for determining the exact condition of the equipment based on previous tests. Where codes exist, the code should be followed.

5. The inspection should produce a detailed analysis of the present condition of the equipment. This will permit the establishment of the rate of corrosion or deterioration and estimation of the safe operating life of the equipment. Through test drilling, corrosion plugs, calipers, and other means of measurement, the life of the equipment may be anticipated so accurately as to permit the replacement of old or defective equipment before it actually fails in service.

6. All test inspections and repairs, when recorded on suitable report forms for future reference, should be distributed to interested company executives.

7. Through the correlation of tests and inspection data, the establishment of design, construction, and maintenance standards for future equipment may be developed, and through the continuous replacement of defective equipment the process may be maintained perpetually new.

8. The inspection personnel should investigate all accidents involving persons or equipment. While this may seem post mortem treatment, the data obtained may become important in developing future construction and control procedure.

This outline may appear to the uninitiated to be extensive, complex, and costly, but when such a procedure is made a definite part of the entire plant production program, it is readily absorbed and, according to published information, can be held well within 0.5 per cent the total plant man-hours. It is a small price to pay for protecting plant and personnel, and guaranteeing the life and safe operation of process pressure equipment.

There is little new in this suggested routine. It is the correlation of many ideas, definitely attached to the operation of chemical process vessels. For those who desire to obtain additional information for guidance in handling the problems in their own plant, I would draw attention to the codes developed within the American Society of Mechanical Engineers, the code for unfired pressure vessels of the American Petroleum Institute and American Society of Mechanical Engineers, the National Safety Council's publication, "Pressure Vessels, Fired and Unfired", Parts I and II.

GETTING THE MOST FROM AUTOMATIC CONTROL

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In automatic control it is the combined characteristics of controller and process that count. They may be suited to each other. Unsatisfactory results may mean that either a more "refined" mode of control or additional control may be required. Again some simple change in the process may solve the problem.

Characteristics of control equipment are generally simple and easily specified. Characteristics of processes are infinitely varied. They not only depend upon the particular application but are frequently subject to wide variation with time. The user of automatic control equipment should be familiar with certain general principles which are helpful in particular control problems. The ideal time to apply these to a process is during the period of design when costly mistakes can be avoided. A brief review of automatic control theory is given, with examples from the fields of temperature and pH control.

The importance of a suitable relation between the control-valve setting and the resultant flow is stressed. Curves are given showing the characteristics of some present-day valves when tested at constant pressure drop. The effect of line drop in determining characteristics in service is shown by families of generally applicable curves plotted on a percentage basis.

AUTOMATIC control has long been applied to pressure, flow, level, and temperature, and recently, to an increasing extent, to such variables as electrolytic conductivity and pH. The great majority of applications have been successful, and, besides freeing operators for other duties, have often paid for themselves many fold in increased production and improved product.

As the use of instruments and automatic controllers has increased, many plants have assigned special men to care for them. These men have usually become very proficient, often understanding a mechanism just as well as the manufacturer who supplied it. Generally they have had a less complete picture of important relations between controller characteristics and process characteristics. To get the most out of automatic control they should be provided with this knowledge, in so far as possible, and, what is just as important, should be given sufficient authority to apply it. Further, they should be consulted when new process equipment is being designed to avoid mistakes which may be difficult to correct later.

What special knowledge should the plant automatic control expert have in addition to knowing the mechanisms employed? First he should understand the modes of control which these mechanisms produce. Then he should know how the results obtainable with those modes depend upon process characteristics. Finally, he should understand the particular process under consideration well enough to analyze it from the point of view of characteristics favorable or unfavorable to control.

Generally speaking, a sufficient knowledge for most practical purposes requires little or no mathematics and is well within the grasp of the average instrument department head. A brief review of the general theory will be given, followed by a discussion of what is called "the effective valve characteristic", a subject of considerable practical importance in both existing and projected automatic control applications.

Examples of Modes of Control

As a basis for discussing the combined effect of controller characteristics and process characteristics, three of the most important modes of control will be considered: two-position control, proportional-position control, and proportional-position plus proportional-speed-floating control.

For brevity the latter will be referred to as proportional-plus-floating control in further discussion.

In explanation of these modes, consider a furnace heated by gas to be controlled at about 600° C. The temperature is automatically measured by suitable means, and from the results of the measurement a valve in the gas line is automatically positioned in accordance with the control employed.

In two-position control the valve automatically takes one of two positions. In the first the rate of heating is always less and in the second always greater than is needed to maintain the desired temperature. (On-and-off control is the special form of two-position control in which, for one position, the heating effect is reduced to zero by closure of the valve.)

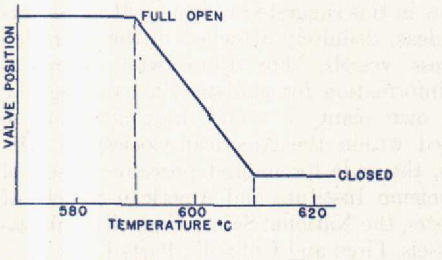


FIGURE 1. RELATION OF TEMPERATURE AND VALVE POSITION IN PROPORTIONAL-POSITION CONTROL

As the temperature is to be held at about 600° C., it may be arranged that when the temperature *rises* to 602° C., the valve will be suddenly moved to the more nearly closed position. The temperature will then fall until at a predetermined temperature—say, 598° C.—the valve will be suddenly moved to provide the higher rate of heating, which rate will continue until 602° C. is again reached; then a new cycle of valve action and temperature change will begin. With two-position control the *time intervals during which* the valve is in each position are automatically so related to each other that the average rate of heat supply corresponds with the average requirement for heat. With this type of control, continuous cycling of the temperature is inevitable.

With proportional-position control the gas valve may be automatically set anywhere between its limits of travel. There is always a definite positional relation between the valve stem and the temperature index as indicated in Figure 1. This curve shows the valve to be full-open when the temperature is 590° C. or lower and closed when the temperature is 610° C. or higher. For intermediate temperatures the valve takes corresponding intermediate positions. The relation between temperature and valve position is here shown to be linear within the working range (590° to 610° C.). This applies to many actual controllers when the valve position is taken as the *position* of the valve stem. As will be pointed out later, the relation between temperature and rate of gas flow, or heating rate, is not usually linear and usually should not be. Figure 1 shows that, if the range of load change requires that in one case the valve must be nearly closed and in another case nearly full-open, the controlled temperature will range from approximately 590° to 610° C. This temperature interval is usually referred to as the proportional-band or throttling range.

It would be possible to so adjust the controller that the valve would be full-open at 595° C. and closed at 605° C., but this might introduce undesirable cycling of the temperature, as will be explained later. When this is the case, the temperature can be held more closely to the desired value only by manually changing the relation between valve position and temperature, while the width of the band for automatic control is held constant. For example, *the entire curve* of Figure 1 could be moved to the left so that the valve would be full-open at 580° C. and closed at 600° C.

Proportional-plus-floating control combines the smooth action and stability of proportional-position control with the ability to control to a specified temperature under all load conditions. This is accomplished by adding to the proportional-position control action a floating mode of control which moves the valve continually, if the temperature deviates from the set point, in a direction depending upon the direction *of the deviation and at a rate proportional to the deviation.* Further explanation of the resultant control action will be given in the next section.

Controllers Applied to a Process

Figure 2 represents two simple temperature processes; *a* is easy to control while *b* is relatively *difficult. Each consists of* a stirred water bath heated by gas. The load or demand, subject to variation, is a flow of water which enters the tank through valve V_h and overflows at *B*. In the control instrument the temperature of thermometer *T* is measured, and from it the position of valve V_c is determined in accordance with the particular mode of control used. For present purposes it is convenient to disregard any time lag in the thermometer or the measuring mechanism and to assume that the control actions are at all times made in accordance with the true temperature of the bath.

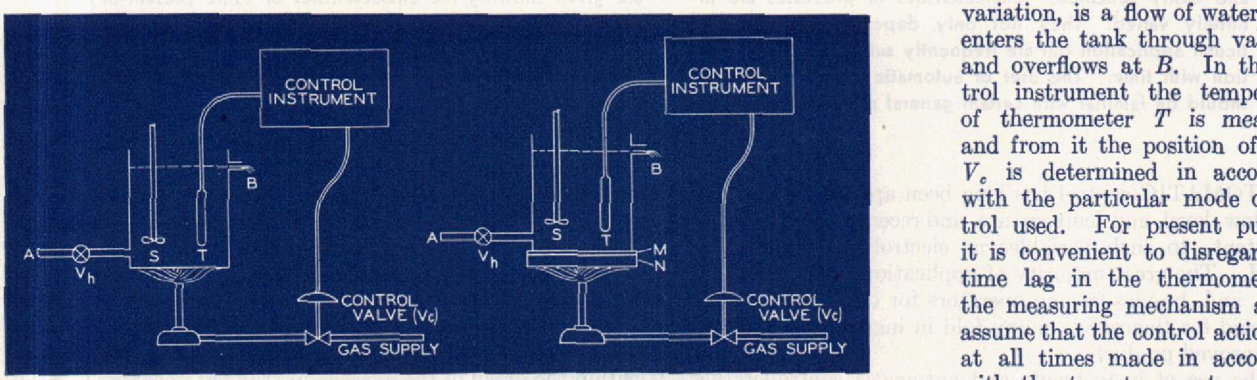


FIGURE 2. EXAMPLES OF PROCESSES EASY TO CONTROL (*a*) AND RELATIVELY DIFFICULT TO CONTROL (*b*)

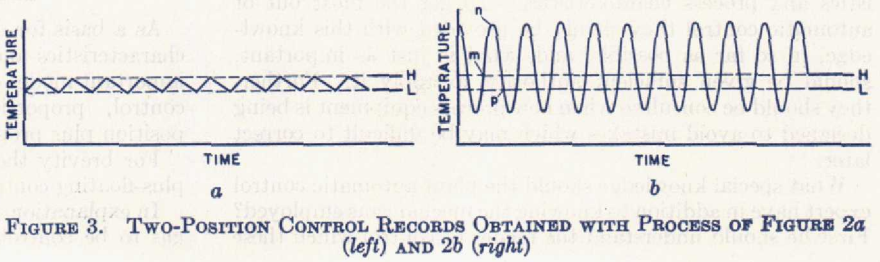


FIGURE 3. TWO-POSITION CONTROL RECORDS OBTAINED WITH PROCESS OF FIGURE 2*a* (*left*) AND 2*b* (*right*)

The only difference between the two processes is that in Figure 2*a* the flame plays directly upon the bottom of the tank, whereas in *b* the heat must first pass through a slab of iron, *N*, and a sheet of asbestos, *M*. The iron is introduced to represent thermal capacity and the asbestos to represent thermal resistance. In *a* we have thermal capacity only, while in *b* we have two thermal capacities separated by a thermal resistance.

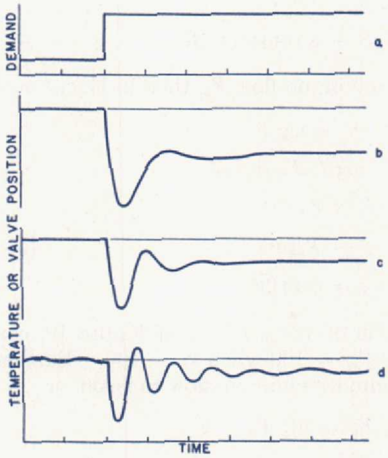


Figure 4. Response to Sudden Disturbance with Proportional-Position Control

Curves *b*, *c*, and *d* are for progressively narrower throttling ranges.

Two-position control applied to Figure 2*a* gives the temperature record of Figure 3*a*. Each time the temperature reaches the value *H*, the gas is turned off; each time it drops to the value *L*, the gas is turned on. The direction of the temperature trend changes immediately in each case. By reducing the temperature interval between *H* and *L*, the range of temperature variation could be reduced. The valve would then move from one position to the other more often, and the frequency of temperature oscillation would be correspondingly increased.

Applied to Figure 2*b*, two-position control gives the temperature record of Figure 3*b*. The valve is opened at temperature *L* and closed at temperature *H* as before, but the direction of temperature trend does not change until some time later. Even though the temperature interval between *H* and *L* were reduced to zero, a similar record would be obtained. The reason is as follows: The iron plate has considerable heat capacity and, because of the presence of the asbestos sheet, must be raised to a temperature well above that of the water before the necessary rate of heat transfer can take place. At *m* (Figure 3*b*) the heat is turned off but the iron plate is considerably hotter than the water, and the heat flow to the water exceeds the heat requirement for a time until point *n* is reached. Similarly, when the heat is turned on at *p*, the full rate of heating is immediately applied to the bottom of the plate, but some time elapses before the water is being heated at a rate sufficient to reverse the temperature trend.

The effect of heat capacity closely associated with the water in the tank may be designated as demand-side capacity lag and that due to the effect of thermal capacity *N* and thermal resistance *M* may be termed as transfer lag. At any given value of demand, a process generally becomes more favorable to control as the demand-side capacity is increased and transfer lag between the automatically adjusted heat supply and the primary measuring element (e. g., thermometer) is decreased. Even if the asbestos sheet were omitted, some overshooting would occur since iron is not a particularly good thermal conductor and every element involves both thermal capacity and thermal resistance.

Proportional-position control and proportional-plus-floating control will be considered in connection with Figure 2*b* (the more difficult process) only. Figure 4 shows how proportional-position control takes care of a sudden change in demand. Curves *b*, *c*, and *d* are for successively narrower proportional bands or throttling ranges. As the temperature change necessary to full-stroke the valve is reduced, the drop in temperature resulting from the increased load is reduced, but the oscillations following the sudden change are increased. These set a limit as to how narrow the throttling range can be made. The more favorable the process time lags are, the narrower the throttling range that can be used without introducing undesirable oscillations.

Figure 5 is an analysis of the response of proportional-plus-floating control to a sudden, sustained, load disturbance. Curve *a* represents the disturbance. Curve *b* is the resulting, experimentally observed temperature change. Curve *e* shows change in rate of heating or valve position. In *e* the valve overshoots its final equilibrium position quickly, then gradually backs off. In doing so it introduces a block of energy, represented roughly by area *A*, which serves two purposes: It gets the water back to temperature by making up for energy lost before the controller can completely do its work, and it provides the energy necessary to bring the iron plate to the new temperature at which the new required rate of heating can be continuously supplied to the water.

Curves *c* and *d* show the separate effects of the two component modes acting simultaneously. The effect of the component corresponding exactly to that of proportional-position control is shown in *c*. It is the same in form as the temperature curve. This is the component which supplies the extra energy necessary for stable return to the control point. The floating component is represented by curve *d*, and this is the component responsible for the shift to the new steady-valve position.

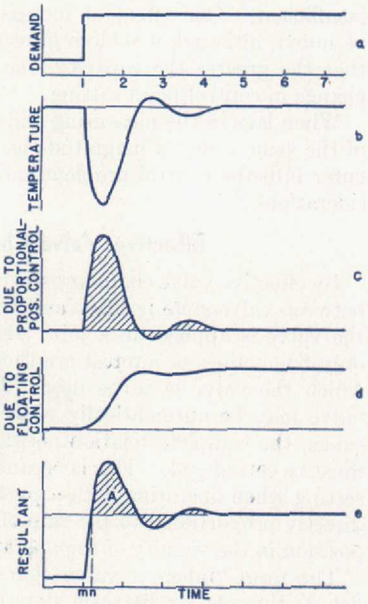


Figure 5. Graphical Analysis of Proportional-plus-Floating Control

Reaction and Distance-Velocity Lag

So far, two types of process lag have been considered, referred to as demand-side capacity lag and transfer lag. Consideration of pH control introduces two more which may be designated as reaction lag and distance-velocity lag. The term "reaction lag" is used to denote delay between the time of introduction of a control agent into a treated solution and the time when chemical equilibrium is reached. It usually enters into the control problem indirectly. In Figure 6 the pH electrodes may be placed very close to the tank; but if the particular reaction is one that requires appreciable time, and although the pH is closely controlled at the point of measurement, a measurement made some distance down the line will show that an additional change has occurred en route. It is natural, then, to want to control at a point far enough away so that the reaction is *essentially complete*. As a result, distance-velocity lag is introduced, representing the time required for the solution to flow from the tank to the point of measurement. This type of lag is particularly unfavorable in that changes occurring in the tank during the delay interval are inevitable, since the controller cannot act until it knows about them. With a tank of sufficient capacity, good control *may be obtained in spite of* considerable distance-velocity lag. Increasing the storage capacity increases the time available for reaction and also cuts down *the rate at which variations in* the inflowing material can change the pH. Both of these are effects favorable to control.

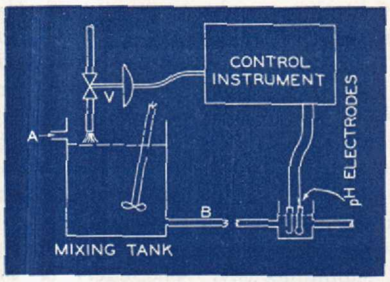


FIGURE 6. SIMPLE ILLUSTRATION OF pH CONTROL

In increasing storage capacity to make a process more favorable to control, it must be kept *in mind that the* desired advantages will not be gained unless provision is made for thorough mixing. Stirring of the type which merely moves the *material* about without quickly distributing incoming material throughout the tank is usually *insufficient*. One effect of increased capacity which should be noted, although it seldom proves to be disadvantageous, is *that the greater the* capacity, the slower is the response to change in control-point setting.

When lags in the measuring *and controlling* equipment are of the same order of magnitude as those of the process, they enter into the control problem and may require *special* consideration.

Effective Valve Characteristic

By effective valve characteristic is meant the actual relation between valve-stem *position and* flow of a control agent when the valve is applied on a job. In two-position control, the only flow values of interest are those *for the two positions* at which the valve is to be used. For controls in which the valve may be automatically positioned anywhere within its *range*, the complete relation between stem position and flow must be considered. This is because the best throttling-range setting *when operating* with a particular flow requirement is directly proportional to the rate of change of flow with stem position in the vicinity of *the operating point*.

The term "inherent valve characteristic" will be used to denote the relation between stem *position and flow* as found by test under conditions of constant pressure drop. Curves B to F of Figure 7 are experimentally determined inherent characteristics of particular valves of the types indicated. All of these approximate more or less closely the form of curve A, which is a *theoretically* perfect equal-percentage characteristic. With this characteristic, equal movements of the valve stem result in equal *percentage changes* in flow, regardless of the actual flow.

"Equal-percentage flow characteristic" *means* a relation such that small equal changes in valve-stem position result in equal percentage changes in flow, regardless of *the actual* flow. If ΔS denotes a small change in stem position and ΔF denotes a corresponding small change in flow F, then the above *statement* is expressed mathematically by

$$\frac{\Delta F}{F} \times 100 = k_1 \Delta S \qquad (1)$$

where k_1 = a constant of proportionality

or

$$\frac{\Delta F}{F} = k_2 \Delta S \qquad (2)$$

where $k_2 = k_1/100$

The corresponding differential equation for infinitely small values of ΔF and ΔS is

$$dF/F = k_2 dS \qquad (3)$$

Integrating Equation 3,

$$\log F = k_2 S + \text{a constant } (K) \qquad (4)$$

Assuming $S = 0$ for minimum flow F_0, then in Equation 4,

$$K = \log F_0$$

and

$$\log F/F_0 = k_2 S$$

$$F = F_0 \epsilon^{k_2 S}$$

or

$$F = F_0 10^{kS} \qquad (5)$$

$$k = 0.4343 k_2$$

To obtain the *equation of curve* $p = 1$ of Figure 10, consider maximum flow $= F_m = 100$ when $S = 100$. Further, consider ratio of maximum to minimum flow *to* be 50, *or*

$$F_m/F_0 = 50; \; F_0 = 2$$

Then Equation 5 becomes

$$F = 2 \times 10^{0.0170S} \qquad (6)$$

Equations 5 and 6 plot as straight lines on semilog paper and thus provide an easy way to obtain intermediate values, corresponding to any particular end values of F_0 and F_m.

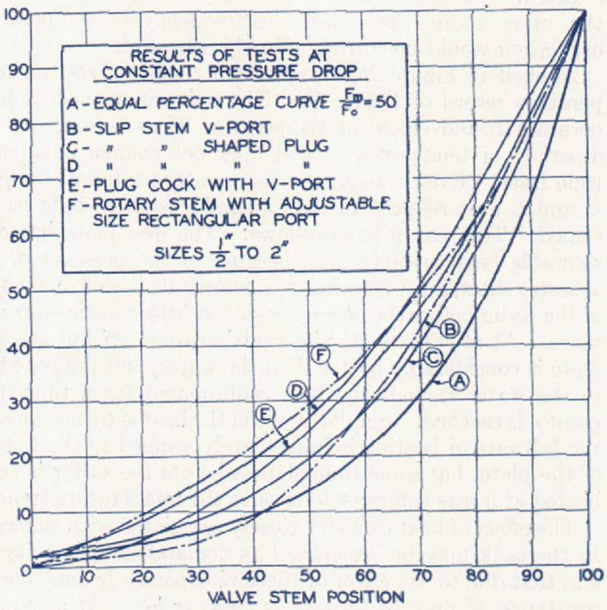


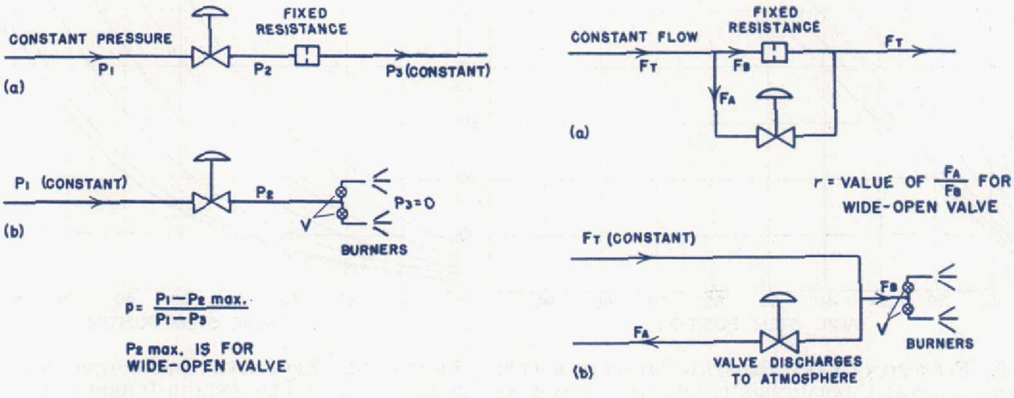
FIGURE 7. TYPICAL INHERENT VALVE CHARACTERISTICS

While, theoretically, valves with perfect equal-percentage characteristics can never reduce the flow below F_0, provision *is made for* tight-seating of practical valves.

Another theoretical characteristic which is approximated by certain valves *is the linear characteristic*. This would be represented by a straight line in Figure 7. (In this and following figures, all flows are *plotted in per cent* of maximum flow and all stem positions in per cent of total movement.)

Effects of Pressure Variation

Many factors, in addition to the inherent characteristic, may influence the effective characteristic. The most common is the effect of variations in pressure drop across the valve with flow through it. The influence of this factor was calculated for the special conditions of installation shown in Figure 8 and for inherent characteristics of both the equal-percentage and linear form. The derived curves apply to turbulent flow of a liquid. The method of derivation is explained later.



A *B*

Figure 8. Methods of Applying Control Valves (*A*), and Schemes for Using a Control Valve in a By-Pass or Return Line (*B*)

Figure 8*A* shows a common arrangement (*b*) for regulating fuel flow to burners. The pressure upstream from the valve is constant. The pressure downstream is determined by drop through the burner system and therefore varies with the rate of flow. When the flow is very small, as with a nearly closed control valve, practically the full available pressure drop appears across the valve. With the valve full-open the pressure drop across it may be only a fraction of the total drop. Figure 8*A* indicates the general arrangement (*a*) which was considered in making calculations; (*b*) is a special case. Curves of Figures 9, 10, and 11 apply to the arrangements of Figure 8*A*.

In Figure 9 the curve marked $p = 1$ is a linear flow characteristic. The letter p designates the ratio of the drop across the valve to the total drop when the valve is delivering 100 per cent flow. When $p = 1$, the entire pressure drop is taken by the valve and the effective characteristic is identical with the inherent characteristic. Curves marked 0.8, 0.6, 0.4, 0.2, 0.1, and 0.05 are for cases in which the drop across the valve at 100 per cent opening is the designated fraction of the total drop. As the proportion of the drop available for the control valve is reduced, the effective characteristic flattens out more and more at the higher flows. As will be brought out in more detail later, this is frequently undesirable with proportional-position or proportional-plus-floating control.

Figure 10 shows that with a valve of inherent equal-percentage characteristic the effective characteristic becomes straighter for smaller values of the ratio p. The fact that this type of inherent characteristic "resists" flattening due to line drops is probably the principal reason that it has proved to be satisfactory for a large percentage of automatic control applications.

In sizing up an automatic control installation applied to an operating process, it is sometimes convenient to be able to determine the effective valve characteristic without experimentally determining the ratio p. If the inherent valve characteristic is known, this is readily done. The curves of Figure 11 are for use when the inherent characteristic is either linear or equal percentage. To use these curves, the operating position of the valve stem is noted together with the drop across the valve and the total drop. The ratio of the former to the latter pressure drop is p'. As an example, assume a linear valve at a stem position of $S = 40$. The drop across the valve is noted to be 30 pounds, and the total drop is 75 pounds. Therefore, $p' = 30/75 = 0.4$. Point $S = 40$, $p' = 0.4$ of Figure 11 nearly falls on the curve marked $p = 0.1$. Curve $p = 0.1$ of Figure 9 may therefore be considered as the effective characteristic.

Another method of applying a control valve is to throttle a return flow as indicated by (*a*) and (*b*) in Figure 8*B*. The arrangement of (*a*) may be considered to represent control of a heat exchanger by by-passing a portion of the heating agent. In this case the resistance to flow is offered by the heat exchanger and any associated orifices or valves. It is assumed that the total flow is constant. Effective characteristics have been calculated for various values of ratio r, which is the ratio of the returned flow to the useful flow when the valve is wide open.

The curves of Figure 12 are for a valve with a linear inherent characteristic. These curves are plotted with the closed position of the valve stem to the right and the full-open position to the left, because in this case the useful flow increases as the valve opening is decreased.

For the curve $r = 16$ of Figure 12 the ratio of maximum to minimum flow is also 16. Larger ratios could be obtained either by decreasing the resistance to flow offered by the valve or by increasing the resistance of the parallel path, but it is evident that an effective characteristic of increasing curvature would result. The curvature of the $r = 16$ characteristic is already too steep for even moderately difficult control jobs unless the valve is to be operated over a narrow range only.

Figure 13 shows the results to be expected when the valve has an inherent equal-percentage characteristic. The curve for $r = 16$ is not far from a linear effective characteristic. In general, better results can be expected with an equal-percentage valve than with a linear valve when the valve is placed as in Figure 8*B*. Comparison of Figures 12 and 13 shows that, by using a valve with an inherent characteristic with somewhat less curvature than the equal-percentage form, effective characteristics ranging between those of the two figures can be obtained.

In determining points for the curves of Figures 9, 10, 12, and 13, it was necessary to make calculations for those based on a linear inherent characteristic only. As will be shown later, the effective characteristics for an equal-percentage valve or, in fact, a valve of any other inherent characteristic is readily obtained from the results for the linear valve.

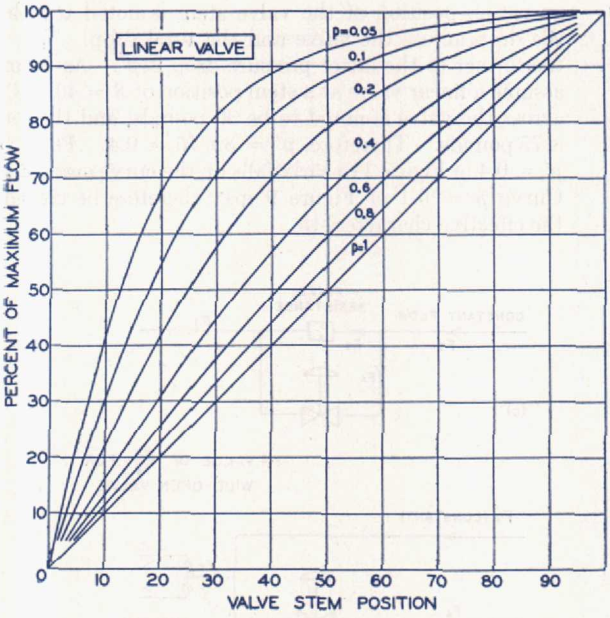


FIGURE 9. EFFECTIVE CHARACTERISTICS OBTAINABLE WITH A LINEAR INHERENT CHARACTERISTIC AND APPLICATION AS IN *FIGURE 8A*

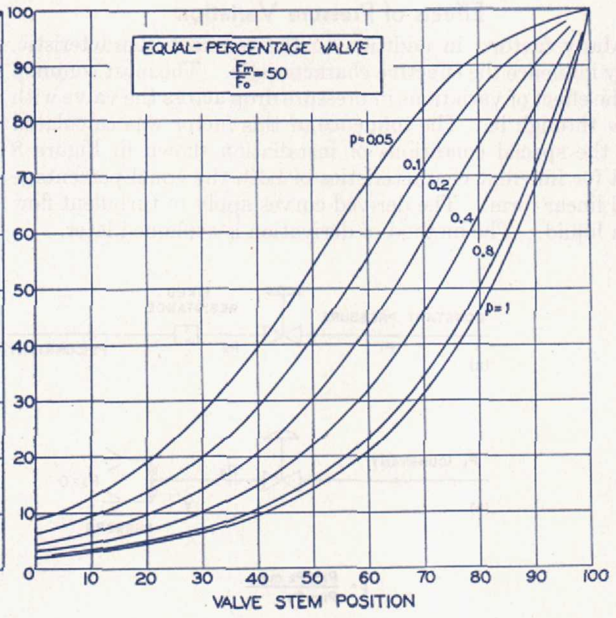


FIGURE 10. EFFECTIVE CHARACTERISTICS OBTAINABLE WITH AN EQUAL-PERCENTAGE INHERENT CHARACTERISTIC AND APPLICATION AS IN FIGURE 8*A*

ARRANGEMENT OF FIGURE 8*A*. The derivation for curves of Figures 9 and 10 is as follows:

F = flow at any stem position S
F_m = maximum flow = 100
P_1 = pressure upstream from valve
P_2 = pressure downstream from valve
P_3 = ultimate constant pressure
k_1, k_2 = constants

For a linear valve

$$F = k_1 S \sqrt{P_1 - P_2} \tag{7}$$

$$F = k_2 \sqrt{P_2 - P_3} \tag{8}$$

$$F_m = 100 k_1 \sqrt{P_1 - P_{2\,\text{max.}}} = 100 \tag{9}$$

$$F_m = k_2 \sqrt{P_{2\,\text{max.}} - P_3} = 100 \tag{10}$$

$$p = \frac{P_1 - P_{2\,\text{max.}}}{P_1 - P_3} \tag{11}$$

Equation 8 may be written in the form,

$$F^2 = k_2^2[(P_1 - P_3) - (P_1 - P_2)] \tag{12}$$

From Equations 9 and 11,

$$(P_1 - P_3) = 1/k_1^2 p \tag{13}$$

From Equation 7,

$$(P_1 - P_2) = \left(\frac{F}{k_1 S}\right)^2 \tag{14}$$

Substituting in Equation 12 from 13 and 14,

$$F^2 = \left(\frac{k_2}{k_1}\right)^2 \left(\frac{1}{p} - \frac{F^2}{S^2}\right) \tag{15}$$

Arranging Equation 10 similar to 12, dividing the result by Equation 9, and substituting from 11,

$$\left(\frac{k_2}{k_1}\right)^2 = \frac{\overline{100}^2\, p}{1 - p} \tag{16}$$

Substituting in 15 from 16, and solving for F,

$$F = \sqrt{\frac{1}{(1 - p) + \overline{100}^2 \left(\frac{p}{S^2}\right)}} \tag{17}$$

Curves of Figure 10 were obtained from those of Figure 9 as follows: Consider, for example, a flow of 50. For a linear valve this would occur at stem position 50. For the equal-percentage valve (curve $p = 1$, Figure 10) it occurs at stem position 82.4. Therefore, all that is necessary is to plot for stem position 82.4 in Figure 10 all points for stem position 50 of Figure 9. Other points for Figure 10 are similarly determined.

By following this same procedure, sets of curves of effective characteristics may be quickly plotted from an inherent characteristic of any form, on a percentage basis.

CURVES OF FIGURE 11. The ordinate p' is defined by the equation,

$$p' = \frac{P_1 - P_2}{P_1 - P_3} \tag{18}$$

From Equations 11 and 18,

$$p' = \frac{(P_1 - P_2)p}{P_1 - P_{2\,\text{max.}}} \tag{19}$$

From Equation 7,

$$(P_1 - P_2) = \left(\frac{F}{k_1 S}\right)^2 \tag{20}$$

From Equation 9,

$$(P_1 - P_{2\,\text{max.}}) = \frac{1}{k_1^2} \tag{21}$$

Substituting in 19 from 20 and 21,

$$p' = F^2 p/S^2 \tag{22}$$

In Figure 11 scales are given for both linear and equal-percentage valves. A special scale can be provided, or a complete series of curves can be drawn, for any valve by merely noting that the corresponding stem positions for the linear and any other valve are those for which the flows given by the inherent characteristics are the same on a percentage basis.

ARRANGEMENT OF FIGURE 8*B*. Derivation for curves of Figures 12 and 13.

F_T = total flow = 100
P = pressure drop across control valve
k_1, k_2 = constants

then

$$F_B = k_1\sqrt{P} \tag{23}$$

and for the case of a linear valve (Figure 12),

$$F_A = k_2\frac{S}{100}\sqrt{P} \tag{24}$$

When $S = 100$,

$$\frac{F_A}{F_B} = \frac{k_2}{k_1} = r \tag{25}$$

$$\frac{F_B}{F_T} = \frac{F_B}{100} = \frac{F_B}{F_A + F_B} \tag{26}$$

From Equations 23, 24, 25, and 26,

$$F_B = \frac{100}{\frac{rS}{100} + 1} \tag{27}$$

Curves of Figure 13 were obtained by first noting corresponding stem positions for linear and equal-percentage characteristics, as mentioned above. Because the most nearly closed position for the theoretical equal-percentage valve considered is 2 per cent and because maximum useful flow occurs at this position, the resulting curves gave maximum flows under 100. The flow values were then multiplied by suitable factors to bring the maximum for each curve up to 100.

Ideal Effective Characteristic

A general definition of the ideal effective characteristic, based on the use of proportional-position or proportional-plus-floating control, may be stated as follows: The ideal effective valve characteristic is that characteristic for which the same experimentally determined, best throttling-range setting is arrived at, regardless of the valve position required by the existing demand. Under the most favorable conditions the ideal characteristic is the same from day to day.

The actual shape of the ideal characteristic depends upon the particular application involved. Practical experience has shown that, when a furnace is operating with a heavy load, a rate of change of the rate of heating with temperature may be used which is several times that permissible at light loads. This assumes that the control valve of Figure 8*A*, for example, handles all of the fuel. However, if a rather unusual case is considered, in which the load is so steady that, say, 90 per cent of the heat may be furnished by independent burners while only 10 per cent passes through the control valve, then it is to be expected that over the narrow range available for automatic control the ideal characteristic would be linear. To take another special case, Ivanoff (7) assumed that a heat exchanger requires a rate of steam flow proportional to the rate of flow of the heated liquid, and at the same time assumed that heat transfer takes place as in a semi-infinite solid. On this basis he concluded that the ideal effective characteristic should be of the equal-percentage form. Most practical cases lie somewhere between these two extremes.

The ideal characteristic will generally be found to be different, depending upon whether it is determined by varying the demand with the control point fixed or by varying the control point only. Therefore, ideal characteristics for both cases cannot exist simultaneously.

While a reasonably good effective characteristic is always desirable, relatively few applications require that it have the ideal form as defined above. A practically ideal characteristic is needed if all of the following requirements are to be met in the face of unfavorable process lags:

1. The temperature is to be controlled as closely as possible under all conditions of operation.
2. Disturbances take place at such a rate that control actions must be as strong as is practical if condition 1 is to be fulfilled.
3. Normal load variations call for valve positions covering the entire range of the valve.

In the absence of any one of the above requirements, satisfactory results may be obtained with an effective characteristic which deviates considerably from the ideal.

Obtaining a Suitable Effective Characteristic

Consider arrangement (*b*) of Figure 8*A* with the full heat supply under control. Assume that up to the present time the fuel has been regulated by means of hand valves *V*.

The desired fuel flow with maximum valve opening must first be decided upon. Then comes the question of how much pressure drop is to be available for the control valve. Assume hand valves *V* to be kept nearly full-open with just enough range remaining for "trimming" purposes. With the hand valves set in this way, the drop across the control valve should be several times as great as the downstream pressure at the highest rate of flow used in normal operation. This will minimize the effect of trimming burners, or slight coking, upon the total flow.

To give a numerical example, the pressure required for the burner system including hand valves, may be 10 pounds at the highest normal flow of 100 gallons per hour; for purposes of bringing the furnace up to temperature quickly, the desired maximum flow may be 200 gallons per hour. Assume a drop across the control valve of 4 × 10 or 40 pounds for 100 gallons per hour. The supply pressure will then be 40 + 10 or 50 pounds. For 100 per cent flow (200 gallons per hour), the pressure downstream from the control valve will be $10 \times (200/100)^2$ or 40 pounds, giving a drop across the valve of 50 − 40 or 10 pounds. The value of *p* which is the ratio of

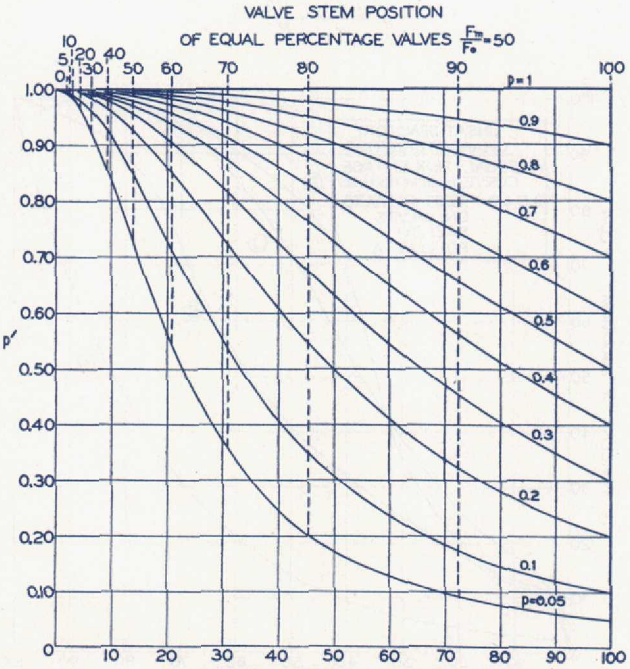


FIGURE 11. RELATION BETWEEN *p* AND *p′* WITH APPLICATION AS IN FIGURE 8*A*

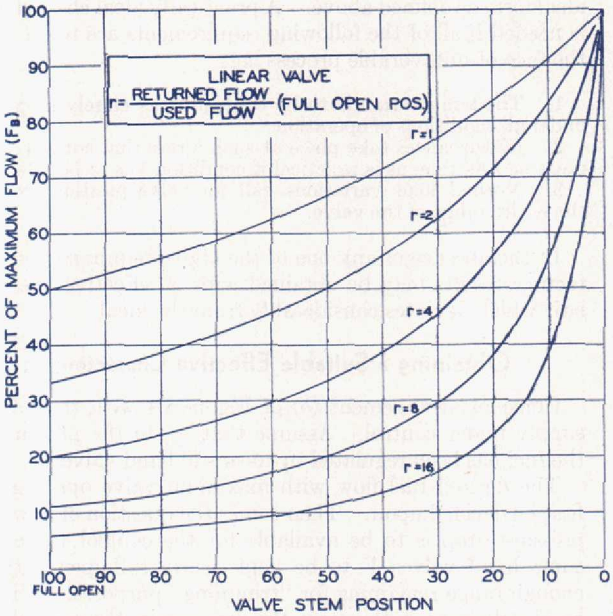


FIGURE 12. EFFECTIVE CHARACTERISTICS OBTAINABLE WITH A LINEAR INHERENT CHARACTERISTIC AND APPLICATION AS IN FIGURE 8B

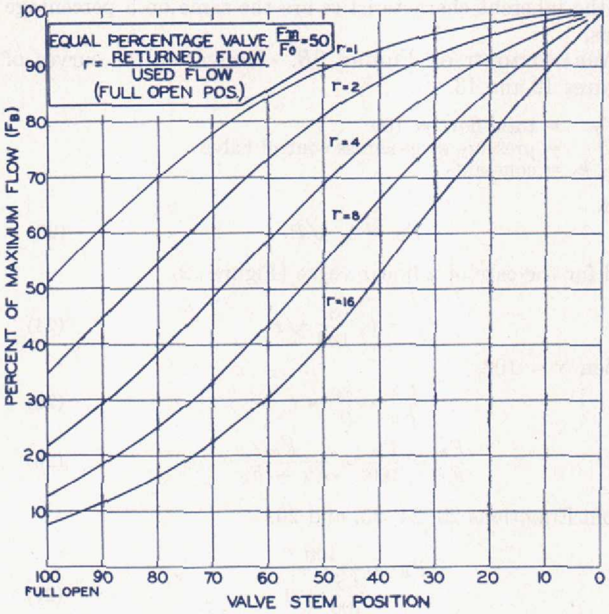


FIGURE 13. EFFECTIVE CHARACTERISTICS OBTAINABLE WITH AN EQUAL-PERCENTAGE INHERENT CHARACTERISTIC AND APPLICATION AS IN FIGURE 8B

this to the total pressure drop is 10/50 or 0.20. For an equal-percentage valve, the effective characteristic is the curve marked 0.2 in Figure 10. For most applications this would be found satisfactory.

Assume now the same requirements but that only linear valves are available in the size required. Figure 9 shows that for the value of $p = 0.2$ and flows of 50 per cent and below, a fairly steep portion of the curve is involved. It may be better then to raise the supply pressure, if possible, to increase p to, say, 0.6. From Figure 8*A*,

$$p = \frac{P_1 - P_{2\text{ max.}}}{P_1}$$

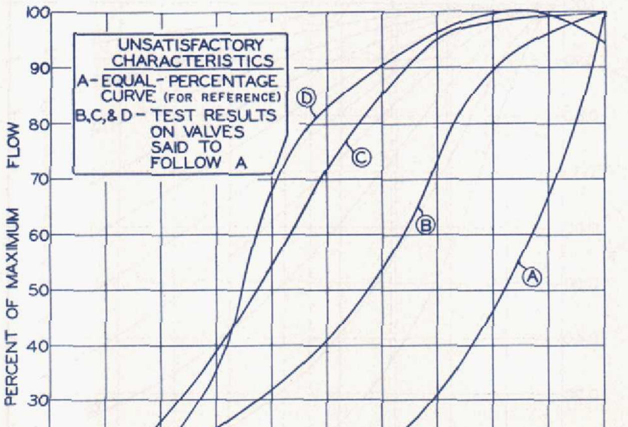


FIGURE 14. EXAMPLES OF UNSATISFACTORY INHERENT CHARACTERISTICS

or $$P_1 = \frac{P_{2\text{ max.}}}{1 - p} = \frac{40}{1 - 0.6} = 100 \text{ lb.}$$

The drop across the valve at the highest normal flow of 100 gallons per hour will now be 100 − 10 = 90, or nine times the downstream pressure.

In the examples considered above, the valve size must be such that with the valve full-open the flow will be 200 gallons per hour at the available pressure drop. This was 10 pounds *in the first case and 60 pounds in the second.*

When the return flow passes through the control valve as indicated in (*b*) of Figure 8*B*, both the burner system and the control valve are *subjected to the same pressure drop at all* times. One way of determining the valve size is to consider the highest pressure the pump is designed to supply and then adjust valves *V* until the desired flow is *obtained at this pressure.* The control valve must be such that it will pass *r* times this flow at the same pressure. The selection of *r* determines the effective characteristic as indicated in Figures 12 and 13. The above procedure gives the smallest usable valve.

Another procedure is to consider the use of an available valve and adjust valves *V* to obtain a suitable ratio, *r*. It is important to consider the possible effect of line drops which may become troublesome if a large valve, requiring a relatively low pressure drop, is used.

It is not always safe to assume that the inherent characteristic of a particular valve is the same as that given by the manufacturer for a line of valves. In Figure 14 curves *B*, *C*, and *D* are the actual characteristics of three valves that were said to follow curve *A* approximately. Curve *D* has a sudden jump-up at the low-flow end which will be troublesome if operation in this region is required. Its effect will be magnified in the effective characteristic and will be greater for smaller values of the ratio *p*. The curve is far too steep in the vicinity of the 30 per cent stem position and unnecessarily flat farther on. The flow reaches a maximum at 85 per cent stem position and then decreases. The characteristics shown in Figure 14 are exceptional cases. In general, the control valves of the principal manufacturers will be found to have reasonably good characteristics. Poor characteristics are usually the result of inaccurate machining or of an effort to build too much flow capacity in a given body size. It is well to be suspicious of control valves of very high capacity ratings. The next larger body size may be well worth the additional cost.

Figure 8 represents relatively simple applications. A factor that was neglected is the possible effect upon the oil flow of the means used for atomization. Another important factor is viscosity. If the oil is very viscous, it is important to provide a suitable heating system which will assure fluidity.

Adjustable resistances to flow, such as valves *V* in Figure 8, must be maintained within a limited range; otherwise the effective characteristic will be seriously altered. Preferably stops should be provided to limit the range of motion. If shutoff valves are required, they should preferably be separate.

Consider a complicated but fairly common case for which the effective characteristic is not readily calculated. The temperature of vapor from a fractionating column is controlled by adjusting the flow rate of reflux supplied by a direct-acting reciprocating pump with the control valve on the steam side of the pump. The flow of reflux is dependent upon the inherent valve characteristic, the steam supply pressure, the pressure drop across the valve, the efficiency of the pump, and the pressure against which it is working. When such a chain is involved, a fixed effective characteristic can be maintained only when each controllable element is held constant. A frequent cause of trouble in this particular case is an oversize or badly worn pump which refuses to function in accordance with any determinable law.

If, instead of a reciprocating pump, a centrifugal pump is used with a control valve in the discharge, and particularly if the pump is driven at essentially constant speed as by an induction motor, the possible variable factors are greatly reduced. The characteristic of the centrifugal pump enters into the effective characteristic.

Testing

Trouble shooting on the more difficult control applications may be greatly facilitated by providing for pressure taps on either side of the control valve and by installing an orifice for flow measurements. The effective valve characteristic can then be readily determined at any time. In the case of the smaller diaphragm-motor operated valves, it is convenient to use a dial gage such as the Ames No. 88, with a range of one inch, to measure the position of the valve stem.

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EDUCATION, EXPERIENCE, AND ENGINEERS

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THOSE of us who have been active in the profession of chemical engineering for a considerable time can remember the days when it was largely applied laboratory chemistry. The training and point of view of most of the older men who were in charge of departments of chemical engineering were to a great extent those of the laboratory chemist. It required a considerable stretch of the imagination to call the courses then given "engineering courses".

However, thanks primarily to the activities of the group at Massachusetts Institute of Technology (and, we believe, also to a certain extent to our group at the University of Michigan), a gradual understanding of the significance of unit operations became, if not actually widespread, at least accepted in a part of the schools and the profession. The crying need of chemical engineering at that time was for a fuller understanding of the theory of these unit operations. Consequently the first thing that most of us fought for was an adequate research program to advance our understanding in this field.

This movement has now progressed to the point where practically every school with chemical engineering courses worthy of the name has a unit operations laboratory and at least one man on the staff who is trying to do some theoretical research in unit operations. One of the indices of the growth in this field is the satisfactory symposia which have been held during Christmas week for a number of years.

At the Symposium on Separation Operations held in Ann Arbor, December, 1939, I was asked to open the program, and some of my statements at that time called forth considerable criticism. This was largely due to the fact that, because I was trying to be brief, I did not amplify or explain my remarks; the present talk is primarily an amplification of those statements and a plea for a new point of view for the teaching of chemical engineering.

What I said that called forth so much criticism was approximately this: "The papers presented at this symposium are not chemical engineering. The type of work they represent may be necessary to the advance of chemical engineering, but it is not all of chemical engineering or even the most important part of it."

What I am trying to get across is the idea that, when a piece of purely theoretical research is finished or when the differential equations have been integrated, the problem is not solved, contrary to the opinion of many of the bright boys who are teaching chemical engineering. At that point the problem has just begun. Any definition of engineering that can possibly be worded will refer to it as an art. An art it is, and an art it must remain for a much longer period than will be