6.4 Actuators: Pneumatic


Types:
A. Linear
   A1. Spring-and-diaphragm
   A2. Piston
B. Rotary
   B1. Cylinder with Scotch yoke
   B2. Cylinder with rack and pinion
   B3. Dual cylinders
   B4. Spline or helix
   B5. Vane
   B6. Pneumohydraulic
   B7. Air motor
   B8. Electropneumatic

Applicable to Valve Sizes:
A1. 0.5 to 8 in. (12 to 200 mm)
A2. 0.5 to 30 in. (12 to 750 mm)
B. 2 to 30 in. (50 to 750 mm)

Standard Spring Ranges:
A1. 3–9, 3–15, 9–15, 6–30 PSIG (20–60, 20–100, 60–100, 40–200 kPa)
A1. 60 PSIG (414 kPa); some higher
A2. 150 PSIG (1035 kPa); accessories may lower ratings

Max. Actuator Pressure Ratings:
B. 250 PSIG (1725 kPa)

Actuator Temperature Ratings:
A1 and A2. −20 to 150°F (−30 to 66°C); some higher
B. −40 to 200°F (−40 to 95°C); special up to 350°F (177°C)

Actuator Areas:
A1. 25 to 500 in.² (0.016 to 0.323 m²)
A2 and B. 10 to 600 in.² (0.006 to 0.38 m²); bore diameters from 2 to 44 in.
   (50 mm to 1.1 m) and strokes up to 24 in. (0.61 m)
6.4 Actuators: Pneumatic

Linear Thrust (Stem Force) Ranges:
A1. 200 to 45,000 lb \(f\) (22 to 10,100 N)
A2. 200 to 32,000 lb \(f\) (22 to 7,200 N); specials up to 186,000 lb \(f\) (41,800 N)

Speeds of Full Stroke:
A1. Small (25 to 100 in.) actuators, 1 to 5 sec.
A2. 0.33 to 24.0 in./sec (8 to 600 mm/sec) or less than 1 to 30 sec/stroke
B. 0.33 to 6.0 in./sec (8 to 150 mm/sec) or 1 to 30 sec/stroke. Dual pistons (Figure 6.4aa) can stroke large rotary valves in 0.5 sec.

For the same size units, spring-and-diaphragm units are the slowest, spring-returned pistons are faster, and double-acting pistons with supply/exhaust at both ends are the fastest

Torque Ranges:
B. 10 to 100,000 ft \(\cdot\) lb \(f\) (0.69 to 6850 N \(\cdot\) m); special units for 5 million ft \(\cdot\) lb \(f\) (343,000 N \(\cdot\) m) have been built

Cost:
Included in control valve cost

Hysteresis, Dead Band, and Linearity:
A and B. Generally within 2%, but when spring-and-diaphragm or rotary piston actuators are operating rotary valves, the linearity is worse

Partial List of Suppliers:
ABB Kent-Introl (www.abb.com)
Actuation Valve & Control Ltd. (www.actuation.co.uk)
Allenair Corp. (www.allenair.com)
Aluma Actuators Inc. (www.aluma-usa.com)
Arca Regler Gmbh (www.arca-valve.com)
Bardiani Valvole SpA (www.bardiani.com)
Bray Valve & Controls (www.bray.com)
Cashco Control Valves & Regulators (www.cashco.com)
China Zhejiang Chaoda Valve Co. Ltd (www.chinavalve.com)
Circor International, Inc. (Leslie Controls) (www.circor.com)
Combraco Industries Inc. (www.combraco.com)
Control Components Inc. (Bailey, CCI, BTG, STI, McCanna, Limitorque) (www.ccivalve.com)
Controlmatics Industrial Products (www.controlmatics.com)
Dresser (Leeden, Masoneilan) (www.dresser.com)
Emerson Process Management (Contex, Fisher, El-O-Matic, Shafer Valve) (www.emersonprocess.com)
Flo-Tork, Inc. (www.flo-tork.com)
Jordan Valve (www.jordanvalve.com)
Koso America Inc. (Hammeldahl) (www.rexa.com)
K-Tork International Inc. (www.ktork.com)
Larox Flowsys Inc. (www.larox.fi)
Nihon Koso (www.koso.co.jp)
Metso Automation USA, Inc. (Neles-Jamesbury, Valmet Automation) (www.metsaoautomation.com)
Mumatics Inc. (www.mumatics.com)
Norrisal Controls (www.norrisal.com)
Parcol SpA (www.parcol.com)
Red Valve Co. Inc. (www.redvalve.com)
Rotork Controls Inc. (www.rotork.com)
Samson Regeltchnick.vb (www.samsom-regeltechnick.nl)
Severn Glocon Ltd. (www.severnglocon.com)
Spirax Sarco Inc. www.spiraxsarco-usa.com
SPX Valves & Controls (Copes Vulcan, Dezurik) (www.spxvalves.com)
Tyco Flow Control Div. (Keystone, Morin, Biffi, Descoti, Sempell, Yarway, Grinnell, MCF) (www.tycovalves-na.com)
Welland & Tuxhorn GmbH (www.welland-tuxhorn.de)
Wier Group plc (Atwood & Morrell, Batley Valve, Blakeborough Controls, Hopkinson, Sebim, Flowguard) (www.wiervalve.com)
Xomax Corp. (www.xomax.com)
Yamatake Corporation (www.yamatake.co.jp)
INTRODUCTION

Pneumatic valve actuators respond to an air signal by moving the valve trim into a corresponding throttling position. This section covers the two basic designs most frequently utilized: the diaphragm and the piston actuator. The discussion of diaphragm- and piston-type actuators is followed by the treatment of pneumatic-rotary and pneumatic-hydraulic actuators.

In connection with the performance of these actuators, an analysis is presented of the various forces positioning the plug, including diaphragm, spring, and dynamic forces generated by the process fluid. An understanding of the interrelationships among these forces will allow the reader to properly size these actuators and make the correct spring selection.

The failure safety of valve actuators and the relative merits of diaphragm vs. piston actuators and the topic of high-speed actuation using pneumatics is also discussed. This section is concluded with a summary of the results of a long-term evaluation of the performance of pneumatically actuated control valves in the field.

DEFINITIONS

An actuator is that portion of a valve that responds to the applied signal and causes the motion resulting in modification of fluid flow. Thus, an actuator is any device that causes the valve stem to move. It may be a manually positioned device, such as a handwheel or lever. The manual actuator may be open-closed, or it may be manually positioned at any position between fully open and fully closed. Other actuators are operated by compressed air, hydraulics, and electricity.

The actuators discussed here are those capable of moving the valve to any position from fully closed to fully open and those using compressed air for power. Of such there are two general types: the spring-and-diaphragm actuator and the piston actuator.

In a spring-and-diaphragm actuator, variable air pressure is applied to a flexible diaphragm to oppose a spring. The combination of diaphragm and spring forces acts to balance the fluid forces on the valve.

In a piston actuator, a combination of fixed and variable air pressures is applied to a piston in a cylinder to balance the fluid forces on the valve. Sometimes springs are used, usually to assist valve closure. Excluding springs, there are two variations of piston actuators: cushion loaded and double acting.

In the cushion-loaded type, a fixed air pressure, known as the cushion pressure, is opposed by a variable air pressure and is used to balance the fluid forces on the valve. In the double-acting type, two opposing variable air pressures are used to balance the fluid forces on the valve.

An actuator can be said to have two basic functions: (1) to respond to the external signal of a controller and cause an inner valve to move accordingly (with the proper selection and assembly of components, other functions can also be obtained, such as a desired fail-safe action) and (2) to provide a convenient support for valve accessory items, such as positioners, limit switches, solenoid valves, and local controllers.

ACTUATOR FEATURES AND SELECTION

Table 6.4a describes the applications and relative advantages of a variety of actuator designs. The table lists both the advantages and the limitations of the various designs. The popularity of the spring/diaphragm actuator is due to its low cost, its relatively high thrust at low air supply pressures, and its availability with “fail-safe” springs. By trapping the pressure in the diaphragm case, it can also be locked in its last position. It is available in springless designs, double diaphragm designs (for higher pressures), rolling diaphragm designs (for longer strokes), and tandem designs (for more thrust).

One of the limitations of this design is the lack of actuator “stiffness” (resistance to rapidly varying hydraulic forces - for example, those caused by flashing). For such applications double-acting piston actuators are used and for extraordinary requirements hydraulic or electromechanical (motor gear) actuators may be preferred. A stiffer spring, 6–30 PSIG (41–210 kPa), in a spring/diaphragm unit is sometimes sufficient to correct the problem.

Linear piston actuators provide longer strokes and can operate at higher air pressures than can the spring/diaphragm actuators. When used to operate rotary valves, the linear piston or spring/diaphragm actuator does not provide a constant ratio of rotation per unit change in air signal pressure; therefore, the use of positioners is always a requirement.

Rotary piston actuators operate at higher air pressure and can provide higher torques, suitable for throttling large ball or butterfly valves. The double-acting version of this actuator does not have a positive failure position, but this can be corrected by extending the piston case and inserting a helical spring. For higher torques (over 1000 ft · lbf [68 Nm]), heavy-duty transfer linkages are required (Scotch yoke or rack and pinion); such units cannot be disassembled and maintained in the field. These actuators also require positioners because the relationship between air signal change and resulting rotation is not linear.

SPRING/DIAPHRAGM ACTUATORS

This discussion is restricted to pneumatic actuators. The external signal, therefore, is an air signal of varying pressure. The air signal range from a pneumatic controller is commonly 0–18 PSIG (0–124 kPa). Signal or actuator input pressure starts at 0 PSIG, not 3 PSIG (21 kPa).

A common mistake is to confuse the 3–15 PSIG (21–104 kPa) range of transmitter output pressure with the signal to a valve. The higher value of 18 PSIG (124 kPa) is fixed only by the air supply to the controller (or positioner),
Actuators: Pneumatic

6.4

and it can easily be set to 20 PSIG (138 kPa) or higher. A variety of other input pressures are used, such as 0–30 or 0–60 PSIG (0–207 or 0–414 kPa).

Both the spring-and-diaphragm and the piston actuator produce linear motion to move the valve. These actuators are ideal for use on valves requiring linear travel, such as globe valves. A linkage or other form of linear-to-rotary motion conversion is required to adapt these actuators to rotary valves, such as the butterfly type.

Steady-State Force Balance

In spring-and-diaphragm actuators the stem positioning is achieved by a balance of forces acting on the stem. These forces are caused by the pressure on the diaphragm, spring travel, rubbing friction, and fluid forces on the valve plug (Figure 6.4b).

Equation 6.4(1) can be derived from a summation of forces on the valve plug adopting the positive direction downward.

\[ PA - KX - P_a A_e = 0 \]

where \( A \) is the effective diaphragm area, \( A_e \) is the effective inner valve area, \( K \) is the spring rate, \( P \) is the diaphragm pressure, \( P_a \) is the valve pressure drop, and \( X \) is stem travel. Equation 6.4(1) applies to a push-down-to-close actuator and valve combination with flow under the plug. This type of actuator is commonly referred to as direct acting.

<table>
<thead>
<tr>
<th>Type of Actuator</th>
<th>Advantage</th>
<th>Disadvantage</th>
<th>Application</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linear spring-and-diaphragm</td>
<td>Low cost</td>
<td>Slow speed</td>
<td>Linear valves 1/2–8 in. (12–200 mm) body size</td>
</tr>
<tr>
<td></td>
<td>Mechanical fail-safe</td>
<td>Moderate thrust</td>
<td>Moderate thrust</td>
</tr>
<tr>
<td></td>
<td>Small package</td>
<td>Simple design</td>
<td>Small package</td>
</tr>
<tr>
<td></td>
<td>Excellent control with or without control devices</td>
<td></td>
<td>Excellent control with or without control devices</td>
</tr>
<tr>
<td>Linear piston</td>
<td>Moderate cost</td>
<td>Large spring compression</td>
<td>Linear valves 1/2–30 in. (12–760 mm) body size</td>
</tr>
<tr>
<td></td>
<td>Moderate thrust</td>
<td>when used for failure</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Small package</td>
<td>Simple design</td>
<td>Simple design</td>
</tr>
<tr>
<td></td>
<td>Excellent control with control device</td>
<td></td>
<td>Excellent control with control device</td>
</tr>
<tr>
<td></td>
<td>Long stroke</td>
<td>High-speed options</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Moderate stiffness</td>
<td></td>
</tr>
<tr>
<td>Rotary spring-and-diaphragm</td>
<td>Moderate cost</td>
<td>Low thrust in spring cycle</td>
<td>Rotary valves 1–6 inch (25–150 mm) body size</td>
</tr>
<tr>
<td></td>
<td>Mechanical fail-safe</td>
<td>Instability</td>
<td>Instability</td>
</tr>
<tr>
<td></td>
<td>Small package</td>
<td>Simple design</td>
<td>Simple design</td>
</tr>
<tr>
<td></td>
<td>Easily reversible</td>
<td>Excellent control with control device</td>
<td>Excellent control with control device</td>
</tr>
<tr>
<td>Rotary piston</td>
<td>Low cost</td>
<td>Slow speed</td>
<td>Rotary valves 1–24 inch (25–600 mm) body size</td>
</tr>
<tr>
<td></td>
<td>Moderate thrust</td>
<td>Large spring compression</td>
<td>Large spring compression</td>
</tr>
<tr>
<td></td>
<td>Small or large package</td>
<td>Good control with control device</td>
<td>Good control with control device</td>
</tr>
<tr>
<td></td>
<td>Mechanical fail-safe option</td>
<td></td>
<td>Mechanical fail-safe option</td>
</tr>
</tbody>
</table>

See Section 4.3 for a detailed treatment of the features of hydraulic and electric actuators.
Another popular actuator configuration is one causing the stem to rise on an increase of air pressure. It is commonly called a reverse acting actuator (Figure 6.4c).

By using the same sign convention, the force balance equation for this valve configuration is given in Equation 6.4(2):

$$-PA + KX - P_v A_v = 0$$  \hspace{1cm} 6.4(2)

If the flow direction is reversed in Figure 6.4b, the equation becomes:

$$PA - KX + P_v A_v = 0$$  \hspace{1cm} 6.4(3)

Likewise, reversing flow direction in Figure 6.4c results in Equation 6.4(4):

$$-PA + KX + P_v A_v = 0$$  \hspace{1cm} 6.4(4)

These equations are simplified because they do not consider friction and inertia. Friction occurs in the valve stem packing, in the actuator stem guide, and in the valve plug guide or guides. Usually, for static valve actuator sizing problems, negligible error is introduced by ignoring the friction terms.

If Equation 6.4(1) is plotted as signal pressure vs. stem travel and if the case of no fluid forces on the plug (bench test) is assumed, then the curve shown in Figure 6.4d is obtained.

Next, consider the case of plug forces due to fluid flow, assuming that the term $P_v$ is constant for all travel positions. This has the effect of shifting the straight line to the right to some position depending on the magnitude of $P_v$. Curves similar to those in Figure 6.4d can readily be drawn for the other valve configurations represented by Equations 6.4(2), 6.4(3), and 6.4(4). The distance between the lines is the force resulting from $A_v$ is the effective inner valve area, $K$ is the spring rate, $P$ is the diaphragm pressure, $P_v$ is the valve pressure drop, intersecting the abscissa is the pressure needed to move the valve plug.

**Actuator Sizing Example**

Let us assume that the forces acting on a 1 in. (25 mm) singleported globe valve are to be evaluated. In that case:

- $A = 46$ in.$^2$ (0.03 m$^2$)
- $X = 5/8$ in. (15.9 mm) full travel
- $K = 885$ ft lb/in. (7.83 N/mm)

If no plug forces exist, Equation 6.4(1) reduces to $PA = KX$. Solving for the pressure change required to obtain full travel from open to closed:

$$P = KX/A = (885)(5/8)/(46) = 12.03$ PSIG (83 kPa)

This is reasonably close to the 12 PSIG (0.83 kPa) desired operating span. Practical considerations of variations in spring constants and in actuator-effective areas usually prevent such a close approach to the desired span, and frequently a ±10% leeway is permitted.

When there are plug forces, it is seen from Equation 6.4(1) that an additional actuator force is required to maintain balance. The actuator pressure required to begin stem motion can be calculated for the case of a 1 in. diameter (25 mm) plug ($A_v = \pi/4$) and 100 PSIG (690 kPa) pressure drop. Equation 6.4(1) can be used to solve for $P$ as follows (stem travel is zero, thus there are no spring forces):

$$P = [(K = 885)(X = 0) + (P_v = 100)(A_v = \pi/4)]/(A = 46) = 1.7$ PSIG (11.8 kPa)  \hspace{1cm} 6.4(5)
This means that the diaphragm pressure must increase to 1.7 PSIG (11.8 kPa) before stem travel begins. This is the distance between the two sloping straight lines in Figure 6.4d.

**Actuator Nonlinearities**

In practice we encounter many nonlinearities, and the ideal curves in Figure 6.4d are not obtained. These nonlinearities are due to several factors, such as the variable effective diaphragm areas. The effective diaphragm area varies with travel and with the pressure level on the diaphragm. Figure 6.4e illustrates this for three different sizes of diaphragms.

Another source of nonlinearity is in the variation of the valve plug forces ($P_vA_v$). Figure 6.4f illustrates the variations in these plug forces for two 4 in. (100 mm), single- and double-ported valves. The figure also shows the effects of flow over and under the plugs of a single-ported valve.

Springs are also nonlinear in that the spring rates vary with travel. By judicious selection of springs, considering their spring rate and travel, the effects of their nonlinearity on the valve assembly can be minimized.

When all of these nonlinearities are considered, a plot of actuator travel vs. diaphragm pressure would not be a straight line as shown in Figure 6.4d, but might be a curve such as the one shown in Figure 6.4g.

A nonlinear curve, such as the one labeled “actual” in Figure 6.4g, is not necessarily objectionable. When used in an automatic control loop, the static nonlinearities are compensated for by the controller. This curve is actually a part of the gain term in the valve’s transfer function, and the other part is the flow characteristic. When a valve positioner is used, the positioner overcomes these nonlinearities, and the result is similar to the ideal curve shown in Figure 6.4g.

**Dynamic Performance of Actuators**

Several control valve subsystems must be analyzed in order to thoroughly evaluate their dynamic performance. The separate systems include:

1. The spring-mass system of the valve’s moving parts.
2. The pneumatic system from controller output to valve diaphragm chamber. If a valve positioner is used, there are two separate pneumatic systems: one from the controller output to the positioner and another from the...
positioner output to the diaphragm chamber. The interconnecting tubing is consideration in all of the pneumatic systems.

**Spring-Mass System Dynamics** Analysis of the spring-and-mass system is only valid for linear systems. It is necessary either to neglect consideration of the nonlinear elements or have a system wherein the nonlinear effects are minor. In the case of control valves with sufficient power in the actuator, the latter case is approached. With such an understanding of the nonlinear effects, we proceed as though valve actuators were linear devices.

The spring-mass system is represented by the following differential equation:

\[
M \frac{d^2X}{dt^2} + b \frac{dX}{dt} + KX = PA - PV,
\]

where \(b\) is the net friction force and \(M\) is mass. The net friction force would include friction due to seals, mechanical rubbing, and viscous friction on the plug.

The static, time-independent terms of Equation 6.4(6) are identical with Equation 6.4(1). The transfer function of the valve actuator is the LaPlace transform of differential Equation 6.4(6):

\[
\frac{X(s)}{P(s)} = \frac{\omega K}{(\omega g K)s^2 + (b/\omega K)s + 1}\]

where \(g\) is the gravitation constant, \(s\) is the LaPlace operator, and \(w\) is the weight of moving parts. This can be written in terminology more useful to instrument engineers using the time constant \(\tau\) (tau) and damping factor \(\zeta\) (zeta).

\[
\frac{x(s)}{P(s)} = \frac{1}{\tau^2 s^2 + 2\zeta\tau s + 1}
\]

The coefficient of the \(s^2\) term in Equation 6.4(7) is the square of the reciprocal of the undamped natural frequency of the spring-mass system. It is a useful number in understanding the relative importance of a control valve’s dynamic components.

Table 6.4h is a list of the natural frequencies of different size valves of the “average” design. It should be noted that even the largest valve with its un-damped natural frequency of Hz = 9 is ten times faster than the typical pneumatic performance of a control valve.

For a more detailed discussion of the dynamics of diaphragm actuators and of the effect of standing pressure waves in the piping on that dynamics, the reader is referred to the discussion by Lynch.

**Resolution and Valve Oscillation** The minimum change in the stem position of a control valve is called its resolution, which limits the ability of the control signal to position the valve exactly at a specific point of its travel. Because of this limitation, valves can continuously oscillate, as the increment of the stem position that can be delivered is larger than what is required. The result is a continuous sequence of overshooting and undershooting the stem position target. The valve’s resolution, this minimum change in position, cannot be very accurately calculated, because of the variations between the valve designs.

The definition of resolution for all pneumatic actuators is the ratio of the change in friction force to the spring rate. The relative resolution \(R\) is calculated by taking the difference between the static and dynamic friction forces \((F_s - F_d)\) and dividing that with the difference between the mechanical spring rate and the spring rate of the trapped air \((K_s - K_v)\). The resolution is calculated or percent of full stroke is then obtained by dividing by the stroke. In equation form the resolution, \(R\), in dimensions of length is:

\[
R = \frac{F_s - F_d}{(K_s - K_v)}
\]

where:
- \(R\) is the resolution in units of either length or percentage of full stroke
- \(F_s\) is the static spring friction
- \(F_d\) is the dynamic (running) spring friction
- \(K_s\) is the spring rate of the trapped air
- \(K_v\) is the spring rate of the spring

The spring rate of the trapped air \((K_v)\) is shown by Equation 6.4(10):

\[
K_v = \frac{1.4PA^2}{V}
\]

where:
- \(A\) is the effective area of the actuator
- \(V\) is the volume of trapped air in each actuator cavity
- \(P\) is the absolute pressure in the actuator

The main contributors to the friction forces are the packing friction on the actuator stem, the valve stem and the effect of any valve internal balancing seals. The motion of the actuator is faster than the time it would take for the air to be exchanged on both sides of the diaphragm or piston. Therefore, the air pressure conditions at each point of travel can
be evaluated in a static manner, and the air exchange dynamics can be ignored.

A number of observations can be made in connection with Equation 6.4(9). One such observation is that the dynamic or running friction ($F_d$) is always less than the static or breakaway friction ($F_s$). The difference usually is 25–35%. Another observation is that the friction forces for a PTFE (Teflon) seal are less than the friction forces generated by higher temperature seals made of fibrous graphite.

The size of these friction forces is much affected by the amount of extra torque applied to tightening packing box seals during installation. Too much compression of the seals will result in high friction forces and in stem travel oscillations.

The mechanical spring rate ($K_s$) is essentially constant. The air spring rate ($K_a$) can be increased by selecting a large effective area ($A$) in the actuator in combination with a small air volume ($V$). Because the air spring rate ($K_a$) is a function of the air volume and because the mechanical spring force changes with the stem movement, the resolution ($R$) will also vary with valve travel.

The relationship between resolution and stem travel for a variety of actuator designs is shown Figure 6.4i. In most designs the use of a spring tends to reduce the resolution. In spring-diaphragm combinations, the resolution is improved (reduced) when the spring opposes the direction of valve stem travel.

Good maintenance is essential to minimize the frictional forces in valve actuators. The positioner, which delivers or exhausts air to/from the actuator, is slower than the speed at which small changes in stem position occur.

Once the valve is installed the only means available to the user to change the resolution is to modify the supply air pressure, up to the limit of the design pressures. Because the air spring rate ($K_a$), which can be calculated by Equation 6.4(10), is small in comparison to the mechanical spring rate ($K_s$), an increase in the air supply pressure is not likely to have much impact on the resolution.

For reasons of competition, the valve manufacturers usually provide the smallest actuator they can for the particular application. Yet, for a small additional expense the user can usually obtain an actuator with a larger effective area and obtain a noticeable improvement in resolution and, therefore, in the controllability of the loop. By so doing, the continuous oscillation of the valve can often be stopped.

As it is shown in Figure 6.4j, the valve resolution can be much reduced (improved) by using a larger actuator. As can be noted from the figure, a doubling of the effective area of a diaphragm actuator cut the actuator resolution nearly in half.

### Safe Failure Position

The valve application engineer must choose between the two readily available fail-safe schemes for control valves, either fail open or fail closed. The choice will be based upon process safety considerations in the event of control valve air failure. Complete plant air failure, controller signal failure, and local air supply failure must all be considered. Local failure is significant when a valve positioner is being used and when piston actuators with cushion loading are used.

The choice must be based on detailed knowledge of the valve application in the overall process or system. Two generalizations are that in a heating application, the valve should fail closed, and in a cooling application it should fail open. There are certainly applications where either failure mode is equally safe; then, considerations of standardization may be used.

Fail-safe involves the selection of actions of actuator and inner valve. Both actuator and inner valve usually offer a choice of increasing air pressure to push the stem down or up, and pushing the stem down may open or close the inner valve. The proper choice of combinations may be made by fail-safe considerations. The process application of the valve must be investigated to determine whether, on instrument air failure, it would be better to have the valve go fully open, fully closed, or remain in its last position.

There may not be much flexibility in the inner valve action. For example, a single-seated top-guided valve must...
push down to close the plug. There is freedom of choice, however, in either single- or double-seated top- and bottom-guided valves. Other valve bodies, such as the Saunders and pinch valve styles, must be of the push-down-to-close type.

Rotary types, such as butterfly and ball valves, may be arranged either way.

The inner-valve flexibility leads to two cases: one in which either inner-valve action is permissible and one in which the inner-valve must be push-down-to-close.

When there is a choice of inner-valve action, overall valve action may be obtained by selecting the suitable inner-valve action and always using increasing air to push down the actuator. This is known as a direct actuator. A direct actuator is preferred because of economy reasons in spring-and-diaphragm actuators. The savings may be in purchase cost. It is also realized in maintenance costs, because there is no actuator stem seal to cause possible leakage and maintenance costs.

When the inner valve must be push-down-to-close, it is necessary to use both direct and reverse actuators to accomplish the desired fail-safe actions. Figure 6.4k summarizes the available diaphragm failure options.

The piston-type actuator is equally suitable for direct or reverse action. If it is the actuator to be used, the application engineer has complete freedom of the choice of selecting the valve action.

The Role of the Positioner

The above description provides a "baseline" for safe valve failure if the actuators are not provided with positioners. By the addition of a positioner, the topic of safe valve failure becomes quite complex. This is because in this case not only the pneumatic signal to the actuator can fail, but also the air supply to the positioner. In order to satisfy these requirements and also to make available the "fail in place" configurations, it is necessary to provide various accessories to either exhaust or trap the actuator air pressure. These accessories include pneumatic pilot valves (see Section 6.2) that, if air pressure is lost, will trip to provide a safe valve action. As far as digital systems are concerned, as of this writing only one digital positioner manufacturer provides a tight shut-off positioner.

<table>
<thead>
<tr>
<th>Valve failure</th>
<th>Fail Open</th>
<th>Fail Closed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator Inner Valve</td>
<td>Direct</td>
<td>Reverse</td>
</tr>
</tbody>
</table>

![Figure 6.4k](image) Overall valve failure positions, which can be achieved by various combinations of direct or reverse actuators and inner valves.

![Figure 6.4l](image) Forces acting on control valve with a spring-and-diaphragm type actuator.

In the cases where only the positioner can send an air signal to the valve actuator, the valve failure position is usually unaffected by the addition of the positioner. An exception to this statement is if a nonbleeding digital positioner is used.

Actually, it is questionable if a bleed can be considered as a means of providing positive failure position. This is because the time to bleed the air out through the positioner can be quite long. This long time that is required to reach a failure position is often unacceptable, and in such cases, one should provide pneumatic pilot valves to quickly trap or exhaust the air.

Pneumatic Response Times

An earlier discussion considered the transfer functions of the spring-and-diaphragm actuator between the actuator pressure to the resulting stem travel. Next we will consider the pneumatic transfer function of the air signal from the controller to the diaphragm actuator (Figure 6.4i).

A short tube behaves linearly as a pure resistance, and the air volume in the actuator above the diaphragm behaves as a capacitance. So the combination is a resistance-capacitance time constant. Some time constant values obtained from tests with very short tubing are given in Table 6.4m.

<table>
<thead>
<tr>
<th>Valve Size</th>
<th>Time Constant</th>
</tr>
</thead>
<tbody>
<tr>
<td>(in.)</td>
<td>(mm)</td>
</tr>
<tr>
<td>1</td>
<td>25</td>
</tr>
<tr>
<td>2</td>
<td>50</td>
</tr>
<tr>
<td>4</td>
<td>100</td>
</tr>
</tbody>
</table>

![Table 6.4m](image) Time Constants for Short Tube Sections
Performance is usually limited by the controller’s or positioner’s ability to supply the required air fast enough. The time constant values in Table 6.41 were obtained from tests in which the air supply was not limiting. These figures show that the valves are capable of fast response. Section 3.1 contains a more detailed discussion of transmission lags and methods of boosting.

**PISTON ACTUATORS**

Piston actuators are either single or double acting. The single-acting actuator, shown in Figure 6.4n, utilizes a fixed air pressure, known as the cushion, to oppose the controller signal. This valve does not have spring or diaphragm area nonlinearities, but it is of course subject to the same plug force nonlinearities (Figure 6.4f) as the spring-and-diaphragm actuator.

In order to use such an actuator for throttling purposes, it is necessary to have a positioner. The positioner senses the actuator motion and causes the valve to move accordingly. It cannot be used as a proportioning travel device without the positioner; consequently, its performance is that of the “ideal” curve in Figure 6.4g.

A double-acting piston actuator is one that eliminates the cushion regulator and uses a positioner with a built-in reversing relay. Thus, the positioner has two air pressure outputs, one connected above the piston and the other below. The positioner receives its signal and senses travel in the same manner as a single-acting positioner. The difference is in the outputs; one pressure increases and the other decreases to cause piston travel.

Table 6.4o provides typical actuator stroking times as a function of actuator sizes, stroking distances, and connecting tube sizes. For closed-loop control applications, the speed of response that is critical is not the full stroking time but rather the time it takes to move the valve about 5% of its full stroke, which is much faster than the values given in Table 6.4o. Therefore, velocity limiting of the loop usually occurs only when the valve is “slow” relative to the controlled variable and when the change in the controller signal to the valve is large.

One advantage of the piston actuator is that higher pressures can be used for motive power. The higher pressure provides better stiffness and resolution. It also provides more force to keep the valve closed. The higher force between the valve plug and the seating surface ensures a tighter shut-off and helps to meet the leakage specifications of the original design.

The piston actuators can also generate longer strokes, because their stroke is limited only by the length of the cylinder used. Such longer strokes are required for many special valve designs that are used in services where cavitation, noise, or pipe vibration is a problem (see Section 6.14).

All of the failure modes are available with the piston actuator. A spring may be used inside or outside of the cylinder to cause the valve to fail in the desired position. The use of outside springs necessitates the use of additional seals and increases mechanical complexity. The use of an inside spring requires a larger cylinder volume that could impact the resolution of the valve.

To obtain a positive failure position without a spring requires a standby air tank to provide the needed power for moving the valve into the failure mode. It also requires control accessories to ensure proper exhaust or lock-up of the air pressures.

**HIGH-SPEED ACTUATORS**

In the past, high-speed actuation was usually provided by hydraulic actuators. Applications that require fast control valve motion include compressor recycle, turbine bypass, and pressure relief applications. Control valves used in starting up or shutting down a process may also require fast actuation.
to protect equipment. Fast speed in these cases usually means full valve stroke in 1 to 2 sec.

In case of pneumatic actuators, the factor that is limiting the speed is the speed at which air can be fed or exhausted from the cylinder. This is accomplished by accessories that direct the air to the cylinder while bypassing the positioner and by increasing the air feed line sizes and the air supply pressure. Because of the increased pressure, the piston actuator is usually preferred in comparison with the diaphragm design, because of its higher pressure rating.

Diaphragm actuators are usually limited to 40 PSIG (275 kPa), with a few designs permitting 60 PSIG (410 kPa) operation. The high-pressure designs can operate with the usual plant air systems of 150 PSIG (1035 kPa) with normal operating pressures of 100 PSIG (690 kPa). This range of air supply pressures is readily available in most plants.

The high-speed pneumatic actuation can be achieved by the use of boosters to feed and exhaust air from the cylinders. These boosters are actuated when the control signal calls for a 10% or so change in valve opening. An actuator that is being fed by two boosters located above and below the piston is shown in Figure 6.4p.

In addition to the boosters, other accessories are also needed to guarantee the required failure positions. There are many disadvantages to these designs, including the difficulty of tuning and calibrating all of the component devices individually so as to maintain a stable operation. In addition, the tubing configuration requires a substantial amount of space around the actuator, and the accessories and tubing provide a lot of handhold and support points for operators, which can lead to damage. These systems can also be sensitive to vibration and to high temperatures radiating from a hot valve.

A newer development is shown in Figure 6.4q. This design required the development of high-capacity servo-valves that can be positioned to very close tolerance and of a programmable electronic controller. The controller can receive the HART or fieldbus protocol of the control system, can maintain the travel of the actuator so that there is no visible overshoot even during the fastest transients, and can calibrate the actuation system within seconds. The higher pressure and stiffness of the actuator allows positioning within 0.25% of the total travel. Dead time and hysteresis are also considerably lower than for the more conventional pneumatic actuators.

Figure 6.4r shows a response curve for an actuator with a 14 in. (355 mm) travel in which full stroke is achieved in 1.3 sec. The dead time is less than 0.140 sec, and there is no visible overshoot when operating a relatively high friction valve. There are no mechanical linkages, tubing, or accessories...
6.4 Actuators: Pneumatic

 According to the supplier, all failure positions can be provided and diagnostic software is also available. (For more details, see Sections 6.8 on valve diagnostics, 6.11 on fieldbus interaction, and 6.12 on intelligent valves.)

 The high-speed pneumatic actuators reduce the need to use hydraulic actuators, when the stroking time of a second or two is sufficient. One pneumatic actuator supplier feels that both electric and hydraulic high-speed actuators are more expensive than the pneumatic ones. (For a detailed discussion of hydraulic and electric actuators refer to Section 6.3.)

RELATIVE MERITS OF DIAPHRAGM AND PISTON ACTUATORS

Table 6.4a compared the features of diaphragm- and piston-type pneumatic actuators. When choosing between piston actuators and the spring-and-diaphragm type, the fail-safe consideration may be the reason for the final selection. If properly designed, the spring is the best way of achieving fail-closed action. Fail-open action is less critical.

Piston actuators may depend upon air lock systems to force the valve closed on air failure. Such systems may work well initially, but there are possibilities for leaks to develop in the interconnecting tubes, fittings, and check valves. Therefore, such piston actuator systems are not considered reliable by many. However, according to one manufacturer, field data on reliability shows the opposite to be true. Air lock systems also add to the actuator’s cost. Piston actuators may also be specified with closure springs to provide positive failure positions.

Valve installation in the line is also a factor to consider. Flow over the plug assists in maintaining valve closure after air failure, but the considerations involving dynamic stability are more important. Therefore, the use of “flow-to-open” valves is recommended for most diaphragm actuators, as the actuator force is usually marginal.

Piston actuators are larger and require more space than do diaphragm actuators. This is particularly the case when the piston is provided with a spring to provide a positive failure position.

Both the diaphragm and piston actuators use manifolds and have become available in modular designs. This has somewhat reduced their costs and lowered their potential for leakage by reducing the number of connections that could leak. An example of the modular design of a linear actuator is shown in Figure 6.4s. The linear actuator includes a positioner, boosters, and adjustable quick exhaust.

Figure 6.4t illustrates the modular design of a rotary piston actuator. The rotary design is the balanced pinion type that can be provided with a plug-in positioner, which can be either the conventional or the smart design. The smart positioners also are available with bus communication options.

Upgrades or to correct errors in the original design. These modular designs are usually provided with position indicators, which are enclosed within the actuator housing. Because of these housings, reliability is better than in designs where the linkages are exposed. Most control modules will eventually include bus control communication and self-diagnostics capability.

Pneumatic vs. Hydraulic Actuators While the purpose of this section is to describe the features of pneumatic actuators and while a detailed discussion of hydraulic and electric actuator features is given in Section 4.3, a few comments
Control Valve Selection and Sizing

from the perspective of a pneumatic piston actuator manufacturer is included here.

The main advantages of hydraulic actuators are speed and stiffness. This is the case because of the high density and the incompressibility of liquids in comparison to air. The speed difference between pneumatic and hydraulic actuators has been narrowed, as the stroking time of some pneumatic piston operators (particularly with dual pistons) is about 1 sec.

With the use of 100 PSIG (690 kPa) or higher air pressures, the piston actuator stiffness and stability have also improved and approach that of the hydraulic actuators. The stem forces provided by a pneumatic piston can equal or exceed those of the hydraulic cylinders, because the area of an oversized air piston can provide higher stem forces than a standard hydraulic actuator. For example, if the hydraulic cylinder area is one 50th of the air piston, while the air pressure is one 20th of the oil pressure, the stem force produced by the hydraulic actuator is actually less than that of the pneumatic actuator.

ROTOR VALVE ACTUATORS

When linear actuators are used to operate rotary valves, a unit change in controller signal will not result in a unit change in rotation unless a positioner is used (Figure 6.4u). By the addition of a positioner, one can guarantee that the ratio between a unit change in controller signal and the resulting rotation will be uniform. On the other hand, loose-fitting actuator lever and follower arms can still create dead play in the actuator, which will lower the responsiveness of the loop when the direction of change in the control signal reverses.

As can be seen in Figure 6.4v, the torques required to rotate ball, butterfly, or plug valves are not linear. These valves are usually used only for on/off applications. When considered for closed-loop throttling control, the nonlinear relationship between the air signal and the resulting rotation makes the use of a positioner essential.

Figure 6.4w shows the nonlinearity in the torque characteristics of double-acting and spring-loaded cylinder actuators.

![Diagram showing components of a pneumatic actuator](image)

**FIG. 6.4u** When linear actuators are used to operate rotary valves, a unit change in controller signal will not result in a unit change in rotation unless a positioner is used. (Courtesy of Flowserv Inc.)

**FIG. 6.4v** The torque characteristics of such rotary valves as ball, butterfly, and plug valves are not linear.
Naturally, the piston actuator should be so selected that its break torque exceeds the peak torque requirement of the valve, which occurs at the beginning and the end of the stroke (break torque).

The torque characteristics of the double-acting piston actuator of Figure 6.4x shows two maximums. One maximum occurs at the closed position and one near the 60–80° position, which corresponds to the peak of the dynamic torque for butterfly valves. Naturally, this actuator too should be so selected that its break torque exceeds the peak torque requirement of the valve, which occurs at the beginning (break torque) and near the end of the stroke.

**Cylinder Type**

Increased use of ball and butterfly valves or plug cocks for control has bred a variety of actuators and applications of existing actuators for powering these designs.

Positioning a quarter-turn valve with a linear output actuator using a lever arm on the valve resolves itself into a problem of mounting and linkages. The actuator can be stationary, with a bushing to restrain lateral movement of the stem. This requires a joint between the stem and a link pinned to the lever arm. The actuator can be mounted on a gimbal mechanism to allow required movement. The actuator can be hinged to allow free rotation to allow for the arc of the lever arm.

Various Scotch yoke designs, such as the one shown in Figure 6.4y, can be used with one, two, or four cylinders. Use of rollers in the slot of the lever arm utilizes the length of the lever arm of the valve opening or closing points.

A rack and pinion can be housed with the pinion on the valve shaft and the rack positioned by almost any linear valve actuator. The rack (Figure 6.4z) can be carried by a double-ended piston or by two separate pistons (Figure 6.4aa) in the same cylinder, where they move toward each other for counterclockwise rotation and away from each other for clockwise rotation.
Similar action is obtained (Figure 6.4bb) by two parallel pistons in separate cylinder bores. Dual cylinders are used in high-pressure actuators used to rotate ball valves as large as 16 in. (400 mm) in less than 0.5 sec.

An actuator similar to the one shown in Figure 6.4cc can be spring-loaded for emergency or positioning operation.

On/off operation of cylinders for quarter-turn valves requires solenoid or pneumatic pilots to inject pneumatic or hydraulic pressure into the cylinders. Open or closed position must be set by stops that limit shaft rotation or piston travel. Thereby the valve rotation is stopped and held in position until reverse action is initiated. Positioning action requires a calibrated spring in the piston or diaphragm actuator, a valve positioner, or a positioning valve system that loads and unloads each end of the cylinder.

Rotation of the valve must be translated to the positioner by gears, direct connection, cam, or linkage. The valve positioner must be the type that includes the four-way valve. The positioning valve system can be a four-way valve with a positioner for use with a pneumatic controller. The piston is sometimes positioned by a servo-system consisting of a servovalve that accepts an electronic signal, a four-way valve to amplify and control the pressure to the cylinder, and a feedback signal from a potentiometer or LVDT.

An electrohydraulic power pack or pneumatic pressure source may be used to furnish pressure to a pair of cylinders, one for each direction of rotation, as shown in Figure 6.4dd.

A multiple helical spline rotates through 90° as pressure below the piston (Figure 6.4ee) moves the assembly upward. A straight spline on the inside of this piston extension sleeve rotates the valve closure member through a mating spline. The cylinder is rated at 1500 PSIG (10 MPa). The actuator will rotate 1 in. (25 mm) and 1 1/2 in. (37.5 mm) valve stems. The mounting configuration is designed to adapt to many quarter-turn valves.

A nonrotating cylinder (Figure 6.4ff), with an internal helix to mate with a helix on a rotatable shaft, creates a form of rotating actuator. Hydraulic or pneumatic pressure in the drive end port (left) causes counterclockwise rotation; clockwise rotation is caused by pressure on the opposite side of the piston. There is a patented seal between the internal and the external bores of the cylinder and the external surface of the shaft. The unit is totally enclosed by seals to protect it from contaminated atmospheres. A hydraulic pump, reservoir, and necessary controls can be mounted integrally.

Injection of pressure on one side of a vane to obtain quarter-turn actuation is straightforward and obtainable with a minimum number of parts. A single vane (Figure 6.4gg) can be used for 400 to 30,000 in. lb (27 to 2060 N · m).

Units can be mounted together for double output, or a double-vane design (Figure 6.4hh) can also be used. The success of the vane actuator as a control device is dependent upon the control systems. By use of an auxiliary pneumatic pressure source, all types of fail-safe actions are possible, although not as positively as with spring loading. Use of line pressure to create hydraulic pressure on the vane is piloted by both manual and automatic methods. Use of a rotary potentiometer to sense position and complete a bridge circuit is necessary for proportional control.

Pneumatic pressure is used to power a rotary motor to drive any of the large gear actuators. Control is by a four-way valve. The motor shown in Figure 6.4ii is running in one position. This will continue until the valve is repositioned or until a cam operates a shut-off valve at one end of the stroke. Reversal of the four-way valve causes reverse operation. An intermediate position causes the motor to stop. The four-way valve can be operated by pneumatic or electric actuators for remote automatic control. A position transmitter will allow adaptation to closed-loop proportional control.
6.4 Actuators: Pneumatic

OTHER PNEUMATIC ACTUATORS

Pneumohydraulic Actuators

An actuator with two double-acting cylinders uses an integrally designed pneumatic or electric powerpack (Figure 6.4jj). This has the advantage of furnishing a constant hydraulic pressure to the cylinders regardless of the power source to the prime mover. Few cylinder sizes are needed to cover a wide range of torque outputs. The prime movers are sized and selected to obtain the actuator speeds desired with the pneumatic pressures or electric voltages available. Multiple auxiliary switches, position transmitter, and positioning devices are adapted to the unit.

Gas pressure is used to create hydraulic pressure using two bottles (Figure 6.4kk). The stability of a hydraulically operated cylinder is utilized in this manner, using line gas pressure and the bottle size for amplification. The manual control valve can be replaced by a variety of electric or pneumatic pilot valves for automatic control. A hand pump is furnished that can take over the hydraulic operation in the absence of gas pressure or malfunction of the pilot controls. This self-sustaining approach to cylinder operation finds wide application for line break shut-off and for the various diverting and bypass operations of a compressor station.

A hydrostatic system consisting of a pneumatic prime mover (Figure 6.4ll) on the shaft of a hydraulic pump to run a hydraulic motor has interesting features. This actuator incorporates many of the features of other high-force geared actuators that rotate a drive sleeve. Torque control consists of a relief valve in the hydraulic line to the motor. This eliminates the reactive force of spring-loaded torque controls. Starting torque occurs

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because hydraulic slippage of the pump allows the motor to reach maximum speed.

Direction and deactivating control is attained with a four-way valve. As many as 16 auxiliary switches, settable at any position, are housed in the unit. Limit switches can be pneumatic or electric. A wide range of torque outputs and speeds is obtained by selection of prime mover, pump, and motor combinations. Initial success of the unit was partially due to its adaptability for retrofit to existing valves. The unit can be manually operated if required.

**Electropneumatic Actuators**

An actuator that defies classification, except that it is pneumatically powered and electrically controlled for proportional...
6.4 Actuators: Pneumatic

Application, is described at this point. Operation of a threaded drive sleeve occurs when spring-loaded pawls create a jogging action on a drive gear. Pressure introduced through one of the external lines selects the pawl to become active when the rocker arm is repetitively rocked by the pneumatic motor. A lead screw positions a sliding block to operate control switches, potentiometer, and position indicators. The lead screw is driven from a small spur gear and bevel gears.

Air is supplied from 60–140 PSIG (414–966 kPa) to give torque outputs up to 360 or 720 ft lb (25 or 50 Nm) using two actuators. Air consumption is from 0.75 SCF–1.70 SCF (0.021–0.048 m³) per revolution at 140 PSIG (966 kPa). Maximum valve stem diameter that can be rotated is 2 in. (50 mm).

Numerous motivating combinations are used for control, including electropneumatic, electric, and fully pneumatic. A wide variety of components can be used to build up these systems, as shown in Figure 6.4mm.

RELIABILITY

Some of the reliability studies on the performance and reliability of pneumatically operated valves were done in the offshore oil industry and in the power industry. These studies did not evaluate only the actuator but the complete valve and actuator system combined.

The offshore study for pneumatically operated control valves was made in 1984 and reported the highest failure rate of 1 failure every 10,000 hours; or a little less than 1 failure per year. A 1990 study made for pneumatically operated control and on/off valves by the Institute of Nuclear Power Operation (INPO) showed a failure rate of 116,000 hours.

In 1997 a summary on the reliability of air-operated valves by the electric power industry was published. The sample consisted of 525 failures from 4,726 component years of operation. This would equate to a failure every 9 years or 79,000 hours of operation. The data covered all types of valves with both piston and diaphragm actuators from eight manufacturers. The study looked at what was called “high duty” (e.g., control valves) vs. “low duty” (e.g., shut-off valves) as well as the impact of intrusive maintenance. A summary of the results

FIG. 6.4jj
Actuator with two double-acting cylinders and power pack.

FIG. 6.4kk
Pneumohydraulic actuator powered by line pressure. (Courtesy of Shafer Valve Co.)
from this study is given below:

- High duty cycle valves exhibit up to 13 times lower reliability than low duty cycle valves.
- Intrusive preventive maintenance causes reliability to decrease by a factor of 6 times.
- The reliability of piston-type actuators was 4 times better than that of diaphragm-type actuators.
- The reliability of diaphragm-actuated valves was 6 times more likely to be impacted by duty cycle than were piston actuators.

These results are a bit surprising because the common perception is that the diaphragm actuator is much more reliable than the piston actuator. It is also possible that by combining all types of valves (on/off and throttling) and by placing emphasis on duty cycle (opening and closing of the valves), the results of this evaluation would differ with one that evaluated only throttling control valves.

**CONCLUSIONS**

Before deciding on the type of control valve actuators to be used in a particular application, the reader is advised to also read Section 6.3, which discusses hydraulic, electric, and digital valve actuators, and to study Table 6.3a, which lists the advantages and disadvantages of electromechanical, electrohydraulic, and servo- or stepping-motor-operated electric actuators. Those who want to learn even more about the features and performance of available valve actuators are advised to study the test reports, books, and articles listed in the Bibliographies of both Sections 6.3 and 6.4.

As far as actuator trends are concerned, pneumatic actuators are still the technology favored by the users of valve actuators. Nevertheless, from 1998 to 2003, the market share of electric actuators increased from 11% to 45% and the market share of hydraulic and electrohydraulic actuators increased from near 0% to 28%. During the same period, use of solenoid valves has dropped by 42%.

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ANSI/ISA-TR75.25.02-2000, “Control Valve Response Measurement from Step Inputs.”