# 7

## *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

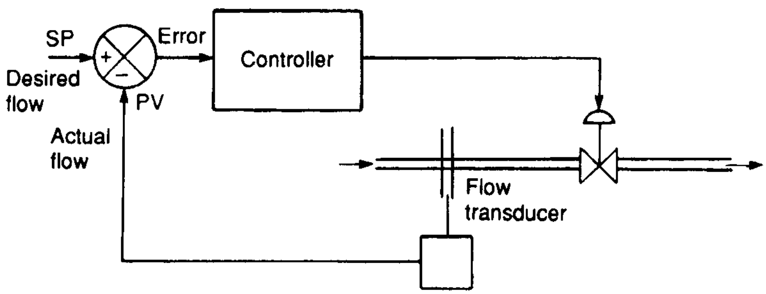



**Figure 7.1** *Closed loop control*

range 4 to 20 mA. If this represents a water flow from 0 to 1500 $1\ min^{-1}$, for example, a flow of $1000\ 1\ min^{-1}$ 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Perhaps surprisingly, pneumatic process control has by no means been superseded by electronic and microprocessor technology, so it is worth looking at the reasons for its popularity.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.

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 to 1.0 bar or the almost identical imperial equivalent 3 to 15 psig. If the water flow of 0 to 1500 $1\ min^{-1}$ is represented pneumatically, $01\ min^{-1}$ is shown by a pressure of 0.2 bar, $1500\ 1\ min^{-1}$ is 1.0 bar, while $10001\ min^{-1}$ 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 to 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 to 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

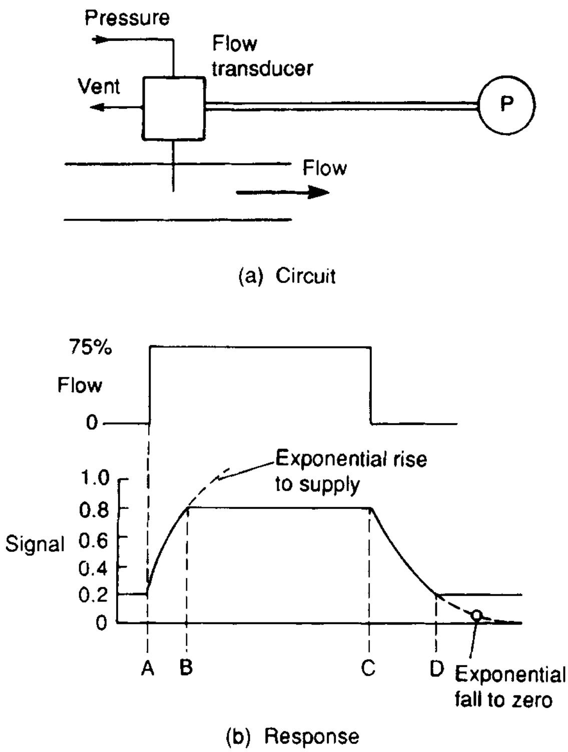



**Figure 7.2** *Advantage of an offset zero*

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 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 (330m $s^{-1}$). If signal lines are over a hundred metres 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 (eg 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 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 to 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.

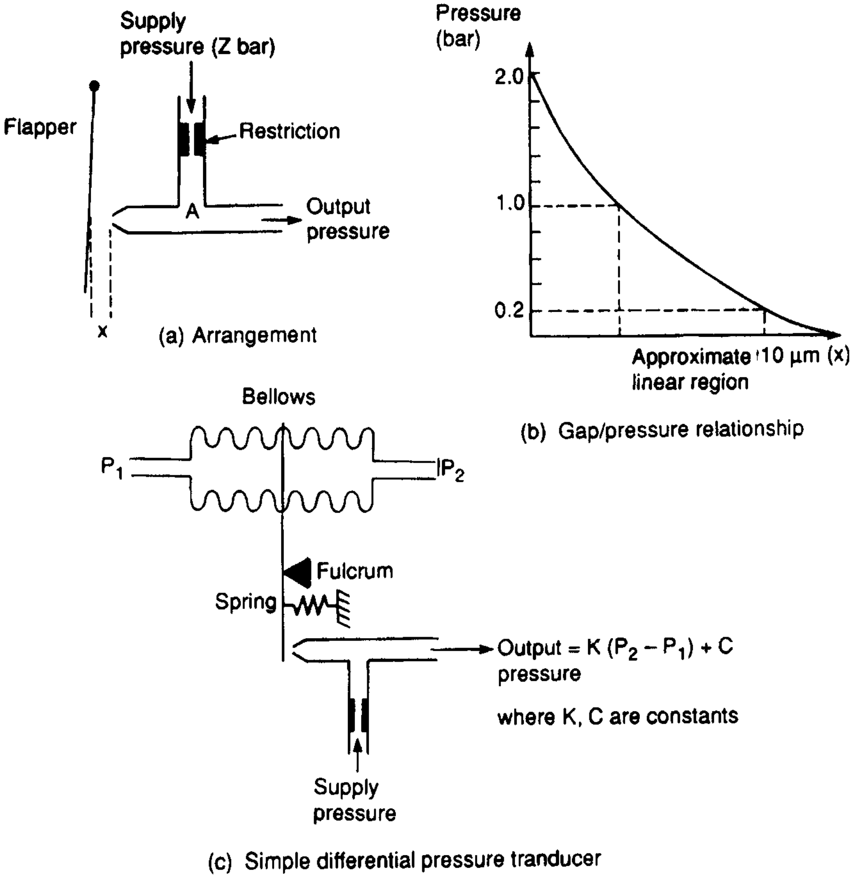



**Figure 7.3** *The flapper-nozzle, the basis of pneumatic process control*

Air loss from the jet (and hence pressure at A) is influenced by the gap x 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 linearised 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

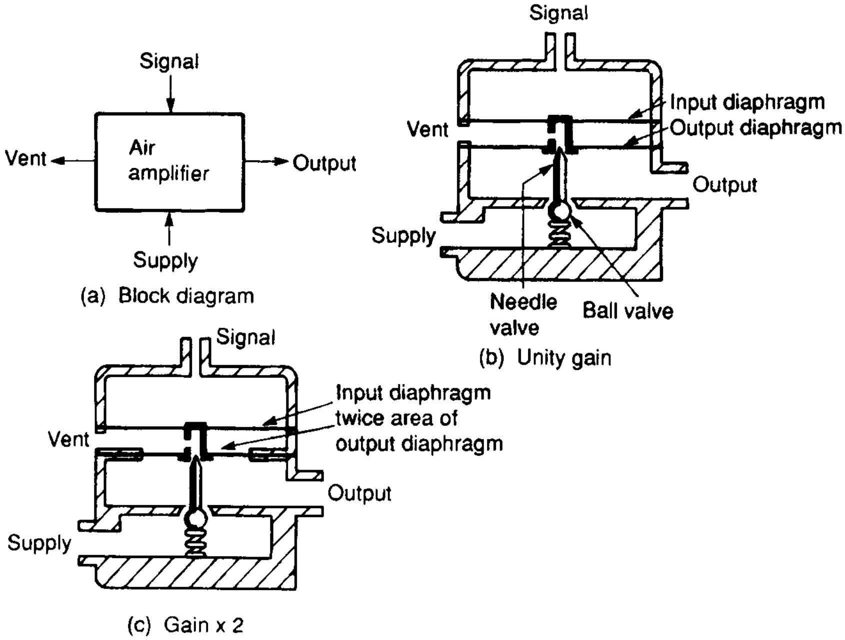



**Figure 7.4** *Volume boosters or air amplifiers*

constant output/input ratio. An amplifier with a gain of two, for example, turns a 0.2 to 1 bar signal range to a 0.4 to 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 output pressure to the lower diaphragm. If output pressure is lower than inlet pressure, the diaphragm is pushed down, closing needle valve and opening 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 stabilises 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 centre of the input signal range, there is no net flow to or from the output port.

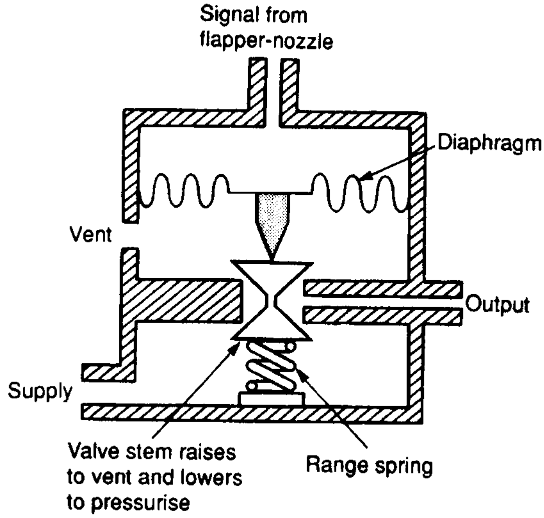



**Figure 7.5** *The air relay*

An air relay is used to linearise a flapper-nozzle, as shown in Figure 7.6. Here, force from the unbalance 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

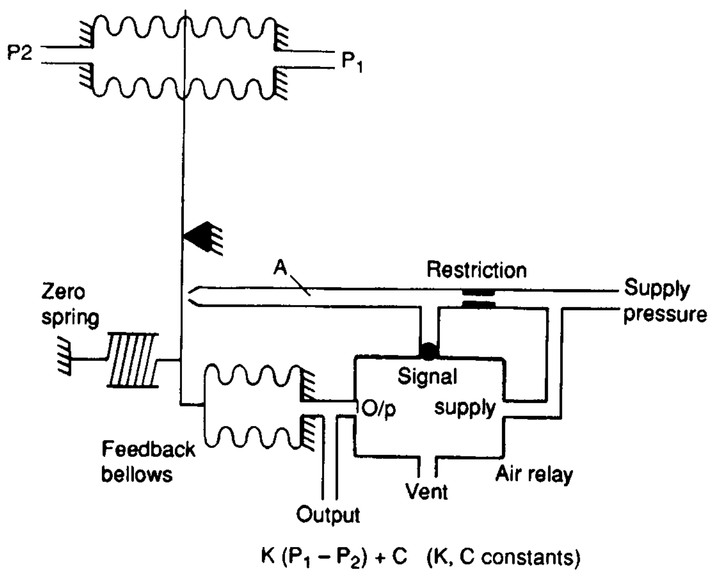



**Figure 7.6** *The force balance principle*

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 centre 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.

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:

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

where K is the gain.

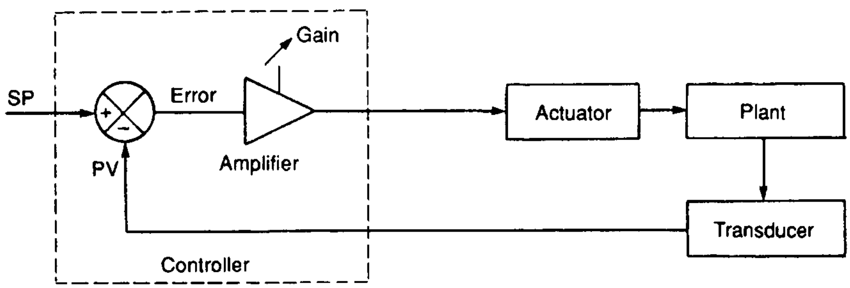



**Figure 7.7** *Proportional only controller*

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:

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

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 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

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

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.

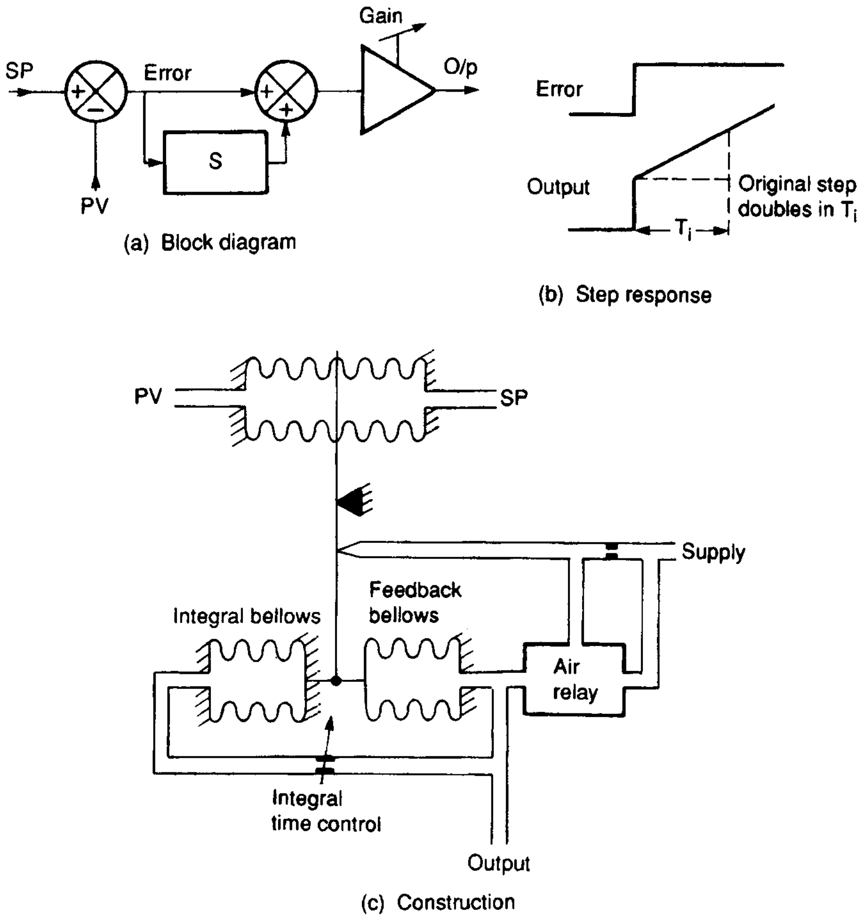



**Figure 7.8** *Proportional plus integral (P + I) controller*

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 7.10 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.

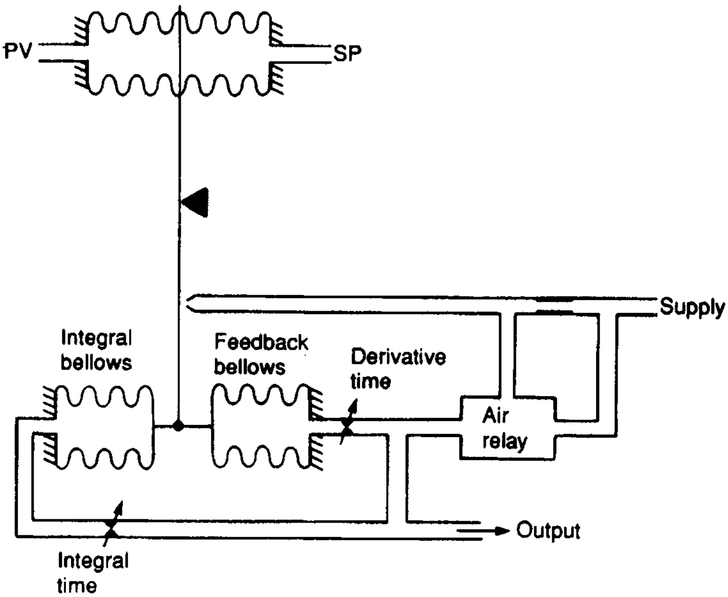



**Figure 7.9** *Three term (P + I + D) controller*

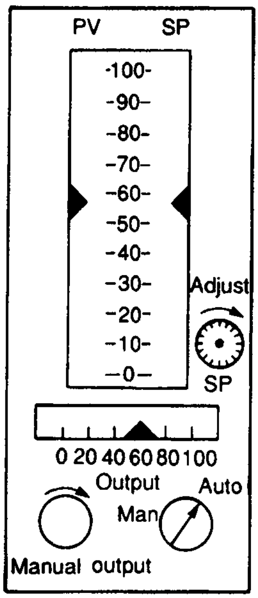



**Figure 7.10** *Front panel of a typical controller*

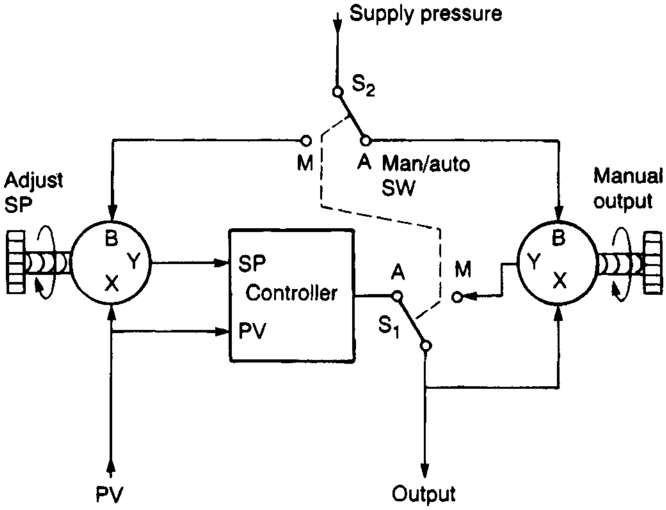



**Figure 7.11** *Internal arrangement giving bumpless transfer*

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 setpoint 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.

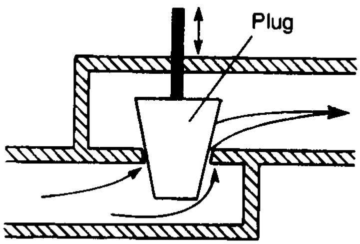
Even with totally electronic or computer-based process control schemes, most valves are pneumatically-operated. Although electrically-operated actuators *are* available, pneumatic devices tend to be cheaper, easier to maintain and have an inherent, and predictable, failure mode.

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.

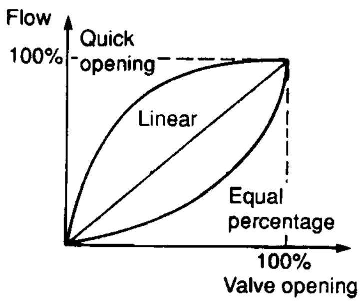
## Flow control valves

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 to 7.14. Of these 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 Fig. 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 approxi-




(a) Construction




(b) Valve characteristics (set by plug profile)

**Figure 7.12** *The plug valve*

mately 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.

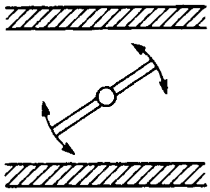


**Figure 7.13** *The butterfly valve*

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.

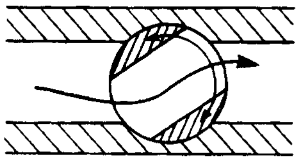


**Figure 7.14** *The ball valve*

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 7.5b, 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

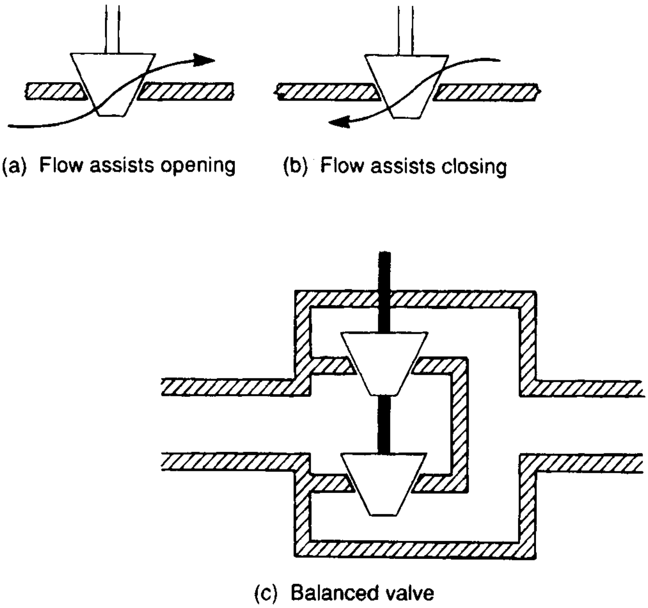



**Figure 7.15** *Dynamic forces acting on a valve*

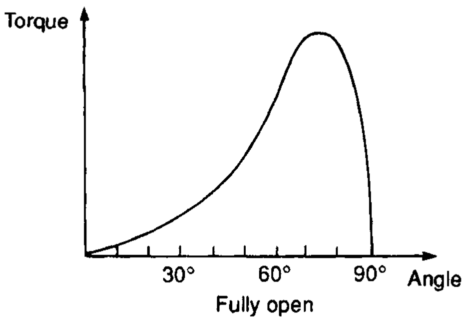



**Figure 7.16** *Torque on a butterfly valve*

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°.

### 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.

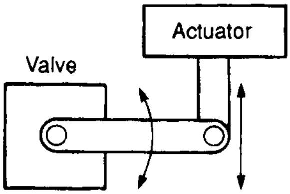



**Figure 7.17** *Conversion from linear actuator motion to rotary valve motion*

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 ($P \times A$) 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.

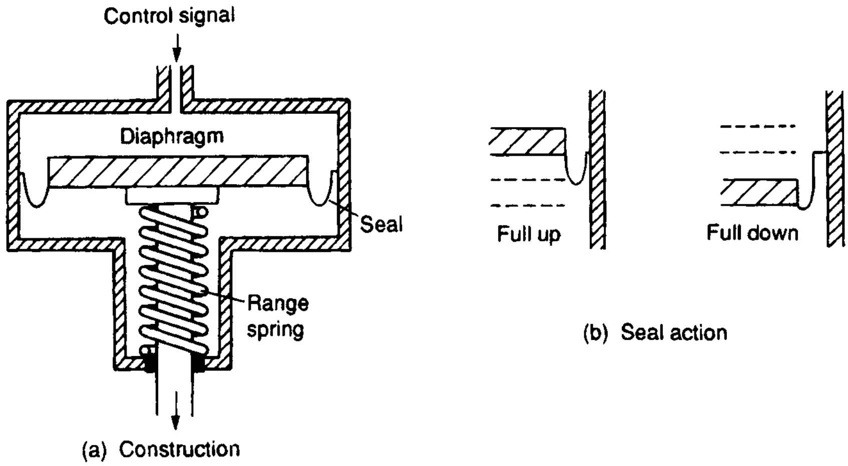



**Figure 7.18** *Fail-up actuator*

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.

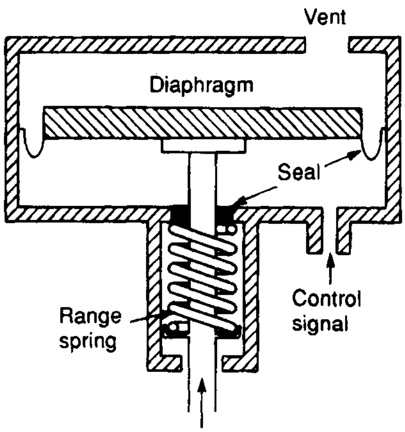



**Figure 7.19** *Fail-down actuator*

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.

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 (ie, 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.

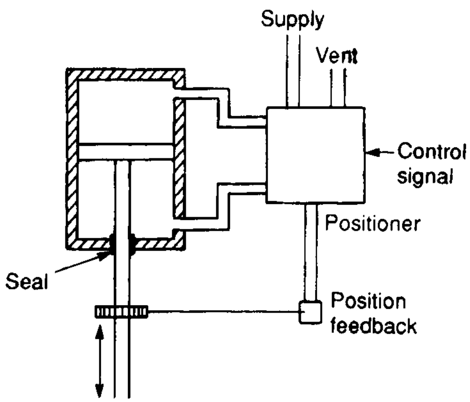



**Figure 7.20** *Double-acting cylinder (holds position on failure)*

### *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 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).

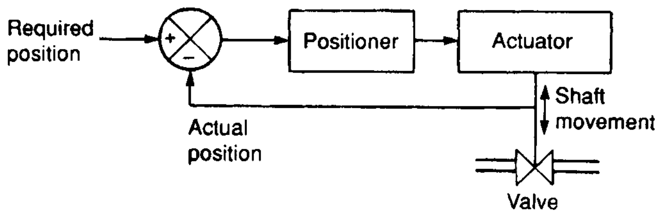



**Figure 7.21** *The valve positioner*

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,

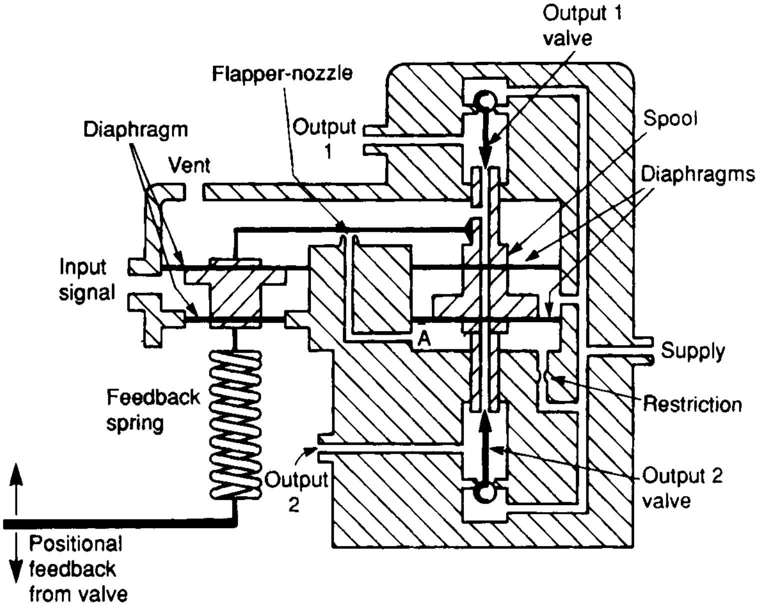



**Figure 7.22** *Force balance valve positioner*

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.

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 illus-

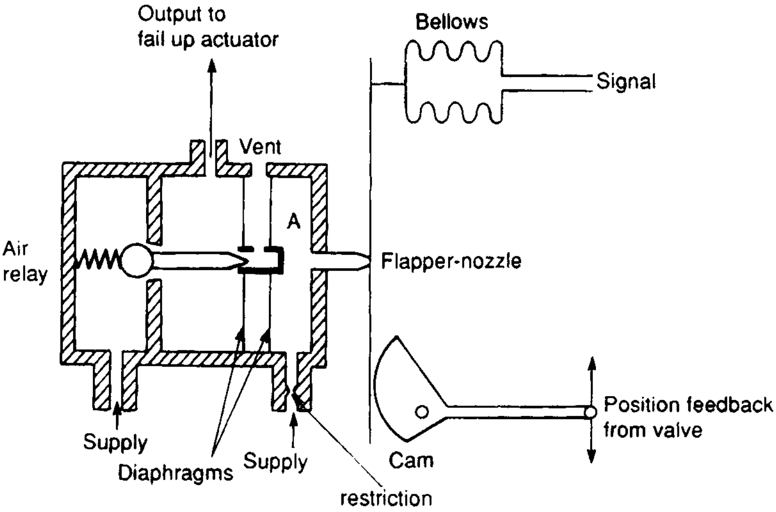



**Figure 7.23** *Motion balance positioner*

trated 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 .

## Converters

The most common process control arrangement is probably electronic controllers with pneumatic actuators and transducers. Devices are therefore needed to convert between electrical analog

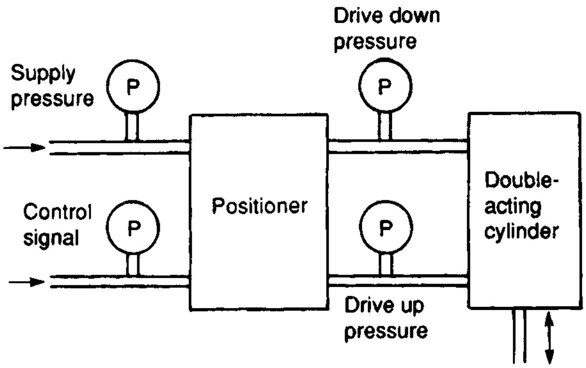



**Figure 7.24** *Pressure indication on a positioner for fault finding*

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 surprisingly, a P–I converter.

## *I–P converters*

Figure 7.25 illustrates a common form of I–P converter based on the familiar force balance principle and the flapper-nozzle. Electrical current is passed through the coil and results in a rotational displacement of the beam. The resulting 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 balance results when the force from the bellows (proportional to output pressure) equals the force from the coil (proportional to input electrical signal).

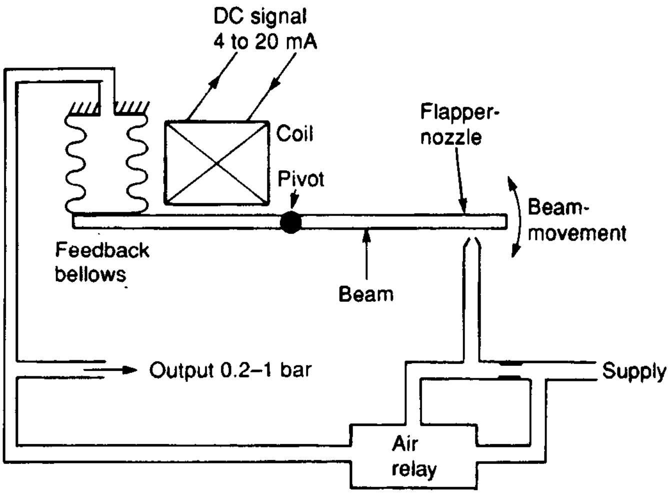



**Figure 7.25** *Current to pressure (I–P) converter*

## *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

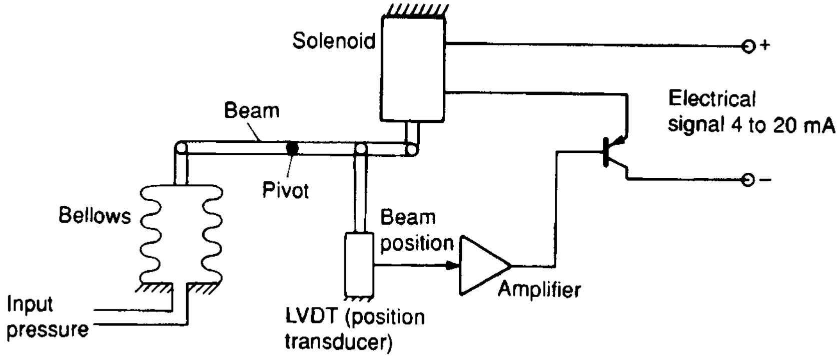



**Figure 7.26** *Pressure to current (P–I) converter*

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 bellows (proportional to input signal pressure).

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 (eg, 15 to 30 V). Often, the current signal of 4 to 20 mA is converted to a voltage signal (commonly in the range 1 to 5 V) with a simple series resistor.

## Sequencing applications

Process control pneumatics is also concerned with sequencing i.e. performing simple actions which follow each other in a simple order or with an order determined by sensors. Electrical equivalent circuits are formed with relays, solid state logic or programmable controllers.

A simple example of a pneumatic sequencing system is illustrated in Figure 7.27, where a piston oscillates continuously between two striker-operated limit switches $LS_1$ and $LS_2$. These shift the main valve $V_1$ with pilot pressure lines. The main valve spool has no spring return and remains in position until the opposite signal is applied. Shuttle valves $V_2$ and $V_3$ allow external signals to be applied via ports Y and Z.

Time is often used to control a sequence (eg, feed a component, wait five seconds, feed next component). A time delay valve is con-

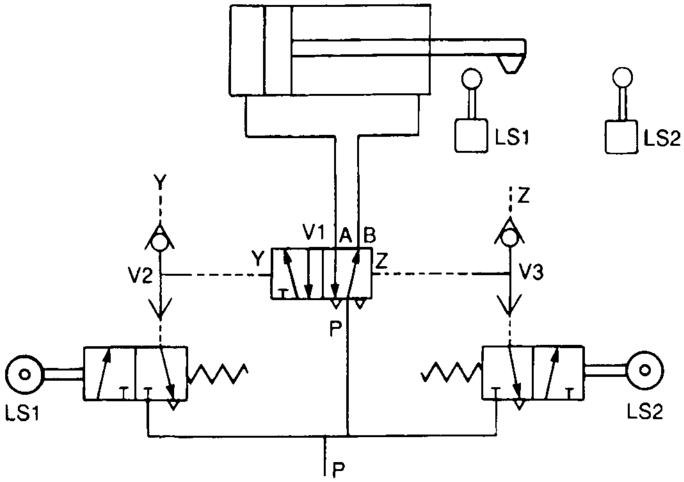



**Figure 7.27** *A sequencing example; the cylinder oscillates between LS1 and LS2*

structed as illustrated in Figure 7.28a. Input signal X is a pilot signal moving the spool in main valve $V_1$, but it is delayed by the restriction valve and the small reservoir volume V. When X is applied, pilot pressure Y rises exponentially giving a delay T before the pilot operating pressure is reached. When X is removed, the non-return valve quickly vents the reservoir giving a negligible off delay. Figure 7.28b shows the response. As shown, the valve is a delay-on valve. If the non-return valve is reversed delay-off action is achieved.

Sequencing valves are used to tie pressure-controlled operations together. These act somewhat like a pilot-operated valve, but the

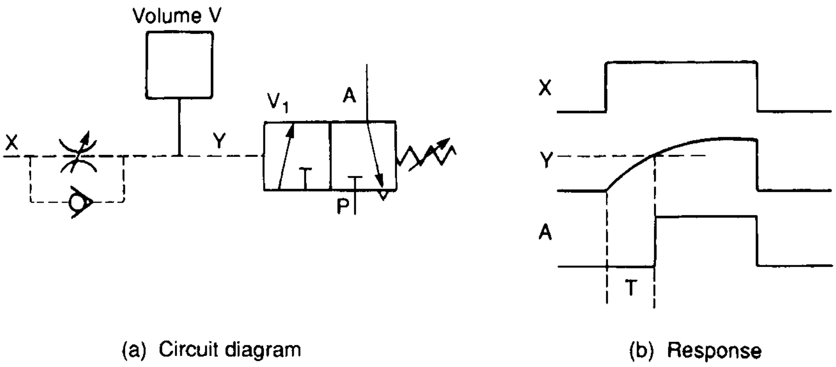



**Figure 7.28** *The time delay (see also Figure 4.28 for construc-*

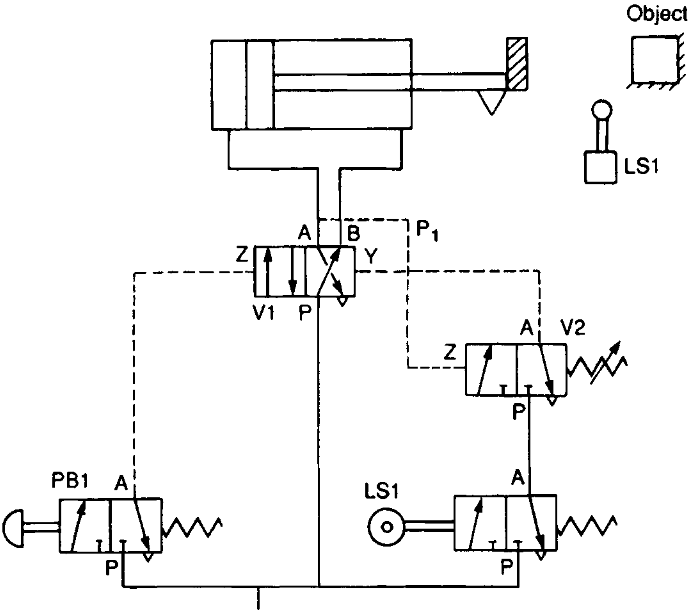



**Figure 7.29** *Sequencing valve application*

designer can control the pressure at which the valve operates. A typical application is shown in Figure 7.29 where a cylinder is required to give a certain force to an object. Valve $V_2$ is the sequence valve and operates at a pressure set by the spring. The sequence is started by pushbutton $PB_1$, which shifts the pilot spool in the main valve $V_1$ causing the cylinder to extend. When the cylinder reaches full extension, limit switch $LS_1$ operates and pressure $P_1$ starts to rise. When the preset pressure is reached sequence valve $V_2$ operates, moving the spool in main valve $V_1$ and retracting the cylinder.

The two applications given so far have used limit switch operated valves to control sequences. Pneumatic proximity sensors can also be used. The reflex sensor of Figure 7.30 uses an annular nozzle jet of air the action of which removes air from the centre 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 (Figure 7.31) 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

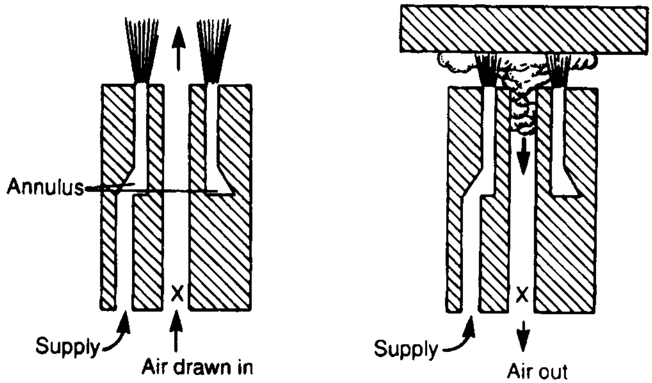



**Figure 7.30** *Reflex proximity switch*

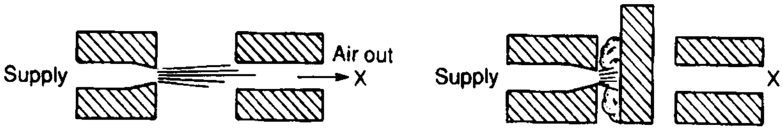



**Figure 7.31** *Interruptible jet limit switch*

pressure rise at signal output X. An object blocking this flow, causes X to fall to atmospheric pressure.

With both types of sensor, air consumption can be a problem. To reduce air usage, low pressure and low flow rates are used. Both of these results in a low pressure signal at X which requires pressure amplification or low pressure pilot valves before it can be used to control full pressure lines.

Logic devices (AND, OR gates and memories) are part of the electrical tool kit for sequencing applications. The pneumatic equivalent (Figure 7.32) uses the wall attachment or Coanda effect. 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 (Figure 7.32a).

This principle is used to give a pneumatic set/reset (S-R) flip-flop memory in Figure 7.32b. 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 7.32c shows a fluidic OR/NOR gate. A small bias pressure keeps the signal on the right-hand wall, which causes it to exit via

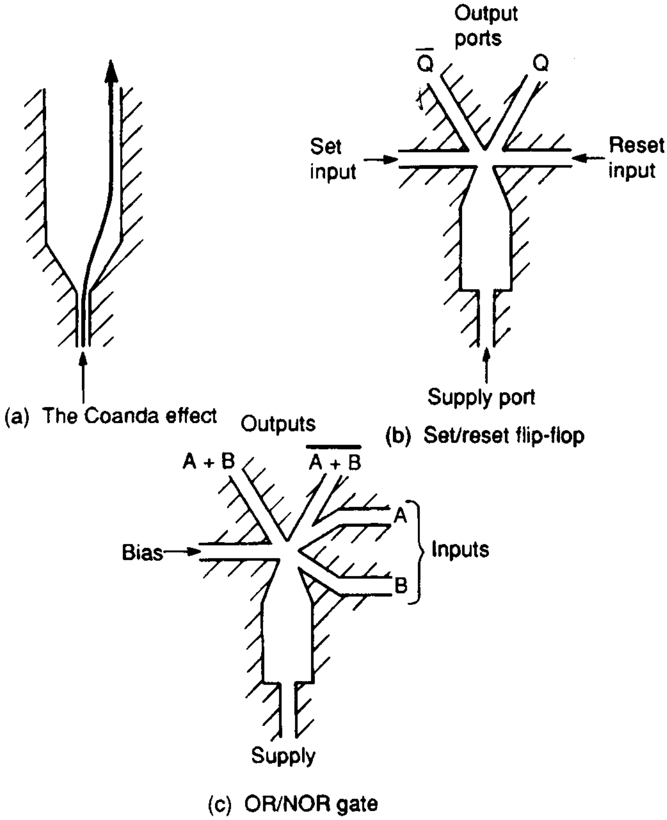



Figure 7.32 Fluidic logic

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.

Logic functions can also be performed by series connections of valves (to give the AND operation) shuttle valves (to give the OR operation) and pilot-operated spools (to give flip-flop memories). Valve $V_1$ in Figure 7.27, for example, acts as an S-R flip-flop memory.