5

VALVE SIZING MADE EASY

VALVE SIZING

This activity can be defined as picking the right size valve for the job or, in certain instances where a valve size is predestined, the correct trim size.

Do not read on if you have a computer fully programmed with ANSI/ISA-75.01.01 equations.[1] But then again, you may want to know what you are doing.

To help matters, about 60 years ago some clever soul invented a valve sizing coefficient called C_v that combined the flow area of the valve orifice, the contraction coefficient, and the head loss coefficient all in one; thus,

$$C_v = q/\sqrt{\Delta P/G_f} \tag{5-1}$$

$$q = C_v\sqrt{\Delta P/G_f} \tag{5-2}$$

where q is the flow rate in U.S. gallons per minute, G_f is the specific gravity of a liquid at a given temperature (water at 60°F has a G_f of one), and ΔP is inlet minus outlet pressure in psi.

Hence, one C_v equals the flow of one U.S. gallon per minute of water at 60°F and under a pressure drop across the valve of 1 psi. Now the same valve will flow 10 gpm at a pressure drop of 100 psi since the flow is proportional to the square root of ΔP, thus following the laws of fluid mechanics. This equation is worthwhile memorizing since you will use it on at least 80% of all sizing problems for liquids. Unfortunately, not all in life is so simple; therefore, we have to recognize two limitations in the use of the above equations.

First, they only apply to turbulent flow, i.e., Reynolds numbers above about 5000. However, you can ignore this limit on required C_v numbers above 0.1 and when the kinematic viscosity is below 40 centistoke (90% of all liquids are). There will be more on this subject—required reading only if you control tar or equally sticky substances and polymers.

Second, in some conditions funny things occur inside the valve which seems to block the flow passages. What really happens is that at a certain ratio of pressure drop to inlet pressure defined by a pressure recovery factor $F_L{}^2$ the flow becomes choked at the orifice. (F_L originally was called *the* "critical flow factor,"[2] C_f and was used as such.) In the case of liquids, this is caused by the evaporation of some of the liquid (also see Cavitation in Chapter 14), and in the case of gases, the onset of sonic velocity. We could not care less about all this were it not for the fact that now the flow is independent of the downstream pressure P_2. Now the flowing quantity can only increase if you use a higher inlet pressure.

To avoid embarrassing errors, you should first start to determine if you have critical (choked) flow or not using the equations below (this also helps to discover any cavitation or noise problems you may have to worry about).

Note, use $(F_{LP} / F_P)^2$ instead of $F_L{}^2$ in the equations below whenever you have a valve between pipe reducers where

$$F_{LP} = 1 \Big/ \sqrt{\frac{1}{F_L^2} + \left(\frac{C_v}{30d^2}\right)^2 \left(1 - \frac{d^4}{D^4}\right)} \qquad (5\text{-}3)$$

here d = valve size (in.) and D = pipe size (in.).

For F_P, see subchapter on Pipe Reducers.

For Liquid Service

SUBCRITICAL FLOW	CRITICAL FLOW	ΔP_s = Maximum ΔP for sizing.
Is if ΔP is less than $F_L{}^2\ (\Delta P_s)$	Occurs when ΔP is more than $F_L{}^2\ (\Delta P_s)$	Use: $P_1 - P_v$ when outlet pressure is higher than vapor pressure.

Volumetric Flow

When outlet pressure is lower or equal to vapor pressure use: $\Delta P_s = P_1 - [0.96 - 0.28\ (P_v\ /\ P_c)]\ P_v$

$$C_v = q\ (G_f / \Delta P)^{0.5} \qquad C_v = \frac{q}{F_L}\left(\frac{G_f}{\Delta P_s}\right)^{0.5} \qquad (5\text{-}4)$$

Flow by Weight

$$C_v = \frac{W}{500\sqrt{G_f \Delta P}} \qquad C_v = \frac{W}{500 F_L \sqrt{G_f \Delta P_s}} \qquad (5\text{-}5)$$

G_f = specific gravity @ flowing temperature (water = 1 @ 60°F)

P_1 = absolute inlet pressure, psia

P_2	=	absolute outlet pressure, psia
ΔP	=	$P_1 - P_2$
P_c	=	pressure at thermodynamic critical point, psia—water is 3206 psia
P_v	=	vapor pressure of liquid at flowing temperature, psia — water = 0.4 psia @ 70°F
q	=	liquid flow rate, U.S. gpm
W	=	flow in pounds per hour

For Gas and Steam Service

Surprising as it may sound, the basic C_v* equation can also be applied to gases and steam. The difference here is that the density of the gas changes with ΔP, and, since this is a gradual process, the relationship $\sqrt{\Delta P}$ to flow is no longer linear but curved. To its credit, the ISA Sizing Committee for ANSI/ISA-75.01.01 has labored hard to predict the exact shape of this curve with the aid of an expansion factor[3], Y, where

$$Y = 1 - (x / 3 F_k X_T) \tag{5-6}$$

Note that Y cannot exceed 1 and that it cannot be less than 0.67 since the latter is exactly the density of the gas, i.e., 67% of the inlet density when the valve is choked (when it sees sonic velocity in the valve orifice).

In the above equation,

x	=	$\Delta P/P_1$ (Note: P_1 *or* P_2 is *always* absolute pressure, i.e., gage psi plus 14.7)
F_k	=	the ratio of the specific heat k of a gas (air = 1.4), $Fk = k$ of gas/1.4
X_T	=	pressure drop ratio. This is the $\Delta P/P_1$ at which there is no longer any flow increase.

Note, that X_T includes some flow increase *after* the onset of choked flow due to changes in the size of the flowing jet (vena contracta).

Now the required flow coefficient becomes:

$$C_v = q(G_g T_1 Z / x)^{0.5} / (1360 P_1 Y) \tag{5-7}$$

* There is evidence that the C_v number on certain valve types, such as angle valves, certain rotary valves, and double-ported globe valves is different between liquid and gases, but we shall ignore this.

Or

$$q = 1360 C_v P_1 Y / \sqrt{G_g T_1 Z / x} \tag{5-8}$$

where:

q	=	flow in scfh
Gg	=	gas specific gravity (air = 1), also ratio of molecular weight of gas to that of air, (air = 28.9)
T_1	=	absolute upstream temperature °R (°F + 460)
Z	=	compressibility factor (ignore if your pressure never exceeds 2000 psia; otherwise, consult handbook tables.)

Now, if this begins to tax the capabilities of your hand-held calculator, you may ignore all of the above and use the simplified version below.[4] (The penalty is a maximum sizing error of ≈8% at the transition from nonchoked to choked flow and not worth the trouble.) It is also strongly recommended that you use these simplified equations to make sure your computer-generated answers are at least in the ball park!

<u>SUBCRITICAL FLOW</u>
When ΔP is less than $F_L^2\,(P_1/2)$

<u>CRITICAL FLOW</u>
When ΔP is more than $F_L^2\,(P_1/2)$

Volumetric Flow

$$C_v = \frac{Q}{963}\sqrt{\frac{G_g T}{\Delta P(P_1 + P_2)}} \qquad C_v = \frac{q\sqrt{G_g T}}{834 F_L P_1}$$

Flow by Weight

$$C_v = \frac{W}{3.22\sqrt{\Delta P(P_1 + P_2) G_g}} \qquad C_v = \frac{W}{2.8 F_L P_1 \sqrt{G_g}}$$

For Saturated Steam

$$C_v = \frac{W}{2.1\sqrt{\Delta P(P_1 + P_2)}} \qquad C_v = \frac{W}{1.83 F_L P_1}$$

For Superheated Steam

$$C_v = \frac{W(1 + 0.0007 T_{sh})}{2.1\sqrt{\Delta P(P_1 + P_2)}} \qquad C_v = \frac{W(1 + 0.0007 T_{sh})}{1.83 F_L P_1}$$

where:

C_v	=	valve coefficient
F_L	=	pressure recovery factor
G_g	=	gas specific gravity (air = 1.0)
P_1	=	absolute upstream pressure, psia
P_2	=	absolute downstream pressure, psia
ΔP	=	pressure drop $P_1 - P_2$, psi
q	=	gas flow rate at 14.7 psia and 60°F, scfh
T	=	flow temperature, °R(460 + °F)
T_{sh}	=	steam superheat, °F
W	=	flow rate, pounds per hr

NOTE: When a valve is installed between reducers, use $C_v F_p$ instead of C_v in capacity tables (see subchapter on Pipe Reducers).

<u>OTHER METHODS</u>

$$C_v = \frac{C_g}{40F_L}; \quad C_v = \frac{C_s}{2F_L}; \quad X_T = 0.84F_L^2$$

$$F_L = \sqrt{K_m}; \quad F_L = C_1/40$$

NOTE: X_T is not strictly related to F_L. While X_T gives you the maximum gas flow, which may happen at orifice pressure ratios P_1 / P_o of 3:1 or more, i.e., above sonic velocity, $F_L{}^2$ tells you the onset of sonic velocity, i.e., where $P_1 / P_{orifice} \approx 2$, and $P_1 / P_{2\ \mathrm{sonic}} = P_1 / (P_1 - 0.5\ F_L{}^2)$.

WHAT ARE THE RIGHT FLOW CONDITIONS?

Using a computer and sizing the C_v number to the fifth place after the decimal point is all well and good, but only if you have the right process conditions. First of all, the valve should be sized to control the maximum flow rate your process is designed for. But what is this exactly? How about emergency conditions? On the other hand, should the valve fail open, would the resultant flow be more than the downstream pressure relief valve could handle? Then again, what is the correct inlet pressure? Can you use the head pressure of the pump from the manufacturer's published curve? (If you do, use the head pressure corresponding to the maximum flow the valve has to pass, and don't ignore the static head at the pump's location.) What is the head loss or pressure drop across a heat exchanger next to the valve? How about line losses? All these are questions that beg only vague responses from your process design people. They don't know either and have to rely on the input from other vendors. The result is usually guesswork with ample safety factors thrown in. Therefore, don't be too concerned about the accuracy of sizing equations. Ninety-nine out of one hundred times, the valve is too large for the job anyhow (see Chapter 8 on equal percentage trim). The remaining one percent that is undersized comes about because someone made a decimal point error in the C_v calculation.

If you want to be clever and save your company energy and money in the bargain, have the process people *ask you* for the valve pressure drop when they design the system (see Chapter 18). Now you are in the driver's seat. Here are the very scientifically derived rules of thumb you should memorize: For ΔP valve sizing, use the higher of 5% of the total system pressure (i.e., static head plus pump head at maximum flow, boiler pressure, etc.), or 5 psi for rotary control valves, or 10 psi for globe valves.

You may wonder why I picked 5 psi for rotary control valves. Well, it just so happens that the flow rate (either liquid or gas) generated by a 5 psi pressure drop through a wide-open control valve with a C_v divided by the valve diameter squared of 17* is all that a pipe (the same size as the valve) can handle, allowing for the customary maximum line velocities of 15 ft/*s* for liquids or 150 ft/*s* for gases [5] as you can see from Figure 5-1. Higher ΔP*'s* mandate a valve smaller than the pipe size and where reducers are required, or when reduced trim is selected.† In any case, you are throwing away dollars for pump, compressor, or boiler horse powers if you assign too much pressure drop to the valve (see chapter 18).

Of course, the above trick can be used in reverse. You may be requested to replace an old steam reducing valve, vintage 1948. All the records are lost, and the only thing you know is the boiler pressure and the valve and pipe size. Say the boiler pressure is 95 psig (110 psia) and the pipe size is 1-1/2 inches. First, look up the maximum flow rate of a 1-1/2 in. steam pipe at, say, 100 psig pressure. You will find it to be approximately 1800 lbs/hr (see Table A-2 in Appendix A). Next, you assume the flow to be choked, i.e., P_2 is less than one-half P_1. In any case, people didn't care much about energy savings in 1948! Now calculate the C_v required, which is about 10.5. So go ahead and select a 1-1/2 in. globe valve with a rated C_v of about 22. The worst that can happen is that the outlet pressure is 80 psig instead of below 50 psig. No problem. Now the C_v required is 16, which is still OK. Therefore, when in doubt, use a full-trim size if flow conditions are unknown, except in special applications such as pH control.

PIPE REDUCERS

One of the competitive strategies of some valve manufacturers is to offer a control valve that has a much higher C_v rating than somebody else's valve. These are usu-

* A butterfly valve at *60°* opening, for example.

† For example, if a 4 in. butterfly valve is installed in a 6 in. pipe, then at the same 17 x d^2 = 272 C_v, the valve pressure drop increases from 5 psi to 25 psi with 15 ft/s fluid velocity in the 6 in. pipe.

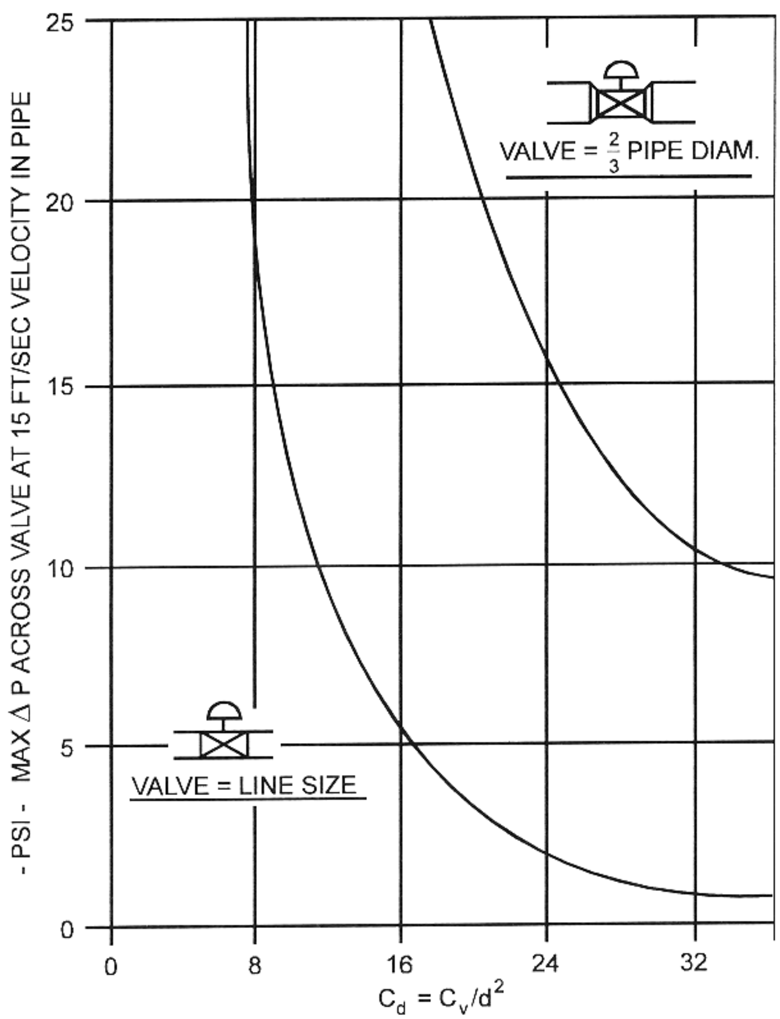


Figure 5-1. Relationship between pressure drop and specific valve flow capacity for water at 15 feet/second velocity in pipe, *d=* valve diameter (inches). This works also well for gas or steam at 150 ft/sec pipe velocity.

ally oversized globe valves with reduced flange connections, for example, a 1-1/2 in. flanged globe valve with a 2 in. orifice size. While there is nothing wrong with this concept (after all, you are getting more weight for your dollars), it nevertheless forces you into using pipe reducers, since with a large C_v your valve can be smaller than the attached pipe. So one thing you have to do is evaluate the extra cost and space requirement. You may also need reducers when you buy rotary control valves, which, due to having less torturous bodies, have inherently more C_v — although with higher *ΔP's,* this cancels out because of low F_L numbers (choked flow is dependent on the product of C_v times F_L*!).*

A word of caution: Never choose a valve that is less than half the pipe size. Too much stress may be created by the movement of the pipes. The only two times in my life when I was genuinely scared were (1) witnessing a live test with steam on a newly designed steam dump valve for nuclear power and (2) seeing a 1 in. steam makeup valve welded into a 10 in. schedule 80 pipe, reducing 1800 psi steam to 10 psig!

There is a good reason why I put this discussion at this particular point in the chapter. Namely, reducers do not affect you that much. Simply remember that the combined flow coefficient of the *valve and reducer* is $C_v F_p$ where F_p is the piping geometry factor (which I used to call R,[6] a term which was used by one manufacturer). Some vendors have published tables showing the reduced $C_v = (C_v F_p)$ due to the pressure losses caused by the pipe reducers. *Make sure the C_v or $C_v F_p$ value of the valve you select is at least 10% higher than the C_v number calculated with the above equations.* For globe valves at rated C_v's of 10 d^2 or less (d is valve size in inches), F_p is practically one, so it can be ignored. Things are more critical with butterfly or ball valves where F_p can be as low as 0.6!

Here is a quick guide to F_p factors. First, select a valve for say ½ the diameter of the pipe, look up the max. C_v of this valve from your vendor's catalog. Now divide the valve catalog C_v by the valve diameter d (inches) squared, then read F_p:

Table 5-1. F_p Values for Valves, D/d = 2

C_v / d^2	10	15	20	25	30	35	40	45
F_p	0.96	0.91	0.85	0.79	0.74	0.68	0.63	0.59

D = pipe diameter (inches)

Next you calculate the required C_v from the given process date and, finally divide this calculated C_v by F_p. This is the final C_v that should be stated on your purchase order. *Note: You may have to start over if the originally selected valve was not large enough, although this is rare.*

For example. You have a 4″ pipe. Next select a 2″ eccentric rotary plug valve with a catalog C_v of 60. C_v/d^2 is 15. This gives an F_p factor of 0.91 from the above table. Now you determine that the process data call for a required C_v of 45. Dividing 45 by 0.91 requires a min. rated C_v of 49.5. This is no problem, since the catalog C_v is 60.

CORRECTING FOR VISCOSITY

Since this is a procedure not often encountered, I put it last. Ignore this unless (a) you have a liquid with a viscosity exceeding 40 centistokes (or = 40 centipoises for liquids), or (b) you need really small valves with a C_v of less than 0.1.

Similar to the reducer correction, we have to increase the C_v from that calculated with the standard sizing equations in order to make up for the additional friction caused by the stickiness of the stuff passing through the valve. This is expressed by the "valve Reynolds number factor," F_R. The valve C_v you *really* need = C_v originally calculated/F_R. More precisely,

$$C_{v\ corrected} = \frac{C_{v\ turbulent}}{F_R} \tag{5-9}$$

where:

$C_{v\ turbulent} = q\sqrt{G_f/\Delta P}$ (see valve sizing formulas)

$$Re_v = \frac{17300 F_d q}{C_{vR} F_L} \qquad (Re_v = \text{valve Reynolds number}^7) \tag{5-10}$$

q	=	flow rate, gpm
C_{vR}	=	rated C_v of valve selected (may have to be increased if $C_{v\ corrected}$ is larger.)
F_L	=	pressure recovery factor
v	=	kinematic viscosity, centistokes (10^{-6}m/s^2) = centipoises/G_f. For typical v values, see Tables 5-2 and 5-3.

$$F_d = \text{valve style modifier, } \frac{d_H}{d_o\sqrt{N_o}};$$

where:

d_H	=	hydraulic diameter of a single flow passage (4 x area/wetted circumference), inch
d_o	=	diameter of equivalent circular orifice of a single flow passage

$$d_o = \sqrt{4C_vF_L/38N_o\pi}\text{, inch}$$

N_o	=	number of equal flow passages

Typical Values for F_d: (from reference 8, also refer to Table 14-2)

For Globe valves (C_v > 0.1):

Single-seated, parabolic plug = 0.46

Double-seated, parabolic plug = 0.32

Butterfly valve = 0.57

Segmented ball valve = 0.98 (wide open only) and 6 in. and above

Eccentric rotary plug valve = 0.42

NOTE: The above F_d values agree with those listed in ANSI/ISA-75.01.01 (IEC 60534-2-1 Mod)-2007.

Example: Bunker C Oil @ 120°F; P_1 = 100 psig; P_2 = 75 psig; G_f= 0.97;
q = 26 gpm; v = 750 centistokes;

$$C_{v\ turbulent} = 26\sqrt{0.97/25} = 5.12$$

Select: 1-1/2 in. globe valve, parabolic plug: rated C_v = 28;* F_L = 0.9;
F_d = 0.46 at rated C_v (see Table 14-2); C_vF_L/d^2 = 11.2

$$Re_v = \frac{17300 \cdot 0.46 \cdot 26}{750\sqrt{28 \cdot 0.9}} = 55 \qquad \text{from Equation (5-10),}$$

F_R is about 0.5 for 11.2 = C_vF_L/d^2 from Figure 5-2 (curve 2)

$$C_{v\ corrected} = \frac{5.1}{0.5} = 10.2 \qquad \text{(OK, 36\% rated } C_v\text{) or choose a reduced port.}$$

The F_R number varies with the internal fluid resistance (K factor) of each valve style (8), i.e., C_v/d_o^2 and the F_L factor (for graphical values, see Figure 5-2).

$$\begin{aligned} F_R &= 0.026\sqrt{Re_vK/F_L} \\ &= 0.78\sqrt{Re_v/(C_{vR}/d^2)F_L} \end{aligned} \tag{5-11}$$

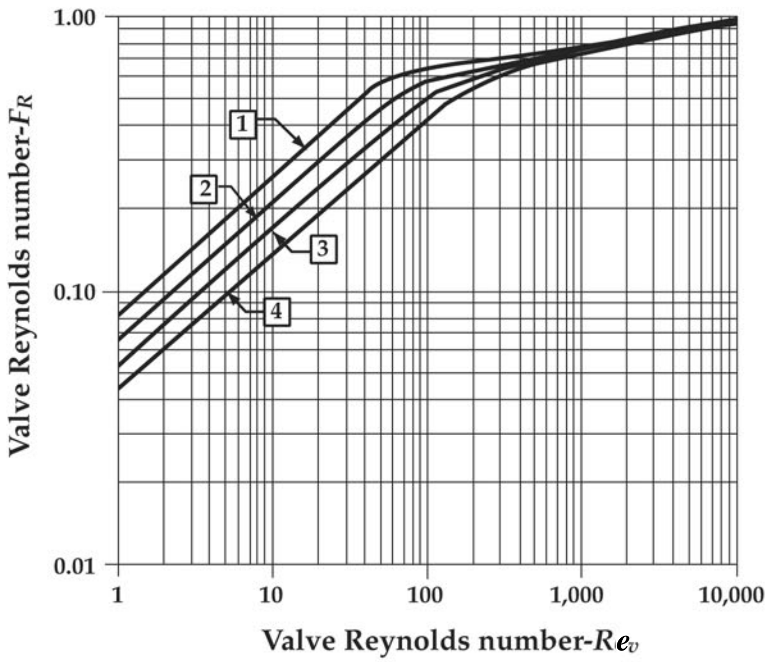
if flow is completely laminar, or

$$F_R = \sqrt[n]{Re_v/10000} \tag{5-12}$$

where $n = 1 + (890d^4/C_{vR}^2\ F_L^{\ 2} + \log (Re_v)$ if flow is transitional.

Since you don't know if the flow is laminar or transitional, calculate F_R with *both* equations, then use the smaller of the two numbers in Equation 5-6 (or get F_R from Figure 5-2). Ignore the viscosity correction completely if Re_v is above 10,000.

* Valve size may be increased if calculations show valve is too small.



After calculating the Reynolds number (Re_v) find F_R from the above graph as function of Valve style and relative flow capacity ($C_d = Cv/d^2$):

Curve 1 is applicable for globe valves with $C_d < 10$.
Curve 2 is applicable for globe valves and eccentric rotary plug valves C_d = 10-15.
Curve 3 is applicable for butterfly valves, C_d = 15-25.
Curve 4 is applicable for ball valves, $C_d > 25$ and for small flow valves $C_d < 1$.

Figure 5-2. Graph[8, 9] of valve Reynolds number factor F_R (*d* = inches)

For control valves with full-size trim, i.e., where the C_v/d^2 exceeds 8 and further assuming a line size valve and assuming the fluid flow is fully laminar, the C_v can be estimated as follows:

$$C_v = 0.194\left(\frac{qG_f v (F_L)^{2.5}}{\Delta P F_d K}\right)^{2/3} \tag{5-13}$$

For globe valves, you may substitute 0.9 for F_L, 10.0 for K, and 0.46 for F_d. This then makes

$$C_{vGlobe} = 0.059\left(\frac{qG_f v}{\Delta P}\right)^{2/3}$$

Example: q = 180 gpm; ΔP = 30 psi; υ = 10,000 cst; G_f = 0.6.

$$C_{vGlobe} = 0.059\left(\frac{180 \cdot 0.6 \cdot 10000}{30}\right)^{2/3} = 64$$

This result means you should select a 3 in. globe valve (rated C_v about 105).

For a contoured ball-type control valve with K = 1.4; F_d = 0.8; F_L = 0.6 (From Table 5-4) Equation 5-13 simplifies to:

$$C_{vBall} = 0.077\left(\frac{qG_f v}{\Delta P}\right)^{2/3}$$

For the above example, this would yield a C_v of 83.7.

Again, a 3 in. size contoured ball valve with a rated C_v of approximately 225 will be appropriate for this job. (Note the uncorrected C_v — assuming turbulent flow is only 25.5!)

If you do not know whether or not the flow is laminar, calculate the required C_v either by the turbulent equation $(C_v = q\sqrt{G_f/\Delta P})$ or by Equation 5-9. Now choose the larger C_v number. Admittedly, this method is not too accurate in the transitional flow area (errors of up to 20% are possible) but is simple and easy to use, at least for an approximation.

Viscosity Conversion

If viscosity is given in centipoises, divide centipoise (μ) by specific gravity (G_f) of liquid to obtain centistokes. If viscosity is given in Saybolt Seconds Universal *(t)*, approximate centistokes by multiplying Saybolt seconds by 0.2 (example: t = 89, v = 18).

Small Flow Valve Sizing

Now for small flow valves $(C_v \le 0.1)$ $F_L \approx 1.0$, we calculate the valve Reynolds number Re_v as before, except we use different F_d factors,[10] namely:

- 0.7 for splined plugs
- 0.3 for short travel diaphragm valves with a flat seating surface
- $F_d = 0.09\ \sqrt{C_v}/(\textit{plug diameter})$, for close clearance tapered plugs

Example: For a 1/8 in. diameter close clearance plug with a C_v = 0.05:

$$F_d = 0.09\sqrt{0.05}/0.125 = 0.16$$

For liquids:

$$Re_v = \frac{17300 \cdot F_d \cdot q}{v\sqrt{C_{vR}F_L}} \tag{5-14}$$

For gases:

$$Re_v = \frac{2153 \cdot F_d \cdot q}{v\sqrt{C_{vR}F_L}} \tag{5-15}$$

where q = flow in scfh at 14.7 psia and 60°F.

Now calculate F_R:

If $Re_v \geq 130$,

$$F_R = \sqrt[n]{Re_v/10000} \quad \text{(Note: } F_R \text{ max.} = 1) \tag{5-16}$$

$$\text{where } n = 1 + K + \log(Re_v) \tag{5-17}$$

and since K for small flow valves = 1, $n = 2 + \log(Re_v)$.

For example:

Fluid: CO_2 gas at 85°F (T = 545°R)

P_1 = 36.3 psia

P_2 = 24.7 psia

v = kinematic viscosity = 8.65 cSt @ 85°F and 14.7 psia (NOTE: Use v at atmospheric pressure if you use q in scfh.)

G_g = specific gravity = 1.516

C_v = valve sizing coefficient (selected) = 0.025

F_L = liquid pressure recovery factor = 1 (from mfr.'s catalog)

F_d = valve style modifier = 0.16 (from mfr.'s catalog)

q = desired max. flow rate = 15 scfh at 85°F and 14.7 psi absolute

T = 460 + 85 = 545°R

From Equation 5-15,

$$Re_v = \frac{2153 \cdot 0.16 \cdot 15}{8.65(1 \cdot 0.025)^{0.5}} = 3778$$

and from Equation 5-16,

$$F_R = \sqrt[5.58]{\frac{3778}{10000}} \; ; \text{ where } n = 2 + \log(3778) = 5.58$$

$$= 0.84$$

Check if the selected valve C_v of 0.025 is sufficient to handle the flow of 15 scfh.

From the turbulent gas equation for sub critical flow* and modified *by* F_R,

$$C_v = q\sqrt{G_g T/\Delta P(P_1 + P_2)}/963F_R$$

$$= 15\sqrt{(1.516 \cdot 545)/(11.6 \cdot 61)}/(963 \cdot 0.84)$$

$$= 0.0193$$

so the selected valve C_v of 0.025 is OK.

Table 5-2. Viscosities and Specific Gravities for Liquids

Liquid	Formula	G_f	T(°F)	v(cSt)
Acetic acid	$HC_2H_3O_2$	1.05	68	1.17
Acetone	C_3H_6O	0.79	68	0.42
Alcohol, ethyl	C_2H_6O	0.79	68	1.52
Alcohol, methyl	CH_4O	0.79	68	0.74
Ammonia	NH_3	0.62	68	0.24
Aniline	C_6H_2N	1.02	60	4.31
Auto oil, SAE 30		0.94	100/210	114/11.2
Benzene	C_6H_6	0.88	68	0.74
Brine, CaCl 25%	CaCl	1.23	68	2.03
Brine, NaC125%	NaCl	1.19	68	1.60
Bromine	Br_2	2.93	60	0.34
Carbon dioxide	CO_2	0.84	-109	0.83
Carbon disulfide	CS_2	1.30	32	0.33
Carbon monoxide	CO	0.84	-314	0.21
Carbon tetrachloride	CCl_4	1.59	77	0.57
Castor oil		0.96	68	1027.00
Chlorine	Cl_2	1.50	-100	0.50

* Remember, there is no sonic velocity if flow is non-turbulent; therefore, critical flow equations don't apply.

Table 5-2. Viscosities and Specific Gravities for Liquids

Citric acid	$C_6H_8O_7$	1.11	68	2.00
Dow Therm A		.79	68	5.60
Dow Therm A		.79	200	1.07
Dow Therm E		.79	68	1.93
Dow Therm E		.79	200	0.53
Ethylene glycol	$(CH_2OH)_2$	1.12	68	16.59
Fluorine	F_2	1.51	-306.4	0.16
Formaldehyde	H_2CO	0.82	60	0.35
Freon 12	CCl_2F_2	1.31	77	0.47
Fuel Oil, #6 (Bunker C)		0.98	85	4080.00
" " " "		0.97	110	1100.00
Gasoline		0.71	60	0.86
Glycerine (100% glycerol)	$C_3H_8O_3$	1.26	68	920.00
Glycol (Ethylene glycol)	$C_2H_6O_2$	1.11	60	17.90
Hydrochloric acid (31.5%)		1.64	60	1.52
Hydrogen	H_2	0.074	-422	0.011
Hydrogen chloride	HCL	0.91	68	2.67
Hydrogen fluoride	HF	0.97	68	0.13
Hydrogen sulfide	H_2S	0.79	60	
Isopropyl alcohol	C_3H_8O	0.79	68	2.94
Linseed oil		0.93	60	55.90
Kerosene		0.82	85	2.99
Mercury	Hg	13.57	60	0.12
Napthalene	$C_{10}H_8$	1.15	68	398.00
Nitric acid (40%)	HNO_3	1.25	68	1.25
Nitrogen	N_2	0.85	-320	0.19
Oxygen	O_2	1.20	-297	0.17
Phosphoric acid	H_3PO_4	1.15	68	1.85
Propylene glycol (60%)	$C_3H_8O_2$	1.04	68	8.99
Sodium hydroxide	NaOH	1.27	68	6.20
Starch (4% Solution)	$(C_6H_{10}O_5)x$	1.02	100	100.00
Sulfuric acid (100%)	H^2SO_4	1.83	68	13.91
Toluene	C_7H_8	0.87	68	0.68
Trichloroethylene	C_2HCl_3	1.46	80	0.39
Turpentine		0.87	60	1.71
Water, fresh	H_2O	1.000	60	1.00
Water,sea (4%saltbywgt)		1.03	60	1.05

Table 5-3. Viscosities and Specific Gravities for Gases at Atmospheric Pressures

Gas	Formula	G_g	T(°F)	v(cSt)
Acetic acid	$HC_2H_3O_2$	2.07	246	5.75
Acetone	C_3H_6O	2.01	212	4.91
Acetylene	C_2H_2	0.907	32	7.99
Air	N_20_2	1.000	64	15.10
Alcohol				
Butanol, butyl	$C_4H_{10}O$	2.56	242	10.94
Ethanol, ethyl	C_2H_6O	1.59	212	7.20
Methanol, methyl	CH_4O	1.11	152	11.77
Propanol, propyl	C_3H_8	2.07	212	4.76
Ammonia	NH_3	0.596	68	13.71
Argon	A	1.379	68	13.38
Benzene	C_6H_6	2.70	58	2.23
Bromine	Br_2	5.52	55	2.21
Carbon dioxide	CO_2	1.5290	68	8.06
Carbon disulfide	CS_2	2.63	68	2.88
Carbon monoxide	CO	0.97	59	14.51
Carbon tetrachloride	CCl_4	5.31	170	3.65
Chlorine	Cl_2	2.486	68	4.44
Cyanogen	C_2N_2	1.806	63	4.51
Ethane	C_2H_6	1.049	63	7.08
Ether	$(C_2H_5)_2O$	2.56	58	2.28
Ethyl chloride	C_2H5CL	2.220	32	3.27
Ethylene	C_2H_4	0.975	68	8.60
Fluorine	F_2	1.31	68	13.98
Freon II	CCl_3F	5.04	68	1.77
Freon 12	CCl_2F_2	4.2	68	2.46
Helium	He	0.138	-431	0.89
Helium	He	0.138	68	117.06
Hydrogen	H_2	0.0695	-431	0.37
Hydrogen	H_2	0.0695	68	104.90
Hydrogen bromide	HBr	2.819	66	5.35
Hydrogen chloride	HCl	1.268	64	9.32
Hydrogen cyanide	HCN	0.93	68	8.95
Iodine	I_2	8.76	338	2.93
Isopropyl alcohol	C_3H_8O	2.08	212	5.55
Krypton	Kr	2.89	59	6.96
Mercury	Hg	6.93	523	11.05

Table 5-3. Viscosities and Specific Gravities for Gases at Atmospheric Pressures

Methane	CH_4	0.554	68	15.41
Methyl acetate	$C_3H_6O_2$	2.56	212	4.14
Methyl chloride	CH_3Cl	1.74	59	4.89
Methyl iodide	CH_3I	4.47	811	14.66
Neon	Ne	0.69	68	37.53
Nitric oxide	NO	1.04	68	15.01
Nitrogen	N_2	0.97	81	15.66
Nitrosyl chloride	ClNO	2.26	59	4.12
Nitrous oxide	N_2O	1.52	80	8.33
Oxygen	O_2	1.10	66	15.23
Pentane (n)	C_5H_{12}	2.49	77	2.30
Propane	C_3H_8	1.52	64	4.32
Propylene (propene)	$C_2H_4(CH_2)$	1.45	62	4.71
Sulfure dioxide	SO_2	2.21	69	4.73
Saturated steam (30 psia)	H2O	0.62	250	12.20
Toluene	C_7H_8	3.18	68	1.83
Xenon	Xe	4.53	68	4.15

NOTE: For pressures other than atmospheric, multiply listed v by 14.7/P_1, where P_1 = actual inlet pressure in psia.

SIZING CONTROL VALVES FOR VISCOUS FLUIDS - AN EXPLANATION

I always wondered how a ball valve with hardly any obstruction can pass the same amount of tar* for a given pressure drop and flow coefficient *(C_v)* than a cage valve with, say, 100 small, drilled holes even though it has the same C_v as the ball valve. Instinct tells us this could not be so despite the fact the previous ISA standard told us so. Not believing it, I decided to do something about it and came up with the following revised method.

Also, the F_R for small flow valves derived from my new equations did not agree with values given in Figure 1 of the former ISA-75.01.01, Flow Equations for Sizing Control Valves (now revised), and yield more accurate results typically increasing the required C_v for small flow valves by a factor of four or more over the current method in Reynolds numbers below 100. The former ISA (single) F_R curve was based on a limited number of tests with full-sized globe valves while the new values are based on air and water tests with small flow valves, or rotary valve types in addition to globe valves.

From data published in Perry's *Chemical Engineers Handbook*, Sixth Edition, Table 5-15, it appears that more streamlined valves (e.g., angle valves and gate

* Ignoring the F_L factor, which affects the Reynolds number.

valves) exhibit F_R factors close to Figure 5-2 curves, while disk and plug-style globe valves (due to their more abrupt flow pattern) tend to favor the former ISA F_R values when corrected with the proper F_d values.

From this data, it appears that the "pseudo C_v", i.e., the "laminar flow C_v,"or $C_vF_RF_L$, is identical for all valve types with full size trim and at low Reynolds numbers ($Re_v < 100$).

From this, it can be stated that the required C_v (valve capacity needed to pass viscous flow for a given Reynolds number) is

$$C_{vReq} = A_p/(0.026\sqrt{K})$$

where 0.026 is the area in square inch per 1 C_v when the head loss coefficient $K = 1$. A_p is the cross-sectional area of the given pipe size in inch^2 (same as valve size!).

Furthermore, from the ISA equation, $C_{vReq} = C_i/F_R$, where C_i is the C_v calculated assuming the flow pattern is completely turbulent. This makes $C_i = A_p\sqrt{Re_v}/F_L$, and since the valve Reynolds Number for liquids is,

$$Re_v = \frac{17300 \cdot F_d \cdot q}{v\sqrt{F_L C_{vReq}}}$$

$$C_{vReq} = 0.194\left(\frac{qG_f v(F_L)^{2.5}}{\Delta P F_d K}\right)^{2/3}$$

This is Equation 5-13.

Since F_L is approximately proportional to the relative valve capacity (C_v/d^2), we may, for estimating purposes only, substitute

$$F_L \cong 0.96 - [0.0025(C_v/d^2)^{1.5}] \text{ in Equation 5-13.}$$

Furthermore, F_d is somewhat related to F_L (for example a large valve, such as a ball valve, has a low F_L but a high F_d). We can thus further estimate that

$$F_d = 0.4/F_L \text{ and, finally, } K = 890\ \mathrm{d}^4 / \mathrm{C_v}^2.$$

This then makes Equation 5-13

$$C_v = 0.00386\left\{\frac{qG_f v\left[0.96 - 0.0025\left(\frac{C_v}{d^2}\right)^{1.5}\right]^{3.5} C_v^2}{\Delta P d^4}\right\}^{2/3}$$

NOTE: The selected C_v should not be less than 10 d^2 and no more than *30* d^2.

I suggest you input the customer's pipe size for *d*. The above equation will give you a ball park C_v number which can aid in selecting a valve type. Using the actual valve coefficients (rated C_v, F_L, F_d), recalculate the required C_v from Equation 5-9.

METRIC UNITS

Normally, it does not bother you that the rest of the world thinks in metric terms. However, an occasion may arise when you will have to do a project for an overseas plant, and you will be confronted with liquid flow rates of cubic meters per hour instead of gpm. Luckily, the C_v numbers work equally as well in Europe. For example:

$$C_v = 1.17q/\sqrt{\Delta P/G_f} \tag{5-18}$$

where:

q = m^3/hr of liquid

ΔP = bar,

G_f = specific gravity (water @ 60°F = 1)

As a matter of fact, in Germany, they go so far as to eliminate the 1.17 constant and then call the flow coefficient K_v; hence, K_v = 0.85 C_v, or C_v = 1.17 K_v.

Complications arise only if you are confronted by newtons per square meter, also called pascal or Pa. To get back to the more familiar bar, divide Pa by 100,000 or one bar = 100,000 Pa, and one bar = 14.5 psi. One kilopascal (kPa) = 0.145 psi. There is also the less often used kg/cm2, which is 0.98 bar or 14.2 psi.

Remember also, that degrees Celsius is used abroad where °F = (°C x 9/5) + 32, and in absolute terms, °K = 273 + °C. Converting flowing units:

1 m^3/hr liquid = 4.4 gpm

1 m^3/hr gas = 35.3 ft^3/hr

kg/hr = 2.2 lbs/hr

You are now fully equipped to work abroad.

WHAT SIZE VALVE TO CHOOSE

After you have labored hard to select the final required valve C_v suitable for maximum flow requirements, and having considered the ***F*** effects *(F_L, F_p, F_R)* you are now confronted with the final choice—the valve size.

If the pressure drop is low, say, 5 to 15 psi, it is always safe to start with a line size valve. Keep in mind that the person in the Piping Design Department already sized the downstream or upstream pipe to pass the design flow rate at a reasonable velocity. Incidentally, valve outlet velocity should not bother you unless there could be a noise problem with high pressure gas or steam reduction (see aerodynamic noise discussion in Chapter 14), or where you have flashing liquids (if the downstream pipe is too long, your valve will cavitate because of downstream pressure buildup caused by high mixed-phase velocity). This then leaves only the C_v rating of the valve. In case you have not yet selected a certain brand because you are partial to the color of the valve, i.e., green or red, here is a hint:

Table 5-4. Typical Flow Coefficients at Rated Travel

Type	C_v/d^2	F_L	C_vF_L/d^2	C_vF_p/d^2		for	
				D/d=1.5		D/d=2	
Single-seated globe valve							
Parabolic plug, flow-to-open	10	0.9	9	9.8	(4.3)	9.6	(2.4)
Double-seated globe valve	12.5	0.9	11.3	12	(5.3)	11.7	(2.9)
High capacity cage valve							
to size 3 in. only	14	0.9	12.5	13.5	(6.0)	12.9	(3.2)
Eccentric rotary plug valve	12 0.	85	10	11.6	(5.1)	11.3	(2.8)
Butterfly valve 60° open	17	0.68	11.5	15.8	(7.0)	15	(3.8)
Butterfly valve 70° open	27	0.57	15.5	23	(10.2)	20.8	(5.2)
Fluted butterfly valve @ 70°	25	0.7	17.5	21.8	(9.7)	19.8	(5.0)
Contoured ball valve @ 90°	25	0.6	15	21.8	(9.7)	19.8	(5.0)
Angle, venturi type,							
flow-to-close	22	0.5	11	19.8	(8.8)	18.3	(4.6)

D = inches

The C_v values are per *valve* diameter (inch) squared. The C_v values in parentheses are per *pipe* diameter (inch) squared. For example: A 6 in. eccentric rotary plug valve has a wide-open, C_v flow coefficient of 6^2 x 12 = 432. However, when installed between reducers in a 12 in. pipe, the available flow coefficient is only 6^2 x 11.3 = 407 (see column C_v/d^2 for D/d = 2), or 12^2 x 2.8 = 403 using the figures in parentheses.

The above values are conservative and vary somewhat from size to size. You may notice that when high-capacity valves are used under choked flow conditions (such as steam pressure reduction), they all tend to lose their high C_v

advantage, as you can see from the $C_v\ F_L/\ d^2$ column! Explain this to your valve vendor.

Starting out with a line size valve, divide your desired maximum C_v value by 0.8. First, this will give you a 10% safety factor and second, it will guard against the ±10% tolerance in the C_v rating of the valve given by the manufacturer. Now divide this new C_v number by the pipe size squared. For example: From the flow data, you require a maximum C_v = 123. Your line size is 4 inches. First divide 123 by 0.8 to obtain 154. Now divide this by $(4\text{ in.})^2$. $154/4^2 = 9.6\ C_v\ /\ d^2$.

From the data given in Table 5-4, you have the following choices:

- A 4 in. single-seated globe valve, $C_v\ /\ d^2 = 10$; C_v rated = $10 \times 4^2 = 160$.
- A butterfly valve at 70° opening (if other considerations such as leakage, noise, etc. permit). Here a 3 in. butterfly valve installed in a 4 in. pipe has a flow capacity higher than $10.2 \times D^2$ (see figures in parentheses), i.e., more than $4^2 \times 10.2 = 163$ C_v.

Having thus determined the size (and type) of valve, you now should verify that you can meet the minimum required flow coefficient that your process requires. Here you have to go to the manufacturer's C_v tables (usually printed in increments of 10% travel). Look up the approximate valve travel when the valve is operating at your minimum C_v. This travel should not be less than 5% of the total valve travel or less than the percentage of the valve dead band (with tight packing).* This is usually 5% with TFE packing to 15% for laminated graphite packing if *no* positioner is used. For a typical TFE-packed valve and no positioner, this limits the valve travel to 10% from the shut-off position. The only exceptions are certain soft seat plugs that "dip" through their seal ring, ball valves, elastomer-lined butterfly valves with angle-seated vanes, or certain diaphragm or pinch valves where there is no "hard" seating action. If a standard valve forces you to throttle below 5% travel, then your only choice is to use a larger and a smaller valve in parallel and split-range the actuator signal, or better yet, use two separate signals from your computer so that the small valve will open first, and then the larger valve starts when the small valve is about 80% open.

In summary:

- Choose a valve that has a rated C_v at least 20% larger than your maximum requirements.
- See that the minimum required C_v occurs at least above 5% of the valve travel (check vendor's catalog).

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* You may have to use 10% if the 5% travel C_v is not known.

ADJUSTABLE TRAVEL - ADJUSTABLE C_v

As mentioned previously, one of the vexing problems confronting the instrument engineer is the fact that the final required flow capacity that the selected valve needs is sometimes less than half of what was specified. The result is an effective signal span of less than 40% (4 - 10.4 mA) in case of a linear plug characteristic and less than 75% (4 - 16 mA) with an equal percentage flow characteristic.

Normally, the controller or the computer does not have a problem controlling the loop with a reduced signal span since the signal resolution is still about one in one thousand for 50% of signal span. However, the minimum controllable flow at 5% travel is higher (as a percentage of the maximum flow), and, worse, the control valve dead band is more than twice as much as a percentage of the usable signal span, thereby affecting loop stability!

It would, therefore, be nice if we could reduce the rated valve C_v after startup, hopefully with the valve staying in the line. This is preferable to sending the valve back to the maintenance shed for installation of a reduced trim.

Well, valves are now available where you can adjust the rated C_v within reasonable limits, and it may be useful to get familiarized with them. The illustration in Figure 5-3 shows such a valve in a schematic way to demonstrate the principle of adjustment in flow capacity. Here, a conventional spring-diaphragm actuator (1) moves a linkage (2) up and down, which, in turn, connects to a valve stem (3). The stem, in turn, moves a valve plug (4) against a seat ring (5). In the normal actuator position, A, the valve plug travels the maximum distance, H, which in turn yields the maximum valve C_v, say 300.

With an equal percentage plug contour, you get a typical "inherent flow characteristic" as shown in Figure 5-3(B). Now assume that after installation we find the valve needs a flow coefficient of only 105 (equivalent to 70% of the present travel). To increase the usable signal span (= actual travel) to, say, 90% from the actual 70% (allowing a 10% safety margin), we can now relocate the actuator (1) along mounting bracket (6) to a new position, B.

Owing to a greater distance L_2 (not to scale) from the pivot point (7), the excursion of the lever (2) is reduced, resulting in a new travel H of 78% of the travel we got at actuator position A. Assuming H was 2 in. at position A, we now get 1.56 in. at B. The equal percentage characteristic at this point yields a C_v *of* 150, as you can see from Figure 5-3(B). With the reduced travel, we can now "spread" the curve over the available 100% signal span or over 1.56 in. plug travel. The result is shown in Figure 5-3(C).

To help you understand how this graph was developed, assume you stroke the plug to 60% travel in actuator position B. Reducing the travel now by 22% (to 78% of 2 in., or 1.56 in. travel) will yield a new plug percentage travel of 0.78 x 0.6 = 0.468 or 46.8%. The actual C_v of the plug at this point is about 37 (from Figure 5-3(B)). However, since 46.8% of the actual plug travel corresponds to 60% of the actuator travel in position B, you have to plot the 37 C_v at the 60% travel position (equivalent to 60% actuator travel, or 60% of signal) in order to construct the curve in Figure 5-3(C).

It is interesting to note that when comparing the two curves at 10% travel, we find a C_v value of 10 for the full travel plug but only 7.5 C_v for the reduced travel

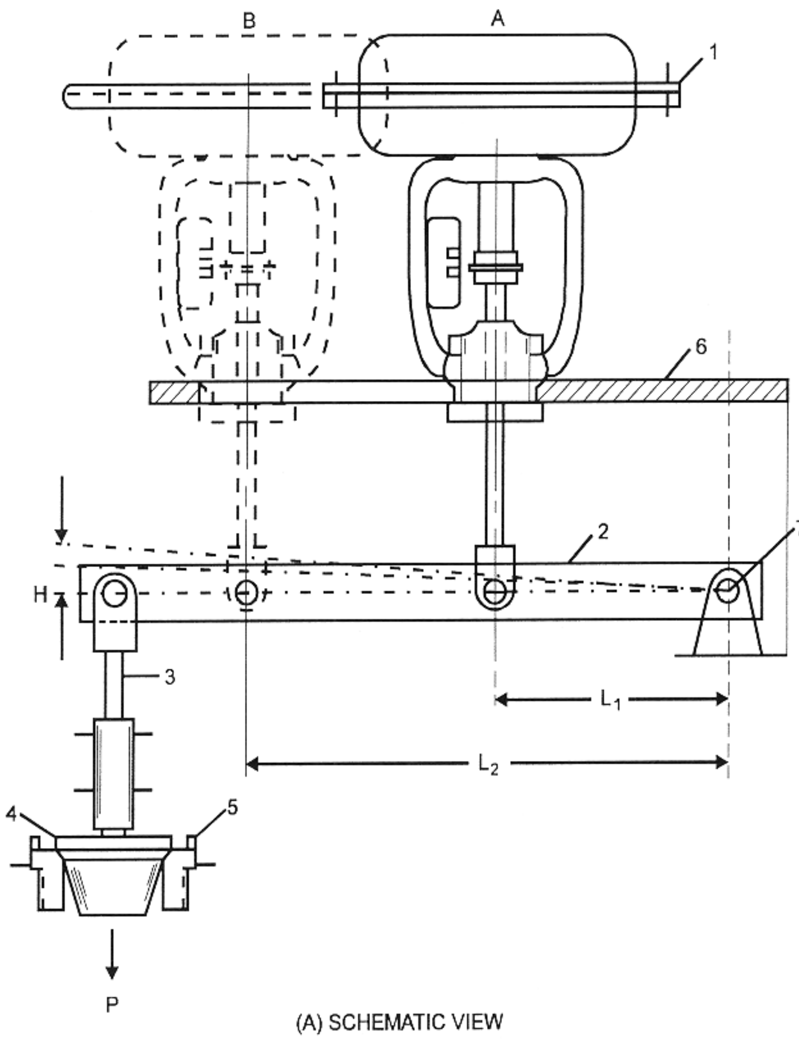

Figure 5-3. Control valve with equal percentage characterized plug and adjustable travel.

plug, yielding a somewhat wider, useful rangeability, in this case, (105 C_v /10 C_v = 10.5:1 over 105 C_v /7.5 C_v = 14:1!).

Of greater significance is the fact that by locating the actuator (1) closer to the valve stem (3), we greatly increased the available plug force P, in the present example by (1/0.78) = 1.28 times. This not only complements the typical increase in pressure drop across the valve with decreasing flow capacity but also reduces the effects of packing box friction on actuator positioning accuracy (dead band). As a matter of fact, you get a double effect: you get an increase in actuator force over the constant packing friction, but you also divide this friction over the larger signal span!

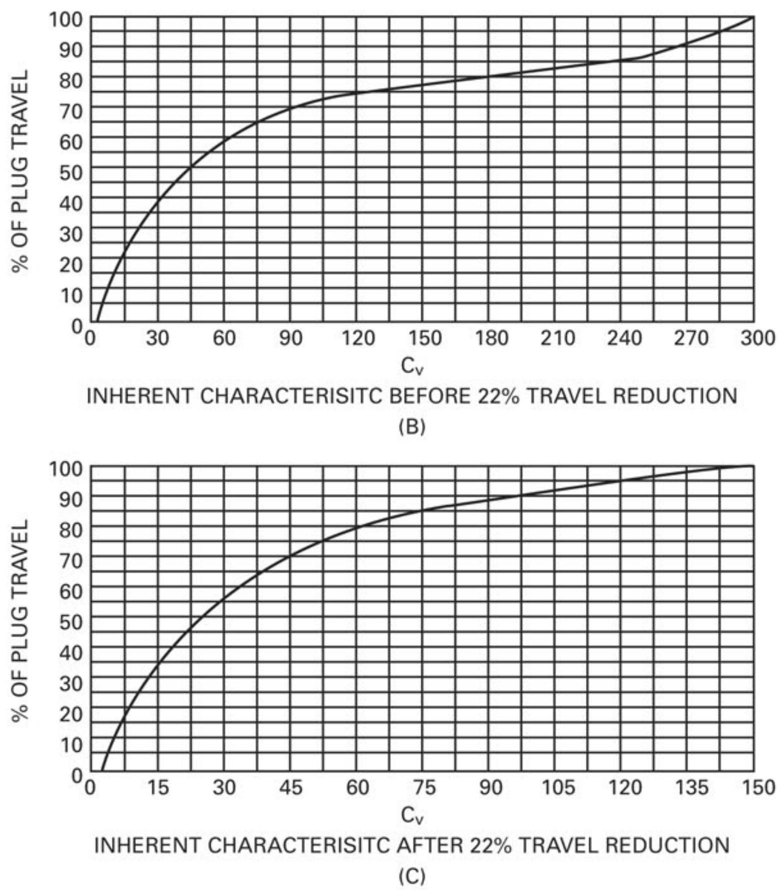

Figure 5-3. (Continued)

Again, using the above example: (a) friction, as seen by the actuator, is reduced to 78%, and (b) lower friction is distributed over 100/78% signal span. Total dead band (as caused by valve friction only) reduction is therefore 39%, [100 x (1 – 0.78^2].

While the above scheme has beneficial effects concerning actuator forces and dead band, we should not expect improvements in the "installed" valve gain. Granted, the gain change (see Figure 8-1) is now spread over a greater signal range; however, the percent of flow at which the "limit in gain change" is exceeded is still about the same.

The above example is conservative since it assumes an equal percentage plug characteristic. If the case had been a linear valve, then with the same C_v decrease, the plug travel would have to be reduced to 40% of the original travel, yielding an actuator force increase of 250% and a packing friction reduction of 84%! Simply reducing the actuator travel of a conventional valve by merely recalibrating the valve positioner will not do the same job. You still get the same actuator force and the same dead band, except the dead band is now divided into a larger signal span, thereby resulting in a lower percentage of signal offset!

The travel of a valve plug or rotary closure member cannot be reduced indiscriminately. For example, with some rotary valves, the seating friction near the point of shut-off becomes very high. Also, most parabolic plugs don't extend their contoured portion all the way to the seating bevel, but are cylindrical for measuring purposes. Therefore, a reduction of valve travel to less than 40% of the original travel does not seem advisable. Also, such travel adjustment is not practical with valves that have an inherent low rangeability, i.e., a low (<20:1) ratio between maximum C_v and minimum C_v.

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