

ENGINEERING DESIGN SEMINAR

SEPTEMBER 6, 1994 – NOVEMBER 11, 1994

INSTRUMENTATION

Table of Contents

INTRODUCTION	1
FEEDBACK CONTROL LOOP	1
History	1
Loop Components	2
Process	3
Measuring Means	5
Temperature	6
Flow	12
Pressure	23
Level	23
Analysis	27
FINAL CONTROL ELEMENT	32
Control Valve Characteristic	45
Hydraulics	47
Control Valve Sizing Formulas	61
CONTROLLER	65
Transmission Means and Response	65
Pneumatic Transmission	66
Digital Transmission	72
Control Modes	74
Controller Tuning	90
SYSTEM REQUIREMENTS	92
ADDITIONAL TOPICS	94
CASCADE SYSTEMS	94
MULTIPLIERS AND SUMMERS	96
SIGNAL SELECTORS	100

Table of Contents (Cont'd)

FEED FORWARD/LEAD-LAG SYSTEMS	107
REBOILER STEAM CONDENSATE CONTROL	108
FRACTIONATOR CONTROLS	111
Heat Input Control	111
Gibbs Phase Rule	111
Pressure Control	112
Total Condensation, Below 5 psig	112
Total Condensation, 5 to 25 psig	113
Total Condensation, More Than 25 psig	113
Total Condensation, Steam Generation	115
Partial Condensation	116
Composition Control	116
Material Balance Control	121
 APPENDIX	
PERFORMANCE AND MEASUREMENT TERMINOLOGY	A-1
ORIFICE BETA RATIO LIMITS	A-5
MEMORANDUM	A-7
SAMPLE CALCULATIONS	A-17
 SIZING STEAM REBOILER VALVES	 A-33

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.

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

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

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.

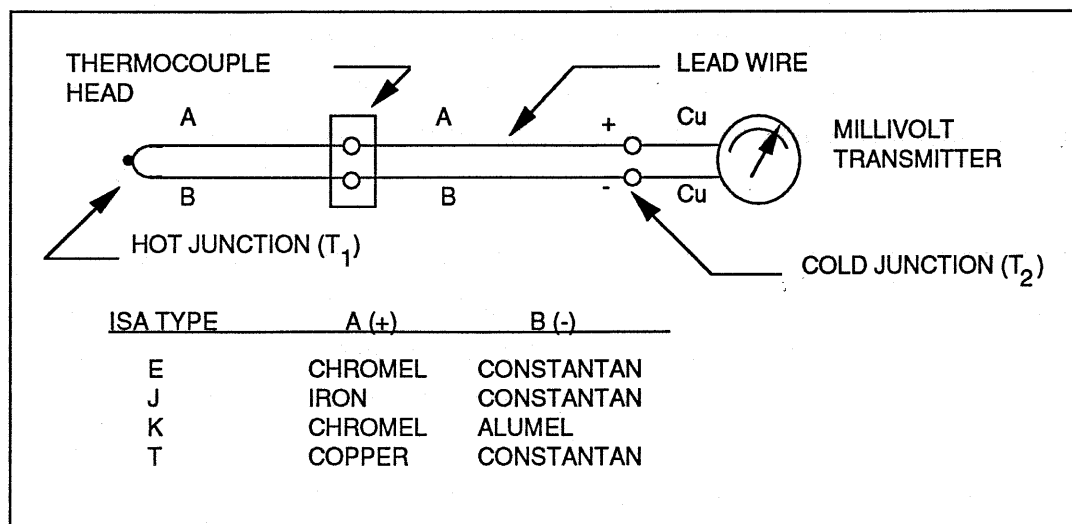


Figure 3 - Thermocouple

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. 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.

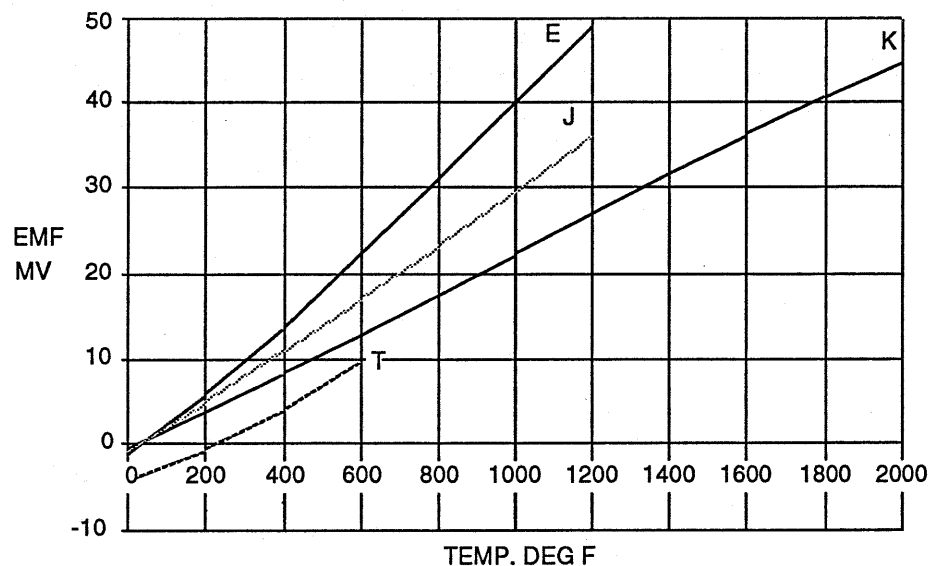


Figure 4 - Thermocouple EMF

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 ± 3 °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

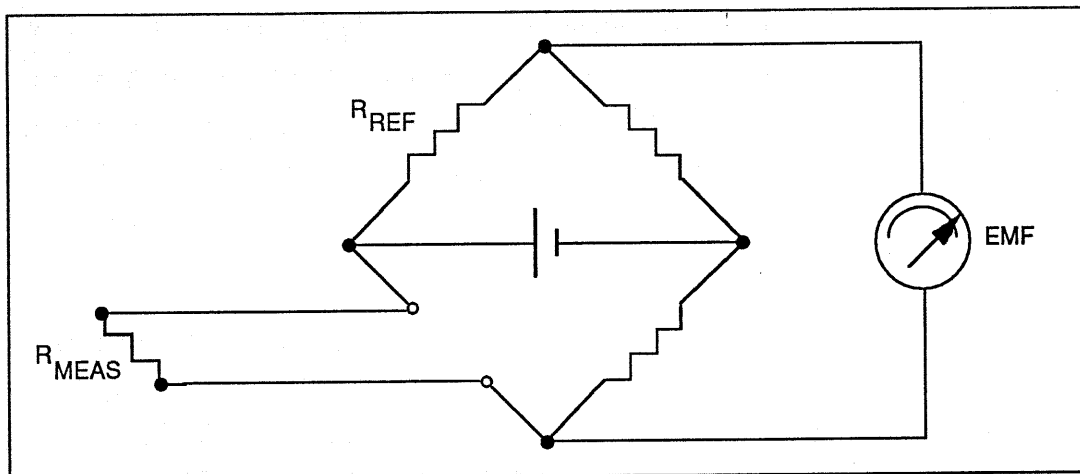
control room. UOP does not feel that this is a good solution. With this design, the cold junction circuitry in the transmitter is exposed to wider ranges of ambient temperature than if the transmitter were located in the control room. Also, any test equipment used to service the transmitter in the field 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.

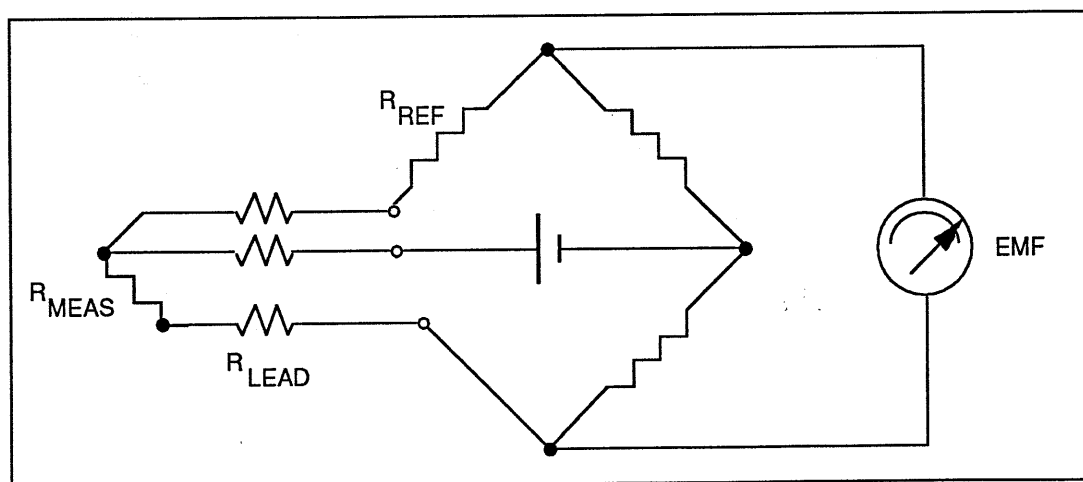
For narrower temperature spans, down to about 5 to 10 °F (2.8 to 5.6 °C), resistance temperature detectors are usually used. The Resistance Temperature Detector, or RTD, is in essence a very long, thin wire - usually nickel or platinum - which has been wound into a small coil. The wire has a stable, 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.



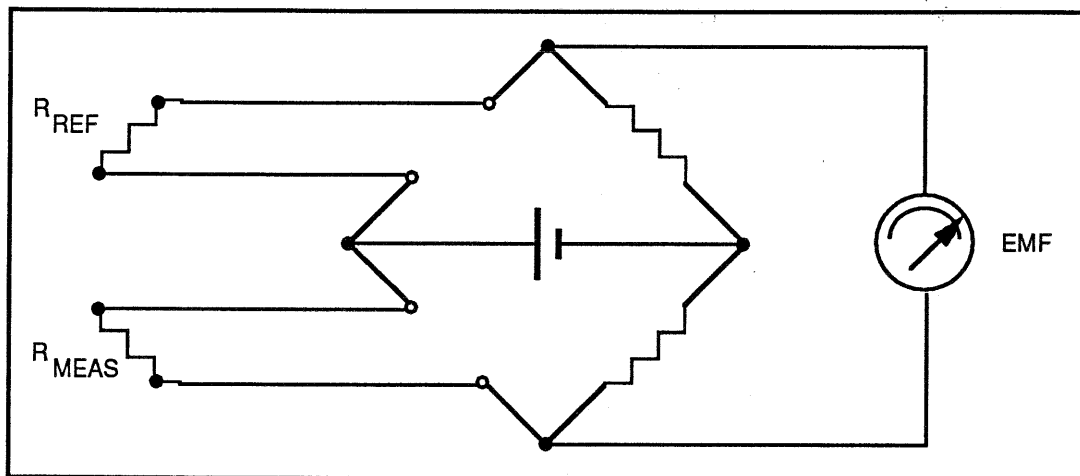
**Figure 5 - Resistance Temperature Detector
Wheatstone Bridge Circuit**

Resistance temperature detectors are more expensive than thermocouples, are not as rugged, 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 temperature measurement, in which one element is used to measure a reference



**Figure 6 - Resistance Temperature Detector
Three-Wire Circuit**



**Figure 7 - Resistance Temperature Detector
Differential Temperature Circuit**

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.

A filled thermal system is another type of primary device used for temperature measurement. 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 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

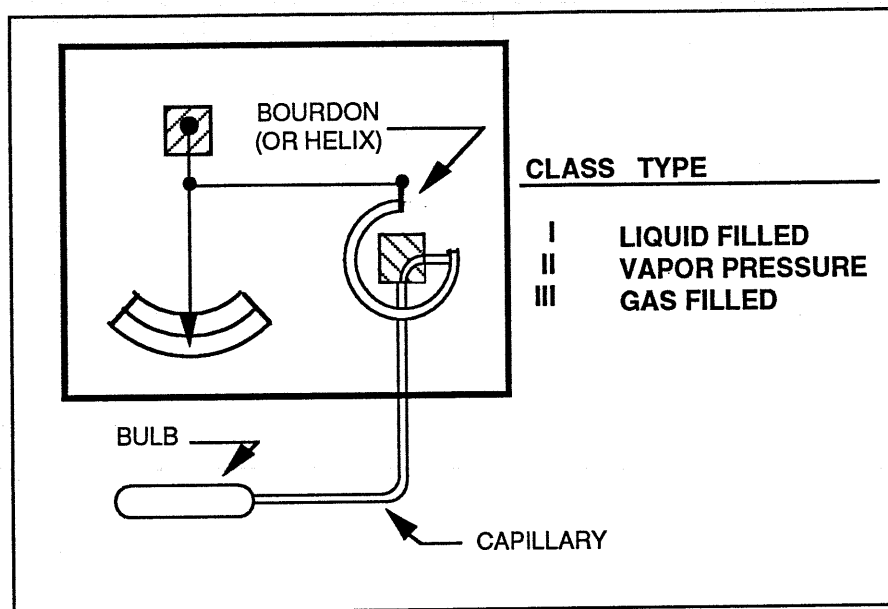


Figure 8 - Filled Thermal System

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 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 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

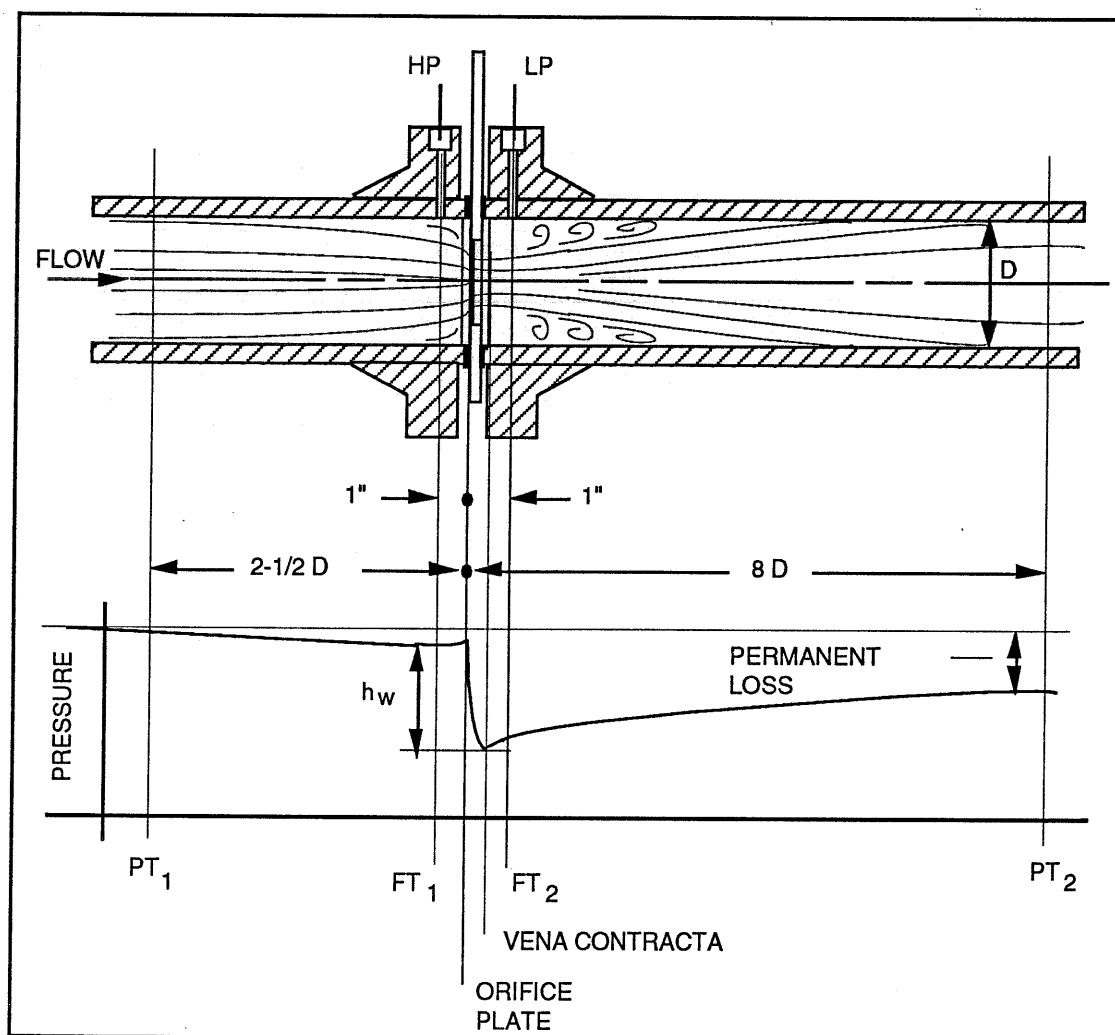


Figure 9 - Orifice Pressure Profile

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 lead 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 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 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

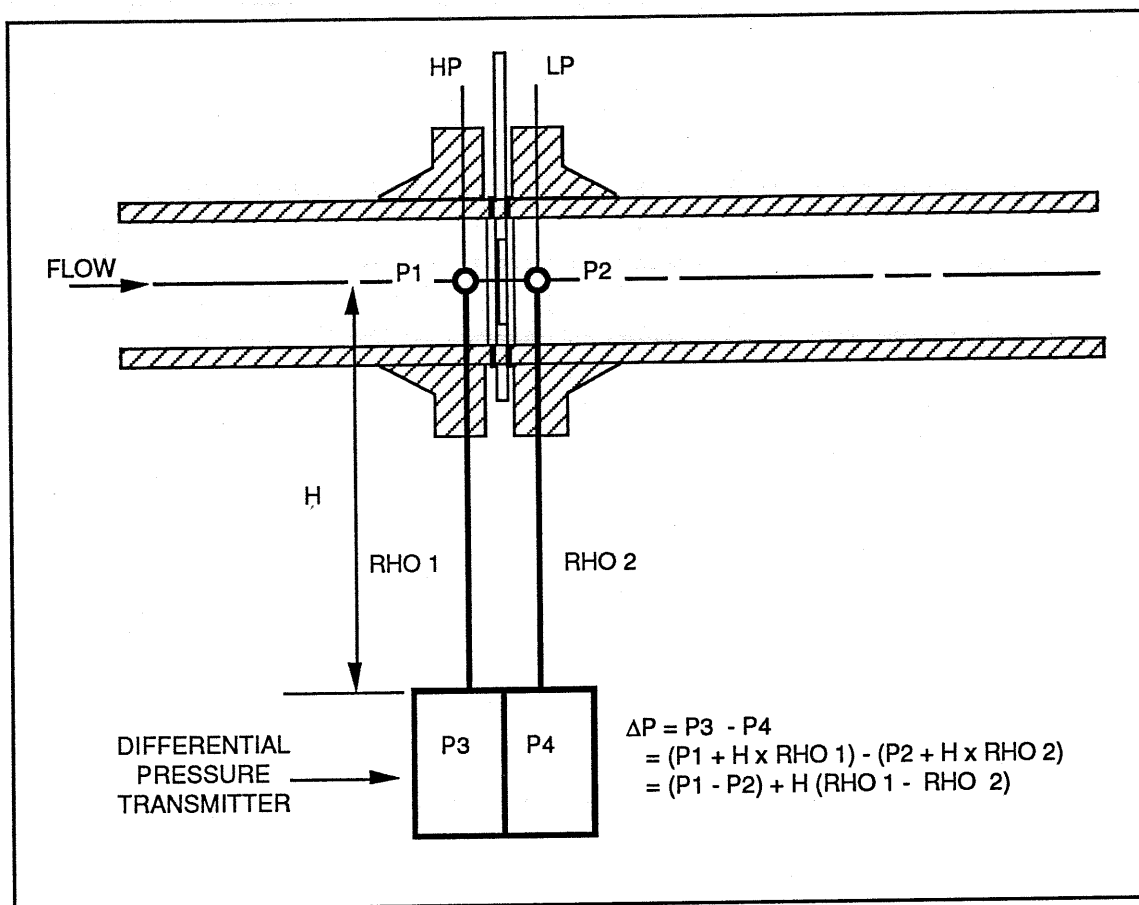


Figure 10 - Transmitter Static Head Effects

inches of H₂O or the nearest even metric equivalent (0 to 2500 mm H₂O, 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 H₂O. 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 H₂O, the permanent loss is 26 inches of H₂O, 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

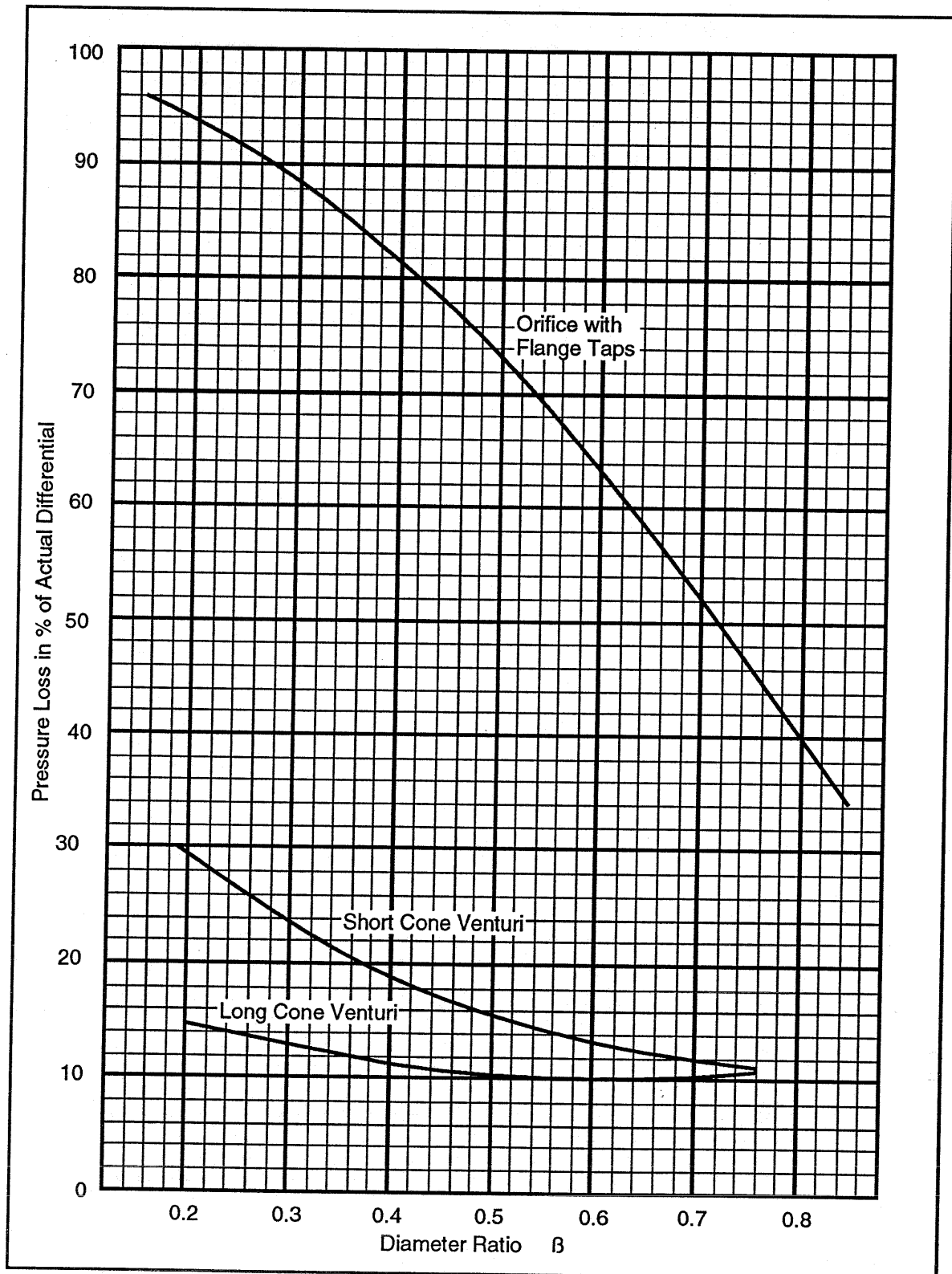


Figure 11 - Overall Pressure Loss

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 at the end of this section. The equations used in the memo to generate the numbers in the table illustrate how the basic orifice 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

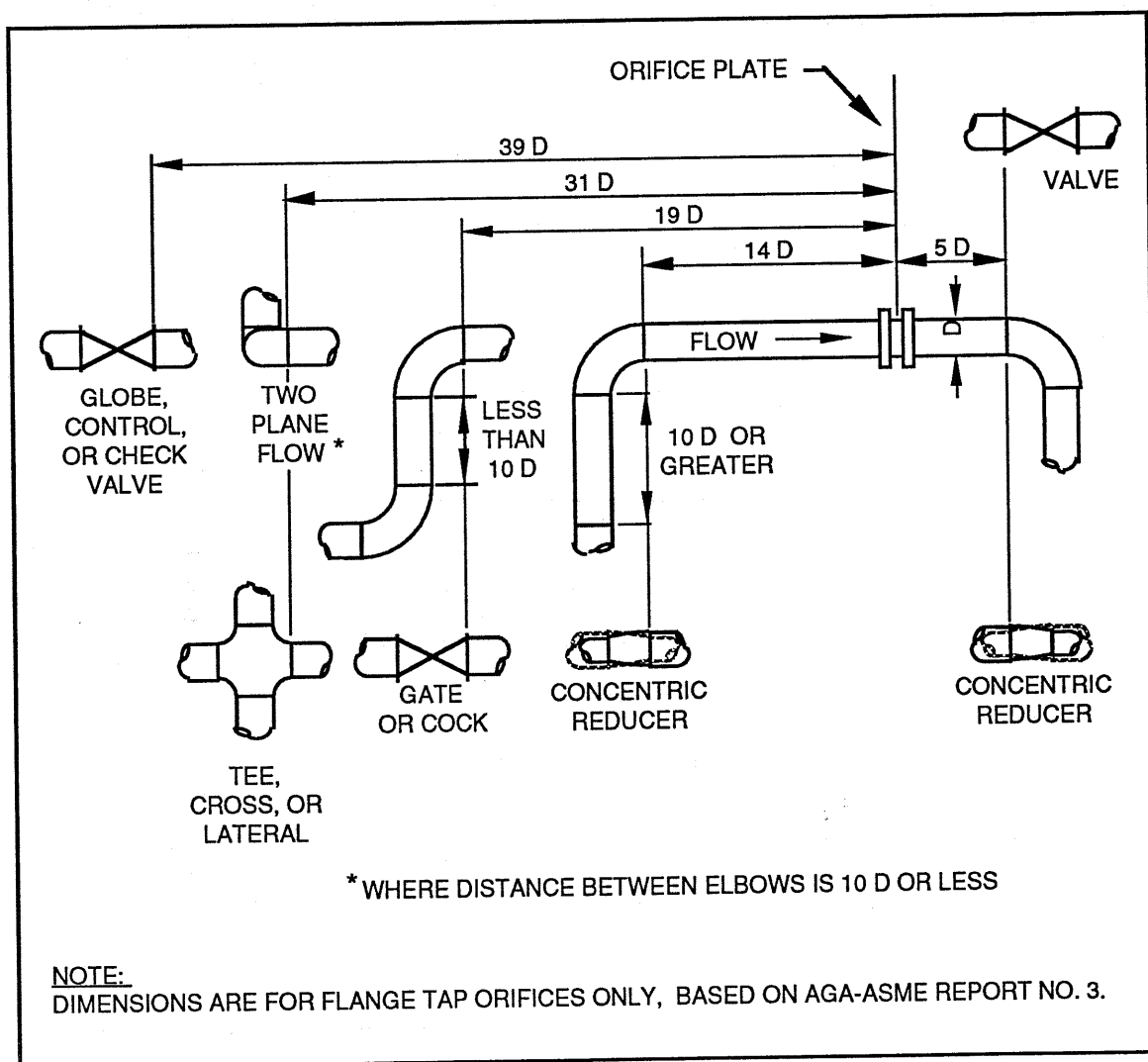


Figure 12 - Orifice Minimum Straight Run Requirements

separation
t h a t
o c c u r s
w i t h a n
o r i f i c e
p l a t e . A s
a r e s u l t ,
t h e
s t a t i c / k i n e t i c
e n e r g y
e x c h a n g e
i s m o r e
n e a r l y

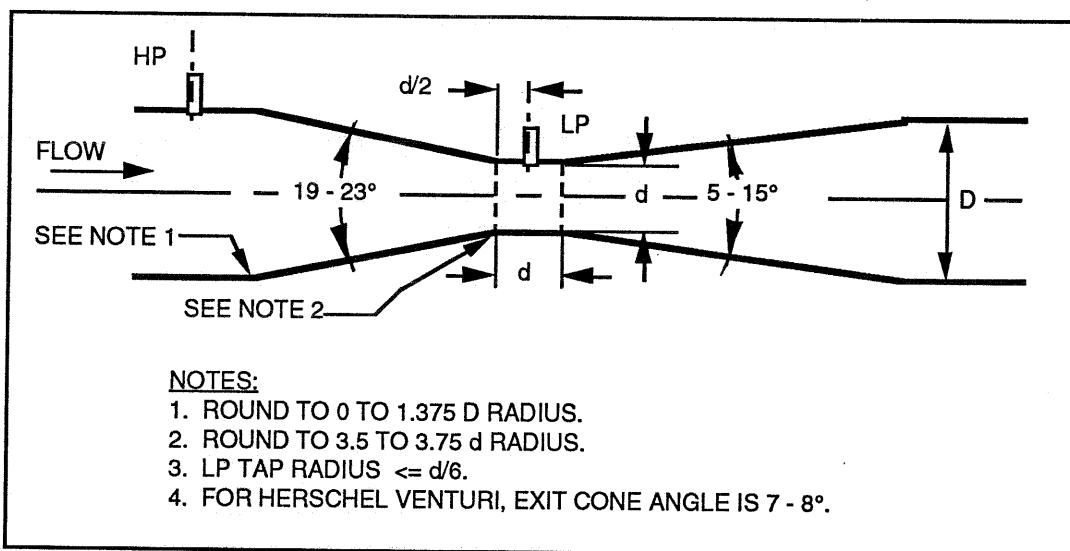


Figure 13 - Dimensions of Classical Venturi Tube

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.

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

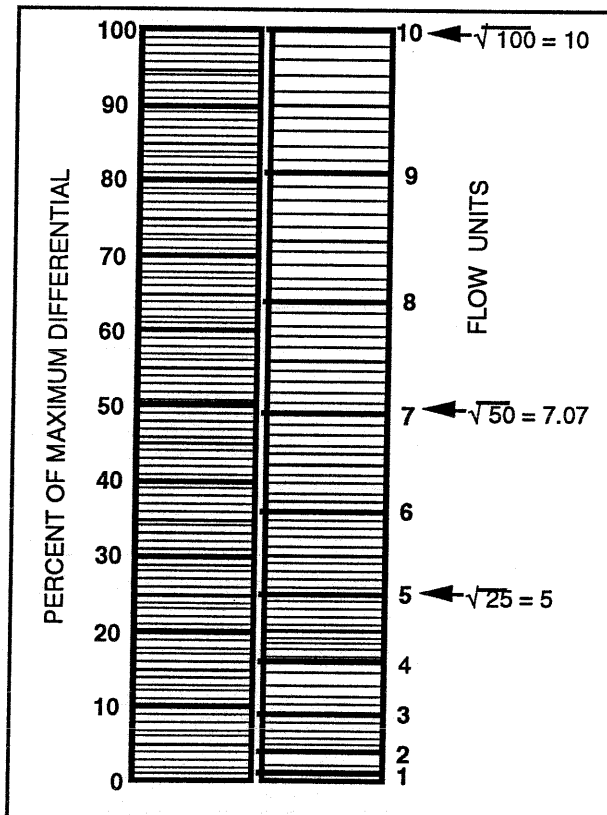


Figure 14 - Square Root Relationship Head vs. Flow

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 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 Doppler 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.

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 condensable 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 condensable 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 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 or compressor. A few manufacturers have transmitters

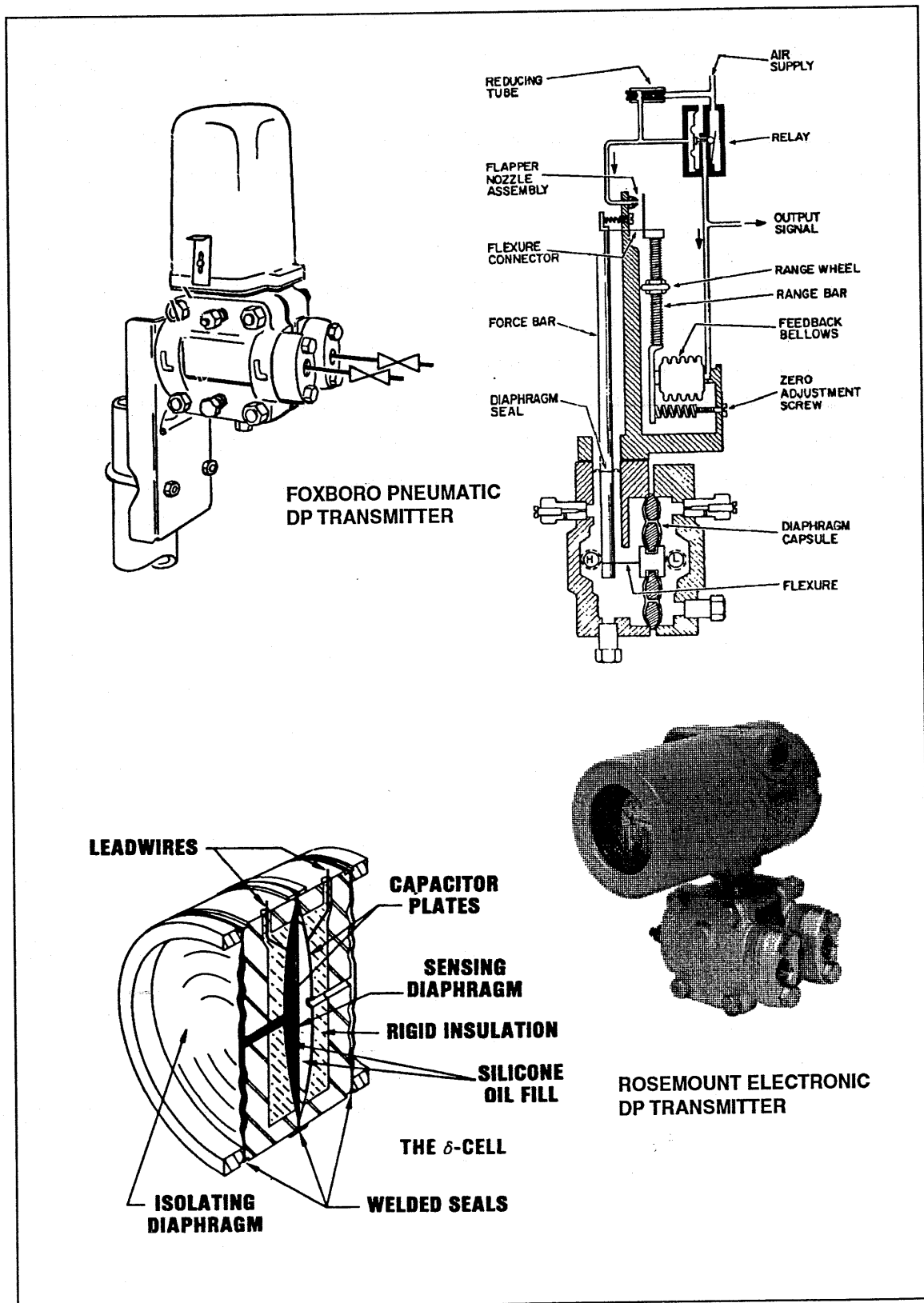


Figure 15 - Diaphragm Type Differential Pressure Instrument

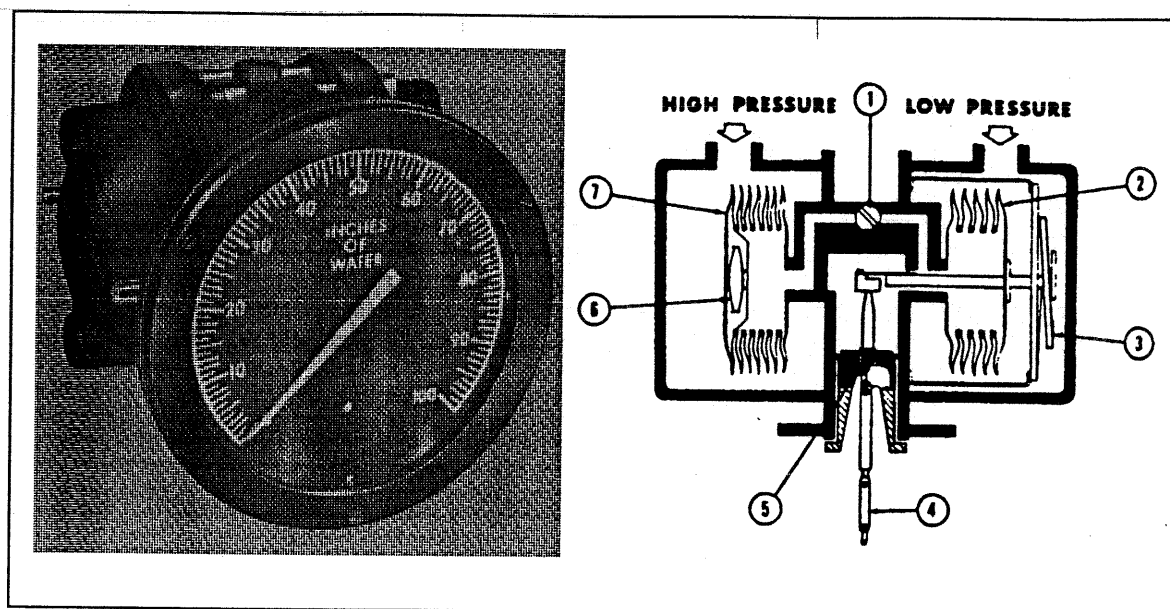


Figure 16 - Bellows Type Differential Instrument

(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

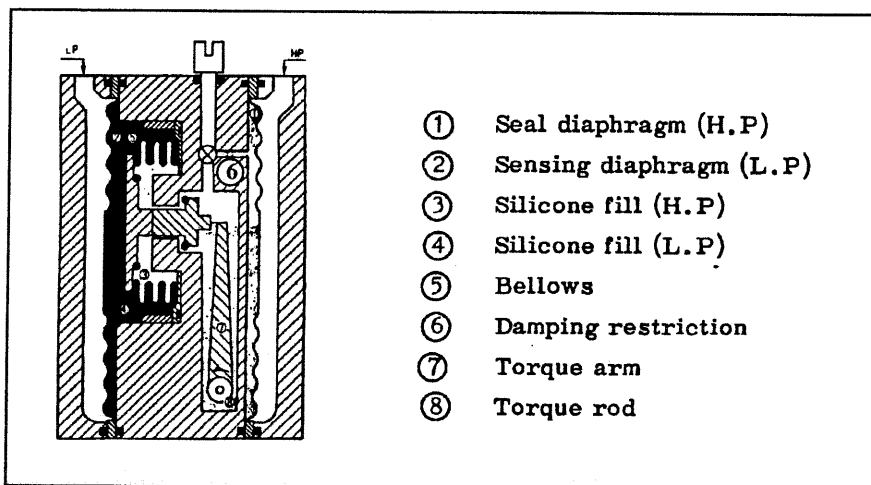


Figure 17 - Diaphragm Type Differential Instrument (with Dampening)

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 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

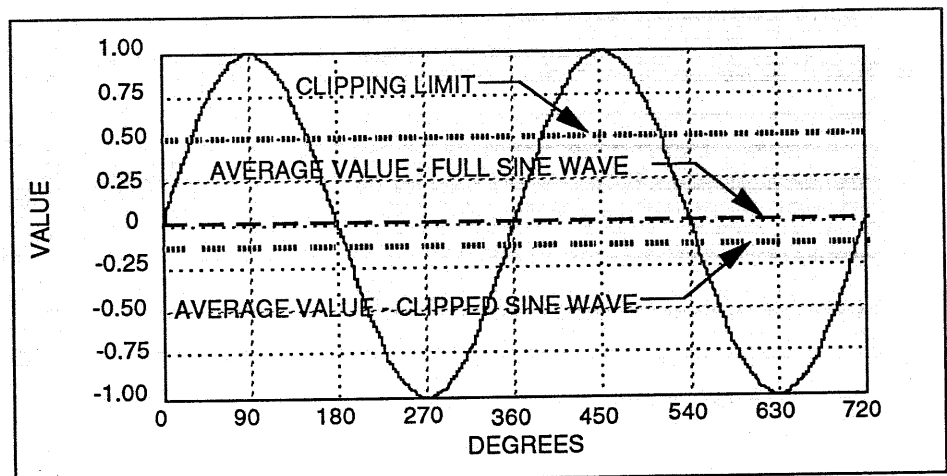


Figure 18 - Sine Wave Clipping

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.

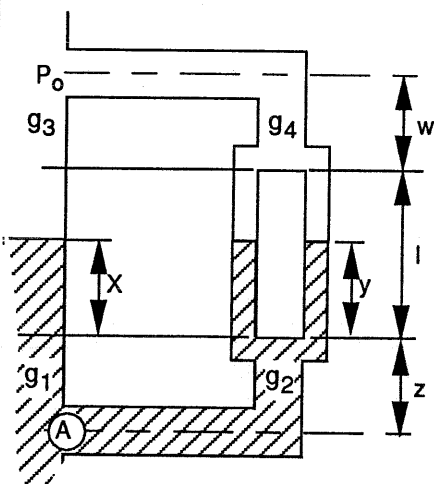
For a displacer in single phase hydrocarbon service, the difference in effective weight over the range of the instrument is the difference between the weight of the displacer alone at 0%, and the weight of the displacer minus the weight of the hydrocarbon displaced at 100%. The weight of the displacer 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 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 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



g = SPECIFIC GRAVITY
 l = LENGTH OF DISPLACER
 w = TOP NOZZLE $\downarrow$ TO TOP OF DISPLACER
 z = BTM NOZZLE $\downarrow$ TO BTM OF DISPLACER
 x = LEVEL IN VESSEL
 y = LEVEL IN DISPLACER
 PB = PROPORTIONAL BAND = 100/GAIN
 g_{set} = SETTING ON SPECIFIC GRAVITY ADJUSTMENT
 P_o = VESSEL PRESSURE AT TOP NOZZLE $\downarrow$

- 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

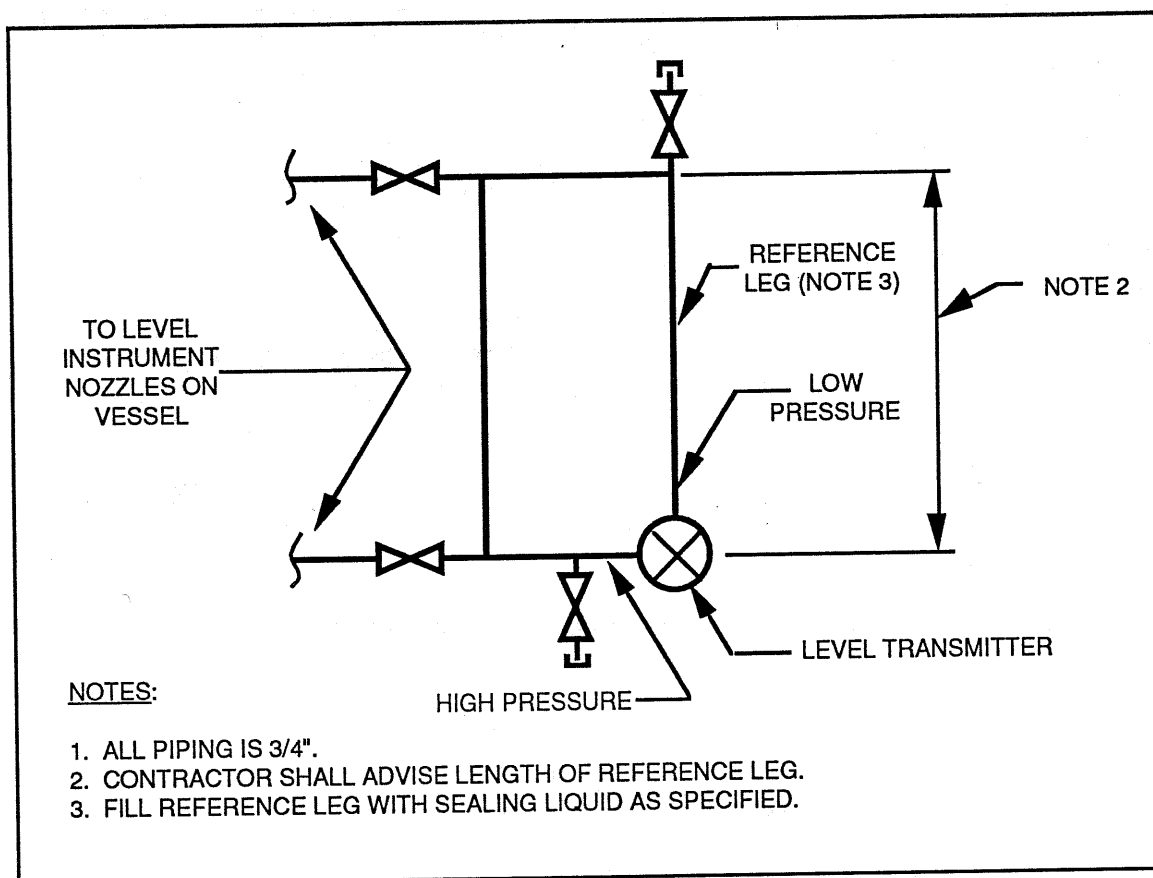


Figure 20 - Differential Pressure Level Transmitter

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

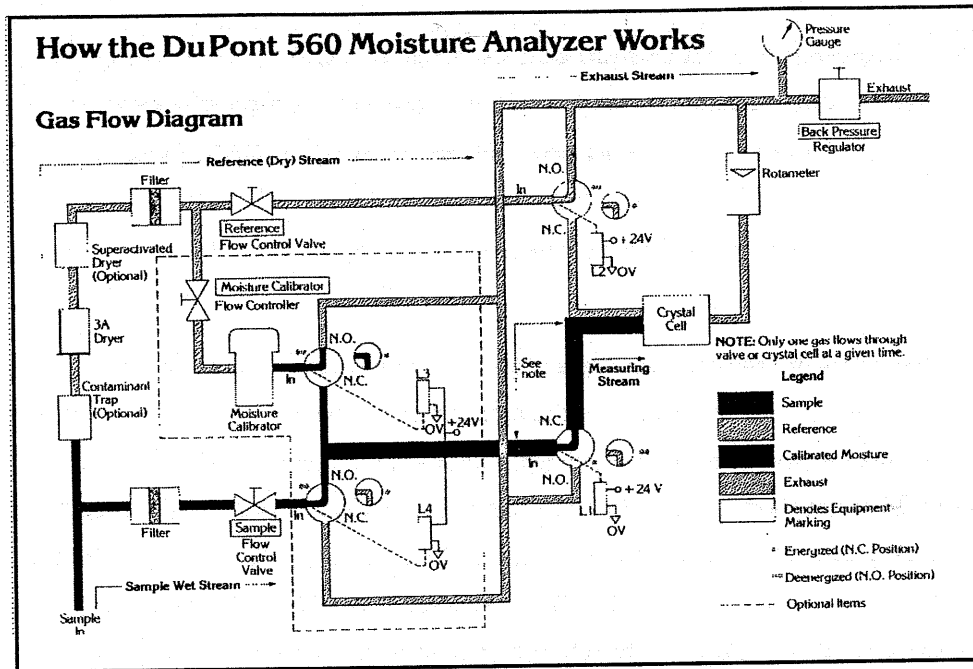


Figure 21 - DuPont Moisture Analyzer

calibration of the analyzer. On the newer DuPont analyzers, a moisture generator is built into the 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 surface area and increases

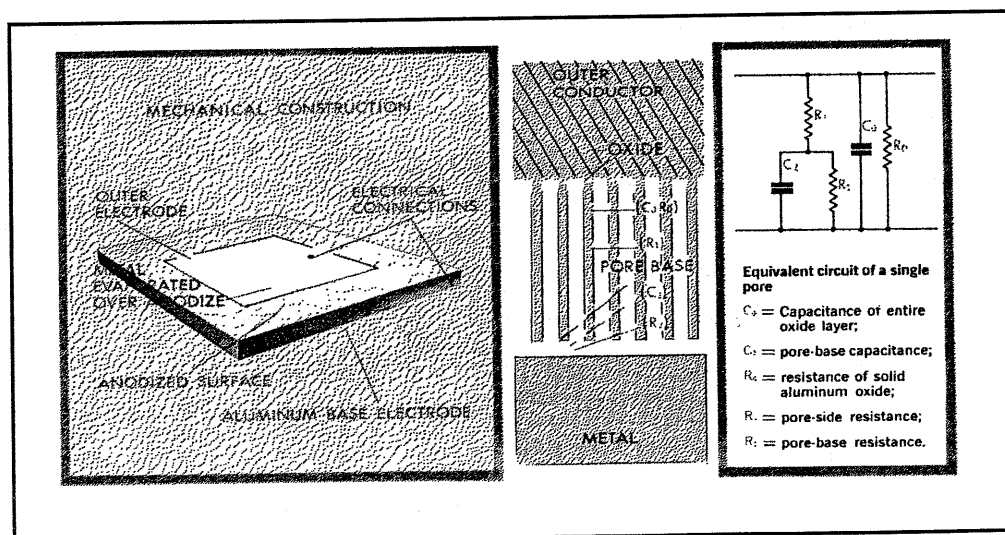


Figure 22 - Panametrics Moisture Analyzer

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 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

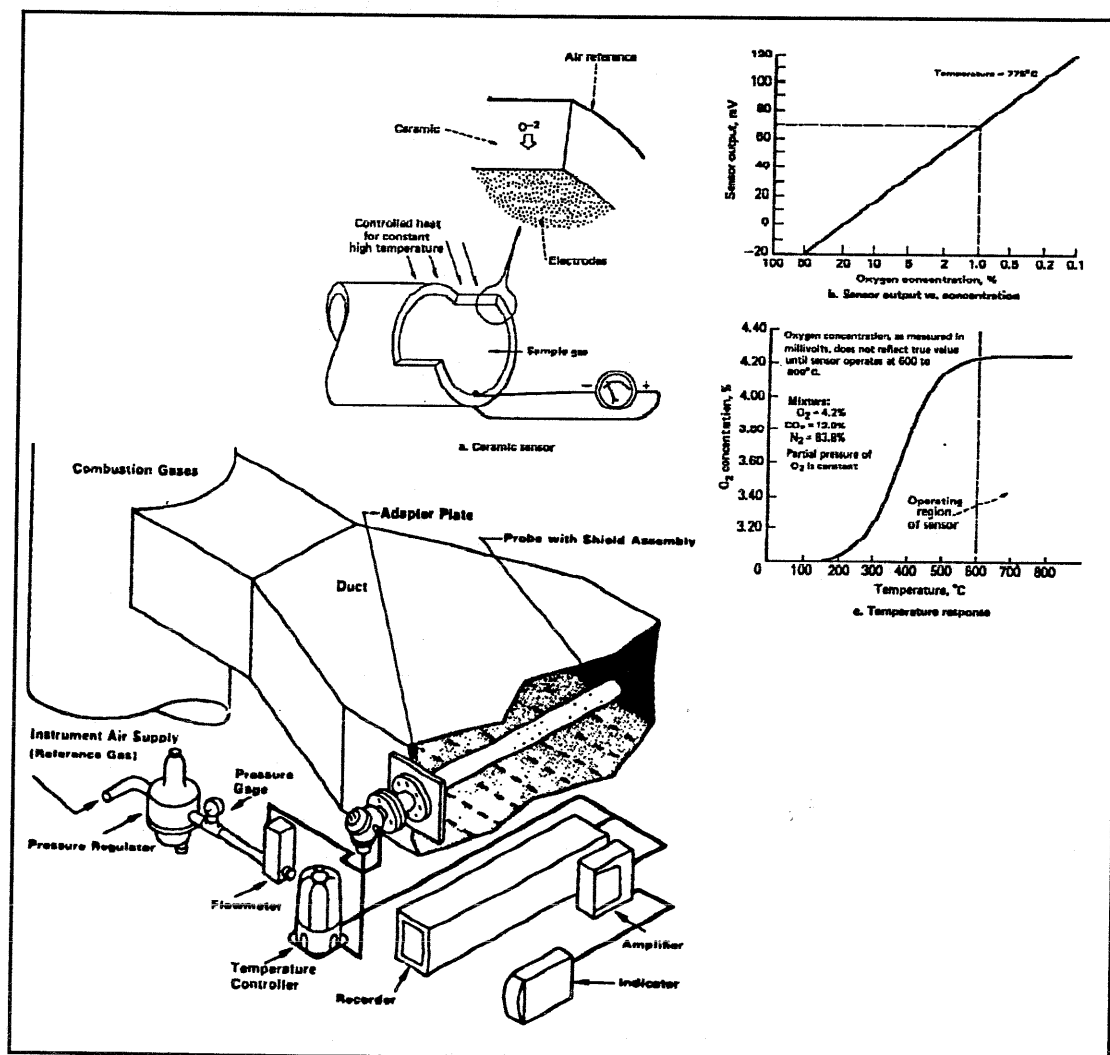


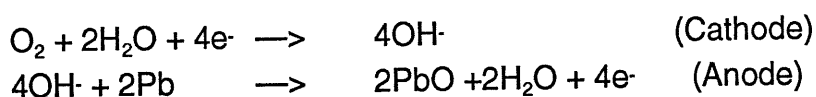
Figure 23 - Zirconium Oxide Oxygen Analyzer (Westinghouse shown)

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 Platforming CCR units, is no longer available. A Thermo 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).

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, 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.

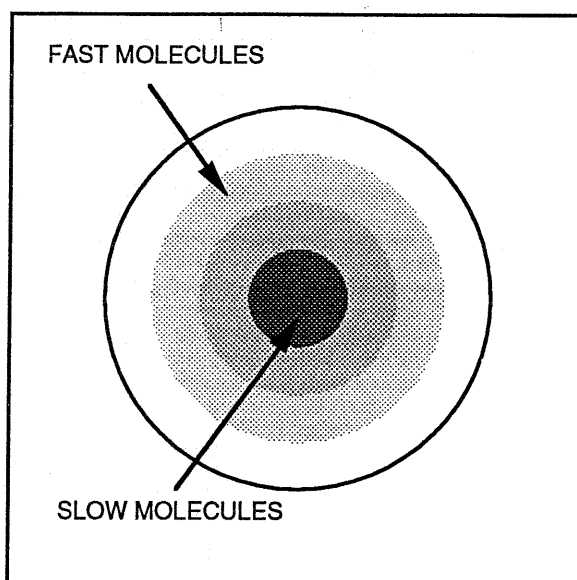


Figure 24 - Paper Chromatography

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

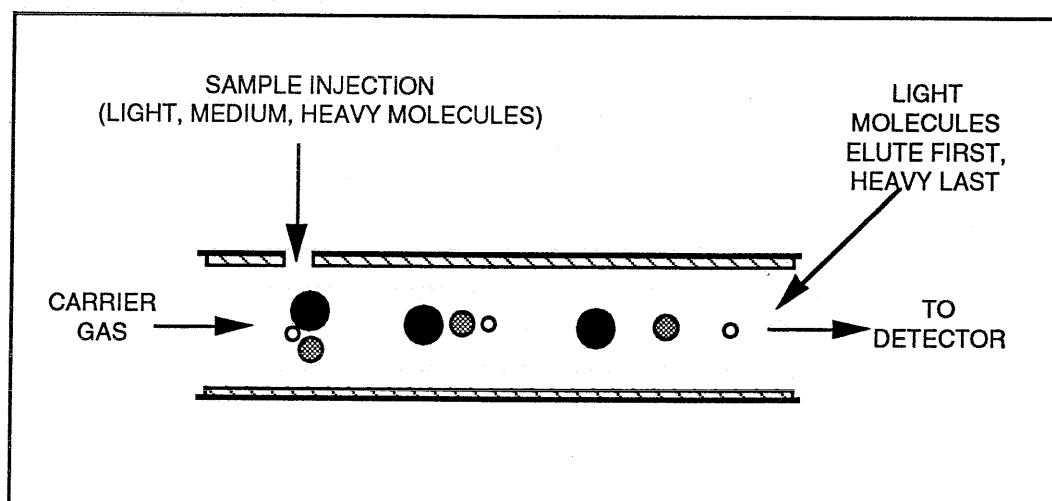


Figure 25 - Gas-Liquid Chromatography

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 studies 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

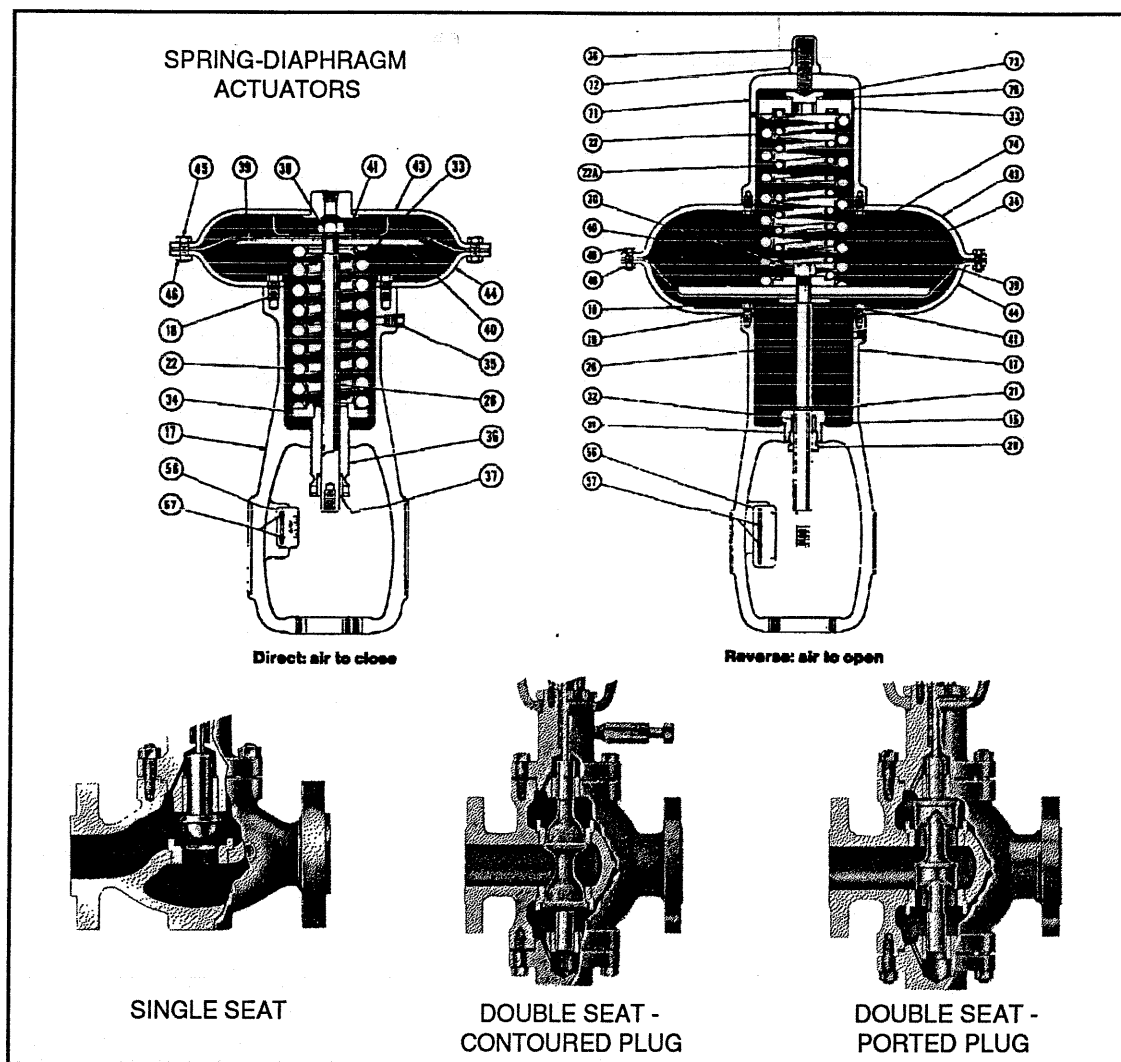


Figure 26 - Globe Type Control Valves

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 of the valve trim, but has the

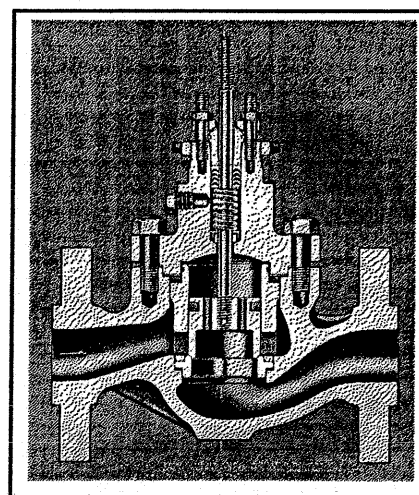


Figure 27 - Cage Type Globe Valve

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 completely 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).

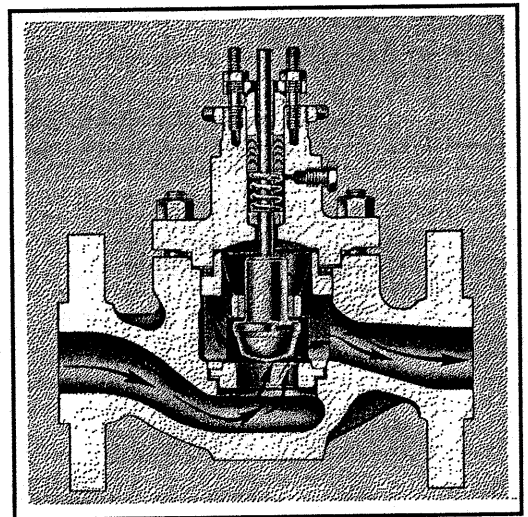


Figure 28 - Fisher Design "EZ"
Globe Valve

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 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.

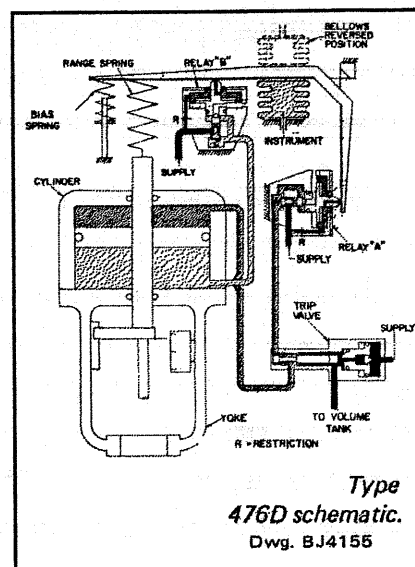
The convenience lies mostly in knowing what control input will just open the valve when the loop is being commissioned. But even if the opening point is correct for one set of operating conditions, changing the operating conditions will change the opening point. The only way to insure that the valve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



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)

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 promoting an actuator design for globe valves called the "SigmaF" 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.

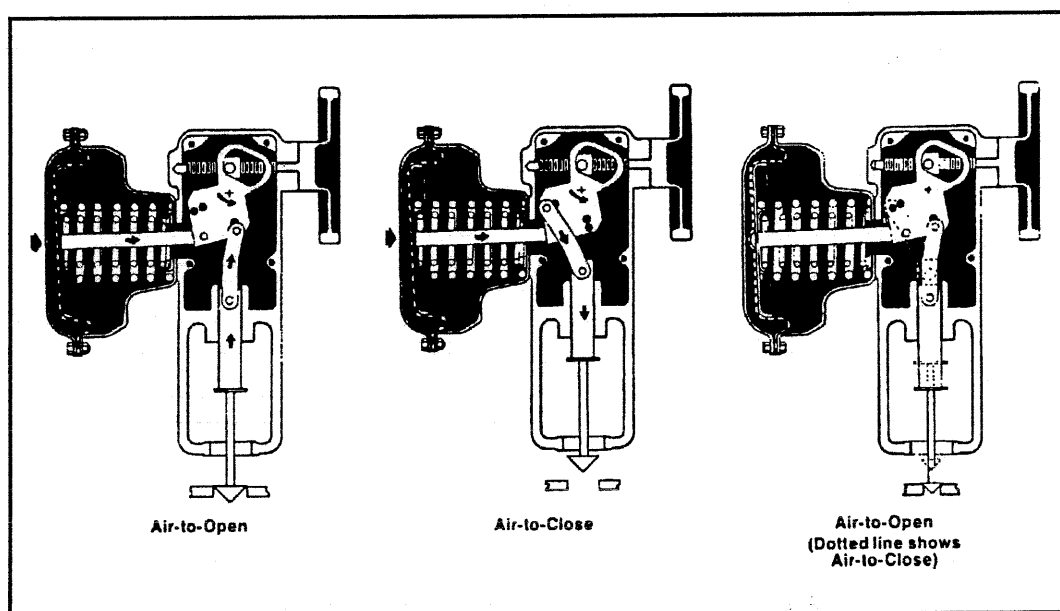
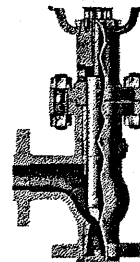


Figure 30 - Masoneilan Σ F Actuator

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 standard globe

71000 Series



Critical Flow Factor – C_v

Trim Size	Flow to Close	Flow to Open
Full Area	.48	.90
.5 factor	.55	.95

300-1500 lb. ANSI (flow to close)

Nominal Trim Size	Full Area	1	1½	2		3	4		6			
	.5 Factor	1	1½	2	3	4		6				
Orifice Dia. (in.)		.500	.750	.812	1.125	1.562	1.687	2.250	2.375	3.187	3.375	4.750
Valve Size (in.)	Reduced Trim	Full Capacity Trim										
1"		11	25									
1½			29	57								
2				57	110							
3						128		254				
4							228		450			
5											510	1000

*300-600 lb. ANSI only

Refer to Bulletin 348E

300-1500 lb. ANSI (flow to open)

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

Valve Size (in.)	Reduced Trim	Full Capacity Trim										
1"	6.3	13.5										
1½			17	30								
2				33	60							
3						74		135				
4							132		240			
5										295	540	

*300-600 lb. ANSI only

Refer to Bulletin 348E

■ Flow Coefficient – C_v
■ Critical Flow Factor – C_v

Figure 31 - Maseonilan 71000 Series Angle Valve

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 combining service, in which two streams enter and one stream exits. Either of these

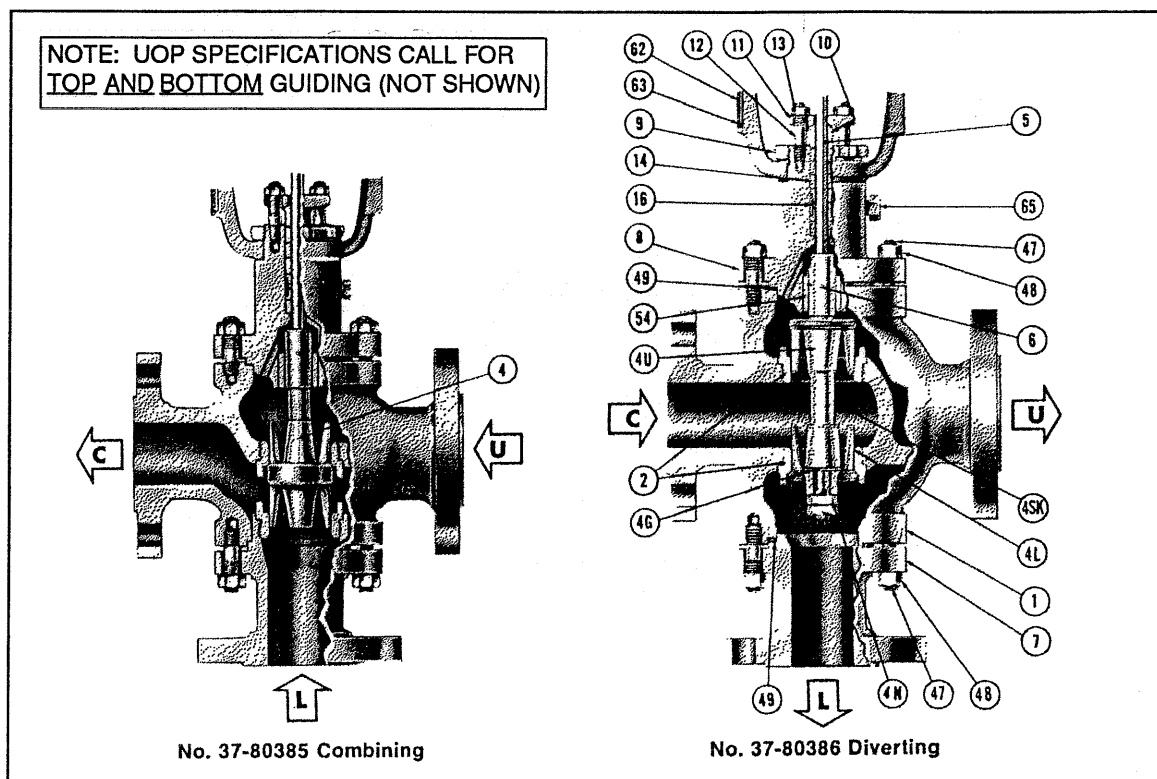


Figure 32 - Three-Way Valves

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 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.

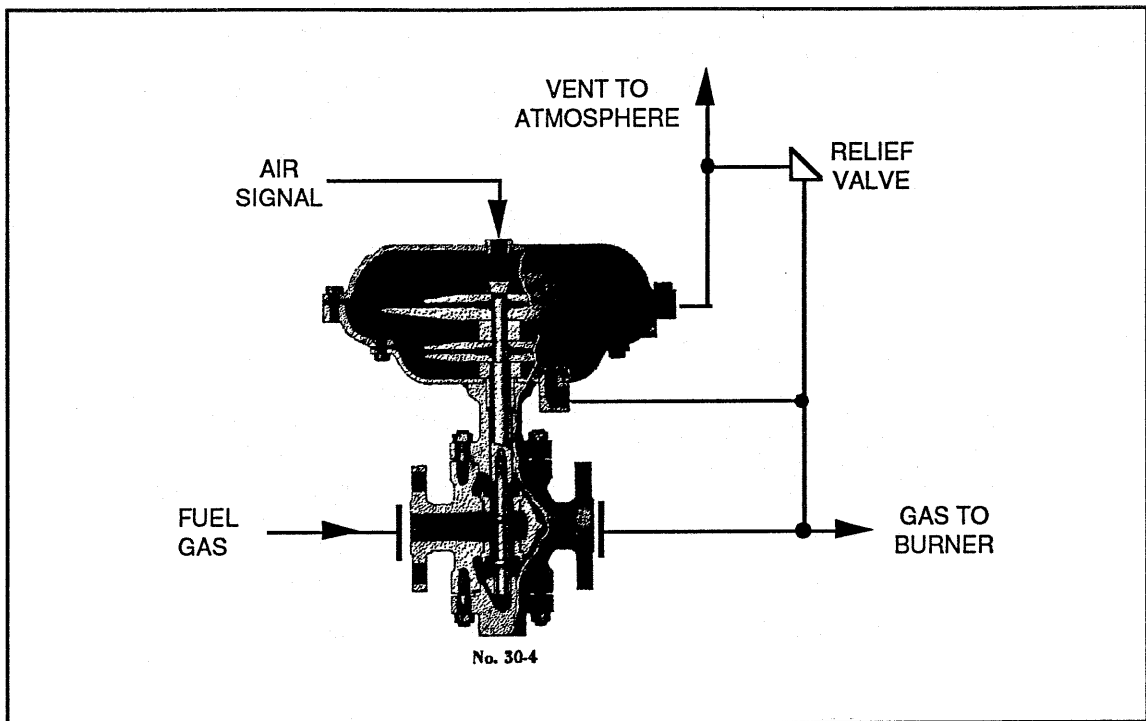


Figure 33 - Double Diaphragm Fuel Gas Valve

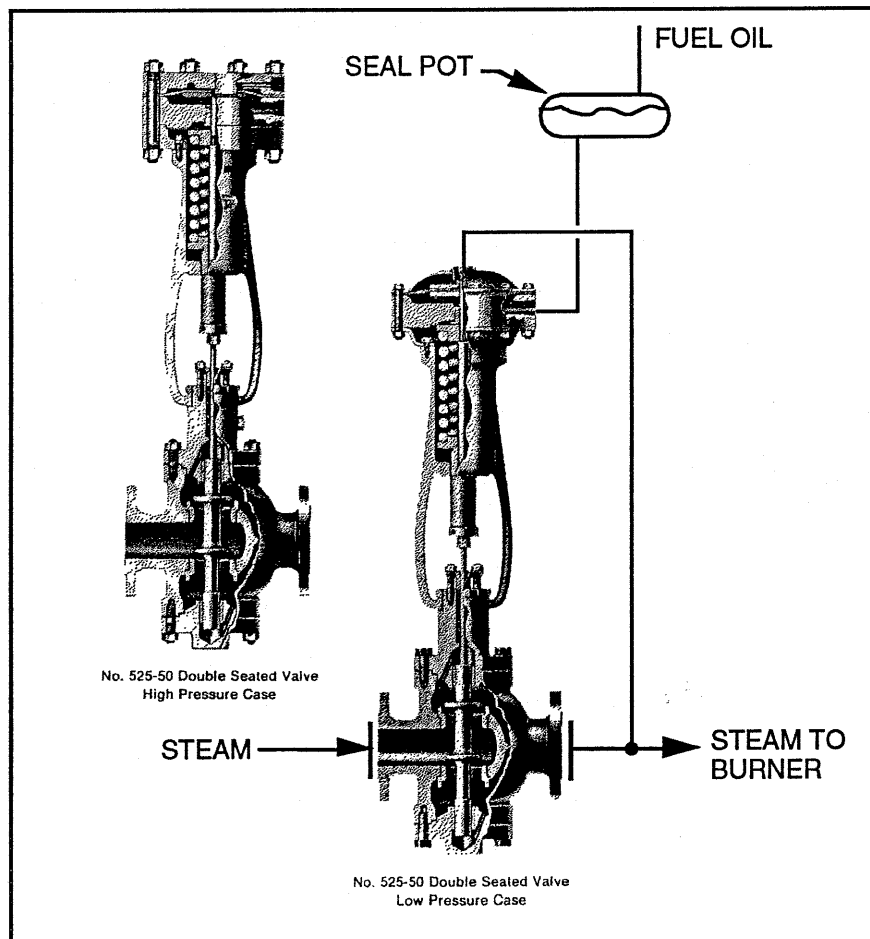


Figure 34 - Atomizing Steam Regulator Valve

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. 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.

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

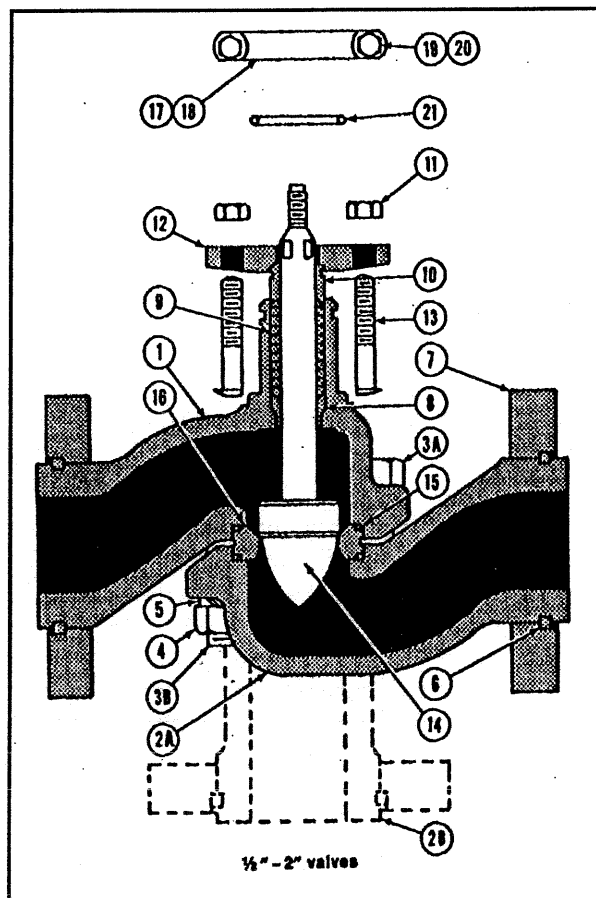


Figure 35 - Split Body Valve

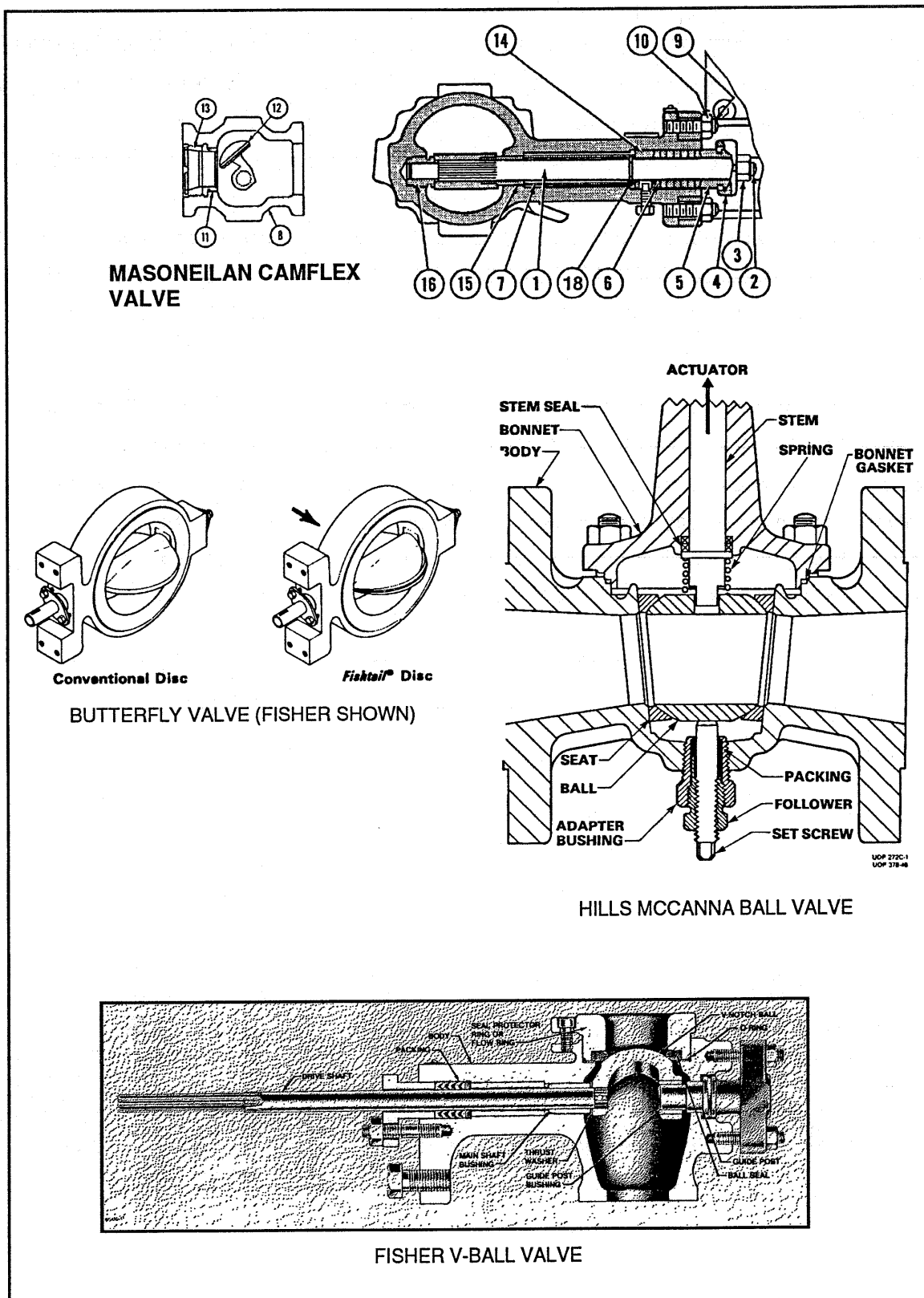


Figure 36 - Rotary Type Control Valves

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 F 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 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

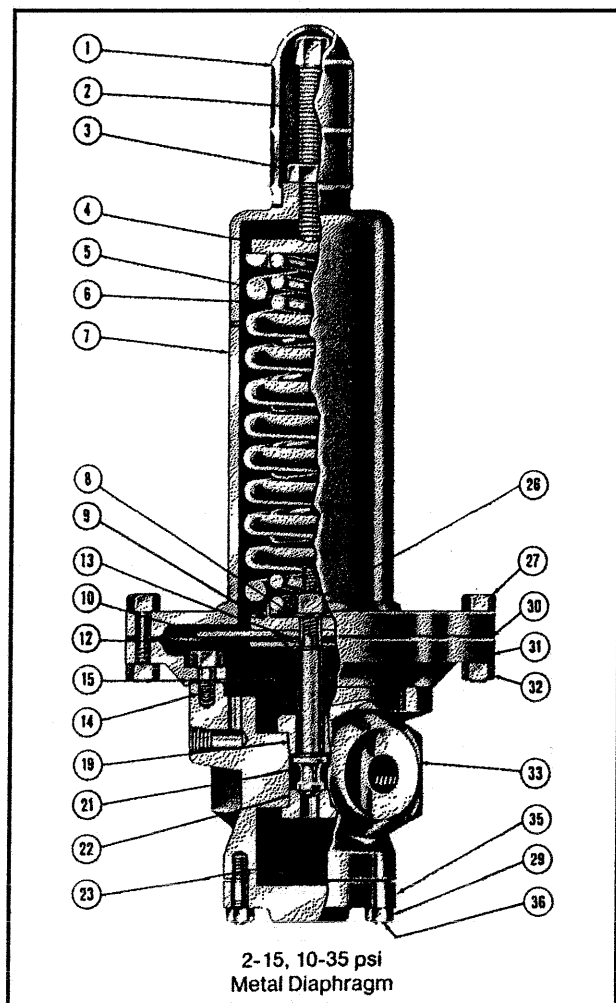


Figure 37 - Masoneilan Regulator (Pilot Gas)

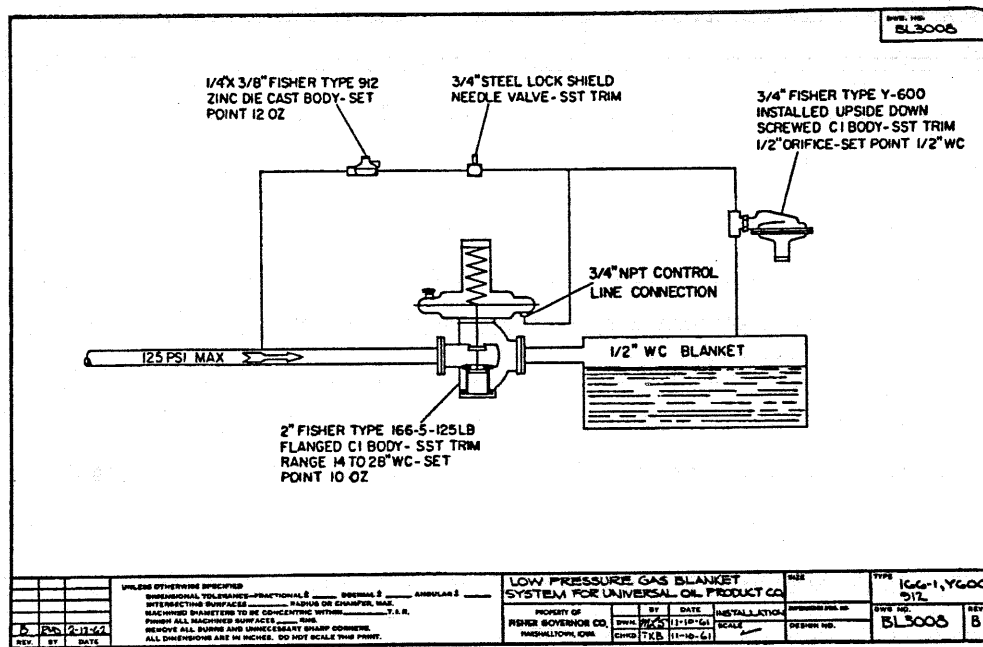


Figure 38 - Tank Gas Blanket Assembly

assembly (Figure 38).

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 H₂O. 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 H₂O 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.

The set point of the 166-5 regulator is 10 ounces, or 17-inches H₂O. A Fisher Type 912 regulator, set at 12 ounces, or 21-inches H₂O, 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 H₂O 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 H₂O, 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.

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.

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

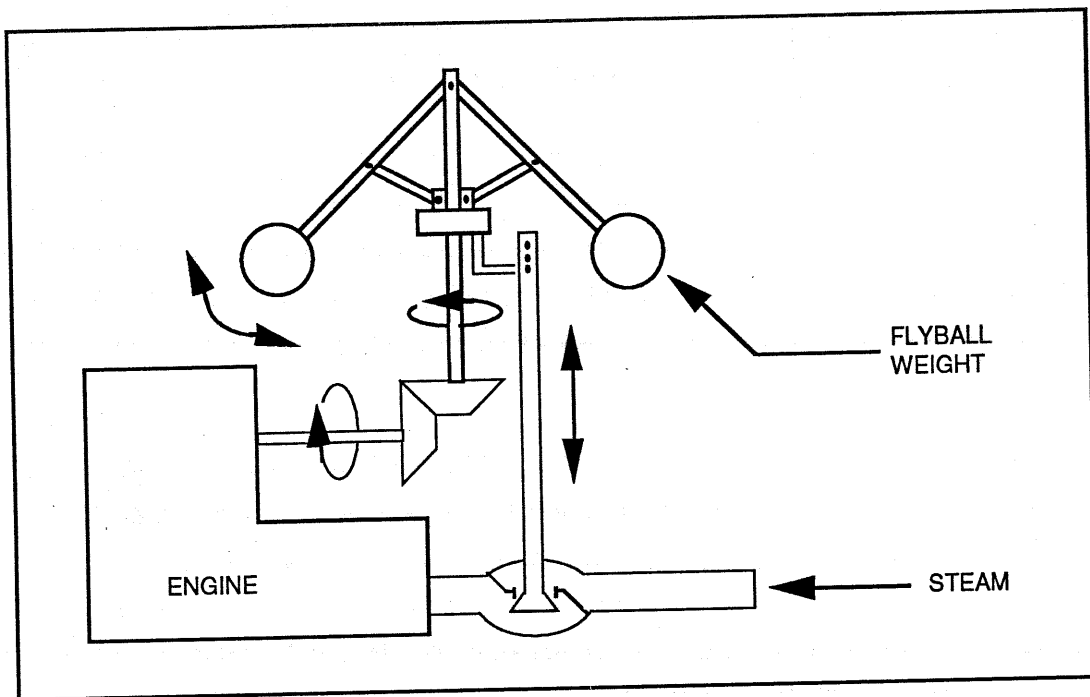


Figure 1 - Flyball Governor

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) and (2) are batch operations, rather than continuous. However, all four examples contain the elements which are diagramed 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 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

Prop. -
 Integ. -
 Deriv. -

PID

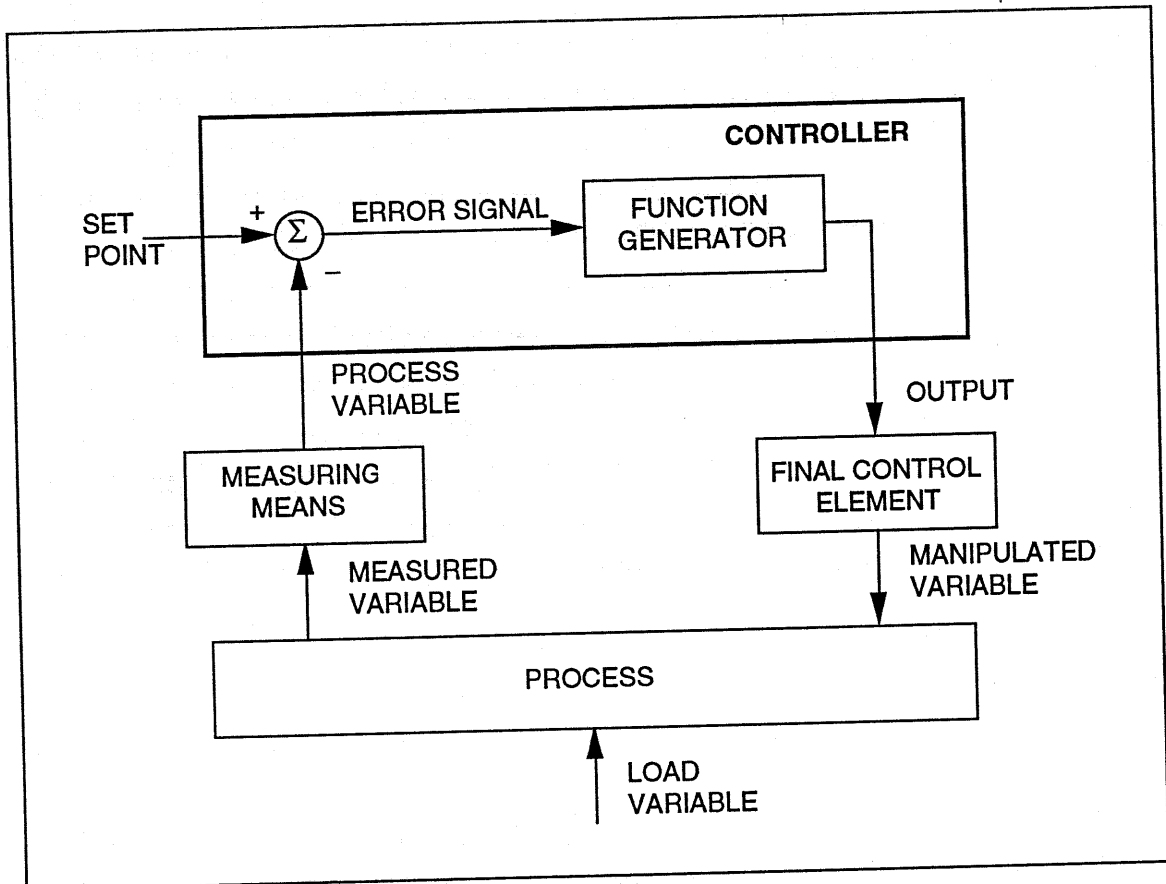


Figure 2 - Feedback Control Loop

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

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 H₂O 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 H₂O 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 H₂O, 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 H₂O, 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 H₂O, and the regulators will close. With the gas flow stopped, the pressure at the sensing point will fall below 1/2-inch H₂O, 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 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

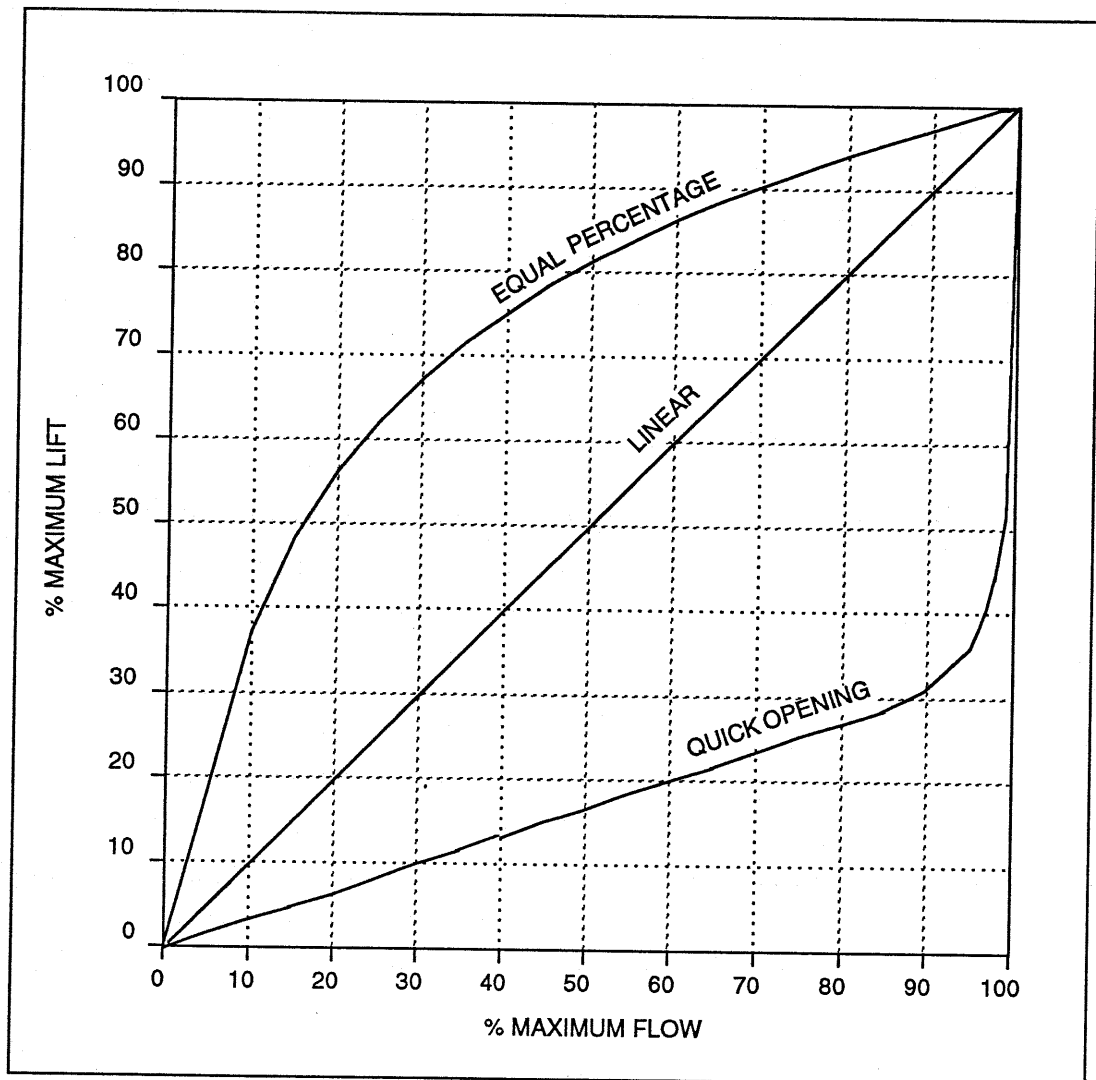


Figure 39 - Control Valve Characteristics

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 the normal flow. As noted earlier, this is less than 122% because the pressure drop of the

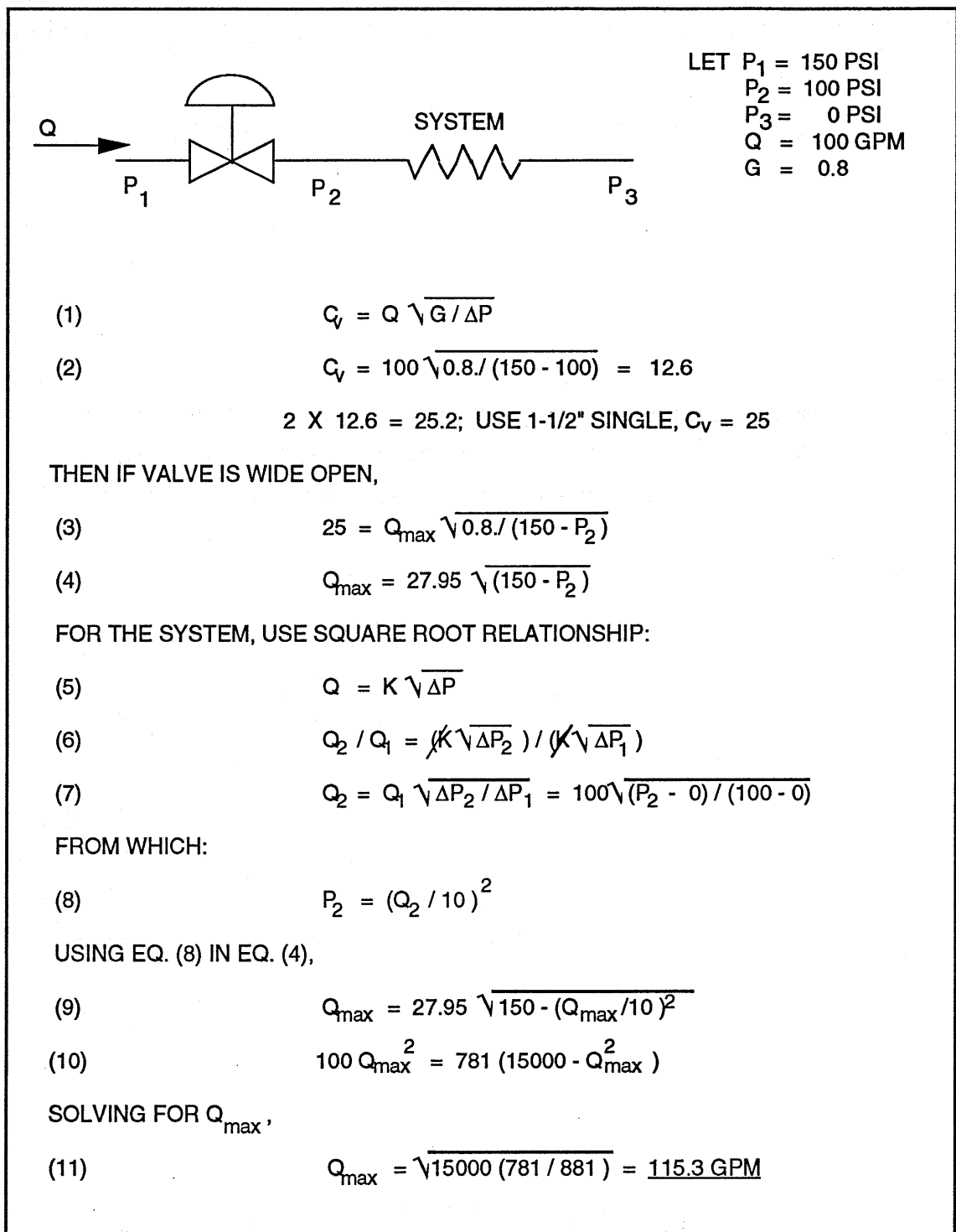


Figure 40 - Maximum Flow with 50% of Friction Drop for Valve
 wide-open valve is not zero.

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.

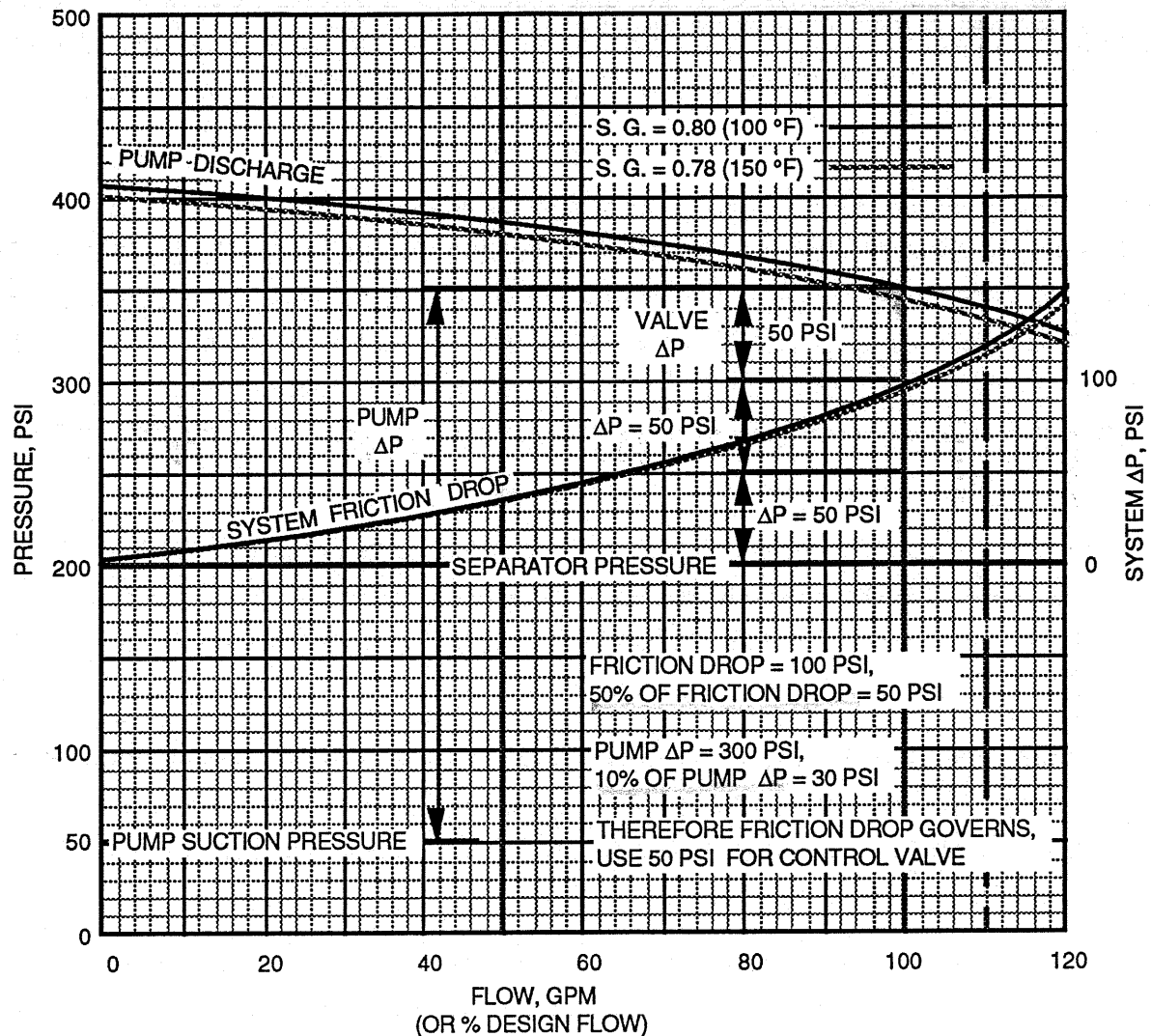
In the case of Figure 41, the process is assumed to be a relatively low pressure process, such as a Platforming Unit. In this case, doubling the valve C_v required at normal conditions yields more than 10% additional flow with the valve wide open, and very near 10% additional even with the change in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



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

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

$$2" \frac{0.5}{85} \times 100 =$$

Figure 41 - Low Pressure Pump, Motor Drive

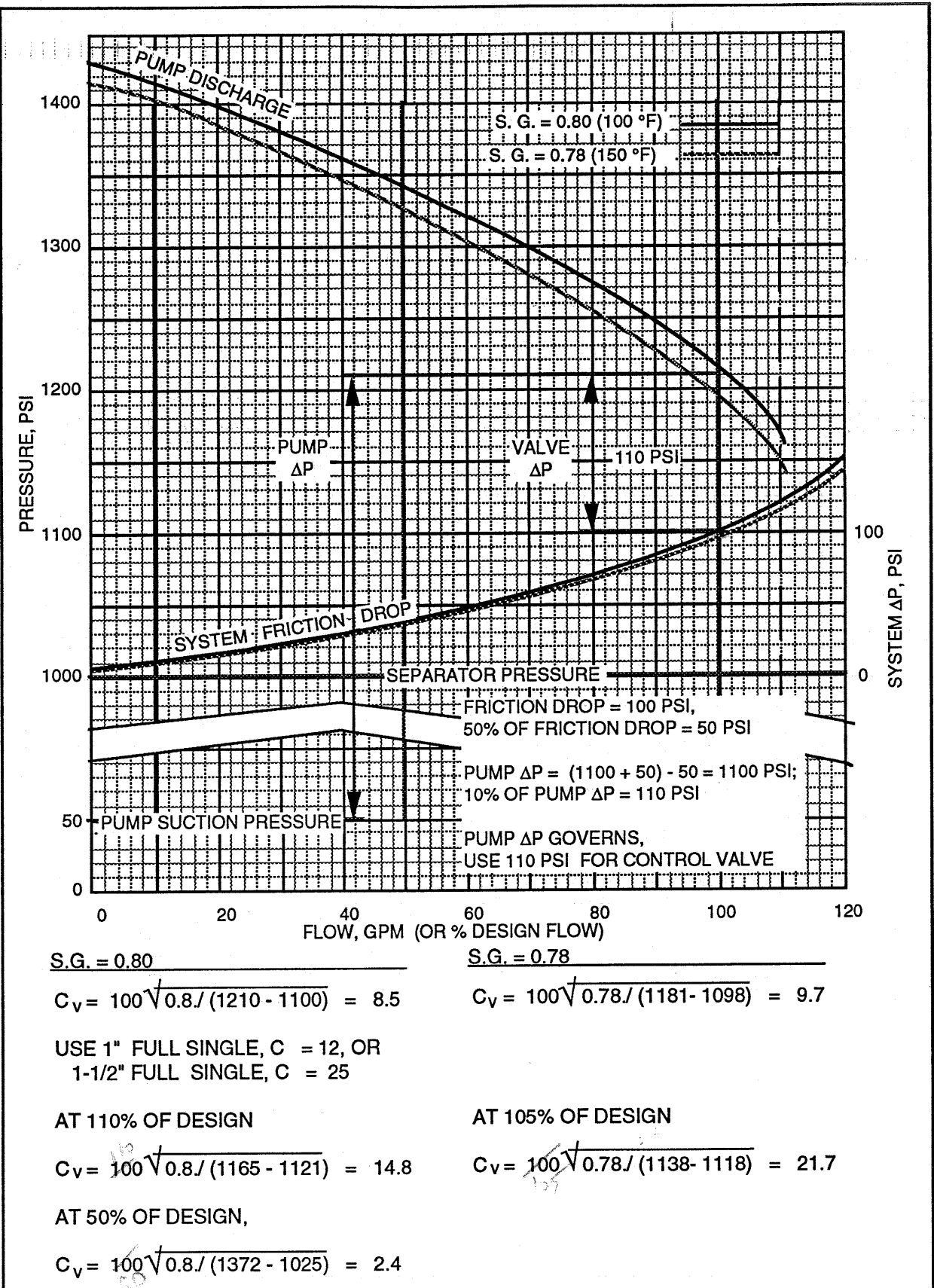


Figure 43 - High Pressure Pump, Motor Drive, Pump Head and Friction Drop Considered

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

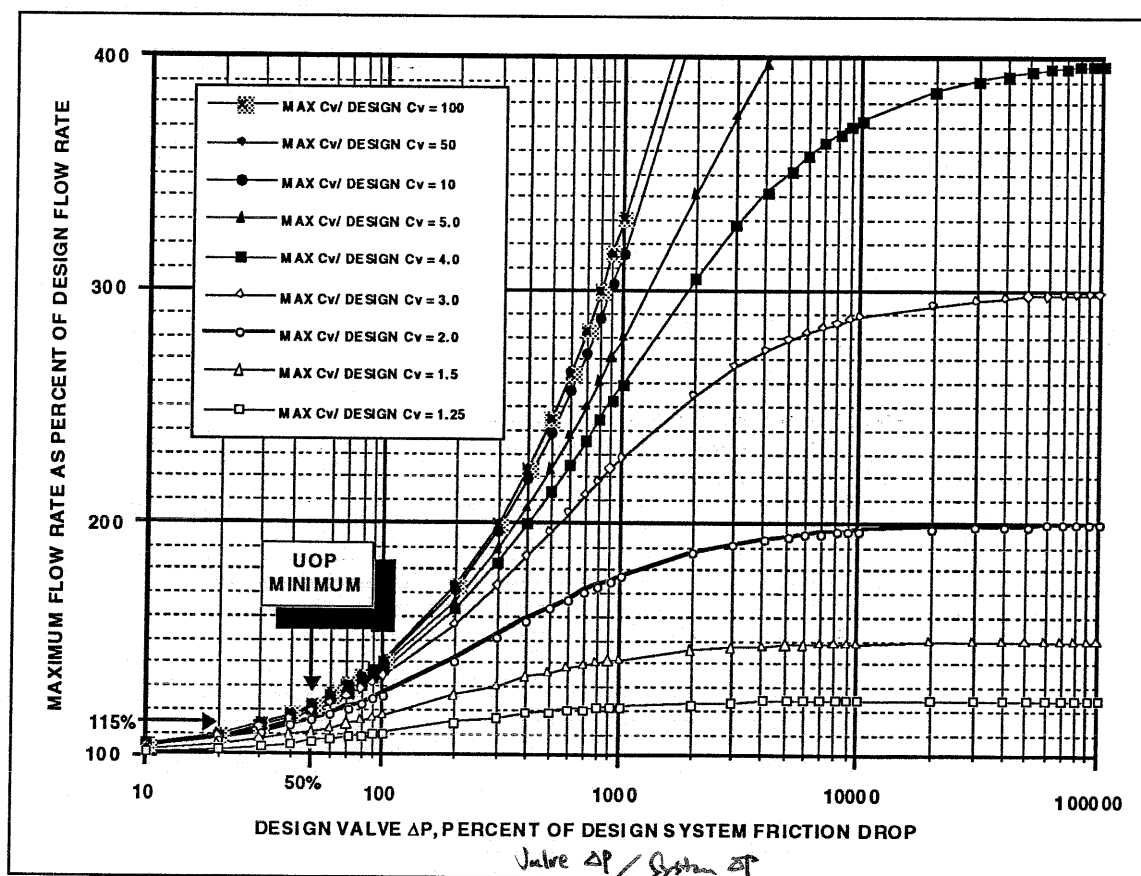


Figure 44 - Control Valve Hydraulics

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 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 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

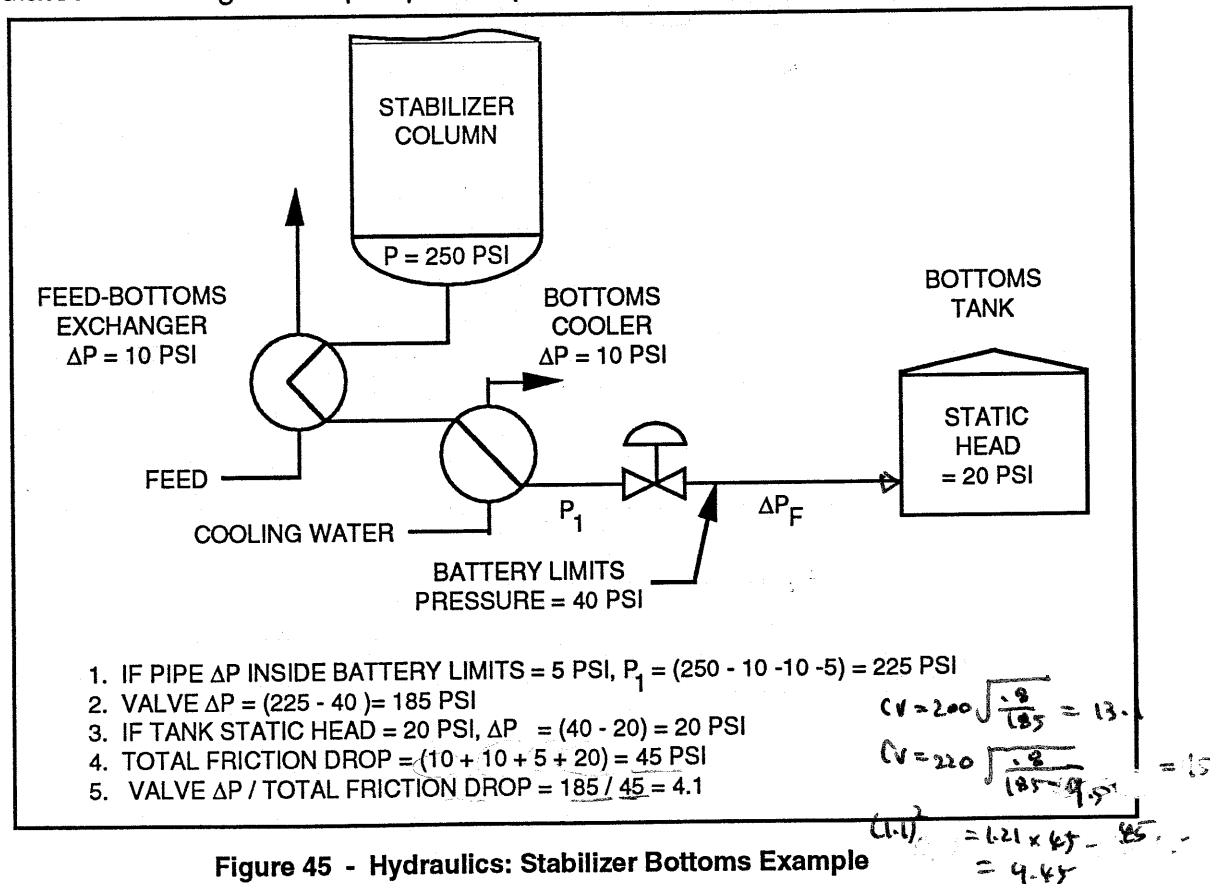


Figure 45 - Hydraulics: Stabilizer Bottoms Example

$(8.5 * 1.25) = 10.6$. 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, 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 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:

$$Q = U A \Delta T$$

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 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

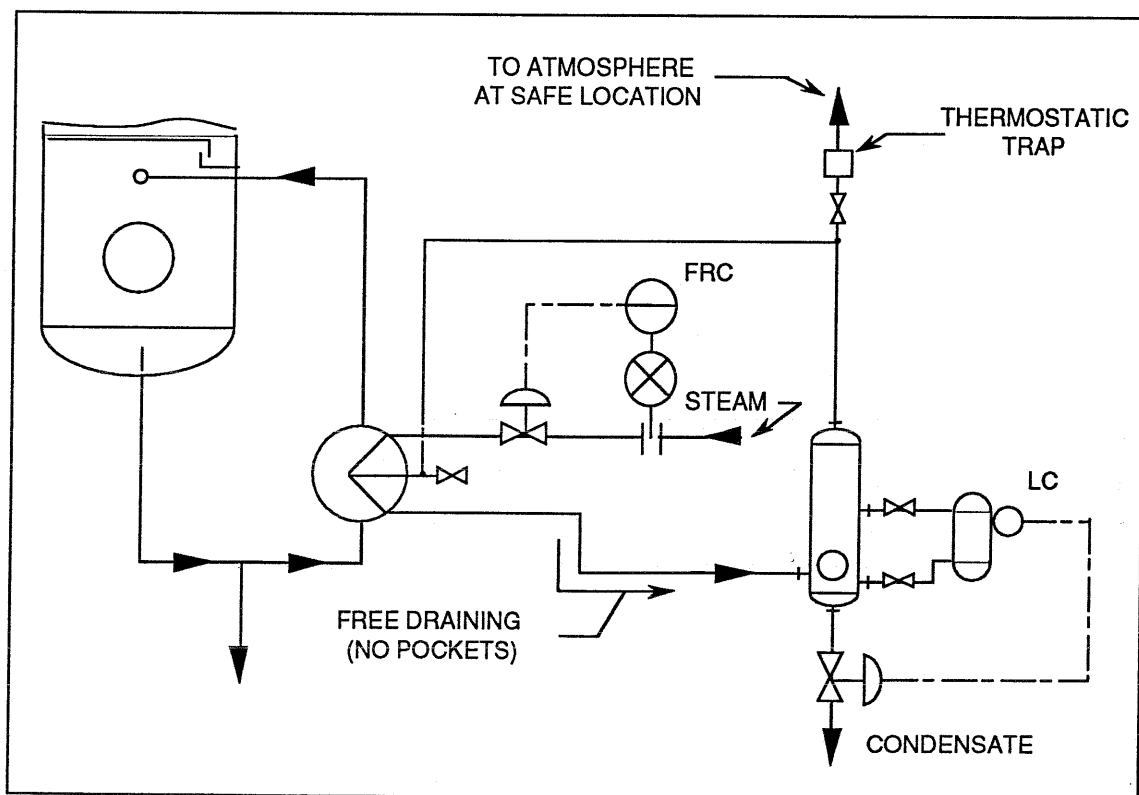


Figure 46 - Steam Reboiler Steam Control

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

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

where

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

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 $C_v^2 (\Delta P_s)$ 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_m , F_1 , 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.