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Actuator Sizing

8.1 Actuator-Sizing Criteria

8.1.1 Introduction to Actuator Sizing

With the automation of process systems, the use of actuators on throttling valves and actuation systems on manual on-off valves has increased dramatically. Generally, actuator sizing is a complex science, involving a number of factors that must be considered to match the correct actuator with the valve. For the valve to open, close, and/or throttle against process forces, proper actuator selection and sizing are critical.

Some users equate valve-body size with the actuator size; for example, a false assumption can be made that a 3-in valve always uses a certain size actuator, whose standard actuator yoke connection matches the valve connection. If all process service conditions and valve designs were equal, this might be possible. However, processes vary widely in terms of pressures, pressure drops, temperatures, shutoff requirements, etc. Valves vary according to motion (linear and rotary), packing friction, balancing (nonbalanced versus pressure-balanced), etc. Because of all the variables between the process and the valve, one valve size may have a number of actuator size options. For this reason, the user cannot simply place any spare actuator on a valve and expect it to work correctly—the actuator will most likely be undersized or oversized for that valve and the process. If the actuator is undersized, the major problem is that it will not be able to overcome the process and valve frictional forces. If the actuator is slightly undersized, it will struggle to overcome the forces working against it, providing sluggish and erratic stroking, as well as possibly not meeting the shutoff requirement. In addition, if the actuator is not stiff enough to hold its position close to the seat or seal, the “bathtub stopper” effect will take place and the closure element will slam into the seat or seal, causing a

water-hammer effect. If the actuator is extensively undersized, it will not be able to open or close or throttle correctly.

If the actuator is oversized, the main disadvantage is that the actuator cost is higher. In addition, the oversized actuator is heavier and taller, which may create seismic, space, or maintenance concerns. From a performance standpoint, the larger actuator may be more sluggish in terms of speed and response. Larger actuators also produce greater thrust, which may damage the internal parts of the valve if the process forces are not present to counter that thrust. For this reason, oversized actuators require the use of a pressure regulator, which may create additional problems of incorrect settings and even slower response.

Generally, actuators have a tendency to be oversized because of the buildup of safety factors that the user and manufacturer add to the design process to ensure adequate “worst-case scenario” protection. If the calculations show a certain actuator size to be marginally or slightly undersized for a given process and valve, most users tend to move to the next larger size. However, because of the safety factors already built into the sizing process, the smaller size may function just as well, if not better, with that process.

8.1.2 Basic Actuator-Sizing Criteria

Actuator-sizing methods vary from manufacturer to manufacturer, depending on the basic design; however, several basic concepts are central to any actuator sizing. First and most importantly, the actuator must have the thrust to overcome the process forces that are operating inside the valve—in particular, the upstream and downstream pressures. In some services, a valve is working in an unbalanced situation where the upstream pressure is working against one side of the closure element, and the downstream pressure is working against the opposite side. These forces can be significant and will require a larger actuator as the force increases. Other valves permit a pressure-balanced design in which the upstream pressure is allowed to act on both sides of the closure element. This allows a minimal amount of process force to act against the element, permitting a smaller actuator.

The actuator must also provide enough force to overcome the process pressures in order to close the closure element, as well as to maintain the shutoff requirements indefinitely, according to the seat leakage classification (Sec. 2.3). The tighter the shutoff requirement, the greater the force must be provided by the actuator. If tight shutoff is not a main consideration, or if the valve is expected to throttle and close rarely, a lower shutoff classification may suffice that will allow the use of a smaller actuator.

The actuator must also overcome any frictional forces between the valve's stem and packing box. This friction can vary from a number of factors: number of rings, packing material, linear versus rotary motion, and packing compression requirements.

The final factor that may create a need for additional force is the design criteria of the valve itself. For example, a linear globe valve may be designed with pressure-balanced trim. Although the process forces are minimized, the seals of the pressure-balanced plug will increase the frictional forces, as well as add to the weight of the plug. In extremely large valves, the weight of the closure or regulating element (especially with globe-style plugs) must be taken into consideration.

Therefore, the forces that must be considered to determine the size and subsequent thrust of the actuator are written as

$$F_{\text{total}} = F_{\text{process}} + F_{\text{seat}} + F_{\text{packing}} + F_{\text{miscellaneous}}$$

where F_{total} = total force (or actuator thrust) required to open, close, or throttle valve

F_{process} = force to overcome unbalanced process pressure

F_{packing} = force required to overcome packing friction

F_{seat} = force to provide correct seat load

$F_{\text{miscellaneous}}$ = force to overcome special design factors, weight, etc.

Another design criteria is the speed requirement of the actuator. In some cases, such as applications in which the process or personnel safety is a concern, the user may want the valve to close in a short time, such as less than a second, as opposed to several seconds. However, excessively fast actuator speed can present multiple problems, including water-hammer effects and position overshoot. Pneumatic actuators are subject to a number of factors that affect air capacity, such as pressure fluctuations, piping and tubing bends, filters, etc. For these reasons, high-speed actuation systems are normally hydraulic or electrohydraulic designs.

8.1.3 Free Air

Because the majority of actuators or actuation systems are pneumatically driven, certain principles concerning air compressibility and volume changes occurring with pressure changes must be understood. The specifications for pneumatically driven equipment, including actuators, are provided using the term *free air*. By definition, free air is the flow or volume rate at standard atmospheric temperature [70°F (21°C)] and pressure [14.7 psia (1 bar)]. Using free air avoids any mis-

understanding regarding changes in volume. Typically, absolute pressure is designated as *psia*, gauge pressure as *psig*, and differential pressure as simply *psi*. For most equipment, the free-air flow rate is expressed in standard cubic feet per minute (scfm).

Because air volume can vary according to changes in pressure, the amount of free air contained in a vessel can be written as

$$V_1 = V_2 \frac{P_2}{P_1}$$

where V_1 = free-air volume (standard cubic feet)

V_2 = vessel volume

P_1 = atmospheric pressure (14.7 psia)

P_2 = absolute vessel pressure (psia)

8.1.4 Supply Flow Rates

For pneumatically driven actuators, determining the correct air supply rate to the actuator is critical to ensure that enough air will be available to operate the actuator and provide the thrust necessary for the application. The relationship between flow rate and pressure drop is demonstrated by the following equations:

$$\Delta P = \frac{L Q_2}{k C_R d^{5.31}} \quad \text{or} \quad Q = \sqrt{\frac{\Delta P k C_R d^{5.31}}{L}}$$

where ΔP = pressure drop (psi)

L = length of tubing or piping (ft)

Q = standard air flow rate (scfm)

k = constant of 35,120

C_R = ratio of line pressure (psia) to atmospheric pressure (14.7 psia)

d = inside diameter of piping or tubing (inches) from Tables 8.1 and 8.2

For example, if a given actuator operates best at 80 psi and must have 4.3 scfm to operate at the required speed, the following parameters apply:

Line pressure	85 psia
Length L of tubing	100 ft
Tubing size	0.25 in

Table 8.1 Piping Values for d and $d^{5.31*}$

Factor	0.25 in.	0.375 in.	0.5 in.	0.75 in.	1.0 in
d	0.364	0.493	0.622	0.824	1.049
$d^{5.31}$	0.0047	0.0234	0.0804	0.3577	1.2892

*All dimensions are outside diameter.

†Data courtesy of Automax, Inc.

The pressure drop ΔP is 5 psi (85 – 80 psi) and the ratio C_R of line pressure to atmospheric pressure is 6.78, which is shown as:

$$C_R = \frac{P_1 + 14.7}{14.7} = \frac{99.7}{14.7} = 6.78$$

Using the calculations above, the flow rate for 0.25-in tubing (from Table 8.2, $d = 0.204$) can be calculated as follows:

$$Q = \sqrt{\frac{\Delta P k C_R d^{5.31}}{L}} = \sqrt{\frac{(5)(35,120)(6.78)(0.204)^{5.31}}{100}} = 1.6 \text{ scfm}$$

Because 1.6 scfm is less than the 4.3 scfm required for the speed requirement, a larger tube size must be chosen. A 0.375-in tube would produce 5.7 scfm, which is more than adequate:

$$Q = \sqrt{\frac{\Delta P k C_R d^{5.31}}{L}} = \sqrt{\frac{(5)(35,120)(6.78)(0.329)^{5.31}}{100}} = 5.7 \text{ scfm}$$

Table 8.2 Tubing Values for d and $d^{5.31*}$

Factor	0.25 in.	0.375 in.	0.5 in.	0.75 in.	1.0 in
d	0.204	0.329	0.430	0.555	0.680
$d^{5.31}$	0.0002	0.0027	0.0113	0.0439	0.129

*All dimensions are outside diameter.

†Data courtesy of Automax, Inc.

8.1.5 Air Usage and Consumption

The user must ensure that the air-supply capacity can meet the needs of all the pneumatic operators involved with a typical process system. This means that the compressor must be sized according to the air requirements of the actuators, which requires knowledge of the air usage and consumption. Correct calculations of the air usage and consumption allow for a more accurate prediction of air requirements and proper sizing of the compressor. In those cases where the air requirements exceed the capacity of the compressor or if the compressor is undersized, the pressure will not be adequate. Overall, this results in sluggish response or not enough thrust to operate the valve.

The term *air usage* refers to the amount of air used by a pneumatic actuator to stroke the valve. After the valve is stroked, the air usage stops until the valve is stroked again. The term *air consumption* refers to those pneumatic instruments that bleed air constantly, such as is the case with positioners. For spring-return (single-acting) diaphragm actuators, no air is used on the spring side of the diaphragm. Therefore, when the actuator is fully stroked, the air usage is the amount of the actuator's free-air volume at the pressure given. For example, using the free-air equation from Sec. 8.1.3, the assumption is made that a single-acting actuator has a volume of 2.1 ft³ at 60 psi of air supply and will stroke six times per hour. The usage per cycle in standard cubic feet is

$$V_1 = V_2 \frac{P_2}{P_1} = 2 \left(\frac{60 + 14.7}{14.7} \right) = 10.2 \text{ scf}$$

The usage per hour (standard cubic feet per hour) involving six strokes per hour can then be calculated:

$$10.2 \text{ scf} \times 6 = 61.2 \text{ scfh}$$

Double-acting actuators use air on both sides of the diaphragm or piston, depending on the design. Ideally, the air volume on both sides would be equal, but this is not the case because one side has less volume due to the actuator stem, travel stop, or fail-safe spring. In this case, the total volumes of the two sides are calculated separately and added together to present air volume per cycle. For example, a rack-and-pinion actuator has 500 in³ on one side and 300 in³ on the opposite side. It will be stroked 12 times an hour with 80 psi of air supply.

Using the conversion factor, cubic inches are converted to cubic feet for the first side:

$$\text{scf} = \frac{\text{in}^3}{1728} = \frac{500}{1728} = 0.29 \text{ scf}$$

The opposite side is converted likewise:

$$\text{scf} = \frac{\text{in}^3}{1728} = \frac{300}{1728} = 0.17 \text{ scf}$$

The combined air volume for the actuator is then 0.46 scf (0.29 + 0.17) and the air usage per cycle is calculated as

$$V_1 = V_2 \frac{P_2}{P_1} = 0.46 \left(\frac{80 + 14.7}{14.7} \right) = 2.96 \text{ scf}$$

The usage per hour (standard cubic feet per hour) involving 12 strokes per hour can then be calculated:

$$2.96 \text{ scf} \times 12 = 35.52 \text{ scfh}$$

If a positioner is used with a single-acting actuator, the air usage can vary considerably since the actuator is throttling between the open and closed positions. Depending on the position movement, which can be large or small, the air usage is directly proportional to the movement. The air usage for an actuator with a positioner can be determined by the following equation:

$$\text{scfh} = \frac{V}{A} [P_s(M_2 - M_1) + 0.4PM_1]N$$

where V = actuator volume (ft^3)

P_s = supply pressure (psia)

M_1 = starting position (fraction of stroke)

M_2 = finished position (fraction of stroke)

A = atmospheric pressure (14.7 psia)

N = number of strokes per hour

For example, a single-acting actuator with a positioner has an air volume of 500 in^3 and is stroked between 10 and 50 percent open. It is

required to stroke eight times an hour using 60 psi of air supply. This would be calculated as

$$\begin{aligned} \text{scfh} &= \frac{V}{A} [P_s(M_2 - M_1) + 0.4 PM_1]N \\ &= \frac{500}{1728} \{[(60 + 14.7)(0.5 - 0.1)] + [0.4(60 + 14.7)(0.1)]\}8 \\ &= 5.26 \text{ scfh} \end{aligned}$$

This calculation provides only the air usage. Because some positioners bleed continually, they provide air consumption, which must be figured into the total air requirement when sizing the compressor. If a positioner is used with a double-acting actuator, as most are, the above equation is modified slightly:

$$\text{scfh} = \frac{V}{A} \{[2P_s(M_2 - M_1)] + [0.4P(1 - M_2 - M_1)]\}N$$

For example, a double-acting actuator with a positioner has an air volume of 300 in³ and is stroked between 20 and 70 percent open. It is required to stroke 12 times an hour using 80 psi of air supply. This would be calculated as

$$\begin{aligned} \text{scfh} &= \frac{V}{A} ([2P(M_2 - M_1)] + [0.4 P(1 - M_2 - M_1)])N \\ &= \frac{300}{1728} \{[2(80 + 14.7)(0.7 - 0.2)] + [0.4(80 + 14.7)(1 - 0.7 - 0.2)]\}12 \\ &= 11.82 \text{ scfh} \end{aligned}$$

Once again, the user should remember that any bleeding of air from the positioner (air consumption) must also be added to the air usage calculation.

8.2 Sizing Pneumatic Actuators

8.2.1 Actuator Force Calculation for Linear Valves

To determine what size of actuator is required for a linear-motion valve, such as a globe control valve, the user must examine the force that the process is applying inside the valve. This force value is known as F_{process} . A major factor in determining the process force is calculating the unbalanced area. The *unbalanced area* is defined as the area of the cage (or sleeve) minus the stem area. The unbalanced area must be greater than the area of the seat. In equation form, it is written as

$$A_{\text{unbalanced}} = A_{\text{cage or sleeve}} - A_{\text{stem}} > A_{\text{seat}}$$

where $A_{\text{unbalanced}}$ = unbalanced area
 $A_{\text{cage or sleeve}}$ = area of the cage or sleeve*
 A_{stem} = area of the plug stem
 A_{seat} = area of the seat

Formulas for calculating the process force are based upon the service conditions as well as three design criteria: The first determination is whether the flow assists with the opening or the closing of the valve. The second determination is whether the valve is unbalanced or pressure-balanced (globe or double-ported valves only). And the third determination is whether the flow is under or over the closure or regulating element (assumed to be a globe valve plug). The following formulas apply for the following valve configurations:

Pressure assists opening, unbalanced trim, flow under the plug:

$$F_{\text{process}} = (P_1 - P_2) A_V + P_2 A_{\text{stem}}$$

Pressure assists opening, unbalanced trim, flow over the plug:

$$F_{\text{process}} = (P_1 - P_2) A_V - P_1 A_{\text{stem}}$$

Pressure assists opening, balanced trim, flow under the plug:

$$F_{\text{process}} = (P_1 - P_2) A_{\text{unbalanced}} - P_2 A_{\text{stem}}$$

*If the valve does not have a cage or sleeve, the area of the top of the plug is used.

Pressure assists opening, balanced trim, flow over the plug:

$$F_{\text{process}} = (P_1 - P_2) A_{\text{unbalanced}} + P_2 A_{\text{stem}}$$

Pressure assists closing, unbalanced trim, flow under the plug:

$$F_{\text{process}} = -[(P_1 - P_2)A_V + P_1 A_{\text{stem}}]$$

Pressure assists closing, unbalanced trim, flow over the plug:

$$F_{\text{process}} = -[(P_1 - P_2)A_V - P_1 A_{\text{stem}}]$$

Pressure assists closing, balanced trim, flow under the plug:

$$F_{\text{process}} = -[(P_1 - P_2) A_{\text{unbalanced}} - P_1 A_{\text{stem}}]$$

Pressure assists closing, balanced trim, flow over the plug:

$$F_{\text{process}} = -[(P_1 - P_2) A_{\text{unbalanced}} + P_1 A_{\text{stem}}]$$

where F_{process} = force to overcome the process pressure unbalance

P_1 = upstream pressure at inlet (psia)

P_2 = downstream pressure at outlet (psia)

A_V = area of the valve port (in²)

A_{stem} = area of the plug stem (in²)

$A_{\text{unbalanced}}$ = unbalanced area (in²)

When the actuator force is used to open the valve, three of the four forces oppose the actuator: the process force, the packing friction force, and any miscellaneous design forces. Because no actuator force is needed for seat loading, that value is not necessary. This can be written as

$$F_{\text{open}} = F_{\text{process}} + F_{\text{packing}} + F_{\text{miscellaneous}}$$

where F_{open} = total force (or actuator thrust) required to open valve

F_{process} = force to overcome the process pressure unbalance

F_{packing} = force required to overcome packing friction

$F_{\text{miscellaneous}}$ = force to overcome special design factors, weight, etc.

If the total force must close the valve, the process force must be a

negative number, as demonstrated in the latter four equations. In other words, because the process pressure is assisting the valve to close, the process force actually decreases, rather than increases, the force required by the actuator. The actuator has to produce only enough force to overcome the combined force produced by the packing friction, seat load, and miscellaneous design forces, minus the process force. In this case, the actuator force requirement may be minimal. This can be written as

$$F_{\text{close}} = F_{\text{seat}} + F_{\text{packing}} + F_{\text{miscellaneous}} - F_{\text{process}}$$

where F_{close} = total force (or actuator thrust) required to close valve
 F_{seat} = force required to provide-correct seat load

In applications where the actuator must open and close the valve, both forces for opening and closing, F_{open} and F_{close} , must be calculated. The largest force of the two is then used to determine the size of the actuator.

After the process force has been determined, the next force to be calculated is the load required by the shutoff classification, which uses the following equation:

$$F_{\text{seat}} = F_{\text{class}} C_{\text{port}}$$

where F_{class} = required seat force of shutoff classification (see Table 2.7)
 C_{port} = circumference of the valve port

The force required to overcome the packing friction, F_{packing} , is provided by the manufacturer. Packing friction is determined by the diameter of the stem and the packing material, assuming correct compression. Overcompressing the packing will add to the packing friction and the force required to overcome it.

After the cumulative forces are calculated, an actuator can be chosen from the manufacturer based on the thrust capabilities of the actuator. The final requirement is that the correct actuator can be mounted on the valve that has been sized for the service. In some applications involving large oversized actuators required for severe services, the yoke-to-bonnet connection may not be a standard and will require modifications.

With pneumatic actuators, the appropriate size and spring will need to be chosen for the application. Most manufacturers provide actuator tables that include the thrust that the actuator can generate. In addi-

tion, the user must choose the desired failure action (fail-open or fail-closed) and yoke-to-bonnet connection. The correct actuator is the smallest actuator that meets the thrust and mounting requirements.

8.2.2 Actuator Force Calculation for Butterfly Valves

A different actuator-sizing criteria must be considered with rotary valves. Critical to rotary actuator sizing is the butterfly valve's torque requirement, in other words, the amount of thrust that the actuator must apply to the shaft to produce a rotational force to operate the valve. In particular, the user must calculate the *seating torque*, which is the torque needed to close the valve against or with the process; the *breakout torque*, which is the torque needed to begin to open the valve; and the *dynamic torque*, which is the torque needed to throttle the valve. When these torque values are known, the correct rotary actuator can be chosen.

The first step in sizing an actuator for a butterfly valve is to determine the orientation of the shaft and the actuator stiffness requirements. Shaft orientation is critical with eccentric butterfly valves. When the shaft is placed on the upstream side of the flow, the process fluid forces the disk into the seal. On the other hand, when the shaft is placed on the downstream side of the flow, the process fluid forces the disk to open. In gas applications, when the butterfly valve is designated to fail-closed, the shaft is generally upstream. If the valve is designated to fail-open, the shaft is downstream. With liquid applications, the disk has a tendency to slam into the seal in fail-closed applications if the actuator is not stiff enough to withstand the process flow. A rotary actuator with insufficient stiffness is likely to cause water-hammer effects; therefore a stiffness calculation must be made by finding the ratio of the maximum pressure drop to the supply pressure:

$$A_s = \frac{P_1 - P_2}{P_s}$$

where A_s = required actuator stiffness

P_1 = upstream pressure at inlet (psia)

P_2 = downstream pressure at outlet (psia)

P_s = Supply pressure

Table 8.3 shows the maximum actuator stiffness values for three sizes of actuators. If the calculated value is larger than the table value, a larger actuator size must be chosen for that size of valve.

For example, a 4-in butterfly valve has an upstream pressure P_1 of 240 psia, and a downstream pressure P_2 of 60 psia, and a supply pressure of 80 psi. The required actuator stiffness ratio is

$$A_s = \frac{P_1 - P_2}{P_s} = \frac{240 - 60}{80} = 2.25$$

Looking at Table 8.3 for 4-in valves, the actuator stiffness is slightly larger than the maximum value for the smallest actuator, size A. Therefore, the next larger size, size B, would be required.

The chosen actuator must also have the necessary force to generate torque for the butterfly valve to close, to open (breakout torque), and

Table 8.3 Actuator Stiffness Factors

Valve Size	Actuator Size A	Actuator Size B	Actuator Size C
2-inch DN 50	4.2		
3-inch DN 80	3.1		
4-inch DN 100	2.0	6.7	
6-inch DN 150		4.3	7.0
8-inch DN 200		2.5	5.9
10-inch DN 250			4.1

to throttle between the open and closed positions. The following equations are used to determine seating and breakout torques:

Shaft downstream, torque required to close the valve:

$$T_{\text{seat}} = -T_P - T_S - T_H - \Delta P_{\text{max}}(C_B + C_O)$$

Shaft downstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + T_H + \Delta P_{\text{max}}(C_B - C_O)$$

Shaft upstream, torque required to close the valve:

$$T_{\text{seat}} = -T_P - T_S - T_H - \Delta P_{\text{max}}(C_B - C_O)$$

Shaft upstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + T_H + \Delta P_{\text{max}}(C_B + C_O)$$

where T_{seat} = seating torque required

T_{breakout} = breakout torque required

T_P = packing torque

T_S = seat torque

T_H = handwheel torque*

ΔP_{max} = maximum pressure drop at shutoff

C_B = bearing (or guide) torque factor

C_O = off-balance torque factor

*Handwheel torque is 0 if no handwheel exists.

The packing torque T_P is the torque required to overcome the rotational friction of the packing on the shaft. The seat torque T_S is the torque required to overcome the friction of the seat on the disk. The bearing torque factor C_B indicates the relationship that as the pressure across the valve increases, the force on the bearing increases proportionally. The handwheel torque T_H is the torque required to overcome the friction of an attached handwheel. If a declutchable handwheel is used, this factor is considered only when the handwheel is in gear. The off-balance torque factor C_O shows the relationship of the off-balance forces between the disk and the mechanical connection in the actuator (which converts the actuator force to torque). Because these torques

and factors vary according to individual valve designs, they are provided by the manufacturer. If the final torque value is a negative value, this indicates that the butterfly disk will have a tendency to resist closing. Conversely, if the value is positive, the disk will have a tendency to resist opening.

When a high pressure drop is expected at any part of the quarter-turn stroke, the net torque output can vary dramatically throughout the shaft rotation. For this reason, the dynamic torque is calculated at various degrees of opening. When the shaft is downstream, a reversal of torque takes place at approximately 75 percent open, which can lead to control problems with the valve. If this happens, the user has the choice of changing the orientation of the shaft to shaft upstream (if possible), or placing the limit stops on the actuator to prevent rotation beyond 70 percent.

The following equations are used when calculating the dynamic torque for butterfly valves in gas services:

To close the valve:

$$T_D = -T_P - \Delta P_{\text{eff}}(C_{BT})$$

To open the valve:

$$T_D = T_P + \Delta P_{\text{eff}}(C_{BT})$$

where T_D = dynamic torque

T_P = packing torque value (from manufacturer)

$\Delta P_{\text{eff}} = \Delta P_{\text{actual}}$ at the flowing condition at the degree of opening
(limited to the ΔP_{choked})

C_{BT} = bearing or guide torque value (from manufacturer)

For liquid applications, the following equations are used:

To close the valve:

$$T_D = -T_P - \Delta P_{\text{eff}}(C_{BT} - C_D)$$

To open the valve:

$$T_D = T_P - \Delta P_{\text{eff}}(C_{BT} - C_D)$$

where C_D = dynamic torque factor (from Table 8.4)

Table 8.4 Butterfly-Valve Torque Factors*,†,‡

Valve Size (in.)	Dynamic Torque Factor vs. Disc Position (Degrees Open)																							
	10°		20°		30°		40°		50°		60°		70°		80°		90°							
	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B						
2	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0						
3	0	0	0	0	0	0	0	1	0	2	0	2												
4	0	0	0	0	0	0	0	0	0	1	1	2	1	4	0	6	-4	4						
6	0	0	0	1	1	1	1	3	2	4	4	9	6	17	0	23	-19	19						
8	0	0	0	1	1	2	2	4	3	7	4	12	1	19	-8	28	-23	23						
10	0	1	1	2	2	6	5	9	7	17	8	28	4	46	-18	65	-55	55						
12	1	2	2	7	5	13	10	20	16	38	18	61	8	101	-40	144	-121	121						
14	1	3	3	10	6	18	15	28	22	54	25	85	12	142	-56	201	-168	168						
16	2	4	4	12	8	23	19	36	27	67	32	106	15	177	-69	250	-211	211						
18	5	10	10	29	19	54	44	83	64	156	73	247	34	411	-161	582	-489	489						
20	5	10	10	31	21	58	47	89	68	168	78	264	6	440	-173	623	-524	524						
24	7	14	14	42	28	78	64	121	92	227	106	358	50	596	-234	845	-710	710						
30	24	49	49	146	97	268	219	414	316	779	365	1230	170	2046	-804	2899	-2436	2436						

*A-shaft downstream; B = shaft upstream.

†Courtesy of Valtek International.

‡Note: When degrees of opening are not known, use highest value of C_d for valve size.

If the final number for the dynamic torque value is a negative number, the disk will resist closing with the flow moving the disk toward the open position. If the dynamic torque number is positive, the disk will resist opening—with the flow moving the disk toward the closed position. From the manufacturer's data and given the necessary available air supply, an actuator with sufficient torque can then be selected. This torque must overcome the seating and breakout torques—as well as the dynamic torque, which is required through the entire stroke of the valve. If the actuator's available torque is less than the dynamic torque, a larger actuator size with more torque force should be selected.

Following selection of the actuator, stiffness should again be checked to prevent the disk from slamming into the seat for those applications with the shaft downstream.

Consideration should be given to whether a spring is necessary to move the disk to a particular failure position (fail-open or fail-closed). For fail-closed applications that do not require a high degree of shut-off, the spring must have adequate torque to overcome the dynamic torque. If the valve requires tight shutoff, the spring must generate enough torque to overcome the required seating torque at the closed position. For fail-open applications, the spring must have enough torque to overcome the required breakout torque at the closed position, as well as to overcome any dynamic torque as it moves through the full stroke to the full-open position. If the spring is incapable of producing enough force to overcome the seating or breakout or dynamic torque, a volume tank could be specified to ensure adequate force to move the valve to the correct position upon loss of air supply.

8.2.3 Actuator Force Calculation for Ball Valves

Because of the ball valve's design with the ball moving into the flow stream, as opposed to a disk that is already in the flow stream, the forces acting on the ball valve (and the torques required) are somewhat different. This requires a different set of torque calculations for seating or breakout:

Shaft downstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + \Delta P_{\text{max}}(C_B + C_S) + T_H$$

Shaft upstream, torque required to open the valve:

$$T_{\text{breakout}} = T_P + T_S + \Delta P_{\text{max}} C_B + (T_S - \Delta P C_S) + T_H$$

where ΔP = actual pressure drop ($P_1 - P_2$)

C_S = seat torque factor

The packing torque T_P is the torque required to overcome the rotational friction of the packing on the shaft. The seat torque T_S is the torque required to overcome the friction of the seat on the disk. The bearing torque factor C_B shows the relationship that as the pressure across the valve increases, the force on the bearing increases proportionately. The handwheel torque T_H is the torque required to overcome the friction of an attached handwheel. If a declutchable handwheel is specified with the actuator, this factor is only considered when the handwheel is in gear. Because these torques and factors vary widely due to design differences, they are usually determined by the manufacturer.

With liquid services, the dynamic torque must also be calculated. As noted in the previous section, dynamic torque is the torque required to overcome the torque on the closure element caused by the fluid dynamic forces on the ball. To calculate dynamic torque, the following equation is used:

$$T_D = T_P + \Delta P_{\text{eff}}(C_D + C_B)$$

where ΔP_{eff} = actual pressure drop across the valve at the flowing condition that occurs when the valve is in the open position (ΔP_{eff} is less than or equal to ΔP_{choked})

C_D = dynamic torque factor (from Table 8.5)

C_B = bearing torque factor (from manufacturer)

Once the seating or breakout and dynamic torques have been calculated, the correct actuator with sufficient torque is then chosen from the manufacturer's tables.

If a spring is required to move the ball to a particular failure position (fail-open or fail-closed), special consideration should be given to sizing the correct spring that can overcome the process forces. For fail-closed applications that do not require a high degree of shutoff, the spring must have adequate torque to overcome the dynamic torque. If the ball valve requires tight shutoff, the spring must generate enough torque to overcome the required seating torque at the closed position. For fail-open applications, the spring must have enough torque to overcome the required breakout torque at the closed position as well as to overcome any dynamic torque as it moves through the full stroke

Table 8.5 Ball-Valve Torque Factors*†

Valve Size (in.)	T _p =Packing Torque (in-lb)					T _s =Seat Torque (in-lb)	C _b =Bearing Torque Factor		C _s =Seat Torque Factor	C _d = Dynamic Torque Factor	
	(1)	(2)	(3)	(4)	(5)		(6)	(7)		60°	90°
1	43	228	421	301	57	20	0.06	0.09	0.1	0.25	0.6
1½	50	280	477	350	63	40	0.06	0.09	0.1	0.5	1.0
2	50	280	477	350	63	60	0.06	0.09	0.15	1.0	2.1
3	57	333	533	399	71	150	0.19	0.28	0.42	4.5	8.0
4	57	333	533	399	71	360	0.38	0.58	0.82	10.0	17.0
6	71	438	646	496	92	540	0.97	1.45	1.64	19.5	30.5
8	71	438	646	496	92	670	1.58	2.37	2.62	52.0	75.5
10	104	648	870	691	151	1100	4.38	6.57	4.55	108.0	165.5
12	104	648	870	691	151	1300	5.61	8.41	6.05	191.0	218.5

*Courtesy of Valtek International.

†(1)PTFE or filled PTFE V-ring packing, (2) grafoil, (3) twin grafoil, (4) asbestos-free packing (AFP), (5) braided PTFE, (6) PTFE lined bearings, (7) metal bearings.

to the full-open position. If the available springs are not capable of producing enough force to overcome the seating, breakout, or dynamic torque, a volume tank should be specified to ensure adequate force to move the valve to the correct position upon loss of air supply.

8.3 Sizing Electromechanical and Electrohydraulic Actuators

8.3.1 Introduction to Actuator Sizing for Electromechanical and Electrohydraulic Actuators

For the most part, electromechanical and electrohydraulic actuators are sized according to the thrust needed to overcome the forces inside the body as shown in the following equation from Sec. 8.1.2:

$$F_{\text{total}} = F_{\text{process}} + F_{\text{seat}} + F_{\text{packing}} + F_{\text{miscellaneous}}$$

where F_{total} = total force (or actuator thrust) required to open, close, or throttle valve

F_{process} = force to overcome process pressure unbalance

F_{packing} = force required to overcome packing friction

F_{seat} = force to provide correct seat load

$F_{\text{miscellaneous}}$ = force to overcome special design factors, weight, etc.

Individual sizing equations to determine actuator size vary widely

depending on the design and application of the actuator and are not specifically included in this section.

8.3.2 Special Considerations

Typically, the application engineers for the electromechanical or electrohydraulic manufacturer will size the actuator based upon the process and frictional forces associated with the valve as well as include some additional thrust for safety considerations. With the high level of engineering required for these actuators, the prevailing thought is better too much actuator than not enough. However, in some cases, the accumulation of safety factors over the sizing process can add anywhere from 25 to 50 percent to the total thrust of the actuator. High costs are associated normally with electromechanical and electrohydraulic actuators. Therefore, if sizing formulas show the thrust requirement to be slightly more than a given size, all safety factors should be reconsidered to check for an impractical accumulation. If that is the case, the smaller actuator size can be considered.

Electromechanical and electropneumatic actuators are normally specified for those applications requiring faster stroking speeds or higher performance than provided normally by pneumatic actuators.

From a sizing standpoint, application engineers use specialized sizing equations to determine the stroking speed, frequency response, and level of precision positioning. Because most applications requiring electromechanical and electrohydraulic actuators are special or severe, services, manufacturers have a tendency to size actuators based on flow rate and pressure drop.