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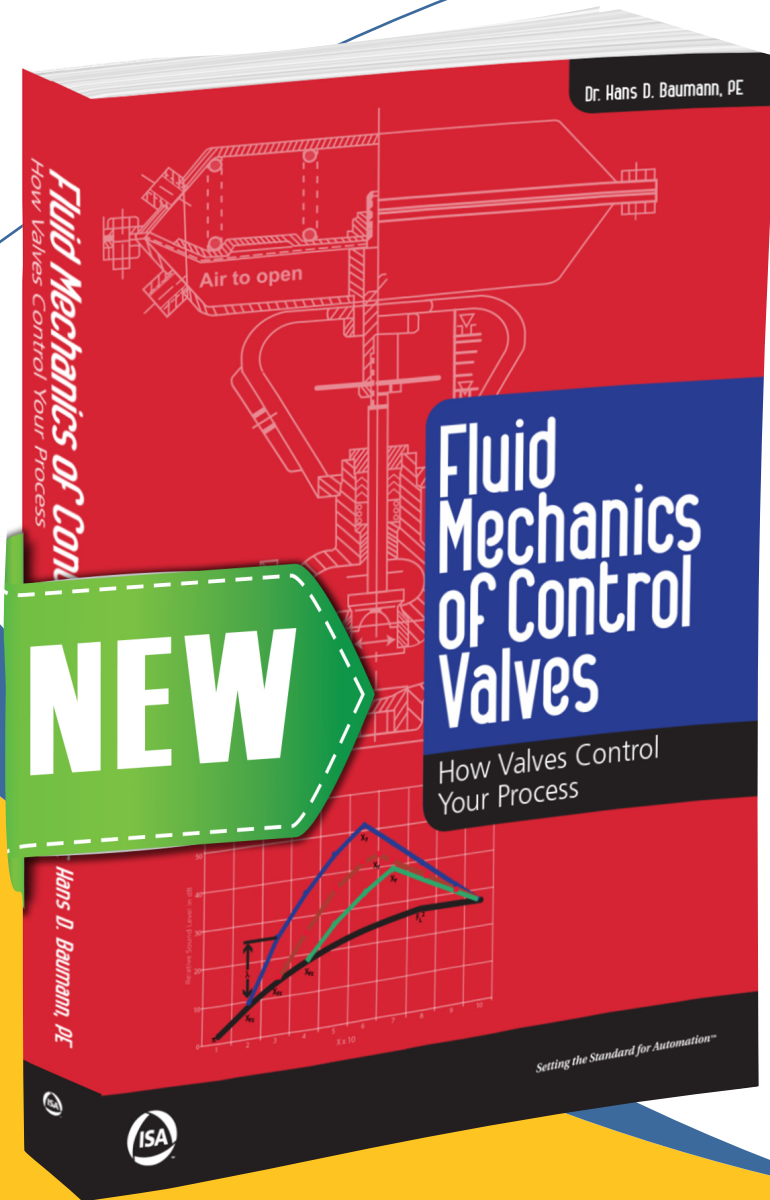
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*Setting the Standard for Automation™*



# Fluid Mechanics of Control Valves

By Hans D. Baumann



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10 9 8 7 6 5 4 3 2

ISBN: 978-1-64331-004-6

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ISA  
67 T. W. Alexander Drive  
P.O. Box 12277  
Research Triangle Park, NC 27709

**Library of Congress Cataloging-in-Publication Data in process**

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# About the Author

Dr. Hans D. Baumann, PE, held managerial positions in Germany, France, and the United States, before being promoted to vice president of engineering at Masoneilan and later to senior vice president at Fisher Controls. He worked as a consultant for international clients prior to starting his own valve-manufacturing company, which he subsequently sold to Fisher Controls.

Baumann's many valve designs have won awards in the United States, France, and Japan, as well as a gold medal in Germany. He is credited with over 105 US patents and many foreign patents. Baumann has published 148 articles and has written 7 books on engineering, management, and historical subjects, including *Control Valve Primer* (now in its fourth edition, including a Japanese translation). In addition, he is the co-author of 8 handbooks on automatic controls and acoustics.

Baumann is an honorary member of the Fluid Controls Institute (FCI) and the Spanish Engineering Society. He is also an ISA Honorary Fellow, a Life Fellow of the American Society of Mechanical Engineers (ASME), and an inductee of the Automation Hall of Fame. ISA selected him as one of 50 outstanding Industrial Innovators.

Besides the above accomplishments, he continues to work as a consultant and to lead R&D projects for select clients.

Hans Baumann was 86 years old when he wrote this book. He can be contacted at [hdbaumann@att.net](mailto:hdbaumann@att.net).

# Introduction

## An Often-Asked Question: What Is a Control Valve?

The simple answer is: *A valve that controls the rate of flow in a piping system as commanded by a pneumatic or electronic signal.*

Unfortunately, it reduces available energy while doing so.

There are other devices available to control fluids, such as liquids or gases using speed-controlled pumps, speed variable blowers, and others. While some of those devices may be somewhat more energy efficient than control valves, none can escape the dictates of the First Law of Thermodynamics, stating that energy cannot be destroyed. Nevertheless, energy always decays to a lower level if it is converted from one form to another (more on this later).

To better understand what I mean by *rate of flow*, consider this example: When one fills a tea kettle, one opens the water faucet fully to begin with and maintains that flow until the kettle is about three-quarters full. Then, one starts closing the faucet to reduce the rate of flow and avoid spillage. One keeps doing this until the flow starts to drip and then closes the valve completely once the kettle is full.

A more technical example would be the case in which a control valve controls the filling of a storage tank with crude oil originating from a pipeline. A level controller is used to sense the tank level and send a signal to the control valve. Here again, the controller will instruct the valve to be fully open until the tank is about 80% full in order to speed up the filling process. Thereafter, the controller instructs the valve to start



closing in proportion to the remaining empty tank space. This would mean that for every 2% of tank level, the valve's travel will be reduced by 10%, until the valve is fully closed when the tank is full. It is important to understand that any control scheme—whether it is the reduction of pressure or control of temperature, level, viscosity, mixing fluids, or pH level—always involves *controlling the rate of flow*.

Valve-sizing equations and noise calculations, which you will encounter later, are now taken for granted, yet they have surprisingly recent origins. Here is a brief history:

- 1940s:  $C_v$ , a coefficient defining the flow capacity of a valve
- 1963:  $F_L$ , a pressure recovery factor defining pressure recovery in a valve
- 1968:  $F_p$ , a factor used to estimate pressure loss in a valve's adjacent pipe reducers
- 1970: The first scientific method to estimate a valve's aerodynamic noise levels
- 1990:  $F_d$ , a factor used to calculate the fluid's jet diameter
- 1994: The first standard to estimate hydrodynamic noise levels of valves

The International Society of Automation (ISA) has played a major role in publishing articles and standards relating to the above advancements.

Did you know that prior to 1973, the pressure rating of flanges and valves was done using pounds of steam as reference (such as the 125 lb designation)? The current class system for pressure rating was not introduced until 1973 by this author.

While it is true that any personal computer can handle complex sizing equations, I still find it useful to have some simple formulas that can be handled with the aid of a calculator or smartphone, if the need arises. This led me to propose shortcuts in the processes of sizing valves and estimating sound levels.

Controlling the rate of flow in a control valve is not always uneventful; one must account for destructive vibration of internal parts (trim), choked-flow conditions, destructive cavitation, or excessive noise. All these phenomena will be discussed in the following chapters.

This book is intended as a guide for young engineering students studying automatic control theory and for more experienced instrument engineers familiar with electronic systems who realize that the final control element has its place and, if selected incorrectly, can damage a complete control system. It may be due to a lack of experienced instrument engineers that current engineers tend to rely more and more on the advice of vendors when it comes to selecting control valves. That may not always be

## 1

# Control Valves

Since the onset of the electronic age, because of the concern of keeping up with ever-increasing challenges by more sophisticated control instrumentation and control algorithms, instrument engineers have paid less and less attention to final control elements—even though all process control loops could not function without them.

Final control elements may be the most important part of a control loop because they control process variables, such as pressure, temperature, tank level, and so on. All these control functions involve the regulation of fluid flow in a system. The control valve is the most versatile device able to do this. Thermodynamically speaking, the moving element of a valve—may it be a plug, ball, or vane—together with one or more orifices, restricts the flow of fluid. This restriction causes the passing fluid to accelerate (converting potential energy into kinetic energy). The fluid exits the orifice into an open space in the valve housing, which causes the fluid to decelerate and create turbulence. This turbulence in turn creates heat, and at the same time, reduces the flow rate or pressure.

Unfortunately, this wastes potential energy because part of the process is irreversible. In addition, high-pressure reduction in a valve can cause cavitation in liquids or substantial aerodynamic noise with gases. One must choose special valves designed for those services.

There are other types of final control elements, such as speed-controlled pumps and variable speed drives. Speed-controlled pumps, while more efficient when flow rates are fairly constant, lack the size ranges, material choices, high pressure and temperature ratings, and wide flow ranges that control valves offer. Advertising claims

touting better efficiency than valves cite as proof only the low-power consumption of the variable-speed motor and omit the high-power consumption of the voltage or frequency converter that is needed.

Similarly, variable-speed drives are mechanical devices that vary the speed between a motor and a pump or blower. These do not need an electric current converter because their speed is mechanically adjusted. Control valves have a number of advantages over speed-controlled pumps: they are available in a variety of materials and sizes; they have a wider rangeability (range between maximum and minimum controllable flow); and they have a better speed of response.

To make the reader familiar with at least some of the major types of control valves (the most important final control element), here is a brief description.

## Valve Types

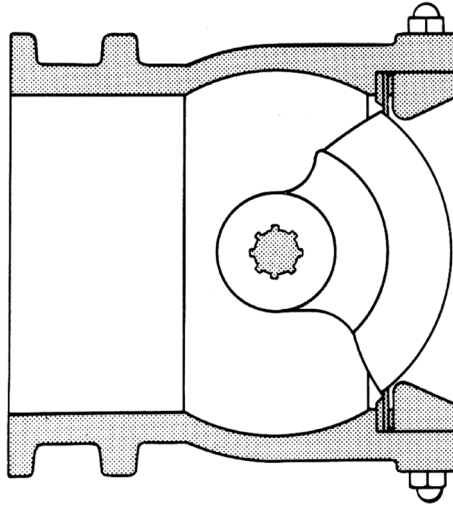
There are two basic styles of control valves: rotary motion and linear motion. The valve shaft of rotary motion valves rotates a vane or plug following the commands of a rotary actuator. The valve stem of linear motion valves moves toward or away from the orifice driven by reciprocating actuators. Ball valves and butterfly valves are both rotary motion valves; a globe valve is a typical linear motion valve. Rotary motion valves are generally used in moderate-to-light-duty service in sizes above 2 in (50 mm), whereas linear motion valves are commonly used for more severe duty service. For the same pipe size, rotary valves are smaller and lighter than linear motion valves and are more economical in cost, particularly in sizes above 3 in (80 mm).

Globe valves are typical linear motion valves. They have less pressure recovery (higher pressure recovery factor  $[F_L]$ ) than rotary valves and, therefore, have less noise and fewer cavitation problems. The penalty is that they have less  $C_v$  ( $K_v$ ) per diameter compared to rotary types.

### Ball Valves

When a ball valve is used as a control valve, it will usually have design modifications to improve performance. Instead of a full spherical ball, it will typically have a ball segment. This reduces the amount of seal contact; thus, reducing friction and allowing for more precise positioning. The leading edge of the ball segment may have a V-shaped groove to improve the control characteristic. Ball valve trim material is generally 300 series stainless steel (see Figure 1-1).

Segmented ball valves are popular in paper mills due to their capability to shear fibers. Their flow capacity is similar to butterfly valves; therefore, they have high pressure recovery in mid- and high-flow ranges.



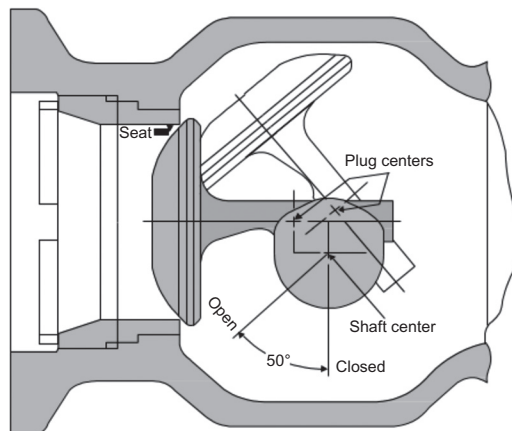
**Figure 1-1.** Segmental ball valve cross section.

*Source: Masoneilan/Dresser (MNI is a division of GE).*

### ***Eccentric Rotary Plug Valves***

Another form of rotary control valve is the eccentric rotary-plug type (see Figure 1-2) with the closure member shaped like a mushroom and attached slightly offset to the shaft. This style provides good control along with a tight shutoff, as the offset supplies leverage to cam the disc face into the seat. The advantage of this valve is tight shutoff without the elastomeric seat seals used in ball and butterfly valves. The trim material for eccentric disc valves is generally 300 series stainless steel, which may be clad with Stellite® hard facing.

The flow capacity is about equal to globe valves. These valves are less susceptible to slurries or gummy fluids due to their rotary shaft bearings.



**Figure 1-2.** Typical eccentric rotary plug valve.

## 2

# Efficiency and Power Consumption of Control Valves

Efficiency and power consumption are directly related to the pressure recovery factor ( $F_L$ ) of a valve type and its travel. Conventional control valves, as described in Chapter 1, typically have  $F_L$  factors ranging between 0.7 and 0.9 for globe valves (due to the interaction of the valve housing and trim; see Chapter 6) and  $F_L$  values of 0.4 to 0.8 for rotary valves. While a lower  $F_L$  factor seems desirable from a power consumption standpoint, such a valve has a lower available flow coefficient ( $C_v$ ) at high to moderate pressure drops.

The First Law of Thermodynamics states that energy is never lost; however, it always degrades to a lower, less usable form. The problem with control valves is that they need pressure drops to accelerate fluids. Because valves are typically installed in a piping system that has its own energy loss (pressure reduction due to friction), the magnitude of this loss is seldom known by the instrument engineer sizing the control valve. Therefore, to play it safe, most control valves are greatly oversized by assuming a lower inlet pressure than what the loop requires. The extra power consumption due to extra pressure is directly related to the valve's pressure drop,  $\Delta P$ , where

$$\text{Power} = Y \cdot 10^{-4} \cdot \Delta P \cdot q \cdot \text{spg in kW}$$

$$q = \text{gpm} \mid \text{spg} = \text{specific gravity (water} = 1) \mid \Delta P \text{ in psi}$$

$$Y = 5.7 \text{ in US customary units and } 344 \text{ if flow is in m}^3/\text{h and } \Delta p \text{ in bar}$$

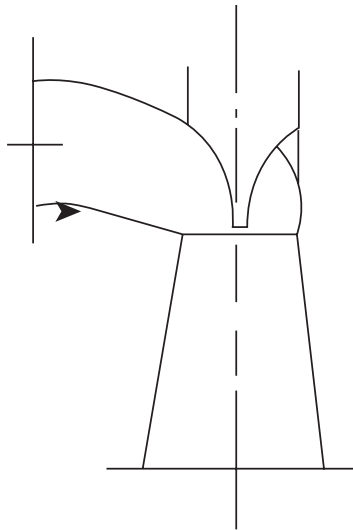
**Example:**

A valve must handle 120 gpm (29 m<sup>3</sup>/h) of oil at a specific gravity of 0.8, and the pressure drop is 50 psi (3.44 bar). The resultant power consumption is 2.7 kW, not including pump efficiency.

If it would be possible to reduce the pressure drop to 5 psi (0.35 bar), which is sufficient to size the valve (ignoring other pressures due to line losses or altitude), the power consumption would be down to 270 W with a corresponding savings in pump horsepower. However, it might mean a larger size valve because the required  $C_v$  would be three times higher. With a yearly energy savings (assuming 24-hour operation) of 21,286 kWh, it might be worthwhile.

While we favor valves having high  $F_L$  values for noise and cavitation concerns, there are cases where an ultralow  $F_L$  factor is highly desirable. Figure 2-1 shows how such a valve might look. Such valves are used for gas fuel control on large gas turbines in power plants due to their low energy loss (pressure loss).

As shown in Figure 2-1, the gas accelerates through the very streamlined inlet section of the housing, producing low pressure loss due to friction. The gas then enters the orifice passing by the streamlined plug to reach sonic velocity. The gas continues to expand through the tapered outlet section to reach supersonic velocities, followed by a process called *isentropic recompression*, where the gas is compressed to about 90% of its



**Figure 2-1.** Sketch of a streamlined angle valve having a high degree of pressure recovery (low head loss).



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# 3

## Basic Functions of a Control Valve

A control valve is an important part of a control loop, whether for controlling pressure in a tank, the temperature in a vessel performing a chemical reaction, or the level in a storage tank. Some of these functions are critical, such as guarding against an exothermic reaction or shutting down a turbine within a split second to avoid a runaway turbine. As these facts indicate, control valves must often be modified to meet specific requirements.

The most important criteria for having a stable control system are very little dead time, good frequency response, and a constant gain.

Dead time is hard to obtain. It is caused first by the time lag between the changes in a control system bringing air pressure into the actuator to move the plug or vane in the valve. Second, there is friction within the valve caused primarily by packings around the valve stem to avoid leakage. This problem has become worse since environmental considerations have resulted in tighter packing. The issue is that it takes an extra force of the actuator to move the stem and therefore more dead time. Even modern intelligent positioners are not much help. Quite often the gain, or the position sensitivity of the positioner, must be degraded to avoid instability of the positioner–valve actuator system. A larger actuator will help to overcome high stem friction. On the minus side, this adds to the time because a larger air volume must be changed. In this situation, amplifying air relays might help.

Rotary control valves offer an advantage because friction in rotary packing is less than in reciprocating types. However, one must make sure that there are no loose

connections between the valve stem and the rotary actuator. Such loose play can add substantial dead time.

The frequency response of the positioner–actuator system is dictated by the timing requirements of the process to be controlled. A good rule is this: the time constant of the controlling element should be either less than one-half or more than twice that of the process (including that of the measuring elements or transducers) to prevent the control loop from “hunting” (unstable cycling). Most modern valve positioners have means to adjust their frequency response to help in such situations.

The combination positioner and actuator form their own control loop independently from the process loop. The set point is the controller signal. The feedback is the valve’s stem position. This loop within a loop can easily become unstable, caused perhaps by too much stem friction. A remedy is to reduce the gain of the positioner (shown in Figure 3-1) between 20 and 67, meaning you get full output pressure to the actuator between 5% of signal change to 1.5%. The typical procedure is to reduce the speed (change in actuator volume per second) whenever the sensitivity (gain) is increased.

Finally, the *gain of the valve* itself (separate from that of the positioner) is a function of the rate of change of the amount of fluid passing the valve per given amount of controller signal change.

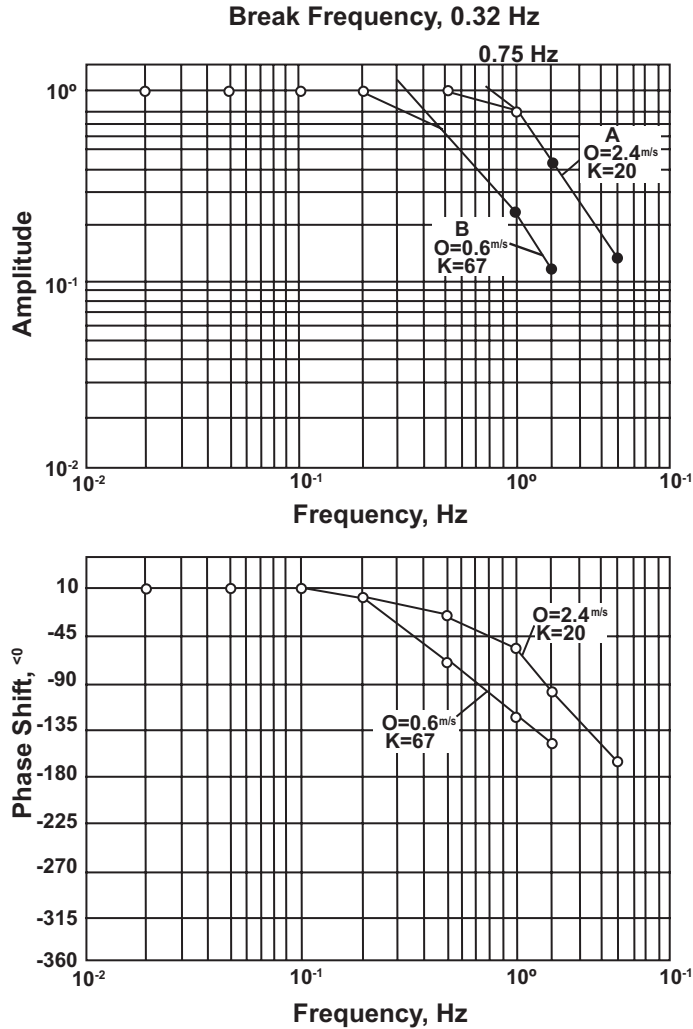
One must distinguish between *inherent gain*, which is when there are no outside fluid restrictions (such as on the bench), and the *installed gain*, which is when the valve is installed in a piping system. The installed gain is the only important one affecting loop stability.

The reason is that any fluid passing through pipes will undergo pressure reduction due to pipe friction, where such change in pressure is a function of the square of the amount of flow change. This results in a nonlinear change in the inlet pressure of the valve and, therefore, the flow rate as a function of valve opening.

The inherent characteristic of a valve can be modified either by installing a special cam in the feedback mechanism of a positioner or, as shown in Figure 3-2, by electronically modifying the control signal from the positioner to the valve.

As shown in Figure 3-2, the upper curve is the installed characteristic of a 6-inch (0.150 m) butterfly valve in a pump and piping system, causing the inlet pressure to drop from 74 to 6 psi (5 to 0.4 bar) or by 6.6 to 1, resulting in an unacceptable gain





**Figure 3-1.** Typical range of dynamic performance of a modern valve positioner with separate gain and speed adjustments.

Source: SAMSON AG.

change from 300 gpm per 10% signal to 45 gpm per 10% signal. After the intelligent positioner's output signal was modified, the gain was reduced to fall within an acceptable range of  $\pm 2$  (see dotted lines).

To overcome this nonlinearity, a special profile of a valve trim has been devised. It is called *equal percentage* inherent characteristic. This means that for each percentage of valve travel, there also is a constant percentage of increase in  $C_v$  or  $K_v$  (metric flow coefficient) of the trim (plug, vane, or ball). For example, in a specific valve plug, the profile is such that for each 10% of travel increase, there is a 15% increase in  $C_v$  or  $K_v$ .

## 4

# Thermodynamic Effects

All throttling actions in a valve follow Bernoulli's law, which states that whenever the velocity of a fluid increases there is a corresponding decrease in pressure. However, temperature change is also involved, depending on the thermodynamic characteristics of the fluid. Energy changes associated with pressure drop through the valve, caused by deceleration of the jet velocity, cause turbulence or cavitation in liquids and turbulence or shock waves for gases. Some of the converted energy also changes the downstream temperature and creates sound.

High-pressure reduction in gases can cause drastic reduction in temperature at the vena contracta resulting in the freezing (*hydrate formation*) of entrained liquids in a process called *auto-refrigeration*. Moist air, for example, can cause ice formation at the trim, which can produce a drastic reduction in the flow capacity of a valve.

In most liquids, there is a slight increase in downstream temperature. Valves handling steam can change superheated steam to saturated steam at a lower temperature following pressure reduction.

## Auto-Refrigeration

We are all familiar with air conditioners. In these devices, a gas such as Freon is compressed at a high pressure and then expanded, causing subfreezing temperatures in a heat exchanger which then cools the air. Not all gases have reduced outlet temperature. Some even have a slightly increased temperature. However, these are exceptions.

# 5

## Valve-Sizing

### In the Beginning...

During the start of a project, or for purposes of cost estimates, the necessary data required to size or select a control valve are usually missing. Therefore, one must make estimates. However, the data concerning the amount of fluid, liquid, or gas are usually available. Consulting Table 5-1 or Table 5-2 gives an idea of the probable pipe size, because line velocities seldom exceed 10 ft/s (3 m/s) for liquids or 150 ft/s (46 m/s) for gaseous fluids.

To obtain flow in kg/h, divide numbers by 2.2. For pressures in bar, divide psi by 14.5.

#### Example:

Need to control 12,000 m<sup>3</sup>/h natural gas,  $G_f = 0.6$ , and at a pressure of 30 psia (2 bara). To convert gas to airflow, divide 12,000 by 0.6 = 20,000 equivalent airflow. Table 5-2 shows that the closest pipe size is 12 in (0.300 m), having a capacity of 22,788 sm<sup>3</sup>/h.

The first approach is to assume that the valve size is equal to the selected pipe diameter. This is a safe assumption; one can always specify a reduced trim, if the valve turned out to be too large.

Later, the project engineer wants to know how much pressure drop to assign for the control valve. The project engineer needs the information (together with the addition

**Table 5-1.** Flow rates in US gpm based on schedule 40 pipe and 10 ft/s fluid velocity (3 m/s).

Pipe diam.	1"	1.75"	2"	3"	4"	6"	8"	10"	12"
gpm	27	62	100	210	380	900	1600	2300	3500
m <sup>3</sup> /h	6.2	14	23	48	87	207	367	528	804

**Table 5-2.** Flow rates in lb/h of saturated steam at a fluid velocity of approximately 150 ft/s in a schedule 40 pipe.

Pressure, psi	Pipe Diameters (Flow in lb/h)								
	1"	1½"	2"	3"	4"	6"	8"	10"	12"
15	200	500	800	1800	3200	5400	12,800	20,000	24,600
30	300	710	1200	2400	4800	9800	19,200	30,000	36,000
60	500	1200	2000	4500	8000	18,000	32,000	50,000	60,000
100	750	1800	3000	6800	12,000	27,200	48,000	75,000	90,000
150	1100	2500	4400	9900	18,000	40,000	72,000	111,000	132,000
200	1500	3300	6000	13,500	24,000	54,000	96,000	150,000	180,000
300	2200	4700	8800	20,000	32,000	80,000	128,000	220,000	265,000
400	2500	6100	10,000	22,500	40,000	90,000	160,000	250,000	300,000
500	3300	7800	13,000	30,000	52,000	120,000	210,000	330,000	400,000

of the expected pipe friction losses) to determine the required pump head (if the fluid is a liquid). Here, the proper answer would be 5% of the expected valve inlet pressure.

This is a safe bet because any piping engineer will always overestimate pipe losses to play it safe. This means that the available valve differential pressure will always be higher than 5%.

Once you have all the pertinent data (such as type of fluid, inlet pressure, temperature, and specific gravity), you can start the actual valve-sizing.

There are two ways of going about this. First, we will discuss the easy way, which you can do on the back of an envelope with the aid of a calculator, risking an error not to exceed 5% to 8%. This should be sufficient for estimating purposes and to get a quote from a vendor. Otherwise, you can always use available computer programs.

Keep in mind that the given pressure conditions are never accurate and that catalog  $C_v$  (or  $K_v$ ) values published by vendors allow a tolerance of  $\pm 10\%$ , due to casting

## 6

# Sound Produced by Throttling

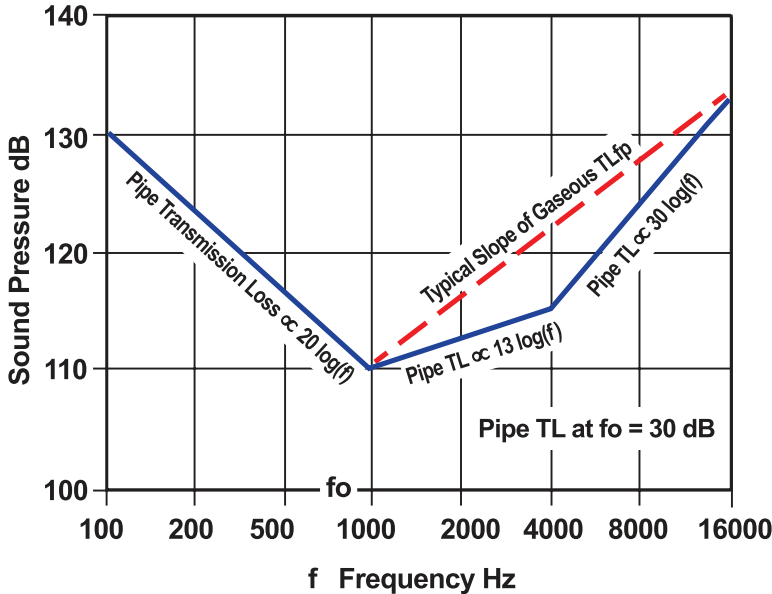
All sound level equations are based on the same basic variables: the mass flow and pressure-derived *mechanical power*; multiplied by an *acoustic efficiency factor* minus the *transmission loss* of a pipe (TL); minus a *correction factor for the distance* between the pipe wall and the observer.

Sound is a by-product of the conversion of kinetic energy (mass flow and velocity) into sound. In all cases, it is the product of energy converted to mechanical power multiplied by an acoustic efficiency factor. Fortunately, most of the acoustic energy is absorbed by the pipe wall. In acoustic terms, this absorbed energy is expressed in decibels (typically 40–60 dB). Regardless, noise produced both by liquids as well as by gases can exceed 120 dBA in some cases (e.g., the sound of a jet engine). Most governments have imposed limitations on how much sound workers can be exposed to—typically 80 dBA for an 8-hour exposure.

Other detrimental factors are pipe damage due to associated mechanical vibration (which is typical if the external sound level exceeds 130 dBA) and damage to pipe-mounted instrumentation.

One of the most misunderstood factors in any sound level calculation is the TL of a pipe. The following graphical representation illustrates a typical example of how the TL is derived (see Figure 6-1).

Figure 6-1 illustrates a typical case with gases in which a given pipe has a coincidence frequency,  $f_o$ , of 1000 hertz (Hz). The transmission (the difference between a



**Figure 6-1.** Graphical presentation of transmission loss slopes.<sup>1</sup>

given frequency spectrum and the external sound pressure of the pipe at the same frequency at 1000 Hz) is assumed to be 30 dB. One may assume further that the peak frequency,  $f_p$ , in which the internal sound pressure is at its maximum is 10,000 Hz. Here, the internal sound pressure,  $L_{pi}$ , is 130 dB. At this frequency, the pipe wall would absorb 55 dB, resulting in 75 dB externally. However, this is not what one would typically hear because most sound escapes from the coincidence frequency. In addition, the human ear is limited to perceiving sound at frequencies above 6000 Hz. As a result, the internal sound pressure gradually decreases (see the red line) until it reaches  $f_o$ . At this point, the  $L_{pi}$  is down to 110 dB [ $130 - 20 \log(10,000/1000)$ ], which is unfortunate because the TL at  $f_o$  is only 30 dB. This leaves the sound pressure at the outer pipe wall remaining at 80 dB (i.e., at a human-sensitive frequency of 1000 Hz). The frequency shift saved the observer 20 dB because 130 dB at 1000 Hz would have resulted in an external sound level of 100 dB.

$L_{pi}$  decreases as a result of a gradual loss of density of the sound waves of the passing gas, proportional to  $L_{pi} \cdot (f)^2$ , where  $f$  is the given pipe frequency. Therefore,  $L_{pi} \cong 20 \log(f_p/f)$ .

<sup>1</sup> The graph in figure 6-1 is meant as a schema only. The energy conversion from constant pressure at high frequencies to low frequencies for liquids, and from high pressure at high frequencies to low pressure at low frequencies for gases, happens simultaneously in all sections of the pipe (at least close to the valve).

## 7

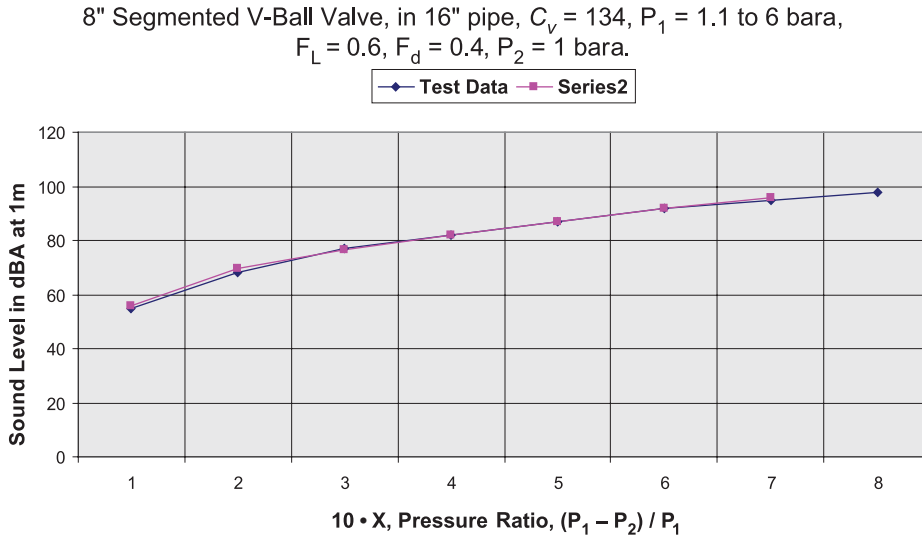
# Estimating the Sound Pressure Level of Gases

Chapter 6, “Sound Produced by Throttling,” claimed that cavitation is an aerodynamic phenomenon where the slope of the cavitating sound increases to the sixth power of static pressure; or  $L_{Ae}$  is proportional to  $60 \log(10X)$ , where  $X$  is the pressure ratio  $(P_1 - P_2)/P_1$ . It stands to reason that the same logic can be applied to the prediction of aerodynamic sound level, and indeed it works, although there is one modifier: there is a difference between the pressure drop inside the valve and the one outside. Inside the valve, the onset of sonic velocity happens at a lower pressure than outside. The reason for this is pressure recovery, the magnitude of which is expressed by the pressure recovery factor,  $F_L$ . To account for this phenomenon, the sixth power relationship must be modified; therefore, the slope of the aerodynamic sound level exterior of the pipe is determined by  $F_L \cdot 60 \cdot \log(10X)$ .

As shown later, this slope fits quite well, even for valves with expanded outlet pipes, as long as the outlet velocity is limited. Figure 7-1 shows test data from an 8-inch V-ball valve with an overlay of the  $F_L \cdot 60 \cdot \log(10X)$  trend line. In this case, the slope is proportional to  $36 \log(10X)$  because the  $F_L = 0.6$ . The two curves overlap quite well. This is a blowdown test where the downstream pressure stays at 1 bara. The variable is the inlet pressure.

The test data in this figure show good agreement with the predicted slope based on  $60 \cdot F_L \log(10X)$ .

To take advantage of the slopes, one needs a *base level*. This proposal uses as a base level a condition where the outlet pressure is exactly one-half of the inlet pressure.



**Figure 7-1.** Sound pressure level during a blow-down test.

Why this ratio? Because at this pressure ratio there is always sonic velocity in the orifice (vena contracta) regardless of the given  $F_L$  factor. The sonic velocity and the density are simply a function of  $P_1$  only. One therefore can dispense with many other equations. The sound level calculation is then very simple, as the following equations demonstrate. Once a base level for  $0.5 P_1$  is established, the sound at other pressure ratios can be established by adding or subtracting  $F_L \cdot 60 \cdot \log(10X)/0.5$ . All one has to do is add the correct downstream pressure (if  $P_1$  is constant) or the variable inlet pressure (if  $P_2$  is constant), and the equations will take care of the rest. First, and based on  $P_1$ , the basic sound level is calculated using  $0.5 P_1$  as the outlet pressure in Equation 7-1. Thereafter, the real  $P_2$  is employed to calculate the sound for the actual  $P_2$  using Equation 7-4.

As in any method, the following input data is required:

- Inlet pressure,  $P_1$ , in bara
- Outlet pressure,  $P_2$ , in bara
- Required flow coefficient,  $C_v$  ( $1.17 K_v$ )
- Pressure recovery factor,  $F_L$
- Valve style modifier,  $F_d$
- Temperature (t) at the valve inlet, in °C



This is an excerpt from the book. Pages are omitted.



## 8

# Ways to Reduce Aerodynamic Noise

Fortunately, there are several ways to reduce aerodynamic noise. First, one should determine if there is a noise problem. If the product of  $C_v \cdot F_L \cdot P_1$  is less than 1000, then the noise is less than 90 dBA. (Here,  $P_1$  is psia; for bar, divide psia by 14.4.)

Quick rules of thumb for noise reduction are as follows:

- Using 1-inch (0.025 m) pipe insulation = 5–10 dB reduction
- Doubling the pipe wall = 6 dB reduction
- Placing the silencer downstream = 10 dB reduction
- Placing the silencer upstream and downstream = 20 dB reduction

More sophisticated methods are discussed in the sections below.

## The Classical Way

This method takes advantage of the sound-absorbing characteristic of pipe. Most of the sound one hears outside the pipe comes through an area where the vibrations inside the pipe match the resonant vibration of the pipe wall. This matching of frequencies is ideal for sound transmission. The frequency where this occurs is called the *coincidence frequency*,  $f_o$  (see Chapter 7, page 81).

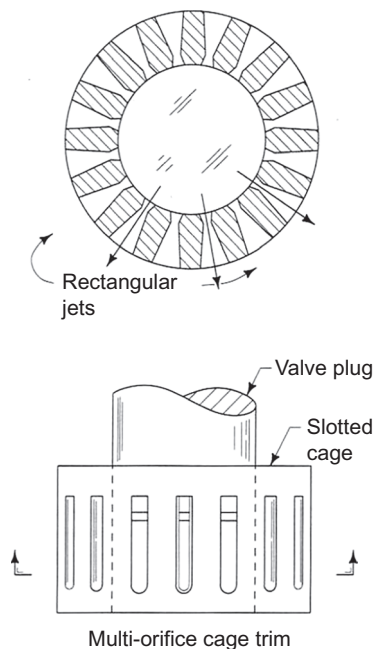
Further, sound pressure inside the pipe, defined as  $L_{pi}$ , decays along the pipe wall both toward the valve and further down the pipe. The decay down the pipe is not

relevant, but the one toward the valve is. This decay happens at a rate of 20 dB per decade, or 6 dB per octave (doubling frequencies). This means that the bigger the differences between the peak frequency produced by the trim of the valve and  $f_o$ , the more sound pressure is coming through the pipe. For example, the peak frequency inside the pipe is 2200 Hz (cps), and the  $L_{pi}$  is 130 dB.

Assuming further that the “hole” in the pipe,  $f_o$ , occurs at 800 Hz, there is a reduction of external sound of  $20 \log(2200/800) = 8.8$  dB, which is quite significant. To this, of course, one must add the pipe’s transmission loss, TL, which in this case may be 40 dB. This makes the total sound pressure level outside the pipe:  $L_{Ae} = L_{pi} - \Delta TL_f - TL = 130 - 8.8 - 40 = 81.2$  dB. To this one must correct for the distance from the pipe wall to the observer (typically 1 m).

The next step is to find ways to increase the peak frequency. The traditional way is to use multihole trims; that is, divide the flow area to produce a multitude of small jets where the peak frequency is proportional to the diameter of the vena contracta ( $D_j$ ) of each hole.

Jet engines use the same principle, dividing the exhaust by forcing the gas through corrugated openings called *nacelles*. This increases the sound frequency, which is better able to excite the air molecules and thereby convert some of the sound energy into heat. Figure 8-1 shows a typical low-noise trim for a globe valve.



**Figure 8-1.** A slotted cage trim for globe valves.



# 9

## What to Do about Hydro-Noise?

First, one must reverse the usual tactic. What is good for gases, such as a low  $F_L$ , is bad for liquids. A higher peak frequency, while desired for gases, is bad for liquid fluids. The reason is the change in transmission loss. Liquid noise frequencies typically occur below 500 Hz (cps) due to the almost 1000 to 1 density change between liquid and gas. This puts the liquid sound frequencies below the pipe's coincidence frequency instead of above it, which is the case for gases.

To avoid cavitation, trims with high  $F_L$  (low pressure recovery trims) are needed. They will retard the onset of cavitation (see Chapter 6 on estimating the  $X_{Fz}$  factor).

For example, for liquid noise applications, a V-port plug is better than a parabolic one.

Multihole plates downstream of valves are effective because they split the pressure drop between the valve and the plate. They do not do much to change the peak frequency because that is determined by the diameter of the downstream pipe. However, when the plates have slotted openings, one must make sure that the largest width of the tapered slot faces upstream. This increases the  $F_L$  of each slot and thereby delays or reduces the cavitation sound level.

One of the most effective ways to reduce turbulent noise is to reduce the throttling velocity. A velocity reduction of 50% decreases the sound level by 7.5 dB. The penalty is that one has to double the flow capacity of a valve or valves to compensate for the reduction in mass flow due to velocity reduction. A multistage, low-noise valve trim is

## 10

# What to Expect of a Good Control Valve

Besides the obvious, such as low price and speedy delivery, when choosing a control valve, one should also look for a standard face-to-face dimension, so it can easily be replaced with a competitor's product if things go wrong. As for functionality, the valve should have a low deadband (basically low friction); good rangeability, because most valves are oversized; and dynamic stability, which means *flow-to-open* for single parabolic or pressure-balanced plugs, and for cage-style globe valves. For rotary valves, such as standard vane butterfly valves and some eccentric rotary plug valves, one should make sure the degree of opening at the maximum  $C_v$  is less than 60 degrees to avoid dynamic instability.

One must beware of valves that have significantly larger flow capacities ( $C_v$ ) than similar competitive products. Such valves have more pressure recovery (lower  $F_L$  numbers) and may need reducers to meet the pipe velocity limitations of the downstream pipe with their added cost, space requirements, and addition of leak-prone flange gaskets. Such reducers do not eliminate the possibility of the valve outlet port velocity exceeding limited Mach numbers.

When specifying positioners, one should ensure a product offers sufficient air capacity (important for larger valves) and low air consumption. Finally, at a minimum, positioners should have speed and gain adjustment. This is important when valve manufacturers are obliged to use environmentally acceptable stem packings that are

# 11

## Fail-Safe Action

One of the many features of spring-diaphragm actuators is that a built-in compression spring can drive a valve to close or, if otherwise desired, to open on failure of the air supply or on signal failure from the controlling instrument. It is up to the instrument engineer to decide what *fail-safe action* a control valve should perform. The criteria should always be the safety of the operating loop. Table 11-1 contains examples.

As a rule, piston-type actuators do not have fail-safe springs. In this case, a standby compressed air cylinder is used that, when controlled by a solenoid valve, will drive the piston in a desired fail-safe position. Hydraulic actuators also need a pressurized auxiliary tank for this purpose.

Electric actuators typically have a battery-powered, source-driven supplementary direct current (DC) motor to move the valve into a safe position. Some models have

**Table 11-1.** Valve handling—action required.

Cooling water to prevent an exothermic reaction in a vessel	Fail-open
Controlling gas to a furnace	Fail-close
Level control in a tank	Fail-close
Steam tracing a fuel oil pipeline	Fail-open
Hot water in a temperature control system	Fail-close
Lubricating oil to a bearing	Fail-open
Bypass valve at a compressor	Fail-open

## 12

# Valves for Sanitary or Aseptic Service

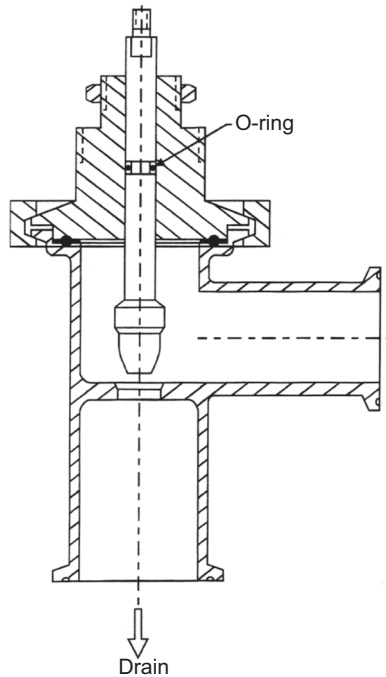
Sanitary valves have been around as long as dairy products have been processed automatically. They are highly polished body subassemblies that can be taken apart rapidly and washed after each use.

Control valves in the bioprocessing industry are another matter. While the former only need to be clean, *bioprocessing valves* are not disassembled but cleaned in place (CIP) and sterilized in place (SIP). Often, such valves are welded to the pipe using orbital welding techniques.

In contrast to automated ON–OFF valves, the use of modulating control valves is increasing, especially in bioprocessing systems that are upgraded from small batch systems to larger production systems. Applications range from pressure reduction of steam used for sterilizing to flow control of high-purity water, in quantities from 0.001 to 500 L/min. Design considerations for such valves need to meet CIP and SIP requirements.

Modulating control valves must have good flow characteristic, high rangeability, and tight shutoff, and they must be cavitation-resistant and self-draining. They also must have low operating friction to ensure stable control.

An early valve type is illustrated in Figure 12-1. This valve meets all requirements of sanitary design. It is self-draining, can be polished to the required finish, usually has a surface roughness (Ra) of 0.15 to 0.20  $\mu\text{m}$ , and can have sufficient seal integrity at the interface between the body and bonnet guiding the stem. However, one difficulty



**Figure 12-1.** Conventional sanitary globe valve. This valve is used in the dairy industry. The modulated construction allows quick disassembly for cleaning.

in making this design acceptable for bioprocessing applications is the sliding O-ring. A continuous cycling of the stem during the sterilizing process is perhaps the only way to accomplish this task, which requires an auxiliary timing control system. The valve plug provides an acceptable flow characteristic. However, the flow-to-close fluid flow direction encounters pressure recovery (low  $F_L$ ), which encourages cavitation and fluid-pressure-change-induced “slamming” against the seat.

Another time-proven design is shown in Figure 12-2. This is a diaphragm valve, which can also be used for nonsanitary applications such as slurries. This valve has good shutoff capabilities, but a major drawback is the poor flow characteristic. Improvements, such as a two-step actuating mechanism, are available from some manufacturers.

Another drawback is the elastomeric diaphragm that will fail in other than very low-pressure steam sterilizing applications. Citric acid might have to be used as a sterilizing medium. To alleviate this problem, coating the rubber with a Teflon layer has helped. This valve also has high pressure recovery due to its streamlined design.

The valve design shown in Figure 12-3 has a number of improvements over the diaphragm valve in Figure 12-2. Here the flow path is more abrupt, resulting in a reduced



# 13

## Inspection and Testing

Control valves are pressure vessels and therefore must abide by the same safety standards and design rules as other components of a piping system.

For example, the minimum wall thickness of a valve housing casting must conform to ASME B16, *Fittings and Valves Package*,<sup>1</sup> and a bonnet flange must meet the same stress requirements as a pipeline flange.

### Materials

All castings should be certified by the foundry as to their chemical composition. Body castings for later use in critical applications, such as nuclear power plants, must be x-rayed for inclusions or cracks prior to machining. Stainless-steel investment castings should be solution annealed to eliminate hard chrome–carbide formation on their surfaces.

Bar stock materials used for critical stem and plugs should be magnetic particle–inspected to look for micro-cracks. All bonnet bolting material should meet ASTM A193/A193M-17, *Standard Specification for Alloy-Steel and Stainless Steel Bolting for High Temperature or High Pressure Service and Other Special Purpose Applications*.<sup>2</sup>

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<sup>1</sup> ASME B16, *Fittings and Valves Package* (New York: ASME [American Society of Mechanical Engineers]).

<sup>2</sup> ASTM A193/A193M-17, *Standard Specification for Alloy-Steel and Stainless Steel Bolting for High Temperature or High Pressure Service and Other Special Purpose Applications* (West Conshohocken, PA: ASTM International).





## 14

# Cybersecurity

It is likely that a plant will be the victim of a cyber attack resulting in a shutdown of its controls system and perhaps even its electrical power supply. It is critical to anticipate such an occurrence. In some plants, the operator in the control room has no idea about the state of the plant operations. As far as control valves are concerned, one must ensure the correct fail-safe action. Consider that there may be no electric power and therefore no compressed air. On the other hand, there still might be steam accumulating in a boiler or hot oil flowing through a pipe. A chemical reactor might have an exothermic reaction (i.e., an explosion) if the cooling water supply fails.

The resultant damage might range from loss of life (e.g., in atomic power plants) to merely annoyance. Cyber attacks that cause the most financial damage are those on electric power stations affecting tens of thousands of households, hundreds of businesses and factories, or a city's transportation system. Thus, power stations are most vulnerable to cyber attack. A recent example is the attack on two Ukrainian electric power plants in 2015. The attackers had it very easy because, as shown in the plant's wiring diagram, the Internet cable was directly connected to the plant's control computers. They gained access despite a firewall between the Internet connection and the utility control system.

*In the author's opinion, it is almost criminal negligence to have an Internet cable connect to any control system, even one behind a firewall.*

For that matter, personal or laptop computers should never have access to a control system either. A case in point is the Bushehr Nuclear Power Plant in Iran that almost

melted down because an agent planted a memory stick containing a Stuxnet virus into an operator's computer, and the virus then found its way into the plant's control system.

Remember that there are other ways to submit operating data to a company's headquarters, such as by phone or fax. A company should consider renting or installing a private communication cable or creating a separate intranet system. The Chinese army is a good example; years ago, it created its own intranet system, separate from the public Internet.

*Remember: Encrypted messages can be broken and firewalls can be bypassed. None of these can guarantee absolute safety and reliability of a control system.*

## 15

# Control Valves as a Source of Loop Instability

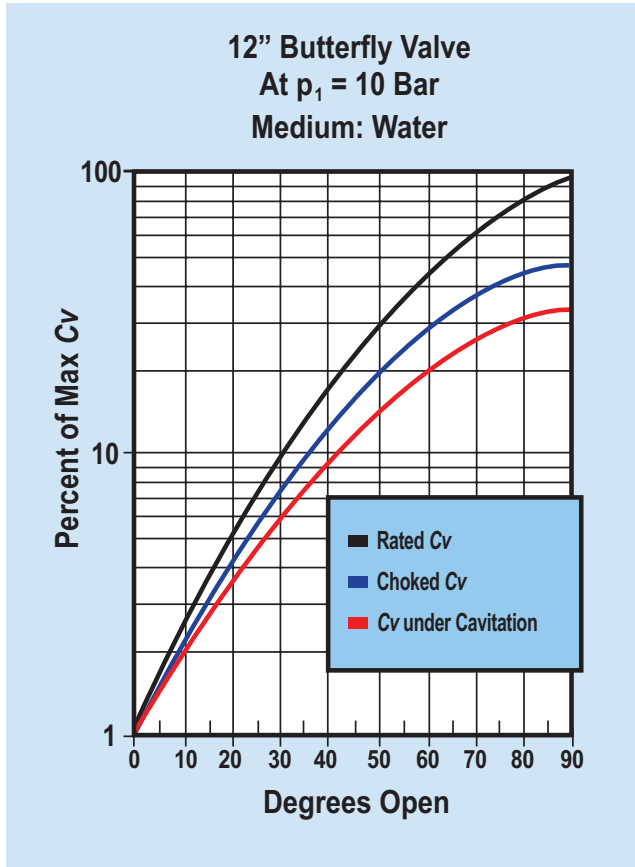
Instability, easily observed by watching widely swinging chart indicators, can have many sources (e.g., changes in process conditions, pump problems, or variations in hydraulic head) but is often caused by misbehaving control valves.

Sometimes problems are caused by seemingly benign actions, such as a maintenance person tightening a stem packing box too tightly while trying to eliminate a leaking valve packing. The sudden increase in friction creates *deadband*, which not only makes the valve position unstable but in turn creates loop instability.

Other typical causes of deadband are loose play between a pin attached to a valve stem and riding in a slot of the valve positioner's feedback arm. In this situation, a 0.030 in (1 mm) clearance can cause 3% deadband on a 1-inch (25 mm) valve travel. Similar problems occur with rotary valves that typically connect the valve stem through a flattened tang to a similar opening in a rotary actuator. A 0.15-inch clearance between the tang and the slot on a 0.75-inch (20 mm) diameter stem can cause an angular error of 1.5 degrees, a deadband of 2% at 60-degree valve travel.

Another mechanical problem is friction caused by galling of the plug guides or stem that, in an extreme case, can freeze the valve motions. Modern digital valve positioners can sense such irregular motion and alert the instrument engineer.

Thermal expansion of valve parts made of different materials can cause problems. This primarily involves globe valves with cage guided trim. The cage might be made of stainless steel while the housing is cast carbon steel. Stainless steel has a thermal



**Figure 15-1.** Various flow characteristics of a typical butterfly valve. Note the slope changes between choked flow conditions [pressure drop exceeds  $F_L^2 \cdot (p_1 - p_v)$ ] and nonchoked conditions. Cavitation typically occurs when the pressure drop exceeds  $X_{Fz} \cdot (p_1 - p_v)$ . However, cavitation does not restrict flow. The red line indicates where cavitation is likely to occur.

expansion factor that is about 1.5 times higher than that of carbon steel. In such cases, the added expansion under high temperature might crush the cage gaskets causing internal as well as external leakage problems, which might also affect the pressure balance capabilities of the plug.

In addition, fluidic problems can alter the installed characteristic of the valve and therefore its open-loop gain or time constant. Figure 15-2, for example, depicts three characteristics of a 12-inch butterfly valve showing a standard inherent flow characteristic without fluidic effects (solid and dashed blue lines). The red line shows the characteristic-altered effects when pressure conditions cause choked flow, which calls for more valve travel to meet a desired flow rate. These effects can create instability and may require adjustments at the process controller, such as altering the proportional



# Appendix A

## Reference Tables

**Table A-1.** Temperature ratings for lined valves.

*Temperature Limitations °F (°C)*

Elastomer	Low Limit	High Limit	Service
Natural rubber	−20 (−30)	160 (70)	Air, water
Butyl	−10 (−24)	800 (150)	Acids, alkalis, steam, alcohols
Silicone	−75 (−60)	200 (90)	High-temperature air
Neoprene	0 (−18)	175 (80)	Air, water, Freon 12, 114, 22
Hypalon	+15 (−9)	225 (110)	Acids, alkalis
EPT Rubber (Nordel)	−30 (−34)	250 (120)	Acids, alkalis
Polyurethane, Adiprene	+10 (−12)	200 (90)	Air, water to 150°F
Fluorocarbon (Vitron, Fluorel)	+20 (−70)	400 (200)	Aromatic, aliphatic hydrocarbons, degreaser, high temperature air and gases
Nitrile (Buna N, Hycar)	+10 (−12)	180 (80)	Freon 11, 12, 13, air, water, alcohols, aromatic hydrocarbons
White Hycar	+20 (−7)	200 (90)	Foodstuffs
White Neoprene	−20 (−30)	175 (80)	Foodstuffs
White Butyl	−10 (−24)	300 (150)	Foodstuffs

**Table A-2.** Common materials for valve bodies.

<b>Metals ASTM</b>	<b>Designations</b>
<b>Bronzes and Brasses</b>	
High Tensile Steam Bronze	B 61
Steam Bronze	B 62
<b>Irons</b>	
Cast Iron	A 126 Class A
Cast Iron	A 126 Class B
Ductile Cast Iron	A 395
Malleable Cast Iron	A 47 Grade 32510
Ni-resist Gray Cast Iron	A 436 Type 2
<b>Steels</b>	
Carbon Steel, Forged	A 105
Carbon Steel, Cast	A 216 Grade WCB
0.15% Moly. Steel, Cast	A 217 Grade WCI
Cr. Moly. Steel, Cast	A 217 Grade WC6
Cr. Moly. Steel, Cast	A 217 Grade WC9
4.6% Cr. Moly. Steel	A 217 Grade C5
8–10% Cr. Moly. Steel	A 217 Grade C12
Carbon Steel, Cast	A 352 Grade LCB
Carbon Moly. Steel, Cast	A 352 Grade LCI
3.5% Nickel Steel, Cast	A 352 Grade LC3
<b>Stainless Steels</b>	
18 Cr. 8 Ni., Cast	A 351 Grade CF8 (Type 304)
Type 304 Bar	A 182 Type F304
Type 303 Bar	A 182 Type F303
18 Cr. 8 Ni. Mo., Cast	A 351 Grade CF8M (Type 316)
Type 316 Bar	A 182 Type F316
18 Cr. 8 Ni. Cb., Cast	A 351 Grade CF8C (Type 347)
18 Cr. 8 Ni. Cb., Bar	A 182 Type F347
18 Cr. 8 Ni. Mo., Forging	A 182 Type F316L
16 Cr. 12 Ni. 2 Mo., Bar	A 479 Type 316L
<b>Nickel Alloys</b>	
Nickel Cast	A 296 CZ-I 00
Nickel Wrought	B 160
Monel Cast	A 296 M-35W
Monel Wrought	B 164 Class A

**Table A-2.** (continued)

Common materials for valve bodies.

<b>Metals ASTM</b>	<b>Designations</b>
Hastelloy “B” Cast	A 296 N-12M
Hastelloy “C” Cast	A 296m CW-12M
Aluminum	
No. 356 T6 Cast	B 26 Grade SG70A T6
Titanium	
Alloy 20, Cast	A 296 Grade CN-7M
Cast	B 367
Wrought	B 381

**Table A-3.** Common trim materials.

<b>Metals</b>	<b>ASTM Designations</b>
Brass Rod	B 16
Naval Brass	B 21 Alloy A
Bronze Rod	B 140 Alloy B
Phosphor Bronze	B 134 Alloy B-2
Type 316 St. Steel Rod	A 479 Type 316
Type 316L St. Steel Rod	A 479 Type 316L
Type 316 St. Steel, Cast	A 351, Grade CF8M
Type 316 St. Steel, Forged	A 182, F316
Type 304 St. Steel Rod	A 182, F304
Type 347 St. Steel Rod	A 182, F347
Type 17-4 St. Steel Rod	A 564
Inconel Rod	B 166
Monel Cast	A 296 Grade M-35

**Table A-4.** Common bolting material.

<b>ASTM Designations</b>		
<b>Metals</b>	<b>Bolts</b>	<b>Nuts</b>
Type 304 St. Steel	A 193 Grade B8, Cl. 1	A 194 Grade 8
Type 347 St. Steel	A 320 Grade B8	A 194 Grade 8
Type 316 St. Steel	A 320 Grade B8	A 194 Grade 8
Carbon Steel	A 193 Grade B7	A 194 Grade 2H
Carbon Moly. Steel	A 193 Grade B7	A 194 Grade 2H
5% Cr. 1/2% Mo, Alloy	A 193 Grade B16	A 194 Grade 4



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