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

A.1 Introduction

Presently valve specification forms lead one to think bigger and tighter is better and project bid processing leads one to think cheaper is better. The result is most valves are oversized and have a design with inherent flaws. Often rotary valves designed for on-off service are used for throttling service by the use of actuators and assemblies 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 ramifications 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 dynamic response. The knowledge presented is intended to give guidance to improve loop performance and should not be taken as requirements. The need for tight shutoff can be met by the use of a separate on-off valve coordinated with the throttling valve.

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 positioner 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. 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} response time should be less than the 20% of the open loop process time constant and inverse of the integrating process gain for a self-regulating and integrating process, respectively for good loop performance.

The response time beyond 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 slewing rate set by actuator volume and positioner flow coefficients with the exhaust rate coefficient generally larger. The time for a full-scale stroke T_v can be estimated by the Y_v fill and exhaust factors exemplified in Tables A.2a and A.2b that depend upon actuator type and volume divided by the corresponding C_v flow coefficients for fill and exhaust rate that depend upon the positioner or volume booster exemplified in Table A.3. The slewing rate (%/sec) is 100% divided by T_v. The effect of restrictor or solenoid valve C_{v2} can be included in a modified C_v flow coefficient [1]. Since the C_v fill and exhaust values are quite different, the response time changes with direction of signal change. Also, the response time changes with the size of signal change. The use of a volume booster on positioner output 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 greatly improve slewing rate as seen in Figure A.1. Volume boosters used instead in positioners can cause serious unsafe instabilities [1]. For larger valves inadequate or restricted air supply will slow valve response as seen in the equations in Section A.11. A local air storage tank might be required.

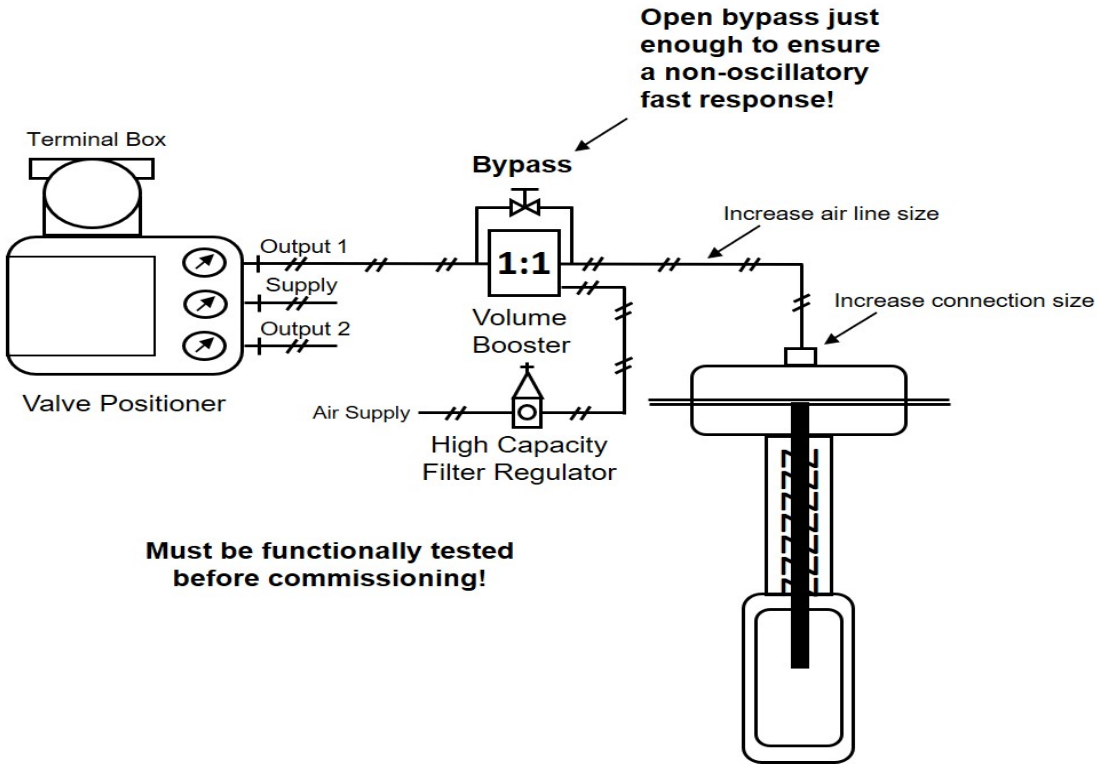

Figure A.1 – Volume booster on positioner output for diaphragm actuator

A.3 Dead time

The valve response dead time for a step change in signal is a combination of a pre-stroke dead time due to actuator volume and fill and exhaust rates and the dead time that dramatically increases for small signal changes and particularly reversals due positioner sensitivity limits. The pre-stroke dead time can be estimated by the X_v fill and exhaust factors exemplified in Tables A.2a and A.2b that depend upon actuator type and volume divided by the corresponding C_v flow coefficients exemplified in Table A.3 for fill and exhaust rate modified as necessary by restrictors or solenoid valves [1]. Higher friction forces will require a larger change in actuator pressure to reversion 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.

The valve response dead time for a ramping signal, which is a characteristic behavior for most control loops due to integral action in respective loop and related loops and the valve response time particularly the slewing rate for large signal changes and large valves, there is an additional dead time in feedback control. The additional deadtime can be estimated as the resolution limit or lost motion dead band divided by the ramp rate [1]. This additional dead time is significant for many types of piston actuators and can be enormous 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 particularly 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 drain effect.

The pre-stroke deadtime can be minimized by volume booster with slightly open bypass valve on positioner output and the size of pneumatic tubing, solenoids, and actuator connections increased. The dead time from resolution limits can be minimized by diaphragm actuators with sensitive positioners. low friction packing, and "flow-to-open" globe valve. The dead time from backlash can be minimized by sliding stem globe valve or rotary valves with splined actuator shaft to stem connections and stem integrally cast with closure member (e.g., ball or disk) [1]. The dead time from shaft windup can generally be reduced by increasing shaft diameter. An increase in actuator size to provide 150% of required maximum force reduces the dead time from resolution and lost motion but slightly increases pre-stroke deadtime.

A.4 Resolution

The stair step response is often the result of piston actuator seal friction, stem packing friction, and seat or seal friction. Movement does not start until the force exceeds the static friction. The movement jumps and does not stop because the sliding friction is less than the static friction. This leads to a stair step response. Piston actuator rack and pinion connections have a resolution set by the gear teeth. The hole pattern of a "drilled hole valve cage" can cause resolution issues. A resolution limit 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 limit. 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, and sensitive processes such as pH, and narrow measurement spans can result in extremely large amplitudes [1].

The limit cycle amplitude (A_o) from a resolution limit is independent of controller tuning and is simply the resolution limit (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 a resolution limit increases as the PID reset time (T_i) increases and the PID gain (K_c) decreases for self-regulating processes [2]:

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

Diaphragm actuators, new packing designs that use modern synthetic products are available for most valves and provide low friction at temperatures once requiring graphite and seat or seal designs that minimize contact particularly after closure member starts to open can greatly improve resolution. The use of external-reset feedback where the actual valve position is used as in input to the PID filter that provides the positive feedback implementation of integral action as detailed in Reference 1 and the ANSI/ISA TR5.9 can stop a limit cycle from resolution on a self-regulating process if the readback is 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 limit.

A.5 Lost Motion

The lack of any response upon signal reversal resulting in dead band where the difference between signal and position is not recovered is termed lost motion. Lost motion can be estimated from the total dead band for a signal reversal 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 position 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. Shaft windup occurs when 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. Backlash and shaft windup cause a limit cycle if there are two or more integrators anywhere (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 is turned on [1][3].

The limit cycle amplitude (A_o) from backlash deadband (DB_v) is inversely proportional to PID gain [3]:

$$A_o = DB_v / K_c$$

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

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

Diaphragm actuators, direct splined shaft to stem connections, large stem diameters, and stems integrally cast with ball or disk can greatly minimize backlash in rotary valves. The use of external-reset feedback can reduce the limit cycle amplitude in an integrating process. For backlash and other sources of lost motion, you can step the PID output when it changes direction and is exceeding a designated noise band. The step would be the backlash dead band. The noise band setting is critical to prevent unnecessary movement of the valve, which can cause excessive wear and upset other loops.

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 pressure 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 Travel Gain

Travel gain is the final change in percent position for a step change in percent signal and is thus dimensionless. If the positioner has a characterization of the input signal, then the output of the characterizer is used as the input signal. Travel gain is particularly affected by resolution and lost motion for small signal changes. 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 set an integral dead band to reduce limit cycles from resolution and to decrease the positioner gain. The positioner can be thought of as a secondary loop where fast immediate response to demands of primary loop is most important with offsets in primary loop from offsets in secondary loop are 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

Flow gain is the final change in flow in engineering units divided by the step change in percent signal. It is affected by travel gain 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 [1].

The flow gain nonlinearity can be reduced by a more precise valve (less 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 fixed 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. Furthermore, the identification of process 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 are severely distorted. Quick opening flow characteristics become so steep, maximum flow is nearly reached below 20% valve position. Linear flow characteristics distort to quick opening characteristics. Equal percentage flow characteristics become very flat until the valve position exceeds 20%. The result is a severe nonlinearity and severe loss of installed rangeability [1].

As noted in flow gain, a signal characterizer can greatly improve loop performance if the installed flow characteristic is well known and fixed and the valve is precise and not oversized. Note that a characterizer does not change impact of some travel nonlinearities, such as resolution on limit cycle amplitude.

A.10 Installed Rangeability Based on Controllability

The inherent rangeability is often stated as the maximum C_v divided by the minimum C_v at low positions that causes inherent flow characteristic to exceed an allowable deviation from theoretical characteristic. A more useful term is the installed rangeability that is maximum controllable flow divided by the minimum controllable flow. The minimum controllable flow is the resolution plus lost motion dead band that is the corresponding flow on the installed flow characteristic near the closed position [1]. For example, if the resolution plus lost motion 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 make valves bigger, cheaper, and tighter 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.

A.11 Specifications

Allowable error (e.g., maximum permissible deviation from setpoint) for minimum, normal and maximum flows is set by the nature of the process to address process efficiency, capacity, quality, and safety requirements. 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 position in tight shutoff rotary valves when the stem is not integrally cast with stem. These valves need sensitive low noise flow measurements in field and travel gages on closure member in shop for tests to identify resolution and backlash. 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 valve test signal may not reveal resolution. ANSI.ISA-S75.25.01 defines the test to identify resolution.

The peak error (E_x) for a step load disturbance denoted as the open loop error (E_o), which is the error if the controller is in manual, can be estimated from the PID tuning settings, open loop 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:

$$E_x = \frac{1.25}{K_c * K_o * \left(1.0 + \frac{0.5 * \theta_o}{(T_i + \Delta t_x + \tau_f)}\right) + 1.0} * E_o$$

The worst-case maximum deviation from setpoint could be approximated as the summation of the limit cycle amplitudes plus the peak error for the largest load disturbance. The loop dynamics with a perfectly linear and instantaneous responding valve, but with the most aggressive PID tuning based on process linearity and dynamics including measurement lags and delays and largest flow gain is first used to estimate peak error. Reasonable amounts of resolution and lost motion are then used to estimate limit cycle amplitudes. Assuming about 20% of the T_{86} response time ends up as additional dead time in a rough first order approximation, the response times, resolution, and lost motion can be increased until the worst-case error approaches the allowable error providing some margin of error providing an indication of specific application requirements. Good tuning software can take into account more accurately the effect of T_{86} response time on tuning in determining the worst-case error.

Specification and test examples given in Table A.1 are based on a broad classification of control loop performance objectives. 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 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 regions 2 and region 3, respectively.

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

Control Objective	Min Test Position %	Resolution %	Lost Motion %	Min Step Size %	Min Step T_{86} sec	Max Step Size %	Max Step T_{86} sec	Min, Max Flow Gain flow eu/%
Tight	2	< 0.2	< 0.2	0.1	< 2	10	< 4	0.8,1.2
Fast	5	< 0.4	< 0.4	0.2	< 1	40	< 2	0.4,2.0
Basic	10	< 1	< 1	0.5	< 4	10	< 8	0.4,2.0
Loose	20	< 2	< 2	5	< 8	10	< 10	0.2,4.0

The following equations can be used with Tables A.2 and A.3 to estimate the pre-stroke dead time (θ_v) and time to stroke from 0 to 100% or vice versa (T_v) using the combined flow coefficient (C_v) taking into account restrictors or solenoid valves.

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

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

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

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

Actuator sq. in.	**Travel inches**	**Pressure psig**	**Spring lb/in**	**Fill Factors X_v**	**Fill Factors Y_v**	**Exhaust Factors X_v**	**Exhaust Factors Y_v**
66	2 1/8	4.5-16	275	0.015	0.338	0.225	0.610
100	3 1/2	6-22	335	0.031	0.861	0.404	1.446
215	4 1/8	6-21	610	0.105	2.276	1.200	3.902
46	3/4	3-15	735	0.012	0.190	0.045	0.256
46	3/4	6-30	1470	0.016	0.226	0.031	0.290
69	3/4	3-15	1100	0.020	0.297	0.071	0.401

69	3/4	6-30	2210	0.028	0.355	0.048	0.457
105	3/4	3-15	1670	0.033	0.466	0.115	0.630
105	3/4	6-30	3320	0.046	0.574	0.078	0.727
156	3/4	3-15	2500	0.046	0.676	0.161	0.913
156	3/4	6-30	5000	0.065	0.811	0.110	1.046
220	2	3-15	1260	0.074	2.004	0.500	2.810
220	2	6-30	2520	0.104	2.243	0.390	3.038
310	2	3-15	1650	0.103	2.790	0.569	3.724
310	2	6-30	3100	0.143	3.379	0.388	4.265
450	2	6-26	4500	0.323	4.386	1.260	6.552
450	2	6-26	4500	0.380	4.586	1.353	6.870

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

Actuator sq. in.	Travel inches	Pressure psig	Spring lb/in	Fill Factors X_v	Fill Factors Y_v	Exhaust Factors X_v	Exhaust Factors Y_v
17	3/4	60	-	0.085	0.050	0.024	0.050
28	3/4	60	-	0.165	0.086	0.035	0.086
56	3/4	60	-	0.296	0.169	0.050	0.169
89	2	60	-	0.715	0.719	0.196	0.719
131	2	60	-	0.995	1.060	0.272	1.060
222	2	60	-	1.730	1.800	0.738	1.800
17	4	60	-	0.020	0.278	0.024	0.278
28	4	60	-	0.051	0.460	0.035	0.460
56	4	60	-	0.099	0.901	0.050	0.901
89	4	60	-	0.181	1.453	0.196	1.453
131	4	60	-	0.227	2.144	0.272	2.144
222	4	60	-	0.603	3.600	0.738	3.600

Table A.3 Examples of Flow Coefficients of Accessories

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	2.30
Volume Booster	1/2 and 1/2 ports	5.32	2.53

A.12 Nomenclature

A_o= limit cycle amplitude (% process variable)

DB_v= dead band from backlash (% stroke)

C_v = flow coefficient for solenoid or booster plus positioner (scfm per psi$^{0.5}$)
C_{v1} = flow coefficient for positioner or booster (scfm per psi$^{0.5}$)
C_{v2} = flow coefficient for solenoid or restriction (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

(1) McMillan, G.K., *Tuning and Control Loop Performance 4th Edition*, Momentum Press, 2014
(2) Hagglund, T., "A Control Loop Performance Manager", *Control Engineering Practice*, vol. 3 no. 11, Elsevier, Amsterdam, Netherlands, 1995
(3) Shinskey, F.G., "Process Control Diagnostics", *Control*, Illinois, USA, assets download, 2002
(4) McMillan, G.K., "Is your control valve an imposter", *Control*, Nov 2012
(5) McMillan, G.K. and Wu, P., "How to specify valves and positioners that do not compromise control", *Control*, March 2016