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Seminario de diseño de Procesos
UOP

Volume N° 1

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Volume N° 11

Subject: Anexos.

- * Process charts (for hidrocarbon
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ENGINEERING DESIGN SEMINAR

FALL 1991

INSTRUMENTATION

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INTRODUCTION

The title of this session of the Engineering Design Seminar is given simply as "INSTRUMENTATION." That title does describe the subject, but it does not really define the content. Instrumentation is a very broad subject. It has much more material than can be covered in the time allotted. Because of this limitation, and also because it is unlikely that process engineers would be interested in all of the material, we must limit the content of our lecture.

We will not, therefore, go into great detail on the subject of control loop hardware - or digital software. We will only touch lightly on control loop stability and controller tuning; these are subjects better suited to a seminar for instrument engineers. Instead, we will look first at a general overview of the Feedback Control Loop. We will then look at the functions of the loop components, and try to learn how they affect the overall performance of the loop. When we have studied all the components in the loop, and touched on stability and tuning, we will look at a few of the more complex systems and applications. Along with the technical aspects of instrumentation and process control, we will try to convey some understanding of the UOP process control philosophy, as well as the background behind it.

FEEDBACK CONTROL LOOP

HISTORY

The first significant application of closed loop, automatic feedback control was the use of a flyball governor on a steam engine built by James Watt in 1788. Similar devices had been used before, for controlling the speed setting of millstones in water-driven flour mills, but Watt was the first to apply the principle to steam engines. In this application, the flyball governor uses the centrifugal force generated by rotating flyball weights, along with a system of links and levers, to operate a valve that controls the steam flow to the engine, similar to the mechanism illustrated in Figure 1.

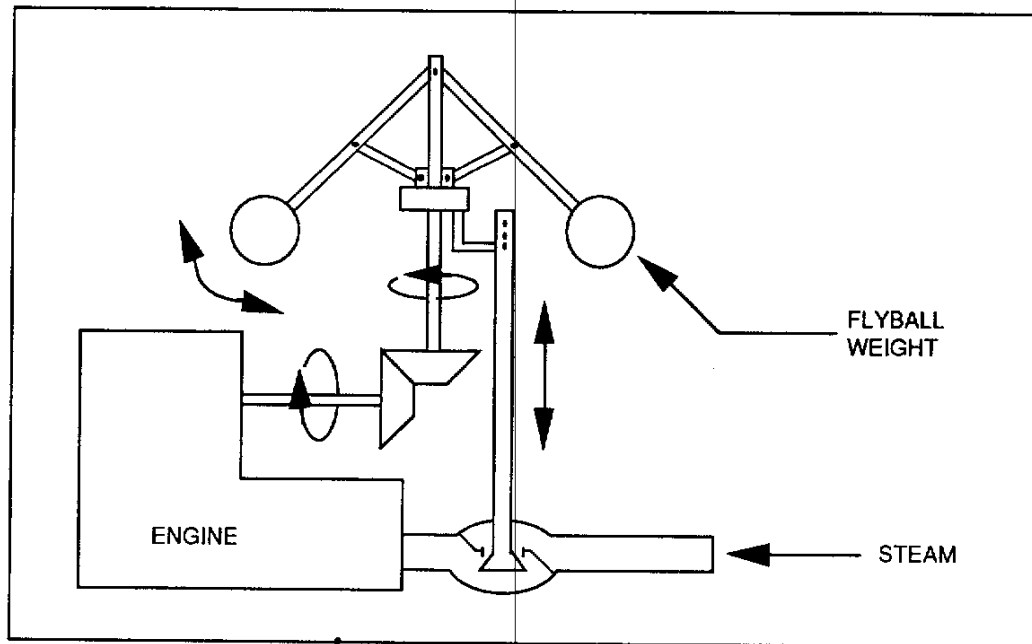


Figure 1 - Flyball Governor

As the speed of the engine in Figure 1 increases, the flyball weights move outward, and the linkage pulls upward on the stem of the steam valve, causing it to close and slow down the engine. The speed set point can be changed by adjusting the length of the linkage connected to the valve stem.

It is interesting to note that before Watt became involved with steam engines, he worked as an instrument-maker. The instruments he made were used for measurement, however, not for control. He studied the craft in London as a youth, and in 1757, at the age of 21, returned home to Glasgow, where he was appointed Mathematical Instrument Maker to the University. Six years later, he was asked to repair a model of a Newcomen steam engine owned by the university, after a London instrument maker had failed to do so. He was successful, and it was that success that inspired his interest in steam engines.

Mechanical feedback controls were not applied to the process industries until about 1900. They did not find general application until the 1930's, as processing moved

from batch to continuous operation and it became more and more difficult to rely on manual control alone. It was during this period that control terms such as Proportional Band and Reset came into general use. As we will see later, these terms derive more or less from a graphical description of the controller operation. Somewhat later, during the Second World War, the increasing use of servomechanisms for positioning and control of guns and related equipment inspired a more detailed study of control theory, and a more mathematical approach to the subject. As a result we now have control terms such as Gain, which is related to Proportional Band, and Integral Time, which is related to Reset. We will return to these terms later, when we discuss the controller's function in the feedback control loop.

LOOP COMPONENTS

Almost always when we mention automatic control, it is three-mode, or PID (Proportional, Integral, and Derivative), feedback control that we mean. There are other, more sophisticated control techniques that are available today, especially with digital systems; however, feedback control accounts for more applications than any other. This is true not only in the process industries, but in other areas as well. There are examples all around us, often with a human component in the loop. For instance, (1) when a person adjusts a water valve to fill a bathtub, feedback control is involved. The person who is filling the tub watches the level, and when it rises to the desired point, operates the water valve and shuts off the water. The level information is fed back visually to the operator, and he takes action accordingly. Likewise, (2) when a float ball in a toilet tank moves a valve to fill the tank to the specified level, feedback control is involved. It is also involved when (3) a person adjusts a valve to control the temperature of a shower bath, and (4) when a thermostat operates a gas valve to control the temperature of a water heater.

These examples are different in some respects. Examples (1) and (3) are - by definition - manual, open loop control, rather than automatic, closed loop; and examples (1)

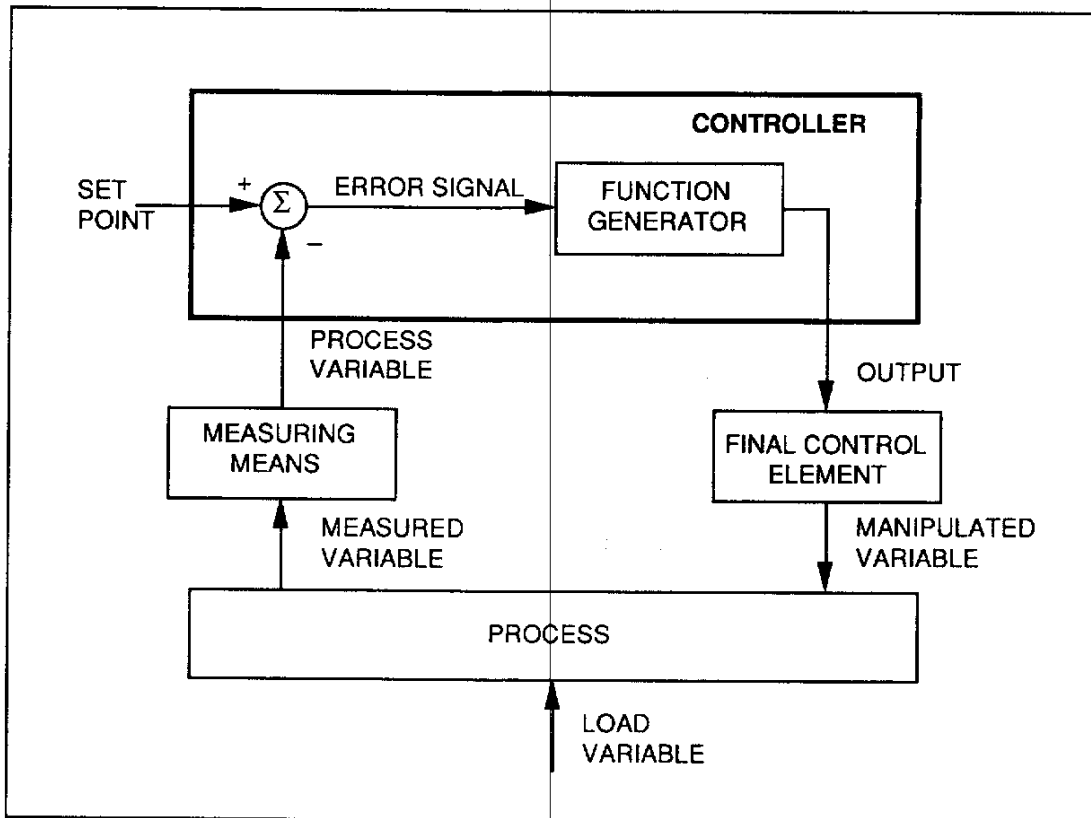


Figure 2 - Feedback Control Loop

and (2) are batch operations, rather than continuous. However, all four examples contain the elements which are diagrammed in Figure 2. In each case, there is a Process, a Measuring Means to measure the Process Variable, and some form of Comparator to compare the value of the Process Variable with the desired value, or Set Point. The resulting difference, or Error Signal, is sent to a Function Generator which operates a Final Control Element. The Final Control Element adjusts a Manipulated Variable as necessary to minimize the Error Signal, and bring the Process Variable closer to the value of the Set Point. Also, each case may be subject to some form of Load Variable, which causes a disturbance in the Process and requires corrective action by the control system. These elements are identified in Table 1.

PROCESS

Although the examples in Table 1 are all related to household plumbing, they do

TABLE 1

Example Number →	(1)	(2)	(3)	(4)
Process	Bathtub	Toilet Tank	Shower	Water Heater
Type	Batch	Batch	Continuous	Continuous
Process Variable	Water Level	Water Level	Water Temp.	Water Temp.
Measuring Means	Eye	Float	Touch	Thermostat
Set Point Adjustment	Hand	Linkage	Hand	Thermostat
Comparator	Brain	Linkage	Brain	Thermostat
Function Generator	Brain	Linkage	Brain	Thermostat
Final Control Element	Water Valve	Water Valve	Water Valve	Gas Valve
Manipulated Variable	Water Flow	Water Flow	Water Flow	Gas Flow
Load Variable	(See Note)	(See Note)	Water Pressure	Water Flow

Note: Normally no load variable; however, load effect could be created by water leakage or variation in water pressure. A Load Variable is any variable, other than the Set Point, that has an affect on the Process Variable.

Table 1 - Feedback Control Loop Examples

illustrate several important points. The first point is that although the processes are different, they all have feedback loops, and the same principles apply to all. In fact, the process may be almost anything: it may be mechanical, electrical, chemical, thermal, hydraulic, or pneumatic; each must follow the same laws of physics and dynamics, and each is described by the same mathematical equations. The coefficients of the equations may differ, but the equations remain the same; this is what forms the foundation for dynamic modeling and computer simulation.

The second point is that in continuous processes there are two external inputs to the control loop: a Set Point and a Load Variable. The control loop must respond to a change in either input. The response to a change in Set Point is called the servomechanism response; the response to a change in Load Variable is called the regulator response.

The regulator response is what separates the instrument engineer from the process engineer. Except for studies in reaction kinetics, the process engineer generally is

concerned only with steady state operation. The instrument engineer knows that true steady state operation never exists. He knows that there are always load changes and upsets in the system which must be countered and corrected, and that the dynamics of the system must be considered.

A third point worth noting is that the Process Variable and the Manipulated Variable very often are not the same. In the case of liquid flow control, they are directly related, but in many other cases, the relationship is not direct; e. g., a fired heater temperature controller may operate a fuel gas valve, which changes the fuel gas burner pressure (the Manipulated Variable), which changes the fuel gas flow through the burner orifice, which changes the temperature in the heater firebox, which (finally) changes the process temperature (the Process Variable). As we shall see later, the more indirect the relationship between the Manipulated Variable and the Process Variable is, the more difficult the controller's task is likely to be.

MEASURING MEANS

To be able to control a process, we must first be able to measure a Process Variable. Generally, this requires a primary element located in the process unit out in the field, which converts some intrinsic property of the process into a measurable signal, and a transmitter which converts the signal from the primary element into a standard instrument signal. Standard instrument signals for transmission from the field to the control room are 3 to 15 psig or the nearest metric equivalent (0.2 to 1.0 kg/cm², 0.2 to 1.0 Bar, etc.) for pneumatic systems, or 4 to 20 mA DC for electronic current systems, corresponding to 0 to 100% of the span of the transmitter.

The transmitted signal is sent to the receiving instrument in the control room. The pneumatic signal used in the control room is identical to the signal used in the field, but the electronic current signal generally passes through a resistor and is converted into a nominal 1 to 5 volt DC signal that is used by the control room instruments. A signal based on current, rather than voltage, is used for transmission from the field

because the transmission wire always has some resistance, which would cause a drop in the voltage and thereby cause an error. Since the wire length varies from one installation to the next, the error also would vary. The error could be minimized if the receiving instrument were designed with a high input impedance, but this would increase the chances of induced noise pickup in the transmission wire, and also add more complication to the circuit design.

Although the primary element is always located in the field, the transmitter need not be. As we will see, a thermocouple that is located in the field may be connected to a millivolt transmitter that is located in the control room. It may be that the transmitter is in the field, but located separately from the primary element, as in the case of a differential pressure transmitter that is near- but separate from- the orifice plate that measures a flow. And it may be that the transmitter and the primary element are located in the same enclosure, as in the case of a Bourdon pressure element located inside the housing of a pressure transmitter.

There are several main types of Process Variable in the process industries, and several varieties of primary element used in each type:

Temperature

Temperature is probably the most important of the main Process Variables, because chemical compositions and reaction rates are closely related to temperature. Other variables, such as flow or pressure, are often controlled primarily to minimize load changes that would otherwise affect the temperature controller, and make its difficult task even more difficult.

The primary element used most often for temperature measurement is the thermocouple. The operation of the thermocouple is based on the Seebeck effect. Seebeck discovered in 1821 that an EMF (electromotive force) is developed when opposite junctions of certain dissimilar pairs of metals are exposed to different temperatures.

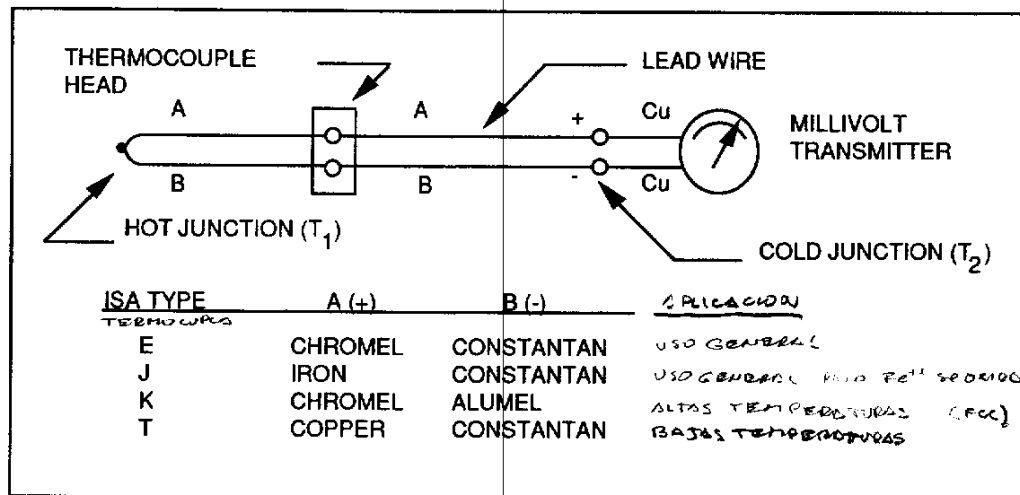


Figure 3 - Thermocouple

The EMF developed varies as the temperature difference between the two junctions varies. The EMF also depends on the thermocouple materials used, but for a given material combination (thermocouple type) the relationship between EMF and temperature difference is predictable. Therefore if the thermocouple type and the cold junction temperature are known, the hot junction temperature can be found: all we need to do is measure the EMF, consult the appropriate thermocouple EMF table to find the corresponding temperature difference, and add the temperature difference to the cold junction temperature. A typical thermocouple circuit is shown in Figure 3.

(We should note that an intermediate junction exists at the thermocouple head in the Figure. This is actually formed from two junctions of opposite polarity, and if the temperature is the same at both junctions, the two EMF's will cancel each other, and the net effect on the overall EMF will be zero.)

The material combinations commonly used for industrial thermocouples are Iron and Constantan (designated ISA Type J), Chromel and Alumel (ISA Type K), Chromel and Constantan (ISA Type E), and Copper and Constantan (ISA Type T). The first material in the combination is electrically positive, the second is negative. Chromel, Alumel, and Constantan are proprietary alloys, but are widely available, so the thermocouple is still a relatively inexpensive device. It is also simple, and therefore fairly

rugged and reliable. There are other material combinations that are used sometimes, such as Platinum/Rhodium and Platinum; but these are more expensive and therefore generally limited to specialized applications, such as lab work.

The thermocouple specified for general use by UOP is the Chromel-Constantan, Type E, thermocouple, fabricated from #14 AWG (American Wire Gauge) wire, which is 0.064 in. (1.63 mm) in diameter. In this configuration, the thermocouple can be used for temperatures up to about 1200 °F (650 °C). For many years, Iron-Constantan thermocouples were the industry standard for temperatures up to about 1000 °F (540 °C); however, iron oxidizes rapidly, especially in the humid coastal atmospheres where refineries are often located. Because of the problems this causes, UOP now specifies Chromel-Constantan thermocouples. Chromel-Constantan thermocouples do not oxidize, and also have the advantage of having the highest EMF for a given temperature, as shown in Figure 4.

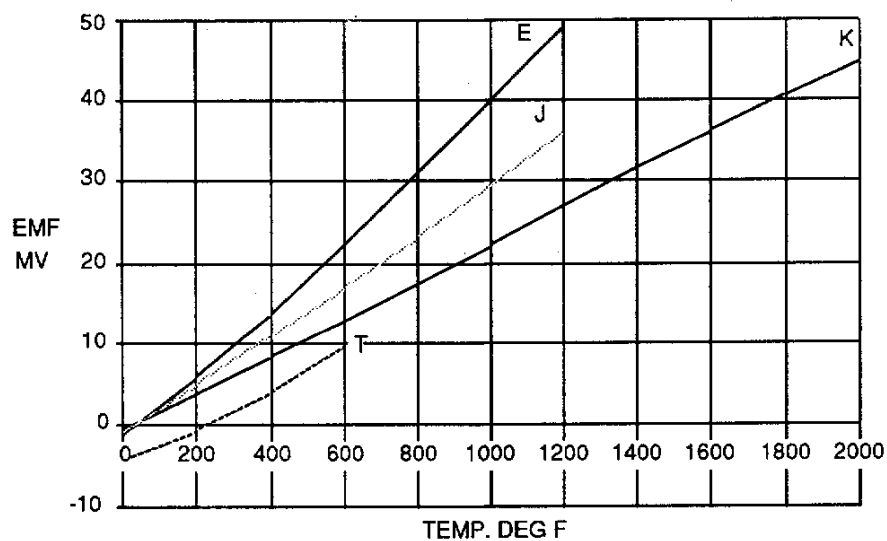


Figure 4 - Thermocouple EMF

For temperatures above 1200 °F, the thermocouple used most often is Chromel-Alumel, Type K. The recommended upper temperature limit for a #14 AWG Chromel-Alumel thermocouple is 2000 °F (1093 °C). The Copper-Constantan, Type T, thermocouple is used primarily for cold temperatures, from 32 °F down to -300 °F (0 °C).

down to -184°C).

For Chromel-Constantan thermocouples, the standard limits of error are $\pm 3^{\circ}\text{F}$ (1.67°C) for temperatures between 32 and 600°F (0 and 316°C), and $\pm 1/2\%$ for temperatures from 600 to 1600°F (316 to 870°C). The percentage applies to the temperature being measured; for example, if the temperature is 1000°F , the allowable error is $1/2\%$ of 1000 , or 5°F .

Thermocouples do have some minor disadvantages. In discussing Figure 3, we noted that the hot junction temperature can be found if the EMF and the cold junction temperature are known. Millivolt transmitters used for thermocouple measurements do not simply measure the EMF developed by the thermocouple; they are also furnished with special cold-junction compensation circuits that, as the name implies, measure and compensate for changes in temperature at the cold junction. If ordinary copper wire is used to connect the thermocouple in the field to the millivolt transmitter in the control room, the cold junction will be located at the thermocouple head, not in the control room, and its temperature will not be known. Therefore it is necessary to use extension wire that is made of the same materials as the thermocouple between the thermocouple head and the millivolt transmitter. This makes the installation more expensive. Some instrument manufacturers now offer millivolt transmitters that are designed to be mounted at the thermocouple in the field, and thus allow copper wire to be used for transmission to the control room. UOP does not feel that this is a good solution. With this design, the cold junction circuitry in the transmitter is exposed to wider ranges of ambient temperature than if the transmitter were located in the control room. Also, any test equipment used to service the transmitter in the field also must be suitable for the wider temperature range. Finally, quite often the thermocouple is not located with service in mind when the plant is built, and access to it may be difficult, and may even be dangerous.

UOP standards for thermocouple extension wire call for the wire to be continuous without any splices from the thermocouple head to the millivolt transmitter. This

avoids the problems that may arise with junction boxes, such as loose or corroded terminals, or connecting wires with the polarities reversed. Also, as indicated earlier, the EMF's from intermediate junctions cancel each other if the temperatures at the two junctions are equal. If they are not for some reason, such as uneven heat from the sun, or drafts from poor weatherstripping, an error will result.

For Chromel-Constantan extension wire, the limits of error are ± 3 °F (1.67 °C) from 0 to 400 °F (-17.8 to 204 °C). So for a thermocouple reading a temperature of 1000 °F (538 °C), the error of the thermocouple could be ± 5 °F (2.78 °C), and the error of the extension wire could be ± 3 °F (1.67 °C), for a total error of ± 8 °F (4.44 °C). As the thermocouple ages, the EMF for a given temperature tends to drift downward, so the overall error may increase. Fortunately, we are generally more interested in reproducibility than in absolute accuracy, and the error does not cause a large problem.

Another disadvantage of the thermocouple is that while the relationship between millivolt transmitter output and EMF is linear, the relationship between temperature and EMF, as shown in Figure 4, is slightly non-linear. Therefore the charts and scales used for thermocouple measurements must be characterized to compensate for the non-linearity. And, lastly, the minimum span available with most millivolt transmitters is between two and five millivolts, depending on the manufacturer. This translates into a minimum temperature span of about 50 to 100 °F (28 to 56 °C), depending on which type of thermocouple is used.

For narrower temperature spans, down to about 5 to 10 °F (2.8 to 5.6 °C), resistance temperature detectors are usually used. The Resistance Temperature Detector, or RTD, is in essence a very long, thin wire - usually nickel or platinum - which has been wound into a small coil. The wire has a stable, accurately known relationship between electrical resistance and temperature. A Wheatstone bridge circuit, the circuit shown in Figure 5, is usually used to accurately measure the resistance, and thereby the temperature.

Resistance temperature detectors are more expensive than thermocouples, are not as

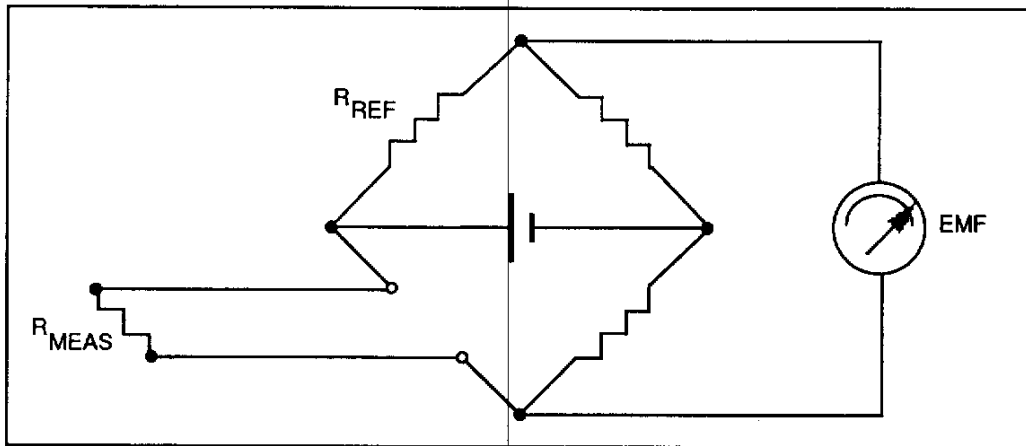


Figure 5 - Resistance Temperature Detector
Wheatstone Bridge Circuit

ugged, and cannot be used at temperatures as high as thermocouples. One advantage that they do have over thermocouples is that ordinary copper wire can be used to connect the RTD to the control room equipment; however, because the circuit is so sensitive to resistance, the resistance of the copper lead wires, which itself is a function of the ambient temperature, also must be taken into account. This generally is done by using three lead wires, connected so the lead wire resistances have equal, counterbalancing effects on the measurement and reference legs of the Wheatstone bridge. Figure 6 illustrates how this can be accomplished.

As we will see later, thermocouples or RTD's sometimes are used for differential tem-

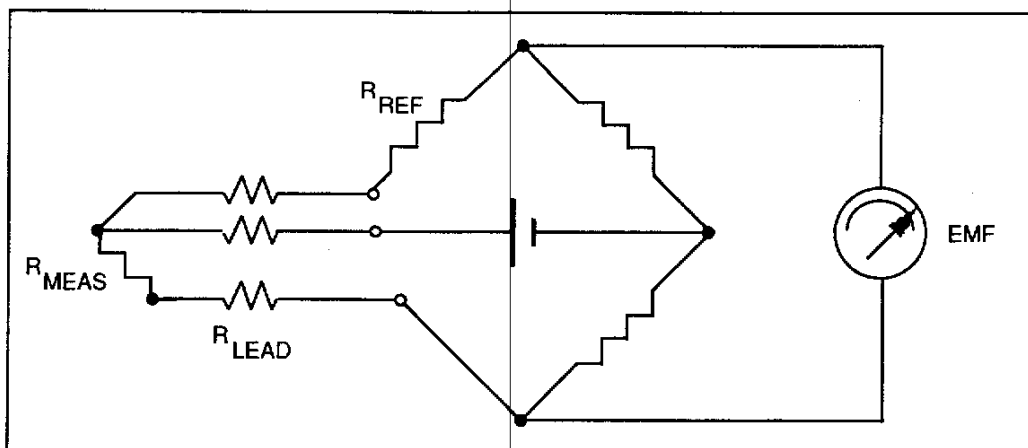


Figure 6 - Resistance Temperature Detector
Three-Wire Circuit

perature measurement, in which one element is used to measure a reference temperature, and the other is used to measure an objective temperature. Millivolt transmitters used for differential thermocouple measurement do not have cold-junction compensation circuits; in this case the reference thermocouple becomes the cold junction. Figure 7 shows how the Wheatstone bridge circuit is modified for differential temperature measurement with resistance temperature detectors. *050 20 PROCESS CONTROL*
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A filled thermal system is another type of primary device used for temperature meas-

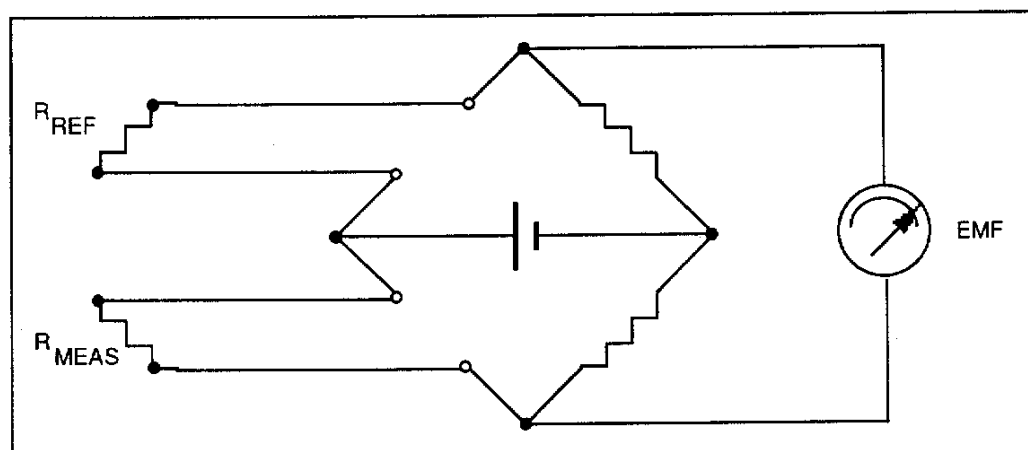


Figure 7 - Resistance Temperature Detector
Differential Temperature Circuit

urement. A common mercury thermometer is one example of a simple filled-system device, but for process measurements, this type of system usually relates the expansion of a liquid, gas, or vapor to the temperature by using a mechanism similar to that shown in Figure 8. Systems that use a liquid fill are subject to errors from the liquid head in the capillary tubing between the bulb and the instrument. The same may be true for the system with the vapor fill. Depending on the relative temperatures of the process and ambient, and the material used to fill the system, the bulb may be filled with vapor, and the capillary filled with liquid, or vice versa. Systems with a gas fill do not have this problem.

At UOP, filled thermal systems are used only for relatively non-critical services such

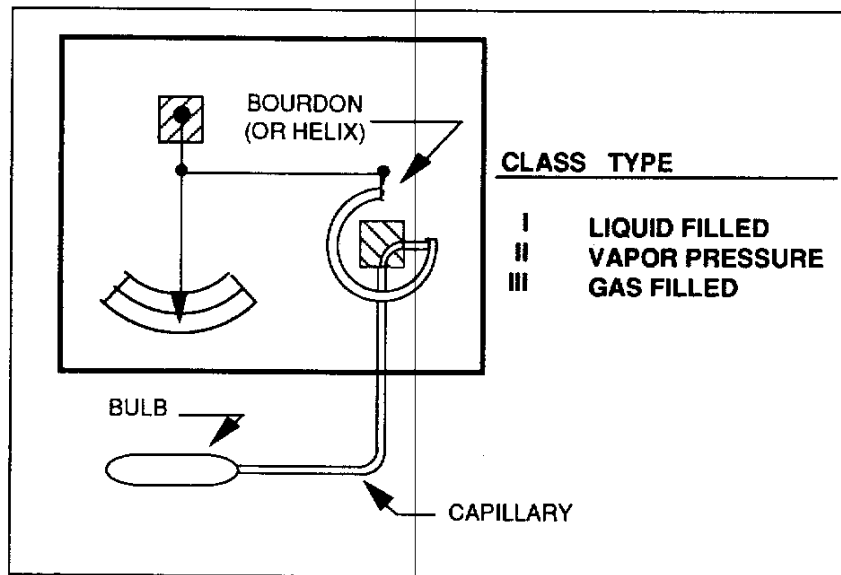


Figure 8 - Filled Thermal System

as air tempering controls, which are used on air fin exchangers in cold climates to recirculate some of the warm air and prevent freezing of the process fluid in the exchanger. The reason is that filled systems are not fail-safe. A ruptured temperature bulb or capillary would result in a low temperature reading. A controller would interpret this as a call for more heat, which could be dangerous. Failure of a resistance bulb results in an open circuit, or infinite resistance, which translates to an infinite temperature and causes the controller to decrease the heat, rather than increase it. Thermocouples are not inherently fail-safe, but a circuit can be added to the millivolt transmitter to drive the output upscale in case of thermocouple burnout, and thus fail in a safe manner.

Flow

Probably 60 to 75% of all process measurements are flow measurements. At UOP, probably 98% of all flow measurements use an orifice plate and a differential pressure instrument. Flow measurement with an orifice plate is based on Bernoulli's relationship between pressure and velocity. The area of the orifice is smaller than the area of the pipe, so the velocity of the fluid must increase as it passes through the orifice. The increase in velocity is accompanied by a decrease in pressure, as the static head is

difference in the lead lines increases, and it also increases as the difference in elevation between the orifice and the transmitter increases. As a percentage of flow, the error

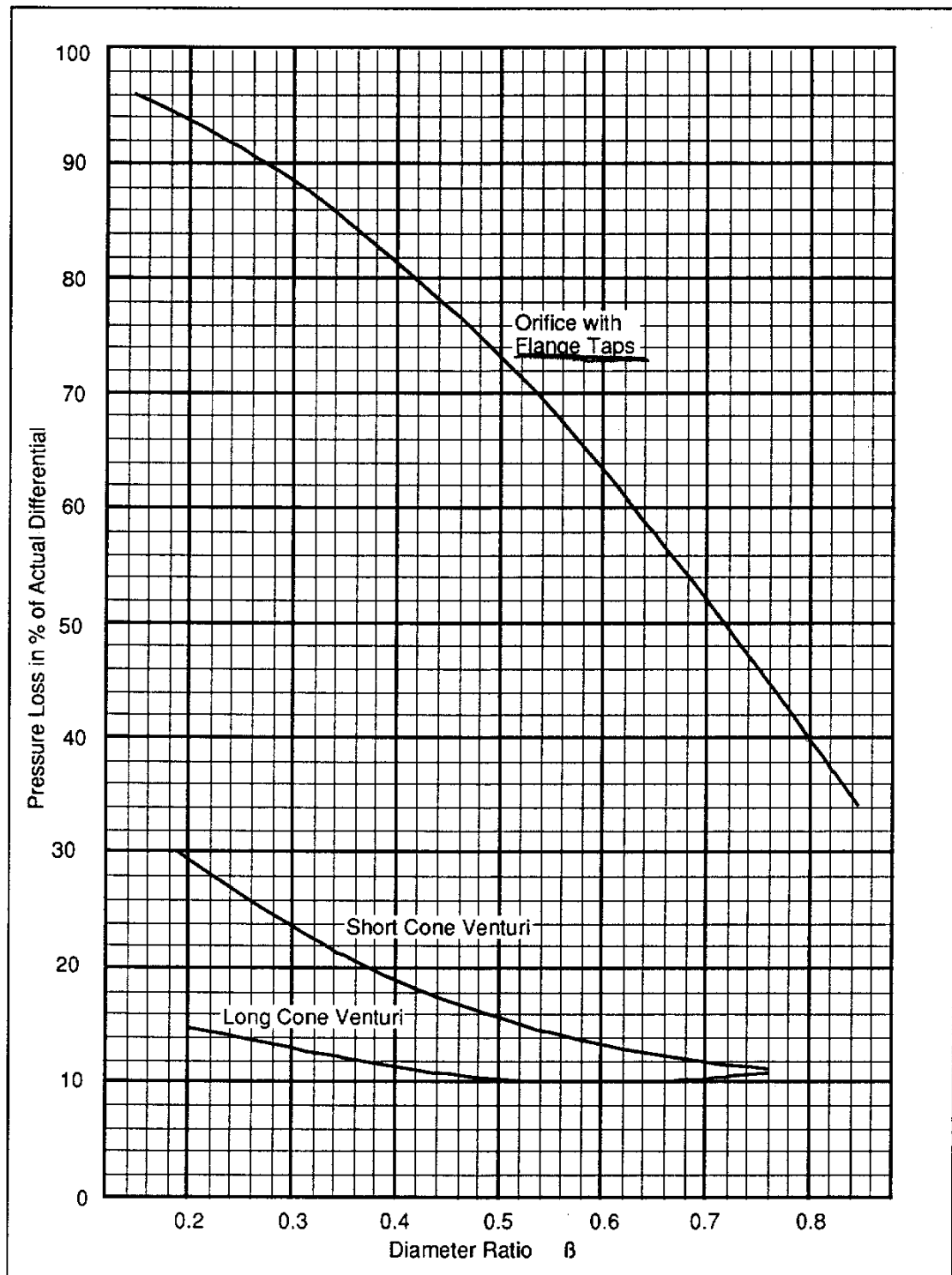


Figure 11 - Overall Pressure Loss

decreases as the orifice differential increases. However, there is an upper limit on the orifice differential that is set by the economics of pumping the fluid. The industry standard differential range is 0 to 100 inches of H_2O or the nearest even metric equivalent (0 to 2500 mm H_2O , 0 to 25 mBar, etc.). The meter maximum generally is chosen so the normal flow is at mid-scale on the 0 to 10 square root chart. This is about 7 chart units, equivalent to 50 inches of H_2O . In terms of permanent loss, the plot in Figure 11 indicates that for a 0.7 β -ratio (ratio of orifice diameter to inside pipe diameter), the permanent loss is about 52% of the operating differential. For an operating differential of 50 inches of H_2O , the permanent loss is 26 inches of H_2O , or 0.94 psi. For a 0.2 β -ratio, the loss would be 94 %, or 1.7 psi.

An orifice is characterized by its β -ratio. This ratio is generally limited to a range of 0.10 to 0.70. Outside of this range there is not much experimental data, and the accuracy of the correlations cannot be confirmed. Most lines sized in accordance with UOP practices will not require β -ratios greater than 0.70. Some lines, however, that are "sized tight", and have higher pressure drop per unit length, may require that a larger size line be used, at least for the meter run portion of the piping (the orifice flanges and the required upstream and downstream straight run). A certain amount of straight run is always required to insure that a fully developed turbulent flow profile exists at the orifice; there must be dynamic equivalence between the orifice and the experimental installation from which the correlations were taken, or the correlations cannot be applied. The straight run requirements for an orifice are dependent on the β -ratio; the requirements used at UOP, shown in Figure 12, are based on a β -ratio of 0.7 to permit future enlargement of the orifice without also requiring alterations to the piping.

For a quick check of the β -ratio when writing specifications, a table of Limiting Values for Orifice Plates has been developed. This table is in the memo titled Orifice Beta Ratio Limits located in the Appendix ^{pag A-5} at the end of this section. The equations used in the memo to generate the numbers in the table illustrate how the basic orifice

converted into velocity head. The fluid passing through the orifice forms a jet which has minimum area (and minimum pressure) at a point called the vena contracta. As the fluid travels downstream from this point, the jet expands and the pressure increases. Approximately eight pipe diameters downstream of the orifice, all possible pressure recovery has taken place; the difference between the pressure at this point and the pressure upstream of the orifice is the permanent pressure loss.

In most installations, flange taps are used to measure the orifice differential pressure. These taps, shown as FT_1 and FT_2 in Figure 9, are located one inch upstream and one inch downstream from the orifice plate. Pipe taps, located 2-1/2 diameters upstream and 8 diameters downstream from the orifice plate, are shown as PT_1 and PT_2 . The

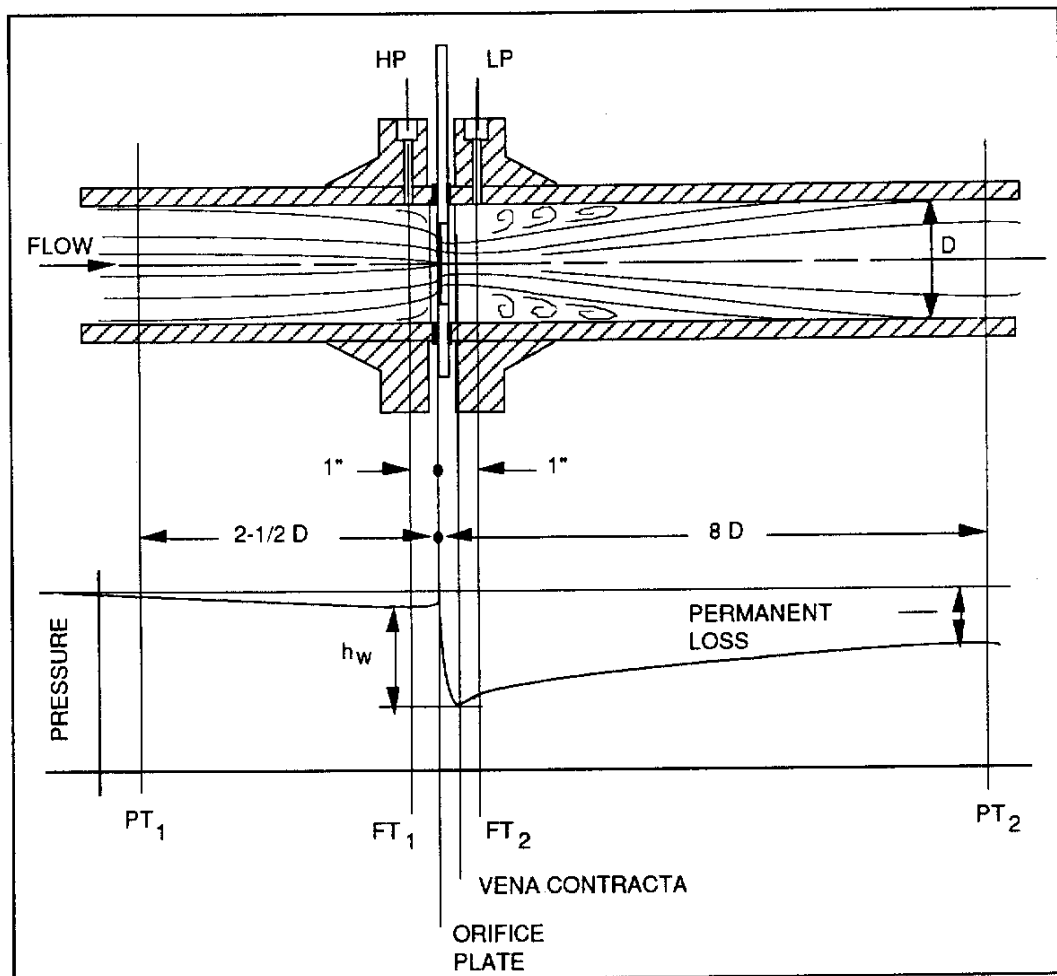


Figure 9 - Orifice Pressure Profile

pipe taps essentially measure the total permanent loss of the orifice. Other tap locations are possible, but flange taps are used most often. One reason is because the orifice flanges are shop fabricated, and can be made to closer tolerances than field fabricated pipe tap installations. Also, as the figure indicates, the differential pressure for a given flow rate is greater for flange taps than it is for pipe taps.

A high differential pressure is advantageous because it tends to minimize the effects of variations in fluid density in the transmitter lead piping. The transmitter generally does not measure the orifice differential directly; it measures the sum of the orifice differential plus the static heads in the transmitter lead lines, as shown in Figure 10. If the static heads are different because the material in the leads lines is of different compositions or at different temperatures, then the differential pressure at the transmitter will not be the same as at the orifice. The resulting error increases as the density

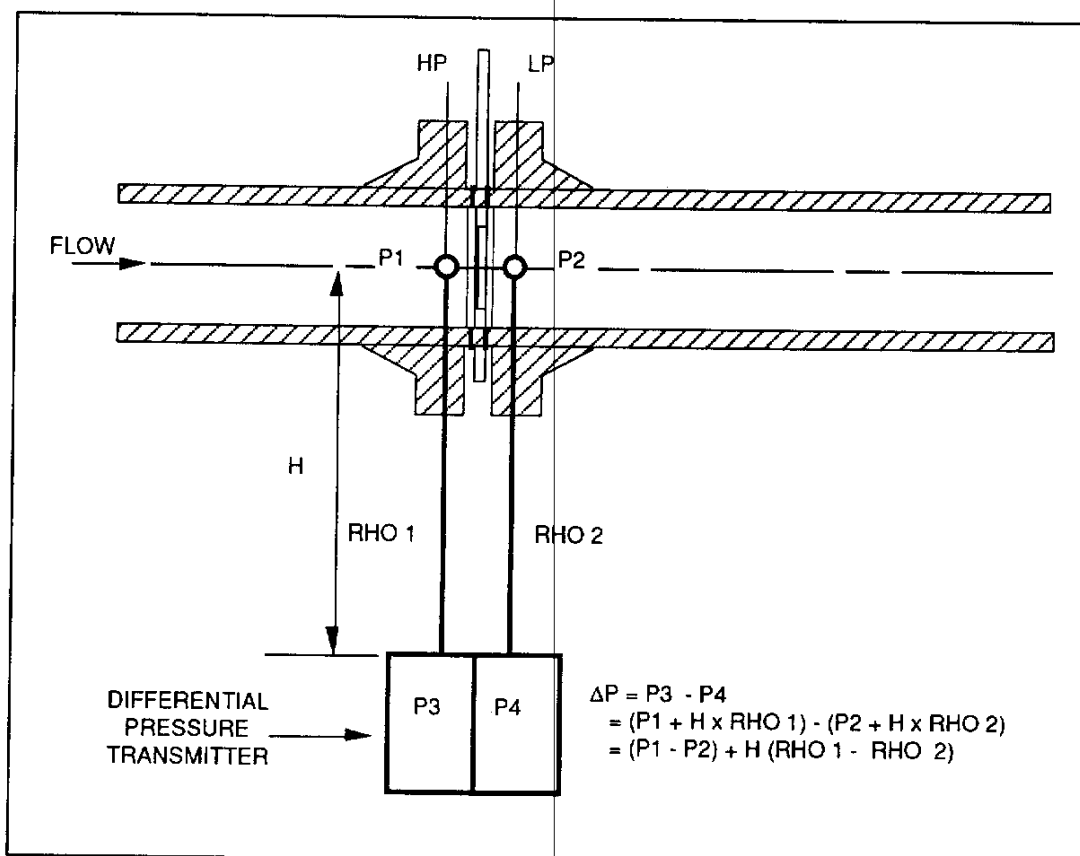


Figure 10 - Transmitter Static Head Effects

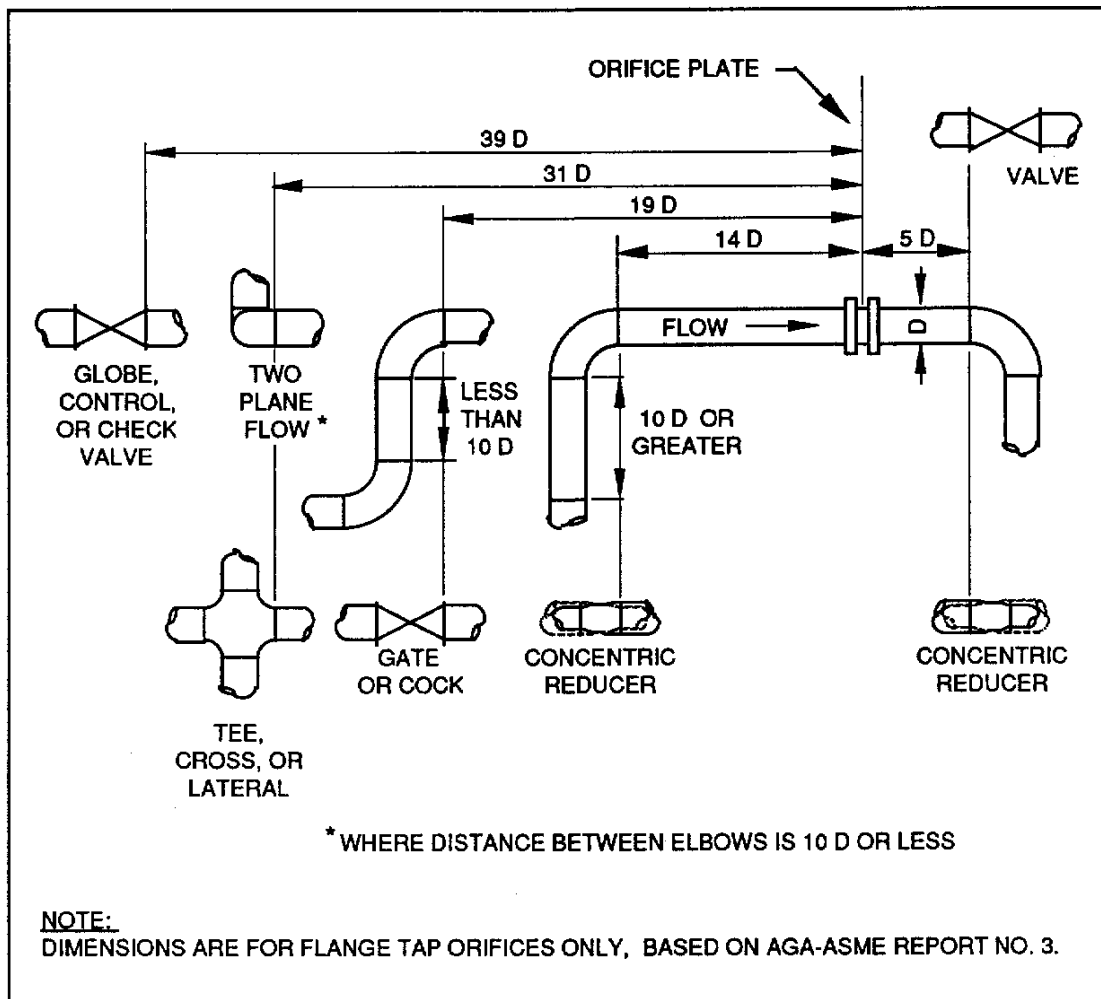


Figure 12 - Orifice Minimum Straight Run Requirements

equations can be manipulated to calculate corrections for conditions other than design.

For compressor suction measurements, or other services where the permanent loss must be held to a minimum, venturi meters are used. The venturi's tapering form is designed to guide the flow streamlines in toward the venturi throat, and to expand them back out to a fully developed velocity profile without developing the turbulence and boundary layer separation that occurs with an orifice plate. As a result, the static/kinetic energy exchange is more nearly reversible, and the energy losses are reduced. The permanent loss for a classical, Herschel type venturi is given in Figure 11 as approximately 10% for a β -ratio of 0.7, versus the 52% noted above for the orifice plate, and operating costs for the venturi will be correspondingly lower. Capital costs

will be higher for the venturi, however, since it is larger and more difficult to fabricate. Also, while the orifice often can be rebored to a larger diameter if the plant throughput is increased, it is not possible to make changes in the throat diameter of a venturi without also changing the inlet and outlet cones and the overall length of the venturi. The dimensional relationships of the venturi are shown in Figure 13.

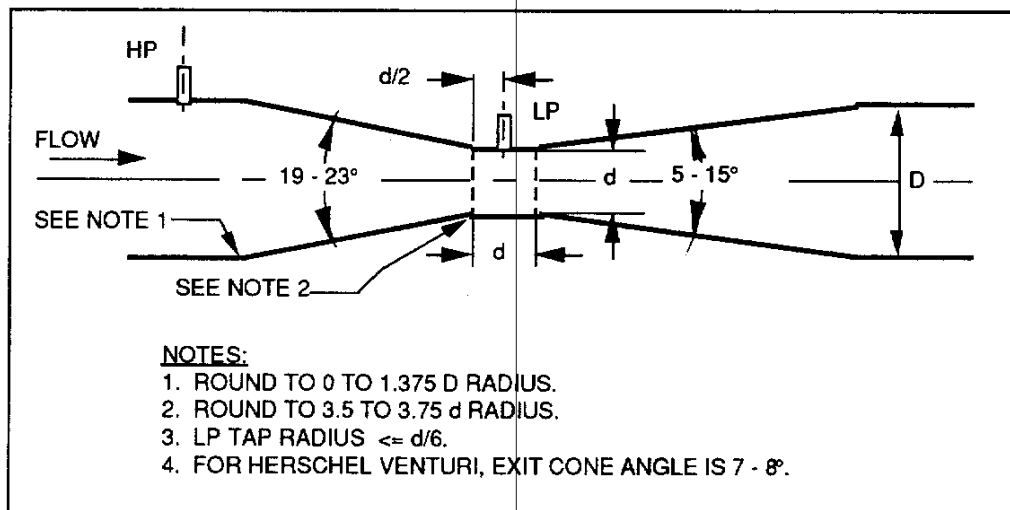


Figure 13 - Dimensions of Classical Venturi Tube

An orifice meter that is installed in accordance with UOP Standards, and used at the design flow rate, can be expected to have an overall accuracy (including the uncertainty of the orifice coefficient and the accuracies of the transmitter and receiver) of about $\pm 2\%$. As can be seen from Figure 14, because of the square root relationship between flow and differential pressure, a given error in differential pressure causes a much larger error in flow at reduced flow rates than it does at the design rate.

If accuracy greater than 2% is required for liquid streams, tank gauges should be used. Alternatively, turbine meters can be used. These have accuracies of about $\pm 0.25\%$, but are more expensive than orifice meters, and have moving parts that require more maintenance.

Other methods of flow measurement are available, but none has the low cost and versatility of the orifice meter. There are "mag" meters, which measure flow by means of

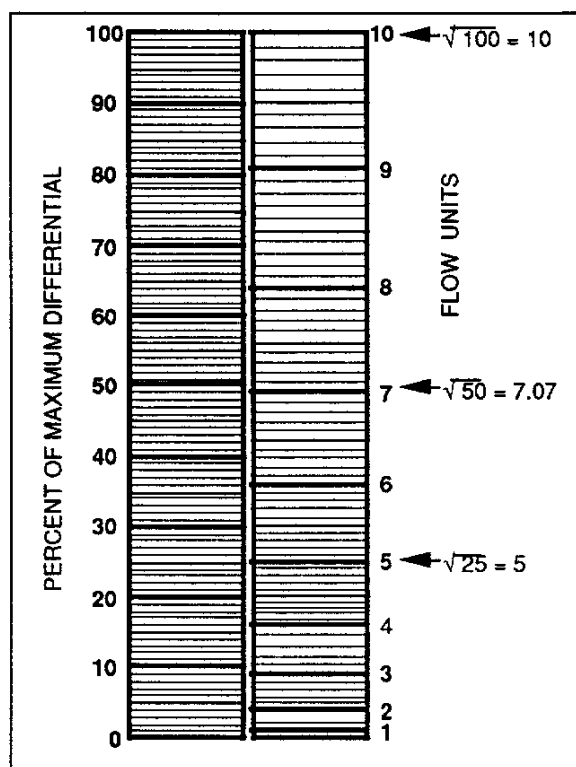


Figure 14 - Square Root Relationship
Head vs. Flow

magnetic induction, but with these the flowing fluid must be electrically conductive. Gases, steam, and most hydrocarbon liquids will not work. There are vortex meters, which use a body in the flowing stream to generate vortices that are shed at a rate proportional to the flow rate. Currently, most of these are limited to temperatures of about 400 °F (204 °C); newer designs are coming out with higher temperature limits, but still not equal to an orifice plate. Ultrasonic meters which use the transit time or the Dop-

pler effect of a sound wave to measure the velocity of a fluid. But these are also temperature limited. In addition, the transit time meter will not work well on dirty fluids, while the Doppler effect meter fails on clean fluids. Rotameters and target meters are used for some special purposes, but range changes present problems for rotameters, and the largest standard target meter for liquids is for a 4-inch line. Positive displacement meters can be used, but these are primarily for flow integration, not for flow rate. Overall, the orifice meter still remains the first choice.

The 2% accuracy of the orifice installation is generally adequate for control purposes. For control, if the flow signal is sufficiently reproducible, greater accuracy is welcome, but not a mandatory consideration. Except for charge streams, most flow controls actually can be considered as "dependent" variables. They are adjusted by the operator, or by another controller, as required to achieve other processing objectives, such as level, temperature, or composition. If the operator can dependably relate a

given temperature to a given flow set point, the absolute value of the flow at that set point is not important to him. For charge and product meters, if greater accuracy is desired, the meters usually can be calibrated against tank gauges, as mentioned earlier.

At present, UOP has only one area, the group of processes known as SORBEX®, where greater accuracy is required. In these processes it is necessary to accurately calculate and control a volumetric flow set point which must change abruptly at intervals ranging from approximately 1.5 to 10 minutes. There are no tanks in this part of the process, and if there were, tank gauges could not be used to calibrate the flowmeters in such a short period of time. For the flows involved in these calculations, turbine meters are used.

The transmitters used with turbine meters furnish a pulse output to the receiving equipment in the control room; the mag flow meter furnishes a current output. The orifice meter uses a differential pressure transmitter to convert the orifice differential into a standard pneumatic or electronic current signal. Many such transmitters measure the force exerted by the differential pressure on a diaphragm. Early transmitter designs used a single diaphragm. To be sensitive to small differentials, the diaphragm must be very thin and flexible. If one side of the transmitter is accidentally vented to atmosphere while the other side is still exposed to full line pressure, the diaphragm will rupture. Most transmitter designs now use a capsule made from two such diaphragms on either side of a heavy support member that can withstand a differential pressure equal to the full design pressure of the transmitter. The capsule is filled with a silicone liquid that, under normal conditions, transmits pressure from one diaphragm to the other through passages in the support (Figure 15). However, if the differential pressure becomes too high, the diaphragm on the high pressure side contacts a stop, and closes a valve or a port. Because the silicone liquid is incompressible, its pressure rises instantly to the same pressure as the process, the differential pressure across the diaphragm becomes zero, and rupture of the diaphragm is prevented.

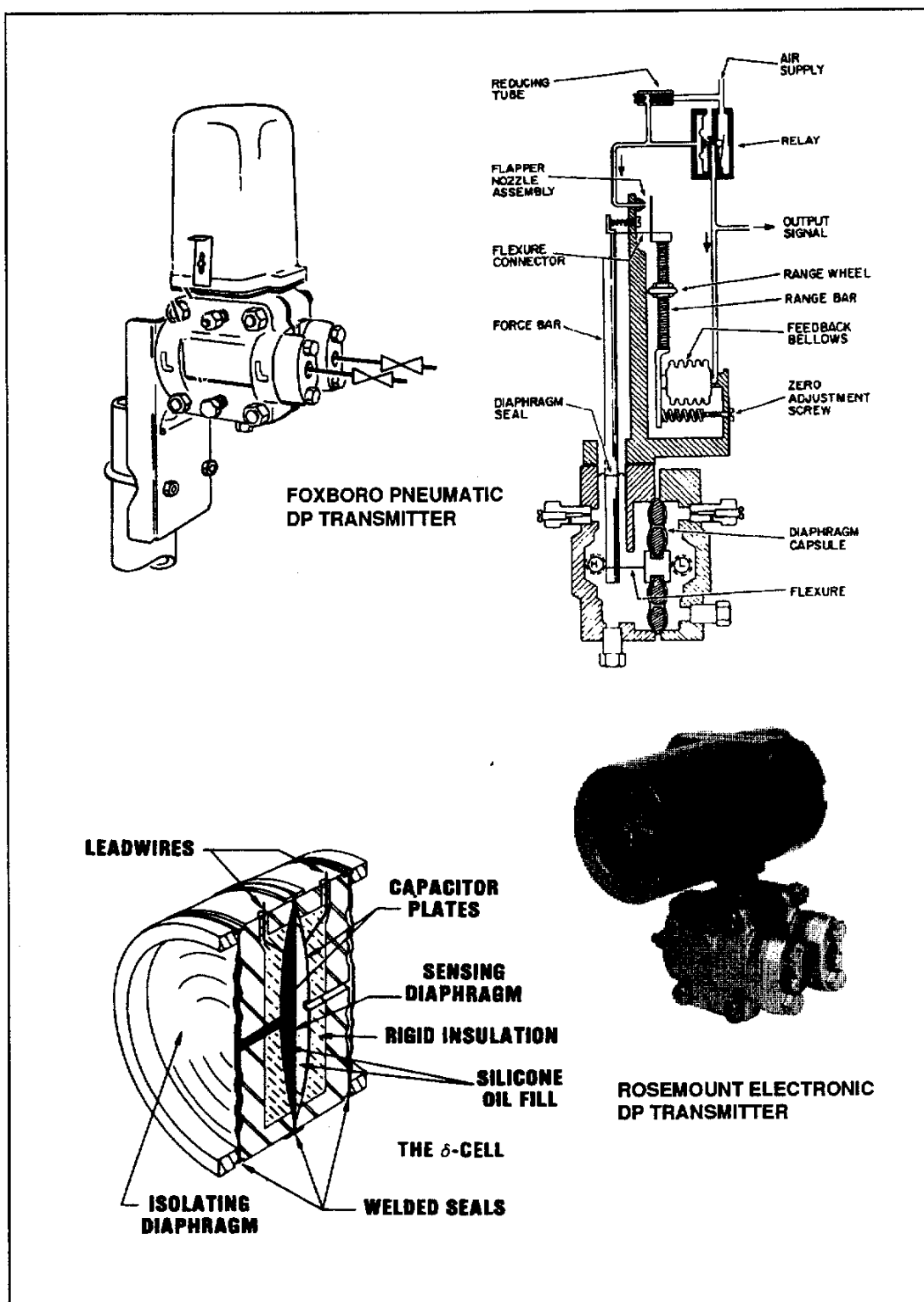


Figure 15 - Diaphragm Type Differential Pressure Instrument

The differential pressure transmitter design above is sometimes called the "force balance" type. The force of the differential pressure on the diaphragm is balanced by an opposing force generated by the output signal acting on a feedback bellows or coil. The actual displacement of the diaphragm as the differential pressure changes from zero to the maximum is therefore only about 0.003 cubic inches. This means that for steam or condensible vapor installations, it is not necessary to use seal pots. Another common design is the "bellows" type (Figure 16). This is actually a "motion balance" instrument; the force of the differential pressure on the bellows is opposed only by a range spring. This results in a larger displacement over the operating range than in the case of the force balance instrument, and if accuracy is desired in steam and condensible vapor installations, seal pots should be provided to maintain equal condensate levels in both lead lines. The diameter and displacement of the bellows in these instruments are such that the differential pressure can drive a pointer or a pen directly, without amplification. This is useful for local indication or recording in the field. Instrument air is required only if the instrument is also a controller.

The two bellows in the bellows type instrument are usually separated by a passage that is equipped with an adjustable restriction. Adjusting the restriction changes the degree

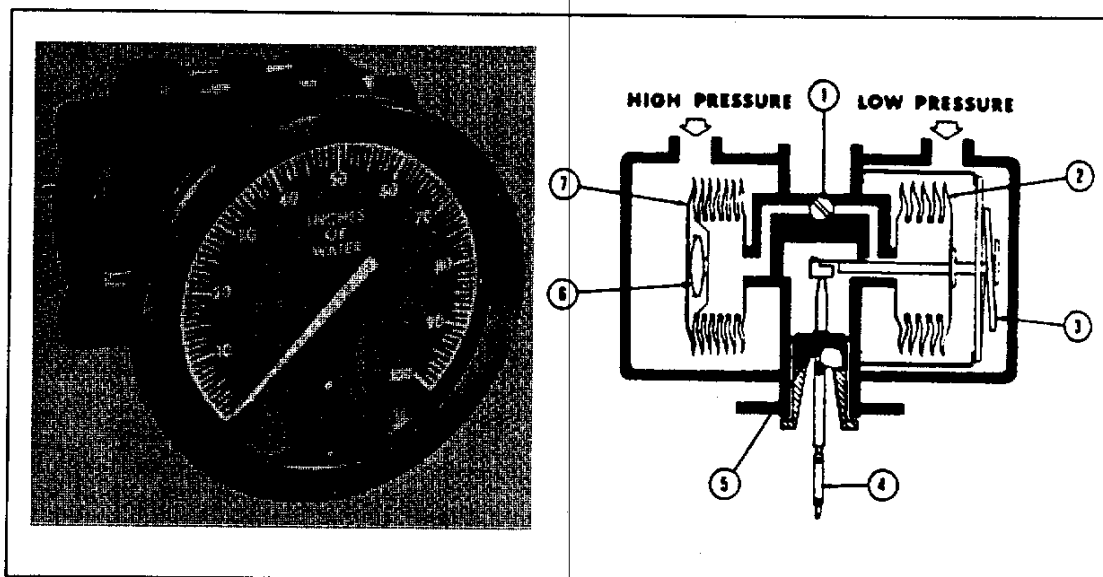


Figure 16 - Bellows Type Differential Instrument

of dampening of fluctuations in the differential pressure. This is useful for pulsating or noisy flows, such as in the discharge from a reciprocating pump

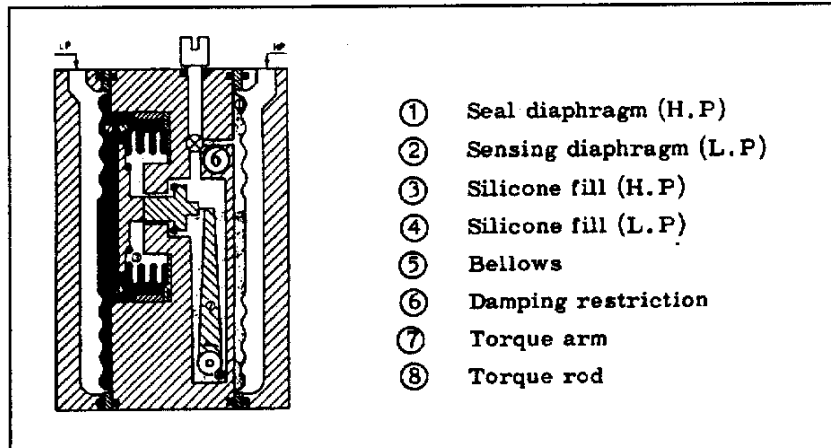


Figure 17 - Diaphragm Type Differential Instrument (with Dampening) or compressor. A few manufacturers have transmitters (Figure 17) that combine the basic force balance design with the externally adjustable damping of the bellows type. Unfortunately, these designs are rapidly becoming extinct.

The reason this is unfortunate is that the transmitters that are replacing them use electronic damping on the output signal, or on the input to the receiving instrument, rather than mechanical damping. Electronic damping is satisfactory as long as the magnitude of the pulsation is within the range limits of the transmitter, but it is impossible to tell from looking at the output signal if this is truly the case. If the magnitude is large enough, or if the transmitter is operating near the top of its range, the transmitter primary mechanism can be driven to its limit, and cause clipping of the signal (Figure 18). When this occurs, an error is introduced into the flow measurement, because, as

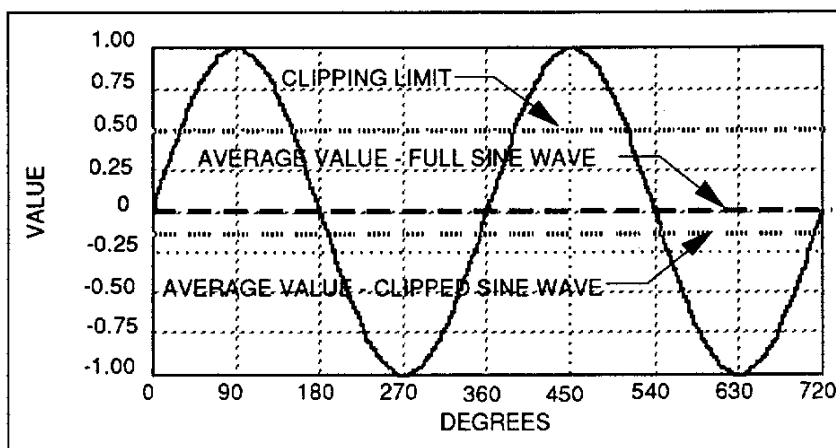


Figure 18 - Sine Wave Clipping

Figure 18 shows, the average of the clipped signal will be less than the actual average. The mechanical damping, with

an externally adjustable restrictor, provides a physical resistance to the pulsation, and reduces the possibility of overtravel and consequent clipping.

Although pneumatic transmitters are essentially limited to the force or motion balance designs, electronic transmitters are able to take advantage of various electrical phenomena to generate the output signal. Some use strain gages to measure the stress on the diaphragm due to the differential pressure; others measure the resonant frequency of a wire that changes with the differential; still others relate the differential to a capacitance that changes with the deflection of an element in the capsule. Any pulsation damping that is done with these transmitters must be done electronically.

Using the silicone filled capsule design to prevent the diaphragm from rupturing solves one problem, but can create another. Except for certain special cases, such as HF Alkylation units, UOP specifies standard stainless steel trim for instruments which are not in the flowing stream, such as pressure gauges, pressure instruments, and differential pressure instruments. For instruments that are in the flowing stream, such as thermowells, control valve internals, and orifice plates, UOP specifies Monel if chlorides or hydrogen sulfide and moisture are present. Some contractors assume that if a Monel orifice plate is specified, the differential pressure transmitter should have Monel internals. Unfortunately, Monel is subject to hydrogen permeation. At elevated pressures during normal operation, hydrogen slowly permeates through the Monel and into the capsule. If the transmitter is the capacitance type mentioned above, the hydrogen changes the dielectric constant of the fill, and causes a shift in the calibration. If the transmitter is the force balance type, it may perform satisfactorily during normal operation, but when the transmitter is depressured for a plant shutdown or for maintenance, the hydrogen will not be able to permeate from the capsule to the atmosphere quickly enough, and the capsule will blow up. In hydrogen-rich services such as Platforming and Hydrotreating, Monel diaphragm capsules are not only unnecessary, they are undesirable.

Pressure

Pressure is very often an "independent" variable, in that the controller set point is usually maintained at a constant value, and is not dependent on other variables. The pressure of a reactor products separator rarely needs adjustment during normal operation. The pressure of a fractionator is held constant so the temperature can be related to composition. The pressures that are adjusted frequently are usually utility pressures, such as fuel gas or fuel oil to a heater.

The primary elements used in pressure transmitters are Bourdon tubes, the same as in pressure gauges; or spiral or helical elements, which are similar to Bourdon tubes; or diaphragm capsules, similar to the differential pressure transmitters described above. For low pressure measurements, a bellows element is sometimes used. If the low pressure side of a differential pressure transmitter is vented to the atmosphere, it will measure a gauge pressure. If the low pressure side is evacuated and sealed to maintain a zero pressure reference, the transmitter can be used to measure an absolute pressure.

Level

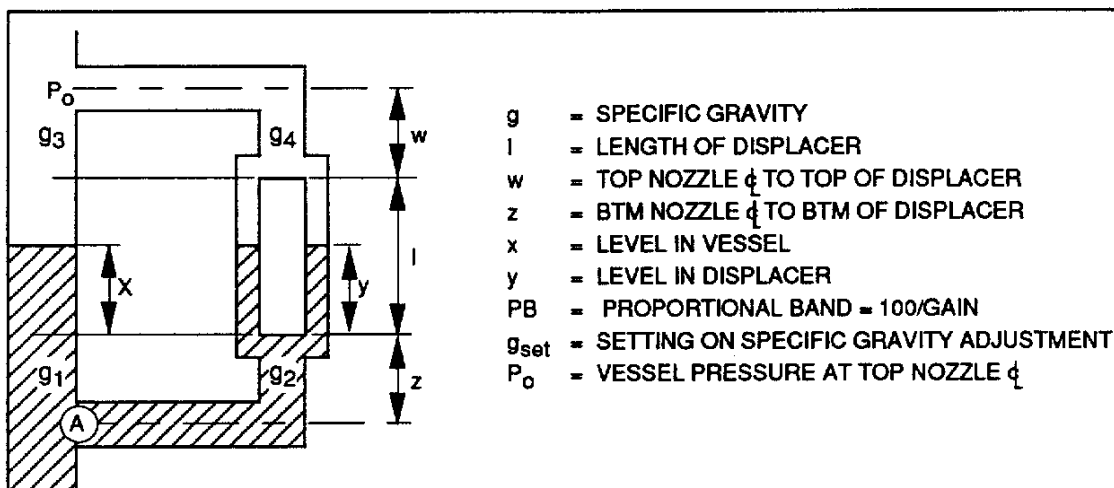
Level is almost always an "independent" variable. Like the pressure controller, once the set point is adjusted, it is rarely changed. At UOP, most level transmitters are the relatively simple, external displacement type. This is an external chamber containing a cylindrical displacer which is often called a "float". Actually, the "float" does not float; the displacer is heavier than the fluid in the chamber. The transmitter has a torque tube and torque arm, which in effect form a spring scale that weighs the displacer. When the chamber is empty, the scale registers the weight of the displacer itself. When the chamber is full to the top of the displacer, the scale registers the weight of the displacer minus the weight of the fluid displaced, in accordance with Archimedes' principle of buoyancy. Because the displacer is cylindrical, the effective weight of the displacer is a linear function of the liquid height on it.

For a displacer in single phase hydrocarbon service, the difference in effective weight

over the range of the instrument is the difference between the weight of the displacer alone at 0%, and the weight of the displacer minus the weight of the hydrocarbon displaced at 100%. The weight of the displacer cancels out, and the difference over the range of the instrument is simply the weight of the hydrocarbon displaced. For a displacer in hydrocarbon/water interface service, the effective weight with the interface at 0% is the weight of the displacer minus the weight of hydrocarbon displaced, and the effective weight at 100% is the weight of the displacer minus the weight of water displaced. Again the weight of the displacer cancels out, but for this service the difference in effective weight over the range of the instrument is the weight of water displaced minus the weight of hydrocarbon displaced. In this case, the instrument is operating on the specific gravity difference between the two phases. (Actually, this is also true for the ordinary level instrument, but the specific gravity of the gas phase is effectively zero, so it is neglected.) If the hydrocarbon phase is too heavy, the measurement could be difficult. For a standard displacement level instrument in interface service, the lower limit of the difference in specific gravities is about 0.2.

The standard displacement type level transmitter has the advantage of a simple zero check: any time the liquid level is below the displacer, the transmitter should show 0% output; the zero setting is not affected by temperature or composition changes which might change the specific gravity. The 100% setting of the standard level instrument is affected by these changes, as are both the 0% and 100% settings of the interface level instrument.

If the vessel temperature is fairly high, the temperature of the fluid in the level instrument may be lower than that in the vessel, and it is possible that the interface in the vessel and the interface in the displacer chamber may be at different levels. This error can be eliminated by using an internal displacement instrument, in which the displacer is located inside the vessel itself, so there is no temperature difference. Unfortunately, maintenance of the instrument then becomes more difficult. The mathematical relationships between internal and external levels for an external displacement instrument



- 1). AT POINT (A), STATIC HEAD IN FLOAT CAGE EQUALS STATIC HEAD IN VESSEL.
- 2). P_o APPEARS IN BOTH STATIC HEAD EQUATIONS, THEREFORE CANCELS.
- 3). DENSITY OF WATER APPEARS WITH SPECIFIC GRAVITY IN BOTH STATIC HEAD EQUATIONS, THEREFORE ALSO CANCELS.

FLOAT CAGE STATIC HEAD = VESSEL STATIC HEAD

$$g_1 (x + z) + g_3 [w + (l - x)] = g_2 (y + z) + g_4 [w + (l - y)]$$

$$g_1 x + g_1 z + g_3 w + g_3 l - g_3 x = g_2 y + g_2 z + g_4 w + g_4 l - g_4 y$$

$$x (g_1 - g_3) + z (g_1 - g_2) + w (g_3 - g_4) + l (g_3 - g_4) = y (g_2 - g_4)$$

SOLVING FOR y YIELDS BASIC EQUATION:

$$y = \frac{x (g_1 - g_3) + z (g_1 - g_2) + (w + l) (g_3 - g_4)}{(g_2 - g_4)}$$

$$\text{OUTPUT \%} = 100 (y / l) (100 / PB\%) [(g_2 - g_4) / g_{set}] + (\text{OUTPUT \% @ } y = 0)^*$$

*NORMALLY = 0

GENERAL SERVICE:

$g_1 = g_2$ (ASSUMES TEMPERATURES EQUAL — IF FLOAT CAGE & PIPING WELL INSULATED, PROBABLY REASONABLE ASSUMPTION); $g_3 = g_4 \approx 0$ (VAPOR DENSITY GENERALLY NEGLIGIBLE).

FLUSHED FLOAT CAGE:

$g_3 = g_4 \approx 0$ (IF g_3 IS CONDENSIBLE VAPOR, MAY NEED TO USE DRY GAS PURGE AS g_4 TO KEEP FROM DILUTING g_2); g_2 IS FLUSH LIQUID GRAVITY, $g_2 > g_1$.

INTERFACE:

$g_2 = g_1 \times$ HEAVY LIQUID EXPANSION FACTOR, $g_4 = g_3 \times$ LIGHT LIQUID EXPANSION FACTOR, WHERE EXPANSION FACTOR = (g @ FLOAT CAGE TEMPERATURE / g @ VESSEL TEMPERATURE).

NOTE:

TOP SIDE FLOAT CAGE CONNECTION: $w = 0$; BOTTOM SIDE FLOAT CAGE CONNECTION: $z = 0$.

Figure 19 - Displacement Type Level Transmitter Equations

are summarized in Figure 19.

Displacement type level instruments are furnished in lengths up to 10 feet. If the same torque tube and torque arm are used for all ranges, the displacer diameter must decrease as the length increases. At the longer lengths, the displacer becomes so long and slender that extra care is required to keep it from being bent during installation and maintenance. Also, the allowable out-of-vertical tolerance decreases, making installation more difficult, and in addition the piping stresses on the chamber due to temperature changes become more of a problem. For level ranges above about 8 feet, a differential pressure instrument, like that used for flow measurement, is often used. The transmitter is installed as shown in Figure 20. Generally, the reference leg is filled with the same material as that inside the vessel. The transmitter measures a differential pressure of zero when the vessel is full, and a negative differential when the vessel is empty. Some transmitter designs require special range suppression

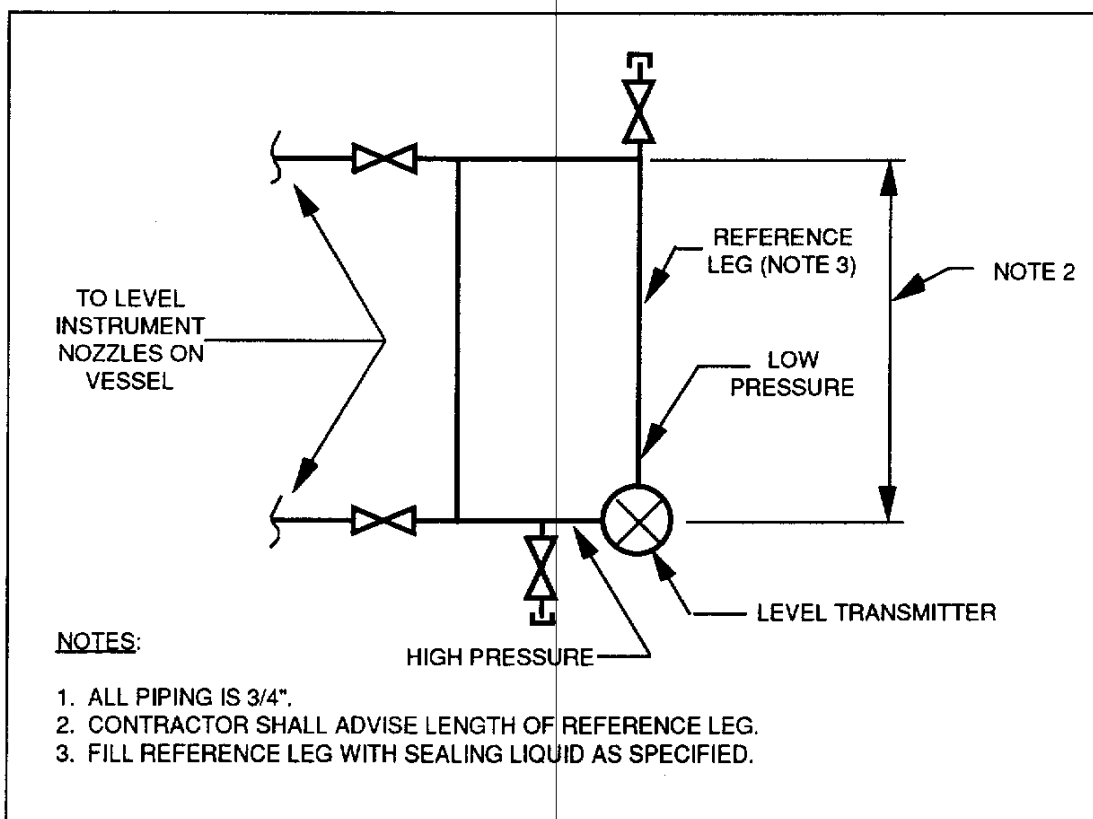


Figure 20 - Differential Pressure Level Transmitter

equipment to accomplish this. The main disadvantage of this type of installation is that if some of the material leaks out of the reference leg, the transmitter indicates a higher level than actually exists. If the transmitter is installed at the bottom of a fractionation column, vapors will condense in the lead piping and keep the reference leg filled; but temperature changes in the reference leg can still cause some error in the indicated level.

When the process material is heavy and has a high viscosity, UOP practice is to heat trace and insulate the piping to maintain the viscosity below 5000 SSU. If the temperature required is more than 150 °F (66 °C), heat tracing the transmitter body could lead to overheating and damage, especially for electronic instruments. For these cases, a transmitter with remote diaphragm seals is used. The remote seals are connected to the transmitter body by capillaries, and the entire system is filled with a silicone oil suitable for high temperatures. The piping to the remote seal can be traced as required. The transmitter body and capillaries are heat traced to maintain the silicone fill at a constant 150 °F (66 °C). This is done to avoid static head variations and the resulting zero errors that would be caused by ambient temperature changes.

For particulate solids, and for difficult liquids such as hydrofluoric acid, a radioactive source and detector system may be used to detect the presence or absence of a level in a vessel by measuring the absorption of a beam of radiation. These are mostly point level measurements, which operate a switch that indicates whether the level is present or not. Proportional measurements, which are actual analog measurements, require either multiple sources or detectors. They are used rarely at UOP; at present the only application of this type is the Surge Hopper level measurement in Continuous Catalyst Regeneration units.

One additional device, which can be used for point or proportional measurement, is the capacitance probe. This is a probe which, with the walls of the vessel, forms a capacitor that uses the fluid in the vessel as the dielectric. Changes in the level

therefore change the capacitance, and the change in capacitance is measured by the associated electronic circuitry. If the probe is installed horizontally, the capacitance change occurs as the level moves over the diameter of the probe, yielding a point measurement; if the probe is installed vertically, the capacitance change occurs as the level moves over the length of the probe, yielding a proportional measurement.

Analysis

Process analyzers are usually open loop devices which depend on the operator to take corrective action if required. The action required may be too complex, or the response time may be too long, to effectively close the loop and let the analyzer act directly on the process. There are many types of analyzers. At UOP the more common types used are moisture analyzers, oxygen analyzers, and chromatographs. Moisture analyzers are used for Platforming unit recycle gas measurements and as monitors for dryer performance. These analyzers operate on two main principles.

The DuPont moisture analyzer (Figure 21) uses a crystal with an absorbent coating. For one part of the operating cycle the crystal is in contact with the moist sample stream, and for the other part of the cycle the crystal is in contact with a dried sample stream which is used as a reference. The moisture in the sample stream is absorbed in the coating on the crystal, changes its mass, and thereby alters its frequency of vibration. The difference in vibrational frequency between the moist state of the crystal and its dried reference state is used as a measure of the moisture content of the sample stream. The Panametrics analyzer (Figure 22) uses a probe which has a thin aluminum oxide coating covered by a gold flashed film to form a capacitor. The oxide adsorbs moisture in proportion to the partial pressure of the water in the sample, and changes the capacitance of the probe. For Platforming recycle gas service, where the moisture content is less than 10 ppm, a moisture generator is specified to provide a sample of known moisture content for calibration of the analyzer. On the newer DuPont analyzers, a moisture generator is built into the analyzer.

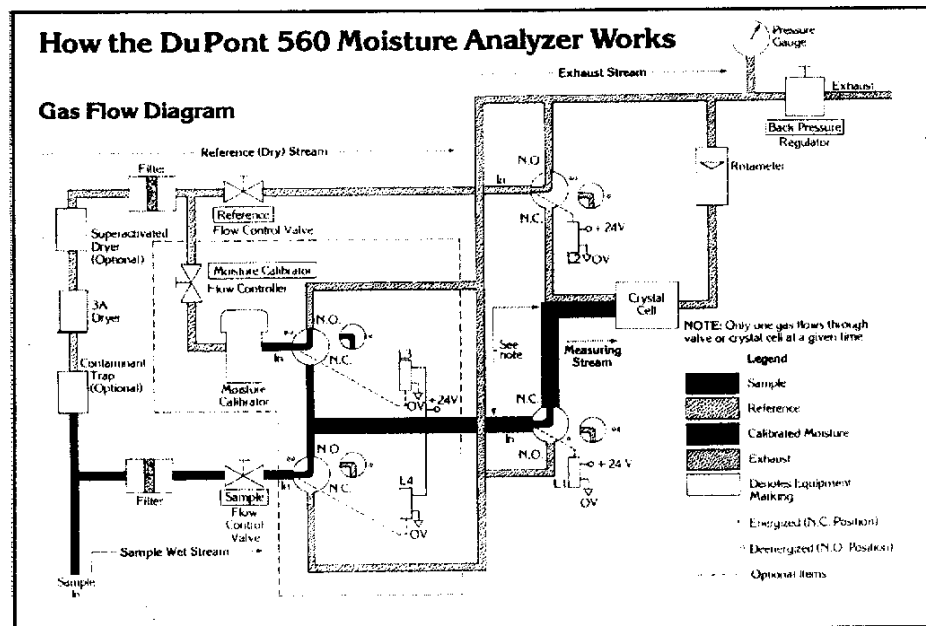


Figure 21 - DuPont Moisture Analyzer

Sampling systems for these low ppm ranges are critical if a reliable reading is to be obtained. The sample tubing to the analyzer must be stainless steel. At low moisture concentrations, the moisture which is adsorbed on the walls can be significant in relation to the moisture in the sample stream. Changes in concentration in the sample stream then encounter a lag, as time is required for the walls to reach a new equilibrium value. Copper tubing has walls which are rough and porous; this adds to the

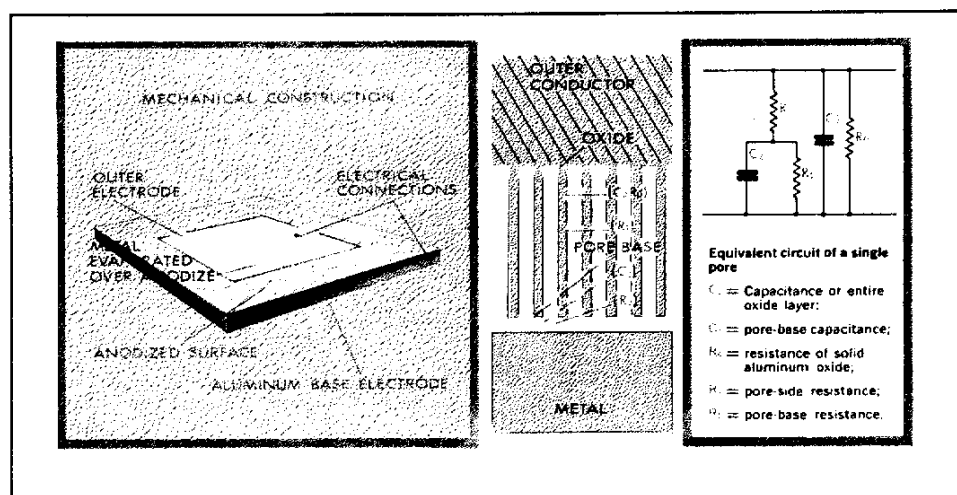


Figure 22 - Panametrics Moisture Analyzer

surface area and increases the adsorption, so stainless steel tubing must be used. Likewise, the longer the sample line, the greater the surface area, with the same detrimental effect on the response, so the sample line must be as short as possible. At UOP, the maximum length allowed is six feet, and is so specified on the Piping and Instrument Diagram. In addition, the sample line should be thoroughly heat traced. Keeping the wall of the tubing hotter than ambient temperatures reduces the adsorption on the walls and keeps the moisture in the flowing stream.

Most analyzers operate on a sample flow so small that even the volume of six feet of 1/4-inch tubing is significant in terms of response. Usually, the sample rate through the tubing is increased by simply bypassing some of the sample around the analyzer. A few analyzers are able to do their analysis in situ, directly in the flowing stream, and

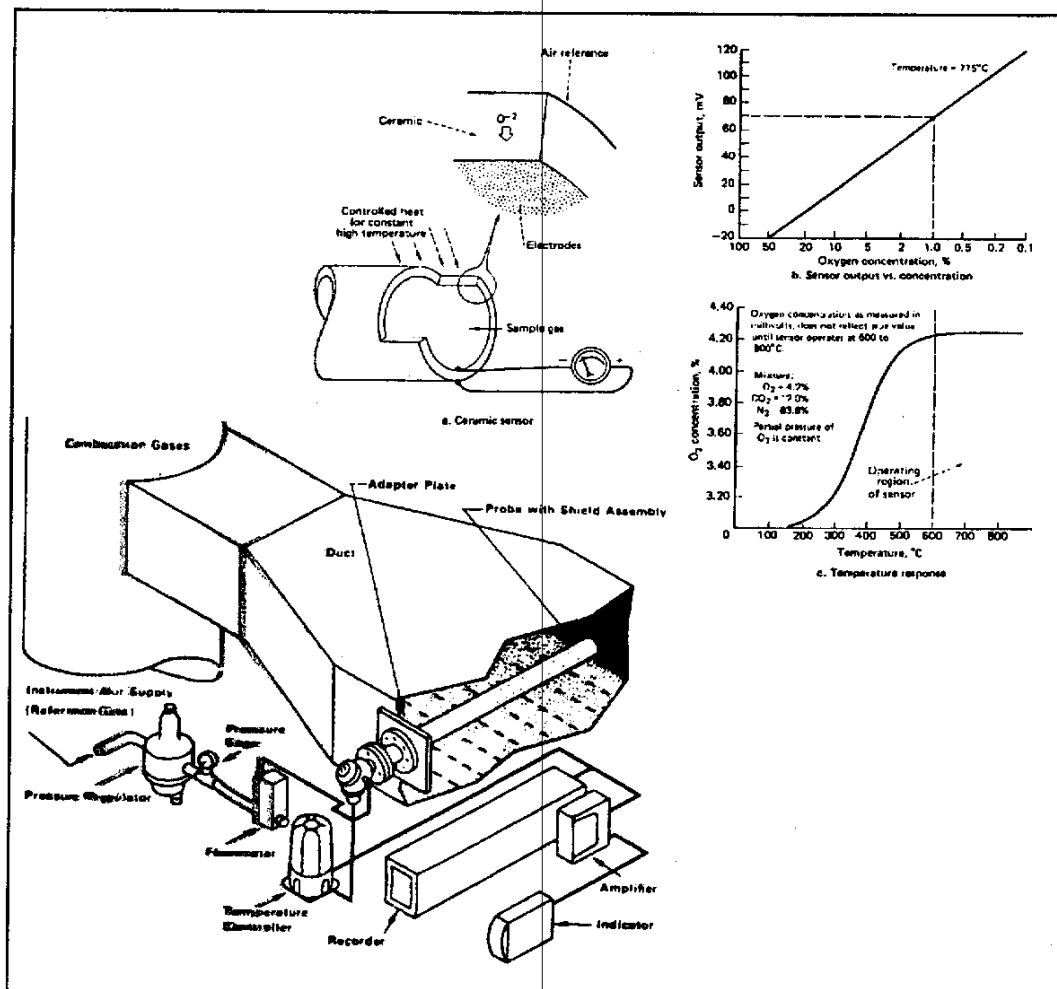


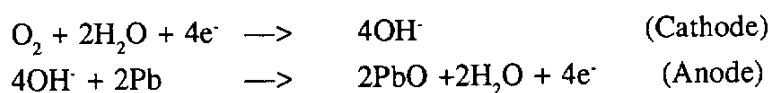
Figure 23- Zirconium Oxide Oxygen Analyzer (Westinghouse shown)

thereby avoid many of the usual sampling problems. One such analyzer is the Westinghouse oxygen analyzer (Figure 23). This analyzer has a sensor which is made from a zirconium oxide ceramic coated with platinum. Instrument air is used as a reference on one side of the ceramic, and the sample gas is on the opposite side. At the operating temperature of about 1500 °F (815 °C), a voltage is developed between the two sides of the ceramic in accordance with the Nernst equation:

$$E = (RT/4F) \ln (P_R / P_S) + C \quad \text{Eq.(1)}$$

where R is the gas constant, T is the absolute temperature, F is Faraday's constant, and P_R and P_S are, respectively, partial pressures of oxygen in the reference gas and the sample. The relatively high operating temperature allows insertion of the sensor directly into the stack of a heater above the convection section. For sampling in the hotter region below the convection section, an extraction type version of the analyzer is available, using an eductor to withdraw the sample from the heater. The zirconium oxide oxygen analyzer has become very popular, and is available from a number of manufacturers. Stack or inert gas analysis is the main application for this type of oxygen analyzer. Since the sensor operates at a high temperature, combustibles in the sample will be burned, consuming part or all of the oxygen in the sample, and giving an incorrect result.

UOP specifies a portable oxygen analyzer for checking during catalyst regeneration. This analyzer may be a Beckman polarographic analyzer, which measures the current from an oxidation-reduction reaction that takes place between the oxygen and a silver anode in a KCl solution. Alternatively, a Teledyne Micro-Fuel Cell type may be used. The fuel cell produces an electrical current proportional to the oxygen concentration in the sample by means of the following chemical reactions:



One other oxygen analyzer, the Hays paramagnetic analyzer, which was used on Plat-

forming CCR units, is no longer available. A Thermox extraction type zirconium oxide analyzer, similar to the Westinghouse analyzer described above, is now specified.

Another type of analyzer that is frequently used, especially in aromatics complexes, is the chromatograph. The chromatograph operates on the differences in rates of adsorption and desorption of different molecular species. It gets its name from paper chromatography, which has been used to analyze multicomponent dyes and separate them into their component colors (Figure 24).

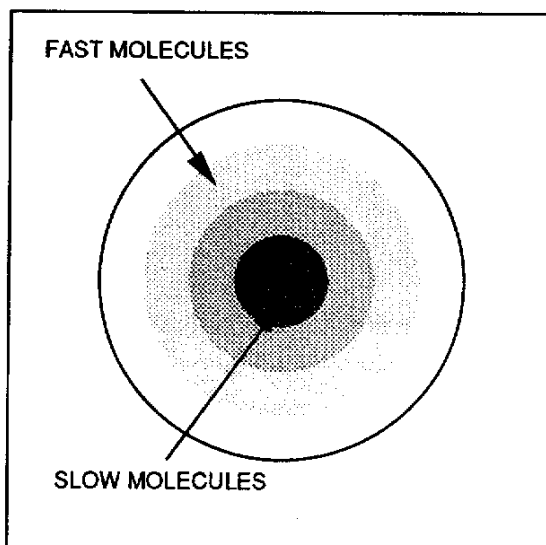


Figure 24 - Paper Chromatography

The branch of chromatography most useful for refinery process analysis is gas-liquid chromatography. This involves a long, thin packed column into which a sample is injected and then is flushed through by a carrier gas (Figure 25). The packing is coated with a liquid which absorbs and then desorbs the sample components. The chromatograph can discriminate between some components in ranges well below 1000 ppm,

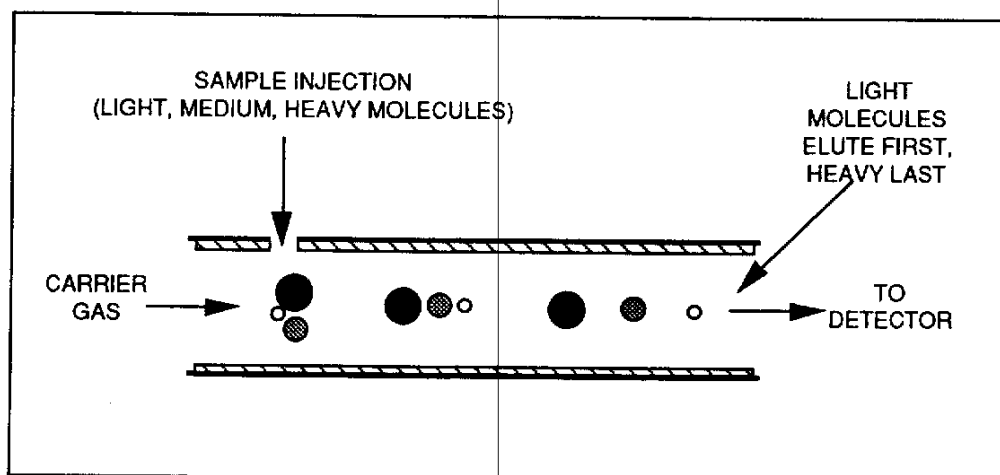


Figure 25 - Gas-Liquid Chromatography

which would require temperature measurements accurate to 0.1 °F or better for equivalent performance. However, the complexity required for that discrimination is costly, and the use of the chromatograph must be carefully considered.

Process chromatographs - and indeed, any process analyzer - should not be considered a substitute for the laboratory. The process chromatograph must be built to handle process conditions, and that generally means that something must be sacrificed in comparison to an equivalent laboratory instrument. Also, while the lab may take 30 minutes of closely attended operation to run a certain analysis, the process instrument generally must be able to complete the analysis - unattended - in 10 minutes or less. Again, something must be sacrificed. The primary consideration is, as it was for flow measurement, reproducibility rather than accuracy. If the plant operation is stable, we can easily relate the laboratory results to the analyzer results; as long as that relationship is constant and reproducible, we can operate almost as well as if the analyzer had the same accuracy as the lab.

The analyzer should not be installed to replace routine samples, but to reduce the need for special samples. Its main value is in an upset. The analyzer can run a sample in a few minutes and indicate where operations are heading so a correction can be made, while a lab sample might take one to two hours from the sample to the result. The justification, therefore, revolves mainly around the probability of an upset occurring, and what that upset means in terms of product quality. If upsets are unlikely and if the off-spec product can easily be blended off or otherwise salvaged, the analyzer is probably not justifiable.

Other types of analyzers that are used include hydrogen analyzers, sometimes used for recycle gas, which compare the thermal conductivity of the sample gas with that of a known reference gas; carbon monoxide and carbon dioxide analyzers for FCC flue gas, which relate the absorption of infrared light in the sample gas to that of a known reference; SO_x and NO_x analyzers which measure stack gas pollutants by means of

chemiluminescence; boiling point analyzers; H_2S analyzers; pH analyzers; conductivity analyzers; and many others.

Almost all of these analyzers are expensive - in terms of maintenance, or initial cost, or both. Analyzers that use wet chemical methods are especially maintenance intensive.

FINAL CONTROL ELEMENT

The final control element used most often in the process industries is the control valve. Occasionally, the controller output signal may adjust the stroke of a pump or the speed of a compressor turbine. And, for a time, energy considerations prompted many stud-

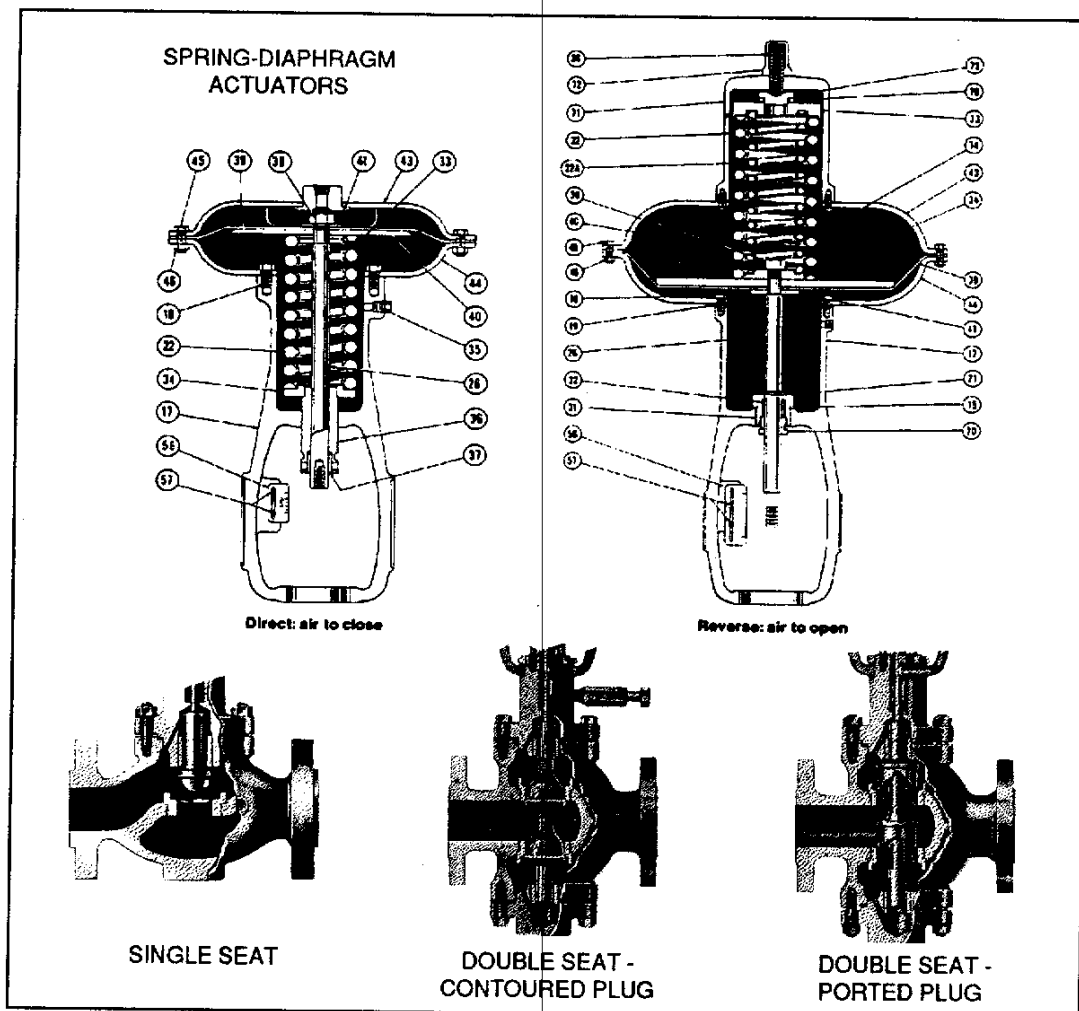


Figure 26 - Globe Type Control Valves

ies into variable-speed electric motor drives and other energy saving devices (the control valve cannot control unless it is given some energy to dissipate). But at present, economics still generally favor the control valve.

The most common type of control valve is the sliding-stem globe type valve with a pneumatic spring-diaphragm actuator (Figure 26). The globe valve is one of the oldest designs. It is so named because the early valves, in particular, were cast with a globular shape to the main portion of the body. Globe valves are available in either single seat or double seat construction, and in a variety of materials. Because globe valves are the most common design and are generally available almost anywhere in the world, these are the valves that are specified most often for UOP process units.

UOP standards for globe type control valves call for top and bottom guiding or heavy duty top guiding. Some early control valve designs used a hollow cylindrical, ported plug, which served as both throttling device and guiding surface, a practice known as "skirt" guiding. To keep the plug in proper alignment, clearances between the guiding surfaces of the plug and seat had to be small. The tight clearances, combined with the location of the guiding surfaces directly in the flowing stream, led to problems with sticking and poor control. Differential expansion due to temperature changes, and fouling or erosion from dirty services caused the guiding surfaces to bind. Using guiding surfaces separate from the flow control surfaces and remote from the flowing stream eliminated the sticking problems. In the early 1970's, control valve manufacturers (notably Fisher Controls) began promoting a "cage" valve design (Figure 27), in which the valve seat ring is held in place by a cage which contains the plug. The cage in turn is clamped in to the body by the valve bonnet. The design permits quick disassembly and change

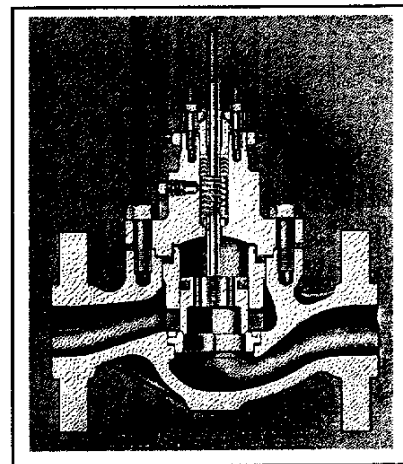


Figure 27 - Cage Type Globe Valve

of the valve trim, but has the same problems as the early skirt guided design.

The design can have some advantages when control valve noise is a problem. Most low noise control valve designs involve a hollow plug or cage which allows the fluid to flow through passages in the wall and impinge on the fluid entering from the opposite side. In this manner, the energy is dissipated in the fluid itself, and not on the valve surfaces, so erosion of the trim and body is reduced, and also the noise level is decreased. This concept fits well with the cage design, but because the passages must be small and torturous to gain the maximum noise reduction, even more than the ordinary cage design, such valves require a clean fluid to operate properly. UOP's preference is to utilize low-db plates or cartridges, if possible, to achieve the noise reduction, rather than cage-guided control valves.

When cage design control valves are proposed for a project, each service should be examined carefully on a case-by-case basis, to evaluate the possibility of problems from dirt or temperature. Fisher does manufacture a valve with a cage design in which the cage incorporates guiding similar to a heavy top guided conventional plug. This Design EZ valve (Figure 28), does not meet UOP standards because the guiding is not heavy enough, but some UOP customers have used it successfully in HF Alkylation services (valves for alkylation services will be discussed later).

Cage type valves are available only in single seat design. Likewise, valves with reduced trim smaller than 3/4-inch are available only in single seat design. Single seated valves are used for applications that require tight shutoff, because it is difficult to shut both seats of a double seated valve simultaneously. Although it is possible to lap the seats of a double seated valve to reduce leakage, if

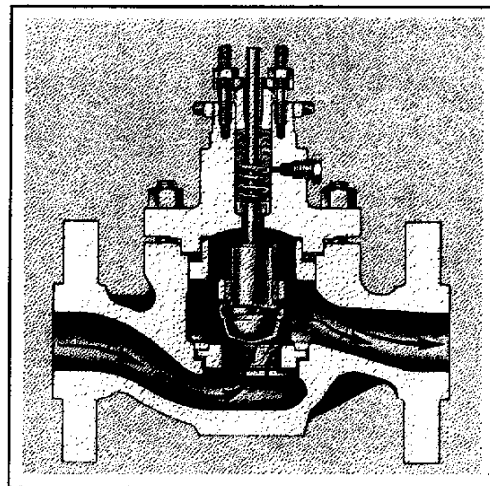


Figure 28 - Fisher Design "EZ"
Globe Valve

the temperature changes, the difference in expansion between the body and the trim will reduce the seating force on one port or the other, and the valve will leak again.

The main advantage of the double seated valve is that the actuator size can be smaller. With a double seated valve, the valve is usually installed so the flow enters between the two seats, and exits upward, flow-to-open, through the upper port, and downward, flow-to-close, through the lower port. The net unbalanced forces due to pressure drop on the area of the plug are therefore relatively small. The forces do not quite cancel each other, because one port must be large enough to allow the other plug to pass through. Also, the hydrodynamic forces do not quite cancel, because the flow directions around the plugs are opposite. But the overall result is that the actuator size can be reduced.

UOP standards require an actuator with a spring that has twice the thrust requirement calculated on the basis that when the valve is open, the pressure drop is as specified, and when the valve is closed, the pressure drop is equal to the full upstream pressure. A suitable actuator with a 3 to 15 psi spring, or an integral multiple of 3 to 15 psi, is then chosen. It is unlikely that most valves will ever be called upon to close against full upstream pressure, but sizing the actuator in this manner provides a thrust at normal conditions which is large in comparison with the plug unbalance forces, and increases the ability to operate at conditions such as reduced throughput, as well as at normal operation.

The 3 to 15 psi spring range specified is the "bench set" range, that is, the range as calibrated on the work bench, with no differential pressure across the plug. It should be noted that when the valve is installed and operating, the range will most likely not be 3 to 15 psi because the valve differential pressure will not be zero. Control valve manufacturers sometimes use non-standard ranges such as 7 to 15 or 6 to 13 psi, to get sufficient closing thrust with an undersized actuator. The non-standard setting is generally intended to give a 3 to 15 psi range under operating conditions. However, if

the actual differential pressure is much different from the design value, the valve stroke range may be quite different from 3 to 15 psi, and it is even possible that the actuator may not be able to fully close—or possibly not fully open—the control valve.

While it is convenient for the operator to have the valve stroke exactly 0 to 100% for a 0 to 100% input signal, as long as the relationship is reasonably close, it is not a critical requirement. Valve position can be classified as a "dependent" variable; during normal operation the valve position will be set as required to control the flow, pressure, or other process variable at the required value. It does not matter to the controller if 60% of full scale output is required to obtain a 50% open valve position. Again, as mentioned earlier, reproducibility is more important.

The convenience lies mostly in knowing what control input will just open the valve when the loop is being commissioned. But even if the opening point is correct for one set of operating conditions, changing the operating conditions will change the opening point. The only way to insure that the valve stroke will match the signal input is to install a positioner.

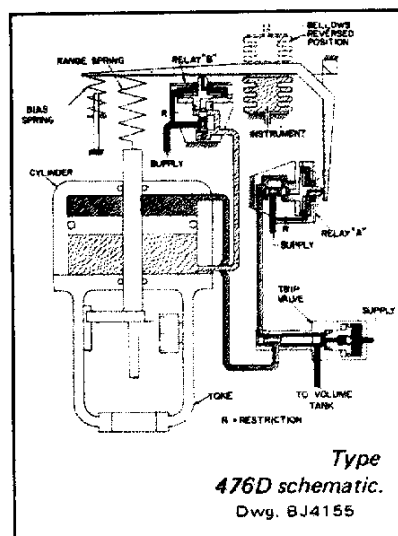
Unfortunately, positioners are not recommended for flow control loops, which make up the majority of control applications. The response of a flow loop, especially with electronic transmission, is quite rapid. The response of the positioner is also very fast. If the two responses are too similar, under certain conditions they can reinforce each other to the extent that the loop becomes unstable.

Some positioners can be used in flow loops. Piston actuators, which are used to meet very high thrust requirements, must have positioners because they have no springs to push against. Instead of springs, they use a regulated air loading pressure beneath the piston. When the positioner output pressure exceeds the loading pressure, the piston moves down; when the positioner output pressure is less than the loading pressure, the piston moves up; when the pressures are equal, the piston does not move. (The unbalance forces on the valve plug are neglected for this discussion.)

The thrust advantage of the piston actuator comes from two sources. The rigid construction of the piston permits the use of higher air pressure than is possible with a diaphragm, generally up to full supply header pressure. Secondly, the loading pressure is regulated; as the piston moves down and compresses the loading air, the loading regulator bleeds off the excess to maintain the controlled pressure. The opposing force therefore does not increase in proportion to the stroke, as it does with the spring diaphragm actuator, and therefore thrust which would be used to compress the spring is available for counteracting plug unbalance forces.

The description above applies to Masoneilan (Annin) Domotor actuators. Fisher actuators (Figure 29) utilize a double-ended positioner which simultaneously increases the pressure on one side of the piston, and decreases the pressure on the other. The positioners used with piston actuators are special designs; standard positioners normally used with spring diaphragm actuators cannot be used with piston actuators.

Another class of actuators should be mentioned in this regard. Masoneilan has been



TYPE 476D — A schematic of the Type 476D is shown.

With the addition of the trip valve, the loading pressure from relay "A" to the cylinder above the piston flows through the trip valve. As long as the supply pressure is normal, the trip valve allows normal automatic operation. When the supply pressure falls below 60% or 75% of the maximum supply pressure, the force of the volume tank pressure acting on the trip valve piston overcomes the force of the reduced supply pressure acting on the other side of the piston. The trip valve inner valve is forced to the right, shutting off the relay output and allowing the volume tank pressure to flow to the top side of the cylinder. The piston is forced to the bottom of its stroke and remains there until the supply pressure is restored.

When the supply pressure is first introduced into the trip valve, a small check valve in the piston opens and allows supply pressure to flow into the volume tank until the volume tank pressure is approximately equal to the supply pressure. The check valve closes when the volume tank is pressurized.

Figure 29 - Pneumatic Piston Actuator (Fisher shown)

promoting an actuator design for globe valves called the " Σ F" actuator (Figure 30). This actuator uses a horizontally mounted spring and diaphragm, with a cam and lever linkage to convert the horizontal motion to a vertical thrust. The translation from horizontal motion to rotary to vertical is not quite linear, and as a result the valve ends up with a non-standard characteristic. (Note: valve characteristics will be discussed later.) Instead of offering plug designs which yield the desired characteristics, Masoneilan initially used a single plug design and different characterized cams in a positioner. So to obtain a certain characteristic, a positioner had to be used, even on flow control loops. Masoneilan now does offer characterized plugs for this valve series. But if a prospective supplier offers only characterized cams to obtain a desired characteristic, it should be made clear before purchasing that if any problems result, the supplier must be willing to assist in their solution.

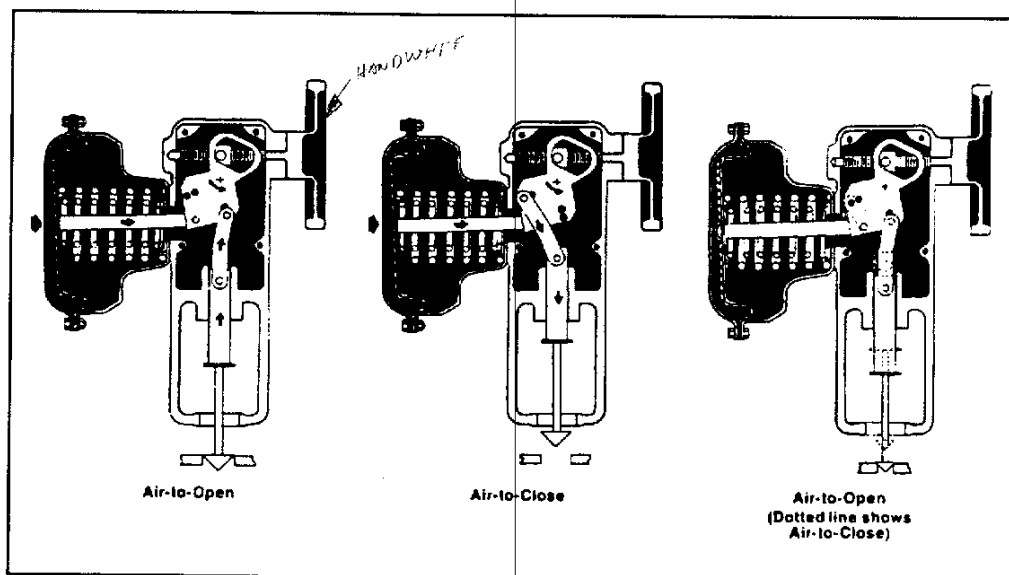


Figure 30 - Masonellian Σ F Actuator

In addition to single and double seat designs, another type of globe valve is the angle valve (Figure 31). Angle valves are used in slurry service in FCC units, and in Vacuum Column bottoms service in Vacuum Distillation units. Installed with the flow tending to close the plug, the angle valve is self cleaning; catalyst and coke particles pass through without causing the clogging and body erosion which would occur if a

71000 Series

Critical Flow Factor – C_f

Trun Size	Flow to Close	Flow to Open
full area	.48	.90
5 factor	.55	.95



10 EXPENSE
20 ALL SILENT
30 CORRUPT
40 ONE REVER
50 CORRUPTION
60 DE NO POSITION
70 FURTHER CAUTION

300-1500 lb. ANSI (flow to close)

Nominal Trim Size	Full Area	1	1½	2	3	4	5					
	.5 Factor	1	1½	2	3	4	5					
On-Rise Dia. (in.)		.500	.750	.812	1.125	1.562	1.687	2.250	2.375	3.187	3.375	4.750

[illegible]

*JDO-600 (b. ANSI) only

Refer to Bulletin 345E

300-1500 lb. ANSI (How to open)

Nominal Trim Size	Full Area	1	1½	2	3	4	6				
	.5 Factor	1	1½	2	3	4	6				
Drifts Dia. (in.)	.500	.750	.812	1.125	1.562	1.687	2.250	2.375	3.187	3.375	4.750

[illegible]

*360-600 lb. ANSI only

Refer to Bulletin 244E

Flow Coefficient = C_v
Critical Flow Factor = C_c

43

Figure 31 - Masonellan 71000 Series Angle Valve

standard globe design were used.

Still another globe type valve is the 3-way valve (Figure 32), used primarily for control of flows through and around heat exchangers. These valves can be arranged either for diverting service, in which one stream enters the valve and two streams exit, or

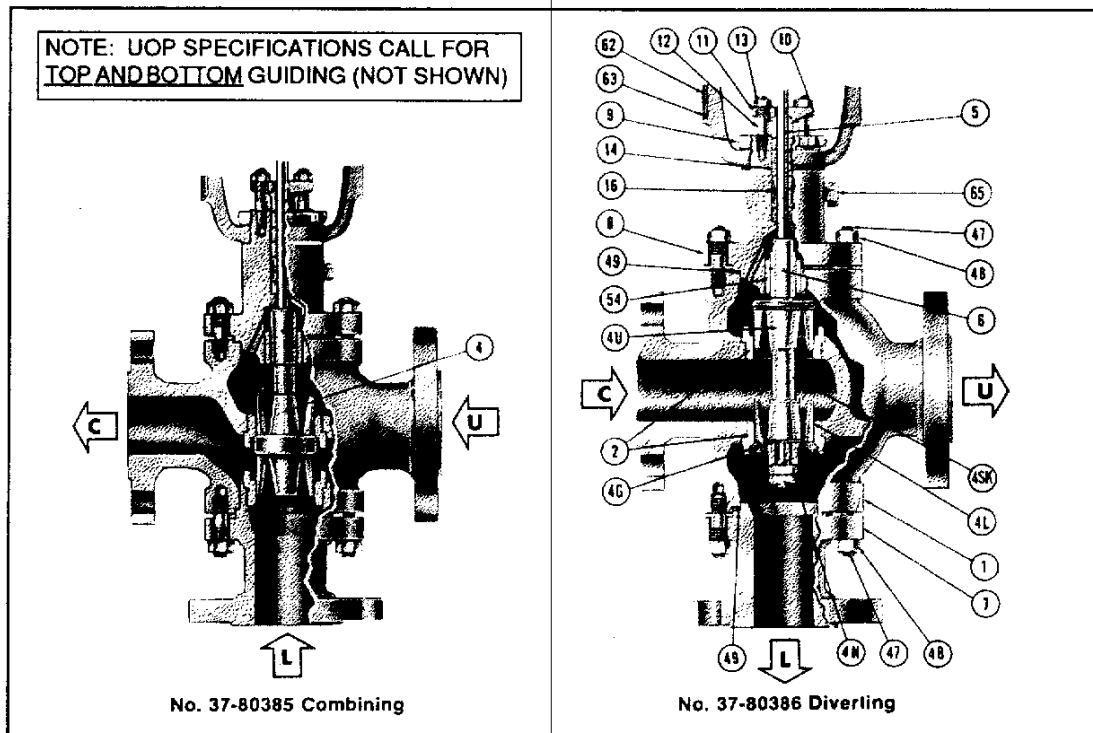


Figure 32 - Three-Way Valves

combining service, in which two streams enter and one stream exits. Either of these functions also could be performed by two separate valves in a split range configuration so one valve is opening when the other valve is closing. The disadvantage of using two separate valves is that it opens up the possibility that a failure might occur which would close both valves simultaneously, and therefore might require the use of a relief valve to protect equipment. If the valves are in reactor effluent service, finding a suitable relief valve could be extremely difficult.

Two other globe type valves have been used extensively on UOP units. One is the double diaphragm fuel gas valve (Figure 33). Instead of a spring-diaphragm actuator, this valve has two diaphragms, one with twice the area of the other. The 3 to 15 psi air signal pushes down on the upper diaphragm and opens the valve. The burner fuel gas pressure downstream of the valve is piped back underneath the lower diaphragm, which has half the area of the upper diaphragm, to oppose the downward force from the air signal. Because of the diaphragm area ratio, the gas pressure must increase to

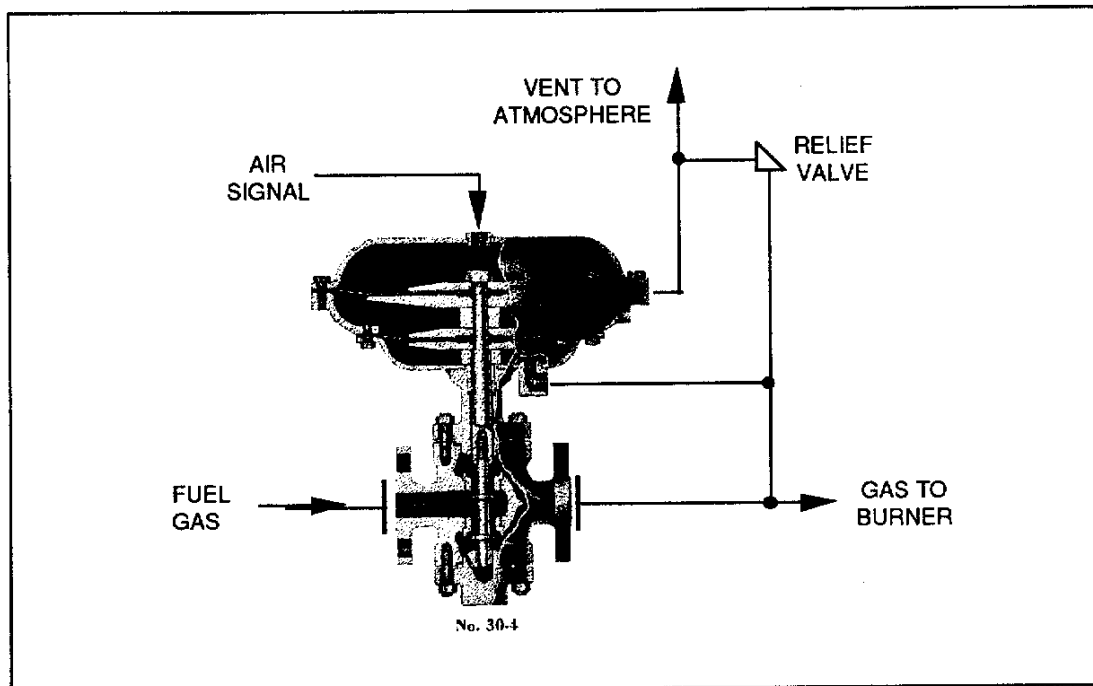
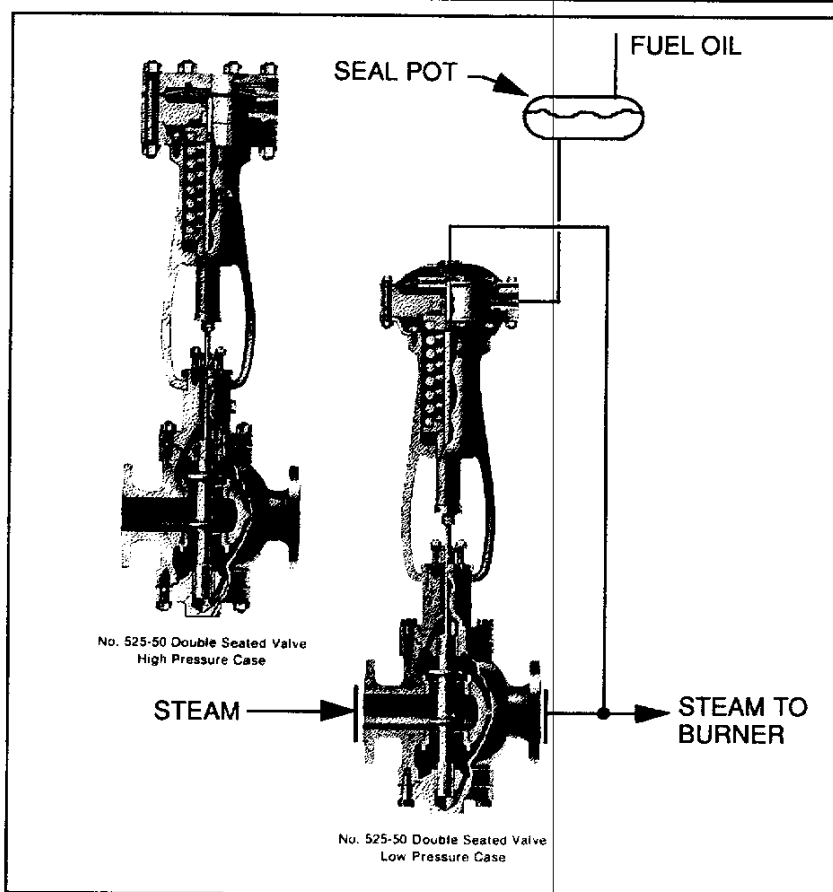


Figure 33 - Double Diaphragm Fuel Gas Valve

two times the air signal pressure before it will balance the downward force and close the valve. Thus a 3 to 15 psi air signal directly controls a 6 to 30 psi fuel gas pressure. Diaphragm ratios of 1 to 1 and 1 to 2 are also available, but are not used as often as the 2 to 1 ratio. The space between the two diaphragms is vented to the atmosphere to prevent any possibility of fuel gas getting into the air supply (or vice versa) if a diaphragm ruptures.

The other type of globe valve is used for control of atomizing steam for fuel oil (Figure 34). This valve has a single diaphragm and spring, but instead of being operated by an air signal, the valve is operated directly by the steam and fuel oil themselves. The steam pressure downstream from the valve is piped above the diaphragm, to oppose the spring and close the valve. Fuel oil pressure is piped underneath the diaphragm via a glycol-water seal. When there is no fuel oil pressure, the steam pressure is set by the opposing spring force alone; as the fuel oil pressure underneath the diaphragm increases, the steam pressure increases by the same amount, and therefore maintains a constant pressure differential between the steam and the oil.



One last specialized globe body design is the split-body valve (Figure 35), which is used for HF Alkylation units. The body of this valve is split horizontally into two halves, and the seat ring is clamped between the two halves to hold it in place.

Figure 34 - Atomizing Steam Regulator Valve

This design makes

it easier to get to all surfaces for cleaning and neutralization of the acid before taking the valve to the shop for maintenance. Standard valves have areas which are difficult to inspect, and the threaded seat ring is very difficult to neutralize properly.

Another fairly common type of control valve in general use is the rotary valve. Instead of having a linear sliding stem, these valves have shafts which rotate to change the position of a vane or plug or ball (Figure 36). The valves in this group include butterfly valves, eccentric plug valves, and ball valves.

Butterfly valves are used by UOP mostly for low pressure drop applications. For a given nominal size, they have more capacity than globe valves, primarily because the flow is straight through the body with essentially no change in direction; also, when the vane is wide open, there is little or no obstruction.

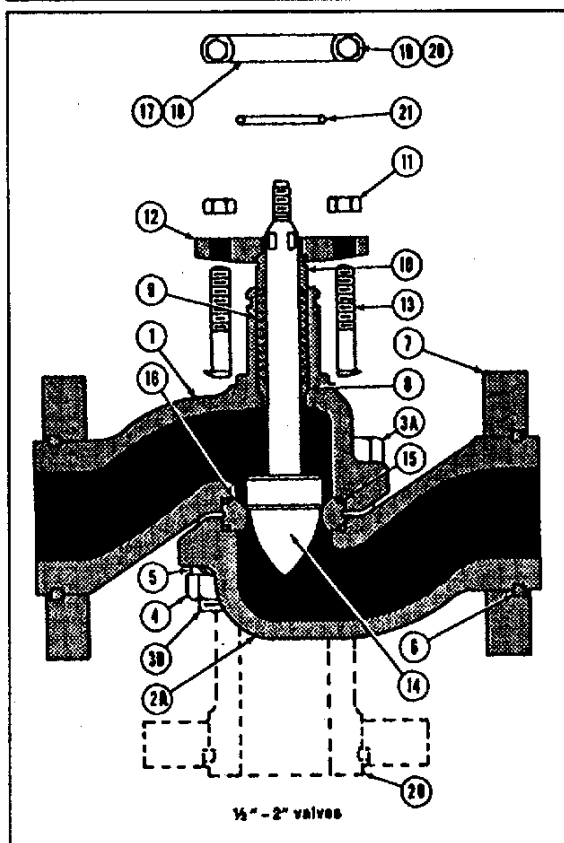


Figure 35 - Split Body Valve

Traditional butterfly valves are generally sized based on their capacity at a 60° vane angle. This is because the capacity at angles beyond 60° is less clearly defined, in part because the shaft of the valve becomes a larger factor in determining the size of the obstruction at these openings. The shaft size is generally what determines the maximum differential pressure for a butterfly valve. It is especially important for butterfly valves in compressor suction lines, where failure of the shaft could result in major damage to the compressor.

Of the valves with eccentric plug designs, probably the best known is the Masoneilan Camflex valve, which has been available since the early 1970's. The Camflex valve is a flangeless design - it fits between two pipe flanges, and does not have flanges of its own. It is a relatively uncomplicated design, but is not available in as many sizes and trim combinations as the standard globe valve. For this reason it can be manufactured and sold more cheaply. Like the early ΣF series valves, it requires a positioner to achieve a standard valve characteristic.

Ball valves at UOP are used for on-off services. The main application is in Continuous Catalyst Regeneration units, for gas-tight shutoff. For control of solids flow in these units - again, on-off, non-throttling - the Fisher Vee-Ball valve is used. The Vee-Ball valve is not a true ball valve; roughly half a hollow ball is used, with a vee-shaped notch at the port of the body to characterize the flow vs. opening relationship. The

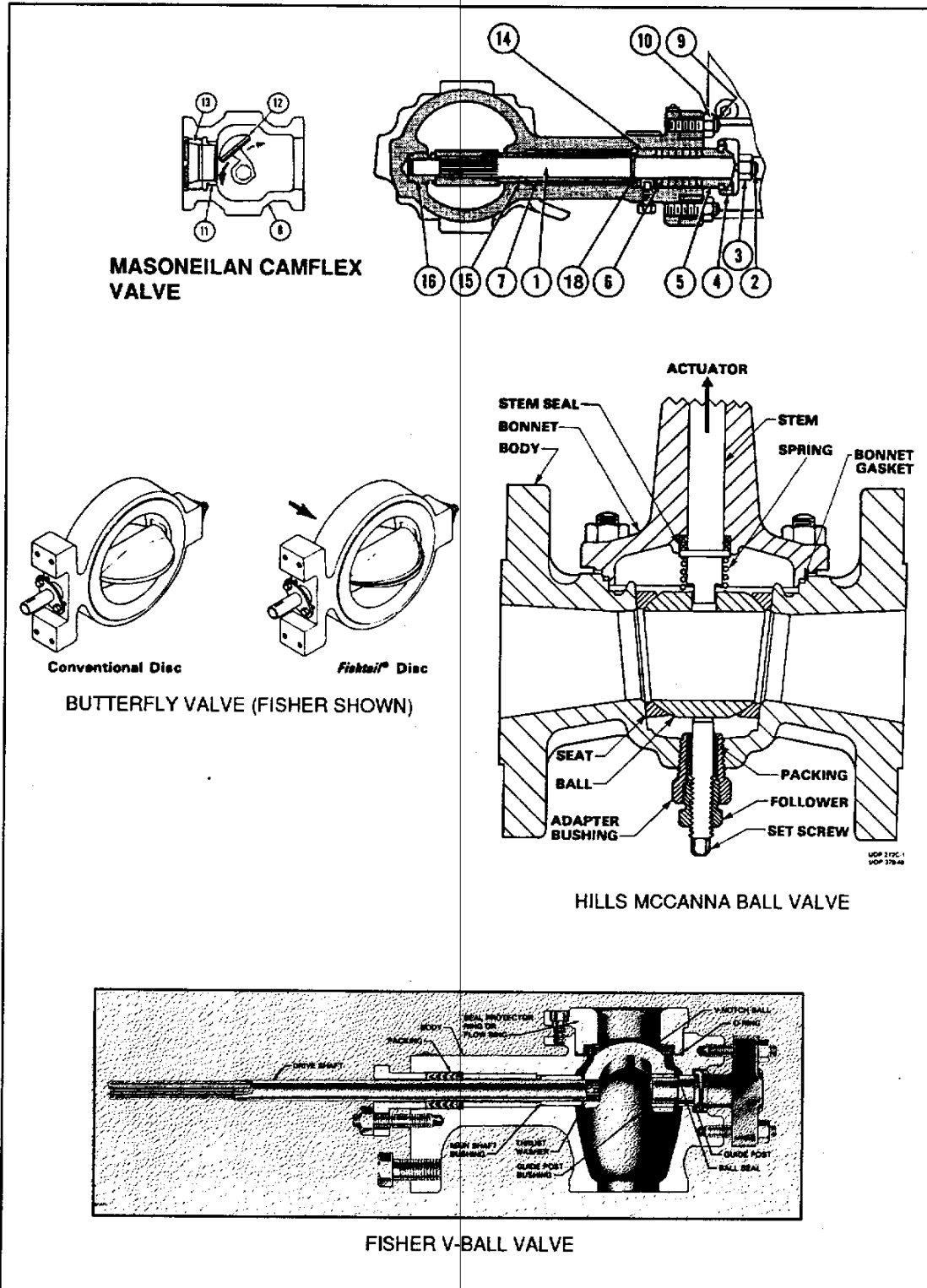


Figure 36 - Rotary Type Control Valves

advantage of the hollow shape in handling solids is that as the valve closes, the material is not trapped in the body of the valve, but falls on through. This minimizes catalyst losses from crushing, and also minimizes valve problems from abrasion by the catalyst. The Vee-Ball valve can also be used for throttling in liquid or gas services.

Regulators (Figure 37) are also a form of control valve. There are many different kinds of regulators. For the most part, their operation is fairly straightforward, and will not be discussed here. However, there is one regulator system that UOP uses which is unusual and which should be explained. This is the storage tank gas blanket assembly (Figure 38).

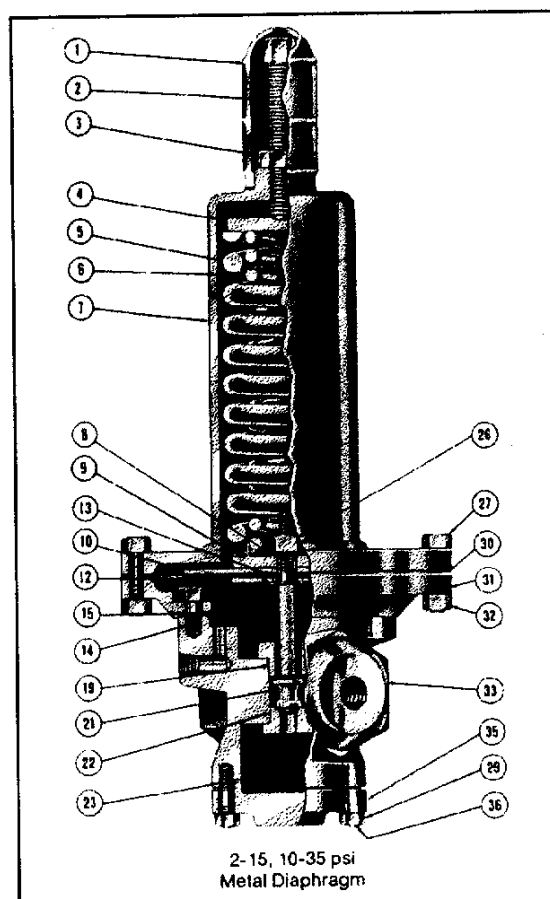


Figure 37 - Masonellian Regulator (Pilot Gas)

The purpose of the gas blanket is to maintain a slight positive pressure on the tank, and thus prevent air from entering. The tank is equipped with a pressure-vacuum relief valve, which opens to vent excess pressure at about 3-inches H_2O . The set point of the gas blanket regulator must be below that pressure, and should be as low as possible. A set point of 1/2-inch of H_2O can be achieved by using a Fisher Y600 regulator, and mounting it upside down, so the weight of the spring does not rest on the diaphragm. However, the Y600 regulator is a small regulator, and does not have the capacity required to blanket a large storage tank. Therefore the Y600 is used as a pilot to operate a larger Fisher 166-5 regulator, which does have the capacity, but does not have the low set point.

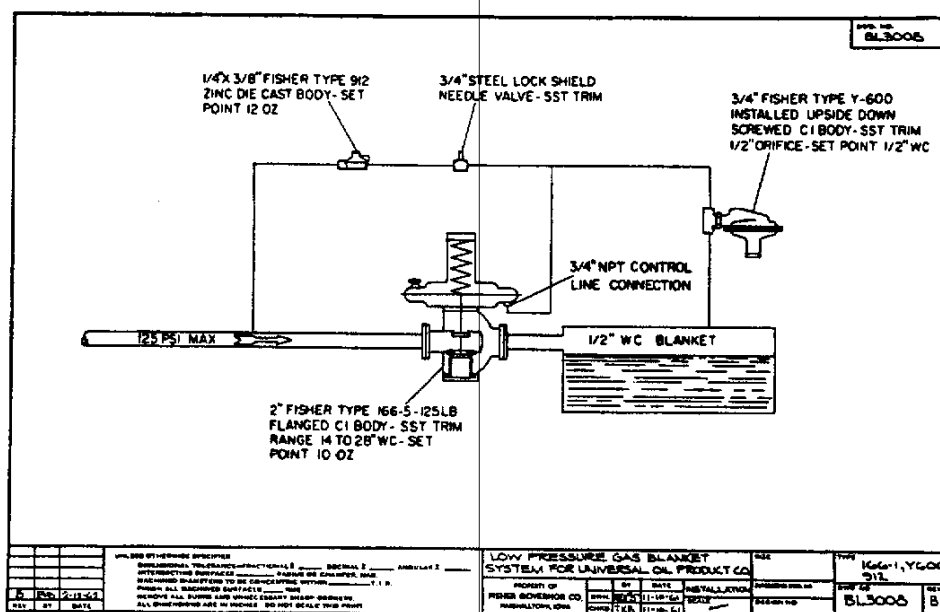


Figure 38 - Tank Gas Blanket Assembly

The set point of the 166-5 regulator is 10 ounces, or 17-inches H_2O . A Fisher Type 912 regulator, set at 12 ounces, or 21-inches H_2O , provides the pilot gas to operate the 166-5 regulator. A needle valve downstream of the Type 912 regulator is adjusted so its capacity is slightly less than that of the Y600 regulator. Therefore when the tank pressure is below 1/2-inch H_2O and the Y600 regulator is open, the pressure sensed at the 166-5 regulator is low and the regulator is wide open, sending blanket gas to the tank. When the pressure in the tank increases to 1/2-inch H_2O , the Y600 regulator closes, the pressure in the sensing line to the 166-5 regulator builds up to 12 ounces, and the regulator closes. Thus the smaller regulator can control a large flow by acting as a pilot for a larger regulator.

It should be noted that this system does not actually control the blanket gas pressure, it only sets the minimum value. If the tank temperature is decreasing, or there is a net flow of liquid out of the tank, the system will add blanket gas to maintain the 1/2-inch H_2O minimum pressure. If the temperature begins to rise or the flow rates change to give a net flow into the tank, the regulator assembly will shut off the flow of blanket gas, but the pressure will rise until it reaches the 3-inch H_2O setting of the pressure-

vacuum relief valve. At other times, the blanket pressure will float between these two extremes.

It should also be noted that the Figure shows a separate connection on the tank for the sensing line. If the sensing connection is on the gas supply line to the tank - as it is when one blanket assembly is shared by multiple tanks - the sizing of the gas supply line downstream of the regulator must be carefully considered. When the gas is flowing, the pressure at the sensing point will be the tank pressure plus the line pressure drop. If the line drop is greater than 1/2-inch H_2O , the system will not control properly. Instead, it will simply oscillate.

If the line is too small and the tank pressure falls below 1/2-inch H_2O , when the system operates and opens the regulators, the gas flow will cause the pressure at the sensing point to increase above 1/2-inch H_2O , and the regulators will close. With the gas flow stopped, the pressure at the sensing point will fall below 1/2-inch H_2O , the regulators will open again, and another cycle will continue. With the low line pressure drop requirement, it is necessary to mount the gas blanket assembly on top of the tank. If the system is installed at grade, it will not function properly.

Control Valve Characteristic

The capacity of a control valve is measured by a valve coefficient called the C_v . The C_v coefficient is used because valves of the same nominal size, of different designs or from different manufacturers, can have different flow capacities. The C_v coefficient provides a common basis for comparison. The C_v of a valve is determined by testing, and is equal to the number of gallons of water that will flow through the valve when the pressure differential across it is one psi. The characteristic of the valve is the relationship between the stroke of the valve (or lift or rotation) and the valve C_v .

The most common valve characteristics are linear and equal percentage, shown in Figure 39. The linear characteristic is, as the name implies, linear: equal changes in

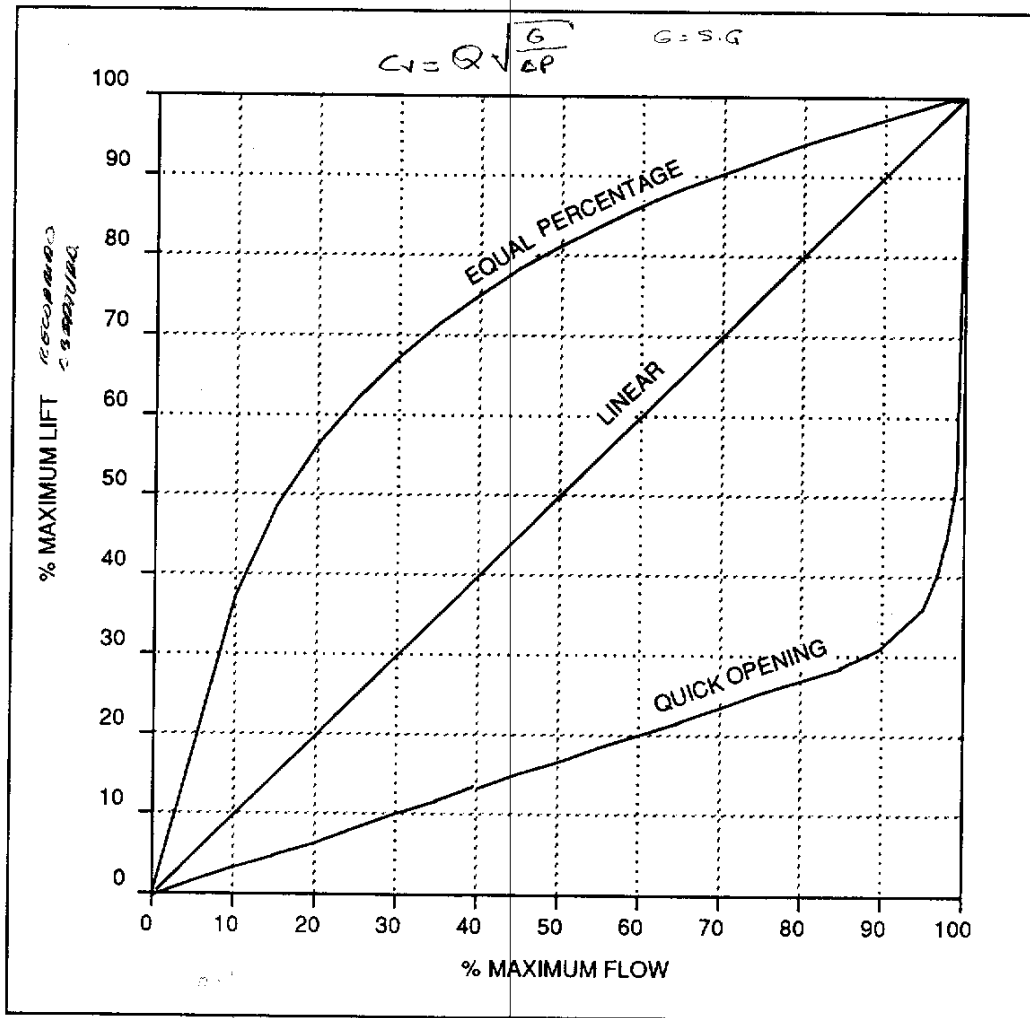


Figure 39 - Control Valve Characteristics

lift yield equal changes in flow (at constant pressure drop). With the equal percentage characteristic, equal changes in lift yield equal percentage changes in flow (again at constant pressure drop). For instance, the Figure shows that a 10% change in lift from 10% to 20% of maximum lift changes the flow from 3.7% to 5.3% of maximum flow. Based on the initial value, this is a 43% change in flow. Likewise, a 10% change in lift from 80% to 90% of maximum lift changes the flow from 48% to 69% of maximum flow. Again, based on the initial value, this is a 44% change in flow, roughly the same as for the 10% to 20% case. The linear characteristic plots as a straight line on linear graph paper; the percentage characteristic plots as a straight line on semilog

graph paper.

UOP uses linear trim for most applications, and percentage trim for applications requiring wide rangeability in terms of C_v requirement. Referring again to the percentage characteristic shown in the Figure, we can approximate the curve fairly well by drawing two intersecting straight tangential lines from the corners at 0% and 100%. In effect, the percentage trim valve can be considered equivalent to two linear valves of different sizes. (If the range of C_v requirements is about 10 to 1, it can probably be handled with one percentage trim valve; if the range is greater, it will probably require two separate, parallel valves. Split range positioners or transducers would be used if the C_v requirement changes frequently; if the operating case does change often, manual shutoff valves in the air tubing should be sufficient.) Percentage trim is also used at UOP for valves in temperature control loops; application of percentage trim to temperature control can be traced back to some of the pioneering studies done on control systems by Ziegler and Nichols in the 1940's.

In actual practice, the characteristic generally is not critical. It might be necessary to alter the controller tuning to obtain optimum control if the plant throughput is changed, but in most cases even that is not likely to be necessary. For reduced trim sizes below one inch, there is often no choice, anyway; in their standard globe designs, Fisher offers only percentage trim, and Masoneilan offers only linear in those sizes.

One additional characteristic, quick opening, is used mainly on double diaphragm fuel gas valves and atomizing steam valves. As the Figure indicates, this is essentially a linear characteristic operating over the first 30% or so of lift. It is used for the two valves mentioned because the actuator's diaphragm does not permit the length of stroke normally used with a standard linear plug.

Hydraulics

When valve characteristic is mentioned, it is usually the inherent characteristic that is implied. The inherent characteristic is the characteristic of the valve with a constant

pressure drop, and under this condition is identical to a plot of % Maximum C_v versus % Maximum Lift. In that respect, it is somewhat analogous to the "bench set" spring range of an actuator. The installed characteristic of the valve takes into account the effect of system hydraulics on the valve pressure drop, and can be very different from the inherent characteristic.

As an example, assume a valve with a linear inherent characteristic is installed in a system which has a very large friction drop compared with the pressure drop allowed for the valve. Assume also that the valve is sized in accordance with the usual UOP practice, which is to calculate the C_v required at normal flow conditions, and then double that Normal Flow C_v to obtain the Approximate Valve C_v . At the normal flow condition, then the valve lift should be about 50%.

The control valve can change the flow only by varying the area of the restriction created by the plug and the seat, and thereby exchanging some of the resistance (pressure drop) of the valve with resistance in the remainder of the system. If the pressure drop allowed for the valve at normal conditions is small compared with the friction drop of the system, even if the valve goes wide open, there is little additional pressure drop available to add to the system drop and increase the flow; so the maximum flow cannot be much greater than the normal flow. Referring to the Figure, this means that the characteristic curve passes through the point of 50% of maximum lift well to the right of the linear characteristic curve, and very near 100% of maximum flow. In contrast, when the valve is just beginning to open and the flow rate is low, the system pressure drop is also low, and almost all the pressure drop available must be taken across the valve. Therefore the flow through the valve is higher than it would be if the valve pressure drop were held constant at the normal flow condition, and again, the characteristic curve is shifted well to the right. In effect, the valve has an installed characteristic which approaches quick-opening, even though the inherent characteristic of the valve is linear.

Since the controller is only concerned with the response to a change in its output, it is the installed characteristic that is important. In this example, the maximum flow is very near the normal flow, and a large change in lift is required to obtain a small percentage change in flow; at lower flow rates, a small change in lift causes a large percentage change in flow. In this situation, the control is not likely to be very good; the optimum controller settings will change too much as the flow rate changes.

Looking at the Figure, we might consider: if the hydraulics of the system in this example can take the linear inherent characteristic, shift it to the right, and distort it into a quick-opening characteristic, what would happen if we started with a percentage inherent characteristic instead of linear? Since it is always to the left of the linear characteristic, shifting the percentage characteristic to the right should make it more nearly linear. That is, in fact, what does happen, and this type of situation is where percentage trim is likely to give better control. But in extreme cases, such as this example, even percentage trim cannot compensate for the faulty hydraulics.

Fortunately, it is usually possible to design the hydraulics so this situation does not occur. As mentioned earlier, the usual practice at UOP is to select a valve with a C_v which is approximately twice the calculated Normal Flow C_v . In conjunction with properly planned hydraulics, this allows the system to pass about 10-15% more flow with the valve wide open than at the design condition. This extra margin provides room for variation from the design conditions, which are really only estimates, and allows the controller operating the valve to operate on either side of the normal output, during an upset, without running into the limits of its range.

To obtain this margin, the following guidelines are used to set the minimum design pressure drop for the control valve:

- 1.) 50% of the system friction drop, exclusive of the control valve, at the normal flow rate;

- 2.) 10% of pump differential (Note: the pump differential here is the differential obtained if the control valve drop is set in accordance with 1.), above); or
- 3.) 25 psi.

The first guideline, 50% of the system friction drop, derives from the square root relationship between differential and flow. If the valve has 50% of the system friction drop, it has 1/3 of the total pressure drop. If the valve goes wide open and takes zero pressure drop, the pressure drop of the system becomes 3/2 of its normal pressure drop, and the flow becomes larger by $(3/2)^{0.5}$, or 1.22 times the normal flow. This is more than the 10-15% margin mentioned above, but in actual practice, the valve pressure drop can never be zero, so the true figure will be somewhat less.

The second guideline is primarily for high head pumps. Its purpose is to allow for variations in head due to design inaccuracies, and for changes in specific gravity due to temperature or composition.

The third guideline is somewhat arbitrary. It is chosen partly because low head process pumps, like control valves themselves, come in sizes which may not exactly match the specification. Also, the 25 psi pressure drop should be large enough, in comparison to the pressure variations that can be expected during normal operation, to ensure that the valve will not have trouble controlling.

The basic control valve C_v equation for liquid service is:

$$C_v = Q (G/\Delta P)^{0.5} \quad \text{Eq.(2)}$$

where

Q = Flow Rate, gpm at flowing temperature

G = Specific Gravity, at flowing temperature

ΔP = Differential Pressure, psi.

Figure 40 shows an example of a typical system which follows the first (friction drop) guideline. For this system, the maximum flow, with the valve wide open, is 115.3% of

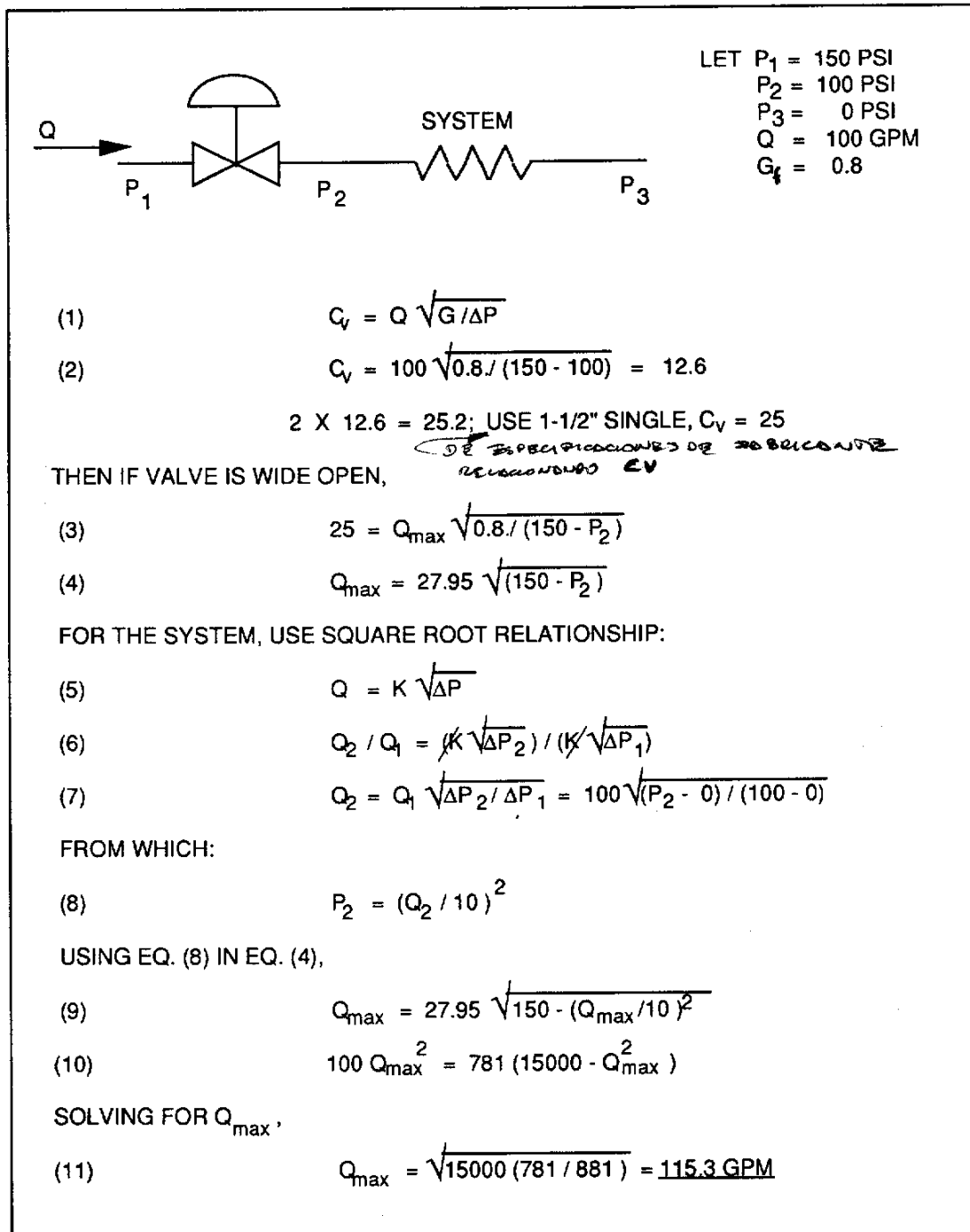


Figure 40 - Maximum Flow with 50% of Friction Drop for Valve

the normal flow. As noted earlier, this is less than 122% because the pressure drop of the wide-open valve is not zero.

The second (pump differential) guideline is illustrated in the three figures that follow

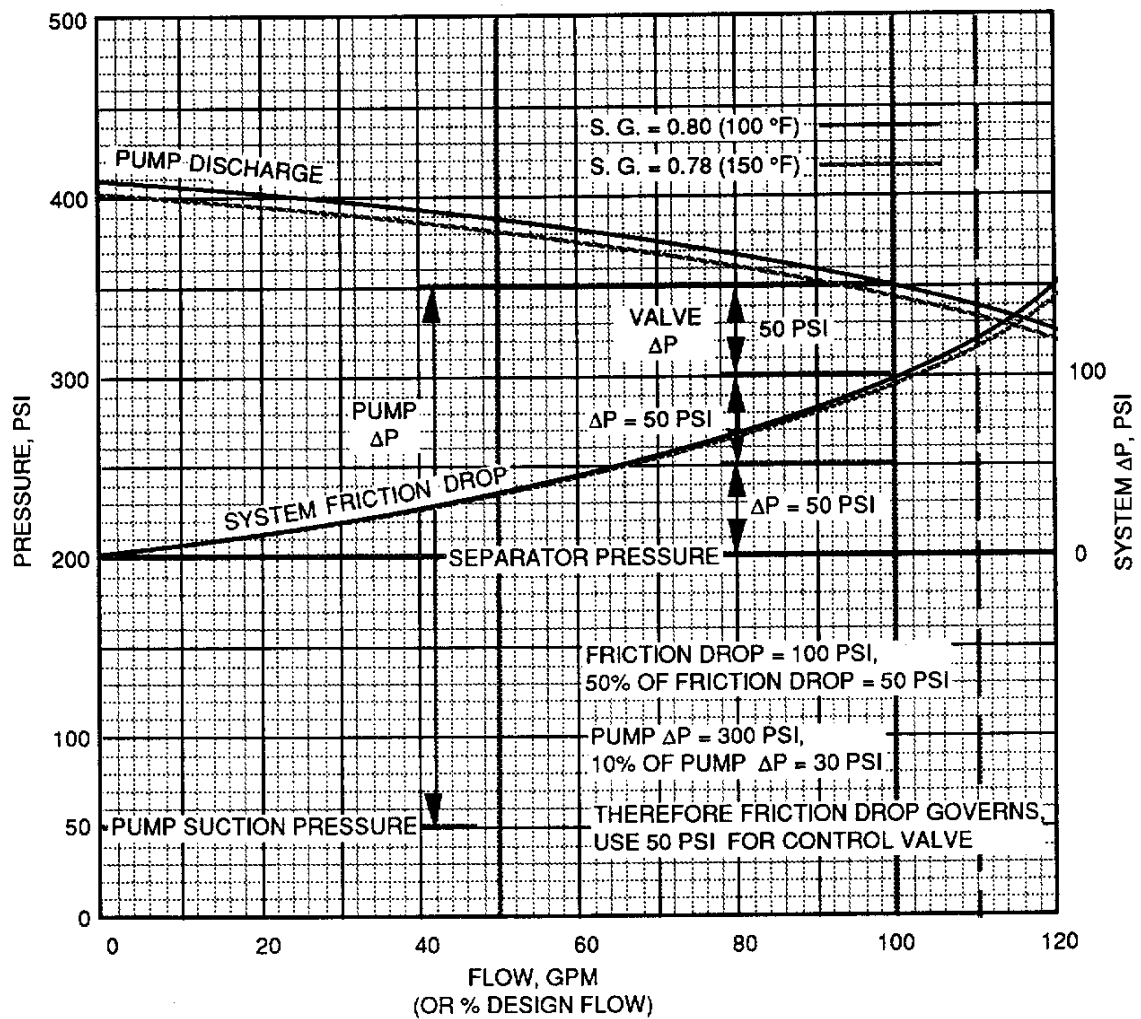
(Figures 41, 42, and 43). In all cases, the specific gravity of the liquid is assumed to be 0.80 at 100 °F, and 0.78 at 150 °F, corresponding to a material with a 42 API Gravity. The system is a typical reactor feed circuit, and consists of the pump suction and discharge piping, along with the combined feed/effluent exchanger, heaters, reactors, products condenser, and their associated piping. The system friction drop is the pressure drop due to flow through this piping and equipment.

We should probably note that the line plotted as System Friction Drop in the figures is actually an oversimplification. Recycle Gas flowing through the heaters, reactors, and exchangers also contributes to the pressure drop, and temperature and phase changes make the system drop a much more complex function than the line that is shown. But it does serve to illustrate the point. We should also note that allowances for fouling of the exchangers or reactors should be included in the friction drop; otherwise the length of run or the throughput will be reduced as fouling occurs.

In the case of Figure 41, the process is assumed to be a relatively low pressure process, such as a Platforming Unit. In this case, doubling the valve C_v required at normal conditions yields more than 10% additional flow with the valve wide open, and very near 10% additional even with the change in specific gravity.

In the case of Figure 42, the pump differential pressure actually should govern, based on the second guideline, but it has purposely been ignored. The result is that it is not possible to reach 110% of the normal flow with the 0.80 specific gravity, and with the 0.78 specific gravity, it is not even possible to reach 105% of normal flow.

The next figure, Figure 43, illustrates several important points. First, it shows that the problems encountered above do not occur when the second guideline, which required 10% of pump differential pressure as minimum drop for the control valve, is followed. If the valve had twice the normal flow C_v , it would have a C_v of 17, it would control up to 110% of normal flow at the 0.80 specific gravity, and it would come fairly close to doing so even at the 0.78 specific gravity. But a valve with a C_v of 17 is not available;



S.G. = 0.80

$$C_v = 100 \sqrt{0.8 / (350 - 300)} = 12.6$$

SAME AS FIG. 40,
USE 1-1/2" SINGLE, $C_v = 25$

AT 110% OF DESIGN

$$C_v = 110 \sqrt{0.8 / (338 - 321)} = 23.9$$

AT 50% OF DESIGN

$$C_v = 50 \sqrt{0.8 / (392 - 225)} = 3.5$$

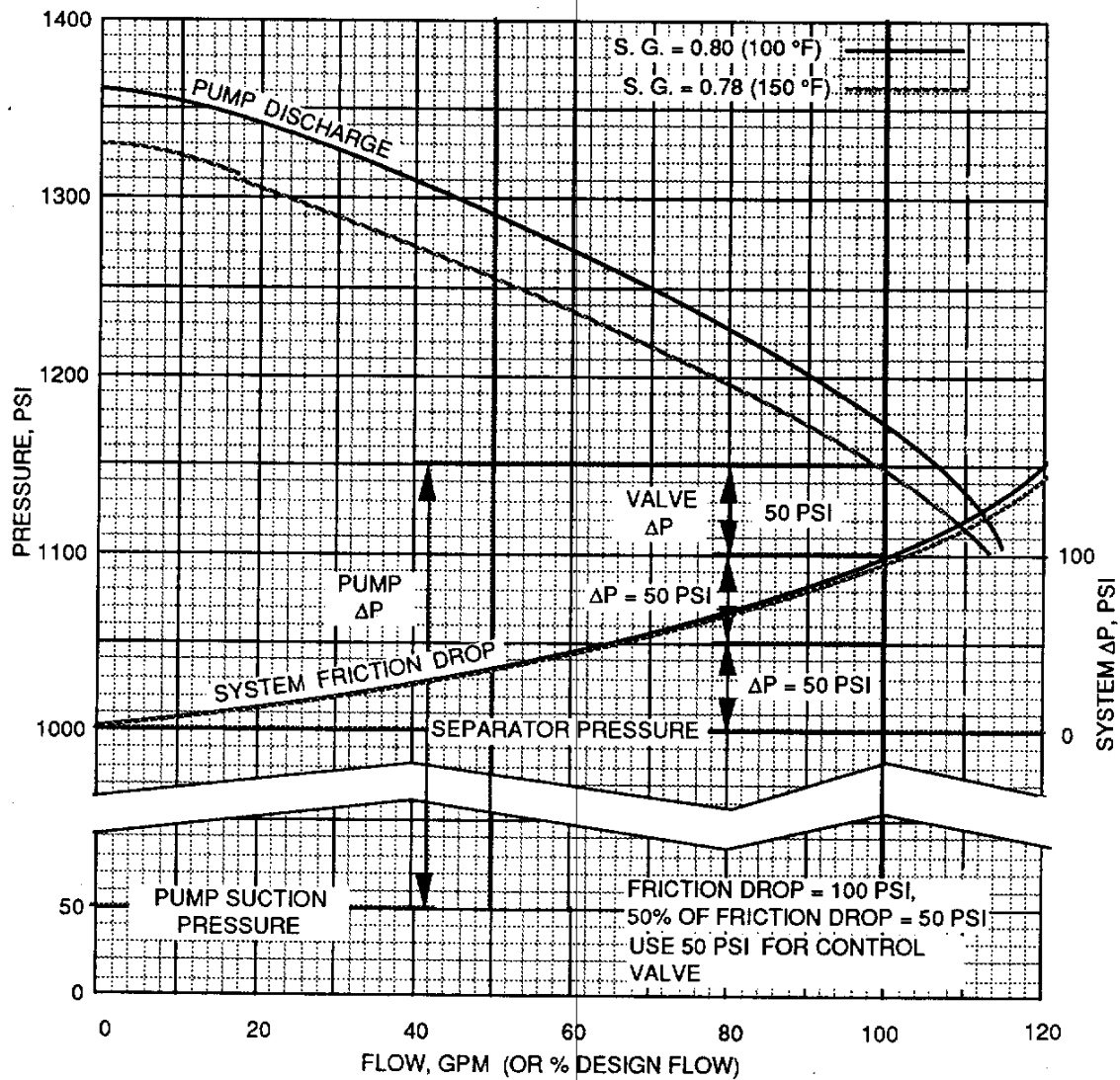
S.G. = 0.78

$$C_v = 100 \sqrt{0.78 / (342 - 298)} = 13.3$$

AT 110% OF DESIGN

$$C_v = 110 \sqrt{0.78 / (331 - 318)} = 26.9$$

Figure 41 - Low Pressure Pump, Motor Drive



S.G. = 0.80

$$C_v = 100 \sqrt{0.8 / (1175 - 1100)} = 12.6$$

SAME AS FIG. 40 AND FIG. 41
USE 1-1/2" SINGLE, C = 25

AT 110% OF DESIGN, ΔP IS NEGATIVE

$$C_v = 110 \sqrt{0.8 / (1135 - 1120)} = 25.4$$

AT 105% OF DESIGN,

$$C_v = 100 \sqrt{0.8 / (1132 - 1110)} = 20.0$$

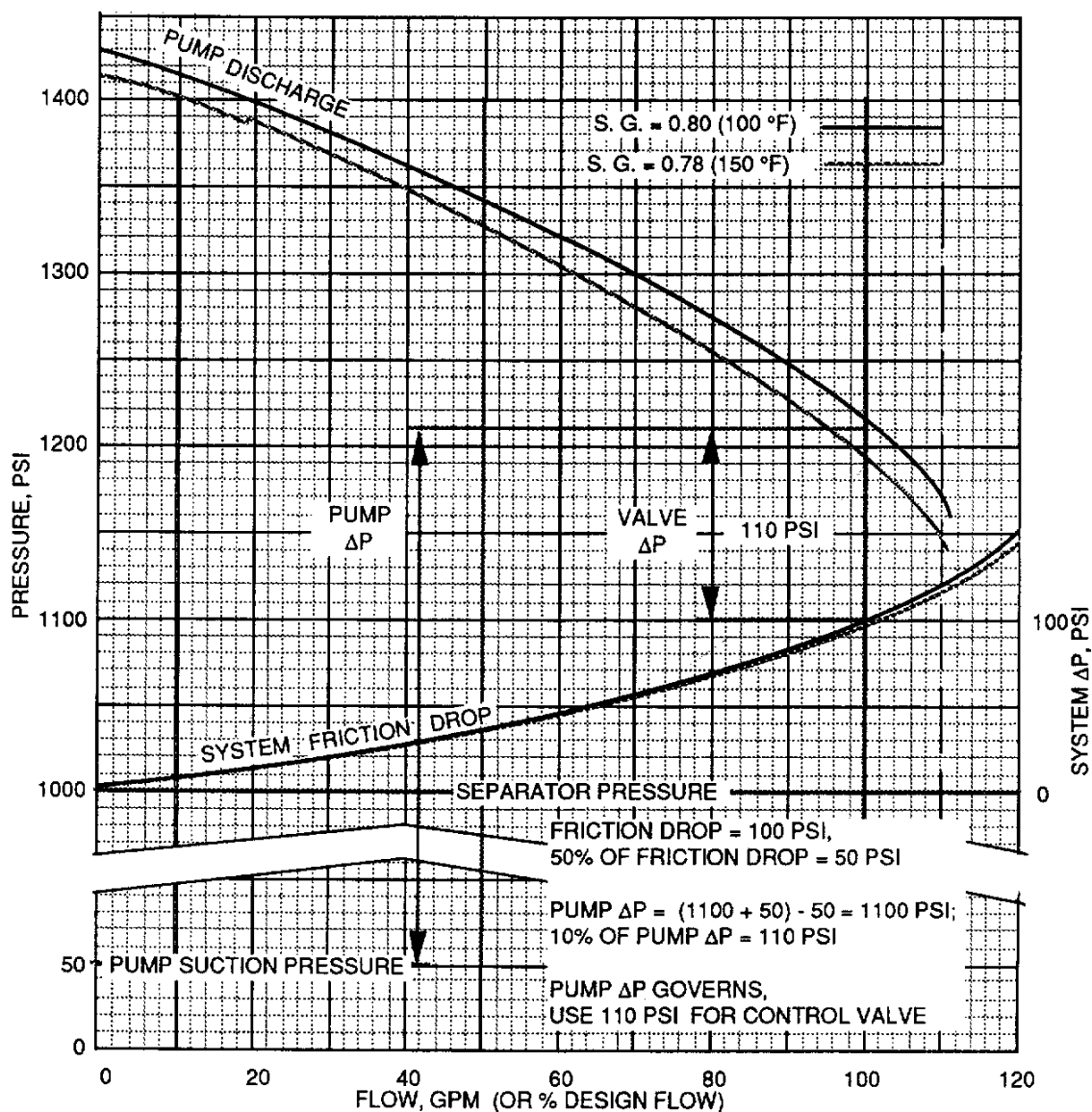
S.G. = 0.78

$$C_v = 100 \sqrt{0.78 / (1150 - 1098)} = 12.25$$

AT 105% OF DESIGN, ΔP IS NEGATIVE

$$C_v = 105 \sqrt{0.78 / (1132 - 1102)} = 16.92$$

Figure 42 - High Pressure Pump, Motor Drive, Only Friction Drop Considered



S.G. = 0.80

$$C_v = 100 \sqrt{0.8 / (1210 - 1100)} = 8.5$$

USE 1" FULL SINGLE, C = 12, OR
1-1/2" FULL SINGLE, C = 25

AT 110% OF DESIGN

$$C_v = 100 \sqrt{0.8 / (1165 - 1121)} = 14.8$$

AT 50% OF DESIGN.

$$C_v = 100 \sqrt{\frac{1345 - 1025}{0.8 \cdot (1372 - 1025)}} = 2.4$$

S.G. = 0.78

$$C_v = 100 \sqrt{0.78 / (1181 - 1098)} = 9.7$$

AT 105% OF DESIGN

$$C_v = 100 \sqrt{0.78 / (1138 - 1118)} = 21.7$$

Figure 43 - High Pressure Pump, Motor Drive, Pump Head and Friction Drop Considered

we must use either a 1" valve with a C_v of 12, which is 1.4 times normal, or a 1-1/2" valve with a C_v of 25, which is 2.9 times normal.

This illustrates the second important point: control valves are not always available in the ideal size for the application. When a problem of this type occurs, it is necessary to analyze the situation closely. In the case above, the 1" valve would be too small to handle 110% of normal flow, so the 1-1/2" valve would appear to be the better choice.

This leads us to the third point. As the flow rate decreases, not only is there a decrease in the system friction drop, but in addition, there is an increase in the pump head. This causes the valve differential pressure to change much faster than it would from friction drop alone, and as the Figure indicates, at 50% of normal flow, the C_v requirement is only 28% of the normal C_v . For the 1-1/2" valve, this is only 9.6% of the maximum C_v . For startup and occasional operation, this is acceptable, but prolonged operation below about 10% of maximum capacity is not recommended. The reduced capacity usually is accompanied by increased pressure drop (in this case, 350 psi, versus 110 psi at normal flow) which leads to increased erosion and wire drawing of the trim. As noted in the discussion on characteristics, in an application with a wide-ranging capacity requirement such as this, consideration should be given to using percentage trim, to provide a more linear installed characteristic.

There is another side to the question of which valve to choose when no valve is available with twice the Normal Flow C_v . It is not always necessary to choose the larger valve as was done in the example above. If there is no pump, and only system friction drop is involved, and if the valve differential pressure at normal flow is large compared with the friction drop, the desired 10-15% additional flow may be obtainable with only 15-20% more additional C_v .

Figure 44 shows how the relationship between design valve differential and design friction drop affects the relationship between flow and C_v , in a system with constant pressure upstream of the valve and downstream from the system. Using 50% of

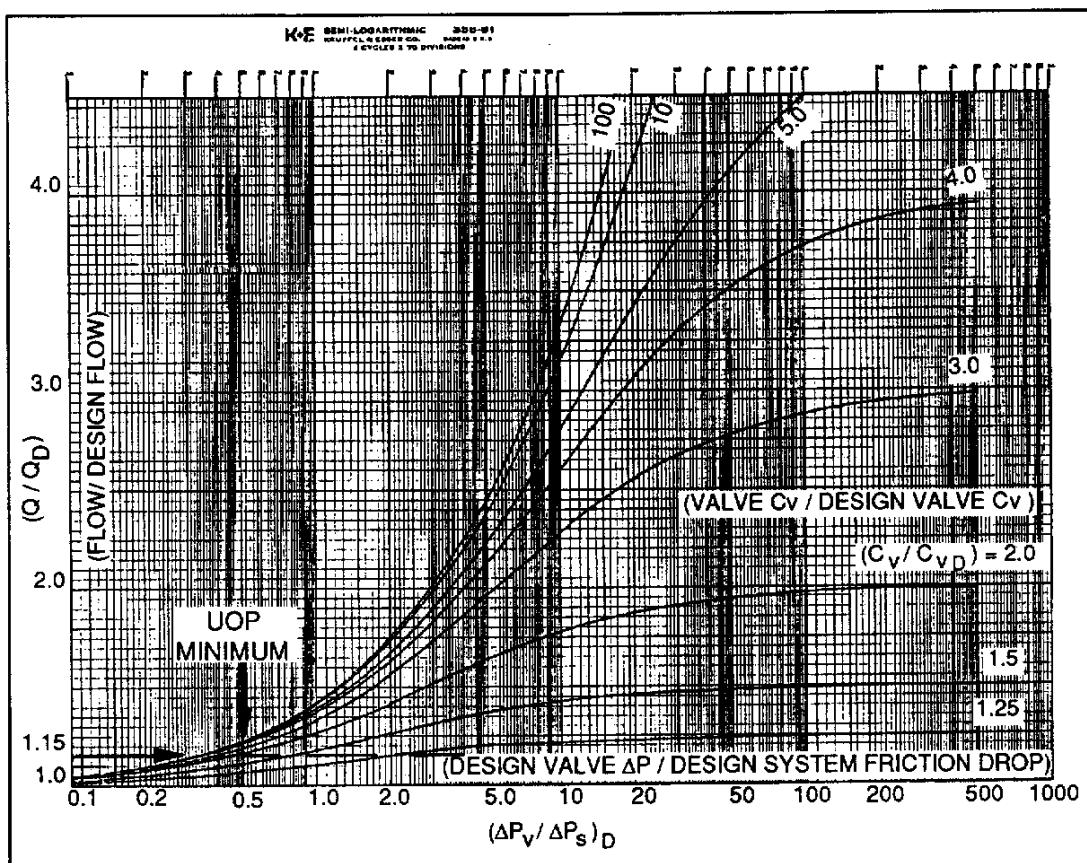


Figure 44 - Control Valve Hydraulics

system friction drop for the valve, in accordance with the guideline, and doubling the resulting normal C_v , yields the same 115% of design flow that was found earlier in Figure 40 (indicated by the arrows in Figure 44). As the ratio of design valve differential to design system drop increases, the ratio of maximum to design flow approaches the ratio of maximum to design C_v .

Many times we do have situations in which the design valve differential is large in comparison to the system friction drop. For instance, consider a stabilizer column operating at 250 psi, with the bottoms product going to storage at 40 psi at the battery limits (Figure 45). Generally, a storage tank has about 20 psi of static head, so the remaining 20 psi of the battery limits pressure must be friction drop. Using 20 psi for the exchangers and 5 psi for the piping inside the battery limits, the total friction drop becomes 45 psi. The control valve therefore has an upstream pressure of 225 psi and a

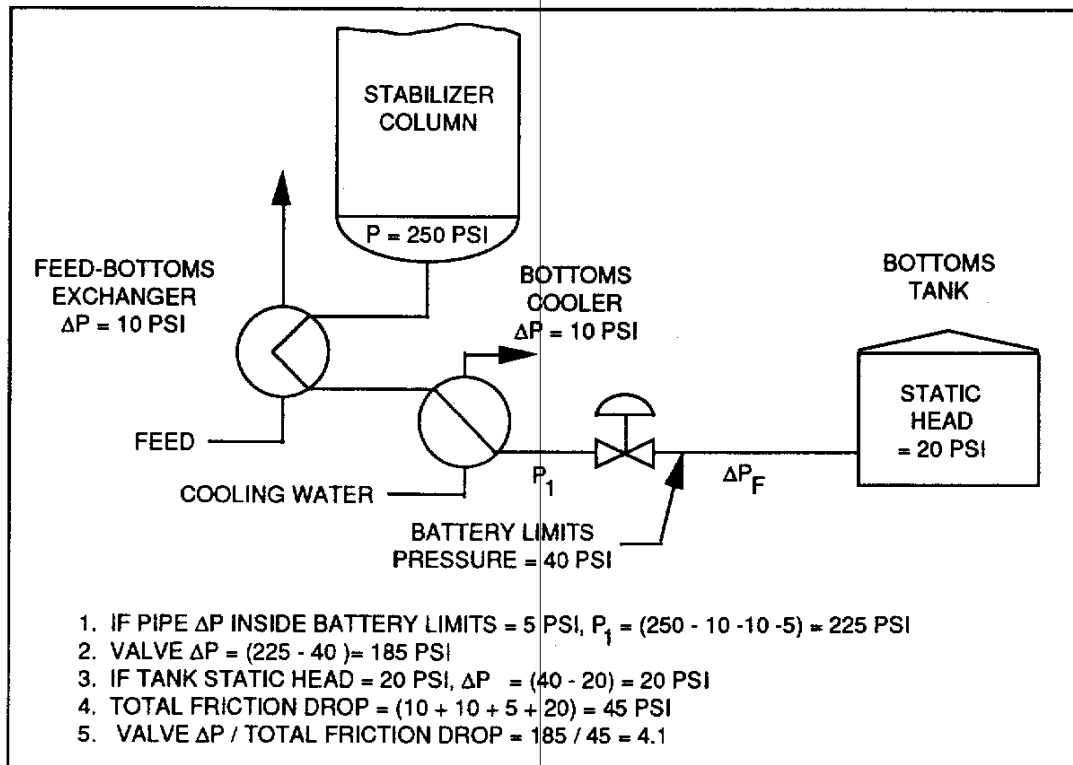


Figure 45 - Hydraulics: Stabilizer Bottoms Example

downstream pressure of 40 psi, and a ratio of valve differential to friction drop of 185 to 45, or 4.1 to 1. For this ratio, Figure 44 indicates that the additional 15% of flow can be met by multiplying the normal C_v by 1.25. If the normal C_v is the same 8.5 that was calculated in the high-head pump example above, the maximum C_v required would be $(8.5 * 1.25) = 10.6$. In this case, the 1" valve with a C_v of 12, which was too small in the pump example, is more than adequate.

Figure 44 also shows very well that trying to compensate for bad hydraulics by oversizing the control valve simply does not work. If the design valve drop is only 20% of the friction drop, even a valve C_v that is 100 times the normal design C_v will only yield a flow that is 110% of design. There is simply no substitute for proper hydraulics.

The guidelines listed - 50% of friction drop, 10% of pump differential, and 25 psi - are used at UOP primarily because the best hydraulics we have for the system are still only a guess. The plant is not yet built, the piping is not laid out, the pumps are not

purchased. For work in which you are doing your own detailed design, the hydraulics should be more accurate; for revamp work, actual flow and pressure drop data should be available. Analysis of the data and the intended range of operation will define the valve sizing requirements better than the general guidelines.

Also, it should be remembered that the guidelines are based on obtaining an additional 10-15% more flow than design by doubling the design C_v . If future operations or other considerations will require flow rates higher than 10-15% over design, the guidelines - and the system hydraulics - will have to be altered accordingly. The guidelines are just that: guidelines. They are not laws which govern every situation.

Probably the guideline which is questioned most often is the third one, which requires a 25 psi minimum pressure drop. Usually the question involves a fractionator which has net gas from the receiver going to the fuel gas system. Unless other considerations are involved, the design pressure of the fractionator is set by the pressure drop taken across the net gas control valve, and there is an advantage in energy savings if the pressure drop is minimized. In a case such as this, we may or may not need the entire 25 psi called for by the guideline; we will not be able to tell until we have analyzed the situation.

The valve pressure drop is a function of the fractionator pressure, the fuel gas system pressure, and the piping friction drop. If we are trying to minimize the fractionator pressure, we will want to minimize the piping pressure drop, so let us assume it can be neglected. The fractionator pressure itself is under our control - the valve we are discussing will be controlling it. So we have a good idea of how much variation to expect in this variable; it should not be much, perhaps 1 or 2 psi. Excursions above the set point only increase the pressure drop across the valve, so we need not consider them in analyzing the minimum drop. Excursions below the set point decrease the pressure drop across the valve - but if the pressure is below the set point, the controller needs to close the valve and reduce the flow anyway, so we can ignore these excursions.

sions, also.

This leaves only the fuel gas system pressure to be considered. If the fuel gas system pressure is also controlled within 1 or 2 psi, we should only need 3 or 4 psi for the normal pressure drop for the control valve. As we will see later, the C_v requirement is essentially a square root function of the differential pressure, therefore if the pressure drop decreases from 3 psi to 1 psi, for the same flow the required C_v increases by the square root of 3, or 30%. If the valve initially is at 50% of maximum capacity, it can easily handle the additional requirement.

Unfortunately, fuel gas system pressure is rarely controlled within 1 to 2 psi; indeed, if the control were that good, we could simply let the fractionator float on the fuel gas system, not take any valve pressure drop, and save the cost of the energy, the control valve itself, and the rest of the loop as well. If we do not allow enough pressure drop for the control valve, that is what will happen, anyway. If we allow 5 psi for the valve, and the fuel gas pressure increases by 5 psi, there is no drop left for the valve. So the fractionator pressure will rise and the valve will go wide open, and the fractionator pressure will follow the fuel gas pressure, until it once again decreases enough to let the valve regain control. For the average fuel gas system, a 5 psi increase in pressure is not infrequent. For variations of this magnitude, a minimum pressure drop of at least 10, and preferably 15, psi should be used - even more if the piping friction drop cannot be neglected as was done in this case.

The best advice regarding hydraulics for control valves is to start with the guidelines, and then analyze the situation to see if there is some special condition that requires modification of the guidelines or prevents their use. Just as there is no substitute for proper hydraulics, there is no substitute for common sense in determining those hydraulics.

One special condition that the guidelines do not cover occurs with steam control valves for steam heaters and reboilers, such as the one shown in Figure 46. The procedure

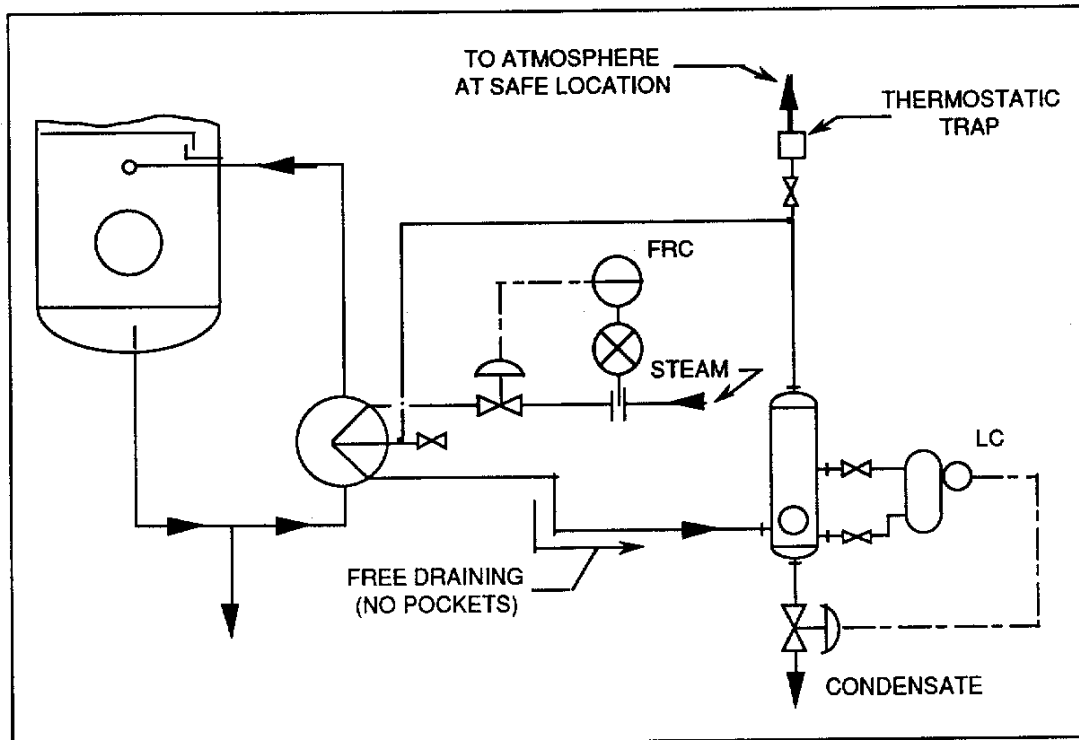


Figure 46 - Steam Reboiler Steam Control

and the derivation of the procedure for this case is covered in the memo titled Sizing Steam Reboiler Control Valves, in the Appendix at the end of this section. In a steam heated reboiler, the control valve varies the heat input by varying the pressure of the steam in the reboiler. The pressure determines the condensing temperature of the steam, and - if conditions on the process side remain constant - the steam temperature determines the LMTD (Log Mean Temperature Difference). Since:

$$Q = U A \Delta T \quad \text{Eq.(3)}$$

where (in English Units)

Q = Heat Transfer Rate, BTU/hr

U = Heat Transfer Coefficient, BTU/hr ft² °F

A = Area, ft²

ΔT = Differential Temperature, °F

setting the steam pressure therefore sets the heat transfer rate. The steam pressure also determines the latent heat of vaporization of the steam. If, as is generally the case,

contributions from superheat and subcooling are small enough to be ignored, knowing the heat transfer rate and the heat of vaporization allows the steam flow rate to be calculated. With the steam flow rate, the header pressure, and the reboiler pressure known, the valve C_v can be calculated. To size the valve, then, all we need to do is to find the LMTD that corresponds to the design heat transfer rate, calculate the reboiler pressure and steam flow rate, and use that information to calculate the size of the valve.

The problem at with this procedure is that at UOP, when the control valve is being sized, the exchanger has not been built, or even designed. UOP does not rate exchangers; this is left to the supplier. The UOP heat exchanger specification is a performance specification only; it tells the supplier what the exchanger must do, but does not tell him in detail how he must build the exchanger to do it. No area is specified, nor is a heat transfer coefficient - only a fouling factor - nor is any contingency. So although Q is known, U and A are not; therefore the LMTD cannot be calculated, and the valve cannot be sized using the procedure above.

Discussions with the heat exchanger group indicate that the actual exchanger will probably be capable of 115 to 120% of design. The transfer coefficient, U , is essentially constant, and when the exchanger is built, the area, A , is fixed, therefore only the LMTD can be changed. So to make maximum use of the exchanger's capability, the maximum LMTD should be 115 to 120% of design, and conversely, the design LMTD should be about 85% of the maximum. The maximum LMTD occurs when the exchanger steam pressure is equal to the steam header pressure, so the control valve design should be based on the LMTD which is equal to 85% of the LMTD which occurs when the reboiler steam pressure is equal to the steam header pressure. We would like to be able to calculate the Normal Flow C_v on this basis, size the control valve in the normal fashion by doubling the C_v , and still be able to obtain at least a 10 to 15% margin over design heat input.

The objective of the memo, Sizing Steam Reboiler Control Valves, was to study this problem. The memo considers four examples which have different steam conditions. The values of U and A are not known, but are constant, so it is possible in each case to calculate a value for C_v/UA , or C_v per "unit" of UA . The C_v/UA was calculated first at the proposed design ΔT (85% of maximum ΔT), and then for other percentages of maximum ΔT . Since U and A are constant, if the value of C_v/UA at these other percentages is ratioed to the value of C_v/UA at design ΔT , the UA terms cancel out. This leaves us with the ratio of the value of the C_v at other percentages of maximum ΔT , to the value of the C_v at design. Similarly, since $Q = UA\Delta T$, and again U and A are constant, the ratio of the value of ΔT at the other percentages to the value of ΔT at design is the same as the ratio of the Q 's at the corresponding conditions. If we plot the C_v ratio versus the percent of maximum ΔT , the result, then is the same as a plot of C_v ratio versus maximum Q .

When the points calculated for the four examples are plotted, they all fall reasonably close to the same curve. The curve shows that at the point where $C_v/C_{v \text{ design}} = 2$, the percent of maximum $\Delta T = 96$. This is 113% of the design value of 85% of maximum ΔT , and is within the desired range of 110 to 115%. Therefore the valve sizing procedure is acceptable.

CONTROL VALVE SIZING FORMULAS

At UOP, the sizing formulas used most often are those in the Masoneilan Handbook for Control Valve Sizing. This is largely because we have generally found the Masoneilan sizing formulas and information simpler and easier to use than those of other manufacturers. As noted earlier in the discussion on hydraulics, detailed, accurate information on system hydraulics is rarely available at UOP, and experience has shown that simple equations are adequate for our purposes.

We have already encountered the basic equation for subcritical liquid flow:

$$C_v = Q (G/\Delta P)^{0.5} \quad \text{Eq. (2)}$$

where

Q = Flow Rate, gpm at flowing temperature

G = Specific Gravity, at flowing temperature

ΔP = Differential Pressure, psi.

Subcritical liquid flow means that there is no cavitation or flashing. The Masoneilan Handbook modifies Equation (2) for critical services where cavitation or flashing occur. Flow is considered critical if

$$P \geq C_f^2 (\Delta P_s) \quad \text{Eq.(4)}$$

where

$$\Delta P_s = P_1 - [0.96 - 0.28 (P/P_c)^{0.5}] P_v \quad \text{Eq.(5)}$$

P_1 = Upstream Pressure, psia

P_v = Vapor Pressure, psia

P_c = Critical Pressure, psia

C_f = Critical Flow Factor

The Critical Flow Factor, C_f , is a function of the geometry of the valve. (An analogous factor, C_{fr} , accounts for the effects of piping reducers at the valve inlet and outlet.) The Critical Flow Factor is used because the flow through a control valve, like the flow through an orifice plate, accelerates past the seat and forms a vena contracta which, as we saw earlier, is the point of minimum pressure. As the fluid decelerates back to normal pipeline velocity, the velocity head is converted back to static head, and some of the pressure drop is recovered. In the discussion on Measuring Means for Flow, we learned that a venturi has a lower permanent loss (better pressure recovery) than an orifice plate. Likewise, a control valve with a smooth, venturi shaped outlet shows a higher percentage of pressure recovery than a valve which has more changes in flow direction and thus more turbulence.

While pressure recovery in a flow measurement device is beneficial, pressure recovery in a control valve is not. A high pressure recovery corresponds to a low value of C_v , which increases the possibility that the ΔP term in Equation (4) will be greater than the $C_v^2(\Delta P_c)$ term, and that critical flow will occur. If the pressure everywhere in the valve is above the liquid vapor pressure, the flow is subcritical, and Equation (2) applies. If the pressure at the valve outlet is below the liquid vapor pressure, some of the liquid vaporizes as it passes through the valve, and flashing occurs: the material downstream from the valve is a mixture of liquid and vapor. If the pressure at the valve outlet is above the liquid vapor pressure, but the pressure at the vena contracta is below the vapor pressure, some of the liquid will vaporize as it passes through the valve, and as the pressure is recovered, the vapor bubbles will collapse until only liquid remains. The collapse of the vapor bubbles is very rapid; the imploding bubbles generate noise, and also create shock waves that can damage the valve and its trim. In addition, the vapor bubbles occupy more volume than the liquid they came from, and thus can limit the flow that the valve can pass.

UOP does not often use Equation (4). Since C_v is a function of valve geometry, and since the valve actually purchased may come from a different line or manufacturer than the one specified, the actual C_v (or equivalent coefficient - K_v , F_v , or whatever) may be quite different. Also, the globe valves usually specified by UOP have C_v factors of 0.85 to 0.95, near enough to 1.00 that whether the C_v factor is considered or ignored, the valve size specified will be the same. So UOP specifications state that factors relating to the geometry of the valve and its installation are not included in sizing the valve. But the vapor pressure and the critical pressure are listed in the specification, and the valve supplier can use them if required.

The main object of the C_v factor is to provide a method of predicting flashing and cavitation in a control valve, and the effects thereof on the C_v requirement. In process

services at UOP, flashing in the valve is already calculated in the process specifications. So the sizing method used is to calculate a C_v for the liquid phase, calculate a C_v for the vapor phase, and sum the two together. This method was in common use before valve manufacturers began using C_f and related factors. Since it has proven to be satisfactory in our experience, UOP still uses it, even though the C_v values calculated may not agree with those calculated by the contractor.

An exception to the policy of ignoring the C_f factor occurs with Masoneilan's 71000 Series angle valve. As mentioned earlier, these valves are specified for slurry services, and where coke particle buildup, etc., may be a problem. With the valve installed flow-to-close, it is self cleaning; particles that would otherwise collect in the body of an ordinary globe valve, as the flow changes direction, are flushed straight through and out the bottom port. A full venturi liner in the throat and outlet area protects the valve body from erosion. Unfortunately, the venturi shape also enhances pressure recovery, and the C_f factor of the valve when used in this manner is a very low 0.48.

In this case, the factor is so significant that, as Figure 31 (Page 45) shows, two separate C_v tables are published for the valve. For the 1" valve, the values listed for C_v and C_f are 13.5 and 0.90, respectively, for flow-to-open, and 25 and 0.48 for flow-to-close. In essence, the C_f factor is built into the C_v table. So to keep the Approximate Valve C_v and the Normal Flow C_v on the same basis, the C_f factor must be used in the C_v calculation, and it should be noted in the specification that the C_v values for that item are based on a C_f factor of 0.48. (The 71000 Series valve is no longer the standard angle valve at Masoneilan, but it is still available, and for the services indicated, it is the angle valve that UOP recommends.)

The liquid service C_v formulas shown above as Equations (2), (4), and (5), are given in Page 3 of the Masoneilan Handbook; formulas for gas services are given on Page 5.

At UOP, the C_f factor is ignored for gas services, also. Page 6 of the Handbook gives equations for gas and vapor services which are intended to be more precise, but are somewhat more complicated, and therefore better suited to calculation by a computer or programmable calculator.

As noted earlier, the reason for ignoring the C_f factor at UOP is that the factor for the actual valve may be different. If the valve type and manufacturer are known in advance, using the manufacturer's equations, with the geometrically dependent factors included, would be appropriate. This would most likely be the case if valves were being purchased for a revamp, and spare parts inventories could be reduced by using the same manufacturer as the original valves.

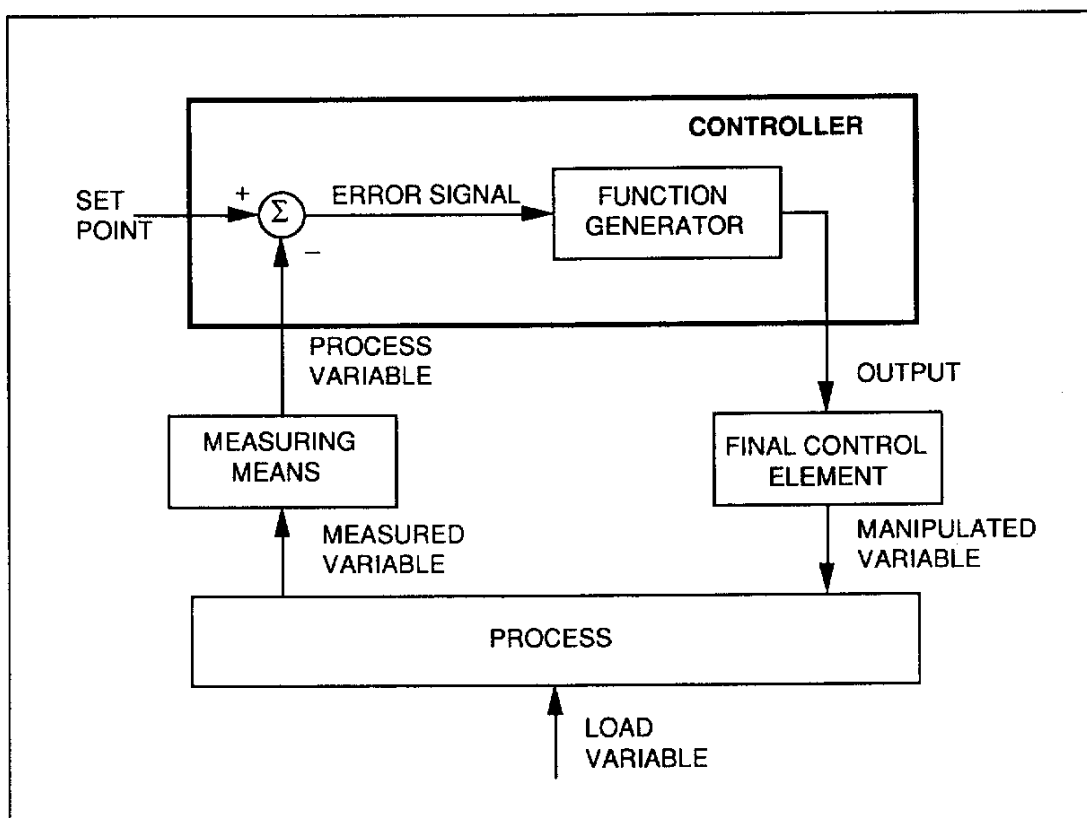


Figure 2 - Feedback Control Loop

CONTROLLER

The Controller in Figure 2 receives the Process Variable signal from the transmitter, or Measuring Means, and generates an Output signal which operates the Final Control Element. In some cases, the means used for transmission of these signals can be important.

Transmission Means and Response

It is not customary to show a block titled "Transmission Means" in the feedback loop diagram, but the means used to transmit the Process Variable to the Controller, and the Controller output to the Final Control Element actually is a part of the loop, and does affect the control.

For many years, process control was almost entirely pneumatic, with a 3 to 15 psig

signal corresponding to 0 to 100% of the signal range. About 1955, electronic control was introduced. Several different signal ranges were used, but currently the industry standard is 4 to 20 mA DC. Electrical transmission and instrumentation is now used more often than pneumatic because it is more compatible with the longer distances associated with centralized control rooms, and with data logging equipment. Since about 1975, the trend has been toward digital transmission and data highways in some portions of the loop.

The means of transmission would not matter if it did not affect the loop response. The transmission of the 4 to 20 mA electrical signal is essentially instantaneous, and therefore it does not affect the overall response of the loop. The problems lie in the pneumatic and digital systems.

Pneumatic Transmission

In any compressible medium, the speed of a pressure change is limited to the speed of sound. The velocity of sound in air at atmospheric conditions is about 1100 feet per second; so if a pneumatic transmitter is located 1000 feet from the controller and the output changes, it will take almost a full second before the input to the controller can even start to change. In addition, each unit length of air tubing has a capacitance (volume), and resistance to flow. The transmitter must fill up each succeeding volume by passing the air through all the preceding resistances. The net result is that once the pressure at the controller input begins to change, it changes in an exponential manner.

Initially, 100% of the pressure change is available to drive the air through the tubing, and the pressure at the controller changes rapidly. As the pressure at the controller changes, and approaches the pressure at the transmitter, there is less driving force to move the air through the tubing, and the rate of change of pressure at the controller decreases. If the pressure at the controller is plotted versus time, the result is as shown in Figure 47.

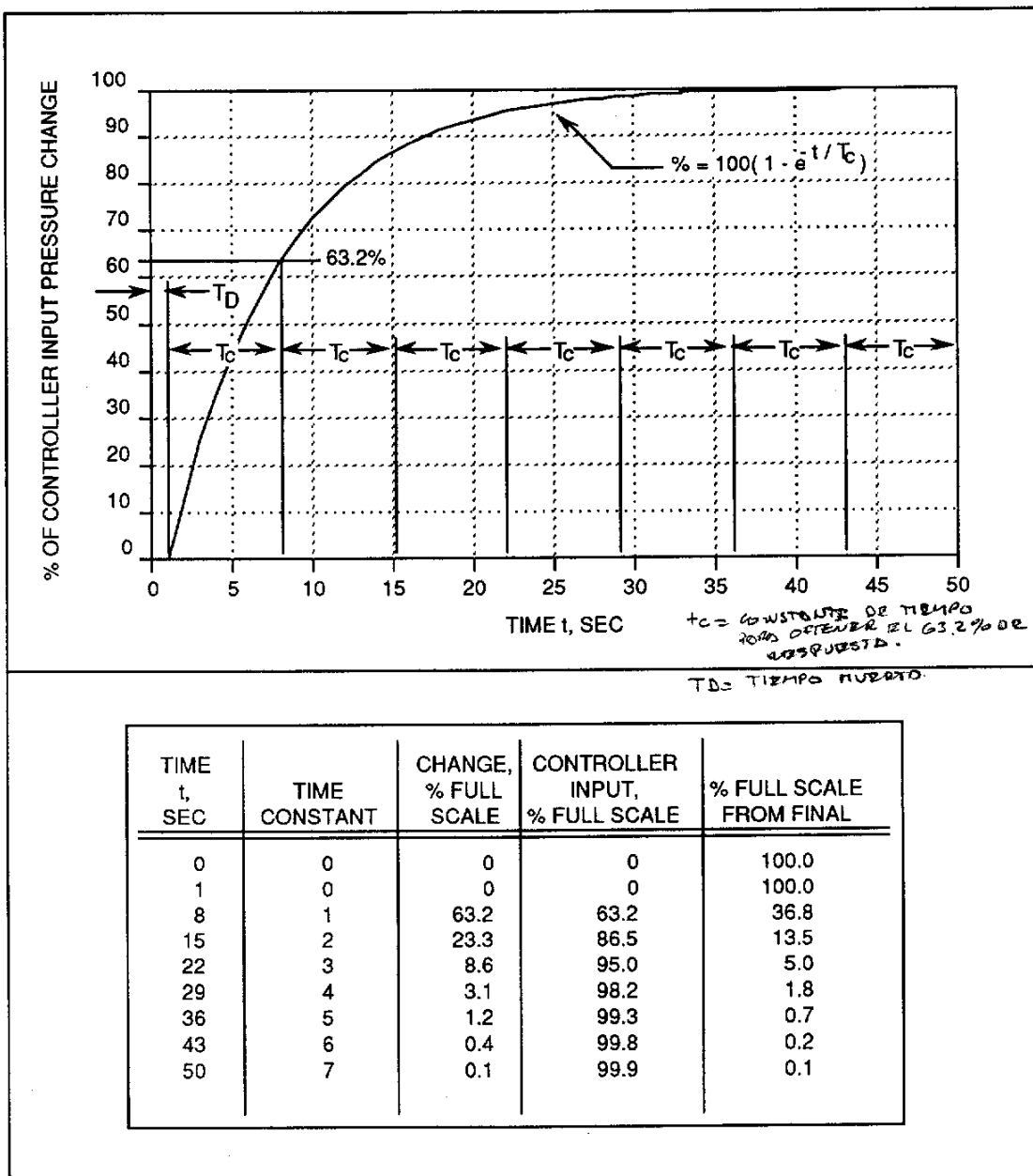


Figure 47 - Pneumatic Tubing Time Response
as First Order Lag + Dead Time
(1000 ft of 1/4-Inch Tube)

The plot in the Figure illustrates the transient response of a single time constant process (also called a First Order Lag) plus a dead time. The single time constant process can be recognized because the steepest slope occurs at the start of the curve. The dead time, T_D , is the time from the initial step change in transmitter output to the

first change in input signal at the controller. The time constant, T_c , is the time required from the first change in controller input signal until it reaches 63.2% of the total change. These two times are important in the determination of system response and stability.

In electrical terms, the time constant, T_c , is equal to RC , resistance times capacitance. For the pneumatic system, R and C are not so easy to define, but it does not matter, because we do not need to measure them. We know that if the system is linear, the time required to make 63.2% of any total change toward the steady state equilibrium value will be one time constant. So we do not have to determine the values of R and C and multiply them together; all we need to do is look at the transient response curve. In the Figure, the controller input changes from 0% to 63.2% of full scale in 7 seconds, therefore the value of T_c must be 7 seconds.

After the first time constant, the controller input signal is 36.8% of full scale away from the final value. In the next time constant (next 7 seconds), the controller input will change 63.2% of the remaining 36.8% of full scale, or 23.3% of full scale. This puts the input at 60.1% of full scale, 39.9% of full scale away from the final value. The sequence is shown in the table in the figure.

As the table shows, it takes four time constants to achieve 98% of the final value, and five time constants to reach 99%. If one time constant is 7 seconds, this means that, if the one second dead time is included, when the transmitter output changes it requires 36 seconds for the controller to almost equal the transmitter output. If this controller operates a valve located near the transmitter, the controller output will have a similar response.

Obviously, a loop with lags like this would be difficult to control. A limit of 300 to 400 feet for pneumatic transmission is more realistic. And even then, UOP's practice is to install controllers for flow and pressure loops in the field, at the control valve, to minimize the transmission lag and improve the response.

Response can be measured directly in terms of time, as defined by the time constant, or in terms of frequency. To determine frequency response for any component, a sinusoidal signal of varying frequency is applied to the input side of the component, and the magnitude and phase shift of the output signal are measured. When the results are plotted, a characteristic response curve results.

The component being tested may be a single element or a complex system. It may be pneumatic or electrical or mechanical, or any of the categories mentioned earlier in this discussion. High fidelity audio equipment is a good example. Manufacturers and magazines publish plots showing frequency response from 20 to 20,000 Hz. The plot that is usually shown is the Magnitude Ratio, or Gain, and a smooth, flat line is representative of an output that faithfully reproduces the input.

Returning to the pneumatic tubing model, let us assume that the transmitter output changes in a sinusoidal manner. If the frequency is very low, the one second dead time becomes insignificant, and the pressure and flow changes are slow enough that the tubing resistance is also insignificant. So at low frequencies, the output essentially follows the input directly. As the frequency increases, the effect of the resistance becomes more pronounced, and the controller input begins to lag behind the transmitter output. Because the controller input lags the transmitter output, the transmitter output has peaked and begun to decrease again before the controller input reaches its peak. Therefore the controller input can never reach the same magnitude as the transmitter output, and the gain, or magnitude ratio, must decrease. As the frequency increases, the lag increases toward 90° , and the magnitude ratio becomes smaller and smaller. The result is the plot shown in Figure 48.

The Figure combines a plot of log gain versus log frequency with a secondary plot of phase shift versus log frequency. It is generally called a Bode plot, after the man who developed it. The gain plot is logarithmic because gains for components in a loop can be multiplied quickly to find the total gain at each frequency by simply graphically

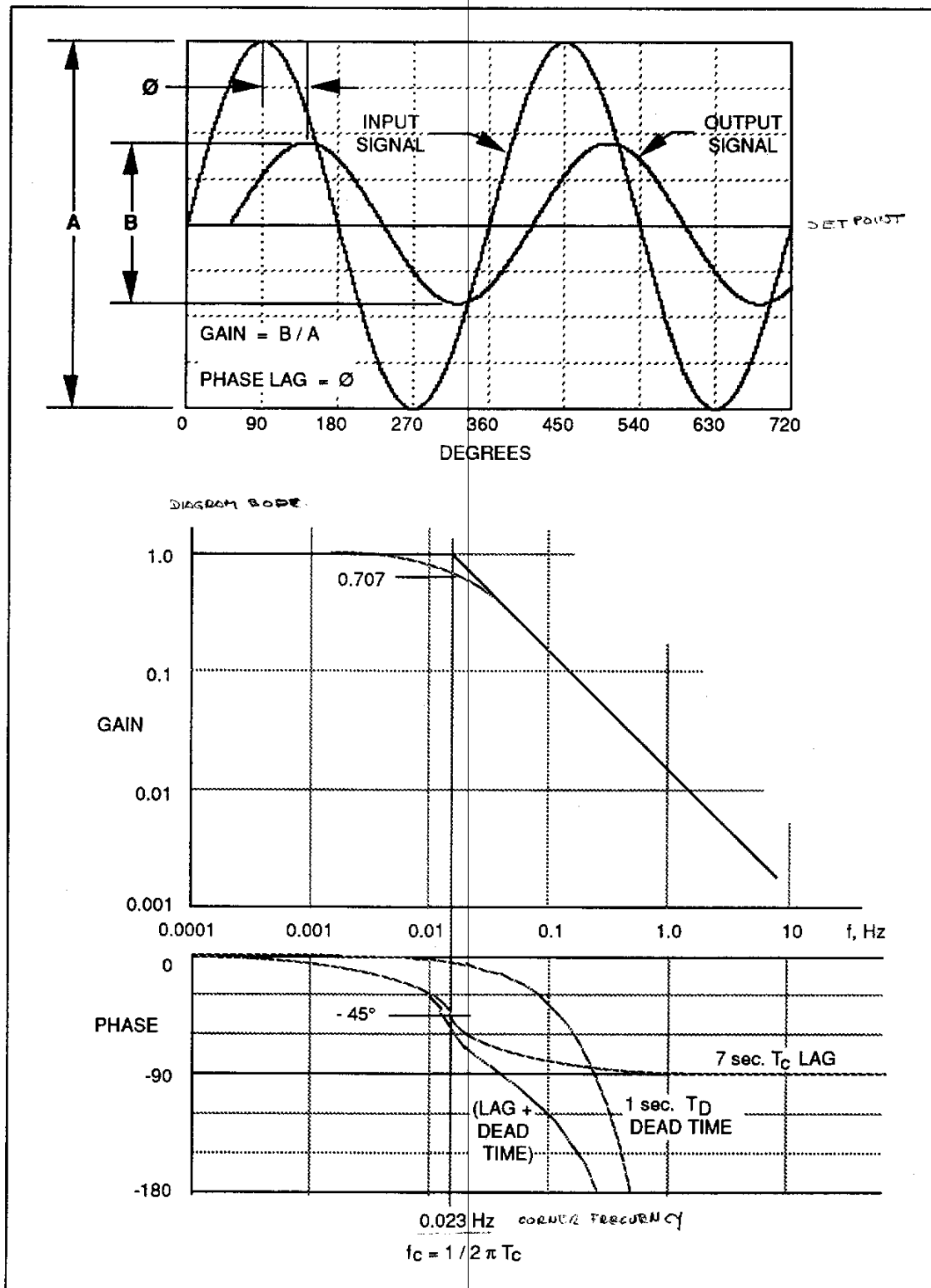


Figure 48 - Pneumatic Tubing Frequency Response

adding the individual gains, for instance by using a pair of dividers, and thereby develop a plot for the entire loop. If two straight lines are drawn asymptotic to the gain curve, the intersection of the two segments occurs at the "break point", or "corner frequency". The frequency at this point is related directly to the time constant. The corner frequency in radians per unit time is equal to the reciprocal of the time constant. Since there are 2π radians in 360° :

$$1/T_c = 2 \pi f \quad \text{Eq. (6)}$$

where

T_c = Time Constant

f = corner frequency, Hz.

The gain at this frequency is 0.707, and the phase shift is -45° . As the frequency increases beyond the corner frequency, the gain decreases logarithmically, and the phase shift approaches -90° asymptotically. It might seem at first that if the gain continually decreases, the phase shift also should continually increase. Studying the Figure helps to show why this is not the case.

As mentioned earlier, at low frequencies, the sine wave at the transmitter output and the sine wave at the controller input are virtually identical. As the frequency increases, the sine wave at the controller input begins to lag behind, displaced by, let us say, N degrees. Also as mentioned earlier, because the controller input lags the transmitter output, the transmitter output has passed its peak and has already begun to decrease when the controller input reaches its peak.

The maximum value of a sine wave occurs 90° after the sine wave crosses zero. Because the frequency of the controller input is the same as the transmitter output, the maximum value of the controller input occurs when the transmitter output sine wave reaches $(90^\circ + N)$ degrees, where N represents the lag. Because the transmitter output has already passed 90° and is decreasing, the amplitude of the controller input peak cannot be greater than the transmitter output at $(90^\circ + N)$ degrees. As the lag, N ,

increases toward 90° , the controller input peak occurs nearer and nearer to 180° on the transmitter output sine wave, and the peak amplitude decreases toward zero. It is not possible to go beyond this point, so the phase lag is limited to -90° .

Dead time in the system causes a pure (no change in gain) phase shift, proportional to frequency. If the dead time is one second and the frequency is one hertz (360° per second), the phase shift is -360° . If the frequency is two hertz (720° per second) the phase shift is -720° , and so on.

Using Equation (6), the 7 second time constant of the 1000 feet of pneumatic tubing we used as an example translates into a corner frequency of 0.023 Hz. This is much too slow for good control. UOP standards require a frequency response of at least one Hz, more than 43 times faster. With the length of tubing shortened to 300 to 400 feet, the dead time is about $1/3$ second, and the time constant is about one second. The one second time constant translates to a corner frequency of about 0.16 Hz, which is still below the standard.

This is the reason that UOP standards require mounting pneumatic controllers for flow and pressure control loops in the field, as near as possible to the transmitter and control valve, so the tubing lengths in the control portion of the loop are minimized. To obtain the proper response from flow and pressure loops with the controller mounted on the control board, the length of tubing would have to be limited to about 150 to 200 feet, which is obviously not practical. Contractors do not like to mount pneumatic controllers in the field, because each field mounted controller requires a "four-pipe" tubing system (Figure 49), versus a less expensive "two-pipe" system (Figure 50) for the board mounted controller.

In contrast, pneumatic controllers for most temperature control loops, and for many level control loops can be mounted on the control board. The temperature signal is from a thermocouple, an electrical device, and the signal to the control room is electronic, so there is no transmission lag. The process response is slow, so there is no

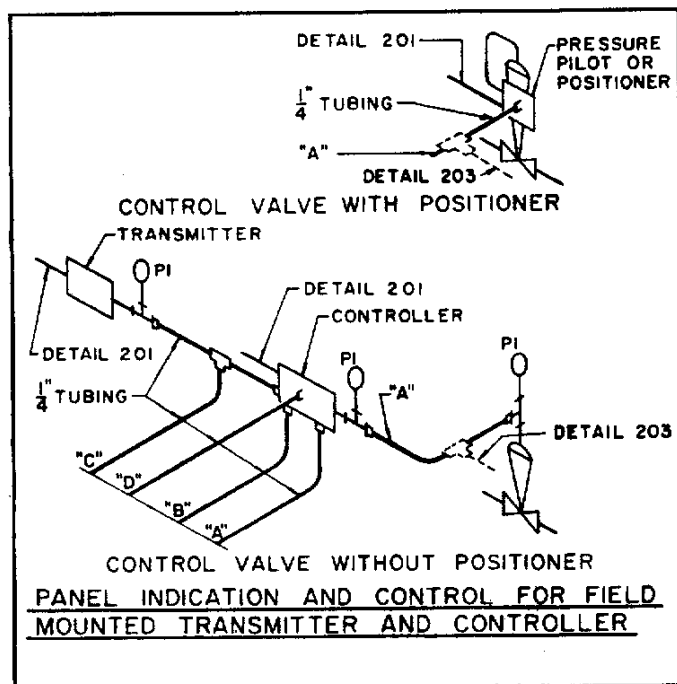


Figure 49 - "Four-Pipe" Installation

NOTES:

"A" = CONTROL VALVE
"B" = CUT-OFF RELAY
"C" = TRANSMITTER
"D" = SET POINT

DETAIL 201 = AIR SUPPLY
PIPING
DETAIL 203 = TWO VALVE
HOOKUP (IF
REQUIRED)

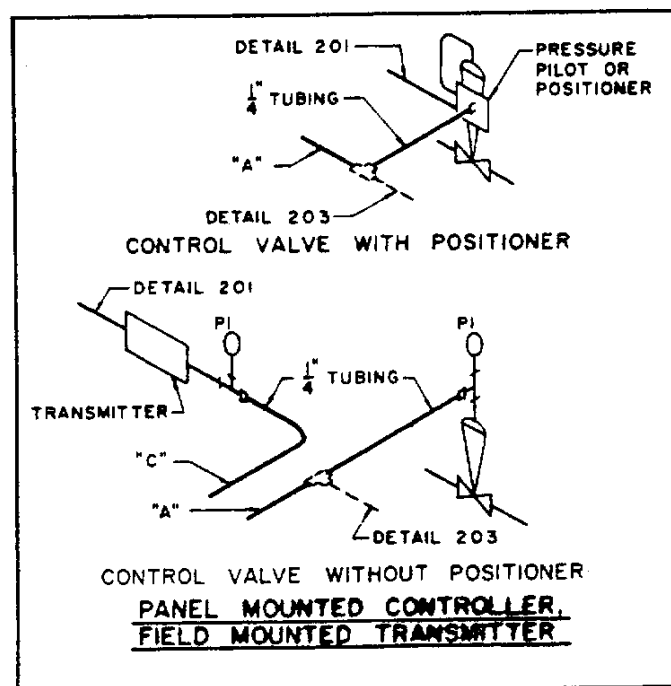


Figure 50 - "Two-Pipe" Installation

great advantage to be gained by mounting the controller in the field. The response of a level control loop is also relatively slow, and - as we will see later - in general the level is only loosely controlled, so the level controller also can be board mounted. (It should be noted that the pneumatic level controllers that UOP does mount on the control board are usually cascade master controllers whose output is the set point to flow controllers; in these cases, the cascade switching must be located on the control board anyway, so there is no advantage to locating the level controllers in the field.)

It is not exactly accurate to classify the pneumatic tubing example as a single time constant, First Order plus Dead Time process. If we consider each unit length of tubing, each has a certain capacitance and a certain resistance to flow, so each has its own time constant. The tubing can be considered to be a long series of single time constant processes, each with a much shorter time constant than seven seconds. To measure and analyze a complex system with many time constants becomes very difficult, and in any case, generally one or two of the time constants dominate, so an approximation is made, and the system is assumed to be either First Order plus Dead Time, or Second Order plus Dead Time. To define a First Order System, we need only determine the single time constant. To define a Second Order System, we need to determine two time constants plus a damping factor. If the damping factor is less than one, the process tends to oscillate.

Usually for the processes we are considering, oscillation will occur only if the loop is closed.; the damping factor for the open loop is greater than one (the system is over-damped) and the transient response is similar to that in Figure 47. The difference is that with a Second Order System, the response curve is distorted into a loose S-shape; the maximum rate of change does not occur at the beginning, as it does for the First Order System. The frequency response of the Second Order System is simply that of two First Order Systems added together.

We will not look at any of these systems in detail. We will take a qualitative look at how a system reacts when various control modes are used, and we will use the First

and Second Order Lag plus Dead Time approximations to do so. But we will not look for quantitative response and stability measurements.

Digital Transmission

Transmission methods for digital systems involve converting the analog input to a binary electrical signal of logic 1's and 0's, and sending it down a data highway to the control room. Since there is only one operating highway, it must be shared among all the inputs. The sharing is accomplished by scanning each input in turn and transmitting its value to the control room. If the process variable is oscillating in a sine wave, and the scanning rate is high compared with the frequency of oscillation, the resulting digital numbers yield a fairly representative picture of the process variable, as shown in Figure 51. As the oscillation frequency of the process variable increases and approaches the scanning frequency, the digital numbers give an increasingly distorted view of the process variable. At certain multiples of the scanning frequency, a phenomenon known as aliasing occurs, in which the signal reconstructed from the digital numbers appears to have a frequency quite different from the actual frequency.

Early digital distributed control systems used a scanning frequency of two hertz. Some people still feel that a scanning frequency equal to twice the oscillation frequency is all that is required; however, as the Figure illustrates, if the scanning frequency is exactly twice the oscillation frequency, and if the scan occurs exactly as the variable crosses the 0° and 180° points in the cycle, the numbers that are decoded will show no oscillation at all, regardless of the actual amplitude of the variable. It would seem, therefore, that to obtain the desired frequency response of one hertz with a digital system, the scanning frequency must be greater than two hertz, not equal to it. Most of the newer digital distributed control systems use scanning frequencies that can be run at 5 or 10 hertz. If a digital system is proposed for a project, the vendor should be able to demonstrate that the system meets or exceeds a one hertz frequency response requirement.

The UOP standard of one hertz is based on experience. In general, the response of the

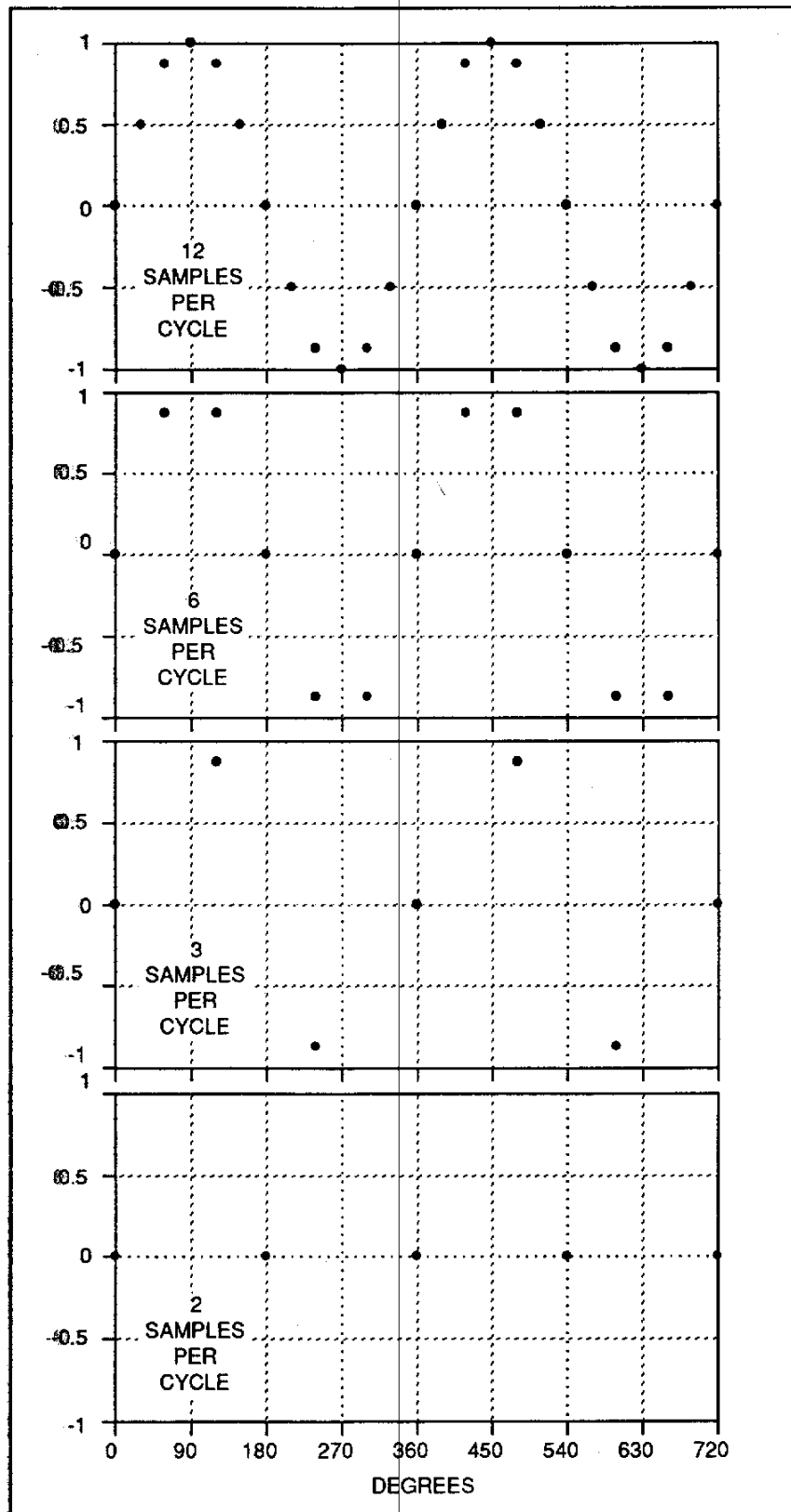


Figure 51 - Digital Sampling

process and of the final control element will be slower than one hertz, so the transmission and control will not be the limiting factors in the loop. For flow loops, however, the response is quite rapid, so the UOP requirement is basically set by flow services. UOP does not recommend the practice of using different scanning frequencies for different services. If it becomes necessary to change the control scheme and reconfigure the system, it is possible that the slot that was configured for a slow temperature service might be needed for a fast flow loop, and with some systems, doing this may be difficult or impossible.

One hertz is, of course, a conveniently round number. For practical purposes, it could just as well be 0.7 or 1.3 hertz. We should note, however, that frequency response that is faster than one hertz is not necessarily better, at least for general process control. Systems with corner frequencies above about 10 hertz are in danger of interference from 50-60 Hz noise that can be picked up from AC power circuits. If the corner frequency is one hertz, the gain at 50 hertz will be down to 0.02, not likely to cause a problem.

Control Modes

The controller in the diagram in Figure 2 consists of a Comparator, which compares the Process Variable with the Set Point and produces an Error Signal, and a Function Generator which operates on the error to produce the Output Signal that manipulates the Final Control Element. The function that is generated is determined by the control mode.

There are three basic control modes that are used at UOP. These modes are Proportional, Integral, and Derivative, often referred to as "PID" control. Other modes are available, especially with digital systems, but they are used less frequently.

The simplest of these three modes is Proportional. For a Proportional Only controller, the output is described by the equation:

$$O_c = G e + B \quad \text{Eq.(7)}$$

where

O_c = Controller Output

G = Gain

e = Error

B = Bias (or Manual Reset)

Also,

$$e = \text{Error} = (PV - SP) \quad \text{Eq.(8)}$$

where

PV = Process Variable

SP = Set Point

The Output, Error, and Bias are measured in terms of Percent of Full Scale. G is the proportional constant, and in terms of Gain, it is dimensionless:

$$\text{Gain} = \Delta \text{Output} / \Delta \text{Input} \quad \text{Eq.(9)}$$

G may also be measured in terms of Proportional Band:

$$\text{PB\%} = (\Delta \text{Input} / \Delta \text{Output}) 100\% \quad \text{Eq. (10)}$$

The Proportional Band is the percentage of full scale that the input must change to cause an output change of 100%. If the transmitter output changes by 50%, and the controller output changes by a full 100%, the controller has a Proportional Band of 50%, or a Gain of 2. Controllers in the control room these days are more often electronic analog or digital, and their Proportional mode adjustments are generally calibrated in terms of Gain; local controllers in the field are more often pneumatic, and their adjustments are generally calibrated in terms of Proportional Band. Obviously, when tuning controllers, it is important to keep the distinction between Gain and Proportional Band in mind, since they are reciprocal functions. A Proportional Band that is smaller (numerically) is actually a larger Gain, and making a move in the wrong

direction could cause the loop to go unstable.

By combining Equations (9) and (10), we find:

$$G = 100 (1/PB) \quad \text{Eq. (11)}$$

At UOP, the Proportional Only control mode is used for most level controls; Figure 52 illustrates its operation. Let us assume the valve in the Figure has a linear installed characteristic and a Cv of 100. If the controller has a Proportional Band of 100%, with the set point set at 50%, and the Bias adjusted to 50%, the controller will operate on the solid line labeled "100% PB". With these settings, when the level is at 0%, from Equations (7), (8), and (11), we have

$$O_c = 100 (1/PB)(PV - SP) + B \quad \text{Eq.(12)}$$

$$= 100 (1/100)(0 - 50) + 50 \quad \text{Eq.(13)}$$

$$= (0 - 50) + 50 = 0 \quad \text{Eq.(14)}$$

Similarly, when the level is at 100%,

$$O_c = 100 (1/PB)(PV - SP) + B \quad \text{Eq.(12)}$$

$$= 100 (1/100)(100 - 50) + 50 \quad \text{Eq.(15)}$$

$$= (100 - 50) + 50 = 100 \quad \text{Eq.(16)}$$

With these Proportional Band and Bias settings on the controller, and the Set Point at 50%, the output follows the level exactly. If we now start a flow of 30 gpm of water into the tank, and allow the level to stabilize, the level will line out at 30%, with the inlet flow at 30 gpm and the outlet flow at 30 gpm.

Once the system has stabilized, two types of changes can occur to upset the balance. One type is under the control of the operator; this is a change in the controller Set Point. The other type is not under the control of the operator; this is an external disturbance, or Load change.

Anything that affects the inlet flow, and anything (other than the controller) that

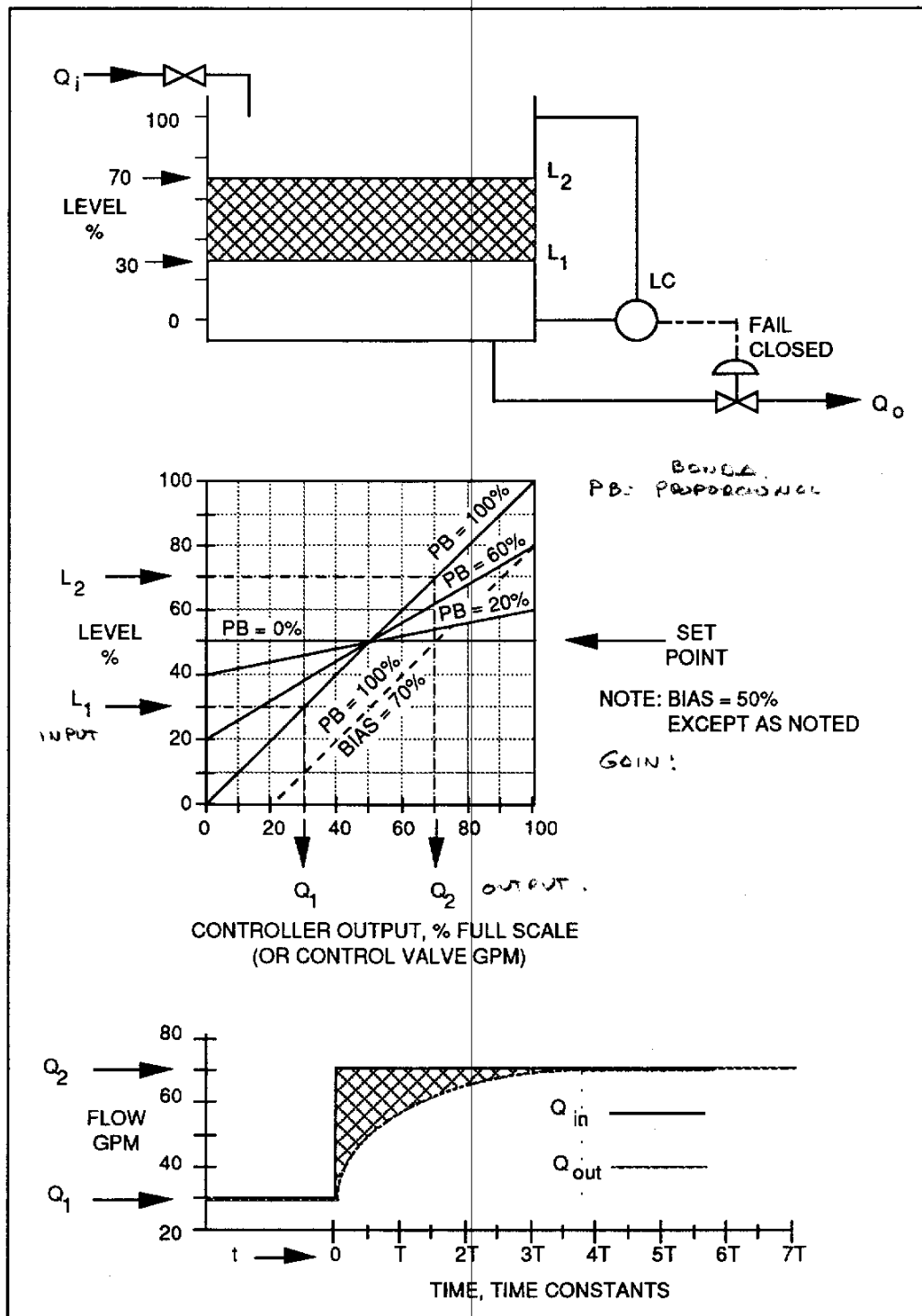


Figure 52 - Level Controller

affects the outlet flow, can be considered a Load change. If something happens upstream of the inlet valve, or downstream of the outlet valve, to change the valve pressure drop and therefore the flow, a Load change has occurred. The tank in Figure 52 is open to the atmosphere, but if the pressure in the tank could change, it would affect the pressure drop of both the inlet and the outlet valves; that, too, would be a Load change, but of a different form. The response of the system to each type of Load change - or to the Set Point change - might not be the same, but each Load change does have an affect on the system.

Let us now assume that a Load change occurs which causes an instantaneous "step" change in inlet flow, from the initial 30 gpm to a new value of 70 gpm. The level is at 30%, so the controller holds the valve lift at 30%, and holds the outlet flow at 30 gpm. We now have 70 gpm flowing into the tank, and 30 gpm flowing out. The level must begin to rise.

The level rises rapidly, at first, but as the level rises, the controller output increases and the outlet valve opens, increasing the outlet flow. As the outlet flow increases, the difference between the inlet and outlet flows decreases, and the level rises more slowly. When the inlet and outlet flows are equal again, the level must stop rising. So with the inlet flow at 70 gpm, and with the controller Proportional Band at 100%, the Set Point at 50%, and the Bias at 50%, the level must rise to 70% to obtain 70 gpm at the outlet.

The steady state Error, the deviation between the Set Point and the lined-out value of the Process Variable is known as the "Offset". In the example above, with a flow rate of 70 gpm, the Offset is equal to 20%, the difference between 70% and 50%. If we reduce the proportional band from 100% to 60%, (increase the Gain to 1.67), the controller output changes from 0% to 100% for a level change from 20% to 80%. Using these values in Equation (12),

$$70 = 100 (1/60)(PV - 50) + 50 \quad \text{Eq. (17)}$$

$$20 = 1.67 PV - 83.3 \quad \text{Eq. (18)}$$

$$PV = 62 \quad \text{Eq. (19)}$$

The Offset has been decreased from 20% to 12%. Reducing the Proportional Band to 20% (increasing the Gain to 5) and applying the same analysis results in a PV of 54%, or an Offset of 4%

Theoretically, we could continue to reduce the Proportional Band until we reached 0% (Gain = ∞) and thereby decrease the Offset to zero. A Proportional Band of 0% is the same as "On-Off" control. Unfortunately, long before we get to 0% Proportional Band, the system will start to oscillate and go unstable. (We will discuss the subject of stability in more detail later.)

We can rewrite Equation (12) to solve for the Offset:

$$\text{Offset} = (PV - SP) = (PB/100) (O_c - B) \quad \text{Eq. (20)}$$

For a given value of O_c , the equation indicates that the Offset can be reduced in two ways: one way is by decreasing the Proportional Band, as we have seen; the second way is by changing the Bias. We can see from Equation (20) that if we set the Offset to zero, the Equation is satisfied if B is equal to O_c . So even if the system requires an outlet flow of 70 gpm, it can be obtained with both the Set Point and Process Variable at 50% by adjusting the Bias to 70%. The problem with this approach, as the Figure indicates, is that even with the level at 0%, the valve will still be 20% open.

For a level controller, offset is not always detrimental. In fact, within limits, it can have certain benefits. With the Proportional Band set at 100% and the Set Point and Bias set at 50%, we saw that when the inlet flow to the tank changed from 30% to 70%, the level itself had to change from 30% to 70%, in order to change the control valve position and the outlet flow. The level changed by accumulating the difference between the inlet and the outlet flow rates. At $t = 0$, when the inlet flow changes from

Q_{i1} to Q_{i2} , the outlet flow, Q_o , has not yet changed, so as indicated earlier, the level increases rapidly. As the level increases, the outlet valve opens, the difference between the inlet and outlet flow rates decreases, and the rate of increase of the level decreases. The shape of the curve of Q_o versus time (and likewise the shape of the curve of level versus time) is similar to the curve of pressure versus time for the pneumatic tubing example we saw earlier, except that the curve of Q_o has no Dead Time. Since the curves are similar, we can see that this system also is a First Order Lag.

The integral of the difference between the inlet and outlet flow curves is equal to the integral of the change in level in the tank. These two areas are indicated in the Figure. The benefit gained from this integration is that the change in the outlet flow is much smoother and more gradual than the change in the inlet flow, because part of the flow change is absorbed in changing the level. If the outlet flow from the tank is the feed to a fractionation column, it is easy to see that the upset to the column will be less if the flow change is smoother. If the outlet flow from that column feeds a second column, using Proportional Only control also for the level controller on the first column will reduce the upset to the second column even more.

As indicated earlier in the discussion on Equation (20), the Offset can be reduced by using a narrower Proportional Band. However, if this is done, the change in level in the tank becomes smaller, and less tank volume is available to absorb and smooth out fluctuations in flow. The Figure shows that result. As the Proportional Band decreases, the shape of the outlet flow versus time comes closer and closer to the shape of the inlet flow change. If the Proportional Band could be set to zero, the Offset would be zero, and the outlet flow would follow the inlet flow exactly.

If our aim were to minimize the Offset and control the level at the Set Point, we could almost always use the level transmitter with the minimum range; a 14" displacer would be sufficient. If we use a 48" displacer, and set a narrow Proportional Band to

minimize the Offset, we are wasting the money spent on the extra length for the displacer and the vessel. If the Proportional Band is set so the control valve is just closed when the level is at 0% and is just open when the level is at 100%, the level is not out of control, and the maximum smoothing of the outlet flow is obtained. To allow for possible zero drift and changes in specific gravity, it is better to set the Proportional Band somewhat narrower, at about 90%. The Set Point should be set at 50%, and the Bias adjusted to give an Output of 50% when the level is at 50%. If this procedure is followed, it should not be necessary to make any further adjustments during normal operation. Except for level instruments in cascade loops, and loops where the level must be maintained at the set point, UOP prefers to mount the level controller locally, in the field, where it is out of the view, and out of the reach, of the control board operator.

It should be noted that with a Proportional Band of 100% on the level controller, a percentage characteristic for the control valve would require a level of approximately 85% to obtain the normal 50% of maximum flow. Since UOP usually uses a setting of 80% for the high level alarm, the alarm would be on during normal operation. Also, at levels of about 60% and higher, the flow changes would be greater for the same change in level than for a valve with a linear characteristic. This would reduce the ability of the level to absorb the changes and smooth out the flow. When possible, the valve used for this type of service should have linear trim.

There are occasions when we do wish to control the level, not just limit it, and where offset is not desirable. In a steam generator, for instance, a change in level changes the head on the tubes, and therefore changes the pressure and temperature of the steam, and the temperature of the process fluid. Low levels also could allow tubes to become uncovered, while high levels could increase water carryover into the steam system. In this case, it is desirable to control the level at the set point, with minimum offset.

With the Proportional Only level controller, we found earlier that we could eliminate the Offset by adjusting the Bias. The Bias is also known as Manual Reset, because it does manually what the Reset action of the Proportional plus Reset controller does automatically. The Reset action moves the controller Output continuously in the direction that minimizes the Offset. Reset action is often measured in "Repeats per Minute". A setting of one Repeat per Minute means that in each one-minute interval, the Reset action will cause a change in Output that is equal to the change in Output that results from the Proportional action alone. The result is easiest to see if we look at the open loop transient response. "Open Loop" means that the feedback loop is open, generally between the Controller and the Final Control Element. Effectively, this is what happens when a control valve sticks and refuses to move.

Figure 53 shows the relationship between Controller Output and Time. The Figure assumes that the Set Point, the Process Variable, and the Controller Output are all at the baseline until Time $t = 0$. At $t = 0$, a Load change occurs and the Process Variable increases by one unit. The Proportional action changes the output immediately, by a number of units that depends on the Proportional Band setting. With no Reset action, the Output would remain constant at this new value. With the Reset action, an additional linear change in Output occurs, beginning at $t = 0$, and continuing thereafter to the physical limit of the Controller Output. Since the Proportional action has changed the Output by one unit, if the Reset contribution changes the Output by one additional unit in one minute, the Reset is one Repeat per minute—that is, after one minute, the Reset action has repeated the Output change made by the Proportional action. If the Reset changes the Output by two additional units in one minute, the Reset is two Repeats per Minute, and so on.

The Reset contribution in this example is linear, increasing with time; it is therefore a simple Integral of time. Actually, the Reset contribution is the time integral of the Error function of the Controller, and because of this, Reset action is also known as the Integral mode, the "I" in "PID". Mathematically, the Output of the Proportional plus

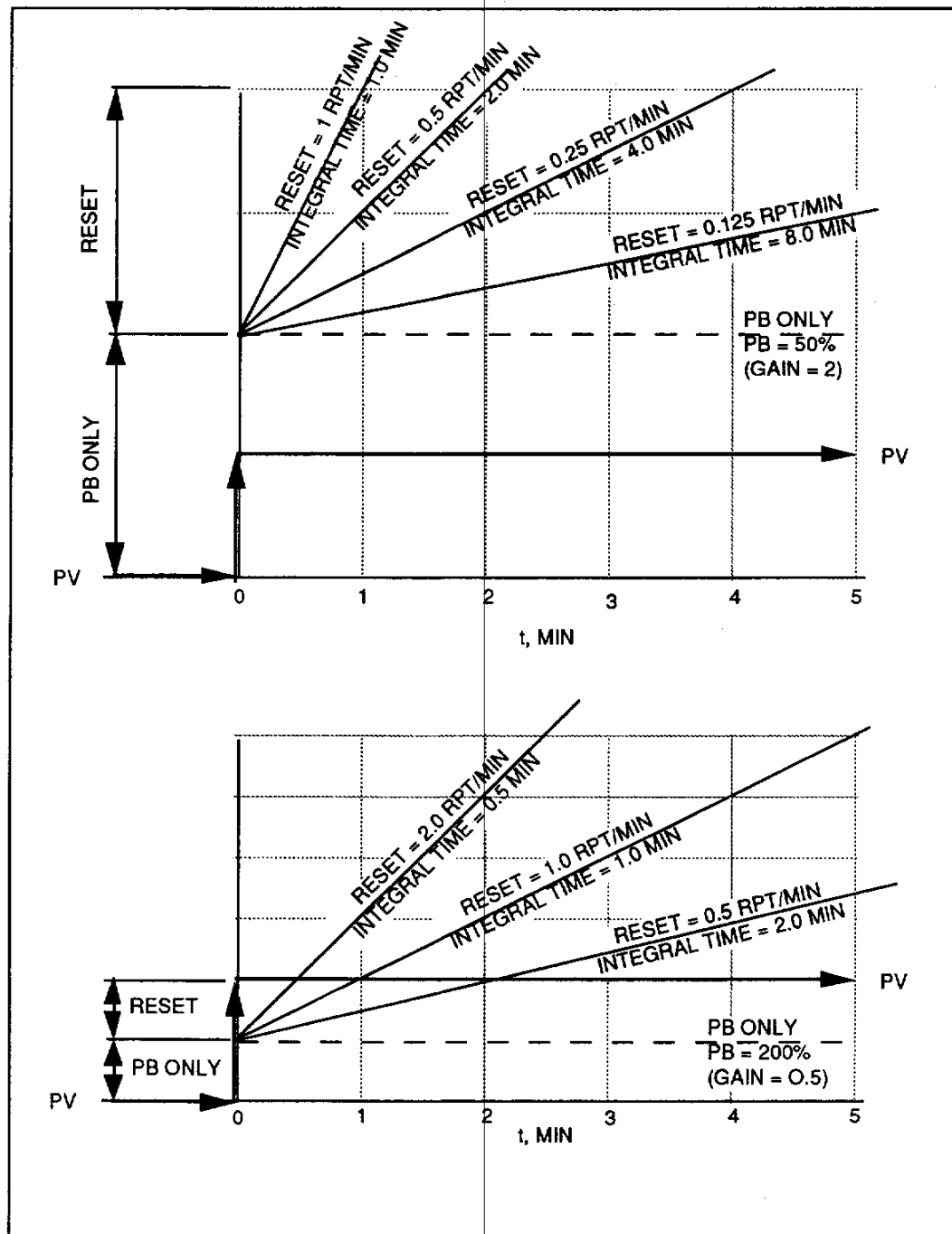


Figure 53 - Open Loop Reset Response

Integral (or Proportional plus Reset) controller can be expressed as

$$O_c = G [e + (1/T_i) \int_0^t e \, dt] \quad \text{Eq.(21)}$$

where

$$G = \text{Gain} = (100/\text{PB}) \quad \text{Eq. (11)}$$

$$e = \text{Error} = (\text{PV} - \text{SP}) \quad \text{Eq. (8)}$$

$$T_i = \text{Integral Time, Minutes per Repeat}$$

Just as Gain is the reciprocal counterpart of Proportional Band, Integral Time in Minutes per Repeat is the reciprocal counterpart of Reset in Repeats per Minute. And, like the Proportional Mode, the same caution to use care when making adjustments applies. Electronic controllers in the control room are almost always calibrated in terms of Integral Time, Minutes per Repeat; pneumatic field mounted controllers may be calibrated either in terms of Minutes per Repeat or Repeats per minute.

If we close the loop on the controller used in the example above, at $t = 0$ the Process Variable will start to change as it did in the example, but as the Offset (error) is maintained, the integral term in Equation (21) will continue to change the Output. As the Output changes, the Process Variable will move back toward the Set Point. When it reaches the Set Point, the Error becomes zero, and the integration stops. The system is now in balance again, with a new value for the Integral term in Equation (21), corresponding to a new value for the Bias, B , in Equation (7). The speed with which the Process Variable returns to the Set Point depends on the Integral Time setting. Directionally, decreasing the Integral Time decreases the time required for the Process Variable to return to the Set Point. However, there are limits set by the stability of the system; these limits will be discussed later.

As we noted in the level controller discussion, the shapes of the curves of level versus time and controller output versus time are similar to the shape of the pressure versus time curve for the pneumatic tubing. Therefore the frequency response plot will also be similar. The values for Gain and Time Constant may be different, but the overall shape will be the same. Ideally, the frequency response plot of a Proportional Only controller is a straight line at constant gain, with no phase shift. Practically speaking, a real controller will have a loss in gain at the higher frequencies because of physical limitations - in fact, as mentioned, this loss has some benefit in that it reduces interfer-

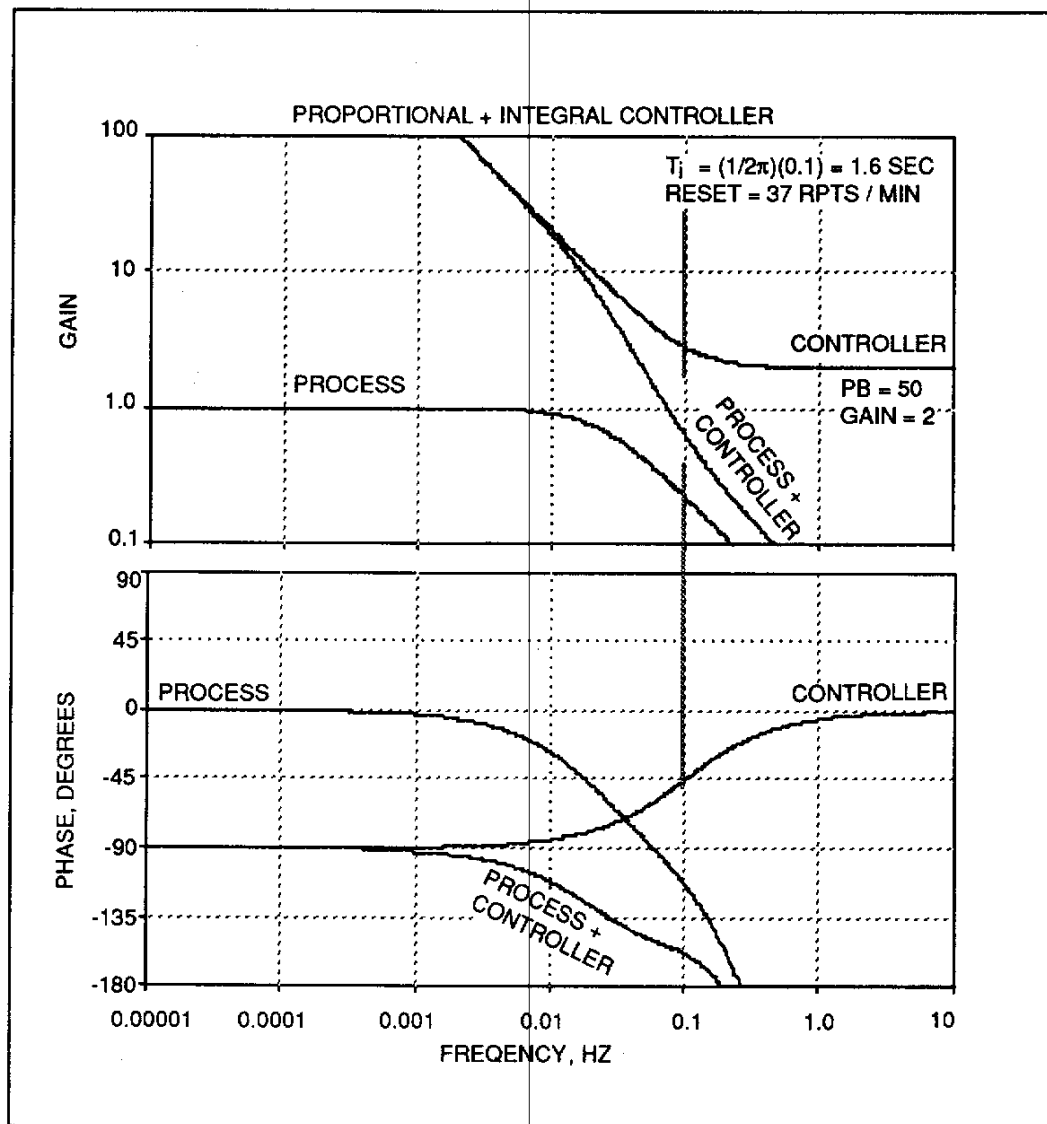


Figure 54 - Proportional + Integral Control

ence from AC electrical power frequencies. Also, because of physical limitations, there will be some phase lag; but this should occur above one hertz, and should not affect normal operation.

The frequency response plot for a Proportional plus Integral controller is shown in Figure 54. The corner frequency corresponds to the Integral Time set on the controller. The second plot repeats the plot of the controller, and also shows the frequency response plot of a First Order, single time constant process similar to the level control

system discussed earlier, but with a Dead Time component similar to that of the pneumatic tubing discussion. As indicated earlier, the combined open loop response of the process and the controller can be found by graphically adding the two individual plots together, because addition on the log scale is equivalent to multiplication on a linear scale. When the combined open loop response is plotted, the stability of the loop can be checked. If the gain is greater than one when the phase shift is -180° , the system will be unstable.

In order to function at all, a feedback control loop must have negative feedback, a 180° phase shift somewhere in the loop. If a conventional flow control loop has a control valve that closes on air failure, the controller must be reverse-acting. If something happens that causes the flow to increase, the flow transmitter output to the controller also increases. Increasing the controller output would cause the valve to open and the flow to increase even more. Therefore the controller itself must provide a 180° phase shift in the loop; when the input is increasing, the output is decreasing, and vice versa. This phase shift takes place in the comparator section of the controller. Conversely, if the flow loop has a control valve that opens on air failure, the controller must be direct-acting. An increase in flow must cause an increase in output from the controller. In this case, the 180° phase shift is in the control valve, rather than in the Controller. Nevertheless, in either case, there is a 180° phase shift somewhere in the loop.

This phase shift is present at all frequencies, even the very low. If, as the frequency increases, an additional lag that reaches 180° is introduced, the total phase shift becomes 360° . The 360° phase shift means that each cycle is exactly in phase with the cycle that precedes it. If the gain of the loop is greater than one when this happens, each oscillation will be larger than the preceding one, and the system will be out of control.

One way to stabilize the system is to reduce the gain. The system will still be

oscillatory, but if the gain is less than one, the oscillations will grow smaller instead of larger. Sometimes, depending on the system, it might also be beneficial to increase the Integral time, T_i , (decrease the Repeats per Minute), and thereby decrease the corner frequency of the Integral mode contribution. If the Integral time is infinite, we have, in effect, a Proportional Only controller.

It is easy to see from this analysis how extremely detrimental Dead Time can be to system performance. As we noted earlier, a Dead Time of one second, by itself, causes a phase shift of -360° at a frequency of one hertz, and -180° at 0.5 hertz. Even if the process has a very short time constant, some phase shift will be present at 0.5 hertz, and a gain greater than one at 0.5 hertz will make the system unstable.

If we say that a process has a "fast response", we really mean that it has little lag, that it has a short Time Constant. The amount of lag in a process can be associated with its ability to store energy. In the pneumatic tubing example, longer lengths of tubing added more capacity for storing energy in the form of compressed air, and increased the lag. Liquid flow is a fast process; the liquid is not compressible, so except for a very small amount in the spring diaphragm control valve actuator, there is little energy storage in the system. A change in valve position is reflected almost instantaneously as a change in process variable. In addition, small disturbances in the system show up as noise on the chart record. What is needed for this is a control mode that will minimize the gain in the high frequency noise region, and maximize the gain at the lower frequencies to reduce the offset.

The Proportional plus Integral controller does just that. Setting a low gain, wide Proportional Band on the controller minimizes amplification of the noise frequencies (it actually reduces their amplitude, since the gain is less than one), and setting the corner frequency of the Integral mode below the noise region provides maximum gain at the lower frequencies. Typical controller settings for a liquid flow loop might be a Gain of 0.5 to 0.7 and an Integral Time of 0.05 Minutes per Repeat (Proportional Band

of 150 to 200% and Reset of 20 Repeats per Minute).

In the case of gas flow loops, the fluid is compressible; but the length of piping between the valve and flowmeter is usually short, and the storage capacity is still fairly low, so similar controller settings would apply. With liquid pressure loops, the noise is not prominent, but, as with the liquid flow loop, the storage capacity is small and therefore the controller settings would be similar to those for liquid flow.

In the case of gas pressure loops, the compressibility becomes significant. The system now includes vessels as well as piping, and the capacity for energy storage is therefore greater. As with liquid pressure loops, the noise is not noticeable; therefore a higher gain can be used than is used on flow loops. However, because the energy storage is greater, more Integral Time is required. Typical settings would be a Gain of 1.5 to 2 and Integral Time of 0.2 to 1.0 minutes (Proportional Band of 50 to 75% and Reset of 1 to 5 Repeats per Minute).

The systems with the greatest amount of energy storage, and therefore the most lag, are temperature loops. Flows and pressures are essentially single time constant, First Order Lags. Temperature loops generally have at least two time constants between the manipulated variable and the process variable. To increase the heat input to a process heater or heat exchanger, the heat must first pass through a film with a transfer coefficient (resistance) on one side of the tube, then through the tube wall (capacitance), through another film (resistance), and into the process fluid (capacitance). There are also a transport lag (dead time) before the process fluid can reach the temperature measuring point and the change in temperature can be detected, and resistances and capacitances in the thermowell and thermocouple.

Actually, the films have capacitance as well as resistance, and the tube wall has resistance as well as capacitance, but overall, the resistances and capacitances generally combine to produce two time constants that dominate all others, and a Second

Order Lag plus Dead Time approximation can be used to evaluate the system.

The lags connected with temperature loops make control more difficult. Since it takes longer for a change in control valve position to have an effect on the temperature, and for that effect to cause a change in the input signal, a way must be found to offset these delays; if the temperature begins to change, the controller must respond immediately. If the controller can be set with high enough gain, the output can change faster than the input indication, and, in effect, catch up to the projected actual input. Part of this high gain is derived from the use of a measurement range with a suppressed zero, and a span (normally, unless there are other considerations) of about 200 °F. For instance, if the operating temperature is 700 °F, the temperature transmitter is specified with a range of 600 to 800 °F, rather than 0 to 1000, as would be used for a pressure loop.

But gain alone is not the entire answer; with dead time present, the phase lag still will cross -180° at some frequency, and the gain must be less than one at that point. To be effective and "catch up" with the process change, the controller needs a high gain at the higher frequencies, where the changes occur the fastest, but must have minimum gain at a phase lag of -180°. To obtain this, Derivative, or Rate action is used. This is the "D" in "PID".

The Derivative contribution is proportional to the rate of change of the error signal:

$$O_c = G [e + (1/T_i) \int_0^t e \, dt + T_d (de/dt)] \quad \text{Eq.(22)}$$

where

$$G = \text{Gain} = (100/PB) \quad \text{Eq.(11)}$$

$$e = \text{Error} = (PV - SP) \quad \text{Eq.(8)}$$

$$T_i = \text{Integral Time, Minutes per Repeat}$$

$$T_d = \text{Derivative Time, Minutes}$$

If the error changes slowly, there is little contribution from the derivative action; but if the change is slow, the effect of the process lag is comparatively unimportant, and less

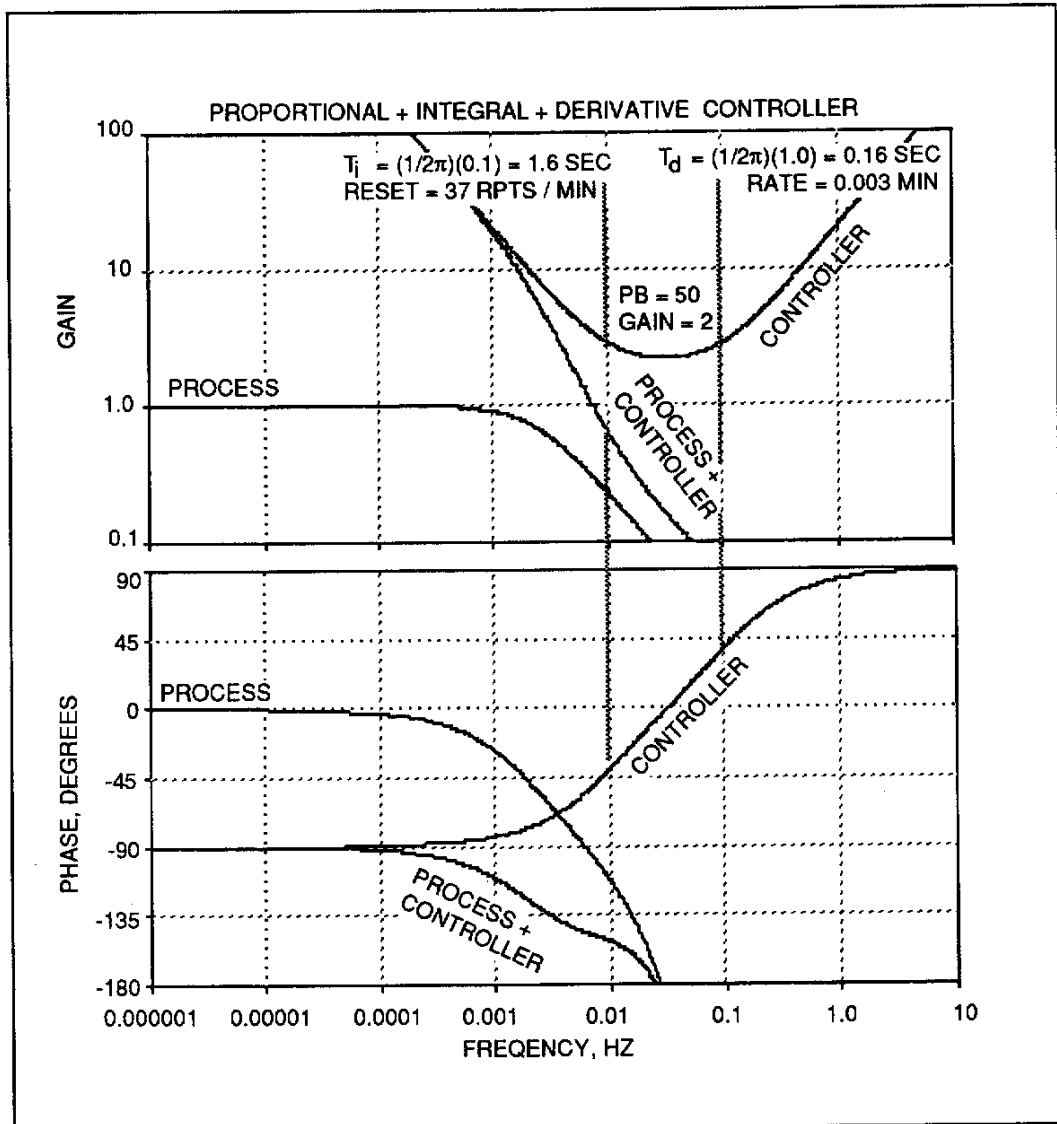


Figure 55 - Proportional + Integral + Derivative Control

contribution from the derivative action is necessary. If the error changes rapidly, the derivative contribution is large, as it should be, but in the closed loop it decreases as the corrective action takes effect and slows down the change in error signal.

The frequency response plot for the PID controller, with the same First Order Lag plus Dead Time system as shown in Figure 54, is shown in Figure 55. The Integral action increases the gain at low frequencies, and the Derivative action increases the gain at high frequencies. In addition the phase lead that results from the Derivative action

helps to counteract the system lag.

The graphical summation of the two frequency response plots is also shown. In effect, the Derivative action cancels out the effect of the shorter time constant and converts the Second Order system into a First Order system. Typical settings for a temperature controller might be a Gain of 5 to 10, Integral Time of 10 to 20 Minutes per Repeat, and Derivative Time of 3 Minutes (Proportional Band of 10 to 12%, Reset Rate of 0.05 to 0.1 Repeats per Minute, and Rate Time of 3 Minutes).

Temperature loops make up the majority of applications for PID controllers, but there are a few others. If the bottoms material from a column has a fairly wide boiling range, with sufficient slope, as shown in Figure 56, there is a well-defined relationship between temperature and percent vaporized. In this case, controlling a constant temperature will maintain a constant percent vaporization. If the bottoms flow to the reboiler heater is controlled at a constant rate by an FRC, a TRC on the outlet, controlling the firing of the heater, controls the percent vaporization, and thereby the heat

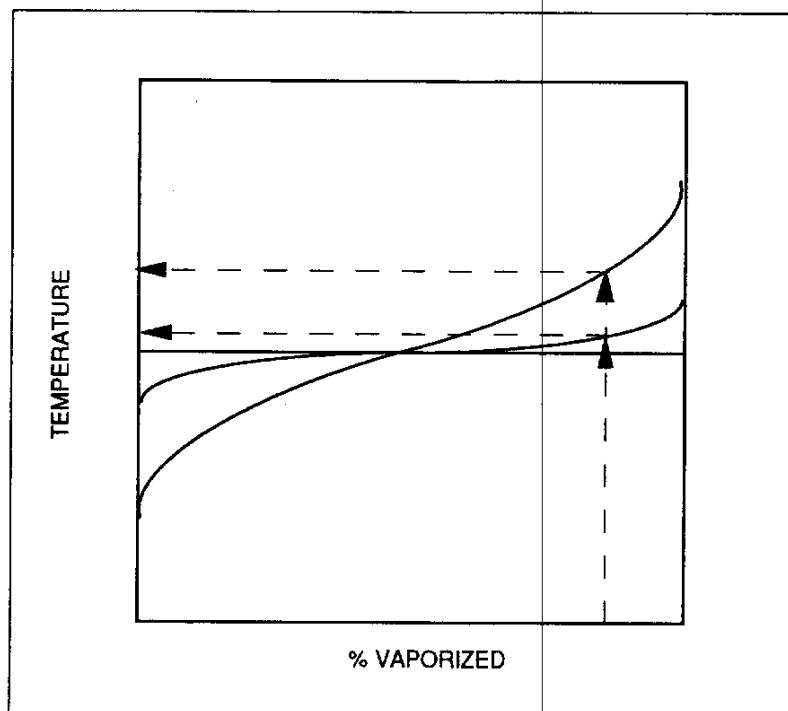


Figure 56 - Boiling Curve

input to the column. However, if the bottoms material is essentially a pure component, the difference in temperature between 0% vaporization and 100% vaporization may be only a few degrees, too narrow a range for reliable control of vaporization by means of temperature.

When the temperature rise across the heater is less than 10 to 20 °F, a PDRCAL controller (Figure 57) is used instead, to measure the differential pressure developed across an eccentric orifice plate. For a given mass flow rate through an orifice, the velocity is higher and the differential pressure is greater for a vapor than it is for a liquid. So for the same flow rate to the heater, a higher percentage of vaporization will result in a higher differential pressure at the PDRCAL.

Thus the PDRCAL controls the heater firing, much like the TRC; it must contend with many of the same lags in the system that the TRC must contend with, and therefore it also benefits from the use of Derivative action. The Derivative mode in this application does have one disadvantage, in that the PDRCAL measurement is, in effect, a flow measurement, and it exhibits the same high frequency noise. The transmitter used for this service must be specified with internally adjustable damping, which will filter out this flow noise. Otherwise, the Derivative mode will not be able to use its corrective action on the process lag without also amplifying the noise.

In a somewhat similar application, a flow controller on the overhead vapor line from a

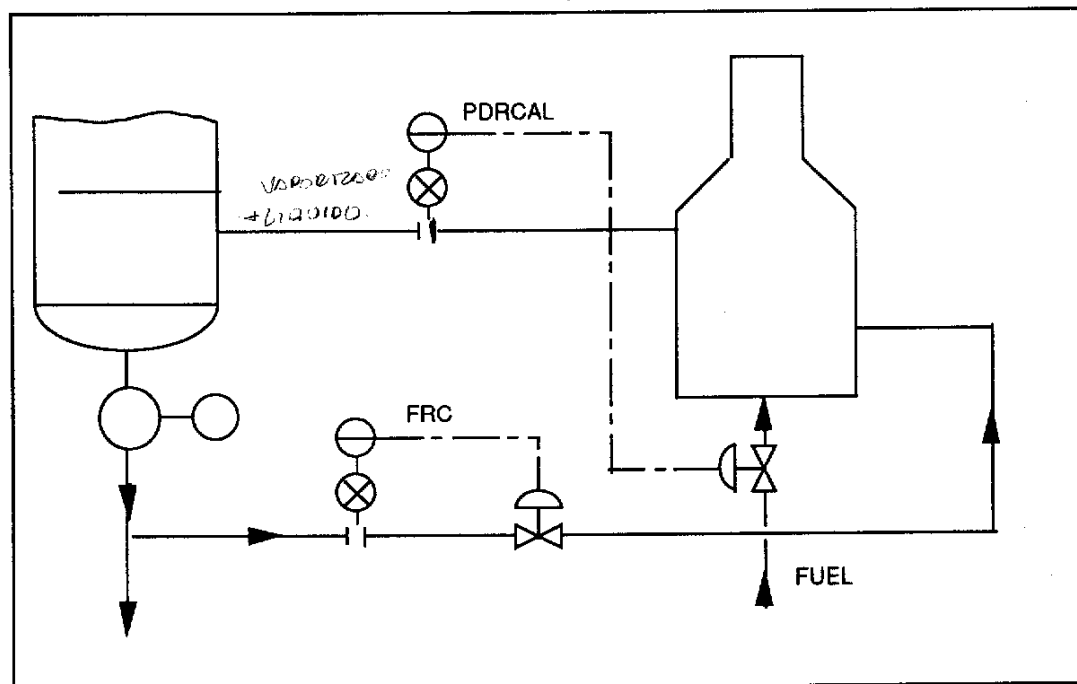


Figure 57 - Fired Reboiler PDRCAL Control

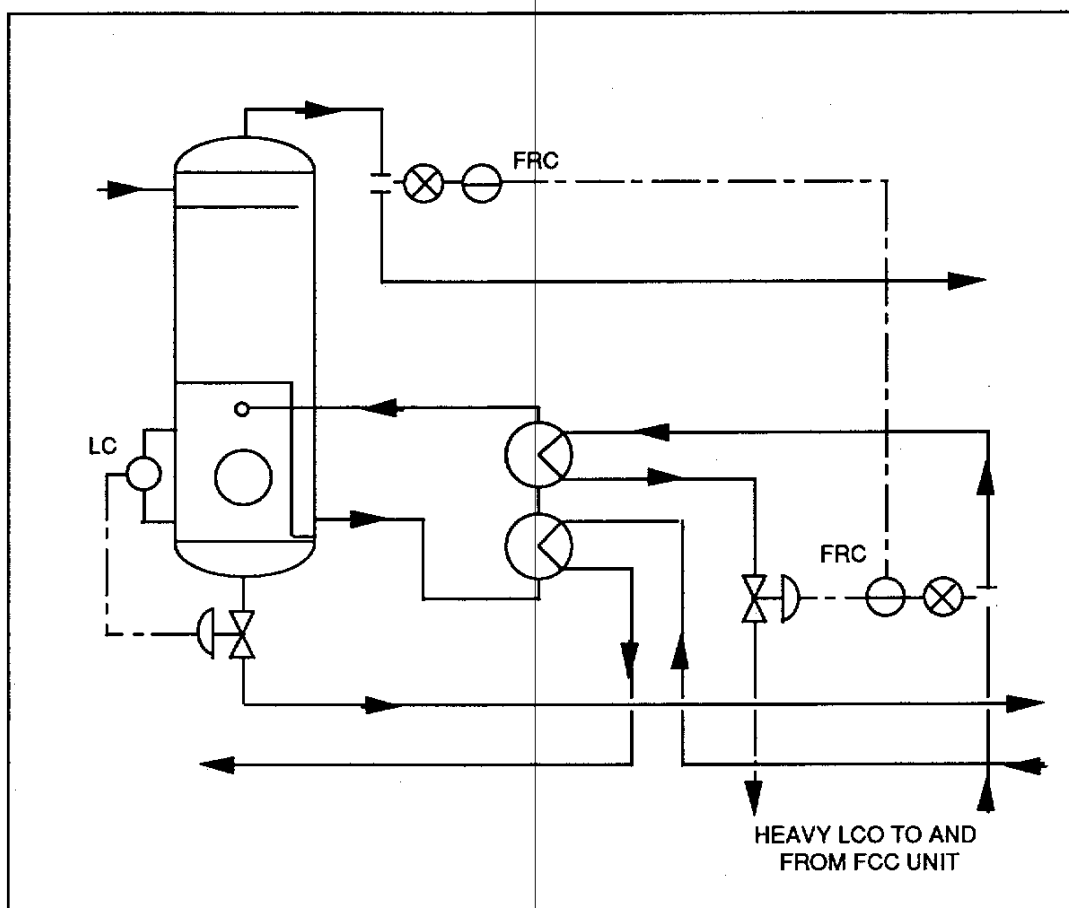


Figure 58 - Gas Concentration Unit Stripper

stripping column (Figure 58) is used to control the stripping rate by adjusting the flow of heating medium to the reboiler. Again, the thermal lags involved in getting the heat into the reboiler, and the transport lag in getting the material up the column, make Derivative action useful. Also, again, the flow transmitter must be well damped to eliminate the high frequency noise.

Still another application for PID control is a Regenerator pressure control loop on some Fluid Catalytic Cracking units which have power recovery turbines on the flue gas. On new plants, when a power recovery turbine is used, it is used in conjunction with a motor/generator in the drive train of the main air blower. The inertia of the drive train allows several seconds for closure of the flue gas valve to the expander in

case of a shutdown. On existing units, the plant layout generally does not permit coupling the expander to the drive train; instead, it is coupled only to a generator. In this case, if the electrical load is lost, the flue gas valve must close quickly (1/2 second) to prevent overspeeding the expander. The Regenerator pressure normally is controlled by bypassing the flue gas around the expander. If the system shuts down, the bypass valve must react quickly to minimize any upset in the Regenerator pressure which could cause a reversal. To give the pressure controller the needed extra response, Derivative action is used.

Controller Tuning

Frequency response plots are useful for control system studies. Given plots for each component of the loop, we can graphically add the responses together and study the effects of changes in controller settings. The obvious problem with this approach lies in getting the data. Aside from the length of time required to obtain it, no one wants his process unit continuously upset while the data gathering operation is taking place. Fortunately, there are other ways to obtain information useful for controller tuning.

Probably the simplest method is one based on the Ziegler-Nichols method. This method is an on-line, or closed-loop, method. It is based on finding the frequency of oscillation of the loop by turning off, or setting to minimum, both the Integral and De-

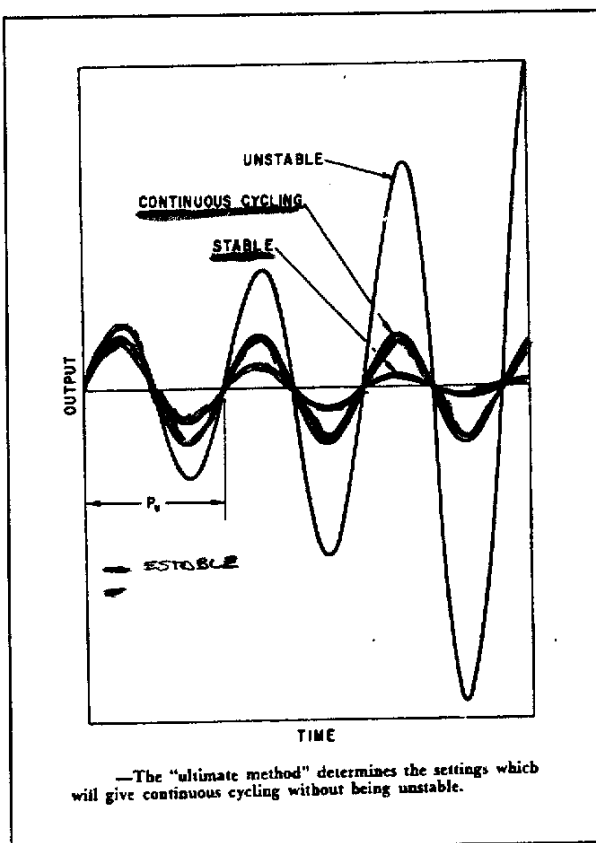


Figure 59 - Controller Tuning

rivative modes, so the controller is effectively a Proportional Only controller. The process is lined out on manual control, an initial setting is made to the gain, the controller is placed in automatic, and a small (2 or 3%) step change is made in the set point. The chart record is examined after a time, and if there is no oscillation, or if the oscillation decreases and dies out, the gain is increased slightly, and the procedure is repeated. If the oscillation increases in amplitude, the gain is too high. The goal is to find a gain setting which will just maintain an oscillation with a constant amplitude (Figure 59). The gain and the period of the oscillation are used to calculate the tuning settings for the controller:

- 1) Set the Gain at 1/2 the Gain which maintains the constant amplitude oscillation.
- 2) Set the Integral Time at about 2 times the period of the oscillation.
- 3) Set the Derivative Time at about 1/2 the period of the oscillation.

For instance, if the constant amplitude oscillation is obtained with a Gain of 10, and has a period of 5 minutes, the settings would be as follows:

- 1) Gain = $0.5 * 10 = 5.0$ (Proportional Band of 20%)
- 2) Integral = $2.0 * 5 = 10.0$ (0.1 Repeats per Minute)
- 3) Derivative = $0.5 * 5 = 2.5$ Minutes.

There are other methods which work off-line, that is, with an open loop. These methods are based on obtaining the data from a step change in controller (manual) output. The record which results is analyzed to determine time constants and dead times.

None of these methods claims to be absolutely accurate; all of them can be expected to require some additional fine tuning to obtain the best control. One reason for this is that there are different definitions of "best". Some methods aim for a "quarter decay ratio" (Figure 60), in which the amplitude of each oscillation is 1/4 of the amplitude of the previous oscillation. Others aim for 5% overshoot; still others for minimum

area under the error curve. What is "best" for one application may not be best for another, so it is up to the user to decide when the tuning is satisfactory.

Most flow and pressure loops respond fast enough to allow them to be tuned fairly well by trial and error in a short time. The tuning methods above are most likely to be useful for temperature loops, and others with long time constants and complicated responses.

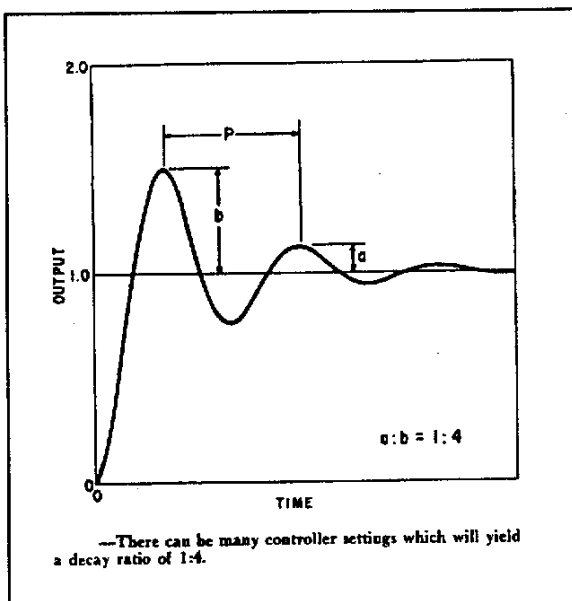


Figure 60 - Quarter-Decay Ratio

SYSTEM REQUIREMENTS

UOP specifications for control systems list several basic requirements that the systems must meet. These requirements apply to analog pneumatic, analog electronic, or digital distributed control systems. They are:

- 1) System frequency response shall be at least one cycle per second.
- 2) Loop integrity shall be such that a single failure will not cause the loss of more than one control loop.
- 3) Records shall be provided for process variables as specified.
- 4) Outputs shall be capable of operation 25% above and below the standard signal range.

The requirement for one cycle per second frequency response has been discussed. It is applied equally to all control loops so the loops can be interchanged easily if a new configuration is required. In digital systems this means that the same scanning frequency is used for all loops.

The loop integrity requirement assumes that the operator can control at least one loop on manual without much loss of product quality or risk of damage to equipment or catalyst. If the other controllers are keeping the rest of the process steady, the one manual loop should also remain fairly steady. If a number of loops are lost at once, they will be very difficult to control, especially if they interact strongly with each other.

Loop integrity requires continuous AC power for electronic and digital control systems; the capacity of the UPS (Uninterruptible Power Supply) must be sufficient to handle the startup of the largest load in the system while the rest of the system is in operation. If a common DC power supply is used for multiple loops, a backup supply with automatic cut-in is required. All components of a loop should be powered from the same source; if power to a temperature transmitter is lost while power to the controller is maintained, the controller will interpret the loss of input signal as a low temperature, and will call for maximum heat input, with possibly disastrous results.

In a digital system, loop integrity requires backup for the microprocessor. Honeywell, for instance, in their TDC systems, uses one basic microprocessor for eight loops; there is an installed backup microprocessor for every eight basic microprocessors. If a fault develops in the basic controller, the backup controller automatically takes over without loss of control.

Manual operation, as used in this context, means manual operation by the control board operator, from the front of the control board. In many electronic systems, the manual control operates through the output section of the controller, and if the controller fails, the manual operation also fails. In this case a separate standby manual station is often used to substitute for the controller. This standby manual station should be operable from the front of the control board, so the process variables still can be monitored.

Records are useful for normal operation because the operator can see changes and

trends more clearly than he can if he only looks at deviations, or columns of numbers. Records are especially useful for upsets, because they help to determine what went wrong and what should be done to avoid similar problems. Smoothed, averaged data, logged every hour, may be of interest to management, but it will not tell the operator everything he needs to know about the plant. UOP will accept hard-wired, analog recorders, or - for digital systems - a plot once each shift (or on demand by the operator) of digital data sampled every 10 seconds and stored. Sampling a loop every 10 seconds amounts to 2880 samples in an eight hour shift. Obviously, for a plant with several hundred loops, this requires considerable memory capacity, and is likely to be practical only in conjunction with a computer.

When a computer is used with a digital distributed system, UOP recommends that the computer be used to generate process graphics with a real time data on one video display, and associated 20 to 30 minute trend records on the adjacent video display. This can be done for subsystems of the plant (a heater, a fractionator, a reactor section) and will give the operator a better idea of what is happening in the unit than the customary overview display. At least three operating stations should be furnished in the console so at least two operators can work at the console in case of an upset. The third station should be equipped with a high speed alarm printer.

The requirement that the output must be capable of operation 25% above and below the standard range relates to the control valve actuator "bench set" range discussed earlier. If a control valve fails closed and has a "bench set" range of 3 to 15 psi, the actual starting point of the stroke will be somewhat less if the valve is installed flow-to-open. In addition, at the starting point of the stroke, the valve is just coming off the seat, and the seating force is zero. If the valve is to shut tightly and not leak, some seating force is required to slightly deform the seat so that all surfaces mate properly. If the valve fails open instead of closed, the diaphragm pressure required to close the valve will be greater than 15 psi, for the same reason.

This has not been a problem for pneumatic systems. Pneumatic controllers and manual

stations generally can operate from 0 psi to very near the 20 psi supply pressure. Also, there is generally some latitude in analog electronic controller outputs. The problems usually arise in digital systems. With most digital systems now available, only 7 to 10% above and below the normal range can be obtained.

The over range available is related to the number of digital steps used for the conversion from a digital number to an analog signal. UOP specifications require 1000 step resolution in the 0 to 100% signal (4 to 20 mA DC) range, and by extension, 1500 steps overall for -25 to 125%. Since the digital to analog conversion is done in a binary fashion, the D/A converter must have more than 10 digital bits to obtain the required minimum of 1500 steps ($2^{10} = 1024$; $2^{11} = 2048$).

ADDITIONAL TOPICS

CASCADE SYSTEMS

The valve positioner was introduced earlier, in the discussion on control valves. Although it was introduced simply as a control valve accessory, the positioner is itself an elementary controller. It has a set point, measurement feedback, and other components of a feedback control loop. The process that this loop is controlling (the control valve) is the final control element of another control loop (flow, pressure, etc.). The load variables associated with this loop are packing box friction, and plug unbalance forces due to pressure drop across the valve. In a temperature control loop with a positioner on the control valve, the temperature controller output is the set point to the positioner. If the temperature controller output changes, the positioner moves the control valve to the designated position (within the limits of the air supply pressure) regardless of packing friction or other outside influences. Without the positioner, the controller output might change, but the valve position might not. The temperature controller would not discover that the valve position was not correct until some time later, and the quality of the control would suffer.

This is actually a classical cascade loop. The output of one controller (the primary, or

master controller) is the set point of the other controller (the secondary, or slave). The main advantage of the cascade system is that the effects of load variables in the secondary loop are neutralized before they can affect the primary loop.

The double diaphragm fuel gas valve was also mentioned in the control valve discussion. This valve, which controls a fuel gas burner pressure of 6 to 30 psi for a corresponding 3 to 15 psi air signal to the upper diaphragm, also, is a self-contained feedback control loop, with the air signal as the set point. Increasing the air signal to the upper diaphragm the control valve increases the burner pressure, which is piped back under the lower diaphragm, and tends to close the valve. When the upward force of the gas pressure increases enough to balance the downward force of the air pressure, the valve stops opening and stabilizes at a new position, with the gas pressure at twice the air signal pressure.

If the air signal is held constant, increasing the fuel gas supply pressure, upstream of the valve, will cause more gas to flow to the burner. This will tend to increase the burner pressure, but the increase in burner pressure will, as above, increase the upward force and close the valve. Once again, the valve will find a new position, with the gas pressure at twice the air signal pressure. (The valve functions as a Proportional Only controller, and does have some offset, so the ratio of fuel gas pressure to air signal is not always exactly 2:1, but the effective Proportional Band is narrow, and the regulation is quite good.)

If a standard spring diaphragm globe valve were used instead of the double diaphragm valve, the increased fuel gas pressure would cause the burner pressure to increase, but there would be no corrective action until the increase in heat affected the process temperature, and the temperature controller reduced the signal to the control valve. By using the temperature controller output as the set point to the burner gas pressure controller (the double diaphragm valve), we have again formed a cascade control system. And once again, the secondary controller counteracts load changes before

they can affect the process variable of the primary controller

Generally, this feature is most useful on systems where there is considerable lag in the primary control loop. If there are no load changes, or if the response of the primary loop is as rapid as that of the secondary, there is little or no advantage in cascading the controllers. Most often, the primary loop is a temperature loop. Fuel oil and fuel gas firing controls are often cascaded from TRC's because of the lag in the temperature loop and the frequent load changes in the fuel systems. An overhead product FRC is often cascaded from a fractionating column tray TRC for similar reasons.

Cascading a temperature controller to a flow controller is done to minimize upsets to the temperature; cascading a level controller to a flow controller sometimes is done to minimize upsets in the flow. In this case, the actual level is not tightly controlled. As noted earlier, we do not object to changes in level caused by changes in inlet flow; in fact, we generally welcome the level change because it helps to absorb upsets and smooth out the flow. This is especially desirable if the flow is coupled to other elements in the process, for example in feeding a fractionator, or exchanging heat with another process stream. Load changes that result in a change in the pressure drop of the control valve, and which would change the flow in a simple level control loop, are neutralized immediately by the flow controller in a level-flow cascade system.

Tuning a cascade loop is not much different from tuning an ordinary control loop. The secondary control loop should be tuned normally, as if it were isolated and alone; then the loops should be put on cascade, and the primary controller should be tuned as if it were isolated and alone. The control settings may not be the same as if the primary controller operated the control valve directly, but there is no reason to change the procedure simply because it is a cascade system.

MULTIPLIERS AND SUMMERS

Multipliers and summers are used for mathematical manipulation of either input or output signals. Sometimes the names of these components are used interchangeably -

a "summing amplifier" may be used for multiplication - but the multiplier's output is usually of the form:

$$O_m = K_1 (X) (Y) + B \quad \text{Eq.(23)}$$

$$O_s = [K_1 (X) + K_2 (Y)] + B \quad \text{Eq.(24)}$$

where

O_m = Multiplier Output

O_s = Summer Output

K_1, K_2 = Scaling Factors

X, Y = Input Signals

B = Bias (or Manual Reset)

In some cases there may be a third variable, Z , or even more variables.

One application for a summing amplifier is in boiler feedwater service. We mentioned earlier that for a steam generator, the level should be controlled at the set point and not allowed to vary. This is not always easy. The steam in the steam drum is in equilibrium with the feedwater. The balance between the vapor phase and the liquid phase in the system is set by the temperature and the pressure of the system, and if the load on the steam header increases, and causes a decrease in pressure in the steam drum, more of the liquid in the steam generator will turn to vapor. The expanding vapor will displace some of the liquid, and the liquid in the steam drum will show an increase in level. This is a phenomenon usually referred to as "swelling" of the level in the drum. As a result of this swelling, a level controller cascaded directly to the boiler feedwater FRC would be decreasing the feedwater flow just at the time it should be increasing the flow to keep up with the added demand. The solution to this problem is shown in Figure 61.

The solution is what is generally termed a "three-element" boiler feedwater control system. This system takes the flow measurement signal from the steam make, and uses essentially that signal, properly scaled, as the set point signal to the boiler feed-

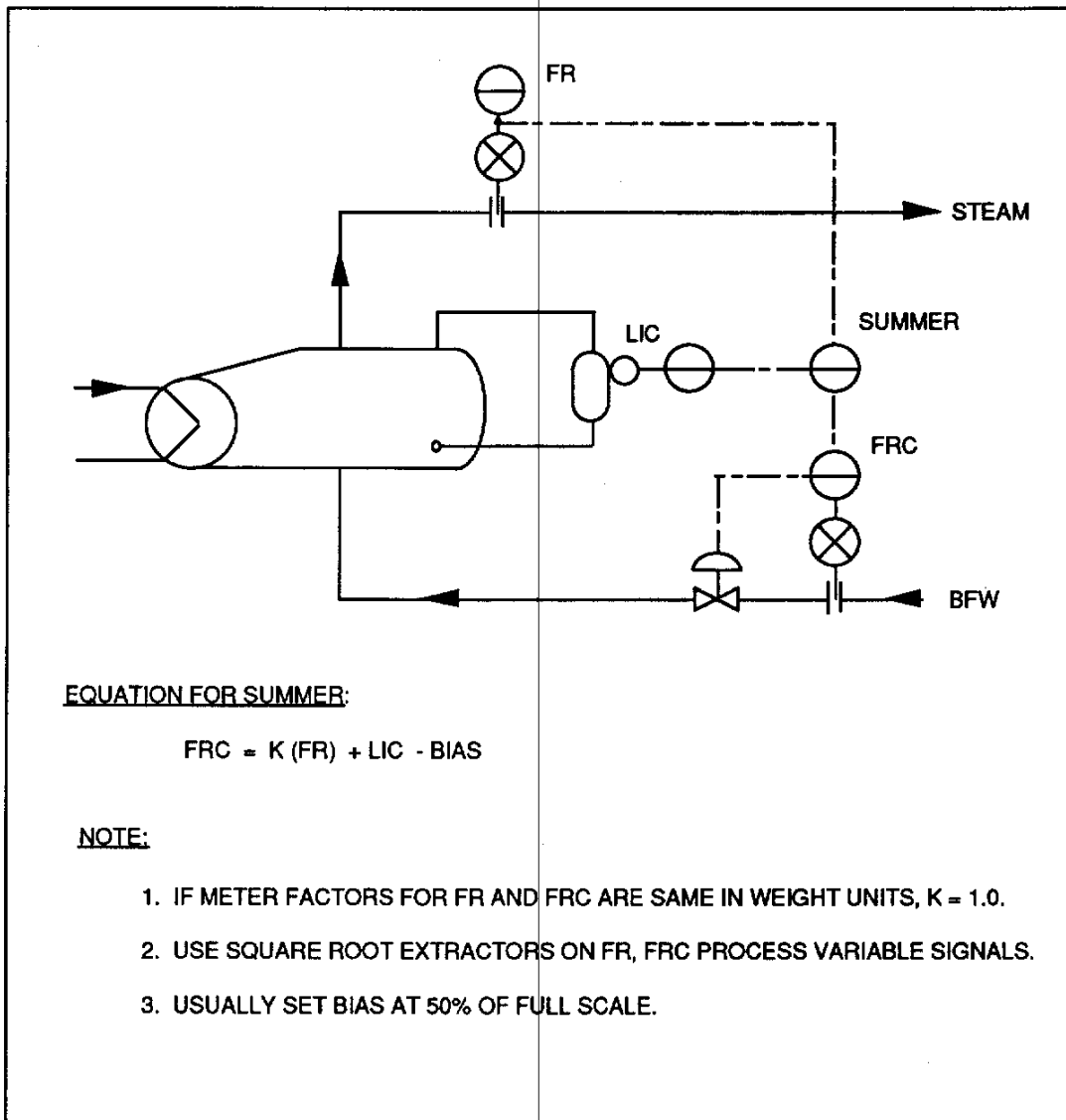


Figure 61 - Three Element Boiler Feedwater Control

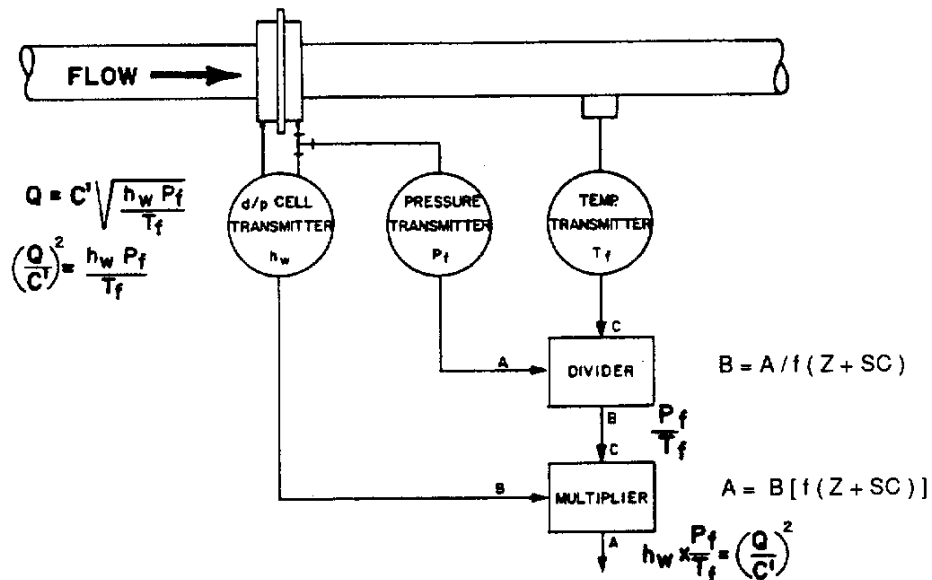
water flow controller. For every pound or kilogram of steam generated, an equal pound or kilogram of feedwater is added to the system. So while the level may fluctuate temporarily, on a long term basis, it should remain constant.

In actual fact, the level will not remain constant. In addition to the steam make leaving the system, there is blowdown, which is not measured. Also, there are inequalities in the meter factors of the steam and feedwater flowmeters, due to installation tolerances and calibration errors. Over a long period of time, unless something was done, the

TEMPERATURE AND PRESSURE COMPENSATION GAS FLOW COMPUTING, TYPICAL EXAMPLE

ORIFICE DESIGN DATA

$h_w = 0-100'' \text{ H}_2\text{O}$
 $P_f = 40 \text{ PSIA}$
 $T_f = 540 \text{ R } (80 \text{ F})$



SYSTEM INFORMATION

PRESS. TRANS. RANGE 0 - 50 PSIA
 MAX. OPERATING PRESS. 45 PSIA
 TEMP TRANS. RANGE 460 - 560 R (0-100 F)
 MIN. OPERATING TEMP 490 R (30 F)

CALCULATIONS

DIVIDER $\frac{P_f}{T_f}$

$$f = \frac{\text{MAX. TEMP. OF TRANS. RANGE}}{\text{MIN. OPERATING TEMP}} \times \frac{\text{MAX. OPERATING PRESS.}}{\text{MAX. PRESS. OF TRANS. RANGE}} = \frac{560 \text{ R}}{490 \text{ R}} \times \frac{45 \text{ PSIA}}{50 \text{ PSIA}} = 1.029$$

$$Z = \frac{\text{MIN. TEMP. OF TRANS. RANGE}}{\text{MAX. TEMP. OF TRANS. RANGE}} = \frac{460 \text{ R}}{560 \text{ R}} = 0.822 \quad S = 1 - Z = 1 - 0.822 = 0.178$$

MULTIPLIER $h_w \times \left(\frac{P_f}{T_f}\right)$

$$f = \frac{\text{MAX. OPERATING PRESS.}}{\text{ORIFICE DESIGN PRESS.}} \times \frac{\text{ORIFICE DESIGN TEMP}}{\text{MIN. OPERATING TEMP}} = \frac{45 \text{ PSIA}}{40 \text{ PSIA}} \times \frac{540 \text{ R}}{490 \text{ R}} = 1.238$$

$$Z = 0 \quad S = 1$$

Figure 62 - Gas Flow Pressure/Temperature Compensation

level would probably drift downward. To offset that tendency, the output signal from the level controller is summed with the steam measurement signal to form the feedwater set point. The level controller is specified with a wide Proportional Band and slow

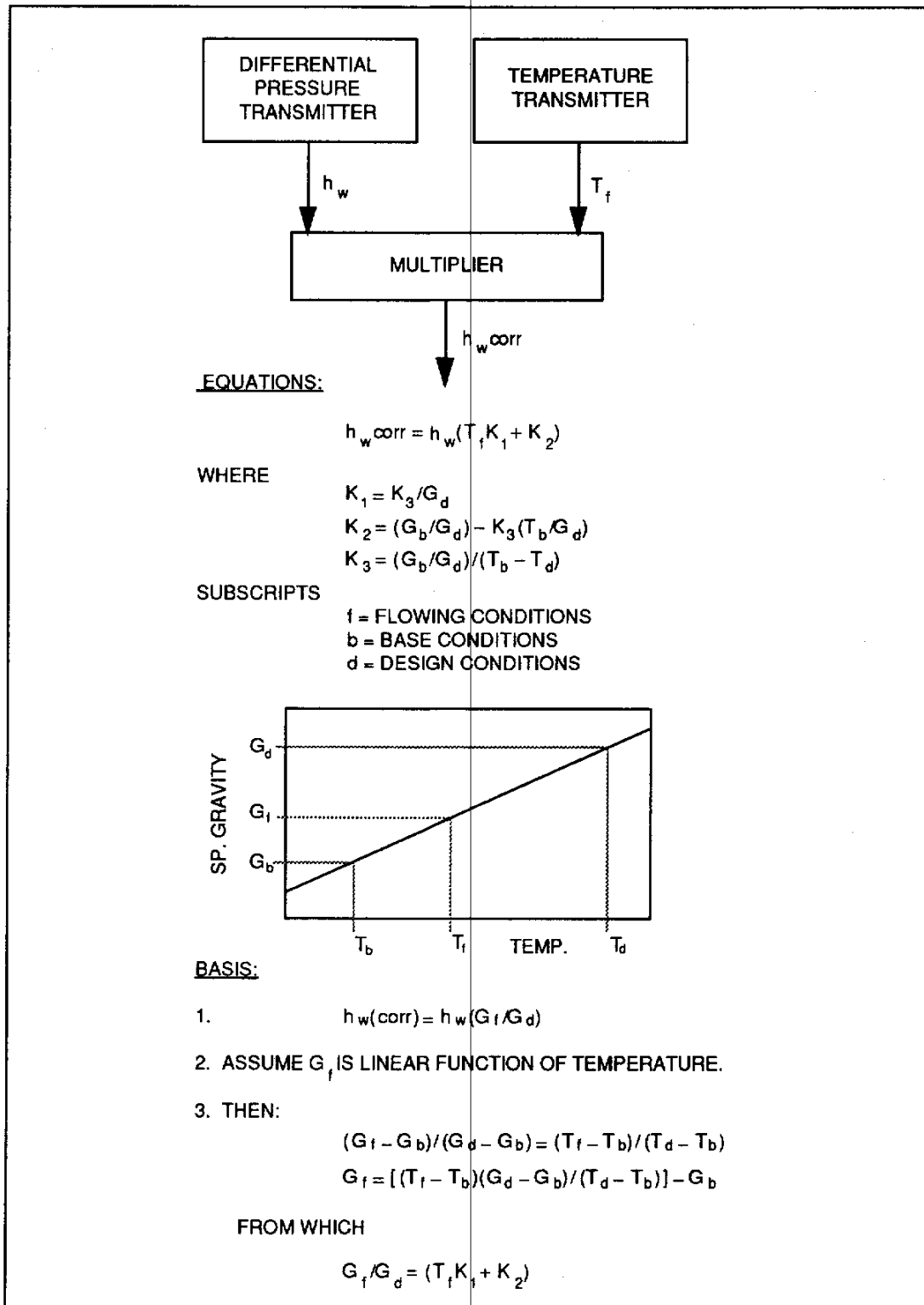


Figure 63 - Liquid Flow Temperature Compensation

Reset, so it makes its correction on the long term error, rather than on the immediate fluctuations. To allow the level controller to operate in both directions, a bias is also added into the system, and adjusted so the level controller output will be about mid-range when the system is balanced. To maintain the 1:1 ratio between change in steam flow and change in water flow throughout the range of flow rates, the square root must be extracted from the flow signals before the summation is accomplished.

Multiplying amplifiers can be used to obtain a gas flow signal that is automatically compensated for changes in pressure and temperature (Figure 62). By proper scaling, the pressure and temperature signals can be modified to corrected signals that represent the quantities $(P_{\text{actual}}/P_{\text{design}})$, and $(T_{\text{actual}}/T_{\text{design}})$. When the differential pressure signal is multiplied by these corrected signals, the result is a flow signal that has been corrected for variations from the design conditions. Since all three variables - differential pressure, pressure, and temperature - are square root functions in the orifice equation, the square root can be extracted by use of a square root chart, just as for an ordinary flow measurement. By relating the flowing specific gravity to the temperature, a similar approach can be used to correct liquid flows for variations in temperature (Figure 63).

Summing amplifiers can also be used instead of valve positioners to obtain split ranges. A 0 to 50% input signal can be used with a summing amplifier to yield a 0 to 100% (or 100 to 0%) signal out. This is useful when split ranging flow controllers, since positioners are not recommended for those services. It is also sometimes necessary to use a summing amplifier to modify the output signal of a controller when signal selectors are used.

SIGNAL SELECTORS

Signal selectors as used at UOP are either high or low signal selectors. These devices are used primarily as override or override controls. Low signal selectors are used most often. The low signal selector takes two signals as inputs, compares them with

each other, and passes the lower signal through as its output. Generally, one of the input signals will be at its maximum value during normal operation. This allows the full range of other input signal to pass through the selector to the output and be used as the controlling signal. If an abnormal situation occurs, the first signal will decrease from its maximum value, and thus limit the output. Figure 64 shows how this limiting action can be used.

The Figure illustrates one possible configuration for a makeup gas compressor control system. Extraneous details, such as coolers, which do not affect the control system have been omitted. In normal operation, Controller C operates Valve C, Controller B operates Valve B, and Controller A operates Valve A. The other inputs to the low signal selectors are higher, so the controller output signals pass through directly to the control valves. If the pressure in the Process decreases, Controller C closes Valve C to decrease the Second Stage spillback and force more gas forward to the Process. This causes the pressure in the Interstage Drum to decrease, and Controller B closes Valve B to decrease the First Stage spillback and force more gas forward to the Second Stage. This causes the pressure in the Suction Drum to decrease, and Controller A closes Valve A.

Without the low signal selectors, if the source of makeup gas failed or was shut off, as the pressure in the Suction Drum decreased, Controller A would close Valve A. With no makeup gas coming in, the pressure in the Suction Drum would continue to decrease, and the pressure in the Interstage Drum would follow. So Controller B on the Interstage Drum would close spillback Valve B to try to increase the pressure. With the spillback valve closed and no makeup gas coming in, very soon the allowable compression ratio of the compressor would quickly be exceeded and damage would result. With the low signal selector, when Valve A closes and the pressure in the Suction Drum continues to fall, the output of Controller A decreases below the output of Controller B, and takes over the operation of spillback Valve B. The valve opens, spills back gas to the suction of the compressor, and the compressor is protected. The operation of Controller B with Valve C is similar.

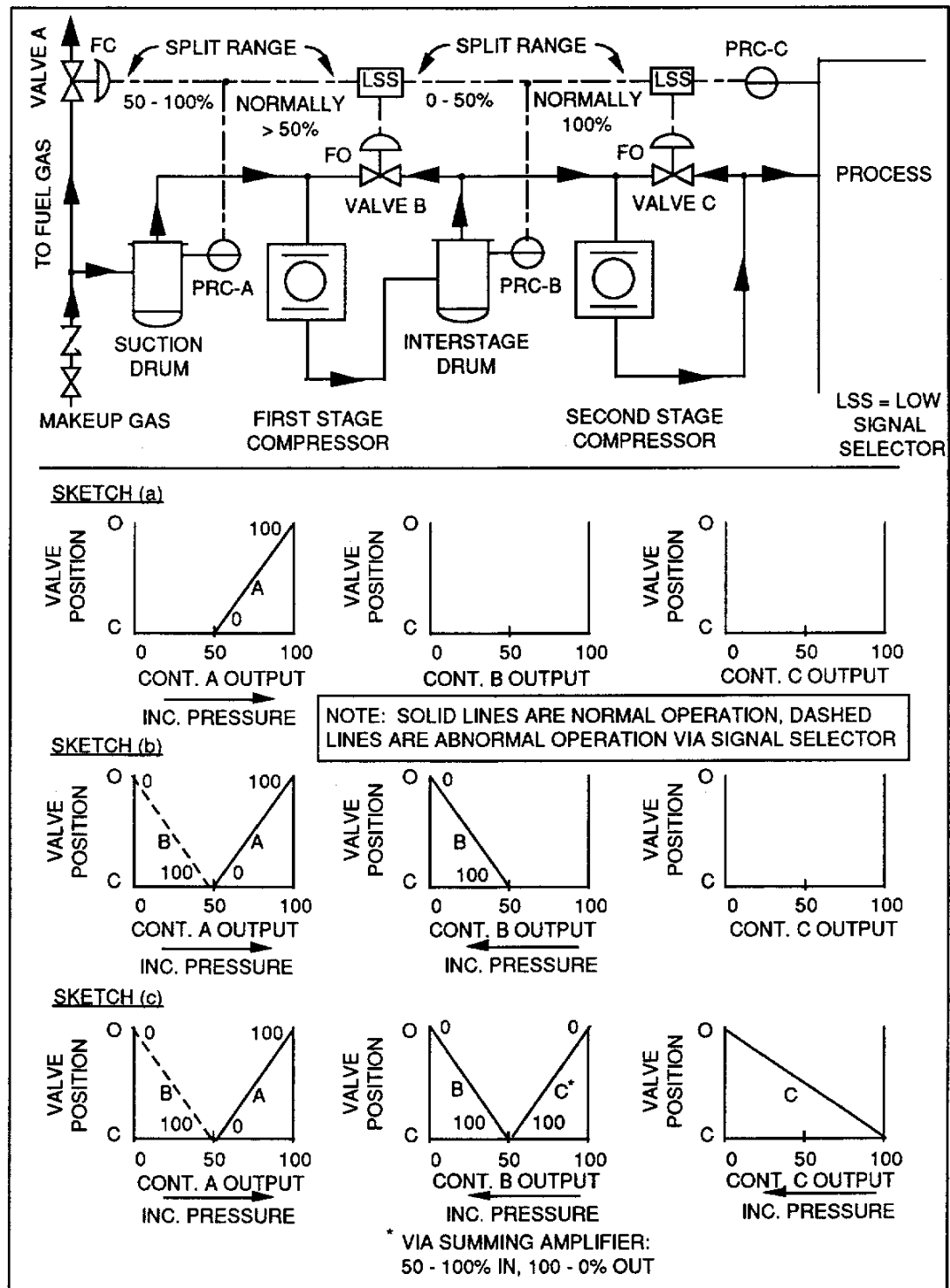


Figure 64 - Makeup Compressor Pressure Control System

Determining the split ranges and controller operation for this type of system can be confusing. It is best to consider one controller at a time. We have established that

each controller operates its own valve during normal operation, and that if makeup gas is lost, Controller A operates Valve B via the low signal selector. This means that in normal operation, Controller A should be in the upper half of its output range, between 50 and 100%. Valve A closes on air failure, so if Controller A is direct acting, increasing Suction Drum pressure will open Valve A, as shown in Sketch (a) in the Figure. Since Valve A is closed at 50% controller output, we would use a valve positioner with 0 to 100% output for a 50 to 100% input.

This leaves the lower half of the Controller A output range to operate Valve B. Valve B opens on air failure; it should be closed (100% air signal to the diaphragm) when the Controller A output is just at 50% (and the process pressure is relatively high), and fully open (0% air signal to the diaphragm) when the Controller A output is at 0% (and the process pressure is low). Therefore the positioner of Valve B must be set with 0 to 100% output for a 0 to 50% input.

Having established the input and output signals of the Valve B positioner, we have also established, by default, the action of Controller B. Valve B is open when the input signal to its positioner is low, and closes as the input signal increases. Since Valve B should be open when the Interstage Drum pressure is high, and close as the pressure decreases, Controller B must be reverse acting, as shown in Sketch (b) of the Figure.

If Valve B is fully closed and the Interstage Drum pressure continues to decrease, Valve C must begin to open and spill back gas to protect the Second Stage Compressor. This means that the output of Controller B must increase above 50%. An increasing signal would function satisfactorily to override the normal signal to the control valve if a high signal selector was used; but since Valve C opens on air failure, only a low signal selector can be used for this purpose. Therefore an amplifier must be added to the control system to convert the 50 to 100% portion of the Controller B output range to a corresponding 100 to 0% input to the low signal selector. (Since we

must provide an amplifier to invert the signal anyway, we may as well use it also to multiply the signal from 50% of the range to a full 100% so we can use a standard positioner on Valve C, rather than a split range positioner.) If no special positioner is used, Valve C will be open at minimum output from Controller C, and closed at maximum output, so Controller C must be reverse acting, as shown in Sketch (c) of the Figure.

There are other ways to arrange the controls for this system. If more amplifiers are used, all splits can be accomplished on the control board, and the split range positioners can be eliminated. With sufficient multiplication and inversion, it is also possible to arrange the system so all three controllers operate with the same action, if that is considered less confusing for the operator and therefore more desirable.

Another system that uses a combination of multipliers and selectors is the heater firing control system shown in Figure 65. Heaters with natural draft usually operate with fairly large quantities of excess air, and the heater fuel flow can be increased by 10% or so over the normal flow rate without exceeding the stoichiometric requirement. Since energy is required to heat the excess air, the efficiency of the heater can be increased if the amount of excess air can be decreased. The problem is that this must be done while insuring that an increase in fuel cannot exceed the stoichiometric amount required for the air that is available; the excess fuel would form a combustible mixture in the flue gas ductwork, and could flash or explode when it subsequently encountered enough oxygen and a source of ignition.

With forced or forced/induced draft systems, the air to the heater is controlled by an FRCAL. If the heater design calls for at least 10% excess air and fuel gas firing only – or 20% excess air and fuel oil firing only, or 15% excess air and combination burners – operator adjustment of the FRCAL set point is considered sufficient. When the heater design calls for less excess air than the percentages above, both the fuel and the air are controlled automatically, as shown in the Figure. The fuel gas density is measured, be-

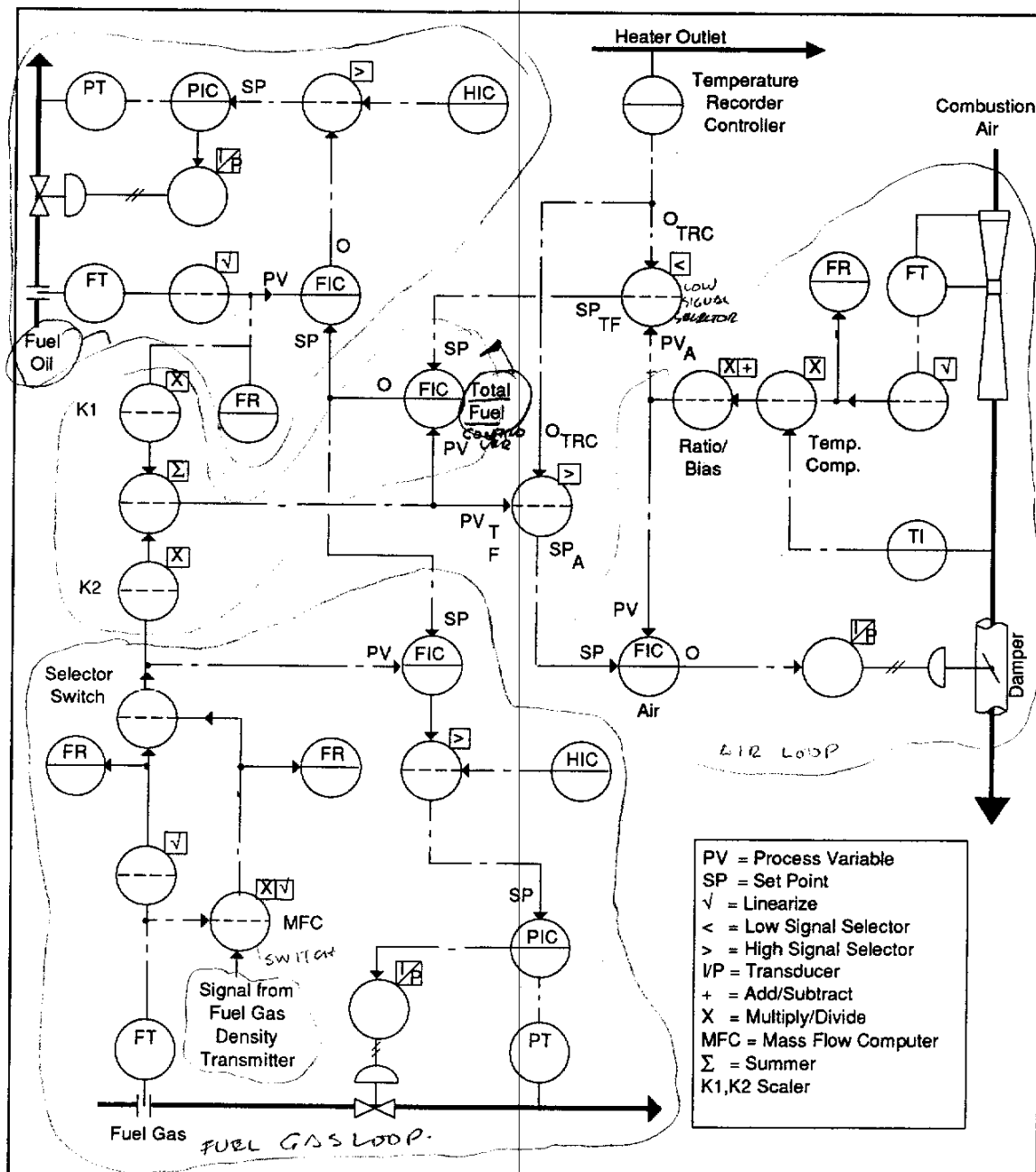


Figure 65 - Fired Heater Fuel-Air Cross-Limiting Control System

cause the composition of the refinery fuel gas is usually subject to considerable variation, and the differential pressure across the orifice is converted into a mass flow rate (a selector switch allows operation with the raw, uncompensated flow signal if the analyzer is out of service). The flow rates of the fuel gas and fuel oil are scaled so both are on a common basis with respect to air requirement, and then are combined into a Total Fuel measurement signal. In normal operation, one fuel is base loaded,

i.e., set at a steady flow rate: the cascade link from the Total Fuel Controller is opened and the controller set point is adjusted by the control board operator. The other fuel is adjusted by the Total Fuel Controller as required by the process.

Scaler K3 is a board mounted ratio controller with adjustable bias. The ratio adjustment multiplies the input signal by the constant number set on the ratio dial. It provides a trim adjustment for changes in fuel heating value; the bias adjustment sets the excess air.

The set point of the Total Fuel Controller comes from the low signal selector, and is the lower of either the air flow measurement signal or the TRC output. The set point of the Air Flow Controller comes from the high signal selector, and is the higher of either the total fuel measurement signal or the TRC output. When the system is in balance, and the Measurement signals are equal to the Set Points, we have:

$$M_{TF} = SP_{TF} = \text{Lower of } M_A \text{ or } O_{TRC} \quad \text{Eq.(25)}$$

$$M_A = SP_A = \text{Higher of } M_{TF} \text{ or } O_{TRC} \quad \text{Eq.(26)}$$

where

M = Measurement Signal

SP = Set Point Signal

O = Output Signal

and subscripts

A = Air

TF = Total Fuel

TRC = Temperature Recorder Controller

We can show that also when the system is in balance, the Air and Total Fuel signals are also equal to each other. If we assume first that O_{TRC} is higher than M_{TF} , then from Eq. (26),

$$M_A = SP_A = O_{TRC} \quad \text{Eq.(27)}$$

But if $M_A = O_{TRC}$, neither is lower than the other, and from Eq.(25),

$$M_{TF} = SP_{TF} = O_{TRC} \quad \text{Eq.(28)}$$

So all four terms - the two measurement signals and the two set point signals - are equal to each other. We can also assume the alternative condition, that O_{TRC} is lower than M_{TF} . In that case, from Eq. (25),

$$M_A = SP_A = M_{TF} \quad \text{Eq.(29)}$$

So if O_{TRC} is lower than M_{TF} , and Eq. (29) tells us that $M_A = M_{TF}$, then O_{TRC} must also be lower than M_A . But then Eq. (25) becomes

$$M_{TF} = SP_{TF} = O_{TRC} \quad \text{Eq.(30)}$$

and again all four terms are equal. The results are the same if we start the analysis with the O_{TRC} and M_A terms, instead of O_{TRC} and M_{TF} .

If the system is balanced and all terms are equal, and the TRC output then increases, the increase will pass through the high signal selector and increase the set point of the Air Flow Controller. The set point of the Total Fuel Controller will remain at the initial value, because it is limited via the low signal selector by the air flow measurement, which has not yet changed. As the air flow increases, the air measurement signal will increase and the Total Fuel Controller set point will increase with it. Conversely, if the system is balanced and all terms are equal, and the TRC output then decreases, the decrease will pass through the low signal selector and decrease the set point of the Total Fuel Controller. The set point of the Air Flow Controller will remain at the initial value, because it is limited via the high signal selector by the total fuel measurement, which has not yet changed. As the fuel decreases, the total fuel measurement signal will decrease, and the Air Flow Controller set point will decrease with it. In either case, the system works to insure that the excess air always exceeds the minimum value set by the controls.

When the TRC output increases, the combustion air change leads the fuel change; when the TRC output decreases, the combustion air change lags the fuel change.

extreme, if the process temperature is too low, it is possible - especially when feed rates are reduced and the heat load is decreased - that the steam pressure required to transfer the heat in the exchanger will not be high enough to push the condensate out of the exchanger, through the steam trap, and into the condensate return header. For instance if the process temperature is only 180 °F, the steam temperature required to obtain the necessary heat transfer rate may be only 200 °F, and the steam pressure only 11.5 psia. This is less than atmospheric. Even if the exchanger used a simple pipe open to the atmosphere instead of a steam trap, the condensate would not have enough pressure to drain out onto the ground.

If flow control of the inlet steam cannot be used, flow control of the outlet condensate may still be possible (Figure 67). This system controls the amount of steam condensed by flooding the tubes in the exchanger and thus varying the area available for heat transfer and condensation. If the system is balanced and the condensate flow rate is decreased, the condensate level in the exchanger rises and covers more tubes. For the same LMTD, reducing the area reduces the heat transfer, which reduces the rate of

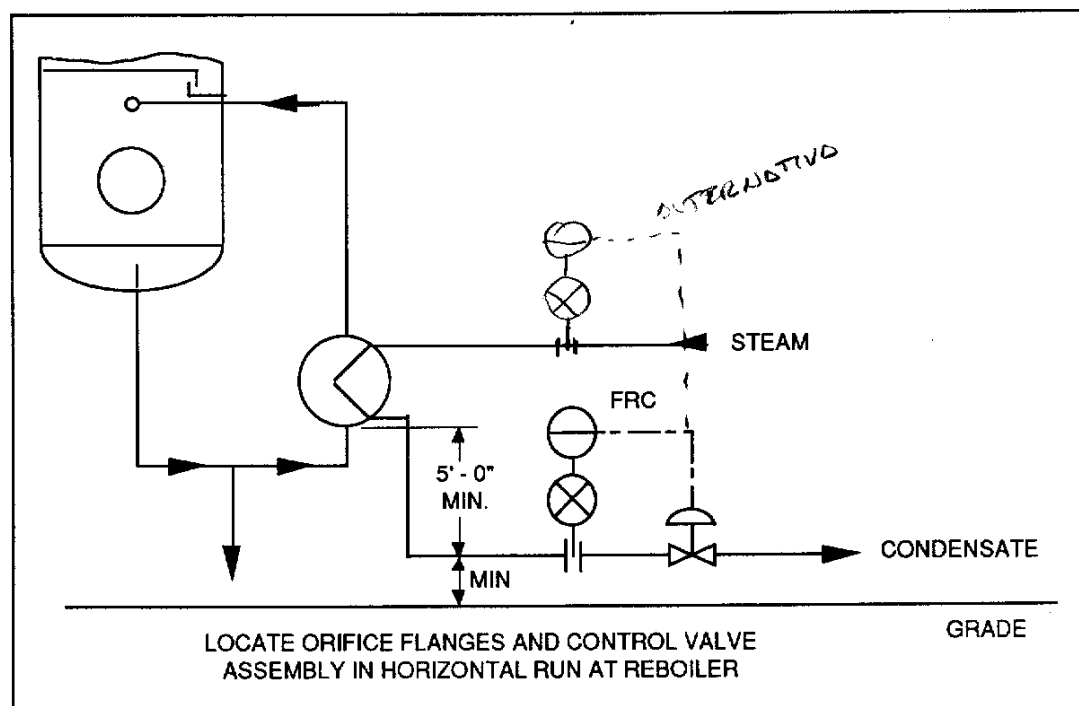


Figure 67 - Steam Reboiler Condensate Control

condensation. When the rate of condensation is equal to the rate of condensate withdrawal, the level stops rising and the system is again in balance.

The response to an increase in condensate flow rate is similar, but with one significant difference. The condensate withdrawal flow rate responds rapidly, but the actual condensation rate does not. The rate of condensation is determined by the heat transfer rate, and the lags associated with that heat transfer limit its ability to respond. If the condensate flow rate is increased too rapidly, the condensation rate cannot keep up, and the exchanger will drain completely. When this happens, the orifice and control valve must try to control a mixed phase flow of steam vapor and slugs of condensate. The two conditions are so different that a constant flow cannot be maintained, and the system becomes uncontrollable. To settle it down, it is necessary to put the controller on manual and allow the condensate level to restabilize in the exchanger before going back to automatic control. The same thing can happen if the flow measurement is on the inlet steam and the control valve is on the outlet condensate; the lag in the system can let the exchanger run dry and cause similar problems.

The condensate leaving the exchanger is, under the worst conditions, saturated, at its flash point. Since the condensate orifice has a pressure drop, the condensate can flash as it passes through the vena contracta. To avoid this, the orifice meter run is located below the exchanger outlet, far enough so the static head of the liquid is greater than the pressure drop from the exchanger to the orifice, and the pressure at the vena contracta is greater than the vapor pressure.

The differential pressure range used for this application is usually 0 to 50 inches of H_2O . If the specific gravity of the condensate is 0.90, at least 55.5 inches of static head must be provided. To account for friction drop in the piping, the difference in pressure between the flange taps and the vena contracta, and for possible changes in density, this dimension is rounded up to 60 inches. The dimension is indicated on the Piping and Instrument Diagram to insure that it will not be missed. The condensate meter run is located a minimum distance from grade. A differential range of 0 to 100

Because of this, the system is often referred to as a "lead-lag" firing control. But as we will see, it is not a true lead-lag system, and it is more correct to refer to it as a "cross limiting" firing control system.

FEED FORWARD/LEAD-LAG SYSTEMS

The emphasis so far has been - and will continue to be - on feedback control systems. In some cases, however, a feed forward/lead-lag system can be useful. Perhaps the most frequent of such applications is a system which measures the feed rate to a

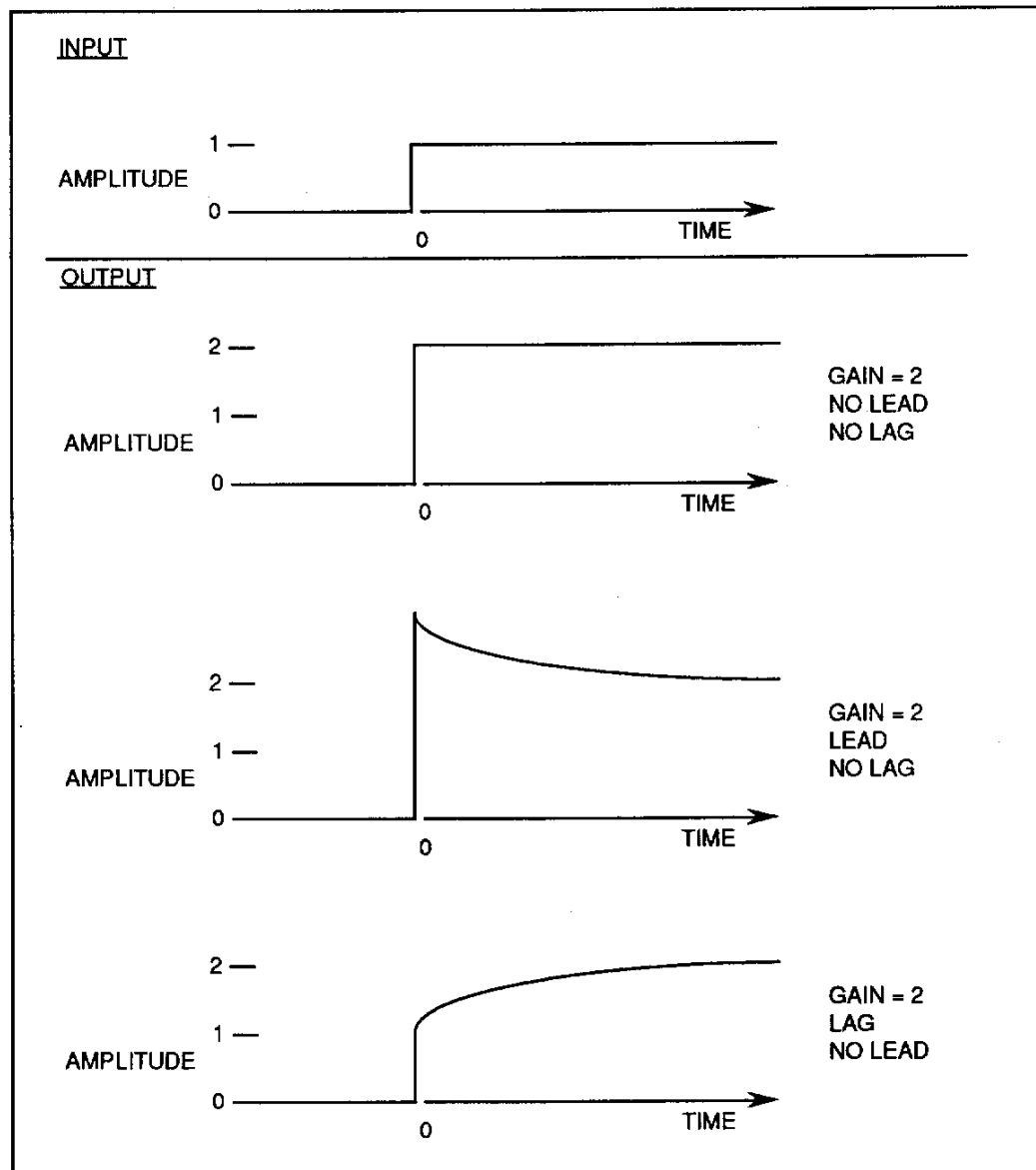


Figure 66 - Lead-Lag Control Systems

fractionator and adjusts the heat input to the reboiler accordingly. This is the feed forward portion of the system.

The problem with feed forward by itself is that the initial response may not be satisfactory. If the column is operating at a low feed rate and a step increase in the feed rate occurs, the feed forward system will increase the heat input. But because of the lag involved in getting the heat into the reboiler and up the column, some of the lighter component in the feed could drop down the column and out the bottom before the full effect of the heat increase is felt.

If a lead-lag network is added to the system, it can be adjusted to provide an initial increase in heat input above that of the feed forward system alone (somewhat similar to the derivative mode in a feedback controller), which will then decay exponentially back to the feed-forward only value. For other applications the lead-lag network can be adjusted to make only a portion of the full change initially, and then make an exponential approach to the final value. These adjustments allow the controls to come nearer to matching the dynamics of the process. Figure 66 illustrates these actions.

REBOILER STEAM CONDENSATE CONTROL

The feed forward system mentioned above can be used with a steam heated reboiler if the control of the heat input is by steam flow to the reboiler. If the control is by condensate flow from the reboiler, the lead-lag system should be used with caution.

When steam is used as a heating medium, the heat is usually controlled by throttling the steam to vary the pressure in the exchanger, as indicated earlier in the control valve discussion, and as shown in Figure 46 (Page 69). A steam trap is used to drain the liquid condensate and insure that it does not build up in the exchanger. There are two extremes which can cause problems with the operation of this system.

If the process temperature is too high for the steam pressure that is available (temperature approach less than about 30 °F, or LMTD less than about 20 °F), there will not be enough pressure drop available for the control valve to control properly. At the other

inches of H_2O could be used, but this would require more static head, which in turn would require a higher elevation for the exchanger; if the exchanger is a fractionator reboiler, a higher elevation would then be required for the entire column, as well.

This system is sometimes used for other condensing vapors, as well as steam. For light hydrocarbons, the specific gravity will be lower than water, and a correspondingly greater static head will be required. In a revamp, if the condensate load is increased to meet new conditions, it probably will not be possible to increase the meter maximum simply by recalibrating the transmitter to a higher differential range, as may be done for other services. In both of these cases it will be necessary to carefully check the available static head to insure that no flashing will occur at the orifice.

FRACTIONATOR CONTROLS

Fractionator controls are subject to three constraints: heat balance, material balance, and the Gibbs Phase Rule. For a simple binary system, with the feed conditions constant, the heat input to the reboiler usually is maintained constant and the reflux flow is varied to control the overhead product composition. Alternatively, the reflux rate could be kept constant and the heat input could be varied to control the bottoms composition. But this method involves more lags connected with the heat transfer equipment, and control is therefore more difficult.

Heat Input Control

The methods used most often for controlling the heat input to a fractionator reboiler have already been discussed in connection with other subjects: fired heaters with TRC or PDRCAL controls, and steam reboilers with flow controls on the steam or the condensate. The other common means of heat input control is by flow control of a hot oil stream to the reboiler. If the control of the hot oil temperature is good, controlling the flow rate provides a reasonably constant heat input. The orifice for the flow measurement is located in the hot oil line to the exchanger. The temperature at this point should be constant, so the specific gravity of the oil should also be constant, and

thus also the flow should be constant. The control valve is located in the hot oil line from the exchanger; this allows the valve to operate at a lower temperature, and it also helps to insure that the exchanger will remain full of oil, regardless of the piping configuration. Since the oil is not compressible and there is no phase change, there is no lag associated with this arrangement, and therefore no control problem.

Gibbs Phase Rule

The principle of the Gibbs Phase Rule provides the basis for most fractionating column composition control systems. The Rule can be written as follows:

$$F = C - P + 2 \quad \text{Eq. (31)}$$

where

F = Number of Degrees of Freedom (e. g., Number of Independent Variables)

C = Number of Components

P = Number of Equilibrium Phases

For a binary system, the trays of the fractionator have two components and two phases. Applying the Phase Rule to this system, we find that there can be only two degrees of freedom:

$$F = 2 - 2 + 2 = 2 \quad \text{Eq. (32)}$$

The independent variables in this system are: (1) pressure, (2) temperature, and (3) composition. Fixing the pressure in the column would leave only one degree of freedom; if the temperature is also fixed, there will be no degrees of freedom left, so fixing the temperature also fixes the composition. For multicomponent systems, the relationship is not precise, as it is for binary systems, but it is still usable as a qualitative indicator.

Pressure Control

At UOP, the method used to control the pressure of a column depends on the cooling medium used on the overhead condenser, on the overhead composition, and on the

magnitude of the overhead pressure itself.

Total Condensation, Below 5 psig (Figure 68)

When there is no net gas and the column pressure is low, the receiver usually is allowed to float on the flare header, since the flare is the lowest safe destination pressure. Upsets which affect the flare pressure will also affect the column, and a

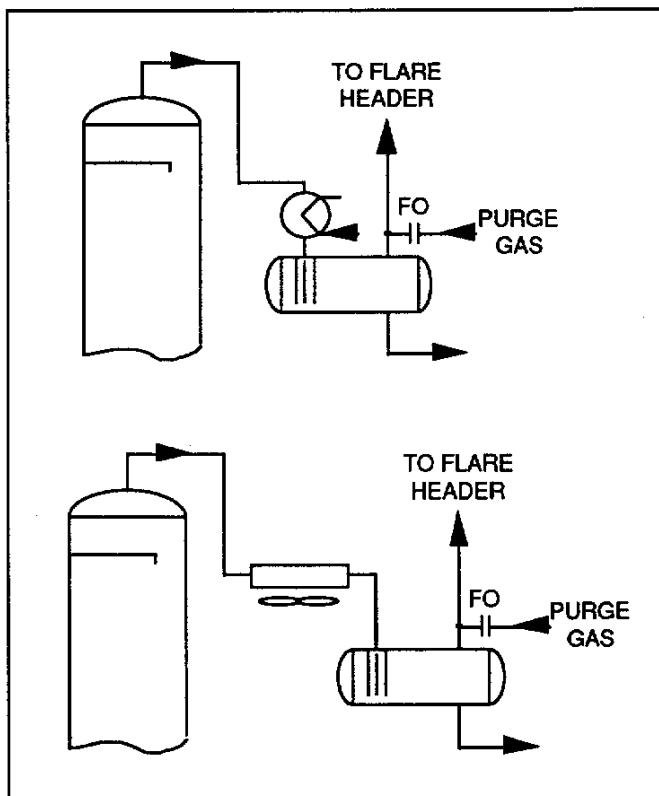


**Figure 68 - Fractionator Pressure Control,
Total Condensation, < 5 psi**

contingency in the reboiler heat source should be provided for this possibility. A gas purge, controlled by a restriction orifice, is provided to sweep the line from the receiver to the flare header, usually at about 0.5 to 1.0 ft/sec, to keep contaminants in the flare header from backflowing into the receiver.

If a water cooled condenser is used in this service, it should be elevated above the receiver, and free draining to it. The non-condensable purge gas must be able

to get back to the condenser, or the condenser will tend to flood and operate in a manner similar to a steam reboiler on condensate flow control. The column pressure then will not change until the flooded area changes, and pressure control will be sluggish and poor.

Total Condensation, 5 to 25 psig (Figure 69)

At pressures in this range, a "push-pull" blanket gas system is used. This is a split range system which opens a valve to bring in blanket gas when the receiver pressure

falls, and opens another valve to vent gas to the flare if the pressure rises. The same considerations as in the case above apply to the condenser in this case, also.

Total Condensation, More than 25 psig (Figure 70)

This system is somewhat different with a water condenser than it is with an air condenser. With the water condenser, the operation is based on the flooded condition described above. The condenser is located below the receiver, with the outlet piping entering the receiver at the bottom so it is sealed in the liquid. The pressure is controlled by a hot vapor bypass control valve, which bypasses hot vapors around the condenser. The differential pressure across the control valve is equal to the

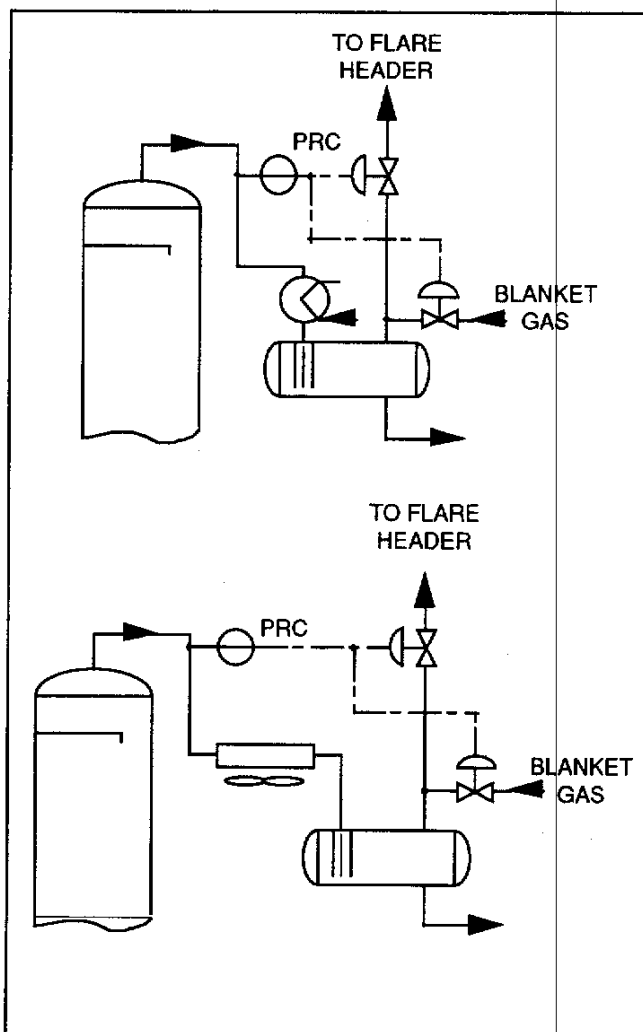


**Figure 69 - Fractionator Pressure Control,
Total Condensation, 5 to 25 psi**

liquid head from the receiver to the exchanger outlet plus the pressure drop. Opening the hot vapor bypass valve increases the pressure in the receiver, which causes the level in the condenser to rise, flood more tubes, and decrease the rate of condensation, thereby increasing the pressure in the column. The hot vapor bypass valve is sized for about 5% of the overhead vapor rate, as a maximum, and the valve is specified with percentage trim. The 5% figure is sufficient to cover the heat loss of the receiver and most column upsets.

The hot vapor performs the function of the non-condensable gas

in the previous systems, but since it actually is condensible, certain precautions must be taken. As long as the surface of the liquid remains smooth and unruffled, a thin layer of hot liquid remains at the liquid-vapor interface, isolating the hot vapor from the colder liquid below. If something happens that disturbs the surface, the colder liquid will contact the hot vapor and the vapor will condense, causing a sudden decrease in the receiver pressure and upsetting the column. If the bypass line is too long and is not protected, when a rainstorm occurs, some of the hot vapors could condense, fall into the receiver, and disturb the surface, causing such an upset. The hot vapor bypass line should be short, free draining in both directions from the hot vapor bypass valve, and insulated from the valve to the receiver. Also, a line (not shown) must be provided to allow venting non-condensibles from the top of the condenser to the receiver, from which they can be vented to the flare.

A different arrangement is used with an air condenser. It generally is not practical to locate the air condenser below the receiver, and in addition the exchanger geometry must be broad and flat to allow air to pass freely over the tubes with minimum pressure drop. A small change in level in the condenser would cause a large change in flooded area, and precise control would be difficult. Instead, a butterfly valve is used upstream of the condenser

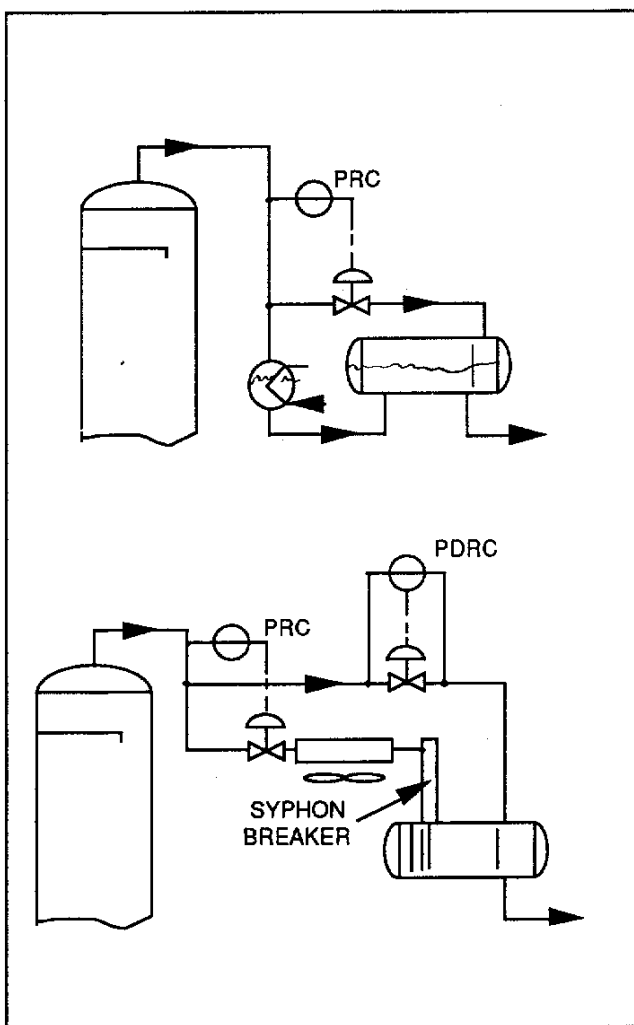


Figure 70 - Fractionator Pressure Control,
Total Condensation, More Than 25 psi

to throttle the vapors and control the column pressure directly. To keep the receiver pressure from changing too much and causing problems for the reflux pump as conditions change in the condenser, a PDIC control loop measures the differential pressure between the column and the receiver, and adjusts a hot vapor bypass valve to hold it constant. The considerations for the hot vapor bypass in this case are much the same as with the water condenser, except that the line from the condenser to the receiver should not be sealed in the liquid. It should be free draining, and an equalizing or siphon-breaker line should be provided to allow hot vapor from the receiver to get back to the condenser outlet and prevent formation of a liquid leg with a significant static head. Since the liquid will be entering the receiver from the top, the receiver should have a baffle to isolate the inlet area and prevent the turbulent liquid from disturbing the surface in the rest of the receiver. Pressure drops used for the valves are generally about 15 psi for the butterfly valve and 10 psi for the hot vapor bypass valve.

Occasionally, systems with air condensers and water trim condensers are used. These can be difficult to operate, especially if the water condenser is located below the receiver. The trim condenser should be sized with at least 30% of the total duty, to insure that there will always be two phases present at the outlet of the air condenser. If only the liquid phase is present, the static head from the air condenser to the receiver

under some conditions could be great enough to make the pressure drop of the hot vapor bypass valve go negative. It is much better to locate the trim condenser a minimum distance above the receiver,

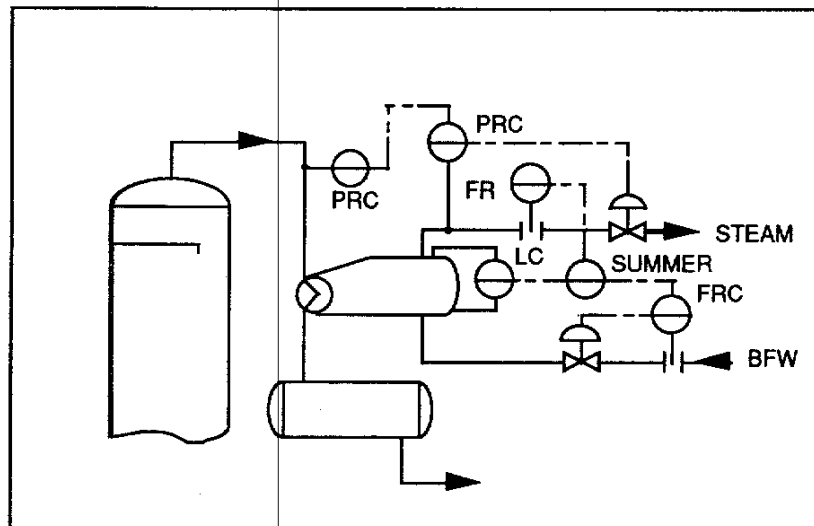


Figure 71 - Fractionator Pressure Control,
Total Condensation, Steam Generation

and use the same arrangement as was described above for the air condenser alone.

Total Condensation, Steam Generation (Figure 71)

If the overhead material has a high boiling point, it may be possible to generate steam with the condensing vapors. In this case, the column pressure controller would be cascaded to the pressure controller of the steam generator. Increasing the pressure in the steam generator would increase the temperature in the steam generator, decrease the heat transfer rate and therefore the condensation, and increase the pressure in the column.

Partial Condensation (Figure 72)

For partial condensation systems that produce net gas streams, the pressure can be controlled by throttling a control valve in the net gas stream. Pressure drop considerations

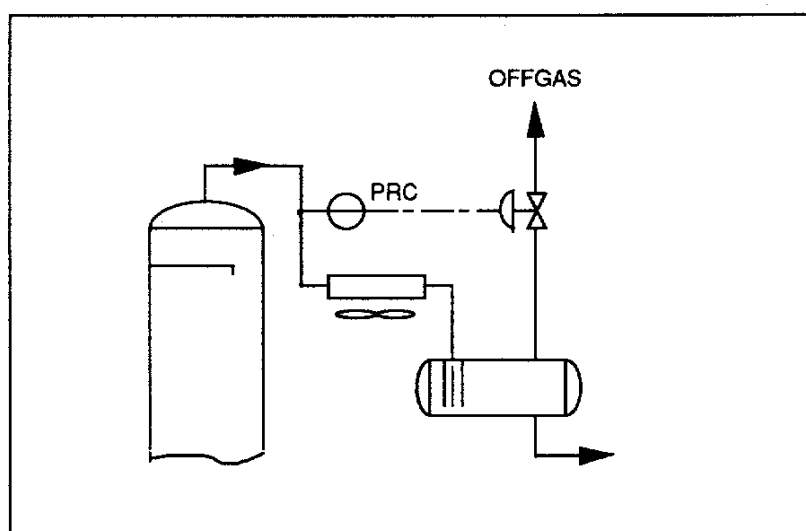


Figure 72 - Fractionator Pressure Control,
Partial Condensation

for the control valve were covered earlier in the control valve discussion. If the column is being used only for stripping, and tray temperatures are not controlled, the pressure control may be on the receiver,

rather than on the overhead line. Water condensers may be located either below or above the receiver in this case.

Composition Control

Although analyzers are used occasionally for composition control of fractionators, such analyzers are expensive, and most often the Gibbs Phase Rule is applied and a

temperature controller is used instead. When the product of interest is the fractionator overhead material, a temperature control in the upper part of the column is usually used. For simple end point control with multicomponent systems, the TRC thermocouple is usually located in the overhead line at the column. For a split between components in a multicomponent system, or a loose spec on a binary system, a TRC on one of the trays is used.

The usual problem in this situation is to decide on which tray to use for control. In general, using the tray that is half way between the feed and the reflux will be satisfactory. Alternatively, tray to tray calculations can be run on a computer, and the variation of product quality with tray temperature can be calculated. If this done for several values of different process variables, such as feed rate or feed composition, the tray which shows the least variation in product quality at a constant temperature can be determined. (This is not necessarily the tray which shows the greatest temperature variation with product quality, as Figure 73 indicates.)

If there is still uncertainty as to the validity of the result, additional thermowells can be installed above and below the tray selected, for alternate use if required.

A benzene column is an example of a column with a tight spec on an essentially binary separation. The column operates at atmospheric pressure, floating on the flare. Most of the upsets caused by pressure fluctuations in the flare header are eliminated by providing a header dedicated to the fractionators and running directly to the flare knockout drum, but some pressure variation is still possible. In addition, even at a constant tray temperature, changes in barometric pressure alone are enough to cause unacceptable variations in the quality of some high purity products

To compensate for these pressure variations, a differential temperature measurement can be used. The reference point is near the top of the column, in the relatively pure component. The change in temperature due to variation in pressure at the control point will be canceled out by a similar change at the reference point, and the change in product composition will be minimized.

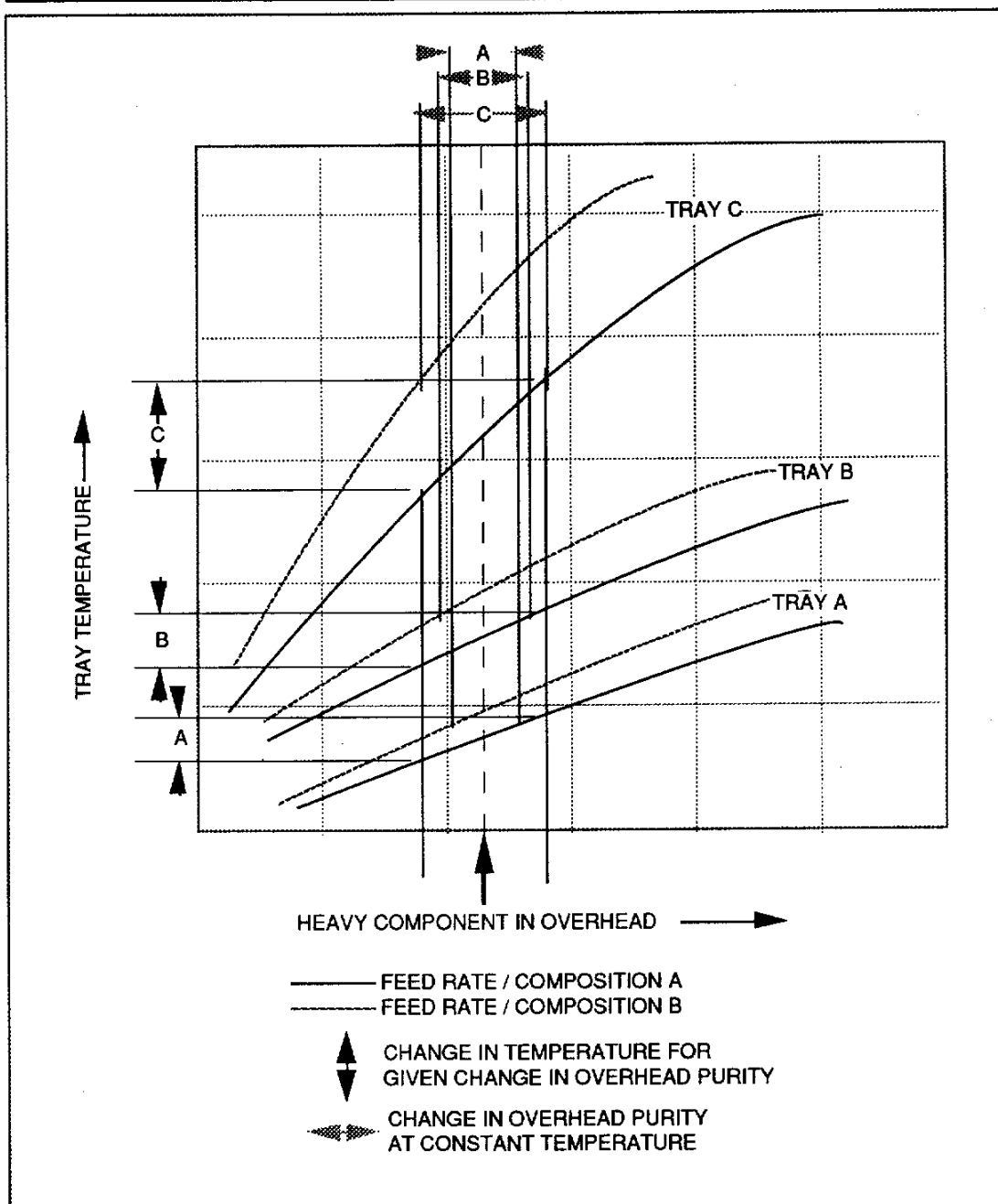


Figure 73 - Fractionator Control Tray Selection

Since the temperature on the control tray is a function of composition and pressure, increasing the reflux to the column changes the composition and decreases the tray temperature; but it also increases the tray loading and the pressure drop, which increases the tray temperature (Figure 74). At low tray loadings, the composition change governs; at high tray loadings, the tray pressure drop governs. The tray temperature is

the sum of the two effects, so as the reflux increases, the temperature first decreases, then passes through a minimum, then increases. If the control point is set too low, below the minimum, the reflux will increase, trying to bring the temperature down to the set point, but it will not succeed, and the column will be out of control. This is less likely to be a problem if there are enough trays between the control tray and the reference tray to make the ΔT contribution from composition relatively large compared to the contribution from the pressure drop.

Another method used occasionally at UOP to accomplish a similar function is to actually measure the pressure on the tray with an absolute pressure transmitter. The pressure measurement and the temperature measurement are then scaled and used in a simple summing network to generate what is, in effect, a composition signal, which is then used by the PcTRC (Pressure-compensated TRC) controller. At normal operating conditions, the PcTRC measurement and set point are at mid-scale. If the temperature increases due to a change in composition, the PcTRC measurement signal increases; if the temperature increases due to a change in pressure, the increase in temperature signal is counteracted by the increase in the pressure signal, and the measurement signal to the PcTRC does not change.

If this system is used, care should be taken to avoid settings on the equipment that could lead to overcompensation by the pressure signal. If the signal overcompensates, an increase in pressure will result in a net decrease in the signal to the PcTRC, indicating that the material on the tray has become lighter. If the PcTRC is in the lower part of the column and adjusts the heat to the reboiler, the decrease in input signal would call for an increase in heat, which would increase the traffic in the column and the pressure on the control tray, and could lead to instability. Figure 75 shows a typical PcTRC system.

Most of the systems mentioned above are used primarily for control of overhead products. A TRC on the bottom of a fractionator can be used to control initial boiling point, much the same as a TRC on the overhead can control the end point. Likewise a TRC in a tray between the feed and the bottom can be used to control the split between

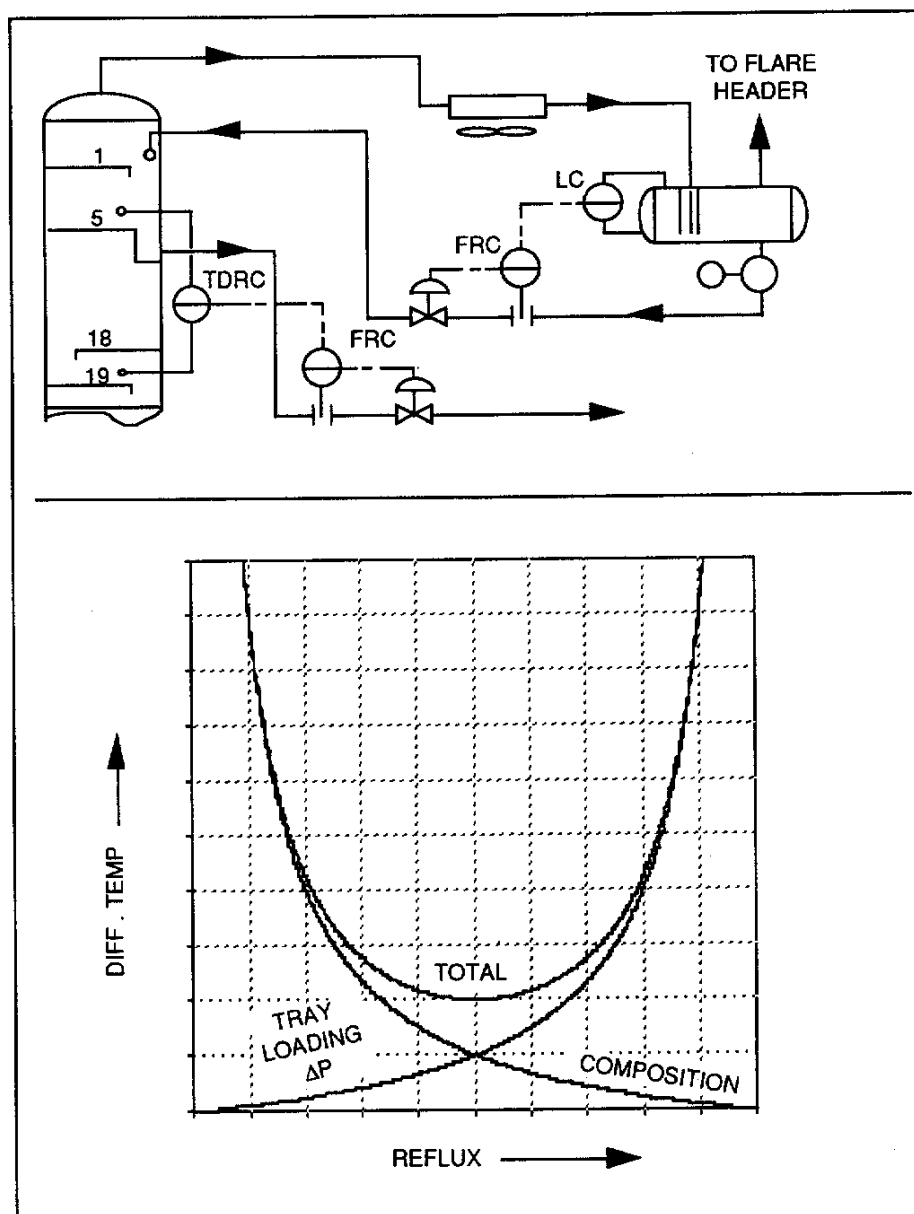


Figure 74 - Differential Temperature Control

components in a multicomponent system. Sidecut streams from crude columns, or others with multiple cuts from a single fractionator, sometimes can be controlled automatically; most often they are adjusted manually, based on lab reports, to control the end point of the sidecut stream. On a benzene column when there may be water or light material in the feed, or where purge gas in the receiver might cause contamination, the net product is taken from a sidecut, a few trays below the top, to insure that it is pure and dry.

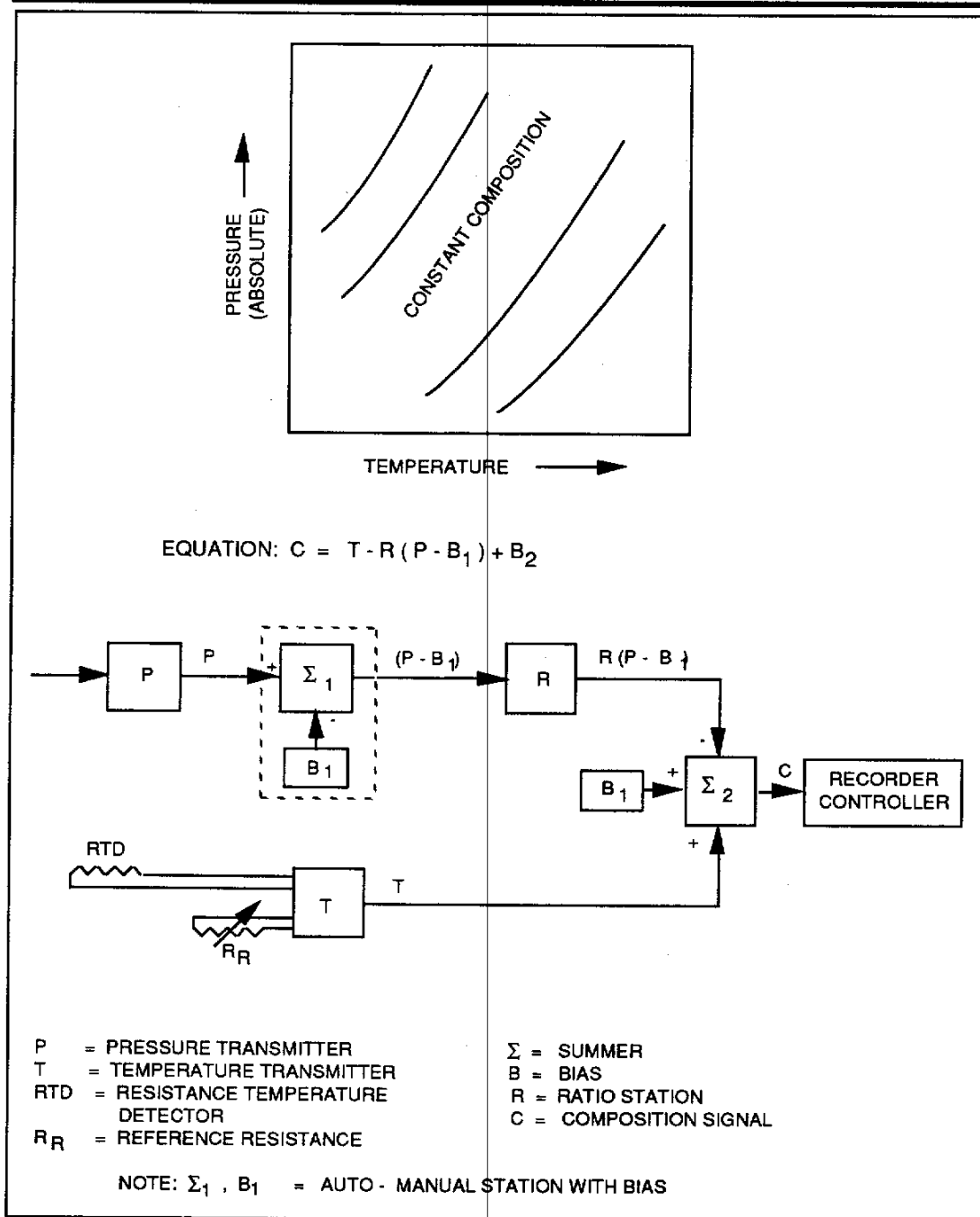


Figure 75 - Pressure Compensated TRC

Material Balance Control

The traditional control configuration for the top of a column has temperature (composition) control adjusting the reflux and the receiver level control adjusting the net product. This method is still used if the receiver must function as surge capacity for

another part of the process, but since about 1975, the preferred configuration at UOP has composition controlling the net product, and the receiver level controlling the reflux.

This configuration (Figure 76), often referred to as the “flywheel” control system at UOP, is preferred because it is more direct and provides better response to changes in heat balance and feed composition. The amount of overhead product that can be withdrawn is determined by the feed composition. If the feed contains 10 moles of light component A and 90 moles of heavy component B, the maximum amount of overhead product that can be withdrawn without affecting the purity is 10 moles. If the feed composition changes to 10.1 moles of A and 89.9 moles of B, the additional light component will travel up the column to the control tray, and the composition controller will increase the net overhead product rate to maintain the composition on the tray. Since the overhead product rate is usually fairly small in comparison to the reflux rate, the resulting change in reflux will be more in proportion to the change in overhead composition than if the reflux were controlled directly. (Actually, “direct” control of the reflux is somewhat difficult, because it is the internal reflux that determines the operation of the column, and the internal reflux is dependent on heat losses and subcooling of the external reflux, as we shall see shortly.)

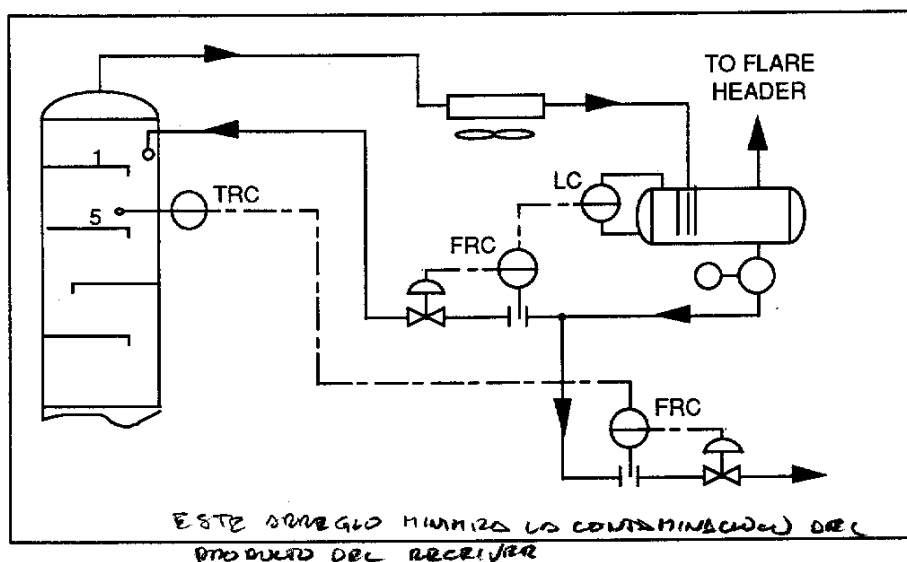


Figure 76 - Flywheel Control

If the column is in balance, and an upset in the reboiler heating medium occurs and increases the heat input, the vapor rate up the column will increase and will cause an increase in the level in the receiver. If the column was recovering the maximum amount of overhead product before the upset, the additional material in the receiver must include more of the heavier component which is being taken overhead as an impurity. The traditional control system will send the contaminated product out to storage until the composition controller can react and increase the reflux. With the flywheel control system, the increased receiver level will immediately increase the reflux; when the composition controller detects heavier material on the control tray, it will decrease the flow of net product, which will increase the level in the receiver even more, and cause another increase in reflux. If the upset is severe, the column may go on total reflux until the composition gradient on the trays is reestablished and the product purity is back to normal.

The proportional band should be used on the receiver level control to minimize the offset and insures that changes in the net product are passed through to the reflux with a minimum amount of delay. Ideally, setting the proportional band to zero percent would completely eliminate the offset; but this would cause problems with stability. Reset action lags the proportional action, and eventually would eliminate the offset, but would not greatly help the response to an upset.

A more recent modification to this system (Figure 77) moves the takeoff for the overhead product to a point between an FRC orifice that measures the total overhead from the receiver and the reflux control valve. Using this location for the product takeoff, a change in net overhead rate by the composition controller is instantly detected by the reflux controller and reflected as a change in reflux rate, without having to affect the level in the receiver. This change improves the response of the control system, and also allows the use of a wider proportional band on the level controller, which, in turn, allows better utilization of the surge capacity of the receiver. The disadvantages of this system are that the net overhead rate must be subtracted from the total overhead

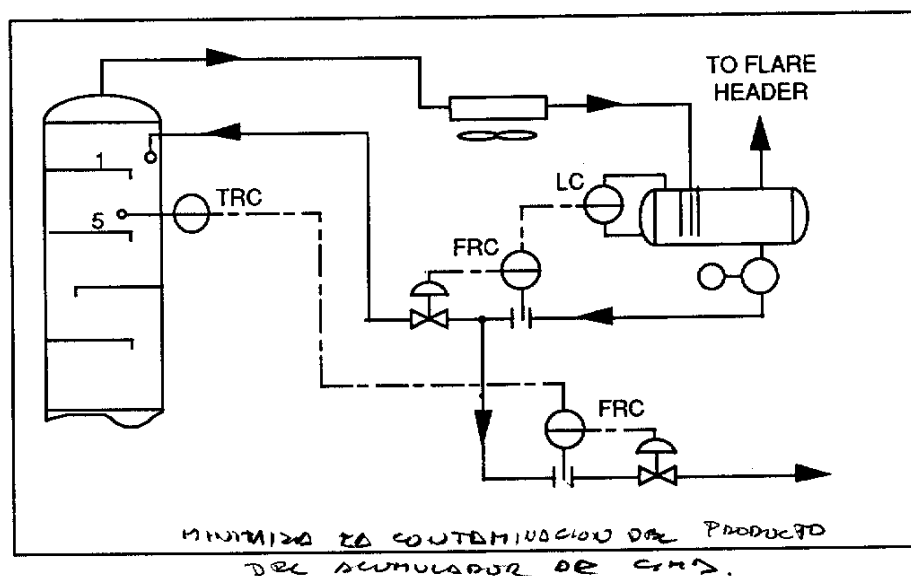
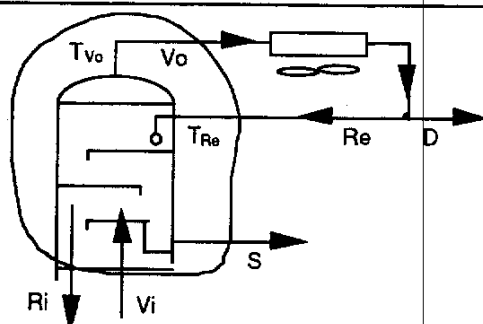


Figure 77 - Modified Flywheel Control

rate to obtain the reflux flow (the reflux itself is not measured directly), and also that it is somewhat more difficult to operate with the controllers on manual control. However, the improvement in control response should make the small inconvenience worthwhile.

Changes in the thermal quality of the reflux itself require additional equipment with the traditional system, but the flywheel system compensates automatically. If the column has an air condenser, and a sudden rainstorm occurs to increase the subcooling of the reflux, although the external reflux may be controlled at a constant value, more vapors will be condensed on the first tray, and the internal reflux will increase. The external reflux will remain at its constant until the control tray thermocouple detects a change in composition, but by this time the column is already upset.

To avoid this upset with the traditional system, equipment must be added to measure not only the external reflux flow, but also the external reflux and overhead vapor temperatures; the internal reflux is then calculated from a heat and material balance around the top tray. (If the column has a sidecut stream, that flow is also used in the calculation, as shown in Figure 78). The calculated internal reflux is then used as the measured variable, and the external reflux flow is adjusted to maintain the internal



V_o = OVERHEAD VAPOR
 V_i = INTERNAL VAPOR
 R_e = EXTERNAL REFLUX
 R = INTERNAL REFLUX
 S = SIDECUT
 D = DISTILLATE

MATERIAL BALANCE:

$$(1) \quad V_i + R_e = V_o + R_i + S$$

ENERGY BALANCE:

$$(2) \quad V_i H_{Vi} + R_e H_{Re} = V_o H_{Vo} + (R_i + S) H_{Ri}$$

FROM (1):

$$(3) \quad V_i = V_o + R_i + S - R_e$$

SUBSTITUTING FROM (3) FOR V_i IN (2)

$$(4) \quad (V_o + R_i + S - R_e) H_{Vi} + R_e H_{Re} = V_o H_{Vo} + (R_i + S) H_{Ri}$$

SINCE BOTH VAPORS ARE SATURATED, WITH ESSENTIALLY SAME PRESSURE TEMPERATURE, AND COMPOSITION:

$$(5) \quad H_{Vi} \approx H_{Vo}$$

SUBSTITUTING FROM (5) FOR H_{Vi} IN (4),

$$(6) \quad (V_o + R_i + S - R_e) H_o + R_e H_{Re} = V_o H_{Vo} + (R_i + S) H_{Ri}$$

$V_o H_o$ CANCELS; COMBINING TERMS:

$$(7) \quad (R_i + S) (H_o - H_{Ri}) = R_e (H_o - H_{Re})$$

SINCE R_i IS SATURATED LIQUID,

$$(8) \quad H_{Vi} = H_{Ri} + \Delta H_{\text{vap}}$$

SUBSTITUTING FROM (5) FOR H_{Vi} IN (8), AND REARRANGING,

$$(9) \quad (H_o - H_{Ri}) = \Delta H_{\text{vap}}$$

IF R_e IS SUBCOOLED,

$$(10) \quad H_o = H_{Re} + C_p (T_{Re} - T_{Vo}) + \Delta H_{\text{vap}}$$

FROM WHICH,

$$(11) \quad (H_o - H_{Re}) = C_p (T_{Re} - T_{Vo}) + \Delta H_{\text{vap}}$$

SUBSTITUTING FROM (9) FOR $(H_o - H_{Ri})$, AND FROM (11) FOR $(H_o - H_{Re})$ IN (7),

$$(12) \quad (R_i + S) (\Delta H_{\text{vap}}) = R_e [C_p (T_{Re} - T_{Vo}) + \Delta H_{\text{vap}}]$$

FROM WHICH

$$(13) \quad R_i = R_e [(C_p / \Delta H_{\text{vap}}) (T_{Re} - T_{Vo}) + 1] - S$$

OR, IN GENERAL TERMS

$$(14) \quad R_i = R_e (1 + K \Delta T) - S$$

WHERE

$$K = (C_p / \Delta H_{\text{vap}})$$

$$\Delta T = (T_{Re} - T_{Vo})$$

NOTE: K HAS UNITS OF T^{-1} .
 $(\text{BTU/LB } ^\circ\text{F}) / (\text{BTU/LB}) = 1 / ^\circ\text{F}$

Figure 78 - Internal R reflux Control

reflux at the set point. With the internal reflux held constant, the control tray temperature does not change, and the upset does not occur.

When the colder external reflux condenses more vapors and increases the internal reflux, the vapor rate to the overhead receiver decreases and the receiver level goes down. If the receiver level controls the external reflux, as in the flywheel control system, the external reflux also will decrease, so the internal reflux will remain more constant. The reaction is the same as for the traditional system with the internal reflux controller, but no extra equipment is required, and there are no calculations to be made, and therefore nothing to introduce errors into the loop. The system is self-correcting.

As mentioned earlier, a narrow proportional band has been used on the receiver level controller when it has been used to control the reflux; this means that the offset is minimized so the changes that affect the receiver level are immediately reflected as a change in the reflux. This also means that if the receiver has a water boot, draining the boot too rapidly can affect the receiver level, and therefore the reflux. If possible, the interface level should be controlled automatically so the draining is smooth and continuous. If it is done manually, extra care may be required to avoid upsetting the column.

If a column has a very high reflux ratio in terms of reflux to distillate, changes in product rate have a relatively small effect on the mass of material in the column, and it takes considerable time for that effect to appear. In this case, automatic composition control of the overhead product rate becomes difficult, and manual adjustment by the operator is generally preferable.

If the overhead of the column is controlled on composition, the bottom of the column is controlled on inventory (level) to close the material balance, and vice versa. For each control configuration discussed for the overhead, there is usually a similar configuration for the bottom. The main difference is that the reflux is varied in the

overhead control arrangements, but the heat input generally remains constant, even when the main controls are at the bottom of the column. Adjusting the heat input automatically to control one of the process variables makes column operation much less stable than if the heat input is kept constant, and is therefore generally not recommended, especially for close fractionation.

If the heat input is kept constant the variables remaining are the net overhead, the reflux, and the net bottoms; and the controls remaining are the composition, the receiver level, and the bottoms level. For the reasons mentioned earlier, it is preferable to have the reflux controlled by the receiver level. That leaves net overhead and net bottoms as variables, and composition and bottoms level as controls. Composition control will probably work best on the smaller flow rate, so if the bottoms stream is larger, using composition to control the net overhead and bottoms level to control the net bottoms will probably be satisfactory. Conversely, if the bottoms stream is smaller, composition control of the net bottoms, and bottoms level control of the net overhead, will probably be suitable. If the net bottoms rate is very small, as in some rerun columns, the conditions are similar to the overhead system with a high R/D, and manual adjustment of the bottoms product rate by the operator is probably best.

The control configurations discussed above do not cover all possibilities, and in any case they are intended as basic guidelines only. But they should be satisfactory as controls for simple fractionators, and as a starting point for control of more complex systems.



ENGINEERING DESIGN SEMINAR

INSTRUMENTATION

APPENDIX





Technical Information

TI
1-1a
September 1970

PERFORMANCE AND MEASUREMENT TERMINOLOGY

There has long been a need for common definitions and terms related to performance and measurements in the field of process instrumentation. The following information is based on Foxboro Engineering Standards and SAMA documents, as well as documents available from ASA, ASME, IEEE, BSI, and IEC. It is hoped that in time all manufacturers, vendors, and users will adopt common terminology; however, in the meantime, this TI presents the definitions as accepted by The Foxboro Company unless otherwise noted.

The following list of terms has been purposely limited to include those which are frequently used. Some terms which have been areas of confusion and misinterpretation in the past have been elaborated on.

Accuracy - A number or quantity which defines the limit of error under reference conditions unless otherwise specified. See Figure 1.

Accuracy includes the combined conformity, hysteresis, and repeatability errors. The units being used are to be stated explicitly. For example, expressed in output units, a typical expression might be $\pm 1\%$; expressed in percent of output span, the expression could be $\pm 0.5\%$ percent of output span; expressed in percent of the upper-range value, it might read $\pm 0.5\%$ percent of upper-range value. For indicators, it might read $\pm 0.5\%$ percent of scale length; or in some cases it is expressed as a percent of actual output reading, such as $\pm 1.0\%$ percent of actual output reading.

Attenuation - A decrease in signal magnitude between two points, or between two frequencies. It is the reciprocal of gain, when the gain is less than one. It may be expressed as a dimensionless ratio or in decibels as 20 times the log of that ratio.

Deadband - The range through which an input can be varied without initiating response of pointer or pen or change in output. NOTE: Sensitivity, Resolution Sensitivity, and Ultimate Sensitivity have been used in the past to denote this or closely related quantities. However, these terms have become ambiguous, and in some cases their specific definitions represent conditions difficult to attain or recognize. Thus none of these terms are now used for any purpose by The Foxboro Company.

Deadtime - The interval of time between initiation of an input change or stimulus and the start of the resulting response. See Figure 6.

Drift - An undesired change in output over a period of time, which change is unrelated to input, operating conditions, or load. It is usually expressed as the change in output over a specified time with fixed input and operating conditions.

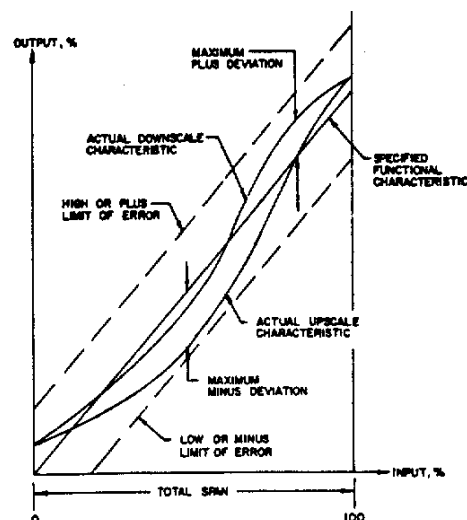


FIGURE 1 ACCURACY

Error - The difference between the indication and the true value of the measured signal. A positive error denotes that the indicated value of the instrument is greater than the true value. That is, error equals indication minus the true value.

Span Error is the difference between the actual span and the specified span and is usually expressed as the percent of specified span.

Zero Error is the error of a device operating under the specified conditions of use when the input is at the lower-range value. See also Zero Shift.

Gain - There are two types of gain, dynamic and static. **Dynamic Gain** is the magnitude ratio of the steady-state amplitude of the output signal from an element or system to the amplitude of the input signal to that element or system, for a sinusoidal input signal. It may be expressed as a ratio, or in decibels as 20 times the log of the ratio for a specified frequency. **Static Gain** is the ratio of an output change to an input change after steady state has been reached. Also referred to as **Zero Frequency Gain** which is the same as the dynamic gain as the signal frequency approaches zero. Either may be expressed as a ratio or in decibels.

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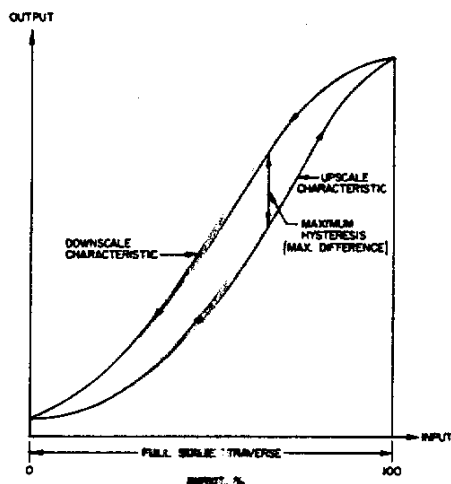


FIGURE 2 HYSTERESIS

Hysteresis - The maximum difference between the upscale and downscale indications of the measured signal during a full range traverse for the same input. See Figure 2. Note: The average difference has, in some cases, been used rather than the maximum difference. This practice is misleading and the values meaningless.

Impedance - Four types of impedance find common usage. Source Impedance is that presented to the input of the device by the source. Input Impedance is the impedance presented by a device to the source. Output Impedance is the impedance presented by a device

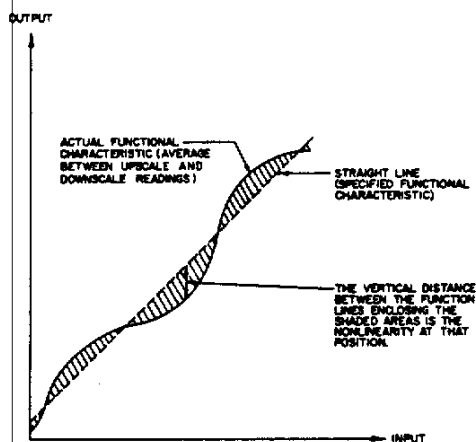


FIGURE 3 LINEARITY

to the load. Load Impedance is that presented to the output of a device by the load.

Interference - Interference is any spurious voltage or current arising from external sources and appearing in the circuits of a device. Two of the more common types of interference are Common Mode and Normal Mode. Common Mode is a form of interference which appears between measuring circuit terminals and ground. Common Mode is sometimes referred to as Longitudinal Mode. Normal Mode interference is a form which appears between measuring circuit terminals. Normal Mode is also referred to as Differential Mode and Transverse Mode.

TABLE 1
ILLUSTRATIONS OF THE USE OF RANGE AND SPAN TERMINOLOGY

TYPICAL RANGE	NAME	RANGE	LOWER RANGE VALUE	UPPER RANGE VALUE	SPAN	SUPPLEMENTARY DATA
0 to +100	---	0 to 100	0	+100	100	---
20 to +100	SUPPRESSED ZERO RANGE	20 to 100	20	+100	80	SUPPRESSION RATIO = .25
-25 to +100	ELEVATED ZERO RANGE	-25 to +100	-25	+100	125	---
-100 to 0	ELEVATED ZERO RANGE	-100 to 0	-100	0	100	---
-100 to -20	ELEVATED ZERO RANGE	-100 to -20	-100	-20	80	---

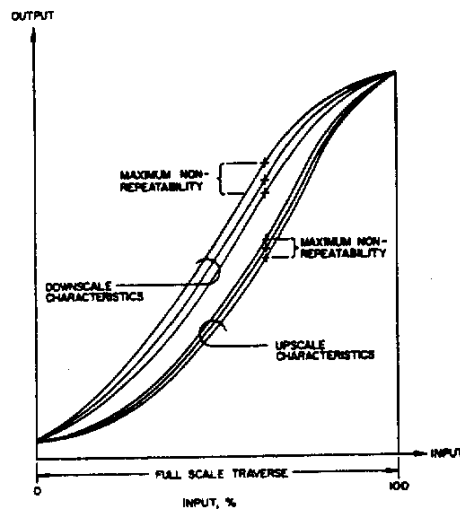


FIGURE 4 REPEATABILITY

Linearity - The closeness to which a curve approximates a straight line. See Figure 3. It is usually expressed as a nonlinearity, that is, a maximum deviation between an average curve and a straight line. The value of nonlinearity is referred to the output unless otherwise stated.

Pen Travel - The length of the path described by the pen in moving from one end of the chart scale to the other. The path may be an arc or a straight line.

Range - The region between the limits within which a quantity is measured, received, or transmitted, expressed by stating the lower- and upper-range values. Unless otherwise modified, the input range is implied.

A range is said to have an **Elevated Zero** when the zero value of the measured variable, measured signal, etc., is greater than the lower-range value.

A range is said to have a **Suppressed Zero** when the zero value of the measured variable is less than the lower-range value. Zero does not appear on the scale.

Note: Usage of the terms Elevation and Suppression by The Foxboro Company appears inconsistent with the newly adopted terminology. However, upon careful examination it is obvious this is not at all true. The SAMA definition speaks of "Elevated Zero Range" and "Suppressed Zero Range" while Foxboro terms are correctly stated as "Elevated Span" and "Suppressed Span."

Thus, the Foxboro term has reference to the calibrated span of the instrument, i.e. "The instrument has an elevated span" while the SAMA term would state "the range has a suppressed zero." Both statements correctly define an instrument with a calibration of perhaps 75-100 psi for a 10 to 50 ma output. (See TI 37-83b.)

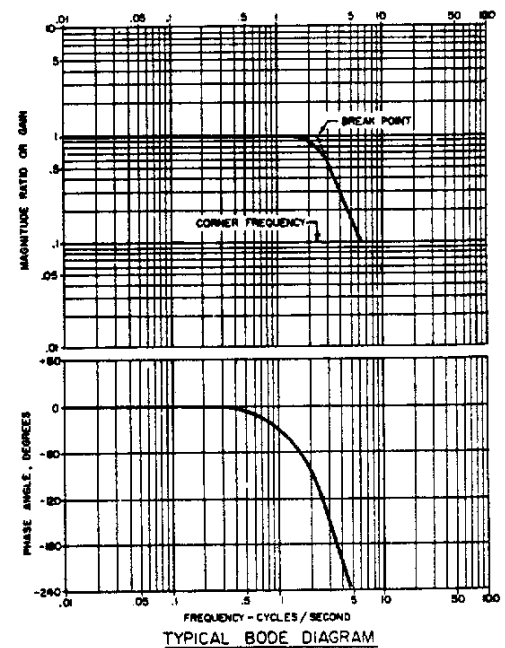


FIGURE 5 TYPICAL BODE DIAGRAM

To avoid confusion, considerable care should be exercised when discussing these nonzero based calibrations. Obviously the usage of the term "Elevation or Suppression" is no longer descriptive enough in itself to simply describe the calibration.

Table I is a convenient reference for the use of range and span terminology.

Repeatability - The closeness of agreement among a number of consecutive measurements of the output for the same value of the measured signal under the same operating conditions, approaching from the same direction for full range traverses. See Figure 4. It is usually expressed as a maximum nonrepeatability in percent of span. It does not include hysteresis.

Reproducibility - The closeness of agreement among repeated measurements of the output for the same value of the input made under the same operating conditions over a period of time, approaching from either direction. It is expressed as a maximum nonreproducibility in percent of span for a specified time. Normally, this implies a long period of time, but under certain conditions, the period may be a short time, during which drift may not be included. Reproducibility includes hysteresis, drift, and repeatability.

Response - Dynamic Response is the general behavior of the output of a device as a function of the input, both with respect to time. There are several dynamic response characteristics that fall into the general category of dynamic response, the two most common being Frequency Response and Step Response.

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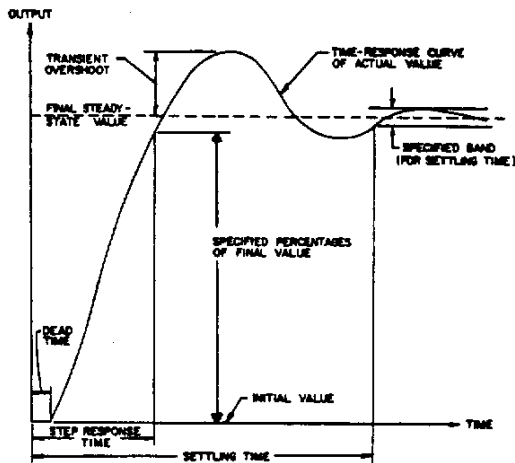


FIGURE 6 TYPICAL TIME RESPONSE OF A SYSTEM TO A STEP INCREASE OF INPUT

Frequency Response characteristic is the frequency-dependent relation, in both amplitude and phase between steady-state sinusoidal inputs and the resulting fundamental sinusoidal outputs.

Frequency Response is usually plotted on a Bode Diagram (See Figure 5), where the upper curve is a plot of magnitude ratio vs frequency, and the bottom curve is phase vs frequency. To establish a magnitude ratio curve an input value is selected (for example, 50 percent of full span) and this value is varied sinusoidally (say ± 5 percent). The input and output is recorded at various frequencies. Magnitude ratios are calculated by dividing the output amplitude by the input amplitude and plotting against the corresponding test frequencies.

When a Bode Diagram is not provided, the Frequency Response of a device may be specified as flat to the frequency at which the magnitude ratio has dropped to 0.707. (Example: Flat to 5 Hz.) Another way to specify Frequency Response is to convert the magnitude ratio to decibel units (dB).

$$\text{decibels} = 20 \log \frac{\text{Amplitude of output}}{\text{Amplitude of input}}$$

and state the frequency at which the plotted curve is down 3 dB. (For the same example as above: Down 3 dB at 5 Hz.)

The Frequency Response of a nonlinear gain device such as a square-root converter varies with the amplitude. Here the response is usually given for a single specified amplitude of input or output signal only.

Step Response Time is the time required for an output of an instrument to change from an initial value to a large specified percentage of a steady state either before or in the absence of overshoot, as a result of a step change to the input. The time is usually stated for 90, 95, or 99 percent change. See Figure 6.

Some confusion exists due to the commonly thought of 63 percent time constant. The 63 percent value has a special mathematical significance for first order elements or systems. Most process control instruments do not exhibit first order characteristics and other figures are more commonly used.

Whereas a single value of step-response time is often given (implying, for example, a 90 percent change in output in the absence of overshoot for an 80 percent input step change) such limited information may not be sufficiently definitive. The full response curve as shown in Figure 6 furnishes complete response information and as such is the most desirable. However, two response time values such as the time for a 50 percent change and the time for a 90 percent change will give a sufficient approximation of the shape of the response curve for most purposes.

Sensitivity - Obsolete term. See Deadband and Gain.

Signal to Noise Ratio - Ratio of signal amplitude to noise amplitude. For sinusoidal signals amplitude may be peak or rms. For nonsinusoidal signals, peak values should be used.

Span - The algebraic difference between the upper- and lower-range values.

Examples:	Range	Span
	0 to 150 F	150 F
	-20 to +200 F	220 F
	20 to 150 F	130 F

Time Constant - For a first order element or system, the time required for the output to complete 63.2 percent of the total rise or decay; as a result of a step change of the input. (See Response.)

Zero Shift - Any parallel shift of the input-output curve. An effect which is equal at all points in the range.

ORIFICE BETA RATIO LIMITS

This memo was written originally for Instrument Department personnel. In writing specifications for orifice meters, the conditions specified should be such that the orifice beta (d/D) ratio will not exceed 0.700. If the beta ratio is too large, some adjustment must be made in the specification, usually to specify a larger pipe size for the meter run, or perhaps to use a larger maximum differential pressure.

The memo presents a quick method of determining whether or not the beta ratio for the conditions in the specification will exceed the 0.700 limit. The section titled DISCUSSION AND DERIVATION lists the basic orifice equations for liquid, gas, and steam flow, and indicates how they might be manipulated to correct for conditions other than design.

Unless indicated otherwise, symbols used in the memo are based on those in Principles and Practice of Flowmeter Engineering, by L. K. Spink:

D	=	pipe diameter, inches
F _a	=	area correction factor = ratio of area of primary device at flowing temperature to that at 68°F (function of temperature and primary device material)
F _c	=	Reynolds Number correction factor (function of pipe Reynolds Number, beta ratio, pipe diameter, and orifice tap location)
F _m	=	manometer correction factor = correction for density of fluid on mercury (applies only to manometer type measurements; F _m = 1.00 if differential pressure transmitter is used.)
F _p	=	correction for compressibility of a liquid (function of reduced temperature and reduced pressure; usually negligible.)
G	=	gas specific gravity = mol. wt. of gas/mol. wt. of air
G _f	=	liquid specific gravity at flowing temperature
G _L	=	liquid specific gravity at base temperature (60°F)
h _m	=	maximum differential pressure, inches of H ₂ O (usually 100 inches of H ₂ O; may also be 50 or 200 inches; other values are used, but rarely)
h _w	=	actual differential pressure, inches of H ₂ O
K	=	SN, = constant (not used in Spink)
K _t	=	temporary combined constant (not used in Spink)
N	=	units constant for liquid flow (194.3 for Q in BPD. 5.667 for Q in gpm, etc.)
p _b	=	pressure at base conditions, psia (usually 14.696 psia)
p _f	=	flowing pressure, psia
p _{sat}	=	steam saturation pressure, psia (not used in Spink)
Q	=	volume flow rate, volume units at base conditions.
S	=	orifice coefficient (function of beta ratio and orifice tap location)
T _b	=	temperature at base conditions, °R (usually 520°R)
T _f	=	temperature at flowing conditions, °R
W _h	=	weight flow rate, lb/hr.

ORIFICE BETA RATIO LIMITS (Cont'd)

Y = gas expansion factor (function of differential pressure; beta ratio; heat capacity ratio, k, where $k = C_p/C_v$, and orifice tap location. For flange taps,

$$Y_1 = 1 - (0.41 - 0.35 \beta^4) h_w / 27.7 p f_1 k$$

subscript "one" indicates upstream tap condition)

Z_b = gas compressibility at base conditions (usually = 1.00)
 Z_f = gas compressibility at flowing conditions
 β = beta ratio = d/D = orifice diameter/pipe inside diameter
 γ_b = specific weight at base conditions, lb/ft³
 γ_f = specific weight at flowing conditions, lb/ft³
 μ = viscosity, cp

NOTE: Symbols with "prime", such as "h'm", are metric unit equivalents of the symbols above.



interoffice correspondence

date April 7, 1981to Instrument Department unit Des Plaines/London/Fieldfrom D. G. Plackmann unit Process Division - Des Plainessubject ORIFICE BETA RATIO LIMITS

A quick but accurate method has been developed for checking orifice beta ratios when writing Differential Instrument (604) specifications. In place of the Foxboro Flow Rule, the method uses the table attached, and - if necessary - a simple calculation done on a calculator.

The table attached lists limiting values of flow rate for various pipe diameters. The limiting values correspond to values of the meter maximum for orifices with flange taps at beta ratios of 0.700. These values serve as standards against which the meter maximum in the Differential Instrument specification can be measured.

The limiting values are based on assumed conditions; $G_f = G_L = G = 1.00$, $T_f = 560^\circ R = 100^\circ F$, $p_f = p_{sat} = 100 \text{ psia} = 85 \text{ psig}$, and $h_m = 100''$ H_2O , in English units, and on their approximate equivalents in metric units. If the conditions in the specification are the same as those assumed for calculation of the limiting values, and if the meter maximum in the specification is less than the limiting value listed for the specified line size, then the beta ratio must be less than 0.700. No further consideration is necessary. If the conditions are not identical, the tabulated value can be adjusted by multiplying by a correction factor. The factor is calculated from the appropriate terms of the Formulas, (A) thru (F), given below the table, and the corrected value is then used for the comparison.

For example, if a liquid flow orifice is specified, with $G_f = 0.65$, $G_L = 0.85$, a 1-1/2" meter run, the standard 100" H_2O differential, and a meter maximum of 1200 BPD, the correction factor, from Formula (A), is

$$\sqrt{G_f/G_L} = \sqrt{0.65/0.85} = 0.9485$$

and the corrected limiting value is

$$1525 \text{ BPD (tabulated)} \times 0.9485 = 1446 \text{ BPD}$$

Continued .../...

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 From: D. G. Plackmann
 Subject: Orifice Beta Ratio Limits
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Since the specified meter maximum is less than the corrected limiting value, the beta ratio is less than 0.700. If the same conditions were specified with a differential range of 0-50" H₂O, however, the correction would be

$$(\sqrt{G_F/G_L}) (\sqrt{h_m/100}) = (\sqrt{0.65/0.85}) (\sqrt{50/100}) = 0.6707$$

and $1525 \text{ BPD} \times 0.6707 = 1022 \text{ BPD}$

In this case, the specified meter maximum is greater than the corrected limiting value; therefore the beta ratio is greater than 0.700, and either the line size or the differential range must be increased.

If a gas flow orifice is specified, with $T_f = 100^\circ\text{F}$, $G = 0.35$, $P_f = 180 \text{ psig}$, a 1-1/2" meter run, the standard 100" H₂O, and a meter maximum of 500,000 SCFD, we know immediately without any calculation that the beta ratio is less than 0.700. The specified meter maximum is less than the limiting value of 615,000 SCFD listed in the table, and the correction factor, from Formula (C), is $\sqrt{(P_f/100)} (560/T_f) (1/G)$. Since P_f is greater than 100 psia, T_f is equal to 560°R, and G is less than 1.00, the value of the correction factor must be greater than 1.00; this can only increase the limiting value, which already is larger than the specified meter maximum.

If saturated steam is specified, with $P_f = P_{\text{sat}} = 50 \text{ psig}$, a 1-1/2" meter run, and a meter maximum of 1000 LB/HR, Formula (E) is used to calculate the correction factor. To minimize variations in the meter factor resulting from changes in the expansion factor, Y , as the flow rate and meter differential change, h_m in inches of water should not exceed P_f in psia. For this example, therefore h_m should be 50" H₂O. The correction then becomes

$$\begin{aligned} & (\text{Psat}/100)^{0.487} \sqrt{h_m/100} = (65/100)^{0.487} \sqrt{50/100} \\ & = 0.8108 \times 0.7071 = 0.5733, \end{aligned}$$

and $1338 \text{ LB/HR (tabulated)} \times 0.5733 = 767 \text{ LB/HR}$

Continued ...

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Since the specified meter maximum is larger than the corrected limiting value, and since the differential range cannot be increased, a larger line size is required.

In addition, factors are given which can be used with the tabulated values to find minimum limiting values for orifice plates, and maximum and minimum limiting values for venturis.

Details and derivation of the limiting values and the correction factors are given in the discussion following the table, all equations are based on Spink, Principles and Practice of Flowmeter Engineering, 9th Edition:

- 1) Eq. (1) for liquids is from Spink, p. 216, (Eq. 22)
- 2) Eq. (10) for steam is from p. 332, (Eq. 45)
- 3) Eq. (7) for gases is derived from (Eq. 45), with (Eq. 47), as follows:

If selected variables in (Eq. 45) are lumped together in a temporary constant, the equation can be rewritten:

$$W_h = 359 K_T \sqrt{\gamma_f} \sqrt{h_w} \quad (a)$$

Converting from weight units to volume units,

$$Q_h = W_h/\gamma_b = 359 K_T (\sqrt{\gamma_f/\gamma_b}) \sqrt{h_w} \quad (b)$$

Substituting for γ from (Eq. 47),

$$Q_h = \frac{359 K_T \sqrt{mw_{pf}/10.73 T_f Z_f} \sqrt{h_w}}{mw_{pb}/10.73 T_b Z_b} \quad (c)$$

Converting molecular weight to specific gravity,

$$G = mw/mw_{air} = mw/28.9644 \quad (d)$$

Continued .../..

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If Eqs. (c) and (d) are combined, and base conditions of 60°F and 1 atm. (519.67°R and 14.696 psia) are assumed, with $Z_b = 1.00$, the result is,

$$Q_h = \frac{359 K_T \sqrt{28.9644 G p_f / 10.73 T_f Z_f} \sqrt{h_w}}{28.9644 G \times 14.696 / 10.73 \times 519.67 \times 1.0} \quad (e)$$

$$Q_h = 7727 K_T \sqrt{p_f / T_f G Z_f} \sqrt{h_w} \quad (f)$$

Replacing K_T with the variables it represents, and letting $h_w = h_m$,

$$Q_m = 7727 S D^2 F_a F_m F_c Y \sqrt{p_f / T_f G Z_f} \sqrt{h_m} \quad (g)$$

Which forms the basis for Eq. (7) in the discussion.

If there are any questions, comments, or corrections, please let me know.

Continued .../...

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LIMITING VALUES FOR ORIFICE PLATES
 FLANGE TAPS, BETA = 0.700

D nom	D, in.	Liquid		Gas		Steam		D nom
		Q max BPD	Q max M ³ /HR	Q max MMSCFD	Q max NM ³ /HR.	W max LB/HR	W max KG/HR	
		(A)	(B)	(C)	(D)	(E)	(F)	
R		677.7	4.454	273.300	302.1	594.8	267.6	
1-1/2	1.500	1525	10.02	0.6150	679.8	1338	602.2	1-1/2
2	1.939	2548	16.75	1.028	1136	2236	1006	2
3	3.068	6379	41.92	2.573	2844	5599	2519	3
4	4.026	10,985	72.19	4.430	4897	9641	4338	4
6	6.065	24,930	163.8	10.05	11,110	21,880	9845	6
8	7.981	43,170	283.7	17.41	19,240	37,890	17,050	8
10	10.020	68,040	447.2	27.44	30,330	59,720	26,870	10
12	12.000	97,590	641.4	39.36	43,510	86,650	38,540	12
14	13.250	119,000	782.0	47.99	53,040	104,400	46,990	14
16	15.250	157,600	1036	63.57	70,260	138,300	62,240	16
18	17.250	201,700	1325	81.34	89,900	177,000	79,640	18
20	19.250	251,100	1650	101.3	112,000	220,400	99,180	20
24	23.250	366,300	2408	147.8	163,300	321,500	144,700	24

$$(A) = R D^2 \left(\sqrt{G_f/G_L} \right) \sqrt{h_m/100}$$

$$(C) = R D^2 \sqrt{(p_f/100) (560/T_f) (1/G)} \sqrt{h_m/100}$$

$$(E) = R D^2 (p_{sat}/100)^{0.487} \sqrt{h_m/100}$$

$$P_{f1} P_{sat} = p_{sia}, T_f = ^\circ R, h_m = \text{"H}_2\text{O}$$

$$(B) = R D^2 \left(\sqrt{G_f/G_L} \right) \sqrt{h'_m/2500}$$

$$(D) = R D^2 \sqrt{(p'_f/7) (311/T'_f) (1/G)} \sqrt{h'_m/2500}$$

$$(F) = R D^2 (p_{sat}/100)^{0.487} \sqrt{h'_m/2500}$$

$$p'_f = \text{kg/cm}^2, p_{sat} = \text{psia}, T'_f = ^\circ K, h'_m = \text{mmH}_2\text{O}$$

$$Q_{min} = Q_{max}/58.23$$

$$Q_{max} = Q_{max} \times 1.911$$

VENTURI

$$Q_{min} = Q_{max}/3.932$$

VENTURI

$$(p_{sat}/100)^{0.487} = \sqrt{4.432/v_g \text{ ft}^3/\text{lb}}$$

$$\text{For } (p'_f, \text{ Bar A/7}), (h'_m, \text{ kPa/25}), \text{ use}$$

$$(B), (F) \times 1.010, (D) \times 1.020$$

Continued .../...

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DISCUSSION AND DERIVATION

When a Differential Instrument specification is written, it is necessary to check the orifice beta ratio to insure that it falls within the limits of 0.10 and 0.70 required by UOP Specification 6-18-0. Traditionally, this check has been performed by using the Foxboro Flow Rule and working back from the meter maximum to find the beta ratio.

To determine the beta ratio with the Flow Rule requires a certain amount of manipulation and effort which, while not excessive, might be used better elsewhere - the value of the beta ratio is not really the information we required; at this point, our only concern is that the beta ratio falls within the specified limits.

It is virtually impossible to judge the value of the beta ratio simply by looking at the data in the specification, hence, if the traditional approach is taken, the Flow Rule offers the simplest solution. There is, however, another approach which allows the evaluation to be made directly from the data, or from a quick calculation based on the data. This method compares the meter maximum in the specification with the meter maximum which would result from the limiting beta ratio.

The method works best for liquid flows, which comprise the majority of Differential Instrument specifications for most process units. Gas flows are slightly more complicated, but still not difficult. Steam flows, if the steam is saturated, can be evaluated - but with slightly less accuracy than for liquids or gases when a correction is required. If the steam is superheated, the density is not a simple function of pressure or temperature; in this case, or for saturated steam if a more accurate limit is required, it is necessary to use the steam tables to obtain the specific volume and calculate a correction.

Liquid Flow

The orifice equation for liquids is

$$Q_m = S N D^2 F_a F_m F_c F_p (\sqrt{G_f / G_L}) \sqrt{h_m} \quad (1)$$

For Q in BPD, N = 194.3. If we assume that $F_a = F_m = F_c = F_p = G_f = G_L = 1.00$, and that $h_m = 100$ " H₂O, we have

$$\frac{Q_m}{\text{BPD}} = 194.3 S D^2 \sqrt{100} \quad (2)$$

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For flange taps at a beta ratio of 0.700, $S = 0.3488$. The equation for the maximum limiting value, therefore, is

$$Q_{\max} = 677.7 D^2 \quad (3)$$

BPD

The assumption that $F_a = F_m = F_c = F_p = 1.00$ is a standard assumption, made also when the Flow Rule is used for the check. The assumption that $G_f = G_L = 1.00$ is made for convenience: the specified values for specific gravities generally fall in the range of 0.7 to 1.0, so the 1.0 assumption is not too far off; and assuming 1.00 for both base and flowing gravities makes it easier to apply a correction when one is necessary.

For liquids with gravities other than 1.00, the correction factor is derived from Eq. (1). Given two sets of conditions, we can write

$$\frac{Q_{m2} = S_2 N_2 D^2 F_{a2} F_{m2} F_{c2} F_{p2} (\sqrt{G_{f2}/G_{L2}}) \sqrt{h_{m2}}}{Q_{m1} = S_1 N_1 D^2 F_{a1} F_{m1} F_{c1} F_{p1} (\sqrt{G_{f1}/G_{L1}}) \sqrt{h_{m1}}} \quad (4)$$

For any given installation, S , N , and D are fixed and constant. Since we assume that $F_a = F_m = F_c = F_p = 1.00$, we are left with

$$\frac{Q_{m2} = (\sqrt{G_{f2}/G_{L2}}) \sqrt{h_{m2}}}{Q_{m1} = (\sqrt{G_{f1}/G_{L1}}) \sqrt{h_{m1}}} \quad (5)$$

When this equation is used to correct the tabulated values of $Q_{\max}$, we have $G_{f1} = G_{L1} = 1.00$, and $h_{m1} = 100$. Therefore in terms of tabulated and specified values,

$$Q_{\max \text{ corr.}} = Q_{\max \text{ tab}} \left(\frac{\sqrt{G_{f \text{ spec}}/G_{L \text{ spec}}}}{\sqrt{G_{f \text{ spec}}/G_{L \text{ spec}}}} \right) \frac{\sqrt{h_{m \text{ spec}}/100}}{\sqrt{h_{m \text{ spec}}/100}} \quad (6)$$

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If necessary, then, the tabulated value can be corrected quickly and accurately with a few entries and operations on the calculator. The main advantage of the method, however, is that in any cases no calculation is necessary. Typical values for the correction factor, with h_m specified at 100 "H₂O, are:

Liquid	GL	T _f , °F	G _f	($\sqrt{G_f/GL}$) $\sqrt{100/100}$
C3	0.51	100	0.48	1.36
C8	0.71	100	0.69	1.17
C8	0.71	350	0.55	1.04
Platformer Charge	0.78	250	0.64	1.03
Parex Charge	0.86	350	0.72	0.99
Fuel Oil	0.97	350	0.86	0.96
Steam Condensate	1.00	450	0.83	0.91
Sulfolane	1.27	350	1.12	0.83

The table indicates that for lighter liquids (e.g., naphtha and lighter) and lower temperatures, the correction is greater than 1.00. For these cases, if the specified meter maximum is less than the tabulated value, there is no need to calculate a correction. Even for heavier liquids and higher temperatures, no correction is required as long as the specified value is less than about 80% of the tabulated value; these cases are more likely to occur on heavy oil units and utilities, and are thus in the minority.

Gas Flow

For gases, the orifice equation is

$$Q_m = 24 \times 7727 \text{ SD}^2 F_A F_M F_C Y \sqrt{p_f / T_f G Z_f} \sqrt{h_m} \quad (7)$$

SCFD

In this case, we assume that $F_A = F_M = F_C = Y = G = Z_f = 1.00$, that $p_f = 100$ psia = 85 psig, that $T_f = 560^\circ\text{R} = 100^\circ\text{F}$, and that $h_m = 100$ "H₂O. Then for a beta ratio of 0.700 and $S = 0.3488$, as for liquids

$$Q_{\max} = 273,300 D^2 \quad (8)$$

SCFD

As Eq. (1) was used via Eq. (4) to develop Eq. (6) for liquid flow, Eq. (7) can be used to develop the following:

$$Q_{\max} = Q_{\max} \sqrt{(p_{f\text{spec}}/100) (560/T_{f\text{spec}}) (1/G)} \sqrt{h_{m\text{spec}}/100} \quad (9)$$

SCFD tab

where p_f is in psia, and T_f is in °R.

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The values assumed for the variables - pressure, temperature, and specific gravity - once again were chosen for convenience and reasonable approximation of the conditions expected in the specifications. Instead of the two somewhat related variables found in the liquid flow equation, in this case there are three independent variables, with a much wider range of variation. This decreases the chance that the values assumed will match those in the specification, and makes it more difficult to determine when a correction is necessary. Typical values of the correction, for $h_m = 100'' \text{ H}_2\text{O}$, are given for various services in the following table:

Gas	Pf, psig	Tf, °F	G	$\sqrt{5.6p/TG}$	$\sqrt{100/100}$
Fuel Gas	50	100	0.77		0.92
Platformer Separator Gas	100	100	0.39		1.72
Platformer Recycle Gas	175	175	0.39		2.07
Isomar Recycle Gas	215	155	0.30		2.64
Depropanizer Net Gas	300	100	1.07		1.72
Naphtha Hydro. Recycle Gas	525	175	0.13		6.05
HDC Unibon Recycle Gas	1075	175	0.16		7.75

Higher pressures and lower specific gravities increase the correction factor. Conditions for Unibon and Hydrotreater services differ so much from those assumed for calculation of the limiting values that it is likely that a correction will be required in most such cases. If calculation of the correction becomes a problem, one or more high pressure cases could be added to the table of limiting values to allow a more direct comparison.

Steam Flow

The orifice equation for steam is

$$\frac{W_m}{\text{LB/HR}} = 359 S D^2 F_a F_m F_c Y \sqrt{Y_f} \sqrt{h_m} \quad (10)$$

Assuming again that $F_a = F_m = F_c = Y = 1.00$, that $h = 100'' \text{ H}_2\text{O}$, and that $\beta = 0.700$ and $S = 0.3488$, we have

$$\frac{W_{max}}{\text{LB/HR}} = 1252 D^2 \sqrt{Y_f} \quad (11)$$

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Values of Y_f may be obtained from its reciprocal, V_g , in the steam tables. For saturated steam at 100 psia, the specific volume, V_g , in ft^3/lb is 4.432. From this we have $\sqrt{Y_f} = 0.4750$, and, using this value in Eq. (11),

$$W_{\max} = 594.8 D^2 \quad \text{LB/HR} \quad (12)$$

For steam flow, as for gas flow, pressure has a significant effect on the value of Y_f , and it would be useful to have an equation which could be used to calculate a correction. Unlike gas flows, however, there is no simple relationship of Y_f versus pressure and temperature available in published literature. The Ideal Gas Law, $PV = nRT$, yields a value for Y_f which is reasonably accurate (within 2%) for saturated steam at 20 psia. At 600 psia, however, the accuracy falls to 22%. The accuracy does improve with increasing superheat, but the simple equation is not satisfactory under all conditions.

Fortunately, a plot of Y_f versus pressure (only) for saturated steam shows a relationship which is approximately linear (0.12 at 50 psia, 0.23 at 100 psia, and 0.44 at 200 psia). From this, an equation can be used to calculate a correction for Eq. (12):

$$\frac{\sqrt{Y_{f \text{ spec}}}}{\sqrt{Y_{f \text{ tab}} (p_s = 100)}} = (p_{\text{sat}}/100)^{0.487} \quad (13)$$

Where p_{sat} is in psia. The correction equation, with a differential range correction included, becomes

$$W_{\max} = 594.8 D^2 (p_{\text{sat}}/100)^{0.487} \sqrt{h_{\text{m spec}}/100} \quad \text{LB/HR} \quad (14)$$

Eq. (13), and therefore also Eq. (14), is accurate within +1%, -3% for $20 \leq p_{\text{sat}} \leq 1000$ psia:

$p_{\text{sat}}, \text{psia}$	$V_g, \text{ft}^3/\text{lb}$	$\sqrt{Y_f/Y_{100}}$	$(p_s/100)^{0.487}$	% Error
20	20.089	0.4697	0.4567	-2.85
50	8.515	0.7215	0.7135	-1.12
100	4.432	1.0000	1.0000	0.00
300	1.5433	1.6947	1.7075	0.75
500	0.9278	2.1856	2.1898	0.19
700	0.6554	2.6005	2.5797	-0.81
1000	0.4456	3.1538	3.0690	-2.76

Continued ...

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If greater accuracy is required, or if the steam is superheated, the correction can be calculated directly from the value of V_g given in the steam tables for the specified pressure and temperature:

$$W_{\max} = 594.8 D^2 \sqrt{4.432/V_{g_{\text{spec}}}} \sqrt{h_{m_{\text{spec}}}/100} \quad (15)$$

LB/HR

Minimum Flow

Because tight line sizes, with maximum allowable friction drop, generally provide the best overall economy, the minimum limiting flow, corresponding to a beta ratio of 0.100, is not generally encountered. On those occasions when a minimum limiting value is required, it can be calculated from the tabulated maximum value. For a beta ratio of 0.100 and flange taps, $S = 0.00599$, and

$$Q_{\min} = Q_{\max} (S_{0.100}/S_{0.700}) = Q_{\max} (0.00599/0.3488) \quad (16)$$

BETA BETA

$$Q_{\min} = Q_{\max} / 58.23 \quad (17)$$

It should be noted that conditions which give a correction greater than 1.00 will make the tabulated maximum values conservative, but will have the opposite effect on the minimum values.

Venturi Meters

Maximum and minimum limiting values for venturi meters also can be calculated from the tabulated values by a procedure similar to the above. For venturis, the beta ratio limits are 0.750 and 0.300; the corresponding values of S are 0.6667 and 0.08858:

$$Q_{\max} = Q_{\max} \times 0.6667/0.3488 = Q_{\max} \times 1.911 \quad (18)$$

VENTURI ORIFICE ORIFICE

$$Q_{\min} = Q_{\max} \times 0.08858/0.3488 = Q_{\max} / 3.938 \quad (19)$$

VENTURI ORIFICE ORIFICE

ENGINEERING DESIGN SEMINAR INSTRUMENTATION										PROJECT SPECIFICATION		
UOP Process Division • UOP Inc. 20 UOP Plaza — Algonquin & Mt. Prospect Roads • Des Plaines, Illinois 60016 • U.S.A.										NUMBER — 604 — SHEET OF BY APP'D DATE		
DIFFERENTIAL INSTRUMENTS												
ITEM NUMBER		1		2								
QUANTITY		1		1								
IDENTIFYING SYMBOL		FRC-1		FR-2								
SERVICE		FRESH FEED		NET GAS								
INST. WITH MTNG. YOKE	TYPE	DIFFERENTIAL RANGE 0-100" H ₂ O		A	0-2500mm H ₂ O	A	0-2500mm H ₂ O					
	TRANSMITTER TYPE		G		G							
	IND.	LARGE CASE	REC.									
CONT. FEAT.	INDICATOR — 6" DIAL											
	TYPE — PNEUMATIC PROPORTIONAL, RESET											
ACCES. SERIES	ACTION		Reverse		Direct							
	OUTPUT INDICATOR TYPE 0-10 SQUARE ROOT											
PRIN. DEVICE	AIR FILTER REGULATOR											
	NAMEPLATE UOP DWG. 6-129 METER FACTOR											
OPERATING CONDITIONS	TYPE — ORIFICE PLATE MATERIAL — 304/316 SS		X		MONEL							
	HOLDING RING MATERIAL —											
UOP DWG.	FLANGES — BY OTHERS SIZE, ANSI CLASS, FACING		6" 300 RF 125 Ra		6" 600 RF 125 Ra							
	LIQUID — BPD AT 60 °F		NORMAL FLOW		66 M ³ /HR		36,500 NM ³ /HR					
		GAS (SCFD) METER MAXIMUM		100 M ³ /HR		50,000 NM ³ /HR						
		FLOW TEMPERATURE, °F		116°C		38°C						
		SP. GRAVITY, 60 °F		0.768		(0.131)						
		SP. GRAVITY, FLOW TEMP.		0.681		—						
		FLOW PRESSURE, PSIG		15.5 kg/cm ²		55.5 kg/cm ²						
		ORIFICE PLATE 6-111 DETAIL		AA		AA						
		INSTRUMENT INSTALLATION		6-116-3		6-115-3						
REMARKS												
TYPE												
A = DIAPHRAGM TYPE B = BELLOWS TYPE (WITH DAMPENER) C = DIAPHRAGM TYPE (WITH DAMPENER) G = ELECTRONIC H = PNEUMATIC												
X = FURNISH INSTALLATION STRAIGHT RUN. UOP DWG. 6-112												
SUPPLIER (OR EQUAL)												
DRAWINGS REFERRED TO IN THIS SPECIFICATION		6-104		6-115		6-116		6-121		6-129		
		X 6-111		6-116		6-119		6-122		6-139		
		X 6-112		6-117		6-120		6-123		6-140		
ALL ITEMS SHALL COMPLY WITH THE REQUIREMENTS, TERMS AND CONDITIONS OF UOP SPECIFICATIONS 6-11, 6-18, 600. GENERAL NOTES, APPLIES TO THIS SPECIFICATION ☐ PROJECT SPECIFICATION												

SAMPLE CALCULATIONSLIQUID FLOW

Basis: Example Specification 604, Item 1, Fresh Feed.

Specifications

When writing specifications for Differential Instruments, we should verify that for the conditions specified, the orifice Beta Ratio will not be greater than 0.700 or less than 0.100.

From Spink (also shown as Equation (1), Page 6, ORIFICE BETA RATIO LIMITS memo):

$$Q = S N D^2 F_a F_m F_c F_p \left(\sqrt{G_f / G_l} \right) \sqrt{h_w} \quad \text{Eq. (A)}$$

We can see from Equation (A) that Q increases as S increases. S is a function of the Beta Ratio (β), and it can be shown that S increases as β increases. Therefore if we can determine the value of $Q_{\max}$ corresponding to $\beta = 0.700$, and if the value of Q_{spec} (the specified Meter Maximum) is less than that value, then the value of β for the specified conditions must also be less than 0.700, i.e.,:

$$\text{If } Q_{\text{spec}} < Q_{\beta=0.700}, \text{ then } \beta_{\text{spec}} < 0.700.$$

The Table, LIMITING VALUES FOR ORIFICE PLATES, on Page 5 of the memo ORIFICE BETA RATIO LIMITS provides an easy way to determine the value $Q_{\beta=0.700}$, which can then be compared with Q_{spec} . The table is based on the pipe I.D.'s listed, and on $\beta = 0.700$; $F_a = F_m = F_c = F_p = 1.00$; $G_f = G_l = 1.00$; and (for metric units) $h'_w = 2500 \text{ mm H}_2\text{O}$. For the 6" line size specified, Column (B) lists the limiting value of $Q_{\max}$ as 163.8 m³/hr. From this, the equivalent limiting value for the conditions specified can be found.

From Equation (A), (also shown as Equation (4), Page 7, ORIFICE BETA RATIO LIMITS memo):

$$\frac{Q_2 = S_2 N_2 D_2^2 F_{a2} F_{m2} F_{c2} F_{p2} \left(\sqrt{G_{f2} / G_{l2}} \right) \sqrt{h_{w2}}}{Q_1 = S_1 N_1 D_1^2 F_{a1} F_{m1} F_{c1} F_{p1} \left(\sqrt{G_{f1} / G_{l1}} \right) \sqrt{h_{w1}}} \quad \text{Eq. (B)}$$

(Note: the metric versions of Equations (A) and (B) are identical to the English equations shown, except that symbols Q' , N' , D' , and h'_w are used to distinguish them from their English counterparts. The only real difference is in the values of N and N' , which are adjusted as required to accommodate changes in the system of measurements.)

In Equation (B), let Conditions 1 represent the conditions used in calculating the value in the table, and let Conditions 2 represent the conditions in the specification. Since $\beta = 0.700$ in both cases, $S_2 = S_1$. Since Q is measured in m^3/hr in both cases, $N'_2 = N'_1$. Since the line size is the same, $D'_2 = D'_1$. The values of F_a , F_m , F_c , and F_p are generally not far from the value of 1.00 that was assumed for them for Condition 1, and they can be ignored. Finally, in both cases, $h'_{w2} = h'_{w1} = 2500 \text{ mm H}_2\text{O}$. If all known values are inserted and equal terms are cancelled, Equation (B) becomes

$$\frac{Q_{\beta=0.700(\text{spec})}}{Q_{\beta=0.700(\text{table})}} = \frac{(\sqrt{G_{\text{fspec}}/G_{\text{lspec}}})}{(\sqrt{1.00/1.00})} = \sqrt{G_{\text{fspec}}/G_{\text{lspec}}} \quad \text{Eq. (C)}$$

and

$$Q_{\beta=0.700(\text{spec})} = Q_{\beta=0.700(\text{table})} \times \sqrt{G_{\text{fspec}}/G_{\text{lspec}}} \quad \text{Eq. (D)}$$

For the conditions in the example specification,

$$Q_{\beta=0.700(\text{spec})} = 163.8 \times \sqrt{0.681/0.768} = 163.8 \times 1.075 = 176 \text{ m}^3/\text{hr} \quad \text{Eq. (E)}$$

Since the Meter Maximum specified for the Fresh Feed is $100 \text{ m}^3/\text{hr}$, which is less than $176 \text{ m}^3/\text{hr}$, β for the conditions specified must be less than 0.700.

Below the table on Page 5 of the memo is the equation:

$$Q_{\min} = Q_{\max} / 58.23 \quad \text{Eq. (F)}$$

$Q_{\min}$ is the value of Q corresponding to $\beta = 0.100$. Since $176 / 58.23 = 3 \text{ m}^3/\text{hr}$, and since $100 \text{ m}^3/\text{hr}$, is greater than $3 \text{ m}^3/\text{hr}$, β for the conditions specified must be greater than 0.100.

Therefore the Meter Maximum specified is acceptable.

Explanation of noninterpolating tables. The noninterpolating tables are designed to permit reading a factor to the accuracy justified by the data without interpolation. The operating data in the columns at the left are the dividing points at which the value of the derived factor on the half space above is just as close as that on the half space below. The derived factor is read from the right-hand column and is always within 0.0005 of the correct value. Tables 7, 8, and 9 are of this type.

TABLE 7

 F_a VALUES, CORRECTION FOR THERMAL EXPANSION OF PRIMARY DEVICE*

Temperature at Orifice, °F.

TABLE 12 (cont.)									Monel	Type 316 or Type 304	Corr. Factor, F_a
S VALUES FOR FLANGE, VENA CONTRACTA, RADIUS, OR CORNER TAPS											
d/D	S	Diff.	d/D	S	Diff.	d/D	S	Diff.			
.515	.1673		.552	.1953		.589	.2268				.992
.516	.1680	.0007	.553	.1961	.0008	.590	.2277	.0009			.993
.517	.1687	.0007	.554	.1969	.0008	.591	.2286	.0009	-281	-325	.994
.518	.1694	.0007	.555	.1977	.0008	.592	.2295	.0009	-191	-152	.995
.519	.1701	.0007	.556	.1985	.0008	.593	.2304	.0009	-108	-86	.996
.520	.1709	.0008	.557	.1993	.0008	.594	.2314	.0010	-33	-24	.997
.521	.1716	.0007	.558	.2002	.0009	.595	.2323	.0009	+38	+38	.998
.522	.1723	.0007	.559	.2010	.0008	.596	.2332	.0009	+104	+101	.999
.523	.1731	.0008	.560	.2018	.0008	.597	.2341	.0009	+168	+158	1.000
.524	.1738	.0007	.561	.2026	.0008	.598	.2350	.0009	230	213	1.001
.525	.1745	.0007	.562	.2034	.0008	.599	.2359	.0010	290	266	1.002
.526	.1753	.0008	.563	.2043	.0009	.600	.2369	.0009	348	318	1.003
.527	.1760	.0007	.564	.2051	.0008	.601	.2378	.0009	405	367	1.004
.528	.1768	.0008	.565	.2060	.0009	.602	.2388	.0010	461	414	1.005
.529	.1775	.0007	.566	.2068	.0008	.603	.2397	.0009	516	463	1.006
.530	.1782	.0007	.567	.2076	.0008	.604	.2407	.0010	571	512	1.007
.531	.1790	.0008	.568	.2085	.0009	.605	.2417	.0010	626	560	1.008
.532	.1797	.0007	.569	.2093	.0008	.606	.2426	.0009	680	607	1.009
.533	.1805	.0008	.570	.2101	.0008	.607	.2436	.0010	733	655	1.010
.534	.1813	.0008	.571	.2110	.0009	.608	.2445	.0009	785	703	1.011
.535	.1820	.0007	.572	.2119	.0009	.609	.2455	.0010	835	750	1.012
.536	.1828	.0008	.573	.2127	.0008	.610	.2464	.0009	886	796	1.013
.537	.1836	.0008	.574	.2136	.0009	.611	.2474	.0010	935	840	1.014
.538	.1843	.0007	.575	.2144	.0008	.612	.2484	.0010			1.015
.539	.1851	.0008	.576	.2153	.0009	.613	.2494	.0010			
.540	.1858	.0007	.577	.2162	.0009	.614	.2504	.0010			
.541	.1866	.0008	.578	.2170	.0008	.615	.2513	.0009			
.542	.1874	.0008	.579	.2179	.0009	.616	.2523	.0010			
.543	.1882	.0008	.580	.2188	.0009	.617	.2533	.0010			
.544	.1890	.0008	.581	.2197	.0009	.618	.2543	.0010			
.545	.1898	.0008	.582	.2205	.0008	.619	.2553	.0010			
.546	.1905	.0007	.583	.2214	.0009	.620	.2563	.0010			
.547	.1913	.0008	.584	.2223	.0009	.621	.2573	.0010			
.548	.1921	.0008	.585	.2232	.0009	.622	.2583	.0010			
.549	.1929	.0008	.586	.2241	.0009	.623	.2593	.0010			
.550	.1937	.0008	.587	.2250	.0009	.624	.2603	.0011			
.551	.1945	.0008	.588	.2259	.0009	.625	.2614	.0010			
.552	.1953	.0008	.589	.2268	.0009	.626	.2624	.0010			

If we had the same conditions specified, but with a Meter Maximum of 200 m³/hr, the value of β would be greater than 0.700, because 200 is greater than 176. To compensate for the higher Meter Maximum, we could increase the differential range to 0 - 5000 mm H₂O. Then we would have, instead of Equation (D):

$$\frac{Q_{\beta=0.700(\text{spec})}}{Q_{\beta=0.700(\text{table})}} = \left[\frac{(\sqrt{G_{\text{fspec}}/G_{\text{lspec}}})}{(\sqrt{1.00/1.00})} \right] \left(\frac{\sqrt{h_{w2}}}{\sqrt{2500}} \right) \quad \text{Eq. (G)}$$

from which:

$$Q_{\beta=0.700(\text{spec})} = 163.8 \left[\frac{(\sqrt{0.681/0.768})}{(\sqrt{1.00/1.00})} \right] \left(\frac{\sqrt{5000}}{\sqrt{2500}} \right) = 249 \text{ m}^3/\text{hr} \quad \text{Eq. (H)}$$

Since $200 < 249$, $\beta_{\text{spec}} < 0.700$. However, a differential range of 5000 mm H₂O is near the maximum for most standard differential instruments. A better choice, if possible, would be to expand the line size to 8". The Table in the memo lists the limiting value for an 8" line as 283.7 m³/hr. Using this in place of the 6" line's 163.8 m³/hr in Equation (E), we find that $Q_{\beta=0.700(\text{spec})} = 305 \text{ m}^3/\text{hr}$. Again, since $200 < 305$, $\beta_{\text{spec}} < 0.700$.

Orifice Bore Calculations

To calculate the orifice bore, the contractor or the orifice plate supplier uses Equation (A), or an equivalent equation, to calculate a value for S , or an equivalent coefficient, and from that value, determine a value for β . For the metric version of Equation (A), with Q' in m³/hr, D' in mm, and h'_w in mm H₂O, the value of N' is 0.0003959. From Table 7 of Spink, for $T = 116^\circ\text{C}$ (241°F), $F_t = 1.003$. Therefore:

$$S = \frac{100}{0.0003959 (154.1)^2 (1.003) (1.0) (1.0) (\sqrt{0.681/0.768}) \sqrt{2500}} = 0.1974 \quad \text{Eq. (I)}$$

From Table 12 of Spink, for this value of S , $\beta = 0.5546$. Since $\beta = d/D$, $d = (D)(\beta) = (154.1)(0.5546) = 85.46 \text{ mm}$.

Corrections for Actual Conditions

Before an orifice plate is installed, it should be checked carefully and its bore measured to insure that the orifice actually in agreement with the specification. If a mistake has been made

in the fabrication of the plate, and the orifice bore is not as calculated, the plate still may be useable. If the actual, measured beta ratio is known, Equation (A) can be used to find the flow rate directly from the operating conditions.

For any flowmeter which uses a 0 to 10 chart,

$$\frac{Q}{Q_{\max}} = \frac{\text{Chart Reading}}{10} \quad \text{Eq. (J)}$$

For a given orifice, S, N, and D are constants. Because there is no mercury in a differential pressure transmitter, and therefore no correction for liquid on mercury is required, $F_m = 1.00$. In general, F_p can be taken as 1.00, and if the temperature does not change much, F_s is constant (Table 7 of Spink shows no change in the value of F_s for 304/316 SS between 38 and 101°F). Therefore in Equation (B), if we let Condition 1 represent the Design Condition as given in the specification, and Condition 2 represent the actual Operating Conditions, cancel out the terms that are equal, and combine the result with Equation (J), we have:

$$\frac{\sqrt{h_w}}{\sqrt{h_m}} = \frac{Q}{Q_{\max}} = \frac{\text{Chart Reading}}{10} \quad \text{Eq. (K)}$$

Combining Equation (K) with Equation (A), and collecting constant terms,

$$Q = KF_c \left(\sqrt{G_f / G_l} \right) (\text{Chart Reading}) \quad \text{Eq. (L)}$$

where

$$K = SND^2 F_a F_m F_p (\sqrt{h_m} / 10)$$

The value of S is determined from the measured value of beta, using Table 12 in Spink, or its equivalent. The value of N is determined by the units of measurement used for Q, D, and h. F_c is a function of Reynolds Number, and is therefore itself a function of Q. F_c may or may not be sufficiently constant to be included in the constant K. We will examine F_c (and F_p) more closely below.

Assume that the measured dimensions of the orifice agree with those calculated from the specification, but the actual operating conditions are:

$T'_f = 95\text{ }^{\circ}\text{C}$	Chart Reading = 6.2 Chart Units
$P'_f = 17\text{ kg/cm}^2\text{ gauge}$	Viscosity = 0.25 cp
$G_f = 0.702$	Critical Temp., $T_c = 570\text{ }^{\circ}\text{K}$
$G_1 = 0.775$	Critical Press, $P_c = 32.7\text{ kg/cm}^2\text{ abs}$

The indicated flow rate (Chart Reading times Meter Factor) is based on the design conditions (the conditions in the specification). To correct this to an actual rate, based on actual conditions, we again use Equation (B), but this time, let Condition 1 represent the Design (Indicated) Condition, and let Condition 2 represent the Actual Condition. Since it is the indicated flow rate that we are correcting, the Chart Reading (h'_w) is the same for Condition 2 as it is for Condition 1. Again, similar to the derivation of Equation (C), above, $S_1 = S_2$, $N'_1 = N'_2$, and $D'_1 = D'_2$. And we also have $F_{m1} = F_{m2} = 1.00$. The indicated flow rate is

$$Q_{ind} = (\text{Chart Reading}) (\text{Meter Factor}) = (\text{Chart Reading}) (\text{Meter Maximum}/10) \quad \text{Eq. (M)}$$

Combining Equation (M) with Equation (B), and cancelling out S , N' , D'^2 , and F_m , we have:

$$\frac{Q_{corr}}{Q_{ind}} = \frac{Q_{corr}}{(\text{Chart Reading}) (\text{Meter Maximum}/10)} = \frac{(F_{a2} F_{c2} F_{p2}) (\sqrt{G_{f2}}/G_{12})}{(F_{a1} F_{m1} F_{c1} F_{p1}) (\sqrt{G_{f1}}/G_{11})} \quad \text{Eq. (N)}$$

From which:

$$Q_{corr} = \frac{(\text{Chart Reading}) (\text{Meter Maximum}) (F_{a2} F_{c2} F_{p2}) (\sqrt{G_{f2}}/G_{12})}{(10) (F_{a1} F_{c1} F_{p1}) (\sqrt{G_{f1}}/G_{11})} \quad \text{Eq. (O)}$$

Equation (O) is the fundamental equation for correction of the indicated flow of an orifice plate in liquid service to account for actual operating conditions. It is often written in the form:

$$Q_{corr} = K (\text{Chart Reading}) (\sqrt{G_{f2}}/G_{12}) \quad \text{Eq. (P)}$$

where F_{a2} , F_{c2} , and F_{p2} are assumed essentially constant, and

$$K = \frac{(\text{Meter Maximum})(F_{a2}F_{c2}F_{p2})}{(10)(F_{a1}F_{c1}F_{p1})(\sqrt{G_{f1}/G_{f2}})} \quad \text{Eq. (Q)}$$

In Item 1, the Meter Maximum is 100 m³/hr. Using this value in Equation (O), and inserting for Condition 1 the design values used in the bore calculation, we have:

$$Q_{\text{corr}} = \frac{(\text{Chart Reading})(100)(F_{a2}F_{c2}F_{p2})(\sqrt{G_{f2}/G_{f1}})}{(10)(1.003)(1.00)(\sqrt{0.681/0.768})} \quad \text{Eq. (R)}$$

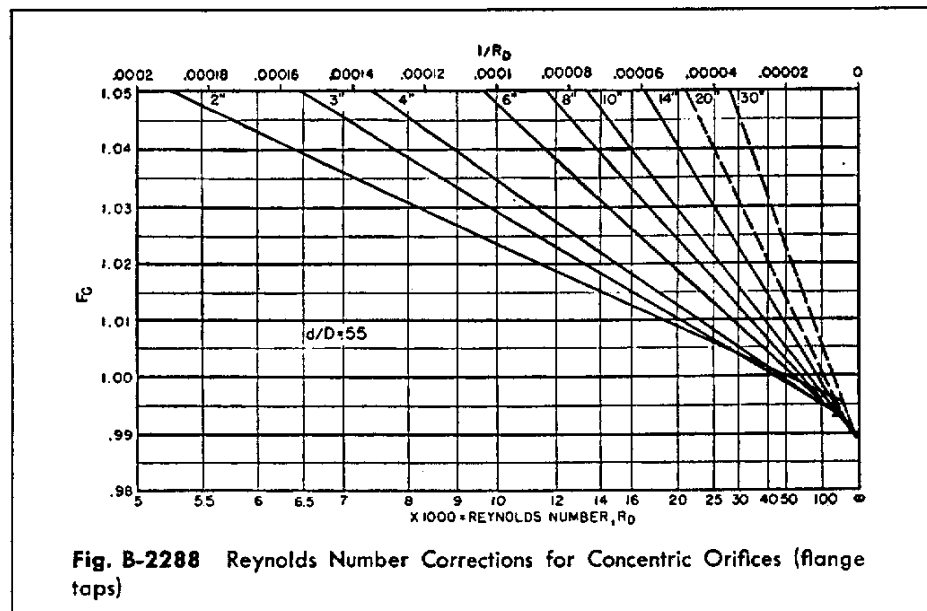
From which:

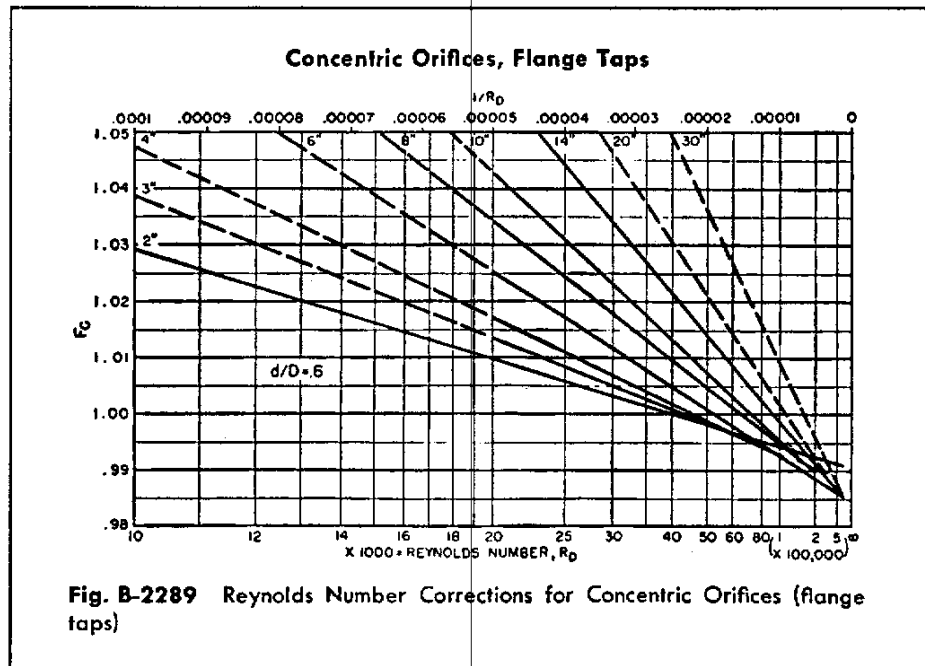
$$Q_{\text{corr}} = (9.279)(\text{Chart Reading})(F_{a2}F_{c2}F_{p2})(\sqrt{G_{f2}/G_{f1}}) \quad \text{Eq. (S)}$$

From the actual operating conditions given above, $G_f = 0.702$, and $G_1 = 0.775$. For evaluation of plant operations, these values would probably come from laboratory data. From Table 7 of Spink, for $T_f = 95^\circ\text{C}$ (203°F), $F_{a2} = 1.002$. This leaves only F_c and F_p to be determined.

For metric units, the Pipe Reynolds Number, R_D , is

$$R_D = \frac{353.7(W', \text{kg/hr})}{(D', \text{mm})(\mu, \text{cp})} = \frac{353.7(Q, \text{m}^3/\text{hr at base temp})(1000)(G_1)}{(D', \text{mm})(\mu, \text{cp})} \quad \text{Eq. (T)}$$





For this example, assume for the moment that the Design Condition Meter Factor is correct, so the 6.2 Chart Units = 62 m³/hr. Then:

$$R_D = \frac{353.7(62)(1000)(0.775)}{(154.1)(0.25)} = 441,000 \quad \text{Eq. (U)}$$

For $D = 6"$, and $R_D = 441,000$, from Spink, Figs. B-2288 and 2289, interpolating between the two figures for $\beta = 0.5546$, we find $F_{c2} = 0.990$. This leaves only F_{p2} .

For the operating conditions shown, $P_r = (17 + 1.033) / 32.7$, and $T_r = (95 + 273) / 570 = 0.65$. From Spink, Fig. B-2477, $F_{p2} = 1.00$.

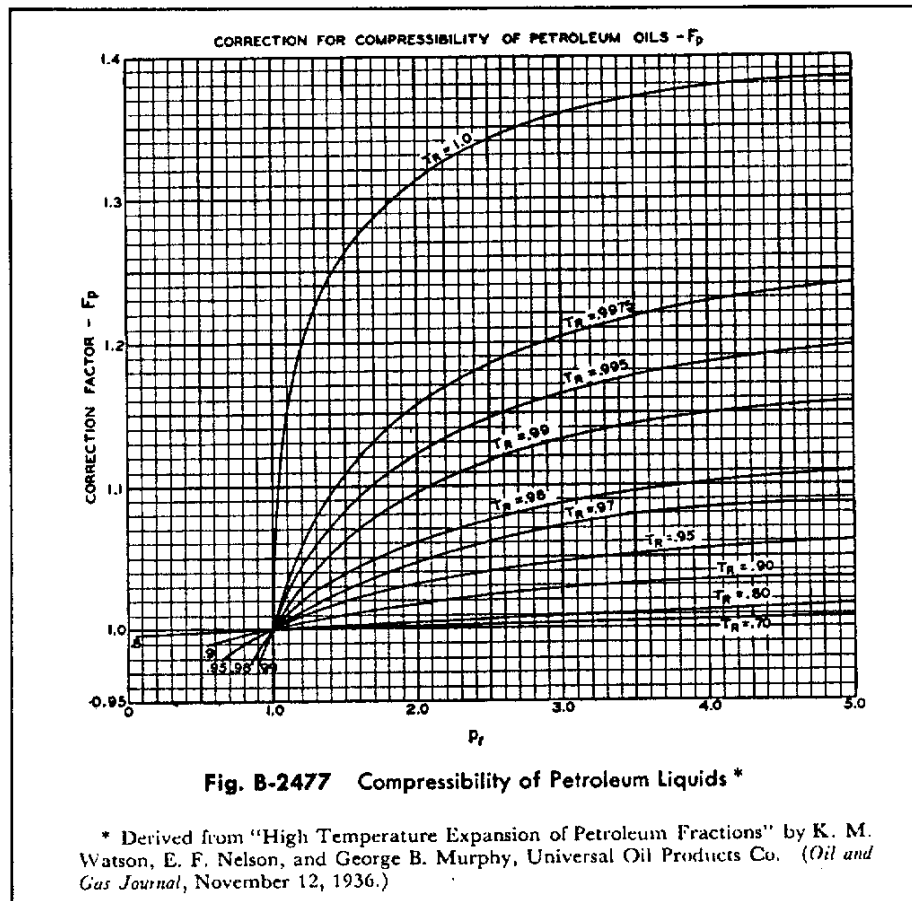
Inserting the values found for F_s , F_c , and F_p in Equation (S), we find:

$$Q_{\text{corr}} = (9.279)(\text{Chart Reading})((1.002)(0.990)(1.0))(\sqrt{0.702/0.775}) \quad \text{Eq. (W)}$$

$$Q_{\text{corr}} = (9.951)(\text{Chart Reading}) \quad \text{Eq. (X)}$$

Liquid Compressibility Factor, F_p

In general, as in the example above, $F_p = 1.00$. Fig. B-2477 of Spink shows that F_p is largest when $T_r = 1.00$; that is, when the operating temperature is equal to the critical temperature,



and the liquid phase and vapor phase are essentially the same. When T_r is less than 0.70, the F_p correction is very close to 1.00.

We are most concerned about the F_p correction when measuring charge and product streams. In these cases the temperature is generally about 40 °C, or 313 °K. If T_c is greater than 447°K when $T = 313$ °K, then T_r will be less than 0.70, and F_p will be essentially 1.00. For pentane, $T_c = 469$ °K. Since T_c generally increases with increasing molecular weight, it is likely that F_p can be ignored for hydrocarbon streams that are C_5 and heavier.

For hydrocarbon streams that are C_4 and lighter, the opposite is true, especially at higher temperatures. If propane, with $T_c = 370$ °K and $P_c = 43.4$ kg/cm²abs, is flowing at 95 °C and 55 kg/cm²g, we would have $T_r = (95 + 273) / 370 = 0.995$, and $P_r = (55 + 1.033) / 43.4 = 1.29$. From Fig. B-2477 of Spink, the value of F_p for these conditions would be 1.06.

A correction factor of 1.06, or 6%, must be considered significant. A correction of 0.6% may, or may not, be significant. The excerpt from Spink on the following page discusses when small corrections are justified. (Note: the "nonviscous flows" mentioned are measurements in which F_c is ignored.)

Reynolds Number Correction, F_c

If operating conditions are stable, it may be acceptable to determine the individual correction factors and combine them into one constant. The correction factor that is most likely to be variable is F_c . Because of the square root relationship between differential pressure and flow, the rangeability of a flow measurement is about 3 to 1 at best, and certainly not more than 5 to 1 (10 Chart Units down to 2 Chart Units). Fig. B-2288 in Spink indicates that for an equivalent 3 to 1 change in R_D , F_c changes more when the pipe size is larger. For the 2" line, changing R_D from 10,000 to 30,000 changes F_c from 1.023 to 1.004. For the 30" line, changing R_D from 30,000 to 90,000 changes F_c from 1.047 to 1.007.

If the change in F_c is relatively large over the 3 to 1 range, it cannot be included in the meter constant. However, if the viscosity and specific gravity are constant, then R_D is a linear function of Q . Fig. B2288 of Spink shows F_c as a linear function of $1/R_D$. So it should be possible to develop a fairly simple relationship for F_c in terms of Chart Reading, which can then be incorporated into the correction for Q . We can write:

$$F_c = (K_1 / \text{Chart Reading}) + K_2 \quad \text{Eq. (Y)}$$

where K_1 and K_2 are constants. For this example, from Fig. B-2288, we would have the following:

- 1) At 6.2 Chart Units, $R_D = 441,000$ and $F_c = 0.990$.
- 2) At 1.0 Chart Units, $R_D = 71,000$ and $F_c = 0.998$.

Using these values in Equation (S), we have:

$$0.990 = (K_1 / 6.2) + K_2 \quad \text{Eq. (Z)}$$

SMALL CORRECTIONS FOR VISCOSITY

When they are warranted. (For exceptions, see p. 139.) On process balances, acceptance tests, and other exacting work, if specific gravity, temperature, and viscosity are stable and measurable, spot corrections (however small) should be made.

Precautions necessary to attain laboratory precision. Unless precautions not ordinarily followed in plant measurement are applied, however, this refinement may prove to be a waste of time. All of the requirements for correct installation should be followed rigidly. The density of the flowing liquid at the flowing temperature should be determined with accuracy. The value of $\sqrt{G_f}$ taken from the usual curves may easily be in error by two or three per cent because of varying expansion characteristics. Unless an accurate measurement of density at the flowing conditions is available, one of the methods described on pages 262-278 should be used for obtaining the value of $\sqrt{G_f}$. If temperatures are above the curves in Figure B-2248, page 141, the magnitude of the compressibility correction F_c should be investigated.

The recorder, indicator, or other instrument should be installed near the elevation of the orifice and the lead lines should be insulated together. If the primary and secondary devices are not at the same elevation, small differences in the temperature of the liquid in the lead lines may completely smother the F_c correction.

For instance, suppose a meter is operating at 15 inches differential and is located 10 feet below the orifice. Assume that the lead lines are filled with 60° A.P.I. gasoline and the temperature of one lead line is 60° F. and the other is 65° F. This difference in temperature would cause an unbalanced head of over 0.3 inch with a corresponding error of more than 1%. If the downstream lead were at the lower temperature the meter would be low, and vice versa. This error does not show up when following the customary method of zeroing the meter.

If the specific gravity of the flowing liquid changes, the meter lead lines should be thoroughly flushed of all the old liquid. Otherwise mixing will occur more rapidly in one lead line than in the other and an error similar to that described in the foregoing paragraph will result.

Unless the flow readings are in the middle or upper part of the chart or unless the meter is mounted in a temperature- and humidity-controlled cabinet, the errors due to humidity expansion of the chart and the temperature expansion of the mercury may outweigh the gain in accuracy attained by applying the F_c factor on nonviscous * flows.

To sum up the precautions necessary for laboratory precision:

1. The d/D ratio should be low.
2. The pipe should be commercially smooth but not polished.
3. The meter lead lines should be short and insulated together.
4. The specific gravity at flowing temperature should be determined precisely.
5. The operating differential should be well out on the scale.
6. The entire installation should be located in a place of uniform ambient temperature, away from sources of radiant heat and protected from direct sunlight.
7. The meter chart should be kept at a reasonably uniform relative humidity or checked against a metal scale.
8. All details of the installation should be made in the strictest accordance with the manufacturer's instructions.

Under the above conditions, application of the F_c factor on nonviscous flows is justified.

$$0.998 = (K_1/1.0) + K_2 \quad \text{Eq. (AA)}$$

Solving Equations (Z) and (AA) simultaneously for K_1 and K_2 we find that $K_1 = 0.01$ and $K_2 = 0.988$, so Equation (Y) becomes:

$$F_c = (0.010/\text{Chart Reading}) + 0.988 \quad \text{Eq. (AB)}$$

Over the 3 to 1 range from 10 Chart Units to 3 Chart Units in this example, the value of F_c only changes from 0.989 to 0.991. For all practical purposes, we may as well assume that F_c is constant at 0.990, as was done in Equation (X). This approach is most useful if composition and temperature are controlled, so viscosity and specific gravity are constant.

GAS FLOW

Basis: Example Specification 604, Item 2, Net Gas.

Specifications

Similar to Equation (7), Page 8, ORIFICE BETA RATIO LIMITS memo (Note: the equation in Spink is not in this form):

$$Q = S N_Q D^2 F_a F_m F_c Y \left(\sqrt{\frac{P_f}{T_f Z_f}} \right) \sqrt{h_w} \quad \text{Eq. (AC)}$$

where, for flange taps, with P_f measured upstream of the orifice,

$$Y = 1 - (0.41 + 0.35 \beta^4) \left(\frac{h_w}{N_Y P_f k} \right) \quad \text{Eq. (AD)}$$

and where N_Q is the appropriate constant for the flow equation, N_Y is the appropriate constant for the Y equation, and $k = C_p / C_v$, the ratio of heat capacities at constant pressure and constant volume. For metric units (Q in NM^3/hr , P_f in kg/cm^2 abs, T_f in $^\circ\text{K}$, and h_w in $\text{mm H}_2\text{O}$), $N_Q = 0.1789$ and $N_Y = 10,000$.

As was the case with the liquid flow example above, when writing specifications we should verify that for the conditions specified, the orifice Beta Ratio will not be greater than 0.700 or less than 0.100. Again, we can use the Table, LIMITING VALUES FOR ORIFICE PLATES, on Page 5 of the memo ORIFICE BETA RATIO LIMITS. For a 6" line, the Table

lists $Q_{\max}$ as 11,110 NM³/hr. This is based on $P'_f = 7 \text{ kg/cm}^2 \text{ abs}$, $T'_f = 311 \text{ }^\circ\text{K}$, $Z_f = 1.00$, and $G = 1.00$. The conditions in the specification are $P'_f = 55.5 \text{ kg/cm}^2 \text{ gauge}$, $T'_f = 38 \text{ }^\circ\text{C}$, and $G = 1.00$.

Also, as was the case with Equation (B) for liquid flow, we can develop an equation for gas flow:

$$\frac{Q_2 = S_2 N_2 D_2^2 F_{a2} F_{c2} Y_2 \left(\sqrt{\frac{P_{f2}}{T_{f2} G_{f2} Z_{f2}}} \right) \sqrt{h_{w2}}}{Q_1 = S_1 N_1 D_1^2 F_{a1} F_{c1} Y_1 \left(\sqrt{\frac{P_{f1}}{T_{f1} G_{f1} Z_{f1}}} \right) \sqrt{h_{w1}}} \quad \text{Eq. (AE)}$$

If we let Conditions 1 represent the conditions used in calculating the value in the table, and let Conditions 2 represent the conditions in the specification, we have, since $S_2 = S_1$, and $N'_2 = N'_1$:

$$\frac{Q'_{\beta=0.700(\text{spec})}}{Q'_{\beta=0.700(\text{table})}} = \frac{Q'_{\max \text{ spec}}}{11,110} = \frac{F_{a2} F_{c2} Y_2 \left(\sqrt{\frac{56.53}{(311)(0.131)(1.00)}} \right) \sqrt{2500}}{(1.00)(1.00)(1.00)(1.00) \left(\sqrt{\frac{7}{(311)(1.00)(1.00)}} \right) \sqrt{2500}} \quad \text{Eq. (AF)}$$

From which, if F_a , F_c , and Y are 1.00, $Q'_{\max \text{ spec}} = 87,000 \text{ NM}^3/\text{hr}$. Since the specified Meter Maximum, 50,000 NM³/hr, is less than $Q'_{\max \text{ spec}}$, the value of β is less than 0.700. Also, since the specified Meter Maximum is greater than $Q'_{\max \text{ spec}} / 58.23$, β is greater than 0.100.

The contractor or orifice plate supplier would use Equation (AC), or its equivalent, to determine the orifice bore. For $T'_f = 38 \text{ }^\circ\text{C}$, and a Monel orifice plate, Spink, Table 7, indicates that $F_a = 1.000$. For $F_c = Y = Z_f = 1.00$, solving for S , Equation (AC) becomes:

$$S = \frac{50,000}{0.1789(154.1)^2(1.000)(1.00)(1.00)(Y) \left(\sqrt{\frac{56.53}{(311)(0.131)(1.00)}} \right) \sqrt{2500}} = \frac{0.1998}{Y} \quad \text{Eq. (AG)}$$

If $Y = 1.00$, $S = 0.1998$. From Table 12 of Spink, for this value of S , $\beta = 0.5576$, from which $d = (D)(\beta) = (154.1)(0.5576) = 85.93 \text{ mm}$.

The memo on ORIFICE BETA RATIO LIMITS indicates that the value of h'_w , in inches of H_2O , should be less than the value of P'_f in psia, to limit the change in Y as the flow rate (and h'_w) change. For metric units, h'_w in mm H_2O and P'_f in kg/cm^2 abs, the ratio h'_w/P'_f should be less than 360. Since in this example $P'_f = 56.53$ kg/cm^2 abs, the differential range of 2500 mm H_2O is well within the limit.

At lower pressures, as h'_w approaches $360 P'_f$, it may be desirable to include the Y correction in the calculation for S , to make the Meter Factor more nearly correct at the design flow rate. UOP Standard Specifications state that the heat capacity ratio, C_p/C_v , may be taken as 1.3, if no value for " k " is listed in the Project Specification. (1.3 is a reasonable approximation for hydrocarbon streams; for air or nitrogen, the value is 1.4.) Since S , and therefore β , is a function of Y , and Y is a function of β , an iterative procedure must be used to solve both Equations (AC) and (AD) simultaneously.

Equation (AG) indicates that when $Y = 1.00$, $\beta = 0.5576$. Using this value in Equation (AD), with $P'_f = 56.53$ kg/cm^2 abs, and $k = 1.3$,

$$Y = 1 - (0.41 + 0.35(0.5576)^4) \left(\frac{h'_w}{(10,000)(56.53)(1.3)} \right) = 1 - 6.039 \times 10^{-7} h'_w$$

Eq. (AH)

If the Y correction is used in determining the orifice bore, it is probably best to use the value of Y (and h'_w) which maximizes the accuracy of the Meter Factor at the normal, design flow rate, rather than at the maximum flow. From Equation (K),

$$\frac{\sqrt{h'_w}}{\sqrt{h'_m}} = \frac{\sqrt{h'_w}}{\sqrt{2500}} = \frac{Q'}{Q_{\max}} = \frac{36,500}{50,000}$$

Eq. (AI)

From which $h'_w = 1332$ mm H_2O . Using this value for h'_w in Equation (AF), $Y = 0.999$. Using 0.999 instead of 1.00 for Y in Equation (AG), $S = 0.2000$. From Table 12 of Spink, for this value of S , $\beta = 0.5577$.

Since this value of β is within 1% of the previous value, we can consider the calculation as having converged. If the difference were greater than 1%, the new β would be used to recalculate another new value for Y , which would in turn be used to determine another new value for β , and so on.

Corrections for Actual Conditions

Assume that the orifice has been measured, that the dimensions are correct, and that it is operating at the following actual conditions:

$$\begin{array}{ll}
 T'_f &= 25^\circ\text{C} = 298^\circ\text{K} & \text{Chart Reading} &= 7.5 \text{ Chart Units} \\
 P'_f &= 50 \text{ kg/cm}^2 \text{ gauge} & \text{Viscosity} &= 0.015 \text{ cp} \\
 Z'_f &= 1.016 & \text{Mol Wt.} &= 3.5 \\
 C_p / C_v &= 1.39
 \end{array}$$

From Table 7 of Spink, for $T = 25^\circ\text{C}$ (77°F) and Monel, $F_a = 1.000$.

Reynolds Number Correction, F_r

To find F_r , we must first find R_D from Equation (T). For gases, the specific weight may be calculated as follows:

$$\gamma'_f, \text{ kg/m}^3 = \frac{\text{mw} (P'_f)}{0.08479 T'_f Z'_f} \quad \text{Eq. (AJ)}$$

At base conditions, $P'_b = 1.0332 \text{ kg/cm}^2 \text{ abs}$, $T'_b = 273.15^\circ\text{K}$, and $Z'_b = 1.000$. Using these values in Equation (AJ), we have:

$$\gamma'_b, \text{ kg/NM}^3 = \frac{\text{mw} (1.0332)}{0.08479 (273.15) (1.000)} = \frac{\text{mw}}{22.414} \quad \text{Eq. (AK)}$$

In this example, with $\text{mw} = 3.5$, $\gamma'_b = 0.1562 \text{ kg/m}^3$. For a Chart Reading of 7.5, and a Meter Maximum of 50,000 NM^3/hr (Meter Factor = 5000),

$$W', \text{ kg/hr} = (Q', \text{ NM}^3/\text{hr}) (\gamma'_b, \text{ kg/NM}^3) = (7.5)(5000)(0.1562) = 5860 \quad \text{Eq. (AL)}$$

Using this value, the pipe diameter, and the viscosity in Equation (T),

$$R_D = \frac{353.7(5860)}{(154.1)(0.015)} = 896,000 \quad \text{Eq. (AM)}$$

From Spink, Fig. 2288 and 2289, $F_c = 0.990$.

As before, if the composition and viscosity are constant, at 10 Chart Units, $R_D = 1,190,000$ and $F_c = 0.990$, while at 3 Chart Units, $R_D = 358,000$ and $F_c = 0.991$. This is a small change,

so we may assume R_D is constant at 0.990. Likewise as before, taking Condition 1 as the Design Condition as given in the specification, and Condition 2 as the actual Operating Condition, with $S_1 = S_2$, $N'_{Q2} = N'_{Q1}$, $D'_2 = D'_1$, $h'_{w2} = h'_{w1}$, $F_{a2} = F_{a1} = 1.00$, $F_{c2} = 0.990$, $F_{c1} = 1.00$, $Y_1 = 0.999$, and $Z_{f1} = 1.00$, Equation (AE) becomes:

$$\frac{Q_{\text{corr}}}{(\text{Chart Reading})(5000)} = \frac{(1.000)(0.989)(Y) \left(\sqrt{\frac{P_{f2}}{T'_{f2} G_{f2} Z_{f2}}} \right)}{(1.00)(1.00)(0.999) \left(\sqrt{\frac{56.53}{(311)(0.131)91.00}} \right)} \quad \text{Eq. (AN)}$$

From which:

$$Q_{\text{corr}} = (4200)(Y) \left(\sqrt{\frac{P'_{f2}}{T'_{f2} G_{f2} Z_{f2}}} \right) (\text{Chart Reading}) \quad \text{Eq. (AO)}$$

For the actual conditions given, solving Equation (K) for h'_w ,

$$h'_w = \left(\frac{\text{Chart Reading}}{\text{Meter Maximum}} \right)^2 (h'_m) = \left(\frac{7.5}{10} \right)^2 (2500) = 1406 \quad \text{Eq. (AP)}$$

Using this value in Equation (AH),

$$Y = 1 - 6.039 \times 10^{-7} (1406) = 0.999 \quad \text{Eq. (AQ)}$$

And Equation (AO) becomes, for the actual conditions given:

$$Q_{\text{corr}} \text{ NM}^3/\text{hr} = (4200)(0.999) \left(\sqrt{\frac{51.03}{(298)(3.5/28.96)(1.016)}} \right) (7.5) \quad \text{Eq. (AR)}$$

$$Q_{\text{corr}} = 37,150 \text{ NM}^3/\text{hr} \quad \text{Eq. (AS)}$$

Steam Flow (or Gas Flow in Weight Units)

For W' in kg/hr, D' in mm, γ'_f in kg/m³, and h'_w in mm H₂O,

$$W' = 0.02151 S D'^2 F_a F_m F_c Y \left(\sqrt{\gamma'_f} \right) \sqrt{h'_w} \quad \text{Eq. (AT)}$$

The procedure used to correct the indicated flow for steam is similar to that used for liquid and lgas flows. For steam service, instead of calculating γ'_f , it is obtained from steam tables.

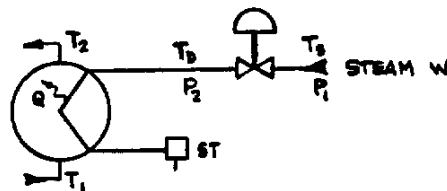
JUNE 27, 1980

TO: INSTRUMENT DEPT.

FROM: O. G. PLACKMANN

SUBJECT: SIZING STEAM REBOILER CONTROL VALVES

THE FOLLOWING IS BASED ON A REPORT DATED FEB. 17, 1966, ON THE SAME SUBJECT. ITS PURPOSE IS TO CONFIRM AND CLARIFY THE RESULTS OF THE ORIGINAL REPORT. THE CONCLUSION IN THE ORIGINAL REPORT WAS THAT THE STANDARD APPROACH TO VALVE SIZING (VALVE $C_v = 2 \times \text{NORMAL FLOW } C_v$) COULD BE USED IF THE DESIGN PRESSURE CHOSEN FOR THE REBOILER STEAM WAS THE PRESSURE WHICH GAVE A ΔT EQUAL TO 85% OF THE MAXIMUM ΔT (ΔT WHEN REBOILER STEAM PRESSURE = STEAM HEADER PRESSURE). USING THE SAME DATA AS THE ORIGINAL REPORT, NEW CALCULATIONS WERE MADE AND THE RESULTS PLOTTED TOGETHER ON ONE GRAPH. THE PLOT INDICATES THAT THE ORIGINAL CONCLUSION IS STILL VALID AND THAT NO CHANGES IN SIZING PROCEDURE ARE REQUIRED.

DATA:

EXAMPLE	P ₁ PSIG	T ₁ °F	T ₂ °F	T _D DESIGN* °F	Q 10 ⁶ BTU/HR	W LB/HR	SPEC 401 PROJECT	ITEM	CUSTOMER
1	20	210	210	250	2.45	2600	8215	SP-E2	DENKA
2	150	205	205	361	1.65	1920	8521	A-E-13	MURPHY
3	400	266	266	422	7.78	10,050	8500	11-E-13	GULF
4	450	320	380	455	12.0	15,600	8521	A-E-B	MURPHY

* ALL DATA FROM EXCHANGER SPECIFICATIONS.

SIZING STEAM REBOILER CONTROL VALVES (CONT'D)

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EQUATIONS:

(1) $Q = UA \Delta T$

(2) $W = Q / h_{fg}$

(3) $C_v = \frac{W(1 + 0.0007 T_{sh})}{1.83 C_F P_1 (Y - 0.148 Y^3)}$

$Y = \frac{1.63}{C_F} \sqrt{\frac{\Delta P}{P_1}}$

WHERE Q = HEAT TRANSFERRED, BTU/HR U = HEAT TRANSFER COEFFICIENT, BTU/HR·FT²·°F A = AREA, FT² ΔT = TEMPERATURE DIFFERENCE, $T_{STEAM} - T_{PROCESS}$, °F h_{fg} = HEAT OF VAPORIZATION, BTU/LB W = STEAM FLOW RATE, LB/HR T_{sh} = STEAM SUPERHEAT, °F C_F = CRITICAL FLOW FACTOR P_1 = STEAM HEADER PRESSURE, PSIA Y = AS DEFINED; $Y_{MAX} = 1.5$ $\Delta P = P_1 - P_2$ P_2 = REBOILER STEAM PRESSURE, PSIAASSUMPTIONS:

- SUPERHEAT IS NEGLECTABLE ($T_{sh} = 0$).
- SUBCOOLING IS NEGLECTABLE.
- HEAT TRANSFER COEFFICIENT (U) IS CONSTANT.
- CRITICAL FLOW FACTOR (C_F) = 1.0.

SIZING STEAM REBOILER CONTROL VALVES (CONT'D)

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DGPCALCULATION PROCEDURE:

- 1) FIND ΔT_{MAX} , ΔT WHEN $T_D = T_S$.
- 2) SELECT APPROPRIATE VALUE OF P_2 OR T_D .
- 3) FIND CORRESPONDING T_D OR P_2 IN STEAM TABLE.
- 4) CALCULATE ΔT .
- 5) $\% \text{ MAX } \Delta T = 100 \times \Delta T / \Delta T_{MAX}$.
- 6) CALCULATE C_V / LB/HR OF STEAM (FOR $W=1$).
- 7) FIND h_{fg} IN STEAM TABLE.
- 8) DIVIDE C_V / LB/HR BY h_{fg} . RESULT IS C_V / BTU/HR.
- 9) MULTIPLY C_V / BTU/HR BY ΔT .

FROM EQ. (1), $\Delta T = Q / UA$. SINCE U AND A ARE CONSTANT, $\Delta T = K Q$, WHERE $K = 1 / UA$. THUS ΔT IS A MODEL OF Q , DIFFERING ONLY BY A CONSTANT SCALING FACTOR. THEREFORE THE PRODUCT OF C_V / BTU/HR AND ΔT IS A MODEL OF THE TOTAL C_V , DIFFERING ONLY BY THE SAME SCALING FACTOR. IN OTHER WORDS,

$$C_V / \text{BTU/HR} \times \Delta T = C_V / \text{BTU/HR} \times K Q \text{ BTU/HR} = C_V K.$$

(THE BTU/HR TERMS CANCEL, AND WE ARE LEFT WITH C_V .)

10) NORMALIZE $C_V K$.

THE $C_V K$ PRODUCT FROM THE PROCEDURE ABOVE IS DIVIDED BY THE $C_V K$ PRODUCT FOUND FOR A ΔT EQUAL TO 85% OF THE MAXIMUM ΔT . (THIS WAS THE ΔT CHOSEN IN THE ORIGINAL REPORT.) WHEN THE $C_V K$ PRODUCTS ARE NORMALIZED, THE K CANCELS OUT, AND THE RESULT IS THE RATIO OF C_V AT THE GIVEN ΔT TO THE C_V AT 85% OF MAXIMUM ΔT .

11) PLOT C_V RATIO VS. $\% \text{ MAX } \Delta T$.

SINCE $\Delta T = Q / UA$, AND $\% \text{ MAX } \Delta T = 100 \times \Delta T / \Delta T_{MAX}$, $\% \text{ MAX } \Delta T$ IS ALSO $100 \times Q / Q_{MAX}$ (UA TERMS CANCEL), AND PLOT CAN

SIZING STEAM REBOILER CONTROL VALVES (CONT'D)

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BE CONSIDERED A PLOT OF C_v RATIOED TO C_v AT 85% MAX Q ,
VS. % MAX Q , SHOWING VARIATION OF C_v WITH Q .

CONCLUSIONS:

THE ORIGINAL REPORT INDICATED THAT THE CONTROL VALVE SHOULD BE ABLE TO INCREASE THE HEAT TRANSFERRED TO AT LEAST 110% OF DESIGN; IT WAS FOUND THAT BY BASING THE DESIGN PRESSURE ON THE PRESSURE WHICH GAVE A ΔT OF 85% MAX ΔT , AND SIZING THE VALVE FOR TWICE THE NORMAL C_v , THE 10% OVER DESIGN HEAT VALUE COULD BE ACHIEVED. INSPECTION OF THE PLOT (PAGE B) CONFIRMS THIS. THE CURVE CROSSES THE 2.0 C_v RATIO LINE AT ~96% MAX ΔT , WHICH IS 113% OF THE 85% MAX ΔT , SO THE 10% REQUIREMENT IS MET. SIZING THE VALVE FOR THREE TIMES THE NORMAL C_v GIVES VERY LITTLE IMPROVEMENT — THE CURVE CROSSES THE 3.0 C_v RATIO LINE AT ABOUT 98.5% MAX ΔT , OR 116% OF THE 85% VALUE. (NOTE THAT THE MAXIMUM POSSIBLE INCREASE, AT 100% MAX ΔT AND INFINITE C_v , WOULD BE $100/85 \times 100$ OR 118%.) HOWEVER, THERE SHOULD NOT BE ANY REQUIREMENT FOR LARGE INCREASES OVER THE DESIGN VALUE, SO THIS LIMITATION SHOULD NOT BE A PROBLEM, AS LONG AS THE EXCHANGER DESIGN ENGINEER UNDERSTANDS THAT OUR PROCEDURE ALLOWS FOR ABOUT 110% OF DESIGN AS MAXIMUM, HE CAN SET HIS DESIGN VALUE ACCORDINGLY.)

DESIGN PROCEDURE:1. CALCULATE MAX ΔT .a) FOR ISOTHERMAL REBOILERS, WITH $T_1 = T_2$, $\Delta T_{\max} = T_s - T_1$.b) FOR NON-ISOTHERMAL REBOILERS, WITH $T_1 \neq T_2$, USE LMTD:

$$LMTD_{\max} = \frac{(T_s - T_1) - (T_s - T_2)}{\ln \left(\frac{T_s - T_1}{T_s - T_2} \right)}$$

SIZING STEAM REBOILER CONTROL VALVES (CONT'D)

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DGP2. FIND DESIGN TEMPERATURE, T_D .a) ISOTHERMAL: $T_D - T_1 = 0.85 (T_3 - T_1)$

$$T_D = 0.85 (T_3 - T_1) + T_1$$

$$b) \text{ LMTD: } \frac{(T_D - T_1) - (T_D - T_2)}{\ln(T_D - T_1 / T_D - T_2)} = 0.85 \left[\frac{(T_3 - T_1) - (T_3 - T_2)}{\ln(T_3 - T_1 / T_3 - T_2)} \right]$$

 $(T_D$'s AND T_3 's CANCEL, LEAVING $(T_2 - T_1)$'s, WHICH ALSO CANCEL.)

$$\ln(T_D - T_1 / T_D - T_2) = \frac{1}{0.85} \ln(T_3 - T_1 / T_3 - T_2)$$

$$\ln(T_D - T_1 / T_D - T_2) = \ln(T_3 - T_1 / T_3 - T_2)^{1/0.85}$$

$$(T_D - T_1 / T_D - T_2) = (T_3 - T_1 / T_3 - T_2)^{1/0.85} = K$$

$$T_D - T_1 = K (T_D - T_2) = K T_D - K T_2$$

$$T_D (1 - K) = T_1 - K T_2$$

$$T_D = \frac{T_1 - K T_2}{1 - K} = \frac{T_1 - T_2 + T_2 - K T_2}{1 - K} = \frac{T_1 - T_2}{1 - K} + \frac{(1 - K) T_2}{1 - K}$$

$$T_D = \frac{T_1 - T_2}{1 - K} + T_2$$

$$T_D = \frac{T_2 - T_1}{K - 1} + T_2 \quad \text{WHERE } K = \left(\frac{T_3 - T_1}{T_3 - T_2} \right)^{1/0.85}$$

3. FROM STEAM TABLES, FIND SATURATION PRESSURE EQUIVALENT TO T_D . THIS IS DESIGN STEAM REBOILER PRESSURE.4. CALCULATE NORMAL FLOW C_V , SIZE VALVE AS USUAL, BY DOUBLING NORMAL FLOW C_V .

SIZING STEAM REBOILER CONTROL VALVES (CONT'D)

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CALCULATIONS:EXAMPLE NO. 1: $T_s = 258.67^\circ\text{F}$, $\Delta T_{\text{MAX}} = 48.67^\circ\text{F}$

P_2	T_D	h_{fg}	ΔT	$C_v/\text{LB}/\text{HR}$	$C_v/\text{BTU}/\text{HR}$	C_v/K	$\frac{C_v/K}{C_v/K_{85\%}}$	%MAX ΔT
30.551	251.37	944.6	41.37	0.029	3.105E-5	0.001285	1.00	85
14.123	210	—	0	—	—	—	0	0
15.289	214	969	4	1.66E-2	1.71E-5	6.84E-5	0.0532	8.2
17.186	220	965.2	10	1.70E-2	1.76E-5	1.76E-4	0.137	20.5
19.275	226	961.3	16	1.76E-2	1.83E-5	2.92E-4	0.228	32.9
21.567	232	957.4	22	1.85E-2	1.93E-5	4.24E-4	0.330	45.2
24.060	238	953.5	28	1.94E-2	2.08E-5	5.83E-4	0.454	57.5
26.827	244	949.5	34	2.23E-2	2.35E-5	7.98E-4	0.621	69.9
29.825	250	945.5	40	2.73E-2	2.89E-5	1.15E-3	0.899	82.2
33.093	256	941.4	46	4.58E-2	4.86E-5	2.24E-3	1.74	94.5
34.245	258	940.1	48	8.52E-2	9.06E-5	4.53E-3	3.39	98.6

EXAMPLE NO. 2: $T_s = 345.85^\circ\text{F}$, $\Delta T_{\text{MAX}} = 160.85^\circ\text{F}$

120.753	341.72	877.6	136.72	4.40E-3	5.02E-6	6.86E-4	1.00	85
17.186	220	965.2	15	3.32E-3	3.44E-6	5.15E-5	0.075	9.3
23.217	236	954.8	31	3.32E-3	3.47E-6	1.08E-4	0.157	19.3
30.864	252	944.2	47	3.32E-3	3.51E-6	1.65E-4	0.241	29.2
40.502	268	933.2	63	3.33E-3	3.57E-6	2.25E-4	0.328	39.2
52.418	284	921.8	79	3.37E-3	3.65E-6	2.89E-4	0.421	49.1
67.013	300	910.1	95	3.45E-3	3.79E-6	3.60E-4	0.525	59.1
84.70	316	898.0	111	3.61E-3	4.02E-6	4.46E-4	0.651	69.0
106.92	332	885.4	127	3.96E-3	4.48E-6	5.69E-4	0.829	79.0
131.17	348	872.4	143	4.90E-3	5.62E-6	8.04E-4	1.17	88.9
160.92	364	858.8	159	1.36E-2	1.58E-5	2.51E-3	3.67	98.6

SIZING STEAM REBOILER CONTROL VALVES (CON'D)

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OGPCALCULATIONS (CON'D):EXAMPLE NO. 3: $T_s = 448.13^\circ\text{F}$, $\Delta T_{\text{MAX}} = 182.13^\circ\text{F}$

P_2	T_0	h_{fg}	ΔT	$C_v/\text{lb/hr}$	$C_v/\text{BTU/hr}$	$C_v \times K$	$\frac{C_v \times K}{C_v \times K_{85\%}}$	%MAX ΔT
311.60	420.81	805.5	154.81	1.80E-2	2.23E-6	3.45E-4	1.00	85
41.858	270	931.8	4	1.32E-3	1.41E-6	5.65E-6	0.016	2.2
55.800	288	918.9	22	1.32E-3	1.43E-6	3.15E-5	0.091	12.1
73.259	306	905.6	40	1.32E-3	1.45E-6	5.62E-5	0.168	22.0
94.84	324	891.8	58	1.32E-3	1.48E-6	8.59E-5	0.249	31.8
121.20	342	877.4	76	1.33E-3	1.52E-6	1.15E-4	0.334	41.7
153.04	360	862.2	94	1.35E-3	1.57E-6	1.48E-4	0.427	51.6
191.12	378	846.4	112	1.40E-3	1.65E-6	1.85E-4	0.535	61.5
236.24	396	827.9	130	1.48E-3	1.79E-6	2.33E-4	0.674	71.4
292.45	415	811.3	149	1.68E-3	2.08E-6	3.09E-4	0.896	81.5
343.72	430	796.0	164	2.10E-3	2.63E-6	4.32E-4	1.25	90.0
401.68	445	780.0	179	4.62E-3	5.92E-6	1.06E-3	3.07	98.3

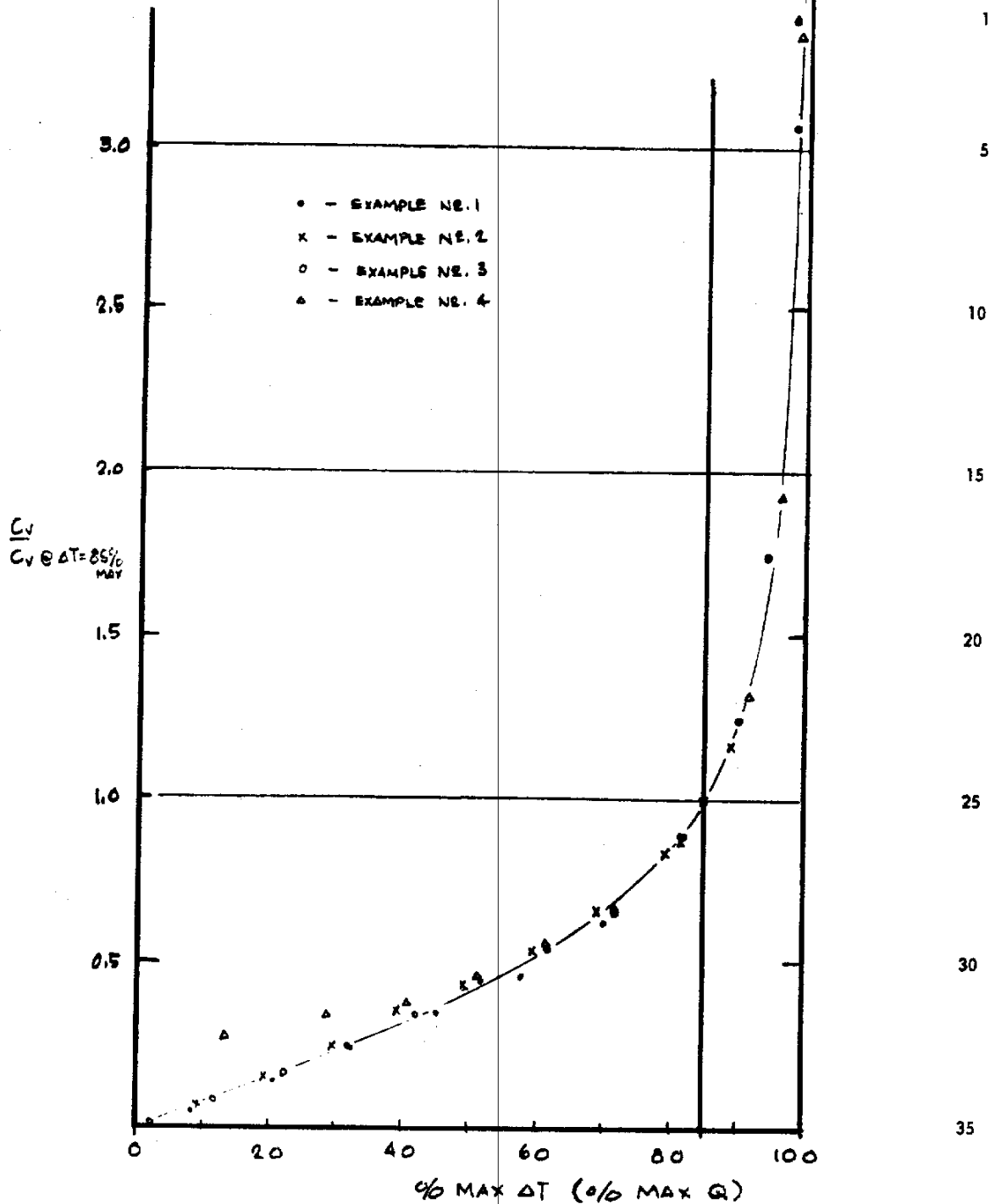
EXAMPLE NO. 4: $T_s = 459.53^\circ\text{F}$, $\Delta T_{\text{MAX}} = 106.73$

397.66	444.00	781.1	90.72	2.01E-3	2.58E-6	3.20E-4	1.00	85
198.14	381	843.7	14.6	1.33E-3	1.46E-6	8.89E-5	0.278	13.7
220.37	390	835.4	30.8	1.25E-3	1.50E-6	1.05E-4	0.329	28.9
247.31	400	826.0	43.3	1.29E-3	1.56E-6	1.25E-4	0.392	40.6
276.75	410	816.3	54.6	1.35E-3	1.65E-6	1.49E-4	0.465	51.2
308.83	420	806.3	65.5	1.43E-3	1.78E-6	1.78E-4	0.557	61.4
343.72	430	796.0	76.1	1.58E-3	1.98E-6	2.18E-4	0.681	71.3
381.59	440	785.4	86.6	1.83E-3	2.34E-6	2.80E-4	0.877	81.1
422.6	450	774.5	96.9	2.49E-3	3.21E-6	4.17E-4	1.31	90.8
444.3	455	768.9	102.	3.50E-3	4.56E-6	6.15E-4	1.92	95.6
457.7	458	765.4	105.	5.91E-3	7.73E-6	1.07E-3	3.34	98.5

Process Division ENGINEERING DEPARTMENT

CUSTOMER _____
 SUBJECT SIZING STEAM REBOILER CONTROL VALVES
Cv REQUIREMENT (NORMALIZED TO Cv REQUIREMENT AT
85% MAX AT) VS. % OF MAX AT

PAGE NO. _____
 DATE 6-27-80
 PROJECT _____
 BY DGP



INTRODUCTION TO ADVANCED PROCESS CONTROL

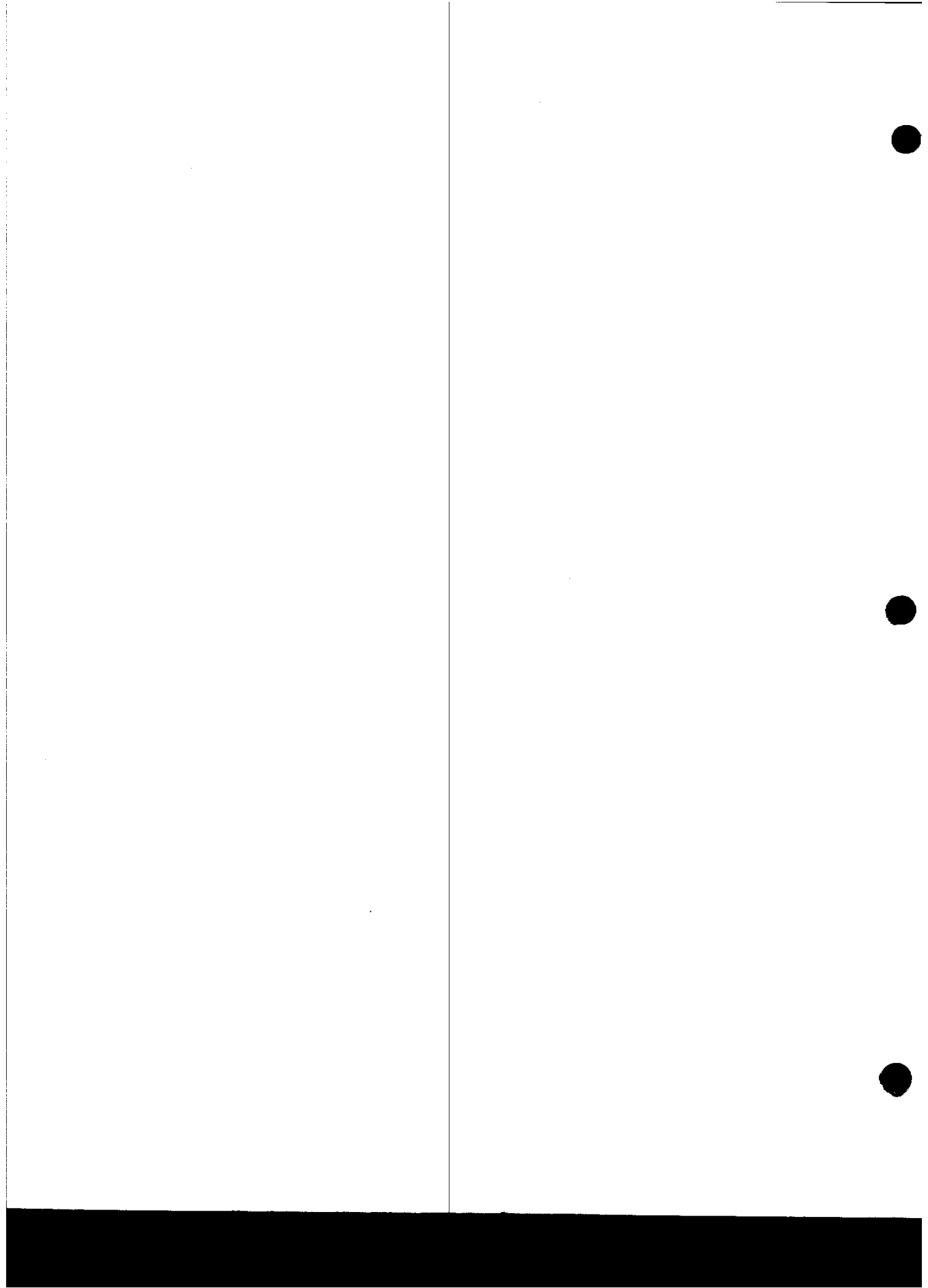
UOP/SETPOINT

ADVANCED PROCESS CONTROL SYSTEMS

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INTRODUCTION TO ADVANCED PROCESS CONTROL

The purpose of advanced control is to augment regulatory control systems through the application of strategies designed to improve profitability and safety. The systems have grown to include unit optimization, multi unit optimization, and production planning and scheduling. The following sections discuss some areas in which APC improves on the performance of regulatory control systems using the UOP advanced control system structure.

Consider the operating characteristics of typical petrochemical plants as they relate to problems in controlling the plant. These characteristics include:

- substantial time delays and lags in dynamic responses
- variables to be controlled which are not measured on-line
- non-linear dynamic responses
- constraints
- interactions
- significant external disturbances

Each of these characteristics present opportunities to improve operation use advanced control methods.

Substantial Time Delays and Lags in Dynamic Response

It is not unusual to see control loops with long time delays, i.e. delayed responses to changes in key operating variables of tens of minutes and major time constants approaching one hour. Even slower loops are relatively frequent. The enhanced capabilities of APC allow for improved control for systems with long response time by the use of predictive control algorithms and the actual unit time constants.

Unmeasured Variables Requiring Control

It is often desirable to control a variable that is not continuously measured - i.e., a composition or important physical property. Periodic laboratory analysis may be used to give some idea of how the process is currently operating relative to the objectives. Operators react by trying to keep the plant as steady as possible. This mode of operation can be improved by the use of correlations employed in APC strategies such as calculated variable control.

Non-Linear Dynamic Response

The observed response of the controlled variable to changes in the manipulated variable is often non-linear. The gain, delay time, and major time constant of the loop can vary depending on the direction the manipulated variable moves, i.e., different responses can be observed for positive and negative moves. The response may vary depending on the operating point of the process, i.e., the feed rate, feed characteristics and catalyst activity. Some loops exhibit hysteresis, i.e. a positive move followed by a negative move of equal magnitude will not return the controlled variable to its original starting point. Unusual dynamic responses which cannot be characterized easily by simple models may also be observed.

The effects of such behavior may lead to frequent retuning of the basic PID controllers. In plants with standard controllers, these loops can often be identified as those which are left in manual under normal operation.

Constraints

Every real control problem has constraints - constraints on manipulated variable moves, constraints on the region where the controlled variable can be operated, and constraints on associated variables that are also affected by the manipulated variable moves. Simple single variable control loops do not anticipate constraints. APC constraint control strategies acknowledge the presence of these constraints and incorporates them into the control scheme.

Interactions

In most processes, a change in one variable can result in changes to several other variables. There is usually a spectrum of effects ranging from major to minor. Compensation for this interaction is accomplished in the APC design by the use of multi variable control strategies thus improving on the traditional PID control.

Significant External Disturbances

Major load changes (disturbances) to processes which affect the operation of loops are often experienced. These disturbances may come from changes in the plant environment or changes in other units. Often these disturbances are "non-stationary" which means their average or mean value changes with time. APC design will inherently minimize the effects of these disturbances by reacting to them before they can impact on key variables.

The net effect of these operating characteristics is to make portions of the plant difficult to control using standard controllers. However, it is possible to improve the operation of the plant through the use of advanced control and on-line optimization techniques.

A. UOP Advanced Process Control Structure

Real life plant controls are always composed of a large number of individual controllers. For these controllers to function in a coordinated fashion there needs to be an overall structure to the design. The natural structure to which most successful industrial control implementers have evolved is a hierarchical one, one in which there are independent control modules with higher level loops passing setpoints to lower level loops.

The structure shown in Figure II.1 is appropriate for considering unit control functions. UOP Advanced Process Control (APC) is structured to improve plant control performance using advanced regulatory control and on-line optimization techniques to provide targets for the basic regulatory controls used to directly operate the plant.

1. Basic Regulatory Control

The basic regulatory controls, generally PID's such as flow, temperature and level controllers, are the primary link between sensed inputs and actuator outputs. Also, included in the basic control level are the following functions:

- real time data acquisition and validation
- conversion of scanned data to engineering units
- communication to external instrumentation, i.e., analyzers, PLC's, etc.
- short term process monitoring and averaging
- alarm detection, display, logging, and alerting
- energy action and shutdown systems
- basic regulatory control loops, display, maintenance and management
- basic sequence control
- basic blending control (ratios)
- historical data storage and trending
- process graphic displays
- communication to higher level functions
- operator man-machine interface.

OVERALL PROCESS CONTROL SYSTEM STRUCTURE

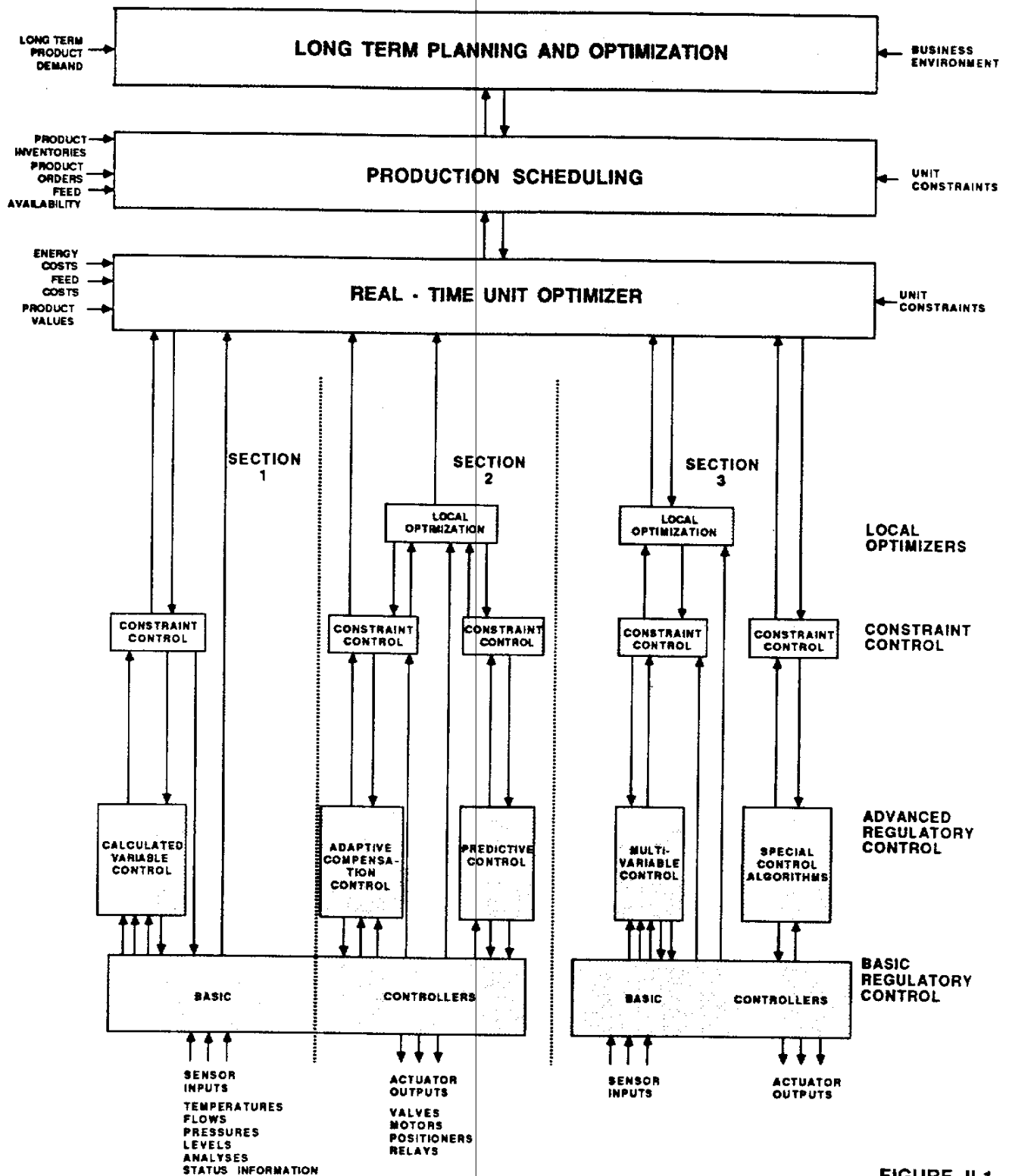


FIGURE II-1

2. Advanced Regulatory Control

The advanced control functions are the levels above the basic regulatory level. These functions may be physically resident in either the DCS equipment or in a process computer.

Functionally, advanced process control is the same as the basic regulatory control, i.e., to hold one or more controlled variables near their individual setpoints in the face of plant disturbances. However the advanced process control algorithms will in general involve coordination of several variables to yield improved profitability and safety.

Calculated Variable Control

With calculated variable control important process variables are controlled that are not directly sensed or measured on the process but are instead calculated from one or more measured inputs. An example would be a TBP cut point on a fractionation column product that is calculated based on one or more temperature and pressure measurements on the column.

Analyzers require periodic calibration and maintenance. The analysis may instead be calculated from a model of the process using reliable temperature, pressure, and flow measurement. Generally, analyzer results are used to update model parameters. This increases the reliability of the system since the controls continue to operate when the analyzers are out of service.

Multivariable Control

As mentioned previously, when trying to control more than one process variable, an interaction or coupling in the process that causes changes in one loop to have an effect on another loop or process variable often exists. The analysis of and compensation for these interactions is the purpose of multivariable control.

Predictive Control

Some process variables are difficult to control. Their response to manipulated variable changes may show long time delays and long lag times. When this occurs, control algorithms that utilize an explicit dynamic model of the process will usually perform better than standard PID algorithms.

Adaptive Control

The sensitivity of many controlled variables to changes in manipulated variables, i.e., the loop gain, is often found to be a function of the operating point of the process. For example, if the feed rate is low, a loop with conventional control at one set of

tuning constants may be sluggish. If the feed rate is high, the same loop with the same tuning constants may be responsive. These observations imply that the tuning for the loop should be a function of the operating point. Adaptive control provides a mechanism to systematically evaluate the desired tuning and to update this tuning on-line.

Constraint Control

Economic analysis of a process often shows that the optimum operating point for many control loops is near constraints. When the process variable is approaching the constraint, it is desirable to have a relatively slow control action. When a constraint is violated, quick control action is desired to bring the process variable back into the desired region. It is desirable to have control algorithms that recognize the unsymmetrical nature of the control objectives near constraints. In addition, multiple constraints often limit operations with each constraint generating a different dynamic response.

3. On-Line Optimization

In some process situations, economic tradeoffs exist between control actions. For example, in many processes it is possible to improve product recovery by increasing energy input into the system. Under these conditions, there is an optimum level of energy input. Clearly, this optimum value will shift with changes in the values of products and energy and with changes in plant constraints. On-line optimization provides the tool to analyze this situation and operate the process at or near the optimum.

Local Optimizers

In some process plants there are a large number of variables which impact on the economics of the unit(s). When such situations occur, there are practical reasons for trying to decompose the problem into a series of simpler problems with fewer independent variables. These simpler problems are solved through local optimizers. This problem typically involves one to four independent variables and a tradeoff between yield and operating costs. A typical example would be the optimization of reflux rate for a distillation column. It is necessary to insure that information is passed to/from the local optimizer to insure coordination and global optimization. The output from all optimizers passes through constraint controllers before implementation to the process.

Global Plant Optimizer

With the local optimizers in place, there still are some variables whose setpoints must be determined to complete the plant optimization. Typically these include feed rates, reactor conditions, and other variables linking major sections of the plant. Optimization at this level is intended to gain the maximum performance from the

whole operation, not just each individual component. The execution time will generally be several multiples of the major time constant of the plant. The global plant optimizer will generally utilize a different model than the local optimizers.

Between all levels of optimization and control there is the necessity to pass information back and forth. Constraints need to be sent to higher level loops to prevent unrealistic setpoints from being returned. Consistent economic information must be passed from one level of optimization to the others to insure coordinated results.

Model Updating

It is also necessary to recognize that all models used for advanced control are only locally accurate representations of the process behavior. As conditions change, the models need to be updated to reflect the current operating conditions. This updating is reflected in Figure II-2.

Data Validation and Reconciliation

Advanced control systems are often dependent on a large number of measurements from the process. Clearly all of these measurements will not be correct all of the time. Detection of incorrect values and obtaining the best estimate of the process conditions is critical to long term successful operation of the system. This is the subject of data validation and reconciliation.

B. Advanced Control Design Philosophy

The following section describes the general UOP advanced control philosophy. The specific unit control strategies presented later in this report will reference the ideas presented in this section.

Several important design considerations affect the performance and reliability of an advanced control system and are the basis for the design of the functions presented for the individual units. These include:

- modularity - individual control functions may need to be turned off due to sensor malfunction or special equipment limitations. The control system should be designed so the modules function independently. If for some reason it is necessary to deactivate one, the remaining modules will still function.
- robustness - the control system should function efficiently under conditions different than that used for design. Alternate calculational procedures should be provided if all of the inputs to a control module are not available. Normal

MODEL UPDATING

**ALL MODELS NEED REGULAR PARAMETER
ADJUSTMENTS TO MAINTAIN ACCURACY
- CAN BE MANUAL OR AUTOMATIC**

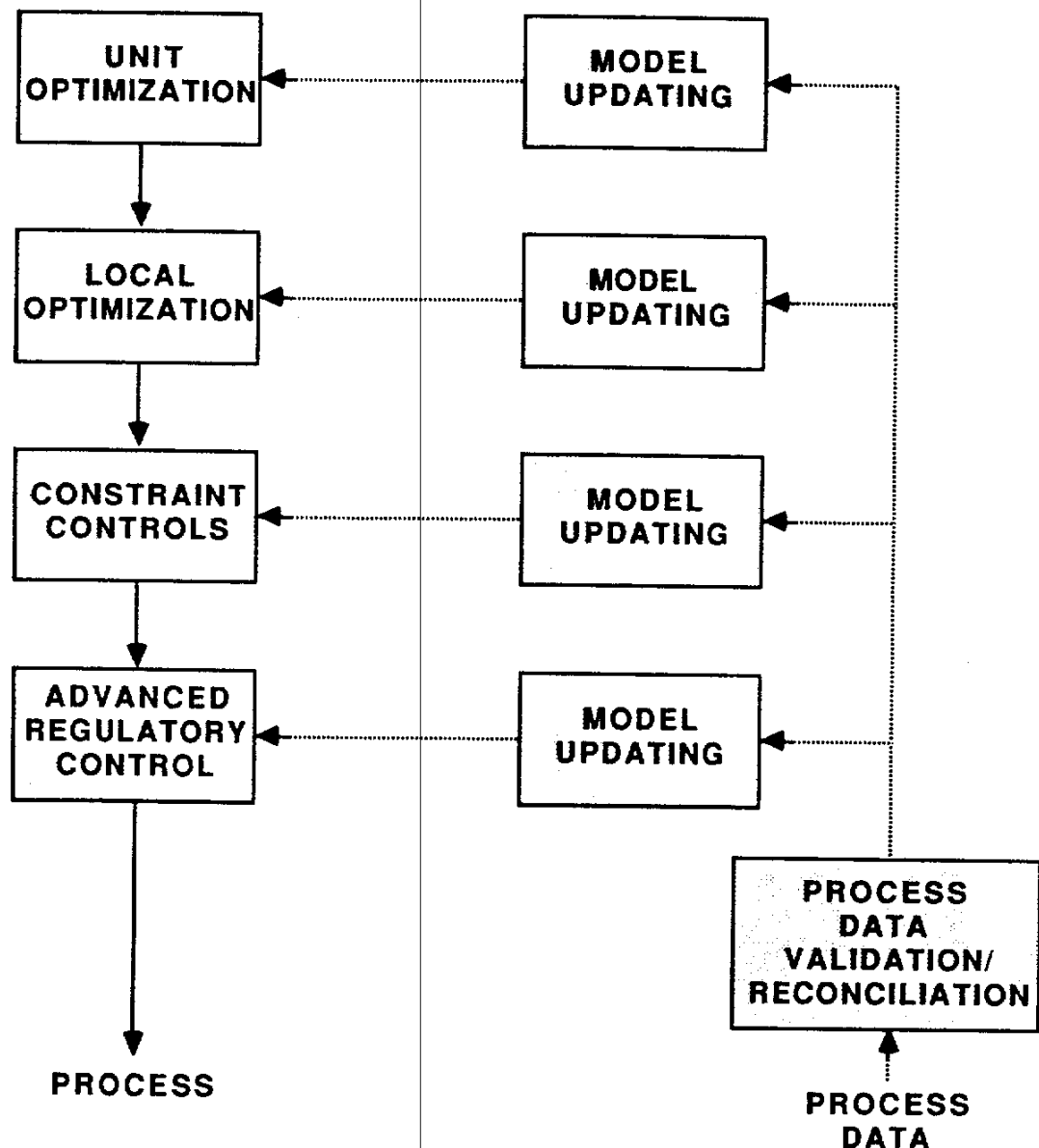


FIGURE II-2

process upsets or changes in the unit operating pattern should not invalidate the control structure.

- simplicity - involved or overly complicated functions should be avoided. An attempt should also be made to provide a straightforward and easily understood mechanism for the operator to enter setpoints for the control function. Control functions should be as simple as permitted by the requirements of the problem. Simplicity contributes to ease of understanding, operator confidence, commissioning and maintenance of the system.

Distillation Control Example

There are numerous applications of distillation column control discussed in this report for the various units. This discussion outlines UOP's general approach to distillation column control.

The function of any distillation column is to separate components in the feed stream. This separation is based on the relative boiling points of each constituent. The primary objective for the operation of a distillation column is usually to control the composition of one or both of the column products within specification.

UOP has extensive experience in successful implementation of advanced control of distillation columns. We employ a hierarchical control structure with three levels:

- Advanced Composition Regulatory Control
- Constraint Control
- Optimization.

Figure II-3 illustrates the control structure.

Each control function is designed as an individual module which can operate independently of other modules. If equipment is out-of-service, then only the control functions directly dependent on this equipment are inactive, and the benefits of the other control functions are still obtained.

By reducing product variation, the probability of producing off-spec material is also reduced. This in itself can save money. However, producing off-spec material is not a problem in most plants because column compositions are usually kept significantly purer than the specification value. The quantitative benefits usually come from moving the operating target closer to specification with no increase in the occurrence of off-spec product. Moving closer to specification allows shifting the material balance to increase production of the more valuable stream and/or reduce the energy required for separation.

ADVANCED CONTROL EXAMPLE DISTILLATION COLUMN CONTROL

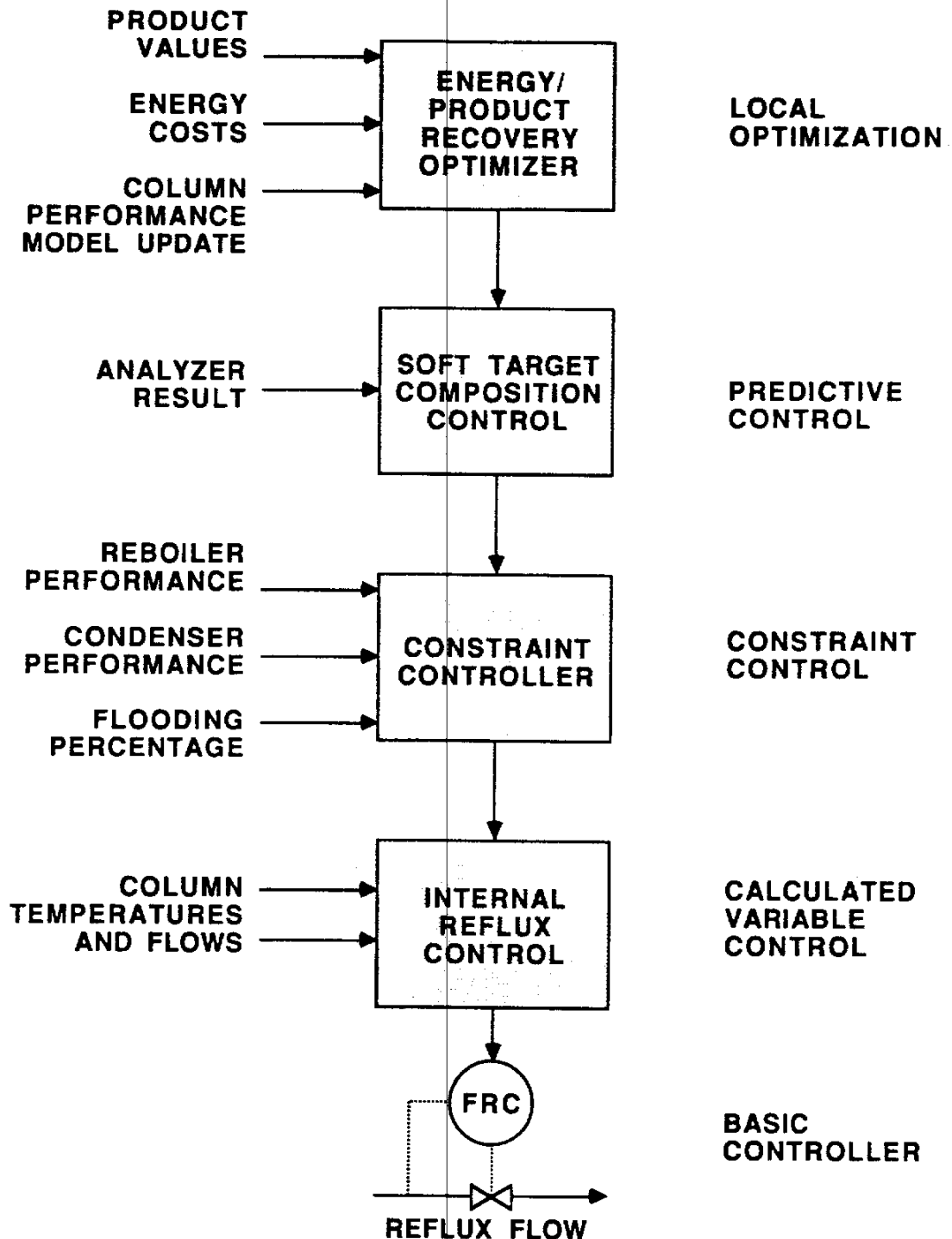


FIGURE II-3

1. Advanced Composition Regulatory Control

Any two-product distillation column has two variables which can be used to manipulate the product compositions: material balance (or distillate/feed ratio) and energy balance (or energy/feed ratio). Either of these variables has an effect on both top and bottom product compositions. It is this interaction that often causes problems when control of both product compositions is attempted. UOP's approach to dual-ended composition control is to choose the most important product for primary composition control and the other product for secondary composition control. The primary composition, or "hard target", is controlled using either a direct or indirect material balance handle; while the energy balance is used for control of the secondary composition, or "soft target". This technique usually provides very tight control of the primary composition because of the good dynamic response to a material balance shift. The secondary composition typically varies somewhat more because of the additional lag associated with the energy balance once the material balance loop is closed. It is the secondary composition that is sacrificed if any of the physical limits on the column are reached. If UOP determine that available interactions warrant multivariable control, this approach is then taken.

Normally, the primary or hard target composition is controlled using either a direct or indirect material balance manipulated variable such as distillate flow rate or column temperature, while the energy balance, i.e. the reboiler, is used for control of the secondary or soft target composition. This technique normally provides very tight control of the primary composition because of the good dynamic response to a material balance shift. The secondary composition typically varies slightly because of the additional lag associated with the energy balance once the material balance loop is closed. Often, interactive moves between the controllers are used to help stabilize the composition controls.

The dynamic responses of product compositions to material and energy balance changes are often characterized by long deadtimes and slow responses. These make control difficult. To compensate for deadtime, UOP has developed proprietary algorithms which include an explicit dynamic model of the process and a prediction of the impact of previous moves made to the process. These predictive control algorithms allow faster control settings and therefore tighter control than is possible with traditional PID control.

Composition control can also be improved by incorporating a properly chosen tray temperature(s) and compensating for pressure variations to maintain specification. This control is especially important when column pressure is not constant, but "floats" -- either because of design or as a part of the control system. UOP has developed models which correct column setpoints for pressure changes.

Another technique of using column temperatures in a control strategy is to incorporate column temperatures into a model to predict composition. An analyzer

is then used to update the model. UOP has found this method to be superior to analyzer control and analyzer-temperature cascade control when a good relationship between temperature and composition exists.

Feedforward controllers are used to minimize disturbances to product compositions. Feed rate feedforward controllers are used to maintain both a distillate-to-feed ratio as well as an internal vapor-to-feed ratio. Feedforward from feed composition is usually not required, but may be necessary when the feed composition changes frequently. On many columns UOP incorporates a lead-lag decoupler between the hard and soft target controllers. Typically, this decoupler is from the soft target to the hard target.

Non-linear level controllers (NLPI) are often installed on levels to minimize feed rate disturbances to downstream units. An NLPI algorithm stabilizes flow by making effective use of the surge capacity of columns and feed tanks. As a result, significant adjustments to feed are only made when the level is far away from a setpoint.

Variables used for advanced control include measured inputs as well as calculated variables. Calculated variables such as reboiler duty and internal flow rates are important in a control scheme because they reflect actual conditions inside a column. By controlling these variables, action is taken immediately in response to variations in the heating or cooling medium. Disturbances are thus prevented from propagating to column temperatures and product compositions.

2. Constraint Control

Once good regulatory control is achieved for the principle composition, a constraint control method is employed for the secondary composition control. A variety of potential constraint limits on a column are measured and compared with preset limit values. A file of these vapor rate projections is maintained in the computer. The most limiting constraint is the target value for the tower internal rate. This would be the value selected to hold the internal tower reflux or internal vapor rate.

All the potential operating limits and constraints for the column are measured and accounted for. These include a measurement relating to condenser limits, tower tray flooding limits based on both jet flooding and downcomer backup, and reboiler limits. They may also be valve positions, pressure drops, or temperature approaches. The specific constraint projections are measured and defined for each column. Constraint control then maximizes reflux, separation, and utilities consumption to maximize the recovery of the more valuable components up to the specified target. It maintains all operating limits within their feasible ranges. This allows higher column and plant capacity.

When columns include side reboilers, two separate sets of constraint controls are

used for the sections above and below the side reboilers. This permits compensation for the subtle interactions which can occur in these columns.

3. Optimization

UOP provides three optimization functions for two product distillation columns. They include:

- Energy Optimization
- Pressure Optimization
- Material Balance Optimization.

Energy Optimization

For many columns it is economically advantageous to reduce the reflux below its maximum physical limit. There often is a smooth tradeoff between the cost of utilities and the incremental value of product recovery. UOP has developed a proprietary algorithm which optimizes energy consumption and determines analytically the economic optimum reflux and boil-up rate based on utility costs and differential values for key components.

In addition, this energy optimizer is integrated with the constraint control module discussed above and projects an economic optimum soft target into the constraint projection table. If the economic optimum is the lowest entry in the table, then it is the value implemented on the column. Through use of this technique, it is possible to make a smooth transition between the constrained and unconstrained optimum case, i.e., the case where economics drive the column to a particular constraint such as maximum condenser duty and the case where the column should not be operating at any limit. The calculations are continually being updated by the control programs, and the column operating point is adjusted to keep the column continually at its optimum point as conditions change. On-line analyzers on both ends of the column are generally required to implement energy optimization.

Pressure Optimization

Optimizing the pressure of operation of the column is important in obtaining minimum utility usage. However, this optimization has often been oversimplified in the past. Pressure interacts with other variables in constraining column operation and the optimum pressure is not always minimum. UOP has developed a pressure

optimization algorithm which is fully integrated with the constraint controls and energy optimizers discussed above. It calculates the optimum pressure and allows an easy transition from constrained to unconstrained operations.

Material Balance Optimization

When light key is more valuable in tops and heavy key is more valuable in bottoms, a smooth optimum material balance split exists for product purities at fixed separation. This can lead to economic primary purity targets which exceed specifications. UOP's material balance optimizer continually checks and solves this case for added potential profits. On-line analyzers on both ends of the column are generally required to implement material balance optimization.

UOP's optimizers always minimize true variable operating costs, based on component values. These values for intermediate streams may be calculated from final product values and downstream recovery performance.

4. Advanced Control Summary

With this complete set of advanced regulatory controls, constraint controls, and on-line optimization algorithms, UOP's distillation control technology provides a cost effective solution for the majority of columns utilized in a modern plant.

This will remove the composition control and material balance optimizer from the hard target control loop. The top block in the typical soft target control loop will be the hot reflux to feed ratio control.

If no on-line analyzers are installed, a few composition models could be implemented. These would require frequent laboratory updates.

5. Engineering Calculations and Information

In addition to the control functions discussed above, the calculational capability of an advanced control system provides an information system that is a valuable tool for plant management and engineers, through timely reporting on the past, current, and projected operation of the plant.

Engineering calculations are provided based on material and energy balances around sections of the tower. In addition, standard engineering principles and procedures are used for correlations such as tray loading, thermodynamic properties, and tower

performance. The calculated parameters are analogous to the calculations a unit engineer might make for a test run. However, they are performed on a frequent basis so they can be used for control and graphical trend display purposes.

C. Advanced Control Benefits

Investments in advanced control and optimization are no different than any other investment in a process plant in that they must be profitable to be justified. There are always a number of projects competing for limited investment capital. Control projects must be able to demonstrate a superior payout to be funded.

The predominant cost is usually the time required for engineering, design, software programming and maintenance. UOP emphasizes installation of controls that are cost effective for their specific application and can be maintained by the plant staff. They stay on-line year after year earning benefits.

There are four items influencing the benefits from advanced control systems, they are: plant characteristics, quantative benefits, how benefits are estimated and qualitative benefits. These are addressed in the following pages.

1. Important Plant Characteristics

The characteristics in a plant that imply that advanced control and on-line optimization may be justified include:

- Frequent Operating Changes
- Large Energy Consumption
- High Product Differential Values
- Integrated Design.

Frequent Operating Changes

Plants that have relatively frequent changes in feed rate, feed composition, product rate and quality demands, and energy supplies are natural candidates for more sophisticated controls. As any of the above variables changes, the optimum operating point is going to change. In addition, changes in these variables can induce significant transients in the process.

Large Energy Consumption

Processes that consume relatively large amounts of energy per unit of product will be natural candidates for more sophisticated controls.

High Product Differential Values

If there is little differential value between the products produced by the process then there will be little incentive to maximize the yield of a particular product. Conversely, if high differential values exist then there will be interest in recovering as much as possible of the higher valued product.

Integrated Design

Rising energy and capital costs have pushed plant designs further in the direction of heat integration. A natural, but unfortunate consequence of this integration is increasing coupling or interactions in the process. Changing a single flow rate can have several secondary effects that propagate through the process and must be corrected by the control system.

2. Quantitative Benefit Areas

Generally the economic payback for the installation of an advanced control system will be in one or more of the following areas:

- Increase in Maximum Capacity
- Reduction of Energy Usage
- Maximization of Valuable Product Yields
- Increase in run length

These are discussed below:

Increase in Maximum Capacity

Full utilization of process capital equipment is essential in today's business climate. Better control often allows more efficient production from the same equipment. Utilizing the capabilities of the advanced control system to monitor a number of constraints, the unit can be safely operated at a maximum capacity which is in excess of that available through manual control.

Reduction of Energy Usage

Close monitoring and management of the energy input and transfers within a plant will often provide significant benefits through reduced energy input requirements per ton of product.

Maximization of More Valuable Product Yields

For most processes producing multiple products, there generally is one that is most valuable. Maximization of the yield of this product is economically beneficial. For plants producing a single important product it is often desirable to maximize yield on feed at a fixed product quality.

Increase in Run Length

A contributor to reduced equipment run lengths is often operating condition excursions, i.e. large deviations from average operating conditions. Steadier operation of the plant can lead to significant increases in run length and minimum equipment downtime.

3. Quantative Benefit Estimates

Quantitative estimates of the expected benefits of advanced control are often obtained through an analysis similar to that shown in Figure II.4. The current operation of the plant is used to evaluate the standard deviation of the controlled variable with conventional controls. The expected reduction in standard deviation with advanced controls is also estimated. The direct economic benefits of reduced process variation depend on the differential value between positive and negative excursions around the mean.

Another benefit of advanced process control is more consistent enforcement of desired process variable targets. For loops with long time lags or delays, it is often found that the process variable of interest is not kept at the specified target. Through advanced process control, the mean value of the process variable is kept nearer the desired value.

Normally, the major benefit from advanced process control comes from the ability to operate closer to constraints with the same or lower probability of constraint violation. As mentioned earlier, the economic optimum operating point for process variables is often found to be at a constraint or at the intersection of constraints. Hence it is economically attractive to operate as close to the constraint as possible. To compensate for process variability, the operator sets a target for the loop which insures that the probability of violation of the constraint is low. Occasional extreme

ADVANCED CONTROL BENEFITS

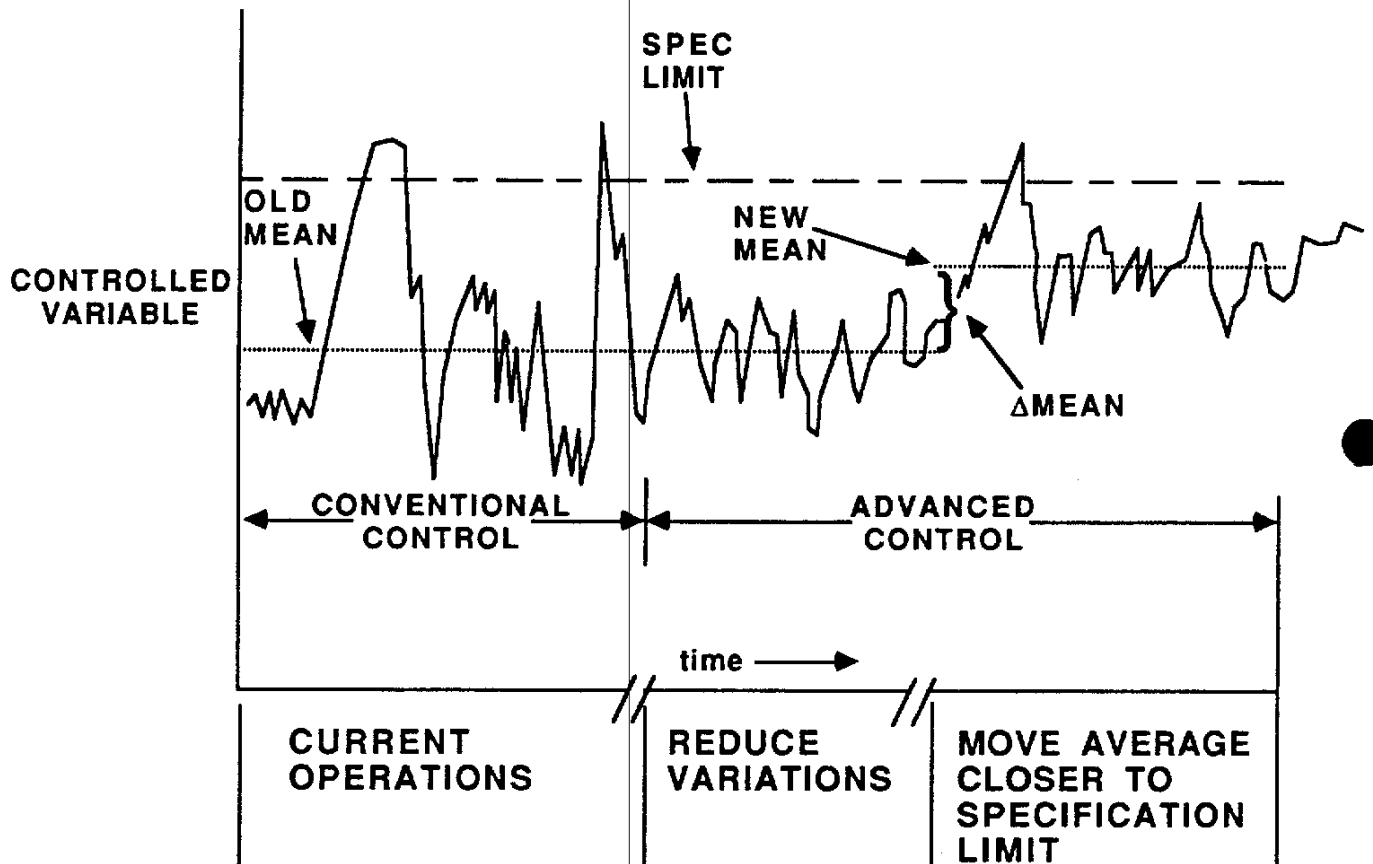


FIGURE II-4

high or low values can severely upset the process. If the variability in the process is reduced through advanced control, the target for the loop can be moved closer to the constraint. The economic benefits of this change correspond to the improved value obtained by moving the target towards the constraint. The ability to operate the plant closer to desired limits will permit more product to be made with the same equipment or the same amount of products with less feedstock and/or energy consumption.

The benefits from on-line optimization can be seen from Figure II.5. If we have complete knowledge about the current state of the plant - process data, mechanical equipment conditions, constraints, etc., and knowledge of the market's current and near-term demand for the plant's products at different pricing levels, a detailed mathematical model of the plant and sufficient computer resources permits frequent on-line execution of an optimizer based on the model. With these conditions we can determine a "true" optimum profitability for the plant. This optimum will shift in time with changes in feedstock availability, market demand, and plant conditions.

The profitability of the plant under normal operator control with modern instrumentation will approximate the "true" optimum with a variable offset. This offset is due to the operator's partial lack of knowledge of plant conditions, market effects, and economic tradeoff between changes in plant operating conditions and the difficulty in making the required number of changes to the basic controllers in a correct dynamically coordinated fashion.

Adding advanced process controls will improve plant profitability as discussed above. Desired operating targets will be implemented in a more consistent fashion with less deviation in performance. Constraint controls will increase profitability further.

On-line optimization advances profitability another level. The direct coupling of economic information to plant decision making through optimization improves the coordination of plant operation with market conditions. Through such coordination, opportunities for short term profitability improvements can be recognized and implemented.

4. Qualitative Benefit Areas

In addition to these quantitative areas, there are a number of areas of improved operation from advanced control systems for which it is difficult to assign specific economic benefits. These are termed intangible benefits and are nevertheless, often of equal importance to the economic benefits. Intangible benefits include:

- Safer Operation
- Steadier Operation

ADVANCED CONTROL BENEFITS

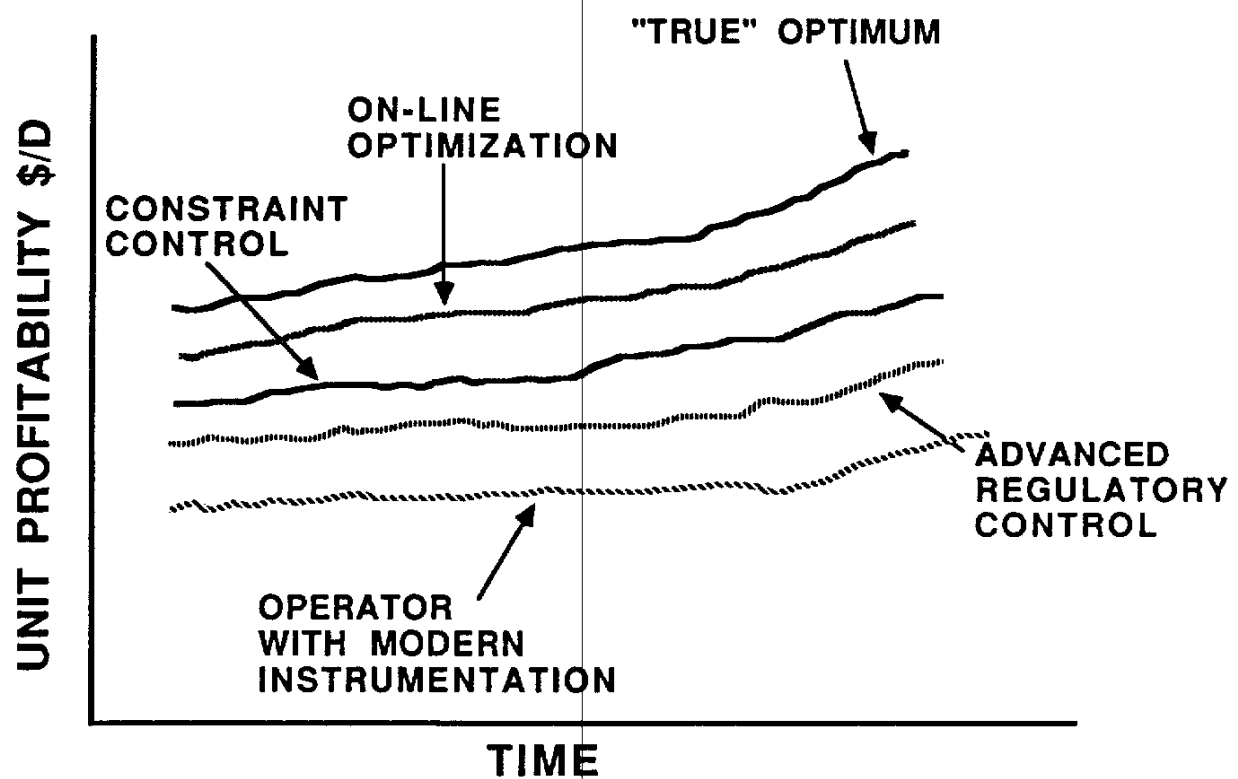


FIGURE II-5

- Improved Flexibility
- Enhanced Product Quality
- Improved Diagnostics
- Better Process Understanding
- Faster Start-up and Shutdown of Equipment
- Improved Recovery from Process Upsets.

Safer Operation

The ability of an advanced control system to monitor a large number of variables and provide alarms based on inferred as well as measured variables can provide an improvement in safety in the plant. Special control actions can be pre-programmed to respond to potentially unsafe conditions.

Steadier Operation

Operating staff prefer a plant that is operating in a stable, smooth fashion. Again advanced controls can often contribute to reducing the effect of disturbances on the process.

Improved Flexibility

Market situations are forcing plants to change product specifications and feed types more frequently than in the past. Under these conditions, improved controls are a requirement to permit the changes to be implemented in a smooth coordinated fashion and to permit operation over a wide range of process conditions.

Enhanced Product Quality

In some cases, companies are seeking market differentiation for their products through improved quality. To attain this higher quality requires improved controls.

Improved Diagnostics

Trends indicating deterioration in performance of equipment can be detected earlier through on-line unit material and energy balances. Data reconciliation techniques can lead to earlier detection of faulty instrumentation.

Better Process Understanding

On-line statistical analysis techniques permit improved understanding of process

behavior. Combined with closer monitoring of operating costs, this improved understanding can pay off in a continuing reduction in operating costs.

Faster Start-Up and Shutdown of Equipment

For equipment which is brought on-line and off-line frequently, advanced controls can enforce consistent procedures while limiting production of off spec product.

Improved Recovery from Process Upsets

Advanced controls are designed to quickly identify and respond to process upsets. This is accomplished by making adjustments of the appropriate magnitude to all necessary variables.



We've Increased the Benefits of Advanced Process Control by a Factor of Two.

UOP and SETPOINT are now working together to bring you the next generation of advanced process control. We've combined UOP's in-depth understanding of its process technologies with SETPOINT's far-reaching experience in advanced control and information systems technology.

The benefits are remarkable:

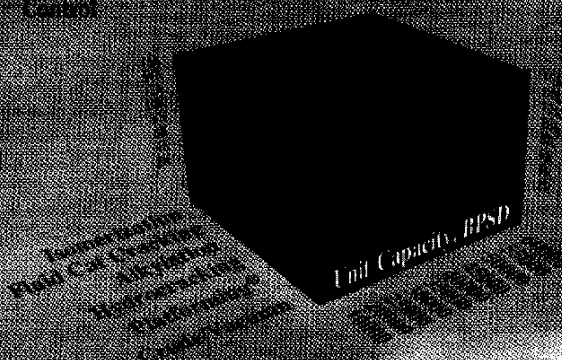
- Maximum profitability with a possible payback of one year or less.
- Improved unit operation using control strategies based on proven algorithms which model the process variables.
- Maximum inherent safety based on full knowledge of process technology.
- Compatibility with every major vendor of instrumentation and computer systems.
- Complete training and support.

UOP and SETPOINT can deliver the technology to make computer-integrated manufacturing a

profitable reality in your plant. Let us develop a benefits estimate to identify where advanced process control can provide improvements.

To make arrangements, or for more information, call John Gobey at UOP, 708-331-3474, or Doug Amos at SETPOINT, 708-331-1500.

UOP/SETPOINT
Advanced Process
Control



UOP

Improving APC
by a factor of two

SETPOINT INC.

INTRODUCTION

This is a general, introductory level overview of process control instrumentation. Application discussions are limited to the simplest instruments. It does not go into great detail on any subject; however, references throughout provide direction for finding details.

The publication is divided into parts which cover the following topics.

- The automatic process control technique including the basic types of control, common combinations, and multiple feedback loop variations
- A comparison of electronic and pneumatic systems
- How a pneumatic transmission system works, including effects of distance
- Major process measurements and the means by which those measurements are made and then transmitted within the process control loop
- Categories of process control equipment, how the equipment operates, what it looks like, and typical applications (NOTE: pneumatic and mechanical instruments are emphasized)
- How a control loop is represented on a functional diagram
- Basic adjustment ("tuning") techniques for feedback loops
- Some major considerations for instrument maintenance and repair
- The terminology of process control technology
- Information on the metric system and unit conversion tables
- Sources of further information

No prior knowledge of process control is required to understand the terminology used. The most common, usually the simplest, application of a concept is given; however, qualifications are provided because process control in practice is not as straightforward as the examples might lead one to believe.

Every attempt has been made to adhere to terminology guidelines provided by the Instrument Society of America (ISA), which has worked long and hard to produce standards which apply to all industries and situations. However, ISA terminology is not always easy for a process control novice to understand, so in some cases generalizations have been made. ISA definitions are provided *in italics* in the Glossary of Terms, Appendix A.

This material can be studied independently or supplemented by other books (refer to the Bibliography, Appendix C). It can also be used as a classroom text. A knowledgeable instructor can expand on topics selected for appropriateness to the industry and processes of a particular plant or mill. A set of the illustrations on slides is available from Foxboro.

This is a revision of a document originally produced by Foxboro in 1951 and used by many as a training aid. In the interest of simplicity, it still covers only the most basic (i.e., pneumatic and mechanical) types of instrumentation. It does not describe recent developments in electronics and digital instrumentation, nor does it describe systems beyond the loop level, even though Foxboro designs and sells products at all these levels. However, references are made throughout the publication to these more advanced forms of process control which can be explored in depth by contacting your Foxboro representative or by using the Bibliography.

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PART I

PROCESS CONTROL

SECTION 1 – PRINCIPLES OF OPERATION

Concepts

A fundamental component of any industrial process control system is the feedback control loop. It consists of the process, the measurement, the controller, and the final control element, as shown in Figure 1. If all these elements are interconnected — that is, if information can be passed continuously around the loop — this is *closed-loop control* and *automatic feedback* generally exists.

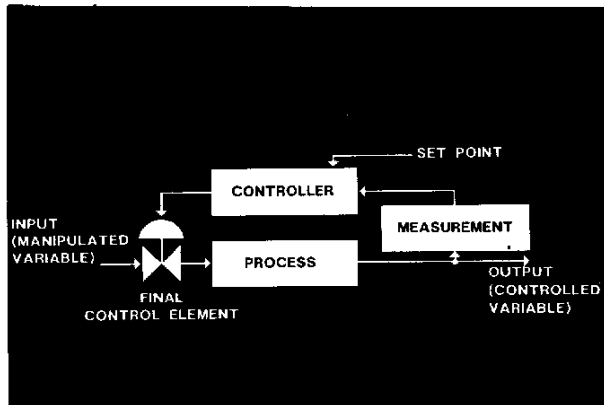


FIGURE 1 — A Closed Feedback Control Loop

This information flow provides the means for control, which allows efficient utilization of raw materials and energy. If a loop is interrupted for any reason — such as when the controller is placed on manual control as shown in Figure 2 — it is considered to be *open-loop control* and automatic control no longer exists.

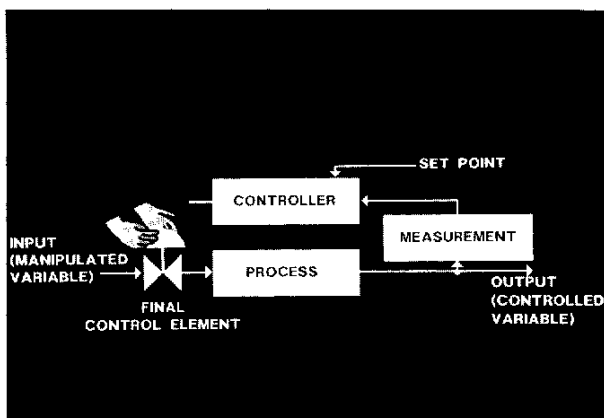


FIGURE 2 — An Open Control Loop

The concept of automatic feedback control is not new. The first such industrial application occurred in 1774 when James Watt used a flyball governor to control the speed of his steam engine.

Understanding of automatic feedback control loops developed slowly at first. Pneumatic transmission systems did not become common until the 1940s, but the past few decades have seen extensive study and development in the theory and application of those concepts.

Automatic feedback control is not used universally. In Figure 2 parts of the system are disconnected, creating open-loop control. Open-loop control does not feed information from the process back to the controller. A familiar example is the domestic washing machine which may be programmed to control a series of operations necessary to wash a load of clothing. It runs through its cycle and, since no information is fed back to the control device regarding the condition of the wash, it shuts down. Only a human agent inspecting the load, and finding it unsatisfactory, can institute corrective action. Open-loop control is seldom encountered in industrial processes and will be given no further consideration.

As stated before, automatic control requires some sort of signal system to close the loop and provide the means for information flow. This means that the controller must be able to move the valve, the valve must be able to affect the measurement, and the measurement signal must be reported to the controller. Without this feedback, you do not have automatic control.

The most common signal systems are electrical and pneumatic, which are compared in Section 2. Because of their simplicity, the pneumatic signal system and the pneumatic and mechanical devices are easily understood and therefore are the ones discussed in detail in this document.

In the discussion of process control loops that follows, several terms are used frequently that should be clarified at this point. Other important terms are *in italics* as they occur in the text and defined then. Additionally, a glossary is provided at the back of this publication for reference (Appendix A).

- The word "process" can refer to many things: it is the raw materials being physically or chemically changed, but it also refers to the collection of equipment (e.g., pipes, tanks, heaters) and functions performed that result in a variable or variables being controlled. (This does not include process control equipment like valves, transmitters, or controllers.)
- The *process load* is the level of material, force, torque, energy, power, or other variable applied to or removed from a process or other component in the system.
- The *set point* is the desired value of the controlled variable; it can be specified by the operator or externally by another instrument.
- A *process upset* is any type of disturbance that causes a control variable to depart from set point.
- A *load change* is a disturbance which causes the controller to reposition its output to restore zero deviation, without any change in set point.
- *Demand* is the requirement made by the control system or the operator on the process.
- The *load response* is the way in which the process behaves after a change in the process load introduces an error into the system.
- *Dead time* is the interval between the initiation of an input change or stimulus and the start of the resulting response.
- *Error*, as applied to automatic control, is the difference between the measured value and the set point. It is also called *deviation*.
- *Steady state* refers to an absence of significant change over a period of time. For example, a process can be at a steady state if disturbances are corrected. It does not have the same meaning as *static* (which means something which normally does not change over time) or *dynamic* (which means something which normally changes over time).
- *Offset* is a steady-state error. A common example (shown in Figure 6) is in a proportional-only control system where a process upset causes the controlled variable to leave the set point and stabilize at another value. That error — the difference between where the output is and where it should be after the system has stabilized — is the offset.
- *Resistance* is anything which opposes the process variable and creates a restriction to the variable — for example, flow from a water tap is determined by the resistance of the tap opening to the flow.
- The *manipulated variable* is that flow, quantity, or other condition which is altered by the automatic control equipment.
- In a feedback control system, the *controlling system* is that portion which compares functions of a directly controlled variable and a set point, and adjusts a manipulated variable as a function of the difference. It includes the reference input elements, summing point, forward and final controlling elements, and feedback elements including sensor. In an automatic control system without feedback, the controlling sys-

tem is that portion which manipulates the controlled system.

- *Capacity* is a measure of capability to store volume, mass, heat, information, or any form of energy or matter.

The use of process control instruments to regulate the process is necessary because process upsets occur. Process upsets create errors, and the function of any controller is to minimize errors as quickly as possible.

Any one of several types of process upsets could disturb a process, requiring a response from the control system to rebalance the process. For example, say we have a tank with the input (inflow) controlled to maintain a tank level at a set point of 50 percent. One type of process upset would be a change in supply, which in this example would happen if the input valve opened more; the control system would have to open the output valve to restore the correct level in the tank. Another type of process upset would be a change in demand, which in this example would happen if the output valve opened more; the control system would have to adjust the input valve to restore the correct level in the tank. Still another type of process upset would occur if the set point were changed, which in this example might be from 50 percent to 60 percent; the control system would have to adjust the input temporarily to reach the correct level (the new set point) in the tank.

For good control, the system should begin to respond quickly, but not change too rapidly. (If the system responds too rapidly, it will overshoot or go past the set point and then have to correct for that error.) Different types of control are employed to make the system do what you want it to, within limits, depending on certain characteristics of the system.

The interactive nature of the parts of a process can be illustrated with the shell-and-tube heat exchanger shown in Figure 3. The temperature of the heated water (the controlled variable) is measured by a temperature

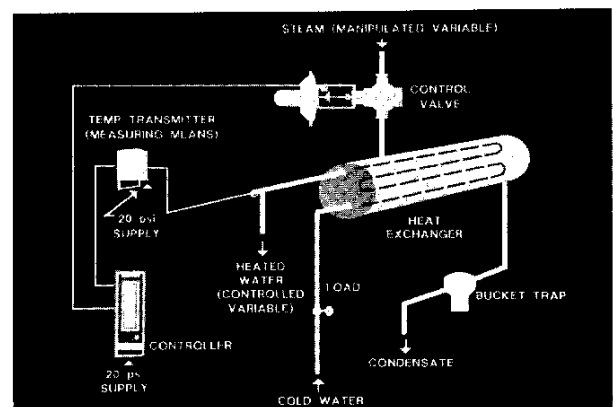


FIGURE 3 — Shell-and-Tube Heat Exchanger Process

transmitter which sends a proportional signal to the controller. The controller makes a comparison between the measurement and the set point; a signal representing a function of that value is transmitted to the valve. If an error has been detected, the valve will be positioned accordingly to allow the required quantity of steam (the manipulated variable) to be admitted to the heat exchanger to bring the heated water temperature to the desired value.

The valve can be *actuated* (caused to change position in response to a changing signal) in several ways, depending on the *control mode*. If the control mode is on/off, the valve is either fully open (usually meaning on) or fully closed (off), changing from one extreme to the other when there is a significant change in the measurement.

If the control mode is *analog* (continuously changing), the position of the valve changes proportionally with each change in the measurement to provide a dynamic balance between supply and demand. There are many varieties of analog control modes — several basic types of control responses and those which combine basic control responses.

Process control loops can be very complex, involving steady-state and dynamic calculations, several types of control modes, and other control loops. The next parts of this publication discuss the system characteristics for which each control mode is best suited, the way that each control mode responds to process error, and typical applications. Later in this publication, you can see the mechanisms that are used to put process control into practice.

Basic Types of Control

On/Off Control (Also Called Two-Position Control)

On/off control is generally both the simplest and the least expensive type of process control and has wide application in industry. A process controlled by an on/off controller almost always has some error in it; in fact, the controller turns on or off only at those times there is no error in the measurement, when the measurement crosses the set point on its way from one extreme error to another. At that point, the valve goes either fully open (on) or closed (off), depending on the direction of the error. The size of the error is not recognized.

No attempt is made to balance the inflow with the outflow; therefore, the energy or material supplied to the process is always either too much or not enough. The measured variable cycles continuously. However, when on/off control is applied to the right type of process, the effect of the cycling is small and acceptable.

On/off control is best applied to a large capacity process that has relatively little dead time and a small mass or energy inflow with respect to the capacity of the system.

A common example would be a typical domestic heating system. A house gets colder than the desired temperature (set point) and the thermostat turns the heater on. The heater supplies enough heat to warm the house to the desired temperature, and the thermostat turns the heater off.

However, there is still enough heat stored in the mass of the house to keep it warm for a while. When the temperature returns to the set point, the thermostat turns the heater on again, but the temperature drops a little more before the heater starts to take effect and warm the house again (dead time).

This cycling is illustrated in Figure 4, which shows the relationship between the temperature of the house (the controlled variable) and the action of the heater (the manipulated variable). Because the mass of the house constitutes a large capacity, the variation in temperature caused by the cyclic effect is so small that it goes unnoticed by the people in the house.

In industry, a typical application for on/off control is the temperature of a large tank or bath. These also have large thermal capacities, with a small source of heat (the energy inflow) warming the liquid in them (the controlled variable) to the desired temperature (the set point).

In either example, the rate of rise (or fall) of the controlled variable is small because the energy inflow is small compared to the large capacity of the system.

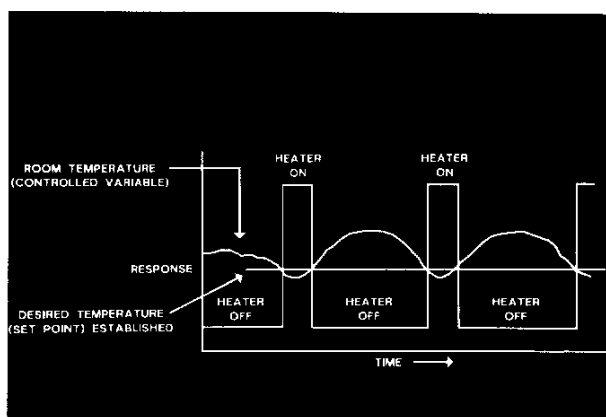


FIGURE 4 — System (House Temperature) Response to a Process Upset (Change in Temperature) with On/Off Control

Proportional Control (Also Called Gain Control or Throttling Control)

On/off control works best on processes with large capacities which change slowly. When a process has small capacity, it usually responds quickly to upsets. Therefore, precise continuous regulation of the manipulated variable is needed. *Proportional control* (abbreviated P) attempts to stabilize the system and avoid fluctuations by responding to the **magnitude** as well as the direction of the error.

The type of process that benefits most from proportional control has a large mass or energy inflow with respect to the capacity and very little dead time.

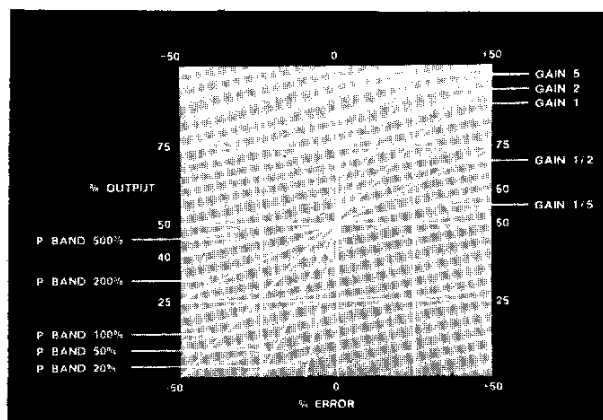
A bathroom shower is an example of a small capacity process. On/off control of water temperature is useless here because turning the controls fully on or fully off (i.e., turning the hot water all the way on and the cold water all the way off, or vice versa) causes too radical a change in output. The energy inflow (hot water) is large with respect to the capacity (total amount of water) of the process. So, we establish a proportion of hot water to cold which can be maintained continuously.

In the shower, as in most process control systems, the final control element is a valve which partially opens or closes to regulate the mass or energy inflow. To provide the appropriate output, the valve travels between fully open and fully closed as positioned by the controller. This valve travel is called the *valve stroke*.

The relationship between the output (valve stroke) and the width of the measurement span is called the *proportional band*. Sometimes called PB or PBand, it is ex-

pressed in percent. For example, a 20 percent proportional band is narrow; it provides sensitive control because 100 percent output change is produced by only 20 percent measurement change. Conversely, a 500 percent proportional band is very wide with only 20 percent of the possible output produced by 100 percent change in measurement.

Figure 5 shows the relationship between the change in output (amount of valve travel) and the amount of error for several different proportional band widths.



NOTE: The typical controller is biased so as to produce 50 percent output, placing the valve at midposition, when the set point and measurement coincide (zero error).

FIGURE 5 — Determination of Proportional Band

In operation, the proportional controller calculates the amount of error between the measurement and set point, amplifies it (the amount of amplification is called the *gain**), and positions the final control element to reduce the error. The magnitude of the corrective action is proportional to the error. Generally, the measurement error is only reduced, not eliminated, leaving offset. A proportional-only controller can completely eliminate offset at only one load condition.

When there is a process upset, such as when flow is suddenly reduced, the valve must change position to keep the controlled variable at a constant level (maintain the set point).

The output from the controller (which controls the valve position) must assume a new value, different from the original (the set point), before equilibrium can again be reached. This new value of the controlled variable

* The relationship between the gain and the proportional band is

$$\text{gain} = \frac{100}{\% \text{ PBand}}$$

is offset from the set point. Figure 6, Curve C, shows system response when the proportional band is at the optimum value, a relatively narrow band in which the oscillations settle out quickly.

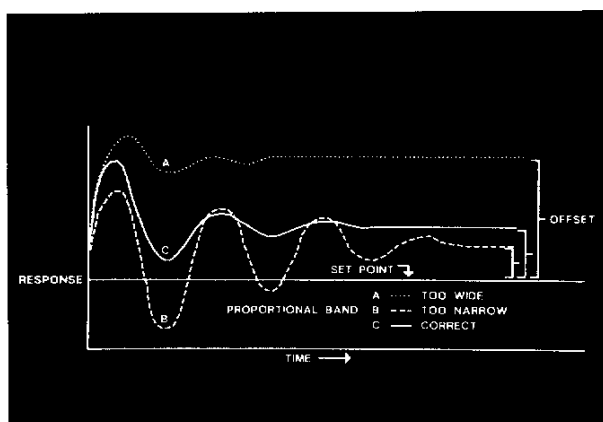


FIGURE 6 — Proportional-Only System Response to a Process Upset with Different Proportional Band Widths

If the proportional band is too wide (insensitive), the offset will be much larger, reducing the amount of control over the process. Narrowing the proportional band (increasing the gain) can reduce the amount of offset, but too narrow a band creates cycling.

Determination of the proportional band setting for optimal control in any loop is discussed later in this publication in Part IV, "After the System Is Installed," under Section 13, "Loop Tuning."

The most important limitation of proportional-only control is that it can accommodate only one fixed relationship between input and output, one control load where input error is zero and one output signal which positions the control valve in the position required to make the error zero.

Pure proportional action is generally adequate for a process that is stable using a narrow proportional band and where a small offset is not detrimental to the operation of the system. For example, noncritical temperature loops with a narrow proportional band or noncritical level control loops with long time constants are good applications for proportional-only control.

However, systems which cannot tolerate even a small amount of offset (such as critical loops) or which need a wide proportional band for stability require an additional control mode: integral action (formerly called reset), which eliminates offset.

Integral Control (Also Called Reset Control)

Integral action avoids the offset created in proportional-only control by bringing the output back to the set point. It is an automatic rebalancing of the system which operates as long as an error exists. Therefore, integral control responds to the duration of the error as well as its magnitude and direction. Integral control is almost never used alone — rather, it is combined with proportional control.

At one time, the system rebalancing had to be done manually; this was called "manual reset." The term "reset" is occasionally still used, although a complete definition of the function includes the mathematical concept of integrating the error until it reaches zero.

Proportional-plus-integral (PI) control is generally used on processes where no amount of offset can be tolerated. Other applications include those where such a wide proportional band would be required for stability that the amount of offset created would be unacceptable.

PI control is applied to almost all processes. The rare exceptions would be gas pressure control and noncritical liquid level control.

When a process upset occurs, the proportional controller registers an error and responds to it as shown in Figure 6. The integral control mode detects the offset error in the proportional mode and tries to eliminate the error.

In a proportional-plus-integral controller, integral action can be expressed in terms of *minutes per repeat* — the amount of time necessary for the integral controller to repeat the open-loop response caused by the proportional mode for a step change in error. The smaller the time value, the faster the integral action. (Some controller manufacturers express integral action in repeats per minute, which is the reciprocal of minutes per repeat.)

Ideally, the minutes per repeat chosen for the integral mode of the controller should bring the control point back to the set point quickly. (The proportional band is specified separately.) If the integral time is too long, the system will not perform at maximum efficiency. If the time is too short, it will overshoot the set point and cycle a bit before returning to the set point; in fact, if

the integral time is too short for the process being controlled, a continuous cycle may result. These relationships are shown in Figure 7.

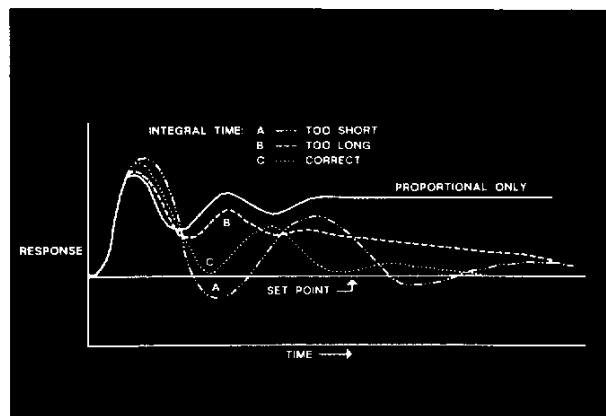


FIGURE 7 — Proportional-plus-Integral System Response to a Process Upset with Different Integral Times

One problem with integral control can occur when a deviation cannot be eliminated over a period of time (as with batch processes when a tank is empty). The controller continues to see an error and tries to correct for it, saturating it and driving the output to its maximum value. This is called *integral windup* (formerly called "reset windup"). When the situation causing the error is corrected, the controller does not immediately return to normal operation; it holds the output — and the valve — at the extreme for a period of time after the deviation has reversed sign.

Things can be done to prevent the problem of integral windup, depending on the type of control system being employed. Refer to the individual sections of this publication on types of control for further information.

Derivative Control (Also Called Rate Action)

The problems noted with proportional-plus-integral control include an extended cycle time and a slight tendency to be unstable. *Derivative control* (abbreviated D) adds stability and reduces response time by acting on the rate of change of the controlled variable.

Derivative control alone cannot bring the signal back to the set point; it must be added to other control modes. Derivative is commonly added to proportional-plus-integral control (see the section following). Derivative added to proportional-only control is called PD control; it does not eliminate offset but reduces its effect by stabilizing the loop and allowing the proportional band to be reduced somewhat.

Because many processes contain multiple capacities, resistances, and other factors that cause delay, there may be a sizable lag between the time that a change occurs in the manipulated variable and the time that the measurement change is detected by the control mechanism.

The faster the error is changing, the greater the initial error will be. For example, a rapid deviation indicates that a drastic change is occurring in the process. Derivative control reduces the initial deviation, because it responds to the rate at which the error is occurring.

Derivative action is expressed in time units (usually minutes). This represents the difference in response time between a proportional-plus-derivative controller and a proportional-only controller (assuming each was subjected to the same process upset and returned to the same set point). Thus, the greater the derivative time the greater the derivative response. Figure 8 shows system response to a process upset with a proportional-plus-derivative controller at different time settings.

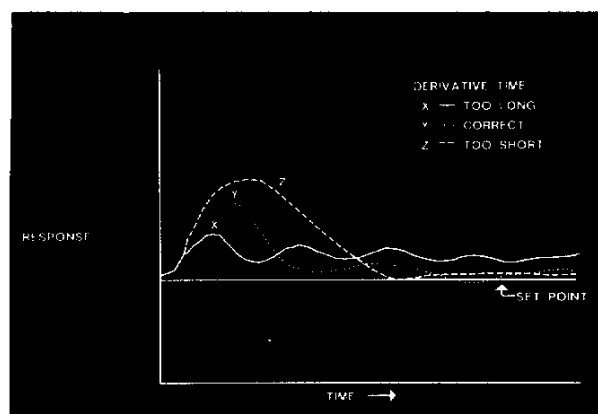


FIGURE 8 — Proportional-plus-Derivative System Response to a Process Upset with Different Derivative Times

Derivative action is used on systems with significant lag and large total capacity. Processes involving temperature are the most common application of derivative control. However, derivative action causes quick control valve movements which may concern some plant operators and maintenance people.

Derivative action is unnecessary on processes which respond fairly quickly to valve motion. It cannot be used at all on processes with much noise in the measurement signal (such as flow); derivative would respond to the noise, resulting in stability problems. By reacting to a rate of input change, derivative action allows the controller to inject more corrective action than proportional action would provide to overcome system inertia.

hot water is applied to raise the temperature to the set point. The batch controller must turn off the hot water before set point is reached and begin cooling to avoid overshoot.

Auto-Selector Control (Also Called Selective Control)

The *auto-selector*, or selective, control system monitors several variables on a process and manipulates one final actuator as necessary to keep the variables within safe operating limits. Any variable may be included in the loop (e.g., pressure, motor load), but all must be affected by the operation of the one valve.

Each variable has an associated controller whose output goes to a device that compares two values, selecting the lowest or highest one (a *low-selector* or *high-selector*, respectively) to direct the valve to respond in such a way that the variable returns to a safe range. However, if another variable exceeds or falls below its set point, that variable's controller output takes over control of the valve.

A common application is a pipeline pumping station (see Figure 15). The control valve is usually wide open for maximum flow unless one of the following conditions occurs.

- The suction pressure drops too low, thus endangering the centrifugal pump with cavitation
- The motor load rises too high, causing the motor to overheat
- The discharge pressure rises too high, exceeding the pressure rating of the downstream pipe

The control system responds to the relationship of the measurement of each variable to its set point, rather than to the measurement itself.

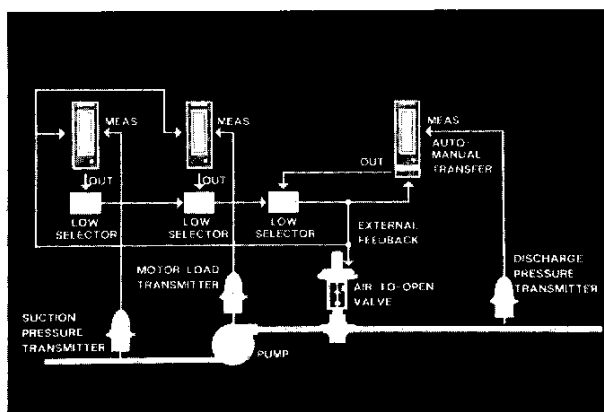


FIGURE 15 — Application of a Three-Variable Auto-Selector Control System Using Low-Selector Relays

If the suction pressure drops below its set point, its controller output will fall below the output of either of

the other two controllers and be selected to throttle the control valve. The valve will close just enough to bring the suction pressure back up to its set point, allowing the flow to remain at the highest level possible consistent with safe operation.

If either the motor load or the discharge pressure rises above its set point, its respective controller output will decrease, immediately assuming control from the suction pressure controller and closing the valve further. External feedback of the selected output to all controllers provides bumpless transfer of control from one controller to the other at the exact time the measurement crosses the set point. A single auto/manual station is available when manual control is required.

The choice of the control valve action, either air-to-open or air-to-close, is usually determined by the fail-safe characteristics of the system. With the *air-to-open* valve shown, an air supply failure will close the valve and shut down the pipeline. If it is more desirable for the valve to fail in the open position so as not to terminate transmission, an *air-to-close* valve would be used.

When an air-to-open valve is chosen, low-selector relays are normally used so that the lowest controller output will determine the valve position. If an air-to-close valve were used in this system, high selectors would be required.

A pneumatically operated batch unit is available as an option on any auto-selector controller used in a low-select system. The purpose of this unit is to provide fast response should an upset arise during a period when none of the controllers is controlling, and the valve is fully open.

Ratio Control

Ratio control provides a means of blending two or more variables in adjustable proportions to obtain a desired mixture. The measurement of the load (the *wild* or *uncontrolled flow*) is the set point to the ratio controller, which adjusts the flow of the controlled variable. A preset ratio regulates the flow of the controlled variable; for example, if the ratio is 2 to 1, for every gallon of the uncontrolled variable flowing, two gallons of the controlled variable are allowed to flow.

Ratio control is applied to processes where the flow of one variable fluctuates, but the desired blend can be maintained satisfactorily by adjusting the related variable (or variables) proportionally.

The most common way of accomplishing ratio control is shown in Figure 16. The signal from the uncontrolled variable (wild flow) is the set point of the ratio controller.

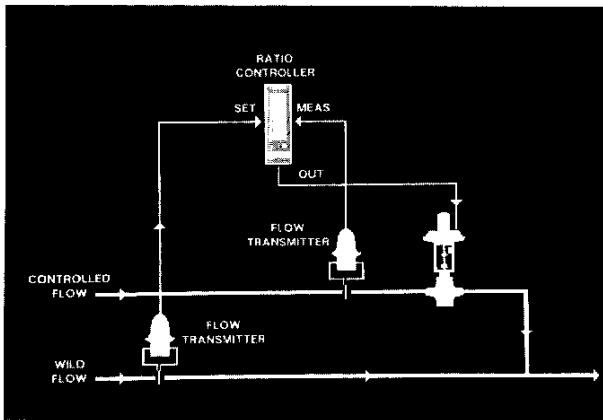


FIGURE 16 — Single Ratio Control System

The value is multiplied by an adjustable factor (the ratio setting). The measurement to the controller is the flow of the controlled variable. The output adjusts the flow of the controlled variable.

The ratio is set to a desired value by a calibrated dial on the controller. Note that the ratio concerns only the flow signals, not the actual amount of the flows, since the measurement ranges of the two flows are not necessarily the same.

The ratio controller can be used with any combination of suitably related process variables. Control action is usually proportional-plus-integral. The response of a system with ratio control to process upsets is the same as the response of the basic control mode used.

Cascade Control

A *cascade control system* consists of one controller (the primary or master) controlling a variable that is to be kept at a constant value by adjusting the set point of a second controller (the secondary or slave) which controls another, faster variable that affects the first variable. The primary controller positions the set point of the secondary controller and it, in turn, manipulates the control valve. The primary variable is slow, most commonly temperature; the secondary variable is as much as ten times as fast, usually flow. The secondary loop is introduced to reduce lags, thus stabilizing inflow to make the whole operation more accurate and responsive.

The secondary controller may be regarded as an elaborate final control element, positioned by the primary controller in the same way a single controller would ordinarily position the control valve. If, for example,

the secondary controller is a flow controller, then the primary controller will not be dictating a valve position but will, instead, be dictating the prescribed flow (set point).

As an example, consider a process consisting of heating a product in tubes by burners supplied with fuel gas. The product temperature must be maintained within narrow limits, made difficult by continuous and erratic variations in fuel pressure. A rise in fuel pressure would increase heat input at the burners, resulting in excessive product temperature.

A conventional feedback control configuration for this sample process is shown in Figure 17. The limitations of this system can be shown by a supply change. If the fuel pressure drops, the product temperature will fall until the single feedback controller could make corrective action to return fuel flow to its original value.

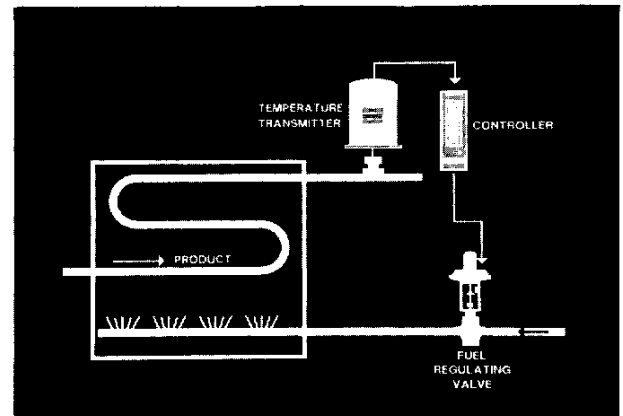


FIGURE 17 — Application of Conventional Control to a Fuel Supply for Burners

Application of the cascade technique (Figure 18) involves simply a different arrangement of conventional equipment. Fuel gas flow is measured by a differential pressure transmitter, which senses a pressure drop across an orifice. This measurement becomes the input to the

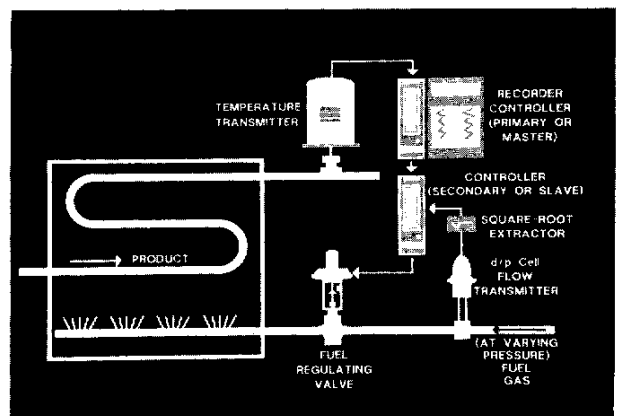


FIGURE 18 — Application of Cascade Control to a Fuel Supply for Burners

Proportional-plus-Integral-plus-Derivative Control (Also Called PID or Three-Mode Control)

Each of the three basic control modes and the combinations discussed so far — proportional (P), proportional-plus-integral (PI), and proportional-plus-derivative (PD) — have limitations which may not be significant if the process and controller are carefully matched.

However, some processes are so difficult to control, or so critical to maintain at set point, that the use of all three modes will be helpful in maintaining desired control. PID control responds to all aspects of process error — direction, magnitude, duration, and rate of change. The output of a PID controller is a linear combination of P, I, and D modes of control.

PID control can be advantageous on many processes. However, its application should be considered carefully because it has limitations with some processes.

Processes that benefit most from PID control have rapid and large disturbances in which derivative action can respond to the rapidity of the changes, and integral action can respond to the duration of them.

Derivative and integral action are complementary in other ways as well. Derivative action permits an increase in proportional gain, offsetting the decrease necessitated by integral action; where integral action tends to increase the period of cycling of a loop, derivative action tends to reduce it, thereby producing the same speed of response as with proportional action but without offset.

Temperature processes, such as the heat exchanger shown in Figure 3, are typical of those applications which can benefit from PID control.

Figure 9 shows the effect of the addition of derivative action to a properly adjusted PI controller. The *period* (time to complete a cycle) is shorter than with proportional-plus-integral control.

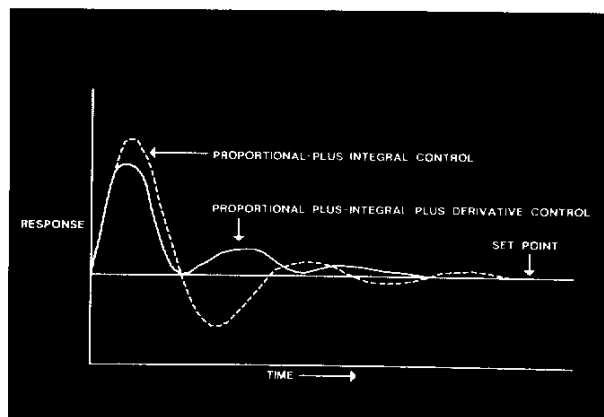


FIGURE 9 — Comparison of System Response to a Process Upset with PI Control and with PID Control

Figure 10 shows the response of a system to a process upset in the primary analog control modes: proportional, integral, and PID. The uncontrolled response is shown for the sake of comparison.

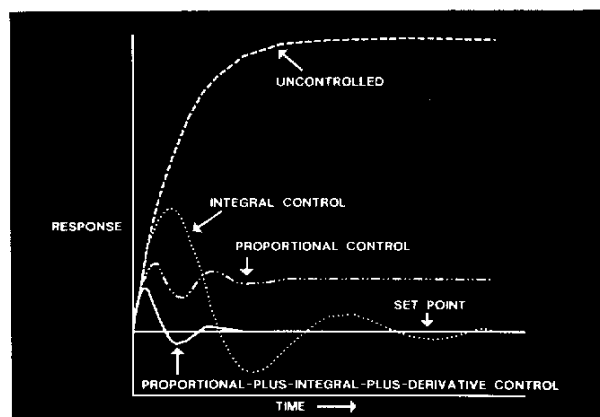


FIGURE 10 — System Response to a Process Upset with Different Modes of Analog Control

Figure 11 shows the response of a system to a change in set point (as would happen in tuning a controller) using the same analog control modes.

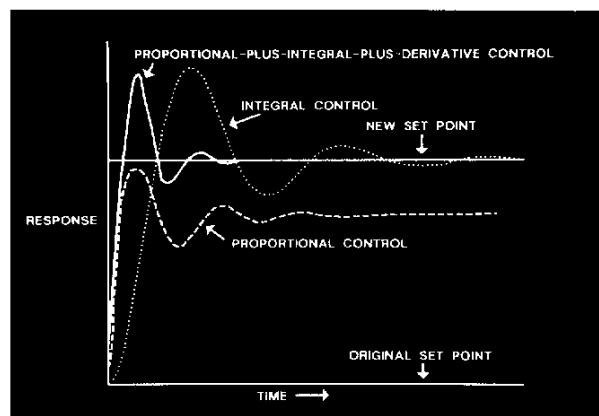


FIGURE 11 — System Response to a Change in Set Point with Different Modes of Analog Control

Combination Types of Control

Batch Control

Batch control was designed for discontinuous processes such as batch reactors, where there are intervals between finishing one production run and beginning the next. Batch tanks are quite common in process plants. It is important that product quality and uniformity be maintained from one batch to another.

At the point when the tanks are empty, an open-loop situation exists. However, the controller will behave as if the loop were still closed and, because the measurement has gone beyond the set point, continue to change its output due to integral action.

When the next production run is started and the tanks start to fill (thereby closing the loop), the measurement must pass the set point (enter the range in which proportional action can begin to function) before the control output signal begins to change. The result is that the measurement overshoots the set point. This condition is called "output saturation due to integral action" or "integral windup." It is shown in Figure 12. Observe that the controller output does not begin to fall until the measurement has crossed the set point following startup.

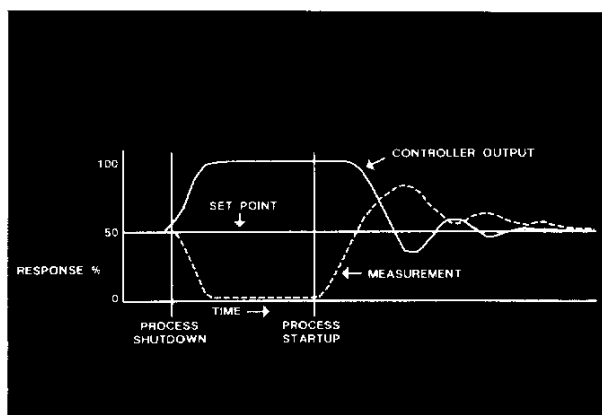


FIGURE 12 — System Response to Process Shutdown and Startup with PI or PID Control Only

To avoid integral windup, a batch controller is specifically designed to begin control action before the measurement reaches the set point and thus avoid overshooting.

When the measurement is within normal control range (output between 0 percent and 100 percent of scale), the standard control mode (typically PI or PID) corrects for any process upset. A "batch unit" or "batch switch" actuates only when the output goes beyond the preset limit (over 100 percent or under 0 percent of scale) and

the standard control mode can no longer return the measurement to set point.

The batch unit dissipates the integral feedback signal, attempting to hold the controller output at the preset limit. The proportional band, which integral action would otherwise force completely above the set point, responds independently to the measurement and is able to react appropriately when the measurement returns to the controller's proportional band range. (Refer to Figure 13.) The tendency to overshoot is overcome and normal PI or PID response is restored. Note, however, that careful tuning of the controller is necessary for the batch unit to function effectively.

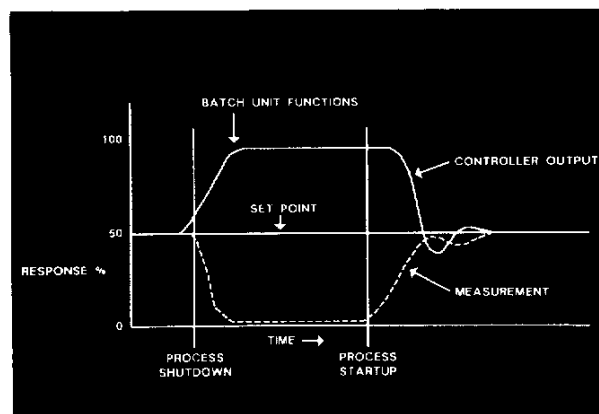


FIGURE 13 — System Response to Process Shutdown and Startup with Batch Control

Batch control is used primarily in discontinuous processes. It is not used on controllers with a fast integral action response time such as flow controllers. It is used on antisurge flow controllers for compressors, however, where speed of response is essential to protect equipment.

A typical batch process is the exothermic reactor shown in Figure 14. After the vessel is charged with reactants,

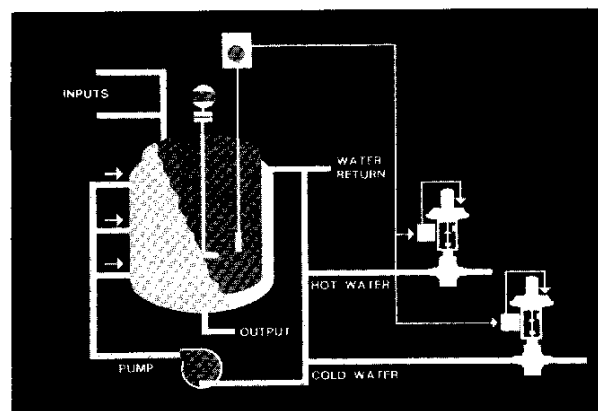


FIGURE 14 — Application of Batch Control to a Batch Chemical Reactor

The advantage of this system is that the flow controller protects the process from the variations that would be caused through varying fuel pressure and allows the primary controller to regulate temperature with improved stability.

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In order for cascade control to be effective, the response of the secondary loop must be considerably faster than the response of the primary loop. Severe instability can result if cascade control is misapplied to a situation where the periods of the primary and secondary loops are approximately equal, because of nearly simultaneous set point and measurement changes in the secondary loop.

System response to a process upset is the same as for the basic control mode, which is determined by the secondary loop (if the secondary process is flow, the control mode is PI; if it is valve position or a relatively fast temperature process, the control mode is P only).

In general, cascade control can dramatically improve systems where: 1) supply upsets are large and frequent; 2) the primary variable responds slowly but with large amplitude to supply upsets; and 3) the supply variable can be controlled in a rapidly responding loop.



PART II

TRANSMISSION SYSTEMS

SECTION 2 — COMPARISON OF ELECTRONIC AND PNEUMATIC SYSTEMS

Before 1938, most measuring and control instruments were connected directly to the process. This required that instruments be locally mounted. The process operators were required to move around the plant taking readings and making adjustments. With a small process plant this worked; but as the plant grew, it became just about impossible. The pneumatic transmission system was devised to overcome this difficulty.

With a transmission system it is possible to locate all recording, indicating, and controlling instruments in a central control room. This permits the operators to continuously supervise the entire plant's operation from one location.

The signal transmission system can be either pneumatic (air pressure) or electronic (electric current). Pneumatic

transmission systems are easy to understand because they use a basic process measurement: pressure. However, electronic systems perform the same functions.

Pneumatic and electronic instruments look more or less identical, both in the field and in the control room. The two signal transmission systems can be used within the same process control system, allowing a great deal of flexibility in instrumenting a plant because equipment can be chosen which suits the application exactly. Pneumatic pressure-to-electric current (P/I , for short) and electric current-to-pneumatic pressure (I/P) converters are readily available.

Of course, there are some differences between electronic and pneumatic systems, both in performance and capability. These are summarized in Table A.

TABLE A — Comparison of Pneumatic and Electronic Systems

FEATURE	PNEUMATIC SYSTEM	ELECTRONIC SYSTEM
Transmission distance	Limited to a few hundred feet	Practically unlimited
Standard transmission signal	3 to 15 psi (20 to 100 kPa) practically universal	4 to 20 mA dc practically universal
Compatibility between instruments supplied by different manufacturers	No difficulty	Occasionally nonstandard signals may require special consideration
Control valve compatibility	Controller output operates control valve operator directly	Pneumatic operators with electropneumatic converters or electrohydraulic or electric motor operator required
Compatibility with digital computer or data logger	Pneumatic-to-electric converters required for all inputs	Easily arranged with minimum added equipment
Reliability	Excellent, if energized with clean dry air	Excellent, under usual environmental conditions
Reaction to very low (freezing) temperatures	Must take precautions to keep air supply completely dry	Electronic portions of control loop are not affected by temperature
Electrical interference effect	No effect	No effect with dc system if properly installed
Operation in hazardous locations (explosive atmosphere)	No inherent hazard	Explosion-proof and intrinsically safe equipment available — power must be removed for most maintenance
Reaction to sudden failure of energy supply	Capacity of system provides safety margin — backup inexpensive	Most failures due to power outages can be prevented through readily available alternate power system such as UPS (Uninterruptible Power Source)
System compatibility	Fair — requires considerable auxiliary equipment	Good — conditioning and auxiliary equipment more compatible to systems approach
Ease and cost of installation	Tends to be labor intensive	On-site labor costs have been decreasing due to electronic modularization
Instrument costs	Relatively stable	Decreasing costs due to microprocessor technology and modularization
Ease and cost of maintenance	Fair — procedures more readily mastered by people with minimum of training	Good — depends upon training and capability of personnel
Dynamic response	Slower, but adequate for most situations	Excellent — frequently, valve becomes limiting factor
Operation in corrosive atmospheres	Air supply becomes a purge for most instruments	Consideration must be given to effects of corrosive gases in the atmosphere around any electronic components
Velocity of transmitted signal	Relatively slow (under 1000 ft/s [300 m/s])	Extremely fast
Measurement of all process variables	A few measurements are difficult, but can be made available	Excellent
Performance of overall control system	Excellent, if transmission distances are reasonable	Excellent — no restrictions on transmission distance
User attitudes (the unmentioned factor that frequently pops up)	Generally regarded as acceptable, but not the latest thing	Often regarded as the latest and most modern approach

SECTION 3 — WORKING WITH A PNEUMATIC SYSTEM

One function of any signal transmission system is to avoid having hazardous materials enter the control room. For example, a pneumatic transmitter may be measuring the pressure of a vessel containing a highly toxic material under very high pressure. A direct pressure measurement would require that this material be piped into the control room. A transmission system, on the other hand, simply requires a signal representing this pressure, so the hazard is avoided in the control room.

Another added advantage is the common signal. All recorders, indicators, alarms and other associated equipment operate from the same signal level, the range of which is generally 3 psi (20 kPa) at the lower scale value to 15 psi (100 kPa) at the upper scale value. This differentiates between a 0 percent signal, which would register as 3 psi, and no signal (as from a failed transmitter), which would register as less than 3 psi. Other pneumatic signal ranges sometimes used are 3 to 27 psi and 40 to 200 kPa. Where an instrument requires a signal level higher than the standard chosen for the plant (positioners often require signals above these ranges), the needed level can be achieved through the use of amplifiers.

The practical advantage of this is that the control room instruments are basically the same. A control panel may contain indicators, recorders and controllers handling pressures, temperatures, flows, and other process variables. The only differences between them are the scales or charts employed.

The air pressure signal is transmitted by tubing, usually 1/4-inch copper or plastic, from field to control room and within the control room between instruments.

Signal Transmission Distances

Unfortunately, it is difficult to recommend allowable distances for pneumatic transmission because there are no "hard and fast" rules to follow. What may be acceptable in one case could be detrimental in another. However, by examining the characteristics of the transmission tubing and its termination, some guidelines can be drawn for individual control loops.

Signal speed is limited by absolute physical properties and by the fact that there is a lag caused by resistance and capacity distributed along the entire length of the line. This distributed lag adds dead time to all processes as shown in Figure 20.

For 1/4-inch tubing (0.188-inch ID), the total dead time is about one second and it takes about five seconds for each 500 feet (150 metres) of tubing for most of the

measurement change to be transmitted to the device at the other end, if the tubing terminates in a small volume such as a bellows. (This is the *time constant*, a mathematical concept, which is defined in the Glossary of Terms, Appendix A.) Terminating in a large volume such as a valve motor greatly increases the time required to transmit the measurement change without materially changing the dead time. Inserting a repeating relay (a pneumatic signal amplifier) or volume booster in the line will tend to reduce the time required to transmit the change without affecting the dead time. Larger diameter tubing has less resistance to flow but greater capacity so, unless it is terminated in a large volume, a marginal advantage can be obtained using 3/8-inch tubing.

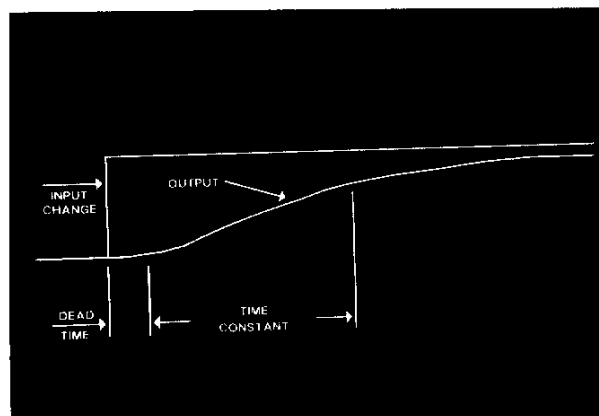


FIGURE 20 — Pneumatic Tubing Transmission Lag

Pneumatic transmission lags most seriously affect the response of flow control loops. A close-coupled pneumatic flow loop will typically oscillate with about a one-second period, as will an electronic flow loop with a pneumatic valve. Adding 100 feet (30 metres) of tubing (to both input and output) will increase the period to perhaps two seconds, and 500 feet (150 metres) will raise the period to eight or ten seconds. If a particular process has few load upsets which are slow to develop, a ten-second period may provide perfectly adequate control. But if a particular flow controller must provide surge protection for a compressor, even a two-second period would be uncomfortably slow.

The point being made here is that the allowable tubing length depends primarily on what process is being controlled and what the controller is being asked to do.

Valve positioners are frequently used to reduce termination volume. Valve positioners should not be used when controlling flow as they tend to destabilize a fast-responding system. Instead, a volume booster mounted on the valve motor will reduce termination volume without loss of stability.

The best response would be achieved by mounting the flow controller on or at the valve, but then extra lines must be run between the control room and the field to transmit the set point and measurement and, if required, the manual regulator output and transfer signals. This doubles the tubing that must be run, as indicated in Figure 21.

As a result of the higher cost of both instrument and installation, relatively few controllers are locally mounted. However, if neither manual control nor valve-position indication are required, and they do not seem to be in most flow loops, a less expensive field-mounted flow controller called a "two-pipe" system (shown in Figure 22) can be justified.

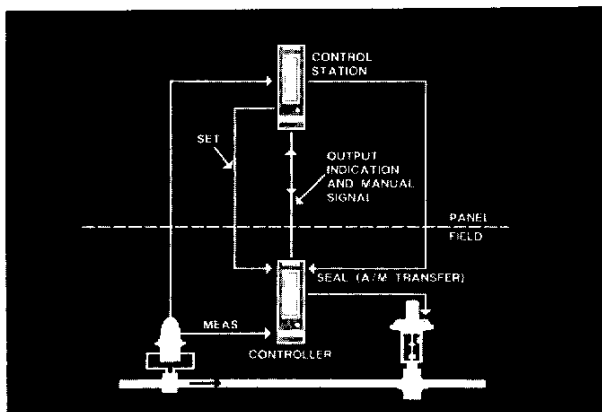


FIGURE 21 — "Four-Pipe" Pneumatic Transmission System

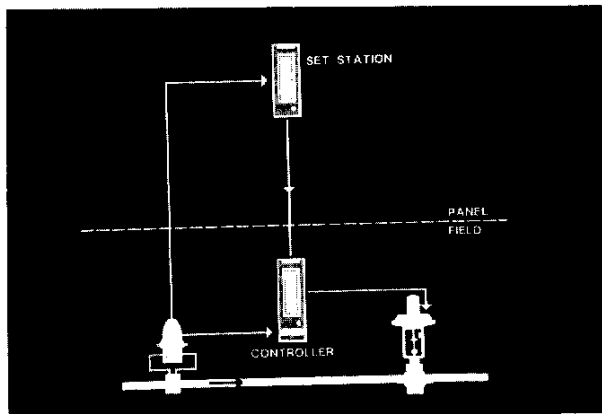


FIGURE 22 — "Two-Pipe" Pneumatic Transmission System

Applications

So far, the discussion of pneumatic transmission signals has concentrated on the requirements of flow loops. Flow measurements can change very fast. Liquid pressure reacts as quickly as flow and, similarly, transmission distances should be kept to a minimum. Minimum transmission distances are in order also when controlling dynamic components such as compressors or combustion systems. If problems arise in keeping transmission distances to less than 100 feet (30 metres), a valve positioner will be helpful in reducing termination volume without loss of stability.

Gas pressure and liquid level processes are slightly slower than flow or liquid pressure processes, making transmission distances less critical. Temperature and composition loops typically display periods in the order of several minutes to an hour or more. In these cases, even 1000 feet (300 metres) of tubing would not noticeably influence performance.

Another thing that can affect the selection of transmission distance is where the loop's set point is determined. If it is set remotely (by another controller, as with cascade control, or by another variable, as with ratio control), consider whether it is more important to have the faster response (and therefore the shorter transmission distance) between the process and the controller or between the remote set point and the controller.

In general, the less critical the measurement, or the less severe the upsets experienced, the more acceptable is the relatively slow response caused by long transmission distances.

PART III

INSTRUMENTATION TO IMPLEMENT PROCESS CONTROL

SECTION 4 – GENERAL INFORMATION

Before you can implement process control, you must have something tangible to control. In the basic feedback control loop, the measurement is that tangible representation of the process. In certain ways, the measurement is the most important part of the control loop: no matter how complex or sophisticated your controller, it needs sensor data.

The sensor can be connected directly to the controller, or a measuring transmitter can be used. Other devices may be inserted between the measurement and the controller to provide signal conditioning. Any of the following may be included in the loop: recorders and/or indicators, computing devices, alarms, integrators, positioners. This array of equipment can be organized into a loop as shown in Figure 23. NOTE: The transmission system — the means by which the signal is passed among the pieces of equipment in the loop — is discussed in Part II of this publication; details on the loop instrumentation are provided in this part.

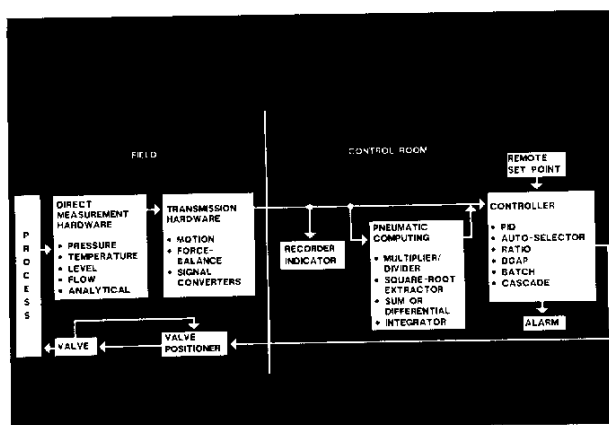


FIGURE 23 — Types of Instrumentation That Can Be Used in a Control Loop

The overall performance of a control system can never surpass that of the individual instruments that make up the measuring system. Good control depends not only on the static accuracy of the measuring system, but also on its dynamic performance. If the measuring system is not capable of responding as fast as the measured variable can change, then the correcting action dictated by the control system will be correspondingly slowed.

It is important to understand the dynamics of processes as well as the advantages and disadvantages of the dif-

ferent types of control equipment. You need to know the requirements of range, corrosion resistance, speed of response, ambient temperature, and economics. Only then can the appropriate instruments be chosen.

The instrumentation required for each measurement point should be specified individually for optimum results. Temperatures can vary from point to point in a line, and flow patterns can change where the piping bends. Instrumentation specification is a very complex subject, which cannot be explored fully here. However, as a guideline, the choice of sensors should consider all of the following:

- the range and the span of the measured variable,
- the accuracy and resolution required to maintain adequate control of the process, and
- the speed or rate of change of the measured variable.

Keep in mind, too, that slight deficiencies in accuracy, resolution, and response time may be magnified when the individual components become part of a control system.

The following sections discuss the different types of process control equipment, the instrument categories within each type, and common instrument mechanisms. Section 5 discusses five basic process variables — pressure, temperature, flow, level, and analytical. Their measurement is explained in terms of mechanical sensors which sense the change in the process and give a direct, proportional, local indication of the measurement. (Section 5 also gives an overview of other process measurements and provides references for them.) Section 6 discusses mechanisms, transmitters, some specialized sensors, and signal converters. Recording and indicating instruments are discussed in Section 7, pneumatic computers in Section 8, and so on through the various parts of the process control loop shown in Figure 23.

The Foxboro equipment for each instrument category is described in detail where appropriate, and the process applications, advantages, and limitations for each are discussed. Options are not discussed; this information can be obtained from specification literature provided by Foxboro. In general, however, most instruments can be provided with special *process-wetted materials* (i.e., materials that come in contact with the process) and with special preparation and cleaning to suit specific

applications. An important option that should be selected with any pneumatic instrument is an *air set* which filters and regulates the air supply to the instrument.

Foxboro recognizes that the future of process control resides in advanced electronic technology, but shows pneumatic and mechanical instrumentation in this publication because it is easier to explain and simpler for the novice to understand. Electronic devices and even microprocessors generally mimic the functional circuits described here because they are basic.

This section discusses only basic pneumatic and mechanical instruments — the “*tried and true*” technologies on which further developments in the field of process control have been based. The more recent (and more common) electronic technologies are referenced

in the comparison tables; further information can be obtained about these instruments from your Foxboro sales representative or in the publications listed in the Bibliography.

This section covers only the measurement and control instrumentation needed to implement basic process control loops. This represents the beginning of what can be accomplished by the application of advanced control developments. Your Foxboro sales representative can give you information on process control systems that incorporate sophisticated control rooms supervising entire plant or mill processes ... even all of the processes in an entire plant or mill, including computers providing management information as well as process information.

SECTION 5 — DIRECT-MEASUREMENT HARDWARE

Pressure Elements

Pressure can be defined as the force exerted over a surface divided by its area. In the United States, pressure is commonly expressed as pounds per square inch (psi), though technically the term is pounds-force per square inch. Very low pressures are commonly expressed as inches of water (inH₂O) because the equivalent values in psi are so small. For example, 1 inH₂O = 0.03606 psi.

To provide a terminology applicable throughout the world and convenient for very low as well as very high pressures, many companies (including Foxboro) recognize the SI* unit pascal (Pa) and multiples of it like kilopascal (kPa); 1 kPa = 6.895 psi. Refer to the Glossary at the end of this publication for a definition of kilopascal, and to the Unit Conversion Tables for the relationships between kPa and different English units of pressure measurement.

The most commonly used direct pressure measurement sensor is the *Bourdon tube*. It is an elliptical-cross-section tube made of somewhat elastic metal and formed into a shape like a "C" or a question mark. One end is open to a pressure source; the other end is sealed and connected to an indicator or, indirectly, to a pressure transmitter. (See Figure 24.) When pressure is applied at the one end, the force tries to straighten out the tube. The amount of movement sustained by the sealed end is proportional to the amount of pressure applied. The basic tube can be twisted into *helical* and *spiral* element shapes, also shown in Figure 24.

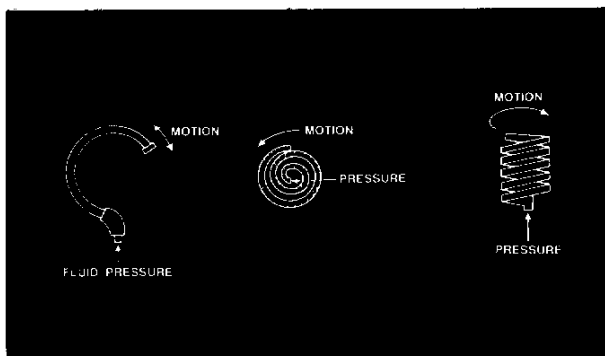


FIGURE 24 — The Bourdon Tube (left), Derived Spiral (center), and Helical (right) Elements

* SI is the common acronym for *Système International d'Unités* (the International System of Units). SI was adopted by the General Conference of Weights and Measures in 1960. It is discussed in Appendix B (Units of Measure) of this publication and in literature listed in the Bibliography (Appendix C).

A *diaphragm* element is a chamber with one side a thin disk made of springy metal and the other side open to pressure (see Figure 25). The center of the disk deforms (is pushed outward, away from the pressure source) in an amount proportional to the pressure applied. This basic element can be expanded into the bellows shape, also shown in Figure 25.

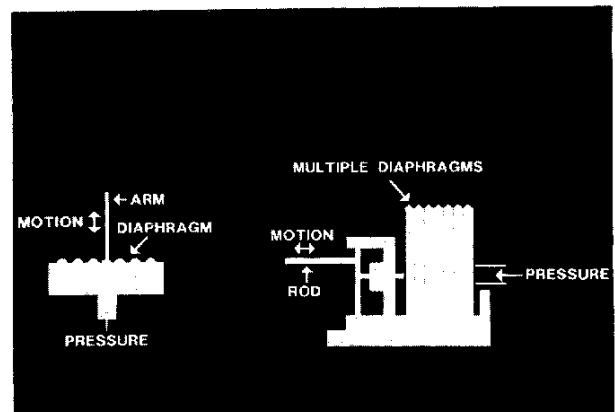


FIGURE 25 — The Diaphragm (left) and Multiple Diaphragm (Bellows) Elements

The pressure elements just discussed come in a wide variety of materials and ranges.

The Bourdon tube, bellows, and diaphragm gauges are commonly used because they have a mechanical motion output that can be connected directly to a position-type pneumatic controller, or can be incorporated into a pneumatic or electronic force-balance transmitter. They are also dependable and durable for long service life. Refer to the information on each type of pressure transmitter for their process applications (e.g., air, steam, liquid, differential pressure).

Unless otherwise specified, pressure elements are measuring *gauge pressure* (sometimes spelled "gage"), which makes no allowance for the fact that the earth is covered by atmosphere. The amount of atmospheric pressure, or "head," varies with elevation; for example, at sea level, atmospheric pressure is about 14.7 psi or 29.9 inHg but in Denver, with an elevation of one mile, atmospheric pressure is 12.5 psi or 25.4 inHg. When atmospheric pressure is taken into consideration in a measurement, we specify the pressure as *absolute*, i.e., gauge pressure plus local atmospheric pressure.

It is highly desirable to install pressure elements with a 1-valve manifold along the pressure line just before the instrument. This will allow the pressure line to be shut off and the instrument removed when necessary for maintenance or repair, without disturbing the process.

Temperature Elements

Heat, technically, is a form of energy added to a system, which cannot be measured directly. But heat causes a change in the quantifiable degree of hotness or coldness of the system, which can be measured; this is known as *temperature*.

Temperature is expressed as degrees of a scale, generally Fahrenheit or Celsius (Centigrade) but also possibly Kelvin or Rankine. (Refer to the Appendices "Units of Measure" and "Unit Conversion Tables" for both the conversion factors and an extensive listing of compared temperatures.)

A *filled thermal system* provides dependable direct measurement. It provides only indication; however, if desired, the measurement can then be transmitted to a recorder and/or controller. Other types of temperature elements — for example, thermocouples (TCs)*, resistance temperature detectors (RTDs)*, thermistors*, and optical or radiation pyrometers* — create a signal (usually electrical) which can be indicated, recorded, or used as the input to a controller.

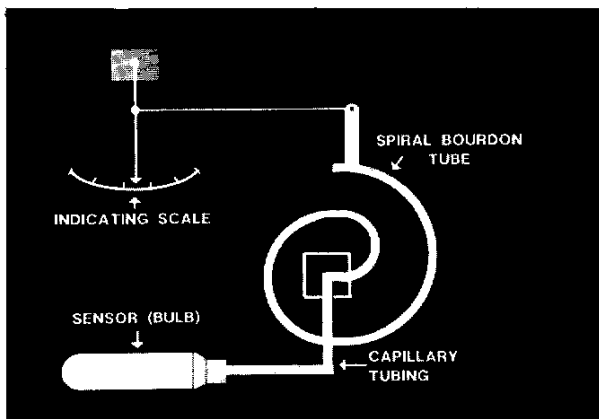


FIGURE 26 — The Parts of a Filled Thermal System

* A *thermocouple (TC)* is composed of two parallel wires of dissimilar metals which are connected at both ends. The metals react differently to changes in temperature and create an electric reaction at the connections (the measuring and reference junctions). The voltage produced can be measured and its magnitude used as a measure of the temperature in question. Different pairs of metals are used for different ranges.

A *resistance temperature detector (RTD)* is made of some material for which the electrical resistivity is a known function of temperature. Different materials are used for different ranges.

Thermistors are made of heat-treated metallic oxides. They can replace RTDs as temperature sensors, trading some benefits for other constraints.

Optical (radiation) pyrometers utilize an optical system to focus energy radiated from a body onto a sensing system. They are used to measure high temperatures and to measure any temperature where contact with the object is impossible.

Filled thermal systems are variations on the common stem thermometer and are based on the thermal expansion of gases and liquids. The basic technique has been used for many, many years.

Filled thermal systems consist of a sensing bulb connected through capillary tubing to a pressure- or volume-sensitive element such as a Bourdon tube which then connects to an indicating scale (Figure 26 shows the relationship of these components). The sensing bulb is generally installed in a *thermowell*, a long, thin metal case that conducts the temperature to the bulb, to protect the bulb from direct exposure to processes that might cause damage to it (e.g., corrosive materials, fluids with suspended abrasive solids, sanitary processes). (Figures 27a and 27b show an actual filled thermal sensor with thermowell, connection tubing, and direct-measuring element.) Filled thermal systems are simple, inexpensive, and, because they require no external source of energy, provide continuous measurement even during power interruptions.

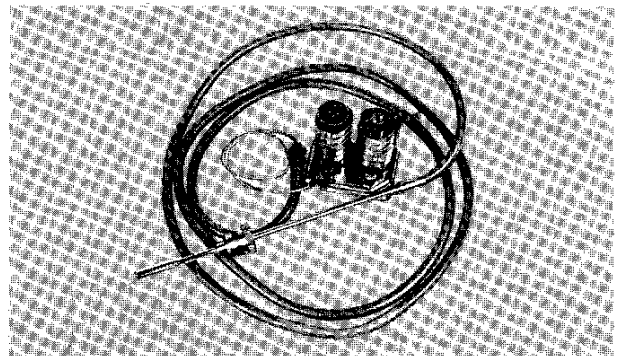


FIGURE 27a — A Filled Thermal System with Sensor, Connecting Tubing, and Mechanical Element

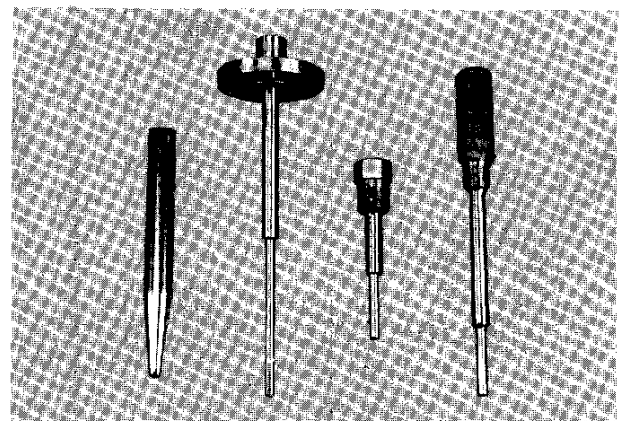


FIGURE 27b — Several Types of Thermowells for a Filled Thermal System

Filled thermal systems are used traditionally in the food, paper, and textile industries. They are ideal for remote locations where power is not available. They have also been applied successfully in industries using cryogenics

(very low temperature, such as that associated with liquid oxygen). The bulb can be protected from potentially corrosive process material or damaging applications by thermometer wells. For service in acids, pickling baths, plating and similar applications where the temperature is not excessive, special plastic-coated bulbs are available. An all-welded stainless steel bulb is available to meet sanitation requirements.

The transmitting medium (filling) in a filled thermal system can be vapor, gas, mercury, or other liquid. Each has different applications. The *Scientific Apparatus Makers' Association (SAMA)* has arranged these systems by number and letter, and has provided specifications for their use. The numerical classifications are I (liquid-expansion), II (vapor-pressure), III (gas-pressure), and V (mercury-expansion). Only Classes I, II, and III are

in wide use anymore and so they are the only ones discussed. A summary of the details following is provided in Table B.

Class I (liquid-expansion) systems are characterized by high accuracy, narrow spans, the smallest sensing bulbs, uniform scales, and capability for differential measurements. They are particularly recommended for measurements at or near ambient temperatures. Class IA devices are *fully compensated* for the effects of ambient temperature changes on the accuracy of the measurement and they have greater overrange protection than other devices. Class IB systems are *case compensated* (i.e., the motion element is compensated but not the capillary). Class I systems are complex and therefore expensive; however, the benefits often outweigh this disadvantage.

TABLE B — Application Data for Temperature-Measuring Systems

	Filled Thermal System — SAMA Class			Thermocouple**	RTD**
	IA,IB	IIA,B,C,D	IIIB		
Principle of Operation	Volume change	Pressure change	Pressure change	Electromotive force	Electrical resistance
Temperature Range	Fairly low to moderate	Low to moderate	Very low to high	Lowest to highest	Low to fairly high
Spans	Moderate	Moderate	Moderate	Narrow to wide	Very narrow to moderate
Response	Slow	Moderate	Moderate	Fast	Moderate
Relative Cost: • Purchase • Installation • Operating	High Moderate Low	Moderate Moderate Low	Moderate Moderate Low	Low Low* Moderate	Low Low* Moderate
Scale	Linear	Nonlinear***	Linear	Nonlinear	Linear
Overrange Capability	Medium	Least	Greatest	Unlimited	Unlimited
Sensitivity	Poor	Poor	Poor	Very Good	Very Good
Accuracy	Moderate	Moderate	Moderate	Poor	Very Good
Fill Medium	Liquid	Vapor	Gas	NA	NA
Distance to Receiver	Short	Short	Short	Long	Long
Sensor Size	Small	Medium	Large	Small	Small
Ambient Temp. Compensation	A: Full B: Case	Not required	B: Case	NA	NA

* General-purpose wiring only. Explosion-proof electrical requirements boost cost to highest.

** Electronic devices only (not detailed in this publication) whose ratings are averages for the various models available. Contact your Foxboro sales representative for further information.

*** Uniform motion or output with temperature may be accomplished for certain ranges by mechanical means.

Class II (vapor-pressure) systems are simple, inexpensive, and inherently accurate devices with long, trouble-free lives, and require no compensation for ambient temperature effects. Instruments follow the vapor-pressure curve of the filled fluid, and associated dials and charts are therefore nonuniform, with more widely spaced increments at high temperatures (see Figure 28). This may or may not be a problem. If the operating temperatures are in the outer portion of the chart, the record will be more readable, but if temperatures vary throughout the range, another system is the better choice. The temperature ranges of the Class II system with respect to ambient (indicated by alphabetical designation) must be strictly observed, or the device may operate erratically. Class IIA and IIB are designed to operate above and below ambient, respectively; IIC temperatures can cross ambient; and IID systems can operate above, below, or at ambient. Class IIA sensors have the fastest response of all filled thermal systems.

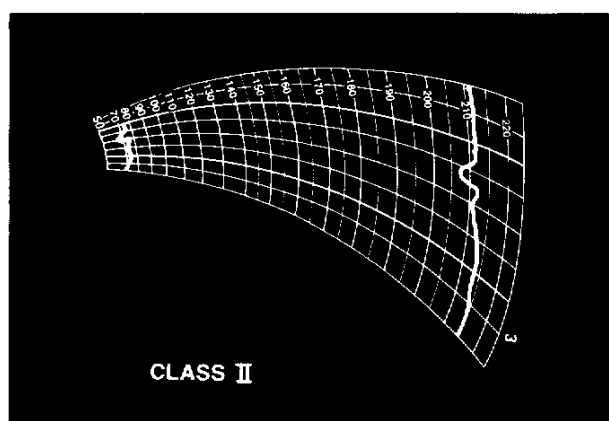
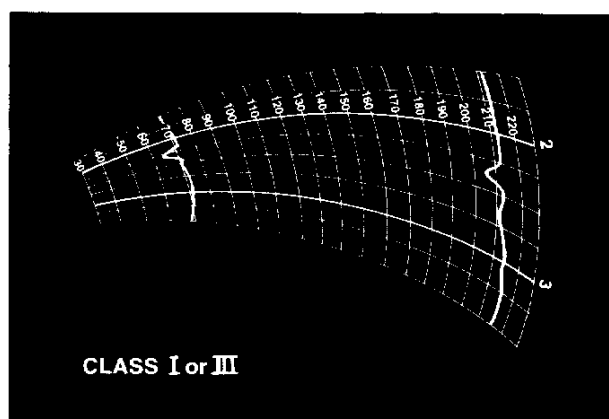


FIGURE 28 — Recording Charts for Filled Thermal Systems

Class III (gas-pressure) systems, which are second to the vapor-pressure devices in simplicity and cost, have the widest range of all filled systems. They generally do not require compensation. They have the largest sensors, but the sensors may be shaped to suit particular applications. For example, in duct-temperature averaging, the bulb may be constructed of a long length of tubing of small cross section. Conventional recorders are not recommended for temperature spans less than 200°F/111°C, but transmitters operating on force-balance principles can be utilized with spans as narrow as 50°F/28°C.

Other types of temperature-measuring devices were mentioned earlier — TCs, RTDs, thermistors, bimetallic devices, optical or radiation pyrometers. Thermocouples and RTDs are discussed in Section 6 of this publication (Type 33C emf-to-Pneumatic Transmitter and Type 34C Resistance-to-Pneumatic Transmitter, respectively). The others can be examined in detail in the books listed in the Bibliography at the back of this publication.

Flow Measurement Devices

There are many aspects to flow measurement, making it a complicated subject. Fluids flow, but they can have varying specific gravities, temperatures, and consistencies. Quantity of flow can be measured volumetrically with a positive displacement meter (e.g., a common household water meter) or indirectly by measuring the flow rate (almost all other applications).

The most common means of measuring process flow rate is by a *differential pressure* system, often called a *head meter system*: a calibrated restriction (called the *primary device*) in the flow line develops a pressure drop measured with sensor (called the *secondary device*).

Figure 29 illustrates, in cross section, the typical line pressure profile created by an orifice plate and how a *mercury manometer* would measure that pressure. (Mercury manometers are seldom used in processes today due to the dangers of mercury, but the device is useful here for illustrative purposes.) This predictable profile is the basis for flow measurement using differential pressure.

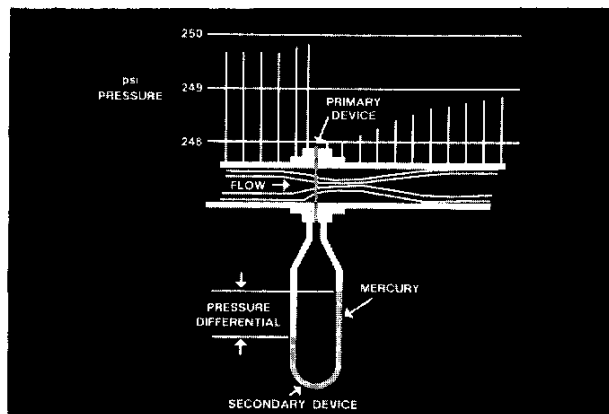


FIGURE 29 — Head Meter Cross Section with Mercury Manometer

The most common primary device is the *orifice plate* held in place in the piping with *flanges*. The most common secondary device is the *differential pressure transmitter*. The differential pressure transmitter senses the change in line pressure caused by the orifice plate, but is in a different package from the mercury manometer just described. A ΔP (differential pressure) transmitter has pressure connections like the manometer, each of which leads to a diaphragm sensor; the movement of the diaphragm indicates the change in pressure. The combination is inexpensive to implement, fits almost any size line, and can be used with most applications. It also can be applied to lines with reverse flows (requiring a measurement that crosses zero, such as -50 to $+50$ gpm), when installed with a *zero elevation* kit.

Of course, economic implications and the configuration and size of the piping are not the only things influencing choice of the primary and secondary devices. Characteristics of the flowing substance (e.g., temperature, solids content, viscosity) must also be considered. For example, the popular ΔP transmitter would be installed differently depending on the flowing substance (see Figures 30a through 30c), and it would not be appropriate at all in some applications (see Table C for a comparison of several primary methods of flow measurement). It is highly recommended that any ΔP transmitter be installed with a 3-valve manifold between the transmitter and the process piping; this isolates the transmitter from the process fluid so that the transmitter can be removed for maintenance or repair without interfering with the process itself.

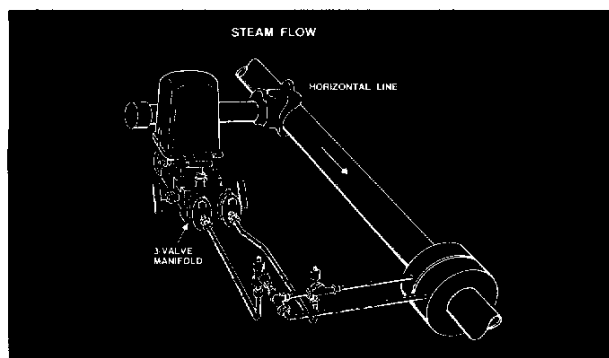
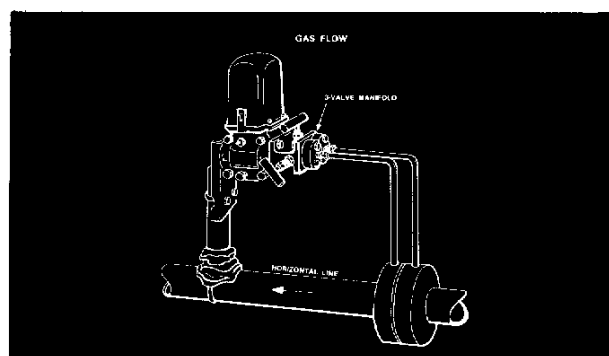
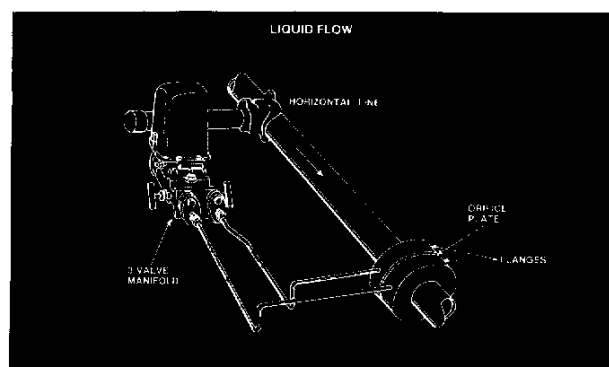
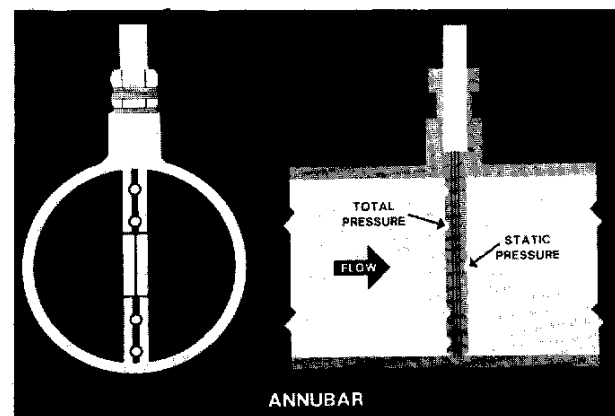
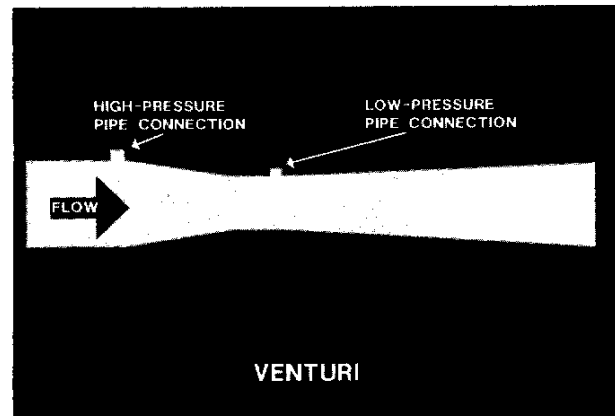
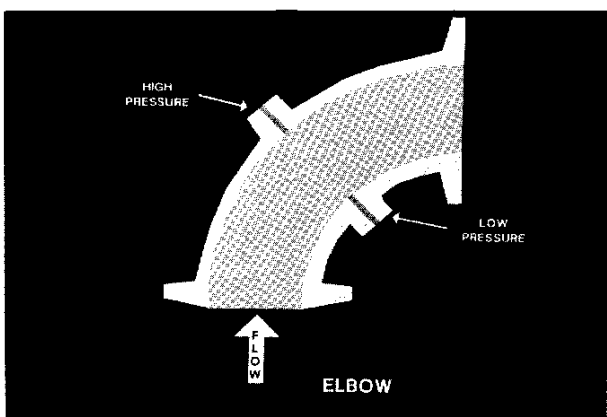
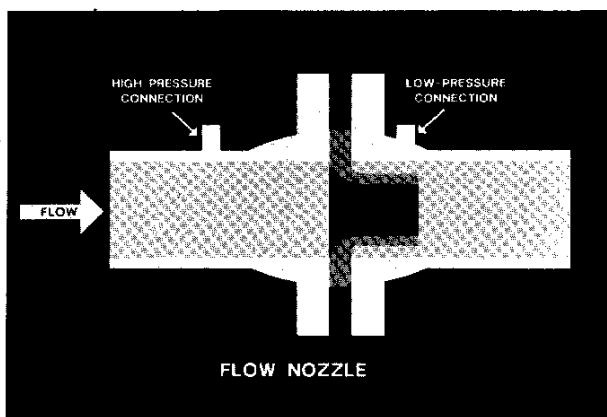
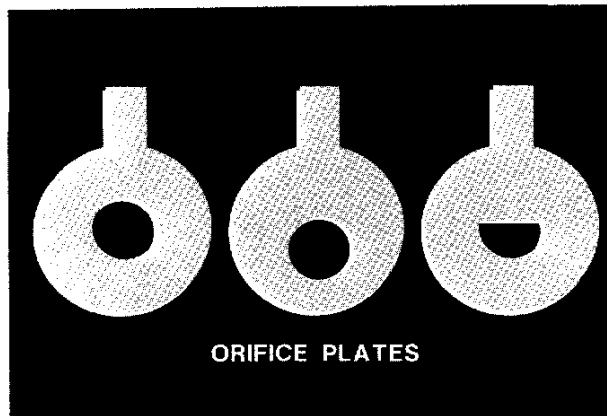


FIGURE 30 — Differential Pressure Meter in Flow Applications for Liquid (Figure 30a, top), Gas (Figure 30b, middle), and Steam (Figure 30c, bottom)

The major limitations of ΔP meters are low *rangeability* (difficulty in reading a wide range of flow rates) and a *square-root relationship in readout* (which must be accommodated in some way*). They also are unable to cope with certain types of fluids (e.g., slurries). Other types of flowmeters may be advantageous when these limitations are insurmountable.

Primary devices (restrictors) include the orifice plate, flow nozzle, elbow, Venturi tube, and Annubar, shown in Figures 31a through 31e.



FIGURES 31a through 31e — Primary Devices

The only Foxboro direct flow-rate measuring device is the Type 37 Diaphragm Measuring Element, described following. It is a secondary device which converts the measured pressure drop into motion via a drive bar.

Most flow-measuring instruments convert the measurement internally into a signal which can be transmitted to a controller or recorder. Typical of these secondary devices are the following, shown in Figures 31f through 31h.

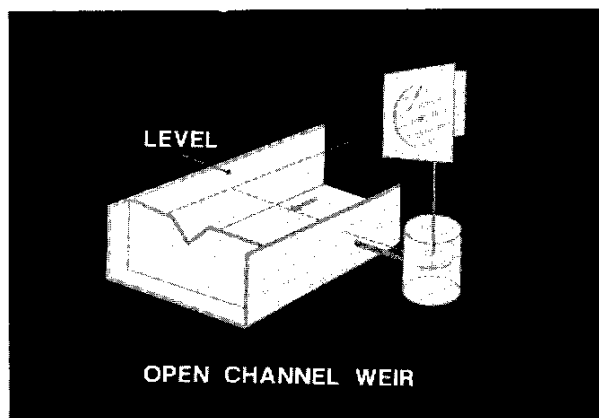
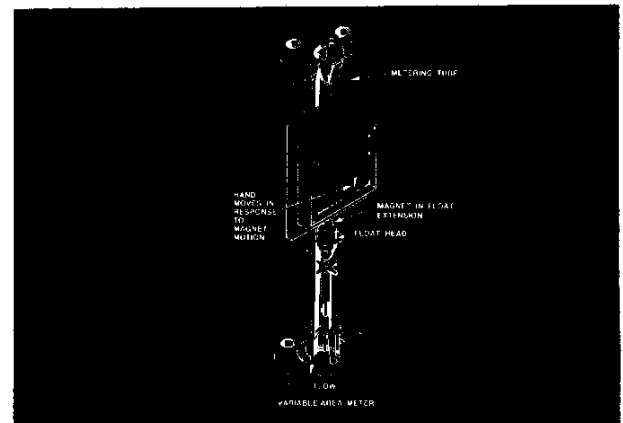
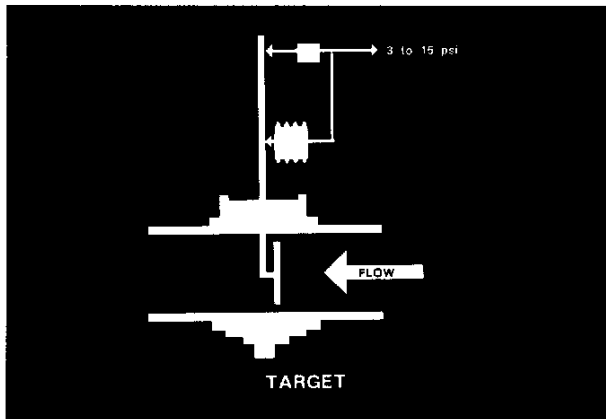
- The *target flowmeter* places a solid disk in the pipe. As the flow passes around it, the force exerted by the flow on the disk produces a flow-related (square-root) signal. The target meter is described in Section 6 of this publication.
- The *variable-area meter (rotameter)* suspends a float denser than the flowing fluid in a tapered tube. Its movement relates to flow and produces a signal that is linear with flow. Advantages: can measure

* Square-root relationships require the use of special output recording charts or the addition of a square-root extractor in the control loop (see Section 8, Pneumatic Computers).

low flow rates. Limitations: The fluid must be reasonably clean and the maximum flow rate is limited by the size of the meter.

- *Open-channel devices*, primarily used for handling water and waste, use a form of primary device called a weir (a plate with a notch in it) or a flume (a restriction in the channel) plus a secondary device to produce a signal that is related to flow. Advantages: flumes can handle waste with suspended solids; weirs can handle large volumes.

Besides these types of flowmeters, several popular and useful types of flowmeters have electrical output signals only and therefore are not detailed in this publication. They include the turbine meter, the vortex meter, and the magnetic flowmeter. Refer to literature listed in the Bibliography for information on them. The most used types of flowmeters are compared in Table C.



FIGURES 31f through 31h — Primary and Secondary Devices

Table C — Application Data for Flow Measurement Devices

	Differential Pressure with					Target	Turbine*	Mag Flow*	Vortex*
	orifice	Venturi	nozzle	Annubar	elbow				
Accuracy	Good	Good	Good	Good	Poor	Fair	V. Good	V. Good	V. Good
Differential Produced	V. Good	Good	Good	Fair	Poor	NA	NA	NA	NA
Pressure Loss	High	Mod-erate	High	Low	None	High	High	None	Low
Service:									
• Dirty	Poor	Good	Good	V. Poor	Poor	V. Good	Poor	V. Good	V. Good
• Corrosive	Good	Good	Good	Good	Good	Good	Poor	V. Good	Good
• Slurries	V. Poor	Good	Poor	Poor	Good	Fair	V. Poor	V. Good	Fair
• Viscous	***	Good	Good	NA	Fair	Good	Fair	V. Good	Poor
• Liquids with Vapors	Good**	Good	Good	Fair	Fair	V. Good	Poor	V. Good	V. Good
• Vapors with Condensate	Good**	Good	Good	Poor	Fair	V. Good	Poor	NA	V. Good
Calibration	Sq. Rt.	Sq. Rt.	Sq. Rt.	Sq. Rt.	Sq. Rt.	Sq. Rt.	Linear	Linear	Linear
Initial Cost	Low	High	Mod-erate	Low	Low	Mod-erate	Mod-erate	High	Low
Ease of Changing Capacity	Good	Fair	Fair	Fair	Fair	Poor	Good	V. Good	Good
Ease of Installation	Good	Fair	Fair	Good	Good	Good	Fair	Fair	V. Good

* Electronic transmitters only, not detailed in this publication. Contact your Foxboro sales representative for further information.

** Some piping restrictions must be observed.

*** Suitability depends on type of orifice plate.

Type 37 Diaphragm Measuring Element

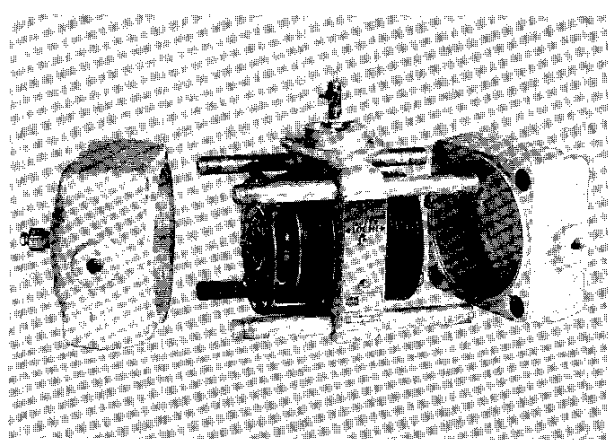


FIGURE 32 — Type 37 Diaphragm Measuring Element

The Type 37 Diaphragm Measuring Element, shown in Figure 32, is a device that provides direct indication of flow rate. It provides an accurate means of measuring low-range differential pressure in a wide variety of applications. The Type 37 has overrange protection and ambient temperature compensation for year-round accuracy and stability. It requires no power for operation and, therefore, will continue operation during power outages.

The element, usually mounted with a recorder in a case, translates differential pressure to angular shaft position. The shaft can be mounted to selected instruments to indicate, record, and control.

Applications include measuring flow of liquid, gas, or vapor by differential pressure, or measuring liquid level within a vessel. A self-draining feature makes it particularly suited to wet gas measurement.

The Type 37 Diaphragm Measuring Element uses two diaphragms connected by a chamber filled with fluid. A pressure increase in the high-pressure chamber compresses the diaphragm assembly (Item 7 in Figure 33) and displaces the fluid through the passage into the low-pressure diaphragm assembly. The linear motion of the diaphragm assembly is transmitted through the link to the inner end of the drive lever assembly (Item 4). The outer end of the drive lever, pivoting on the bellows seal pivot point (Item 5), moves to drive the pen arm. This motion continues until the force of the range spring (Item 3) equals the force of the differential pressure in the diaphragm assemblies.

The bimetallic temperature compensator (Item 6) inside the high-pressure diaphragm assembly adjusts the capacity of the low-pressure diaphragm assembly to match the volume change in the liquid filling due to ambient temperature effects. Precise damping* adjustments can be made at the damping screw (Item 1), while the instrument is in service, to provide accurate control of instrument response.

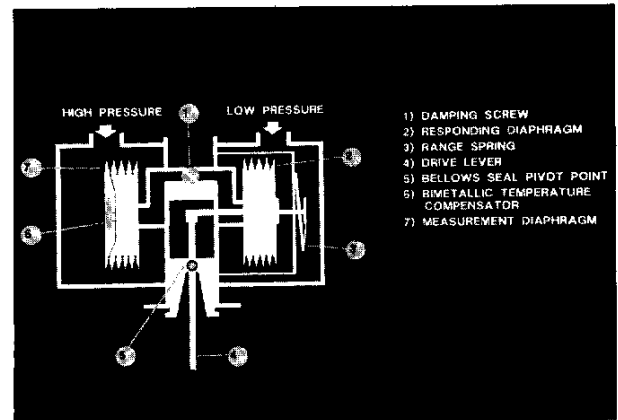


FIGURE 33 — Type 37 Diaphragm Mechanism

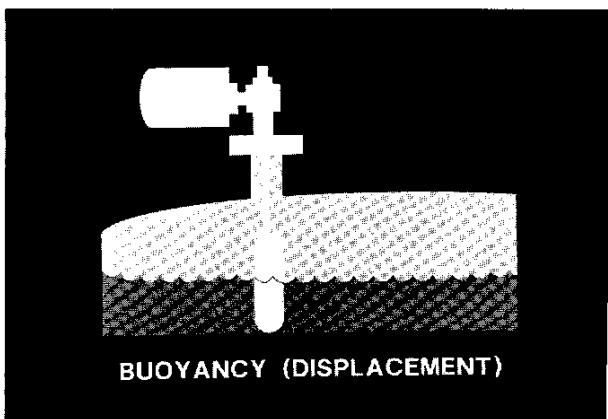
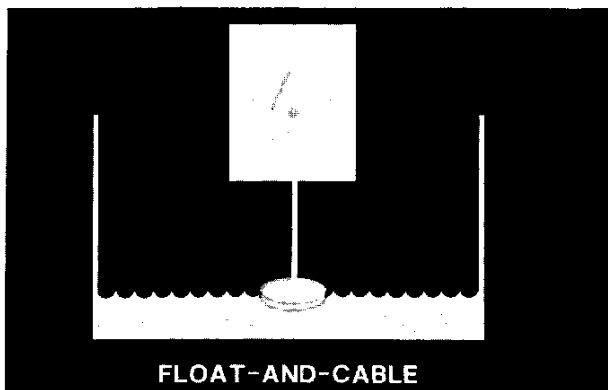
* Damping affects the time required for a system to stabilize after an abrupt process upset.

Level Measurement Devices

The typical process plant contains many tanks, vessels, and reservoirs. Their function is to store or process materials. Accurate measurement of the contents of these containers is vital. The material in the tanks is usually liquid, but occasionally it may consist of solids. Of course, gases have no level.

Level measurement of liquids may indicate the precise level at all times or it may indicate only when the level reaches a predetermined point. This discussion is limited to the former type of measurements.

An analog signal indicates either actual volume (gallons or litres) or percent of volume. In general, the most common level measurement is differential pressure, which uses measuring instruments already discussed.

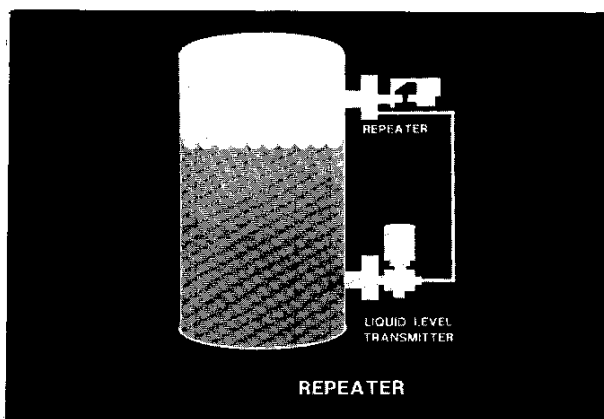
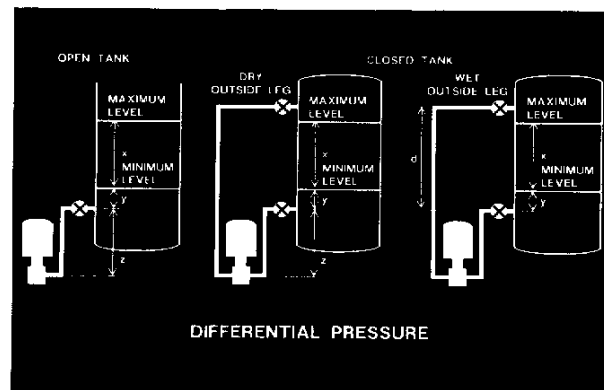


FIGURES 34a through 34d — Simple Level Measurement Devices (Figures 34e and 34f on next page)

* The *specific gravity* (SG) of a liquid is its quantification of relative density. It is the ratio of the density of a fluid at a particular temperature to the density of water at a particular temperature. Both temperatures (and even the quality of the water) are generally indicated; for example, for scientific purposes, the reference is double-distilled water at 4°C (39.2°F). For most applications, distilled water is implied and 15.6°C (60°F) is the reference. (The American Petroleum Institute, called API, has published much data and many curves and tables on specific

While the discussion of the various level measurement methods that follow includes primary advantages and disadvantages for each type, some problems occur in level measurement that complicate the choice of sensor. One of the problems is possible waves (called *hydraulic resonance*) that can occur in any vessel with an open surface, noise, and boiling liquid/condensing vapor thermal interfaces. Another potential problem is variable *specific gravity**. However, these are beyond the scope of this publication. The reader is advised to consult some of the more detailed books listed in the Bibliography at the end of this publication for specific information.

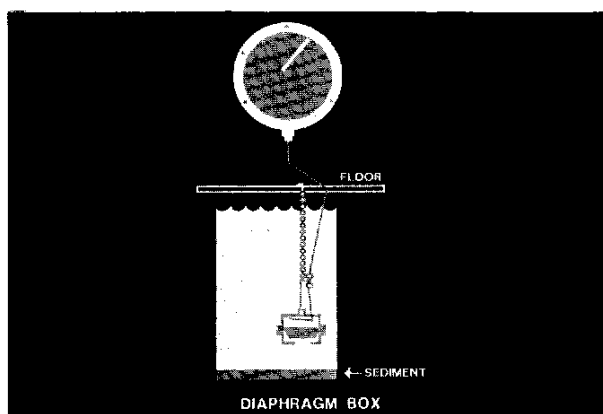
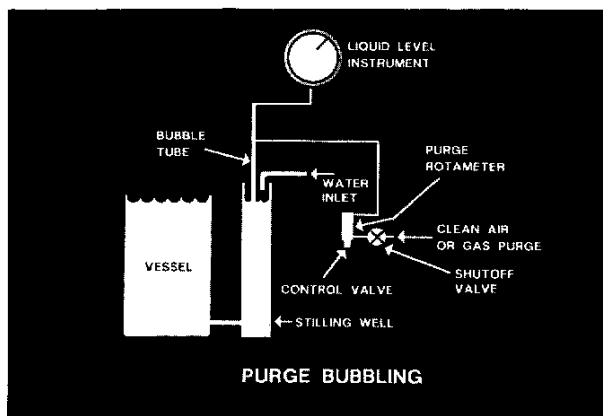
This discussion of level measurement will be confined to open (vented) tanks and closed (*pressurized** or *evacuated**) tanks with none of the problems noted above. Liquid level measurement sensors for such applications include the following and are shown in Figures 34a through 34f.



gravities of many liquids.) Specific gravity can vary with temperature and, to a much smaller degree, with pressure. If there is a wide variation in either of these conditions, a corrective factor (beyond the scope of this publication) must be introduced into control calculations.

A *pressurized vessel* is a closed vessel which may operate at pressures above atmospheric.

An *evacuated vessel* is a closed tank from which all gas has been removed.



FIGURES 34e and 34f — Simple Level Measurement Devices (continued)

- *Float-and-cable*, indication-only instruments which convert liquid level to mechanical motion. These are direct-measuring elements primarily used in open tanks. Advantages: simplicity of operation and insensitivity to density changes. Limitations: reasonably clean liquids only, possible problems with turbulence.
- *Displacer (buoyancy)*, a type of force-balance transmitter with a heavy element (displacer) immersed in the liquid which senses the force pushing it up (buoyancy). The force is directly related to amount of liquid displaced. Advantages: wide range of elements available for differing applications; application to level, liquid-gas interface or density measurement; and application to special situations such as glass-lined tanks, high temperature service, tanks where pressure connections are not practical, and density applications. Limitation: difficulty in measuring extremely turbulent process conditions.
- *Pressure transmitters (head* meters)*, which sense the pressure at the bottom of the tank and compare it to atmospheric pressure (for open tanks) or to the pressure at the top of the closed tank. This is the most common type of level measurement, with

variations available to accommodate almost any process application (four of the most common variations, all for closed tanks — dry leg, wet leg, repeaters, and pressure seals — are explored in Table D).

- *Purge bubbling or bubble tube* is a form of pressure measurement which measures the pressure required to force a gas into a liquid at a point beneath the surface (changes in level cause the air pressure in the bubble tube to build or drop). It is good in applications with corrosive or solids-bearing liquids.
- The *diaphragm box*, used primarily for water level measurement in open vessels, is very much like the diaphragm element discussed in the section on pressure, except that this is larger, actually a box, sealed and filled with air. One side is a slack, thin, flexible diaphragm which deforms in response to pressure; an attached instrument senses the deformation and indicates it.

Table D summarizes the applications for the primary level measurement methods discussed above.

More elaborate forms of liquid level measurement include capacitance, conductance, radiation (nucleonic), ultrasonic, and thermal methods. These forms are beyond the scope of this publication but can be examined in more detail in the books listed in the Bibliography.

Density is important in process plants as an inference of composition, of concentration of chemicals in a solution, or of solids in suspension. Many methods are available to determine concentration, most utilizing the effect of density changes on weight as a measurement. Note that, for measurements to be reliable, liquid in the tank must be of uniform density; this is usually accomplished by agitating the liquid sufficiently to maintain a uniform mixture without causing undue velocity effects.

Density measurement uses some of the methods of level measurement, such as displacement/buoyancy, differential pressure, ultrasonics, radiation, and temperature. A method used specifically for density measurement is vibration, in which a reed, plate, or filled cylinder is immersed in the liquid and vibrated from externally applied energy. The natural frequency of vibration is measured, since it changes with density.

* *Head* (properly, hydrostatic head) is the term used for the result of a pressure equation. In level measurement, the equation is

$$H = SG \times d \quad \text{where } H = \text{head}$$

SG = specific gravity
d = the maximum height of the liquid above the transmitter sensor

TABLE D — Application Data for Level Measurement Devices

	Buoyancy	Purge Bubbling	Head				
			Vented	Dry Leg*	Wet Leg*	Repeater	Seals
Tank	Open or closed, pressurized or evacuated	Open	Open	Closed — pressurized	Closed — pressurized	Closed — pressurized or evacuated	Closed — pressurized or evacuated
Service:							
• Clean	Good	Good	Good	Good (pipe- or flange-mounted)	Good	Good	Good
• Viscous	Fair	Fair	Good	Flange-mtd only	Good	Good	Good
• Corrosive Fluids	Fair	Fair	Good	Flange-mtd only	V. Good	Good	V. Good
• Corrosive Vapors	Fair	Fair	Fair	NA	V. Good	Good	V. Good
• Condensing Vapors	Good	NA	Poor	Poor	V. Good	Good	NA
• Negative Pressure	Good	NA	NA	Good	Good	Good, if modified	Poor
• Vacuum	NA	NA	NA	NA	NA	NA	V. Good
Ease of Installation	Good	Fair	Good	Good	Fair	Fair	Good
Cost	High	Low	Low	Low	Low	Moderate	Moderate
Ease of Service	Difficult	Simple	Simple	Moderate	Moderate	Moderate	Moderate

* A wet leg is a long process connection filled with liquid; a dry leg is a long process connection filled with gas.

Analytical Measurement Devices

Analytical measurements seek to define the contents of a process stream and thus enable control of its composition. Some analytical measurements are physical; others are electrochemical.

The physical types include humidity, density, differential vapor pressure, boiling-point rise, chromatography, ultraviolet, infrared, and turbidity. Others, such as spectroscopy, osometry, and polarography, are laboratory techniques and not usually associated with automatic control.

Many other analytical measurements, such as conductivity (often called specific conductance), *pH* (hydrogen ion concentration), *ORP* (oxidation-reduction potential, also called Redox), and specific ion concentration, are electrochemical. In general, electrochemical measurements measure activity rather than concentration.

The application of these analytical measurements to process control is briefly described below. However, since most analytical sensors produce an electrical signal, their

full description is beyond the scope of this publication. Contact your Foxboro sales representative or refer to the Bibliography for more information.

- humidity (the amount of moisture in gases, expressed in several ways — relative humidity, absolute humidity, and dew point)
- density (which was discussed in the section on level measurement)
- differential vapor pressure
- boiling-point rise
- chromatography (used for gas analysis)
- ultraviolet
- infrared
- turbidity
- conductivity (measures the activity of all ions present)
- *pH* (a measure of the degree of acidity or alkalinity of an aqueous solution)
- *ORP* (determines the oxidizing or reducing properties of a chemical reaction, and applies to any material which exchanges electrons in a chemical reaction)
- specific ion concentration

SECTION 6 — PNEUMATIC TRANSMISSION HARDWARE

The direct-measuring elements discussed earlier are purely mechanical interconnections of components with the measurement available only at the instrument. Where the measurement is used only as a reference, direct-measuring elements may be sufficient.

It is more common, though, for measurements from all field instruments to be brought to a central control area where control strategy can be performed. This requires the addition of air pressure — pneumatics — to the measuring element so that the measurement can be transmitted to the control room.

A pneumatic transmitter internally converts the measurement into a pneumatic signal which can be transmitted. There are two primary types of transmitter mechanisms: position and force-balance. Their standard configurations, as described below, or variations thereon are used in the rest of the instruments described in this publication. (Variations are explained in the description of each product.)

The basic transmitter mechanisms comprise a sensor (usually a diaphragm or bellows as described earlier in this publication), a flapper-nozzle assembly to detect errors, a bellows to provide feedback, and a relay to amplify the output signal and to rebalance the system. Other components, such as span and zero adjustments, are typically added.

The basic error detection mechanism used in pneumatic controllers and transmitters is the *flapper-nozzle unit*. A reducing tube drops the full-scale reference air supply to the minimum scale (e.g., 3 psi/20 kPa in a 3 to 15 psi/20 to 100 kPa instrument) and allows it to flow through a tube to the open nozzle. If the nozzle is unobstructed by the flapper, as in steady state (on the left in Figure 35), the pressure in the tubing will be low since the air may flow freely from the nozzle. If the flapper moves closer to the nozzle (as after a change in measurement, shown on the right), the outflow of air will be obstructed, causing the pressure in the tubing to rise, possibly to full-scale value (here, 15 psi/100 kPa).

The air pressure in the tubing is very low, so a *relay* (amplifier) is installed in the line to amplify the signal. In this way, a small change in the flapper movement (approximately 0.0006 inch), representing a small change in the input signal (say 1 psi/7 kPa), can produce a proportionally much larger and essentially linear relay output (in this case, 12 psi/80 kPa on a 3 to 15 psi/20 to 100 kPa scale).

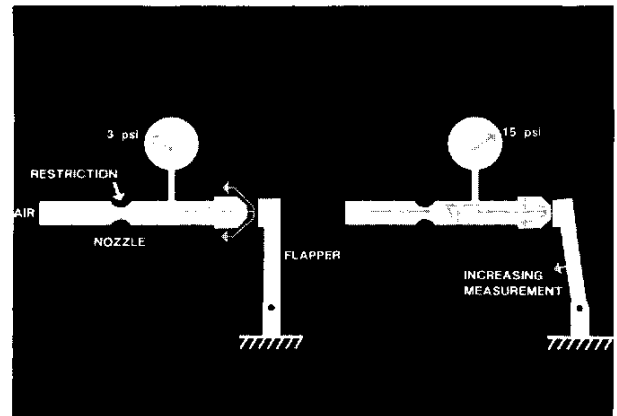


FIGURE 35 — Flapper-Nozzle Assembly in Original State (left) and with Nozzle Obstructed Due to Change in Measurement (right)

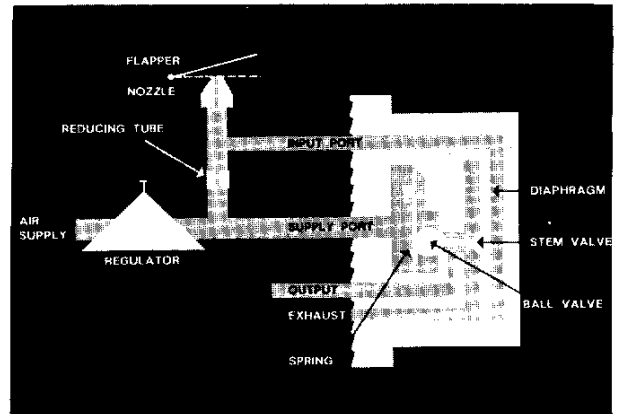


FIGURE 36 — Pneumatic Relay

The air supply enters the relay (Figure 36) through a supply port to provide a reference for diaphragm movement. The input signal (determined by the flapper-nozzle relationship) enters the relay through the input port. The input signal pressure causes the diaphragm to deform, moving its attached stem valve. As the input signal increases, the stem pushes the ball valve against a spring, allowing the air supply to pass through the relay body to the output.

When input pressure is in midrange, a proportional part of the supply pressure is exhausted. Further increased input pressure causes the stem valve to close off the exhaust port entirely so the supply pressure equals the output pressure. Decreasing input pressure reverses this until zero input allows no supply pressure to reach the output, at which point the output is the air pressure of the exhaust. Because the diaphragm has a large area, small pressure changes on its surface result in a significant force to move the ball valve.

Position-Type Pneumatic Transmitters

In a position-type pneumatic transmitter, the sensor is linked to the flapper of the flapper-nozzle assembly, often through an indicating pointer (shown in Figure 37). The nozzle is connected to the input of the relay, and the feedback bellows is connected to the output of the relay.

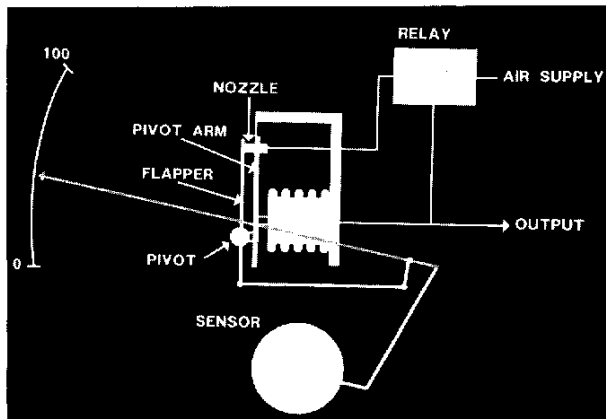


FIGURE 37 — Position-Type Mechanism

In a steady state, all of these components maintain constant relative positions: the sensor and pointer rest; air is exhausted through the nozzle at the flapper and establishes a pressure; and the air pressure in the relay is amplified and fed through tubing to the output and to the bellows which is connected to the flapper linkages.

When a measurement increase causes the sensor to move, the linkage causes the flapper to change and this changes the flapper-nozzle pressure. Increased pressure in the relay changes the output pressure, which also changes pressure in the bellows. The bellows expands. This movement reaches the flapper through linkages; however, it is counter to the movement from the sensor. The two movements eventually balance, restoring the position of the flapper relative to the nozzle.

In general, the position-type mechanism depends on movements of components to indicate a change in measurement and to restore balance to the system.

* A bell element, used to measure very low ranges of differential pressure with good accuracy, looks something like a bell on top of a box. Process pressure is applied to the top and bottom of the "box"; a diaphragm between them registers minute changes in pressure. A hair element, a band of specially treated human hairs which change in length as humidity changes, indicates direct measurement of relative humidity.

45 Series Pneumatic Transmitter

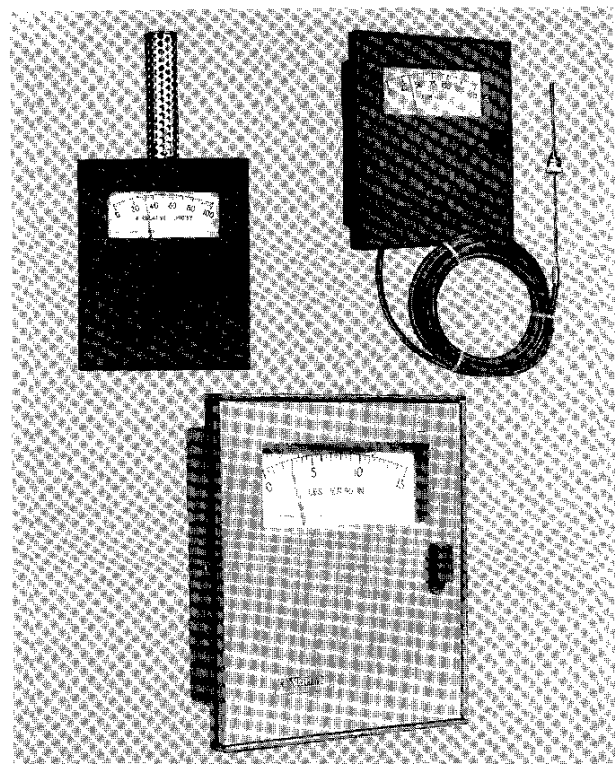


FIGURE 38 — 45 Series Pneumatic Indicating Transmitters for Humidity Measurement (left top), for Temperature Measurement (right top), and for Vacuum, Pressure, Differential Pressure, and Level Measurement (bottom)

Due to the variety of measuring elements available with this instrument, the 45 Series Pneumatic Indicating Transmitter can measure most basic process variables. It transmits a standard proportional pneumatic output signal to remote receivers such as controllers and recorders. The direct-measurement element selected provides continuous local indication, even upon loss of air supply. The case is specially designed to provide protection from corrosive atmospheres, weather, and flame; the interior parts are protected from damage due to dust, dirt, or fumes by a continuous flow of air (called an air purge).

The measuring elements which can be selected include: filled thermal systems Class I, II, and III; bellows, spiral, helical, and diaphragm pressure elements; diaphragm, bell*, and mercury manometer differential pressure elements; hair humidity measuring elements*; and certain RTDs and thermocouples (which require an electrical connection).

The operation of the 45 Series Pneumatic Transmitters is based on the standard Foxboro position-type mechanism with slight variations from unit to unit due to the sensing element attached to the pointer.

44 Series Pressure or Temperature Transmitter

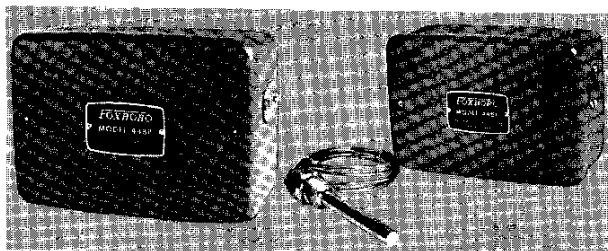


FIGURE 39 — 44BP Pressure Transmitter (left) and 44BT Temperature Transmitter (right)

The 44BP and 44BT Pneumatic Transmitters measure a range of pressures or temperatures, respectively, transmitting a standard pneumatic signal proportional to the measurement. (Nonstandard pneumatic signals are also available.) There is no local indication on the standard unit, although an optional indicator is available on the 44BP. Compact and weatherproof, the 44B Transmitters provide high accuracy at low cost.

The 44BP offers a number of pressure ranges and spans for many industrial environments through a selection of direct-measuring elements and element materials. Pressure seals* can also be used.

The 44BT uses standard filled thermal systems to offer a number of temperature ranges and spans.

Both instruments operate on the same principle, receiving a direct measurement (from a sensor to the spiral element shown in Figure 40) and converting it to a signal using the standard Foxboro position mechanism described previously in this publication.

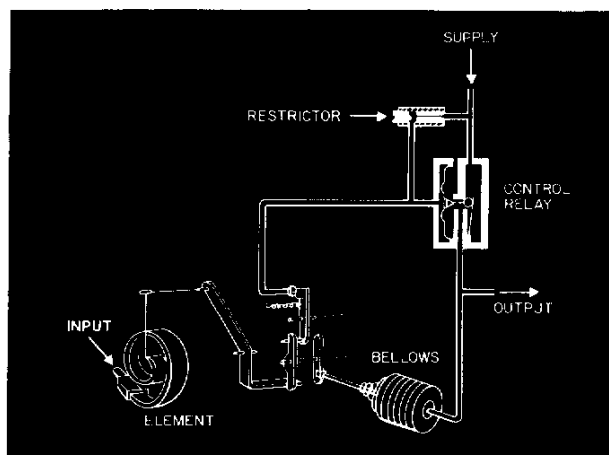


FIGURE 40 — Position Mechanism Operating the 44BP/44BT Transmitters

* Mechanical pressure seals are used to isolate corrosive or viscous products, or products that tend to solidify, from the measuring element. A diaphragm contacts the process; its movement is transferred through the filling fluid in connecting tubing to the indicating or transmitting mechanism.

Type CP Position Transmitter

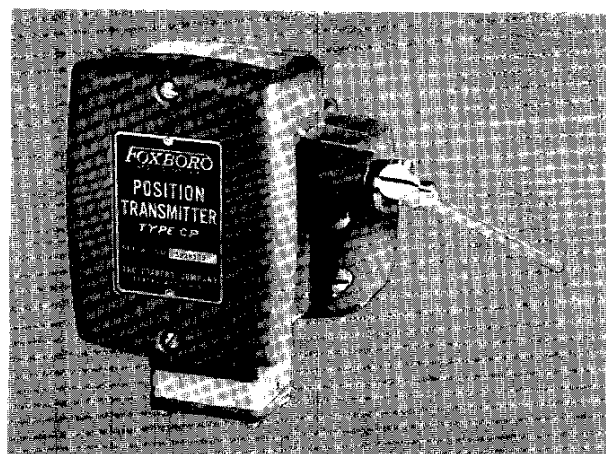


FIGURE 41 — Type CP Position Transmitter

The Type CP Pneumatic Position Transmitter has an operating lever that rides on a moving edge or surface or which can be linked to another moving point like a valve stem. Changes in angular position are measured and a proportional standard pneumatic output signal is generated. This output signal may be used to operate pneumatic indicators, recorders, or controllers. An internal air purge and gasketed cover protect the mechanism from damage due to vapor or dust.

The Type CP Position Transmitter uses a variation of the standard position-type mechanism. It is illustrated in Figure 42. When the lever (F) is not in motion, the system is in balance. Clockwise rotation of the operating lever rotates the flexure assembly (B) so that its ball tip moves away from the flapper (C), causing the flapper to cover the nozzle (D) and restrict the air flowing from the nozzle. The restricted air pressure is amplified by the relay (E) and directed to the feedback bellows (A) and to the output. The bellows expands, pushing the ball tip of the flexure assembly so that it restores the original relationship of the flapper to the nozzle.

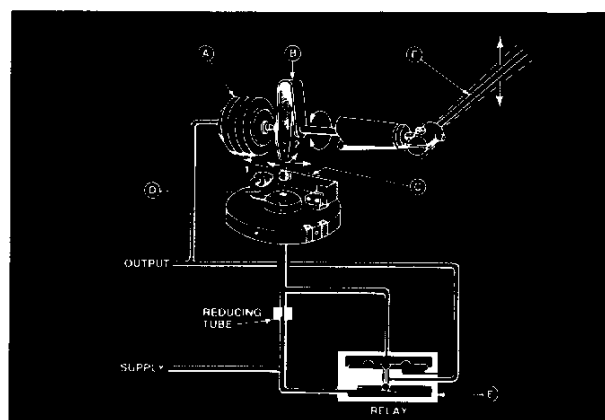


FIGURE 42 — Operating Mechanism of Type CP Position Transmitter

Force-Balance Pneumatic Transmitters

Figure 43 shows a force-balance transmitter mechanism. The sensor is the bellows capsule at the bottom. (Other direct-reading elements could be used, depending on the measurement range.) A rigid bar (the force bar), attached to the sensor at one end and to a flexure connector at the other, replaces the pointer and linkages in the flapper-nozzle assembly. The flexure moves horizontally with change in measurement, bringing the flapper closer to the nozzle. As with the position-type mechanism, the nozzle pressure is transferred to one side of the relay. This changes the output pressure and the amount of pressure transferred to the feedback bellows.

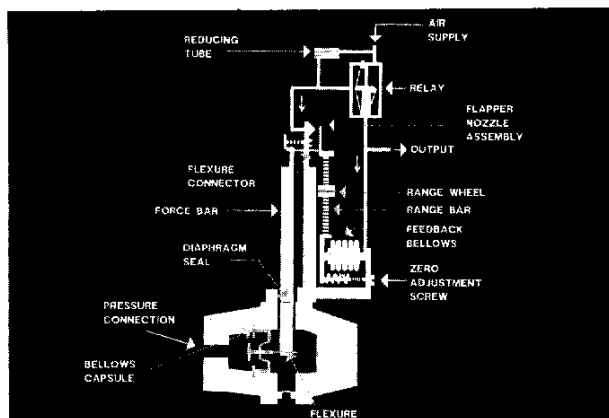


FIGURE 43 — Force-Balance Mechanism for Gauge Pressure Measurement

Other modifications from the position-type mechanism include the reducing tube between the nozzle and the relay, the span adjustment between the feedback bellows and the flapper, and the reference adjustment in tandem with the feedback bellows.

When the pressure measurement at the capsule changes, the movement in the capsule is transferred to the force bar. It moves slightly, using the diaphragm seal as a fulcrum. This moves the flexure, causing a slight change in the flapper-nozzle relationship which feeds back to the relay as in the motion mechanism. The range bar is repositioned by the feedback, acting as a pivot to reposition the flexure which restores balance to the system. The output pressure established by this balance is the transmitted signal and is proportional to the pressure applied to the measurement capsule.

In general, the force-balance mechanism uses the force and opposing force of air pressure to indicate a change in measurement and to reestablish the balance of the system. This mechanism is able to measure pressure ranges from a few inches of water to thousands of psi, converting the measured pressure into a proportional signal that may be transmitted over a reasonable dis-

tance. A receiving instrument is then employed to convert the signal into a pen or pointer position or a measurement input signal to a controller.

Total motion in a force-balance system is extremely small, resulting in minimum hysteresis and excellent repeatability.

Force-balance transmitters generally have an accuracy of ± 0.5 percent of span. Their ranges also generally can be suppressed or elevated. Refer to the current data sheets on each product for specifications.

12A Series Pneumatic Temperature Transmitters

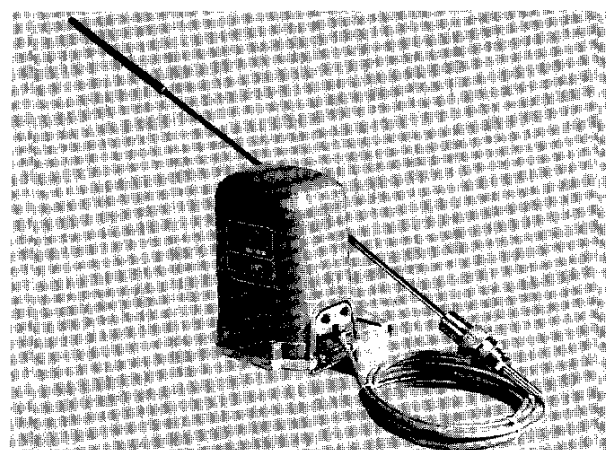


FIGURE 44 — 12A Pneumatic Temperature Transmitter

The 12A Pneumatic Temperature Transmitter is a non-indicating measuring device using a standard Class IIIB filled thermal system. Its standard pneumatic output signal is proportional to the temperature at the sensor and is transmitted to a pneumatic indicator, recorder, or controller.

The 12A is compensated for effects of ambient temperature. The exhaust of the relay purges the air in the housing to protect transmitter components from corrosive atmospheres. The instrument zero can be elevated or suppressed 100 percent; the range can be changed in the field with only a few simple adjustments.

When transmission distances are long or the capacity of the thermal process is large, an optional derivative unit reduces the response time of the 12A Transmitters by compensating for thermal lags.

The 12A Transmitter uses a variation (shown in Figure 45) of the standard force-balance mechanism. In this mechanism, the force bar both relays the movement from the process (thermal system capsule) and acts as the flapper to the nozzle. Movement of the force bar

is moderated by the base elevation spring (which establishes the relationship between nozzle pressure and the transmitter output) and the temperature compensating bellows.

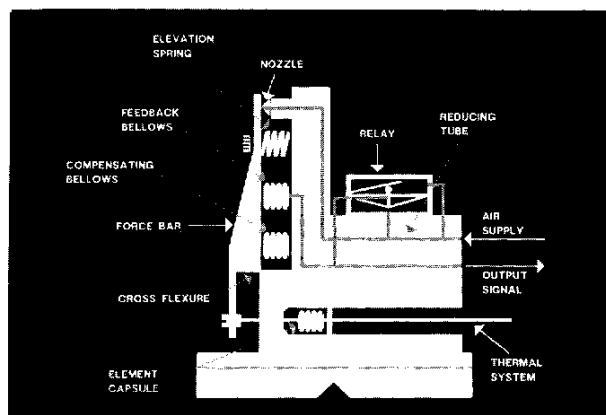


FIGURE 45 — Force-Balance Mechanism for the 12A Series Pneumatic Temperature Transmitter

33C Series emf* Pneumatic Transmitter

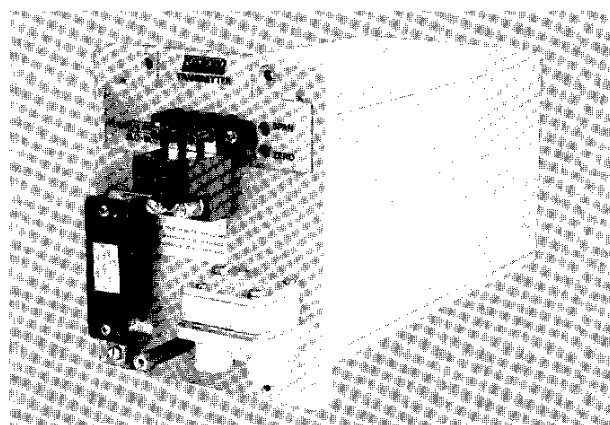


FIGURE 46 — 33C Transmitter

The 33C emf* Transmitter receives a dc millivolt or voltage signal from a thermocouple, tachometer*, or other electronic device. It converts the electrical input to a proportional standard pneumatic output that is linear with respect to the input. The ambient-temperature-compensated output is typically used as an input to a pneumatic controller or recorder.

The 33C can be adjusted to accept practically any standard voltage input used worldwide. It accommodates spans from very narrow to very wide, although some wide spans require an optional voltage divider on the input circuit. Elevated and suppressed zero ranges are available. Intrinsically safe* thermocouple input leads can be used in hazardous locations.

The electric-to-pneumatic conversion utilizes a unique *galvanometric** motor (see Figure 47) which consists

of a coil wound with fine copper wire suspended on flexures around a magnet. An electric input signal received from the thermocouple causes the coil to start to rotate around the magnet (although its movement is limited by the flexures). The rotation of the coil causes the integral flapper (see Figure 48) to move toward or away from the pneumatic nozzle. From this point, the mechanism behaves like the standard force-balance mechanism described earlier.

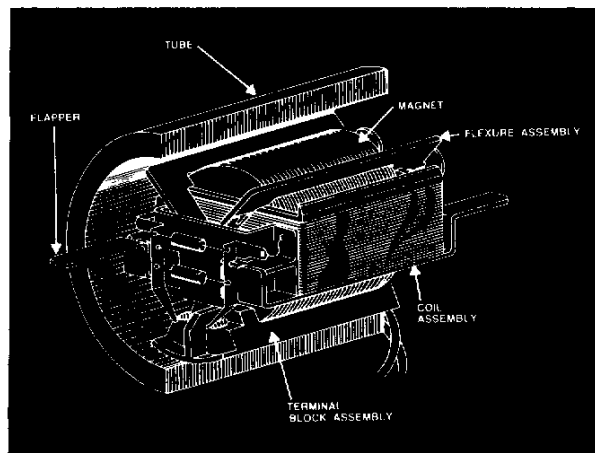


FIGURE 47 — Unique Galvanometric Motor of the 33C

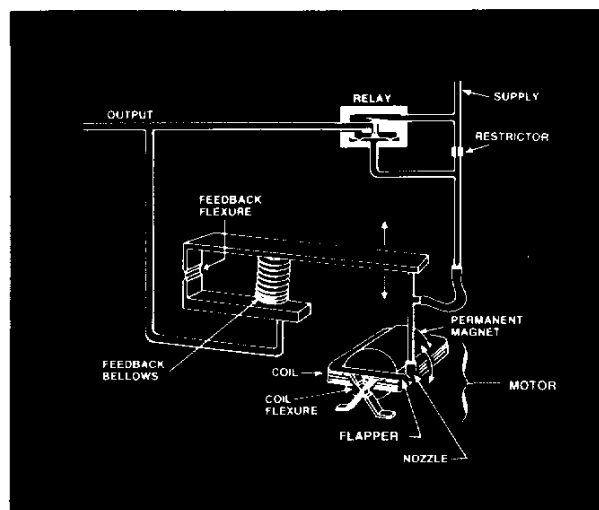


FIGURE 48 — Force-Balance Mechanism of the 33C

* *Electromotive force* (abbreviated emf) is generated in a closed circuit of two dissimilar metals when their junctions are at different temperatures. This direct action of heat, producing an electric signal, is used in thermocouples.

A *tachometer* senses rotary motion of a shaft and produces a signal proportional to its speed.

Intrinsic safety is a term applied to electrical equipment and wiring that have been made incapable of releasing sufficient energy, even under abnormal conditions, to cause ignition of a hazardous atmospheric mixture.

"*Galvanometric*" indicates a type of instrument for detecting or measuring a small electric current by movements of a magnetic needle or of a coil in a magnetic field.

34C Series Resistance Pneumatic Transmitter

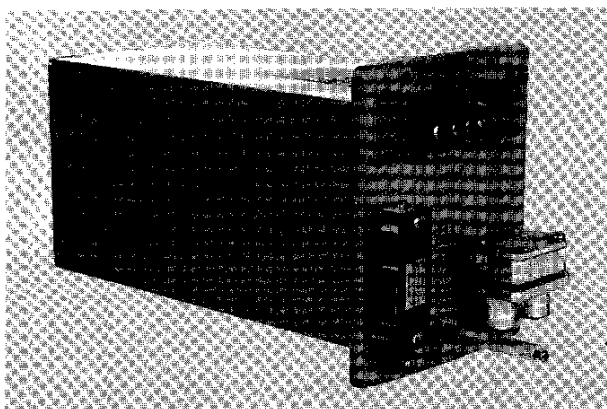


FIGURE 49 — 34C Resistance Pneumatic Transmitter

The 34C Transmitter receives a resistance input from a resistance temperature detector (RTD), measuring either temperature or temperature difference. Internal circuitry modifies the input so it is linear with temperature. The input signal is converted to a standard pneumatic output which can then be used as an input to a recorder or controller.

Intrinsically safe RTD input leads can be used in hazardous locations. Either a nickel or platinum RTD can be used to provide a wide range of temperature spans. The transmitter provides long-term, dependable, universal operation. The 34C is protected from radio frequency interference (RFI).

The RTD sensor is connected to an electronic circuit called a bridge. It amplifies the changing measurement signal and conditions it before the signal is applied to a galvanometric motor (position-type mechanism) like the one used in the 33C emf (thermocouple) Transmitter. (Refer to the product description for 33C for the principle of operation of the galvanometric motor.)

15 and 13 Series Pneumatic d/p Cell Transmitters

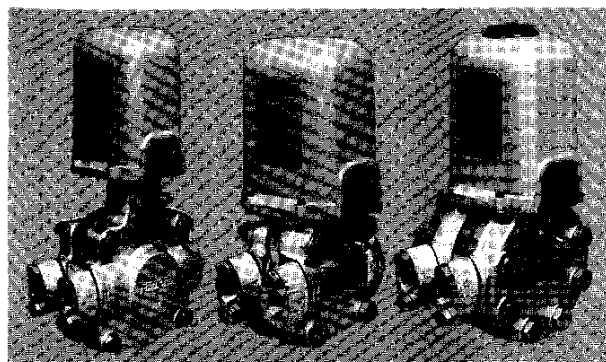


FIGURE 50 — 15A (left), 13A (center), and 13H (right) Pneumatic Differential Pressure Transmitters

The 15 and 13 Series Pneumatic d/p Cell Transmitters measure low to very low ranges of differential pressure, liquid level in pressurized vessels, or liquid level or density in closed or open tanks. The transmitters produce proportional pneumatic output signals that are transmitted to indicating, recording, or controlling devices. Span and zero can be elevated or suppressed.

The many variations of 15 and 13 Series Transmitters reflect sensors with different range capabilities and the way the sensor contacts the process. The 15A, 13A, and 13H, shown in Figure 50, measure very low, low, and medium ranges of differential pressure, respectively. These transmitters can be used in flow, liquid level, and density applications. Their sensors can be interchanged in the field if process requirements change.

The 15F and 13FE, shown in Figure 51, are *flange-mounted* — that is, attached to a tank with an integral rim. The sensing diaphragm contacts the process directly through a hole in the tank wall; if the wall is particularly thick, the sensor can be extended. This allows application to sticky, viscous, or otherwise difficult-to-measure

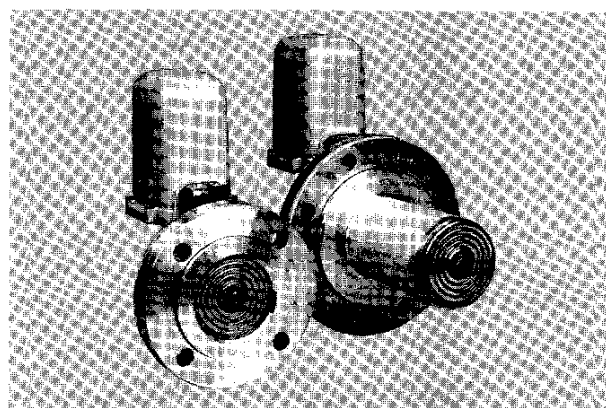


FIGURE 51 — Low-Range Liquid Level or Density Transmitters — 15F with Flush Diaphragm (left) and 13FE with Extended Diaphragm (right)

sure processes. The 15F, 13F (which looks very much like the 15F), and 13FE Transmitters measure very low, low, and medium ranges of pressure, respectively.

The 13DMP, shown in Figure 52, and 13DEMP (which looks similar to the 13DMP) have sensors on pressure seals which may or may not be flange-mounted. As with the 15F, 13F, and 13FE, they can be applied to viscous or sticky processes; however, the 13DMP and 13DEMP measure differential pressure (in low and medium ranges, respectively) and can be used at higher temperatures.

As shown in Figure 53, 15 and 13 Series Transmitters use a diaphragm capsule instead of a bellows element; it is connected to the standard force-balance mechanism described previously. The high process pressure is applied to one side of the diaphragm and the low process pressure is applied to the other, reference side. The capsule configuration shown is for the 15A; the diaphragm configuration changes slightly for the 13A, 13H, and 13DMP/DEMP Transmitters.

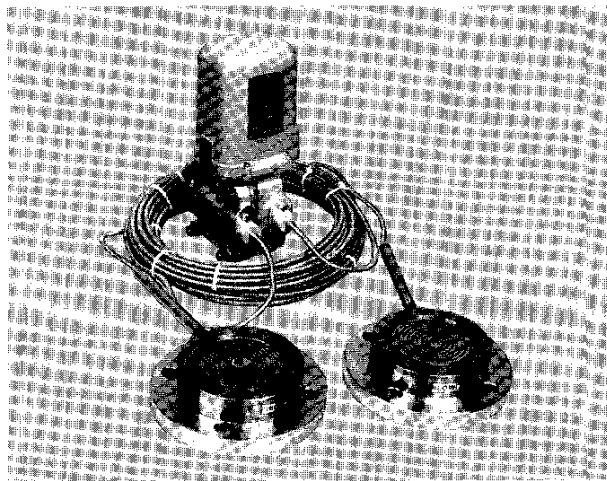


FIGURE 52 — Type 13DMP Pneumatic d/p Cell Transmitter with Matched Pressure Seals for Liquid Level Measurement

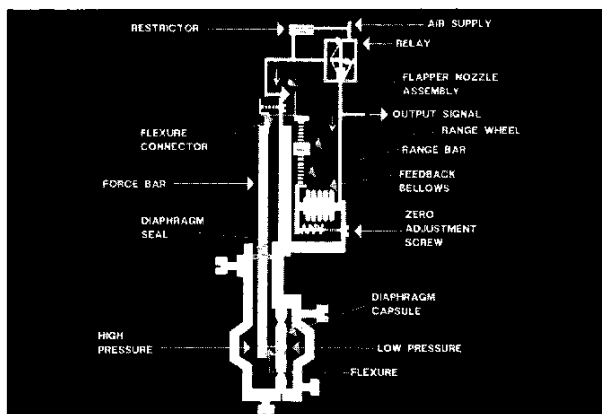


FIGURE 53 — Operating Mechanism for the 15A Transmitter (Operating Mechanisms for 13A and 13H Transmitters Are Similar)

11 Series Pneumatic Absolute, Differential, and Gauge Pressure Transmitters

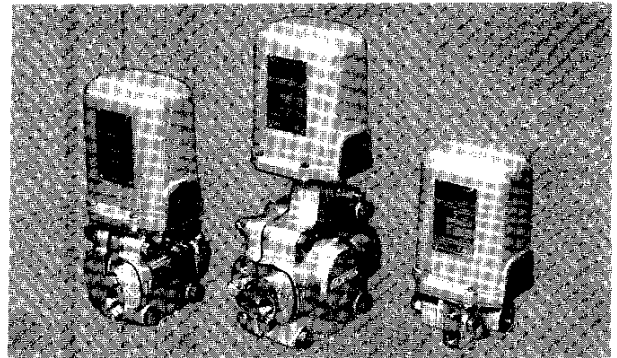


FIGURE 54 — 11AL (left), 11AM (center), and 11AH (right) Absolute Pressure Transmitters

The 11 Series Pressure Transmitters measure pressure and transmit a proportional standard pneumatic signal to a controller or recorder. They measure low to very high ranges of absolute pressure (11AL, 11AM, and 11AH, shown in Figure 54), gauge pressure (11GM and 11GH, shown in Figure 55), and differential pressure (11DM, shown in Figure 56).

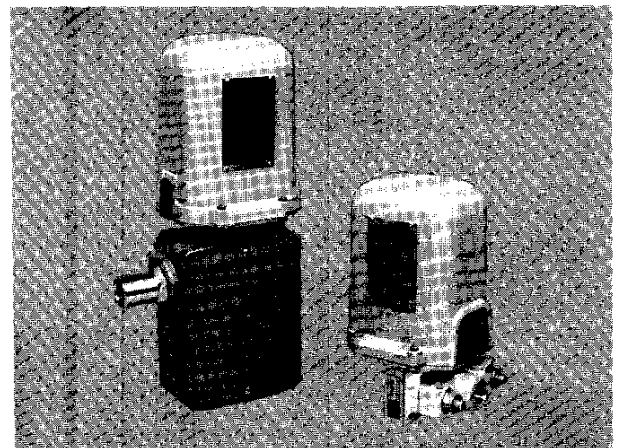


FIGURE 55 — 11GM (left) and 11GH Gauge Pressure Transmitters

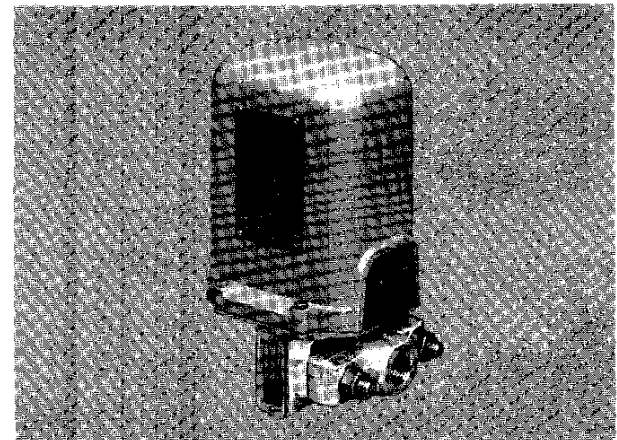


FIGURE 56 — 11DM Differential Pressure Transmitter

All 11 Series Transmitters have wide rangeability and allow unusual amounts of zero suppression and elevation (as much as 1000 percent). They have excellent over-range capabilities and are ruggedly constructed units designed to operate out-of-doors in normal industrial atmospheres.

The 11A Series Absolute Pressure Transmitters have particularly good accuracy, even in narrow suppressed-zero ranges. To provide even greater flexibility to process application than recalibration would allow, the 11A Transmitters have interchangeable sensors. The sensors measure low to medium-high ranges of pressure.

The 11G Series Gauge Pressure Transmitters also have particularly good accuracy, maintaining their precision of measurement even in highly suppressed or elevated ranges. Although not interchangeable, the standard sensors for the 11GM and 11GH Transmitters provide the widest range of pressure measurement — from medium-high to very high pressures.

The 11DM Differential Pressure Transmitter has the broadest zero suppression and elevation capabilities for application to a wide range of processes. It measures medium-high to high process pressures.

All 11 Series Transmitters use the reliable, standard force-balance mechanism discussed earlier in this publication. The 11AH, 11DM, and 11G Transmitters use the bellows sensor shown in Figure 43. 11AL and 11AM Transmitters use a diaphragm sensor with one side exposed to the pressure being measured and the other side evacuated, as shown in Figure 57.

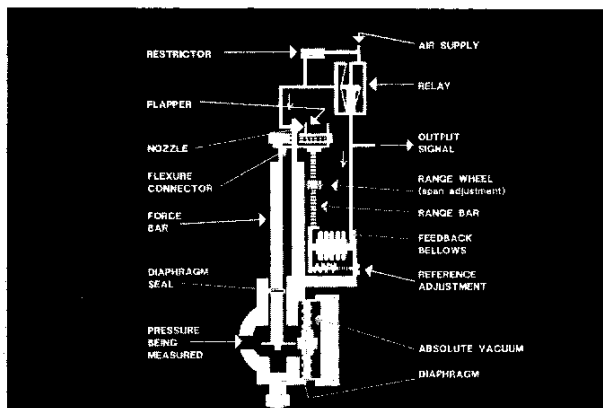


FIGURE 57 — Operational Mechanism for the 11AL and 11AM Transmitters

13T Series Pneumatic Target Flowmeter

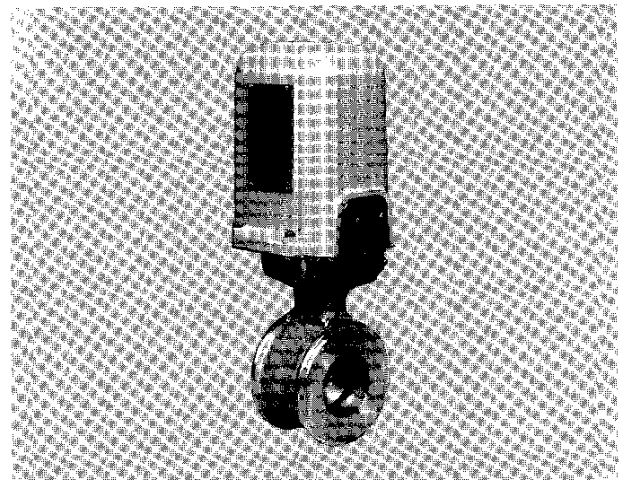


FIGURE 58 — 13T Pneumatic Target Flowmeter

The 13T Target Flowmeter measures liquid, vapor, gas, dirty fluids, light slurries, and high viscosity liquids. It converts force exerted on a target in the flow stream into a standard pneumatic signal that is proportional to the square root of the flow rate. A solid disk suspended in the pipe (the target) connects directly to the force bar of the force-balance mechanism.

Steam and water flow measurements in outdoor installations are common applications for target flowmeters. The complications of condensate pots, heat tracing, seal fluids, and antifreeze compounds to prevent freezing in cold weather are avoided with target flowmeters. (The transmitter body has a short length that helps to eliminate steam jacketing requirements in high melting point liquid applications.) Some high viscosity and high melting point liquids that have been successfully measured by target flowmeters include aromatic tar, asphalt, molten sulphur, polyethylene-water slurry, catalytic cracker bottoms, styrene rubber solution, coal tar, crude oil, corn syrup, molasses, wheat starch, and tallow.

Because the target is integral to the transmitter, no separate primary device (as needed with the orifice meter, described in Section 5) is required. This reduces installation costs and simplifies servicing. Target flowmeters also have wider application capability than orifice meters.

19C Series Pneumatic Consistency Transmitter

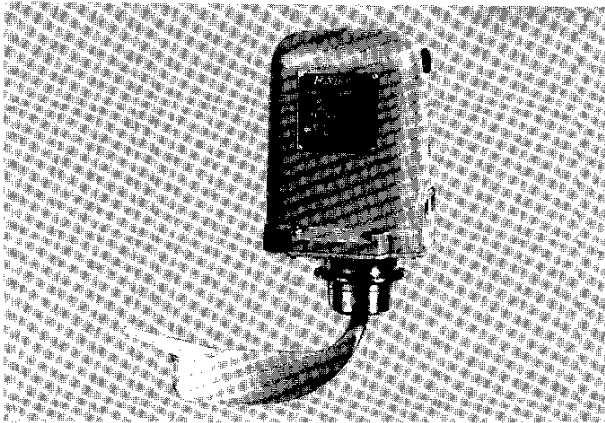


FIGURE 59 — 19C Pneumatic Consistency Transmitter

The 19C Pneumatic Consistency Transmitter converts the forces acting on its blade to a standard pneumatic output signal which is related to the consistency of stock flowing past the blade. It is extremely sensitive to very small variations in consistency of the flowing fluid. This instrument is designed for use on pulp and paper stock whose actual consistency is determined by laboratory testing*. The 19C Transmitter measures changes related to consistency, rather than consistency itself.

Paper stock and similar slurries carrying suspended particles pose special measurement problems. They are *non-Newtonian* fluids — fluids whose viscosities are not constant and therefore their flow pattern (velocity profile*) is not predictable. However, research has shown that there is a relationship between force acting on the blade (caused by the consistency of the fluid and essentially independent of velocity) and consistency.

* More information on determination of consistency can be found in Foxboro technical documentation on this product. The *velocity profile* of a fluid depends on the viscosity of the fluid, the texture of the pipe walls, and the flow speed (velocity). If there were no viscosity and the pipe walls were perfectly smooth, the fluid would travel at the same speed at all points in a cross section of the pipe. Since these are ideal and unattainable conditions, any fluid travels at least a little bit slower at the pipe walls (due to friction) than in the middle of the pipe; this causes a flow pattern to be created that is effectively a parabola which is called the velocity profile (see Figure 60). Flow is further characterized as being *laminar* (in which the generally viscous, slow-moving fluid is in layers that do not mix) or *turbulent* (in which the generally less viscous, faster moving fluid moves in an agitated, complex, and random way). The velocity profile of a *Newtonian fluid* (a homogeneous fluid with no or only fine, spherical suspended particles) is the basic reference condition for all flowmeters. NOTE: all gases and most liquids are Newtonian fluids.

The pivoting movement of the scimitar blade is transmitted through a pivot point to the force bar of the force-balance mechanism. Besides the pivot at the diaphragm seal and other minor modifications (such as the size of the feedback bellows), the 19C Transmitter uses the same force-balance mechanism described in Section 5.

There are some limitations on the size of the pipeline, depending on the type of fluid. The output signal is noisy and may require modification of the transmission line or the addition of other equipment. However, this is an invaluable piece of equipment for very specific applications. Refer to technical documentation on this instrument for more details.

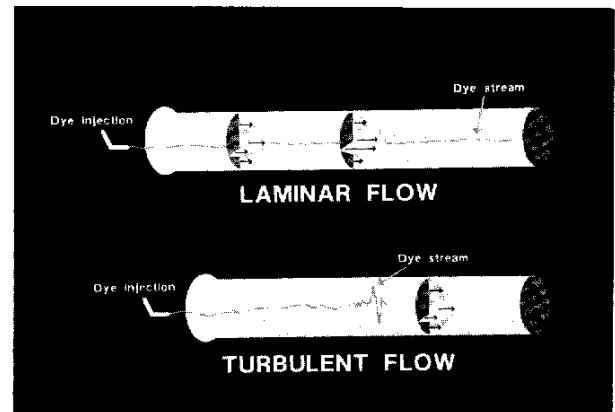


FIGURE 60 — Velocity Profiles for Laminar and Turbulent Flow

17B Series Pneumatic Buoyancy Transmitters

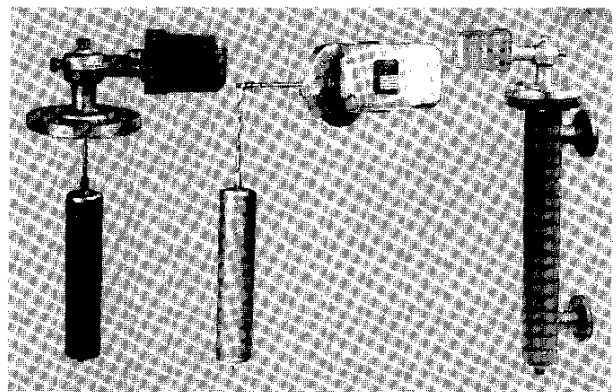


FIGURE 61 — 17BT (left), 17BS (center), and 17BC (right) Pneumatic Buoyancy Transmitters

Buoyancy transmitters measure liquid level, liquid density, or interface level in a closed tank, using the buoyant force on a displacer as the measurement. The measured value is then transmitted as a proportional standard pneumatic signal. Any of the three 17B Pneumatic Buoyancy Transmitters can be used for any

application. They are chosen for the most advantageous mounting position (see Figures 62a and 62b), rather than for differences in specifications.

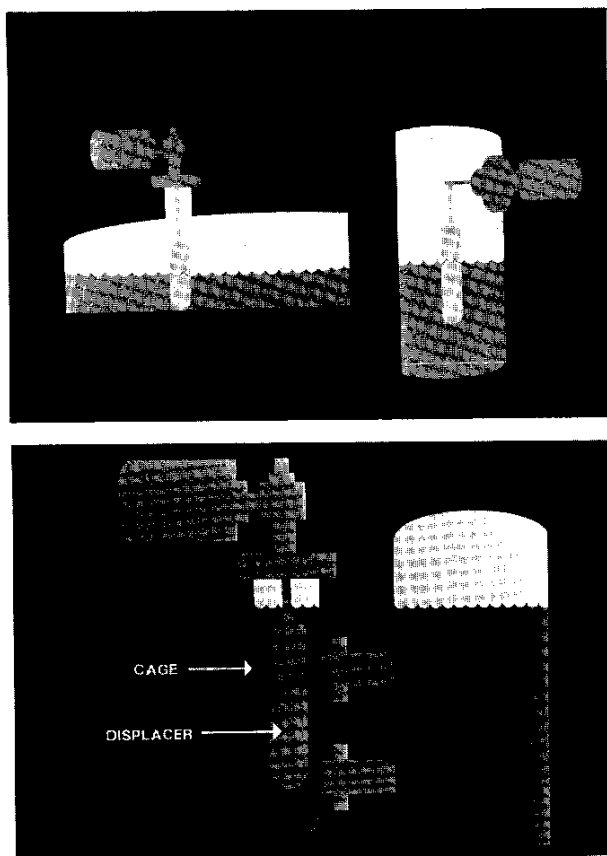


FIGURE 62 — Mounting for the 17BT (Figure 62a, top, left), 17BS (Figure 62a, top, right), and 17BC (Figure 62b, bottom) Pneumatic Buoyancy Transmitters

The 17B Transmitters are usually applied to glass-lined vessels, vessels in which a lower connection is not possible, applications with changing pressures or levels, or high temperature applications. The vessel may be vented, pressurized, or evacuated.

Tank configuration and space available determine choice of the three versions of the 17B Transmitters. The 17BT is top-mounted, the 17BS is side-mounted, and the 17BC is cage-mounted (to isolate the displacer from any turbulence present in the vessel which would otherwise cause the displacer to swing and produce erratic measurements or possibly cause physical damage to the displacer, transmitter, or vessel).

In addition to these mounting possibilities, a wide selection of materials for process-wetted parts and a selection of displacer lengths and diameters is provided so that the 17B Series Pneumatic Buoyancy Transmitters can be used in almost any application.

In liquid level applications (see Figure 63, left), the displacer is partially above the liquid and partially in the liquid, no matter what the level; as the level rises, the displacer is buoyed slightly upward. In liquid density applications (Figure 63, center), the displacer is completely submerged in the liquid; as the surrounding liquid becomes more dense, the displacer is buoyed slightly upward. In interface level applications (the measurement of the levels of two immiscible liquids — Figure 63, right), the displacer is completely submerged beneath the top liquid and extends into the bottom liquid; as the level of the heavier (bottom) liquid rises, the displacer is buoyed slightly upward.

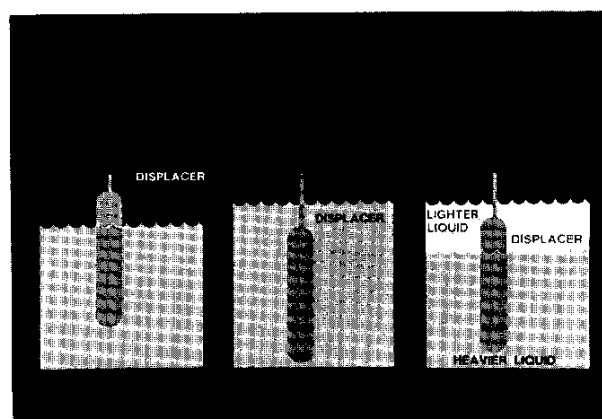


FIGURE 63 — Installation of Displacers in Liquid Level (left), Liquid Density (center), and Interface Level (right) Applications

The 17B Transmitters use the standard force-balance mechanism described earlier, modified slightly at the process connection as shown in Figure 64. (Although this shows side mounting and liquid level measurement, it applies to top and cage mounting, and any appropriate application, as well.) The standard force bar ends at the diaphragm seal, which keeps out process fluid or vapor. A continuation of the force bar, called the lower force bar, extends horizontally into the vessel; from it,

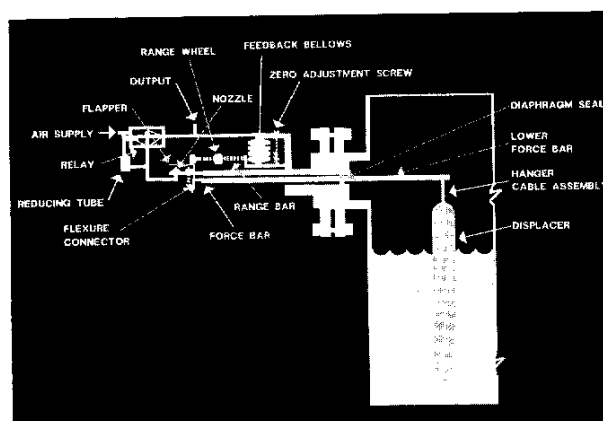


FIGURE 64 — Force-Balance Mechanism for 17B Pneumatic Buoyancy Transmitters (compare to Figure 43)

the displacer is hung from a connecting hanger cable assembly that absorbs torsional stress in the displacer. The buoyant force on the displacer is applied to the end of the lower force bar and transmitted through the fulcrum of the diaphragm seal to the range bar and the flapper-nozzle assembly.

16A Series Pneumatic Speed Transmitter

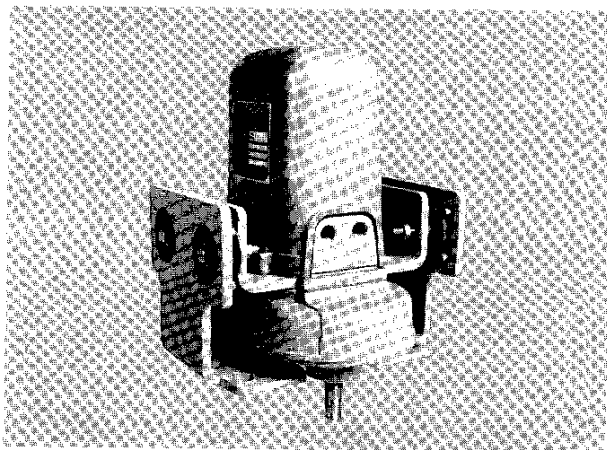


FIGURE 65 — 16A Pneumatic Speed Transmitter

The lightweight, compact, 16A Pneumatic Speed Transmitter can be mounted on machines, compressors, engines, or drives to measure rotational speed and transmit a linear, proportional, standard pneumatic signal. (Optional external speed changing mechanisms, such as gears and sprockets, extend its capability to practically any speed input range with the same output.) Typically, the 16A is used with a controller for smooth regulation of desired speed or with a recorder.

Normal performance — providing unusually high, sustained accuracy and 100 percent overrange protection — is achieved during vibrations up to 5 g (five times the force of gravity). Such applications include mounting the 16A on hoists, fans, pumps, blowers, dry feeder belts, centrifuges, belt drives, coating machines, rollers, paper machines, mixers, and screw conveyors. An optional vibration isolation mount allows normal performance during vibrations between 5 and 50 g. Typical high vibration applications are internal combustion engines, natural gas compressor engines, centrifugal pumps, agitators, and compressors.

This instrument is inherently nonhazardous, as operation is entirely pneumatic and has an air purge. It is ideal for corrosive, explosive, and dusty atmospheres.

The 16A Pneumatic Speed Transmitter uses a variation of the standard force-balance mechanism described in

Section 5. (See Figure 66.) The rotating process unit turns a shaft connected to a series of gears. These connections reduce the process speed proportionally, turning a magnet beneath a nonmagnetic disk. Just as in a speedometer in a car, the magnetomotive pull on the disk tends to turn the disk in the same direction, producing a force proportional to the speed of rotation. The force moves a flapper relative to a stationary nozzle, producing a restoring force through the relay output acting on a feedback unit.

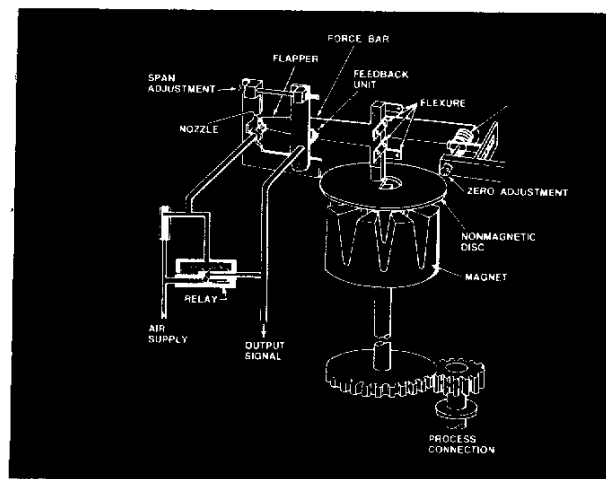


FIGURE 66 — Force-Balance Mechanism for the 16A Speed Transmitter

13R, 13RE, 17R Series Repeaters

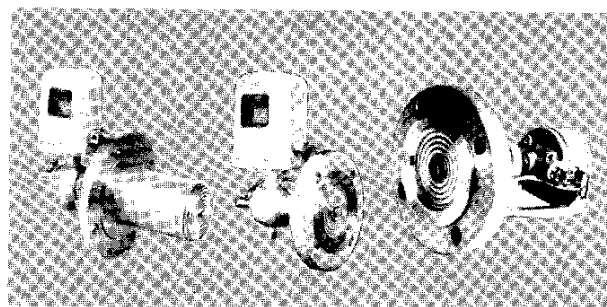


FIGURE 67 — 13RE Repeater (left) Has Extended Diaphragm; 13R (middle) and 17R (right) Are Flush-Mounted

Repeaters measure pressure with a process-contacted diaphragm (like a flange-mounted transmitter or pressure seal) and transmit the measured pressure through a pipe "leg" to another instrument. Their most common application is as a replacement for the wet leg or dry leg process connection to a differential pressure transmitter at the bottom of a closed tank filled with a process fluid that cannot or should not be conveyed through the piping. Repeaters are ideal for corrosive, viscous, or thermoplastic liquids or slurries, and there is a sanitary version for food and drug processes. Repeaters can also

be used independently for level or density measurement in an open tank.

The repeater is mounted above the maximum level of the liquid with the differential pressure liquid level transmitter mounted near the bottom of the tank. (Figure 34 in Section 5 shows how repeaters and differential pressure "legs" are installed.)

Process pressure is applied to one side of a large diaphragm. Supply pressure against the other side of the diaphragm creates a feedback system which creates the output. (The supply pressure must exceed the process pressure being repeated by 10 psi/68.9 kPa.)

Typical repeater operation is shown in Figure 68 (the force-balance mechanism of the 17R). The primary components of the mechanism are a flanged sensing head (to detect the variable pressure) and a pilot valve (to deliver an air pressure to just balance the force on the sensing head).

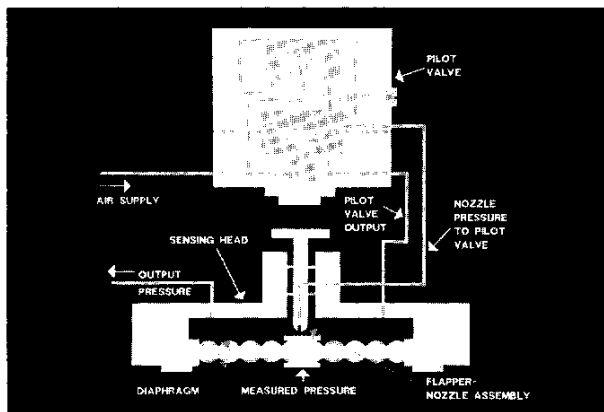


FIGURE 68 — Force-Balance Mechanism for the 17R Repeater

The sensing head consists of a diaphragm capsule and variation of the standard flapper-nozzle assembly. As the measured pressure (the pressure to be repeated) changes, the flapper-nozzle assembly registers variations in the clearance between the flapper-nozzle and the diaphragm. This causes a back pressure to be conveyed to one side of the small internal diaphragm in the pilot valve; plant air applied to the other side of the small diaphragm is changed and affects the pilot valve output. This is fed back to the measuring diaphragm both for transmission (repeating) and for balancing.

* Most electronic transmission systems are based on *current* (the measurement of energy flow between points of unequal voltage as measured in milliamperes; the most common range for process control is expressed as 4 to 20 mA dc). However, some systems use *voltage*

Signal Converters

Many processes require both pneumatic and electronic* instrumentation. Signal converters allow electronic signals (such as those from analytical instruments) to be incorporated into a pneumatic control system, or allow pneumatic signals (such as those used in hazardous atmospheres of a plant) to be incorporated into an electronic system. Signal converters therefore allow great flexibility in control system design. (Signals can also be converted from analog form to the digital form used by electronic computers and back again, but this is beyond the scope of this publication. Contact your Foxboro sales representative or refer to the Bibliography for more information.)

Signal converters can be mounted in or near the control room or in the field. Almost all valves (the most common final control element, discussed in Section 10 of this publication) are pneumatic, and frequently they require signal converters; these can be mounted in the field. (A positioner is a kind of field-mounted signal converter — it converts whatever signal it receives, either pneumatic or electronic, to a position. See Section 10 of this publication for more information.) Remote pneumatic loops sending a signal to a centralized electronic control room generally use field-mounted signal converters (though the converters may require protective housing). However, when a control room has been changed from pneumatic to electronic and satisfactory pneumatic transmission lines exist, the signal converters are usually in the control room.

Fortunately, the most common pneumatic and electrical signals are in the same ratio, and this simplifies the conversion from one system to another. The pneumatic 3 to 15 psi and 20 to 100 kPa and electrical 4 to 20 mA dc signal ranges all have "live" zeroes — that is, a signal representing an input of zero (3 psi, 20 kPa, or 4 mA) that is different from no input at all (0 psi, 0 kPa, or 0 mA) — and a full-scale value that is five times the zero value.

The applications, benefits, and principles of operation for individual pneumatic-to-electronic and electronic-to-pneumatic signal converters are discussed in the following pages. (Note that instruments with integral electronic-to-pneumatic conversion — such as transmitters receiving inputs from RTDs or thermocouples — have already been discussed. Refer to the 34C and 33C, respectively.) A "shorthand" is sometimes used in this section, representing pneumatic-to-electronic conversion as P/I and electronic-to-pneumatic conversion as I/P.

(the measurement of the concentration of electrons at a point in a circuit, the difference in concentrations at various points causing the energy to flow from one point to another; the most common voltage range for process control is expressed as 0 to 10 V dc).

Pneumatic-to-Electric Converters

E92 Series Pneumatic-to-Current or -Voltage Converter

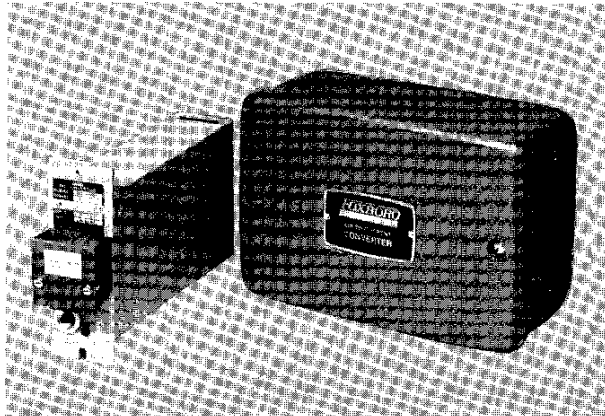


FIGURE 69 — E92R (Rack-Mounted, left) and E92F (Field-Mounted, right) Pneumatic-to-Current or -Voltage Converters

The E92 Pneumatic-to-Current or -Voltage Converter receives a standard pneumatic transmission signal and transmits a proportional standard current or voltage dc signal. The E92 is ideally suited to interface pneumatic transmitters with electronic recorders and controllers. The field-mounted version has a durable, weatherproof enclosure; the rack-mounted version is compact for high-density installation.

Air pressure is sensed at a bellows and this moves a spring-adjusted lever much like the force bar in the force-balance mechanism. However, instead of moving a flap-per in relation to a nozzle, the movement of the E92 lever initiates an electric signal when it moves in and out of a magnetic coil. The signal is then amplified, conditioned, and compared to a reference value before being outputted as a current or voltage value. Figure 70 is a

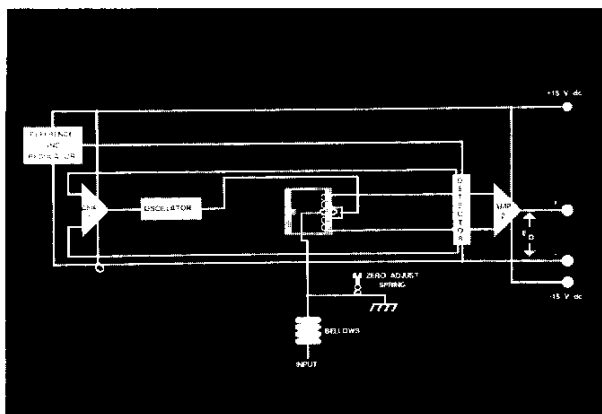


FIGURE 70 — Basic Signal Conversion Path for E92 Pneumatic-to-Current or -Voltage Converter

drawing of the basic signal conversion path, both pneumatic and electrical. (A full explanation of the electrical circuit is beyond the scope of this publication, but can be found in support literature for this Foxboro product.)

7010A Dual Pneumatic-to-Current Converter

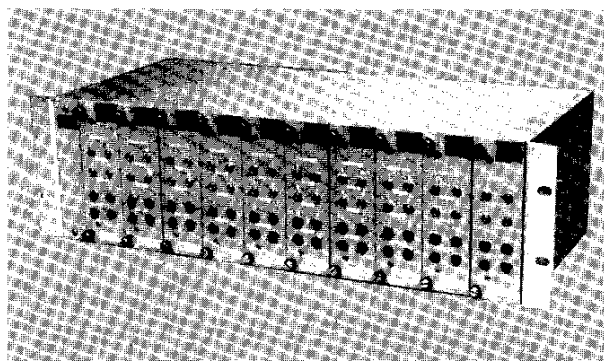


FIGURE 71 — Ten 7010A Dual Pneumatic-to-Current Converters in High-Density Rack for Panel Mounting

The 7010A P/I Converter receives standard pneumatic signals from one or two separate sources (generally field-mounted sensors and transmitters) and converts each to a proportional standard 4 to 20 mA dc signal. Because it is a rack-mounted instrument, the output generally goes to an electronic controller.

With its capacity to convert two field signals at once and its compact size, the 7010A P/I Converter allows high-density installation, saving a considerable amount of space. The "nest," or special rack shelf provided, automatically makes all necessary air and electrical connections for much simplified installation and later maintenance.

Electric-to-Pneumatic Converters

E69 Series Current-to-Pneumatic Signal Converters

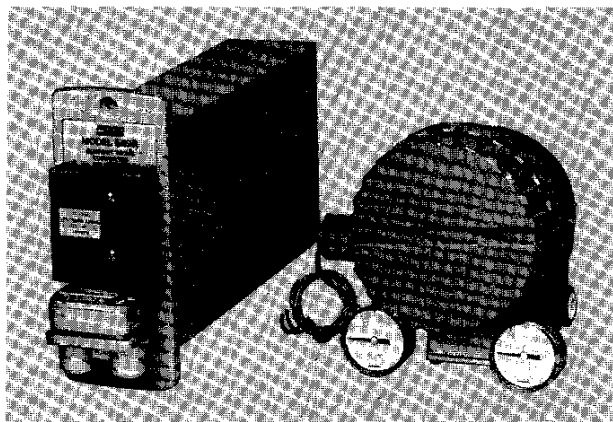


FIGURE 72 — E69R (Rack-Mounted, left) and E69F (Field-Mounted, right) Current-to-Pneumatic Converters

The E69 I/P Converters accept a standard current input signal (4 to 20 mA dc or 10 to 50 mA dc) and convert it to any of several pneumatic output signal ranges (standard 3 to 15 psi and 20 to 100 kPa plus others — see product literature for details). The output signal can be used as the input to various pneumatic instruments. (Another E69 connects directly to final actuators and is discussed in Section 10 of this publication.)

The field-mounted E69F is designed to minimize the effects of vibration and shock normally encountered in process applications. It is also unaffected by normal atmospheric dust and corrosion. The rack-mounted E69R has a compact, vibration-resistant design with easily accessible adjustments.

The E69 I/P Converters use the same position-balance galvanometric motor as the 33C emf Pneumatic Transmitter discussed earlier in this section. The current input signal to the motor is converted to a position and then, through a feedback bellows and flexure, produces a pneumatic signal for use with an external device. (See Figure 73.)

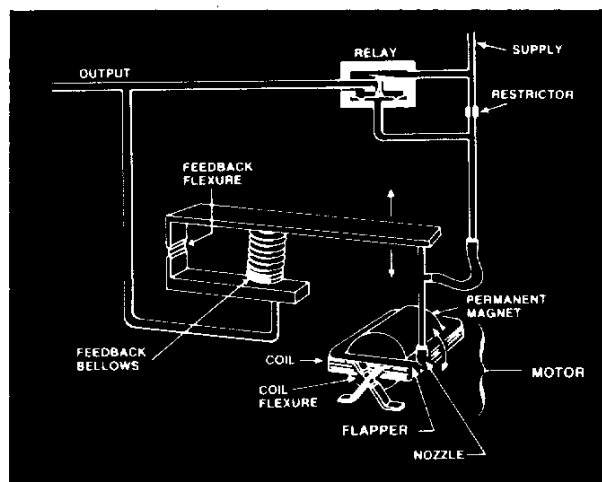


FIGURE 73 — Galvanometric Motor and Force-Balance Mechanism for the E69 I/P Converter

SECTION 7 — RECORDING AND INDICATING HARDWARE

Recorders and indicators receive a measurement value (a standard pneumatic signal) and convert it to pen or pointer position. They provide the operator at all times with a visual display of controller input variables or other important process data. A history of selected variables' values can also be kept.

The most important function of recorders and indicators is to allow the operator to assume manual control of the process when required. They can also be used as a double check on automatic controls or for testing and adjustment purposes.

An indicator can identify the process measurement value fairly accurately on a scale through the use of a pointer, colored "ribbon," or other means — even at a distance. A recorder tracks the process measurement value continuously through a pen's tracings on a chart, which are read most accurately at close range. Some instruments provide the best of both and have an indicator scale as well as a recording chart, and some controllers have recorders built into them.

The chart or scale specifies the process measurement range and units. The standard signal may have the same units but will almost always be a different range, and the two should not be confused. For example, in a recorder with a 3 to 15 psi input signal and a chart range of 0 to 100 psi, a process measurement value of 0 psi will produce a 3 psi signal; a mid-range process measurement value of 50 psi will produce a 9 psi signal; and a full-range process measurement value of 100 psi will produce a 15 psi signal.

Recorders usually track up to four variables on one chart through the use of separate pens, each with a different color ink. Each recorder has a chart drive, moving the chart under the pens at a specific speed. Although all variables combined in one recorder must use the same time scale, a control room (or other place in the plant) can have different recorders with different time scales.

Recorder charts come in many variations, depending on which of two primary types they are (round or strip), time scale, and units. The appropriate type for any recorder is chosen on the basis of how fast the chart is traveling, how much historical chart should be visible at one time and how much of it should be easily accessible, readability desired, the space available, any possible governmental requirements, and the frequency of maintenance desired. Generally, some trade-off of these is required.

Time scale has to do with how fast the chart travels, and it can vary from 3/4 inches per minute to 3/4 inches per hour for strip charts and from full rotation per 24 hours to full rotation per 7 days for circular charts. Generally, the faster the process changes, the faster the chart should move to be able to record those changes. A circular chart, which takes up more control panel space than a strip chart, can display more data than a strip chart, but the circular chart holds less historical information. (The recorded part of a *strip chart* is stored in the recorder case, so that it can be removed and reviewed as necessary.) Records can have uniform or nonuniform increments of units, as shown in Figure 28; this is chosen based on the area of the chart in which most data should occur to enhance readability. Choice of recorder and chart can be complicated.

Round Chart Recorders

Model 40P Pressure Recorder

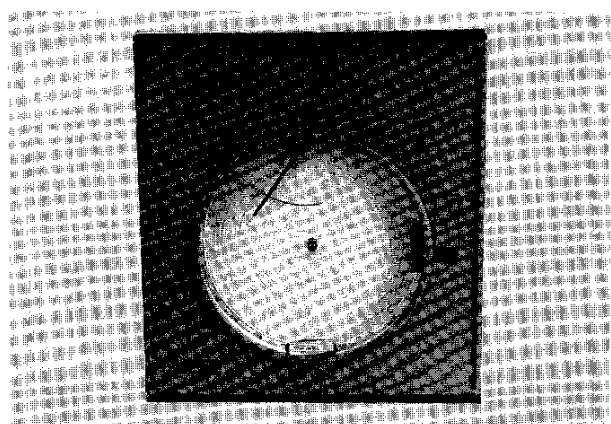


FIGURE 74 — Model 40P Pressure Recorder

The Model 40P Recorder directly measures up to four similar process variables with integral elements, records the value(s) on a circular chart, and transmits proportional standard pneumatic signal(s) representing the value(s) to a controller. (NOTE: there is a Model 40P Indicating Controller available; see Section 9.)

An extensive selection of charts provides virtually all ranges, mathematical relationships, and time scales desired. (Refer to the Foxboro chart and dial catalog for details.) Pneumatic, mechanical, and electrical chart drives in several speeds can be selected. Elevated and suppressed zeroes are available.

A variety of flow, pressure, and temperature elements is available. Flow measurements are made with the Type 37 Diaphragm Element in a selection of materials; it is mounted on the back of the Model 40P case. Pressure measurements can be made with a diaphragm, bellows, spiral, double spiral, or helical element in a variety of materials for either gauge or absolute pressure ranges; these are mounted inside the Model 40P case. Temperature measurements are made with filled thermal systems (most classes are available). All of these elements, their operation, and their benefits are discussed in Section 5.

For a drawing of the Model 40P components with circular chart superimposed, see Figure 75.

A linkage establishes the difference between the direct process measurement and the set point. The error positions a proportioning lever which pivots to change the flapper-nozzle relationship. (The error also positions the measurement pen, which registers the change on the circular chart.) The output pressure created by the flapper-nozzle and the relay is fed to the control mechanism (proportional, integral, and/or derivative), which rebalances the flapper-nozzle assembly.

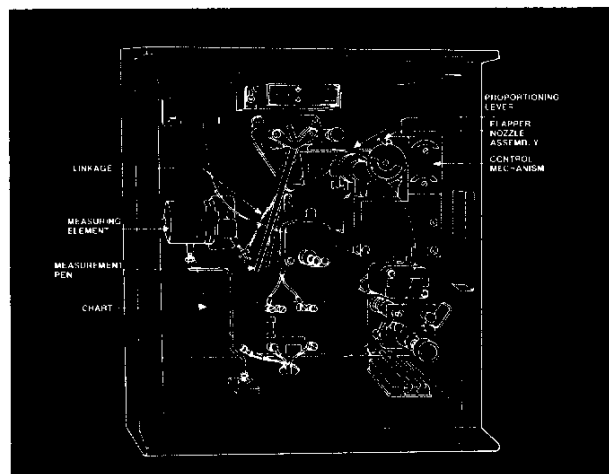


FIGURE 75 — Parts of the Model 40P Recorder

Strip Chart Recorders

Model 120 Pneumatic CONSOTROL Recorders

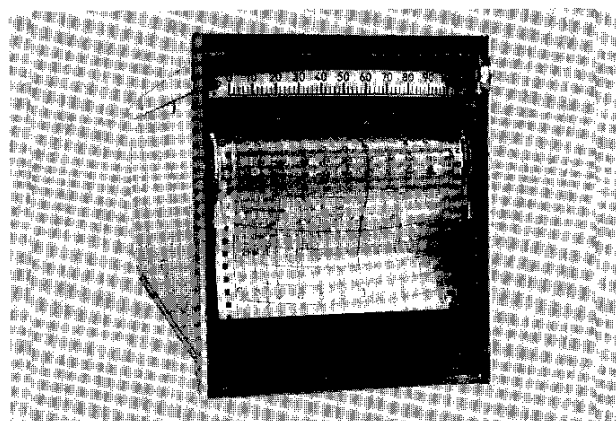


FIGURE 76 — Model 120 Pneumatic Recorder

The Model 120 Pneumatic CONSOTROL Recorder records as many as four process measurements received as standard pneumatic transmission signals from remote transmitters. Large, brightly colored pens move beneath a scale, clearly indicating the measurement. The Model 120 Recorder is part of a complete Foxboro "family" of panel-mounted instruments called the CONSOTROL 100 Line. (The Model 110 Indicator in this section and the Model 130 series of controllers discussed in Section 9 are part of this "family.")

Two types of strip charts are available for the Model 120 Recorder, both 4 in/102 mm wide. SCAN-FOLD charts provide an easily accessible record that accordion-folds into the recorder case; roll charts provide a longer record.

The Model 120 Pneumatic CONSOTROL Recorder uses a variation on the standard position mechanism described in Section 5. The transmission signal is applied to a bellows (see Figure 77); movement of the bellows causes the linkage to pivot and ultimately move the pen point across the chart. One mechanism is provided for each measurement.

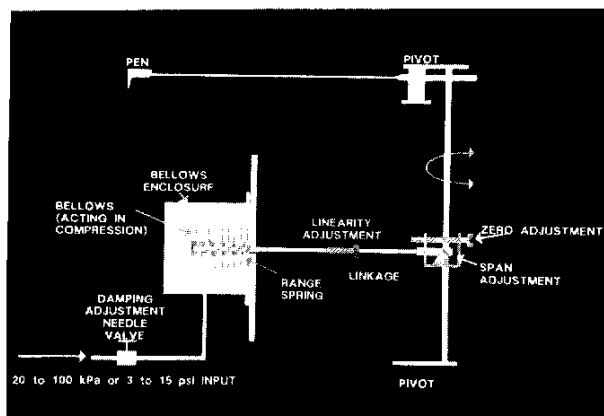


FIGURE 77 — Position Mechanism for the Model 120 Recorder

Model 127S Pneumatic Indicating Recorders

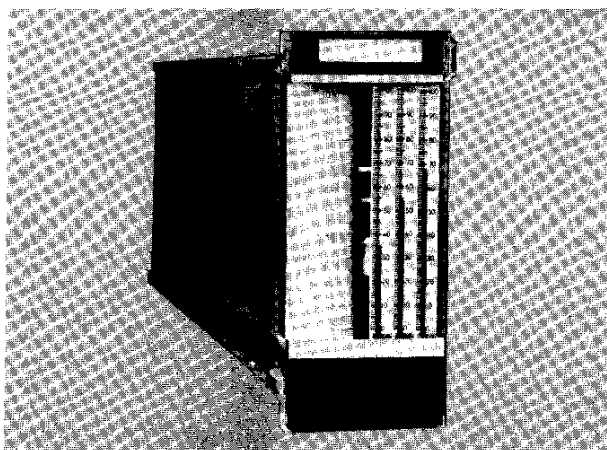


FIGURE 78 — Model 127S Pneumatic Indicating Recorder

The Model 127S Pneumatic Indicating Recorders receive one, two, or three standard pneumatic input signals from remote transmitters, continuously record the values on a chart, and simultaneously display the values on color-keyed vertical ribbon indicators. The ribbons make highly visible displays, and each can have a different scale.

At standard speed, the roll chart provides a total of 30 days of chart record, with two hours of record visible from the front of the recorder and ten additional hours of record visible when the recorder is partially withdrawn

from its panel location. The chart drive, which can be pneumatic or electric, is designed to allow easy access to the record. Several chart drive speeds are available.

The Model 127S Recorder is designed for simplified maintenance with, among other features, color-keyed connecting tubing.

The process measurement input signal is applied to a bellows that is part of a variation of the force-balance mechanism described in Section 5 and seen in Figure 79. Called the force-balance detector, it yields an output that is an error signal (the difference between the input signal and the current pen position) rather than a measurement signal. The signal is applied to the pneumatic drive assembly, which moves the piston and causes the drum to turn. The drum is coupled to the pen assembly through the drive cable.

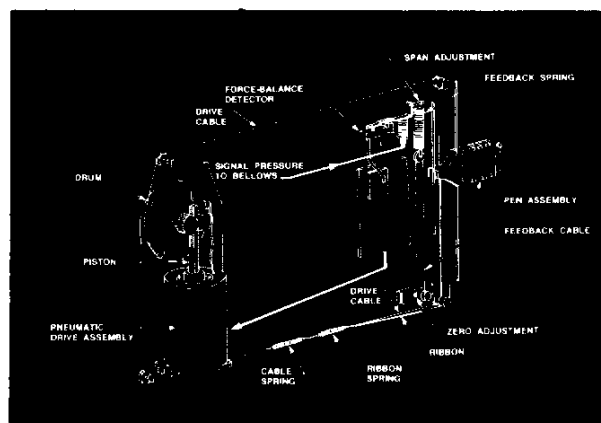


FIGURE 79 — Operating Mechanism of the Model 127S Recorder

Meanwhile, movement of the force-balance detector changes the position of the feedback spring. The feedback cable, connected to the feedback spring through a set of pulleys, provides a small countering force to that of the drive cable at the pen assembly to increase the overall accuracy of the instrument. Pen position puts tension on the connected cable and ribbon springs, which, in turn, moves the ribbon indicator.

Indicators

Model 110 Pneumatic CONSOTROL Indicators

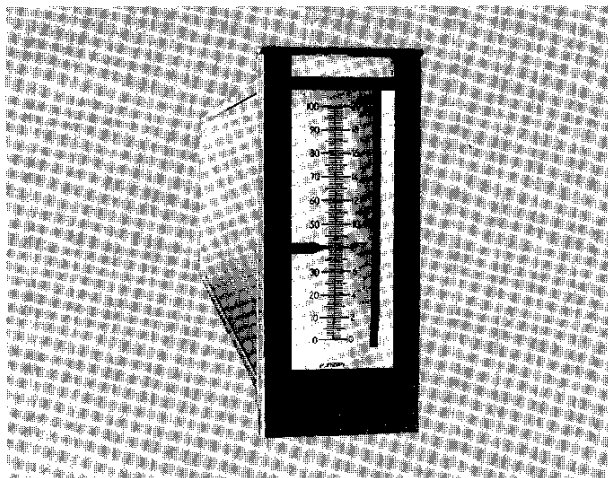


FIGURE 80 — Model 110 Pneumatic CONSOTROL Indicator

The Model 110 Pneumatic CONSOTROL Indicator receives proportional pneumatic signals from one, two, or three process measurements and continuously indicates the values. Brightly colored pointers provide an easily readable display against large black numbers on a white background.

The Model 110 Indicator uses a mechanism much like the position-type mechanism seen in Section 5, with added span, zero, and pointer angularity adjustments.

Model 40P Pressure Indicator

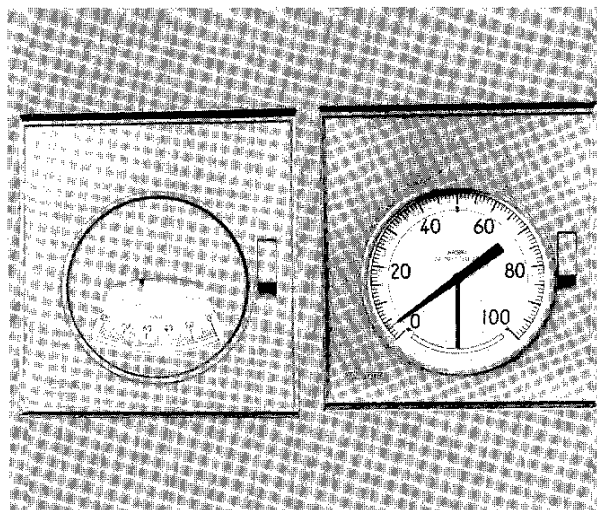


FIGURE 81 — Model 40P Indicators with Sector Scale for One or Two Variables (left) and Concentric Scale for One Variable (right)

The Model 40P Indicator directly measures one or two similar process variables with integral elements and displays the value(s) on a concentric scale or a sector scale. (NOTE: there is a Model 40P Indicating Controller available; see Section 9.)

An extensive selection of dials provides virtually all ranges desired. (Refer to the Foxboro chart and dial catalog for details.) Elevated and suppressed zeroes are available.

A variety of flow, pressure, and temperature elements is available. Flow measurements are made with the Type 37 Diaphragm Element which is available in a selection of materials; it is mounted on the back of the Model 40P case. Pressure measurements can be made with a diaphragm, bellows, spiral, double spiral, or helical element in a variety of materials for either gauge or absolute pressure ranges; these are mounted inside the Model 40P case. Temperature measurements are made with filled thermal systems (most classes are available). All of these elements, their operation, and their benefits are discussed in Section 5.

The operating mechanism is much the same as the one used in the Model 40P Recorder shown in Figure 75.

SECTION 8 — PNEUMATIC COMPUTERS

"Computer" is a general term for a piece of equipment that accepts one or more standard input signals, modifies and/or combines the signals in some predetermined mathematical way, and transmits the resultant output to a receiver.

The pneumatic computers discussed in this section are small, dedicated (i.e., single-purpose) units. Each performs only one function. Some computers (like these) are analog, others are digital. Digital computers, both small- and large-scale ones that can accept many inputs and perform many functions, are beyond the scope of this publication. Refer to the Bibliography for further reading suggestions.

The pneumatic computers described here accept a standard pneumatic signal and transmit a standard pneumatic signal to a controller or totalizer. They can be mounted in the field or in a control room. They are economical and ideal for a relatively small or uncomplicated system.

Integrators are a form of computer that sums the flow of one variable in time and provides a readout of the current sum. Integrators frequently incorporate a square-root extractor since the most common input signal is from a differential pressure measuring device, which produces a signal that is the square value of flow rate. The summing process is the mathematical function of integration.

Multiplier/Divider

Model 556 Pneumatic Analog Computer

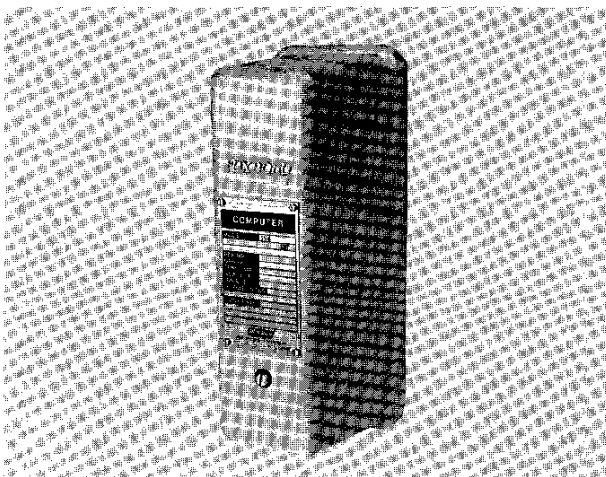


FIGURE 82 — Model 556 Pneumatic Analog Computer

The Model 556 Pneumatic Analog Computer accepts one or two standard pneumatic input signals, modifies it or them in accordance with a mathematical function (multiplication, division, squaring, or square-root extraction), and provides a proportional standard pneumatic output signal. Each unit can perform all four functions and can be changed in the field when control schemes are altered. Applications include mass flow computing, stock blending, flow proportioning, and others used in feedforward and feedback control systems.

Multiplication and division functions can incorporate zero elevation; they also have scaling flexibility for applications such as pressure and temperature compensation for gas flows, ratio computations, and Btu computations.

Scaling factors are applied to the Model 556 Computer to provide a means of expanding or reducing the output for a given set of input conditions. Span and zero factors are also available for scaling elevated ranges.

The Model 556 Computer uses a variation of the force-balance mechanism described in Section 5. Figure 83 shows how this mechanism is modified for the multiplication function, which has two inputs. (The mechanism for square-root extraction is essentially the same as for the Model 557, shown in Figure 85; diagrams of the mechanisms for the other functions, which are slightly different, can be found in the technical documentation on the Model 556.)

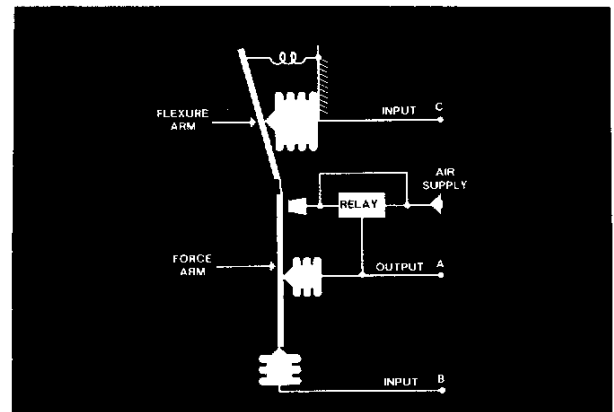


FIGURE 83 — Force-Balance Type Mechanism for the Multiplication Function ($A = B \times C$) of the Model 556 Pneumatic Computer

Square-Root Extractor

Model 557 Pneumatic Square-Root Extractor

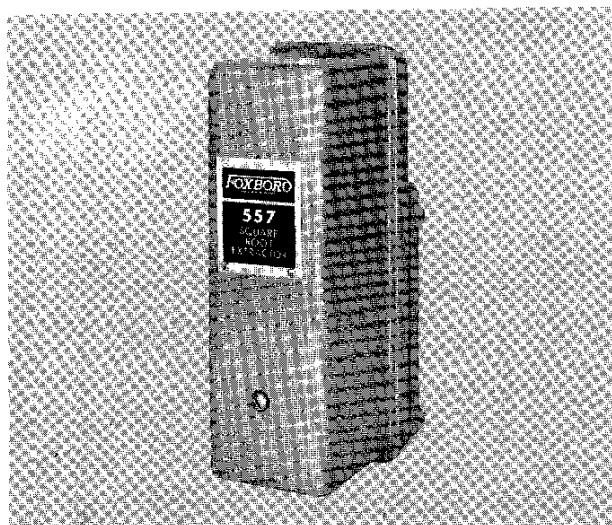


FIGURE 84 — Model 557 Pneumatic Square-Root Extractor

The Model 557 is a pneumatic analog computer like the Model 556, except that it is limited to one mathematical function. It receives a standard pneumatic input signal and provides a standard pneumatic output signal proportional to the square root of the input signal. A typical application for this is converting the pressure drop across an orifice, measured by a differential pressure transmitter, to a linear flow-rate signal.

Because the inherently high gain of any square-root extractor can cause some rather large errors at very low input signals, the Model 557 Extractor is calibrated so that the output signal will drop to zero output (0 psi or 0 kPa) whenever the input signal decreases to one percent of span. In cases where zero output cannot be tolerated (as when the outputs of two square-root extractors are the inputs to another instrument that requires two valid inputs), an optional high-signal selector effectively bypasses the square-root extractor under low flow conditions, producing an output that is approximately 3 psi or 20 kPa.

The Model 557 Pneumatic Square-Root Extractor uses a variation of the force-balance mechanism described in Section 5 and shown in Figure 85. (It is essentially the same mechanism used with the Model 556 multiplier/divider function.)

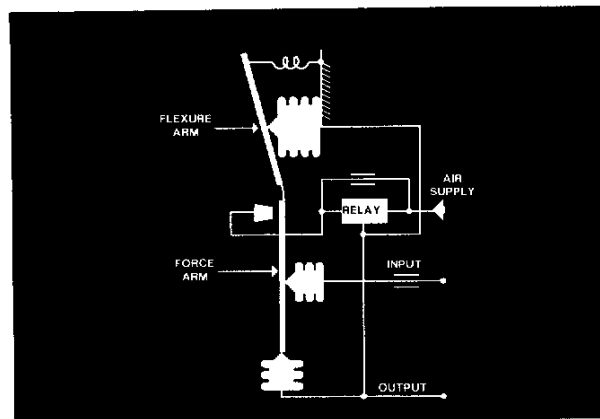


FIGURE 85 — Force-Balance Type Mechanism for the Model 557 Pneumatic Square-Root Extractor

Sum or Differential

Model 136 Pneumatic Summing Unit and Dynamic Compensator

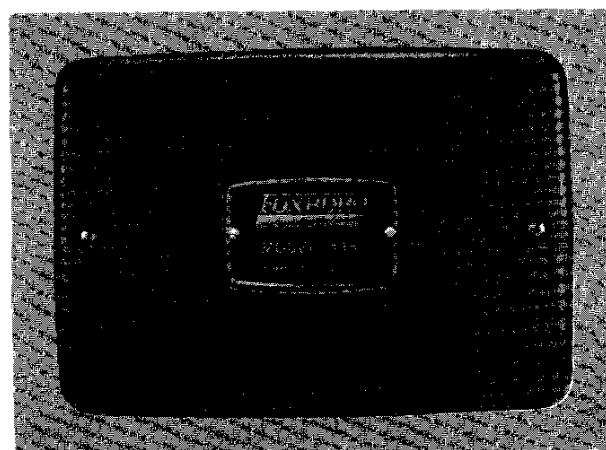


FIGURE 86 — Model 136 Pneumatic Summing Unit and Dynamic Compensator

Model 136 pneumatic analog computers receive one, two, or three standard pneumatic inputs, modify it or them according to a mathematical function selected, and produce a standard pneumatic output signal. The Dynamic Compensators* are most commonly used to modify the process response in a feedforward* loop (lead*, lag*, or impulse* functions) for improved control. Pneumatic Summing Units are used to condition one signal by applying a fixed bias*, either positive or negative, and to amplify, reduce, or reverse the signal; they are also used for adding, subtracting, and averaging.

* See footnotes on next page.

The following functions are available for the Model 136 Computer.

Universal	$P = B + g(A - C)$
Average	$P = \frac{B + gA}{1 + g}$
Differential	$P = K_1 + g(A - C)$
Add	$P = B + g(A - K_1)$
Subtract	$P = B - g(C - K_1)$
Amplify or Reduce	$P = g(A - K_1) + K_2$
Reversal	$P = g(K_1 - C) + K_2$
Impulse	$P = A + \frac{B}{g} \left(\frac{Ts}{1 + Ts} \right)$
Lead/Lag	$P = \frac{1 + Ts}{1 + \frac{Ts}{g}}$

where: P = output and feedback signal
 g = gain
 T = lead time
 A, B, C = input signals
 K_1, K_2 = constants
 s = Laplace operator*

The operating mechanism for the Model 136 function summing three inputs [$P = B + g(A - C)$], shown in Figure 87, is a force-balance mechanism. A circular disk, which acts as the flapper of the flapper-nozzle assembly, rests on four bellows. Process inputs are applied to the opposite side of three of the bellows; their expansion or contraction tilts the disk with respect to the fulcrum, and this affects the air pressure at the nozzle. As the disk tilts toward the nozzle, nozzle pressure rises, is amplified by the relay, and fed back to the output bellows which rebalances the disk.

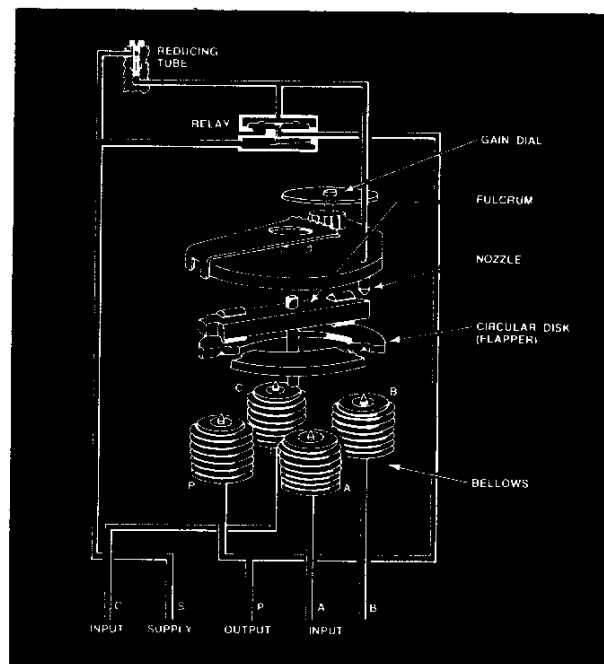


FIGURE 87 — Force-Balance Mechanism for the Model 136 Pneumatic Summing Unit

* *Feedforward control* is a control mode in which information concerning one or more conditions that can disturb the controlled variable is converted, outside of any feedback loop, into corrective action to minimize deviations of the controlled variable.

Lead action causes the output of a device to advance ahead of its input signal, exhibiting a phase lead and gain which increase with increasing frequency of the input signal.

Lag action causes the output of a device to fall behind its input signal, exhibiting a phase lag which increases and a gain which decreases with increasing frequency.

An *impulse* function is a form of lead action having zero steady-state gain.

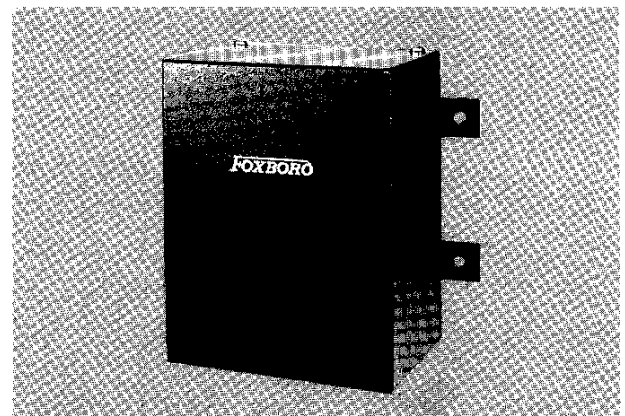
A *dynamic compensator* is a device consisting of lead, lag, impulse, and/or dead-time functions, provided to balance control action against the dynamic response of the process being controlled.

Bias is a signal or a constant added to or subtracted from an input signal to provide elevation or suppression.

A *Laplace operator* is an algebraic substitute for the differential operator (d/dt) or time derivative in a differential equation.

Model 14A Pneumatic Flow Integrators receive a standard pneumatic signal (usually from a differential pressure transmitter), extract the square root, and continuously provide a readout of the total flow to that time (*totalize* the flow). For example, if the flow is being read in gallons, a count will be made on the readout after each gallon of flow has been measured, showing **current total flow**; average flow can be determined by dividing the readout by the time (minutes, hours, or days since the counter was reset). If the flow is large, the integrator can be set to register one count after each 10 or 100 units (e.g., gallons) has passed through the meter.

Model 224E Receiver-Integrator



The Model 224E Pneumatic Receiver-Integrator is basically a signal converter with signal conditioning: it receives a standard pneumatic signal from a transmitter and produces a time-duration pulse output to a motor-driven counter. The output is either a linear or square-root function of the input (other relationships are optional). The Model 224E Integrator can be combined with timers to provide rate control, batch control, batch sampling, and other functions.

The operating mechanism (Figure 91) consists of a bellows, linkages, constant-speed motor-driven cam, and electronic switching circuit. The input is applied to the bellows which adjusts the positioning arm and linkages (including the cam lever). The electric circuit consists of a neon lamp striking a photoconductive cell, reducing electrical resistance, and creating a voltage. This is interrupted when cam rotation moves the cam follower and linkages so that the vane intercepts the beam of the neon lamp for a period of time proportional to the value of the measurement; this increases the circuit resistance and closes the switch for a comparable period.

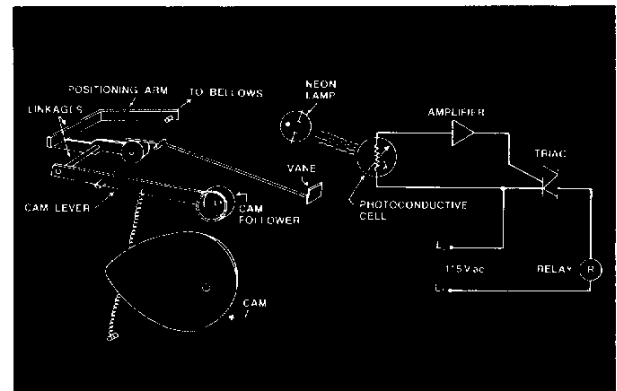


FIGURE 91 — Operating Mechanism of the Model 224E Receiver-Integrator

SECTION 9 — CONTROLLERS AND ALARMS

A *controller* is a device that receives measurement information from sensors and a set point from an operator (or another controller or a computer), determines the desired response, and sends an appropriate signal to keep the measurement at a constant acceptable value. It is the "brain" of the control system, where the control strategy resides, and is usually mounted in the control room.

Any plant or mill may have a combination of pneumatic, electronic, and digital controllers, all of which provide specific benefits. Pneumatic controllers are particularly appropriate in situations where environmental conditions make electronic controllers not acceptable.

Controller selection involves many things and must be made only after an analysis of the process. Although the subject is too complicated to be described in full here, some of the topics include safety conditions (which determine controller action, discussed following), process dead time and capacity (which determine the control mode, discussed in Section 1 and summarized in Table E following), and things like precision of control required and plant economics, which further narrow the choice.

The *controller action* — whether an increasing signal to the controller produces an increasing or decreasing output signal, called *direct* or *reverse action*, respectively — depends on the position the valve or other final actuator must assume should the plant air supply (or power, with electronic systems) fail and on the direction in which the controlled variable moves when the valve opens. For example, if for safety reasons the valve shuts completely upon loss of power, the valve is called *air-to-open* (or, sometimes, *fail-closed*) and if the controlled variable increases when the valve opens, the controller has *increase-decrease* action. Then a rising measurement will cause the valve to close.

Most automatic controllers allow transfer to manual control if desired or required. This means that there are actually two control units in one instrument, requiring the presence of an automatic balancing unit so that the transfer can be smooth ("bumpless"). The auto/manual unit has a transfer switch which is usually located at the bottom of the instrument face.

Alarms are often installed in controllers to provide an audible and visual signal when a variable has reached some critical level.

Table E — Selection of Controller Modes

CHARACTERISTIC	MODE				
	On/Off	P-only	PI	PD	PID
Transfer Lag	Minimum	Small	Any	Small	Any
Dead Time	Minimum	Small	Any	Small	Any
Capacitance	High	Moderate	Any	Moderate	Any
Reaction Rate	Slow	Slow	Any	Slow	Any
Load Changes	Any	Small	Any	Small	Any
Cost	Lowest	Low	Moderate	Moderate	Highest
Noise Level	None	Low	Any	Low	Low
Batch or Continuous Operation	Either	Either	Continuous*	Batch	Continuous*

* Requires windup protection for batch operation

Controllers

Model 43AP Pneumatic Indicating Controllers

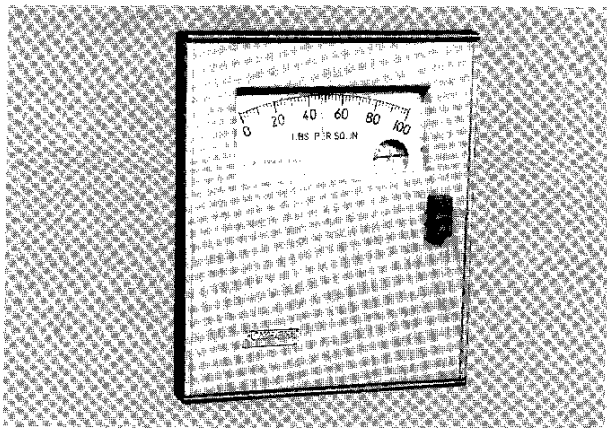


FIGURE 92 — Model 43AP Pneumatic Indicating Controller

The Model 43AP Controllers measure a process variable, indicate its value, act on it in accordance with the control mode chosen, and transmit a standard pneumatic signal to a final control element. The Model 43AP Controller may be mounted in a panel, on a wall or pipe, or directly on the valve. Measurement of pressure, temperature, vacuum, or differential pressure can be done with an integral element. Measurement and set point are clearly visible from a distance; output is also indicated, though on a smaller gauge. With an extremely wide variety of measuring elements, ranges, and options available, the Model 43AP can be applied to virtually any process.

Since it uses a direct-measuring element, the process value indication is not disturbed by power failures. Optional automatic/manual transfer also assures uninterrupted operation.

Control modes available are on/off, proportional-only, PD, PI, PID, differential gap, and automatic shutdown. The Model 43AP can also be applied to batch processes with an optional function.

The Model 43AP Controllers use a combination of mechanisms. A position mechanism indicates the directly measured variable and, by balancing the measurement against the set point, produces an error signal for the control mechanism. A control relay similar to the one in Figure 36 indicates the output and provides feedback to a flapper-nozzle assembly. A proportioning lever with center pivot conveys the error signal applied at one end of the lever to a striker attached to the flapper-nozzle assembly at the other end of the lever (all units except automatic shutdown). Various combinations of bellows, restrictors, and springs provide the various control modes (all units except on/off).

Figure 93 shows a diagram of the most complex combination — that of the PID controller with auto/manual

transfer. All the basic control mode mechanisms are variations of this. Because their operation is different, the automatic shutdown mechanism is shown in Figure 94 and the differential gap mechanism is shown in Figure 95.

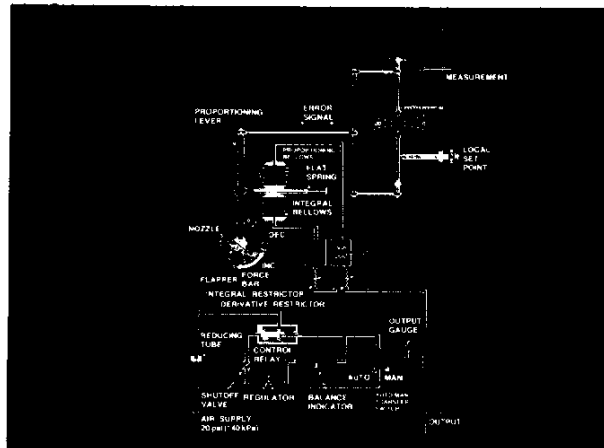


FIGURE 93 — Operating Mechanism for Model 43AP Auto/Manual Transfer Unit (PID control)

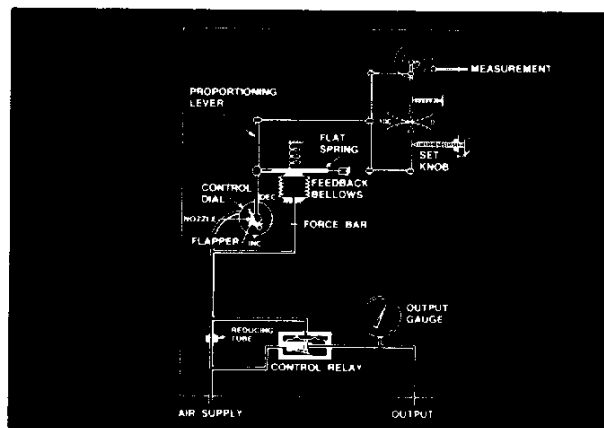


FIGURE 94 — Operating Mechanism for Model 43AP Unit with Differential Gap

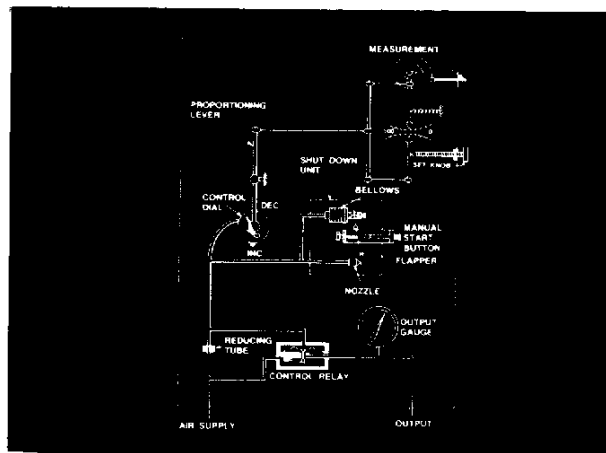


FIGURE 95 — Operating Mechanism for Model 43AP Automatic Shutdown Unit (proportional-only control)

Model 130 Pneumatic CONSOTROL Controllers

The CONSOTROL Control System is a "family" of pneumatic controllers, recorders, indicators, and auxiliary devices. They are characterized by bold displays for easy reading at a distance; by separate automatic and manual functions for continued operation even during servicing; by precision force-balance control operating mechanisms for high accuracy and reliability; by simple, low-cost panel mounting; by flexibility of application; and by simplified servicing due to easily accessible adjustments and connections.

CONSOTROL Model 1275 Recorders, Model 110 Indicators, Model 120 Recorders, and Model 136 Computers have already been described. The Model 130 Controllers — the heart of the system — are described following, as are auxiliary devices Model 135 Manual Loading, Switching and Bias Stations, Model 138 Remote-Mounted Controllers, and the alarm and flasher systems for all CONSOTROL instruments.

For the Model 130 Indicating Controllers:

All units allow virtually bumpless transfer between automatic and manual control. Set points generally can be adjusted manually (locally) or by a pneumatic signal from a remote location; some units allow either type of adjustment, and one unit offers computer setting of the set point.

All inputs and outputs are standard pneumatic transmission signals. Pneumatic connections are made through a special device which seals off the input signal when the station is removed from operation. The stations can be pulled partially out of their panel mounting for access to adjustments, records, etc., without disrupting operation.

Options generally offered include "batch" modification, alarms, flashers, and special preparation for specific industry applications.

Model 130 Indicating Controllers are comprised of seven basic units (shown in Figure 98). The local set-point unit (A) is the basic position mechanism shown in Figure 37, with the pointer registering the set point on Indicator F. (The remote set-point unit has the same basic mechanism but more components.) The function of the derivative unit (B) is the same as that discussed in Section 1 of this publication. The automatic control unit (C) is a specialized form of force-balance mechanism and is shown in Figure 99. The arrangement of the bellows beneath the force-balance floating disk changes with the control action. The manual control unit (D) is a variation on the position mechanism shown in Figure 37, except that a manual thumbwheel drives the action of the mechanism

rather than a sensor. (The pointer registers the output on Indicator G.) A transfer switch on the face of the Model 130 Controllers allows the operator to select the driving action to be used: either the manual control unit via the thumbwheel or the automatic balancing unit (E) via a series of bellows and relays. (See Figure 100.)

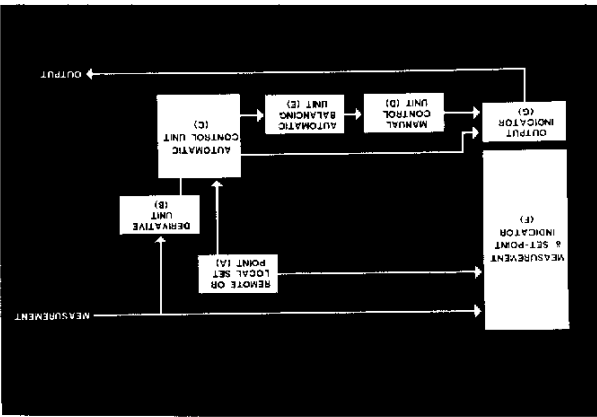


FIGURE 98 — The Seven Basic Components of a Model 130 Controller

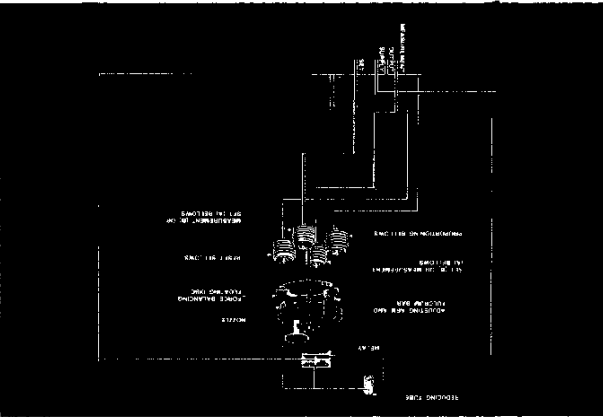


FIGURE 99 — Force-Balance Automatic Control Unit of the Model 130 Controller

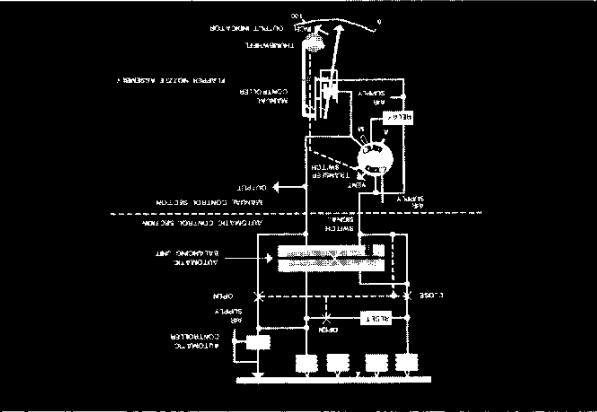
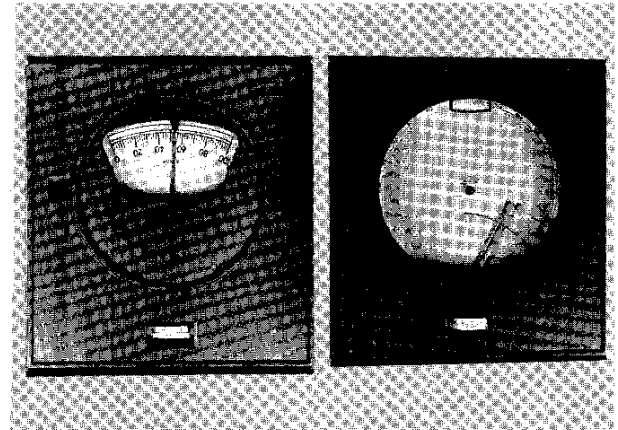


FIGURE 100 — Manual Control Unit and Automatic Balancing Units of the Model 130 Controllers

FIGURE 96 — Model 40P Recording Controller for One or Two Variables (left) and Model 40P Indicating Controller for One Variable (right)



Model 40P Pneumatic Recording or Indicating Controller

The center pivot of the proportioning lever in the differential gap unit (Figure 95) is positioned by a positive feedback loop which is immediately responsive to the flapper-nozzle back pressure. This causes the output to "lock up" at the supply pressure when the measurement exceeds the predetermined neutral zone or gap. When the measurement decreases below the gap, the output "locks up" at zero pressure. The feedback loop incorporates just one bellows opposing the flat spring.

The automatic shutdown unit is essentially an on/off controller with a shutdown unit and manual start button added. The center pivots of both proportioning levers are secured to the case (Figure 94). However, the on/off unit has its flapper-nozzle assembly at the control dial. The automatic shutdown unit has the flapper of its assembly connected to a bellows that in its normally expanded condition covers the nozzle to keep the output at the supply pressure; when the output falls below the set point, the bellows contracts and opens the nozzle to make the output pressure zero. The flapper remains in its open position until the manual start button is pushed against it.

The center pivot of the proportioning lever in Figure 93 is positioned by two feedback bellows responding to the output signal. The PID control feedback loop incorporates opposing proportioning and integral bellows on either side of a flat spring, fed by integral and derivative restrictors. PI control feedback incorporates the proportioning bellows and the derivative restrictor. On/off control has no feedback loop; the proportioning lever is secured to the case.

The operating mechanisms for Model 40P Controllers are much like those described for Model 43AP Controllers. As shown in Figure 97, motion of the direct-measuring element is conveyed through linkages to both the recording pen and the proportioning lever. The lever, whose pivot point is established by a feedback loop, operates on a flapper-nozzle assembly. A deviation changes the flapper-nozzle relationship, creating both the output pressure and the feedback force. (The feedback loop is comprised of a flat spring between a pair of bellows fed by restrictors.)

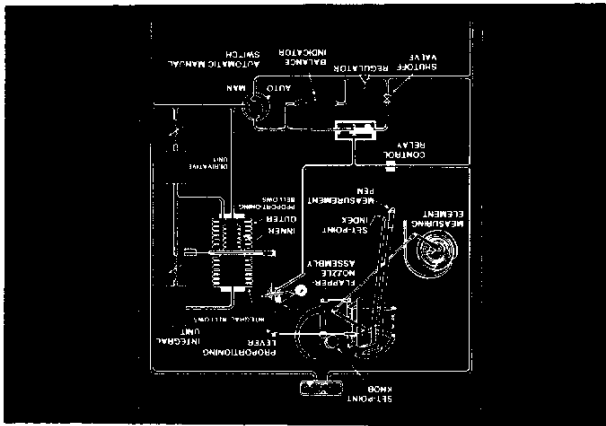
The Model 40P Controller directly measures one or two process variables with integral elements. It records or indicates the value(s), acts upon the signal in accordance with the chosen control mode, and transmits standard pneumatic signal(s) to a final control element. A wide selection of process measurement ranges can be accommodated. Elevated and suppressed zeroes are available.

All of the basic control modes discussed in Part I are available — on/off, proportional-only, PI, PD, and PID — plus a narrow band variation of proportional action. The standard Model 40P Controller records or indicates and controls one variable; optionally, it can record and control one variable which activates two control mechanisms (called a *duplex controller*) or it can record two independent variables each with a control mechanism (called a *dual controller*).

A variety of flow, pressure, and temperature elements is available. Flow measurements are made with the Type 37 Diaphragm Element in a selection of materials; it is mounted on the back of the Model 40P case. Pressure measurements can be made with a diaphragm, bellows, spiral, double spiral, or helical element in a variety of materials for either gauge or absolute pressure ranges; these are mounted inside the Model 40P case. Temperature measurements are made with filled thermal systems (most classes are available). All of these elements, their operation, and their benefits are discussed in Section 5.

The Model 40P Controller directly measures one or two process variables with integral elements. It records or indicates the value(s), acts upon the signal in accordance with the chosen control mode, and transmits standard pneumatic signal(s) to a final control element. A wide selection of process measurement ranges can be accommodated. Elevated and suppressed zeroes are available.

FIGURE 97 — Operating Mechanism for the Model 40P Recording Controller with Auto/Manual Switch (PID control mode)



Model 130M, 130P, 130F Pneumatic CONSOTROL Indicating Controllers

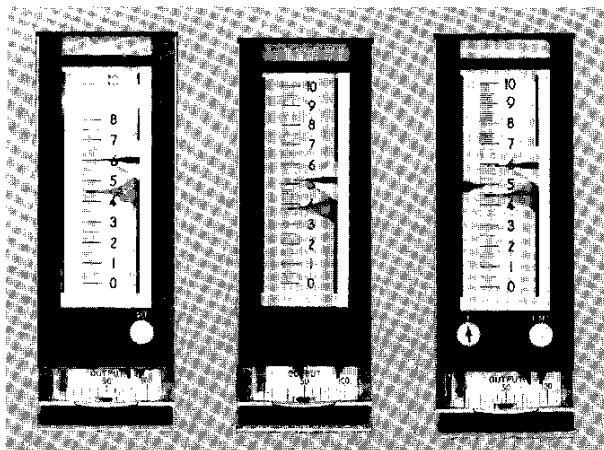


FIGURE 101 — Model 130M/P/F Controllers Have Basic Control Modes

Model 130M, 130P, and 130F Controllers receive a standard pneumatic signal, perform basic modes of control (on/off, proportional-only, PD, PI, and PID), and transmit a standard pneumatic signal to a final control element. As an option, a record can be made on a chart inside the controller case.

Set points can be adjusted manually (Model 130M), pneumatically (Model 130P), or both ways (Model 130F).

Model 130K Pneumatic CONSOTROL Computer-Set Indicating Controllers

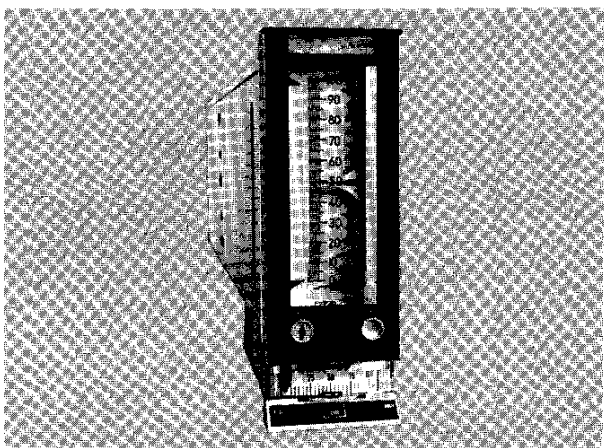


FIGURE 102 — Model 130K Computer-Set Controller

The Model 130K Pneumatic CONSOTROL Computer-Set Indicating Controller receives a standard pneumatic input signal, acts upon it by one of the basic control modes, and produces a standard pneumatic output sig-

nal. It also receives a digital input (pulse or contact closure) from a computer to establish its set point at a value that will optimize the performance of the process. In this way, the Model 130K Controller provides supervisory set-point analog control of a process by an electronic digital computer monitoring the whole process.

Since feedback is so important to determination of the set-point value, three types of feedback loops are offered as options: pneumatic feedback of the set-point signal (accomplished with a pneumatic-to-voltage converter), voltage feedback proportional to the set-point signal, or voltage feedback proportional to the process measurement (which acts as transmitter). Control of the set-point value can be switched from the computer to the local operator.

A converter receives a pulsed signal representing the optimized set point which the converter transforms, via a digital electronic circuit, into a signal to a motor which mechanically positions the set-point pointer. (For details, refer to product literature on this product.) The pointer position is transmitted to the pneumatic force-balance mechanism to use in creating an error signal for control action.

Model 130B Pneumatic CONSOTROL Indicating Batch Controllers

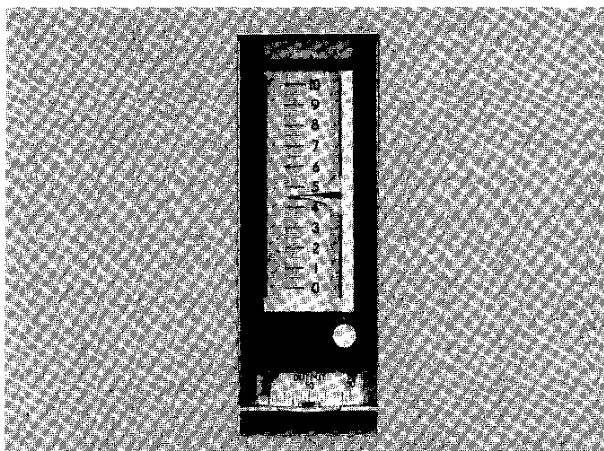


FIGURE 103 — Model 130B Batch Controller

The Model 130B Batch Controller is essentially a Model 130M Controller with a batch option built in. It is used for processes which may be off control (or shut down) for long periods. (As noted in Section 1, a controller receiving a zero signal for a long period — as would happen if a standard Model 130M Controller were applied to this type of process — will saturate or “wind up” and then severely overshoot the set point when restarted.) The batch unit unloads the integral feedback bellows so that the controller has no integral action until the deviation is smaller than the proportional band.

Model 130MA/FA and 130MS/FS Pneumatic CONSOTROL Auto-Selector Control System

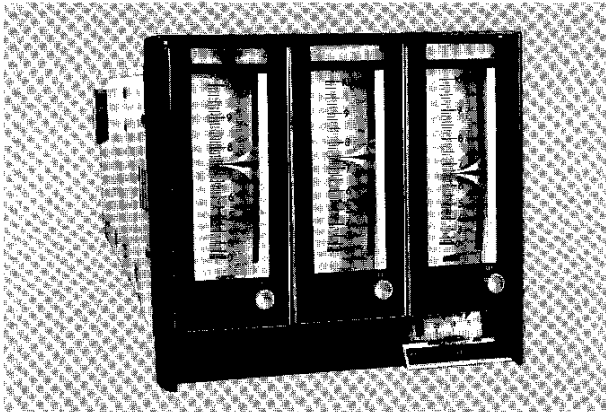


FIGURE 104 — Three-Variable Model 130 Auto-Selector Control System Showing Two Model 130FA Units to the Left of a Model 130MS Unit

The CONSOTROL Auto-Selector Control System enables two or more related variables to be held within pre-selected limits by positioning one control element. The variables are continuously measured, and control is transferred to whichever is tending to exceed its preset limit. (Refer to Section 1 for more information on auto-selector control.)

The system is comprised of one or more Model 130MA or 130FA Controllers without auto/manual transfer, one Model 130MS or 130FS Controller with auto/manual transfer, and separate high- or low-signal selectors. Each controller receives a standard pneumatic input signal from a process variable. All controllers send output signals to the selector system, which selects the lowest or highest output for transmission to the common auto/manual unit and thence to the final operator. The selected output is fed back to all controllers to avoid windup of those not selected.

Model 130MS and 130MA Controllers have local adjustment of set point only; Model 130FS and 130FA Controllers have local or remote adjustment of set point.

Model 130Z Pneumatic CONSOTROL Ratio Controllers

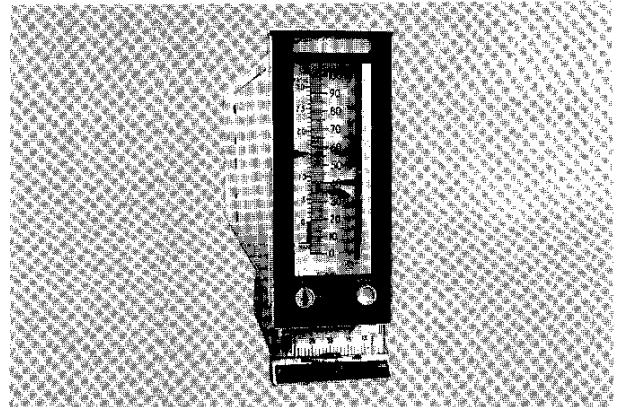


FIGURE 105 — Model 130Z Ratio Controller Showing the Ratio Setting (left side), the Controlled Variable (red inner pointer on right side), and the Uncontrolled ("Wild") Variable (black on white pointer on right side)

The Model 130Z Pneumatic CONSOTROL Indicating Ratio Controller maintains one process variable (the controlled variable) at some preset ratio to an uncontrolled ("wild") process variable. In essence, the set-point signal for the controlled variable is the wild variable's value times the selected ratio. (Refer to Section 1 for more information on ratio control.)

Either the wild variable or the set point can be displayed on the set-point indicator, as selected by a knob under the scale. If the ratio pointer is set below zero, its pointer enters a dark band on its scale called "MAN." In this band, the ratio knob drives the controller set point directly, unaffected by variations in the wild input. The selected ratio value is always displayed on the left side.

Several displays — in linear or square-root ranges — are available, depending on the relationship desired between wild and controlled flow. Either PI or PID control modes are available.

Model 130MC Pneumatic CONSOTROL Single-Station Cascade Controller

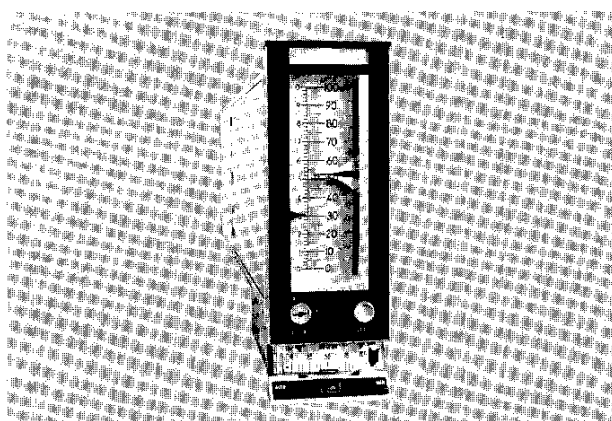


FIGURE 106 — Model 130MC CONSOTROL Cascade Controller

The Model 130MC Pneumatic CONSOTROL Single-Station Cascade Controllers maintain a normally slow-responding primary variable at a predetermined set point by manipulating a second, normally fast-responding variable in a closed loop. The output from the secondary controller drives the final control element.

Essentially, there are two control units here, both receiving standard pneumatic inputs from the process. The output of the primary variable's control unit becomes the set point for the secondary control unit. (Refer to Section 1 for more information on cascade control.)

Both control units are contained in the Model 130MC Controller. The primary variable and its set point are displayed on the right side of the instrument face, the secondary variable or its set point on the left side. PI control is applied to the secondary variable; PI or PID control is available for the primary variable.

If desired, the secondary variable can be controlled directly by redirecting the primary variable's set point to the secondary controller. An auto/manual transfer station allows manual control of the output of this unit. The secondary measurement is internally connected to the integral mode of the primary controller to avoid windup.

Model 130M-N Pneumatic CONSOTROL Indicating Gap Controllers

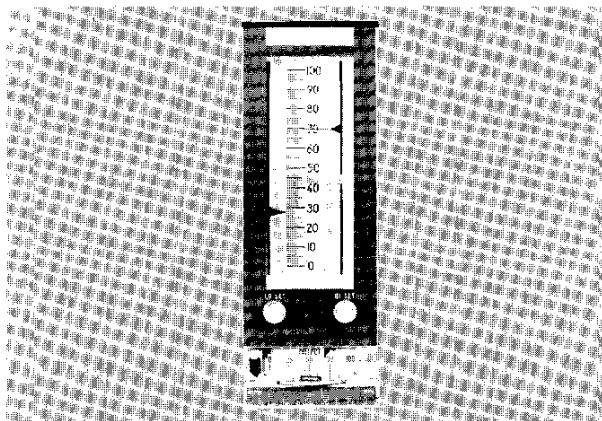


FIGURE 107 — Model 130M-N CONSOTROL Gap Controllers

There are two different control actions available.

- 1) The Model 130M-N7 Gap Controller provides *differential gap control* — a form of on/off control in which no change in output takes place while the process measurement is within a specified range (between set points). The output remains at maximum or minimum value until the controlled variable crosses the gap, causing the output to reverse. The controlled variable must then cross the band in the opposite direction before the output is restored to its original condition.

A typical application for differential gap control is filling a tank to the upper set point, then allowing it to empty until the lower set point is reached.

- 2) The Model 130M-N94 Controller provides standard PI control when the measured variable is outside of a gap or dead zone formed by the two set points. Within the dead zone, the proportional band is multiplied by a factor of approximately 50. This controller is useful when the disturbances to the valve are to be minimized, as long as the controlled variable lies between the two set points.

Model 130MD, 130PD, 130FD Pneumatic CONSOTROL Control Stations with Model 138 Pneumatic CONSOTROL Remote-Mounted Controllers

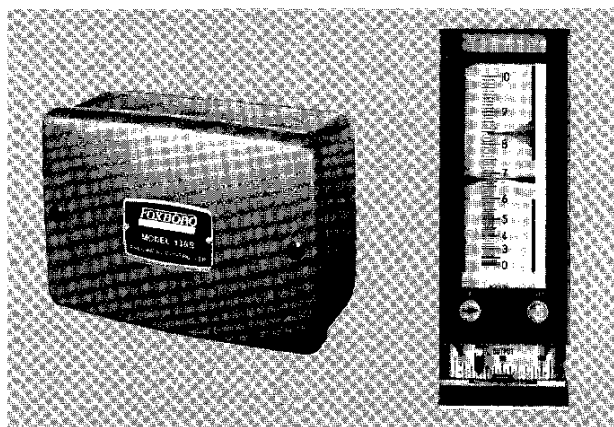


FIGURE 108 — CONSOTROL Model 138 Remote-Mounted Controller (left) with Companion Panel-Mounted Model 130MD/PD/FD Control Station (right)

The Model 138 Controller and companion Model 130MD/PD/FD Control Station are used when long distances between the process and the control room create unacceptably long time lags. To compensate for the slow response and reduced control this situation causes, the Model 138 Controller is mounted in the field near the process and provides standard modes of control. It transmits its output to the Model 130MD/PD/FD Control Station in the control room for convenience of the operator. The control room operator can change the set point and transfer between automatic and manual control.

The different control stations indicate different set points — the Model 130MD has only a local (i.e., internally generated) set point; the Model 130PD indicates only a remotely generated set point; and the Model 130FD has a two-position switch to allow either local or remote setting of the set point.

The Model 138 Controller has the standard force-balance operating mechanism. The Model 130MD/PD/FD Control Station has a variation of the position mechanism discussed in Section 6. The most complicated version (for the Model 130FD, shown in Figure 109) has two mechanisms, one of which receives a standard pneumatic input signal from the remote controller (the remote set-point receiver) and converts the signal into a pointer position. The other mechanism receives a signal from the internal (local) set-point generator and concurrently transmits the signal to the remote controller and converts the signal into a pointer position.

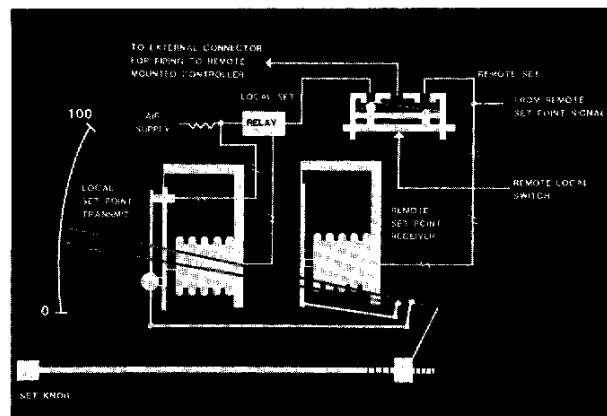


FIGURE 109 — Operating Mechanism of the Model 130FD Control Station

Model 135 Pneumatic CONSOTROL Auxiliary Stations

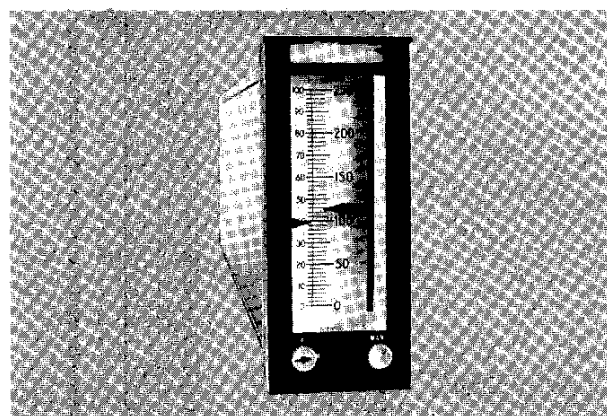


FIGURE 110 — Model 135 CONSOTROL Auxiliary Stations Provide Manual Loading, Switching, and Bias Functions

Model 135 Auxiliary Stations generate and indicate an adjustable standard pneumatic signal and transmit to a pneumatic receiver (e.g., a final control element or a remote controller). A variety of models allows the input and indication of one or two process variables, adds the capability to switch signal transmission between the process measurement and the generated value, and allows the input signal to be biased up to 100 percent. Model 135 Auxiliary Stations are usually used in control loops where a single controller regulates two or more final operators.

The Model 135 Auxiliary Stations use a variation of the position mechanism discussed in Section 6 to indicate the generated signal and produce a transmission signal.

The mechanism shown in Figure 111 indicates the optional functions that can be added to the basic one. (The option of displaying one or two process measurements adds a set of bellows/flapper-nozzle mechanisms and a pointer for each measurement; see Figure 109 for a similar arrangement.)

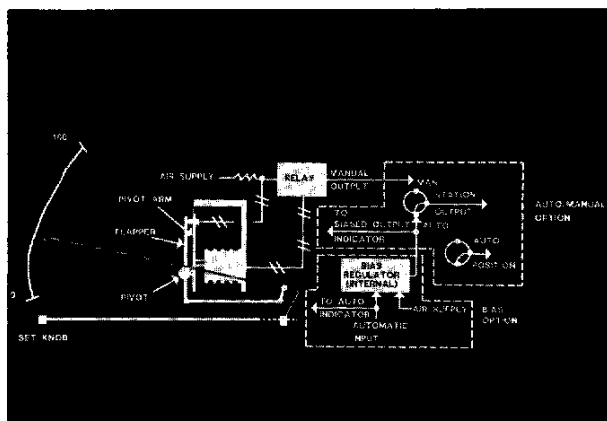


FIGURE 111 — Operating Mechanism of the Model 135 Auxiliary Station Showing Indication of Generated Signal, Auto/Manual Option, and Bias Option

Alarm Systems

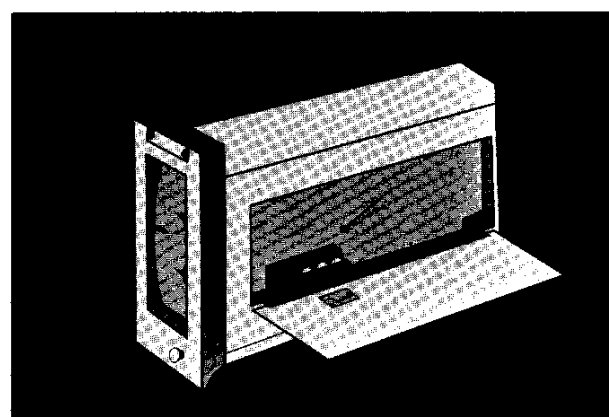
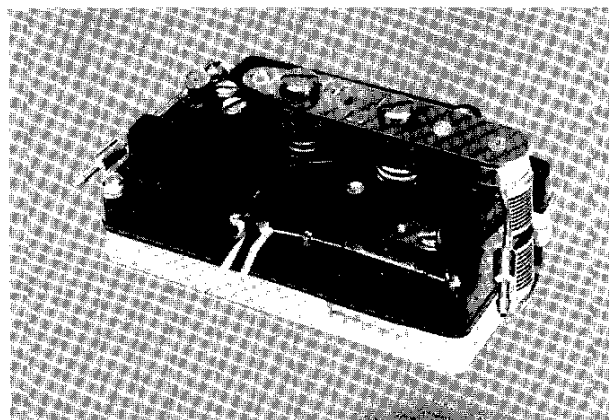


FIGURE 112 — Standard Alarm Used in CONSOTROL and Other Instruments (top) and Typical Installation Location (bottom)

Alarm actuators (signals to external audible alarms) and alarm flashers (blinking lights) are offered mounted in almost all Foxboro controllers, auxiliary stations, recorders, and indicators. The alarms are connected directly to standard pneumatic input signal lines; they are independent of instrument operation or display. Alarms can be absolute (comparing one incoming signal to the predetermined set points) or deviation (comparing the difference between two incoming signals to the predetermined set points).

Incoming signals are compared to predetermined critical high and low set points; when the input exceeds a set point in either direction, actuator and/or flasher signals initiate operation of the audible and/or visual alarms. To turn the alarm off and/or stop the light from flashing, the operator touches a contact switch on the front of the instrument. (The light stays on as long as the alarm condition exists.)

NOTE: Alarms and flashers require an electrical connection.

SECTION 10 — CONTROL VALVES AND POSITIONERS

A *final control element* is a general term for a device in a process line whose movement causes the value of the measured variable to change. The movement is determined by the signal received at the final control element from the controller. The most common final control element is a valve; others are dampers, louvers, variable-speed pumps, conveyor belts, and motor relays.

Each final control element has an internal mechanism to convert the signal format received (e.g., air pressure) into force and/or motion. This mechanism is called the *actuator* and it is relatively inexpensive and fast. It can be used by itself in a loop because it provides direct linear or rotary motion to the process. (In this direct physical connection to the process, it is the counterpart of the sensors discussed in Section 5.) Examples of stand-alone pneumatic actuators are shown later in this section.

Generally the actuator is part of a more complex element like a valve. Many valves have diaphragm actuators which look like the bell pressure elements described earlier (45 Series). The force of the pneumatic input signal is opposed by a spring; the equilibrium established between the force of the air and the force of the spring positions a stem which, in turn, positions the plug of the valve.

Although actuators alone provide fairly accurate positioning, a problem caused by hysteresis* may develop due to friction in the motion of the valve stem. Loops with a high proportional band or applications which cause only small changes in flow are particularly susceptible. To compensate for this, a *positioner* can be added to the valve.

A positioner is a signal converter which acts as a feedback controller to the valve for highly accurate positioning of the valve plug. Besides compensating for hysteresis, other typical applications include those with high pressure drop, highly viscous fluids, fluids that may cause gumming, fluids carrying sediment, loops where the response speed of the valve needs to be increased, temperature control, or pH control. Valve positioners should NOT be used when the process responds faster than the valve (e.g., flow) or when the use of a positioner would require increasing the proportional band substantially (instead, use a volume booster).

It is *extremely important* that the final control element in a loop be chosen carefully with respect to the process. More details follow, and further information can be found in literature listed in the Bibliography.

* *Hysteresis* is the difference between the position assumed for a value on increasing signal and the position assumed for that same value on decreasing signal.

Valve Positioners

Type C Vernier VALVACTOR Positioner

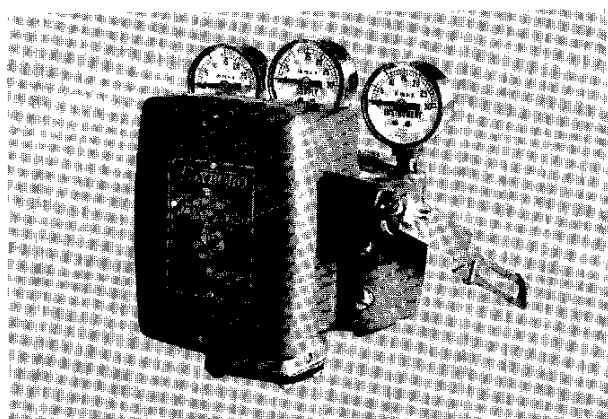


FIGURE 113 — Type C Vernier VALVACTOR Positioner

The Type C Vernier VALVACTOR Positioner enhances the response of an actuator to pneumatic input signal changes. It receives a standard pneumatic signal from a controller or signal converter and acts on it to drive the valve on which it is mounted. The Type C Positioner is used to reduce the effects of valve friction, the delay in response due to long signal lines, and the effects of changing line pressure.

The positioner may be adjusted for either increasing or decreasing output (i.e., air-to-open or air-to-close). Full stem motion can be provided using as little as one-third of the normal controller output; this allows operation of two or three control valves from one output signal.

Gauges mounted on the positioner indicate controller output, positioner output, and supply pressure. An optional bypass switch allows the valve to be driven directly by the input signal.

The VALVACTOR Positioner operates on a simple motion-balance principle. Figures 114a and 114b show the specialized type of motion mechanism the Type C uses and how it looks in the case.

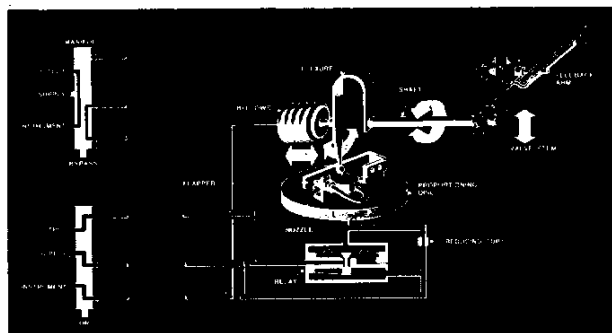


FIGURE 114a — Motion Mechanism for the Type C Positioner

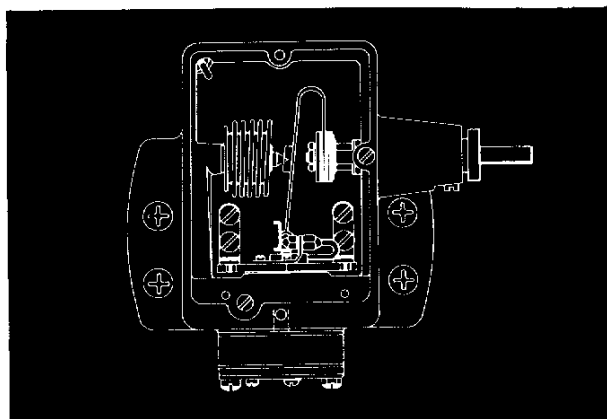


FIGURE 114b — Motion Mechanism in Type C Case

Model E69P Current-to-Pneumatic Valve Positioner

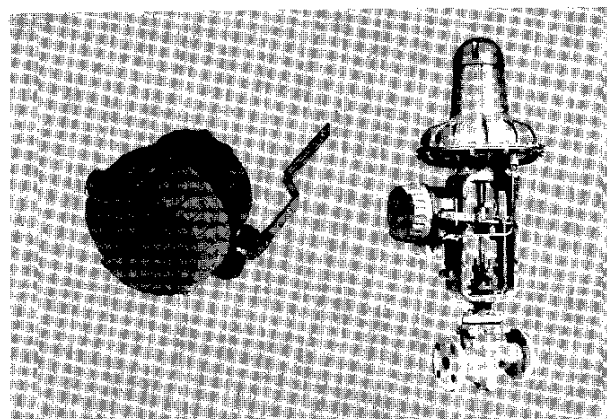


FIGURE 115 — Model E69P Current-to-Pneumatic Valve Positioner Shown by Itself, left, and Mounted on a Valve, right

The Model E69P Positioner, mounted directly on a pneumatic valve yoke, drives the valve stem to follow a standard direct current input signal. Although the standard action is *direct* (increasing signal producing increasing response) and air-to-open, the positioner can be provided with *reverse action* (increasing signal producing decreasing response) and/or air-to-close action.

Normal vibration of pipelines does not affect the operation of the Model E69P Positioner. It is also intrinsically safe (defined elsewhere), explosion-proof*, and dust-ignition-proof*.

* An *explosion-proof* enclosure is capable of withstanding an explosion of a gas or vapor which may occur within it; also, it will not cause a flammable atmosphere surrounding the enclosure to ignite, either as a result of an explosion that may occur within the enclosure or from the temperature of the enclosure.

A *dust-ignition-proof* enclosure excludes ignitable amounts of dust or amounts that might affect performance; also, it will not allow anything generated within the enclosure to cause ignition of dust or other accumulations on or around the enclosure.

The Model E69P Positioner uses the galvanometric motor described with the Model E69F and E69R Pneumatic-to-Current Signal Converters and the Model 33C emf-to-Pneumatic Transmitter. The feedback loop (shown in Figure 116) differs slightly.

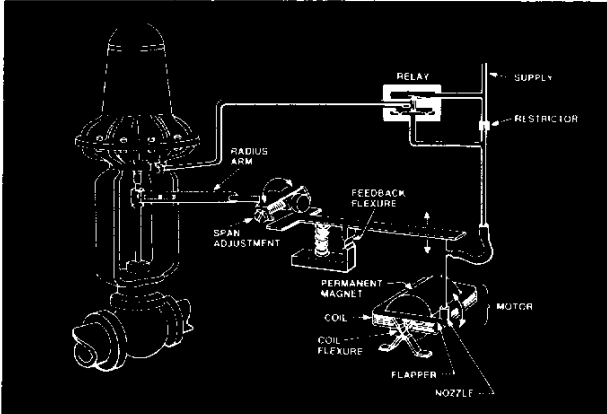


FIGURE 116 — Operation of the Model E69P Current-to-Pneumatic Valve Positioner

A dc milliampere input is applied to a coil and is positioned in the field of a permanent magnet. The coil exerts a tangential thrust proportional to the direct current input signal flowing through it. This thrust moves the flapper with respect to the nozzle, thereby changing the air output pressure of a pneumatic relay. The output pressure is applied to the pneumatic actuator which moves the valve stem. (Refer to the brief description of this action at the beginning of this section.) The motion of the valve stem is fed back to the nozzle by a direct mechanical linkage (the radius arm).

Valve Selection

Valve Sizing

Control valves are sized for two reasons:

- *economics* — If a valve is too small, it will not pass the required flow. The valve will have to be discarded and replaced by a larger, properly sized valve. Similarly, if the valve is too large, it will obviously pass the required flow but it will be more expensive than a properly sized smaller valve.
- *control* — An undersized valve will never deliver the full flow rate, thus sharply narrowing the controllable flow range. An oversized valve will be throttling near the closed position, and the full control range of the valve will not be utilized. When the plug throttles very close to the seat, high fluid velocities occur which can cause erosion.

The accepted method of valve sizing in the United States is the "C_v" approach. C_v is a capacity rating coefficient which is defined as the number of U.S. gpm of 60°F water which will flow through the wide-open valve when there is a pressure drop of one psi across the valve.

Foxboro uses the three following basic formulae for C_v calculation.

1. Liquids: $C_v = Q \sqrt{\frac{G}{\Delta P}}$

2. Gases*: $C_v = \frac{Q}{1360} \sqrt{\frac{T_f G}{\Delta P(P_2)}}$

3. Steam and Vapors*: $C_v = \frac{W}{63.3} \sqrt{\frac{v}{\Delta P}}$

Where: Q or W = flow rate (liquid in gpm, gases in scfh, vapors in lb/h)

G = specific gravity (available in engineering or valve handbooks)**

T_f = flowing temperature in degrees Rankine (°F + 460)

ΔP = pressure drop in psi (P₁ - P₂)

P₁ = upstream pressure at valve inlet in psia

P₂ = downstream pressure at valve discharge in psia

v = downstream specific volume in cubic ft/lb (available in engineering or valve handbooks)

Other Considerations

Under normal conditions, fluid passing through a valve will undergo a pressure drop across the valve orifice. This pressure drop can cause problems called flashing*** and cavitation***. While these conditions are too complicated to detail in this publication, you should be aware that problems can occur. (Refer to the books listed in the Bibliography.)

No matter how carefully the control system for the process loop has been designed, a mismatched control valve can cause the system to be unstable and difficult to control effectively. Valve characteristics help compensate for this. Two standard characteristics (linear and equal percentage***) accommodate most requirements, though characteristics can be customized for special situations.

Rangeability, the ratio of maximum controllable flow to minimum controllable flow, is stated for all valves. However, this does not reflect real flow conditions. A better guide is a second ratio, rated C_v to minimum controllable C_v; this ratio should be less than the stated rangeability of the valve. When the rangeability requirements rule out a valve that would otherwise be suitable for the application, it may be feasible to use two control valves in sequence. (Sequencing two valves to extend rangeability must be done according to a very specific procedure to retain the necessary characteristic and avoid bumping during transfer from one valve to another.)

* There is an important limitation imposed on the value of P used for vapor and gas sizing. It may never exceed one-half of the absolute inlet pressure, P₁, even though the valve will absorb up to 100 percent of the inlet pressure. If the pressure drop is greater than (1/2)P₁, use (1/2)P₁ for both ΔP and the downstream pressure, P₂. Remember to use this adjusted downstream pressure (1/2)P₁ in determining the downstream specific volume (v) under these conditions.

** For gases, specific gravity is molecular weight divided by 29.

*** Cavitation occurs in a valve when the pressure drop across the orifice first results in the pressure being lowered below the liquid's vapor pressure, then recovering to above the vapor pressure. The pressure recovery causes an implosion.

Flashing occurs when a fluid enters the valve as a liquid and exits partially as a vapor.

Equal percentage characteristic means that, theoretically, for equal increments in valve opening, an equal percent change in flow will occur. In practice, the gain can and does vary from the theoretical. (Tables available from Foxboro provide extensive data.) Generally, equal percentage valves are used on applications with highly varying pressure drop conditions such as those found in temperature control, while *linear characteristic* valves (where flow is directly proportional to valve travel at a constant pressure drop) are most often used in flow, liquid level, and gas pressure control.

Both the stability and the responsiveness of the loop are affected by process dynamics, and linearity and valve characteristics. Guidelines to these process dynamics (available elsewhere) can help you in selection of valve characteristics.

Valve selection depends on various aspects of the process: fluid composition (e.g., corrosiveness, viscosity, any solids in it), line pressure, and process temperature. The valve with the widest application is the globe; it is also the most expensive. Table F and the valve descriptions give basic application information. But first become familiar with some valve terminology.

TABLE F — Application Data for Valves

	Globe	Needle	Ball	Butterfly	Saunders
Characteristic and Rangeability	= % or linear (50:1)	= % or linear (approx. 35:1)	Approx. = % (50:1)	Approx. = % (30:1)	Approx. linear (3:1 conventional 15:1 dual range)
Use on Slurries or Dirty Solids-Bearing Fluids	Very poor	Very poor	Reasonably good	Poor	Very good
Relative Cost	High to very high	High	Medium	Lowest for large sizes	Medium
Maintenance Rating	Good; easy to work on	Good; easy to work on	Good; disassembly may be difficult	Good	Fair
Tight Shutoff	Yes, if single soft seat is used	Yes, if single soft seat is used	Yes, if lined	Yes, if lined	Yes
Rating as a Control Valve	Excellent	Excellent	Excellent, if characteristic is suitable	Good, if characteristic is suitable	Poor to fair (generally used only when dirty flow dictates)

Terminology

Figure 117 identifies the primary components of a single-seat globe valve, the most commonly used type. Other valve types are variations of this.

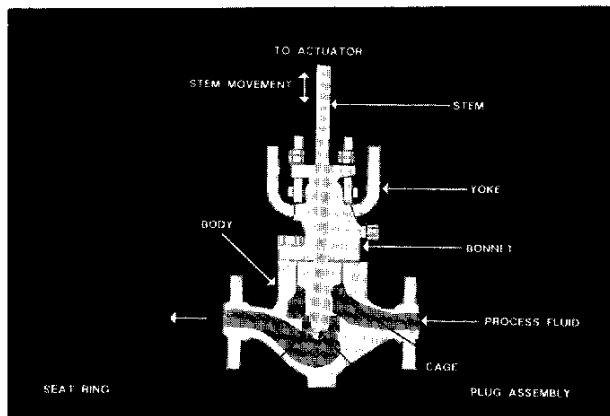


FIGURE 117 — Typical Single Seat Globe Valve Assembly

The process fluid enters the valve *body* and passes through an opening in the *seat ring* created by the valve *plug*, which is positioned by the *stem* movement in response to the actuator. A single “soft” *seat ring* allows the fully closed plug to create a leaktight seal for most applications. A double-seat valve (with the second seat ring at the top of the cage) is better for high-pressure applications, but does not provide a tight seal.

The design of the valve plug identifies the type of valve: within the globe-style valves, the plug can be *balanced* or *unbalanced*, *poppet* (on/off) or *V-port*. A *needle valve* is a smaller version of the globe, with a pointed plug that looks much like a needle. Other valve styles have slightly different components that perform the same functions: a *ball valve* has a sphere (with a path through it) that turns within the valve body; a *butterfly valve* has a disk that turns on an axis within the valve body; and a *Saunders valve*, sometimes called a diaphragm valve, has a set of diaphragms instead of the plug and seat ring.

Valves

V1 STABILFLO Control Valves

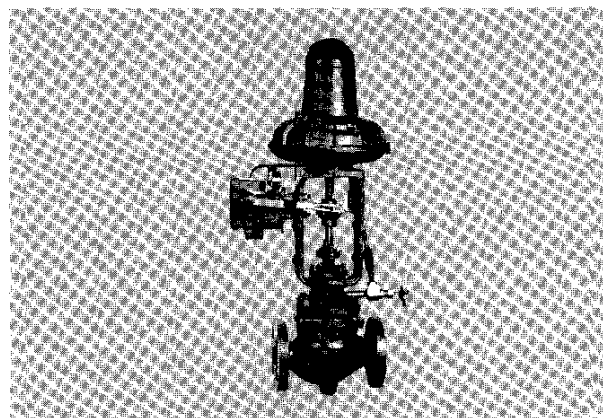


FIGURE 118 — V1 Globe-Style Control Valve with Positioner

The V1 Globe-Style Control Valve is extremely versatile, providing a variety of materials, interchangeable parts, minimal hysteresis, and wide rangeability. It handles a high capacity, yet has exceptionally tight shutoff. With actuators, the V1 provides fast response with linear or equal percentage valve characteristics.

Two versions are available: a stem-guided version with unbalanced inner valve, single seat, and contoured plug; and a cage-guided version with balanced inner valve, single seat, and balanced plug.

Model V4A Parabolic Needle Low-Flow Control Valve

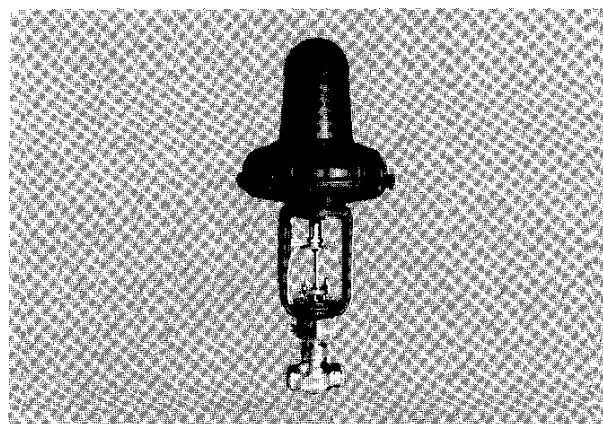


FIGURE 119 — Model V4A Parabolic Needle Low-Flow Control Valve

The stem-guided V4A Needle Valve is specifically designed for low-flow service in laboratories and pilot plants as well as industry. Despite the small size of the valve, its rugged construction assures accuracy under high process pressure conditions.

The pointed valve plug (needle) provides equal percentage characteristics through 35 to 1 rangeability (i.e., equal increments of stem travel produce equal percentage increases in flow from full capacity to 1/35 of full capacity).

Models V2000 and G2 Saunders Valves

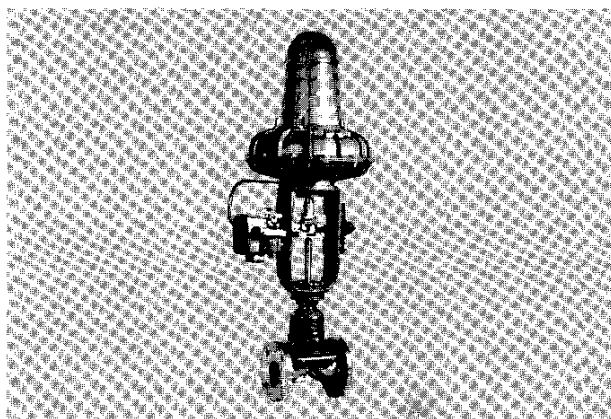


FIGURE 120 — Models V2000 and G2 Saunders Valve

The V2000 and G2 Control Valves are used for low pressure, hard-to-handle fluids and tight shutoff applications involving corrosive or dirty fluids. Protective body linings and plastic diaphragms can be provided for applications ranging from chemical to sanitary to pulp stock and slurries. The V2000 provides limited throttling control and the G2 provides on/off control.

The design (Figure 121) was originally based on the open-channel flowmeter weir to provide smooth passages for fluids with suspended solids. The V2000 Valve comprises a two-piece compressor assembly that provides independent control over two areas of the valve diaphragm. The first increments of stem travel raise only the inner compressor from the weir, allowing flow

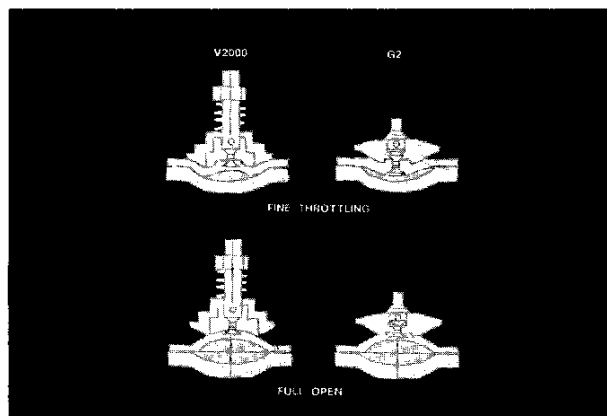


FIGURE 121 — Operation of the V2000 and G2 Saunders Valves

through a contoured opening in the center of the valve. The outer compressor raises only when further stem travel brings the inner compressor into contact with it and both are raised as a unit. (In contrast, the G2 Valve one-piece compressor allows only a single stage of opening.)

Model V9000 Ball Control Valves

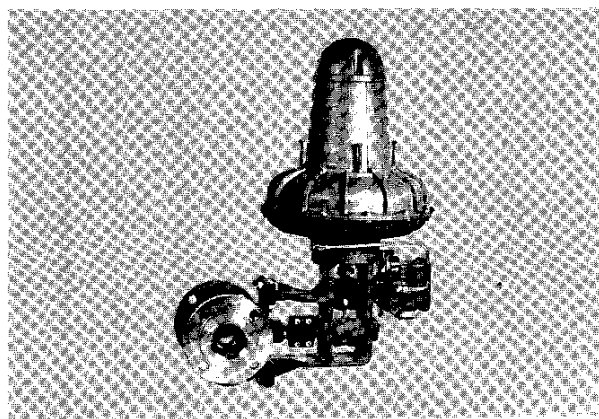


FIGURE 122 — Model V9000 Ball Control Valve

Model V9000 Ball Control Valves have a streamlined flow passage with straight-through unobstructed design for exceptional flow capacity. They can be applied to a wide range of processes. They are ideal for pulp stock, other nonabrasive slurries, and also heavy oils, chemicals, and viscous fluids. V9000 Ball Valves are highly controllable with inherent equal percentage characteristics and a 50 to 1 rangeability. Diaphragm actuators (P50 or P110 Series, described in this section) are provided with all V9000 Ball Valves.

Ball valves in general provide bidirectional flow, high-flow capacity, leaktight shutoff, and simple construction with a minimum of parts. V9000 Ball Valves have the additional advantages of positive, long-lasting shutoff, reliable stem sealing to eliminate stem leakage, and safety features like a blowout-proof stem and special metal-to-metal sealing for application in processes with flammable fluids.

The ball valve (shown in Figure 123) has a cylindrical hole through its center which lines up with the connecting piping in the fully open valve position. The valve is fully closed when the ball rotates 90 degrees in response to stem rotation, so that the cylindrical hole no longer lines up with the connecting piping.

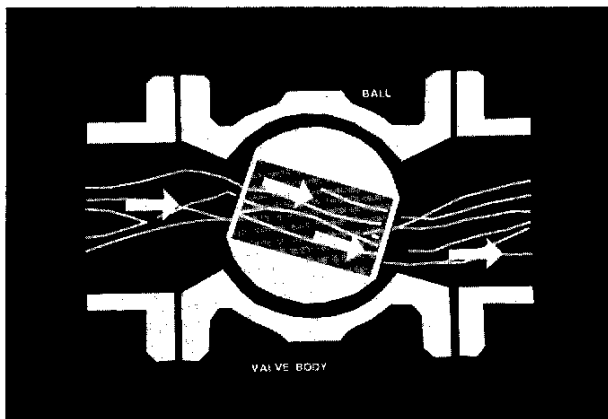


FIGURE 123 — Operation of the V9000 Ball Valve from the Top (valve stem would come toward viewer from ball)

V3000 Series Butterfly Control Valves

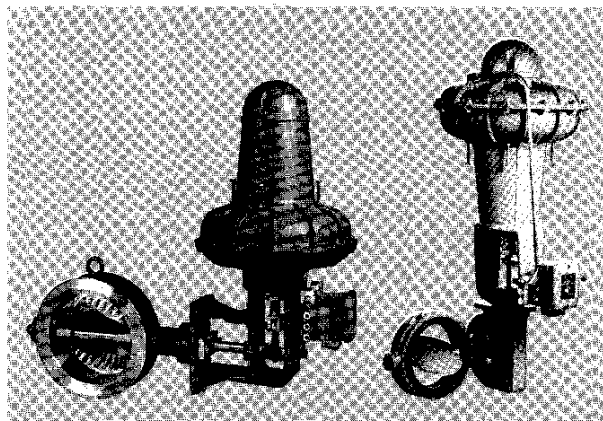


FIGURE 124 — Heavy-Duty V3000 Unlined Butterfly Valve (left) and V3400 Lined Butterfly Valve (right)

The V3000 Series Butterfly Control Valves provide precise control of virtually all common liquids, vapors, and gases through several vane designs, with the availability of liners for different applications. The valves feature high capacity (for line sizes up to 16 inches), rangeability of up to 100 to 1, very low leakage, modified equal percentage flow characteristics, and minimum hysteresis. A P50 or P110 Series Actuator (see descriptions following) is supplied with each valve.

The V3000 Butterfly Valve (shown in Figure 124) has a unique vane design: it has teeth. This produces lower torque than conventional designs for stable operation with a smaller actuator. It also reduces noise, resists cavitation, and increases rangeability. The V3000 Valve provides throttling control.

The V3400 Butterfly Valve vane design provides bubble-tight shutoff. The vane and valve body have metal-reinforced liners for resistance to blowout and distortion. The V3400 Valve (shown in Figure 124) also provides throttling control.

The V3500 Butterfly Valve has a flat, smooth, specially lined vane that, in conjunction with the specially lined body, is qualified for food service. It was designed to optimize resistance to contamination and corrosion with the bubble-tight shutoff of the V3400. The V3500 Valve provides proportional control.

The V3600 Butterfly Valve has the flat vane design of the V3500 but with liners suited to corrosive or fibrous fluid applications. It also has bubble-tight shutoff and provides throttling control.

Actuators

20F/P50 and 30C/P110 Diaphragm Lever Actuators

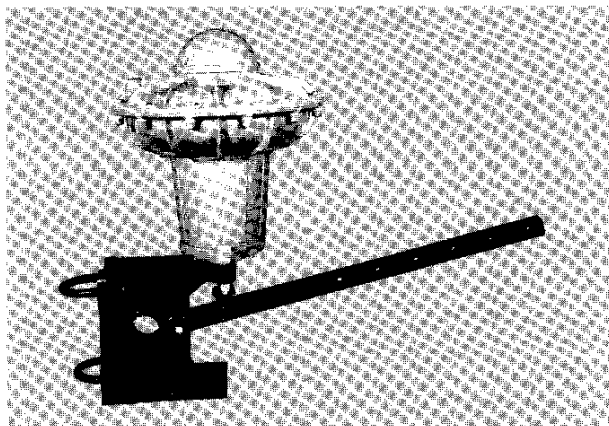


FIGURE 125 — P50 Actuator (topworks) with 20F Lever System (P110 Actuator and 30C Lever System Is Slightly Larger)

The 20F/P50 and 30C/P110 Lever Actuator systems convert a pneumatic input signal into mechanical motion or force to actuate dampers, variable-speed drives, weight-loaded regulators, governors, rheostats, and other similar devices. They are lightweight for simplified installation, with high-speed performance and reversible action (either air-to-open or air-to-close).

The Types P50 and P110 are pneumatic spring diaphragm type actuators and are frequently used alone mounted on a valve yoke to actuate the valve stem.

The Types 20F and 30C levers have six and nine holes, respectively, to connect to the final control device being driven. They are designed to produce maximum possible diaphragm stroke. The lever systems are vinyl-coated for resistance to industrial corrosive atmospheres, and have low friction seals for minimum hysteresis.

2A Cylinder Actuator with Power Positioner

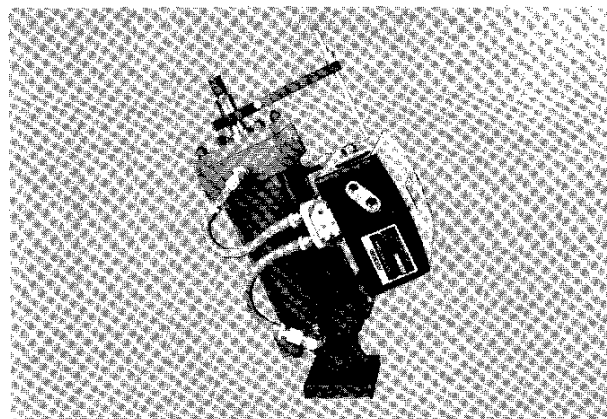


FIGURE 126 — 2A Cylinder Actuator with Power Positioner

Cylinders are used to actuate dampers, gates, shutters, doors, and large control valves like the Saunders type. Throttling action is provided by power positioners (the box and tubing shown in Figure 126), which convert a standard pneumatic input signal to full supply on the cylinder piston. The combination provides smooth, fast, friction-free operation. Many mounting configurations are available.

If throttling action is not needed and only piston motion is desired, air can be piped directly to the cylinder.

PART IV

AFTER THE SYSTEM IS INSTALLED

SECTION 11 — USING A LOOP DIAGRAM

So far in this publication, all instruments have been illustrated with photographs or drawings. Even the figures in Section 1 showed actual process equipment. Frequently, however, process control systems are shown with *loop diagrams*. Their interconnected loops provide a kind of shorthand to show the relationships between parts of a system. It is important to understand this shorthand because the information is useful in understanding loop functions and identifying loop components.

Loop diagrams may include engineering and construction details, connections, locations, energy sources, size of vessel, shape of vessel, length of piping, size of piping, and process information like the fluid or medium being controlled. A common type of loop diagram is a process and instrument diagram (*P & I diagram*). It indicates the flow of information throughout the system relative to process vessels, piping, prime movers, and other equipment.

These diagrams are generally supplied by the contractor, plant designer, or someone at the plant. Although the plant instrument engineer is usually responsible, technicians should have some knowledge of how to interpret loop diagrams both for use in *fault location* (troubleshooting) and in describing or designing instrument loops.

To make instrument and equipment identification uniform, the *Instrument Society of America* (ISA) has adopted a set of symbols. Their guidelines (detailed in their publication S5.1) are generally used by Foxboro and are shown in this publication where appropriate.

Devices and device functions can be expressed by a single shape, by a shape plus an abbreviation (either a letter or a symbol), or by combinations of shapes and abbreviations. Typical ISA symbols for instruments and process equipment are shown in Figure 127. The table following lists abbreviations and their usual meanings. (Note that letters can have many meanings, depending on their position in the tag, device function, or user designation.) Figure 128 shows combinations representing multifunction instruments.

Other symbols used by the process industry may be found in publications of various industry standards organizations (particularly the *Scientific Apparatus Makers' As-*

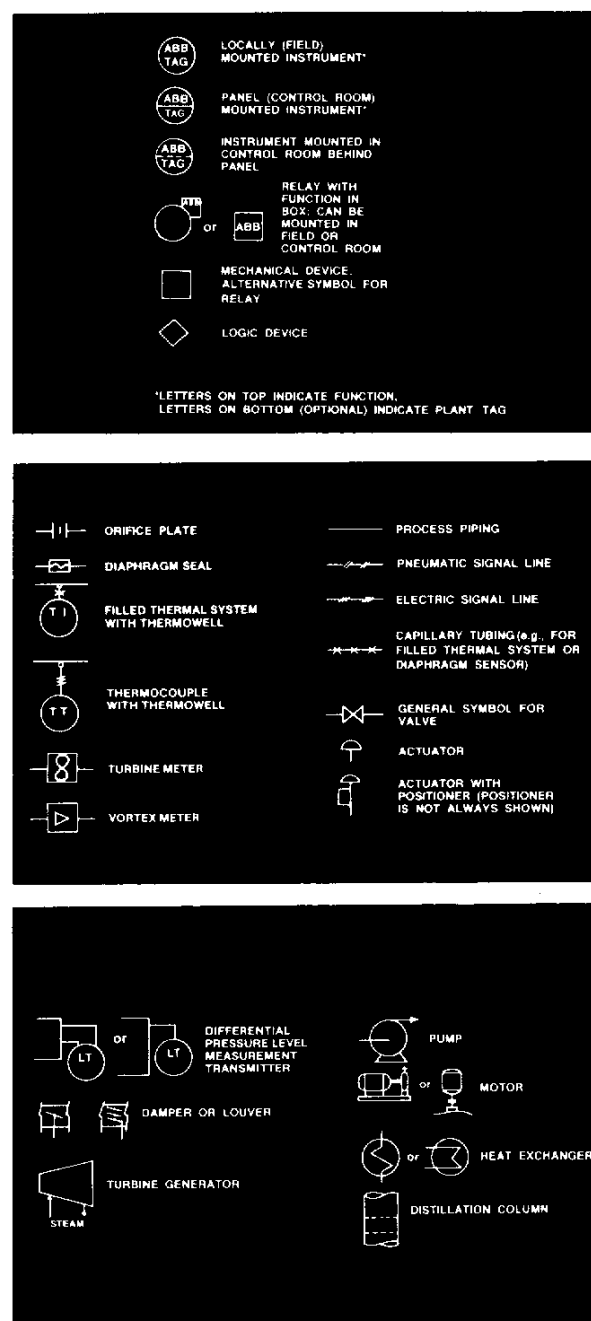


FIGURE 127 — Typical ISA Symbols

sociation — SAMA). These vary somewhat from the ISA standard, and it is suggested that you obtain any publications that might apply to your industry. Once you become familiar with the symbols and abbreviations that go in them, it will be easy to read loop diagrams.

A relatively simple loop — a temperature control loop on a heat exchanger — is shown in both pictorial and loop diagram forms in Figures 129a and 129b. A more extensive, interactive process (combustion control in a boiler) is illustrated in Figure 130.

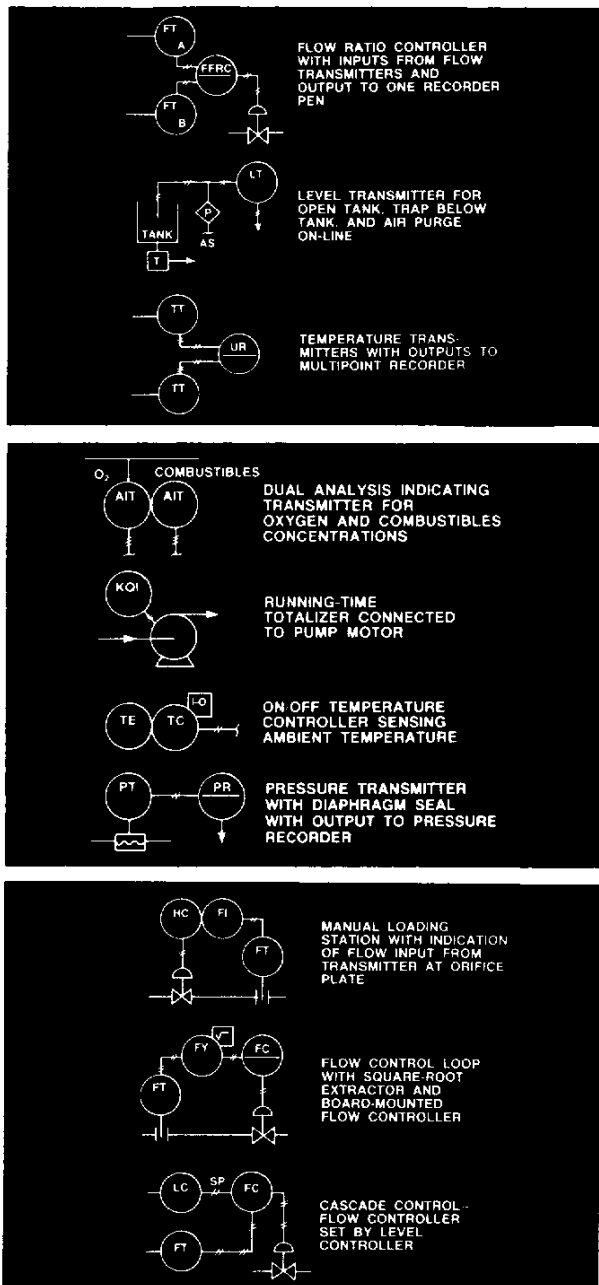


FIGURE 128 — Typical Combinations of ISA Symbology

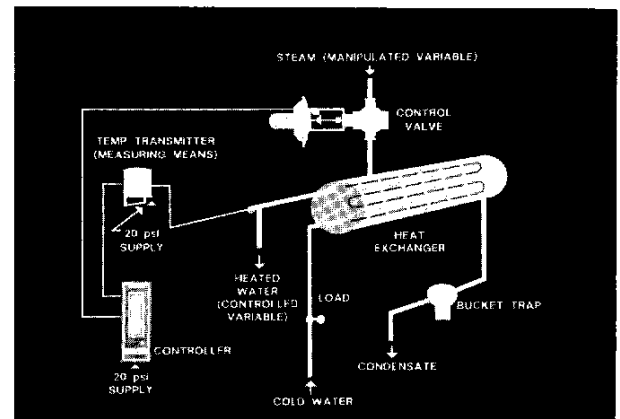


FIGURE 129a — Shell-and-Tube Heat Exchanger in Pictorial Form

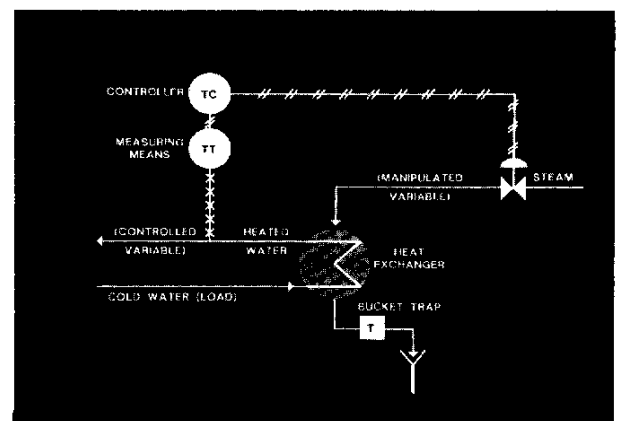


FIGURE 129b — Shell-and-Tube Heat Exchanger in Loop Diagram Form

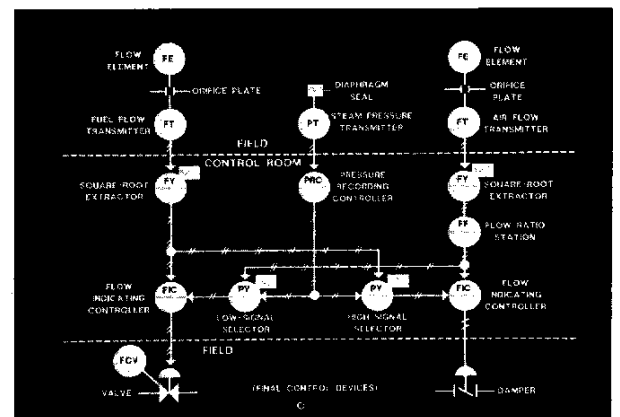


FIGURE 130 — ISA Symbology Describing an Interactive Control Loop (Boiler Firing Control)

TABLE G — Meanings of ISA Abbreviations

Letter	Used in			
	Circle*	Relay Superscript	Square/Diamond	Elsewhere
A AS	Analysis; Alarm	Analog		Air Supply
B	Burner, Combustion			
C	—; Control			
D d/dt	—; Differential	Digital Derivative	Digital	
E ES	Voltage (emf); Sensor	Voltage		Electric Supply
F FF FQ	Flow Rate; Ratio (Fraction) Flow Ratio Flow Quantity			
G GS	—; Glass, Viewing Device			Gas Supply
H HS	Hand; High	High; Hydraulic		Hydraulic Supply
I IA I-O	Current (electric); Indicate	Current On/Off	Interlock	Instrument Air
J	Power; Scan			
K	Time, Time Schedule; Time Rate- of-Change, Control Station	Proportional		
L	Level; Light, Low	Low		
M	—; Momentary, Middle or Intermediate			
N NS		Number of analog inputs or value of exponent		Nitrogen Supply
O	—; Orifice (Restriction)	Electromagnetic, Sonic		
P PA PD	Pressure, Vacuum; Point (Test) Connection Pressure Differential	Pneumatic	Purge or Flush	Plant Air
Q	Quantity; Integrate, Totalize			
R REV	Radiation; Record	Resistance (Elec.) Reverse action	Reset	
S SP SS	Speed, Frequency; Safety, Switch		Solenoid	Set Point Steam Supply
T TD	Temperature; Transmit Temperature Differential	Time	Trap	
U	Multivariable; Multifunction			
V	Vibration, Machinery Analysis; Valve/Damper/Louver			
W WD WS	Weight, Force; Well Weight/Force Differential			Water Supply
X	—; X Axis	Multiply, analog input variable		
Y	Event, State or Presence; Y Axis, Relay, Compute			
Z ZD	Position, Dimension; Z Axis, Driver/Actuate/Unclassified Final Control Element Gauging, Deviation			

TABLE G Continued

Symbol	Used in			
	Circle*	Relay Superscript	Square/Diamond	Elsewhere
Σ		summing		
Σ/n		averaging		
Δ		difference		
$\int$		integral		
$\div$		dividing		
$\sqrt{\quad}$		square-root extraction		
x^n		exponential		
$f(x)$		nonlinear or unspecified function		
$f(t)$		time function		
$>$		high selecting		
$<$		low selecting		
$\uparrow$		high limiting		
$\downarrow$		low limiting		
$-k$		reverse proportional		
∇		velocity limiter		
$\left. \begin{matrix} + \\ - \\ \pm \end{matrix} \right\}$		bias		
$*/\circ$		convert		

* When used in a circle, meanings before the semicolon generally apply if the letter is in the first position, with meanings after the semicolon applying to succeeding letters.

NOTE: Letters can be given meanings other than those shown here for users' convenience, but ANY CHANGE IN MEANING MUST BE SPECIFIED ON THE DRAWING.

SECTION 12 — MAINTENANCE AND REPAIR

What Instrument Maintenance Entails

The following tasks are generally considered to be part of the function of a maintenance technician.

Preventive Maintenance: Comprises both everyday upkeep and periodic servicing

Emergency Service: work performed on a rush basis when breakdowns occur requiring fault location (sometimes called troubleshooting).

Commissioning or New Installation Checkout, Startup, and Tuning: verifying system and instrument operation, assisting during periods of critical operation, and tuning or characterizing instruments to suit process requirements.

Repair and Modification: bench work to correct defects or rescale or recalibrate instruments; also changing or updating the piping, wiring, location, or anything else unique to individual instruments or systems.

Each of these tasks occupies varying amounts of the technician's time and requires different procedures and different types and degrees of skill. Each will be discussed separately. Following this is a discussion of the physical environment in which these tasks are performed.

Preventive Maintenance

Everyday Upkeep: chart changing, inking, cleaning, and all other tasks involved in day-to-day instrument operation. This is done primarily as directed by manufacturers' instructions.

Periodic Servicing: work performed on a regular basis per instrument manufacturers' recommendations as amended by company policy (visual inspection, cleaning, leak checks, calibration and performance checks, seal replacement, valve packing, etc.).

Emergency Service

This generally comprises three or four steps: 1) determine the problem area (e.g., system design, instrument failure, or external influence); 2) correct problem on-site or, if this is not possible, replace the faulty unit; 3) check out and restart the system; and 4) if the unit was removed from the field and replaced with a new one, take the unit back to the shop where the fault can

be located and the instrument repaired and recalibrated during a slack period or the unit can be returned to the manufacturer. As with other types of maintenance tasks, for emergency service the technician often depends on documentation supplied by the manufacturer and possibly modified by the plant engineers to reflect special applications.

One note is perhaps in order. Process control systems are very expensive, so they are usually installed because the systems they will be controlling are even more expensive. System downtime can cost hundreds of dollars per minute, so it is important that emergency servicing be done as efficiently as possible.

Since hundreds of malfunctions are possible within a system, to isolate a problem you must understand the way the system operates. This includes the type of measurements, the way the different parts of the system interrelate, and the instruments used in the particular system. This information can be gathered from observation, referring to this publication and others, looking at functional block diagrams for the system, and reading the documentation available for the system.

The most logical approach to discovering the source of a problem is to take the available documentation and start eliminating potential sources of problems. For example, if the controller outputs are incorrect, you would want to check the air supply (if it is a pneumatic instrument) and the inputs. Examine the inputs by type and plant area; manually check any suspect inputs. If these are all right, check both the input and output portions of the controller.

Commissioning (New Installation Checkout, Startup, and Tuning)

Instruments should be installed and calibrated according to manufacturers' directions. Piping and wiring should be done according to plant specifications. (There are many helpful reference documents on good installation practices published by trade organizations.) The process is then checked out, the objective being to determine whether the equipment is installed correctly and whether the equipment as installed can achieve a balance between process supply and demand. Rarely can this balance be achieved without further adjustment. The loop tuning that accomplishes this balance is covered in Section 13.

Repair and Modification

The *calibration* of a pressure instrument is done by comparing its input and output against standards. The instrument is then adjusted to make it agree with the standard. Success in calibration depends not only on one's ability to adjust the instrument, but also on the quality of the standard.

Field standards must be reasonably convenient to use and must satisfy the *accuracy* requirements for the instrument under calibration. A 2500 mm (100 in) water column, for example, is extremely accurate but not practical to set up in a plant. For practical reasons, most field standards are test gauges.

A good quality test gauge will be designed, constructed, and calibrated with care, so that it is accurate to within ± 0.25 percent of span. This is adequate for field calibration of most of the pneumatic and mechanical equipment described in this publication and even for most electronic equipment. The test gauge should provide the standard pressure regulator, scale, and manifold connections described in "The Maintenance Shop" following.

Under some conditions, a *manometer* (an instrument used to measure the pressure of gases or vapors) may be used in the field. This usually occurs when a low pressure range is to be calibrated and no other suitable standard is readily available.

The Maintenance Shop

The dust, grit, iron filings, and chips of conductive material found in corrosive, dust-laden atmospheres become extremely harmful to an instrument when the protective covers are removed for servicing. Magnets, circuit boards, close-tolerance pivots, relays, and switch contacts are easily damaged by tiny bits of foreign material. Therefore, whenever possible, instruments should be brought back to the repair shop for work.

Shop areas should be kept conspicuously clean, and dust-producing jobs such as filing, sanding, and grinding should be well isolated from the calibration and repair area. As further protection, dust covers should always be on except during actual work on instruments.

Test Equipment and Standards

Instruments are devices designed to act upon input information in a specified manner and to provide a predictable output. Inputs may represent temperature, pres-

sure, vacuum, current, voltage, manual adjustments, etc. Outputs may represent pen position, force, resistance or millivoltage, pneumatic or electronic signals suitable for transmission to other equipment, and so on. But no matter what function an instrument is to perform, its desired reaction to a given set of circumstances is known beforehand. Thus, a trained technician can readily diagnose or adjust the instrument's response by regulating its input and observing the resulting output. Such tests, known as instrument performance tests or calibration checks, make use of the setup shown in block diagram form in Figure 131.

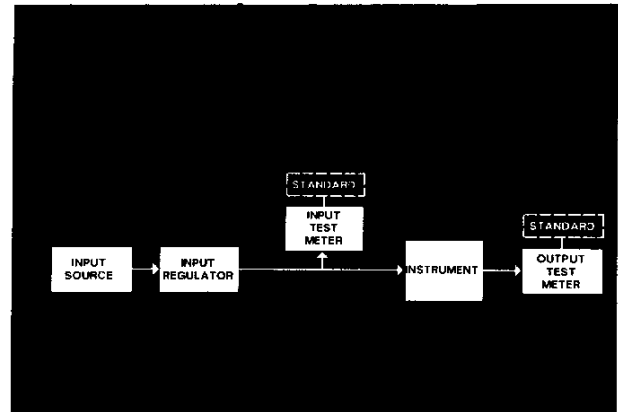


FIGURE 131 — Block Diagram of Performance Test Setup

If, for example, the instrument being tested were a pressure transmitter, the input source would be a source of pressure, the input regulator would be a calibration manifold capable of locking up the appropriate input pressure as observed at the input test meter (such as a deadweight tester or mercury manometer), and the instrument response would be read on the output test meter.

Any sort of equipment will suffice for the input source and input regulator as long as it is capable of establishing and holding the input signal at desired points throughout the instrument's operating range. For example, the process itself is often used as an input source and regulator. Input and output test meters, on the other hand, determine the accuracy basis of the performance test which, in turn, determines instrument accuracy and thus process performance.

Accuracy really boils down to the question: How much error is acceptable and practical? Instruments and test equipment are, therefore, rated by an allowable band of error generally expressed as a percentage of span or full scale above and below the nominal value. If, for example, the span of an instrument was 100 psi with a range of 0 to 100 psi, a ± 1 percent accuracy would mean that for any value of input the output would be

no more than 1 percent, or 1 psi, over or under what it should be. For example, if the output should be 50 psi, the actual output would be no greater than 51 psi and no less than 49 psi. Pneumatic instruments are usually rated at ± 0.5 percent of span; electronic instruments are often rated at ± 0.25 percent of span.

To today's instrument technician, one or two special purpose calibrators capable of receiving and transmitting a standard transmission signal are adequate for testing a majority of instruments in the plant. Following are the standard transmission signals commonly used by the instrument industry.

<i>Pneumatic (Pressure)</i>	<i>Electronic (Current)</i>
3 to 15 psi (20 to 100 kPa)	4 to 20 mA dc
3 to 27 psi	10 to 50 mA dc

Since this publication has dealt with pneumatic instruments, only the pneumatic signal calibration setup will be discussed here.

The pneumatic standard signal requires only a source of pressure (such as a hand pump or air supply), an inexpensive calibration manifold, and test gauges of suitable range and accuracy.

The pneumatic calibration bench setup shown in Figure 132 can produce a pneumatic signal for the input test meter shown in Figure 131.

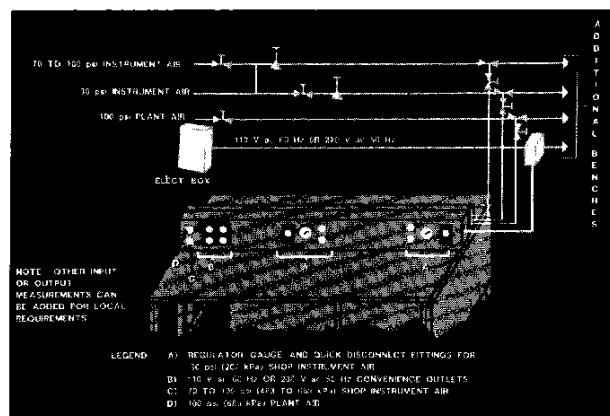


FIGURE 132 — Calibration Bench Setup

Special tools needed for assembling, aligning, and calibrating specific instruments can be purchased from instrument manufacturers in kit form. One such kit available from Foxboro is shown in Figure 133.

Table H, showing Measuring Element Test Equipment following, relates types of measurements to the test equipment commonly used for each.



FIGURE 133 — Manufacturer's Tool Kit

TABLE H — Measuring Element Test Equipment

Measurement			Test Equipment			
Input Variable (Measurement)	Sensing Element Type	Element Output Variable	Input Equipment (Source)	Regulator	Input Test Gauge	Standard
Pressure or Vacuum	Diaphragm	Motion	Compressor, Pressure Tanks, Hydraulic Pump, Deadweight Tester	Calibration Manifold	U-tube Manometer Deadweight Tester Test Gauges	U-Tube Manometer Deadweight Tester
	Helical					
	Spiral					
	Bellows	or				
	C-Tube					
	Capsule	Force				
Differential Pressure (Flow Applications)	Opposed Bellows or Diaphragm	Motion	Same as Pressure or Vacuum			
	Differential Pressure Transmitter	Force				
Flow	Restrictors	Diff. Pressure	Flow Laboratory Equipment			
	Magnetic Flowmeter	emf				
	Vortex Flowmeter	Frequency				
	Target Meter	Force	Special Weights			
	Float-and-Cable	Motion	Scale			
Temperature	Filled Systems	Motion or Force	Temp. Baths, Dry Ice, Elec. Heaters, etc.	Electronic Controls As Required	Test Thermometer	Certified Thermometer
	Thermocouple	emf	Lab. Furnace or Process			
	Resistance Bulbs	Resistance	Temp. Baths			
Speed	Tach. Gen.	emf	Var. Speed			
	Contact Closure	Frequency	Motors, Synchronous			
	Torque	Pressure	Motors			
Level	Air Purge	Pressure	Process Tank	Level Contr.	Tank Height	Yardstick
	Diaphragm Box	Pressure	Process Tank	Level Contr.	Tank Height	Yardstick
	Capacitance Probe	Capacity	Process Tank	Level Contr.	Tank Height	Yardstick
	Buoyancy	Force	Weights			
	Pressure and Diff. Pressure	Motion or Force	See Pressure and Differential Pressure			
	Float-and-Cable	See Flow	See Flow			
Humidity	Dew Point Element	Temperature	See Temp — Resis.		Sling Psychrometer	Sling Psychrometer
	Hair Element	Motion	—	—		
ANALYTICAL						
Ion-Selective (pH, etc.)	Electrodes	emf	Precision mV Source — Known Sample Solutions		Laboratory Tests	Laboratory Standards
ORP, Redox, etc.	Electrodes	emf				
Conductivity		Probe	Resistance Decade Box	Resistance		

SECTION 13 — LOOP TUNING

The closed-loop control system attempts to achieve a balance between supply and demand by comparing the controlled variable to the set point and regulating the supply to an amount which will maintain the desired balance. *Tuning* the controller adjusts it so it can achieve that balance as quickly as possible. This is done when the instrument is first put in service and, later, on a periodic basis as part of preventive maintenance.

When tuning, remember that each controller is part of a closed loop: all the parts of the loop are interactive, so adjustments made at the controller must reflect the behavior of other devices in that loop. The controller response must be matched to that of the process. There are several procedures for doing this, some mathematical, most using trial and error.

A simple, three-step method for tuning most *three-mode controllers** follows. (Batch controllers and once-through processes are special cases discussed after the three-mode and two-mode controllers.) This three-step procedure is based on a simple test to determine the natural period of oscillation of the process.

Step 1: Set the integral time of the controller at its maximum and the derivative time at its minimum, thereby providing proportional-only control. Then reduce the proportional band until oscillation begins. Measure the period of this oscillation (also called the *natural period*) as the time between two successive crests or valleys. (See Figure 134.)

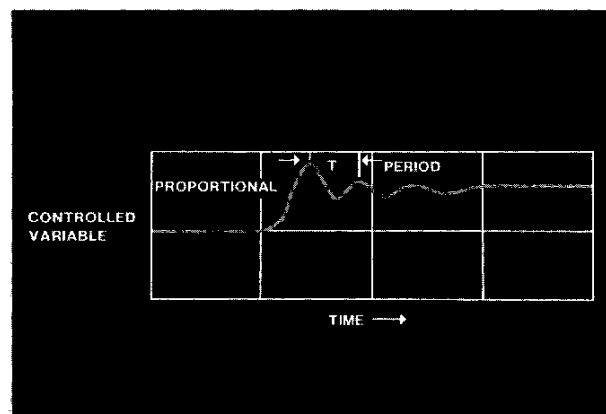


FIGURE 134 — Period of Oscillation with Proportional-Only Controller After First Tuning Step

Step 2: Set the derivative time at 0.15 times the natural period, and the integral time at 0.4 times the natural period. Observe the new period of oscillation; there should be a 25 percent decrease (see Figure 135). If the new period of oscillation is shorter than this, reduce the derivative time; if the period is longer, increase the integral time.

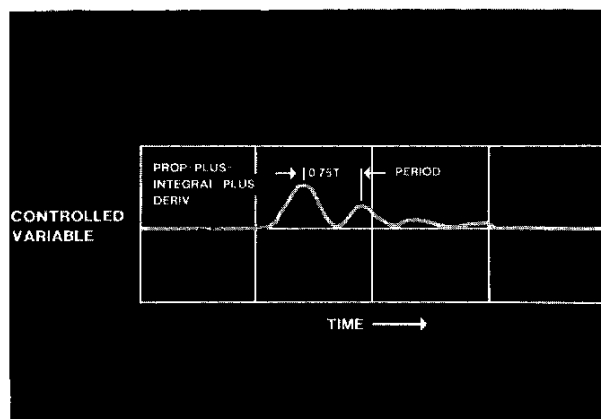


FIGURE 135 — Period of Oscillation for Correctly Tuned PID Controller After Second Tuning Step

Step 3: Finally, readjust the proportional band to achieve the desired degree of damping (the amount of correction to a process upset which, when too much or too little, shows up as either overshoot or sluggishness respectively).

When adjusting a two-mode (proportional-plus-integral) controller, a slightly different method should be used since integral mode introduces phase lag that is not counteracted by derivative. The procedure follows.

Step 1: Set the integral time of the two-mode controller at its maximum and the derivative time at its minimum, thereby providing proportional-only control, just as with the three-mode controller. Then reduce the proportional band until oscillation begins and measure this period (see Figure 134).

* *Three-mode controller* is another term for a proportional-plus-integral-plus-derivative controller (discussed in Part I of this publication).

Step 2: Set the integral time to the natural period. The period of oscillation should increase about 40 percent (ideally, 43 percent). If the period is longer than this, increase the integral time. (See Figure 136.)

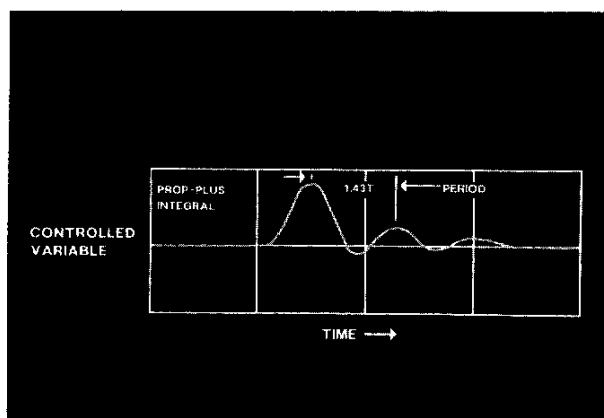


FIGURE 136 — Period of Oscillation for Correctly Tuned PI Controller After Second Tuning Step

Step 3: Finally, adjust the proportional band until the desired degree of damping is achieved. Adding integral will always increase the proportional band required for stable control.

Some consideration must be given to processes with variable dynamic characteristics. Once-through processes such as tubular heat exchangers exhibit a natural period that varies inversely with flow. In such situations, one combination of controller settings cannot be ideal for all flow rates. Integral time should be set according to the lowest anticipated flow rate, and derivative time according to the highest.

Some batch controllers, because of their mechanical arrangement, will become unstable if equal values of integral and derivative time are used. Always keep their integral time at least twice the derivative time.

Other methods of tuning include step-analysis (a method of finding time constants) and frequency-response analysis. These are beyond the scope of this publication. However, if you wish to explore these topics, refer to the books listed in the Bibliography.

PART V APPENDICES

Appendix A — Glossary of Terms

NOTE: Boldface type indicates definitions from ANSI/ISA Standard S51.1-1979*. Italicized words are defined in this Appendix. For definitions of advanced control or exclusively electronic or digital terms, please refer to the ANSI/ISA standard.

A/M STATION — See *Automatic/Manual Station*.

ABSOLUTE PRESSURE — Pressure relative to a vacuum (absolute zero, the ambient pressure of outer space). When using U.S. customary units (e.g., psi), the pressure measurement is assumed to be *gauge pressure* (compare definitions) unless specified as absolute (e.g., psia). This is done when ambient pressure (usually atmospheric pressure, approximately 14.7 psi) must be taken into account. NOTE: No distinction is made in the basic metric (SI) pressure measurement unit, kPa, between absolute and gauge pressure. However, the identifying phrase "absolute pressure" may be added.

ABSOLUTE PRESSURE TRANSMITTER — A differential pressure transmitter with process pressure applied to one side only; the other side of the sensor is evacuated to provide a zero absolute pressure reference.

ACCURACY — **In process instrumentation, degree of conformity of an indicated value to a recognized accepted standard value, or ideal value.** It is usually expressed as a percentage *deviation* (of *span*, *reading*, or *upper range value*) from the standard or ideal value. For example, force-balance transmitters generally have an accuracy of ± 0.5 percent of *span*.

ACTUATE — Cause to change position in response to a changing signal.

ACTUATOR — Hardware device which initiates corrective action by adjusting the *final control element* in response to a signal from the *controller*. It converts the *control system* signal (air pressure or current) into mechanical motion. (When a *feedback signal* is included, the device performing this function is called a servomechanism.)

AIR PURGE — A small, continuous flow of air over the interior parts of an instrument to protect them from being damaged due to dust, dirt, or fumes.

AIR SET — A device that filters and regulates the air supply of a pneumatic instrument.

AIR-TO-CLOSE/AIR-TO-OPEN — Description of control valve fail-safe action. Upon failure of air supply, an air-to-close valve (sometimes called fail-open) will lock in the open position so transmission will not be disturbed; an air-to-open (fail-closed) valve will fail in the closed position, stopping flow.

ALARM — 1) A device that responds to a signal and warns of failure, limits, or other specified conditions, by means of some audible or other indication. 2) A device with an audible or visible signal that indicates an abnormal or out-of-limit condition in a plant or *control system*. See also *Annunciator*.

AMBIENT PRESSURE — **The pressure of the medium surrounding a device.**

AMBIENT TEMPERATURE — **The temperature of the medium surrounding a device.** 1) It is assumed that the device in question is not exposed to significant radiant energy sources, and 2) if the device generates heat, that heat is considered to contribute to the temperature of the medium surrounding the device.

AMPERE — The unit of measure of *current*.

AMPLIFICATION — The ratio of the change in an output signal to the change in the input signal which produced it.

AMPLIFIER — **In process instrumentation, a device that enables an *input signal* to control power from a source independent of the signal and thus be capable of delivering an output that bears some relationship to, and is generally greater than, the *input signal*.**

* Reprinted from ANSI/ISA-S51.1-1979 with permission of the copyright holder Instrument Society of America. Complete copies of the standard are available from ISA, 67 Alexander Drive, P.O. Box 12277, Research Triangle Park, NC 27709.

AMPLITUDE — The difference between the extreme upper and lower points of a cycle. For example, in proportional control, the ideal value creates one-quarter amplitude *damping*, in which each half cycle is one-half the amplitude of the previous half cycle, thereby reducing the cycling.

AMPLITUDE RATIO — The ratio of the magnitude of a steady-state sinusoidal output relative to the input.

ANALOG — The continuous and observable representation of a variable reflecting any and all changes in value. A system whose behavior is mathematically analogous to that of some other system and therefore has the same *dynamic* equations. In the way it is viewed, analog appears like a movie, while *digital* (compare definitions) appears like a series of snapshots.

ANNUNCIATOR — A visual or audible signalling device which indicates conditions of associated circuits.

ANSI — American National Standards Institute.

ANTISURGE — A control system which protects a compressor from operating in the surge region, where it could become damaged by pulsations.

AUTO-SELECTOR CONTROL — A control system with two or more interdependent inputs, of which one, the lowest or highest, is selected to perform control by manipulating a single valve.

AUTOMATIC CONTROL SYSTEM — **A control system that operates without human intervention.**

AUTOMATIC CONTROLLER — See *Controller*.

AUTOMATIC FEEDBACK CONTROL — See *Feedback Control*.

AUTOMATIC/MANUAL STATION — **A device which enables an operator to select an automatic signal or a manual signal as the input to a controlling element. The automatic signal is normally the output of a controller while the manual signal is the output of a manually operated device. Sometimes called an auto/manual station or A/M station.**

BACKUP — An alternative means to maintain control over the process in the event of a failure in the primary control system; it can be *analog* or *digital*.

BALANCE — Matching manual mode output with automatic mode output before switching between the modes to provide "bumpless" transfer.

BATCH CONTROL — Automatic regulation of a *discontinuous process* where the nature of the process puts special demands on the control equipment.

BATCH PROCESS — A system in which the input and output flows are intermittent and periodic, sometimes called a *discontinuous process*; the converse of a *continuous process*.

BATCH SWITCH — The mechanism on a batch controller that actuates when the output goes beyond a preset limit (over 100 percent or under 0 percent of scale) to prevent the controller output from saturating or "winding up."

BATCH UNIT — See *Batch Switch*.

BELL ELEMENT — A bell-shaped device containing a diaphragm, used for accurate measurement of very low ranges of differential pressure.

BIAS — A signal or a constant added to or subtracted from an input signal to provide elevation or suppression.

BLOCK DIAGRAM — See *P & I Diagram*.

BOURDON TUBE — A mechanical pressure-measuring device that is a curved or twisted metallic tube, flattened in cross section. One end of the tube is closed, and the fluid pressure to be measured is applied to the other end. As the pressure is increased, the tube becomes more nearly circular in cross section, and tends to straighten. The motion of the free (closed) end of the tube is a measure of the internal pressure. It may be made in any one of a number of shapes, the most common being the C tube, the spiral, and the helical.

BUBBLE TUBE — See *Purge Bubbling*.

BUMPLESS TRANSFER — What occurs when transfer of control is made from the automatic to manual (or manual to automatic) modes of a controller without changing the output of the controller.

BUOYANCY TRANSMITTER — A measuring device that uses the force exerted upon a sealed container (displacer) to determine the level or density of the surrounding fluid.

C_v — A capacity rating coefficient which is defined as the number of USgpm of 60°F water which will flow through a valve at a specified opening with a pressure drop of 1 psi across the valve.

CALIBRATE — To ascertain outputs of a device corresponding to a series of values of the quantity which the device is to measure, receive, or transmit. Data so obtained are used to: 1) determine the locations at which scale graduations are to be placed; 2) adjust the output, to bring it to the desired value, within a specified tolerance; 3) ascertain the *error* by comparing the device output reading against a standard. Calibration is the process of determining the *accuracy* of a device by applying carefully quantified inputs to the device, registering the output for each input, comparing the measured output to a standard, and correcting the device, if necessary, to make the output match the standard.

CAPACITY — A measure of the maximum quantity of energy or material which can be stored in a system.

CASCADE CONTROL — Control in which the output of one *controller* is the *set point* for another *controller*.

CASE-COMPENSATED — When only the mechanism of a temperature-measuring device is corrected for the effects of ambient temperature. Compare to *Fully Compensated*.

CAVITATION (in a valve) — Vaporization in a valve when the pressure drop across it results in the pressure falling below the liquid's vapor pressure before recovering to above the vapor pressure. The pressure recovery causes an implosion.

CLOSED LOOP — A signal path which includes a forward path, a *feedback* path and a *summing point*, and forms a closed circuit. Compare to *Open Loop*.

COMMISSIONING — For a new installation, the process of verifying system and instrument operation including starting critical processes and *tuning* instruments as needed to suit the process.

COMPUTER — A general term for a piece of equipment that accepts one or more standard input signals, modifies and/or combines them in some predetermined mathematical way, and transmits the resultant output to a receiver. Computers can range from small, dedicated (i.e., single-purpose) units like the ones discussed in this publication to large units that supervise many loops and use the information to prepare reports.

CONDUCTIVITY (Electrolytic) — An analytical measurement of the activity (or concentration) of ions present in a fluid.

CONFORMITY — Of a curve, the closeness to which it approximates a specified curve (e.g., logarithmic, parabolic, cubic, etc.). **NOTES:** 1) It is usually measured in terms of non-conformity and expressed as conformity (e.g., the maximum *deviation* between an average curve and a specified curve). 2) As a performance specification, it is assumed to mean independent conformity (the maximum *deviation* of the calibration curve [the average of upscale and downscale values] from a specified characteristic curve [a graph which shows the ideal values at steady state] so positioned as to minimize the maximum *deviation*), unless otherwise specified.

CONSISTENCY — As used in the Pulp & Paper Industry, a percentage by weight of dry fibrous material in any combination of stock (fibrous material plus additives) in water.

CONTINUOUS PROCESS — A process in which, for an extended period of time, uninterrupted flows of fluids enter and products leave a system; the converse of a *batch process*. Compare to *Discontinuous Process*.

CONTROL ACTION — Of a *controller* or of a *controlling system*, the nature of the change of the output effected by the input. See *Direct Action* and *Reverse Action* for the relationship between input and output. See also *Control Mode*.

CONTROL LOOP — A group of interconnected instruments which measures the process variable, compares that value to a predetermined desired value, and applies to the process variable any change necessary to make the process value match the desired value. See also *Open Loop* and *Closed Loop*.

CONTROL MODE — A specific type of *control action*. See separate definitions of *Proportional Control*, *Integral Control*, *Derivative Control*, *PI Control*, *PID Control*, *Batch Control*, *Auto-Selector Control*, *Ratio Control*, *Cascade Control*, *Feedforward Control*, and *Differential Gap Control* types of control action.

CONTROL POINT — The value at which the controlled system or process settles out or stabilizes. It may or may not agree with the *set point* (instruction) applied to the *controller*.

CONTROL SYSTEM — A system in which deliberate guidance or manipulation is used to achieve a prescribed value of a variable.

CONTROL VALVE — A final controlling element, through which fluid passes, which adjusts the size of flow passage as directed by a signal from a *controller* to modify the rate of flow of the fluid.

CONTROLLED VARIABLE — A directly controlled variable in a control loop is the variable the value of which is sensed to originate a *feedback signal*. An indirectly controlled variable is one which does not originate a *feedback signal* but which is related to, and influenced by, the directly controlled variable. The level, quantity, or other condition in the process which is to be directly measured and controlled. The amount of energy or material coming out of a process. Compare to *Manipulated Variable*.

CONTROLLER — A device which operates automatically to regulate a *controlled variable*. This maintains a selected variable within prescribed limits.

CONTROLLING SYSTEM — That portion of a feedback control system which compares a *controlled variable* to a set point and adjusts a *manipulated variable* as a function of the difference. It includes the reference input elements, summing point, forward and final controlling elements, and feedback elements including sensor.

CONVERTER — See *Signal Converter*.

CURRENT (Electric) — The measure of energy flow between points of unequal voltage (compare definitions) in amperes (A). The most common range of transmission signal for electrical process control instruments is 4 to 20 mA dc.

CYCLING — A periodic change in the *controlled variable*, usually resulting in equal excursions above and below the *control point*; oscillation.

DAMPING — (noun) **The progressive reduction or suppression of oscillation in a device or system.** Damping affects the time required for a system to stabilize after an abrupt process upset; a system adjusted to respond as fast as possible without overshooting is said to be "critically damped," one that overshoots is "underdamped," and one that is slower than critical is "overdamped." Damping can be affected by properties of the measured medium (e.g., viscosity or magnetism).

DEAD BAND — In process instrumentation, the *range* through which an *input signal* may be varied, upon reversal of direction, without initiating observable change in *output signal*. Dead band is usually expressed in percent of *span*.

DEAD TIME — The interval of time between initiation of an input change or stimulus and the start of the resulting observable response.

DEMAND — The requirement made by the control system or the operator on the process. Compare to *Load*.

DENSITY — The unit of measurement properly called mass per unit volume. It is important in process plants as an inference of composition, of concentration of chemicals in a solution, or of concentration of solids in suspension.

DERIVATIVE CONTROL ACTION — In process instrumentation, *control action* in which the *output* is proportional to the rate of change of the *input*.

DEVIATION — Any departure from a desired value or expected value or pattern. Compare to *Error*.

DIAGNOSTIC MAINTENANCE — See *Fault Location*.

DIFFERENTIAL GAP CONTROL — Control in which the *output* of a *controller* remains at a maximum or minimum value until the *controlled variable* crosses a band or gap, causing the output to reverse. The *controlled variable* must then cross the gap in the opposite direction before the *output* is restored to its original condition.

DIFFERENTIAL PRESSURE (d/p) — Measurement of one pressure with respect to another pressure. A typical example is measuring the pressures of a flowing fluid on each side of an obstruction in a closed pipe. See also *Head*, *Primary Element*, *Secondary Device*.

DIGITAL — Representation of system variables by a limited number of discrete values. Compare to *Analog*.

DIRECT ACTION — An increasing *input* to an instrument producing an increasing *output*. Compare to *Reverse Action*.

DIRECTLY CONTROLLED VARIABLE — See *Controlled Variable*.

DISCONTINUOUS PROCESS — Any process in which part of the system does and should shut down completely for a period of time before resuming operation. Compare to *Continuous Process*.

DISPLACER — See *Buoyancy Transmitter*.

DRIFT — An undesired change in *output* over a period of time, which change is unrelated to the *input*, *environment*, or *load*.

DRY LEG — A long process connection to a differential pressure transmitter filled with gas. In liquid level measurement, the transmitter is near the bottom of the tank with one process connection open to the liquid; the other process connection is open to the gas above the liquid at the top of the tank. Compare to *Wet Leg*.

DUAL CONTROLLER — A device that can record and control two independent variables with two control mechanisms.

DUPLEX CONTROLLER — A device that can record and control one variable with two control mechanisms.

DUST-IGNITION-PROOF — A term used to express the safety of a product for use in specific industrial atmospheres.

DYNAMIC — Referring to something (response, calculations) that changes as a function of time. Compare to *Static* and *Steady-State*.

DYNAMIC COMPENSATOR — A device provided to counteract sources of *error* due to changing operating conditions. A device consisting of *lead*, *lag*, *impulse*, and/or *dead time* functions, provided to balance *control action* against the *dynamic response* of the process being controlled.

ELECTROMOTIVE FORCE (emf) — An electrical signal generated in a closed circuit of two dissimilar metals when their junctions are at different temperatures (thermocouple construction).

ELEVATED-ZERO RANGE — A *range* in which the zero value of the *measured variable*, *measured signal*, etc. is greater than the *lower range value*. Note: The zero may be between the lower and upper range values, at the *upper range value*, or above the upper range value. Examples of a 100 psi range with elevated zero are -50 to +50 psi, -100 to 0 psi, and -150 to -50 psi, respectively. Compare to *Suppressed-Zero Range*.

emf — See *Electromotive Force*.

ENVIRONMENTAL AREA — A basic qualified location in a plant with specified environmental conditions dependent on severity. Note: Environmental areas include air conditioned areas; control room areas, heated and/or cooled; sheltered areas (process facilities); outdoor areas (remote field sites).

EQUAL PERCENTAGE CHARACTERISTIC — In theory, for equal increments of valve opening, an equal percent change in flow will occur.

EQUILIBRIUM — The condition of a system when all inputs and outputs (*supply* and *demand*) have stabilized and are in balance.

ERROR — In process instrumentation, the algebraic difference between the indication and the *ideal value* of the *measured signal*. Note: a positive error denotes that the indication of the instrument is greater than the *ideal value*. It is often expressed as a percentage of *span* or *upper range value*. Error has several aspects which process control instrumentation corrects: direction, magnitude, duration, and rate of change. Compare to *Deviation*.

EVACUATED VESSEL — A closed tank from which all gas has been removed.

EXOTHERMIC — Characterized by evolution of heat.

EXPLOSION-PROOF — A term used to qualify the safety of a product for use in specific industrial atmospheres.

FAULT LOCATION — The process of diagnosing, locating, and correcting malfunctions.

FEEDBACK SIGNAL — In process instrumentation, the return signal which results from a measurement of the *directly controlled variable*.

FEEDBACK CONTROL — Control in which a *measured variable* is compared to its desired value to produce an actuating *error* signal which is acted upon in such a way as to reduce the magnitude of the *error*.

FEEDFORWARD CONTROL (Predictive Control) — Control in which information concerning one or more conditions that can disturb the *controlled variable* is converted, outside of any *feedback loop*, into corrective action to minimize *deviations* of the *controlled variable*. Note: The use of feedforward control does not change system *stability* because it is not part of the *feedback loop* which determines the *stability* characteristics.

FIELD-MOUNTED — An instrument mounted in the plant rather than in the control room — e.g., a sensor mounted on a pipe or near one.

FILLED THERMAL SYSTEM — An assembly consisting of a bulb, capillary tube, and *Bourdon tube* (or derived form of direct-measuring element), containing a temperature-responsive gas or liquid.

FINAL CONTROL ELEMENT (Final Controlling Element) — The forward controlling element which directly changes the value of the *manipulated variable*. Component of a *control system* (such as a valve) which directly changes the value of the *manipulated variable*.

FLANGE — A rim integral to an instrument used for mounting it to a tank or to a pipe with flanges on it.

FLAPPER-NOZZLE — The basic error detection assembly in pneumatic controllers and transmitters.

FLASHING — Vaporization of volatile liquids by either heat or vacuum. What occurs when a fluid enters a valve as a liquid and exits as a vapor.

FLOWMETER — **A device that measures the rate of flow or quantity of a moving fluid in an open or closed conduit. It usually consists of both a *primary* and a *secondary device*.**

FLUID — Anything capable of flowing; a substance tending to flow or conform to the outline of its container. Liquid, gas, or steam.

FORCE-BALANCE MECHANISM — A method of transforming a direct measurement to a signal. A force produced by a sensor creates an imbalance which is amplified and converted to a restoring or balancing force which is related to the output signal.

FULLY COMPENSATED — When both the motion mechanism and the capillary of a temperature-measuring device are corrected for the effects of ambient temperature. Compare to *Case-Compensated*.

GAIN (Magnitude Ratio) — **For a linear system or element, the ratio of the magnitude (*amplitude*) of a *steady-state* sinusoidal output relative to the causal input; the length of a phasor from the origin to a point of the transfer locus in a complex plane. In non-linear systems, gains are often *amplitude-dependent*. The ratio of change in *output* divided by the change in *input* that caused it. The gain of a loop is the product of the gains of all the elements in the loop.**

GALVANOMETRIC — Capable of detecting or measuring a small electric current by movements of a magnetic needle or of a coil in a magnetic field.

GAUGE (GAGE) PRESSURE — Measurement of one pressure with respect to *ambient* atmospheric pressure. It does not correct for atmospheric pressure, which changes with elevation. Compare to *Absolute Pressure*.

HAIR HYGROMETER — A direct-measuring instrument indicating relative humidity which has a sensing element fabricated from specially treated human hairs that change in length with humidity.

HEAD (Hydrostatic Head) — Pressure resulting from gravitational forces on liquids; measured in terms of the depth below a surface of the liquid.

HEAD METER — A *differential pressure* transmitter for measuring liquid level.

HEAT — Energy flowing between bodies of different temperatures.

HIGH-SELECTOR — A device that compares two values of *input* and allows the higher value to be the *output*. Compare to *Low-Selector*.

HUMIDITY — The amount of moisture in gases expressed as relative humidity, absolute humidity, or dew point.

HYDRAULIC RESONANCE — Waves that develop in a tank due to turbulence and differential pressure that interfere with accurate measurement of the level in the tank.

HYSTERESIS — **That property of an element evidenced by the dependence of the value of the output, for a given excursion of the input, upon the history of prior excursions and the direction of the current traverse. Note: it is usually determined by subtracting the value of *dead band* from the maximum measured separation between upscale going and downscale going indications of the measured variable (during a full range traverse, unless otherwise specified) after *transients* have decayed. Some small reversal of output may be expected for any small reversal of input; this distinguishes hysteresis from *dead band*. The difference between upscale and downscale results in instrument response when subjected to the same input approached from the opposite direction. A nonlinearity usually attributed to flexibility and loose fits in linkage and to backlash in gear trains.**

I/P — Abbreviation representing *current-to-pneumatic* signal conversion.

IDEAL VALUE — **The value of the indication, output, or ultimately controlled variable of an idealized device or system. It is assumed that an ideal value can always be defined even though it may be impossible to achieve.**

IMMISCIBLE — Incapable of being mixed to attain homogeneity.

IMPULSE FUNCTION — A form of *lead* action having zero *steady-state* gain.

INDICATOR — A measuring instrument in which only the present value of the *measured variable* is visually indicated. A device which displays information; technically, it takes information that is in a form which cannot be directly interpreted by humans and converts it into a form which can be directly interpreted by humans.

INDIRECTLY CONTROLLED VARIABLE — See *Controlled Variable*.

INPUT SIGNAL — A signal applied to a device, element, or system. Incoming signal to measuring instrument, control units, or system.

INSTRUMENT — In process measurement and control, this term is used broadly to describe any device that performs a measuring, controlling, recording, or computing function.

INSTRUMENTATION — A collection of *instruments* or their application for the purpose of observation, measurement or control.

INTEGRAL CONTROL (previously called Reset) — **Control action in which the output is proportional to the time integral of the input; i.e., the rate of change of output is proportional to the input.** Integral control is used most often in combination with proportional control.

INTEGRAL TIME (previously called Reset Time) — The time required to produce a change in controller output equal to the change in *error* input. The proportionality constant in the equation relating the rate of change of controller output to the *error* for integral control.

INTEGRAL WINDUP (previously called Reset Windup) — The overcharging, in the presence of a continuous *error*, of the integral capacitor (bellows, in a pneumatic controller) which must discharge through a long *time constant* and which prevents a quick return to the desired *control point*.

INTEGRATOR — A form of computer that sums the flow of a fluid over time through the instrument and provides a digital readout of the current sum.

INTRINSIC SAFETY — A term applied to electrical equipment and wiring that have been made incapable of releasing sufficient electrical or thermal energy — even under abnormal conditions — to cause ignition of a specific hazardous atmospheric mix in its most easily ignited concentration.

ISA — The Instrument Society of America.

LAG — Any delay from instantaneously complete response to an input signal. It causes the output of a device to fall behind its input signal, exhibiting a *phase lag* which increases with increasing frequency.

LAMINAR FLOW — Flow in which a generally viscous, slow-moving fluid separates into layers that do not mix.

LAPLACE OPERATOR — An algebraic substitute for the differential operator (d/dt) or time derivative in a differential equation.

LEAD — Action which causes the output of a device to advance ahead of its input signal, exhibiting a phase lead and gain which increase with increasing frequency of the input signal.

LINEAR — A straight-line relationship between one variable and another. When used to describe the output of an instrument, it means that the output is proportional to the input. In a valve, it is the characteristic where flow is directly proportional to valve travel. Compare to *Nonlinear*.

LINEARITY — The closeness to which a curve approximates a straight line. It relates to a calibration curve (the average of upscale and downscale readings) and is usually expressed as the *deviation* between the curve and a straight line.

LIVE ZERO — A signal representing an input of zero value coming from the process (e.g., 3 psi from an instrument with a 3 to 15 psi output) as contrasted with a signal representing no input at all (e.g., 0 psi).

LOAD (Process Load) — The level of material, force, torque, energy, power, or other variable applied to or removed from a process. Compare to *Demand*.

LOAD CHANGE — A disturbance in *load* which moves the controlled variable and causes the controller to reposition its output to restore zero deviation, without any change in set point.

LOAD RESPONSE — The way in which a controlled process behaves after a change in the process load introduces an error into the system.

LOCALLY MOUNTED — Applies to an instrument mounted next to the process it is measuring.

LOOP — See both *Closed Loop* and *Open Loop*.

LOOP DIAGRAM — A representation of the various parts of a control loop, showing interconnected elements.

LOOP TUNING — Adjusting the instrumentation in the loop (primarily the controller) so that the process can achieve a balance between supply and demand as quickly and efficiently as possible.

LOW-SELECTOR — A device that compares two values of *input* and allows the lower value to be the *output*. Compare to *High-Selector*.

LOWER RANGE VALUE — **The lowest value of the measured variable that a device is adjusted to measure.** For example, if a *calibrated* range is 0 to 100 psi, 0 psi is the lower range value.

mA dc — See *Current*.

MAGNITUDE RATIO — See *Gain*.

MANIPULATED VARIABLE — **A quantity or condition which is varied as a function of the actuating error signal so as to change the value of the directly controlled variable.**

MANUAL CONTROL — A mode of control in which the controller output is adjusted by the operator (the adjustment can be made directly at the controller or remotely). Usually provided as *backup* in case the automatic mode of control fails for any reason.

MEASURED SIGNAL — **The electrical, mechanical, pneumatic or other variable applied to the input of a device. It is the analog of the measured variable produced by a transducer (when such is used). For example, in a flowmeter the measured signal may be a differential pressure which is the analog of the rate of flow through the orifice.**

MEASURED VARIABLE — **A quantity, property, or condition which is measured**, for example, temperature, pressure, rate of flow, level, or speed.

MEASUREMENT — The present value of a variable such as flow rate, pressure, level, or temperature.

MULTICAPACITY PROCESS — A process consisting of several storage elements connected in series such as a distillation column containing several equilibrium stages or trays.

NEWTONIAN FLUID — A *fluid* wherein the ratio of flow to force applied is constant. The basic reference condition for all flowmeters. NOTE: all gases and most liquids are Newtonian fluids.

NOISE — **An unwanted component of a signal or variable.** It causes a fluctuation in a signal that tends to obscure its information content. For example, flow is considered noisy since turbulence creates high-frequency variations in addition to the flow-rate variations. See also *Radio-Frequency Interference*.

NOMINAL VALUE — The value assigned for the purpose of convenient designation but existing in name only; the stated or specified value as opposed to the actual value.

NON-NEWTONIAN FLUID — A *fluid* whose *viscosity* is not constant and, therefore, the pattern of its *velocity profile* is not predictable.

NONLINEAR — A relationship between two or more variables which cannot be described as a straight line. When used to describe the output of an instrument, it means that the output is of a different magnitude than the input (e.g., a square-root relationship). Compare to *Linear*.

NORMAL OPERATION CONDITIONS — The range of environmental circumstances (e.g., *ambient* pressure, *ambient* temperature, electromagnetic fields, gravitational force, inclination, power supply variation, radiation, shock, vibration) within which an instrument is designed to operate. Documentation for an instrument usually indicates the operating limits (the extremes allowable) within which it can maintain a specified degree of *accuracy*.

OFFSET — **The steady-state deviation when the set point is fixed.** Offset develops in a control loop having no integral action, when the controller output must move to a value other than its bias point to balance the process load.

ON/OFF CONTROL — A mode of control in which the *output* (usually expressed in terms of valve position) is either fully on (the valve fully open) or fully off (the valve fully closed).

OPEN-CHANNEL DEVICE — Primarily used for handling water and waste, it is a flow-measuring device comprising a *primary device* called a weir or a flume (a plate with a notch in it or a restriction in a channel, respectively) and a *secondary device* (see definition) to produce a signal related to flow rate.

OPEN LOOP — **A signal path without feedback.**

OPEN TANK — A vessel open to ambient air. Compare to *Evacuated Vessel*.

OPTICAL (Radiation) PYROMETER — Device which utilizes an optical system to focus energy radiated from a body onto a sensing system. It is used to measure high temperatures where contact with the object is impossible.

OPTIMIZATION (Optimizing Control) — **Control that automatically seeks and maintains the most advantageous value of a specified variable, rather than maintaining it at one set value.**

ORIFICE PLATE — The most common *primary device* for measuring fluid flow; a circular disk with a hole in it (several shapes and placements of the hole are possible) inserted in a pipe between two *flanges*.

ORP — Abbreviation for oxidation-reduction potential. A type of analytical measurement that determines the oxidizing or reducing state of a chemical reaction; it applies to any chemical reaction in which electrons are transferred.

OSCILLATION — See *Cycling*.

OUTPUT SIGNAL — The signal provided by an instrument; for example, the signal that the *controller* delivers to the valve operator is the controller output.

OUTPUT SATURATION — The condition where the output of a device has reached its maximum or minimum possible value.

OVERRANGE — **Any excess value of the input signal above its upper range value or below its lower range value.** The overrange limit is **the maximum input that can be applied to a device without causing damage or permanent change in performance.**

OVERRANGE CAPABILITY — The ability of an instrument to handle infrequently occurring signals beyond its specified range.

OVERSHOOT — To exceed the desired or *steady-state* value when attempting to correct for an *input*; the amount by which the *output* exceeds the desired value.

P & I DIAGRAM — A drawing of a plant showing, in symbolic form, all piping and instrumentation.

P/I — Abbreviation representing pneumatic-to-current signal conversion.

PERIOD — The interval of time required for a cyclic motion or phenomenon to complete a cycle.

pH — An analytical measurement of the activity of hydrogen ions in a solution; a measure of the degree of acidity or alkalinity of an aqueous solution.

PHASE LAG — A delay between the input and output of a device expressed in the portion of a complete sinusoidal cycle as it passes through the device.

PI CONTROL (Proportional-plus-Integral Control) — **Control action in which the output is proportional to a linear combination of the input and the time integral of the input.**

PID CONTROL (Proportional-plus-Integral-plus-Derivative Control) — **Control action in which the output is the linear combination of the input, the time integral of the input, and the time rate-of-change of the input (i.e., the linear combination of the effects of proportional, integral, and derivative control modes).**

PNEUMATIC SIGNAL AMPLIFIER — See *Relay*.

POSITION-TYPE MECHANISM — A series of linkages connecting a direct sensor to an indicator, transmitter, or recorder. Also called motion mechanism.

POSITIONER — A device used to position an *actuator* with regard to a signal. The positioner compares the input signal with a mechanical *feedback* link from the *actuator*, then produces the force necessary to move the *actuator* output until the mechanical output position *feedback* corresponds with the pneumatic signal value.

PRESSURE — Force exerted over a surface divided by its area. It can be specified as *ambient pressure* (the pressure of the medium surrounding a device), *maximum working pressure* or *MWP* (the maximum total pressure permissible in a device under any circumstances during operation, at a specified temperature — the designed safe limit for regular use), *operating pressure* (the actual pressure at which a device operates under normal conditions), *process pressure* (the pressure at a specified point in the process medium), *rupture pressure* (the pressure, determined by a test, at which a device will burst), *static pressure* (the steady-state pressure applied to a device; in the case of a differential pressure device, the process pressure applied equally to both connections), or *surge pressure* (the operating pressure plus the increment above operating pressure to which a device may be subjected for a very short time during pump starts, valve closings, etc.).

PRESSURE SEAL — A mechanical device that isolates a corrosive or viscous product, or product that tends to solidify, from the measuring element. It has a diaphragm which contacts the product directly and tubing filled with fluid on the other side of the diaphragm which conveys the diaphragm pressure to the indicator or transmitter.

PRESSURIZED VESSEL — A closed vessel which may operate at pressures above atmospheric.

PRIMARY ELEMENT — **The system element that quantitatively converts the *measured variable* energy into a form suitable for measurement. Note: for transmitters not used with external primary elements, such as magnetic flowmeters, the sensing portion is the primary element.**

PROCESS — (verb) To subject to a special process or treatment (as in the course of manufacture). (noun) The equipment for which supply and demand must be balanced — the system under control, excluding the equipment that does the controlling. See *Process Control*.

PROCESS CONTROL — **The regulation or manipulation of variables influencing the conduct of a *process* in such a way as to obtain a product of desired quality and quantity in an efficient manner.**

PROCESS LOAD — See *Load*.

PROCESS UPSET — Any type of disturbance that causes a control variable to depart from set point.

PROCESS-WETTED MATERIALS — Any part of a sensor or other instrument which comes in direct contact with the process fluid. Frequently, these materials require special characteristics (such as corrosion resistance) which are not needed by the rest of the instrument.

PROPORTIONAL BAND — **The change in input required to produce a full range change in the output of a *controller* due to *proportional control action*. The reciprocal of *gain* expressed as a percentage.**

PROPORTIONAL CONTROL — ***Control action* in which there is a continuous linear relation between the *output* and the *input*.**

PURGE BUBBLING — A form of pressure measurement which measures the pressure required to force a gas into a liquid at a point beneath the surface (changes in level cause the air pressure in the bubble tube to increase or decrease).

RADIO-FREQUENCY INTERFERENCE (RFI) — **A special case of electromagnetic interference from radio transmitters causing a spurious effect in the circuits or elements of a device.**

RANGE — **The region between the limits within which a quantity is measured, received, or transmitted, expressed by stating the lower and upper range values. Note: Unless otherwise modified, input range is implied. See *Elevated-Zero Range* and *Suppressed-Zero Range*.**

RANGEABILITY — The ratio of maximum controllable flow (usually the upper scale value) of an instrument to minimum controllable flow (e.g., 10:1 or 50:1); it is a way of expressing the ability of an instrument to accommodate a wide variety of conditions. Rangeability is particularly important for valves.

RATE ACTION — See *Derivative Control*.

RATIO CONTROL — A *control mode* which maintains an easily controlled variable in a preset ratio to a second, often wildly fluctuating, variable for purposes of maintaining satisfactory control.

REDOX (Reduction-Oxidation Potential) — See *ORP*.

RELAY (Pneumatic Amplifier) — A device which converts a small change in the input signal to a large change in the output signal. Typically, a 7 kPa (1 psi) change in input will produce approximately an 80 kPa (12 psi) change in output.

RELIABILITY — **The probability that a device will perform its objective adequately, for the period of time specified, under the operating conditions specified.**

REPEATABILITY — **The closeness of agreement among a number of consecutive measurements of the output for the same value of the input under the same operating conditions, approaching from the same direction, for full range traverses. It does not include *hysteresis*. The ability of an instrument to produce exactly the same result every time it is subjected to the same conditions. Usually expressed in percent of *span*.**

REPEATER — A mechanical means of transferring a pressure measurement made by a process-contacted diaphragm through a length of piping to one side of a differential pressure sensor. Generally used in closed tank, *dry leg* applications for the upper tank measurement, the repeater's diaphragm protects the dry leg from fluids that might solidify or corrode.

RESET CONTROL — See *Integral Control*.

RESET WINDUP — See *Integral Windup*.

RESISTANCE (Fluid) — A restriction to flow that accounts for the dissipation of energy and limits flow. For example, flow from a water faucet is limited by the resistance of the faucet, which is a function of the size of the opening.

RESISTANCE TEMPERATURE DETECTOR (RTD) — A sensor consisting of wire which changes its resistance as a function of temperature. The resistance-temperature relationship is well-established so that a specific resistance can be interpreted as a predictable measurement of temperature.

RESPONSE TIME — The interval required for an input to result in a change in output under normal operating conditions. Compare to *Time Response*.

REVERSE ACTION — An increasing input to an instrument producing a decreasing output. Compare to *Direct Action*.

RFI — See *Radio-Frequency Interference*.

ROTAMETER — See *Variable Area Meter*.

RTD — See *Resistance Temperature Detector*.

SAMA — The Scientific Apparatus Makers' Association.

SECONDARY DEVICE — A device that converts the effect produced by a *primary element* into a standard signal.

SELECTIVE CONTROL — See *Auto-Selector Control*.

SET POINT — An input variable which sets the desired value of the *controlled variable*. It may be manually set, automatically set, or programmed, and it is expressed in the same units as the *controlled variable*. Sometimes called the reference point.

SI — The common abbreviation for *Système International d'Unités* (International System of Units), which is being accepted and implemented by industries worldwide. Its implementation is overseen by the International Organization for Standardization (ISO) to ensure consistency of use.

SIGNAL — In process instrumentation, a physical variable, one or more parameters of which carry information about another variable (which the signal represents). Information in the form of a transmission medium (pneumatic pressure, electric current, or mechanical position) that is conveyed from one control loop component to another.

SIGNAL CONVERTER — A *transducer* which converts one transmission signal to another.

SPAN — The algebraic difference between the *upper* and *lower range values*. For example, a range of 0 to 150°C has a 150°C span. The example is the span of a zero-based range; the lower range value may be above or below zero. See also *Range*, *Elevated-Zero Range*, and *Suppressed-Zero Range*.

SPAN ADJUSTMENT — Means provided in an instrument to change the slope of the input-output curve. For example, if the output of a transmitter (for a specified level of input) is supposed to be 0 to 100 psi but actually is 0 to 105 psi, the instrument can be adjusted to produce the desired range. Compare to *Zero Adjustment*.

SPECIFIC GRAVITY — The quantification of relative density of a fluid. The ratio of the density of a liquid at a particular temperature to the density of water at a particular temperature. Also the molecular weight of a gas divided by the molecular weight of air.

SQUARE-ROOT EXTRACTOR — A device whose output is the square root of its input signal, both expressed over the range of 0 to 1.0.

STABILITY — The property of a system for which any bounded (limited) input signal results in a bounded output signal.

STATIC — Referring to something (primarily behavior) which is either not a function of time or which takes place over a sufficient length of time that dynamic changes are of minor importance. Compare to *Dynamic*.

STEADY-STATE — A characteristic of a condition, such as value, rate, periodicity, or amplitude, exhibiting only negligible change over an arbitrary long period of time. It may describe a condition in which some characteristics are *static*, others *dynamic*. The net (long-term) output of a device at some point after it has responded to a disturbance by a temporary change in the input. Compare to *Dynamic*.

STEP CHANGE — A change from one level to another in virtually zero time.

SUPERVISORY CONTROL — Control in which the control loops operate independently subject to intermittent corrective action such as set-point changes from an external source.

SUPPRESSED-ZERO RANGE — A *range in which the zero value of the measured variable is less than the lower range value. (Zero does not appear on the scale.)* An example of a 100 psi range that is suppressed would be 20 to 120 psi. Compare to *Elevated-Zero Range*.

SYSTEM — Without modifiers, the term "system" generally encompasses all components, including process, measurement, control, operation, and final actuation, which provide deliberate guidance or manipulation to achieve a prescribed value of a variable. With modifiers, terms like "controlled system," "controlling system," "directly controlled system," and "indirectly controlled system," refer to specific parts of the overall system.

TC — See *Thermocouple*.

TACHOMETER — A device which senses rotary motion, usually of a shaft, and produces a signal proportional to the rotary speed.

TARGET FLOWMETER — A flow-sensing transmitter which combines a *primary device* (a solid disk suspended in a pipe — called a target) and a *secondary device*. As flow passes around the disk, the force exerted by the flowing fluid on the disk produces a flow-related (square) signal.

TEMPERATURE — The change in the quantifiable degree of hotness or coldness of a system. It can be further specified as *ambient temperature* (the temperature of the medium surrounding a device) or *process temperature* (the temperature of the process medium at the sensing element).

THERMISTOR — A temperature-measuring device, similar to a *resistance temperature detector*, made of heat-treated metallic oxide.

THERMOCOUPLE (TC) — A device for measuring temperature in which two electrical conductors of dissimilar metals (as copper and iron) are joined at the point where heat is to be applied and the free ends are connected to an electrical measuring instrument (as an ammeter) which by registering the amount of thermoelectric current being produced at the juncture of the dissimilar conductors indicates the temperature at that point.

THERMOWELL (Thermometer Well) — Long, thin metal case into which a temperature sensor fits to protect the sensor from direct exposure to process conditions that might damage it (e.g., corrosive materials, fluids with suspended abrasive solids, high temperatures, locations exposed to direct sunlight). Some can also be used in sanitary processes.

THREE-MODE CONTROL — See *PID Control*.

THROTTLING CONTROL — See *Proportional Control*.

TIME CONSTANT — The value T in an exponential response term $A \exp(-t/T)$ or in one of the transform factors $1 + sT$, $1 + j\omega T$, $1/(1 + sT)$, $1/(1 + j\omega T)$ where:

s = complex variable

t = time

T = time constant

$j = \sqrt{-1}$

ω = angular velocity, radians per unit time.

Note: For the output of a first-order system forced by a step or an impulse, T is the time required to complete 63.2% of the total rise or decay; at any instant during the process, T is the quotient of the instantaneous rate of change divided into the change still to be completed. In higher order systems, there is a time constant for each of the first-order components of the process.

TIME RESPONSE — An output expressed as a function of time, resulting from the application of a specified input under specified operating conditions. Compare to *Response Time*.

TOTALIZER — A device which accumulates incremental measurements of flow, providing an indication of total flow to that point.

TRANSDUCER — An element or device which receives information in the form of one quantity and converts it to information in the form of the same or another quantity. For example, a thermocouple converts temperature into millivoltage. **Note:** This is a general term and definition and as used here applies to specific classes of devices such as primary element, signal transducer (converter), and transmitter.

TRANSIENT — The behavior of a variable during transition between two *steady-states*.

TROUBLESHOOTING — See *Fault Location*.

TUNING — Adjusting a controller to respond as desired to a specific process.

TURBULENT FLOW — Flow in which a not very viscous, generally fast-moving fluid moves in an agitated, complex, and random way.

TWO-POSITION CONTROL — See *On/Off Control*.

UPPER RANGE VALUE — The highest value of the *measured variable* that a device is adjusted to measure. For example, if a *calibrated* range is 0 to 100 psi, 100 psi is the upper range value.

VALVE POSITIONER — See *Positioner*.

VALVE STROKE — The distance the valve stem moves between the fully open and fully closed positions of a valve.

VARIABLE-AREA METER (Rotameter) — One of the few instruments that measures flow rate directly, by the placement of a float denser than the fluid it is to measure in a tapered tube, whose position relates directly (linearly) to flow.

VELOCITY PROFILE — A flow velocity pattern created in a pipe by the combined factors of *fluid viscosity*, texture of the pipe walls, and flow speed; it is important in determining the suitability of a flowmeter for a specific application. (Theoretically, if there were no viscosity and the pipe walls were perfectly smooth, the fluid would travel at the same speed at all points across a cross section of the pipe. In practice, any fluid travels a little slower at the pipe walls due to friction than in the middle of the pipe, creating a roughly parabolic pattern.)

VISCOSITY — The resistance of a fluid to flowing.

VOLTAGE — Electrical potential or potential difference expressed in volts.

WET LEG — A long process connection filled with liquid. Compare to *Dry Leg*.

ZERO — The point which represents no variable being transmitted (0 percent of the upper range value). Frequently, the lower range value is at zero (for example, a range of 0 to 100 psi). See *Elevated-Zero Range*, *Suppressed-Zero Range*, and *Zero Adjustment*.

ZERO ADJUSTMENT — Means provided in an *instrument* to produce a parallel shift of the *input-output curve*. For example, if the output of a transmitter (for a specified level of input) is supposed to be 0 to 100 psi but actually is 2 to 102 psi, the instrument can be adjusted to produce the desired output. Compare to *Span Adjustment*.

ZERO ELEVATION — For an *elevated-zero range*, the amount the *measured variable* zero is above the *lower range value*. It may be expressed either in units of the *measured variable* or in percent of *span*.

ZERO SUPPRESSION — For a *suppressed-zero range*, the amount the *measured variable* zero is below the *lower range value*. It may be expressed either in units of the *measured variable* or in percent of *span*.



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APPENDIX B — Units and Conversion Factors

The International System of Units (SI, for *Système International d'Unités*) is rapidly becoming the most commonly used in the world. It derives nearly all quantities needed in all technologies from only seven base units: the *metre* (m), for length; the *kilogram* (kg), for mass (what is usually called weight); the *second* (s), for time; the *ampere* (A), for electric current; the *Kelvin* (K), for thermodynamic temperature; the *mole* (mol), for amount of substance; and the *candela* (cd), for luminous intensity. There are also two supplementary units, the *radian* (rad), for plane angle, and the *steradian* (sr), for solid angle. More information on the properties of these units and their conversion factors can be found in documents published by ISO and others, listed in the Bibliography of this publication.

Of course, other units have been used in metric practice. However, the only proper SI units are those derived from the units listed above. The characteristic of being able to produce all needed quantities from base and supplementary units is called "coherence." For example, area is measured as length times length (square metres).

To take maximum advantage of the SI system, only base, supplementary, or derived SI units should be used. The appropriate units for quantities commonly used in pneumatic process control are listed in Table I, along with the base or supplementary units from which they are derived. Other unit derivations can be found in trade standards listed in the Bibliography.

As a logical system, SI units use terms and symbols to abbreviate numbers and show relationships between any number and its unit. For example, 1 000 000 (one million) metres is expressed as one megametre or one Mm. The most common terms and symbols are listed in Table II.

One problem with any system is the accuracy with which it can or should be applied. Any digit that is used to indicate accuracy is significant. For example, 4.001 201 has seven significant digits; all that may be required is 4.001 or even 4. In Table IV — Conversion Factors, where possible, five significant figures have been used. This represents an acceptable degree of accuracy for most process control situations.

To assist in preserving the advantage of SI as a coherent system, it is advisable to minimize the use of units from other systems. Throughout this document the preferred term for expressing each category of units has been marked with an asterisk.

It is also desirable not to mix unit symbols with unit names or abbreviations (including the name "per" and its symbol, "/"). Some examples of proper and improper usage are listed below.

PROPER USAGE

joules per kilogram
J/kg
kilometres per second
km/s
litres per minute
L/min

IMPROPER USAGE

joules/kilogram
joules/kg
kilometres/second
km/second
litres/minute
L/m (because "m"
alone means "metre")

All units in the following tables are listed in alphabetical order and are cross-referenced to commonly used units in both the U.S. customary and metric systems. Notes useful to understanding the units and their categories have been included in Table III — Categories of Units.

In some units, the preferred form may pose too great a magnitude for all applications. For example, while kilogram is the proper term for mass, a very small amount is more easily expressed in terms of grams. Similarly, kilowatts are usually used instead of watts and kilopascals instead of pascals. The expression of speed (which is an aspect of velocity, discussed later) takes this concept a little further; the proper term is metres per second, but common usage expresses traffic speeds as kilometres per hour in SI countries.

Pressure and mass are two particularly appropriate examples, since each is affected (at least very slightly) by gravity. For example, many pressure and differential pressure instruments use forms of springs as measuring elements, which measure force directly; these are called "gravity-independent." However, the pressure standards used to calibrate the springs, such as dead-weight testers which measure the force of gravity on a column of mercury or other substance of fixed mass, are often "gravity-dependent." Pneumatic systems cancel the effect of gravity when the same type of pressure standard is used for both input and output. Electronic systems have a gravity-independent output (current or voltage). They must have either a gravity-independent input or be calibrated in a way that accommodates the local gravitational force (either by incorporating a correction factor or by calibrating the pressure instrument at the location where it will be used).

The unit category mass has other problems. The U.S. customary system of units has long used the same term — pounds — to mean both units of mass (the invariant quantity of matter in a body) and weight (the force of gravity on that mass, which varies with location). The variations in the effect of gravity over the surface of the Earth are slight. However, since the U.S. customary system does not account for them, it poses an inherent problem when a high degree of accuracy is required or when related facilities are at widely dispersed locations.

Complicating the problem is the fact that force units (which more closely reflect weight) often incorporate

mass terminology (for example, pounds-force or kilograms-force). Even pressure units sometimes use mass terminology (e.g., pounds per square inch). The SI system provides the means to incorporate the effect of gravity, establish a common terminology, and distinguish pure mass from force (mass accelerated over a distance), pressure (force per unit area), density (mass per unit volume), and flow (mass per unit time). Even energy, power, and torque units are partially derived from mass, but mass is not a significant enough factor for the mass vs. force issue to be of concern. Refer to the table of proper SI units to see how they all relate.

Notes About Units

The following is general information about the unit categories and helpful hints for working with individual units.

ABSOLUTE VISCOSITY: Also called “dynamic viscosity” or just “viscosity.”

ACCELERATION: 1. “Metres per second squared” (the term preferred in SI guidelines) is also called “metres per second per second.”
2. The acceleration of gravity is about 10 m/s^2 .

ANGULAR VELOCITY: 1. The SI unit for this is defined in terms of a supplemental unit, the radian: rad/s.
2. The terms “revolutions per minute” and “revolutions per second” are properly abbreviated “r/min” and “r/s,” respectively, rather than “rpm” and “rps.”
3. This category is also called rotational frequency, primarily in specifications on rotating machinery, when the revolution per second and revolution per minute are widely used as units.

AREA: 1. The term “hectare” (abbreviated “ha”) is used as an alternative name for square hectometre and is restricted to the measurement of large land areas. Agricultural engineers use the term to relate machines to field sizes.
2. The square metre is also called “centare.”
3. Although the centimetre (cm) is rarely used to indicate length (metre or millimetre is preferred), the square centimetre (cm^2) is often used to indicate area because the interval between the square metre (m^2) and square millimetre (mm^2) is so great (1 000 000 to 1).

ENERGY: 1. This unit category includes “quantity of heat” and “work.”

2. The use of the calorie was discontinued by SI on January 1, 1978.

3. The kilowatt-hour and variations thereon (e.g., MW-h, GW-h), although not proper SI units (the joule is the proper one), are in widespread use for measurement of electric energy.

4. The units based on the electronvolt (eV, keV, MeV, and GeV) are also improperly but widely used in atomic and nuclear physics and in accelerator technology (the joule should be used).

5. The joule is equivalent to one watt-second.

FORCE: 1. The use of the kilogram-force (once widely used in Europe) was discontinued by SI on January 1, 1978.

2. The kilogram-force is also called “kilopond.”

KINEMATIC VISCOSITY: The SI unit, the square metre per second (m^2/s), is equivalent to the English unit Stoke (St) and the SI unit square millimetre per second (mm^2/s) is equivalent to the English unit centiStoke (cSt).

LENGTH: 1. The smaller units in this category (like the metre and millimetre) are easy to learn because they can be related to items contacted every day. For example, a U. S. dime is about one millimetre thick, a U. S. quarter is about 25 millimetres wide, and the height of most home doorways is about two metres. However, the kilometre is harder to visualize and is therefore easier to learn by memorization. Following is a list of common values.

miles or mph	km or km/h
10	16
25	40
50	80
55	90
62	100
75	120
100	160

2. The millimetre is used all over the world on industrial engineering drawings.
3. The use of centimetre is generally restricted to body measurements, clothing sizes, and textile weights.
4. The micrometre (sometimes called "micron") is the preferred unit to express surface finish.

- MASS:** 1. The kilogram and gram will generally replace the use of the pound and ounce, respectively.
2. Two aspirin, an American dollar bill, and one paper clip each weigh about one gram.
 3. A kilogram is the weight of one litre of water.
 4. The alternate name of "tonne" is "metric ton." A tonne is equal to one megagram.
 5. A load-supporting rating (e.g., floor load) should be expressed in kilograms.
 6. This unit category is also called "weight."

MASS PER UNIT TIME: This unit category is also called "flow" and "mass flow."

- MASS PER UNIT VOLUME:** 1. This unit category is also called "density," "mass density," and "mass capacity."
2. One part per million is equal to one milligram per litre or one gram per cubic metre, referenced as "by weight in water" at a specified temperature.

- PLANE ANGLES:** 1. The radian, a supplementary SI unit, is the proper unit for this category. The decimalized degree (defined as $[\pi \div 180]$ rad) is not proper but is widely used. Although the minute and the second are still widely accepted, their use is discouraged because they require an extra conversion step.
2. The plane angle is also called just "angle."

- POWER:** 1. There are several types of horsepower. The one usually assumed is electric horsepower (unless otherwise stated).
2. Boiler horsepower is primarily used to rate the size of small industrial boilers.
 3. The use of the calorie was discontinued by SI on January 1, 1978.
 4. Power is also called "heat flow rate" and "radiant flux."

- PRESSURE:** 1. Although the pascal is the proper SI unit for pressure, the kilopascal (kPa) is recognized for use in all fields except high vacuum measurement of absolute pressure, for which the pascal may be more convenient.
2. The kPa is used for measurement of both gauge and absolute pressure (gauge pressure is absolute pressure minus ambient pressure [ambient pressure is usually atmospheric pressure]). However, when absolute pressure is intended, the unit kPa should be followed by the word "absolute."
 3. The bar is a convenient multiple of the pascal, the proper SI term for pressure ($1 \text{ bar} = 10^5 \text{ Pa}$), but its use is discouraged. The millibar is and will continue to be widely used in meteorology; however, the kilopascal should be used in presenting meteorological data to the public and for other pressure applications.
 4. The mmHg is also called "torr." (The torr was once widely used in Europe but its use, as well as use of the kilogram-force per square centimetre, was discontinued by SI on January 1, 1978.)
 5. This unit category is also called "stress" and "force per unit area."

SOLID ANGLE: The steradian is a supplementary SI unit.

- TEMPERATURE:** 1. Technically, this unit is called "thermodynamic temperature."
2. The proper SI unit for this category is the Kelvin, NOT the degree Kelvin. For example, a temperature would be correctly expressed as 283K or, less properly but more commonly, as 10°C (though not 10C). But be careful not to confuse the abbreviation for Kelvin with the designation for 1000, as in a 10K ohm resistor.
 3. Degrees Celsius was called Degrees Centigrade and it is the most commonly encountered form of temperature measurement.
 4. One degree Celsius as a temperature interval is equal to one Kelvin unit.
 5. Kelvin is the absolute temperature scale in the metric (Celsius) system.
 6. Degrees Rankine is the absolute temperature scale in the English (Fahrenheit) system.

TIME: 1. The second is the proper SI unit of time. However, a coherent system of time measurement is not practical (e.g., a solar day cannot be conveniently divided into kiloseconds). Therefore, the noncoherent system now in use with minutes, hours, days, and years will continue to be used indefinitely.

2. Time units can be defined as mean or sidereal: mean time closely approximates actual star movement but is modified slightly to provide regularity of measurement; sidereal units are based on actual movement of stars but do not break down into neat units (e.g., a sidereal day is 23 hours, 56 minutes, and 4.09 seconds long — expressed in mean time).

3. Note that the SI symbol for “year” is “a.”

TORQUE: 1. The units in this category are mathematically the same as those in the category “bending moment,” although the application of the units is different.

2. Torque is also called “moment of force.”

3. The preferred unit for this category, the N·m, is the same as the definition for the energy unit joule ($J = N \cdot m$), but the two should not be used interchangeably since they have different applications.

4. The use of the unit kilogram-force times metres was discontinued by SI on January 1, 1978.

VELOCITY: 1. The best way to learn the commonly used velocity measurement kilometres per hour (although metres per second is the proper unit) is by memorizing comparable values. (Refer to the kilometres-to-miles list under “LENGTH.”)

2. This unit category includes “speed.” (Velocity, a vector, is magnitude plus direction while speed, a scalar, is magnitude only.)

VOLUME: 1. Although litres are frequently used as a substitute for quarts, at least in the U. S., it is not technically correct to do so. According to SI guidelines, the cubic metre should be used instead. However, the litre will probably continue to be used for measurement of displacement of an internal combustion engine and for the volume of space in a refrigerator or the trunk of a car.

2. A litre is equivalent to a cube 10 cm on each side (a cubic decimetre).

3. A millilitre is equivalent to a cubic centimetre.

4. The symbol for litre has been a lower case l, but is changing to upper case L to avoid confusion with the number one (1).

5. Water and gas supplies for homes and factories — in fact, almost anything now measured in cubic feet — will be measured in cubic metres, the proper SI unit.

6. Automotive fuel consumption is expressed in countries using the metric system as litres per 100 kilometres or kilometres per litre rather than miles per gallon. A conversion estimate from mpg to L/100 km is achieved by dividing 235 by the mpg (e.g., if you normally get 20 mpg, you will get approximately 12 L/100 km); to go from mpg to km/L, divide the mpg by 2.35 (e.g., 20 mpg = 8.5 km/L).

7. This unit category is also called “capacity.”

VOLUME PER UNIT TIME: This unit category is also called “instantaneous volume velocity” or just “volume velocity.” The unit category “mass per unit time” also includes flow.

TABLE I — Proper SI Units

QUANTITY	NAME OF UNIT	SYMBOL OF DERIVED UNIT	UNIT EXPRESSED AS BASE OR SUPPLEMENTARY UNITS
Absolute Viscosity	Pascal Second	Pa·s	Pa × s
Acceleration	Metre Per Second Squared		m/s ²
Angular Velocity	Radian Per Second		rad/s
Area	Square Metre		m ²
Energy	Joule	J	N·m (kg × m ² × s ⁻²)
Force	Newton	N	kg × m × s ⁻²
Kinematic Viscosity	Square Metres Per Second		m ² /s
Length	Metre		m
Mass	Kilogram		kg
Mass Per Unit Time	Kilogram Per Second		kg/s
Mass Per Unit Volume	Kilograms Per Cubic Metre		kg/m ³
Plane Angle	Radians	rad	m × m ⁻¹
Power	Watt	W	J/s (kg × m ² × s ⁻³)
Pressure	Pascal	Pa	N/m ² (kg × m ⁻¹ × s ⁻²)
Solid Angle	Steradian	sr	m ² × m ⁻²
Temperature	Kelvin Celsius	°C	K K - 273.15
Time	Second		s
Torque	Newton-Metre	N·m	kg/s ² × m
Velocity	Metres Per Second		m/s
Volume	Cubic Metres		m ³
Volume Per Unit Time	Cubic Metres Per Second		m ³ /s

TABLE II — Multiplication Factors

MULTIPLICATION FACTOR	PREFIX	SYMBOL
1 000 000 000 000 = 10^{12}	tera	T
1 000 000 000 = 10^9	giga	G
1 000 000 = 10^6	mega	M
1 000 = 10^3	kilo	k
100 = 10^2	hecto [†]	h
10 = 10^1	deca [†]	da
0.1 = 10^{-1}	deci [†]	d
0.01 = 10^{-2}	centi [†]	c
0.001 = 10^{-3}	milli	m
0.000 001 = 10^{-6}	micro	μ
0.000 000 001 = 10^{-9}	nano	n
0.000 000 000 001 = 10^{-12}	pico	p

[†]These prefixes should be avoided where possible.

NOTE: In some cases, a prefix symbol is the same as a unit symbol, so it is important to look at the position of each symbol in the term to determine its meaning. For example, in ms the "m" means "milli" (millisecond), but in km the "m" means "metre" (kilometre). Also to avoid confusion, it is important that no more than one prefix be used when forming the decimal multiple or submultiple of a derived unit. For example, mμm should be expressed as nm. Refer to Table I — Categories of Units for appropriate abbreviations.

Another general rule is to use SI prefixes to indicate order of magnitude and eliminate nonsignificant digits and leading zeroes. This also provides a convenient conversational alternative to the powers-of-ten notation preferred

in computation. For example,
 12 300 mm becomes 12.3 m
 12.3×10^3 m becomes 12.3 km
 0.001 230 μA becomes 1.23 nA

Another point illustrated by the numbers above is the place-holding value of the comma. Outside the United States the comma is sometimes used instead of the decimal point (e.g., the American 0.00123 would be written 0,00123). To avoid confusion, recommended international practice uses a space instead of a comma when dividing numbers into groups of three digits (a decimal point is still used to indicate a break between numbers greater than one and numbers less than one). This applies to groupings of numbers on either side of zero. For example, 12,300.001230 could be written as 12 300.001 230.

TABLE III — Categories of Units

For more information on the SI system, particularly the units not being covered by this text, refer to the industry standards represented by these numbers: ISO 1000 (incorporated in ISO Standards Handbook 2), ANSI Z210.1-1976, IEEE Standard 268-1976, ASTM E 380-76.

UNIT SYMBOL	UNIT (CATEGORY)	UNIT SYMBOL	UNIT (CATEGORY)
a	year (time)	g/L	grams per litre (mass per unit volume)
acre	acres (area)	g/cm ³	grams per cubic centimetre (mass per unit volume)
atm	atmospheres (pressure)	g/m ³	grams per cubic metre (mass per unit volume)
bars	bars (pressure)	gal, Imp.	Imperial gallons (volume)
bars absolute	bars absolute (pressure)	gal, U.S.	U.S. gallons (volume)
bbl	barrels (volume)	gr	grains (mass)
Btu	British thermal units (energy)	ha	hectares (area)
Btu/h	British thermal units per hour (power)	hp	horsepower (power)
Btu/min	British thermal units per minute (power)	hp-h	horsepower-hours (energy)
cal	calories (energy)	in	inches (length)
cfh	cubic feet per hour (volume per unit time)	in ²	square inches (area)
cfm	cubic feet per minute (volume per unit time)	in ³	cubic inches (volume)
cfs	cubic feet per second (volume per unit time)	in ³ /min	cubic inches per minute (volume per unit time)
cm	centimetres (length)	inH ₂ O at 60°F	inches of water (pressure)
cm ²	square centimetres (area)	inHg at 0°C	inches of mercury (pressure)
cm ³	cubic centimetres (volume)	J*	Joules (energy)
cm ³ /s	cubic centimetres per second (volume per unit time)	K*	Kelvin (temperature)
cmHg at 0°C	centimetres of mercury (pressure)	kPa	kilopascals (pressure)
cm/s	centimetres per second (velocity)	kcal	kilocalories (energy)
cP	centipoises (absolute viscosity)	kcal/min	kilocalories per minute (power)
cSt	centistokes (kinematic viscosity)	kg*	kilograms (mass)
dwt	pennyweights (mass)	kg/h	kilograms per hour (mass per unit time)
dynes	dynes (force)	kg/m ³ *	kilograms per cubic metre (mass per unit volume)
ft	feet (length)	kg/min	kilograms per minute (mass per unit time)
ft ²	square feet (area)	kg/s*	kilograms per second (mass per unit time)
ft ² /s	square feet per second (kinematic viscosity)	kgf	kilograms-force (force)
ft ³	cubic feet (volume)	kgf × m	kilograms-force times metre (torque)
ft H ₂ O at 60°F	feet of water (pressure)	kgf/cm ²	kilograms-force per square centimetre (pressure)
ft-lbf	foot-pounds-force (energy)	km	kilometres (length)
ft-lbf/h	foot-pounds-force per hour (power)	km/h	kilometres per hour (velocity)
ft-lbf/min	foot-pounds-force per minute (power)	kn	knots (velocity)
ft-lbf/s	foot-pounds-force per second (power)	kW	kilowatts (power)
ft/min	feet per minute (velocity)	kW-h	kilowatt-hours (energy)
ft/s	feet per second (velocity)	L	litres (volume)
ft/s ²	feet per second squared (acceleration)	L/s	litres per second (volume per unit time)
g	grams (mass)	lb or lb, av.	avoirdupois pounds (mass)
		lb, troy	troy pounds (mass)
		lb/Imp. gal	pounds per Imperial gallon (mass per unit volume)

* Indicates proper SI unit.

UNIT SYMBOL	UNIT (CATEGORY)
lb/U.S. gal	pounds per U.S. gallon (mass per unit volume)
lb/ft ³	pounds per cubic foot (mass per unit volume)
lb/ft-s	pounds per foot second (absolute viscosity)
lb/h	pounds per hour (mass per unit time)
lb/in ³	pounds per cubic inch (mass per unit volume)
lb/min	pounds per minute (mass per unit time)
lb/s	pounds per second (mass per unit time)
lbf	pounds-force (force)
lbf × ft	pounds-force times feet (torque)
lbf-s/ft ²	pound-force-seconds per square foot (absolute viscosity)
m*	metres (length)
m ² *	square metres (area)
m ² /s	square metres per second (kinematic viscosity)
m ³ *	cubic metre (volume)
m ³ /h	cubic metres per hour (volume per unit time)
m ³ /min	cubic metres per minute (volume per unit time)
m ³ /s*	cubic metres per second (volume per unit time)
m/min	metres per minute (velocity)
m/s*	metres per second (velocity)
m/s ² *	metres per second squared (acceleration)
mi	land miles (length)
mi ²	square miles (area)
mi, nautical	nautical miles (length)
mbar	millibars (pressure)
million U.S. gpd	million U.S. gallons per day (volume per unit time)
mm	millimetres (length)
mmHg at 0°C	millimetres of mercury (pressure)
MPa	megapascals (pressure)
mph	miles per hour (velocity)
MW	megawatts (power)
N*	Newtons (force)
N·m*	Newton-metres (torque)

UNIT SYMBOL	UNIT (CATEGORY)
oz, av.	avoirdupois ounces (mass)
oz, Imp. fluid	Imperial fluid ounces (volume)
oz, troy	troy ounces (mass)
oz, U.S. fluid	U.S. fluid ounces (volume)
oz/in ³	ounces per cubic inch (mass per unit volume)
P	poises (absolute viscosity)
Pa*	Pascal (pressure)
Pa·s*	Pascal second (absolute viscosity)
ppm	parts per million (mass per unit volume)
psi	pounds-force per square inch (pressure)
psia	pounds-force per square inch, absolute (pressure)
pt	pint (volume)
qt	quart (volume)
rad	radians (plane angle)
rad/s*	radians per second (angular velocity)
r/min	revolutions per minute (angular velocity)
r/s	revolutions per second (angular velocity)
short ton	short tons (mass)
sr	steradians (solid angle)
St	stokes (kinematic viscosity)
t	metric tons (mass)
ton	long tons (mass)
U.S. gph	U.S. gallons per hour (volume per unit time)
U.S. gpm	U.S. gallons per minute (volume per unit time)
U.S. gps	U.S. gallons per second (volume per unit time)
W*	Watts (power)
W-h	Watt-hours (energy)
yd	yards (length)
μm	micrometres (length)
"	seconds (plane angle)
'	minutes (plane angle)
°	degrees (plane angle)
°/s	degrees per second (angular velocity)
°C*	degrees Celsius (temperature)
°F	degrees Fahrenheit (temperature)
°R	degrees Rankine (temperature)

TABLE IV — Conversion Factors

HOW TO USE THIS TABLE: Find the unit you want to convert from listed in capital letters at the left-hand margin and multiply it by the number indicated to arrive at the unit listed to the right of the number. Where appropriate, additional information has been provided.

EXAMPLE:

BRITISH THERMAL UNITS			International Table	Btu	(energy)
unit converting from			qualifying information	abbreviation	unit category
conversion factors	$\times 1.055$	$\times 10^3$	$= \text{J}^*$		
	$\times 2.928$	$\times 10^{-4}$	$= \text{kW-h}$		
	$\times 3.928$	$\times 10^{-4}$	$= \text{hp-h}$		
	$\times 0.252$		$= \text{kcal}$		
	$\times 7.780$	$\times 10^2$	$= \text{ft-lbf}$		

unit converting to
* indicates proper SI unit

Additional information { NOTE: There are several definitions of Btu, differing only past the third significant digit. If four or more significant digits are needed, refer to the appropriate handbook.

DERIVING UNITS: Many categories have several units related by a power of ten (e.g., Pa and kPa) or by a factor of 60 (e.g., ft/s, ft/min, and ft/h). Generally, conversion factors are provided for only the proper SI unit or the unit most easy to use. There are several shortcuts to deriving units not listed; following is one reliable method.

Suppose you have a volume per unit time of 1 cfm and you want to express that in m^3/s . Look up Cubic Feet Per Hour and read, "Divide by 60 and refer to Cubic Feet Per Minute." Look up Cubic Feet Per Minute and find the conversion factor to m^3/s ($\text{cfm} \times 4.7195 \times 10^{-4} = \text{m}^3/\text{s}$). String these together and get

$$(1 \text{ cfm}) \times \frac{4.7195 \times 10^{-4}}{60} = 0.000007865833 \text{ or } 7.8658 \times 10^{-6} \text{ m}^3/\text{s}.$$

Suppose you have a volume per unit time of 1 m^3/s and you want to express that in cfm. Look up Cubic Metres Per Second and find that the closest conversion factor for cfm is cfm ($\text{m}^3/\text{s} \times 2.1189 \times 10^3 = \text{cfm}$). Then look up Cubic Feet Per Minute

and find the conversion factor to cfm ($\text{cfm} \times 60 = \text{cfh}$). String these together and get

$$(1 \text{ m}^3/\text{s}) \times (2.1189 \times 10^3) \times 60 = 127134 \text{ or } 1.2713 \times 10^5.$$

SCIENTIFIC NOTATION: Remember when using a calculator that positive powers move the decimal point to the right of the 1 (e.g., $10^3 = 1000.0$) and negative powers move the decimal point to the left of the 1 (e.g., $10^{-3} = 0.0010$).

PRESSURE UNITS: Where density is specified or implied, it is based on the following:

Density of water at 60°F = 62.3707 lb/ft³
Density of mercury at 0°C = 13.5955 g/cm³.

MAGNITUDE OF FIGURES: Where practical, conversion factors over 10^4 or under 10^{-4} have not been included. (Factors have been included for all proper SI units no matter what the multiple.) However, avoid mixing prefixes within one document or equation (e.g., don't use kPa, Pa, and MPa together).

ACRES, U.S. Survey (area)

$\times 4.0469 \times 10^3$	$= \text{m}^2$
$\times 4.3560 \times 10^4$	$= \text{ft}^2$
$\times 1.5625 \times 10^{-3}$	$= \text{mi}^2$
$\times 0.4047$	$= \text{ha}$

ATMOSPHERES, Standard at Sea Level Pressure — atm (pressure)

$\times 1.0132 \times 10^5$	$= \text{Pa}^*$
$\times 14.696$	$= \text{psia}$
$\times 7.60 \times 10^2$	$= \text{mmHg at } 0^\circ\text{C}$
$\times 29.921$	$= \text{inHg at } 0^\circ\text{C}$
$\times 4.0716 \times 10^2$	$= \text{inH}_2\text{O at } 60^\circ\text{F}$
$\times 33.930$	$= \text{ftH}_2\text{O at } 60^\circ\text{F}$
$\times 1.0132$	$= \text{bars absolute}$
$\times 1.0332$	$= \text{kgf/cm}^2 \text{ absolute}$

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

BARRELS, Petroleum — bbl (volume)

$\times 0.1590$	$= \text{m}^3$
$\times 9.702 \times 10^3$	$= \text{in}^3$
$\times 5.6146$	$= \text{ft}^3$
$\times 42$	$= \text{gal, U.S.}$
$\times 34.972$	$= \text{gal, Imp.}$
$\times 1.5898 \times 10^2$	$= \text{L}$

BARS (pressure)

$\times 10^5$	$= \text{Pa}^*$
$\times 14.504$	$= \text{psi}$
$\times 7.5006 \times 10^2$	$= \text{mmHg at } 0^\circ\text{C}$
$\times 29.530$	$= \text{inHg at } 0^\circ\text{C}$
$\times 4.0184 \times 10^2$	$= \text{inH}_2\text{O at } 60^\circ\text{F}$
$\times 33.486$	$= \text{ftH}_2\text{O at } 60^\circ\text{F}$
$\times 0.9869$	$= \text{atm}$
$\times 10^3$	$= \text{mbar}$
$\times 1.0197$	$= \text{kgf/cm}^2$

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

BRITISH THERMAL UNITS, International Table — Btu (energy)

$\times 1.054 \times 10^3$	$= \text{J}^*$
$\times 2.929 \times 10^{-4}$	$= \text{kW-h}$
$\times 3.928 \times 10^{-4}$	$= \text{hp-h}$
$\times 0.252$	$= \text{kcal}$
$\times 7.780 \times 10^2$	$= \text{ft-lbf}$

NOTE: There are definitions of Btu other than the International Table, but they differ only past the third decimal place. If four or more decimal places are needed, refer to the appropriate handbook.

BTU PER HOUR, International Table — Btu/h (power)

$\times 0.293$	$= \text{W}^*$
$\times 1.667 \times 10^{-2}$	$= \text{kW/min}$
$\times 3.93 \times 10^{-4}$	$= \text{hp}$
$\times 4.20 \times 10^{-3}$	$= \text{kcal/min}$
$\times 12.961$	$= \text{ft-lbf/min}$

NOTE: See note under Btu.

BTU PER MINUTE, International Table — Btu/min (power)

NOTE: Multiply by 60 and refer to Btu Per Hour.

CALORIES, International Table — cal (energy)

NOTE: Divide by 1000 and refer to Kilocalories.

CENTARES — ca (area)

NOTE: Refer to Square Metres*.

CENTIMETRES — cm (length)

NOTE: Divide by 100 and refer to Metres*.

CENTIMETRES OF MERCURY, at 0°C — cmHg (pressure)

NOTE: Multiply by 10 and refer to Millimetres of Mercury.

CENTIMETRES PER SECOND — cm/s (velocity)

NOTE: Divide by 100 and refer to Metres Per Second*.

*Indicates proper SI unit

CENIPOISES — cP (absolute viscosity)
NOTE: Divide by 100 and refer to Poises.

CENTISTOKES — cSt (kinematic viscosity)
NOTE: Divide by 100 and refer to Stokes.

CUBIC CENTIMETRES — cm ³ (volume)	
× 10 ⁻⁶	= m ³ *
× 6.1024 × 10 ⁻²	= in ³
× 3.5315 × 10 ⁻⁵	= ft ³
× 3.3814 × 10 ⁻²	= oz, U. S. fluid
× 3.5195 × 10 ⁻²	= oz, Imp. fluid
× 2.6417 × 10 ⁻⁴	= gal, U. S.
× 2.1997 × 10 ⁻⁴	= gal, Imp.
× 10 ⁻³	= L
× 2.1134 × 10 ⁻³	= pt
× 1.0567 × 10 ⁻³	= qt

CUBIC CENTIMETRES PER SECOND — cm ³ /s (volume per unit time)	
× 10 ⁻⁶	= m ³ /s*
× 3.6614	= in ³ /min
× 2.1189 × 10 ⁻³	= cfm
× 10 ⁻³	= L/s
× 1.5850 × 10 ⁻²	= U. S. gpm
× 2.2824 × 10 ⁻⁵	= million U. S. gpd

CUBIC FEET — ft ³ (volume)	
× 2.832 × 10 ⁻²	= m ³ *
× 1.728 × 10 ³	= in ³
× 9.5751 × 10 ²	= oz, U. S. fluid
× 9.9661 × 10 ²	= oz, Imp. fluid
× 7.4805	= gal, U. S.
× 6.229	= gal, Imp.
× 28.317	= L
× 0.1781	= bbl

CUBIC FEET PER HOUR — cfh (volume per unit time)
NOTE: Divide by 60 and refer to Cubic Feet Per Minute.

CUBIC FEET PER MINUTE — cfm (volume per unit time)	
× 4.7195 × 10 ⁻⁴	= m ³ /s*
× 1.6990	= m ³ /h
× 1.728 × 10 ³	= in ³ /min
× 1.667 × 10 ⁻²	= cfs
× 60	= cfh
× 0.4719	= L/s
× 7.4805	= U. S. gpm
× 1.0772 × 10 ⁻²	= million U. S. gpd

CUBIC FEET PER SECOND — cfs (volume per unit time)
NOTE: Multiply by 60 and refer to Cubic Feet Per Minute.

CUBIC INCHES — in ³ (volume)	
× 1.6387 × 10 ⁻⁵	= m ³ *
× 5.787 × 10 ⁻⁴	= ft ³
× 0.5541	= oz, U. S. fluid
× 0.5767	= oz, Imp. fluid
× 4.329 × 10 ⁻³	= gal, U. S.
× 3.605 × 10 ⁻³	= gal, Imp.
× 1.639 × 10 ⁻²	= L

CUBIC INCHES PER MINUTE — in ³ /min (volume per unit time)	
× 2.7312 × 10 ⁻⁷	= m ³ /s*
× 5.787 × 10 ⁻⁴	= cfm
× 2.7312 × 10 ⁻⁴	= L/s
× 4.3290 × 10 ⁻³	= U. S. gpm

*CUBIC METRES — m ³ (volume)	
× 6.1024 × 10 ⁴	= in ³
× 35.315	= ft ³
× 3.3814 × 10 ⁴	= oz, U. S. fluid
× 3.5195 × 10 ⁴	= oz, Imp. fluid
× 2.6417 × 10 ²	= gal, U. S.
× 2.1997 × 10 ²	= gal, Imp.
× 10 ³	= L
× 6.2898	= bbl

CUBIC METRES PER HOUR — m³/h (volume per unit time)
NOTE: Divide by 3600 and refer to Cubic Metres Per Second.*

CUBIC METRES PER MINUTE — m³/min (volume per unit time)
NOTE: Divide by 60 and refer to Cubic Metres Per Second.*

*CUBIC METRES PER SECOND — m ³ /s (volume per unit time)	
× 60	= m ³ /min
× 3.600 × 10 ³	= m ³ /h
× 10 ⁶	= cm ³ /s
× 2.1189 × 10 ³	= cfm
× 10 ³	= L/s
× 15.850 × 10 ³	= U. S. gpm
× 22.824	= million U. S. gpd

DEGREES, Angular — ° (plane angles)	
× 1.745 × 10 ⁻²	= rad
× 60	= ', angular
× 3.600 × 10 ³	= ", angular

*DEGREES CELSIUS — °C (temperature)	
(C × 9/5) + 32	= °F
C + 273.15	= K
(C × 9/5) + 491.67	= °R

NOTE: See also TABLE V — Temperature Conversions

DEGREES CENTIGRADE — see Degrees Celsius (temperature)

DEGREES FAHRENHEIT — °F (temperature)	
(F - 32)/1.8	= °C*
(F + 459.67)/1.8	= K
F + 459.67	= °R

NOTE: See also TABLE V — Temperature Conversions

DEGREES KELVIN — see Kelvin (temperature)

DEGREES RANKINE — °R (temperature)	
(R/1.8) - 273.15	= °C*
R - 459.69	= °F
R/1.8	= K

DEGREES PER SECOND, Angular — °/s (angular velocity)	
× 1.7453 × 10 ⁻²	= rad/s*
× 0.1667	= r/min (rpm)

DYNES (force)	
× 10 ⁻⁵	= N*
× 1.0197 × 10 ⁻⁶	= kgf
× 2.2481 × 10 ⁻⁶	= lbf

FEET — ft (length)	
× 0.3048	= m*
× 12	= in
× 0.3333	= yd
× 1.894 × 10 ⁻⁴	= mi, statute
× 1.6458 × 10 ⁻⁴	= mi, nautical

FEET OF WATER, at 60°F — ftH ₂ O (pressure)	
× 2.9863 × 10 ³	= Pa*
× 0.4331	= psi
× 22.399	= mmHg at 0°C
× 0.8818	= inHg at 0°C
× 12	= inH ₂ O at 60°F
× 2.9473 × 10 ⁻²	= atm
× 2.9863 × 10 ⁻²	= bar
× 3.0452 × 10 ⁻²	= kgf/cm ²

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

FEET PER MINUTE — ft/min (velocity)	
× 5.0800 × 10 ⁻³	= m/s*
× 1.8288 × 10 ⁻²	= km/h
× 1.1364 × 10 ⁻²	= mph
× 1.6667 × 10 ⁻²	= ft/s
× 9.8750 × 10 ⁻³	= kn

FEET PER SECOND — ft/s (velocity)
NOTE: Multiply by 60 and refer to Feet Per Minute.

FEET PER SECOND SQUARED — ft/s ² (acceleration)	
× 0.3048	= m/s ² *

FOOT-POUNDS-FORCE — ft-lbf (energy)	
× 1.3558	= J*
× 3.7662 × 10 ⁻⁷	= kW-h
× 1.285 × 10 ⁻³	= Btu
× 5.0505 × 10 ⁻⁷	= hp-h
× 3.238 × 10 ⁻⁴	= kcal

*Indicates proper SI unit

FOOT-POUNDS-FORCE PER HOUR — ft-lbf/h (power)

NOTE: Divide by 60 and refer to Foot-Pounds-Force Per Minute.

FOOT-POUNDS-FORCE PER MINUTE — ft-lbf/min (power)

$\times 2.2597 \times 10^{-2}$	= W*
$\times 7.716 \times 10^{-2}$	= Btu/h
$\times 3.030 \times 10^{-5}$	= hp
$\times 3.2405 \times 10^{-4}$	= kcal/min
$\times 60$	= ft-lbf/h
$\times 1.667 \times 10^{-2}$	= ft-lbf/s

FOOT-POUNDS-FORCE PER SECOND — ft-lbf/s (power)

NOTE: Multiply by 60 and refer to Foot-Pounds-Force Per Minute.

GALLONS, Imperial — gal (volume)

$\times 4.546 \times 10^{-3}$	= m ³ *
$\times 2.774 \times 10^{-2}$	= in ³
$\times 0.1605$	= ft ³
$\times 1.537 \times 10^{-2}$	= oz, U. S. fluid
$\times 160$	= oz, Imp. fluid
$\times 1.2009$	= gal, U. S.
$\times 4.546$	= L
$\times 2.859 \times 10^{-2}$	= bbl

GALLONS, U. S. — gal (volume)

$\times 3.7854 \times 10^{-3}$	= m ³ *
$\times 2.31 \times 10^{-2}$	= in ³
$\times 0.1337$	= ft ³
$\times 128$	= oz, U. S. fluid
$\times 1.3323 \times 10^{-2}$	= oz, Imp. fluid
$\times 0.8327$	= gal, Imp.
$\times 3.7854$	= L
$\times 8$	= pt
$\times 4$	= qt
$\times 2.3810 \times 10^{-2}$	= bbl

GALLONS PER HOUR, U. S. — U. S. gph (volume per unit time)

NOTE: Divide by 60 and refer to Gallons Per Minute, U. S.

GALLONS PER MINUTE, U. S. — U. S. gpm (volume per unit time)

$\times 6.3090 \times 10^{-5}$	= m ³ /s*
$\times 2.31 \times 10^{-2}$	= in ³ /min
$\times 0.1337$	= cfm
$\times 60$	= U. S. gph
$\times 1.667 \times 10^{-2}$	= U. S. gps
$\times 6.309 \times 10^{-2}$	= L/s
$\times 1.4400 \times 10^{-3}$	= million U. S. gpd

GALLONS PER SECOND, U. S. — U. S. gps (volume per unit time)

NOTE: Multiply by 60 and refer to Gallons Per Minute, U. S.

GRAINS, Avoirdupois or Troy — gr (mass)

$\times 6.480 \times 10^{-5}$	= kg*
$\times 6.480 \times 10^{-2}$	= g
$\times 2.2857 \times 10^{-3}$	= oz, av.
$\times 2.0833 \times 10^{-3}$	= oz, troy
$\times 1.4286 \times 10^{-4}$	= lb, av.
$\times 1.7361 \times 10^{-4}$	= lb, troy
$\times 4.1667 \times 10^{-2}$	= dwt

GRAMS — g (mass)

$\times 10^{-3}$	= kg*
$\times 3.5274 \times 10^{-2}$	= oz, av.
$\times 3.2151 \times 10^{-2}$	= oz, troy
$\times 2.2046 \times 10^{-3}$	= lb, av.
$\times 2.6792 \times 10^{-3}$	= lb, troy
$\times 15.432$	= gr
$\times 0.6430$	= dwt

GRAMS PER CUBIC CENTIMETRE — g/cm³ (mass per unit volume)

NOTE: Divide by 1000 and refer to Kilograms Per Cubic Metre.*

GRAMS PER CUBIC METRE — g/m³ (mass per unit volume)

NOTE: Divide by 1000 and refer to Kilograms Per Cubic Metre.*

GRAMS PER LITRE (g/L) — see Kilograms Per Cubic Metre* (mass per unit volume)**HECTARES — ha (area)**

$\times 10^4$	= m ² *
$\times 3.861 \times 10^{-3}$	= mi ²
$\times 2.4711$	= acre

HORSEPOWER, Boiler — boiler hp (power)

$\times 9.8095 \times 10^3$	= W*
$\times 3.3446 \times 10^4$	= Btu/h
$\times 13.1548$	= hp (mechanical)
$\times 1.407 \times 10^2$	= kcal/min
$\times 4.3411 \times 10^5$	= ft-lbf/min

HORSEPOWER, Mechanical — hp (power)

$\times 7.457 \times 10^2$	= W*
$\times 2.543 \times 10^3$	= Btu/h
$\times 10.694$	= kcal/min
$\times 3.30 \times 10^4$	= ft-lbf/min
$\times 1.0139$	= metric hp
$\times 7.6018 \times 10^{-2}$	= boiler hp

NOTE: In most conversions, this is the type of horsepower assumed unless otherwise stated.

HORSEPOWER, Metric — metric hp (power)

$\times 7.3550 \times 10^2$	= W*
$\times 2.51 \times 10^3$	= Btu/h
$\times 0.9863$	= hp (mechanical)
$\times 10.55$	= kcal/min

HORSEPOWER-HOURS — hp-h (energy)

$\times 2.6845 \times 10^6$	= J*
$\times 0.7457$	= kW-h
$\times 2.546 \times 10^3$	= Btu
$\times 6.416 \times 10^2$	= kcal
$\times 1.98 \times 10^6$	= ft-lbf

INCHES — in (length)

$\times 2.54 \times 10^{-2}$	= m*
$\times 8.3333 \times 10^{-2}$	= ft
$\times 2.7778 \times 10^{-2}$	= yd
$\times 1.5783 \times 10^{-5}$	= mi, statute

INCHES OF MERCURY, at 0°C — inHg (pressure)

$\times 3.3864 \times 10^3$	= Pa*
$\times 0.4912$	= psi
$\times 25.4$	= mmHg at 0°C
$\times 13.608$	= inH ₂ O at 60°F
$\times 1.1340$	= ftH ₂ O at 60°F
$\times 3.3421 \times 10^{-2}$	= atm
$\times 3.3864 \times 10^{-2}$	= bar
$\times 3.4532 \times 10^{-2}$	= kgf/cm ²

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

INCHES OF WATER, at 60°F — inH₂O (pressure)

$\times 2.4886 \times 10^2$	= Pa*
$\times 3.6094 \times 10^{-2}$	= psi
$\times 1.8666$	= mmHg at 0°C
$\times 7.3486 \times 10^{-2}$	= inHg at 0°C
$\times 8.333 \times 10^{-2}$	= ftH ₂ O at 60°F
$\times 2.4560 \times 10^{-3}$	= atm
$\times 2.4886 \times 10^{-3}$	= bar
$\times 2.5377 \times 10^{-3}$	= kgf/cm ²

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

*** JOULES — J (energy)**

$\times 2.778 \times 10^{-7}$	= kW-h
$\times 9.485 \times 10^{-4}$	= Btu
$\times 3.725 \times 10^{-7}$	= hp-h
$\times 2.390 \times 10^{-4}$	= kcal
$\times 0.7376$	= ft-lbf

KELVIN — K (temperature)

K - 273.15	= °C*
1.8K - 459.67	= °F
1.8K	= °R

KILOCALORIES, International Table — kcal (energy)

$\times 4.184 \times 10^3$	= J*
$\times 1.1622 \times 10^{-3}$	= kW-h
$\times 3.9683$	= Btu
$\times 1.5586 \times 10^{-3}$	= hp-h
$\times 3.0860 \times 10^3$	= ft-lbf
$\times 10^3$	= cal

KILOCALORIES PER MINUTE, International Table — kcal/min (power)

$\times 69.733$	= W*
$\times 2.3810 \times 10^2$	= Btu/h
$\times 9.3514 \times 10^{-2}$	= hp
$\times 3.0860 \times 10^3$	= ft-lbf/min

*Indicates proper SI unit

* KILOGRAMS — kg (mass)

× 10 ³	= g
× 35.274	= oz, av.
× 32.151	= oz, troy
× 2.2046	= lb, av.
× 2.6792	= lb, troy
× 1.5432 × 10 ⁴	= gr
× 6.4301 × 10 ²	= dwt
× 9.8420 × 10 ⁴	= long ton
× 1.1023 × 10 ³	= short ton
× 10 ³	= t

* KILOGRAMS PER CUBIC METRE — kg/m³ (mass per unit volume)

× 10 ³	= g/m ³
× 10 ³	= g/cm ³
× 3.6127 × 10 ⁵	= lb/in ³
× 8.3454 × 10 ³	= lb/U. S. gal
× 1.0022 × 10 ²	= lb/imp. gal
× 9.9908 × 10 ³	= ppm in H ₂ O at 60°F

KILOGRAMS PER HOUR — kg/h (mass per unit time)

NOTE: Divide by 3600 and refer to Kilograms Per Second.*

KILOGRAMS PER MINUTE — kg/min (mass per unit time)

NOTE: Divide by 60 and refer to Kilograms Per Second.*

* KILOGRAMS PER SECOND — kg/s (mass per unit time)

× 1.3228 × 10 ²	= lb/min
× 60	= kg/min
× 3.600 × 10 ³	= kg/h

KILOGRAMS-FORCE — kgf (force)

× 9.8067	= N*
× 2.2046	= lbf
× 9.8067 × 10 ⁵	= dynes

KILOGRAMS-FORCE PER SQUARE CENTIMETRE — kgf/cm² (pressure)

× 9.8067 × 10 ⁴	= Pa*
× 14.223	= psi
× 7.3556 × 10 ²	= mmHg at 0°C
× 28.959	= inHg at 0°C
× 3.9406 × 10 ²	= inH ₂ O at 60°F
× 32.838	= ftH ₂ O at 60°F
× 0.9678	= atm
× 0.9807	= bar

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

KILOGRAMS-FORCE TIMES METRES — kgf × m (torque)

× 9.8067	= N·m*
× 7.2330	= lbf × ft

KILOMETRES — km (length)

NOTE: Multiply by 1000 and refer to Metres.*

KILOMETRES PER HOUR — km/h (velocity)

× 0.2778	= m/s*
× 0.6214	= mph
× 54.681	= ft/min
× 0.5400	= kn

KILOPASCALS — kPa (pressure)

NOTE: Multiply by 1000 and refer to Pascals.*

KILOPONDS — see Kilograms-force (force)

KILOWATTS — kW (power)

NOTE: Multiply by 1000 and refer to Watts.*

KILOWATT-HOURS — kW-h (energy)

× 3.600 × 10 ⁶	= J*
× 10 ³	= W-h
× 3.4095 × 10 ³	= Btu
× 1.3410	= hp-h
× 8.5918 × 10 ²	= kcal
× 2.6552 × 10 ⁶	= ft-lbf

KNOTS, International — kn (velocity)

× 0.5144	= m/s*
× 1.852	= km/h
× 1.1508	= mph
× 1.0127 × 10 ²	= ft/min

*Indicates proper SI unit

LITRES — L (volume)

× 10 ⁻³	= m ³ *
× 61.024	= in ³
× 3.5315 × 10 ⁻²	= ft ³
× 33.814	= oz, U. S. fluid
× 35.195	= oz, imp. fluid
× 0.2642	= gal, U. S.
× 0.2200	= gal, imp.
× 6.2898 × 10 ⁻³	= bbl

LITRES PER SECOND — L/s (volume per unit time)

× 10 ⁻³	= m ³ /s*
× 3.6614 × 10 ³	= in ³ /min
× 2.1189	= cfm
× 15.850	= U. S. gpm
× 2.2824 × 10 ⁻²	= million U. S. gpd

MEGAPASCALS — MPa (pressure)

NOTE: Multiply by 1 000 000 and refer to Pascals.*

MEGAWATTS — MW (power)

NOTE: Multiply by 1 000 000 and refer to Watts.*

* METRES — m (length)

× 10 ⁶	= μm
× 10 ³	= mm
× 10 ²	= cm
× 10 ⁻³	= km
× 39.370	= in
× 3.2808	= ft
× 1.0936	= yd
× 6.2137 × 10 ⁻⁴	= mi, statute
× 5.3996 × 10 ⁻⁴	= mi, nautical

METRES PER MINUTE — m/min (velocity)

NOTE: Divide by 60 and refer to Metres Per Second.*

* METRES PER SECOND — m/s (velocity)

× 60	= m/min
× 10 ²	= cm/s
× 3.6	= km/h
× 2.2369	= mph
× 1.9685 × 10 ²	= ft/min
× 1.9438	= kn

* METRES PER SECOND SQUARED — m/s² (acceleration)

× 3.281	= ft/s ²
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METRIC TONS — see Tonnes (mass)

MICROMETRES — μm (length)

× 1.000 × 10 ⁻⁶	= m*
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MICRONS — see Micrometres (length)

MILES, Statute — mi (length)

× 1.6093 × 10 ³	= m*
× 6.3360 × 10 ⁴	= in
× 5.280 × 10 ³	= ft
× 1.760 × 10 ³	= yd
× 0.8690	= mi, nautical

MILES, International Nautical — mi (length)

× 1.852 × 10 ³	= m*
× 7.2913 × 10 ⁴	= in
× 6.0761 × 10 ³	= ft
× 2.0254 × 10 ³	= yd
× 1.1508	= mi, statute

MILES PER HOUR, Statute — mph (velocity)

× 0.4470	= m/s*
× 1.6093	= km/h
× 88	= ft/min
× 0.8690	= kn

MILLIBARS — mbar (pressure)

× 10 ⁻³	= bars
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MILLILITRES — see Cubic Centimetres (volume)

MILLIMETRES — mm (length)

NOTE: Divide by 1000 and refer to Metres.*

MILLIMETRES OF MERCURY, at 0°C — mmHg (pressure)

$\times 1.3332 \times 10^2$	= Pa*
$\times 1.9337 \times 10^{-2}$	= psi
$\times 0.10$	= cmHg at 0°C
$\times 3.9370 \times 10^{-2}$	= inHg at 0°C
$\times 0.5357$	= inH ₂ O at 60°F
$\times 4.4644 \times 10^{-2}$	= ftH ₂ O at 60°F
$\times 1.3158 \times 10^{-3}$	= atm
$\times 1.3332 \times 10^{-3}$	= bar
$\times 1.3595 \times 10^{-3}$	= kgf/cm ²

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

MILLION GALLONS PER DAY, U. S. — million U. S. gpd (volume per unit time)

$\times 4.3813 \times 10^{-2}$	= m ³ /s*
$\times 1.6042 \times 10^6$	= in ³ /min
$\times 92.834$	= cfm
$\times 43.813$	= L/s
$\times 6.9444 \times 10^2$	= U. S. gpm

MINUTES, Angular — ' (plane angles)

$\times 2.9089 \times 10^{-4}$	= rad*
$\times 1.667 \times 10^{-2}$	= °, angular
$\times 60$	= ", angular

* NEWTONS — N (force)

$\times 0.1020$	= kgf
$\times 0.2248$	= lbf
$\times 10^5$	= dynes

* NEWTON-METRES — N-m (torque)

$\times 0.1020$	= kgf $\times$ m
$\times 0.7376$	= lbf $\times$ ft

OUNCES, Avoirdupois — av. oz (mass)

$\times 2.8350 \times 10^{-2}$	= kg*
$\times 28.350$	= g
$\times 0.9115$	= oz, troy
$\times 0.0625$	= lb, av.
$\times 7.595 \times 10^{-2}$	= lb, troy
$\times 4.375 \times 10^2$	= gr
$\times 18.229$	= dwt

OUNCES, Fluid, Imperial — oz (volume)

$\times 2.8412 \times 10^{-5}$	= m ³ *
$\times 1.7339$	= in ³
$\times 1.0034 \times 10^{-3}$	= ft ³
$\times 0.9608$	= oz, U. S. fluid
$\times 7.5060 \times 10^{-3}$	= gal, U. S.
$\times 6.25 \times 10^{-3}$	= gal, Imp.
$\times 2.8412 \times 10^{-2}$	= L

OUNCES, Fluid, U. S. — oz (volume)

$\times 2.9574 \times 10^{-5}$	= m ³ *
$\times 1.8047$	= in ³
$\times 1.0444 \times 10^{-3}$	= ft ³
$\times 1.0408$	= oz, Imp. fluid
$\times 7.8125 \times 10^{-3}$	= gal, U. S.
$\times 6.5053 \times 10^{-3}$	= gal, Imp.
$\times 2.9573 \times 10^{-2}$	= L

OUNCES, Troy — troy oz (mass)

$\times 3.1103 \times 10^{-2}$	= kg*
$\times 31.103$	= g
$\times 1.0971$	= oz, av.
$\times 8.3333 \times 10^{-2}$	= lb, troy
$\times 6.857 \times 10^{-2}$	= lb, av.
$\times 4.80 \times 10^2$	= gr
$\times 20$	= dwt

PARTS PER MILLION, by weight (mass) in water at 60°F — ppm or ppm in H₂O at 60°F (mass per unit volume)

$\times 9.9908 \times 10^{-4}$	= kg/m ³ *
$\times 3.6094 \times 10^{-6}$	= lb/in ³
$\times 8.3377 \times 10^{-6}$	= lb/U. S. gal
$\times 1.0013 \times 10^{-5}$	= lb/Imp. gal

* PASCALS — Pa (pressure)

$\times 10^{-3}$	= kPa
$\times 10^{-6}$	= MPa
$\times 1.4504 \times 10^{-4}$	= psi
$\times 7.5006 \times 10^{-3}$	= mmHg at 0°C
$\times 2.9530 \times 10^{-4}$	= inHg at 0°C
$\times 4.0186 \times 10^{-3}$	= inH ₂ O at 60°F
$\times 3.3488 \times 10^{-4}$	= ftH ₂ O at 60°F
$\times 9.8692 \times 10^{-6}$	= atm
$\times 10^{-5}$	= bar
$\times 1.0197 \times 10^{-5}$	= kgf/cm ²
$\times 10$	= dynes/cm ²

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

PENNYWEIGHTS — dwt (mass)

$\times 1.5552 \times 10^{-3}$	= kg*
$\times 1.5552$	= g
$\times 5.4857 \times 10^{-2}$	= oz, av.
$\times 5.00 \times 10^{-2}$	= oz, troy
$\times 3.4286 \times 10^{-3}$	= lb, av.
$\times 4.167 \times 10^{-3}$	= lb, troy
$\times 24$	= gr

PINTS, Fluid — pt (volume)

$\times 4.7316 \times 10^{-4}$	= m ³ *
$\times 28.875$	= in ³
$\times 1.671 \times 10^{-2}$	= ft ³
$\times 16$	= oz, U. S. fluid
$\times 16.653$	= oz, Imp. fluid
$\times 0.125$	= gal, U. S.
$\times 0.1041$	= gal, Imp.
$\times 0.4732$	= L
$\times 0.5$	= qt

POISES — P (absolute viscosity)

$\times 0.1000$	= Pa-s*
$\times 100$	= cP
$\times 2.0885 \times 10^{-3}$	= lbf-s/ft ²
$\times 0.0672$	= lb/ft-s

POUNDS, Avoirdupois — lb (mass)

$\times 0.4536$	= kg*
$\times 4.5359 \times 10^2$	= g
$\times 16$	= oz, av.
$\times 14.583$	= oz, troy
$\times 1.2153$	= lb, troy
$\times 7.00 \times 10^3$	= gr
$\times 2.9167 \times 10^2$	= dwt
$\times 5.00 \times 10^{-4}$	= short ton
$\times 4.464 \times 10^{-4}$	= long ton
$\times 4.536 \times 10^{-4}$	= t

POUNDS, Troy — lb (mass)

$\times 0.3732$	= kg*
$\times 3.732 \times 10^2$	= g
$\times 12$	= oz, troy
$\times 13.166$	= oz, av.
$\times 0.8229$	= lb, av.
$\times 5.760 \times 10^3$	= gr
$\times 2.40 \times 10^2$	= dwt
$\times 4.1143 \times 10^{-4}$	= short ton
$\times 3.6735 \times 10^{-4}$	= long ton
$\times 3.7324 \times 10^{-4}$	= t

POUNDS PER CUBIC FOOT — lb/ft³ (mass per unit volume)

$\times 16.018$	= kg/m ³ *
$\times 5.787 \times 10^{-4}$	= lb/in ³
$\times 0.1337$	= lb/U. S. gal
$\times 0.1605$	= lb/Imp. gal
$\times 1.6033 \times 10^4$	= ppm inH ₂ O at 60°F

POUNDS PER CUBIC INCH — lb/in³ (mass per unit volume)

$\times 2.7680 \times 10^4$	= kg/m ³ *
$\times 1.728 \times 10^3$	= lb/ft ³
$\times 2.31 \times 10^2$	= lb/U. S. gal
$\times 2.774 \times 10^2$	= lb/Imp. gal
$\times 2.7705 \times 10^7$	= ppm inH ₂ O at 60°F

*Indicates proper SI unit

POUNDS PER HOUR — lb/h (mass per unit time)
NOTE: Divide by 60 and refer to Pounds Per Minute.

POUNDS PER IMPERIAL GALLON — lb/gal (mass per unit volume)
 $\times 99.776 = \text{kg/m}^3$
 $\times 3.6047 \times 10^{-3} = \text{lb/in}^3$
 $\times 0.8327 = \text{lb/U.S. gal}$
 $\times 9.9868 \times 10^4 = \text{ppm in H}_2\text{O at } 60^\circ\text{F}$

POUNDS PER MINUTE — lb/min (mass per unit time)
 $\times 7.5599 \times 10^{-3} = \text{kg/s}^*$
 $\times 1.667 \times 10^{-2} = \text{lb/s}$
 $\times 60 = \text{lb/h}$

POUNDS PER SECOND — lb/s (mass per unit time)
NOTE: Multiply by 60 and refer to Pounds Per Minute.

POUNDS PER U.S. GALLON — lb/gal (mass per unit volume)
 $\times 1.1983 \times 10^2 = \text{kg/m}^3$
 $\times 4.3290 \times 10^{-3} = \text{lb/in}^3$
 $\times 1.2010 = \text{lb/imp. gal}$
 $\times 1.1994 \times 10^5 = \text{ppm in H}_2\text{O at } 60^\circ\text{F}$

POUNDS-FORCE — lbf (force)
 $\times 4.4482 = \text{N}^*$
 $\times 0.4536 = \text{kgf}$
 $\times 4.4482 \times 10^5 = \text{dynes}$

POUNDS-FORCE TIMES FEET — lbf $\times$ ft (torque)
 $\times 1.3558 = \text{N-m}^*$
 $\times 0.1383 = \text{kgf} \times \text{m}$

POUNDS-FORCE PER SQUARE INCH — psi (pressure)
 $\times 6.895 \times 10^3 = \text{Pa}^*$
 $\times 51.715 = \text{mmHg at } 0^\circ\text{C}$
 $\times 2.036 = \text{inHg at } 0^\circ\text{C}$
 $\times 27.705 = \text{inH}_2\text{O at } 60^\circ\text{F}$
 $\times 2.3088 = \text{ftH}_2\text{O at } 60^\circ\text{F}$
 $\times 6.8046 \times 10^{-2} = \text{atm}$
 $\times 6.895 \times 10^{-2} = \text{bar}$
 $\times 7.031 \times 10^{-2} = \text{kgf/cm}^2$

NOTE: Where a qualifying temperature is noted, the values for this unit vary with temperature.

QUARTS, Fluid — qt (volume)
 $\times 9.4635 \times 10^{-4} = \text{m}^3$
 $\times 57.75 = \text{in}^3$
 $\times 3.342 \times 10^{-2} = \text{ft}^3$
 $\times 32 = \text{oz, U.S. fluid}$
 $\times 33.31 = \text{oz, imp. fluid}$
 $\times 0.25 = \text{gal, U.S.}$
 $\times 0.2082 = \text{gal, imp.}$
 $\times 0.9464 = \text{L}$

* RADIANS — rad (plane angles)
 $\times 57.296 = ^\circ, \text{angular}$
 $\times 3.4377 \times 10^3 = ', \text{angular}$
 $\times 2.0626 \times 10^5 = ", \text{angular}$

* RADIANS PER SECOND — rad/s (angular velocity)
 $\times 57.296 = ^\circ/\text{s}$
 $\times 9.5493 = \text{r/min (rpm)}$

REVOLUTIONS PER MINUTE — r/min (angular velocity)
 $\times 0.1047 = \text{rad/s}^*$
 $\times 6 = ^\circ/\text{s}$
 $\times 1.667 \times 10^{-2} = \text{r/s}$

NOTE: A common variation of the short form of this category is rpm.

REVOLUTIONS PER SECOND — r/s (angular velocity)
NOTE: Multiply by 60 and refer to Revolutions Per Minute.

SECONDS, Angular — '' (plane angles)
 $\times 4.8481 \times 10^6 = \text{rad}^*$
 $\times 2.778 \times 10^4 = ^\circ, \text{angular}$
 $\times 1.667 \times 10^{-2} = ', \text{angular}$

SQUARE CENTIMETRES — cm² (area)
NOTE: Divide by 10 000 and refer to Square Metres.*

SQUARE FEET — ft² (area)
 $\times 9.2903 \times 10^{-2} = \text{m}^{2*}$
 $\times 1.44 \times 10^2 = \text{in}^2$
 $\times 3.5870 \times 10^{-8} = \text{mi}^2$
 $\times 2.2957 \times 10^{-6} = \text{acre}$
 $\times 9.29 \times 10^{-6} = \text{ha}$

SQUARE INCHES — in² (area)
 $\times 6.4516 \times 10^{-4} = \text{m}^{2*}$
 $\times 6.944 \times 10^{-3} = \text{ft}^2$

* SQUARE METRES — m² (area)
 $\times 10^4 = \text{cm}^2$
 $\times 1.550 \times 10^3 = \text{in}^2$
 $\times 10.764 = \text{ft}^2$
 $\times 2.4711 \times 10^{-4} = \text{acre}$
 $\times 10^{-4} = \text{ha}$
 $\times 1 = \text{ca}$

SQUARE MILES — mi² (area)
 $\times 2.5900 \times 10^6 = \text{m}^{2*}$
 $\times 6.40 \times 10^2 = \text{acre}$
 $\times 2.5900 \times 10^2 = \text{ha}$

STOKES — St (kinematic viscosity)
 $\times 10^{-4} = \text{m}^2/\text{s}^*$
 $\times 1.076 \times 10^{-3} = \text{ft}^2/\text{s}$
 $\times 10^2 = \text{cSt}$

TONNES — t (mass)
 $\times 10^3 = \text{kg}^*$
 $\times 2.2046 \times 10^3 = \text{lb, av.}$
 $\times 2.679 \times 10^3 = \text{lb, troy}$
 $\times 0.9842 = \text{long ton}$
 $\times 1.1023 = \text{short ton}$

TONS — long ton (mass)
 $\times 1.016 \times 10^3 = \text{kg}^*$
 $\times 2.240 \times 10^3 = \text{lb, av.}$
 $\times 2.722 \times 10^3 = \text{lb, troy}$
 $\times 1.120 = \text{short ton}$
 $\times 1.016 = \text{t}$

TONS — ton or short ton (mass)
 $\times 9.072 \times 10^2 = \text{kg}^*$
 $\times 2 \times 10^3 = \text{lb, av.}$
 $\times 2.4306 \times 10^3 = \text{lb, troy}$
 $\times 0.8929 = \text{long ton}$
 $\times 0.9072 = \text{t}$

TORR — see Millimetres of Mercury (pressure)

* WATTS — W (power)
 $\times 10^{-3} = \text{kW}$
 $\times 10^{-6} = \text{MW}$
 $\times 3.414 = \text{Btu/h}$
 $\times 1.3410 \times 10^{-3} = \text{hp}$
 $\times 1.432 \times 10^{-2} = \text{kcal/min}$
 $\times 44.2357 = \text{ft-lbf/min}$

WATT-HOURS — W-h (energy)
NOTE: Divide by 1000 and refer to Kilowatt-hours.

YARDS — yd (length)
 $\times 0.9144 = \text{m}^*$
 $\times 36 = \text{in}$
 $\times 3 = \text{ft}$
 $\times 5.682 \times 10^{-4} = \text{mi, statute}$
 $\times 4.937 \times 10^{-4} = \text{mi, nautical}$

*Indicates proper SI unit

TABLE V — Temperature Conversions

Fahrenheit and Celsius (Centigrade)

C	*	F	C	*	F	C	*	F	C	*	F	C	*	F
-273.15	-459.67		-17.2	1	33.8	10.6	51	123.8	43	110	230	266	510	950
-268	-450		-16.7	2	35.6	11.1	52	125.6	49	120	248	271	520	968
-262	-440		-16.1	3	37.4	11.7	53	127.4	54	130	266	277	530	986
-257	-430		-15.6	4	39.2	12.2	54	129.2	60	140	284	282	540	1004
-251	-420		-15.0	5	41.0	12.8	55	131.0	66	150	302	288	550	1022
-246	-410		-14.4	6	42.8	13.3	56	132.8	71	160	320	293	560	1040
-240	-400		-13.9	7	44.6	13.9	57	134.6	77	170	338	299	570	1058
-234	-390		-13.3	8	46.4	14.4	58	136.4	82	180	356	304	580	1076
-229	-380		-12.8	9	48.2	15.0	59	138.2	88	190	374	310	590	1094
-223	-370		-12.2	10	50.0	15.6	60	140.0	93	200	392	316	600	1112
-218	-360		-11.7	11	51.8	16.1	61	141.8	99	210	410	321	610	1130
-212	-350		-11.1	12	53.6	16.7	62	143.6				327	620	1148
-207	-340		-10.6	13	55.4	17.2	63	145.4				332	630	1166
-201	-330		-10.0	14	57.2	17.8	64	147.2				338	640	1184
-196	-320		-9.4	15	59.0	18.3	65	149.0				343	650	1202
-190	-310		-8.9	16	60.8	18.9	66	150.8	100	212	413	349	660	1220
-184	-300		-8.3	17	62.6	19.4	67	152.6				354	670	1238
-179	-290		-7.8	18	64.4	20.0	68	154.4				360	680	1256
-173	-280		-7.2	19	66.2	20.6	69	156.2				366	690	1274
-169	-273	-459.4	-6.7	20	68.0	21.1	70	158.0				371	700	1292
-168	-270	-454	-6.1	21	69.8	21.7	71	159.8				377	710	1310
-162	-260	-436	-5.6	22	71.6	22.2	72	161.6	104	220	428	382	720	1328
-157	-250	-418	-5.0	23	73.4	22.8	73	163.4	110	230	446	388	730	1346
-151	-240	-400	-4.4	24	75.2	23.3	74	165.2	116	240	464	393	740	1364
-146	-230	-382	-3.9	25	77.0	23.9	75	167.0	121	250	482	399	750	1382
-140	-220	-364	-3.3	26	78.8	24.4	76	168.8	127	260	500	404	760	1400
-134	-210	-346	-2.8	27	80.6	25.0	77	170.6	132	270	518	410	770	1418
-129	-200	-328	-2.2	28	82.4	25.6	78	172.4	138	280	536	416	780	1436
-123	-190	-310	-1.7	29	84.2	26.1	79	174.2	143	290	554	421	790	1454
-118	-180	-292	-1.1	30	86.0	26.7	80	176.0	149	300	572	427	800	1472
-112	-170	-274	-0.6	31	87.8	27.2	81	177.8	154	310	590	432	810	1490
-107	-160	-256	0	32	89.6	27.8	82	179.6	160	320	608	438	820	1508
-101	-150	-238	0.6	33	91.4	28.3	83	181.4	166	330	626	443	830	1526
-95.6	-140	-220	1.1	34	93.2	28.9	84	183.2	171	340	644	449	840	1544
-90.0	-130	-202	1.7	35	95.0	29.4	85	185.0	177	350	662	454	850	1562
-84.4	-120	-184	2.2	36	96.8	30.0	86	186.8	182	360	680	460	860	1580
-78.9	-110	-166	2.8	37	98.6	30.6	87	188.6	188	370	698	466	870	1598
-73.3	-100	-148	3.3	38	100.4	31.1	88	190.4	193	380	716	471	880	1616
-67.8	-90	-130	3.9	39	102.2	31.7	89	192.2	199	390	734	477	890	1634
-62.2	-80	-112	4.4	40	104.0	32.2	90	194.0	204	400	752	482	900	1652
-56.7	-70	-94	5.0	41	105.8	32.8	91	195.8	210	410	770	488	910	1670
-51.1	-60	-76	5.6	42	107.6	33.3	92	197.6	216	420	788	493	920	1688
-45.6	-50	-58	6.1	43	109.4	33.9	93	199.4	221	430	806	499	930	1706
-40.0	-40	-40	6.7	44	111.2	34.4	94	201.2	227	440	824	504	940	1724
-34.4	-30	-22	7.2	45	113.0	35.0	95	203.0	232	450	842	510	950	1742
-28.9	-20	-4	7.8	46	114.8	35.6	96	204.8	238	460	860	516	960	1760
-23.3	-10	14	8.3	47	116.6	36.1	97	206.6	243	470	878	521	970	1778
-17.8	0	32	8.9	48	118.4	36.7	98	208.4	249	480	896	527	980	1796
			9.4	49	120.2	37.2	99	210.2	254	490	914	532	990	1814
			10.0	50	122.0	37.8	100	212.0	260	500	932	538	1000	1832

*In the center column, find the temperature to be converted. The equivalent temperature is in the left column, if converting to Celsius, and in the right column, if converting to Fahrenheit.

INTERPOLATION VALUES

C	*	F	C	*	F
0.56	1	1.8	3.33	6	10.8
1.11	2	3.6	3.89	7	12.6
1.67	3	5.4	4.44	8	14.4
2.22	4	7.2	5.00	9	16.2
2.78	5	9.0	5.56	10	18.0

C	*	F	C	*	F	C	*	F	C	*	F
543	1010	1850	821	1510	2750	1099	2010	3650	1377	2510	4550
549	1020	1868	827	1520	2768	1104	2020	3668	1382	2520	4568
554	1030	1886	832	1530	2786	1110	2030	3686	1388	2530	4586
560	1040	1904	838	1540	2804	1116	2040	3704	1393	2540	4604
566	1050	1922	843	1550	2822	1121	2050	3722	1399	2550	4622
571	1060	1940	849	1560	2840	1127	2060	3740	1404	2560	4640
577	1070	1958	854	1570	2858	1132	2070	3758	1410	2570	4658
582	1080	1976	860	1580	2876	1138	2080	3776	1416	2580	4676
588	1090	1994	866	1590	2894	1143	2090	3794	1421	2590	4694
593	1100	2012	871	1600	2912	1149	2100	3812	1427	2600	4712
599	1110	2030	877	1610	2930	1154	2110	3830	1432	2610	4730
604	1120	2048	882	1620	2948	1160	2120	3848	1438	2620	4748
610	1130	2066	888	1630	2966	1166	2130	3866	1443	2630	4766
616	1140	2084	893	1640	2984	1171	2140	3884	1449	2640	4784
621	1150	2102	899	1650	3002	1177	2150	3902	1454	2650	4802
627	1160	2120	904	1660	3020	1182	2160	3920	1460	2660	4820
632	1170	2138	910	1670	3038	1188	2170	3938	1466	2670	4838
638	1180	2156	916	1680	3056	1193	2180	3956	1471	2680	4856
643	1190	2174	921	1690	3074	1199	2190	3974	1477	2690	4874
649	1200	2192	927	1700	3092	1204	2200	3992	1482	2700	4892
654	1210	2210	932	1710	3110	1210	2210	4010	1488	2710	4910
660	1220	2228	938	1720	3128	1216	2220	4028	1493	2720	4928
666	1230	2246	943	1730	3146	1221	2230	4046	1499	2730	4946
671	1240	2264	949	1740	3164	1227	2240	4064	1504	2740	4964
677	1250	2282	954	1750	3182	1232	2250	4082	1510	2750	4982
682	1260	2300	960	1760	3200	1238	2260	4100	1516	2760	5000
688	1270	2318	966	1770	3218	1243	2270	4118	1521	2770	5018
693	1280	2336	971	1780	3236	1249	2280	4136	1527	2780	5036
699	1290	2354	977	1790	3254	1254	2290	4154	1532	2790	5054
704	1300	2372	982	1800	3272	1260	2300	4172	1538	2800	5072
710	1310	2390	988	1810	3290	1266	2310	4190	1543	2810	5090
716	1320	2408	993	1820	3308	1271	2320	4208	1549	2820	5108
721	1330	2426	999	1830	3326	1277	2330	4226	1554	2830	5126
727	1340	2444	1004	1840	3344	1282	2340	4244	1560	2840	5144
732	1350	2462	1010	1850	3362	1288	2350	4262	1566	2850	5162
738	1360	2480	1016	1860	3380	1293	2360	4280	1571	2860	5180
743	1370	2498	1021	1870	3398	1299	2370	4298	1577	2870	5198
749	1380	2516	1027	1880	3416	1304	2380	4316	1582	2880	5216
754	1390	2534	1032	1890	3434	1310	2390	4334	1588	2890	5234
760	1400	2552	1038	1900	3452	1316	2400	4352	1593	2900	5252
766	1410	2570	1043	1910	3470	1321	2410	4370	1599	2910	5270
771	1420	2588	1049	1920	3488	1327	2420	4388	1604	2920	5288
777	1430	2606	1054	1930	3506	1332	2430	4406	1610	2930	5306
782	1440	2624	1060	1940	3524	1338	2440	4424	1616	2940	5324
788	1450	2642	1066	1950	3542	1343	2450	4442	1621	2950	5342
793	1460	2660	1071	1960	3560	1349	2460	4460	1627	2960	5360
799	1470	2678	1077	1970	3578	1354	2470	4478	1632	2970	5378
804	1480	2696	1082	1980	3596	1360	2480	4496	1638	2980	5396
810	1490	2714	1088	1990	3614	1366	2490	4514	1643	2990	5414
816	1500	2732	1093	2000	3632	1371	2500	4532	1649	3000	5432

APPENDIX C — Bibliography

Basic Process Control Information (General Applications)

* *Instrumentation for Process Measurement and Control* (Third Edition), by Norman A. Anderson (1980)

(a detailed survey of various types of control systems, the range of electronic, pneumatic, and analytical instruments used in a control system, control loop adjustment and analysis, and process computers)

* *Process-Control Systems* (Second Edition), by F.G. Shinskey

(explains the elements of a process control loop and how they contribute to stability and ease of regulation; other topics include nonlinear characteristics and contributions of additional loops and adaptive, decoupling, and feedforward controls)

* "Basic Control Data," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-1)

(articles on pneumatic transmission distances, avoiding reset integral windup in cascade systems, rules for adjusting controllers, interaction between controller settings, an index of controllability, automatic selection of manipulated variables, self-regulation, and external reset)

* "Control Loop Interaction," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-3)

(articles on interaction between control loops and multiple-output systems)

* "Basic Concepts, Terminology and Techniques for Process Control," an article by Lewis M. Gordon, published in *Chemical Engineering Magazine* (Foxboro Publication 623)

* "Feedback Control Modes," an article by Lewis M. Gordon, published in *Chemical Engineering Magazine* (Foxboro Publication 623)

* "Feedback Methods for Process Control Systems," an article by Thomas J. Myron, Jr., published in *Chemical Engineering Magazine* (Foxboro Publication 623)

* "Direct-Synthesis and Adaptive Controls," an article by Paul C. Badavas, published in *Chemical Engineering Magazine* (Foxboro Publication 623)

* "How to Use Feedback Loops to Meet Process Conditions," an article by Thomas C. Kurth, published in *Chemical Engineering Magazine* (Foxboro Publication 623)

Advanced Process Control Topics (General Applications)

* *Energy Conservation Through Control*, by F.G. Shinskey

(discusses: the thermodynamic aspects of energy conservation; minimizing power lost while exercising control over processes; compressor and refrigeration controls; applying energy-converting controls to the mass-transfer operations of evaporation, drying, and distillation; modifying heating, ventilating, and air-conditioning systems controls)

* *pH and pION Control in Process and Waste Streams*, by F.G. Shinskey

(discusses ion-measuring systems including the relationship between measured pH and the concentration of acids and bases, other measurable ions, and oxidation-reduction potential; also describes how to achieve control over ionic species, how to build a controllable plant, and how to estimate the plant's performance)

* *Controlling Multivariable Processes*, by F.G. Shinskey

(for people wanting to develop greater skills in control-system design)

* "Specific Control Problems," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-2)

(articles on sampled-data control, processes with variable periods, adaptive control, end-point control of batch processes, and batch reactor control systems)

* These books and articles were written by experts at Foxboro and are available through your Foxboro sales representative. For other references, contact your local library or trade association.

Your Foxboro sales representative can recommend other publications and technical articles to suit your specific needs.

- * "Controlling Unstable Processes," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-4)

(articles on positive feedback and control, limit cycles and expanding cycles, diagnosing control problems, and controls for four unstable processes — the steam jet, a heat exchanger, parallel heat exchangers, and the exothermic reactor)

Specialized Process Control Topics (Specific Applications)

- * *An Introduction to Paper Industry Instrumentation*, by J.R. Lavigne

(basic considerations for instruments and control in pulp and paper processes; computers are introduced)

- * *Instrumentation Applications for the Pulp and Paper Industry*, by J.R. Lavigne

(applying the basic knowledge of Lavigne's *An Introduction to Paper Industry Instrumentation*; also how to increase operating efficiency through quality improvement)

- * *Distillation Control for Productivity and Energy Conservation*, by F.G. Shinskey

(incentives for improving control and the constraints to be encountered along the way; how to evaluate the effects of control, absorption, stripping, and aeolotropic distillation; control of reboilers and condensers; product quality control; loop and decoupling interaction; and optimization)

- * *Instrumentation & Control in Water Supply and Wastewater Disposal*, by R.H. Babcock
- (answers a variety of difficult technical questions)

- * "Humidity and Dryer Control," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-6)

(articles on humidity/dew point/wet-bulb temperature and three types of dryers — batch, continuous adiabatic, and steam-tube)

- * "pH Control Problems," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-7)

(articles on nonlinear control for pH, the importance of mixing in pH control, leakage problems with pH electrodes, the dynamic response of potentiometric electrodes, and difficulties of pH control)

Instrumentation

- * *Flow Measurement Engineering Handbook*, by Richard Miller

(explains, theoretically and practically, all types of flowmeters now in use and how to select them for your application; translates the latest ISO and ANSI standards into practical U.S. and SI unit engineering equations developed to accommodate modern calculators; discusses fluid properties, accuracy calculations, and common influence parameters)

- * "Control Valve Engineering Handbook," Foxboro Publication 237

(application information and copious tables)

- * "Flow and Level Control," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-5)

(articles on analog blending systems, accuracy and rangeability of head flowmeters, estimating systems accuracy, characterizers for flumes and weirs, and liquid level control)

- * "Control Valves and Motors," one of a series of short publications by F.G. Shinskey (Foxboro Publication 413-8)

(articles on when you have the wrong valve characteristic, sequencing valves for wide range, when to use valve positioners, and dynamic response of valve motors)

After the System Is Installed

- "Instrumentation Symbols and Identification," ISA Standard S5.1, updated in 1985

(the standard for the process industries in the United States)

- * "Tuning Process Controllers," an article by Thomas B. Kinney, published in *Chemical Engineering Magazine*

(Foxboro Publication 623)

* These books and articles were written by experts at Foxboro and are available through your Foxboro sales representative. For other references, contact your local library or trade association.

Appendices

ISO Standards Handbook 2
(units of measurement, International Organization
of Standardization)

*Guidelines for Writers of SI Metric Standards and Other
Documents*, American National Metric Council
(a great deal of practical information on using the
SI system including topics such as "Thinking
Metric," rounding, working with uncertainties, and
case histories on implementing an SI system)

Metric Practice, American National Standard ANSI
Z210.1-1976 (also known as ASTM E 380-76 and IEEE
Std. 268-1976)

*SI Units and Recommendations for the Use of Their
Multiples and of Certain Other Units*, International
Standard ISO 1000

(background data on the SI system and detailed lists
of every recognized SI unit by application)

* *Units and Conversion Tables for Process Control*,
Foxboro Publication 73
(engineering data such as Properties of Steam and
Saturated Water, Liquid Gravity Tables and Weight
Factors, Stainless Steel Composition)

* These books and articles were written by experts at Foxboro and
are available through your Foxboro sales representative. For other
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