POWERS CONTROL VALVE
Selection and Sizing

Application Eng. Form
AE-1
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- INTRODUCTION -

The control valve is the most important single element in any fluid handling system, and it is for this reason that the control valve selection for air conditioning, or for industrial processes is a choice requiring more than a casual approach. In addition, the demand by modern applications for more precise flow control, as well as for a longer service life with minimum maintenance for the valve and all other system components (such as pumps and compressors), emphasize the need for correct valve sizing.

To select the proper valve, an extensive knowledge of the process and the components of the system usually is necessary. Valves are no longer sized simply on the basis of available pipe sizes, but rather on the actual process requirements. However, rules have been developed from experiments and experience which greatly simplify control valve selection while making it more exact. The purpose of this application form is to provide a better understanding, as well as a more complete presentation, of control valve selection and sizing methods.

- CONTROL VALVE TERMINOLOGY -

There are several terms which are used to describe control valves. The following are the most common.

Flow Characteristic – The relationship between stem position (inches) and fluid flow (gpm), usually expressed in percent of maximum, with a constant pressure drop across the valve. The four basic flow characteristics can be classified as quick opening, linear, modified linear and equal percentage. See figure 1.

Flow Coefficient (C_v) – A precise rating of the size of a valve in terms of its fluid handling capacity rather than its nominal pipe thread connection size. C_v is defined as the flow of water (U.S. gpm) at 60°F through a control valve in the full open position with a 1 psi pressure differential across the valve. Sizing formulas for steam or gas give an equivalent water valve size for the fluid in question. C_v values for valves are listed in water capacity tables under 1 psi pressure differentials.

Rangeability – Ratio of maximum controllable flow to minimum controllable flow at a constant pressure drop. The minimum controllable flow is determined from the flow characteristic curve, and is the smallest flow which does not deviate from the flow curve. Minimum controllable flow should not be confused with shutoff leakage.

Turndown – Ratio of maximum usable flow on the particular application to the minimum controllable flow at a constant pressure drop. Since a control valve is usually oversized, it may never open more than 75% of its maximum stroke. For example, if a valve is capable of controlling flows from 50 gpm to 1 gpm, but the maximum flow used is 30 gpm, the turndown is 30 to 1. In this example, the rangeability is 50 to 1.

Travel Coefficient – Ratio of maximum controllable flow to the flow at any intermediate stem position at a constant pressure drop. If a valve has a maximum controllable flow of 100 gpm, and the flow is 50 gpm at 75% travel, and 20 gpm at 50% travel, the travel coefficient is 2:1 at 75% travel, and 5:1 at 20% travel.

Spring Range – The spring range refers to the change in air pressure which must be applied to the valve top to move the valve through its full stroke (with 0 or nominal ΔP across valve). Powers valves are available with 5 psi or 10 psi spring ranges. The initial spring tension determines the starting pressure which initiates valve motion. With the spring adjusted for a 3 psi starting pressure, full valve travel is completed by 8 psi with a 5 psi spring, and 13 psi with a 10 psi spring, provided the pressure drop is small. On large valves, a 5 psi spring may act as a 7 or 8 psi spring, depending on the pressure drop across the valve. Large single seat valves require large actuator forces to seat the disc for shutoff. These high differential pressures across the valve are caused by the unbalanced fluid pressures which exert large forces on the valve disc. For this reason, valve positioners are recommended for large valves designed for large pressure drops. See appropriate sizing sheets to determine top closing pressure required at various pressure differentials.

Valve Action – Control valves are either normally open (spring opened) or normally closed (spring closed). A normally open valve opens with a decrease in air pressure in the top; a normally closed valve closes with a decrease in air pressure in the top. The type of valve action is determined by the application, and is usually analyzed on the basis of desirable valve position (open or closed) with supply air or power failure. For example, you would select a normally open valve for controlling the flow of steam into a preheat coil. A normally open valve will prevent coil freeze-up with supply air failure.

Critical Pressure Ratio – When sizing valves for controlling steam or other *compressible* fluids such as air or gas, the flow (SCFM or Lb./Hr.) depends upon the initial pressure and pressure differential. This is because of the change in specific volume (cu. ft./lb.) with changing pressures. Because of the thermodynamic properties of compressible fluids, increasing pressure differential will not increase fluid flow beyond a maximum value which occurs when a critical pressure ratio is reached.

For saturated steam flowing through an orifice or valve, with supply pressure P_1 and reduced pressure

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P_2, the critical pressure ratio is $P_2/P_1 = 0.546$, the pressures being expressed in absolute values (gauge pressure + 14.7 psia).
Critical pressure ratios vary slightly for various gases, but are assumed equal to 0.5. Hence, for control valve sizing, the actual flow does not increase if the reduced absolute pressure P_2 is equal to or less than ½ of the absolute inlet pressure P_1.

Wire Drawing – The effect of a fluid acting on a valve seat or poppet when the valve is almost closed. The poppet, positioned close to the seat, must throttle the fluid flowing at high velocities or turbulent conditions because of the large pressure drop across the valve. The effect of wire drawing appears as a small eroded area or thin "wire drawn" slit on the valve poppet or seat. An oversized valve increases the possibility of wire drawing.

Flashing – Flashing or boiling of a fluid takes place when the saturation pressure of the fluid is reduced below the value required to maintain the fluid in its liquid state. For example, (refer to steam tables) water at 274°F can exist only if pressurized to at least 30 psig. Any pressure less than 30 psig will cause "flashing" or boiling, resulting in noise, valve damage, reduced capacity, and other undesirable effects.

– CONTROL VALVE DESIGN –

Valve Components

Actuator – A part of a control valve which causes valve motion in response to an external signal.

Body – That portion of a control valve which regulates the flow of fluid.

Trim – All portions of a valve in contact with the fluid flowing (except valve body castings, caps and bonnets). Seats, discs, stem, throttling plug, etc., are all trim components. The term "trim" usually denotes trim material.

Disc – That portion of the valve trim which makes contact with the valve seat when the valve is closed.

Plug – That portion of the valve trim which characterizes the flow of fluid. The disc is often an integral part of the throttling plug.

Port – The cylindrical or annular area of a valve body between the disc or plug which determines the flow rate of the valve at various stem positions.

Extension Bonnet – Bonnet extensions are used on diaphragm control valves to prevent stem friction or seizing due to freezing of atmospheric moisture at the packing. Freezing is caused by low body temperatures.

Fin Radiating Bonnets – Fin radiating bonnets are used on diaphragm control valves where body operating temperatures exceed 450F. Radiating bonnets reduce packing box temperatures and minimize the danger of excessive drying of the packing and lubricant.

Valve Flow Characteristics

The four basic flow characteristics are;
1. Quick opening
2. Linear
3. Modified linear
4. Equal percentage

ports, that can be characterized in any of the four styles listed above. See figure 1 for a typical characteristic flow curve.

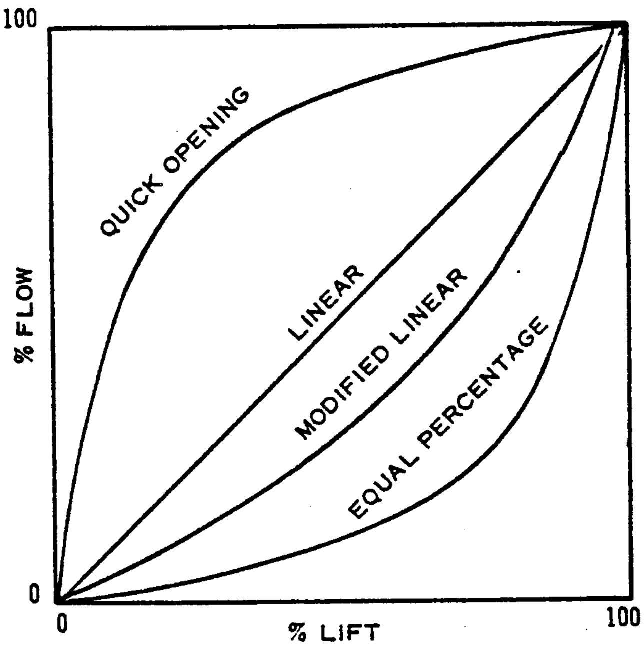


FIGURE 1

Selecting the proper valve body plug depends on actual flow conditions. The most important variables are pressure and temperature drops, process load changes and controller sensitivity.

1. Quick Opening Poppet – The quick opening valve is used primarily in self-contained regulators or in on-off type control systems or systems with constant pressure drop across a control valve. The quick opening valve obtains full capacity with a relatively small travel. The flow characteristic is nearly linear up to 70% of valve lift. Tight shutoff is obtained by a composition disc which can be replaced if damaged. Powers quick opening composition disc valves control the flow of steam, hot or chilled water, oil, air, gases, and other fluids for controlling steam convertors, preheat coils, liquid level, and instantaneous heaters. See figure 1 for a typical flow characteristic curve.

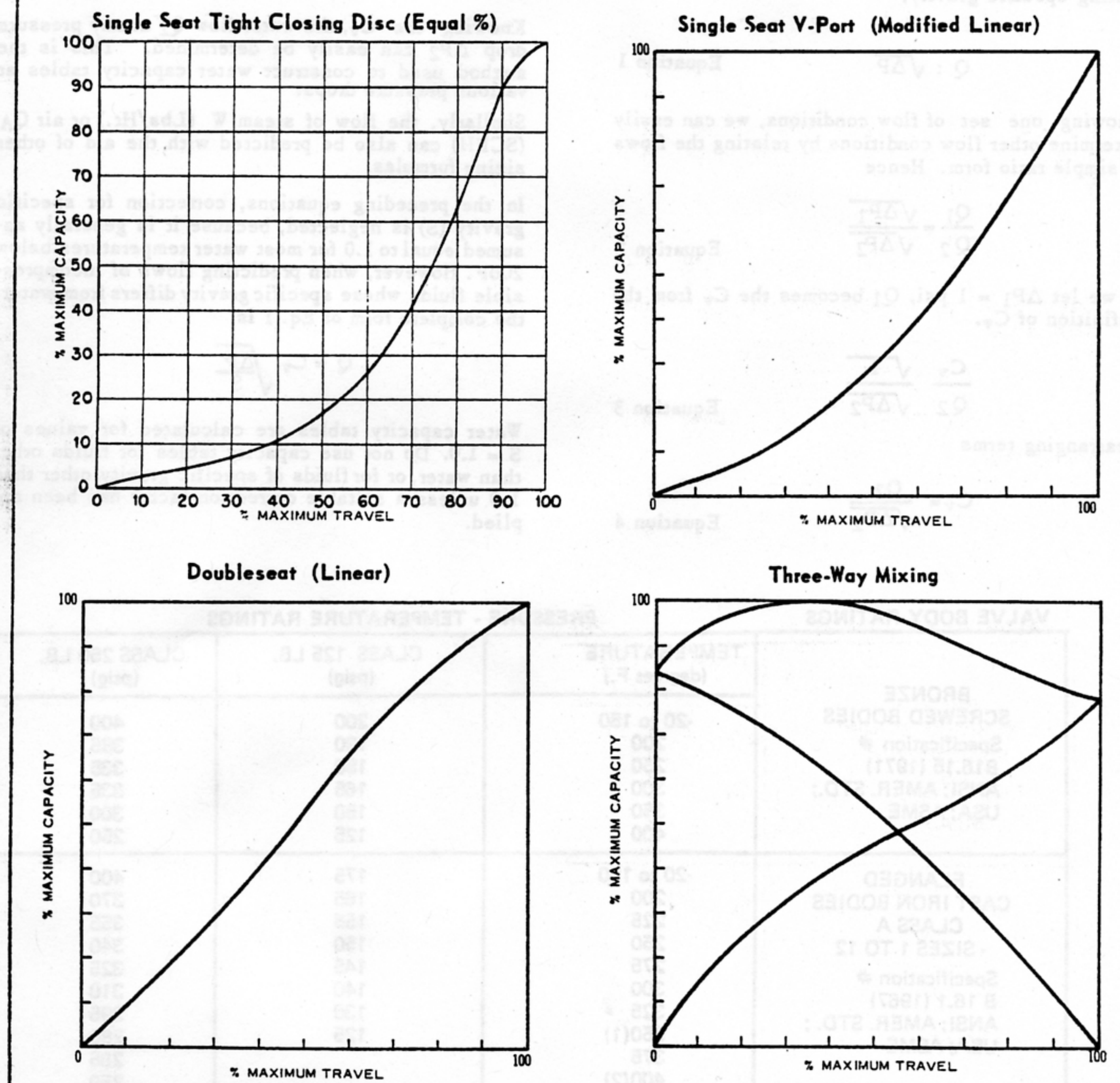
2. Linear Plug – The double seat (balanced) valve is used where high pressure differentials in single seat valves would normally require large valve actuators and positioners because of the unbalanced fluid forces. The double seated valve provides a suitable linear flow characteristic for all plug positions from fully open to fully closed. The double seat plug construction prevents tight shutoff, but leakage is generally less than 1% of maximum flow. The double seat valve often has a higher capacity rating for its nominal size.
Powers double seat valves are used for steam, hot and chilled water, oil, air, gases and other fluids, where high differential pressures are encountered or large capacities are required. Applications include steam convertors, instantaneous heaters, preheat coils and pump bypass control. See figure 2 for a typical flow characteristic curve.

3. Modified Linear Plug – Commonly referred to as

ability is important. At low flows, a large stem movement results in a small change in flow a useful characteristic in the control of heat exchangers. The V-Port design of this type valve permits easier positioning of the plug for a given flow condition. See figure 2 for a typical flow characteristic curve.

Powers V-port valves are used for accurate control of steam, hot and chilled water, oil, air, gas and other fluids where high rangeability and good control at low flow rates are required in steam convertors, instantaneous heaters, heating coils, and other industrial applications such as meat processing, paper, lumber and textile mills.

4. Equal Percentage Plug – This valve plug is used extensively where a high degree of control accuracy is desirable with wide variations in pressure, flow rates, load changes, and other variables such as long time lags. Maximum rangeability is achieved by the exponentially increasing flow rates. This characteristic compensates for the nonlinear heat output characteristic of water coils. For this reason, Powers equal-percentage valve plugs are particularly recommended for control of hot or chilled water coils, although steam, oil, air, gas and other fluids can also be controlled. See figure 1 for a typical flow characteristic curve.



TYPICAL FLOW CHARACTERISTIC CURVES FOR FLOWRITE VALVES

For actual characteristics of each pipe size refer to Technical Literature or contact Application Engineering

Determination of C_v and Valve Flow Characteristics

Determination of C_v — Since the physical configuration of the control valve dictates its resistance to the flow of fluids, the actual flow rating must be determined by specifically testing each valve size and style. An incompressible fluid, water, is universally used in this test under the conditions stated in the section titled "Control Valve Terminology." Valves made by different manufacturers will vary in flow rating even though listed as the same nominal pipe thread size.

The flow of water (Q) through a restriction (valve) can be shown to be directly proportional to the square root of the pressure differential (ΔP) across the valve. The mathematical equivalent of this statement, neglecting specific gravity, is

$$Q : \sqrt{\Delta P} \qquad \text{Equation 1}$$

Knowing one set of flow conditions, we can easily determine other flow conditions by relating the flows in simple ratio form. Hence

$$\frac{Q_1}{Q_2} = \frac{\sqrt{\Delta P_1}}{\sqrt{\Delta P_2}} \qquad \text{Equation 2}$$

If we let $\Delta P_1 = 1$ psi, Q_1 becomes the C_v from the definition of C_v.

$$\frac{C_v}{Q_2} \quad \frac{\sqrt{1}}{\sqrt{\Delta P_2}} \qquad \text{Equation 3}$$

Rearranging terms

$$C_v = \frac{Q_2}{\sqrt{\Delta P_2}} \qquad \text{Equation 4}$$

Since it is difficult to maintain a 1 psi differential across a control valve with any accuracy, Eq. 4 provides a simple method to calculate the C_v of a control valve.

By measuring the flow of water per unit time Q_2 (gpm) with a constant pressure differential ΔP_2 (usually 9, 16 and 25 psi) the C_v can easily be calculated by Eq. 4.

Having determined the C_v of a control valve, water tables can easily be tabulated by rearranging Eq. 4. Hence,

$$Q_2 = C_v \sqrt{\Delta P_2} \qquad \text{Equation 5}$$

Knowing the C_v, the water flow Q_2 at any pressure drop ΔP_2 can easily be determined. This is the method used to construct water capacity tables at various pressure drops.

Similarly, the flow of steam W (Lbs/Hr.) or air Q_A (SCFH) can also be predicted with the aid of other sizing formulas.

In the preceding equations, correction for specific gravity (S) is neglected, because it is generally assumed equal to 1.0 for most water temperatures below 200F. However, when predicting flows of incompressible fluids whose specific gravity differs from water, the complete form of Eq. 1 is

$$Q = C_v \sqrt{\frac{\Delta P}{S}}$$

Water capacity tables are calculated for values of S = 1.0. Do not use capacity tables for fluids other than water, or for fluids of specific gravity other than 1.0 unless a suitable correction factor has been applied.

VALVE BODY RATINGS	PRESSURE - TEMPERATURE RATINGS		
	TEMPERATURE (degrees F.)	CLASS 125 LB. (psig)	CLASS 250 LB. (psig)
BRONZE SCREWED BODIES Specification # B16.15 (1971) ANSI; AMER. STD.; USA; ASME	-20 to 150	200	400
	200	190	385
	250	180	365
	300	165	335
	350	150	300
	400	125	250
FLANGED CAST IRON BODIES CLASS A - SIZES 1 TO 12 Specification # B 16.1 (1967) ANSI; AMER. STD.; USA; ASME	-20 to 150	175	400
	200	165	370
	225	155	355
	250	150	340
	275	145	325
	300	140	310
	325	130	295
	350(1)	125	280
	375		265
	400(2)		250

Table 1.

(1) - 353F (Max) to reflect the temperature of saturated steam of 125 psig.
(2) - 406F (Max) to reflect the temperature of saturated steam of 250 psig.

Determination of Flow Characteristics – With a constant pressure drop across the valve, the valve stem is positioned in small increments from closed to full open. The flow at each position is carefully measured and recorded.
C_V values are determined in this same test with valve in the full open position.

Valve Body Pressure – Temperature Ratings

The number stamped on the valve body shows the valve body rating for saturated steam conditions. The packless bellows, packing lubricant, disc material, or other trim components could also restrict valve ratings to a lesser value. For long valve and packing life, the body temperature and pressure should not exceed the valve body rating. Even though design safety factors will allow for greater temperatures and pressures, it is generally advisable to select a valve material which is within the pressure/temperature rating.

Although body temperatures should not exceed the valve rating, higher valve pressures can be tolerated for liquid or gas media, provided the fluid temperature is below the maximum rating of the valve.

Table 1 summarizes standard ratings for bronze and cast iron valve bodies.

CONTROL VALVE SIZING – Valve Capacity

The sizing of a valve is very important if it is to render good service. If it is undersized it will not have sufficient capacity. If it is oversized, the controlled variable may cycle, and the seat and disc will be subjected to wire drawing because of the restricted opening.
Systems are designed for the most adverse conditions expected (i.e. - coldest weather, greatest load, etc.). In addition, system components (boiler, chiller, pumps, coils, etc.) are limited to sizes available and frequently have a greater capacity than system requirements. Correct sizing of the control valve for *actual expected conditions* is considered *essential* for good control.

The most important variables which must be considered are:

1. What medium will the valve control? Water? Air? Steam? Oil? What effects will specific gravity and viscosity have on the valve size?
2. What will the inlet pressure be under maximum load demand? What is the inlet temperature?
3. What pressure drop (differential) will exist across the valve under maximum load demand?
4. What maximum capacity should the valve handle?
5. What is the maximum pressure differential the valve top must close against?

When these are known, a valve can be selected by formula (C_v method) or water and steam tables. In any case, the valve size should not exceed the line size and it should preferably be one or two sizes smaller.

The above sizing procedures are also applicable to the No. 11 regulator. Each style and size has its own C_V value as well as flow characteristic curve. Drawings containing this information are available from Application Engineering.

Pressure Drop for Water Flow

A pressure drop must exist across a control valve if flow is to occur. The greater the drop, the greater the flow at any fixed opening. But the pressure drop across a valve also varies with the disc position – from minimum when fully open, to 100% of the system drop when fully closed.

To size a valve properly, it is necessary to know the full flow pressure drop across it. The pressure drop across a valve is the difference in pressure between the inlet and outlet under flow conditions. When it is specified by the engineer and the required flow is known, the selection of a valve is simplified. But when this pressure drop is not known, it must be computed or assumed.

A basic rule of control valve sizing is; "The higher the percentage of drop that can be made to occur across the wide open valve in relation to the percentage of pressure drop through the line and process, the better will be the control obtained."

However, the valve should also meet system static pressure, differential pressure, and temperature requirements for best service and long life. The valve should be sized with a reasonably high pressure drop when fully open so that the fluid flow is determined by the *valve* through its entire travel. This is necessary to preserve its designed flow characteristic. If the pressure drop across the valve when fully open is not a large enough percentage of the total system drop, the gradual closing of the valve will cause a correspondingly *large increase* in fluid velocity. Since the percentage drop varies as the square of fluid velocity, the pressure drop will increase as the valve closes and thus maintain a relatively constant flow. Therefore, in this case, there is little change in fluid flow until the valve actually closes, forcing the valve's characteristic toward a quick opening form. Figure 3 shows flow-lift curves for a linear valve with various percentages of design pressure drop. Note the improved characteristic as pressure drop approaches 100% of system pressure drop at full flow.

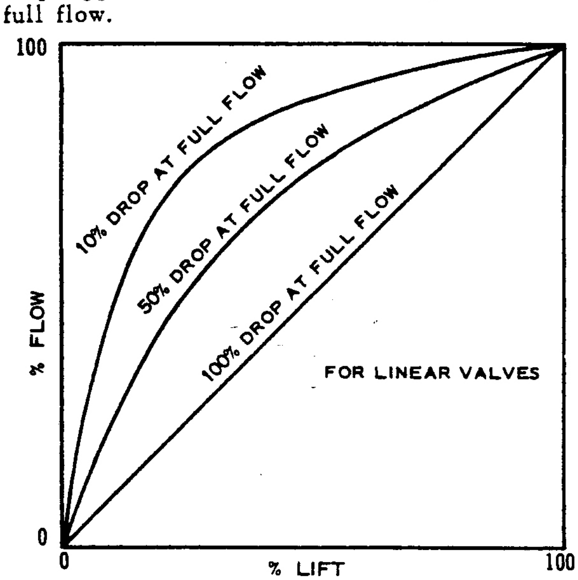


FIGURE 3

It is important to realize that the flow characteristic for any particular valve, such as the linear characteristic shown in figure 3 is applicable only if the pressure drop remains nearly constant across the valve for full stem travel. In most systems, however, it is clearly impractical to take 100% of the system drop across the valve.

For this reason, a good working rule is that, at maximum flow, 25 – 50% of the total system pressure drop should be absorbed by the control valve. Although this generally results in larger pump sizes.

it should be pointed out that the initial equipment cost is offset by a reduction in control valve size, and results in improved controllability of the system. Reasonably good control can be accomplished with pressure drops of 15 – 30% of total system pressures. A drop of 15% can be utilized if the variation in flow is small.

Importance of Adequate Valve Pressure Drop Allowance – It is important to allow sufficient control valve pressure drop while the system is in its design stage since changes are almost impossible to make after piping and pumps have been installed. A valve should have a large pressure drop (resistance) when in the open position. The reason is that the *change in* valve resistance must be kept as small as practicable. If the pressure differential allocated to the valve is 10% or less of the total system drop, the valve selected will be so large that it cannot throttle effectively until near its fully closed position. The result is poor control and excessive trim wear.

Pressure Drop for Steam Flow

The same reasoning in selecting a valve for control of water flow applies to selecting a valve for control of steam flow. The most important consideration is in the selection of a pressure drop.

First, the correct maximum capacity of the coil must be determined. Ideally, there should be no safety factor in this determination and it should be based on the actual BTU heating requirements rather than on the condensing capacity. The valve size must be based on the actual supply pressure at the valve. When the valve is fully open, the outlet pressure will assume a value such that the valve capacity and coil condensing rate are in balance. If this outlet valve pressure is relatively large (small pressure drop), then as the valve closes, there will be no appreciable reduction in flow until the valve is nearly closed. To achieve better controllability, the smallest valve (largest pressure drop) should be selected. With the valve outlet pressure much less than the inlet pressure, a large pressure drop results. There will now be an immediate reduction in capacity as the valve throttles. For steam valves, generally the largest possible pressure drop should be taken, without exceeding the critical pressure ratio. Therefore, with a few exceptions, the steam pressure drop should approach 50% of the *absolute* inlet pressure.

Examining the pressure drops, under "Recommended Pressure Drops for Control Valve Sizing" one might be concerned about the steam entering the coil at 0 psig when a large drop is taken across the control valve. Steam flow through the coil will still take place because the pressure in the coil will drop to vacuum pressures due to condensation of the steam. Consequently, a pressure differential will still exist. In this case, proper steam trapping and condensate piping is essential.

Selection of Valve Pressure Drop for HTHW Systems

High temperature water systems are generally considered as those operating with supply water temperatures and pressures from 325F (80 psig) to 425F (310 psig). Since water can exist in its liquid state be pressurized over the saturation pressure to prevent flashing. This is accomplished with the use of some inert gas (nitrogen or air) to pressurize the expansion tank from 15–45 psi over saturation conditions.

The valve must be capable of withstanding the high temperatures and pressures usually encountered for these systems. Generally, single seated valves are used to reduce the leakage normally encountered when using double seated valve.

Since HTHW systems are designed for temperature drops of 100F and higher, the location of the valve should be carefully considered. If the valve is located after the coil, the valve can be designed for lower temperatures and static pressures. In any case, the pressure drop selected must never reduce the water pressure below its saturation pressure, or flashing will occur.

For example, let us size a valve for a HTHW system with consideration given to the location of the valve with respect to the coil. Assume the water is at 325F (80 psig) as it leaves the boiler, and that it is pressurized 30 psi over saturation conditions. The coil is designed for a 100F temperature drop.

Valve on Entering Side of Coil – Water entering the valve is at 325F and 110 psig. As previously discussed, a 30% pressure drop is required for good controllability. With a 33 psi drop (.3 x 110), the water leaving the valve would be at 325F (neglecting piping heat loss) and 77 psig, or below the saturation pressure of the water. Flashing will occur in this situation.

Valve on Leaving side of Coil – Water entering the valve is at 225F and 100 psig (10 psi and 100F drop through heating coil). With a 30 psi drop (.3 x 100), the water leaving the valve would be at 225F and 70 psig well above the saturation pressure of 4 psig for water at 225F.

Recommended Pressure Drops for Valve Sizing

Steam

1. With gravity flow condensate removal and inlet pressure less than 15 psig, use a pressure drop equal to the inlet gauge pressure.
2. With vacuum return system up to 7" Hg vacuum and an inlet pressure less than 2 psig, a pressure drop of 2 psi should be used. With an inlet pressure of 2–15 psig, use a pressure drop equal to the inlet gauge pressure.
3. With an inlet pressure greater than 15 psig, use a pressure drop equal to 50% of inlet absolute pressure. Example: Inlet pressure is 20 psig (35 psia). Use pressure drop of 17.5 psi.
4. An exception to item 3 is the case where a coil size is selected on the basis that line pressure and temperature is available in the coil of a heating and ventilating application. Under these conditions, a very minimum pressure drop is desired. In this case, use the following pressure drop:

Initial Pressure	Pressure Drop
15 psi	5 psi
50 psi	7½ psi
100 psi	10 psi

Water

1. With potential differential pressure less than 20 psig, use a pressure drop equal to 5 psi.
2. With potential differential pressure greater than 20 psig, use a pressure drop equal to 25% of total system pressure drop (maximum pump head), but not exceeding the maximum rating of the valve.

Flow Characteristic Selection

The heat output of a water to air coil is not linearly related to flow. As the water flow is reduced, there is typically a greater temperature drop of the water and a high output of heat transfer is maintained. Figure 4 shows the typical heat capacity of a coil vs. flow.

If the coil is designed for a large water temperature drop, the output flow characteristic becomes more linear. See figure 7. For example, when water flow to a 20F drop coil is reduced from 100% to 50%, there is only a 15% reduction in heat output; while with a 50F drop coil, the reduction is 35% in output.

See Application Engineering Form AE-13 for a comprehensive analysis of Hot Water Systems.

See Application Engineering Form AE-16 for a description of Medium (MTHW) and High (MTHW) Temperature Hot Water Systems.

We have seen how pressure drop affects the valve flow characteristic in a closed system, how the pump head varies with flow, and how the coil characteristic is non-linear. With all these factors considered, it is obvious that a valve must have a good characteristic in addition to being properly sized with an adequate pressure drop.

If a coil system with linear output (BTU vs. valve lift) is required, an equal percentage valve should be used. This is particularly important if the heating or cooling load varies during operation.

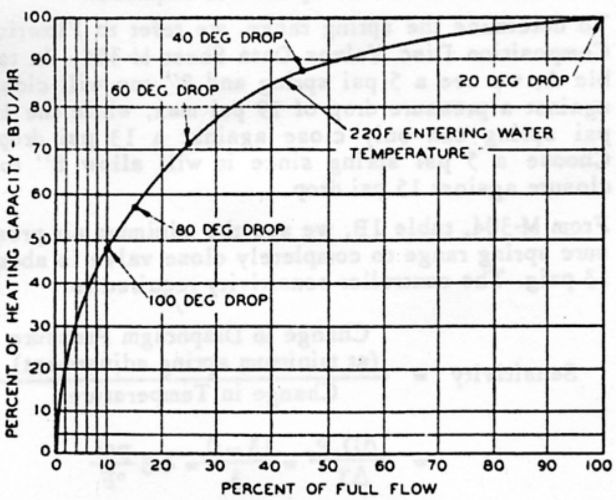


FIGURE 4

The equal percentage valve offers the best control since its characteristic, combined with the coil characteristic, gives a coil output which is linear with respect to valve lift. Note the poor control obtained with a quick opening valve. For practical purposes, linear or modified linear valves will provide excellent control if the valves are sized with sufficient pressure drop (30 – 50% of total system pressure drop).

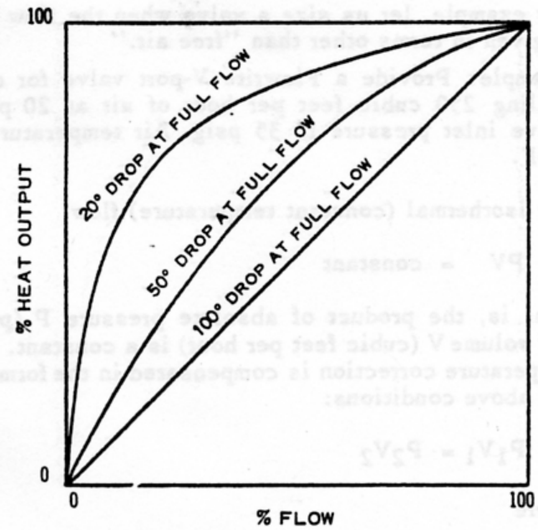


FIGURE 5

Effect of Viscosity on Control Valve Sizing

A fundamental property of any fluid is its viscosity. Viscosity is the measure of the resistance of the fluid itself to the relative motion of its flow layers. As the fluid viscosity increases for a given flow, the C_V or valve size must be increased.

Several units are used to measure viscosity. Saybolt Universal Seconds (SSU) and Centipoise are two common terms, but either can be expressed in terms of the other. We will be referring to SSU terms for our control valves.

Fluid Viscosity varies with temperature, and it is important to calculate valve sizes at the actual flow temperature. For example, the SSU factor for turbine oil at 100F is about 420 SSU, while at 210F the value is 60 SSU. The correction factor is shown in the table. At 100F, the correction is $1.45C_V$, while at 210F the correction is only $1.11C_V$.

Viscous fluids usually require corrections for specific gravity. The complete C_V equation for viscous fluids is given by:

$$C_V = K_r Q \sqrt{\frac{S}{\Delta P}}$$

Where

K_r = Viscosity correction factor from table at actual flow temperature

Q = Flow, GPM

S = Specific gravity at actual flow temperature

ΔP = Valve design pressure drop, psi

See table 2 for Viscosity Factors

Determination of Flow Rate of Gases

To determine the correct valve size for controlling the flow of compressible fluids such as air, the volumetric flow must be known. The sizing formula for gas flow can only be used when the amount of standard cubic feet per hour is known.

The term "standard cubic feet per hour" (SCFH) refers to air at 14.7 psia and 68F. This is sometimes referred to as "free air". Often the gas flow is not given in SCFH units and conversion must be made.

For example, let us size a valve when the flow rate is given in terms other than "free air."

Example: Provide a Flowrite V-port valve for controlling 250 cubic feet per hour of air at 20 psig. Valve inlet pressure is 35 psig. Air temperature is 100F.

For isothermal (constant temperature) flow,

$$PV = \text{constant}$$

That is, the product of absolute pressure P (psia) and volume V (cubic feet per hour) is a constant. The temperature correction is compensated in the formula. For above conditions:

$$P_1V_1 = P_2V_2$$

Where

$P_1 = 20 + 14.7 = 34.7$ psia

$V_1 =$ 250 cubic feet per hour

$P_2 =$ atmospheric pressure $= 14.7$ psia

$V_2 =$ standard cubic feet per hour (SCFH)

$$Q_{a2} = V_2 = \frac{P_1}{P_2}V_1 = \frac{(34.7)}{(14.7)}(250) = 590 \text{ SCFH}$$

The correct valve size is determined by

$$C_v = \frac{Q_a\sqrt{(G)\ (T+460)}}{1360\sqrt{(\Delta P)\ P_2}} \quad \frac{590\sqrt{1\ (560)}}{1360\sqrt{15(34.7)}} = 0.45$$

From Flow Tables, choose a 1/2B" Flowrite V-port valve with a $C_v = 0.5$.

To determine the volumetric flow of free air (or any other compressible gas) multiply the volumetric flow at the reduced pressure to atmospheric pressure. For absolute pressures:

$$\text{SCFH} = \frac{P_2}{P_1}Q_a = \frac{P_2}{14.7}\ Q_a$$

For gauge pressures:

$$\text{SCFH} = \frac{P_2 + 14.7}{14.7}\ Q_a$$

Caution: In the above example, *if* the flow was given as "250 *standard* cubic feet per hour of air at 20 psig"; the flow rate has already been converted into "free air" and conversion is not necessary. The valve size for this condition would be selected as follows:

$$C_v = \frac{250\sqrt{1\ (560)}}{1360\sqrt{15\ (34.7)}} = 0.19$$

From Flow Tables, choose a 1/2A" Flowrite V-port

SELECTION OF SPRING RANGE – FOR CONTROL VALVES –

The most important factor affecting selection of a spring range is whether the valve will be capable of closing against the design pressure drop. When either the 5 or 10 psi spring will handle the pressure drop involved (and the top pressure is not excessive, i.e. 16–25 psig when the controller supply is only 15 psig), other factors may decide what spring should be used.

Whether a 5 or 10 psi spring should be used for a given installation depends primarily on the sensitivity of the controller. Thus, the throttling range will be 5F regardless of whether a 2 psi/F sensitivity is used with a 10 psi valve spring, or a 1 psi/F sensitivity is used with a 5 psi valve spring. Because load fluctuations for various applications are often unknown, it is difficult to predict what controller sensitivity will be used. Knowing the desired throttling range of the process, the minimum sensitivity of the controller can be calculated for various spring characteristics;

$$\text{Sensitivity} = \frac{SR}{TR}$$

Where

SR = Spring range
TR = Process throttling range, F

Example: A 4" Flowrite composition disc normally open valve is required to control a given flow with a pressure drop of 15 psi. Controller air supply is 15 psig and sensitivity is adjustable from ½ to 3 psi/F. The desirable throttling range is 3F. What spring should be used? What top size is required?

To determine the spring range, we refer to Flowrite Composition Disc Valves Data Sheet M-304. In table A, we see a 5 psi spring and 8" top will close against a pressure drop of 19 psi max, while the 10 psi spring can only close against a 13 psi drop. Choose a 5 psi spring since it will allow 8" top closure against 15 psi drop.

From M-304, table 1B, we see the minimum air pressure spring range to completely close valve is about 13 psig. The controller sensitivity required is:

$$\text{Sensitivity} = \frac{\text{Change in Diaphragm Pressure (at minimum spring adjustment)}}{\text{Change in Temperature}}$$

$$= \frac{\Delta D.P.}{\Delta T} = \frac{13-0}{3} = 4.3\frac{\text{psi}}{°F}$$

Since the controller cannot be adjusted for this sensitivity, we will select a 12" top and 5 psi spring. (A 10 psi spring is not available).

From table 1A, the 12" top can close against pressure drops up to 58 psi. From table 1B, the minimum air pressure spring range to completely close valve is 8 psig. The controller sensitivity is:

$$\text{Sensitivity} = \frac{\Delta D.P.}{\Delta T} = \frac{8-0}{3} = 2.67\frac{\text{psi}}{°F}$$

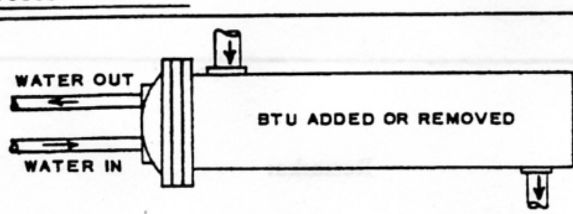
In addition to consideration of throttling range, controller sensitivity and shutoff requirements, other factors may decide what spring to choose if either (5 or 10 psi) spring satisfies the above requirements:

1. Use of the 10 psi spring is recommended where less sensitivity reduces cycling of controlled variable and increases useful valve life.
2. Use of the 5 psi spring is recommended where:
   a. Sequence operation (split ranging of steam valves or alternate heating-cooling of a coil) is desirable.
   b. Where large load changes cause excessive offset in control, where the controller sensitivity is easily adjusted.
   c. When the controller is equipped with automatic reset.
3. Use of the 5 psi spring also:
   a. Allows greater flexibility in adjustment and starting pressures.
   b. Allows valve to close off against greater pressure differentials.

Refer to AB 187 for a more complete description of valve spring selection.

– SIZING FORMULAS & TABLES –

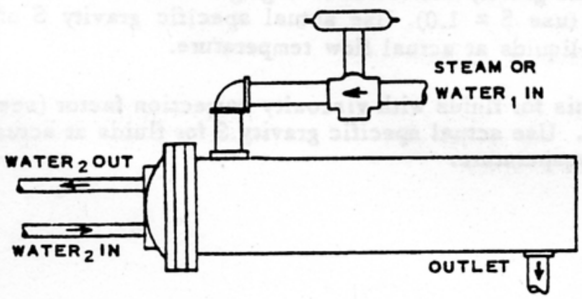
Process Formulas



For heating or cooling water

$$GPM = \frac{BTU/HR}{(^{\circ}F \text{ water temp. rise or drop}) \times 500}$$

$$GPM = \frac{CFM \times .009 \times (\text{change in enthalpy of air - in BTU/\#Air.})}{^{\circ}F \text{ water temperature change}}$$

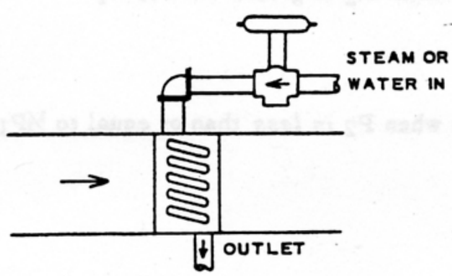


For heating water with steam

Lbs. steam/hr. = 0.50 x GPM x ($^{\circ}$F water temp. rise)

For heating or cooling water with water

$$GPM_1 = GPM_2 \times \frac{(^{\circ}F \text{ water}_2 \text{ temp. rise or drop})}{(^{\circ}F \text{ water}_1 \text{ temp. rise or drop})}$$



For heating air with steam coils

$$\text{Lbs. steam/hr.} = 1.08 \times (^{\circ}F \text{ air temp. rise}) \times \frac{CFM}{1000}$$

For heating air with water coils

$$GPM = 2.16 \times \frac{CFM \times (^{\circ}F \text{ air temp. rise})}{1000 \times (^{\circ}F \text{ water temp. drop})}$$

For radiation

Lbs. steam/hr. = 0.24 x sq. ft. EDR(Low pressure steam)

$$GPM = \frac{\text{Sq. Ft. EDR}}{50}$$ (Assume 20°F water TD)

Conversion Factors

1 lb/sq. in. . . . . 2.04 inches Mercury
1 lb/sq. in. . . . . 2.3 feet water
1 lb/sq. in. . . . . 27.7 inches water
1 Kg/sq. cm. . . . . 14.2 lb/sq. in.
1 U.S. Gallon Water . . . . 231 cubic inches
1 U.S. Gallon Water . . . . 8.33 lbs.
1 cubic foot . . . . 1728 cubic inches
1 cubic foot water . . . . 62.4 lbs. water
1 cubit foot water . . . . 7.5 U.S. Gallons
1 cubic meter . . . . 264 U.S. Gallons
1 U.S. Gallon Water . . . . 0.83 Imperial Gallon
1 Liter . . . . 0.264 Gallons
1 lb. Water . . . . 454 Grams
1 lb. Water . . . . 7000 grains
1 lb. Steam/hr. . . . . 1000 BTU/hr.
1 Ton (Refrigeration) . . . . 12,000 BTU/hr.
1 EDR (Steam) . . . 240 BTU/hr. (Coil temp. = 215F)
1 EDR (Water) . . . 200 BTU/hr. (Coil temp. = 197F)
1 MBH . . . . 1000 BTU/hr.
1 Watt . . . . 3.41 BTU/hr.

Valve Sizing Formulas

The following definitions apply in the following formulas:

C_v Valve flow coefficient, U.S. GPM with $\Delta P = 1$ psi.

P_1 Inlet pressure at maximum flow, psia (abs.).

P_2 Outlet pressure at maximum flow, psia (abs.)

ΔP $P_1 - P_2$ at maximum flow, psi.

Q Fluid flow, U.S. GPM.

Q_a Air or gas flow, standard cu. ft. per hr. (SCFH) at 14.7 psia and 60F.

W Steam flow, lbs. per hr. (Lb/Hr.).

S Specific gravity of fluid relative to water @60F.

G Specific gravity of gas relative to air at 14.7 psia and 60F.

T Flowing air or gas temperature (°F).

K 1 + (0.0007 x °F superheat), for steam.

V_2 Specific volume, cu. ft. per lb., at outlet pressure P_2 and absolute temperature (T + 460).

K_r Viscosity correction factor (see table) for fluids.

1. For Liquids (Water, oil, etc.):

Remarks:

$$C_v = Q\sqrt{\frac{S}{\Delta P}}$$

Specific gravity correction is negligible for water below 200F (use S = 1.0). Use actual specific gravity S of other liquids at actual flow temperature.

$$C_v = K_r Q\sqrt{\frac{S}{\Delta P}}$$

Use this for fluids with viscosity correction factor (see table). Use actual specific gravity S for fluids at actual flow temperature.

2. For Gases (Air, natural gas, propane, etc.)

$$C_v = \frac{Q_A\sqrt{G\,(T+460)}}{1360\sqrt{\Delta P\,(P_2)}}$$

Use this when P_2 is *greater* than ½ P_1.

$$C_v = \frac{Q_A\sqrt{G\,(T+460)}}{660\,P_1}$$

Use this when P_2 is *less* than or equal to ½P_1.

3. For Steam (Saturated or superheated)

$$C_v = \frac{WK}{2.1\sqrt{\Delta P\,(P_1+P_2)}}$$

Use this when P_2 is *greater* than ½P_1.

$$C_v = \frac{WK}{1.82\,P_1}$$

Use this when P_2 is *less* than or equal to ½P_1.

4. For Vapors other than Steam

$$C_v = \frac{W}{63.4}\sqrt{\frac{V_2}{\Delta P}}$$

When P_2 is less than or equal to ½P_1, use the value of ½P_1 in place of ΔP and use P_2 corresponding to ½P_1 when determining specific volume V_2.

Viscosity Factors

SAYBOLT* UNIV. SECONDS S.S.U.	ENGLER TIME – SECONDS	KINEMATIC VISCOSITY –	C_V CORRECTION FACTORS (K_r)
46,350	–	10,000	–
37,080	–	8,000	–
27,810	–	6,000	–
18,540	–	4,000	–
13,900	–	3,000	–
11,590	–	2,500	–
9,270	–	2,000	1.93
6,950	10,800	1,500	1.90
4,635	7,100	1,000	1.82
3,708	5,700	800	1.78
2,781	4,250	600	1.74
1,854	2,820	400	1.67
1,390	2,120	300	1.63
1,159	1,760	250	1.61
927	1,400	200	1.57
695	1,050	150	1.53
464	700	100	1.45
371	555	80	1.42
278	420	60	1.37
186	290	40	1.30
141	225	30	1.25
119	191	25	1.22
97.8	157	20	1.20
77.4	127	15	1.16
58.9	97	10	1.11
52.1	85.5	8	1.08
45.6	76.0	6	1.07
39.1	67.5	4	1.05
36.0	62.5	3	1.03
32.6	58.0	2	–
31.6	55.5	1.5	–
31.3 ← PURE WATER AT 60F →		1.1	–

The relation between kinematic and absolute viscosity:

$$\text{Centistoke} = \frac{\text{Centipoise}}{\text{Specific Gravity}}$$

*Redwood time (seconds) approximately same as S.S.U.

Specific Gravity of Water

Temp. t (°F)	Abs. Pressure	Specific Gravity – S (W = 62.4 lb/ft @ 60F)	$\sqrt{S}$
60	–	1.000	1.000
100	–	0.993	0.999
150	–	0.981	0.985
200	–	0.963	0.981
250	30	0.942	0.971

Temp. t (°F)	Abs. Pressure	Specific Gravity – S (W=62.2 lb/ft @ 60F)	
300	67	0.920	0.959
350	135	0.891	0.944
400	247	0.860	0.927
450	423	0.827	0.910

TABLE 3

STEAM PRESSURE – TEMPERATURE CHART

VACUUM inches Hg	ABSOLUTE PRESSURE psi	TEMPERATURE degrees Fahrenheit
29.74	0.0886	32
29.67	0.1217	40
29.56	0.1780	50
29.40	0.2562	60
29.18	0.3626	70
28.89	0.505	80
28.50	0.696	90
28.00	0.946	100.00
27.88	1	101.83
25.85	2	126.15
23.81	3	141.52
21.78	4	153.01
19.74	5	162.28
17.70	6	170.06
15.67	7	176.85
13.63	8	182.86
11.60	9	188.27
9.56	10	193.22
7.52	11	197.75
5.49	12	201.96
3.45	13	205.87
1.42	14	209.55

GAUGE PRESSURE psi	ABSOLUTE PRESSURE psi	TEMPERATURE degrees Fahrenheit
0.0	14.70	212.0
0.3	15	213.0
1.3	16	216.3
2.3	17	219.4
3.3	18	222.4
4.3	19	225.2
5.3	20	228.0
6.3	21	230.6
7.3	22	233.1
8.3	23	235.5
9.3	24	237.8
10.3	25	240.1
11.3	26	242.2
12.3	27	244.4
13.3	28	246.4
14.3	29	248.4
15.3	30	250.3
16.3	31	252.2
17.3	32	254.1
18.3	33	255.8
19.3	34	257.6
20.3	35	259.3
21.3	36	261.0
22.3	37	262.6
23.3	38	264.2
24.3	39	265.8
25.3	40	267.3
26.3	41	268.7
27.3	42	270.2
28.3	43	271.7
29.3	44	273.1
30.3	45	274.5

GAUGE PRESSURE psi	ABSOLUTE PRESSURE psi	TEMPERATURE degrees Fahrenheit
31.3	46	275.8
32.3	47	277.2
33.3	48	278.5
34.3	49	279.8
35.3	50	281.0
36.3	51	282.3
37.3	52	283.5
38.3	53	284.7
39.3	54	285.9
40.3	55	287.1
41.3	56	288.2
42.3	57	289.4
43.3	58	290.5
44.3	59	291.6
45.3	60	292.7
46.3	61	293.8
47.3	62	294.9
48.3	63	295.9
49.3	64	297.0
50.3	65	298.0
51.3	66	299.0
52.3	67	300.0
53.3	68	301.0
54.3	69	302.0
55.3	70	302.9
56.3	71	303.9
57.3	72	304.8
58.3	73	305.8
59.3	74	306.7
60.3	75	307.6
61.3	76	308.5
62.3	77	309.4
63.3	78	310.3
64.3	79	311.2
65.3	80	312.0
66.3	81	312.9
67.3	82	313.8
68.3	83	314.6
69.3	84	315.4
70.3	85	316.3
71.3	86	317.1
72.3	87	317.9
73.3	88	318.7
74.3	89	319.5
75.3	90	320.3
76.3	91	321.1
77.3	92	321.8
78.3	93	322.6
79.3	94	323.4
80.3	95	324.1
81.3	96	324.9
82.3	97	325.6
83.3	98	326.4
84.3	99	327.1
85.3	100	327.8
87.3	102	329.3
89.3	104	330.7
91.3	106	332.0
93.3	108	333.4
95.3	110	334.8

GAUGE PRESSURE psi	ABSOLUTE PRESSURE psi	TEMPERATURE degrees Fahrenheit
97.3	112	336.1
99.3	114	337.4
101.3	116	338.7
103.3	118	340.0
105.3	120	341.3
107.3	122	342.5
109.3	124	343.8
111.3	126	345.0
113.3	128	346.2
115.3	130	347.4
117.3	132	348.5
119.3	134	349.7
121.3	136	350.8
123.3	138	352.0
125.3	140	353.1
127.3	142	354.2
129.3	144	355.3
131.3	146	356.3
133.3	148	357.4
135.3	150	358.5
137.3	152	359.5
139.3	154	360.5
141.3	156	361.6
143.3	158	362.6
145.3	160	363.6
147.3	162	364.6
149.3	164	365.6
151.3	166	366.5
153.3	168	367.5
155.3	170	368.5
157.3	172	369.4
159.3	174	370.4
161.3	176	371.3
163.3	178	372.2
165.3	180	373.1
167.3	182	374.0
169.3	184	374.9
171.3	186	375.8
173.3	188	376.7
175.3	190	377.6
177.3	192	378.5
179.3	194	379.3
181.3	196	380.2
183.3	198	381.0
185.3	200	381.9
190.3	205	384.0
195.3	210	386.0
200.3	215	388.0
205.3	220	389.9
210.3	225	391.9
215.3	230	393.8
220.3	235	395.6
225.3	240	397.4
230.3	245	399.3
235.3	250	401.1
245.3	260	404.5
255.3	270	407.9
265.3	280	411.2
275.3	290	414.4
285.3	300	417.5