Control Valve Dynamic Specification

(Version 3.0, 11/98)

1.0 Competitive Marketplace

The global market's continuing demand for quality and uniformity in manufactured products means there is even greater focus being given to process control equipment and its performance. EnTech Control Engineering Inc. has specialized in the optimization of process performance, particularly in pulp and paper manufacturing where product uniformity specifications are now approaching 1%, and product can be rejected when it deviates outside of these limits. Equally important is the fact that process variability impacts operating constraints causing lower manufacturing efficiency and throughput, and thereby reducing the economic potential for the plant. Plant process variability audits frequently find that product variability is increased by individual control loops that *limit cycle* because their control valves are unable to track their controller output signals closely enough (Figure 1). ***This undesirable behaviour of control valves is the biggest single contributor to poor control loop performance and the destabilization of process operation.***

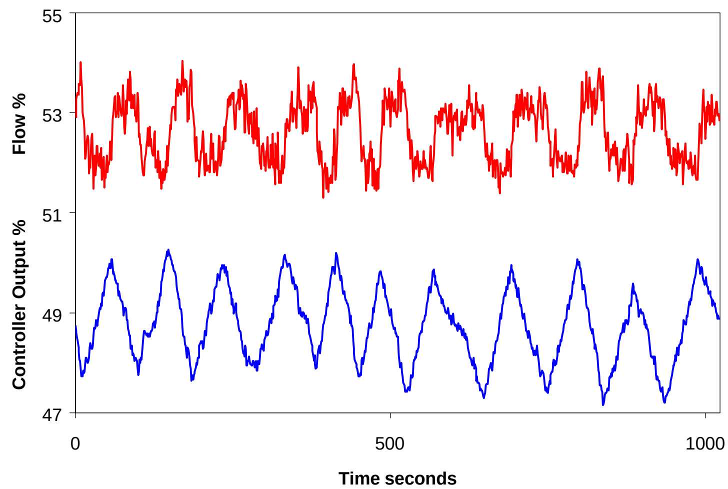

Figure 1 – Typical Control Valve Induced *Limit Cycle*

Control Valve Dynamic Specification - Purpose

The purpose of the Control Valve Dynamic Specification is to define the degree to which control valves can be nonlinear and still allow acceptable process control to be achieved in the highly competitive process industry environment. Minimizing the impact of the control valve on process variability is a key consideration. Intended uses of this specification include: in-process control valve end-use performance; control valve sizing; purchase requirements; and control valve design, manufacture and maintenance requirements. The specification has three parts 1) Nonlinear, 2) Dynamic Response, and 3) Valve Sizing. Parts 1) and 2) - nonlinear and dynamic response, deal with issues such as *dead band* and speed of response, and are intended for the control valve manufacturer. A given control valve can be expected to meet one of the categories called out in the first two parts of the specification. The third part – valve sizing, is intended for the process/instrumentation-engineering designer who is selecting and sizing a control valve for a particular process application. A given valve selection and process design can be expected to meet one of the categories called out in the third part of the specification.

About Version 3.0

The original EnTech Control Valve Dynamic Specification was issued in 1992 and was last updated in 1994 (Version 2.1). Although it targeted performance in pulp and paper processes, it quickly migrated to other industries such as chemicals, hydrocarbons, food processing and energy where similar problems exist. It has been adopted by valve manufacturers as a performance guideline for design. More currently it has provided the impetus for the formation of the ISA SP75.25 subcommittee, which is preparing an ISA standard for small step change performance of control valves. Version 3.0 aligns the language with ISA terminology, considers the end user's process control requirements, defines a valve step response performance index, and broadens the applicability to all process industries where flow regulation affects throughput of quality products. Version 3.0 replaces all previous versions. Sections 1, 2 & 3 give background, Section 4 is the specification, and Section 5 gives testing methods. Note: *italicized* words are defined in Section 6.0 at the end of the document.

Control Valve System

The Specification considers the control valve as a dynamic system, from input signal through to the *flow coefficient* that determines the fluid flow in the pipe. The *control valve system* includes the actuator, drive train, positioner and valve, under normal process operating conditions. The key to determining performance is that there is a measured change in a *process variable* in response to small input step-changes (1% and less). This indicates that the **valve *flow coefficient* has actually changed in the pipe.** Valve *stem* movement is not an adequate indication, especially for rotary valves, and may even be in error for sliding *stem* valves if there is instability in the fluid passing through the valve body. However, valve *stem* indication, is considered to be a good measure of *control valve system speed of response* for step changes large enough to cause valve motion. (In the text the words *valve* and *control valve* are used to mean *control valve system where* ambiguity is avoided).

2.0 Control Problem Definition

Most control valves are used as final control elements in feedback control loops with PID control algorithms. The dynamic response of the *control valve system* is inherently nonlinear in a complex way and has the potential to create the following problems for the control loop:

1. For very small input signal changes, valve nonlinearities and variable *dead time* cause ***limit cycles***. Figure 1 displays a typical *limit cycle* where the tendency of the valve to stick or delay forces the controller to continue correcting for the error from setpoint. **Once a *limit cycle* occurs, effective control is lost and unwanted process variability is created**.
2. The ***speed of response* of the *control valve system*** must be sufficiently fast to allow the desired control loop *speed of response* to be achieved.
3. The *control valve system* response often introduces *dead time* into the loop, which can vary with the magnitude of the valve input signal**. *Dead time* is extremely destabilizing** for a control loop. Variable *dead time* even more so.
4. For larger input changes valve nonlinearities cause the valve **dynamic response** to be **inconsistent**, making it difficult or impossible to tune the controller for consistent performance. **For effective control the *control valve system* must deliver a consistent dynamic response over a specified range of step sizes.**

Linear Control

Most feedback controllers are essentially linear. If all elements of a control loop were also linear, there would be far fewer control problems. What does 'linear' mean in the control context? A linear dynamic system responds to its input signal with the same dynamic response (same gain, *time constants*, *dead time* etc.) regardless of the size of the change in the input signal. Due to their mechanical nature *control valve systems* are highly nonlinear, and this is a major source of problems for control loop performance.

Ideal *Control Valve System* Step Response

Ideally, a *control valve system* should respond to a step change in a fashion which allows the control loop the greatest possible chance of controlling the process effectively without inadvertently generating additional variability. This ideal step response would be much **like a first order response** and would rise monotonically to its final value. It would have a *time constant* and a ***T86* suitably fast to satisfy the control loop speed of response** needed for the process application. It would reach steady state at a time ***Tss*** that would be equal to five time constants or **2.5 times *T86.*** It would have **zero *dead time***, **no *ringing or hunting***, **zero *overshoot***, and a **travel gain of 1.0.** Such a response would appear to be essentially linear to the control loop. It would be **free of all characteristics that could potentially generate variability** though ***overshoot***, ***hunting*** and ***dead time***.

2.1 Speed of Response

The *speed of response* of a *control valve system* can be gauged by its *approximate time constant* t'. The *speed of response* of a control loop is usually determined by the desired closed loop *time constant*, often referred to as *Lambda* (l), which should be selected to allow the process manufacturing objectives to be met. In order to ensure control loop stability and robustness margins consistent with low process variability operation, the *speed of response* (*time constant*) of all of the internal dynamic elements in the control loop including the process (non-integrating), the transmitter and the *control valve system,* should be at least five times faster than that of the control loop. In some cases the *control valve system* is the slowest or dominant dynamic in the control loop, and as a result, it determines how the loop can be tuned. It is this limiting case that forms the argument for the relationship between the control loop expected *speed of response* and the *control valve system speed of response* slow limit in this specification. Unfortunately because of their nonlinear nature, *control valve systems* have a variable *speed of response*. When the valve is slower than an expected *speed of response*, it can destabilize the control loop and cause oscillations to occur. On the other hand when the valve is faster than this expected speed of response, this seldom has a harmful effect. Hence a *control valve system* with a *speed of response* no slower than a certain slow limit is capable of being used successfully in any control loop, as long as the *speed of response* limit of the *control valve system* is at least five or more times faster than the intended *speed of response* of the control loop.

Perhaps 80% of flow and pressure control loops in most industrial plants can be tuned for a *speed of response* range from 5 seconds to one minute. This range is determined by the typical dynamics of many existing control valves, transmitters and distributed control systems. These loops would all work satisfactorily if their control valves had an effective *time constant* of 1 second. In some cases however it is critically important to achieve a faster *speed of response*, such as a hydraulic header pressure control loop which may need a *speed of response* as fast as one second, and hence a minimum control valve *speed of response* of 0.2 seconds. At the other end of the scale, many flow and pressure loops are tuned for 10 seconds and slower, while other variables including many temperature and tank level controllers are often tuned as slow one minute or even one hour. To satisfy a control loop *speed of response* of one minute requires a minimum control valve *speed of response* of 12 seconds. Based on this, four classes of control valve *speed of response* can be defined, and are shown in Table I:

Table I - Control Valve *Speed of response* Classes

Control Loop *Speed of response* λ	Control Valve Maximum *Time constant* τ'
Very Fast (1 second)	0.2 seconds
Fast (5 seconds)	1 seconds
Nominal (10 seconds)	2 seconds
Slow (1 minute)	12 seconds

2.2 Measuring the Control Valve Dynamic Response

The *control valve system* is expected to produce consistent dynamic responses over a certain range of input signal step sizes. The *speed of response* of the valve system can be measured via the *stem* or *shaft* position, and requires a transducer to be mounted on the valve. This must be calibrated to agree with the input signal, and must have a measurement *time constant* at least 20 times faster than the valve (*T86*). A typical step response is shown in Figure 2. The response often has *dead time* (*Td*) which may vary considerably. Prior to analyzing the dynamic response the initial and final values should be established for both the input signal and the stem position. For the input signal and stem position the initial and final values should be averaged over the initial and final steady state periods of the response. The measurement of ***T86***, the time at which the response crosses 86.5% of the step change, captures the majority of the total dynamic response including the *dead time*. The amount of *dead time* is of interest and should be recorded. To avoid ambiguity, *dead time* can be measured as the time after the step change where the response crosses 10% of the full value of the response. After ***T86***, the settling behaviour becomes of interest. The response may or may not *overshoot*. It may ring or hunt by *overshooting and undershooting* several times. The rate of change may slow down considerably as it approaches steady state. It may or may not reach the right steady state value. The initial *overshoot* is the point where the stem position reaches its maximum value after the step change (in either the up or down direction). The ***% overshoot*** is calculated as the amount over the steady state value expressed as a percentage of the change in steady state value of the stem position (the term overshoot applies to both increasing and decreasing steps as in Figure 6). The ***% undershoot*** (not present in Figure 2 – see Figure 6 which *overshoots, undershoots* and *overshoots*) is calculated as the amount under the steady state value expressed as a percentage of the change in steady state value of the stem position. Overshoots and undershoots over 1% should be measured and counted. The ***travel gain*** is calculated by dividing the change in steady state value of the *stem* position by the change in input signal. Ideally, the travel gain should have a value of 1.0. The time at steady state (***Tss***) is the point where the *stem* position reaches within plus and minus 1% of the steady state value.

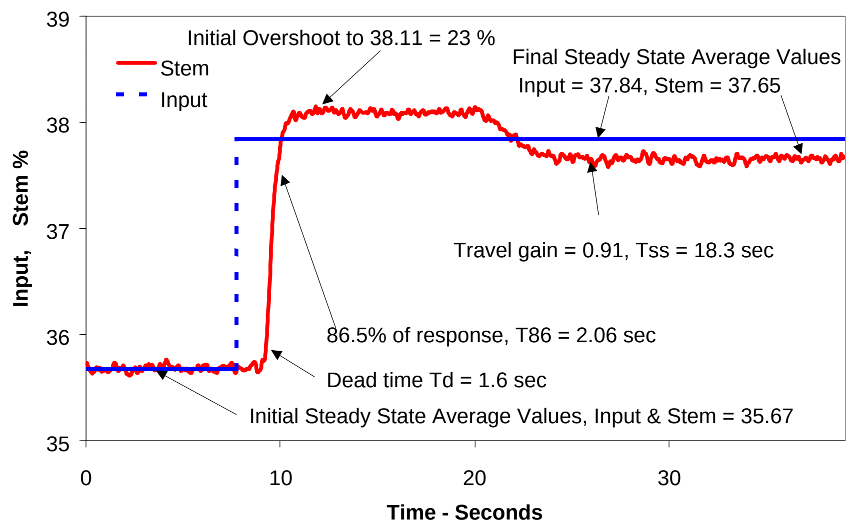

Figure 2 – Step Change Speed of Response
Note: *T86*, Initial Overshoot, Travel Gain, *Tss*

2.3 *Tss as a Function of T86*

A linear first order system reaches steady state in four to five *time constant*s. In four *time constant*s the step response has reached 98.2% of its final value, while in five *time constants* it has reached 99.3% of the final value. Ideally, the settling behaviour of the valve response should be as close to linear as possible, hence the ***Tss*** upper limit should be no longer than 2.5 times ***T86***. A slower settling time will also tend to de-stabilize the control loop. Hence in order to have a speed of response which is fast enough for a givem control loop desired speed of response, the control valve system should have both *T86* and *Tss* values which are equal to or faster than their respective specification limits.

2.4 *T86*, *Dead time* and Control Performance

Whereas *T86* is a convenient way of capturing the valve step response time, it is important to recognize the consequences of various valve response dynamics with the same *T86*. Figure 3 shows three idealized valve step responses all with the same *T86* of 2.8 seconds. Response #1 is an ideal first order response with a *time constant* of 1.4 seconds. Such a response, if it were possible, would be ideal for a valve and would allow the loop to be tuned for a *Closed Loop Time Constant* (λ) of 7 seconds, which is five times slower than the *time constant* of the valve. (In fact because of its ideal nature it would be safe to tune it even faster).

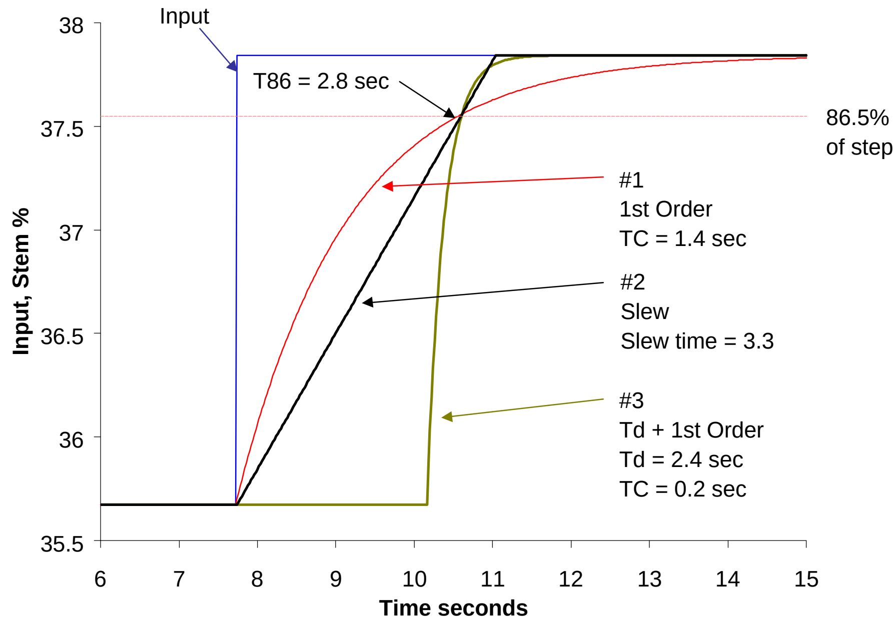

Figure 3 T86 for Various Responses

Response #2 is more typical of an electric valve driven by a fixed speed motor. The response reaches steady state in 3.3 seconds for this step change. The *dead time* is zero, and the response is roughly equivalent to a first order *time constant* of 1 second. Hence the loop could be tuned for a *closed loop time constant* of 5 seconds. Response #3 is much more typical of a pneumatic control valve, and includes 2.4 seconds of *dead time*. *Dead time* is the most destabilizing dynamic parameter for a control loop. The *T86* of 2.8 seconds is 86% *dead time*. *Dead time* in a control loop causes resonance to occur, in which the loop has a tendency to cycle at its natural frequency and amplify process variability. The frequency of the cycle is determined by the amount of *dead time* and the closed loop *time constant*. The amount of resonance or amplification can be expressed in dB's, amplitude ratio or as a percentage. It expresses how much bigger the variability that already exists in the process at the natural frequency would be as a result of the loop's control action. The faster the tuning, the stronger this tendency. Table II quantifies the relationship. Based on this result it is advisable to limit the *Closed Loop Time Constant* (λ) to 4 x *Td*, in order to limit the resonance to less than 20%. For the example of Figure 3, Response #3, since *T86* is mainly *dead time* it is advisable to limit the tuning to 11 seconds (4 x *T86*). A simple rule can be generated from these three results as summarized in Table III. Put in other words, all *T86*'s are not equal. A control loop can achieve effective process control as long as the control valve *speed of response* is at least five times faster than that of the control loop. As *T86* is twice τ', it means that *T86* for the control valve should be 2.5, or more, times faster than the fastest λ planned for the control loop, as long as the *Td* / *T86* ratio is not high. If it the ratio is high, then *T86* should be faster still by a factor of 2.5 / 4 or 62.5% in order to handle the additional *dead time*. This allows Table I to be restated in terms of the above discussion as shown in Table IV below:

Table II – Loop Resonance as a Function of λ and *Td*

Closed Loop TC λ	Resonance		Period of Oscillation
	%	dB	
2 x Td	35%	+2.6	6.0 x Td
3 x Td	26%	+2.0	6.6 x Td
4 x Td	19%	+1.5	6.9 x Td
5 x Td	16%	+1.3	7.2 x Td

Table III - λ as a Function of *T86* and Td

Td/ *T86* Ratio	λ (minimum)
Low (<0.5)	2.5 x *T86*
High (>0.5)	4 x *T86*

Table IV - Control Valve *Speed of response* Classes

Control Loop *Speed of response* *l*	Control Valve *T86* *Td / T86 < 0.5*	Control Valve *T86* *Td / T86 > 0.5*
1 second	0.4 seconds	0.25 seconds
5 seconds	2 seconds	1.25 seconds
10 seconds	4 seconds	2.5 seconds
1 minute	24 seconds	15 seconds

3.0 Control Valve Nonlinearities

Control valve systems have nonlinear behaviour that can be categorized as follows:

1. control valve tracking nonlinearities,
2. flow characteristic nonlinearities.

3.1 Control Valve Tracking Nonlinearities

Control valve tracking nonlinearities represent the inability of the *control valve system* (valve, actuator, and positioner) to faithfully track changes in the input signal, and to ensure that changes in *flow coefficient* actually occur as a result. Tracking nonlinearities consist of *dead band* and *step resolution*, which combine in a complex way to produce *total hysteresis*. This determines the degree to which the valve *closure member* (trim, plug, etc.) fails to track step changes in the input signal. Ideally, the valve system should track input changes with a *travel gain* of 1.0. However, due to the mechanical nature of the valve system (clearances, flexibility, and static friction, dynamic friction), it stands to reason that it is impossible to execute very small step changes uniformly. This specification is intended to quantify the valve behaviour for step changes that approach the ultimate limit of movement.

3.1.1 Nonlinear Regions

Tracking nonlinearities are caused by problems in the positioner/actuator/drive train part of the *control valve system* and prevent the valve *closure member* from following the input signal in a linear and repeatable fashion. For a **pneumatically actuated control valve**, there are four regions of nonlinear operation (referred to as *Regions A, B, C* and *D*). For a very small input signal step change (say 0.1%), the *closure member* does not move at all (*Region A* – less than *dead band* or *step resolution*) in a reasonable time after the step change. Above some initial threshold (say 0.1% to 1%), motion occurs (*Region B*), but due to the nonlinearities and other effects the responses are not consistent. For larger step changes the *closure member* moves in a more consistent manner, and it is possible to classify responses in this region (*Region C*) such that their *step response times (T86*) all fall below the acceptable limit which is required for effective control. For step changes which are larger still (*Region D*), it is likely that the motion of the *control valve system* will become velocity limited (steps of say 10% and greater), hence causing the *step response times (T86*) to become progressively longer.

The *control valve system* transitions continuously through *Regions A, B, and C* as the control loop regulates the process. Under normal process regulation it will transition though *Regions A, B* and will penetrate slightly into *Region C,* as most of the control moves made by a controller are small under normal process regulation. The controller must transition through *Region A,* as here the loop is essentially open due to the fact that the *control valve system* does not respond. The controller will also transition through *Region B* as here the responses are inconsistent and may have very long *dead times*. Only once the controller output transitions into *Region C* can it be expected that the *control valve system* will respond reliably enough for feedback control to work. For this reason it is expected that the controller output will have frequent but shallow penetrations into *Region C*. Only for major setpoint changes or very large process disturbances will the control valve transition through *Region C* and into *Region D.* As a process disturbance occurs, the control loop takes corrective action. Initially, this action tends to be small (inside *Region A*). However, because the *control valve system* will not respond to these small changes, the controller will "wind-up" and produce larger control actions which eventually reach *Region B*. Here the control valve moves but not consistently. Sometimes small step changes result in a long *dead time* before the valve actually moves, again causing the controller to "wind-up". When the valve does finally move after the *dead time*, it is trying to match an input signal (controller output) which actually exceeds the value needed to have the *process variable* achieve setpoint. It is this that causes the *limit cycle* to occur. This variable *dead time* phenomenon produces a region of local instability where the *dead time* is far too long for the existing controller tuning. The variable *dead time* phenomenon is a very common mechanism for inducing control valve *limit cycles*. The amplitude of such *limit cycles* is a complex function of the nonlinearities causing *Region A*, as well as the variable *dead time* of *Region B*. An important point is that the behaviour of the *control valve system* just inside *Region C* is key to determining the effectiveness of a control loop under the condition of normal regulation (some 98% of the time). The actual degree of penetration into *Region C* is a function of *process gain*, the controller tuning and the amount of noise present in the control loop.

For large setpoint changes or major process disturbances, the controller will make larger changes (*Region C*) in the valve input signal. The valve responds to each of these changes relatively quickly and with reasonable consistency. As long as the valve response is fast enough for the controller tuning that has been installed, the loop response will be as anticipated by the controller and effective control will be established. Should the valve be faster than expected, this will not generally upset the controller. When even larger control corrections are needed, the control valve may become velocity limited (*Region D*) and take progressively longer to complete larger changes. This will appear to the controller as if the process has a slower *time constant* than anticipated, with the result that the control loop will tend to oscillate and cause increased variability.

Hydraulically actuated control valves have a similar behaviour to that described above, except that *Regions A* and *B* will likely be much narrower than for a pneumatic valve.

Electrically actuated valves using fixed speed electric motors usually have a narrow *dead band* that applies for small step changes. Here the motor is turned off. This *dead band* determines *Region A*. Depending on the electric motor increase/decrease control logic, there may not be a *Region B*. Beyond this point, electric valves are velocity limited for steps of all sizes as they move at a fixed speed. As long as the step response time *T86* is less than a user-specified limit this defines an acceptable *Region C.* The point at which *T86* exceeds the high limit defines the start of *Region D.*

User Selection of Minimum and Maximum Step Sizes

The specification requires the valve user to specify the desired minimum and *maximum step size*s that are to apply in *Region C*. These limits determine the range of controller output step sizes over which the *control valve system* dynamic response should consistently conform to the dynamic response specification limits (*T86*, *overshoot*, *travel gain* and *Tss*) and will allow the control loop to operate in a near linear fashion as a result. Also, the test procedure identifies what the actual *minimum step* size is at which *T86* (and other parameters) actually meet the specification limits. This point is the upper limit of *Region B* and the lower limit of *Region C*. Clearly, the *minimum step size* is the most important as it determines the limits of effective control, as well as the amplitude of a potential *limit cycle*. Under normal conditions of regulatory control, the controller output transitions through *Regions A* and *B* and far enough into *Region C* to cause the *control valve system* to respond and allow feedback control to occur. The amount of penetration into *Region C* varies inversely with *process gain*, and directly with controller gains and process noise. If the process gain is high, the tuning fast and the process noise substantial, the controller output will continually be making changes that are large enough to be inside *Region C* all the time. In this case there will be no limit cycle and the *control valve system* will not impact control performance in any adverse way. The minimum step size determines where the lower limit of *Region C* should occur. It in turn depends on the size of the valve nonlinearities (*dead band*, *step resolution*, *total hysteresis)* in *Region A*, as well as the size of *Region B*. If the user wants to have a tighter *minimum step size*, this determines *Region B*. In turn it also requires tighter limits to be set for the nonlinear parameters in *Region A*. Roughly a factor of two can be applied to the *total hysteresis* (*Region A*) in order to estimate a feasible value for the *minimum step size*, although this is clearly very dependent on the design of the valve system. Alternatively, the *total hysteresis* must be smaller than the specified *minimum step* size, and a factor of one half can be used to estimate a reasonable *total hysteresis* limit from the minimum step size.

Selection of the *maximum step size* is far less important for regulatory control. The *maximum step size* determines the ability of the control loop to handle large changes with consistent dynamics as well as small ones. Large step changes occur only at certain times, such as when the control loop is responding to major setpoint changes, large disturbances or some form of sequence such as process start-up, shutdown or product transition. In-process testing will not normally allow step sizes larger than some practical limit, such as 10%, to be applied under process operating conditions. Hence, under these conditions it is only possible to imply conformance for large step changes by extrapolation. For instance, if *T86* as measured, is well below the specification limit and is decreasing as step changes increase (see Figure 4), then it is likely that it will also meet specification for 10% and possibly even for 50% changes.

This concept is illustrated in Figure 4, which shows how the *step response time T86* might vary with step size for a given pneumatic valve. For very small step sizes, *T86* is expected to be very long. In fact in *Region A* where no motion occurs, *T86* is infinitely long. In *Region B*, it is expected that small step changes will cause ever longer *dead time*s. As the step size becomes larger, *T86* is expected to become much smaller. Then as the step size becomes larger still, *T86* is expected to become progressively larger as the valve system becomes velocity limited. In the example of Figure 4, the user specified parameters are consistent with the default values for minimum and maximum step size (2% and 10%), as well a control loop *speed of response* (l) of 10 seconds, which calls for "consistent movement" as follows:

1. *T86* less than 4 seconds for step sizes ranging from 2.0% to 10%. Since in Figure 4 the *T86* vs. step size curve crosses 4 seconds at a step sizes of 1.4% and 15.9%, this requirement is far exceeded.
2. A *travel gain* of 1.0 +/- 0.2 for all of the step changes specified in 1. above.
3. An *overshoot* of less than 20% for all of the step changes specified in 1. above.

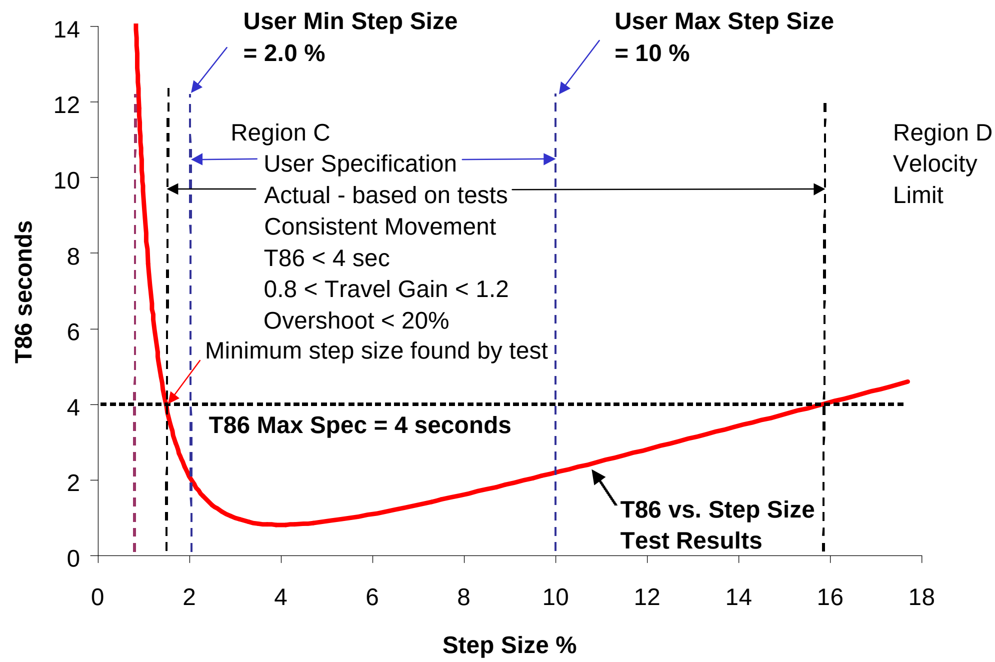

Figure 4 – *Region C* – Consistent Responses

Figure 4 also illustrates how the user specification of minimum and maximum step sizes may differ with the actual performance of the control valve system. Clearly the example of Figure 4 exceeds the minimum and maximum step size specifications, which for illustration use the specification default values of 2% and 10%. The example valve actually conforms to 1.4% and 15.9%. The Actual minimum step size is the boundary between *Regions B* and *C*. This value is the real measure of the *control valve system* nonlinear performance and is measured during the testing.

Figure 4 illustrates the expected results for a typical pneumatic valve only. Other results are also possible. A fixed speed electrically actuated valve is expected to have a fixed dead band when due to the fact that the motor must be deactivated when the valve is at rest. When the valve is moving it will do so at fixed speed. This translates into a characteristic that would parallel the Region *A* demarcation line in Figure 4 to a minimum *T86* value for the smallest step change the valve can execute. From this point there would be a line of rising T86 with step size, which would cross the T86 limit at a *T86* value which would be the demarcation between *Regions C* and *D*.

The pneumatic valve illustration in Figure 4 assumes that *control valve systems* have the tendency to have longer *T86* values as the step size becomes smaller at the bottom of *Region C*. The specification recognizes only four control loop speed of response classes (1, 5, 10 and 60 seconds) with eight T86 limits (0.25, 0.4, 1.25, 2, 2.5, 4, 15, 24) which in turn are a function of *dead time*. These limits have been designed to handle typical pneumatic valve characteristics. *As control valve system* designs continue to improve, this may no longer be the case. In an ideal design, as soon as the valve starts to move for the smallest step size possible (*Region A* upper limit), it will do so at a *T86* which is well below the *T86* limit for the speed of response class. In this case the testing methods should record the longest *T86* observed, as this is a real measure of the true performance of the *control valve system*.

3.1.2 *Control Valve System* Step Response Performance Index - Weighting Factor *W*

The specification sets out various limits for speed of response *T86, Tss*, *% overshoot* and *travel gain* which apply to *Region C*. Two c*ontrol valve systems* can meet the specification limits yet differ in their relative step response performance, hence it is important to provide a means of measuring these differences. For instance a valve with a 19% overshoot and one with a 0% overshoot both pass the specification overshoot limit of 20%, yet clearly the performance of the second valve is better than the first. A *control valve system* step response performance index has been designed. This consists of a system weighting factors that weight the deviations of each step response from the ideal first order response in various ways. The performance index is designed to measure the performance of the *control valve system* under normal regulatory conditions. As a result the weighting factors are applied only over a very narrow range of step sizes from the minimum step size to double the minimum step size, as this is where the control valve system is expected to operate most of the time. The absolute value of the minimum step size is also factored into the index. Weighting factors are applied to four phenomena are listed in Table V below.

Table V – Weighting Factors – Step Sizes ranging from Minimum to 2 x Minimum

Phenomenon	Most Desirable Value	Specification Limit	Typical Weighting *W* at Limit
Td/T86 **Ratio**	0	1.0	30
Tss/T86	2.5	expect 5 and more	20
Overshoots & Undershoots	0%	20%	30
Travel Gain	1.0	0.8 – 1.2	20
Combined	0	All limits	Approx. 100

The performance index is calculated using Equation 1) below, which combines weighting factors based on *T86*:

_W(T86_) = (_Td/T86_) x 30 + Abs(_Tss/T86_ – 2.5) x 8
+ (sum of overshoots and undershoots (%)) x (min. step size %) x 0.75
+ Abs(1 – _travel gain_) x 50) x (min. step size %) …….1)

Example: *Td*=2, *T86*=3, Tss=12, 1st *O/S*=12%, other overshoots, undershoots = 0, *Travel gain* = 0.9, minimum step size = 2%. For these statistics *W(T86)* = 0.67 x 30 + (4 -2.5) x 8 + 12 x 1 x 2 + 0.1 x 2 x 50 = 66

The weighting factors have been designed to produce a valve performance index of approximately 100, for a *control valve system* which just passes the specification at the default *minimum step* size of 2%, with all undesirable values at or near the specification limits, or other large excursions as outlined in Table V. The first two terms in the performance index relate to speed of response. The second two terms relate to travel. The last two terms have been weighted by the absolute value of the minimum step size. In this way a valve which meets a minimum step size of 0.5% is much better than one with a 2% minimum step size. The minimum step size value has two meanings. Prior to testing only the specification limit is known. After testing is complete, the actual value (Upper limit of *Region* B) should be known. If available, the actual value is to be used in Equation 1) and 2), otherwise the specification limit value is to be used. The use of the step size as part of the weighting factor on overshoot produces a much lighter penalty for a 0.5% minimum step size than for 2%, and a better valve performance rating as a result. More serious problems, such as *dead time* and *overshoot* have been given a relative weight of 30 points each at their maximum values and 2% minimum step size. Less serious problems, such as *Tss* and *travel gain* have a weighting of 20 points each at their high or maximum allowed values and 2% minimum step size. Overshoot is further weighted by the sum of the % overshoots and % undershoots, as this indicates a stronger tendency to oscillate and to create process variability.

The performance of a given *control valve system* can also be judged based on its intended use in a control loop of given closed loop time constant, *Lambda*. If a control valve with a *T86* of 3 seconds, a relatively high *Td/T86* ratio and a relatively slow *Tss* is to be used in a control loop with a one minute *Lambda* value, clearly, the relative importance of the *Td* and *Tss* weights is not very high (the *Lambda* values which should be used for this purpose are: 1, 5, 10 and 60 seconds and correspond to the speed of response classes in the specification). These factors are taken into account Equation 2 below, which weights the time-based penalties on the basis of the control loop *Lambda* value, as opposed to *T86*:

W(Lambda) = ((_Td/T86_) x 30 + Abs(_Tss/T86_ – 2.5) x8) x (_T86_ x 2.5 / Lambda)
+ (sum of overshoots and undershoots (%)) x (min. step size %) x 0.75
+ Abs(1 – _travel gain_) x 100) x (min. step size %) …….2)

Example: using the example given above for Lambda = 60 seconds, *W(Lambda)* = 38, whereas *W(T86)* = 66

The total score is the sum of both time response and overshoot/gain scores. Example scores for the responses of Figures 2, 3 and 6 for a minimum step size of 2% are given in Table VI below for illustration:

Table VI – Weighting Factors for Figure 2, 3 and 6 Example Responses

	Figure 2	Fig 3, Resp # 1 Ideal 1st Order	Fig 3, Resp # 2 Electric Drive	Fig 3, Resp # 3 Pneumatic	Figure 6
Td	1.6	0	0	2.4	2
T86	2.06	2.8	2.8	2.8	2.5
Tss	18.3	7	3.3	3.8	70
Overshoots & Undershoots	23 %	0 %	0 %	0 %	16%+5%+4%
Travel Gain	0.91	1.0	1.0	1.0	1.0
W(T86)	**118 = v poor**	**0 = perfect**	**11 = excellent**	**35 = vg**	**265 = v bad**
W(Lambda=10)	**82 = poor**	**0 = perfect**	**7= excellent**	**24 = vg**	**180 = bad**

The response of Figure 2 has relatively high *dead time*, a large *overshoot*, and a long settling time. The resulting score of 118 is very poor and is indicative of these problems. If this valve is to be used in a control loop with a Lambda value of 10 seconds, the score is slightly improved (poor), as the *dead time* and settling time contributions are not as vital. In Figure 3, Response #1 is an ideal first order response and has a perfect score of zero. Response #2 is typical of an electric valve. It has zero *dead time*, and no *overshoot*. Its settling time is surprisingly short as compared to the ideal value of 2.5 times *T86*. Its score excellent (very and close to zero). Response #3 is typical of a pneumatic valve, and has a high *dead ti*me ratio, but no *overshoot*. The score is quite low (35) signifying a very good performance, but not as good as Response #2. Figure 6 shows a response with a modest initial overshoot (16%), which is followed by an undershoot of 5% and an overshoot of 4%. This is represents excessive ringing and also has an extremely long settling time. The resulting score is very high signifying a very unsatisfactory (very bad) result.

3.2 Flow Characteristic Nonlinearities

Valve selection and valve sizing is generally carried out by an engineering design firm during the engineering phase of a capital project. It is also carried out by plant staff when re-sizing, replacing or re-selecting a control valve. The *control valve system* that is selected will determine the fineness of the control capability (*total hysteresis*, *minimum step size*, *Region C*, etc.) as well as the *valve characteristics*. The size of the valve will determine the installed *valve flow coefficient, flow gain, and* range of *process gain*. The process/instrumentation designer making these two decisions determines: the capability of this control valve to be used for effective control; the amplitude of the potential limit cycle; and the degree of difficulty of tuning the resulting control loop. In turn these decisions will determine how effective the control loop will be and how much unwanted process variability it will potentially create. This section of the specification is devoted to helping the process/instrumentation designer to make informed decisions regarding both selections with a view of minimizing the impact on process variability.

Every control valve has a fluid flowing through it, hence a flow value can be determined under given process conditions. During the project design phase the sizing of flow streams is carried out, and the design drawings and flow sheets typically show the minimum, nominal and maximum expected flow figures for the process design. The process designer has access to these figures. The fluid flow may or may not be measured by the final process instrumentation. Instead, the instrumentation may measure pH, tank level, or temperature. Nonetheless, the performance of the control valve selected for the application will determine the *step resolution* of the *control valve system*, while the valve sizing will determine the *flow gain*, which in turn will determine the *flow resolution*. This will be the case even if the flow is not measured. The installed *flow gain*, together with the process characteristics and transmitter span will result in the “installed *process gain* characteristic” for the control loop. In turn this determines:

1. The effective *flow gain (Kf)* of the control valve, which together with the *minimum step size*, determine the minimum expected amplitude of the potential flow *limit cycle*. The *minimum step size* is a function of the valve system selection, while the *flow gain* at the operating point is a function of the valve sizing and piping design. The amplitude of the *limit cycle* determines the process variability that the control valve is capable of generating. The worst case on a percentage basis occurs at the minimum design flow value.
2. The *flow gain (Kf)* together with the span of the transmitter impact the control loop *process ga*in (*Kp)*, which in turn determines the rangeability of the loop,
3. The degree to which the *flow gain* varies over the operating range of the process determines the degree of difficulty of tuning the control loop over the operating range of the process.

The *flow gain* for the control valve measures the ratio of flow change to input signal change for an input signal step change. It is determined by the *flow coefficient* for the valve, the fluid characteristics, the upstream and downstream pressures, and the shape and slope of the installed control *valve characteristic*, which is typically nonlinear. However the degree to which the installed *valve characteristic* changes is determined by the chosen characteristic for the valve (equal percentage, linear, quick opening etc.) and the relative pressure drops across the valve and the rest of the fluid transport system. Ideally an installed control loop *process gain* of 1.0 is desirable for a *self regulating process*, as this allows for full valve travel over the full span of the transmitter. However, in many cases valves are severely oversized in the quest for additional operating flexibility. This results in control valves with very high *process gains*, which are nearly closed during normal operation. The effective operating range for these control valves is hence very narrow. The *step resolution* now becomes a significant fraction of this range, which magnifies the valve tracking nonlinearities making good control impossible. Also a large *limit cycle* usually results.

3.2.1 Flow Resolution and Process Resolution

The ability of the *fluid flowing* through the valve and the *process variable* to track changes in controller output signal, or valve input signal, can be defined respectively as the ***flow resolution*** and the ***process resolution***. *Flow resolution* applies to all valves even if the flow is not measured, as all valves have a fluid flowing through them. *Flow resolution* can be calculated based on the *step resolution* and the *flow gain.* *Process resolution* applies to all control loops based on the actual measurement that is made. If the process measurement happens to be flow*,* then the *flow resolution* and the *process resolution* are the same thing*. Flow resolution* is the product of valve *step resolution* and *flow gain (Kf),* while the *process resolution* is the product of the *step resolution* and *process gain (Kp)*. Given a certain *step resolution*, the lower the *flow gain* and *process gain*, the lower the *flow resolution* and *process resolution* become. Hence, to some extent a **high *step resolution* can be compensated for by a low *process gain*.**

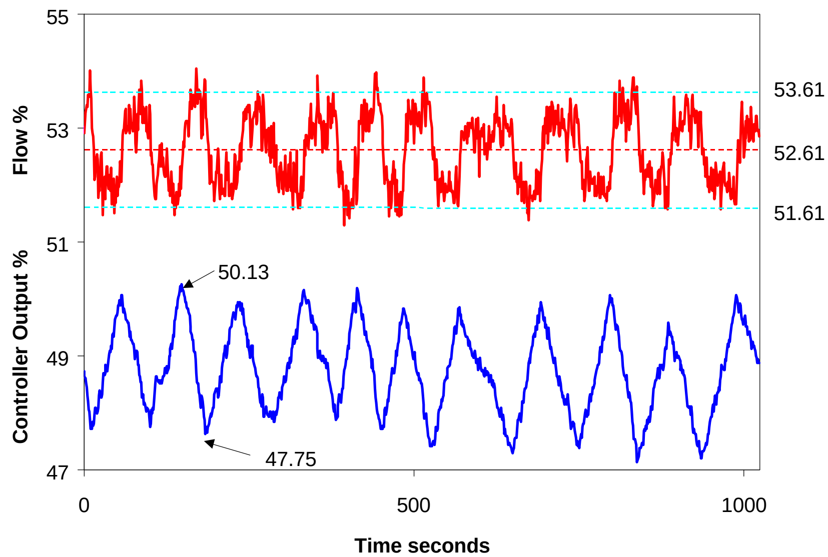

Figure 5 - *Limit cycle* Amplitude

3.2.2 *Limit Cycle* Amplitude – *Process Variable*

It is difficult to predict the exact amplitude of a *limit cycle* in either the fluid flow or the *process variable* that results from valve nonlinearities. The amplitude is a complex function of the *dead band*, *step resolution*, *total hysteresis* and variable *dead time* that apply inside *Regions A* and *B*. A safe estimate however is that the controller output *limit cycle* peak-to-peak excursion is unlikely to be smaller than the *minimum step size*, or the upper limit of *Region B*. Hence, the amplitude (one half of peak-to-peak) of the flow *limit cycle* can be predicted as at least one half of the *minimum step size* times the *flow gain*, while that of the *process variable* is at least one half of the *minimum step size* times the *process gain*. The amplitude of the flow *limit cycle* is very important as it determines the degree to which the control valve will impact process variability. The flow *limit cycle* amplitude can be expressed as a percentage of the mean flow at the operating point. The worst case occurs at the minimum design flow.

This can be illustrated by referring to Figure 5, which is a repeat of Figure 1. The flow signal oscillates from approximately 53.61% down to 51.61% of span. The peak-to-peak of the flow *limit cycle* is about 2% of span, while the amplitude is about 1% of span. The mean value is 52.61% of span. Hence the **amplitude of the flow *limit cycle* is 1.9% of mean value** (1% x 100% / 52.61%)**. This value represents the de-stabilizing potential on process variability of this control valve selection, sizing and flow**. The controller output is cycling from approximately 50.13% down to 47.75%. The cycle appears to be a triangular wave caused by the controller integral action ramping the controller output as long as the valve is stuck, or is slow to respond due to a long *dead time*. Hence, it is likely that the actual valve *stem* position is switching at the instants that the controller output reaches its furthest excursion, and at approximately the same values. The peak-to-peak of the controller output *limit cycle* is about 2.38%, and is a good estimate of the *minimum step size* and Region B. It is not possible to estimate the size of the nonlinearities of *Region A* from this number, except that the *total hysteresis* is less than this value. The *process gain Kp* has the value 0.84 (2% / 2.38%).

3.3 Varying *Process Gain* & Controller Tuning

Flow characteristic nonlinearities are a function of valve *closure member* characteristics and the pressure drop distribution between the control valve and the rest of the process piping, pumps and other equipment. **These nonlinearities cause the control loop *process gain* to vary with the operating point** and this **further complicates the control loop tuning and performance problem**, since a fixed gain controller cannot cope well with these changes. Ideally, the *valve characteristics* and sizing should be chosen to minimize how much the *process gain* varies over the operating range. Ideally the *process gain* should not change by more than a factor of two, for good control, as reasonable tuning can usually absorb this factor. As a last resort, the variation in *process gain* can be compensated for through nonlinear compensation techniques in the controller, or positioner. Although this is possible it is seldom very robust.

3.4 Documentation – Specification Sheet

At time of purchase the expected performance of a *control valve system* should be documented in a specification sheet, for *control valve systems* or "valve packages" assembled by a valve manufacturer or supplier. When a user assembles a *control valve system* from components (control valves, actuators, positioners), the user should attempt to document the performance based on in-process tests, after the valve system has been placed in service. The actual performance of a *control valve system*, as installed, should be documented in a specification sheet. The parameters called out in this specification should be reported.

3.5 User Selection of Specification Parameters

The specification has been written taking into account the majority of process applications. Nevertheless control valve system applications outside of the anticipated performance will occur, and it is up to the user to determine appropriate performance requirements. Each part of the specification contains an extra space for the user for this reason. Users are encouraged to make informed decisions on the basis of the principles outlined in this document.

3.6 Applicability

The specification applies to control valves. It does not apply to on/off, hand, solenoid, blocking or switching valves.

3.7 Passing the Specification – Responsibility and Dependency on Instrumentation

The specification contains three parts: Nonlinear, Dynamic Response and Valve Sizing. The first two parts apply to the performance of a particular *control valve system* design and it is the responsibility of a *control valve system* vendor to pass these requirements. The third part – valve sizing – applies to the suitability of a particular *control valve system* for use in a specific process control application, and it is the responsibility of process/instrumentation designer making these design selections to conform with the specification.

Three signals are needed to test the *control valve system* in-process: the valve input or controller output signal, the process measurement (*PV*) and the *stem* or shaft position. The PV signal is needed to test the Nonlinear and Valve Sizing part of the specification, while the *stem* or shaft position is needed to test the Dynamic Response part of the specification. Should these signals be unavailable, or should suitable testing equipment be unavailable then testing cannot be carried out. Although the *PV* is always available, it is not always suitable for testing purposes (integrating process variables, or very slow measurements are not suitable). A stem or shaft transducer is needed to test against the Dynamic Step Response part of the specification. In most cases this can be fitted to the control valve system as a temporary installation, providing plant safety regulations allow. A stem transducer could be installed as a permanent installation in such cases.

The stem transducer signal and the valve input signal must be sampled at a rate that is at least twenty times faster than *T86* for the *control valve system.* For the four control loop speed of response classes the corresponding sampling rates are: 12 msec, 62 msec, 125 msec and 0.75 sec. These sampling rates are faster those for a typical DCS system and require a high-speed data collection system to be used for this purpose.

4.0 SPECIFICATION

The Control Valve Dynamic Specification is organized in three sections: Nonlinear, Dynamic Step Response and Valve Sizing. Each category has a number of recommendations, a default value, and an extra space is provided for a user-specified selection. **If no control loop application knowledge is available, the default values should be used**. **The performance of a *control valve system* shipped as a package should be documented in a specification sheet, including the parameters called out in this specification. (**Footnotes for references in the tables appear on the next page).

4.1 NONLINEAR SPECIFICATION – (Responsibility - Control Valve System Vendor)

The nonlinear specification sets the maximum allowed *dead band, step resolution* and *total hysteresis*. The *total hysteresis* influences the potential *minimum step size*, which in turn determines the amplitude of the potential controller output *limit cycle*. The *minimum step size* together with the *flow gain* determine the amplitude of the potential *PV limit cycle*. Three classes are given: nominal, fine and very fine. Default values are provided for both rotary valves and sliding *stem* valves.

Valve Tracking Nonlinearities (% input signal)

Class	Nominal - 1%	Fine - 0.5%	V Fine – 0.1%	DEFAULT Rotary Valves	DEFAULT Sliding *Stem*	User
***Dead Band* (%)**	0.6 [1]	0.3	0.06	**0.6**	**0.3**	
***Step Resolution* (%)**	0.4 [1]	0.2	0.04	**0.4**	**0.2**	
***Total Hysteresis* (%)**	1.0 [1]	0.5	0.1	**1.0**	**0.5**	

4.2 DYNAMIC STEP RESPONSE SPECIFICATION – (Responsibility - Control Valve System Vendor)

STEP SIZE RANGE

The dynamic response specification sets the ranges over which consistent dynamics are to be achieved (*Region C*)

The step size range is set from minimum to maximum. *Minimum step size* depends on the *total hysteresis*, and the magnitude of *Region B*. It is valve design dependent and is likely to be about double the *total hysteresis*. Values are given for *nominal*, *fine*, and *very fine*. The finer, the more capable the valve design. Default values are given for rotary and sliding *stem* valves.

***Minimum Step Size* (%)**

Nominal	Fine	Very Fine		DEFAULT Rotary Valves	DEFAULT Sliding *Stem*	User
2.0 [1]	1.0	0.2		**2.0**	**1.0**	

The *Maximum step size* determines the upper range over which the valve is nearly linear and depends on the size of *Region D*. Values are given for nominal, wide and very wide. The wider, the more capable the valve design.

***Maximum Step Size* (%) [7]**

Nominal	Wide	Very Wide		DEFAULT	User
10	50	100		**10**	

STEP RESPONSE - *REGION C* – Consistent Dynamics

The step response specification sets *T86*, *% Overshoot, Travel Gain*, *Tss.* Each class is based on the fastest control loop *speed of response* (*l*) available, given the valve ***T86*** and ***Tss*** as specified. The four classes include: Very Fast (1 second), Fast (5 seconds), Nominal (10 seconds), Slow (1 minute). The default is set for 5 sec. The valve performance index W calculates a composite weighting for responses ranging from the minimum step size to twice this value.

***T86 Step Response Time (*seconds**) by Fastest Loop *Speed of Response* Class (Function of *Td / T86 Ratio*)

Class	1 second	5 seconds	10 seconds	1 minute	DEFAULT	User
Td / T86 < 0.5	0.4 [2]	2 [3]	4 [4]	24	**2**	
Td / T86 > 0.5	0.25	1.25 [6]	2.5 [5]	15	**1.25**	

***Tss* Steady State Time *(*seconds)** by Fastest Loop *Speed of Response* Class (Function of *Td / T86 Ratio*)

Class	1 second	5 seconds	10 seconds	1 minute	DEFAULT	User
Td / T86 < 0.5	1	5	10	60	**5**	
Td / T86 > 0.5	0.63	3.1	6.3	38	**3.2**	

Travel gain

Nominal	DEFAULT	User
0.8 to 1.2	**0.8 to 1.2**	

%*Overshoot* (% of step change)

Nominal	DEFAULT	User
20	**20**	

***Valve Performance Index W* - Weighting Factor** - (Based on Equations 1 and 2) 0=perfect, 100=poor

***W(T86)* Equation 1 =**		***W(Lambda)* Equation 2 =**	

4.3 VALVE SIZING SPECIFICATION – (Responsibility – Process/Instrumentation Designer)

Flow Characteristic Nonlinearities:

This section of the specification is intended as a guideline for control valve sizing calculations. Valve sizing is an integral part of the general design and sizing of the whole fluid transport system, and is concerned with designing the pressure drops taken across each of the elements of the system, over the operating range of the process. The process designer has the ability to balance various aspects of the design as the flow changes over the operating range. For example in a hydraulic system, the selection of the pump characteristics and piping dimensions determine the pressure drop across the control valve. This, together with the selection of the control valve *flow coefficient* and *valve characteristic* determines the *flow gain,* as well as the manner in which the *flow gain* varies over the operating range. The *flow coefficient* selection also determines the range of operation of the flow, given the available pressure drops. The higher the *flow coefficient*, the larger the flow range. The *flow gain* in part determines the *process gain* for the control loop, while the installed characteristic of the valve determines the amount by which the *process gain* varies over the operating range of the process.

The selection of a given *control valve system* determines the *minimum step size* (Section 4.2)*,* as well as the *total hysteresis* (Section 4.1). The amplitude of a potential *limit cycle* in the controller output is likely to be at least one half of the *minimum step size*. The amplitude of a potential *limit cycle* in the fluid flow represents the potential for the control valve to create unwanted process variability, and is a key measure of the control valve performance. This estimate of the potential amplitude of the flow *limit cycle* is the product of one half the *minimum step size* times the installed *flow gain*. The flow *limit cycle* amplitude can hence be determined by selecting both the *minimum step size* and the *flow gain* in combination. A high *minimum step size* implies a *control valve system* with mediocre tracking performance. This can be partially offset by selecting a *low flow* gain. The lower the *flow gain*, the lower the flow *limit cycle* amplitude for a given *minimum step size*. The process designer can specify a nominal value for the flow *limit cycle* amplitude, based on product uniformity considerations. The instrumentation designer is then free to achieve this specification by selecting a *control valve system* with a small *minimum step size* (better tracking performance) if the *flow gain* is high. Alternatively, the designer can re-size the valve, or change the pressure drop profile in the fluid system to reduce the *flow gain* if the *minimum step size* is high. The design process is a combination of both options.

The flow *limit cycle* can best be expressed as the amplitude of the potential process variability, on a percentage basis, by calculating the *limit cycle* amplitude as a percentage of the nominal flow. The *flow gain %* is the *flow gain* in flow units / valve travel %, divided by the flow at the minimum design operating point and expressed as a percent. The *Flow Limit Cycle (%)* is the *minimum stem size* times the *flow gain (%).* The designer should consider the worst case in the process design (highest or lowest flow).

Maximum Allowed Flow *Limit Cycle* Amplitude (% of Nominal Flow)

				DEFAULT	DEFAULT	
	Nominal	**Fine**	**Very Fine**	**Rotary Valves**	**Sliding *Stem***	**User**
***Minimum Step Size* (%)**	2.0	1	0.2	**2.0**	**1**	
***Flow Gain* (%)**	1.0	1.0	1.0	**1.0**	**1.0**	
Flow *Limit Cycle* (%)	1.0	0. 5	0.1	**1.0**	**0. 5**	

The control loop *process gain* is a function of the *flow gain*, the relationship of the flow in the pipe to the measured process variable, and the span of the transmitter used to make the process measurement. Ideally, the *process gain* should be approximately equal to unity (% *PV* / % valve travel) for good design. The amount by which the *process gain* varies over the operating range of the process, determines the degree to which the control loop will be difficult to tune. Poor tuning leads to control loop cycling and higher process variability. Ideally the *process gain* range should be limited to +/- a factor of two.

Variation in *Process Gain* (*Kp*),

				DEFAULT	DEFAULT	
	Nominal	**High**	**Low**	**High**	**Low**	**User**
Nominal *Kp* (%/%)	1.0 [1]	2.0 [1]	0.5 [1]	**2.0**	**0.5**	

Footnotes

1 Equivalent to Version 2.1 — Combined *Backlash* & *stiction* limit of 1%
2 Closest Version 2.1 Equivalent — 0-2 inch valve *T86* = 1.43 x T63 = 0.43 sec.
3 Closest Version 2.1 Equivalent — 6-12 inch valve *T86* = 1.43 x T63 = 1.7 sec.
4 Closest Version 2.1 Equivalent — 20+ inch valve *T86* = 1.43 x T63 = 3.4 sec.
5 Closest Version 2.1 Equivalent — 12-20 inch valve *T86* = 1.43 x T63 = 2.5 sec.
6 Closest Version 2.1 Equivalent — 2-6 inch valve *T86* = 1.43 x T63 = 0.86 sec.
7 In-process testing step size limited to ~10%. Can only imply conformance to larger values by extrapolating slope of Fig. 4.

5.0 TESTING PROCEDURES – IN-PROCESS

In-process testing for *dead band*, *step resolution* and *total hysteresis* can be done for most control loops with reasonably fast non-integrating *process dynamics* such as flow or pressure. The key criterion is to be able to measure, observe and interpret changes in the *process variable* as a result of stepping the control valve input signal. Where this is not feasible, such as on a tank level measurement, it may be possible to install a downstream pressure transducer, or in some cases an ultra-sonic flow measurement can be used. The tests are organized to measure control valve nonlinearities (*dead band, step resolution*) and response dynamics (*T86*, *Td, O/S,* K_T*, Tss).* The tests for nonlinearities are carried out using the *process variable*. The tests for response dynamics depend on a *stem* or *shaft* transducer being available. The tests are organized to maximize testing efficiency and to take as little time as possible. The tests include:

1. **Test 1 - Initial Test** consists of a few step tests that are larger than the *minimum step size*[1]. The test can be done in a few minutes. If based on this test the valve fails to meet the requirements no further tests are needed.
2. **Test 2 - Increasing Step Test** consists of a sequence of step changes of increasing magnitude. The test takes about ten minutes[1]. The test allows initial estimates of the *dead band* to be bracketed and some measurements of *T86, Td, O/S,* K_T *and Tss* to be made. It allows the *minimum step size* to be measured, and the valve performance index W to be calculated. If the valve fails to meet the criteria being measured, then further tests are not needed. Also, if the test indicates that the *dead band* is very small in comparison with the requirement, it may be possible to avoid doing the Test 3 - Small Step Test, which is quite time consuming.
3. **Test 3 - Small Step Test** consists of a sequence of small step changes carried out in the same direction, with several reversals of direction. The test is designed to accurately measure the *dead band* and *step resolution* and is required to pass the Nonlinear part of the specification. The test is time consuming and requires about an hour to complete[1].
4. **Test 4 - Medium Step Test** is similar to the Test 2 - Increasing Step Test and is intended to provide accurate measurements of *T86, Td, O/S,* K_T *and Tss* and the valve performance index *W*. The test requires about twenty minutes to complete[1].

Some form of process measurement is always available even though some measurements may not be suitable for in-process testing to be carried out. In many circumstances the availability of a *stem* transducer involves temporarily fitting a test transducer to the stem or shaft. In some cases this cannot be done for safety reasons. If a stem transducer is not available the speed of response measurements cannot be made, hence only the first three tests can be carried out, and only dead band, step resolution, total hysteresis, flow gain and process gain can be determined.

[1] Test times quoted are based on a *T86* of 4 seconds. Longer test times are required for longer *T86*'s

5.1 General Preparation

1) Connect the channels of a data collection device to the controller output (valve input) signal and *process variable* signals at the terminal strip. The *process measurement* will be used to determine conformance with the *dead band*, *step resolution,* total *hysteresis* and *process gain* specification limits. If the process measurement is a flow or pressure, this is ideal. Other measurements that imply flow, such as concentration or pH, can also be used. If the measurement is very slow, such as many temperatures, or if the measurement is an *integrating variable* such as a tank level, then it is advisable to install a pressure measurement downstream of the valve. If the fluid is a liquid, in some cases an ultrasonic flow measurement mounted externally on the pipe can be made to work. The process measurement can be filtered to reduce the effect of noise. A filter *time constant* of 20% of wait time between steps is adequate, as steady state changes of the process measurement are of interest. Set up to collect data at a rate of about 1 sample per second and start data collection.

2) In addition, install a valve *stem*, or valve *shaft* position measurement transducer. This transducer should have a *time constant* that is at least 20 times faster than the expected step response of the valve being tested (*T86*). Connect this signal to the data collection device. If the same data collection device is to be used for the process measurement and for the valve *stem*, then all of the channels should be collected at the fast rate dictated by the *stem* transducer. If a separate data collection device is to be used for the *stem* transducer, then this device should also measure the valve input signal in order to have the same time reference for each step change. Calibrate the transducer so that it agrees closely with the zero and span of the input signal. The *stem* or *shaft* transducer will be used to measure the *speed of response* of the *control valve system*. Set up to collect data at a rate at least 20 times faster than the expected valve step response time (*T86*).

3) Clearly establish the performance criteria which should apply to this *control valve system*, including: *dead band*, *step resolution*, *total hysteresis*, *minimum step size*, *maximum step size*, *T86*, *Td/T86,* O/S, *travel gain* limits, *Tss*, *flow limit cycle amplitude, flow gain, process gain, range of process gain*. Write down these values for future reference.

4) The control valve will be tested at or near the operating point at which it is working at the time of the test. The process operator must be consulted in order to determine the magnitude of excursions that is acceptable under the operating conditions. If it is a requirement to test the valve at a specific position, for instance at 10% open, then this must be done at the time when the valve can be safely operated at this opening.

5) At the control console or controller faceplate, prepare to initiate a series of steps. Put the loop in manual mode and allow the process to stabilize.

6) Each step in controller output (valve input) should be as abrupt as possible (square edged step). This is best done by entering a new value via a keyboard in the control system. Some control systems only allow the controller output to be slewed via up/down push buttons. This method does not provide a square edged step, but rather a ramp, and the consequence is that response dynamics will be inaccurate. If a slew button must be used, each step should be made as a single push of approximately the right duration to achieve the step size needed.

7) Test the data collection process, and ensure that high quality time synchronous data can be extracted to a common spreadsheet software program for data manipulation, interpretation and presentation. All the data manipulation and presentation in this document has been done using Microsoft^R Excel 97.

5.2 Test 1 - Initial Test

The purpose of Test 1 is to determine that the valve tracking appears to work and that further tests are warranted. Measure the input signal, the *stem* position and the *process variable*. Ideally a stem position transducer is available and parameters such as *T86* can be measured. If this is not the case and only the process measurement is available, the test is still worth doing, as at least it will determine that the process signal changes with input signal. Hence, it will bracket the *dead band* and will also provide a measure of the *process gain*. Carry out a few (at least two) step tests with an amplitude greater than the *minimum step size* (well inside *Region C*). Ensure that the step tests allow adequate time for any unusual behaviour to develop such as *ringing*. For this reason the wait time between steps should be at least a few minutes. Plot the resulting response as shown in Figure 6. The example shows a single downward step of –2.7%. Even though the *step response time T86* is an acceptable 2.5 seconds and the *overshoot* is under the 20% limit, the response exhibits excessive oscillations or "*ringing*" which causes the *time to steady state* (*Tss*) to be 70 seconds. This is 28 times *T86* and is unacceptable because the specification requires the ratio of *Tss* / *T86* to be 2.5 or less, in order to ensure a fairly quick settling mode. Calculate the weighting factors using Equations 1) and 2). (The result is 265 or very bad). A test result like this would suggest that further tests are not warranted until the cause of the *ringing* is cured. Proceed to the next test only if Test 1 is clearly successful.

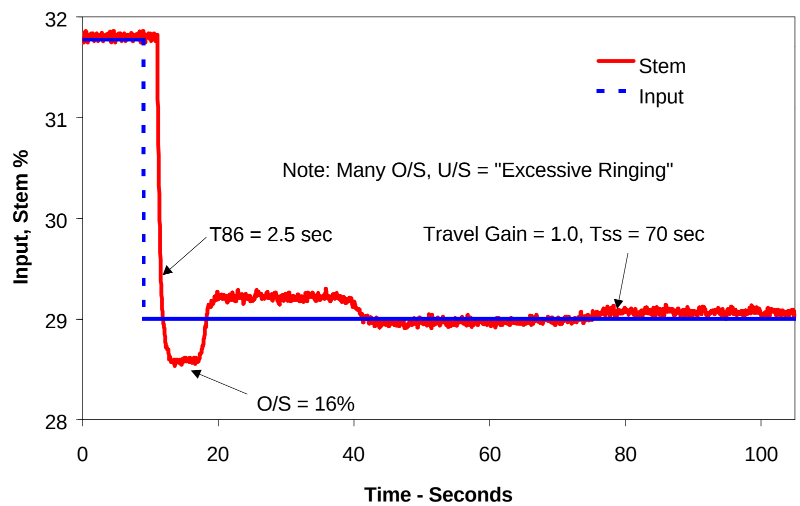

Figure 6 – Test 1 - Initial Test Example, Step size = -2.7%
***T86*=2.5 sec. O/S = 16%. Excessive Ringing. K_T=1.0. *Tss*=70 sec**

5.3 Test 2 - Increasing Step Size Test

The purpose of this test is to provide initial meaningful measurements of *minimum step size, dead band*, *T86, Td/T86* ratio, *overshoot*, travel gain, *Tss* and *process gain*. Measure the input signal to the valve, the *stem* or *shaft* position and the process signal. If the stem position measurement is not available, the test is still worth doing as at least it will determine the input signal step sizes for which the process signal changes, thus bracketing the bracket the *dead band,* and it will also provide a measure of the *process gain*. The test involves a sequence of step tests starting with 0.25% and doubling the amplitude successively until 8% is reached (or whatever maximum size the operator will allow) as shown in Figure 7. Each step is repeated four times in an up, down, down, up pattern. Ensure that there is adequate wait time between steps. In Figure 7 the steps are applied every 20 seconds. For the valve in Figure 7 the expected *T86* is 4 seconds, hence a 20 second wait time should be adequate. For slower valves, a longer wait time would be needed.

However, regardless of the expected *T86,* there are other considerations that should also be taken into account in choosing the wait time. The behaviour exhibited in the *Test 1 - Initial Test* should be taken into account. As well, the slowness of the process measurement should also be considered for there should be adequate time for the *PV* signal to actually change and settle. From the process measurement (*PV*) it will be possible to bracket the magnitude of the *dead band* effect, as it will be obvious that the *PV* did not change for some of the small steps, whereas it did for the larger ones. The *stem* position can be used to measure *T86*, *Td/T86* ratio, O/S and K_T and *Tss* for each test. Analyze each test and prepare a spreadsheet as shown in Table VII below. Calculate the weighting factors, using both Equations 1 and 2 (if loop application knowledge available), for the step tests which pass the *Region C* criteria and which fall into the step size range from minimum to twice the minimum step size. It is this range of input step sizes which best measures the regulation performance of the *control valve system*. Calculate an average value for all of the weighting factors. Use all of the step tests involved – not just the good ones. Table VII shows the analysis for 6 of the 24 step tests of Figure 7 as an example.

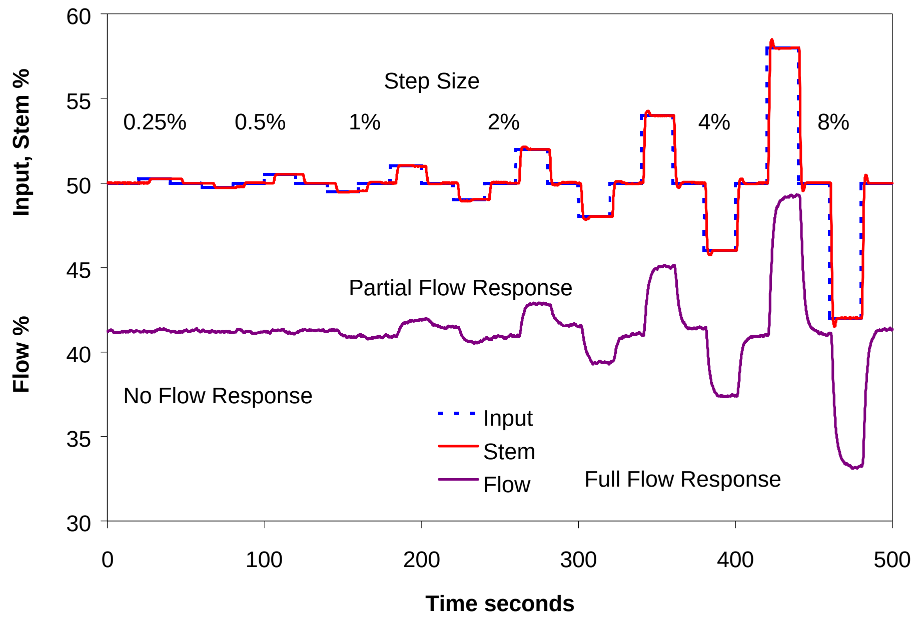

Figure 7 – Test 2 - Increasing Step Test

Table VII – Step Test Analysis

Specification Limits														
Dead Bnd	***Stp Res***	***Tot Hyst***												
0.50%	**0.30%**	**1.00%**												
	Min Step	***Max Step***	***T86***		***Tss/T86***	***O/S %***		***Tr Gain***		***Kp***			***Lambda***	
Spec	**2.00%**	**10.00%**	**4 or 2.5**		**2.5**	**20**		**0.8 - 1.2**		**0.5 - 2.0**			**10**	
Actual	**1.43%**													
												Valve Performance		
		W weighting factors=	30		8	0.75		50				**Index W**		
Step #	**Step Size**	***T86***	***Td/T86***	***Tss***	***Tss/T86***	**1 st O/S %**	**#o/s,u/s**	***Tr Gain***		***Kp***	***Region***	***W(T86)***	***W(Lamb)***	**Comment**
	%	sec		sec								Eqn 1	Eqn 2	
1	0.25	7.67	0.84	9.4	1.23	0	0	1.05		0.00	*A*			Hys> 0.25
6	0.5	5.95	0.82	7.5	1.26	0	0	1.09		0.00	*A*			Hys> 0.5
10	1	4.13	0.74	13	3.15	5	0	0.99		0.51	*B*			
14	**2**	**3.04**	**0.77**	**9.4**	**3.10**	**5**	0	**0.97**		**0.65**	***C***	**30**	**23**	**Hys < 2**
18	4	2.18	0.56	6.2	2.84	8	0	0.99		0.91	***C***	**20**	**11**	
22	8	1.96	0.48	5.1	2.60	8	0	1.00		1.01	***C***			
	Region C Average Values		**Avg=**		**Avg=**			**Avg=**		**Avg=**		**Avg=**	**Avg=**	
			0.60		**2.85**			**0.99**		**0.85**		**25**	**17**	
		Min=	**0.48**	**Min=**	**2.60**		**Min=**	**0.97**	**Min=**	**0.65**				
		Max=	**0.77**	**Max=**	**3.10**		**Max=**	**1.00**	**Max=**	**1.01**				

On the basis of Figure 7 and the analysis of Table VII, it is clear that the *total hysteresis* caused by *dead band* or *step resolution* is greater than 0.5% and definitely less than 2% (based on *process gain*). As the *minimum step size* has been set at 2%, the valve in question meets this aspect of the specification.

It is critically important to accurately estimate the actual minimum step size. This is the point at which *T86* vs. step size curve crosses the specification limit for *T86* (*Region* - example 4 seconds), and to also determine that this point is within the specification limit for the minimum step size (example 2%). Figure 8 shows a plot of the T86 versus step size results for the 24 step tests of Figure 7. The result is 1.43%, which indicates that this control valve system exceeds the specification limit of 2%.

It is natural to expect there to be a fair amount of scatter in the test results. It must also be expected that the point at which the T86 vs. step size test results cross the T86 limit is straddled by two sets of adjacent points. This can best be resolved by calculating a regression trend line as shown in Figure 8. The trend line should clearly pass through the adjacent clusters of test results.

The weighting factors have been calculated for step sizes ranging from the minimum of 2% to 4% (double the minimum step size). The responses are generally well shaped have fairly low valve performance index of 36, indicating quite a good result (zero is perfect and 100 is poor). However, the *Td/T86* ratio is clearly greater than 0.5 for steps of 4% and less, hence this is a consideration which would require the faster 2.5 second *T86* limit to be imposed, as opposed to the 4 second value in order to minimize *dead time*. On this basis, the test valve is borderline since the 2% steps have a *T86* of 3 seconds.

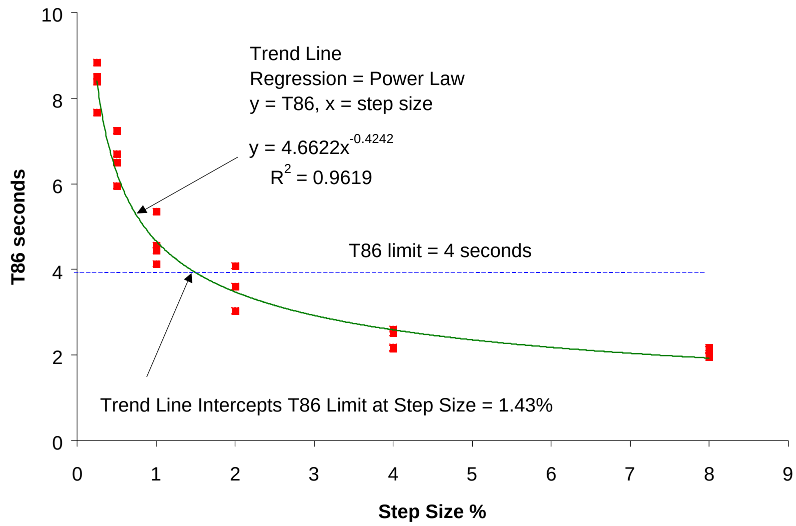

Figure 8 – *T86* vs. Step Size Plot for step test of Figure 7

5.4 Test 3 - Small Step Test

The purpose of the Test 3 - Small Step Test is to measure the *dead band* and *step resolution* as accurately as possible. Should a *stem* transducer not be available, this is the only accurate test that can be carried out. Measure the input signal and the *process variable*. There is no need to measure the *stem* position, as the step sizes will be less than the expected *step resolution* in magnitude, and only steady state changes are of interest. The signals should be filtered to reduce the impact of noise. A filter *time constant* of 10 seconds is adequate, as long as the wait time between steps is at least 50 seconds.

The test involves a sequence of small steps applied first in one direction, and then in the other as shown in Figure 9. The size of the steps must be smaller than the expected magnitude of either the *dead band* or *step resolution*, so that these parameters can be accurately bracketed. A suggested step size is one half of the smaller of dead band or step resolution. The valve in the Figure 9 example is the same as used in Figure 7. A step size of 0.2% was used in Figure 9. The absolute maximum and minimum values of valve position depend on the process operator. However, consideration must also be given to the number of steps and the length of the test. Figure 9 shows nearly 50 steps, each with a wait time of one minute, for a total duration of 50 minutes. As well, it is important to perform at least two full cycles of the input signal from minimum to maximum for adequate repeatability.

Figure 9 is relatively easy to analyze. The first three "up" steps caused the flow to change by 0.4%. Little can be said about this result as the valve *dead band* was simply being taken up in the up direction. After the first reversal in direction, it took five "down" steps to cause the flow to change. Based on this, the *dead band* is less than 1% (5 x 0.2%). After this first change in flow in the down direction, it took two "down" steps to cause a change in flow. Based on this the *step resolution* is less than 0.4%. This is also true of all subsequent changes in flow, which occurred in the same direction. All of the reversals in direction required 5 steps before the flow changed, except for the reversal which started at about second 1000, which required six steps. Hence the *dead*

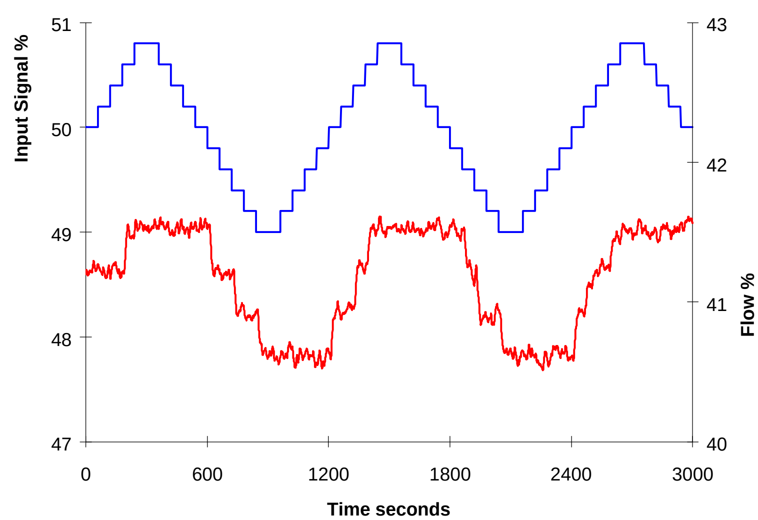

Figure 9 – Test 3 - Small Step Test

band is less than 1.2%. The largest value should be taken. The *total hysteresis* can be estimated as the sum of the *dead band* and step resolution, or less than 1.6% (1.2% + 0.4%).

Test 3 of Figure 9 involved nearly 50 steps, took 50 minutes, produced 13 changes in flow. It yielded the conclusion that *dead band* is less than 1.2%, the *step resolution* is less than 0.4%, and the *total hysteresis* is less than 1.6%. Given the **specification limits are: *dead band* less than 0.6%, *step resolution* less than 0.4% and *total hysteresis* is less than 1%**, the valve in question fails.

X-Y Plot

Another way to view the same data is an X-Y plot as shown in Figure 10. To generate such a plot involves estimating steady state data after each step change has settled. This involves taking a filtered value or an average of both the input signal and the *process variable* just before the next step is applied. Once this data is generated it can be plotted as an X-Y plot using most commonly spreadsheet software programs. The data shown in Figure 10 is the same as in Figure 9. The run starts at an input signal of 50.0% and a flow of 41.3%, and two full cycles are executed. If the valve were ideal, the flow would follow the "ideal flow" line as shown in Figure 10. This line goes through the initial point on the X-Y plot and has a slope equal to the *process gain* (*Kp*). However, Figure 10 shows a tendency for the input signal to change without causing any change in flow. This is a result of the *dead band.* From Figure 10, two estimates of the *dead band* can be made: 0.8% and 1.0%. Since 1% is the larger number, this should be reported. Once motion is initiated the relationship between input signal and flow is not very regular. If the *step resolution* were infinitely fine, the figure should have straight sides parallel to the ideal flow line. The irregularity of the figure is a measure of the *step resolution*. The *total hysteresis* is the total width of the figure as measured from the Low-Low line to the Hi-Hi line (lines passing through the points furthest away from the ideal flow). This is estimated as 1.12% in Figure 10, and is also the estimated value of the *total hysteresis*. The minimum *hysteresis* is the distance between the Hi-Low and Low-Hi lines (lines passing through the points nearest to the ideal flow). This is estimated as 0.41% in Figure 10. An estimate of the effective *step resolution* is half the difference between the total and minimum *hysteresis*, or 0.36% ((1.12% - 0.41)/2). Hence the final results based on Figure 10 are: a *dead band* of 1.0%, a *step resolution* of 0.36% and a *total hysteresis* of 1.12%. Based on this interpretation the valve in question fails the specification on *dead band* and *total hysteresis* limits*.*

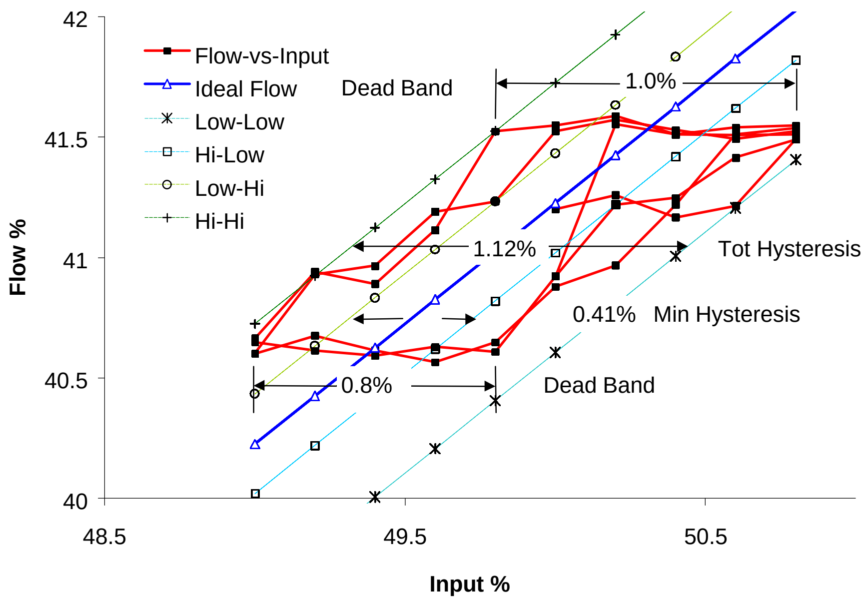

Figure 10 – X-Y Plot for Test 3 - Small Step Test of Figure 5. *Hysteresis*: Max = 1.12%, Min = 0.41%, Dead *Band*=1%, *Step Resolution*=0.36%, *Total Hysteresis*=1.12%

No matter what the results, the interpretation will be to some extent subjective. Valve users will wish to argue that slightly questionable results represent failure, while valve suppliers will argue the opposite. The key issue is that there are not very many tests available on which to base statistics. Hence, the interpretation of the data requires some judgment.

5.5 Test 4 - Medium Step Test

The Test 4 - Medium Step Test is intended to accurately measure the dynamic response of the valve system, specifically: *minimum step size, T86, Td / T86 Ratio,* ***%*** *overshoot*, % undershoot, travel gain, *Tss*. Measure the input signal and the *stem* or *shaft* position. Use the fast sampling rate for the measurements. A measurement of the *PV* is not required. The test is similar to the Test 2 - Increasing

Step Test, except that the step sizes for the test should start inside *Region B*. The number of tests depends on how many statistics are needed. The size of the steps should start at an initial value which is less than (one half) the minimum step size found during in Test 2, and the step size should increase up to a maximum value set by the process operator. In this way the steps will transition out of *Region B* into *Region C*. Hopefully, *Region D* will also be reached. If not, then the shape of the resulting *T86* vs. step size curve (Figure 4) should indicate if the *maximum step size* requirement could be met, based on the slope of the line. Special emphasis should be placed on step sizes that are typical of normal control loop regulation, and for which control valve performance index can be calculated. This involves step sizes ranging from the *minimum step size* (as found in Test 2 - Increasing Step Test) to double this value. A suggested number is 40 step tests. Suggested step sequence is shown in table VIII:

Table VIII – Test 4 - Medium Step Test Suggested Sequence

Step Size (n x Minimum Step Size)	Number of Steps	Calculate W
o.5	4	
1.0	10	Y
1.5	10	Y
2	10	Y
4	4	
8	4	
Maximum	2	
	Total = 40	

Each of the step tests should be analyzed as in Section 5.3, and a similar table should be built. Determine the *minimum step size* using regression as shown in Figure 8. If this value of *minimum step size* differs by more than 30% from the value used when starting this test, repeat the test using the new value to select step sizes. A figure similar to Figure 4 should be plotted. For the valve to pass the specification, there should be consistent dynamic responses in *Region C*, as defined by the limits on *minimum step size*, *T86*, *Td/T86 ratio*, travel gain, *overshoot* and *Tss*. Performance index weighting factors should be calculated for small step sizes as indicated.

6.0 Definition Section:

The terms defined below are common with previous EnTech control valve specifications and terminology adopted by ISA. Where possible, ISA terminology has been adopted in order to make the specification commonly understood.

<u>Amplitude Ratio:</u> for sinusoidal signals the ratio of output amplitude to input amplitude.

<u>Approximate Time Constant (t')</u>: approximate *time constant* for the *control valve system* step response, (*T86* divided 2).

<u>Backlash</u>: mechanical lost motion caused by looseness in a mechanism. *Backlash* causes *dead band* to be observed.

<u>Closed Loop Time Constant (Lambda (l)):</u> the *time constant* of a control loop as tuned. The user is free to tune a control loop to a desired l value in order to achieve process-manufacturing objectives. The value of l which can be safely achieved is limited by the *speed of response* of the process, the transmitter and the control valve.

<u>Closure Member</u>: the part of the control valve that is in direct contact with the flowing fluid (valve trim, plug, disc, ball etc.) and which causes progressive throttling of the fluid.

<u>Control Valve Assembly:</u> same as a *control valve system*.

<u>Control Valve Package:</u> same as a *control valve system*.

<u>Control Valve System:</u> a system consisting of control valve, actuator, positioner and any other components needed to allow the control valve closure member track the input signal.

<u>Dead time (T_d):</u> the time period after an input signal step change and prior to the start of a response. For practical purposes, *dead time* can be estimated by measuring the time that the response crosses 10% (T_{10}) of the full steady state change.

<u>Dead Band</u>: the range through which the input signal can be changed, without causing a change in output. *Dead band* results from various phenomena, such as *backlash* and *shaft* deflection, and causes the valve system to require extra input change after a reversal of direction before actual movement is resumed. In Versions 2.1 and earlier, this phenomenon was referred to as "*backlash*". The term *dead band* has been adopted to fully comply with ISA terminology.

<u>dB:</u> decibel, unit to measure attenuation of sinusoidal signals as 20 Log (Amplitude Ratio)

<u>Flow Coefficient:</u> the coefficient which determines the flow through a valve given the fluid, valve opening, upstream and downstream pressures (and temperatures for some fluid).

<u>Flow Gain (Kf):</u> for a step change in input signal, the ratio of the change in flow passing through the control valve and the change in input signal (flow units /% valve travel).

<u>Flow Gain % (Kf%):</u> the flow gain expressed as a percentage of the minimum design flow under a given operating condition (normally there are minimum, nominal and maximum flows design figures). (flow units /% valve travel / nominal flow) x 100%).

<u>Flow Resolution (Rf):</u> the minimum change in the fluid flow that the control valve can produce (*step resolution x flow gain).*

<u>Hunting</u>: a tendency for a feedback system to oscillate about its setpoint.

<u>Hysteresis</u>: the combined effect of *dead band* and *step resolution* that prevents changes in *flow coefficient* response in spite of movement in the input signal. Hysteresis is seen as displacement in an X-Y plot (see Figure 10) in the input signal dimension.

_Integrating Variable_*:* a process variable, such as tank level whose rate of change is determined by the net flow into a vessel of a certain volume. Moving a control valve causes the rate of change of the *integrating variable* to alter.
_Linear Dynamics_*:* a single set of dynamic parameters (gain, *time constants*) of a linear transfer function model.
_Limit Cycle_*:* a sustained control loop cycle or '*hunting*' caused by a nonlinear element in the loop such as the control valve. *Limit cycling* can also be caused by the integral action of valve positioners and can occur in the valve system itself.
_Nonlinear Dynamics_*:* dynamic parameters (gain, *time constants*) which change over the operating range.
Minimum Step Size: in this specification, the minimum user specified value of input signal step size for which the *control valve system* is to conform to the Dynamic Response Specification limits (lower limit of *Region C*). Also, value form test results.
Maximum Step Size: in this specification, the maximum user specified value of input signal step size for which the *control valve system* is to conform to the Dynamic Response Specification limits (upper limit of *Region C*).
Overshoot (o/s): the amount by which the step response initially exceeds the final steady state value (% of step change).
Process Gain (Kp): for a step change, the ratio of the change in *process variable* to the change in input signal (process change % of span / valve input change %).
Process Resolution (Rp): the minimum change in the *process variable* that the control valve can produce (*Rs x Kp).*
Process Variable (PV): the process measurement used for feedback control by the loop to which the control valve is connected.
Process Dynamic: the way a *process variable* responds, over time, to a change in controller output. This is best characterized by the step response, modeled by a transfer function and expressed in terms of *process gain*, *time constants* and *deadtime.*
Regions A, B, C, D: nonlinear regions defined as input signal step size varies from small to large, over which a *control valve system* exhibits different nonlinear behaviour. *Region A* – no movement, *Region B* – inconsistent movement, *Region C* – consistent movement, in which values are within specification limits, Region *D* – velocity limited movement.
_Ringing_*:* a tendency for a dynamic system to oscillate after a step change.
Self Regulating Process: a process whose step response reaches a new steady state value after an input step change.
Step Resolution (Rs): the minimum step change in input signal to which the *control valve system* will respond while moving in the same direction. The phenomenon is caused by the tendency for a control valve to "stick" after coming to rest. Sometimes referred to as *stiction*, or *stick-slip*. An ISA term for this does not yet exist.
_Shaft_*:* for a rotary valve the actuator member that transmits torque to rotate the *closure member*.
_Stem_*:* actuator member that forces the *closure member* to move through the range of valve travel.
Speed of Response: of a dynamic non-integrating system can be gauged by measuring the *time constant* of a first order model which approximates the dynamic response of the actual system. For process control, *T86* provides a fair "goodness of fit".
Step Response Time (T86)**:** the time after an input signal step change until the output has reached 86.5% of the final steady state value. Since a first order linear system reaches 86.5% of the step response value in two *time constants*, *T86* divided by two provides a useful *approximate time constant* value for the valve system.
_Stiction_*:* a term used in control literature meaning a tendency to *stick-and-slip*, due to the presence of high static friction. The phenomenon causes a limited resolution of the resulting control valve motion. ISA terminology has not settled on a suitable term yet, however the term Resolution has been proposed.
Stick-Slip: the tendency of a *control valve system* to stick while at rest, and to suddenly slip after force has been applied.
Travel Gain (K_T): the ratio of valve *stem* or *shaft* change (% travel) to valve input signal change. In some positioner designs logic exists to compensate for control *valve characteristics* (equal percentage, quick opening). In this specification the term *travel gain* implies a nominal value (for which 1.0 is the ideal) which has been normalized for any such compensation.
_Time constant_*:* the time required for a first order linear system to reach 63.2% of the full change after a step change.
Time to Steady State (Tss): time at which the *stem* position within 1% of its steady state value.
Total Hysteresis: term used in the specification to denote the combined effect of *dead band* and *step resolution*. Can be measured by performing Test 3 - Small Step Test - see Figure 10. Can be estimated as *dead band* plus *step resolution*.
Undershoot: amount by which the stem position falls below its final steady state value (% of step change) after an overshoot.
Valve Characteristic: the characteristic curve as the *flow coefficient* varies with valve travel (equal percentage, linear, etc.)
W(Lambda): valve step response performance index weighting factor calculated by Equation 2) which weights the relative step response performance in the presence of dead time, varying settling time, overshoot and travel gain, using the closed loop time constant Lambda as a basis.
_W(T86):_ valve step response performance index weighting factor calculated by Equation 1) which weights the relative step response performance in the presence of dead time, varying settling time, overshoot and travel gain, using T86 as a basis.

References

1) EnTech - Control Valve Dynamic Specification (Version 2.1, 3/94),
2) ANSI/ISA-S51.1-1979 - Process Instrumentation Terminology.
3) ISA Standard-S75.13-1989 Method of Evaluating Performance of Positioners with Analog Input Signals & Pneumatic Output.

For more information:

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