Hydraulics and Pneumatics

A Technician's and Engineer's Guide

Third edition

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Preface

Machines should work, people should think

The IBM Pollyanna Principle

Practically every industrial process requires objects to be moved, manipulated or be subjected to some form of force. This is generally accomplished by means of electrical equipment (such as motors or solenoids), or via devices driven by air (pneumatics) or liquids (hydraulics).

Traditionally, pneumatics and hydraulics are thought to be a mechanical engineer's subject (and are generally taught as such in colleges). In practice, techniques (and, more important, the fault-finding methodology) tend to be more akin to the ideas used in electronics and process control.

This book has been written by a process control engineer as a guide to the operation of hydraulic and pneumatic systems. It is intended for engineers and technicians who wish to have an insight into the components and operation of a pneumatic or hydraulic system. The mathematical content has been deliberately kept simple with the aim of making the book readable rather than rigorous. It is not, therefore, a design manual, and topics such as sizing of pipes and valves have been deliberately omitted.

The second edition was updated to include recent developments such as the increasing use of proportional valves, and an expanded section on industrial safety.

The third edition includes many changes based on suggestions from readers including a new chapter on sequencing applications and an appendix of hydraulic and pneumatic symbols. I would, in particular, like to thank Don MacDonald and Kerry Jones of IPS International (www.ips-international.com) for useful advice and the use of the simulation software described in Chapter 9.

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Fundamental Principles

INDUSTRIAL PRIME MOVERS

Most industrial processes require objects or substances to be moved from one location to another, or a force to be applied to hold, shape or compress a product. Such activities are performed by prime movers, the workhorses of manufacturing industries.

In many locations all prime movers are electrical. Rotary motions can be provided by simple motors, and linear motion can be obtained from rotary motion by devices such as screw jacks or rack and pinions. Where a pure force or a short linear stroke is required a solenoid may be used (although there are limits to the force that can be obtained by this means).

Electrical devices are not, however, the only means of providing prime movers. Enclosed fluids (both liquids and gases) can also be used to convey energy from one location to another and, consequently, to produce rotary or linear motion or apply a force. Fluid-based systems using liquids as transmission media are called hydraulic systems (from the Greek words *hydra* for water and *aulos* for a pipe, descriptions which imply fluids are water although oils are more commonly used). Gas-based systems are called pneumatic systems (from the Greek *pneumn* for wind or breath). The most common gas is simply compressed air, although nitrogen is occasionally used.

The main advantages and disadvantages of pneumatic or hydraulic systems both arise out of the different characteristics of low-density compressible gases and (relatively) high-density incompressible liquids. A pneumatic system, for example, tends to have a 'softer' action than a hydraulic system which can be prone to producing noisy and wear-inducing shocks in the piping. A liquid-based hydraulic system, however, can operate at far higher pressures than a pneumatic system and, consequently, can be used to provide very large forces.

To compare the various advantages and disadvantages of electrical pneumatic and hydraulic systems, the following three sections consider how a simple lifting task could be handled by each.

A BRIEF SYSTEM COMPARISON

The task considered is how to lift a load by a distance of about 500 mm. Such tasks are common in manufacturing industries.

1

An electrical system

With an electrical system we have three basic choices: a solenoid, a DC motor or the ubiquitous workhorse of industry, the AC induction motor. Of these, the solenoid produces a linear stroke directly but its stroke is normally limited to a maximum distance of around 100 mm.

Both DC and AC motors are rotary devices and their outputs need to be converted to linear motion by mechanical devices such as wormscrews or rack and pinions. This presents no real problems; commercial devices are available comprising motor and screw.

The choice of motor depends largely on the speed control requirements. A DC motor fitted with a tacho and driven by a thyristor drive can give excellent speed control, but has high maintenance requirements for brushes and commutator.

An AC motor is virtually maintenance free, but is essentially a fixed-speed device (with speed being determined by number of poles and the supply frequency). Speed can be adjusted with a variable frequency drive, but care needs to be taken to avoid overheating, as most motors are cooled by an internal fan connected directly to the motor shaft. We will assume a fixed speed raise/lower is required, so an AC motor driving a screwjack would seem to be the logical choice.

Neither type of motor can be allowed to stall against an end of travel stop (this is not quite true; specially designed DC motors, featuring good current control on a thyristor drive together with an external cooling fan, *can* be allowed to stall), so end of travel limits are needed to stop the drive.

We have thus ended up with the system shown in Figure 1.1 comprising a mechanical jack driven by an AC motor controlled by a reversing starter. Auxiliary equipment comprises two limit switches, and a motor overload protection device. There is no practical load limitation provided screw/gearbox ratio, motor size and contactor rating are correctly calculated.

A hydraulic system

A solution along hydraulic lines is shown in Figure 1.2. A hydraulic linear actuator suitable for this application is the ram, shown schematically in Figure 1.2a. This consists of a movable piston connected directly to the output shaft. If fluid is pumped into pipe A, the piston will move up and the shaft will extend; if fluid is pumped into pipe B, the shaft will retract. Obviously some method of retrieving fluid from the non-pressurized side of the piston must be incorporated.

The maximum force available from the cylinder depends on fluid pressure and cross-sectional area of the piston. This is discussed further in a later section but, as an example, a typical hydraulic pressure of 150 bar will lift 150 kg cm^{-2} of piston area. A load of 2000 kg could thus be lifted by a 4.2 cm diameter piston.

A suitable hydraulic system is shown in Figure 1.2b. The system requires a liquid fluid to operate; this is expensive and messy and, consequently, the piping must act as a closed loop, with fluid transferred from a storage tank to one side of the piston, and returned from the other side of the piston to the tank. Fluid is drawn from the tank by a pump which produces fluid flow at the required 150 bar.

(a) Electric circuit

(b) Physical layout
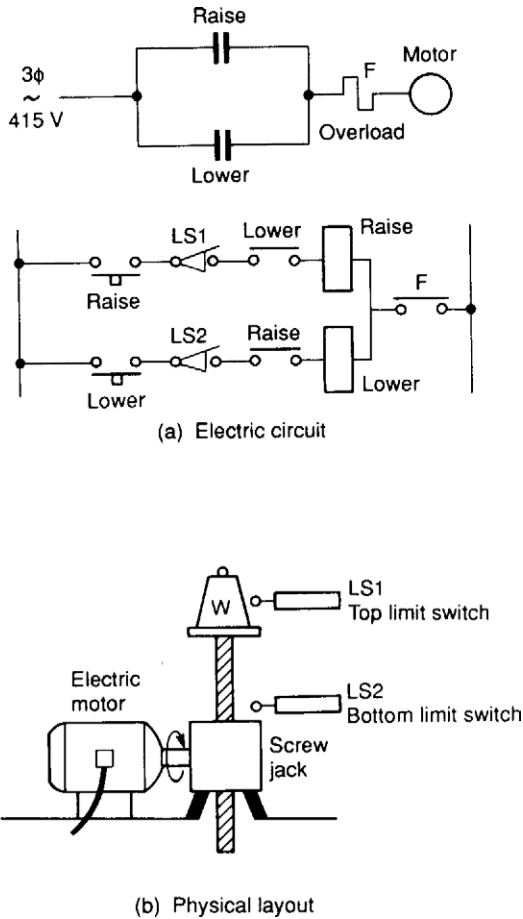
FIGURE 1.1 Electrical solution, based on three-phase motor

Such high-pressure pumps, however, cannot operate into a dead-end load as they deliver constant volumes of fluid from input to output ports for each revolution of the pump shaft. With a dead-end load, fluid pressure rises indefinitely, until a pipe or the pump itself fails. Some form of pressure regulation, as shown, is therefore required to spill excess fluid back to the tank.

Cylinder movement is controlled by a three-position changeover valve. To extend the cylinder, port A is connected to the pressure line and port B to the tank. To reverse the motion, port B is connected to the pressure line and port A to the tank. In its center position the valve locks the fluid into the cylinder (thereby holding it in position) and dead-ends the fluid lines (causing all the pump output fluid to return to the tank via the pressure regulator).

There are a few auxiliary points worthy of comment. First, speed control is easily achieved by regulating the volume flow rate to the cylinder (discussed in a later section). Precise control at low speeds is one of the main advantages of hydraulic systems.
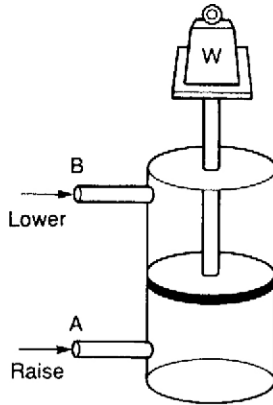
(a) Hydraulic cylinder
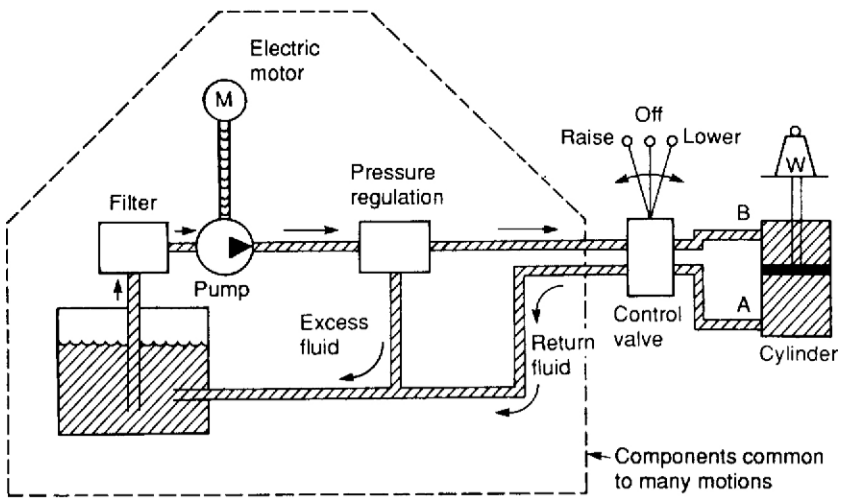
(b) Physical components

FIGURE 1.2 Hydraulic solution

Second, travel limits are determined by the cylinder stroke and cylinders, generally, can be allowed to stall at the ends of travel so no overtravel protection is required.

Third, the pump needs to be turned by an external power source, almost certainly an AC induction motor which, in turn, requires a motor starter and overload protection.

Fourth, hydraulic fluid needs to be very clean, hence a filter is needed (shown in Figure 1.2b) to remove dirt particles before the fluid passes from the tank to the pump.

One final point worth mentioning is that leaks of fluid from the system are unsightly, slippery (hence hazardous) and environmentally very undesirable. A major failure can be catastrophic.

At first sight Figure 1.2b appears inordinately complicated compared with the electrical system of Figure 1.1, but it should be remembered that all parts enclosed in the broken-lined box in Figure 1.2 are common to an area of plant and not usually devoted to just one motion as we have drawn.

A pneumatic system

Figure 1.3 shows the components of a pneumatic system. The basic actuator is again a cylinder, with maximum force on the shaft being determined by air pressure and piston cross-sectional area. Operating pressures in pneumatic systems are generally much lower than those in a hydraulic system, 10 bar being typical, which will lift $10\,kg\,cm^{-2}$ of piston area, so a $16\,cm$ diameter piston is required to lift the $2000\,kg$ load specified in the previous section. Pneumatic systems therefore require larger actuators than hydraulic systems for the same load.

The valve delivering air to the cylinder operates in a similar way to its hydraulic equivalent. One notable difference arises out of the simple fact that air is free; return air is simply vented to the atmosphere.

Air is drawn from the atmosphere via an air filter and raised to required pressure by an air compressor (usually driven by an AC motor). The air temperature is raised considerably by this compressor. Air also contains a significant amount of water vapor. Before the air can be used it must be cooled, and this results in the formation of condensation. So, the air compressor must be followed by a cooler and air treatment unit.

Compressibility of a gas makes it necessary to store a volume of pressurized gas in a reservoir, to be drawn on by the load. Without this reservoir, a slow
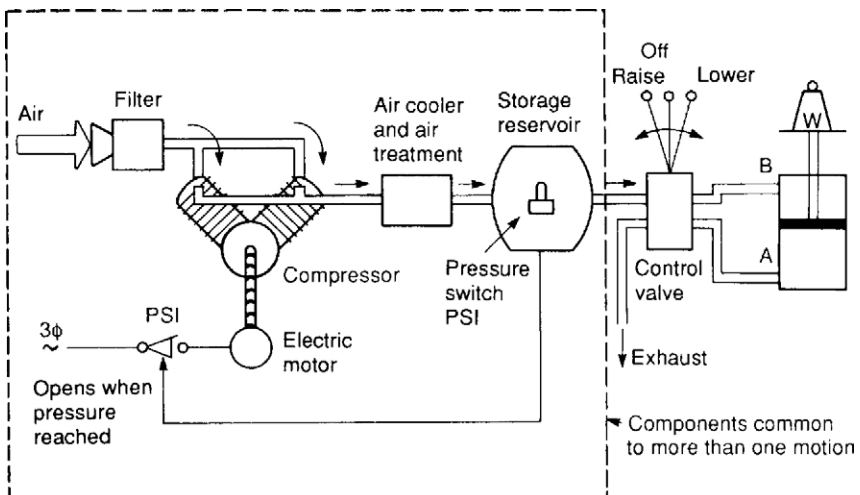
FIGURE 1.3 Pneumatic solution

exponential rise in pressure results in a similar slow cylinder movement when the valve is first opened. The air treatment unit is thus followed by an air reservoir.

Hydraulic systems require a pressure regulator to spill excess fluid back to the tank, but pressure control in a hydraulic system is much simpler. A pressure switch, fitted to the air reservoir, starts the compressor motor when pressure falls and stops it again when pressure reaches the required level.

The general impression is again one of complexity, but units in the broken-lined box are again common to one plant or even a whole site. Many factories produce compressed air at one central station and distribute an air ring main to all places on the site in a similar way to other services such as electricity, water or gas.

A comparison

Table 1.1 gives superficial comparisons of the various systems discussed in the previous sections.

DEFINITION OF TERMS

There is an almost universal lack of standardization of units used for measurement in industry, and every engineer will tell tales of gauges indicating, say, velocity in furlongs per fortnight. Hydraulics and pneumatic systems suffer particularly from this characteristic, and it is by no means unusual to find pressure indicated at different locations in the same system in bar, kpascal and psi.

TABLE 1.1   Comparisons of electrical, hydraulic and pneumatic systems

	Electrical	**Hydraulic**	**Pneumatic**
Energy source	Usually from outside supplier	Electric motor or diesel driven	Electric motor or diesel driven
Energy storage	Limited (batteries)	Limited (accumulator)	Good (reservoir)
Distribution system	Excellent, with minimal loss	Limited, basically a local facility	Good, can be treated as a plant wide service
Energy cost	Lowest	Medium	Highest
Rotary actuators	AC and DC motors. Good control on DC motors. AC motors cheap	Low speed. Good control. Can be stalled	Wide speed range. Accurate speed control difficult
Linear actuator	Short motion via solenoid. Otherwise via mechanical conversion	Cylinders. Very high force	Cylinders. Medium force
Controllable force	Possible with solenoid and DC motors Complicated by need for cooling	Controllable high force	Controllable medium force
Points to note	Danger from electric shock	Leakage dangerous and unsightly. Fire hazard	Noise

There is, however, a welcome (and overdue) movement to standardization on the International System (SI) of units, but it will be some time before this is complete. The engineer will therefore encounter many odd-ball systems in the years to come.

Any measurement system requires definition of the six units used to measure:

- length:
- mass;
- time;
- temperature;
- electrical current;
- light intensity.

Of these, hydraulic/pneumatic engineers are primarily concerned with the first three. Other units (such as velocity, force, pressure) can be defined in terms of these basic units. Velocity, for example, is defined in terms of length/time.

The old British Imperial system used units of foot, pound and second (and was consequently known as the *fps system*). Early metric systems used centimeter, gram and second (known as the *cgs system*), and meter, kilogram and second (the *mks system*). The mks system evolved into the *SI system* which introduces a more logical method of defining force and pressure (discussed in later sections). Note that units given real persons' names (e.g. newton) use lower case letters, but have capital letter symbols (e.g. N).

The conversion tables below convert TO the equivalent SI unit. To convert FROM SI units divide by the conversion factor. To convert between two non-SI units use a two-stage process: first multiply by the first conversion factor to convert to SI units then divide by the second conversion factor to give the required value in the new units. For example, in Table 1.2 to convert from kips to cwt multiply by 453.59 then divide by 50.802.

TABLE 1.2 Mass

From	Multiply by	Comments
Grams	1×10^{-3}	
Tonnes	1000	Also known as metric ton
Short tons	907.185	1 short ton = 2000 pounds
Long tons	1016.05	1 long ton = 2240 pounds
Kips	453.59	1 kip = 1000 pounds
Hundredweight (cwt)	50.802	1 cwt = 112 lbs
Pounds	4.5359×10^{-4}	1 pound = 16 ounces
Ounces	2.8350×10^{-5}	
Grains	6.48×10^{-7}	

SI unit is the kilogram: kg

Conversions between the basic units of mass, length and volume are given in Tables 1.2–1.4.

TABLE 1.3 Length (distance)

From	Multiply by	Comments
Millimeters (mm)	10^{-3}	
Centimeters (cm)	10^{-2}	
Kilometers (km)	1000	
Miles	1609.35	1 mile = 1760 yards
Nautical mile (UK)	1853	
Yards	0.9144	
Feet	0.3048	
Inches	2.54×10^{-2}	1 inch = 25.4 mm
Mil (thou)	2.54×10^{-5}	1 mil = 0.001 inches
Micron	10^{-6}	
Angstrom	10^{-10}	
Fermi	10^{-15}	

SI unit is the meter: m (also spelt metre in some texts)

TABLE 1.4 Volume

From	Multiply by	Comments
Liters	10^{-3}	1 liter = 1000 cc
Cubic cm (cc)	10^{-6}	
Imperial gallons	4.5461×10^{-3}	1 gallon of water weighs 10 lb
Quart (qt)	1.1365×10^{-3}	4 quarts = 1 gallon
Pints	5.6826×10^{-4}	8 pints = 1 gallon
US gallons	3.7854×10^{-3}	1 US gallon = 231 cubic inches
Cubic yards	0.76455	
Cubic feet	2.8317×10^{-2}	
Cubic inches	1.6387×10^{-5}	
Fluid ounces	2.8413×10^{-5}	20 fluid oz = 1 pint

SI unit is the cubic meter: m^3

Mass and force

Pneumatic and hydraulic systems generally rely on pressure in a fluid. Before we can discuss definitions of pressure, though, we must first be clear what is meant by everyday terms such as weight, mass and force.

We all are used to the idea of weight, which is a *force* arising from gravitational attraction between the mass of an object and the earth. The author weighs 75 kg on the bathroom scales; this is equivalent to saying there is 75 kg *force* between his feet and the ground.

Weight therefore depends on the force of gravity. On the moon, where gravity is about one-sixth that on earth, the author's weight would be about 12.5 kg; in free fall the weight would be zero. In all cases, though, the author's *mass* is constant.

The British Imperial fps system and the early metric systems link mass and weight (force) by defining the unit of force to be the gravitational attraction of unit mass at the surface of the earth. We thus have a mass defined in pounds and force defined in pounds force (lbs f) in the fps system, and mass in kilograms and force in kg f in the mks system.

Strictly speaking, therefore, bathroom scales which read 75 kg are measuring 75 kg f, not the author's mass. On the moon they would read 12.5 kg f, and in free fall they would read zero.

If a force is applied to a mass, acceleration (or deceleration) will result as given by the well-known formula:

$$F = ma \qquad\qquad (1.1)$$

Care must be taken with units when a force F is defined in lbs f or kg f and mass is defined in lbs or kg, because resulting accelerations are in units of g, acceleration due to gravity. A force of 25 kg f applied to the author's mass of 75 kg produces an acceleration of 0.333 g.

The SI unit of force, the newton (N), is defined not from earth's gravity, but directly from expression 1.1. A newton is defined as the force which produces an acceleration of 1 m s^{-2} when applied to a mass of 1 kg.

One kg f produces an acceleration of 1 g (9.81 m s^{-2}) applied to a mass of 1 kg. One newton produces an acceleration of 1 m s^{-2} when applied to a mass of 1 kg. It therefore follows that:

$$1 \text{ kg f} = 9.81 \text{ N}$$

but as most instruments on industrial systems are at best 2% accurate it is reasonable (and much simpler) to use:

$$1 \text{ kg f} = 10 \text{ N}$$

for practical applications.

Table 1.5 gives conversions between various units of force.

A newton is the force which gives a mass of 1 kg an acceleration of one meter per second per second.

TABLE 1.5 Force

From	Multiply by	Comments
Ton f (ton weight)	9964	
lb f (lb weight)	4.448	
oz f (ounce weight)	0.278	
Poundal	0.1383	fps unit of force
kp	9.807	
Pond (p)	9.807×10^{-2}	
kg f (kg weight)	9.807	
g f (gramme weight)	9.807×10^{-2}	
Dyne	10^{-5}	cgs unit of force
SI unit is the newton: N		

A poundal is the force which gives a mass of one pound an acceleration of one foot per second per second.

A dyne is the force which gives a mass of one gram an acceleration of one centimeter per second per second.

Weight, often confused with both force and mass, is the force that arises from the action of gravity on a mass.

Pressure

Pressure occurs in a fluid when it is subjected to a force. In Figure 1.4 a force F is applied to an enclosed fluid via a piston of area A. This results in a pressure P in the fluid. Obviously increasing the force increases the pressure in direct proportion. Less obviously, though, decreasing piston area also increases pressure. Pressure in the fluid can therefore be defined as the force acting per unit area, or:

$$P = \frac{F}{A} \tag{1.2}$$
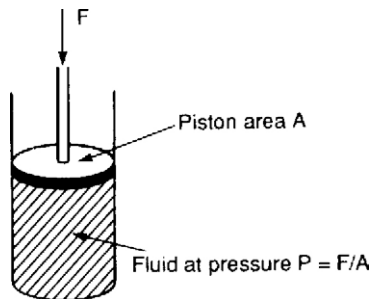
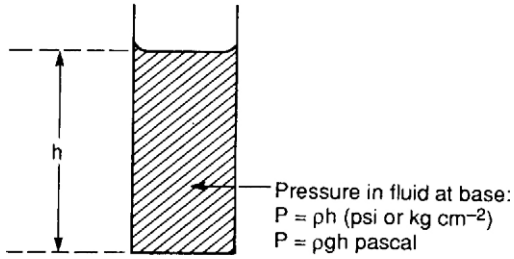**FIGURE 1.4** Pressure in a fluid subjected to a force

FIGURE 1.5 Head pressure in a fluid

Although expression 1.2 is very simple, there are many different units of pressure in common use. In the Imperial fps system, for example, F is given in lbs f and A is given in square inches to give pressure measured in pound force per square inch (psi).

In metric systems, F is usually given in kg f and A in square centimeters to give pressure in kilogram/force per square centimeter (kg f cm^{-2}).

The SI system defines pressure as the force in newtons per square meter (N m^{-2}). The SI unit of pressure is the pascal (with $1\,\text{Pa} = 1\,\text{N m}^{-2}$). One pascal is a very low pressure for practical use, however, so the kilopascal ($1\,\text{kPa} = 10^3\,\text{Pa}$) or the megapascal ($1\,\text{MPa} = 10^6\,\text{Pa}$) are more commonly used.

Pressure can also arise in a fluid from the weight of a fluid. This is usually known as the head pressure and depends on the height of fluid. In Figure 1.5 the pressure at the bottom of the fluid is directly proportional to height h.

In the Imperial and metric systems head pressure is given by:

$$P = \rho h \tag{1.3}$$

where ρ is the density and h the height (both in the correct units) to give P in psi or kg cm^{-2}.

In the SI system expression 1.3 is rearranged as:

$$P = \rho g h \tag{1.4}$$

where g is the acceleration due to gravity (9.81 ms^{-2}) to give the pressure in pascal.

Pressure in a fluid can, however, be defined in terms of the *equivalent* head pressure. Common units are millimeters of mercury and centimeters, inches, feet or meters of water. The suffix *wg* (for *water gauge*) is often used when pressure is defined in terms of an equivalent head of water.

We live at the bottom of an ocean of air, and are consequently subject to a substantial pressure head from the weight of air above us. This pressure, some 15 psi, 1.05 kg fm^{-2}, or 101 kPa, is called an atmosphere, and is sometimes used as a unit of pressure.

It will be noted that 100 kPa is, for practical purposes, one atmosphere. As this is a convenient unit for many applications 100 kPa (10^5 Pa or 0.1 MPa)

has been given the name *bar*. Within the accuracy of instrumentation generally found in industry one bar is the same as one atmosphere.

There are three distinct ways in which pressure is measured, shown in Figure 1.6. Almost all pressure transducers or transmitters measure the pressure *difference* between two input ports. This is known as *differential pressure,* and the pressure transmitter in Figure 1.6a indicates a pressure of $P_1 - P_2$.

In Figure 1.6b the low-pressure input port is open to atmosphere, so the pressure transmitter indicates pressure above atmospheric pressure. This is known as *gauge pressure,* and is usually denoted by a 'g' suffix (e.g. psig). Gauge pressure measurement is almost universally used in hydraulic and pneumatic systems (and has been implicitly assumed in all previous discussions in this chapter).

Figure 1.6c shows the pressure transmitter measuring pressure with respect to a vacuum. This is known as *absolute pressure* and is of importance when the compression of gases is considered. The relationship between absolute and gauge pressure is illustrated in Figure 1.7. Pressure measurement and gas compression are discussed in later sections. Table 1.6 compares units of pressure. A typical hydraulic system operates at 150 bar, while typical pneumatic systems operate at 10 bar.

Work, energy and power

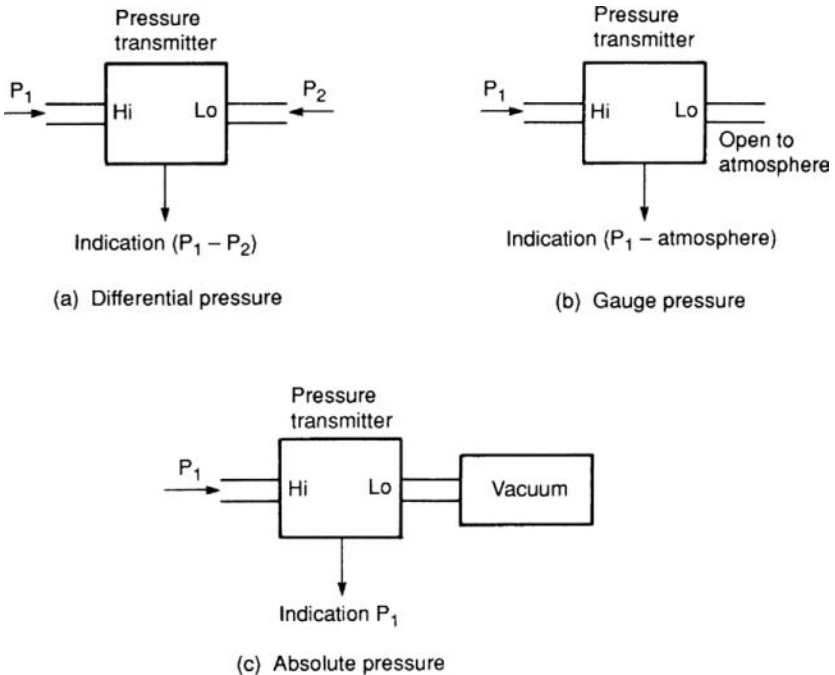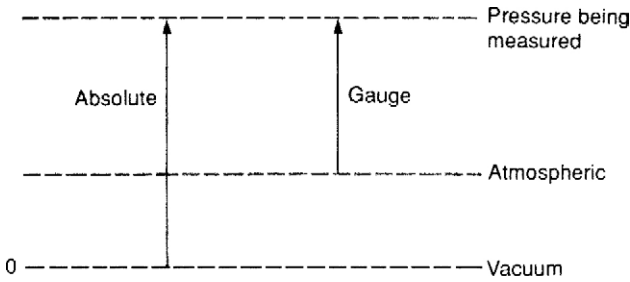Work is done (or energy is transferred) when an object is moved against a force, and is defined as:

(a) Differential pressure

(b) Gauge pressure

(c) Absolute pressure

FIGURE 1.6 Different forms of pressure measurement

FIGURE 1.7 Relationship between absolute and gauge pressures

$$\text{Work} = \text{force} \times \text{distance moved} \tag{1.5}$$

In the Imperial fps system expression 1.5 gives a unit of ft lb f. For metric systems the unit is cm kg f. The SI unit of work is the joule, where $1\,J = 1\,N\,m$ ($= 1\,m^2\,kg\,s^{-2}$). Table 1.8 compares these, and other, units of work.

Power is the rate at which work is performed:

$$\text{Power} = \frac{\text{work}}{\text{time}} \tag{1.6}$$

TABLE 1.6 Pressure

From	Multiply by	Comments
Kilopascal (kPa)	10^3	Practical unit
Atmosphere (at)	1.013×10^5	1 at approx 14.5 psi
Bar	10^5	1 bar approx 1 at
N m^{-2}	1	
kp cm^{-2}	9.807×10^3	
kg f m^{-2}	9.807	
kg f cm^{-2}	9.807×10^4	1 kg f cm^{-2} approx 1 at
Inches water gauge	248.9	in WG
mm water gauge	10.34	mm WG
Inches mercury	3385	in Hg
mm mercury (torr)	131	mm Hg
lb f ft^{-2}	47.88	
lb f in^{-2} (psi)	6895	

SI unit is the pascal: Pa
One pascal is defined as one newton force per square meter ($1\,Pa = 1\,N\,m^{-2} = 1\,kg\,m^{-1}\,s^{-2}$)
The pascal is not a very practical unit and common units in industry are the kilopascal (kPa), the bar and lb f in^{-2} (psi)

The SI unit of power is the watt, defined as $1\,\mathrm{J\ s^{-1}}$. This is by far the most common unit of power, as kW is almost universally used for the measurement of electrical power.

The Imperial system uses horse power (HP) which was used historically to define motor powers. One horse power is defined as $550\,\mathrm{ft\ lb\ f\ s^{-1}}$. Table 1.7 compares units of power.

A British Thermal Unit (Btu) is the energy required to raise the temperature of one pound of water by one degree Fahrenheit.

A calorie is the energy required to raise the temperature of one gram of water by one degree Centigrade.

A kilocalorie (often called a Calorie on food packaging) is 10^3 calories.

Work can be considered as the time integral of power (often described loosely as *total power used*). As electrical power is measured in watts or kilowatts $(1\,\mathrm{kW} = 10^3\,\mathrm{W})$, the kilowatt hour (kW h) is another representation of work or energy.

TABLE 1.7 Power

From	Multiply by	Comments
Watts	10^{-3}	
Horse power (HP)	0.7457	1 HP = 745.7 watts
German HP (ps)	0.7355	
French HP (ch, CV)	0.7355	
kcal s^{-1}	4.1868	
ft lb f s^{-1}	1.36×10^{-3}	
Btu s^{-1}	1.055	

SI unit is the kilowatt: kW

TABLE 1.8 Energy (also work and heat)

From	Multiply by	Comments
kW h	3.6×10^6	1000 watts × 60 min × 60 sec
Therm	1.055×10^8	1 therm = 10^5 Btu
British Thermal Unit (Btu)	1.055×10^3	
ft lb f	1.356	
ft poundal	0.0421	
calorie (cal)	4.187	

SI unit is the joule: J
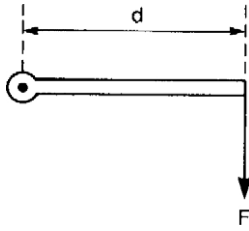One joule = one watt second

FIGURE 1.8 Definition of torque

Torque

The term *torque is* used to define a rotary force, and is simply the product of the force and the effective radius as shown in Figure 1.8. We thus have:

$$T = F \times d \tag{1.7}$$

In the Imperial system the unit is lb ft, in metric systems the unit is kg f m or kg f cm, and in SI the unit is N m.
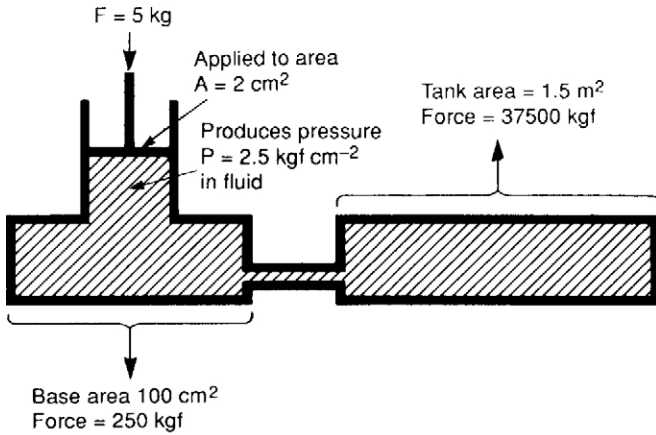
PASCAL'S LAW

Pressure in an enclosed fluid can be considered uniform throughout a practical system. There may be small differences arising from head pressures at different heights, but these will generally be negligible compared with the system operating pressure. This equality of pressure is known as *Pascal's law,* and is illustrated in Figure 1.9 where a force of 5 kg f is applied to a piston of area 2 cm². This produces a pressure of 2.5 kg f cm^{-2} at every point within the fluid, which acts with equal force per unit area on the walls of the system.

Suppose the base of the left hand tank is 0.1×0.1 m to give a total area of 100 cm². The total force acting on the base will be 250 kg f. If the top of the right hand tank is 1 m×1.5 m, a surprisingly large upwards force of 37,500 kg f is developed. Note, the size of the connecting pipe has no effect. This principle explains why it is possible to shear the bottom off a bottle by applying a small force to the cork, as illustrated in Figure 1.9b.
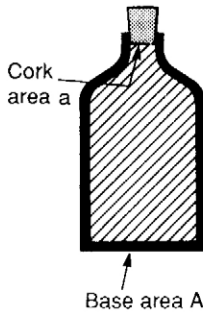
TABLE 1.9 Torque

From	Multiply by
kg f m	9.807
lb ft	1.356
lb in	0.1129
oz in	7.062×10^{-3}

SI unit is the newton meter: N m

F = 5 kg

Applied to area
A = 2 cm²

Produces pressure
P = 2.5 kgf cm⁻²
in fluid

Tank area = 1.5 m²
Force = 37500 kgf

Base area 100 cm²
Force = 250 kgf

(a) Forces and pressure in closed tanks

Cork
area a

Base area A

(b) Pressure in a bottle

FIGURE 1.9 Pressure in an enclosed fluid

The applied force develops a pressure, given by the expression:

$$P = \frac{f}{a} \qquad (1.8)$$

The force on the base is:

$$F = P \times A \qquad (1.9)$$

from which can be derived:

$$F = f \times \frac{A}{a} \qquad (1.10)$$

Expression 1.10 shows an enclosed fluid may be used to magnify a force. In Figure 1.10 a load of 2000 kg is sitting on a piston of area 500 cm² (about 12 cm radius). The smaller piston has an area of 2 cm². An applied force f given by:
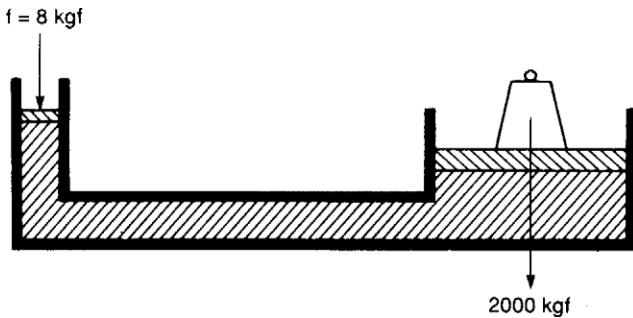
f = 8 kgf

2000 kgf

FIGURE 1.10 Mechanical advantage

$$f = 2000 \times \frac{2}{500} = 8 \text{ kgf} \tag{1.11}$$

will cause the 2000 kg load to rise. There is said to be a *mechanical advantage* of 250.

Energy must, however, be conserved. To illustrate this, suppose the left hand piston moves down by 100 cm (one meter). Because we have assumed the fluid is incompressible, a volume of liquid 200 cm² is transferred from the left hand cylinder to the right hand cylinder, causing the load to rise by just 0.4 cm. So, although we have a force magnification of 250, we have a movement reduction of the same factor. Because work is given by the product of force and the distance moved, the force is magnified and the distance moved reduced by the same factor, giving conservation of energy. The action of Figure 1.10 is thus similar to the mechanical systems of Figure 1.11, which also exhibit mechanical advantage.
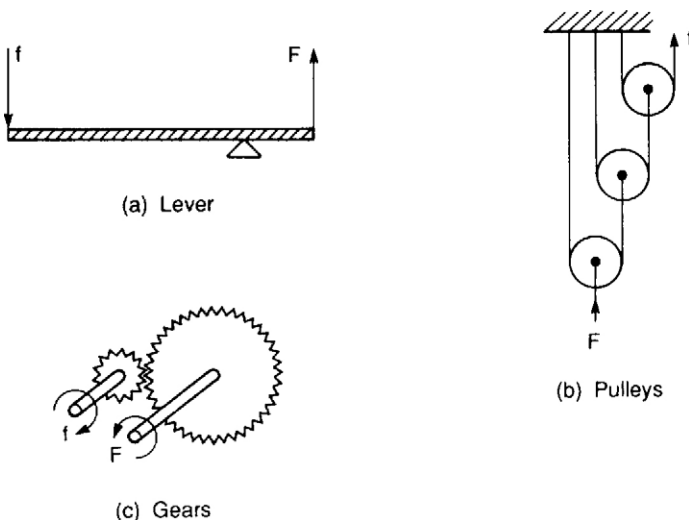
(a) Lever

(b) Pulleys

(c) Gears

FIGURE 1.11 Examples of mechanical advantage where a small input force f produces a larger output force F

(a) Raising the load

(b) Lowering the load

(c) Both valves open
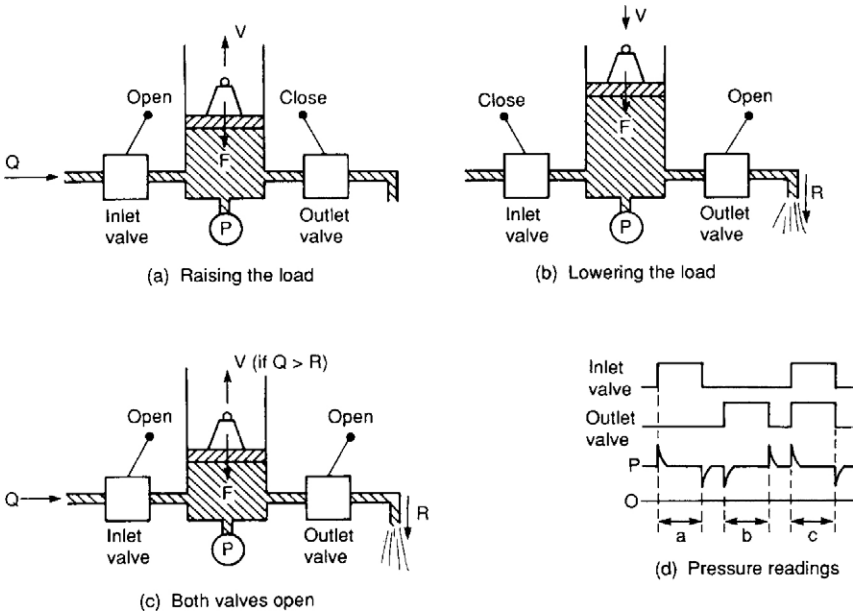
(d) Pressure readings

FIGURE 1.12   The relationships between force, pressure, flow and speed

The principle of Figure 1.10 is widely used where a large force is required with small movement. Typical examples are clamps, presses, hydraulic jacks and motor car brake and clutch operating mechanisms.

It should be noted that pressure in, say, a cylinder is determined solely by load and piston area in the steady state, and is not dependent on velocity of the piston once a constant speed has been achieved. Relationships between force, pressure, flow and speed are illustrated in Figure 1.12.

In Figure 1.12a, fluid is delivered to a cylinder at a rate of Q cm^3 s^{-1}. When the inlet valve is first opened, a pressure spike is observed as the load accelerates, but the pressure then settles back to a steady value of $P=F/A$ kg f cm^{-2}, where A is the area of the piston in cm^2 and F is measured in kg f. The load rises with a velocity $V=Q/A$ cm s^{-1} and velocity can obviously be controlled by adjusting flow rate Q.

In Figure 1.12b, the inlet valve has been closed, and the outlet valve opened allowing R cm^{-3} s^{-1} to flow out of the cylinder. There is again a pressure spike (negative this time) as the load accelerates downwards, but the pressure reverts to $P=F/A$ once the steady speed $V=R/A$ cm s^{-1} is achieved.

Finally, in Figure 1.12c both valves are open. The net flow is $(Q - R)$ giving a cylinder velocity $(Q - R)/A$, which can be positive (rising) or negative (falling) dependent on which flow is the largest. The steady state pressure, however, is unchanged at $P=F/A$.

PRESSURE MEASUREMENT

Behavior of a fluid can generally be deduced from measurements of flow or pressure. A flow transducer or transmitter has to be plumbed, in line, into a pipe, whereas pressure transmitters can be added non-intrusively as tappings to the side of a pipe. The basic fault-finding tool in both pneumatic or hydraulic systems is therefore a pressure gauge. Often this is a simple gauge which can be plugged into various parts of the system via a flexible connection.

These test pressure gauges invariably measure gauge pressure with the simple Bourdon pressure gauge shown in Figure 1.13. This consists of a flattened C-shaped tube which is fixed at one end, shown in Figure 1.13a. When pressure is applied to the tube it tends to straighten, with the free end moving up and to the right. For low pressure ranges a spiral tube is used to increase the sensitivity.

This movement is converted to a circular pointer movement by a mechanical quadrant and pinion. If an electrical output signal is required for remote indication, the pointer can be replaced by a potentiometer, as shown in Figure 1.13b.

Hydraulic and pneumatic systems tend to exhibit large pressure spikes as loads accelerate or decelerate (a typical example being shown in Figure 1.12c). These spikes can be irritating to the observer, can mislead, and in extreme cases could damage a pressure indicator. The response of a pressure sensor can be dampened by inclusion of a snubber restriction, as shown in Figure 1.13c.

Bourdon gauge-based transducers are generally robust but are low-accuracy (typically $\pm 2\%$) devices. As the limit of visual resolution of a pointer position is no better than $\pm 2\%$ anyway, ruggedness of these transducers makes them ideal for plant-mounted monitoring.

Where more accurate pressure measurement is required, transducers based on the force balance principle of Figure 1.14 are generally used. This is essentially a differential pressure transducer, in which the low-pressure inlet (LP) is left open to atmosphere and the high-pressure (HP) inlet connects to the system. The signal given (HP – LP) is thus gauge pressure.

A pressure increase in the system deflects the pressure-sensitive diaphragm to the left. This movement is detected by the displacement transducer which, via a servo amplifier, leads to an increase in current in the balance coil.

Because the force from the balance coil always exactly balances the force arising from the pressure difference between LP and HP, current through the transducer is directly proportional to the differential pressure.

Remote indicating transducers are generally arranged with a remote power supply and the indicator and/or recorder connected into one line as in Figure 1.15 to give a two-wire system. A signal range of 4–20 mA is commonly used, with the 4 mA zero level providing a current supply for the transducer's servo amplifier and also indicating circuit continuity (0 mA indicating an open circuit fault condition).
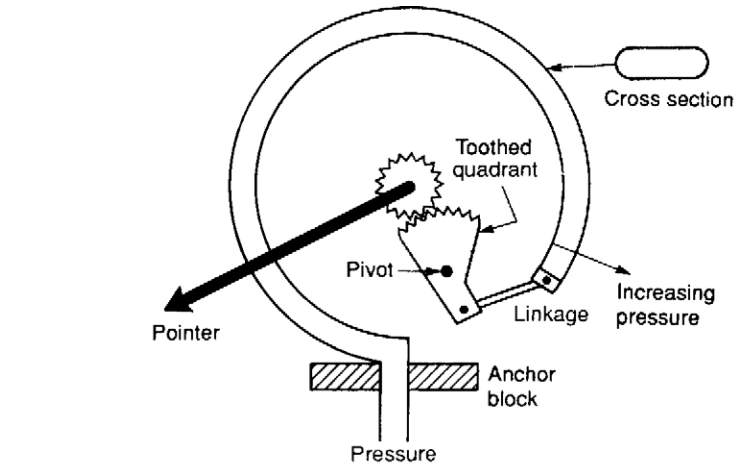
(a) Bourdon tube gauge construction
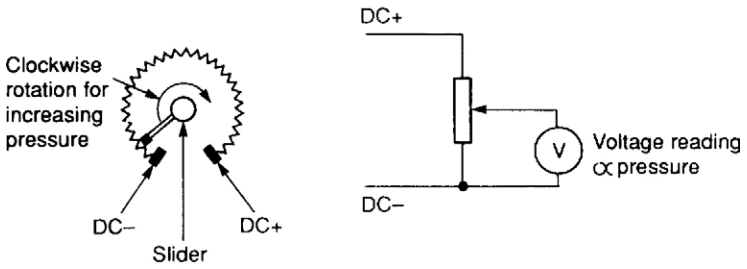
(b) Electrical signal from Bourdon gauge
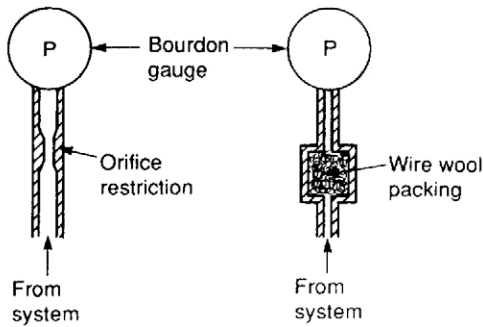
(c) Snubber restrictions
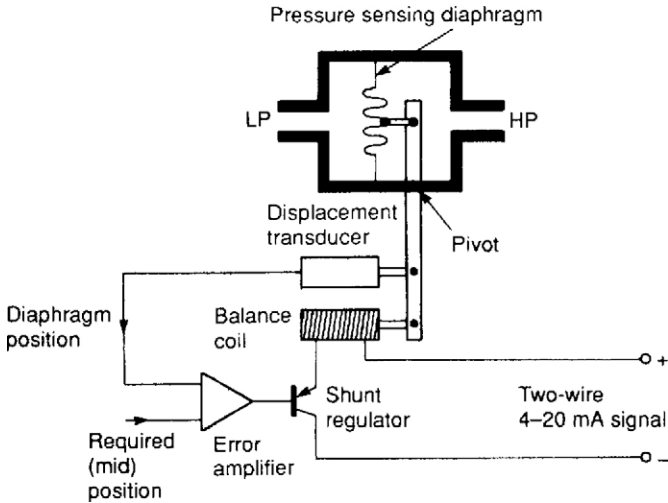
FIGURE 1.13 The Bourdon pressure gauge

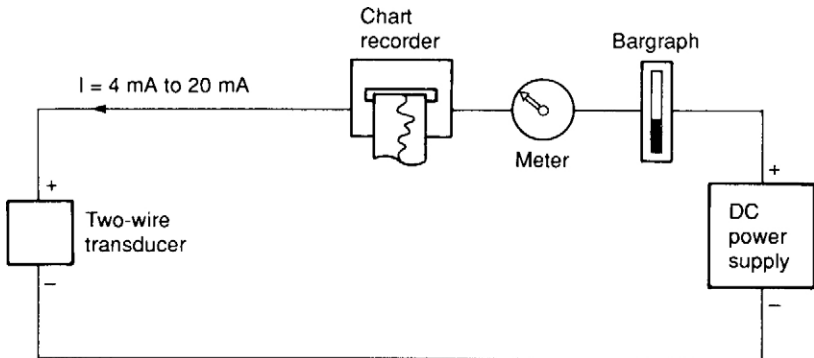**FIGURE 1.14** Force balance pressure transducer

FIGURE 1.15 Advantages of two-wire transducers

FLUID FLOW

Hydraulic and pneumatic systems are both concerned with the flow of a fluid (liquid or gas) down a pipe. Flow is a loose term that generally has three distinct meanings:

- *Volumetric flow is* used to measure volume of fluid passing a point per unit of time. Where the fluid is a compressible gas, temperature and pressure must be specified or flow normalized to some standard temperature and pressure (a topic discussed later). Volumetric flow is the most common measurement in process control.
- *Mass flow* measures the mass of fluid passing the point in unit time.
- *Velocity of flow* measures linear speed (in m s^{-1}, say) past the point of measurement. Flow velocity is of prime importance in the design of hydraulic and pneumatic systems.

TABLE 1.10 Volumetric flow rate

From	Multiply by
Imperial gallons/min	7.756×10^{-5}
Imperial gallons/hour	1.263×10^{-6}
US gallons/min	6.309×10^{-5}
ft^3/sec	2.831×10^{-2}
ft^3/min (cfm)	4.719×10^{-4}
m^3/min	1.667×10^{-4}
$m^3/hour$	2.778×10^{-8}
liters/sec	10^{-3}
liters/min (lpm)	1.667×10^{-5}
SI unit: cubic meters per sec, $m^3 s^{-1}$. This is a rather impractical unit	

Types of fluid flow are illustrated in Figure 1.16. At low flow velocities, the flow pattern is smooth and linear with low velocities at the pipe walls and the highest flow at the center of the pipe. This is known as *laminar* or *streamline* flow.

As flow velocity increases, eddies start to form until at high flow velocities complete turbulence results as shown in Figure 1.16b. Flow velocity is now virtually uniform across the pipe.

The nature of the flow is determined by the *Reynolds number,* R_c, given by the expression:

$$R_c = \frac{vd\rho}{\eta} \tag{1.12}$$

where v is flow velocity, d is pipe diameter, ρ the fluid density and η the viscosity. The Reynolds number is a ratio and hence dimensionless. If $R_c < 2000$, flow is laminar. If $R_c > 10^5$, flow is turbulent.

A turbulent flow is generally preferred for products in process control as it simplifies volumetric flow measurement (with differential pressure flowmeters – see later). Turbulent flow, however, increases energy loss through friction
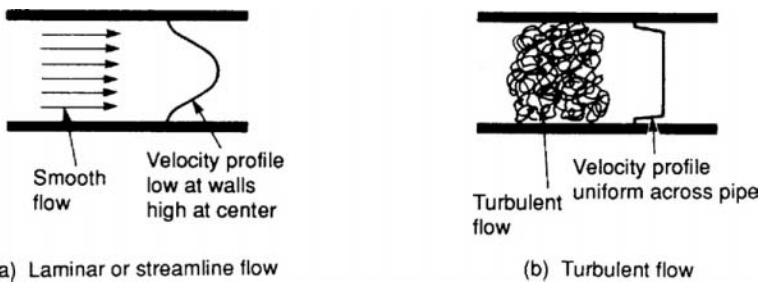

(a) Laminar or streamline flow   (b) Turbulent flow

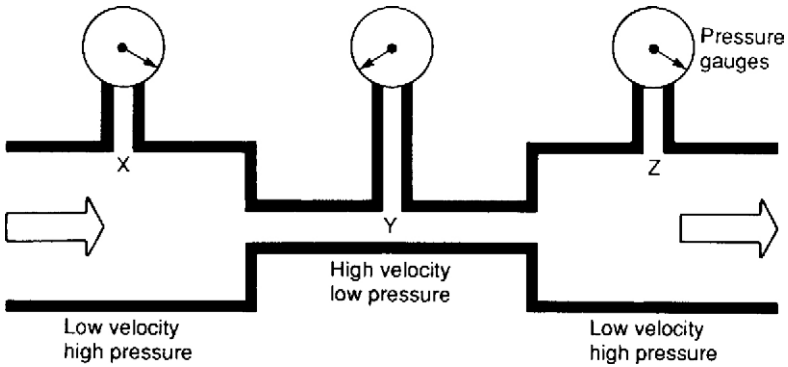**FIGURE 1.16** Types of fluid flow

FIGURE 1.17 Relationship between flow and pressure

and may lead to premature wear. Cavitation (formation and collapse of vapor bubbles) occurs with turbulent liquid flow and may result in pitting on valve surfaces.

Energy in a unit mass of fluid has three components:
- kinetic energy from its motion, given by $v^2/2$ where v is flow velocity;
- potential energy from the height of the fluid;
- energy arising from the pressure of the fluid, given by P/ρ where P is the pressure and ρ the density.

Fluid is passing along a pipe in Figure 1.17. Neglecting energy losses from friction, energies at points X, Y and Z will be equal. The flow velocity at point Y, however, is higher than at points X and Z because of the smaller pipe diameter. Potential energy at each point is constant because the pipe is horizontal, so we can write:

$$\underset{\text{Energy at X}}{\frac{v_x^2}{2} + \frac{P_x}{\rho}} = \underset{\text{Energy at Y}}{\frac{v_y^2}{2} + \frac{P_y}{\rho}} = \underset{\text{Energy at Z}}{\frac{v_z^2}{2} + \frac{P_z}{\rho}} \qquad (1.13)$$

We have implied an incompressible fluid by assuming the density, ρ, is constant throughout. Expression 1.13 becomes more complicated for a gas as different densities have to be used at each point.

The net result of the expression is fluid pressure falls as flow velocity rises. Note, though, that the pressure recovers as flow velocity falls again at point Z.

The simplest method of measuring flow (known as a variable area flowmeter) uses a float in a vertical tube arranged as in Figure 1.18. The obstruction of the float causes a local increase in the fluid velocity which causes a differential pressure drop across the float, resulting in an upward force. The weight of the float obviously causes a downward force. The float therefore rises or falls depending on which force is the largest. The area around the float, however, increases the higher the float rises because of the tube taper. This increase in area decreases the pressure drop across the float and the upwards force. The float therefore settles at a vertical position where the weight of the float and the
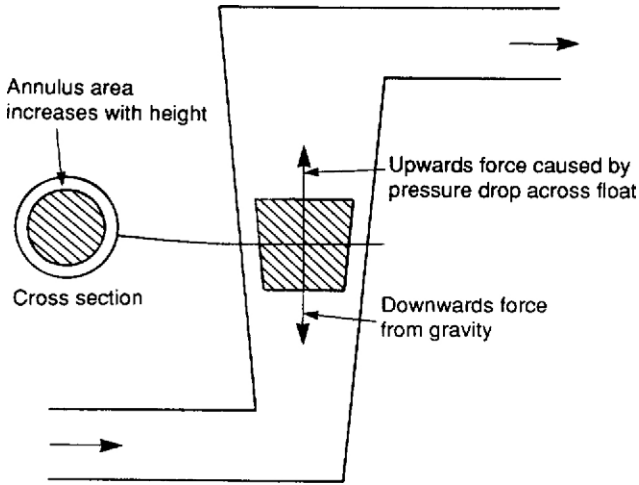
FIGURE 1.18 Variable area flowmeter

upwards force from the differential pressure exactly match. Flow rate can therefore be determined from the float position.

A remote indicating flowmeter can be constructed from a pipe-mounted turbine, as shown in Figure 1.19. Fluid flow causes the propeller to rotate, its rotational speed being proportional to flow rate. Blade rotation is counted electronically by an external inductive proximity detector to give an electrical signal for remote indication of the flow rate.

Finally, the classical method of measuring flow returns directly to expression 1.13 by locally increasing flow velocity with a deliberately introduced restriction, as shown in Figure 1.20. Typical obstructions are an orifice plate or a venturi. These increase flow velocity, causing a pressure drop which can be measured to give a differential pressure related to the flow. Unfortunately,
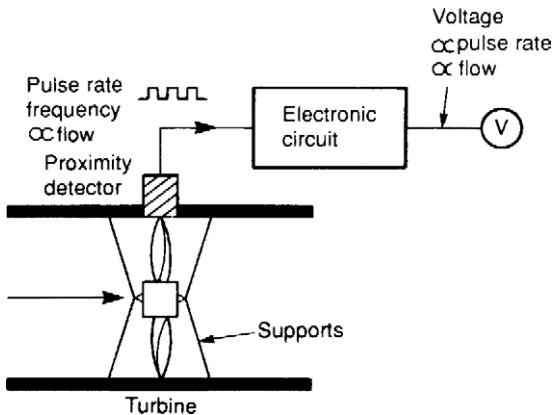
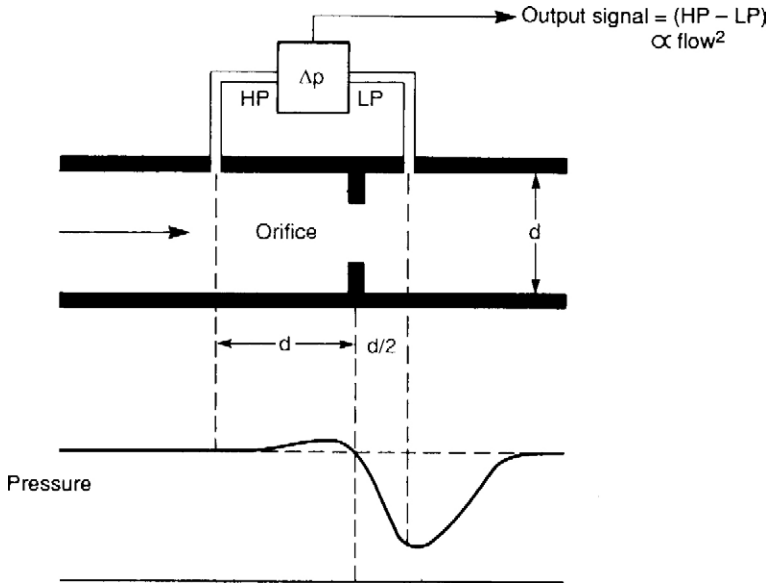**FIGURE 1.19** Turbine flowmeter

FIGURE 1.20 Orifice plate flowmeter

the differential pressure is proportional to the square of the flow rate, so a linearizing square root extractor circuit is required to give a linear signal. Although differential pressure flow measurement is widely used to measure the flow rates of process material, the technique is not widely used in hydraulic and pneumatic systems.

It will be apparent that all flow measurement systems are intrusive to various degrees, and cannot be tapped in as easily as pressure measurement can. Fault finding in hydraulic and pneumatic systems is therefore generally based on pressure readings at strategic points.

TEMPERATURE

Fluid behavior is determined to some extent by its temperature. A later section discusses the relationship between pressure and temperature in a gas.

Temperature scales

A temperature scale is established by choosing two observable physical effects which are dependent upon temperature and assigning numerical values to them. The Fahrenheit and Celsius (previously known as Centigrade) scales use the freezing and boiling points of water as the two reference points:

	Fahrenheit	Celsius
Freezing point	32	0
Boiling point	212	100

From which:

$$F = \left(9 \times \frac{C}{5}\right) + 32 \qquad (1.14)$$

and:

$$C = (F - 32) \times \frac{5}{9} \qquad (1.15)$$

The SI unit of temperature is the Kelvin. This defines the lowest theoretical temperature (called *absolute zero*) as 0 K, and the triple point of water (0.01 °C) as 273.16 K. It should be noted that temperatures in Kelvin do not use the degree (°) symbol. These apparently odd numerical values make a temperature change of 1 K the same as 1 °C, and:

$$K = °C + 273.1 \qquad (1.16)$$

The Celsius scale is most widely used in industry, but the Kelvin scale is important in determining the changes in gas pressure or volume with temperature.

Temperature measurement

There are four basic ways of measuring temperature based on temperature-dependent physical properties.

Expansion of a substance with temperature can be used to produce a change in volume, length or pressure. This is probably the most common type of temperature measurement in the form of mercury or alcohol-in-glass thermometers. A variation is the bimetallic strip shown in Figure 1.21, where two dissimilar metals have different coefficients of expansion which cause the strip to bend according to the temperature. This technique is the basis of most on/off thermostats used for temperature control or alarm annunciation. A bimetallic spiral can be used to construct an indicating thermometer.

Electrical resistance changes with temperature. A platinum wire with resistance 100 ohms at 0 °C will have a resistance of 138.5 ohms at 100 °C. Temperature sensors based on this principle are known as RTDs (for resistance temperature detector) or PT100 sensors (from PT, for platinum, and 100 for 100 ohms at 0 °C). Semiconductor devices called thermistors have more dramatic changes, the char-
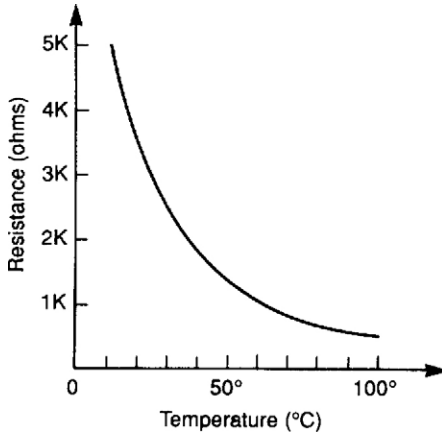

FIGURE 1.21 Bimetallic strip

FIGURE 1.22 Typical resistance temperature curve for NTC thermistor
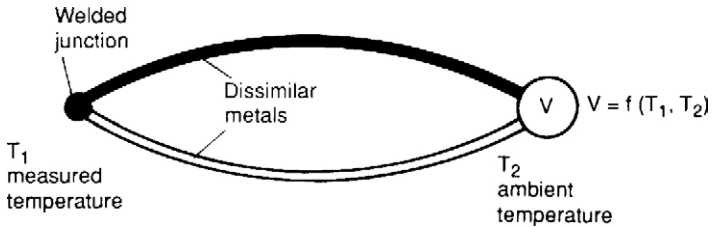

FIGURE 1.23 The thermocouple

acteristics of a typical device being shown in Figure 1.22. The response, however, is non-linear which makes thermistors more suitable for alarm/control application than temperature indication.

Thermocouples, the principle of which is shown in Figure 1.23, use the small difference in contact potentials between different metals to give a voltage which depends on the temperature difference between the measurement and reference points. Although widely used in process control, the technique is rarely encountered in pneumatic and hydraulic systems.

The final method, called pyrometry, uses the change in radiated energy with temperature. As this has a minimum temperature measurement of about 400 °C, it is totally unsuitable for the systems we shall be discussing.

GAS LAWS

For all practical purposes, liquids used in hydraulic systems can be considered incompressible and insensitive to changes in temperature (provided the temperature remains within some quite broad limits). The gas in a pneumatic system is very sensitive to changes in pressure and temperature, and its behavior is determined by the gas laws described below.

In the following expressions it is important to note that pressures are given in absolute, not gauge, terms and temperatures are given in absolute degrees
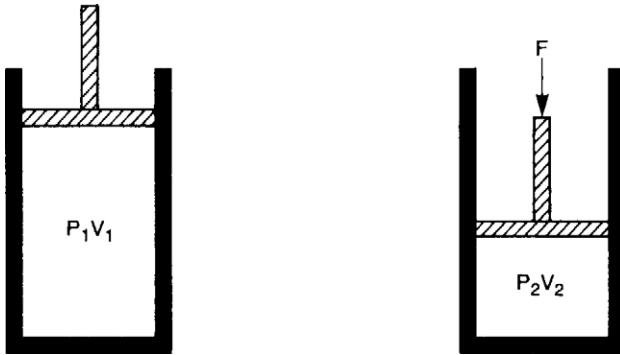
FIGURE 1.24 Boyle's law

Kelvin, not in degrees Celsius. If we discuss, say, a liter of air at atmospheric pressure and 20 °C being compressed to three atmospheres gauge pressure, its original pressure was one atmosphere, its original temperature was 293 K and its final pressure is four atmospheres absolute.

Pressure and volume are related by Boyle's law. In Figure 1.24 we have a volume of gas V_1 at pressure P_1 (in absolute units, remember). This gas is compressed to volume V_2, which will result in a rise of pressure to P_2, where:

$$P_1 V_1 = P_2 V_2 \tag{1.17}$$

provided the temperature of the gas does not change during the compression. A reduction of pressure similarly leads to an increase in volume.

In practice, compression of a gas is always accompanied by a rise in temperature (as is commonly noticed when pumping up a bicycle tire) and a reduction in pressure produces a temperature fall (the principle of refrigeration). For expression 1.17 to apply, the gas must be allowed to return to its original temperature.

In Figure 1.25, on the other hand, the temperature of a fixed volume of gas is controlled by a heater. A rise in temperature from T_1 to T_2 results in an increase in pressure from P_1 to P_2, where:

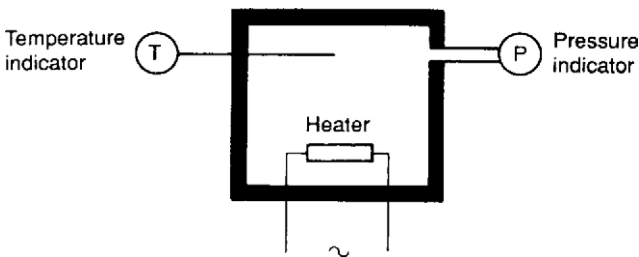$$\frac{P_1}{T_1} = \frac{P_2}{T_2} \tag{1.18}$$

FIGURE 1.25 Relationship between temperature and pressure

Again it should be remembered pressure and temperature are in absolute terms. Although expression 1.18 gives the change in pressure resulting from a change in temperature, it also applies to changes in temperature resulting from a change in pressure provided no heat is lost from the system. In a pneumatic air compressor, the temperature of the outgoing compressed air is considerably elevated by the increase in pressure, resulting in the need for the compressor to be followed by an air cooler.

Expressions 1.17 and 1.18 are combined to give the general gas law:

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

(1.19)

where P_1, V_1, T_1 are initial conditions and P_2, V_2, T_2 are final conditions. As before, expression 1.19 assumes no heat is lost to, or gained from, the environment.

Chapter | two

Hydraulic Pumps and Pressure Regulation

A hydraulic pump (Figure 2.1) takes oil from a tank and delivers it to the rest of the hydraulic circuit. In doing so it raises oil pressure to the required level. The operation of such a pump is illustrated in Figure 2.1a. On hydraulic circuit diagrams a pump is represented by the symbol in Figure 2.1b, with the arrowhead showing the direction of flow.

Hydraulic pumps are generally driven at constant speed by a three-phase AC induction motor rotating at 1500/1000 rpm in the UK (with a 50 Hz supply) and

(a) Operation of a pump

(b) Pump symbol, arrow shows direction of flow
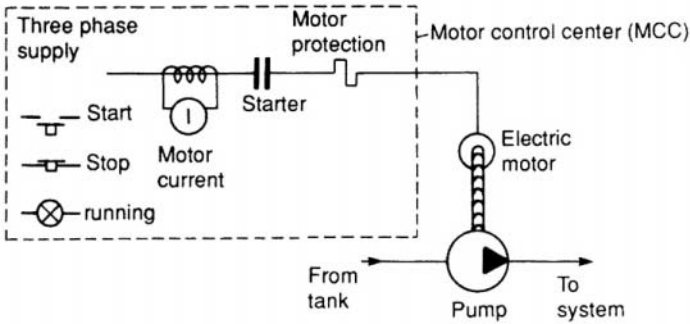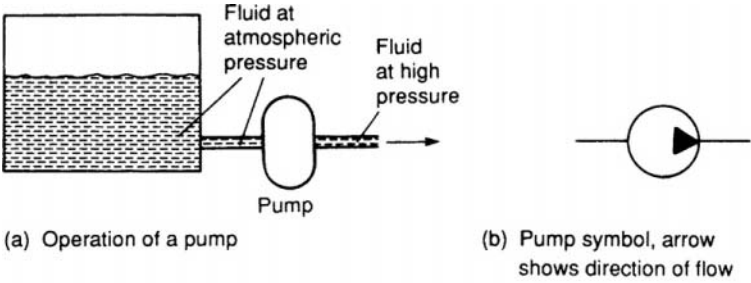
(c) Pump associated components

FIGURE 2.1 The hydraulic pump
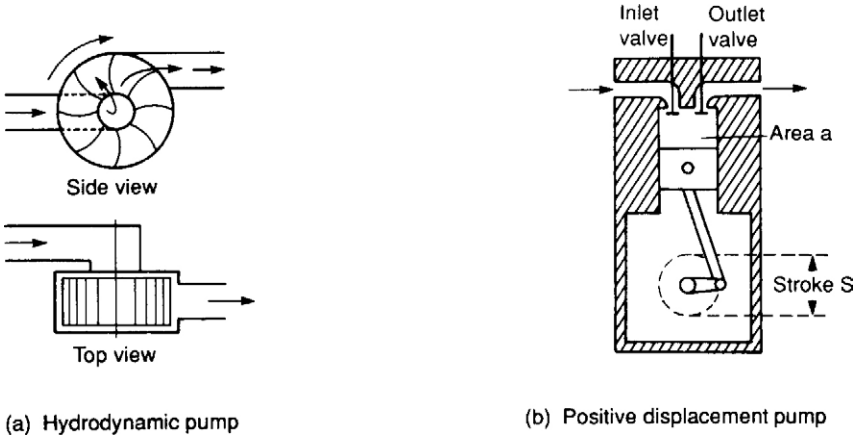
(a) Hydrodynamic pump

(b) Positive displacement pump

FIGURE 2.2 Types of hydraulic pump

at 1200/1800 rpm in the USA (with a 60 Hz supply). Often pump and motor are supplied as one combined unit. As an AC motor requires some form of starter, the complete arrangement illustrated in Figure 2.1c is needed.

There are two types of pump (for fluids) or compressor (for gases), illustrated in Figure 2.2. Typical of the first type is the centrifugal pump of Figure 2.2a. Fluid is drawn into the axis of the pump, and flung out to the periphery by centrifugal force. Flow of fluid into the load maintains pressure at the pump exit. Should the pump stop, however, there is a direct route from outlet back to inlet and the pressure rapidly decays away. Fluid leakage will also occur past the vanes, so pump delivery will vary according to outlet pressure. Devices such as that shown in Figure 2.2a are known as hydrodynamic pumps, and are primarily used to shift fluid from one location to another at relatively low pressures. Water pumps are a typical application.

Figure 2.2b shows a simple piston pump called a positive displacement or hydrostatic pump. As the piston is driven down, the inlet valve opens and a volume of fluid (determined by the cross-section area of the piston and the length of stroke) is drawn into the cylinder. Next, the piston is driven up with the inlet valve closed and the outlet valve open, driving the same volume of fluid to the pump outlet.

Should the pump stop, one of the two valves will always be closed, so there is no route for fluid to leak back. Exit pressure is therefore maintained (assuming there are no downstream return routes).

More important, though, is the fact that the pump delivers a fixed volume of fluid from inlet to outlet each cycle, regardless of pressure at the outlet port. Unlike the hydrodynamic pump described earlier, a piston pump has no inherent maximum pressure determined by pump leakage: if it drives into a dead-end load with no return route (as can easily occur in an inactive hydraulic system with all valves closed) the pressure rises continuously with each pump stroke until either piping or the pump itself fails.

Hydraulic pumps are invariably hydrostatic and, consequently, require some method of controlling system pressure to avoid catastrophic pipe or pump failure. This topic is discussed further in a later section.

A hydraulic pump is specified by the flow rate it delivers, called the displacement (usually given in liters min^{-1}, gallons min^{-1} or liters per rev), and the maximum pressure the pump can withstand. These are normally called the pump capacity (or delivery rate) and the pressure rating.

Pump data sheets specify required drive speed (usually 1500/1000 rpm (50 Hz) or 1800/1200 (60 Hz) corresponding to the common standard speeds of a three-phase induction motor). Pump displacement is directly related to drive speed; at a lower than specified speed, pump displacement is reduced and pump efficiency falls as fluid leakage (called slippage) increases. Pump displacement cannot, on the other hand, be expected to increase by increasing drive speed, as effects such as centrifugal forces, frictional forces and fluid cavitation will drastically reduce service life.

Like any mechanical device, pumps are not 100% efficient. The efficiency of a pump may be specified in two ways. First, volumetric efficiency relates actual volume delivered to the theoretical maximum volume. The simple piston pump of Figure 2.2b, for example, has a theoretical volume of A × s delivered per stroke, but in practice the small overlap when both inlet and outlet valves are closed will reduce the volume slightly.

Second, efficiency may be specified in terms of output hydraulic power and input mechanical (at the drive shaft) or electrical (at the motor terminals) power.

Typical efficiencies for pumps range from around 90% (for cheap gear pumps) to about 98% for high-quality piston pumps. An allowance for pump efficiency needs to be made when specifying pump capacity or choosing a suitable drive motor.

The motor power required to drive a pump is determined by the pump capacity and working pressure. From expression 1.6:

$$Power = \frac{work}{time}$$
$$= \frac{force \times distance}{time} \qquad (2.1)$$

In Figure 2.3, a pump forces fluid along a pipe of area A against a pressure P, moving fluid a distance d in time T. The force is PA, which, when substituted into expression 2.1, gives:

$$Power = \frac{P \times A \times d}{T}$$

but A×d/T is flow rate, hence:

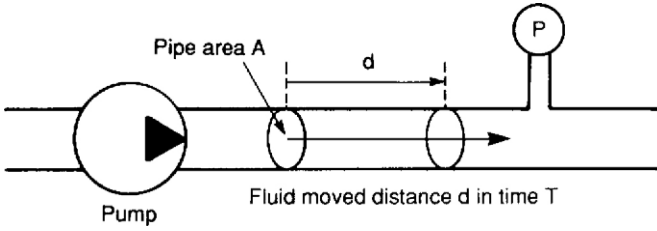$$Power = pressure \times flow\ rate \qquad (2.2)$$

FIGURE 2.3 Derivation of pump power

Unfortunately, expression 2.2 is specified in impractical SI units (pressure in pascals, time in seconds, flow in cubic meters). We may adapt the expression to use more practical units (pressure in bar, flow rate in liters min^{-1}) with the expression:

$$Power = \frac{Pressure \times flow\ rate}{600}\ kW \qquad (2.3)$$

For Imperial systems (pressure in psig, flow rate in gallons min^{-1}), the expression becomes:

$$Power = \frac{pressure \times flow\ rate}{1915}\ kW \qquad (2.4)$$

For fully Imperial systems, motor power in horsepower can be found from:

$$Horsepower = 0.75 \times power\ in\ kW \qquad (2.5)$$

Hydraulic pumps such as that in Figure 2.1 do not require priming because fluid flows, by gravity, into the pump inlet port. Not surprisingly this is called a self-priming pump. Care must be taken with this arrangement to avoid sediment from the tank being drawn into the pump.

The pump in Figure 2.4 is above the fluid in the tank. The pump creates a negative (less than atmospheric) pressure at its inlet port, causing fluid to be pushed up the inlet pipe by atmospheric pressure. This action creates a fluid lift which is, generally, incorrectly described as arising from pump suction. In reality fluid is *pushed* into the pump.
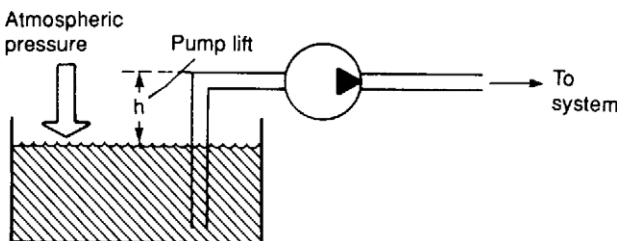
FIGURE 2.4 Pump lift

Maximum pump lift is determined by atmospheric pressure and is given by expressions 1.3 and 1.4. In theory a lift of about 8 m is feasible but, in practice, would be accompanied by undesirable side effects such as cavitation (formation and destructive collapse of bubbles from partial vaporization of fluid). The lift should be as small as possible and around 1 m is a normal practical limit.

Fluid flow in the inlet line always takes place at negative pressure, and a relatively low flow velocity is needed to reduce these side effects. The design should aim for a flow velocity of around 1 m s^{-1}. Examination of any hydraulic system will always reveal pump inlet pipes of much larger diameters than outlet pipes.

PRESSURE REGULATION

Figure 2.5a shows the by now familiar system where a load is raised or lowered by a hydraulic cylinder. With valve V_1 open, fluid flows from the pump to the cylinder, with both pressure gauges P_1 and P_2 indicating a pressure of F/A. With valves V_1 closed and V_2 open, the load falls with fluid being returned to the tank. With the load falling, gauge P_2 will still show a pressure of F/A, but at P_1 the pump is dead-ended, leading to a continual increase in pressure as the pump delivers fluid into the pipe.

Obviously some method is needed to keep P_1 at a safe level. To achieve this, pressure-regulating valve V_3 has been included. This is normally closed (no connection between P and T) while the pressure is below some preset level

(a) Circuit diagram

(b) Pressure recording
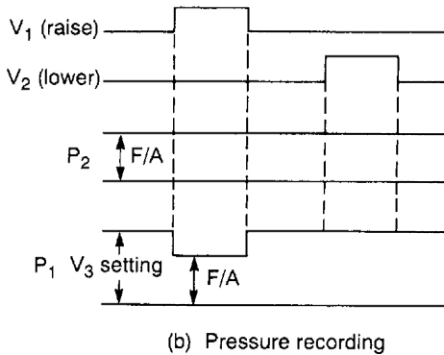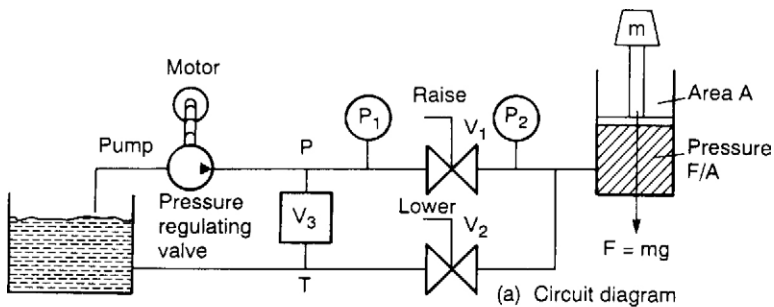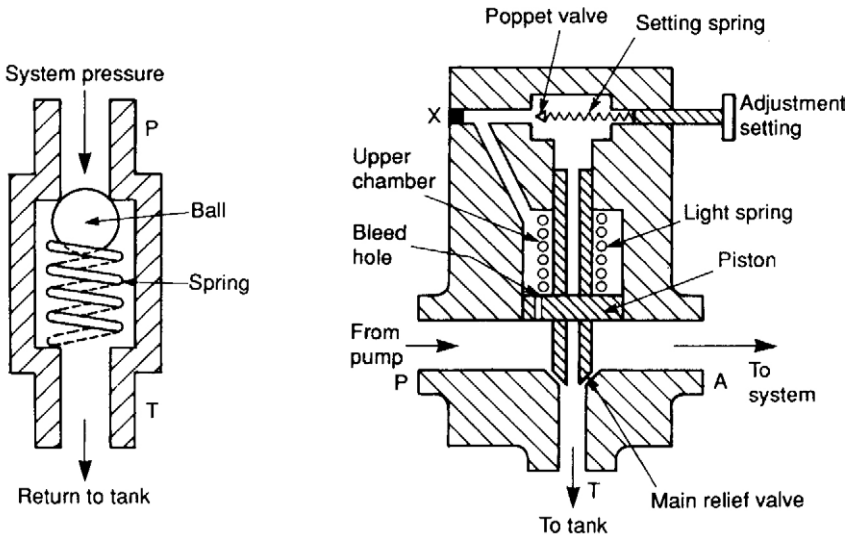
FIGURE 2.5 Action of pressure regulation

(called the cracking pressure). Once the cracking pressure is reached valve V_3 starts to open, bleeding fluid back to the tank. As the pressure increases, valve V_3 opens more until, at a pressure called the full flow pressure, the valve is fully open. With valve V_1 closed, all fluid from the pump returns to the tank via the pressure-regulating valve, and P_1 settles somewhere between the cracking and full flow pressures.

Cracking pressure of a relief valve *must* be higher than a system's working pressure, leading to a fall in system pressure as valve V_1 opens and external work is performed. Valve positions and consequent pressure readings are shown in Figure 2.5b.

The simplest form of pressure regulation valve is the ball and spring arrangement of Figure 2.6a. System pressure in the pipe exerts a force of $P \times a$ on the ball. When the force is larger than the spring compressive force the valve will crack open, bypassing fluid back to the tank. The higher the pipe pressure, the more the valve opens. Cracking pressure is set by the spring compression and in practical valves this can be adjusted to suit the application.

The difference between cracking and full flow pressure is called the pressure override. The steady (non-working) system pressure will lie somewhere within the pressure override, with the actual value determined by pipe sizes and characteristics of the pressure-regulating valve itself.

If the quiescent pressure is required to be precisely defined, a small pressure override is needed. This pressure override is related to spring tension in a simple relief valve. When a small, or precisely defined, override is required, a balanced piston relief valve (shown in Figure 2.6b) is used.

(a) Simple regulator   (b) Balanced piston relief valve

FIGURE 2.6 Pressure regulation

The piston in this valve is free moving, but is normally held in the lowered position by a light spring, blocking flow to the tank. Fluid is permitted to pass to the upper chamber through a small hole in the piston. The upper chamber is sealed by an adjustable spring-loaded poppet. In the low-pressure state, there is no flow past the poppet, so pressures on both sides of the piston are equal and spring pressure keeps the valve closed.

When fluid pressure rises, the poppet cracks and a small flow of fluid passes from the upper chamber to the tank via the hole in the piston center. This fluid is replenished by fluid flowing through the hole in the piston. With fluid flow there is now a pressure differential across the piston, which is acting only against a light spring. The whole piston lifts, releasing fluid around the valve stem until a balance condition is reached. Because of the light-restoring spring a very small override is achieved.

The balanced piston relief valve can also be used as an unloading valve. Plug X is a vent connection and, if removed, fluid flows from the main line through the piston. As before, this causes the piston to rise and flow to be dumped to the tank. Controlled loading/unloading can be achieved by the use of a finite position valve connected to the vent connection.

When no useful work is being performed, *all* fluid from the pump is pressurized to a high pressure then dumped back to the tank (at atmospheric pressure) through the pressure-regulating valve. This requires motor power defined earlier by expressions 2.3 and 2.4, and represents a substantial waste of power. Less obviously, energy put into the fluid is converted to heat, leading to a rise in fluid temperature. Surprisingly, motor power will be higher when no work is being done because cracking pressure is higher than working pressure.

This waste of energy is expensive, and can lead to the need for heat exchangers to be built into the tank to remove the excess heat. A much more economic arrangement uses loading/unloading valves, a topic discussed further in a later section.

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 center, 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.
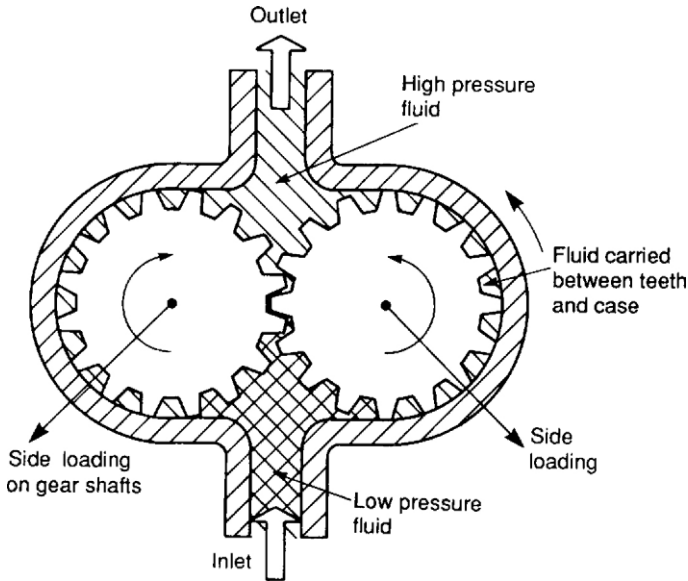
FIGURE 2.7 Gear pump

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. Gear pumps are compact and cheap but are noisy.

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. The operational method of a gear pump means that high displacements can be achieved by multiple parallel gear assemblies driven from a single shaft and motor. 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 center line as shown. Volumetric efficiency of gear pumps at 90% is lowest of the three basic 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, a lobe pump.

Figure 2.9a 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 molding. Yet another variation on the theme is the gerotor pump of Figure 2.9b, where the crescent molding is dispensed with by using an internal gear with one less tooth than the outer gear wheel.
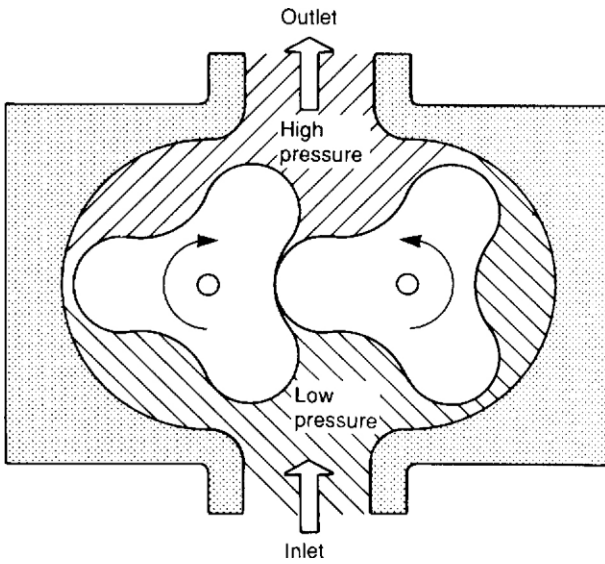
FIGURE 2.8 The lobe pump
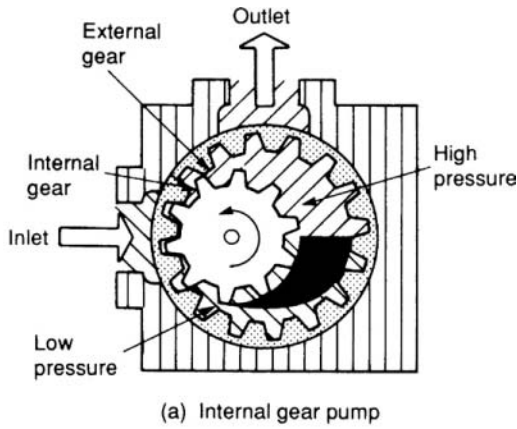
(a) Internal gear pump
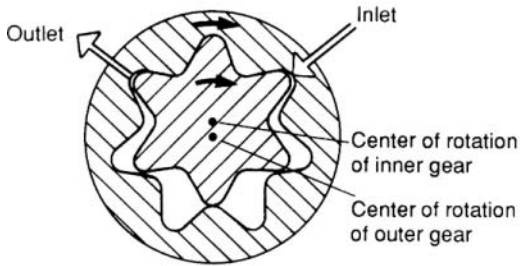
(b) Gerotor pump

FIGURE 2.9 Further forms of gear 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. Centrifugal force also helps to seal the vane tips. Vane pumps are much quieter than gear pumps.

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 hous-

(a) Unbalanced vane pump

(b) Balanced vane pump

FIGURE 2.10   Vane pumps

ing itself. There is still, however, leakage between rotor faces and body sides. Pump displacement is determined by vane throw, vane cross-sectional area and speed of rotation. Pump displacement can again be increased by having multiple pump assemblies driven from a single shaft and motor. Many manufacturers build vane pumps with pump assemblies as replaceable cartridges which can be quickly changed.

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.

Displacement 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 displacement 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 displacement 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 displacements and the user has to choose the next largest size. Figure 2.11 shows a vane pump with adjustable displacement, set by the positional relationship between rotor and inner casing, with the inner casing position set by an external screw.
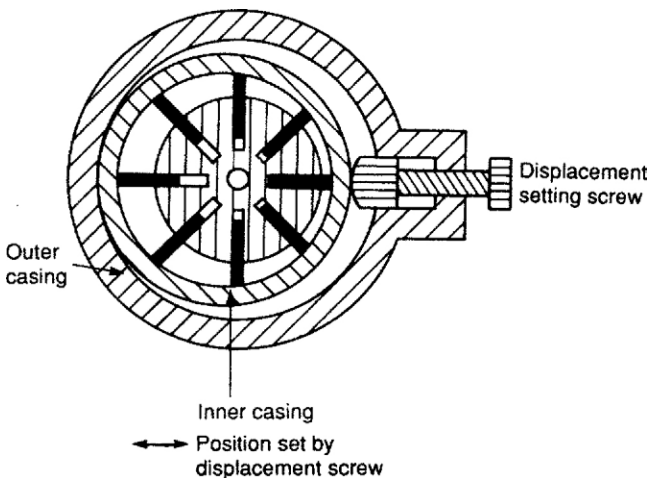
FIGURE 2.11 Variable displacement vane pump

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 cylinders and pistons to smooth out fluid delivery, and much ingenuity goes into designing multicylinder pumps which are surprisingly compact.

The displacement of a piston pump can be easily calculated:

Q = (number of pistons) × (piston area) × (piston stroke) × (drive speed)

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.

The pump of Figure 2.13 uses the same principle, but employs a stationary cam and a rotating cylinder block. This arrangement does not require multiple inlet and outlet valves and is consequently simpler, more reliable, and cheaper. Not surprisingly most radial piston pumps have this construction. Like gear and vane pumps, radial piston pumps can provide increased displacement by the use of multiple assemblies driven from a common shaft.

An alternative form of piston pump is the axial design of Figure 2.14, where multiple pistons are arranged in a rotating cylinder. The pistons are stroked by a fixed angled plate called the swash plate. Each piston can be kept in contact with the swash plate by springs or by a rotating shoe plate linked to the swash plate.
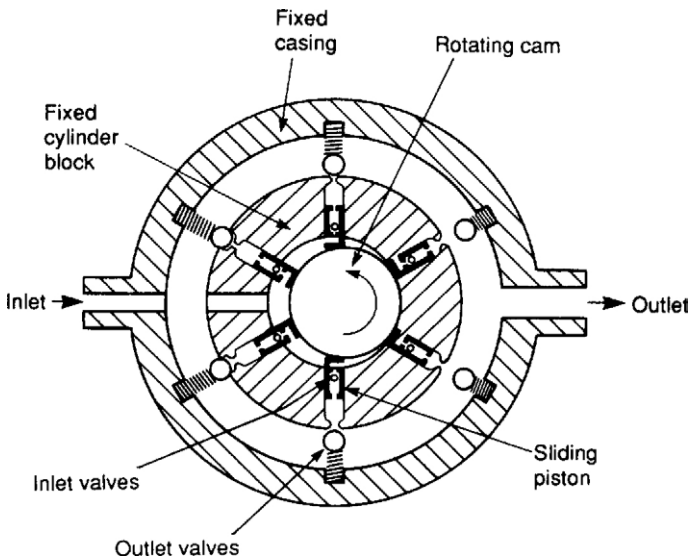
FIGURE 2.12 Radial piston pump
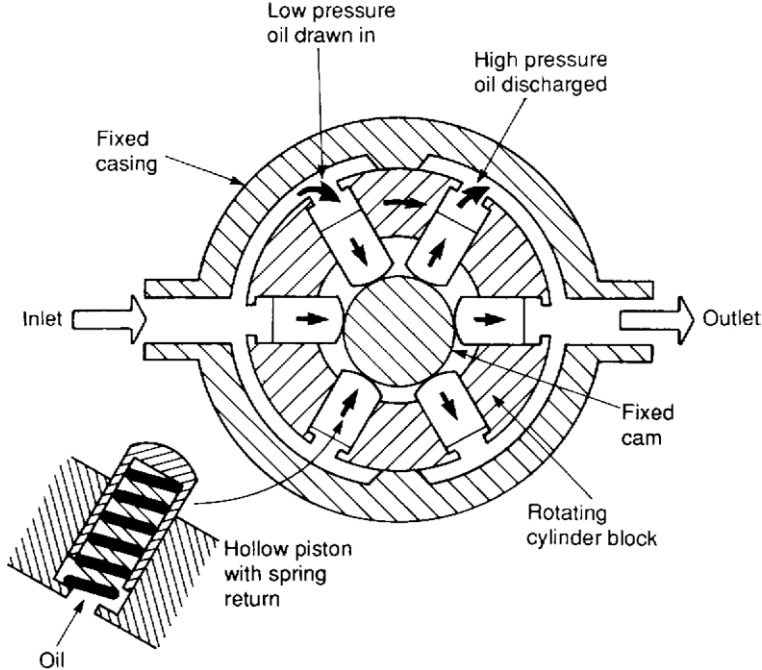
FIGURE 2.13 Piston pump with stationary cam and rotating block
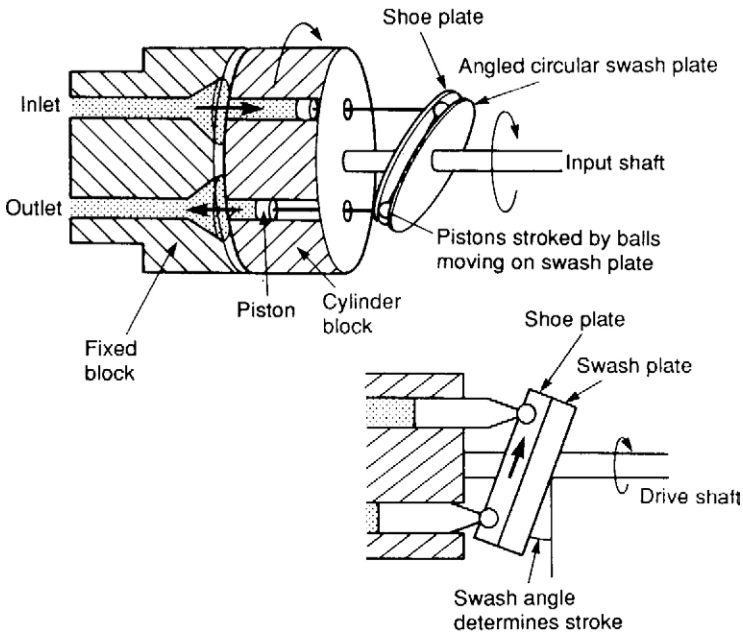
FIGURE 2.14 Axial pump with swash plate
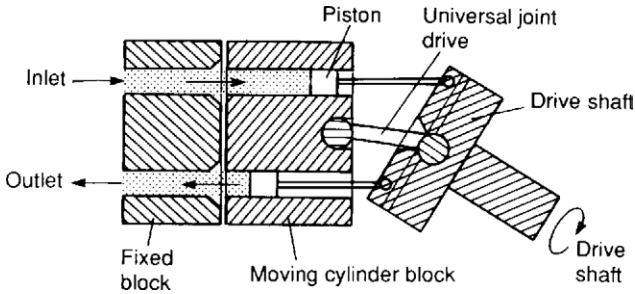
FIGURE 2.15 Bent axis pump

Pump displacement is controlled by altering the angle of the swash plate; the larger the angle, the greater the displacement. With the swash plate vertical displacement is zero, and flow can even be reversed. Swash plate angle (and hence pump displacement) can easily be controlled remotely with the addition of a separate hydraulic cylinder.

An alternative form of axial piston pump is the bent axis pump of Figure 2.15. Stroking of the pistons is achieved because of the angle between the drive shaft and the rotating cylinder block. Pump displacement can be adjusted by altering the drive shaft angle.

Piston pumps have very high volumetric efficiency (over 98%) and can be used at the highest hydraulic pressures. They are, though, bulky and noisy. Being more complex than vane and gear pumps, they are correspondingly more expensive and maintenance requires more skill. Table 2.1 gives a comparison of the various types of pump.

The figures in Table 2.1 are typical values and manufacturers' catalogs should be checked for a specific application. The displacement of gear, vane and radial piston pumps can be increased with multiple assemblies. Specialist pumps are available for pressures up to about 7000 bar at low flows. The

TABLE 2.1 Comparison of hydraulic pump types

Type	Maximum pressure (bar)	Maximum flow (l min⁻¹)	Variable displacement	Positive displacement
Centrifugal	20	3000	No	No
Gear	200	375	No	Yes
Vane	200	400	Yes	Yes
Axial piston (swash plate)	350	750	Yes	Yes
Axial piston (valved)	500	1500	Yes	Yes
In-line piston	1000	100	Yes	Yes

delivery from centrifugal and gear pumps can be made variable by changing the speed of the pump motor with a variable frequency (VF) drive.

Combination pumps

Many hydraulic applications are similar to Figure 2.16, where a workpiece is held in place by a hydraulic ram. There are essentially two distinct requirements for this operation. As the cylinder extends or retracts a large volume of fluid is required at a low pressure (sufficient just to overcome friction). As the workpiece is gripped, the requirement changes to a high pressure but minimal fluid volume.

This type of operation is usually performed with two separate pumps driven by a common electric motor as shown in Figure 2.17. Pump P_1 is a high-pressure low-volume pump, while pump P_2 is a high-volume low-pressure pump. Associated with these are two relief valves RV_1 and RV_2 and a one-way check (or non-return) valve which allows flow from left to right, but blocks flow in the reverse direction.

A normal (high-pressure) relief valve is used at position RV_1 but relief valve RV_2 is operated not by the pressure at point X, but remotely by the pressure at point Y. This could be achieved with the balanced piston valve of Figure 2.6 or,
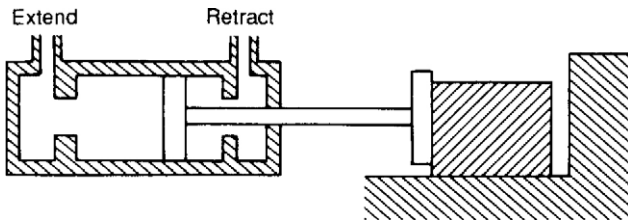
FIGURE 2.16 A clamping cylinder. A large flow, but low pressure, is needed during extension and retraction, but zero flow and high pressure are needed during clamping
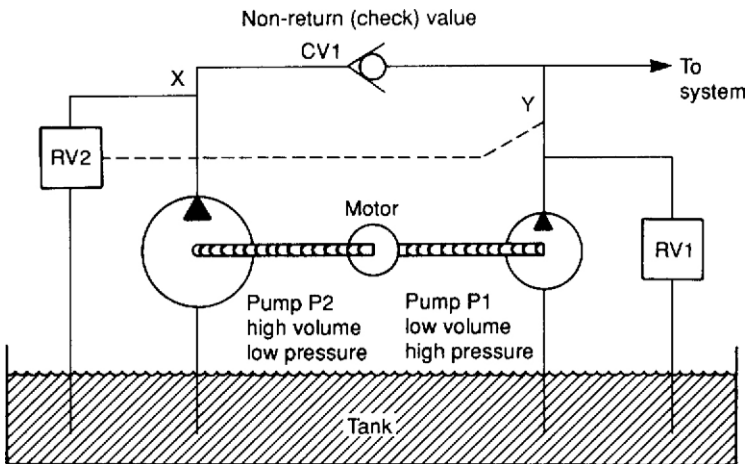
FIGURE 2.17 Combination pump

more commonly, the sequence valve of Figure 4.27. In low-pressure mode both relief valves are closed and both pumps P_1 and P_2 deliver fluid to the load, the majority coming from pump P_2 because of its higher displacement.

When the workpiece is gripped, the pressure at Y rises, and relief valve RV_2 opens, causing all the fluid from pump P_2 to return straight to the tank and the pressure at X to fall to a low value. Check valve CV_1 stops fluid from pump P_1 passing back to the tank via relief valve RV_2; consequently pressure at Y rises to the level set by relief valve RV_1.

This arrangement saves energy as the large volume of fluid from pump P_2 is returned to the tank at a very low pressure, and only a small volume of fluid from pump P_1 is returned at a high pressure. Pump assemblies similar to that shown in Figure 2.17 are called combination pumps and are manufactured as complete units with motor, pumps, relief and check valves prefitted.

LOADING VALVES

Expression 2.2 shows that allowing excess fluid from a pump to return to the tank by a pressure relief valve is wasteful of energy and can lead to a rapid rise in temperature of the fluid as the wasted energy is converted to heat. It is normally undesirable to start and stop the pump to match load requirements, as this causes shock loads to pump, motor and couplings.

In Figure 2.18, valve V_1 is a normal pressure relief valve regulating pressure and returning excess fluid to the tank as described in earlier sections. The additional valve V_2 is opened or closed by an external electrical or hydraulic signal. With valve V_2 open, all the pump output flow is returned to the tank at low pressure with minimal energy cost.

When fluid is required in the system the control signal closes valve V_2, pressure rises to the setting of valve V_1, and the system performs as normal. Valve V_2 is called a pump loading or a pump unloading valve according to the interpretation of the control signal sense.

PUMP PROBLEMS

Cavitation is the formation of vapor bubbles in a flowing liquid. When the bubble rapidly collapses, it produces an intense shock wave which rapidly erodes contact surfaces. If a pump suffers from cavitation it will be very noisy and suf-
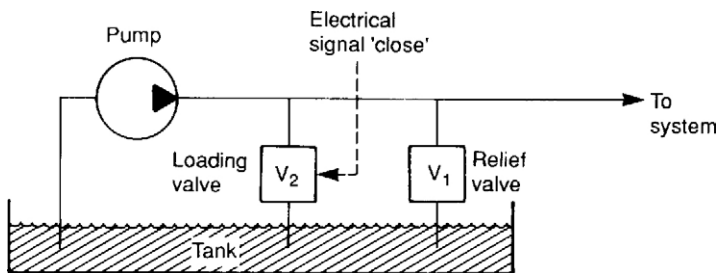
FIGURE 2.18 Loading valve

fer from early failure. Causes of cavitation are high-viscosity fluid, low temperature, pump overspeed (e.g. wrong motor fitted), inlet line too small or blocked inlet strainer.

Aeration occurs when air is drawn into the fluid leading to spongy imprecise operation. Possible causes are worn pump shaft seal, low oil level (allowing a vortex to form), return line above liquid level entrapping air or leak on pump inlet line allowing air to be drawn in. Leaks on the pressure side of the pump do not, generally, cause aeration.

No pump is 100% efficient and all suffer from some degree of *leakage*. Surprisingly a pump can suffer a major loss of displacement and still work satisfactorily, as discussed in Chapter 9. Leakage generally results in low speed rather than complete failure. Fault finding is simplified if a flow/pressure sensor (such as the UCC System 20) is fitted between the pump and the relief valve.

FILTERS

Dirt in a hydraulic system causes sticking valves, failure of seals and premature wear. Even particles of dirt as small as $20\,\mu m$ can cause damage (1 micron is one millionth of a meter; the naked eye is just able to resolve $40\,\mu m$). Filters are used to prevent dirt entering the vulnerable parts of the system, and are generally specified in microns or meshes per linear inch (sieve number).

Inlet lines are usually fitted with strainers inside the tank, but these are coarse wire mesh elements only suitable for removing relatively large metal particles and similar contaminants. Separate filters are needed to remove finer particles and can be installed in three places, as shown in Figure 2.19a–c.

(a) Inlet line filter

(b) Pressure line filter

(c) Return line filter
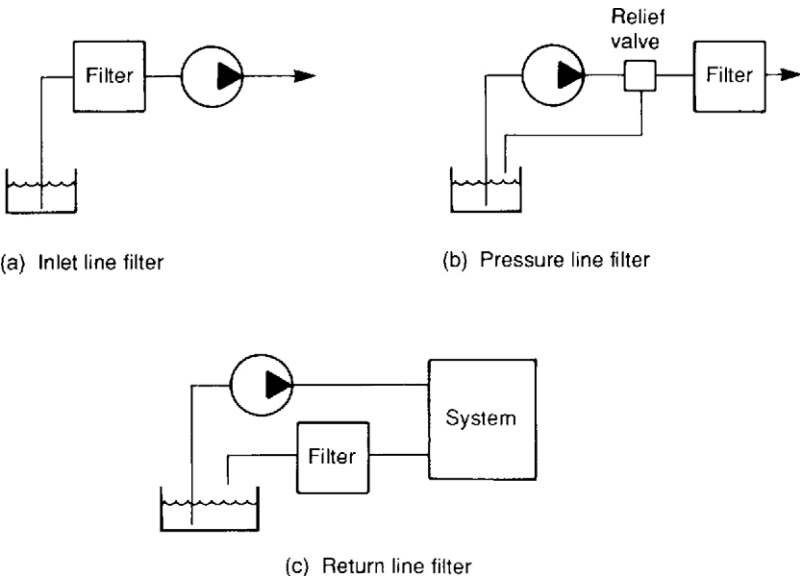
FIGURE 2.19 Filter positions

Inlet line filters protect the pump, but must be designed to give a low pressure drop or the pump will not be able to raise fluid from the tank. Low pressure drop implies a coarse filter or a large physical size.

Pressure line filters placed after the pump protect valves and actuators and can be finer and smaller. They must, however, be able to withstand full system operating pressure. Most systems use pressure line filtering.

Return line filters may have a relatively high pressure drop and can, consequently, be very fine. They serve to protect pumps by limiting the size of particles returned to the tank. These filters only have to withstand a low pressure. Filters can also be classified as full or proportional flow. In Figure 2.20a, all flow passes through the filter. This is obviously efficient in terms of filtration, but incurs a large pressure drop. This pressure drop increases as the filter becomes polluted, so a full flow filter usually incorporates a relief valve which cracks when the filter becomes unacceptably blocked. This is purely a safety feature, though, and the filter should, of course, have been changed before this state was reached as dirty unfiltered fluid would be passing round the system.

In Figure 2.20b, the main flow passes through a venturi, creating a localized low-pressure area. The pressure differential across the filter element draws a proportion of the fluid through the filter. This design is accordingly known as a proportional flow filter, as only a proportion of the main flow is filtered. It is characterized by a low pressure drop, and does not need the protection of a pressure relief valve.

Pressure drop across the filter element is an accurate indication of its cleanliness, and many filters incorporate a differential pressure meter calibrated with a green (clear), amber (warning), red (change overdue) indicator. Such types are called indicating filters.

Filtration material used in a filler may be mechanical or absorbent. Mechanical filters are relatively coarse, and utilize fine wire mesh or a disc/screen arrangement as shown in the edge type filter of Figure 2.21. Absorbent filters are based on porous materials such as paper, cotton or cellulose. Filtration size

(a) Full flow filter   (b) Proportional flow filter
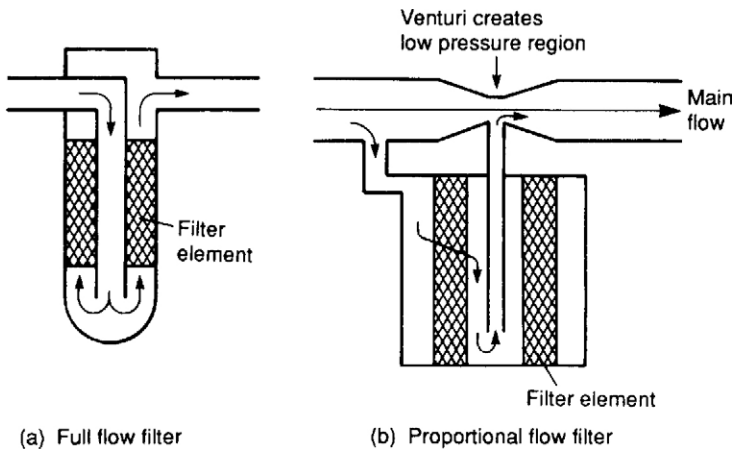
FIGURE 2.20 Filter types
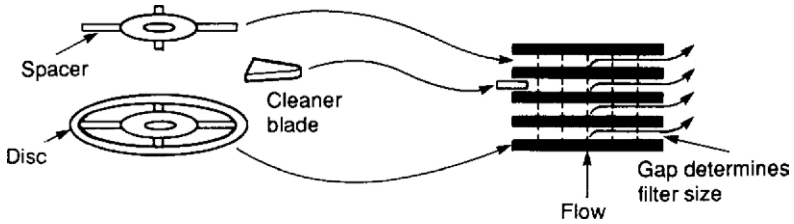
FIGURE 2.21 Edge type filter
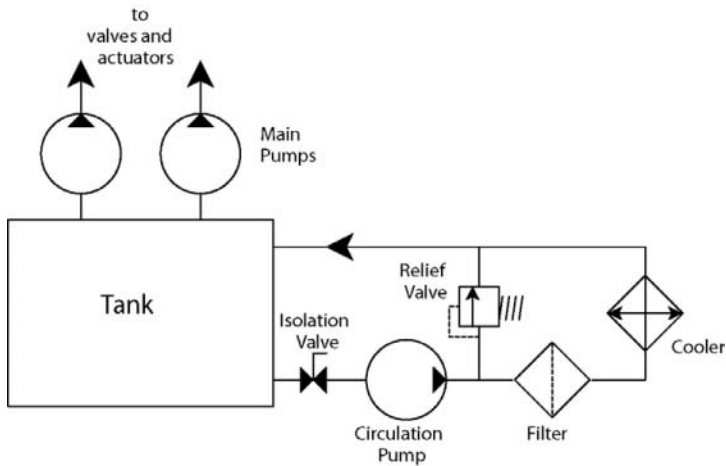
FIGURE 2.22 A circulation pump used to filter and clean the fluid when the draw from the main pumps is small

in an absorbent filter can be very small as filtration is done by pores in the material. Mechanical filters can usually be removed, cleaned and re-fitted, whereas absorbent filters are usually replaceable items.

In many systems where the main use is the application of pressure the actual draw from the tank is very small, reducing the effectiveness of pressure and return line filters. Here a separate circulating pump may be used as shown in Figure 2.22 to filter and cool the oil. The running of this pump is normally a precondition for starting the main pumps. The circulation pump should be sized to handle the complete tank volume every 10–15 minutes.

Note the pressure relief valve – this is included to provide a route back to the tank if the filter or cooler is totally blocked. In a real-life system additional hand isolation and non-return valves would be fitted to permit changing the filter or cooler with the system running. Limit switches and pressure switches would also be included to signal to the control system that the hand isolation valves are open and the filter is clean.

Chapter | three

Air Compressors, Air Treatment and Pressure Regulation

The vast majority of pneumatic systems use compressed atmospheric air as the operating medium (a small number of systems use nitrogen obtained commercially from liquid gas suppliers). Unlike hydraulic systems, a pneumatic system is 'open'; the fluid is obtained free, used and then vented back to atmosphere.

Pneumatic systems use a compressible gas, hydraulic systems an incompressible liquid, and this leads to some significant differences. The pressure of a liquid may be raised to a high level almost instantaneously, whereas pressure rise in a gas can be distinctly leisurely. In Figure 3.1a, a reservoir of volume two cubic meters is connected to a compressor which delivers three cubic meters of air (measured at atmospheric pressure) per minute. Using Boyle's law (expression 1.17) the pressure rise shown in Figure 3.1b can be found.

Pressure in a hydraulic system can be quickly and easily controlled by devices such as unloading and pressure-regulating valves. Fluid is thus stored at atmospheric pressure and compressed to the required pressure as needed. The slow response of an air compressor, however, precludes such an approach in a pneumatic system and necessitates storage of compressed air at the required pressure in a receiver vessel. The volume of this vessel is chosen so there are minimal deviations in pressure arising from flow changes in loads and the compressor is then employed to replace the air used, averaged over an extended period of time (e.g. a few minutes).

Deviations in air pressure are smaller, and compressor control is easier if a large receiver feeds many loads. A large number of loads statistically results in a more even flow of air from the receiver, also helping to maintain a steady pressure. On many sites, therefore, compressed air is produced as a central service which is distributed around the site in a similar manner to electricity, gas and water.

Behavior of a gas subjected to changes in pressure, volume and temperature is governed by the general gas equation given earlier as expression 1.19 and reproduced here:

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \qquad (3.1)$$

51
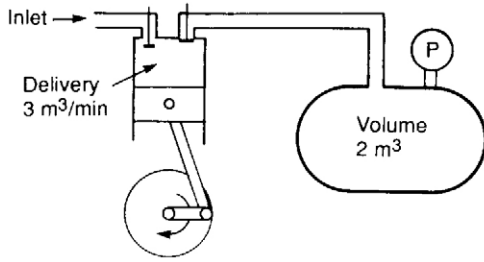
(a) Components

t (min)	Volume (at NTP)	P Abs	P gauge
0	2	1	0
1	5	2.5	1.5
2	8	4	3
3	11	5.5	4.5

(b) Response

FIGURE 3.1 Compressibility of a gas

where pressures are given in absolute terms and temperatures are measured in degrees Kelvin.

A compressor increases air pressure by reducing its volume, and expression 3.1 predicts a resultant rise in temperature. A pneumatic system must therefore incorporate some method of removing this excess heat. For small systems, simple fins on the compressor (similar in construction to an air-cooled internal combustion engine) will suffice. For larger systems, a separate cooler (usually employing water as the heat-removing medium) is needed.

Atmospheric air contains water vapor, the actual amount varying from day to day according to humidity. The maximum amount of water vapor held in a given volume of air is determined by temperature, and any excess condenses out as liquid droplets (commonly experienced as condensation on cold windows). A similar effect occurs as compressed air is cooled, and if left the resultant water droplets would cause valves to jam and corrosion to form in pipes. An after-cooler must therefore be followed by a water separator. Often aftercoolers and separators are called, collectively, primary air treatment units.

Dry cool air is stored in the receiver, with a pressure switch used to start and stop the compressor motor, maintaining the required pressure.

Ideally, air in a system has a light oil mist to reduce chances of corrosion and to lubricate moving parts in valves, cylinders and so on. This oil mist cannot be
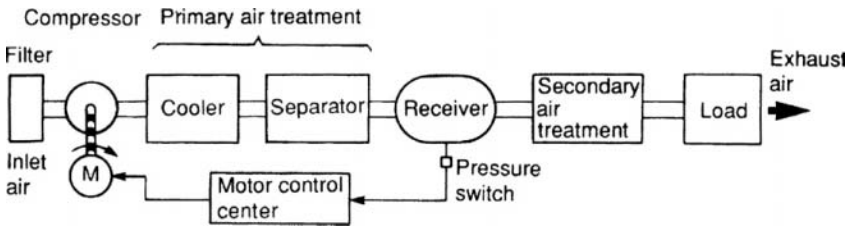
FIGURE 3.2 Component parts of a pneumatic system

added before the receiver as the mist would form oil droplets in the receiver's relatively still air, so the exit air from the receiver passes through a unit which provides the lubricating mist along with further filtration and water removal. This process is commonly called secondary air treatment.

Often, air in the receiver is held at a slightly higher pressure than needed to allow for pressure drops in the pipe lines. A local pressure regulation unit is then employed with the secondary air treatment close to the device using air. Composite devices called service units comprising water separation, lubricator and pressure regulation are available for direct line monitoring close to the valves and actuators of a pneumatic system.

Figure 3.2 thus represents the components used in the production of a reliable source of compressed air.

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 pressure* (STP) are commonly used, although differences between them are small for industrial users.

The technical normal condition is:

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

and the physical normal condition is:

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

The term normal temperature and pressure (NTP) is also used.

Required delivery volume of a compressor (in $M^3 \text{ min}^{-1}$ or $ft^3 \text{ min}^{-1}$, 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 Figure 3.3c.
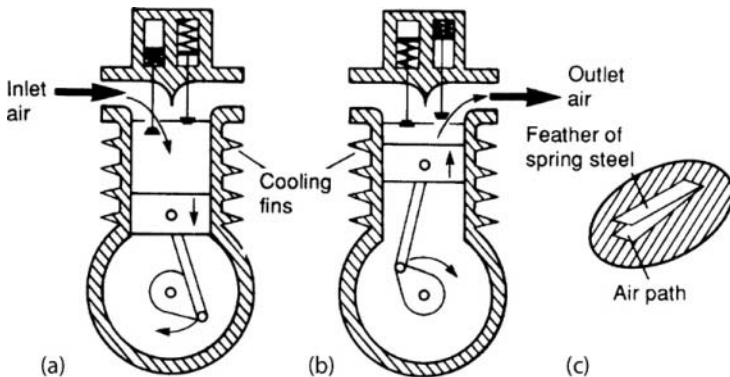
FIGURE 3.3 Single cylinder compressor
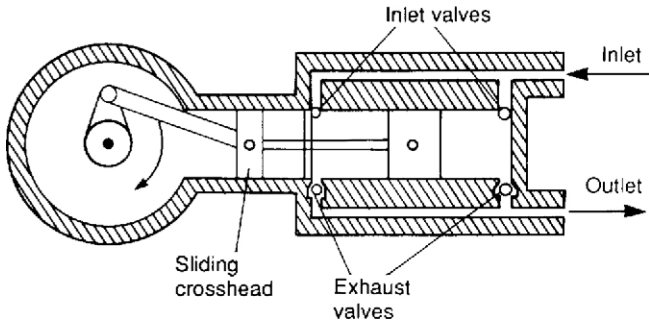
FIGURE 3.4 Double-acting compressor

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 multicylinder compressor is used. These are usually classified as vertical or 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 single-acting 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.

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 single-acting compressor, the compressed air temperature can rise to over 200 °C 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. Figure 3.5 shows an example. As cooling
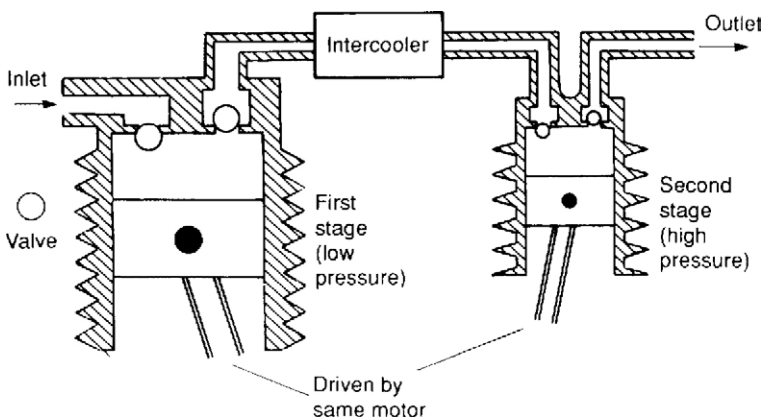
FIGURE 3.5 Two-stage compressor
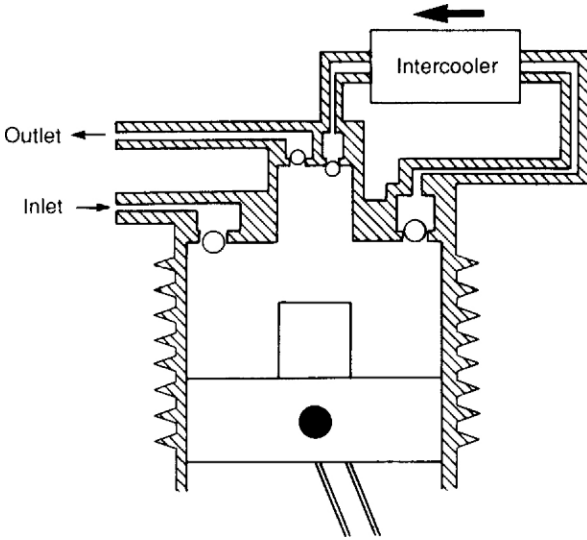
FIGURE 3.6 Combined two-stage compressor

(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 10–15 bar, but multistage compressors are available for pressures up to around 50 bar.

Multistage compressors can be manufactured with multicylinders as shown in Figure 3.5 or, more compactly, with a single cylinder and a double diameter piston as shown in Figure 3.6.

There is contact between pistons and air, in standard piston compressors, which may introduce small amounts of lubrication oil from the piston walls into the air.

This very small contamination may be undesirable in food and chemical industries. Figure 3.7 shows a common way of giving a totally clean supply by incorporating a flexible diaphragm between piston and air.
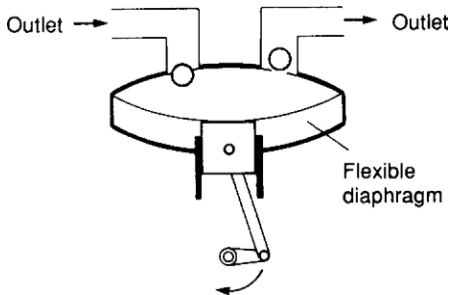
FIGURE 3.7 Diaphragm compressor, used where air must not be contaminated
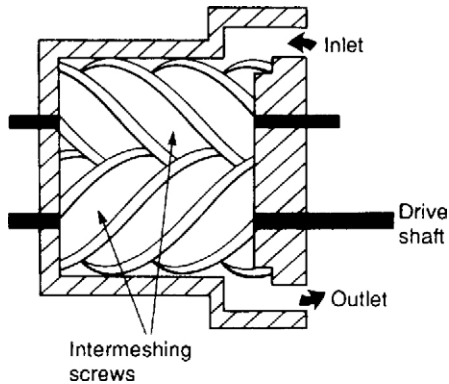
FIGURE 3.8 Dry screw rotary compressor

Screw compressors

Piston compressors are used where high pressures (>20 bar) and relatively low volumes (<10,000 m^3 h^{-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 m^3 h^{-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 Figure 3.8 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.

Rotary compressors

The vane compressor, shown in Figure 3.9, 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.
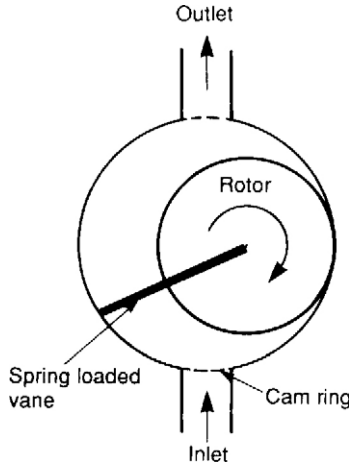
FIGURE 3.9 Vane compressor

Figure 3.10 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.

Dynamic compressors

A large volume of air (up to 5000 m³ min⁻¹) is often required for applications such as pneumatic conveying (where powder is carried in an air stream), ventila-
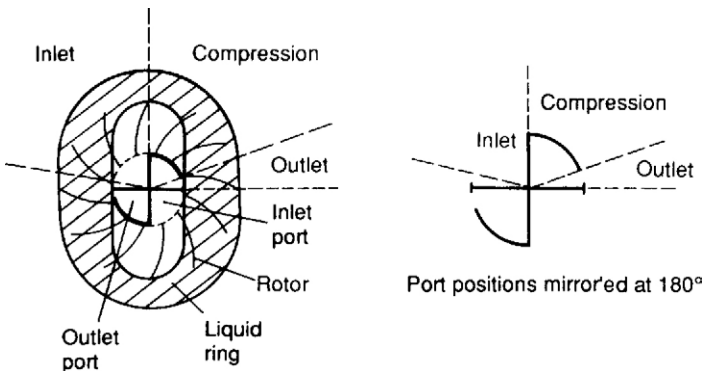
FIGURE 3.10 Liquid ring compressor
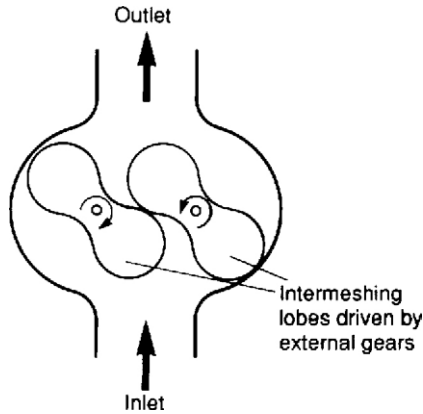
FIGURE 3.11 Lobe compressor

tion 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
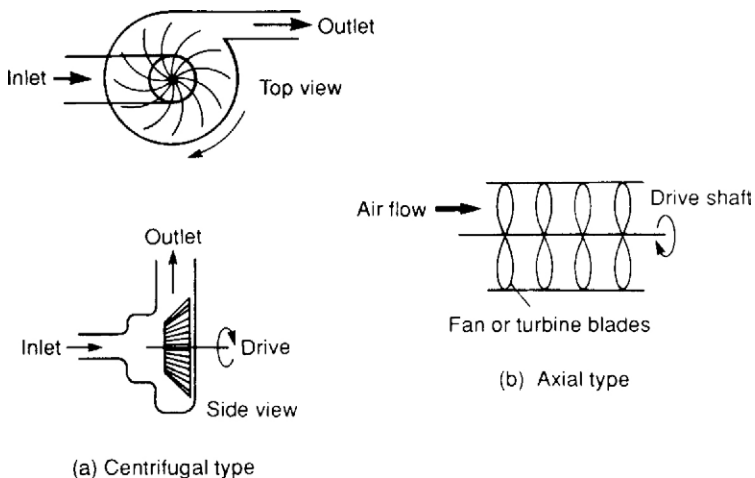
FIGURE 3.12 Non-positive displacement compressors (blowers)

air entry velocity to subsequent stages, thereby converting air kinetic energy to pressure energy.

Positive displacement compressors use oil to lubricate the close machined parts and to maintain the air seal. Dynamic compressors have no such need, and consequently deliver very clean air.

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.

Figure 3.13 shows essential features of a 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 pressurized 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 Figure 3.14. 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.
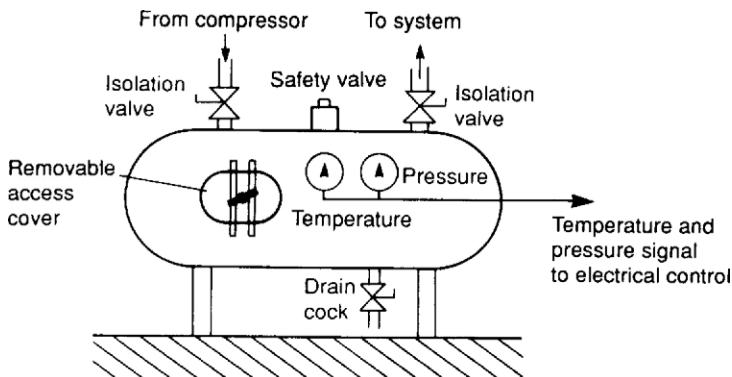
FIGURE 3.13 Compressed air receiver
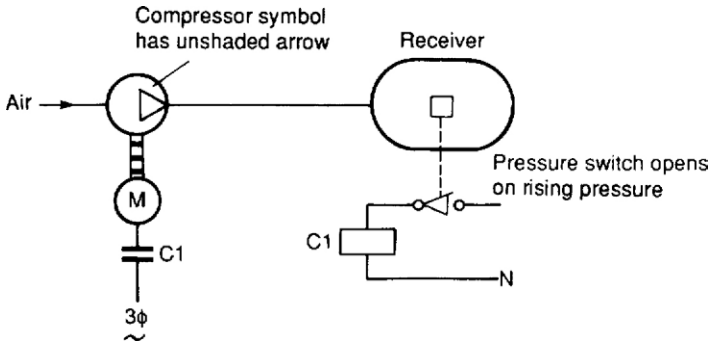
FIGURE 3.14 Receiver pressure control via motor start/stop

In Figure 3.15 another method of pressure control is shown, where the compressor runs continuously and an exhaust valve is fitted to the compressor outlet. This valve opens when the required pressure is reached. A non-return valve prevents air returning from the receiver. This technique is known as exhaust regulation.

Compressors can also be controlled on the inlet side. In the example of Figure 3.16, 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 3.15 and 3.16 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 meters volume and a working pressure of 8 bar delivers 25 cubic meters of air (at STP) to a load, pressure in the receiver falls to approximately 5.5 bar.
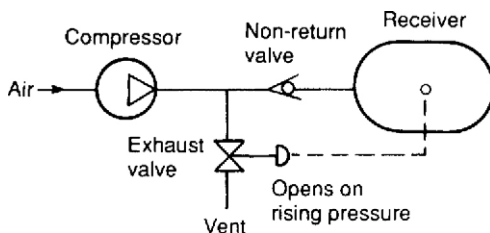
FIGURE 3.15 Receiver pressure control using compressor outlet valve
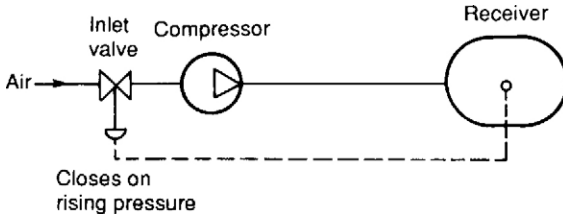
FIGURE 3.16 Receiver pressure control using compressor inlet valve

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.

AIR TREATMENT

Atmospheric air contains moisture in the form of water vapor. We perceive the amount of moisture in a given volume of air as the humidity and refer to days with a high amount of water vapor as 'humid' or 'sticky', and days with low amounts of water vapor as 'good drying days'. The amount of water vapor which can be held in a given volume depends on temperature but does *not* depend on pressure of air in that volume. One cubic meter at 20 °C, for example, can hold 17 grams of water vapor. The amount of water vapor which can be held in a given volume of air rises with temperature, as shown in Figure 3.17.

If a given volume of air contains the maximum quantity of water vapor possible at the air temperature, the air is said to be *saturated* (and we would perceive it as sticky because sweat could not evaporate from the surface of the skin). From Figure 3.17, air containing 50 grams of water vapor per cubic meter at 40 °C is saturated.

Moisture content of unsaturated air is referred to by relative humidity, which is defined as:

$$\text{Relative humidity} = \frac{\text{water content per cubic meter}}{\text{maximum water content per cubic meter}} \times 100\% \quad (3.2)$$

Air containing 5 grams of water vapor per cubic meter of air at 20 °C has, from Figure 3.17, a relative humidity of 30%.

Relative humidity is dependent on both temperature and pressure of the air. Suppose air at 30 °C contains 20 grams of water vapor. From Figure 3.17 this corresponds to 67% humidity. If the air is allowed to cool to 20 °C it can only hold 17 grams of water vapor and is now saturated (100% relative humidity).
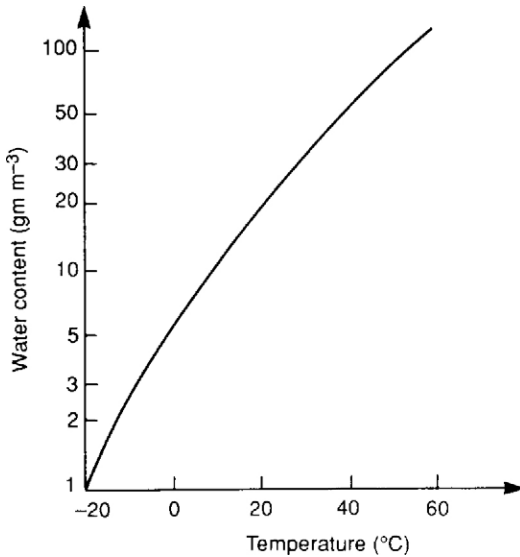
FIGURE 3.17 Moisture content curve

The excess 3 grams condenses out as liquid water. If the air is cooled further to 10°C, a further 8 grams condenses out.

The temperature at which air becomes saturated is referred to as the 'dew point'. Air with 17.3 grams of water vapor per cubic meter has, for example, a dew point of 20°C.

To see the effect of pressure on relative humidity, we must remember the amount of water vapor which can be held in a given volume is fixed (assuming a constant temperature). Suppose a cubic meter of air at atmospheric pressure (0 bar gauge or 1 bar absolute) at 20°C contains 6 grams of water vapor (corresponding to 34% relative humidity). If we wish to increase air pressure while maintaining its temperature at 20°C, we must compress it. When the pressure is 1 bar gauge (or 2 bar absolute) its volume is 0.5 cubic meters, which can hold 8.6 grams of water vapor, giving us 68% relative humidity. At 2 bar gauge (3 bar absolute) the volume is 0.33 cubic meters, which can hold 5.77 grams of water vapor. With 6 grams of water vapor in our air, we have reached saturation and condensation has started to occur.

It follows that relative humidity rises quickly with increasing pressure, and even low atmospheric relative humidity leads to saturated air and condensation at the pressures used in pneumatic systems (8–10 bar). Water droplets resulting from this condensation can cause many problems. Rust will form on unprotected steel surfaces, and the water may mix with oil (necessary for lubrication) to form a sticky white emulsion, which causes valves to jam and blocks the small piping used in pneumatic instrumentation systems. In extreme cases water traps can form in pipe loops.

When a compressed gas expands suddenly there is a fall of temperature (predicted by expression 1.19). If the compressed air has a high water content, a rapid expansion at exhaust ports can be accompanied by the formation of ice as the water condenses out and freezes.
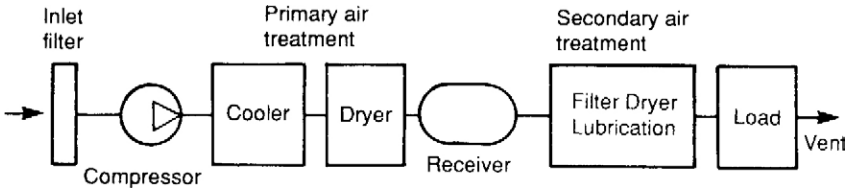
FIGURE 3.18 Three stages of air treatment

Stages of air treatment

Air in a pneumatic system must be clean and dry to reduce wear and extend maintenance periods. Atmospheric air contains many harmful impurities (smoke, dust, water vapor) and needs treatment before it can be used.

In general, this treatment falls into three distinct stages, shown in Figure 3.18. First, inlet filtering removes particles which can damage the air compressor. Next, there is the need to dry the air to reduce humidity and lower the dew point. This is normally performed between the compressor and the receiver and is termed primary air treatment.

The final treatment is performed local to the duties to be performed, and consists of further steps to remove moisture and dirt and the introduction of a fine oil mist to aid lubrication. Not surprisingly this is generally termed secondary air treatment.

Filters

Inlet filters are used to remove dirt and smoke particles before they can cause damage to the air compressor, and are classified as dry filters with replaceable cartridges (similar to those found in motor car air filters) or wet filters where the incoming air is bubbled through an oil bath then passed through a wire mesh filter. Dirt particles become attached to oil droplets during the bubbling process and are consequently removed by the wire mesh.

Both types of filter require regular servicing: replacement of the cartridge element for the dry type, cleaning for the wet type. If a filter is to be cleaned, it is essential the correct detergent is used. Use of petrol or similar petrochemicals can turn an air compressor into an effective diesel engine – with severe consequences.

Filters are classified according to size of particles they will stop. Particle size is measured in SI units of micrometers (the older metric term *microns* is still common), one micrometer ($1\,\mu m$) being 10^{-6} meter or 0.001 millimeter. Dust particles are generally larger than $10\,\mu m$, whereas smoke and oil particles are around $1\,\mu m$. A filter can have a nominal rating (where it will block 98% of particles of the specified size) or an absolute rating (where it blocks 100% of particles of the specified size).

Microfilters with removable cartridges passing air from the center to the outside of the cartridge case will remove 99.9% of particles down to $0.01\,\mu m$, the limit of normal filtration. Coarse filters, constructed out of wire mesh and called

strainers, are often used as inlet filters. These are usually specified in terms of the mesh size, which approximates to particle size in micrometers as follows:

Mesh size	μm
325	30
550	10
750	6

Air dryers

An earlier section described how air humidity and dew point are raised by compression. Before air can be used, this excess moisture has to be removed to bring air humidity and dew point to reasonable levels.

In bulk air systems all that may be required is a simple after-cooler similar to the intercoolers described earlier, followed by a separator unit where the condensed water collects and can be drained off.

Figure 3.19a shows a typical water trap and separator. Air flow through the unit undergoes a sudden reversal of direction and a deflector cone swirls the air (Figure 3.19b). Both of these cause heavier water particles to be flung out to the walls of the separator and to collect in the trap bottom, from where they can be drained. Water traps are usually represented on circuit diagrams by the symbol in Figure 3.19c.

(a) Construction

(b) Swirl introduced by deflector cone

(c) Symbol
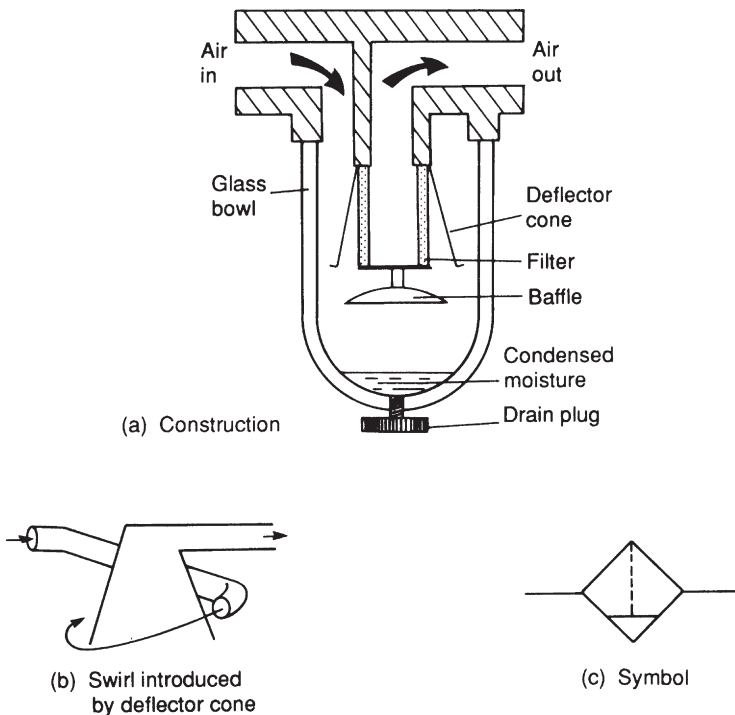
FIGURE 3.19 Air filter and water trap
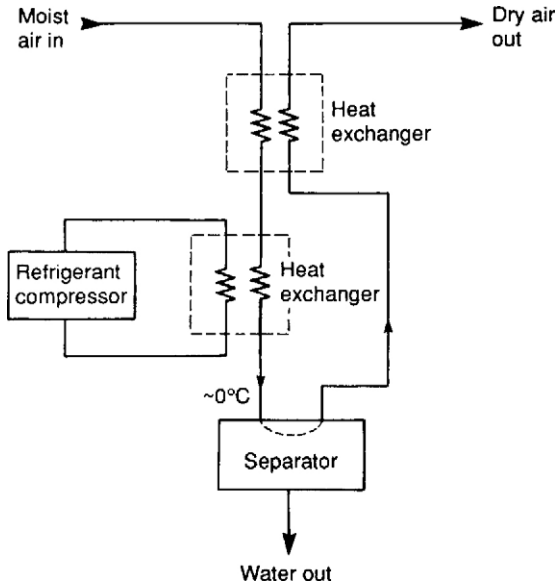
FIGURE 3.20 Refrigerated dryer

Dew point can be lowered further with a refrigerated dryer, the layout of which is illustrated in Figure 3.20. This chills the air to just above 0°C, condensing almost all the water out and collecting the condensate in the separator. Efficiency of the unit is improved with a second heat exchanger in which cold dry air leaving the dryer pre-chills incoming air. Air leaving the dryer has a dew point similar to the temperature in the main heat exchanger.

Refrigerated dryers give air with a dew point sufficiently low for most processes. Where absolutely dry air is needed, chemical dryers must be employed. Moisture can be removed chemically from air by two processes.

In a deliquescent dryer, the layout of which is shown in Figure 3.21, a chemical agent called a desiccant is used. This absorbs water vapor and slowly dissolves to form a liquid which collects at the bottom of the unit, where it can be
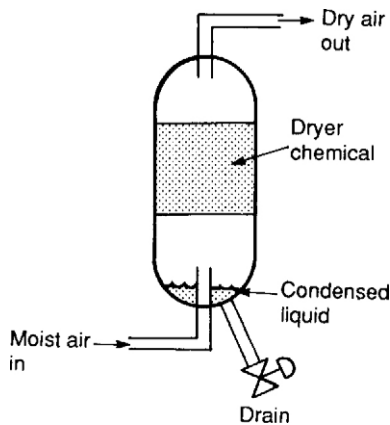
FIGURE 3.21 Deliquescent dryer
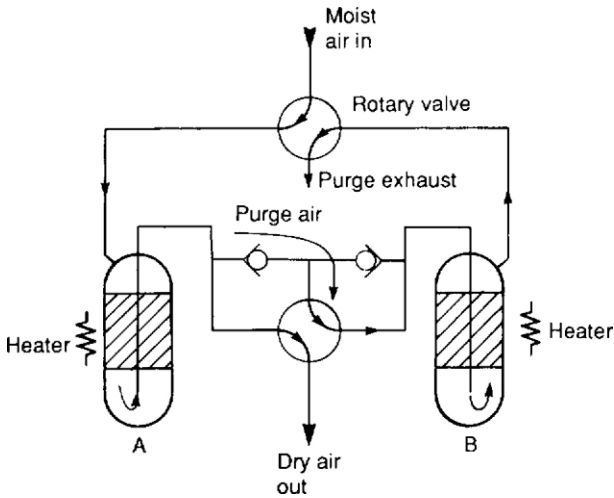
FIGURE 3.22 Adsorption dryer

drained. The desiccant material is used up during this process and needs to be replaced at regular intervals. Often, deliquescent dryers are referred to as absorbtion dryers, a term that should not be confused with the next type of dryer.

An adsorption dryer collects moisture on the sharp edges of a granular material such as silicon dioxide, or with materials which can exist in hydrated *and* dehydrated states (the best known is copper sulfate but more efficient compounds are generally used). Figure 3.22 shows construction of a typical adsorption dryer. Moisture in the adsorption material can be released by heating, so two columns are used. At any time, one column is drying the air while the other is being regenerated by heating and the passage of a low purge air stream. As shown, column A dries the air and column B is being regenerated. The rotary valves are operated automatically at regular intervals by a time clock. For obvious reasons adsorption dryers are often referred to as regenerative dryers.

Lubricators

A carefully controlled amount of oil is often added to air immediately prior to use to lubricate moving parts (process control pneumatics are the exception as they usually require dry unlubricated air). This oil is introduced as a fine mist, but can only be added to thoroughly clean and dry air or a troublesome sticky emulsion forms. It is also difficult to keep the oil-mist-laden air in a predictable state in an air receiver, so oil addition is generally performed as part of the secondary air treatment.

The construction of a typical lubricator is shown with its symbol in Figure 3.23. The operation is similar to the principle of the petrol air mixing in a motor car carburettor. As air enters the lubricator its velocity is increased by a venturi ring, causing a local reduction in pressure in the upper chamber. The pressure differential between lower and upper chambers causes oil to be drawn up a riser tube, emerging as a spray to mix with the air. The needle valve adjusts the pressure differential across the oil jet and hence the oil flow rate.
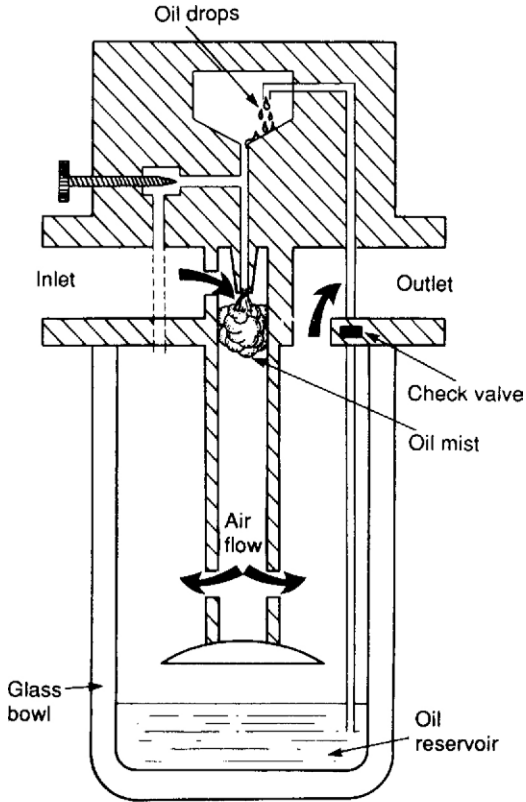
(a) Construction
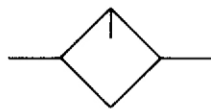
(b) Symbol for lubricator

FIGURE 3.23 Lubricator

The air–oil mixture is forced to swirl as it leaves the central cylinder, causing excessively large oil particles to be flung out of the air stream.

Air classification

Different applications require different air qualities; very small particles, for example, can cause unsightly blemishes with paint spraying for motor vehicle bodies, and the fine tolerances in process control pneumatics can easily be blocked by high oil content or particles. At the other extreme it is wasteful to have expensive high-quality air for workshop machine tools. Standard ISO 8573-1 defines different air qualities, as shown in Table 3.1.

TABLE 3.1 Air classification for different applications

Class	Particle size (μm)	Particle density (mg m^{-3})	Dew point (°C)	Oil content (mg m^{-3})	Typical application
1	0.1	0.1	−70	0.01	Paint spraying
2	1	1	−40	0.1	Process control and instrumentation
3	5	5	−20	1	Miniature solenoid valves
4	15	8	3	5	Standard factory pneumatics
5	40	10	7	25	Machine tools

PRESSURE REGULATION

Flow velocities in pneumatic systems can be quite high, which can lead to significant flow-dependent pressure drops between the air receiver and the load.

Generally, therefore, air pressure in the receiver is set higher than the required load pressure and pressure regulation is performed local to loads to keep pressure constant regardless of flow. Control of air pressure in the receiver was described in an earlier section. This section describes various ways in which pressure is locally controlled.

There are essentially three methods of local pressure control, illustrated in Figure 3.24. Load A vents continuously to atmosphere. Air pressure is controlled by a pressure regulator which simply restricts air flow to the load. This type of regulator requires some minimum flow to operate. If used with a dead-end load which draws no air, the air pressure will rise to the main manifold pressure. Such regulators, in which air must pass through the load, are called non-relieving regulators.

Load B is a dead-end load, and uses a pressure regulator which vents air to atmosphere to reduce pressure. This type of regulator is called a three-port (for
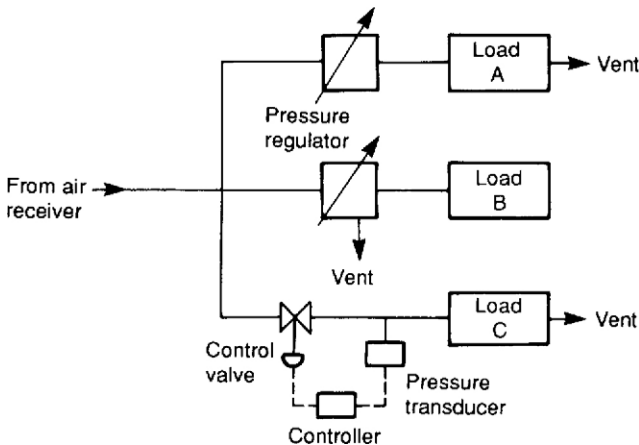
FIGURE 3.24 Three types of pressure regulator
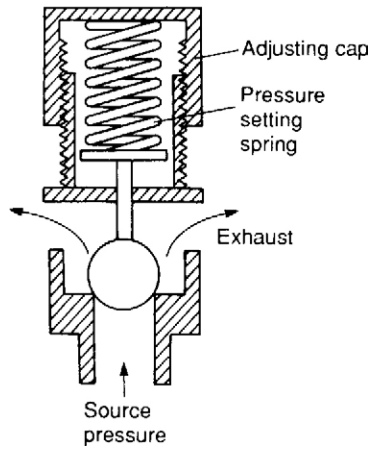
FIGURE 3.25 Relief valve

the three connections) or relieving regulator. Finally, load C is a large-capacity load whose air volume requirements are beyond the capacity of a simple in-line regulator. Here a pressure control loop has been constructed comprising pressure transducer, electronic controller and separate vent valve. This technique can also be used if the pressure-regulating valve cannot be mounted locally to the point at which the pressure is to be controlled.

Relief valves

The simplest pressure-regulating device is the relief valve shown in Figure 3.25. This is not, in fact, normally used to control pressure but is employed as a back-up device should the main pressure control device fail. They are commonly fitted, for example, to air receivers.

A ball valve is held closed by spring tension, adjustable to set the relief pressure. When the force due to air pressure exceeds the spring tension, the valve cracks open releasing air and reducing the pressure. Once cracked, flow rate is a function of excess pressure, an increase in pressure leading to an increase in flow. A relief valve is specified by operating pressure range, span of pressure between cracking and full flow, and full flow rate. Care is needed in specifying a relief valve because in a fault condition the valve may need to pass the entire compressor output.

A relief valve has a flow/pressure relationship and self-seals itself once the pressure falls below the cracking pressure. A pure safety valve operates differently. Once a safety valve cracks, it opens fully to discharge all the pressure in the line or receiver, and it does not automatically reclose, needing manual resetting before the system can be used again.

Non-relieving pressure regulators

Figure 3.26 shows construction of a typical non-relieving pressure regulator. Outlet pressure is sensed by a diaphragm which is pre-loaded by a pressure-setting spring. If outlet pressure is too low, the spring forces the diaphragm and poppet down, opening the valve to admit more air and raise outlet pressure.
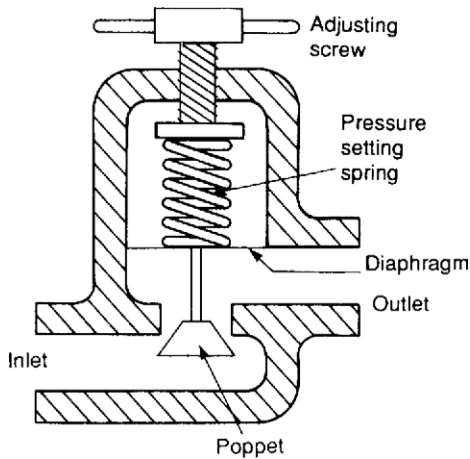
FIGURE 3.26 Non-relieving pressure regulator

If the outlet pressure is too high, air pressure forces the diaphragm up, re-
ducing air flow and causing a reduction in air pressure as air vents away through
the load. In a steady state the valve will balance, with the force on the diaphragm
from the outlet pressure just balancing the preset force on the spring.

Relieving pressure regulators

A relieving regulator is shown in Figure 3.27. Outlet pressure is sensed by a
diaphragm preloaded with an adjustable pressure-setting spring. The diaphragm
rises if the outlet pressure is too high, and falls if the pressure is too low.

If outlet pressure falls, the inlet poppet valve is pushed open, admitting more
air to raise pressure. If the outlet pressure rises, the diaphragm moves down,
closing the inlet valve and opening the central vent valve to allow excess air to
escape from the load, thereby reducing pressure.

In a steady state the valve will balance, *dithering* between admitting and
venting small amounts of air to keep load pressure at the set value.

Both the regulators in Figures 3.26 and 3.27 are simple pressure regulators
and have responses similar to that shown in Figure 3.28, with outlet pressure
decreasing slightly with flow. This drop in pressure can be overcome by using a
pilot-operated regulator, shown in Figure 3.29.

Outlet pressure is sensed by the pilot diaphragm, which compares outlet pres-
sure with the value set by the pressure-setting spring. If outlet pressure is low the
diaphragm descends, while if outlet pressure is high the diaphragm rises.

Inlet air is bled through a restriction and applied to the top of the main dia-
phragm. This space can, however, be vented to the exit side of the valve by the
small ball valve.

If outlet pressure is low, the pilot diaphragm closes the ball valve, causing
the main diaphragm to be pushed down and more air to be admitted to the load.

If outlet pressure is high, the pilot diaphragm opens the ball valve and the
space above the main diaphragm de-pressurizes. This causes the main diaphragm
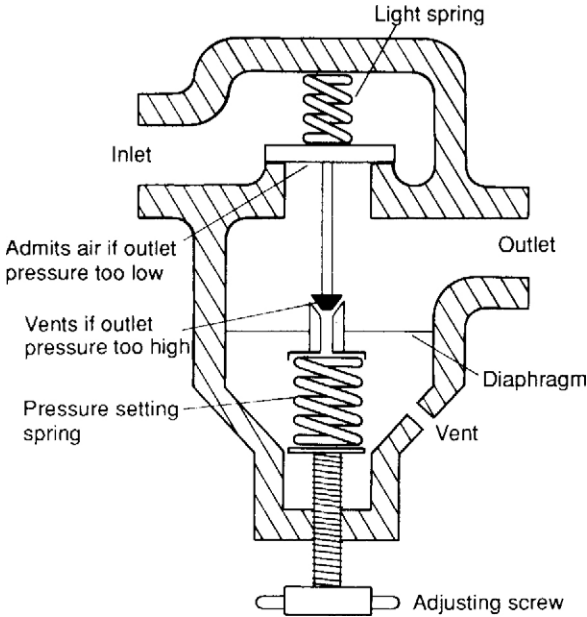
FIGURE 3.27 Relieving pressure regulator

to rise, opening the central vent allowing air to escape from the load and pressure to be reduced.

Action of pilot diaphragm and inlet air bleed approximates to integral action, giving a form of P+I (proportional plus integral) control. In the steady state, the outlet pressure equals the set pressure and there is no pressure drop with increasing flow.

SERVICE UNITS

In pneumatic systems a moisture separator, a pressure regulator, a pressure indicator, a lubricator and a filter are all frequently required, local to a load or system. This need is so common that combined devices called *service units* are available. Individual components comprising a service unit are shown in Figure 3.30a, while the composite symbol of a service unit is shown in Figure 3.30b.
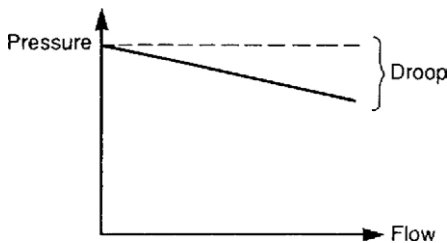
FIGURE 3.28 Response of simple pressure regulators
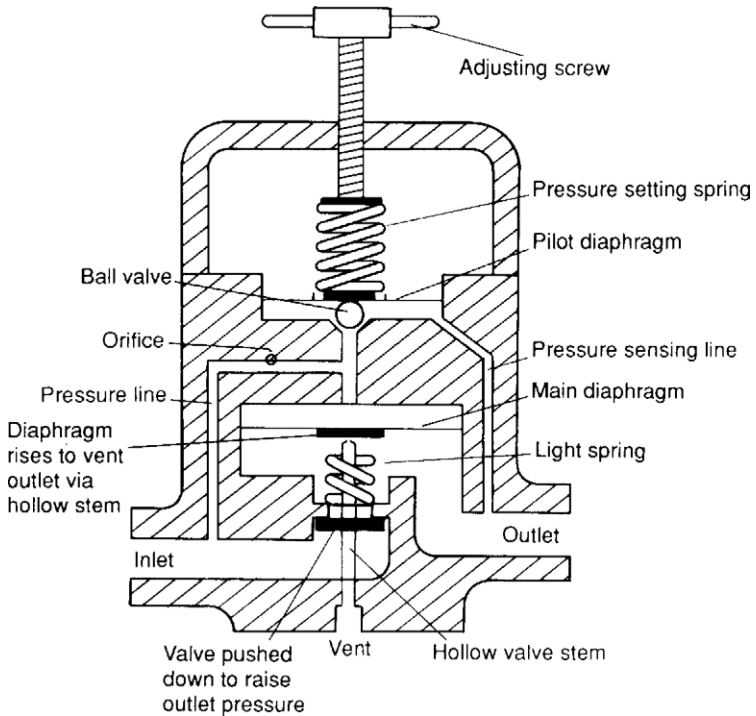
FIGURE 3.29 Pilot-operated regulator
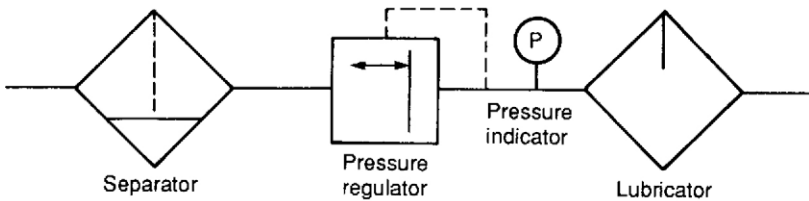
(a) Symbols for individual components
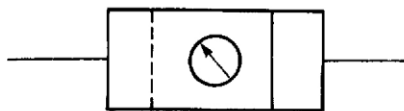
(b) Composite symbol

FIGURE 3.30 The service unit

Chapter | four

Control Valves

Pneumatic and hydraulic systems require control valves to direct and regulate the flow of fluid from compressor or pump to the various load devices. Although there are significant practical differences between pneumatic and hydraulic devices (mainly arising from differences in operating pressures and types of seals needed for gas or liquid) the operating principles and descriptions are very similar.

Although valves are used for many purposes, there are essentially only two types of valve. An infinite position valve can take up any position between open and closed and, consequently, can be used to modulate flow or pressure. Relief valves described in earlier chapters are simple infinite position valves.

Most control valves, however, are only used to allow or block flow of fluid. Such valves are called finite position valves. An analogy between the two types of valve is the comparison between an electric light dimmer and a simple on/off switch. Connections to a valve are termed 'ports'. A simple on/off valve therefore has two ports. Most control valves, however, have four ports, shown in hydraulic and pneumatic forms in Figure 4.1.

In both the load is connected to ports labeled A, B and the pressure supply (from pump or compressor) to port P. In the hydraulic valve, fluid is returned to the tank from port T. In the pneumatic valve return air is vented from port R.

Figure 4.2 shows internal operation of valves. To extend the ram, ports P and B are connected to deliver fluid and ports A and T connected to return fluid. To retract the ram, ports P and A are connected to deliver fluid and ports B and T to return fluid.

Another consideration is the number of control positions. Figure 4.3 shows two possible control schemes. In Figure 4.3a, the ram is controlled by a lever with two positions: extend or retract. This valve has two control positions (and the ram simply drives to one end or the other of its stroke). The valve in Figure 4.3b has three positions: extend, off, retract. Not surprisingly the valve in Figure 4.3a is called a two-position valve, while that in Figure 4.3b is a three-position valve.

Finite position valves are commonly described as a port/position valve where *port* is the number of ports and *position* is the number of positions. Figure 4.3a therefore illustrates a 4/2 valve, and Figure 4.3b shows a 4/3 valve. A simple block/allow valve is a 2/2 valve.

The numbers of ports and positions do not, however, completely describe the valve. We must also describe its action. Figure 4.4 shows one possible action for the 4/3 valve of Figure 4.3b. Extend and retract connections are similar, but in the off position ports P and T are connected – unloading the pump back to the

(a) Hydraulic system
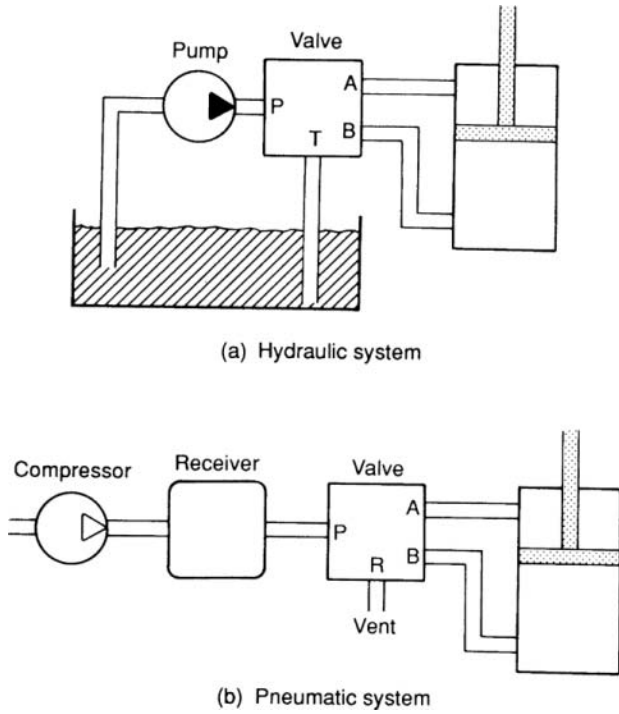
(b) Pneumatic system

FIGURE 4.1 Valves in a pneumatic and a hydraulic system

Extend   Retract
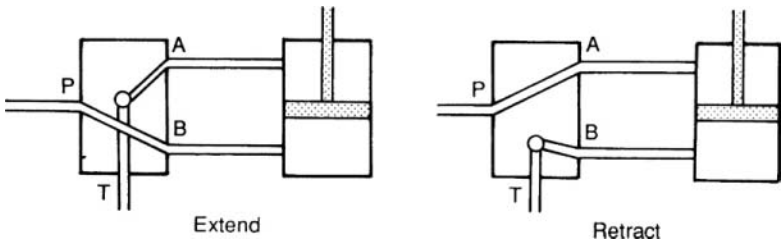
FIGURE 4.2 Internal valve operation

tank without need of a separate loading valve, while leaving the ram locked in position. (This approach could, of course, only be used where the pump supplies one load.) Other possible arrangements may block all four ports in the off position (to maintain pressure), or connect ports A, B and T (to leave the ram free in the off position). A complete valve description thus needs number of ports, number of positions *and* the control action.

GRAPHIC SYMBOLS

Simple valve symbols have been used so far to describe control actions. From the discussions in the previous section it can be seen that control actions can easily become too complex for representation by sketches showing how a valve is constructed.

(a) Two position valve   (b) Three position valve
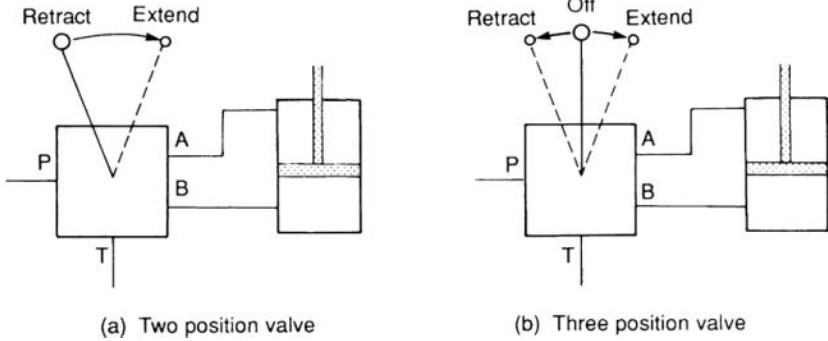
FIGURE 4.3 Valve control positions
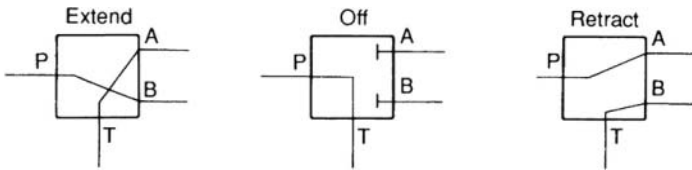
FIGURE 4.4 Possible valve action for a 4/3 valve

A set of graphic symbols has therefore evolved (similar, in principle, to symbols used on electrical circuit diagrams). These show component function without showing the physical construction of each device. A 3/2 spool valve and a 3/2 rotary valve with the same function have the same symbol, despite their totally different constructions.

Symbols are described in various national documents: DIN24300, BS2917, ISO1219 and the new ISO5599, CETOP RP3 plus the original American JIC and ANSI symbols. Differences between these are minor.

A valve is represented by a square for each of its switching positions. Figure 4.5a thus shows the symbol of a two-position valve, and Figure 4.5b a three-position valve. Valve positions can be represented by letters a, b, c and so on, with 0 being used for a central neutral position.

Ports of a valve are shown on the outside of boxes in normal unoperated or initial position. Four ports have been added to the two-position valve symbol shown in Figure 4.5c. Designations given to ports are normally:

Port	Designation	ISO 5599
Working lines	A, B, C and so on	2,4
Pressure (power) supply	P	1
Exhaust/return	R, S, T and so on (T, for tank, is normally used for hydraulic systems, R and S for pneumatic systems)	3,5 Pilot exhaust 82, 84
Control (pilot) lines	Z, Y, X and so on	12, 14

(a) Two position
valve

(b) Three position
valve

(c) 4/2 valve
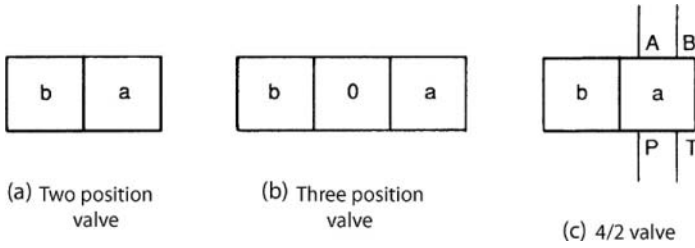
FIGURE 4.5 Basis of graphic symbols

(a) 4/2 valve

(b) 4/3 valve center off
(load isolated)

(c) 4/3 valve, load free
in center

(d) Pneumatic valve with vent
(pneumatic valve often
presented with unshadded
arrow heads)
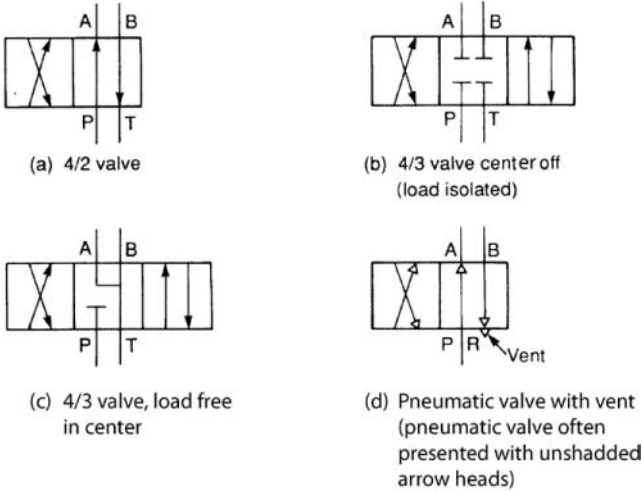
FIGURE 4.6 Valve symbols

Arrow-headed lines represent direction of flow. In Figure 4.6a, for example, fluid is delivered from port P to port A and returned from port B to port T when the valve is in its normal state a. In state b, flow is reversed. This valve symbol corresponds to the valve represented in Figures 4.2 and 4.3a.

Shut-off positions are represented by **T**, as shown by the central position of the valve in Figure 4.6b, and internal flow paths can be represented as shown in Figure 4.6c. This latter valve, incidentally, vents the load in the off position.

In pneumatic systems, lines commonly vent to atmosphere directly at the valve, as shown by port R in Figure 4.6d.

Figure 4.7a shows symbols for the various ways in which valves can be operated. Figure 4.7b thus represents a 4/2 valve operated by a pushbutton. With the pushbutton depressed the ram extends. With the pushbutton released, the spring pushes the valve to position a and the ram retracts.

Actuation symbols can be combined. Figure 4.7c represents a solenoid-operated 4/3 valve, with spring return to center.

Infinite-position valve symbols are shown in Figure 4.8. A basic valve is represented by a single square as shown in Figure 4.8a, with the valve being shown in a normal, or non-operated, position. Control is shown by normal actuation symbols: in Figure 4.8b, for example, the spring pushes the valve right, decreasing flow, and

Push button

Roller limit SW

Spring

Pressure line (pilot)

Lever

Detent (holds position)
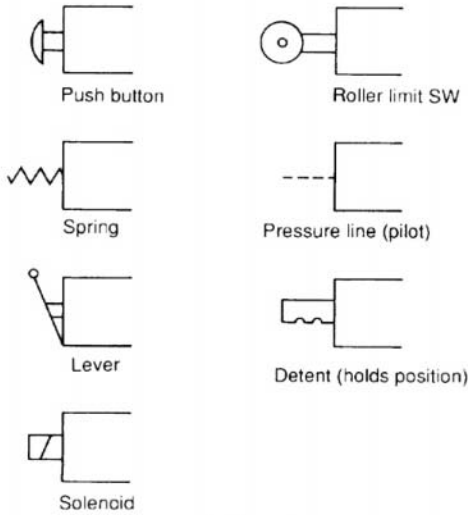
Solenoid

(a) Actuation symbols
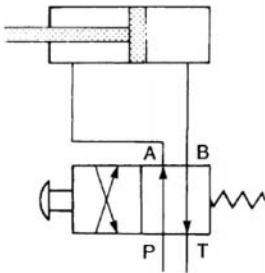
(b) Pushbutton extend, spring retract
when pushbutton released

A B

P T
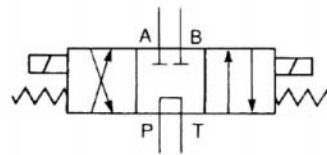
(c) 4/3 valve, solenoid operated,
spring return to center. Pressure
line unloads to tank and load
locked in center position

FIGURE 4.7 Complete valve symbols

pilot pressure pushes the valve left, increasing flow. This represents a pressure-relief valve which would be connected into a hydraulic system as shown in Figure 4.8c.

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. Figure 4.9 shows the construction and symbol of a simple 2/2 normally closed valve, where depression of the pushbutton lifts the

(a) Infinite position valve

(b) With actuation symbols

(c) Pressure relief valve
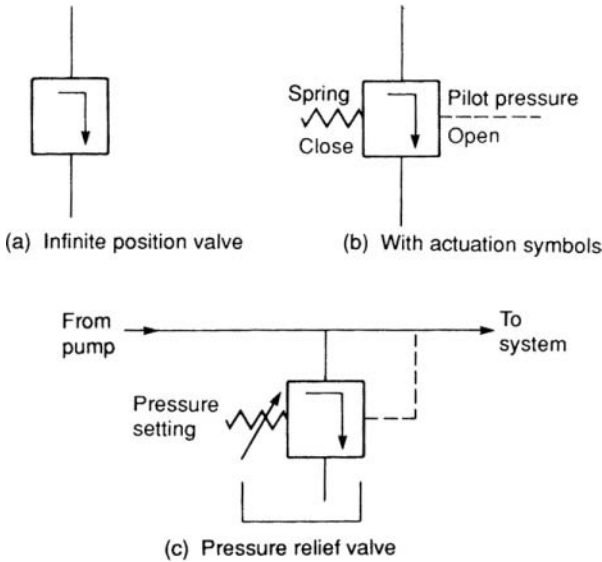
FIGURE 4.8 Infinite position valve symbols

(a) Construction
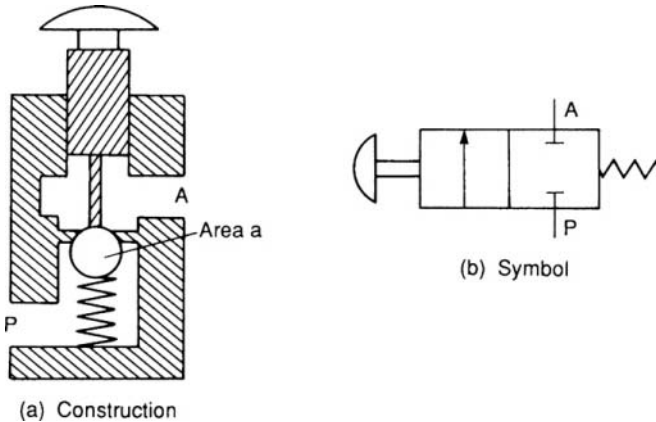
(b) Symbol

FIGURE 4.9 Simple 2/2 poppet valve

ball off its seat and 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.

Figure 4.10 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 Figure 4.11 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,

(a) Construction

(b) Symbol
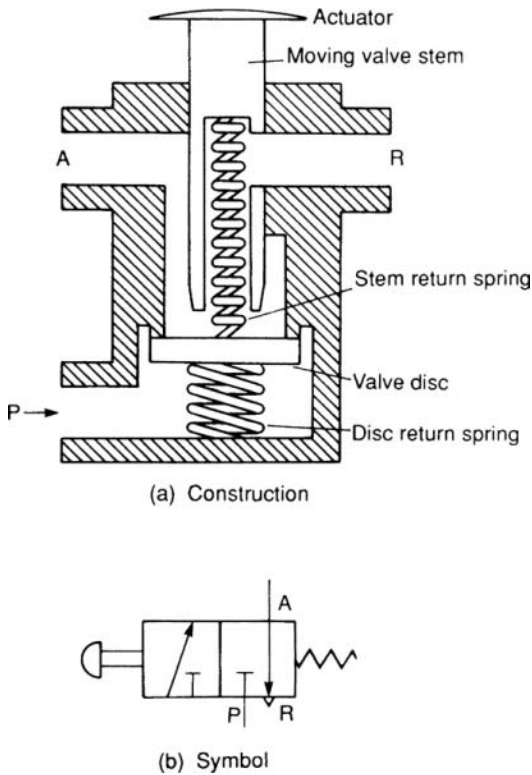
FIGURE 4.10   A 3/2 poppet valve

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.

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

(a) Construction

(b) Symbol
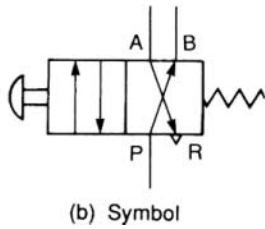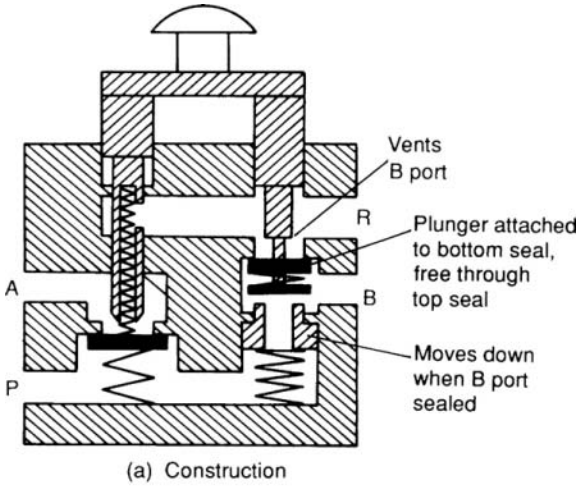
FIGURE 4.11 A 4/2 poppet valve

(a) Symbol

(b) Construction
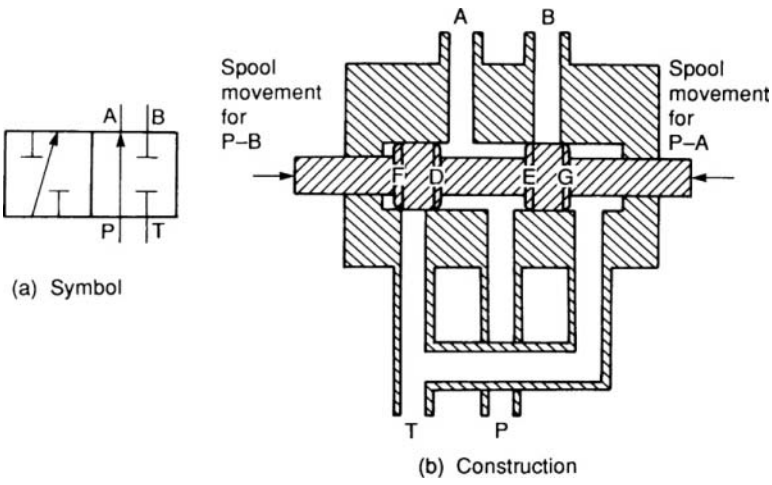
FIGURE 4.12 Two-way spool valve

(a) Symbol

(b) Construction

(c) Symbol showing
transient state
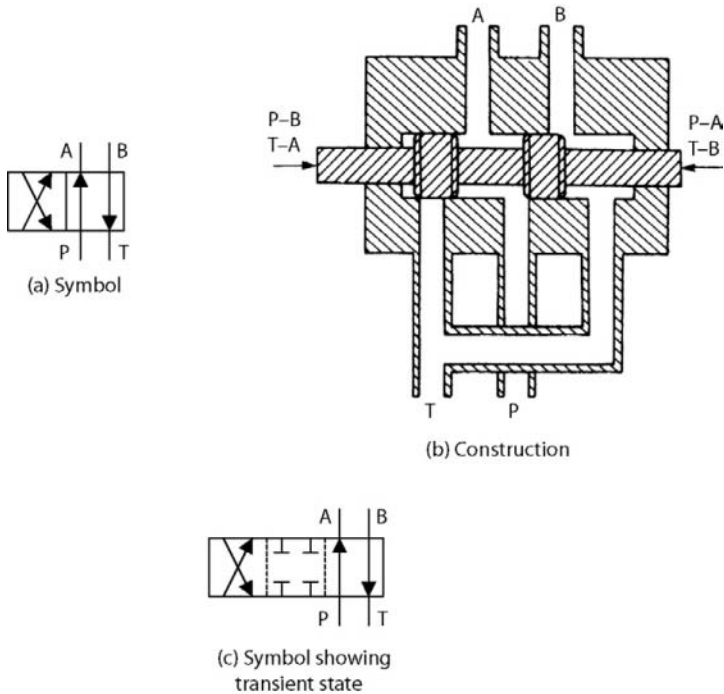
FIGURE 4.13 Four-way spool valve

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.

Figure 4.13 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.

As a valve goes from one state to another the port connections will go through a transitional state. This temporary state may be important in some cases, for example the designers would not want a cylinder supporting a load to be connected to the tank even for a brief moment. If the transitional state is important the valve can be ordered accordingly and the transitional state shown on the diagram. Figure 4.13c shows that the transitional state, denoted by the dashed central lines, blocks all four ports. Note that although this is a two-position valve, with the inclusion of the transitional state it is shown by three blocks.

Figure 4.14 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-centering can easily be provided if springs are mounted at the end of the spool shaft.
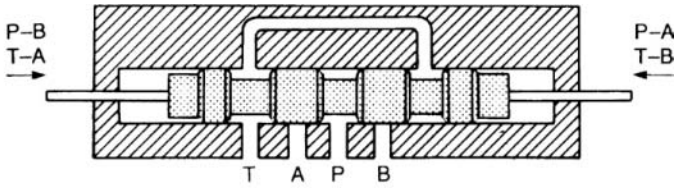
(a) Construction of center off valve
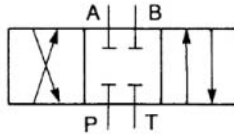
(b) Symbol
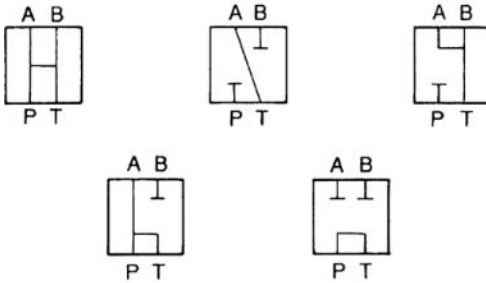
(c) Common center position connections

FIGURE 4.14   Three-position four-way valves

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 that of a 110 V solenoid. 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 plug top. This allows a fault to

(a) 4/3 way valve

(b) Symbol
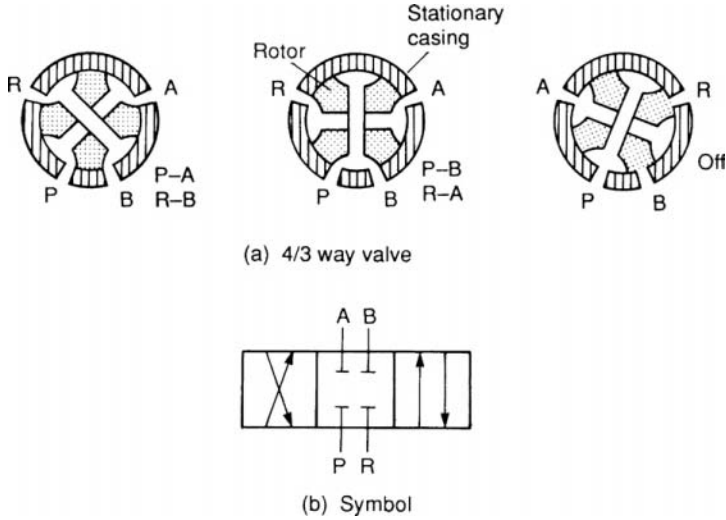
FIGURE 4.15 Rotary valve

be quickly identified as either an electrical or hydraulic problem. Fault finding is discussed further in Chapter 9.

A solenoid can exert a pull or push of about 5–10 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.

Rotary valves

Rotary valves consist of a rotating spool which aligns with holes in the valve casing to give the required operation. Figure 4.15 shows the construction and symbol of a typical valve with center 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.

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 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.
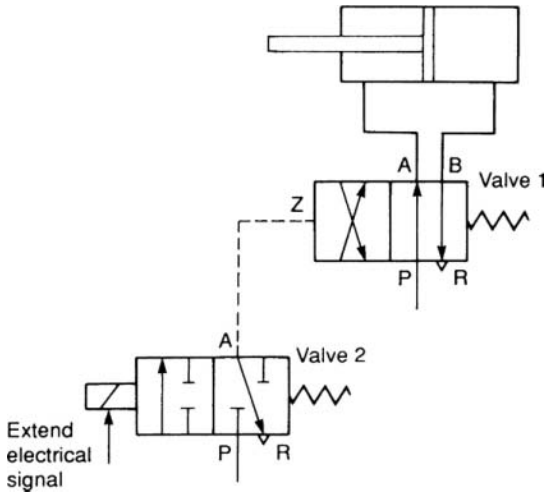
FIGURE 4.16 Pilot-operated valve

In Figure 4.16, pilot port Z is depressurized with the solenoid de-energized, and the ram is retracted. When the solenoid is energized valve 2 changes over, pressurizing Z, causing valve 1 to energize 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 ready-made 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
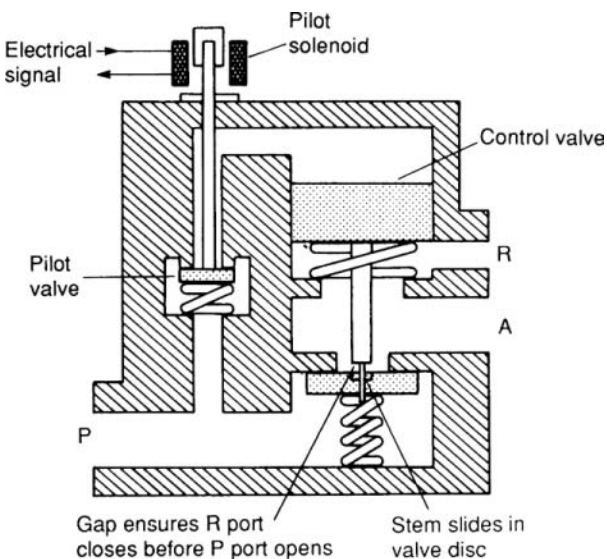
FIGURE 4.17 Construction of a pilot-operated 3/2 valve

Chapter | five

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 X 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:

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

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.
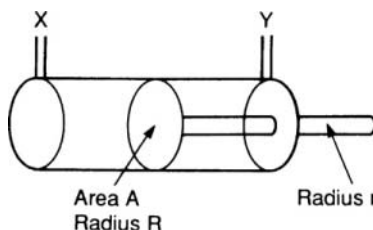
FIGURE 5.1 A simple cylinder
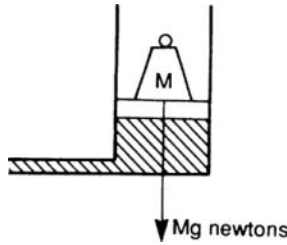
FIGURE 5.2 A mass supported by a cylinder

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.

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:

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

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 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.)

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.
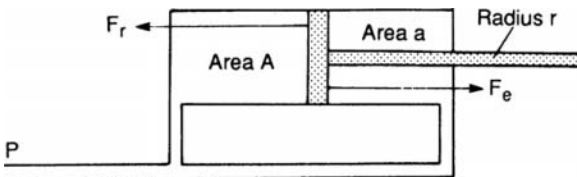
FIGURE 5.3 Pressure applied to both sides of piston
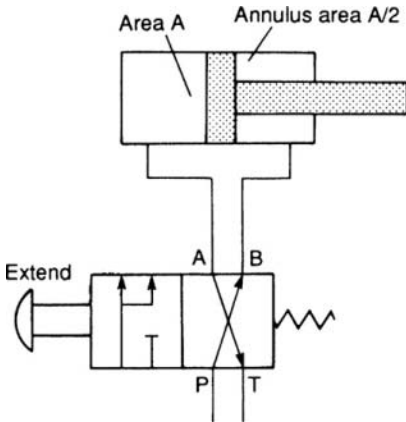
FIGURE 5.4 Cylinder with equal extend/ retract force
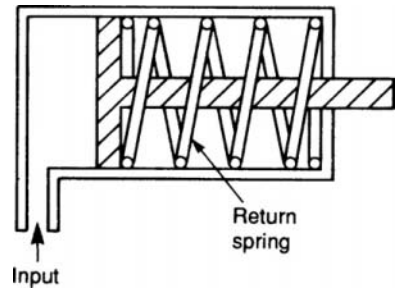
FIGURE 5.5 Single-acting cylinder

Single-acting cylinders are simple to drive (particularly for pneumatic cylinders with quick exhaust valves (see Chapter 4)) but the extend force is reduced and, for spring-return cylinders, the 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. Suppose a cylinder of area A has moved a distance d. This has required a volume V of fluid where:

$$V = Ad \tag{5.3}$$

(a) Construction

(b) Typical application
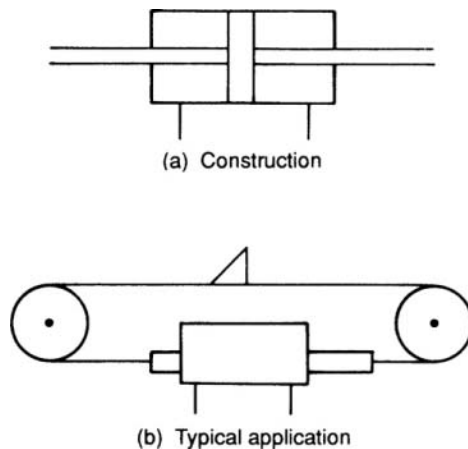
FIGURE 5.6 Double rod cylinder (with equal extend/retract force)

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:

$$V_f = \frac{Ad}{t}$$

$$= Av \tag{5.4}$$

The flow rate units of expression 5.4 depend on the units being used. If d is in meters, v in meters min^{-1} and A in meters^2, flow rate is in meters^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 normalized to atmospheric pressure by using Boyle's law (given in expression 1.17).

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

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

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 5.7 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.

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 5.8.
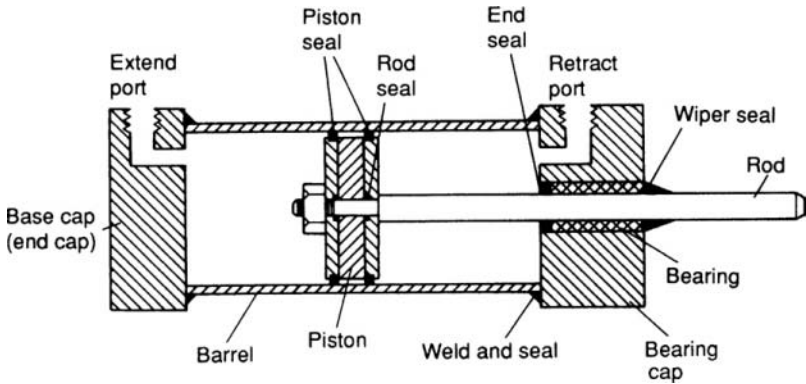
FIGURE 5.7 Construction of a typical cylinder

(a) Enlarged view of bearing cap
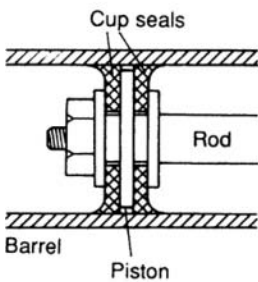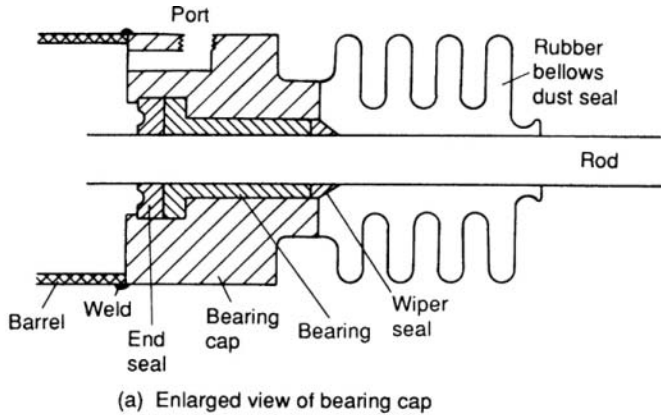
(b) Cup seals
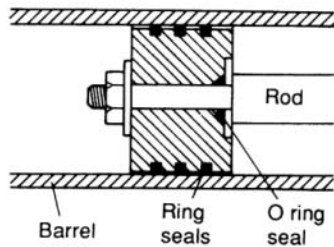
(c) Ring seals and O ring

FIGURE 5.8 Cylinder constructional details

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, a stainless steel, aluminum 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. Alternatively the rod may be plated with chromium which is then polished to give a smooth, corrosion-resistant surface.

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 5.8a) 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 Figure 5.8b and c. Leakage can occur around the rod, so seals are again needed. These can be cup seals (as in Figure 5.8b) which combine the roles of piston and rod seal, or a static O ring around the rod (as in Figure 5.8c).

End caps are generally cast (from iron or aluminum) 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 5.9, for example, exhaust fluid flow is unrestricted until the plunger 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 5.9 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 5.10a 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 5.10b.
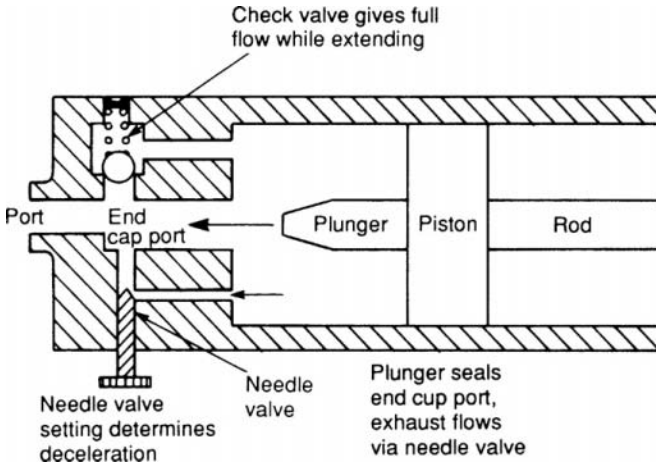
FIGURE 5.9 Cylinder cushioning

(a) Cylinder with a 30 cm stroke

(b) Cylinder with a 60 cm stroke and stop tube
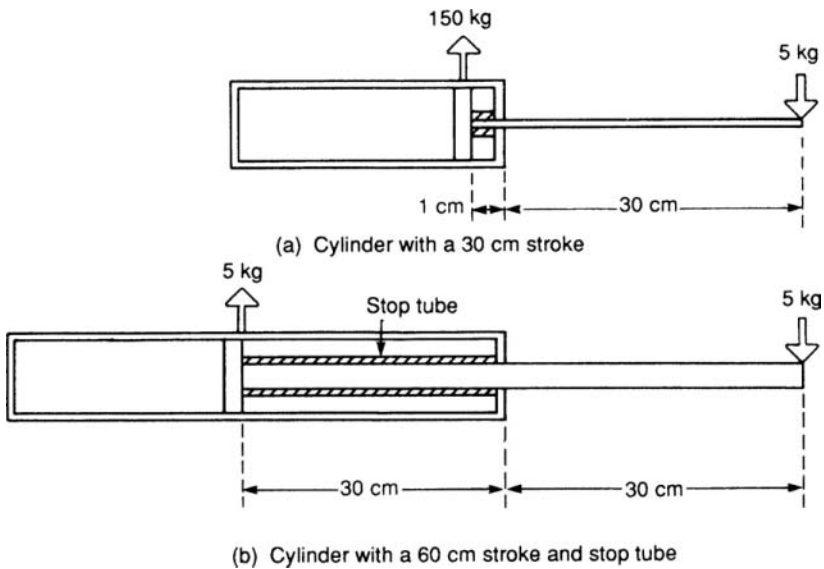
FIGURE 5.10 Side loads and the stop tube

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 5.11 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.
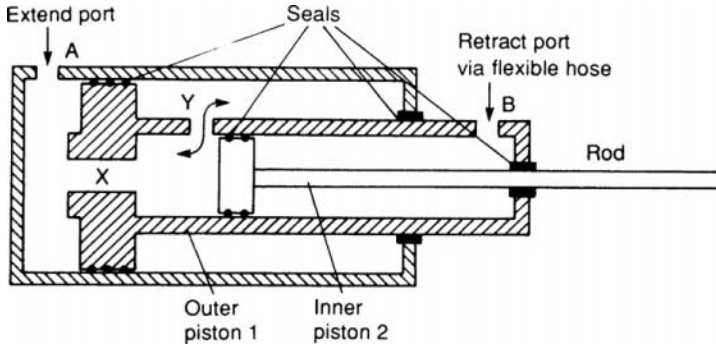
FIGURE 5.11 Two-stage telescopic piston

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 5.12. Pressure is initially applied to port 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}).

When fully extended a conventional cylinder and rod occupies at least twice the length of the stroke. In many applications, such as automatic sliding doors, there is insufficient space to mount a piston/rod cylinder assembly.

Rodless cylinders mount a piston follower on the outside of the cylinder. The simplest construction, shown in Figure 5.13a, uses a magnetic external follower to track the position of a magnetic piston. The maximum force that this type of cylinder can provide is set by the breakaway force between the piston and the follower.
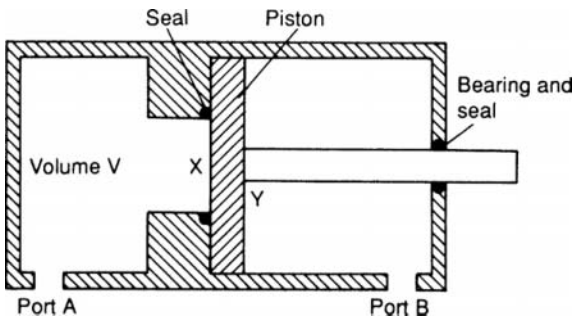
FIGURE 5.12 An impact cylinder
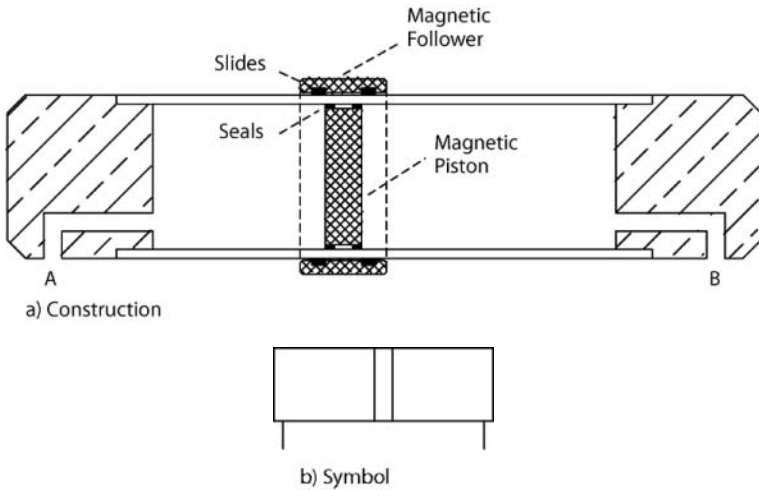
FIGURE 5.13 Rodless cylinders: (a) construction of a magnetic rodless cylinder; (b) symbol

An alternative, but more complex, design has a physical connection between the piston and the follower. This connection link passes through a slot in the cylinder wall which is sealed by magnetic strip seals either side of the link. Although capable of providing the full force of which the cylinder is capable, the seal is vulnerable to dust intrusion which may cause leaks from the seal.

Sequencing applications, particularly those controlled by PLCs, often need to know the position of cylinders. Although conventional limit switches can be mounted to moving parts of the controlled plant it is often cheaper and simpler to use cylinders with integral end of travel limit switches. The piston in the cylinder is made of magnetic material or has embedded magnets and operates simple reed switches mounted on the outside of the cylinder barrel. The reed switches are usually strapped to the barrel with jubilee clips to allow accurate setting of the actuation position. Chapter 8 describes sequencing applications in detail.

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 5.14b requires a cylinder mount which can pivot.

Figure 5.15 shows various mounting methods using these two basic types. The effects of side loads should be considered on non-centerline 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 pressurized. The extending force is given by the expression:

(a) Clamp using front
flange mounting

(b) Pusher using rear trunnion mounting
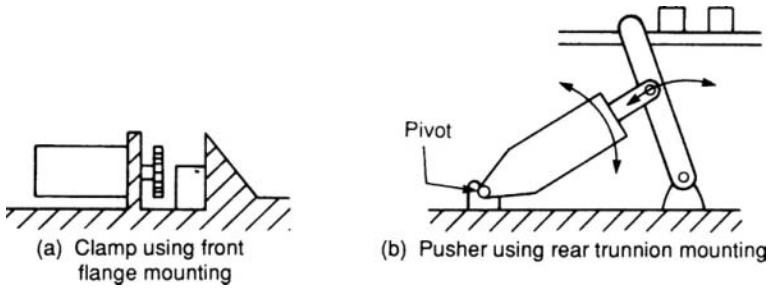
FIGURE 5.14 Basic mounting types

Rear trunnion

Rear flange

Centre trunnion

Front flange

Foot

Threaded
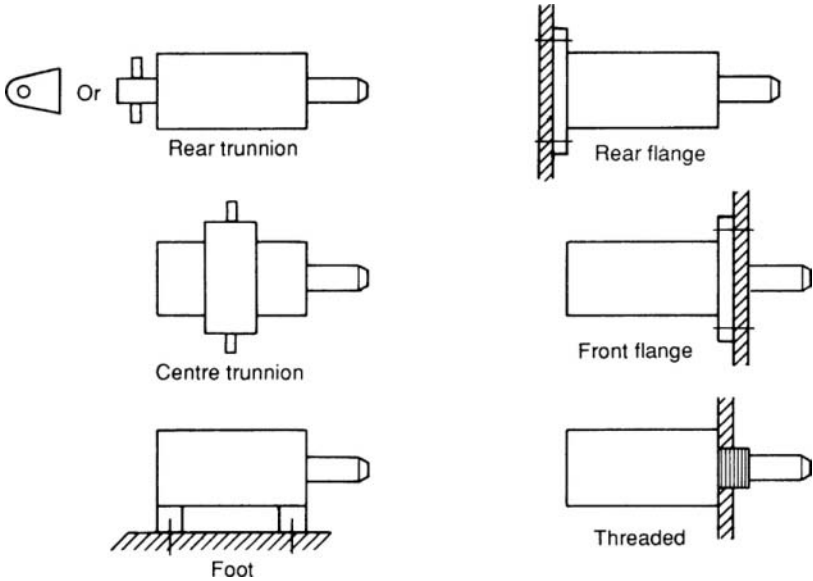
FIGURE 5.15 Methods of cylinder mounting

$$F = P_1 A - P_2 a \qquad (5.6)$$

To lift the load at all, F must be greater than $Mg + f$ where M is the mass and f the static frictional force.

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 $P_1 A$ is larger than $P_2 a$, 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:

$$\text{Acceleration} = \frac{F_a}{M} \qquad (5.7)$$

where $F_a = P_1 A - P_2 a - Mg - f$.
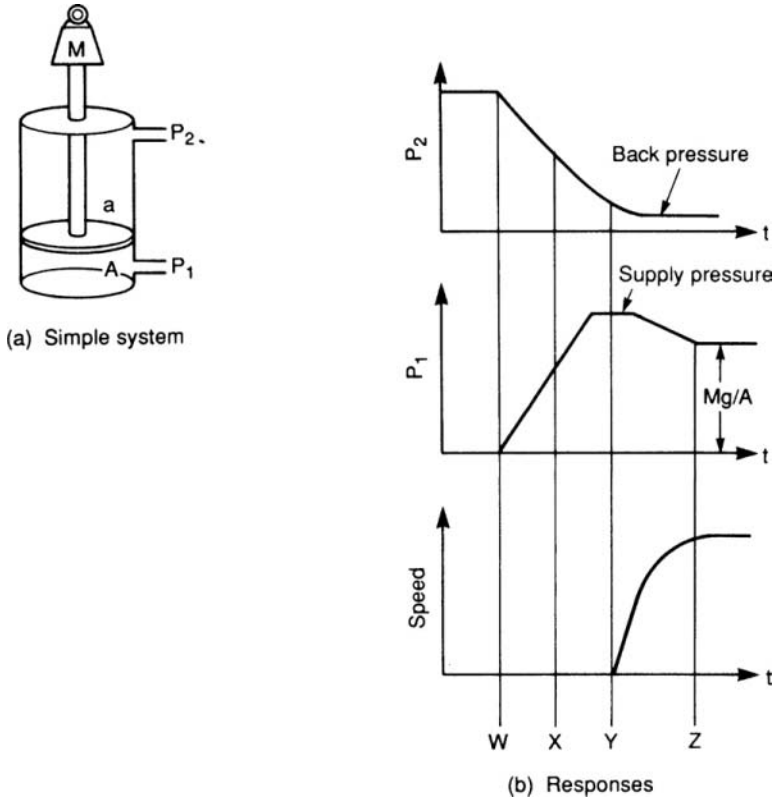
FIGURE 5.16 Cylinder dynamics

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:

$$P_1 = \frac{Mg + f + P_2 a}{A}$$

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 depressurizing 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 $P_1 A$ 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.

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 molded 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.

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.

(a) Simple static seal
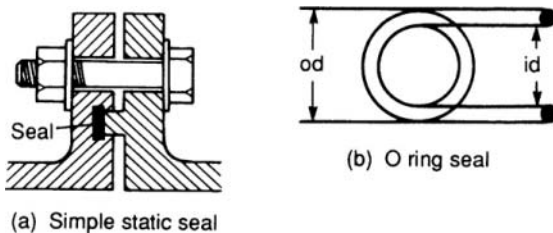
(b) O ring seal

FIGURE 5.17 Static seals

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.

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.

Seals are manufactured from a variety of materials, the choice being determined by the fluid, its operating pressure and the likely temperature range. The
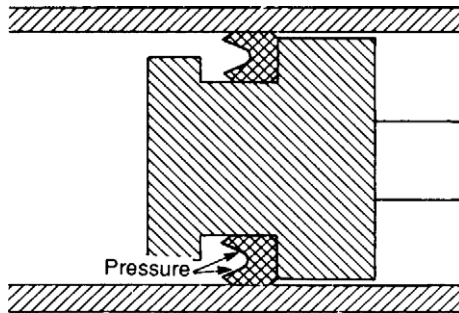
FIGURE 5.18 The U ring seal
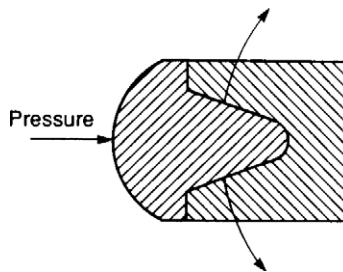
FIGURE 5.19 The composite seal

Creep

Pressure

Anti-extrusion ring

Pressure

(a) Unsecured seal creeps into gap

(b) Backup ring prevents creep
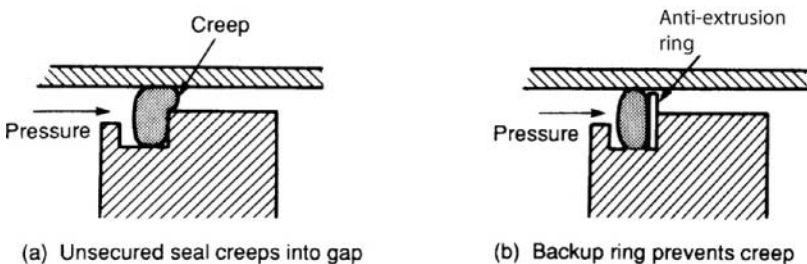
FIGURE 5.20 Anti-extrusion ring
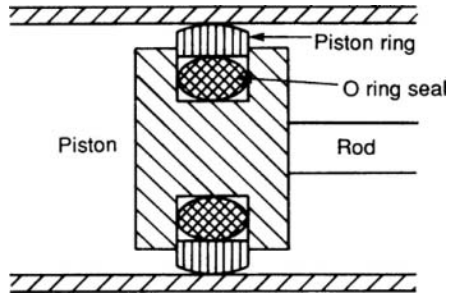
FIGURE 5.21 Combined piston ring and O ring seal (not to scale)

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.

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 meters from a shaft center. The device is then producing a torque T given by the expression:

$$T = Fd \ Nm \tag{5.8}$$
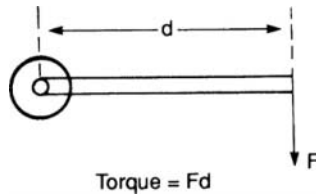
Torque = Fd

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 in Figure 5.23. Obviously, torque is dependent on the applied pressure; increasing the pressure results in increased torque, as shown.

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

$$P = \frac{TR}{9550}\,kW \qquad (5.9)$$

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

$$P = \frac{TR}{5252}\,hp \qquad (5.10)$$

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

$$P = \frac{TR}{63024}\,hp \qquad (5.11)$$

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.

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:

(a) Pneumatic actuator   (b) Hydraulic actuator
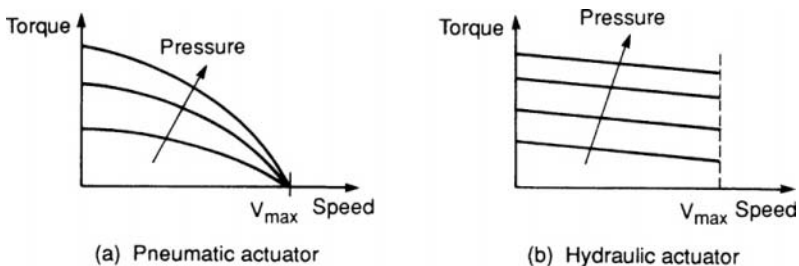
FIGURE 5.23 Torque/speed curves for rotary actuators

Chapter | six

Hydraulic and Pneumatic Accessories

HYDRAULIC RESERVOIRS

A hydraulic system is closed, and the oil used is stored in a tank or reservoir to which it is returned after use. Although probably the most mundane part of the system, the design and maintenance of the reservoir is of paramount importance for reliable operation. Figure 6.1 shows details of a typical reservoir.

The volume of fluid in a tank varies according to temperature and the state of the actuators in the system, being minimum at low temperature with all cylinders extended, and maximum at high temperature with all cylinders retracted. Normally the tank volume is set at the larger of four times the pump draw per minute or twice the external system volume. A substantial space must be provided above the fluid surface to allow for expansion and to prevent any froth on the surface from spilling out.

The tank also serves as a heat exchanger, allowing fluid heat to be removed. To obtain maximum cooling, fluid is forced to follow the walls of the tank, from the return line to pump suction inlet, by a baffle plate down the tank center line.
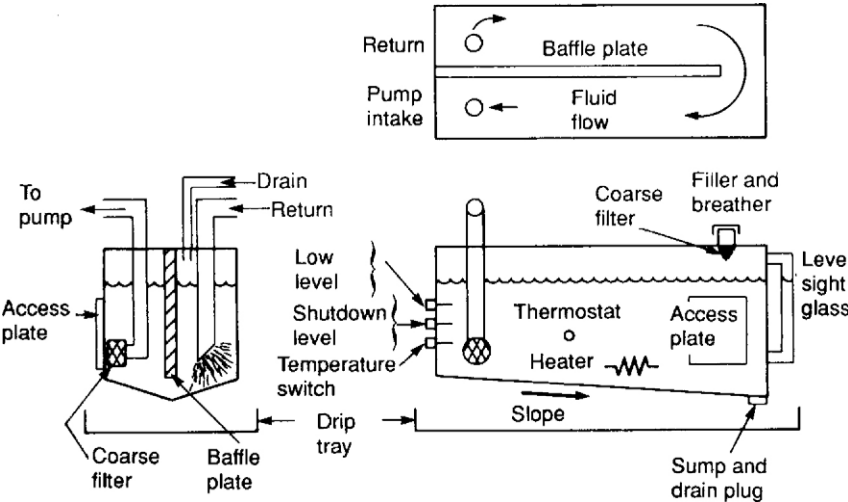
FIGURE 6.1 Construction of a hydraulic reservoir

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Typically this baffle plate will be two-thirds of the height of the fluid surface. This plate also encourages any contamination to fall to the tank bottom before reaching the pump inlet, and allows any entrapped air to escape to the surface. The main return line should enter from the top of the tank to preclude the need for a check valve and end below the minimum tank level to prevent air being drawn into the oil. The return flow should emerge into the tank through a diffuser with a low velocity of around $0.3\,\text{m s}^{-1}$ to prevent disturbance of any deposits at the base of the tank. The flow should be directed at the tank wall to assist cooling. Pump inlet lines should be at least 100 mm above the base of the tank to avoid debris being sucked into the pump.

If, as is commonly the case, the external components outside the tank are below the oil level in the tank the return line should be equipped with a removable anti-siphonage plug. This should be removed to allow air into the return line before any external components are disconnected. Without this precaution a siphon back-flow can occur which is very difficult to stop. If you have never encountered it before the sudden and apparently unstoppable flow of oil from the return pipe on disconnection can be very surprising.

Low-pressure returns (such as drains from motors or valves) must be returned above fluid level to prevent back pressure and formation of hydraulic locks.

Fluid level is critical. If it is too low, a whirlpool forms above the pump inlet, resulting in air being drawn into the pump. This air results in maloperation, and will probably result in pump damage.

A level sight glass is essential to allow maintenance checks to be carried out. The only route for oil to leave a hydraulic system is, of course, by leaks, so the cause of any gross loss of fluid needs investigation. In all bar the smallest and simplest systems, two electrical float switches are generally included, giving a remote (low-level) warning indication and a last ditch (very-low-level) signal which leads to automatic shutdown of the pump before damage can occur.

The temperature of fluid in the tank also needs monitoring and as an absolute minimum a simple visual thermometer should be included. The ideal temperature range is around 45–50 °C and, usually, the problem is keeping the temperature down to this level. Ideally an electrical over-temperature switch is used to warn the user when oil temperature is too high.

When the system is used intermittently, or started up from cold, oil temperature can be too low, leading to sluggish operation and premature wear. A low-temperature thermostat and electrical heater may be included to keep the oil at an optimum temperature when the system is not in use.

Reservoirs are designed to act as collecting points for all the dirt particles and contamination in the system and are generally constructed with a V-shaped cross-section forming a sump. A slight slope ensures contamination collects at the lower end where a drain plug is situated. Often magnetic drain plugs are used to trap metallic particles.

Reservoirs should be drained periodically for cleaning, and a removable man access plate is included for this purpose. This is *not* the most attractive of jobs!

Oil is added through a filler cap in the tank top. This doubles as a breather allowing air into and out of the tank as the volume of fluid changes. A coarse filter below the breather prevents contamination entering the tank as fluid is added.

Tank air filters are commonly forgotten in routine maintenance. The oil in a typical tank changes considerably during operation as temperatures change and actuators operate. This change in volume is directly reflected in air changes both in and out of the tank. The only route for this air flow is through the filters. If these become blocked the tank may become pressurized and fail disastrously.

Reservoirs are generally constructed from welded steel plate with thin side walls to encourage heat loss. The inside of the tank is shot blasted then treated with protective paint to prevent formation of rust particles.

At some time in the life of a hydraulic system there will eventually be oil spillage around the tank, whether from leakage, over-enthusiastic filling or careless maintenance. It is therefore good practice to put substantial drip trays under reservoir pumps and associated valves to limit oil spread when the inevitable mishaps occur.

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 Figure 6.2 a system operates intermittently at a 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 the 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
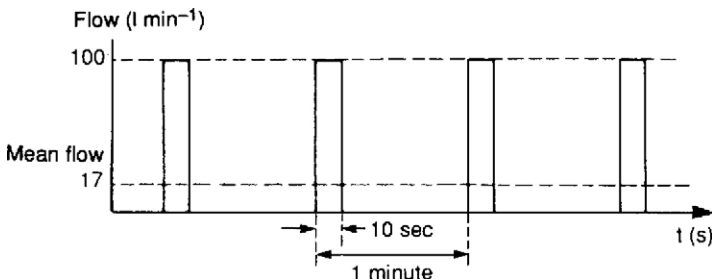
FIGURE 6.2 A simple system with uneven demands. To supply this without an accumulator a 100 L min^{-1} is required although the mean flow is only 17 L min^{-1}

(a) Circuit diagram

(b) Timing chart
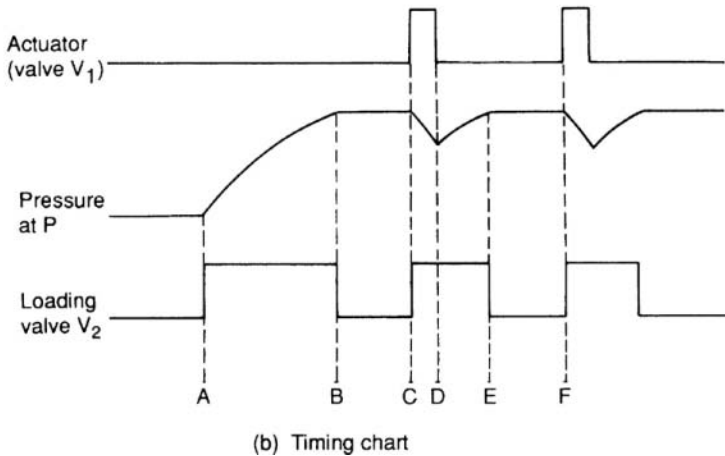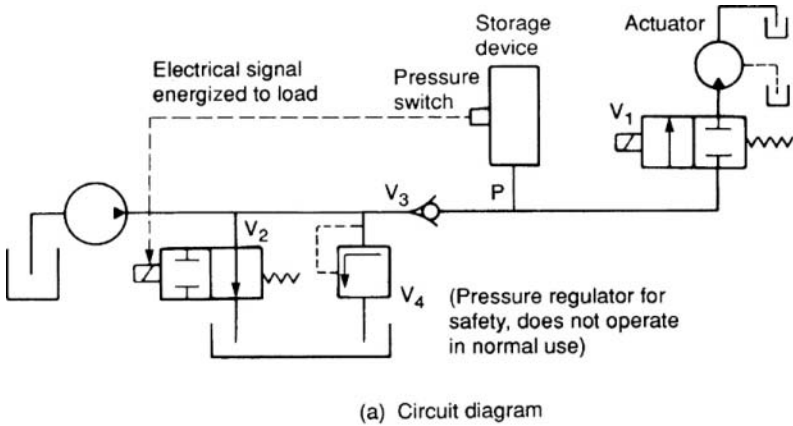
FIGURE 6.3 System with an accumulator

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 F.

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 s 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
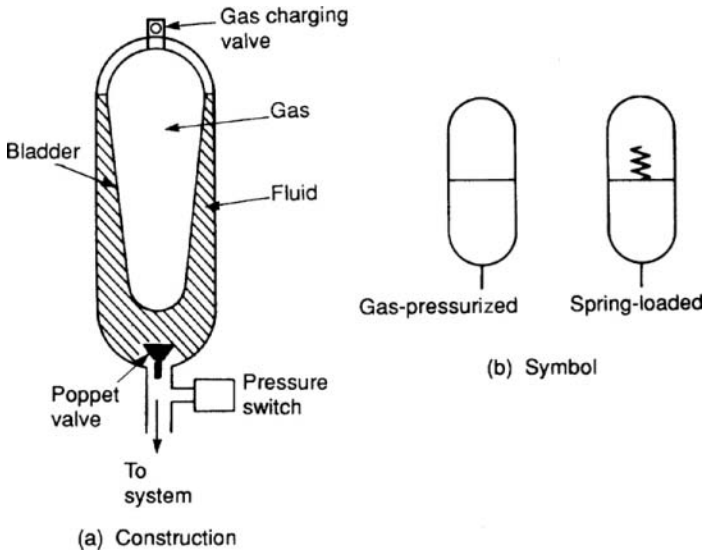
FIGURE 6.4 The accumulator

loads for 17% of the time, with an accumulator a 10 hp motor loads for about 90% of the time.

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 gas-filled bladder accumulator shown in Figure 6.4. 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 Figure 6.2, 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 the volume of the 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.
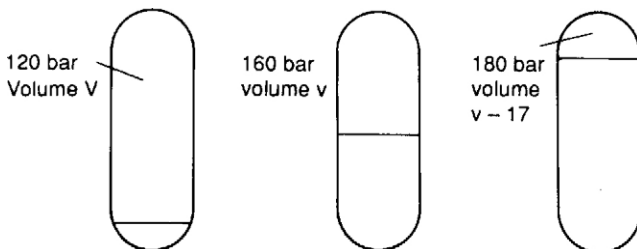
FIGURE 6.5 Sizing an accumulator

From Figures 6.5b and c using Boyle's law:

$$160\,v = 180\,(v-17)$$

which reduces to:

$$n = 153 \text{ liters}$$

From Figure 6.5a:

$$120V = 160 \times 153$$

or:

$$V = 204 \text{ liters}$$

Hence an accumulator of around 250 liters 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. Figure 6.6 shows typical blowdown circuits.

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 pressurized vessel and as such requires certification if it contains more than 250 bar·liters. It will require a recorded expert visual inspection every 5 years and a full volumetric pressure test every 10 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 °C. 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

(a) Manual (b) Automatic
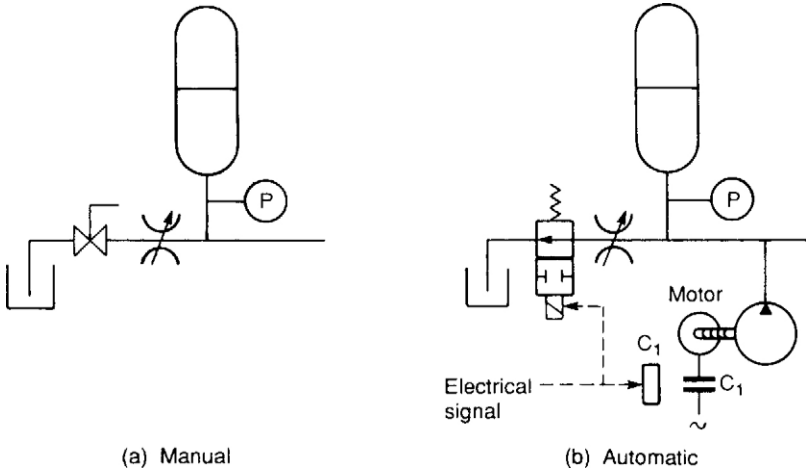
FIGURE 6.6 Accumulator blowdown circuits. In each case flow from the accumulator is restricted to prevent an explosive decompression

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.

Based on a cubic tank where

$$L^3 = v$$

Surface area $= 5 \times L^2$ (to allow for air gap at top and poor heat transfer from base)

Heat loss approx. 0.3 kW m^{-2}
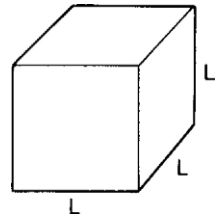
TABLE 6.1 Heat loss for various tank volumes. These are only approximate as few tanks are pure cubes

Vol (l)	L (m)	Surface area (m²)	Heat loss (kW)
250	0.63	1.98	0.5
500	0.8	3.2	1.0
1000	1.0	5.0	1.5
2000	1.25	7.8	2.5
10,000	2.15	23.1	15.0

(a) Shell and tube heat exchanger

(b) Air cooler
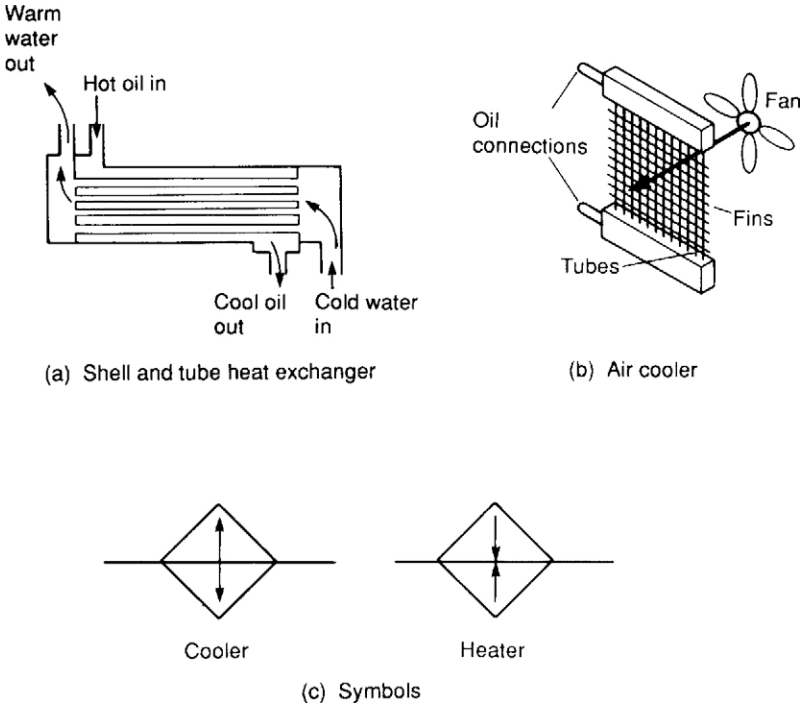
(c) Symbols

FIGURE 6.7 Coolers and heat exchangers

Figure 6.7 shows two types of cooler and their symbols. Water cooling is most common and Figure 6.7a shows the usual form of a shell and tube heat exchanger which is fitted in the return line to the tank. Note that the cooling water flows in the opposite direction to the oil (giving rise to the term: counter-flow cooler). If the system is open to atmosphere and liable to stand unused in cold weather, protection must be included to prevent frost damage which can result in water-contaminated oil.

Air cooling is also common, shown in Figure 6.7b, with fans blowing air through a radiator matrix similar to those in motor cars (but, obviously, with a far higher pressure rating). Air cooling is noisy and occupies more space than a water cooler, but does not have the danger of contamination from leaks inside a water cooler.

HYDRAULIC FLUIDS

The liquid in a hydraulic system is used to convey energy and produce the required force at the actuators. Very early systems used water (in fact the name hydraulic implies water) but water has many disadvantages, the most obvious of which are its relatively high freezing point of $0\,°C$, its expansion when freezing, its corrosive (rust formation) properties and poor lubrication. Modern fluids designed specifically for hydraulic circuits have therefore been developed.

(a) Unlubricated,
metal-to-metal contact

(b) Lubricant separates
surfaces
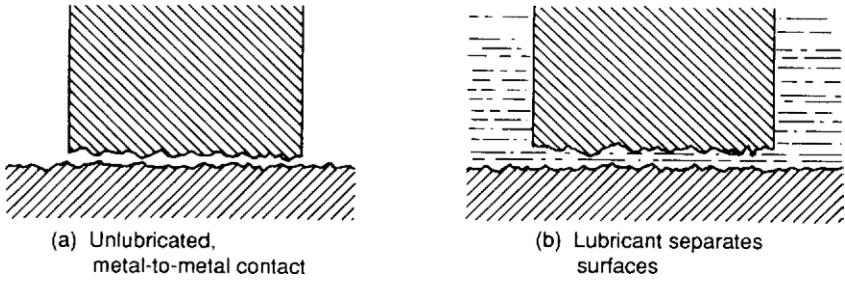
FIGURE 6.8 Need for lubrication from hydraulic fluid

The fluid conveys power in a hydraulic circuit, but it must also have other properties. Chapter 5 described the seals found in actuators. Moving parts in valves do not have seals; instead they rely on fine machining of spools and body to form the seal in conjunction with the fluid. Despite fine machining, irregularities still occur on the surface, shown in exaggerated form in Figure 6.8a. The fluid is required to pass between the two surfaces, holding them apart as in Figure 6.8b, to reduce friction and prevent metal-to-metal contact which causes premature wear. Sealing and lubrication are therefore two important properties of hydraulic fluid.

The temperature of hydraulic fluid tends to rise with the work done, an ideal operating temperature being around 50 °C (a useful quick check is to touch pipes in a system: the hand can be left indefinitely on metal at 40 °C; can touch metal at 50 °C but long contact is distinctly uncomfortable; but cannot be left for more than a second or so on metal at 60 °C. If you cannot touch the pipes, the oil is too hot!). The fluid must be able to convey heat from where it is generated (valves, actuators, frictional losses in pipes) and must not be affected itself by temperature changes.

The fluid can cause deterioration of components. An extreme case is water causing rust, but less obvious reactions occur. A water–glycol fluid, for example, attacks zinc, magnesium and cadmium – all fairly common materials. Some synthetic fluids interact with nitrile and neoprene, and special paint is needed on the inside of the reservoir with some fluids. The fluid must therefore be chosen to be compatible with the rest of the system.

The fluid itself comes under attack from oxygen in air. Oxidation of fluid (usually based on carbon and hydrogen molecules) leads to deleterious changes in characteristics and the formation of sludge or gum at low-velocity points in the system. The resulting oxidation products are acidic in nature, leading to corrosion. The fluid of course must be chemically stable and not suffer from oxidation. The temperature of fluid strongly influences the rate of oxidation, which rises rapidly with increasing temperature.

The most common hydraulic fluid is petroleum-based oil (similar to car engine oil) with additions to improve lubrication, reduce foaming and inhibit rust. With the correct additives it meets all the requirements and does not react adversely with any common materials.

TABLE 6.2 ISO designations for fire-resistant fluids

	ISO designation
High water fluids	HFA
Water in oil (invert emulsions)	HFB
Water–glycols	HFC
Synthetics	HFD

Its one major disadvantage is flammability; petroleum oils readily ignite. Although few (if any) hydraulic systems operate at temperatures that could ignite the oil, a major leak *could* bring spilt oil into contact with an ignition source. The probability of leakage needs consideration if petroleum oils are to be used.

If safety dictates that a fire-resistant fluid is required there are four basic choices (Table 6.2).

The most common form is a water-in-oil emulsion (roughly 40% water, 60% oil). Oil-in-water emulsions are sometimes used, but their lubricating properties are poor. Both types of mixture have a tendency to form rust and to foam, but these characteristics can be overcome by suitable additives. Both types also need regular checking to ensure the correct oil/water ratio is being maintained.

Another non-flammable fluid is a water–glycol mix. This consists of roughly equal proportions of water and glycol (similar to car antifreeze) plus additions to improve viscosity (see below), inhibit foaming and prevent rust to which water-based fluids are vulnerable. Glycol-based fluids interact with many common materials (particularly zinc, cadmium and magnesium), so the system components must be carefully chosen.

High water content fluids (HFA) use around 95% water with 5% additives, making them totally non-flammable. They are often called 95/5 microemulsion. Their use needs some care as they have very low viscosity, for all practical purposes the same as water, making applications using them prone to leaks at joints and seals. Unlike normal fluids small external leaks can be difficult to see as at the normal 40–50 °C operating temperature the fluid evaporates away without leaving any trace.

Spool valves have an inherent leakage and this can be problematical with low-viscosity fluids such as HFA. Cartridge valves, described in Chapter 4, are therefore often used with HFA.

The high water content makes precautions against rust very important. Any 95/5 components removed from service must be protected against exposure to air. Some manufacturers will not honor their warranties when 95/5 fluid has been used.

Regular checks on water-based fluids are essential to maintain the correct oil/water ratio, the presence of the correct amount of additives and the pH level, all of which will be changed by evaporation. As evaporation increases with temperature water-based fluids are normally operated at a lower temperature than petroleum fluids.

Synthetic fluids based on chemicals such as phosphate esters are also non-flammable and can be used at very high temperatures. These tend to have high densities, which limit the height allowed between tank and pump inlet without cavitation occurring, and do not operate well at low temperatures. Systems with synthetic fluids usually require heaters in the tank to preheat fluid to operating temperature. Synthetic fluids are the most expensive form of hydraulic oil.

Viscosity

The properties of a liquid are largely determined by its resistance to flow, which is termed its viscosity. In non-scientific terms we talk about treacle having high viscosity, and water having low viscosity. Both extremes bring problems; a low-viscosity fluid flows easily and wastes little energy, but increases losses from leakage. A viscous fluid seals well, but is sluggish and leads to energy and pressure losses around the system. Hydraulic fluid has to hit a happy medium between these extremes, so some way of defining viscosity is required.

There are basically two techniques of specifying viscosity. Absolute viscosity measures the shear force between two plates separated by a thin fluid film, shown in Figure 6.9. The most common unit is the poise (a cgs unit) which is the measure of shear force in dynes, for surface areas of $1\,cm^2$ separated by $1\,cm$ of fluid. The centipoise (0.01 poise) is a more practical unit.

Kinematic viscosity is the most common, and most practical, way of measuring viscosity. It is measured by recording the time a fixed quantity of oil takes to flow through a capillary tube. It is defined with a unit called the stokes which is the absolute viscosity (in poise) divided by the density (in $g\ cm^{-3}$). A practical unit is the centi-stokes; a typical hydraulic fluid will have a viscosity of around 40 centi-stokes and low-viscosity fluid such as HFA about 1 centi-stoke. Not surprisingly this much lower viscosity means that HFA is very prone to leaks.

The poise and the stokes are units denoting scientific definitions of viscosity. In hydraulics, all that is really needed is a relative comparison between different liquids. This is achieved with the practical experiment shown in Figure 6.10, where a fixed volume of oil is heated to a test temperature then allowed to drain out through a fixed-sized valve. The time taken to drain in seconds is a measure of the viscosity (being high for high-viscosity liquids and low for low-viscosity liquids).

The test of Figure 6.10 (generally performed at $100\,°F$ and $210\,°F$ with a volume of $60\,cm^3$) gives viscosity in saybolt universal seconds (SUS). The
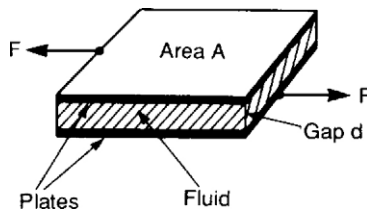
FIGURE 6.9 Scientific definition of viscosity in terms of shear force
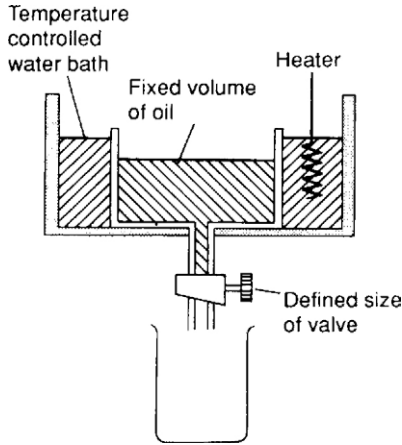
FIGURE 6.10 Practical definition of viscosity

Fahrenheit basis of these definitions comes from the American origin. Hydraulic fluid normally has a viscosity between 150 and 250 SUS defined at 100 °F, although higher values are used in high-temperature applications.

Viscosity can also be given by similar tests for engine oils devised by the American Society of Automotive Engineers (SAE). These give Winter numbers with suffix W (e.g. 10W, 20W) defined at 0 °F, and Summer numbers defined at 210 °F. An oil rating of 10W SAE, for example, covers the range 6000–12,000 SUS at 0 °F, while 30SAE covers the range 58–70 SUS at 210 °F.

Viscosity decreases with increasing temperature, and this is given in SAE units in the form SAE 10W50, for example. This variation in viscosity with temperature is defined by the viscosity index, a unit based on an arbitrary scale from zero (poor, large variation in viscosity with temperature) to 100 (good, small variation with temperature). The range zero to 100 was chosen to relate to standards obtainable with practical fluids rather than some absolute measurable standard. Most hydraulic oils have a viscosity index of about 90.

The reliability of a hydraulic system is strongly influenced by the state of fluid. Contamination from dirt or the products of oxidation and deterioration of a fluid's lubrication ability will lead to rapid wear and failure.

Pour point

The pour point temperature is the lowest temperature at which a fluid will flow. In hydraulic systems the pour point temperature should be at least 10°C below the lowest temperature at which the system will operate.

Environmental fluids

The environmental effects of hydraulic leaks can be large. Leaked oil permeates down through the ground, eventually reaching the water table where it can spread beyond the boundaries of the offending plant. The growing interest in

the environment has therefore led to the introduction of hydraulic fluids with minimal environmental impact. There are two considerations: toxicity, which is the harm that fluid can cause to animals and plants, and biodegradability, which describes the speed at which the fluid changes to innocuous products.

Three different base oils are used as environmentally safe hydraulic fluids. The commonest, and cheapest, is vegetable oil (such as rape seed oil). This has excellent natural biodegradability. In most environmentally sensitive applications vegetable oil is the main choice.

Other oil types are polyglycols and synthetic esters. These are more expensive than vegetable oil, markedly so for synthetic esters, and are not as environmentally friendly. They are used where the characteristics of vegetable oil, such as a limited temperature range and poor water tolerance, preclude its use. They can also be used as a blend with vegetable oil.

Whichever base oil is used additives will still be required to provide lubrication, reduce foaming, etc. These additives must also have low toxicity and be biodegradable.

PNEUMATIC PIPING, HOSES AND CONNECTIONS

The various end devices in a pneumatic system are linked to the air receiver by pipes, tubes or hoses. In many schemes the air supply is installed as a fixed service similar, in principle, to an electrical ring main, allowing future devices to be added as required. Generally, distribution is arranged as a manifold (as in Figure 6.11a) or as a ring main (as in Figure 6.11b). With strategically placed isolation valves, a ring main has the advantage that parts of the ring can be isolated for maintenance, modification or repair without affecting the rest of the system.

Pneumatic systems are vulnerable to moisture and, to provide drainage, the piping should be installed with a slope of about 1% (1 in 100) down from the res-

(a) Manifold

(b) Ring

(c) Prevention of water ingress
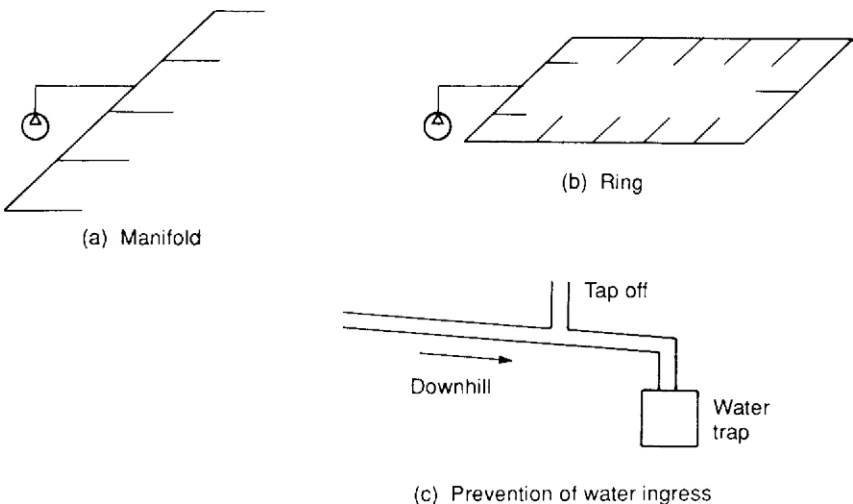
FIGURE 6.11 Pneumatic piping

ervoir. A water trap fitted at the lowest point of the system allows condensation to be run off, and all tapoffs are taken from the top of the pipe (Figure 6.11c) to prevent water collecting in branch lines.

The pipe sizing should be chosen to keep the pressure reasonably constant over the whole system. The pressure drop is dependent on maximum flow, working pressure, length of line, fittings in the line (e.g. elbows, T-pieces, valves) and the allowable pressure drop. The aim should be to keep air flow non-turbulent (laminar or streamline flow). Pipe suppliers provide tables or nomographs linking pressure drops to pipe length and different pipe diameters. Pipe fittings are generally specified in terms of an equivalent length of standard pipe (a 90 mm elbow, for example, is equivalent in terms of pressure drop to 1 meter of a 90 mm pipe). If an intermittent large load causes local pressure drops, installation of an additional air receiver by the load can reduce its effect on the rest of the system. The local receiver is serving a similar role to a smoothing capacitor in an electronic power supply, or an accumulator in a hydraulic circuit.

If a pneumatic system is installed as a plant service (rather than for a specific well-defined purpose) pipe sizing should always be chosen conservatively to allow for future developments. Doubling a pipe diameter gives four times the cross-sectional area, and pressure drops lowered by a factor of at least ten. Retrofitting larger size piping is far more expensive than installing original piping with substantial allowance for growth.

Black steel piping is primarily used for main pipe runs, with elbow connections where bends are needed (piping, unlike tubing, cannot be bent). Tubing, manufactured to a better finish and more accurate inside and outside diameters from drawn or extruded flexible metals such as brass, copper or aluminum, is used for smaller-diameter lines. As a very rough rule, tubing is used below 25 mm and piping above 50 mm – diameters in between are determined by the application. A main advantage of tubing is that swept angles and corners can be formed with bending machines to give simpler and leak-free installations, and minimizing the pressure drops associated with fittings.

Connections can be made by welding, threaded connections, flanges or compression tube connectors. (Examples of compression fittings are illustrated in Figure 6.12.)

Welded connections are leak-free and robust, and are the prime choice for fixed main distribution pipe lines. Welding does, however, cause scale to be deposited inside the pipe, which must be removed before use.

Threaded pipe connections must obviously have male threads on the pipes. Industry suffers from a lack of consistency in many areas, and not least of these is in standards for screw threads. In pneumatics and hydraulics piping the following pipe fittings may be encountered: American National Pipe Thread (NPT); British Standard Pipe Thread (BSP); Metric Pipe Thread (M10 to M45); Unified Pipe Thread (UNF). Of these NPT and BSP are the commonest.

The choice between these is determined by the standards already chosen for a user's site. Taper threads are cone-shaped and form a seal between the male and
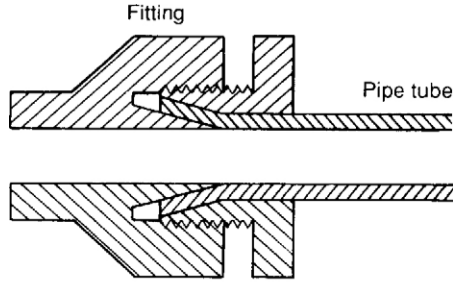
(a) Flanged connector
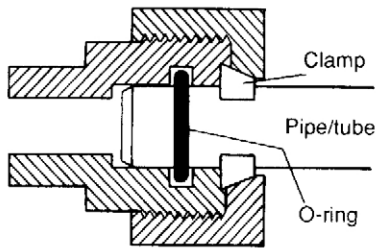
(b) O-ring

FIGURE 6.12 Compression fittings

female parts as they tighten, with assistance from a jointing compound or plastic tapes. Parallel threads are cheaper, but need an O-ring to provide the seal.

A pipe run can be subject to shock loads from pressure changes inside the pipe, and there can also be accidental outside impacts. Piping must therefore be securely mounted and protected where there is a danger from accidental damage. In-line fittings such as valves, filters and treatment units should have their own mounting and not rely on piping on either side for support.

At the relatively low pressure of pneumatic systems (typically 5–10 bar), most common piping has a more than adequate safety margin. Pipe strength should, however, be checked – as a burst air line will scatter shrapnel-like fragments at high speed.

Plastic tubing is used for low-pressure (around 6 bar) lines where flexibility is needed. One advantage of plastic tubing is the availability of multiple colors allowing easy tracing of pipe runs around a machine. Connections are usually made with barbed push-on connectors, illustrated in Figure 6.13. or by quick-release push-in, push-out connectors. Care should be taken when quick-release couplings are used because a volume of compressed air may be trapped downstream of the coupling and released explosively when the pipe is open ended.

Where flexibility is needed at higher pressure, hosing can be used. Pneumatic hoses are constructed with three concentric layers: an inner tube made of
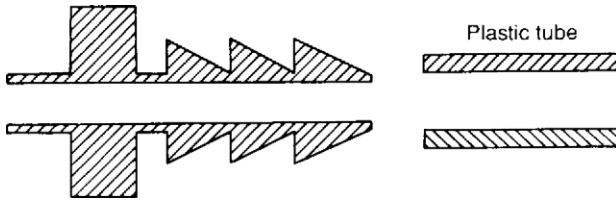
FIGURE 6.13 Barbed connector for plastic tube

synthetic rubber surrounded by a reinforcement material such as metal braiding. A plastic outer layer is then used to protect the hosing from abrasion.

Hose fittings need care in use, as they must clamp tightly onto the hose, but not so tightly as to cut through the reinforcement. Quick-disconnect couplings are used where hoses are to be attached and disconnected without the need of shut-off valves. These contain a spring-loaded poppet which closes the outlet when the hose is removed. There is always a brief blast of air as the connection is made or broken, which can eject any dirt around the connector at high speed. Extreme care must therefore be taken when using quick-disconnect couplings.

COST OF AIR LEAKS

Air leaks from a compressed air system are very noisy and may even be hazardous to health. They are also surprisingly expensive. The table below shows the required compressor power and the corresponding annual costs of various sizes of leaks. It is assumed that the air is at a standard 7 bar in a factory which runs 24 hours a day for 50 weeks a year and pays 10 p per kWh for their electricity.

Hole diam.	Leakage	Comp. power	Annual cost
mm	$l\,s^{-1}$	kW	
0.5	0.2	0.06	£50
1.0	0.8	0.24	£200
1.5	1.8	0.54	£450
2.0	3.1	0.93	£800
6.0	28.2	8.50	£7200

Note that the cost rises as the square of the diameter of the leakage hole; twice the diameter gives four times the cost. Fortunately the noise makes leaks relatively easy to find.

SILENCERS

When a cylinder or other actuator moves at high speed air has to be quickly removed from the non-pressurized port of the actuator. This air invariably vents to atmosphere. When the cylinder is large or the speed of motion is fast the exhaust

a) Silencer with adjustable
discharge rate

b) Reclassifier (filter/silencer)
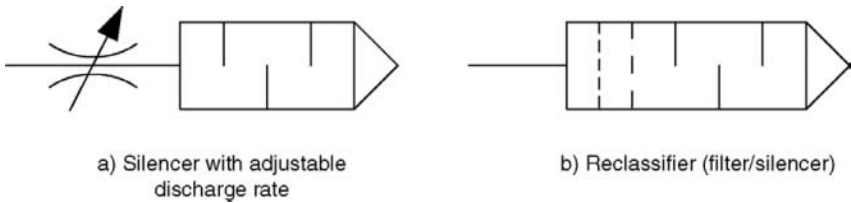
FIGURE 6.14 Silencer and filter silencer symbols

air exits at high speed with a pistol-like crack. This noise is very intrusive and can even be of a level which can cause hearing damage.

One solution is to put flow restrictors in the exhaust line to reduce the velocity and extend the time over which the air blast takes place. Alternatively, specially designed silencers can be fitted to the actuator or valve. These are generally constructed as a porous mesh through which the air loses velocity and exhausts over a wide area, reducing the force and the noise. Often a silencer has a built-in adjustable flow restrictor.

A related device is an exhaust filter/silencer (also known as a reclassifier), which is used when the exhausting air has a significant oil content. The silencer incorporates a filter which captures the oil in the air to reduce airborne contamination. The captured oil runs down into a sump which must be manually drained at regular intervals.

Symbols for a silencer and filter/silencer are shown in Figure 6.14.

HYDRAULIC PIPING, HOSING AND CONNECTIONS

The differences between hydraulic and pneumatic piping primarily arise from the far higher operating pressures in a hydraulic system.

Particular care has to be taken to check the pressure rating of pipes, tubing, hosing and fittings, specified as the bursting pressure. A safety factor is defined as:

$$\text{Safety factor} = \frac{\text{bursting pressure}}{\text{working pressure}}$$

Up to 60 bar, a safety factor of eight should be used, between 60 and 150 bar a safety factor of six is recommended, while above 150 bar a safety factor of four is required. This may be compared with pneumatic systems where safety factors of around 40 are normally obtained with simple standard components.

The choice of piping or tubing is usually a direct consequence of pressure rating. These can be manufactured as welded or drawn (seamless) pipe. Welded pipe has an inherent weakness down the welded seam, making seamless pipes or tubing the preferred choice for all but the lowest-pressure hydraulic systems.

Hydraulic piping is specified by wall thickness (which determines the pressure rating) and outside diameter (OD, which determines the size of fittings to be used). It follows that for a given OD, a higher-pressure pipe has a smaller inside diameter (ID). American piping is manufactured to American National Standards Institute (ANSI) specifications, which define 10 sets of wall thickness

as a schedule number from 10 to 160. The higher the number, the higher the pressure rating. 'Standard' piping is schedule 40.

Pipes should be sized to give a specified flow velocity according to the expected flow. Typical flow velocities are 7–8 m s^{-1} for a pressure line, and 3–4 m s^{-1} for a return line. The lower velocity is specified for the return line to reduce the back pressure. For a similar reason the velocity in a pump suction line should be in the range 1.5–2 m s^{-1}. At the point of exit from the return line diffuser into the tank the velocity should be very low, below 0.3 m s^{-1}, to prevent stirring up any contamination at the base of the tank.

Like pneumatic piping, joints can be made by welding, with compression fittings (similar to those in Figure 6.12 but of higher pressure rating) or threaded connections and flanges. Particular care needs to be taken to avoid leaks at joints; in pneumatic systems a leak leads to loss of downstream pressure and perhaps an objectionable noise whereas a hydraulic leak loses expensive fluid and creates an oil pool which is a fire and safety hazard.

Flexible hosing is constructed in several concentric layers, with the inner tubing being chosen to be compatible with the hydraulic fluid and its temperature. One (or more) braided reinforcing layers are used. At higher pressures the braiding will be wire. The outer layer is designed to resist abrasion and protect the inner layers. Hoses are generally manufactured complete with fittings. Hydraulic hoses, like pneumatic hoses, must be installed without twists (which can lead to failure at the fittings).

Quick-disconnect hydraulic connections are available, but the higher pressure, risk of spillage and danger of introducing dust into the system restricts their usage.

HYDRAULIC AND PNEUMATIC FUSES

A fuse protects a system against component failures. There are actually two distinct types of fuse and their operation, and effects, are totally different.

The first type of fuse protects against plant damage from over-pressurization. Mounted, for example, in a line directly from pump outlet to tank it is a simple thin metal plate which will rupture at a specified pressure. Once ruptured the entire pump delivery goes straight back to the tank, leaving minimal system pressure. Once this type of fuse has blown the system has to be shut down and the fuse replaced. Note, though, that the sudden and catastrophic loss of pressure can cause unwanted movement and safety hazards.

The second type of fuse is common in aircraft and mobile plant hydraulics and protects against a failure of a pipe or connection which results in a major leak. It detects higher than normal flow in a line and shuts to prevent further leakage. Usually the fuse has to be manually reset. Although it does totally shut down one motion it is probable that other motions can still continue, allowing the plant to be brought into a safe state. Whilst working well for major leaks they are not capable of detecting small dripping leaks.

Pneumatic fuses also protect a pneumatic system against major leaks. If a flexible hose for a pneumatic tool fails or a coupling becomes detached the sudden very high air flow will cause the hose to thrash around with the potential to cause major injury. Like hydraulic fuses they have to be reset manually once triggered.

Chapter | seven

Process Control Pneumatics

If some industrial process is to be automatically controlled, there will be many process variables (e.g. temperature, flow, pressure, level) which need to be measured and kept at the correct value for safety and economical operation. In Figure 7.1, for example, water flow in a pipe is to be kept at some preset value.

In Figure 7.1 the flow is measured to give the current value (usually termed PV for process variable). This is compared with the required flow (called SP for set point) to give an error signal, which is passed to a controller. This adjusts the actuator drive signal to move the valve in the direction to give the required flow (i.e. PV = SP, giving zero error). The arrangement of Figure 7.1 is called *closed loop control* because a loop is formed by the controller, actuator and measuring device.

In many plants, closed loop control is achieved by electronics, or even computer, techniques with the various signals represented by electric currents. A common standard uses a current within the range 4–20 mA. If this represents a water flow from 0 to 1500 L min^{-1}, for example, a flow of 1000 L min^{-1} is represented by a current of 14.67 mA.

Electrical representation, and electronic devices, are not the only possibility, however. Process control history goes back before the advent of electronics (some early examples being speed governors on steam engines and an early servosystem for ships' rudders designed by Isambard Kingdom Brunel). Much of the original process control work was based around pneumatic devices, with the various signals represented by pneumatic pressures.

Perhaps surprisingly, pneumatic process control has by no means been superseded by electronic and microprocessor technology, so it is worth looking at the reasons for its popularity. First and foremost is safety. Much process control is done in chemical or petrochemical plants where explosive atmospheres are common. If electrical signals are used, great care must be taken to ensure no possible fault can cause a spark, which could ignite an explosive atmosphere. While this can be achieved, the result is complex and maintenance may be difficult (test instruments must also be classified safe for use in an explosive atmosphere).

A pneumatic system contains only air, so it presents no hazard under these conditions. No particular care needs to be taken with installation, and maintenance work can be carried out 'live' with simple non-electrical test instruments.

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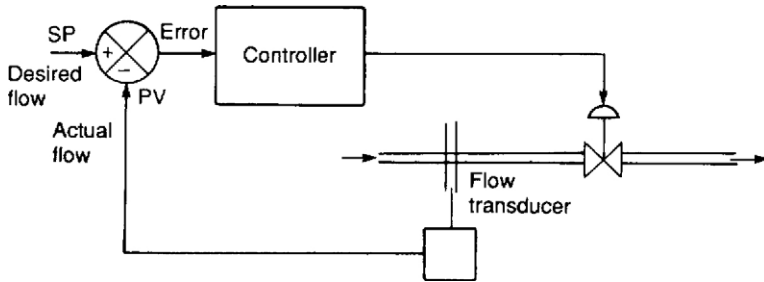
FIGURE 7.1 Closed loop control

A great deal of design and application experience has evolved over the years, and this base of knowledge is another major reason for the continuing popularity of pneumatic control. Companies with a significant investment in pneumatic control and a high level of staff competency are unlikely to change.

Many devices in the loop are, in any case, best provided by pneumatic techniques. Although electrical actuators are available, most valves are driven by pneumatic signals – even when transducer and controller are electronic.

SIGNALS AND STANDARDS

Signals in process control are generally represented by a pressure which varies over the range 0.2–1.0 bar or the almost identical imperial equivalent 3–15 psig. If the water flow of 0–1500 L min^{-1} is represented pneumatically, 0 L min^{-1} is shown by a pressure of 0.2 bar, 1500 L min^{-1} is 1.0 bar, while 1000 L min^{-1} is 0.733 bar.

The lower-range pressure of 0.2 bar (3 psig in the imperial range) is known as an offset zero and serves two purposes. First is to warn about damage to signal lines linking the transmitter and the controller or indicator (the 4 mA offset zero of electrical systems also gives this protection). In Figure 7.2a a pneumatic flow transmitter is connected to a flow indicator. A pneumatic supply (typically, 2–4 bar) is connected to the transmitter to allow the line pressure to be raised. The transmitter can also vent the line to reduce pressure (corresponding to reducing flow). If the line is damaged it is probably open to atmosphere giving a pressure of 0 bar, regardless of the transmitter's actions. As the indicator is scaled for 0.2–1 bar, a line fault therefore causes the indicator to go offscale, negatively. Loss of the pressure supply line causes a similar fault indication.

The offset zero also increases the speed of response. In Figure 7.2b a sudden increase in flow is applied to the transmitter at time A. The flow transmitter connects the supply to the line, causing an exponential increase in pressure (with a time constant determined by the line volume). The pressure rises towards the supply pressure, but at time B the correct pressure of 0.8 bar is reached, and the transmitter disconnects the supply.

The pressure stays at 0.8 bar until time C, when the flow rapidly falls to zero. The transmitter vents the line and the pressure falls exponentially towards 0 bar (with time constant again determined by line volume). At time D, a
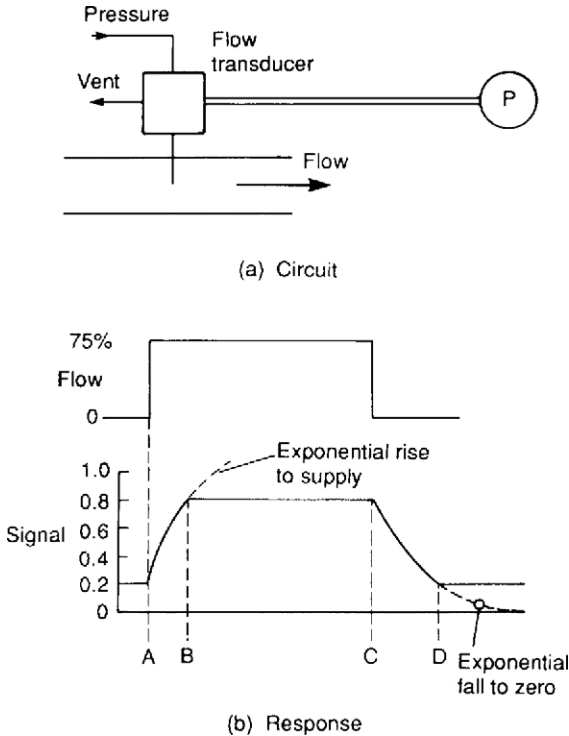
FIGURE 7.2 Advantage of an offset zero

pressure of 0.2 bar is reached (corresponding to zero flow) and the transmitter stops venting the line. For increasing indication, the offset zero has little effect, but for decreasing indication, the transmitter would need to completely vent the line without an offset zero to give zero indication. With a first-order lag response, this will theoretically take an infinite time, but even with some practical acceptance of error, time CD will be significantly extended.

Speed of response is, in any case, the Achilles heel of pneumatic signals. With an infinitely small time constant (given by zero volume lines), the best possible response can only be the speed of sound (330 m s^{-1}). If signal lines are over a hundred meters or so in length, this transit delay is significant. To this is added the first-order lag caused by the finite volume of the line, and the finite rate at which air flows into or out of the line under transmitter control. For a fast response, line volume must be small (difficult to achieve with long lines) and the transmitter able to deliver or vent large flow rates. In practice, time constants of several seconds are quite common.

THE FLAPPER-NOZZLE

Most properties (e.g. flow, pressure, level, error, desired valve position) can be converted to a small movement. The heart of all pneumatic process control devices is a device to convert a small displacement into a pressure change, which
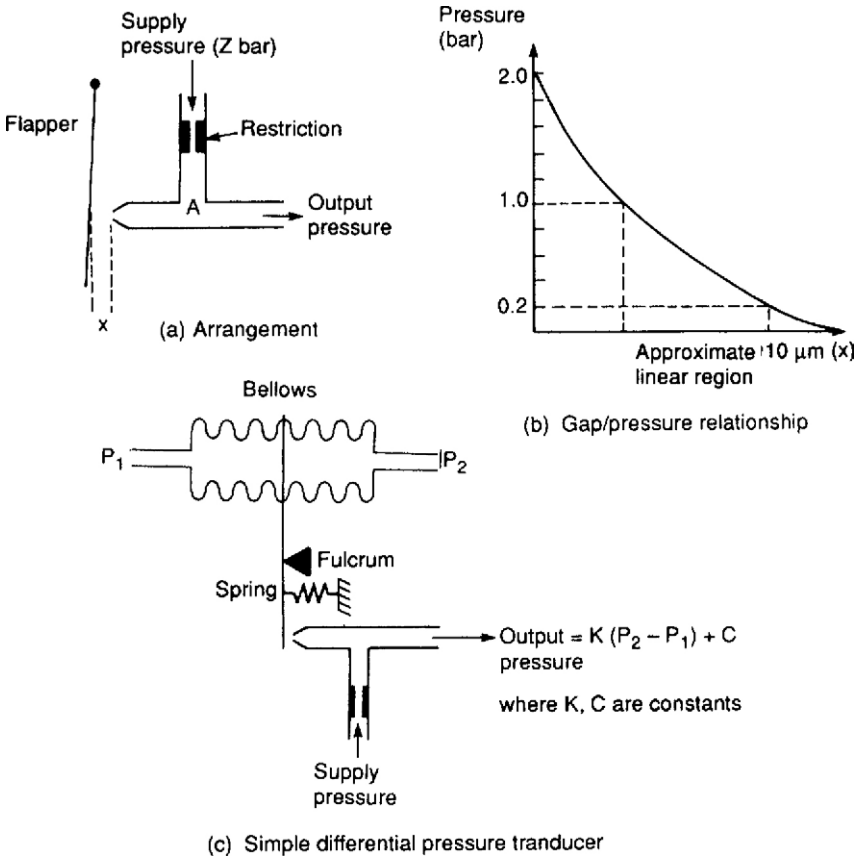
FIGURE 7.3   The flapper-nozzle, the basis of pneumatic process control

represents the property causing the displacement. This is invariably based on the flapper-nozzle, whose arrangement, characteristic and application are illustrated in Figure 7.3.

An air supply (typically, 2–4 bar) is applied to a very fine nozzle via a restriction, as shown in Figure 7.3a. The signal output side of the nozzle feeds to a closed (non-venting) load, such as an indicator. Air escapes as a fine jet from the nozzle, so the pressure at A is lower than the supply pressure because of the pressure drop across the restriction.

Air loss from the jet (and hence pressure at A) is influenced by the gap between the nozzle and movable flapper; the smaller the gap, the lower the air flow and higher the pressure. A typical response is shown in Figure 7.3b, illustrating the very small range of displacement and the overall non-linear response. The response can, however, be considered linear over a limited range (as shown) and the flapper-nozzle is generally linearized by use of a force balance system as described later.

Figure 7.3c shows a very simple differential pressure transducer which may be used as a flow transmitter by measuring the pressure drops across an orifice plate. The difference in pressure between P_1 and P_2 causes a force on the flapper.

Assuming $P_1 > P_2$ (which is true for the direction of flow shown), the top of the flapper is pushed to the right until the force from $(P_1 - P_2)$ is matched by the force from the spring extension. Flapper-nozzle gap, and hence the output pressure, is thus determined by the differential pressure and the flow through the orifice plate.

The arrangement of Figure 7.3c is non-linear, and incapable of maintaining output pressure to a load with even a small loss of air. Even with a totally sealed load the minimal air flow through the restriction leads to a first-order lag response with a very long time constant. A flapper-nozzle is therefore usually combined with an air amplifier, or volume booster, which takes a pressure as the input and gives a linearly related pressure output – with an ability to supply a large volume of air. When combined with the force balance principle described later, the inherent non-linearity of the flapper-nozzle can be overcome.

VOLUME BOOSTERS

An air amplifier is illustrated in Figure 7.4. It is provided with an air supply (typically 2–4 bar) and an input signal pressure. The amplifier admits air to, or vents air from, the output to maintain a constant output/input ratio. An amplifier with a gain of two, for example, turns a 0.2–1 bar signal range to a 0.4–2 bar range. Output pressure, controlled by the amplifier, has the ability to provide a large air volume and can drive large-capacity loads.

A unity gain air amplifier is shown in Figure 7.4b. It consists of two equal-area linked diaphragms, which together operate a needle and ball valve arrangement. The low-volume input signal is applied to the upper diaphragm and the
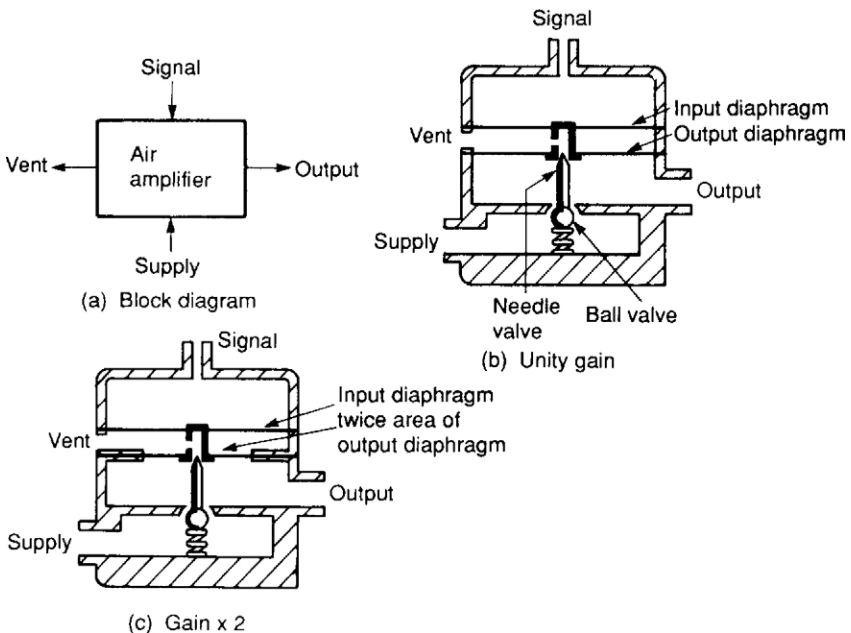
FIGURE 7.4 Volume boosters or air amplifiers

output pressure to the lower diaphragm. If output pressure is lower than inlet pressure, the diaphragm is pushed down, closing the needle valve and opening the ball valve to pass supply air to the load and increase output pressure.

If the output pressure is high, the diaphragm is forced up, closing the spring-loaded ball valve and opening the needle valve to allow air to escape through the vent and reduce output pressure. The amplifier stabilizes with output and input pressures equal.

The input port has a small and practically constant volume, which can be controlled directly by a flapper-nozzle. The output pressure tracks changes in inlet pressure, but with the ability to supply a large volume of air.

An air amplifier balances when forces on the two diaphragms are equal and opposite. Equal-area diaphragms have been used in the unity gain amplifier of Figure 7.4b. The area of the input diaphragm in the amplifier of Figure 7.4c is *twice* the area of the output diaphragm. For balance, the output pressure must be twice the input pressure, giving a gain of two. In general, the amplifier gain is given by:

$$\text{Gain} = \frac{\text{input area}}{\text{output area}}$$

THE AIR RELAY AND THE FORCE BALANCE PRINCIPLE

Air amplifiers balance input pressure and output pressure. An air relay, on the other hand (illustrated in Figure 7.5), balances input pressure with the force from a range spring. An increasing input signal causes air to pass from the supply to the load, while a decreasing input signal causes air to vent from the load. In the center of the input signal range, there is no net flow to or from the output port.
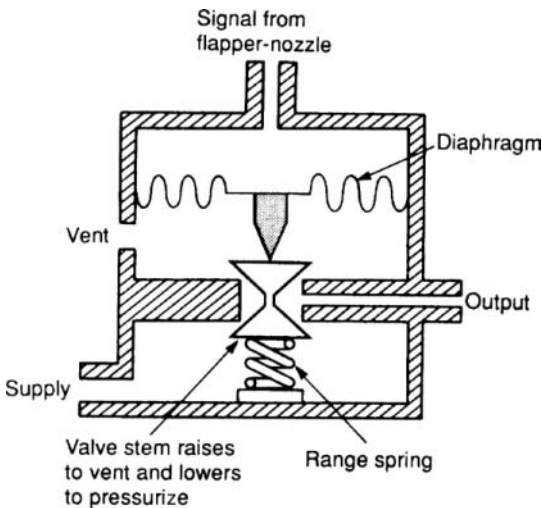
FIGURE 7.5 The air relay
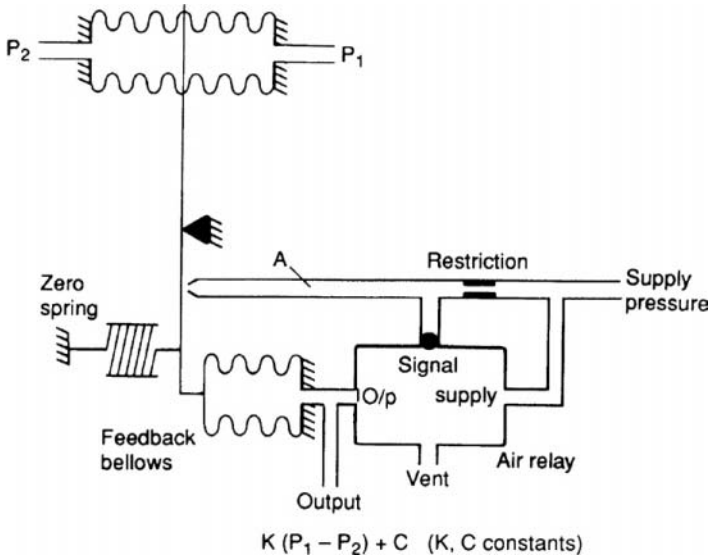
FIGURE 7.6 The force balance principle

An air relay is used to linearize a flapper-nozzle, as shown in Figure 7.6. Here, force from the imbalance in input pressures P_1 and P_2 is matched exactly by the force from the feedback bellows whose pressure is regulated by the air relay.

Suppose flow in the pipe increases, causing pressure difference $P_1 - P_2$ to increase. Increased force from the bellows at the top decreases the flapper gap causing pressure at the air relay input to rise. This causes air to pass to the feedback bellows, which apply a force opposite to that from the signal bellows.

The system balances when the input pressure from the flapper-nozzle to the air relay (point A) is at the center of its range, at which point the air relay neither passes air nor vents the feedback bellows. This corresponds to a fixed flapper-nozzle gap.

Figure 7.6 thus illustrates an example of a feedback system where the pressure in the feedback bellows is adjusted by the air relay to maintain a constant flapper-nozzle gap. The force from the feedback bellows thus matches the force from the input signal bellows, and output pressure is directly proportional to $(P_1 - P_2)$. The output pressure, driven directly from the air relay, can deliver a large air volume.

The arrangement in Figure 7.6 effectively operates with a fixed flapper-nozzle gap. This overcomes the inherent non-linearity of the flapper-nozzle. It is known as the force balance principle and is the basis of most pneumatic process control devices.

PNEUMATIC CONTROLLERS

Closed loop control, discussed briefly earlier, requires a controller which takes a desired (set point) signal and an actual (process variable) signal, computes the error then adjusts the output to an actuator to make the actual value equal the desired value.
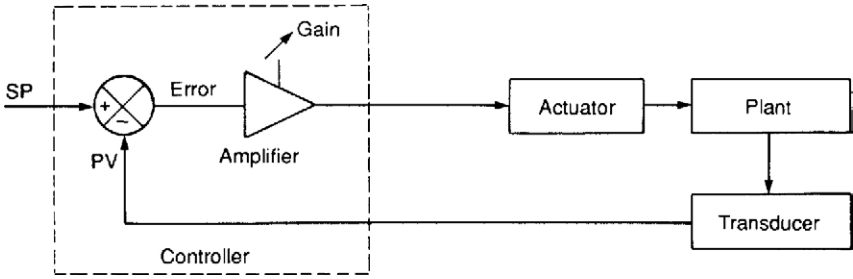
FIGURE 7.7 Proportional only controller

The simplest pneumatic controller is called a proportional only controller, shown schematically in Figure 7.7. The output signal here is simply the error signal multiplied by a gain:

$$\begin{aligned} OP &= K \times error \\ &= K \times (SP - PV) \end{aligned} \tag{7.1}$$

where K is the gain.

Comparison of the controller in Figure 7.7 with the force balance transmitter in Figure 7.6 shows that the differential pressure measurement $(P_1 - P_2)$ performs the same function as error subtraction $(SP - PV)$. We can thus construct a simple proportional only controller with the pneumatic circuit of Figure 7.6. Gain can be set by moving the pivot position.

The output of a proportional controller is simply K×error, so to get any output signal, an error signal must exist. This error, called the offset, is usually small, and can be decreased by using a large gain. In many applications, however, too large a gain causes the system to become unstable.

In these circumstances a modification to the basic controller is used. A time integral of the error is added to give:

$$OP = K \left(error + \frac{1}{T_i} \int error\ dt \right) \tag{7.2}$$

Controllers following expression 7.2 are called proportional plus integral (P+I) controllers, illustrated in Figure 7.8. The constant T_i, called the integral time, is set by the user. Often the setting is given in terms of $1/T_i$ (when the description repeats/min is used). A controller following expression 7.2 has a block diagram shown in Figure 7.8a, and responds to a step response as shown in Figure 7.8b. As long as an error exists, the controller output creeps up or down to a rate determined by T_i. Only when there is no error is the controller output constant. Inclusion of the integral term in expression 7.2 removes the offset error.

A pneumatic P+I controller can be constructed as shown in Figure 7.8c. Integral bellows oppose the action of the feedback bellows, with the rate of change of pressure limited by the T_i setting valve. The controller balances the correct

(a) Block diagram

(b) Step response

(c) Construction
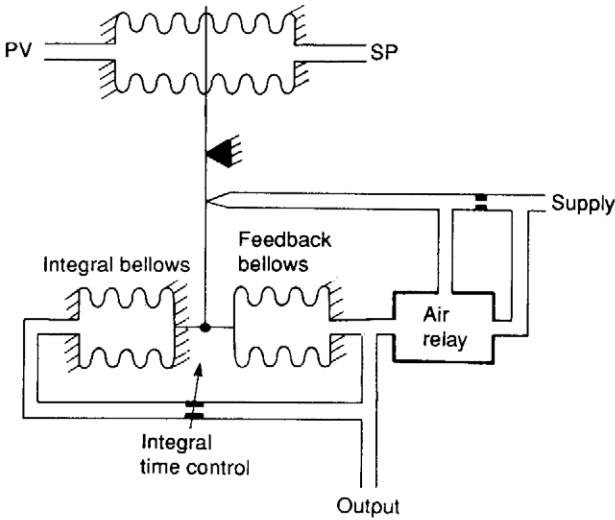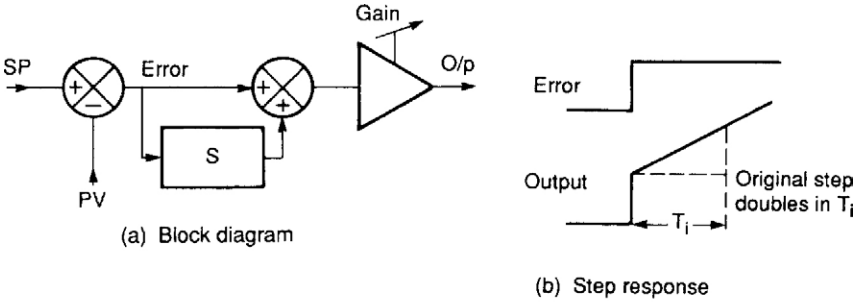
FIGURE 7.8 Proportional plus integral (P + I) controller

flapper-nozzle gap to give zero error, with PV = SP and equal forces from the integral and feedback bellows.

A further controller variation, called the three-term or P+I+D controller uses the equation:

$$OP = K \left(error + \frac{1}{T_i} \int error \, dt + \frac{1}{T_d} \frac{d \, error}{dt} \right) \qquad (7.3)$$

where T_d is a user-adjustable control, called the derivative time. Addition of a derivative term makes the control output change quickly when SP or PV are changing quickly, and can also serve to make a system more stable.

Pneumatic three-term control can be achieved with the arrangement of Figure 7.9, where the action of the feedback bellows has been delayed. The three user-adjustable terms in expression 7.3 (gain K, integral time T_i, derivative time T_d) are set by beam pivot point and two bleed valves to give the best plant response. These controls do, however, interact to some extent – a failing not shared by electronic controllers.
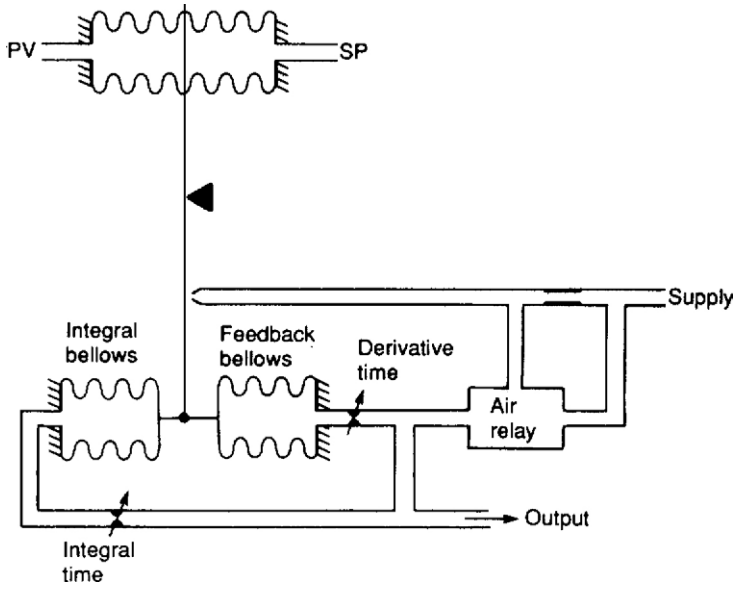
FIGURE 7.9 Three-term (P + I + D) controller

Figure 7.10 represents the typical front panel of a controller. Values of SP, PV and controller output are displayed and the operator can select between automatic and manual operation. The desired value (SP) can be adjusted in auto or the controller output set directly in manual. The operator does not have access to K, T_i, T_d setting controls; these are adjusted by the maintenance technician.
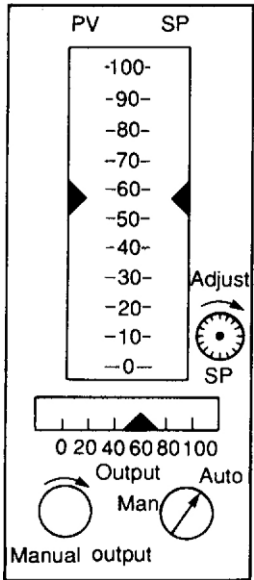
FIGURE 7.10 Front panel of a typical controller
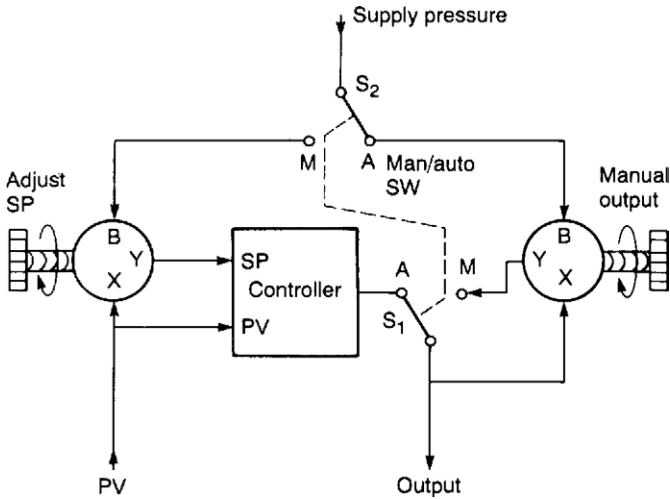
FIGURE 7.11 Internal arrangement giving bumpless transfer

Internally the controller is arranged as shown in Figure 7.11. Setpoint and manual output controls are pressure regulators, and the auto/manual switch simply selects between the controller and manual output pressures. If the selection, however, just switched between P_c and P_m there would be a step in the controller output. The pressure regulators are designed so their output Y tracks input X, rather than the manual setting when a pressure signal is applied to B. The linked switch S_2 thus makes the set point track the process variable in manual mode, while manual output P_m tracks the controller output in automatic mode. 'Bumpless' transfers between automatic and manual can therefore be achieved.

PROCESS CONTROL VALVES AND ACTUATORS

In most pneumatic process control schemes, the final actuator controls the flow of a fluid. Typical examples are liquid flow for chemical composition control, level control, fuel flow for temperature control and pressure control. In most cases the actual control device will be a pneumatically actuated flow control valve.

Even with totally electronic or computer-based process control schemes, most valves are pneumatically operated. Although electrically operated actuators *are* available, pneumatic devices tend to be cheaper, easier to maintain and have an inherent, and predictable, failure mode.

It is first useful to discuss the way in which fluid flow can be controlled. It is, perhaps, worth noting that these devices give full proportional control of fluid flow, and are *not* used to give a simple flow/no-flow control.

Flow control valves

All valves work by putting a variable restriction in the flow path. There are three basic types of flow control valves, shown in Figures 7.12–7.14. Of these
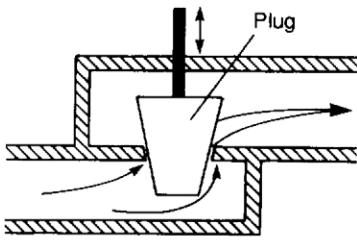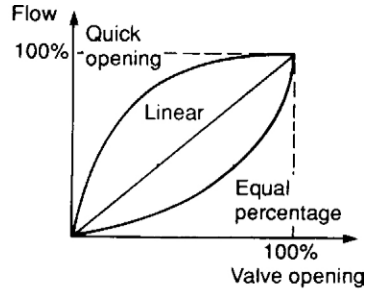
(a) Construction   (b) Valve characteristics
(set by plug profile)

FIGURE 7.12 The plug valve

the plug or globe valve (Figure 7.12) is probably most common. This controls flow by varying the vertical plug position, which alters the size of the orifice between the tapered plug and valve seat. Normally the plug is guided and constrained from sideways movement by a cage, not shown in Figure 7.12a for simplicity.

The valve characteristics define how the valve opening controls flow. The characteristics of the globe valve can be accurately predetermined by machining the taper of the plug. There are three common characteristics, shown in Figure 7.12b. These are specified for a constant pressure drop across the valve, a condition which rarely occurs in practical plants. In a given installation, the flow through a valve for a given opening depends not only on the valve, but also on pressure drops from all the other items and the piping in the rest of the system. The valve characteristic (quick opening, linear, or equal percentage) is therefore chosen to give an approximately linear flow/valve position relationship for this particular configuration.

A butterfly valve, shown in Figure 7.13, consists of a large disc which is rotated inside the pipe, the angle determining the restriction. Butterfly valves can be made to any size and are widely used for control of gas flow. They do, however, suffer from rather high leakage in the shut-off position and suffer badly from dynamic torque effects, a topic discussed later.

The ball valve, shown in Figure 7.14, uses a ball with a through hole which is rotated inside a machined seat. Ball valves have an excellent shut-off characteristic with leakage almost as good as an on/off isolation valve.
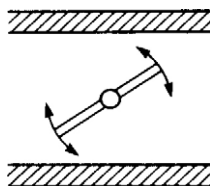
FIGURE 7.13 The butterfly valve
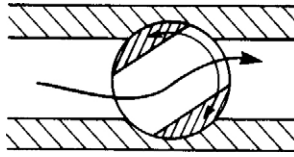
FIGURE 7.14 The ball valve

When fluid flows through a valve, dynamic forces act on the actuator shaft. In Figure 7.15a, the flow assists opening (and opposes the closing) of the valve. In Figure 7.15b, the flow assists the closing (and opposes the opening) of the valve. The latter case is particularly difficult to control at low flows as the plug tends to slam into the seat. This effect is easily observed by using the plug and chain to control flow of water out of a household bath.

The balanced valve of Figure 7.15c uses two plugs and two seats with opposite flows and gives little dynamic reaction onto the actuator shaft. This is achieved at the expense of higher leakage, as manufacturing tolerances cause one plug to seat before the other.

Butterfly valves suffer particularly from dynamic forces, a typical example being shown in Figure 7.16. As can be seen, maximum force occurs just before the fully open position, and this force acts to open the valve. It is not unknown for an actuator to be unable to move a butterfly valve off the fully open position and it is consequently good practice to mechanically limit opening to about 60°.
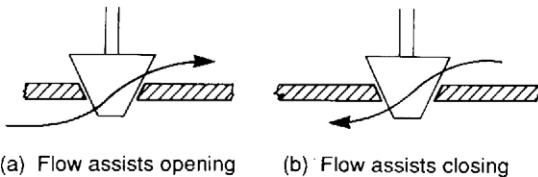
(a) Flow assists opening   (b) Flow assists closing
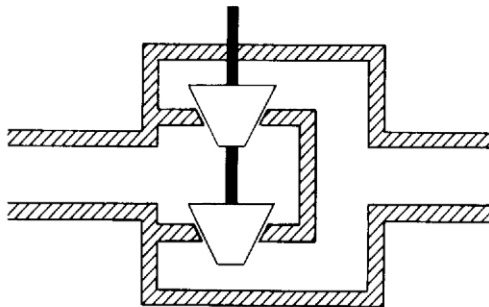
(c) Balanced valve

FIGURE 7.15 Dynamic forces acting on a valve
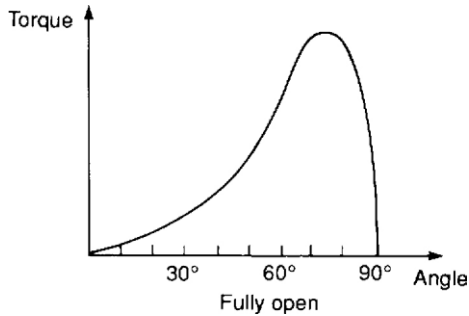
FIGURE 7.16 Torque on a butterfly valve

Actuators

The globe valve of Figure 7.12 needs a linear motion of the valve stem to control flow, whereas the butterfly valve of Figure 7.13 and the ball valve of Figure 7.14 require a rotary motion. In practice all, however, use a linear displacement actuator – with a mechanism similar to that in Figure 7.17 used to convert a linear stroke to an angular rotation if required.

Pneumatic valve actuators are superficially similar to the linear actuators of Chapter 5, but there are important differences. Linear actuators operate at a constant pressure, produce a *force* proportional to applied pressure and are generally fully extended or fully retracted. Valve actuators operate with an applied pressure which can vary from, say, 0.2 to 1 bar, producing a *displacement* of the shaft in direct proportion to the applied pressure.

A typical actuator is shown in Figure 7.18. The control signal is applied to the top of a piston sealed by a flexible diaphragm. The downward force from this pressure (P×A) is opposed by the spring compression force and the piston settles where the two forces are equal, with a displacement proportional to applied pressure. Actuator gain (displacement/pressure) is determined by the stiffness of the spring, and the pressure at which the actuator starts to move (0.2 bar say) is set by a pre-tension adjustment.

Figure 7.18b illustrates the action of the rubber diaphragm. This 'peels' up and down the cylinder wall so the piston area remains constant over the full range of travel.

The shaft of the actuator extends for increasing pressure, and fails in a fully up position in the event of the usual failures of loss of air supply, loss of signal or rupture of the diaphragm seal. For this reason such an actuator is known as a fail-up type.
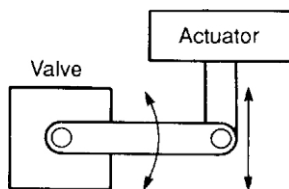
FIGURE 7.17 Conversion from linear actuator motion to rotary valve motion

(a) Construction

(b) Seal action
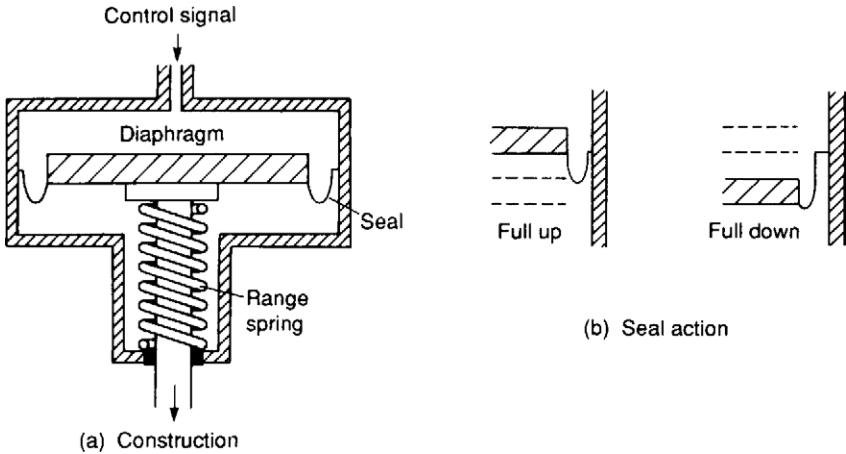
FIGURE 7.18 Fail-up actuator

In the actuator of Figure 7.19, on the other hand, signal pressure is applied to the bottom of the piston and the spring action is reversed. With this design the shaft moves up for increasing pressure and moves down for common failure modes. This is known as a fail-down or reverse-acting actuator.

One disadvantage of this design is the need for a seal on the valve shaft.

Where safety is important, valve and actuator should be chosen to give the correct failure mode. A fuel valve, for example, should fail closed, while a cooling water valve should fail open.

Valve actuators tend to have large surface areas to give the required force, which means a significant volume of air is above the piston. Valve movement leads to changes in this volume, requiring air to be supplied from, or vented by, the device providing the pressure signal. A mismatch between the air requirements of the actuator and the capabilities of the device supplying pressure signal results in a slow, first-order lag response.
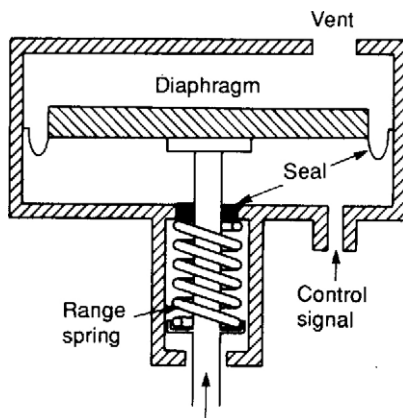
FIGURE 7.19 Fail-down actuator
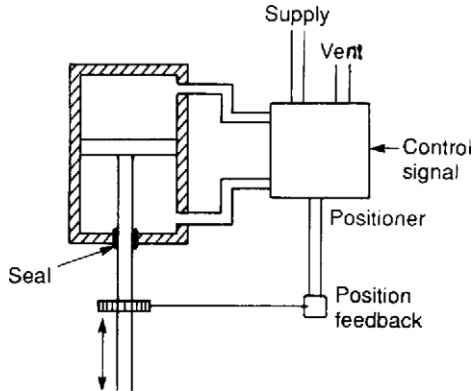
FIGURE 7.20 Double-acting cylinder (holds position on failure)

The net force acting on the piston in Figures 7.18 and 7.19 is the sum of force from the applied pressure, the opposing spring force *and* any dynamic forces induced into the valve stem from the fluid being controlled. These dynamic forces therefore produce an offset error in valve position. The effect can be reduced by increasing the piston area or the operating pressure range, but there are limits on actuator size and the strength of the diaphragm seal. In Figure 7.20 a double-acting piston actuator operating at high pressure is shown. There is no restoring spring, so the shaft is moved by application of air to, or venting of air from, the two sides of the piston. A closed loop position control scheme is used, in which shaft displacement is compared with desired displacement (i.e. signal pressure) and the piston pressures adjusted accordingly. The arrangement of Figure 7.20 is called a valve positioner, and correctly positions the shaft despite dynamic forces from the valve itself.

Valve positioners

A valve positioner is used to improve the performance of a pneumatically operated actuator, by adding a position control loop around the actuator as shown in Figure 7.21. They are mainly used:
- to improve the operating speed of a valve;
- to provide volume boosting where the device providing the control signal can only provide a limited volume of air. As noted previously a mismatch
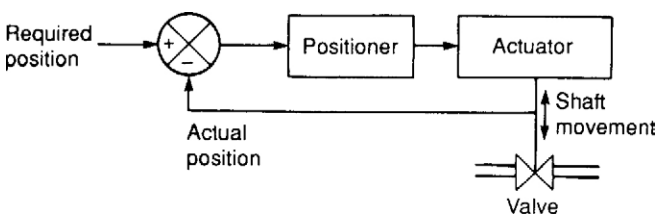
FIGURE 7.21 The valve positioner
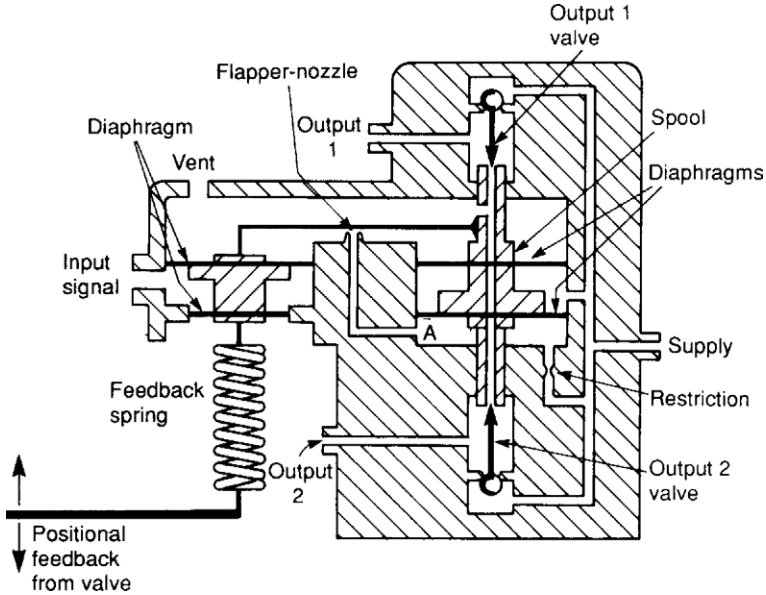
FIGURE 7.22 Force balance valve positioner

between the capabilities of driver and the requirements of an actuator results in a first-order lag response with a long time constant;
- to remove offsets resulting from dynamic forces in the valve (described in the previous section);
- where a pressure boost is needed to give the necessary actuator force;
- where a double-acting actuator is needed (which cannot be controlled with a single pressure line).

There are two basic types of valve positioner. Figure 7.22 shows the construction of a valve positioner using a variation of the force balance principle described earlier. The actuator position is converted to a force by the range spring. This is compared with the force from the signal pressure acting on the input diaphragm. Any mismatch between the two forces results in movement of the beam and a change in the flapper-nozzle gap.

If the actuator position is low, the flapper-nozzle gap decreases, causing a rise in pressure at point A. This causes the spool to rise, connecting supply air to output 1, and venting output 2, resulting in the lifting of the actuator. If actuator position is high, the flapper-nozzle gap increases and pressure at A falls, causing the spool to move down applying air to output 2 and venting output 1, which results in the actuator lowering. The actuator thus balances when the range spring force (corresponding to actuator position) matches the force from the input signal pressure (corresponding to the required position), giving a constant flapper-nozzle gap.

The zero of the positioner is set by the linkage of the positioner to the valve shaft and the range by the spring stiffness. Fine zero adjustment can be made by a screw at the end of the spring.
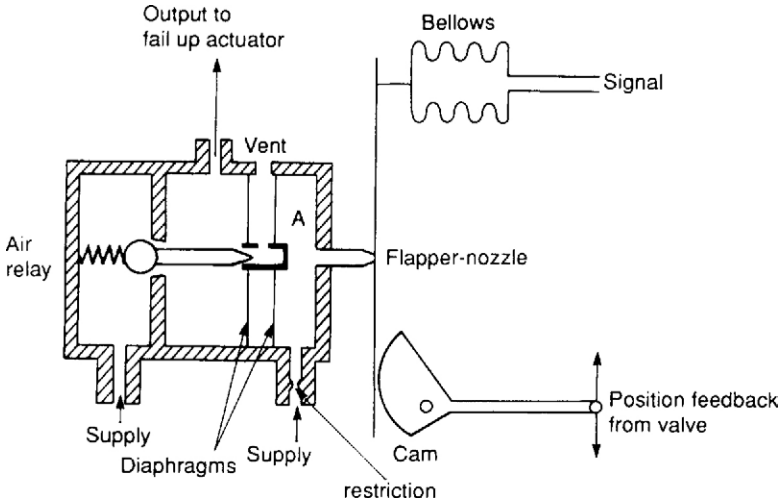
FIGURE 7.23 Motion balance positioner

The second type of positioner, illustrated in Figure 7.23, uses a motion balance principle. The valve shaft position is converted to a small displacement and applied to one end of the beam controlling the flapper-nozzle gap. The input signal is converted to a displacement at the other end of the beam. The pressure at A resulting from the flapper-nozzle gap is volume boosted by an air relay which passes air to, or vents air from, the actuator, to move the shaft until the flapper-nozzle gap is correct. At this point, the actuator position matches the desired position.

Positioners are generally supplied equipped with gauges to indicate supply pressure, signal pressure and output pressures, as illustrated in Figure 7.24 for a double-acting actuator.

Often, bypass valves are fitted to allow the positioner to be bypassed temporarily in the event of failure with the signal pressure sent directly to the actuator.
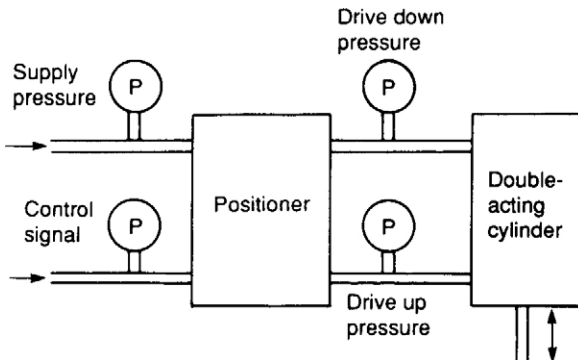
FIGURE 7.24 Pressure indication on a positioner for fault finding

CONVERTERS

The most common process control arrangement is probably electronic control-lers with pneumatic actuators and transducers. Devices are therefore needed to convert between electrical analog signals and the various pneumatic standards. Electrical to pneumatic conversion is performed by an I–P converter, while pneumatic to electrical conversion is performed by a device called, not surpris-ingly, a P–I converter.

I–P converters

Figure 7.25 illustrates a common form of I–P converter based on the famil-iar force balance principle and the flapper-nozzle. Electrical current is passed through the coil and results in a rotational displacement of the beam. The result-ing pressure change at the flapper-nozzle gap is volume-boosted by the air relay and applied as a balancing force by bellows at the other end of the beam. A bal-ance results when the force from the bellows (proportional to output pressure) equals the force from the coil (proportional to input electrical signal).

P–I converters

The operation of a P–I converter, illustrated in Figure 7.26, again uses the force balance principle. The input pressure signal is applied to bellows and produces a deflection of the beam. This deflection is measured by a position transducer such as an LVDT (linear variable differential transformer). The electrical signal corresponding to the deflection is amplified and applied as current through a coil to produce a torque which brings the beam back to the null position. At balance, the coil force (proportional to output current) matches the force from the bel-lows (proportional to input signal pressure).
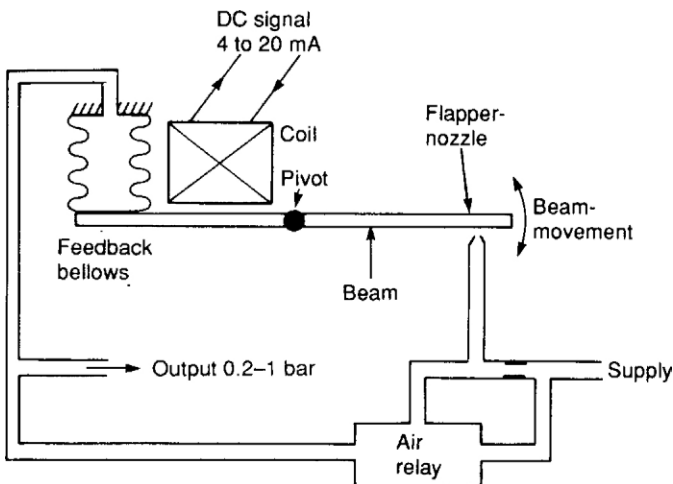
FIGURE 7.25 Current to pressure (I–P) converter
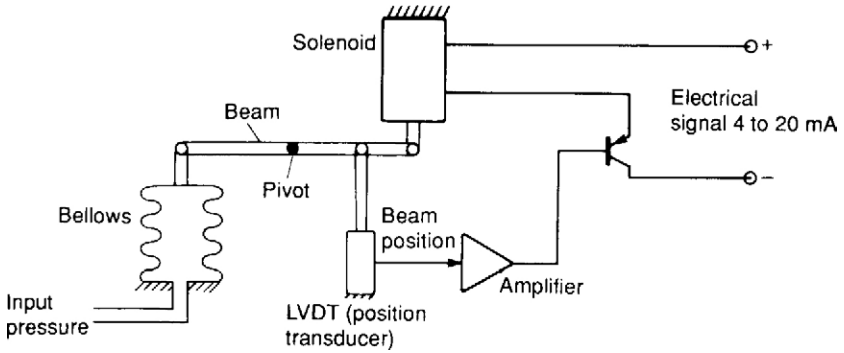
FIGURE 7.26 Pressure to current (P–I) converter

The zero offset (4 mA) in the electrical signal is sufficient to drive the amplifier in Figure 7.26, allowing the two signal wires to also act as the supply lines. This is known as two-wire operation. Most P–I converters operate over a wide voltage range (e.g. 15–30 V). Often, the current signal of 4–20 mA is converted to a voltage signal (commonly in the range 1–5 V) with a simple series resistor.

Chapter | eight

Sequencing Applications

Many industrial processes such as packaging lines consist of pneumatic actuators which follow a predetermined sequence of operations. These sequences can be considered as a series of steps, each step resulting in the operation of pneumatic solenoids. Progression between steps is controlled by sensors on the controlled plant, e.g. photo-cells, stroke limit switches, timers, etc. A typical simple sequence is shown in Figure 8.1, where a product is delivered on a conveyor, moved to the correct position by a horizontal cylinder then stamped by a vertical cylinder.

Associated with these actuators is a series of sensors: a photocell, or similar device, to say a product is present and positional limits to sense the end of stroke for each cylinder.

To aid the design and to assist fault finding it is usual to draw a sequence diagram showing the interaction between the sensors and actuators. Figure 8.1 could be represented by Figure 8.2 where A+ denotes cylinder A extending and A– denotes cylinder A retracting.

Sequence systems are built around two-position pneumatic shuttle valves. It is important to note that these have no spring return and hold position when the operating signals are removed. A typical example is shown in Figure 8.3. If pneumatic pressure is applied to port X the spool shifts to the right, extending the cylinder. This condition will be held even if the signal is removed from port X. If a signal is now applied to port Y the spool will shift to the left and the cylinder will retract. Again this condition will be held even if the signal is removed from port Y. The valve can be considered to be a memory device with its state being an indication of which signal was last applied. Obviously signals should not be applied to opposing sides of the valve at the same time.

PNEUMATIC LIMIT SWITCHES

Sequences are driven by limit switches which pass, or block or vent pneumatic pressure. On circuit diagrams limit switches are normally shown in the rest (normally unstruck) state. Figure 8.4a thus vents when unstruck and passes pressure when struck. Figure 8.4b passes pressure when unstruck and vents when struck.

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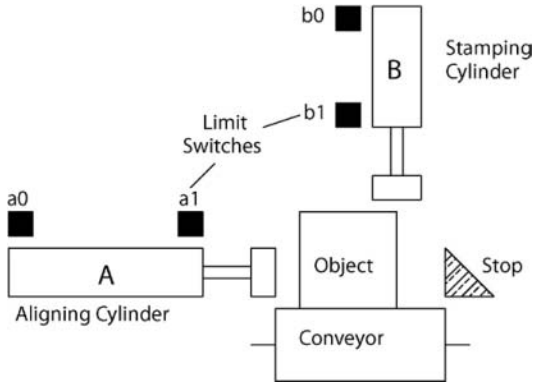
FIGURE 8.1 A simple sequencing system
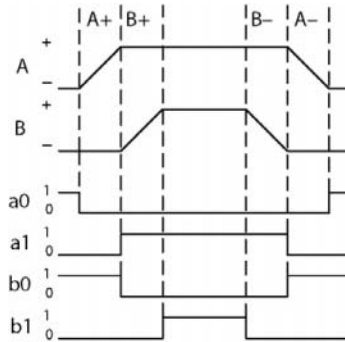
FIGURE 8.2 A simple sequence diagram
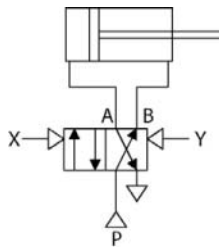
FIGURE 8.3 A shuttle valve with no spring return

If a limit is struck in the rest state (for example, the back-retracted state for a cylinder) it is represented by Figure 8.4c or d. Note the different spool positions between the two sets of symbols a,b and c,d.

The position of limits is normally shown on a circuit diagram by filled black squares. A simple example of a pneumatic sequencing system is illustrated in Figure 8.5, where the cylinder oscillates continuously between two striker-op-
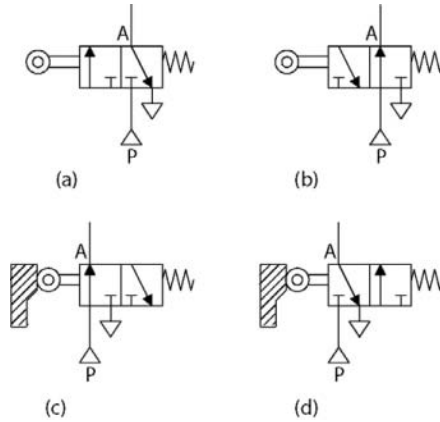
FIGURE 8.4 Representation of pneumatic limits: (a) pressurizes when struck; (b) vents when struck. (c), (d) show limits which are struck in the rest state

erated limit switches a0 and a1. The two limits shift the spool of the valve to extend and retract the cylinder.

In some circuits, as we shall see later, it is required to only get a signal from a limit switch when it is struck in a certain direction. These are known as one-way or unidirectional or pulsed limits. A typical construction is shown in Figure 8.6a. The limit has two linked arms A and B. Arm A actually controls the pneumatic signals. Arm B is spring loaded into the vertical position but can move over a 90° arc to the left. If the striker moves from right to left it will only fold down arm B and not operate the limit. If the striker moves from left to right arm B cannot fold down, pushing arm A down and operating the limit. The limit can thus be described as effective from left to right. One-way limits have the symbol in Figure 8.6b and are shown on circuit diagrams with an arrow showing the effective direction.
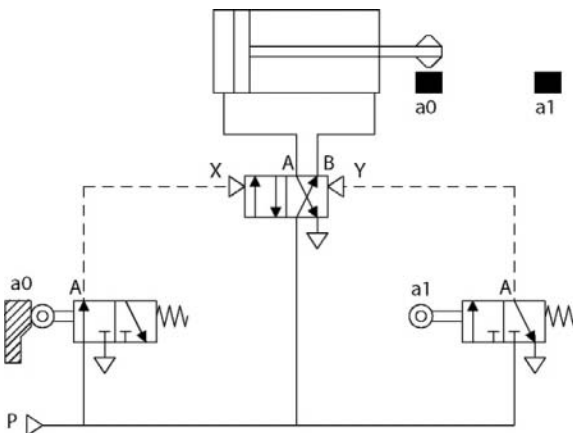
FIGURE 8.5 Sequencing system using limits
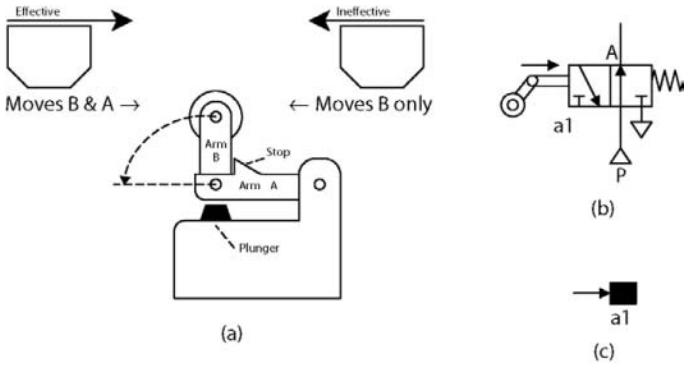
FIGURE 8.6 One-way (unidirectional) limit: (a) construction; (b) symbol
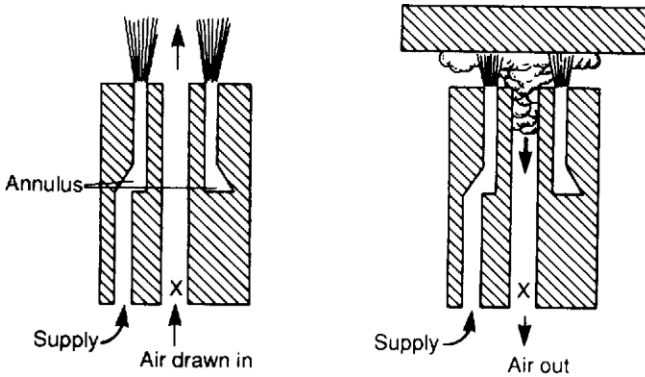
FIGURE 8.7 Reflex proximity switch

Pneumatic proximity sensors can also be used. The reflex sensor of Figure 8.7 uses an annular nozzle jet of air, the action of which removes air from the center bore to give a light vacuum at the signal output X. If an object is placed in front of the sensor, flow is restricted and a significant pressure rise is seen at X. Another example is the interruptible jet sensor shown in Figure 8.8, which is simple in operation but uses more air. A typical application could be sensing the presence of a drill bit to indicate 'drill complete' in a pneumatically controlled machine tool. With no object present, the jet produces a pressure rise at signal output X. An object blocking this flow causes X to fall to atmospheric pressure.

With both of these sensors, air consumption can be a problem. To reduce air usage, low pressure and low flow rates are used. Both of these result in a low-
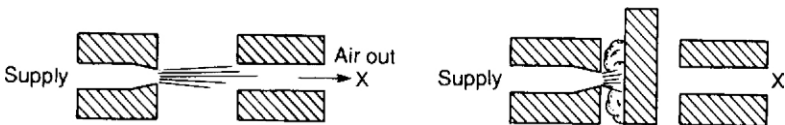
FIGURE 8.8 Interruptible jet limit switch
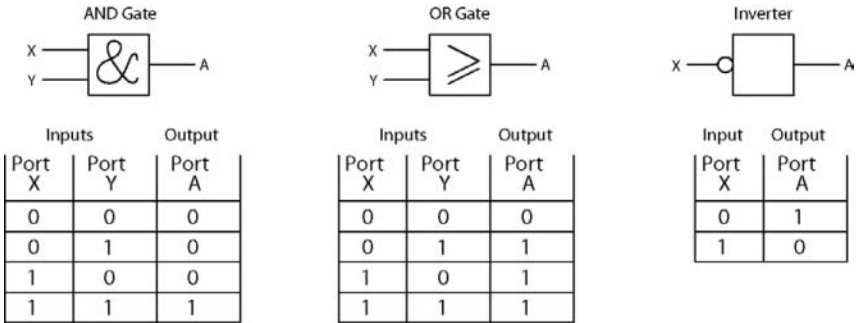
FIGURE 8.9 Logic element symbols

pressure output signal which requires pressure amplification or low-pressure pilot valves before it can be used to control full pressure lines.

LOGIC ELEMENTS

Many sequences are built around expressions such as:

'If (W is present AND X is present AND Y is NOT present)
OR Z is present then operate actuator Q'   (8.1)

These are known as logic statements and are built around the three logic elements AND, OR, NOT. The logic symbols and operational tables for these three elements are shown in Figure 8.9. An AND gate thus gives an output pressure if, and only if, pressure is applied to both of its two inputs. The logic statement 8.1 above could thus be represented by Figure 8.10 which is, in effect, a piping diagram.

An AND gate can be constructed as in Figure 8.11a or b. In Figure 8.11a there is a movable unsprung spool which will shift left or right and block off a pressure signal which is applied on its own. If two pressures are applied simultaneously the spool will shift to allow the lowest pressure through and block the highest pressure. Figure 8.11a is often used as the symbol for an AND gate rather than the logic symbol in Figure 8.9.

The alternative AND gate of Figure 8.11b is a simple spring return valve where an output pressure is given if the spool has been shifted to the right AND a pressure is present at the base.
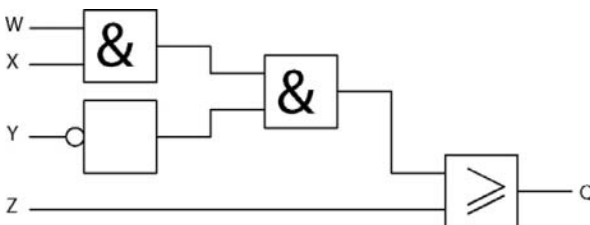
FIGURE 8.10 Representation of statement 8.1
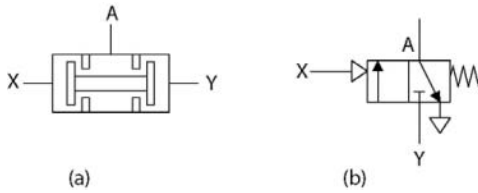
FIGURE 8.11 (a, b) Common forms of AND gate valves
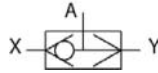
FIGURE 8.12 An OR gate valve
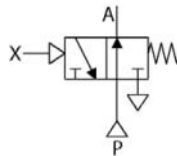
FIGURE 8.13 A NOT gate. Note this requires a separate pressure source

An OR gate can be constructed with a ball shuttle valve as in Figure 8.12. Again this symbol is often used on circuit diagrams instead of the earlier logic symbol of Figure 8.10. The element will pass the higher of the two input pressures.

A NOT gate (often called an inverter) is constructed with a spring return shuttle valve as in Figure 8.13. Note that, unlike the AND and OR gates, a NOT gate requires a separate pressure source.

Logic devices (AND, OR gates and memories) can also be constructed using the wall attachment or Coanda effect although these are relatively rare. The effect is based on the fact that a fluid stream exiting from a jet with a Reynolds number in excess of 1500 (giving very turbulent flow) tends to attach itself to a wall and remain there until disturbed as shown in Figure 8.14a.

This principle can be used to give a pneumatic set/reset (S–R) flip-flop memory in Figure 8.14b. If the set input is pulsed, the flow attaches itself to the right-hand wall, exiting via output Q. If the set input is then removed the Coanda effect keeps the flow on this route until the reset input is pulsed.

Figure 8.14c shows a fluidic OR/NOR gate. A small bias pressure keeps the signal on the right-hand wall, which causes it to exit via the right-hand port. If signal A or B is applied (at higher pressure than the bias) the flow switches over to the (A+B) output. When both A and B signals are removed, the bias pressure switches the flow back again.

TIMERS

Sequences often use time as part of the control. For example, a sequence could be: '*Extend cylinder B, wait five seconds after limit b1 has made then extend cylinder C.*'
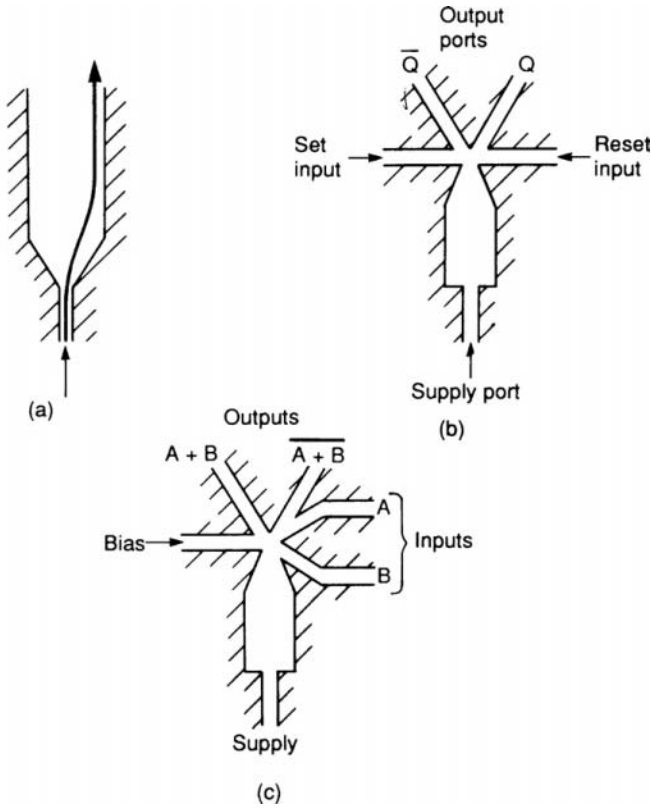
FIGURE 8.14 Fluidic logic: (a) the Coanda effect; (b) set/reset flip-flop; (c) OR/NOR gate

There are three types of timer, shown in Figure 8.15. Timer 8.15a is a DE-LAY-ON, commonly known as a TON. Timer 8.15b is a DELAY-OFF, commonly known as a TOF. Timer 18.15c is a ONE-SHOT or PULSE timer.

All use a similar principle with a small reservoir which is charged (or discharged) via an adjustable restriction which sets the time. The pressure in the
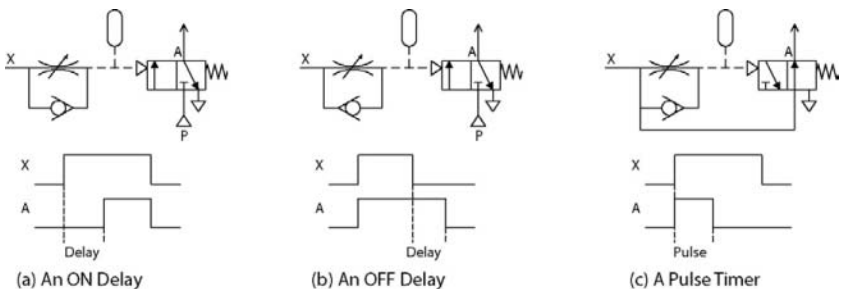
FIGURE 8.15 Type of timers: (a) DELAY-ON (TON); (b) DELAY-OFF (TOF); (c) ONE-SHOT (PULSE) timer

reservoir shifts the valve spool. Note the TOF timer requires a separate pressure supply.

MORE COMPLEX SEQUENCES

The earlier sequence of Figures 8.6 and 8.7 translated easily into a practical circuit. In many cases, though, designing a circuit is less straightforward.

Suppose we have two cylinders, A and B, and four positional limits, but now the sequence required is A+, B+, B–, A–. Figure 8.16 shows the start of the design with shuttle valve Va controlling cylinder A and shuttle valve Vb controlling cylinder B. Signal Xa extends the cylinder and will be controlled by limit a0 (saying the cylinder has retracted). Similarly signal Ya retracts the cylinder and will be controlled by limit b0 (saying cylinder B has retracted). However the limits a0 and b0 are piped and arranged examination will show that there is a point in the proposed cycle where one or both shuttle valves will have pressure applied to both sides of the valve at the same time. This problem is often known as a 'locked-in', maintained or trapped signal. There are several different ways in which this problem can be overcome.

The first solution is to add one-shot pulse timers to the signals from limits b0 and a1. These pulses only last sufficiently long to shift the valve spool and prevent valves Va and Vb receiving pressure on both sides. This solution is shown in Figure 8.17.

The second solution uses the one-way limit shown earlier in Figure 8.8. Here, one-way limits are used for b0 and a1 which are active in the direction shown. These limits are arranged so the striker travels through the limit and produces a pulse in one direction but has no effect in the other direction. Effectively one-way limits are producing a single pulse when struck and the result is identical to that achieved with one-shot pulse timers in Figure 8.17. The one-way limit solution is shown in Figure 8.18.

The third solution notes that the problem arises because cylinder B has to both extend and retract within the cycle of cylinder A. This observation
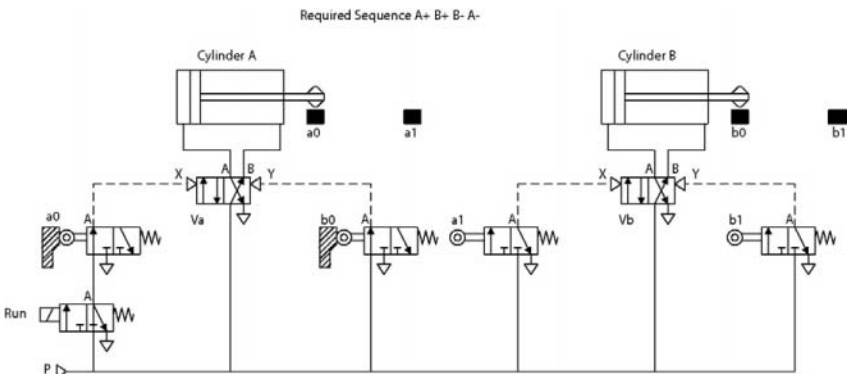
FIGURE 8.16 A more complex sequence which cannot be achieved with a simple approach

Required Sequence A+ B+ B- A-
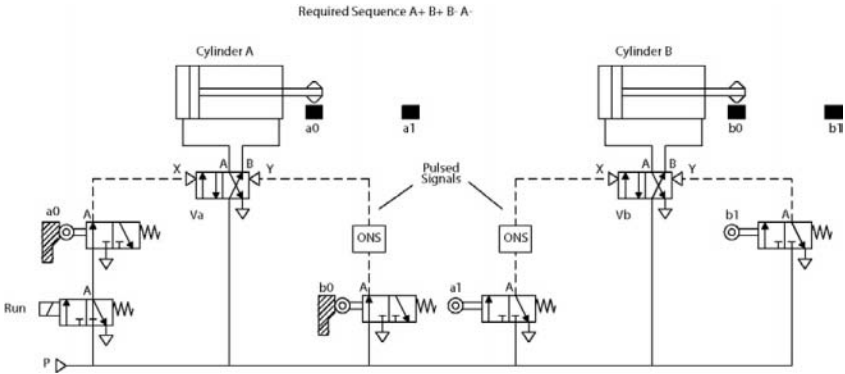
FIGURE 8.17 A solution using pulsed timers

Sequence A+ B+ B- A-
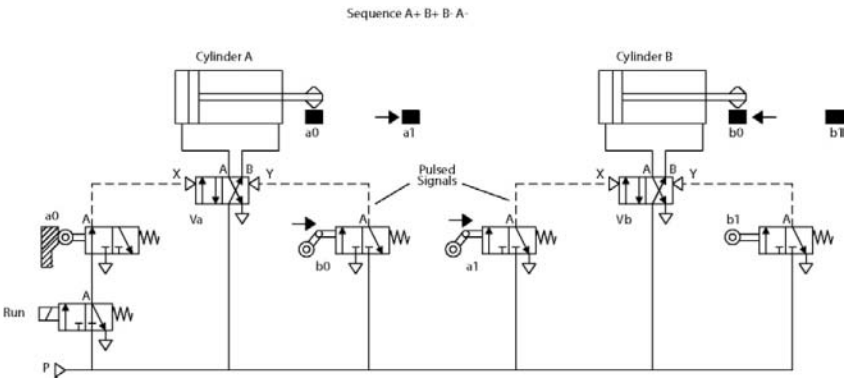
FIGURE 8.18 A solution using one-way limits

applies to any pneumatic sequence, if an actuator has to complete two move-
ments within a single movement of another actuator a lock-in will probably
result. To overcome this problem the cycle is split into two groups, as shown
in Figure 8.19:

Group 1   A+, B+
Group 2   B−, A−

A group selector valve is used which delivers a pressure supply as below:

Group 1   To the start solenoid valve and limit a1
Group 2   To limit b0 and port Y of valve Vb

The sequence can be described in words as follows:

1. Initially both cylinders are retracted and a0 has moved valve Vg to apply
   pressure to group 1.
2. The start solenoid is energized and Va extends cylinder A. Limit a0 is un-
   struck then limit a1 is struck.
3. Limit a1 operates Vb and extends cylinder B.
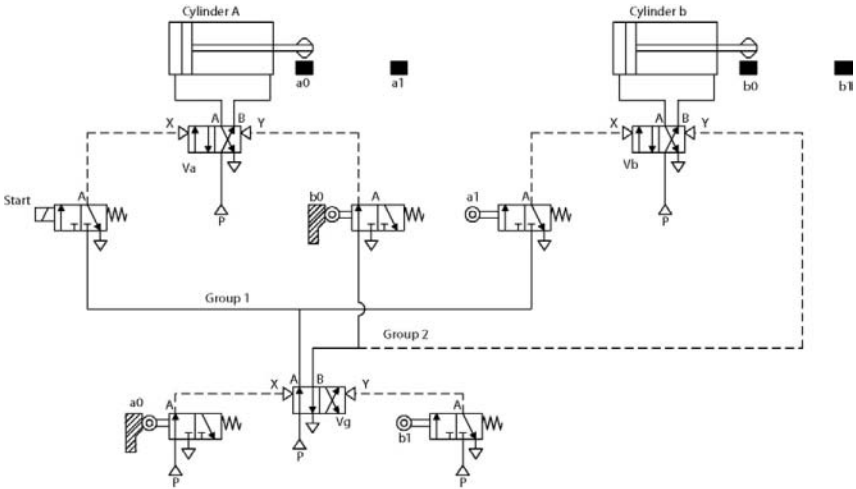
FIGURE 8.19 A solution using a group selector valve

4. Limit b1 is struck which shifts Vg to apply pressure to group 2.
5. Valve Vb now shifts and retracts cylinder B. Note there is no opposing signal via limit a1 because group 1 is depressurized.
6. Cylinder B retracts and limit b0 is struck. This operates Va, retracting cylinder A causing limit a0 to be struck when cylinder A is retracted.
7. Limit a0 shifts Vg depressurizing group 2 and repressurizing group 1 ready for the next cycle. We are now back at step 1.

The method of breaking a sequence down into smaller groups is a common way of handling very complex sequences.

PRESSURE-CONTROLLED SEQUENCES

Many simple applications can be controlled without limit switches by using the pressure rise when a cylinder reaches end of stroke to operate a shuttle valve. In Figure 8.20 a pulse on the solenoid valve shifts valve Va, causing cylinder A
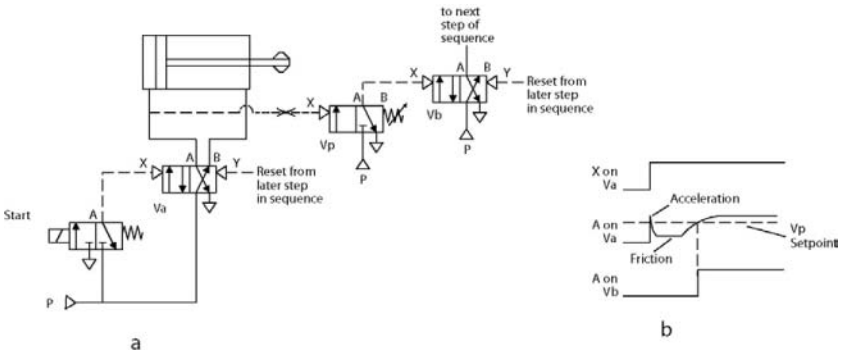
FIGURE 8.20 A pressure-driven sequence

to extend. There is no load on the cylinder so the extend pressure is low. When the cylinder reaches the end of the stroke (or the load is gripped), the pressure will rise as shown. This rise in pressure shifts valve Vb for the next step in the sequence.

MODULAR SEQUENCE VALVES

Most sequences can be considered to be a series of states. Each state causes one or more actions and the sequence progresses from state to state as signals from limit switches and similar devices occur. The sequence can be represented by a diagram such as Figure 8.21a, which is known as a state transition diagram, or STD, which shows the states, the actions performed in each state and the input signals which step the sequence from one state to the next state. For example, a state could extend a cylinder and the step onto the next state could be made by the cylinder extended limit.

A state can thus be represented by Figure 8.21b. It has three inputs: start, action(s) complete and reset (the last signal coming from the succeeding state); and three outputs: action (to the actuator(s)), state complete (to start on the next state) and reset (to the previous state).

Modular sequence valves provide the features needed to provide one state of an STD. Each state has one valve which is a single unit pre-made as in Figure 8.22 with a shuttle valve, an AND gate and an OR gate. The start command (from the completion of the previous state) shifts the shuttle valve and provides the action signal to the actuator(s). This signal also goes via the OR gate to reset the previous state. When the actuator has completed its action the action complete
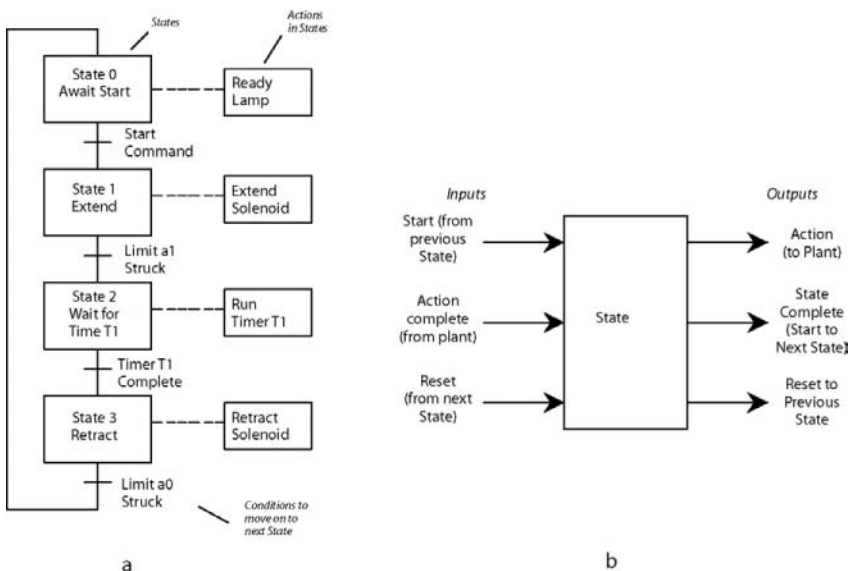
FIGURE 8.21 State transition diagrams (STDs): (a) a typical STD; (b) representation of one state
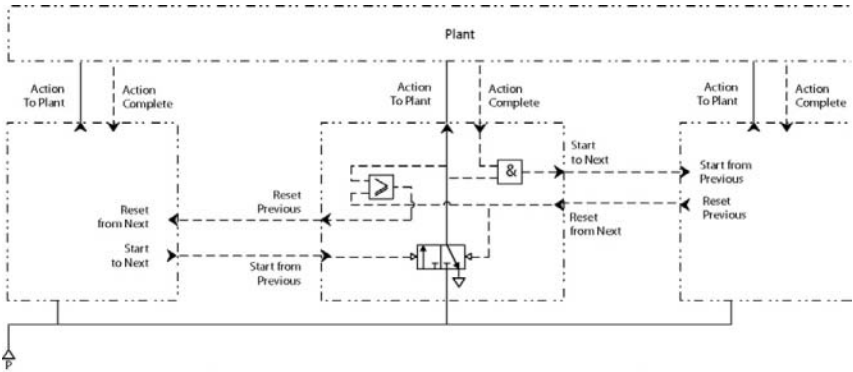
FIGURE 8.22 A modular sequence valve

signal from the plant AND the action signal give the complete signal which acts as the start signal for the next valve. The reset signal from the next valve shifts the first shuttle valve back, removing the signal to the actuator(s) ready for the next sequence.

The action output signal can be combined with other action outputs from other states by AND and OR gates in cases where an identical action is required in more than one state.

PROGRAMMABLE CONTROLLERS

Increasingly pneumatic and hydraulic sequences are controlled by program-mable controllers (PCs, not to be confused with personal computers). They are often called PLCs for programmable logic controllers, although strictly speaking this is a registered trademark of Allen Bradley.

A PC consists of a computer adapted for process control and programmed in a manner which is easy for first-line technicians to understand. Signals from the plant (e.g. limit switches, photocell, pushbuttons, switches, etc.) are connected to a range of input cards and outputs to the plant (e.g. solenoids, contactors, lamps, etc.) are driven by output cards.

These signals are identified by an address which is based on their connection points into the PLC. For example, on an SLC500 the form is:

I:<slot>/<bit> for inputs and
O:<slot>/<bit> for outputs.
Typical addresses could be:

| I:09/08 | Input signal 8 on the card in slot 9 |
| O:12/14 | Output signal 14 on the card in slot 12 |

Other manufacturers' schemes are similar, for example Siemens use I & Q (E & A in Europe for Eingang and Ausgang).

PCs also have a vast range of built-in functions: storage bits, timers, counters, sequencers, integer and floating point numbers, and much much more. With these it is quite straightforward to construct a hydraulic or pneumatic system
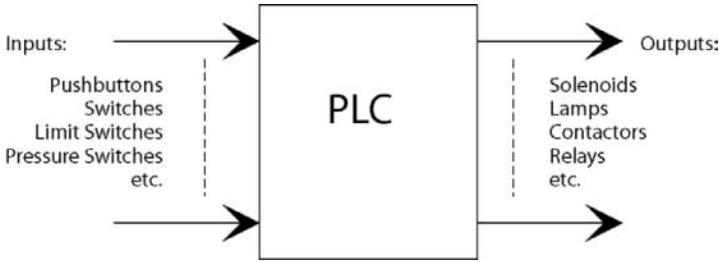
FIGURE 8.23 Representation of a PLC system

where all the solenoids and limit switches are solely connected back to a PC as shown in Figure 8.23.

This brings many advantages. The first is cost; PCs are very cheap. A simple PLC with eight inputs and four outputs costs, at the time of writing in early 2010, less than 100 Euros. It also simplifies the hydraulic or pneumatic piping, reducing installation costs in most applications. Second, it is easy to modify the system and make changes without re-piping. Finally, fault finding becomes much simpler because each input can be clearly seen and each solenoid can be tested on its own.

PCs are commonly programmed using something called 'Ladder Logic'. This mimics a relay circuit diagram using the American symbols of -] [- for a normally open contact (made when a signal is present) and -]/[- for a normally closed contact (made when a signal is absent).

Figure 8.24a shows a very simple PLC system with three inputs and two outputs. The two solenoids control a single cylinder and the system is required to follow the sequence of Figure 8.24b, i.e. extend, pause, retract, pause and repeat until run is de-selected. When run is de-selected the cylinder retracts to the back limit switch.

The program to achieve this is shown, written for an SLC500 PLC in Figure 8.25. The program is called a Ladder Diagram because it looks like the rungs of a ladder, and each statement is called a 'Rung'.

Rungs 0 and 2 control the two solenoids. Rungs 1 and 3 provide the time delays for the pauses at the end of stroke. Note that in Rungs 0 and 2 the solenoids hold themselves on via a branch around the initiating timer signal.
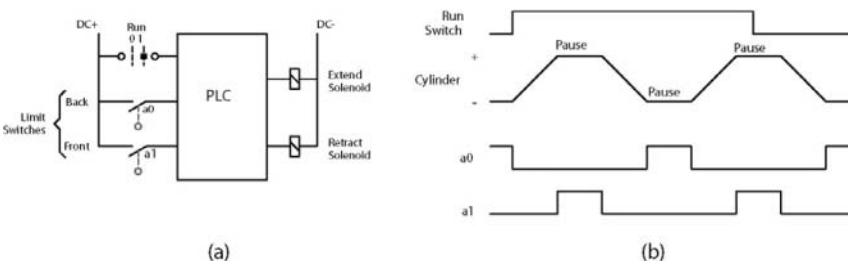
FIGURE 8.24 A simple PLC-controlled sequence: (a) connections of plant signals; (b) required sequence
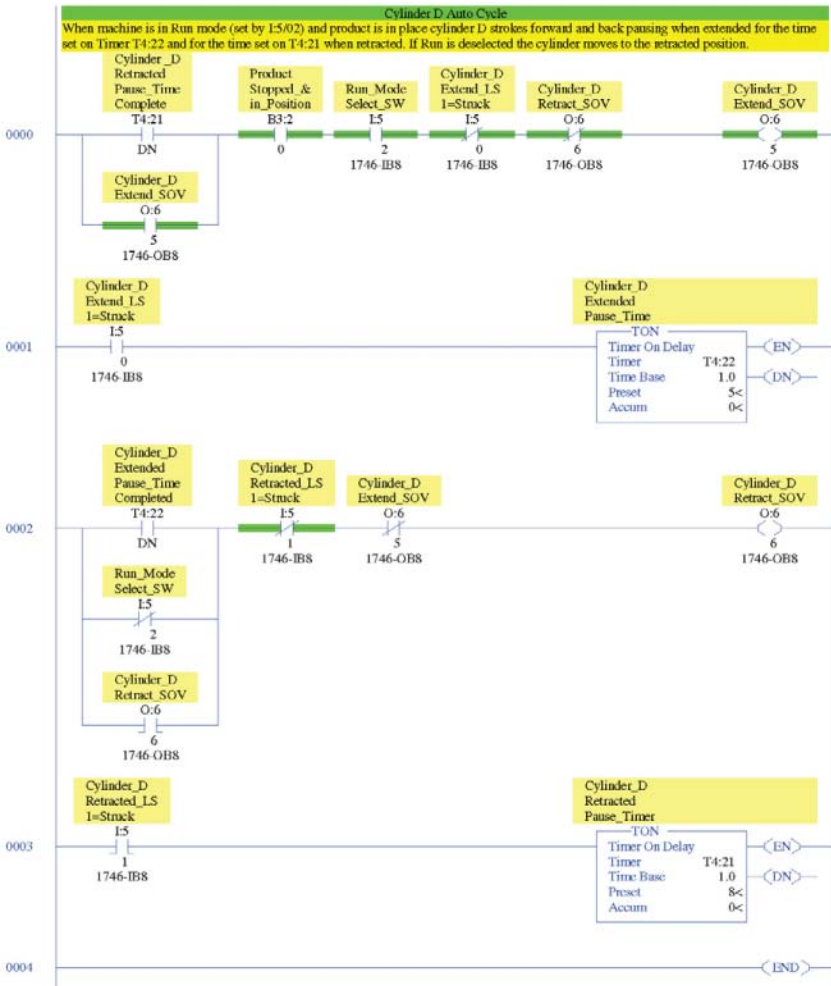
FIGURE 8.25 SLC500 PLC program for sequence. Signals which are present are shown highlighted which aids fault-finding. 1746-IB8 etc. below a signal are the card types which are producing the signal. B3:2/0 is an internal storage bit from another part of the program

Ladder Logic is not the only programming method. The IEC 1131 standard defines five programming methods:

- Ladder Rung (LD)
- Function Blocks (FBD) (AND gates, OR gates, SR memories, maths blocks, etc.)
- Statement List (STL) (simple text, e.g. LD X1 OR X2 ST Y0)
- Structured Text (ST) (structured high level language, similar to Pascal)
- Sequential Function Chart (SFC) (sequence diagram similar to Figure 8.26b).

Whichever method is used the programming system will allow the state of the inputs and outputs to be observed from the programming device and changes can be made quickly and simply (e.g. adding manual controls to the example above).
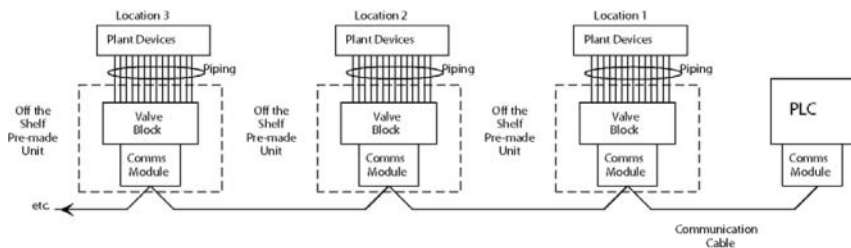
FIGURE 8.26 A distributed system. Remote I/O units are mounted at convenient locations to minimize plant cabling

DISTRIBUTED SYSTEMS

Early PLCs existed as a single control panel to which all signals had to cabled. Modern PLCs allow the input/output cards (called I/O) to be mounted remotely from the PLC processor, communication to/from all the I/O in one location taking place serially down single small cables. A hydraulic or pneumatic system will have the I/O cards pre-mounted at the manufacturers and will be delivered pre-wired and pre-tested to the project. A distributed system will look similar to Figure 8.26.

There are many advantages to this approach. It is, of course, much cheaper to install as the cabling is greatly reduced. Units such as skids, motor control centers and control desks can be tested prior to delivery, shortening the commissioning time. The system can also be easily expanded at a later date.

There are many standards for distributed systems. Amongst the most common are DeviceNet, Profibus DP, ASi-bus, Modbus and ControlNet, all of which allow the I/O to be mounted remotely from the processor. Many manufacturers of pneumatic and hydraulic equipment now offer valve manifolds complete with built-in distributed system interfaces which greatly simplifies both piping and cabling.

Chapter | nine

Safety, Fault-Finding and Maintenance

SAFETY

Most industrial plant has the capacity to maim or kill. It is therefore the responsibility of all people, both employers and employees, to ensure that no harm comes to any person as a result of activities on an industrial site.

Not surprisingly, this moral duty is also backed up by legislation. It is interesting that most safety legislation is reactive, i.e. responding to incidents which have occurred and trying to prevent them happening again. A prime example of this is the CDM regulations which arose because of the appalling safety record in the construction industry.

Safety legislation differs from country to country, although harmonization is underway in Europe. This section describes safety from a British viewpoint, although the general principles apply throughout the European community and are applicable in principle throughout the world. The descriptions are, of course, a personal view and should only be taken as a guide. The reader is advised to study the original legislation before taking any safety-related decisions.

Most safety legislation has a common theme. Employers and employees are deemed to have a *duty of care* to ensure the health, safety and welfare of the employees, visitors and the public. Failure in this duty of care is called *negligence*. Legislation defines required actions at three levels:

- *Shall* or *must* are absolute duties which have to be obeyed without regard to cost. If the duty is not feasible the related activity must not take place.
- *If practicable* means the duty must be obeyed if feasible. Cost is not a consideration. If an individual deems the duty not to be feasible, proof of this assertion will be required if an incident occurs.
- *Reasonably practicable* is the trickiest as it requires a balance of risk against cost. In the event of an incident an individual will be required to justify the actions taken.

There is a vast amount of safety legislation with varying degrees of authority. Acts (e.g. the Health and Safety at Work Act (HASWA)) are statutes passed by full parliamentary procedures and are enforced by criminal law. Often acts such as HASWA (called Enabling Acts) are arranged to allow supplementary regulations to be made by the Secretary of State without going through the full parliamentary procedure.

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Regulations are introduced under an enabling act. They have the same power and status as acts. Most British safety regulations have been made under the Health and Safety at Work Act 1974.

Approved Codes of Practice (ACOPs) are documents written to define safe working methods and procedures by organizations such as CENELEC and the British Standards Institute. They are approved by the Health and Safety Commission. Whilst they are not mandatory (i.e. there can be no prosecution for not following them), failure to follow ACOPs may be viewed as a contributory factor in investigations of an incident.

Codes of Practice are guidance codes provided by trade unions and professional organizations. These do not have the semi-legal status of ACOPs, but contain good advice. Again, though, implementation or otherwise can be given in evidence in court.

In Europe there is a serious attempt to have uniform legislation throughout the EU. At the top level is EC Regulations which override national legislation. Of most relevance are EC Directives which require national laws to be implemented.

In Britain the primary legislation is the Health and Safety at Work Act 1974 (HASWA). It is an enabling act, allowing other legislation to be introduced. It is wide ranging and covers everyone involved with work (both employers and employees) or affected by it. In the USA the Occupational Safety and Health Act (OSHA) affords similar protection.

HASWA defines and builds on general duties to avoid all possible hazards, and its main requirement is described in section 2(1) of the act:

It shall be the duty of every employer to ensure, so far as is reasonably practicable, the health, safety and welfare at work for his employees

This duty is extended in later sections to visitors, customers, the general public and (upheld in the courts) even trespassers. The onus of proof of *Reasonably Practicable* lies with the employer in the event of an incident.

Section 2(2) adds more detail by requiring safe plant, safe systems of work, safe use of articles and substances (i.e. handling, storage and transport), safe access and egress routes, safe environment, welfare facilities and adequate information and training.

If an organization has five or more employees it must have a written safety policy defining responsibilities and employees must be aware of its existence and content (section 2(3)). Employers must consult with worker safety representatives.

The act is not aimed purely at employers; employees also have duties described in sections 7 and 8 of the act. They are responsible for their own and others' safety and must co-operate with employers and other people to ensure safety, i.e. they must follow safe working practices. They must not interfere with any safety equipment (e.g. tampering with interlocks on movable guards).

The act defines two authorities and gives them power for the enforcement of the legislation (sections 10–14 and 18–24). The Health and Safety Commission is the more academic of the two, and defines policy, carries out research,

develops safety law and disseminates safety information. The Health and Safety Executive (HSE) implements the law by inspection and can enforce the law where failings are found. Breaches of HASWA amount to an indictable offence and the HSE has the power to prosecute the offenders.

The powers of HSE inspectors are wide. They can enter premises without invitation and take samples, photographs, documents, etc. People, as well as organizations, may be prosecuted if a safety failing or incident arises because of neglect by a responsible person.

The HSE also has the power to issue notices against an organization. The first, an Improvement Notice, is given where a fairly minor safety failing is observed. This notice requires the failing to be rectified within a specified period of time. The second, a Prohibition Notice, requires all operations to cease immediately and not restart until the failing is rectified and HSE inspectors withdraw the notice.

It is all but impossible to design a system which is totally and absolutely fail-safe. Modern safety legislation, such as the Six Pack, recognizes the need to balance the cost and complexity of the safety system against the likelihood and severity of injury. The procedure, known as *risk assessment*, uses common terms with specific definitions:

Hazard	The potential to cause harm
Risk	A function of the likelihood of the hazard occurring and the severity
Danger	The risk of injury.

Risk assessment is a legal requirement under most modern legislation, and is covered in detail in standard prEN1050 *'Principles of Risk Assessment'*.

The first stage is identification of the hazards on the machine or process. This can be done by inspections, audits, study of incidents (near misses) and, for new plant, by investigation at the design stage. Examples of hazards are: impact/ crush, snag points leading to entanglement, drawing in, cutting from moving edges, stabbing, shearing (leading to amputation), electrical hazards, temperature hazards (hot and cold), contact with dangerous material and so on. Failure modes should also be considered, using standard methods such as HAZOPS (Hazard and Operability Study, with key words Too much of and Too little of), FMEA (Failure Modes and Effects Analysis) and Fault Tree Analysis.

With the hazards documented the next stage is to assess the risk for each. There is no real definitive method for doing this, as each plant has different levels of operator competence and maintenance standards. A risk assessment, however, needs to be performed and the results and conclusions documented. In the event of an accident, the authorities will ask to see the risk assessment. There are many methods of risk assessment, some quantitative assigning points, and some using broad qualitative judgments.

Whichever method is used there are several factors that need to be considered. The first is the severity of the possible injury. Many sources suggest the following four classifications:

Fatality   One or more deaths.
Major   Non-reversible injury, e.g. amputation, loss of sight, disability.
Serious   Reversible but requiring medical attention, e.g. burn, broken joint.
Minor   Small cut, bruise, etc.

The next step is to consider how often people are exposed to the risk. Suggestions here are:

Frequent   Several times per day or shift.
Occasional   Once per day or shift.
Seldom   Less than once per week.

Linked to this is how long the exposure lasts. Is the person exposed to danger for a few seconds per event or (as can occur with major maintenance work) several h? There may also be a need to consider the number of people who may be at risk, often a factor in petrochemical plants.

Where the speed of a machine or process is slow, or there is a lengthy and obvious (e.g. noisy) start-up, the exposed person can easily move out of danger in time. There is obviously less risk here than with a silent high-speed machine which can operate before the person can move. From studying the machine operation, the probability of injury in the event of failure of the safety system can be assessed as: *certain, probable, possible, unlikely.*

From this study, the risk of each activity is classified. This classification will depend on the application. Some sources suggest applying a points scoring scheme to each of the factors above then using the total score to determine *high, medium* and *low* risks. Maximum possible loss (MPL), for example, uses a 50-point scale ranging from 1 for a minor scratch to 50 for a multi-fatality. This is combined with the frequency of the hazardous activity (F) and the probability of injury (again on a 1–50 scale) in the formula:

$$\text{Risk rating (RR)} = F \times (MPL + P)$$

The course of action is then based on the risk rating.

An alternative and simpler (but less detailed approach) uses a table as in Figure 9.1 from which the required action can be quickly read.

Likelihood of incident	Severity of outcome
9. Almost certain	9. Fatality
8. Very likely	8. Permanent total incapacity
7. Probable	7. Permanent severe incapacity
6. Better than even chance	6. Permanent slight incapacity
5. Even chance	5. Off work for > 3 weeks but subsequent recovery
4. Less than even chance	4. Off work for 3 days to 3 weeks with full recovery
3. Improbable	3. Off work for less than 3 days with full recovery
2. Very improbable	2. Minor injury, no lost time
1. Almost impossible	1. Trivial injury
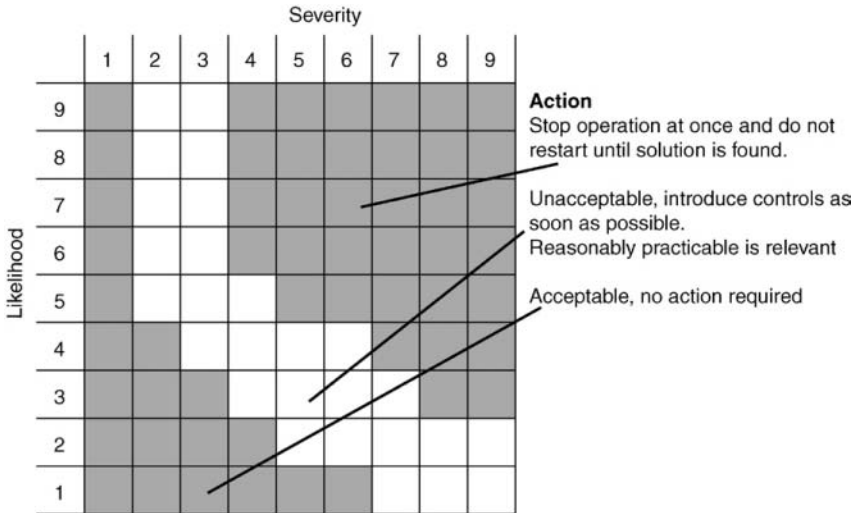
FIGURE 9.1 A typical risk assessment table. Although this is based on a real application, it should not be applied elsewhere without supporting study and documentation. The main point of a risk assessment is identifying and reducing the risks associated with a specific task

There is, however, no single definitive method, but the procedure used must suit the application and be documented. The study and reduction of risks is the important aim of the activity.

The final stage is to devise methods of reducing the residual risk to an acceptable level. These methods will include removal of risk by good design (e.g. removal of trap points), reduction of the risk at source (e.g. lowest possible speed and pressures, less hazardous material), containment by guarding, reducing exposure times, provision of personal protective equipment and establishing written safe working procedures which must be followed. The latter implies competent employees and training programs.

There is a vast amount of legislation covering health and safety, and a list is given below of those which are commonly encountered in industry. It is by no means complete, and a fuller description of these, and other, legislation is given in the third edition of the author's *Industrial Control Handbook*. An even more detailed study can be found in *Safety at Work* by John Ridley, both books published by Butterworth-Heinemann.

Commonly encountered safety legislation:
Health and Safety at Work Act 1974 (the prime UK legislation)
Management of Health and Safety at Work Regulations 1992
Provision and Use of Work Equipment Regulations 1992 (PUWER)
Manual Handling Regulations 1992
Workplace Health, Safety and Welfare Regulations 1992
Personal Protective Equipment Regulations 1992
Display Screen Equipment Regulations 1992
(the previous six regulations are based on EC directives and are known collectively as 'the six pack')

Machinery Directive 2006/42/EC (see note below)

EN 286-1:1998+A2:2005 Simple pressure vessels designed to contain air or nitrogen

BS EN574:1996+A1:2008 Two handed control

EMC Directive 1993 Electromagnetic interference

Low Voltage Directive 73/23/EEC and 2006/95/EC

Safety of Fluid Power Systems, Hydraulics. EN982 1996 (see note below)

Safety of Fluid Power Systems, Pneumatics. EN983 1996 (see note below)

Reporting of Injuries, Diseases and Dangerous Occurrences Regulations (RIDDOR) 1995

Construction (Design and Management) Regulations (CDM) 1994

Electricity at Work Regulations 1990

Control of Substances Hazardous to Health (COSHH) 1989

Noise at Work Regulations 1989

Ionising Radiation Regulations 1985

Safety Signs and Signals Regulations 1996

Highly Flammable Liquids and Liquefied Petroleum Gas Regulations 1972

Fire Precautions Act 1971

Safety Representative and Safety Committee Regulations 1977

Health and Safety Consultation with Employees Regulations 1996

Health and Safety (First Aid) Regulations 1981

Pressure Systems and Transportable Gas Containers Regulations 1989

The Machinery Directive (formerly 98/37/EC) is implemented in the UK as the Supply of Machinery (Safety) Regulations 1998 and requires manufacturers of ready to use equipment, machine or plant to state the equipment meets all the Essential Health and Safety Requirements (ESHR) of the relevant directives and legislation. The manufacturer gives the equipment a CE conformity mark. Note that compliance is a lot more than assembling pre-made units which are individually CE marked; CE plus CE does not make CE. Generally the manufacturer will provide a file showing compliance and listing safety procedures, safe ways of working etc.

The Health and Safety Executive website has many excellent publications which can be downloaded free as PDF files. The important EN982 and EN983 books and others can be found here:

http://www.hse,gov.uk/pubns/

British Standards can also be found on the British Standards website:

http://shop.bsifgroup.com/

As hydraulic and pneumatic systems are nowadays invariably linked to programmable controllers (PLCs), the reader should also consult the occasional paper OP2 'Microprocessors in Industry' published by the HSE in 1981 and the two later booklets 'Programmable Electronics Systems in Safety Related Applications', Book 1, an Introductory Guide and Book 2, General Technical Guidelines, both published in 1987. These also can be found on the HSE website.

Electrical systems are generally recognized as being potentially lethal, and all organizations must, by law, have procedures for isolation of equipment, permits to work, safety notices and defined safe-working practices. Hydraulic and pneumatic systems are no less dangerous, but tend to be approached in a far more carefree manner. High-pressure air or oil released suddenly can reach an explosive velocity and can easily maim, blind or kill. Unexpected movement of components such as cylinders can trap and crush limbs. Spilt hydraulic oil is very slippery, possibly leading to falls and injury. It follows that hydraulic and pneumatic systems should be treated with respect and maintained or repaired under well-defined procedures and safe-working practices as rigorous as those applied to electrical equipment.

Some particular points of note are:

- before doing *anything,* think of the implications of what you are about to do, and make sure anyone who could be affected knows of your intentions. Do not rush in, instead, *think*;
- anything that can move with changes in pressure as a result of your actions should be mechanically secured or guarded. Particular care should be taken with suspended loads. Remember that fail open valves will turn *on* when the system is depressurized;
- never disconnect pressurized lines or components. Isolate and lock-off relevant legs or depressurize the whole system (depending on the application). Apply safety notices and locks to inhibit operation by other people. Ideally the pump or compressor should be isolated and locked off at its MCC. Ensure accumulators in a hydraulic system are fully blown down. Even then, make the first disconnection circumspectly;
- in hydraulic systems, make prior arrangements to catch oil spillage (from a pipe replacement, say). Have containers, rags and so on ready and, as far as is possible, keep spillage off the floor. Clean up any spilt oil before leaving;
- where there is any electrical interface to a pneumatic or hydraulic system (e.g. solenoids, pressure switches, limit switches) the control circuits should be isolated, not only to remove the risk of electric shock, but also to reduce the possibility of fire or accidental initiation of some electrical control sequence. Again, *think* how things interact;
- after the work is completed, leave the area tidy and clean. Ensure people know that things are about to move again. Check there is no one in dangerous areas and sign off all applied electrical, pneumatic or hydraulic isolation permits to work. Check for leaks and correct operation;
- many components contain springs under pressure. If released in an uncontrolled manner these can fly out at high speed, causing severe injury. Springs should be released with care. In many cases manufacturers supply special tools to contain the spring and allow gradual and safe decompression.

CLEANLINESS

Most hydraulic or pneumatic faults are caused by dirt. Very small particles nick seals, abrade surfaces, block orifices and cause valve spools to jam. In hydraulic and pneumatic systems cleanliness is next to Godliness. Dismantling a valve in

an area covered in swarf or wiping the spool on an old rag kept in an overall pocket does more harm than good.

Ideally components should not be dismantled in the usual dirty conditions found on site, but returned to a clean workshop equipped with metal-topped benches. Too often one bench is used also for general mechanical work: it needs little imagination to envisage the harm metal filings can do inside a pneumatic or hydraulic system.

Components and hoses are delivered with all orifices sealed with plastic plugs to prevent dirt ingress during transit. These should be left in during storage and only removed at the last possible moment.

Filters exist to remove dirt particles, but only work until they are clogged. A dirty filter bypasses air or fluid, and can even make matters worse by holding dirt particles then releasing them as one large collection. Filters should be regularly checked and cleaned or changed (depending on the design) when required.

Oil condition in a hydraulic system is also crucial in maintaining reliability. Oil which is dirty, oxidized or contaminated with water forms a sticky gummy sludge, which blocks small orifices and causes pilot spools to jam. Oil condition should be regularly checked and suspect oil changed before problems develop.

FAULT-FINDING INSTRUMENTS

Electrical fault-finding is generally based on measurements of voltage, current or (less often) resistance at critical points in the circuit. Of these, voltage is easier to measure than current unless ammeters or shunts have been built into the circuit, and resistance measurement usually requires the circuit to be powered down and the device under test disconnected to avoid sneak paths. An electronic circuit is given in Figure 9.2. This converts a voltage input V_i to a current signal I, where $I = V_i/R$. Such a circuit is commonly used to transmit an instrumentation signal through a noisy environment. A typical checking procedure could be:

Voltage checks:	A	(input signal)
	B and C	(amplifier ± 15 V supply)
	D	(return voltage should equal A)
	E	(across load, 15 V indicates open circuit load, 0 V indicates short-circuit load).
Followed by:		
Current checks:	X, Y	(X should equal Y and both equal A/R)
Resistance checks:	F, G	(for open- or short-circuit load or resistance).

In pneumatic or hydraulic systems, pressure measurement is equivalent to electrical voltage measurement, while flow measurement is equivalent to current measurement. There is no direct simple measurement equivalent to electrical
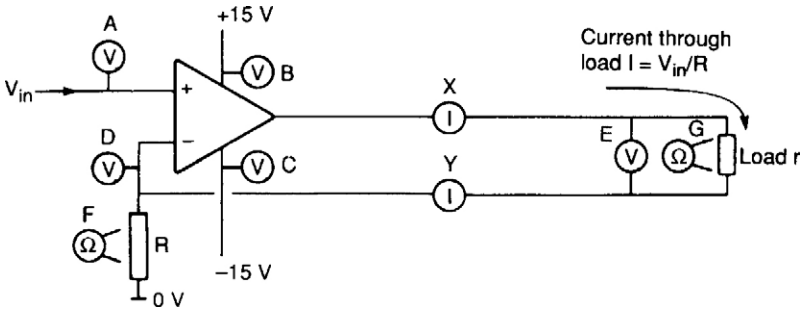
FIGURE 9.2 Test measurement points on an electronic circuit

resistance. Pressure tests and (to a lesser extent) flow tests thus form the bases of fault-finding in pneumatic or hydraulic systems.

There is, however, a major difference in the ease of access. Electrical systems abound with potential test points; a voltage probe can be placed on practically any terminal or any component, and (with a little more trouble) a circuit can be broken to allow current measurements to be made.

In fluid systems, oil or gas is contained in pipes or hoses, and measurements can only be made at test points which have been built in as part of the original design. Test points can be plumbed in on an ad hoc basis but this carries the dangers of introducing dirt from cutting or welding, and in hydraulic systems any air introduced will need to be bled out. The designer should, therefore, carefully consider how faults in the system can be located, and provide the necessary test points as part of the initial design.

By far the most common technique is a built-in rotary pressure select switch, as shown in Figure 9.3, which allows pressure from various strategic locations to be read centrally. An alternative technique uses quick-release connections, allowing a portable pressure meter to be carried around the system and plugged in where required.
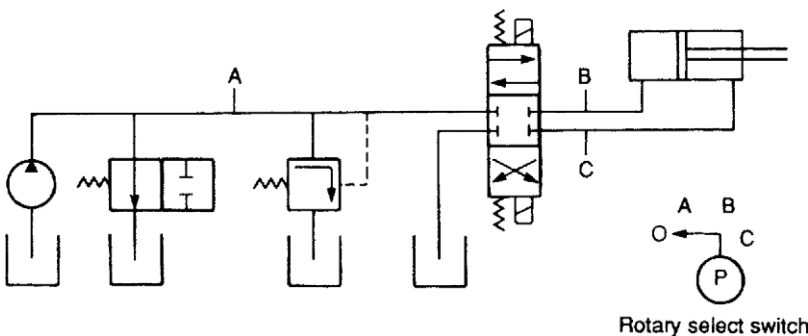
FIGURE 9.3 The commonest hydraulic and pneumatic test system, a rotary select switch
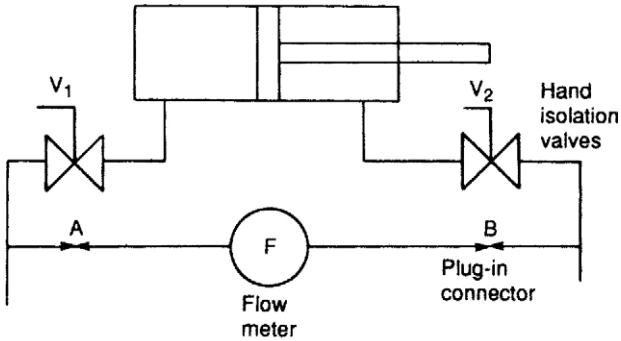
FIGURE 9.4   Checking flow available at an actuator

Flow measurement is more difficult, as the basic flow transducer needs to be built in. Portable flow meters can be used, as shown in Figure 9.4, where the flow available for a cylinder is checked by closing hand valves V_1 and V_2 while connecting a flow meter between quick-release connections A and B.

The UCC System 20 is an invaluable three-in-one 'plumbed-in' test instrument which provides measurement of pressure, flow and temperature at the installed point with a plug-in test meter. The inclusion of such a device should be provided immediately after every pump (but before the first relief valve) to allow pump delivery to be checked and at crucial points such as the pressure lines to a critical cylinder or motor. Remember, with hydraulics and pneumatics the test points have to be designed in.

An indicator in the plug of a solenoid valve will show voltage is arriving at the solenoid (see Figure 9.9) but this is not a foolproof indication that the solenoid itself is operating. The coil may, for example, be open circuit or there is a loose connection inside the plug. RS components sell a very cheap and useful solenoid tester (part number 214-338), which illuminates when held in a strong magnetic field. About the size of a fountain pen, it can be touched onto the body of a solenoid to see if the solenoid really is being energized.

FAULT-FINDING

Fault-finding is often performed in a random and haphazard manner, leading to items being changed for no systematic reason beyond 'Fred got it working this way last time'. Such an approach may work eventually (when every component has been changed!) but it is hardly the quickest, or cheapest, way of getting a faulty system back into production. In many cases more harm than good results, both with introduction of dirt into the system, and from ill-advised 'here's a control adjustment, let's twiddle it and see if that makes any difference' approach. There must be a better way.

There are three maintenance levels. First-line maintenance is concerned with getting faulty plant running again. When the cause of a fault is found, first-line

staff have the choice of effecting a first-line, on-site, repair (by replacing a failed seal, say) or changing the complete faulty unit for a spare. This decision is based on cost, time, availability of spares, technical ability of staff, the environment on site and company policy.

Second-line maintenance is concerned with repair to complete units changed by first-line maintenance staff. It should be performed in clean and well-equipped workshops. Work is usually well-defined and is often a case of following manufacturers' manuals.

The final level is simply the return of equipment for repair by the manufacturer. The level at which this is needed is determined by the complexity of equipment, ability of one's staff, cost and the turn-round time offered by the manufacturer.

Of these three levels, first-line maintenance is hardest as work is ill-defined, pressures from production staff are great and the responsibility high. Unfortunately, it is too often seen as a necessary evil.

Fault-finding is, somewhat simplistically, represented by Figure 9.5. All the evidence on the fault gathered so far is evaluated, and possible causes considered. The simplest test to reduce the number of possibilities is then performed and the cycle repeated until the fault is found.

The final steps in Figure 9.5 are concerned with fault recording and fault analysis. Any shift crew (which performs almost all the first-line repairs) only sees one-quarter of all faults. The fault recording and analysis process shows if there is any recurring pattern in faults, indicating a design or application problem. Used diplomatically, the records may also indicate shortcomings in crews' knowledge and a need for training.

Modern plants tend to be both complex and reliable. This means that a maintenance crew often sees a plant in detail for the first time when the first fault occurs. (Ideally, of course, crews should be involved at installation and commissioning stages – but that is another story!) It is impossible to retain the layout of all bar the simplest of systems in the mind, so it is essential to have schematic diagrams readily available.

Equally important, readings at each test point should be documented when the system is working correctly. It is not much use to know pressure at TP_3 is 15 bar, the motor draws 75 A or flow to rotary actuator C is 1500 L min^{-1} under fault conditions, without knowing what the normal readings are.

It can often be difficult to decide what a fault really is; usually the only information is simply 'the Firkling Machine is not working'. The first diagnostic step is, therefore, to establish what is *really* wrong – whether there is one fault or several from a common cause. A quick visual and manual check should be made for any obvious aberrations: noise, vibration, heat, leaks, unusual motor current.

From maintenance records it should be possible to see if any recent work has been done or if this is a recurring fault. Recent work is always suspect – particularly if the unit has not been used since the work was done. Some points to check if work has recently been performed are:
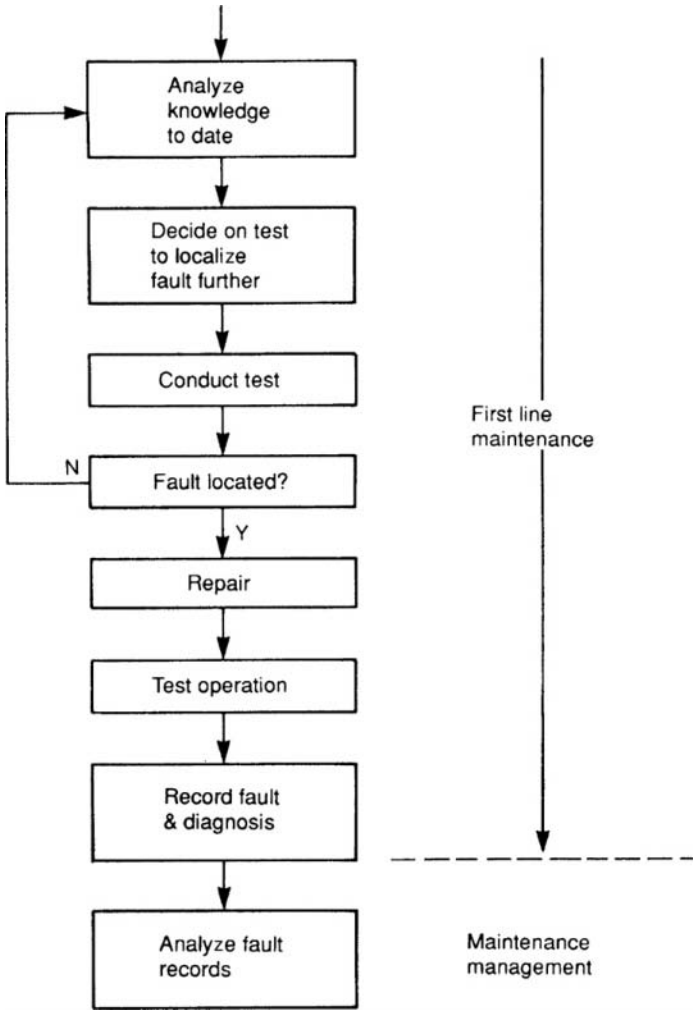
FIGURE 9.5 Fault-finding process

- whether the correct units were fitted. Stores departments are not infallible and lookalike units may have been fitted in error. Non-return valves can sometimes be fitted the wrong way round;
- whether all handvalves are correctly open or shut. Many systems are built with standby pumps or compressors with manual changeover. These are (in the author's experience) a constant source of trouble after a change-over (one invariable characteristic seems to be one less valve handle than there are valves!). Valves can also creep open. Figure 9.6 shows a common fault situation with two hydraulic units, one in use and one on standby. If any of the hand isolation valves V_1–V_4 are set incorrectly open on the main or standby units, flow from the duty unit returns direct to tank via
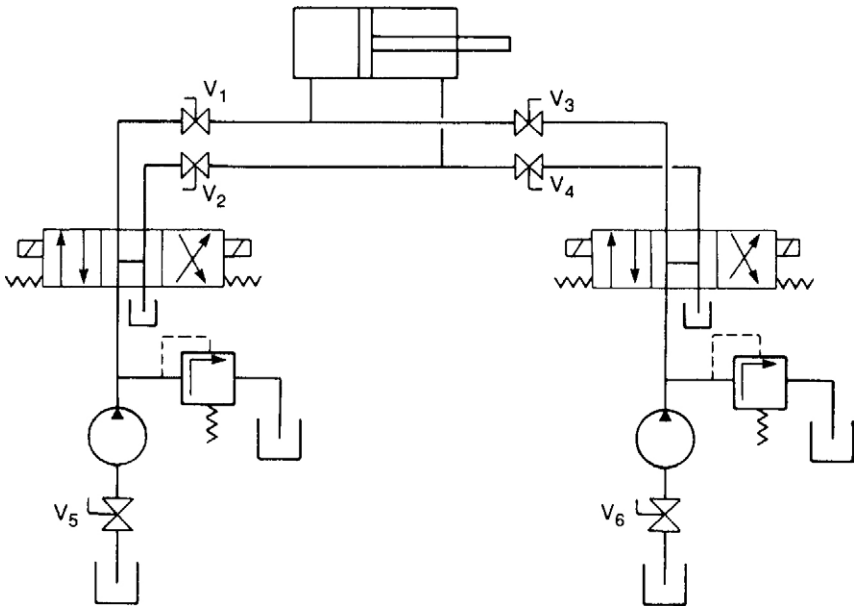
FIGURE 9.6 A common source of trouble: a main/standby system with hand isolation valves. Wrong setting of valves leads to many obscure faults

the center position of the standby directional valve and the actuator will not move;

- after electrical work, check the direction of rotation of the pump or compressor. Most only operate in one direction, usually defined with an arrow on the casing, and may even be damaged by prolonged reverse running;
- have any adjustments been 'twiddled' or not set correctly after an item has been changed? On many directional valves, for example, the speeds of operation from pilot to main spool can be set by Allen key adjustments. If these are maladjusted, the main spool may not move at all.

If no recent work has been done, and these quick checks do not locate a fault, it is time to start the fault-finding routine of Figure 9.5. One advantage of pneumatic systems is their natural break into distinct portions: (1) a supply portion up to and including the receiver and (2) one or more application portions after the receiver.

The pressure gauge on the receiver allows a natural fault-finding split.

Problems generally fall into three types: a lack of force, low speed (or no speed), or erratic operation. Lack of force or no movement is generally a pressure-related fault. Low speed arises from a flow fault. Erratic operation can arise from sticking valves or from air in a hydraulic system.

Usually pressure monitoring is much easier than flow monitoring but is often misunderstood. A typical example of fault-finding using pressure test points is given in Figure 9.7. Up to time A the system unloads via the solenoid-operated unloading valve V_1. When valve V_1 energizes, pressure rises to the setting

(a) Circuit diagram

(b) Pressure readings
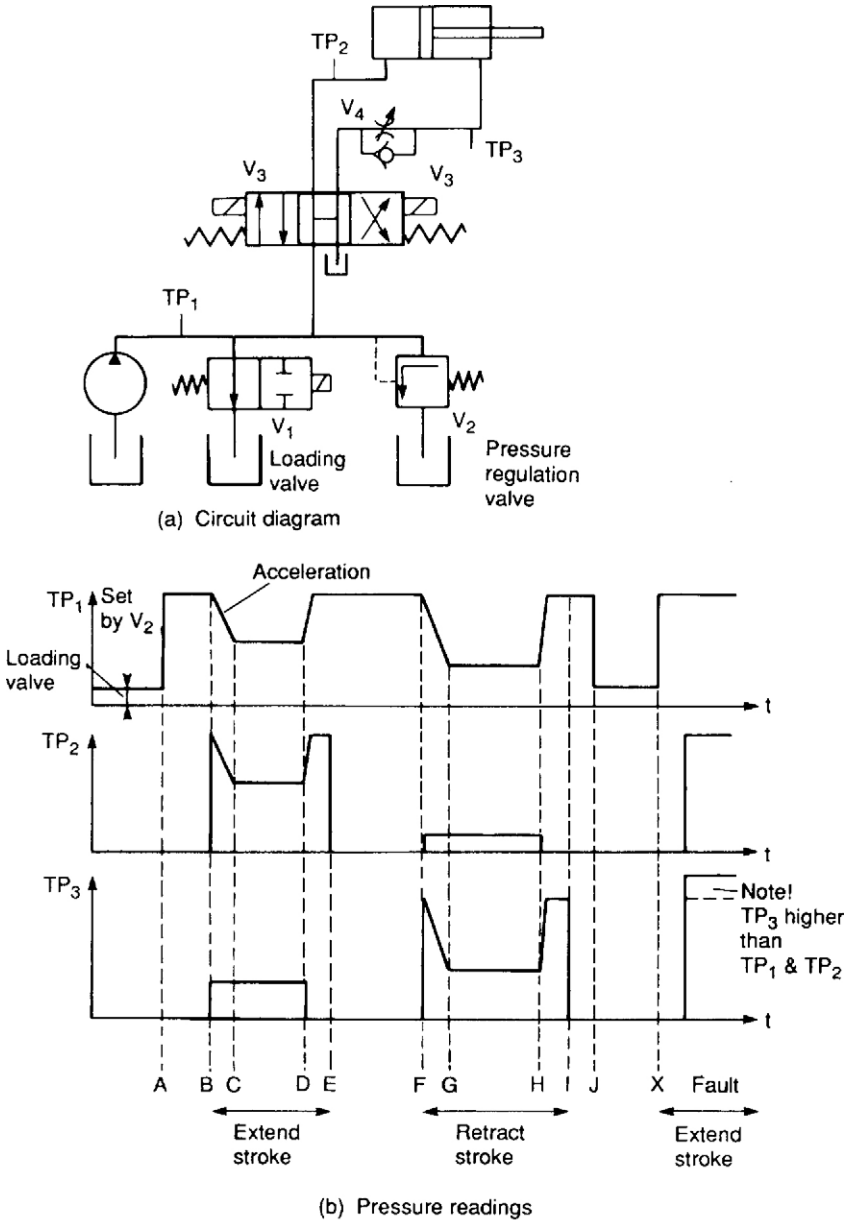
FIGURE 9.7 Fault-finding with pressure test points

of the relief valve V_2. At time C, directional valve V_3 calls for the cylinder to extend. Pressure *falls* as the cylinder accelerates, until the cylinder is moving at constant speed when P=F/A. At time D, the cylinder reaches the end of travel, and the pressure rises back to the setting of the relief valve. Directional valve V_3 de-energizes at E. Note the low pressures in return line test points.

A similar retract stroke takes place during time F to I. The pressure between G and H is lower than between C and D, because friction alone opposes the movement.

The loading valve comes off at time J.

It is important to monitor return line pressure. A fault exists from time X onwards; the return line from the cylinder is blocked, possibly because the spool in the meter-out flow control valve has jammed, allowing no fluid to return. At time Y the directional valve operates causing a rise in pressure on test point TP_2 to the setting of the relief valve. Because of the blockage in the return line, point TP_3 also rises to a *higher* pressure because of the lower annulus area on the return side of the piston ($P_1A = P_2a$, remember!).

Pressure is therefore a good indication of what is going on in a system, the pressure being the *lowest* demanded by the loading/unloading valves, the relief valve(s) or the load itself.

Observation of pressure, though, is not the whole story. A pump or actuator can have major internal leakage caused by wear and still function and produce normal pressure readings. The pump on the later Figure 8.10 can deliver 60 l per minute and the normal pressure required to raise the cylinder, measured at T6, is 80 bar. Suppose there is a leak on the seal of the actuator which passes 55 l per minute. The pump will deliver 60 l per minute, 55 l of which will go straight back to the tank via the leaking seal. The remaining 5 l will, however, still cause the cylinder to rise, albeit at a much reduced speed. Because the piston is supporting the load the pressure at T6 will be unchanged at 80 bar. The leak has not changed the observed pressure.

Similarly if, again because of wear, the pump has much reduced efficiency and is only delivering 10 l per minute, it can still develop 80 bar at T6 and cause the cylinder to rise, again at a lower than normal speed.

A hydraulic pump is a positive displacement device (see Chapter 2). This has useful implications when fault-finding. If a pump is working its flow *must* be getting back to the tank via some route. If it does not the pressure will rise and the oil will eventually go everywhere! Tracking the oil flow route by as simple a method as following warm pipes by hand can sometimes indicate what is wrong.

Remember these basic facts for fault-finding:

- Knowing the delivery rate of the pump is vital and would immediately identify the source of the leaks described above. If there is not a UCC system 20 or similar flow sensor immediately after the pump but before the first relief valve consider installing one as soon as possible.
- Hydraulic pumps are invariably positive displacement pumps. If a pump is delivering fluid it must be going somewhere.
- Acceleration is determined by pressure.
- Force is determined by pressure.
- Velocity (speed) is determined by flow.
- The pressure at any point is determined by the *lowest* pressure the system can provide under the current conditions.
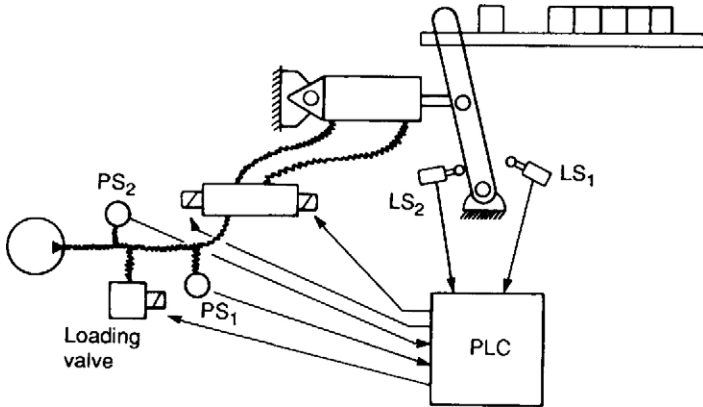
FIGURE 9.8 A typical sequencing application

The interface with the electrical control can cause confusion. The control sequence should be clearly understood. Figure 9.8 shows a typical electrical/hydraulic scheme used to build a tight pack of objects. An object is placed onto the skid, and its presence noted by a proximity detector connected as an input to a programmable controller (PLC).

When the PLC sees an object, it energizes the loading valve, and causes the cylinder to extend. The cylinder extends until the front limit switch LS $_j$ makes (for the first few objects) *or* the pressure switch PS$_2$ makes (indicating a full stalled pack) *or* timeout (indicating some form of fault). The cylinder then returns to the back limit LS$_2$ *or* a timeout (again indicating a fault) when the loading valve is de-energized. The PLC also monitors pump action via pressure switch PS$_1$, which is made whenever the loading valve is energized.

A knowledge of the complete system, both electrical and hydraulic, is required to fault-find on this application. Fault-finding involves checking the sequence by monitoring the state of electrical outputs to solenoids and inputs from limit switches.

All solenoid valves should have an indicator in the plug tops to allow electrical signals to be observed local to the valves. Indicator blocks which fit between plug and valve are available for retro fitting onto systems without this useful feature. It should be remembered, though, that indications purely show electrical voltage is present – it does not, for example, identify an open circuit solenoid coil.

Solenoids can operate on AC (usually 110 V AC) or DC (usually 24 V DC). DC solenoids have totally different operating characteristics. An AC solenoid has a very high inrush current producing a high initial force on the pilot spool. As the spool moves in, the inductance of the coil rises and current falls to a low holding current (and a low force on the pilot spool). If the pilot
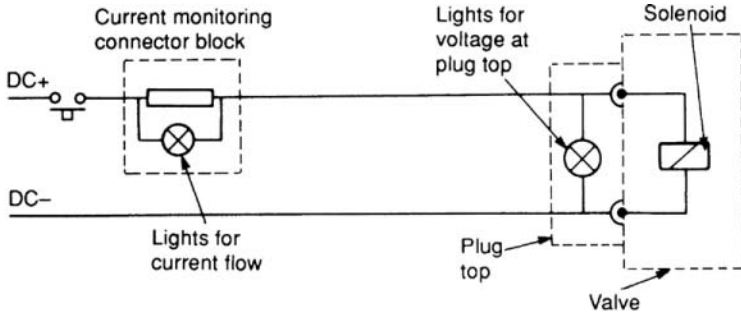
FIGURE 9.9 Monitoring a DC solenoid
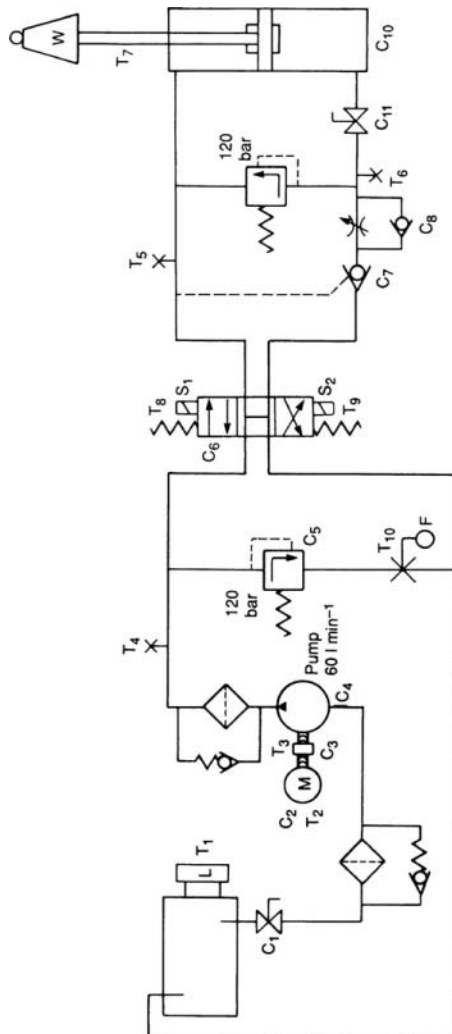
FIGURE 9.10 Hydraulic circuit for diagnostic chart
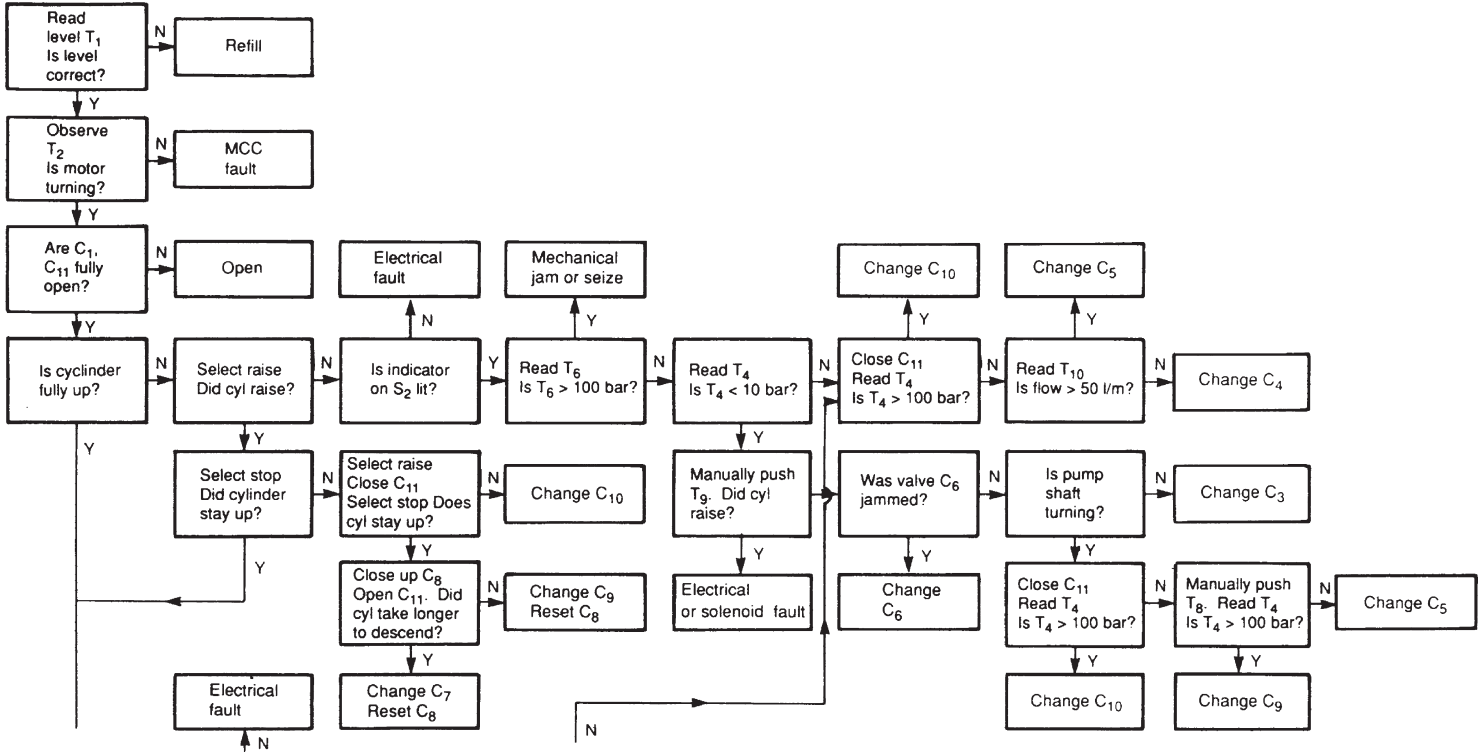
FIGURE 9.11 Fault-finding flowchart for circuit of Figure 9.10
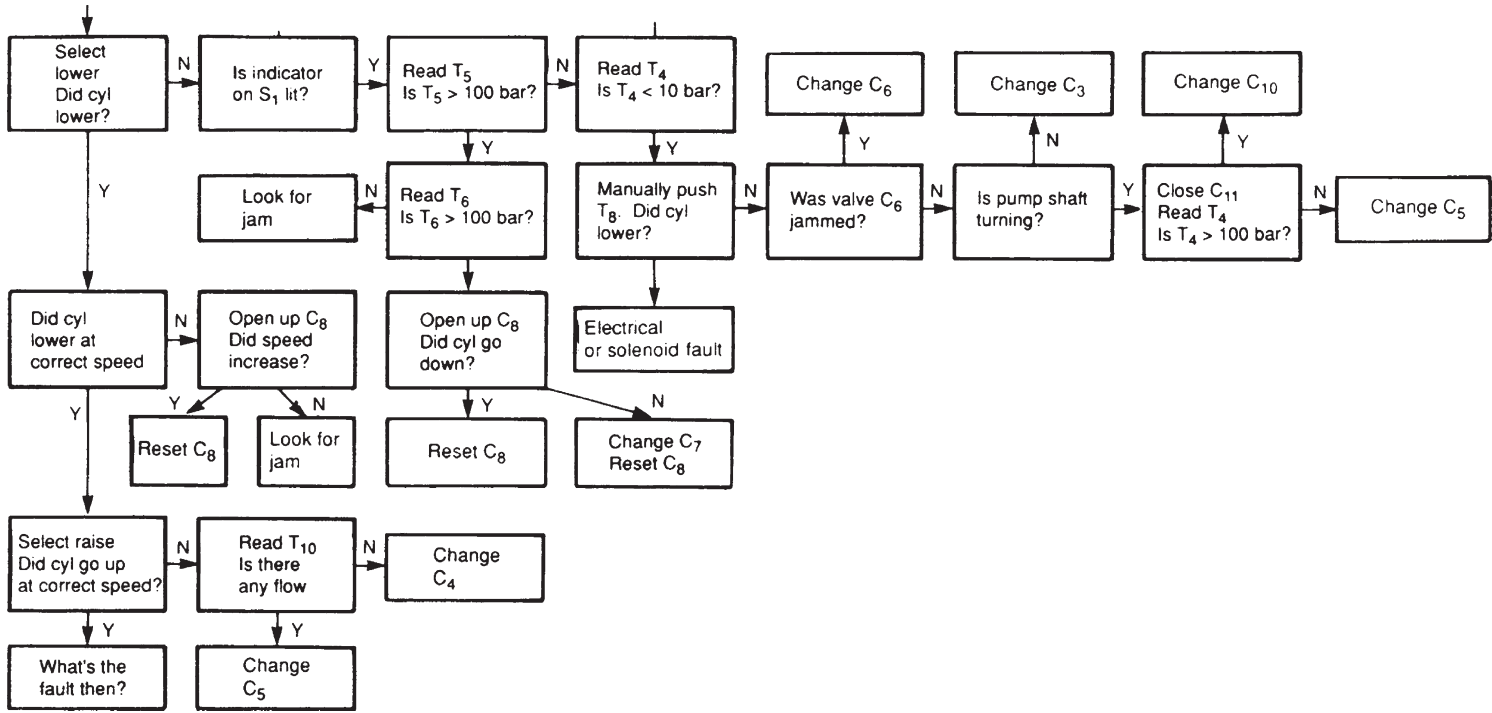
FIGURE 9.11 Continued

spool jams the current remains high, causing the protection fuse or breaker to open or the solenoid coil to burn out if the protection is inadequate. Operation of a 110 volt solenoid system with cold oil is best undertaken with a pocketful of fuses.

The current in a DC solenoid is determined by the coil resistance and does not change with pilot spool position. The solenoid does not, therefore, give the same 'punch' to a stiff spool but will not burn out if the spool jams. Current in a DC solenoid also tends to be higher, requiring larger-size cables, particularly if a common return line is used from a block of solenoids.

A useful monitoring device is the through-connector with an integral current indicator, shown in Figure 9.9. This *does* give indication of an open circuit coil and combined with the indicator on the plug top can help find most electrical faults.

On most electrically operated valves the pilot spool can be operated manually by pushing the spool directly with a rod (welding rod is ideal!). Electrical signals should, however, be disabled when operating valves manually, as pushing in the opposite direction to the solenoid can cause the coil to burn out.

Designers of a system can simplify maintenance by building in a fault-finding methodology from the start. This often takes the form of a flowchart. Figure 9.10 shows a typical system, which can be diagnosed by following the flowchart of Figure 9.11. Such charts cannot solve every problem, but can assist with the majority of common faults. If transducers can be fitted to allow the system to be monitored by a computer or programmable controller, Figure 9.11 could form the basis of a computer-based expert system.

PREVENTIVE MAINTENANCE

Many production people think a maintenance department exists purely to repair faults as they occur (the common image being a team sitting in the workshop waiting for the phone to ring). The most important part of a maintenance department's responsibility, however, is performing routine planned maintenance. This provides regular servicing of equipment, checks for correct operation and identifies potential faults – which can be corrected before they interrupt production. A personal analogy is the 6000 mile service for motor cars. As an often overlooked side benefit, planned maintenance trains the maintenance craftsmen in the operation and layout of the plant for which they are responsible.

A planned maintenance schedule can be done on a calendar basis (work done daily, weekly, monthly and so on) or on an operation-based schedule (work done after so many hours operation, or so many cycles) with time run or number of cycles recorded by control equipment. Different parts of the system may have differing maintenance schedules. Identifying what work needs to be done and the basis of the schedule for each item is the art of planned maintenance. It depends heavily on the nature of the plant; air filters in a dust-filled steel works, say, require checking more often than in a clean food factory.

With the advent of the desktop personal computer many excellent computer-based maintenance planning programs are available. These produce fully detailed work schedules on a shift-by-shift basis, and flag urgent work. The user still, however, has to specify the work to be done and the basis of schedules.

In hydraulic systems it is generally thought that oil problems (level in the tank, contamination by dirt, air or water) are responsible for around three-quarters of faults. Regular checks on oil condition and level are therefore of utmost importance. Any sudden change in level should be investigated.

Oil temperature should also be checked regularly. High temperatures arise from heat produced by flow discharging with a high pressure drop. Apart from the obvious possible fault with a heat exchanger (no water flow, for example) other possible causes are incorrect operation of relief or unloading valves (i.e. the pump on load continuously), internal leakage or too high a fluid viscosity.

System pressure should be recorded and checked against design values. Deviations can indicate maladjustment or potential faults. Too high a pressure setting wastes energy and shortens operational life. Too low a pressure setting may cause relief valves to operate at pressures below that needed by actuators, leading to no movement. Pressure deviation can also indicate developing faults outside the system. The fouling of a component moved by an actuator, for example, may cause a rise of pressure which can be observed before a failure occurs.

Motor currents drawn by pumps and compressors should also be checked both in working and unloading states (ideally, indication of motor currents should be available on a panel local to the motor). Changes in current can indicate a motor is working harder (or less) than normal.

Filters are of prime importance in both hydraulic and pneumatic systems. The state of most hydraulic filters is shown by a differential pressure indicator connected across the filter element. Obviously filters should be changed before they become blocked. Inlet air filters on pneumatic systems also need regular cleaning (but *not* with flammable fluids such as petrol or paraffin). A record should be kept of filter changes.

Many checks are simple and require no special tools or instruments. Visual checks should be made for leaks in hydraulic systems (air leaks in pneumatic systems generally can be detected from the noise they make!). Pipe runs and hosing should be visually checked for impact damage and to ensure all supports are intact and secure. Connections subject to vibration should be examined for tightness and strain. It is not unknown for devices such as pumps and compressors to 'walk' across the floor dragging their piping with them.

Where the device examined follows a sequence, the operation should be checked to ensure all ancillary devices, such as limit switches, are operating. The time to perform sequences may be worth recording as a lengthening of sequence times may indicate a possible developing fault due to, say, leakage in a cylinder.

Actuators have their own maintenance requirements given in manufacturers' manuals. Seals and bushing in cylinders, for example, require regular checking and replacement if damaged. Cylinder rods should be examined for score marks which

can indicate dust ingress. Actuators which move infrequently under normal duty can be operated to check they still work (and also to help lubricate the seals).

Treat leaks from around the rods of cylinders with urgency. If oil is leaking out round the neck seal on the extend stroke, dirt is being drawn into the system on the return stroke and a minor leak can soon turn into a major system failure.

Pneumatic preventive maintenance is very similar to hydraulic maintenance (although obviously there is no hydraulic oil to check). Other points such as piping, filters, fittings, sequences and so on need checking in the same way.

Compressors have their own maintenance requirements. Many are belt-driven, and require belt condition and tension to be checked at regular intervals. Crankcase oil level and the air breather should also be checked.

The compressor is normally sized for the original capacity plus some reserve for future additions. A compressor will thus start life on a low-duty cycle, which increases as further loads are added. When compressor capacity is reached, the compressor will be on a 100% duty cycle. Any additional load results in a fall of system pressure in the receiver. Leaks also cause a rise in compressor duty cycle, as will any loss of compressor efficiency. Duty cycle of the compressor thus gives a good indication of the health and reserve capabilities of the systems.

Compressor efficiency is determined largely by the condition of valves, piston rings and similar components subject to friction wear. These should be examined at intervals given in manufacturers' instruction manuals.

Other common pneumatic maintenance checks are validation of safety valve operation on the receiver, replenishment of oil in the air lubrication and drainage of water from air dryers.

COMPUTER SIMULATION

Computer simulations are becoming increasingly important for the design of fluid-based systems and for training purposes. All the illustrations in this section have been made using Automation Studio, a sophisticated software package from the Canadian company Famic Technology, which can simulate systems based on hydraulics, pneumatics, electrical, digital and PLC control.

Simulations are designed with a simple drag-and-drop procedure. Components are selected from a drop-down library list, placed on the screen then connected with a mouse, as shown in Figure 9.12.

For a training aid, default values can be chosen for cylinders, valves and pumps, etc. For serious design work all components can be individually specified along with the diameters and lengths of piping. Figure 9.13 shows the specification of some of the design parameters for a cylinder. In addition to the parameters shown, the angle of the cylinder, frictional forces, load inertia and mass, and other detailed characteristics can be entered.

With the system designed on screen it can be tested. The animated operation of the circuit is displayed along with flows and pressures at selected points. Trending graphs can be added as in Figure 9.14 to allow transients and timings to be observed.
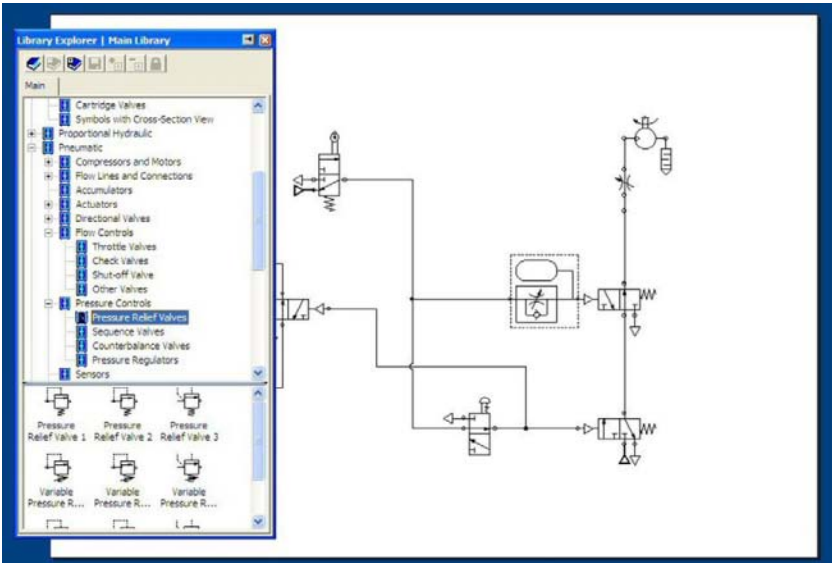
FIGURE 9.12 Building a simulation. Components are selected from the drop-down library and placed on the screen
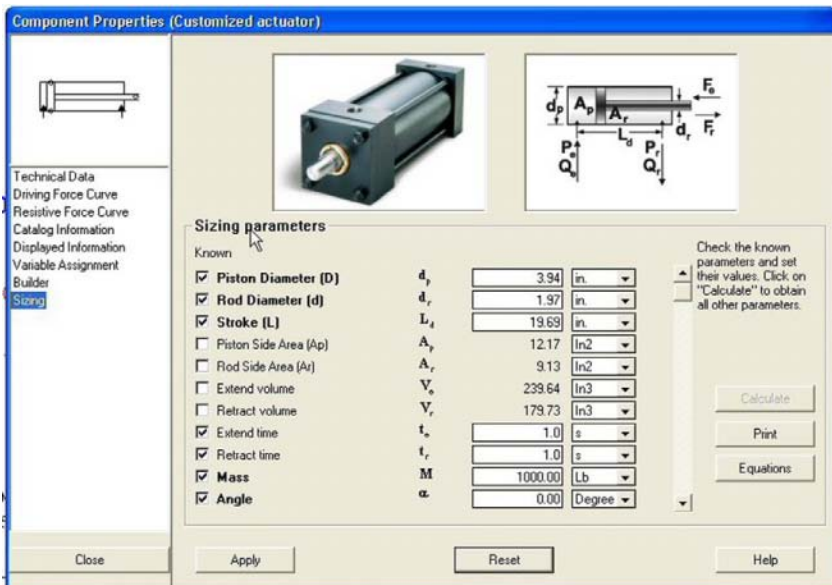
FIGURE 9.13 Part of the detailed specification of a cylinder to be used in a design
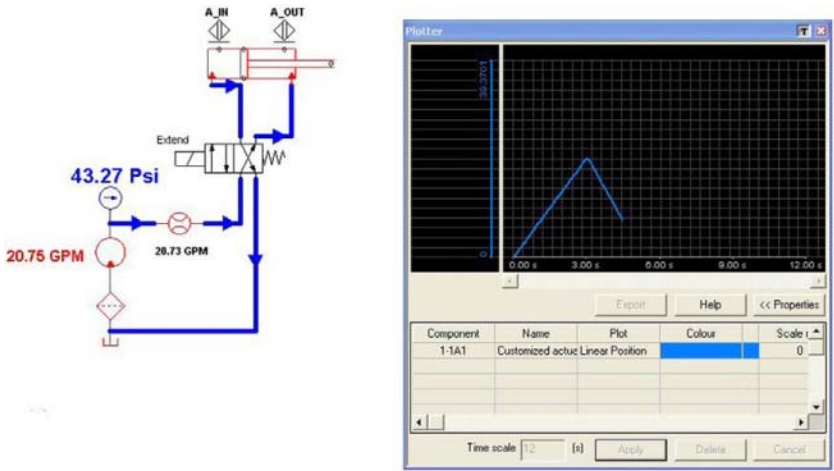
FIGURE 9.14 An animated display of a circuit with trending graph of cylinder position
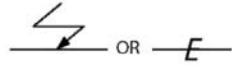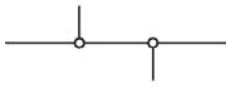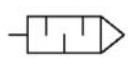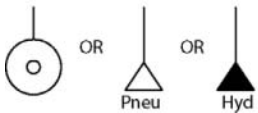
Appendix

HYDRAULIC AND PNEUMATIC SYMBOLS

Like electrical circuit diagrams, graphical representations of hydraulic and pneumatic systems are a shorthand way of representing the operation of a circuit by using standard graphical symbols. Although the diagrams show the relationship between components they are not a piping diagram.

Hydraulics and pneumatics use the same general symbols, the difference being that energy triangles, found on pumps and motors for example, are filled black on hydraulic diagrams and left clear on pneumatic diagrams.

The symbols listed below are generally in accordance with ISO 1219, although manufacturers do tend to use variations on this theme.

Main	Reservoir (tank)
Pilot	Pressurized Tank
Drain	Return line above oil level
Enclosure around multiple units, e.g. an operating unit	Return line below oil level
Electrical Line (OR)	Hydraulic Accumulator
Flexible Hose	Pneumatic Exhaust (Tapped, Untapped)
Connection Points	Silencer
Crossing Lines	Reclassifier (Filter/Silencer)
Pressure Source (OR Pneu OR Hyd)	

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Symbol	Description
	Fixed Displacement Pump
	Fixed Displacement Motor (uni-directional)
	Variable Displacement Pump
	Variable Displacement Motor (uni-directional)
	Pump with two flow directions
	Bi-directional Motor
	Hand Pump
	Rotary Actuator (limited angle of rotation)
M	Electric Motor
	Internal Combustion engine (e.g. petrol)
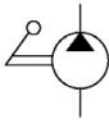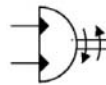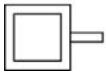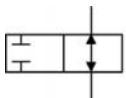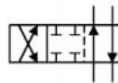
Symbol	Description
	2 port, 2 position Normally Open
	4 port, 2 position DCV (all ports blocked)
	2 port, 2 position Normally Closed
	4 port, 3 position DCV All ports blocked in center
	3 port, two position Normally Open
	4 port, 3 position DCV Pump unloads to tank in center
	3 port, two position Normally Closed
	5 port, 3 position DCV all ports blocked in center
	4 port, two position Directional Control
	4 port proportional (throttling) DCV with center off position
	5 port, two position Directional Control
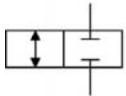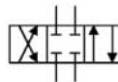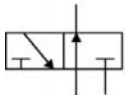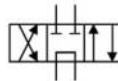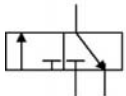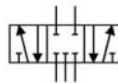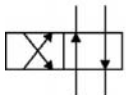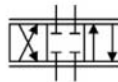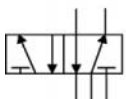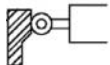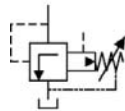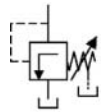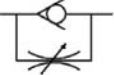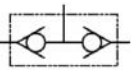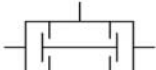
Pushbutton	Direct Acting Solenoid
Pull-button	Spring (usually for return)
Manual Lever	Detent (holds position)
Foot Pedal	Pneumatically Operated
Plunger	Hydraulically Operated
Cam or Limit Struck in rest state	Pneumatic Pilot
Roller (Cam or Limit) Unstruck in rest state	Hydraulic Pilot
One-Way Limit Arrow shows active direction	Solenoid with Hydraulic Pilot
Representation of limit on layout drawing	Proportional Solenoid with Spool Position Feedback
Representation of One-Way Limit on layout drawing	Combined Solenoid and Spring, e.g. spring return to center

Pressure Relief Valve	Throttle
	Adjustable Throttle
Pilot Operated Pressure Relief Valve	One-Way Adjustable Throttle
Sequence Valve	Shuttle Valve (OR Gate)
Check Valve	AND Gate
Spring Assisted Check Valve	
Pilot Assisted Check Valve	Shut-Off Valve

	Single Acting Cylinder, external return (e.g. gravity)
	Single Acting Cylinder spring return
	Double Acting Cylinder
	Double Acting Cylinder with dual rod
	Rodless Cylinder (e.g. magnetic coupling)

	Double Acting Cylinder with return cushioning
	Double Acting Cylinder with cushioning on both directions
	Double Acting Cylinder with adjustable cushioning
	Double Acting Telescopic Cylinder

	Filter
	Manual Drain Valve
	Auto Drain Valve
	Filter with Auto Drain Valve
	Lubricator
	Dryer
	Cooler
	Heater

	Water Cooled Heat Exchanger
	Air Cooled Heat Exchanger
	Level Sensor
	Combination Service Unit Filter, Pressure Regulator, Lubricator
	Pressure Gauge
	Temperature Indicator
	Flow Meter

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