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Fundamentals of Three-Mode Controllers

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This paper provides a basic insight into what a controller does and how it does it. The three modes of control are simply explained. Emphasis is upon understanding the fundamentals rather than specific hardware. Finally, a simple procedure for the proper tuning of a controller is stated and thoroughly explained.

Fundamentals of Three-Mode Controllers

Introduction

Controllers in one form or another have been around the process industries for a number of years. In fact, they are such a familiar sight in most industrial operations that they frequently suffer from being taken for granted. Yet, the quality of performance provided by a control system is determined by the performance of the controller and the other elements in the loop. The controller, with its various adjustments, is the one element in the control loop that allows any measure of operating flexibility. For optimum performance, it is necessary to use the controller properly. This requires a thorough understanding of some fundamental relationships.

Most standard controllers have three common modes of operation:

1. Proportional mode
2. Reset or integral mode
3. Rate or derivative mode

These three modes of control can operate individually or in various combinations, regardless of whether the controller is pneumatic or electronic. Thus, it is possible to have a three mode controller which combines proportional plus integral plus derivative control modes. This is frequently designated as a PID controller. A two mode controller combining proportional plus integral control modes is called a PI controller.

The Control Problem

Before discussing the intricacies of controller hardware, the control problem itself can be examined by looking at the behavior of an elementary gas pressure system as in Figure 1. The object of this system is to maintain the controlled pressure (P_2) within limits while supplying the load with the flow that it requires.

The regulator is placed upstream of the valve or other device that represents the load on the system. This load is frequently of such a nature that it is continually varying its demand for gas from the regulator. If the load flow decreases, then the regulator flow must decrease also; otherwise, the regulator will put too much gas into the system and the pressure P_2 will rise. On the other hand, if the load flow increases, then the regulator flow must increase also. If it does not, P_2 will fall off due to a shortage of gas in the pressure system.

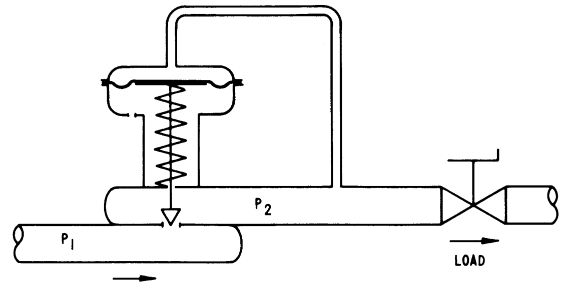

Figure 1. Elementary Control System

During steady-state conditions, when the load demand is not changing, the valve plug is stationary. The system is in equilibrium with the pressure force on the diaphragm balancing the spring force. For simplicity, forces on the valve plug due to the pressure drop across the port are assumed to be insignificant. This assumption in no way affects the conclusions that are drawn from this analysis.

The pressure force on the diaphragm (F_d) is developed by the sensed pressure (P_2) acting against the diaphragm area (A). The relationship between these variables can be expressed in a simple equation.

$$F_d = P_2 A$$

If the area of the diaphragm (A) is assumed to be relatively constant, the only way to change the force on the diaphragm (F_d) is by changing the pressure (P_2). If the symbol—delta (Δ)—is used to represent a change in a variable, then the equation can be rewritten to express the change in diaphragm pressure force as a result of a change in pressure.

$$(\Delta F_d) = (\Delta P_2) A$$

The spring force (F_S) is developed from the compression (X) that exists in the spring. The amount of force that is developed is a function of the spring rate (k), or stiffness of the spring.

$$F_s = kX$$

Since the stiffness of the spring (k) is a constant, the only way to change the spring force (ΔF_S) is to change the compression (ΔX).

$$(\Delta F_s) = k(\Delta X)$$

If the valve plug in Figure 1 moves through its full travel (T), the spring must change its compression by the same amount ($\Delta X = T$). The change in spring force that results from this full travel is

$$(\Delta F_s) = kT$$

Remember, however, that the pressure on the diaphragm is balancing the spring force! If the spring force changes as a result of full travel of the valve plug, then the diaphragm force (F_d) also must change by the same amount.

$$(\Delta F_d) = (\Delta F_s)$$

Since the area is constant, the change in diaphragm force is a direct function of the change in the diaphragm pressure (ΔP_2).

$$(\Delta P_2)\,A = kT$$

This fundamental equation can be rearranged by dividing both sides by the area (A). The result is an equation that tells how much the pressure on the diaphragm must change (ΔP_2) in order to allow the valve plug to move through its full travel (T).

$$(\Delta P_2) = \frac{kT}{A}$$

In Figure 1, assume that the load demand is very small and that the valve plug is operating very near its seat. In this condition, the system has been adjusted so that the downstream pressure (P_2) is 30 psig. This is the pressure that we are trying to maintain in the system. This pressure is called the SETPOINT pressure. As long as the load demand remains low, the valve plug will be in equilibrium near its seat and the downstream pressure will remain at the 30 psig setpoint value.

Now, suppose that the load demand abruptly increases to its maximum. To supply the needed gas to the system, the regulator valve plug must move to its wide open position. The only way for this to happen is to decrease the pressure on the diaphragm. Since the downstream pressure (P_2) is acting on the diaphragm, P_2 must decrease enough to allow the valve plug to open wide.

The amount that P_2 must decrease to fully open the valve is called DROOP or OFFSET, since the controlled pressure is offset by this amount from the setpoint. The amount of offset is a function of the particular hardware involved and can be calculated from the previous formula.

$$(\Delta P_2) = \frac{kT}{A}$$

For the purpose of supplying a numerical example, assume the following hypothetical system parameters and calculate the amount of offset that occurs.

k = 1200 lbs./in. (spring rate)

T = 0.25 in. (total valve travel)

A = 60 in^2. (effective diaphragm area)

$$(\Delta P_2) = \frac{(1200\ lbs./in.)\ (0.25\ in.)}{(60\ in.^2)}$$

$$= 5\,psi.\ (full\ load\ offset)$$

In most systems, if there was a full load offset of this amount, we would say that the system had rather low sensitivity, or that the loop gain was low, or perhaps that the proportional band was rather broad. All of these expressions are used to describe a control system that has a rather insensitive response to load changes.

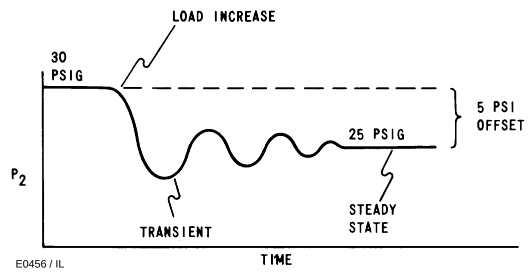

Figure 2. Variation in Controlled Pressure After an Abrupt Full Load Upset

Figure 2 illustrates how this example system responds to the abrupt step change in load that has been introduced. Because the system is underdamped, the controlled pressure (P_2) tends to drop a lot more than the calculated 5 psi during the transient condition. Remember, this calculation was based upon a steady-state equilibrium situation. Following the transient condition, the system eventually comes to equilibrium with the steady offset of 5 psi.

There are two very important points that can be inferred from this example. First, remember that the offset occurs because the pressure on the diaphragm must decrease in order to allow the valve plug to open and accommodate the increased demand for gas flow. This means that a small increase in load flow will only require a small amount of offset to move the valve plug; whereas, a large increase in load flow will require a correspondingly large offset. As the offset equation implies, the amount of offset (ΔP_2) required to open the valve plug the proper amount is proportional to the change in load. For this reason, this type of control action is referred to as PROPORTIONAL CONTROL. Secondly, because the offset is needed to open the valve for the larger flow, it should be obvious that the offset must continue to exist as long as the load continues at the larger flow rate. The controlled pressure (P_2) will only return to the setpoint value when the load returns to its original setpoint flow rate.

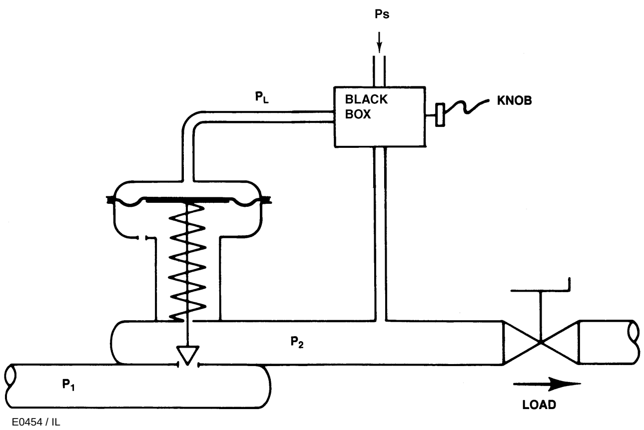

Figure 3. Black Box Added to the Control System

Thus, with proportional control an offset must always exist anytime that the load flow is something other than that for which the system was tuned or adjusted.

Now revise the system slightly, as shown in Figure 3, by adding a black box device in the controlled pressure line to the diaphragm. This black box device has three principal connections and one adjustment knob. The purpose of this box is to provide an output loading pressure signal (P_L) that is some function of the input measured pressure (P_2). The third connection is simply a supply pressure (P_S) input which provides a source of energy to operate the black box. Sometimes this source of supply (P_S) is obtained from the upstream pressure (P_1) and other times it is obtained from a separate compressed air source.

The knob on the black box is used to adjust the sensitivity of the relationship between P_2 and P_L. Assume that the knob has been adjusted for a sensitivity or gain of ten. This means that a change in controlled pressure (ΔP_2) results in a change in loading pressure (ΔP_L) that is ten times as large i.e., (ΔP_L) = 10 x (ΔP_2). The black box provides a GAIN in amplitude of the pressure change. For the present adjustment of the knob, there is a gain of ten.

Now, return to the previous control problem. Assume that the system is operating at a very low load flow and the controlled pressure (P_2) is at the setpoint value of 30 psig. If an abrupt full load flow change is experienced as before, the valve must open wide to pass the greater flow. As before, the loading pressure (P_L) on the diaphragm must decrease by 5 psi to allow the valve plug to open wide. Even though the diaphragm loading pressure change (ΔP_L) must still be 5 psi, the controlled pressure change (ΔP_2) is only 0.5 psi because of the gain of ten in the black box. As a result, there is a terrific improvement in performance as seen by comparing Figure 4 with Figure 2.

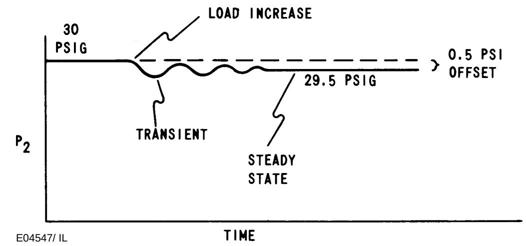

Figure 4. Improved Performance Due to Higher Gain

By adding the black box with its gain of ten, we can say that we have increased the loop sensitivity, or that we have raised the loop gain, or perhaps that we have narrowed the proportional band. Figure 4 shows that this increased loop gain not only improves the quality of the steady-state control, but it also greatly improves the quality of the transient control.

Eureka! We have found the secret of good transient and steady-state control! All that is needed is to make the loop gain as high as possible by adjusting the knob on the black box. If the loop gain could be made high enough (like infinite), there would be no deviation from the controlled pressure (P_2) regardless of load changes. Obviously, we can't obtain infinite gain, but it certainly seems at first glance that the ideal situation would be to have the loop gain as high as possible.

The Control Dilemma

But, wait! Don't be too hasty. Imagine how the system might behave if it did have very high loop gain.

High loop gain simply means that any signal variation or disturbance will be greatly amplified as the control loop tries to correct for the disturbance. In a steady-state operating condition there is no disturbance to the system, hence there is nothing to amplify, and the high loop gain can cause no problem. In fact, it works to our advantage because it minimizes the steady-state offset.

When a disturbance does come along, high loop gain also tries to improve the transient performance by greatly amplifying the feedback signal which causes the system to respond quickly and accurately. If the loop gain is too high however, the system is far too sensitive to these disturbances and it has a tendency to overcorrect severely. This results in a system that oscillates back and forth in an unstable manner.

The system can be returned to a stable condition simply by lowering the gain of the loop. This makes the system less sensitive to the disturbances and the tendency to overcorrect is eliminated. But, lowering the gain does result in poorer transient and steady-state control.

It appears that we have a definite control dilemma! Keep the loop gain high for good performance and risk an unstable system or lower the loop gain for stability and suffer the poorer system performance.

What To Do?

It was seen previously that in the steady-state condition with no disturbances to amplify, the system would tolerate the high loop gain and achieve the benefit of minimum steady-state offset. Furthermore, if a rapid load disturbance occurs, it can also tolerate the high gain *momentarily* in order to help the valve plug response. This will greatly improve the transient response of the system, but as soon as the valve plug starts to move the gain must be cut enough to keep the system stable.

Using the gain adjustment knob on the side of the black box, the gain of the loop can be varied to avoid the control dilemma! Start by adjusting the knob for high gain in the steady-state to take advantage of the good steady-state control accuracy. When a load disturbance occurs, momentarily keep the gain high to get good initial transient response, then quickly lower the gain enough to keep the system stable. Finally, as the disturbance begins to die out, slowly increase the gain again in order to decrease the steady-state offset.

Theoretically, one could manually operate the system in this manner and have the best of both worlds, but he would have to stay continually on the job and keep his reflexes sharp. Fortunately, it is possible to build a simple device that will do this job automatically. It is called a three mode controller!

Whether electronic or pneumatic, all conventional three mode controllers operate on exactly the same principle; i.e., the principle of automatically varying the gain of the system, when a disturbance occurs, to obtain good, stable control. Let's investigate how a controller does this job by looking at a typical pneumatic controller.

Proportional Control

The basic nozzle-flapper amplifier is a common sight in industry. It's the key element in many different types of pneumatic instruments. Figure 5 shows the basic operation of this important device.

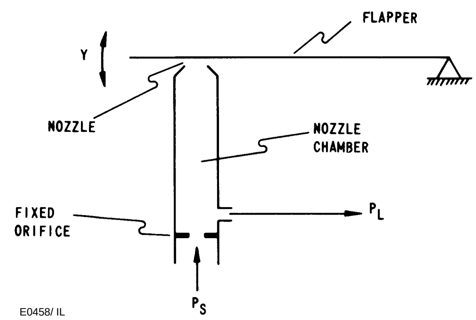

Figure 5. Basic Nozzle-Flapper Amplifier

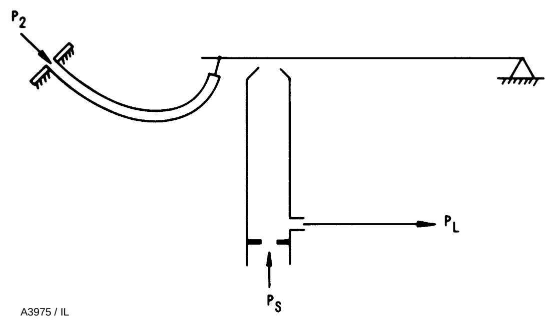

Figure 6. Nozzle-Flapper Amplifier with Bourdon Tube Sensor

A source of supply pressure is fed into the nozzle chamber through a fixed restriction. It is immaterial where this supply pressure (P_S) comes from. Most frequently it is obtained from a separate compressed air system which provides a source of energy to many different instruments at the same time. A nozzle opening, which is larger than the fixed orifice, allows the flow to escape. As the free end of the pivoted flapper undergoes an input motion (Y), the flow area of the nozzle changes and varies the pressure in the nozzle chamber accordingly. This nozzle pressure can then be used as an output loading pressure (P_L) to some other device.

To convert this nozzle-flapper amplifier into a pressure controller, some type of pressure sensor, such as the Bourdon tube shown in Figure 6, can be incorporated into the device. The Bourdon tube is curved and has an oval shaped cross-section. One end of the tube is fastened securely to the frame and is open so that it can receive the input controlled pressure (P_2). The free end of the tube is sealed and is attached to the movable end of the flapper.

As the input pressure (P_2) increases, the Bourdon tube tries to straighten itself and results in a flapper motion that tends to restrict the flow out of the nozzle. This causes a resulting increase in the output loading pressure (P_L). A typical system such as this is designed so that the output pressure (P_L) is rather sensitive to changes in the input pressure (P_2). In other words, it is a rather high gain pneumatic amplifier.

As the device now stands, the high gain that exists is a constant. It cannot be changed or adjusted without changing the mechanical design. In order to make this

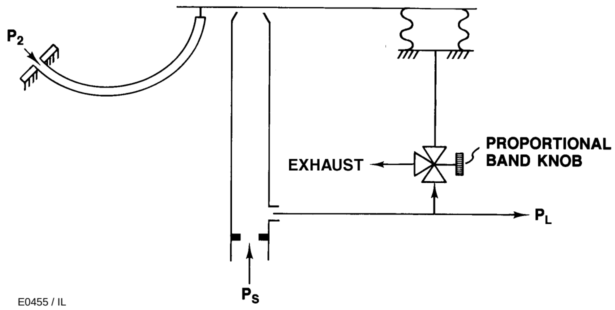

Figure 7. Pneumatic Proportional Controller

unit useful as a controller, some type of adjustment must be provided for the gain as indicated in Figure 3. A conventional method of providing for adjustable gain on this device is shown in Figure 7.

The fixed pivot on the flapper is removed and replaced by a bellows. The output loading pressure (P_L) is tapped and fed to an adjustable three-way valve. By adjusting this valve, the percentage of pressure signal(P_L)**)** that is sent to the bellows can be varied. In essence, the three-way valve acts as a splitter. Part of the output signal going to the three-way valve is fed to the bellows and the remainder is exhausted to the atmosphere.

When a change occurs in the input pressure (P_2), the action of the nozzle-flapper changes the output pressure (P_L). Part of this output pressure change is fed to the bellows in such a way as to *decrease* the effect that P_2 has upon P_L. This is technically known as *negative* feedback. The more negative feedback used, the more the sensitivity or gain of the device is decreased.

As an example, assume that an increase in P_2 occurs. This causes a restriction of the nozzle flow which increases P_L. Part of the increase in P_L is fed to the bellows. This backs the flapper away from the nozzle thereby decreasing the effect of P_2 upon the system.

If the gain control is adjusted so that none of the change in P_L is registered in the bellows, then the system will respond just as the high gain, fixed pivot system in Figure 6. As more and more of the changes in P_L are fed back to the bellows, the gain of the unit is cut accordingly. The negative feedback adjustment on the controller is called the GAIN or PROPORTIONAL BAND control.

The unit shown in Figure 7 is a typical pneumatic proportional controller. This device is the black box that was added to the system in Figure 3. When it is installed in an operating system, it must always have some offset in the controlled pressure (P_2) in order to accommodate changes in the load flow. For this reason, it is called a *proportional* controller. The offset

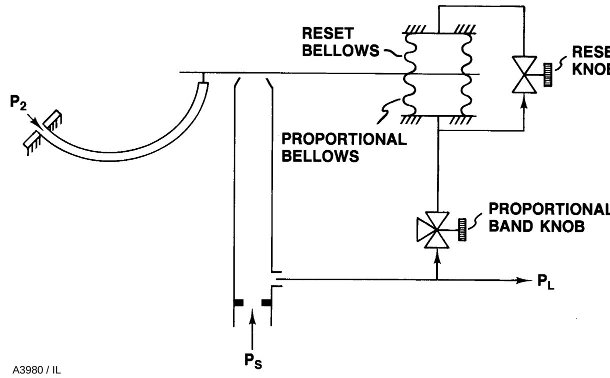

Figure 8. Proportional Plus Reset Controller

required is *proportional* to the amount of change in load flow. The amount of offset required in any case can be decreased by increasing the gain of the system. If the gain is sufficiently high, the steady-state offset can be reduced to a negligible amount. The only catch in this logic is that problems with stability are encountered.

The way to solve the instability problem with the proportional controller is to adjust the gain to a lower value; i.e., broaden the proportional band setting on the controller. When this is done, we must accept the larger steady-state offset that results from the lower gain.

Reset Control

The main reason for adding the proportional bellows in Figure 7 was to provide a means for cutting the gain of the controller. A previous section suggested the desirability of being able to automatically vary the gain of the controller when a load disturbance occurs. The second bellows shown in Figure 8 provides a means of accomplishing this.

The second bellows, known as the reset bellows, is located opposite the proportional bellows. Its operating characteristics match those of the proportional bellows. If the pressures in these two bellows are the same, one cancels the effect of the other, and again the system has high gain. Indeed, this is the real secret behind the action of a proportional plus reset controller.

Figure 8 shows that the pressure being sent to the proportional bellows is also sent to the new reset bellows. Before this pressure can reach the reset bellows, however, it must first pass through an adjustable needle valve known as the reset control. It should be noted that this reset control is *not* a three-way valve like the gain control. The sole purpose of this reset valve is to provide an adjustable time delay: the more restriction adjusted into the valve, the longer it will take a change in pressure to register in the reset bellows.

The signal which goes to the reset bellows is technically known as *positive* feedback. The more positive the feedback, the greater the increase in the sensitivity or gain of the device. In the present example, the positive and negative feedback signals are canceling each other in the steady-state condition. In the transient condition, the reset restriction temporarily delays the positive feedback signal so that the negative feedback can achieve its gain cutting effect.

A simple illustration will help explain how this system works. Assume a steady-state operating condition with everything in equilibrium. In this situation, the pressures in both bellows are equal and are canceling each other. Also, a high gain exists to provide the best possible steady-state accuracy.

If a step decrease occurs in the load flow of the pressure control system, the controlled pressure (P_2) increases and attempts to close the control valve. This increase in P_2 registers in the Bourdon tube of the controller in Figure 8 and results in an increase in P_L. Instantaneously, part of this increase in P_L passes through the gain control three-way valve and registers in the proportional bellows. The restriction of the reset valve prevents this pressure increase from immediately reaching the reset bellows. At this stage of the operation, the pressure in the proportional bellows is much greater than that in the reset bellows. This has the effect of immediately reducing the gain of the controller. This gain reduction, during the transient portion of the disturbance, prevents the system from oscillating in an unstable manner.

As the transient disturbance decays and the system once again approaches equilibrium, the pressure signal to the proportional bellows has time to begin bleeding through the reset restriction to the reset bellows. As a result, the pressure in the reset bellows slowly increases and cancels the gain cutting effect of the proportional bellows. When a steady-state condition is finally reached, the pressures in the two bellows are again equal and the controller is back to the same high gain as before.

If the gain in the steady-state is high enough, the amount of steady-state offset is so small that, for all practical purposes, P_2 is returned to the original setpoint. That is why this type of controller action has earned the name RESET action. After a load disturbance occurs, it appears that the controller resets the controlled pressure (P_2) back to the setpoint. Those who are more theoretically oriented may refer to this same controller action as INTEGRAL

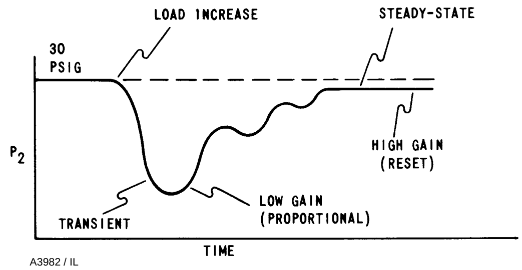

Figure 9. Control Performance with a Proportional Plus Reset Controller

control because of the mathematical relationship that exists.

Regardless of the name you prefer to use, the net result is the same. You sacrifice the high gain of the controller during the transient condition in order to keep the system stable. As a result, low gain performance is obtained during the transient condition as previously illustrated in Figure 2. As the transient condition decays, the reset action slowly increases the controller gain. By the time a steady-state condition is reached, the high gain results in the good steady-state performance illustrated in Figure 4. The net result of this proportional plus reset action is a performance curve similar to Figure 9.

To get the best performance from the system and still keep it stable, the restriction of the reset control must be adjusted. The reset signal must be delayed just enough to match the recovery characteristics of the process under control. If the gain increases too rapidly, the system will be unstable. In this case, we say that the reset is too fast! On the other hand, if the gain increases too slowly, the system will be very sluggish and will not achieve the best possible control.

In addition to the reset adjustment, the proportional control also must be set properly. The gain during the transient condition must be sufficiently low to maintain stability, but not so low that the transient deviation in P_2 is unnecessarily large. The proper adjustment of these controls for a given process is called controller tuning and will be discussed in more detail later.

Rate Control

When a load disturbance occurs, a pressure change is immediately sent to the proportional bellows, but the reset valve temporarily isolates this pressure change from the reset bellows. This has the effect of *immediately* reducing the gain of the controller so that the system remains stable. Unfortunately this immediate gain reduction also means that the transient control performance suffers as seen in Figure 9.

This transient control can be significantly improved by delaying the gain cutting effect just long enough to

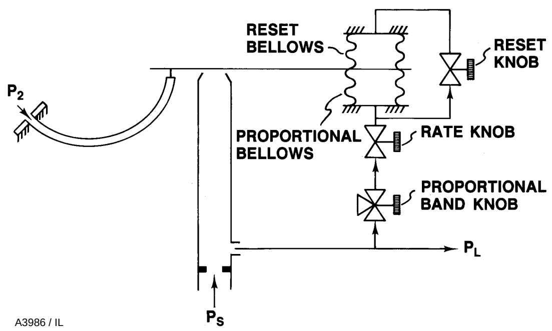

Figure 10. Proportional Plus Reset Plus Rate Controller

start the system responding to the load disturbance, but not so long that the system becomes unstable. The amount of delay needed will have to be adjustable, since it depends upon the dynamics of the particular system under control.

Figure 10 shows the addition of another valve restriction in the pressure line feeding both the controller bellows. This valve is known as the RATE CONTROL valve. Like the reset control, the rate control restriction can be varied to provide just the proper amount of time delay before the gain cutting effect begins. Once the pressure change has time to bleed across the rate valve restriction, the controller acts just as explained for the two mode control action.

The control action just described is known as rate action, because its effect is usually influenced by the rate at which the load disturbance occurs. If the disturbance occurs rather slowly, the rate valve offers no significant delay to the pressure signal. On the other hand, if the load disturbance is very rapid, the delay caused by the rate valve is comparatively more significant. The theorists refer to this controller action as DERIVATIVE control.

How does the gain of this three mode controller vary when it encounters a step change in the load? Initially, in the steady-state, the gain is high. When the step change in load occurs, the rate control momentarily maintains the high gain to improve the initial transient response of the system. Before this high gain has an opportunity to make the system unstable, the pressure signal to the proportional bellows cuts the gain during the major portion of the transient condition. Finally, as the system again approaches steady-state, the reset control slowly increases the gain back to its high steady-state value. This improves the steady-state control characteristics. Figure 11 illustrates how this gain variation of the three mode controller improves both the transient and the steady-state control.

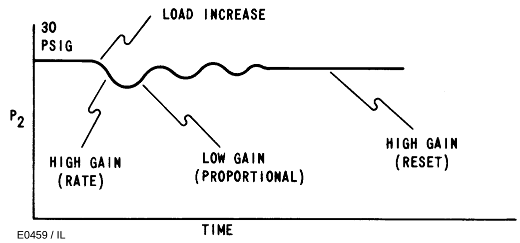

Figure 11. Control Performance with a Three Mode Controller

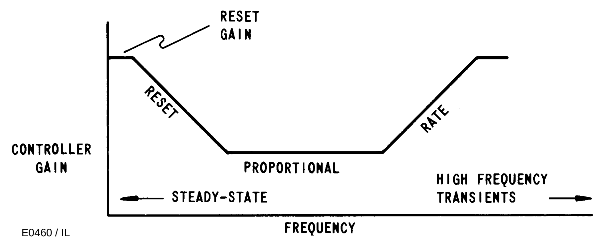

Figure 12. Frequency Response of an Ideal Three Mode Controller

Some Dynamic Considerations

By plotting the gain of the three mode controller versus the *frequency* of signals to which it must respond, some rather interesting and useful conclusions can be derived. This type of plot is known as a frequency response curve.

Figure 12 shows such a frequency response plot for an ideal three mode controller.

The low end of the frequency scale approaches what has been referred to as steady-state. Previously it was shown that as steady-state is approached, the reset action of the controller steadily increases the gain. This can be seen quite clearly in Figure 12. This increase in gain continues until a maximum is reached where the two bellows are completely balancing each other. This maximum gain is known as the RESET GAIN. The higher the reset gain, the closer the controlled variable (P_2) will be held to the desired setpoint in the steady-state.

At the other end of the frequency scale occur the rather rapid, or high frequency, transient changes in the load disturbance. The resulting high frequency changes in nozzle pressure are effectively isolated from the proportional bellows by the rate control valve. This keeps the gain of the controller at its maximum value. As the frequency of the signal decreases or slows down, the rate valve restriction becomes less and less significant and the gain gradually decreases. This is shown clearly in Figure 12.

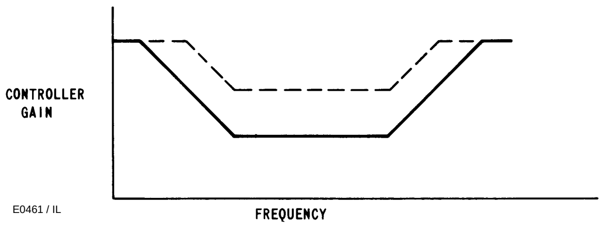

Figure 13. Effect of Gain, or Proportional Band, Adjustment on Frequency Response

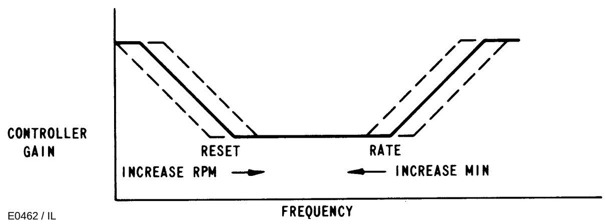

Figure 14. Effect of Rate and Reset Adjustments on Frequency Response

The lowest gain that the controller will reach is determined by the setting of the proportional band or gain control valve. Figure 13 shows how the depth of the notch in the gain curve can be varied with the gain or proportional band control.

By making the notch deeper, we say that we have cut the gain or broadened the proportional band. This is shown as the solid line at the bottom of the curve. If we narrow the proportional band, or increase the gain, we make the notch less deep as indicated by the broken line in the figure. In either case, it is worth noting that the width at the bottom of the notch remains the same when adjusting the proportional band or gain control.

The width of the notch is adjusted by the rate and reset controls as indicated in Figure 14. The left side of the notch is adjusted with the reset control. A common set of measurement units on the reset control is repeats-per-minute (RPM). Increasing the RPM brings the reset action into play sooner, i.e. at a higher frequency, and the left side of the notch moves to the right.

The right side of the notch is adjusted with the rate control. A common set of measurement units on the rate control is minutes (MIN). Increasing the MIN lets the rate action maintain the gain longer, i.e. down to a lower frequency, and the right side of the notch moves to the left.

Increasing both the rate and reset values narrows the notch, and decreasing these values broadens the notch. These two controls, in conjunction with the proportional band control, theoretically give a method of adjusting the notch and moving it around in virtually any manner that we choose. Let's digress for a minute and discuss why this ability might be important.

In every control loop there is a critical frequency where disturbances tend to reinforce themselves. If conditions are not satisfactory when a disturbance occurs, the loop will be unstable and will cycle at this critical frequency. This frequency is sometimes referred to as the system cycling frequency and can be thought of as loosely analogous to the resonant frequency of a simple mechanical device. In order to keep the loop from being unstable, the system loop gain must be sufficiently low at this critical frequency. At other frequencies, the gain can be maintained at much higher values in order to obtain the best possible control performance.

The three mode controller just discussed is simply a pneumatic notch filter with low gain at the system cycling frequency and high gain everywhere else. The whole purpose of tuning such a controller is to make certain that this gain notch is centered about the critical frequency of the particular system. Since the characteristic cycling frequency varies from system to system, the adjustability provided by the three mode controller gives the flexibility needed to control many different types of systems.

Controller Tuning Basics

Now, having discussed the effect of the various controls on the dynamics of the controller, a technique can be recommended for adjusting these controls.

There have been many theoretical studies to determine formulas and techniques for calculating the controller settings. These studies certainly have their usefulness in the general investigation of controller theory, but they have definite limitations when applying them to real processes. These studies usually make a number of simplifying assumptions that render results of limited value in real systems. In addition, the criteria for defining a properly tuned controller are very subjective and vary significantly from one instance to the next.

Experienced control people agree that the most realistic method of tuning a controller is for the operator to tune it around the actual system being controlled. This has the advantage of not requiring any assumptions about the dynamic nature of the process. Furthermore, this technique allows the operator to tune the system according to established company policy or his own subjective tastes, whichever is the determining factor. The following technique has been developed to accomplish this goal.

Every control loop has a characteristic critical frequency. This is the frequency where the gain must

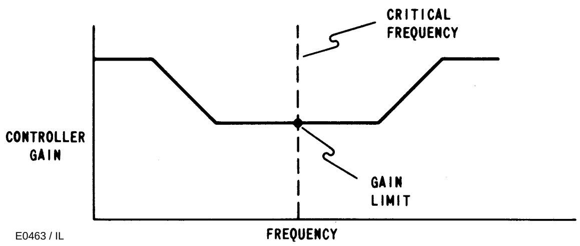

Figure 15. Tuning the Proportional Band Control

be kept sufficiently low for the system to be stable. It is not necessary to know this critical frequency; it is only necessary to realize that at this frequency there is some value of the controller gain that will keep the loop stable. The objective is to tune the controller for maximum performance, yet keep the gain at the critical frequency below this limiting value.

If the controller is already in service, it is usually wise to record the current controller settings before beginning the tuning process.

The first step is to broaden the frequency response notch of the controller enough so that the critical frequency, whatever its value, is included within the notch. At the same time, the proportional band is broadened enough so that the gain will be sufficiently low at this critical frequency.

The next step is to narrow the proportional band in small increments. After each adjustment, disturb the process slightly and watch the response of the controlled variable. At first, the system responds rather sluggishly, but as the proportional band gets narrower, the system becomes more and more oscillatory.

Eventually a condition is reached where the system continues to cycle when disturbed. This is the condition where the bottom of the notch has been raised to the limiting value of gain at the critical frequency. Figure 15 illustrates this situation. Once this condition is reached, double the proportional band setting, i.e. cut the gain in half. This again makes the system stable and provides a small gain margin.

Once the proportional band is adjusted, the next step is to increase the reset setting (RPM) in small increments. Again, disturb the process slightly after each adjustment. The process behaves in much the same manner as before. Again a condition is reached where the system sustains the oscillation. At this point, the left side of the notch is far enough to the right so that the limiting gain has been reached at the critical frequency. This is illustrated in Figure 16.

Once this condition is reached, move the reset control to a setting that is one-third of the limiting reset value

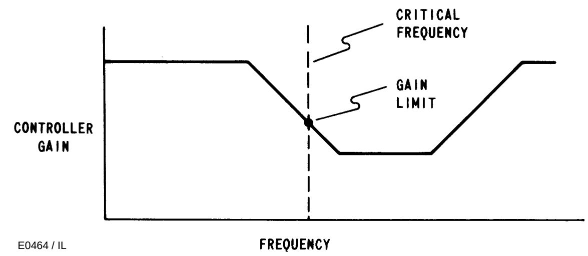

Figure 16. Tuning the Reset Control

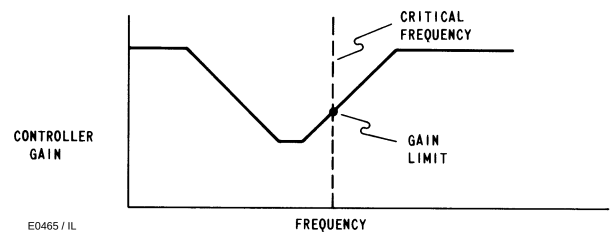

Figure 17. Tuning the Rate Control

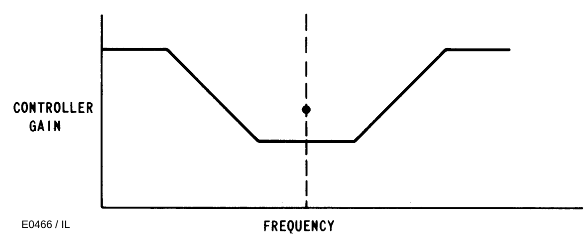

Figure 18. Properly Tuned Controller

(RPM) where the system cycled. This again makes the system stable with a reasonable gain margin.

Finally, the rate setting (MIN) is likewise increased in small increments while the process is disturbed slightly after each adjustment. As before, a point is reached where the oscillation continues. This is the condition where the right side of the notch is far enough to the left so that the limiting gain has again been reached at the critical frequency. This is illustrated in Figure 17.

In order to stabilize the system with a reasonable gain margin, the rate control is adjusted to a value that is one-third of the limiting rate (MIN) where the system cycled. The controller is now properly tuned around the critical frequency of the actual system being controlled. The final result is shown in Figure 18.

When using this method of tuning the controller, it is necessary to make several small disturbances to the process. This can be done in any manner that is convenient. Disturbing the flapper on a pneumatic controller or changing the setpoint are common methods.

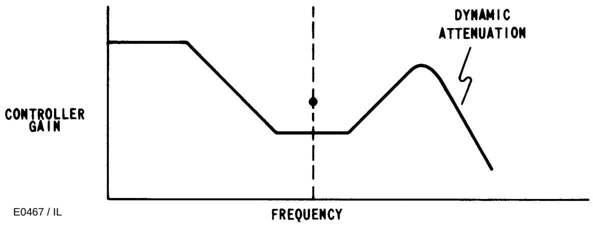

Figure 19. High Frequency Attenuation

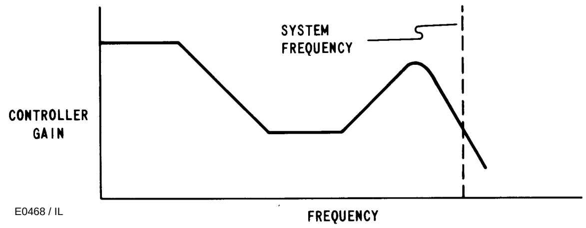

Figure 20. Controller Misapplied on Fast System

Once the controller is tuned, an experienced operator considers any large load changes that might be expected to occur during normal operation of the system. He investigates the response of the system to these large changes and makes any final, necessary adjustments. This procedure is highly recommended since many systems respond differently to large disturbances than to small disturbances.

More Dynamic Considerations

So far, the discussion has been limited to an ideal controller. In reality, the ideal frequency response of Figure 18 cannot be achieved in most practical controllers. At high frequencies, a real controller is not capable of responding as expected of an ideal controller. Such realities as inertia of moving parts, friction, etc., cause a high frequency attenuation of the controller response as indicated in Figure 19.

The three controller tuning adjustments permit moving and shaping the notch to whatever is needed for operating any process; however, if the notch is moved into the area of dynamic attenuation, the notch characteristics disappear. Figure 19 indicates that the dynamic attenuation is beginning to affect the rate action of the controller. If the notch is moved far enough to the right, the rate action can disappear entirely. Thus, if the characteristic cycling frequency (i.e. critical frequency) of the system were slightly higher than that shown in Figure 19, we would be unable to form the notch around the critical frequency.

One of the basic principles of controller application is to make certain that the response of the controller is faster than the characteristic cycling frequency of the system. If the response of the system is much faster than that of the controller, the characteristic frequency of the system will fall in the region of dynamic attenuation of the controller. This is illustrated in Figure 20.

If the controller is misapplied to a fast system as indicated here, the operator will discover that the tuning controls have absolutely no effect upon the performance of the system. The notch can be moved around anywhere within the limits of the controller response, but the gain of the controller remains unchanged at the system critical frequency.

The most frequent cause of this problem is a controller whose response has been slowed down due to excessive loading on the controller output. This can be caused by feeding the output of the controller directly into the diaphragm casing of a large actuator or into a long transmission line. The solution is to unload the controller output by using a booster or a positioner in conjunction with the actuator. The decision on which to use is a function of loop dynamics and is discussed elsewhere. In either case, the small volume of the positioner input bellows, or the input chamber of the booster, places much less loading upon the controller than the large diaphragm actuator casing. As a result, the controller response is a great deal faster, and it can then be properly tuned around the characteristic frequency of the system.

Conclusion

The proper selection, use, and tuning of a controller is a complex business. As a result, the controller is probably the most misunderstood and misused of all control components. It is sincerely hoped that this paper has helped eliminate some of the confusion.

The fundamentals of controller operation are actually rather straightforward, but as in everything else, there are many important considerations that can be learned only through experience.

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