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Valve Sizing

7.1 Introduction to Valve Sizing

7.1.1 The Importance of Correct Valve Sizing

Generally, valve sizing is based on the standard thermodynamic laws of fluid flow. The application of these laws is affected by the particular function of the valve plus the type and severity of the service. Simple on-off block valves are expected to pass nearly 100 percent of the flow without a significant pressure drop, since they are not expected to control the flow other than to shut it off. On the other hand, throttling services are expected to produce a certain amount of flow at certain positions of opening and take a particular pressure drop. Therefore, the science of valve sizing is almost always directed toward sizing throttling valves.

With manually operated on-off block valves, the valve is often expected to pass full flow. If the valve's internal flow passage or closure element is sized smaller than the upstream piping, flow will be restricted from that point forward. This will cause the valve to take a pressure drop and pass less flow, defeating the major purpose of the on-off valve. If the on-off block valve is sized larger than the upstream piping, installation costs are more expensive (since increasers are required). The larger valve is also more expensive. On the other hand, throttling valves, which are intended to take a pressure drop and to reduce the flow, may have a seat that is significantly less in diameter than the upstream port. Determining the flow through this diameter is the science behind valve sizing. If a throttling valve is sized too small, the maximum amount of flow through the valve will be limited and will inhibit the function of the system. If a throttling valve is sized too large, the user must bear the added cost of installing

a larger valve. Another major disadvantage is that the entire flow control may be accomplished in the first half of the stroke, meaning that a minor change in position may cause a large change in flow. In addition, because regulation occurs in the first half of the stroke, flow control is extremely difficult when the regulating element is operating close to the seat. The ideal situation is for the throttling valve to utilize the full range of the stroke while producing the desired flow characteristic and maximum flow output.

Throttling valves are rarely undersized because of the number of safety factors built into the user's service conditions and the manufacturer's sizing criteria. Because of these safety factors, a large number of throttling valves actually end up being oversized. This happens because the user provides a set of service conditions that are usually the maximum conditions of the service (temperature, pressure, flow rate, etc.). The manufacturer then adds its own safety factors into the sizing equations. The valve manufacturer does this to avoid the error of undersizing, which is less forgiving than oversizing. Although not ideal, an oversized valve is still workable.

7.1.2 Valve-Sizing Criteria for Manual Valves

The basic function of manual on-off block valves is quite simple: to pass full flow while the valve is open or to shut off or divert the full flow when closed. Therefore, the valve size can sometimes be determined simply by the size of the piping, which has already been sized by the system engineers. Manual-valve manufacturers often provide sizing charts that indicate the relationship between the flow-rate requirement (Q) are the minimum and maximum valve size that can pass the given flow rate.

An important choice in manual-valve sizing is whether the valve should be full bore or reduced bore. In many cases this is more a function of the valve's purpose to pass full flow or to take a slight pressure drop. If the valve is installed in an application that must allow the passage of a pig to clean or scour the pipeline, the valve chosen must be full bore, since the pig is the same size as the inside diameter of the pipe. Another application calling for full-bore manual valves is one installed in slurries or services with entrained materials or particulates. If the valve has a reduced bore, these particulates or slurries have a tendency to settle and become trapped at the narrowed constriction. A full-bore valve has no such restriction, allowing for free passage of the foreign material without collection. Full-bore manual

valves are also chosen for services with high velocities, for which a restriction would increase the chance of erosion as well as increase the velocity further.

The service conditions generally required for correct manual-valve sizing are maximum and minimum temperatures, pressures, flow rates, and specific volume (steam applications). Not only are the extremes important, but also the average operating conditions are important. The specific volume is normally provided to the user by commonly published steam tables, which show the specific volume in cubic feet per pound. Most steam tables provide the data in *pounds per square inch absolute* (psia), which does not take atmospheric pressure (14.7 psi or 1.03 bar) into account. On the other hand, *pounds per square inch gage* (psig) accounts for this adjustment for the atmospheric pressure. The metric equivalent for psig is *barg*.

7.1.3 Valve-Sizing Criteria for Check Valves

The most critical element of check-valve sizing is that a sufficient pressure drop and minimum flow exist for the check valve to open. Without a pressure drop, the closure element will not open and the valve will remain closed, which is what happens when a pump fails to maintain a proper flow or flow reverses.

The minimum pressure drop required for check valves to open is typically 1 psi (0.07 bar). This minimum pressure drop is needed to maintain the open position of the closure element without failing. If the pressure drop falls to less than 1 psi, the closure element will float back and forth, which is commonly called “flutter.” As the disk moves toward the seat, the opening narrows and pressure rebuilds, which causes the disk to open higher. This low-pressure drop situation will cause this cycle to repeat until the pressure drop is increased, causing wear of the moving parts and shortening the life of the check valve. The maximum pressure drop is approximately 10 psi (0.7 bar), depending on the size of the check valve. Higher pressure drops lead to severe erosion of the check valve’s closure element.

Check-valve manufacturers provide the cracking pressure of their check valves. The cracking pressure is the minimum pressure required to open the check valve and is a fixed number associated with the style and size of the check valve. It can vary anywhere from 0.1 to 0.5 psi (0.01 to 0.03 bar). Generally, cracking pressures are of little concern unless the pressures in the process are extremely low or the pressure drop is small (less than 1 psi). However, the cracking pressure can be

important if the valve is installed in a vertical line, where the check valve must open against gravitational forces in addition to the process pressure. Smaller lines have higher cracking pressures than larger lines. This is because the larger the line, the larger the process force must be against the component's mass in the check valve.

Unless the flow experiences a wide range of flow during the service, check valves are sized for minimum flow, which in turn determines the valve size. This is done using manufacturer's sizing charts. If the size provided for the minimum flow is equivalent to or greater than the pipeline size, the pipeline size should be used for the valve size. For example, if the manufacturer's literature calls for a 4-in check valve, yet the pipe size is 3-in line, a 3-in check valve should be satisfactory. The larger, oversized valve will not benefit the flow rate yet is more expensive and would require the installation of increasers. If the suggested valve size for the minimum flow is smaller than the pipeline, reducers must be installed and the smaller-sized check valve installed.

The user should ensure that the flow rates are within the parameters of the check-valve design. High flow rates can increase the frequency of vortices and currents, which will increase the pressure drop across the valve as well as cause valve wear. Insufficient flow will cause the valve to flutter. The flow must be sufficient to overcome the closed position of the check valve—whether it be gravity, weight of the closure element, line orientation, or spring force.

As a general rule, the maximum liquid flow velocity for check valves is 11 ft/s (3.4 m/s). The minimum liquid flow velocity is normally 6 to 7 ft/s (1.8 to 2.1 m/s), although some designs (such as a double-disk check valve) can operate at 3 ft/s (0.9 m/s).

7.1.4 Valve-Sizing Criteria for Throttling Valves

Throttling valves require a systematic method of determining the required flow through the valve, as well as the size of the valve body, the body style, and materials that can accommodate (or tolerate) the process conditions, the correct pressure rating, and the proper installed flow characteristic. The industry standard for determining the flow capacity of a throttling valve is ANSI/ISA Standard S75.01, which contains the equations required to predict the flow of incompressible (liquid) and compressible (gas) process fluids. Because of the compressibility issues between liquids and gases, equations have been formulated for each and are included in this chapter.

Proper selection of the valve is based on the service conditions of the process. For correct sizing, the following conditions are needed: the upstream pressure; the maximum and minimum temperatures; the type of process fluid; the flow rate that is based upon the maximum flow rate, the average flow rate, and the minimum flow rate; vapor pressure; pipeline size, schedule, and material; the maximum, average, and minimum pressure drop; specific gravity of the fluid; and the critical pressure.

7.2 Valve-Sizing Nomenclature

7.2.1 Upstream and Downstream Pressures

In process systems, most valves are designed to either pass or restrict the flow to some extent. In order for the process to flow in a particular direction through a valve, the upstream and downstream pressures must be different; otherwise, the pressure would be equal and no flow would occur. By definition, the *upstream pressure* is the pressure reading taken before the valve, while the *downstream pressure* is the pressure reading taken after the valve.

7.2.2 Pressure Drop

The resulting difference between the upstream and downstream pressures is called the *pressure drop* (or the *pressure differential*). The pressure drop allows for the flow of fluid through the process system from the upstream side of the valve to the downstream side. In theory, the greater the pressure drop, the greater the flow through the valve.

7.2.3 Flow Capacity

The most commonly applied sizing coefficient is known as the *valve coefficient* (C_v), which is defined as one U.S. gallon (3.8 liters) of 60°F (16°C) water that flows through a valve with 1.0 psi (0.07 bar) of pressure drop. This general equation is written several ways, but two of the most common methods are

$$C_v = Q \sqrt{\frac{S_g}{\Delta P}}$$

or

$$C_v = Q \sqrt{\frac{\Delta P}{S_g}}$$

where C_v = required flow coefficient for the valve

Q = flow rate (in gal/min)

S_g = specific gravity of the fluid

ΔP = pressure drop (psi)

When calculated properly, C_v determines the correct trim size (or area of the valve's restriction) that will allow the valve to pass the required flow while allowing stable control of the process throughout the stroke of the valve.

7.2.4 Actual Pressure Drop

Another term for pressure drop, *actual pressure drop* (ΔP), is defined as the difference between the upstream (inlet) and downstream (outlet) pressures. When the choked and actual pressure drops are compared and the actual pressure drop is smaller, it is used in the C_v sizing equation.

7.2.5 Choked Pressure Drop

As the C_v equation is examined, the assumption is made that if the pressure drop is increased, the flow should increase proportionately. A point exists, however, where further increases in the pressure drop will not change the valve's flow rate. This is what is commonly called *choked flow*.

As illustrated in Fig. 7.1, with liquid applications having a constant upstream pressure, the flow rate Q is related to the square root of the pressure drop with a proportional and constant C_v . When the valve begins to choke, the flow-rate curve falls away from the linear relationship. Because of the choked condition, the flow rate will reach a maximum condition due to the existence of cavitation in liquids or sonic velocity with gases.

Depending on the valve style, this departure from the linear relationship will occur at different regions of the line, with some being more gradual and others more abrupt. For example, globe-style valves tend to handle higher pressure drops without choking, as opposed to rotary valves, which tend to choke and cavitate at smaller pressure drops.

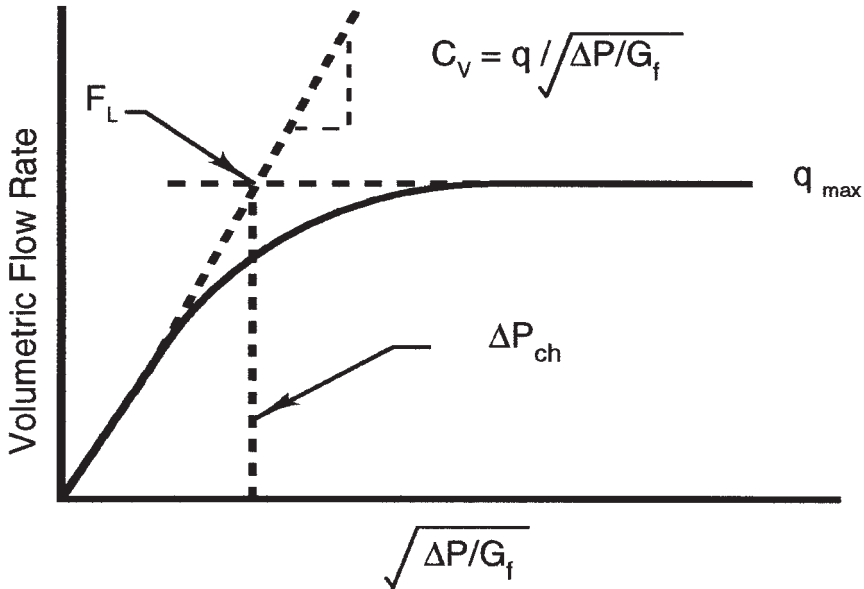


Figure 7.1. Maximum flow rate occurring due to choked conditions. (Courtesy of Valtek International)

For simplicity, the term *choked pressure drop* ΔP_{choked} is used to show the theoretical point where choked flow occurs, intersecting the linear lines of the constant C_v and the maximum flow rate Q_{max} . This point is known as the liquid pressure-recovery factor F_L , which is discussed in more detail in this section. The ANSI/ISA sizing equations for liquids use F_L to calculate the theoretical point where choked flow occurs (ΔP_{max}) so that the valve can be sized without the difficulty of the process being choked.

For gas applications, the terminal pressure-drop ratio x_T is used to describe the choked pressure drop for a particular valve.

7.2.6 Allowable Pressure Drop

The *allowable pressure drop* ΔP_a is chosen from the smaller of the actual pressure drop or the choked pressure drop and is used in the determination of the correct C_v . When determining the C_v of a liquid application, the following must be considered to see if the allowable pressure drop should be used: first, if the inlet pressure P_1 is fairly close to the vapor pressure; second, if the outlet pressure P_2 is fairly close to the vapor pressure; and third, if the actual pressure drop is fairly large when compared to the inlet pressure P_1 . If any one of the above three conditions

exists, the user should calculate the allowable pressure drop and compare it against the actual pressure drop, using the smaller value.

7.2.7 Incipient and Advanced Cavitation

With liquid applications, when the fluid passes through the narrowest point of the valve (vena contracta), the pressure decreases inversely as the velocity increases. If the pressure drops below the vapor pressure for that particular fluid, vapor bubbles begin to form. As the fluid moves into a larger area of the vessel or downstream piping, the pressure recovers to a certain extent. This increases the pressure above the vapor pressure, causing the vapor bubbles to collapse or implode. This two step-process—creation of the vapor bubbles and their subsequent implosion—is called *cavitation* and is a leading cause of valve damage in the form of erosion of metal surfaces.

As the pressure drops, the point where vapor bubbles begin to form is called *incipient cavitation*. The pressure level where cavitation is occurring at its maximum level is called *advanced cavitation*. During advanced cavitation, the flow is choked and cannot increase, which affects the flow capacity of the valve as well as its function. The point where advanced cavitation occurs can be predicted. To do this, the pressure drop must be determined, using the liquid cavitation factor F_L . A detailed discussion about the causes and effects of cavitation is found in Sec. 9.2.

7.2.8 Flashing Issues

When the downstream pressure does not recover above the vapor pressure, the vapor bubbles remain in the fluid and travel downstream from the valve, creating a mixture of liquid and gas. This is called *flashing*. Problems typically associated with flashing are higher velocities and erosion of valve components. Section 9.3 provides a more detailed discussion about flashing and its effects.

7.2.9 Liquid Pressure-Recovery Factor

A critical element in liquid sizing is the *liquid pressure-recovery factor* F_L , which predicts the effect the geometry of a valve's body will have on the maximum capacity of that valve. F_L is used to predict the amount of pressure recovery occurring between the vena contracta and the outlet of the body.

The liquid pressure-recovery factor is determined by the manufacturer through flow testing that particular valve style. F_L factors can vary significantly depending on the internal design of the valve. Valves from the same basic design (for example, butterfly valves) may have varying F_L factors depending on the unique internal designs of the manufacturer. Generally, rotary valves, especially ball and butterfly valves, allow for a high recovery of the fluid following the vena contracta. Therefore, they tend to cavitate and choke at smaller pressure drops than globe valves. For the most part, globe valves have better F_L factors and are able to handle severe services when compared to rotary valves.

7.2.10 Liquid Critical-Pressure Ratio Factor

The *liquid critical-pressure ratio factor* F_F is important to liquid sizing because it predicts the theoretical pressure at the vena contracta, when the maximum effective pressure drop (or in other words, the choked pressure drop) occurs across the valve.

7.2.11 Choked Flow

With liquid services, the presence of cavitation or flashing expands the specific volume of the fluid. The volume increases at a faster rate than if the flow increased due to the pressure differential. At this point, the valve cannot pass any additional flow, even if the downstream pressure is lowered.

With gas and vapor services, choked flow occurs when the velocity of the fluid achieves sonic levels (Mach 1 or greater) at any point in the valve body or downstream piping. Following the basic laws of mass and energy, as the pressure decreases in the valve to pass through restrictions, velocity increases inversely. As the pressure lowers, the specific volume of the fluid increases to the point where a sonic velocity is achieved.

Because of the velocity limitation [Mach 1 for gases and 50 ft/s (12.7 m/s) for liquids], the flow rate is limited to that which is permitted by the sonic velocity through the vena contracta or the downstream piping.

7.2.12 Velocity

As a general rule, smaller valve sizes are better equipped to handle higher velocities than larger-sized valves, although the actual sizes

vary according to the valve style. For liquid services, the general guideline for maximum velocity at the valve outlet is 50 ft/s (12.7 m/s), while gas services are generally restricted to Mach 1.0. When cavitation or flashing is present, creating a higher velocity associated with the liquid–gas mixture, the maximum velocity is usually restricted to 500 ft/s (127 m/s). Some exceptions exist, however, for liquids. In services where temperatures are close to saturation point, the velocity must be less—approximately 30 ft/s (7.6 m/s). This lower velocity prevents the fluid from dropping below the vapor pressure, which will lead to the formation of vapor bubbles. The rule of 30 ft/s (7.6 m/s) is also applicable to those valve applications that must have a full flow rate with minimal pressure drop. A valve in which the pressure drop falls below the vapor pressure and advanced cavitation is occurring should be restricted to 30 ft/s (7.6 m/s) to minimize the cavitation damage that would spread from the valve into the downstream piping. Ideally, the user would try to restrict the pressure recovery and allow the subsequent cavitation damage to be contained in the body and not downstream into the piping. In essence, the valve body is sacrificed and the piping is saved.

7.2.13 Reynolds-Number Factor

Some processes are characterized by nonturbulent flow conditions in which laminar flow exists (such as oils). Laminar fluids have high viscosity, operate in lower velocities, or require a flow capacity requirement that is extremely small. The *Reynolds-number factor* F_R is used to correct the C_v equation for these flow factors. In most cases, if the viscosity is fairly low (for example, less than SAE 10 motor oil), the Reynolds-number factor is insignificant.

7.2.14 Piping-Geometry Factor

The flow capacity of a valve may be affected by nonstandard piping configurations, such as the use of increasers or reducers, which must be corrected in the C_v equation using the *piping-geometry factor* F_P .

Standardized C_v testing is conducted by the valve manufacturer with straight piping that is the same line size as the valve. The use of piping that is larger or smaller than the valve, or the close proximity of piping elbows, can decrease these values and must be considered during sizing.

7.2.15 Expansion Factor

With gas services, the specific weight of the fluid varies as the gas moves from the upstream piping and through the valve to the vena contracta. The *expansion factor* Y is used to compensate for the effects of this change in the specific weight of the gas. The expansion factor is important in that it takes into account the changes in the cross-sectional area of the vena contracta as the pressure drop changes in that region.

7.2.16 Ratio of Specific Heats Factor

Because the C_v equation for gases is based upon air, some adjustment must be made for other gases. The *ratio of specific heats factor* F_K is used to adjust the C_v equation to the individual characteristics of these gases.

7.2.17 Terminal Pressure-Drop Ratio

With gases, the point where the valve is choked (which means that increasing the pressure drop though lowering the downstream pressure cannot increase the flow of the valve) is predicted by the *terminal pressure-drop ratio* x_T . Similar in many respects to the liquid pressure-recovery factor F_L , the terminal pressure-drop ratio is affected by the geometry of the valve's body and varies according to valve style and individual size.

7.2.18 Compressibility Factor

Because the density of gases varies according to the temperature and pressure of the fluid, the fluid's compressibility must be included in the C_v equation. Therefore the *compressibility factor* Z is included in the equation and is a function of the temperature and pressure.

7.3 Body Sizing of Liquid-Service Control Valves

7.3.1 Basic Liquid Sizing Equation

The liquid C_v sizing equation is a general-purpose equation for most liquid applications, using the actual pressure drop (upstream pressure

minus downstream pressure) to calculate the flow capacity. For non-laminar liquids,

$$C_v = \frac{Q}{F_p} \sqrt{\frac{S_g}{\Delta P_a}}$$

where C_v = valve-sizing coefficient

F_p = piping geometry function

Q = flow rate (gal/min)

S_g = specific gravity (at flowing temperature)

ΔP_a = allowable pressure drop across the valve (psi)

For sizing purposes, the liquid C_v equation can be determined step by step by following Secs. 7.3.2 through 7.3.14.

7.3.2 Actual-Pressure-Drop Calculation

Before the allowable pressure drop is determined, the actual pressure drop should be determined by using the following equation:

$$\Delta P = P_1 - P_2$$

where ΔP = actual pressure drop (psi)

P_1 = upstream pressure (at valve inlet, psia)

P_2 = downstream pressure (at valve outlet, psia)

7.3.3 Choked Flow, Cavitation, and Flashing Determination

The choked flow point must be predicted using the following equation:

$$\Delta P_{\text{choked}} = F_L^2(P_1 - F_F P_v)$$

where ΔP_{choked} = choked pressure drop

F_L = liquid pressure-recovery factor

F_F = liquid critical-pressure-ratio factor

P_v = vapor pressure of the liquid (at inlet temperature, psia)

The liquid pressure-recovery factor F_L is usually provided by the manufacturer. Table 7.1 provides typical F_L values for throttling linear globe and rotary valves.

Table 7.1 Typical F_L Factors^{*,†}

Valve Style	Flow Direction	Trim Area	F_L
Linear globe	Over seat	Full area	0.85
	Over seat	Reduced area	0.80
	Under seat	Full area	0.90
	Under seat	Reduced area	0.90
Butterfly	60° open	Full area	0.76
	90° open	Full area	0.56
Ball	60° open	Full area	0.78
	90° open	Full area	0.66

^{*}Data courtesy of Valtek International.

[†]Note: All values provided are full-open.

Continuing with the ΔP_{choked} equation, the liquid critical-pressure ratio factor F_F is determined by using the following equation:

$$F_F = 0.96 - 0.28 \sqrt{\frac{P_V}{P_C}}$$

where P_C = critical pressure of the liquid (psia)

Critical pressures for common gases and liquids are found in Table 7.2.

If the calculation for the choked pressure drop ΔP_{choked} is a smaller value than the actual pressure drop ΔP , the ΔP_{choked} value should be used for the actual pressure drop ΔP_a in the C_v equation. To determine at what pressure drop advanced cavitation begins, the following equation should be used:

$$\Delta P_{\text{cavitation}} = F_i^2(P_1 - P_V)$$

Table 7.2 Critical Pressures for Common Process Fluids

Liquid	Critical Pressure <i>psia/bar</i>
Ammonia	1636.1/112.8
Argon	707.0/48.8
Benzene	710.0/49.0
Butane	551.2/38.0
Carbon Dioxide	1070.2/73.8
Carbon Monoxide	507.1/35.0
Chlorine	1117.2/77.0
Dowtherm A	547.0/37.7
Ethane	708.5/48.8
Ethylene	730.5/50.3
Fuel Oil	330.0/22.8
Gasoline	410.0/28.3
Helium	32.9/2.3
Hydrogen	188.1/13.0
Hydrogen Chloride	1205.4/83.1
Isobutane	529.2/36.5
Isobutylene	529.2/36.5
Kerosene	350.0/24.1
Methane	667.3/46.0
Nitrogen	492.4/33.9
Nitrous Oxide	1051.1/72.5
Oxygen	732.0/50.5
Phosgene	823.2/56.8
Propane	615.9/42.5
Propylene	670.3/46.2
Refrigerant 11	639.4/44.1
Refrigerant 12	598.2/41.2
Refrigerant 22	749.7/51.7
Seawater	3200.0/220.7
Water	3208.2/221.2

Table 7.3 Typical F_i Factors^{*,†}

Valve Style	Flow Direction	Trim Area	F_i
Linear globe	Over seat	Full area	0.75
	Over seat	Reduced area	0.72
	Under seat	Full area	0.81
	Under seat	Reduced area	0.81
Butterfly	60° open	Full area	0.65
	90° open	Full area	0.49
Ball	60° open	Full area	0.65
	90° open	Full area	0.44

^{*}Data courtesy of Valtek International.

[†]Note: All values provided at full open.

where $\Delta P_{\text{cavitation}}$ = pressure drop with advanced cavitation
 F_i = liquid cavitation factor

Typical liquid cavitation factors for common valve styles are found in Table 7.3.

7.3.4 Specific-Gravity Determination

The value for the fluid's specific gravity S_g should be determined using the operating temperature and a reference table for specific-gravity data.

7.3.5 Approximate-Flow-Coefficient Calculation

Using the values calculated to this point, the approximate flow capacity should be calculated, using the C_v sizing equation for liquids from

Sec. 7.3.1. For this calculation, the assumption should be made that the piping-geometry factor F_p is 1.0. When the valve is not operating in a laminar flow—due to high viscosity, low velocity, or low flow—the effects of nonturbulent flow can be ignored.

7.3.6 Approximate Body Size Selection

Using the manufacturer's C_v tables, the smallest-sized body that can accommodate the calculated C_v should be selected. Typical C_v data are found in Fig. 7.2.

7.3.7 Reynolds-Number-Factor Calculation

The following equation can be used to determine the Reynolds-number factor:

$$\text{Re}_V = \frac{N_4 F_d Q}{\nu \sqrt{F_L C_v}} \left(\frac{F_L^2 C_v^2}{N_2 d^4} + 1 \right)^{0.25}$$

where Re_V = valve Reynolds number

N_4 = 17,300 (when Q is in gal/min and d in inches)

F_d = valve style modifier (see Table 7.4)

ν = kinematic viscosity (centistokes, μ/S_g)

C_v = valve flow coefficient (from Sec. 7.3.1)

N_2 = 890 (when d is in inches)

d = valve inlet diameter (inches)

Valve Type: Mark One, Unbalanced Body Rating: Class 900-1500 Trim Characteristics: Quick Open Flow Direction: Flow Over														
SIZE	TRIM NO	STROKE	F_L	PERCENT OPEN										SEAT AREA
				100	90	80	70	60	50	40	30	20	10	
1.00	.81	.75	.87	9.0	8.9	8.9	8.7	8.6	8.5	7.5	5.7	3.5	1.9	.52
1.50	1.25	1.00	.85	24	24	24	24	24	21	18	13	8.7	4.9	1.23
2.00	1.62	1.50	.87	41	41	40	40	39	39	34	26	15	8.1	2.06
3.00	2.00	2.00	.86	106	105	105	104	104	94	81	62	39	21	5.41
4.00	3.50	2.50	.87	185	185	183	181	178	162	139	105	68	37	9.62
6.00	5.00	3.00	.85	382	382	381	380	355	317	270	210	140	75	19.63

Note: All C_v values are shown at 100% open. For each valve size below, the full area values are shaded. Reduced trim values follow, in descending order.

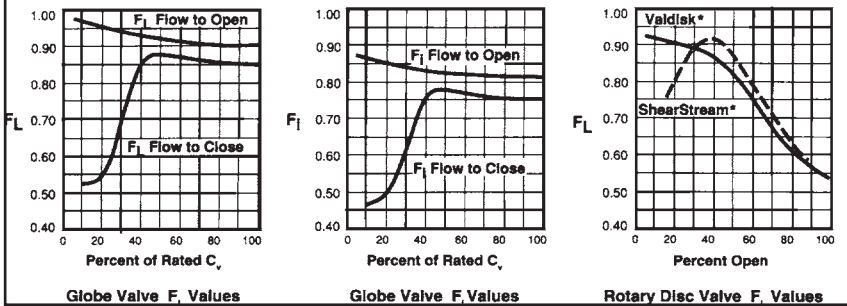
Figure 7.2. Typical manufacturer's C_v data. (Courtesy of Valtek International)

Table 7.4 Valve Recovery Coefficient and Incipient Cavitation Factors*

Valve Type	Flow Direction	Trim Size	F_L	F_i	x_T	F_d
Globe	Over Seat	Full Area	0.85	0.75	.70	1.0
	Over Seat	Reduced Area	0.80	0.72	.70	1.0
	Under Seat	Full Area	0.90	0.81	.75	1.0
	Under Seat	Reduced Area	0.90	0.81	.75	1.0
Valdisk	60° Open	Full	0.76	0.65	.36	.71
Rotary Disc	90° Open	Full	0.56	0.49	.26	.71
ShearStream	60° Open	Full	0.78	0.65	.51	1.0
Rotary Ball	90° Open	Full	0.66	0.44	.30	1.0
CavControl	Over Seat	All	0.92	0.90	N/A	$.2/\sqrt{d}$
MegaStream	Under Seat	All	~1.0	N/A	~1.0	$(n_s/25d)^{2/3}$ **
ChannelStream	Over Seat	All	~1.0	0.87 to 0.999	N/A	.040*
Tiger-Tooth	Under Seat	All	~1.0	0.84 to 0.999	~1.0	.035*

* Typical ** n_s = number of stages

NOTE: Values are given for full-open valves. See charts below for part-stroke values



*Courtesy of Valtek International.

If the valve Reynolds number (Re_v) is equal to or greater than 40,000 ($Re_v \geq 40,000$), 1.0 should be used for the Reynolds-number factor F_R . The following equation is used to find F_R if the valve Reynolds number is less than 40,000 ($Re_v \leq 40,000$):

$$F_R = 1.044 - 0.358 \left(\frac{C_{vS}}{C_{vT}} \right)^{0.655}$$

through use of these equations:

$$C_{vS} = \frac{1}{F_S} \left(\frac{Q\mu}{N_S \Delta P} \right)^{0.667}$$

$$F_S = \frac{F_d^{0.667}}{F_L^{0.333}} \left(\frac{F_L^2 C_v^2}{N_2 d^4} + 1 \right)^{0.167}$$

where C_{vS} = laminar flow C_v
 C_{vT} = turbulent flow C_v (the C_v is used from the liquid C_v equation in Sec. 7.3.1)
 F_S = laminar or streamline flow factor
 N_2 = 890 (when d is in inches)
 N_4 = 17,300 (when Q is in gal/min and d is in inches)
 N_5 = 47 (when Q is in gal/min and ΔP is in psi)
 μ = absolute viscosity (centipoise)

7.3.8 Flow-Coefficient Recalculation

Flow is considered to be laminar when the Reynolds-number factor F_R is less than 0.48 ($F_R < 0.48$). That means that the C_v is the same as the $C_{vS'}$ which is determined from the equation in Sec. 7.3.7.

If F_R is larger than 0.98 ($F_R > 0.98$), the flow is determined to be turbulent and assumed to be equal to 1.0 ($F_R = 1.0$). At this point, the C_v is determined from the standard C_v liquid sizing equation found in Sec. 7.3.1. The piping-geometry factor F_p is not required in this situation and should not be figured into the C_v equation.

If F_R falls between 0.48 and 0.98, the flow is determined to be in a transitional stage, which is calculated using the following equation:

$$C_v = \frac{Q}{F_R} \sqrt{\frac{S_g}{\Delta P}}$$

where F_R = Reynolds-number factor
 S_g = specific gravity (at flowing temperature)

7.3.9 Piping-Geometry-Factor Calculation

The inside diameter of the piping is required to determine the piping-geometry factor F_p . In the event that the pipe size is not provided or known, the body size determined from Sec. 7.3.6 should be used to determine the pipe size. Tables 7.5 and 7.6 can be used to find the piping-geometry factors. Table 7.7 provides F_p for valves with reducers (or increasers) on both the inlet and outlet of the valve. Table 7.8 provides F_p for a valve with the reducer (or increaser) on the valve outlet only. The maximum effective pressure drop (defined as ΔP_{choked}) can be affected by the use of increasers and reducers.

Table 7.5 Piping-Geometry Factors for Valves with Reducers and Increases on Both Ends*,†

C_v / d^2	d / D				
	0.50	0.60	0.70	0.80	0.90
4	0.99	0.99	1.00	1.00	1.00
6	0.98	0.99	0.99	1.00	1.00
8	0.97	0.98	0.99	0.99	1.00
10	0.96	0.97	0.98	0.99	1.00
12	0.94	0.95	0.97	0.98	1.00
14	0.92	0.94	0.96	0.98	0.99
16	0.90	0.92	0.95	0.97	0.99
18	0.87	0.90	0.94	0.97	0.99
20	0.85	0.89	0.92	0.96	0.99
25	0.79	0.84	0.89	0.94	0.98
30	0.73	0.79	0.85	0.91	0.97
35	0.68	0.74	0.81	0.89	0.96
40	0.63	0.69	0.77	0.86	0.95

*Courtesy of Valtek International.

†Note: The maximum effective pressure drop (ΔP choked) may be affected by the use of reducers and increasers. This is especially true of butterfly valves.

7.3.10 Final-Flow-Coefficient Calculation

After the piping-geometry factor F_p is determined, it should be applied to the liquid C_v equation (Sec. 7.3.1) and the final C_v calculated.

7.3.11 Valve Exit-Velocity Calculation

As discussed in Sec. 7.2.12, the general rule for velocities in liquids is that the velocity should be limited to 50 ft/s (15.2 m/s), although this may vary according to the size of the valve—smaller valves can handle

Table 7.6 Piping-Geometry Factors for Valves with Reducers and Increases on Outlet Only*,†

C_v / d^2	d / D				
	0.50	0.60	0.70	0.80	0.90
4	1.00	1.00	1.00	1.00	1.00
6	1.01	1.01	1.01	1.01	1.01
8	1.01	1.02	1.02	1.02	1.01
10	1.02	1.03	1.03	1.03	1.02
12	1.03	1.04	1.04	1.04	1.03
14	1.04	1.05	1.06	1.05	1.04
16	1.06	1.07	1.08	1.07	1.05
18	1.08	1.10	1.11	1.10	1.06
20	1.10	1.12	1.12	1.12	1.08
25	1.17	1.22	1.24	1.22	1.13
30	1.27	1.37	1.42	1.37	1.20
35	1.44	1.65	1.79	1.65	1.32
40	1.75	2.41	3.14	2.41	1.50

*Courtesy of Valtek International.

†Note: d = valve port inside diameter in inches; D = internal diameter of the piping in inches.

higher velocities, while larger valves handle lower velocities. To calculate the exit velocities from the valve, the following equation is used:

$$V = \frac{0.321Q}{A_v}$$

where V = velocity (ft/s)

A_v = flow area of valve body port (square inches) from Table 7.9

If the exit velocity exceeds the acceptable velocity for that given application, a larger valve size may be chosen to prevent damage from erosion. If a larger body size is chosen, the piping-geometry factor F_p will have to change, requiring a new C_v calculation.

Table 7.7 Piping-Geometry Factors, with Reducers or Increaseers on Both Inlet and Outlet of Valve*,†

C_v/d^2	0.50 d/D	0.60 d/D	0.70 d/D	0.80 d/D	0.90 d/D
4	0.99	0.99	1.00	1.00	1.00
6	0.98	0.99	0.99	1.00	1.00
8	0.97	0.98	0.99	0.99	1.00
10	0.96	0.97	0.98	0.99	1.00
12	0.94	0.95	0.97	0.98	1.00
14	0.92	0.94	0.96	0.98	0.99
16	0.90	0.92	0.95	0.97	0.99
18	0.87	0.90	0.94	0.97	0.99
20	0.85	0.89	0.92	0.96	0.99
25	0.79	0.84	0.89	0.94	0.98
30	0.73	0.79	0.85	0.91	0.97
35	0.68	0.74	0.81	0.89	0.96
40	0.63	0.69	0.77	0.86	0.95

*Data courtesy of Valtek International.

†Note: d = inside diameter of valve port (inches); D = inside diameter of piping (inches).

7.3.12 Trim-Size Selection

Control-valve manufacturers provide tables that outline the C_v s for a certain valve style, flow direction, body pressure rating, flow characteristic, size of the valve seat or the seal, and length of stroke. Some charts may be broken down to percentages of opening, since some throttling services may not utilize the entire stroke.

Using the manufacturer's C_v table based upon the correct criteria (body size, flow characteristic, flow direction, etc.), the correct size of the valve opening (of the seat or the seal) should be chosen. This open-

Table 7.8 Piping-Geometry Factors with Reducer or In increaser on Outlet of Valve^{*,†}

C_v/d^2	0.50 d/D	0.60 d/D	0.70 d/D	0.80 d/D	0.90 d/D
4	1.00	1.00	1.00	1.00	1.00
6	1.01	1.01	1.01	1.01	1.01
8	1.01	1.02	1.02	1.02	1.01
10	1.02	1.03	1.03	1.03	1.02
12	1.03	1.04	1.04	1.04	1.03
14	1.04	1.05	1.06	1.05	1.04
16	1.06	1.07	1.08	1.07	1.05
18	1.08	1.10	1.11	1.10	1.06
20	1.10	1.12	1.12	1.12	1.08
25	1.17	1.22	1.24	1.22	1.13
30	1.27	1.37	1.42	1.37	1.20
35	1.44	1.65	1.79	1.65	1.32
40	1.75	2.41	3.14	2.41	1.50

^{*}Data courtesy of Valtek International.

[†]Note: d = inside diameter of valve port (inches); D = inside diameter of piping (inches).

ing and its dimension are often called the *trim number*. A globe-style valve will have a number of trim-number options, including one that is a *full-area trim number*, the largest sized diameter opening for that particular size. The valve may also have several *reduced-area trim numbers*, which are progressively smaller in diameter and allow smaller C_v s in the same body size.

7.3.13 Flashing-Velocity Calculation

As described in Sec. 7.2.8, if the valve outlet pressure is lower than the vapor pressure, the vapor bubbles that are formed remain in a gaseous state, providing a downstream flow that has a combined liquid–gas mixture. This results in increased velocity and difficult control situations. Since the application is found to be flashing, certain measures

Table 7.9 Valve Port Areas^{*,†}

Valve Size (inches)	Valve Outlet Area, A_v (Square Inches)						
	Class 150	Class 300	Class 600	Class 900	Class 1500	Class 2500	Class 4500
1/2	0.20	0.20	0.20	0.20	0.20	0.15	0.11
3/4	0.44	0.44	0.44	0.37	0.37	0.25	0.20
1	0.79	0.79	0.79	0.61	0.61	0.44	0.37
1 1/2	1.77	1.77	1.77	1.50	1.50	0.99	0.79
2	3.14	3.14	3.14	2.78	2.78	1.77	1.23
3	7.07	7.07	7.07	6.51	5.94	3.98	2.78
4	12.57	12.57	12.57	11.82	10.29	6.51	3.98
6	28.27	28.27	28.27	25.97	22.73	15.07	10.29
8	50.27	50.27	48.77	44.18	38.48	25.97	19.63
10	78.54	78.54	74.66	69.10	60.13	41.28	28.27
12	113.10	113.10	108.43	97.12	84.62	58.36	41.28
14	137.89	137.89	130.29	117.86	101.71	70.88	50.27
16	182.65	182.65	170.87	153.94	132.73	92.80	63.62
18	233.70	226.98	213.82	194.83	167.87	117.86	84.46
20	291.04	283.53	261.59	240.53	210.73	143.14	101.53
24	424.56	415.48	380.13	346.36	302.33	207.39	143.14
30	671.96	660.52	588.35	541.19	476.06	325.89	
36	962.11	907.92	855.30				
42	1320.25	1194.59					

^{*}Data courtesy of Valtek International.

[†]Note: To find approximate fluid velocity in the pipe, use the equation $V_p = V_v A_v / A_p$, where V_p = velocity in pipe, A_v = valve outlet area, V_v = velocity in valve outlet, and A_p = pipe area.

To find equivalent diameters of the valve or pipe inside diameter use $d = \sqrt{4A_v/\pi}$, $D = \sqrt{4A_p/\pi}$.

must be taken to prevent undue damage and premature wear to the valve, such as using special trims or hardened materials. Flashing applications must be limited to a velocity of 500 ft/s (152 m/s), unless special modifications are made to the valve-body design to accommodate the increased volume and velocity. Either of the following equations can be used to calculate flashing velocity, depending on the flow-rate measurement (lb/h or gal/min):

$$V = \frac{0.040}{A_v} w \left[\left(1 - \frac{x}{100\%} \right) V_{f2} + \frac{x}{100\%} V_{g2} \right]$$

$$V = \frac{20}{A_v} Q \left[\left(1 - \frac{x}{100\%} \right) V_{f2} + \frac{x}{100\%} V_{g2} \right]$$

where V = velocity (ft/s)

w = liquid flow rate (lb/h)

V_{f2} = saturated liquid specific volume (ft³/lb at outlet pressure P_2)

V_{g2} = saturated vapor specific volume (ft³/lb at outlet pressure P_2)

x = percentage of liquid mass flashed to vapor (Sec. 7.3.14)

7.3.14 Percentage of Flashing Calculation

To calculate the percentage of the liquid flashing into gas, the user should have access to steam tables, which provides a listing of enthalpies and specific volumes. To make this calculation, the following equation should be used:

$$x = \left(\frac{h_{f1} - h_{f2}}{h_{fg2}} \right) \times 100\%$$

where h_{f1} = enthalpy of saturated liquid at inlet temperature

h_{f2} = enthalpy of saturated liquid at outlet pressure

h_{fg2} = enthalpy of evaporation at outlet pressure

7.3.15 Liquid Sizing Example A

For this example, the following service conditions are given in Imperial units:

Liquid	Water
Critical pressure P_C	3206.2 psia
Temperature	250°F
Upstream pressure P_1	314.7 psia
Downstream pressure P_2	104.7 psia
Flow rate Q	500 gal/min
Vapor pressure P_V	30 psia
Specific gravity S_g	0.94
Kinematic viscosity ν	0.014 cS
Pipeline size	4 in (ANSI Class 600)
Valve	Globe, flow-to-open
Flow characteristic	Equal percentage

The actual pressure drop ΔP is calculated using the C_v equation for liquids (Sec. 7.3.1):

$$\Delta P = P_1 - P_2 = 314.7 \text{ psia} - 104.7 \text{ psia} = 210 \text{ psi}$$

Choked flow can be checked by finding the liquid pressure-recovery factor F_L from Table 7.4, which is 0.90. Then, the liquid critical-pressure ratio factor (F_F) is calculated by using the equation found in Sec. 7.3.3.

$$F_F = 0.96 - 0.28 \sqrt{\frac{P_V}{P_C}} = 0.96 - 0.28 \sqrt{\frac{30}{3206.2}} = 0.93$$

After determining F_L and F_F , these numbers are used in the choked pressure drop (ΔP_{choked}) equation from Sec. 7.3.3:

$$\Delta P_{\text{choked}} = F_L^2 (P_1 - F_F P_V) = (0.90)^2 [314.7 - (0.93)(30)] = 232 \text{ psi}$$

A comparison should be made between the actual pressure drop ΔP of 210 psi and the choked pressure drop ΔP_{choked} of 232 psi. Since the actual pressure drop is smaller than the choked pressure drop, the actual pressure drop will be used to size the valve.

By using the equation in Sec. 7.3.3, the advent of incipient cavitation should be checked:

$$\Delta P_{\text{cavitation}} = F_L^2 (P_1 - P_V) = (0.81)^2 (314.7 - 30) = 187 \text{ psi}$$

In this example, the actual pressure drop (ΔP) of 210 psi is greater than the pressure drop associated with incipient cavitation ($\Delta P_{\text{cavitation}}$) of 187 psi. This can be interpreted to mean that, although cavitation is occurring in the service, the cavitation is not causing the flow to choke. In this case, the user should begin considering methods to deter the cavitation damage, such as special trims or hardened materials. With a specific gravity of 0.94 and assuming the piping-geometry factor F_p is 1.0 (Sec. 7.3.9), the C_v should be calculated using the original liquid sizing equation (Sec. 7.3.1):

$$C_v = \frac{Q}{F_p} \sqrt{\frac{S_g}{\Delta P_a}} = \frac{500}{1} \sqrt{\frac{0.94}{210}} = 33.4$$

The required valve is a globe valve with flow-under-the-plug trim design, equal-percentage flow characteristic, and ANSI Class 600 pressure class. The manufacturer's C_v tables should be examined to deter-

mine the smallest valve available that would allow the flow of $33.4C_v$ through the flow area of the seat or seal. In this case, the assumption is made that, according to the charts, a 2-in valve body would be the smallest size with a trim number available to pass the required C_v .

At this point, the Reynolds-number factor F_R is calculated by using the equation from Sec. 7.3.7:

$$\begin{aligned} \text{Re}_v &= \frac{N_4 F_d Q}{v \sqrt{F_L C_v}} \left(\frac{F_L^2 C_v^2}{N_2 d^4} + 1 \right)^{0.25} \\ &= \frac{(17,300)(1)(500)}{(0.014) \sqrt{(0.90)(33.4)}} \left(\frac{(0.90)^2 (33.4)^2}{(890)(2)^2} + 1 \right)^{0.25} \\ &= 114 \times 10^6 \end{aligned}$$

Because the Reynolds-number factor F_R is significantly larger than 40,000 (114×10^6 versus 40,000), the calculated C_v remains 33.4 and is used in further calculations. With a 2-in body tentatively chosen for this application and a 4-in pipeline, the calculation of the piping-geometry factor F_p is made using Table 7.5 with the following numbers:

$$\frac{d}{D} = \frac{2}{4} = 0.5$$

and

$$\frac{C_v}{d^2} = \frac{33.4}{2^2} = 8.35$$

According to the table, the piping-geometry factor (F_p) should be 0.97. Now, the F_p of 0.97 can be inserted into the C_v equation to determine the final C_v :

$$C_v = \frac{Q}{F_p} \sqrt{\frac{S_g}{\Delta P_a}} = \frac{500}{0.97} \sqrt{\frac{0.94}{210}} = 34.5$$

Using Table 7.9, for a 2-in valve in ANSI Class 600 service, the valve outlet area A_v is 3.14 in^2 . Using this number and a flow rate of 500 gal/min (1892 liters/m), the velocity through the valve can be calculated as

$$V = \frac{0.321Q}{A_v} = \frac{0.321(500)}{3.14} = 51 \text{ ft/s (130 m/s)}$$

The velocity of 51 ft/s exceeds the limit of 50 ft/s for liquids. Since the service is cavitating, damage will most likely occur to the valve body. At this point, the only option to lower the velocity is to choose the next larger valve size, a 3-in body with reduced trim. Using a 3-in body and an A_v of 7.07, the velocity is significantly lowered to acceptable levels:

$$V = \frac{0.321Q}{A_v} = \frac{0.321(500)}{7.07} = 23 \text{ ft/s (5.8 m/s)}$$

Despite the lower velocity with the 3-in body, cavitation remains a concern and some material or design action should be taken to prevent damage. Another option that may reduce the cost of a larger valve would be to use an expanded outlet body—for example, a 2 × 4-in expanded outlet valve (since the piping is 4 in). Because of the velocity issue, which required the changing of the valve size to 3 in, the C_v equation will need to be recalculated using a new piping-geometry factor F_p :

$$\frac{d}{D} = \frac{3}{4} = 0.75$$

and

$$\frac{C_v}{d^2} = \frac{33.4}{3^2} = 3.71$$

With a piping-geometry factor F_p of 1.00 (interpolated from Table 7.5), the revised C_v for a 3-in body is

$$C_v = \frac{Q}{F_p} \frac{S_g}{\Delta P_a} = \frac{500}{1} \frac{0.94}{210} = 33.4$$

7.3.16 Liquid Sizing Example A (with Flashing)

For this example, the same service conditions as the previous example are provided, except that the temperature is increased by 100°F from 250 to 350°F. Using the saturated steam temperatures in the steam

tables, the saturation pressure for water at 350°F is 134.5 psia. Because the saturation pressure (134.5 psia) is significantly higher than the downstream pressure of the valve (104.7 psia), the service is flashing. Because of the flashing, the percent flash x must be calculated:

$$x = \left(\frac{h_{f1} - h_{f2}}{h_{fg2}} \right) \times 100\% = \left(\frac{321.8 - 302.3}{886.4} \right) \times 100\% = 2.2\%$$

where h_{f1} = 321.8 Btu/lb at 350°F (from the saturation temperature table)

h_{f2} = 302.3 Btu/lb at 105 psia (from the saturation pressure table)

h_{fg2} = 886.4 Btu/lb at 105 psia (from the saturation pressure table)

The equation from Sec. 7.3.13 must then be used to determine the velocity from a 3-in valve:

$$V = \frac{20}{A_v} Q \left[\left(1 - \frac{x}{100\%} \right) V_{f2} + \frac{x}{100\%} V_{g2} \right]$$

$$V = \frac{(20)(500)}{7.07} \left[\left(1 - \frac{2.2\%}{100\%} \right) 0.0178 + \frac{2.2\%}{100\%} 4.234 \right] = 156 \text{ ft/s}$$

where V_{f2} = 0.0178 ft³/lb at 105 psia (from the saturation pressure table)

V_{g2} = 4.324 ft³/lb at 105 psia (from the saturation pressure table)

From Sec. 7.2.12, the maximum velocity for flashing services is 500 ft/s. The calculated velocity of this service is 156 ft/s, which is far below the maximum level. Once again, however, the presence of flashing should be considered by selecting hardened materials or special trim features.

7.3.17 Liquid Sizing Example B

In this second liquid example, the following service conditions are provided in Imperial units:

Liquid	Ammonia
Critical pressure P_C	1638.2 psia

Valve Sizing

Temperature	20°F
Upstream pressure P_1	149.7 psia
Downstream pressure P_2	64 psia
Flow rate Q	850 gal/min
Vapor pressure P_V	465.6 psia
Specific gravity S_g	0.65
Kinematic viscosity ν	0.02 cS
Pipeline size	3 in (ANSI Class 600)
Valve	Globe, flow-to-close
Flow characteristic	Linear

The actual pressure drop ΔP is calculated as follows:

$$\Delta P_a = P_1 - P_2 = 149.7 \text{ psia} - 64.7 \text{ psia} = 85 \text{ psi}$$

Choked flow is checked by determining the liquid pressure-recovery factor F_L from Table 7.4, which is 0.85. The liquid critical-pressure ratio factor F_F can then be calculated by using the following equation found in Sec. 7.3.3.

$$F_F = 0.96 - 0.28 \frac{P_V}{P_C} = 0.96 - 0.28 \sqrt{\frac{45.6}{1638.2}} = 0.91$$

After determining that F_L (= 0.85) and F_F (= 0.91), these numbers are inserted in the choked-pressure-drop ΔP_{choked} equation from Sec. 7.3.3:

$$\Delta P_{\text{choked}} = F_L^2(P_1 - F_F P_V) = (0.85)^2[149.7 - (0.91)(45.6)] = 78.2 \text{ psi}$$

In comparing the actual pressure drop ΔP of 85.0 psi and the choked pressure drop ΔP_{choked} of 78.2 psi, the choked pressure drop is smaller than the actual pressure drop. Therefore, the smaller of the two numbers—the choked pressure drop—is used to size the valve. Because the valve is choked, the service is also cavitating. Therefore, checking for incipient cavitation $\Delta P_{\text{cavitation}}$ is not necessary. In this case, the user should plan to use special anticavitation trim inside the valve as well as hardened materials to avoid the erosion of metal parts associated with cavitation.

With a specific gravity of 0.65 and assuming a piping-geometry factor F_p of 1.0 (Sec. 7.3.9), a preliminary C_v can be calculated using the original liquid sizing equation (Sec. 7.3.1):

$$C_v = \frac{Q}{F_p} \sqrt{\frac{S_g}{\Delta P_a}} = \frac{850}{1} \sqrt{\frac{0.65}{78.2}} = 77.5$$

From the conditions of this example, the preferred valve is a globe valve with flow-over-the-plug trim design, a linear flow characteristic, and ANSI Class 600 pressure classification. The manufacturer's C_v tables can then be examined to estimate the smallest valve available that would allow the flow of $77.5C_v$ through the flow area of the seat. In this case, the assumption is made that the manufacturer's C_v tables show that a 3-in valve body would be the smallest size with a trim number that would pass the required C_v .

Because the flow is cavitating, it is turbulent when exiting the valve. Because of the turbulent flow, the Reynolds-number factor F_R is assumed to be $F_R = 1.0$ and no further calculations are required.

Since a 3-in body was chosen initially for this application and the pipeline is determined to be a 3-in line, the piping-geometry factor F_p will be 1.0 (no reducers or increasers are required). Because $F_p = 1.0$, the C_v calculation made earlier does not change because of the piping geometry and remains at 77.5.

Using Table 7.9, for a 3-in valve in ANSI Class 600 service, the valve outlet area A_v is 7.07 in². Using this number and a flow rate of 850 gal/min, the velocity through the valve can be calculated as

$$V = \frac{0.321Q}{A_v} = \frac{0.321(850)}{7.07} = 39 \text{ ft/s}$$

The velocity of 39 ft/s is below the limit of 50 ft/s for liquids. Therefore, a 3-in body is acceptable for this application, although the cavitating service will need to be dealt with through modifications to the valve, such as special trim or hardened materials.

7.4 Body Sizing of Gas-Service Control Valves

7.4.1 Basic Gas Sizing Equations

The basic difference between liquid sizing and gas sizing deals with the compressibility of gases. Because of their compressibility, gases have a tendency to expand as the pressure drop occurs through the vena contracta. In turn, this lowers the specific weight of the gas. This changing specific weight must be taken into account during the sizing process using a special factor called the *expansion factor* Y .

Depending on the given service conditions or variables, one of four gas sizing equations is used. The numerical constants included in each equations deal with unit conversion factors.

$$w = 63.3F_p C_v Y \sqrt{x P_1 \gamma_1}$$

$$Q = 1360F_p C_v P_1 Y \sqrt{\frac{x}{G_g T_1 Z}}$$

$$w = 19.3F_p C_v P_1 Y \sqrt{\frac{x M_w}{T_1 Z}}$$

$$Q = 7320F_p C_v P_1 Y \sqrt{\frac{x}{M_w T Z}}$$

where w = gas flow rate (lb/h)

F_p = piping-geometry factor

C_v = valve sizing coefficient

Y = expansion factor

x = pressure-drop ratio

γ_1 = specific weight at inlet service conditions (lb/ft³)

Q = gas flow (scfh)

G_g = specific gravity or gas relative to air at standard conditions

T_1 = absolute upstream pressure (°R = °F + 460)

Z = compressibility factor

M_w = molecular weight

P_1 = upstream absolute pressure (psia)

One of the four gas sizing equations should be selected based on the available data for the given service conditions.

7.4.2 Choked-Flow Determination

The terminal pressure-drop ratio x_T is determined by taking the appropriate value from Table 7.10. The ratio of specific heats factor F_K can be calculated by using the following equation:

$$F_K = \frac{k}{140}$$

Table 7.10 Typical x_T Factors^{*,†}

Valve Style	Flow Direction	Trim Area	X_T
Linear globe	Over seat	Full area	0.70
	Over seat	Reduced area	0.70
	Under seat	Full area	0.75
	Under seat	Reduced area	0.75
Butterfly	60° open	Full area	0.36
	90° open	Full area	0.26
Ball	60° open	Full area	0.51
	90° open	Full area	0.30

^{*}Data courtesy of Valtek International.

[†]Note: All values provided at full-open.

where F_K = ratio of specific heats factor
 k = ratio of specific heats

The ratio k of specific heats can be found for common gases in Table 7.11, which is provided for quick reference.

The ratio x of actual pressure drop to absolute inlet pressure is determined by using the following equation:

$$x = \frac{\Delta P_a}{P_1}$$

where x = ratio of actual pressure drop to absolute inlet pressure

ΔP = actual pressure drop (psi)

P_1 = upstream pressure (at inlet, psia)

P_2 = downstream pressure (at outlet, psia)

If the value for x is less than the value for $F_K x_{T'}$, choked flow is not occurring. Inversely, when x reaches or exceeds the value of $F_K x_{T'}$, the

Table 7.11 Physical Data for Common Gas Services

Gas	Molecular Weight (M_w)	Critical Temperature ($^{\circ}R^*$)	Critical Pressure (psia/bar)	Ratio of Specific Heats (k)
Air	28.97	227°	492/33.9	1.40
Ammonia	17.00	730°	1636/112.8	1.31
Argon	39.95	271°	707/48.8	1.67
Carbon Dioxide	44.01	547°	1070/73.8	1.29
Carbon Monoxide	28.01	239°	507/35.0	1.40
Ethane	30.07	549°	709/48.9	1.19
Ethylene	28.10	508°	731/50.4	1.24
Helium	4.00	9°	33/2.3	1.66
Hydrogen	2.02	59°	188/13.0	1.40
Methane	16.04	343°	667/46.0	1.31
Natural Gas	16.04	343°	667/46.0	1.31
Nitrogen	28.00	227°	492/33.9	1.40
Oxygen	32.00	278°	732/50.5	1.40
Propane	44.10	665°	616/42.5	1.31
Steam	18.02	1165°	3208/221.2	1.33

*°R = °F + 460.

flow is choked. If the flow is choked, the value $F_K x_T$ should be used instead of x , if x is used in the chosen gas sizing equation.

7.4.3 Expansion-Factor Calculation

Because of the compressibility of gases, the expansion factor Y must be determined by using the following equation. If choked flow is occurring, the value $F_K x_T$ should be used instead of x .

$$Y = 1 - \frac{x}{3F_K x_T}$$

where Y = expansion factor

x_T = terminal pressure-drop ratio

7.4.4 Compressibility-Factor Determination

The compressibility factor Z is determined by calculating the reduced-pressure value P_r and the reduced-temperature value T_r :

$$P_r = \frac{P_1}{P_c}$$

where P_r = reduced pressure

$$T_r = \frac{T_1}{T_c}$$

where T_r = reduced temperature

T_1 = absolute upstream temperature

T_c = absolute critical temperature

Once the reduced pressure P_r and reduced temperature T_r are known, the compressibility factor Z can be determined with either Fig. 7.3 or 7.4.

7.4.5 Flow-Coefficient Calculation

Using the factors determined to this point, a preliminary C_v is calculated by using the applicable gas sizing C_v equation. For this equation, the piping-geometry factor F_p should be assumed to be 1.0.

7.4.6 Approximate Body-Size Selection

Using the manufacturer's C_v tables, the smallest sized body is selected that can accommodate the calculated preliminary C_v .

7.4.7 Piping-Geometry-Factor Calculation

When the pipeline size has not been determined or is unknown, for calculation purposes the body size that was determined from Sec. 7.4.6 is used as pipeline size. The inside diameter of the piping is required

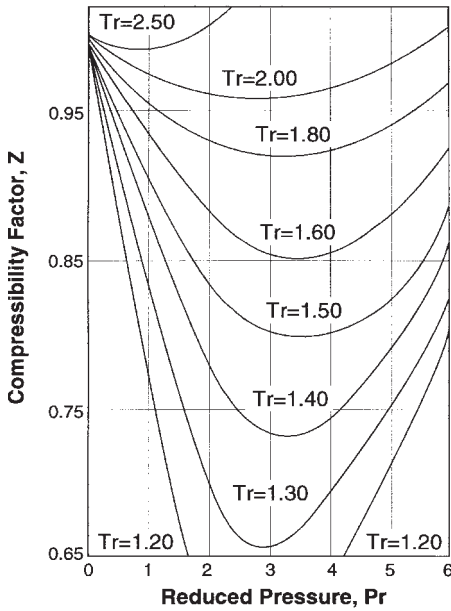


Figure 7.3. Compressibility factors, reduced pressures 0 to 6. (Courtesy of Valtek International)

to determine the piping-geometry factor F_p . Tables 7.5 and 7.6 are used to find the piping-geometry factors. Table 7.5 provides F_p for valves with reducers (or increasers) on both the inlet and outlet of the valve. Table 7.6 provides F_p for a valve with the reducer (or increaser) on the valve outlet only.

7.4.8 Final-Flow-Coefficient Calculation

Using the piping-geometry factor F_p , the final C_v is calculated, using one of the four equations provided. Usually, the C_v will be close to the preliminary C_v chosen earlier. Therefore, the body size will most like stay the same, unless high velocities are present.

7.4.9 Valve Exit Mach-Number Calculation

With the flow coefficient known, as well as the body size, the exit velocity of the gas from the valve is determined in Mach numbers. The

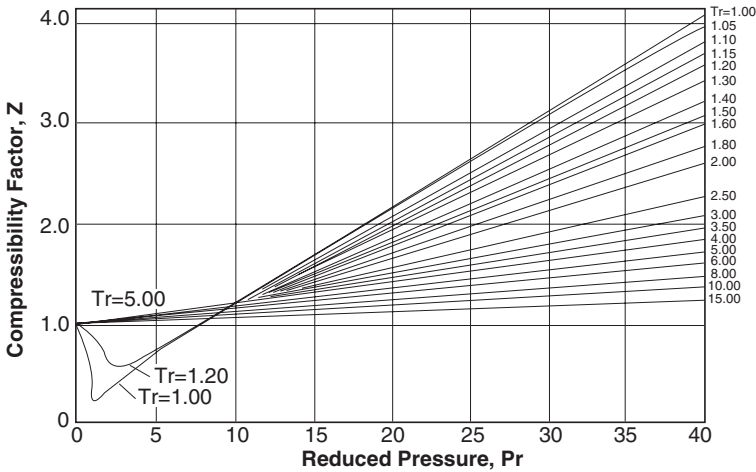


Figure 7.4. Compressibility factors, reduced pressures 0 to 40.
(Courtesy of Valtek International)

following two equations are used for calculating velocities in gas services:

$$M_{\text{gas}} = \frac{Q_a}{5574 A_v \sqrt{\frac{kT}{M_w}}}$$

$$M_{\text{gas}} = \frac{Q_a}{1036 A_v \sqrt{\frac{kT}{G_g}}}$$

where M_{gas} = Mach number for gas service

Q_a = actual flow rate (cfh instead of scfh)

A_v = applicable flow area of body port (square inches) from Table 7.9

k = ratio of specific heats

T = absolute temperature ($^{\circ}\text{R}$ or $^{\circ}\text{F} + 460$)

M_w = molecular weight

G_g = specific gravity at standard conditions relative to air

The following velocity equation is used for air service:

$$M_{\text{air}} = \frac{Q_a}{1225 A_v \sqrt{T}}$$

where M_{air} = Mach number for air service

To convert scfh to cfh, the following equation is used:

$$\frac{P_a Q_a}{T_a} =$$

where P_a = actual operating pressure

Q_a = actual volume flow rate (cfh)

T_a = actual temperature (°R or °F + 460)

P_s = standard pressure (14.7 psi)

Q = standard volume flow rate (scfh)

T_s = standard temperature (520°R)

The following velocity equation is used for steam service:

$$M_{\text{steam}} = \frac{wv}{1514A_v \sqrt{T}}$$

where M_{steam} = Mach number for air service

w = mass flow rate (lb/h)

v = specific volume at flow conditions (ft³/lb)

Once the exit velocity has been calculated and is found to exceed Mach 0.5, the possibility of excessive vibration and noise will become evident because of the turbulence caused in the valve. The velocity limit for valves is near Mach 1. If noise is occurring in the valve and a special antinoise trim is used in the valve, the velocity is normally limited to Mach 0.33. If the high velocity exceeds the Mach-0.5 limit for noise generation, a larger valve body will need to be chosen. If the velocity approaches Mach 1.0 in this situation, a larger body size should also be chosen.

7.4.10 Trim-Size Selection

Valve manufacturers provide tables that outline the flow coefficients for a certain valve style, flow direction, body-pressure rating, flow characteristic, size of the valve seat (either full or reduced area) or the seal, and stroke. Some charts may be broken down to percentages of opening, since some throttling services may not utilize the entire stroke. Depending on the style of the valve, a trim number is offered with a predetermined flow area that allows the passage of flow equal to the C_v maximum.

7.4.11 Gas Sizing Example A

For this example, the following service conditions and equipment requirements are given in Imperial units:

Gas	Steam
Temperature	450°F
Upstream pressure P_1	140.0 psia
Downstream pressure P_2	50.0 psia
Flow rate Q	10,000 lb/h
Critical pressure P_C	3206.2 psia
Critical temperature T_C	705.5°F
Molecular weight M_W	18.03
Specific volume	10.41
Ratio k of specific heats	1.33
Pipeline size	2 in (ANSI Class 600)
Valve	Globe, flow-to-open
Flow characteristic	Equal percentage

Of the four C_V equations given for gas sizing (Sec. 7.4.1), the following equation is appropriate for the provided service conditions:

$$w = 19.3F_P C_v P_1 Y \sqrt{\frac{x M_W}{T_1 Z}}$$

From Table 7.4, the pressure-drop ratio x_T for a globe valve with flow-to-open action is 0.75. The user should check for choked flow by calculating the ratio of specific heats factor F_K :

$$F_K = \frac{k}{1.40} = \frac{1.33}{1.40} = 0.95$$

The ratio of actual pressure drop to absolute inlet pressure x is now calculated with the following equation:

$$x = \frac{\Delta P}{P_1} = \frac{140 - 50}{140} = 0.64$$

The value $F_K x_T$ can then be calculated as

$$(0.95)(0.75) = 0.71$$

Because the ratio of actual pressure drop to absolute inlet pressure x is less than the combined value $F_K x_T$, choked flow is not occurring and the value x is used with the remaining calculations.

The expansion factor Y is now calculated using the following equation:

$$Y = 1 - \frac{x}{3(F_K x_T)} = 1 - \frac{0.64}{3(0.71)} = 0.70$$

The compressibility factor Z can be determined by using the equations for the reduced-pressure factor P_r and the reduced-temperature factor T_r .

$$P_r = \frac{P_1}{P_C} = \frac{140}{3208.2} = 0.04$$

$$T_r = \frac{T_1}{T_C} = \frac{450 + 460}{705.5 + 460} = 0.78$$

With the aid of these two numbers and Fig. 7.3, the compressibility factor Z is found to be 1.0. Assuming that the piping-geometry factor F_p is 1.0, the appropriate C_v equation should be used to calculate a preliminary C_v :

$$w = 19.3 F_p C_v P_1 Y \sqrt{\frac{x M_W}{T_1 Z}} \quad \text{or} \quad C_v = \frac{w}{19.3 F_p P_1 Y} \sqrt{\frac{T_1 Z}{x M_W}}$$

$$C_v = \frac{10,000}{(19.3)(140)(0.70)} \sqrt{\frac{(910)(1.0)}{(0.64)(18.02)}} = 47.0$$

From the manufacturer's C_v tables, the smallest valve body should be chosen that will pass the required C_v of 47. For assumption purposes, a 2-in valve is the smallest size that will accommodate a C_v of 47. Because the 2-in body is the same size as the pipeline size, the piping-geometry factor F_p is 1.0 and C_v remains the same. In this case, the preliminary C_v becomes the final C_v .

At this point, the exit velocity should be calculated to ensure that it is within the velocity limits of Mach 0.5 for noise or Mach 1.0 for maximum velocity. The valve outlet of a 2-in valve is 3.14 (from Table 7.9). From the steam tables, v is found to be 10.41 ft³/lb and T is 414°F.

Therefore, the following velocity equation should be used for steam service:

$$M_{\text{steam}} = \frac{wv}{1514 A_v \sqrt{T}} = \frac{(10,000)(10.41)}{(1515)(3.14)\sqrt{414 + 460}} = 0.74$$

Because Mach 0.74 is greater than the noise limit of Mach 0.5, the turbulence will most likely create noise in the valve, and preventative measures may be necessary, such as special trim, insulation, or isolation of the valve. Because the velocity did not exceed the limit of Mach 1.0, a larger valve size is not necessary and the final C_v remains the same.

7.4.12 Gas Sizing Example B

For the second gas example, the following service conditions and equipment requirements are provided in Imperial units:

Gas	Natural gas
Temperature	65°F
Upstream pressure P_1	1314.7 psia
Downstream pressure P_2	99.7 psia
Flow rate Q	2,000,000 scfh
Critical pressure P_C	672.9 psia
Critical temperature T_C	342.8°F
Molecular weight M_W	16.04
Ratio k of specific heats	1.31
Pipeline size	Unspecified (ANSI Class 600)
Valve	Globe, flow-to-open
Flow characteristic	Linear

Of the four C_v equations given for gas sizing from Sec. 7.4.1, the following equation is best for the provided service conditions:

$$Q = 7320 F_P C_v P_1 Y \sqrt{\frac{x}{M_W T}}$$

Referring to Table 7.4, the pressure-drop ratio x_T for a globe valve with flow-to-open action is 0.75. A choked-flow condition should be checked first by calculating the ratio of specific heats factor F_K :

$$F_K = \frac{k}{1.40} = \frac{1.31}{140} = 0.94$$

The ratio x of actual pressure drop to absolute inlet pressure is determined by using the following equation:

$$x = \frac{\Delta P}{P_1} = \frac{1314.7 - 99.7}{1314.7} = 0.92$$

The value $F_K x_T$ can then be calculated as follows:

$$(0.94)(0.75) = 0.70$$

Because the combined value $F_K x_T$ is less than the ratio of actual pressure drop to absolute inlet pressure x , choked flow is occurring and $F_K x_T$ is used with the remaining calculations. The expansion factor Y is now calculated using the following equation:

$$Y = 1 - \frac{x}{3(F_K x_T)} = 1 - \frac{0.70}{3(0.70)} = 0.67$$

Before the compressibility factor Z can be determined, the reduced-pressure factor P_r and the reduced-temperature factor T_r must be calculated with the following equations:

$$P_r = \frac{P_1}{P_C} = \frac{1314.7}{667.4} = 1.97$$

$$T_r = T \frac{1}{T_C} = \frac{65 + 460}{342.8} = 1.53$$

Using the P_r and the T_r factors with Fig. 7.3, the compressibility factor Z is found to be approximately 0.86. With the assumption that the piping-geometry factor F_p is 1.0 and that x is now replaced by the combined value $F_K x_T$, the chosen C_v equation is used to calculate a preliminary C_v :

$$Q = 7320 F_p C_v P_1 Y \sqrt{\frac{F_K x_T}{M_w T Z}} \quad \text{or} \quad C_v = \frac{Q}{7320 F_p P_1 Y} \sqrt{\frac{M_w T Z}{F_K x T}}$$

$$C_v = \frac{2,000,000}{(7320)(1.0)(1314.7)(0.667)} \sqrt{\frac{(16.04)(525)(0.86)}{0.70}} = 32$$

From the manufacturer's C_v tables, the user should find the smallest valve body that will pass the required C_v of 32. For this example, a 1.5-in valve is assumed to be the smallest size that will accommodate the preliminary C_v of 32. Because the pipeline size is unspecified, the user must assume that the piping-geometry factor F_p is 1.0 and the final C_v remains the same as the preliminary C_v .

The exit velocity is now calculated to ensure that the 1.5-in body will handle the velocity limit of Mach 1.0. If the velocity exceeds Mach 0.5, noise will most likely be generated. From Table 7.9, the valve outlet area A_v of a 1.5-in body is 1.77. Since the fluid is natural gas, the following velocity equation for gas service is used after converting scfh to cfh (Sec. 7.4.9):

$$M_{\text{gas}} = \frac{Q_a}{5574 A_v \sqrt{\frac{kT}{M_w}}} = \frac{297,720}{(5574)(1.77) \sqrt{\frac{(1.31)(65 + 460)}{16.04}}} = \text{Mach 4.61}$$

Because a Mach number exceeding sonic velocity (Mach 1.0) at the outlet of the valve is not possible, a larger valve size must be chosen to lower the velocity to below Mach 1.0.

The chosen valve would ideally handle a velocity of Mach 0.5 or less. To find the correct valve size to handle the process at Mach 0.5, the velocity equation should be used—except the user should solve for the unknown factor, which is the valve outlet area A_v :

$$A_v = \frac{Q_a}{5574 M_{\text{gas}} \sqrt{\frac{kT}{M_w}}} = \frac{297,720}{(5574)(0.5) \sqrt{\frac{(1.31)(65 + 460)}{16.04}}} = 16.3 \text{ in}^2$$

The valve outlet area A_v can then be used to solve the size of the valve:

$$A_v = p d^2 \quad \text{or} \quad d = \sqrt{\frac{4 A_v}{\pi}} = \sqrt{\frac{(4)(16.3)}{3.14}} = 4.6 \text{ in}$$

Because a 4-in valve would be too small and a 5-in valve does not exist, a 6-in valve is necessary. This valve will need a reduced trim to accommodate a C_v of 32.