technical monograph 49

Control Valve Impact on Loop Performance

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Process Management

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THE CONTROL OBJECTIVE

A manufacturing company is in business to realize a profit through the sale of a quality product, where a quality product is defined as one that conforms to specifications. A deviation from the established specifications can mean lost revenue due to wasted materials and resources, the reprocessing of off-spec product to meet specifications, or selling off-spec product at a lower price.

Therefore it's a reasonable assumption that the ultimate goal of every manufacturer is to avoid any deviation, or at best, to minimize any deviation and its impact on the manufacturing process. To that end, control systems typically are put in place to yield products that are on spec, time after time, without variation.

UNDERSTANDING THE STATISTICS OF VARIATION

Within a manufacturing environment the expectation is that machines will operate as required and materials will be within their own tolerance limits.

However, reality shows that there are unique causes of variation, such as equipment malfunctions, fouling or operator errors. Even raw materials may have minor variations in characteristics and performance.

As each machine, process and raw material varies randomly within its own tolerance band, the cumulative effect on the overall process is a band of performance values (PV) that form a distribution about some average. The distribution of data typically follows a "normal" or "Gaussian" frequency distribution, which is the familiar bell-shaped curve shown below.

The process variable values are plotted along the horizontal axis, and the probability of occurrence of each particular value is plotted in the vertical direction.

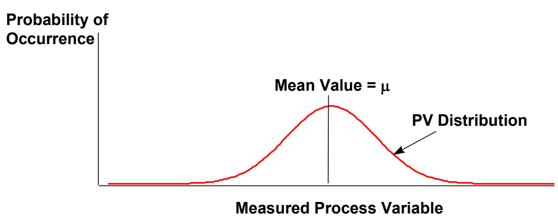

Figure 1.

The closer the data points are to the mean value (i.e., the set point.), the more frequently they will tend to occur. Conversely, those data points which represent larger deviations from the mean will tend to occur with less frequency.

On average, about the same number of deviations will be above the mean value as will be below it, thereby forming a curve that is symmetrical.

Mathematicians have several basic measurements or metrics they can use to describe the spread of the data, which is the amount of variation about the mean value. The metric most commonly used in the process control industry is standard deviation (σ), which often is simply referred to as "sigma."

$$\sigma = \sqrt{\sigma^2} = \sqrt{\frac{\sum (x_i - \mu)^2}{n-1}}$$

Equation 1: Standard Deviation (Sigma)

In layman's terms, standard deviation can be visualized simply as the average deviation from the mean.

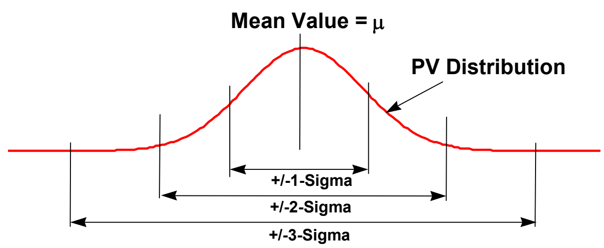

Figure 2.

Thus, sigma (σ) is a statistically derived parameter that describes the "spread" of the data about the mean value. The larger the value of σ, the greater the spread.

The area under the PV distribution curve represents the percentage of the total population between any two points on the curve. Thus, sigma is a parameter which tells how much of the total population is contained in a given region centered about the mean value:

- $\pm$ 1-sigma contains 68.26% of the total population
- $\pm$ 2-sigma contains 95.45% of the total population
- $\pm$ 3-sigma contains 99.73% of the total population

In performance testing, 2-sigma (2σ) is the metric used most often to indicate valve performance and to compare one valve against another in terms of reducing process variability.

For example, assuming a manufacturer wishes to ensure that at least 95% of product meets the minimum specification, the set point of the process must be adjusted to at least a 2-sigma distance above the lower limit specification as shown in the figure below.

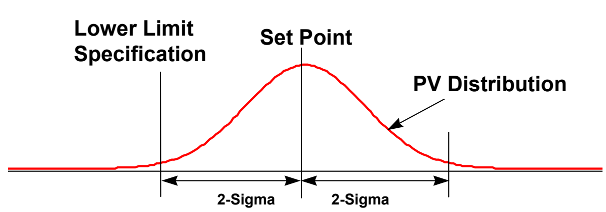

Figure 3.

This adjustment of set point will ensure that only 2.275% of the finished product will fail to meet the specifications on the first try. (Note that the tail of products above the 2-sigma limit on the upper end also meet the specification.) While this setting will produce acceptable results, it also means that 97.725% of the finished product is being produced at quality levels that exceed the lower limit specification. This may cost significant dollars in excessive catalysts, raw materials and energy use.

However, the manufacturer can reduce the degree to which products must exceed the specification limit by decreasing the “spread” of the distribution (i.e., going to a lower value of sigma as shown in the following diagram.)

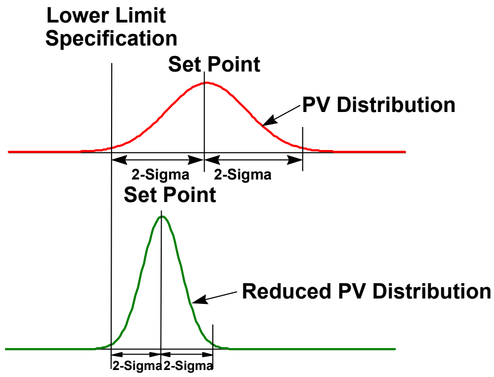

Figure 4. Reduced process variability

This is referred to as a reduction in “variance” or “process variability.”

This reduction in process variability allows the manufacturer to move the process set point closer to the lower limit specification, which;

- Produces a more uniform product
- Improves manufacturing efficiencies
- Improves profitability

Thus, reducing process variability is really the manufacturer’s goal.

Manufacturers tend to understand well that spending resources on elaborate systems of instrumentation are often justified to achieve this type of reduction in process variability. What is not as well understood is that significant reductions in process variability can be achieved simply by choosing the right control valve for the application and ensuring that the control loop is tuned for the most effective reduction of process variability.

Surveys have shown that as many as 80% of the control loops in plants are not producing the level of product uniformity of which they are capable.

The reason for this is twofold:

- The right process control equipment is not being utilized.
- The process control equipment that is installed is not being operated at optimum performance.

Research data show that process variability can be reduced by as much as 50% by the straightforward application of:

- Good engineering design practices
- Intelligent selection of valves and instruments
- Careful system installation
- Good instrument maintenance
- Proper loop tuning methods

In general, process variance can be reduced by:

- Maintaining stable loops
- Minimizing the effects of load disturbances
- Avoiding unwanted changes in loop gain
- Reducing the effects of nonlinearities in the loop
- Proper valve maintenance

THE IMPACT OF NONLINEARITIES

What does "linear" mean in the process control context? A linear dynamic element responds to its input signal in a uniform fashion over its entire operating range with unchanging dynamic behavior. It does so regardless of the size or amplitude of a change in the input signal. If the time constant of a step response remains the same for step inputs with amplitudes of 0.01%, 1%, and 100%, then the element is considered "linear." If the time constant varies with the amplitude of the input signal, the element is "nonlinear."

Nonlinearities become a major source of problems for control loop tuning and performance because the apparent process dynamics to which the control loop tuning was matched initially, keep changing with process conditions. As a result, the variability of the loop no longer remains stable. That is why it is imperative to find the source of nonlinearities in the process and eliminate or reduce them. Historically, the control valve is a good place to start.

Following is a list of typical valve nonlinearities:

- Friction
- Backlash
- Relay dead zone
- Spool valves
- Shaft windup
- Fluid turbulence
- Sample time in transmitters and controllers
- Processes which change gain with change in throughput
- Valve characteristics that change valve gain with throughput

Many of these nonlinearities are extremely complicated to handle or explain mathematically, but the general observation can be made that they all introduce undesirable phase shift and gain effects that place limitations on the performance of the system.

System stability is always a concern, and nonlinearities can definitely influence stability. However, a more insidious effect of nonlinearity is the undesirable influence it has on process variability.

As a result, these nonlinearities cause the need to be more conservative in tuning, which limits how much the process variability can be reduced. This is why it is so important to review the control hardware components of the loop first to determine what nonlinearities are present. Then effort can be made to reduce or eliminate them through more judicious component selection.

Friction

Friction is a force that tends to oppose the relative motion between two surfaces that are in contact. The friction force is a function of the force pressing these two surfaces together and the characteristic nature of the two surfaces. Friction has two components: static friction and dynamic friction.

Static friction is the force that must be overcome before there is any relative motion between the two surfaces. Stick/slip or "stiction" are colloquial terms that are often used to describe static friction. Static friction is one of the major causes of dead band in a valve assembly. Dead band is the range through which an input signal can be varied, upon reversal of direction, without an observable change in the output signal. (Note: This is not the same thing as dead time.)

Once relative motion begins between the two parts, static friction no longer applies. Now, dynamic friction is the force that must be overcome to maintain the relative motion. Running friction and sliding friction are colloquial terms that are sometimes used to describe dynamic friction.

Static friction is nearly always significantly larger in magnitude than dynamic friction. In other words, it usually takes more force to get an element to break out of its static condition and begin moving, whereas less force is usually required to keep it moving. This means that static friction is a much more significant problem in process control than dynamic friction.

Process disturbances do not cause errors that continually move in the same direction. They typically take the form of random oscillatory disturbances that are continually stopping and reversing direction. This means that the control elements also must continually follow this same stop-and-reverse-direction pattern. Every time the control element has to stop, static friction can become a factor.

While friction cannot be eliminated completely, good design practices can have a dramatic impact on the amount of friction present in a control valve. The places where friction can exist in a control valve include packing, guides or other sliding surfaces, bearings in rotary-motion valves, seals in balanced, cage-guided globe valves, seals in ball valves, and linings in butterfly valves. Actuators have friction in their guide bushings, piston seals, stem seals, and in motion conversion devices. Friction can be particularly significant if there is misalignment between stems and bearings or other sliding surfaces.

Piston actuators with very large, high-pressure O-Rings can contribute significant friction, particularly after they have been in service for awhile. O-rings typically are lubricated when first installed, but after only a few hours of operating in a typical high temperature environment, they tend to get stiff, and the friction factors can increase by as much as 400 percent or more.

__Backlash__
Backlash is the general name given to a form of dead band that results from a temporary discontinuity between the input and the output of a device when the input changes direction. Slack, or looseness of mechanical connection is a typical cause.
One of the biggest sources of backlash comes from lost motion between the power element of the actuator and the valve closure member. Linear motion valves are not measurably affected by backlash except when poorly designed stem connectors existf. In rotary motion valves, there are several points where backlash can occur. Linkage connecting the rotary valve to a linear actuator will have clearance at the joints, and wear will cause the clearance to increase resulting in more lost motion. Some rotary valves are fitted with a scotch yoke or rack-and-pinion drive mechanism that introduces backlash. Actuators that have gear trains are also subject to backlash in the gears.

__Relay Dead Zone__
A relay is a device that acts as a power amplifier. It takes an electrical, pneumatic, or mechanical input signal and produces a large volume of air or hydraulic fluid to the actuator. The relay can be an internal component of the positioner or a separate valve accessory mounted between the controller and the valve.

Most usually, relays are constructed like a miniature version of a control valve, and they suffer from the same type of dead band characteristics as a control valve. Since relays are small and an unimposing part of the overall valve assembly, they often are overlooked as a source of dead band that causes process variability.

__Spool Valve Dead Zone__
Many manufacturers of spool valve positioners intentionally build some dead band into their instruments to reduce the large amount of steady-state air consumption. In other words, some change in input occurs without any corresponding change in output because a movement of the spool valve is required to uncover a particular port before any output change can occur.

__Shaft Windup__
Shaft windup results when one end of a rotary valve shaft turns and the other does not. This typically occurs in rotary style valves where the actuator is connected to the valve closure member by a relatively long and small diameter shaft. While seal friction in the valve holds the control element end of the shaft in place, rotation of the shaft at the actuator end is absorbed by twisting of the shaft until the actuator input transmits enough torque to overcome the friction.

This phenomenon results in dead band since there is no output during this input motion. Of course, if the input signal was to reverse direction, the shaft would simply untwist and begin twisting up in the opposite direction until the friction could be overcome. Until then, there would be no change in the output at all.

Dead band is one of the worst nonlinearities facing the process control engineer. When a load disturbance occurs, the process variable will deviate from the set point by some small percentage. This deviation will attempt to initiate a corrective action through the controller and back through the process. Unfortunately, if this initial change in controller output percentage is smaller than the process dead band, it produces no corresponding corrective change in the process variable. Only when the controller output has changed enough to progress through the dead band does a corresponding change in the process variable occur.

Any time the controller output reverses direction the controller signal must again pass through the dead band before any corrective change in the process variable will occur. The presence of dead band in the process essentially ensures that the process variable deviation from the set point will have to continue to get larger and larger until it is big enough to get through the dead band. Only then can a corrective action occur.

Since most control actions for regulatory control consist of small changes (e.g., 1% or less), a control valve with excessive dead band may not even respond to many of these small changes. At a minimum, a well-engineered control valve should be able to respond to signals of 1% or less to provide effective reduction in process variability. Ideally, the valve should be able to respond to signals as low as ¼ %. Unfortunately, it is not uncommon for audits to show a high percentage of valves with dead bands ranging from 2% to as high as 5%.

Regardless of its source, dead band in a valve assembly is undesirable when it comes to maintaining good process variability. This should be self-evident, because when a disturbance produces an error in the process variable, the valve assembly cannot correct for the error until the error becomes large enough to get through the dead band of the valve and produce a corrective action.

DEAD BAND MEASUREMENT

In any practical sense, it is not possible to measure the effects of all these dead band sources individually. What is important to know is the cumulative effect on the process performance. This brings up two extremely important points:

1. **The cumulative effect of all the dead band sources acting together must be measured.**

2. **The cumulative dead band in a realistic live process must be measured.**

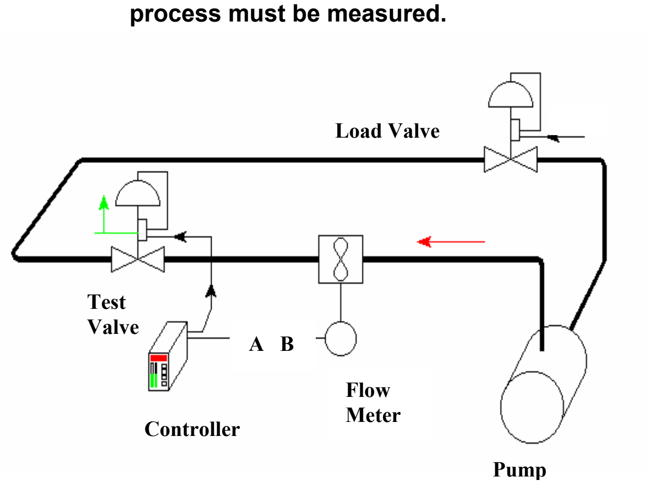

Figure 5: Open-loop, step-test flow loop

Figure 5 shows a standard test system that Emerson uses to perform cumulative dead band measurement tests on a realistic liquid-flow process. The test loop provides a method for measuring the dead band performance of a variety of test valves under realistic dynamic load conditions. It is imperative that these tests be done in a consistent manner so that direct comparisons can be made from one test valve to another under identical conditions.

Water is used in a 4-inch loop to ensure fast process response and eliminate compressibility effects. Emerson also has other loop sizes for testing smaller control valves. For the purposes of this paper only the larger loop for high flows is discussed.

The pump is programmed to provide constant flow at either of two different flow levels, one being a simulated high flow process condition and the other a simulated low flow process condition. The load valve is a high performance, Fisher, cage-guided, sliding-stem valve that is fixed in the appropriate position for the selected load flow condition. The load valve is a permanent fixture of the loop, along with the pump, the flow meter, and the controller.

A turbine meter is used to measure the flow to ensure accuracy and speed of response. Notice, however, that the flow meter signal is only measured at the point labeled (B) in Figure 5. This signal is not sent back to the controller to regulate the flow. This is an open-loop test setup. The reason for this will become clear later.

The test valve represents the piece of hardware to be tested for dynamic performance. Once it has been thoroughly tested, it will be removed from the loop and another test valve will be inserted and subjected to identical tests. This may be another Fisher valve model or it may be a valve from another manufacturer. It makes little difference. They are all treated the same. When the test valve is installed for testing, it is installed, adjusted, and calibrated carefully and precisely according to the manufacturer's instructional literature that came with the valve. This is to ensure that the valve has a legitimate opportunity to perform as it was designed.

The controller is a Fisher digital valve controller that is capable of sending a pure step signal to the test valve ranging in magnitude from as small as .125% or smaller to a full 100%.

The Objective

The objective is to determine if the *complete* valve assembly is capable of responding to small disturbances in the process variable and if it can make the necessary corrective actions without interference from dead band in the valve assembly. To determine this, the valve assembly must be installed in the loop under normal flow conditions.

To provide precise measurable, controlled test conditions, the controller is placed in manual (hence, open-loop) so that a series of small step inputs of variable size can be applied to the valve assembly.

The step inputs are a series of two steps up and two steps down at levels of 0.5, 1.0, 2.0, 5.0, and 10 percent as shown in Figure 6. During this test, the input signal is recorded, the stem position signal is recorded, and the process variable (flow) signal is recorded on the same chart for comparison.

This test does not make a precise measurement of the dead band with this test, but it easily and accurately brackets the magnitude of the dead band by comparing the process variable (flow) response to the input signal. If the process variable movement faithfully follows the two-step input, then the dead band of the assembly is less than the percent input applied. For example, the process variable may not faithfully respond to the 0.5% signal, but does respond faithfully to the 1.0% signal. This indicates that the dead band of the assembly is somewhere between 0.5% and 1.0%, which is normally expressed as being less than 1.0%.

It is important to recognize that the process variable (not the stem position) is the important parameter to observe when making dead band measurements. Some manufacturers incorrectly use bench test stem position measurements as the key parameter when reporting dead band of their valves.

With some valves, such as rotary valves, it is possible for the actuator stem position to change without any corresponding change in valve movement. From a process variability point of view, simply changing the stem position will not make a correction to the process variable until the valve disk actually moves. For rotary valves, the majority of the valve assembly dead band is usually in the valve shaft, bearings, and disc seals. These are often highly affected by the process pressure and flow and would be totally ignored in a bench test dead band measurement test.

The Procedure

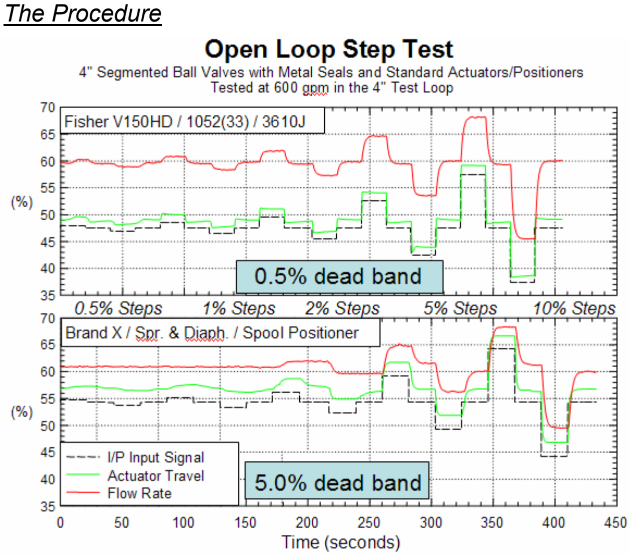

Figure 6. Open-Loop Dead Band Measurement Test

As seen in Figure 6, the step inputs are applied in an alternating series of two steps up and two steps down, beginning at input levels of 0.5 percent, and then progressing to 1.0, 2.0, 5.0, and 10 percent. During this test, the input signal is recorded, the stem position signal of the test valve is recorded, and the process variable (flow) signal is recorded on the same chart for comparison.

Actually, Figure 6 contains the charts for two separate dead band tests. The top chart illustrates a valve with a dead band performance ten times better than the valve in the bottom chart. This is a dramatic illustration of how dead band can make a major difference in the dynamic performance of two different valve assemblies.

Notice in the top chart that the process variable (flow) is responding faithfully to each of the input steps, including the 0.5% steps. Every time the input moves, the process variable changes accordingly. There is no way of knowing how small the dead band really is, but from this test it can be concluded that it is 0.5% or better for the top valve.

On the other hand, the valve test illustrated in the bottom chart shows that the process variable (flow) doesn't respond at all to either the 0.5% or the 1.0% input steps. There is finally some response to the 2% steps, however flow is not responding to all of the steps. It isn't until the input steps get to 5% that the process variable responds faithfully to each of the input steps. Again, the precise dead band is not known, but it must be somewhere between 2% and 5%. In most cases, this would classify as a 5% valve assembly.

Noted earlier was that some manufacturers determine the valve assembly dead band based on stem motion being the output, rather than process variable. This tends to make the valves look better than they really are in actual practice. It should be obvious that stem motion alone cannot correct for an error in the process unless there is a corresponding motion of the valve disk.

In the bottom valve pictured in Figure 6, note that the valve stem responds to the 0.5% signal changes, but there is absolutely no change in the flow through the valve. Using stem motion to rate the assembly for dead band would suggest that the valve was capable of 0.5% performance, when in reality it is only good for 5% performance as far as affecting process variability.

DEAD BAND GUIDELINES

These guidelines are based on process performance measurements using open-loop step tests as indicated in Figure 6.

Valve Style	Total Dead Band (%)
Sliding-Stem	0.5
Rotary	1.0

Table 1: Dead Band Guidelines for Control Valves

DEAD BAND VERSUS HYSTERESIS

Hysteresis is not an important factor when dealing with control valves, but it will be treated here anyway because there is so much confusion about the distinction between dead band and hysteresis. These two terms are often lumped together or even used interchangeably as though they were the same thing, but they are not. They are really two different phenomena.

Pure Dead Band
Figure 7 illustrates a case where the tested element is perfectly linear except for some pure dead band.

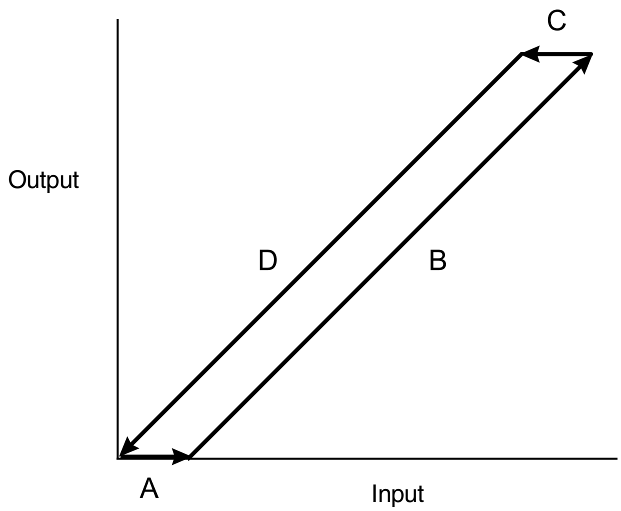

Figure 7. Pure dead band element

The curve in Figure 7 is generated by a quasi-static test performed by making small incremental input changes to the device and then waiting to see what change occurs in the output as the result of each input change. Making a whole series of these stop-and-go tests and recording the input versus output for each point generates the curve.

When in the dead band zones of the device (zones A and C), there is no corresponding change in the output for each incremental change in the input. Once through the dead band and the output does start to respond. It does so in a linear fashion as long as the input signal continues to change in the same direction.

On the other hand, as soon as there is a change in direction with the input, the input has to go through a period of dead band again where there is no corresponding change in the output (Zone C).

The reason for this dead band zone typically can be attributed to some kind of looseness in a mechanical linkage between the input and the output. An example might be a linkage pin in an enlarged, out-of-round hole. In this situation, when the input tries to change the output, nothing happens until the linkage pin comes in contact with the edge of the hole. The output then begins to change linearly with the input. This continues until the input moves in the opposite direction, and the pin must then once again move through the looseness in the linkage before anything happens at the output. Some individuals refer to this type of dead band as “backlash.” It is also a common problem in many types of gear assemblies, particularly when the gear teeth become worn.

Pure Hysteresis
Figure 8 illustrates a case where the tested element is perfectly linear except for some hysteresis.

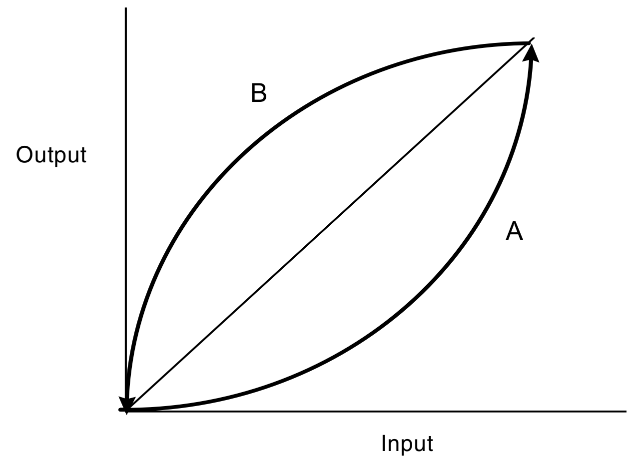

Figure 8. Pure Hysteresis Element

It is important to note that each change in input does not cause a corresponding change in the output.

Like the dead band curve, the hysteresis curve in Figure 8 is generated by making small incremental input changes to the device and then observing the change that occurs in the output as the result of each input change. Plotting the input versus output for each point generates the curve in Figure 8.

It is important to note that there is one characteristic that clearly distinguishes hysteresis from dead band. With pure hysteresis there is a change in output for each change in input. Note however, that the relationship between the input and output is not a linear one. Further, when the input is decreasing (B) the relationship is not the same as when the input is increasing (A).

The reason for the nonlinearity of these curves and for the separation between the up and down curves is that some of the input energy is being converted to other forms of energy (heat, for example) and is not available to produce a change in output. The separation between the up and down curve is an indicator of the dissipated energy called “hysteresis.”

Typical sources of hysteresis in control valves are such things as springs, I/P electromagnets, positioner bellows, actuator diaphragms, etc. A simplistic way to think about the energy dissipation that occurs with hysteresis is to visualize the molecular friction internal to the device. For example, repeatedly and rapidly flexing a metal spring causes it to become warm. This is due to the energy being dissipated in the spring due to hysteresis.

The hysteresis effect is particularly pronounced in transformers for example where the alternating current is rapidly and frequently changing the magnetic field in the core.

Applying these same general principles to all of the devices listed here helps in understanding why they are sources of hysteresis, but there is one important point of caution that needs to be pointed out to keep things in perspective.

While hysteresis may be present to some degree in control valve assemblies, it is usually a VERY MINOR problem, almost to the point of being negligible. Consider, for example, how infrequently and slowly the springs, bellows, diaphragms, etc. in a control valve get flexed during the course of their normal operation. Can you imagine that any of these elements are any warmer than their surroundings due to hysteresis generated heat? If you're not certain, try touching an actuator spring the next time you're out in the plant. Believe me, dead band is a far more serious problem than hysteresis.

DEAD BAND VERSUS RESOLUTION

The small-step, open-loop test is another way to look at dead band of a valve. It can provide a more precise measurement, and it also has the advantage of being able to measure the resolution of the valve assembly. The test uses the same test setup as shown in Figure 5 except that the test is conducted in a different manner as shown in Figure 9.

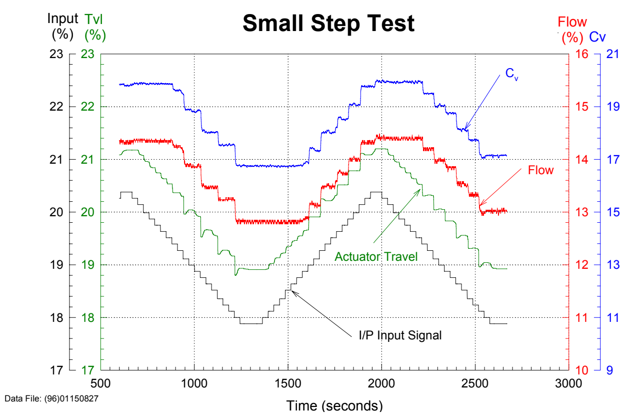

Figure 9. Small step test for measuring dead band and resolution

In this test, the valve assembly is subjected to a series of uniform step changes in the I/P signal input that are each .125% in magnitude. This is a quasi-static test with pauses of 30 seconds after each step.

The bottom input curve represents 20 small input steps in one direction with about a 2-minute pause and then taking 20 steps in the other direction.

After the change in direction occurs, dead band can be measured as the number of small steps required at the input before the flow signal begins to change direction. For this valve, it would appear that the dead band is approximately nine small steps, or 1.125%. This is a much more precise measurement than the technique that was used earlier.

Resolution on the other hand is a measure of how sensitive the valve is to incremental changes when moving in the same direction. On both the decreasing steps and the increasing steps for this valve, it would appear that after the valve has gotten through the dead band, it takes approximately three small input steps before the valve responds with a step change in the flow. This valve would be characterized as having approximately .375% resolution.

DEAD TIME

When most process control operators talk about controlling the "process," they are talking about everything in the loop except for the controller (i.e., they are using the controller to control everything else in the loop collectively.)

For purposes of loop tuning, these processes can be represented by a first-order lag response with the addition of some dead time. Without dead time in the loop, the response to a step input to an open loop, such as that shown in Figure 5, would be a smooth exponential increase to some final value, with no overshoot.

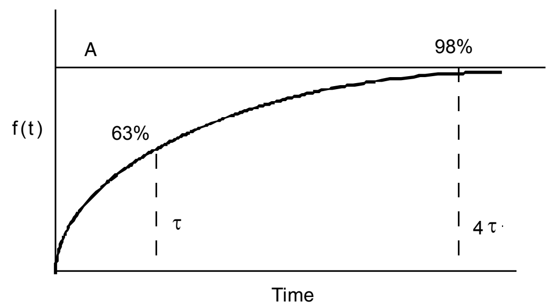

Figure 10. First order lag with no dead time

How fast this response occurs would be an inherent characteristic of the process, which in Figure 5, includes the test valve. The parameter used to identify the speed of response of any first-order lag is called "Tau," the time constant (τ).

$$f(t) = A\left[1 - e^{-\left(\frac{t}{\tau}\right)}\right]$$

Equation 2. First Order lag equation with no dead time

To gain an appreciation of what this time constant means, let the time parameter (t) in equation (2) be equal to the time constant (τ). By doing so, the response at that time will be equal to 0.63A. In other words, at the end of one time constant of time, the output response will be at 63% of the final value of the response to the step input. This suggests that we have a convenient way to measure the time constant of any process (i.e., simply subject the process to a step input and then measure the time it takes for the output response to achieve 63% of the final steady-state value.) However, it is not quite that easy due to the presence of dead time.

While on the subject, consider letting the time variable (t) be equal to four times the time constant (i.e., $t = 4\tau$.) At this point the output response will have reached 98% of its final value. In other words, the response to the step input is nearly over.

Previously mentioned was that most real processes have some dead time in addition to the first order lag characteristics. Dead time is one of the most common nonlinearities in any process loop.

Dead time, as might be supposed, is time when nothing appears to be happening due to transport lag times, dead zones in a relay, long digital sampling times, etc. These types of dead time will exhibit themselves on the measured response as follows, where T_d is the dead time.

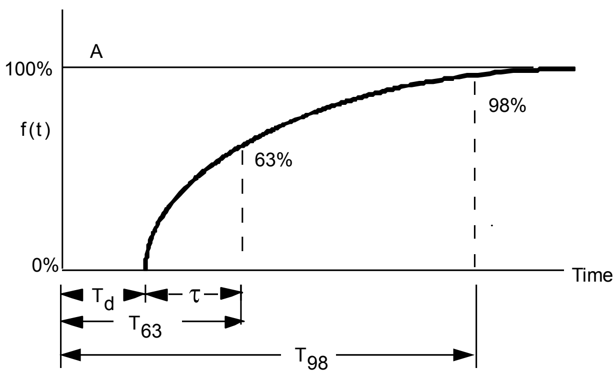

Figure 11. First order lag with dead time

The above diagram provides an opportunity to discuss terms that frequently cause some confusion. These terms are Tee-63 (T_{63}), Tee-98 (T_{98}) and dead time (T_d). T_{63} represents the time in which the response function reaches 63% of the final response, including the dead time.

The dead time (T_d) is the time that occurs after the step input in which there is no output response. This diagram should make it clear that T_{63} and τ are not the same unless there is no dead time since T_{63} includes the dead time.

By the same token, T_{98} also includes the dead time. T_{98} represents the time in which the response function reaches 98% of the final response. Use of T_{98} as a measurement parameter should be discouraged not only for the reasons mentioned previously, but also for the deleterious effects that dead time has on the first-order characteristics of the response. In many processes, dead time can be a rather significant factor compared to the time constant of the process. In these cases it can begin to severely destabilize the loop, amplify the disturbances, cause overshoot, and make measurements such as T_{98} relatively meaningless.

MEASURING DEAD TIME

The same set up shown in Figure 5 can be used to measure the dead time of the process.

The presence of dead time requires that some precautions be observed to ensure the most accurate representation of the process.

Due to any dead band nonlinearity present, the system is likely to respond differently to a decreasing step input than it would to an increasing step input.

Therefore, a two-step change procedure is used to deal with the dead band. One step is designed to position the valve at the dead band and the second step is designed to take the valve through the dead band.

To ensure that the system is positioned properly to begin an increasing step input test, the system should be subjected to an up-down/down-up cycle before conducting the first actual up-step measurement.

Figure 6 illustrates this kind of two-step test. Figure 12 takes one of the steps from this two-step test and amplifies the scale to make it easier to read. (Because of its large size, it is placed at the end of this paper.) For simplification, notice that only the valve assembly input signal (i.e., the controller output) and the process variable (i.e., the flow rate) are shown.

To measure the dead time, first estimate the time when the input signal was applied (approximately 239.9 sec) and the time when the output response appears to begin (approximately 240.1 sec). The difference between these two numbers is the dead time (i.e., T_d = 0.2 sec.)

Finding the time constant (tau) first requires finding the time when the output (flow) reached 63% of its total change, once it actually started to change. The first step is to determine how much change actually occurred in the output flow. Using a straightedge to draw an asymptote to the flow curve in Figure 12 provides an estimate that the final flow rate is approximately 65.2%. Since the change began at 60.4%, this gives a total output change of 4.8%. Finding 63% of this change gives (0.63)(4.8%) = 3.0%. Adding the beginning 60.4% to this means that the 63% point is reached at a reading of 63.4% on the chart. This occurs at a time of approximately 242 sec.

To determine tau, subtract the time when the output first began to respond (240.1 sec) from the 63% time (i.e., τ = 242.0 - 240.1 = 1.9 sec.)

The T_{63} time is determined by adding the dead time to the time constant (i.e., T_{63} = 0.2 sec + 1.9 sec = 2.1 sec.)

It is important to note that because of nonlinearities in the valve assembly, these parameters will likely be different for decreasing steps than they are for increasing steps. Also, accuracy can be improved by taking more than one measurement in each direction. For this reason, it is common to make determinations of these parameters for at least two different up-steps and two different down-steps and averaging the values together to obtain the most realistic set of parameters.

To ensure further accuracy, this entire sequence should be repeated at least twice to obtain a total of three up-step measurements.

Three down-step measurements are also obtained during this same sequence.

Final values for the process dead time (T_d) and the process time constant (τ) are then obtained by averaging the six measurements (three up and three down) for each of these two parameters.

RESPONSE TIME GUIDELINES

Valve Size		T_d	τ
(inches)	(cm)	(sec)	(sec)
0-2	0-5	0.1	0.2
>2-6	>5-15	0.2	0.4
>6-12	>15-30	0.4	0.8
>12-20	>30-50	0.6	1.2
>20+	>50+	0.8	1.6

Table 2. Guidelines for dead time and time constant

It should be noted that some guidelines in the past have listed values for T_{63} instead of τ. This practice can give unfair advantage to the equipment being measured. Since dead time is included in the T_{63} measurement, any unused margin in the dead time measurement would automatically and unfairly accrue to the T_{63} measurement. Both dead time and time constant should stand on their own, independent of each other.

GAIN

Maintaining proper loop gain is the secret of good control. It is what allows the loop operator to reduce the process variability by clustering the measured process variable measurements tightly around the set point as shown in the lower diagram of Figure 4. It is important not only to find the proper loop gain, but it is imperative to maintain that loop gain as steady as possible over a wide range of operating conditions.

To illustrate what is meant by loop gain, return to the example used to generate the chart in Figure 12. A 5% step input was applied at the controller input (point A in Figure 5), which is shown in Figure 12 as the step input going from 64.8% to 69.8%. This represents a 5% disturbance that has been introduced into the steady state flow rate. The controller and the rest of the elements in the loop will try to correct for this disturbance.

The loop responds by making a first-order type flow response (at point B in Figure 5) that has been plotted in Figure 12. To measure the gain of this loop response, the apparent final value of the process variable (approximately 65.2%) is subtracted from the starting value (approximately 60.4%) to get a total output change of 4.8%.

Since the input change was 5% in this example, the loop gain is simply the ratio of the output change over the input change (4.8/5.0 = 0.96.) In other words, the "sensitivity" or gain of the loop is such that it removed 96% of the disturbance in one pass around the loop.

If the loop were to be instantly reconnected at points A & B in Figure 2, the remaining 4% of the disturbance would go around the loop again and another 96% would be removed. This process would continue until all evidence of the original disturbance was removed or until it became so small that it was dissipated by pipe friction or other energy losses in the system.

The lower the loop gain, the longer it takes the loop to bring the system back into stable control. The ideal, of course, would be to have the loop gain be equal to a value of one so that the disturbance is completely eliminated on the first pass around the loop.

What happens if the loop gain is greater than 1.0? Again return to the test setup in Figure 5 and put in a 5% step input at the controller input (Point A). Assume that the loop responds with a flow change of 10% at Point B. This results in a loop gain of 2.0. The initial 5% disturbance is now a 10% disturbance. If the loop was instantly reconnected as done previously, the disturbance would go round and round the loop, getting larger and larger each time.

While this appears bad on the surface, there may be some extenuating circumstances that will help out. In real life operations, it is not likely to experience a single, unidirectional step disturbance in isolation. Instead, there will be literally thousands of disturbances of various magnitudes that are both increasing and decreasing in a random pattern. Figure 13 shows a load disturbance recorded from a typical plant. Many of these will cancel each other out. Others will likely be of such low magnitude that they may be consumed by friction or other energy losses in the system.

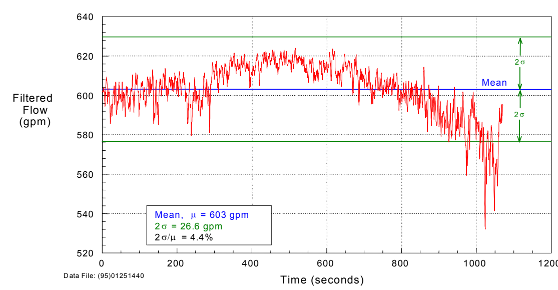

Figure 13. Typical plant load disturbance

Most control experts agree that a control loop can operate in a relatively stable mode as long as the loop gain does not get any higher than 2.0 for a period of time. If it gets any higher than that, the loop is certain to become unstable. Keep in mind that the loop will not provide as good a process variability at this gain setting as it would at an ideal gain of 1.0.

At the low end of the gain curve effective control requires that the loop gain should never go below a value of 0.5. To simplify the application of control valves the term “control range” references the range of valve travel over which a control valve can maintain the installed loop gain between the normalized values of 0.5 and 2.0.

NOTE: Normalized means that the values are expressed as a ratio of percent rated output versus input spans.
For years the industry has relied upon terms such as rangeability, turndown ratio, etc. to judge the ability of a control valve to provide proper control. In actuality, these terms are meaningless for this purpose since they have nothing to do with the issue of control. Instead, the term "control range" directly relates to the issue of good control.
Theoretically, the loop could be tuned for optimum performance at some desired set point flow condition. However, since the gain of a typical loop changes with flow, process variability optimization becomes difficult. There is also danger that the loop gain might change enough to cause instability, limit cycling, or other dynamic difficulties.

To overcome these difficulties, a means must be found to compensate for these changes in loop gain and maintain a relatively constant gain over the entire range of flows. As a minimum, it is important to stay within the range of 0.5 to 2.0. This compensation is often referred to as "characterization."

By far the best process performance occurs when the required flow characterization is obtained through changes in the valve trim rather than through use of cams or other methods.

VALVE CHARACTERIZATION

The concept of valve characterization has been around for a long time and may well be one of the most talked about concepts in the industry. At the same time, it may well be one of the least understood. There are so many legends and rules-of-thumb that have developed over the years that it is hard to tell what is useful and what is not.

First, exactly is meant by valve characterization (i.e., the things that characterize the flow through a given valve.) Obviously, the size and style of the valve, as well as its internal flow geometry make a difference in flow.

Other factors that can make a difference are external, such as the type of flowing fluid and its properties, the inlet pressure to the valve, the pressure drop across the valve, and the valve travel (valve opening.)

INHERENT VALVE CHARACTERISTICS

When a valve manufacturer, such as Emerson, develops a new valve style, they take a representative sample of the new valve and perform a valve characterization test according to established industry standards. During this test, all the factors previously mentioned are held constant except for the valve opening. The flow-versus-travel results (with constant pressure drop) are plotted to obtain what is called the *inherent characteristic curve.*

Inherent means that it is part of the "constitution or essential character" of the device. In other words, the valve will always exhibit this same flow characteristic no matter when or where it is tested, or regardless of who tests it, *as long as it is tested under the same conditions with constant pressure drop.*

Although there have been various minor modifications over the years, there are only three basic inherent flow characteristics which are illustrated in Figure 14.

Before studying these inherent characteristic curves, it is useful to define what is meant by the term "valve gain." As always, gain is defined as the ratio of the change in output over the change in input. From the perspective of the control valve, the input is a change in the stem position (i.e., the travel.) The resulting output is a change in the flow. Thus the valve gain is simply the ratio of the change in flow divided by the corresponding change in travel. A close look at Figure 14 shows that this is simply the definition of the slope of the curve at any point.

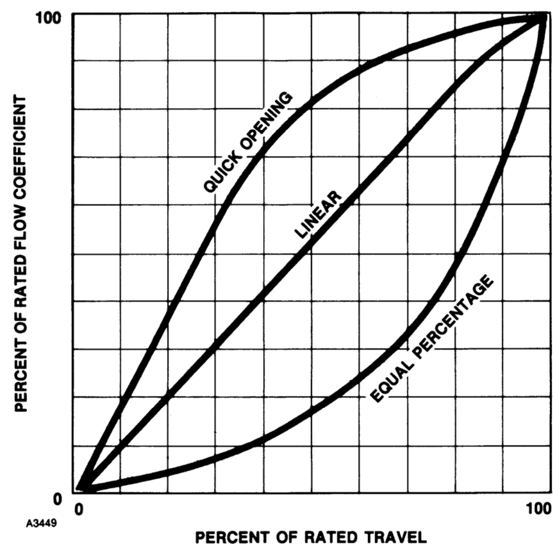

Figure 14. Inherent flow characteristics

The *linear flow characteristic* curve shows that the flow rate is directly proportional to the valve travel. This proportional relationship produces a characteristic with constant slope so that with constant pressure drop, the valve gain will be the same at all flows. The linear valve characteristic is commonly specified for liquid level control and for certain flow control applications requiring constant gain.

The *quick-opening flow characteristic* provides for maximum change in flow rate at low valve travels with a nearly linear relationship with a slope about 1.5 times that of the linear characteristic for the first 70% of flow capacity. Additional increases in valve travel give sharply reduced changes in flow rate, and when the valve nears the wide-open position, the change in flow rate approaches zero as it asymptotically increases to full flow capacity. In a control valve, the quick opening valve plug is used primarily for on-off service, and is often found in regulators and relief valves where the valve can establish significant flow quickly with minimum travel.

This characteristic is sometimes used in applications where a linear characteristic should normally be specified. This is not a good idea because of the dramatically wide variation in the valve gain over its travel range. It goes from very high gain at lower travel range to virtually zero gain at high travel ranges. This could make it very difficult to keep the loop gain within the 0.5 to 2.0 control range.

With the *equal-percentage flow characteristic*, equal increments of valve travel produce equal percentage changes in the existing flow. The change in flow rate is always proportional to the flow rate just before the change in valve opening is made. Reference to Figure 14 shows that when the equal-percentage valve is nearly closed, a 10% increase in travel produces very little increase in flow, but at greater flows, each 10% increase in travel produces greater and greater changes in flow, with a dramatic increase in flow during the final 10% travel.

A flow characteristic should not be considered inherent to a particular style of valve because many globe style valves, for example, have internal trim parts (plugs and cages) which can be easily changed to produced any of the characteristics shown in Figure 14. This is not so easy for other valve styles, however. For example, rotary valves, such as high-recovery ball valves and butterfly valves typically will exhibit inherent equal-percentage characteristics and there is no way to modify trim parts to change that. One caveat applies here. It has been common practice in the industry to select line-sized butterfly valves in many applications. As a result, these valves are typically over-sized, which means small movements of the valve disk will produce large changes in flow. Thus, for all outward appearances, the valve acts like a quick-opening valve.

INSTALLED VALVE CHARACTERISTICS

In actual service, the pressure drop across the valve is likely to vary all over the place. To understand how varying pressure drop can change the valve's flow characteristics, consider a simple example of a linear inherent characteristic valve.

Figure 15 illustrates a number of linear flow versus travel curves for constant pressure drops across the valve. Logic suggests that at any given valve travel, the highest pressure drops across the valve would give the greatest flow. The steeper the curve, the higher the pressure drop as indicated in the drawing. Since the pressure drops are held constant throughout the travel, the flow gain remains constant. The result is linear flow curves throughout the range regardless of the pressure drop, as long as the pressure drop is held constant.

To illustrate how varying the pressure drop can affect the flow characteristic, assume that the same linear valve body is installed in a process where the valve pressure drop (ΔP) decreases with load flow. In Figure 15, a locus of points has been plotted that defines the flow rate as a function of travel. These points are found by selecting the appropriate ΔP curve for each increment of travel.

Connecting this locus of points generates what is known as the *installed valve characteristic* curve. Note that this installed valve characteristic curve is not linear, even though the valve itself is *inherently* linear.

The *installed* characteristic is not the same all the time, even for the same valve, since it depends entirely upon the application in which the valve is installed.

This example illustrates that the installed valve characteristic is both a function of the inherent valve characteristic and the process characteristics. While in this instance a linear inherent characteristic was used for simplicity, exactly the same procedure can be used for each of the other inherent characteristics.

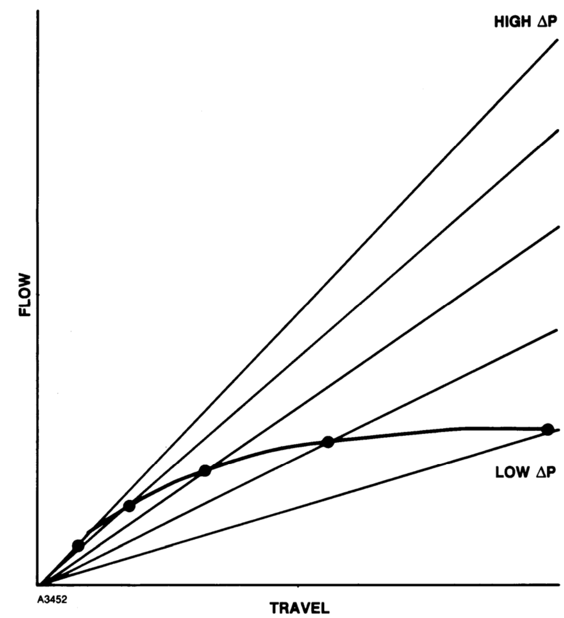

Figure 15. Installed characteristic curve for linear valve with decreasing pressure drop

INSTALLED VALVE GAIN

In the discussion of inherent valve characteristics it was stated that flow gain (flow/travel) was simply the slope of the inherent characteristic curve. This is true for the installed gain as well. In most cases, it is actually the installed valve gain that is of most interested. In Figure 15 the installed gain can be obtained directly from the measured slope of the installed valve characteristic

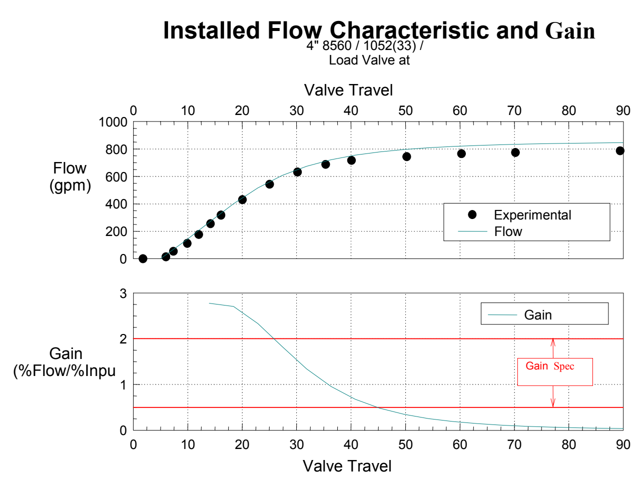

Figure 16. Installed flow characteristic and installed gain curves

curve. In fact, this is often how the installed gain is obtained for the entire process loop. A flow versus valve travel measurement is made in a loop similar to Figure 5, the data analyzed and the slope (valve gain) calculated at each valve travel. This data is then plotted on a graph similar to Figure 16. The top curve is the installed characteristic flow curve. The dots represent actual measured data sample points and the solid line is the flow curve calculated from the measured data sample points.

Once the installed characteristic flow curve is determined, the curve is used to calculate the slope (installed gain) at each travel point. Installed gain versus travel is plotted below the installed characteristic flow curve.

On the lower graph, there are three horizontal lines. The middle line shows the ideal gain of 1.0. The other two lines, at gains of 2.0 and 0.5 show the upper and lower limits of the control range that was defined earlier.

Figure 16 represents an eccentric disk valve with an installed flow characteristic curve that is far from linear, and the installed gain deviates considerably from the ideal value of 1.0. The effectiveness of good control is lost when the installed process gain gets outside the control range.

Note that the eccentric disk valve in this process only has a control range of about 20 degrees of travel (from 25 degrees to 45 degrees.) In the higher travel region, above the control range, the installed flow curve has flattened out, and the installed gain is approaching zero. There is virtually no ability to control at all in this region. Wide ranges in valve travel can be made with essentially no change in flow.

Conversely, at the lower end of the valve travel, below the control range, the installed gain is tending to get very high. This makes the loop so sensitive to changes that loop instability is a real threat.

What this means is that the eccentric disk valve has been incorrectly applied in this process. Two things could be wrong. The first is that the wrong valve style has been used. The second is that the valve may be incorrectly sized for the application. Oversizing of valves is often a major problem in the industry, partly due to a tendency to use line-size valves.

It cannot be overemphasized that valve style has a major impact on the installed process gain and control range. In general, globe valves tend to have the widest control range and the best performance in terms of process variability. Test results show that globe valves have nearly triple the control range of conventional butterfly valves.

Conventional butterfly style valves tend to have the narrowest control range and the lowest performance in terms of process variability reduction. Other valve styles tend to fall somewhere in between the two extremes of conventional butterfly valves and globe valves.

EFFECT OF VALVE SIZING ON CONTROL RANGE AND INSTALLED PROCESS CHARACTERISTICS

As stated, proper sizing of the control valve also plays an important role in achieving a wide control range and a good installed process characteristic.

It is possible to take a globe valve capable of achieving a control range of 70% of its travel range and reducing it down to only 20% of its travel range simply by oversizing it. This is what often happens when line size valves are used instead of using one size smaller with swages.

Regardless of its inherent flow characteristics, when a valve is oversized, the installed flow characteristics tend to appear more like a quick opening valve. As soon as the disc leaves the seat the result is a lot more flow than normally would be expected. The valve doesn't have to travel very far before it provides all the flow that is needed. Thus, the control range is very small, and the sensitivity or flow gain is very high.

A valve is much more effective and controls much more smoothly when the control range covers a larger percentage of the available travel.

SUMMARY

There are several key points to remember when working to achieve effective loop performance:

- Control valves can play a major role in reduction of process variability in the loop.

- The importance of dead band and dead time cannot be over emphasized when it comes to reducing process variability.

- Proper valve selection means managing the installed gain by matching the valve to the process. For example, conventional butterfly valves have a very narrow control range compared to other valve styles.

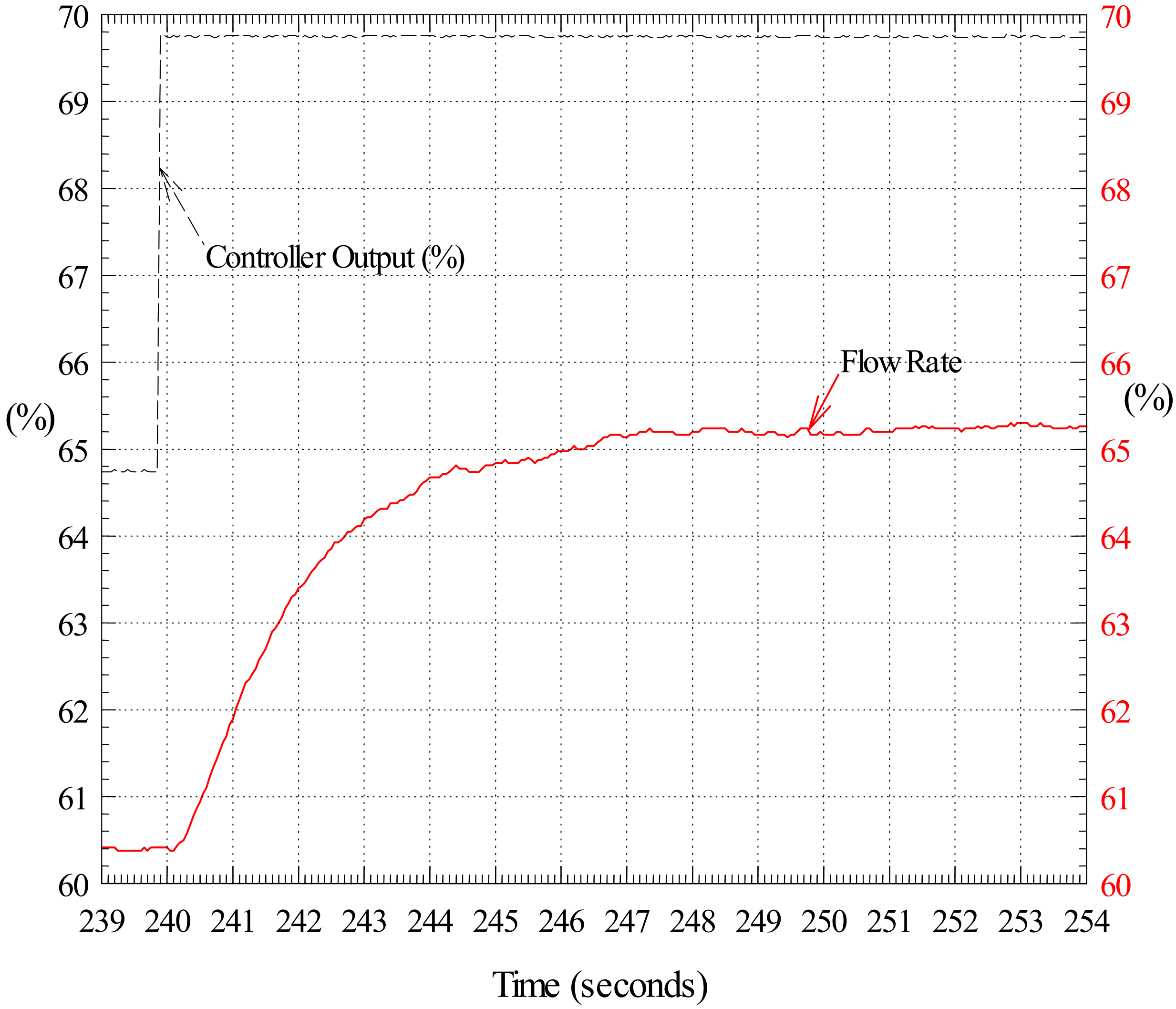

Figure12. Dead time and time constant measurement

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