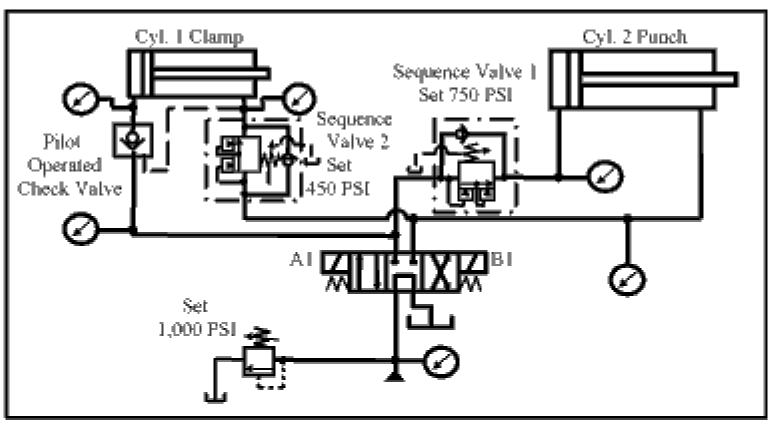
Figure 14-4. Kick-down sequence valve



Fluid from the inlet flows through the control orifice and up to the adjustable poppet where it is blocked. The resulting pressure tries to open the poppet while equal pressure and a light spring *acting on the opposite side hold it shut. When pressure increases enough to unseat the adjustable poppet and more flow starts passing the poppet than going through the control orifice, the pressure imbalance lets the poppet raise. When the poppet moves enough to let trapped fluid go through the bypass orifice, pressure on top of the poppet drops off -- because the bypass orifice is larger than the control orifice. At this point, the only force acting to hold the poppet shut is spring force and backpressure at the outlet port. When flow stops, the poppet closes again due to pressure equalization and spring force on the poppet.*

The circuit in ***Figure 14-5*** *is the same as in 14-3 except it incorporates kick-down sequence valves in place of standard sequence valves. Cyl. 2 will not extend in this circuit until pressure on Cyl. 1 has reached 750 psi. The difference is when a kick-down sequence valve opens at its pressure setting, it allows fluid to pass at 50 psi plus whatever it takes to overcome downstream resistance. This means the whole circuit from the pump to all actuators is 50 psi plus Cyl. 2's resistance. The pilot-operated check valve at Cyl. 1's cap-end port keeps it pressurized at near full force, while Cyl. 2 extends at low force. Energy waste is very low so heat buildup is minimal. (Other sequence valve circuits can be found in the e-book Fluid Power Circuits Explained by the author of this manual, which will be launched in the next few months.)*

Figure 14-5. Hydraulic circuit with kick-down sequence valves

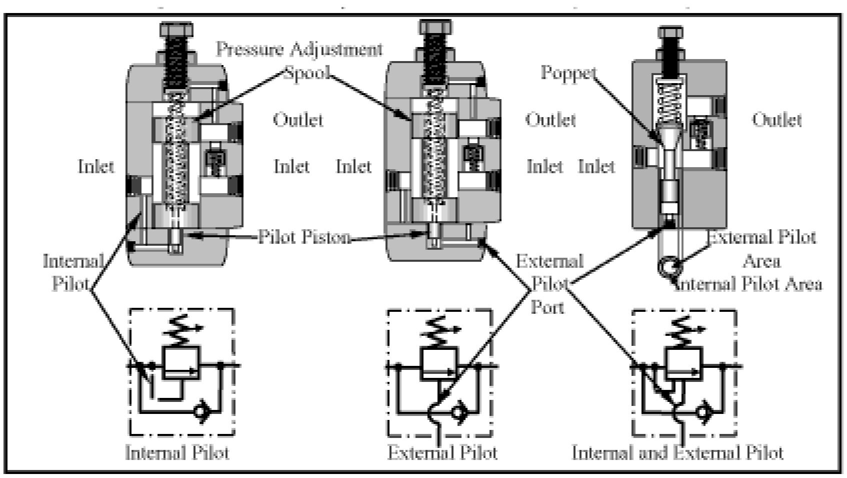


Counterbalance valves

*The fourth and last normally closed pressure control valve found in hydraulic circuits is the counterbalance valve. Cylinders with external forces -- such as weight from a platen, machine members, or tooling -- acting against them will overrun when cycled if oil flowing out of them is not restricted. A meter-out flow control circuit is one way to control overrunning loads but it has one main drawback. A flow control's speed is fixed except for manual adjustment or when using an infinitely variable proportional type. Because flow is fixed, the actuator will continue at the same speed – even when working flow to it increases or decreases. Thus, control is minimal and there could be high energy waste. (****Figure 13-8*** *shows a meter-out flow control circuit for running away loads.)*

*A counterbalance valve keeps an actuator from running away regardless of flow changes because it responds to pressure signals, not flow. A counterbalance valve is almost the same as a sequence valve except it normally does not have an external drain connection. The cutaways and symbols in **Figure 14-6** depict the physical makeup of three different counterbalance valves and how they are represented on a schematic drawing.*

Figure 14-6. Three types of counterbalance valves



The two cutaways and symbols on the left are spool designs with internal and external pilots. The valve on the right is a poppet design that is both Internally and externally piloted. Each valve type has advantages in different circuit arrangements that will be discussed later. A counterbalance valve usually has a bypass check valve for reverse flow because its most common use is in controlling actuators with running away or overrunning loads.

An internal pilot-operated counterbalance valve shifts to allow excess fluid to flow to the outlet when pressure at the inlet increases to the pressure set by the pressure adjustment. Pressure at the inlet never drops below set pressure when there is flow at the outlet. Flow from the inlet to the outlet is just enough so that backpressure on the actuator never drops below set pressure. This means the actuator moves only as fast as it is supplied and stops when Inlet flow ceases.

Pressure adjustment on the Internal-piloted counterbalance valve is usually made by first screwing the pressure adjustment all the way in. To assure that the valve is capable of high enough pressure, start the pump and raise the load a small amount. Then center the directional valve -- which connects the cylinder rod-end port to tank -- to see if it holds. If the load holds, next raise the load in increments -- checking for load stop every few inches. With the load suspended, start reducing set pressure on the counterbalance valve slowly until the load creeps forward. When the load starts drifting down slowly, increase pressure until movement stops, then turn the pressure adjustment another quarter to half turn higher. This method of adjusting usually wastes less energy while it always stops and holds the load.

The main disadvantage of an internal pilot-operated counterbalance valve is that backpressure is constant and it holds back even when the actuator needs maximum force. Another disadvantage is that to maintain optimum performance, an Internal-piloted counterbalance valve must be readjusted every time the load changes. The valve's main advantage is that it produces smooth cylinder action while advancing to the work.

An external pilot-operated counterbalance valve shifts to allow excess fluid flow to the outlet when pressure at the opposite cylinder port reaches the pressure set by the pressure adjustment. Pressure at the inlet never drops below load-induced pressure plus pressure set on the pressure adjustment when there is flow at the outlet. Flow from inlet to outlet is just enough that the actuator moves only as fast as it is supplied and stops when flow to the actuator ceases.

Pressure adjustment on the external pilot-operated counterbalance valve can be made on a test stand by setting the pressure adjustment at 100 to 200 psi. If pressure must be set on the machine, set the pressure adjustment higher than 200 psi and lift the load a small distance to make sure it stops and holds. If it holds, continue to raise the load high enough to have some time for the next step. Now, power the load down and observe pump pressure. Pump pressure while lowering the load should not exceed 200 psi. Continue this action until pump pressure is between 100 and 200 psi while the load is lowering. This method of adjusting usually wastes less energy while always stopping and holding the load.

The main disadvantage to an external pilot-operated counterbalance valve is that it may cause lunging or even stop cylinder action while advancing to the work. The main advantage is that backpressure is only present when the actuator is advancing to the work. At work contact, pressure at the actuator inlet increases and forces the counterbalance valve wide open, thus eliminating all backpressure. Another advantage is that an external pilot-operated counterbalance valve does not need to be readjusted when the load changes.

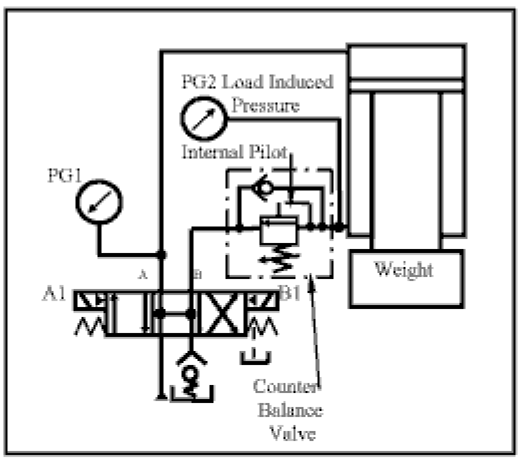
Internal and external pilot-operated counterbalance valves shift when pressure at the internal pilot area reaches the pressure set on the pressure adjustment and allows excess flow to go to the outlet. Pressure at the Inlet never drops below set pressure when there is flow at the outlet. Flow from the inlet to the outlet is just enough that backpressure on the actuator never drops below set pressure. This means the actuator moves only as fast as it is supplied and stops when Inlet flow ceases.

Pressure adjustment on an internal and external pilot-operated counterbalance valve is usually made by first screwing the pressure adjustment all the way in. To assure that the valve is capable of high enough pressure, start the pump and raise the load a small amount. Then center the directional valve that has the cylinder rod-end port connected to tank -- to see if it holds. If the load holds, then raise the load in increments -- checking for load stop every few inches. With the load suspended, start reducing set pressure slowly until the load creeps forward. When the load starts drifting down slowly, increase pressure until movement stops, then turn the pressure *adjustment another quarter to half turn higher. This method of adjusting usually wastes less energy while always stopping and holding the load.*

An internal and external pilot-operated counterbalance valve lowers loads smoothly and opens fully when pressure at the actuator inlet increases upon contact with the work. The valve does need to be readjusted when loads change, but this is a small price to pay for good control.

***Figure 14-7** depicts a vertically oriented cylinder with rod facing down and a load trying to extend it. To keep the cylinder from running away, the counterbalance valve must resist the load-induced pressure from the weight. The load-induced pressure can be calculated and the counterbalance valve could be preset at 100 to 150 psi higher on a test stand, but pressure adjustment is usually done at the machine (as mentioned earlier).*

Figure 14-7. Internally pilot-operated counterbalance valve circuit



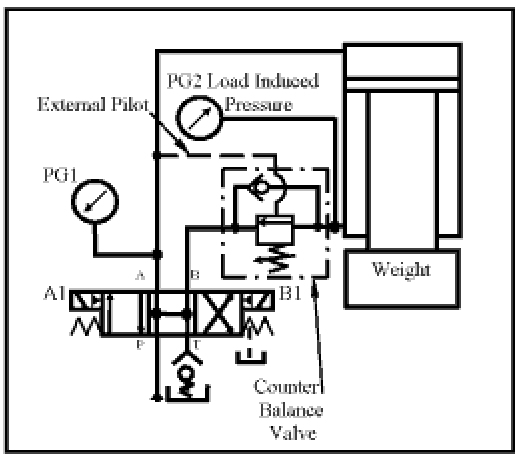
Notice that the directional control valve has ports A and B connected to tank in the center condition. There is no chance of extra pressure buildup in the pilot line while the circuit is at rest. If ports A or B were blocked, pressure could build and pilot the counterbalance valve open, allowing the cylinder to drift.

Energizing solenoid A1 sends pump flow to the cylinder cap end. As pressure builds there, pressure also increases in the rod end. When pressure at the cylinder rod end reaches 100 to 150 psi above the load-induced pressure, the cylinder starts to extend as fast as the pump fills the cap end. When flow increases, cylinder speed increases and when flow decreases, cylinder speed decreases.

As stated in the counterbalance valve explanation, backpressure at the cylinder rod end is present during the entire extend stroke. As a result, at work contact cylinder force is reduced by counterbalance pressure times the cylinder's rod-end area. The total weight of the platen and tooling on a press plus the amount of added pressure at the counterbalance valve cannot be used *to do work. Energy is expended to raise the weight but it is not recouped during the work cycle. Energizing solenoid B1 sends fluid around the counterbalance valve through the bypass check valve and on to the cylinder rod end to retract it.*

*The circuit in **Figure 14-8** shows the same cylinder with an external pilot-operated counterbalance valve. An externally piloted valve can be set at approximately 100 to 200 psi regardless of load-induced pressure in the cylinder. This is especially convenient in applications where loads constantly change. It is also the best use of energy because the counterbalance valve opens fully when the cylinder meets resistance so the weight is able to do some work. Because backpressure on the cylinder rod end is zero, more force is available.*

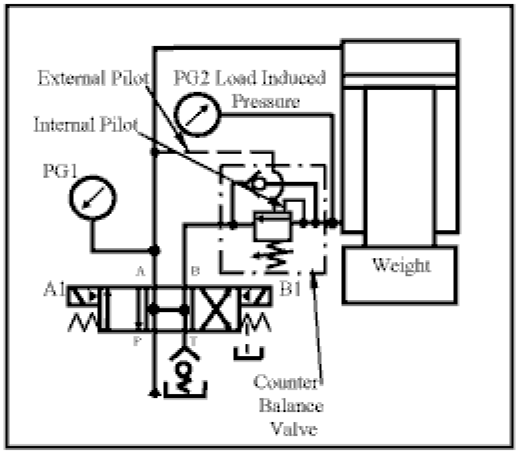
Figure 14-8. Externally pilot-operated counterbalance valve circuit



Energizing solenoid A1 sends fluid to the cylinder's cap end to start it extending. As pressure builds in the cylinder cap end, it pressurizes the external pilot and opens the counterbalance valve The valve only opens enough to let fluid out when the cap end is at pilot pressure. If pilot pressure is set too low, the counterbalance valve may quickly open too far -- allowing the cylinder to run away and pilot pressure to drop. At this point, the counterbalance valve shuts abruptly and the cylinder stops. Almost immediately, pressure again builds at the cylinder cap end, the counterbalance valve reopens, and the same scenario repeats until the cylinder meets resistance. A meter-in flow control in the external pilot line can help, but is very difficult to set. Energizing solenoid B1 sends fluid around the counterbalance valve through the bypass check valve and on to the cylinder rod end to retract it.

*The internal and external pilot-operated counterbalance valve in **Figure 14-9** incorporates the best features of both valves. The internal pilot provides a smooth advance stroke at low force, while the external pilot opens the valve fully to eliminate backpressure from the cylinder rod end when it contacts the workpiece. (Like the internally piloted valve. this version must be reset at each load change to maintain its efficiency and keep energy losses low.)*

Figure 14-9. Internally and externally pilot-operated counterbalance valve circuit



The symbols in these example circuits show a direct-acting pressure control valve. Several suppliers offer a pilot-operated version that is more stable and has less pressure differential between cracking and full flow operation.

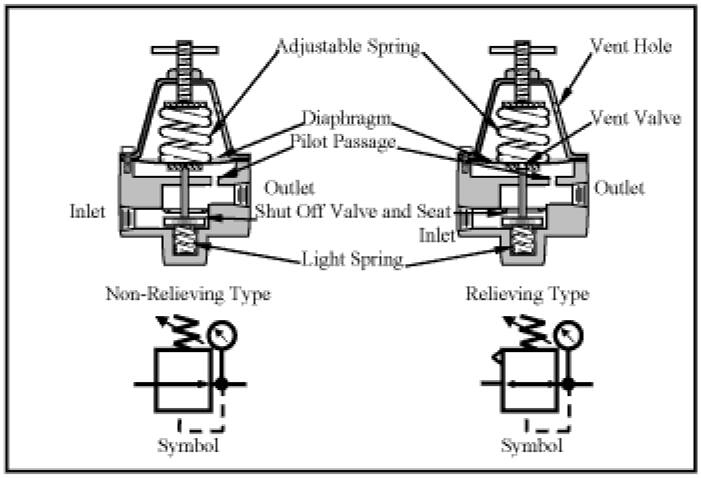
The circuits shown here work equally well with hydraulic motors, except that a counterbalance valve will not stop and hold a running away load on a motor without creep. All hydraulic motors have internal leakage that increases as the motor wears. The counterbalance valve may not have any bypass but fluid will slip by the motor parts no matter what its design.

There are no counterbalance valves for air circuits. Air circuits depend on meter-out flow controls to keep an actuator from running away. Usually an air circuit uses a 2-position valve that keeps pressure on the retract side at rest so it stays in place at end of stroke. When a load must be stopped in mid-stroke, a 3-position valve with cylinder ports blocked in center is the common method of trying to do this. There also is available a pilot-operated check valve for air service that gives some control for stopping and holding a pneumatic cylinder in mid-stroke.

Air line regulators

Most plant air systems produce pressures between 90 and 125 psi, while most air circuits are designed to operate at 75 to 85 psi. Other systems may operate at pressures as low as 15 to 20 psi. To accommodate these ranges, some method is needed to reduce the system pressure without wasting energy. A relief valve that would release plant air to atmosphere and try to lower the whole system it is not a good solution. The air line regulator shown in ***Figure 14-10*** *reduces outlet pressure by shutting off flow when downstream pressure tries to go above the regulator's setting. There is very little energy loss because air merely expands from its elevated pressure to meet the lower pressure requirement. In other words, an air compressor operating at 120 psi only has to run about a third as often when regulated or reduced to 40 psi.*

Figure 14-10. Air line regulators (or reducing valves)



This points out the main reason why an air line regulator should be set just high enough to do the job at hand. Without a regulator, not only does it cost more to operate a machine, but the machine tries to run a repetitive cycle with fluctuating pressure, therefore different forces and speeds.

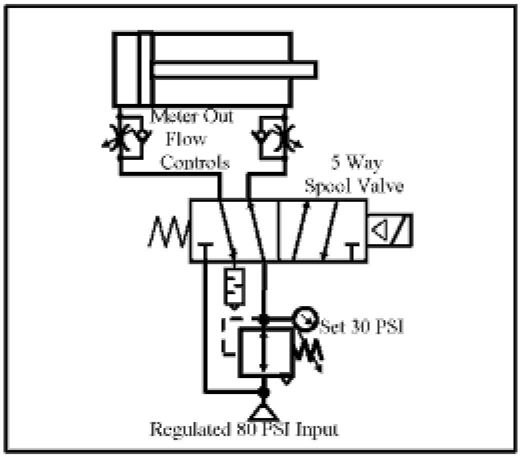
The cutaway views and symbols in ***Figure 14-10*** *show two common direct-acting air line regulators. They are normally available in sizes from 1/8-in. through 2-in. pipe thread. (Larger sizes are built but they usually are pilot-operated from a small direct-acting regulator.) Air flows freely from inlet to outlet until the outlet pressure reaches the set pressure. The adjustable spring holds the shut-off valve off its seat by extending the diaphragm during free flow. As pressure at the outlet continues to build, it passes through the pilot passage to the underside of the diaphragm. At set pressure, the diaphragm pushes the adjustable spring back, allowing the shut-off valve to seat. The light spring pushes the shut-off valve closed. Pressure at the outlet now is stable at its reduced setting -- as long as the inlet pressure is equal to or higher than the outlet. Any pressure drop at the outlet reduces pressure under the diaphragm and the adjustable spring again pushes the shut-off valve open to let more air in.*

If there is a possibility of the reduced pressure line seeing excess pressure for any reason, use the relieving-type regulator shown on the right in ***Figure 14-10****. This valve closes a hole through the diaphragm's center section with the shut-off valve's stem. After reaching set pressure, the shut-off valve cannot move up. Any extra pressure buildup under the diaphragm raises its center section off the shut-off valve's stem and allows air to flow to atmosphere through the vent hole. This feature should not be used as a relief valve function where pressure increases during every cycle -- it is only for occasional overpressure situations.*

Every pneumatically powered machine should have a regulator set for the lowest pressure that will produce good products. Costly overpressure should be eliminated in every case. Use an air line regulator anytime a job can be done at a pressure lower than plant air supply.

Another application for air line regulators that can save compressor output is reducing pressure on the return stroke of actuators that can use low power to retract. Many cylinders need high force to extend and do work, but the retract portion of the cycle needs very low force. An air line regulator positioned as shown in ***Figure 14-11*** *can save air during part of every cycle on many cylinder operations in most circuits.*

Figure 14-11. Pneumatic circuit with air-saving regulator

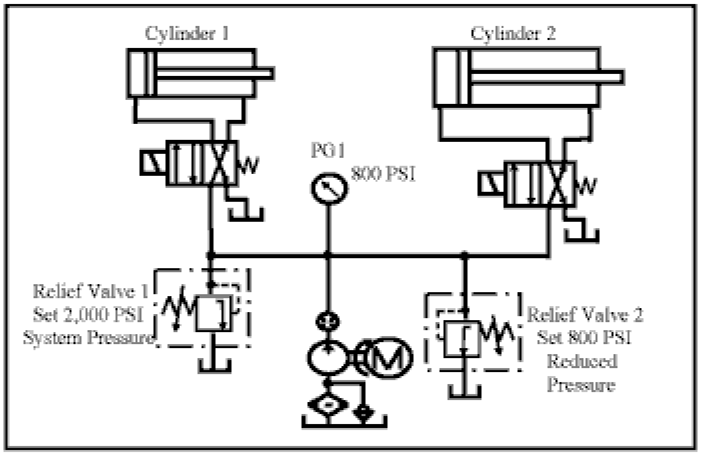


A 5-way spool valve, piped with a dual-pressure inlet as shown, can give normal cycle time while conserving plant compressed air. Return pressure is set on the regulator supplying the cylinder rod end at the lowest possible pressure that maintains cycle integrity. A reduction as small as 20 psi below working pressure can pay for the regulator in a short time. Shifting the 5-way valve starts the cylinder extending. (There will be a brief lunge as the lower-pressure air in the rod end compresses to hold back against the higher pressure in the cap end.) To control cycle time, adjust cylinder speed with the rod-end meter-out flow control. When the 5-way valve shifts again to return the cylinder, the meter-out flow control on the cap end must be adjusted for a faster rate because return power is limited.

Pressure-reducing and reducing//relieving valves

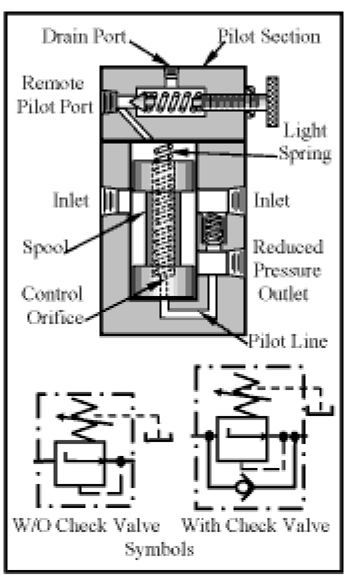
There are times in multi-actuator hydraulic circuits when system pressure is too high for some actuators while others need maximum force. One suggested remedy is the circuit in ***Figure 14-12****. Cylinder 1 needs 2000 psi to maintain force, while Cylinder 2 can damage the product when pressure exceeds 800 psI. Adding Relief Valve 2 (set at 800 psi) takes care of Cylinder 2's overpressure, but limits the entire circuit to 800 psi. Pressure in a circuit with more than one relief valve will never be higher than the setting of the lowest valve. The correct way to have two or more pressures in a single circuit is to incorporate reducing valves. (****Figure 14-14*** *diagrams a circuit using a reducing valve to give two pressures.)*

Figure 14-12. Dual-pressure hydraulic circuit with two relief valves



*The cutaway and symbol in **Figure 14-13** depicts a pilot-operated reducing valve that allows flow from the inlet to the reduced-pressure outlet until pressure reaches the setting on the direct-acting relief valve in the pilot section. Unlike the other four pressure controls (relief, unloading, sequence, and counterbalance valves), a reducing valve is normally open and blocks flow at set pressure.*

Figure 14-13. Pilot-operated reducing valve



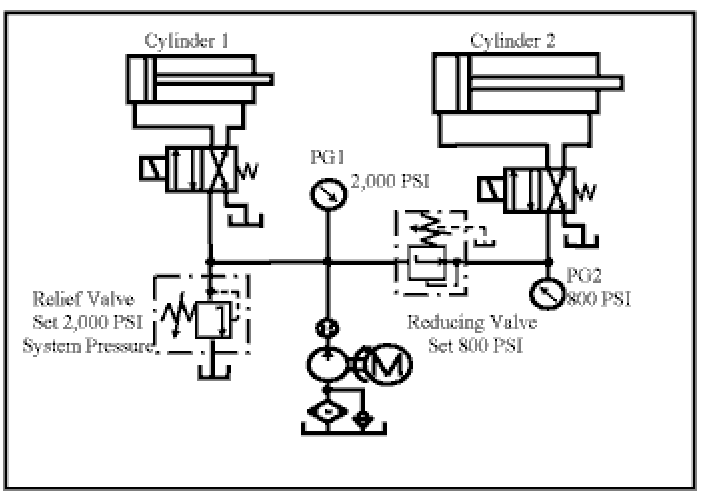
The normally closed direct-acting relief valve in the pilot section traps fluid from the reduced-pressure outlet port through the control orifice on top of the spool when pressure is below its setting. The spool stays in the normally open position because pressure on both ends balances it hydraulically while the light spring keeps it pushed down. As pressure at the reduced-pressure *outlet port continues to increase, it finally starts to open the direct-acting relief valve in the pilot section. Some fluid then flows to tank through the drain port. When flow through the direct-acting relief valve is more that the control orifice can handle, pressure on top of the spool drops and pressure on the bottom of the spool pushes it closed. The spool never closes completely because there is flow through the drain port anytime pressure at the outlet is lower than at the inlet. Drain port flow amounts to about 60 to 90 cim. This flow is all wasted energy and it can cause a system to overheat if more reducing valves are installed than necessary. When pressure drops below the direct-acting relief valve's setting in the pilot section, the valve closes and forces the spool to the open position.*

A reducing valve is normally open so it appears reverse flow should not be a problem. However, when the valve is working, it is almost closed -- and it can be held closed by back flow when the actuator starts to return. Anytime a reducing valve must pass reverse flow, select a valve with an integral bypass check valve to eliminate the possibility of blocked return flow.

It also is very important to have a free-flow drain port with very low (or even no) backpressure. Backpressure in the drain port adds to the setting of the direct-acting relief valve and can cause erratic results when drain pressure fluctuates. (Our next e-book, Fluid Power Circuits Explained, discusses how this drain port can be used advantageously in a dual-pressure circuit. This book will be launched in the next few months.)

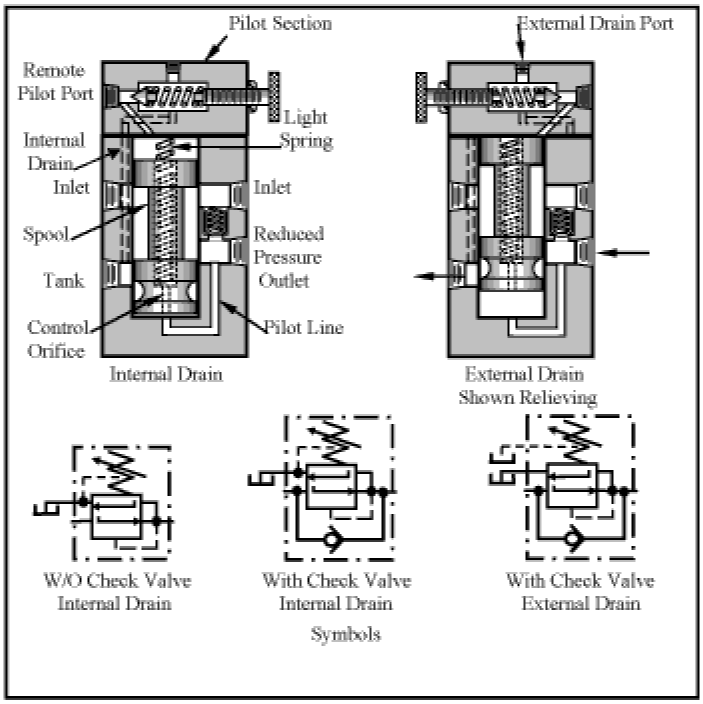
The modified circuit in Fig, 14-14 allows two pressures without lowering system pressure (as happened in ***Figure 14-13****). A pressure-reducing valve in place of Relief Valve 2 makes it possible to set pressure for Cylinder 2 without affecting pressure at Cylinder 1. This reducing valve never has reverse flow so a bypass check valve is not required.*

Figure 14-14. Dual-pressure hydraulic circuit using a relief valve



When it is working, a reducing valve is nearly closed and will pass very little reverse drain flow unless it has a bypass check valve. Even then, reverse flow must be at a pressure greater than that at the inlet. If this much pressure cannot be tolerated, use the reducing/relieving valve depicted in ***Figure 14-15****.*

Figure 14-15. Pilot-operated reducing/relieving valves



Reducing/relieving valves function exactly like reducing valves -- until an external force starts to increase pressure at the reduced-pressure outlet above the pressure set by the pilot section. When outlet pressure is 4 to 6% above set pressure, the spool moves up until the outlet is connected to tank. Any fluid at pressure above set pressure returns to tank, so outlet pressure does not continue to climb. Tank flow comes only from the reduced-pressure outlet, not from the pump through the inlet. When excess pressure at the outlet drops, the reducing/relieving valve continues to perform its reducing function.

Note that the left cutaway view has an internal drain for the pilot section. This saves connecting a separate drain line for pilot flow. However, when backpressure in the tank line is high or may fluctuate due to other return functions, it adds to the pilot-section setting and can elevate pressure at the reduced-pressure outlet above allowable rates. When tank-line backpressure may be high or when pressure fluctuations cannot be tolerated, use a valve with an external drain. When reverse flow is necessary, specify a model with an integral bypass check valve for piping convenience.

Figure 14-16. Modular sequence, counterbalance, and reducing/relieving valves

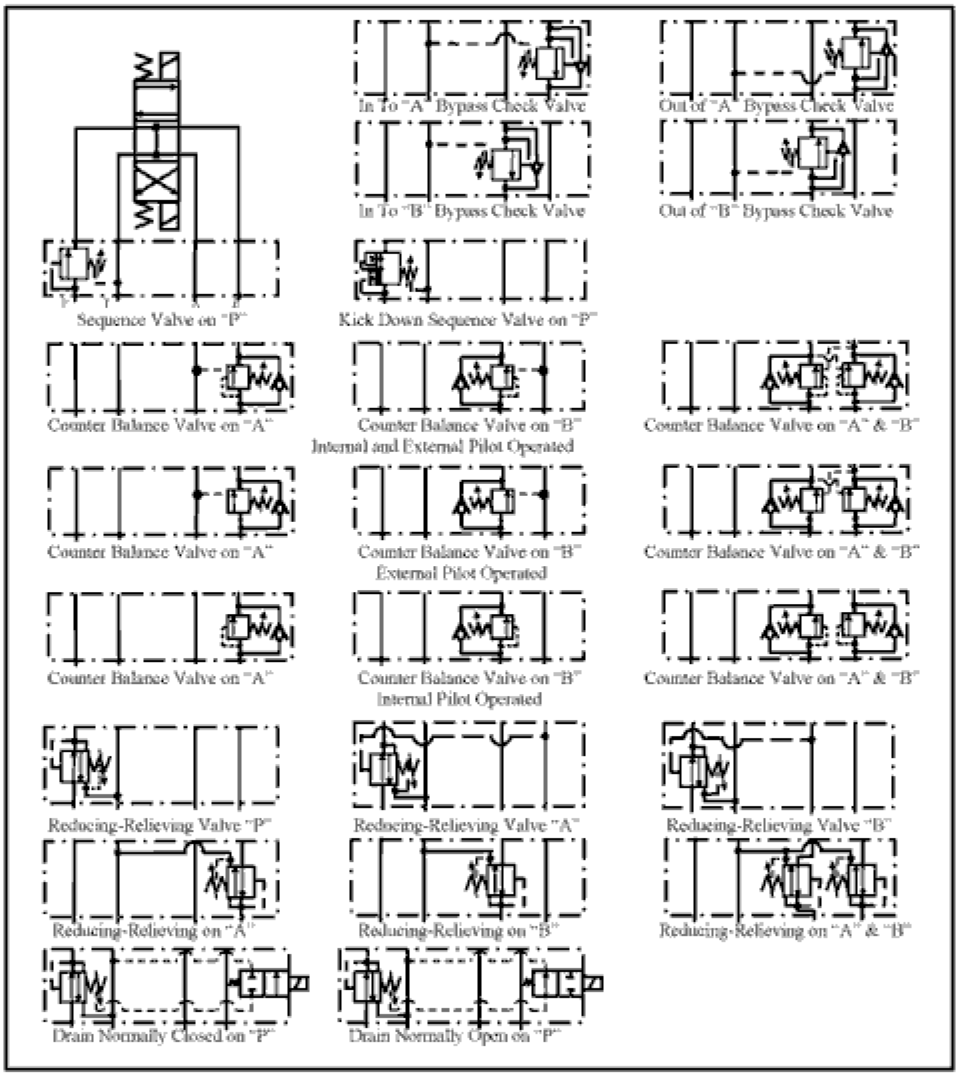


Figure 14-16 *shows most of the modular valve configurations for sequence, counterbalance, and reducing valves. Modular valves simplify piping and eliminate many connections that can generate backpressure or add potential leakage points.*

Sequence Valves

There are times when two or more cylinders need to stroke in a planned sequence. With two or more cylinders controlled by a single directional valve, the cylinder with the lowest resistance always strokes first. If the actuator with the least resistance is first in the sequence, the circuit runs smoothly without other valving.

When the cylinder that must move first has the highest resistance, a single directional control will not work. A separate directional valve for each cylinder is one way to sequence such a circuit. Energizing one solenoid extends the first cylinder. When the first cylinder contacts a limit switch, it energizes a second solenoid, causing the next cylinder to stroke. With this type of sequencing circuit, the first cylinder may lose holding power when the second directional valve shifts. It may require other valves to make sure the first cylinder generates and maintains the force required both before and during the second cylinder's stroke.

Another way to force fluid to take the path of greatest resistance is to use a pressure-control valve called a sequence valve.

Figure 20-1 shows the schematic symbol for an internally piloted sequence valve. A sequence valve symbol is similar to a relief valve symbol. The main difference is that a sequence valve always has an external drain line — and often has a bypass check valve for reverse flow.

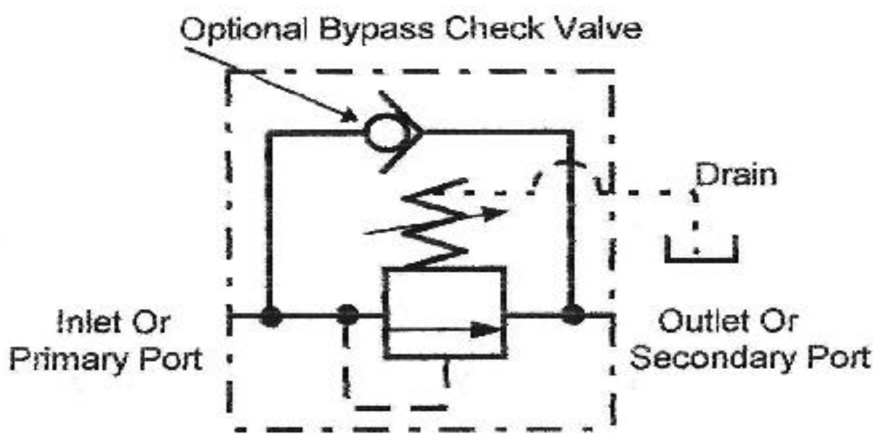


Figure 20-1. Internally piloted sequence valve.

A sequence valve is a pressure-operated, normally closed, poppet or spool valve that opens at an adjustable set pressure. Some designs use a spring acting directly on the spool or poppet, others are pilot-operated. A sequence valve always has an external drain port to keep from trapping leakage oil. Trapped fluid modifies set pressure at best or keeps the valve from opening at worst. For reverse flow capabilities, use the integral bypass check valve shown in the symbol.

Sequence valves may be internally pilot-operated as in Figure 20-1.This is the standard arrangement for the pilot source. Fluid at the inlet port of the valve cannot pass to the secondary circuit or outlet port, until reaching set pressure. Upon reaching set pressure, the valve opens enough to let excess pump flow pass on to the second operation.

The primary circuit never drops below the sequence valve setting as long as the primary pressure is equal to or greater than the sequence pressure setting. Pressure at the outlet port of the sequence valve is that required to overcome resistance in the secondary circuit when it is not above relief valve or pressure compensator setting.

Figure 20-2 pictures the symbol for an externally piloted sequence valve. In some circuits the pilot signal to open the valve is from a source other than the line feeding it. An external pilot-operated sequence valve opens and allows flow when a remote operation reaches a certain pressure.

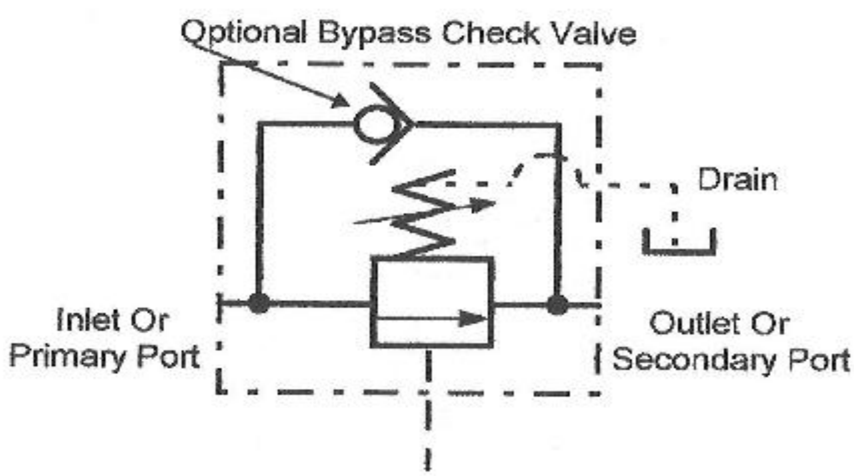


Figure 20-2. Externally piloted sequence valve

Sequence valves produce heat in a hydraulic system. With a pressure setting of 800 psi and resistance in the secondary circuit of 150 psi, there is a 650-psi pressure drop across the valve. This pressure drop results in heat, because its energy does not do useful work. Most sequence circuits require a heat exchanger, especially when they cycle rapidly.

Many older machines use sequence circuits because at the time they were designed there was a lack of understanding of electrical controls. Sequence circuits are unreliable and difficult to set up and maintain. Some older circuits have one directional valve and up to six sequence valves. With this many adjustments to make, it is hard to keep the cycle operating consistently.

Another potential problem with a sequence valve circuit is that actuator position cannot be assured. When a sequence valve shifts, the only sure thing is that pressure has reached a certain level. Pressure build up could be from a damaged or stalled cylinder or a kinked line. When it is necessary to positively locate an actuator, always use a limit switch or limit valve. When it is only necessary to know that pressure has built, a sequence valve in the line keeps fluid from the next action until the limit switch is contacted and pressure increases.

Figure 20-3 shows the symbol for a kick-down sequence valve. Its operation is different from a normal sequence valve. After a kick-down sequence valve reaches set pressure, flow passes through unrestricted. Pressure may have to reach 900 psi before flow passes through the valve, but when it starts passing, a kick-down sequence opens fully. A pressure drop of more than 50 psi across a kick-down sequence valve keeps it full open. (Note that a kick-down sequence valve causes less heat generation, but does not hold pressure on the primary circuit.)

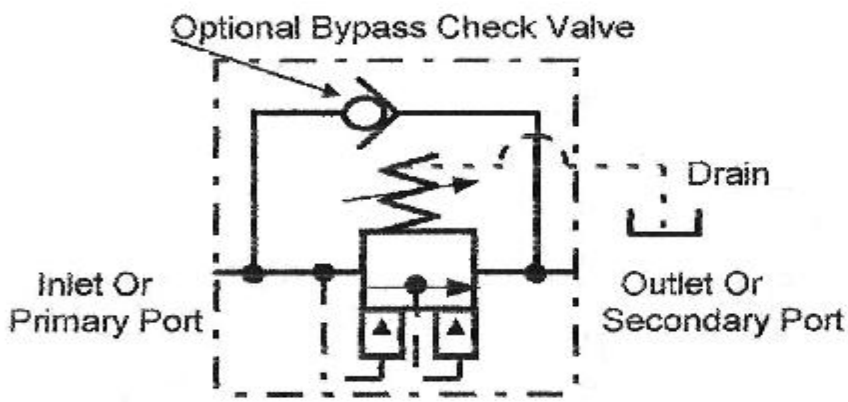


Figure 20-3. Kick-down sequence valve.

Figures 20-14 through 20-17 show a circuit using a kick-down sequence valve operating two cylinders. A pilot-operated check valve, added to the inlet of the first cylinder, maintains pressure on the first cylinder while the second cylinder strokes at low pressure.

Another use for a kick-down sequence is unloading a pump after the circuit reaches maximum pressure. A kick-down sequence valve keeps unloading the pump until pressure drop across it falls below 50 psi. (See further explanation in conjunction with Figures 20-23.)

When using flow controls with sequence circuits, meter-in flow control is the only workable option. Chapter 10 covering flow controls, explains the reasons for this.

Figures 20-4 through 20-11 provide schematic drawings for a two-cylinder sequence circuit. One 4-way directional control valve controls both cylinders. The sequence is: cylinder1 extend, *cylinder 2 extend, cylinder 2 retract, and cylinder 1 retract. Cylinder 2 will not extend until pressure at cylinder 1 reaches 600 psi.*

A good feature of a sequence circuit: if cylinder 1 is a clamp, it does not matter how thick the part is. Cylinder 2 will not extend until cylinder 1 securely clamps any thickness part. On the other hand, if the clamp cylinder locks up for any reason before contacting the part, pressure will build and allow cylinder 2 to cycle. Any sequence circuit may fail to operate correctly at any time because of outside influences.

Two-cylinder sequence circuit

Figure 20-4 shows a two-cylinder sequence circuit at rest. The schematic drawing notes the valve pressure settings. Gauges are placed to show working pressures as the sequence progresses.

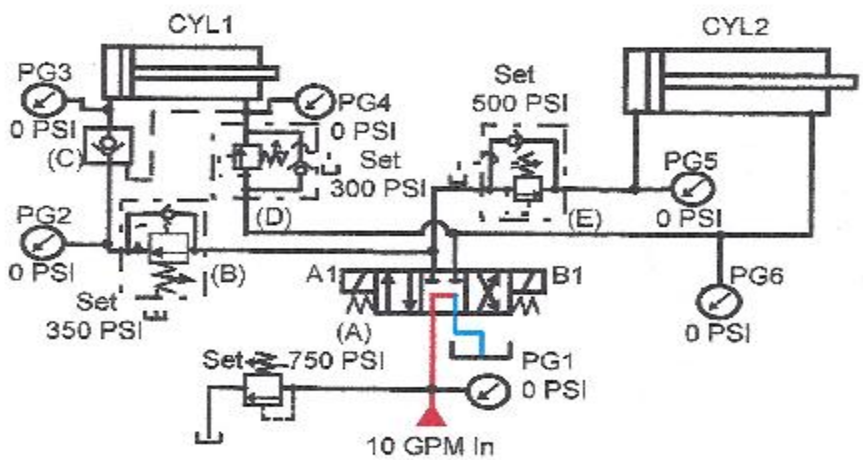


Figure 20-4. Two-cylinder sequence circuit.

In Figure 20-5, solenoid A1 is energized and CYL1 is extending. Pressure on gauges PG1-2 and 3 show the pressure required (100 psi) to move CYL1. Even if CYL2 required only 25 psi to move, sequence valve E keeps fluid from it. CYL1 extends until it contacts a part.

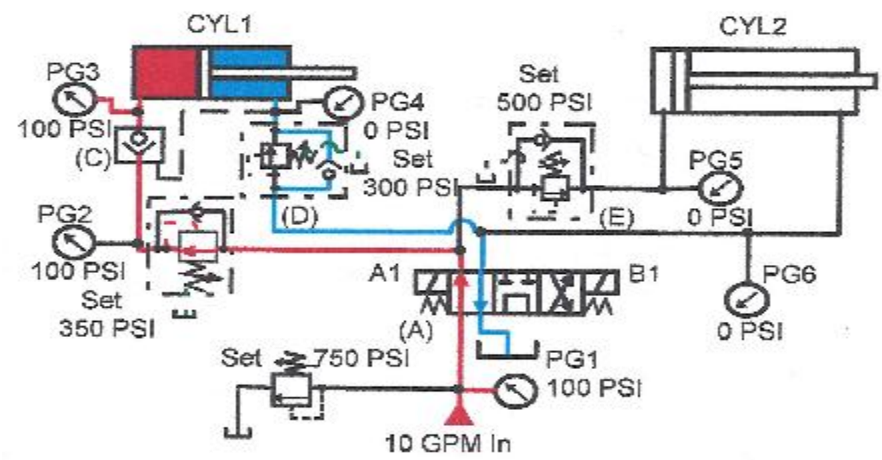


Figure 20-5. Two-cylinder sequence circuit

When CYL1 makes part contact, Figure 20-6, system pressure increases rapidly. As pressure passes through 300 psi (as seen on gauges PG1, 2, and 3), CYL2 is still stationary. Pressure continues to climb at CYL1 until it reaches 350 psi. When pressure is 350 psi at CYL1, reducing valve (B) shuts and holds. Pressure in the rest of the circuit continues to climb until it reaches 500 psi.

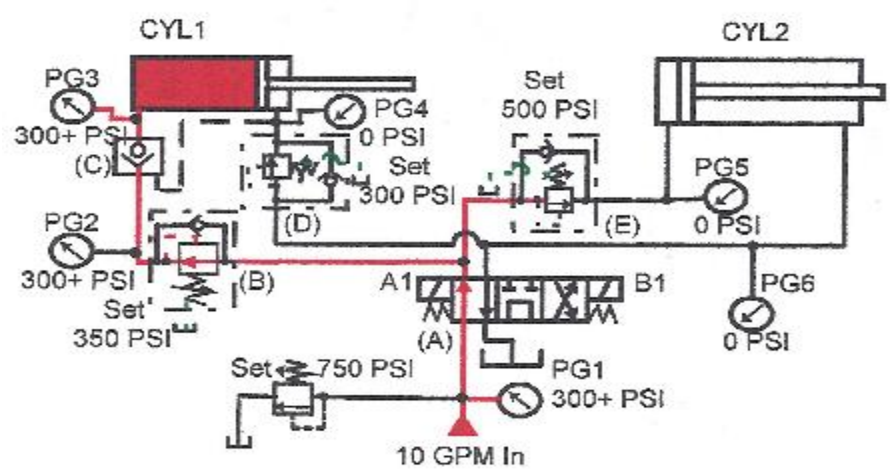


Figure 20-6. Two-cylinder sequence circuit.

When pressure reaches 500 psi, as shown in Figure 20-7, sequence valve Eopens enough to let excess pump fluid flow to CYL2. If pressure at CYL1 drops for any reason, sequence valve E shuts enough to keep system pressure at 500 psi or higher if possible. Now gauge PG1 reads 500 *psi while gauges PG2 and 3 read 350 psi, and gauge PG5reads whatever it takes to move CYL2. Pressure at CYL2 changes with load variations.*

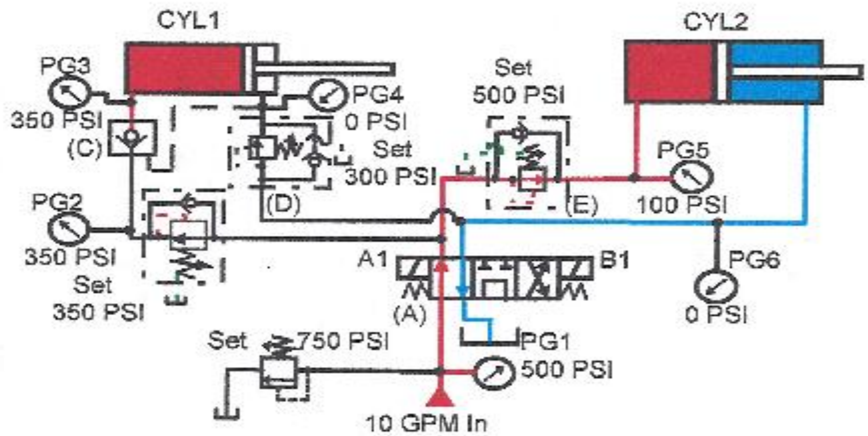


Figure 20-7. Two-cylinder sequence circuit.

When pressure on CYL2 is less than 500 psi, the pressure drop across sequence valve (E)generates heat. When pressure on CYL2goes higher than 500 psi, there is no energy loss, hence no heat. Because of reducing valve (B), pressure at CYL1 stays at 350 psi no matter how high system pressure climbs.

When CYL2 bottoms out, as in Figure 20-8, pressure at gauges PG1 and PG5 goes to 750 psi and the system relief valve starts dumping fluid to tank. Pressure at CYL1 stays at 350 psi because reducing valve (B) will not let it go higher. Reducing valve (B) prevents CYL1 from crushing the part while CYL2 does its work.

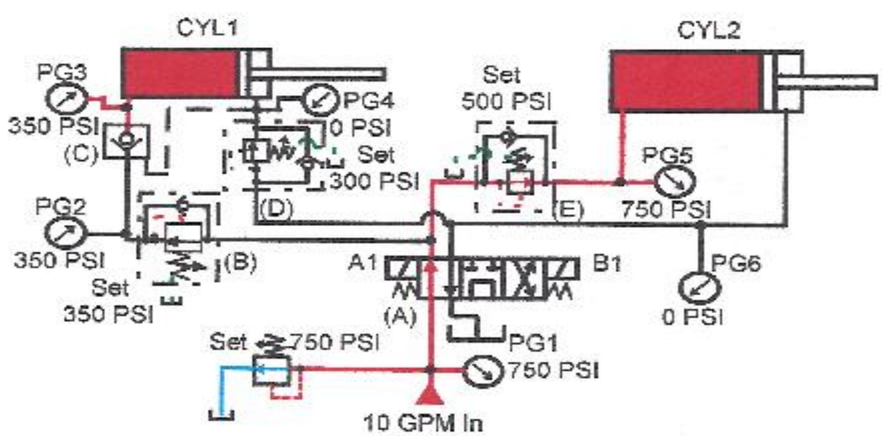


Figure 20-8. Two-cylinder sequence circuit.

Energizing solenoid B1 on directional valve (A) , Figure 20-9, starts returning the cylinders to their at-rest positions. In this circuit, CYL2 retracts first, while CYL1 holds under pressure when the directional valve shifts. Pilot-operated check valve (C) traps oil in the cap end of CYL1 — note gauge PG3 — so it does not relax and release the part. Oil now goes to CYL2 and sequence valve (D) . CYL2 retracts first because it only takes 100 psi to move it, while the pressure setting at sequence valve (D) is 300 psi. Pressure at CYL2 changes as it retracts, but never goes higher than 200 to 250 psi.

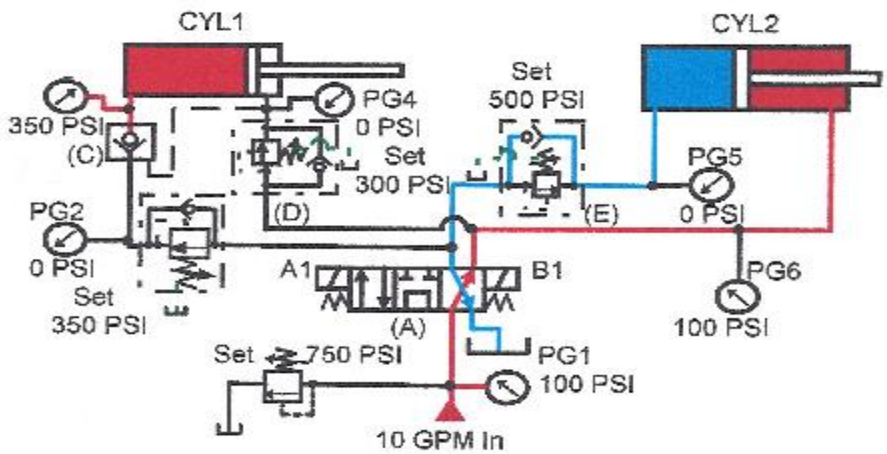


Figure 20-9. Two-cylinder sequence circuit.

When CYL2 fully retracts, system pressure increases rapidly, as seen in Figure 20-10. Pressure in the rod end of CYL2 finally increases to 300 psi. CYL1 still has approximately 350 psi in its cap end due to pilot-operated check valve (C) . (Pressure at CYL1 may drop due to leaks at seals or piping with the circuit described here.) With a short cycle time, pressure drop is minimal. If decreasing pressure is a problem, tee a small accumulator in the line between pilot-operated check valve (C) and the cylinder. See a leakage make-up circuit using an accumulator in Chapter 1, Figures 1-24 to 1-27.

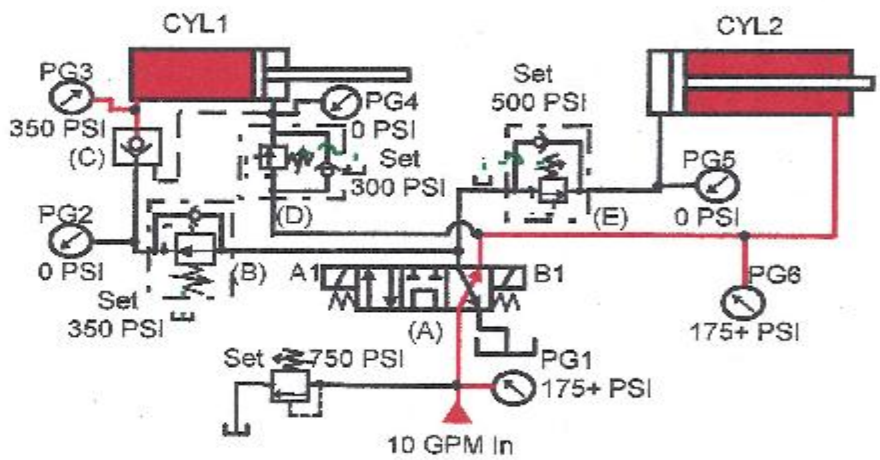


Figure 20-10. Two-cylinder sequence circuit.

When pressure reaches 300 psi in Figure 20-11, CYL1 starts to retract. Because of sequence valve (D) , pressure at the rod end of CYL2 stays at 300 psi. When oil passes through sequence valve (D) , it first sends a pilot signal to open pilot-operated check valve (C) . After valve (C) opens, CYL1 can retract. Pressure at the rod end of CYL1 is whatever it takes to stroke the cylinder home.

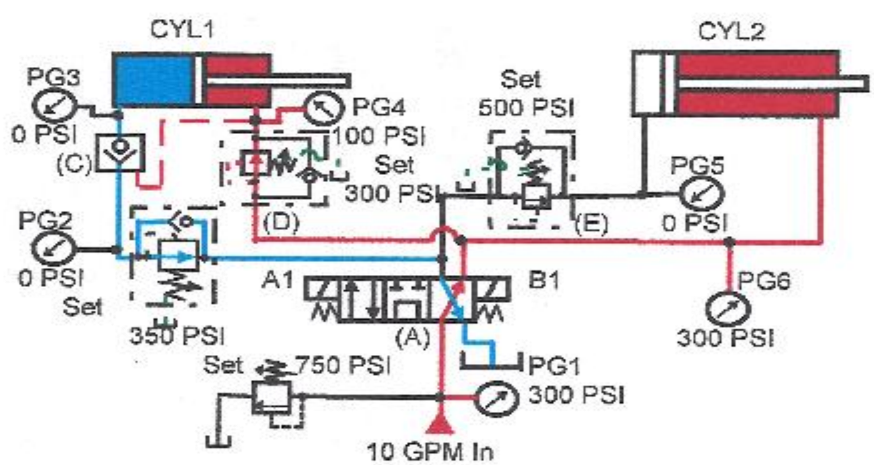


Figure 20-11. Two-cylinder sequence circuit.

Simple sequence circuit using modular valves

Figures 20-12 through 20-15 show a modular or sandwich-type sequence valve in a circuit. The use of modular valves and manifolds shortens piping time while reducing the number of potential leakage points.

Figure 20-12 pictures the system at rest. A fixed-displacement pump, unloading at no pressure through a tandem-center valve, is the power source. This circuit has some heat generation but makes sure the Clamp cylinder never goes below a certain pressure while the Work cylinder extends and retracts. Also, the Work cylinder cannot even try to extend until CYL1 makes a limit switch.

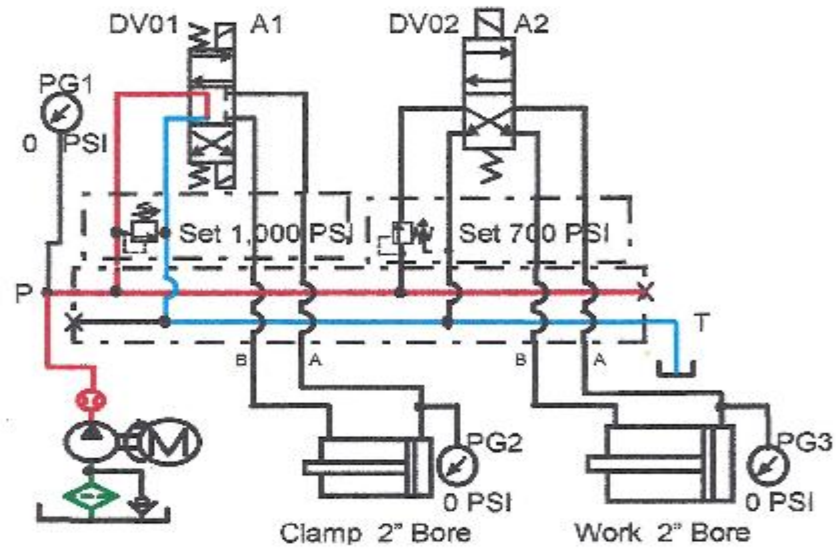


Figure 20-12. Sequence circuit to maintain clamp force.

In Figure 20-13, solenoid A1 on directional valve DV01 is energized. Pump flow goes to the Clamp cylinder, extending it to the work. Because this requires only low pressure and uses all the pump flow, there is no heat generation.

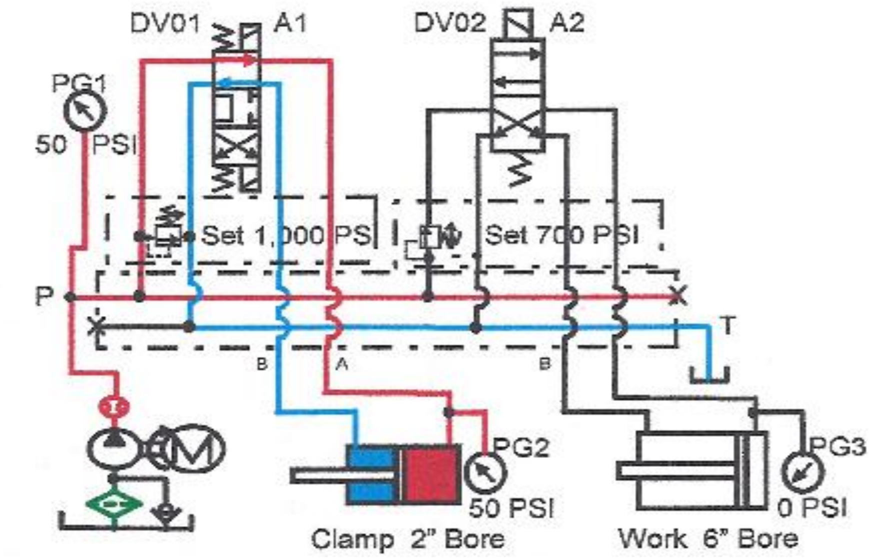


Figure 20-13. Sequence circuit to maintain clamp force.

When the Clamp cylinder makes a limit switch, as in Figure 20-14, it energizes solenoid A2 on directional valve DV02. The modular sequence valve under DV02 assures that the Clamp cylinder sees at least 700 psi before the Work cylinder extends. If the Work cylinder only requires 450 psi to extend, the 300-psi energy loss generates heat. With this sequence circuit, Clamp cylinder pressure cannot drop below 700 psi while the Work cylinder strokes. Directional valve DV02, shifted by a limit switch, ensures that the Clamp cylinder contacts the part before the Work cylinder cycles.

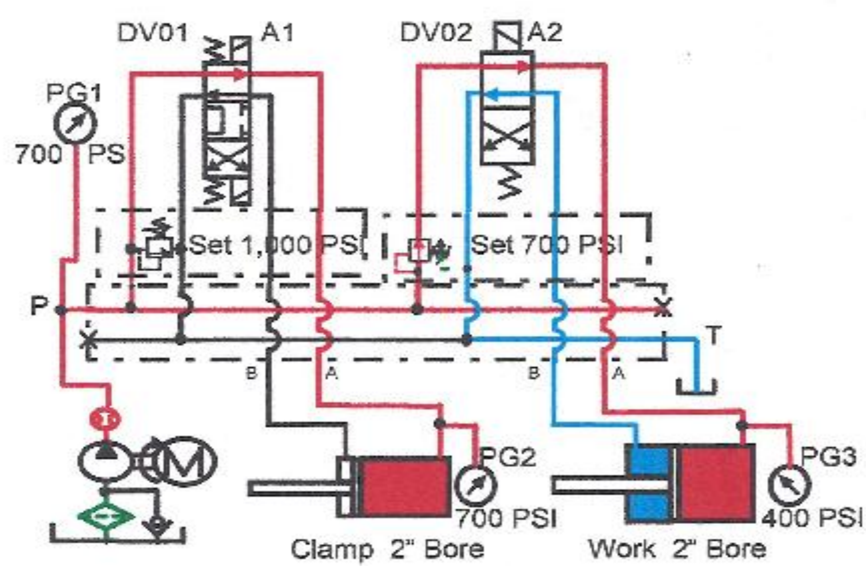


Figure 20-14. Sequence circuit to maintain clamp force.

To retract the Work cylinder, deenergize solenoid A2 as in Figure 20-15. This directs oil from the pump to the rod end of the Work cylinder. The Clamp cylinder still has 700 psi on it to hold the part firmly while the Work cylinder returns home.

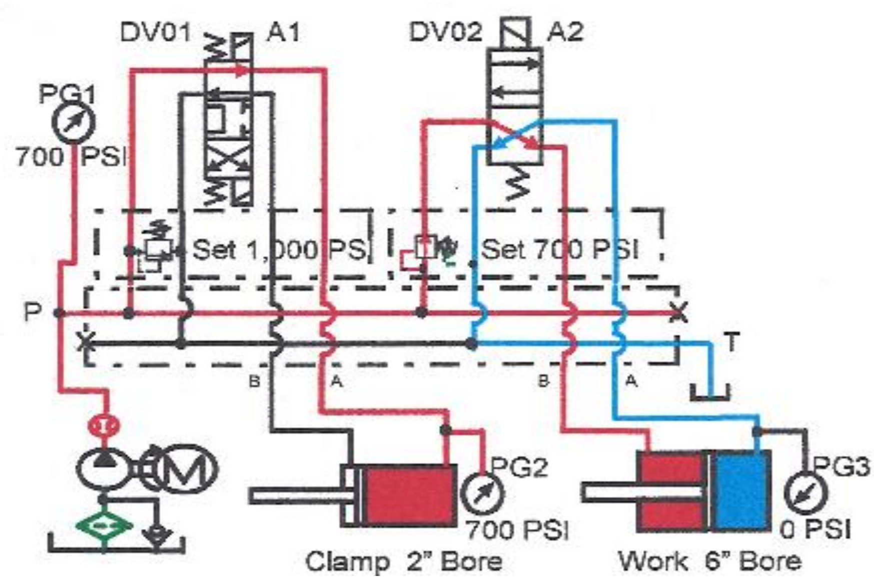


Fig.20-15. Sequence circuit to maintain clamp force.

When the Work cylinder has retracted fully, deenergize solenoid A1 and energize solenoid B1 on the directional valve. The Clamp cylinder returns, DV01 deenergizes, and the cycle ends.

Adding more directional valves like DV02 and more modular sequence valves will ensure the proper pressure for more working functions. A single screw-in cartridge sequence valve — added to the bar manifold in the pump line between the clamp valve and the working cylinder valves — can eliminate multiple sequence valves. The extra cost of special manifolds for this arrangement is a good investment.

Sequence circuit with kick-down sequence valve

Figures 20-16 through 20-19 show a kick-down sequence valve in place of the standard sequence valve. With a kick-down sequence valve, add the modular pilot-operated check valve shown in the cap-end line of the Clamp cylinder. A pilot-operated check valve blocks pressurized fluid in the cap end of the clamp cylinder when the kick-down sequence valve opens.

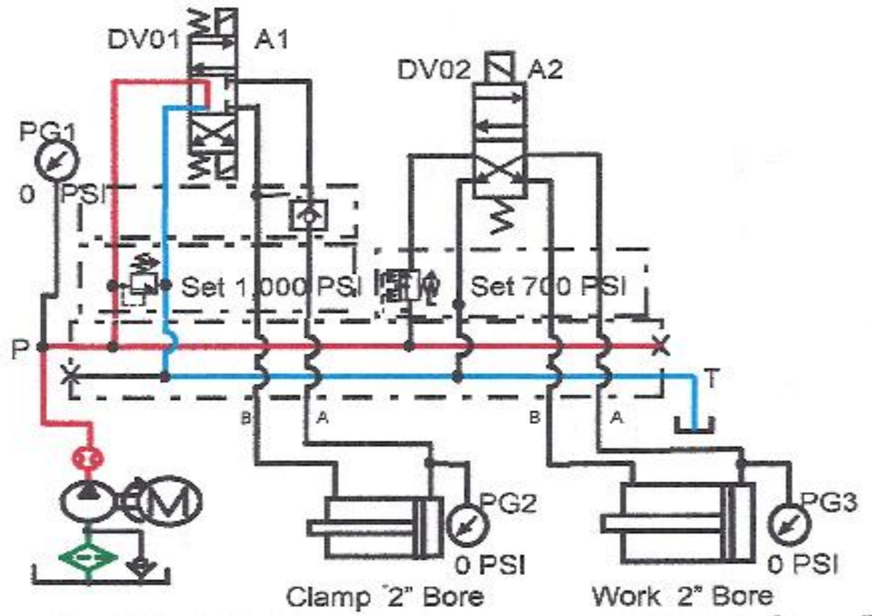


Figure 20-16. Kick-down sequence circuit to maintain clamp force.

In Figure 20-17, solenoid A1 on directional valve DV01 is energized. Pump flow goes to the Clamp cylinder through the pilot-operated check valve to extend the Clamp cylinder. Because this takes low pressure and all pump flow, there is no heat generation.

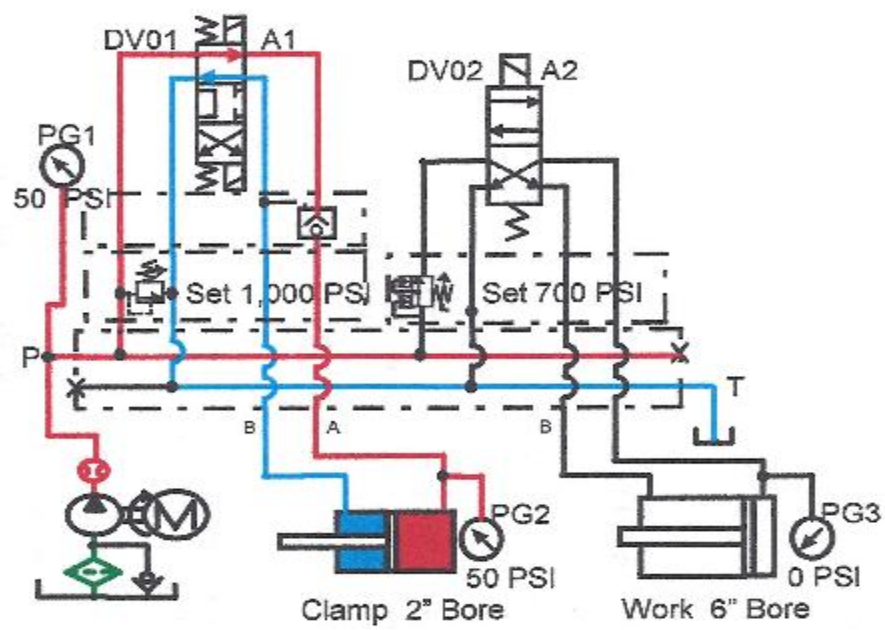


Figure 20-17. Kick-down sequence circuit to maintain clamp force.

After the work is contacted, Figure 20-18, energize solenoid A2 on directional valve DV02. The modular kick-down sequence valve under DV02 causes 700-psi pressure to build in the Clamp cylinder before the Work cylinder extends. If the Work cylinder only requires 450 psi to advance to the work, system pressure drops to 450 psi with minimal heating.

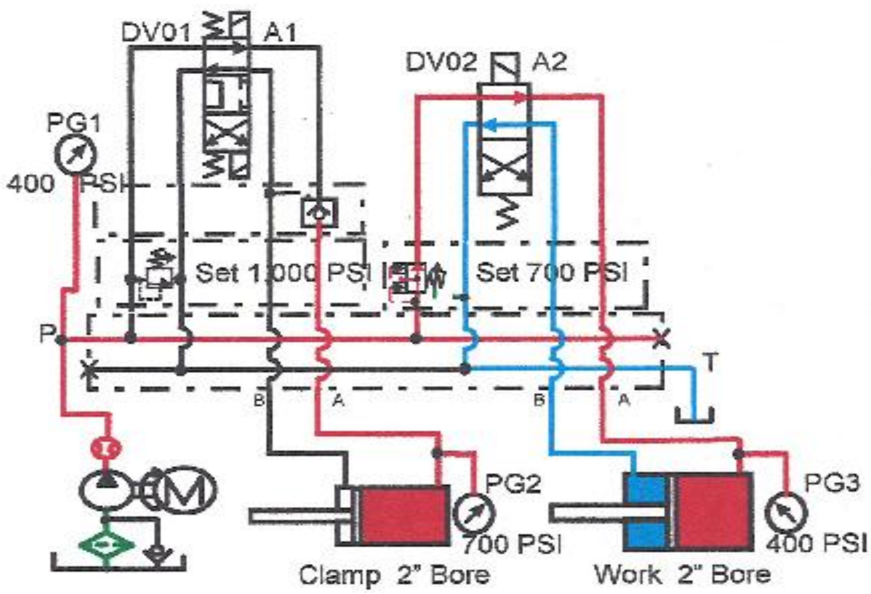


Figure 20-18. Kick-down sequence circuit to maintain clamp force.

The Clamp cylinder maintains force because the modular pilot-operated check traps 700-psi fluid in it. With a short cycle time, pressure drop is minimal. If decreasing pressure is a problem, tee a small accumulator in the line between the manifold and the cylinder's cap-end line. See a leakage make up circuit using an accumulator in Chapter 1, Figures 1-24 to 1-27.

To retract the Work cylinder, deenergize solenoid A2 as in Figure 20-19. This sends pump flow to the rod end of the Workcylinder. The pilot-operated check still maintains 700 psi or higher on the Clamp cylinder, holding the work firmly while the Work cylinder retracts.

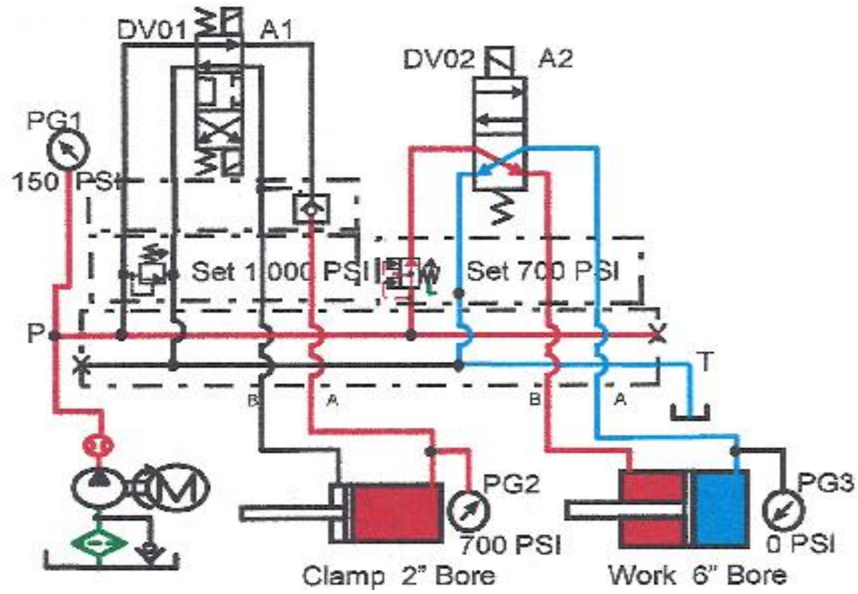


Figure 20-19. Kick-down sequence circuit to maintain clamp force.

After the Work cylinder retracts fully, deenergize solenoid A1 and energize solenoid B1 on the directional valve, retracting the Clamp cylinder to its home position. Pilot pressure from port B opens the pilot-operated check valve, allowing trapped fluid to leave the cap end of the Clamp cylinder. After the Clamp cylinder returns, DV01 deenergizes and the cycle ends.

Pump unloading with kick-down sequence valve

Figures 20-20 through 20-23 show a kick-down sequence valve automatically unloading a pump at the end of a cycle. The kick-down sequence valve and a single-solenoid directional valve can replace an open- or tandem- center, 3-position valve unloading circuit. This circuit simplifies the electrical controls because it only uses one solenoid.

A punch cylinder application, using a single-solenoid, two-position, spring-return valve and a fixed-displacement pump could operate this way with little heat generation. Figure 20-20 shows the circuit at rest. The pump, unloading through kick-down sequence valve (A) at about 50 psi, is ready for a cycle.

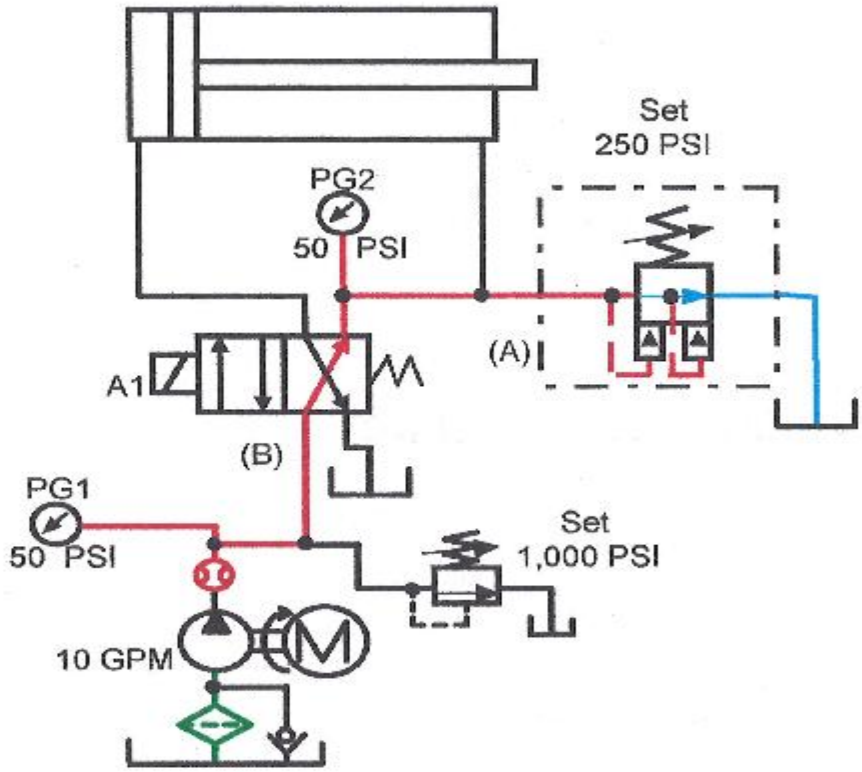


Figure 20-20. Pump-unloading circuit with kick-down sequence valve — at rest with pump running.

Energizing solenoid A1 on directional valve (B), Figure 20-21, directs oil to the cap end of the punch cylinder. The cylinder extends at the pressure required to move it. Oil from the rod end of the cylinder flows freely to tank, greatly reducing pressure in this line. The pressure drop allows kick-down sequence valve (A) to close (or reset) for the retract cycle.

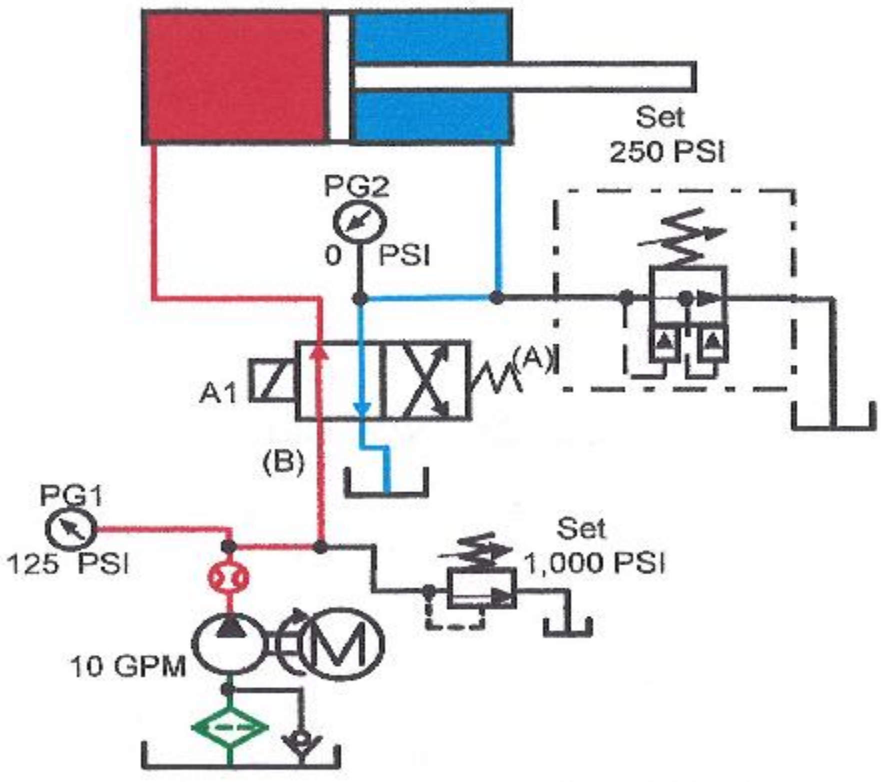


Figure 20- 21 Pump-unloading circuit with kick-down sequence valve — cylinder extending.

The cylinder extends until it meets the part. Pressure builds until the punch goes through the part. A limit switch then deenergizes solenoid A1 on directional valve (B) . Directional valve (B) spring returns to normal and cylinder travel reverses and retracts.

The cylinder retracts at the pressure required to move it, Figure 20-22. Kick-down sequence valve (A) stays closed because its setting is higher than the pressure retracting the cylinder. The cylinder retracts until the end of stroke or until it contacts a dead stop.

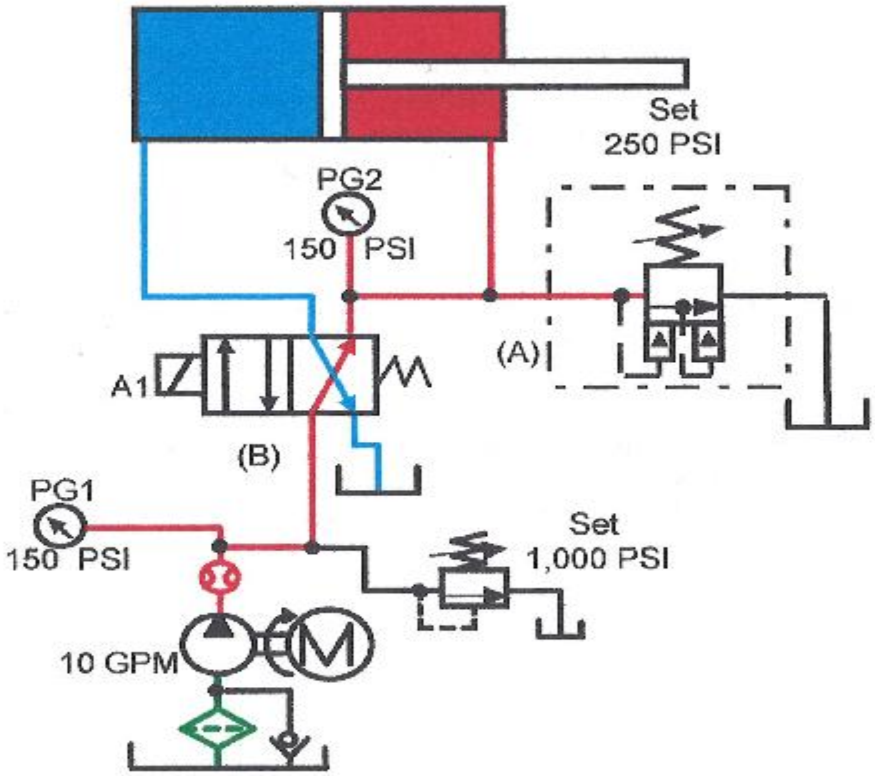


Figure 20-22. Pump-unloading circuit with kick-down sequence valve — cylinder retracting.

When the retracting cylinder stops, Figure 20-23, pressure builds in its rod end. When pressure reaches the setting of kick-down sequence valve (A) , the valve opens and unloads the pump to tank at approximately 50 psi. The circuit has returned to the conditions of Figure 20-20.

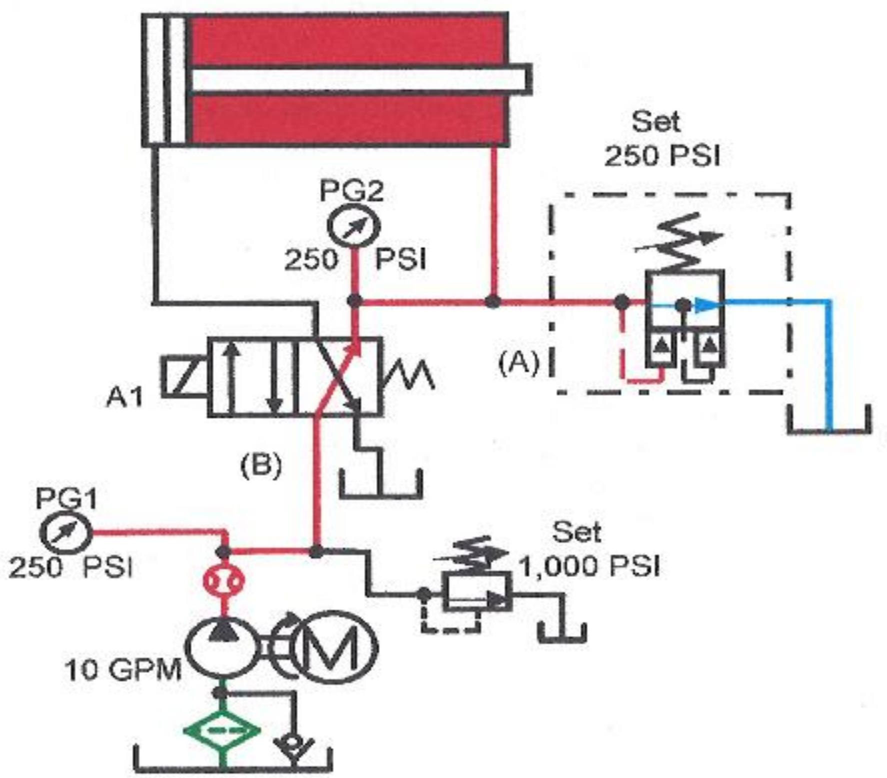


Figure 20-23. Pump-unloading circuit with kick-down sequence valve — pump just starting to unload.

A kick-down sequence valve is a unique component that can simplify the electrical control of systems with one or two cylinders. At the same time, there is minimal energy loss and heat generation.

CAUTION: *When using any pressure-control valve, the only thing certain when they operate is that they have reached set pressure.*

Pressure-compensated pump with self-adjusting overpressure relief valve
Some designers use a relief valve with a pressure-compensated pump to reduce pressure spikes as the pump rapidly goes from full flow to no flow. Figure 1-16 in Chapter 1 shows and explains a circuit using a pressure-compensated pump and a relief valve. Figures 1-17 to 1-19 show another circuit using an accumulator to protect the pump. The accumulator circuit almost eliminates pressure spikes — plus it gives faster actuator response at the start of a cycle. However, there are pressure-compensated pump circuits that need overpressure protection that an accumulator may not be capable of providing alone. The schematic diagrams in Figures 20-20 through 20-24 show a circuit with a cylinder opposed by a greater force than its pressure capabilities. When an outside force starts pushing back against the cylinder, pressure in its cap

end increases. Without a relief valve in the circuit, pressure might easily exceed the rating of the valves, piping, pump, and cylinder. This happens because a pressure-compensated pump compensates to no flow at set pressure, but will not allow reverse flow to relieve pressure above its setting.

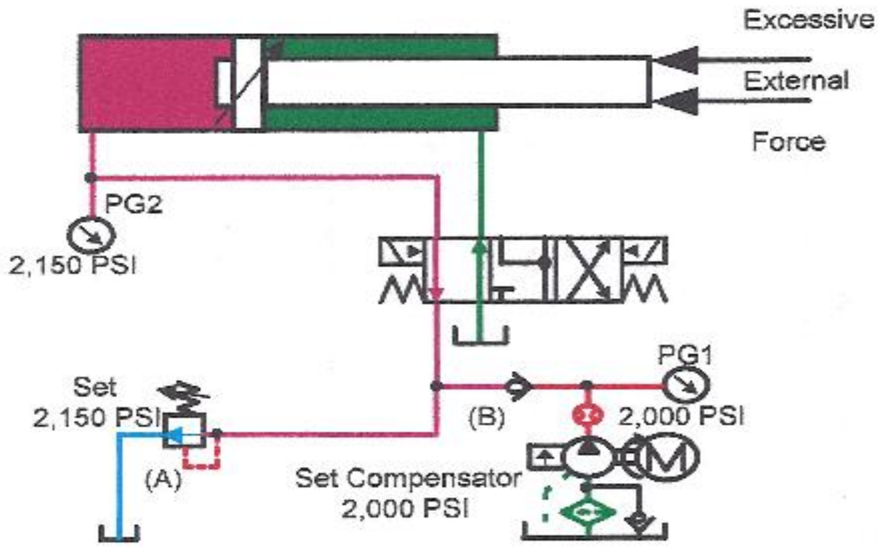


Figure 20-24. Pressure-compensated pump with safety relief valve — at rest with pump running.

Relief valve (A), installed anywhere in the pressure line, protects the system when set 150 to 200 psi higher than the pump compensator. The inlet to a pressure-compensated pump should never see a pressure higher than its compensator setting. Adding check valve (B) at the pump outlet assures that pressure at the pump never goes above the compensator setting. However, relief valve (D) may cause problems, as stated in Chapter 1, page ACC7. Figures 20-25 through 20-27 show an overpressure protection circuit that is less prone to problems.

Figure 20-25 shows internally piloted, externally drained, low-pressure sequence valve (A) teed into the pump outlet. (Valve (A) has a low-pressure spring rate between 50 and 250 psi.) The outlet of sequence valve (A) goes directly to tank. Isolation check valve (B) in the pump outlet line before the sequence valve keeps reverse flow and excess pressure away from the pump. Pilot line (C) from the pump outlet before check valve (B) goes to the external drain port of sequence valve (A).

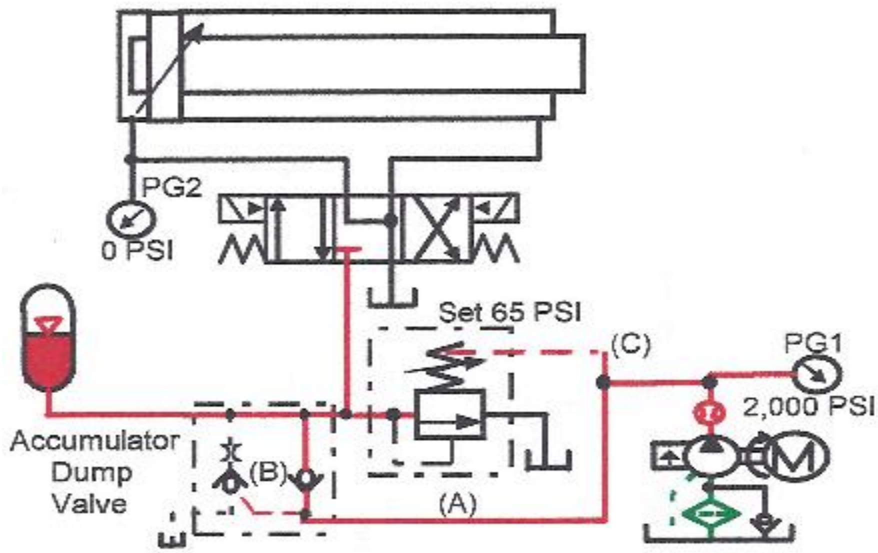


Figure 20-25. Pressure-compensated pump with safety relief valve — cylinder extending and being pushed back.

With the circuit at rest and the pump running, system pressure is the pump compensator setting. Pump pressure at the internal pilot is trying to open sequence valve (A), but at the same time holding it closed through the external drain port. With a sequence-valve spring setting of 65 psi, it will not open to tank until pressure after check valve (B)goes at least 65 psi higher than pump compensator setting, as in Figure 20-26.

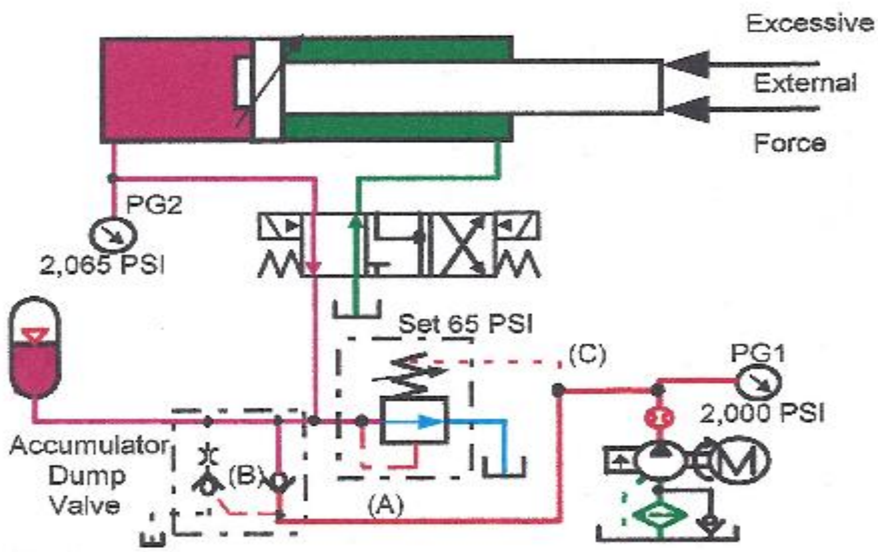


Figure 20-26. Pressure-compensated pump with safety relief valve —

cylinder extending and being pushed back.

The main reason this overpressure relieving circuit is better than a circuit with a standard relief valve is that adjusting the pump compensator not only changes system pressure, but also automatically raises relief pressure. The pump never relieves to tank and the circuit always relieves when pressure in it increases more than the sequence-valve spring setting.

With this overpressure circuit there is no protection from pressure spikes when the pump compensator has to work quickly. Using the accumulator shown in the schematic diagrams protects the pump when it has to compensate rapidly. The accumulator also makes the circuit more responsive at cycle start.

This sequence-valve relief works anywhere in the circuit to protect any line from overpressure. Figure 20-27 shows a sequence valve teed into the cap end line of a cylinder with excess external force. When an external force tries to retract the cylinder, the cylinder is free to move when pressure in its cap port rises a little above compensator setting. Any other time the sequence valve stays closed because its inlet never sees pressure higher than compensator setting.

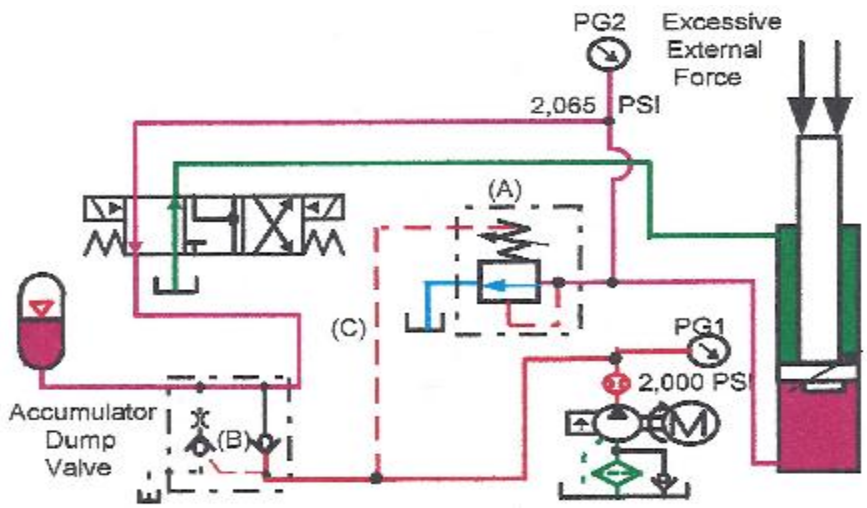
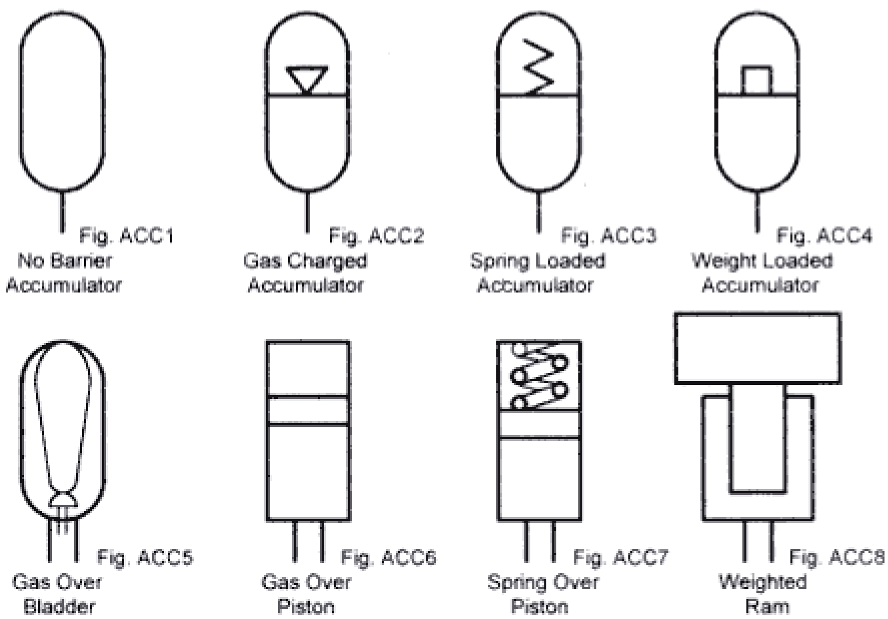


Figure 20-27. Pressure-compensated pump with safety relief valve — cylinder extending and being pushed back.

Hydraulic accumulators make it possible to store useable volumes of non-compressible fluid under pressure. A 5-gal container completely full of oil at 2000 psi will only discharge a few cubic inches of fluid before pressure drops to 0 psi. The same container filled with half oil and half nitrogen gas would discharge over 11/2 gal of fluid before pressure dropped to 1000 psi.

Figures 1-1 through 1-4 *show symbols used for different types of accumulators.* ***Figures 1-5 through 1-8*** *are simplified cutaways showing construction of different types of accumulators.*

Fig-1-1 to 1-8



All accumulators except ***Figure 1-4*** *will have a pressure decrease as fluid discharges. A weight-loaded accumulator maintains pressure until all oil is used.*

When using an accumulator, it is necessary to install a manual or automatic function to de-pressurize all fluid before working on the circuit. Several manufacturers make automatic discharge valves that work well. These automatic discharge valves are explained at the end of this section.

Most hydraulic accumulators are used in one of four applications: 1. Supplement pump flow in circuits with medium to long delays between cycles. 2. Hold pressure in a cylinder while the pump is unloading or stopped. 3. Have a ready supply of pressurized fluid in case of power failure. 4. Reduce shock in high velocity flow lines or at the outlet of pulsating piston pumps.

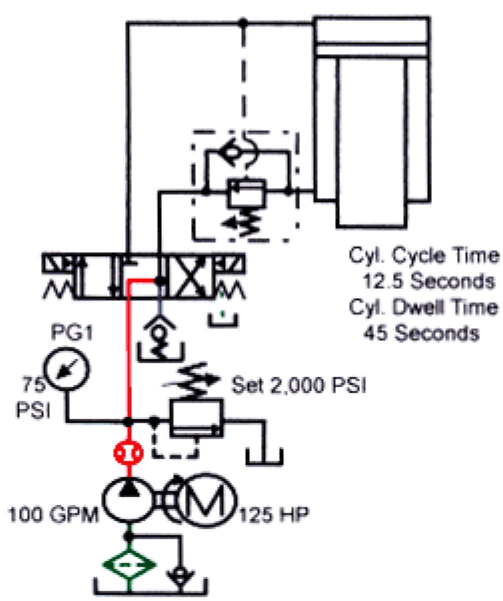
The following circuit images show some circuits using accumulators for the operations mentioned in 1 to 4 above. Other accumulator circuits and information follow.

Using accumulators to supplement pump flow

Some hydraulic circuits require a large volume of oil for a short time; for example to move a large cylinder rapidly to clamp a part. After clamping, the circuit needs little or no additional fluid for period of time while curing takes place. When a circuit has extended dwell time, an accumulator can be used to downsize the pump, motor, tank, and relief valve. The cost of accumulators usually offsets savings on these smaller components, but downsizing saves on operating costs.

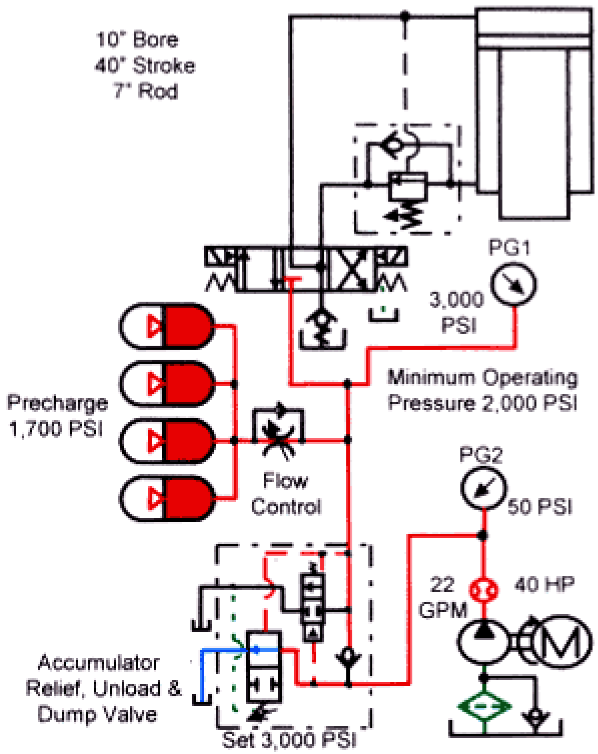
*The conventional pump, directional valve, and cylinder pictured in **Figure 1-9** show horsepower and flow requirements needed for a 12.5 sec cycle time. The advance cycle requires full power, while returning the cylinder needs minimal force. Reduction of the pump and motor size is not possible if the cylinder cycles rapidly. However, if there was a 45 sec wait between cycles, the pump and motor could be almost 70% smaller with an accumulator circuit.*

Fig-1-9



*This reduced flow and horsepower are possible when using accumulators and the circuit shown in **Figure 1-10**. The extra expense of the accumulators offsets the reduced price for the power unit, but operating cost is less for the life of the machine. The directional valve and piping from the accumulators to the cylinder still has to handle the 125 gpm flow.*

Fig-1-10



Using a gas charged accumulator in a pump supplementing circuit will increase maximum system pressure. The extend portion of the cycle needs at least 2000 psi working pressure, which requires filling the accumulators with fluid above 2000 psi so they can discharge oil and not drop below minimum pressure. The maximum system pressure should be as high as can be tolerated. The higher the maximum allowable system pressure, the smaller the accumulators. The drawback of high pressure is that the circuit is at this pressure when the cycle starts. If this higher pressure can cause damage or other problems, it should be lowered to a safe level.

Accumulator circuits normally have flow controls because there is a volume of oil at elevated pressure that can discharge almost instantaneously. Placing a flow control at the accumulator outlet allows free flow from pump to accumulator and adjustable flow to system.

*The circuit in **Figure 1-10** has a minimum pressure of 2000 psi and a maximum pressure of 3000 psi. This pressure is the limit of most hydraulic components. A 22-gpm pump driven by a 40-hp motor now meets the force and cycle time specified. All pump flow continuously goes to the circuit instead of being unloaded most of the time as in conventional circuits.*

As the cylinder cycles, the accumulators supply fluid at a rate set by the flow control. Pump flow adds to accumulator flow to set the required cycle time. Cylinder cycling could be made faster than specified by increasing outlet flow from the accumulator.

*The fixed-volume pump in **Figure 1-10** unloads through a special accumulator relief/unload/dump valve, which sends all pump flow to the accumulators and cylinder until the system reaches set pressure. After reaching set pressure, the valve opens and unloads the pump to tank at approximately 50 psi. The pump will continue to unload until the system pressure drops about 15%. This pressure drop might be from leakage or at the start of a new cycle. Any time pressure drops, the pump will load and stay loaded until pressure tries to go above 3000 psi. With this valve, stored oil in the accumulators automatically discharges to tank when the pump stops, which makes the circuit safe to work on shortly after locking and tagging off the pump.*

Fig-1-11

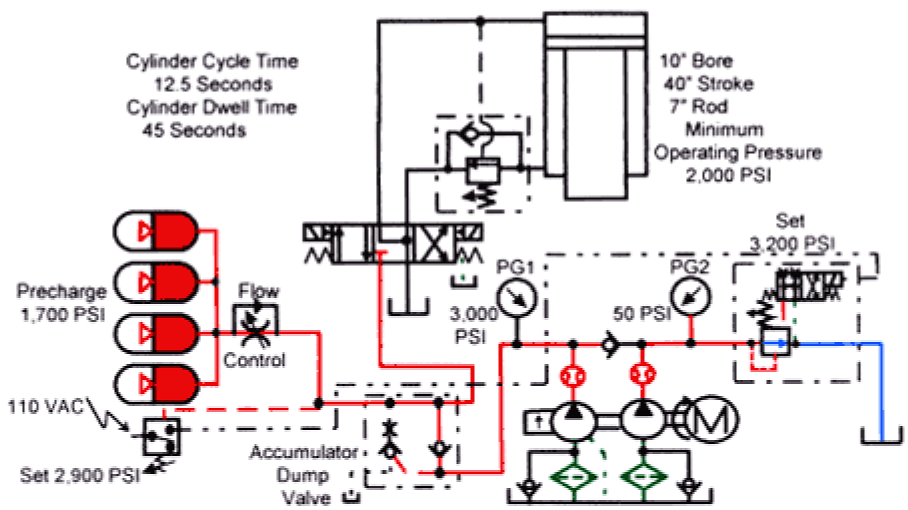


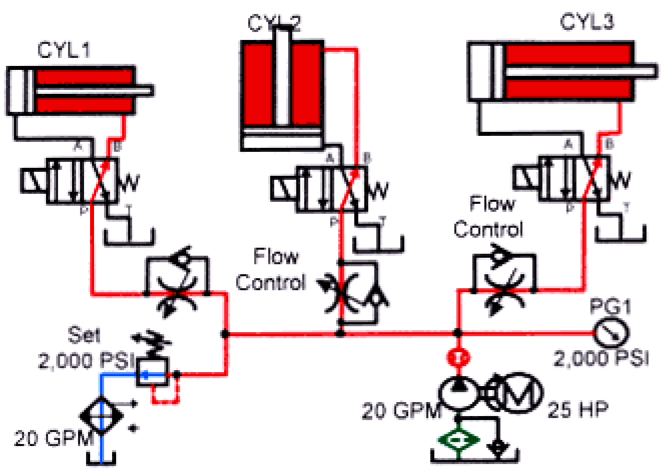
***Figure 1-11** shows a variation of the accumulator circuit in **Figure 1-10**. Here a 1-gpm fixed-volume pump and a 5-gpm pressure-compensated pump supply oil until the accumulators fill. A pressure switch, set at about 2900 psi, unloads the fixed-volume pump through a solenoid-operated relief valve. After the fixed-volume pump unloads, the pressure-compensated pump finishes filling the accumulators and holds maximum pressure without fluctuations and with minimal heating.*

The accumulator dump valve in this circuit will stay closed as long as the pumps are running. When the pumps stop, this valve quickly and automatically discharges the accumulators to tank.

Full-time pressure with fixed-volume pumps

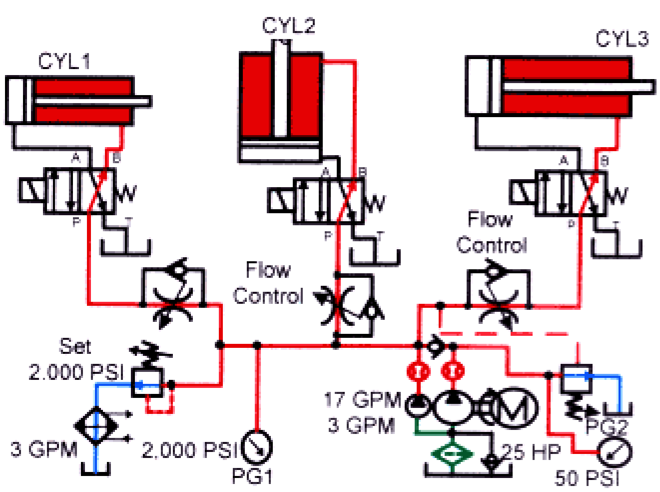
*Some circuits need pressure at all times to hold position or maintain force. The circuit in **Figure 1-12** holds pressure on the cylinders when they stop, but excessive heat generation makes it a poor choice. Flow controls keep pressure in the circuit while a cylinder is moving.*

Fig-1-12



*Some designers use the circuit shown in **Figure 1-13** to simultaneously reduce energy loss and maintain holding pressure. This double-pump circuit provides high flow (to move the cylinders rapidly) and low flow (for pressure holding). While the system is at holding pressure, the high-flow pump goes to tank through an unloading valve. Only the low-flow pump goes across the relief valve. Although energy loss is drastically reduced, it is still excessive.*

Fig-1-13



*The circuits shown in **Figures 1-14** and 1-15 use a small accumulator to hold pressure on the actuators while unloading the pump at minimum pressure. This makes it possible to use a less expensive fixed-volume pump instead of a pressure-compensated pump, with little or no energy loss or heat generation.*

Fig-1-14

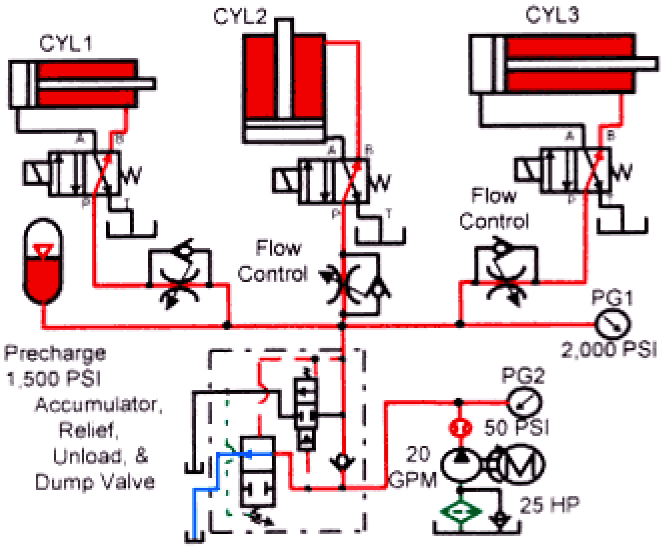
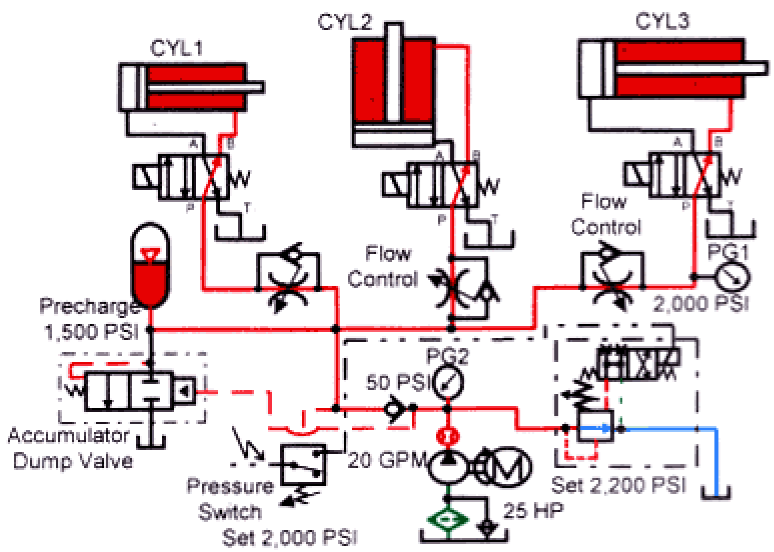


Fig-1-15



*The pump in **Figure 1-14** unloads through an accumulator relief/unload/dump valve. This valve sends all pump flow to the accumulator and cylinders until the system reaches set pressure. After it reaches set pressure, the valve opens and unloads the pump to tank at approximately 50 psi. The pump will continue to unload until the system pressure drops about 15%. This pressure drop* *might be from leakage or it could be at the start of a new cycle. The pump loads again and fills the circuit until pressure tries to go above 2000 psi. While the pump unloads, the accumulator makes up for any leakage so pressure at the cylinders only drops about 15% maximum. The length of time the pump unloads depends on the size of the accumulator and the amount of system leakage. With the accumulator relief/unload/dump valve, stored oil in the accumulator discharges to tank when the pump stops. This makes the circuit safe to work on shortly after locking and tagging out the pump.*

Notice the variation of the above pressure holding circuit in ***Figure 1-15****. Here the pump unloads through a normally open, solenoid-operated relief valve controlled by a pressure switch. The accumulator and actuators fill from the pump until system pressure reaches 2000 psi. At 2000 psi, the pump unloads through a solenoid operated relief valve at approximately 50 psi. The main advantage of the circuit in Figure 1-15 is that pressure drop is adjustable by more or less than the fixed 15% allowed by the unloading valve in* ***Figure 1-14****.*

To have a safe accumulator circuit, it is necessary to have a means to discharge stored energy at shutdown. The circuit in Figure 1-15 uses a high-ratio pilot-to-close check valve. The pilot ratio is about 200:1, which means 25 psi in the pilot line can hold as much as 5000 psi in the circuit. Most unloading circuits have at least 25 psi while unloaded, so this valve works well. When the pump shuts off, pressure drops to zero, the pilot-to-close check valve opens, and stored energy dumps to tank.

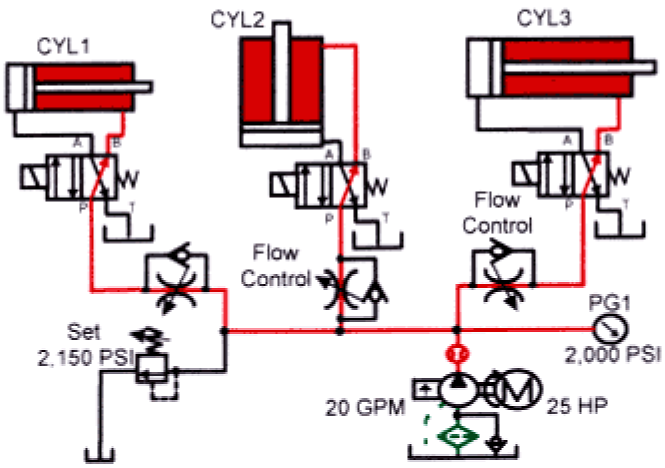
Another way to automatically discharge the accumulator at shutdown is with a normally open, solenoid-operated, 2-way directional valve. This directional valve connects to the accumulator pressure line and on to tank. Starting the pump motor also energizes the solenoid on the normally open 2-way valve, causing it to close. As long as the pump runs, this valve blocks the flow path to tank. When the pump stops, the solenoid is deenergized, and the valve shifts to port stored energy to tank.

CAUTION! ALWAYS CHECK AN ACCUMULATOR CIRCUIT FOR PRESSURE BEFORE WORKING ON IT. NEVER ASSUME THE AUTOMATIC UNLOADING SYSTEM WORKED!

Accumulators used for fast response and over-pressure control of pressure-compensated pumps

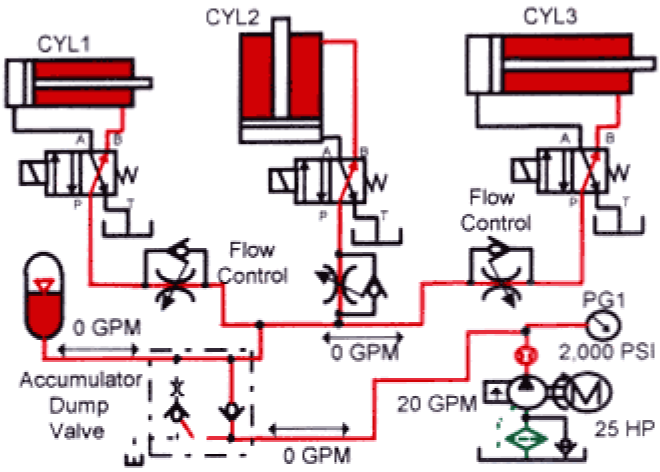
*Because most pressure-compensated pump circuits have closed-center or two-position directional valves (such as the one shown in **Figure 1-16**), they stay at full-pressure, no-flow until a valve shifts. After any directional valve shifts to start an actuator's movement, pressure in the circuit starts to drop. When the pump sees a pressure drop, its internal mechanism starts shifting as fast as possible to start fluid flowing. Pump shifting times vary, but no matter how fast they shift, the actuator's initial response will be slowed down.*

Fig-1-16



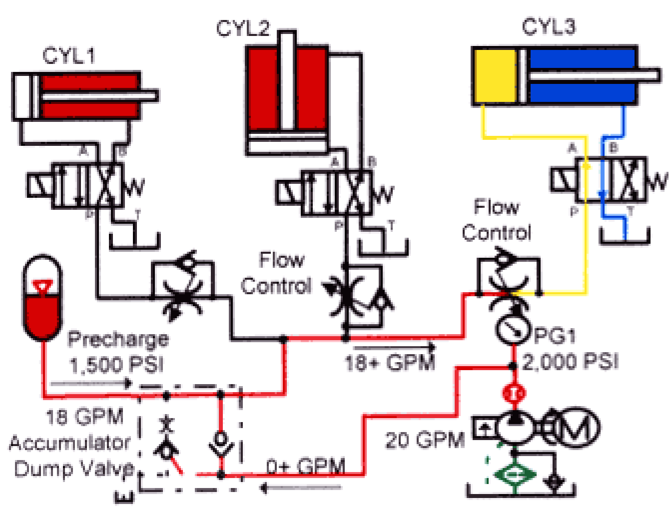
*With an accumulator installed, as shown in **Figure 1-17**, the pump is still at no-flow when the circuit is at rest. However, there is a ready supply of oil at pressure available. As a cylinder starts to cycle, as seen in **Figure 1-18**, fluid flows directly to the actuator from the accumulator and pressure starts to drop. This pressure drop causes the pump to go on stroke, but now pressure drop is minimal. The cylinder takes off quickly and smoothly, and the pump has time to respond to the flow need.*

Fig-1-17



On the other end of the cycle, if the pump is at full flow and all valves center or all the actuators hit the end of stroke, the flow requirement suddenly drops to zero. The pressure-compensated pump is still flowing at the maximum rate and pressure starts to climb. The pump will continue at full flow until pressure reaches 80-98% of the compensator setting. There has been zero flow needed for some time, but the pump does not know this until pressure is near maximum. When pressure reaches compensator setting, the pump starts to shift to no flow. All pump flow during shifting time has no place to go, so this excess flow generates a pressure spike of five to ten times the compensator setting. This pressure spike can cause premature failure of the pump, plumbing, and actuators.

Fig-1-18

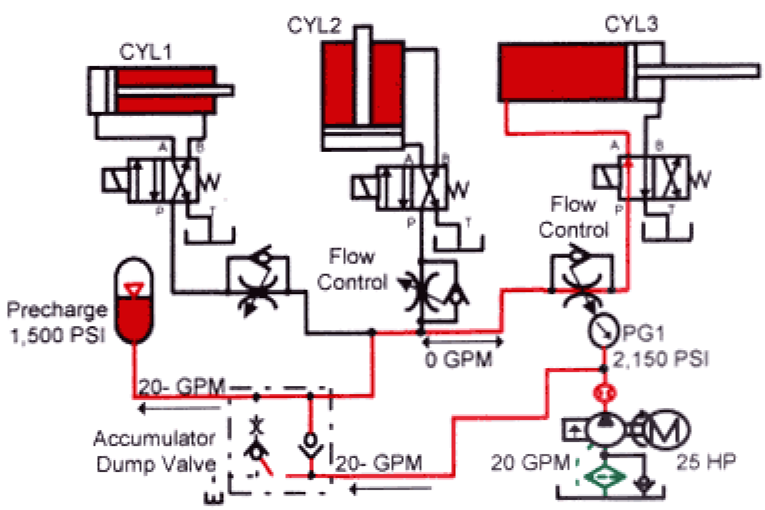


*A common fix for this pressure spike is to add a relief valve near the pump outlet, set 150 to 200 psi higher than the pump compensator (as shown in **Figure 1-16**). This relief valve should reduce the pressure spike, but it does not lower it as much as it appears. A relief valve remains closed until pressure reaches 90 to 98% of its setting. Once the relief reaches maximum pressure, it starts to open, but by the time it actually relieves, the pressure may be 11/2 to 3 times its set pressure. This reduced spike is better, but still is not as good as what an accumulator could provide.*

Other problems can occur with relief valves. For example, if the relief valve setting is at or near the pump-compensator setting, the pump can start oscillating on-off flow. As the pump nears its pressure-compensator setting and starts to compensate, the relief valve starts to relieve. A flow path is created when the relief valve begins to open, so downstream pressure drops, causing the pump to go back on stroke. The drop in pressure allows the relief valve to close, so downstream pressure builds up again. This oscillation cycle repeats rapidly, causing damage to the pump and possible line failure due to shock. In another example, if the relief valve setting is lower than the pump compensator, all pump flow goes to tank at relief pressure, generating excess heat. To avoid these problems, use the correct procedure when setting pressures on a relief valve used to reduce pressure spikes.

*An accumulator absorbs excess pump flow with minimal pressure override or shock. While fluid from the pump compensates from full flow to no flow, as seen in **Figure 1-19**, it has a direct path to the accumulator. Because the accumulator has a compressible gas in it, it takes in the small amount of excess flow produced while the compensator is reacting. Pressure increase from this additional fluid is imperceptible.*

Fig-1-19



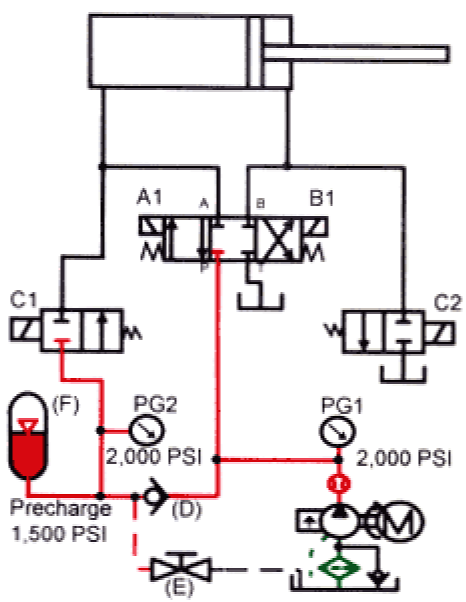
To size an accumulator for fast response of the circuit, plan to have somewhere between 1 and 5 sec of actuator flow before pressure drops below the minimum it takes to move it. A rule of thumb is to have 1 gal of accumulator for every 10 gpm of pump flow.

Using an accumulator as an emergency power supply

A conventional hydraulic system will not operate unless the pump is running. Some machines must be able to cycle to a safe condition after a power or pump failure. Use an accumulator to store enough energy to move the actuators to a safe condition after the pump quits. The operator or setup person can manually cycle the machine into a safe condition by using the stored energy.

The hopper gate cylinder shown in Figure 1-20 must close in case of a power failure. If the gate stays open, the entire hopper could overflow the truck under it, then dump on the ground. This circuit uses a pressure-compensated pump that maintains pressure with minimal heating during normal operation. An accumulator F stores the first pump flow, check valve D stops accumulator back flow, and normally open directional valves C isolate the accumulator from the cylinder and tank during normal operation.

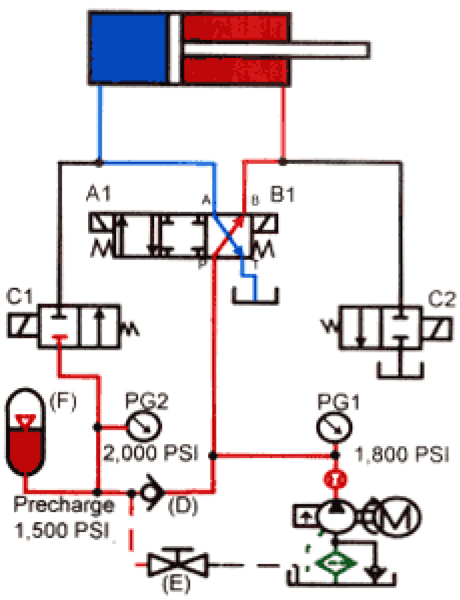
Fig-1-20



The gate cylinder needs at least 1500 psi, so the pump compensator is set for 2000 psi. This ensures that the accumulator has enough fluid to extend the cylinder when necessary. Because the solenoids on valves C are energized by the pump start command, the accumulator is completely isolated from the cylinder and tank as long as the pump runs. When solenoid B of the

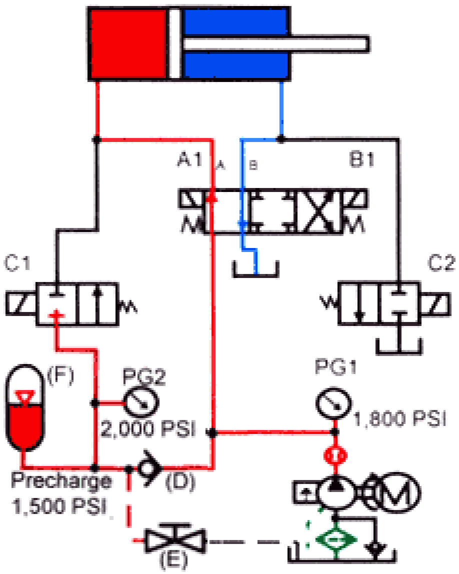
*4-way directional valve shifts (as seen in **Figure 1-21**), the gate opens as fast as the pump moves it.*

Fig-1-21



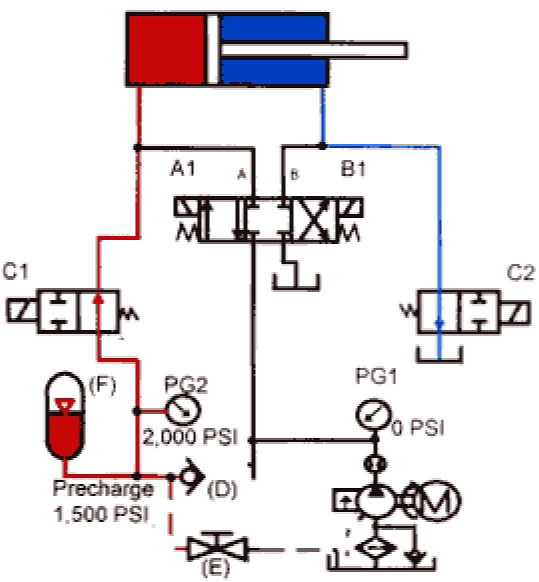
*When solenoid A shifts the 4-way directional valve, as seen in **Figure 1-22**, the gate closes as fast as the pump moves it. When the power is on, the cylinder extends or retracts partially or all the way at the operator's command.*

Fig-1-22



*If the gate cylinder is partially or completely open and power fails, the circuit automatically goes to the condition shown in **Figure 1-23**. In this condition the pump stops, the 4-way directional valve centers, and the normally open 2-way shutoff valves C open.*

Fig-1-23



When power fails, the accumulator has a direct path to the cap end of the cylinder while rod-end oil flows to tank. The cylinder will extend and close the gate using the stored energy in the *accumulator.* ***Place warning signs at the gate indicating this equipment can operate at any time without operator intervention.***

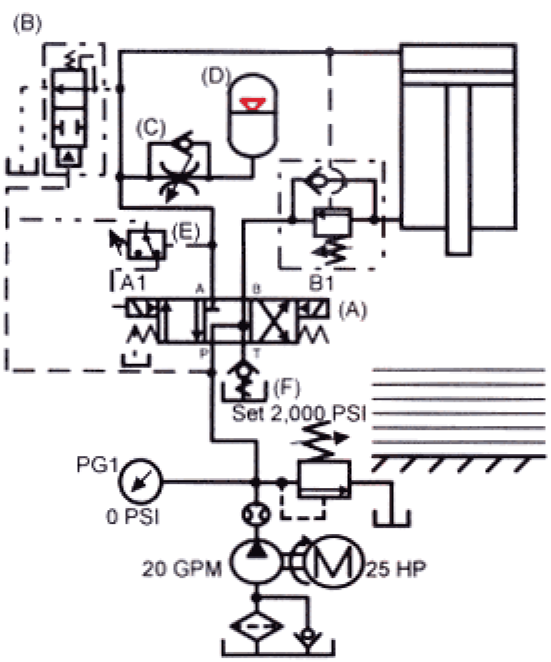
When using an accumulator for emergency power supply it is difficult to automatically drain it during normal operation. Automatically draining the accumulator would defeat its purpose as an emergency power supply. Add a manual drain valve E, with ***warning signs to tell maintenance persons to manually drain the accumulator before working on the gate circuit.***

Size emergency-power accumulators to hold enough oil to move all actuators to the home position before pressure drops to dangerous levels. Most manufacturers provide formulas in their catalogs and offer several offer excellent computer programs to size accumulators for emergency-power supplies.

Using accumulators for leakage makeup

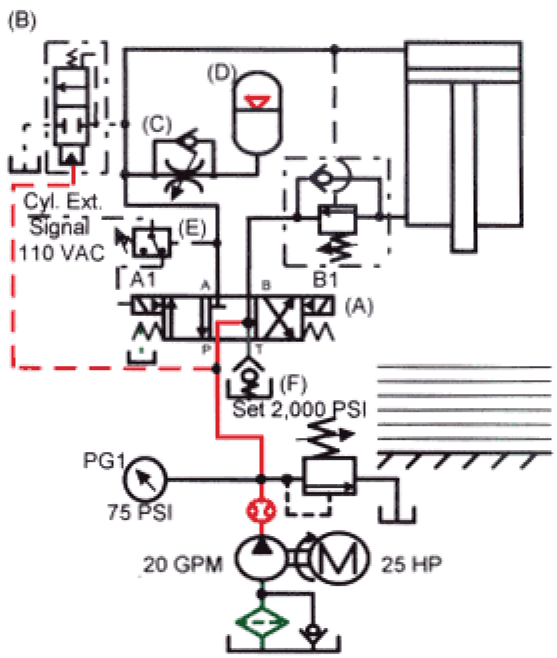
Some hydraulic circuits, such as in laminating presses, need to hold at pressure for long periods. A pressure-compensated pump could maintain pressure, but energy loss from pump leakage generates heat. Another way to hold pressure for long periods is with a fixed-volume pump and an accumulator. ***Figure 1-24*** *shows a press cylinder that must stay extended under pressure for several minutes.*

Fig-1-24



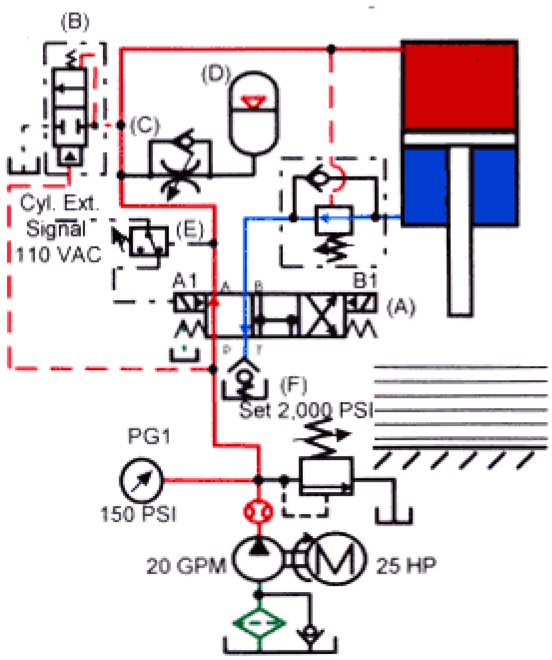
Tee small accumulator D into the cylinder cap-end line through flow control C. Flow control C allows the accumulator to fill quickly but discharge slowly when directional valve A centers or shifts to retract the cylinder. Flow control C should pass enough flow to let the accumulator discharge quickly without system shock when directional valve A shifts to retract the cylinder. Any oil left in the accumulator when the directional valve centers will make the cylinder extend a small amount. Tee dump valve B into the cylinder cap-end line to automatically discharge the accumulator when the pump stops. Tee pressure switch E into the cap-end cylinder line to set pump load and unload pressure. Pressure switch E sets high and low pressures to control maximum and minimum tonnage.

Fig-1-25



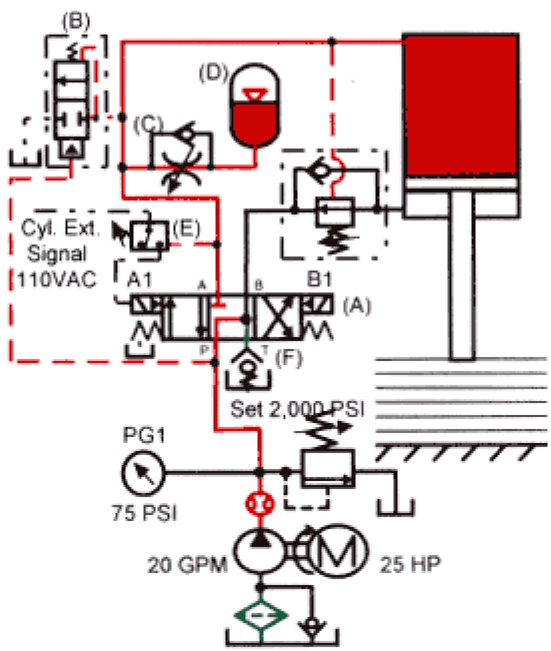
When the pump starts, ***Figure 1-25****, backpressure check valve F gives 75 psi pressure, closing accumulator dump valve B and supplying pilot oil for solenoid pilot-operated directional valve A. When directional valve A shifts, the cylinder starts to extend,* ***Figure 1-26****, at whatever pressure it takes to overcome the counterbalance valve. The signal to the extend coil of directional valveA goes through the normally closed contacts on pressure switch E. Because gas pre-charge pressure in the accumulator is approximately 85% of working pressure, no fluid will enter it yet.*

Fig-1-26



When the cylinder contacts the work, ***Figure 1-27****, pressure increases and oil fills the accumulator. Upon reaching the maximum working pressure set by pressure switch E, the normally closed contacts open, de-energizing the solenoid on directional valve A. Directional valve A spring centers, the pump unloads, and oil stored in the accumulator maintains pressure while making up for cylinder and valve leakage.*

Fig-1-27



Bypass at the cylinder seals and/or valve causes pressure to drop slowly to the low-pressure setting of pressure switchE. This low-pressure setting is normally adjustable but must be high enough to keep the parts firmly together. Upon reaching the low-pressure setting, pressure switch E shifts, allowing the normally closed contacts to shift directional valve A to refill the accumulator. Upon reaching maximum working pressure, directional valve A again spring centers to unload the pump, while the accumulator holds its pressing force and makes up for leaks.

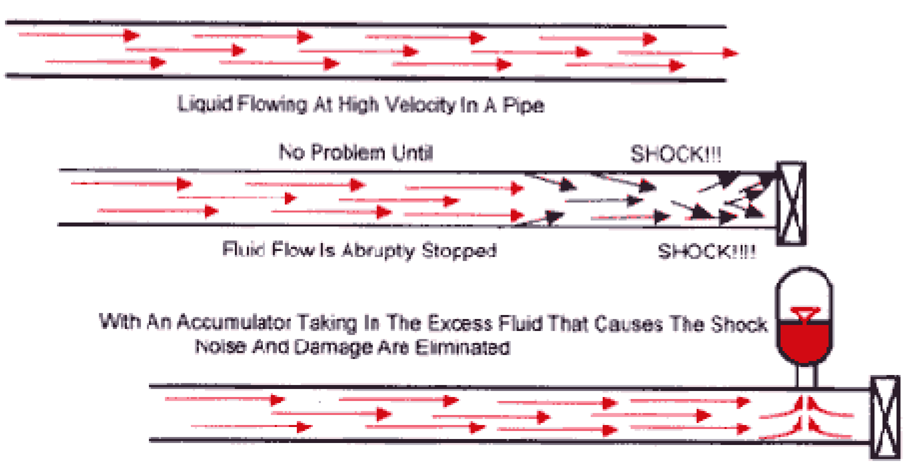
A pilot-operated check valve in the cap-end cylinder line between the directional valve and the pressure switch would have less leakage than the blocked port of the spool valve. With a pilot-operated check valve and resilient seals in the cylinder, it is possible to maintain pressure for 2 to 5 min or more. Use an all-ports-open directional valve with the pilot-operated check valve. This accumulator circuit maintains pressure in the cylinder while unloading the pump. It also conserves energy while using an inexpensive fixed-volume pump.

Using accumulators as shock absorbers

Accumulators can reduce damage from shock in some circuits if correctly applied. In other applications, an accumulator may add shock by releasing stored energy too quickly.

The top half of ***Figure 1-28*** *illustrates one way shock is produced. Flow velocity in a hydraulic circuit may be 25 to 30 fps and not cause any problems. However, if oil flow stops abruptly, as seen in* ***Figure 1-28's*** *middle example, damaging shock can rip out tubing, blow seals, and split pump housings with ease. A column of moving fluid has a lot of energy that can get out of control.*

Fig-1-28

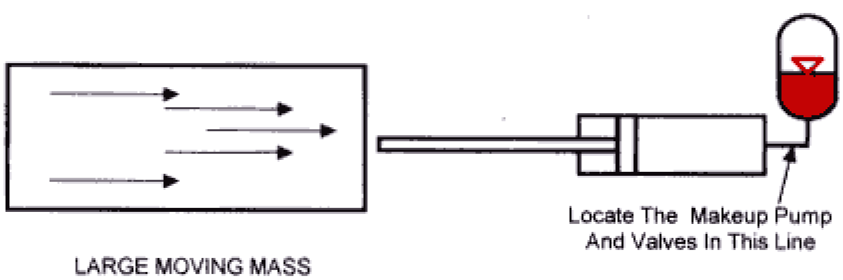


*The third example in **Figure 1-28** shows a small accumulator teed into the line at the shut off that stops flow suddenly. An accumulator spreads the shock energy over a short period of time and eliminates the potential for damage.*

To absorb flow shock, the accumulator is usually pre-charged at about 70 to 80% of system pressure. At this pre-charge pressure, only a small amount of fluid enters the accumulator subsequent to a shock situation. There is also little fluid transfer to take away from or add to the normal pump flow.

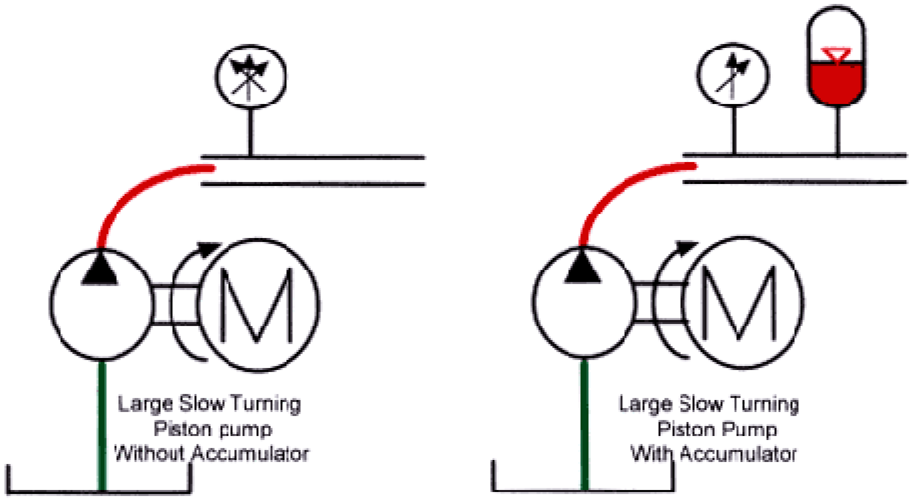
*When it is necessary to stop a heavy load, such as shown in **Figure 1-29**, try using an accumulator and a hydraulic cylinder. The accumulator's pre-charge pressure holds the cylinder extended, thus making it ready for the advancing mass. When the load contacts the cylinder, it mechanically forces it to retract. As the cylinder retracts, fluid flows into the accumulator and gas pressure increases. As pressure increases, the higher resistance slows the mass more. After the load decelerates, the cylinder might try to push the part away. Add valves between the accumulator and the cylinder to control the shock absorber after it finishes decelerating the load.*

Fig-1-29



*Some large, slow-turning piston pumps send a shock wave into the circuit every time a piston discharges oil. On the left of **Figure 1-30**, the piston pump does not have an accumulator at the discharge port. Pressure at the gauge will fluctuate from less than system pressure to well above it without an accumulator.*

Fig-1-30



*On the right side of **Figure 1-30**, adding a small accumulator reduces discharge flow and shock damage. A portion of the sudden discharge flow from an advancing piston goes into the accumulator and discharges smoothly while waiting for the next stroke. The pre-charge pressure for this type of accumulator circuit is about 60 to 75% of maximum system pressure.*

Accumulator manufacturers have formulas in their brochures to calculate any situation mentioned here. Some suppliers have computer programs that do all the math after asking for circuit parameters

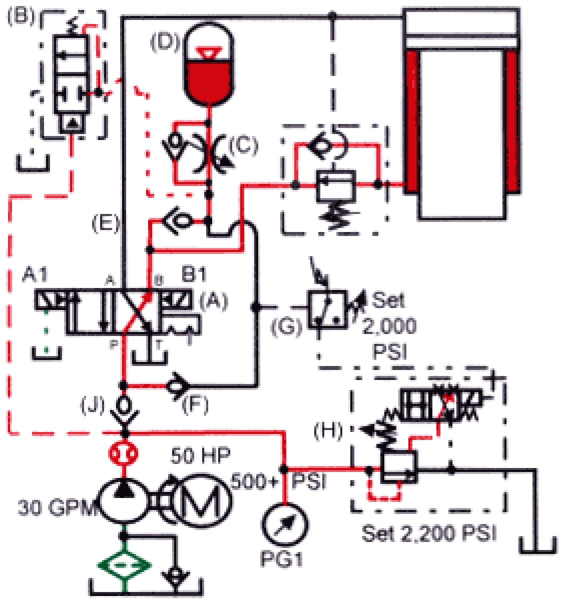
Pump supplementing circuit with full pressure when work is contacted

In some cases, a pump-supplementing accumulator circuit can speed up cylinder extension and/or retraction without having to go above working pressure. Normally in a pump-supplementing circuit, the relief valve is set as high as possible above the working pressure to store ample fluid. As the cycle progresses, oil from the accumulator and pump move the actuator quickly, but circuit pressure drops steadily. If pressure drops below the actuator's need, the pump must refill the accumulator before the cycle finishes. To overcome this problem, a larger pump and/or more accumulators are necessary.

The next circuit shows an accumulator arrangement that provides high volume to move the cylinder rapidly with the relief valve set at working pressure. The accumulator and pump supply volume to fill the large bore cylinder as it extends. The cylinder then moves to working pressure while a check valve isolates the accumulator.

Like all accumulator circuits, there must be time for refilling between cycles, as shown in ***Figure 1-31****. Pre-charge the accumulator to a pressure slightly higher than it takes to retract the cylinder. The cylinder will then retract when directional valve A and normally open, solenoid-operated relief valve H shift. (Also see* ***Figure 1-34****.) The large piston rod reduces the return volume, although retract pressure will be higher. When the cylinder fully retracts, pressure climbs and the accumulator starts to fill through check valve E and the bypass check valve around flow control C. Piston-type accumulators are best for this circuit because they can have a low pre-charge pressure and a high final pressure without internal damage. The accumulator can discharge a large volume of oil because the pressure in it is not important when the cylinder needs full tonnage.*

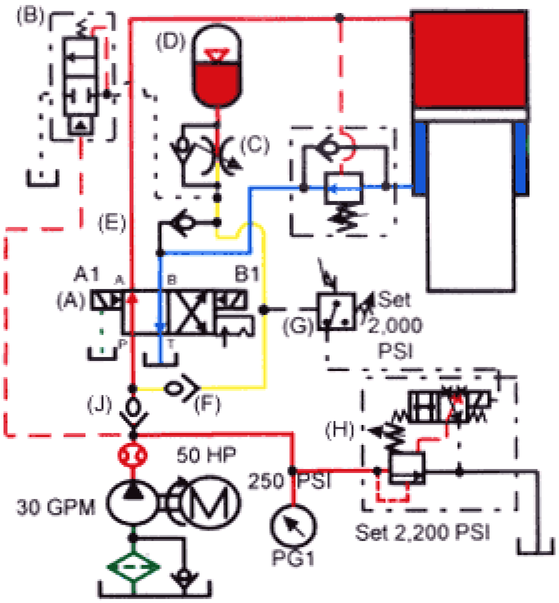
Fig-1-31



When pressure in the circuit reaches 2000 psi, pressure switch G de-energizes the solenoid on normally open, solenoid-operated relief valve H, unloading the pump to tank.

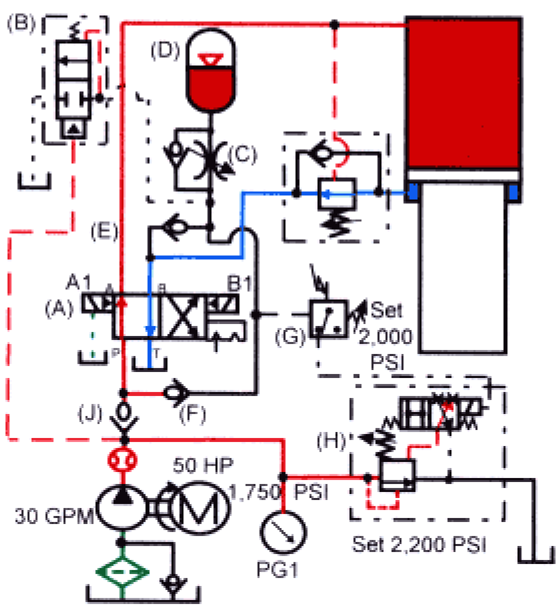
When directional valve A and normally open, solenoid-operated relief valve H shift, ***Figure 1-32****, pump flow and accumulator flow provide a large volume of oil to quickly stroke the cylinder to the work. Because accumulators can discharge at a very high rate, use flow control C to set the desired advance speed. Pressure in the circuit will fall as the cylinder extends and will be well below working pressure before the cylinder meets the work.*

Fig-1-32



*When the cylinder contacts the work, **Figure 1-33**, check valve F keeps pump flow from going to the accumulator. The pump will continue filling the cylinder and pressure will build to whatever it takes to do the work. Check valve F blocks flow to the accumulator to isolate it during the high-pressure work stroke.*

Fig-1-33



When directional valve A shifts to the retract position, Figure 1-34, pump flow goes to the cylinder rod end. The accumulator pre-charge is high enough to force all pump flow to the cylinder, causing it to quickly retract.

Fig-1-34

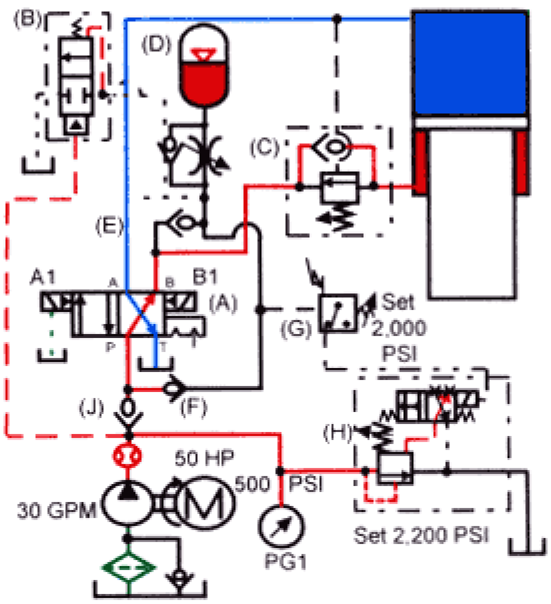
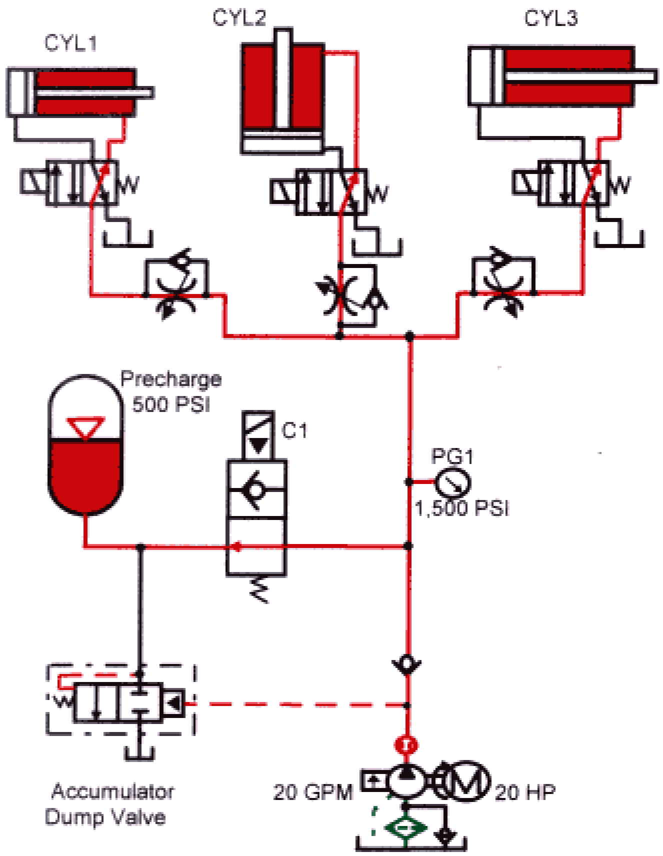


Figure 1-31 *shows the cylinder reaching the top of the stroke. The accumulator now accepts all pump flow through check valve E until pressure switch G unloads the pump.*

Two other pump-supplementing circuits with full pressure when work is contacted

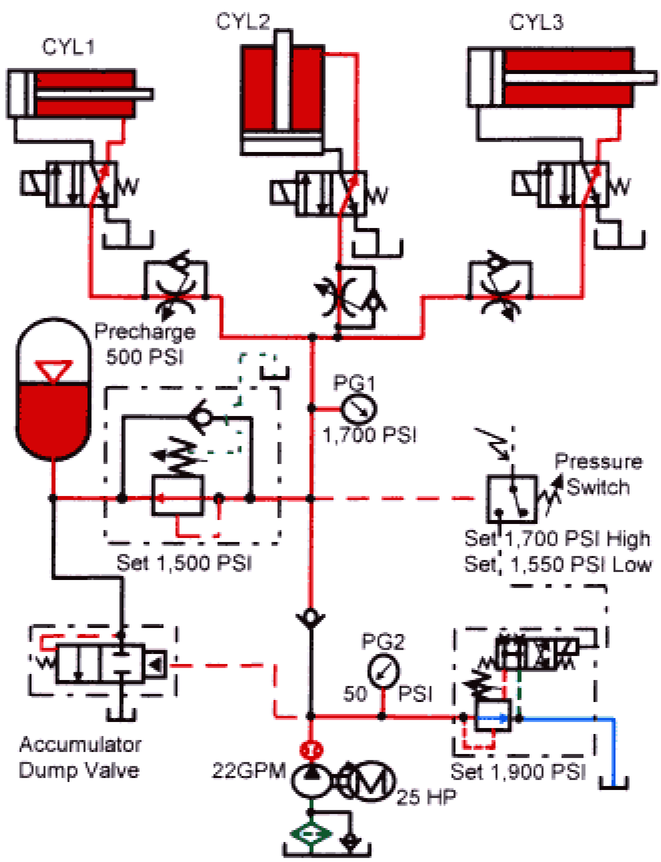
Figures 1-35 and 1-36 *depict two more ways to use an accumulator for volume and still have immediate high pressure for doing work. Either circuit works equally well with the two pump types shown.*

Fig-1-35



These circuits would normally require a piston-type accumulator. Notice the pre-charge is less than one-third of maximum pressure. The large pressure difference would squeeze the bladder in a bladder-type accumulator so much that holes caused by chafing would allow the nitrogen gas to leak. The minimum pressure in the circuit could be even lower than shown here. If the actuators can move at 300 psi, then use 150 to 200 psi pre-charge.

Fig-1-36



The circuit in Figure 1-35 uses a pressure-compensated pump and a normally open, poppet-type, 2-way directional valve. All flow goes directly to the accumulator, filling it to maximum pressure with the pump operating. When the cylinders start to cycle, flow from the pump and accumulator move them rapidly. When the cylinders contact the work, pressure is well below the required amount. To get full force, energize solenoid C1. This stops pump flow to the accumulator and raises the cylinders to full pressure. De-energize solenoid C1 when the cylinders finish their work to allow the accumulator to refill.

Energizing solenoid C1 when the actuators are moving is possible with a correctly designed poppet valve. Notice the blocked position of the valve has a check valve symbol, meaning it only stops flow to the accumulator. This type of poppet valve provides accumulator volume to the actuators when pressure is low. However, maximum pressure is immediately available when the cylinders meet resistance. De-energize solenoid C1 at the end of the cycle to refill the accumulator. Some poppet-type directional valves have a very high pressure drop when flowing through the closed check valve. Use a brand designed for low pressure drop in this circuit.

*The circuit in **Figure 1-36** has a fixed-volume pump with a normally open,, solenoid-operated relief valve and pressure switch to unload the pump at maximum pressure. Minimum system* *pressure for this circuit is 1500 psi. Therefore, it is important to set the sequence valve in front of the accumulator to this pressure. Set the pressure switch to unload the pump at 1700 psi. Then set the normally open, solenoid-operated relief valve at approximately 1900 psi. Because no oil can go to the accumulator if the system pressure is below 1500 psi, the actuators will always have maximum force anytime they meet resistance. When the cylinders are moving to and from the work, pump and accumulator flow can combine to give rapid movement at reduced pressure. Flow from the accumulator can always go to the cylinders through the bypass check valve. Fluid only goes to the accumulator when pump flow is greater than the system requires. This circuit fills the accumulator anytime the cylinders stop or anytime required volume is less than pump output.*

There will be some heating of the oil while the accumulator is filling until system pressure reaches 1500 psi or above. One advantage is that no control circuitry is necessary, even while the accumulator fills anytime actuator volume is less than pump flow.

Non-invasive way to check accumulator pre-charge

It is important to check accumulator pre-charge pressure at regular intervals. Check a new installation each shift for a few days to see if there is a gas-pressure loss. It the gas charge is holding, check pre-charge pressure weekly for the next month. If all is well at the end of a month, then monthly checks should be more than satisfactory.

The normal way to check pre-charge pressure is: (1). Shut down the system. (2). Attach a gauge and charging kit to the accumulator. (3). Open the gas valve and check the pressure reading.

However, this procedure is time consuming, allows some gas to discharge, and may damage the charging valve, which can result in a continuous leak. Outlined below is a simple, non-invasive way to check accumulator pre-charge pressure to see if gas is leaking.

Fig-1-37

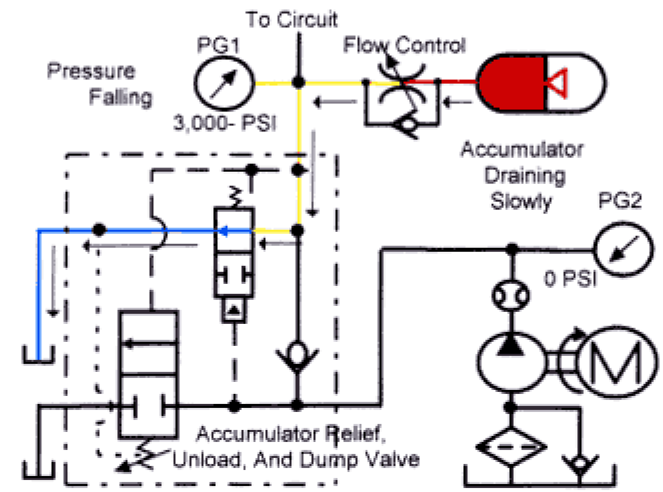
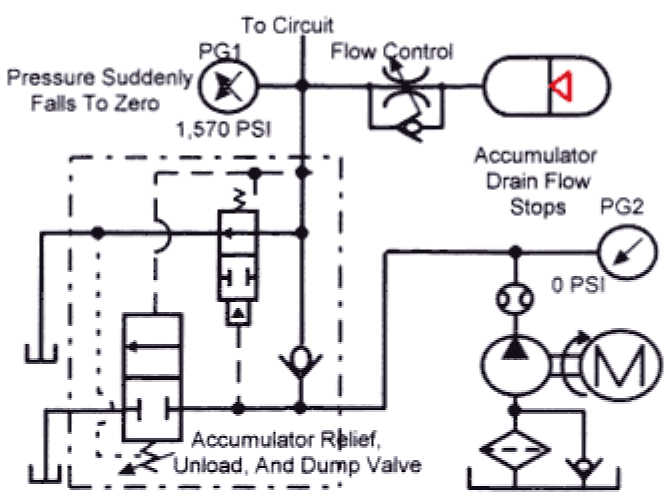


Figure 1-37 *shows a partial accumulator circuit. This figure shows an operating hydraulic system, just as the pump stops. At this point, the accumulator relief/unload/dump valve is open, draining pressurized oil stored in the accumulator. As fluid in the accumulator discharges, pressure at gauge PG1 starts dropping. By controlling the flow with a fixed orifice or a flow control, pressure deteriorates slowly when there is oil in the accumulator.*

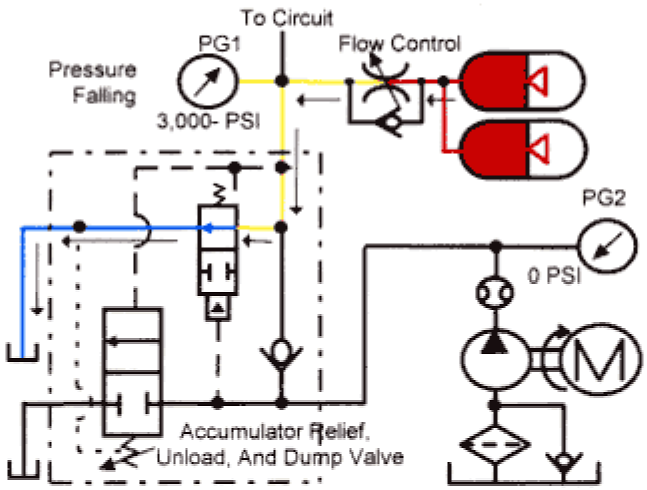
Fig-1-38



:

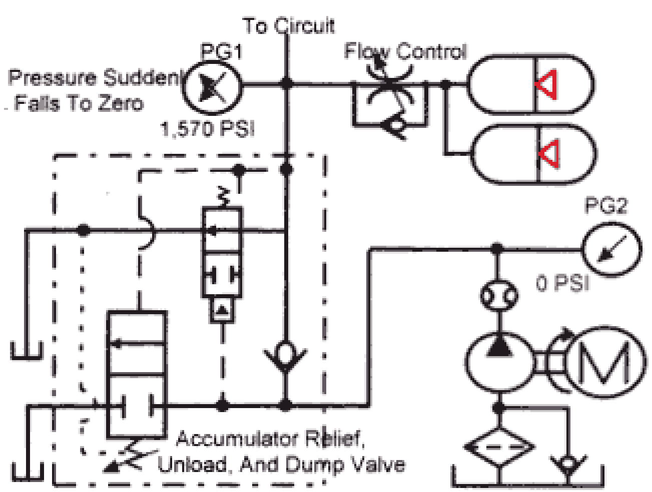
When all fluid is out of the accumulator, ***Figure 1-38****, pressure at gauge PG1 will suddenly drop to zero. Carefully note gauge pressure when it suddenly drops. The pressure seen at the sudden drop is the present pre-charge pressure of the accumulator. This reading is only as accurate as the gauge and the person reading it. It is not a perfect reading, but will be close enough to see if a full-fledged check is needed.*

Fig-1-39



*If there is more than one accumulator on the machine, as in **Figures 1-39 and 1-40**, this test will show the lowest pre-charge pressure. When a low pre-charge pressure shows up, check each accumulator individually until finding those at a lower pressure than required.*

Fig-1-40



Another way to check pre-charge pressure is to note the gauge's pressure reading when turning on the pump. With an accumulator in the circuit, the first pressure reading should be pre-charge pressure. It is difficult to obtain an accurate reading this way with glycerin-filled or orifice-dampened gauges in the circuit. The gauge should also be at or close to the accumulator to keep line losses from adding to the reading.

Hydraulic type accumulator dump valves

When using an accumulator, there must be a way to discharge stored oil before safely working on the circuit. Even when using the accumulator for emergency power supply, install a manual drain valve for safe operation.

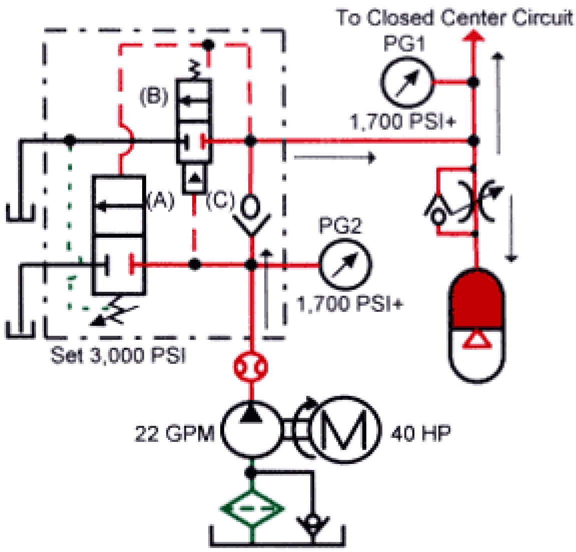
A manual drain valve with a gauge near it is the best way to ensure a safe operation. Mark the manual drain valve and place warning signs at all hydraulic component locations indicating there is an accumulator in the circuit and to open the manual drain before performing maintenance.

A common way to discharge stored energy is to use a normally open, solenoid-operated, 2-way directional valve teed into the pressure line with its outlet hooked to tank. Wire the solenoid on the 2-way valve to close when the pump is running. Any time the pump stops, the 2-way solenoid valve de-energizes and discharges stored oil to tank.

A solenoid-operated valve works well in most cases but can cause problems. First, if the valve fails to close or only partially closes, oil dumps across it, generating heat and making it operate sluggishly or not at all. Second, if the valve fails to open when the pump stops, the circuit is unsafe. This is a safety hazard for an inexperienced person who might not detect the problem. Third, additional wiring creates additional costs.

If the circuit uses a fixed-volume pump as shown in ***Figures 1-41 through 1-44****, use an accumulator relief/unload/dump valve for most applications. This valve has an integral adjustable 2-way unloading valve A to unload the pump when reaching set pressure. Also, there is a pilot valve to close shut-off B that stays closed while the pump is running and opens any time the pump stops. Isolation check valve (C) keeps accumulator oil from back flowing to the pump when it stops.*

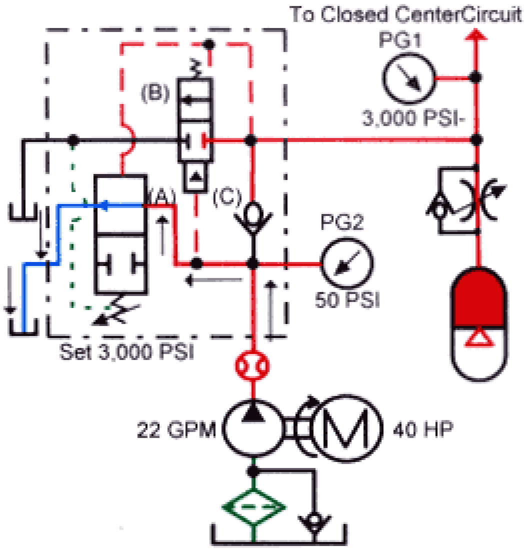
Fig-1-41



*In **Figure 1-41** the pump has just started, so pressure jumps to accumulator pre-charge pressure and all flow goes to the accumulator through check valve C. Pilot-operated 2-way shut-off B pilots closed when the pump is running. The pilot-operated, adjustable-spring shut-off Astays closed until set pressure is reached.*

*Pressure continues to climb until the accumulator is full, as seen in **Figure 1-42**. When pressure reaches that set on 2-way adjustable-spring valve A, it opens, unloading the pump to tank at low pressure. Even while unloading there is enough pressure to keep pilot-operated 2-way shut-off B closed.*

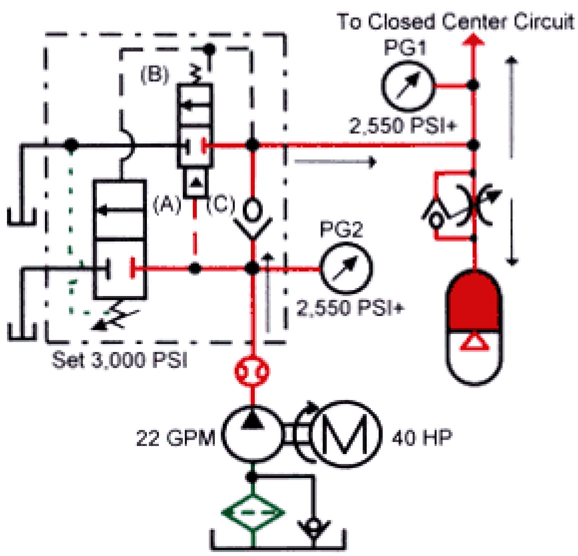
Fig-1-42



*When pressure in the circuit drops approximately 15%, **Figure 1-43**, unload-valve A closes, again forcing oil to the circuit and accumulator. The pump will load and fill the system any time pressure drops about 15%. This pump load pressure is non-adjustable so it will not work for all circuits.*

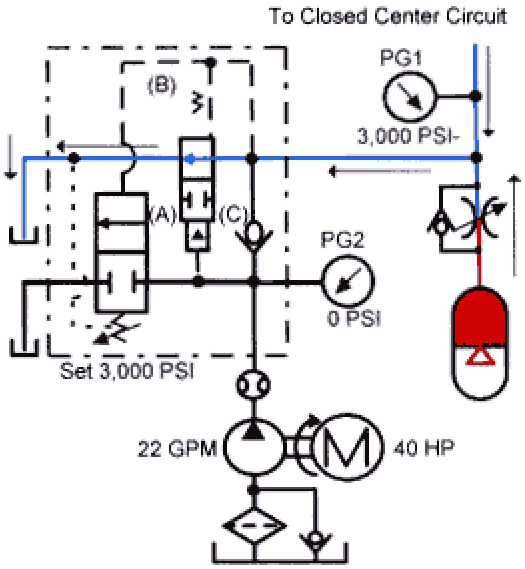
Some manufacturers offer an accumulator relief/unload/dump valve with an adjustable differential setting. Setting these valves' load-unload pressure by more or less than the 15% differential is possible.

Fig-1-43



*When the pump shuts off, as in **Figure 1-44**, pilot pressure to 2-way valve B drops, allowing it to open. Now all stored fluid from the accumulator has a path directly to tank. The accumulator will quickly discharge, making it safe to work on the circuit.*

Fig-1-44



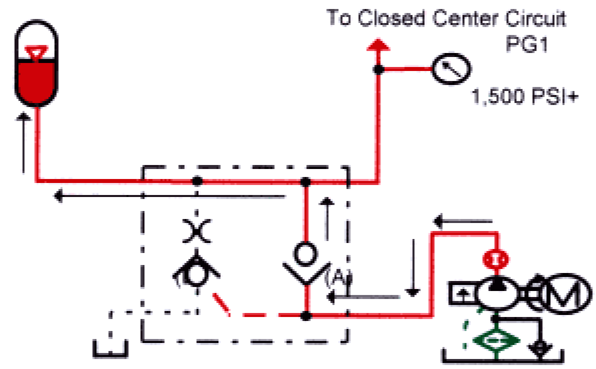
CAUTION! ALWAYS CHECK AN ACCUMULATOR CIRCUIT FOR PRESSURE BEFORE WORKING ON IT. NEVER ASSUME THE AUTOMATIC UNLOADING SYSTEM WORKED!

Hydraulic-type accumulator dump valves (continued)

*When using an accumulator with a pressure compensated pump, the packaged dump valve shown works well. (See **Figures 1-45 through 1-48**.)*

*A pressure-compensated pump maintains pressure while flow changes to meet the needs of the circuit. When the first actuator in the system starts to move, there is no flow for it until pressure drops. As pressure drops, a pressure-compensated pump will go on stroke quickly but there will be a slight pause before flow actually starts. The addition of the small accumulator shown in **Figure 1-45** nearly eliminates the startup pause. This enhances system response while reducing cycle time and pressure fluctuations.*

Fig-1-45



On the other end of the cycle, if the pump is at full flow and all the valves center or all the actuators hit end-of-stroke, flow requirement suddenly goes to zero. The pressure compensated pump is still flowing at maximum and pressure starts to climb. The pump will continue at full flow until pressure reaches 80 to 98% of the compensator setting. There has been zero flow needed for some time, but the pump does not know this until pressure is near maximum. When pressure reaches the compensator setting, the pump starts to shift to no-flow. All pump flow during the shifting time has no place to go, so this excess flow makes a pressure spike of five to ten times the compensator setting. This pressure spike can cause premature failure of the pump, plumbing, and actuators. An accumulator as shown will take in this small volume of oil to minimize the spike.

As with any accumulator installation, safety is important. When shutting a circuit down for maintenance, always drain the accumulators. A manual drain valve works, but the automatic drain shown on the facing page is better. When the pump starts -- and as long as it is running -- a pilot valve closes check valve B to block the drain port. Check valve A isolates the pump from accumulator back flow when it stops or fails. There is no electrical wiring needed, so the accumulator dump valve is invisible to the control circuitry.

*The pump is just starting in **Figure 1-45**, so pressure immediately climbs to accumulator pre-charge pressure. Flow continues until the accumulator is full and system pressure is at its maximum. Pilot-to-close check valve B blocks the drain path to tank when the pump starts. The drain path stays closed as long as the pump is running.*

Fig-1-46

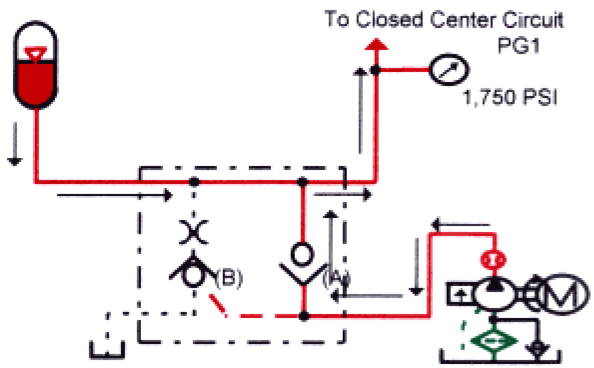


__Figure 1-46__ shows flow while the circuit is working. Accumulator and/or pump flow will go to the actuators to quickly start them and move them through their cycle. During the working part of the cycle, the accumulator smooths out flow fluctuations, while reducing pressure drops and spikes.

With the system at rest as shown in __Figure 1-47__, pump flow is zero and the accumulator is full and ready for another cycle.

Fig-1-47

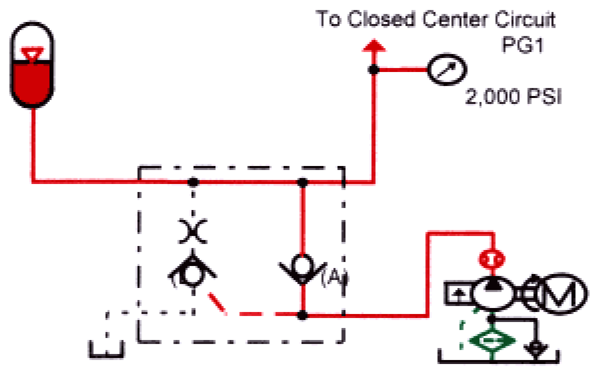
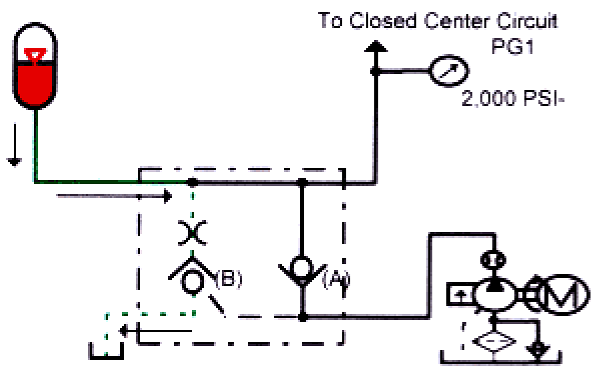


__Figure 1-48__ shows how the circuit responds when the pump stops. Check valve A closes to stop back flow and pump motoring. Pressure to pilot-to-close check B drops out, allowing it to open. All accumulator volume now has a path to tank through an orifice that keeps flow at a reasonable rate. In a very short time the accumulator's stored energy dissipates, making it safe to work on the system.

Fig-1-48



CAUTION! ALWAYS CHECK AN ACCUMULATOR CIRCUIT FOR PRESSURE BEFORE WORKING ON IT. NEVER ASSUME THE AUTOMATIC UNLOADING SYSTEM WORKED!

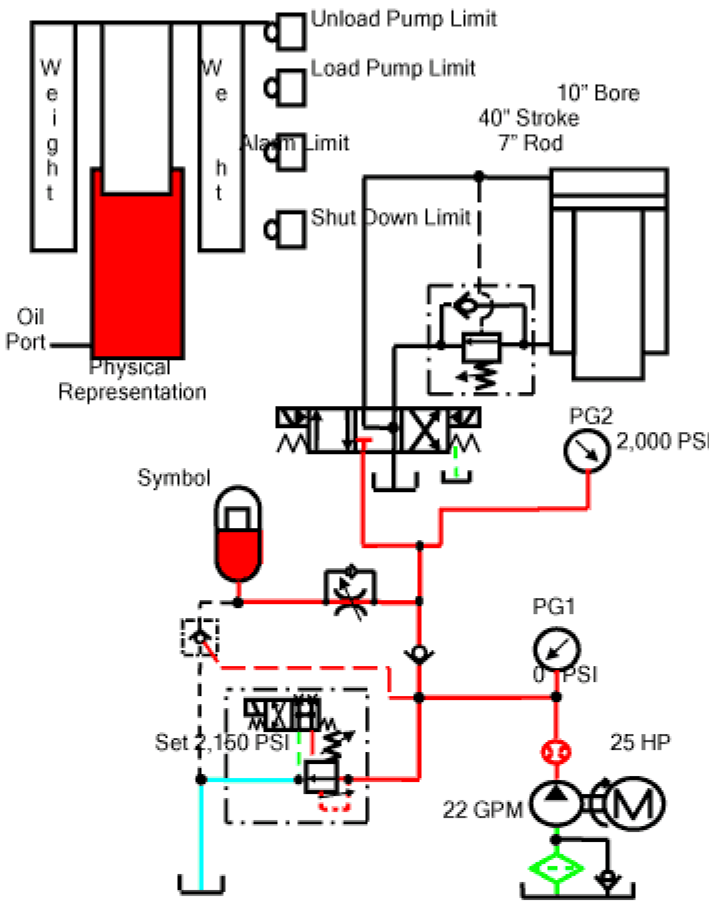
Linear pressure-type accumulators

The following circuits use accumulator types with little or no pressure drop as they discharge fluid.

Gas- or spring-loaded accumulators lose pressure as fluid discharges and the gas or spring expands. In a typical circuit using this type of accumulator, the maximum system pressure must be higher than working pressure to allow for this pressure drop. Some circuits cannot operate at these elevated pressures or may need high pressure for the entire stroke. Therefore, they can't use gas or spring loaded accumulators.

*The circuit in **Figure 1-49** shows a weight-loaded accumulator, a fixed-volume pump, and a normally open, solenoid-operated relief valve that can replace either circuit shown in **Figures 1-10 and 1-11.** Notice the maximum pressure and working pressure are at 2000 psi. This is possible because the weight-loaded accumulator does not lose pressure as fluid discharges. Until the accumulator piston reaches bottom, system pressure stays constant.*

Fig-1-49

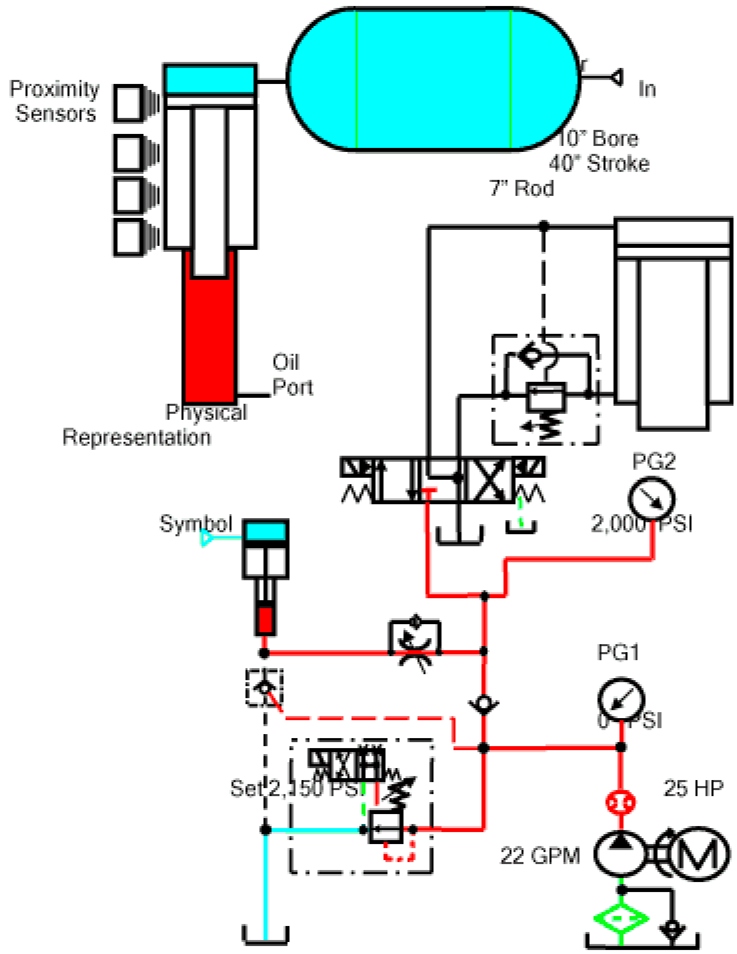


With a weight-loaded accumulator, the amount of weight on a given piston area sets maximum pressure. To raise or lower maximum pressure, weight must be added or taken off. Set the relief valve on this type circuit 100 to 150 psi higher than system pressure so it does not bypass during normal operation.

The main disadvantage of a weight-loaded accumulator is its physical size. An accumulator for the circuit shown in ***Figure 1-49*** *would require a 10-in. ram with a 60-in. stroke for the cylinder to have full force for its entire cycle. This size accumulator needs almost 160,000 lb of weight on the ram to get the required volume and pressure stated. A block of concrete approximately 1080 ft3 in size or about 10 X 10 X 11 ft would be necessary to meet this need. Such high mass eliminates the use of this type accumulator for mobile equipment and also rules out many industrial applications. Using a smaller accumulator ram with a longer stroke reduces weight, but you must make sure column strength is adequate when reducing ram diameter.*

The air-cylinder-loaded accumulator shown in ***Figure 1-50*** *works the same as a weight-loaded accumulator. There is a slight pressure drop as fluid starts to flow due to piston and ram seal friction but this is usually not enough to cause problems.*

Fig-1-50



Physical size can also be a problem with air-cylinder-loaded accumulators, especially when using low air pressure. Most plant systems operate at 100 to 125 psi so the unit required to handle the cylinder in ***Figure 1-50*** *might be a 40-in. bore air cylinder driving a 9-in. ram with a 75-in. stroke. Using air pressure at 250 psi could reduce the accumulator to a 30-in. air cylinder driving a 10_-in. ram for a 55-in. stroke. In either case, these accumulators are still too large for mobile equipment and for many industrial applications.*

Air-cylinder-loaded accumulators work best and are more economical to operate using a surge tank for the air cylinder. Surge tanks provide fast flow for discharging high oil volumes with minimal pressure drop. They also make it possible to use a small air compressor because it only has to make up for leaks after the system gets up to pressure. Size the surge tank to allow for a 3- to 8-psi drop when the accumulator discharges during a normal cycle.

Electrical and electronic devices, such as relay logic circuits, programmable controllers, or computers, normally control fluid power circuits. Fluid power systems can also 0be controlled with "Air Logic." These controls perform any function normally handled by relays, pressure or vacuum switches, time delays, counters, and limit switches. While the circuitry is similar, compressed air is the control medium instead of electrical current.

Environments high in dust or moisture are excellent places for air logic controls because practically no danger from explosion or electrical shock is possible even in these atmospheres. Water can splash on the controls with no effect on the operation. If there is danger of explosion, air controls can not ignite the materials involved.

Air logic can also be used on machines that have cylinders or fluid motors, but no type of electrical device. In such instances, two services are required because the machine is powered by air but controlled electrically. In cases where electrical and mechanical maintenance come under different labor grades, air logic is also ideal because different technicians work on different aspects of the machine -- one works on the circuit and the other handles the machine parts that are electrically driven.

Air logic does have its disadvantages; most common is the lack of understanding of how the components work and how to read the schematic drawing. If an air controlled machine fails, very few persons can work on it. Also, air logic with long control lines will have a noticeably slower cycle. Control lines longer than 10 to 15 ft fill and exhaust slowly when compared to electrical signals. In addition, air quality must be above average for long life.

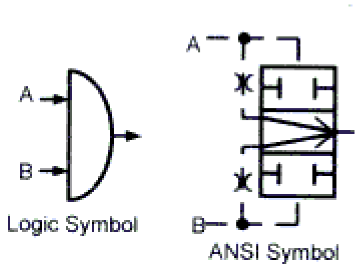
Air logic controls are basically miniaturized 3- and 4-way air valves. The actions of the valves provide on or off functions like relays or switches. They also exhaust the spent signal. The symbols used for air logic are similar to electronic symbols. Some manufacturers use modified electrical symbols and ladder diagrams to show circuitry.

The following is an explanation of the basic logic components showing the ANSI logic symbol and ISO graphic symbol for a comparable directional control valve.

And, or, and not symbols

Figures 2-1 and 2-2 *show two types of "and" elements, which must receive two inputs before it provides an output. This ensures that two functions have completed before there is a command to continue the cycle. This can also be described by saying "this input, this input, and that input must be present before getting an output. Connect "and" inputs in a series when using more than two inputs. The first "and" receives signals' one and two while the output of this element hooks to one input of the second "and." The other input of the second "and" receives the third signal, making three inputs necessary before giving an output.*

Figure 2-1: Passive "and" element



*Some manufacturers supply both types of elements. This gives you **Figure 2-1** "and" with **Figure 2-2** designated as "yes." The difference in elements is that the "and" in **Figure 2-1** uses the lower of the two inputs as an output. This is a passive element. In contrast, a "yes" element has two inputs which obtain an output, but the designer has the choice of which input pairs with the output. Using this feature can amplify a weak signal. The weak signal pilots the valve open while the through signal comes from a full pressure supply. The "yes" in this situation is an active element.*

Figure 2-2: Active "and" element

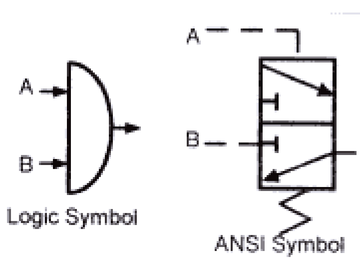
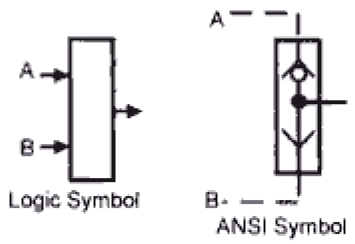


***Figure 2-3** shows the symbol for an "or" element. A shuttle valve serves the same purpose as an "or" element. Both inputs to an "or" element provide an output. A pilot signal from two different sources can pass through to start the next function. This can also be described by saying this signal or that signal provides an output. An "or" element differs from an inline "tee" because an "or" passes either input to the output but does not allow the inputs to pass to eachother.*

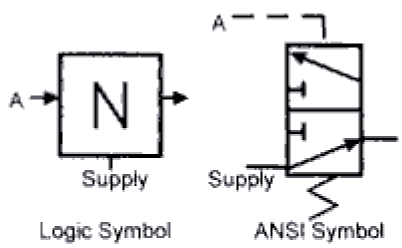
Figure 2-3: "Or" element



Stacking "or" elements allows for more than two inputs. Use an extra "or" element for each input after the first two signals.

***Figure 2-4** shows the symbol for a "not" element, which is a normally open 3-way valve. An input signal or pressure supply will go through the valve until there is a pilot signal at port A. Pressurizing port A blocks supply and exhausts the output signal to atmosphere. "Not" elements will block a signal or supply as long as there is pilot pressure on the A port. The "not" always returns to a normally open condition without a pilot signal.*

Figure 2-4: "Not" element



*Replace a limit switch with a "not" element to indicate a cylinder is at the end of stroke. Pressure from the cylinder port goes to port A of the "not" element, holding it closed. As the cylinder moves to the work, pressure stays steady because of the meter-out flow control. When the cylinder contacts the work, the signal on port A drops, the "not" element opens and sends a signal to start the next operation. See **Figure 2-21** and accompanying text to review a circuit using "not" elements to replace limit valves.*

The cylinder can stop at any position and the "not" output signal will indicate its nonmovement. This will always happen whether the cylinder stopped where it should have or if it stalled by some other means. Because this can happen, take care when using a "not" element to replace a limit switch. In contrast, this feature can be advantageous when clamping different sized parts. Use a "not" element for applications where different work locations stop the cylinder.

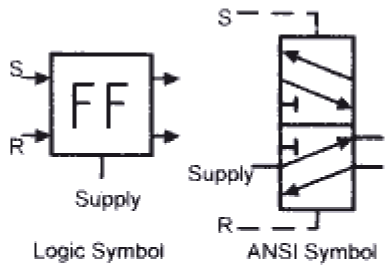
Most manufacturers supply a different pilot ratio for a "not" element used as a limit switch. The valve function is the same but it shifts at much lower pressure. Some manufacturers make a *special "not" element that mounts directly to a cylinder port. A port-mounted meter-out flow control used in conjunction with this special "not" element makes a compact installation.*

Caution! Pressure control valves only show pressure buildup. When a positive location must be made, use limit valves.

Flip-flop circuits

"Flip flop" elements, with their symbol shown in Figure 2-5 are double piloted 5-way valves that send supply air to either outlet port with a signal at pilot ports S or R. Supply can be system pressure or air from another logic element. The main use for a "flip flop" is to eliminate the first pilot signal to a directional control valve. This allows a second signal on the directional valve's opposite pilot port to shift it back. "Flip flops," sometimes called "memory" elements, stay in their last shifted position even with no air supply. Whether the signal maintains or drops out, output from the "flip flop" stays the same.

Figure 2-5: "Flip flop" element

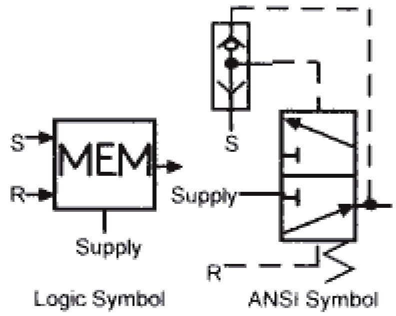


The S and R signals stand for "set" and "reset." The "set" signal shifts the "flip flop" for a function; whether the signal continues or not, the element stays shifted. The "reset" signal returns the "flip flop" back to its original position until the next cycle.

"Flip flop" can also be used to set up a new cycle, allowing the operator to momentarily push the start buttons. Use this same "flip flop" to eliminate unwanted signals and set up the circuit for cycle completion as required.

Figure 2-6 *shows another valve actually called a "memory" element, which is a normally closed 3-way valve with a built in shuttle valve. The shuttle valve uses the "memory's" output air to hold it shifted once it receives an S "set" signal. A momentary "set" signal gives continuous pilot output. An R "reset" signal shifts the "memory" element to normally closed and exhausts output air. In addition, turning supply pressure off returns a "memory" element to its start position.*

Figure 2-6: "Memory" element

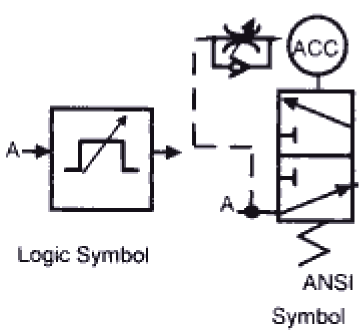


There are three different types of time delays in air logic control. Fixed- or adjustable-time delays are common in both normally closed and normally open configurations. Some time delays use an orifice and accumulator chamber for delays up to one minute. Some manufacturers use air actuated diaphragms and orifices that eliminate system pressure fluctuation inaccuracies.

*A "one-shot" timer, shown in **Figure 2-7**, is sometimes called an "impulse timer." A "one shot" timer takes a signal and passes it on to the circuit. At the same time, the input signal goes through an orifice to an accumulator tank. The setting of the orifice and size of the accumulator give a certain time delay before the normally open 3-way valve closes. After a "one shot" times out and closes, it remains closed as long as it has an input signal.*

***Figure 2-7** shows an adjustable time delay before it loses its output. Leaving off the sloping arrow in the symbols makes it a preset time delay. Times range from to 2 or more sec on valves with preset time delays.*

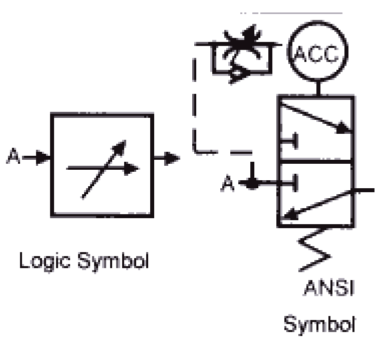
Figure 2-7: "One shot" element



*Many circuits uses "one shots" to eliminate opposing signals. When a valve receives a signal to extend to a cylinder, it resists a return pilot signal to itself until loss of the first pilot. Using a "one shot" element drops the extend signal shortly after iniatiation. However, when the short duratoinj signal meets a hard-to-shift valve, the time may not always be long enough to move the valve spool. The cycle will stall if the valve does not have time to shift. For best results, use a "flip flop" to drop unwanted signals after it performs a task. **Figures 2-17 to 2-20** and accompanying text further describe "flip flop" valves.*

Passing a signal through the element after timing stops is done with an adjustable, normally closed "time-on" time delay. Figure 2-8 shows the symbol for this element. A "time-on" delay is a preset fixed timer without the sloping arrow. Most anti-tie down circuits use a fixed time delay, thus forcing the operator to actuate both palmbuttons concurrently.

Figure 2-8: "On delay timer" element



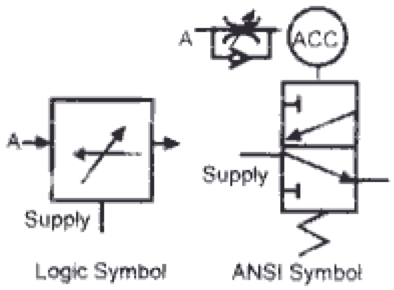
*The symbol in **Figure 2-8** shows an input A moving towards the blocked port of a 3-way directional valve. Signal A also moves to a meter-in flow control to fill an accumulator. After the accumulator is filled, pilot pressure shifts the 3-way valve, allowing air to pass on to the next operation. As long as the input signal stays on the time delay stays open.*

Some brand of "time-on" delays use shop air to the normally closed port A of the 3-way valve while the signal to the timing section comes from another logic element or limit valve. This allows a strong passing signal to travel long distances or to quickly shift several other logic elements.

With a built-in accumulator tank, the time delay length is usually unjer one-to-one and one-half minutes. With added external accumulators, time delays up to 5 min are possible. The repeatability of long time delays using accumulators is poor. Often, diaphragm type timers go to 3 min with good repeatability.

*With a normally open 3-way valve in place of a normally closed 3-way, the delay is "time off." **Figure 2-9** shows the symbol for a "time off" delay timer. A continuous input to the supply gives an output until a set time after receiving a signal at A. When A receives a signal, the time delay starts and continues timing. When the accumulator fills, it closes the normally open 3-way valve and exhausts the signal. As before, a preset, non-adjustable time delay is available.*

Figure 2-9: "Off delay timer" element



"Time-on" and "time-off" delays often are identical in appearance. The part number may be the only way to tell these units apart.

To get different functions, connect air logic elements together like the examples in ***Figures 2-10 and 2-11****. These two common pairs might be familiar to anyone using air logic. A "nand" element, shown in* ***Figure 2-10****, uses an "and" to signal a "not." The term "nand" means "not this and this." As long as there are not signals at A and B, air passes. If signals A and B are present, the "not" closes and exhausts the output signal.*

A "nor" element, shown in Figure 2-11, uses an "or" to signal a "not." The term "nor" means not this or this. As long as there is not a signal at A or B, air passes through the "not." If a signal is present at either A or B), the "not closes and exhausts the output signal.

Figure 2-10: "Nand" element

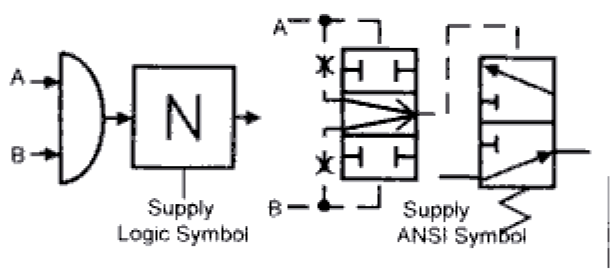
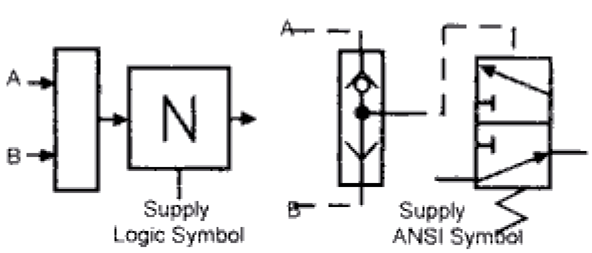


Figure 2-11: "Nor" element



Some other commonly used air logic elements include:

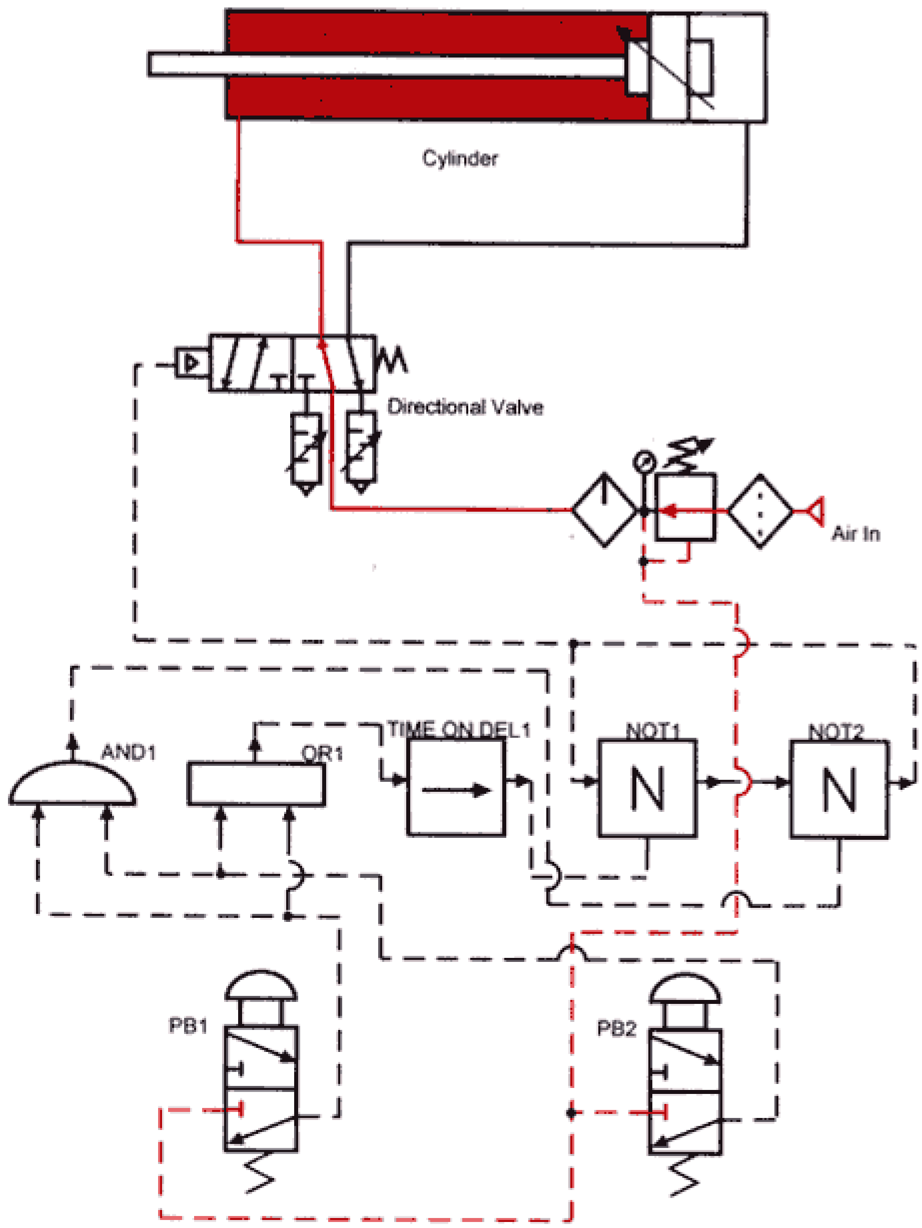
- Amplifiers to detect a low pressure signal (down to 3-in. water column) and send it on as an 80 psi signal.
- Pressure or vacuum sequence elements shift after reaching a set pressure or vacuum.
- Programmable controllers are combination elements that are used to design complex circuits with minimum knowledge of circuit design.
- Air-operated indicators show circuit condition and/or function. Several colors are available but none emit light.

The following text and images depict examples of air logic circuits, showing how some basic circuits perform machine control functions.

Anti-tie down air logic circuit, using logic symbols

*The two-hand, anti-tie down circuit schematic in **Figure 2-12** uses ANSI air logic symbols to simplify schematic drawings. However, most mechanics do not understand the hardware behind the symbols. An electrician may recognize the symbols but often does not understand how air logic functions. So like most problems with hydraulics and pneumatics, changing parts, turning knobs, and swapping lines continues until the machine starts working or an expert is called.*

Figure 2-12: Anti-tie down air logic circuit using logic symbols



*To make the cylinder in **Figure 2-12** extend, depress both palm buttons at the same time and hold them shifted. Tying either palm button more than one second before actuating the second palm button keeps the cylinder from moving. Depressing the second palm button within one second after the first palm button makes the cylinder extend and stay. Letting up on either or both of the palm buttons causes the cylinder to retract. This means that if the operator tries to use one of his hands to adjust or hold a part, the cylinder retracts. To start another cycle, release both palm buttons to reset the time delay. Both of the operator's hands must stay on the palm buttons when the cylinder is extending.*

Notice that the AND1 and OR1 elements on the left receive signals from the palm buttons at the same time. AND1 uses both signals to get an output while ORr1 gives an output when depressing either palm button.

Actuating palm button PB2 sends a signal through OR1 to start TIME ON DEL1. After approximately one half to one second, TIME ON DEL1 opens, sending a signal through normally open NOT1 to close normally open NOT2. Depressing PB1 after NOT2 closes gives an AND1 output, but it cannot go through to shift the directional valve. Actuating either palm button separately blocks the signal to shift the directional valve at PB2.

Shifting both palm buttons concurrently sends a signal through OR1 starting TIME ON DEL1. At the same time, an output from AND1 passes through NOT2, shifting the directional valve to extend the cylinder. The output ofNOT2 also closes NOT1, blocking the output from TIME ON DEL1. Depressing and holding both palm buttons extends the cylinder and keeps it there.

Releasing one palm button while the cylinder is extending drops one output of AND1. When AND1 drops out, the directional valve spring returns, the cylinder retracts, NOT1 opens, and TIME ON DEL1 output closes NOT2. Depressing the released palm button again leaves the cylinder retracted because TIME ON DEL1 closes NOT2. To start another cycle, release both palm buttons to reset TIME ON DEL1. ***Figure 2-13*** *shows this operation using ISO valve symbols.*

Anti-tie down air logic circuit, using ISO symbols

Figures 2-13 to 2-16 *show the previous anti-tie down circuit in ISO symbols. Most people understand ISO symbols since they show valve function more clearly.*

The circuit is at rest in ***Figure 2-13****. PB1 and PB2 are not actuated, so there is no signal being sent to the directional valve. This schematic is what the machine supplier sends with his documentation for the machine.*

Figure 2-13: Anti-tie down air logic circuit at rest,
air on

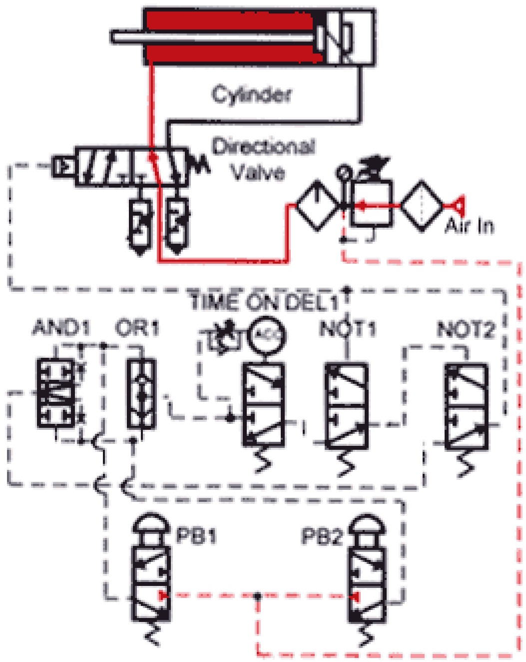
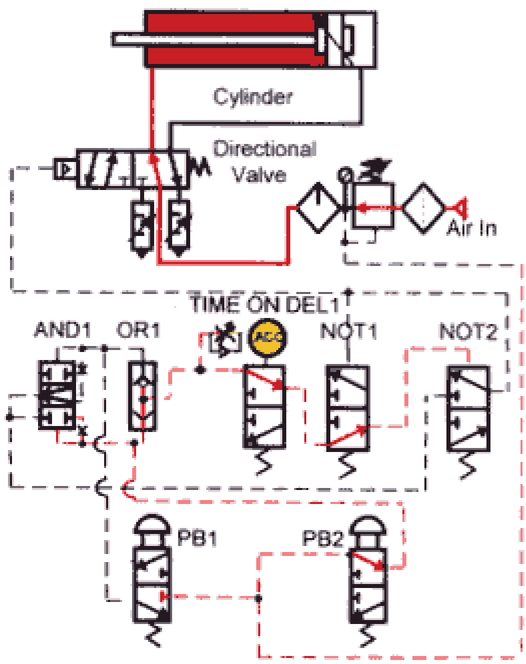


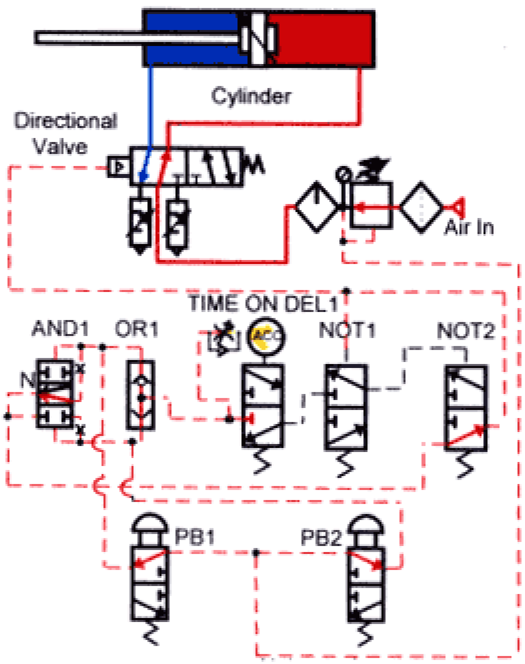
Figure 2-14 *shows the circuit when depressing only one palm button. Here, PB2 sends an air signal to one port of the and element. The and element does not send any output because it needs two signals. The air signal from PB2 does go through the or element and starts TIME ON DEL1 timing. If PB1 is not shifted within a short time, TIME ON DEL 1 times out. When TIME ON DEL1 times out, it sends a signal through NOT1 and closes NOT2. After TIME ON DEL1 closes NOT2, the signal from PB1 through the and element becomes blocked at NOT2. Either palm button gives the same results. This protects the operator because depressing both palm buttons is the only way to cycle the machine. Before a cycle is possible in this condition, the palm buttons must exhaust any air signal from the or element, resetting TIME ON DEL 1.*

Figure 2-14: Anti-tie down air logic circuit, one palm button depressed



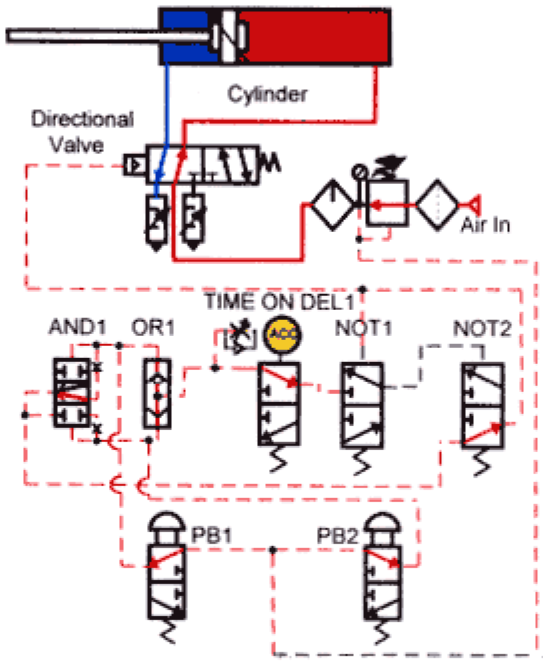
*Depressing both palm buttons simultaneously gives the results shown in **Figure 2-15**. Either one of the signals to the or element starts TIME ON DEL 1but one of the signals to the and element passes through NOT2, on to the directional valve. The signal from NOT2 also pilots NOT1 closed, blocking the output from TIME ON DEL 1. As long as both buttons stay shifted, the cylinder extends and holds.*

Figure 2-15: Anti-tie down air logic circuit, both palm buttons just actuated



After TIME ON DEL 1times out, the circuit changes to the one shown in ***Figure 2-16****. Output from TIME ON DEL 1L1 stops at NOT1 because the start signal from the and element is holding it closed. With this circuit the operator has to keep both hands on the palm buttons to make the cylinder extend.*

Figure 2-16: Anti-tie down air logic circuit, both palm buttons actuated, time delay timed out



The following circuits show other uses for these elements and how more complex circuits use other logic valves.

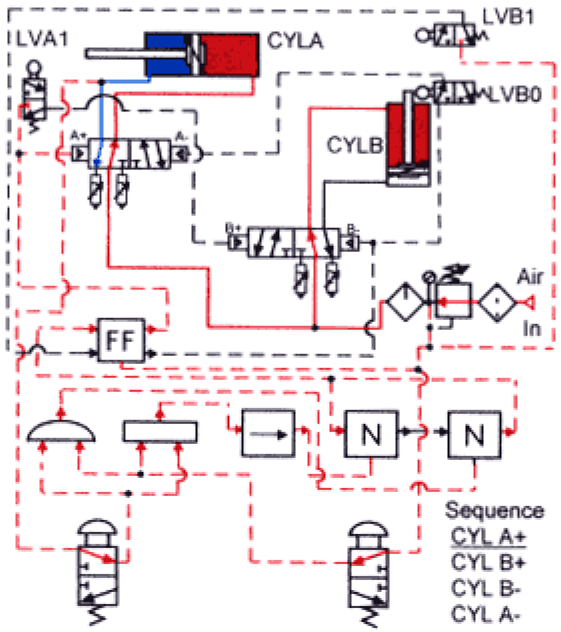
Anti-tie down, non-repeat, flip flop air logic circuit

Figures 2-17 through 2-20 *show a two cylinder circuit with CYLA extending (A+, CYLB extending (B+), CYLB retracting (B-), and CYLA retracting (A-).*

Notice CYLB retracts immediately after extending, which means there would be an extend signal opposing a retract signal if the circuit only has limit valves for control. Using a "one shot" valve to stop the opposing signal works, but is less reliable than the "flip flop," FF circuit shown here.

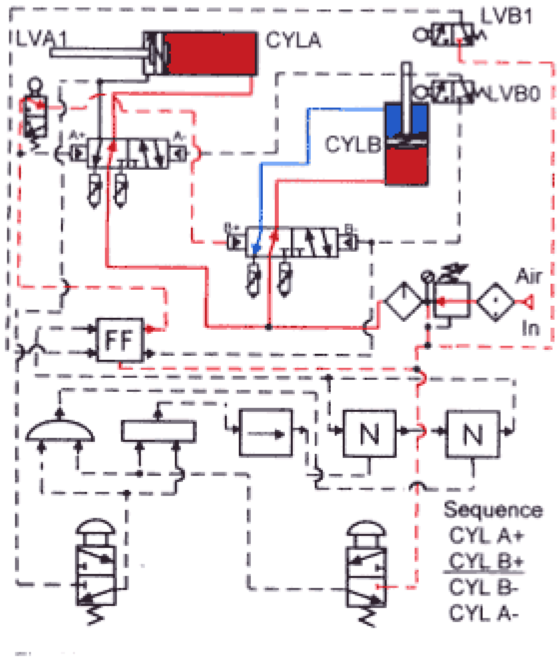
Figure 2-17 *shows both palm buttons depressed, causing the output of the anti-tie down circuit (see the previous explanation of an anti-tie down circuit) to shift FF. The output of the top port of FF sends a signal to shift a doubled pilot valve and extend CYLA. The FF output will also supply the normally closed port of limit valve LVA1. Shifting the FF also drops the signal from limit valve LVB0 that retracted CYLA. CYLA extends until it contacts limit valve LVA1.*

Figure 2-17: Anti-tie down, non-repeat flip flop circuit, cylinder *A* extending



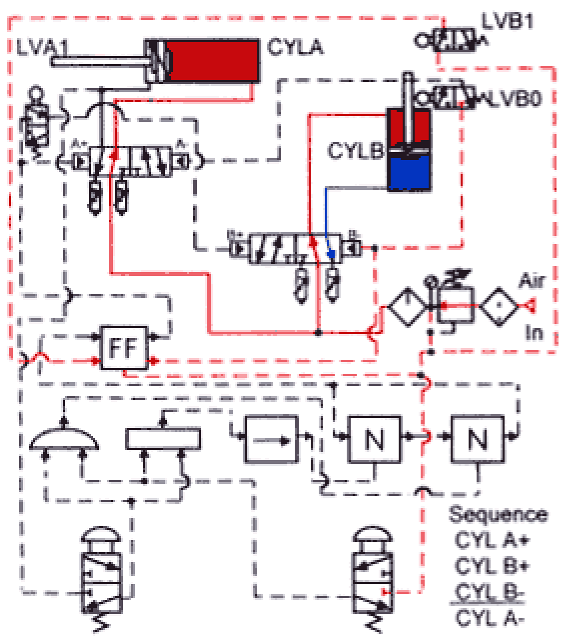
*When CYLA contacts LVA1, **Figure 2-18**, air from the top port of FF passes through it and shifts a double piloted valve making CYLB extend. The signal to retract CYLB came from the bottom port of FF that is now exhausting to atmosphere. CYLB continues to extend until it contacts limit valve LVB1.*

Figure 2-18: Anti-tie down, non-repeat flip flop circuit, cylinder *B* extending



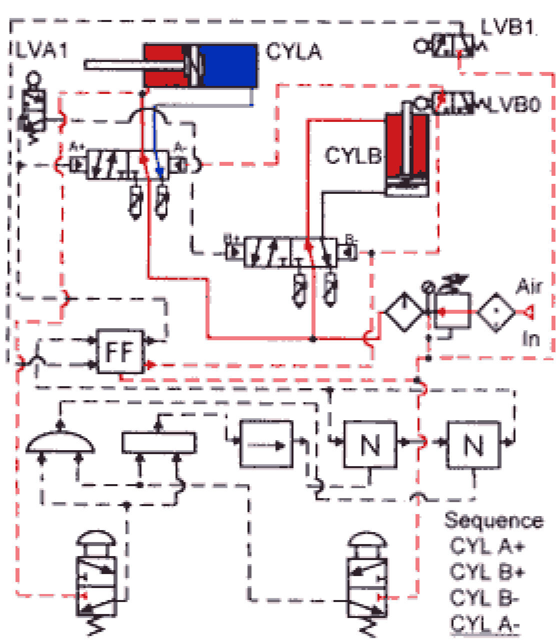
The normally closed inlet port of LVB1 has a constant air supply, so when CYLB contacts it, ***Figure 2-19****, it shifts FF back to starting position. A signal from the bottom port of the FF shifts a double-piloted valve to retract CYLB and supplies air to the normally closed port of limit valve LVB0. After FF shifts back to the starting condition, it drops the extend signals to both double piloted directional valves. This makes it possible to shift the double-piloted valves to retract the cylinders. CYLB continues to retract until it contacts LVB0.*

Figure 2-19: Anti-tie down, non-repeat flip flop circuit, cylinder *B* retracting



A signal from LVB0 shifts the double-piloted valve to retract CYLA as shown in ***Figure 2-20****. This cylinder can retract since its extend signal dropped out when FF shifted from LVB0. CYLA retracts to home position and ends the cycle.*

Figure 2-20: Anti-tie down, non-repeat flip flop circuit, cylinder *A* retracting



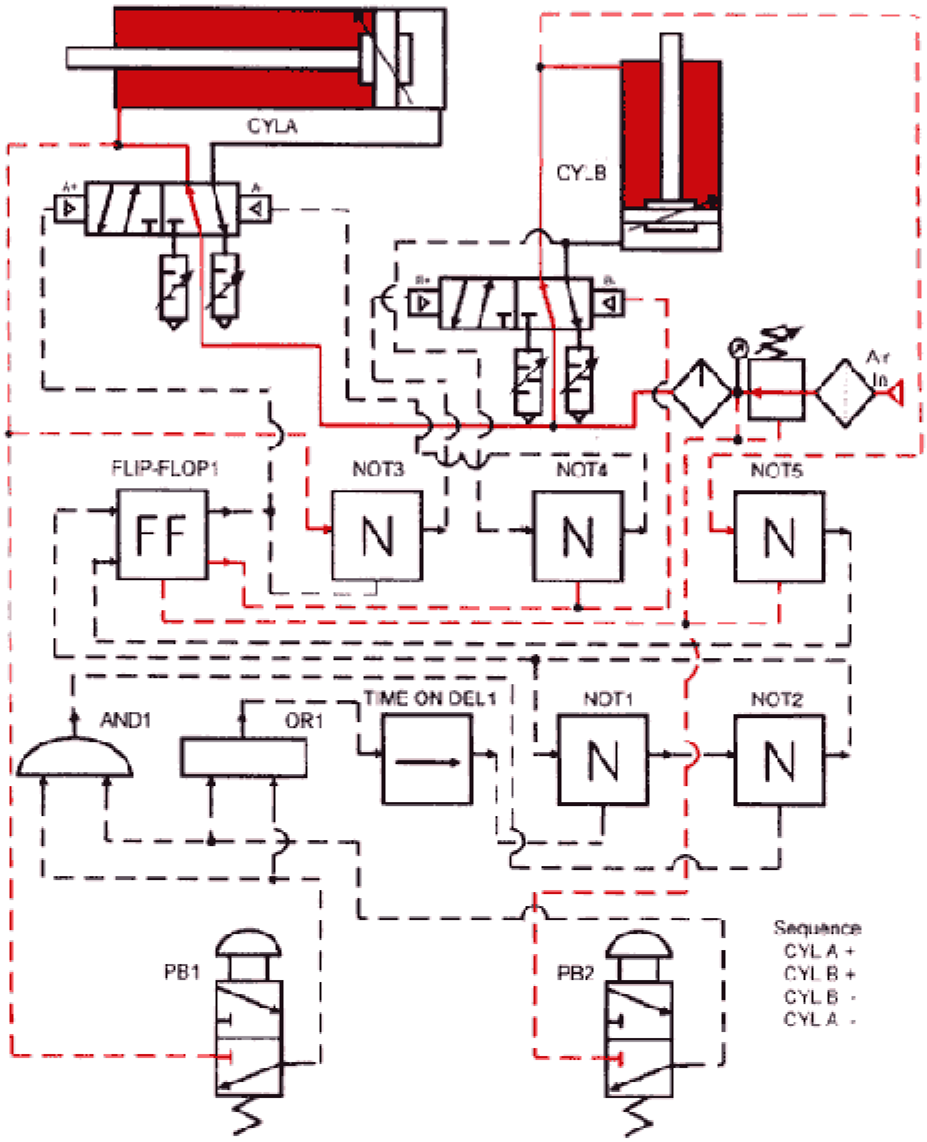
The nonrepeat feature is possible because when the circuit is in the at rest position, there is a supply to the left palm button from the rod end port of CYLA. After the cycle starts and CYLA reaches the end of its stroke, the left palm button loses its supply. Whether the operator lets off the palm buttons or not, loss of air to the left palm button disables the anti-tie down circuit.

With both palm buttons supplied with direct shop air, if the operator kept the palm buttons shifted all during the cycle, the machine would probably stall after CYLB extended. The nonrepeat feature adds little cost, but may save lost production.

Using modified "not" elements as limit valves

The circuit in ***Figure 2-21*** *operates the same as* ***Figures 2-17 through 2-20*** *on the preceding page. The only difference is pressure controlled "not" elements replace limit valves.*

Figure 2-21: Anti-tie down, non-repeat and flip flop circuit, using modified "not" elements as limit valves



"Not" elements can replace limit valves when the movement they are detecting is not critical. "Not" limits operate any time the cylinder has a pressure drop. The pressure drop could be end of stroke or any place the cylinder stops for any reason. ***If actuator position is critical, always use limit valves.***

Using a standard "not" to replace limit valves works, but the special low pressure "not" is best. Some manufacturers call this an "inhibitor", others, a "pressure trip release." Whatever the name, the modification causes the valve to shift at a lower differential pressure. This keeps a reduced backpressure at the cylinder port from giving a premature signal.

Using "not" elements in place of limit valves makes installation and plumbing easier, but can make troubleshooting more difficult. Placing the "not" elements in the control box works, but cylinder port mounting is best. No matter the location, they must read the air between the *cylinder port and a meter-out flow control. This location ensures they see backpressure when the cylinder is moving.*

Because a "not" is normally open, pressure holding the cylinder in position and backpressure from a meter-out flow control when the cylinder is moving give the signal to hold it shut. When the cylinder stops, pressure drops, allowing the "not" to open and send a signal to continue the cycle.

*The circuit in **Figure 2-21** uses "not3" to tell CYLB to extend, "not5" to tell CYLB to retract, and "not4" to tell CYLA to retract.*

Since the "not" works on loss of pressure, a cylinder with leaking seals can keep it from shifting. After a slowly moving cylinder stops, slow deterioration of pressure may delay the output signal.

Loss pressure valves have many benefits. For example, it does not matter when the cylinder contacts the part. Whether the part is 1 or 20 in. thick, when the cylinder makes contact, there will be an indication. In addition, air pressure changes have little or no effect on them as the "not" only reads minimum pressure. Finally, maximum pressure setting does not affect a "not" like it does a sequence valve. "Not" elements are a preferred choice over sequence valves because sequence valves only work with meter-in flow controls. Any air cylinder has better control with a meter-out circuit and overrunning loads require meter-out flow control.

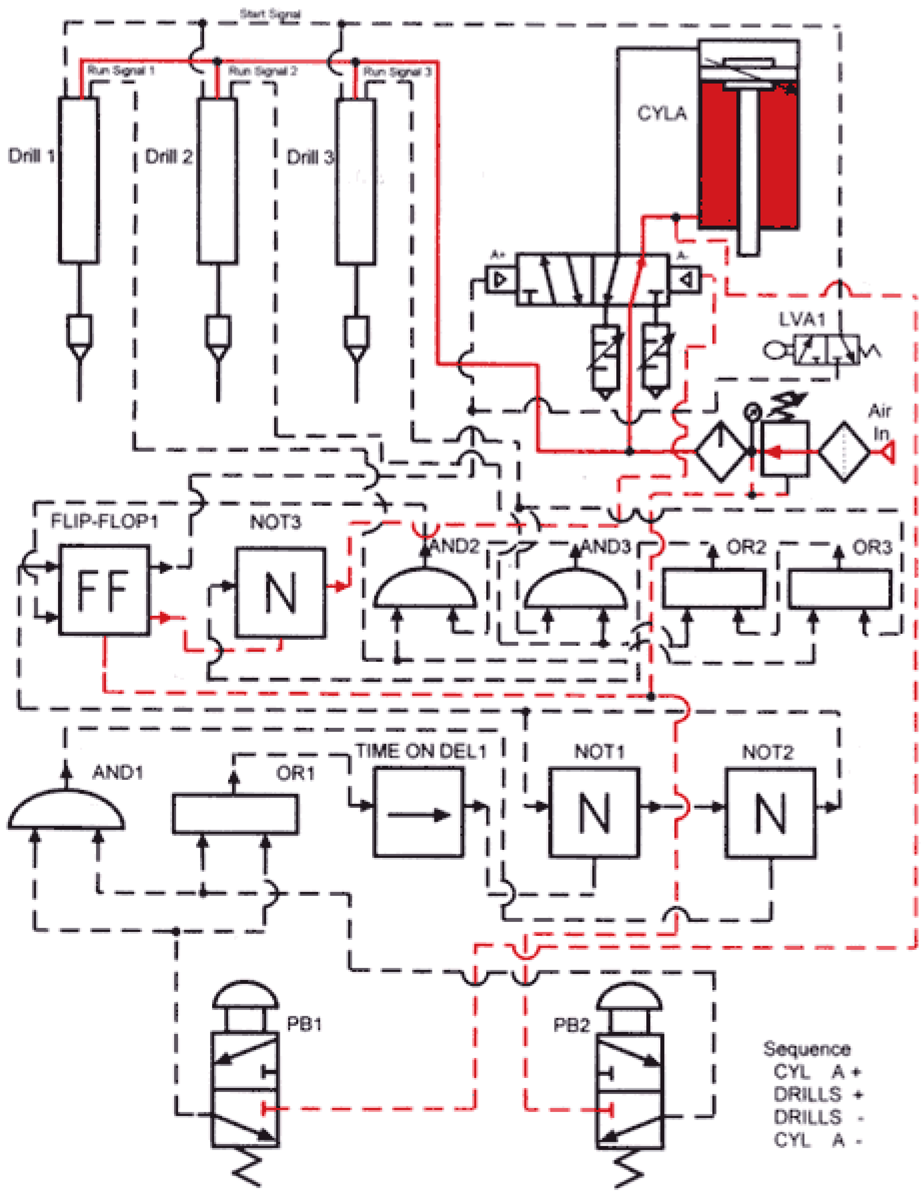
Always use loss of pressure controls with caution since they can operate any time cylinder pressure drops below their minimum shifting pressure.

Anti-tie down, non-repeat and flip flop air logic circuit with automatic cycling air drills

***Figure 2-22** shows a clamp cylinder, CYLA, and three self-contained automatic air powered drills controlled with air logic.*

*Some circuits clamp a part then start the drills with a "one shot" element. As long as all the drills start there is no problem. However, if any drill fails to cycle, parts may come off the fixture with one or more holes missing. When double drilling is necessary, part costs' and scrap increase. The circuit in **Figure 2-22** eliminates this problem with air logic elements and piping.*

Figure 2-22: Anti-tie down, non-repeat and flip flop circuit, with automatic cycling air drills



When the anti-tie down circuit shifts "flip flop" FF, a signal from its top port goes to extend clamp cylinder CYLA. FF top port output also supplies the normally closed port of limit valve LVA1. CYLA extends, clamps the part, and shifts limit valve LVA1. Use a limit valve here since the drills could sling a loosely clamped part out of the fixture. A pressure operated "not" circuit could allow premature cycling of the drills, resulting in damage and safety concerns.

After clamping the part, FF output goes through limit valve LVA1 to the drills' start ports. When the drills start they give a run signal when they move from home position. This run signal remains on until the drills fully retract. On most of these types of air-operated drills, the output is the same air that turns the air motor in the drill. One brand of air drill has an output when at rest and exhausts that signal when it starts.

The drill run signals go to the input ports of two "ands" and two "ors". When the two cascaded "ands" have three signals indicating all the drills are moving, their output shifts FF back to its starting position and exhausts the drill start signal. The three inputs to the two cascaded "ors" pass through to close "not3" so the clamp will not open until all drills have retracted. Output from the lower port of FF goes to the inlet of "not3" to set up clamp CYLA open sequence.

The drills will continue forward until they meet their internal limit valves and retract. The run signals drop out as each drill finishes and retracts to home position. When the last drill is home, the run signal from the last "or" element exhausts and "not3" opens. When "not3" opens, its output shifts the clamp valve to retract CYLA.

If starting of one of the drills is sluggish, the run start signal stays on until it moves. If a drill fails to start, the run signal stays on and the running drill stay extended. In either case, the operator knows when a problem exists. If one of the drills hangs in the part, the clamp will not open until the drill is free to retract. For every added drill, use another "and" and "or" element. With air indicators installed in each drill run signal line, picking out a nonrunning drill is easy.

Air-over-oil systems

Compressed air is suitable for low power systems, but air compressibility makes it difficult to control actuators smoothly and accurately. Some low power systems need smooth control, rigidity, or synchronization capabilities normally associated with oil hydraulics. All of these features are available to low power circuits by using compressed air as power and oil for control. Purchased or special built air-over-oil circuits provide smooth control when power requirement is low.

Some manufacturers make self-contained air-powered cylinders with built-in oil cylinders and reservoirs. Air provides thrust while oil controls speed and/or mid stroke stopping. Some units have two speed capabilities as well as stop and hold options. ARO Corp.'s "Coaxial Cylinder" and Schrader Bellows' "Air Motor-Hydro-Check" are two such units. Both of the above companies also make a bolt-on control unit that mounts on a standard air cylinder and controls speed and/or position.

Coupling low-pressure hydraulic cylinders with air-over-oil tanks is another common way to create an air-over-oil system. These tanks hold more than enough oil to stroke the cylinder one way. Having an air valve piped to the air-over-oil tanks forces oil from the tanks into the cylinder. Add flow controls and stop valves to the oil lines to give smooth accurate cylinder control. Air-over-oil tanks do not intensify the oil, no matter the tank diameter or length. The amount of air pressure supplied is the highest possible oil pressure available.

Tandem cylinders can also control oil and air power. The single-rod cylinder of the tandem runs on air, while the double-rod cylinder is full of oil. Because volume is equal in both ends of the double-rod cylinder, oil flows from end-to-end through flow controls and/or shut off valves for accurate speed and stopping control.

When designing with air-over-oil systems, take care sizing the oil lines. Most air-oil circuits operate at 100 psi or less, so any pressure drop in the circuit can reduce the force drastically. If oil lines are undersized, cylinder movement will be very slow. Size most air-over-oil circuit oil lines for about 2 to 4 ft/sec velocity. This low speed requires large lines and valves but is necessary if average travel speed with maximum force is important.

Bleeding of air from the oil chambers can present another common problem with air-over-oil circuits. Any trapped air in the oil will make the cylinder spongy. This compressibility makes accurate midstroke stopping and smooth speed control hard to attain. When using an air-over-oil tank system, it is best to mount the tanks higher than the cylinder they feed. All lines between the cylinder and the tanks should slope up to them. Also, if possible, let the cylinders make full strokes to purge the air. With dual oil tank systems, incorporate a means for equalizing tank level in the design.

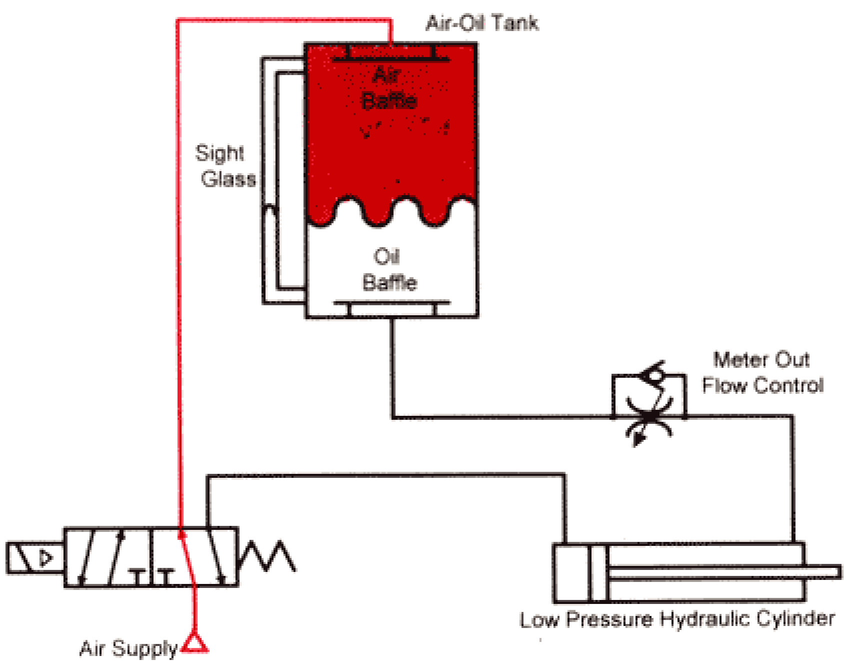
The cylinder seals must be as leak-free and provide the lowest amount of friction as possible. Any leakage past the seals can cause tank overflow, oil misting, and loss of control. The *following examples can be used to provide low power with smooth control rather than oil hydraulics.*

Be sure to read Chapter 13 on Intensifiers to learn more about air-over-oil circuits that also require short, high-force work strokes.

Air-over-oil tank systems

***Figure 3-1** shows a single tank air-over-oil system with an air valve piped to an air-over-oil tank to power the rod end with its other port connected directly to the cylinder cap end. A flow control valve in the oil line controls the cylinder advance speed. The cylinder's extend speed is very slow and controllable with this setup. Also, external forces trying to extend the cylinder cannot make it move faster than the flow control allows. If force requirements keep changing while the cylinder is moving, speed will change very little as long as the cylinder has enough force.*

Figure 3-1: Single tank air-over-oil system



Leakage of air past the piston seal can be a problem in a single tank air-oil circuit. Any air mixed with the oil can cause erratic movement and reduced cylinder control. Oil leaking past cylinder seals can cause misting and the environmental problems associated with it.

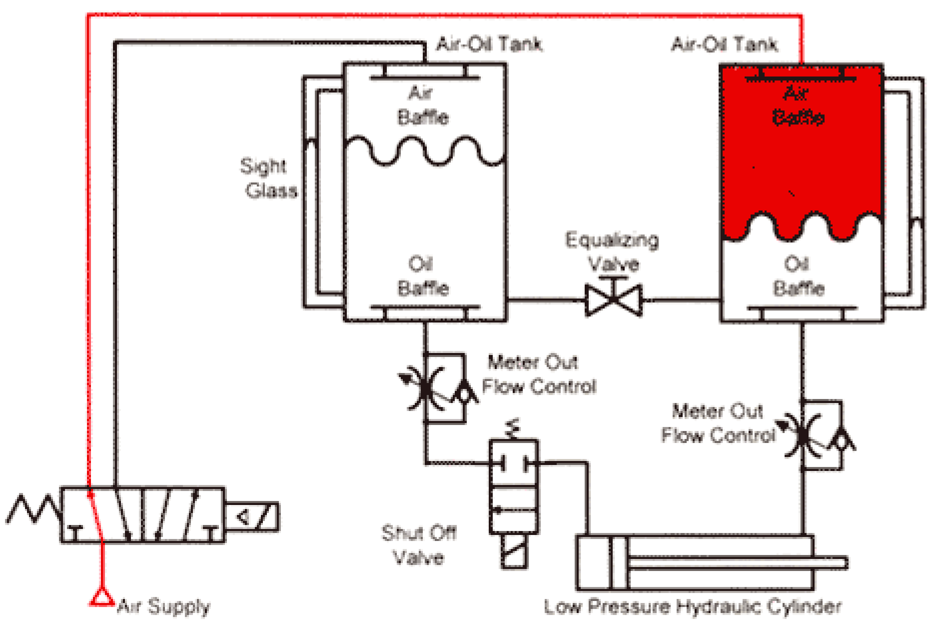
This circuit is good for replacing a straight air cylinder that is moving a large load. When load force requirements on the extend stroke are variable, a straight air cylinder may stop or lunge uncontrollably. With oil in the rod end of the cylinder, movement will be smooth. Even if the cylinder stops, it will not lunge when it starts again.

*Notice the **baffles** in the air oil tank. The top baffle keeps incoming air from shooting into the oil, causing aeration. The bottom baffle stops any vortexing that might allow air to enter the cylinder as the tank level lowers.*

***Sight gauges** are also necessary to monitor oil level on the air-over-oil tank. Purchased air tanks usually come with baffles, sight gauges, and a fill port. Some purchased air-over-oil tanks feature air cylinder tubing with two cylinder caps held in place with tie rods. Air-over-oil tanks should be sized to run approximately 75% full for optimum performance.*

***Figure 3-2** shows a double tank system. Two tanks eliminate the aeration problem caused by leaking cylinder seals. They will also allow accurate flow or stop control in both directions of travel. A twin tank system is more expensive, and may slightly reduce maximum cylinder speed.*

Figure 3-2: Double tank air-over-oil system



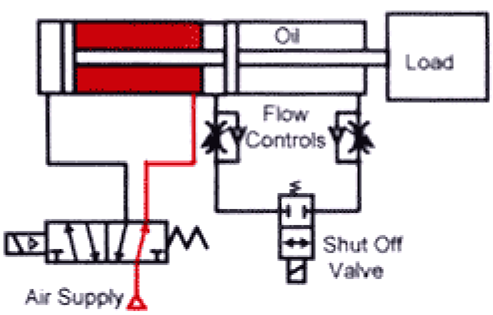
*When using two oil tanks, always put a line with an **equalizing valve** between the tanks. This line should be below the lowest tank oil level to allow easy adjustment of tank oil level when cylinder seals bypass.*

*When filling air-over-oil tanks, stroke the cylinder toward the low one. This action exhausts the tanks' air for fill-plug removal and allows replenishing without overfilling. Also there is a minimum of trapped air in the piping and the cylinder. **Never fill a tank to capacity when the cylinder is stroked away from it. If oil gets into the cylinder as well as the tank, it will overflow as soon as the cylinder strokes**.*

Install a solenoid or pilot-operated shut-off valve in the oil lines to the cylinder for accurate control. A ***shut-off valve*** *makes it possible to precisely position and hold the cylinder for long periods. Since oil is practically non-compressible, a cylinder will not move even if outside forces push against it. If excessive outside force is possible, install a relief circuit to protect the cylinder and machine structure. Tandem cylinders in air-over-oil circuits*

Figure 3-3 *shows a tandem cylinder piped for speed control and mid-stroke stopping. A tandem cylinder unit provides the same results as an air-over-oil tank system. An air valve piped to the single-rod cylinder gives directional control. Install* ***meter-out flow controls*** *and a* ***shut-off valve*** *in the line between ports on the double-rod cylinder. The double-rod end cylinder is completely full of oil and sealed against leakage. A small* ***makeup tank*** *attached to the transfer line, with a* ***check valve*** *between it and the cylinder port, allows oil to enter the cylinder when needed.*

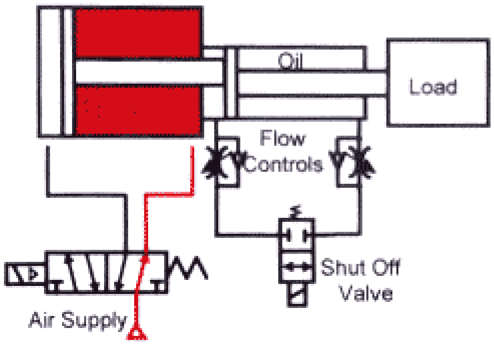
Figure 3-3: Tandem cylinder air-over-oil circuit



This tandem cylinder arrangement operates smoothly and gives accurate mid-stroke stopping control. A tandem cylinder is about the same price as a dual tank system. However, space requirement at the workstation more than doubles. Also, long-stroke tandems can be hard to mount and maintain. Normal piston seal leakage is no problem, but rod seal leaks can let oil out and/or air in, causing a house keeping problem and spongy cylinder movement.

Figure 3-4 *shows a large-bore, high-force air cylinder with a small-diameter hydraulic cylinder in tandem with it. An unmatched tandem cylinder has ample oil for smooth control and/or precise positioning. Unmatched tandems offer two big advantages: less cost and lower oil flow.*

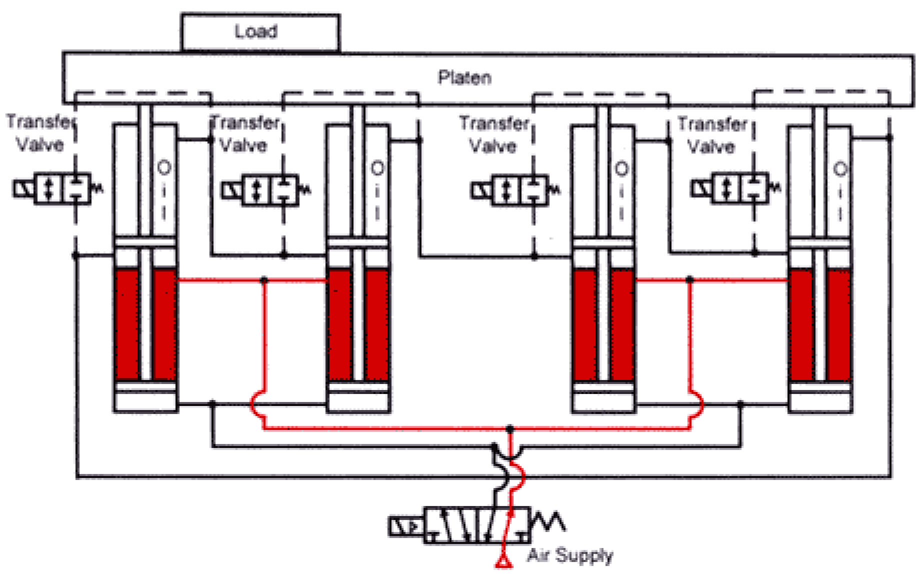
Figure 3-4: Unmatched tandem cylinder air-over-oil circuit



When specifying unmatched tandem cylinders, make sure the piston rod sizes are compatible between the air and oil cylinders. Rod size usually dictates the smallest bore hydraulic cylinder possible. Also, check for excessive pressure in the oil cylinder that could damage it and flow-control or stop valves. The higher oil pressure comes from intensification due to different bore sizes. Also, a vertically mounted cylinder with a heavy load adds load-induced pressure to intensification, possibly damaging components rated for air or low-pressure hydraulic service.

Figure 3-5 *shows an accurate way of synchronizing two or more air cylinders with fluid power. Four tandem cylinders are hooked to one platen, which will stay level throughout the stroke. Force from all four cylinders is available to move the load, regardless of its position.*

Figure 3-5: Tandem cylinder air-over-oil circuit for synchronizing two or more air cylinders



Notice the ***transfer valves*** *between the tandem cylinder's cross port lines. Always return a fluid-type synchronizing circuit to a positive home position often, preferably after every cycle. When* *the air cylinders return the platen to home position, the* ***transfer valves*** *open to allow the cylinders to resynchronize. Doing this after each cycle eliminates any cumulative position error from seal leakage.*

When using unmatched tandem cylinders in a synchronizing circuit, check for overpressure from intensification and load-induced pressure as mentioned above. Placing a heavy load over one cylinder provides pressure intensification from the other actuators. All other cylinders will transfer their excess energy through the oil lines to the loaded cylinder. The loaded cylinder would then be capable of pushing with four times its normal force. One-fourth of this force would be from its own air cylinder, while three times more force comes from energy transfer. This could overpressure the cylinders or valves, causing failure.

Figure 3-6: Opposing tandem cylinder air-over-oil circuit for synchronizing

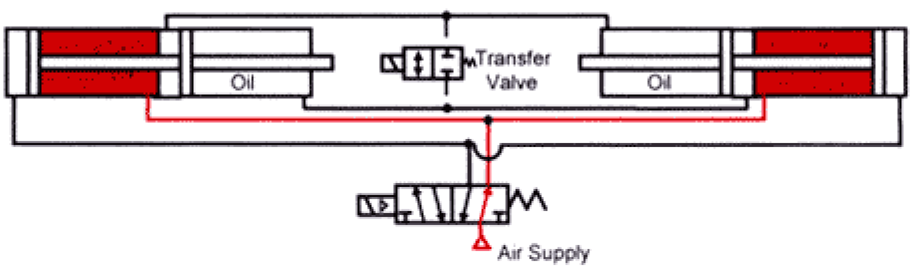


Figure 3-6 *shows opposing tandem cylinders. These cylinders will meet exactly in center. Drilling a center hole in odd sized parts is one possible use for this circuit.*

Counterbalance valves

Some actuators with running-away (or overrunning) loads will let the load free fall when the directional valve that controls the actuator shifts to lower the load. Cylinders with large platens and tooling or hydraulic motors on winch drives are two examples of such actuators. When the directional valve shifts, an overrunning load forces the actuator to move faster than pump flow can fill it. Oil at high velocity leaves one end while the opposite side starves for fluid. A vacuum void forms in the inlet side of the actuator that must be filled before applying force. Any running-away or overrunning load needs some method to retard its movement.

A meter-out flow control is one way to control a running-away load at a constant speed. Unless pump flow never changes, setting flow precisely on this type control is critical. Setting the flow control for minimum pump flow will waste energy when pump flow is high. Setting the flow control for maximum pump flow lets the cylinder run ahead when pump flow is low. Incorporating a pressure control valve called a counterbalance is a better way to control running-away loads. A counterbalance keeps an actuator from running away even with variable flow rates.

Fig. 5-1: Internally piloted counterbalance valve

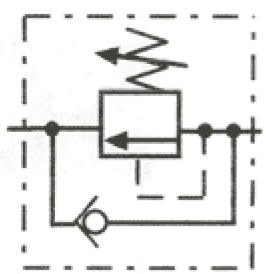


***Figure 5-1** shows the symbol for an internally piloted counterbalance valve. Use an internally piloted counterbalance to hold a load back when the actuator does not need full power at the end of stroke. This type of counterbalance valve retards flow continuously, so it resists flow even after work contact stops the actuator. Note that it is necessary to adjust an internally piloted counterbalance valve every time the load changes. The following circuits show these characteristics and how to design around them.*

Fig. 5-2: Externally piloted counterbalance valve

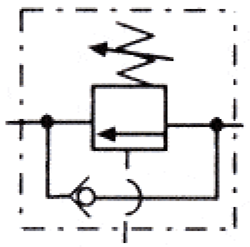


Figure 5-2 *shows the symbol for an externally piloted counterbalance valve. This valve's pilot supply is from a source other than the controlled load. An externally piloted counterbalance does not waste energy at the end of stroke and does not require adjustment for changing loads. However, an externally piloted counterbalance valve does waste a little more energy when moving the load to the work.*

Fig. 5-3: Internally/externally piloted counterbalance valve

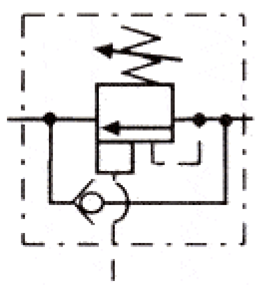


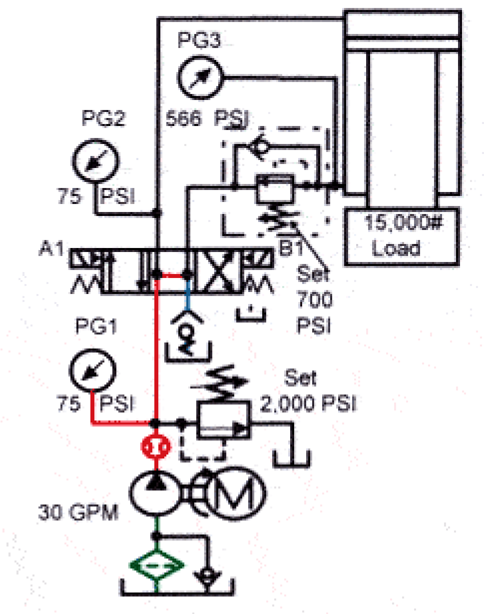
Figure 5-3 shows the symbol for an internally/externally piloted counterbalance valve. This valve has the best of both systems. As the load extends, internal pilot supply gives smooth control with little energy loss. After work contact, as system pressure builds, the external pilot fully opens the counterbalance to relieve all backpressure in the cylinder.

Counterbalance valves are manufactured in both spool and poppet designs. Spool designs leak at a rate of 3 to 5 in.3/min, thus allowing actuator creep. Poppet designs leak at only 3 to 5 drops/min, minimizing cylinder creep. (Because hydraulic motors bypass internally, a counterbalance only works with a moving load. The designer should apply a braking method to hold a hydraulic motor at rest.)

Internally piloted counterbalance valve

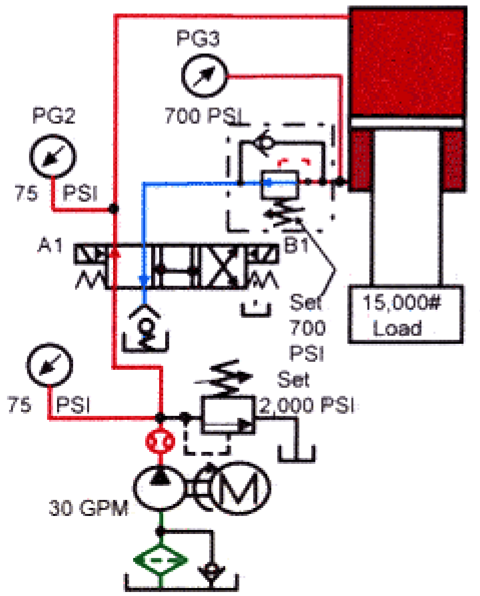
Figure 5-4 pictures a circuit with a running-away load. This circuit demonstrates the operation of an internally piloted counterbalance valve. The cylinder in Figure 5-4 has a static pressure of 566 psi in the rod end due to the 15,000-lb load on the 26.51- in.2 area. (15,000 / 26.51 = 566 psi). An open-center directional valve unloads the pump and keeps backpressure off the counterbalance valve outlet and pilot port. The cylinder holds in any position if the *counterbalance valve is set correctly and does not leak. Set the counterbalance approximately 100 to 150 psi higher than the load-induced pressure.*

Fig. 5-4: Internally piloted counterbalance valve at rest with pump running



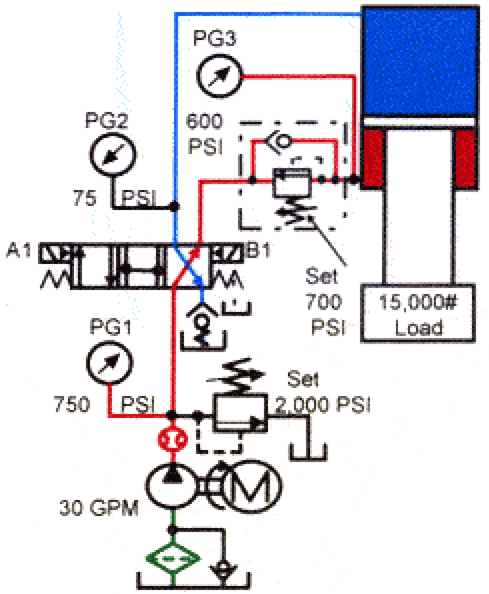
Normal procedure for setting a counterbalance valve is to turn the adjusting screw to its highest pressure before raising the cylinder. After starting the pump, energize the directional valve and carefully raise the load a short distance. With the load suspended, deenergize the directional valve. A working counterbalance will hold the load suspended and gauge PG3 will show the load-induced pressure. Now start lowering the counterbalance pressure setting slowly. When the cylinder begins to creep downward, increase the pressure until creeping stops. Then continue turning the adjusting control in the same direction another to turn. After setting the counterbalance this way, power the cylinder down and notice the pressure reading on gauge PG3. Pressure should be approximately 700 to 750 psi. Any time the cylinder loading changes, repeat the above process. Resetting the counterbalance valve keeps the cylinder from running away and reduces energy loss with a lighter load.

Fig. 5-5: Internally piloted counterbalance valve with cylinder extending



*When the directional valve shifts to extend the cylinder in **Figure 5-5**, oil from the pump flows into the cap end of the cylinder and pressure starts to build. When pressure in the cap end of the cylinder reaches about 75 psi, the cylinder should start to stroke. (This is because it builds an extra 140 psi in the rod end, adding to the load's 566 psi.) At this point the cylinder starts to extend and continues to move as long as the pump supplies oil at 75 psi or higher to the cylinder cap end. If pump flow changes, cylinder speed changes also.*

Fig. 5-6: Internally piloted counterbalance valve with cylinder retracting

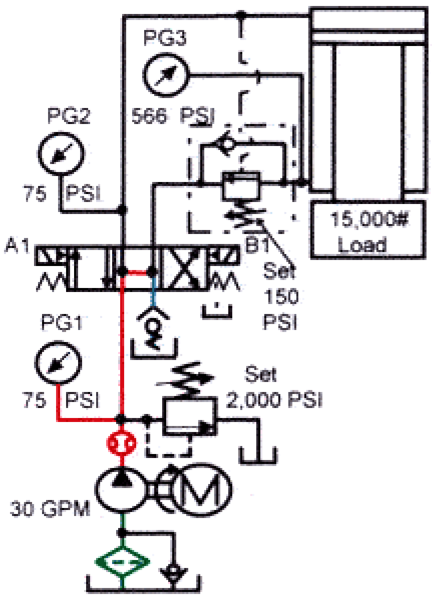


While the cylinder is retracting, ***Figure 5-6****, pump flow bypasses the counterbalance valve through the integral bypass check. The counterbalance valve offsets the potential energy of the weight on the rod end of a cylinder. The 15,000-lb force in this figure cannot do useful work when using an internally piloted counterbalance valve. (See* ***Figures 5-13 through 5-15*** *for energy loss and heat generation for different types of counterbalance circuits.)*

Externally piloted counterbalance valve

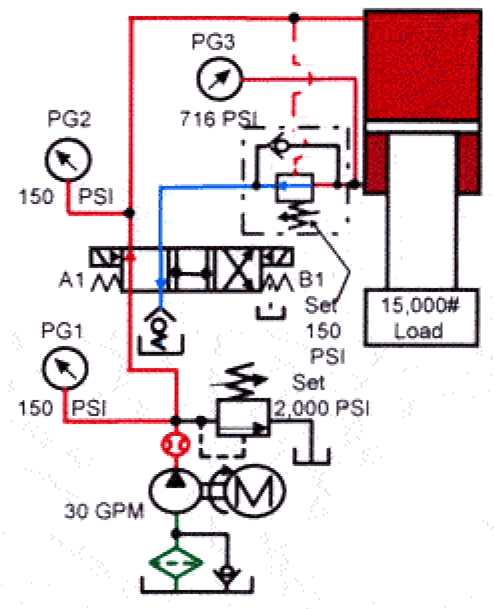
Figure 5-7 *shows an externally piloted counterbalance valve circuit. (This is the same cylinder arrangement shown previously.) Notice the counterbalance setting of 150 psi with this circuit. Because this is an externally piloted counterbalance, it operates at a much lower setting. If the load changes, the counterbalance setting does not change. Load-induced pressure does not affect set pressure. Theoretically, set pressure could be 1 psi and the load would not move. The cylinder would only extend when the counterbalance valve's external pilot port sensed 1 psi at the cylinder cap end.*

Fig. 5-7: Externally piloted counterbalance valve at rest with pump running



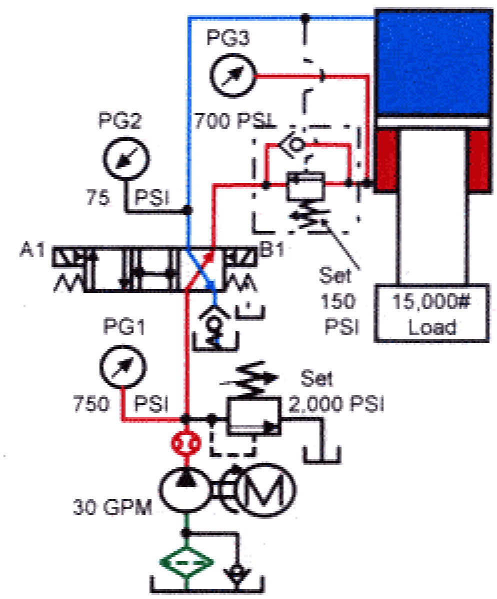
On most externally piloted counterbalance valves, the minimum pressure setting is 100 to 200 psi. This keeps the counterbalance valve from hunting. Hunting starts when the valve sees enough pressure to open, but then opens too wide. The cylinder runs away when the valve opens too much and pilot pressure drops. When pilot pressure drops, the counterbalance valve closes and the cylinder stops. After the cylinder stops, pilot pressure builds again. The process repeats and continues to the end of stroke. The higher the load-induced pressure, the greater the hunting problem. (On some systems it is possible to add an orifice in the pilot line to slow the pilot supply response and reduce hunting. This orifice fix is difficult to get right and may cause other circuit problems.)

Fig. 5-8: Externally piloted counterbalance valve with cylinder extending



*Energizing the directional valve to extend the cylinder in **Figure 5-8** sends pilot pressure to the counterbalance valve from the cap end cylinder line. Once pressure in the cap end cylinder line reaches 150 psi, the counterbalance opens and the cylinder extends. As long as there is enough pilot pressure to keep the counterbalance open, the cylinder moves forward. Increasing, decreasing, or stopping pump flow causes the cylinder to respond accordingly, but never to run away.*

Fig. 5-9: Externally piloted counterbalance valve with cylinder retracting



When the cylinder meets the load, pressure in the pilot port of the counterbalance continues to increase. When pilot pressure goes above the counterbalance setting, the valve opens fully and drops all backpressure on the cylinder rod end. With no backpressure on the rod end, the weight energy generates extra downward force. The externally piloted system saves energy by eliminating all backpressure when performing work. ***Figure 5-9*** *shows the flow paths after the directional control valve shifts to retract the cylinder.*

Internally/externally piloted counterbalance valve

Some manufacturers make counterbalance valves with internal and external pilots. These internally/externally piloted valves provide the best of both systems. They use the internal pilot to lower the load smoothly and the external pilot to drop all backpressure when performing work, thus avoiding loss of down force. In addition, internally/externally piloted counterbalance valves don't hunt.

Fig. 5-10: Internally/externally piloted counterbalance valve at rest with pump running

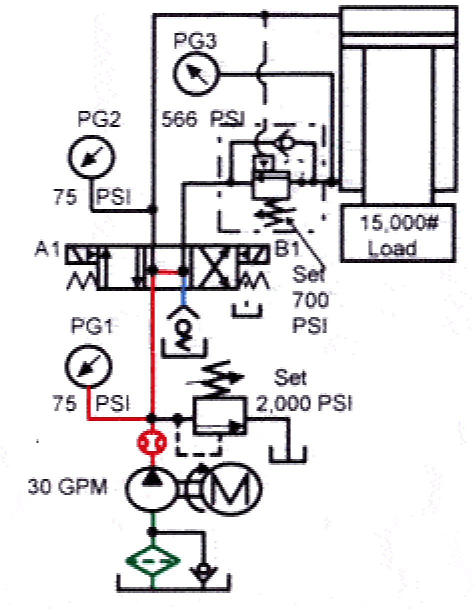
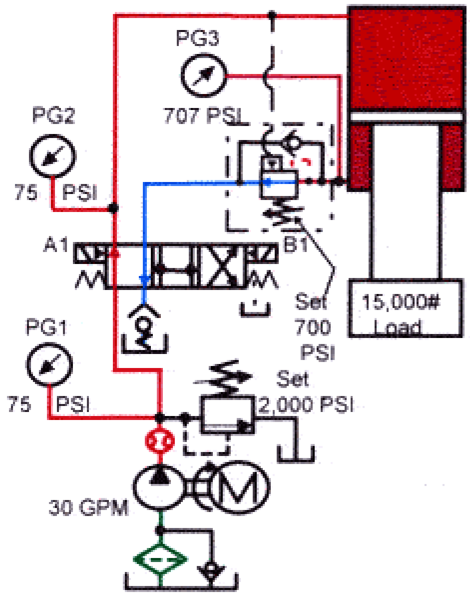


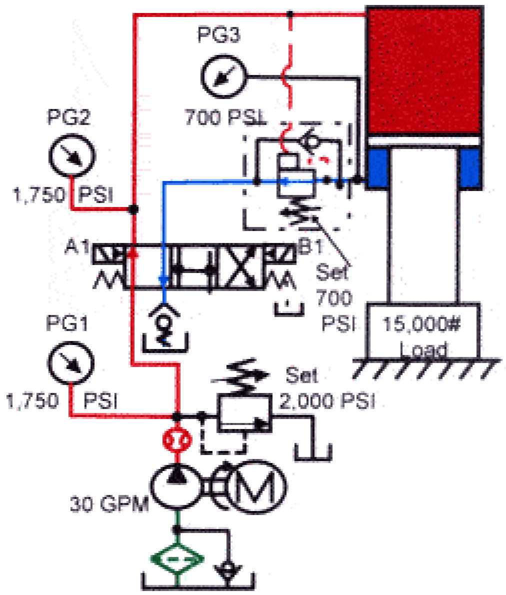
Figure 5-10 *shows a schematic drawing with an internally/externally piloted counterbalance valve. In the at rest condition, the external pilot drains to tank through the directional valve. The internal pilot has static pressure from the load-induced pressure on the rod end area. Setting the counterbalance pressure approximately 25% higher than static pressure (1.25 X 566 = 707 psi) means that when pressure at the cylinder cap end rises to approximately 75 psi, the cylinder starts to stroke.*

Fig. 5-11: Internally/externally piloted counterbalance valve with cylinder extending



*When the directional valve shifts, **Figure 5-11**, the cylinder begins to extend. Internal pilot pressure opens the counterbalance valve enough for the cylinder to move. Pilot pressure built by the pump pushing against the cylinder keeps the counterbalance valve open. The cylinder continues to extend smoothly at a controlled rate. If flow to the cylinder cap end changes or even stops, cylinder speed responds accordingly. When the cylinder meets resistance, the external pilot takes over, **Figure 5-12**, and opens the counterbalance fully at approximately 250 psi. With the counterbalance valve open to tank, backpressure against the cylinder rod end drops, allowing full thrust.*

Fig. 5-12: Internally/externally piloted counterbalance valve with cylinder pressing against work



*To see how cylinder thrust changes with different counterbalance valve pilot options, look at **Figures 5-13, 5-14, and 5-15**. These circuits show each type of counterbalance piloting system in the working condition -- with pressures, forces, and effective force listed.*

Machine thrusts with different counterbalance valve pilot options

An internally piloted counterbalance valve controls cylinder extension smoothly, but reduces thrust during the working portion of the cycle.

Fig. 5-13: Internally piloted counterbalance valve with cylinder pressing

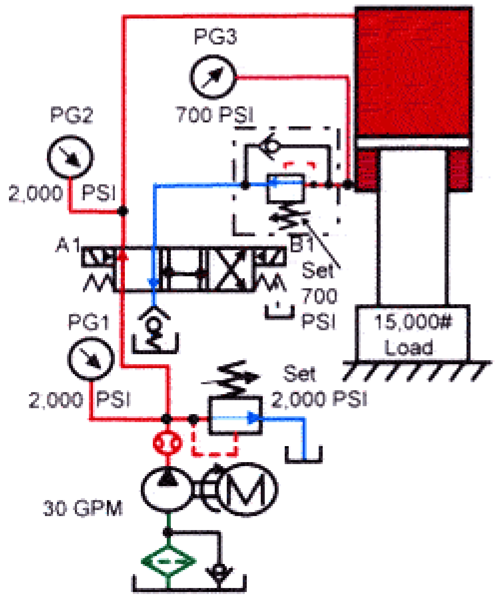
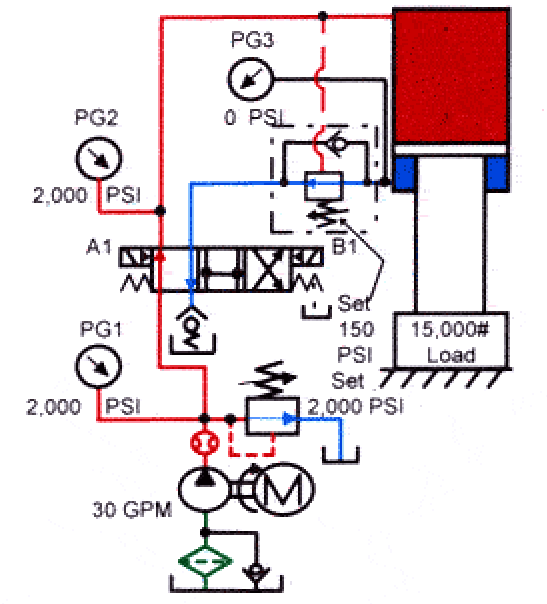


Figure 5-13 *shows the maximum force from different types of counterbalance circuits while acting on a part with the cylinder stalled. System pressure of 2000 psi acting on the cylinder cap end produces 100,540 lb of thrust. (50.27 in.2 X 2,000 psi = 100,540 lb.) The 15,000-lb weight on the rod end increases the resulting downward force to 115,540 lb. The 716 psi acting against the 26.51-in.2 rod end area produces an upward acting force of 18,981 lb. (716 psi X 26.51 in.2 = 18,981 lb.) The net effective downward acting force is 96,559 lb. If the upward acting force could be reduced or eliminated, the cylinder could do more useful work.*

The counterbalance valve more than cancels the weight of platen and tooling that gives an energy loss of approximately 16%. Approximately 7.5 tons of force from the rod end weight must be raised during every cycle but does not do any work as the cylinder extends.

Fig. 5-14: Externally piloted counterbalance
valve with cylinder pressing



Externally piloting the counterbalance valve, ***Figure 5-14****, requires about twice as much pressure to extend the cylinder. However, upon reaching the work, the loss of backpressure on the cylinder increases the cylinder force and more than makes up for the loss.*

The schematic shown in ***Figure 5-14*** *has the same downward force as* ***Figure 5-13*** *-- a total of 115,540 lb. The difference is that there is no upward force in* ***Figure 5-14****. The resultant downward force of 115,540 lb is an increase of 16% over the circuit with an internally piloted counterbalance valve. This saves most of the energy expended to raise the load on the return cycle.*

If at all possible, a counterbalance valve should be externally piloted. As explained previously, there are some instances where a cylinder might chatter as it extends if its circuit uses an externally piloted counterbalance valve. This chatter usually applies to circuits with high load-induced pressure or when the counterbalance valve is mounted at a distance from the cylinder port. The best practice is to mount the counterbalance valve directly on or very close to the cylinder port. Note that if a conductor between the cylinder port and valve breaks, the cylinder will free fall. That is why it is always good practice to use an external safety device to protect the operator and machine.

Fig. 5-15: Internally/externally piloted counterbalance valve with cylinder pressing

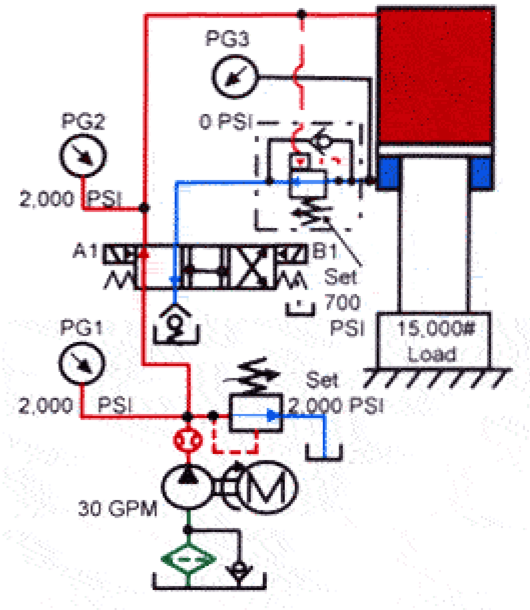
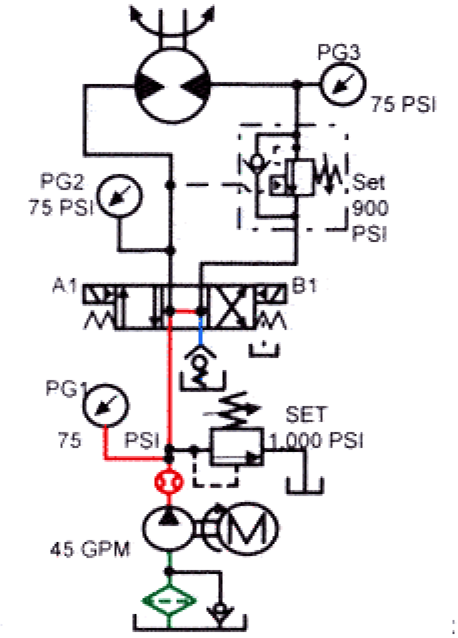


Figure 5-15 *shows a schematic diagram of the best counterbalance circuit. This circuit has a counterbalance valve with internal and external pilot supply. As the cylinder extends, the lower-pressure internal pilot gives a smooth descent at reduced pump pressure. The end result is the same as the externally piloted valve of* ***Figure 5-14****. When the cylinder contacts the work, all upward force is eliminated, minimizing energy loss.*

Hydraulic motor brake valve

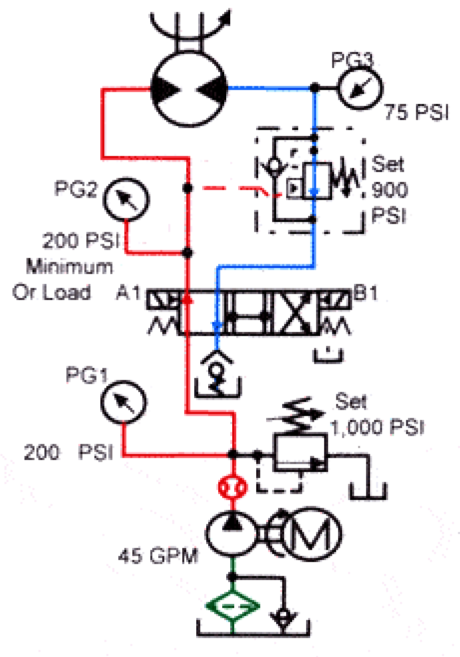
Excessive backpressure can damage a fast-turning hydraulic motor during an emergency stop situation. An open-center valve will eliminate backpressure, but the motor will continue to turn until it coasts to a stop. For a fast, non-shock stop, use a special counterbalance valve, called a brake valve. ***Figures 5-16, 5-17, and 5-18*** *illustrate a hydraulic motor circuit that uses a brake valve.*

Fig. 5-16: Internally/externally piloted brake valve at rest with pump running



The brake valve is an internally/externally piloted valve with different pilot areas. Some designs take one eighth of the pilot pressure at the external pilot port as that set at the internal pilot port. This means, for example, that setting the internal pilot at 900 psi requires only 113 psi at the external pilot to open the valve. The actuator could be a hydraulic motor or a fast moving horizontally mounted cylinder with an over running load. In either case, a brake valve eliminates damage from stopping the actuator abruptly.

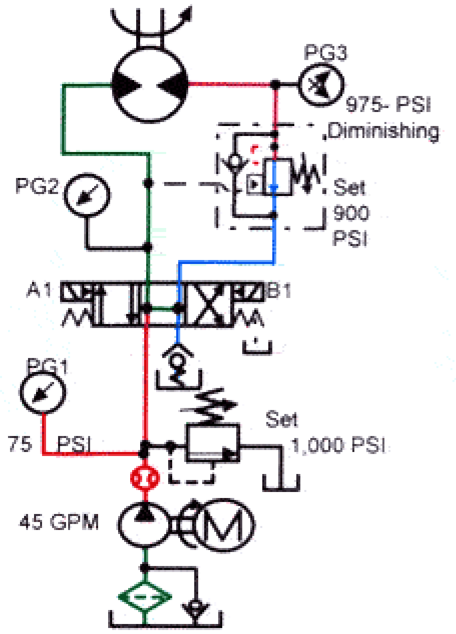
Fig. 5-17: Internally/externally piloted brake valve with hydraulic motor running



*With the hydraulic motor moving, **Figure 5-17**, external pilot supply opens the brake valve at low pressure. As long as pressure required to move the motor is greater than the external pilot pressure needed, there is little or no energy loss. A brake valve appears virtually nonexistent as the motor runs under load. If the hydraulic motor tries to run away, say on a loaded winch, a brake valve holds against the load until the motors down port sees at least 113 psi. The load will lower only as fast as fluid enters the motor down port. A brake valve counterbalances when necessary and allows almost free flow under load.*

***NOTICE:** Using a counterbalance or brake valve in a hydraulic motor circuit will not keep the motor from creeping when stopped. No matter how leak-free the counterbalance valve is, the internal bypass in the motor will let it slowly turn. Use an external braking system to hold any overrunning load driven by a hydraulic motor.*

Fig. 5-18: Internally/externally piloted brake valve with hydraulic motor stopping



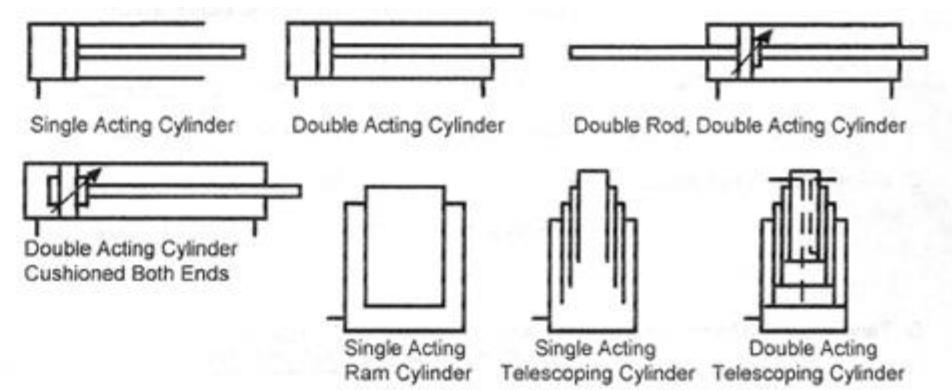
*When the directional valve shifts back to its center position, **Figure 5-18**, external pilot pressure drops and the brake valve begins to close. The hydraulic motor now acts like a pump, trying to force oil through the brake valve. As the brake valve starts closing, internal pilot pressure builds to 900 psi, forcing fluid through the brake valve at a 900-psi pressure drop. This 900-psi backpressure decelerates the actuator smoothly and rapidly. Setting the valve pilot pressure higher makes the stop faster and more abrupt. A lower pilot pressure setting makes the stopping time longer but smoother. In any case, stopping action is smoother and quicker than it would be without the brake valve.*

The difference between this circuit and a setup using a cross-port relief valve is that setting the brake valve at a pressure lower than system pressure does not affect normal actuator operation. Also, it eliminates the danger of cavitating an externally drained hydraulic motor. Note how the path around the brake valve with a bypass check valve allows reverse free flow for opposite rotation. To stop the motor quickly in the opposite direction of rotation, install another brake valve in the opposite motor line.

Fluid power cylinders

Approximately 85% of fluid power circuits incorporate some form of cylinder (or linear actuator). The cylinder converts pneumatic or hydraulic pressure into thrust to perform useful work. Both air and hydraulic cylinders come in ram, telescoping, single-acting/spring-return, double-acting, double rod end, rodless, and tandem types. Figure 6-1 shows the symbols for several of these types. Each machine has specific requirements that challenge the designer to determine which type of actuator to use.

Figure 6-1. Some representative cylinder symbols . . . using the “complete symbols."



Throughout this manual, many circuits show cylinders in a variety of applications. An explanation accompanies each example – noting the pumps, valves, and peripheral hardware used to do the work. Every design description also attempts to cover the limitations of a particular circuit and show other ways to perform the same task. This section covers several types of cylinder applications that do not fall under a particular heading.

Normally air cylinder circuits are less expensive than hydraulic circuits because there is no need for a power unit. An air compressor usually is part of the plant facility and compressed air is a commodity similar to electrical power. However, the cost of operating an air-powered machine may be four to seven times more than a hydraulically operated one.

Another disadvantage of air is the fluid’s compressibility. Hydraulic circuits are very rigid, while air circuits are quite spongy. This lack of control makes it almost impossible to accurately stop and hold an air cylinder in mid-stroke with standard air valves alone. After an air cylinder stops, it may start creeping or be forced out of position almost immediately.

When it comes to brute force, air cylinders fall far behind hydraulic cylinders because they normally operate only at 80 to 100 psi. Getting high force from low pressure requires large areas . . . with attendant large valves and piping. A general rule might be to look at hydraulics when an operation requires a 5-in. bore or larger air cylinder to develop the required force. However, another factor is how often the cylinder must cycle. Air circuits with very low cycle rates and long holding times could be more economical than hydraulics, but the faster the cycle *time, the more it costs to operate an air cylinder. Another consideration is the operating environment. Around food or medicine, potential contamination from hydraulic oil could be a serious problem. Look at each application to see which fluid system best suits it.*

Sizing hydraulic cylinders

Chart 6-1 provides an exercise in sizing a simple, single-cylinder hydraulic circuit with straightforward parameters. The example covers the basic requirements for sizing a hydraulic cylinder to power a specific machine.

TOO SIZE A HYDRAULIC CYLINDER
THE FOLLOWING INFORMATION MUST BE KNOWN

1. Maximum force required——— Usually in pounds or tons of force
2. Total stroke required——— Usually in inches
3. High pressure stroke required——— Usually in inches
4. Total cylinder cycle time——— Usually in seconds
5. Maximum pressure allowed——— Arbitrarily decided by the engineer

SAMPLE PROBLEM

1. Maximum force required——— 50,000 pounds
2. Total stroke required——— 42 inches
3. High pressure stroke required——— Total cylinder stroke
4. Total cylinder cycle time——— 10 seconds
5. Maximum pressure allowed——— 2,000 PSI

A. Minimum cylinder bore $\sqrt{\dfrac{\text{Maximum force required}/\text{Maximum PSI allowed}}{.7854}}$

$$\sqrt{\frac{50{,}000\text{ pounds}/2{,}000\text{ PSI}}{.7854}} = 5.641\text{" diameter or a } \underline{6\text{" Bore}}$$

B. Pump capacity in GPM = $\dfrac{\text{Piston area (Sq. In.) X Stroke (Inches) X 60 Seconds}}{\text{Cycle time (Seconds) X 231 (Cubic inches/gallon)}}$

$$\text{GPM} = \frac{28.275 \text{ X } 84\text{" X } 60}{10 \text{ X } 231} = 61.7 \text{ GPM or a } \underline{65\text{ GPM pump}}$$

*84" is for cylinder extend and retract. Rod displacement is disregarded in this example.

C. Electric motor horse power = HP = GPM X PSI X .000583

HP = 65 X 2,000 X .000583 = 75.79 or a <u>75 HP motor</u>

D. Tank size = 2-3 times pump GPM = 2 X 65 = 130 gallons = 150 gallon tank
3 X 65 = 195 gallons = <u>200 gallon tank</u>

CYL2

Of course, in the real world of circuit design, experience, knowing the process, the environment, the skill of the user, how long will the machine be in service, and other items will affect cylinder and power unit choices.

Before designing any cylinder circuit it is necessary to know several things. The first is the required force. Usually, the force to do the work is figured with a formula. In instances where there is no known mathematical way to calculate force, use a mock-up part on a shop press or on *a prototype machine to estimate the force requirements. If all else fails, an educated guess may suffice. (The sample problem in the chart requires a force of 50,000 Llb.)*

The second requirement is the total cylinder stroke. Stroke length is part of machine function, but it is needed to figure pump size. Use a stroke of 42 in. in this problem.

Third, how much of the stroke requires full tonnage? If only a small portion of the stroke needs full force, a hi-lo pump circuit and/or a regeneration circuit could reduce first cost and operating cost. This cylinder requires full tonnage for the complete 42-in. stroke.

Fourth, what is the total cylinder cycle time? Make sure the time used is only for cycling the cylinder. While load, unload, and dwell are part of the overall cycle time, they should not be included in the cylinder cycle time when figuring pump flow. Use a cylinder cycle time of 10 seconds for this problem.

Finally, choose maximum system pressure. This is often a matter of preference of the circuit designer. Standard hydraulic components operate at 3000 psi maximum, so choose a system pressure at or below this pressure. If the company that will operate the machine has operating and maximum pressure specifications, adhere to them. Remember that lower working pressures require larger pumps and valves at high flow to get the desired speed.

In the example in Chart 6-1, the square root of the maximum thrust, divided by the maximum system pressure, divided by 0.7854 gives a minimum cylinder bore of 5.641 in. Obviously, a standard 6-in. bore cylinder should suit this system.

To figure pump capacity, take the cylinder piston area in square inches, times the cylinder stroke in inches, times 60 seconds, divided by the cycle time in seconds, times 231 cubic inches per gallon. This indicates a minimum pump flow of 61.7 gpm. A 65-gpm pump is the closest standard flow available. Never undersize the pump because this formula figures the cylinder is going at maximum speed the whole stroke. In the real world, the cylinder must accelerate and decelerate for smooth operation, so travel speed after acceleration and before deceleration should actually be higher than this formula indicates.

Figure horsepower by multiplying flow in gpm by pressure in psi by a constant of 0.000583. This comes out to 75.79 hp . . . and is close to a standard 75-hp motor. This should provide sufficient horsepower because the system pressure does not have to go to 2000 psi with the cylinder size used.

The tank size should be at least two to three times pump flow. For the example, 3 X 63 equals 195 gallons. A 200-gal tank should be satisfactory. When using single-acting cylinders or unusually large piston rods, size the tank for enough oil to satisfy cylinder volume without starving the pump.

Sizing pneumatic cylinders

The procedure for sizing air cylinders is very similar to that for sizing hydraulic cylinders. One major difference: most plant air systems operate around 100 to 120 psi with approximately 80 psi readily available at the machine site. This gives little or no leeway for selecting operating pressure.

Also, because a compressor is part of the plant facilities, the number of cubic feet per minute (cfm) of air available for a single air circuit usually is many times that required. It is good practice though, to check for adequate flow capacity at the machine location.

The other items needed to design an air circuit are maximum force required, cylinder stroke, and cycle time. With this information, sizing cylinders, valves, and piping is simple.

TOO SIZE A PNEUMATIC CYLINDER
THE FOLLOWING INFORMATION MUST BE KNOWN

1. Maximum force required——— Usually in pounds or tons of force
2. Total stroke required———Usually in inches
3. Total cylinder cycle time ———Usually in seconds
4. System operating pressure———Arbitrarily 80 PSI

SAMPLE PROBLEM

1. Maximum force required——— 150 pounds
2. Total stroke required———14 inches
3. Total cylinder cycle time ——— 4 seconds
4. System operating pressure———80 PSI

A. Minimum cylinder bore $\sqrt{\dfrac{\text{Maximum force required X 1.25 or 2} / \text{Maximum PSI allowed}}{.7854}}$

$$\sqrt{\frac{150 \text{ pounds X } 1.25 / 80 \text{ PSI}}{.7854}} = 1.72\text{" diameter or a } \underline{\quad} \text{ 2" Bore}$$

B. SCFM required = $\dfrac{\text{V (Volume in.}^3\text{) X Compression Ratio (PSI + 14.7 / 14.7)}}{\text{Time in seconds to fill cylinder X 28.8}}$

$$\text{SCFM} = \frac{2 \text{ X } 2 \text{ X } .7854 \text{ X } 28^* \text{ X } (80 + 14.7 / 14.7)}{4 \text{ X } 28.8} = \underline{4.92 \text{ SCFM}}$$

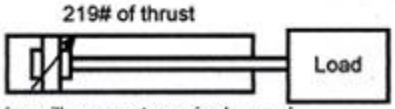
*28" is for cylinder extend and retract. Rod displacement is disregarded in this example

C. Minimum valve C $\dfrac{^{(\text{Flow SCFM})}Q}{_{(\text{Constant})}22.48} \sqrt{\dfrac{T_1^{(\text{Abs. Temp.})} \text{ X } G^{(\text{Spec. Gravity [1 for air]})}}{\Delta P^{(\text{Pressure drop})} \text{ X } (P_2^{(\text{Outlet pressure})} + P_A^{(\text{Atmospheric pressure})})}}$

Minimum valve C_v $\dfrac{4.92}{22.48} \sqrt{\dfrac{460 \text{ X } 1}{10 \text{ X } (70 + 14.7)}}$ Minimum valve C_v = $\underline{.161}$ = 1/8" ported valve

219# of thrust

2" Bore @ 80 PSI



Cylinder will move at nominal speed

CYL4

To calculate the cylinder bore required, use the formula given at A in Chart 6-2. Notice the 1.25 multiplier on the force line. For an air cylinder to move at a nominal rate, it needs approximately 25% greater thrust than the force required to just offset the load. When the cylinder must move rapidly, provide a force up to twice that required to simply balance the load.

The reason for this added force can be illustrated by the example of filling an empty tank from a tank at 100 psi. When air first starts to transfer, the high pressure difference between the two tanks produces fast flow. As the pressures in the tanks get closer, the rate of transfer slows. The last 5 to 10 psi of transfer takes a long time. As the tank pressures get close to equal, there is less reason for transfer because the pressure difference is so low.

At a system pressure of 80 psi, if an air cylinder needs 78 psi to balance the load, there is only a 2-psi differential to move fluid into the cylinder. If the cylinder moves at all, the motion will be very slow and intermittent. If pressure differential increases – either from higher inlet pressure or lower load – the cylinder starts to move smoothly and steadily. The greater the differential, the faster the cylinder strokes. (Note that once cylinder force is twice the load balance, any increase in speed due to higher pressure is minimal.)

Substituting the 1.25 multiplier in the formula produces a cylinder bore of 1.72 in. minimum. Choose a 2-in. bore cylinder because it is the next standard size greater than 1.72 in.

To size the valve, use the flow coefficient (or Cv) rating formula. (The Cv factor is an expression of how many gallons of water pass through a certain valve . . . from inlet to outlet . . . at a certain pressure differential.) Valve manufacturers use many ways to report Cv and some may be confusing. Always look at the pressure drop allowed when investigating the Cv, to be able to compare different brands intelligently.

The formula indicates that a valve with 1/8-in. ports is big enough to cycle the 2-in. bore cylinder out 14 in. and back in 4 seconds.

There are many charts in data books as well as valve manufacturers' catalogs that take the drudgery out of sizing valves and pipes. There are several computer programs as well to help in proper sizing of components.

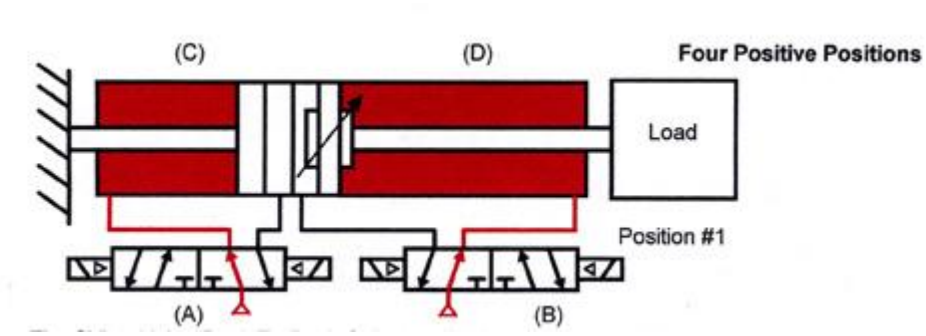
Cylinder circuits with four positive stopping positions

To stop a cylinder stroke accurately at different points in its travel, use a hydraulic servo system. Particularly for constantly changing intermediate stopping positions, a servo system works best. However, with only one constant mid-stroke stopping point, the circuit shown in Figure 6-2 will work well. A pair of cylinders with different strokes is attached at their cap ends. (This arrangement might be as simple as two off-the-shelf cylinders with their cap end flanges bolted together. Many manufacturers furnish this cylinder arrangement as a unit, using long tie rods to make the mechanical connection.) Because the cylinders have different strokes, it is possible to stop the load accurately at four positions. For instance, if cylinder C has a 2-in. stroke and cylinder D has a 4-in. stroke, the positions are home, and two, four, and six inches from home. If both cylinders have the same stroke, the positions are home, half extended, and full extended.)

This positioning arrangement works the same with air or hydraulic circuits, and always requires two valves. Air cylinders might bounce at fast speeds, but would quickly settle at an exact *position. Note that the cylinders also move, so use flexible lines and provide some way to guide the cylinders.*

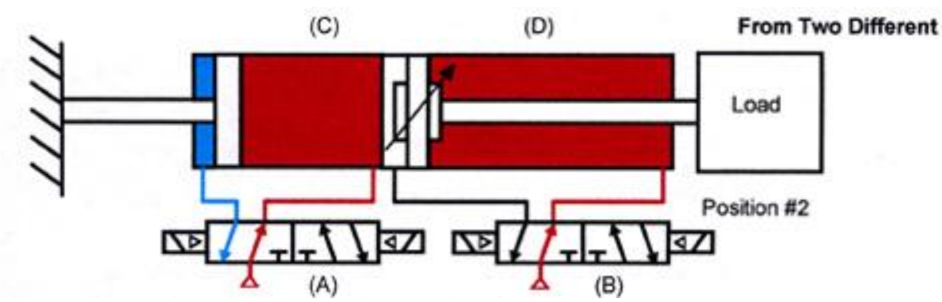
Figure 6-2 shows the circuit at rest. The valves could be double-solenoid (as shown), single-solenoid/spring-return, or spring-centered. The cylinders are both fully retracted, in Position #1.

Figure 6-2. Two cylinders mounted back-to-back for multiple positive stopping positions – at rest.



When valve A shifts, as in Figure 6-3, cylinder C strokes to Position #2. This position is always the same because the piston bottoms out against the cylinder's head end. Adjusting the rod attachment can make slight position variations. Machine wear could make such adjustments necessary.

Figure 6-3. Two cylinders mounted back-to-back for multiple positive stopping positions – position 1.



In Figure 6-4, valve A shifts to retract cylinder C while valve B shifts to extend cylinder D . This accurately places the load in Position #3. Finally, both cylinders extend as shown in Figure 6-5, moving the load to Position #4.

Figure 6-4. Two cylinders mounted back-to-back for multiple positive stopping positions – position 2.

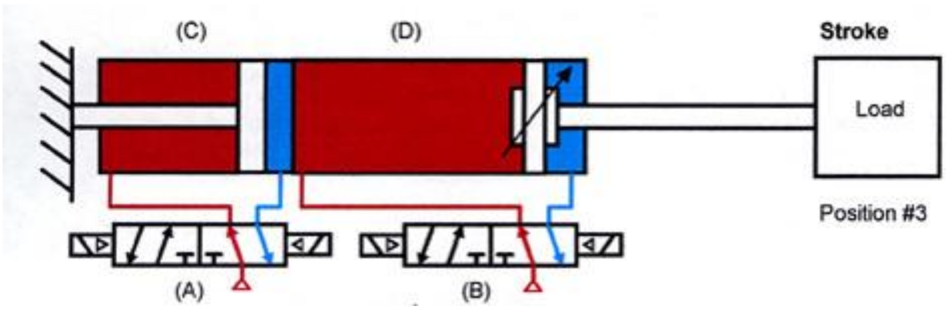
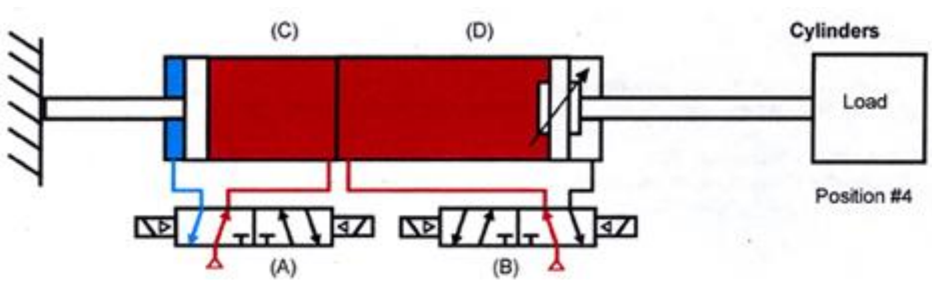


Figure 6-5. Two cylinders mounted back-to-back for multiple positive stopping positions – position 3.



After both cylinders extend fully, they can return to home or either of the mid-stroke positions as required. (The circuit designer might choose air logic or electrical controls, with palm buttons numbered one through four – to allow an operator to pick any cylinder position at any time.)

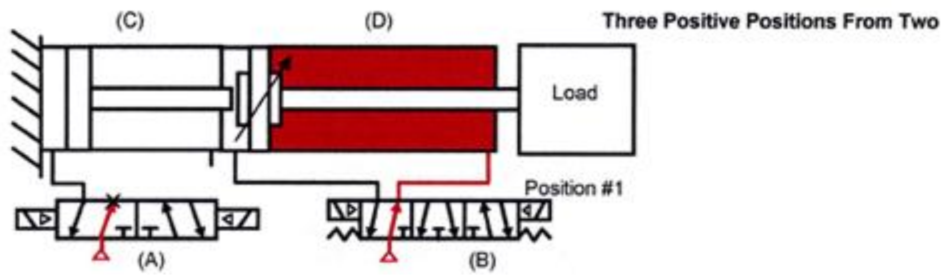
Using more than two cylinders can provide a greater number of stopping positions, but controlling more positions requires more circuitry. This still may be is less expensive than a servo system. Lower cost and easier maintenance may offset the greater versatility of a servo system in some applications.

Air or hydraulic tandem-cylinder circuits with three positive stopping positions

A tandem cylinder consists of two double-acting cylinders in one envelope. It has four fluid ports, and the piston rods may be attached or unattached, depending on the application. Most unattached-rod tandem cylinders have unequal strokes, while attached rod tandems have equal strokes. Some tandem cylinders have different bores, again depending on the need.

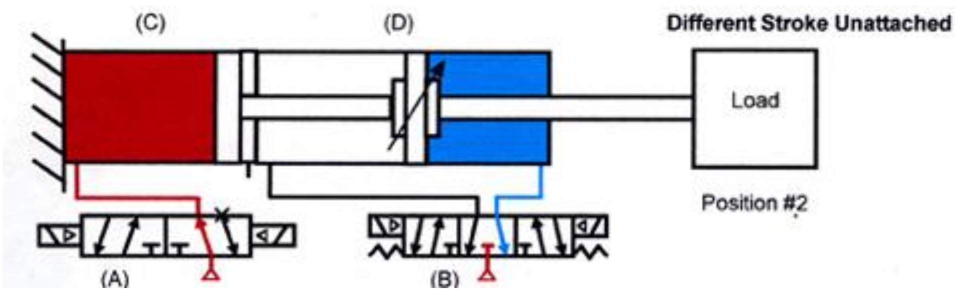
Figure 6-6 shows a rigidly mounted, unattached tandem cylinder in a multi-positioning circuit, with the cylinder and valving at rest. This circuit produces three positive positions. Note that the load must be resistive – or made that way with valving. Cylinder C has a 2-in. stroke and cylinder D has a 6-in. stroke. This combination gives a positive home position, plus two inches, and six inches extended. Valve A could be single-solenoid/spring-return or a double-solenoid detented (as pictured). Valve B must allow cylinder D to float – to avoid reducing the force of the stroke to Position #2.

Figure 6-6. Using unattached tandem cylinders for multiple positive stopping positions – at rest.



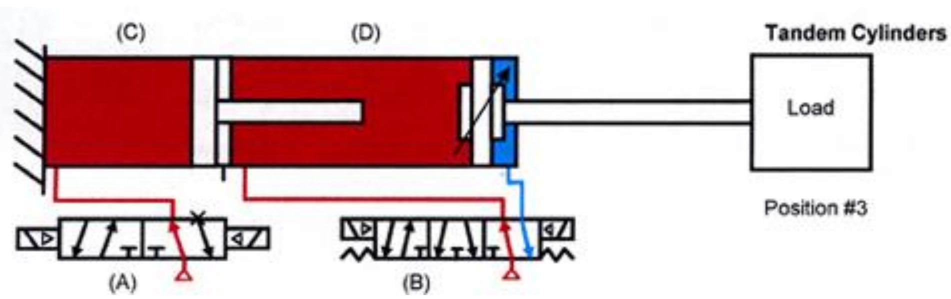
Shifting valve A, as shown in Figure 6-7, extends cylinder C through its full stroke, moving cylinder D and the load to Position #2. If travel speed is too fast and/or resistance is low, cylinder D may overshoot Position #2. If this occurs, add a flow control for air or a counterbalance valve for hydraulic service to offer resistance while cylinder C is stroking.

Figure 6-7. Using unattached tandem cylinders for multiple positive stopping positions – position 1.



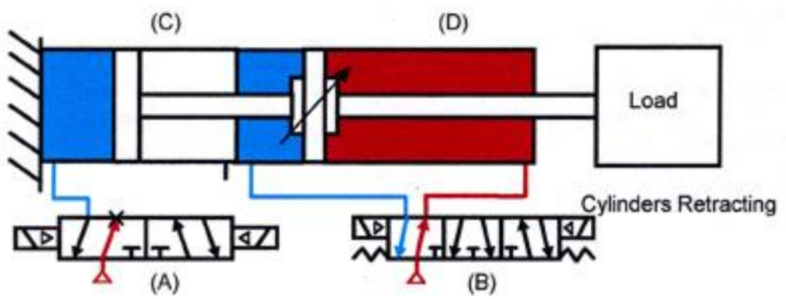
To extend the tandem cylinder fully, valve B shifts, as in Figure 6-8, porting fluid to the cap end of cylinder D . Cylinder D then extends fully to Position #3. Positions #2 and #3 are positive. They will be rigid in a hydraulic circuit and typically spongy in an air circuit.

Figure 6-8. Using unattached tandem cylinders for multiple positive stopping positions – position 2.



To retract the load, both valves return to home position, Figure 6-9. Cylinder D retracts fully and also pushes cylinder C home. Vent cylinder C's rod port to atmosphere if it is air operated, or drain the port to tank on a hydraulic cylinder.

Figure 6-9. Using unattached tandem cylinders for multiple positive stopping positions – cylinders retracting.



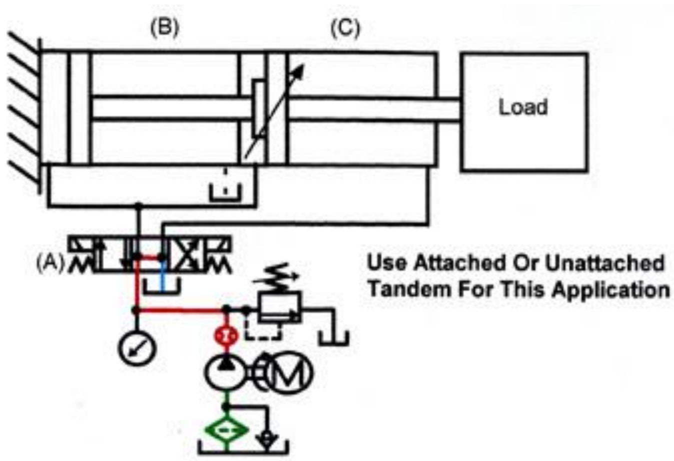
Assembling more than two cylinders this way creates more positive stopping positions when needed. Always make the first cylinder the one with the shortest stroke, with each added cylinder's stroke longer.

Using tandem cylinders to increase force

On occasion, a cylinder already in service is undersized for a new material or product, and there is no room in its location for a larger-diameter cylinder. One way to produce more force is to use a tandem cylinder with the same bore and mounting dimensions as the original cylinder. A tandem cylinder almost doubles the force of the single cylinder. The tandem cylinder mounts exactly as before, with the same rod diameter and thread. The only dimensional difference is that the tandem cylinder is more than twice as long. (Normally an attached tandem cylinder is best for doubling force, although not in all cases.)

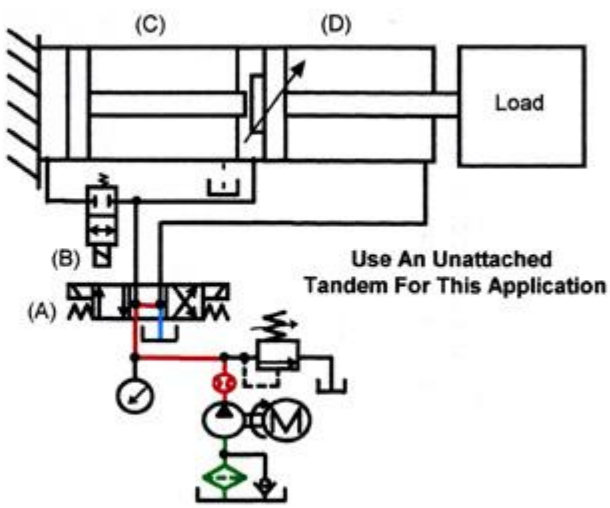
Figure 6-10 shows a tandem cylinder circuit that produces additional force on the extension stroke. Normally the retraction stroke needs minimal force, so vent or drain the rod side of the single-rod cylinder. Piped this way, fluid volume only increases on the extension stroke. With a 6-in. cylinder bore and a 2-in. rod diameter, the tandem cylinder's force is 90% more than the original single cylinder.

Figure 6-10. Using tandem cylinders in increase force.



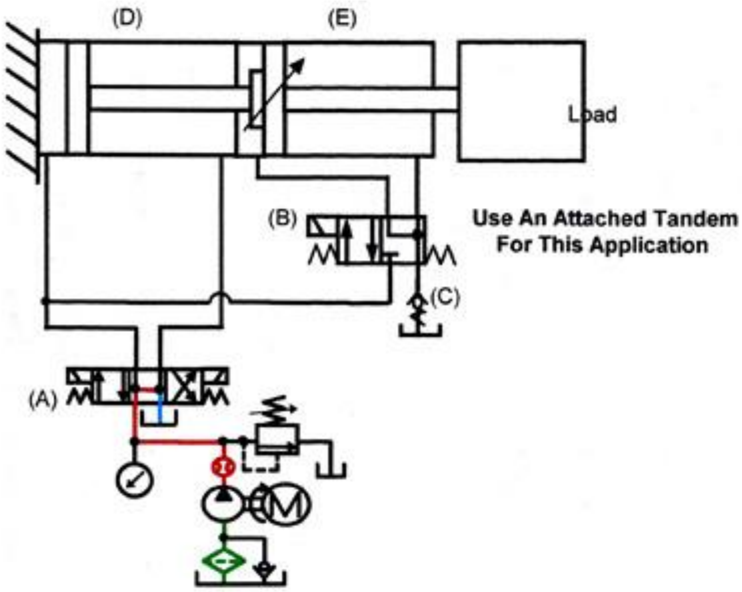
The circuit in Figure 6-11 uses an unattached tandem cylinder in a circuit that allows standard force or increased force as needed. For low force only, energize valve A. Oil volume and force are the same as in the original circuit. For almost double force, energize valves A and B.

Figure 6-11. Using tandem cylinders in increase force – dual force option #1.



The dual-force circuit in Figure 6-12 uses almost the same volume of oil as the single cylinder it replaces. Pipe directional valve A (supplied by the pump) to single rod-end cylinder D that is part of an attached tandem cylinder. When directional valve A shifts to extend the cylinder, oil flows to cylinder D. As cylinder D extends, it moves cylinder E. Cylinder E is fitted with a flow line from rod end to cap end through directional valve B. All the oil in cylinder E transfers to the opposite side of the piston, so the cylinder is full for the double-force portion of the stroke. Check valve C holds backpressure in the transfer circuit while cylinder E is moving and allows oil to flow to tank during the extra-force portion of the cycle. Extra force comes in when directional valve B shifts, sending oil to the push side of cylinder E's piston and allowing the opposite end to flow to tank.

Figure 6-12. Using tandem cylinders in increase force – dual force option #2.



When changing to a tandem cylinder for extra force, always check the rod diameter for column strength. All manufacturers show maximum force capabilities for a given rod diameter. When rod size increases, maximum force decreases due to less area on the double rod end cylinder. When using an oversize rod, purchase it with an undersize thread rod so it attaches directly to the machine member without modification.

Caution*: make sure the cylinder mounting can withstand the extra thrust. Most cylinder manufacturers' literature gives maximum force capabilities for a given mounting style. Because certain mounting styles have a lower pressure rating, a tandem cylinder may only accept slightly more than half the rated pressure. Change the mounting style if the reduced pressure generates too little force. Also, realize the extension speed of the double force portion of a tandem cylinder arrangement is approximately half the speed of a single cylinder.*

Circuit with unmatched tandem cylinders for high speed and force

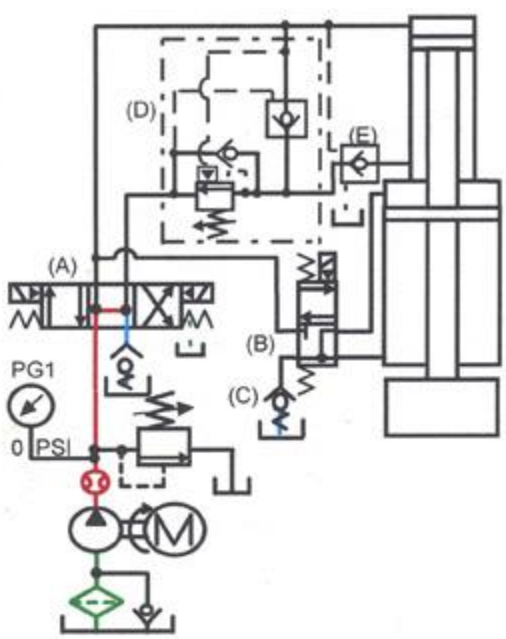
Many press applications require long strokes for loading parts with only a small portion of the stroke operating at high tonnage at the end. A 10-in. bore cylinder might be required for tonnage, while a 4-in. bore cylinder could provide all the force necessary to move to and from the work. Conventional circuitry often uses high volume at low pressure and high pressure at low volume for an application of this type. A regeneration circuit (Chapter 17 will cover regeneration circuits) could reduce the high-volume pump flow by half, but fast cycling still requires high flow.

Large cylinders with prefill valves and push back cylinders are one way to overcome the requirement for large fluid volumes. (Chapter 7 will explain decompression and prefill valves.)

Due to their high cost, prefill valves normally are found only in circuits with 20-in. or larger bore cylinders.

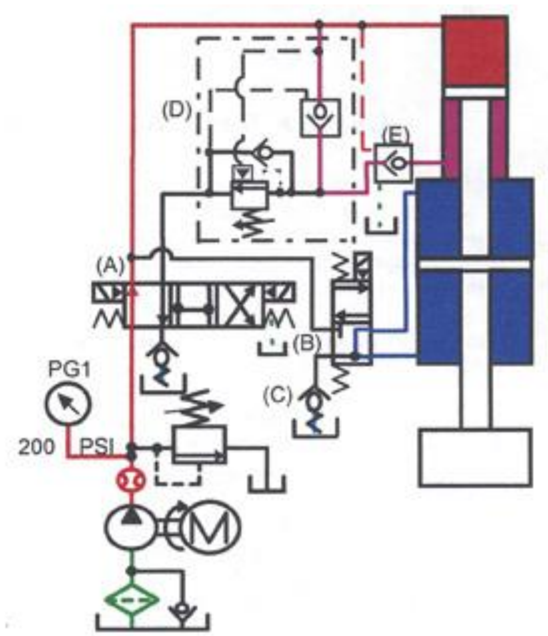
The circuit in Figure 6-13 illustrates another way to operate at high speed for extension and retraction at low force, with high tonnage available at any point along the extension stroke. The unmatched tandem cylinder has attached piston rods so the small-bore cylinder can retract both the large-bore cylinder and the load. The small-bore cylinder needs only a small volume of fluid to extend and retract at high speed, while both cylinders can produce high tonnage.

Figure 6-13. Using an unmatched tandem cylinder for high speed and high force – at rest with pump running.



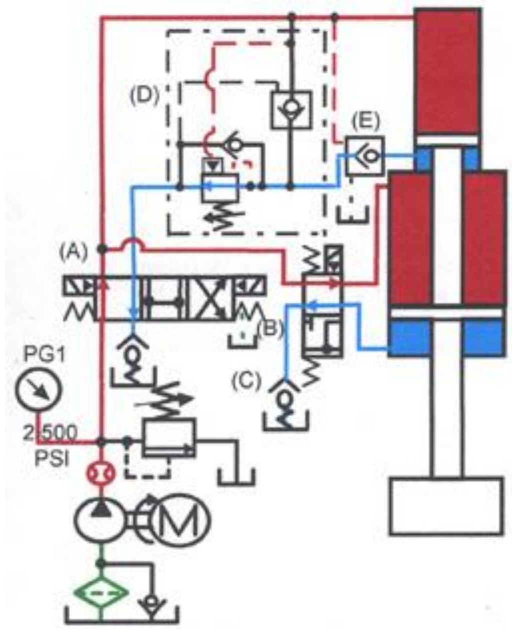
Energizing the extend solenoid on valve A in Figure 6-14 causes the small-bore cylinder to extend rapidly, in regeneration. This moves the large-bore cylinder and platen downward. As the platen lowers, oil in the large-bore cylinder transfers through valve B to the large-bore cylinder's opposite end.

Figure 6-14. Using an unmatched tandem cylinder for high speed and high force – fast-forward mode.



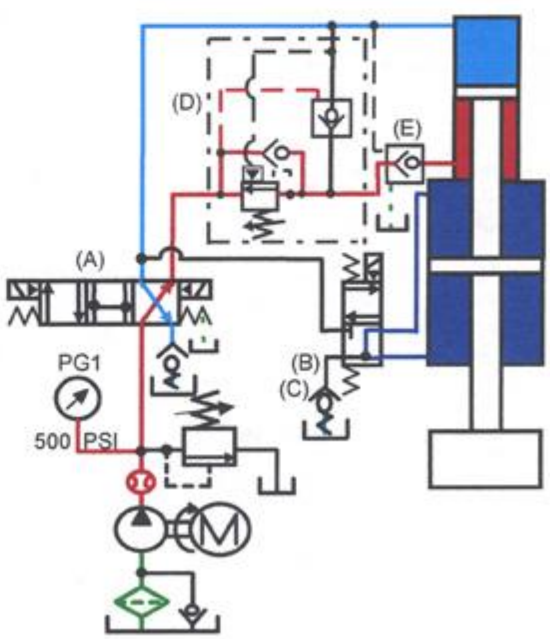
When the load meets resistance or contacts a limit switch, Figure 6-15, valve B's solenoid also energizes – sending pump flow to both cylinders. During this part of the cycle, speed slows and tonnage increases. (The large-bore cylinder transferred oil during the high-speed portion of the cycle to ready it for the high-force portion of the stroke.) During the high-force portion of the cycle, oil from the mounting end of the large cylinder returns to tank. Because the large bore cylinder exhausts during this part of the cycle, it receives fresh oil for every high-pressure stroke.

Figure 6-15. Using an unmatched tandem cylinder for high speed and high force – high-force mode.



To retract the cylinder at high speed, energize the retract solenoid on valve A, Figure 6-16. The pump retracts the small-bore cylinder, which also retracts the large-bore cylinder and platen. While the large-bore cylinder retracts, fluid in it again flows from end to end, so the cylinder stays full. Backpressure check valve C in the tank line keeps oil from draining to tank when it is lower than the cylinder.

Figure 6-16. Using an unmatched tandem cylinder for high speed and high force – fast-retraction mode.



Note externally drained pilot-operated check valve E at the rod end of the small-bore cylinder. With a running-away load, some means is needed to hold the cylinder in place while the circuit is at rest. This cylinder might free fall when the directional valve centers without some way to keep it from trying to regenerate. If the load is heavy, use an externally drained counterbalance valve to stop the pilot-operated check valve from chattering.

One potential problem with this arrangement is the length of the tandem cylinder. For long strokes, the more-than-double length of the tandem cylinder could cause height or length interference. Also, the rod size of the large-bore cylinder determines the smallest bore of the small cylinder. For example: if the double rod-end cylinder has a 10-in. bore with a 5-in. rod, then the smallest single-rod cylinder would require a 7-in. bore.

For the arrangement just shown and sized, the force at 3000 psi is approximately 292,000 lb. A pump flow of 30 gpm would result in a cylinder cycle time of about 15 seconds . . . with a 40-in. travel stroke and a 3/4-in. tonnage stroke.

Short closed height with double-length movement using two cylinders

Some machines need long strokes but lack space to mount long-stroke cylinders. Using telescoping cylinders is feasible for some applications, but high cycle rates usually eliminate them from consideration. Also, most telescoping cylinders are single-acting and depend on gravity or other outside forces to return them. Another drawback to telescoping cylinders is that the smallest-diameter ram must able to generate enough force to move the load. This means all other sections must be larger so they will need to be supplied with high flow for high speed.

Figures 6-17 and 6-18 show two air cylinders facing in opposite directions with their bodies attached side by side. This orientation makes the total stroke additive, while the retracted length is that of a single cylinder. (Assuming that both cylinders have a 20-in. stroke, the platen's starting position in Figure 6-17 is about 20 inches lower with this arrangement than it would be with a single 40-in. stroke cylinder.) Many applications use standard NFPA-design cylinders in such an arrangement. With this circuit there is constant force and speed, compact mounting, and double-acting operation. The only special requirement is to specify valves that give smooth action and control. If the circuit used two directional valves, the platen could have three positive positions (if required). With different stroke lengths, these cylinders could stop the platen positively in four positions.

Figure 6-17. Using two cylinders for double stroke from half the height – both cylinders retracted.

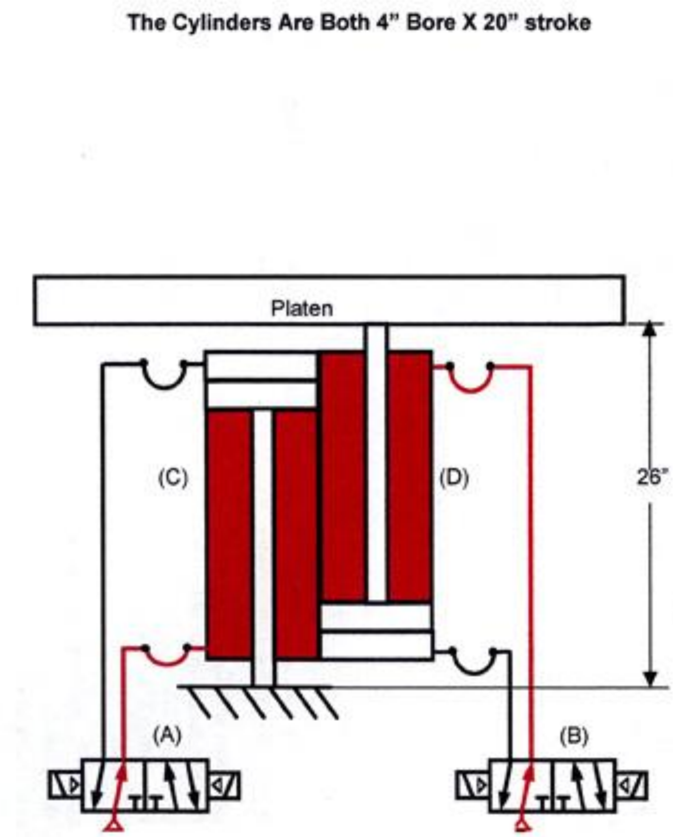
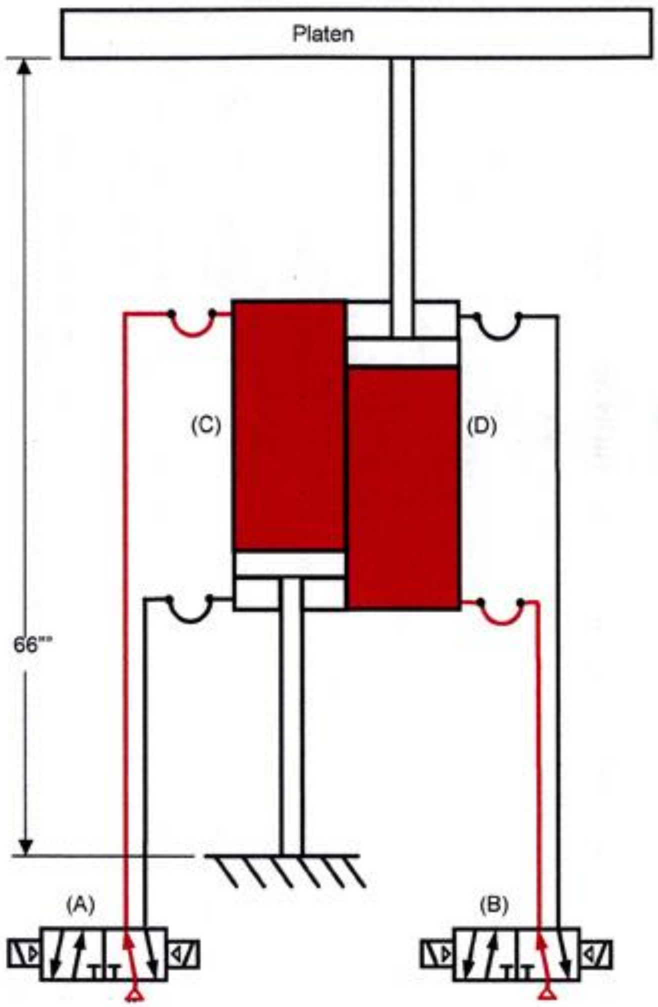


Figure 6-18. Using two cylinders for double stroke from half the height – both cylinders extended.



Using a single valve for extra stroke only requires meter-out flow controls at each cylinder port for near-simultaneous movement. This arrangement works smoothly and eliminates jerking when the cylinders bottom out at different times.

With hydraulic cylinders, use a spool-type flow divider for simultaneous movement and closely synchronized end of stroke stopping. (Chapter 11 will cover flow divider circuits.)

Because both cylinder bodies move, use flexible fluid lines. Also, arrange to guide the platen or machine member to keep excess side-loading off the cylinders.

Side-by-side cylinder mounting does not work as well in high-force applications because the higher side-load forces will wear out bushings and cause premature seal leakage. The side-by-side configuration works best in low-force pneumatic applications.

Why decompression is necessary in hydraulic systems

In high-pressure circuits with large-bore, long-stroke cylinders -- and the accompanying large pipes and/or hoses -- there is a good chance for system shock. In circuits with large components, when high-pressure oil rapidly discharges to tank, decompression shock results.

Decompression shock is one of the greatest causes of damage to piping, cylinders, and valves in hydraulically powered machines. The energy released during decompression breaks pipes, blows hoses, and can instantly displace cylinder seals. Damage from decompression shock may take time to show up because the energy released by a single shock may be small. After repeated shocks however, weaker parts in the circuit start to fail.

The potential for decompression shock is usually easy to determine beforehand and the design can be revised to avoid it. Shock from decompression normally occurs at the end of a pressing cycle when valves shift to stop pressing and retract the cylinder. The compressibility of the oil in the circuit, cylinder tube expansion, and the stretching of machine members -- all add to stored energy. The more energy stored, the worse the effects of decompression. Any time stored energy is a problem in a hydraulic system, a simple decompression circuit will add reliability and extend the system's service life.

One type of decompression shock that is hard to overcome occurs when a cylinder builds tonnage, then breaks through the work. Because pressure is resistance to flow, once the resistance is removed, the oil expands and decompresses rapidly. Such is the case when punching holes in a part. Punching applications pose one of the worse shock conditions any hydraulic circuit meets. To help reduce this type shock, keep piping as short as possible and anchor it rigidly. Some manufacturers offer resisting cylinders that slow the working cylinder's movement at breakthrough. These special cylinders may reduce or eliminate decompression shock.

Another type of shock occurs when oil flowing at high velocity comes to a sudden stop. This might happen when a cylinder bottoms out or when a directional valve shifts to a blocked condition. Whatever the cause, the effect is the same as trying to stop a solid mass moving at high speed. Use an accumulator or deceleration valve to control shock caused by a sudden flow stop. (See Chapter 1 on accumulators.)

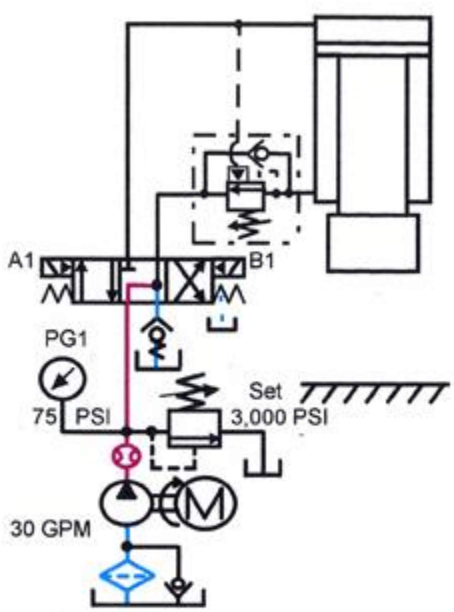
The ensuing text describes applications where decompression shock might cause a problem. Also shown is the operation of some typical decompression circuits.

When using a decompression circuit, cycle time becomes longer. Instead of the cylinder immediately retracting after finishing its working stroke, there is a short delay while stored energy dissipates. (It may be possible to arrange to decrease cylinder traverse time to make up for decompression time.) In any case, the added cycle time, if necessary, will decrease down time and maintenance problems.

Press circuit without decompression

Figure 7-1 shows a schematic diagram for a typical medium- to large-bore cylinder without provision for decompression. A 50-in.-bore cylinder always needs a decompression circuit -- while cylinders with bores under 10 in. may get by without one. The main criteria are the volume and pressure of the stored fluid. The more high-pressure oil in a circuit, the greater the decompression shock. Long lengths of hose also cause and/or amplify decompression shock. It is best to install a decompression circuit when there is any chance it may be necessary. The expense of a decompression circuit is minimal and only adds to the cycle time if used.

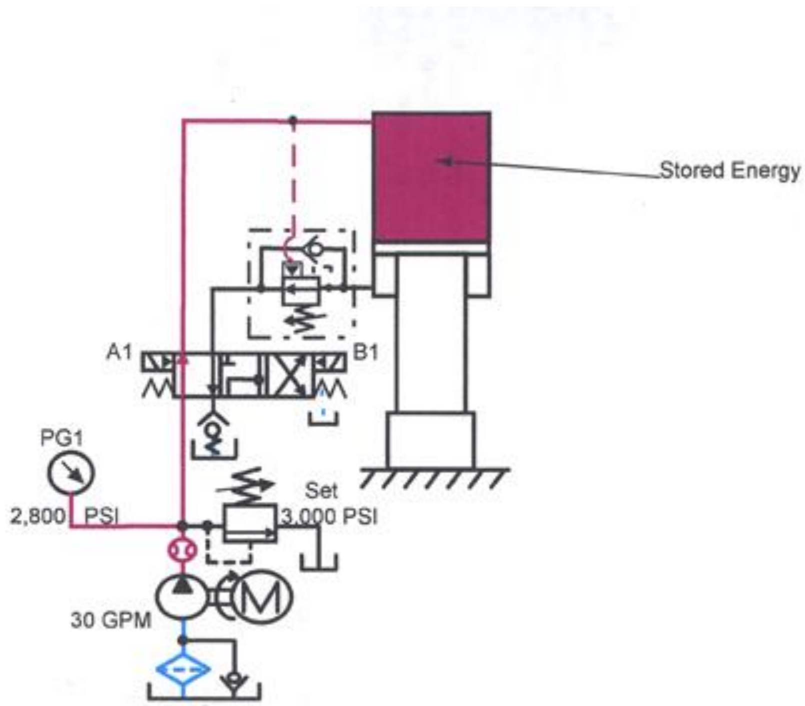
Fig. 7-1. Press circuit without decompression protection – at rest with the pump running.



The circuit in Figure 7-1 has a directional valve with an all-ports-open center condition. The pump unloads to tank when the valve shifts to this center condition. The cylinder stays retracted because there is a counterbalance valve on the rod port.

In Figure 7-2 the cylinder is pressing at a working pressure of 2800 psi. The 10-in. bore by 40-in. stroke cylinder holds approximately 3141 in.3 of oil. Added to this is another 800 in.3 of oil is in the pipe between the valve and the cylinder's cap end. At a compressibility of approximately 1/2% per thousand psi, and allowing another 1/2% per thousand psi for physical expansion of the cylinder and pipe, plus frame stretch, total volume expansion could be up to 1% per thousand psi. Multiplying (0.01) X (2800 psi) X (3941 in.3) indicates that there are approximately 110 in.3 of extra oil in the cylinder when pressing at 2800 psi.

Fig. 7-2. Press circuit without decompression protection – while extended cylinder is at full *tonnage.*



When the directional valve shifts to retract the cylinder, a large portion of the 110 in.3 of extra oil rapidly flows to tank. Every corner this fast moving fluid turns and every restriction it meets causes system shock. The shock only lasts a few milliseconds during each cycle but the damage accumulates. In a small system like this one, the shock may not be audible or give a noticeable jerk to the pipes. However each shock adds to the last one, and the damage eventually shows up in leaking fittings or broken machine members.

Press circuit with decompression

The circuit depicted in Figure 7-4 is the same as in Figures 7-1, 7-2, and 7-3, but a decompression circuit has been added. Also, the directional valve's center condition has ports P, B, and T interconnected, while port A is blocked. A pressure switch and a single-solenoid directional valve (the decompression valve) are added to the basic circuit to make decompression automatic and adjustable. The cylinder is at full tonnage in Figure 7-4, ready for decompression before beginning to retract.

Fig. 7-3. Press circuit without decompression protection – cylinder just starting to retract.

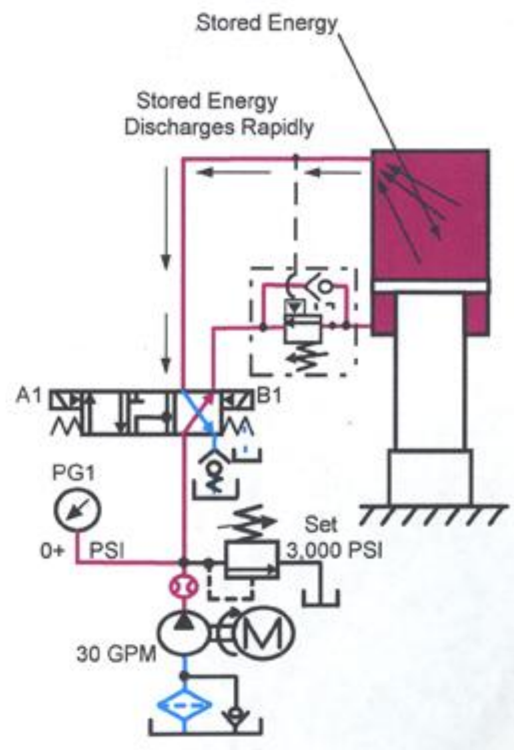
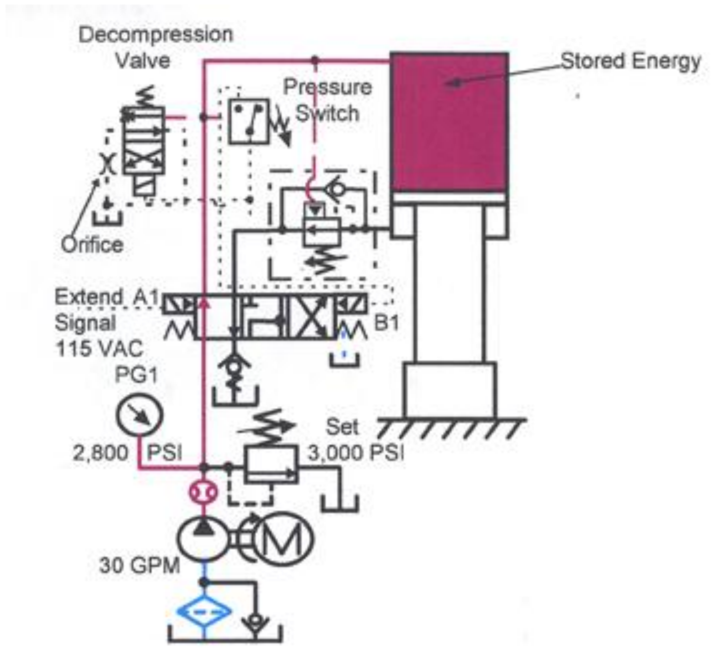


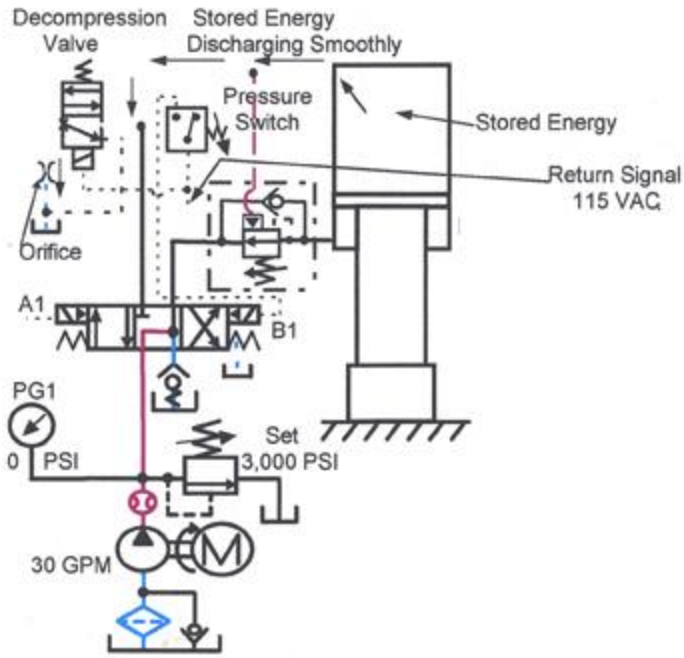
Fig. 7-4. Press circuit with decompression protection – while extended cylinder is at full tonnage.



In this circuit, the signal to the retract solenoid on the directional valve passes through the normally closed contacts on the pressure switch. With a pressure switch setting of 350 psi, the retract solenoid will not be energized until pressure in the cap end of the cylinder lowers to that level and the contacts close. Set the shift pressure of the pressure switch high enough to shorten the decompression time as much as possible, yet still low enough to eliminate decompression shock.

In Figure 7-5, the extend solenoid on the directional valve has just been deenergized, and a 115-VAC signal to retract the cylinder is on, but is blocked at the pressure switch's open contacts. The 115-VAC signal does go to the decompression valve's solenoid and that valve shifts, opening a path to tank for any stored energy. Until pressure in the cap end of the cylinder deteriorates to the pressure switch setting, the cylinder sits still. The main flow of trapped oil in the cylinder is stopped at the directional valve's blocked A port. This part of the cycle completely eliminates all shock damage -- although it does add to cycle time.

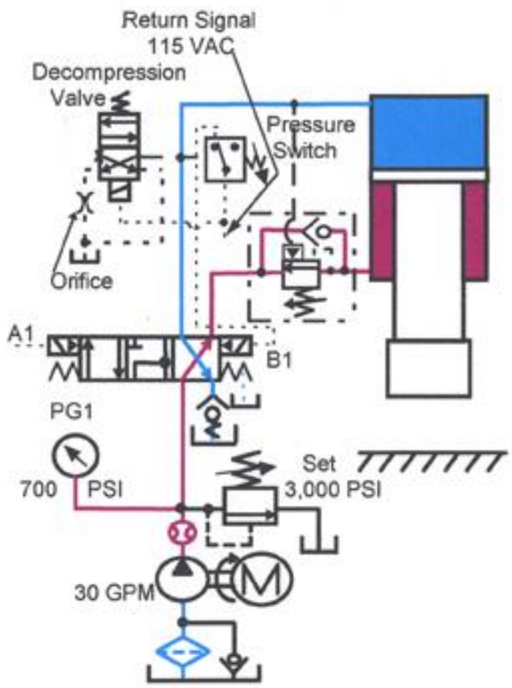
Fig. 7-5. Press circuit with decompression protection – while cylinder is decompressing.



Note the orifice in the line going to tank from the decompression directional valve. A fixed or adjustable orifice works equally well here. The orifice size determines the length of decompression time. If the orifice is too large, shock is less but may still be enough to cause damage. If the orifice is too small, there is no shock but cycle time may slow.

When pressure in the cylinder's cap end drops to the pressure switch setting -- as in Figure 7-6 -- the pressure switch shifts to its normal condition. The normally closed contacts on the pressure switch pass a signal to the retract solenoid on the directional valve, and the cylinder retracts.

Fig. 7-6. Press circuit with decompression protection – while cylinder is retracting.



Large press circuit with prefill valve and decompression

On presses with large-bore cylinders or rams, oil compressibility is a problem. Another problem can be how to fill the ram as it approaches the work at high speeds and how to empty the ram when it retracts rapidly. The circuit in Figures 7-7 through 7--12 shows how to use a prefill valve to fill and empty a large ram. This type of prefill valve also can decompress the ram automatically without electrical controls.

Fig. 7-7. Press circuit with prefill and decompression valves – at rest with pump running.

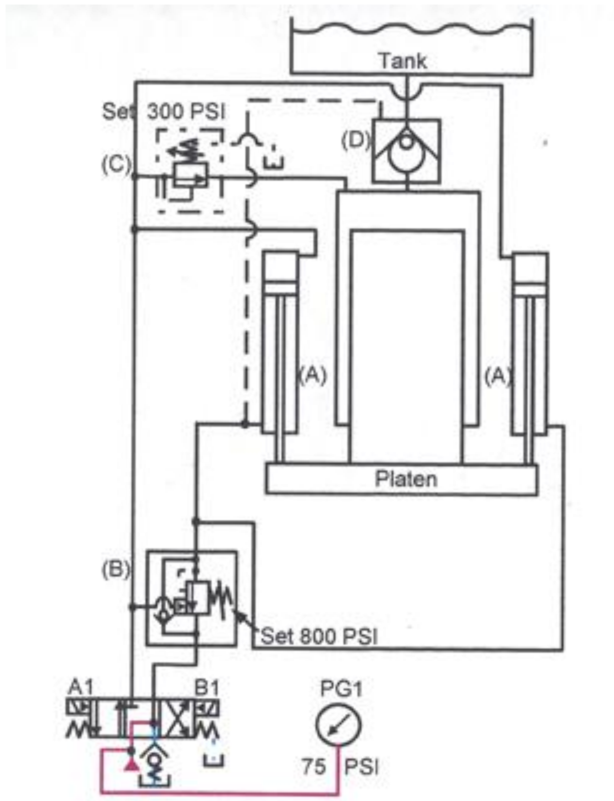
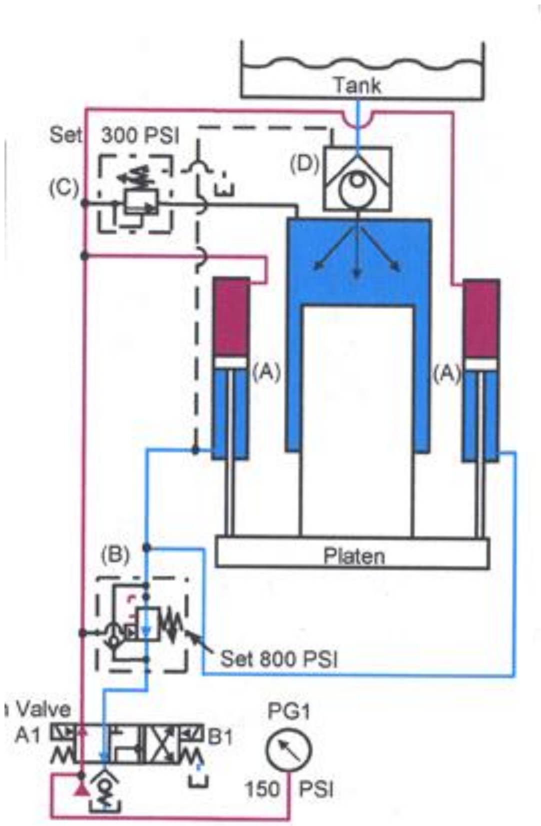


Figure 7-7 shows the parts of a typical high-tonnage press. Small double-acting cylinders A (sometime called outriggers or pull-back cylinders) rapidly extend and retract the large ram. A small volume of oil cycles the outriggers for fast advance and return. Counterbalance valve B keeps the outriggers from running away and sequence valve C directs all fluid to the outriggers until the platen meets resistance. As the ram advances, vacuum opens prefill valve D, sucking fluid out of the tank to fill the large volume. Piloting the prefill valve open on retract first decompresses trapped oil, then allows free return flow to tank from the ram.

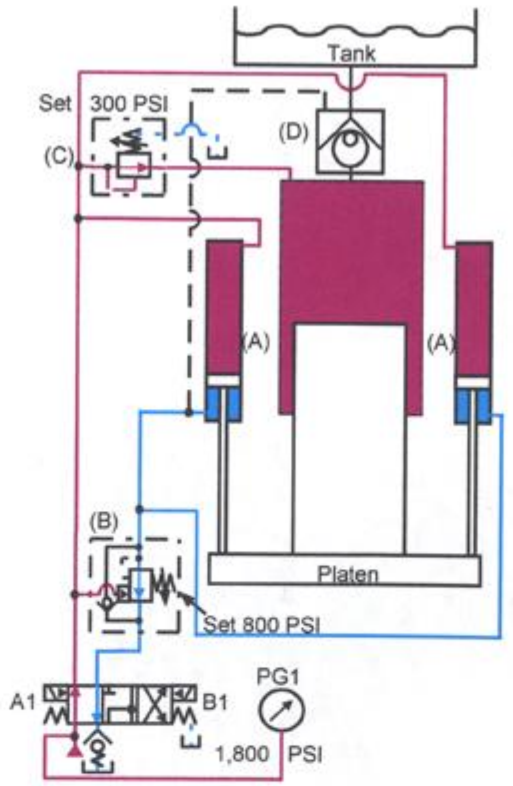
Figure 7-8 shows the cylinder extending toward the work. Pump flow to the outriggers A increases the rod-end pressure of these cylinders to open counterbalance valve B. When B opens, the platen starts forward and the ram pulls a vacuum in the cylinder tube. This vacuum sucks prefill valve D open and oil flows from the tank to fill the ram void. As the ram extends, the cylinder tube continues filling from the tank through D.

Fig. 7-8. Press circuit with prefill and decompression valves – ram extending in fast-forward mode.



When the platen meets resistance, forward movement stops and pressure increases in the outrigger cylinders, Figure 7-9. When the ram stops, prefill valve D closes and pressure build-up opens sequence valve C, oil from the pump flows to the ram and outriggers simultaneously. The press can develop full tonnage during this part of the cycle. Ram speed during full tonnage is relatively slow because the pump flow is low in relation to ram volume. However, the horsepower requirement is at a minimum while the overall cycle is fast.

Fig. 7-9. Press circuit with prefill and decompression valves – ram pressing.



The outrigger cylinders must produce enough force while retracting to raise the platen and ram, as well as to discharge the volume of oil displaced by the ram. If the outriggers have a 2:1 rod-area ratio, use a regeneration circuit on the forward stroke for faster speed or add a small pump.

Replacing the sequence valve with a normally closed 2-way directional valve allows the use of a limit switch to tell the ram to slow before contacting the work. Also, using a bi-directional pump to control direction, speed, acceleration, and deceleration is common for large cylinders on presses or some other machines.

When the press completes its work stroke and reaches full tonnage, Figure 7-10, it is ready to retract. Pressure in the circuit is 2800 psi and the trapped oil contains a large amount of stored energy. To retract the press, deenergize the directional valve's forward solenoid and energize its retract solenoid. The sequence valve closes when the directional valve shifts, and fluid in the cap ends of the outrigger cylinders flows to tank.

Fig. 7-10. Press circuit with prefill and decompression valves – ram generating full tonnage.

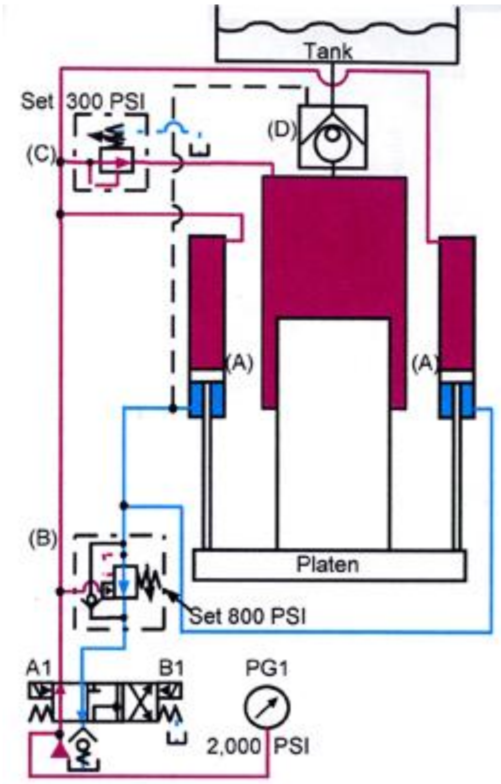
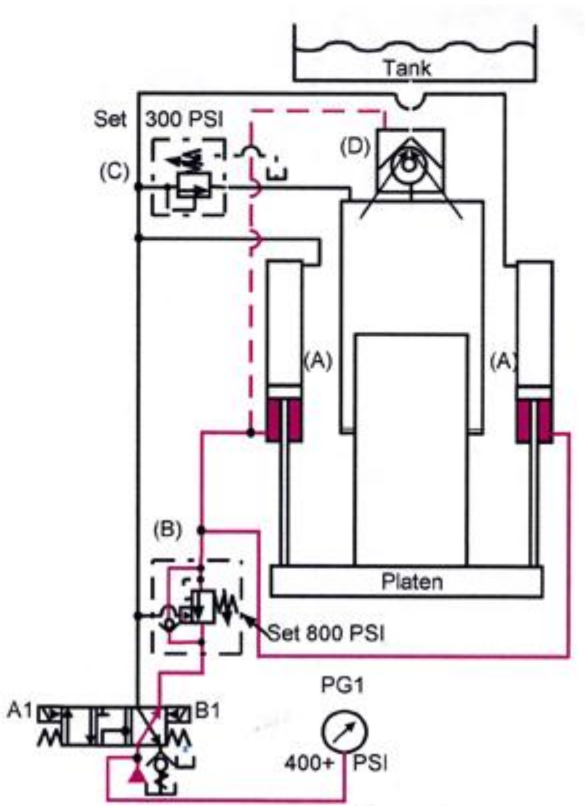


Figure 7-11 shows the press in decompression mode. Fluid from the pump flows to the outrigger cylinders' rod ends and to the pilot port of the prefill valve. A prefill valve operates almost the same as a pilot-operated check valve. Pilot pressure opens the flow poppet for reverse flow when needed. However, on a prefill valve, the ratio of the pilot-piston area to the flow-poppet area is the reverse of a normal pilot-operated check. Most pilot-operated check valves have 3 to 4 times more pilot-piston area than flow-poppet area. On a prefill valve, the pilot-piston area is only about 1/10th of the flow-poppet area. This reverse area ratio keeps the flow poppet closed until most of the backpressure against it dissipates. Another feature of the prefill valve is that inside the main poppet of the prefill valve is a smaller poppet. The area of this small poppet is only 1/16th the area of the pilot piston, so it opens easily -- even with high pressure trapped inside the ram. The flow capability of the small poppet gives a quick, smooth decompression when it is piloted open.

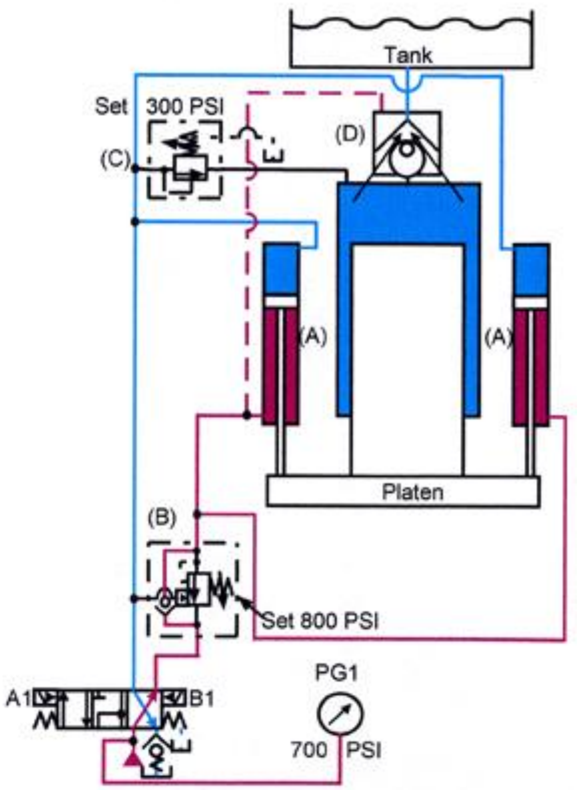
Fig. 7-11. Press circuit with prefill and decompression valves – ram decompressing.



As pressure builds on the rod sides of the outrigger cylinders, pressure in the pilot line to the prefill valve also increases. When pilot pressure is high enough to open the small poppet, decompression flow lowers pressure in the ram at a controlled rate. When ram pressure is low enough, pilot pressure opens the main prefill poppet. Low shifting pressure and flow of the inner poppet allows the prefill valve to meet most system requirements.

When the main prefill poppet opens, Figure 7-12, the ram freely retracts at high speed. Pump flow into the rod end volumes of the outrigger cylinders determines the ram's retraction speed. The prefill valve allows fast ram movement in both directions of travel. This same prefill valve often has the option of automatic decompression as shown here. (Some manufacturers make prefill valves with large spools or sliding sleeves. They operate differently, but the end results are basically the same.)

Fig. 7-12. Press circuit with prefill and decompression valves – ram retracting rapidly.



Simple decompression for a single-cylinder circuit

Figures 7-13 through 7-16 depict a simple but effective decompression circuit for an application with a single valve and cylinder. There are no separate decompression valves to operate. This circuit can be adjustable and is easy to set up and maintain.

In Figure 7-13 the circuit is at rest. Pressure switch A keeps the directional valve from retracting the cylinder until a safe minimum pressure is reached. Check valve B blocks pump flow from the cylinder while retracting. Directional valve C unloads the pump, blocks main cylinder flow during decompression, and extends and retracts the cylinder. Adjustable or fixed orifice D controls decompression speed.

Fig. 7-13. Simple decompression circuit – at rest with pump running.

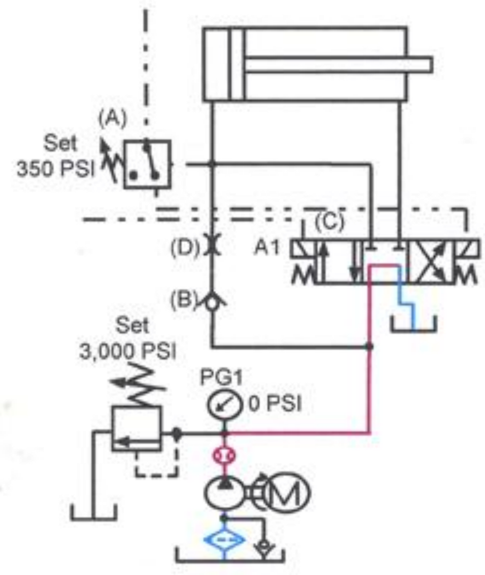
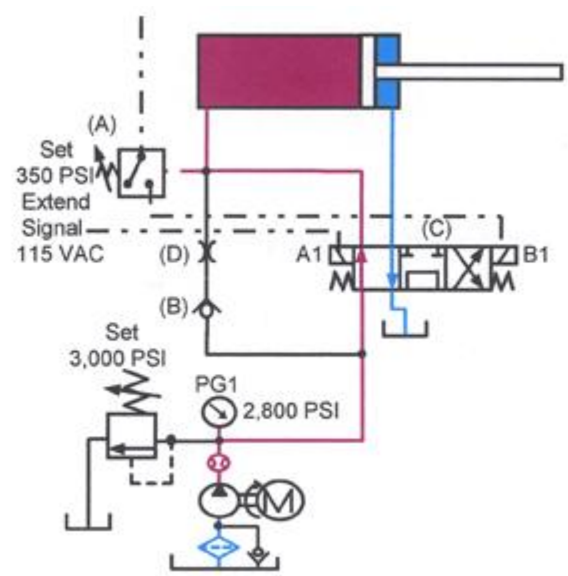


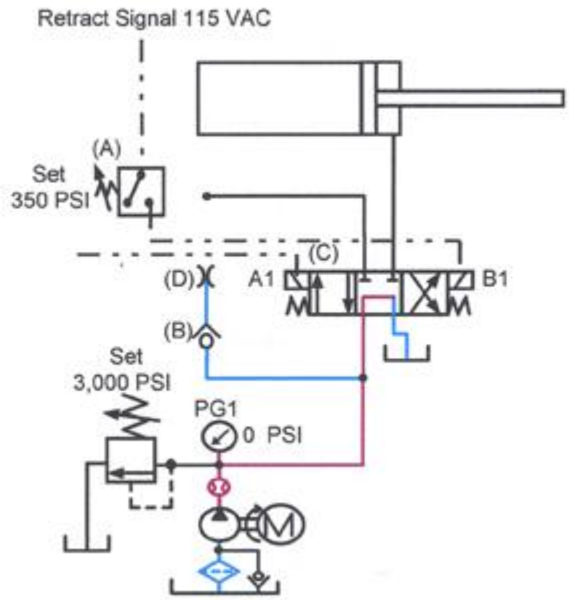
Figure 7-14 shows the cylinder meeting resistance and pressure increasing in the circuit.

Fig. 7-14. Simple decompression circuit – cylinder extended and generating full force.



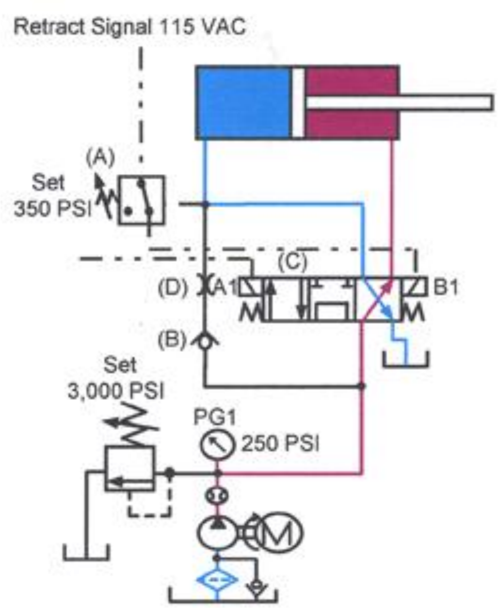
To retract the cylinder, Figure 7-15, the extend solenoid is deenergized and a retract signal goes to the normally closed contacts of pressure switch A. (These normally closed contacts are open at this time because pressure in the cylinder cap end is well above the 350-psi setting.) Directional valve C shifts to its center position; the pump unloads; and trapped fluid decompresses through orifice D and check valve B. The pump-to-tank condition of tandem-center directional valve C allows decompression flow while centered. Decompression lowers pressure in the cylinder cap end quickly, without shock, until pressure reaches the setting of pressure switch A.

Fig. 7-15. Simple decompression circuit – cylinder decompressing.



When the contacts on the pressure switch close, they pass a signal to the retract solenoid on directional valve C. The valve shifts and the cylinder retracts as shown in Figure 7-16. This circuit requires no special electrical controls while eliminating decompression shock.

Fig. 7-16. Simple decompression circuit – cylinder retracting after decompression.



Another way to control the decompression portion of the cycle uses a time-delay relay. If the signal to retract comes from a time-on delay, set it for enough time to allow orifice D to *decompress the cylinder before sending a retract signal to directional valve C. This type of control always gives an exact cycle time. Set the time long enough to make sure decompression takes place under any operating conditions. This usually makes the cycle longer than necessary, so it may not be a satisfactory arrangement for all machines.*

Fluid Motor Circuits

One way to change fluid energy into useful work is through air or hydraulic motors. These fluid motors produce rotary power that can drive conveyors, operate long transfers, power fan blades, run a winch, drill and tap a hole, and handle many other applications.

Compared to electric motors, quick reversal of rotation or stalling does not damage a fluid motor. Changing motor speed (within the limits of its specifications) does not adversely affect torque. Repeated starting and stopping air or hydraulic motors does not cause damage or overheating. Also, hydraulic motors commonly operate at low speed without a gear reducer. These features — along with compact size — make fluid motors the best choice in many applications.

To size a fluid motor, calculate torque instead of horsepower. However, remember that fluid motors have approximately 50% less starting torque than electric motors of the same horsepower. Make allowance for this reduced torque if the motor must start under load.

Vane-design air motors are common in low-torque/high-speed applications. For some high-torque applications, vane motors with integral gear reducers work well. Piston-type motors are larger and more expensive, but they produce high torque at low speeds. A less expensive, more-compact gerotor-design air motor that operates in the lower-speed range also is available.

Air motors operate safely in most environments and save space at the work site. Air motor efficiency is very low — only in the 15 to 30% range. Minimizing run time by starting and stopping the motor helps offset low efficiency in many applications.

Air motor speed varies with load changes. Power level and torque also vary widely with speed change. Without a good silencer or muffler, air motors are loud.

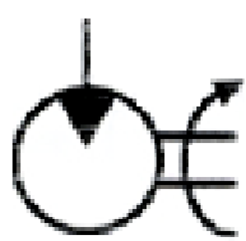


Fig. 12-1. Uni-directional hydraulic motor.

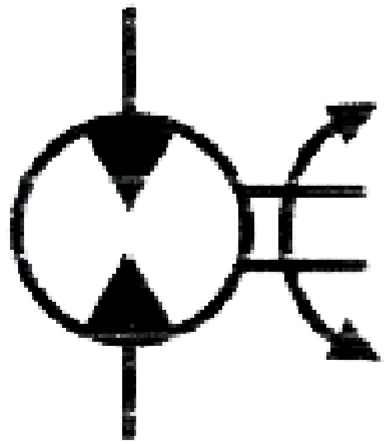


Fig. 12-2. Bi-directional hydraulic motor.

Hydraulic motors, on the other hand, can produce speeds from 1 to 5000 rpm. With proper valving they produce little shock, and their efficiency is in the 80 to 95% range. They also operate safely in most environments and save space at the work site. They are very rugged. Unlike cylinders with resilient seals, hydraulic motors always leak or bypass internally. To stop and hold over-running loads, use a brake or other external device.

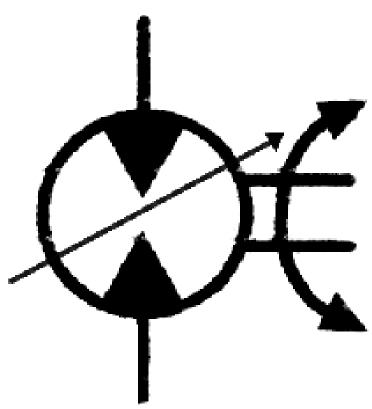


Fig. 12-3. Variable-displacement, bi-directional hydraulic motor.

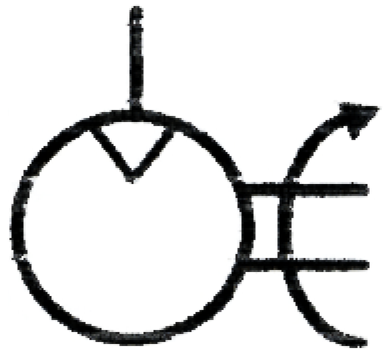


Fig. 12-4. Uni-directional air motor.

Figure 12-1 and Figure 12-4 show the symbols for a uni-directional hydraulic and air motor. Figures 12-2, 12-3, and 12-5 show motors designed for bi-directional operation. Figure 12-3 shows a variable-displacement bi-directional hydraulic motor symbol.

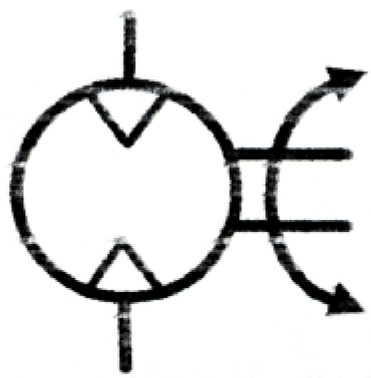


Fig. 12-5. Bi-directional air motor.

Sizing a hydraulic motor system

To size a hydraulic motor system, the following information must be known:

1. Maximum torque required (usually in in.- or ft-lb)

2. Maximum speed required (usually in revolutions per minute, rpm)

3. How motor will be stopped (coast, braked, decelerated, other)

4. Maximum pressure allowed (in psi, arbitrarily decided by the engineer or designer)

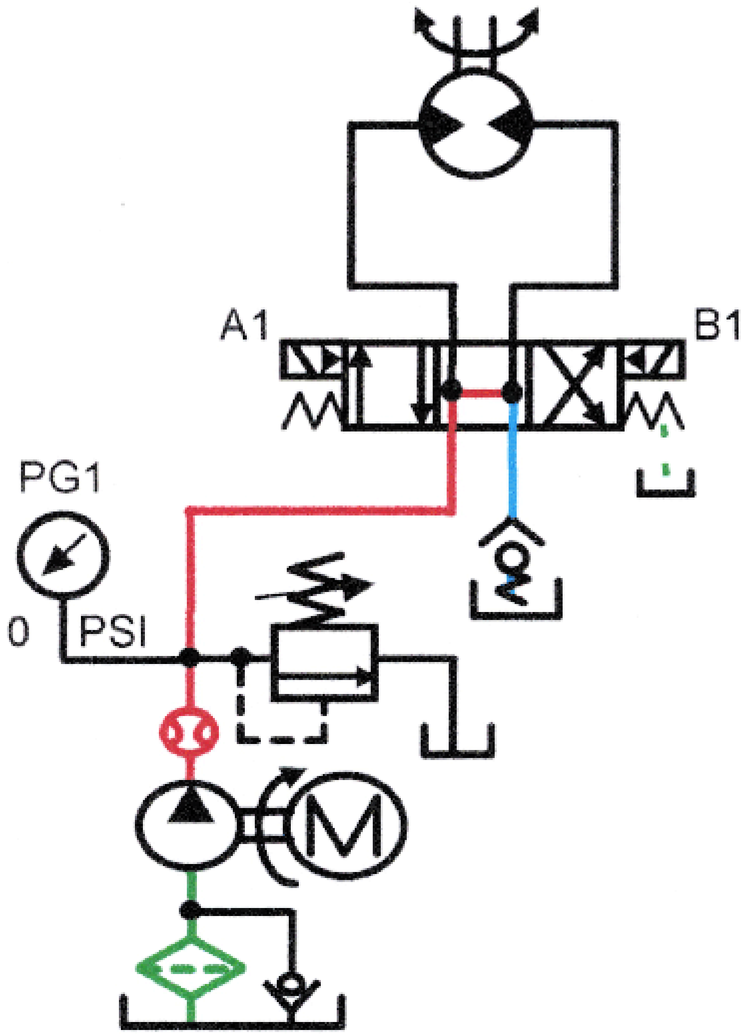


Fig. 12-6. Bi-directional hydraulic motor at rest with pump running.

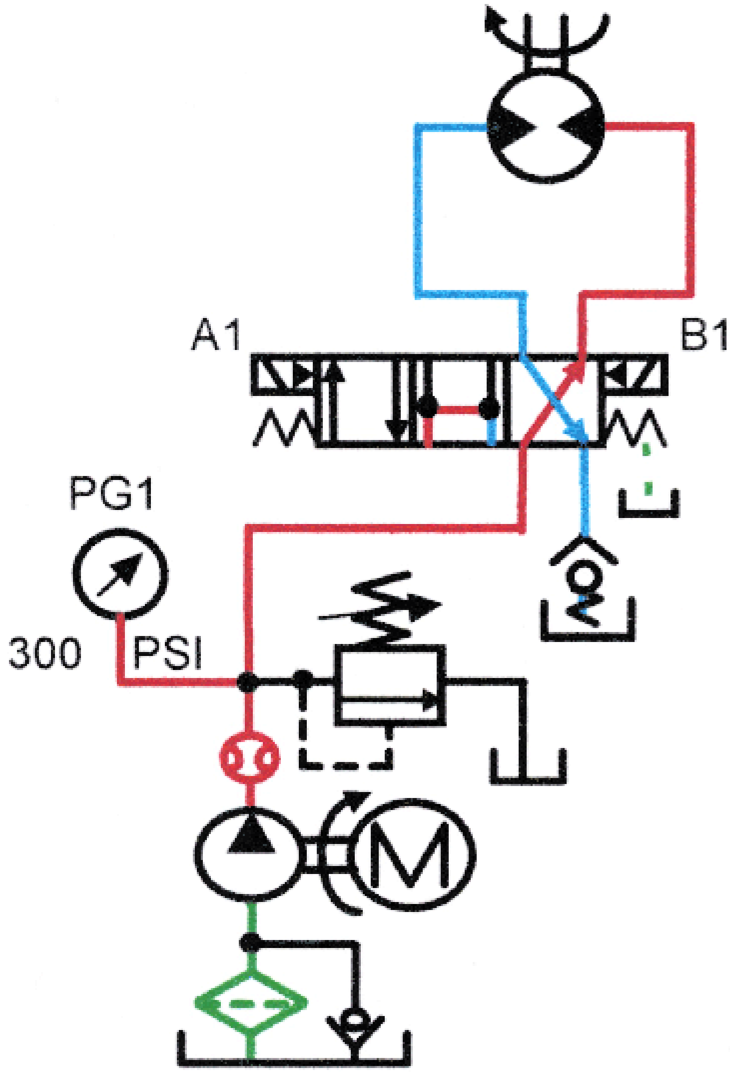


Fig. 12-7. Bi-directional hydraulic motor running clockwise.

Sample problem – see Figures 12-6 through 12-9

Maximum torque = 5400 in.-lb Maximum pressure = 3000 psi

Maximum speed = 220 rpm Motor will coast to a stop

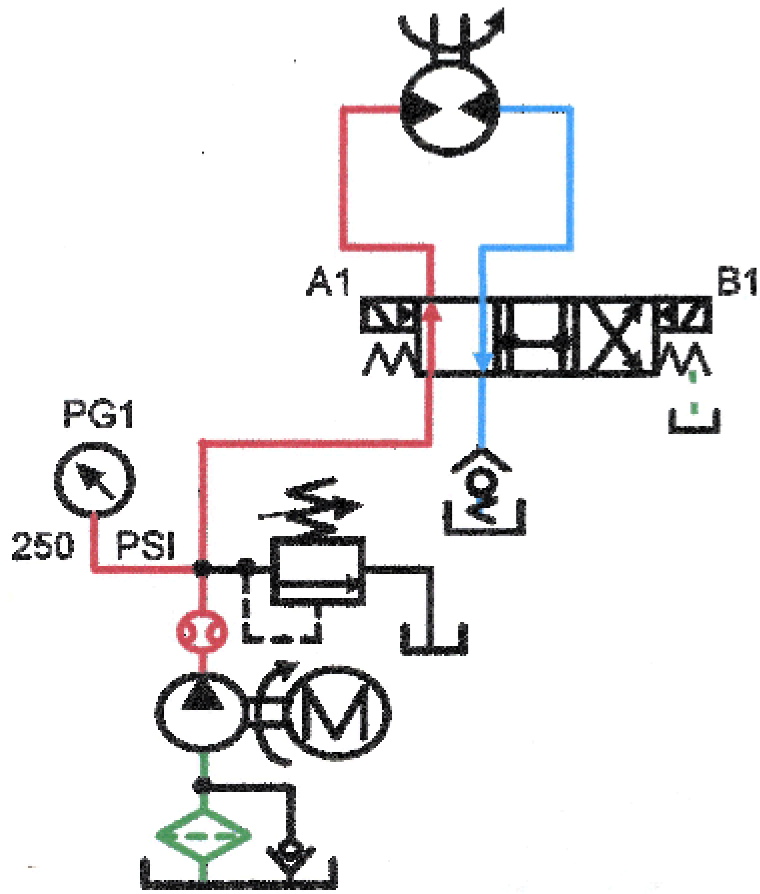


Fig. 12-8. Bi-directional hydraulic motor running counterclockwise.

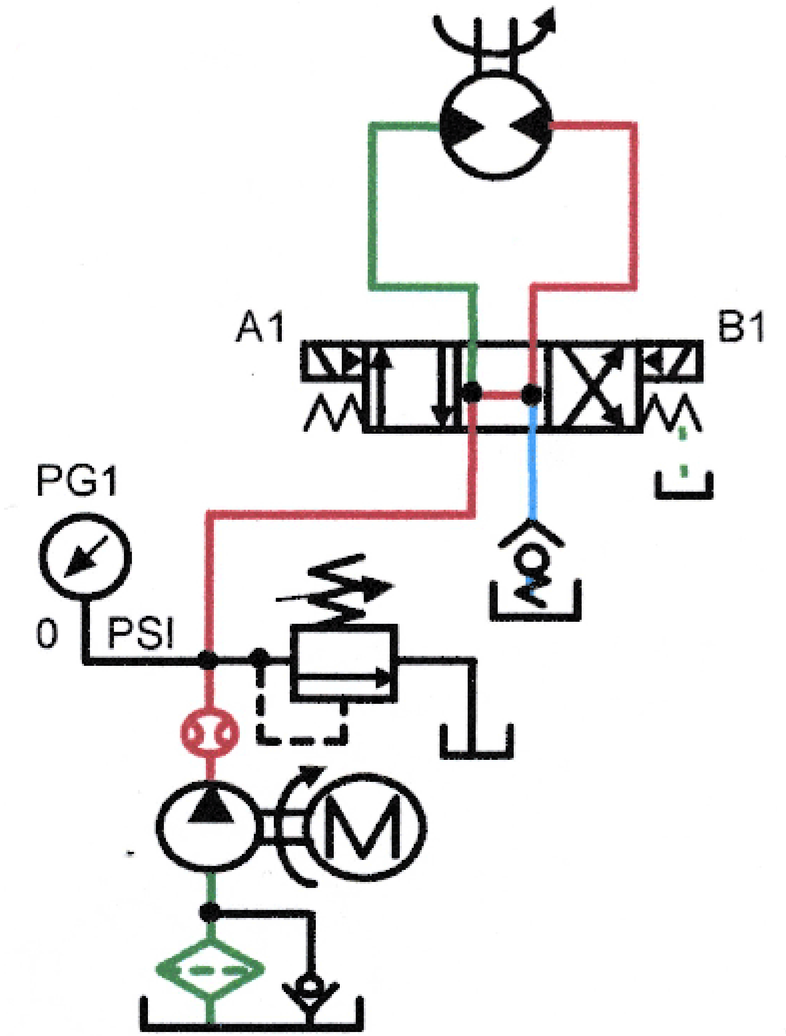


Fig. 12-9. Bi-directional hydraulic motor stopping.

A. Calculate minimum cubic inches per revolution (cir)

$$CIR = \frac{2\pi \text{ X TORQUE } ^{\text{(Inch Pounds)}}}{\text{psi}}$$

$$CIR = \frac{2 \text{ X } 3.1416 \text{ X } 5{,}400}{3{,}000} = 11.31 \text{ CIR Minimum}$$

Choose 14.4 CIR from supplier's catalog

B. Pump capacity $GPM = \frac{CIR \times rpm}{231_{\text{Cubic Inches/Gallon}}}$

$$GPM = \frac{14.4 \times 220}{231} = 13.7 \text{ GPM minimum}$$

Choose 15 GPM from supplier's catalog

C. Electric motor $HP = GPM \times psi \times .000583$

$$HP = 15 \times 3{,}000 \times .000583 = 26.24 \text{ HP}$$

Choose 25 HP from supplier's catalog

D. Tank size = 2 or 3 times pump GPM

Tank size = 2 X 15 = 30 gallons
Tank size = 3 X 15 = 45 gallons

Choose 50-gal. tank from supplier's catalog

Sizing an air motor system

To size an air motor system the following information must be supplied:

1. Maximum torque required — Usually in foot-pounds (or horsepower could be substituted)
2. Maximum speed required — Usually in revolutions per minute, rpm
3. How motor is stopped — Coast, braked, decelerated, other
4. Maximum pressure allowed — Usually 80 to100 psi

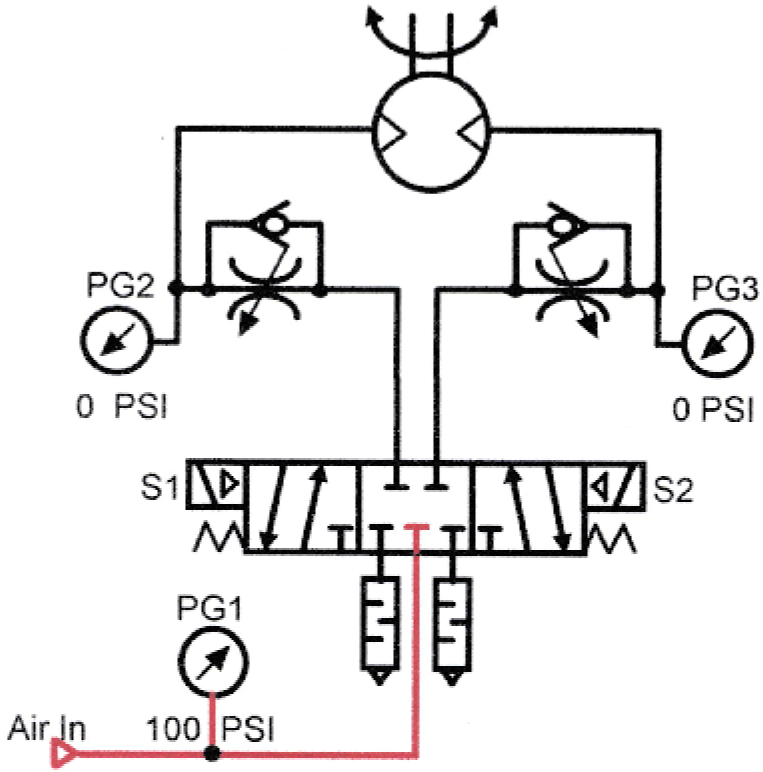


Fig. 12-10. Bi-directional air motor at rest.

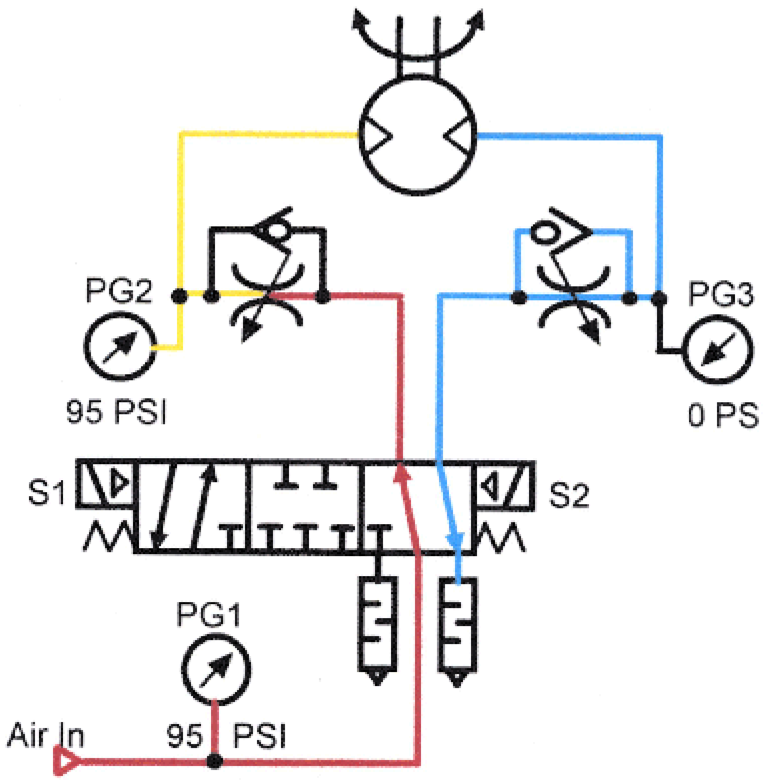


Fig. 12-11. Bi-directional air motor just starting to rotate.

Sample problem – see Figures 12-10 through 12-13

1.	Maximum torque required	10 ft-lb
2.	Maximum speed required	150 rpm
3.	How to stop motor	Retarded
4.	Maximum pressure available	90 psi

A. Minimum horsepower required

$$HP = \frac{TORQUE^{\text{(Foot Pounds)}} \times rpm}{5{,}252}$$

$$HP = \frac{10 \times 150}{5{,}252} = .286\ HP$$

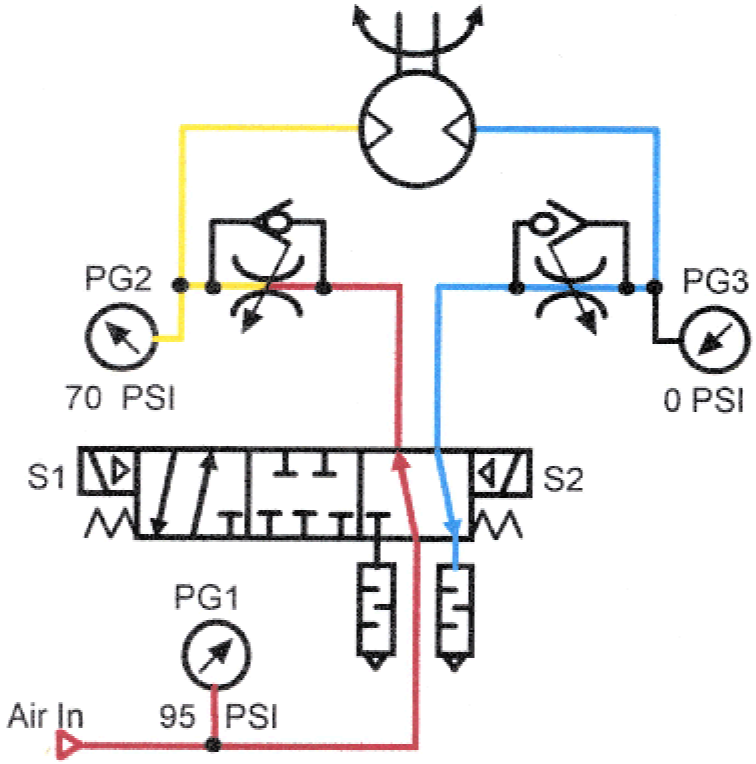


Fig. 12-12. Bi-directional air motor running at full torque.

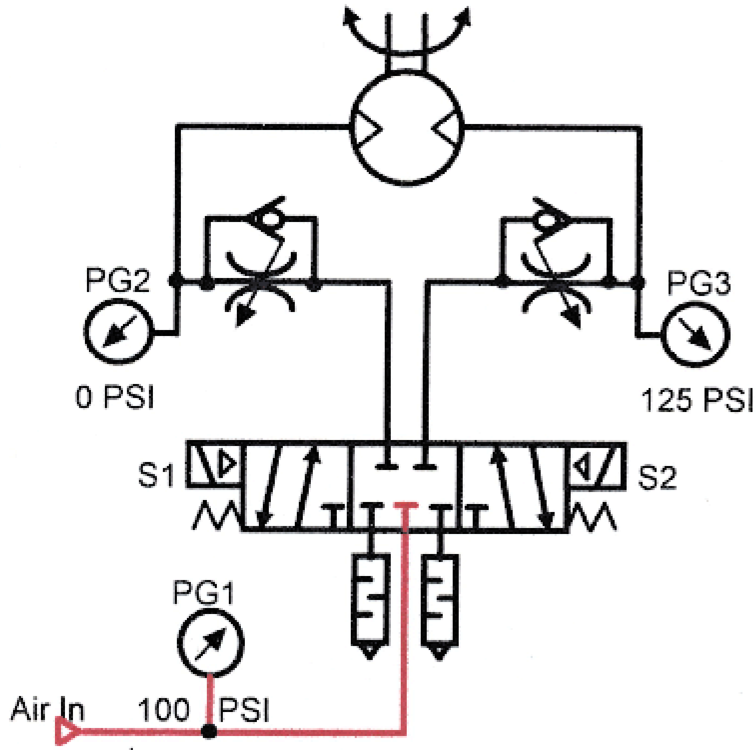


Fig. 12-13. Bi-directional air motor stopping with backpressure.

Directional controls for hydraulic motors

Figure 12-14 shows a solenoid-operated control circuit for a small, uni-directional hydraulic motor. Energizing the solenoid starts and runs the motor. Deenergizing the solenoid allows the motor to coast to a stop. Depending on the type and amount of load, starting and stopping the motor may be anywhere between smooth and abrupt.

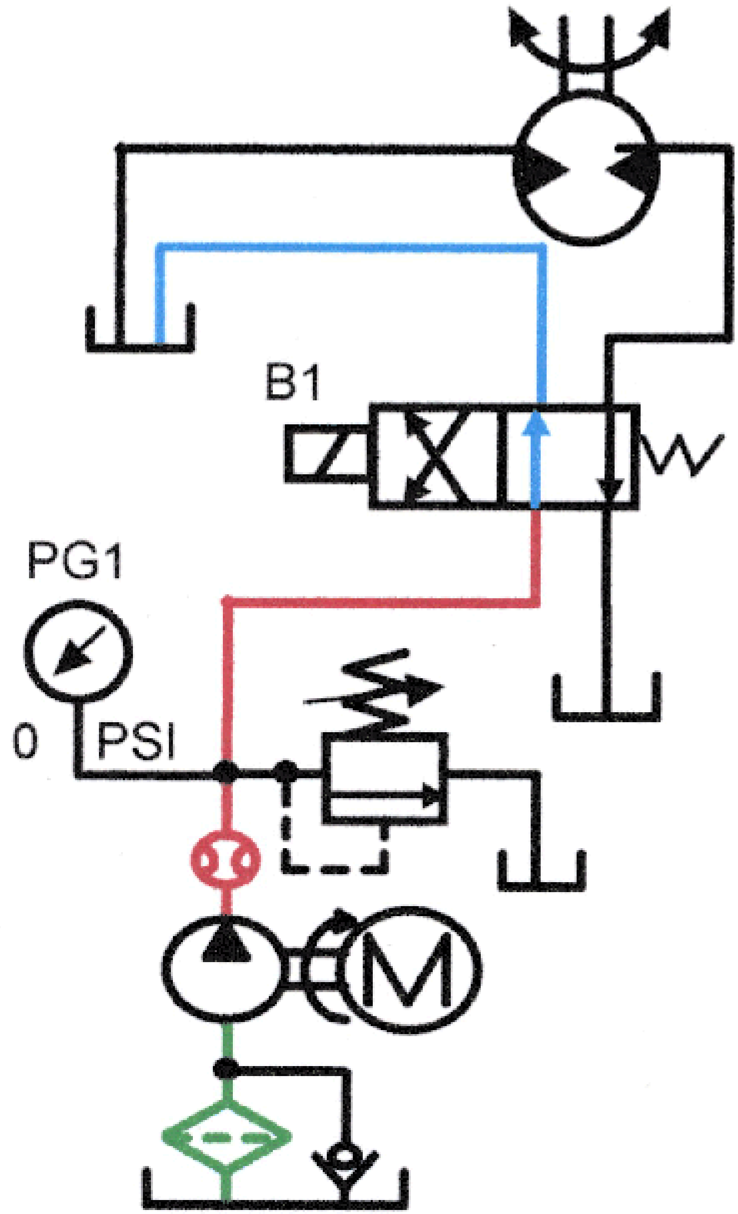


Fig. 12-14. Small hydraulic motor circuit with single direction of rotation.

Figure 12-15 shows how to control a large uni-directional hydraulic motor via a normally open, solenoid-operated relief valve. This circuit is less expensive than one that uses a large directional valve and a relief valve. Solenoid-operated relief valves discharge pump flow to tank at about 20 to 50 psi, normally keeping the motor stopped. Energizing the solenoid on the relief valve causes it to start closing. The closing of the relief valve builds pressure and the motor starts to turn. If pressure tries to go higher than the relief setting, this valve stays partially open, *which gives the motor time to accelerate. When the motor is at maximum speed, pressure drops, and the relief valve closes completely, directing all pump flow to the motor inlet. If the system pressure tries to go higher than the relief setting, the valve opens to protect the circuit. De-energizing the solenoid on the relief valve allows the motor to coast to a stop without cavitation.*

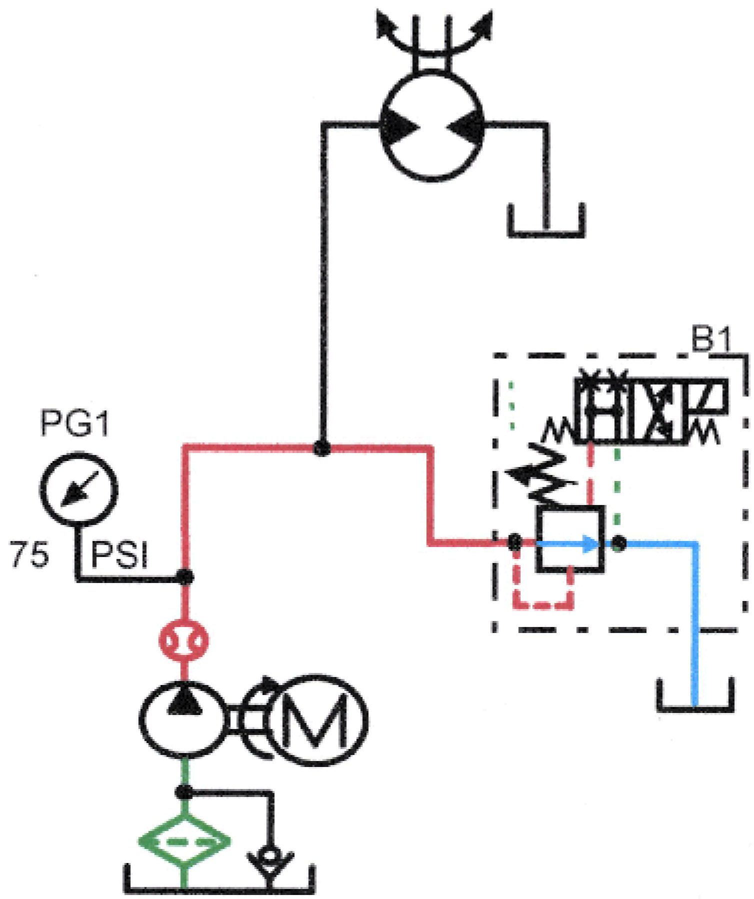


Fig. 12-15. Large hydraulic motor circuit with single direction of rotation.

Note that if the motor can turn at the low unloading pressure (20 to 50 psi), it may never stop completely. Use a brake valve at the motor outlet to keep it from turning if this situation occurs.

Figure 12-16 shows two hydraulic motors in a parallel circuit. Supplying two or more motors from a single valve lets the fluid follow the path of least resistance. Synchronizing the motors with a flow divider (Chapter 11) or by a mechanical linkage would keep them together. With parallel circuits, all motors have full torque but only get a portion of pump flow. In other words, the motors have equal power at reduced speed.

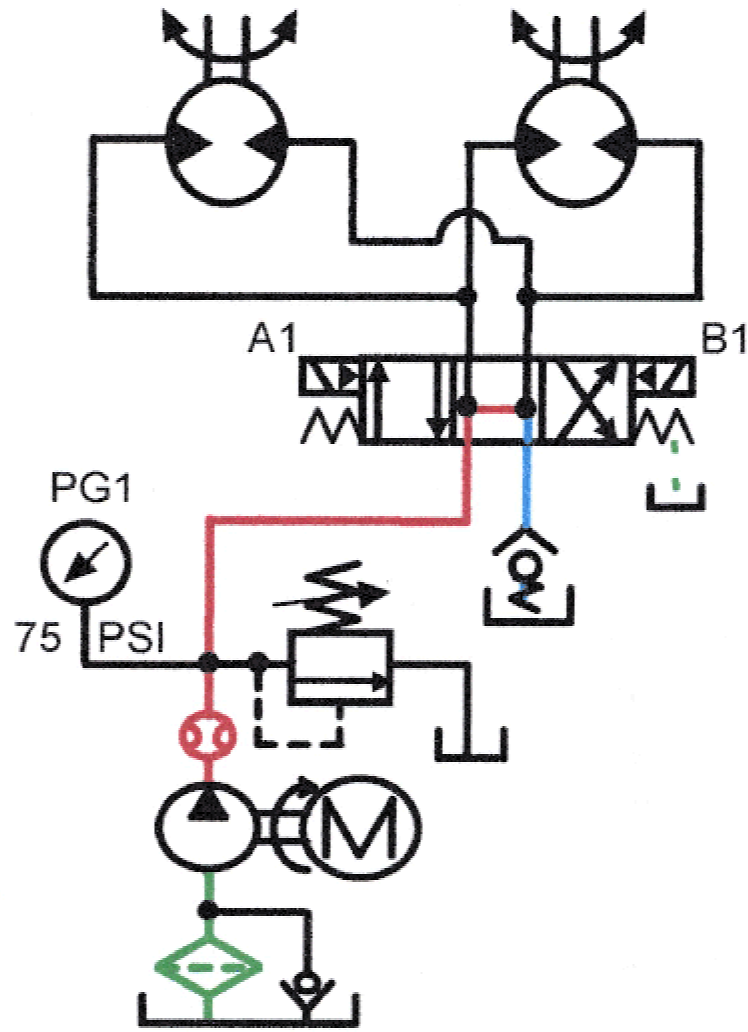


Fig. 12-16. Parallel hydraulic motor circuit with bi-directional rotation.

The series circuit in Figure 12-17 has the outlet of the first motor piped to the inlet of the second. This series circuit gives nearly perfect speed synchronization, but reduces each motor's power.

The series circuit works well with hydraulic motors because outlet flow is almost identical to inlet flow. This near-equal flow provides tolerable synchronization, but generates backpressure on the inlet and outlet ports of the leading motor. All hydraulic motors have some internal leakage that normally goes to the low-pressure or tank-side outlet through internal check valves. With motors in series, backpressure to run the downstream motors can cause excessive case pressure that may blow out the shaft seal. Some hydraulic motors are available with high-pressure seals that eliminate the blowout problem. Another option is to select motors with *external case drains (as shown in the Figure) that allow leakage oil to return to tank at little or no backpressure. When external case drains, remember that leakage in the leading motors will cause the downstream motors to run more slowly.*

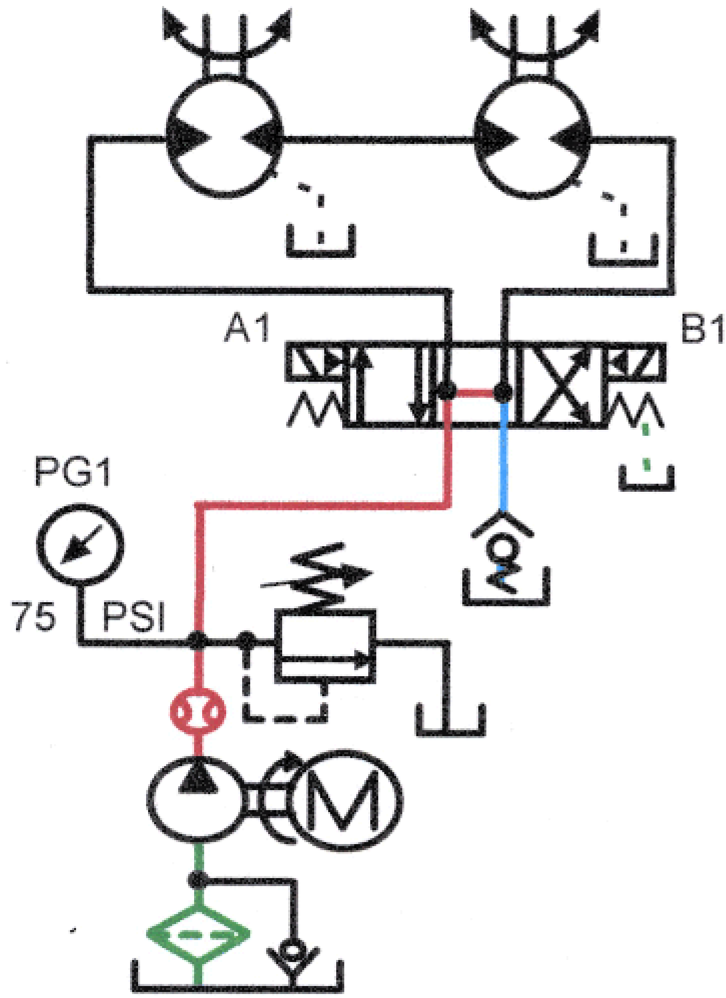


Fig. 12-17. Series hydraulic motor circuit with bi-directional rotation.

All of these circuits show open-centered valves for smooth stopping or coasting of the motors. This works well if the motor has little or no tendency to coast or if coasting is allowable. Next, we'll look at some circuits for rapidly and smoothly stopping hydraulic motors with over-running loads.

Circuits for hydraulic motors with over-running loads

Figure 12-18 depicts the most common way to slow a hydraulic motor with an over-running load. Notice that the directional valve has blocked ports A and B. This valve's center condition makes a motor stop abruptly, which could cause high shock and physical damage. Shock occurs because the motor becomes a fixed-displacement pump without a relief valve to protect it from over pressure. The cross-port relief valve shown in the Figure allows fluid from the over-pressured port to go to the opposite motor port. The reason for piping the outlet of the reliefs to the opposite motor port is to keep it from cavitating as the motor decelerates. Cross-port reliefs are available with both relief valves in a common housing to save piping time and reduce potential leakage points.

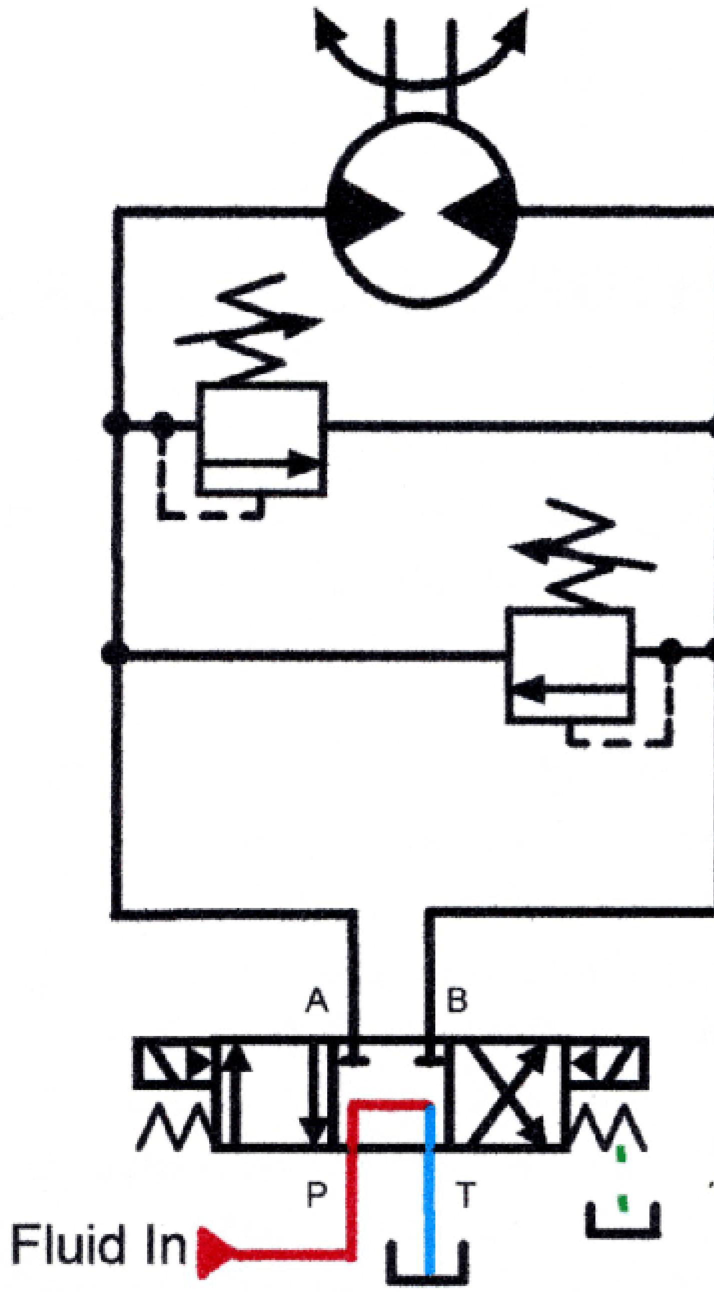


Fig. 12-18. Dual cross-port relief valves.

Set the pressure for the cross-port relief valve the same or higher than system pressure. This gives full starting torque and smooth controlled stopping. When the cross-port relief valve pressure setting is lower than system pressure, it reduces starting and maximum running torque. The only reason for a lower setting at the cross-port reliefs is for longer coasting time. The higher the pressure setting of the cross-port reliefs, the more quickly the motor stops.

By deenergizing the directional valve with the motor at full speed, outlet flow from the motor is blocked and pressure increases. When pressure reaches the setting of the cross-port relief valve, the valve opens and allows flow to the opposite motor port. Backpressure equal to or greater than what it took to start the motor now holds back against the load. The energy of the over-running load dissipates over a short period of time to eliminate shock.

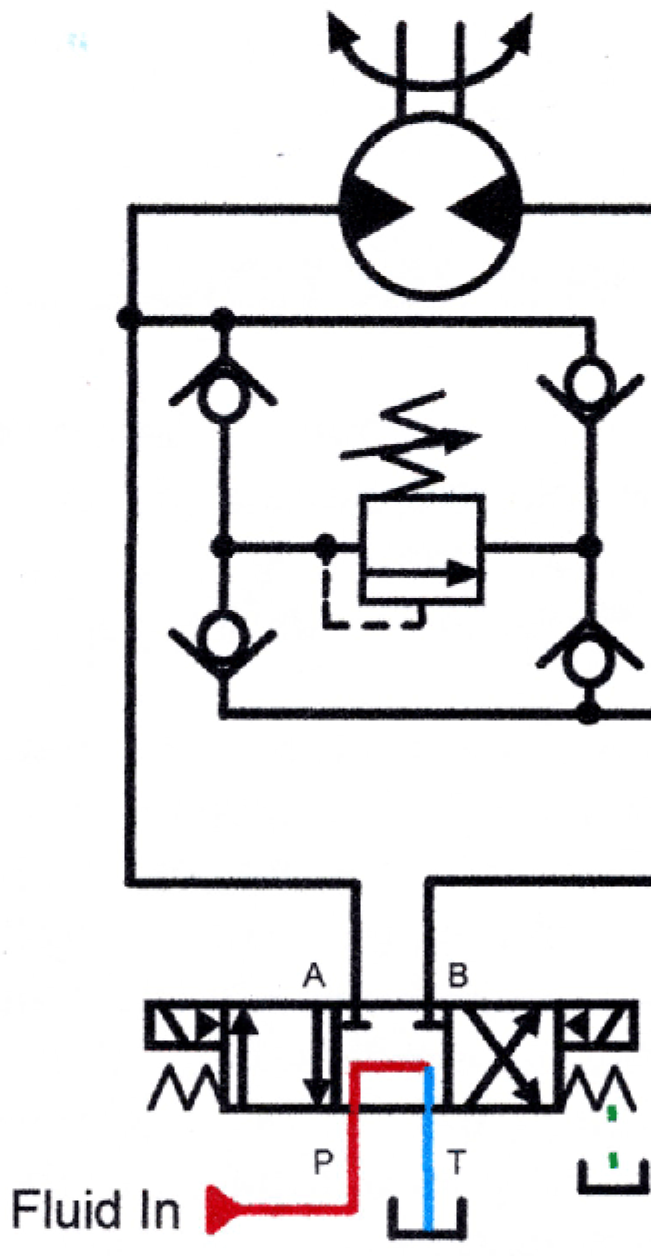


Fig. 12-19. Single cross-port relief valve with check valves.

Dual cross-port relief valves allow different stopping times for the two directions of rotation. If this feature is not required, use the single relief and check valves shown in Figure 12-19. With this circuit, pressure is set only once for both directions of rotation. Notice that the check valves direct flow through the relief from either motor port to the opposite motor port. This check valve arrangement is available in a manifold to save piping time.

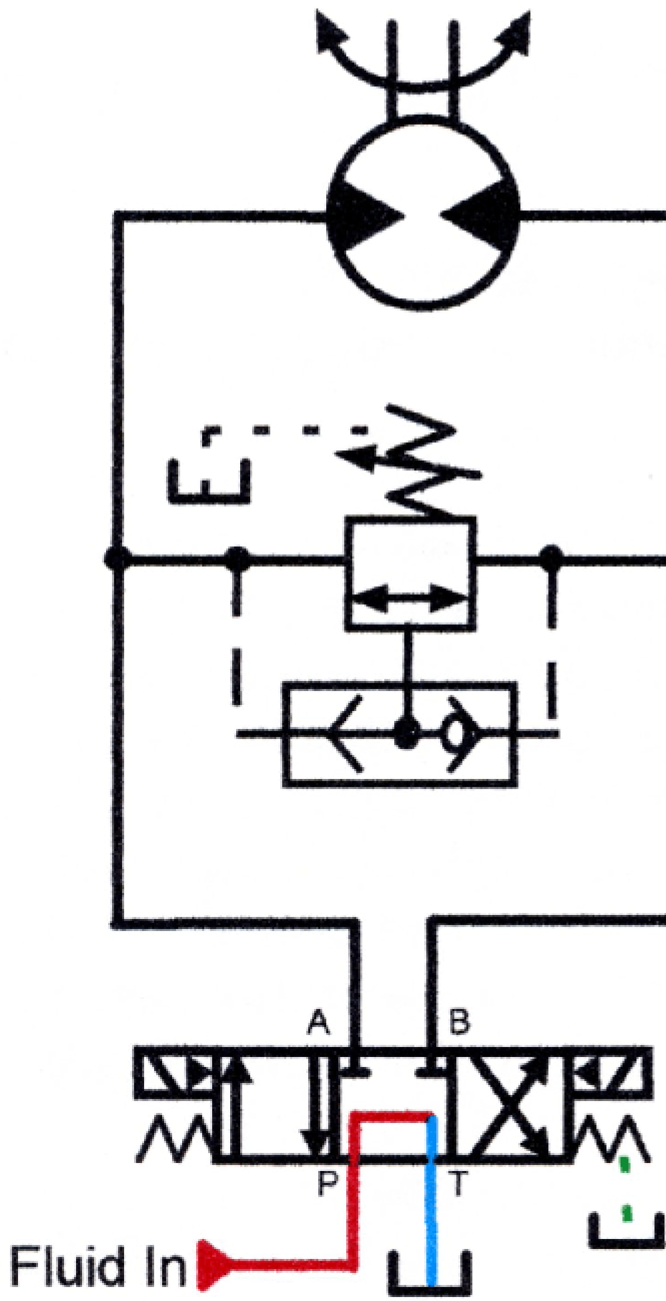


Fig. 12-20. Sequence valve for cross-port relief.

Figure 12-20 shows an externally piloted and drained pressure control valve that slows the motor in both directions. A shuttle valve feeds the pilot port from either motor port to open the pressure control valve. The external drain line allows internal leakage to return to tank. As Figure 12-20 indicates, the pressure control must be set equal to or higher than the system pressure. For longer stopping times, use the piping arrangement in Figure 12-21.

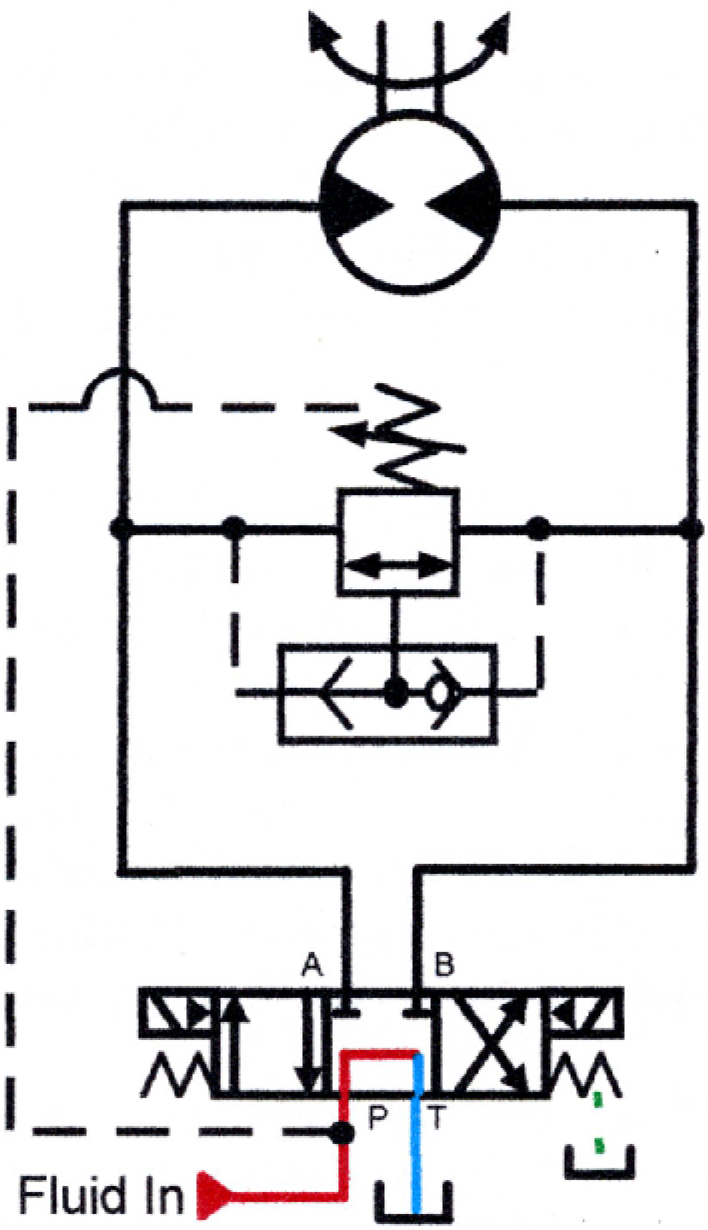


Fig. 12-21. Externally piloted and drained sequence valve for

cross-port relief.

In Figure 12-21, the drain line from the pressure-control valve is piped to the pump line of the directional valve. With the drain line piped this way, the setting of the pressure control is unimportant when the motor is running. Working pressure at the drain port of the pressure control valve adds to the set pressure. Shifting the directional valve keeps the cross-port relief valve from opening when the motor is running.

When the directional valve shifts to its center position, pump flow dumps to tank and the drain line pressure drops, allowing the cross-port relief to operate. This provides pressure lower than system pressure at the cross-port relief valve, so the over-running load takes longer to stop.

Brake valves to control hydraulic motors with over-running loads

Figure 12-22 depicts brake valves (sometimes called over-center valves) piped in the lines between the motor and the directional valve. To control the motor in both directions of rotation, install two brake valves as shown.

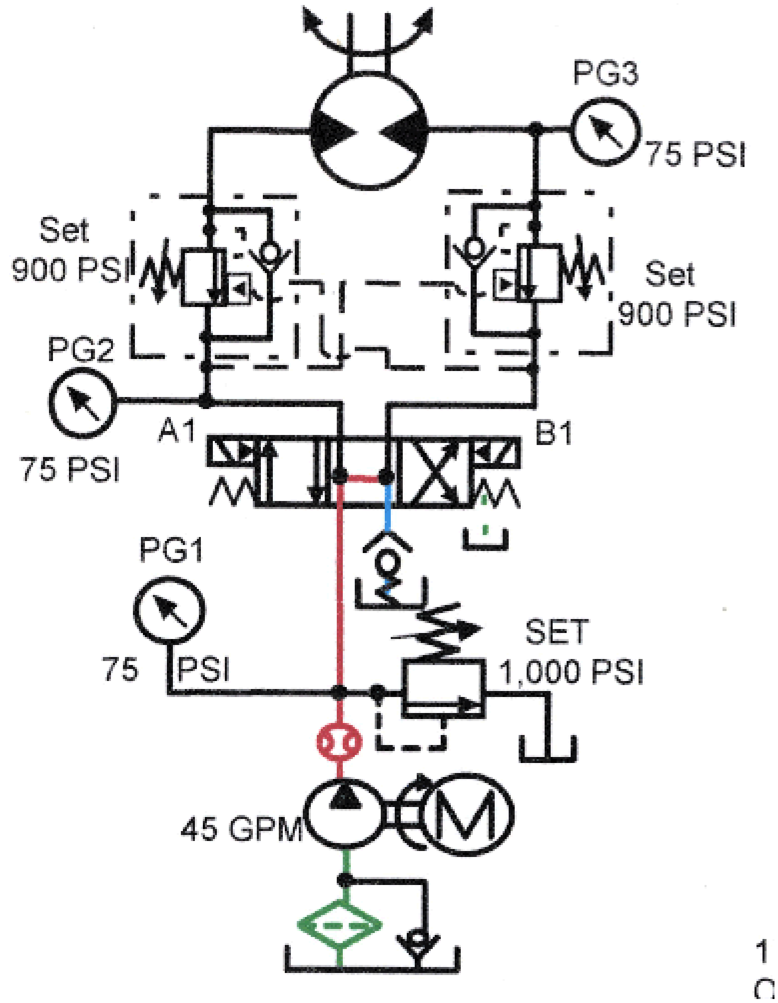


Fig. 12-22. Internally and externally piloted brake valve – at rest with pump running.

A brake valve is an internally and externally pilot-operated pressure-control valve. The internal pilot works on a small area and thus requires a high pilot pressure. The external pilot works on a larger area at a much lower pilot pressure. A common ratio for pilot areas is 8:1. With an 8:1 pilot-area difference, the valve allows fluid to pass when the internal pilot pressure is 1000 psi or the external pilot pressure is 125 psi. Always note the internal pilot pressure setting on the schematic diagram. The internal pilot setting indicates the amount of backpressure at the motor during deceleration.

Brake valve pressure setting is independent of system pressure. This means a lower or higher pressure on the brake valve does not affect the system's maximum operating pressure.

Brake valves are necessary when a hydraulic motor's load tries to make it go faster than the pump or control circuit feeds it. In a winch application, a directional valve shifts to lower a load. Without a brake valve, the winch load falls freely, the motor cavitates, and the circuit is unsafe. Most winch applications require only one brake valve.

Figure 12-22 shows a brake-valve circuit at rest. The brake valves stay closed because their pressure settings are high enough to stop the load. Notice the internal pilot lines on each brake valve that sense pressure in the motor outlet lines. Also, each brake valve has an external pilot line from the opposite motor flow line. Bypass check valves in each brake valve allow free-flow return during reverse flow.

NOTE: Never depend on any valves to stop and hold a hydraulic motor. Hydraulic motors always have internal leakage and will continue to turn slowly without some external braking arrangement. Spring-on, pilot-to-release, multiple-disk brakes are one method of holding a hydraulic motor stationary.

Figure 12-23 shows the brake-valve circuit after shifting the directional valve. Oil entering the left motor port must be at least 112 psi (1/8 of the 900-psi setting) to open the right brake valve. When the load is over-running, inlet pressure to the motor stays at 112 psi to hold the outlet brake valve open. When pressure at the inlet drops below 112 psi, the opposite brake valve closes to retard motor movement. During this portion of the cycle, backpressure keeps the motor from running away. When pressure tries to go above 112 psi, the opposite brake valve opens wide -- dropping all backpressure. As the motor is turning under power, the external pilot supply controls the brake valve.

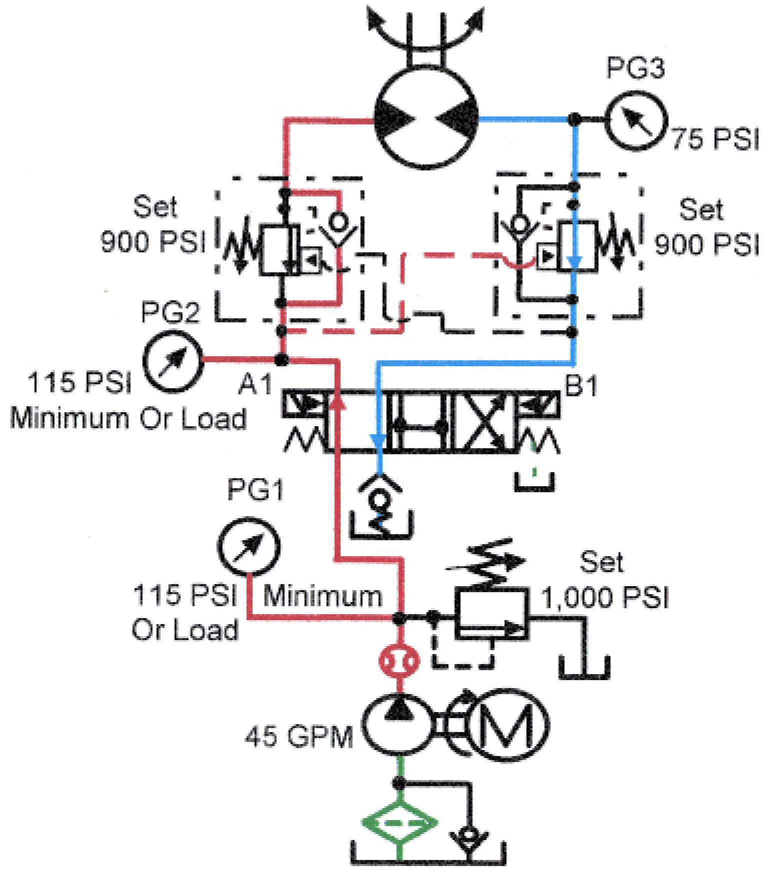


Fig. 12-23. Internally and externally piloted brake valve – with hydraulic motor running.

When the directional valve centers, as depicted in Figure 12-24, external pilot supply to the brake valve drops. The brake valve tries to close, causing pressure to increase at the motor outlet port. When outlet pressure reaches 900 psi, it forces the brake valve open and holds back against the over-running load. Because 900 psi is more than enough to stop the load, the motor decelerates to a smooth controlled stop.

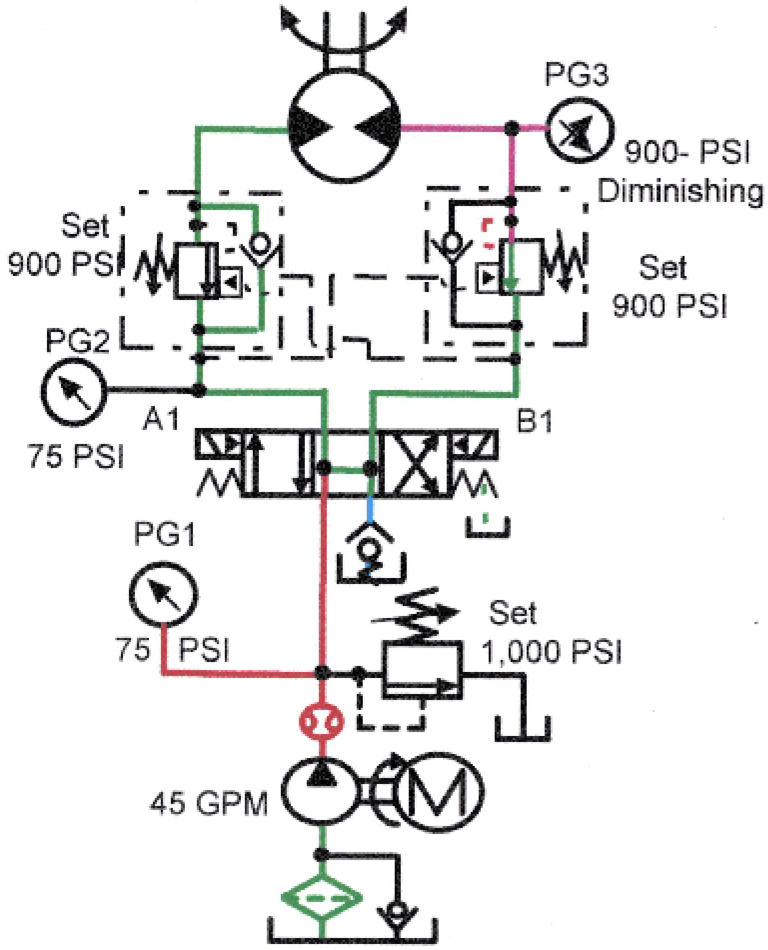


Fig. 12-24. Internally and externally piloted brake valve – with hydraulic motor stopped.

The brake valve only holds the motor back if it is trying to run away. This means little or no energy loss as the motor turns under load — and no running away when the load tries to turn the motor. This action is identical to that of a counterbalance circuit for a cylinder.

Because each brake valve is independent, a different pressure setting at each valve is possible.

Controlling the speed of hydraulic motors

When flow controls set the speed of a hydraulic motor, the result is nearly the same as with a cylinder. The main difference is that cylinders normally have positive seals while motors always have internal leakage. With meter-in and bleed-off circuits, internal leakage causes motor speed to fluctuation as pressure varies.

Figure 12-25 shows a motor circuit with meter-in flow controls. This is the preferred way to control the speed of hydraulic motors. Use a meter-in circuit whenever possible because the motor's internal leakage passes to the low-pressure port through internal check valves. This type circuit makes a case drain or high-pressure shaft seal unnecessary. There is little or no backpressure at the outlet of the motor to cause high pressure at the shaft seal.

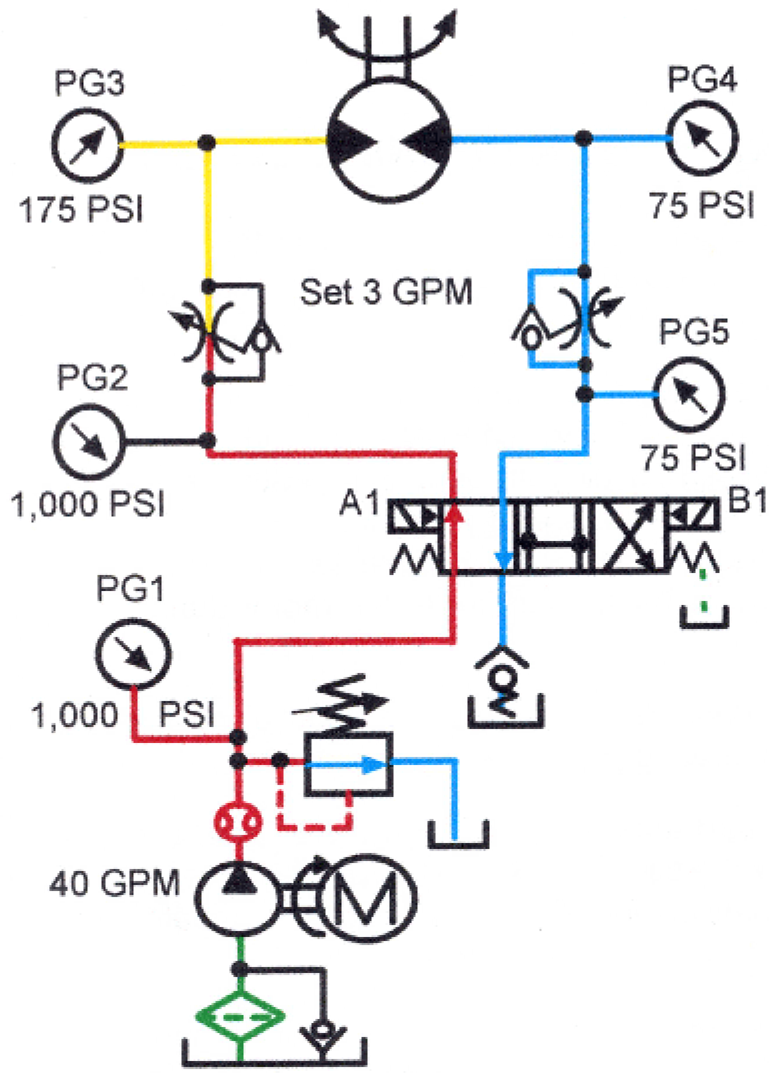


Fig. 12-25. Meter-in flow-control circuit for a hydraulic motor.

When supplied with 3 gpm at 100 psi, the motor turns at low torque. Its speed is approximately 200 rpm. When the motor load increases and pressure drop across it climbs, increased internal leakage causes speed to slow by as much as 10 to 30%. (The lower the efficiency of the motor, the greater the change in speed.) Motors that turn very slowly may even stop as pressure drop *increases. Fast-turning motors may lose speed but continue rotating. As discussed in the section on flow controls, a pressure-compensated valve keeps a constant flow to the motor but cannot offset its internal leakage.*

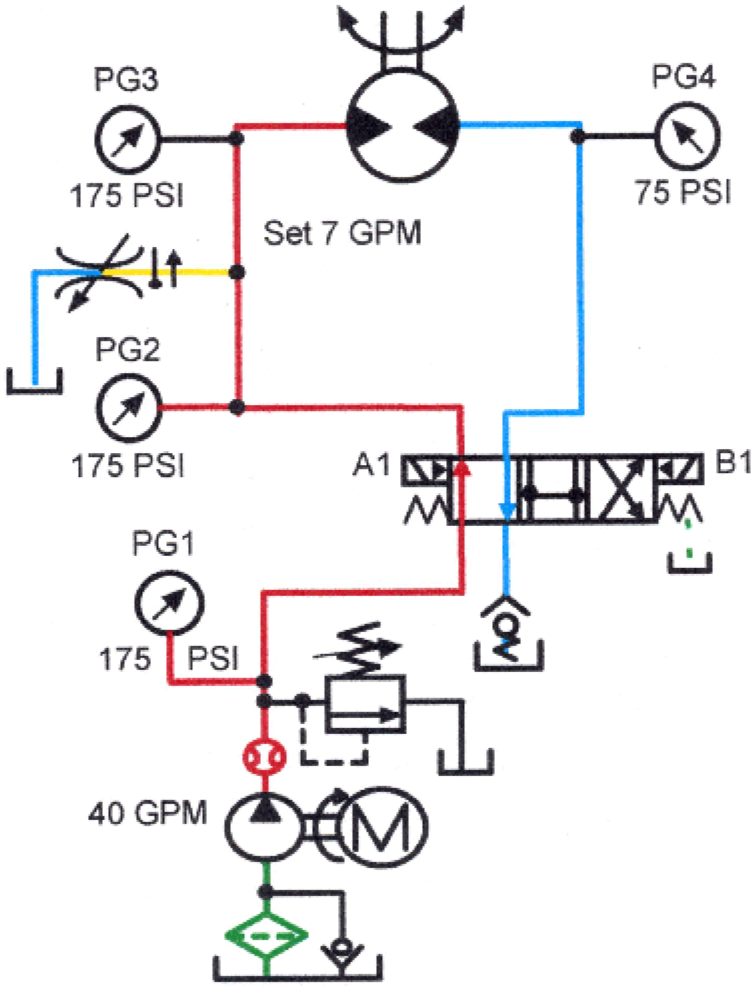


Fig. 12-26. Bleed-off (or bypass) flow-control circuit for a hydraulic motor.

The bleed-off circuit shown in Figure 12-26 is even less effective than the meter-in circuit. With bleed-off flow control, the motor inefficiency combines with the pump inefficiency to produce an even greater loss of rotational speed as the motor loads. The circuit shows a pressure-compensated flow control bypassing 7 gpm. The motor operates at 3 gpm and 100 psi. As the motor loads and pressure increases, internal leakage in the motor and pump results in a greater drop in speed than with the meter-in circuit.

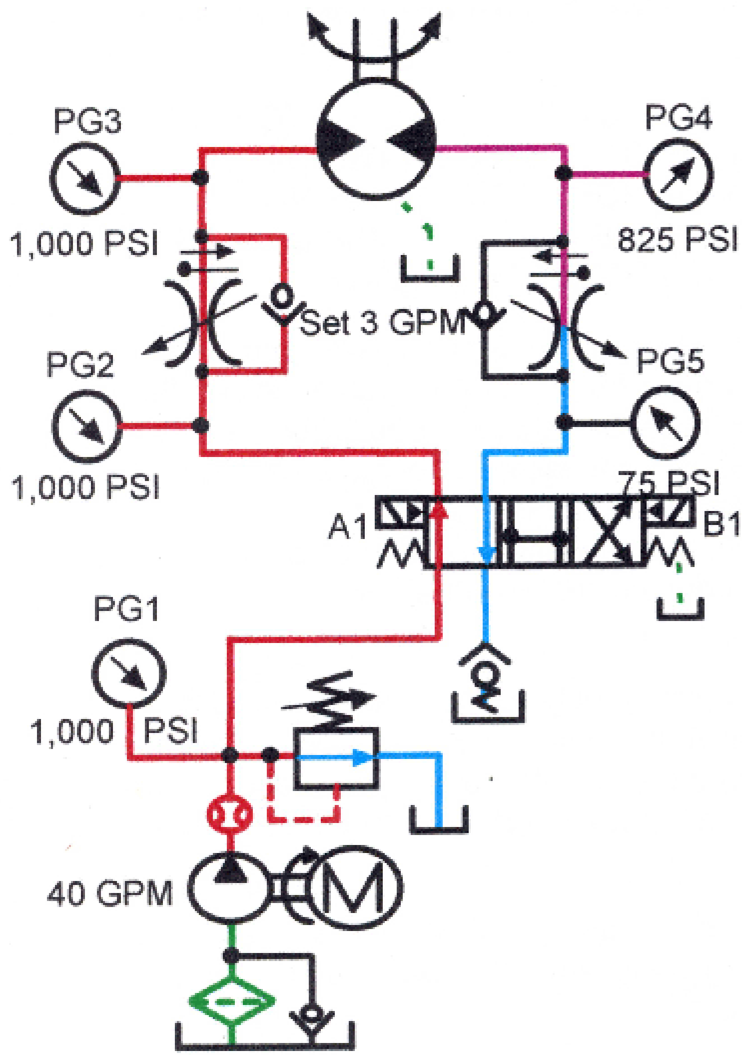


Fig. 12-27. Meter-out flow-control circuit with an external drain.

The meter-out circuit shown in Figure 12-27 provides the most accurate speed control. Note that heat generation is high, even with a pressure-compensated pump, but the resulting speed control is very accurate. Use pressure-compensated flow controls and motors with external case drains for the greatest accuracy and longest motor life. Oil enters the motor at low torque and the pressure drop across it is low. Even though high inlet pressure with high backpressure causes high internal leakage, pump input makes up for it. Accurately controlled fluid leaving the motor keeps speed constant because the pressure-compensated flow control maintains a steady flow. When the motor loads, pressure drop across it decreases. Internal leakage decreases but the amount of oil leaving the outlet port stays constant. As long as outlet flow does not change, speed stays the same.

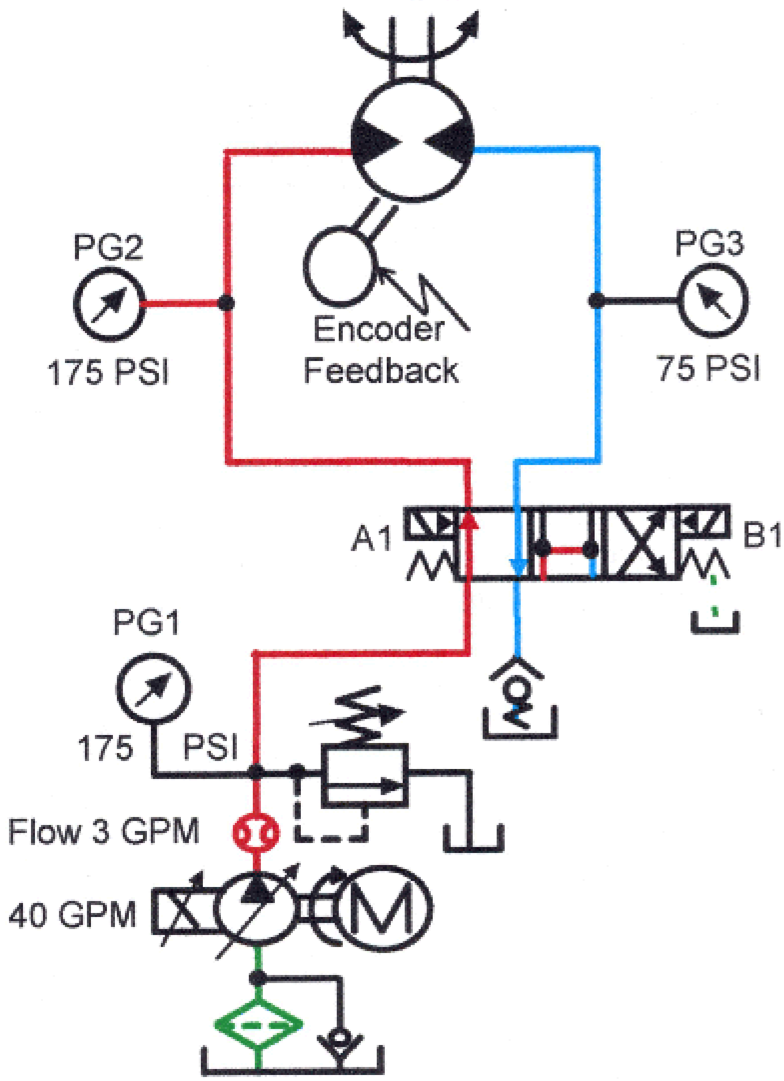


Fig. 12-28. Meter-in flow-control circuit with a servo-controlled variable-displacement pump.

The circuit in Figure 12-28 generates only minimum heat, but speed fluctuates with load changes when using a variable-displacement pump. The only energy loss is from the inefficiency of the pump, valve, and motor. As the load increases, pressure at the motor inlet climbs. The increased pressure causes greater slip at the pump and motor, so motor speed slows. As shown in the Figure, a servo-controlled pump with electronic feedback from the hydraulic motor eliminates the speed change problem and gives extremely accurate speed control.

Simple torque limiter for air motors

Some circuits reduce the inlet pressure to limit air motor torque on a device. Reducing inlet pressure causes the motor to stall when the workload exceeds its torque. Limiting torque with this method is not reliable or repeatable. Torque levels vary by as much as 10 to 20% according to how fast the motor turns, the mass involved, and the pressure required. The faster the motor turns and the heavier the part being turned, the more the torque overrides. Adjusting inlet pressure reduces motor torque but will not overcome the variations that weight and speed cause. Also, adjusting inlet pressure to a very low level makes motor performance erratic, with an ever-changing torque. Finally, at lower pressures, reduced speed may increase cycle time.

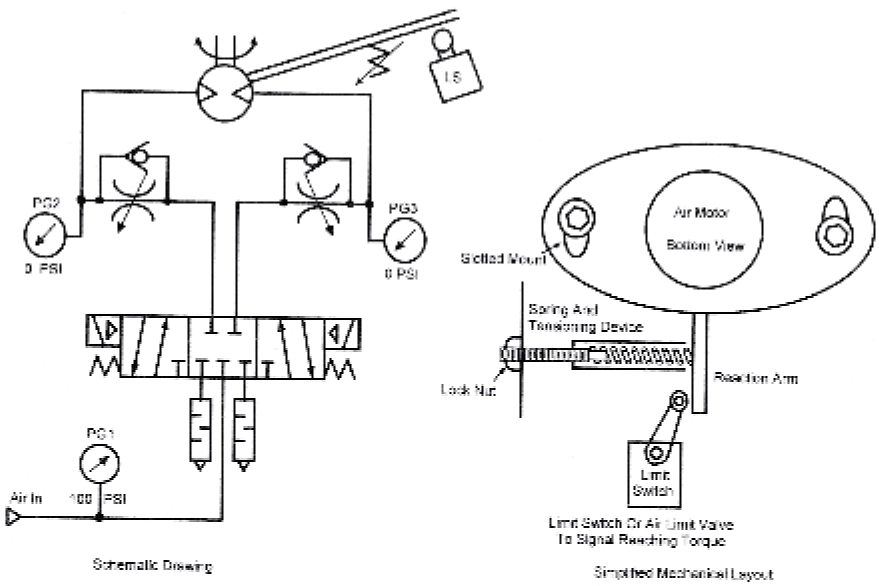


Fig. 12-29. Simple unidirectional torque limiter for an air motor.

To overcome these problems, try the diagrammed circuit and simplified mechanical layout shown in Figure 12-29. With this circuit, the motor operates at full torque, while meter-in flow controls set its maximum speed. An adjustable spring force holds the motor in position by opposing its torque reaction. As the motor turns to tighten a device, torque is low until near the end of the operation. During this part of the cycle, the adjustable spring resists motor torque, keeping its housing stationary. When motor torque increases to match the amount set by the spring tension, the housing starts to rotate. When torque reaches spring setting, the motor housing moves the reaction arm against the limit switch, shifting the directional valve to center and stopping the motor. This mechanism — designed with minimal friction or binding — gives reliable and repeatable results. Air limit valves or electric limit switches work equally well with this arrangement. Use air limits with snap action shifting for good repeatability.

Hydraulic motor driven by a bi-directional pump

Figure 12-30 shows a variable-displacement bi-directional pump driving a bi-directional hydraulic motor in a closed-loop system. This circuit is commonly known as a hydrostatic transmission. The bi-directional hydraulic pump can produce up to full flow from either port, eliminating the need for directional or flow control valves. When the pump strokes to produce flow from one port, the motor starts turning a given direction. With controlled pump stroking, the motor starts and accelerates smoothly. The amount of pump stroke controls motor rpm and pressure is just enough to make the motor turn. With high-efficiency components, this system generates little heat, eliminating the need for large reservoirs while it gives infinitely variable speed and torque.

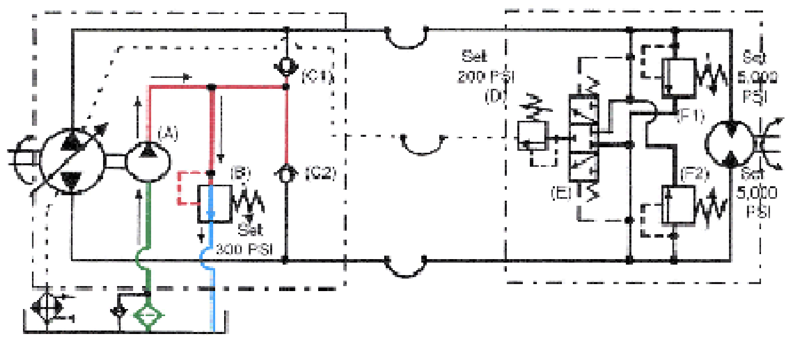


Fig. 12-30. Bidirectional hydraulic motor driven by a bidirectional pump (shown with motor at rest and pump running).

Due to internal leakage of the pump and motor, the circuit is not a true closed loop. Without leakage makeup, the circuit would quickly run dry, operate intermittently, and damage its parts. To keep the closed loop full, fixed-displacement charge pump A supplies flow through check valves C1 and C2 to the low-pressure side of the circuit. Charge oil keeps the pump inlet fully supplied, preventing cavitation. When the motor stops, excess oil from the charge pump goes to tank through relief valve B. Oil from the charge pump often operates the pump stroking mechanism, plus other circuits on the machine.

In Figure 12-31, the motor is running forward. Flow from the pump goes to the motor inlet, making it turn. Flow from the motor outlet returns to the opposite side of the pump. Check valve C1 stays closed (due to the working pressure) while check valve C2 opens to allow charge oil to make up for leakage and supply the closed-loop with cooled clean fluid. Shuttle valve E shifts when the motor runs, porting excess charge fluid through relief valve D, the motor case, the pump case, and back to tank. Charge flow sends spent oil through the motor and pump case to *cool them, then through a heat exchanger to remove excess heat. Flow goes through relief valve D because its setting is 100 psi less than relief valve B.*

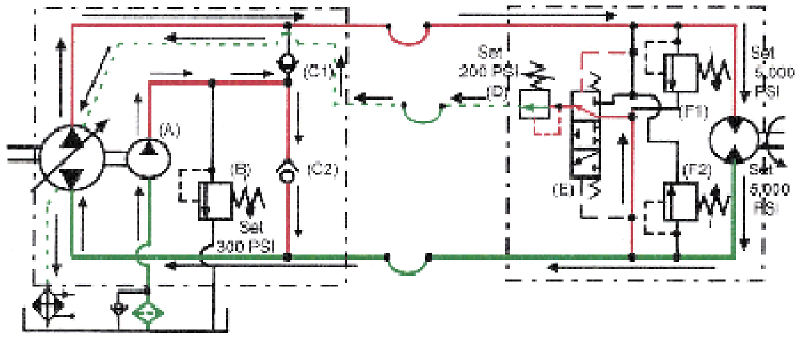


Fig. 12-31. Bidirectional hydraulic motor driven by a bidirectional pump (shown with motor running forward).

While the motor runs, speed is infinitely variable and system pressure fluctuates with any load changes. The only heat generation in this circuit comes from the charge pump going across the relief valve, plus inefficiencies in the pump, motor, and valves.

When the motor meets a load large enough to stall it, relief valve F1 opens as shown in Figure 12-32. This directs pump flow around the stalled motor, protecting the system from overpressure. If relief valve F1 stays open for any length of time, the wasted energy quickly over-heats the closed-loop piping.

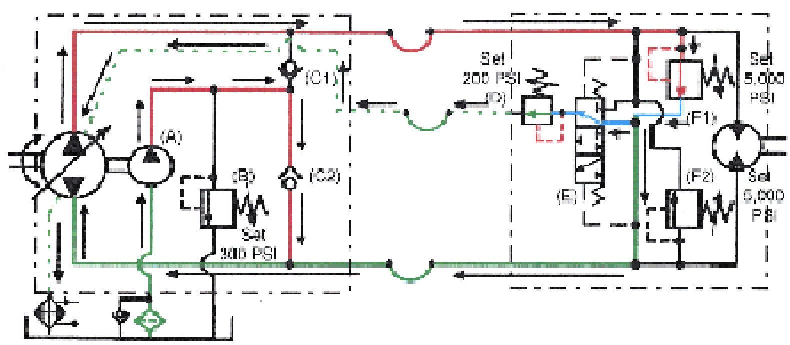


Fig. 12-32. Bidirectional hydraulic motor driven by a bidirectional pump (shown with motor stalled).

Relief valves F1 and F2 also protect the motor when the pump goes to zero stroke and an external force continues to drive the motor. When an external force drives a motor, the motor *becomes a pump. Pressure at its outlet climbs until it reaches 5000 psi. At 5000 psi, one of the relief valves opens, allowing oil to flow to the opposite motor port. This protects the motor and circuit from excess pressure and possible cavitation. While the externally driven motor bypasses, relief valve backpressure decelerates it, stopping the load quickly and smoothly.*

Closed-loop motor circuits give infinitely variable control of torque and speed with minimal shock and heat generation.

Pressure- or fatigue-testing machines often require high pressure for long periods of time. Other circuits might need a small volume of high-pressure fluid for a short period while most of the cycle only needs low pressure. Other machines can use air cylinders to manipulate a part but need very high pressure to perform one operation. Some manufacturers make high-pressure rotary pumps — rated up to approximately 10,000 psi — but these pumps are expensive and may heat the fluid. Another choice for low-volume/high-pressure circuits is an intensifier.

When a circuit calls for a small volume of high-pressure oil or air, consider using an intensifier — sometimes called a booster. Most cylinder manufacturers build air- or hydraulic-powered intensifiers. Or you can use off-the-shelf cylinder parts to assemble your own booster. Also, intensification is a natural function of single-rod cylinders and motor-type flow dividers.

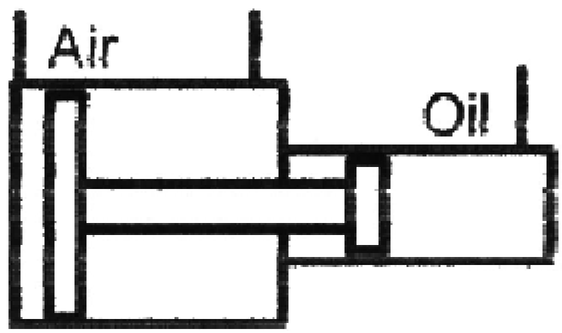


Fig. 13-1. Air-oil intensifier symbol.

Figure 13-1 pictures the symbol for an air-oil intensifier. While the symbol shows two pistons with different diameters, the actual intensifier consists of a piston pushing a rod. The large-area air piston pushes a small-area hydraulic ram against trapped oil. The difference between the two areas gives high-pressure capability at the small ram. This capability is indicated by the area ratio. If the air piston has a 5-in. diameter and the oil piston has a 1-in. diameter, the area ratio is 25:1. With this area ratio, 80 psi acting on the air piston produces 2000 psi at the hydraulic piston.

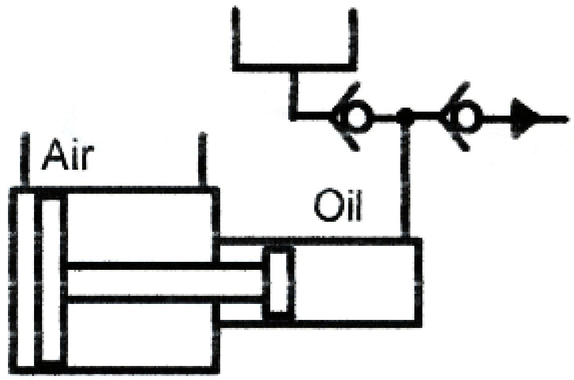


Fig. 13-2. Reciprocating air-oil intensifier.

Stroke length dictates the maximum volume of high-pressure fluid from an intensifier configured as in Figure 13-1. The booster in Figure 13-2 produces the same pressure but an unlimited volume. A reciprocating intensifier takes fluid from a reservoir and forces it into the circuit. In effect, the reciprocating intensifier is a single-piston pressure-compensated pump. The area ratio and air pressure determine the maximum hydraulic pressure. This pump is close to 100% efficient, so oil heating is not a problem. Intensifiers do not need relief valves because they stall at maximum pressure.

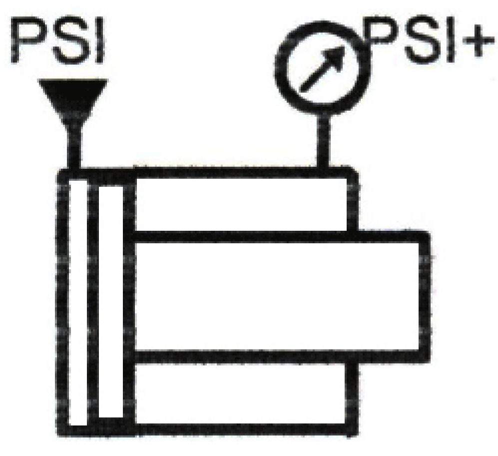


Fig. 13-3. Oversize-rod intensifier.

The oversize-rod cylinder shown in Figure 13-3 also is an intensifier. Any single-rod cylinder intensifies pressure with the rod end port blocked. The larger the rod diameter, the greater the *intensification. For low intensification — say 1.5 to 2 times system pressure — a single-rod cylinder is inexpensive and readily available.*

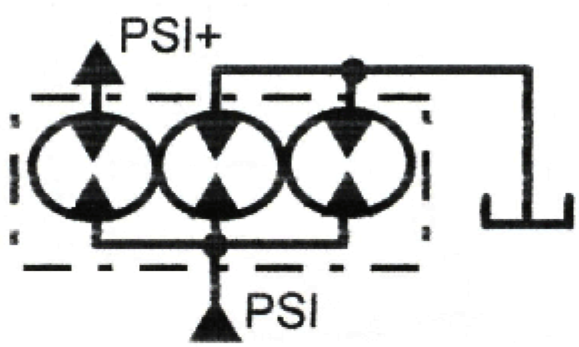


Fig. 13-4. Motor-type flow-divider/intensifier.

Figure 13-4 depicts the symbol for a motor-type flow divider used as an intensifier. This type intensifier produces a continuous flow of higher-pressure oil at a reduced flow rate. The reduced flow rate is the same ratio as the pressure increase. (A 2:1 intensifier reduces the flow by 50%.) A motor-type flow divider intensifier is less efficient than a piston-type intensifier and is not recommended for applications with long holding periods.

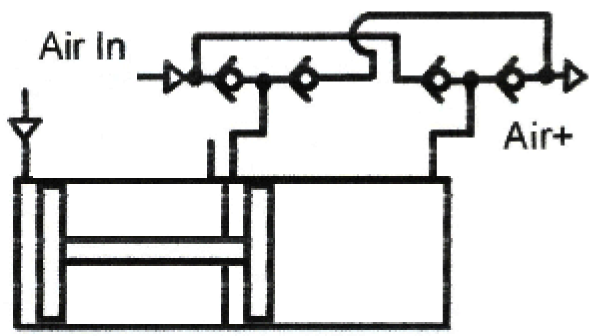


Fig. 13-5. Air-to-air intensifier.

Figure 13-5 shows the symbol for an air-to-air intensifier. These intensifiers produce small volumes of higher-pressure air from the plant air supply. Ratios up to 4:1 are common. Hydraulically driven designs with higher ratios are available from some manufacturers.

Intensifier circuit using standard cylinders

The schematic diagram in Figures 13-6 through 13-9 suggests how to use standard cylinders as an air-hydraulic intensifier. This is a quick way to get high ratio intensification for a rush job. A 6-in. bore air cylinder driving a 1.5-in. bore hydraulic cylinder gives an intensification ratio of 16:1. With 80-psi input air, hydraulic output pressure is approximately 1300 psi.

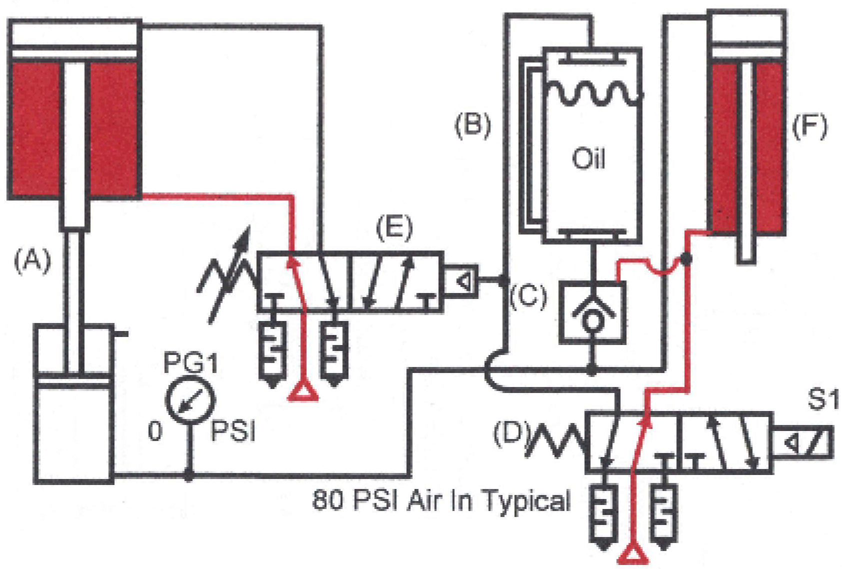


Fig. 13-6. Air-oil intensifier circuit using standard cylinders. System on and ready.

Mount the cylinders to a beam or machine member and pipe them as shown in the Figures 13-6. This circuit allows a hydraulic cylinder to operate at low pressure during extension and retraction, with a short high-pressure work stroke to clamp, punch, or do other work. The circuit includes shop-made intensifier A, air-oil tank B, air-pilot-operated hydraulic check valve C, solenoid-operated 5-way air valve D, sequence operated 5-way air valve E, and work cylinder F. With solenoid S1 deenergized, the cylinder and intensifier stay fully retracted, ready for a work stroke.

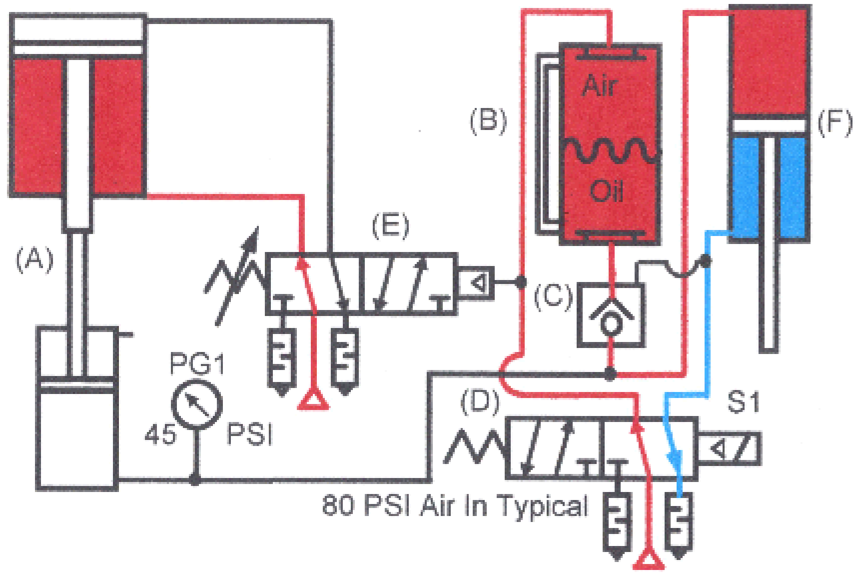


Fig. 13-7. Air-oil intensifier circuit using standard cylinders. Work cylinder advancing at low pressure.

Energizing solenoid S1 on valve D, as in Figure 13-7, directs air to air-oil tank B and exhausts the rod end of cylinder F. Oil from the air-oil tank free-flows through check valve C to extend the cylinder rapidly. Pressure in the line to the cylinder's cap end remains low as the cylinder moves toward the work, so sequence valve E stays in its normal position. The cylinder extends until it contacts the work.

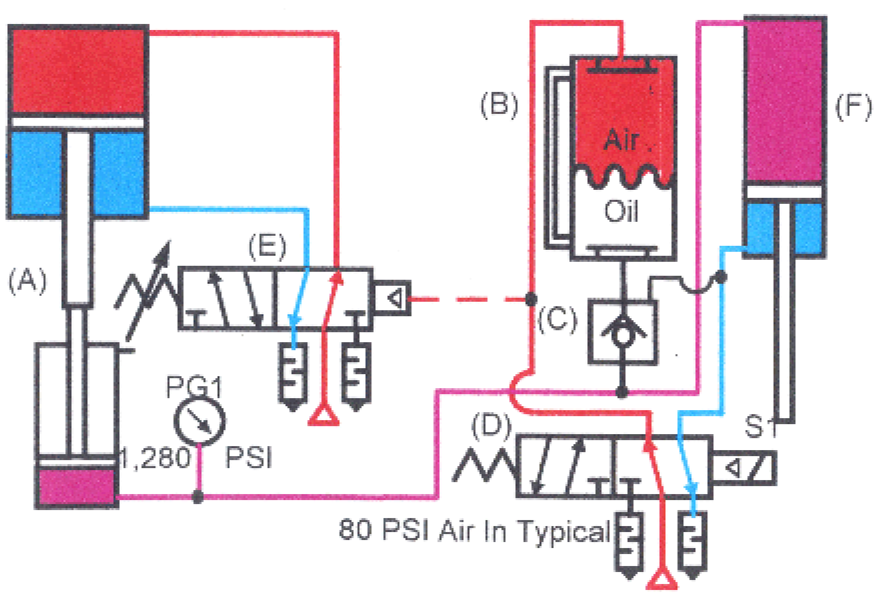


Fig. 13-8. Air-oil intensifier circuit using standard cylinders. Work cylinder holding at high pressure.

After the cylinder contacts the work, pressure in its cap-end port increases. Figure 13-8 shows the circuit condition after this pressure buildup shifts sequence valve E. When sequence valve Eshifts, air goes to the cap end of the 6-in. cylinder on intensifier Aand exhausts from its rod end. Cylinder Aextends to stroke the 1-1/2-in. hydraulic cylinder. This forces high-pressure oil to the cap end of work cylinder F. Check valve C is held closed by its spring to block high-pressure oil from going to air-oil tank B. Pressure in the cap end of cylinder F rises to approximately 1300 psi — and is available to power any high-force operation.

The intensifier's hydraulic cylinder must provide enough oil to move the work cylinder through its high-pressure stroke. A 3.25-in. bore work cylinder with a high-pressure work stroke of 0.75 in. requires a minimum 6.22 in.3 intensifier volume. Calculate volume by multiplying the area of the working cylinder by the length of the high-pressure work stroke. To figure the minimum intensifier stroke, divide the volume required for the work cylinder by the area of the intensifier. In this example, the minimum intensifier stroke is 3.5 in. To make sure there is always enough high-pressure oil to do the job, add 1.0 to 1.5 in. to the intensifier stroke to allow for oil compressibility, hose stretch, and possible future needs. Choose an intensifier stroke of at least 5 in. for this application.

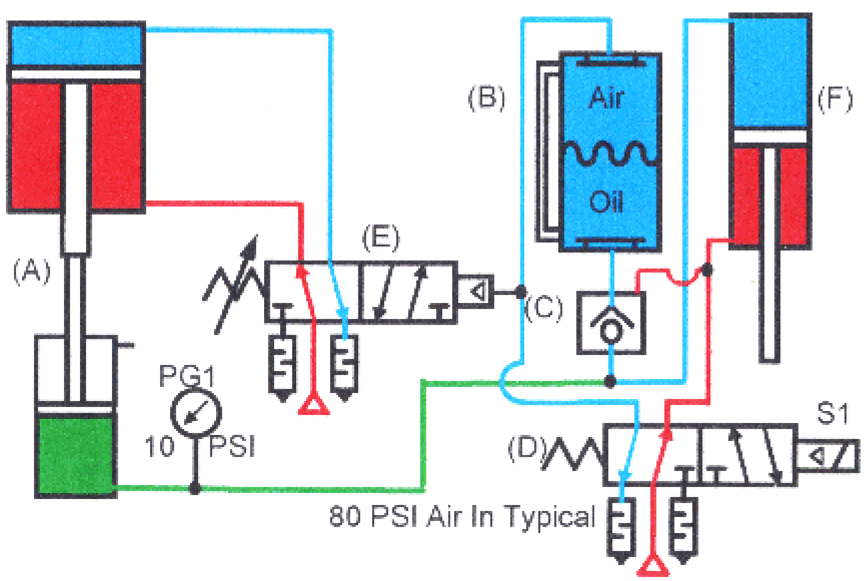


Fig. 13-9. Air-oil intensifier circuit using standard cylinders. Work

cylinder retracting at low pressure.

Deenergizing solenoid S1 on valve D, Figure 13-9, directs air to the rod end of cylinder F and to the pilot port of air-pilot-operated check valve C. Check valve C opens, providing oil from the cap end of cylinder F with a free path to tank. Pilot pressure to sequence valve E drops when valve D shifts. When sequence valve E returns to its normal position, intensifier A retracts and fills the intensifier cylinder with oil for the next cycle.

Notice that as cylinder F retracts, only 80-psi air pressure drives it. There is ample hydraulic pressure to extend the cylinder for the high-force work stroke, but only air pressure to retract it. If a higher retracting force is needed (to disengage tooling or for other reasons), external help or other circuit changes may be necessary.

Adjust hydraulic pressure to the cylinder with a regulator in the air line connected to sequence valve E. With a regulator to adjust the air pressure, changing hydraulic force is simple.

Hydraulic cylinder F should have resilient seals that keep oil from leaking to the air side or air to the oil side. Some circuits use two air-oil tanks on cylinder F to prevent aeration of the oil. (Chapter 3 has information about sizing and hooking up air-oil tanks.)

Three-head intensifier circuit with tandem cylinder

Several manufacturers produce 3-head intensifiers that eliminate external pilot-operated check valves. The first head on a 3-head intensifier has an air seal on its rod facing the air side and a hydraulic seal facing the oil side. The second head has an oil port into the rod chamber and a resilient seal facing the third head. The third head has a welded-on oil chamber that the piston rod enters. When the piston rod advances, it displaces oil from this chamber to create high pressure. The ratio of the air-piston area to the rod area intensifies the pressure by up to 40:1, or even higher. A standard 5-in. bore air cylinder with a 1-in. diameter piston rod produces 25:1 intensification. (This is a standard size for several manufacturers.) Three-head intensifiers supply a small volume of oil for short high-pressure work strokes. Calculate the high-pressure oil volume by multiplying the rod area by the stroke length after the rod passes the seal between the second and third head.

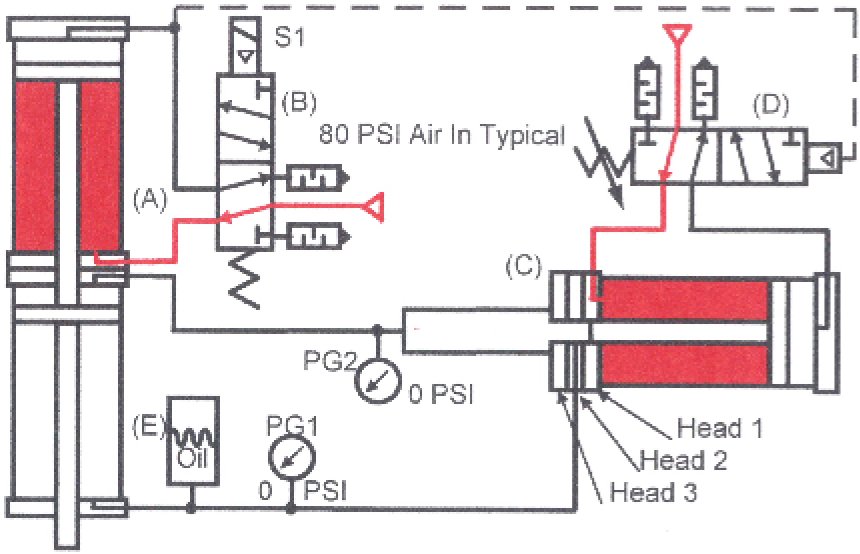


Fig. 13-10. Tandem-cylinder air-oil intensifier circuit with typical 3-head intensifier. System is on and ready.

Figure 13-10 shows how 3-head intensifier C pressurizes air-oil tandem cylinder A. This circuit provides rapid low-force advance and retract strokes, with a short high-force work stroke when the cylinder meets resistance. Solenoid-operated directional valve B extends the air part of the tandem cylinder. Sequence valve D operates intensifier C. Sealed expansion tank Ereceives oil from the tandem cylinder while it extends at high pressure. For an expansion tank, mount an air filter with a clear bowl upside down, and remove the filter element. The transparent bowl makes it easy to check oil levels. This circuit eliminates air-oil tanks to make the system more compact. Figure 13-10 shows the circuit at rest.

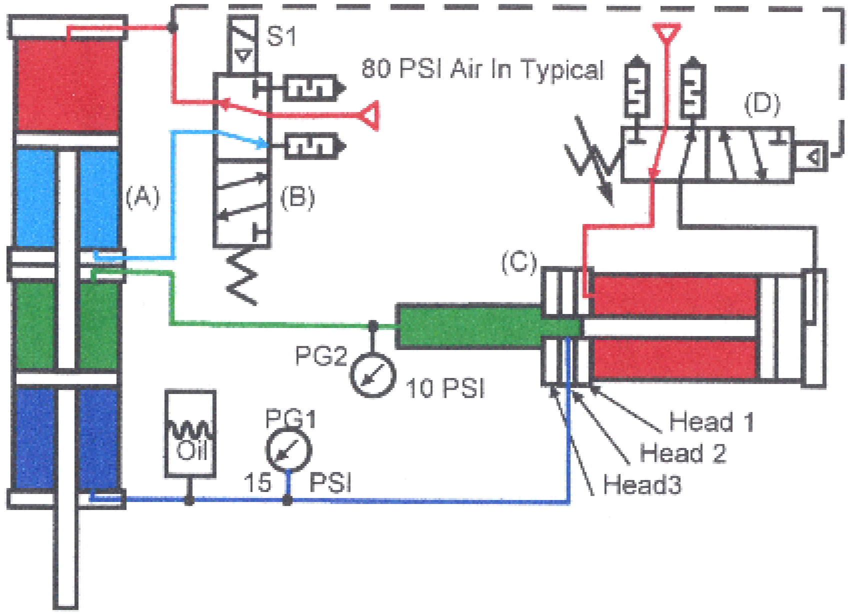


Fig. 13-11. Tandem-cylinder air-oil intensifier circuit with typical 3-head intensifier. Tandem cylinder is advancing rapidly.

Shifting solenoid S1 on valve B, as in Figure 13-11, makes the air-oil tandem cylinder advance rapidly to the work. Oil in the double rod-end cylinder transfers from front to back through the center head of intensifier C. (Keep these transfer lines short with oil velocity below 4 fps to minimize pressure drop.) As the cylinder advances, pressure at the cap port stays low. Adjust the spring on sequence valve D to cycle the intensifier after the tandem cylinder contacts the work.

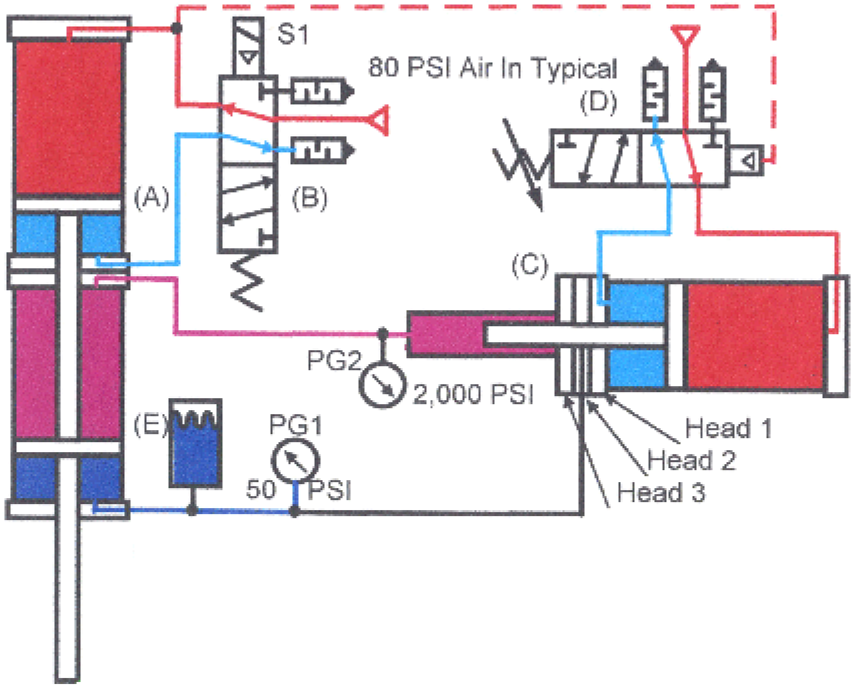


Fig. 13-12. Tandem-cylinder air-oil intensifier circuit with typical 3-head intensifier. High pressure in tandem cylinder.

At this contact, sequence valve D shifts to start intensifier C stroking forward, as in Figure 13-12. When the intensifier rod passes through the seal between heads 2 and 3, pressure intensification begins on the back of the double rod-end cylinder. As the tandem cylinder extends, trapped oil from the front chamber goes into expansion tank E. (Pressure increases slightly in the tank because the air trapped above the oil is compressed.) Use a tank with three to four times the volume displaced by the cylinder during the high-pressure work stroke. As the intensifier continues to stroke, increased pressure performs the work.

It is important that the intensifier contains enough oil to move the tandem cylinder through its high-pressure stroke. If the double-rod cylinder has a 3.25-in. bore with a 1.375-in. rod, and the high-pressure stroke is 0.375 in., then a minimum of 2.55 in.3 of oil is needed. Add considerations for oil compressibility plus line and cylinder tube expansion to the cylinder high-pressure stroke volume. Remember: line expansion is greater when using flexible hose. Determine the volume of oil in the high-pressure portion of the piping and cylinder, and then increase this volume by 0.5% per thousand psi of pressure. Often it requires 0.5 to 1.5 in.3 of oil at 2000 psi to make up for oil compressibility. Calculate oil compressibility and add it to the stroke volume so the intensifier does not bottom out before the oil reaches the desired high *pressure. On most 3-head intensifiers, add 2.0 in. to the stroke required for volume to make up for oil that bypasses the rod before it enters the high-pressure seal between head 2 and head 3.*

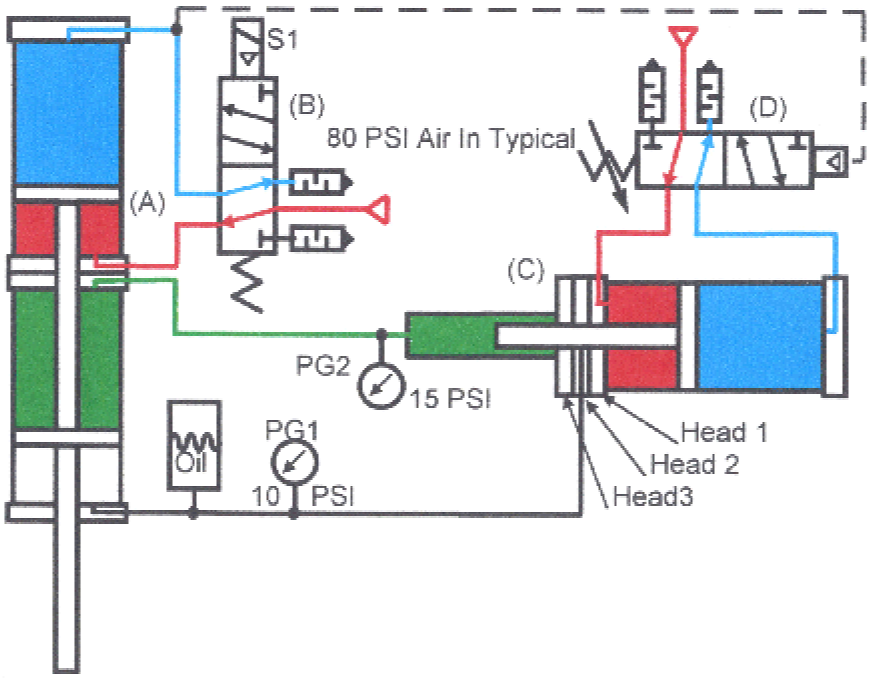


Fig. 13-13. Tandem-cylinder air-oil intensifier circuit with typical 3-head intensifier. Tandem cylinder is retracting rapidly.

Figure 13-13 shows the intensifier and cylinder retracting. Deenergizing solenoid S1 on valve B lets sequence valve D spring-return to its normal condition. The intensifier starts retracting at high speed, but the tandem cylinder moves slowly. When the intensifier passes the high-pressure seal between heads 2 and 3, the tandem cylinder quickly returns to its home position.

Reciprocating intensifier for increased volume

A single-stroke intensifier produces a limited volume of high-pressure fluid. Pressure stops building when a single-stroke intensifier reaches the end of its stroke. If cylinder seals or piping leak, a single-stroke intensifier may build pressure, but then quickly lose it. When a circuit needs unlimited high-pressure volume at low flow, use a reciprocating intensifier.

Figures 13-14 through 17 show a reciprocating intensifier powering a cylinder that must hold clamping pressure for days. Reciprocating intensifier A, air-oil tank B, pilot-operated check D, solenoid valve E, and sequence valve F advance cylinder C to the work rapidly. This arrangement can hold as much as 3200 psi for long periods without wasting energy or generating heat.

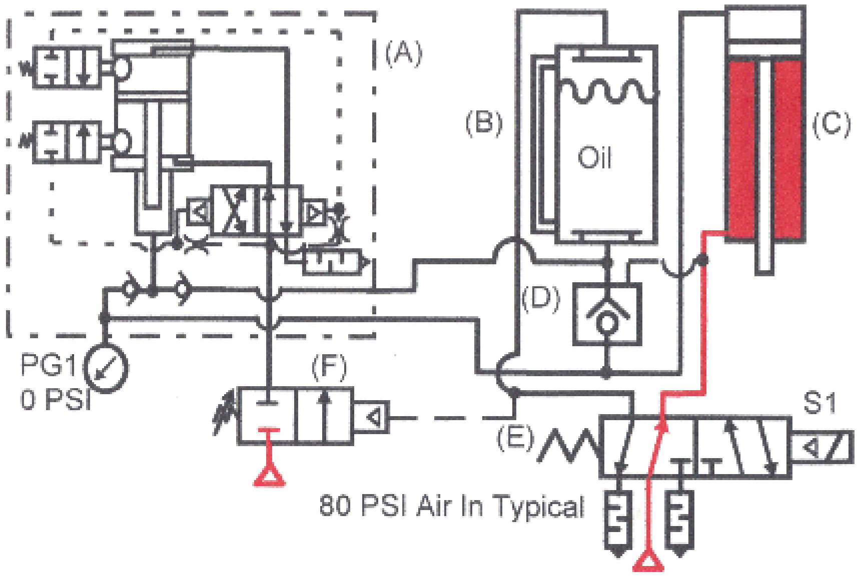


Fig. 13-14. Air-oil intensifier circuit with purchased reciprocating intensifier. System is on and ready.

Several companies assemble reciprocating intensifiers with a directional valve, limit valves, and check valves in a unit. Special-order units may come with air-oil tanks, special valves, or accumulators — all pre-piped for operation. When supplied with pressurized air, the unit in Figure 13-14 cycles and pumps oil until it reaches a maximum pressure. Other units operate from a pilot signal whenever the machine requires intensified pressure. For even higher pressures, dual or triple air pistons give higher ratios. Double-acting intensifiers increase oil volume while using less air. Most manufacturers offer single-acting intensifiers as standard and double-acting intensifiers as an option. When a machine needs a low to medium volume of high-pressure oil and has long holding times, use a reciprocating intensifier.

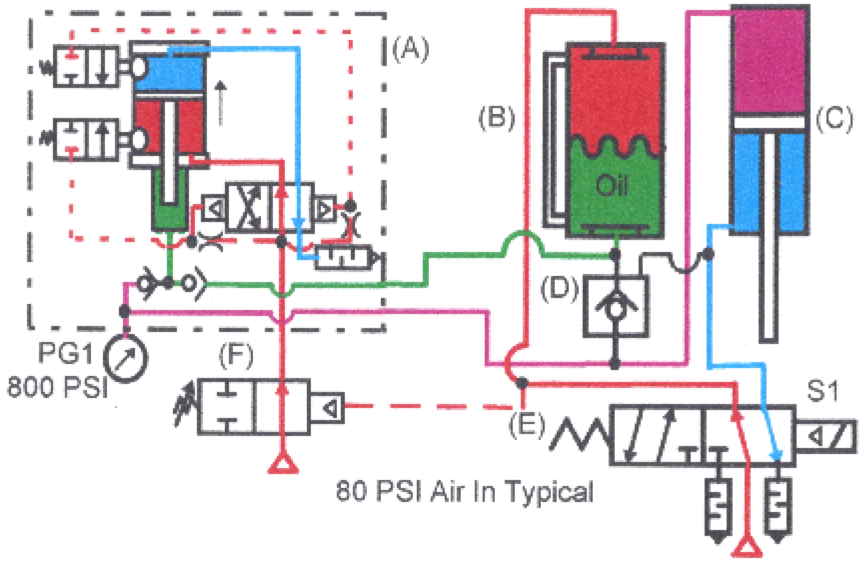


Fig. 13-15. Air-oil intensifier circuit with purchased reciprocating intensifier. Intensifier is filling with oil.

The circuit changes as shown in Figure 13-15 after the cylinder contacts the work. The intensifier starts cycling because pressure buildup shifts sequence valve F to open. Pilot-operated check valve D closes, blocking pressure fluid from going to tank. Pressure in cylinder C is already at 800 psi. As the intensifier retracts, suction draws oil in through the right-hand check valve to fill the oil chamber. Its spring and the pressure already in the work cylinder hold the left-hand check valve closed. A reciprocating intensifier is a low-volume, single-piston, pressure-compensated pump that continues to move fluid until it reaches maximum pressure. Because output from the intensifier is intermittent, cylinder movement is jerky, as is the rate of pressure increase.

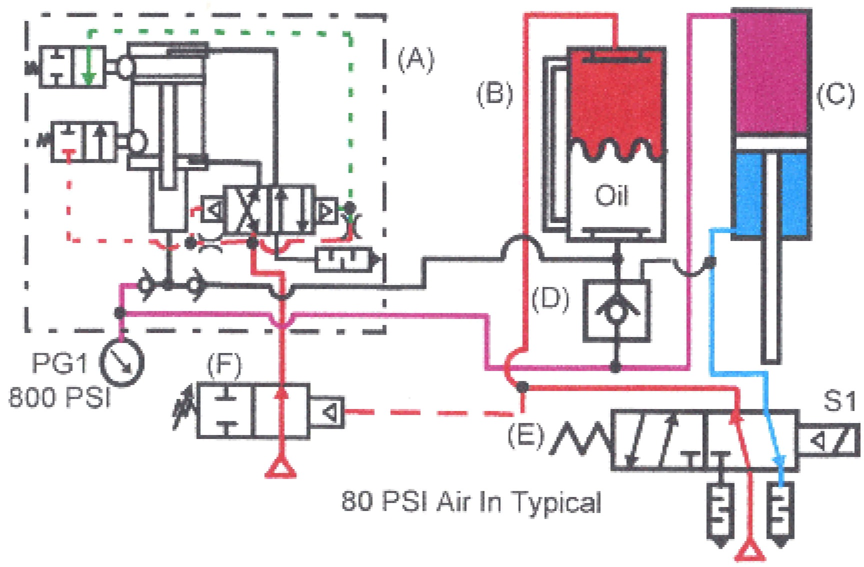


Fig. 13-16. Air-oil intensifier circuit with purchased reciprocating intensifier. Intensifier is reversing direction.

Figure 13-16 shows the intensifier changing from filling mode to pumping mode. The reciprocating air piston depresses the upper cam valve, reducing pressure on the right end of the double-bleed valve and causing it to shift. Both check valves close at this time, trapping oil in the cylinder. The intensifier now starts its pumping stroke.

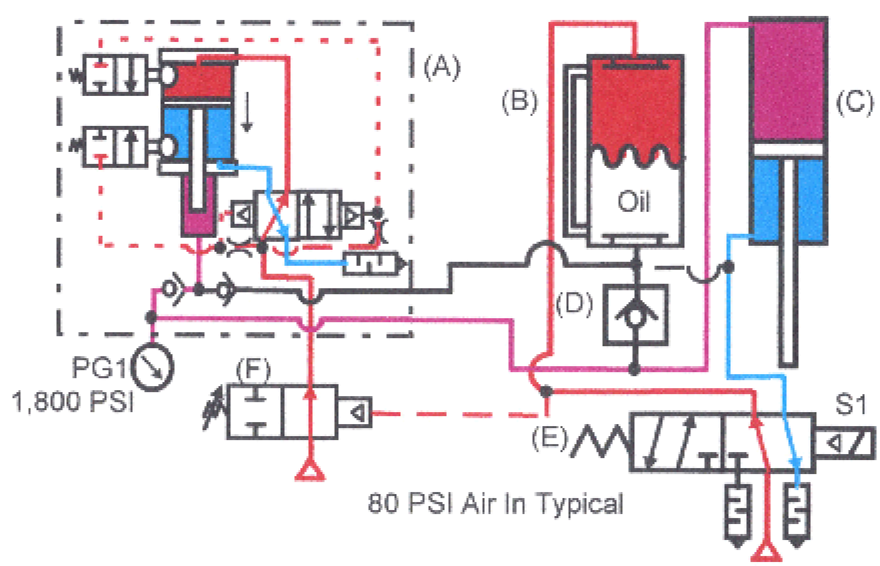


Fig. 13-17. Air-oil intensifier circuit with purchased reciprocating

intensifier. Intensifier is filling work cylinder with high-pressure oil.

The intensifier is extending and discharging fluid through the left check valve to the actuator in Figure 13-17. Fluid fills the actuator and pressure increases. The intensifier continues to reciprocate until it reaches maximum pressure. At maximum pressure, the intensifier stalls but continues to make up for internal or external leakage.

With the addition of an accumulator, a reciprocating intensifier could supply cylinders or motors that operate intermittently. The accumulator stores oil during machine idle time, and then discharges it at high flow without pulses for short periods. Use flow controls to slow the rapid uncontrolled movements likely to occur when using an accumulator.

Oversize-rod cylinder as an intensifier

There are times when the operating pressure of a hydraulic system is too low to produce enough force on a cylinder. The pump's rated pressure may be inadequate or the electric motor has too little horsepower for the higher pressure. Also, other actuators in the system may not be able stand higher pressure. One answer to this problem is a hydraulic cylinder piped as an intensifier.

When a single-rod cylinder extends, pressure in the rod end intensifies if there is any resistance to flow out of it. Resistance could be from a flow control, counterbalance valve, or simply a restriction. The amount of intensification depends on the area differential of the cap end to the rod end of the cylinder. A typical 4.0-in. bore cylinder with a 2.5-in. oversize rod is sold as a 2:1 ratio. All standard interchangeable cylinders use standard bore and rod sizes that are close to but not greater than a 2:1 ratio. The 4.0-in. bore, 2.5-in. rod combination actually has 1.64:1 area differential. With the rod-end port blocked, a 1.64:1-ratio cylinder produces 1640 psi at the rod end if the cap-end pressure is 1000 psi. This intensified fluid might cause problems in a typical circuit, but could supply a small volume of higher-pressure oil for a short, high-force work stroke from a cylinder.

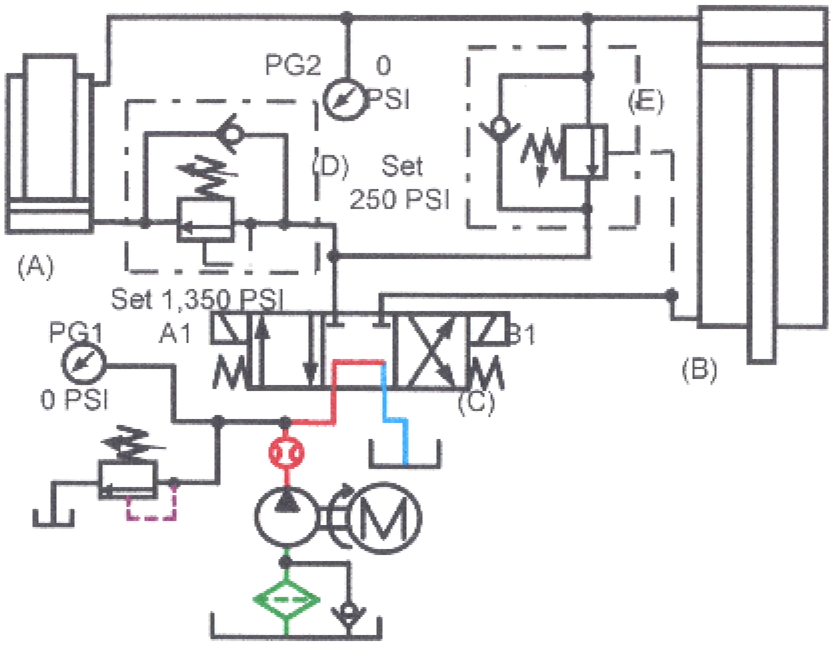


Fig. 13-18. 2:1 rod cylinder serving as an intensifier. At rest with pump running.

The volume of oil entering and leaving the intensifier cylinder has the same ratio as the intensification. In the 1.64:1 example above, with a cylinder cap-end flow of 10 gpm, pressure intensified flow from the rod end is 6.1 gpm. The larger the cylinder rod, the higher the intensified pressure — and the lower the flow.

Figure 13-18 shows a schematic diagram of an oversize-rod cylinder used as an intensifier. Intensifier cylinder A has 5.0-in. bore with a 3.5-in. diameter rod. The area of the cap side is 19.64 in.2 and the rod annulus area is 10.01 in.2, giving a ratio of 1.96:1. Every 100 psi in the cap end produces 196 psi in the rod end. Also, 10 gpm entering the cap end pushes 5.1 gpm from the rod end. Stroke length of intensifier cylinder A must give enough volume to move work cylinder B through its high-pressure work stroke. If cylinder B has a 10.0-in. bore and a 0.5-in. stroke, the required volume is approximately 40 in. 3. Dividing the 40-in.3 work-stroke volume by a 10-in.3 intensifier volume indicates that a minimum stroke of 4 in. is needed from cylinder A. To allow for oil compressibility and leakage, specify an intensifier stroke of 6 to 8 in.

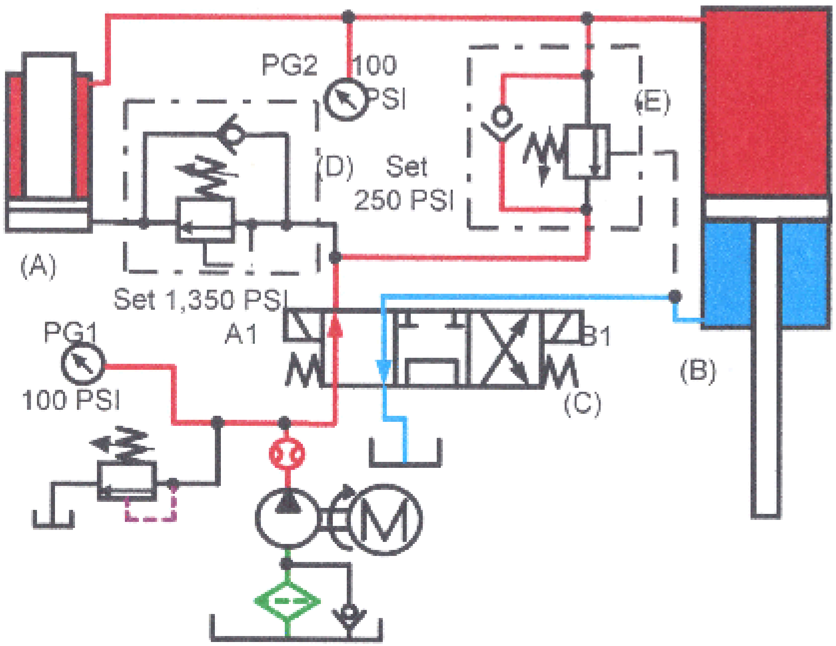


Fig. 13-19. 2:1 rod cylinder serving as an intensifier. Work cylinder is extending rapidly.

The cycle is automatic because sequence valves D and E control extension and retraction of the intensifier. Cycle time is slightly slower than the original low-force circuit.

Figure 13-19 shows solenoid A1 on directional valve C energized. Fluid flows directly to work cylinder B through the free-flow check on sequence valve E. Work cylinder B advances rapidly toward the work at low pressure.

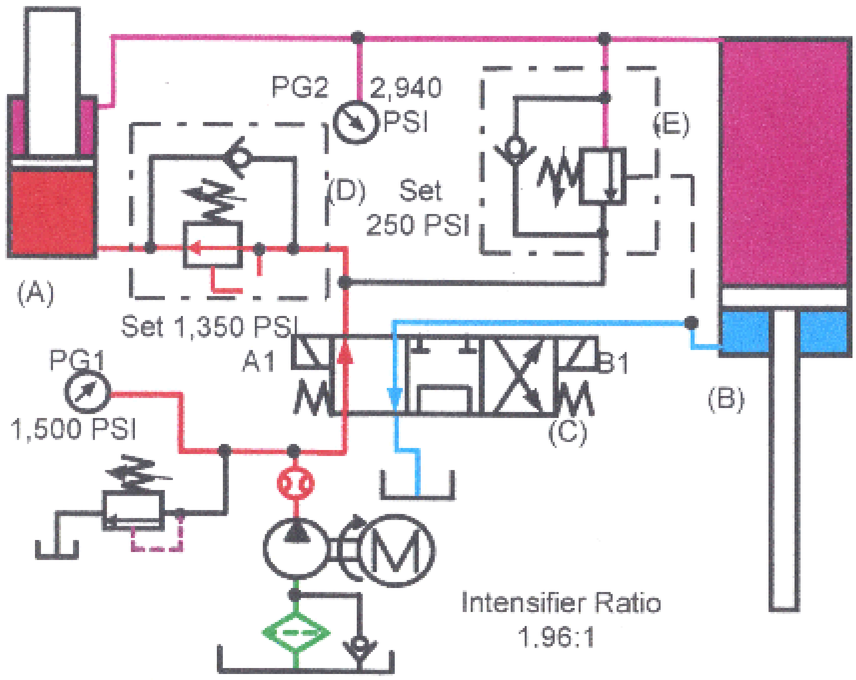


Fig. 13-20. 2:1 rod cylinder serving as an intensifier. Work cylinder is extending under high pressure.

At work contact, pressure builds to the setting of sequence valve D, Figure 13-20. Intensifier cylinder A extends and pressurizes oil in the cap end of work cylinder B to approximately twice system pressure. Before the intensifier bottoms out, it must give enough volume to complete cylinder B's work stroke. For long holding cycles, calculate valve and cylinder leakage, then add extra intensifier stroke so pressure holds.

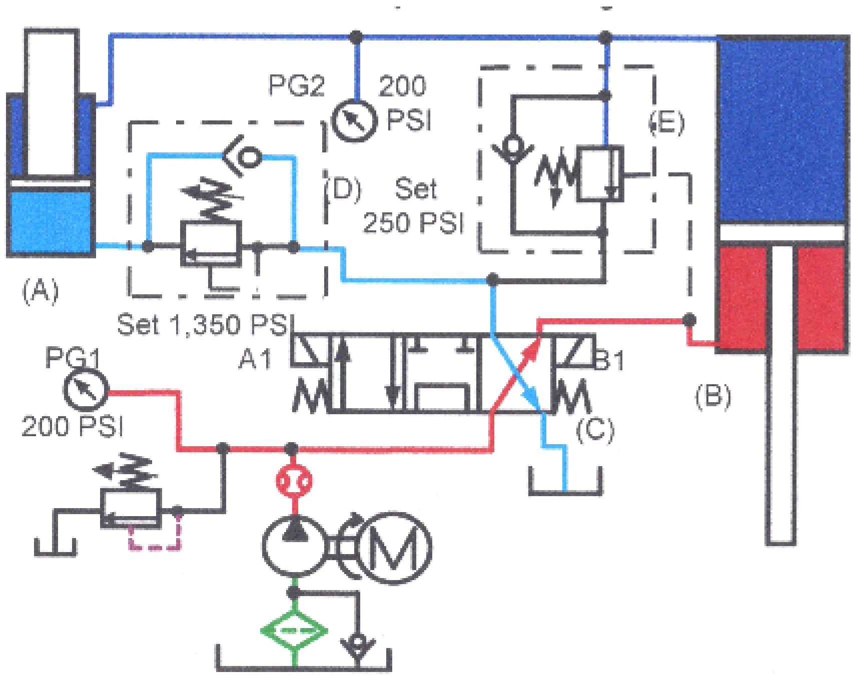


Fig. 13-21. 2:1 rod cylinder serving as an intensifier. Both cylinders retracting.

To retract work cylinder B, energize solenoid B1 to direct oil to its rod end, as in Figure 13-21. As cylinder B retracts, sequence valve E forces oil from its cap end to retract intensifier cylinder A. This saves pump fluid and retracts the intensifier within normal cycle time. When intensifier cylinder B retracts fully, external pilot-operated sequence valve E opens and the remainder of the oil in the work cylinder cap end goes to tank. The only added cycle time is while the intensifier boosts pressure in the work cylinder.

Motor-type flow divider as an intensifier

A motor-type flow divider intensifies pressure at one outlet when the other outlet is at a lower or no pressure. In the case of a 2-outlet motor flow divider with equal displacements, when inlet pressure is 1000 psi, one outlet can be at 2000 psi while the other outlet is 0 psi. While pressure doubles, flow from the intensified outlet is one half that at the inlet. The energy from the zero outlet motor transfers to the other motor to produce intensified pressure.

With more than one section going to tank, say from a 4-outlet divider with three outlets to tank, intensification would be four times. While the intensified fluid is four times inlet pressure, volume is only one-fourth inlet flow.

Using motor dividers with unequal sections is another way to get high intensification. If the motor in one section discharges 3 gpm to tank and the other section sends 1 gpm, intensification is still 4:1.

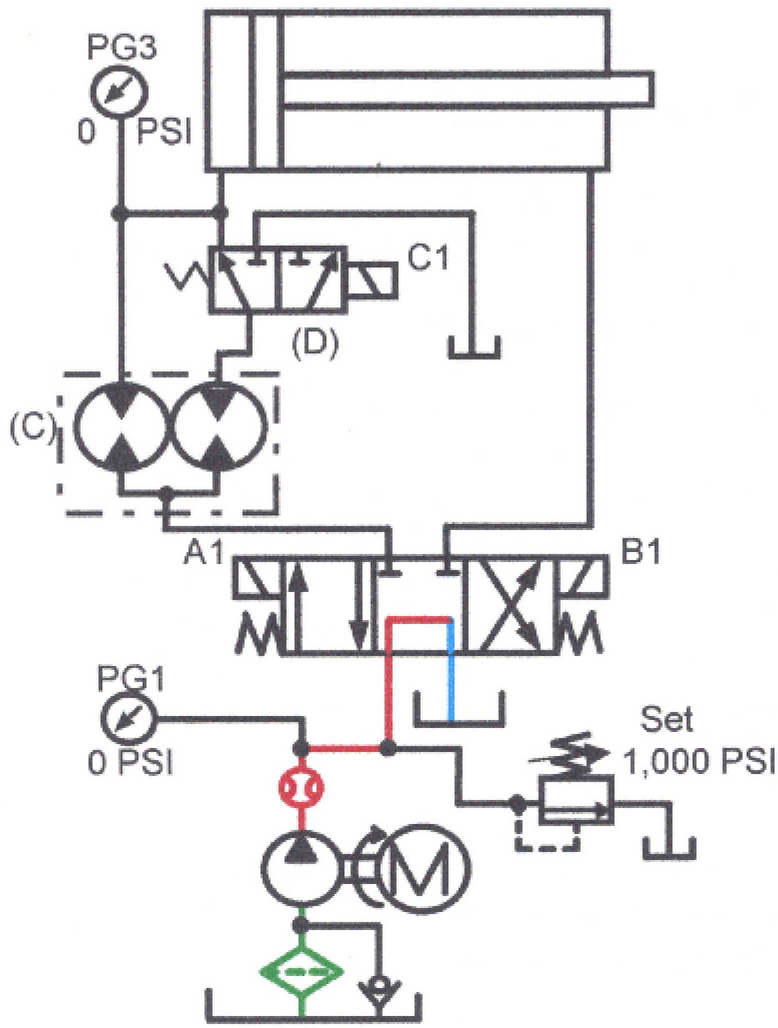


Fig. 13-22. Motor-type flow divider used as an intensifier. At rest with pump running.

Figures 13-22 through 25 show how to use this feature of motor-type flow dividers in a circuit. This circuit has equal flow divider C and 3-way directional valve D in the cylinder cap end line. In the at-rest condition, both outlets of the flow divider connect to the cap-end port.

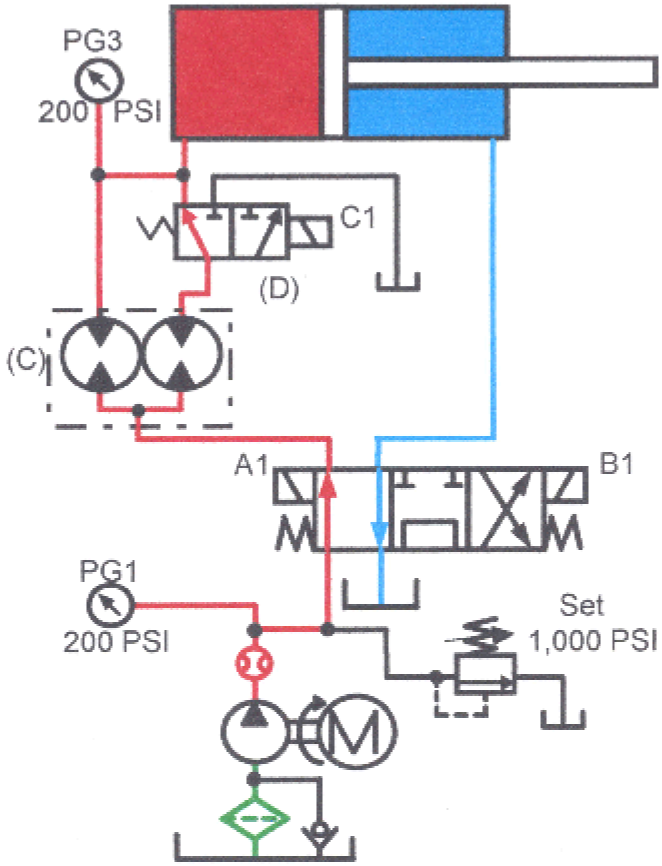


Fig. 13-23. Motor-type flow divider used as an intensifier. Cylinder extending at full speed.

In Figure 13-23, the cylinder is extending at full speed and low thrust. Shifting solenoid A1 of the directional valve ports oil through one side of the divider and 3-way valve to the rod-end port. Fluid from the other side of the divider goes directly to the cylinder rod-end port. Size the pump and valves to provide enough flow for the speed required in the fast-forward portion of the cycle. Normally, motor horsepower is low for a cylinder moving a light load.

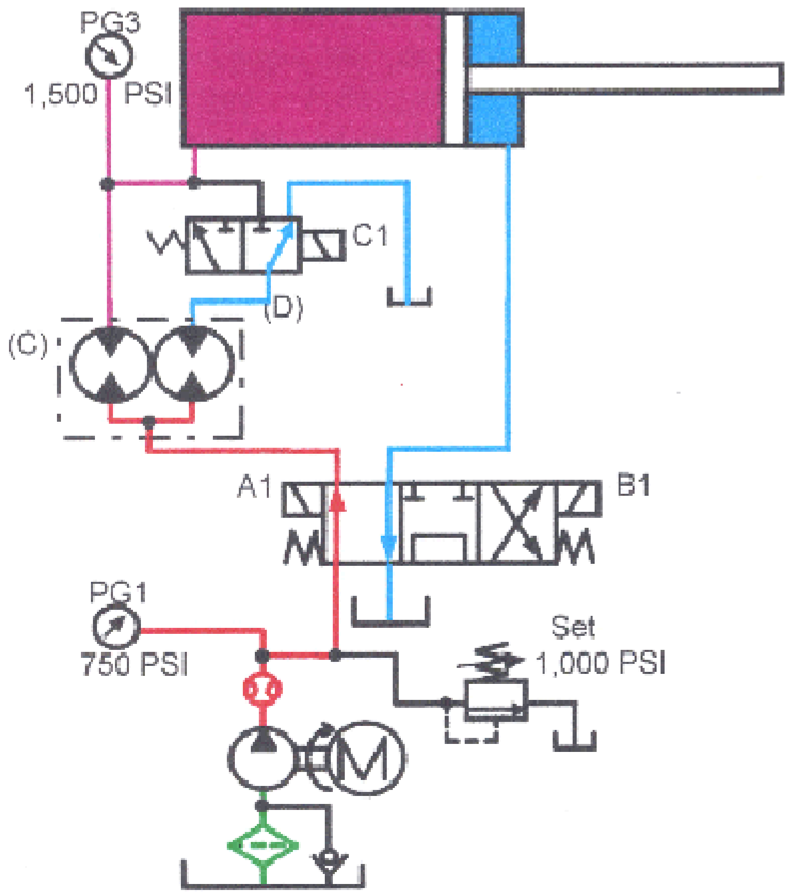


Fig. 13-24. Motor-type flow divider used as an intensifier.
Cylinder extending at full power.

When the cylinder makes a limit switch, as in Figure 13-24, it energizes solenoid C1 on the 3-way valve. When the valve shifts, oil from one section of the motor flow divider goes to tank. Pressure doubles, while cylinder speed drops to half what it was before energizing solenoid C1.

This circuit works best on actuators that do not stall. Using this setup for a fast advance and clamping operation might result in excess heat from internal leakage in the flow divider during the clamping part of the cycle.

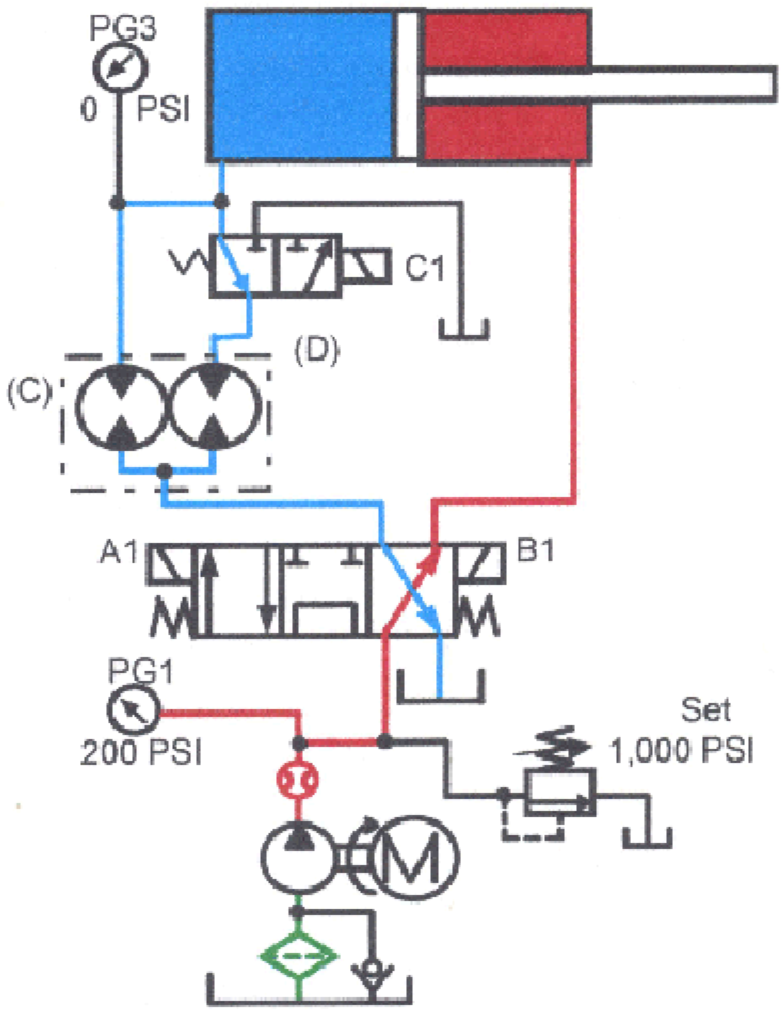


Fig. 13-25. Motor-type flow divider used as an intensifier. Cylinder retracting.

Energizing solenoid B1, Figure 13-25, makes the cylinder retract. Oil from the cap-end port flows through both sections of the flow divider, then back to tank through the directional valve.

When using a motor-type flow divider as an intensifier, make sure it is capable of operating at the elevated pressure. Pressure rating of an inexpensive gear motor flow dividers may be only 2000 psi intermittent and 1500 psi continuous. Some gerotor flow dividers go as high as 4500 psi intermittent and 3000 psi continuous.

Special air-oil intensifier cylinder

Some manufacturers build self-contained, air-driven, high-force hydraulic cylinders. These units look like a very long stroke air cylinder. Typically, they have 2 to 10 in. total strokes with 1.0- to 1.5-in. high-force strokes. They often replace a hydraulic unit on a machine that needs high tonnage for one operation on an otherwise air-powered circuit. Because these special intensifiers are self-contained, they only require an air supply and a signal to start them. They have sealed reservoirs so they operate in any position. They normally have an indicator to monitor oil volume for preventive maintenance. According to bore size and stroke length, cycle rates go as high as 150 per minute. The bigger the bore and longer the stroke, the fewer the cycles per minute.

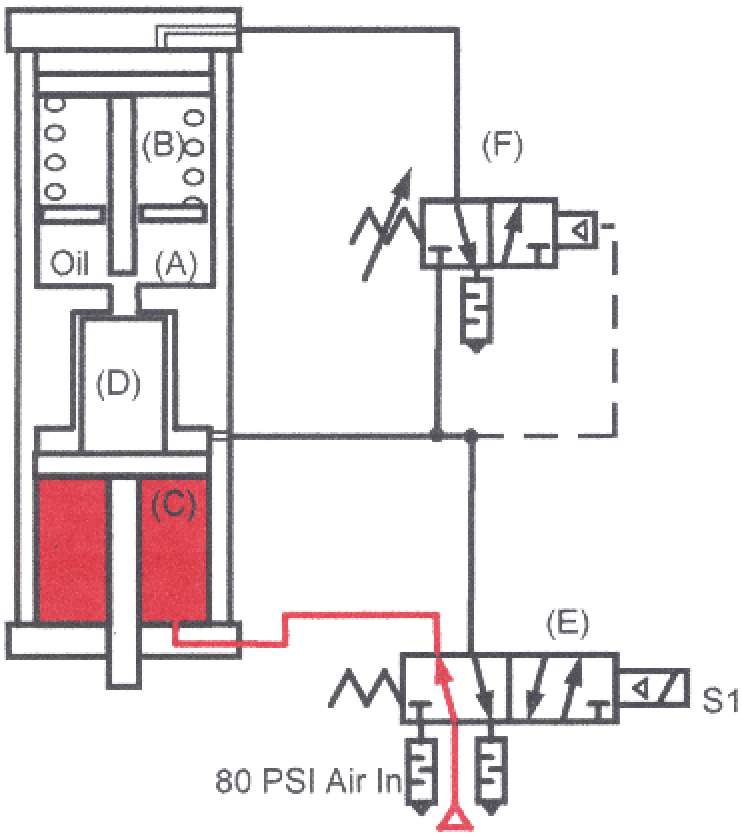


Fig. 13-26. Special air-oil intensifier cylinder. System on and ready.

As with other air-oil devices, return power is only cylinder net rod-end area multiplied by air pressure. The unit may have 50 tons to punch a hole but only 0.5 ton to retract the punch. For high retraction force use springs or urethane strippers, or add short-stroke return cylinders.

Figure 13-26 has a cutaway view of the intensifier cylinder at rest. (This view only shows function, not necessarily an actual assembly.) Air piston and rod C with attached hydraulic ram D move rapidly at low force to the work and return the tooling at the end of the cycle. Ram D is the area that intensified oil pushes on to get the short, high-force work stroke. Spring-loaded, floating piston A forms the top of a variable-volume, sealed oil tank. Spring-return air piston B drives its piston rod into trapped oil to intensify pressure for the work stroke. Directional control valve E cycles the advance and return strokes of cylinder C, and supplies air to pilot sequence valve F, starting the high-pressure work stroke.

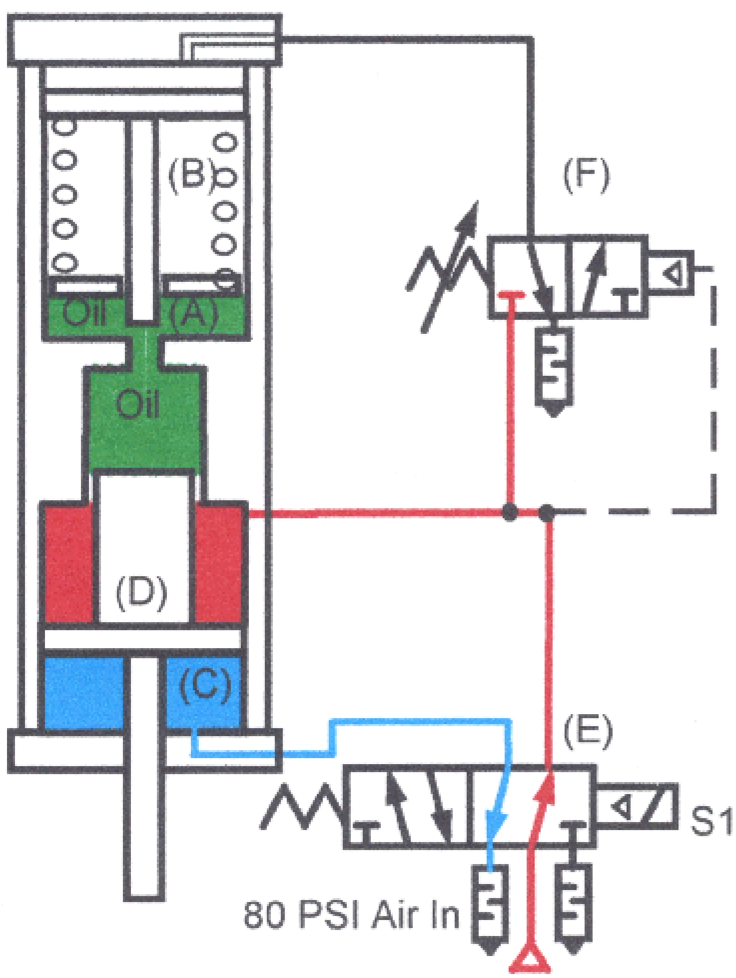


Fig. 13-27. Special air-oil intensifier cylinder. Fast advance at low force.

When directional valve Eshifts, as in Figure 13-27, air piston-and-rod C extends the tooling to the work rapidly. As the piston-and-rod extend, ram D advances and fills with oil from the variable-volume tank. Vacuum forms in the chamber behind ram D, and the spring behind the *floating piston forces oil into the void. Piston-and-rod C continues to advance and oil transfers until the work is met. This low-force advance stroke moves quickly (and uses air flow controls when necessary). Seals on ram D separate oil and air.*

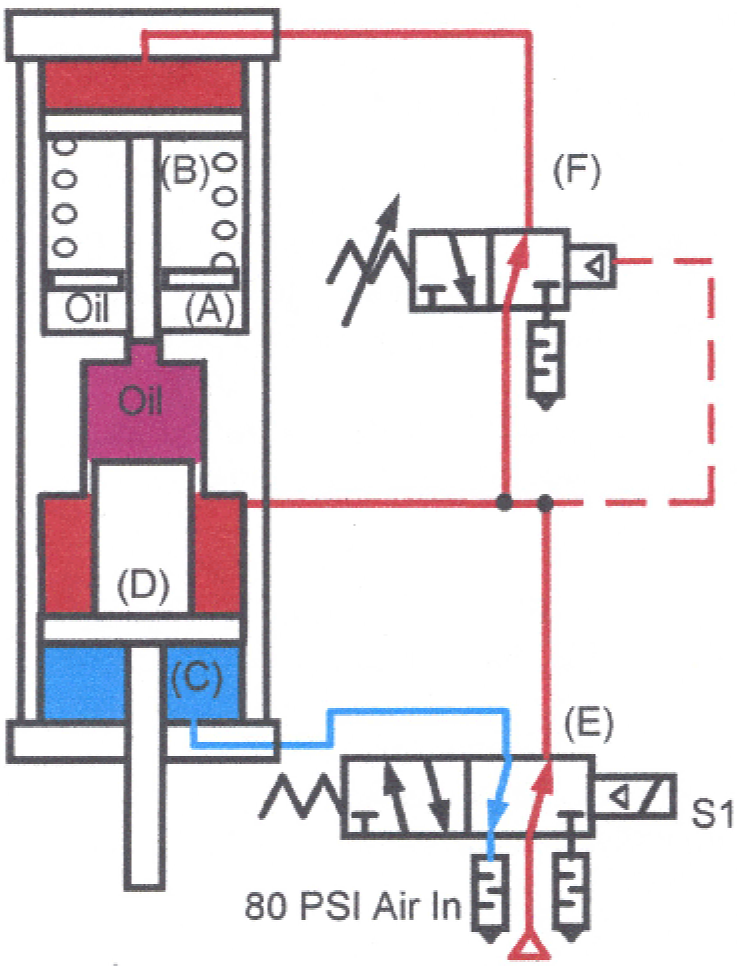


Fig. 13-28. Special air-oil intensifier cylinder. Starting high-pressure cycle.

Figure 13-28 shows the intensifier after contacting the work. When air piston-and-rodC stop against the work, pressure build-up behind the piston shifts sequence valve F. When sequence valve F shifts, shop air extends spring-return air piston B. The first movement of the spring-loaded air piston advances the rod to the flow port connecting the tank to the chamber behind hydraulic ram D. As the rod enters this flow port, it passes through a resilient seal, stopping flow to tank and sealing the chamber behind ram D. This action automatically isolates the low-pressure chamber — eliminating the need for a pilot-operated check valve.

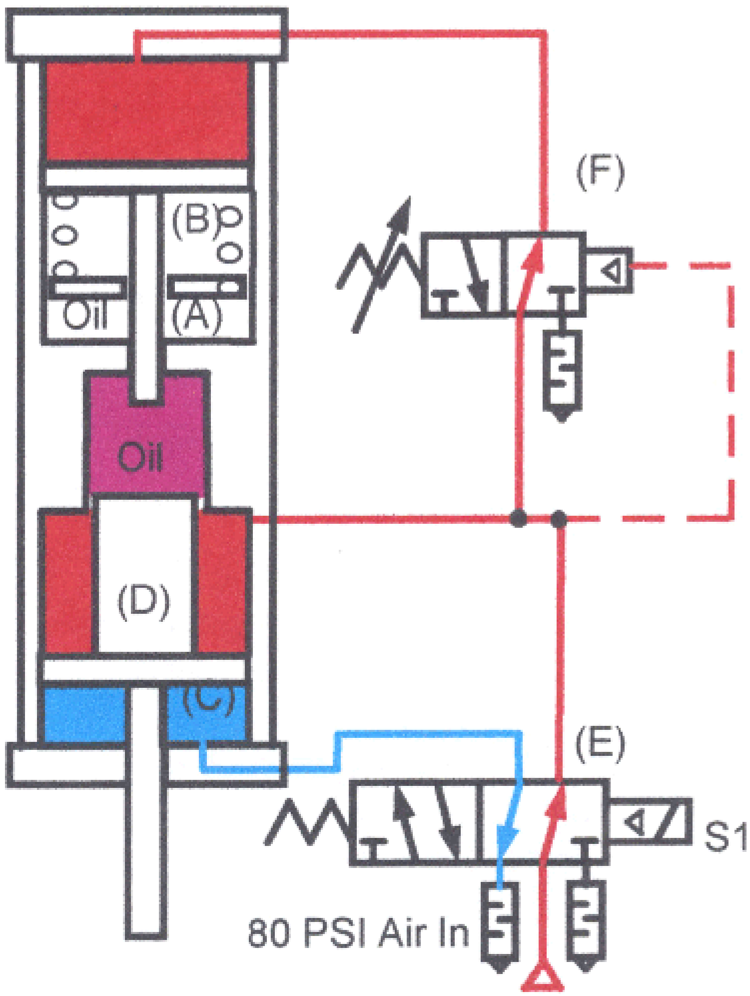


Fig. 13-29. Special air-oil intensifier cylinder extending at low speed with high force.

As the spring-return air piston continues to extend, as in Figure 13-29, the rod displaces oil in the chamber behind hydraulic ram D. In this case, the area of spring-return piston B is 15 times the area of the rod entering the sealed chamber. The air piston and rod continue to displace oil and move hydraulic ram D until the pressure behind the ram becomes 15 times greater than the air pressure on the piston. The stroke of spring-return piston B and the diameter of its rod set the maximum high-pressure work stroke. The higher the intensification ratio and the shorter the stroke, the less the high-pressure stroke capability.

Deenergizing directional valve E allows the spring loaded air piston and the work cylinder to return home. The work cylinder returns slowly while spring return air piston B retracts past the high-pressure seal.

Air-to-air intensifiers

Instead of buying a high-pressure compressor when only a small volume of compressed air is needed, consider using an air-to-air intensifier. Air-to-air intensifiers are small self-contained units that operate automatically as long as they have a supply of compressed air. Figure 13-30 shows a generic schematic of a simple air-to-air intensifier made from stock cylinders and valving. The arrangement has two cylinders connected at their rod ends and mounted on a beam, with limit switches or limit valves, a directional control valve, and four check valves. As long as compressed air is supplied to the intensifier, it takes in atmospheric air, compresses it, and sends it to a receiver and/or the system.

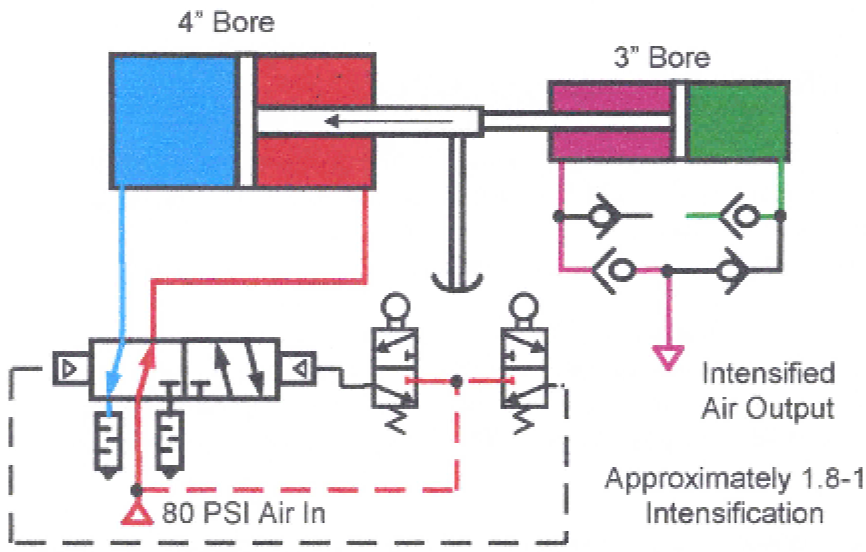


Fig. 13-30. Typical piping arrangement for air-to-air intensifier (shown running with air on).

As the cylinders in Figure 13-30 stroke to the left, the intensifier takes in atmospheric air at the cap end of the 3.0-in. bore cylinder. Compressed air discharges from the rod end to a high-pressure receiver. After the cylinders fully stroke to the left, a limit valve pilot-shifts the directional control valve to stroke the cylinders to the right. When the cylinders stroke to the right, the opposite check valves take in and discharge air. Reciprocation continues until outlet pressure from the 3.0-in. cylinder reaches approximately twice the inlet pressure at the 4.0-in. bore cylinder.

The piping arrangement in Figure 13-30 produces less intensified air per compressor horsepower than the circuits in Figures 13-31 and 13-32. Taking in and compressing atmospheric air to a higher pressure gives a minimal high-pressure volume for each stroke. When compressing a gas, reducing volume by one half doubles absolute pressure. If the 3-in.

bore cylinder has a 6-in. stroke and intake pressure is 14 psia, then as the cylinder moves through 3 in. of stroke, pressure climbs to 28 psia. As the 3-in. bore cylinder continues to stroke, pressure goes to 56 psia 1.5 in. from the end and to 112 psia 0.75 in. from the end. The cylinder finally starts discharging 160-psia air about 0.625 in. from the end of its stroke. Volume entering the high-pressure receiver is minimal for each stroke, and continues to decrease as the pressure level increases.

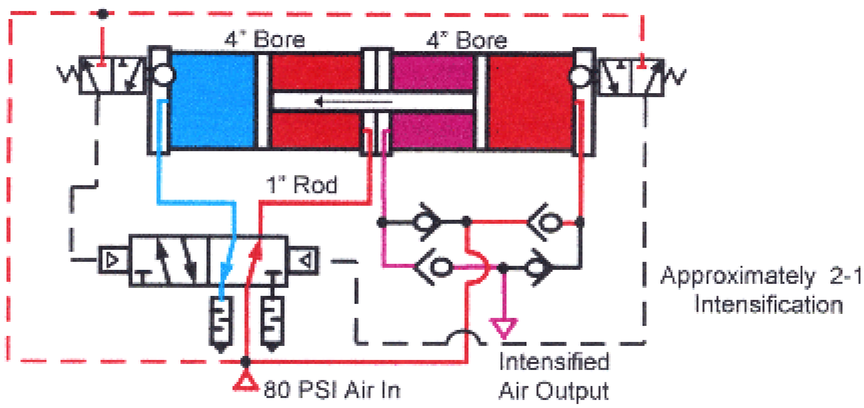


Fig. 13-31. Air-saving piping arrangement for air-to-air intensifier (shown running with air on).

Using shop-air pressure in the intensifying cylinder, Figures 13-31 and -32, greatly reduces this high-pressure/low-flow problem. First, the high-pressure receiver starts with 80 psig and the air in the intensifying cylinder starts at 94 psia. This circuit discharges intensified air for more than half its stroke, making it a smaller, more-efficient package.

Notice also, the approximately 2:1 intensification from a 4-in. cylinder driving a 4-in. cylinder. This is possible because two areas, pressurized by shop air, push against one area of the intensifier cylinder. The actual intensification of the unit in Figure 13-31 is 2.06:1 when stroking to the left, and 1.93:1 when stroking to the right.

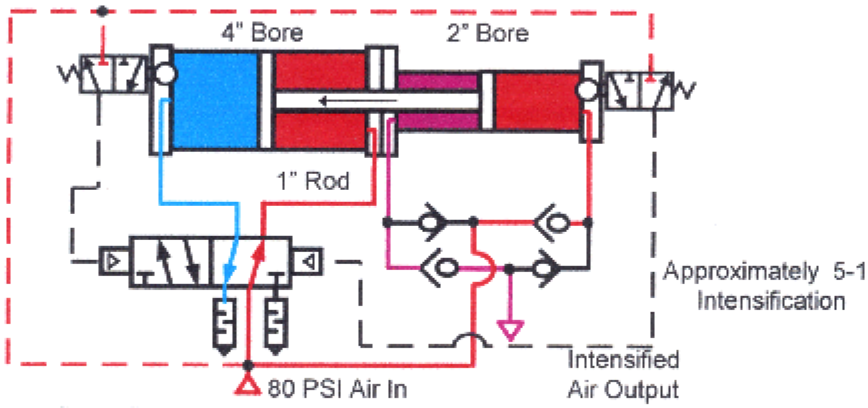


Fig. 13-32. Air-saving piping arrangement for air-to-air intensifier (shown running with air on).

For higher pressure use a smaller-bore intensifier cylinder or a larger-bore driving cylinder. Figure 13-32 depicts a 4-in. bore driving cylinder and a 2-in. bore intensifier cylinder. This combination increases inlet air pressure about five times. The actual intensification ratios are 6.33:1 as the cylinders stroke left, and 4.74:1 as the cylinders stroke right.

When specifying cylinders to build an air-to-air intensifier, be careful not exceed their pressure rating. Pre-lubed cylinders are best for this type of operation because they keep excess lubricator oil out of the high-pressure circuit.

As an air-to-air intensifier pumps air to a maximum pressure, the volume decreases as the pressure increases. It is best to operate the intensifier to produce a pressure 15 to 20% higher than the system needs, with a regulator to set the maximum pressure at the work.

When one branch of a fluid power circuit must operate at a lower pressure, use a reducing valve to provide it. Reducing valves control their outlet or downstream pressure only.

Air line regulators, Figure 16-1, reduce pressure for a pneumatic circuit. Because air in the supply line to a machine is at maximum pressure, energy can be saved by reducing pressure whenever possible. With a compressor setting between 115 and125 psi and a machine requirement of 70 psi, approximately 40% of the input energy would be lost without a properly adjusted regulator. The air-driven machine will work at the higher pressure, but it consumes more compressor horsepower than necessary.

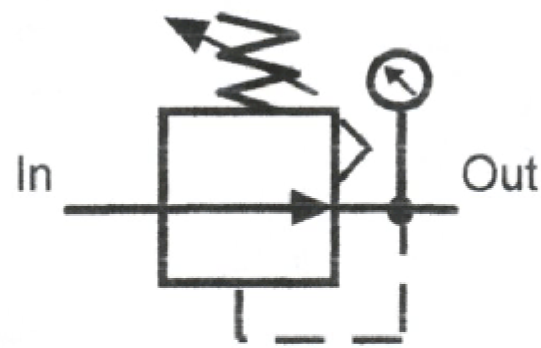


Figure 16-1. Self-relieving type air line regulator.

Another use for air line regulators is on retraction strokes of air cylinders. Reducing pressure on a cylinder's retraction stroke saves air and thus consumes less compressor horsepower.

In multiple actuator circuits, it is often impossible to size all actuators to operate at maximum system pressure. For example: when a cylinder needs 5000 lb of force and one standard bore produces only 4712 lb at maximum pressure, the designer must go to the next larger standard bore. However, the next larger bore produces 7363 lb of force, which can cause machine or part damage. Instead, install a pressure-reducing valve in the branch circuit with the over-sized cylinder, as in Figure 16-2, to lower that branch's pressure to generate the required cylinder force.

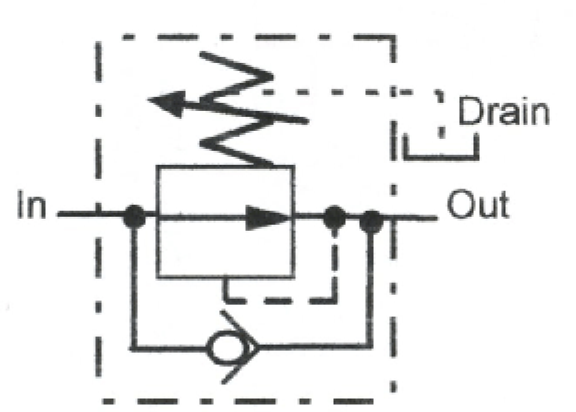


Figure 16-2. Pressure-reducing valve with bypass check valve.

A standard reducing valve is normally open. When downstream pressure goes higher than its setting, the valve closes, blocking flow. If pressure downstream tries to increase — say due to resistance from an opposing cylinder — a reducing valve also blocks reverse flow. Escalating pressure in the downstream line continues until something bursts or gets mechanically damaged.

Figure 16-3 shows the symbol for a reducing-relieving valve. A reducing-relieving valve sets maximum outlet pressure, then relieves fluid to tank when outlet pressure tries to go higher. The overpressure could be due to outside forces or possibly high temperature in some environments. A reducing-relieving valve has an integral relief valve with a full-flow line to tank. When pressure in the downstream circuit rises 3 to 5% above reduced pressure, trapped fluid relieves to tank. Adjusting the reduced pressure automatically sets the maximum relief pressure.

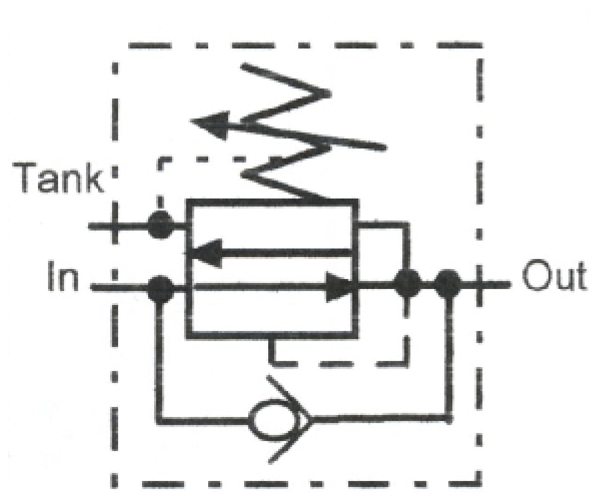


Figure 16-3. Pressure-reducing-relieving valve with bypass check valve.

Hydraulic reducing valves always have a drain line open to tank for control oil flow. Drain oil flows when reducing valve outlet is lower than its inlet. This generates a small amount of heat in the system. Blocking the drain line forces the valve wide open and lets outlet pressure rise to system pressure.

Multiple pressures in one circuit

Figure 16-4 has a schematic diagram for two cylinders that need different pressures. One option a novice designer might use is to add a second relief valve. However, second relief valve B reduces pressure in the whole circuit. System pressure cannot go above 400 psi — making high-pressure relief valve A useless.

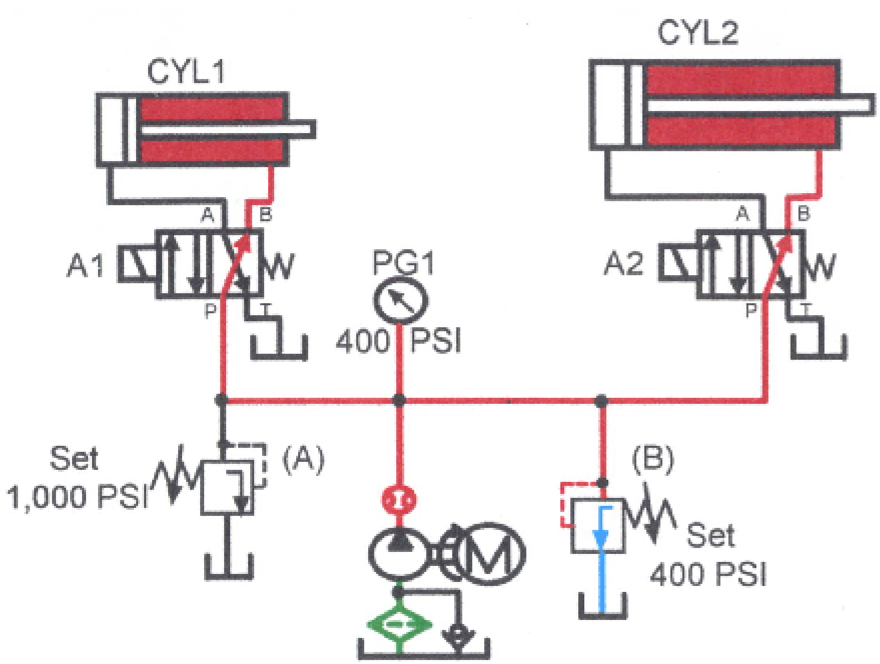


Figure 16-4. Using relief valves for two pressures.

In Figure 16-5, reducing valve C replaces relief valve B. Now each cylinder operates at a different pressure. Note that there is no bypass check valve on reducing valve C. When the system does require reverse flow through the reducing valve, the bypass check valve can be omitted. However, for a circuit with reverse flow always use a bypass check.

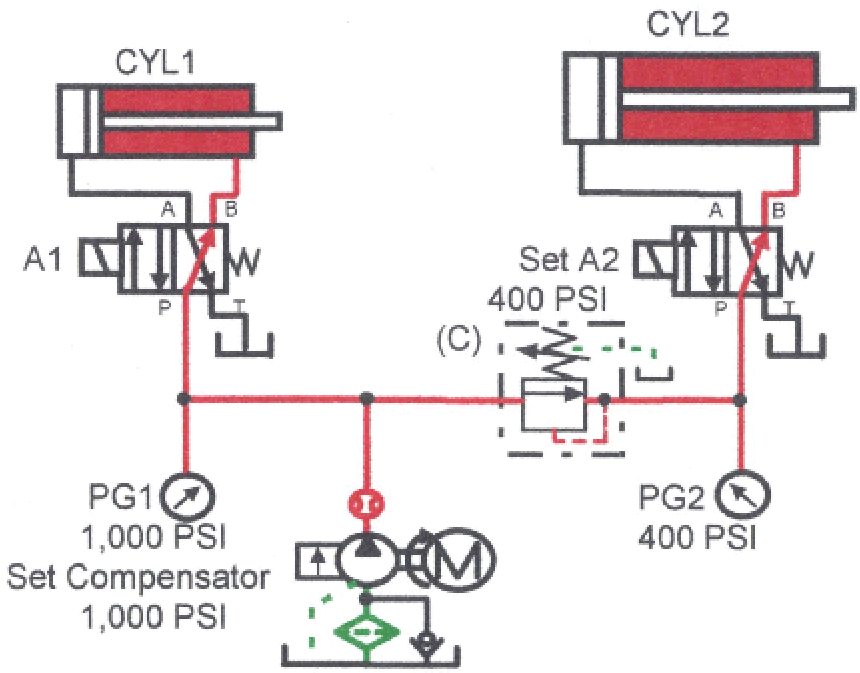


Figure 16-5. Using a reducing valve for two pressures.

With the reducing valve installed in the line that feeds the directional control valve, pressure at both ends of the cylinder is reduced. Also, when the pump is at pressure, reducing valve drain line flow is constant. Drain flow amounts to 20 to 70 in.3 minimum, and produces heat. With several reducing valves in a system, drain line flow might require a larger pump and a heat exchanger.

Figures 16-6 and 16-7 show the preferred location for a reducing valve. In Figure 16-6, the circuit is at rest. There is no drain flow with the reducing valve in the line between the directional valve and the actuator. This arrangement eliminates oil heating and provides extra flow to other actuators. When both ends of the cylinder need pressure reduction and/or different pressures, use the arrangement in Figure 16-7. The components cost more up front but the energy it saves often pays for the extra reducing valve.

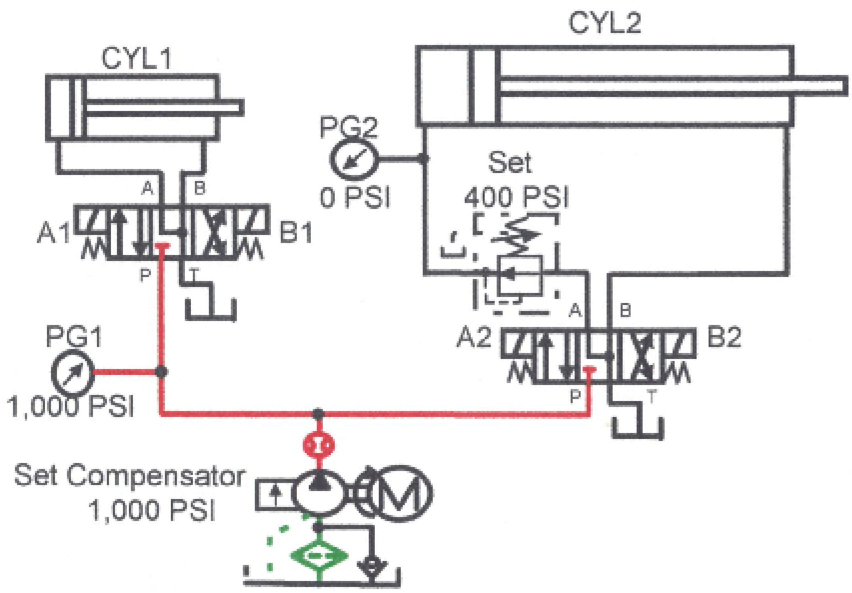


Figure 16-6. Using a reducing valve for two pressures (circuit at rest with pump running).

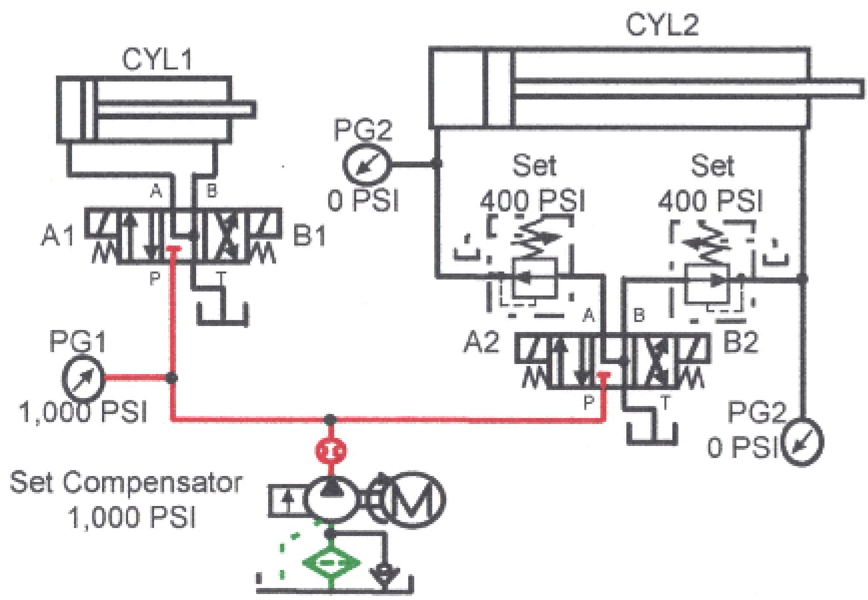


Figure 16-7. Using two reducing valves for two pressures (circuit at rest with pump running).

A reducing valve is normally open from inlet to outlet, but closes when reaching the outlet pressure setting. When an actuator at reduced pressure reverses suddenly, the reducing valve does not have time to open. Oil forced out of the cylinder that tries to go back through the reducing valve keeps pressure on the outlet, holding it shut. A small pilot-drain flow in this *blocked reverse flow condition allows very slow reverse cylinder movement. A reducing valve with a bypass check may try to stay closed but will not block flow, so the cylinder reverses easily.*

Two-pressure circuit with a pressure-reducing valve
Always connect the drain line of a pressure-reducing valve to a free-flow tank line. Backpressure in the drain line adds to the spring setting, thus raising the set pressure. A constant backpressure can be offset by a lower spring setting, avoiding a problem. With intermittent and/or fluctuating backpressure, the reduced outlet pressure changes when the backpressure changes.

Some circuits require a reduced pressure to position a part, then full pressure to do the work. A reducing valve easily gives two pressures by opening or blocking the drain line. Figures 16-8 through 16-11 show a simple way to get two pressures using a reducing valve and a normally open 2-way directional valve.

Figure 16-8 shows a normally open 2-way directional control valve piped in the drain line. There is no leakage from the drain port in the at-rest condition.

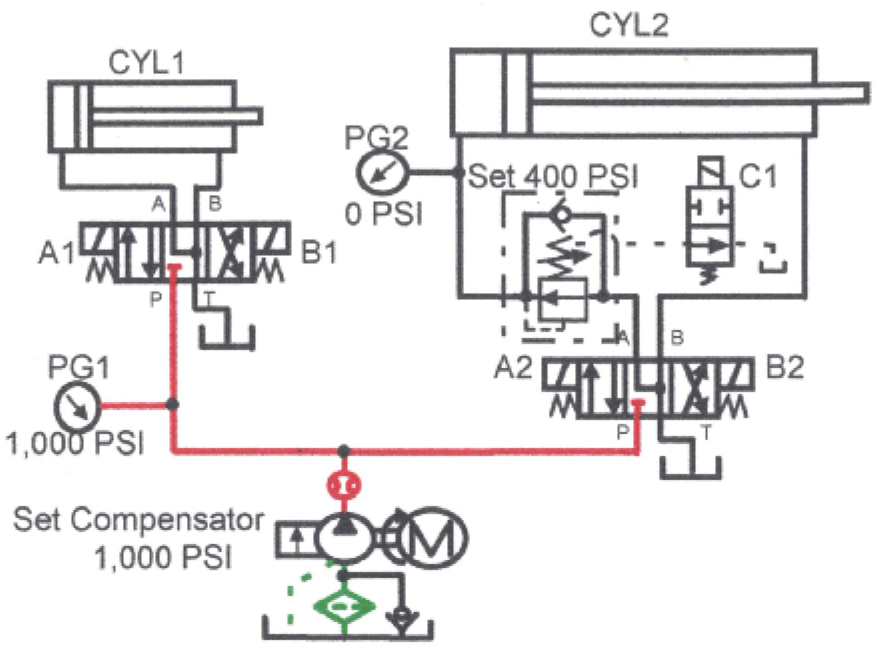


Figure 16-8. Using a reducing valve for dual pressure (circuit at rest with pump running.). (contacting work at low pressure).

Figure 16-9 shows the directional valve on CYL2 shifted to advance the cylinder to the work at low pressure. During this part of the cycle the reducing valve stays open.

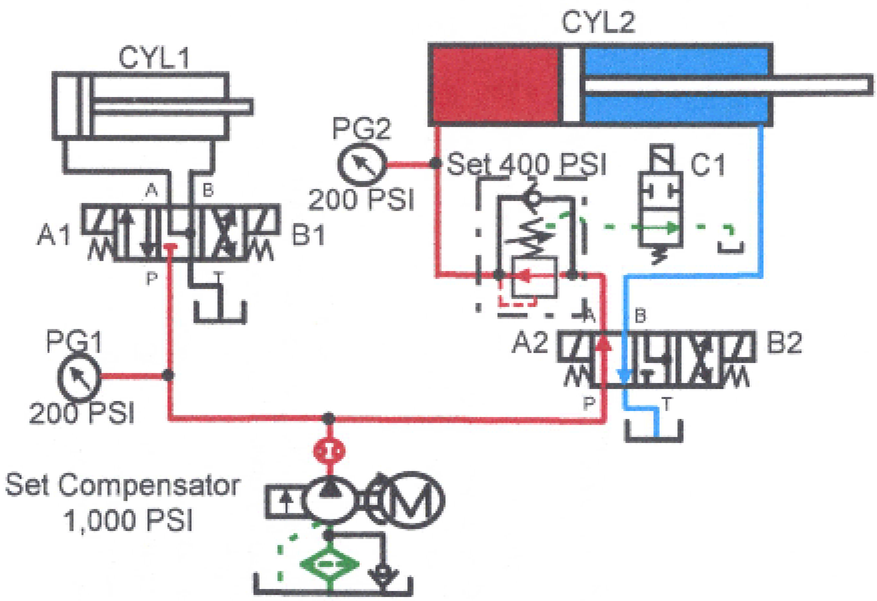


Figure 16-9. Using a reducing valve for dual pressure (cylinder 2 extending at low pressure).(pressing work at high pressure).

Figure 16-10 shows the cylinder contacting the work with pressure at the reducing valve setting. The low pressure continues as long as required. During this time the operator can check part alignment or other details. If a problem is detected, the operator simply reverses the cylinder to realign any out of place or problem components.

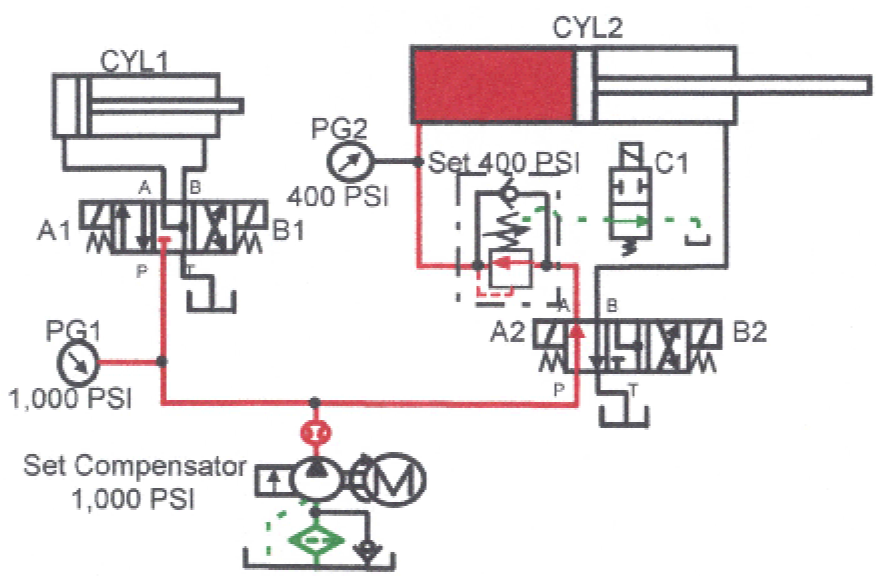


Figure 16-10. Using a reducing valve for dual pressure (cylinder 2 contacting work at low pressure).

After determining all is well, the operator energizes the solenoid on the 2-way directional valve as shown in Figure 16-11. This blocks drain flow from the reducing valve. Blocking drain flow at the reducing valve causes it to open fully. Backpressure in the blocked drain line, plus the internal valve spring, pushes and holds the spool open. When the reducing valve opens, full system pressure goes to the cylinder to generate high force. This action poses no problem to the reducing valve. This circuit is a reliable way to get two pressures for an actuator.

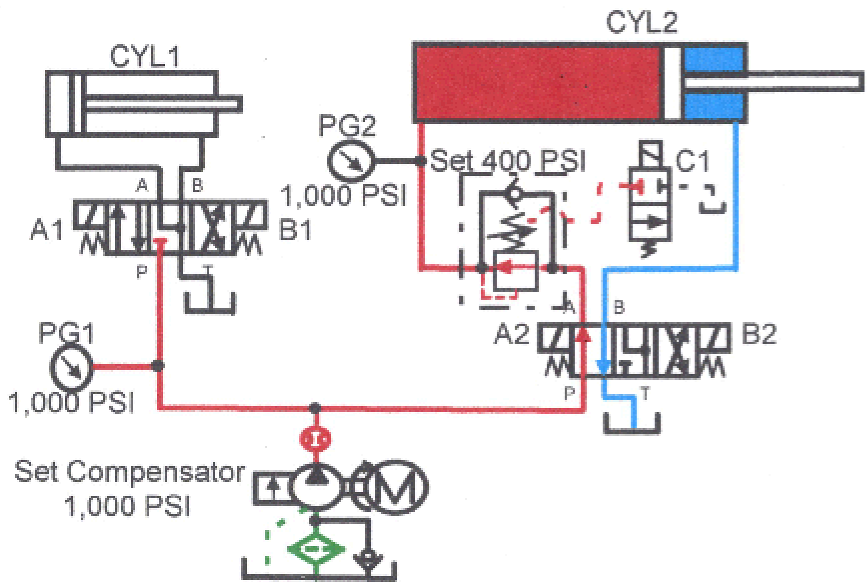


Figure 16-11. Using a reducing valve for dual pressure (cylinder 2 pressing work at high pressure).

Remotely operating a pressure-reducing valve

Pilot-operated reducing valves have a remote pressure-control port. Connecting this port to other pressure valves allows pressure to be changed from a remote location. For example, Figure 16-12 shows a reducing valve with a directional valve and two remote relief valves connected to the remote-control port. With the directional control valve in its center position, set the pressure with the knob on the reducing valve. This setting is always the highest reduced pressure for the circuit.

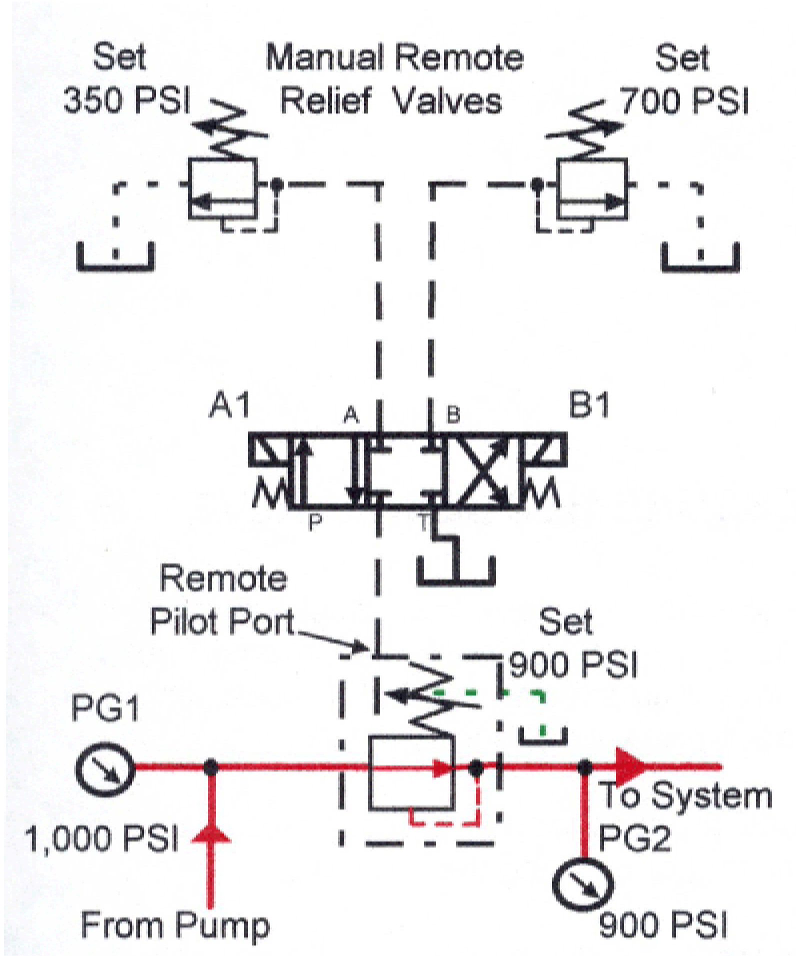


Figure 16-12. Using remote pilot port for three different system pressures (no solenoids energized).

Energizing solenoid A1 of the directional valve, as in Figure 16-13, connects the remote pilot port to the remote relief valve SET 350 psi. Pressure in the system now drops to and holds at 350 psi. Energizing solenoid B1 of the directional valve, as in Figure 16-14, connects the remote pilot port to the remote relief valve SET 700 psi. Pressure in the system now rises to 700 psi and holds at that level.

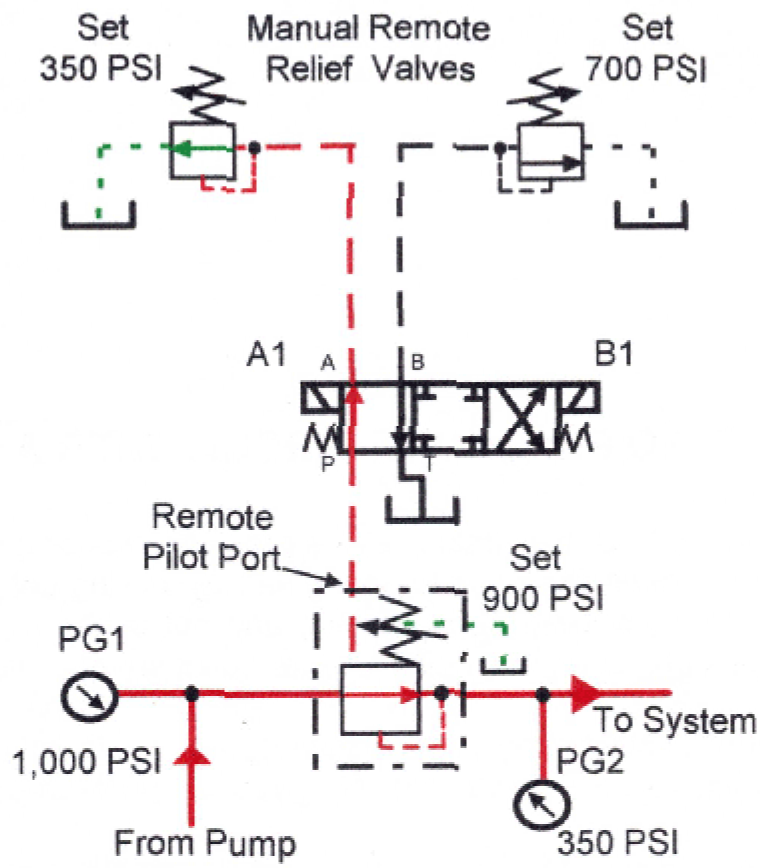


Figure 16-13. Using remote pilot port for three different system pressures (solenoid A1 energized).

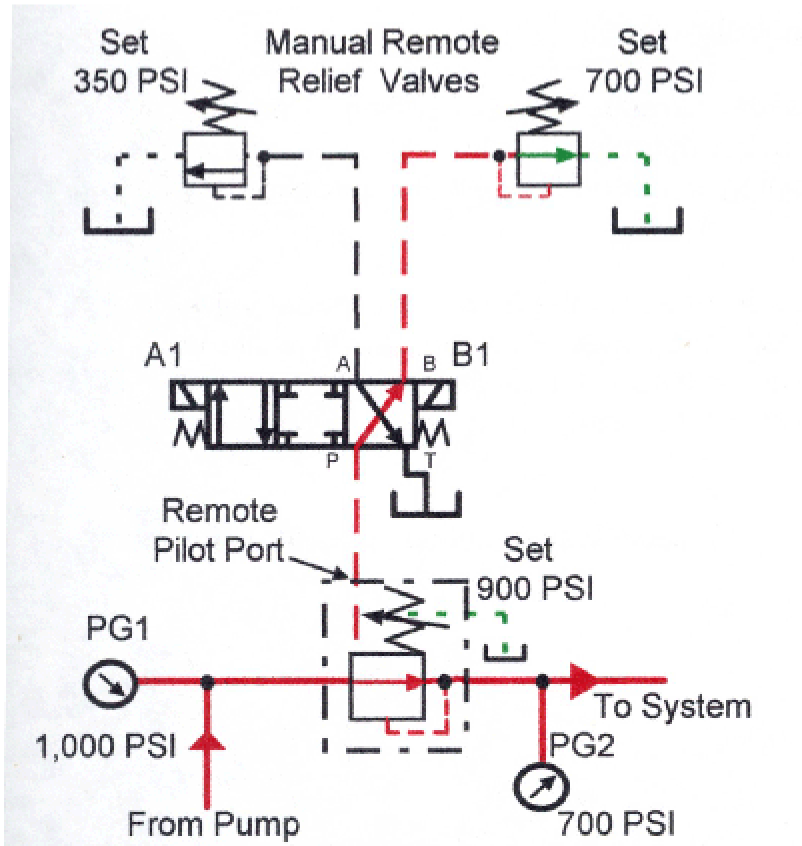


Figure 16-14. Using remote pilot port for three different system pressures (solenoid B1energized).

Figure 16-15 shows the reducing valve's remote port connected to an infinitely variable electrically modulated relief valve. An electronically controlled relief valve changes the reduced pressure infinitely with a remote electrical controller.

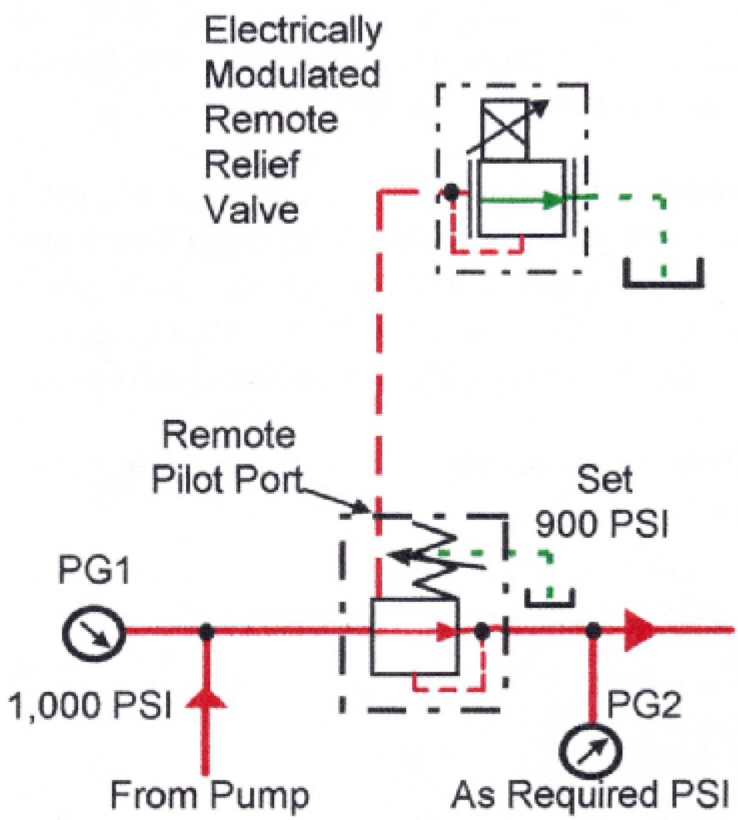


Figure 16-15. Using remote pilot port with a servo press controller for infinitely variable pressure.

Pressure-reducing-relieving valves
When it is possible for an external force to increase pressure in a reduced-pressure circuit, use a reducing-relieving valve. Most modular valves now have the reducing-relieving function. When in doubt, specify reducing-relieving valves where they are required.

Figure 16-16 shows a large-bore cylinder opposing a smaller-bore cylinder. With a standard reducing valve, oil in the cap end of the 2-in. bore cylinder (CYL1) is blocked after reaching reduced pressure. With a 6-in. bore CYL2 opposing CYL1, pressure could increase to 9000 psi in its cap end. Pressures this high could cause machine damage and be unsafe.

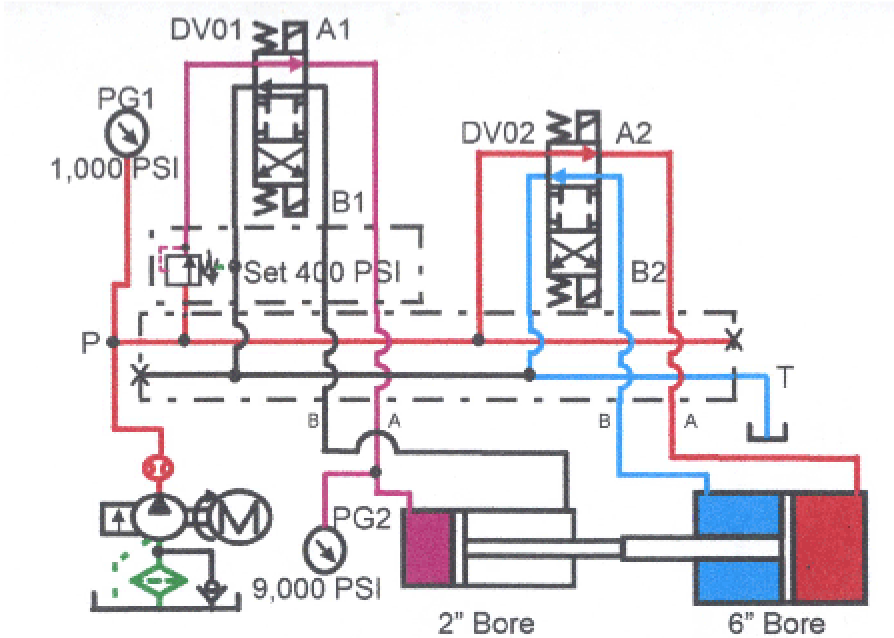


Figure 16-16. Using a reducing valve in circuit with unmatched opposing cylinders (both extending and locked up).

Figure 16-17 shows a reducing-relieving module installed. Now, pressure in the end of cylinder CYL1 only increases to 430 psi. At 430 psi, the relief function takes over and the cylinder retracts.

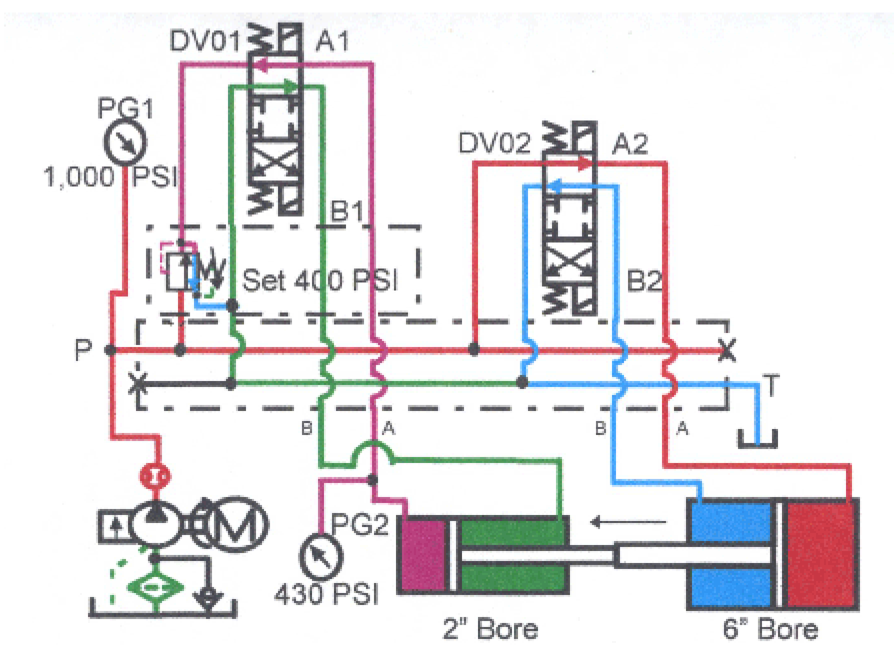


Figure 16-17. Using a reducing valve in circuit with unmatched

opposing cylinders (larger cylinder driving smaller back).

A cylinder in a high-temperature location may have a similar problem. (Normally, hydraulic systems are not installed in areas with excessive heat, but it is a possibility.) With the cylinder extended at reduced pressure, as in Figure 16-18, heat could cause pressure at a conventional reducing valve outlet to increase and cause failure. Figure 16-19 shows how a reducing-relieving valve allows any heat-expanded oil to relieve to tank.

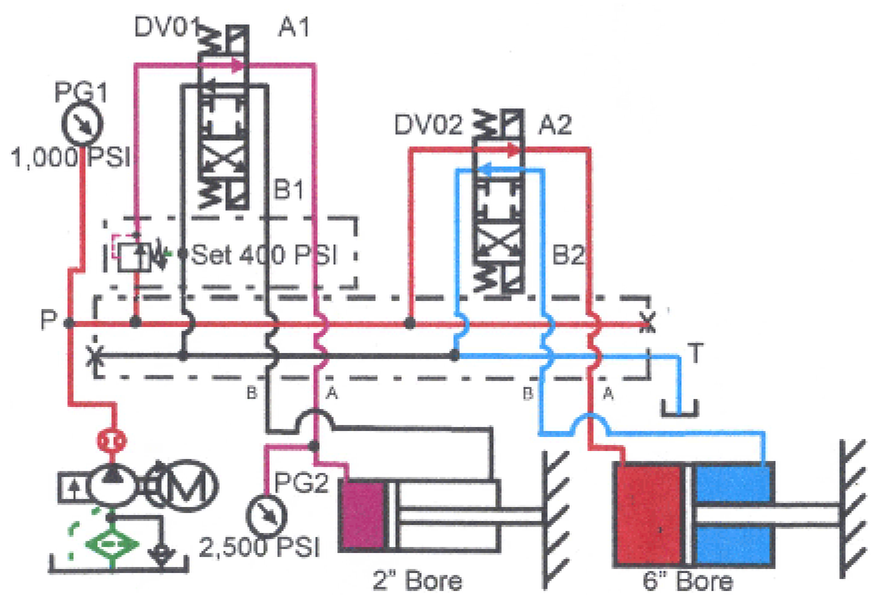


Figure 16-18. Using a reducing valve with cylinder in high-temperature area (cylinder stalled).

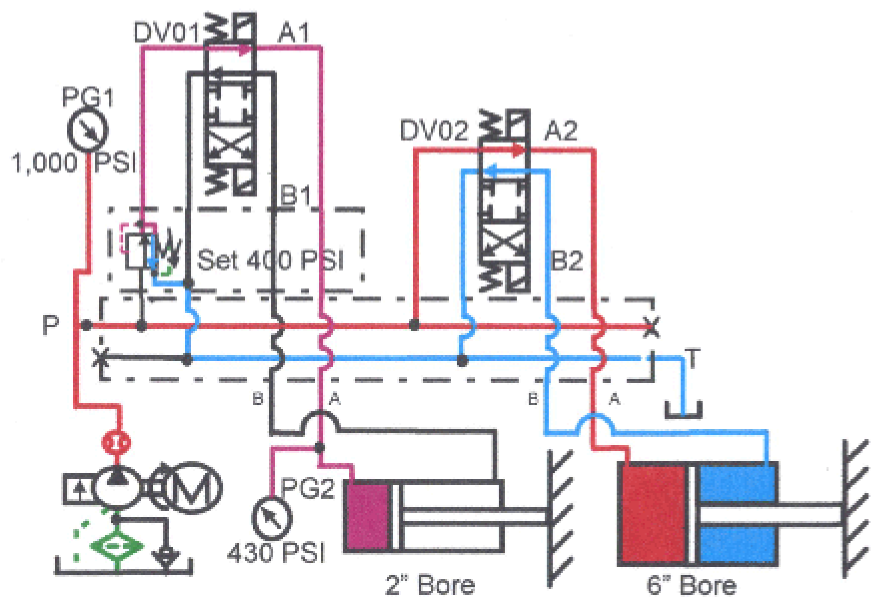


Figure 16-19. Using a reducing-relieving valve with cylinder in high-temperature area (cylinder stalled).

With slow heat build-up in a location with conventional ambient temperatures, oil expansion that raises pressure is slow enough to pass through the normal drain function.

All pilot-operated reducing valves have a drain line that bypasses control oil. There is always a small amount of oil passing through it. When drain flow is sufficient to handle backpressure from outside forces or heat, a reducing-relieving valve may be unnecessary. If in doubt, specify a reducing-relieving valve for safety's sake.

Modular pressure-reducing-relieving valves

When buying modular reducing valves or reducing-relieving valves, different options help reduce heat in a circuit while still maintaining good control.

Figures 16-20 and 16-21 show a reducing-relieving valve in the pump port line. The valve has an internal pilot that maintains reduced pressure at the outlet port. This means there is heat-generating flow from the drain line whenever the pump is running.

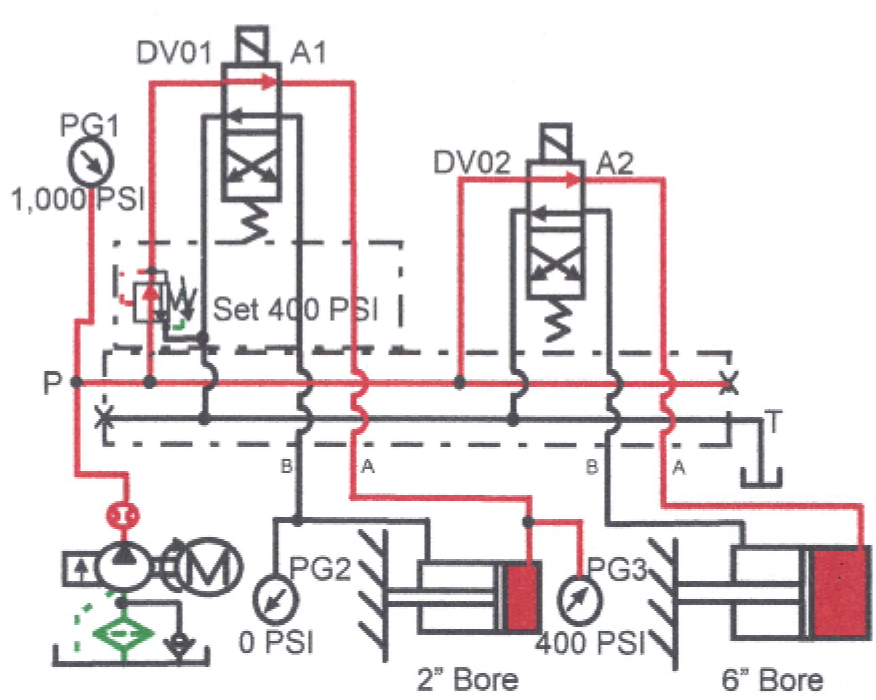


Figure 16-20. Reducing valve circuit using a reducing-relieving valve on port P — with both cylinders contacting a load.

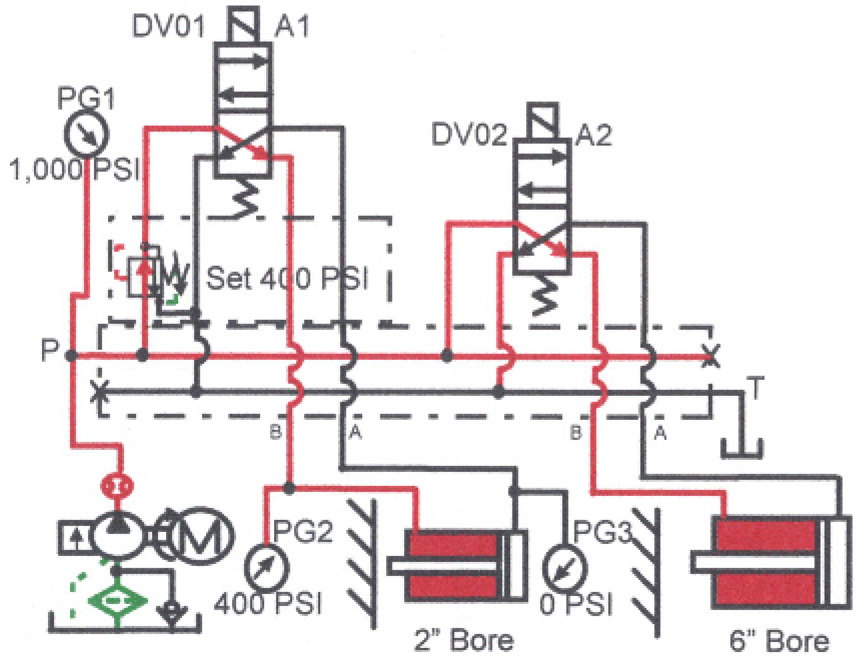


Figure 16-21. Reducing valve circuit using a reducing-relieving valve on port P — with both cylinders retracted.

Remotely piloting the reducing-relieving valve from port A, as in Figure 16-22, reduces pressure only on the extension stroke of the cylinder. While the cylinder retracts and holds, as in Figure 16-23, the reducing valve drain is not bypassing oil. (Some manufacturers put the reducing valves directly in the A or B ports and use bypass checks for reverse free flow.)

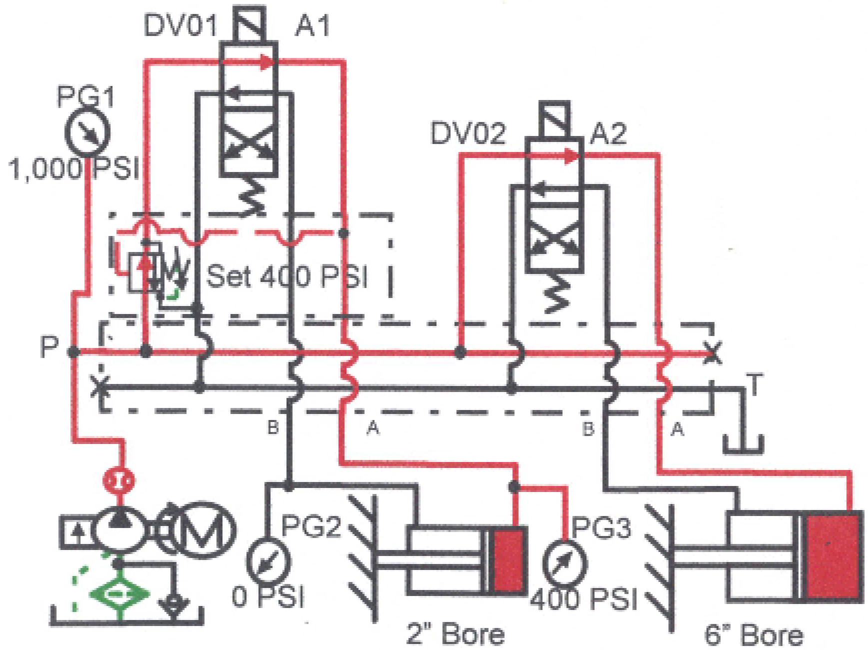


Figure 16-22. Reducing valve circuit with reducing-relieving valve on port A — with both cylinders contacting a load.

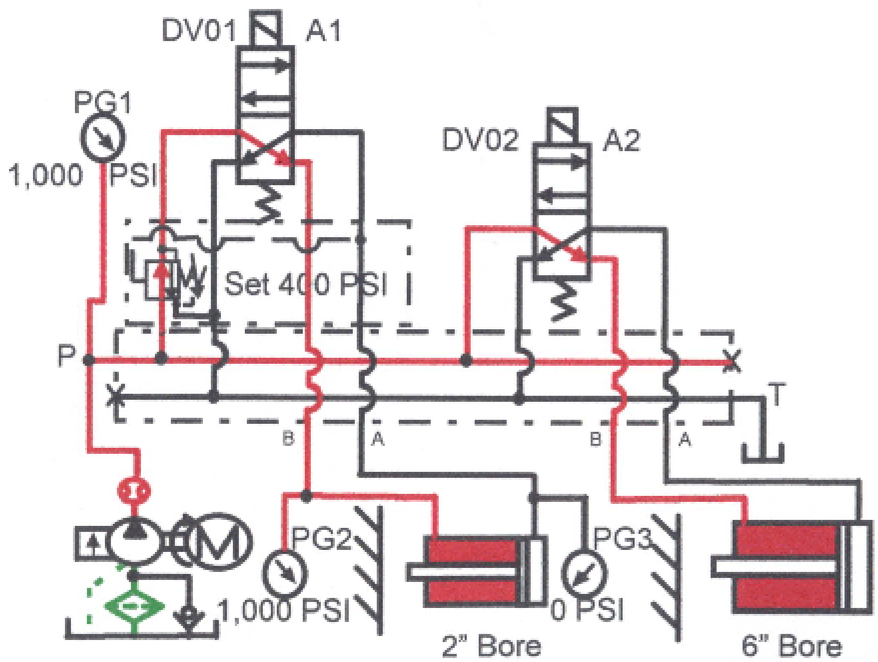


Figure 16-23. Reducing valve circuit with reducing-relieving valve on port A — with both cylinders retracted.

In either case, drain flow only takes place during a small portion of the cycle. This may sound unnecessary, but some circuits have multiple reducing valves. Excess drain flow can cause heating and fluid waste. (Pilot flow cannot be used to operate other actuators.)

Extra time spent on circuit design pays off in energy-efficient systems that perform better in the work place

Regeneration Circuits

A regeneration circuit can double the extension speed of a single-rod cylinder without using a larger pump. This means that regeneration circuits save money because a smaller pump, motor, and tank can produce the desired cycle time. It also means that the circuit costs less to operate over the life of the machine.

A regeneration circuit can also replace a double rod-end cylinder in some circuits. With equal rod diameters, a double-rod cylinder's area is the same on both ends. Equal areas mean identical force and speed both ways at a given pressure and flow. Reciprocating tables often use double rod-end cylinders for this reason. When the main function of a double rod-end cylinder is equal speed and power in both directions of travel, replace it with a regeneration circuit.

A double rod-end cylinder costs more than a cylinder with a single oversize rod; the extra rod needs space in which to move; and the second rod seal is another potential leakage source. To eliminate these objections, use the full-time regeneration circuit shown in Figures 17-6 and 17-8. Extension and retraction speed (as well as thrust) is the same, without the extra rod and its problems.

One disadvantage to using cylinders with a single oversize rod is that speed and thrust are not identical if the rod diameter ratio is not exactly 2:1. Most cataloged 2:1 rod diameters are only close to that ratio. A standard NFPA 3.25-in. bore cylinder comes with a 2.00-in. diameter rod as a 2:1 differential. If using this cylinder in a full-time regeneration circuit, speed is about 21% faster on the extension stroke, with about 21% less force than the retraction stroke.

This chapter explains regeneration principles and shows several common circuit designs, as well as some uncommon and unique circuits.

Regeneration principles

Applying fluid flow to both ports of a single-rod cylinder makes it extend -- or at least try to extend. Because areas on opposite sides of the piston are unequal, the cap end of the cylinder always has more force than the rod side.

Figure 17-1 shows forces and speeds that result when using a cylinder with a 10-in.^{2} piston area and a 2-in.^{2} rod area. This cylinder has a rod differential of approximately 1.25:1.

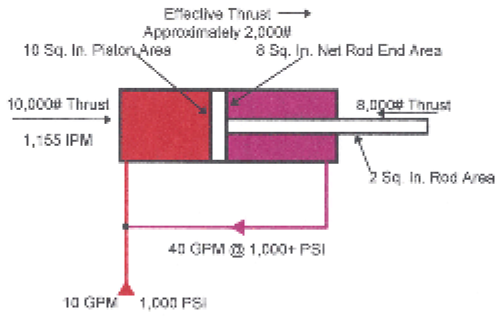


Fig. 17-1. Regeneration flows and forces for a cylinder with a small-diameter rod.

Applying 10-gpm flow to both ports of the cylinder makes it extend at a rate of 1155 ipm. The effective cylinder thrust as it moves forward at 1000 psi is 2000 lb. The same cylinder, with 10 gpm flowing into the cap end while the rod end is connected to tank, would extend at 231 ipm or 1/5 the speed of the regeneration circuit.

Figure 17-2 shows the forces and speeds of this same cylinder as it retracts. Thrust is 8000 lb, but speed is only 288 ipm.

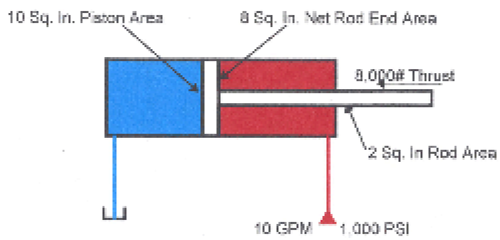


Fig. 17-2. Cylinder with a small-diameter rod retracting.

Often when a cylinder with a small rod is piped for regeneration, it will not even try to extend at a reasonable working pressure. This is because the extension force is too low to overcome cylinder friction, machine force requirements, and pressure drop due to high flow from rod end to cap end. Even if the cylinder does extend, low force may make it useless. Notice also that extension speed is very fast and retraction speed is very slow.

Most regeneration circuits use cylinders with a 2:1 rod ratio. A 2:1 rod cylinder has a rod area equal to half the piston area. In actual use, extend and retract speed and power is identical with an area differential of exactly 2:1. All of the cylinders in the following examples have 2:1 ratios.

Regeneration with a 2:1 rod

Figure 17-3 shows a 2:1 rod cylinder in regeneration. Notice the difference in force and speed using the same flow and pressure as Figure 17-1. These figures assume an exact 2:1 rod. In actual practice only specially made cylinders have an exactly 2:1 area ratio.

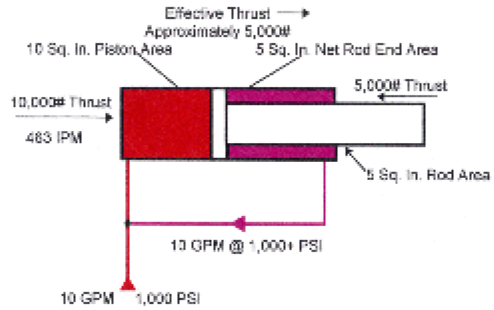


Fig. 17-3. Regeneration flows and forces for a cylinder with a 2:1 rod diameter.

With a 2:1 rod, force and speed on the extension and retraction stroke is identical. As Figure 17-4 shows, a regenerating cylinder acts like a ram-type cylinder. In effect, a cylinder in regeneration does not need a piston. Because both ports connect to the same power source, the effect of the piston area outside the rod diameter is cancelled. Pressure on both sides of the annulus area around the rod makes the piston area here useless. As the cylinder extends in regeneration, force is half while speed doubles.

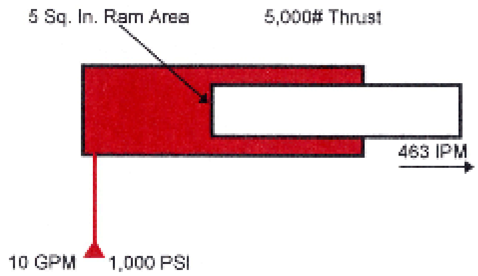


Fig. 17-4. Ram-type cylinder extending. (same force and speed as regeneration.)

During regeneration there is no flow to tank. All oil from the pump goes to the cylinder. All fluid from the rod end of the cylinder mixes with pump flow and goes to the cap end. The exchange of rod end fluid, added to pump oil going to the cap end, doubles the cylinder's speed.

Figure 17-5 shows force and speed when a 2:1 oversize rod cylinder retracts. Pump flow of 10 gpm enters the rod port and the cylinder retracts using the 5-in.2 annulus area. Speed during retraction is the same as when the cylinder regenerates forward. Flow from the cylinder cap end port is twice pump flow as it retracts because cap end area is twice rod end area. A conventional regeneration circuit has double flow through the valving while the cylinder extends and retracts.

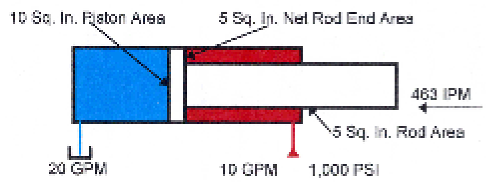


Fig. 17-5. Cylinder with 2:1 rod retracting.

The rest of this section shows many regeneration circuits with explanations of how they work. Different circuits have various features that make them better for certain applications. Some circuits are just a different way of connecting the same parts.

Full-time regeneration

The regeneration circuit in Figures 17-6 through 17-8 replaces a double-rod-end cylinder circuit used to produce equal speed and power in both directions of travel. The schematic diagram shows a tandem-center, 3-position, 4-way directional valve, connected to a 2:1 rod cylinder. Port A of the directional valve connects to the cap-end port of the cylinder. The rod-end port of the cylinder tees into port P of the directional valve. In the at-rest condition shown in Figure 17-6, the pump unloads to tank while the cylinder sits still.

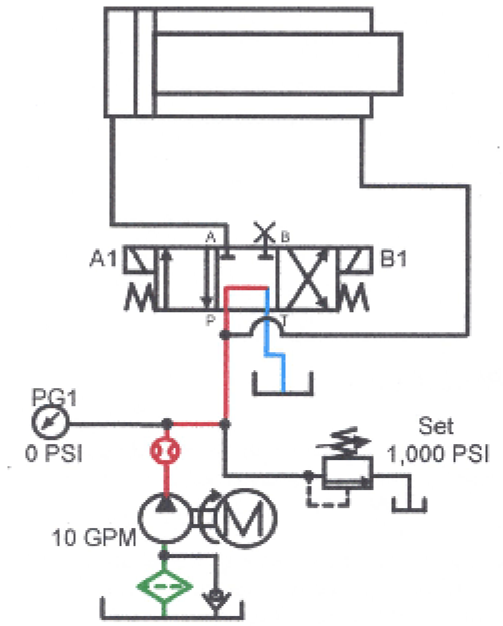


Fig. 17-6. Full-time regeneration circuit at rest, with pump running.

To extend the cylinder, energize solenoid A1 as in Figure 17-7. Energizing solenoid A1 connects the pump to the cap end of the cylinder to make it extend. Oil leaving the rod end of the cylinder mixes with pump flow and regenerates to the cap end of the cylinder through the directional valve. Supplementing pump flow with rod flow makes the cylinder extend twice as fast as a conventionally piped circuit.

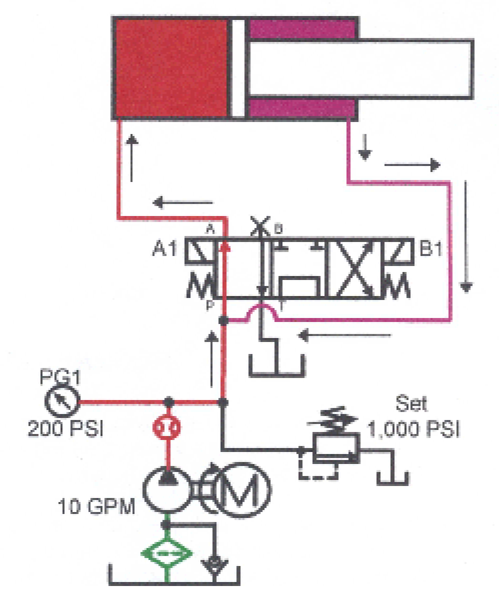


Fig. 17-7. Full-time regeneration circuit extending in regeneration with solenoid A1 energized.

At first it appears that the cylinder only goes 50% faster. If 10 gpm goes in the cap end of a 2:1 rod cylinder, only 5 gpm comes out of the rod end. This puts 15 gpm into the cap end for a 50% increase in speed. However, once 15 gpm goes into the cap end, 7.5 gpm comes out of the rod end. Now there is 17.5 gpm going into the cap end. Continuing this scenario shows that flow from the cylinder's rod end -- as it extends in regeneration -- almost immediately reaches 10 gpm to produce double cylinder speed.

Figure 17-8 shows the cylinder retracting. Pump flow goes directly to the rod-end port and the cylinder retracts. Oil from the cap-end port returns to tank through the directional valve. Note that in a full-time regeneration circuit, the directional valve handles twice pump flow in both directions of travel. If the valve were sized for just pump flow, excess pressure drop would slow cylinder speed and add to system heating.

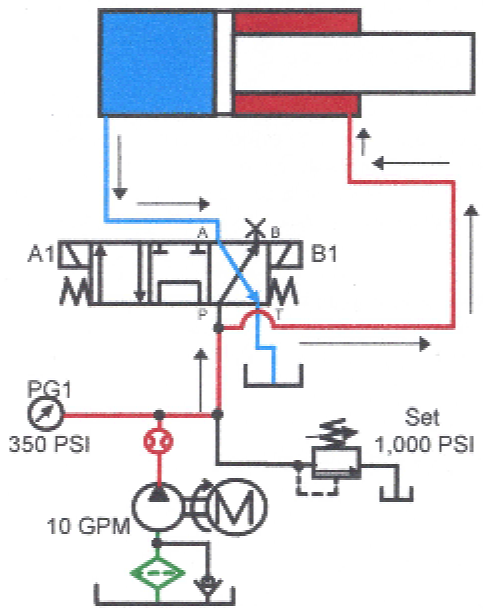


Fig. 17-8. Full-time regeneration circuit retracting with solenoid B1 energized.

In Figure 17-9, a needle valve has been teed into the rod-end line and connected to port B. Use this needle valve to adjust maximum speed when the cylinder extends. This circuit is useful when standard speed is too slow but double speed is too fast. It is a bleed-off flow-control circuit, which means opening the valve reduces speed. Also, for accurate speed control, use a pressure-compensated flow-control valve in this circuit.

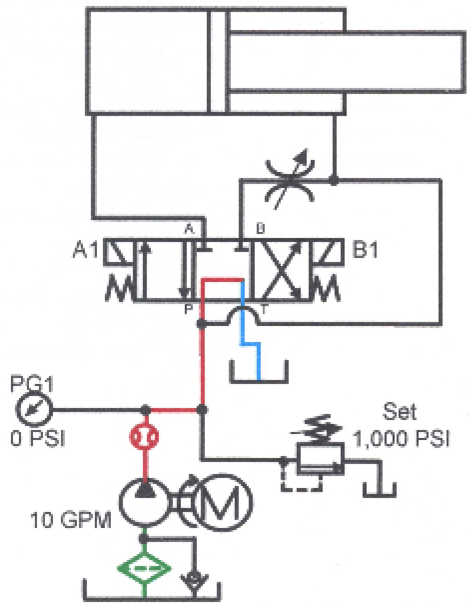


Fig. 17-9. Full-time regeneration circuit with needle valve to adjust extension speed.

Regeneration circuit – pressure-actuated to full thrust

Regeneration circuits move a cylinder rapidly to the work, but only have half power at work contact. For machines that need full cylinder force to do their jobs, divert regeneration flow to tank after work contact. The full force portion of the stroke is only half as fast as approach speed. The half-speed work stroke poses little problem because it usually is a small portion of the cycle.

Figures 17-10 through 17-13 show a regeneration circuit that changes to full thrust when pressure reaches a predetermined setting. The addition of externally piloted sequence valve A and check valve B to the circuit makes this possible. When pressure rises, sequence valve A shifts to connect the rod end of the cylinder to tank, eliminating backpressure. Because sequence valve A has an external pilot, it opens fully as pressure builds in the cylinder's cap end.

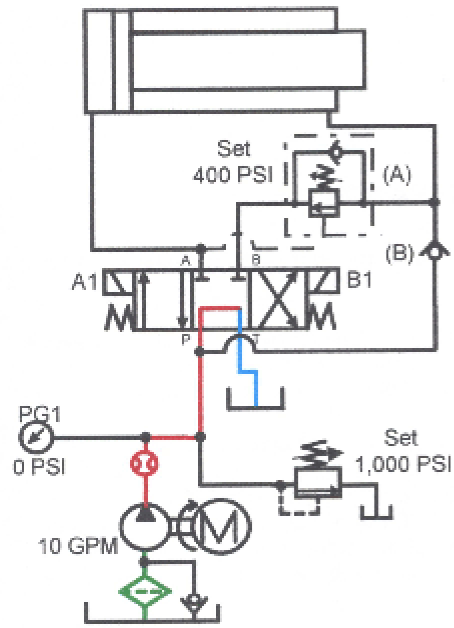


Fig. 17-10. Regeneration circuit that is pressure-activated to full thrust. Shown at rest, with pump running.

Figure 17-11 shows the cylinder regenerating forward. During this part of the cycle pump oil goes to the cap end of the cylinder through the directional valve. Oil from the rod end free flows through check valve B to mix with pump flow -- passing through the directional valve to the cylinder's cap end also. Sequence valve A stays closed because its pressure setting is higher than the pressure it takes to move the regenerating cylinder. The setting of sequence valve A is easy to establish while the circuit runs. Start with a high-pressure setting and slowly lower it as the cylinder extends until speed starts to slow. When speed slows, raise the setting until speed is maximum, then go another half turn higher. The cylinder continues to extend at twice speed and half power until it meets added resistance.

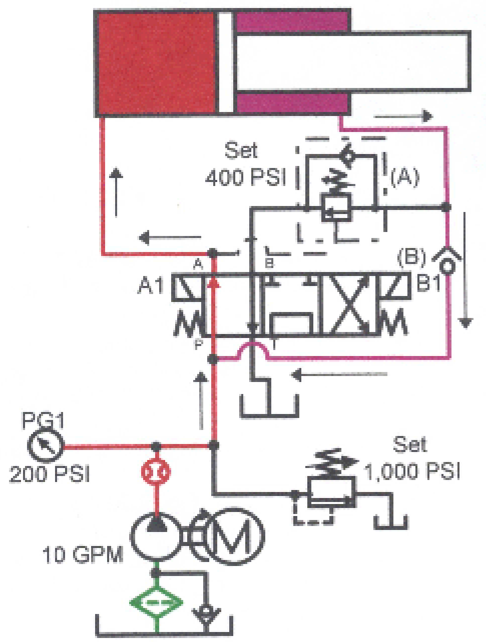


Fig. 17-11. Regeneration circuit that is pressure-activated to full thrust. Shown extending with solenoid A1 energized.

When extra resistance raises cylinder cap-end pressure high enough to pilot sequence valve A open, as in Figure 17-12, oil from the cylinder's rod end goes directly to tank. With this flow going to tank, forward speed decreases as rod-end pressure drops. No backpressure means the cylinder is capable of full tonnage but only moves half as fast. During this part of the cycle, check valve B blocks pump flow from going to tank through sequence valve A.

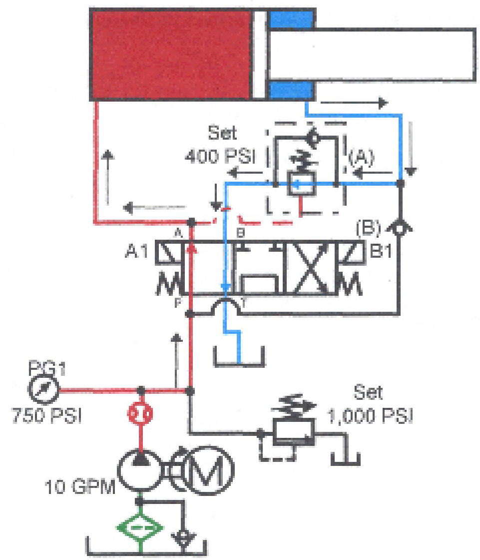


Fig. 17-12. Regeneration circuit, pressure-activated to full thrust. Solenoid A1 energized. Shown extending at full force.

To retract the cylinder, energize solenoid B1 of the directional valve as in Figure 17-12. Pump flow now goes through the directional valve, around the free-flow check valve in sequence valve A, and into the rod end of the cylinder. As the cylinder retracts, oil from the cap end goes to tank through the directional valve.

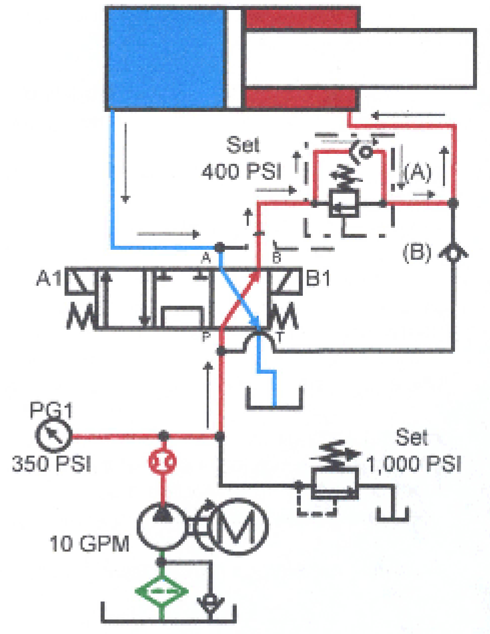


Fig. 17-13. Regeneration circuit, pressure-activated to full thrust. Solenoid B1 energized. Shown retracting.

This sequence circuit provides rapid advance speed, followed by full force to clamp, punch, or operate with different size parts. The cylinder moves rapidly until it contacts a part, then changes to full force. As with any sequence circuit there is no guarantee pressure build-up came from work contact. Anytime pressure increases enough, the circuit automatically shifts out of regeneration into full force. Another down side of this circuit is that the cylinder hits the part at full speed. This impact might cause damage to the machine, tooling, or parts. To avoid these problems, consider a circuit that drops out of regeneration electrically.

Regeneration circuit – solenoid-actuated to full thrust
*Figures 17-14 through 17-17 show a standard regeneration circuit that changes to full thrust and half speed on demand. Shifting a solenoid-operated valve any place in the stroke slows the cylinder's advance and sets up the circuit for full tonnage. Addition of solenoid pilot-*operated 2-*way valve A and check valve B to the circuit makes this possible. When the cylinder reaches a limit switch, 2-way valve A shifts to connect the rod end of the cylinder to tank, eliminating backpressure.*

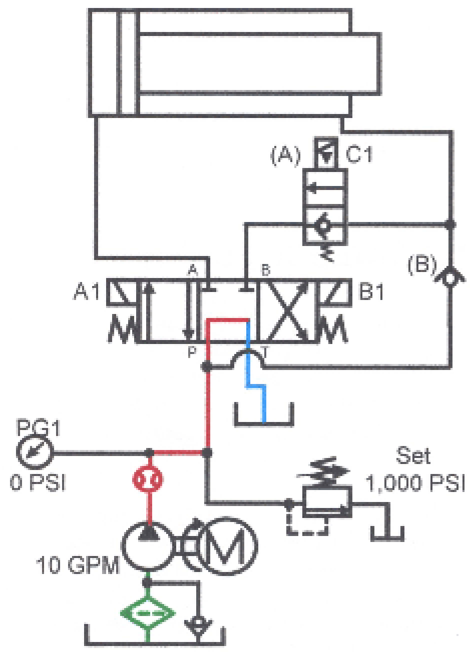


Fig. 17-14. Regeneration circuit, solenoid-activated to full thrust. Shown at rest with pump running.

In Figure 17-15, cylinder is depicted extending in regeneration. During this part of the cycle, oil from the pump goes to the cap end of the cylinder. Oil from the rod end flows freely through check valve B to mix with pump flow and go back into the cylinder. Two-way directional valve A remains closed. The cylinder advances at double speed and half power.

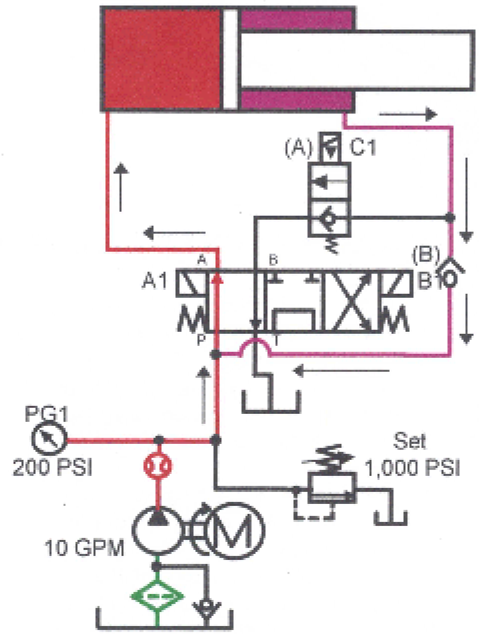


Fig. 17-15. Regeneration circuit, solenoid-activated to full thrust. Solenoid A1 energized. Shown extending in regeneration.

When the moving cylinder makes the limit switch, an electrical signal energizes the solenoid on valve A as in Figure 17-16. Valve A opens and oil from the cylinder's rod end flows directly to tank. With the cylinder's rod-end flow going to tank, pressure drop there allows the cylinder to produce full force while stroking at half speed. During this part of the cycle, check valve B blocks pump flow from going to tank through 2-way valve A. With this arrangement, cylinder speed is slowed before contacting the part because the limit switch drops it out of regeneration. The result: less shock to the system and damage to the machine, tooling, or parts.

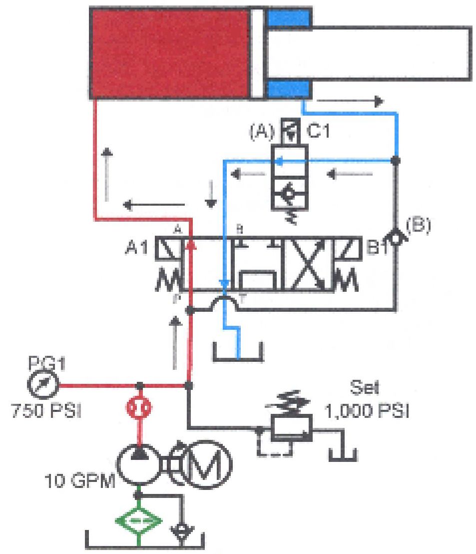


Fig. 17-16. Regeneration circuit, solenoid-activated to full thrust. Solenoids *A1* and *C1* energized. Shown extending at full force.

To retract the cylinder, energize solenoid B1 of the directional valve as in Figure 17-17. Pump flow goes through the 4-way directional valve and flows freely through the check valve in 2-way valve A, then into the rod end of the cylinder. As the cylinder retracts, oil from the cap end flows to tank through the 4-way directional valve. (Note: when using a 2-way valve without reverse free-flow capability, either leave the solenoid on the 2-way valve energized, or install a bypass check around it.)

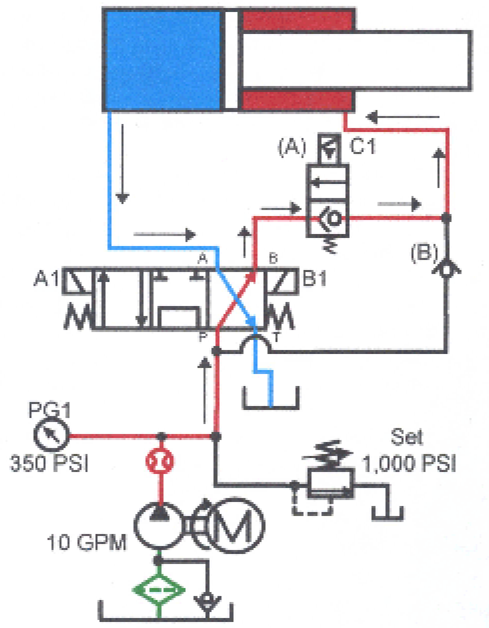


Fig. 17-17. Regeneration circuit, solenoid-activated to full thrust. Solenoid B1 energized. Shown retracting.

Other regeneration circuits

Some of the following circuits use a solenoid or a pressure-control valve to disable regeneration but the placement of the valves is different. Some of these circuits have specific advantages; some are just another way of doing the same thing.

Figures 17-18 through 17-21 show a different way to pipe a solenoid-actuated regeneration circuit. (The results are the same as the previous circuit. There is no particular advantage to this configuration.) Figure 17-18 depicts a standard valve and cylinder circuit with the addition of 2-way normally open directional valve A and check valve B. At rest, the pump unloads through the tandem-center directional valve and 2-way valve A.

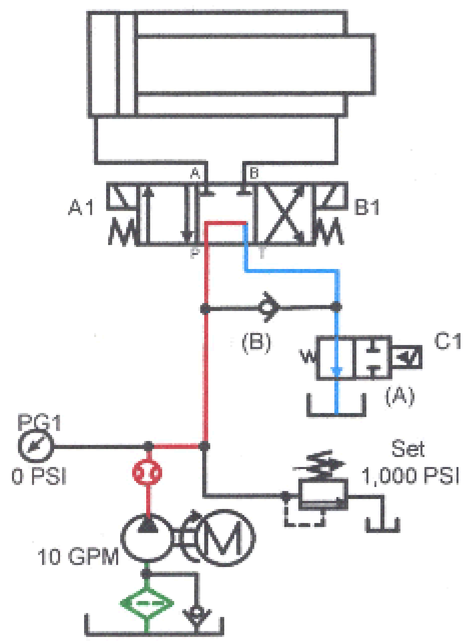


Fig. 17-18. Regeneration circuit, solenoid-activated to full thrust. Shown at rest with pump running.

In Figure 17-19, both solenoid A1 on the 4-way valve and solenoid C1 on the 2-way valve are energized. Pump flow goes through the 4-way valve to the cylinder's cap-end port. Oil returning from the cylinder goes through check valve B to mix with pump flow and regenerates to the cylinder's cap end. Cylinder speed is twice that of a conventional circuit, but force is one half.

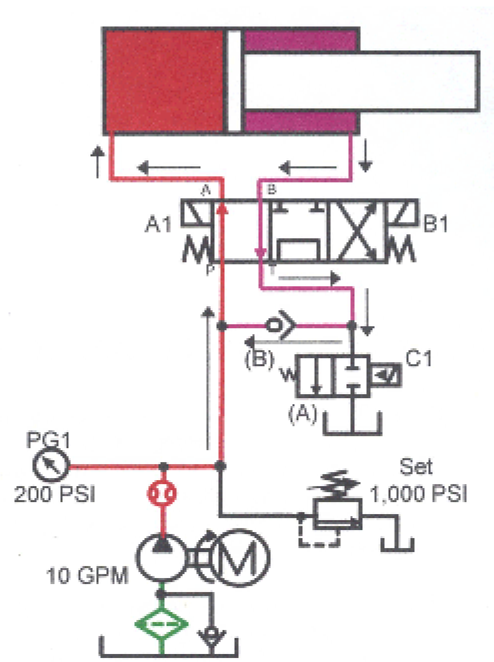


Fig. 17-19. Regeneration circuit, solenoid-activated to full thrust. Solenoids A1 and C1 energized. Shown extending in regeneration.

When the cylinder makes the limit switch, valve A shifts, Figure 17-20, opening a free path for oil from the rod end to flow to tank. During this part of the cycle, check valve B closes to prevent pump flow from going to tank also. Now the cylinder is capable of full thrust at half speed.

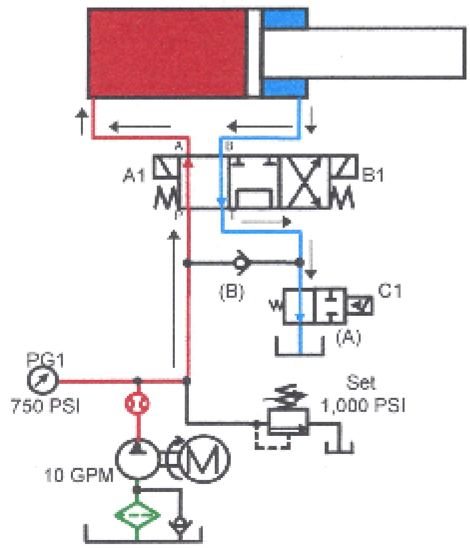


Fig. 17-20. Regeneration circuit, solenoid-activated to full thrust. Solenoid *A1* energized. Shown extending at full force.

In Figure 17-21, solenoid B1 on the 4-way valve is energized to retract the cylinder. Pump oil goes to the rod end of the cylinder while oil from the cap end returns to tank through the 4-way and the 2-way valves.

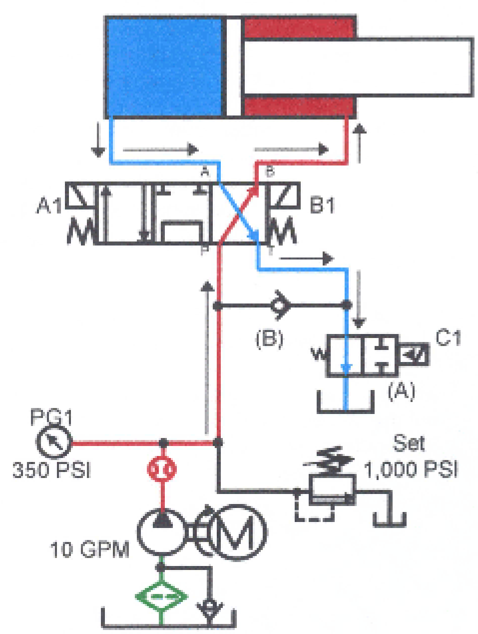


Fig. 17-21. Regeneration circuit,

solenoid-activated to full thrust. Solenoid B1 energized. Shown retracting.

Caution: make sure the tank line of the 4-way directional valve used in this circuit is capable of operating with the amount of backpressure generated.

Regeneration circuit – solenoid-actuated to full thrust

Figure 17-22 shows another way to alter and operate the previous solenoid-actuated regeneration circuit. Add 2-way normally open directional valve A to the standard valve and cylinder circuit. At rest, the pump unloads through the open-center directional valve and valve A. This circuit also has counterbalance valve B to resist a running away load situation. Note that counterbalance valve B has an external drain port. The external drain is necessary on this circuit because outlet flow from the counterbalance valve sees pressure while oil regenerates. Backpressure from regeneration would prevent an internally drained counterbalance valve in this circuit from opening. The cylinder would not extend with the counterbalance valve closed.

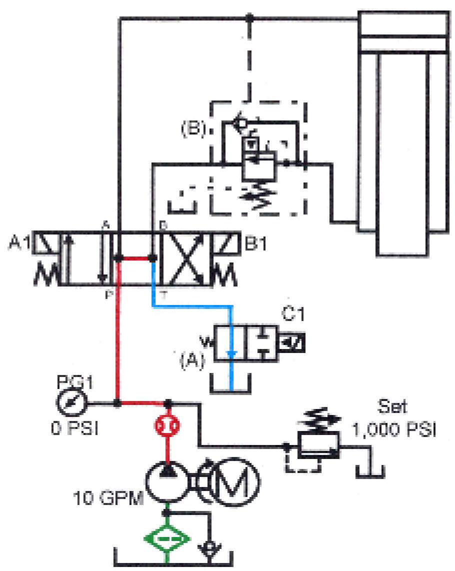


Fig. 17-22. Regeneration circuit, pressure-activated to full thrust. Shown at rest with pump running.

In Figure 17-23, solenoid C1 on directional valve A is energized to start the cylinder forward. Pump flow goes through the 4-way directional valve to the cylinder. Oil returning from the cylinder's rod end flows through the open-center 4-way directional valve, mixes with pump flow, *and enters the cylinder's cap end. Cylinder speed is twice as fast as a conventional circuit while force is only half. When the cylinder contacts the limit switch, it deenergizes solenoid C1 and energizes solenoid A1 on the 4-way directional valve, as in Figure 17-24. Pump oil still flows to the cylinder's cap end, but flow from the rod end now passes freely to tank. The cylinder is now capable of full thrust at half speed.*

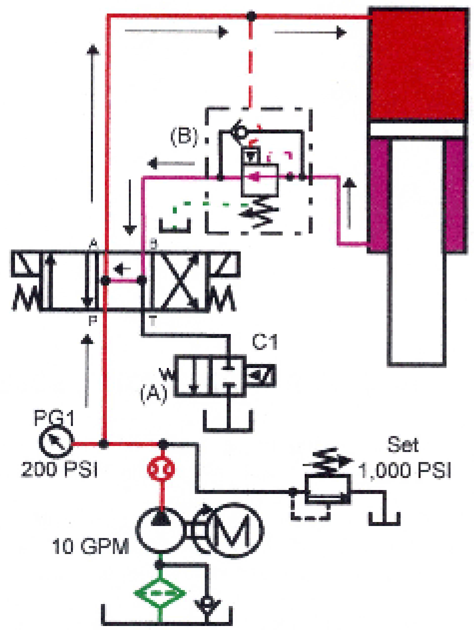


Fig. 17-23. Regeneration circuit, solenoid-activated to full thrust. Solenoid C1 energized. Shown extending in regeneration.

In Figure 17-25, solenoid B1 on the 4-way directional valve is energized. Pump oil now goes to the rod end of the cylinder while oil from the cap end returns to tank through the 4-way and the 2-way directional valves. Note: while this hook-up eliminates the check valve, it complicates the electrical circuit slightly.

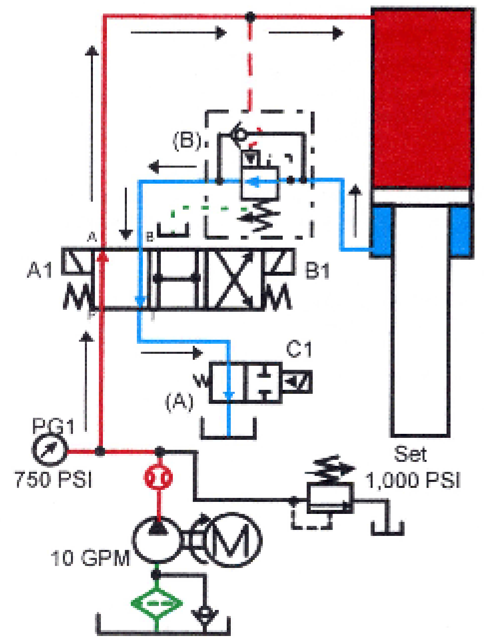


Fig. 17-24. Regeneration circuit, solenoid-activated to full thrust. Solenoid *A1* energized, Shown extending at full force.

CAUTION: Make sure the tank line of the 4-way directional valve used in this circuit is capable of operating with the amount of backpressure generated.

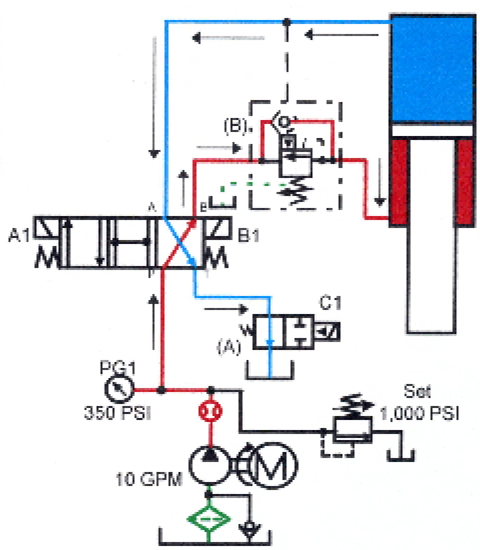


Fig. 17-25. Regeneration circuit, solenoid-activated to full thrust

solenoid B1 energized retracting.

Regeneration circuit – solenoid-actuated to full thrust

Figures 17-26 through 17-29 show another way to pipe a solenoid-actuated regeneration circuit. The results are the same as the previous circuit, but the components it uses eliminate most of the piping and potential leaks.

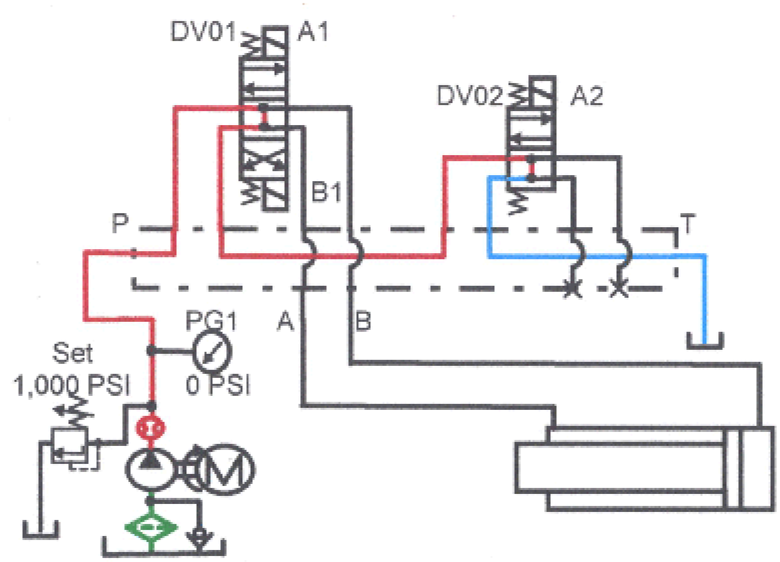


Fig. 17-26. Regeneration circuit, solenoid-activated. Using modular valves on a bar manifold. Shown at rest with pump running.

Figure 17-26 indicates a series bar manifold with two open-center directional valves: a double-solenoid 4-way and a single-solenoid 3-position valve. At rest, the pump unloads through the open centers on the directional valves. The single-solenoid valve takes the place of the 2-way valve in the previous circuit.

Energizing solenoid A2 on the single-solenoid directional valve, as in Figure 17-27, sends pump flow through the open center of the 4-way valve to the cylinder's cap end. Oil returning from the cylinder's rod end mixes with pump flow through the open center of the double-solenoid 4-way and goes into the cylinder's cap end. Speed is twice that of a conventionally piped circuit while the force is only half.

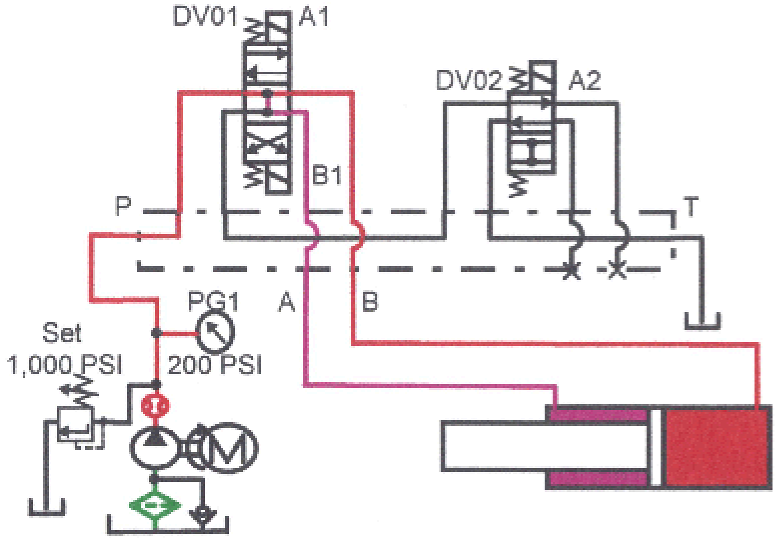


Fig. 17-27. Regeneration circuit, solenoid-activated. Using modular valves on a bar manifold. Solenoid A2 energized. Shown extending in regeneration.

When the cylinder makes the limit switch, Figure 17-28, it deenergizes solenoid A2 on the single-solenoid valve and energizes solenoid A1 on the 4-way directional valve. Pump oil still goes to the cylinder's cap end but flow from the rod end passes freely to tank. Cylinder speed is half of that of regeneration, but now full cylinder force is available.

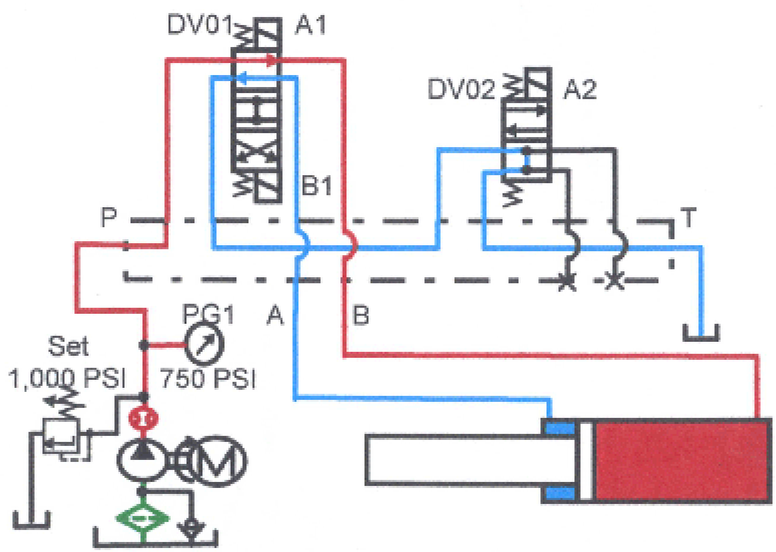


Fig. 17-28. Regeneration circuit, solenoid-activated. Using modular valves on a bar manifold. Solenoid *A1* energized. Shown extending at full force.

Figure 17-29 shows solenoid B1 on the 4-way directional valve energized for retraction. Pump oil goes to the rod end of the cylinder and oil from the cap end returns to tank through the 4-way and the open center of the single-solenoid directional valve.

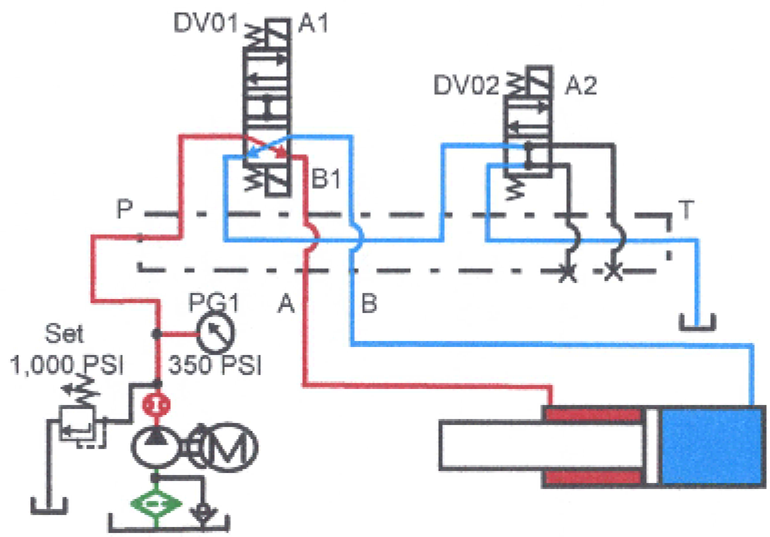


Fig. 17-29. Regeneration circuit, solenoid-activated. Using modular valves on a bar manifold. Solenoid B1 energized. Shown retracting.

The advantages of this circuit are ready availability of the valves, shorter installation time, and fewer potential leaks.

CAUTION: again make sure the tank line of the 4-way directional valve used in this circuit is capable of operating with the amount of backpressure generated.

Regeneration circuit – pressure actuated to full thrust

Large cylinders at high flows need large, expensive directional valves to handle the double flow of a regeneration circuit. Also, when cylinders with regeneration circuits are located at some distance from the directional valve, pressure drop at high regeneration flows might be a problem. The circuit in Figure 17-30 uses pressure-control valves near the cylinder to handle regeneration flows, so that the directional valve only sees pump flow. This circuit drops out of *regeneration after pressure increases at load contact. Valve B opens under low pressure to route rod-end oil to the cap end. Its operation is like a sequence valve so it uses an external drain. In this circuit the external drain goes to tank through the rod-end port. Porting the drain here keeps the sequence valve from opening when the cylinder retracts. Valve C direct oil from the rod end to the cap end until load pressure builds. This valve is similar to a counterbalance valve. (Set pressure on valve C high enough to make sure the cylinder regenerates.) Pressure valve A dumps part of cap-end flow to tank when the cylinder retracts. This keeps part of the high return flow away from the 4-way valve to reduce backpressure during cylinder return.*

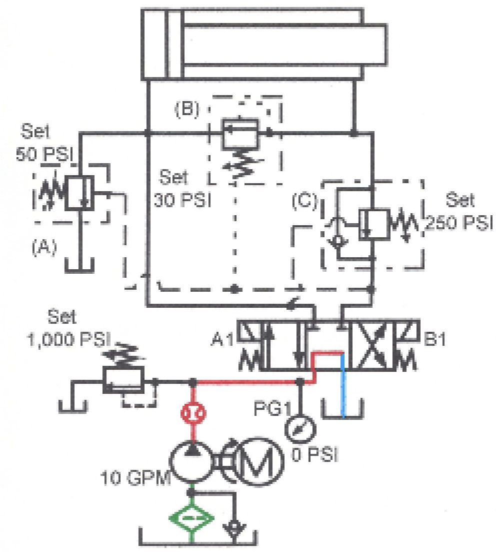


Fig. 17-30. Regeneration circuit, pressure activated to full thrust. Shown at rest, pump running.

Figure 17-31 shows the cylinder extending in regeneration. Energizing solenoid A1 shifts the 4-way directional valve to send oil to the cylinder's cap end. Oil from the cylinder's rod end opens low-pressure sequence valve B and regenerates to the cap end. Counterbalance valve C stays closed because its pressure setting is higher than regeneration pressure. Set sequence valve B as low as possible to minimize backpressure. Set counterbalance valve C high enough to make sure the cylinder regenerates -- and low enough to open easily when the cylinder meets resistance. Valve A stays closed when the cylinder extends because its pilot supply comes from the rod port after counterbalance valve C.

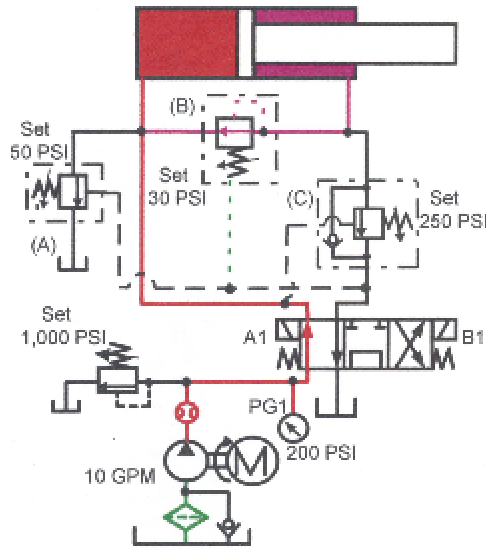


Fig. 17-31. Regeneration circuit, pressure activated to full thrust. Solenoid A1 energized. Shown extending in regeneration.

When the cylinder meets resistance, Figure 17-33, pressure builds up in the cap-end port and opens externally piloted counterbalance valve C. When counterbalance valve C opens, it lets rod-end flow go directly to tank. Pressure drop at sequence valve B lets it close to stop regeneration. This drops all backpressure at the cylinder's rod end so it is capable of full thrust.

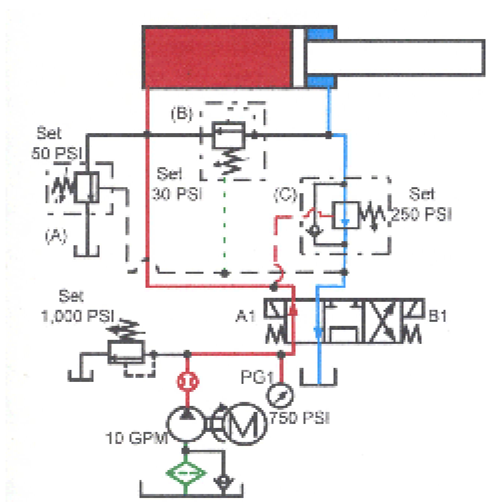


Fig. 17-32. Regeneration circuit, pressure activated to full thrust.

Solenoid A1 energized. Shown extending at full force.

To retract the cylinder, energize solenoid B1 of the 4-way directional valve, as in Figure 17-33. Oil flows through the bypass check of counterbalance C to the cylinder's rod end. Rod-end pressure at the external drain line of sequence valve B holds it closed so fluid cannot bypass to tank. The cylinder retracts with part of the cap-end oil going through the directional valve and excess flow through valve A directly to tank. Pressure-control valve A gets an external pilot signal from the cylinder's rod end line to open it for retraction.

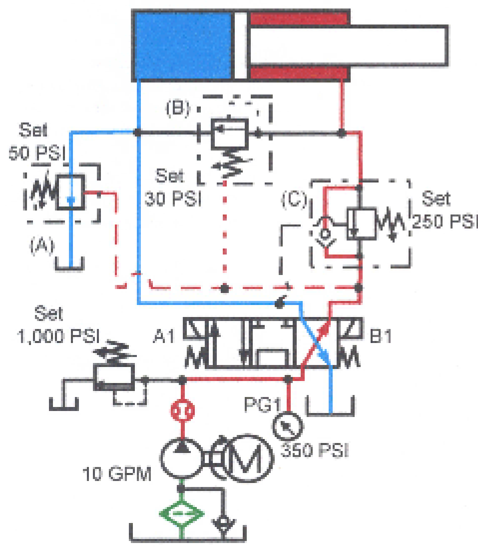


Fig. 17-33 Regeneration circuit, pressure activated to full thrust. Solenoid B1 energized. Shown retracting.

In the at rest condition the cylinder could extend due to external forces because fluid can pass through sequence valve B from rod to cap at any pressure above 30 psi. If this happens, add a pilot-operated check valve in the line between sequence valve B and the rod port.

Regeneration circuit – solenoid-actuated to full thrust

The circuit in Figures 17-34 through 17--37 is the same as in Figures 17-30 through 17-33 except that the change from regeneration to full thrust is solenoid controlled. The circuit in Figure 17-34 uses pressure-control valves near the cylinder to handle the high regeneration *flows, so the directional valve only sees pump flow. This circuit drops out of regeneration after making the limit switch.*

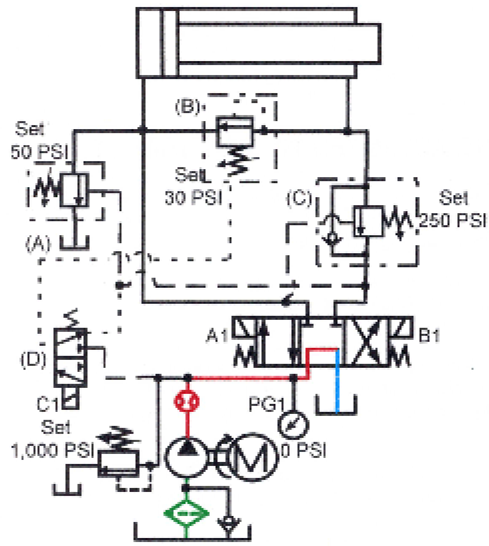


Fig. 17-34. Regeneration circuit, pressure activated to full thrust. Shown at rest, pump running.

Pressure valve B opens under low pressure to route oil from the rod end to the cap end. The valve's operation is like a sequence valve, so use an external drain. In this circuit, the external drain goes through 3-way directional valve D to the rod-end port, then to tank. With the drain here, sequence valve B will not open when the cylinder retracts. Directional valve D closes sequence valve B from an electrical command to stop regeneration.

Valve C directs oil from the rod end to the cap end until pressure rises enough to pilot it open. This valve acts as a counterbalance, so set it high enough to make sure the cylinder regenerates, but low enough to minimize energy loss.

Pressure valve A dumps part of the cap end flow to tank while the cylinder retracts. This keeps high flow away from the 4-way valve to reduce backpressure during the return stroke.

Figure 17-35 shows the cylinder extending in regeneration. Solenoid A1 of the 4-way directional valve shifts it, porting oil to the cylinder's cap end. Oil from the rod end passes through sequence valve B to the cap end. Valve C stays closed because pressure during regeneration is not enough to pilot it open. (Set valve C high enough to make sure the cylinder regenerates, but low enough to operate easily after energizing 3-way valve D. Valve A stays closed while the cylinder extends because its pilot oil comes from the rod port after counterbalance valve C.

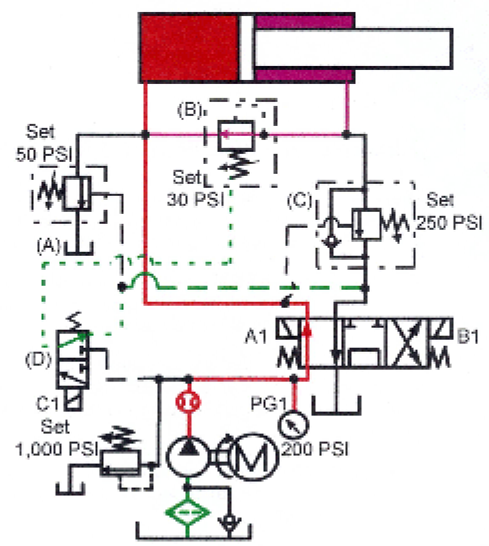


Fig. 17-35. Regeneration circuit, solenoid-activated to full thrust. Solenoid A1 energized Shown extending in regeneration.

When the cylinder contacts the limit switch in Figure 17-36, 3-way directional valve D shifts and sends pump pressure to the drain port of sequence valve B. Pressure in the drain port of sequence valve B closes it to block flow from the rod end to the cap end. Pressure rise in the cap port opens externally piloted counterbalance valve C to give rod-end oil a path to tank. Because oil from the rod end has a free path to tank, the cylinder now is capable of full thrust.

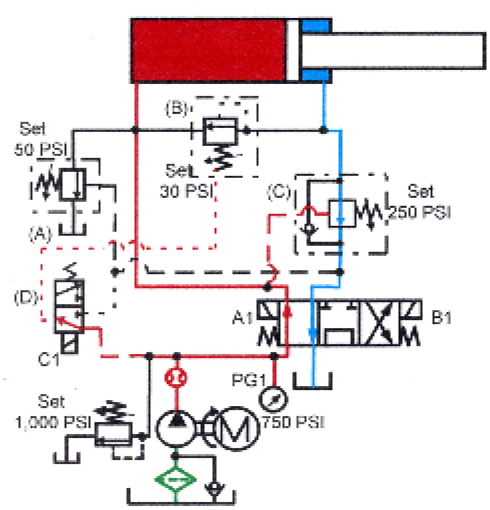


Fig. 17-36. Regeneration circuit, pressure

activated to full thrust. Solenoids *A1* and *C1* energized. Shown extending at full force.

To retract the cylinder, energize solenoid B1 of the 4-way directional, as in Figure 17-37. Oil goes through the bypass check of counterbalance C to the cylinder's rod-end port. The cylinder now retracts, with part of the oil from the cap end going to tank through the directional valve and part through dump valve A. Pilot pressure from the cylinder's rod-end line opens pressure valve A to dump excess fluid.

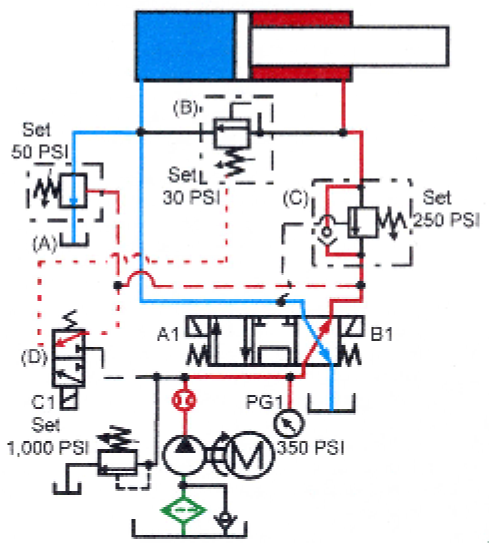


Fig. 17-37. Regeneration circuit, pressure activated to full thrust. Solenoid B1 energized. Shown retracting.

Regeneration circuit with poppet-type cartridge valves

For circuits with flows above 60 to 100 gpm, poppet-type cartridge valves are an option. This type of valve, sometime called a logic valve is a simple pilot-to-close check valve. Because check valves are 2-way valves, four of them are needed to operate a double-acting cylinder. The main advantage of using logic valves in a regeneration circuit is the way they handle high flows. Another reason is, larger cartridges handle double flow during regeneration and cylinder retract in these flow paths only.

With a 150-gpm pump cycling a 2:1 rod cylinder, flow to the cylinder is 150 gpm. Flow from the rod end of the cylinder extending is, 75 gpm, and 300 gpm from the cap end when retracting. With a conventional 4-way valve this 150-gpm-pump circuit requires a 300-gpm valve. A valve this size is expensive, big, and overkill for all but the regeneration and retract part of the cycle.

In the circuit covered by Figures 17-38 through 17--41, size logic valve CV1 to handle 75 gpm, logic valve CV2 for 150 gpm, and logic valves CV3 and CV4 for 300 gpm. Smaller cartridges are less expensive and take less space, while larger cartridges give high flow at low pressure drop.

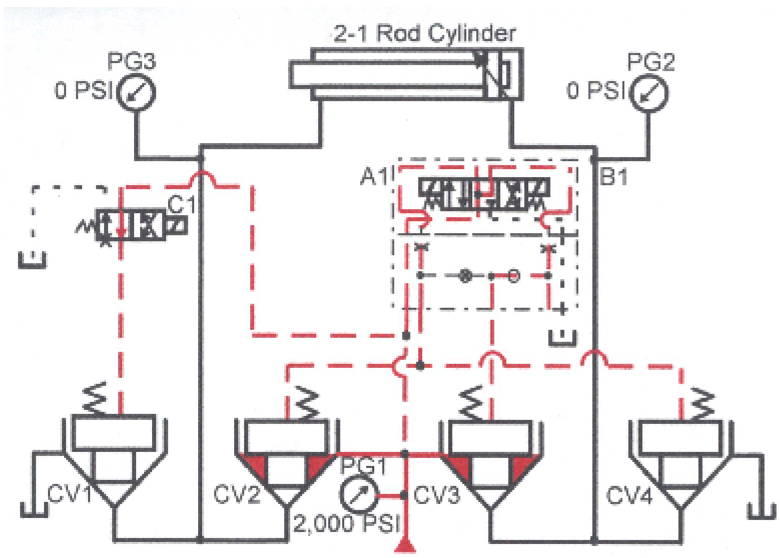


Fig. 17-38. Slip-in cartridge valves in a solenoid-activated to full force regeneration circuit. Shown at rest, pump running.

Figure 17-38 shows the circuit at rest with the pump running. Pilot pressure through two directional valves holds all logic valves closed. A regeneration circuit requires two pilot valves for limit switch changing from regeneration to full force. Use a pressure-control valve in place of the single-solenoid pilot valve when transition from regeneration to full force is due to pressure build up at work contact.

To extend the cylinder in regeneration, energize solenoid A1 of the 4-way double-solenoid directional valve, as in Figure 17-39. This drops pilot pressure on logic valve CV3, allowing pump flow to go to the cylinder's cap end. Oil from the cylinder's rod end cannot get out of logic valve CV1 so pressure increases there until it forces logic valve CV2 to open. Then oil flows through logic valve CV2, mixes with pump flow, and both pass through logic valve CV3 to the cylinder. The cylinder extends at twice speed and half force till it contacts a limit switch.

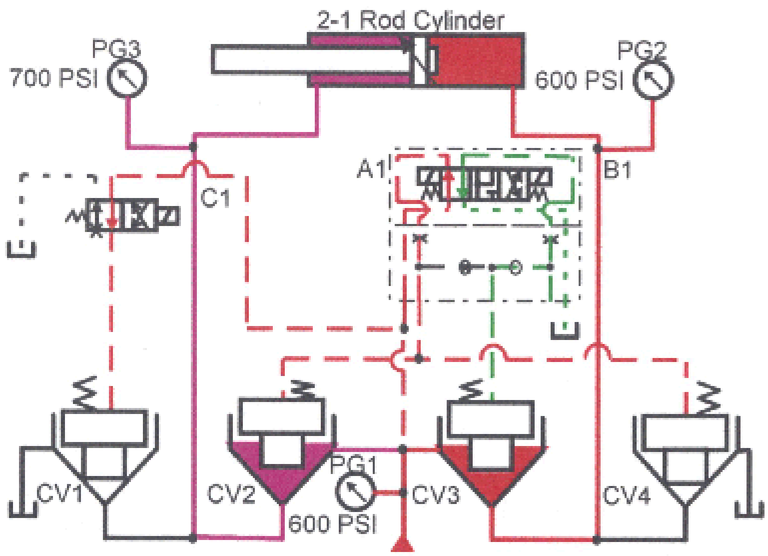


Fig. 17-39. Slip-in cartridge valves in a solenoid-activated to full force regeneration circuit. Solenoid *A1* energized. Shown extending in regeneration.

When the cylinder rod contacts the limit switch, it energizes solenoid C1 of the single-solenoid directional valve to let logic valve CV1 open, as in Figure 17-40. When logic valve CV1 opens, logic valve CV2 closes because regeneration pressure drops. With oil free flowing to tank through logic valve CV1, the cylinder is capable of full thrust at half speed.

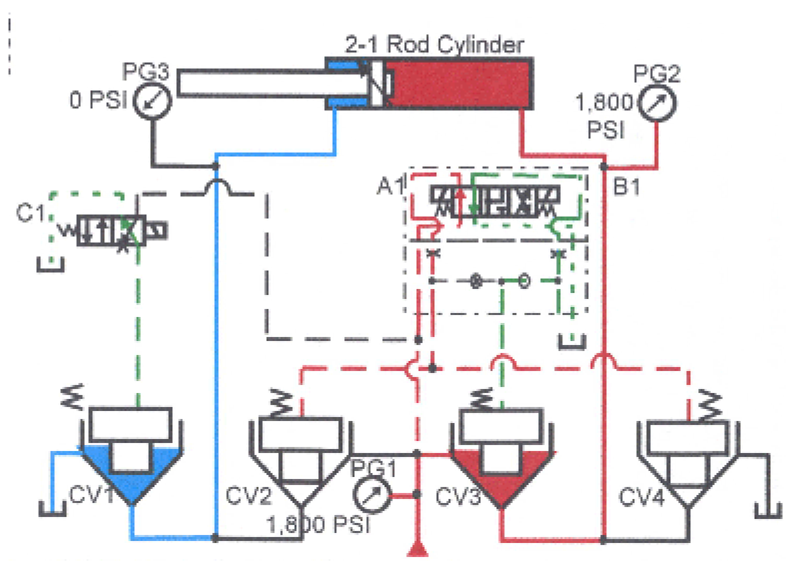


Fig. 17-40. Slip-in cartridge valves in a solenoid-activated to full force regeneration circuit. Solenoids A1 and C1

energized. Shown extending at full force.

Figure 17-41 shows solenoid B1 of the double-solenoid directional valve energized and the logic valve positioned to retract the cylinder. With solenoid B1 energized, pilot pressure drops on logic valve CV2 and CV4. Logic valves CV2 and CV4 open to pass pump flow to the cylinder's rod end while cap end oil goes to tank.

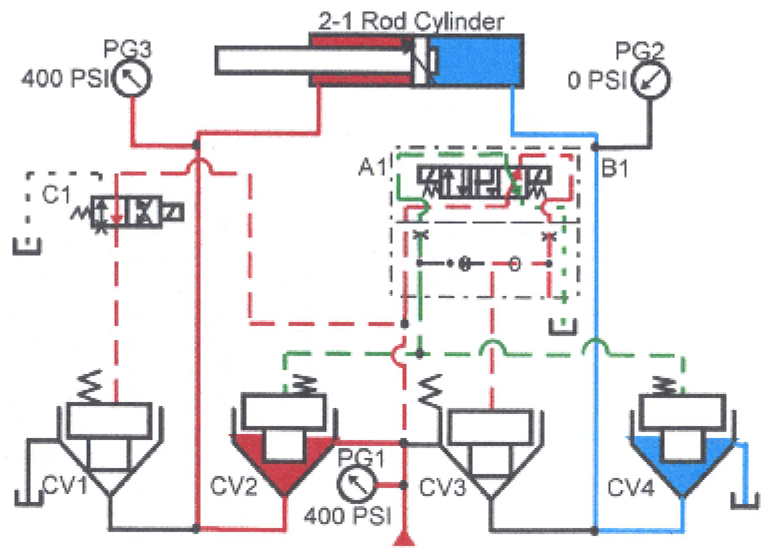


Fig. 17-41. Slip-in cartridge valves in a solenoid-activated to full force regeneration circuit. Solenoid B1 energized. Shown retracting.

A high-flow circuit normally costs less, and has lower pressure drop when using cartridge valves.

Motor-type flow divider in full-time regeneration circuit

Figures 17-42 through 17-44 show a unique regeneration circuit using a motor-type flow divider. This regeneration circuit works best on small-rod cylinders and produces exactly twice the speed on double-rod cylinders and hydraulic motors.

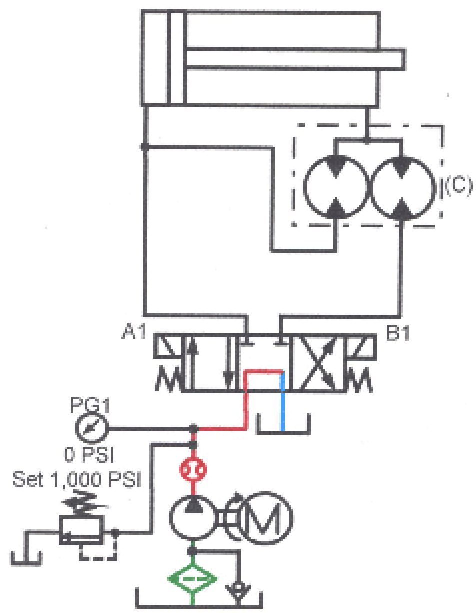


Fig. 17-42. Full-time regeneration circuit using a motor-type flow divider. Shown at rest, pump running.

A motor- type flow divider consists of two or more hydraulic motors in the same housing mounted on a common shaft so they all rotate at the same speed. The motors have a common inlet but separate outlets. If the motors are the same size, any fluid to the inlet divides equally at the outlets. Motors with different cir provides proportionally different outlet flows.

Normally, flow-divider circuits split the inlet flow to synchronize actuator movements, but they also can be used to increase flow for a regeneration circuit.

Figure 17-42 shows a regeneration circuit with a full-time motor-type flow divider in the at-rest condition. Piped between the cylinder's rod end-port and the directional valve is equal-motor flow divider C. Its normal inlet port connects to the cylinder, one outlet connects to the directional valve, and the other outlet is teed into the cylinder's cap-end line.

Figure 17-43 shows solenoid A1 energized so flow from the pump goes past the teed-in flow-divider line to the cylinder's cap end. As the cylinder extends, oil from the rod end enters the flow divider. The flow divider splits that oil. Half flows to tank at no pressure, and the other half flows to the cylinder's cap-end tee at pressure high enough to mix it with pump flow. As the cylinder starts forward, speed quickly increases to almost double. Maximum cylinder speed directly relates to the rod size. The larger the rod, the slower the speed. With a double-rod end cylinder, speed exactly doubles. As with any regeneration circuit, speed increases but force decreases.

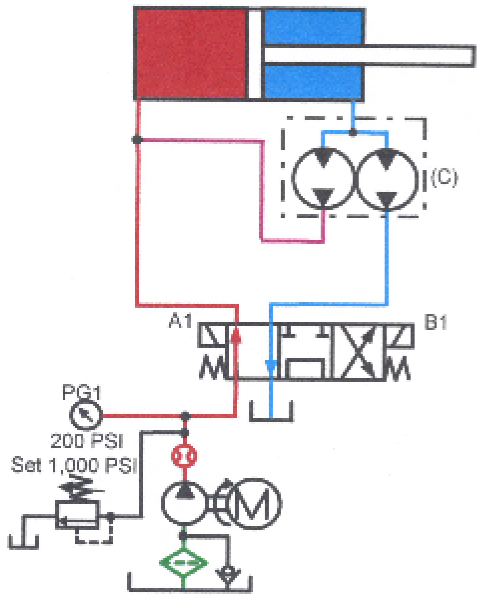


Fig. 17-43. Full-time regeneration circuit using a motor-type flow divider. Solenoid A1 energized. Shown extending in regeneration.

Figure 17-36 shows the cylinder retracting. Energizing solenoid B1 shifts the 4-way directional valve to send pump flow to one outlet of the flow divider. Both of the motors in the flow divider turn at the rate of flow from the pump. During this part of the cycle, the motor with its inlet teed into the cap-end line acts as a pump. Pump flow, plus the same flow from the second motor, makes the cylinder retract twice as fast as a conventional circuit. Again, cylinder thrust is only half that of a conventional circuit.

This flow-divider regeneration circuit doubles the speed without making the pump work harder. Size the pump, valve, tank, and piping up to the regeneration circuit according to pump flow. The only high flows are at or very near the cylinder.

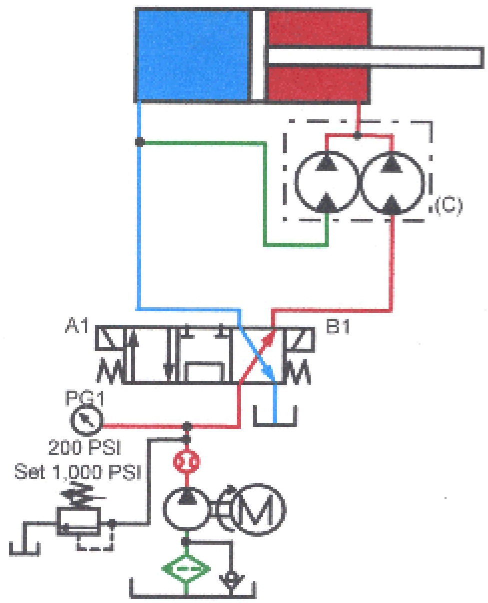


Fig. 17-44. Full-time regeneration circuit using a motor-type flow divider. Solenoid B1 energized. Shown retracting.

Using a motor with a higher cir on the left side of the flow divider increases speed even more. The limit is when pressure to run the cylinder at the faster rate, exceeds the relief valve setting. When using unmatched motors, make sure the line from the cylinder cap end to the motor will handle the higher suction flow.

Motor-type flow-divider regeneration circuit – pressure actuated to full thrust

When it is necessary to get out of regeneration and into full thrust, add other valving to the motor-type flow-divider regeneration circuit. Figure 17-45 shows such a regeneration circuit at rest. Piped between the 4-way directional valve and the cylinder is equal flow divider C. Its normal inlet port connects to the cylinder; one outlet connects to the directional valve; the other outlet passes flow through pilot-operated check valve E to a tee in the cylinder's cap-end line. Check valve E gets its pilot signal from the cylinder's rod-end line before the flow-divider port. Teed into the line between the flow divider and check valve E is the inlet to sequence valve D. The outlet of sequence valve D tees into the cylinder's rod-end line. Sequence valve D is internally drained and gets its external pilot signal from the cylinder's cap-end line.

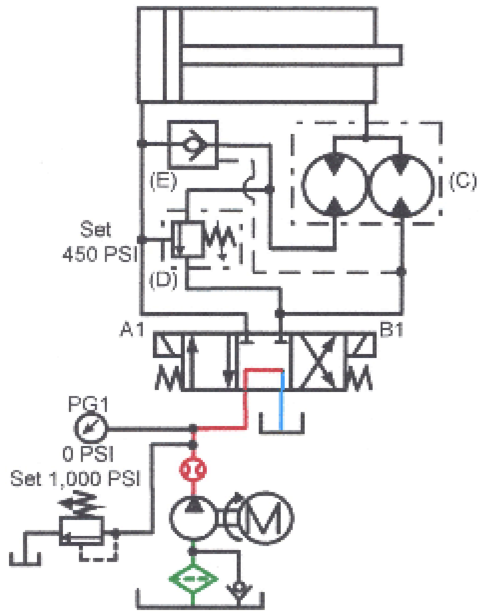


Fig. 17-45. Pressure activated to full thrust regeneration circuit using a motor-type flow divider. Shown at rest, pump running.

Figure 17-46 shows solenoid A1 energized, with flow from the pump going past the teed in the flow-divider line to the cylinder's cap end. As the cylinder extends, oil from the rod end enters the flow divider. The flow divider splits that oil. Half goes to tank at no pressure and half flow freely through pilot-operated check E, then goes to the cylinder's cap-end tee at a pressure high enough to mix with pump flow. As the cylinder starts forward, speed quickly increases to almost double. Maximum cylinder speed directly relates to the rod size. The larger the rod, the slower the speed. With a double-rod end cylinder, speed exactly doubles. As with any regeneration circuit, speed increases but force decreases.

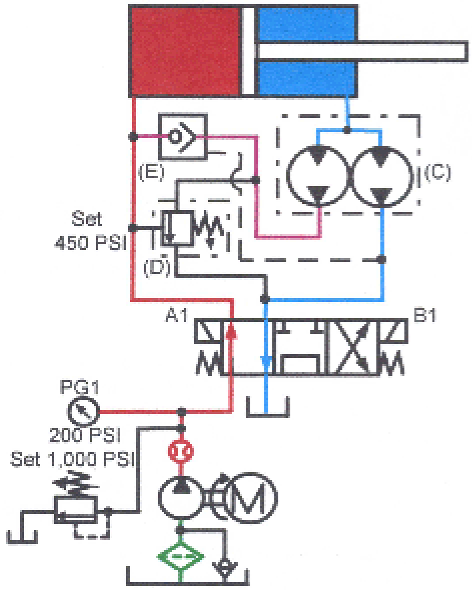


Fig. 17-46. Pressure activated to full thrust regeneration circuit using a motor-type flow divider. Solenoid A1 energized. Shown extending in regeneration.

When the cylinder meets resistance, pressure builds. Figure 17-47 shows the cylinder against the work. Pressure build-up in the cap-end line pilots sequence valve D open. When sequence valve D opens, oil from both sides of the flow divider returns to tank at no pressure. At the same time, pilot-operated check E closes to keep the pump from relieving to tank. With the rod end of the cylinder hooked to tank and the pump feeding the cap end, the cylinder has full thrust.

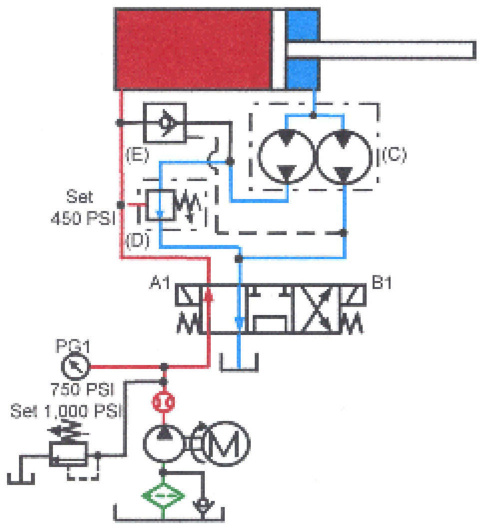


Fig. 17-47. Pressure activated to full thrust regeneration circuit using a motor-type flow divider. Solenoid A1 energized. Shown extending at full force.

In Figure 17-48, the cylinder is retracting. Energizing solenoid B1 of the 4-way directional valve sends pump flow to one outlet of the flow divider. Both of the motors in the flow divider turn at the rate of flow from the pump. During this part of the cycle, the motor with its inlet teed into the cap end line becomes a pump. Pilot pressure from the cylinder's rod end port, opens pilot operated check valve E to allow this flow. Pump flow, plus the same flow from the second motor, makes the cylinder retract twice as fast as a conventional circuit. However, cylinder thrust is only half that of a conventional circuit.

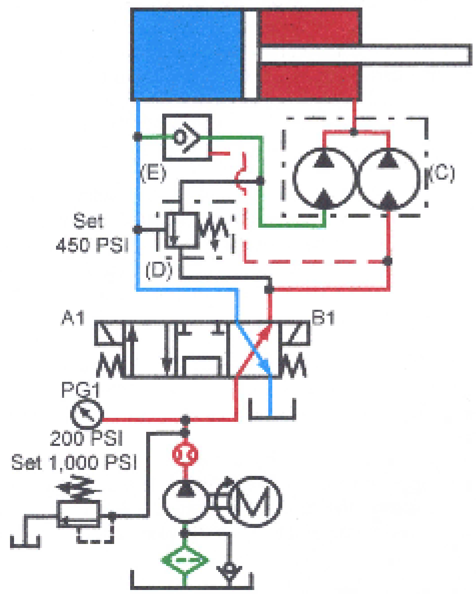


Fig. 17-48. Pressure activated to full thrust regeneration circuit using a motor-type flow divider. Solenoid B1 energized. Shown retracting.

Motor-type flow-divider regeneration circuit -- solenoid actuated to full thrust

When it is necessary to get out of regeneration and into full thrust, add other valving to the motor-type flow-divider regeneration circuit.

The regeneration circuit in Figure 17-49 is solenoid actuated to full thrust. Piped between the 4-way directional valve and the cylinder is equal flow divider C. Its normal inlet port connects to the cylinder; one outlet connects to the directional valve; and the other outlet passes flow freely through pilot-operated check valve E to a tee in the cylinder's cap-end line. Check valve E gets its pilot signal from the cylinder's rod-end line upstream from the flow-divider port. Teed into the line between the flow divider and pilot-operated check E is the inlet to normally closed valve D. The outlet of directional valve D tees into the cylinder's rod-end line. Directional valve D is direct-solenoid-operated and does not need a pilot supply.

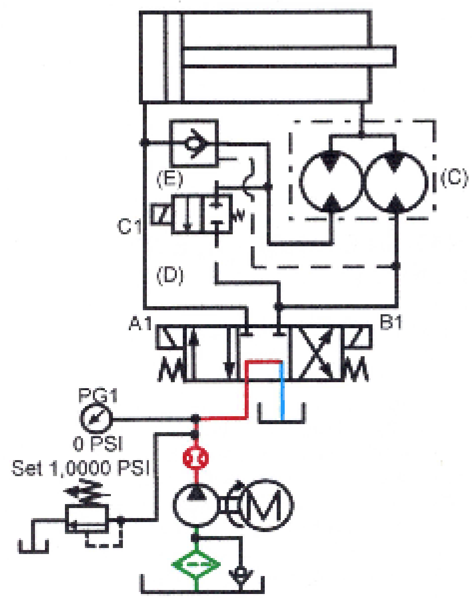


Fig. 17-49. Solenoid-activated to full thrust regeneration circuit using a motor-type flow divider. Shown at rest, pump running.

Energizing solenoid A1 of the main directional valve, as in Figure 17-50, sends flow from the pump to the cylinder past the teed-in flow-divider line. As the cylinder extends, oil from the rod end enters the flow divider. Oil entering the flow divider splits -- half of it going to tank at no pressure, and half of it flowing freely through pilot-operated check valve E to the cylinder's cap end at a pressure high enough to mix with pump flow. As the cylinder moves, speed almost doubles. The amount of speed increase is directly related to the rod size. The larger the rod, the slower the speed. A double-rod end cylinder would go exactly twice as fast. As with any regeneration circuit, the speed increases but force decreases.

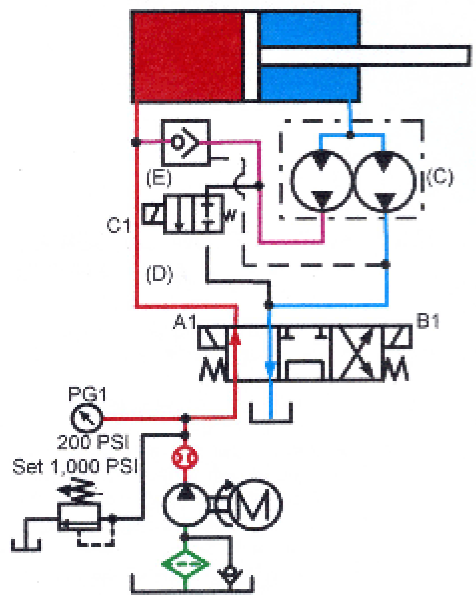


Fig. 17-50. Solenoid-activated to full thrust regeneration circuit using a motor-type flow divider. Solenoid A1 energized. Shown extending in regeneration.

When the cylinder rod contacts the limit switch in Figure 17-51, the switch sends an electrical signal to 2-way valve D, causing it to shift open. When valve D opens, oil from both sides of the flow divider returns to tank at no pressure. The cylinder slows before contacting the work with this arrangement. At the same time, pilot-operated check E closes to keep the pump from bypassing to tank also. With the rod end of the cylinder connected to tank and the pump feeding the cap end, the cylinder has full thrust.

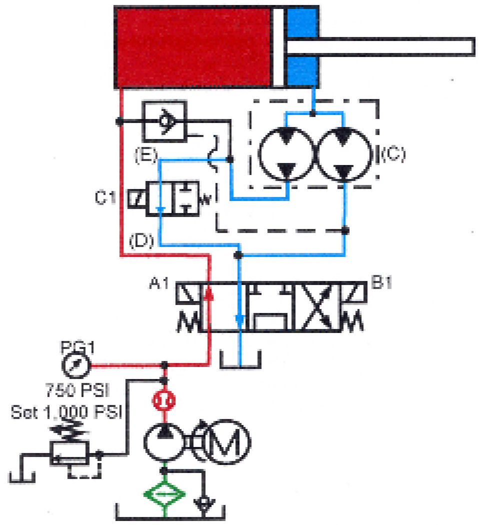


Fig. 17-51. Solenoid-activated to full thrust regeneration circuit using a motor-type flow divider. Solenoids A1 and C1 energized. Shown extending at full force.

Figure 17-52 shows the cylinder retracting. Energizing solenoid B1 of the 4-way directional valve sends pump flow to one outlet of the flow divider. Both of the motors in the flow divider turn at the rate of flow from the pump. During this part of the cycle, the motor with its inlet teed into the cap-end line becomes a pump. Pilot pressure from the cylinder's rod-end port opens check valve E to allow this flow to pass. Pump flow, plus the same flow from the second motor, makes the cylinder retract twice as fast as a conventional circuit. However, cylinder thrust is only half that of a conventional circuit.

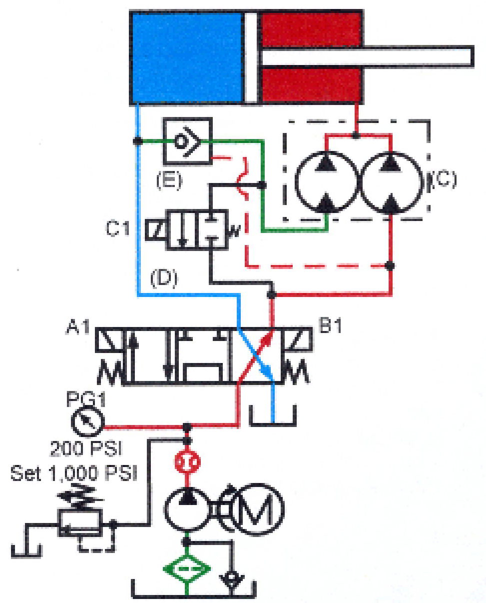


Fig. 17-52. Solenoid-activated to full thrust regeneration circuit using a motor-type flow divider. Solenoid B1 energized. Shown retracting.

Figure 17-52 shows a purchased manifold regeneration circuit from Sun Hydraulics. It is available as an inline manifold that mounts close to the cylinder or as a sandwich module that goes between a directional valve and sub-plate. Although operation of both units is the same, the inline type in these figures has less pressure drop because it normally mounts closer to the cylinder.

Figure 17-53 shows a 2:1 rod-diameter cylinder controlled by a 4-way directional valve. Regeneration manifold A, mounted close to the cylinder, gives double extension speed. The regeneration manifold contains an internal and external pilot-operated counterbalance valve and a pilot-to-close check valve.

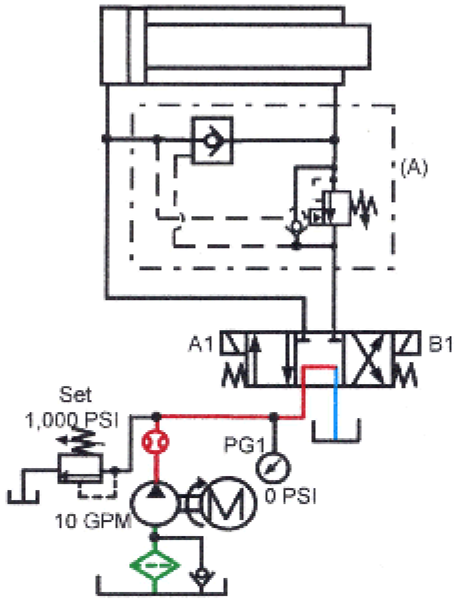


Fig. 17-53. Regeneration circuit, pressure activated to full thrust, Shown at rest, pump running.

Figure 17-54 shows solenoid A1 of the directional valve energized, sending oil to the cylinder's cap end. As the cylinder extends, oil from the rod end regenerates to the cap-end line through the pilot-to-close check valve. Set the counterbalance valve pressure high enough to keep oil from going to tank while the cylinder approaches the work. The cylinder advances at approximately twice normal speed but only half of its normal force.

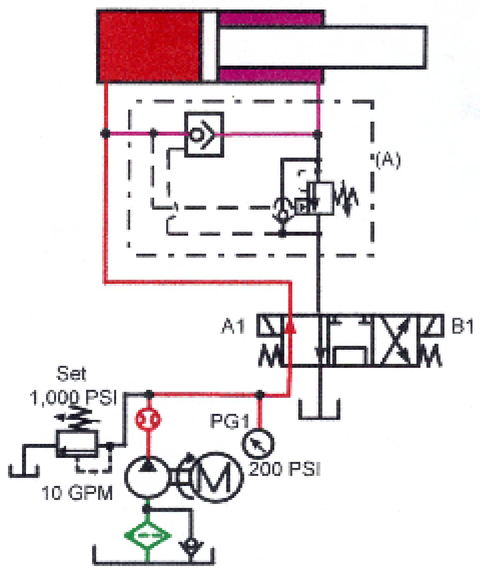


Fig. 17-54. Regeneration circuit, pressure activated to full thrust. Solenoid A1 energized. Shown extending in regeneration.

When the cylinder contacts the work, Figure 17-55, pressure rises in the cylinder's cap end. When pressure increases enough, the counterbalance valve opens the rod end to tank and the pilot-to-close check valve shuts. The cylinder is now capable of full force although the speed is half of regeneration travel. The cylinder advances at working force until it stalls or the directional valve shifts to stop or return it.

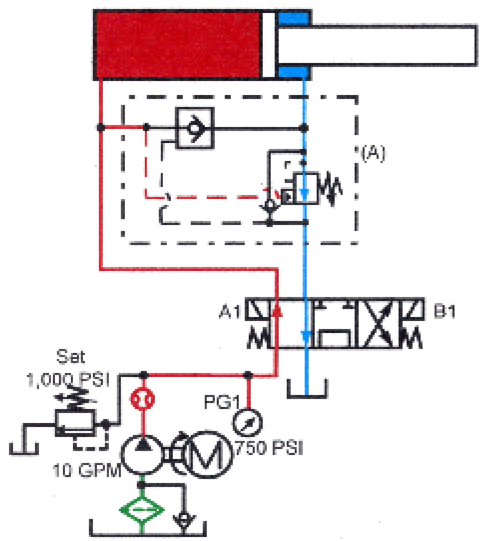


Fig. 17-55. Regeneration circuit, pressure activated to full thrust. Solenoid *A1* energized. Shown extending at full force.

Energizing solenoid B1 of the directional valve sends oil from the pump to the cylinder's cap end, as in Figure 17-56. Oil bypasses the counterbalance valve through its integral check valve and flows to the cylinder's rod end. As the cylinder retracts, pilot pressure from the rod-end line holds the pilot-to-close check valve shut so fluid cannot return to tank through the cap-end line.

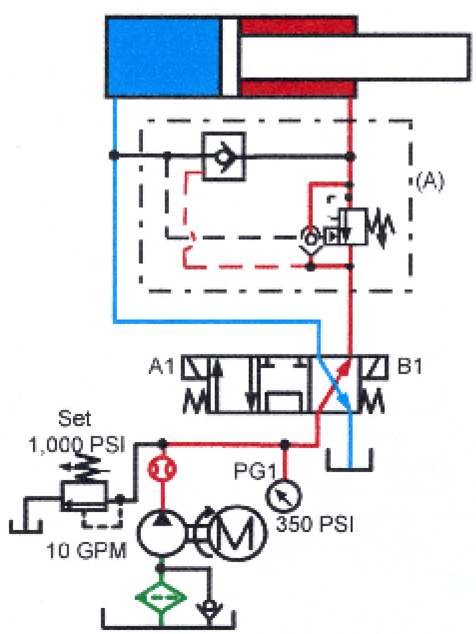


Fig. 17-56. Regeneration circuit, pressure activated to full thrust. Solenoid B1 energized. Shown retracting.

This package is convenient and easy to use. It eliminates many leakage points, is easy to setup, and simple to troubleshoot.

Two-hand, anti-tie-down circuit with manual regeneration

Figures 17-57 through 17-62 show a manually operated regeneration circuit using a pair of lever-controlled directional valves. To make the cylinder move with this circuit, the operator must have both hands on the levers. Regeneration takes place when the operator shifts the valves as explained below.

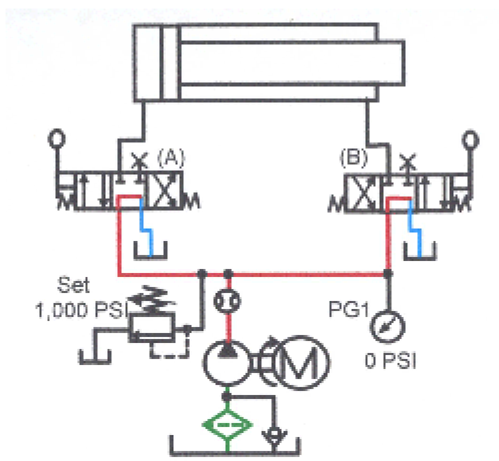


Fig. 17-57. Manual two-hand anti-tie-down circuit with regeneration. Shown at rest with pump running.

Figures 17-58 and 17-59 show how the circuit reacts when the operator shifts only one valve. Pump oil always has a path to tank through the idle valve so the cylinder does not move. Shifting one of the valves and then tying it down allows the cylinder to extend when the second valve is shifted. However, the cylinder cannot retract until the operator releases the tied-down valve, and shifts it to the retract position.

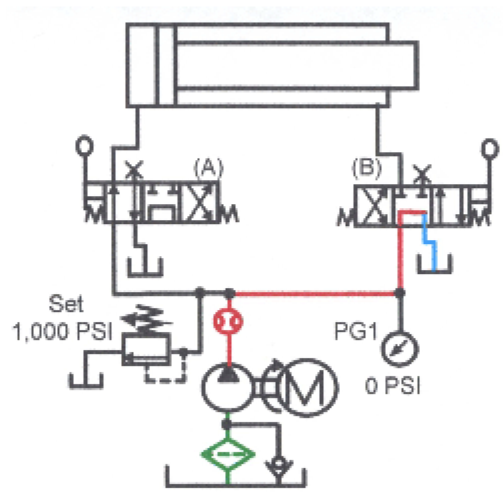


Fig. 17-58. Manual two-hand anti-tie-down circuit with regeneration. Shown with valve *A* actuated.

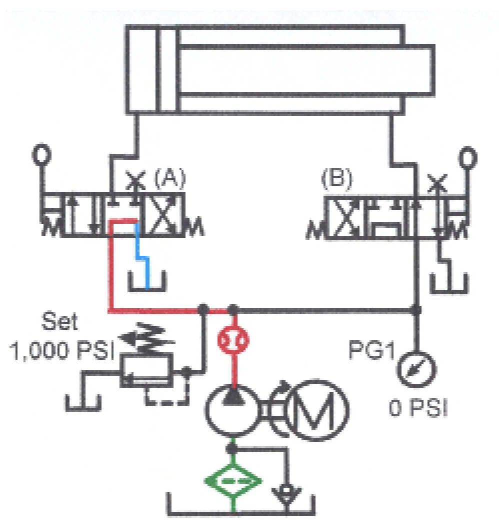


Fig. 17-59. Manual two-hand anti-tie-down circuit with regeneration. Shown with valve B actuated.

In Figure 17-60, the valves are shifted to extend the cylinder at regeneration speed. Valve sends pump oil to the cylinder's cap end while valve B directs rod-end oil to mix with the pump flow. During this part of the stroke the cylinder extends at half force and double speed. The operator controls when the cylinder goes out of regeneration into slow speed and full force.

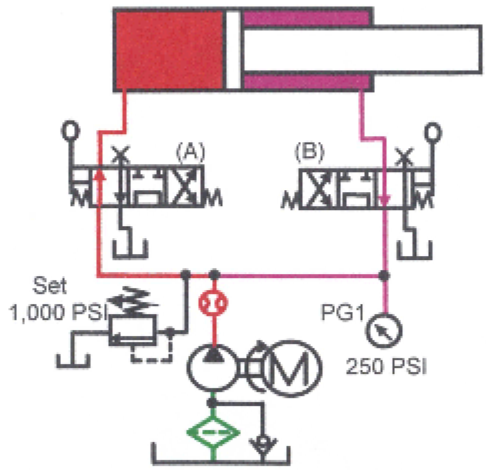


Fig. 17-60. Manual two-hand anti-tie-down circuit with regeneration. Shown extending in regeneration.

Figure 17-61 shows the valves shifted to give full thrust. Pump flow goes to the cylinder's rod end and oil from the cap end goes directly to tank.

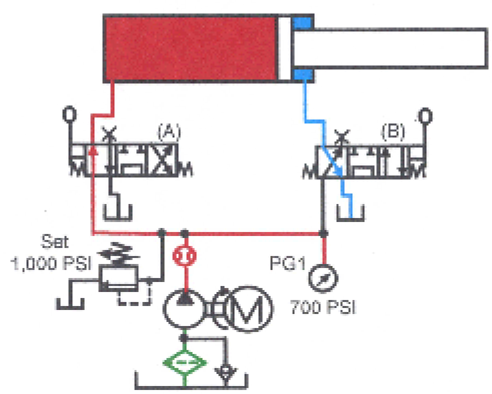


Fig. 17-61. Manual two-hand anti-tie-down circuit with regeneration. Shown extending at full force.

To use this circuit for the anti-tie down feature only, add a check valve in the pump line to valve B, The check valve allows pump flow to go to valve B, but blocks regenerating flow from the cylinder from going to the other manual valve.

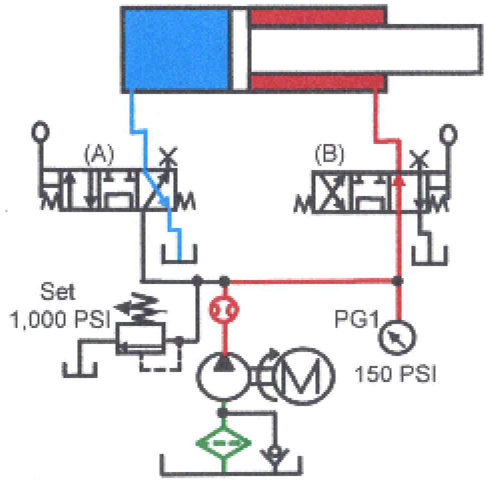


Fig. 17-62. Manual two-hand anti-tie-down circuit with regeneration. Shown retracting.

Some machine actions require rotary motion for only a portion of a turn. Using a hydraulic motor to perform a partial-turn function is expensive and it is difficult to accurately stop a motor at a specified degree of rotation. A clevis-mounted cylinder, attached to an arm and keyed to a shaft, produces rotary action, but is limited to 90° or less. At 90° rotation, a cylinder/lever arrangement has half torque or less when it starts and nears the end of stroke. To obtain partial- or multiple-turn rotary action and/or accurate stopping of rotary output, use one of the rotary actuators shown in this chapter. Figure 19-1 pictures the symbols for air- and hydraulic-operated rotary actuators.

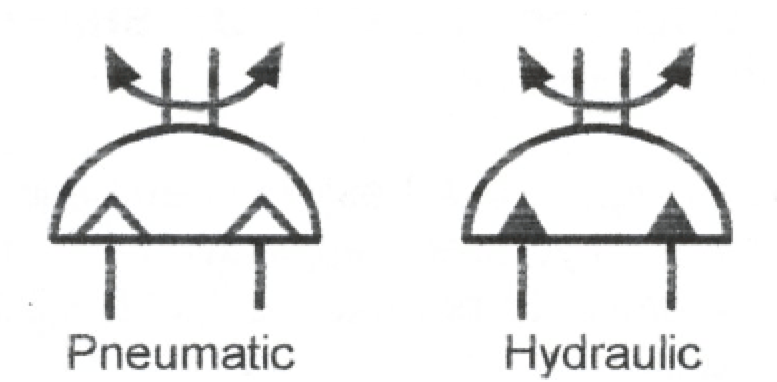


Fig. 19-1. Rotary actuator symbols.

Figure 19-2 provides simplified cutaway views of vane-type rotary actuators. The figure depicts both single- and double-vane-types. The vanes attach to an output shaft and have seals around their periphery. When fluid pressure on a given vane area pushes it through the body cavity, the output shaft turns with a given torque. The maximum rotation of vane rotary actuators is limited to approximately 280° in a single-vane model and approximately 100° in the double-vane configuration.

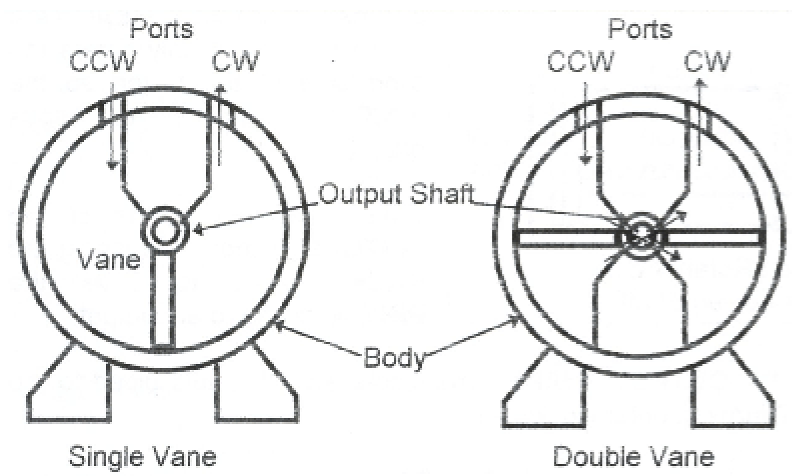


Fig. 19-2. Vane-type rotary actuator.

A double-vane rotary actuator sends fluid to the push side of the opposite vane through drilled passages in the shaft, as shown by dashed lines and arrows. Pressurized fluid at the CW port turns the output shaft clockwise. Pressurized fluid at the CCW port turns the output shaft counterclockwise.

Most vane-type rotary actuators operate at lower pressure and torque limits of 2500 to 5000 in. lb. Some manufacturers do make units that operate at up to 3000 psi, with torque in excess of 700,000 in. lb.

Vane-type rotary actuators have no effective way of internally cushioning or limiting the degree of rotation. An external method must be used to limit rotation or cushion the load. Some manufacturers offer a valve and stroke-limiting package that makes rotation degrees adjustable and gives variable deceleration and cushioning. Check manufacturers' catalogs for more information on these packages.

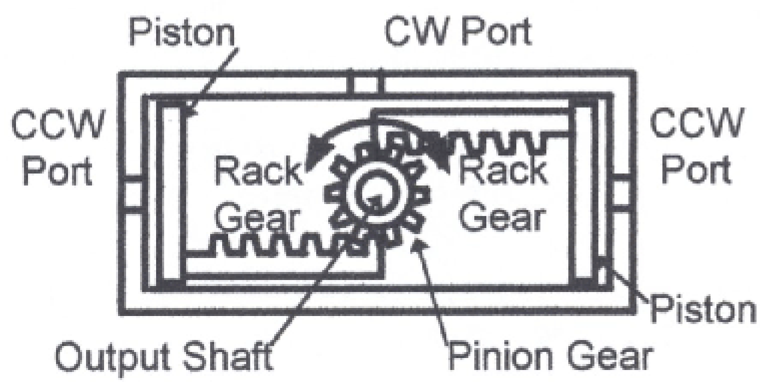


Fig. 19-3. Rack-and-pinion rotary actuator with sealed output shaft.

Figure 19-3 illustrates one design of a rack-and-pinion type rotary actuator. This cutaway view shows a double-rack design that has fluid in the area where the pinion runs. This configuration requires a high-pressure shaft seal but assures that the rack and pinion is well lubricated. With fluid piped to the CW port, the output shaft turns clockwise. With fluid piped to the CCW port, the output shaft turns counterclockwise. This design works best in pneumatic or low-pressure hydraulic applications. The torque range usually does not exceed 2500 to 3500 in. lb.

The cutaway view in Figure 19-4 shows another style rack-and-pinion type rotary actuator. This design has opposing pistons with a rack gear as the piston rod. Fluid only enters the blind side of the piston so the pinion shaft never sees pressure. When fluid enters one of the piston cavities, that piston moves, pushing the rack gear to drive the pinion, and producing rotary output. With *fluid piped to the CW port, the output shaft turns clockwise. With fluid piped to the CCW port, the output shaft turns counterclockwise.*

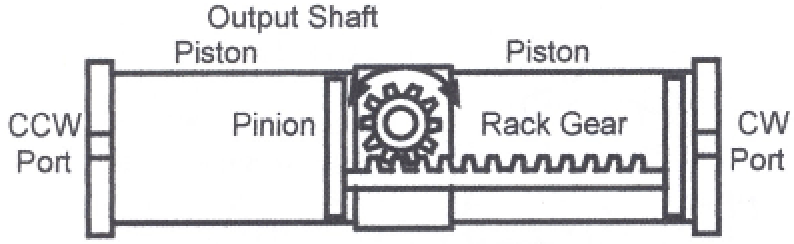


Fig. 19-4. Rack-and-pinion rotary actuator with low-pressure sealed shaft.

The rack-and-pinion design rotary actuators shown in Figure 19-4 are available with a second rack gear and pistons mounted on the opposite side of the pinion. This double-piston setup produces twice the torque in both directions of rotation.

Optional stroke limiters select a precise stopping point at any degree of rotation less than maximum. Also available are cushions that decelerate rotation speed near the end of the stroke. Cushions are adjustable and not affected by the stroke limiter option in the same rotary actuator. This type rotary actuator is available with an optional hollow output shaft.

Rack-and-pinion rotary actuators operate equally well on pneumatic or hydraulic pressure (up to 3000 psi). They generate torque up to 200,000 in. lb for air service, and up to 15,000,000 in. lb and higher for hydraulic service. Output shafts turn any number of degrees up to five rotations according to piston and rack gear size.

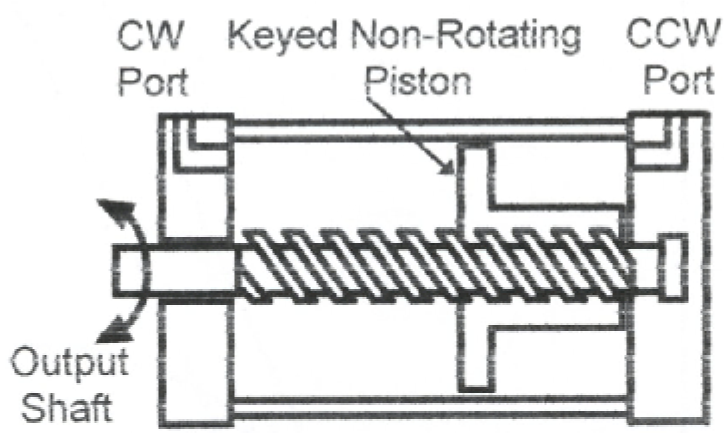


Fig. 19-5. Simplified cutaway view of spiral -shaft rotary actuator.

Figure 19-5 shows a simplified cutaway view of a spiral-shaft rotary actuator. (There are several variations of spiral-type rotary actuators, but all function similar to this diagram.) The spiral-shaft rotary actuator has a keyed, non-rotating piston with a hollow rod. The hollow rod has a set of internal spiral grooves that mesh with the spiral shaft. The spiral-grooved shaft only has rotational movement and extends through the housing as an output shaft. With fluid piped to the CW port, the output shaft turns clockwise. With fluid piped to the CCW port, the output shaft turns counterclockwise.

One available option is a stroke limiter that allows a precise stopping at any degree less than maximum. Also available are cushions to decelerate rotation speed near the end of stroke. Some manufacturers make this type rotary actuator with an integral cylinder that adds linear movement to the output shaft.

The spiral-shaft rotary actuators in Figure 19-5 operate equally well on air or hydraulic power. They operate at pressures up to 3000 psi and produce torque up to 20,000 in. lb for air service, and up to 5,000,000 in. lb for hydraulic service. Output shafts normally rotate 360° with more turns available on special order.

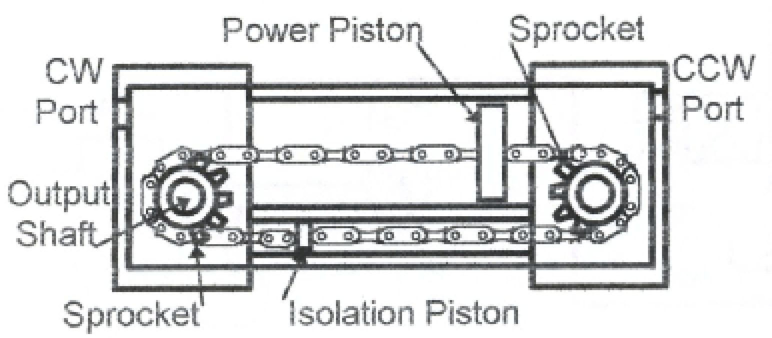


Fig. 19-6. Chain-and-sprocket rotary actuator.

Figure 19-6 shows a simplified cutaway view of a chain-and-sprocket rotary actuator. It consists of a large-diameter power piston with a roller chain attached to both sides. The roller chains go around a sprocket at both ends and attach to both sides of a smaller isolation piston. When pressurized fluid enters a port, it pushes against both pistons with equal force. Because the power piston has more area, it moves away from incoming fluid. The smaller isolation piston regenerates into the incoming pump flow. (To find the effective working area, subtract the area of the isolation piston from the area of the power piston.) With fluid piped to the CW port, the output shaft turns clockwise. With fluid piped to the CCW port, the output shaft turns counterclockwise.

Moving a load rapidly without shock

Raising a well-guided table 24 in. in 1.5 sec; then, after removing a part, lowering it in 1 sec is a challenge. The part cannot stand high starting forces and must stay on the table at the end of stroke.

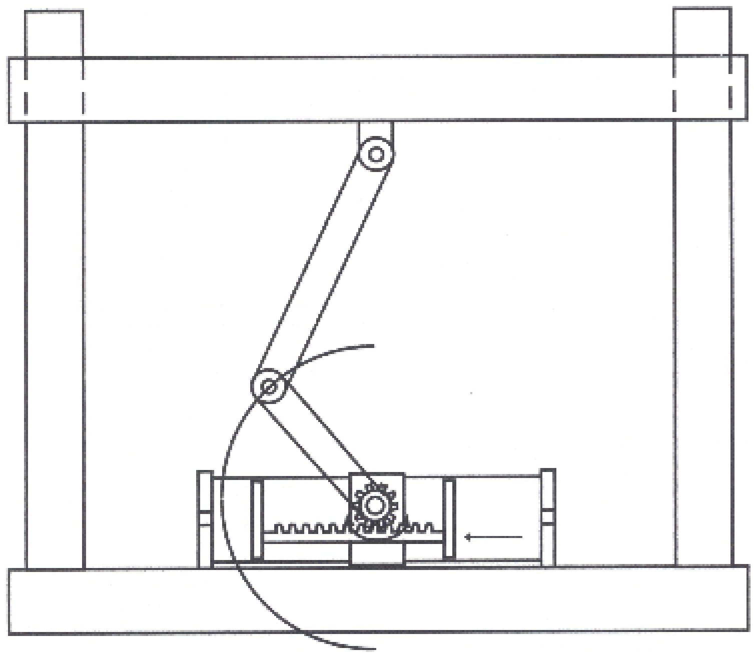


Fig. 19-7. Pictorial layout of rotary actuator that rapidly and smoothly lifts and lowers table without shock.

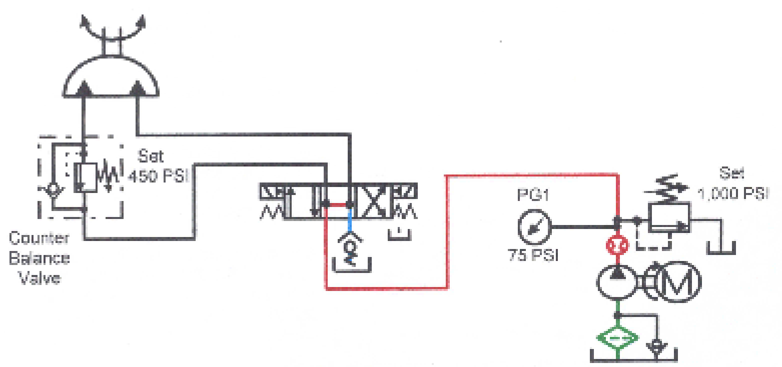


Fig. 19-8 Schematic layout of rotary actuator that rapidly and smoothly lifts and lowers table without shock.

The schematic and pictorial layout in Figures 19-7 and 19-8 shows a way to move something -- such as a lift table -- through a fixed travel rapidly, with little shock. (For other ways to move this type load, using a straight cylinder, see Chapter 14.) The action here is similar to a crankshaft-type press, where the platen and tooling move rapidly and smoothly from top dead center, through bottom dead center, and back with little or no shock.

The layout in Figure 19-7 shows a hydraulic rotary actuator with a 6-in. arm attached to the output shaft. For a large table, choose a rack-and-pinion actuator with a through shaft and put an arm on both sides. Attach the 6-in. arms with pivot pins to 12-in. arms that also connect to the table with pivot pins.

Porting fluid to the right end of the rotary actuator makes the piston drive the rack gear, thus turning the pinion. The pinion starts at low torque and accelerates quickly. As the pinion turns, the table moves up -- slowly at first and then constantly gains speed until mid stroke. From mid stroke on, the table decelerates and actually stops just as the rotary actuator finishes its stroke. The action is fast and smooth without jerks. There is no need for position indicators, shock absorbers, or other devices to position the table. Use a rotary actuator with cushions to minimize shock.

Even though this is a running-away load on the return stroke, motion remains smooth. The schematic drawing in Figure 19-8 shows a counterbalance valve in the lowering line to produce hydraulic resistance that offsets the table and/or part weight. Set the counterbalance valve high enough to offset the weight and stop free fall. Setting the counterbalance valve too high uses excess energy, resulting in unnecessary oil heating.

If the load is resistive part way and running away the rest of the stroke, use a counterbalance valve with an external and internal pilot. The counterbalance valve in Figure 19-8, with internal piloting only, adds extra resistance even with a resistive load. This might cause the circuit to operate at higher pressure than needed and possibly stall. (See Figures 19-20-23, for a full explanation of a resistive, overrunning load circuit.)

With a 360° rotary actuator, action is like the cycle of a crankshaft-type press. For each direction the rotary actuator strokes, the table completes a lift or lower motion. This same mechanism and circuit can move a horizontal load with the same results

Some typical applications for rotary actuators

Figure 19-9 shows a I80° turnover station powered by a rotary actuator. The shaft of the turnover attaches directly to the actuator shaft without gearing or other devices. Because the load is resistive part way and running away for the rest of the stroke, use an internally and externally piloted counterbalance valve for good control. If speed is the same for the whole stroke, use a meter-out flow control in place of the counterbalance valve in a hydraulic circuit.

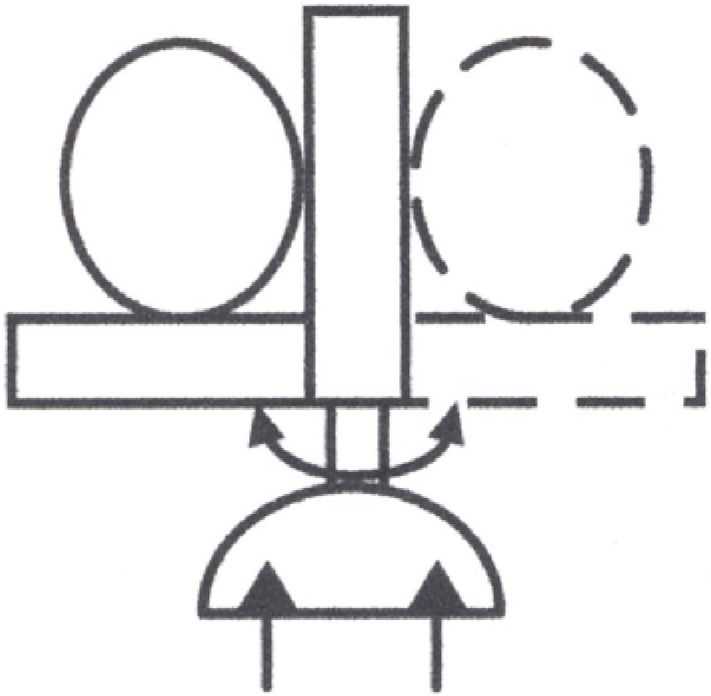


Fig. 19-9. Turnover device.

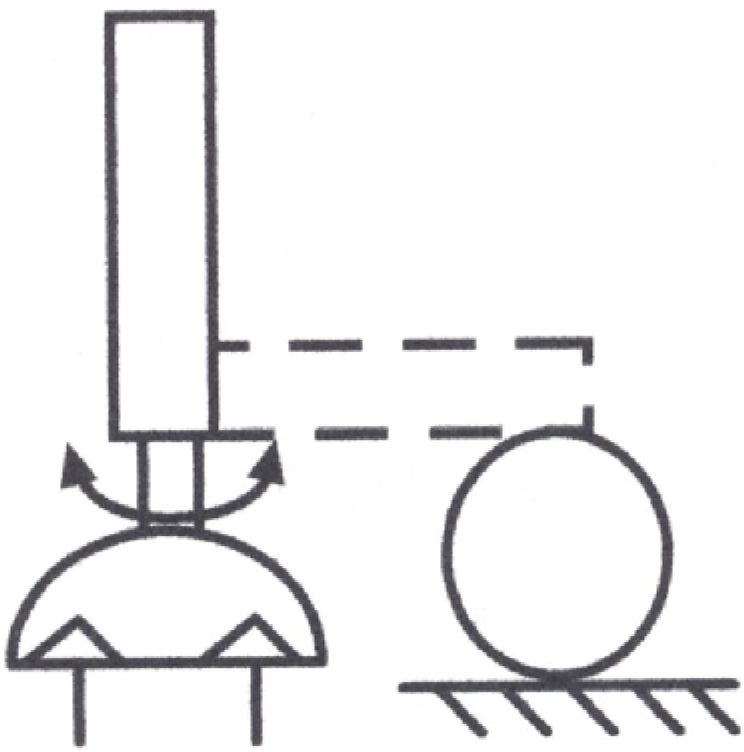


Fig. 19-10. Clamping device

The clamping device in Figure 19-10 allows the part to be loaded from the top without interference. This is an advantage for some machining operations. Calculate the downward force of the clamp arm by dividing its length into the torque of the output shaft. When using a 180° rotary actuator, the clamp arm is completely out of the way for loading.

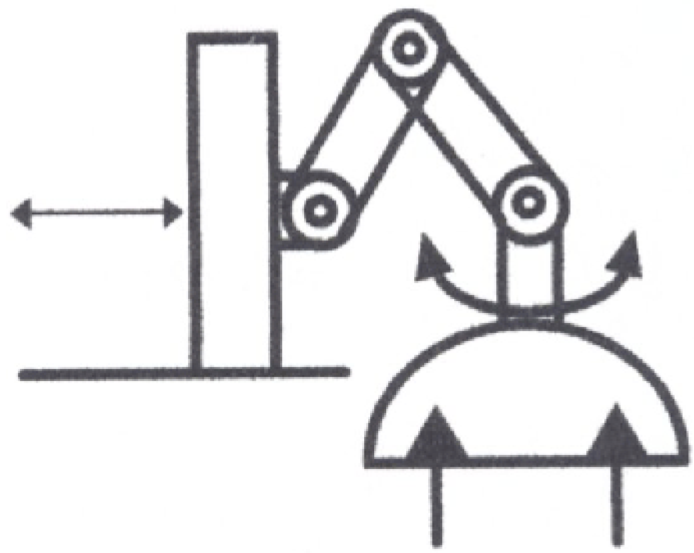


Fig. 19-11. Toggle mechanism

The toggle arms in Figure 19-11 rapidly move a platen to and from the workstation. The arms also multiply the force as they reach the work and get into a straight line. This mechanism only works when the closed height does not change for a given setup.

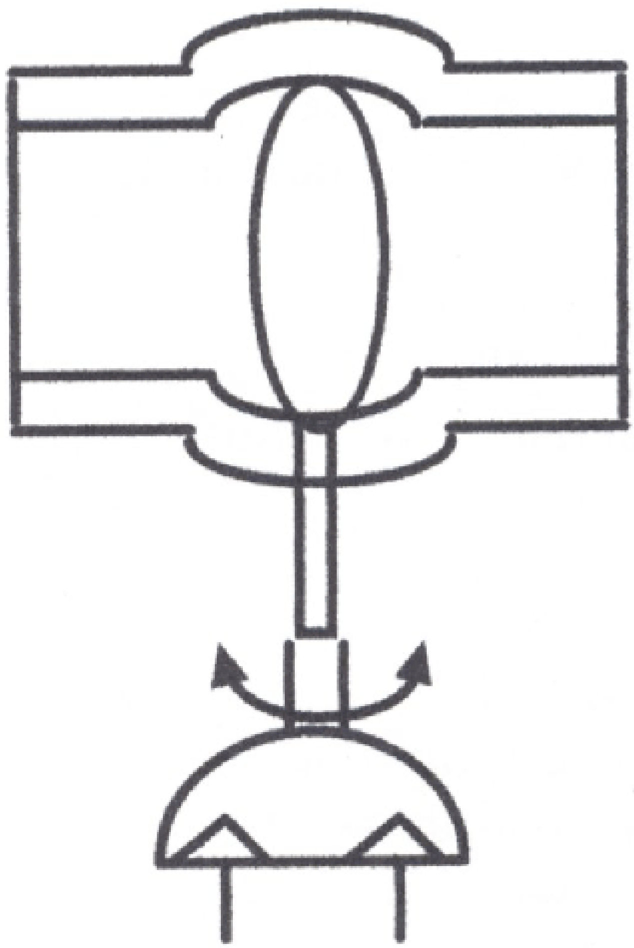


Fig. 19-12. Valve actuator.

Many 90°-turn rotary actuators operate butterfly or ball valves at remote locations, as in Figure 19-12. With adjustable stops or control valve circuits, these actuators give partial movement to control product flow.

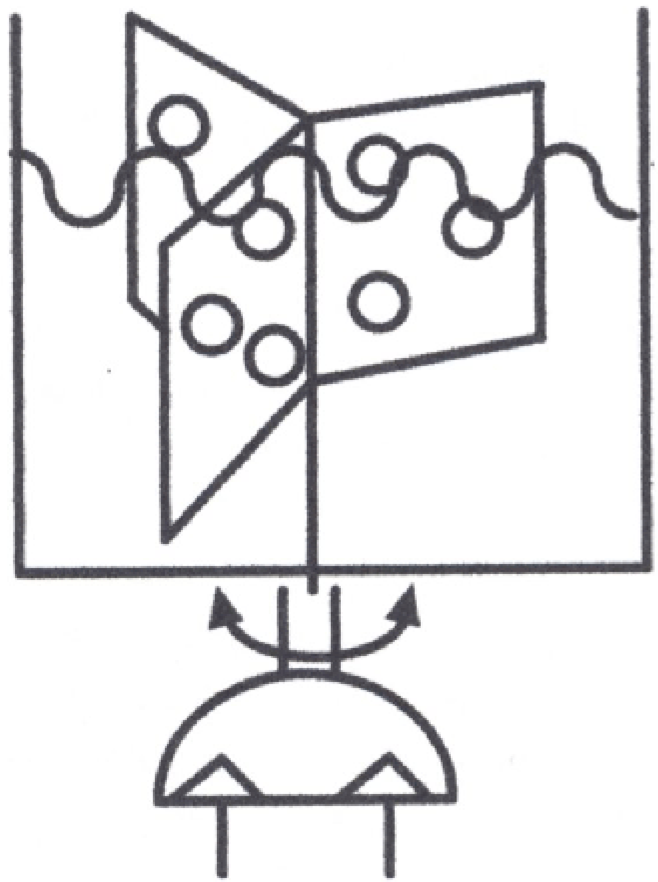


Fig. 19-13. Agitator.

For an agitator drive, especially in an explosive atmosphere, use the rotary actuator setup shown in Figure 19-13. Select an air pilot-operated pneumatic or hydraulic directional valve and pneumatic limit valves to cycle the unit for explosion-proof applications.

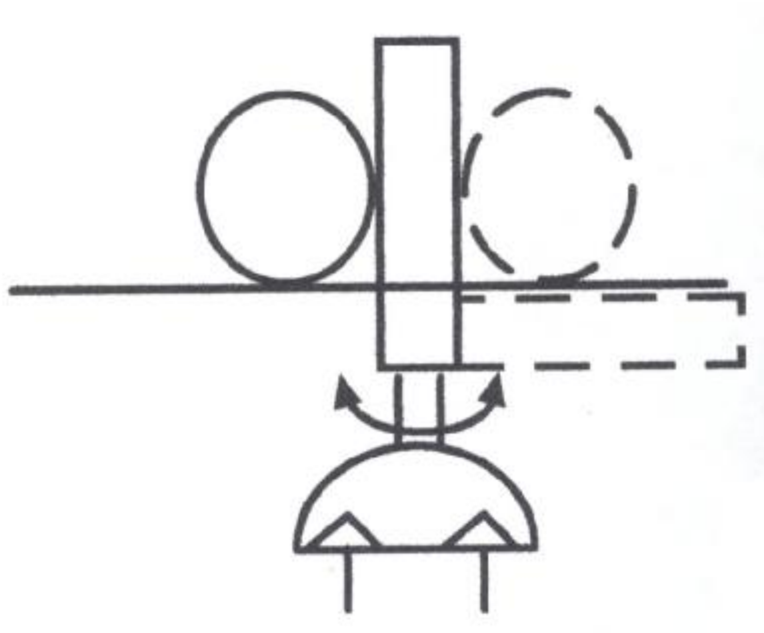


Fig. 19-14. Part-stop device

The swinging stop in Figure 19-14 holds parts on a conveyor while they stack up to a specified number. When the required parts have accumulated, the actuator swings the stop out of the way to let them pass.

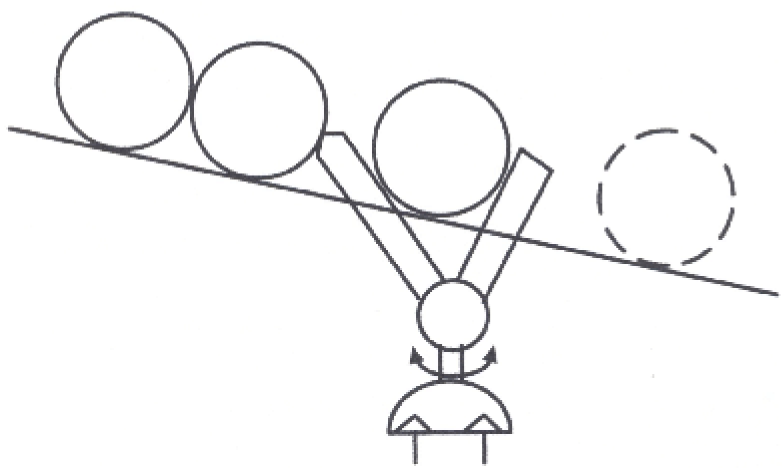


Fig. 19-15. Part-release device

Figure 19-15 shows a single-part release. As the rotary actuator swings through approximately 90°, it releases one part while holding back all others.

There are many applications where rotary actuators save time and cost over other fluid power devices. They are often not as expensive as the components they replace, and they reduce maintenance costs for the life of the machine.

More circuits for rotary actuators

To control rotary actuators consider all of the hydraulic-motor circuits in Chapter 12. Many cylinder circuits work well also, but a rotary actuator has the advantage of equal areas in both directions of travel. Note that these equal areas make it impossible to use a conventional regeneration circuit, but the flow-divider regeneration circuits shown in Chapter11 work well.

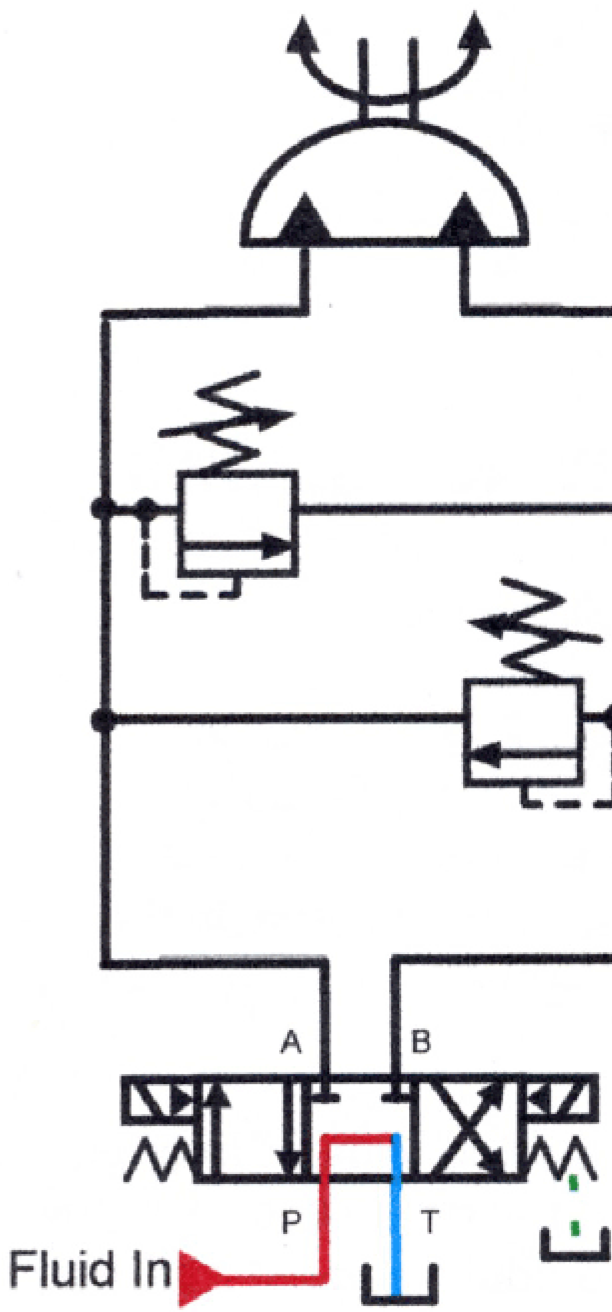


Fig. 19-16. Dual cross-port relief valves

The circuit in Figure 19-16 shows a rotary actuator with cross-port relief valves to protect it from excess pressure when the valve centers with ports A and B blocked. If the rotary actuator has an overrunning load, sudden stops cause damaging pressure spikes. The dual cross-port relief valve shown in this example allows the actuator to decelerate quickly, without shock and its resulting damage.

Set the cross-port relief valves pressure so that it is equal to or higher than the system relief or pump-compensator setting. A pressure setting equal to the system relief valve gives a stopping distance equal to the distance it takes to accelerate the load. Higher pressure settings stop the load in less distance. Always set cross-port relief valve pressure lower than the actuator's maximum operating pressure.

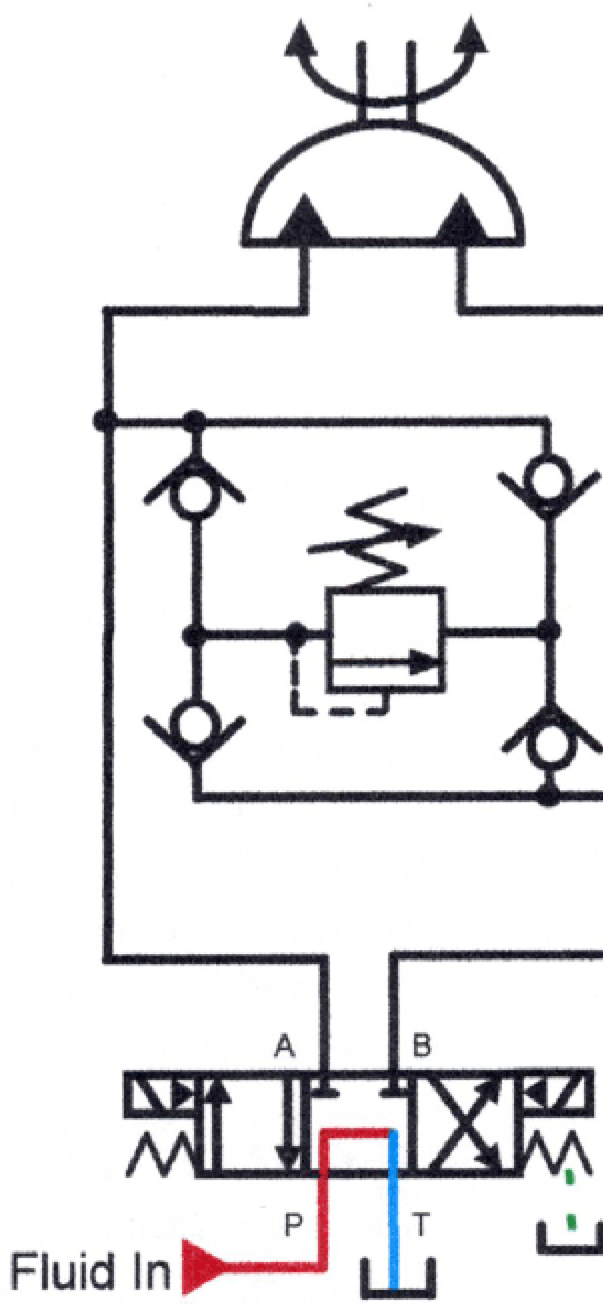


Fig. 19-17. Single cross-port relief with check valves.

In Figure 19-17, a single relief valve with four check valves gives the same cross-port relief function without having to make two adjustments. Otherwise, this circuit works the same as the one in Figure 19-16.

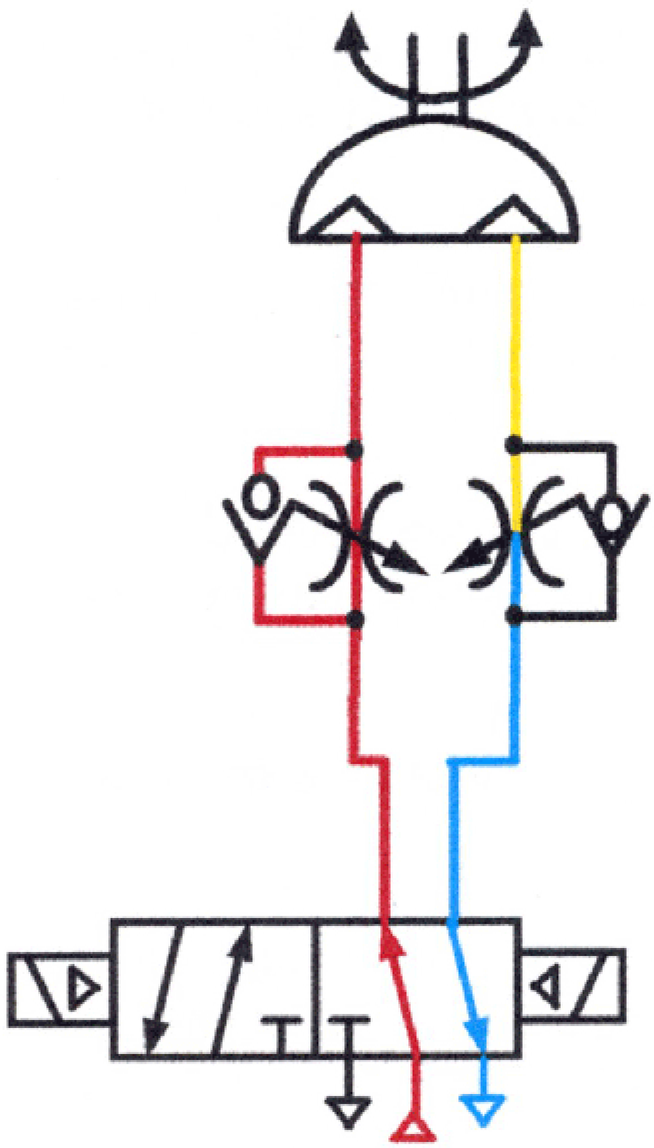


Fig. 19-18. Gate-valve operator with 2-position directional valve and meter-out flow controls.

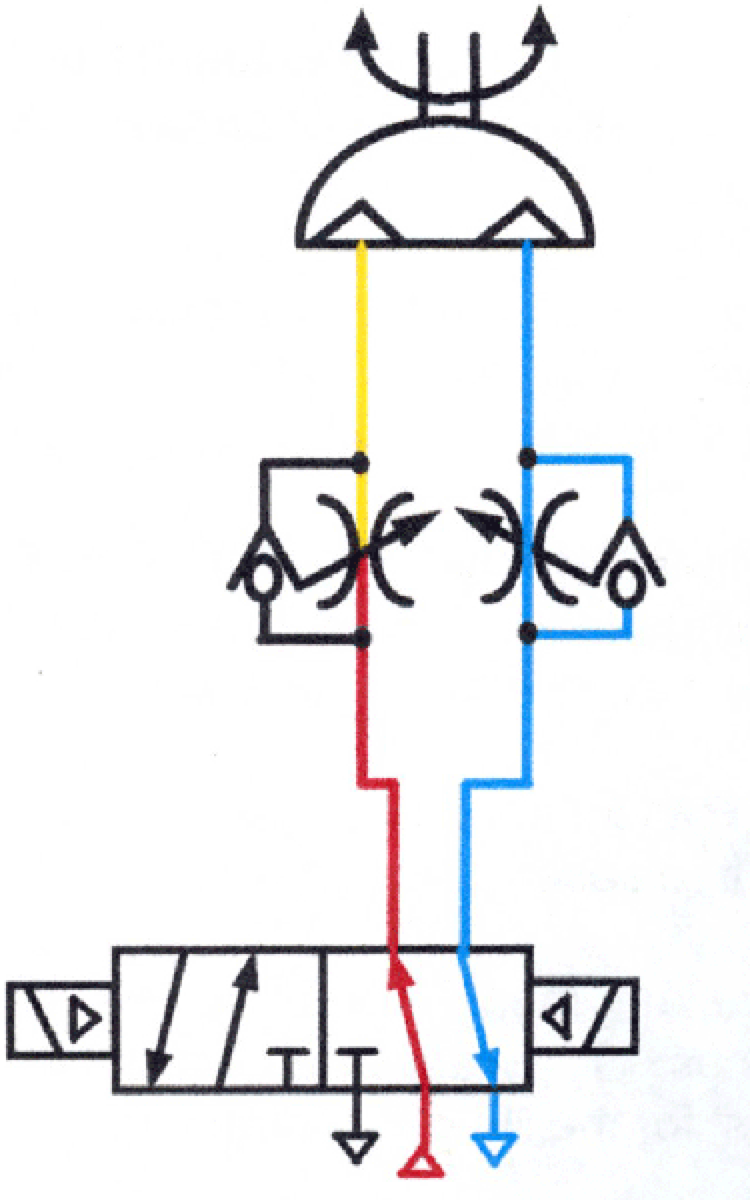


Fig. 19-19. Gate valve operator with 2-position directional valve and meter-in flow controls

---

Flow control circuits shown in Figures 19-18 and 19-19 control the speed of the rotary actuator. All of the advantages and disadvantages of the different flow control circuits apply here as well as to cylinders. (See Chapter 10 on flow controls for the ways to control an actuator's speed.)

Resistive to over-running load application

In Figure 19-20, a turnover table driven by a rotary actuator starts off as a resistive load and goes to over-running at about mid stroke. This is not an unusual application for a rotary *actuator. When speed of rotation is the same for the whole stroke, use a meter-out flow control circuit for this application. When using variable speed such as fast traverse and slowdown, use the counterbalance circuit shown and described.*

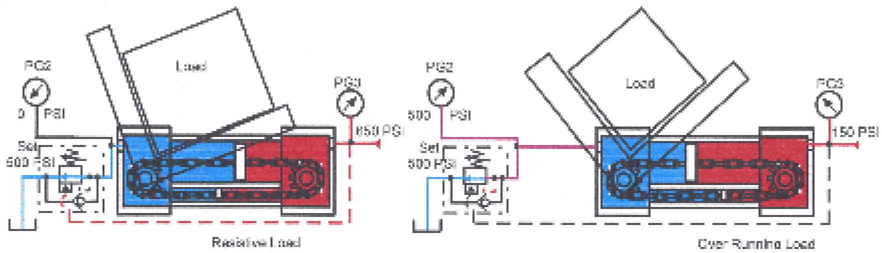


Fig. 19-20 Pictorial representation of resistive, overrunning load application with high-low pump circuit for rapid traverse and slowdown.

To make it possible to keep pressure high enough to move the load and also have a slow down at the end of travel, use the circuit shown in Figures 19-21 through 19-23. This circuit has both a high-volume pump and a low-volume pump for fast travel, but uses only the low-volume pump for slow speed. Normally open solenoid operated relief valve B unloads the high-volume pump near the end of the stroke. Counterbalance valves C and D slow the actuator when the high-volume pump unloads.

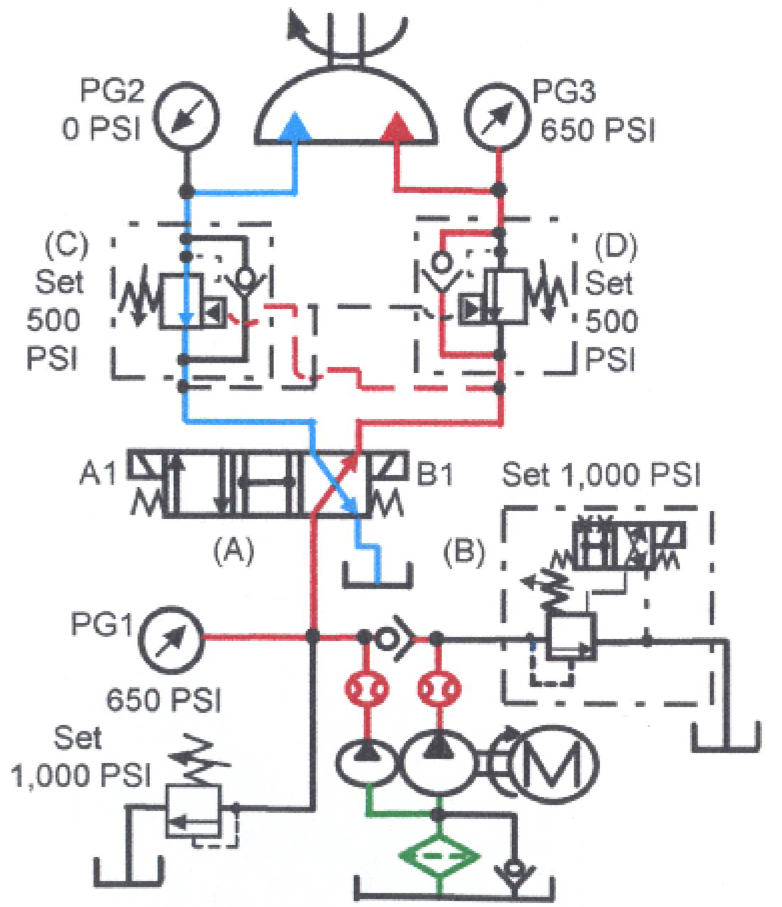


Fig. 19-21. Fast traverse in resistive mode.

In Figure 19-21, both pumps lift the load rapidly. Energizing the solenoid on normally open solenoid-operated relief valve B and solenoid B1 on directional valve A sends flow from both pumps to the actuator. Fluid from both pumps raises the load through the resistive part of the cycle. (Note that gauges PG1 and PG3 indicate 650 psi.) The external pilot port of counterbalance valve C in the outlet line of the actuator senses that working pressure, causing it to open fully. Pressure at gauge PG2 shows 0-psi backpressure in the outlet line of the actuator, so there is no wasted energy. As long as the rotary actuator lifts the load, external pilot pressure keeps counterbalance valve C open, allowing free flow to tank.

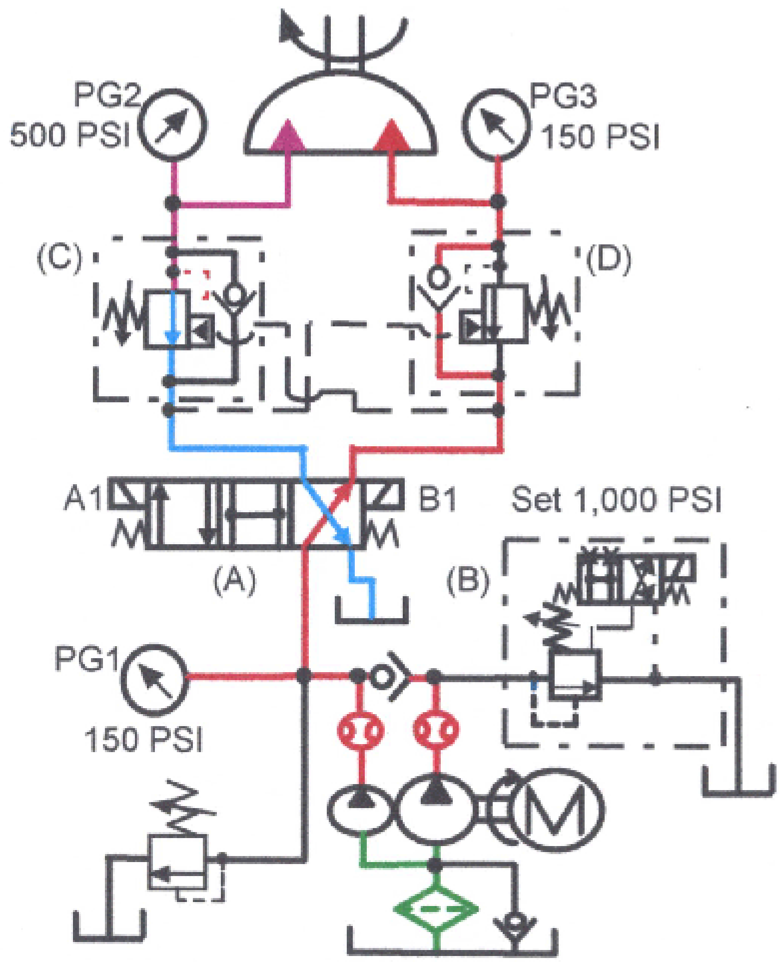


Fig. 19-22. High-low pump circuit for fast traverse and slowdown of a resistive, over-running load.

When the load goes over center, as in Figure 19-22, the rotary actuator tries to run away. When this happens, external pilot pressure to counterbalance valve C drops and it tries to close. When the counterbalance valve closes enough to restrict movement of the actuator, pressure at its inlet increases. The pressure increase is partially from the over-running load while the rest is from the pump pushing on the other side of the actuator piston. As the load goes over center and continues on, pressure at PG2 steadily increases while pressure at PG3 decreases. With a counterbalance valve creating resistance, it takes pump flow and its resulting pressure to keep the rotary actuator moving. Speed is as fast as pump flow dictates.

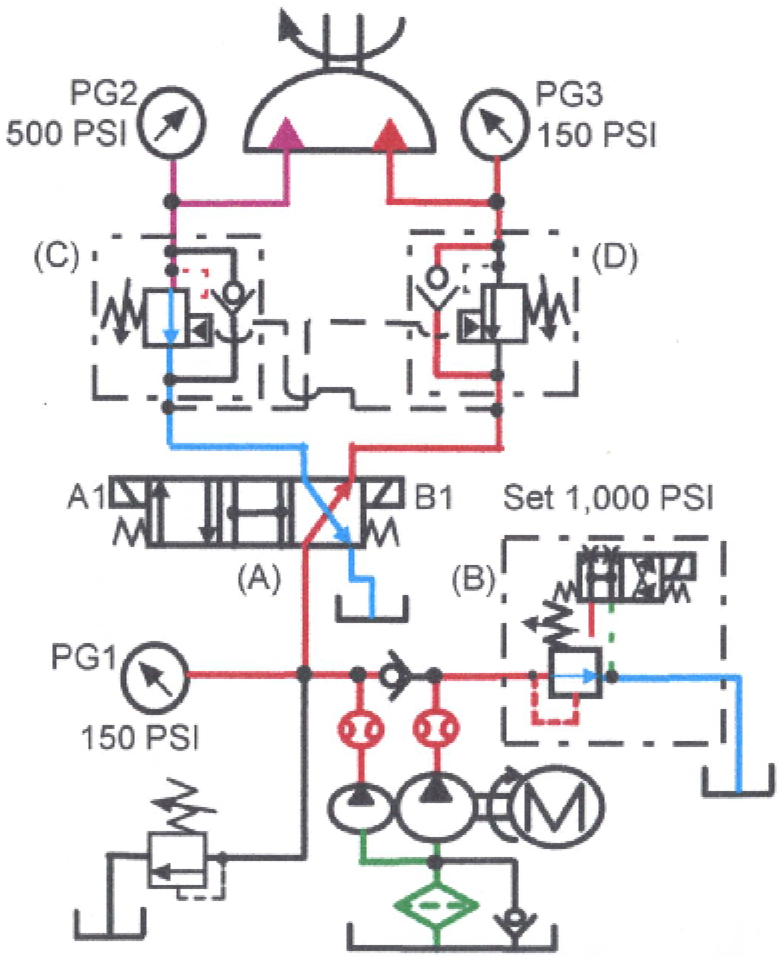


Fig. 19-23. Slow down in over-running mode.

In Figure 19-23, the solenoid on normally open solenoid-operated relief valve B is deenergized near the end of the stroke. This unloads the high-volume pump at low pressure. Decreased flow to the actuator causes a pressure drop at its inlet and outlet. Counterbalance valve C again tries to close, holding the load back until the low-volume pump starts pushing it at a slower rate. Slow down is quick and smooth. This circuit allows the turnover to move rapidly during most of the cycle, while the slow down eliminates shock at stopping.

The schematic diagram shows two counterbalance valves to handle a load in each direction. If the carrier returns without a load, set counterbalance D lower, to reduce the energy requirement on the return stroke.

Servovalve Circuits

When a cylinder or fluid motor application needs precise control of position, speed, or force, an on/off solenoid or proportional solenoid valve will not do the job. Some rolling mills control metal thickness to a tolerance of ±0.0005 in. This is with metal passing through the rolls at 2500 to 3000 ft/min and more. To hold these kinds of tolerances requires more than a go, no-go hydraulic control valve.

Servo directional valves are the only hydraulic valves capable of controlling oil flow and/or pressure rapidly and precisely. Servo directional valves are 4-way, 3-position spool valves with all ports blocked in the center position. Usually, servovalve spools are controlled by high-pressure pilot oil. Many spools have feedback sensing to give repeatable positioning from a given input.

Servovalve spools differ from on/off or proportional valve spools because they have no overlap in center condition. Spool overlap makes proportional valves (and the actuators they control) respond slowly. With no overlap or underlap, any servovalve spool movement gives immediate flow and actuator response. The more closely the spool and body lands match at all four sealing areas, the more responsive the valve. This type of spool is difficult to manufacture, which makes the valve expensive.

Servo systems control actuators to very close tolerances in regard to position, speed, or force. Often a single circuit uses a combination of these functions. A cylinder may have to rapidly approach the work piece, then penetrate it to precise depth at a controlled rate.

While servovalves are very fast and precise, their electronic control is what really makes a servo system work so well. When a signal to move a cylinder starts an action, feedback from its movement modifies valve input to make it match control input. Regardless of pressure drop, fluid viscosity, load, or friction, feedback signals modify valve-spool position to make the cylinder perform exactly as the input signal commands. The only time the actuator falls behind is when it is underpowered.

Figures 21.1 and 21.2 show the schematic symbols for a typical servovalve as established by the American National Standards Institute and the International Standards Organization. Both symbols have parallel lines on both sides of the position envelopes. These parallel lines indicate a valve spool with infinite positions. The symbol shows a blocked center (P to A, B to T and P to B, A to T), but the spool seldom shifts all the way to either of these positions. Spools can shift any amount in either direction, producing increasing or decreasing flow to and from the actuator to move it in either direction.

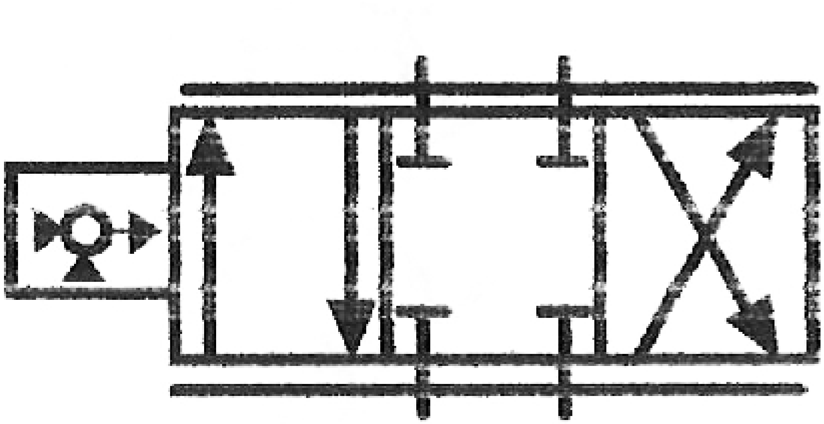


Figure 21-1. ANSI servovalve symbol.

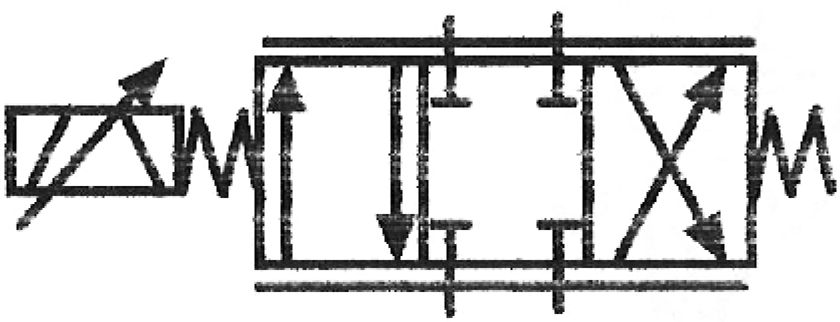


Figure 21-2. ISO servovalve symbol.

Simple mechanical servocircuit

Figure 21.3 shows a simple mechanical servocircuit that controls rudder movement on tugboats. The rudder on a tugboat is big and directly in the prop wash, so the operator must have help in moving and controlling it. The lever-operated hydraulic valve in this circuit directs hydraulic power to move the rudder via a double-acting cylinder. If the valve is in the pilothouse, it does not show rudder position. Without knowing the rudder angle, engaging the propellers might be disastrous in some situations.

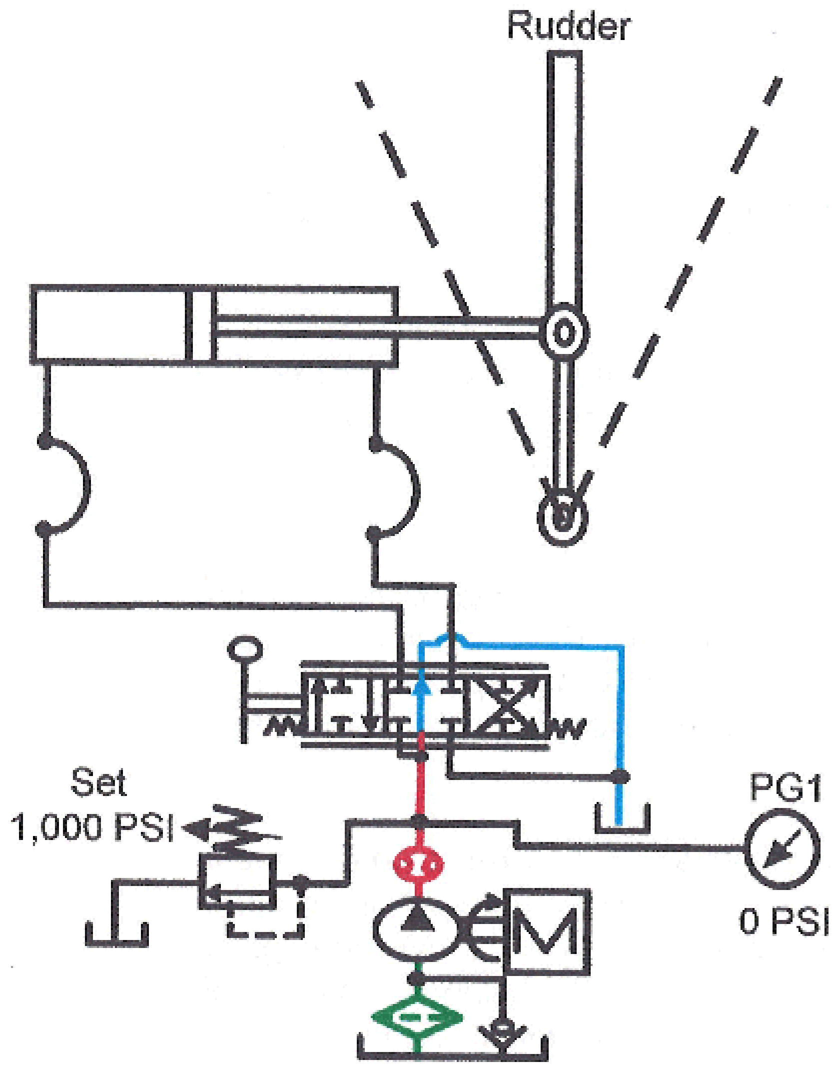


Figure 21-3. Simple manual rudder-control circuit -- at rest with pump running.

Figures 21.4 through 21.6 show an inexpensive manual rudder-control circuit. This circuit uses the same lever-operated control valve in Figure 21.3, but here it mounts on the rod of the double-acting cylinder. The operator controls the valve from the pilothouse with a lever called a "tiller." A cable and pulley system connects the tiller to the valve. This all sounds a little crude but it works quite well on small boats.

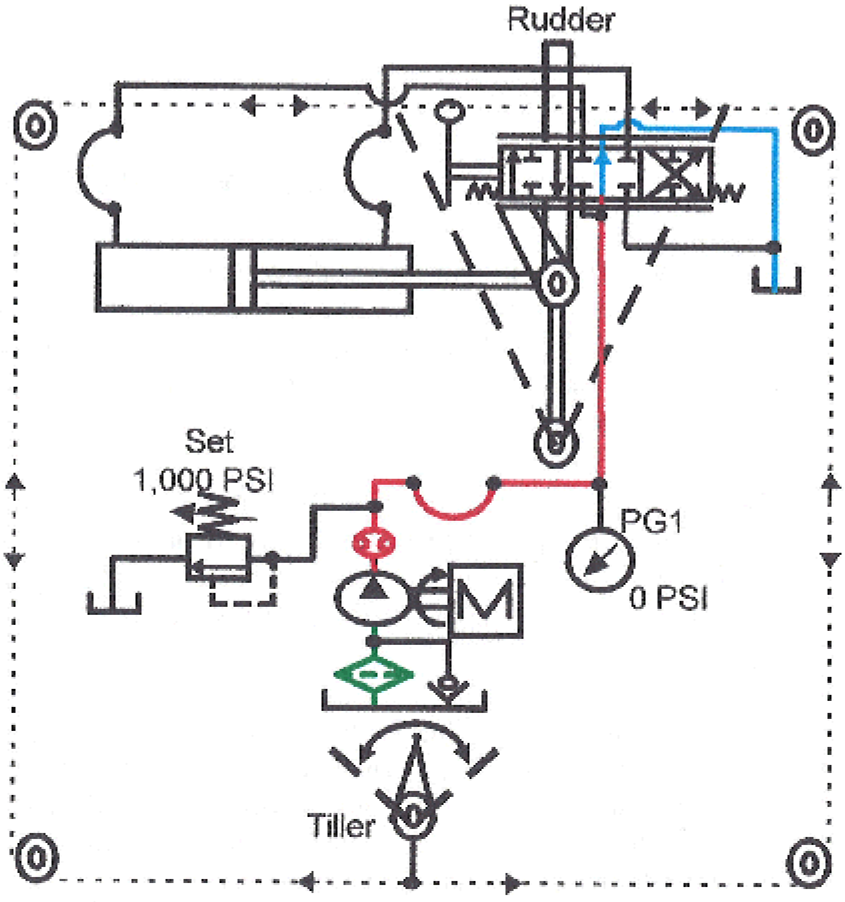


Figure 21-4. Mechanical servo rudder-control circuit -- at rest with pump running.

The clevis-mounted double-acting cylinder attaches to the boat frame and the rudder lever. The lever-operated valve mounts directly to the cylinder rod so it moves with the rudder lever. When the operator moves the tiller to the right, as in Figure 21.5, the lever on the valve moves to the right. When the lever moves, it shifts the directional valve and ports oil from the pump to the cylinder's cap end and returns oil to tank from the rod end. The cylinder moves the rudder to the right as long as the operator keeps moving the tiller.

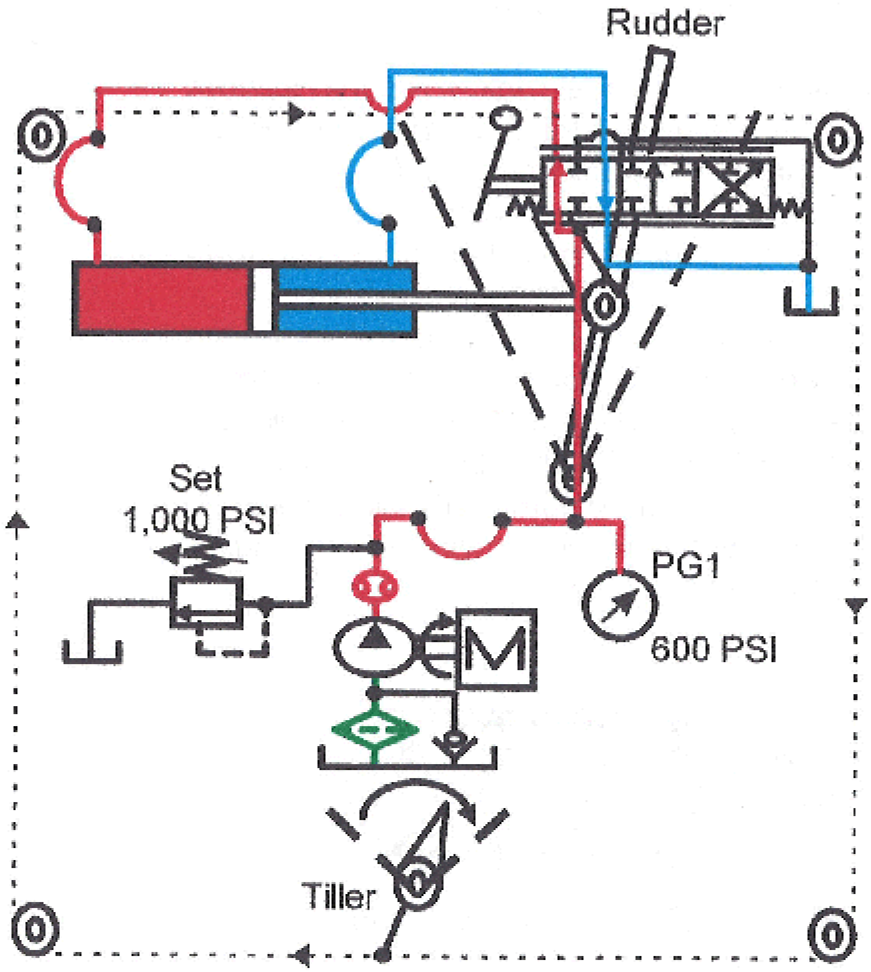


Figure 21-5. Mechanical servo rudder-control circuit -- rudder moving right.

When the operator stops moving the tiller, as in Figure 21.6, the directional valve, moving with the cylinder rod, catches up and centers. When tiller movement ceases, the rudder stops and holds. The rudder and tiller stay in this position until the operator steers in a different direction. At all times the operator knows rudder position by looking at the tiller angle.

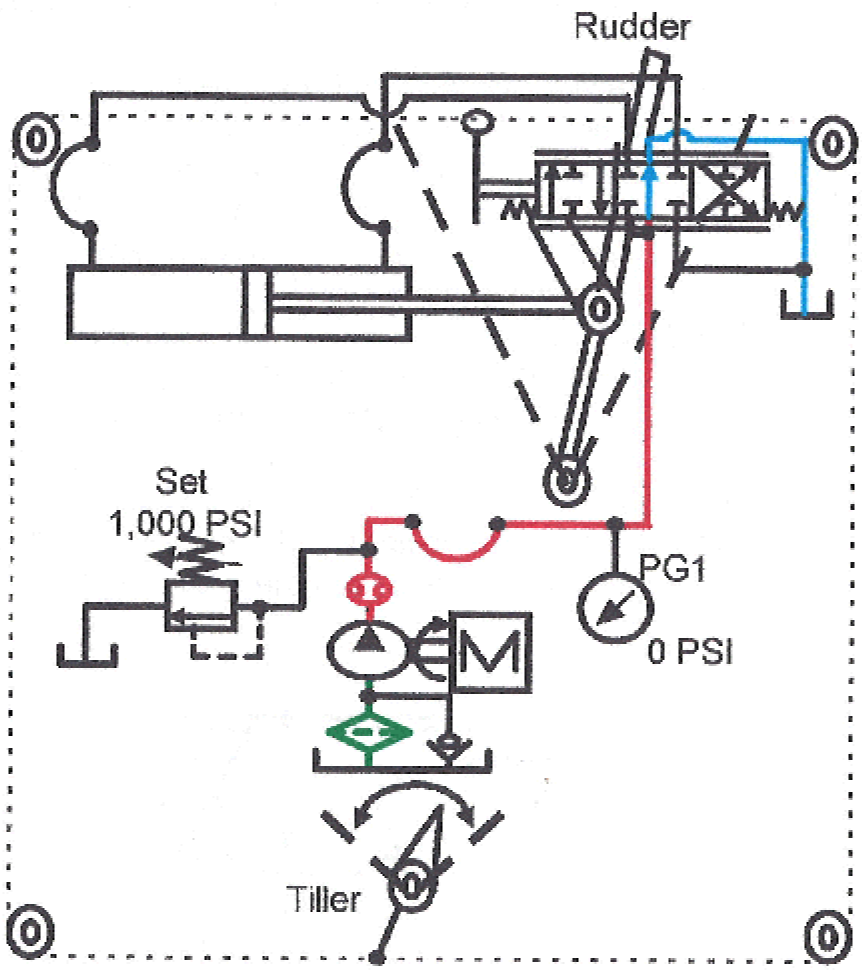


Figure 21-6. Mechanical servo rudder-control circuit -- rudder stopped and holding.

The mechanical servosystem is nothing more than a force multiplier. In this case, the formerly hard-to-move rudder now moves with slight manual force. At the same time, the tiller position indicates the rudder angle because of mechanical linkage feedback.

An automobile power-steering system uses similar circuitry. Steering wheel movement shifts a directional valve that powers a cylinder to move the steering mechanism. When the steering wheel moves, front wheel angle changes. When steering wheel motion ceases the front wheels stop and hold.

The rudder control circuit shown here might be adapted to control a pressing action where cylinder movement follows the motion of the operator's hand. This gives accurate position with a great amount of force from the operator's intuitive feel.

Servovalves for accurate positioning of actuators

The schematic drawing in Figure 21.7 shows the general arrangement for a typical servocircuit that accurately controls cylinder position. When a cylinder must quickly go to many different locations with an accuracy of less than ±0.020 in., a servocircuit is the best way to control it.

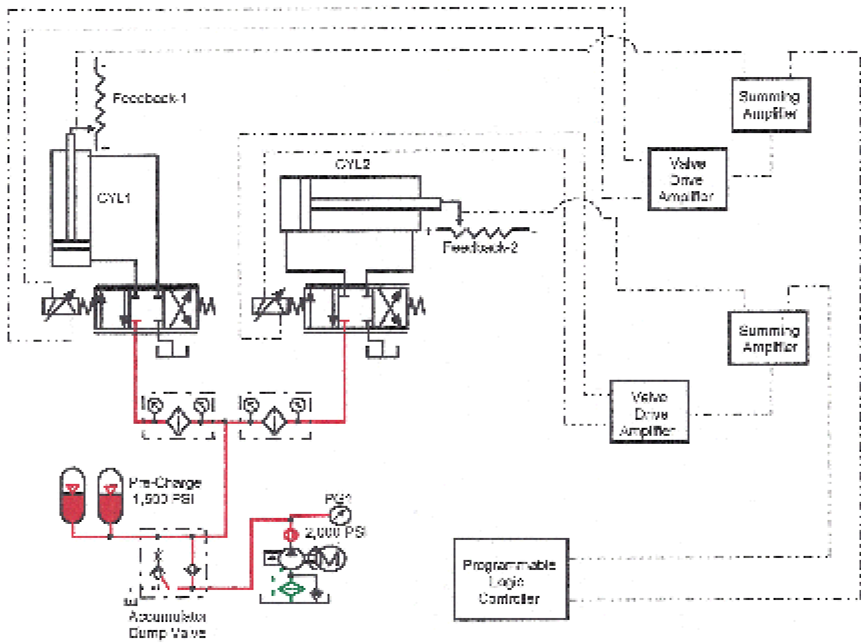


Figure 21-7. Pressure-compensated piston pump with accumulators in an electrical closed-loop positioning circuit.

Notice that the hydraulic power unit has a pressure-compensated pump with accumulators. This arrangement holds constant pressure and has ample volume for short bursts of high flow. Without the accumulators, there is a sharp pressure drop when a cylinder starts moving. Fixed-displacement pumps and accumulators work also, but the power unit shown here is best overall.

Place the servovalve as close as possible to the actuator (preferably attaching it directly to it). Use rigid piping when the valve cannot mount directly on the actuator. Flexible lines between the servovalve and the actuator can negatively affect the accuracy and stability of the circuit.

Always install pressure filters in the lines to the servovalves. One pressure filter after the pump might be sufficient when the power unit is close to the valves. Separate filters are advisable when there is some distance to the servovalves. Use a cleanliness level of 1 to 5 µm in a servocircuit. Even normal pump-wear contamination quickly plugs orifices and sticks spools in most servovalves. Do not use a bypass-type pressure filter in a servovalve circuit. Even with a 125-psi bypass spring, contaminated fluid can get around the filter during a normal cycle. It is *better to shut the machine down with a clogged filter than with a contaminated servovalve and a dirty filter.*

Because the cylinders in this arrangement must stop accurately at many different locations, the circuit includes a feedback transducer at the cylinder rod. When the PLC commands the cylinder to go to a certain location, the PLC sends a signal to the servovalve control card. The servo control card sends an output to the servovalve that starts the cylinder moving. As the cylinder moves, the feedback transducer constantly sends position information to the servo control card. When the cylinder approaches the predetermined position, it slows and stops within a few thousandths of an inch of that location every cycle. Because electronic hardware controls the speed and position of the cylinder, fluid viscosity, load, pressure drop, or machine friction have no effect. The control card modifies the sevovalve shift to offset external or internal system changes as long as the actuator has ample power to overcome them.

In essence the electronics modify servovalve output according to actual actuator movement to get the desired accuracy. A servovalve controls oil flow as a 4-way directional valve would, but it has the ability to change flow continuously. Response time of the servovalve to the electronic controllers' changes is the important thing. Less-expensive, more dirt-tolerant servovalves offer less-accurate control.

With the circuit in Figure 21.7, cylinder positioning at any location within its stroke is attainable with repeatable accuracy to thousandths of an inch.

Servovalves for accurate control of position and velocity

The schematic diagram in Figure 21.8 shows servovalves controlling the velocity and position of a cylinder. The cylinder in this circuit has position and speed control, while the hydraulic motor only has speed control. All previous information about hydraulic power unit type, valve location, and filters applies to this circuit — or any other servo application.

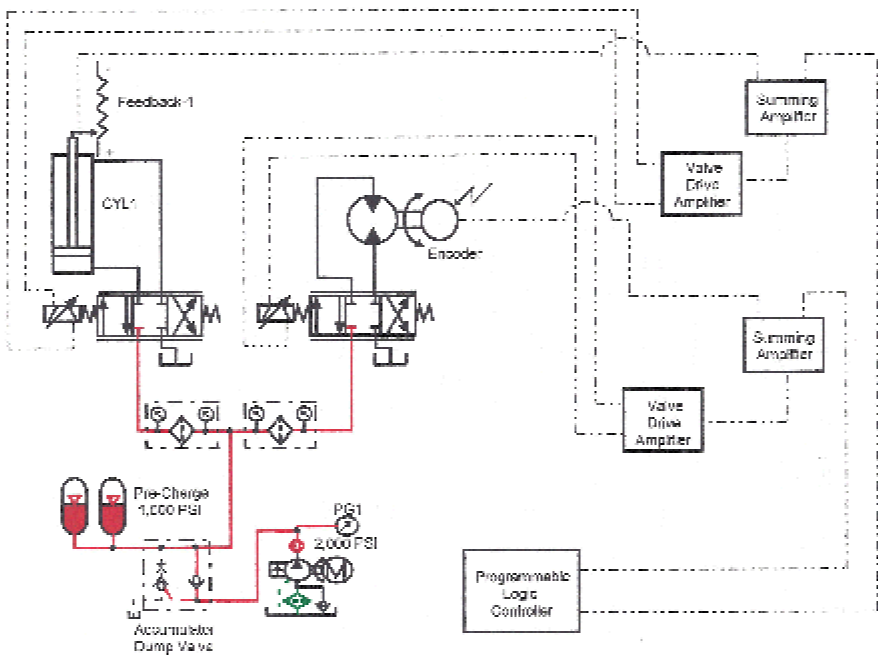


Figure 21-8. Pressure-compensated piston pump with accumulators in an electrical closed-loop positioning and velocity circuit.

The cylinder in this circuit has accurate positioning as does the cylinder in Figure 21.7, but this cylinder has controlled speed as well. A milling operation requires accurate speed control but also may need depth control. When fast accurate positioning at multiple locations is important, use a servovalve.

When the PLC sends a signal to start the cylinder moving, it smoothly ramps up to any speed desired. A servovalve allows for accurate velocity change anywhere within the stroke when the controller calls for it. At the end of stroke, the cylinder decelerates rapidly and smoothly to an accurate stopping position, without shock. Again, the servovalve performs the 4-way function while the electronic controls change speed and position. The servovalve must respond fast enough to follow the controllers' outputs or the cylinder position and/or speed will not match the machine requirements.

The hydraulic motor in Figure 21.8 must turn at a constant rate regardless of load or changes in pressure drop or fluid thickness. Even with a pressure- and temperature-compensated flow control, motor speed varies as pressure changes. Internal slippage in the motor is greater at higher pressures, so speed decreases even with constant input flow.

With a servovalve feeding the hydraulic motor and a feedback device giving the servocontrol card continuous speed information, motor speed is consistent. The only time motor speed varies is when it stalls at relief valve pressure.

As before, electronics handles all input and modifications to get the desired speed. A servovalve controls oil flow as a 4-way directional valve does, but it also has the ability to change flow as needed. It is the response of the servovalve to the electronic controllers' changes that is most important. Less-expensive, more dirt-tolerant servovalves have less accuracy.

With the circuit in Figure 21.8, cylinder speed is fast, and the cylinder stops in a precise position without shock. The hydraulic motor maintains the set speed regardless of load or input fluctuations — until it stalls from lack of torque. All motions are repeatable.

Servovalves for accurate control of position and force
Figure 21.9 shows a schematic diagram with a servovalve controlling the force of an actuator. The vertical cylinder in this circuit has position control, while the horizontal cylinder has force control. All the information about hydraulic power unit type, valve location, and filters, applies to this circuit or any other servo application.

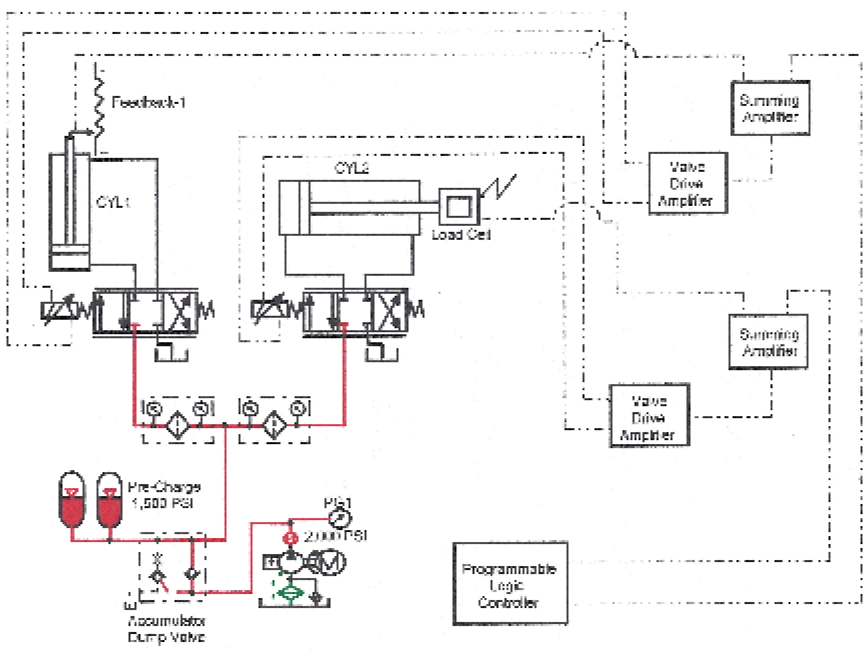


Figure 21-9. Pressure-compensated piston pump with accumulators in an electrical closed-loop positioning and force circuit.

*The vertical cylinder in this circuit has accurate positioning like the cylinder in Figure 21.7***,** *but this cylinder has controlled speed as well. An application might be a milling operation that* *requires accurate speed control but may need depth control as well. When fast, accurate positioning at multiple locations is important, use a servovalve.*

When the PLC sends a signal to stroke the cylinder, it smoothly ramps up to any speed desired. A servovalve allows for accurate velocity change anywhere along the stroke when the controller calls for it. At the end of stroke, the cylinder decelerates smoothly, rapidly, and accurately to the commanded stopping position without shock. Again, the servovalve does the 4-way function while the electronic controls change speed and position. The servovalve must respond quickly enough to follow the controller's output signals or cylinder position and/or speed will not match the machine requirements.

The horizontal cylinder in Figure 21.9 must hold a constant force against a part, regardless of the load or other changes such as pressure drop or fluid viscosity. Even with a constant pressure source, fluctuations in cylinder friction, machine friction, or rod-end backpressure continuously affect cylinder force. To produce consistent cylinder force, use a servovalve to operate the cylinder and load-cell feedback to continuously modify the valve's spool position. Force stays exactly as set, regardless of system changes -- up to relief valve pressure.

As before, the electronics handle all the input and modifications to set and maintain the desired force. A servovalve controls oil flow as a 4-way directional valve, but has the ability to change flow as needed. It is the response of the servovalve to the electronic controller's changes that is most important. Less-expensive, more dirt-tolerant servovalves offer less-accurate control.

With the circuit in Figure 21.9, the vertical cylinder accurately reaches and maintains any position at any speed. The horizontal cylinder holds any force desired up to maximum pressure.

Stepper-motor-driven servovalves for cylinders

Figure 21.10 shows a simplified cutaway view of a stepper-motor-driven servovalve controlling a hydraulic cylinder. As it receives current pulses, the stepper motor turns in increments of a revolution. Stepper motors may require anywhere from 100 to 500 pulses per revolution. A stepper-motor drive is reliable and repeatable, and produces high torque.

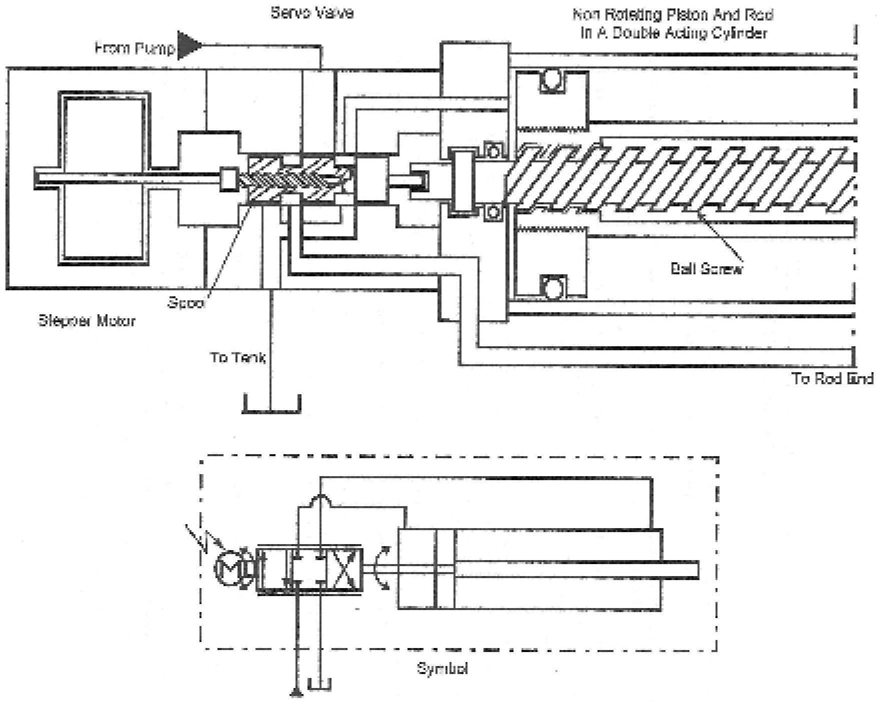


Figure 21-10. Stepper-motor driven servovalve controlling a hydraulic cylinder.

This type servovalve is more dirt-tolerant than other designs. It does not require specific electronics, does not need feedback transducers, and is easy to troubleshoot. This valve may be a stand-alone unit for acceleration and/or deceleration circuits, or for controlling flow -- with or without feedback. Like other servovalves, it has little or no land overlap and a precisely fitted spool to reduce leakage. There are no control orifices to plug, so fluid cleanliness is not as important as with a standard servovalve.

Feedback to a stepper-motor-driven servovalve is mechanical and internal -- similar to the rudder control in Figures 21.4, 21.5, and 21.6. This means that when the cylinder meets resistance it cannot overcome, it will stall. When the cylinder stalls, there is no external feedback to show it has not made its complete stroke. Adding a limit switch or another external signal source helps this problem, but now the circuit resembles a standard on/off solenoid-valve setup.

The response of a stepper motor drive is a little better than the best proportional valves, but not equal to top-of-the-line servovalves.

In the cutaway view, a stepper motor drives a threaded shaft in a threaded spool. The spool can move in and out, but it cannot rotate unless the feedback ball screw in the piston rod turns. Electric pulses to the stepper motor turn the screw in the spool, making the spool shift. Spool *shift ports fluid to the cylinder's cap end, making the cylinder extend. When the non-rotating piston and rod start forward, the internal ball screw turns the spool. The ball screw's mechanical linkage turns the spool in the reverse direction of the stepper motor, shifting the spool to stop cylinder movement. When the stepper motor turns, the cylinder extends. The faster the stepper motor receives pulses, the faster the cylinder travels. When the stepper motor stops turning and shifting the spool, the cylinder continues until the ball screw brings the spool back to center. Reversing rotation of the stepper motor reverses all the actions above, including cylinder direction.*

From the above explanation, it is obvious that pulsing the stepper motor a certain number of times at a given rate strokes the cylinder to a certain position at a preset speed. If external forces try to move the cylinder out of its position, spool shift — caused by rotation of the ball screw in the piston rod — ports oil to offset these forces.

Stepper-motor-driven servovalves for motors

Figure 21.11 is a simplified cutaway view of a stepper-motor-driven servovalve controlling a hydraulic motor. As it receives current pulses, the stepper motor turns in increments of a revolution. Stepper motors may require anywhere from 100 to 500 pulses per revolution. A stepper-motor drive is reliable and repeatable, and produces high torque.

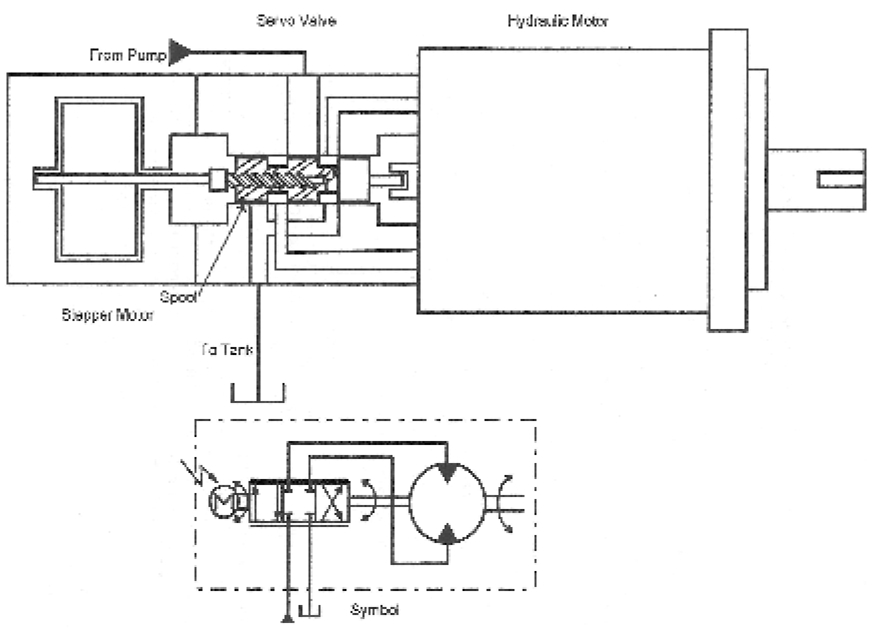


Figure 21-11. Stepper-motor driven servovalve controlling a hydraulic motor.

This type servovalve is more dirt tolerant than other designs, does not require specific electronics, does not need feedback transducers, and is easy to troubleshoot. It may be used as a stand-alone valve for acceleration and/or deceleration circuits or to control flow -- with or without feedback. Like other servovalves, it has little or no land overlap. It has a precisely fitted spool to reduce leakage. There are no control orifices to plug, so fluid cleanliness is not as important as with a standard servovalve.

Feedback to a stepper-motor-driven servovalve is mechanical and internal, similar to the rudder control in Figures 21.4, 21.5, and 21.6. This means that when the motor meets resistance it cannot overcome, it will stall. When the motor stalls, there is no external feedback to show it has not made its predetermined position. Adding a limit switch or other external means helps this problem, but now the circuit resembles a standard on/off solenoid-valve setup.

The response of a stepper-motor driven servovalve is a little better than the best proportional valves, but not equal to top-of-the-line servovalves.

In the cutaway view, a stepper motor drives a threaded shaft in a threaded spool. The spool can move in and out, but it cannot rotate unless feedback from the rotating hydraulic motor turns it. Electric pulses to the stepper motor turn the screw in the spool, making the spool shift. Spool shift ports fluid to the hydraulic motor, making it turn. When the hydraulic motor starts to rotate, it turns the spool. The mechanical linkage turns the spool in reverse of the stepper motor, shifting the spool to stop hydraulic motor rotation. When the stepper motor turns, the hydraulic motor rotates. The faster the stepper motor receives pulses, the faster the hydraulic motor turns. When the stepper motor stops turning and shifting the spool, the hydraulic motor continues to rotate until it brings the spool back to its center position. Reversing rotation of the stepper motor reverses all the actions above -- including hydraulic motor's rotation direction.

From the above explanation it is obvious that pulsing the stepper motor a certain number of times at a given rate, turns the hydraulic motor a certain number of revolutions at a preset speed. If external forces try to move the hydraulic motor from its off position, spool shift — caused by feedback rotation — ports oil to offset these forces

Synchronizing Cylinder Circuits

Some machines with multiple cylinders require that the cylinder strokes be perfectly synchronized for the machine to operate properly. If all the loads, line sizes and lengths, and friction of the cylinders and machine members are identical, they may stroke at the same time and rate. While line sizes and lengths, and machine loading can be controlled to some extent, friction changes constantly. Thus, when cylinders have to stroke together, use some method to synchronize them.

One way of synchronizing cylinders is with external mechanical hardware. Some common mechanisms are racks and pinions, crankshafts, cables and pulleys, and chains and sprockets. The accuracy of these methods depends on the strength of the hardware and the position of the load. Mechanical methods are the most common way to accurately synchronize air cylinders. One advantage of mechanical synchronization is that the cylinders can operate anyplace in the stroke without getting out of phase. The accuracy of mechanical synchronization is about ±0.005 to 0.010 in. -- depending on load variation and strength of the mechanism used.

The most accurate way to synchronize hydraulic cylinders is with servovalves. Servovalves independently control each cylinder with electronic position feedback, and compare each actuator's position with all others. This is the most expensive way to synchronize cylinders but the most accurate. Actuator position within ±0.001 to 0.002 in. of each other is attainable using good servo practices. (This type of synchronizing also works well with cylinders that never go to a home position.)

This chapter deals with ways to synchronize cylinders by using other fluid power components. These circuits show how to arrange the components to hold multiple cylinder positions in close proximity to each other. The simplest circuit uses only flow controls to build resistance to hold the fast cylinder back. The accuracy of flow-control synchronizing is only fair to poor. Some of the more complex ways -- such as using tandem cylinders or a master-slave cylinder arrangement -- hold relative position as low as ±0.010 to 06 in.

To use fluid-power components to synchronize cylinders, all cylinders must come to a positive dead stop at the end of each cycle. Leakage in cylinder seals or valving causes minor position differences after each stroke. When the cylinders all bottom out or meet a positive, level stop, the error of each cycle cannot accumulate. This is the main reason not to use fluid-power synchronizing with cylinders that operate only in mid-stroke.

When testing cylinder synchronization on a machine, always start the circuit with the cylinders detached from the machine. Cycle the cylinders without any load attached. This allows a safe time for air purging and valve adjustment. Any sudden or out-of-control moves will not affect machine members.

Synchronizing with flow controls

The circuit in Figure 22-1 has no controls except the directional valve. If the pipes are all the *same relative size and all the same length; if the load is centered; and if friction of all parts is identical, the cylinders might travel exactly together. Some of these variables are controllable, but things like friction may change even during a single cycle. With the setup in Figure 22.1, the cylinders actually move one at a time until they hit end of stroke or bind up mechanically.*

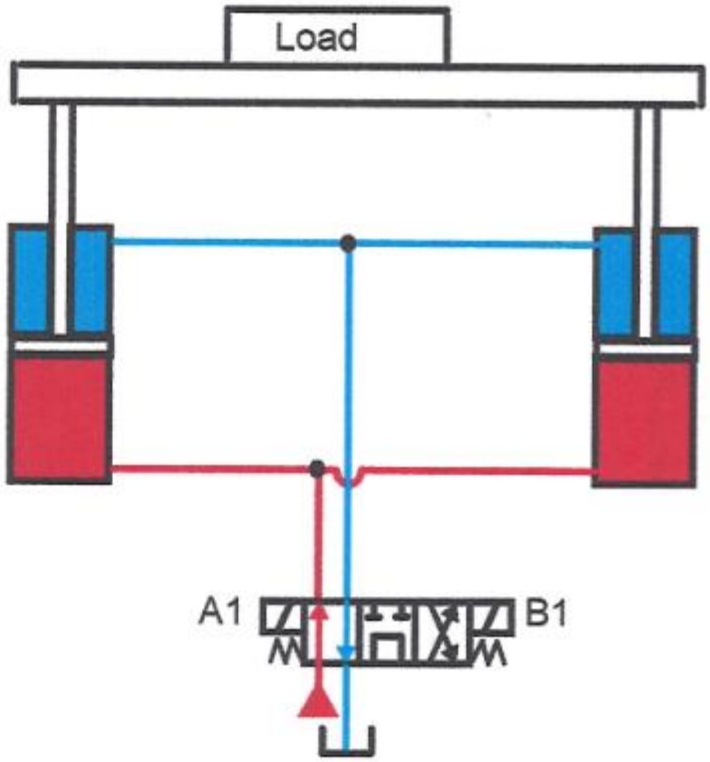


Figure 22-1. Two cylinders in parallel with equal loading.
Synchronization is possible if everything matches perfectly.

With the off-center load shown in Figure 22-2, the cylinder farthest from the load would extend until it stroked out or locked up -- before the opposite cylinder starts to stroke.

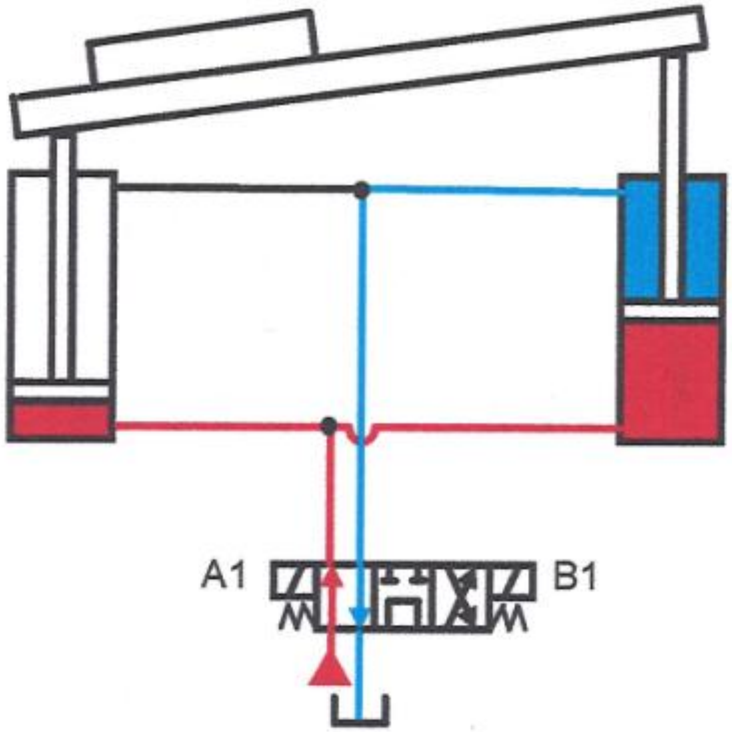


Figure 22-2. Two cylinders in parallel with unequal loading. Cannot synchronize even when everything matches perfectly.

Adding meter-out flow controls to each cylinder port, as in Figure 22-3, adds variable resistance for each cylinder. The added resistance may need to be changed throughout the day because of many factors that affect cylinder movement.

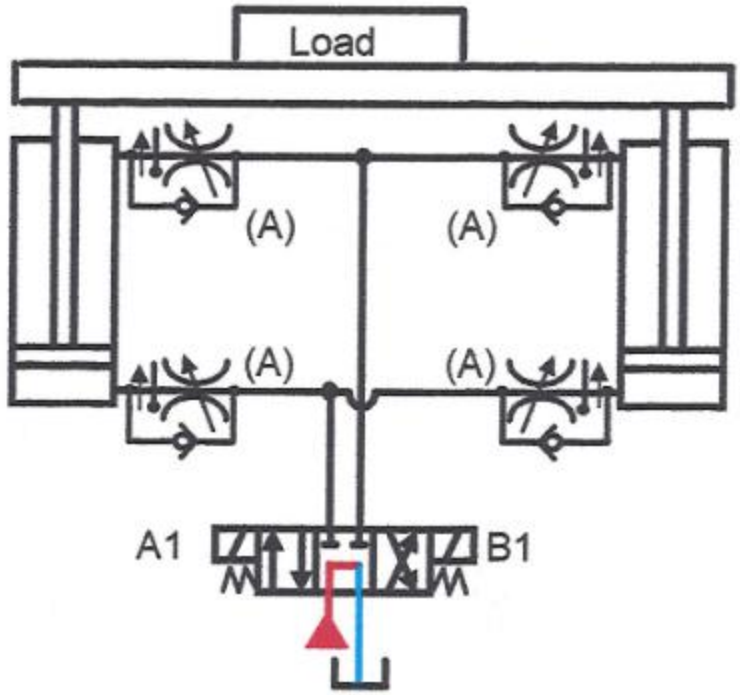


Figure 22-3. Meter-out flow-control circuit for synchronizing. Flow controls can compensate for mismatched parts and loading.

Flow-control synchronizing circuits work with air or hydraulic cylinders. For air cylinders, the problem of compressibility increases potential instability. However, without going to a mechanical or hydraulic option like the tandem-cylinder circuit described in Chapter 3, it is the only way to synchronize air cylinders using fluid power alone.

With flow controls, the cylinders stay reasonably synchronized only if load position does not change. If the load moves, cylinder force must change to maintain synchronization. If load position change is infrequent, resetting flow controls is an option.

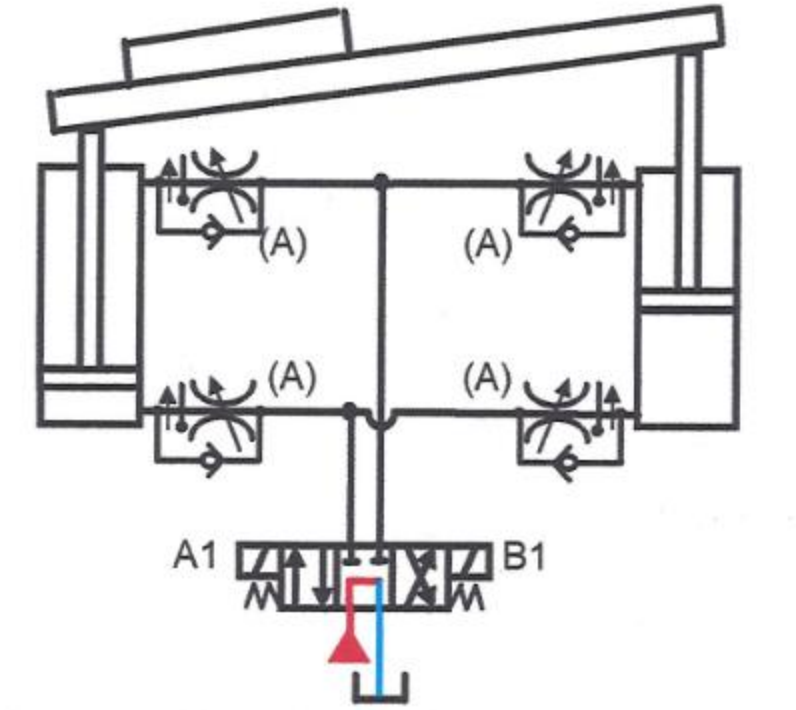


Figure 22-4. Meter-out flow-control circuit for synchronizing. May allow platen to drift when stopped in mid-stroke with unbalanced load.

Even with hydraulics, another problem with uneven loads is what happens if the cylinder does not stroke all the way. If the cylinder stops in mid stroke, as in Figure 22-4, oil from the loaded cylinder can transfer to the opposite cylinder and throw the platen out of synchronization. Figure 22-5 shows pilot-operated check valves added to the cap-end lines to overcome oil transfer during mid-stroke stopping. With these check valves in place, oil cannot transfer when the cylinders stop in mid-stroke, so the cylinders maintain their positions.

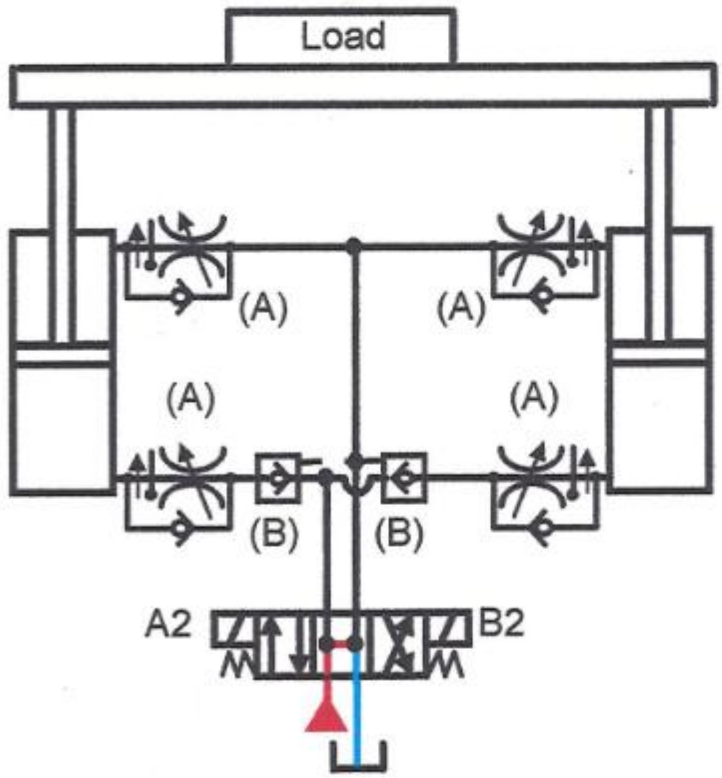


Figure 22-5. Meter-out flow-control circuit for synchronizing -- with pilot-operated check valves to stop cylinder drifting after mid-stroke stop.

Another problem with flow-control synchronization is the maximum lifting force. With two identical cylinders positioned parallel to each other, the platen should be able to lift twice each cylinder's force. However, this is only true if the load is centered. With a double load positioned over one cylinder, that cylinder would stall while the opposite cylinder tries to extend. When using flow-control synchronization, size each cylinder to carry the whole load if the load might get off center.

When controlling hydraulic cylinders, it is best to use pressure-compensated flow controls. Pressure-compensated flow controls maintain a constant flow when load differences cause a change in pressure drop.

Double-rod end cylinders in series

Figure 22-7 shows a fairly accurate way of synchronizing cylinders using double-rod end cylinders piped in series. Oil from the directional valve extends the first cylinder, the first cylinder's top port supplies oil to extend the second cylinder, and the second cylinder's top port connects to the other port of the directional valve. In this arrangement, oil trapped between the *cylinders must have a means of replenishing or draining. As this circuit operates, cylinder seal leakage either depletes the trapped volume or adds to it. Either situation alters synchronization adversely.*

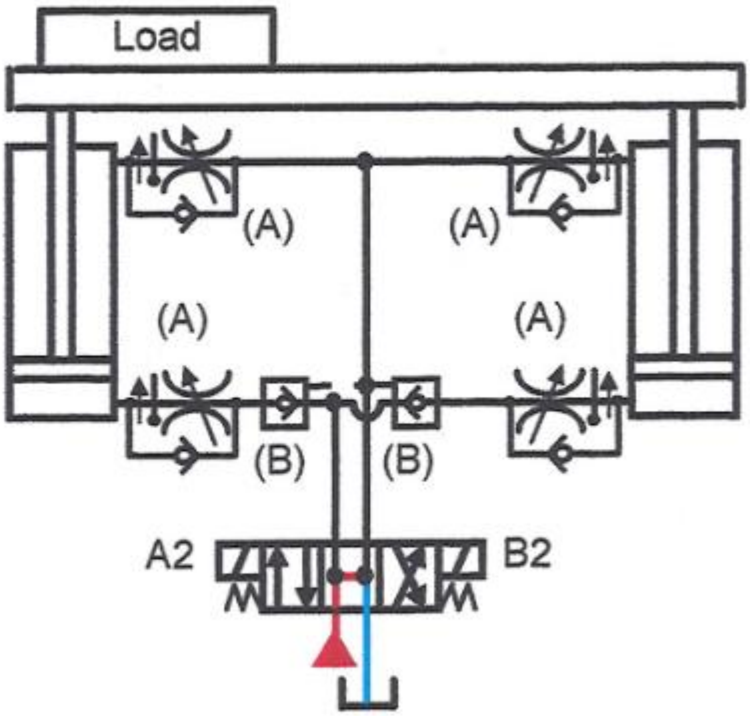


Figure 22-6. Meter-out flow-control circuit for synchronizing. Can only raise a load equal to the force of one cylinder.

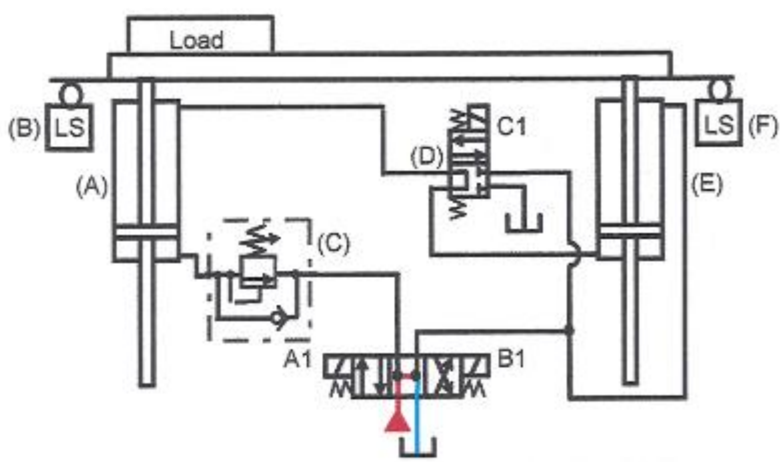


Figure 22-7. Synchronizing circuit with double-rod end cylinders in series flow -- at rest with pump running.

In Figure 22-7, when 2-position, spring-centered, single-solenoid, tandem-center leveling valve D is deenergized, it allows oil to flow from cylinder (A) to cylinder (E). The valve is deenergized while the cylinders extend and retract to do work. (Figure 22-10 shows how the cylinders are leveled at the end of a cycle.)

Energizing solenoid A1 of the main directional valve, as in Figure 22-8, sends oil to cylinder (A) , causing it to extend. Oil from the opposite end of cylinder (A) flows through leveling valve (D) to the push end of cylinder (E). Oil from the opposite end of cylinder (E) flows to tank through the main directional valve. When the trapped volume is completely full and if all seals do not leak, the cylinders synchronize nearly perfectly, regardless of load position.

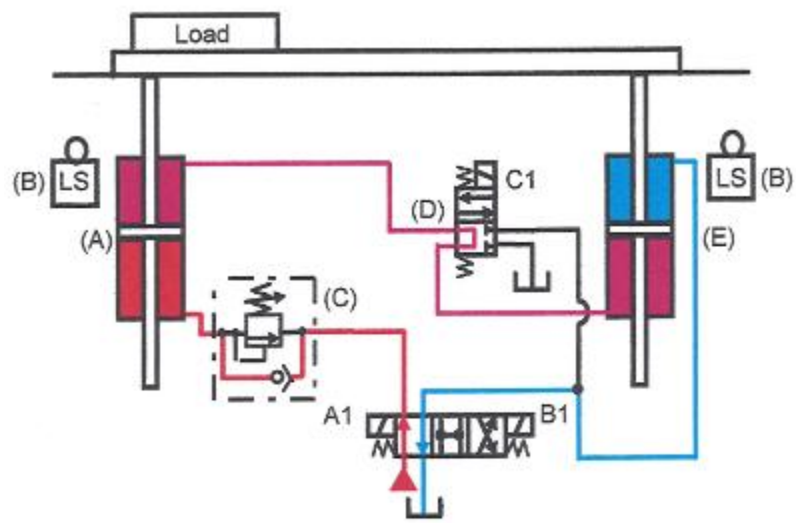


Figure 22-8. Synchronizing circuit with double-rod end cylinders in series flow. Cylinders are extending.

To retract the cylinders, energize solenoid B1 of the main directional valve as in Figure 22-9. This sends oil to the retract side of cylinder (E). Oil from the opposite end of cylinder (E) flows through leveling valve (D) to the top of cylinder (A). Oil from the opposite end of cylinder (A)flows to tank through the counterbalance valve and main directional valve.

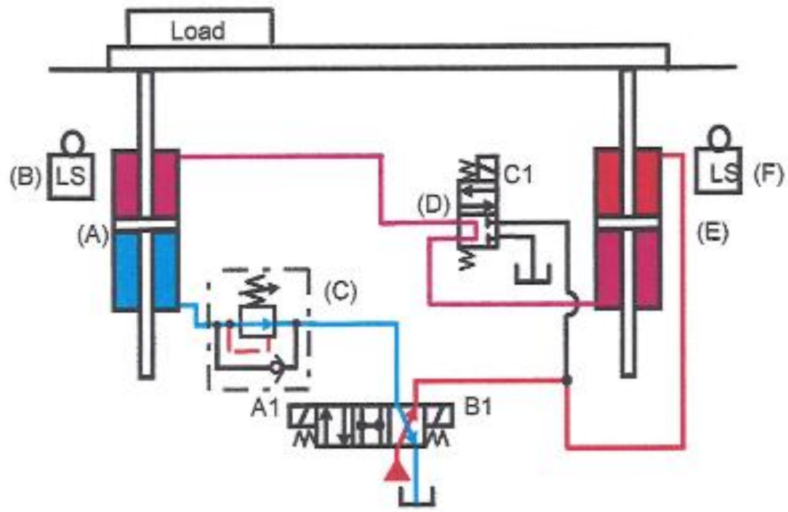


Figure 22-9. Synchronizing circuit with double-rod end cylinders in series flow. Solenoid B1 energized, cylinder retracting.

Figure 22-10 shows how the cylinders maintain synchronization as they cycle. When the platen nears bottom, it contacts limit switches B and F. If the switches make simultaneously, no leveling occurs. If one limit switch makes before the other, the cylinders obviously are out of synchronization, so solenoid C1 on the leveling valve energizes. With solenoids B1 and C1 energized, pump oil flows to the retract sides of cylinders (A) and (E), forcing them to retract fully. Cylinders (A) and (E) can retract because the extend sides of both cylinders have a direct path to tank. When both limit switches make, the leveling valve and retract solenoids deenergize. (This leveling circuit also works for horizontally mounted cylinders.)

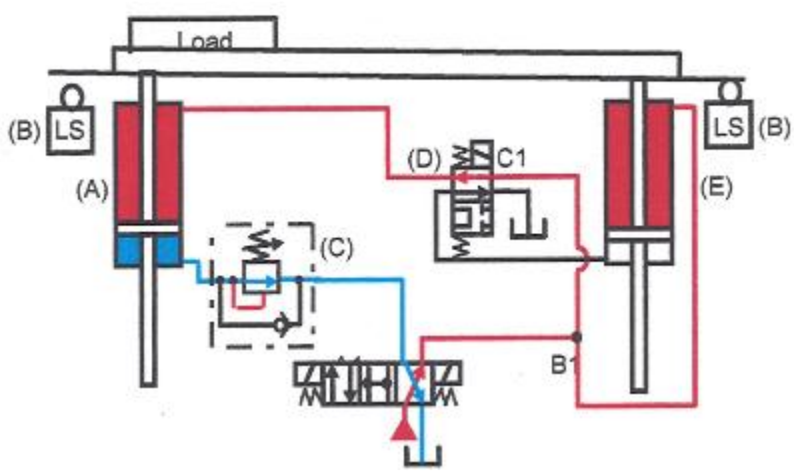


Figure 22-10. Synchronizing circuit with double-rod end

cylinders in series flow. Solenoids B1 and C1 energized, cylinders leveling.

With series cylinder synchronizing, load placement is not important. The cylinders stay level regardless of load position or weight. The only things a heavy off-center load might cause are more seal leakage, or oil volume changes due to compressibility.

It is important to note that, because the cylinders are in series, they each have to be able to lift the total load. No matter the load placement, or the number of cylinders in series, each one must be capable of lifting the entire load. At the same time only one cylinder's volume is considered when calculating pump flow.

Other ways to use cylinders in series

To save cost, reduce potential leakage at the extra rod seals, and eliminate space needed for the second rod, use the circuit in Figure 22-11. The cylinders in this circuit oppose one another, so one extends while the other retracts. This is one way to synchronize single-rod cylinders in a series circuit. Connecting identical rod end volumes together allows series synchronization the same as double-rod end cylinders. Space for the top cylinder could be a problem on some machines so the circuit in Figure 22-12, although more expensive, works equally well. (Use the same tandem-center valve makeup circuit as seen in Chapter 21, figures 7-10 to level the cylinders after each stroke.)

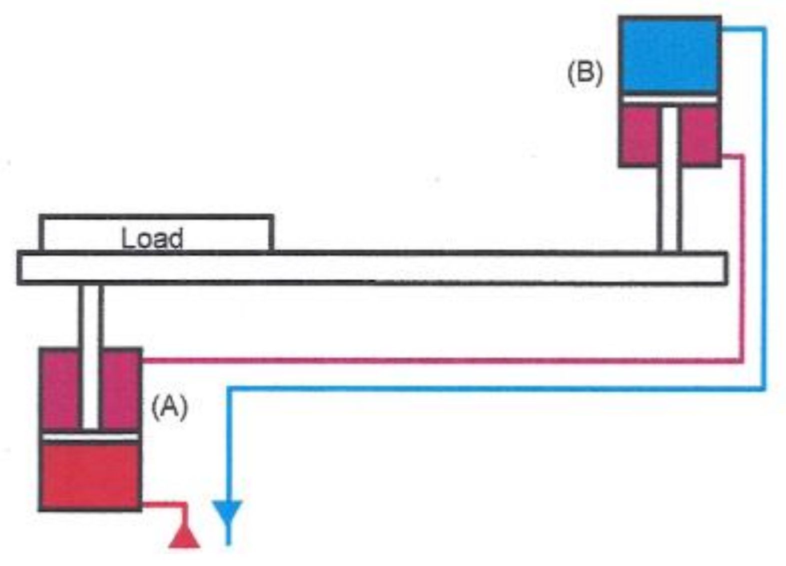


Figure 22-11. Alternative cylinder positions for a series-flow synchronizing circuit using single-rod end cylinders.

Mounting is more conventional using three single-rod cylinders piped as in Figure 22-12. The only purpose of cylinder (B) is to connect equal areas. This design is still less expensive than two double-rod cylinders and it has one less leak source. This circuit requires make up valves that allow cylinder (C) to retract, cylinder (A) to retract without cavitation, and cylinder (B) to stroke if the other two do not reach home position simultaneously.

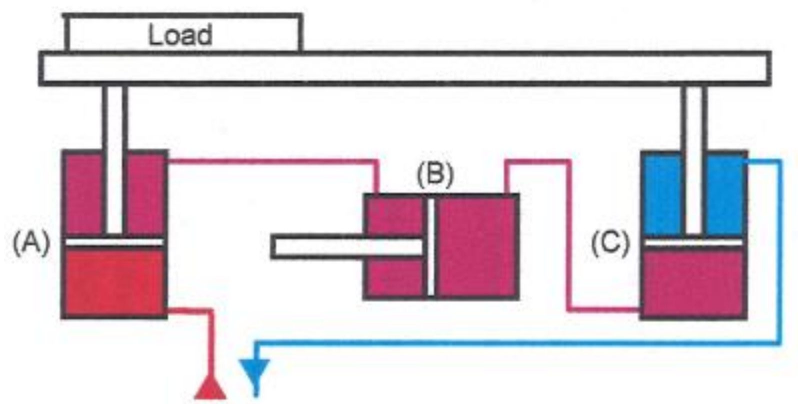


Figure 22-12. Alternative cylinder positions for a series-flow synchronizing circuit using single-rod end cylinders.

Figures 22-13 through 14 show how to attain reasonable synchronization with a set of equalizing flow controls on single-rod end cylinders in series. The cylinders are extending in Figure 22-13. Oil from the directional valve goes through needle valve (C) to the cap end of cylinder (B), thus controlling its speed. At the same time, some bleed oil from the directional valve goes through needle valve (D) to the cap end of cylinder (A). Set needle valve (D) to make up for lower oil volume as it transfers from the rod end of cylinder (B) to the cap end of cylinder (A). Without needle valve (D), cylinder (A) would lag every cycle and be out of synchronization. Changing flow at needle valve (C) means readjusting needle valve (D) also. Both needle valves work best if they are pressure compensated. This is a problem in this circuit because there is bi-directional flow. Refer to Chapter 10, Figure 10-4 to see a pressure-compensated needle valve piped for bi-directional flow.

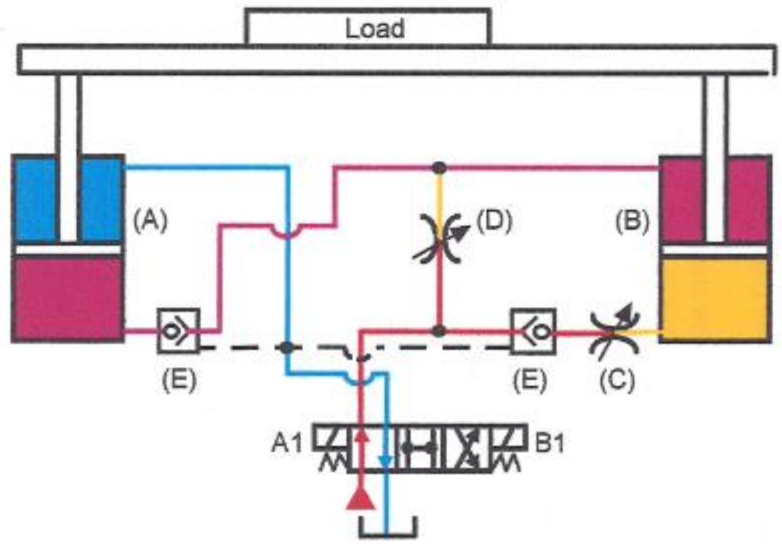


Figure 22-13. Alternative cylinder in series flow synchronizing circuit using single-rod end cylinders. Solenoid *A1* energized, cylinders extending.

To retract the cylinders, the directional valve shifts as in Figure 22-14, porting oil to the rod end of cylinder (A). As cylinder (A) retracts, oil from its cap end transfers to the rod end of cylinder (B). Excess oil volume from cylinder (A) goes directly to tank through needle valve (D). Needle valve (C) controls the up and down speeds of the platen.

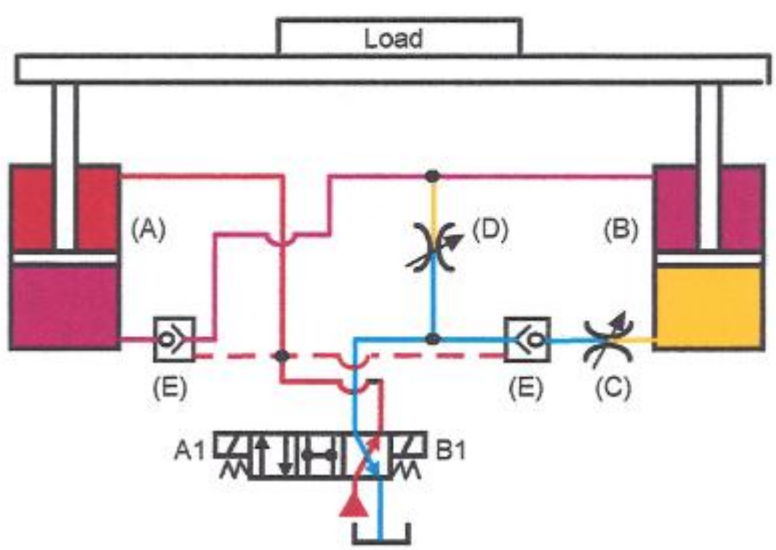


Figure 22-14. Alternative cylinder in series flow

synchronizing circuit using single rod end cylinder. Solenoid B1 energized, cylinders retracting.

Each cylinder in a series circuit must be powerful enough to lift the entire load. When load position changes, it affects synchronization due to the resulting change in pressure drop across needle valve (D). An off-center load that is too heavy for one cylinder to lift still allows oil transfer through needle valve (D), throwing the platen out of synchronization. Add pilot-operated check valves (E)if the cylinders must stop in mid stroke. Without these pilot-operated checks, oil transfer through needle valve (D) allows the cylinders to drift.

Double-pump-and-valve synchronizing circuit

Figures 22-15 through 18 illustrate a common way of synchronizing cylinders. Many designers use this circuit and consider it to be one of the best ways to synchronize cylinders. It is reasonably accurate, but may allow the cylinders to get out of phase in certain conditions.

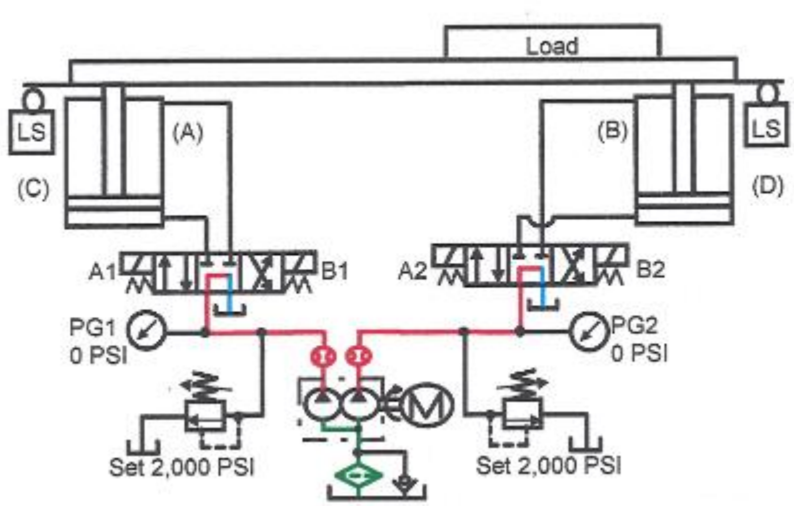


Figure 22-15. Double-pump-and-valve synchronizing circuit -- at rest with pump running.

The two pumps in Figure 22-15 have identical flow. They are attached to two double-solenoid, spring-centered valves that are piped to two matching cylinders. Both pumps have a relief valve set at the same maximum pressure. Because both pumps have the same flow and both cylinders use the same volume, the cylinders will stroke at approximately the same rate.

The cylinders are shown extending in Figure 22-16. Energizing solenoids A1 and A2 on the directional valves simultaneously causes the cylinders to extend at the same rate. If one *cylinder's load needs more pressure, the pump for that side continues to feed nearly the same flow until the relief valve dumps.*

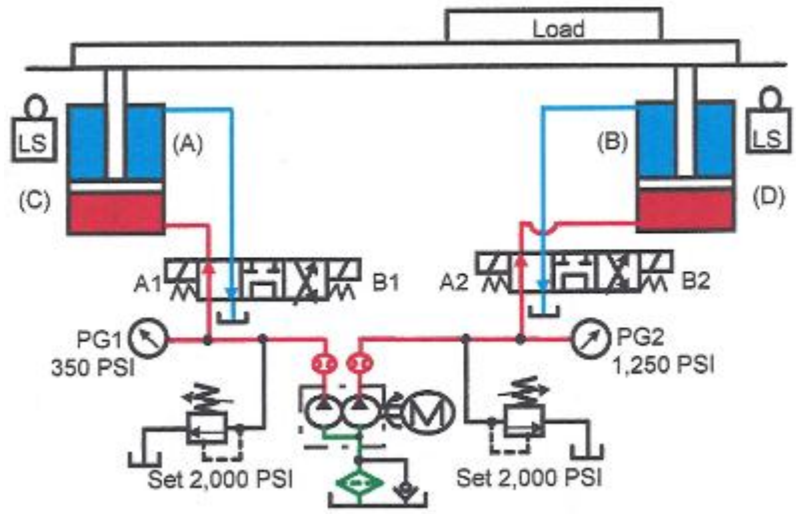


Figure 22-16. Double-pump-and-valve synchronizing circuit. Solenoids A1 and A2 energized, cylinders extending.

To retract the cylinders, energize solenoids B1 and B2 on both directional valves simultaneously, as in Figure 22-17. The cylinders retract at the same rate.

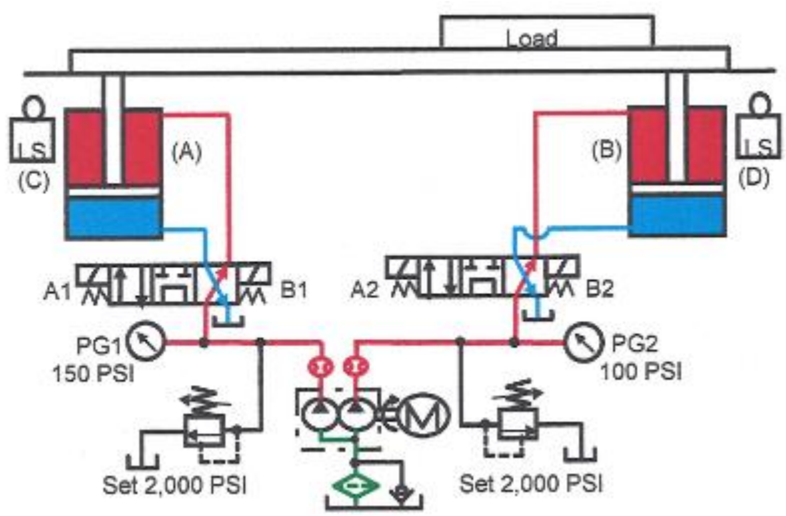


Figure 22-17. Double-pump-and-valve synchronizing circuit. Solenoids *B1* and *B2* energized, cylinders retracting.

Should the cylinders get out of phase, Figure 22-18 shows how they re-synchronize. Because a separate pump and valve control each cylinder, separate limit switches drop out the retract solenoids after the cylinders reach home. This leveling happens automatically during each cycle, so position errors do not accumulate.

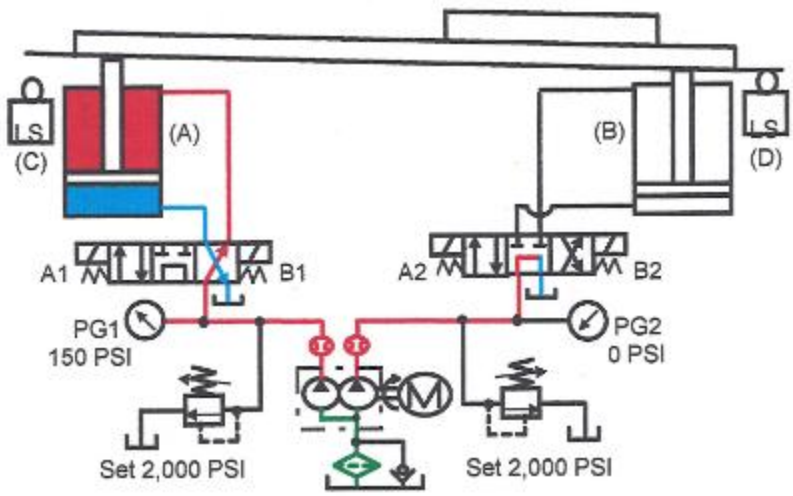


Figure 22-18. Double-pump-and-valve synchronizing circuit.
Solenoid B1 energized, cylinder (A) leveling.

A major problem with this synchronizing circuit is the difficulty of finding two identical pumps. Even pumps manufactured at the same time often have slightly different flows. Any flow variation of the pumps lets the cylinders get out of phase. Another problem is efficiency. As pressure climbs, pump efficiency allows more slip oil, valves leak more, and some cylinder seals bypass more. All of these losses add up to poor performance especially if the cylinders have long strokes.

On top of that, what happens if one solenoid is sluggish or fails to operate? This makes one cylinder start late or not start at all. Starting late causes the cylinders to be out of phase; not starting at all may damage the machine.

This circuit has the same force problem as a flow-control synchronizing circuit. Each cylinder has to be able to lift the entire load. If the load on this circuit gets too heavy for one cylinder, its pump dumps across the relief valve and the cylinder stops. Again the other cylinder continues extending until it damages itself or the machine.

Double-pump-and-valve synchronizing circuit improvement
The circuit changes shown in Figure 22-19 overcome most of the problems mentioned about Figures 22.-5 through 18. Instead of two cylinders as before, use two or more pairs of cylinders. Connect half of the cylinders to each pump/valve combination. Pipe port A of directional valve

(E) to the caps of cylinders (A) and (C). Hook port B of directional valve (E) to the rod ports of cylinders (B) and (D). Pipe port A of directional valve (F) to the cap end of cylinders (B) and (D) with its B port hooked to the rod ports of cylinders (A) and (C). Piping the circuit this way uses one pump and valve to extend two cylinders, while this same valve retracts the cylinders extended by the other pump and valve.

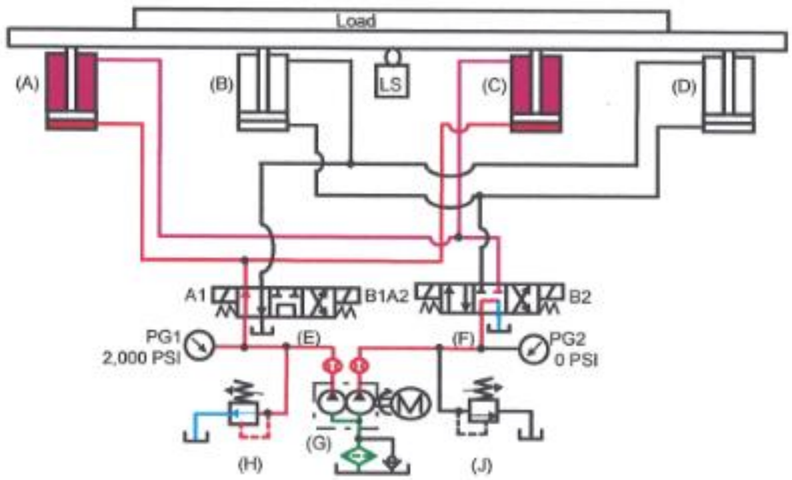


Figure 22-19. Modified double-pump-and-valve synchronizing circuit. Solenoid A1 on left-hand valve shifted to show condition if solenoid A2 is sluggish or fails to shift.

Should a solenoid fail, as in Figure 22-19, the platen will not move because, while cylinders (A) and (C) may be trying to extend, oil from their rod end ports cannot get back to tank through valve (F). Also, blocked inlet flow to cylinders (B) and (D) at valve (F) prevents them from stroking -- although leakage past the spool in valve (F) may allow minor movement.

After both directional valves shift and the cylinders are stroking as in Figure 22-20, the pairs of cylinders try to stay level. If pump (G) produces higher flow, cylinders (A) and (C) try to run ahead. Because cylinder (B) is between them, it will either hold the other cylinders back or be dragged along by them. The platen must be strong enough to transmit this differential cylinder loading without flexing.

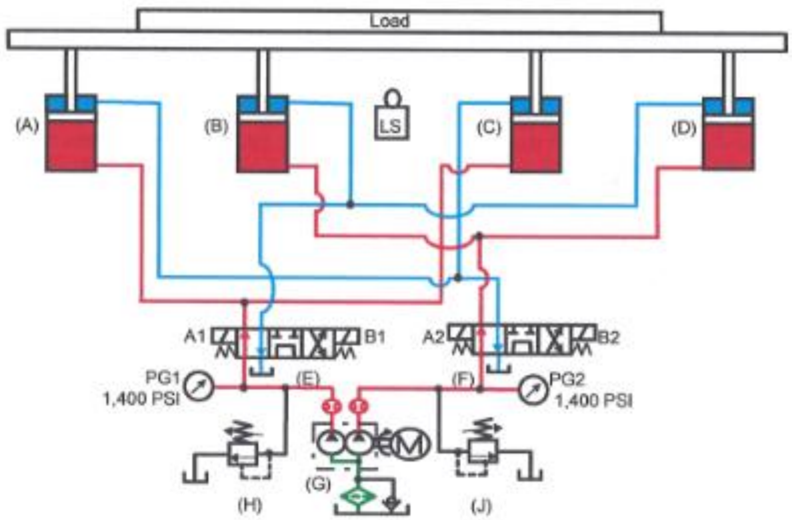


Figure 22-20. Modified double-pump-and-valve synchronizing circuit. Solenoids A1 and A2 energized, cylinders extending.

This circuit is less load-sensitive because the load is always over a pair of cylinders operated by different pumps. Both pumps will relieve to tank before the load stops moving. However, the lightly loaded cylinders can move ahead in relation to the stiffness of the platen and the distance between cylinders.

Use only one limit switch for this cylinder arrangement. To re-phase the cylinders, shift both directional valves to send the cylinders to home position. One relief valve bypasses fluid until the lagging cylinders reach the positive stop.

Spool-type flow-divider synchronizing circuit

Spool-type flow dividers split flow from a single conductor into two separate flows. The split flows may be at different rates if needed, but for cylinder synchronization, they usually are equal. Spool-type flow dividers basically consist of two pressure- compensated flow controls in one body. In this arrangement, each flow control's pressure drop modifies the opposite flow output. Because these flow controls constantly look at each other's pressure drop, they split flow relatively well. (Most manufacturers claim about ±5%, depending on the pressure differential at the outlets.)

One problem with spool-type flow dividers is that they do not allow reverse flow. Even if they did, there would be no guarantee of equal flow. A spool-type flow divider/combiner allows forward and reverse flow, and equally splits or combines the two flows. Normally a flow divider/combiner is the component of choice in cylinder synchronizing circuits. Figure 22-21 *shows a spool-type flow divider/combiner synchronizing circuit. It is similar to a double-pump circuit, but only uses one pump and valve. Flow is split downstream from the single directional valve.*

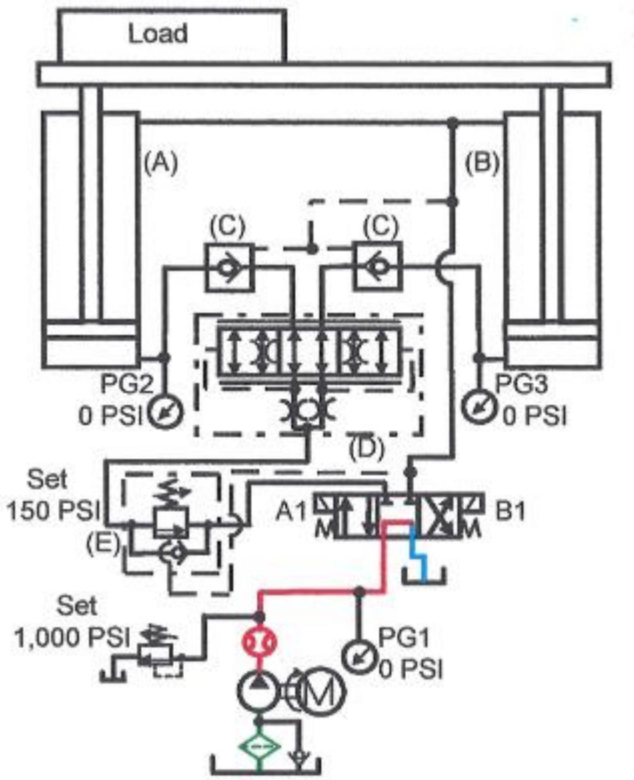


Figure 22-21. Spool-type flow-divider/combiner synchronizing circuit -- at rest with pump running.

In Figure 22-22, the cylinders are extending. Shifting solenoid A1 on the directional valve sends oil to the flow divider, which sends half pump flow to each cylinder. Even when there is a pressure difference at the cylinders, flows are close to equal. The cylinders extend at about the same rate even with an off-center load. Each cylinder must develop enough force to lift the load above it. If one cylinder reaches its force limit and stops, the opposite cylinder tries but does not completely stop -- due to internal leakage past the flow divider spool. (Figure 22-24 shows the condition of the flow divider when cylinder (B) stalls as it retracts.)

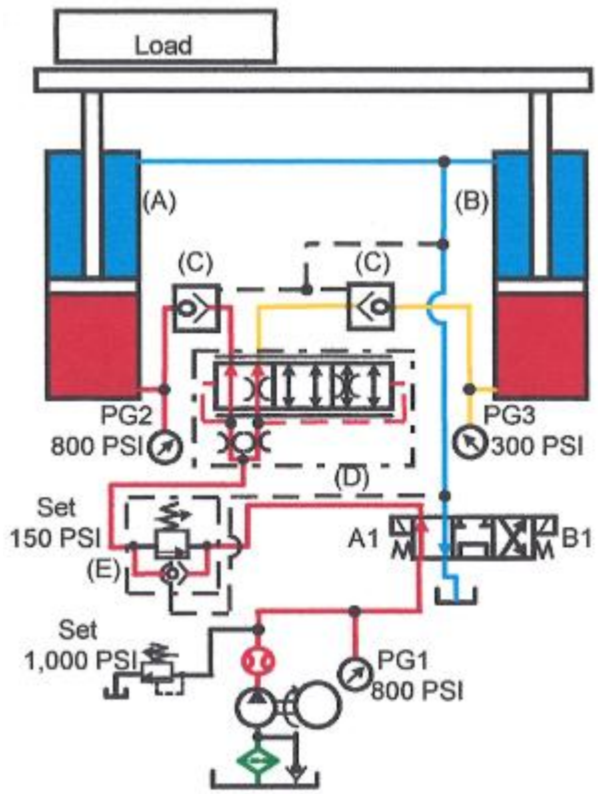


Figure 22-22. Spool-type flow-divider/combiner synchronizing circuit. Solenoid A1 energized, extending.

Figure 22-23 shows the circuit after energizing solenoid B1 on the 4-way directional valve. Oil flows to the cylinder rod ends while fluid from the cylinder cap ends combines equally at the flow divider and flows on to tank. The flow divider holds back the cylinder that wants to get ahead -- thus maintaining synchronization. When the cylinders reach bottom, they re-phase automatically if the directional valve is left in the down mode long enough. Internal leakage in the flow divider spool allows the lagging cylinder to continue stroking. (Some flow-divider brands have integral bypasses that operate when the pressure differential reaches a pre-set limit.)

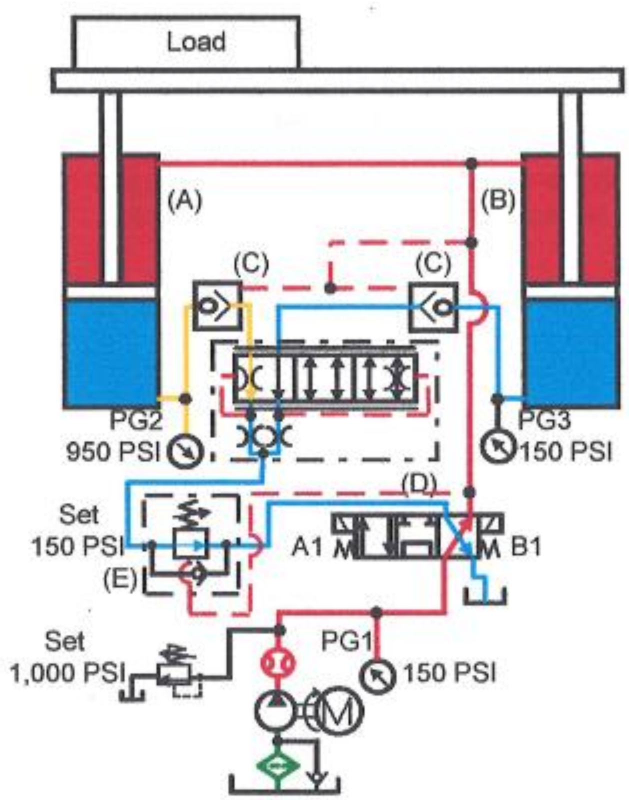


Figure 22-23. Spool-type flow-divider/combiner synchronizing circuit. Cylinder (B) bound up.

Because the flow divider has a common path internally, fluid can flow between the cap end ports. If the cylinders need to stop in mid-stroke, always use pilot-operated check valves (C) to prevent oil transfer. Control an overrunning load with counterbalance valve (E) between the flow divider and directional valve.

Spool-type flow dividers waste energy. Notice the gauge reading at each cylinder as it extends, PG2 shows 800 psi while PG3 reads 300 psi. In this situation, gauge PG1 at the pump reads 800 psi. The 500-psi drop across the right side of the flow divider generates heat when the cylinders extend.

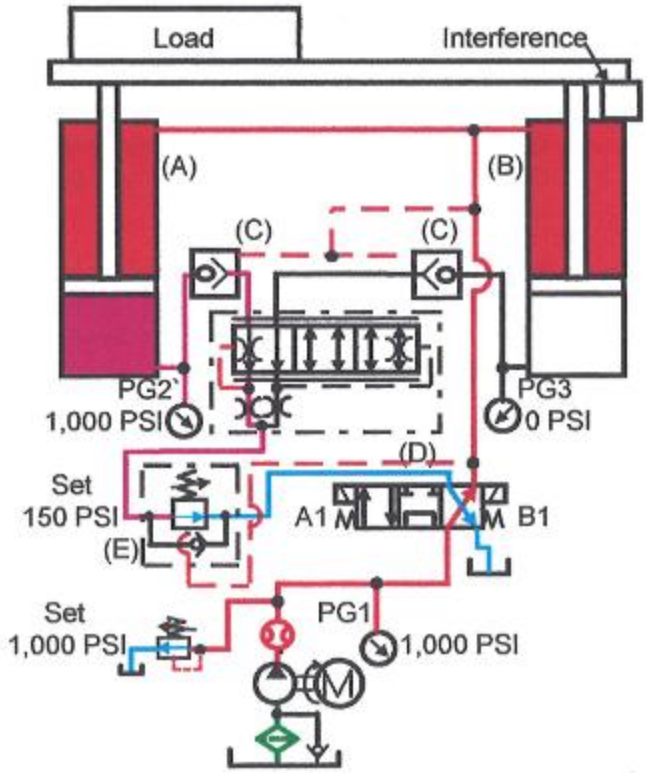


Figure 22-24. Spool-type flow-divider/combiner synchronizing circuit. Solenoid B1 energized, retracting.

Spool type flow dividers only split flow into two outputs. It would take three spool-type flow dividers to split flow four ways.

Motor-type flow divider synchronizing circuit
Motor-type flow dividers do not waste energy and are more versatile. One motor-type flow divider can splits flow from a pump and run two or more cylinders in unison. Plus, they offer multiple outlets -- up to ten or more -- and can pass unequal flows when required.

A motor-type flow divider consists of two or more hydraulic motors in one housing. The motors have a common shaft. Thus, when one motor turns, all motors turn. The motors share a common inlet but have separate outlets. Fluid from the pump enters all motors at once, rotating then in unison. If the motors are the same size, output from each section is an equal portion of inlet oil. Because a mechanical motor -- instead of an orifice -- splits flow, there is no energy loss due to different outlet pressures. Figure 22-25 shows a motor-type flow divider synchronizing two cylinders. The flow divider is installed between the directional valve and the cylinders in this circuit.

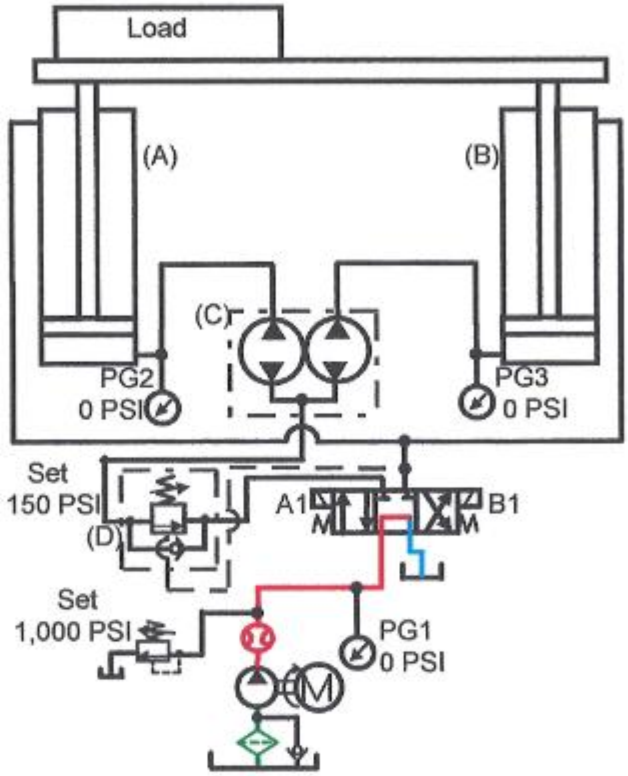


Figure 22-25. Motor-type flow divider synchronizing circuit -- at rest with pump running.

In Figure 22-26, solenoid A1 is energized to shift the 4-way directional valve. This sends oil to the flow divider, which sends equal volumes to each cylinder. The accuracy of motor-type flow dividers depends on the amount of pressure difference at the outlets. The motors have internal slippage that increases as pressure drop increases. The greater the pressure difference, the greater the flow difference and loss of synchronization.

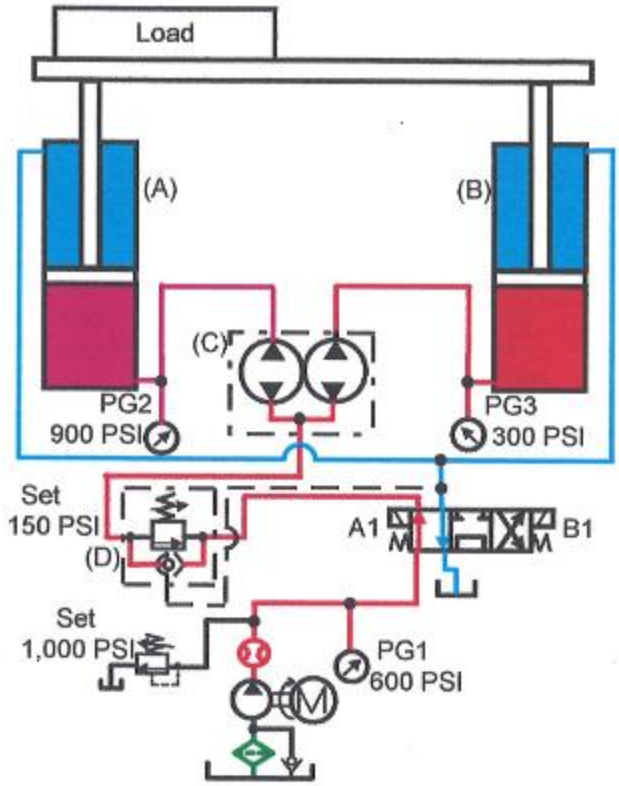


Figure 22-26. Motor-type flow divider synchronizing circuit. Solenoid A1 energized, extending.

In Figure 22-27, the cylinders are retracting. Energizing solenoid B1 on the directional valve sends oil from the pump to the cylinders' rod ends. As the cylinders retract, oil flows from the cylinders' cap ends through the flow divider to tank. The flow divider combines the cylinder flows and maintains synchronization when the cylinders travel freely.

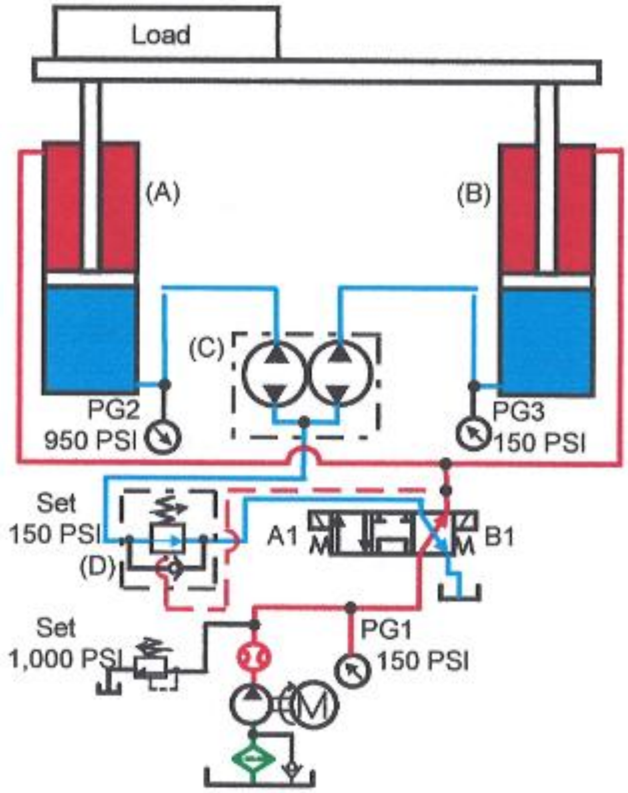


Figure 22-27. Motor-type flow divider synchronizing circuit. Solenoid B1 energized, retracting.

If one cylinder binds up and stops traveling, as in Figure 22-28, all oil from the pump goes to the free-moving cylinder. The flow divider section that is not getting oil from the stopped cylinder continues to turn and cavitate, causing the free cylinder to retract at twice speed. When there is a chance of cylinder binding, install a motor-type flow divider at both ends of the cylinders. A flow divider on the rod end forces the binding cylinder to synchronize or stalls them both.

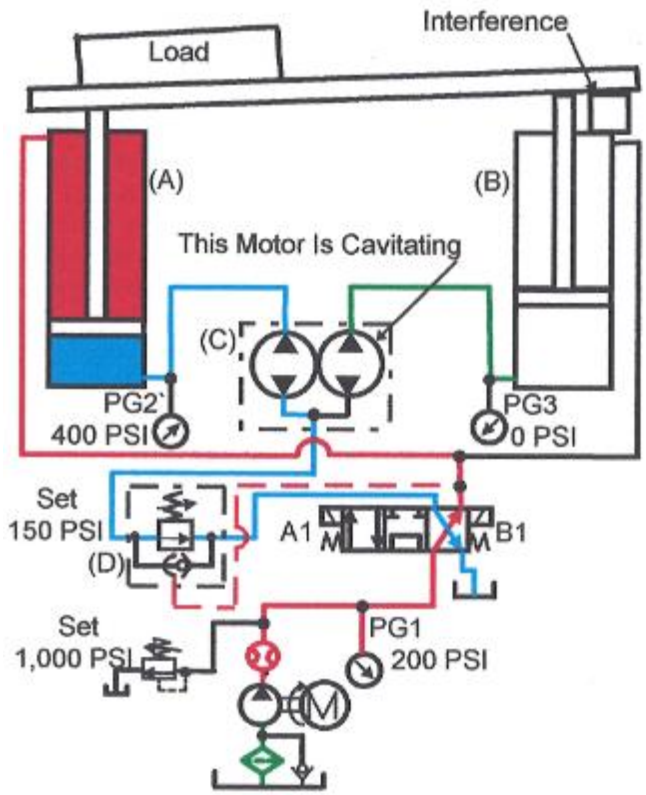


Figure 22-28. Motor-type flow divider synchronizing circuit. Cylinder (B) bound up.

The internal slip of motor-type flow dividers is usually sufficient to level the cylinders. Another option is integral relief valves that allow fluid to bypass a motor at a predetermined adjustable pressure.

As mentioned, an advantage of motor-type flow dividers is that they waste little energy. Notice the gauge values in Figure 22-26. The left cylinder requires 900 psi, while the right cylinder only needs 300 psi. With those conditions, the inlet pressure to a spool-type flow divider has to be 900 psi. With a motor-type flow divider, the inlet pressure only has to be 600 psi. Because the motor-type flow divider has a mechanical link through a common shaft, energy transfer between sections lowers the required pressure at the inlet.

Another advantage is that motor-type flow dividers with two, three, even ten or more outlets are common. Instead of stacking 2-outlet spool-type dividers, use only one multiple-outlet motor-type flow divider for many circuits.

One caution: motor-type flow dividers will intensify outlet pressure as they operate. (See Chapter 11 for an explanation of motor-type flow divider intensification.) With a 2-outlet equal-

flow divider, if relief valve pressure is over half the maximum rated pressure of any component it feeds, install a relief valve at each outlet. The outlet relief valves protect the cylinders, valves, and lines from excess pressure.

Master-and-slave cylinder synchronizing circuit

Figures 22-29 through 32 show one of the most accurate ways to hydraulically synchronize cylinders. Figure 22-29 shows the circuit at rest. Cylinder (C) -- mechanically linked to two cylinders (D) -- provides the driving force. The (D) cylinders have the same bore, stroke, and rod diameter as working cylinders (A) and (B). One cylinder (D) connects to cylinder (A), while the other cylinder (D) connects to cylinder (B). In case of external leakage, makeup check valves (H) let oil into the dead areas of cylinders (A), (B), and(D) at low pressure. A 75-psi backpressure check valve in the tank line gives sufficient pressure to make sure the trapped oil volume stays full. Leveling valves (J) through (M) retract the cylinders to home position when they get out of phase. Limit switches (F) and (G) indicate cylinder home positions and operate the leveling valves when the cylinders get out of synchronization. Counterbalance valve (E) stops the cylinders from running away while they retract.

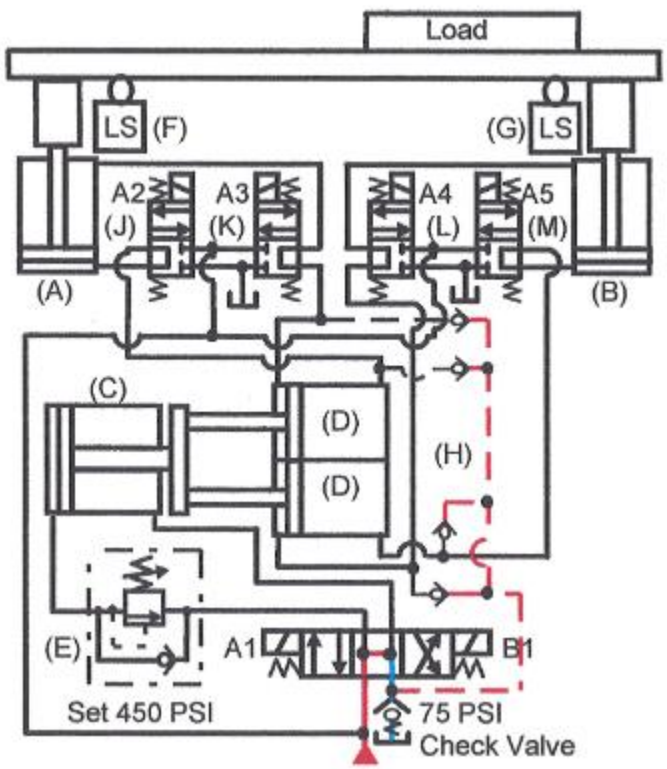


Figure 22-29. Master-and-slave-cylinder synchronizing circuit -- at rest with pump running.

Force from cylinder (C) is enough to do the whole operation by itself. This cylinder produces all the force and passes it on to slave cylinder (D), then to the working cylinders (A) and (B).

The location of the load on the platen affects synchronization only slightly. Energy transfer from the master/slave linkage moves the same volume of oil regardless of pressure. Cylinder (A)operates at twice pressure with the load above it as with a centered load. To protect the cylinders from overpressure, set the relief valve for no more than half the cylinder pressure rating.

Figure 22-30 shows solenoid A1on the 4-way directional valve shifted to extend cylinder (C). Cylinder (C) pushes cylinders (D), and oil from the cap ends of cylinders (D)flows equally to the cap ends of cylinders (A)and (B). Oil from the rod ends of cylinders (A)and (B)returns to the rod sides of cylinders (D). Cylinders (A)and (B) extend in unison if cylinder (C) has enough power to do the job. If one working cylinder stalls, both stop.

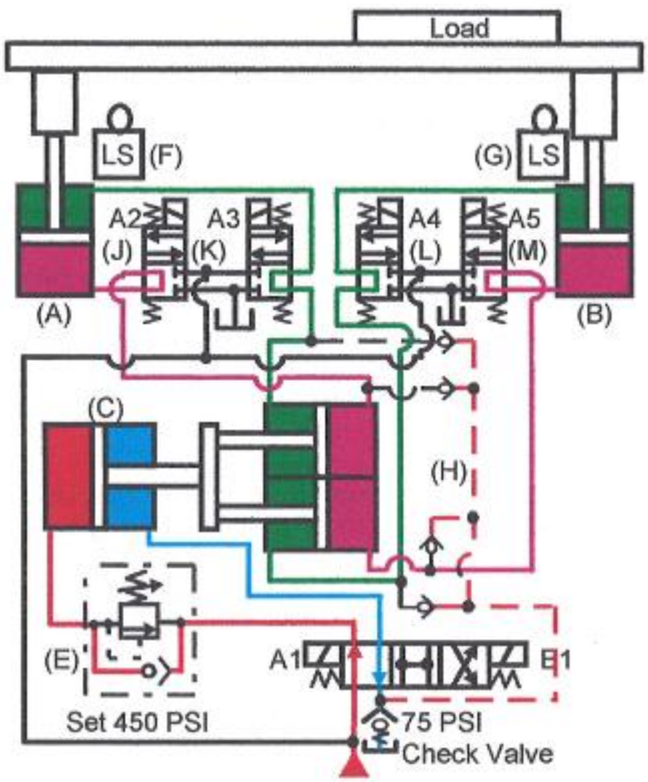


Figure 22-30. Master-and-slave-cylinder synchronizing circuit. Solenoid A1 energized, extending.

To retract the working cylinders, energize solenoid B1 on the 4-way directional valve as in Figure 22-31. Cylinder (C) then retracts and pulls both slave cylinders (D) back, forcing working cylinders (A) and(B) to retract also.

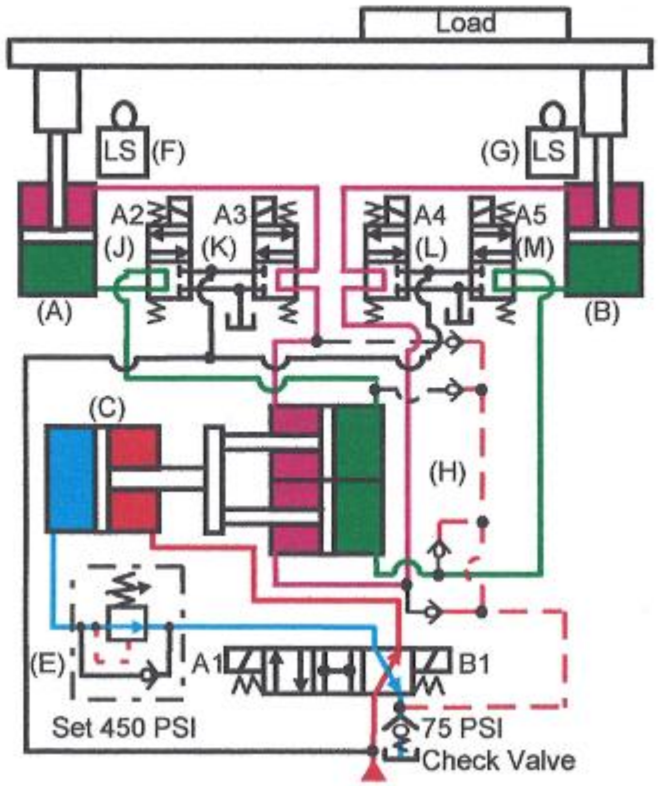


Figure 22-31. Master-and-slave-cylinder synchronizing circuit. Solenoid B1 energized, retracting.

If the working cylinders get out of synchronization, the circuit diagram in Figure 22-32 shows how they level. While solenoid B1 on the 4-way directional valve stays shifted, energize solenoids A2 through A5 on directional valves (J) through (M). This directs pump oil to the rod sides of cylinders (A), (B) and(C), and to the cap sides of both (D) cylinders. At the same time, oil from the cap sides of cylinders (A), (B) and(C) and the rod sides of both cylinders (D) flows to tank. In this condition, the pump forces all cylinders to their home positions, ready for the next cycle.

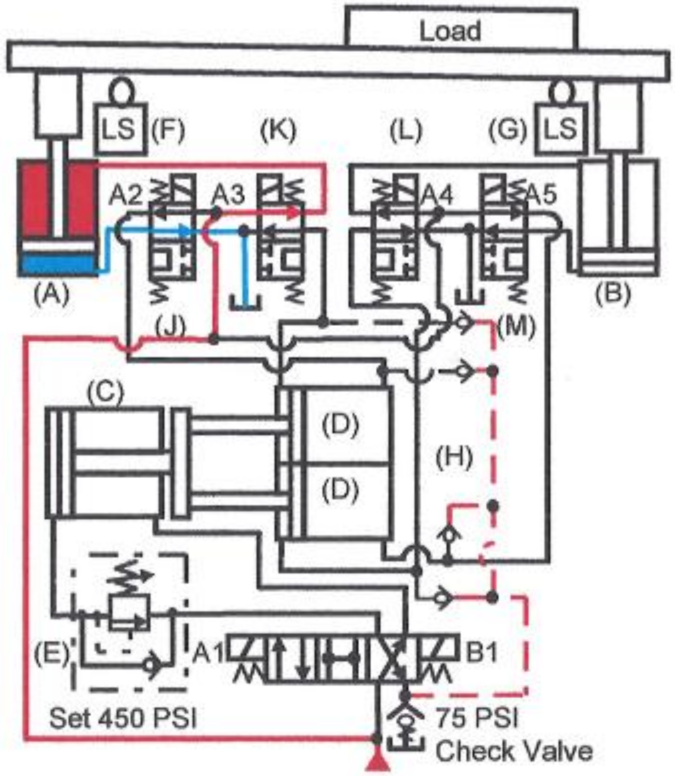


Figure 22-32. Master-and-slave-cylinder synchronizing circuit. Solenoids B1, A2, A3, A4, and A5 energized, leveling.

This circuit is an accurate but expensive way to synchronize cylinders. One advantage is that the master and slave cylinder can be located remotely, to leave the work area less cluttered. Also, energy transfer minimizes the required cylinder size and still handles off-center loads.

Tandem-cylinder synchronizing circuit

Figure 22.-3 shows another very accurate way to synchronize cylinders. The tandem cylinders in this circuit must meet in center even when they run into unequal forces.

Tandem cylinders consist of two cylinders in one housing. They have four ports and the back cylinder is single-rod end while the front cylinder is double-rod end. Because the front cylinder is double-rod end, it has equal areas and volumes on both sides of the piston.

Notice the 4-way directional valve supplies the single-rod cylinders in a conventional manner. The double-rod cylinders have the front port of the left cylinder connected to the back port of the right cylinder and the front port of the right cylinder connected to the back port of the left cylinder.

The tandem cylinders move in unison and transfer energy because hydraulic flow ties them together. If either cylinder stalls, both cylinders stop. Before the cylinders stop, energy transfers through the tandem cylinders and tries to force the lagging cylinder to do its work. The lagging cylinder may see as much as double force before stalling.

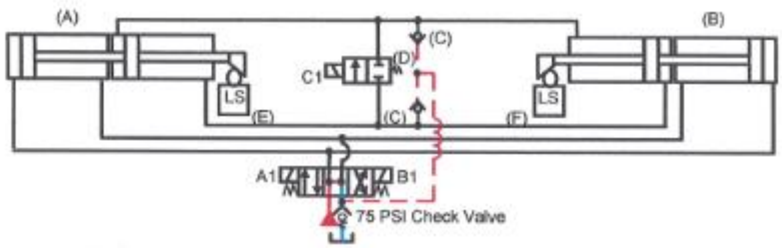


Figure 22-33. Tandem-cylinder synchronizing circuit -- at rest with pump running.

---

*The two check valves **(C)**, fed from a 75-psi backpressure check valve in the tank line, allow makeup oil into the trapped volume of the tandem cylinders. The pump makes up for leakage in the trapped volume through check valves **(C)**. Makeup pressure is equal on both sides of both cylinders so the 75 psi has no effect on them. Always furnish bleed ports at both ends of the tandem cylinders to purge any trapped air.*

*The 2-way, normally closed directional valve **(D)** between the tandem cylinder connecting lines opens to level the cylinders at one end of the stroke. Leakage at the cylinder piston seals may allow the cylinders to get out of phase. Valve **(D)** opens when limit switches **(E)** and**(F)** do not make simultaneously as the cylinders retract. When one limit makes first, valve **(D)** opens and allows fluid transfer from one end of the double-rod end cylinder to the other, until both limits make.*

*In Figure 22-34, solenoid **A1** of the 4-way directional valve is energized and the cylinders extend. As they extend, oil transfer in the tandem cylinders maintains near perfect synchronization. If either cylinder tries to lag, power transfers hydraulically through the tandem cylinder lines to keep them in unison. When the load is too great for both cylinders, they stall.*

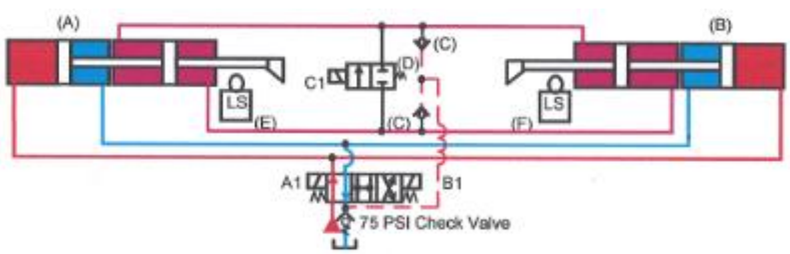


Figure 22-34. Tandem-cylinder synchronizing circuit.
Solenoid A1 energized, cylinders extending.

---

*Figure 22-35 shows the cylinders retracting. Energizing solenoid **B1** of the 4-way directional valve sends fluid to the rod ends of the single-rod cylinders. As the cylinders retract, the double-rod end cylinders cross piping keeps the machine in synchronization, the same as when they extend.*

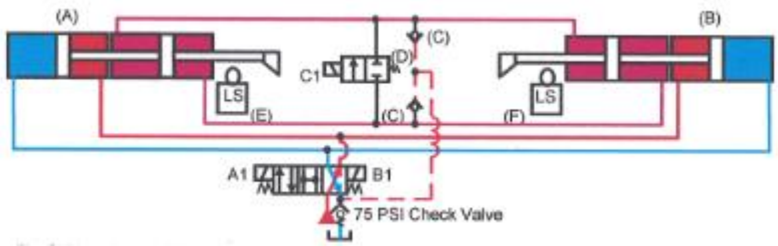


Figure 22-35. Tandem-cylinder synchronizing circuit.
Solenoid B1 energized, cylinders retracting.

---

*When the cylinders get close to home, they level or re-phase when necessary, as pictured in Figure 22-36. Limit switches **(E)** and **(F)** both have to make to center the 4-way directional valve. If one limit switch makes early, solenoid **C1** of 2-way directional valve **(D)**energizes, allowing the lagging cylinder to transfer oil until it makes its limit switch.*

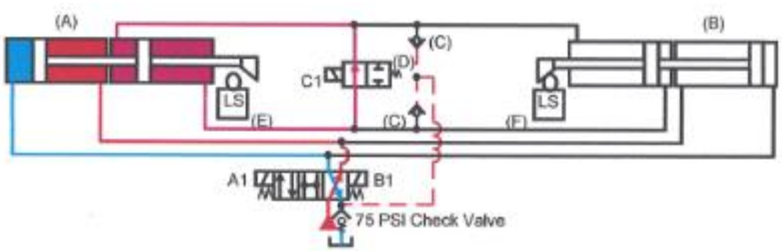


Figure 22-36. Tandem-cylinder synchronizing circuit.
Solenoids B1 and C1 energized, cylinder (A) leveling.

This synchronizing circuit works equally well with air as the power source to the single-rod end cylinders. Use oil in the tandem cylinders because it does not compress. Oversize the oil flow lines for a velocity of 2 to 4 fps to maintain a reasonable speed. Install a makeup oil tank with check valves to feed the tandem cylinders when necessary

High-efficiency circuit operating 100-ton trim press
Figure 23-1 is a schematic circuit diagram for a 100-ton trim press, shown at rest with the pump running. This press has a 24-in. total stroke with a 1.0-in. high-tonnage stroke. It completes a dry cycle in about 10 seconds.

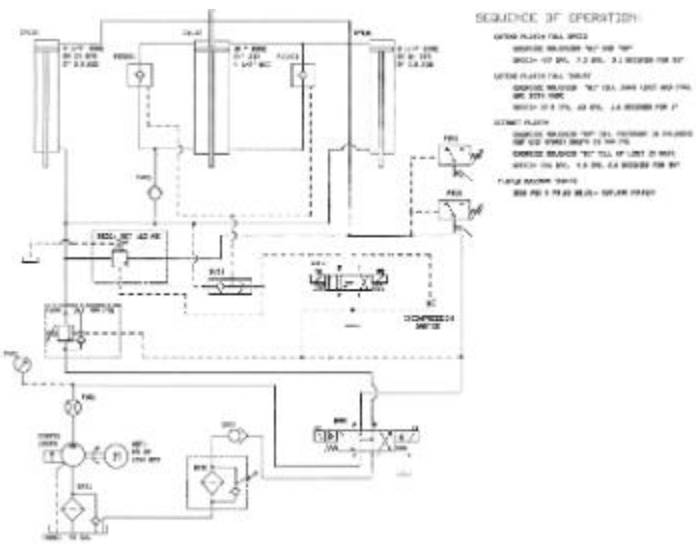


Fig. 23-1. Circuit for 100-ton trim press. Shown at rest with pump running.
Select figure to enlarge.

*Pump PUMP01 is rated at 13 gpm. Its compensator is set at approximately 3000 psi. Output from the pump flows through flow meter FM01 to directional valve DV01, which has ports "P," "B," and "T" connected in center position. From there, fluid flows to backpressure check valve CK02 and 3-**micron** filter RF01. Pressure at gauge PG01 holds between 75 and 100 psi with the press at rest, providing ample pilot pressure for directional valve DV01.*

Port "A" on directional valve DV01 connects to the cap ends of two 3.25-in. bore double-acting cylinders CYL01 and the extend side of double-rod end cylinder CYL02. The rod sides of these cylinders connect to port "B" on the directional valve.

Directional valve DV02 supplies pilot pressure to sequence valve SEQ01 and two pilot-operated check valves POCK01 for cylinder regeneration during fast advance. Directional valve DV02 also decompresses the cylinders before retracting.

Externally piloted sequence valve SEQ01 connects the cap and rod ports of the 3.25-in. bore cylinders in a regeneration circuit for a fast advance stroke.

Check valves POCK01 allow oil to flow from one end of the double-rod end cylinder to the other during fast advance and retract. Shuttle valve SV01 directs pilot pressure to the pilot-operated check valves from two sources.

Counterbalance valve CB01 holds the platen in position at rest, and keeps it from running away during the high-tonnage advance stroke. (Because it has an external pilot, set it one time for all platen and tooling weights.) External piloting also eliminates all cylinder rod-end backpressure during the high-tonnage portion of the cycle.

Check valve CK01 allows oil out of double-rod end cylinder CYL02 during the high-tonnage stroke, but prevents flow from entering it as the platen retracts.

Pressure switch PS01 ends the pressing cycle at a set tonnage. After the pressing cycle is completed, pressure switch PS02 keeps directional valve DV01 from shifting until the cylinders decompress.

Platen extending rapidly
Energizing solenoid A1 on directional valve DV01 and solenoid A2 on directional valve DV02 moves the cylinders to the positions shown in Figure 23-2.

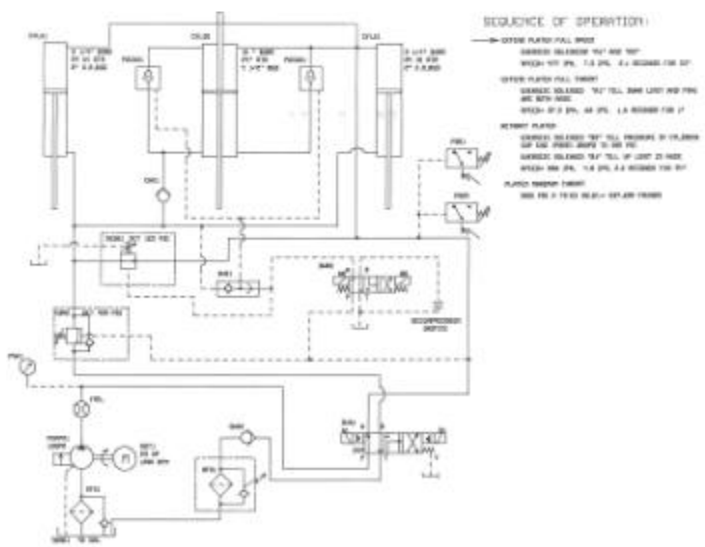


Fig. 23-2. Circuit for 100-ton trim press. Shown with platen moving fast forward.
Select figure to enlarge.

Pump flow goes to the cap ends of cylinders CYL01 and they begin to extend. At the same time, pilot pressure opens sequence valve SEQ01 and pilot-operated check valves POCK01. Oil now regenerates from the rod ends of the 3.25-in.-diameter cylinders through SEQ01 to their cap ends for high speed at low volume. Oil in the bottom of CYL02 transfers to the top, keeping it full, ready for high-tonnage pressing.

The pressure setting of counterbalance valve CB01 is high enough to make sure oil regenerates during this portion of the cycle, but low enough not to waste energy after regeneration is disabled. (About 400 psi works well on presses in this type of service.)

The platen extends at approximately 7.5 ips for about 3.1 sec. to produce 23 in. of travel.

Platen at full tonnage
A limit switch or encoder indicates that the platen is in close proximity to the part and the circuit shifts as shown in Figure 23-3.

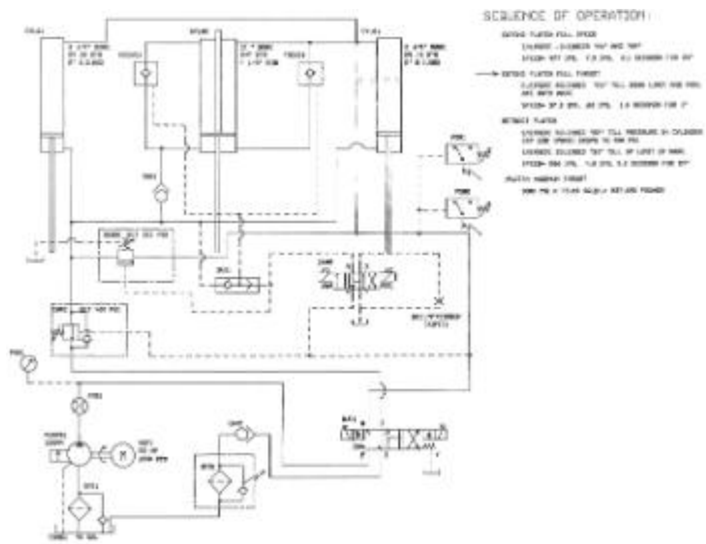


Fig. 23-3. Circuit for 100-ton trim press. Shown as platen develops full tonnage.
Select figure to enlarge.

Solenoid A1 on directional valve DV01 stays energized, but solenoid A2 on DV02 deenergizes.

Now sequence valve SEQ01 and pilot-operated check valves POCK01 close. The 3.25-in.-diameter cylinders no longer regenerate, so forward speed slows.

The double-rod end cylinder can still transfer fluid through the pilot-operated check valves so the platen does not slow to pressing speed until it contacts the work. Counterbalance valve CB01 opens during the slow-down part of the cycle, letting fluid go to tank, but not allowing the platen to run away.

At contact with the work, pilot-operated check valves POCK01 close, so pump flow goes to all three cylinders. Forward speed slows to approximately 0.6 ips, so it takes about 1.75 sec. to travel 1.0 in. Force during this portion of the stroke is more than 237,000 lb at 3000 psi.

Externally piloted counterbalance valve CB01 opens fully during the trimming part of the cycle to eliminate all backpressure on the cylinders.

During part trim, oil from the bottom of the double-rod end cylinder flows to tank through check valve CK01 and counterbalance valve CB01 along with fluid from the other two cylinders.

When pressure reaches the setting on pressure switch PS01, directional valve DV01 deenergizes to unload the pump and set up for decompression.

Cylinders decompressing

At the end of the high-pressure trim cycle, there may be a lot of stored energy in the cylinders and piping. (Also, if the tie rods on the press stretch, more energy is stored.) If directional valve DV01 shifts immediately to retract the cylinders, stored energy would send oil to tank with enough velocity and force to damage pipes and valves. In a short time, shock damage to plumbing and components could make the machine inefficient and messy.

To avoid this, after trimming the part, the circuit changes as shown in Figure 23-4. Directional valve DV01 centers to unload the pump and block oil in the cylinders' cap ends. Energizing solenoid B2 on directional valve DV02 opens a controlled flow path from the cap end of the cylinders to tank. Stored energy dissipates rapidly through this path to tank without decompression shock.

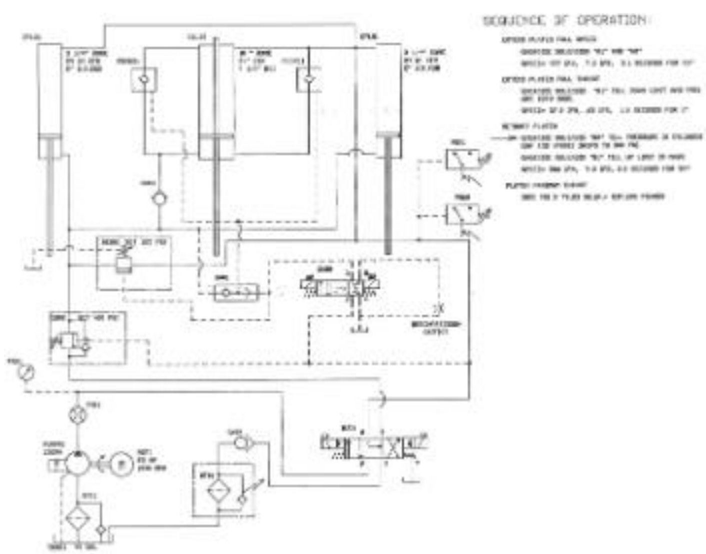


Fig. 23-4. Circuit for 100-ton trim press. Shown with cylinders decompressing.
Select figure to enlarge.

This adds time to the overall cycle, but greatly increases machine life and reliability. (Always figure in the added time of decompression when designing a circuit with stored energy.)

Decompression can take from 0.25 to 2.00 sec., depending on fluid volume and pressure. Offset the added cycle time with a faster cylinder stroke.

When pressure in the cap-end line of the cylinders reaches approximately 300 psi, pressure switch PS02 energizes solenoid B1 of directional valve DV01 to retract the platen.

Platen retracting at high speed
After the cylinders decompress, solenoid B1 on directional valve DV01 energizes as shown in Figure 23-5. Pump flow goes to the rod ends of 3.25-in.-diameter cylinders CYL01 through the bypass check valve on counterbalance valve CB01 to retract the platen.

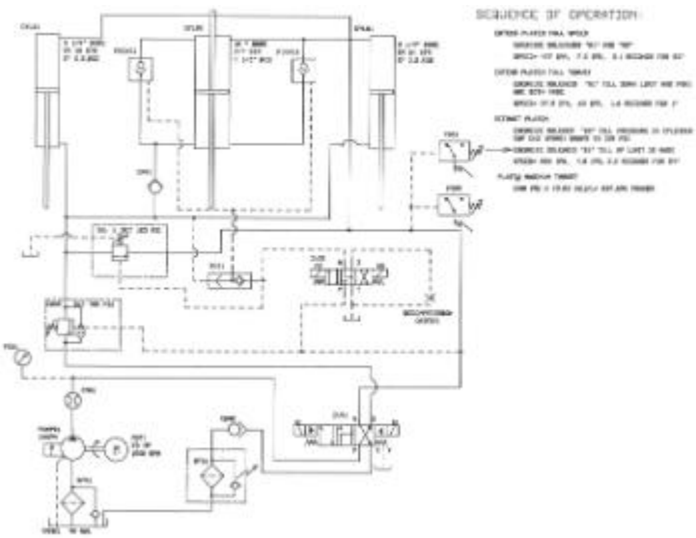


Fig. 23-5. Circuit for 100-ton trim press. Shown with platen retracting at high speed.
Select figure to enlarge.

Check valve CK01 blocks pump flow from the bottom of double-rod end cylinder CYL02 during the retract cycle.

Pilot oil from the cylinder rod end line opens pilot-operated check valves POCK01 through shuttle valve SV01. This action opens a path for oil transfer from top to bottom of the double-rod end cylinder.

The platen retracts at a rate of approximately 4.75 ips, and travels 24 in. in about 5 sec. Oil in the double-rod end cylinder transfers from top to bottom through pilot-operated check valves POCK01 to keep it full for the next cycle.

It appears the double-rod end cylinder keeps the same fluid in it all the time -- and might overheat. In actual operation, this cylinder gets some fresh oil from the pump and sends an equal *amount to tank during each cycle. If the high-force stroke of the cycle is 0.5 in., cylinder volume is replaced completely every 48 cycles.*

The reason for a circuit design such as shown here is to reduce pump displacement as well as valve and piping size while maintaining a fast cycle. A conventional cylinder regeneration and hi-lo pump circuit requires approximately 70 gpm to meet the cycle time of this circuit.

Another fast-cycle option is a prefill valve and a cylinder with a 9.0-in. oversize rod. This circuit could meet or exceed the above cycle time. A prefill circuit lets the platen lower from its own weight, so the extend part of the stroke could be faster.

This schematic diagram uses several of the basic circuits shown in other parts of this manual. Refer to the appropriate sections for a refresher on different valve functions and use.

High-efficiency circuit operating increment feeder for cardboard sheets
The circuit diagrammed in Figure 23-6 cycles a cylinder slowly in incremental steps as it extends, and then drops it very quickly to pick up another load. Feeding sheets of paperboard or plywood onto a conveyor is a common use for this circuit.

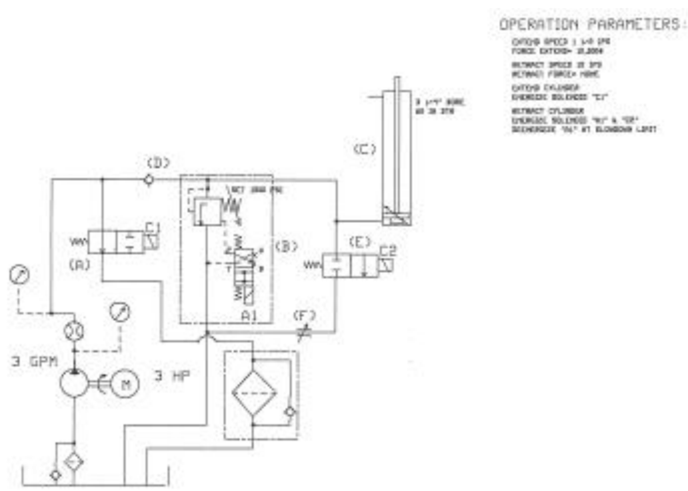


Fig. 23-6. Circuit operating increment feeder for cardboard sheets. Shown at rest with pump running. Select figure to enlarge.

A small (3-gpm), fixed-displacement pump sends oil through a flow meter, normally open 2-way directional valve (A), and a filter to tank. In the condition shown in the diagram, the pump unloads to tank at almost no pressure, continually filtering all flow.

The pump can be small because it only has to extend the cylinder a short distance each feed cycle. Because pump flow is immediately available, it is always ready to raise the load.

Check valve (D) keeps cylinder fluid from going to tank while the pump unloads. This same check valve allows free flow to the cylinder when required.

Normally closed solenoid-operated relief valve (S) protects the system from excess pressure and gives a high-flow path to tank to lower the table for another load. It also decelerates the cylinder as it nears bottom and eliminates shock damage to the hydraulic components and the machine.

Normally closed 2-way directional valve (E) and needle valve (F) give a slow-down bypass to the normally closed solenoid-operated relief valve to make sure the cylinder bottoms out smoothly.

Cylinder extending
Energizing solenoid C1 on directional valve (A) sends pump flow to cylinder (C), making it extend as diagrammed in Figure 23-7. This action takes place as fast as directional valve (A) shifts because the pump is at full flow. Cylinder (A) extends at 1.50 ips at a maximum force of 10,000 lb, as long as solenoid C1 stays energized.

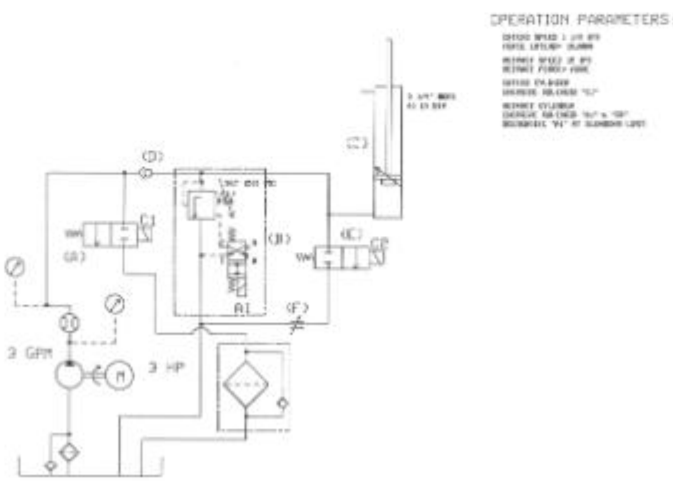


Fig. 23-7. Circuit operating increment feeder for cardboard sheets. Shown with cylinder extending. Select figure to enlarge.

If the cylinder meets a resistance that calls for more than 1500 psi, pump flow goes to tank through normally closed solenoid-operated relief valve (B) and the cylinder stalls.

Solenoid C1 shifts as often as necessary to raise the load.

When the table gets all the way up, a limit switch signals the control circuit to lower for another load.

Cylinder retracting at high speed
To lower the table fast, the valves shift as shown in Figure 23-8.

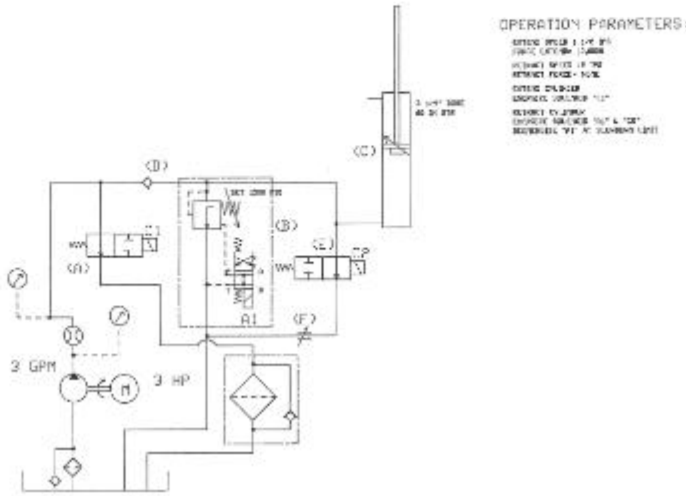


Fig. 23-8. Circuit operating increment feeder for cardboard sheets. Shown with cylinder retracting at high speed.
Select figure to enlarge.

Solenoid A1 on normally closed solenoid-operated relief valve (S) energizes to vent it. This allows the valve to open with about 20-psi backpressure. The 0.75-in. valve in this circuit passes about 30 gpm at 20- to 30-psi pressure drop, so the cylinder retracts rapidly from machine weight. On this particular machine, the weight gave it a speed of approximately 12 ips or about a 5-sec lowering time. Increasing load-induced pressure on the cylinder would make it even faster.

The solenoid on normally closed 2-way valve (E) also energizes during fast retract of the cylinder. It does not add much speed because its function is to let the cylinder descend slowly through the last 0.5 to 1.5 in. of travel. (Set the slow-down limit switch high enough to decelerate the cylinder before it hits bottom.) Fluid viscosity or weight changes give more or less flow to increase or decrease the slow-down distance.

Retracting cylinder slows down
The circuit in Figure 23-9 shows the valve positions as the retracting cylinder nears the end of stroke and makes the slow-down limit switch.

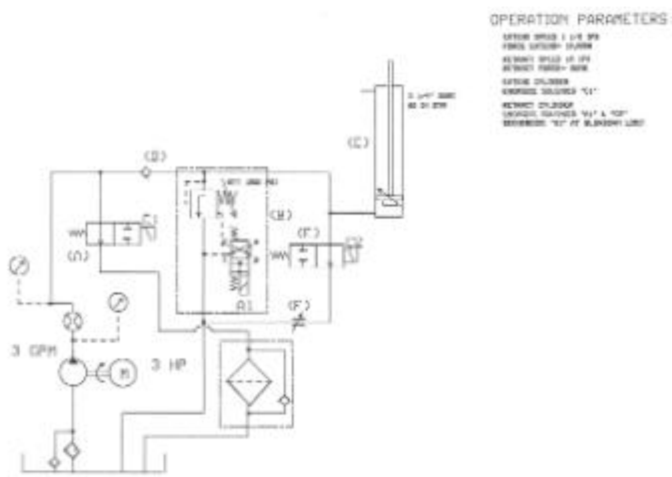


Fig. 23-9. Circuit operating increment feeder for cardboard sheets. Shown as retracting cylinder slows down.
Select figure to enlarge.

All solenoids are deenergized except C2 on normally closed 2-way (E). When solenoid A1 on normally closed solenoid-operated relief valve (S) deenergizes, the valve starts to close. When this happens, pressure from the load and high -flowing fluid builds to relief setting. The relief valve resists flow but does not close completely all at once (like a directional valve would). Backpressure at the cylinder causes it to decelerate smoothly and quickly with little or no shock.

With relief pressure set low, deceleration takes longer. With relief pressure set high, deceleration is quicker. In any event, cylinder slowing does not consistently stay the same.

Always position the slow-down limit far enough from end of stroke to decelerate the cylinder before it reaches bottom. This means it could stop before bottoming out when the only control is normally closed solenoid-operated relief valve (S).

Normally closed 2-way valve (E) provides another controllable flow path around the normally closed solenoid-operated relief valve. Set needle valve (F) to let the cylinder move quickly, but not fast enough to shock the machine when it reaches bottom. Use a cylinder cap-end cushion as a final shock absorber to shorten slow-down travel.

The slow-down part of the stroke is usually less than 2 in., so it poses no time problem.

After the cylinder bottoms out, solenoid (C2) on normally closed directional valve (E) deenergizes, ending the cycle.

Opposing trim cylinders with synchronous movement
Two cutter heads, driven by opposing cylinders, must meet near perfectly in center regardless of

differing loads. If the cylinders are out of phase, parts can be damaged and the profiled cutters may break.

An original circuit used two pumps and directional valves to synchronize the cylinders. The machine was useable but parts often were under par and cutting tools had to be changed frequently.

The schematic diagram in Figure 23-10 shows the new circuit now on the machine. Now the cutters move almost perfectly in unison, and produce a better part with no tooling damage.

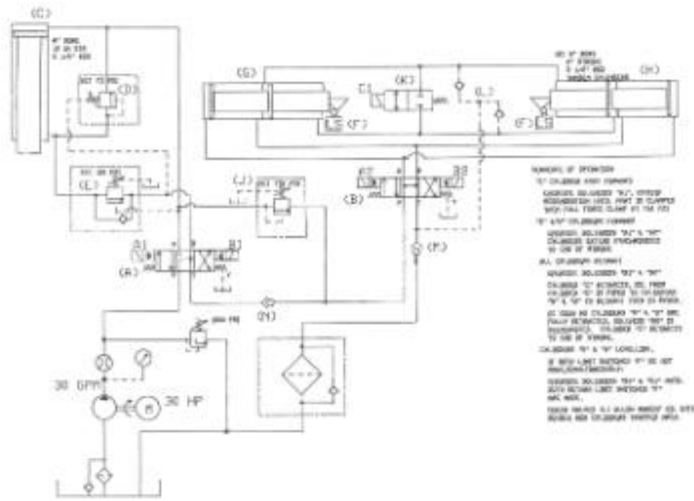


Fig. 23-10. Circuit operating opposing trim cylinders with synchronous movement. Shown at rest with pump running.
Select figure to enlarge.

A single 30-gpm pump replaced two 21-gpm pumps with a cycle time decrease of 1.5 seconds.

Clamp cylinder (C) has an oversize rod and uses a regeneration circuit for fast, low-volume extension. Counterbalance valve (E) keeps the vertical clamp cylinder from running away. Sequence valve (J) assures that the part stays tightly clamped as trim cylinders (G) and (H) advance to the work.

Both directional valves (A) and (B) have pump-to-tank center conditions, so the pump unloads until they shift.

Check valve (N) prevents backflow to the clamp cylinder's rod end while the trim cylinders work.

Check valve (M) provides backpressure to supply pilot oil to both solenoid pilot-operated directional valves and makeup fluid to the tandem cylinders.

Two check valves (L) allow fluid into the double-rod end cylinder's trapped area to make up for leakage. Pressure here also keeps the seals energized to reduce leaks.

Normally closed 2-way directional valve (K) re-synchronizes the trim cylinders if they get out of phase. It operates automatically any time limit switches (F) do not make simultaneously.

Clamp cylinder (C) extending in regeneration
Energizing solenoid A1 on directional valve (A) starts clamp cylinder (C) forward, as seen in Figure 23-11. Oil from the rod end of cylinder (C) regenerates to the cap end and almost doubles its speed. (Because the cylinder needs minimum force to move to the part, a regeneration circuit such as this works well and uses a smaller pump.)

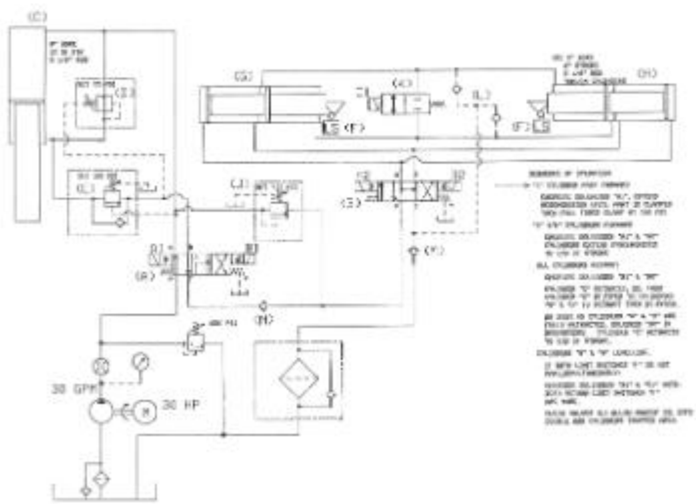


Fig. 23-11. Circuit operating opposing trim cylinders with synchronous movement. Shown with clamp cylinder (C) extending in regeneration.
Select figure to enlarge.

Set counterbalance valve (E) high enough to force rod end oil to regenerate through sequence valve (0) until the cylinder meets the work.

Clamp cylinder (C) advances at fast speed until it contacts the work. At work contact, pressure increases and opens counterbalance valve (E). This action drops backpressure on the rod end of cylinder (C), giving full force to clamp the part. When valve (E) opens, sequence valve (O) closes, shutting off the regeneration path.

Pressure must climb to 750 psi before oil can pass sequence valve (J) and flow on to trim cylinder valve (B). Sequence valve (J) makes sure the part has ample clamp force while the trim cylinders advance to the work. Pressure in clamp cylinder (C) never drops below 750 psi if sequence valve (J) is set correctly.

Cylinder (C) extended at full force, cylinders (G) and (H) extending in synchronization
After clamping the part, energize solenoid A2 on directional valve (B) as shown in Figure 23-12. Sequence valve (J) maintains 750 psi at the clamp cylinder but allows pump flow to go to the trim cylinders.

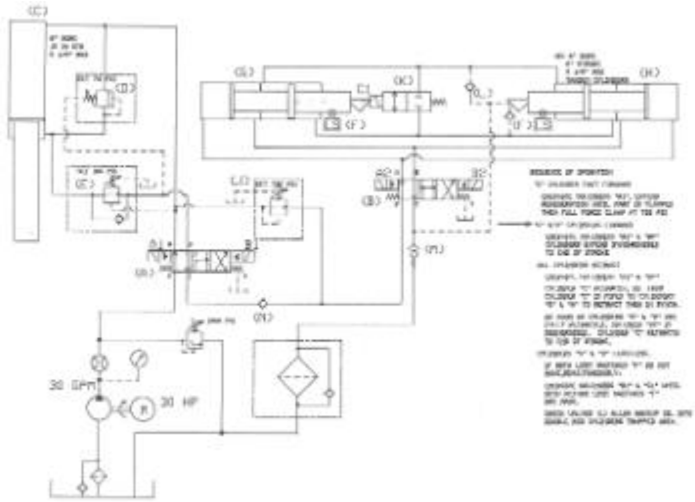


Fig. 23-12. Circuit operating opposing trim cylinders with synchronous movement. Shown with clamp cylinder (C) extended at full force, trim cylinders (G) and (H) extending in synchronization.
Select figure to enlarge.

Trim cylinders (G) and (H) now advance in synchronization. They stroke together regardless of load difference until both cylinders stall at relief pressure. The reason trim cylinders (G) and (H) stay together is that oil from each of the double-rod end cylinders transfers to its opposing cylinder's opposite end. If one cylinder moves an inch, it is only because the opposing cylinder also moved that distance. If one cylinder needs to develop more force, energy from the opposing cylinder transfers to it. Neither cylinder stalls until resistance against it is greater than both cylinders can overcome.

In case of leakage from the double-rod end cylinders, backpressure check valve (M) forces pump fluid into the trapped area through check valves (L).

Except for piston or rod seal bypass, the trim cylinders always stay synchronized. In case they do get out of phase, normally closed 2-way directional valve (K) allows them to re-synchronize.

All cylinders retracting
After trimming the part, the valves shift to the positions shown in the schematic diagram in Figure 23-13. Energizing solenoids B1 and B2 on directional valves (A) and (B) directs pump flow to the rod end of cylinder (C). Pump flow goes around the bypass check in counterbalance

valve (E) to the cylinder's rod end. Sequence valve (O) is held closed by its light spring and pressure from retracting cylinder (C).

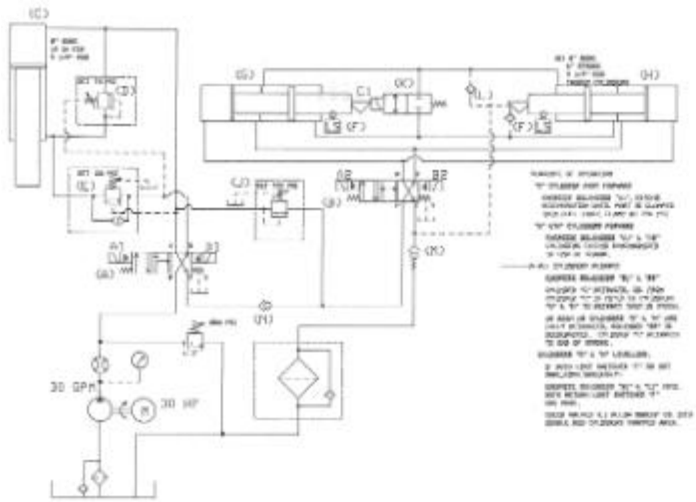


Fig. 23-13. Circuit operating opposing trim cylinders with synchronous movement. Shown with all cylinders retracting after trimming the part.
Select figure to enlarge.

Oil from the cap end of cylinder (C) goes through directional valve (A), check valve (N), and directional valve (B) to the rod ends of cylinders (G) and (H). The pump only has to retract cylinder (C). Return flow from cylinder (C) retracts the trim cylinders. This is possible because cylinder (C) has more than enough oil to fill the rod ends of the trim cylinders at the low force required to retract them.

All three cylinders continue retracting quickly because cylinder (C) has an oversize rod. When trim cylinders (G) and (H) bottom out, solenoid B2 on directional valve (B) deenergizes so clamp cylinder (C) can complete its retraction stroke.

When cylinder (C) bottoms out, the limit switch indicates the end of the cycle and deenergizes solenoid B1 on directional valve (A). Now the pump unloads because the circuit is back to the "At rest with pump running" condition.

Cylinders (G) and (H) stay synchronized because they must both make their limit switches (F) before directional valve (B) centers.

Cylinders (G) and (H) re-synchronizing
During each cycle, trim cylinders (G) and (H) return to make limit switches (F). If one limit switch makes first, that indicates the cylinders are out of synchronization. The schematic diagram in Figure 23-14 shows how the cylinders re-synchronize if this happens.

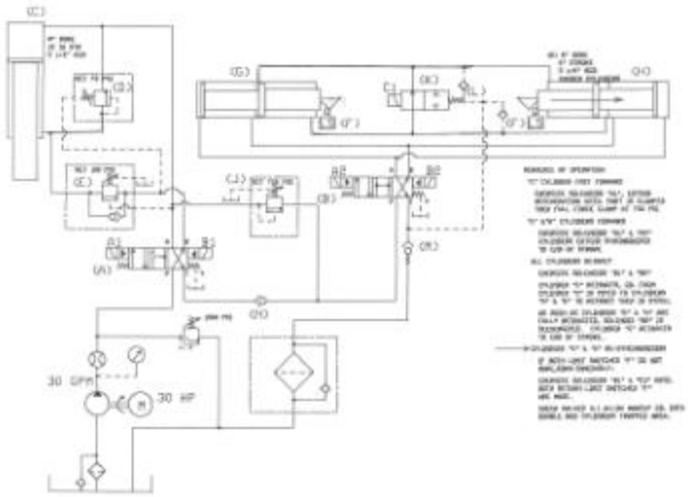


Fig. 23-14. Circuit operating opposing trim cylinders with synchronous movement. Shown with trim cylinders (G) and (H) re-synchronizing. Select figure to enlarge.

In the diagram, trim cylinder (H) is late making its limit switch. When this condition occurs, the control circuit automatically energizes solenoid C1 on normally closed directional valve (K). Because all cylinders are in the retract mode, pressure on the rod end of cylinder (H) keeps it retracting. It can move by itself because open directional valve (K) provides a flow path from back to front of the double-rod end cylinders during this part of the cycle.

When both limit switches (F) make, solenoid C1 on directional valve (K) deenergizes. Solenoid B2 on directional valve (B) deenergizes, letting it center, and allowing clamp cylinder (C) to keep retracting.

Because re-synchronizing is automatic, there is never a build up of stroke error, which could allow damage to cutters or parts.
