technical monograph 7

The Dynamics of Level and Pressure Control

Charles B. Schuder

Research Engineer
Fisher Controls International, Inc.

FISHER®

FISHER-ROSEMOUNT™

The Dynamics of Level and Pressure Control

Part I - Liquid Level

The control of liquid level is relatively easy where properly sized and characterized valves are used and the vessel capacity is large as compared to valve capacity. Ratios of valve capacity (Gallons/Minute) to vessel capacity (Gallons/Inch) up to 100:1 ordinarily present no problems and ratios as high as 500:1 can usually be handled by careful selection and application of components.

The open loop transfer function of these systems can be used to predict the stability of a proposed installation or to size a process vessel so that the system will meet predetermined performance requirements.

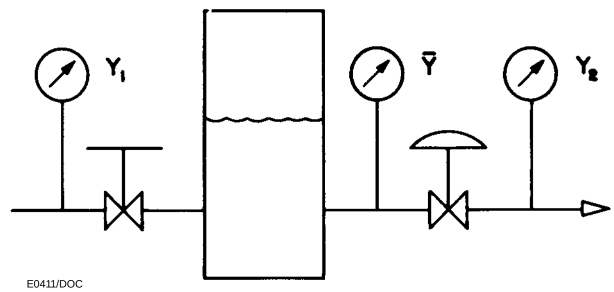


Figure 1. Level Control Process

The Process

Figure 1 shows a typical level control process. The dynamics are the same for either inflow or outflow control. Neglecting self-regulation, we have:

$$\frac{dY}{dt} = \frac{Q}{C}$$

Giving for a transfer function:

$$\frac{\mathcal{L}(Y)}{\mathcal{L}(Q)} = \frac{1}{SC} \qquad (1)$$

Considering self-regulation:

$$\frac{dY}{dt} = \frac{1}{C}\ (Q - YZ)$$

Where Z is $\frac{\partial Q}{\partial Y}$ at fixed valve position. Our transfer function is:

$$\frac{\mathcal{L}(Y)}{\mathcal{L}(Q)} = \frac{1}{CS + Z} = \frac{1/Z}{\frac{C}{Z}\ S + 1} \qquad (2)$$

For self-regulation at the outlet:

$$\overline{Q} = B\sqrt{\overline{Y} - Y_2}$$

$$Z = \frac{\partial Q}{\partial Y} = \frac{\partial \overline{Q}}{\partial \overline{Y}} = \frac{B}{2\sqrt{\overline{Y} - Y_2}}$$

$$But\ \ B = \overline{Q}\,/\sqrt{\overline{Y} - Y_2}$$

$$\therefore\ Z_{out} = \frac{\overline{Q}}{2(\overline{Y} - Y_2)}$$

In the same manner at the inlet:

$$Z_{in} = \frac{\overline{Q}}{2(Y_1 - \overline{Y})}$$

Y = Liquid Level Head, Ft.
Y_1 = Upstream Head, Ft.
Y_2 = Downstream Head, Ft.
Q = Control Valve Flow at Constant Head, $Ft.^3/Sec.$
C = Vessel Capacity, $Ft.^3/Ft.$
Z = Self-Regulation Factor, $Ft.^2/Sec.$

NOTE: $\overline{Q}$ and $\overline{Y}$ are steady state values while Q and Y are deviations from equilibrium.

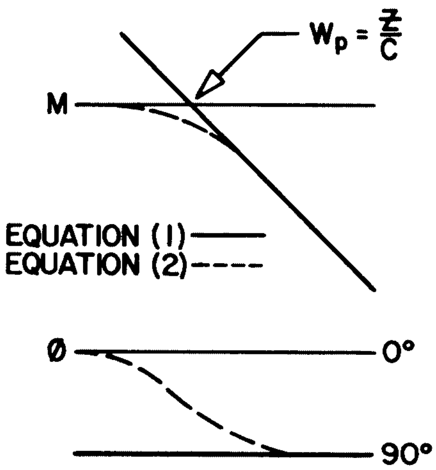
And for self-regulation at both points:

$$Z = \frac{\overline{Q}}{2(Y_1 - \overline{Y})} = \frac{\overline{Q}}{2(\overline{Y} - Y_2)} \qquad (3)$$

Z, then, is easily calculated directly from the process operating conditions. Where the inflow enters the vessel above the liquid level, the first term in Eq. (3) should be omitted. It should also be omitted where there is no inlet restriction as in the case of heater drainers.

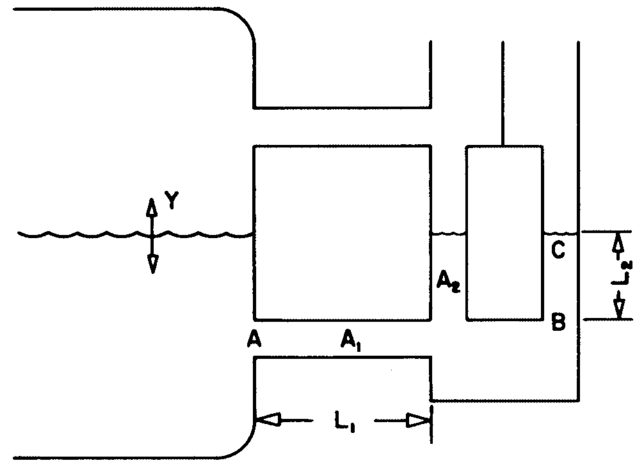
We now have the process transfer functions, but have yet to establish the conditions under which the simpler function, Eq. (1), can be used in place of Eq. (2). A simple rule can be developed from an examination of a Bode plot of the two transforms as shown in Figure 2.

Obviously, if the break frequency, ωp of the self-regulated process occurs considerably ahead of the earli-



E0412/DOC

Figure 2. Process Transfer Functions



E0413/DOC

Figure 3. Float Cage Nomenclature

est break frequency of the other system components, then self-regulation can be neglected and Eq. (1) can be used. A separation of 10 is adequate for most analysis work so the requirement for use of Eq. (1) can be written as:

$$\frac{C}{Z} > 10 \; T_m \qquad (4)$$

where T_m is the largest time constant of the other system components. In most cases Tm will be the reciprocal of the undamped natural frequency of the float cage. It will be found that there are very few installations where self-regulation will need to be considered.

The Measuring Element

The measuring element most commonly found in process level control is the externally mounted displacer as shown in Figure 3. This component introduces a highly underdamped second order lag which is of major importance in the system analysis. The undamped natural frequency of the cage may be calculated, but the damping ratio must be obtained experimentally.

To find the undamped natural frequency we neglect friction and apply Newton's Second Law to the fluid mass in Sections 1 and 2.

$$P_a \; A_1 - P_b \; A_1 = A_1 \; L_1 \; \varrho U^{11}$$

$$P_b \; A_2 - P_c \; A_2 = A_2 \; L_2 \; \varrho X^{11}$$

"U" and "X" are the displacements in Section 1 and 2 respectively. Making the substitutions:

$$U^{11} = X^{11} \frac{A_2}{A_1}, \; P_c = W_x, \; P_a = W_y$$

and combining the two equations gives:

$$X^{11} \frac{L_2 + L_1 \left(\frac{A_2}{A_1}\right)}{g} + X = Y$$

so that the undamped natural frequency of the cage is:

$$W_n = \sqrt{\frac{g}{L_2 + L_1\left(\frac{A_2}{A_1}\right)}} \; Radians/Sec. \qquad (5)$$

where:
G = Acceleration of Gravity, Ft./Sec.^{2}
L_1 = Length of Liquid Equalizing Line, Ft.
L_2 = Wetted Length of Displacer, Ft.
A_1 = Area of Liquid Equalizing Line, Ft.^{2}
A_2 = Area of Annular Ring between Displacer and Cage, Ft.^{2}

The dynamics of the displacer have been neglected because its natural frequency will always be much higher than that of the cage. Float cages should be installed so as to provide a high natural frequency. Eq. (5) shows that this can be done by using a minimum length of large diameter pipe for the liquid equalizing line.

Damping ratios have been obtained experimentally for a number of installations and are summarized in Table 1. All displacers had a 100 cu. in. volume and were installed in cages made from 4" schedule 80 pipe. The equalizing connection was made up with a plug cock, an elbow, and 2 feet of 2-inch pipe.

Table 1. Damping Ratios

Displacer Length, Inch	Fluid	Damping Ratio
14	Water	0.058
14	Solvent	0.052
14	Hyraulic Oil (M.I.H. 10)	0.330
32	Water	0.062
60	Water	0.055
14	Ca Cl Brine (1.05 S.G.)	0.054
14	Ca Cl Brine (1.10 S.G.)	0.050
14	Ca Cl Brine (1.20 S.G.)	0.047
14	Ca Cl Brine (1.30 S.G.)	0.047

Data has also been obtained for other equalizing line sizes and configurations, but the damping ratios found were not significantly different from those shown in the table. For low viscosity fluids such as water a damping ratio of 0.05 or 0.06 should be a satisfactory value to use in most system performance calculations.

Improved damping ratios can be obtained from cages which have damping orifices built in to the liquid connection. These designs reduce the resonant peak and permit loop gains about six times as great as those, which can be utilized with conventional cages. A damping ratio of 0.3 is typical of current damped-cage designs, which use a one-half inch diameter orifice for this purpose.

From the above considerations, the transfer function is seen to be:

$$\frac{\mathcal{L}(X)}{\mathcal{L}(Y)} = \frac{1}{\left(\frac{S}{W_n}\right)^2 + \frac{2\zeta S}{W_n} + 1} \qquad (6)$$

Control Components

The transfer functions of the controller and actuator will vary widely depending on installation and type of equipment selected.

Both actuators and controllers can have time constants ranging from a few hundredths of a second up to several seconds. Control equipment manufacturers can normally supply frequency response information, although it is often necessary to modify such data for the transmission system being used.

Gain

The open loop gain "K" for all components except the process will be the steady state flow through the control valve produced by a one foot level change. This can be written as:

$$K = \frac{100R}{L(PB)}$$

where:

L = Displacer Length, Ft.
(PB) = Proportional Band, Percent
R = Valve Capacity, Ft.^{3}/Sec.

At this point we are assuming a linear "valve + system" characteristic, i.e., the valve gain (Ft.^{3}/Sec. per psi control pressure) is constant at all loads. The valve itself may or may not be linear depending on the system flow-pressure relationship. This linear characteristic will give optimum control in systems with negligible self-regulation. Where self-regulation is an important factor., see Eq. (4), the valve characteristics must be modified to maintain constant relative stability over the entire flow range.

The need for such modification is much more likely to arise in pressure control than in liquid level control so the problem will be treated in detail in that section.

Where the valve, as installed, does not have constant gain, the value of "R" to use is obtained by multiplying the slope of the "valve + system" curve (Δ% flow/Δ% lift) by the maximum valve capacity.

When we include the process in our open loop gain equation, we obtain, neglecting self-regulation:

$$K_v = \frac{100R}{CL(PB)}$$

and considering self-regulation:

$$K_p = \frac{100R}{ZL(PB)}$$

Open Loop Analysis

Our complete open loop transfer function for a level control system is, without self-regulation:

$$\frac{K_v}{S\left[\left(\frac{S}{W_n}\right)^2 + \frac{2\zeta S}{W_n} + 1\right]} \; x \left[\begin{matrix}\text{Transfer Function}\\ \text{of Control}\\ \text{Components}\end{matrix}\right] \qquad (1A)$$

and with self-regulation:

$$\frac{K_p}{\left(\frac{C}{Z}\;S + 1\right)\left[\left(\frac{S}{W_n}\right)^2 + \frac{2\zeta S}{W_n} + 1\right]} \; x \left[\begin{matrix}\text{Transfer}\\ \text{Function of}\\ \text{Control}\\ \text{Components}\end{matrix}\right] \qquad (2A)$$

To illustrate the general characteristics of these systems we can make a polar plot of Eq. (1A) assuming that a first order lag in the actuator dominates the control element dynamics.

Figure 4 shows such a plot based on a float cage natural frequency of 5 radians/second and an open loop gain, K_v, of 0.5.

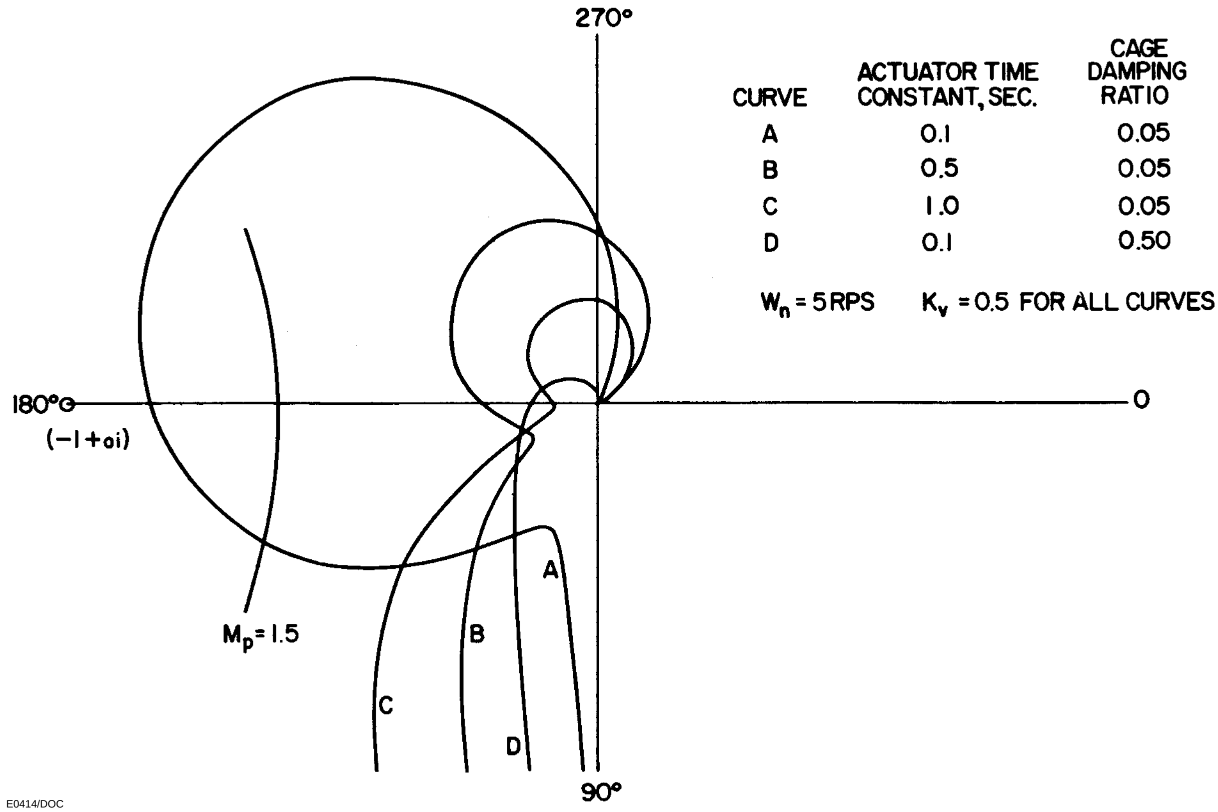


Figure 4. Typical Response Curve

Curves A, B, and C illustrate the effect of the actuator time constant on system stability. The use of a fast actuator, curve A, would make it necessary to reduce the gain by using a wide proportional band or large capacity vessel. A slower actuator will permit the use of higher gain, but a very slow actuator, curve C, will again require a gain decrease. The operating frequency of the system drops slightly as slower actuators are used. When we reach a point where the low frequency portion of the curve touches the M_p circle before the loop produced by the float cage, a large reduction in operating frequency will occur. The system will become sluggish and its response to load changes may be unsatisfactory.

Retaining the fast actuator and damping the float cage, curve D, results in a system that can use a relatively high gain without an appreciable sacrifice in response.

If we would replot the systems using a considerably lower float cage natural frequency, all of the curves would encircle the (–1, oi) point, indicating instability. It is very important to install the float cage so as to obtain a high natural frequency and take advantage of the attenuation offered by the integrating action of the process.

Application

Dynamic analysis can be a useful tool in designing control systems and in improving the performance of existing systems. In system design the open-loop transfer function is plotted, and the gain adjusted to give required stability. The value of K_V thus obtained is substituted in the gain equation and minimum tank size or proportional-band determined. For existing systems the curve is plotted using the actual gain. A study of the plot will indicate the changes required to improve performance.

It is essential that the open-loop response be shown on a Nyquist diagram or other gain-angle chart. The two gain peaks which occur in many level control loops can produce oscillatory systems even though adequate phase and gain margins are provided.

The methods outlined have been applied by the author to a variety of level-control systems. In all cases the calculated gain for a pre-determined relative stability was within $\pm 20\%$ of the measured gain. In most cases the error was considerably less. This error amounts to about $\pm 10\%$ discrepancy in predicting vessel diameters, and is probably representative of the accuracy we can expect when working with control equipment data modified for our system.

With certain simplifying assumptions it is possible to reduce the open-loop analysis to a simple relationship which can be used as a rough guide in the design of level control systems. The necessary assumptions are: (1) ideal cage installation, (2) 14” displacer, (3) response of controller and actuator flat well beyond 0.5 cps, and (4) properly sized and characterized control valve. Performing the analysis for these conditions gives, for an undamped cage:

$$C = \frac{Q_m}{K_u\,(PB)} \qquad (7)$$

and for a damped cage (ς= 0.3)

$$C = \frac{Q_m}{K_d\,(PB)} \qquad (8)$$

The value of C obtained will be the minimum vessel capacitance that will give satisfactory control for the expected maximum flow rate (Q_m) and the desired percent proportional band (PB). Table 2 gives values of the constants K_d and K_u for different units of vessel capacitance and flow rate. Longer displacers will permit higher controller gain (or smaller vessels), but not in direct proportion because cage natural frequency will be lower. Deviations from the other assumptions will generally call for lower controller gains.

Table 2. K_u and K_d Values

Units of Vessel Capacitance, C	**Ft^2**	**In^2**	**Gal/In**
Units of Maximum Flow, Q_m	**Ft^3/Sec**	**Gal/Min**	**Gal/Min**
K_u	1.25×10^{-3}	3.91×10^{-3}	0.9
K_d	7.8×10^{-3}	2.43×10^{-2}	5.6

PART II - PRESSURE

Most pressure control systems will give satisfactory performance without the benefit of dynamic analysis in either process design or control equipment selection. In many systems, however, the pressure must be held within very close limits, even during fast load changes. Here, dynamic analysis is essential for intelligent system design.

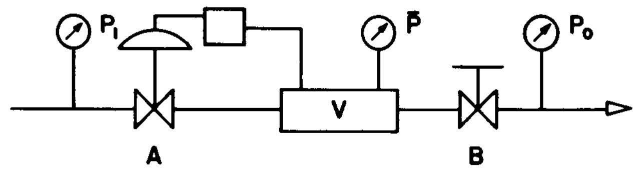


Figure 5. Basic Control System

P1 = Upstream Pressure, PSFA
P = Controlled Pressure, PSFA
P_o = Downstream Pressure, PSFA
F = Control Valve Flow at Constant Pressure Drop, Pounds/Sec.
V = Volume Downstream of Valve ”A”, Ft.^{3}
C = Sonic Velocity, Ft./Sec.
k = Ratio of Specific Heats, C_p/C_v
ρ = Fluid Density, Slugs/Ft.^{3}
Z = Self-Regulation Factor, Ft.^{2}/Sec.
g = Acceleration of Gravity, Ft./Sec.^{2}
NOTE: $\overline{F}$ and $\overline{P}$ are steady state values while F and P are deviations from equilibrium.

The System

Pressure control systems present a wide variety of component configurations, but in most cases they can be reduced to the elements shown in Figure 5. Valve “A” is shown controlling the intermediate pressure with valve “B” as the source of load disturbance. As far as the system dynamics are concerned, valve “B” could be working as a back pressure valve with valve “A” introducing the load changes. We could also have each valve controlling its respective downstream pressure as in the case of two-stage gas regulation. In this case our analysis would be valid for the loop involving valve “A” (neglecting interaction) and the system controlled by valve “B” would be analyzed by shifting our notation one step downstream. The volume shown between the valves could be a receiver, a short length of pipe, or with admitted error a lumped distribution system. The lumped system concept is valid for frequencies well below $\frac{C}{1}$.

The Process

The analysis of the process is based on two assumptions:

1. The controlled fluid is a perfect gas, which compresses and expands isentropically in the control volume.

Most pressure control systems have operating frequencies, which are too high to permit appreciable transfer of heat from the gas to the pipe or vessel wall. The effects of friction should be negligible, except for long lines. The irreversible expansion, which occurs immediately downstream of the valve orifice, does not effect the validity of this assumption.

2. The processes upstream and downstream of the control system can be considered as large pressure sources and sinks so that P_1 and P_0 do not vary with fluctuations in P. We will, however, permit P_1 and P_0 to vary with load.

Before writing the process differential equation we need a relationship between the net inflow to the volume and the controlled pressure.

By equating the net mass flow into the volume to the rate of change of mass within the volume we obtain:

$$F_{net} = gV \frac{d\rho}{dt} \qquad (1)$$

And from our assumption of isentropic flow:

$$\frac{P}{\rho k} = A\ Constant, \ \frac{d\rho}{dP} = \frac{\rho}{PK} = \frac{1}{C^2}$$

$$\frac{d\rho}{dt} = \frac{1}{C^2}\frac{dP}{dt}$$

Substituting in (1):

$$F_{net} = \frac{gV}{C^2}\frac{dP}{dt}$$

Solving for $\frac{dP}{dt}$ and including self-regulation:

$$\frac{dP}{dt} = \frac{C^2}{gV}(F - PZ) \qquad (2)$$

where Z is a self-regulation factor, $\frac{\partial F}{\partial P}$ at a fixed valve position. An expression for Z can be obtained in terms of the system operating conditions by differentiating an equation, which relates valve flow to the upstream and downstream pressures.

There are a number of such equations in use, and, although they are considerably different in form, they all show good correlation with experimental data.

One equation, for subsonic flow, which is convenient for our use is:

$$\overline{F} = B\sqrt{\frac{(P_1 - \overline{P})\overline{P}}{T_1}} \qquad (3)$$

The numerical value of the constant "B" is unimportant since it can be eliminated by the substitution of known operating conditions.

Using, with Eq. (3), the relationship $F = B_1 P_1 \ / \ \sqrt{T_1}$ for sonic flow we can develop a table of Z factors for all flow combinations. Table 3 is based on K = 1.4 and a critical pressure ratio of 0.5 obtained from pressure measurements outside the valve body.

Table 3. Self Regulation Factors

Case		Z
I	$\overline{P} < 0.5\,P_1$, $P_o < 0.5\,\overline{P}$	$\frac{.86\,\overline{F}}{\overline{P}}$
II	$\overline{P} > 0.5\,P_1$, $P_o > 0.5\,\overline{P}$	$-\frac{\overline{F}}{2}\frac{(P_1 - 2\overline{P})}{(P_1 - \overline{P})\overline{P}} + \frac{\overline{F}}{2.8}\frac{\left[1 + .4\left(\frac{P_o}{\overline{P}}\right)\right]}{(\overline{P} - P_o)}$
III	$\overline{P} > 0.5\,P_1$, $P_o < 0.5\,\overline{P}$	$-\frac{\overline{F}}{2}\frac{(P_1 - 2\overline{P})}{(P_1 - \overline{P})\overline{P}} + \frac{.86\,\overline{F}}{\overline{P}}$
IV	$\overline{P} < 0.5\,P_1$, $P_o > 0.5\,\overline{P}$	$\frac{\overline{F}\left[1 + .4\left(\frac{P_o}{\overline{P}}\right)\right]}{2.8(\overline{P} - P_o)}$

Taking the LaPlace transform of both members of Eq. (2), we obtain our process transfer function:

$$\frac{\mathcal{L}(P)}{\mathcal{L}(F)} = \frac{1/Z}{\frac{gV}{C^2 Z}S + 1} \qquad (4)$$

In some systems self-regulation can be neglected so that the transfer function is simply:

$$\frac{\mathcal{L}(P)}{\mathcal{L}(F)} = \frac{C^2}{gVS} \qquad (5)$$

and the condition that must be satisfied before using this simpler form is:

$$\frac{gV}{C^2 Z} \gg T_m \qquad (6)$$

where T_m is the largest time constant of the control components.

Open Loop Analysis

The open loop gain, K, of the system components, other than the process, will be the flow through the control valve produced by a 1 PSF change in control pressure. This can be written as.:

$$K = \frac{100RM}{bu}$$

where:
R = Maximum Valve Capacity, #/Sec.
M = Slope of "Valve + System" Curve, (%ΔF/%ΔY). "Y" is valve lift.
b = Proportional Band of Controller, Percent
u = Rating of Measuring Element (Bourdon Tube or Bellows), PSFG

The two open loop transfer functions for the complete pressure control loop are —

with self-regulation:

$$\frac{\frac{100RM}{buZ}}{\frac{gV}{C^2Z}\; S+1} \; X \begin{bmatrix} \textit{Transfer function of} \\ \textit{control components} \\ \textit{at unity gain.} \end{bmatrix} \qquad (4A)$$

and without self-regulation:

$$\frac{100RMC^2}{SgV\,bu} \; X \begin{bmatrix} \textit{Transfer function of} \\ \textit{control components} \\ \textit{at unity gain.} \end{bmatrix} \qquad (5A)$$

For the system with negligible self-regulation the gain and time constants are independent of load so that a linear "valve + system" characteristic will give optimum control.

For systems with appreciable self-regulation the selection of optimum valve characteristics becomes more difficult.

To see the nature of this problem we can write Eq. (4A) including a second order lag, critically damped, for the control components and the self-regulation factor from Table 2 for a case I process.

$$\frac{\dfrac{100\,RM}{.86bu\left(\frac{F}{P}\right)}}{\left[\dfrac{gVS}{C^2\left(\frac{F}{P}\right).86}\right] + (T_cS+1)^2} \qquad (7)$$

We note that both the open loop gain and the process time constant, $\frac{gV}{C^2Z}$, vary inversely with load. If the process time constant is large as compared to T_c, a linear "valve + system" characteristic will give constant relative stability even though the gain approaches infinity at low loads.

If, however, we have a system such that the process time constant is negligible at all loads (for example, a zero volume system), then, for constant relative stability, we must provide a characteristic such that M/F is constant. This is obtained by a percentage characteristic.

If we use "Y" as the ratio of valve lift to maximum lift, we can write for a percentage characteristic:

$$M = \frac{d\left(\frac{\bar{F}}{R}\right)}{dY} = \left(\frac{\bar{F}}{R}\right)N \qquad N,\ a\ constant$$

Solving gives:

$$\ln\left(\frac{\bar{F}}{R)}\right) = NY + A\ constant$$

With the conditions that F/R = 1 when Y = 1 and F/R= 1/r when Y = 0

Gives N = lnr (r is rangeability)

We can now make the substitution of 1n r for the quantity $MR/\bar{F}$ in our transfer function so that the system stability becomes independent of maximum valve capacity. The stability does, however, depend upon valve rangeability. If, for example, the process could tolerate a valve rangeability of 10 instead of 50, the proportional band could be narrowed by the ratio ln 50/1n 10 = 1.7.

Selection of Valve Characteristics

Few systems will meet the requirements outlined above for the use of percentage or linear characteristics. We could develop some general rules to serve as a guide in valve selection, but it would be preferable to plot the exact characteristic as dictated by the open loop transfer function and base our decision on this curve.

To obtain the optimum valve characteristic from the open loop transfer function we can proceed as follows:

1. Write the system transfer function at 5%, 15% – 95% load. Other increments could be used depending on the accuracy required.
2. At each point, calculate the permissible gain, G_p, for a pre-selected relative stability. The use of a Nichols' chart for this step is desirable although calculations based on phase or gain margins will provide reasonable accuracy for most systems.
3. Calculate the required slope, M, of the valve characteristic curve at each point from the equation:

$$M = \frac{bu\,Z\,Gp}{100R} \qquad (8)$$

Since we want the curve of the valve itself rather than "valve + system" curve, we should use a value of "R" in each calculation based on the pressures available at that load.

The selection of a proportional band at this point is arbitrary. No matter what value we select, the final curve will give the multiplying factor to use in determining the setting at which the system will operate. The curve will be easier to plot, however, if we calculate a proportional band at the 45% or 55% point from:

$$b = \frac{100\,R}{uZ\,G_p} \qquad (9)$$

and use this value in calculating M at the other points.

4. Plot the valve characteristic curve on %F – %Y coordinates starting from 0, 0 and extending the curve by the calculated slopes to 100% load. The value of %Y at the 100% point determines the correction factor to apply to the proportional band used in Step 3.

To illustrate this procedure we can assign numerical values in Eq. (7) and plot the optimum valve characteristic for the system.

Let:

P_1 = 500 PSIG at no load dropping (with the square of the load) To 450 PSIG at the full load.
P = 85 PSIG
V = 10 $Ft.^3$
C = 1120 Ft./Sec. (Air)
T_c = 0.5 Sec.
P_o = 0 PSIG
U = 100 PSI
R = 5 #/Sec. at 500 PSI Inlet

Our open loop transfer function with P_1 in units of PSIA is:

$$\frac{\dfrac{1.13\, P_1\, M}{b\ \overline{F}}}{\left(\dfrac{4.3}{\overline{F}}\ S + 1\right)\ (0.5S + 1)^2}$$

The process time constant $4.3/\overline{F}$ at 45% load is 1.9 sec. so that our permissible gain at this point, based on a 45° phase margin, is 3.85. P_1 will be down to 505 PSIA at this load so that a convenient proportional band to use in the calculations is, by Eq. (9):

$$b = \frac{100.5\left(\frac{505}{515}\right)}{100\left(\frac{2.25}{100}\right) .86\ (3.85)} = 66\%$$

The next step is to prepare a work sheet as shown in Table 4 with the data needed to calculate M. A sample calculation at the 25% load points shows:

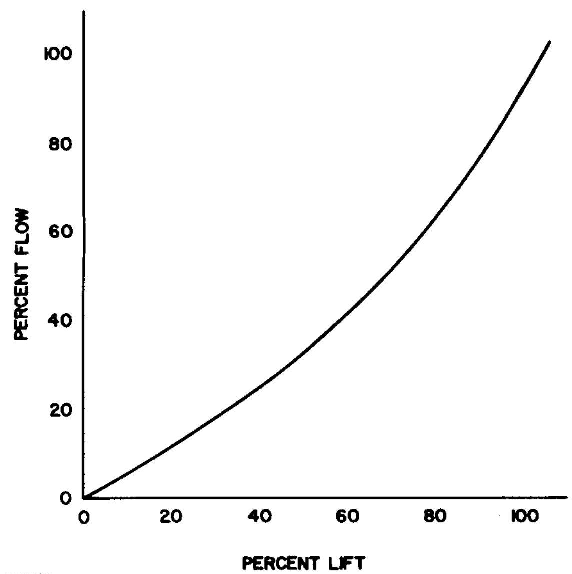
$$T_{process} = \frac{4.3}{\overline{F}} = \frac{4.3}{1.25} = 3.44\ sec.$$

$$g_p = 5.3 \text{ for } 45° \ Phase\ \ Margin$$

$$M = \frac{bG_p\overline{F}}{1.13P_1} = \frac{66 \times 5.3 \times 1.25}{1.13 \times 512} = 0.76$$

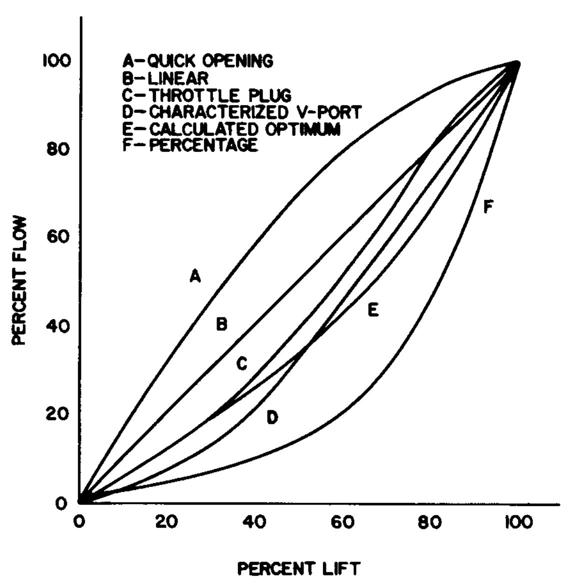
Table 4. Work Sheet

% Load	Flow #/Sec	P_1 PSIA	$T_{process}$, Sec	G_p	M
5	0.25	515	17.20	19.00	0.54
15	0.75	514	5.74	7.70	0.66
25	1.25	512	3.44	5.30	0.76
35	1.75	509	2.46	4.43	0.89
45	2.25	505	1.91	3.85	1.00
55	2.75	500	1.56	3.55	1.14
65	3.25	494	1.32	3.33	1.28
75	3.75	498	1.15	3.24	1.45
85	4.25	479	1.01	3.05	1.58
95	4.75	470	0.91	3.00	1.77



E0416 / IL

Figure 6. Calculated Characterristic Curve



E0417 / IL

Figure 7. Valve Characterristics

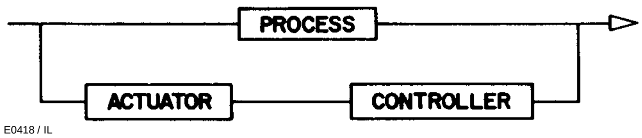


Figure 8. Block Diagram of System

Using the values of M from the work sheet we can plot the curve shown in Figure 6. The 100% load point occurs at 104% lift so that the proportional band at which the system will operate is 66 x 1.04 = 69%. This wide band would be unacceptable for nearly all applications so that we would add reset action or, if transient response was critical, we would redesign the system using a larger volume or faster control components. In a case where we would accept considerable drop in pressure, the change in P should be included in the calculation of Z.

If we expand our lift axis so that the curve passes through (100,100) we can compare the calculated characteristic with typical standard characteristics as shown in Figure 7.

The purpose of valve characterizing is to permit us to operate at a single, minimum proportional band setting at all loads. Consequently, our selection should be based on this minimum setting. The linear valve has a slope of 1 at all loads whereas our stem cannot tolerate a slope above 0.53 at 5% load. Therefore, our proportional band setting with a linear valve would have to be 130%. Calculating these values for all the valves gives approximately:

Custom Valve (Calculated)	69%
Linear	130%
V-Port	95%
Throttle Plug	95%
Quick Opening	195%
Percentage (50:1)	150%

On this basis we would probably select a V-Port or throttle plug inner valve. A special characteristic would be difficult to justify when we consider the differences, which may exist, between our idealized system and the actual system.

Transient Response

The block diagram of Figure 8 shows the system components arranged to obtain transient response following a load change.

The open loop analysis tells us very little about transient response because of the dynamic elements in the feedback path. If the time constant of the process is considerably larger than the time constants of the control components, the approximate overshoot can be obtained from the system damping ratio. Many pressure processes, however, have time constants of only a few hundredths of a second — much faster than most control components. In these cases, the initial overshoot can be excessive and, if transient performance is critical, a closed loop response must be plotted.

Transient response is normally of secondary importance in the analysis of pressure control systems. Most of the systems in service today are subject to such low magnitude or slow load changes that their poor transient response is of little significance.

The author has applied the open loop analysis procedure to an experimental system set up in the laboratory where the operating pressures and system volumes could be varied. Very good correlation was obtained in most of the test, although several tests at very low loads gave errors in predicted gain as high as 25%. Good results have been obtained in the analysis of a number of field installations although too few systems have been checked to justify a figure for accuracy that we might expect from these methods.

Liquids and Vapors

The transfer functions for the control of liquid systems with open surge tanks are the same as those for liquid level. For closed systems, the transfer function is the same as for compressible fluid systems with zero volumes. We could calculate a process time constant for these systems, but it would be insignificant in comparison to the control equipment lag. The self-regulation factor is:

$$Z = -\frac{\overline{F}}{2(P_1 - \overline{P})} + \frac{\overline{F}}{2(\overline{P} - P_o)}$$

so that the system requires percentage characteristics. Transient response is always poor. When a load change occurs, the pressure goes all the way to its self-regulation value and then moves back toward the control point at a rate depending on the speed of the control components.

Transfer functions for vapor systems can be obtained in the same manner as those for a perfect gas. Self-regulation factors are similar in form to those we have developed, but the capacitance will depend on $\partial\rho/\partial P$ as obtained from thermodynamic property curves.

Fisher Controls International, Inc.
205 South Center Street
Marshalltown, Iowa 50158 USA
Phone: (641) 754-3011
Fax: (641) 754-2830
Email: fc-valve@frmail.frco.com
Website: www.fisher.com

D350399X012 / Printed in U.S.A. / 1965

FISHER®

FISHER-ROSEMOUNT™