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Control Valve Modeling and Control
Improving valve response for better process control loop performance

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Biography

Greg retired from Solutia Inc. as a Senior Fellow after 33 years specializing in improving loop performance. Greg contracted as a consultant in DeltaV R&D for 12 years and is presently a Principal Senior Software Engineer in Simulation R&D for Emerson Automation Solutions focusing on dynamic modeling using the Digital Twin. Greg is the author of more than 100 articles and papers and 20 books. Greg's most recent books are *Advances in Reactor Measurement and Control (2015), Process/Industrial Instruments and Controls Handbook 6th Edition (2019), New Directions in Bioprocess Modeling and Control 2nd Edition (2020)*, and *Advanced pH Measurement and Control 4th Edition (2022)*. Greg is an ISA Fellow and received the ISA "Kermit Fischer Environmental" Award for pH control in 1991, the Control Magazine "Engineer of the Year" Award for the Process Industry in 1994 and was inducted with Greg Shinskey into the "Process Automation Hall of Fame" in 2001. Greg was honored as one of InTech's 50 most influential innovators and received the ISA Life Time Achievement Award in 2010.

Abstract

Control valves are a significant source of nonlinearity and oscillations in most control loops. Not understood are the details and consequences of the dynamic response of control valves and how proportional-integral-derivative (PID) controller features and tuning can mitigate the problems. Techniques are presented for how to simply model the many complexities of the dynamic response to account for the diversity of control valves including the effects of different actuator, positioner, connections, packing, seat-seal, and internal closure member designs. Common mistakes in the valve specification and testing are noted that are leading to disastrous results that are occurring with increasing prevalence driven by a lack of understanding and a mistaken desire to increase valve capacity, reduce valve cost, and minimize leakage. Methods for better valve specification and testing are detailed. Figures show the effect of valve to system pressure drop ratio on the installed characteristic. Equations show how the installed flow characteristic and the resolution and lost motion near the closed position determine the actual valve rangeability. Equations are also presented to estimate the limit cycle amplitude and period for a given valve resolution, lost motion, and flow gain and PID controller tuning. Figures are presented of test results to show the effect of valve response on flow and level processes and what can be done with PID tuning and key features most notably external-reset feedback to reduce oscillations. Best practices are presented for control valves and variable frequency drives often mistakenly touted as more linear, fast, and precise.

1 Introduction

Presently, the absence of valve response requirements, the need to fill in a leakage class on valve specification forms, an emphasis on minimizing cost and in some cases, pressure drop, and a perception that excess capacity is good for future capability may lead one to think that valves typically designed for on-off service are a good option for throttling control because of lower cost, tighter shutoff, and lower pressure drop. Often these valves designed for on-off service employ actuators and assemblies including linkages and shaft connections with severe inherent limitations that greatly reduce control loop performance. This annex provides the sources, consequences, fixes, and examples of valve response nonlinearities to understand the ramifications of such a decision and concludes with examples of specifications and tests to help a good throttling valve meet application performance objectives. A broad view of nonlinearities is taken to include anything that changes the valve's response. The knowledge presented is intended to give guidance (including examples of specifications and tests in Table 1) to improve loop performance and should not be taken as requirements. The goal is to make suppliers and users aware of the impact of valve response on loop performance so that better decisions are made as to the offering and selection of throttling control valves. The need for tight shutoff can be met by a separate on-off valve coordinated with the throttling valve.

The response metrics need to be based on the change in effective flow coefficient reflecting the actual movement of the internal closure member. Due to lost motion in positioner readback, actuator shaft to stem, and stem to internal closure member (e.g., ball or disk) connections, the readback of valve position may not be representative of actual closure member position. Consequently, bench tests may need a travel indicator attached to the actual closure member. Since what we are really interested in is the change in effective flow coefficient and that process temperature and pressure can affect resolution and lost motion, response tests done with a precise low noise flow measurement in a pilot or actual plant or flow lab may provide the most representative response metrics.

2 The Perfect Storm

We have the strange situation where the performance of many control valves and control loops has deteriorate as the technologies for automation progressed. If we go back to the 1960s before the advent of electronic controllers, the PID algorithm in pneumatic controllers had an inherent ability to deal with valve response issues. Furthermore, the use of diaphragm actuators, globe valves, and sensitive high gain pneumatic positioners minimized valve performance issues. The lack of understanding of the valve dynamics and the consequences has lead to the following "perfect storm" of mistakes:

1. Volume boosters instead of positioners recommended for fast loops
2. Rangeability statements based on inherent flow characteristic rigor
3. Lack of understanding of loss of rangeability due to installed flow characteristic and resolution (stiction) near the closed position
4. Piston actuators instead of diaphragm actuators recommended
5. Use of so called "high performance" (e.g., rotary on-off valves in piping spec) to minimize cost and leakage and maximize capacity
6. Severe lost motion in rotary valves from positioner to shaft, shaft to stem, and stem to internal closure member (e.g., ball) connections
7. Lack of recognition of limit cycles caused by lost motion when there is 2 or more integrators besides those limit cycles from resolution
8. Positioners with poor sensitivity (spool positioners with 2% resolution) and single stage relay that cause huge dead time for steps < 0.4%
9. Lying smart positioners due to lost motion and shaft windup
10. Lack of understanding of terms per ISA-TR75.25.01 that replace old terms (lost motion for backlash and hysteresis) (resolution for stiction)
11. Use of integral action in positioners to eliminate offsets in open loop tests reducing gain, which increases response time and limit cycles
12. Valve specification sheets that have entries for leakage, packing, and capacity but none for response time, lost motion, and resolution
13. Common practice of tests doing 10% or 25% steps instead 0.2% steps
14. Reluctance of valve companies to do tests below 20% position to show effect of poorer resolution (greater stiction) near seal and seat
15. Lack of sensitive, low noise, and high rangeability flow measurement to see actual flow response that includes lost motion for small steps
16. Actuators marginally sized to lower cost resulting in poorer precision
17. Purchase of "high performance" valve companies by throttling valve companies resulting in their offering to lower cost in bids
18. Loss of external reset feedback capability in most PID algorithms
19. Slow updates of actual valve position by HART and wireless
20. Misconception that variable frequency drives (VFDs) have a faster, more linear, and more precise response (see McGraw Hill Handbook on slide 28 and best practices in paper for VFD issues and solutions)

3 The Step Response Time

The time to 86% of the final valve response (T_{86}) for a step change in signal is critical for many loops. This response time often increases with actuator size and step size due to slewing rate. The response time can greatly increase for small step sizes for many pneumatic positioner and actuator designs, particularly as the signal reverses direction. The dead time part of the response time increases for these positioner designs and systems with significant resolution limits and lost motion often aggravated by higher friction. Constant speed actuators (e.g., electric and electro-hydraulic) may result in a fast T_{86} for small steps and a slower T_{86} for larger steps. The dead time part of the response time is most detrimental especially in terms of the peak error for a load disturbance because a control loop cannot start a correction until the valve starts to respond [1]. The response time is critical for compressor surge control and most pressure control loops. The large T_{86} response time for small signal reversals can cause a limit cycle when the longer T_{86} response time significantly slows the overall control loop step response time. In general, the T_{86} valve step response time should be less than 10% of the desired closed loop time constant for self-regulating processes or arrest time for integrating processes to enable good loop performance. In cases where the valve T_{86} cannot be much faster than the primary process time constant in a self-regulating process, the valve T_{86} is the dominant time constant in the loop and may cause limit cycling.

For pneumatically actuated valves, the portion of the response to a step input change after the dead time for small signal changes is a mixture of small lags set by positioner design and tuning. For large signal changes, there is an additional response time that is the result of a maximum slewing rate set by actuator volume and positioner or volume booster flow coefficients with the exhaust flow coefficient generally larger.

The use of a volume booster on the positioner output (as seen in Figure 1) with booster bypass opened just enough to stop position hunting by enabling the positioner to see part of the actuator volume that is much larger than the booster volume, can make valve response faster without causing oscillations. Volume boosters used instead of positioners mistakenly advocated for fast processes can cause serious unsafe instabilities [1]. Without a positioner to react, a volume booster driven by an I/P output to a diaphragm actuator has resulted in fail open butterfly discs slamming shut due to the booster reacting to flow forces without correction due to positive feedback [1]. A person can actually change a large rotary valve butterfly disk position by simply grasping the actuator shaft and moving the shaft up or down. Boosters can artificially lower the effective pneumatic stiffness because when the valve begins to move changing the pressure in the actuator due to diaphragm flexure, they will exhaust or fill rapidly to keep the pressure where it was but not the valve travel. Volume boosters also have a significant dead band. For larger valves, inadequate or restricted air supply will slow valve response.

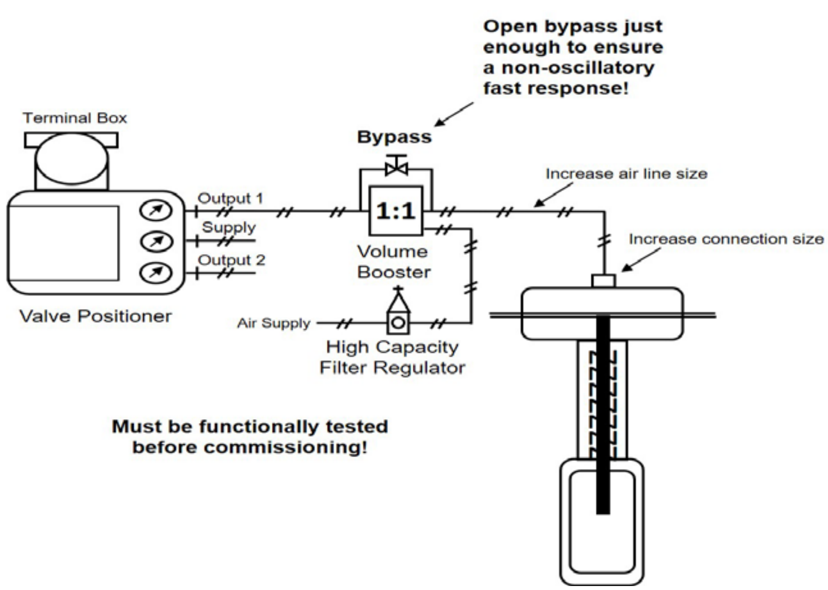

Figure 1. Volume booster on positioner output for diaphragm actuator

4 Dead Time

For pneumatically actuated valves, the valve response dead time for a step change in signal is a combination of pre-stroke dead time and the dead time due to positioner sensitivity limits interacting with friction induced dead band. The pre-stroke dead time depends on actuator volume and fill & exhaust rates, and is only applicable when moving from an end point cutoff. This dead time can be estimated by the (X_v) fill and exhaust factors exemplified for an actuator type and volume that is divided by the corresponding (C_v) flow coefficients. During mid travel reversals, dead band induced from positioner sensitivity and friction is inversely related with step size, and can increase dramatically for small signal changes. Higher friction forces require a larger change in actuator pressure to reverse direction, and thus more dead time. In general, the valve dead time should be less than 10% of the total loop dead time for good loop performance.

There are additional sources of dead time due to gradual changes rather than step changes in controller output. The gradual change can be approximated as a ramp, and the additional dead time can be estimated as the lost motion and resolution divided by the average ramp rate in the controller output. For a reversal in direction of controller output, the additional dead time occurs for the deadband, which is the sum of resolution and lost motion. For steps continuing in the same direction, the additional dead time is the result of resolution. The ramp rate in controller output can originate from integral action in the controller manipulating the valve or from an effective ramp rate in the controlled variable from disturbances that often come from other loops and other final control elements. These disturbances generally exhibit a gradual rather than a step change due to slewing rate of the valve or velocity limit in the variable frequency drive setup and the integral action in the controller creating the disturbance. Disturbances to temperature control loops tend to exhibit a gradual change due to volume, and thermowell and heat transfer lags. The smoothing effect of a well-mixed volume for continuous processes is from a primary process time constant, which is the volume divided by the throughput flow. Hence, the time constant depends upon level and production rate. For batch processes there is also a smoothing effect by an increase in volume via the consequential decrease in integrating process gain. This additional dead time is significant for many types of piston actuators due to friction from internal piston cylinder rust and seals that can get worse with time and can be very large for valves with large amounts of packing and seal or seat friction and lost motion from play in linkages and connections (backlash) and from shaft windup. The dead time is usually greatest near the closed position that is particularly true for valves designed for lower leakage due to higher internal valve friction. A “flow-to-close” globe valve can also delay opening and create instabilities near the closed position due to “bath tub or sink drain” effect where fluid forces suck the plug into the seat potentially causing seat damage and water hammer besides control problems. Procedure automation and state-based control for automation of startups, transitions, and dealing with abnormal operation, and safety instrumented systems can create large sustained step changes in valve signal that would not pose these concerns as to additional dead time and instabilities.

The pre-stroke dead time can be minimized by using a volume booster with a slightly open bypass valve on positioner output and by increasing the size of pneumatic tubing, solenoids, and actuator connections as seen in Figure 1. The additional dead time from resolution can be minimized by diaphragm actuators with sensitive positioners, low friction packing, and “flow-to-open” globe valve. The additional dead time from backlash can be minimized by the use of sliding stem globe valves or in rotary valves with the use of better clamped actuator shaft to stem connections (e.g., splined connection) and zero clearance stems to closure member connection (e.g., integrally cast), and zero clearance drivetrains (e.g., rod end bearings) [1]. The additional dead time from shaft windup can generally be reduced by increasing shaft diameter possibly offered by different valve models or manufacturers to reduce windup. An increase in actuator size to provide 150% of required throttling stiffness (e.g., torque or thrust) can improve the resolution and decrease the shaft windup, which reduces the dead time from resolution and lost motion but slightly increases pre-stroke dead time.

5 Resolution

For pneumatically actuated valves, the stair step response seen in Figure 1 of the Technical Report is often the result of the difference between static and dynamic friction of piston seals, stem packing, and valve seat or seal components, which can be worse due to wear and corrosion. Movement does not start until the force exceeds the static friction. The movement of the internal closure member (e.g., plug, disk, or ball) jumps and does not stop because the dynamic (sliding) friction is less than the static friction. This leads to a stair step response. Clearance between gear teeth of piston actuator rack and pinion connections worsens resolution caused by difference in static and dynamic friction. The hole pattern of a “drilled hole valve cage” can cause resolution issues of the flow coefficient and thus the process response. A non-zero resolution causes a limit cycle if there is one or more integrators anywhere (e.g., PID, positioner, process) [1,4,5]. The limit cycle amplitude for a self-regulating process is the open loop gain multiplied by the resolution. The open loop gain is the product of the valve travel gain, valve flow gain, process gain, and measurement gain. Steep installed flow characteristics, oversized valves, sensitive processes such as pH, and narrow measurement spans can result in extremely large amplitudes in the limit cycle of the process [1].

The limit cycle amplitude (A_o) from resolution is independent of controller tuning and is simply the resolution (R_v) multiplied by the open loop gain (K_o) for a self-regulating process [2]:

$$A_o = R_v * K_o$$

The limit cycle period (T_o) from resolution increases as the PID reset time (T_i) increases and the PID gain (K_c) decreases for self-regulating processes [2]:

$$T_o = 4 * T_i * \{Max[2, 1/(K_o * K_c)] - 1\}$$

Actuators designed for greater throttling stiffness, possibly by much higher operating air pressures in diaphragm actuators, or higher crossover pressure in piston actuators, provide greater thrust and enable their use for larger valves and higher process operating pressures. New packing designs that use modern synthetic products including live-loading are available for most valves and provide low friction and less difference between static and dynamic friction at temperatures once requiring graphite. Seat or seal designs that minimize contact particularly after the closure member starts to open can greatly improve resolution. The use of external-reset feedback (e.g., dynamic reset limit) where the readback of actual valve position is used as an input to the PID filter that provides the positive feedback implementation of integral action as detailed in Reference 1 and ISA-5.9 can stop a limit cycle from resolution on a self-regulating process. However, the readback must be indicative of the actual internal closure member position and be timely and precise [4,5]. The use of integral dead band can stop a limit cycle on a self-regulating process if the integral dead band setting is larger than resolution. Both methods can result in an offset between the process setpoint and process variable in closed loop control which may or may not provide better process control performance that the limit cycle. Each case must be evaluated to determine the benefit of these methods. Limit cycles reduce packing life but may be averaged out by relatively large back mixed process volumes to provide a controlled variable closer to setpoint. It is better, however, to address the root cause of a deterioration in resolution.

6 Lost Motion

Lost motion is the magnitude of the percent offset between the percent position and percent signal input after a reversal of input signal minus the initial offset. Lost motion can be estimated as the dead band minus the resolution. Major sources of lost motion are friction, backlash and shaft windup. Lost motion from friction is proportional to friction forces and inversely proportional to I/P and/or positioner gain. Backlash is often due to play in linkages seen in piston link-arm and scotch yoke actuators and in pinned or keylock shaft to stem and stem to ball or disc connection for rotary valves. Lost motion is also caused by shaft windup in rotary valves when the piston or diaphragm actuator shaft twists before it moves and increases with friction in packing and seat or seal [4,5]. A reversal in signal requires a reversal in twist causing lost motion. Lost motion causes a limit cycle if there are two or more integrators anywhere in the control loop (e.g., PID, positioner, process). This consequence was first documented by Shinskey for an integrating process and a PID controller in Reference 3 and was extrapolated in Reference 1 with test results shown in Reference 5. The limit cycle amplitude depends upon tuning unless external-reset feedback (e.g., dynamic reset limit) is turned on [1][3]. The following equations for backlash dead band may be applicable in terms of relative effects for other sources of lost motion.

The limit cycle amplitude (A_o) from lost motion backlash dead band (DB_v) is inversely proportional to PID gain as shown in the following equation for integrating processes [3]:

$$A_o = DB_v / K_c$$

The limit cycle period (T_o) from lost motion increases as the reset time (T_i) increases and the PID gain decreases as exemplified by the following equation for integrating processes [3]:

$$T_o = 5 * T_i * [1 + (2/K_c^{0.5})]$$

Converting linear actuation to rotary motion using a zero-clearance drivetrain (e.g., lever arms with rod end bearings), clamped actuator to valve stem connections including clamped splined connections, large stem diameters, and zero clearance stem to flow element connections (e.g., stems integrally cast with ball or disk or plug, taper pins) can greatly minimize lost motion in rotary valves. The use of external-reset feedback can reduce the limit cycle amplitude for a PID controller with integral action in an integrating process. For backlash and other sources of lost motion, you can configure the PID to automatically step the PID output when it changes direction and is exceeding a designated noise band. The step size would be the expected lost motion. Normally the step would not be done for PID in manual mode so as not to interfere with response tests. The noise band setting is critical to prevent unnecessary movement of the valve, which can cause excessive wear and upset other loops. Also, a step larger than the lost motion can create a disturbance from excessive motion. Since the lost motion is often a function of operating conditions, an accurate compensating step size is challenging.

7 Installed Flow Characteristic

For installations with a low valve pressure drop to system pressure drop ratio (e.g., < 0.1), inherent flow characteristics develop severely distorted installed flow characteristics. The distortion results in linear inherent flow characteristics approaching a quick-opening flow characteristics with a large flow gain and 50% of maximum flow reached below 20% valve position. The distortion results in an equal percentage inherent flow characteristics having a nearly zero flow gain below 5% valve position. There is a severe loss of linearity for linear inherent flow characteristic and severe loss of installed rangeability for both characteristics [1]. Figures 2 and 3 for linear and equal percentage trims in systems with no appreciable change in static pressure or phases, show how the installed flow characteristic distorts with ratio of the valve pressure drop to system pressure drop providing an alert to misguided attempts to minimize pressure drop not realizing the consequential loss in rangeability [1]. Most valve rangeability statements are erroneous because they do not account for the installed characteristic and effect of lost motion and resolution (worse near seat and seal). In these figures, the pressure drop ratio ΔP_R = 0.0625 corresponds to valve drop at maximum flow being 6.25% of the system drop which is close to 5% drop cited to minimize energy use in attempt to discourage replacement of valves with variable frequency drives (VFDs). Not recognized is that VFDs have their own nonlinearity problems [1].

As noted in flow gain, a signal characterizer can greatly improve loop performance if the installed flow characteristic is well known and constant and the valve is precise and not oversized. However, a characterizer does not change impact of some travel nonlinearities, such as resolution on process limit cycle amplitude. The controller tuning needs to be improved based on better linearity to see all the benefits. The signal characterizer is preferably done in the controller for visibility and maintainability.

8 Installed Rangeability Based on Controllability

Inherent rangeability that is often stated as the maximum valve C_v divided by the minimum valve C_v (the point the inherent flow characteristic exceeds an allowable deviation from theoretical characteristic at low valve position) is susceptible to being much larger than what is actually experienced. For more information see ISA-75.11.01 Inherent Flow Characteristic and Rangeability of Control Valves. A more useful term is installed rangeability, the maximum controllable flow divided by the minimum controllable flow. The minimum controllable flow is the lost motion and resolution dead band that is the corresponding flow on the installed flow characteristic near the closed position [1]. For example, if the dead band is 0.4%, the minimum controllable flow would be the flow from the installed flow characteristic at 0.4% position. The resulting installed rangeability raises awareness as to the consequences of trying to select valves that have large capacity, tighter shutoff and lower price, and appear to use less energy. Rangeability is greatly improved in valves that are more precise and optimally sized with the valve to system pressure drop ratio greater than 0.25 for an equal percentage inherent flow characteristic and a valve to system pressure drop ratio greater than 0.5 for a linear inherent flow characteristic [1].

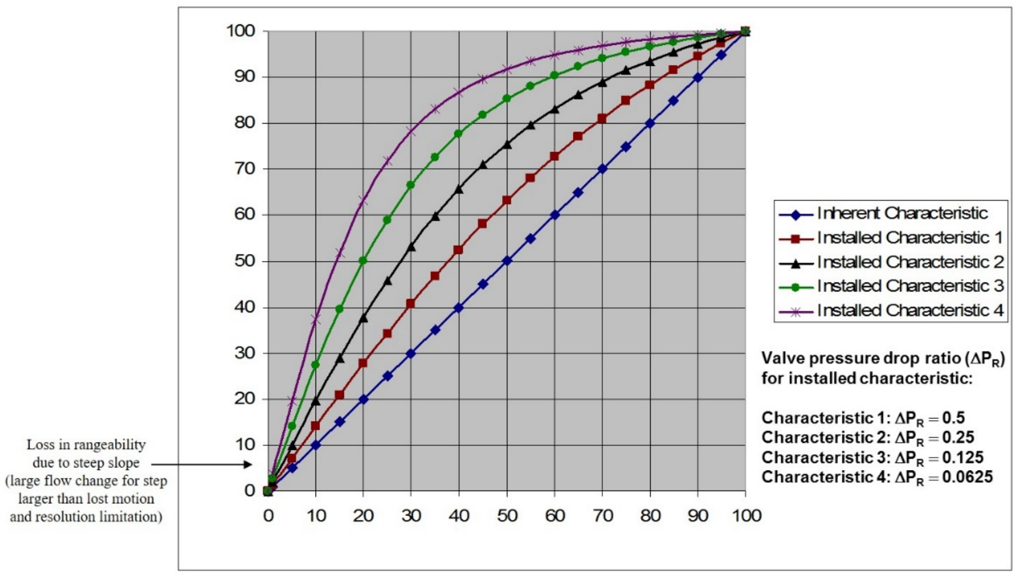

Figure 2. Installed flow characteristic of linear inherent flow characteristic

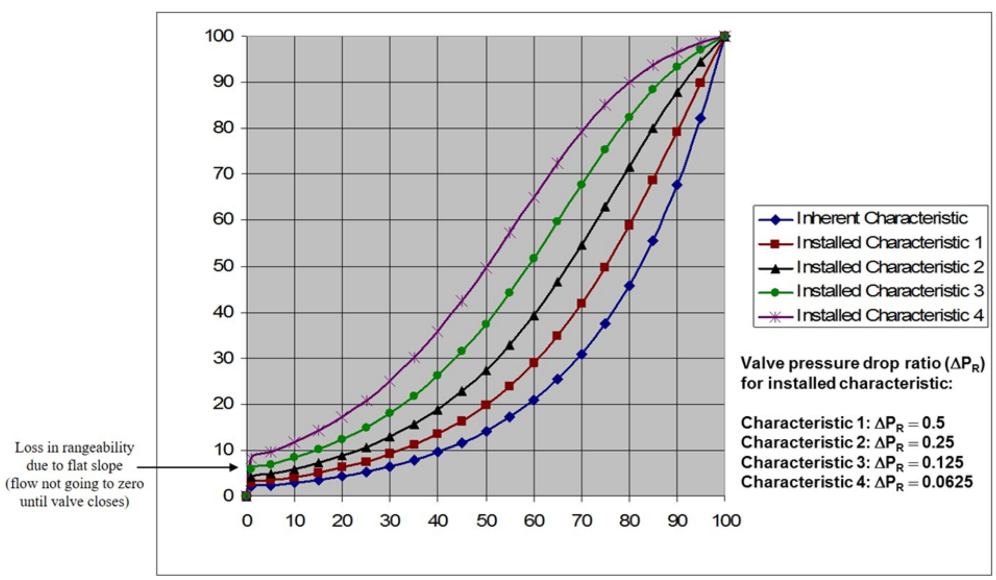

Figure 3. Installed flow characteristic of equal percentage inherent flow characteristic

9 Specifications

The user can address requirements for process efficiency, capacity, quality, and safety by setting allowable error (e.g., maximum permissible deviation from setpoint) for minimum, normal and maximum flows. The limit cycle amplitude and peak error based on loop dynamics and tuning that includes valve response to meet the allowable error can be used as the goals for the various nonlinearities. Test step sizes can be approximated from these goals and signal starting points based on minimum, normal and maximum flows. Closure member position may not move for several percent changes in stem or shaft position in tight shutoff rotary valves when the stem is not rigidly connected with the closure member. These valves need sensitive low noise flow measurements in the field and travel gauges on the closure member for shop tests, to identify resolution and lost motion. Large step sizes and starting points can be approximated based on fastest and largest disturbance and allowable peak errors. Note that fast ramp rates of the valve test signal may not reveal resolution. ANSI.ISA-S75.25.01 defines the test to identify resolution.

Understanding the effect of valve response on the peak error (E_x) for a step load disturbance can provide guidance in the specification of valve response requirements, and an example method is shown below for generating requirements for an existing loop. The peak error, quantified as a fraction of the open loop error (E_o) and process response time, is the error if the loop is in manual and received the step disturbance. This can be estimated from the tuning settings, open loop process gain (K_o), the total loop dead time (θ_o), PID execution rate (Δt_x), and signal filter time (τ_f), by the following equation for a PI controller for a self-regulating process [1]:

$$E_x = \frac{1.5}{K_c * K_o * \left(1.0 + \frac{0.5 * \theta_o}{(T_i + \Delta t_x + \tau_f)}\right) + 1.0} * E_o$$

The use of derivative action can reduce the peak error. The effect can be estimated by decreasing the numerator from 1.5 to 1.25. Also, the use of derivative action may enable a reduction in the integral time.

The worst-case maximum deviation from setpoint can be approximated by summing the limit cycle amplitude and peak error for the largest load disturbance. Peak error is estimated first assuming a linear and instantaneous responding valve, the most aggressive PID tuning based on process dynamics and nonlinearities, and the largest open loop gain. Estimates for valve resolution and lost motion from Table 1 are then used to estimate process limit cycle amplitudes. As a rough approximation, assume 20% of the valves T_{86} response time contributes additional dead time (Δt_x). Then the valve's response time, resolution, and lost motion can be iterated until the worst-case error approaches the allowable error plus some design margin based on application requirements. Tuning software can more accurately account for the effects of dead time and T_{86} response time in determining the worst-case error. Approximating some systems as having simple low order dynamics may not be adequate, and dynamic simulations may help provide the knowledge needed.

Table 1. Examples of Specifications and Tests for Different Loop Performance Objectives

Control Objective	Min Test Position %	Max Resolution %	Max Lost Motion %	*Region 3 Max T_{86} sec	*Region 3 Lower Limit Step Size %	*Region 3 Upper Limit Step Size %	*Region 3 Min, Max Travel Gain	Average Overshoot % of step size	**Min, Max Valve Flow Gain Δ%Flow/ Δ%Travel
Tight	2	0.1	0.1	2	0.25	10	0.8, 1.2	20	0.6, 1.4
Fast	5	0.4	0.4	1	1.0	40	0.8, 1.2	20	0.4, 2.0
Basic	10	1	1	4	2.5	10	0.8, 1.2	20	0.4, 2.0
Loose	20	2	2	8	5	10	0.8, 1.2	20	0.2, 4.0

* The lower limit of Step Response Region 1 is 0 and its upper limit is equal to the maximum dead band which is equal to the lower limit of region 2. The upper limit of region 2 is equal to the lower limit of region 3. The lower limit of region 4 is the upper limit of region 3 and the upper limit of region 4 is equal to 100%.

** Use the flow at maximum 100% open as the span to convert flow from EU to %Flow/%Travel. The valve flow gain is the slope of the installed %flow characteristic at the operating point as exemplified in Figures 2 and 3. Note that flow gain is based on %Travel, not % input signal.

Examples of response specification given in Table 1 are based on a broad classifications of control loop performance objectives. Specification terminology should match 75.25 metrics. The “Tight Control” example often needed for pH systems, particularly with strong acids and bases, has the smallest resolution and lost motion requirements, step size, and minimum test position. The “Fast Control” example, often needed for surge and pressure control, has the fastest response time for large step changes and the largest max step test size. The “Basic Control” example, acceptable for most flow and level control and many temperature applications, has specifications that could be met by most control valves designed for throttling service. The “Loose Control” example is for loops where variability is not important, and there is a desire to minimize valve cost. The minimum and maximum step sizes in Table 1 correspond to region 3. There may be additional requirements associated with region 4. The valve flow gains shown in Table 1 are expressed as the % change in valve’s maximum flow divided by percent change in travel. The range of acceptable valve flow gains is based on the installed flow characteristic and step changes significantly larger than the minimum step size.

10 Best Practices [6]

1. Use sizing software with physical properties for worst case operating conditions
2. Include effect of piping reducer factor on effective flow coefficient
3. Select valve location and type to eliminate or reduce damage from flashing
4. Preferably use a sliding stem valve (size permitting) to minimize backlash and stiction unless crevices and trim causes concerns about erosion, plugging, sanitation, or accumulation of solids particularly monomers that could polymerize and for single port valves install “flow to open” to eliminate bathtub stopper swirling effect
5. If a rotary valve is used, select valve with splined shaft to stem connection, integral cast of stem with ball or disk, and minimal seal friction to minimize backlash and stiction
6. Use Teflon and for higher temperature ranges use Ultra Low Friction (ULF) packing
7. Compute the installed valve flow characteristic for worst case operating conditions
8. Size actuator to deliver more than 150% of the maximum torque or thrust required
9. Select actuator and positioner with threshold sensitivities of 0.1% or better
10. Ensure total valve assembly dead band is less than 0.4% over the entire throttle range
11. Ensure total valve assembly resolution is better than 0.2% over the entire throttle range
12. Choose inherent flow characteristic and valve to system pressure drop ratio that does not cause the product of valve and process gain divided by process time constant to change more than 4:1 over entire process operating point range and flow range
13. Tune positioner aggressively for application without integral action with readback that indicates actual plug, disk or ball travel instead of just actuator shaft movement
14. Use volume boosters on positioner output with booster bypass valve opened enough to assure stability to reduce valve 86% response time for large signal changes
15. Use small (0.2%) as well as large step changes (20%) to test valve 86% response time
16. Use ISA standard and technical report relaxing expectations on travel gain and 86% response time for small and large signal changes, respectively
17. Counterintuitively increase PID gain to reduce oscillation period and/or amplitude from backlash, stiction and from poor actuator or positioner sensitivity
18. Use external reset feedback of accurate and fast valve position readback to stop oscillations from poor valve precision and slow response time
19. Use input and output chokes and isolation transformers to prevent EMI from inverter
20. Use PWM to reduce torque pulsation (cogging) at low speeds
21. Use inverter duty motor with class F insulation and 1.15 service factor and totally enclosed fan cooled (TEFC) motor with a constant speed fan or booster fan or totally enclosed water cooled (TEWC) motor for high temperatures to prevent overheating
22. Use a NEMA Design B instead of Design A motor to prevent a steep torque curve
23. Use bearing insulation or path to ground to reduce bearing damage from Electronic Discharge Machining (EDM) that is worse for the 6-step voltage older drive technology
24. Size pump to prevent operation on the flat part of the pump curve
25. Use a recycle valve to keep the pump discharge pressure well above static head at low flow and a low speed limit to prevent reverse flow for highest destination pressure
26. Use at least 12 bit signal input cards to improve the resolution limit to 0.05% or better

27. Use drive and motor with a generous amount of torque for the application so that speed rate-of-change limits in the VFD setup do not prevent changes in speed being fast enough to compensate for the fastest possible disturbance
28. Minimize dead band introduced into the drive setup, causing delay and limit cycling
29. For tachometer control, use magnetic or optical pickup with enough pulses per shaft revolution to meet the speed resolution requirement
30. For tachometer control, keep speed control in the VFD to prevent cascade rule violation where the secondary speed loop is not 5 times faster than the primary process loop
31. To increase rangeability to 80:1, use fast cascade control of speed to torque in the VFD to provide closed loop slip control as detailed in Resource 9
32. Use external reset feedback of accurate and fast speed readback to stop oscillations from poor VFD resolution and excessive dead band and rate limiting
33. Use foil braided shield and armored cable for VFD output spaced at least one foot from signal wires with never any crossing of signal wires, ideally via separate cable trays

11 References

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