A Rational Approach to Control Valve Sizing

The sizing of control valves usually involves a trade-off between pump head and control valve size: the bigger the valve, the smaller the pressure drop across the valve and the lower the pump head. Thus, large valves are desirable from a steady-state energy-consumption viewpoint. However, dynamic rangeability is improved by designing the system for small, high pressure drop control valves. Control valves are often sized on the basis of heuristics, the most common one being to make the pressure drop over the control valve at design conditions one-third of the total system pressure drop. In some applications, this rule-of-thumb works well, but in others it is totally inadequate. A more rational procedure is presented in this paper. The idea is to specify the maximum and minimum flow rates required. Then the sizes of both the pump and the control valve can be directly calculated.

In a majority of chemical engineering processes, the final control element is an automatic control valve which throttles the flow of a manipulated variable. Sizing of control valves is one of the more controversial subjects in process control. The flow rate through a control valve depends on the size of the valve, the pressure drop over the valve, the stem position, and the fluid properties. The design equation for liquids (nonflashing) is

$$F = C_v f_{(x)} \left(\frac{\Delta P_v}{\text{sp.gr.}} \right)^{1/2} \tag{1}$$

where F is the flow rate in gpm, C_v is the valve size coefficient, x is the valve stem position (fraction of wide open), $f_{(x)}$ is the fraction of total flow area (from valve flow characteristic curve), sp.gr. is the specific gravity (relative to water), and ΔP_v is the pressure drop over the valve in psi. More detailed equations are available in publications of the control valve manufacturers (for example, the *Masonielan Handbook for Control Valve Sizing* (1977)) that handle flows of gases, flashing liquids, and critical flows with either English or SI units.

The sizing of control valves is a good example of one of the many engineering trade-offs that must be made in designing a plant. Connell (1987) gives a good description of the problem. Consider the process sketched in Figure 1. The engineer's job is to size both the centrifugal pump and the control valve. The bigger the control valve, the less pressure drop it will take. This means a lower head pump can be used and energy costs will be lower because the power consumed by the motor driving the pump will be less. So from a steady-state energy viewpoint, the pressure drop over the control valve at design conditions should be low and the valve should be big.

However, from a dynamic control point of view, the pressure drop over the valve should be large and the valve should be small, as will be illustrated below. This type of design gives a system with more rangeability; i.e., the magnitudes of both the positive and negative changes that can be made in the flow rates are increased.

Example

These effects are most easily seen in a specific example. Suppose the flow rate at design conditions is 100 gpm, the pressure in the feed tank is atmospheric, the pressure drop over the heat exchanger (ΔP_H) at the design flow rate is 40 psi, and the pressure in the final tank is 150 psig. Let us assume that we will have the control valve half open ($f_{(x)} = 0.5$) at the design flow and that the specific gravity of the liquid is 1.

Two different designs will be examined to show why it is desirable from a dynamic point of view to take more pressure drop over the control valve. In case 1 we will size the valve so that it takes a 20 psi pressure drop at design flow when it is half open. This means that the pump must produce a differential head of 150 + 40 + 20 = 210 psi at design. In case 2 we will size the valve so that it takes a 80 psi pressure drop at design. Now a higher head pump will be needed: 150 + 40 + 80 = 270 psi.

By use of eq 1, both control valves can be sized.

Case 1: (design valve ΔP = 20 psi)

$$100 = C_{v1}(0.5)(20)^{1/2} \Rightarrow C_{v1} = 44.72$$

Case 2: (design valve ΔP = 80 psi)

$$100 = C_{v2}(0.5)(80)^{1/2} \Rightarrow C_{v2} = 22.36$$

Naturally the control valve in case 2 is smaller than that in case 1.

A. Maximum Capacity. Now let us see what happens in the two cases when we open the control valve all the way: $f_{(x)} = 1$. Certainly the flow rate will increase, but how much? From a control point of view, we may want to be able to increase the flow substantially. This unknown maximum flow rate is F_{max}.

As the flow rate is increased through the system, the pressure drop over the heat exchanger increases as the square of the flow rate:

$$\Delta P_H = 40(F_{max}/F_{des})^2 = 40(F_{max}/100)^2 \tag{2}$$

The higher flow rate might also reduce the head that the centrifugal pump produces if we are out on the pump curve where head is dropping significantly with throughput. For simplicity, let us assume for the moment that the pump curve is flat. This means that the total pressure drop (ΔP_T) across the heat exchanger and the control valve is constant. Therefore, the pressure drop over the control valve must decrease as the pressure drop over the heat exchanger increases:

$$\Delta P_v = \Delta P_T - \Delta P_H \tag{3}$$

Substituting in the appropriate numbers for the two cases yields the following results.

Case 1: (20 psi design with C_{v1} = 44.72)

$$\Delta P_T = \Delta P_v^{des} + \Delta P_H^{des} = 20 + 40 = 60 \text{ psi}$$

$$F_{max,1} = (44.72)(1.0)[60 - 40(F_{max,1}/100)^2]^{1/2} \tag{4}$$

Solving for $F_{max,1}$ gives 115 gpm. So the maximum flow through the valve is only 15% more than the design flow rate if a 20 psi pressure drop over the valve is used at design flow rate. The valve pressure drop at $F_{max,1}$ is

$$\Delta P_{v,1}^{max} = 60 - 40(115/100)^2 = 7.1 \text{ psi}$$

Note that the pressure drop through the valve at the maximum flow rate is less than at the design flow rate.

Case 2: (80 psi design with C_{v2} = 22.36)

$$\Delta P_T = 80 + 40 = 120 \text{ psi}$$

$$F_{max,2} = (22.38)(1.0)[120 - 40(F_{max,2}/100)^2]^{1/2} \tag{5}$$

Solving for $F_{max,2}$ yields 141 gpm. So the maximum flow

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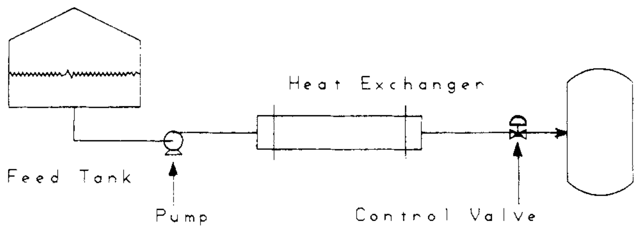

Figure 1. Process for pump and control valve design.

through this valve, which has been designed for a higher pressure drop, is over 40% more than design. The valve pressure drop at $F_{max,2}$ is

$$\Delta P_{v,2}^{max} = 120 - 40(141/100)^2 = 40.5 \text{ psi}$$

We can see from the results above that the valve that has been designed for the larger pressure drop can produce *more* of an increase in the flow rate at its maximum capacity.

B. Minimum Turndown. Now consider what happens when we want to reduce the flow. Control valves do not work too well when they are less than about 10% open. They can become mechanically unstable, shutting off completely and then popping partially open. The resulting fluctuations in flow are undesirable. Therefore, we want to design for a minimum valve opening of 10%. Let us see what the minimum flow rates will be in the two cases considered above when the two valves are pinched down so that $f_{(x)} = 0.1$.

In this case, the lower flow rate will mean a decrease in the pressure drop over the heat exchanger and therefore an *increase* in the pressure drop over the control valve.

Case 1: (20 psi design)

$$F_{min,1} = (0.1)(44.72)[60 - 40(F_{min,1}/100)^2]^{1/2} \quad (6)$$

Solving gives $F_{min,1}$ = 33.3 gpm.

Case 2: (80 psi design)

$$F_{min,2} = (0.1)(22.36)[120 - 40(F_{min,2}/100)^2]^{1/2} \quad (7)$$

Solving gives $F_{min,2}$ = 24.2 gpm.

These results show that the minimum flow rate is lower for the valve that was designed for a larger pressure drop. So not only can we *increase* the flow more but we also can *reduce* it more. Thus, the *turndown* (the ratio of F_{max} to F_{min}) of the big ΔP valve is larger:

turndown ratio for 20 psi design valve = 115/33.3 = 3.46

turndown ratio for 80 psi design valve = 141/24.2 = 5.83

This specific example demonstrates the increase in rangeability provided by designing for higher pressure drops over control valves.

It should be noted that this problem of control valve and pump sizing is independent of considerations of types of control valve trim (linear or equal percentage) and installed versus inherent valve characteristics. These aspects, which are discussed in many of the standard texts (for example, Shinskey (1979)), affect how the flow changes as stem position is changed over its entire range, but they do not affect the maximum and minumum flows through the system.

Existing Methods

Two methods have been proposed to resolve this conflict between the steady-state energy consumption and rangeability.

A. Common Heuristic. A commonly used heuristic recommends that the pressure drop over the control valve at design should be 50% of the total pressure drop through the system. Although widely used, this procedure often leads to poorly controlled processes.

In the numerical example considered above, if the heat-exchanger pressure drop is 40 psi, the control valve pressure drop at design would be set at 20 psi. This is case 1 considered above. The results showed that the maximum increase in the flow rate was only 15%, and the minimum flow rate was one-third of the design.

In some situations, this amount of rangeability is perfectly acceptable, e.g., for a feed flow rate into a process unit. However, in other situations it is very important to be able to substantially increase the flow rate above the design conditions. For example, the cooling water to the jacket of an exothermic chemical reaction may have to be doubled or tripled to handle dynamic upsets.

B. Connell Procedure. Connell (1987) proposed a method for determining the design pressure drop over a control valve that used the maximum flow rate and some heuristic "safety factors". These factors provide additional valve pressure drop to compensate for possible uncertainty in pressure drops through process equipment and changes in the pressure levels in the vessels at the inlet or exit of the pump/valve circuit. Connell did not consider the turndown problem. A comparison of valve and pump designs using Connell's method and using the proposed procedure will be given later in this paper.

Recommended Design Procedure

A more logical approach is to base the design of the control valve and the pump on producing a process that can attain both the maximum and the minimum flow conditions. The design flow conditions are only used to get the pressure drop over the heat exchanger (or fixed resistance part of the process).

The designer must specify the maximum flow rate that is required under the worst conditions *and* the minimum flow rate that is required. Then the valve flow equations for the maximum and minimum conditions give two equations and two unknowns: the pressure head of the centrifugal pump (ΔP_P) and the control valve size (C_v).

To illustrate the procedure, consider the design a control valve for admitting cooling water to a cooling coil in an exothermic chemical reactor. The normal flow rate is 50 gpm. To prevent reactor runaways, the valve must be able to provide 3 times the design flow rate. Because the sales forecast could be overly optimistic, a minimum flow rate of 50% of the design flow rate must be achieveable. The pressure drop through the cooling coil is 10 psi at a design flow rate of 50 gpm. The cooling water is to be pumped from an atmospheric tank. The water leaving the coil runs into a pipe in which the pressure is constant at 2 psig.

The pressure drop through the coil depends on the flow rate (F):

$$\Delta P_c = 10(F/50)^2 \quad (8)$$

The pressure drop over the control valve is the total pressure drop available (which we don't know yet) minus the pressure drop over the coil.

We will consider three cases: the first when the pump curve is flat, the second when the pump head decreases linearly with flow rate, and the third when the pump curve is parabolic.

A. Flat Pump Curve. If the pump curve is flat, ΔP_T is constant:

$$\Delta P_v = \Delta P_T - 10(F/50)^2 \quad (9)$$

We can write two equations, one for the maximum flow conditions and one for the minimum flow conditions. At the maximum conditions,

$$150 = C_v(1.0)[\Delta P_T - 10(150/50)^2]^{1/2} \quad (10)$$

At the minimum conditions,

$$25 = C_v(0.1)[\Delta P_T - 10(25/50)^2]^{1/2} \quad (11)$$

Solving simultaneously for the two unknowns yields the control valve size ($C_v = 21.3$) and the pump head ($\Delta P_P = \Delta P_T + 2 = 139.2 + 2 = 141.2$ psi).

At the design conditions (50 gpm), the valve fraction open (f_{des}) will be given by

$$50 = 21.3 f_{des}(139.2 - 10)^{1/2} \Rightarrow f_{des} = 0.206 \quad (12)$$

The pressure drop over the valve at design conditions is 129.2 psi, at maximum flow is 49.2 psi, and at minimum flow is 136.7 psi.

This example illustrates how the proposed method determines simultaneously both the control valve size and the pump head.

B. Linear Pump Curve. To illustrate the effect of changing centrifugal pump head with flow rate on control valve design, consider the example discussed above but now assume that the pump curve drops off with a constant slope of −0.1 psi/gpm. This means that there is more pressure drop available over the valve at low flow rates and less pressure drop at high rates:

$$\Delta P_T = \Delta P_P^0 - 0.1F - 2 \quad (13)$$

where ΔP_P^0 is the pump head at zero flow.

Now the pressure drop over the control valve is

$$\Delta P_v = \Delta P_P^0 - 0.1F - 2 - 10(F/50)^2 \quad (14)$$

At the maximum conditions,

$$150 = C_v(1.0)[\Delta P_P^0 - 0.1(150) - 2 - 10(150/50)^2]^{1/2} \quad (15)$$

At the minimum conditions,

$$25 = C_v(0.1)[\Delta P_P^0 - 0.1(25) - 2 - 10(25/50)^2]^{1/2} \quad (16)$$

Solving simultaneously for the two unknowns yields the control valve size ($C_v = 20.0$) and the pump head ($\Delta P_P^0 = 163.2$ psi). The valve pressure drop at design is now 146.2 psi, compared to the flat pump curve valve pressure drop of 129.2 psi.

C. Parabolic Pump Curve. To be more realistic, let us consider a pump curve that is parabolic. As a specific example, suppose we can describe the curve by a second-order polynomial that goes through three points: zero flow, half design flow, and design flow. We define the pump head at zero flow to be ΔP_P^0, which we have to determine. We assume that the shape of the pump curve is known so that the change in pump head from zero flow to design flow is ΔP_{des} and the change in pump head from zero flow to half of design flow is $\Delta P_{des/2}$. See Figure 2. The equation giving the pump head at any flow rate (F) is

$$\Delta P_P = \Delta P_P^0 + a_1 F + a_2 F^2 \quad (17)$$

where the constants a_1 and a_2 can be calculated from the known drop-off in the pump curve as flow changes from zero to half design to design:

$$a_1 = \frac{\Delta P_{des} - 4\Delta P_{des/2}}{F_{des}} \quad (18)$$

$$a_2 = \frac{2(2\Delta P_{des/2} - \Delta P_{des})}{F_{des}^2} \quad (19)$$

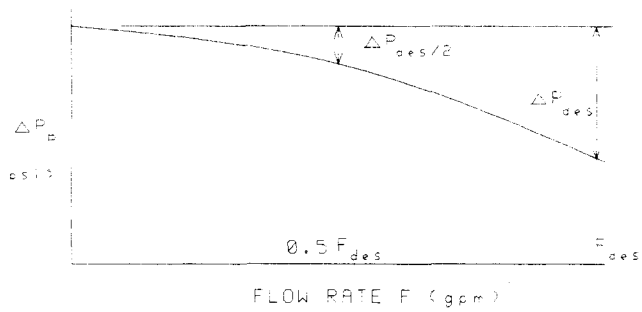

Figure 2. Parabolic pump curve.

To take a specific case, assume that the pump curve drops 2.5 psi from zero flow to half design flow (50 gpm in our numerical example) and 10 psi from zero flow to design flow (100 gpm). Evaluation of the constants gives $a_1 = 0$ and $a_2 = -0.001$. For this case, the pump head is given by

$$\Delta P_P = \Delta P_P^0 - 0.001F^2 \quad (20)$$

At the maximum conditions,

$$150 = C_v(1.0)[\Delta P_P^0 - 0.001(150)^2 - 2 - 10(150/50)^2]^{1/2} \quad (21)$$

At the minimum conditions,

$$25 = C_v(0.1)[\Delta P_P^0 - 0.001(25)^2 - 2 - 10(25/50)^2]^{1/2} \quad (22)$$

Solving simultaneously for the two unknowns yields the control valve size ($C_v = 19.1$) and the pump head at zero flow ($\Delta P_P^0 = 176.0$ psi). The valve pressure drop at design is 154 psi.

Thus, the effect of a variable pump head is to require a larger pressure drop over the valve at design conditions and a smaller control valve.

The method discussed above produces designs for both the control valve and the pump. A similar procedure was used by Papastathopoulou and Luyben (1988) for the simultaneous design of the control valve and the heat-exchanger area in the design of flooded-reboiler systems.

Comparison with Connell's Method

One of the examples given by Connell (1987) was a process that had three heat exchangers, orifice, piping, and a furnace. The total pressure drop of this fixed resistance part of the process at design flow rate was 124 psi. The maximum flow rate was set at 1.2 times the design flow rate. Connell recommended a 76 psi valve pressure drop at design flow rate. Assuming a flat pump curve, the total pressure drop available for the process equipment and the control valve was 200 psi.

At the maximum flow rate, the equipment pressure drop was $124(1.2)^2 = 178.6$ psi. Thus, the valve pressure drop at maximum flow was 21.4 psi. The valve (C_v) can be calculated for this maximum condition: $C_v = 1.2F_{des}/21.4^{1/2}$. The valve fraction open at design is

$$f_{des} = \frac{F_{des}}{[1.2F_{des}/21.4^{1/2}]76^{1/2}} = 0.442 \quad (23)$$

The ratio of the minimum flow rate to the design flow rate can be found by solving the equation given below:

$$F_{min} = 0.1[1.2F_{des}/21.4^{1/2}][200 - 124(F_{min}/F_{des})^2]^{1/2} \quad (24)$$

$$F_{min}/F_{des} = 0.352$$

Thus, the Connell procedure for this process gives a system in which the flow can be increased to 120% and reduced to 35% of the design.

In the proposed method, the desired minimum flow must be explicitly specified. If the minimum flow rate were set at 35% of the design, the two methods would give identical results. If, however, the minimum flow rate were given as 60% of the design, the results from the Connell method would not change since it does not consider turndown. However, the proposed method would give a design that required only a 60.2 psi pressure drop over the valve at design flow rate.

Limitations

The control valve/pump sizing procedure proposed above is not without its limitations. The two design equations for the maximum and minimum conditions in general terms (for a flat pump curve) are

$$F_{max} = C_v[\Delta P_T - (\Delta P_H)_{des}(F_{max}/F_{des})^2]^{1/2} \quad (25)$$

$$F_{min} = f_{min}C_v[\Delta P_T - (\Delta P_H)_{des}(F_{min}/F_{des})^2]^{1/2} \quad (26)$$

where ΔP_T is the total pressure drop through the system at design flow rates, $(\Delta P_H)_{des}$ is the pressure drop through the fixed resistances in the system at design flow, f_{min} is the minimum valve opening, and F_{des} is the flow rate at design. Solving these two equations for ΔP_T gives

$$\frac{\Delta P_T}{(\Delta P_H)_{des}} = \frac{\dfrac{F_{max}{}^2 - F_{min}{}^2}{F_{des}{}^2}}{1 - \left(\dfrac{f_{min}F_{max}}{F_{min}}\right)^2} \quad (27)$$

It is clear from eq 27 that as the second term in the denominator approaches unity, the required pressure drop goes to infinity! So there is a limit to the achievable rangeability of a system.

Let us define this term as the rangeability index of the system, $\mathcal{R}$:

$$\mathcal{R} \equiv \frac{f_{min}F_{max}}{F_{min}} \quad (28)$$

The parameters on the right side of eq 28 *must* be chosen such that $\mathcal{R}$ is less than unity.

This can be illustrated using the numbers from the example above. If the minimum flow rate is reduced from 50% of the design (where ΔP_T was 139.2 psi) to 40%, the new ΔP_T becomes 202 psi. If F_{min} is reduced further to 35% of the design, ΔP_T is 335 psi. In the limit as F_{min} goes to 30% of the design, the rangeability index becomes

$$\mathcal{R} \equiv \frac{f_{min}F_{max}}{F_{min}} = \frac{(0.1)(150)}{15} = 1$$

and the total pressure drop required goes to infinity.

This limitation can be reduced, if a larger turndown ratio is required, by using a value of f_{min} below 0.1. This can be accomplished by using two control valves in parallel (one large valve and one small valve) that are set up in a split-range configuration. The small valve opens first and then the large valve opens as the signal to the two valve changes over its full range.

Conclusion

A simple procedure is proposed that provides a rational approach to the problem of designing control valve and pumping systems. The method is based on guaranteeing that both maximum and minimum flow rates can be attained by the system. The designer does not have to rely on heuristics but must define the largest and the smallest flow rates required.

Nomenclature

a_i = coefficients in parabolic pump curve
C_v = control valve sizing coefficient
f_{des} = fraction of valve opening at design flow rate
f_{max} = fraction of valve opening at maximum flow rate
f_{min} = fraction of valve opening at minimum flow rate
F_{des} = design flow rate, gpm
F_{max} = maximum flow rate, gpm
F_{min} = minimum flow rate, gpm
$f_{(x)}$ = fraction of valve opening
$\mathcal{R}$ = rangeability index
x = valve stem position (fraction open)
ΔP_{des} = change in pump head from zero flow to design flow, psi
$\Delta P_{des/2}$ = change in pump head from zero flow to half of design flow, psi
ΔP_H = pressure drop over heat exchanger, psi
ΔP_T = total pressure drop over heat exchanger and control valve, psi
ΔP_v = pressure drop over control valve, psi
ΔP_P = pressure head produced by pump, psi
ΔP_P^0 = pressure head produced by pump at zero flow, psi

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