ANSI/ISA-TR75.25.02 Annex A - Valve Response and Control Loop Performance
- Sources, Consequences, Fixes, and Specifications

A.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 A.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.

A.2 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 an 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 if it is not consistent.

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 flow coefficients with the exhaust rate coefficient generally larger.

The use of a volume booster on the positioner output (as seen in Figure A.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 as seen in the equations in Section A.11. A safety factor of 1.4 is used for sizing the air regulator to ensure adequate air supply, despite degradation over time. A local air storage tank may be required.

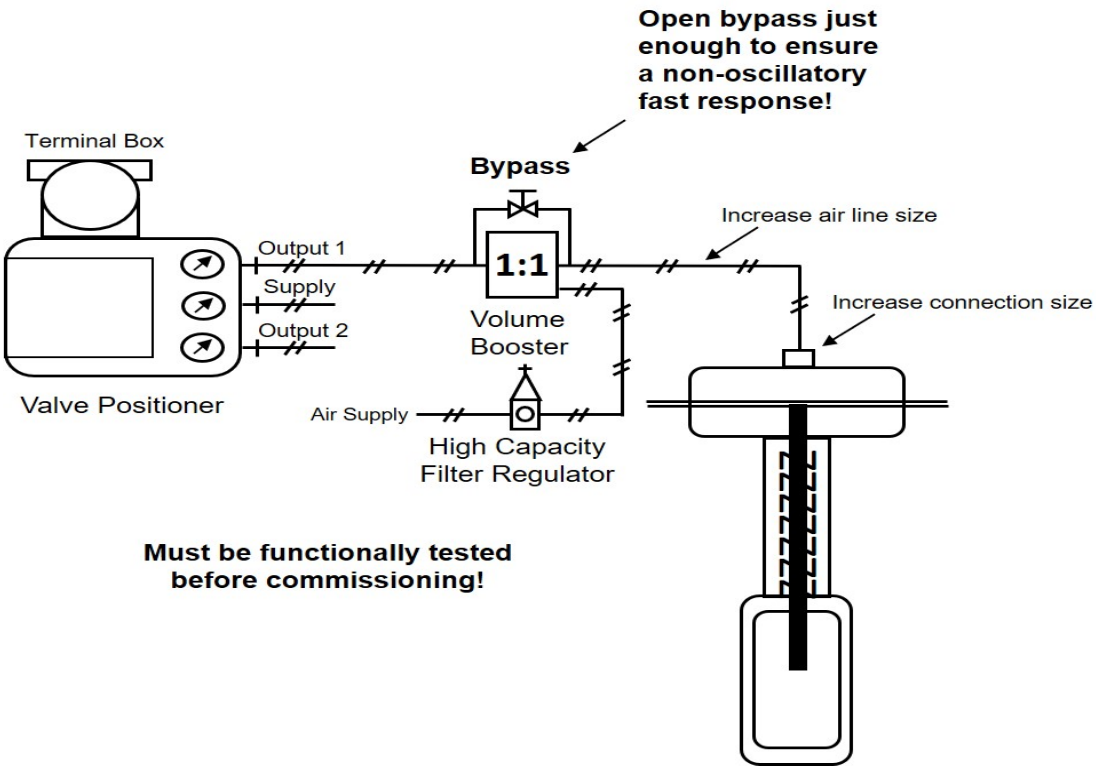

Figure A.1 – Volume booster on positioner output for diaphragm actuator

A.3 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 in Tables A.2a and A.2b for an actuator type and volume that is divided by the corresponding (C_v) flow coefficients exemplified in Table A.3. 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 A.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.

A.4 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.

A.5 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.

A.6 Hysteretic Error

Hysteretic error is the width of the bowing of the curve in the plot of position versus signal for an increasing and decreasing signal over the travel range. It does not include any dead band and is thus independent of resolution and lost motion. The hysteretic error is generally quite small compared to dead band [1]. A common source is the flexure of an actuator diaphragm and spring. The travel gain would be slightly higher for high valve positions for an increasing signal and low valve positions for a decreasing signal.

Hysteretic error can be minimized by higher throttling stiffness of diaphragm actuators and piston actuators that result in a dynamically stiffer system. Gain scheduling could potentially help but is not justifiable because there are other much larger nonlinearities and uncertainties.

A.7 Valve Travel Gain

Valve travel gain is the final change in closure member position divided by the step change in signal both expressed in percent of full scale. If the positioner has a characterization of the input signal, then the change in output of the characterizer is used as the change in input signal. Valve travel gain is particularly affected by resolution and lost motion for signal changes slightly larger than the resolution or lost motion since the change in closure member position is reduced. The use of integral action in the positioner may improve travel gain but the ability to help the PID reject fast load disturbances is reduced from the need to decrease the positioner gain and set an integral dead band to reduce limit cycles from resolution. The positioner can be thought of as a secondary loop where fast immediate response to demands of primary loop is most important. Errors in the primary loop from offsets in the secondary loop can be quickly eliminated by feedback control correction of positioner signal [1].

Travel gain is best improved by making the valve more precise (better resolution and less lost motion).

A.8 Flow Gain

The flow gain (product of travel gain and valve flow gain) contribution to the open loop gain is the final change in flow in engineering units divided by the step change in percent signal. It is affected by travel gain, input characterization in the positioner and installed flow characteristic. The result is often a severe nonlinearity especially for small signal changes due to travel gain nonlinearity and for a low valve pressure drop to system pressure drop ratio due to installed flow characteristic nonlinearity seen in Figures A.2 [1].

The flow gain nonlinearity can be reduced by a more precise valve (better resolution and less lost motion), minimal excess capacity, and a more linear installed flow characteristic. Given a precise and properly sized valve and a well-known and constant installed flow characteristic, signal characterization can greatly reduce the flow gain nonlinearity. The PID gain can then generally be increased since it is no longer set to deal with the steepest slope (highest gain) of the installed flow characteristic. The higher PID gain can decrease the dead time from resolution and lost motion by increasing the rate of change of the PID output signal. The increase in the change in signal on the flatter portions of the installed flow characteristic also helps to reduce the dead time from resolution and lost motion by the magnification of the change in signal by the signal characterizer. Furthermore, the identification of open loop gain depends less on step size due to less local changes in flow gain making tuning more accurate [1].

A.9 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 A.2 and A.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].

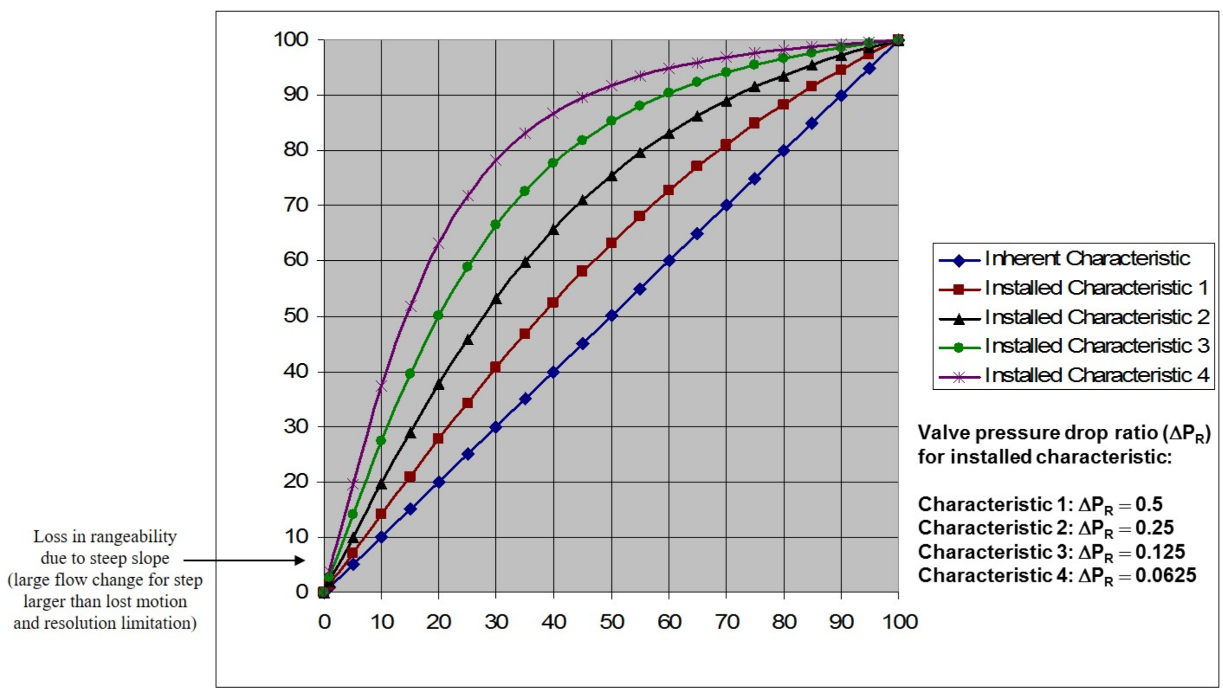

Figure A.2 - Installed flow Characteristic of Linear Control Valve

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.

A.10 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 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% valve 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].

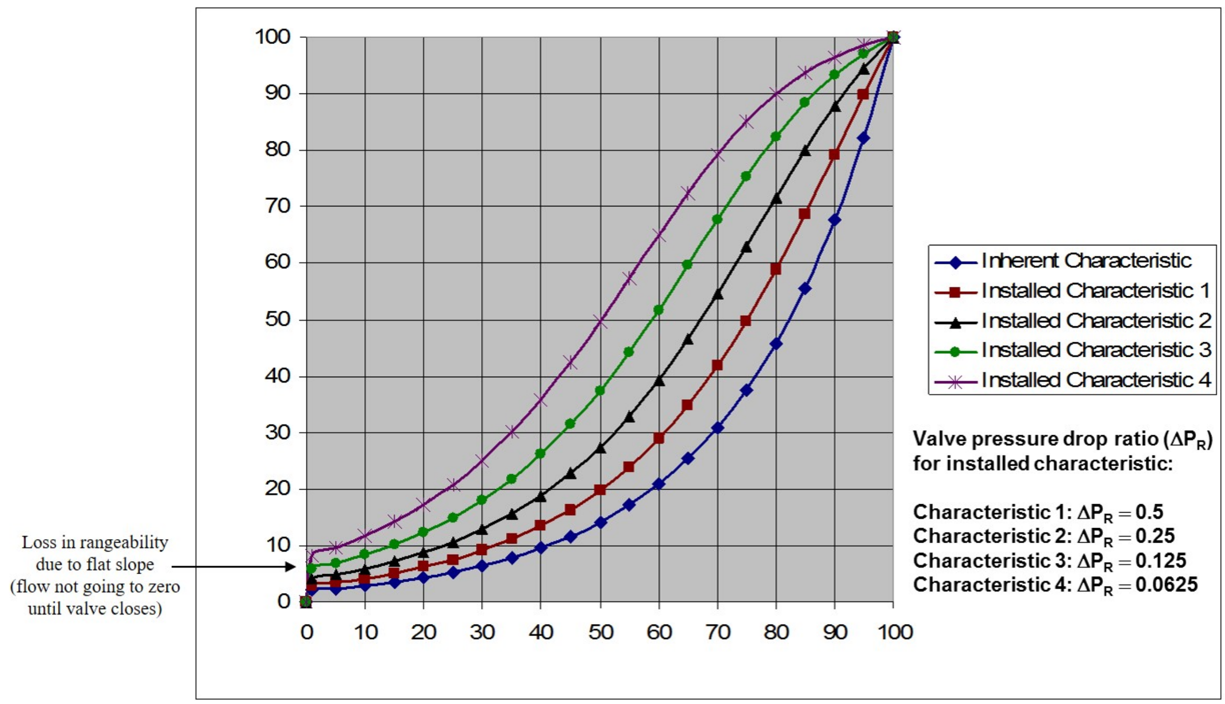

Figure A.3 - Installed flow Characteristic of Equal Percentage Control Valve

A.11 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 A.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 A.1 Examples of Specifications and Tests for Different Control 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 A.2 and A.3. Note that flow gain is based on %Travel, not % input signal.

Examples of response specification given in Table A.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 A.1 correspond to region 3. There may be additional requirements associated with region 4. The valve flow gains shown in Table A.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.

The following equations can be used to estimate pre-stroke dead time (θ_v) and full-scale stroke time from 0 to 100% or from 100% to 0% (T_v) [1]. The (X_v and Y_v) fill and exhaust factors exemplified in Tables A.2a and A.2b depend upon actuator type and volume are divided by the corresponding fill and exhaust flow coefficients (C_v) that depend upon the positioner or volume booster exemplified in Table A.3. The effect of restrictors or solenoid valves (C_{v2}) can be included in a combined (C_{vc}) flow coefficient. To include the effect of additional restrictions such as air tubing size and actuator connection, the combined flow coefficient can be computed for a successive series of pairings of flow coefficients. For example, to include air tubing size, its effective flow coefficient would be combined with solenoid valve. The result would then be combined with the effective flow coefficient of the actuator connection. This final combined flow coefficient is combined with the flow coefficient for the positioner or booster. If there is no positioner or booster, the flow coefficient of the I/P is used. Since the (C_v) for fill and exhaust values can be quite different, particularly for boosters, the response time changes with direction of signal change.

For large step changes (e.g.,10% or more) stroking time dominates the T_{86} response time and the response time can be estimated by multiplication of the 100% stroking time by a factor that is the step size divided by 100%. This estimation is particularly useful for compressor surge control, startups, setpoint response in batch operations, and temperature controllers with large PID gain and derivative settings. These estimates are provided for process control guidelines and may not be adequate for fine control or specifying valve performance.

$$\theta_v = \frac{X_v}{C_v}$$

$$T_v = \frac{Y_v}{C_v}$$

$$C_{vc} = \sqrt{\frac{C_{v1}^2 * C_{v2}^2}{C_{v1}^2 + C_{v2}^2}}$$

Table A.2a1 Examples of Diaphragm Actuator Pre-stroke Dead Time and Stroking Time Factors

USC (United States Customary Units) [1]

Actuator Area	Travel	Supply Pressure	Bench Range	Spring Rate	Fill Factors X_v	Fill Factors Y_v	Exhaust Factors X_v	Exhaust Factors Y_v
sq. in.	inches	psig	psig	lb/in				
69[1]	1	35	6 - 30	1656	0.035	0.294	0.027	0.282
105[3]	2	35	6 - 30	1260	0.055	0.301	0.47	0.313
215[3]	3	35	14 - 30	1760	0.080	0.338	0.066	0.320
46[2]	3/4	20	3 - 15	735	0.012	0.19	0.045	0.256
46[2]	3/4	35	6 - 30	1470	0.016	0.226	0.031	0.29
69[2]	3/4	35	6 - 30	2210	0.028	0.355	0.048	0.457
105[2]	3/4	20	3 - 15	1670	0.033	0.466	0.115	0.63
105[2]	3/4	35	6 - 30	3320	0.046	0.574	0.078	0.727
156[2]	3/4	20	3 - 15	2500	0.046	0.676	0.161	0.913
156[2]	3/4	35	6 - 30	5000	0.065	0.811	0.11	1.046
220[4]	2	20	3 - 15	1260	0.074	2.004	0.5	2.81
220[4]	2	35	6 - 30	2520	0.104	2.243	0.39	3.038
273[3]	3	20	3 - 15	1092	0.048	0.302	0.044	0.297
273[3]	3	35	6 - 30	2184	0.111	0.371	0.086	0.341
450[4]	2	35	6 - 26	4500	0.323	4.386	1.26	6.552
450[4]	2	35	6 - 26	4500	0.380	4.586	1.353	6.87
38[1]	0.75	45	3 - 15	571	0.037	0.250	0.036	0.247
38[1]	0.75	45	7 - 28	999	0.042	0.256	0.036	0.245
38[1]	0.75	45	15 - 35	1999	0.052	0.267	0.036	0.245
38[2]	0.75	45	3 - 15	571	0.037	0.250	0.036	0.247
38[2]	0.75	50	7 - 28	999	0.049	0.266	0.043	0.254
38[2]	0.75	50	15 - 35	999	0.060	0.278	0.043	0.254
77[1]	0.75	45	3 - 15	1142	0.032	0.244	0.030	0.240
77[1]	0.75	45	7 - 28	1999	0.037	0.249	0.030	0.238
77[1]	0.75	45	15 - 35	1999	0.046	0.260	0.030	0.238
77[2]	1.5	45	3 - 15	571	0.060	0.271	0.056	0.264
77[2]	1.5	45	7 - 28	999	0.070	0.281	0.054	0.261

77[2]	1.5	45	15 - 35	999	0.087	0.301	0.054	0.261
108[3]	1.5	45	3 - 15	799	0.057	0.267	0.053	0.261
108[3]	1.5	45	7 - 28	1399	0.066	0.277	0.051	0.257
108[3]	1.5	45	15 - 35	1399	0.083	0.295	0.050	0.257
108[4]	2.5	45	3 - 15	533	0.084	0.294	0.077	0.285
108[4]	2.5	45	7 - 28	933	0.098	0.309	0.074	0.280
108[4]	2.5	70	22 - 55	1532	0.271	0.497	0.152	0.365
232[3]	3	45	3 - 15	857	0.109	0.319	0.100	0.307
232[3]	3	45	6 - 29	1713	0.121	0.331	0.092	0.297
232[3]	3	45	11 - 20	696	0.144	0.355	0.100	0.308
465[3]	3	45	3 - 15	1713	0.109	0.318	0.099	0.306
465[7]	3	45	3 - 15	1713	0.035	0.239	0.039	0.242
465[7]	3	45	6 - 29	3426	0.039	0.243	0.037	0.239
465[7]	4	50	17 - 38	2398	0.080	0.284	0.056	0.259

X_v=actuator factor for pre-stroke dead time (seconds * scfm per $psi^{0.5}$)
Y_v=actuator factor for stroking time (seconds * scfm per $psi^{0.5}$)
1. Air To Open, Positioner, 1/4" Instrument Tubing
2. Air To Close, Positioner, 1/4" Instrument Tubing
3. Air To Open, Positioner, 3/8" Instrument Tubing
4. Air To Close, Positioner, 3/8" Instrument Tubing
5. Air To Open, Positioner, 1/2" Volume Booster with 1/2" Instrument Tubing
6. Air To Close, Positioner, 1/2" Volume Booster with 1/2" Instrument Tubing
7. Air To Open, Positioner, 3/4" Volume Booster with 3/4" Instrument Tubing
8. Air To Close, Positioner, 3/4" Volume Booster with 3/4" Instrument Tubing

Table A.2a2 Examples of Diaphragm Actuator Pre-stroke Dead Time and Stroking Time Factors

SI (International System of Units) [1]

Actuator Area	**Travel**	**Supply Pressure**	**Bench Range**	**Spring Rate**	**Fill Factors X_v**	**Fill Factors Y_v**	**Exhaust Factors X_v**	**Exhaust Factors Y_v**
cm^2	mm	bar	bar	N/mm				
445[1]	25	2.4	0.41 – 2.0	7366	0.035	0.294	0.027	0.282
677[3]	50	2.4	0.41 – 2.0	5604	0.055	0.301	0.47	0.313
1387[3]	76	2.4	0.96 – 2.0	7829	0.080	0.338	0.066	0.320
1006[2]	19	1.4	0.20 - 1.0	3269	0.012	0.19	0.045	0.256
1006[2]	19	2.4	0.41 - 2.0	6539	0.016	0.226	0.031	0.29

445[2]	19	2.4	0.41 - 2.0	9831	0.028	0.355	0.048	0.457
677[2]	19	1.4	0.20 - 1.0	7429	0.033	0.466	0.115	0.63
677[2]	19	2.4	0.41 - 2.0	14768	0.046	0.574	0.078	0.727
1006[2]	19	2.4	0.20 - 1.0	11121	0.046	0.676	0.161	0.913
1006[2]	19	2.4	0.41 - 2.0	22241	0.065	0.811	0.11	1.046
1419[4]	51	1.4	0.20 - 1.0	5605	0.074	2.004	0.5	2.81
1419[4]	51	2.4	0.41 - 2.0	11210	0.104	2.243	0.39	3.038
1761[3]	76	1.4	0.20 – 1.0	4857	0.048	0.302	0.044	0.297
1761[3]	76	2.4	0.41 – 2.0	9714	0.111	0.371	0.086	0.341
2902[3]	51	2.4	0.41 - 1.8	20017	0.323	4.386	1.26	6.552
2902[3]	51	2.4	0.41 - 1.8	20017	0.38	4.586	1.353	6.87
250[1]	19	3.1	0.2 - 1.0	2540	0.037	0.250	0.036	0.247
250[1]	19	3.1	0.5 - 1.9	4444	0.042	0.256	0.036	0.245
250[1]	19	3.1	1.0 - 2.4	8892	0.052	0.267	0.036	0.245
250[2]	19	3.1	0.2 - 1.0	2540	0.037	0.250	0.036	0.247
250[2]	19	3.1	0.5 - 1.9	4444	0.049	0.266	0.043	0.254
250[2]	19	3.1	1.0 - 2.4	4444	0.060	0.278	0.043	0.254
500[1]	19	3.1	0.2 - 1.0	5080	0.032	0.244	0.030	0.240
500[1]	19	3.1	0.5 - 1.9	8892	0.037	0.249	0.030	0.238
500[1]	19	3.1	1.0 - 2.4	8892	0.046	0.260	0.030	0.238
500[2]	38	3.1	0.2 - 1.0	2540	0.060	0.271	0.056	0.264
500[2]	38	3.1	0.5 - 1.9	4444	0.070	0.281	0.054	0.261
500[2]	38	3.1	1.0 - 2.4	4444	0.087	0.301	0.054	0.261
700[3]	38	3.1	0.2 - 1.0	3554	0.057	0.267	0.053	0.261
700[3]	38	3.1	0.5 - 1.9	6223	0.066	0.277	0.051	0.257
700[3]	38	3.1	1.0 - 2.4	6223	0.083	0.295	0.050	0.257
700[3]	64	3.1	0.2 - 1.0	2371	0.084	0.294	0.077	0.285
700[3]	64	3.1	0.5 - 1.9	4150	0.098	0.309	0.074	0.280
700[3]	64	4.8	1.5 - 3.8	6815	0.271	0.497	0.152	0.365
1500[3]	76	3.1	0.2 - 1.0	3812	0.109	0.319	0.100	0.307
1500[3]	76	3.1	0.4 - 2.0	7620	0.121	0.331	0.092	0.297
1500[3]	76	3.1	0.8 - 1.4	3096	0.144	0.355	0.100	0.308
3000[3]	76	3.1	0.2 - 1.0	7620	0.109	0.318	0.099	0.306
3000[3]	76	3.1	0.2 - 1.0	7620	0.035	0.239	0.039	0.242
3000[3]	76	3.1	0.4 - 1.0	15240	0.039	0.243	0.037	0.239
3000[7]	102	3.4	1.2 - 2.6	10667	0.080	0.284	0.056	0.259

X_v=actuator factor for pre-stroke dead time (seconds * Nm^3/m per $bar^{0.5}$)
Y_v=actuator factor for stroking time (seconds * Nm^3/m per $bar^{0.5}$)
1. Air To Open, Positioner, 6mm Instrument Tubing
2. Air To Close, Positioner, 6mm Instrument Tubing
3. Air To Open, Positioner, 10mm Instrument Tubing
4. Air To Close, Positioner, 10mm Instrument Tubing
5. Air To Open, Positioner, 12mm Volume Booster with 12mm Instrument Tubing
6. Air To Close, Positioner, 12mm Volume Booster with 12mm Instrument Tubing
7. Air To Open, Positioner, 19mm Volume Booster with 18mm Instrument Tubing
8. Air To Close, Positioner, 19mm Volume Booster with 18mm Instrument Tubing

Table A.2b1 Examples of Piston Actuator Pre-stroke Dead Time and Stroking Time Factors

USC (United States Customary Units) [1]

Actuator Area	Travel	Supply Pressure	Bench Range	Spring Rate	Fill Factors	Fill Factors	Exhaust Factors	Exhaust Factors
sq. in.	inches	psig	lb/in	lb/in	X_v	Y_v	X_v	Y_v
17[1]	3/4	60	-	-	0.085	0.05	0.024	0.05
28[1]	3/4	60	-	-	0.165	0.086	0.035	0.086
56[1]	3/4	60	-	-	0.296	0.169	0.05	0.169
89[1]	2	60	-	-	0.715	0.719	0.196	0.719
131[1]	2	60	-	-	0.995	1.06	0.272	1.06
222[1]	2	60	-	-	1.73	1.8	0.738	1.8
17[1]	4	60	-	-	0.02	0.278	0.024	0.278
28[1]	4	60	-	-	0.051	0.46	0.035	0.46
56[1]	4	60	-	-	0.099	0.901	0.05	0.901
89[1]	4	60	-	-	0.181	1.453	0.196	1.453
131[1]	4	60	-	-	0.227	2.144	0.272	2.144
222[1]	4	60	-	-	0.603	3.6	0.738	3.6
25[1]	1.5	60	7 - 20	189	0.120	0.392	0.104	0.362
25[1]	1.5	60	11 - 29	267	0.132	0.406	0.105	0.359
25[1]	1.5	60	18 - 48	456	0.154	0.434	0.102	0.347
25[1]	1.5	60	-	-	0.100	0.367	0.112	0.366
50[1]	3	60	3 - 14	164	0.183	0.441	0.172	0.420
50[1]	3	60	7 - 26	287	0.206	0.467	0.169	0.411
50[1]	3	60	11 - 41	447	0.228	0.493	0.158	0.393
50[1]	3	60	-	-	0.169	0.426	0.178	0.432
100[3]	4	60	4 - 17	300	0.236	0.491	0.212	0.455

100[3]	4	60	12 - 37	561	0.291	0.551	0.200	0.435
100[3]	4	70	19 - 60	911	0.433	0.719	0.233	0.470
100[3]	6	60	-	-	0.212	0.565	0.220	0.571
200[3]	4	60	2 - 9	300	0.220	0.473	0.210	0.457
200[3]	4	60	6 - 18	561	0.246	0.502	0.209	0.453
200[3]	4	60	10 - 30	911	0.272	0.530	0.203	0.442
200[2]	6	60	-	-	0.310	0.561	0.321	0.572
300[3]	4	60	4 - 12	561	0.230	0.485	0.238	0.498
300[7]	4	60	4 - 12	561	0.076	0.294	0.073	0.289
300[4]	4	60	4 - 12	561	0.225	0.479	0.213	0.461
300[7]	4	60	6 - 18	911	0.083	0.203	0.071	0.286
300[8]	4	60	6 - 18	911	0.082	0.300	0.073	0.288
300[7]	6	60	-	-	0.102	0.320	0.109	0.331
300[8]	6	60	-	-	0.104	0.322	0.100	0.317

X_v=actuator factor for pre-stroke dead time (seconds * scfm per psi$^{0.5}$)
Y_v=actuator factor for stroking time (seconds * scfm per psi$^{0.5}$)
1. Air To Open, Positioner, 1/4" Instrument Tubing
2. Air To Close, Positioner, 1/4" Instrument Tubing
3. Air To Open, Positioner, 3/8" Instrument Tubing
4. Air To Close, Positioner, 3/8" Instrument Tubing
5. Air To Open, Positioner, 1/2" Volume Booster with 1/2" Instrument Tubing
6. Air To Close, Positioner, 1/2" Volume Booster with 1/2" Instrument Tubing
7. Air To Open, Positioner, 3/4" Volume Booster with 3/4" Instrument Tubing
8. Air To Close, Positioner, 3/4" Volume Booster with 3/4" Instrument Tubing

Table A.2b2 Examples of Piston Actuator Pre-stroke Dead Time and Stroking Time Factors

SI (International System of Units) [1]

Actuator Area	Travel	Supply Pressure	Bench Range	Spring Rate	Fill Factors	Fill Factors	Exhaust Factors	Exhaust Factors
cm^2	**mm**	**bar**	**bar**	**N/mm**	**X_v**	**Y_v**	**X_v**	**Y_v**
110[1]	19	4.2	-	-	0.085	0.05	0.024	0.05
181[1]	19	4.2	-	-	0.165	0.086	0.035	0.086
361[1]	19	4.2	-	-	0.296	0.169	0.05	0.169
574[1]	51	4.2	-	-	0.715	0.719	0.196	0.719
845[1]	51	4.2	-	-	0.995	1.06	0.272	1.06
1432[1]	51	4.2	-	-	1.73	1.8	0.738	1.8

110[1]	102	4.2	-	-	0.02	0.278	0.024	0.278
181[1]	102	4.2	-	-	0.051	0.46	0.035	0.46
361[1]	102	4.2	-	-	0.099	0.901	0.05	0.901
574[1]	102	4.2	-	-	0.181	1.453	0.196	1.453
845[1]	102	4.2	-	-	0.227	2.144	0.272	2.144
1432[1]	102	4.2	-	-	0.603	3.6	0.738	3.6
161[1]	38	4.14	0.5 - 1.4	841	0.120	0.392	0.104	0.362
161[1]	38	4.14	0.8 - 2.0	1188	0.132	0.406	0.105	0.359
161[1]	38	4.14	1.2 - 3.3	2028	0.154	0.434	0.102	0.347
161[1]	38	4.14	-	-	0.100	0.367	0.112	0.366
323[3]	76	4.14	0.2 - 1.0	730	0.183	0.441	0.172	0.420
323[3]	76	4.14	0.5 - 1.8	1277	0.206	0.467	0.169	0.411
323[3]	76	4.14	0.8 - 2.9	1988	0.228	0.493	0.158	0.393
323[3]	76	4.14	-	-	0.169	0.426	0.178	0.432
645[3]	102	4.14	0.3 - 1.2	1334	0.236	0.491	0.212	0.455
645[3]	102	4.14	0.8 - 2.6	2495	0.291	0.551	0.200	0.435
645[3]	102	4.83	1.3 - 4.1	4052	0.433	0.719	0.233	0.470
645[3]	102	4.14	-	-	0.212	0.565	0.220	0.571
1290[3]	102	4.14	0.1 - 0.6	1334	0.220	0.473	0.210	0.457
1290[3]	102	4.14	0.4 - 1.2	2495	0.246	0.502	0.209	0.453
1290[3]	102	4.14	0.7 - 2.1	4052	0.272	0.530	0.203	0.442
1290[3]	102	4.14	-	-	0.310	0.561	0.321	0.572
1935[3]	102	4.14	0.3 - 0.8	2495	0.230	0.485	0.238	0.498
1935[7]	102	4.14	0.3 - 0.8	2495	0.076	0.294	0.073	0.289
1935[4]	102	4.14	0.3 - 0.8	2495	0.225	0.479	0.213	0.461
1935[7]	102	4.14	0.41 - 1.2	4052	0.083	0.203	0.071	0.286
1935[8]	102	4.14	0.41- 1.2	4052	0.082	0.300	0.073	0.288
1935[7]	152	4.14	-	-	0.102	0.320	0.109	0.331
1935[8]	152	4.14	-	-	0.104	0.322	0.100	0.317

X_v=actuator factor for pre-stroke dead time (seconds * Nm^3/m per $bar^{0.5}$)
Y_v=actuator factor for stroking time (seconds * Nm^3/m per $bar^{0.5}$)
1. Air To Open, Positioner, 6mm Instrument Tubing
2. Air To Close, Positioner, 6mm Instrument Tubing
3. Air To Open, Positioner, 10mm Instrument Tubing
4. Air To Close, Positioner, 10mm Instrument Tubing
5. Air To Open, Positioner, 12mm Volume Booster with 12mm Instrument Tubing

6. Air To Close, Positioner, 12mm Volume Booster with 12mm Instrument Tubing
7. Air To Open, Positioner, 19mm Volume Booster with 18mm Instrument Tubing
8. Air To Close, Positioner, 19mm Volume Booster with 18mm Instrument Tubing

Table A.3 Examples of Flow Coefficients of Accessories [1]

Accessory Type	Connection Sizes, inches	Supply C_v	Exhaust C_v
Positioner	-	**0.37**	**0.31**
I/P Transducer	-	**0.39**	**0.36**
Volume Booster	**3/8 and 3/8 ports**	**3.74**	**2.29**
Volume Booster	**3/8 and 1/2 ports**	**3.74**	**2.52**
Volume Booster	**1/2 and 3/8 ports**	**5.32**	**2.3**
Volume Booster	**1/2 and 1/2 ports**	**5.32**	**2.53**
Positioner, Smart HART	-	**0.482**	**0.482**
Positioner, Smart HART	-	**0.370**	**0.370**
Volume Booster	**3/4 and 3/4 ports**	**5.00**	**3.00**

Tables A.2a1, A.2b1, and A.3 were expanded from their first appearance in the ISA book *Tuning and Control Loop Performance,* 1st edition, 1984

A.12 Nomenclature

A_o= limit cycle amplitude (% process variable)

DB_v= dead band from lost motion (% stroke)

C_{vc} = combined flow coefficient for components in series (scfm per psi$^{0.5}$)
C_{v1} = flow coefficient for component 1 (scfm per psi$^{0.5}$)
C_{v2} = flow coefficient for component 2 (scfm per psi$^{0.5}$)
E_o= open loop error (% process variable)

E_x= peak error (% process variable)

K_c= PID controller gain (dimensionless)

K_o= self-regulating process open loop gain (dimensionless)

R_v= resolution (% stroke)

T_i= PID integral time (sec)

T_o= oscillation period (sec)

T_v = time for a full-scale stroke of control valve (sec)
X_v = actuator factor for pre-stroke dead time (seconds $*$ scfm per psi$^{0.5}$)
Y_v = actuator factor for stroking time (seconds $*$ scfm per psi$^{0.5}$)
Δt_x= PID execution rate (sec)

τ_f= signal filter time (sec)

θ_o= total loop dead time (sec)

θ_v = valve pre-stroke dead time (sec)

References

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