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UNIT I FLUID POWER SYSTEMS AND FUNDAMENTALS

UNIT II HYDRAULIC SYSTEM & COMPONENTS

UNIT III DESIGN OF HYDRAULIC CIRCUITS

UNIT IV PNEUMATIC SYSTEMS AND COMPONENTS

UNIT V DESIGN OF PNEUMATIC CIRCUITS

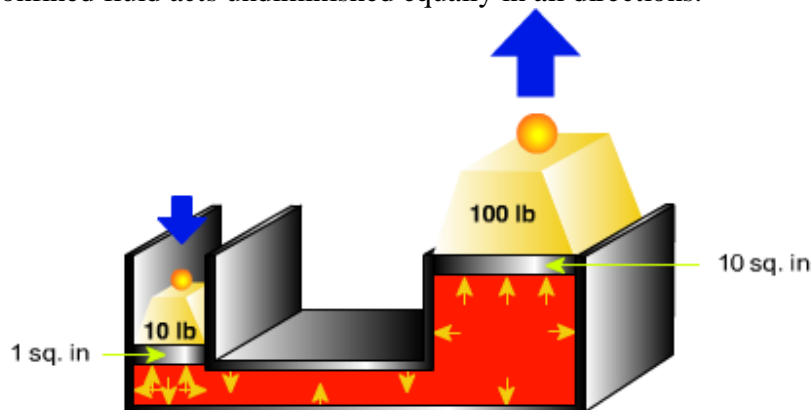
UNIT I FLUID POWER SYSTEMS AND FUNDAMENTALS

What is fluid power?

Fluid power is energy transmitted and controlled by means of a pressurized fluid, either liquid or gas. The term fluid power applies to both hydraulics and pneumatics. Hydraulics uses pressurized liquid, for example, oil or water; pneumatics uses compressed air or other neutral gases. Fluid power can be effectively combined with other technologies through the use of sensors, transducers and microprocessors.

How fluid power works

Pascal's Law expresses the central concept of fluid power: "Pressure exerted by a confined fluid acts undiminished equally in all directions."



An input force of 10 pounds (44.8 N) on a 1-square-inch (6.45 cm^2) piston develops a pressure of 10 pounds per square inch (psi) (68.95 kN/m^2 or 68.95 KPa) throughout the container. This pressure will allow a 10-square-inch piston to support a 100-pound (444.8 N) weight. The forces are proportional to the piston areas.

The advantages of fluid power

Fluid power systems provide many benefits to users including:

- **Multiplication and variation of force**-Linear or rotary force can be multiplied from a fraction of an ounce to several hundred tons of output.
- **Easy, accurate control**-You can start, stop, accelerate, decelerate, reverse or position large forces with great accuracy. Analog (infinitely variable) and digital (on/off) control are possible. Instantly reversible motion-within less than half a revolution-can be achieved.
- **Multi-function control**-A single hydraulic pump or air compressor can provide power and control for numerous machines or machine functions when combined with fluid power manifolds and valves.

- **High horsepower, low weight ratio**-Pneumatic components are compact and lightweight. You can hold a five horsepower hydraulic motor in the palm of your hand.
- **Low speed torque**-Unlike electric motors, air or hydraulic motors can produce large amounts of torque (twisting force) while operating at low speeds. Some hydraulic and air motors can even maintain torque at zero speed without overheating.
- **Constant force or torque**-This is a unique fluid power attribute.
- **Safety in hazardous environments**-Fluid power can be used in mines, chemical plants, near explosives and in paint applications because it is inherently spark-free and can tolerate high temperatures.
- **Established standards and engineering**-The fluid power industry has established design and performance standards for hydraulic and pneumatic products through NFPA, the National Fluid Power Association and ISO, the International Organization for Standardization.

Fluid power applications

- **Mobile:** Here fluid power is used to transport, excavate and lift materials as well as control or power mobile equipment. End use industries include construction, agriculture, marine and the military. Applications include backhoes, graders, tractors, truck brakes and suspensions, spreaders and highway maintenance vehicles.
- **Industrial:** Here fluid power is used to provide power transmission and motion control for the machines of industry. End use industries range from plastics working to paper production. Applications include metalworking equipment, controllers, automated manipulators, material handling and assembly equipment.
- **Aerospace:** Fluid power is used for both commercial and military aircraft, spacecraft and related support equipment. Applications include landing gear, brakes, flight controls, motor controls and cargo loading equipment.

Fluid power products




Fluid power products are sold as individual components or as systems for the original equipment manufacturing, maintenance, repair and replacement markets.

A typical fluid power system includes the following components:




- Hydraulic pump or air compressor, which converts mechanical power to fluid power.
- Cylinder or motor, which converts fluid power to linear or rotary mechanical power.
- Valves, which control the direction, pressure and rate of flow.
- Filters, regulators and lubricators, which condition the fluid.
- Manifolds, hose, tube, fittings, couplings, etc., which conduct the fluid between components.
- Sealing devices, which help contain the fluid.
- Accumulators and reservoirs, which store the fluid.
- Instruments such as pressure switches, gauges, flow meters, sensors and transducers, which are used to help monitor the performance of a fluid power system.

FLUID POWER SYMBOLS.


Lines

	-continuous line - flow line
	-dashed line - pilot, drain
	-envelope - long and short dashes around two or more component symbols.

Circular

	-large circle - pump, motor
	-small circle - Measuring devices
	-semi-circle - rotary actuator



Square

	-one square - pressure control function
	-two or three adjacent squares - directional control

Diamond

	-diamond - Fluid conditioner (filter, separator, lubricator, heat exchanger)
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Miscellaneous Symbols

	-Spring
	-Flow Restriction

Triangle

	-solid - Direction of Hydraulic Fluid Flow
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-open - Direction of Pneumatic flow

BASICS OF HYDRAULICS

"What is hydraulics ?"

Hydraulics is the transmission and control of forces and motions through the medium of fluids.

Short and simple.

Hydraulic systems and equipment have wide-spread application throughout industry.

For example:

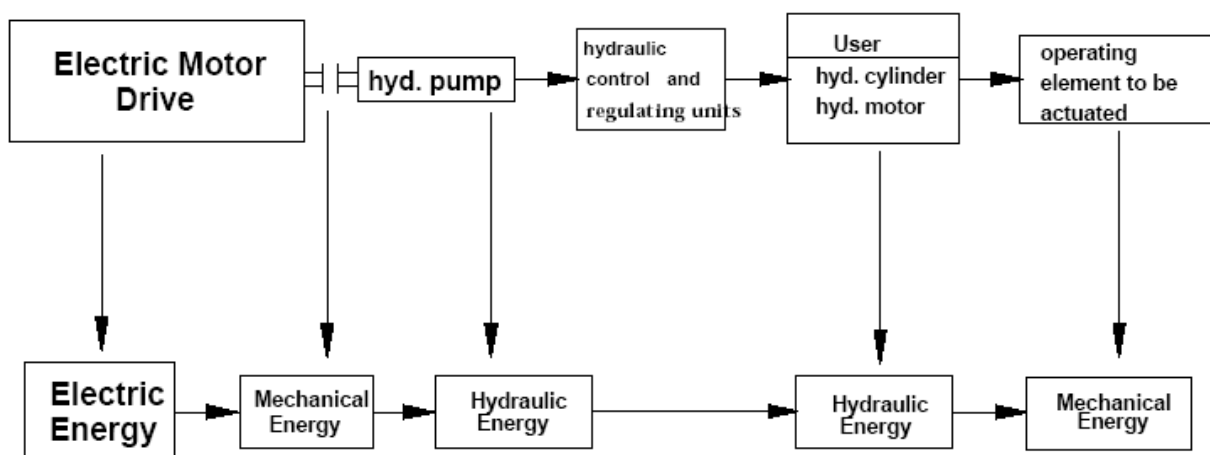
- machine tool manufacturing
- press manufacturing
- plant construction
- vehicle manufacturing
- aircraft manufacturing
- shipbuilding
- injection molding machines

Hydraulic to Electrical Analogy

Hydraulics and electrics are analogous, because they both deal with flow, pressure and load. The components in each type of circuit perform similar functions and therefore can be related, a few examples are listed below:

H Hydraulic Pumps	↔	C Compressor
H Hydraulic Motor	↔	H Electric Motor
H Directional Control Valve	↔	S Switches
H Hoses	↔	W Wires
C Cylinders	↔	S Solenoid
C Check Valves	↔	D Diodes
H Relief Valves	↔	C Circuit Breaker
A Accumulator	↔	C Capacitor
H Reservoir	↔	T Transducer

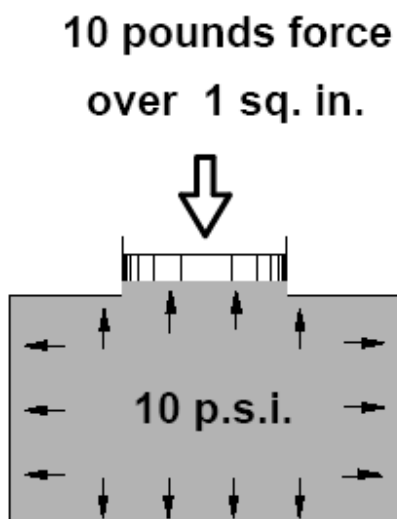
Conversion of Energy in Hydraulics



Various forms of energy are converted to accomplish mechanical movement in the injection molding machine. Electrical energy is converted to mechanical energy, which in turn is converted to hydraulic energy to operate and control the moving components of the machine. The hydraulic energy is converted to mechanical energy to achieve the final desired result, which may be "mold clamping pressure" or "material injection". The figure above summarizes the energy conversions for an injection molding machine. Click on the thumbnail for a larger view.

Pascal's Law

Pascal's Law states that a pressure acting on a confined fluid is transmitted equally and undiminished in all directions. In the figure below, a 10 pound force acting on a 1 square inch area generates a pressure of 10 pounds per square inch (psi) throughout the container acting equally on all surfaces.



This principle is important to remember, that the pressure in any portion of an hydraulic system is equal throughout that system. This statement is valid with the omission of the force of gravity, which would have to be added, according to the fluid level. Due to the pressures that hydraulic systems operate at, this smaller amount need not be considered e.g. a 32 foot head of water approximately equals 14.5 psi. (a 10 meter head of water approximately equals 1 bar.)

Force Transmission in Hydraulics

One of the main advantages on the use of hydraulic to power our molding machinery is the efficient transmission of forces.

If force F_1 is exerted on surface A_1 , pressure p is created. ($p = \frac{F_1}{A_1}$).

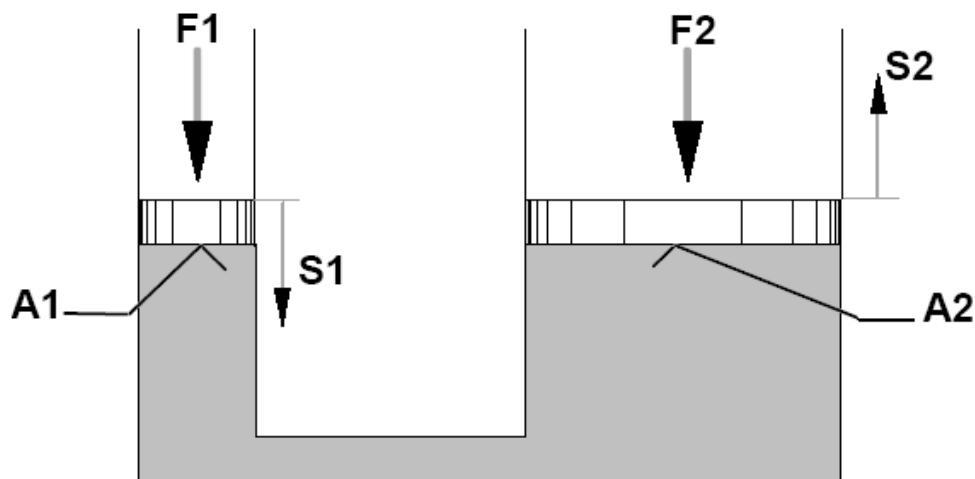
Since pressure affects all parts of the system equally, therefore pressure p is applied to surface area A_2 .

Force F_2 will equal pressure p x surface area A_2 ($F_2 = p \times A_2$), and transposing that formula for $p = \frac{F_2}{A_2}$.

Since $p = \frac{F_1}{A_1}$ therefore $\frac{F_2}{A_2} = \frac{F_1}{A_1}$.

In the diagram below, the following relationships hold: $\frac{S_1}{S_2} = \frac{A_2}{A_1} = \frac{F_2}{F_1}$

Where
 S = piston stroke
 A = piston area
 F = force



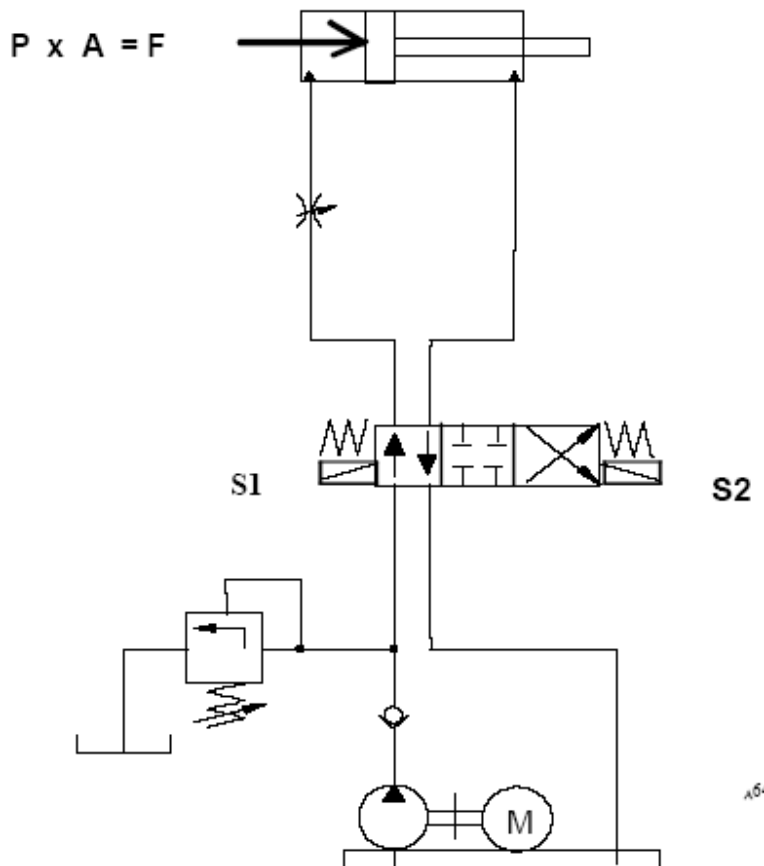
If $A_1 = 1$ square inch and $A_2 = 10$ square inches, then a force of $F_1 = 10$ pounds can support a force of $F_2 = 100$ pounds.

However, the strokes of the pistons are inversely proportional to their surface areas .

If the smaller piston were moved in the direction of S_1 by 10 inches, then the larger piston will only move 1 inch in the direction of S_2 .

Area and Force

As the clamp piston is moved forward during the clamp close function, the pressure developed acts upon the clamping piston which has a certain size or area.



A basic formula in hydraulics states that pressure multiplied by area to which that pressure is applied equals force. i.e. pressure \times area = force

$$p \times A = F$$

The formula can be manipulated to calculate any one of the three variables p , A or F , if any of the other two variables are known.

As follows:

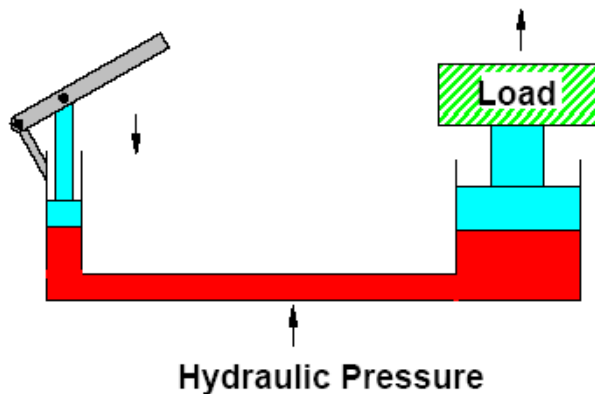
$$p \times A = F$$

$$F / p = A$$

$$F / A = p$$

Pressure

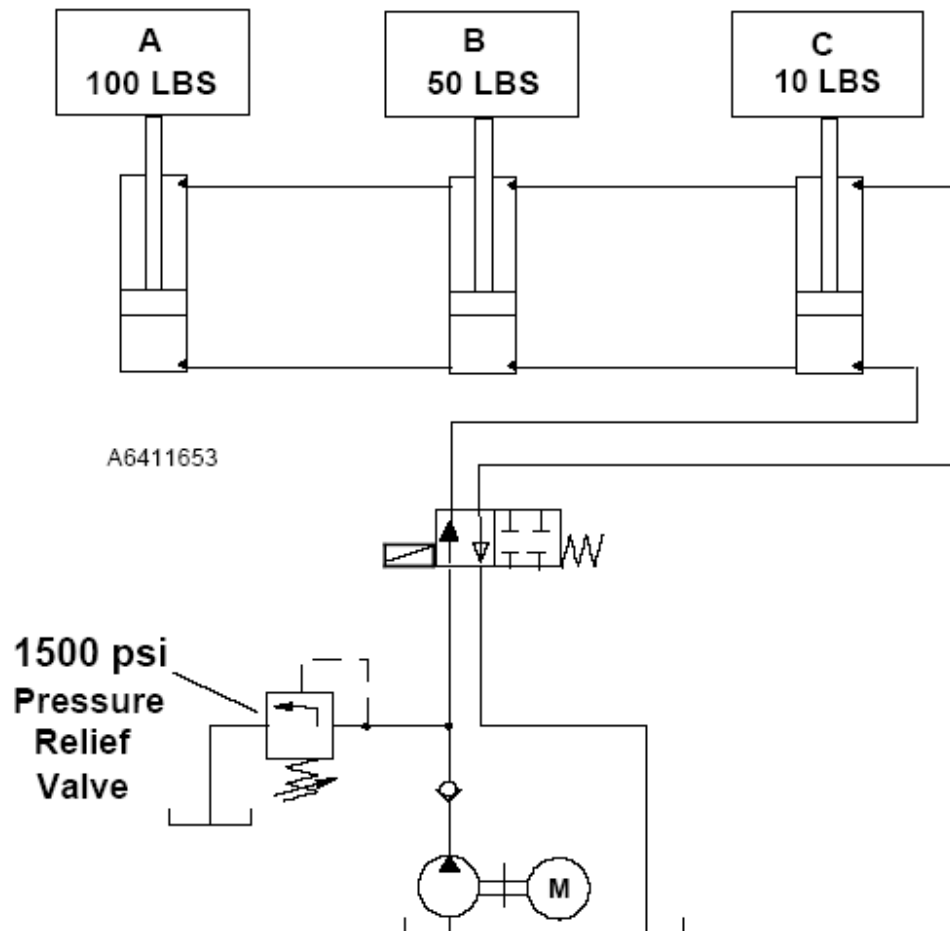
Hydraulic pressure is generated when a flowing fluid meets resistance which is generally related to the load that is being moved.



A force is applied via the lever to produce system pressure ($p = F/A$ or $F = p \times A$). If more force is applied, the system pressure rises until the load moves, if the load remains constant the pressure will increase no further. The load can therefore be moved if the necessary pressure is generated. The speed at which the load moves will be dependent upon the volume of fluid which is fed to the load cylinder. For example, as the mold is opening or closing, the pressure generated in the system represents the resistance of the toggle lever to movement. Adding to that resistance would be the weight (i.e. mass) of the mold and toggle lever and also the friction between the toggle lever bushings and the tiebars. When the two mold halves touch and the toggle begins to straighten out, the increasing pressure represents that which is required to stretch the tiebars in the generation of a particular clamp force. Similarly when injecting material into the mold the pressure generated in the injection system represents the resistance of the injection ram to movement. Adding to that resistance would be the mass of the injection ram and screw, the friction between all moving components and the resistance of the plastic melt as it is forced quickly into the mold cavity.

Pressure Control

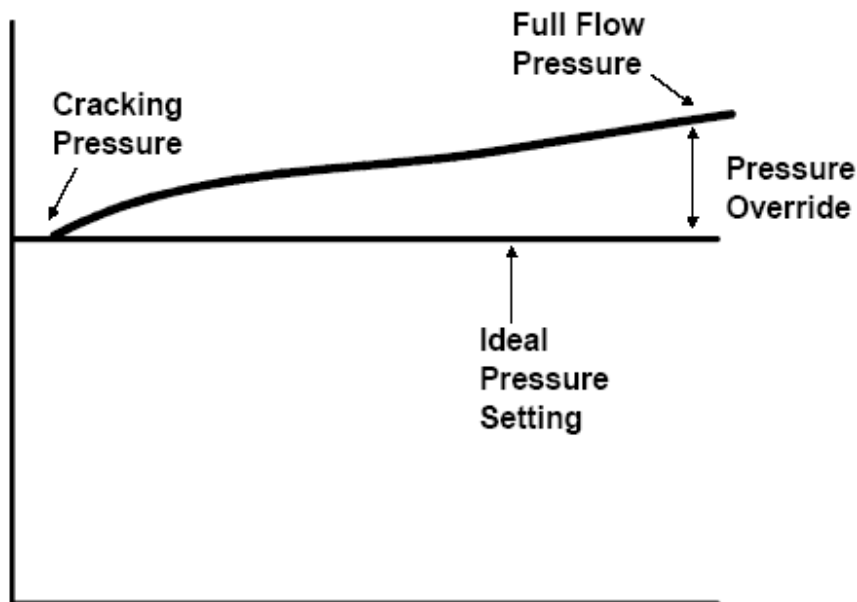
In order to safeguard the system, pressure relief valves are installed. The valves serve to limit the amount of pressure that can develop in the hydraulic system since the various hydraulic components are expensive and they are subject to pressure limitations before failure occurs. One characteristic of fluid flow that is important to note here is that flow occurs always in the path of least resistance. Pressure would continue to rise in the circuit consistent with the load being moved. The pressure relief valve is always set to allow flow to travel through the relief valve well before pressure rises above safe levels and causes damage to the system and its components. In other words, the path of least resistance is employed here to safeguard the system after the other movements have taken place.



- Load "C" would move first, because it is the lightest.
- Load "B" would move next, because it is the second lightest.
- Load "A" would move last, because it is the heaviest.

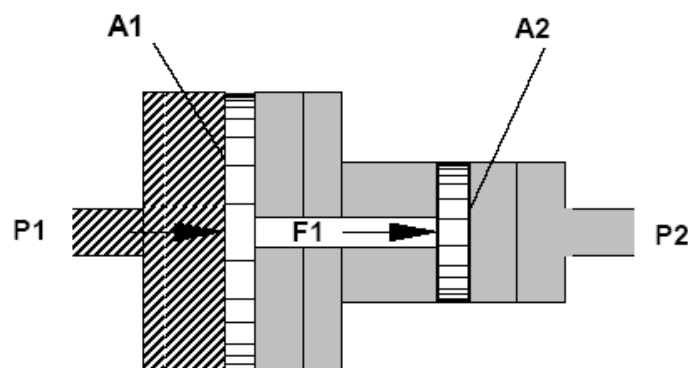
Pressure Override

An extremely important concept to understand about pressure relief valves is their pressure override characteristics. Pressure override is the difference between the pressure at which the relief valve just starts to crack open and the pressure at the full open position. For direct acting pressure relief valves this pressure differential can be as high as 30% and proportional pressure relief valves range from 10% - 20%.



Pressure Intensification

Another important concept to keep in mind is that of pressure intensification. This law of hydraulics is often forgotten when troubleshooting hydraulic circuits.



For example, if two pistons of different size are connected by a rod, the pressure existing on the smaller area will always be greater. This principle also applies to the cap side and the rod side of a normal double acting piston.

If $P_1 = 1,000$ psi and $A_1 = 10$ square inches, then $F_1 = 10,000$ pounds of force.

If $F_1 = 10,000$ pounds of force and if $A_2 = 5$ square inches, then $P_2 = 2,000$ psi.

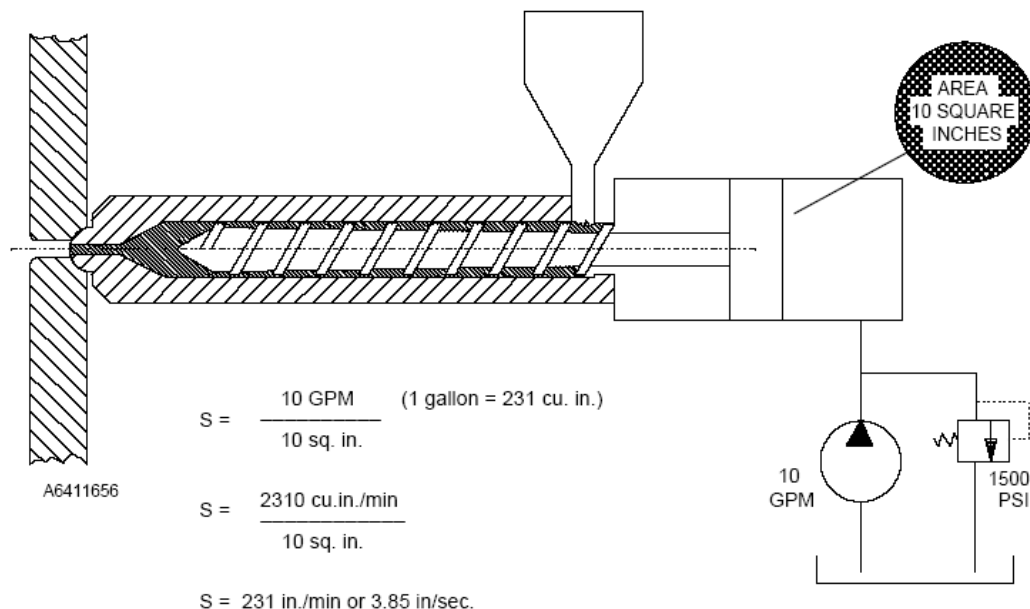
Speed in Hydraulics

The speed of a hydraulic component can be calculated based on the formula below:

$$S = \frac{V/t}{A}$$

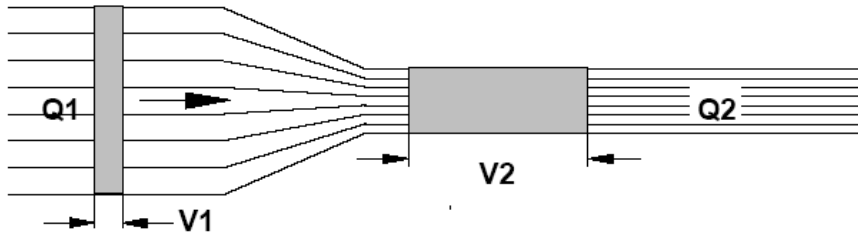
S = piston speed
 V/t = volume of oil flow time (i.e., gpm)
 A = piston area

For example, given the conditions below the injection piston, therefore the screw, will move at 3.85 inches per second. However, this speed will not be possible if the pressure relief valve opens.



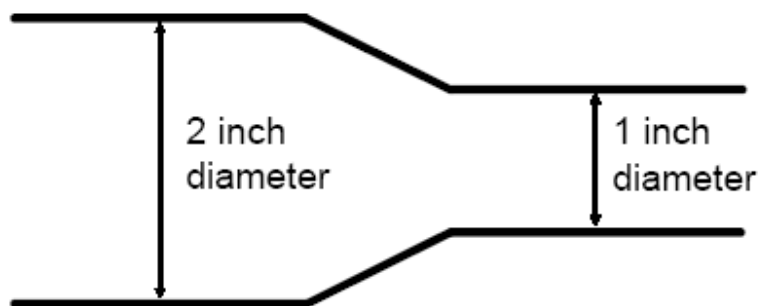
Hydrodynamics

As well as understanding the concept of speed in hydraulics, it is also important to have some insight into flow characteristics. For example, the drawing below shows that when oil is flowing through different diameter pipes an equal volume flows in an equal unit of time. If that is true and if the shaded quantity Q1 equals the shaded quantity Q2, then velocity V2 must be greater than velocity V1.



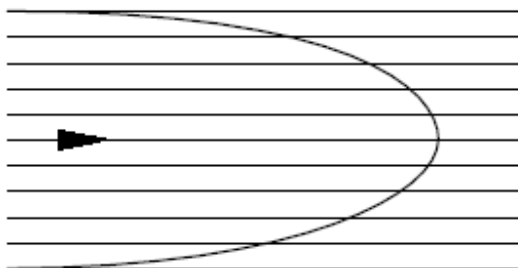
As the diameter of the pipe decreases, the flow rate will increase. Specifically, if the pipe diameter decreases by one half in the direction of oil flow, the cross sectional area will decrease by four times, and visa versa. Oil flow velocity through different pipe sizes can be calculated using the formula:

$$\text{Velocity (feet per second)} = \text{GPM} / 3.117 \times \text{area}$$



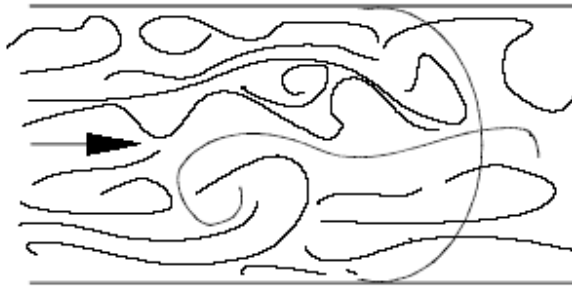
The same gallons per minute will have to travel 4 times faster through the smaller pipe.

Another important concept in hydrodynamics is how fluids flow based on certain critical flow speeds or as the result of meeting restrictions to flow such as bends in the pipe or system components.



LAMINAR FLOW

One goal in the initial design of hydraulic power transmission systems is to encourage laminar flow as much as possible since an increase in turbulence will increase flow resistance and hydraulic losses as well. The diagram below illustrates the concept of turbulent flow.

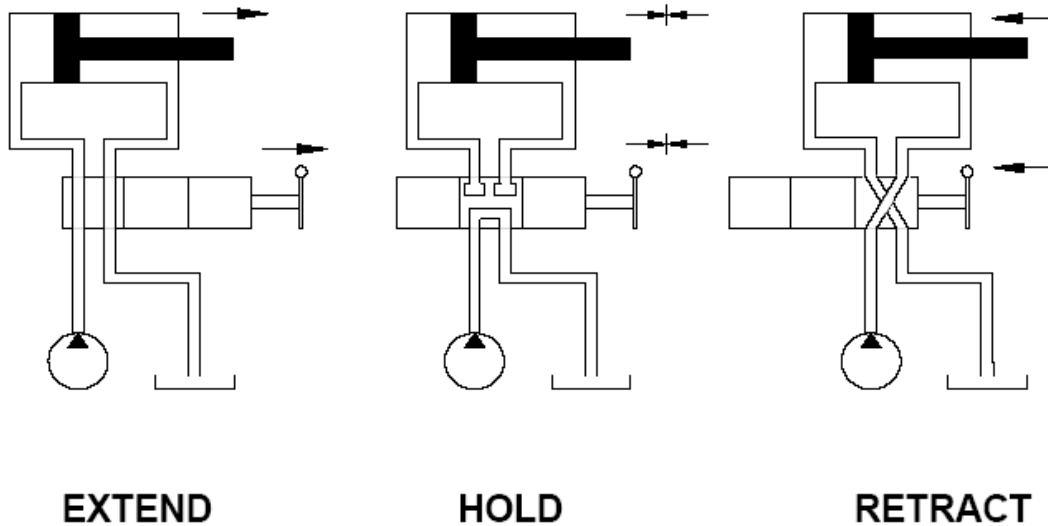


TURBULENT FLOW

Although turbulent flow is wasteful in most hydraulic applications, it is desirable to have turbulence in the oil flow as it travels through the heat exchanger for cooling purposes. If turbulence exists as the oil flows through the heat exchanger, more of the oil molecules come into contact with the heat exchanger cooling tubes and more efficient cooling is the result.

Directional Control

One of the main advantages of hydraulic based systems is that the oil flow direction is easily controlled. The drawing below shows a piston being extended, held stationary and then retracted, simply by changing the position of a directional valve. Even though the drawing is simple in nature, it still demonstrates the principle involved in directional control. In addition to simple directional control valves, we also employ proportional directional control valves on some machines to control the clamp opening and closing function.



Reynold's number

The Reynolds Number is a non dimensional parameter defined by the ratio of dynamic pressure (ρu^2) and shearing stress ($\mu u / L$) and can be expressed as

$$\text{Re} = (\rho u^2) / (\mu u / L)$$

$$= \rho u L / \mu$$

$$= u L / \nu \quad (1)$$

where

Re = Reynolds Number (non-dimensional)

ρ = density (kg/m^3 , lb_m/ft^3)

u = velocity (m/s, ft/s)

μ = dynamic viscosity (Ns/m^2 , $\text{lb}_m/\text{s ft}$)

L = characteristic length (m, ft)

ν = kinematic viscosity (m^2/s , ft^2/s)

Darcy's Equation

The frictional head loss can be calculated using a mathematical relationship that is known as Darcy's equation for head loss. The equation takes two distinct forms. The first form

of Darcy's equation determines the losses in the system associated with the length of the pipe.
(3-14)

$$H_f = f \frac{L}{D} \frac{v^2}{2g}$$

where: f = friction factor (unitless)

L = length of pipe (ft)

D = diameter of pipe (ft)

v = fluid velocity (ft/sec)

g = gravitational acceleration (ft/sec²)

Example: Darcy's Head Loss Equation A pipe 100 feet long and 20 inches in diameter contains water at 200°F flowing at a mass flow rate of 700 lbm/sec. The water has a density of 60 lbm/ft³ and a viscosity of 1.978×10^{-7} lbf-sec/ft². The relative roughness of the pipe is 0.00008. Calculate the head loss for the pipe.

UNIT II HYDRAULIC SYSTEM & COMPONENTS

Pump types

There are essentially three different types of positive displacement pump used in hydraulic systems.

Gear pumps

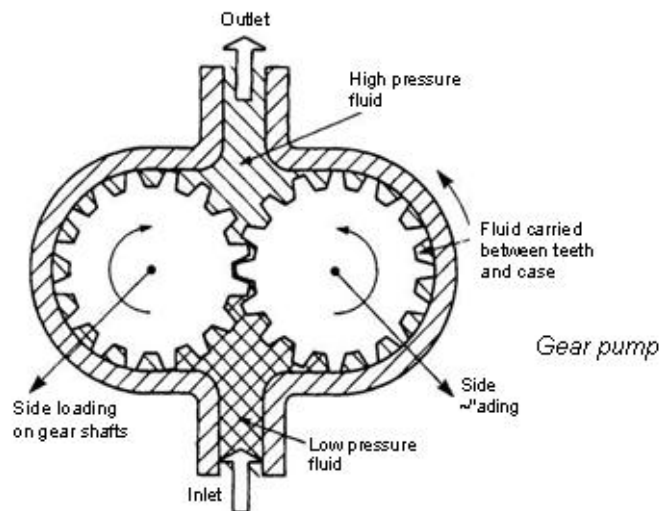
The simplest and most robust positive displacement pump, having just two moving parts, is the gear pump. Its parts are non-reciprocating, move at constant speed and experience a uniform force. Internal construction, shown in Figure 2.7, consists of just two close meshing gear wheels which rotate as shown. The direction of rotation of the gears should be carefully noted; it is the *opposite* of that intuitively expected by most people.

As the teeth come out of mesh at the centre, a partial vacuum is formed which draws fluid into the inlet chamber. Fluid is trapped between the outer teeth and the pump housing, causing a continual transfer of fluid from inlet chamber to outlet chamber where it is discharged to the system.

Pump displacement is determined by: volume of fluid between each pair of teeth; number of teeth; and speed of rotation. Note the pump merely delivers a fixed volume of fluid from inlet port to outlet port for each rotation; outlet port pressure is determined solely by design of the rest of the system.

Performance of any pump is limited by leakage and the ability of the pump to withstand the pressure differential between inlet and outlet ports. The gear pump obviously requires closely meshing gears, minimum clearance between teeth and housing, and also between the gear face and side plates. Often the side plates of a pump are designed as deliberately replaceable wear plates. Wear in a gear pump is primarily caused by dirt particles in the hydraulic fluid, so cleanliness and filtration are particularly important.

The pressure differential causes large side loads to be applied to the gear shafts at 45° to the centre line as shown. Typically, gear pumps are used at pressures up to about 150 bar and capacities of around 150 gpm (6751 min^{-1}). Volumetric efficiency of gear pumps at 90% is lowest of the three pump types.



There are some variations of the basic gear pump. In Figure 2.8, gears have been replaced by lobes giving a pump called, not surprisingly

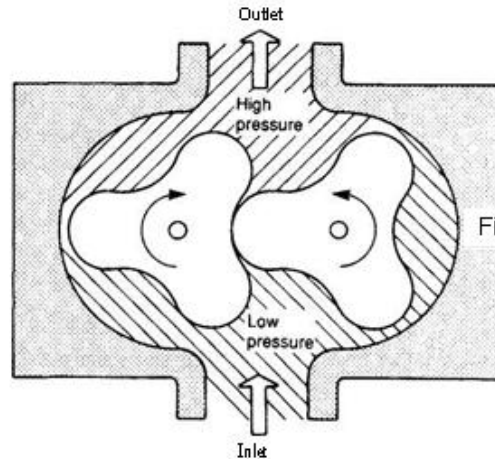
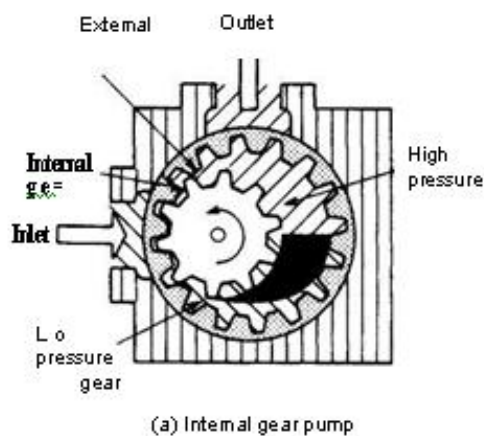
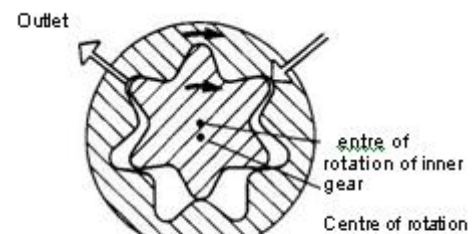


Figure 2.8 The lobe pump

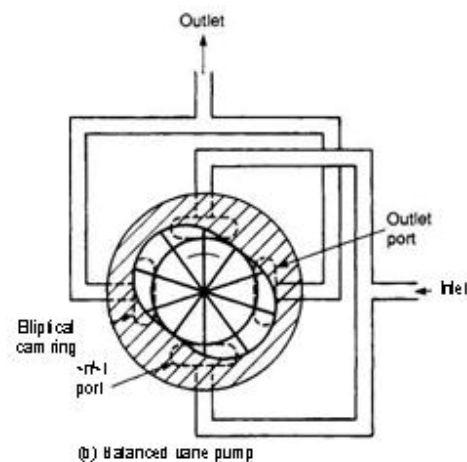


(a) Internal gear pump



(b) Gerotor pump

The Figure is another variation called the internal gear pump, where an external driven gear wheel is connected to a smaller internal gear, with fluid separation as gears disengage being performed by a crescent-shaped moulding. Yet another variation on the theme is the gerotor pump of Figure 2.9b, where the crescent moulding is dispensed with by using an internal gear with one less tooth than the outer gear wheel. Internal gear pumps operate at lower capacities and pressures (typically 70 bar) than other pump types.

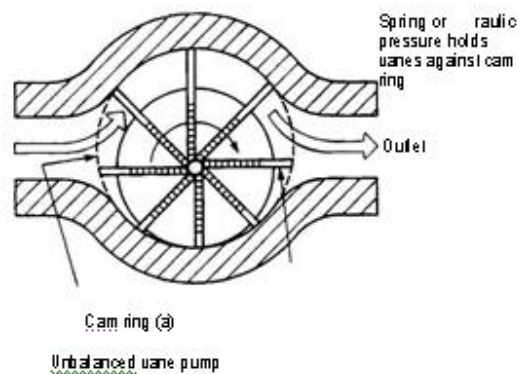


(c) Balanced vane pump

Vane pumps

The major source of leakage in a gear pump arises from the small gaps between teeth, and also between teeth and pump housing. The vane pump reduces this leakage by using spring (or hydraulic) loaded vanes slotted into a driven rotor, as illustrated in the two examples of Figure 2.10. In the pump shown in Figure 2.10a, the rotor is offset within the housing, and the vanes constrained by a cam ring as they cross inlet and outlet ports. Because the vane tips are held against the housing there is little leakage and the vanes compensate to a large degree for wear at vane tips or in the housing itself. There is still, however, leakage between rotor faces and body sides. Pump capacity is determined by vane throw, vane cross sectional area and speed of rotation.

The difference in pressure between outlet and inlet ports creates a severe load on the vanes and a large side load on the rotor shaft which can lead to bearing failure. The pump in Figure 2.10a is consequently known as an unbalanced vane pump. Figure 2.10b shows a balanced vane pump. This features an elliptical cam ring together with two inlet and two outlet ports. Pressure loading still occurs in the vanes but the two identical pump halves create equal but opposite forces on the rotor, leading to zero net force in the shaft and bearings. Balanced vane pumps have much improved service lives over simpler unbalanced vane pumps.



Capacity and pressure ratings of a vane pump are generally lower than gear pumps, but reduced leakage gives an improved volumetric efficiency of around 95%.

In an ideal world, the capacity of a pump should be matched exactly to load requirements. Expression 2.2 showed that input power is proportional to system pressure and volumetric flow rate. A pump with too large a capacity wastes energy (leading to a rise in fluid temperature) as excess fluid passes through the pressure relief valve.

Pumps are generally sold with certain fixed capacities and the user has to choose the next largest size. Figure 2.11 shows a vane pump with adjustable capacity, set by the positional relationship between rotor and inner casing, with the inner casing position set by an external screw.

Piston pumps

A piston pump is superficially similar to a motor car engine, and a simple single cylinder arrangement was shown earlier in Figure 2.2b. Such a simple pump, however, delivering a single pulse of fluid per revolution, generates unacceptably large pressure pulses into the system.

Practical piston pumps therefore employ multiple cylinder and pistons to smooth out fluid

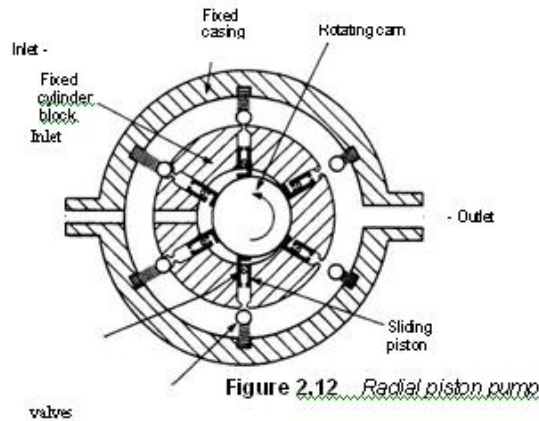


Figure 2.12 Radial piston pump

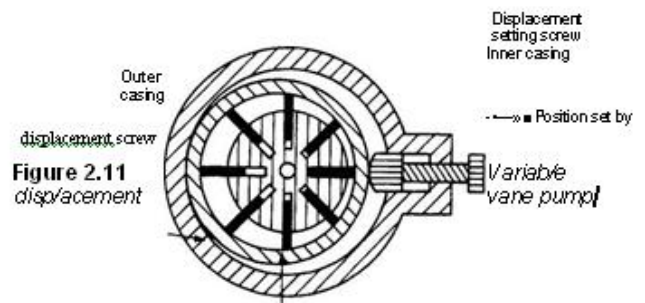
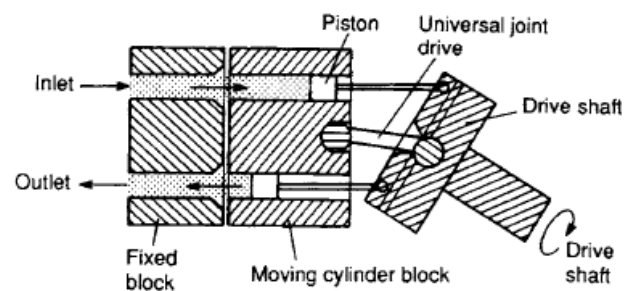
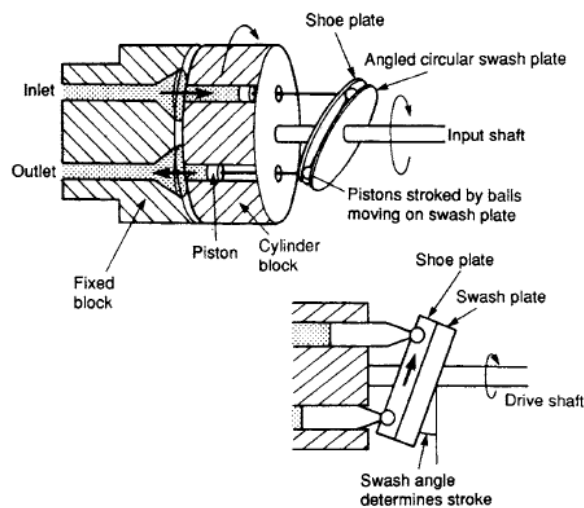


Figure 2.11 displacement

delivery, and much ingenuity goes into designing multicylinder pumps which are surprisingly compact.

Figure 2.12 shows one form of radial piston pump. The pump consists of several hollow pistons inside a stationary cylinder block. Each piston has spring-loaded inlet and outlet valves. As the inner cam rotates, fluid is transferred relatively smoothly from inlet port to the outlet port.



Actuators

A hydraulic or pneumatic system is generally concerned with moving, gripping or applying



force to an object. Devices which actually achieve this objective are called actuators, and can be split into three basic types.

Linear actuators, as the name implies, are used to move an object or apply a force in a straight line. Rotary actuators are the hydraulic and pneumatic equivalent of an electric motor. This chapter discusses linear and rotary actuators.

The third type of actuator is used to operate flow control valves for process control of gases, liquids or steam. These actuators are generally pneumatically operated and are discussed with process control pneumatics in Chapter 7.

Linear actuators

The basic linear actuator is the cylinder, or ram, shown in schematic form in Figure 5.1. Practical constructional details are discussed later. The cylinder in Figure 5.1 consists of a piston, radius R , moving in a bore. The piston is connected to a rod of radius r which drives the load. Obviously if pressure is applied to port X (with port Y venting) the piston extends. Similarly, if pressure is applied to port Y (with port Z venting), the piston retracts.

The force applied by a piston depends on both the area and the applied pressure. For the extend stroke, area A given by πR^2 . For a pressure P applied to port X, the extend force available is:

$$F_c = P \pi R^2 \quad (5.1)$$

cylinder

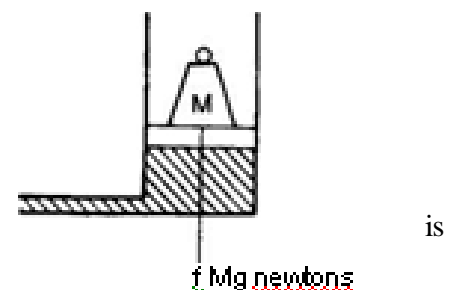


Fig A mass supported by a

The units of expression 5.1 depend on the system being used. If SI units are used, the force is in newtons.

Expression 5.1 gives the maximum achievable force obtained with the cylinder in a stalled condition. One example of this occurs where an object is to be gripped or shaped.

In Figure 5.2 an object of mass M is lifted at constant speed. Because the object is not accelerating, the upward force is equal to Mg newtons (in SI units) which from expression 5.1 gives the pressure in the cylinder. This is lower than the maximum system pressure; the pressure drop occurring across flow control valves and system piping. Dynamics of systems similar to this are discussed later.

There are five basic parts in a cylinder; two end caps (a base cap and a bearing cap) with port connections, a cylinder barrel, a piston and the rod itself. This basic construction allows fairly simple manufacture as end caps and pistons are common to cylinders of the same diameter, and only (relatively) cheap barrels and rods need to be changed to give different length cylinders. End caps can be secured to the barrel by welding, tie rods or by threaded connection. Basic constructional details are shown in Figure 5.9.

The inner surface of the barrel needs to be very smooth to prevent wear and leakage. Generally a seamless drawn steel tube is used which is machined (honed) to an accurate finish. In applications where the cylinder is used infrequently or may come into contact with corrosive materials, stainless steel, aluminium or brass tube may be used.

Pistons are usually made of cast iron or steel. The piston not only transmits force to the rod, but must also act as a sliding bearing in the barrel (possibly with side forces if the rod is subject to a lateral force) and provide a seal between high and low pressure sides. Piston seals are generally used between piston and barrel. Occasionally small leakage can be tolerated and seals are not used. A bearing surface (such as bronze) is deposited on to the piston surface then honed to a finish similar to that of the barrel.

The surface of the cylinder rod is exposed to the atmosphere when extended, and hence liable to suffer from the effects of dirt, moisture and corrosion. When retracted, these antisocial materials may be drawn back inside the cylinder. Heat treated chromium alloy steel is generally used for strength and to reduce effects of corrosion.

A wiper or scraper seal is fitted to the end cap where the rod enters the cylinder to remove dust particles. In very dusty atmospheres external rubber bellows may also be used to exclude dust (Figure 5.9a) but these are vulnerable to puncture and splitting and need regular inspection. The bearing surface, usually bronze, is fitted behind the wiper seal.

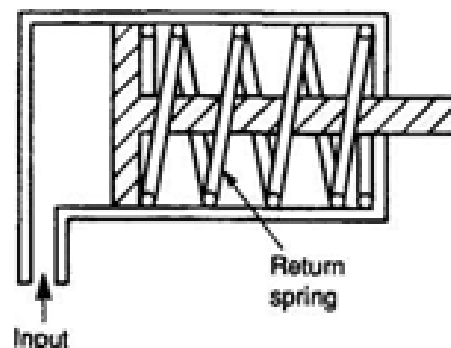


Fig Single-acting cylinder

An internal sealing ring is fitted behind the bearing to prevent high pressure fluid leaking out along the rod. The wiper seal, bearing and sealing ring are sometimes combined as a cartridge assembly to simplify maintenance. The rod is generally attached to the piston via a

threaded end as shown in Figures 5.9b and c. Leakage can occur around the rod, so seals are again needed. These can be cap seals (as in Figure 5.9b) which combine the roles of piston and rod seal, or a static O ring around the rod (as in Figure 5.9c).

End caps are generally cast (from iron or aluminium) and incorporate threaded entries for ports. End caps have to withstand shock loads at extremes of piston travel. These loads arise not only from fluid pressure, but also from kinetic energy of the moving parts of the cylinder and load.

These end of travel shock loads can be reduced with cushion valves built into the end caps. In the cylinder shown in Figure 5.10, for example, exhaust fluid flow is unrestricted until the plunger

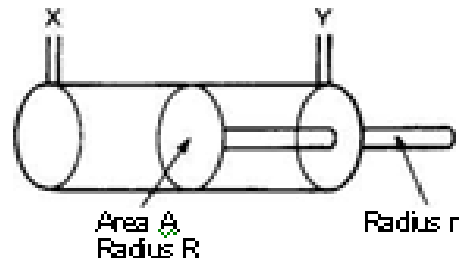


Fig A simple cylinder

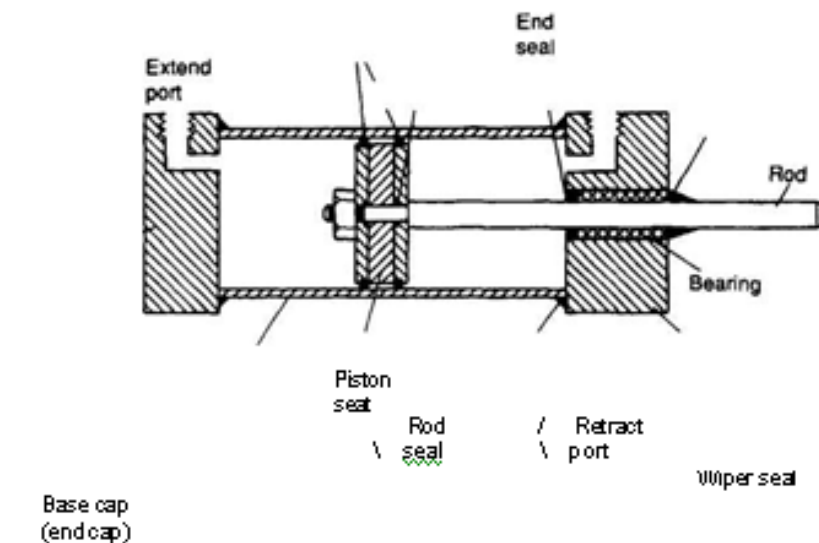


Figure 5.8 Construction of a typical cylinder

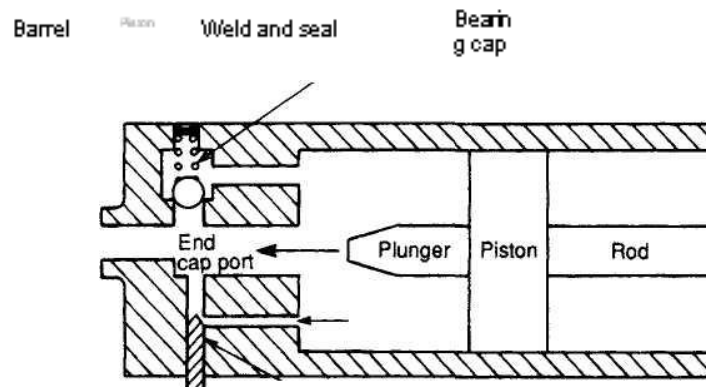


Fig Cylinder cushioning

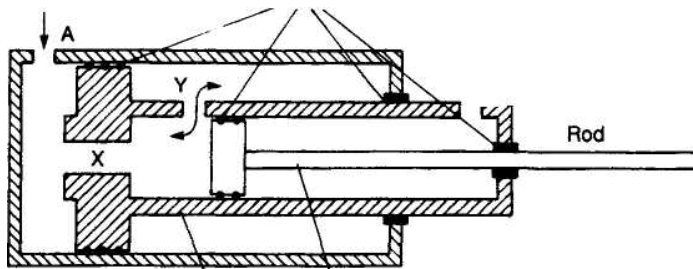


Fig Two-stage telescopic piston

The stroke of a simple cylinder must be less than barrel length, giving at best an extended/retracted ratio of 2:1. Where space is restricted, a telescopic cylinder can be used.

Figure 5.12 shows the construction of a typical double-acting unit with two pistons. To extend, fluid is applied to port A. Fluid is applied to both sides of piston 1 via ports X and Y, but the difference in areas between sides of piston 1 causes the piston to move to the right.

To retract, fluid is applied to port B. A flexible connection is required for this port. When piston 2 is driven fully to the left, port Y is now connected to port B, applying pressure to the right-hand side of piston 1 which then retracts.

The construction of telescopic cylinders requires many seals which makes maintenance complex. They also have smaller force for a given diameter and pressure, and can only tolerate small side loads.

Pneumatic cylinders are used for metal forming, an operation requiring large forces. Pressures in pneumatic systems are lower than in hydraulic systems, but large impact loads can be obtained by accelerating a hammer to a high velocity then allowing it to strike the target.

Such devices are called impact cylinders and operate on the principle illustrated in Figure 5.13. Pressure is initially applied to port

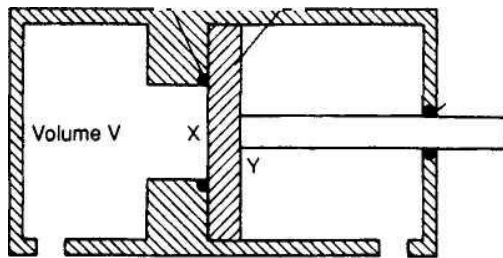


Fig An impact cylinder

B to retract the cylinder. Pressure is then applied to both ports A and B, but the cylinder remains in a retracted state because area X is less than area Y. Port B is then vented rapidly. Immediately, the full piston area experiences port A pressure. With a large volume of gas stored behind the piston, it accelerates rapidly to a high velocity (typically 10 m s^{-1}).

Rotary actuators

Rotary actuators are the hydraulic or pneumatic equivalents of electric motors. For a given torque, or power, a rotary actuator is more compact than an equivalent motor, cannot be damaged by an indefinite stall and can safely be used in an explosive atmosphere. For variable speed applications, the complexity and maintenance requirements of a rotary actuator are similar to a thyristor-controlled DC drive, but for fixed speed applications, the AC induction motor (which can, for practical purposes, be fitted and forgotten) is simpler to install and maintain.

A rotary actuator (or, for that matter, an electric motor) can be defined in terms of the torque it produces and its running speed, usually given in revs per minute (rpm). Definition of torque is illustrated in Figure 5.22, where a rotary motion is produced against a force of F newtons acting at a radial distance d metres from a shaft centre. The device is then producing a torque T given by the expression"

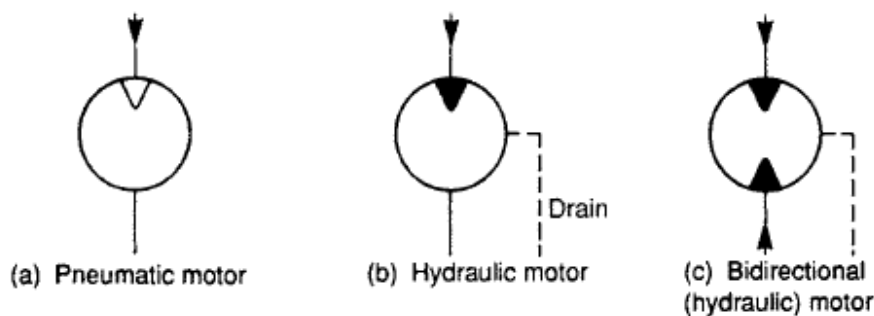


Fig Rotary actuator symbols

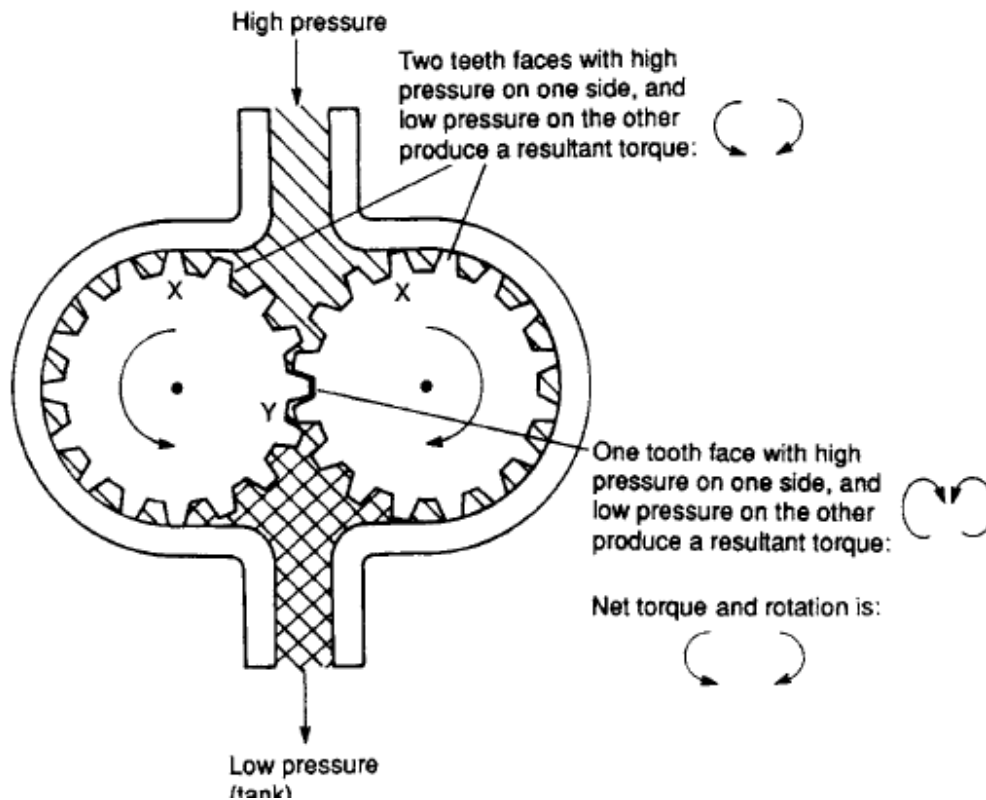


Fig A gear motor

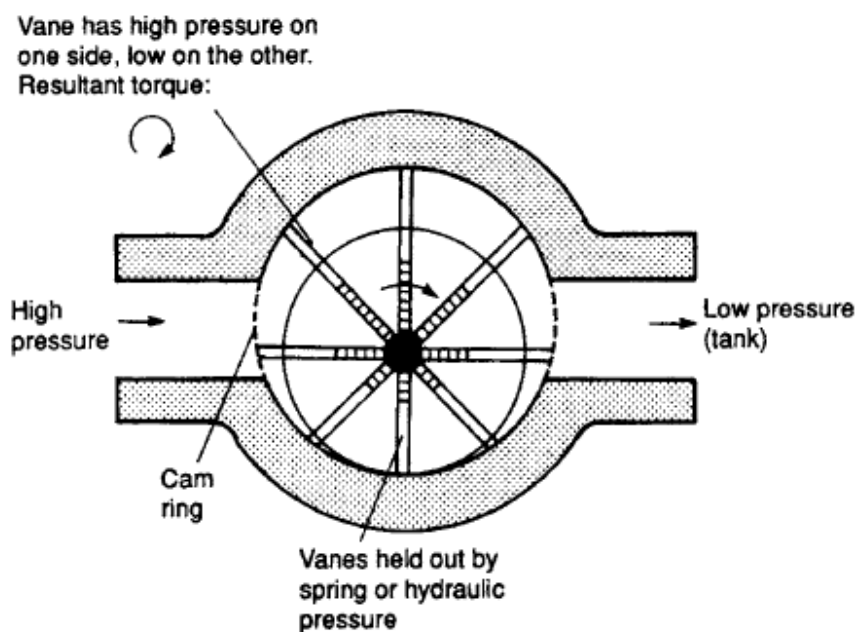


Fig A vane motor

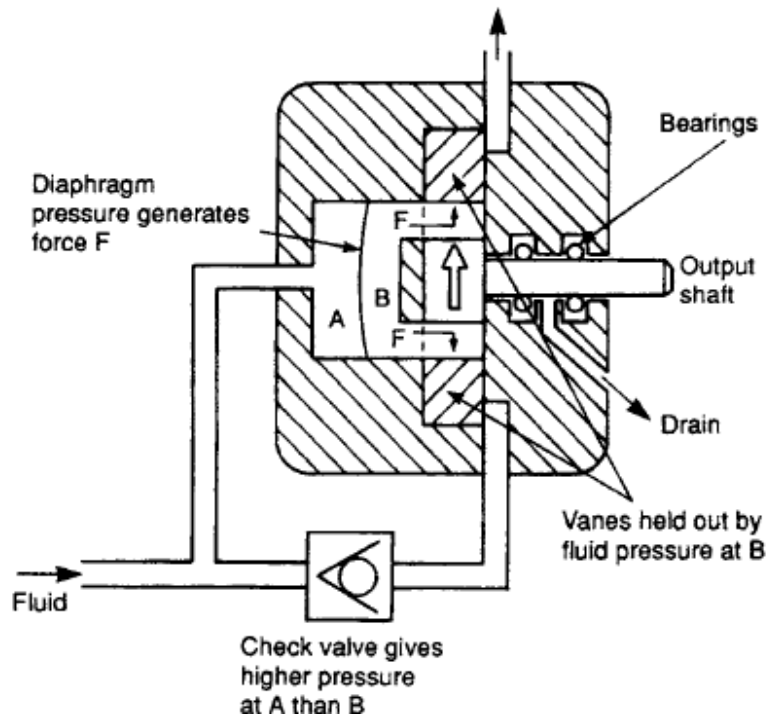


Fig Vane operation in hydraulic motor

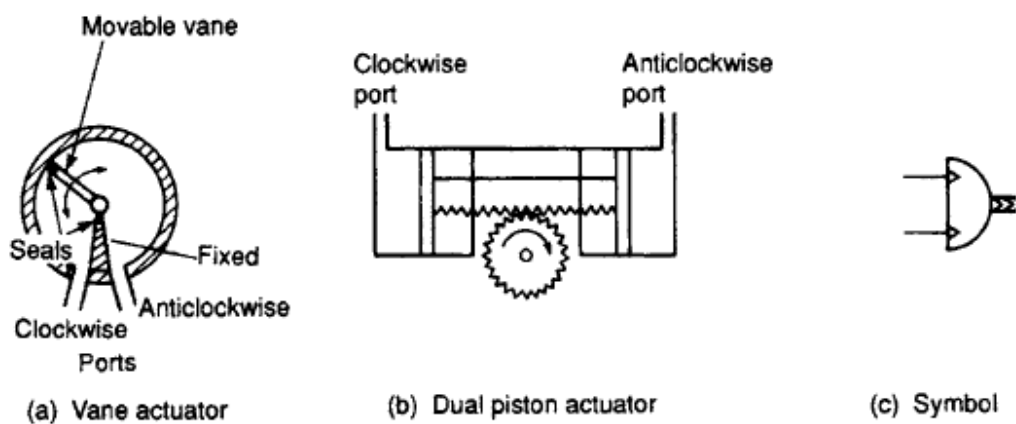


Fig Limited motion rotary actuators

Application

The operational speed of an actuator is determined by the fluid flow rate and the actuator area (for a cylinder) or the displacement (for a motor). The physical dimensions are generally fixed for an actuator, so speed is controlled by adjusting the fluid flow to (or restricting flow from) the

actuator. Rotary actuator speed can also be controlled by altering swash plate angle. The compressibility of air, normally advantageous where smooth operation is concerned, makes flow control more difficult for pneumatic than hydraulic systems. Although techniques described below can be applied in pneumatics, precise slow-speed control of a pneumatic actuator is achieved with external devices described later. There are essentially four ways in which fluid flow can be controlled. The first is shown in Figure 5.30, where a pump delivers a fluid volume V per minute. Because the pump is a fixed displacement device this volume of fluid *must* go either back to the tank or to the actuator. When the control valve moves from its centre position, the actuator moves with a velocity:

$$v = \frac{V}{A}$$

where A is the piston area. If pump delivery volume V can be adjusted (by altering swash plate angle, say,) and the pump feeds no other device, no further speed control is needed.

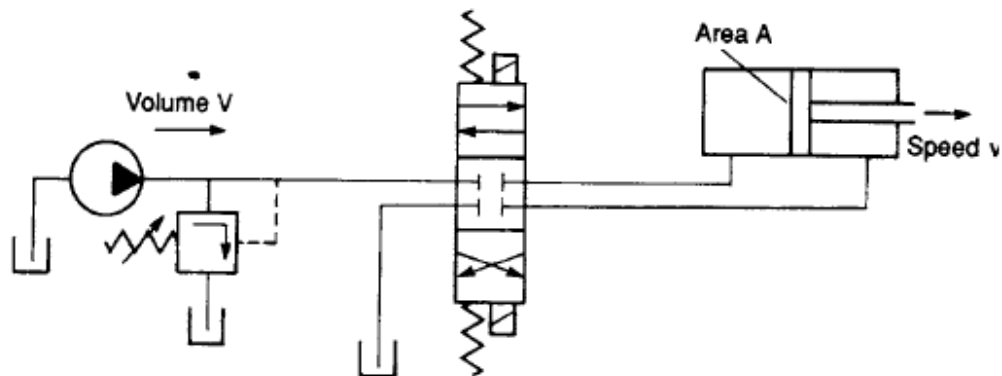


Fig Speed control by pump volume

UNIT III DESIGN OF HYDRAULIC CIRCUITS

Types of control valve

There are essentially three types of control valve; poppet valves spool valves and rotary valves.

POPPET VALVES

In a poppet valve, simple discs, cones or balls are used in conjunction with simple valve seats to control flow. allows fluid to flow from port P to port A. When the button is released, spring and fluid pressure force the ball up again closing the valve. Fig shows the construction and symbol of a disc seal 3/2 poppet. With the pushbutton released, ports A and R are linked via the hollow pushbutton stem. If the pushbutton is pressed, port R is first sealed, then the valve disc pushed down to open the valve and connect ports P and A. As before, spring and fluid pressure from port P closes the valve. The valve construction and symbol shown in Fig is a poppet changeover 4/2 valve using two stems and disc valves. With the pushbutton released, ports A and R are linked via the hollow left-hand stem and ports P and B linked via the normally-open right hand disc valve.

When the pushbutton is pressed, the link between ports A and R is first closed, then the link between P and B closed. The link between A and P is next opened, and finally the link between B and R opened. When the pushbutton is released, air and spring pressure puts the valve back to its original state. Poppet valves are simple, cheap and robust, but it is generally simpler to manufacture valves more complicated than those shown in Figure 4.11 by using spool valves. Further, a major disadvantage of poppet valves is the force needed to operate them. In the poppet valve of Figure 4.10, for example, the force required on the pushbutton to operate the valve is $P \times a$ newtons. Large capacity valves need large valve areas, leading to large operating force. The high pressure in hydraulic systems thus tends to prevent use of simple poppet valves and they are, therefore, mainly found in low pressure pneumatic systems.

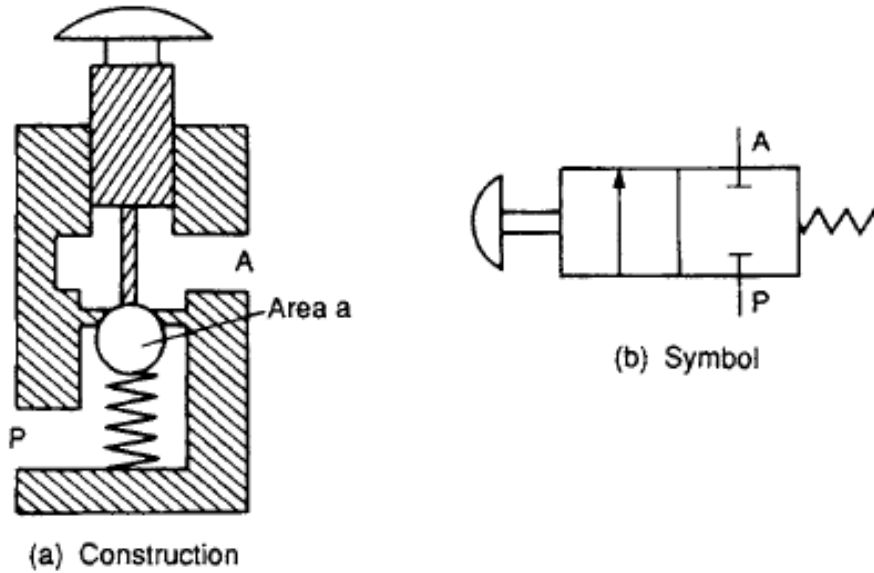


Fig Simple 2/2 poppet valve

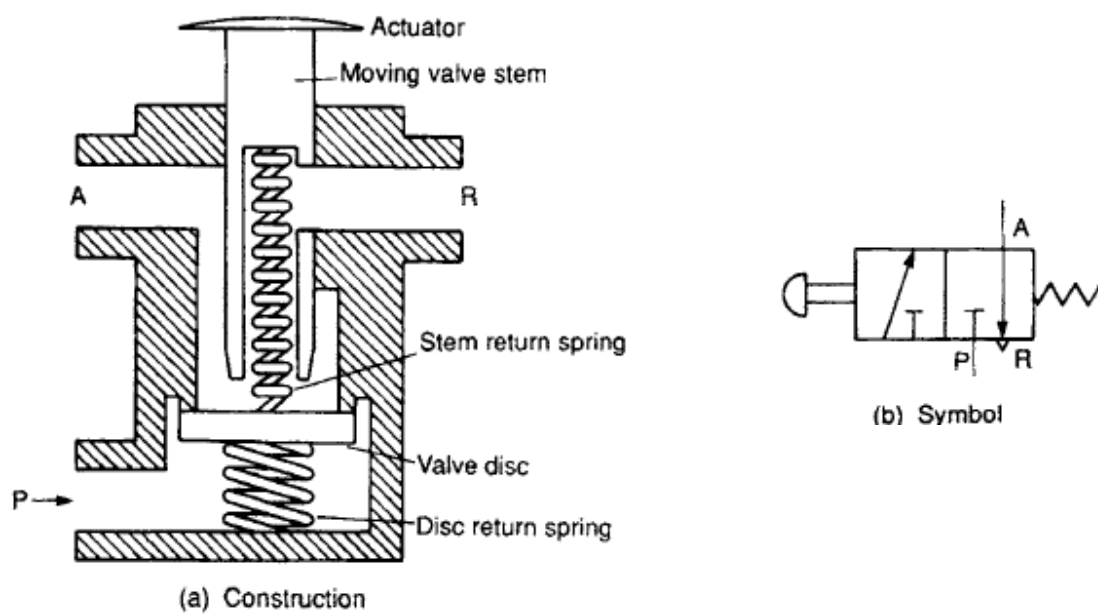


Fig A 3/2 poppet valve

SPOOL VALVES

Spool (or slide) valves are constructed with a spool moving horizontally within the valve body, as shown for the 4/2 valve in Figure 4.12. Raised areas called 'lands' block or open ports to give the required operation.

The operation of a spool valve is generally balanced. In the valve construction in Figure 4.12b, for example, pressure is applied to opposing faces D and E and low tank pressure to faces F and G. There is no net force on the spool from system pressure, allowing the spool to be easily moved

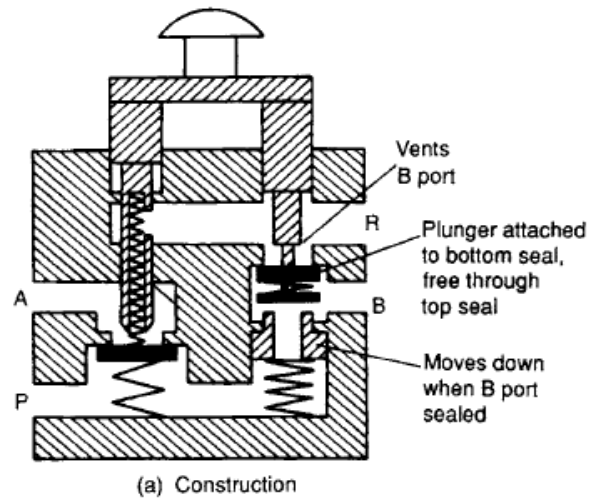
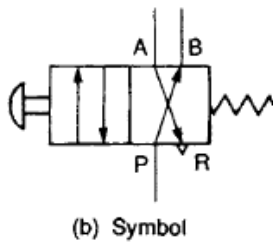


Fig A 4/2 poppet valve

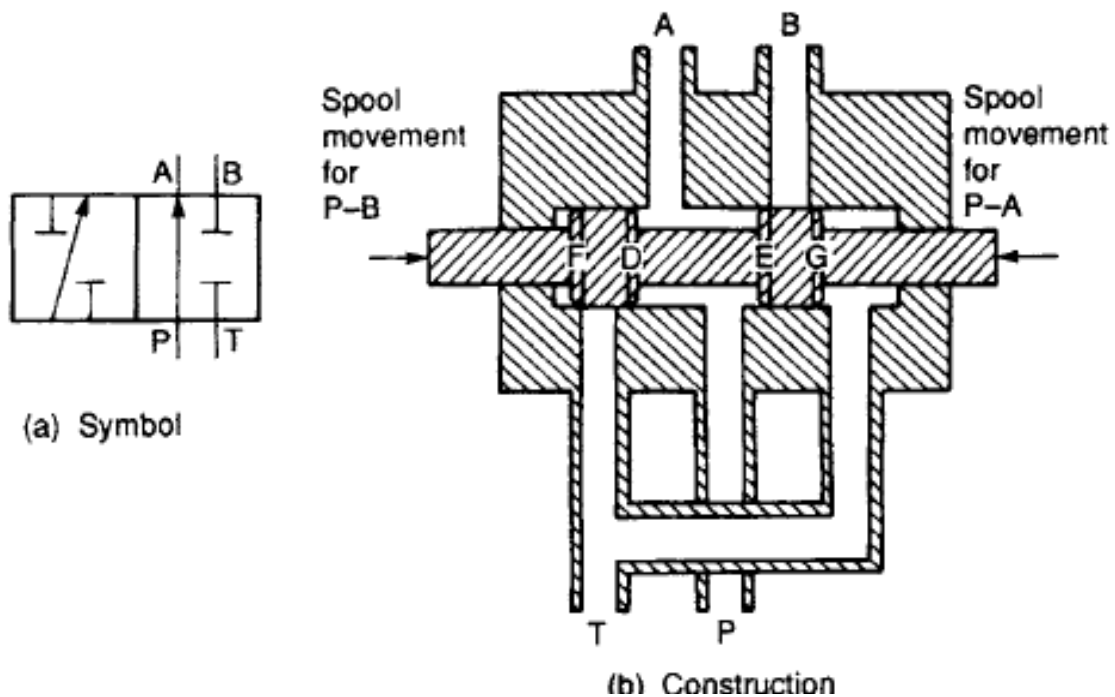


Fig Four-way spool valve

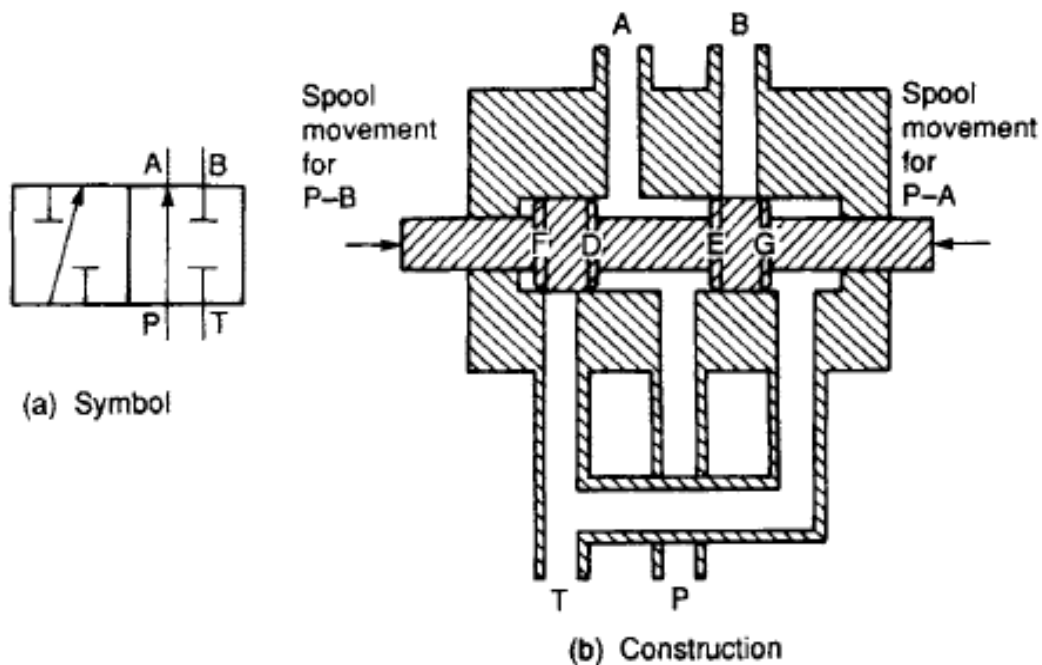


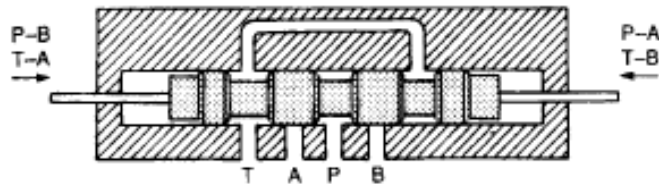
Fig Four-way spool valve

Fig is a changeover 4/2 spool valve. Comparison of the valves shown in Figures 4.12 and 4.13 shows they have the same body construction, the only difference being the size and position

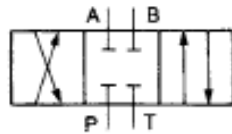
of lands on the spool. This is a major cost-saving advantage of spool valves; different operations can be achieved with a common body and different spools. This obviously reduces manufacturing costs. Fig shows various forms of three position changeover valves; note, again, these use one body with different functions achieved by different land patterns. Spool valves are operated by shifting the spool. This can be achieved by button, lever or striker, or remotely with a solenoid. Self-centring can easily be provided if springs are mounted at the end of the spool shaft. Solenoid-operated valves commonly work at 24 V DC or 110 V AC. Each has its own advantages and disadvantages. A DC power supply has to be provided for 24 V DC solenoids, which, in large systems, is substantial and costly. Operating current of a 24 V solenoid is higher than a 110 V solenoid's. Care must be taken with plant cabling to avoid voltage drops on return legs if a common single line return is used.

Current through a DC solenoid is set by the winding resistance. Current in an AC solenoid, on the other hand, is set by the inductance of the windings, and this is usually designed to give a high inrush current followed by low holding current. This is achieved by using the core of the solenoid (linked to the spool) to raise the coil inductance when the spool has moved. One side effect of this is that a jammed spool results in a permanent high current which can damage the coil or the device driving it. Each and every AC solenoid should be protected by an individual fuse. DC solenoids do not suffer from this characteristic.

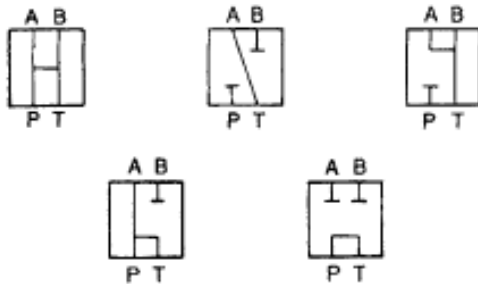
A burned out DC solenoid coil is almost unknown. Whatever form of solenoid is used it is very useful when fault finding to have local electrical indication built into the solenoid plugtop. This allows a fault to be quickly identified as either an electrical or hydraulic problem. Fault finding is discussed further in Chapter 8. A solenoid can exert a pull or push of about 5 to 10 kg. This is adequate for most pneumatic spool valves, but is too low for direct operation of large capacity hydraulic valves. Here pilot operation must be used, a topic discussed later.



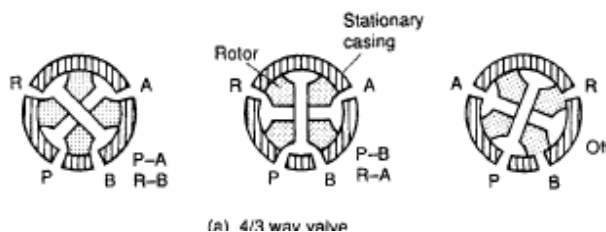
(a) Construction of centre off valve



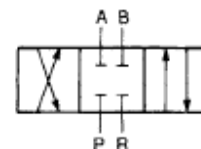
(b) Symbol

*Fig Three position four-way valves***ROTARY VALVES**

Rotary valves consist of a rotating spool which aligns with holes in the valve casing to give the required operation. Fig shows the construction and symbol of a typical valve with centre off action. Rotary valves are compact, simple and have low operating forces. They are, however, low pressure devices and are consequently mainly used for hand operation in pneumatic systems.



(a) 4/3 way valve



(b) Symbol

*Fig Rotary Valves***PILOT-OPERATED VALVES**

With large capacity pneumatic valves (particularly poppet valves) and most hydraulic valves, the operating force required to move the valve can be large. If the required force is too large for a solenoid or manual operation, a two-stage process called pilot operation is used.

The principle is shown in Figure 4.16. Valve 1 is the main operating valve used to move a ram. The operating force required to move the valve, however, is too large for direct operation by a solenoid, so a second smaller valve 2, known as the pilot valve, has been added to allow the main valve to be operated

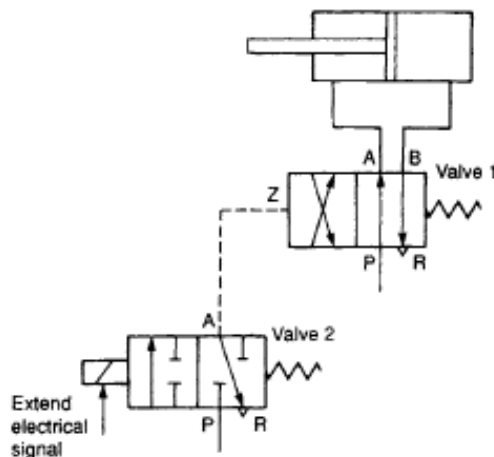


Fig Pilot-operated valve

by system pressure. Pilot pressure lines are normally shown dotted in circuit diagrams, and pilot ports on main valves are denoted Z, Y, X and so on. In Fig pilot port Z is depressurised with the solenoid deenergised, and the ram is retracted. When the solenoid is energized valve 2 changes over, pressurising Z; causing valve 1 to energise and the ram to extend. Although pilot operation can be achieved with separate valves it is more usual to use a pilot/main valve assembly manufactured as a complete

readymade unit. Figure 4.17 shows the operation of a pilot-operated 3/2 pneumatic valve. The solenoid operates the small pilot valve directly. Because this valve has a small area, a low operating force is required. The pilot valve applies line pressure to the top of the control valve causing it to move down, closing the exhaust port. When it contacts the main valve disc there are two forces acting on the valve stem. The pilot valve applies a downwards force of $P \times D$, where P is the line pressure and D is the area of the control valve. Line pressure also applies an upwards force $P \times E$ to the stem, where E is the area of the main valve. The area of the control valve, D , is greater than area of the main valve E , so the downwards force is the larger and the valve opens. When the solenoid de-energises, the space above the control valve is vented. Line and spring pressure on the main valve causes the valve stem to rise again, venting port A.

A hydraulic 4/2 pilot-operated spool valve is shown in Fig The ends of the pilot spool in most hydraulic pilot-operated valves are visible from outside the valve. This is useful from a maintenance viewpoint as it allows the operation of a valve to be checked. In extreme cases the valve can be checked by pushing the pilot spool directly with a suitably sized rod (welding rod is ideal !). Care must be taken to check solenoid states on dual solenoid valves before attempting manual operation. Overriding an energised AC solenoid creates a large current which may damage the coil, (or blow the fuse if the solenoid has correctly installed protection).

CHECK VALVES

Check valves only allow flow in one direction and, as such, are similar in operation to electronic diodes. The simplest construction is the ball and seat arrangement of the valve in Fig, commonly used in pneumatic systems. The right angle construction in Fig is better suited to the higher pressures of a hydraulic system. Free flow direction is normally marked with an arrow on the valve casing. A check valve is represented by the graphic symbols in Fig. The symbol in Fig is rather complex and the simpler symbol in Fig is more commonly used.

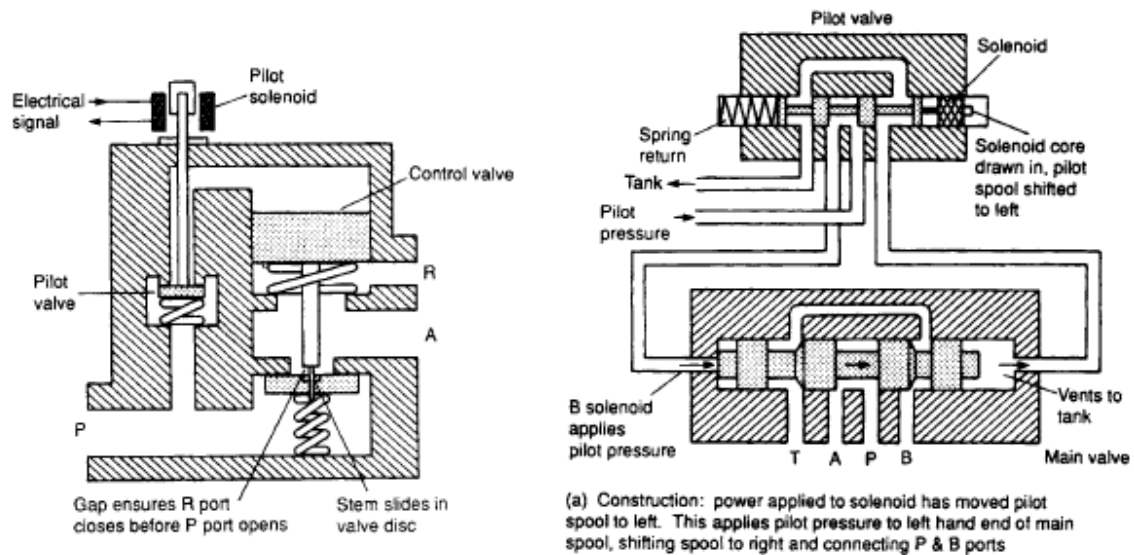


Fig Construction of a pilot-operated 3/2 valve

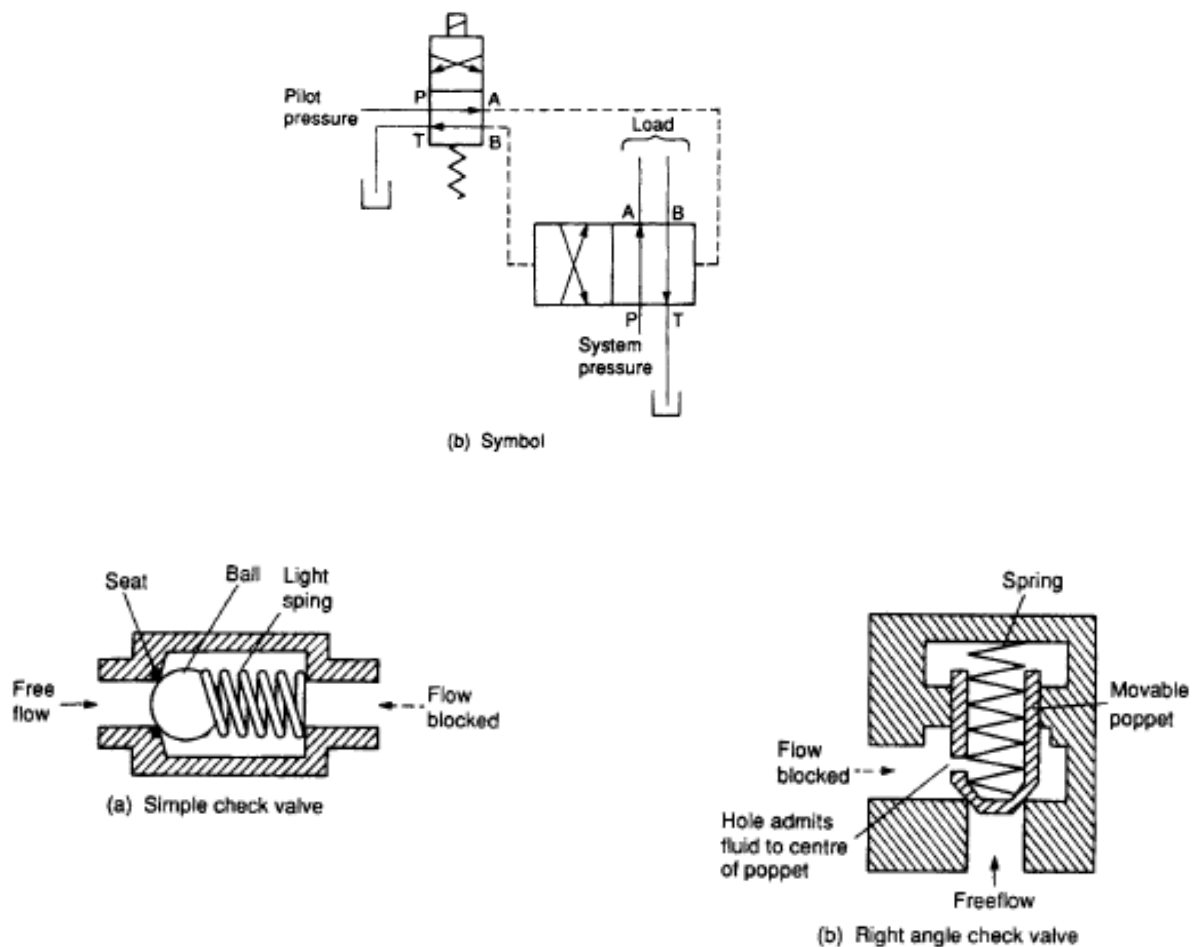


Fig Pilot-operated valve
Fig Check valves

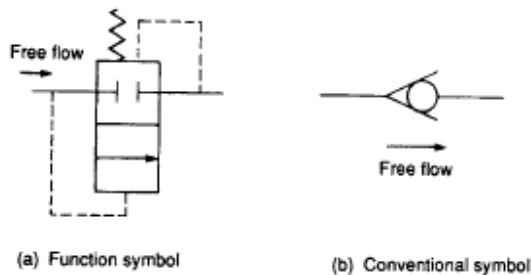


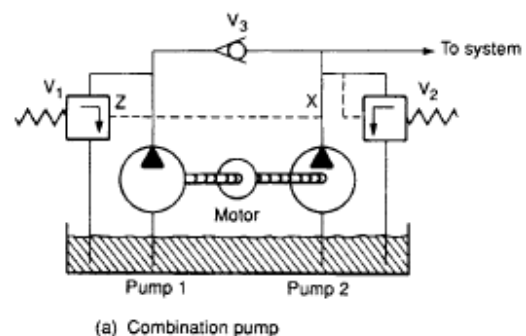
Fig Check valve symbols

illustrates several common applications of check valves. Fig shows a combination pump, used where an application requires large volume and low pressure, or low volume and high pressure. A typical case is a clamp required to engage quickly (high volume and low pressure) then grip (minimal volume but high pressure). Pump 1 is the high volume and low pressure pump, and pump 2 the high pressure pump. In high volume mode both pumps deliver to the system, pump 1 delivering through the check valve V 3. When high pressure is required, line pressure at X rises operating unloading valve V 1 via pilot port Z taking pump 1 off load. Pump 2 delivers the required pressure set by relief valve V 2, with the check valve preventing fluid leaking back to pump 1 and V1. Fig shows a hydraulic circuit with a pressure storage device called an accumulator (described in a later chapter). Here a check valve allows the pump to unload via the pressure regulating valve, while still maintaining system pressure from the accumulator.

A spring-operated check valve requires a small pressure to open (called the cracking pressure) and acts to some extent like a low pressure relief valve. This characteristic can be used to advantage. In Fig pilot pressure is derived before a check valve, and in Fig a check valve is used to protect a blocked filter by diverting flow around the filter when pressure rises. A check valve is also included in the tank return to prevent fluid being sucked out of the tank when the pump is turned off.

PILOT-OPERATED CHECK VALVES

The cylinder in the system in Fig should, theoretically, hold position when the control valve is in its centre, off, position. In practice, the cylinder will tend to creep because of leakage in the control valve. Check valves have excellent sealage in the closed position, but a simple check valve cannot be used in the system in Fig because flow is required in both directions. A pilot-operated check is similar to a basic check valve but can be held open permanently by application of an external pilot pressure signal. There are two basic forms of



(a) Combination pump

pilot-operated check valves, shown in Fig They operate in a similar manner to basic check valves, but with pilot pressure directly opening the valves. In the 4C valve shown in Fig inlet pressure assists the pilot. The symbol of a pilot-operated check valve is shown in Fig The cylinder application of Figure 4.22 is redrawn with pilotoperated check valves in Fig The pilot lines are connected to the pressure line feeding the other side of the cylinder. For any cylinder movement, one check valve is held open by flow (operating as a normal check valve) and the other is held open by pilot pressure. For no required movement, both check valves are closed and the cylinder is locked in position.

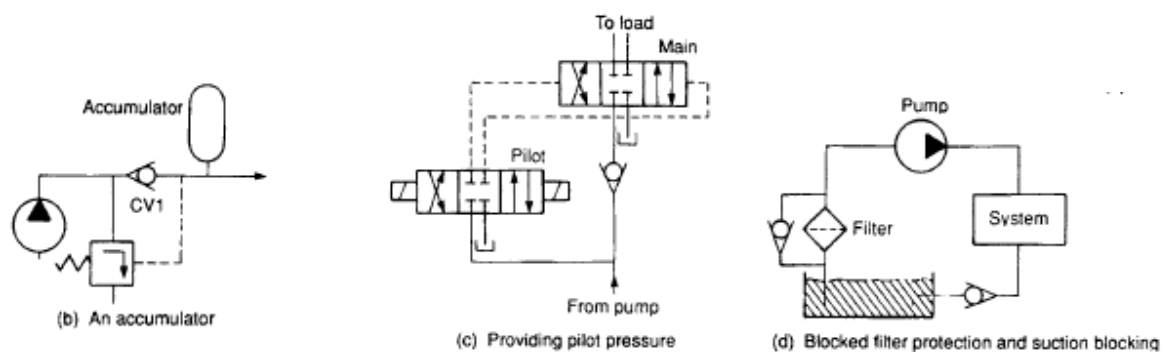


Fig Check valve applications

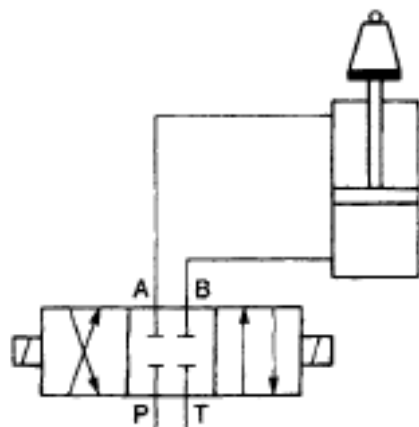


Fig System requiring a check valve. In the off position the load 'creeps'

RESTRICTION CHECK VALVES

The speed of a hydraulic or pneumatic actuator can be controlled by adjusting the rate at which a fluid is admitted to, or allowed out from, a device. This topic is discussed in more detail in Chapter 5 but a speed control is often required to be direction-sensitive and this requires the inclusion of a check valve.

A restriction check valve (often called a throttle relief valve in pneumatics) allows full flow in one direction and a reduced flow in the other direction. Figure 4.24a shows a simple hydraulic valve and Figure 4.24b a pneumatic valve. In both, a needle valve sets restricted flow to the required valve. The symbol of a restriction check valve is shown in Fig shows a typical application in which the cylinder extends at full speed until a limit switch makes, then extend further at low speed. Retraction is at full speed. A restriction check valve V2 is fitted in one leg of the cylinder. With the cylinder retracted, limit-operated valve V3 is open allowing free flow of fluid from the cylinder as it extends. When the striker plate on the cylinder ram hits the limit, valve V3 closes and flow out of the cylinder is now restricted by the needle valve setting of valve V2. In the reverse direction, the check valve on valve V2 opens giving full speed of retraction.

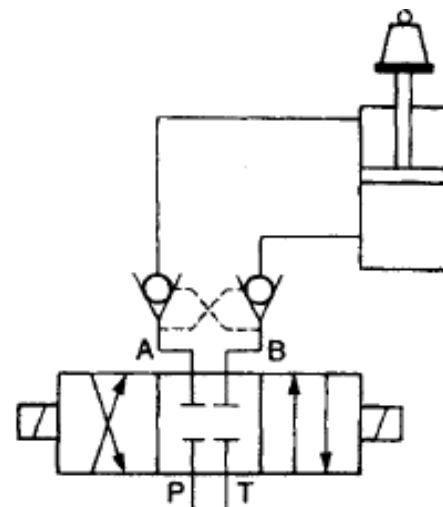
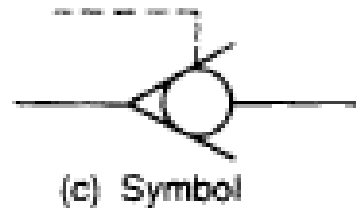
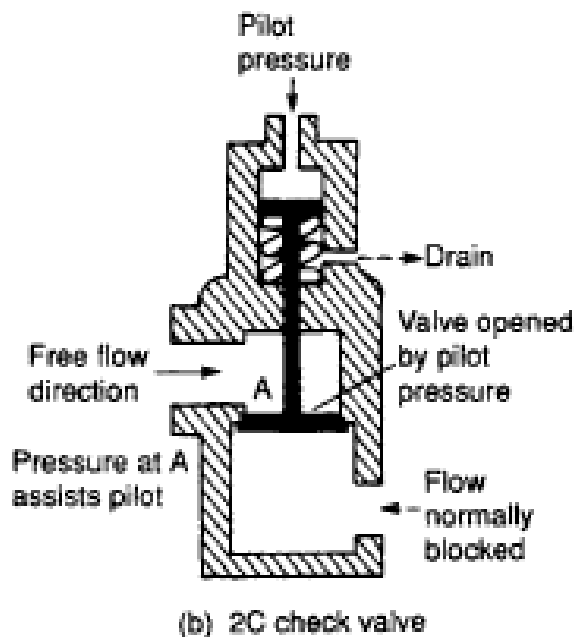
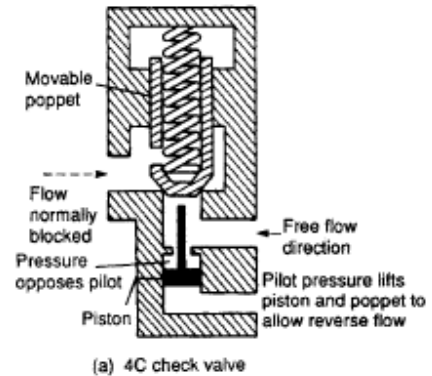


Fig Pilot-operated check valves

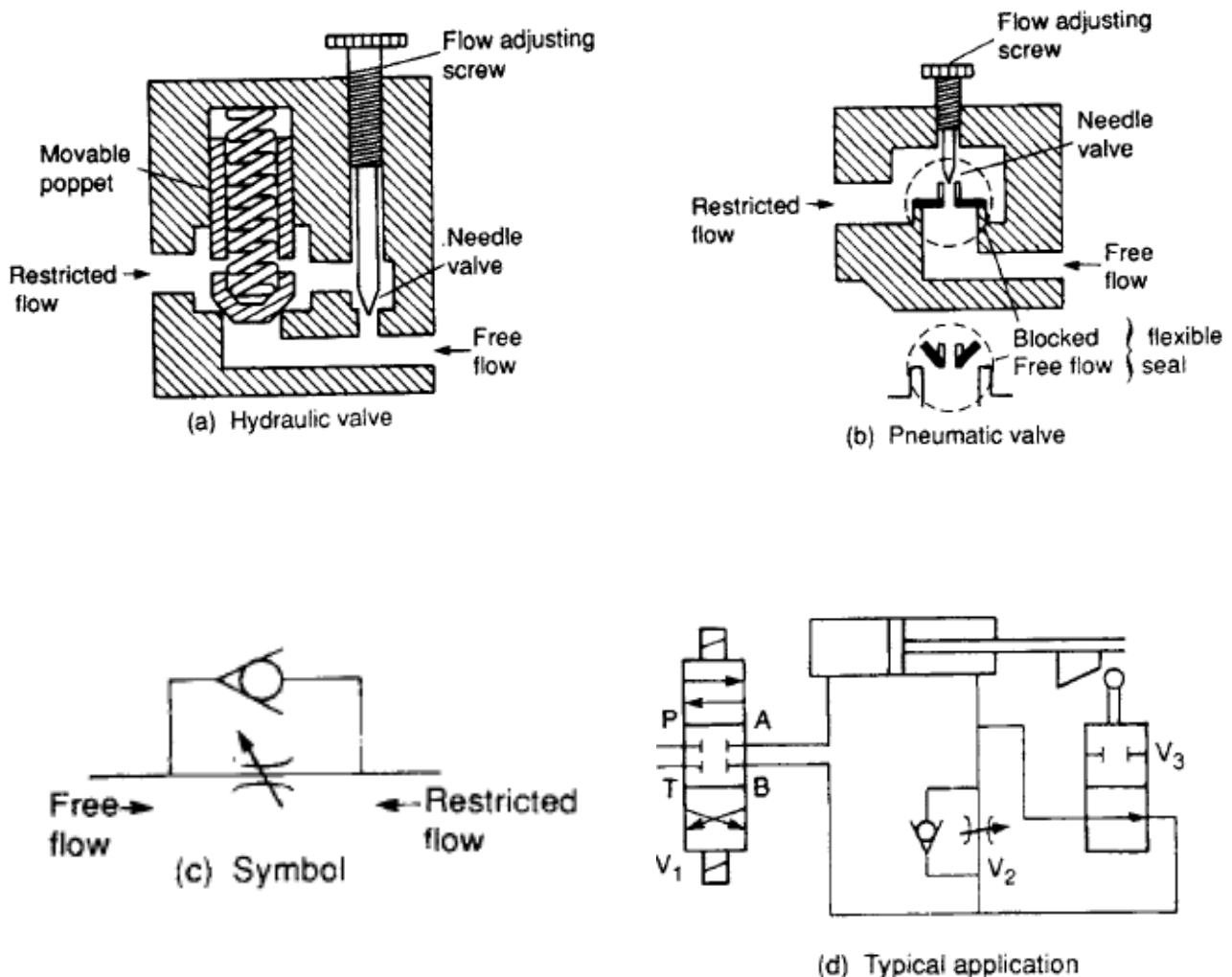
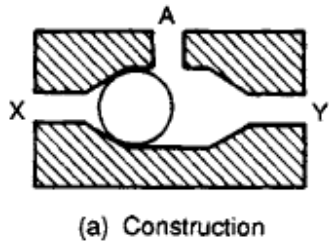


Fig Restriction check valve

SHUTTLE AND FAST EXHAUST VALVES

A shuttle valve, also known as a double check valve, allows pressure in a line to be obtained from alternative sources. It is primarily a pneumatic device and is rarely found in hydraulic circuits. Construction is very simple and consists of a ball inside a cylinder, as shown in Figure 4.25a. If pressure is applied to port X, the ball is blown to the right blocking port Y and linking ports X and A. Similarly, pressure to port Y alone connects ports Y and A and blocks port X. The symbol of a shuttle valve is given in Fig A typical application is given in Fig where a spring return cylinder is operated from either of two manual stations. Isolation between the two stations is provided by the shuttle valve. Note a simple T-connection cannot be used as each



valve has its A port vented to the exhaust port. A fast exhaust valve (Figure 4.26) is used to vent cylinders quickly. It is primarily used with spring return (single-acting) pneumatic cylinders. The device shown in Figure 4.26a consists of a movable disc which allows port A to be connected to pressure port P or large exhaust port R. It acts like, and has the same symbol as, a shuttle valve.

A typical application is shown in Fig Fast exhaust valves are usually mounted local to, or directly onto, cylinders and speed up response by avoiding any delay from return pipes and control valves. They also permit simpler control valves to be used.

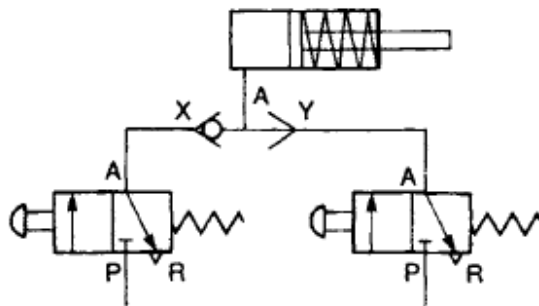
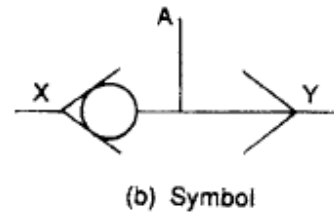


Fig Pneumatic shuttle valve

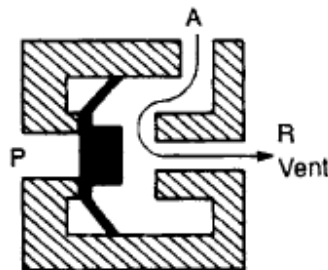
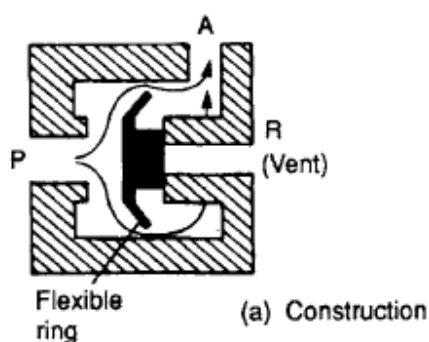
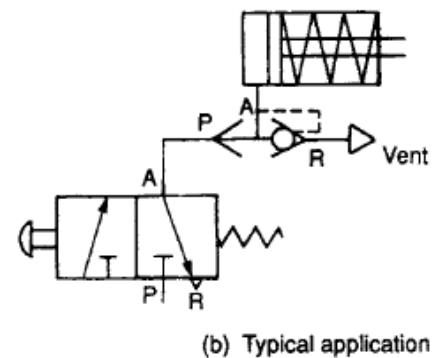
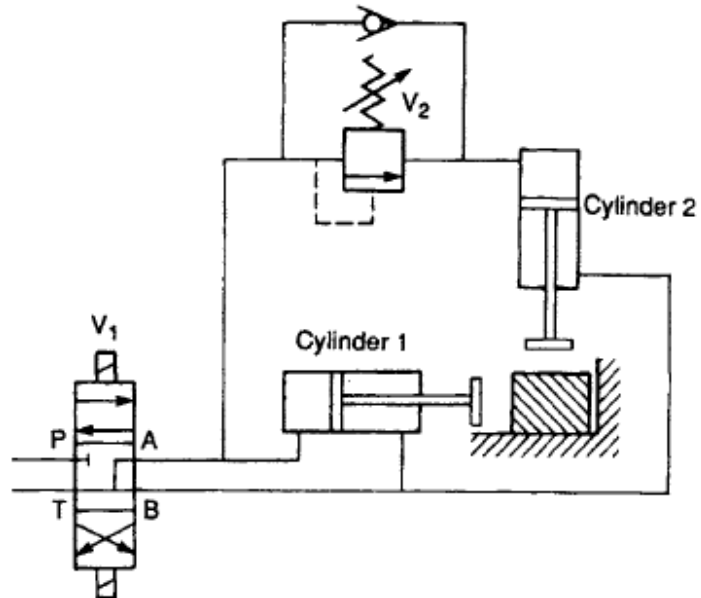


Fig Fast exhaust valve



SEQUENCE VALVES

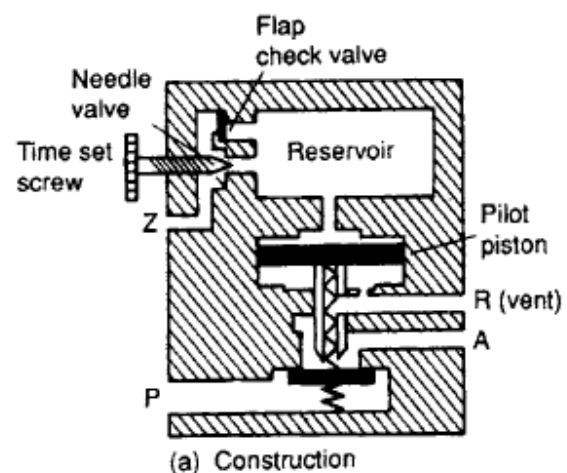
The sequence valve is a close relative of the pressure relief valve and is used where a set of operations are to be controlled in a pressure related sequence. Figure 4.27 shows a typical example where a workpiece is pushed into position by cylinder 1 and clamped by cylinder 2. Sequence valve V 2 is connected to the extend line of cylinder 1. When this cylinder is moving the workpiece, the line pressure is low, but rises once the workpiece hits the end stop. The sequence valve opens once its inlet pressure rises above a preset level. Cylinder 2 then operates to clamp the workpiece. A check valve across V 2 allows both cylinders to retract together.



TIME DELAY VALVES

Pneumatic time delay valves are used to delay operations where time-based sequences are required. Fig shows construction of a typical valve. This is similar in construction to a 3/2 way pilot-operated valve, but the space above the main valve is comparatively large and pilot air is only allowed in via a flowreducing needle valve. There is thus a time delay between application of pilot pressure to port Z and the valve operation, as shown by the timing diagram in Figure 4.28b. The time delay is adjusted by the needle valve setting. The built-in check valve causes the reservoir space above the valve to vent quickly when pressure at Z is removed to give no delay off.

The valve shown in Fig is a normally-closed delay-on valve. Many other time delay valves (delay-off, delay on/off, normally- open) can be obtained. All use the basic principle of the air reservoir and needle valve. The symbol of a normally-dosed time delay valve is in Fig



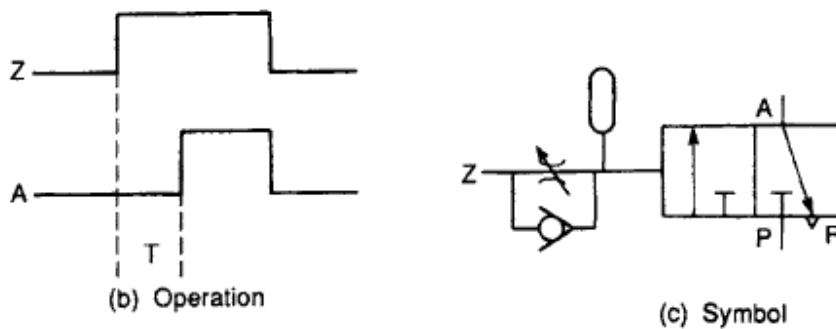


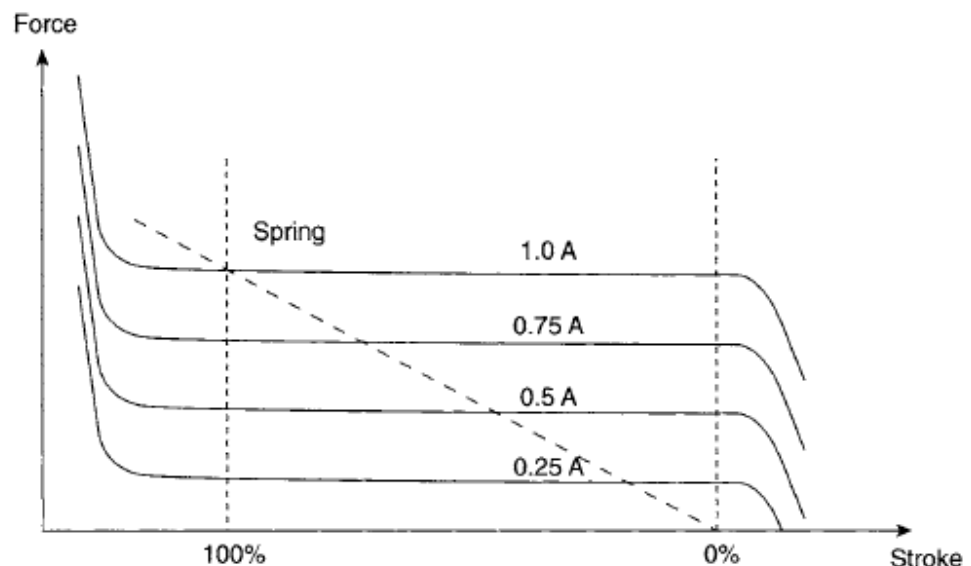
Fig Pneumatic time delay valve

PROPORTIONAL VALVES

The solenoid valves described so far act, to some extent, like an electrical switch, i.e. they can be On or Off. In many applications it is required to remotely control speed, pressure or force via an electrical signal. This function is provided by proportional valves. A typical two position solenoid is only required to move the spool between 0 and 100% stroke against the restoring force of a spring. To ensure predictable movement between the end positions the solenoid must also increase its force as the spool moves to ensure the solenoid force is larger than the increasing opposing spring force at all positions.

A proportional valve has a different design requirement.

The spool position can be set anywhere between 0% and 100% stroke by varying the solenoid current. To give a predictable response the solenoid must produce a force which is



dependent solely on the The relationship between coil current force and stroke for a proportional valve solenoid. Note the flat part of the curve and the linear relationship between current and force current and not on the spool position, i.e. the force for a given current must be constant over the full stroke range. Furthermore, the force must be proportional to the current. Fig shows a typical response. The force from the solenoid is opposed by the force from a restoring spring, and the spool will move to a position where the two forces are equal. With a current of 0.75 A, for example, the spool will move to 75% of its stroke. The spool movement in a proportional

valve is small; a few mm stroke is typical. The valves are therefore very vulnerable to stiction, and this is reduced by using a 'wet' design which immerses the solenoid and its core in hydraulic fluid.

A proportional valve should produce a fluid flow which is proportional to the spool displacement. The spools therefore use four triangular metering notches in the spool lands as shown on Fig. As the spool is moved to the right, port A will progressively link to the tank and port B to the pressure line. The symbol for this valve is also shown. Proportional valves are drawn with parallel lines on the connection sides of the valve block on circuit diagrams. Fig gives equal flow rates to both A and B ports. Cylinders have different areas on the full bore and annulus sides

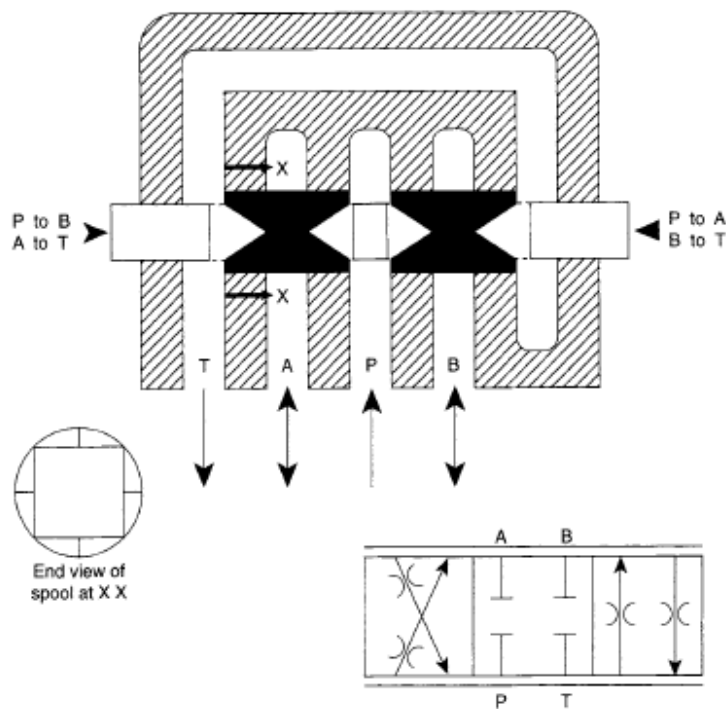


Fig Construction and symbol for a proportional valve. When used with a cylinder with 2:1 full bore to annulus area ratio, half the V cutouts will be provided on one of the P lands

To achieve equal speeds in both directions, the notches on the lands must have different areas. With a 2:1 cylinder ratio, half the number of notches are used on one side. Figure 4.31 shows the construction and symbol for a restricted centre position valve. Here the extended notches provide a restricted (typically 3%) flow to tank from the A and B ports when the valve is in the centre position. So far we have assumed the spool position is determined by the balance between the force from the solenoid and the restoring force from a spring. Whilst this will work for simple applications, factors

such as hydraulic pressure on the spool and spring ageing mean the repeatability is poor. Direct solenoid/spring balance is also not feasible with a pilot/main spool valve. What is really required is some method of position control of the spool. To achieve this, the spool position must be measured. Most valves use a device called a Linear Variable Differential Transformer (or LVDT) shown on Fig. The LVDT consists of a soft iron core whose position is to be measured surrounded by three electrical windings. A high frequency (typically a few kHz) AC signal is applied to the centre winding which induces voltages into the other two windings. When the core

is central, V 1 and V 2 are equal but opposite in phase giving zero volts at V 3.

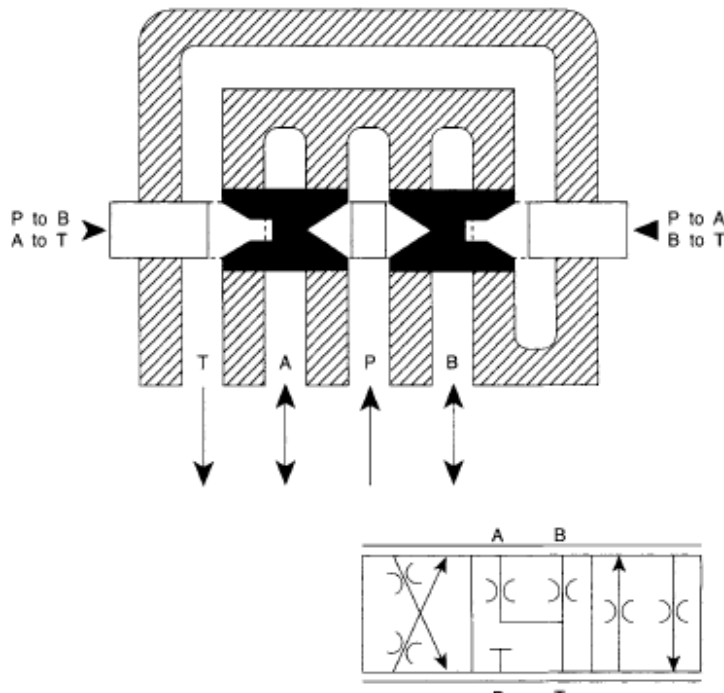
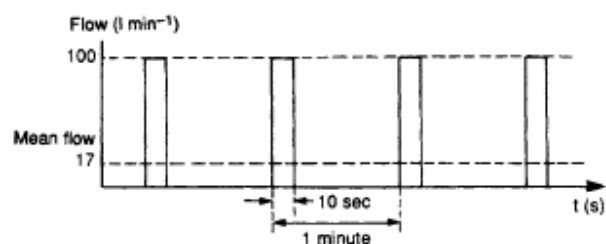


FIG Construction and symbol for a proportional valve with A and B ports linked to tank in the null position

HYDRAULIC ACCUMULATORS

In a simple hydraulic system, the pump size (delivery rate and hence motor power) is determined by the maximum requirements of the actuators. In Fig a system operates intermittently at a

Fig a simple system with uneven demands. To supply this without an accumulator a 100 l min⁻¹ is required although the mean flow is only 17 l/min



pressure of between 150 and 200

bar, needing a flow rate of 100 l min⁻¹ for 10 s at a repetition rate of 1 minute. With a simple system (pump, pressure regulator and loading valve) this requires a 200 bar, 100 l min⁻¹ pump (driven by about a 50 hp motor) which spends around 85% of its time unloading to tank. In Figure 6.3a a storage device called an accumulator has been added to the system. This can store, and release, a quantity of fluid at the required system pressure. In many respects it resembles the operation of a capacitor in an electronic power supply. The operation is shown in Figure 6.3b. At time A the system is turned on, and the pump loads causing pressure to rise as the fluid is delivered to the accumulator via the non-return valve V 3. At time B, working pressure is reached and a pressure switch on the accumulator causes the pump to unload. This state is maintained as non return valve V 3 holds the system pressure. The actuator operates between

times C and D. This draws fluid from the accumulator causing a fall of system pressure. The pressure switch on the accumulator puts the pump on load again but it takes until time E before the accumulator is charged ready for the next actuator movement at time E. An accumulator reduces pump requirements. The original system required a 100 l min^{-1} pump. With an accumulator, however, a pump only needs to provide 17 l min^{-1} (that is, 100 l min^{-1} for 10 secs every minute). Pump size, and hence motor size, have been reduced by a factor of six with obvious cost and space savings, plus gains in ancillary equipment such as motor starters and cabling. There is no gain in the energy used; with the simple system a 50 hp motor loads for 17% of the time, with an accumulator a 10 hp motor loads for about 90% of the time.

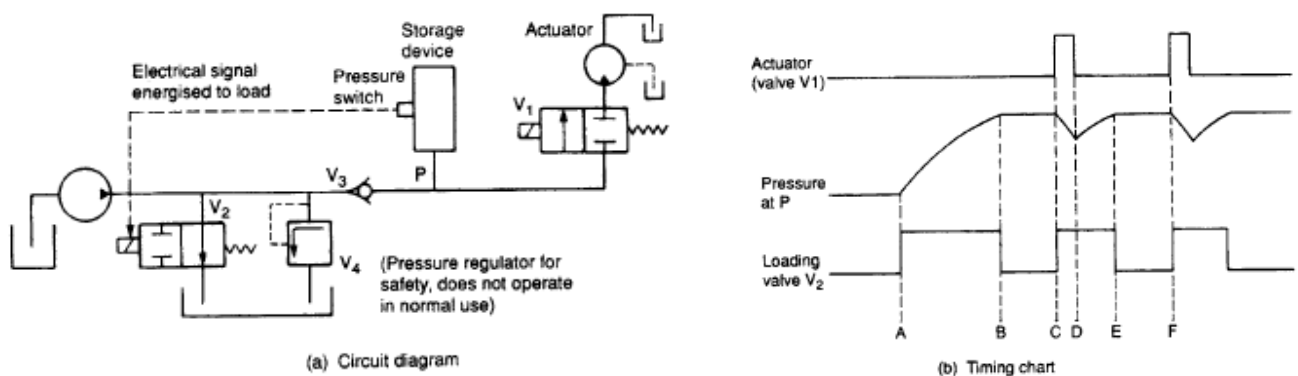


Fig System with an accumulator

Most accumulators operate by compressing a gas (although older and smaller accumulators may work by compressing a spring or lifting a weight with a cylinder). The most common form is the gasfilled bladder accumulator shown in Fig. Gas is precharged to some pressure with the accumulator empty of fluid when the whole of the accumulator is filled with gas. A poppet valve at the accumulator base prevents the bladder extruding out into the piping. Accumulators are sized by Boyle's law and a knowledge of the demands of the actuators. For the example system of Fig.

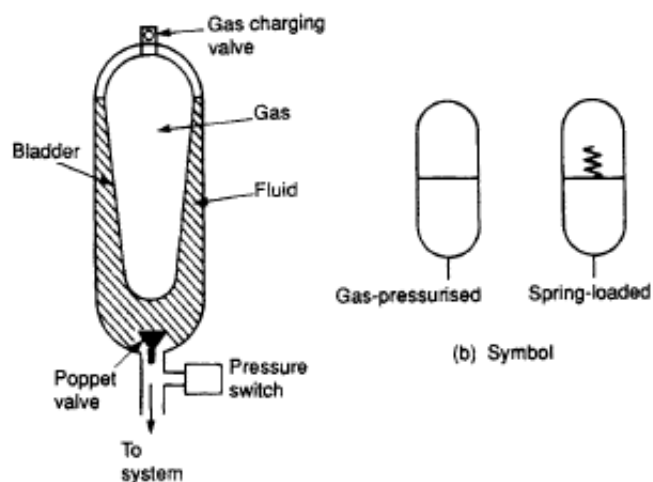


Fig The accumulator

Assuming a precharge of 120 bar, a charged accumulator pressure of 180 bar and a fall to a pressure to 160 bar during the removal of 17 liters of fluid: let V be volume of accumulator. This gives us the three states illustrated in Figure 6.5 to which Boyle's law can be applied to find the required accumulator volume. From Fig b and c using Boyle's law:

Which reduces to: $160v = 180(v - 17)$ $v = 153$ litres

From Fig a: $120V = 160 \times 153$ or $V = 204$ litres

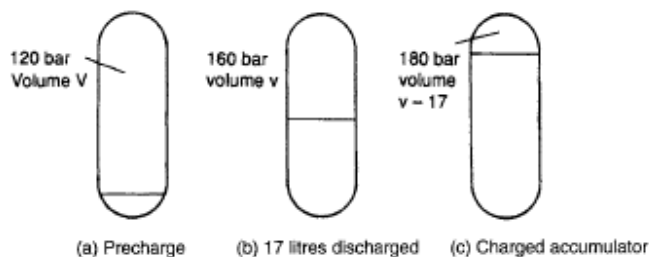


Fig Sizing an accumulator

Hence an accumulator of around 250 litres is required, with a precharge of 120 bar and a pressure switch set at 180 bar. Accumulators can also be used to act as 'buffers' on a system to absorb shocks and snub pressure spikes. Again the accumulator acts in similar manner to a capacitor in an electronic circuit. An accumulator, however, brings an additional danger into the system, as it is possible for high pressures to exist in the circuit even though the pump has been stopped. If a coupling is opened under these circumstances the accumulator discharges all its fluid at working pressure. The author speaks from personal experience of having committed this cardinal sin and being covered in oil for his mistake! Extreme care should therefore be taken when working on circuits with accumulators. Normally a manual or automatic blowdown valve is included to allow the accumulator pressure to be released. The pressure gauge should be observed during blowdown and no work undertaken until it is certain all pressure has been released. Fig shows typical blowdown circuits.

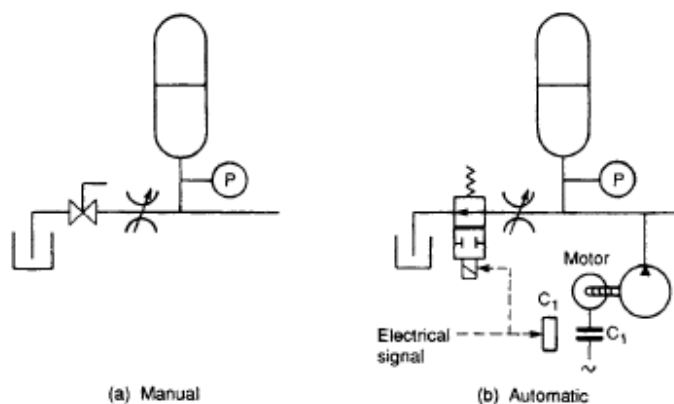


Fig Accumulator blowdown circuits. In each case flow from the accumulator is restricted to prevent an explosive decompression

Once a system has warmed up, a quick check can be made on the state of an accumulator with the flat of the hand. There should always be a significant temperature difference between the gas

and the hydraulic oil and the oil/gas split can be detected by the temperature change on the body of the accumulator. If the whole body is the same temperature something has gone severely wrong with the gas bladder. An accumulator is a pressurised vessel and as such requires certification if it contains more than 250 bar.litres. It will require a recorded expert visual inspection every five years and a full volumetric pressure test every ten years.

HYDRAULIC COOLERS AND HEAT EXCHANGERS

Despite the occasional use of heaters mentioned earlier, the problem with oil temperature is usually keeping it *down* to the required 50~ In small systems, the heat lost through reservoir walls is sufficient to keep the oil cool, but in larger systems additional cooling is needed. Table 6.1 shows typical heat losses from various sizes of reservoirs. It should be noted that the relationship between volume and heat loss (surface area) is non-linear, because surface area increases as the square of the linear dimensions, whereas volume increases as the cube.

Table Heat loss for various tank volumes. These are only approximate as few tanks are pure cubes

Vol(l)	L(m)	Surface area (m ²)	Heat loss (kW)
250	0.63	1.98	0.5
500	0.8	3.2	1.0
1,000	1.0	5.0	1.5
2,000	1.25	7.8	2.5
10,000	2.15	23.1	15.0

UNIT IV PNEUMATIC SYSTEMS AND COMPONENTS

COMPRESSOR TYPES

Like hydraulic pumps, air compressors can be split into positive displacement devices (where a fixed volume of air is delivered on each rotation of the compressor shaft) and dynamic devices such as centrifugal or axial blowers. The vast majority of air compressors are of the positive displacement type. A compressor is selected by the pressure it is required to work at and the volume of gas it is required to deliver. As explained in the previous section, pressure in the receiver is generally higher than that required at the operating position, with local pressure regulation being used. Pressure at the compressor outlet (which for practical purposes will be the same as that in the receiver) is called the working pressure and is used to specify the compressor. Pressure at the operating point is called, not surprisingly, the operating pressure and is used to specify valves, actuators and other operating devices. Care should be taken in specifying the volume of gas a compressor is required to deliver. Expression 3.1 shows the volume of a given mass of gas to be highly dependent on pressure and temperature. Delivery volume of a compressor is defined in terms of gas at normal atmospheric conditions. Two standards known as standard temperature and pressures (STP) are commonly used, although differences between them are small for industrial users.

The *technical normal condition* is:

$$P = 0.98 \text{ bar absolute, } T = 20^\circ\text{C}$$

and the *physical normal condition* is:

$$P = 1.01 \text{ bar absolute, } T = 0^\circ\text{C}$$

The term *normal temperature and pressure* (NTP) is also used. Required delivery volume of a compressor (in $\text{M}^3 \text{min}^{-1}$ or $\text{ft}^3 \text{min}^{-1}$, according to the units used) may be calculated for the actuators at the various operating positions (with healthy safety margins to allow for leakage) but care must be taken to ensure this total volume is converted to STP condition before specifying the required compressor delivery volume. A compressor delivery volume can be specified in terms of its theoretical volume (swept volume multiplied by rotational speed) or effective volume which includes losses. The ratio of these two volumes is the efficiency. Obviously the effective volume should be used in choosing a compressor (with, again, a safety margin for leakage). Required power of the motor driving the compressor is dependent on working pressure and delivery volume, and may be determined from expressions 2.2 and 2.5. Allowance must be made for the cyclic on/off operation of the compressor with the motor being sized for on load operation and not averaged over a period of time.

PISTON COMPRESSORS

Piston compressors are by far the most common type of compressor, and a basic single cylinder form is shown in Figure 3.3. As the piston descends during the inlet stroke (Figure 3.3a), the inlet valve opens and air is drawn into the cylinder. As the piston passes the bottom of the stroke, the inlet valve closes and the exhaust valve opens allowing air to be expelled as the piston rises (Figure 3.3b) Figure 3.3 implies that the valves are similar to valves in an internal combustion engine. In practice, spring-loaded valves are used, which open and close under the action of air pressure across them. One common type uses a 'feather' of spring steel which moves above the inlet or output port, as shown in Fig

A single cylinder compressor gives significant pressure pulses at the outlet port. This can be overcome to some extent by the use of a large receiver, but more often a multi cylinder compressor is used. These are usually classified as vertical or

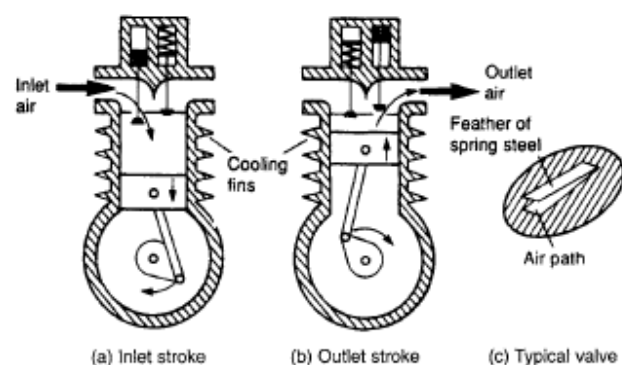


Fig Single cylinder compressor

horizontal in-line arrangements and the more compact V, Y or W constructions. A compressor which produces one pulse of air per piston stroke (of which the example of Figure 3.3 is typical) is called a singleacting compressor. A more even air supply can be obtained by the double acting action of the compressor in Figure 3.4, which uses two sets of valves and a crosshead to keep the piston rod square at all times. Double-acting compressors can be found in all configurations described earlier.

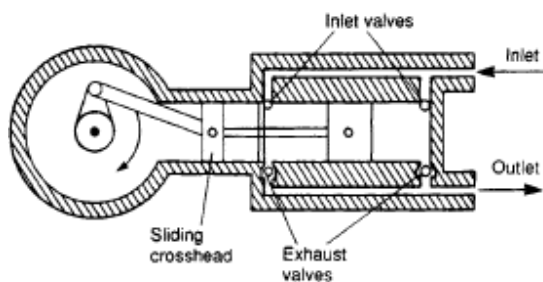


Fig Double-acting compressor

Piston compressors described so far go direct from atmospheric to required pressure in a single operation. This is known as a single stage compressor. The general gas law (expression 1.19) showed compression of a gas to be accompanied by a significant rise in gas temperature. If the exit pressure is above about 5 bar in a singleacting compressor, the compressed air temperature can rise to over 200° and the motor power needed to drive the compressor rises accordingly. For pressures over a few bar it is far more economical to use a multistage compressor with cooling between stages. Fig shows an example. As cooling (undertaken by a device called an intercooler) reduces the volume of the gas to be compressed at the second stage there is a large energy saving. Normally two stages are used for pneumatic pressures of 10 to 15 bar, but multistage compressors are available for pressures up to around 50 bar. Multistage compressors can be manufactured with multicylinders as shown in Fig or, more compactly, with a single cylinder and a double diameter piston as shown in Fig There is contact between pistons and air, in standard piston compressors, which may introduce small amounts of lubrication oil

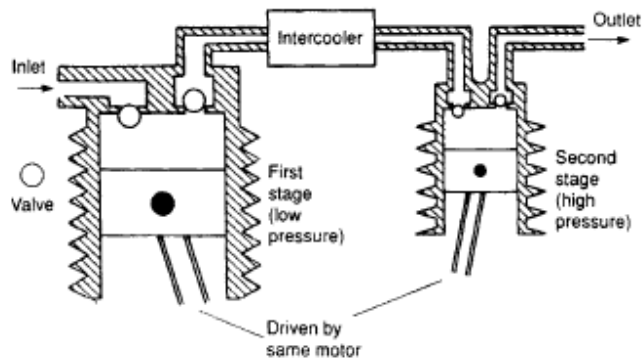


Fig Two-stage compressor

from the piston walls into the air. This very small contamination may be undesirable in food and chemical industries. Fig shows a common way of giving a totally clean supply by incorporating a flexible diaphragm between piston and air.

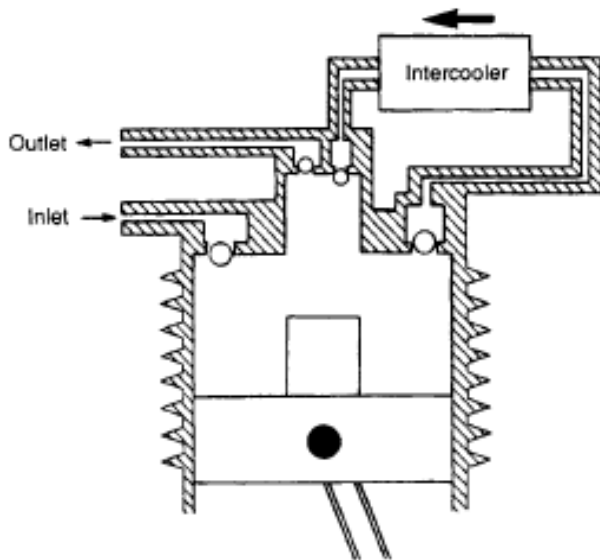


Fig Combined two-stage compressor

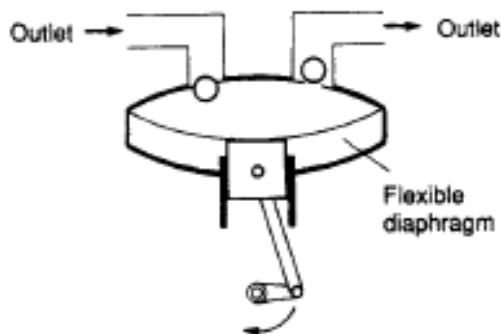


Fig Diaphragm compressor, used where air must not be contaminated

SCREW COMPRESSORS

Piston compressors are used where high pressures (> 20 bar) and relatively low volumes ($< 10,000 \text{ m}^3 \text{ hr}^{-1}$) are needed, but are mechanically relatively complex with many moving parts. Many applications require only medium pressure (< 10 bar) and medium flows (around $10,000 \text{ m}^3 \text{ hr}^{-1}$). For these applications, rotary compressors have the advantage of simplicity, with fewer moving parts rotating at a constant speed, and a steady delivery of air without

pressure pulses. One rotary compressor, known as the dry rotary screw compressor, is shown in Fig and consists of two intermeshing rotating screws with minimal (around 0.05 mm) clearance. As the screws rotate, air is drawn into the housing, trapped between the screws and carried along to the discharge port, where it is delivered in a constant pulse-free stream. Screws in this compressor can be synchronized by external timing gears. Alternatively one screw can be driven, the second screw rotated by contact with the drive screw. This approach requires oil lubrication to be sprayed into the inlet air to reduce friction between screws, and is consequently known as a wet rotary screw compressor. Wet screw construction though, obviously introduces oil contamination into the air which has to be removed by later oil separation units.

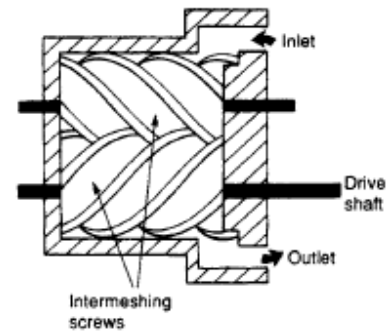


Fig Dry screw rotary compressor

ROTARY COMPRESSORS

The vane compressor, shown in Fig operates on similar principles to the hydraulic vane pump described in Chapter 2, although air compressors tend to be physically larger than hydraulic pumps. An unbalanced design is shown, balanced versions can also be constructed. Vanes can be forced out by springs or, more commonly, by centrifugal force. A single stage vane compressor can deliver air at up to 3 bar, a much lower pressure than that available with a screw or piston compressor. A two-stage vane compressor with large low pressure and smaller high pressure sections linked by an intercooler allows pressures up to 10 bar to be obtained.

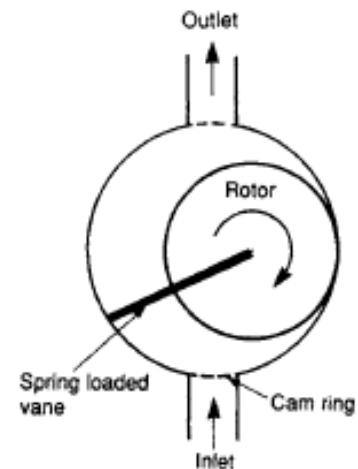


Fig Vane compressor

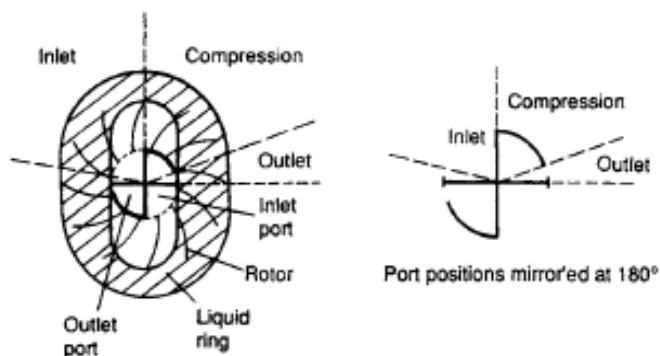


Fig Liquid ring compressor

Fig shows a variation on the vane compressor called a liquid ring compressor. The device uses many vanes rotating inside an eccentric housing and contains a liquid (usually water) which is flung out by centrifugal force to form a liquid ring which follows the contour of the housing to give a seal with no leakage and minimal friction. Rotational speed must be high (typically 3000 rpm) to create the ring. Delivery pressures are relatively low at around 5 bar. The lobe compressor of Figure 3.11 (often called a Roots blower) is often used when a positive displacement compressor is needed with high delivery volume but low pressure (typically 1-2 bar). Operating pressure is mainly limited by leakage between rotors and housing. To operate efficiently, clearances must be very small, and wear leads to a rapid fall in efficiency.

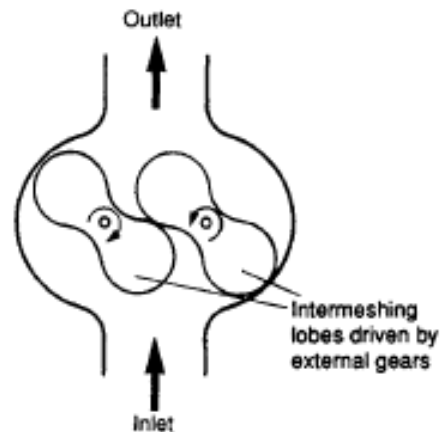


Fig Lobe compressor

DYNAMIC COMPRESSORS

A large volume of air (up to $5000 \text{ m}^3 \text{ min}^{-1}$) is often required for applications such as pneumatic conveying (where powder is carried in an air stream), ventilation or where air itself is one component of a process (e.g. combustion air for gas/oil burners). Pressure in these applications is low (at most a few bar) and there is no need for a positive displacement compressor. Large volume low pressure air is generally provided by dynamic compressors known as blowers. They can be subdivided into centrifugal or axial types, shown in Figure 3.12. Centrifugal blowers (Figure 3.12a) draw air in then fling it out by centrifugal force. A high shaft rotational speed is needed and the volume to input power ratio is lower than any other type of compressor. An axial compressor comprises a set of rotating fan blades as shown in Figure 3.12b. These produce very large volumes of air, but at low pressure (less than one bar). They are primarily used for ventilation, combustion and process air. Output pressures of both types of dynamic compressor can be lifted by multistage compressors with intercoolers between stages. Diffuser sections reduce air entry velocity to subsequent stages, thereby converting air kinetic energy to pressure energy.

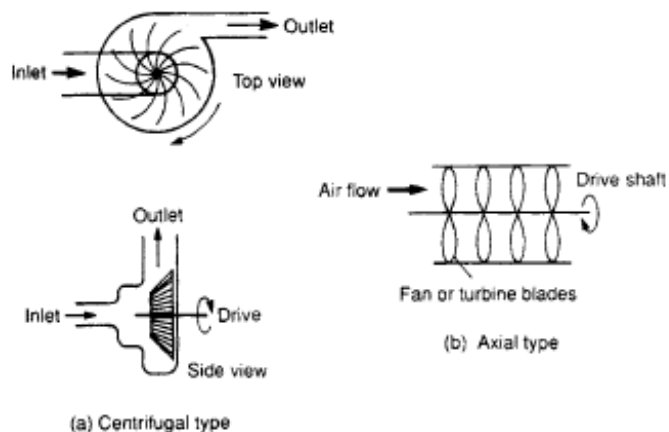


Fig Non-positive displacement compressors (Blowers)

AIR RECEIVERS AND COMPRESSOR CONTROL

An air receiver is used to store high pressure air from the compressor. Its volume reduces pressure fluctuations arising from changes in load and from compressor switching. Air coming from the compressor will be warm (if not actually hot!) and the large surface area of the receiver dissipates this heat to the surrounding atmosphere. Any moisture left in the air from the compressor will condense out in the receiver, so outgoing air should be taken from the receiver top. Fig shows essential features of a receiver.

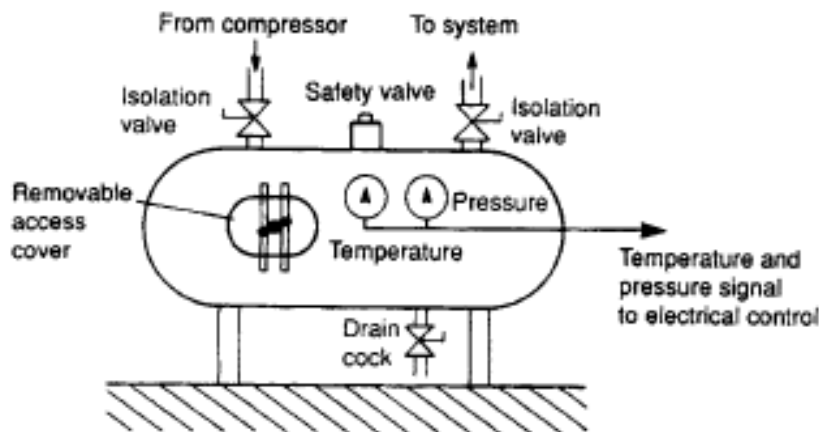


Fig Compressed air receiver

They are usually of cylindrical construction for strength, and have a safety relief valve to guard against high pressures arising from failure of the pressure control scheme. Pressure indication and, usually, temperature indication are provided, with pressure switches for control of pressure and high temperature switches for remote alarms. A drain cock allows removal of condensed water, and access via a manhole allows cleaning. Obviously, removal of a manhole cover is hazardous with a pressurised receiver, and safety routines must be defined and followed to prevent accidents. Control of the compressor is necessary to maintain pressure in the receiver. The simplest method of achieving this is to start the compressor when receiver pressure falls to some minimum pressure, and stop the compressor when pressure rises to a satisfactory level again, as illustrated in Fig. In theory two pressure switches are required (with the motor start pressure lower than the motor stop pressure) but, in practice, internal hysteresis in a typical switch allows one pressure switch to be used. The pressure in the receiver cycles between the start and stop pressure settings.

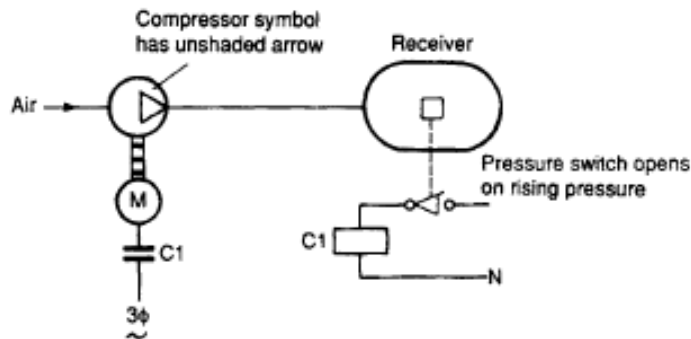


Fig Receiver pressure control via motor start/stop

Compressors can also be controlled on the inlet side. In the example of Fig an inlet valve is held open to allow the compressor to operate, and is closed when the air receiver has reached the desired pressure, (the compressor then forms a near vacuum on its inlet side). The valves in Figures can be electrically-operated solenoid valves controlled by pressure switches, or can be pneumatic valves controlled directly by receiver pressure. The control method is largely determined by flow rates from receiver to the load(s) and the capacity of the compressor. If the compressor has significant spare capacity, for example, start/stop control is commonly used. If compressor capacity and load requirements are closely matched, on the other hand, short start/stop cycling may cause premature wear in the electrical starter for the compressor motor. In this situation, exhaust or inlet regulation is preferred. Air receiver size is determined by load requirements, compressor capacity, and allowable pressure deviations in the receiver. With the compressor stopped, Boyle's law (expression 1.17) gives the pressure decay for a given volume of air delivered from a given receiver at a known pressure. For example, if a receiver of 10 cubic metres volume and a working pressure of 8 bar delivers 25 cubic metres of air (at STP) to a load, pressure in the receiver falls to approximately 5.5 bar. With the compressor started, air pressure rises at a rate again given by expression 1.17 (with the air mass in the receiver being *increased* by the difference between the air delivered by the compressor and that removed by the load). These two calculations give the cycle time of the compressor when combined with settings of the cut-in and drop-out pressure switches. If this is unacceptably rapid, say less than a few minutes, then a larger receiver is required. Manufacturers of pneumatic equipment provide nomographs which simplify these calculations. An air receiver is a pressure vessel and as such requires regular visual and volumetric pressure tests. Records should be kept of the tests.