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Advice on How to Specify the Best Control Valve Characteristics

What is the best inherent flow characteristic for controllability, considering the effect of valve resolution and lost motion, process gain, piping system design, system pressure drops, fouling, flashing, and cavitation? This discussion is part of the Ask the Automation Pros series from the International Society of Automation, a regularly occurring column by Greg McMillan, who is an industry consultant, author, and 2010 ISA Life Achievement Award recipient.

By Gregory K. McMillan, Michael Taube, Michel Ruel, Mark Darby and R. Russell Rhinehart

Greg McMillan's Question

What is the best inherent flow characteristic for controllability, considering the effect of valve resolution and lost motion, process gain, piping system design, system pressure drops, fouling, flashing, and cavitation?

Michael Taube's Response

A control valve is but one part of a much larger system, which includes the pump/compressor (if there is one), piping, other "users" of the process stream, and other equipment (i.e., heat exchangers). Thus, choosing a valve based on its published characteristics alone is a very limited, even illusory way of achieving (or predicting) the desired behavior. Some process simulation vendors have recognized this and provide the means to model piping networks, which can be quite useful in assessing different control valves over the entire operating range (and conditions) to which it is subjected. This should really be a part of every process and control engineer's "toolbox," even if only implemented in a spreadsheet!

The important issue to recognize in such an analysis is that the process is rarely at the "design point", thus one must recognize the changes in up/downstream pressure, frictional losses and control valve characteristics over the entire range—from zero flow to "maximum" flow. This is, strictly speaking, a steady-state analysis, which does not require knowledge or recognition of dynamic behavior. Hence, the emphasis of being a process and control engineer's tool.

Michel Ruel's Response

When selecting a process control valve, I focus primarily on overall process gain and linearity. While I use software tools provided by valve manufacturers, the key is accurately estimating the pressure drops at different flow rates, specifically at minimum, normal, and maximum flows. The valve's inherent characteristic should be chosen to ensure the installed valve operates as linearly as possible.

As a rule of thumb, I aim for a maximum overall process gain of 3, with the ratio of maximum to minimum overall process gain kept below 3.

Some engineers may size the valve wrongly to ensure operation within 30% to 80% of its capacity, without considering factors like linearity and process gain. In such cases, an equal percentage valve characteristic can be used to minimize the impact of these errors. Since process characteristics have not been taken into account, this control loop will not perform.
Finally, it's important to account for resolution, deadband, backlash, and other potential issues to ensure the valve meets the expected performance standards.

Mark Darby's Response

An equal percentage valve is often recommended for processes in which the process gain is inversely proportional to flow, which also applies to a load input (a throughput variable). This means the process gain decreases as flow increases. A typical example is temperature control in a plug flow volume where the temperature controller is manipulating the flow to the volume for a load upset, which is the case for jackets, coils, and heat exchangers. If the controller outputs to an equal percentage valve, its valve characteristic (proportional to flow) will help offset the varying process gain, even if the pressure drop across the control valve is constant (in which, normally, one expects to use a linear valve). A modified equal percentage valve may help near the closed position.

But this result would not hold if the TIC outputs to an FIC in a cascade. Use of an equal percentage valve here could introduce a nonlinearity. For a flow loop cascade, an equal percentage valve would make sense if the pressure drop across the control valve dropped significantly over the required flow range.

Greg McMillan's Response

Process engineers are particularly good at detailing system pressure drops and available valve pressure drops. Often not recognized is that flows have to be manipulated beyond their steady state position to reach the setpoint for integrating and runaway process loops and for a fast response in many other loops. Also, in an attempt to save energy, the valve-to-system pressure drop is so low that the installed characteristic is severely distorted and there is a terrible loss in rangeability. Not understood is that a variable frequency drive (VFD) may have a worse flow characteristic and rangeability, in an attempt to minimize inverter and motor costs. (Look for the next "Ask the Automation Pros installment for guidance on VFD selection and installation.)

My hope is that process and control engineers will each bring their expertise to the table to get the best control valve characteristic. If most of the system pressure drop is due to the control valve and the process gain is not inversely proportional to flow, which is the case for most reagent valves manipulated by pH controllers and steam header valves manipulated by pressure controllers, a linear flow characteristic is required. For most other situations, an equal percentage characteristic is best. A quick opening characteristic is sometimes touted for pressure relief control, providing on-off control, but the nonlinearity and loss of rangeability are undesirable for throttling control.

The equal-percentage characteristic offers these practical advantages over the linear characteristic:

- Minimum effect of stick-slip near the closed position
- Maximum rangeability and linearity for low valve to system pressure drops
- Maximum forgiveness of under-sizing and over-sizing

- Beneficial process gain compensation from valve flow gain proportional to flow for a process gain inversely proportional to flow for jacket, coil, and heat exchanger temperature control but not for cases where vessel temperature directly manipulates a valve because the increase in process gain at low flows is offset by increase in process time constant.
- Since it tends to keep plug away from seat, potential corrosion at low flows is reduced.

More information is available in ISA-TR75.25.02-2024, *Control Valve Response Measurement from Step Inputs*. This technical report is a reference for ANSI/ISA-75.25-01, *Test Procedure for Control Valve Response Measurement from Step Inputs*. It describes a control valve's characteristic response to step input signal changes, discusses the factors that affect this response, and discusses the impact of the response on the quality of process control.

Russ Rhinehart's Response

Most of my experience has been in the academic side of understanding phenomena. So, I think that I can share some fundamentals that may help some folks understand the principles and issues that may underlie the best practices that other responses report.

The valve equation for liquids is $F = C_v f(x)\sqrt{\frac{\Delta P_v}{G\xi}}$, where F is the volumetric flow rate through the valve, ΔP_v is the pressure drop across the valve, and G is the fluid specific gravity. C_v is the valve capacity coefficient, the volumetric flow rate at a unity value of ΔP_v when the valve is fully open. x is the valve stem position as a fraction of fully open, $0 \leq x \leq 1$. $f(x)$ is called the valve characteristic, the fraction of flow rate at any stem position, x, to the flow rate when fully open, $x = 1$. The equation is dimensionally consistent because of the factor $\xi = 1$, with the units of that on ΔP_v. Because ξ has a unity value, it is usually omitted. This relation is for turbulent flow through the valve inner orifice, of an incompressible fluid, with no cavitation, degassing, or choking.

If the valve is linear, then $f(x) = x$, and with constant ΔP_v, the flow rate will be linearly proportional to x. If the valve has an equal-percent characteristic, this means that the flow rate makes a relative change proportional to the valve stem position, $\frac{\Delta F}{F} = \alpha x$. With constant ΔP_v and some fun math, this becomes $f(x) = R^{x-1}$, where R is a dimensionless value called valve rangeability. Oddly, a true equal-percent valve cannot shut off. If $x = 0$ then $f(x = 0) = R^{0-1} = \frac{1}{R}$ then $F = C_v \frac{1}{R}\sqrt{\frac{\Delta P_v}{G}} \neq 0$. So, manufacturers provide "modified equal percent" that do actually shut off when $x = 0$. Valve rangeability, R has values from about 15 to 50, and R is part of what you would specify in choosing an equal percent valve.

As nomenclature, $f(x)$ is termed the inherent valve characteristic, the flow characteristic when the valve is in a test bed and ΔP_v is constant. As shortened notation, I'll call the valve linear or equal percent, but really it is the property of the trim (plunger) of the inner orifice of the valve under constant ΔP_v.

If ΔP_v is constant, then $F \propto f(x)$. Figure 1 reveals how flow rate (as a fraction of maximum flow) depends on valve stem position for linear and modified equal percent valves with $R = 15$ and $R = 50$.

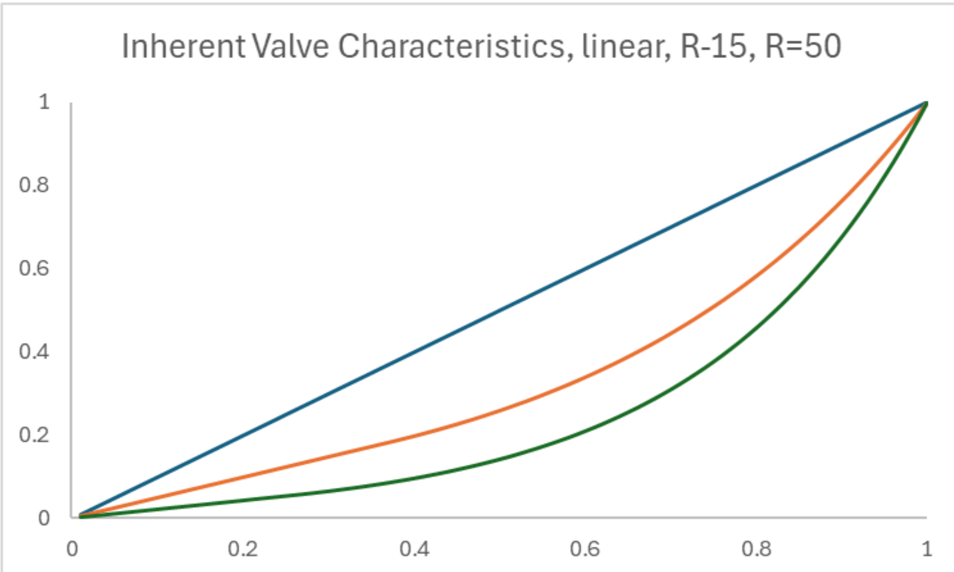

Figure 1. Inherent Valve Characteristics – Linear (upper curve), R=15, R=50 (lower curve)

Process gain is the change in flow rate divided by the change in controller output. If the stem position is proportional to the controller output, $x = CO/100$, then the linear valve would have a constant gain, which would be desirable for controller tuning. Alternately, the equal percent valves have a high gain at high flow rates and a low gain at low flow rates. Why would one want this tuning aggravation?

The answer lies in the fact that when installed in a pipeline, the pressure drop across the valve is not constant. Consider a centrifugal pump and flow control valve in a process line, with a pump-inlet-to-line-outlet pressure and head differential of ΔP_{system}. When the valve is closed, there is no flow, 1) the dead-headed pump has its maximum outlet pressure, and 2) there is no pressure loss in the line. Then the pressure drop across the valve has the maximum pump head. However, when the valve is fully open 1) the pump curve reveals that the head decreases, and 2) the line friction losses ΔP_f are at a maximum. So, the pressure drop across the valve is lower. The ideal friction loss and pump curve relations could be modeled as $\Delta P_f = K \frac{1}{2} \frac{\rho}{g_c} \left(\frac{F}{\pi r^2}\right)^2$, and $\Delta P_p = \frac{H_0 \rho g}{g_c}\left[1 - \left(\frac{F}{F_o}\right)^p\right]$. At steady state the pressure losses balance the pump pressure increase, $0 = \Delta P_p - \Delta P_s - \Delta P_v - \Delta P_f$, and substituting models and solving for how F depends on x,

$$F = C_v f(x) \sqrt{\frac{\frac{H_0 \rho g}{g_c} - \Delta P_s}{G\xi - \left[\frac{H_0 \rho g F^{p-2}}{g_c \quad F_o{}^p} + K \frac{1}{2} \frac{\rho}{g_c}\left(\frac{1}{\pi r^2}\right)^2\right][C_v f(x)]^2}} \qquad (1)$$

If the line losses are small relative to the pressure loss across the valve and the pump maximum capacity, F_o, is high relative to the flow rate, then the bracketed denominator term in Equation (1), is inconsequential, which reveals $F \propto f(x)$, and a linear valve makes $F \propto x = CO/100$, which is desirable. However, if either the line losses are substantial or the F_o is low, then the pressure drop across the valve has a substantial dependency on x, and for a linear valve $F(x)$ will have a quick opening character. Figure 2 illustrates the quick opening character of the installed characteristic with a linear valve. The installed characteristic is different from the inherent characteristic because the other influences in the system cause ΔP_v to change with flow rate. The equal percent installed characteristics somewhat linearize the installed valve responses.

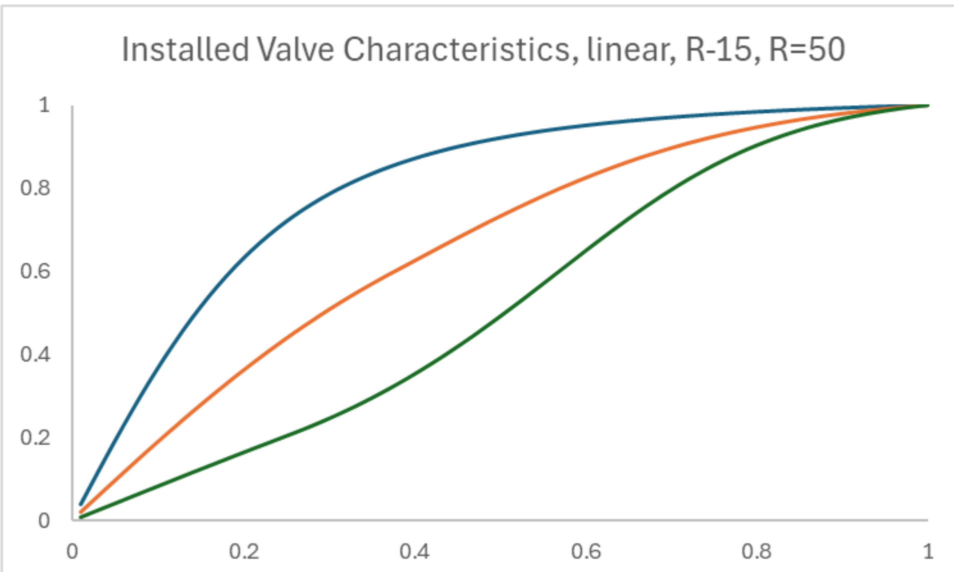

Figure 2. Installed Valve Characteristics – Linear (upper curve), R=15, R=50 (lower curve)

When the pressure drop across the valve has a substantial dependency on x, an equal percent valve will substantially linearize the $F(x)$ relation. For the factors chosen in the Figure 2 illustration (C_v, H_0, F_0, K, ρ, R) the linear valve has a gain ratio of 65 over the full range. For the equal percent valve with R=12 the gain ratio is 6, and for R=50 it is 7.

The reader may have a preference for another of the many alternate models for the pump curve and for the flow losses in a pipeline. Using the common Moody-Darcy squared relation between friction losses and flow rate, $\Delta P_f = f \frac{L_{equ}}{d} \frac{1}{2} \frac{\rho}{g_c} \frac{\dot{Q}^2}{(\pi r^2)^2}$. But, because f is effectively constant in fully rough elements and proportional to $Re^{-.25}$ (the Blasius relation is $f = 0.3164 Re^{-.25}$ for smooth sections), the Hazen-Williams studies reveal a 1.85 power better matches practice. $\Delta P_f = f \frac{L_{equ}}{d} \frac{1}{2} \frac{\rho}{g_c} \frac{\dot{Q}^{1.85}}{(\pi r^2)^{1.85}}$ However, this is dimensionally inconsistent. To fix, include a coefficient C with a value of unity and the units of time/length to the 0.15 power. $\Delta P_f = Cf \frac{L_{equ}}{d} \frac{1}{2} \frac{\rho}{g_c} \frac{\dot{Q}^{1.85}}{(\pi r^2)^{1.85}}$.

Alternately, use the sum of k-factors for the fully rough elements and the Blasius relation for the straight lines. $\Delta P_f = \left(\sum k + 0.3164 Re^{-.25} \frac{L}{d}\right) \frac{1}{2} \frac{\rho}{g_c} \frac{\dot{Q}^2}{(\pi r^2)^2}$. One step more rigorous is to use the Colebrook-White relation for the friction factor in straight pipe runs that accounts for surface roughness. $\frac{1}{\sqrt{f}} = -2log_{10}\left(\frac{\varepsilon}{3.7d} + \frac{2.51}{Re\sqrt{f}}\right)$, then $\Delta P_f = \left(\sum k + f\frac{L}{d}\right) \frac{1}{2} \frac{\rho}{g_c} \frac{\dot{Q}^2}{(\pi r^2)^2}$.

Regardless, the impact is the same: If the changing flow rate has a substantial impact on ΔP_v, an equal percent valve can substantially linearize the installed characteristic. You need to specify the R-value.

The basic models above are for turbulent flow. Check the Reynolds Number to see if flow is turbulent, transition or laminar.

If there is coarse discretization in the valve positioning (perhaps due to D/A or i/P digital bit capacity, or stiction, or lost motion) then the valve stem position will only have a limited number of finite positions. Where the valve gain is low, realizable x increments will lead to small F increments, and the discretization may not be noticeable. But where the valve gain is high, there will be large intervals between possible F values. A more linear gain will permit more equal F intervals throughout the entire range.

Figure 3 illustrates the impact of discretization of realizable stem position values. The vertical lines each represent an x interval of 0.1, which is very large, but selected for visual impact. The horizontal lines reveal the relative flow rate impact for the linear valve. The upper flow rate discretization would not be visible within measurement noise. At lower x-values, however, the x discretization has nearly 30% of full range flow rate impact. This would make flow control for low flow rates nearly impossible. Although the coarse discretization in the figure is quite extreme, the concept remains for smaller Δx values. To not clutter the graph with the impact on $R = 15$ or $R = 50$ equal percent valves, the lines are omitted. However, the improvement in uniformity of ΔF to Δx impact with equal percent valves should be obvious to the reader.

Figure 3 supports the concepts and equations in Greg McMillan's response on the topic of minimum fractional flow coefficients.

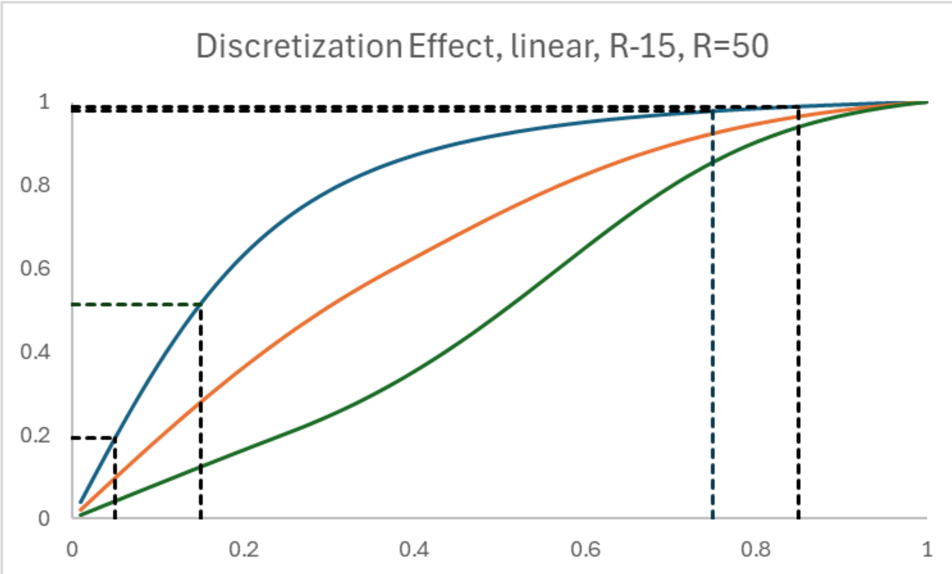

Figure 3. x discretization impact on F

Part of specifying a valve includes the C_v. If the C_v is small, then the pressure drop across the valve will be large, and a linear characteristic could be best. If the C_v is large, then an equal percent characteristic might substantially linearize the response.

One should also consider the impact of valve choices on the size (initial cost) and power (annual expense) of the pump. A large C_v may permit a lower head pump, which would be less expensive initially and also have lower operating expense, but this would lead to an equal percent valve choice.

Pumps and valves come in sizes provided by the manufacturers. One should not invent C_v, R, H_0, or F_0, specifications to optimize linearization and pump expense, but use selections from the manufacturer's device specifications in the analysis to best balance all issues – linearize the $F(x)$ response with minimal costs.

There are many other features to seek in a valve. For instance, if the line pressure has pulses or the valve is operating in the near-closed position, a double-acting trim will minimize chatter or leakage.

Further, the downstream side of the inner orifice, where the fluid velocity is very high, will have the lowest dynamic pressure, which might lead to cavitation or degassing. In such a case, choose an anti-cavitating trim style or place the valve within an expander/contractor assembly to keep the valve exit pressure high. This will change the pressure drop across the valve, which may reshape the best inherent characteristic decision.

There are probably many situations that the system will encounter other than the nominal ones: Piping configuration will change, and erosion, corrosion, sand, blocked screens, or such will cause changes in the pipe friction losses, modeled here as the K factor. Pump impeller wear will change the head-to-flow-rate characteristics. System entrance and exit pressures will likely change over time. Whatever values are used in the analysis (even the manufacturer's specifications), they will likely only approximate reality. So, the analysis should be performed with a variety of possible values of all the variables, and the selection of pump and valve capacity and characteristics that best optimize all relevant issues for possible cases. Michael Taube's response recommends using simulation to explore all such cases.

The analysis outlined here is for steady state. However, it may be important to also consider dynamic issues. For instance, use valve actuator dynamics to model the time to change the valve stem position. If the valve has a quick opening characteristic (see Figure 2), it will take a longer time to make flow rate changes at high flow rates because the stem needs to travel further, and less time to make the same flow rate change at low flow rates. If the time for the valve to move the flow rate to new values is variable and dominates the flow rate dynamics, this may affect the integral time-constant in a controller.

Using $F = ma$ on the fluid in the pipe sections will reveal 1) the lag in flow rate due to fluid inertia, and 2) pressure excursions at various points in the pipeline. For example, a valve that moves rapidly when closing could create overpressure (water hammer) at the valve entrance causing pipe or joint bursts, or very low pressure at the valve exit causing degassing or cavitation. And, if the pipeline is long, the time for the fluid to accelerate to a final value might dominate the dynamics. But if the line is short, the fluid may accelerate very fast, making the valve dynamics control.

The momentum balance on the fluid can be modeled by Newton's Law, $\vec{F} = ma/g_c$. m is the mass of fluid in the pipeline, effectively $m = \pi r^2 L\rho$. a is the average acceleration, $\frac{dv}{dt} = \frac{1}{\pi r^2}\frac{dF}{dt}$, where F is the volumetric flow rate. $\vec{F}$ is the sum of all forces on the fluid. $\vec{F} = \pi r^2\left(\Delta P_s + \Delta P_p - \Delta P_v - \Delta P_f\right)$. Combining, the fluid acceleration is a function of fluid and pipe properties and the pressure changes. $\frac{dF}{dt} = \frac{\pi r^2 g_c}{L\rho}\left(\Delta P_s + \Delta P_p - \Delta P_v - \Delta P_f\right)$

Combining:

$$\frac{dF}{dt} = \frac{\pi r^2 g_c}{L\rho}\left(\Delta P_s + H_0\rho g/g_c\left[1 - \left(\frac{F}{F_0}\right)^p\right] - \left(\frac{F}{C_v f(x)}\right)^2 G\xi \; - K\,\frac{1}{2}\frac{\rho}{g_c}\frac{F^2}{(\pi r^2)^2}\right) \qquad (2)$$

If the valve has a pneumatic actuator, the valve stem position responds to the target value in a first-order-ish manner. Typical τ_{valve} time-constant values are about 1 second.

$$\tau_{valve}\frac{dx}{dt} + x = x_{target} \qquad (3)$$

Figure 4 illustrates pressure excursions (red trace) when the controller output changes (blue trace) over time for a long enough pipeline to make the fluid momentum substantial. At a time of 2 seconds, the valve opens (relatively rapidly compared to the time for the fluid to change speed), creating a low-pressure excursion; and at a time of 6 seconds, the valve closes (relatively rapidly) and the relatively rapid momentum change creates a pressure pulse.

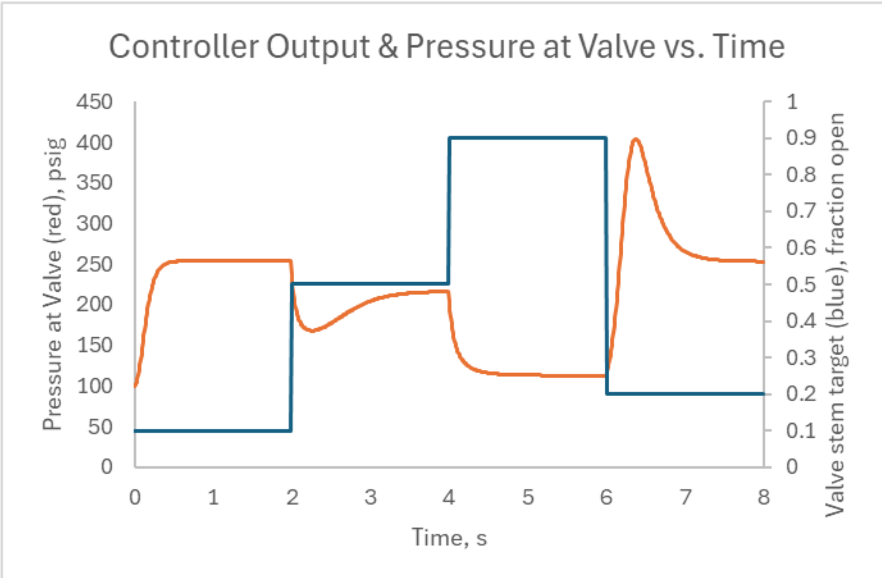

Figure 4. Pressure excursions at the valve entrance (red trace) and valve target position (blue trace) over time for a long pipeline in which fluid momentum is substantial.

Again, simulation permits you to explore such dynamic effects.

Finally, the valve characteristic might not be of dominant importance. If the flow rate time to change is relatively constant, then gain scheduling or output characterizing a PI controller may be fully adequate compensation for nonlinear gains. If both the gain and process time-constant change, a model-based control algorithm may solve the issues that we conventionally seek to solve by using high pressure loss at the valve or nonlinear trim characteristic.

Greg McMillan's Follow-up Thoughts

Not commonly understood is that the increase in process time constant for temperature and composition control offsets the increase in process gain for a decrease in flow to a well-mixed volume in the tuning of a PID controller that is manipulating the main flow to the volume. For this case, the equal percentage characteristic for a constant valve pressure drop that gives a valve flow gain proportional to flow is undesirable. However, most temperature and composition control loops on well-mixed volumes are manipulating a flow loop or secondary loop. Also, while pH is a composition control loop, its process gain is dominated by the titration curve.

Finally, I would alert users to the problem stemming from conventional thinking that valve rangeability is determined based when the deviation of an inherent valve characteristic from theoretical characteristic exceeds some specification and that we do not need to be concerned about the increases in friction of plug, ball, or disc near the closed position particularly for high flow, low cost, low leakage valves, termed "high performance valves". In startup, batch control, pH control, pressure control, and split range control, valves may be operating near the closed position and the loss in installed rangeability from poor resolution and lost motion causes control problems.

I have personally seen rotary valves in the piping spec designed for tight shutoff used in throttling service that would not open till the signal exceeded 8% trying to come off closed position despite a smart positioner because it was being lied to due poor readback from key-lock shaft to stem connection and pinned stem to disc or ball connection. The oscillations in the plant were thought to be due to tuning.

I have seen valves that would not open till the signal exceeded 20% due to bench settings and a positioner being omitted by the supplier using a rule that fast loops should not use positioners. I have seen where a well-designed throttling control valve with a smart positioner had its 0.4% resolution deteriorate to 4% because the actuator stocked with the valve was marginal in size. When I tried to change the specification to include a larger actuator, the delivery increased to 8 weeks.

I have seen where a graphoil and tightened packing has caused resolution to deteriorate to 6% or more. There are many more stories of how bad it can get because valve specifications do not require a valve to actually move in response to a change in signal. Instead, valve specifications emphasize cost and value of tight shutoff and large capacity, leading to on-off valves posing as throttling valves, undersized actuators, graphoil instead of new high temperature low friction packing, rack and pinion and link arm piston actuators, and cheap or poorly tuned positioners.

To get the best valve response, we need throttling valves (globe valves or rotary valves with no seal, splined shaft to stem, and integral cast stem and disc or ball) with low friction packing, preferably diaphragm actuators sized to provide 150% or more maximum thrust requirement, and smart positioners tuned with aggressive proportional and moderated integral action.

The control valve characteristic can be made more linear by a signal characterizer on PID output that provides a piecewise linear fit of the installed flow characteristic that then calculates the X axis (percent valve signal) from the installed flow characteristic Y axis (percent flow capacity). The operator needs to see the resulting output signal after characterization, and there should be an option when the controller is in manual mode for the characterizer to be bypassed so that operations and maintenance can check and test the stroking of the valve.

Also, it would help if users understood that per ISA-TR75.25.02-2024, stick-slip and stiction show up as resolution limit, backlash and shaft windup show up as lost motion, and that deadband can be a combination of the effects of resolution and lost motion. Furthermore, a resolution limit causes a limit cycle for one or more integrators in a control system or process, and lost motion causes a limit cycle for two or more integrators in a control system or process.

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ISA Fellow **Michel Ruel** is a recognized expert in process control and control performance monitoring. Now retired, he led a team that implemented innovative and highly effective control strategies across a wide range of industries, including mining and metals, aerospace, energy, pulp and paper, and petrochemicals. An accomplished author of numerous books and publications, Ruel is also a software designer specializing in instrumentation and process control. He is the founding president of Top Control Inc. and has contributed to projects in multiple countries. In addition, Ruel is a sought-after lecturer for various professional associations.

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ISA Fellow **R. Russell Rhinehart** has experience in both the process industry (13 years) and academe (31 years). He is a fellow of ISA and AIChE, and a *CONTROL* Automation Hall of Fame inductee. He served as president of the American Automatic Control Council and Editor-in-Chief of *ISA Transactions*. Now "retired" Russ is working to disseminate engineering techniques with his website (www.r3eda.com), short courses, books, and monthly articles. His 1968 B.S. in ChE and M.S. in NucE are both from the University of Maryland. His 1985 Ph.D. in ChE is from North Carolina State University.