5

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 is given by πR^2. For a pressure P applied to port X, the extend force available is:

$$F_c = P \pi R^2. \tag{5.1}$$

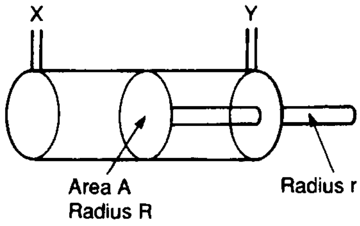


Figure 5.1 *A simple cylinder*

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.

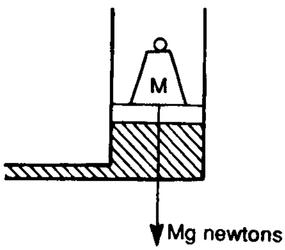


Figure 5.2 *A mass supported by a cylinder*

When pressure is applied to port Y, the piston retracts. Total piston area here is reduced because of the rod, giving an annulus of area A_a where:

$$A_a = A - \pi r^2$$

and r is the radius of the rod. The maximum retract force is thus:

$$F_r = P\,A_a = P(A - \pi r^2). \tag{5.2}$$

This is lower than the maximum extend force. In Figure 5.3 identical pressure is applied to both sides of a piston. This produces an

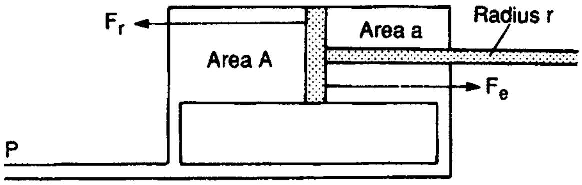


Figure 5.3 *Pressure applied to both sides of piston*

extend force F_c given by expression 5.1, and a retract force F_r given by expression 5.2. Because F_c is greater than F_r, the cylinder extends.

Normally the ratio A/A_a is about 6/5. In the cylinder shown in Figure 5.4, the ratio A/A_a of 2:1 is given by a large diameter rod. This can be used to give an equal extend and retract force when connected as shown. (The servo valve of Figure 4.40 also uses this principle.)

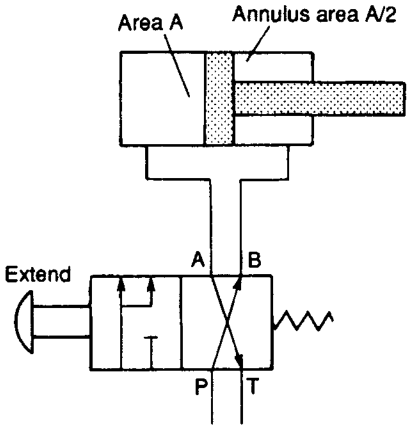


Figure 5.4 *Cylinder with equal extend/retract force*

Cylinders shown so far are known as double-acting, because fluid pressure is used to extend and retract the piston. In some applications a high extend force is required (to clamp or form an object) but the retract force is minimal. In these cases a single-acting cylinder (Figure 5.5) can be used, which is extended by fluid but retracted by a spring. If a cylinder is used to lift a load, the load itself can retract the piston.

Single-acting cylinders are simple to drive (particularly for pneumatic cylinders with quick exhaust valves (see Chapter 4)) but the

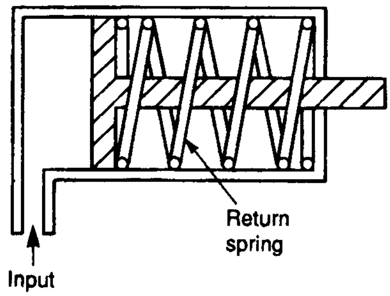


Figure 5.5 *Single-acting cylinder*

extend force is reduced and, for spring-return cylinders, the Figure length of the cylinder is increased for a given stroke to accommodate the spring.

A double rod cylinder is shown in Figure 5.6a. This has equal fluid areas on both sides of the piston, and hence can give equal forces in both directions. If connected as shown in Figure 5.3 the piston does not move (but it can be shifted by an outside force). Double rod cylinders are commonly used in applications similar to Figure 5.6b where a dog is moved by a double rod cylinder acting via a chain.

The speed of a cylinder is determined by volume of fluid delivered to it. In the cylinder in Figure 5.7 the piston, of area A, has moved a distance d. This has required a volume V of fluid where:

$$V = A\,d \tag{5.3}$$

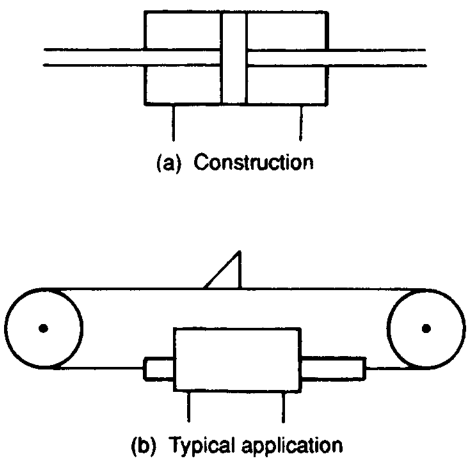


Figure 5.6 *Double rod cylinder (with equal extend/retract force)*

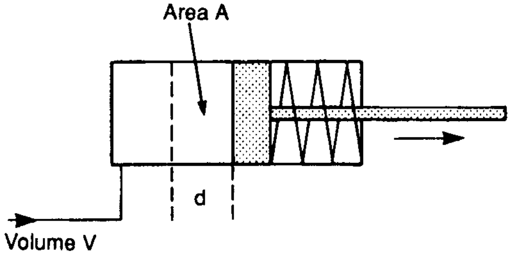


Figure 5.7 *Derivation of cylinder speed*

If the piston moves at speed v, it moves distance d in time t where:

$$t = d/v$$

Flow rate, V_f, to achieve speed v is thus:

$$V_f = \frac{A\,d}{t} = A\,v \qquad (5.4)$$

The flow rate units of expression 5.4 depend on the units being used. If d is in metres, v in metres min^{-1} and A in $metres^2$, flow rate is in $metres^3\ min^{-1}$.

In pneumatic systems, it should be remembered, it is normal to express flow rates in STP (see Chapter 3). Expression 5.4 gives the fluid volumetric flow rate to achieve a required speed at working pressure. This must be normalised to atmospheric pressure by using Boyle's law (given in expression 1.17).

The air consumption for a pneumatic cylinder must also be normalised to STP. For a cylinder of stroke S and piston area A, normalised air consumption is:

$$\text{Volume/stroke} = S\,A\frac{(P_a + P_w)}{P_a} \qquad (5.5)$$

where P_a is atmospheric pressure and P_w the working pressure. The repetition rate (e.g. 5 strokes min^{-1}) must be specified to allow mean air consumption rate to be calculated.

It should be noted that fluid pressure has no effect on piston speed (although it does influence acceleration). Speed is determined by piston area and flow rate. Maximum force available is unrelated to flow rate, instead being determined by line pressure and piston area. Doubling the piston area while keeping flow rate and line pressure constant, for example, gives half speed but doubles the maximum force. Ways in which flow rate can be controlled are discussed later.

Construction

Pneumatic and hydraulic linear actuators are constructed in a similar manner, the major differences arising out of differences in operating pressure (typically 100 bar for hydraulics and 10 bar for pneumatics, but there are considerable deviations from these values).

Figure 5.8 shows the construction of a double-acting cylinder. Five locations can be seen where seals are required to prevent leakage. To some extent, the art of cylinder design is in choice of seals, a topic discussed further in a later section.

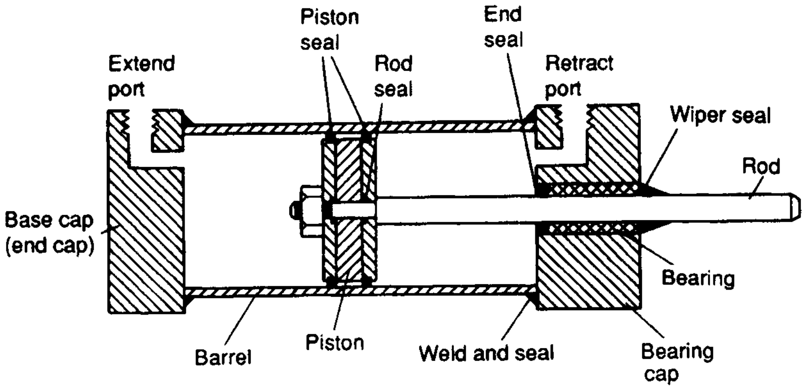
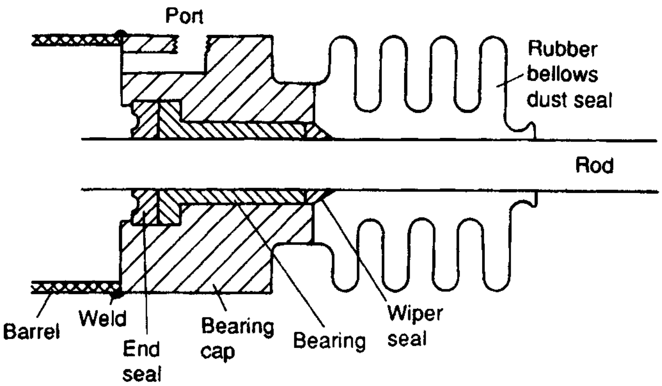


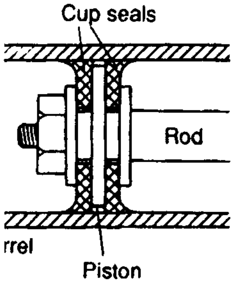
Figure 5.8 *Construction of a typical cylinder*

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.

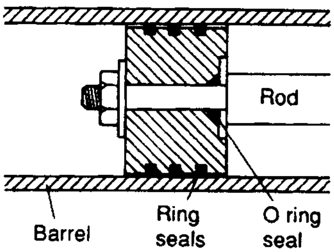
The inner surface of the barrel needs to be very smooth to prevent wear and leakage. Generally a seamless drawn steel tube is used which is machined (honed) to an accurate finish. In applications



(a) Enlarged view of bearing cap



(b) Cup seals



(c) Ring seals and O ring

Figure 5.9 *Cylinder constructional details*

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 barrel to cause problems 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.

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

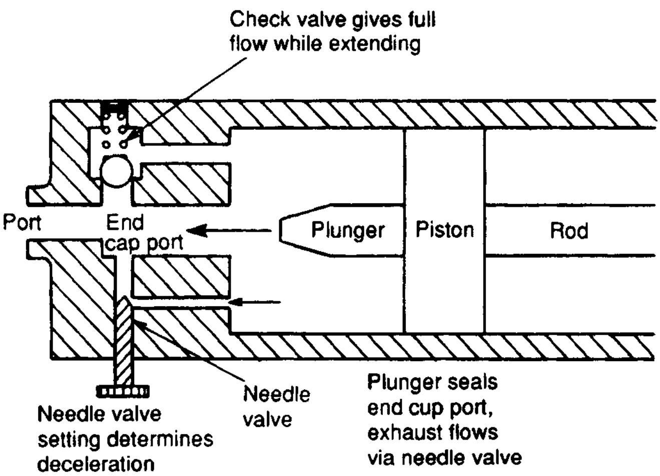


Figure 5.10 *Cylinder cushioning*

enters the cap. The exhaust flow route is now via the deceleration valve which reduces the speed and the end of travel impact. The deceleration valve is adjustable to allow the deceleration rate to be set. A check valve is also included in the end cap to bypass the deceleration valve and give near full flow as the cylinder extends. Cushioning in Figure 5.10 is shown in the base cap, but obviously a similar arrangement can be incorporated in bearing cap as well.

Cylinders are very vulnerable to side loads, particularly when fully extended. In Figure 5.11a a cylinder with a 30 cm stroke is fully extended and subject to a 5 kg side load. When extended there is typically 1 cm between piston and end bearing. Simple leverage will give side loads of 155 kg on the bearing and 150 kg on the piston seals. This magnification of side loading increases cylinder wear. The effect can be reduced by using a cylinder with a longer stroke, which is then restricted by an internal stop tube as shown in Figure 5. 11b.

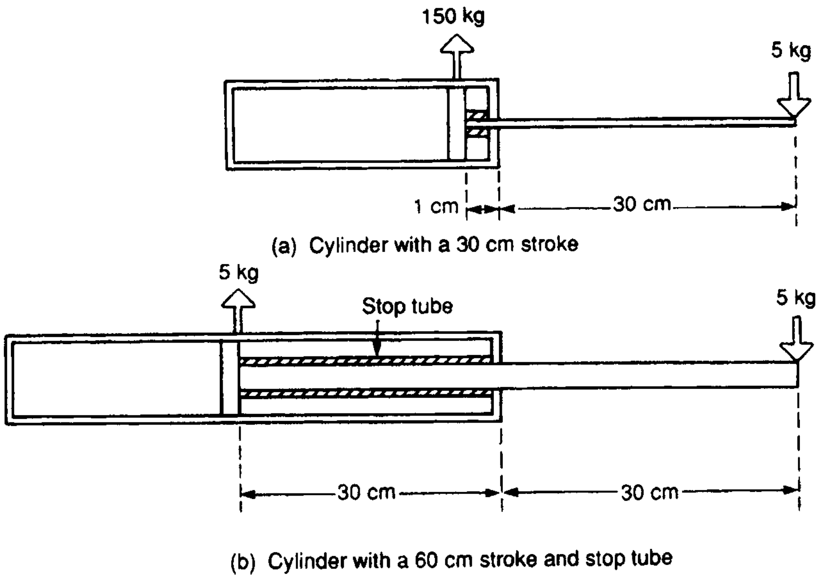


Figure 5.11 *Side loads and the stop tube*

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

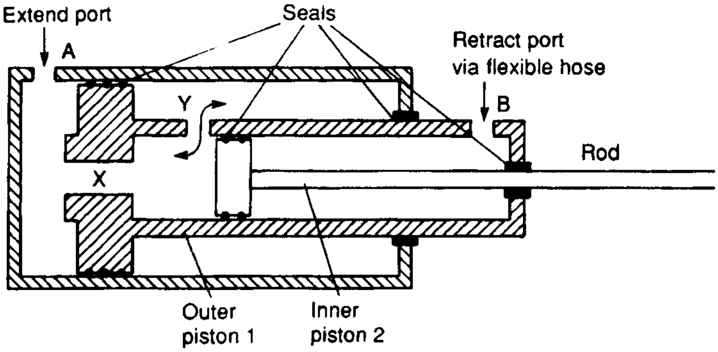


Figure 5.12 *Two-stage telescopic piston*

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

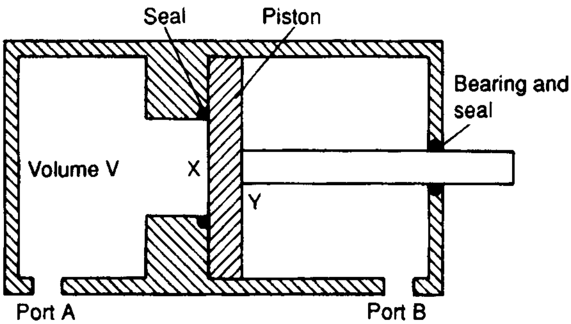
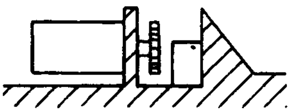


Figure 5.13 *An impact cylinder*

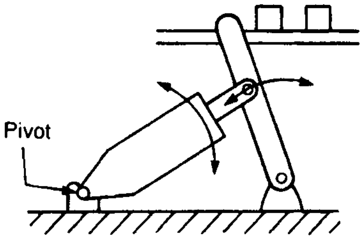
B to retract the cylinder. Pressure is then applied to both ports A and B, but the cylinder remains in a retracted state because area X is less than area Y. Port B is then vented rapidly. Immediately, the full piston area experiences port A pressure. With a large volume of gas stored behind the piston,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

Mounting arrangements

Cylinder mounting is determined by the application. Two basic types are shown in Figure 5.14. The clamp of Figure 5.14a requires a simple fixed mounting. The pusher of Figure 5.15b requires a cylinder mount which can pivot.



(a) Clamp using front flange mounting



(b) Pusher using rear trunnion mounting

Figure 5.14 *Basic mounting types*

Figure 5.15 shows various mounting methods using these two basic types. The effects of side loads should be considered on non-centreline mountings such as the foot mount. Swivel mounting obviously requires flexible pipes.

Cylinder dynamics

The cylinder in Figure 5.16a is used to lift a load of mass M. Assume it is retracted, and the top portion of the cylinder is pressurised. The extending force is given by the expression:

$$F = P_1A - P_2a. \quad (5.6)$$

To lift the load at all, F>Mg+f where M is the mass and f the static frictional force.

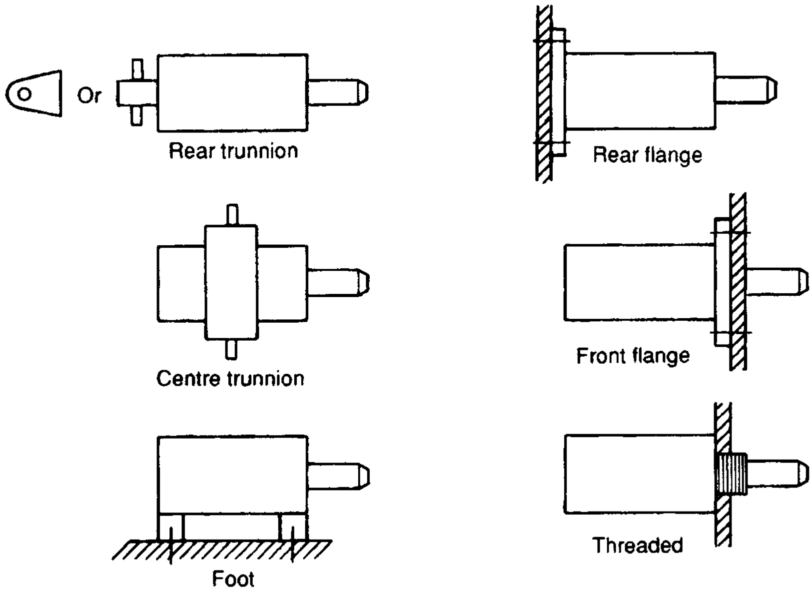


Figure 5.15 *Methods of cylinder mounting*

The response of this simple system is shown in Figure 5.16b. At time W the rod side of the cylinder is vented and pressure is applied to the other side of the piston. The pressure on both sides of the piston changes exponentially, with falling pressure P_2 changing slower than inlet pressure P_1, because of the larger volume. At time X, extension force P_1A is larger than P_2a, but movement does not start until time Y when force, given by expression 5.6, exceeds mass and frictional force.

The load now accelerates with acceleration given by Newton's law:

$$\text{acceleration} = \frac{F_a}{M} \ . \tag{5.7}$$

Where $F_a = P_1A - P_2a - Mg - f$.

It should be remembered that F_a is not constant, because both P_1 and P_2 will be changing. Eventually the load will reach a steady velocity, at time Z. This velocity is determined by maximum input flow rate or maximum outlet flow rate (whichever is lowest). Outlet pressure P_2 is determined by back pressure from the outlet line to tank or atmosphere, and inlet pressure is given by the expression:

$$P_1 = \frac{Mg + f + P_2a}{A}$$

The time from W to Y, before the cylinder starts to move, is called the 'dead time' or 'response time'. It is determined primarily by the decay of pressure on the outlet side, and can be reduced by de-pressurising the outlet side in advance or (for pneumatic systems) by the use of quick exhaust valves (described in Chapter 4).

The acceleration is determined primarily by the inlet pressure and the area of the inlet side of the piston (term P_1A in expression 5.6). The area, however, interacts with the dead time – a larger area, say, gives increased acceleration but also increases cylinder volume and hence extends the time taken to vent fluid on the outlet side.

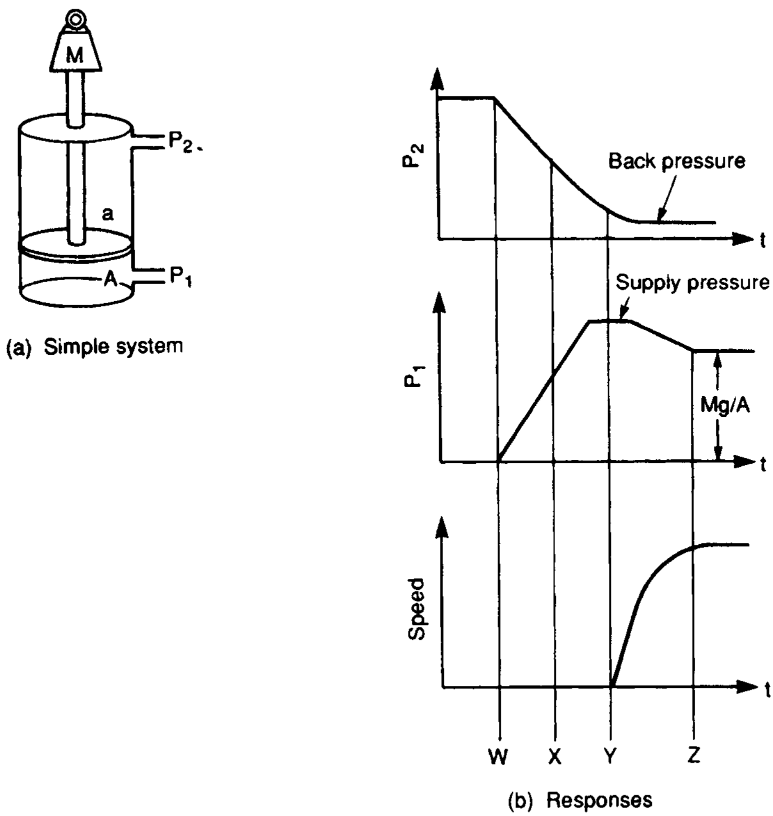


Figure 5.16 *Cylinder dynamics*

Seals

Leakage from a hydraulic or pneumatic system can be a major problem, leading to loss of efficiency, increased power usage, temperature rise, environmental damage and safety hazards.

Minor internal leakage (round the piston in a double-acting cylinder, for example) can be of little consequence and may even be deliberately introduced to provide lubrication of the moving parts.

External leakage, on the other hand, is always serious. In pneumatic systems, external leakage is noisy; with hydraulic systems, external loss of oil is expensive as lost oil has to be replaced, and the resulting pools of oil are dangerous and unsightly.

Mechanical components (such as pistons and cylinders) cannot be manufactured to sufficiently tight tolerances to prevent leakage (and even if they could, the resultant friction would be unacceptably high). Seals are therefore used to prevent leakage (or allow a controlled leakage). To a large extent, the art of designing an actuator is really the art of choosing the right seals.

The simplest seals are 'static seals' (Figure 5.17) used to seal between stationary parts. These are generally installed once and forgotten. A common example is the gasket shown in a typical application in Figure 5.17a. The O ring of Figure 5.17b is probably the most used static seal, and comprises a moulded synthetic ring with a round cross section when unloaded. O rings can be specified in terms of inside diameter (ID) for fitting onto shafts, or outside diameter (OD) for fitting into bores.

When installed, an O ring is compressed in one direction. Application of pressure causes the ring to be compressed at right angles, to give a positive seal against two annular surfaces and one flat surface. O rings give effective sealing at very high pressures.

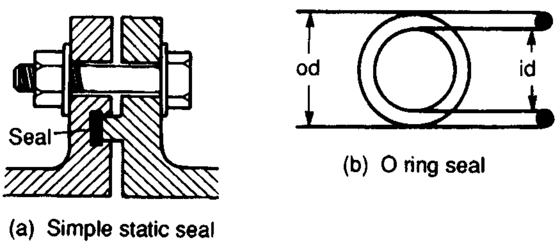


Figure 5.17 *Static seals*

O rings are primarily used as static seals because any movement will cause the seal to rotate allowing leakage to occur.

Where a seal has to be provided between moving surfaces, a dynamic seal is required. A typical example is the end or cup seal shown, earlier, in Figure 5.9a. Pressure in the cylinder holds the lip of the seal against the barrel to give zero leakage (called a 'positive seal'). Effectiveness of the seal increases with pressure, and leakage tends to be more of a problem at low pressures.

The U ring seal of Figure 5.18 works on the same principle as the cup seal. Fluid pressure forces the two lips apart to give a positive seal. Again, effectiveness of the seal is better at high pressure. Another variation on the technique is the composite seal of Figure 5.19. This is similar in construction to the U ring seal, but the space between the lips is filled by a separate ring. Application of pressure again forces the lips apart to give a positive seal.

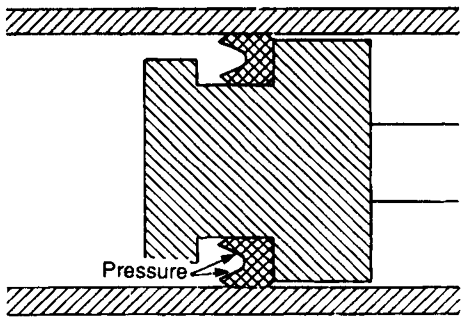


Figure 5.18 *The U ring seal*

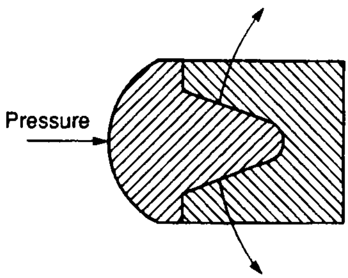


Figure 5.19 *The composite seal*

At high pressures there is a tendency for a dynamic seal to creep into the radial gap, as shown in Figure 5.20a leading to trapping of the seal and rapid wear. This can be avoided by the inclusion of an anti-extrusion ring behind the seal, as in Figure 5.20b.

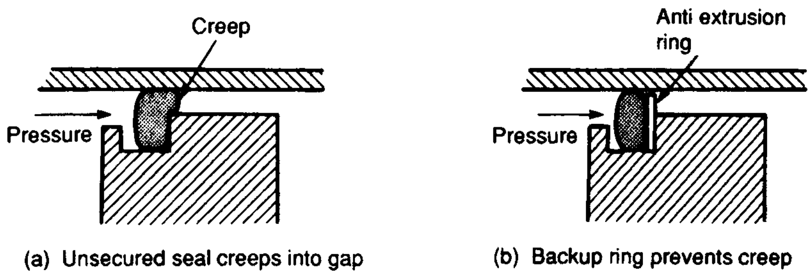


Figure 5.20 *Anti-extrusion ring*

Seals are manufactured from a variety of materials, the choice being determined by the fluid, its operating pressure and the likely temperature range. The earliest material was leather and, to a lesser extent, cork but these have been largely superseded by plastic and synthetic rubber materials. Natural rubber cannot be used in hydraulic systems as it tends to swell and perish in the presence of oil.

The earliest synthetic seal material was neoprene, but this has a limited temperature range (below 65°C). The most common present-day material is nitrile (buna-N) which has a wider temperature range (–50°C to 100°C) and is currently the cheapest seal material. Silicon has the highest temperature range (–100°C to +250°C) but is expensive and tends to tear.

In pneumatic systems viton (–20°C to 190°C) and teflon (–80°C to +200°C) are the most common materials. These are more rigid and are often used as wiper or scraper seals on cylinders.

Synthetic seals cannot be used in applications where a piston passes over a port orifice which nicks the seal edges. Here metallic ring seals must be used, often with the rings sitting on O rings, as illustrated in Figure 5.21.

Seals are delicate and must be installed with care. Dirt on shafts or barrels can easily nick a seal as it is slid into place. Such damage may not be visible to the eye but can cause serious leaks. Sharp edges can cause similar damage so it is usual for shaft ends and groove edges to be chamfered.

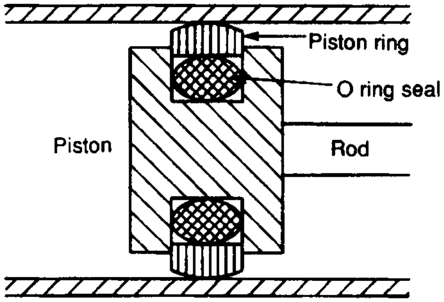


Figure 5.21 *Combined piston ring and O ring seal (not to scale)*

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:

$$T = Fd \text{ Nm.} \tag{5.8}$$

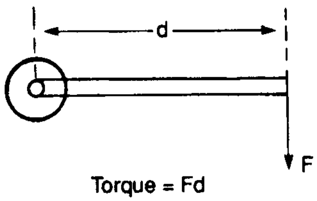


Figure 5.22 *Definition of torque*

In Imperial units, F is given in pounds force, and d in inches or feet to give T in lbf ins or lbf ft. It follows that 1 Nm = 8.85 lbf ins.

The torque of a rotary actuator can be specified in three ways. Starting torque is the torque available to move a load from rest. Stall torque must be applied by the load to bring a running actuator to rest, and running torque is the torque available at any given speed. Running torque falls with increasing speed, typical examples being shown on Figure 5.23. Obviously, torque is dependent on the applied pressure; increasing the pressure results in increased torque, as shown.

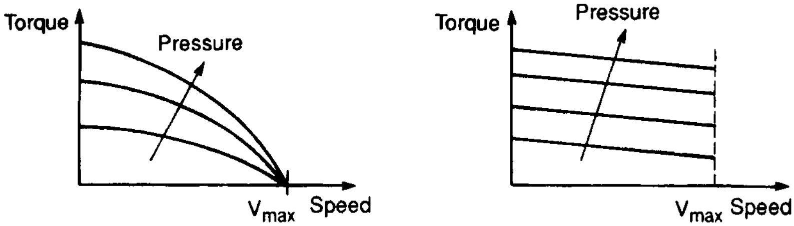


Figure 5.23 *Torque/speed curves for rotary actuators*

The output power of an actuator is related to torque and rotational speed, and is given by the expression:

$$P = \frac{T\,R}{9550}\,\text{kw.} \tag{5.9}$$

where T is the torque in newton metre and R is the speed in rpm. In Imperial units the expression is:

$$P = \frac{T\,R}{5252}\,\text{hp.} \tag{5.10}$$

where T is in lbsf ft (and R is in rpm) or:

$$P = \frac{T\,R}{63024}\,\text{hp.} \tag{5.11}$$

where T is in lbsf ins.

Figure 5.23 illustrates how running torque falls with increasing speed, so the relationship between power and speed has the form of Figure 5.24, with maximum power at some (defined) speed. Power like the torque, is dependent on applied pressure.

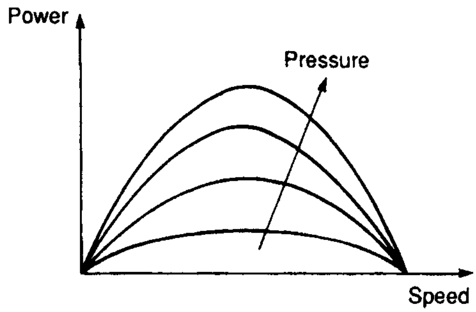


Figure 5.24 *Power/speed curve for pneumatic rotary actuator*

The torque produced by a rotary actuator is directly related to fluid pressure; increasing pressure increases maximum available torque. Actuators are often specified by their torque rating, which is defined as:

$$\text{torque rating} = \frac{\text{torque}}{\text{pressure}}$$

In Imperial units a pressure of 100 psi is used, and torque is generally given in lbf ins.

The allowable pressure for an actuator is defined in terms of pressure rating (maximum applicable pressure without risk of permanent damage), and pressure range (the maximum and minimum pressures between which actuator performance is defined).

Fluid passes through an actuator as it rotates. For hydraulic actuators, displacement is defined as the volume of fluid used for one motor rotation. For a given design of motor, available torque is directly proportional to displacement. For pneumatic actuators, the air usage per revolution at a specified pressure is generally given in terms of STP (see Chapter 3).

Rotational speed is given by the expression:

$$\text{rotational speed} = \frac{\text{fluid flow rate}}{\text{displacement}}$$

With the torque rate and displacement fixed for a chosen motor, the user can control maximum available torque and speed by adjusting, respectively, pressure setting and flow rate of fluid to the actuator.

Constructional details

In electrical systems, there are many similarities between electrical generators and electric motors. A DC generator, for example, can be run as a motor. Similarly, a DC motor can be used as a generator. Similar relationships exist between hydraulic pumps and motors and between pneumatic compressors and motors. This similarity is extended as manufacturers use common parts in pumps, compressors and motors to simplify users' spares holdings.

The similarity between pumps, compressors and motors extends to graphic symbols. The schematic symbols of Figure 5.25 are used to show hydraulic and pneumatic motors. Internal leakage always occurs in a hydraulic motor, and a drain line, shown dotted, is used to return the leakage fluid to the tank. If this leakage return is inhibited the motor may pressure lock and cease to rotate or even suffer damage.

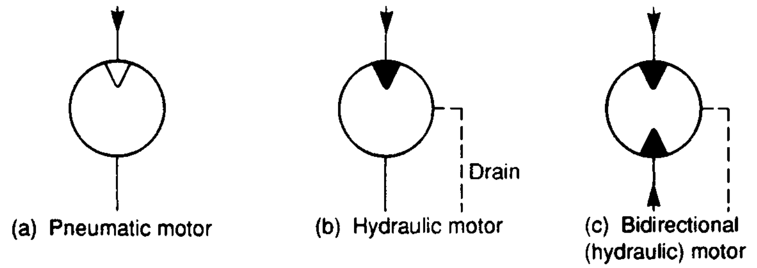


Figure 5.25 *Rotary actuator symbols*

There are three basic designs of rotary or pump compressor; the gear pump, the vane pump and various designs of piston pump or compressor described earlier in Chapter 2. These can also be used as the basis of rotary actuators. The principles of hydraulic and pneumatic devices are very similar, but the much higher hydraulic pressures give larger available torques and powers despite lower rotational speeds.

Figure 5.26 shows the construction of a gear motor. Fluid enters at the top and pressurises the top chamber. Pressure is applied to two gear faces at X, and a single gear face at Y. There is, thus, an imbalance of forces on the gears resulting in rotation as shown. Gear motors suffer from leakage which is more pronounced at low speed. They thus tend to be used in medium speed, low torque applications.

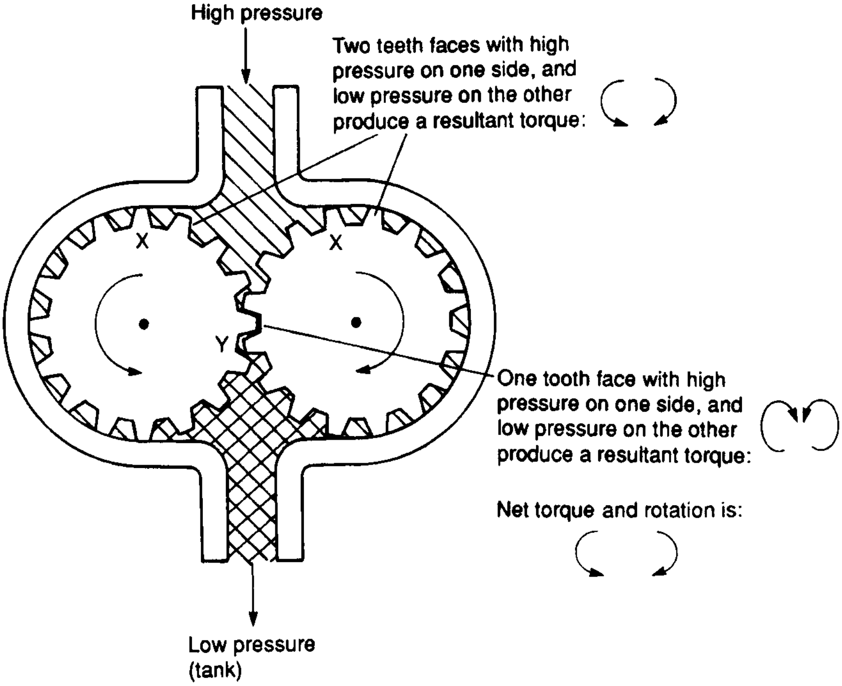


Figure 5.26 *A gear motor*

A typical vane motor construction is illustrated in Figure 5.27. It is very similar to the construction of a vane pump. Suffering from less leakage than the gear motor, it is typically used at lower speeds. Like the vane pump, side loading occurs on the shaft of a single vane motor. These forces can be balanced by using a dual design similar to the pump shown in Figure 2.10b. In a vane pump, vanes are held out by the rotational speed. In a vane motor, however, rotational speed is probably quite low and the vanes are held out, instead, by fluid pressure. An in-line check valve can be used, as in Figure 5.28, to generate a pressure which is always slightly higher than motor pressure.

Piston motors are generally most efficient and give highest torques, speeds and powers. They can be of radial design similar to the pump of Figures 2.12 and 2.13, or in-line (axial) design similar to those of Figures 2.14 and 2.15. Radial piston motors tend to be most common in pneumatic applications, with in-line piston motors most common in hydraulics. The speed of the piston motor can be varied by adjusting the angle of the swash plate (in a similar manner to which delivery volume of an in-line piston pump can be varied).

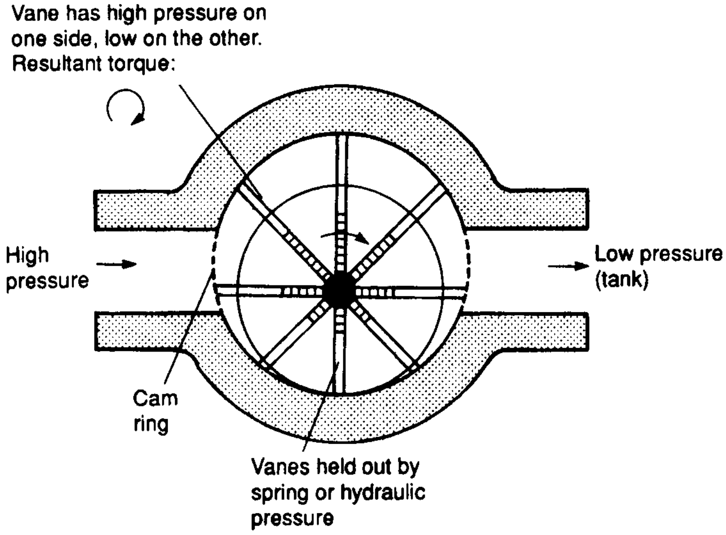


Figure 5.27 *A vane motor*

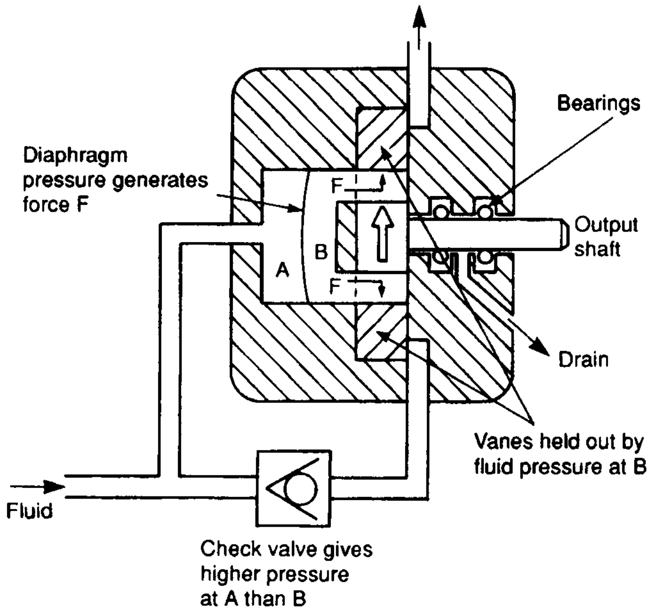


Figure 5.28 *Vane operation in hydraulic motor*

Turbine-based motors can also be used in pneumatics where very high speeds (up to 500,000 rpm) but low torques are required. A common application of these devices is the high-speed dentist's drill.

All the rotary actuators described so far have been pneumatic or hydraulic equivalents of electric motors. However, rotary actuators with a limited travel (say 270°) are often needed to actuate dampers or control large valves. Some examples are illustrated in Figure 5.29. The actuator in Figure 5.29a is driven by a single vane coupled to the output shaft. In that of Figure 5.29b, a double-acting piston is coupled to the output shaft by a rack and pinion. In both cases the shaft angle can be finely controlled by fluid applied to the ports. These have the graphic symbol shown in Figure 5.29c.

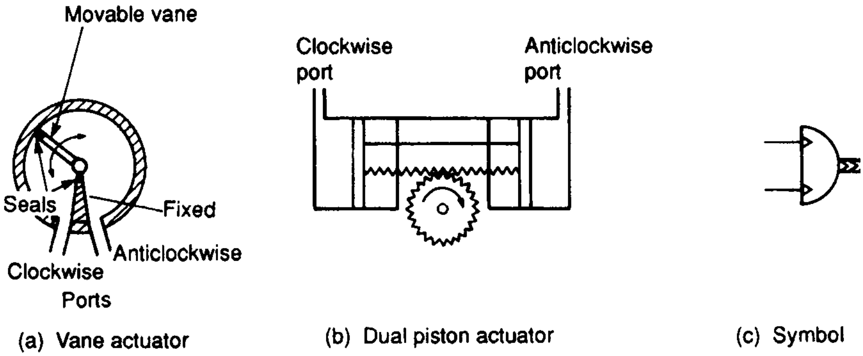


Figure 5.29 *Limited motion rotary actuators*

Application notes

Speed control

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.

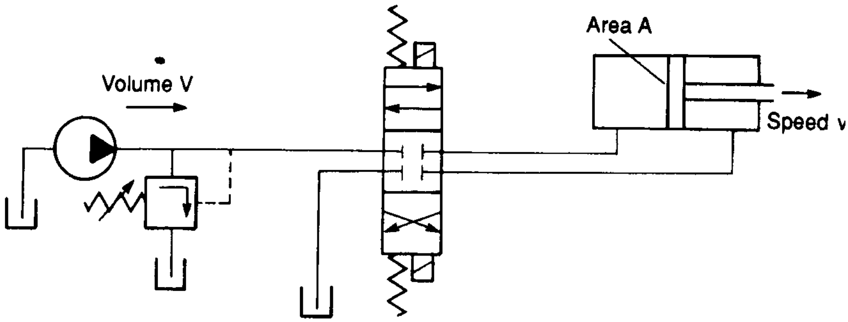


Figure 5.30 *Speed control by pump volume*

Most systems, however, are not that simple. In the second speed control method of Figure 5.31, a pump controls many devices and is loaded by a solenoid-operated valve (see Chapter 2). Unused fluid goes back to the tank via relief valve V_3. The pump output is higher than needed by any individual actuator, so a flow restrictor is used to set the flow to each actuator. This is known as a 'meter in' circuit, and is used where a force is needed to move a load. Check valve V_1 gives a full-speed retraction, and check valve V_2 provides a small back pressure to avoid the load running away. The full pump delivery is produced when the pressure reaches the setting of relief valve V_3, so there is a waste of energy and unnecessary production of heat in the fluid.

If the load can run away from the actuator, the third speed control method; the 'meter out' circuit of Figure 5.32 must be used. As

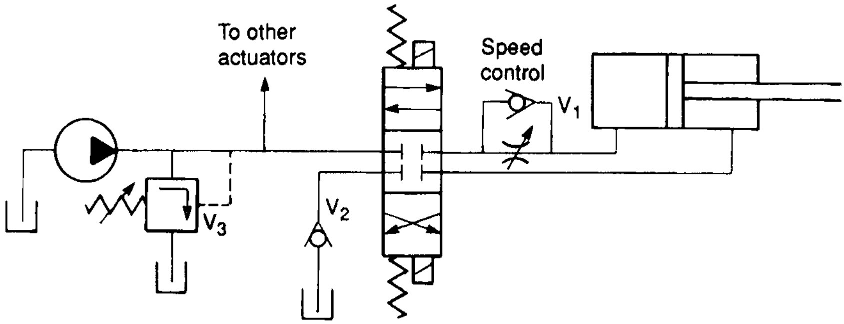


Figure 5.31 *Meter in speed control*

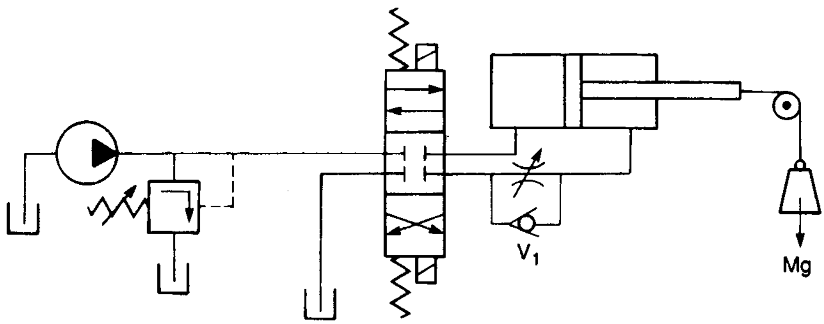


Figure 5.32 *Meter out speed control for overhauling load*

drawn, this again gives a controlled extension speed, and full retraction speed (allowed by check valve V_1). As before, the pump delivers fluid at a pressure set by the relief valve, leading to heat generation.

Finally, in the fourth speed control method of Figure 5.33, a bleed-off valve V_1 is incorporated. This returns a volume v back to

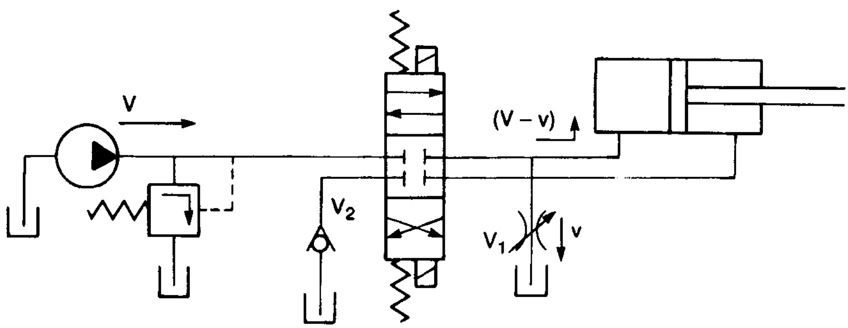


Figure 5.33 *Bleed-off speed control*

the tank, leaving a volume V-v to go to the actuator (where V is the pump delivery volume). Pump pressure is now determined by the required actuator pressure, which is lower than the pressure set on the relief valve. The energy used by the pump is lower, and less heat is generated. The circuit can, however, only be used with a load which opposes motion. Check valve V_2 again gives a small back pressure.

Any unused fluid from the pump is returned to the tank at high pressure leading to wasted energy; even with the more efficient 'bleed'-off circuit. One moral, therefore, is to have a pump delivery volume no larger than necessary.

Figures 5.31 to 5.33 imply flow, and hence speed, is set by a simple restriction in piping to the actuator. While a simple restriction reduces flow and allows speed to be reduced, in practice a true flow control valve is needed which delivers a fixed flow regardless of line pressure or fluid temperature.

An ideal flow controller operates by maintaining a constant pressure drop across an orifice restriction in the line, the rate being adjusted by altering orifice size. The construction of such a device is shown in Figure 5.34. The orifice is formed by a notch in a shaft which can be rotated to set the flow. The pressure drop across the orifice is the difference in pressure between points X and Y, and is applied to the moveable land. The pressure at X, in conjunction with the spring pressure, causes a downward force, while pressure at Y causes an upward force. If the land moves up the flow reduces,

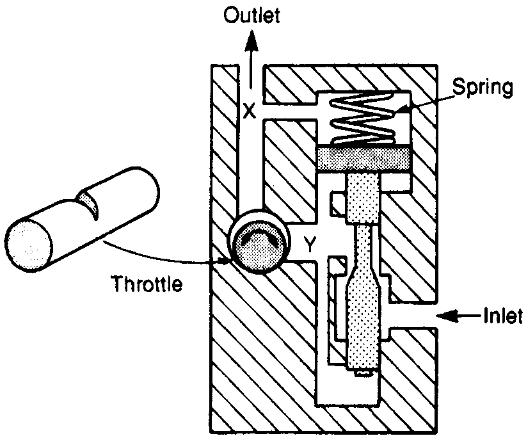


Figure 5.34 *Pressure compensated flow control valve*

if the land moves down the flow increases. The piston thus moves up and down until the pressure differential between X and Y matches the spring compressive force. The device thus maintains a constant pressure drop across the orifice, which implies constant flow through the valve, and is known as a pressure-compensated flow control valve.

Flow control valves can also be adversely affected by temperature changes which alter the viscosity of the oil. For this reason more complex flow control valves often have temperature compensation. Symbols for various types of flow control valves are given in Figure 5.35.

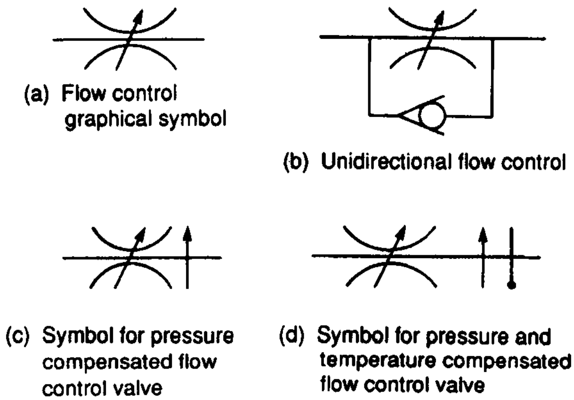


Figure 5.35 *Flow control valves*

Discussions in this section have, so far, been concerned with hydraulic systems as compressibility of air makes speed control of pneumatic actuators somewhat crude. If a pneumatic actuator is required to act at a slow controlled speed an external *hydraulic* damper can be used, as shown in Figure 5.36. Oil is forced from one side of the hydraulic piston to the other via an adjustable flow control valve. Speeds as low as a few millimetres a minute can be accurately controlled in this manner, although the technique is physically rather cumbersome.

Actuator synchronisation

Figure 5.37 illustrates a common problem in which an unbalanced load is to be lifted by two cylinders The right-hand cylinder is

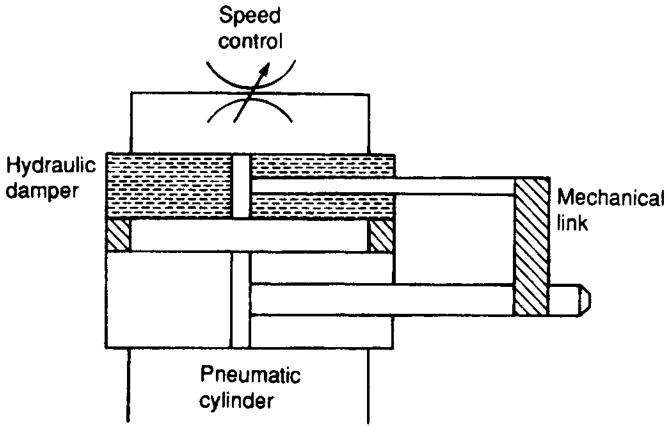


Figure 5.36 *Speed control of pneumatic cylinder*

subject to a large force F, the left-hand cylinder to a smaller force f. The right-hand piston requires a pressure of F/A to lift, while the left-hand piston needs f/A. When lift is called for on valve V_1, the pressure rises to the lower pressure f/A, and only the left-hand piston moves. The unbalanced load results in faulty operation. A similar result can occur where two, or more, cylinders operate against ill-defined frictional forces.

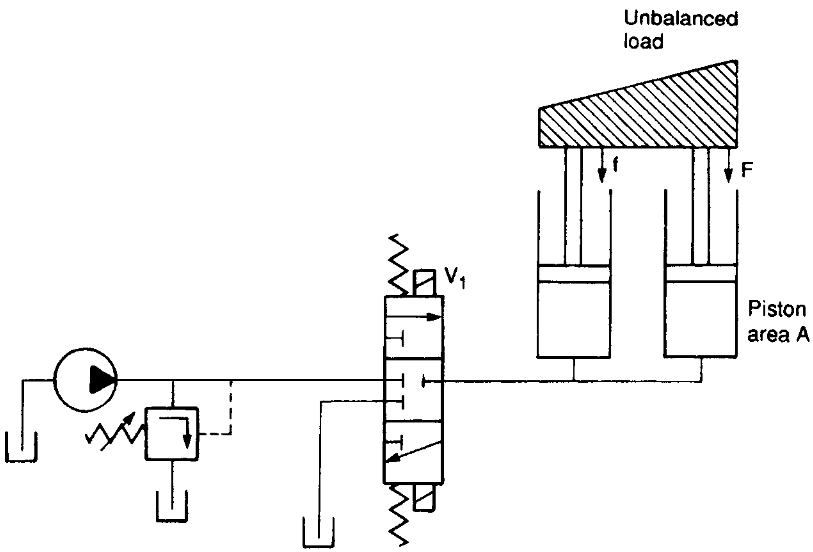


Figure 5.37 *Linked cylinders with unbalanced load*

One simple solution is the inclusion of flow regulating valves. A flow control valve can set, and hold, fluid flow to within about ±5% of nominal value, resulting in a possible positional error of 10% of the stroke. This may, or may not, be acceptable, and in the example of Figure 5.37 the cylinders would, in any case, align themselves at each end of the stroke. (When the most lightly laden and hence fastest travelling piston reaches the end of its stroke, the system pressure will rise.) This solution is not acceptable if good positional accuracy is required or rotary actuators without end stops are being driven.

The flow divider valve of Figure 5.38 works on a similar principle, dividing the inlet flow equally (to a few percent) between two outlet ports. The spool moves to maintain equal pressure drops across orifices X and Y, and hence equal flow through them.

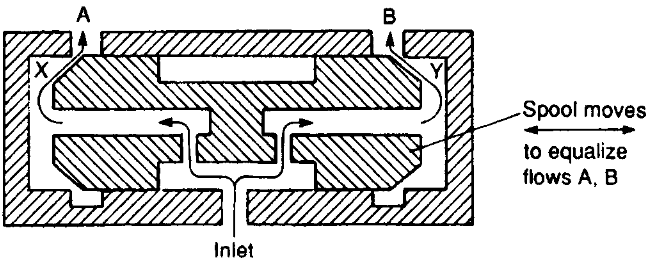


Figure 5.38 *Flow divider valve*

The displacement of a hydraulic or pneumatic motor can be accurately specified, and this forms the basis of an alternative flow divider circuit of Figure 5.39. Here fluid for two cylinders passes through two mechanically coupled motors. The mechanical coupling ensures the two motors rotate at the same speed, and hence equal flow is passed into each cylinder.

The two cylinders in Figure 5.40 are effectively in series with fluid from the annulus side of cylinder 1 going to the full bore side of cylinder 2. The cylinders are chosen, however, so that full bore area of cylinder 2 equals the annulus area of cylinder 1. Upon cylinder extension, fluid exits from cylinder 1 and causes cylinder 2 to extend. The two cylinders move at equal speed because of the equal areas.

There is, though, an unfortunate side effect. Pressure P_2 in cylinder 2 is F/a. Fluid on the full bore side of cylinder 1 has to lift the piston against force f plus the force from P_2 acting on the annulus side of the piston. Pressure P_1 is (F+f)/A; higher than would be

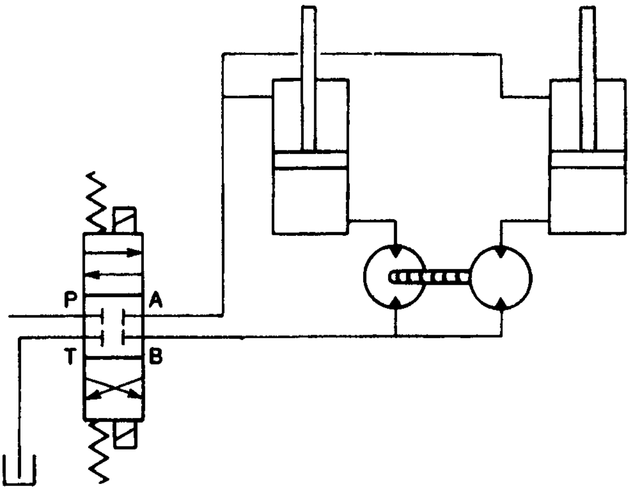


Figure 5.39 *Cylinder synchronisation with linked hydraulic motors*

required by two independent cylinders acting in parallel. The rotational speed of motors with equal displacement can similarly be synchronised by connecting them in series. Inlet pressure of the first motor is again, however, higher than needed to drive the two motors separately or in parallel.

None of these methods gives absolute synchronisation, and if actuators do not self-align at the ends of travel, some method of driving actuators individually should be included to allow intermittent manual alignment. The best solution, however, is usually to include some form of mechanical tie to ensure actuators experience equal loads and cannot get out of alignment.

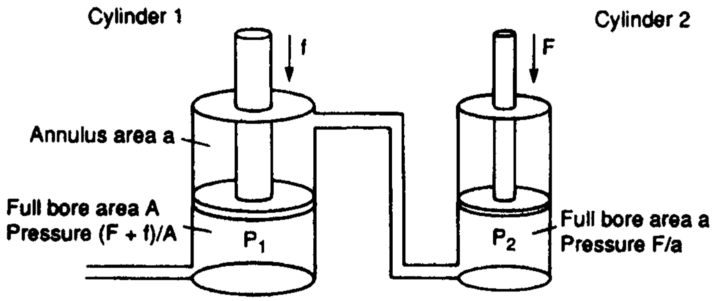


Figure 5.40 *Cylinder synchronisation with series connection*

Regeneration

A conventional cylinder can exert a larger force extending than retracting because of the area difference between full bore and annulus sides of the piston. The system in Figure 5.41 employs a cylinder with a full bore/annulus ratio of 2:1, and is known as a differential cylinder.

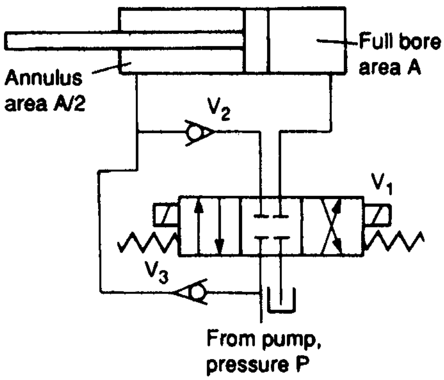


Figure 5.41 *Regeneration circuit*

Upon cylinder extension, line pressure P is applied to the right-hand side of the piston giving a force of P × A, while the left-hand side of the piston returns oil via valve V_3 against line pressure P producing a counter force P × A/2. There is thus a net force of P × A/2 to the left. When retraction is called for, a force of P × A/2 is applied to the left-hand side and fluid from the right-hand side returns to tank at minimal pressure. Extension and retraction forces are thus equal, at P × A/2.

Counterbalance and dynamic braking

The cylinder in Figure 5.42 supports a load which can run away when being lowered. Valve V_2, known as a counterbalance valve, is a pressure-relief valve set for a pressure higher than F/2 (the pressure generated in the fluid on the annulus side of the piston by the load). In the static state, valve V_2 is closed and the load holds in place.

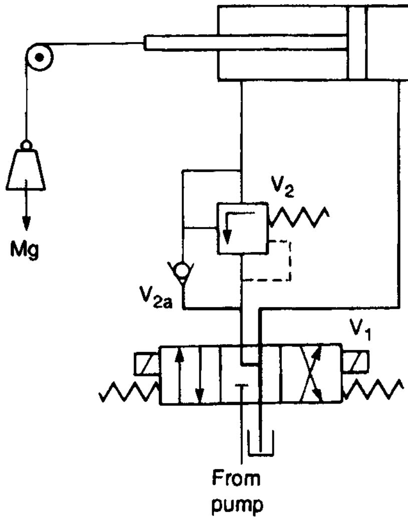


Figure 5.42 *Counterbalance circuit*

When the load is to be lowered line pressure is applied to the full bore side of the piston through valve V_1. The increased pressure causes valve V_2 to open and the load to lower. Check Valve V_2a passes fluid to raise the load.

Counterbalance valves can also be used to brake a load with high inertia. Figure 5.43 shows a system where a cylinder moves a load with high inertia. Counterbalance valves V_2 and V_3 are included in the lines to both ends of the cylinder. Cross-linked pilot lines (shown dotted as per convention) keep valve V_2 open when extending and valve V_3 open when retracting. At constant cylinder speed, therefore, valves V_2 and V_3 have little effect.

To stop the load valve V_1 is moved to its centre position, the pump unloads to tank and pilot pressure is lost, causing valves V_2 and V_3 to close. Inertia, however, maintains some cylinder movement. If, for example, the cylinder had been extending, inertia keeps it moving to the left – raising pressure on the piston's annulus side until valve V_2 reaches its pressure setting and opens. A constant deceleration force Pa (where P is the setting of valve V_1 and a is the annulus area) is applied to the load. On deceleration, fluid passes to the full bore side of the cylinder through check valve V_3a.

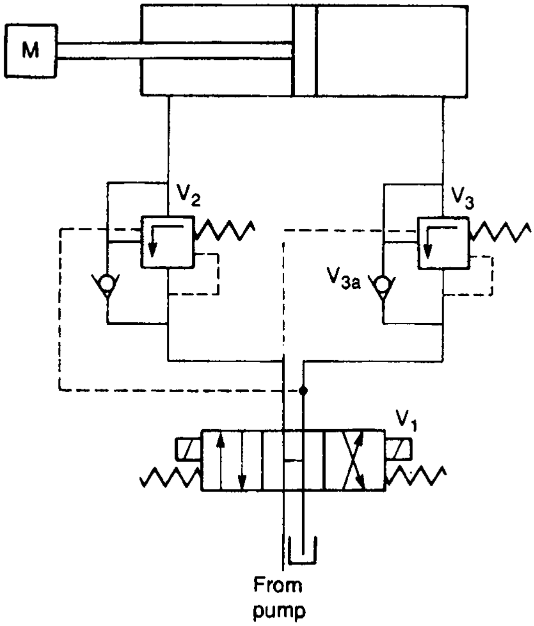


Figure 5.43 *Braking a high inertia load*

Pilot-operated check valves

Directional control valves and deceleration valves have a small, but definite, leakage and can only be used to hold an opposing load in position for short periods (of the order of minutes rather than hours) without the energy wasting procedure of permanently applying pressure to the cylinder.

A check valve can be constructed with zero leakage. The pilot-operated check valve (described in Chapter 4) can thus be used to 'lock' an actuator in position. Figure 5.44 shows a typical example. Valve V_2 passes fluid normally when extending, but closes when valve V_1 is in its centre position. In this state, energy is saved by unloading the pump to tank. Pilot line pressure opens valve V_2 when the load is to be lowered. Counterbalance valve V_3 gives a controlled lowering but also ensures sufficient line pressure exists on the annulus side of the cylinder to give the pilot pressure needed to open valve V_2.

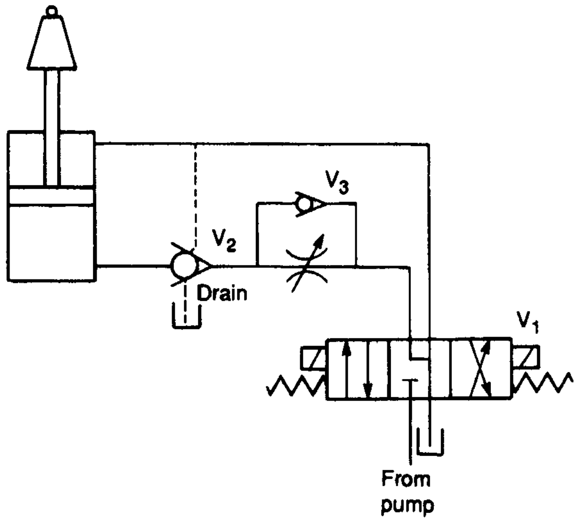


Figure 5.44 *Pilot-operated check valve used to hold an over-hauling load*

Pre-fill and compression relief

Figure 5.45 shows the hydraulic circuit for a large press. To give the required force, a large diameter cylinder is needed and, if this is driven directly, a large capacity pump is required. The circuit shown (known as a pre-fill circuit) uses a high level tank and pilot operated check valve to reduce the required pump size.

The cross-head of the press is raised and lowered by small cylinders C_1 and C_2. When valve V_1 is switched to lower, the pressure on the full bore sides of cylinders C_1 and C_2 is low and valve V_3 is closed. Valve V_4 is a counterbalance valve, giving a controlled lower. As cylinders C_1 and C_2 extend, cylinder C_3 also extends because it is mechanically coupled, drawing its fluid direct from the high level tank via pilot valve V_2.

When the cross-head contacts the load, the pressure on the full bore side of cylinders C_1 and C_2 rises. This causes valve V_3 to open, full line pressure to be applied to cylinder C_3 and check valve V_2 to close. Full operating force is now applied to the load via cylinder C_3.

When the cross-head is raised, pressure is applied to the annulus side of cylinders C_1 and C_2. This opens check valve V_2, allowing fluid in cylinder C_3 to be returned directly to tank.

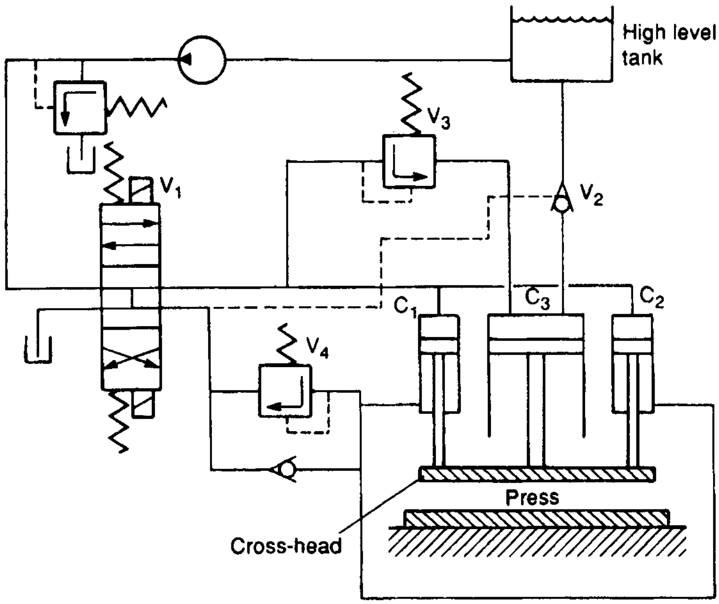


Figure 5.45 *Pre-fill circuit*

High pressure hydraulic circuits like this require care both in design and in maintenance. For most practical purposes, hydraulic fluid can be considered incompressible. In reality, it compresses by about 0.8% per 100 bar applied pressure. When high pressure and large volumes of oil are present together, sudden release of pressure can result in an explosive release of fluid. The design must, therefore, allow for the gradual release of high pressure, high volume fluid.

Large volume, high pressure valves are thus fitted with a central damping block as illustrated in Figure 5.46 to return fluid to tank slowly.

Figure 5.47 shows a common decompression circuit. When the cylinder extends, fluid passes to the full bore side via check valve V_3 as usual, with fluid pressure rising once the load is contacted. This rise in pressure keeps valve V_2 closed. When valve V_1 is returned to its centre position, the pressure decays via restriction valve RV_1. Once the pressure decays to a safe level, set by valve V_2, this valve opens allowing pressure to decay fully.

Valve V_4 is included to protect against a quick change from high pressure extend to retract, without a pause to allow the pressure to decay. When the full bore side of the cylinder is pressurised, valve V_4 is held open causing the pump to unload to tank if retract is

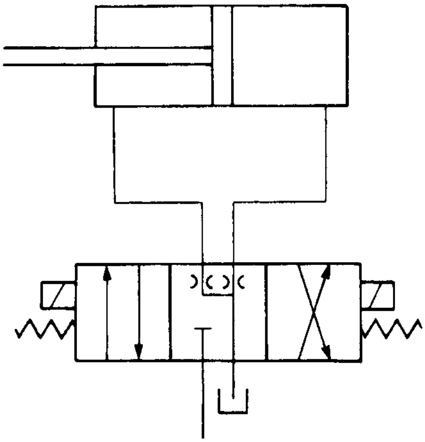


Figure 5.46 *Valve with central damping block*

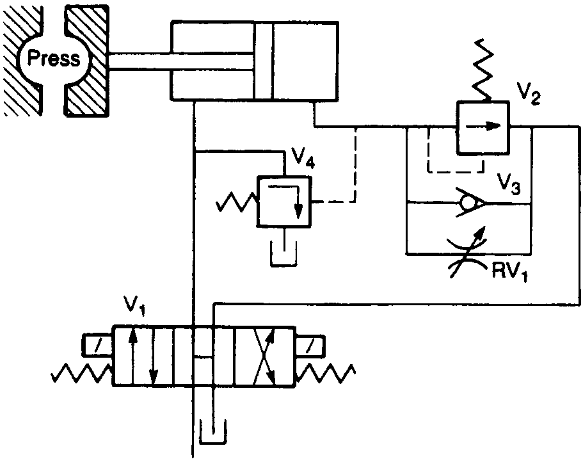


Figure 5.47 *Controlled decompression circuit*

requested before decompression is complete. Once pressure on the full bore side decays, valve V3 closes and the cylinder can retract as normal.

Bellows actuator

Many applications require a simple lift function, for example to raise a disappearing stop on a set of rollers. This function is usually provided by a pneumatic cylinder which requires space and mounting lugs. A simple alternative is the bellows of Figure 5.48. In the de-energized state the bellows are deflated and the load falls under gravity. When air is passed to the bellows they inflate lifting the load. The actuator requires minimal space in its de-energized state and is simple to mount. The only disadvantage is that the load falls under gravity and is not driven down.

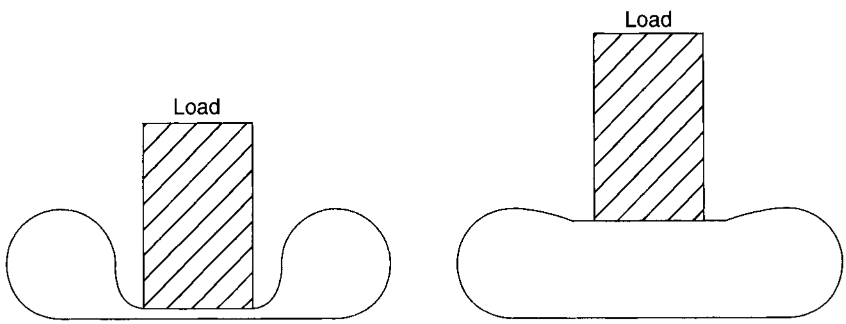


Figure 5.48 *The use of pneumatic bellows gives a simple way of raising and lowering a load*