Bulletin OZ1000 03/04

Masoneilan Control Valve Sizing Handbook

Masoneilan

Experience, Knowledge & Technology...In Control

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Engineering Data

Foreword

This handbook on control valve sizing is based on the use of nomenclature and sizing equations from ANSI/ISA Standard S75.01.01 and IEC Standard 60534-2-1. Additional explanations and supportive information are provided beyond the content of the standards.

The sizing equations are based on equations for predicting the flow of compressible and incompressible fluids through control valves. The equations are not intended for use when dense slurries, dry solids or non-Newtonian liquids are encountered.

Original equations and methods developed by Masoneilan are included for two-phase flow, multistage flow, and supercritical fluids.

Values of numerical factors are included for commonly encountered systems of units. These are United States customary units and metric units for both kilopascal and bar usage.

The principal use of the equations is to aid in the selection of an appropriate valve size for a specific application. In this procedure, the numbers in the equations consist of values for the fluid and flow conditions and known values for the selected valve at rated opening. With these factors in the equation, the unknown (or product of the unknowns, e.g., $F_p C_V$) can be computed. Although these computed numbers are often suitable for selecting a valve from a series of discrete sizes, they do not represent a true operating condition. Some of the factors are for the valve at rated travel, while others relating to the operating conditions are for the partially open valve.

Once a valve size has been selected, the remaining unknowns, such as F_p, can be computed and a judgement can be made as to whether the valve size is adequate. It is not usually necessary to carry the calculations further to predict the exact opening. To do this, all the pertinent sizing factors must be known at fractional valve openings. A computer sizing program having this information in a database can perform this task.

Flow Coefficient C_V

The use of the flow coefficient, C_V, first introduced by Masoneilan in 1944, quickly became accepted as the universal yardstick of valve capacity. So useful has C_V become, that practically all discussions of valve design and characteristics or flow behavior now employ this coefficient.

By definition, the valve flow coefficient, C_V, is the number of U.S. gallons per minute of water at 60°F that will pass through a given flow restriction with a pressure drop of one psi. For example, a control valve that has a maximum flow coefficient, C_V, of 12 has an effective port area in the full open position such that it passes 12 gpm of water with one psi pressure drop. Basically, it is a capacity index upon which the engineer can rapidly and accurately estimate the required size of a restriction in any fluid system.

Operating Conditions

The selection of a correct valve size, as determined by formula, is always premised on the assumption of full knowledge of the actual flowing conditions. Frequently, one or more of these conditions is arbitrarily assumed. It is the evaluation of these arbitrary data that really determines the final valve size. **No formulas, only good common sense combined with experience, can solve this problem**.

There is no substitute for good engineering judgement. Most errors in sizing are due to incorrect assumptions as to actual flowing conditions. Generally speaking, the tendency is to make the valve too large to be on the "safe" side (commonly referred to as "oversizing"). A combination of several of these "safety factors" can result in a valve so greatly oversized it tends to be troublesome.

Specific Gravity

In the flow formulas, the specific gravity is a square root function; therefore, small differences in gravity have a minor effect on valve capacity. If the specific gravity is not known accurately, a reasonable assumption will suffice. The use of .9 specific gravity, for example, instead of .8 would cause an error of less than 5% in valve capacity.

Pressure Drop Across the Valve

On a simple back pressure or pressure reducing application, the drop across the valve may be calculated quite accurately. This may also be true on a liquid level control installation, where the liquid is passing from one vessel at a constant pressure to another vessel at a lower constant pressure. If the pressure difference is relatively small, some allowance may be necessary for line friction. On the other hand, in a large percentage of control applications, the pressure drop across the valve will be chosen arbitrarily.

Any attempt to state a specific numerical rule for such a choice becomes too complex to be practical. The design drop across the valve is sometimes expressed as a percentage of the friction drop in the system, exclusive of the valve. A good working rule is that 50% of this friction drop should be available as drop across the valve. In other words, one-third of the total system drop, including all heat exchangers, mixing nozzles, piping etc., is assumed to be absorbed by the control valve. This may sound excessive, but if the control valve were completely eliminated from such a system, the flow increase would only be about 23%. In pump discharge systems, the head characteristic of the pump becomes a major factor. For valves installed in extremely long or high-pressure drop lines, the percentage of drop across the valve may be somewhat lower, but at least 15% (up to 25% where possible) of the system drop should be taken.

Remember one important fact, the pressure differential absorbed by the control valve in actual operation will be the difference between the total available head and that required to maintain the desired flow through the valve. It is determined by the system characteristics rather than by the theoretical assumptions of the engineer. In the interest of economy, the engineer tries to keep the control valve pressure drop as low as possible. However, a valve can only regulate flow by absorbing and giving up pressure drop to the system. As the proportion of the system drop across the valve is reduced, its ability to further increase flow rapidly disappears.

In some cases, it may be necessary to make an arbitrary choice of the pressure drop across the valve because meager process data are available. For instance, if the valve is in a pump discharge line, having a discharge pressure of 7 bar (100 psi), a drop of 0.7 to 1.7 bar (10 to 25 psi) may be assumed sufficient. This is true if the pump discharge line is not extremely long or complicated by large drops through heat exchangers or other equipment. The tendency should be to use the higher figure.

On more complicated systems, consideration should be given to both maximum and minimum operating conditions. Masoneilan Engineering assistance is available for analysis of such applications.

Flowing Quantity

The selection of a control valve is based on the required flowing quantity of the process. The control valve must be selected to operate under several different conditions. The maximum quantity that a valve should be required to pass is 10 to 15% above the specified maximum flow. The normal flow and maximum flow used in size calculations should be based on actual operating conditions, whenever possible, without any factors having been applied.

On many systems, a reduction in flow means an increase in pressure drop, and the C_V ratio may be much greater than would be suspected. If, for example, the maximum operating conditions for a valve are 200 gpm and 25 psi drop, and the minimum conditions are 25 gpm and 100 psi drop, the C_V ratio is 16 to 1, not 8 to 1 as it would first seem. The required change in valve C_V is the product of the ratio of maximum to minimum flow and the square root of the ratio of maximum to minimum pressure drop, e.g.,

$$\frac{200 \times \sqrt{100}}{25 \times \sqrt{25}} = \frac{16}{1}$$

There are many systems where the increase in pressure drop for this same change in flow is proportionally much greater than in this case.

Liquid Flow Equations

Flow of Non-vaporizing Liquid

The following equations are used to determine the required capacity of a valve under fully turbulent, non-vaporizing liquid flow conditions. Note F_p equals unity for the case of valve size equal to line size.

volumetric flow

$$C_v = \frac{q}{N_1 \; F_p} \sqrt{\frac{G_f}{p_1 - p_2}}$$

mass flow

$$C_v = \frac{w}{N_6 \; F_P \; \sqrt{(p_1 - p_2) \; \gamma_1}}$$

Choked Flow of Vaporizing Liquid

Choked flow is a limiting flow rate. With liquid streams, choking occurs as a result of vaporization of the liquid when the pressure within the valve falls below the vapor pressure of the liquid.

Liquid flow is choked if

$$\Delta p \geq F_L^2 \; (p_1 - F_F \; p_v)$$

In this case, the following equations are used.

volumetric flow

$$C_v = \frac{q}{N_1 \; F_{LP}} \sqrt{\frac{G_f}{p_1 - F_F \; p_v}}$$

mass flow

$$C_v = \frac{w}{N_6 \; F_{LP} \; \sqrt{(p_1 - F_F \; p_v) \; \gamma_1}}$$

Nomenclature

C_V = valve flow coefficient
N = numerical constants based on units used (see Table 1)
F_p = piping geometry factor (reducer correction)
F_F = liquid critical pressure factor = $0.96 - 0.28\sqrt{\frac{p_v}{p_c}}$
F_L = liquid pressure recovery factor for a valve
F_{LP} = combined pressure recovery and piping geometry factor for a valve with attached fittings
K_i = velocity head factors for an inlet fitting, dimensionless
p_c = pressure at thermodynamic critical point
q = volumetric flow rate
G_f = specific gravity at flowing temperature (water = 1) @ 60˚F/15.5˚C
p_1 = upstream pressure
p_v = vapor pressure of liquid at flowing temperature
p_2 = downstream pressure
w = weight (mass) flow rate
γ_1 = specific weight (mass density) upstream conditions

Numerical Constants for Liquid Flow Equations

Constant		**Units Used in Equations**				
N		**w**	**q**	**p, Δp**	**d, D**	**γ_1**
N_1	0.0865	-	m^3/h	kPa	-	-
	0.865	-	m^3/h	bar	-	-
	1.00	-	gpm	psia	-	-
N_2	0.00214	-	-	-	mm	-
	890.0	-	-	-	in	-
N_6	2.73	kg/h	-	kPa	-	kg/m^3
	27.3	kg/h	-	bar	-	kg/m^3
	63.3	lb/h	-	psia	-	lb/ft^3

Table 1

Liquid Pressure Recovery Factor F_L

The liquid pressure recovery factor is a dimensionless expression of the pressure recovery ratio in a control valve. Mathematically, it is defined as follows:

$$F_L = \sqrt{\frac{p_1 - p_2}{p_1 - p_{vc}}}$$

In this expression, p_{vc} is the pressure at the vena contracta in the valve.

Liquid pressure recovery factors for various valve types at rated travel and at lower valve travel are shown in product bulletins. These values are determined by laboratory test in accordance with prevailing ISA and IEC standards.

Combined Liquid Pressure Recovery Factor F_{LP}

When a valve is installed with reducers, the liquid pressure recovery of the valve reducer combination is not the same as that for the valve alone. For calculations involving choked flow, it is convenient to treat the piping geometry factor F_p and the F_L factor for the valve reducer combination as a single factor F_{LP}. The value of F_L for the combination is then F_{LP}/F_p where :

$$\frac{F_{LP}}{F_p} = \sqrt{\frac{p_1 - p_2}{p_1 - p_{vc}}}$$

The following equation may be used to determine F_{LP}.

$$F_{LP} = F_L \left(\frac{K_i \; F_L^2 \; C_v^2}{N_2 \; d^4} + 1 \right)^{-1/2}$$

where $K_i = K_1 + K_{B1}$ (inlet loss and Bernoulli coefficients)

Cavitation in Control Valves

Cavitation, a detrimental process long associated with pumps, gains importance in control valves due to higher pressure drops for liquids and increased employment of high pressure recovery valves (e.g. butterfly and ball valves).

Cavitation, briefly, is the transformation of a portion of liquid into the vapor phase during rapid acceleration of the fluid in the valve orifice, and the subsequent collapse of vapor bubbles downstream. The collapse of vapor bubbles can produce localized pressure up to 100,000 psi (7000 bar) and are singly most responsible for the rapid erosion of valve trim under high pressure drop conditions.

It is, therefore, necessary to understand and to prevent this phenomenon, particularly when high pressure drop conditions are encountered.

Cavitation in a control valve handling a pure liquid may occur if the static pressure of the flowing liquid tends to decrease to a value less than the fluid vapor pressure. At this point, continuity of flow is broken by the formation of vapor bubbles. Since all control valves exhibit some pressure recovery, the final downstream pressure is generally higher than the orifice throat static pressure. When downstream pressure is higher than vapor pressure of the fluid, the vapor bubbles revert back to liquid. This two-stage transformation is defined as cavitation.

The pressure recovery in a valve is a function of its particular internal geometry. In general, the more streamlined a valve is, the more pressure recovery is experienced. This increases the possibility of cavitation.

The pressure recovery factor, F_L, is useful for valve sizing purposes to predict limiting choked flow rate under fully cavitating conditions. However, the use of F_L can be misleading to predict limiting pressure drop at which damaging cavitation will result.

An enhanced cavitation prediction method is described in the ISA Recommended Practice ISA-RP75.23-1995 “Considerations for Evaluating Control Valve Cavitation”. The recommended practice is based on the “Sigma” method, where sigma is defined as:

$$\sigma = \frac{(P_1 - P_V)}{(P_1 - P_2)}$$

The determination of sigma is based on cavitation energy levels, not on choked flow. Laboratory testing using high-frequency vibration data establishes sigma values. These sigma values then define different operational regimes for a specific product as illustrated below.

Cavitation Prediction "Sigma" Regimes

Four different operational regimes for each product and lift position.

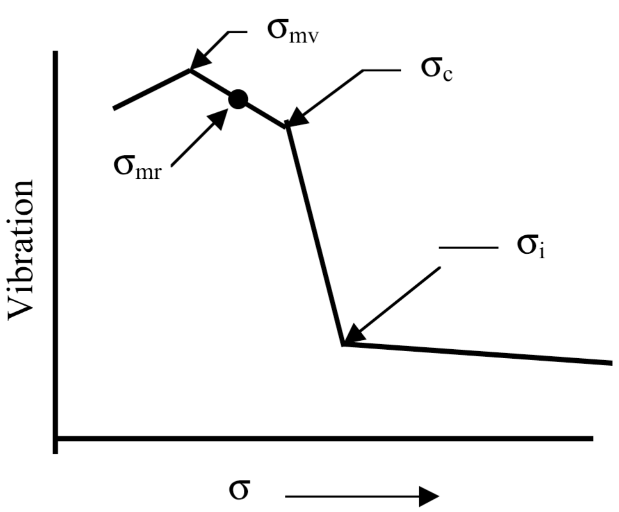
σ_i = Inception
σ_c = Constant
σ_{mv} = Maximum Vibration

Regime envelopes vary for each product and lift, and are based on laboratory testing.

σ_{mr} = Manufacturer's Recommended Limit

A series of tests have to be run on multiple valve sizes, and at multiple upstream pressures to establish performance curves for each product line.

Typical Sigma Curve



Characteristics of the different cavitation regimes are:

Incipient Cavitation:

- Onset of cavitation
- Detect using high frequency vibration measurement
- Very local phenomenon
- Transient: random "ticks" sound
- Low level cavitation: usually not damaging
- Occurs prior to loss of capacity

Constant Cavitation:

- More regular cavitation events
- Lower frequency sound and vibration sensed: "rumbling" sound
- Some damage to surfaces may occur: dependent upon valve and trim styles, and materials

Maximum Cavitation:

- Highest vibration amplitude: sounds like "marbles" or "gravel"
- Vigorous, large scale cavitation
- Predicted by steady flow pressure distribution ($\approx F_L$)
- Very high damage potential

Manufacturer's Recommended Limit:

- Valve style dependent
- Provided by manufacturer from combination of:
  - Application experience
  - Laboratory testing (Cavitation damage testing of aluminum parts)
- Varies with:
  - Size
  - Pressure
- Other application considerations:
  - Materials, usage duration and frequency, fluid properties
  - Fluid velocity
- Testing required for each product line:
  - Sigma curves established for each lift point
  - Minimum of two valve sizes of like geometry tested to establish size scaling factors
  - Minimum of two upstream pressures used to establish pressure scaling factors
  - The use of these two scaling factors allows the application of a particular valve geometry at various pressures and sizes while allowing the same cavitation energy levels to occur

Factors Impacting Cavitation Damage

Valve Size
Larger valves increase the extent of the cavitating region. Larger and more damaging bubble size.

Damage Scales with:

$$SSE = \left(\frac{d_v}{d_{ref}}\right)^{0.068\left(\frac{C_v}{{d_v}^2}\right)^{0.25}}$$

Driving Pressure
High pressure is more damaging
Quantified by exponent 'a'

Damage is proportional to: $(P_1 - P_2)^a$

'a' exponent is from testing at multiple P_1 levels

Scaling varies with valve Style and Geometry

Additional Factors Impacting Cavitation Damage: (Not scaled by ISA- RP75.23)

Fluid Properties

- Fluid Surface Tension
  - Higher tension, higher collapse energy, more damaging.
  - Water has very high surface tension.
  - Ammonia also has high surface tension.
- Fluid with multiple constituents
  - Multiple vapor pressures are less damaging as only a portion of the liquid cavitates at service condition.
  - Hydrocarbon mixtures are less damaging.
- Fluid with non-condensable gases
  - Favorable: Gas "cushions" bubble implosion, reducing overpressure and damage.
  - Unfavorable: Cavitation inception occurs "earlier", at higher application "sigma" over a larger region. Presence of gas or solid particles 'foster' the formation of bubbles.
- Temperature
  - Impacts gas solubility and degree of cushioning (favorable).
  - Pressure of vaporization, (unfavorable), higher temperature, higher P_v, increased cavitation possibility. Higher temperature decreases surface tension (favorable).

Valve Materials of Construction

- In most instances, ***ALL MATERIALS WILL EVENTUALLY FAIL!***
- Stain-Hardening: Material toughens as it plastically deforms, this is a positive trait.
- Ductility: Ability to deform vs. fracture. Ductile materials exhibit greater resistance than brittle materials.
- Hardness: This is the most important quality, the ability to resist surface pressures. The higher the hardness the greater the resistance.
- Temperature effects material properties, higher temperatures decrease material yield strength levels.

Cavitation Can Worsen Corrosion and Chemical Attack on Materials

- Cavitation weakens material facilitating corrosion attack (and visa versa).
- Cavitation expedites removal of weakened material.
- Cavitation removes protective oxide layers, greatly accelerating additional material removal.

Additional Considerations:
Some designs can allow a degree of cavitation to occur, however, by controlling the location and energy levels, damage is avoided (Cavitation Containment Designs). For these designs the following considerations, along with the Sigma index, are also important and additional limitations are applied:

- Inlet and inter-stage pressure levels
- Valve body velocity
- Trim velocity
- Sound power levels

Calculation Method

1. Calculate Applications using Service Conditions

$$\sigma = \left(\frac{P_1 - P_v}{P_1 - P_2}\right)$$

2. Calculate Operating C_V

3. From Product Rating @ C_V Find σ_{mr}

4. Scale σ_{mr} to Service Conditions

4.1 Calculate Size Scaling Effect SSE

$$SSE = \left(\frac{d}{d_r}\right)^b$$

d_r = Ref. Valve Size
d = Application Valve Size
b = Size Scaling Exponent

4.2 Calculate Pressure Scaling Effect PSE

$$PSE = \left(\frac{(P_1 - P_v)}{(P_1 - P_v)_r}\right)^a$$

$(P_1 - P_v)_r$ = Reference from Testing

4.3 σ_{mr} Scaled to Service Conditions and Valve Size

$$\sigma_v = \left[(\sigma_{mr})SSE - 1\right] PSE + 1$$

5. IF σ >/= σ_V Valve is OK for Application
IF σ < σ_V Valve is Not Acceptable for the Application

Note: See Nomenclature page 10

Calculation Example

Conditions: Water, P_1 = 275 psia, P_2 = 75 psia, P_V= 4.0
C_V req'd = 21, 3 inch Pipe Line

1. $\sigma = \left(\frac{275 - 4}{275 - 75}\right)$ = 1.36

2. Try 2 Inch Camflex @ C_V = 21, F-T-O

3. σ_{mr} = 1.15 @ C_V = 21

4. Scale σ_{mr} to Service conditions

4.1 SSE = $\left(\frac{2}{1}\right)^{0.132}$ = 1.096

4.2 PSE = $\left(\frac{275 - 4}{100}\right)^{0.4}$ = 1.49

4.3 $\sigma_V = \left[(\sigma_{mr})1.096 - 1\right] 1.49 + 1$ = 1.39

5. As σ(1.36) < σ_V(1.39), – Valve is Not Acceptable –

Try 3 Inch Camflex in the 3 Inch Line
@ C_V = 21, σ_{mr} = 1.06

New SSE SSE = $\left(\frac{3}{1}\right)^{0.132}$ = 1.156

PSE = 1.49

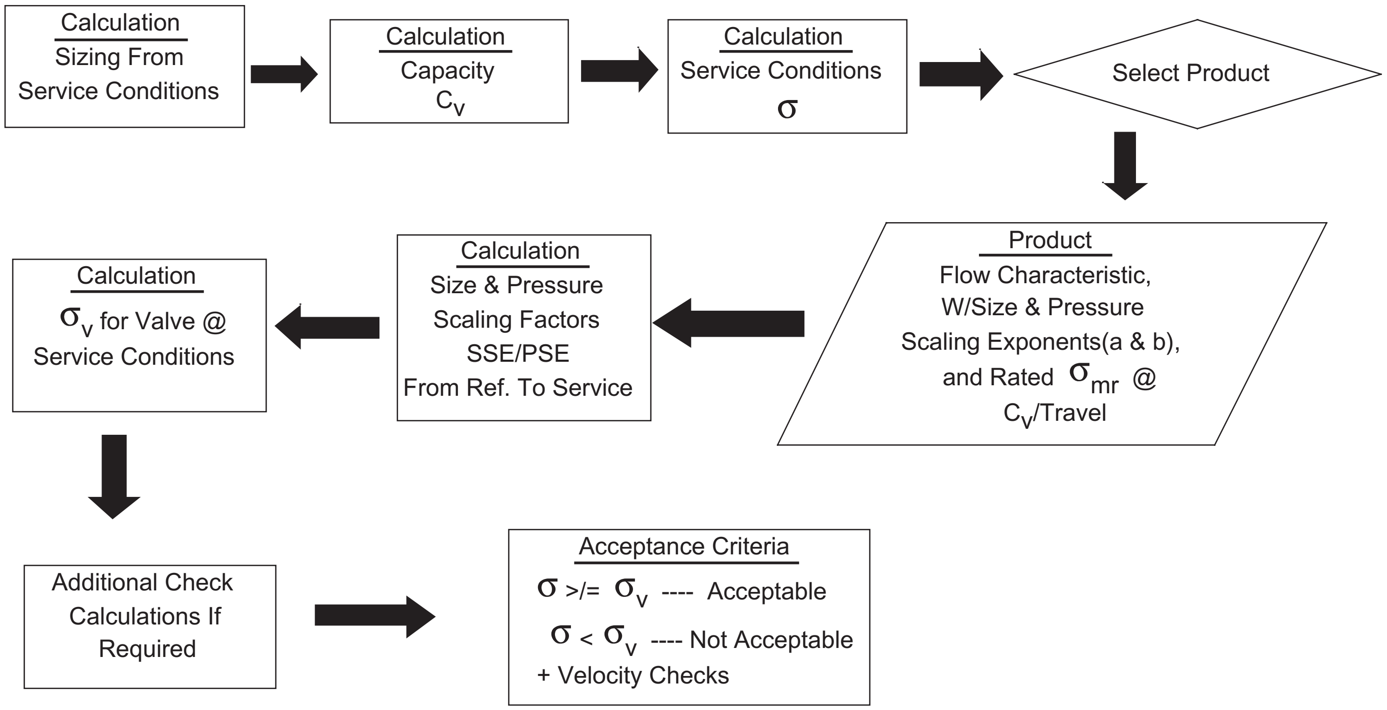
New σ_V

$$\sigma_V = \left[(\sigma_{mr})1.156 - 1\right] 1.49 + 1 = 1.34$$

As σ(1.36) > σ_V(1.34), – Valve is Acceptable –

Note: Also Check Body Velocity on Camflex

Calculation Flow Chart



Nomenclature

a	Empirical characteristic exponent for calculating PSE
b	A characteristic exponent for calculating SSE; determined from reference valve data for geometrically similar valves.
C_V	Valve flow coefficient, $C_V = q(G_f/\Delta P)^{1/2}$
d	Valve inlet inside diameter, inches
d_r	Valve inlet inside diameter of tested reference valve, inches
F_L	Liquid pressure recovery factor
P_1	Valve inlet static pressure, psia
P_2	Valve outlet static pressure, psia
PSE	Pressure Scale Effect
P_v	Absolute fluid vapor pressure of liquid at inlet temperature, psia
SSE	Size Scale Effect
σ	Cavitation index equal to $(P_1-P_V)/(P_1-P_2)$ at service conditions, i.e., σ (service)
σ_c	Coefficient for constant cavitation; is equal to $(P_1-P_V)/\Delta P$ at the conditions causing steady cavitation.
σ_i	Coefficient for incipient cavitation; is equal to $(P_1-P_V)/\Delta P$ at the point where incipient cavitation begins to occur.
σ_{mr}	Coefficient of manufacturer's recommended minimum limit of the cavitation index for a specified valve and travel; is equal to minimum recommended value of $(P_1-P_V)/\Delta P$.
σ_{mv}	Coefficient of cavitation causing maximum vibration as measured on a cavitation parameter plot.

Effect of Pipe Reducers

When valves are mounted between pipe reducers, there is a decrease in actual valve capacity. The reducers cause an additional pressure drop in the system by acting as contractions and enlargements in series with the valve. The Piping Geometry Factor, F_p, is used to account for this effect.

Piping Geometry Factor

$$F_p = \left(\frac{C_V^2 \; \Sigma K}{N_2 \; d^4} + 1 \right)^{-1/2}$$

Pipe Reducer Equations

Loss Coefficients

inlet $$K_1 = 0.5 \left[1 - \left(\frac{d}{D_1} \right)^2 \right]^2$$

outlet $$K_2 = \left[1 - \left(\frac{d}{D_2} \right)^2 \right]^2$$

Bernoulli Coefficients

$$K_{B1} = 1 - \left(\frac{d}{D_1} \right)^4$$

$$K_{B2} = 1 - \left(\frac{d}{D_2} \right)^4$$

Summation

$$\Sigma K = K_1 + K_2 + K_{B1} - K_{B2}$$

When inlet and outlet reducers are the same size, the Bernoulli coefficients cancel out.

Nomenclature

C_V = valve flow capacity coefficient

d = valve end inside diameter

D_1 = inside diameter of upstream pipe

D_2 = inside diameter of downstream pipe

F_p = piping geometry factor, dimensionless

K_1 = pressure loss coefficient for inlet reducer, dimensionless

K_2 = pressure loss coefficient for outlet reducer, dimensionless

K_{B1} = pressure change (Bernoulli) coefficient for inlet reducer, dimensionless

K_{B2} = pressure change (Bernoulli) coefficient for outlet reducer, dimensionless

ΣK = $K_1 + K_2 + K_{B1} - K_{B2}$, dimensionless

Equations for Non-turbulent Flow

Laminar or transitional flow may result when the liquid viscosity is high, or when valve pressure drop or C_V is small. The Valve Reynolds Number Factor is used in the equations as follows :

volumetric flow $$C_v = \frac{q}{N_1 \; F_R} \sqrt{\frac{G_f}{p_1 - p_2}}$$

mass flow $$C_v = \frac{w}{N_6 \; F_R \; \sqrt{(p_1 - p_2) \; \gamma_1}}$$

The valve Reynolds number is defined as follows :

$$Re_v = \frac{N_4 \, F_d \, q}{\nu \, F_L^{1/2} \; C_v^{1/2}} \left(\frac{F_L^2 \; C_v^2}{N_2 \, d^4} + 1 \right)^{1/4}$$

The Valve Reynolds Number Re_v is used to determine the Reynolds Number Factor F_R. The factor F_R can be estimated from curves in the existing ISA and IEC standards, or by calculation methods shown in the standards. Iteration is required in the method shown in the IEC standard.

Nomenclature

C_V = valve flow capacity coefficient

d = nominal valve size

F_d = valve style modifier, dimensionless

F_L = Liquid pressure recovery factor

F_R = Reynolds number correction factor, dimensionless

G_f = specific gravity at flowing temperature (water = 1) @ 60˚F/15.5˚C

Δp = valve pressure drop

q = volumetric flow rate

Re_v = valve Reynolds number, dimensionless

w = weight (mass) flow rate

γ = mass density of liquid

ν = kinematic viscosity, centistokes

Numerical Constants for Liquid Flow Equations

Constant		Units Used in Equations				
N		w	q	p, Δp	d, D	γ_1
N_1	0.0865	-	m^3/h	kPa	-	-
	0.865	-	m^3/h	bar	-	-
	1.00	-	gpm	psia	-	-
N_2	0.00214	-	-	-	mm	-
	890.0	-	-	-	in	-
N_4	76000	-	m^3/h	-	mm	-
	17300	-	gpm	-	in	-
N_6	2.73	kg/h	-	kPa	-	kg/m^3
	27.3	kg/h	-	bar	-	kg/m^3
	63.3	lb/h	-	psia	-	lb/ft^3

Table 2

Gas and Vapor Flow Equations

volumetric flow

$$C_v = \frac{q}{N_7 \; F_p \; p_1 \; Y} \sqrt{\frac{G_g \; T_1 \; Z}{x}}$$

or

$$C_v = \frac{q}{N_9 \; F_p \; p_1 \; Y} \sqrt{\frac{M \; T_1 \; Z}{x}} \quad *$$

mass flow

$$C_v = \frac{w}{N_6 \; F_p \; Y \; \sqrt{x p_1 \; \gamma_1}} \quad *$$

or

$$C_v = \frac{w}{N_8 \; F_p \; p_1 \; Y} \sqrt{\frac{T_1 \, Z}{x \; M}} \quad *$$

Gas expansion factor

$$Y = 1 - \frac{x}{3 \, F_k \; x_T}$$

Pressure drop ratio

$$x = \frac{\Delta p}{p_1}$$

Ratio of specific heats factor

$$F_k = \frac{k}{1.40}$$

*The IEC 534-2 equations are identical to the above ISA equations (marked with an *) except for the following symbols:

k (ISA) corresponds to γ (IEC)
γ_1 (ISA) corresponds to ρ_1 (IEC)

Nomenclature

C_V = valve flow coefficient
F_k = ratio of specific heats factor, dimensionless
F_P = piping geometry factor (reducer correction)
p_1 = upstream pressure
p_2 = downstream pressure
q = volumetric flow rate
N = numerical constant based on units (see table below)
G_g = gas specific gravity. Ratio of gas density at standard conditions
T_1 = absolute inlet temperature
M = gas molecular weight
x = pressure drop ratio, $\Delta p/p_1$ Limit $x = F_k x_T$
Z = gas compressibility factor
Y = gas expansion factor, $Y = 1 - \frac{x}{3 \, F_k \, x_T}$
x_T = pressure drop ratio factor
γ_1 = (Gamma) specific weight (mass density), upstream conditions
w = weight (mass) flow rate
k = gas specific heat ratio

Numerical Constants for Gas and Vapor Flow Equations

Constant		Units Used in Equations				
N		w	q*	p, Δp	γ_1	T_1
N_6	2.73	kg/h	-	kPa	kg/m^3	-
	27.3	kg/h	-	bar	kg/m^3	-
	63.3	lb/h	-	psia	lb/ft^3	-
N_7	4.17	-	m^3/h	kPa	-	K
	417.0	-	m^3/h	bar	-	K
	1360.0	-	scfh	psia	-	R
N_8	0.948	kg/h	-	kPa	-	K
	94.8	kg/h	-	bar	-	K
	19.3	lb/h	-	psia	-	R
N_9	22.5	-	m^3/h	kPa	-	K
	2250.0	-	m^3/h	bar	-	K
	7320.0	-	scfh	psia	-	R

*q is in cubic feet per hour measured at 14.73 psia and 60˚F, or cubic meters per hour measured at 101.3 kPa and 15.6˚ C.

Table 3

Multistage Valve Gas and Vapor Flow Equations

volumetric flow

$$C_v = \frac{q}{N_7 \; F_p \; p_1 \; Y_M} \sqrt{\frac{G_g \; T_1 \; Z}{x}}$$

or

$$C_v = \frac{q}{N_9 \; F_p \; p_1 \; Y_M} \sqrt{\frac{M \; T_1 \; Z}{x}}$$

mass flow

$$C_v = \frac{w}{N_6 \; F_p \; Y_M \; \sqrt{x \; p_1 \; \gamma_1}}$$

or

$$C_v = \frac{w}{N_8 \; F_p \; p_1 \; Y_M} \sqrt{\frac{T_1 \; Z}{x \; M}}$$

$$Y_M = 1 - \frac{x_M}{3 \; F_k \; x_T}$$

$$x_M = F_M \frac{\Delta p}{p_1} \text{ , limit } x_M = F_k \, x_T$$

$$F_k = \frac{k}{1.40}$$

F_M = Multistage Compressible Flow Factor
(F_M = 0.74 for multistage valves)

X_M = Pressure drop ratio factor for multistage valves

Ratio of Specific Heats Factor F_k

The flow rate of a compressible fluid through a valve is affected by the ratio of specific heats. The factor F_k accounts for this effect. F_k has a value of 1.0 for air at moderate temperature and pressures, where its specific heat ratio is about 1.40.

For valve sizing purposes, F_k may be taken as having a linear relationship to k. Therefore,

$$F_k = \frac{k}{1.40}$$

Expansion Factor Y

The expansion factor accounts for the changes in density of the fluid as it passes through a valve, and for the change in the area of the vena contracta as the pressure drop is varied. The expansion factor is affected by all of the following influences :

1. Ratio of valve inlet to port area
2. Internal valve geometry
3. Pressure drop ratio, x
4. Ratio of specific heats, k
5. Reynolds Number

The factor x_T accounts for the influence of 1, 2 and 3; factor F_k accounts for the influence of 4. For all practical purposes, Reynolds Number effects may be disregarded for virtually all process gas and vapor flows.

As in the application of orifice plates for compressible flow measurement, a linear relationship of the expansion factor Y to pressure drop ratio x is used as below :

$$Y = 1 - \frac{x}{3 \, F_k \; x_T}$$

Two-Phase Flow Equations

Two-phase flow can exist as a mixture of a liquid with a non-condensable gas or as a mixture of a liquid with its vapor. The flow equation below applies where the two-phase condition exists at the valve inlet.

The flow equation accounts for expansion of the gas or vapor phase, and for possible vaporization of the liquid phase. It utilizes both the gas and liquid limiting sizing pressure drops.

The flow equation for a two phase mixture entering the valve is as follows.

Note : F_p equals unity for the case of valve size equal to line size.

$$C_V = \frac{w}{N_6 F_p}\sqrt{\frac{f_f}{\Delta p_f \gamma_f} + \frac{f_g}{\Delta p_g \gamma_g Y^2}}$$

Use the actual pressure drop for Δp_f and Δp_g, but with the limiting pressure drop for each individually as follows :

$$\Delta p_f = F_L^2 \; (p_1 - F_F \; p_v)$$

$$\Delta p_g = F_k \; x_T \; p_1$$

The use of this flow equation results in a required C_V greater than the sum of a separately calculated C_V for the liquid plus a C_V for the gas or vapor phase. This increased capacity models published two-phase flow data quite well.

For the hypothetical case of all liquid flow ($f_f = 1$), the flow equation reduces to the liquid flow equation for mass flow.

For the hypothetical case of all gas or vapor flow ($f_g = 1$), the flow equation reduces to the gas and vapor flow equation for mass flow.

Nomenclature

C_V = valve flow coefficient
f_f = weight fraction of liquid in two-phase mixture, dimensionless
f_g = weight fraction of gas (or vapor) in two-phase mixture, dimensionless
F_F = liquid critical pressure factor = $0.96 - 0.28 \sqrt{\frac{p_v}{p_c}}$
F_k = ratio of specific heats factor, dimensionless
F_L = liquid pressure recovery factor
F_p = piping geometry factor (reducer correction)
p_1 = upstream pressure
p_v = vapor pressure of liquid at flowing temperature
Δp_f = pressure drop for the liquid phase
Δp_g = pressure drop for the gas phase
w = weight (mass) flow rate of two-phase mixture
x_T = pressure drop ratio factor
Y = gas expansion factor, $Y = 1 - \frac{x}{3 \; F_k \, x_T}$
γ_f = specific weight (mass density) of the liquid phase at inlet conditions
γ_g = specific weight (mass density) of the gas or vapor phase at inlet conditions

Numerical Constants for Liquid Flow Equations

Constant		Units Used in Equations				
N		w	q	p, Δp	d, D	γ_1
N_6	2.73	kg/h	-	kPa	-	kg/m^3
	27.3	kg/h	-	bar	-	kg/m^3
	63.3	lb/h	-	psia	-	lb/ft^3

Table 4

Choked Flow (Gas and Vapor)

If all inlet conditions are held constant and pressure drop ratio x is increased by lowering the downstream pressure, mass flow will increase to a maximum limit. Flow conditions where the value of x exceeds this limit are known as choked flow. Choked flow occurs when the jet stream at the vena contracta attains its maximum cross-sectional area at sonic velocity.

Values of x_T for various valve types at rated travel and at lower valve travel are shown in product bulletins. These values are determined by laboratory test.

When a valve is installed with reducers, the pressure ratio factor x_{TP} is different from that of the valve alone x_T. The following equation may be used to calculate x_{TP} :

$$\mathbf{x_{TP}} = \frac{\mathbf{x_T}}{\mathbf{F_p}^2}\left(\frac{\mathbf{x_T}\ \mathbf{K_i}\ \mathbf{C_v}^2}{\mathbf{N}_5\ \mathbf{d}^4} + 1\right)^{\mathbf{-1}} \quad , \qquad \text{where}$$

$$\mathbf{K_i = K_1 + K_{B1}} \text{ (inlet loss and Bernoulli coefficients)}$$

The value of N_5 is 0.00241 for d in mm, and 1000 for d in inches.

Supercritical Fluids

Fluids at temperatures and pressures above both critical temperature and critical pressure are denoted as supercritical fluids. In this region, there is no physical distinction between liquid and vapor. The fluid behaves as a compressible, but near the critical point great deviations from the perfect gas laws prevail. It is very important to take this into account through the use of actual specific weight (mass density) from thermodynamic tables (or the compressibility factor Z), and the actual ratio of specific heats.

Supercritical fluid valve applications are not uncommon. In addition to supercritical fluid extraction processes, some process applications may go unnoticed. For instance, the critical point of ethylene is 10˚C (50˚F) and 51.1 bar (742 psia). All ethylene applications above this point in both temperature and pressure are supercritical by definition.

In order to size valves handling supercritical fluids, use a compressible flow sizing equation with the weight (mass) rate of flow with actual specific weight (mass density), or the volumetric flow with actual compressibility factor. In addition, the actual ratio of specific heats should be used.

Compressibility Factor Z

For many real gases subjected to commonly encountered temperatures and pressures, the perfect gas laws are not satisfactory for flow measurement accuracy and therefore correction factors must be used.

Following conventional flow measurement practice, the compressibility factor Z, in the equation PV = ZRT, will be used. Z can usually be ignored below 7 bar (100 psi) for common gases.

The value of Z does not differ materially for different gases when correlated as a function of the reduced temperature, T_r , and reduced pressure, p_r , found from Figures 1 and 2.

Figure 2 is an enlargement of a portion of Figure 2. Values taken from these figures are accurate to approximately plus or minus two percent.

To obtain the value of Z for a pure substance, the reduced pressure and reduced temperature are calculated as the ratio of the actual absolute gas pressure and its corresponding critical absolute pressure and absolute temperature and its absolute critical temperature.

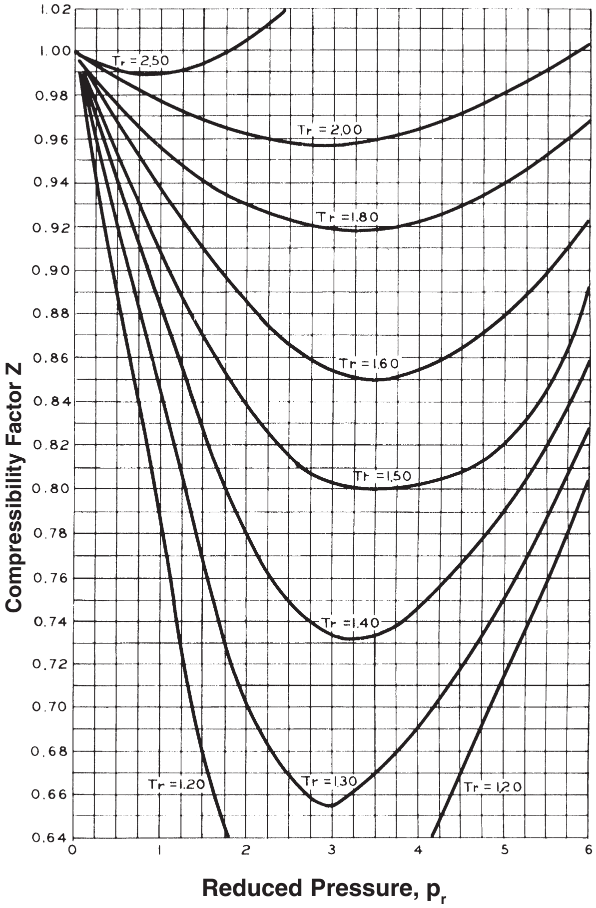
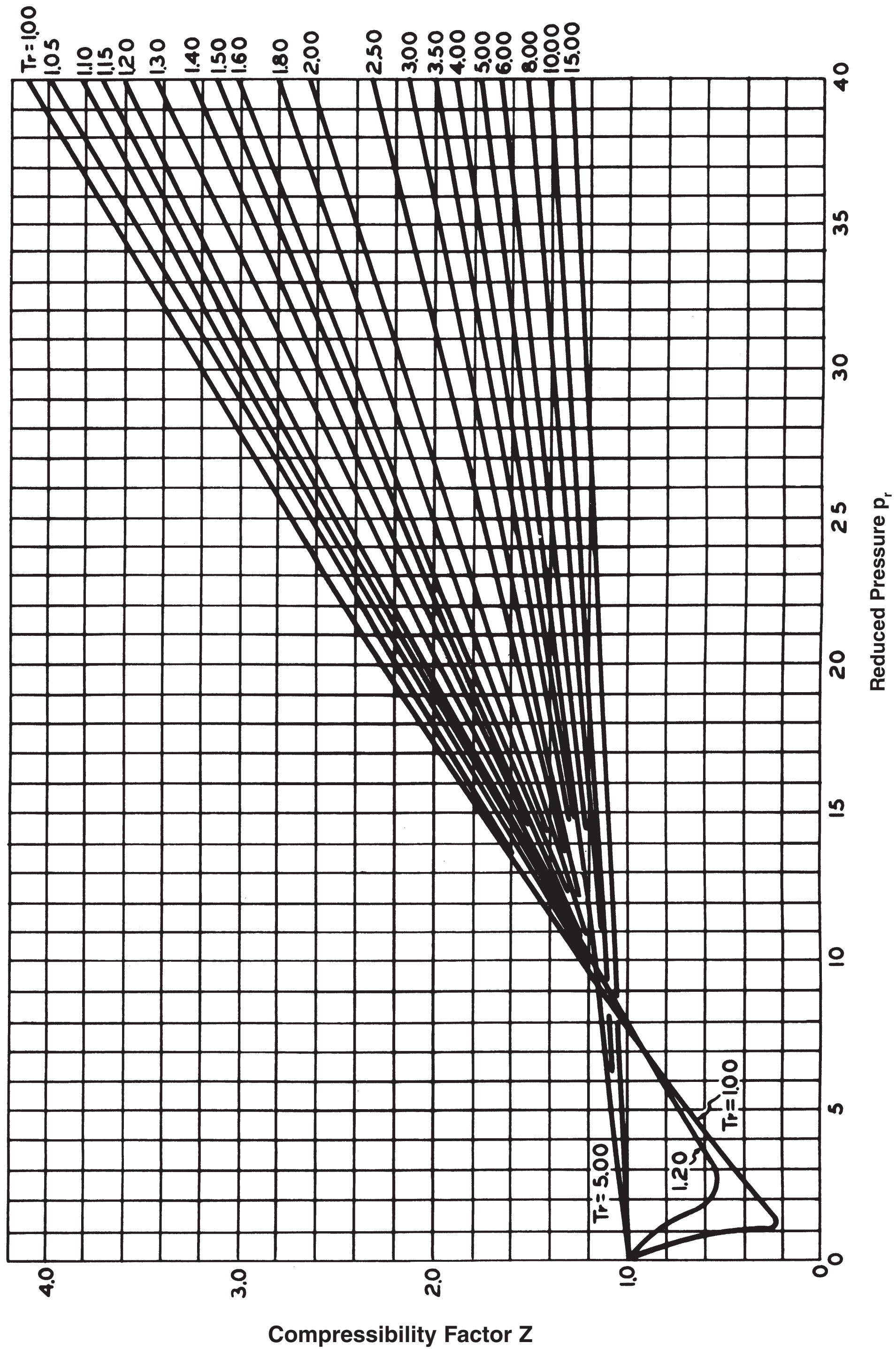


Figure 1
Compressibility Factors for Gases with Reduced Pressures from 0 to 6

(Data from the charts of L. C. Nelson and E. F. Obert, Northwestern Technological Institute)

The compressibility factor Z obtained from the Nelson-Obert charts is generally accurate within 3 to 5 percent. For hydrogen, helium, neon and argon, certain restrictions apply. Please refer to specialized literature.

Compressibility



$P_r = \dfrac{\text{inlet pressure (absolute)}}{\text{critical pressure (absolute)}}$ $T_r = \dfrac{\text{inlet temperature (absolute)}}{\text{critical temperature (absolute)}}$

Figure 2
Compressibility Factors for Gases with Reduced Pressures from 0 - 40
See Page 15 for critical pressures and temperatures

(Reproduced from the charts of L. C. Nelson and E. F. Obert, Northwestern Technological Institute)

Thermodynamic Critical Constants and Density of Elements, Inorganic and Organic Compounds

Element or Compound	Critical Pressure - p_c		Critical Temperature - T_c		k *
	psia	bar (abs)	˚F	˚C	C_p / C_V
Acetic Acid, CH_3-CO-OH	841	58.0	612	322	1.15
Acetone, CH_3-CO-CH_3	691	47.6	455	235	-
Acetylene, C_2H_2	911	62.9	97	36	1.26
Air, O_2+N_2	547	37.8	-222	-141	1.40
Ammonia, NH_3	1638	113.0	270	132	1.33
Argon, A	705	48.6	-188	-122	1.67
Benzene, C_6H_6	701	48.4	552	289	1.12
Butane, C_4H_{10}	529	36.5	307	153	1.09
Carbon Dioxide, CO_2	1072	74.0	88	31	1.30
Carbon Monoxide, CO	514	35.5	-218	-139	1.40
Carbon Tetrachloride, CCl_4	661	45.6	541	283	-
Chlorine, Cl_2	1118	77.0	291	144	1.36
Ethane, C_2H_6	717	49.5	90	32	1.22
Ethyl Alcohol, C_2H_5OH	927	64.0	469	243	1.13
Ethylene, CH_2=CH_2	742	51.2	50	10	1.26
Ethyl Ether, C_2H_5-O-C_2H_5	522	36.0	383	195	-
Fluorine, F_2	367	25.3	-247	-155	1.36
Helium, He	33.2	2.29	-450	-268	1.66
Heptane, C_7H_{16}	394	27.2	513	267	-
Hydrogen, H_2	188	13.0	-400	-240	1.41
Hydrogen Chloride, HCl	1199	82.6	124	51	1.41
Isobutane, (CH_3) CH-CH_3	544	37.5	273	134	1.10
Isopropyl Alcohol, CH_3-CHOH-CH_3	779	53.7	455	235	-
Methane, CH_4	673	46.4	-117	-83	1.31
Methyl Alcohol, H-CH_2OH	1156	79.6	464	240	1.20
Nitrogen, N_2	492	34.0	-233	-147	1.40
Nitrous Oxide, N_2O	1054	72.7	99	37	1.30
Octane, CH_3-$(CH_2)_6$-CH_3	362	25.0	565	296	1.05
Oxygen, O_2	730	50.4	-182	-119	1.40
Pentane, C_5H_{12}	485	33.5	387	197	1.07
Phenol, C_6H_5OH	889	61.3	786	419	-
Phosgene, $COCl_2$	823	56.7	360	182	-
Propane, C_3H_8	617	42.6	207	97	1.13
Propylene, CH_2=CH-CH_3	661	45.6	198	92	1.15
Refrigerant 12, CCl_2F_2	582	40.1	234	112	1.14
Refrigerant 22, $CHClF_2$	713	49.2	207	97	1.18
Sulfur Dioxide, SO_2	1142	78.8	315	157	1.29
Water, H_2O	3206	221.0	705	374	1.32

* Standard Conditions

Table 5

Thermodynamic Critical Constants and Density of Elements, Inorganic and Organic Compounds

Element or Compound	Density - lb/ft^3 14.7 psia & 60˚F		Density - kg/m^3 1013 mbar & 15.6˚C		Mol Wt
	Liquid	Gas	Liquid	Gas	
Acetic Acid, CH_3-CO-OH	65.7		1052.4		66.1
Acetone, CH_3-CO-CH_3	49.4		791.3		58.1
Acetylene, C_2H_2		0.069		1.11	26.0
Air, O_2+N_2		0.0764		1.223	29.0
Ammonia, NH_3		0.045		0.72	17.0
Argon, A		0.105		1.68	39.9
Benzene, C_6H_6	54.6		874.6		78.1
Butane, C_4H_{10}		0.154		2.47	58.1
Carbon Dioxide, CO_2		0.117		1.87	44.0
Carbon Monoxide, CO		0.074		1.19	28.0
Carbon Tetrachloride, CCl_4	99.5		1593.9		153.8
Chlorine, Cl_2		0.190		3.04	70.9
Ethane, C_2H_6		0.080		1.28	30.1
Ethyl Alcohol, C_2H_5OH	49.52		793.3		46.1
Ethylene, CH_2=CH_2		0.074		1.19	28.1
Ethyl Ether, C_2H_5-O-C_2H_5	44.9		719.3		74.1
Fluorine, F_2		0.097		1.55	38.0
Helium, He		0.011		0.18	4.00
Heptane, C_7H_{16}	42.6		682.4		100.2
Hydrogen, H_2		0.005		0.08	2.02
Hydrogen Chloride, HCl		0.097		1.55	36.5
Isobutane, $(CH_3)_2$ CH-CH_3		0.154		2.47	58.1
Isopropyl Alcohol, CH_3-CHOH-CH_3	49.23		788.6		60.1
Methane, CH_4		0.042		0.67	16.0
Methyl Alcohol, H-CH_2OH	49.66		795.5		32.0
Nitrogen, N_2		0.074		1.19	28.0
Nitrous Oxide, N_2O		0.117		1.87	44.0
Octane, CH_3-$(CH_2)_6$-CH_3	43.8		701.6		114.2
Oxygen, O_2		0.084		1.35	32.0
Pentane, C_5H_{12}	38.9		623.1		72.2
Phenol, C_6H_5OH	66.5		1065.3		94.1
Phosgene, $COCl_2$		0.108		1.73	98.9
Propane, C_3H_8		0.117		1.87	44.1
Propylene, CH_2=CH-CH_3		0.111		1.78	42.1
Refrigerant 12, CCl_2F_2		0.320		5.13	120.9
Refrigerant 22, $CHClF_2$		0.228		3.65	86.5
Sulfur Dioxide, SO_2		0.173		2.77	64.1
Water, H_2O	62.34		998.6		18.0

Table 5 (cont.)

Liquid Velocity in Commercial Wrought Steel Pipe

The velocity of a flowing liquid may be determined by the following expressions :

U.S. Customary Units

$$v = .321 \frac{q}{A}$$

Where
v = velocity, ft/sec
q = flow, gpm
A = cross sectional area, sq in

Metric Units

$$v = 278 \frac{q}{A}$$

Where
v = velocity, meters/sec
q = flow, meters^3/hr
A = cross sectional area, sq mm

Figure 3 gives the solution to these equations for pipes 1" through 12" over a wide flow range on both U.S. Customary and Metric Units.

Steam or Gas Flow in Commercial Wrought Steel Pipe

Steam or Gas (mass basis)

To determine the velocity of a flowing compressible fluid use the following expressions :

U.S. Customary Units

$$v = .04 \frac{WV}{A}$$

Where
v = fluid velocity, ft/sec
W = fluid flow, lb/hr
V = specific volume, cu ft/lb
A = cross sectional area, sq in

Metric Units

$$v = 278 \frac{WV}{A}$$

Where
v = fluid velocity, meters/sec
W = fluid flow, kg/hr
V = specific volume, m^3/kg
A = cross sectional area, mm^2

Figure 4 is a plot of steam flow versus static pressure with reasonable velocity for Schedule 40 pipes 1" through 12" in U.S. Customary and Metric Units.

Gas (volume basis)

To find the velocity of a flowing compressible fluid with flow in volume units, use the following formulas :

U.S. Customary Units

$$v = .04 \frac{F}{A}$$

Where
v = fluid velocity, ft/sec
F = gas flow, ft^3/hr at flowing conditions*
A = cross sectional area, sq in

*Note that gas flow must be at flowing conditions. If flow is at standard conditions, convert as follows :

$$F = \frac{\text{std ft}^3}{\text{hr}} \times \frac{14.7}{p} \times \frac{T}{520}$$

Where
p = pressure absolute, psia
T = temperature absolute, R

Metric Units

$$v = 278 \frac{F}{A}$$

Where
v = fluid velocity, meters/sec
F = gas flow, meters^3/hr at flowing conditions*
A = cross sectional area, sq mm

*Note that gas flow must be at flowing conditions. If flow is at standard conditions, convert as follows :

$$F = \frac{\text{std meters}^3}{\text{hr}} \times \frac{1.013}{p} \times \frac{T}{288}$$

Where
p = pressure absolute, bar
T = temperature absolute, K

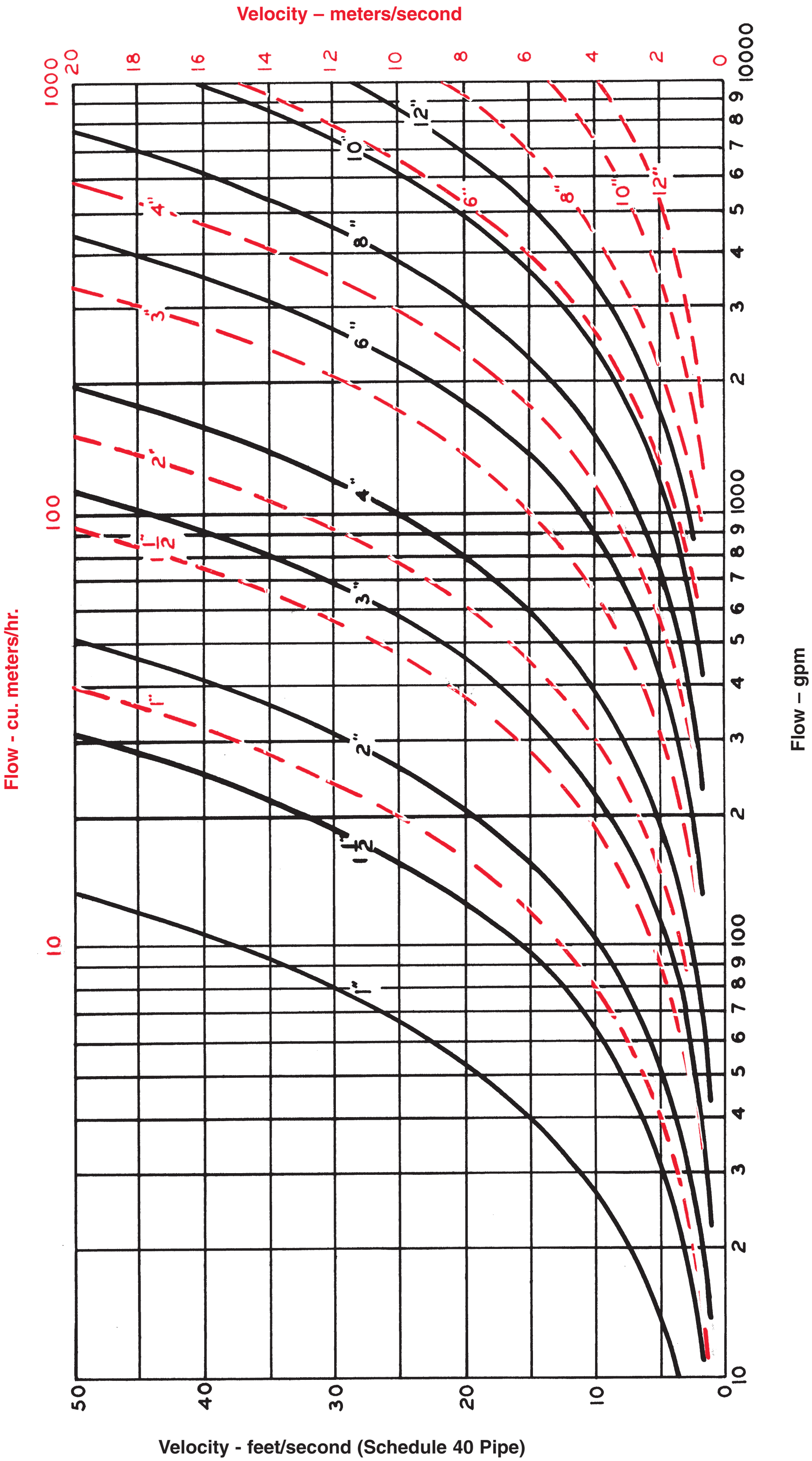


Figure 3

Liquid Velocity vs Flow Rate

☐ **U.S. Customary Units**

☐ **Metric Units**

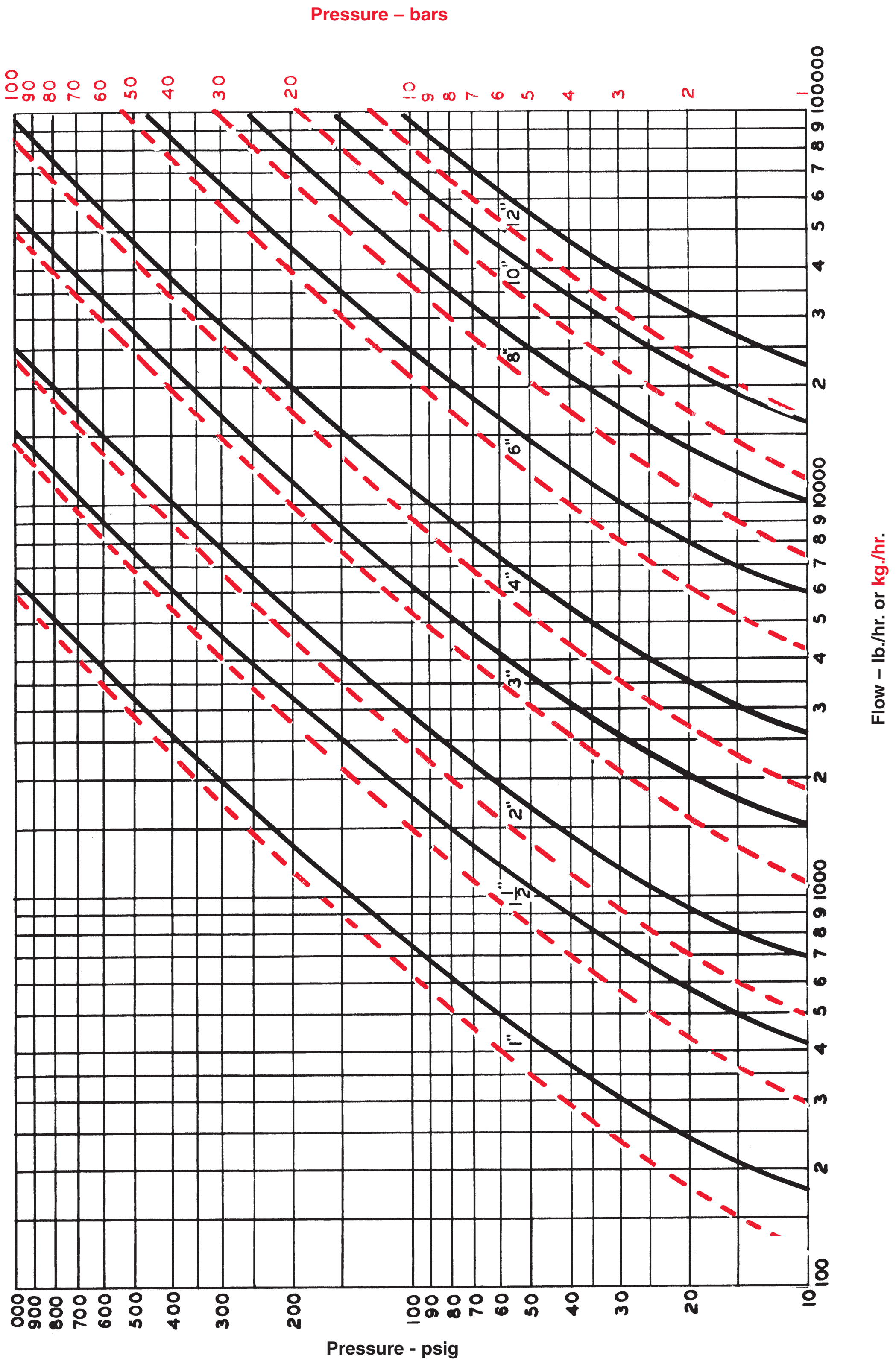


Figure 4
Saturated Steam Flow vs Pressure
for 1" to 12" Schedule 40 Pipe

☐ **U.S. Customary Units**

☐ **Metric Units**

Velocity -- 130 to 170 feet per second --
-- 50 to 60 meters per second --

Commercial Wrought Steel Pipe Data (ANSI B36.10)

	Nominal Pipe Size		O.D.	Wall Thickness		I.D.	Flow Area	
	mm	inches	inches	mm	inches	inches	mm^2	sq in
Schedule 10	350	14	14	6.35	0.250	13.5	92200	143
	400	16	16	6.35	0.250	15.5	121900	189
	450	18	18	6.35	0.250	17.5	155500	241
	500	20	20	6.35	0.250	19.5	192900	299
	600	24	24	6.35	0.250	23.5	280000	434
	750	30	30	7.92	0.312	29.4	437400	678
Schedule 20	200	8	8.63	6.35	0.250	8.13	33500	51.9
	250	10	10.8	6.35	0.250	10.3	53200	82.5
	300	12	12.8	6.35	0.250	12.3	76000	117.9
	350	14	14.0	7.92	0.312	13.4	90900	141
	400	16	16.0	7.92	0.312	15.4	120000	186
	450	18	18.0	7.92	0.312	17.4	152900	237
	500	20	20.0	9.53	0.375	19.3	187700	291
	600	24	24.0	9.53	0.375	23.3	274200	425
	750	30	30.0	12.70	0.500	29.0	426400	661
Schedule 30	200	8	8.63	7.04	0.277	8.07	33000	51.2
	250	10	10.8	7.80	0.307	10.1	52000	80.7
	300	12	12.8	8.38	0.330	12.1	74200	115
	350	14	14.0	9.53	0.375	13.3	89000	138
	400	16	16.0	9.53	0.375	15.3	118000	183
	450	18	18.0	11.13	0.438	17.1	148400	230
	500	20	20.0	12.70	0.500	19.0	183200	284
	600	24	24.0	14.27	0.562	22.9	265100	411
	750	30	30.0	15.88	0.625	28.8	418700	649
Schedule 40*	15	½	0.84	2.77	0.109	0.622	190	0.304
	20	¾	1.05	2.87	0.113	0.824	340	0.533
	25	1	1.32	3.38	0.133	1.05	550	0.864
	32	1¼	1.66	3.56	0.140	1.38	970	1.50
	40	1½	1.90	3.68	0.145	1.61	1300	2.04
	50	2	2.38	3.91	0.154	2.07	2150	3.34
	65	2½	2.88	5.16	0.203	2.47	3100	4.79
	80	3	3.50	5.49	0.216	3.07	4700	7.39
	100	4	4.50	6.02	0.237	4.03	8200	12.7
	150	6	6.63	7.11	0.280	6.07	18600	28.9
	200	8	8.63	8.18	0.322	7.98	32200	50.0
	250	10	10.8	9.27	0.365	10.02	50900	78.9
	300	12	12.8	10.31	0.406	11.9	72200	112
	350	14	14.0	11.13	0.438	13.1	87100	135
	400	16	16.0	12.70	0.500	15.0	114200	177
	450	18	18.0	14.27	0.562	16.9	144500	224
	500	20	20.0	15.06	0.593	18.8	179300	278
	600	24	24.0	17.45	0.687	22.6	259300	402

***Standard wall pipe same as Schedule 40 through 10" size. 12" size data follows.**

300	12	12.8	9.53	0.375	12.00	72900	113

Table 6

Commercial Wrought Steel Pipe Data (ANSI B36.10) (continued)

	Nominal Pipe Size		O.D.	Wall Thickness		I.D.	Flow Area	
	mm	inches	inches	mm	inches	inches	mm^2	sq in
Schedule 80*	15	1/2	0.84	3.73	0.147	0.546	150	0.234
	20	3/4	1.05	3.91	0.154	0.742	280	0.433
	25	1	1.32	4.55	0.179	0.957	460	0.719
	32	1 1/4	1.66	4.85	0.191	1.28	820	1.28
	40	1 1/2	1.90	5.08	0.200	1.50	1140	1.77
	50	2	2.38	5.54	0.218	1.94	1900	2.95
	65	2 1/2	2.88	7.01	0.276	2.32	2700	4.24
	80	3	3.50	7.62	0.300	2.90	4200	6.61
	100	4	4.50	8.56	0.337	3.83	7400	11.5
	150	6	6.63	10.97	0.432	5.76	16800	26.1
	200	8	8.63	12.70	0.500	7.63	29500	45.7
	250	10	10.8	15.06	0.593	9.56	46300	71.8
	300	12	12.8	17.45	0.687	11.4	65800	102
	350	14	14.0	19.05	0.750	12.5	79300	123
	400	16	16.0	21.41	0.843	14.3	103800	161
	450	18	18.0	23.80	0.937	16.1	131600	204
	500	20	20.0	26.16	1.03	17.9	163200	253
	600	24	24.0	30.99	1.22	21.6	235400	365
Schedule 160	15	1/2	0.84	4.75	0.187	0.466	110	0.171
	20	3/4	1.05	5.54	0.218	0.614	190	0.296
	25	1	1.32	6.35	0.250	0.815	340	0.522
	32	1 1/4	1.66	6.35	0.250	1.16	680	1.06
	40	1 1/2	1.90	7.14	0.281	1.34	900	1.41
	50	2	2.38	8.71	0.343	1.69	1450	2.24
	65	2 1/2	2.88	9.53	0.375	2.13	2300	3.55
	80	3	3.50	11.13	0.438	2.62	3500	5.41
	100	4	4.50	13.49	0.531	3.44	6000	9.28
	150	6	6.63	18.24	0.718	5.19	13600	21.1
	200	8	8.63	23.01	0.906	6.81	23500	36.5
	250	10	10.8	28.70	1.13	8.50	36600	56.8
	300	12	12.8	33.27	1.31	10.1	51900	80.5
	350	14	14.0	35.81	1.41	11.2	63400	98.3
	400	16	16.0	40.39	1.59	12.8	83200	129
	450	18	18.0	45.21	1.78	14.4	105800	164
	500	20	20.0	50.04	1.97	16.1	130900	203
	600	24	24.0	59.44	2.34	19.3	189000	293
Double Extra Strong	15	1/2	0.84	7.47	0.294	0.252	30	0.050
	20	3/4	1.05	7.82	0.308	0.434	90	0.148
	25	1	1.32	9.09	0.358	0.599	180	0.282
	32	1 1/4	1.66	9.70	0.382	0.896	400	0.630
	40	1 1/2	1.90	10.16	0.400	1.10	610	0.950
	50	2	2.38	11.07	0.436	1.50	1140	1.77
	65	2 1/2	2.89	14.02	0.552	1.77	1600	2.46
	80	3	3.50	15.24	0.600	2.30	2700	4.16
	100	4	4.50	17.12	0.674	3.15	5000	7.80
	150	6	6.63	21.94	0.864	4.90	12100	18.8
	200	8	8.63	22.22	0.875	6.88	23900	37.1

***Extra strong pipe same as Schedule 80 through 8" size. 10" & 12" size data follows.**

250	10	10.8	12.70	0.500	9.75	48200	74.7
300	12	12.8	12.70	0.500	11.8	69700	108

Table 6

Temperature Conversion Table

°C		°F	°C		°F
-273	-459.4		43.3	110	230
-268	-450		46.1	115	239
-240	-400		48.9	120	248
-212	-350		54.4	130	266
-184	-300		60.0	140	284
-157	-250	-418	65.6	150	302
-129	-200	-328	71.1	160	320
-101	-150	-238	76.7	170	338
-73	-100	-148	82.2	180	356
-45.6	-50	-58	87.8	190	374
-42.8	-45	-49	93.3	200	392
-40	-40	-40	98.9	210	410
-37.2	-35	-31	104.4	220	428
-34.4	-30	-22	110	230	446
-31.7	-25	-13	115.6	240	464
-28.9	-20	-4	121	250	482
-26.1	-15	5	149	300	572
-23.2	-10	14	177	350	662
-20.6	-5	23	204	400	752
-17.8	0	32	232	450	842
-15	5	41	260	500	932
-12.2	10	50	288	550	1022
-9.4	15	59	316	600	1112
-6.7	20	68	343	650	1202
-3.9	25	77	371	700	1292
-1.1	30	86	399	750	1382
0	32	89.6	427	800	1472
1.7	35	95	454	850	1562
4.4	40	104	482	900	1652
7.2	45	113	510	950	1742
10	50	122	538	1000	1832
12.8	55	131	566	1050	1922
15.6	60	140	593	1100	2012
18.3	65	149	621	1150	2102
21.1	70	158	649	1200	2192
23.9	75	167	677	1250	2282
26.7	80	176	704	1300	2372
29.4	85	185	732	1350	2462
32.2	90	194	762	1400	2552
35	95	203	788	1450	2642
37.8	100	212	816	1500	2732
40.6	105	221			

Note : The temperature to be converted is the figure in the red column. To obtain a reading in °C use the left column; for conversion to °F use the right column.

Table 7

Metric Conversion Tables

Multiply	By	To Obtain
Length		
millimeters	0.10	centimeters
millimeters	0.001	meters
millimeters	0.039	inches
millimeters	0.00328	feet
centimeters	10.0	millimeters
centimeters	0.010	meters
centimeters	0.394	inches
centimeters	0.0328	feet
inches	25.40	millimeters
inches	2.54	centimeters
inches	0.0254	meters
inches	0.0833	feet
feet	304.8	millimeters
feet	30.48	centimeters
feet	0.304	meters
feet	12.0	inches
Area		
sq. millimeters	0.010	sq. centimeters
sq. millimeters	$10.^{-6}$	sq. meters
sq. millimeters	0.00155	sq. inches
sq. millimeters	1.076 x 10^{-5}	sq. feet
sq. centimeters	100	sq. millimeters
sq. centimeters	0.0001	sq. meters
sq. centimeters	0.155	sq. inches
sq. centimeters	0.001076	sq. feet
sq. inches	645.2	sq. millimeters
sq. inches	6.452	sq. centimeters
sq. inches	0.000645	sq. meters
sq. inches	0.00694	sq. feet
sq. feet	9.29 x 10^4	sqs. millimeters
sq. feet	929	sq. centimeters
sq. feet	0.0929	sq. meters
sq. feet	144	sq. inches
Flow Rates		
gallons US/minute GPM	3.785	liters/min
gallons US/minute	0.133	ft^3/min
gallons US/minute	8.021	ft^3/hr
gallons US/minute	0.227	m^3/hr
gallons US/minute	34.29	Barrels/day (42 US gal)
cubic feet/minute	7.481	GPM
cubic feet/minute	28.32	liters/minute

Multiply	By	To Obtain
Flow Rates		
cubic feet/minute	60.0	ft^3/hr
cubic feet/minute	1.699	m^3/hr
cubic feet/minute	256.5	Barrels/day
cubic feet/hr	0.1247	GPM
cubic feet/hr	0.472	liters/min
cubic feet/hr	0.01667	ft^3/min
cubic feet/hr	0.0283	m^3/hr
cubic meters/hr	4.403	GPM
cubic meters/hr	16.67	liters/min
cubic meters/hr	0.5886	ft^3/min
cubic meters/hr	35.31	ft^3/hr
cubic meters/hr	150.9	Barrels/day
Velocity		
feet per second	60	ft/min
feet per second	0.3048	meters/second
feet per second	1.097	km/hr
feet per second	0.6818	miles/hr
meters per second	3.280	ft/sec
meters per second	196.9	ft/min
meters per second	3.600	km/hr
meters per second	2.237	miles/hr
Weight (Mass)		
pounds	0.0005	short ton
pounds	0.000446	long ton
pounds	0.453	kilogram
pounds	0.000453	metric ton
short ton	2000.0	pounds
short ton	0.8929	long ton
short ton	907.2	kilogram
short ton	0.9072	metric ton
long ton	2240	pounds
long ton	1.120	short ton
long ton	1016	kilogram
long ton	1.016	metric ton
kilogram	2.205	pounds
kilogram	0.0011	short ton
kilogram	0.00098	long ton
kilogram	0.001	metric ton
metric ton	2205	pounds
metric ton	1.102	short ton
metric ton	0.984	long ton
metric ton	1000	kilogram

Some units shown on this page are not recommended by SI, e.g., kilogram/sq. cm should be read as kilogram (force) / sq. cm

Table 8

Metric Conversion Tables (continued)

Multiply	By	To Obtain
	Volume & Capacity	
cubic cm	0.06102	cubic inches
cubic cm	3.531 x 10^{-5}	cubic feet
cubic cm	10.$^{-6}$	cubic meters
cubic cm	0.0001	liters
cubic cm	2.642 x 10^{-4}	gallons (US)
cubic meters	10.^{6}	cubic cm
cubic meters	61,023.0	cubic inches
cubic meters	35.31	cubic feet
cubic meters	1000.0	liters
cubic meters	264.2	gallons
cubic feet	28,320.0	cubic cm
cubic feet	1728.0	cubic inches
cubic feet	0.0283	cubic meters
cubic feet	28.32	liters
cubic feet	7.4805	gallons
liters	1000.0	cubic cm
liters	61.02	cubic inches
liters	0.03531	cubic feet
liters	0.001	cubic meters
liters	0.264	gallons
gallons	3785.0	cubic cm
gallons	231.0	cubic inches
gallons	0.1337	cubic feet
gallons	3.785 x 10^{-3}	cubic meters
gallons	3.785	liters
	Pressure & Head	
pounds/sq. inch	0.06895	bar
pounds/sq. inch	0.06804	atmosphere
pounds/sq. inch	0.0703	kg/cm^2
pounds/sq. inch	6.895	kPa
pounds/sq. inch	2.307	ft of H_2O (4 DEG C)
pounds/sq. inch	0.703	m of H_2O (4 DEG C)
pounds/sq. inch	5.171	cm of Hg (0 DEG C)
pounds/sq. inch	51.71	torr (mm of Hg) (0 DEG C)
pounds/sq. inch	2.036	in of Hg (0 DEG C)

Multiply	By	To Obtain
	Pressure & Head	
atmosphere	14.69	psi
atmosphere	1.013	bar
atmosphere	1.033	Kg/cm^2
atmosphere	101.3	kPa
atmosphere	33.9	ft of H_2O
atmosphere	10.33	m of H_2O
atmosphere	76.00	cm of Hg
atmosphere	760.0	torr (mm of Hg)
atmosphere	29.92	in of Hg
bar	14.50	psi
bar	0.9869	atmosphere
bar	1.020	Kg/cm^2
bar	100.0	kPa
bar	33.45	ft of H_2O
bar	10.20	m of H_2O
bar	75.01	cm of Hg
bar	750.1	torr (mm of Hg)
bar	29.53	in of Hg
kilogram/sq. cm	14.22	psi
kilogram/sq. cm	0.9807	bar
kilogram/sq. cm	0.9678	atmosphere
kilogram/sq. cm	98.07	kPa
kilogram/sq. cm	32.81	ft of H_2O (4 DEG C)
kilogram/sq. cm	10.00	m of H_2O (4 DEG C)
kilogram/sq. cm	73.56	cm of Hg
kilogram/sq. cm	735.6	torr (mm of Hg)
kilogram/sq. cm	28.96	in of Hg
kiloPascal	0.145	psi
kiloPascal	0.01	bar
kiloPascal	0.00986	atmosphere
kiloPascal	0.0102	kg/cm^2
kiloPascal	0.334	ft of H_2O
kiloPascal	0.102	m of H_2O
kiloPascal	0.7501	cm of Hg
kiloPascal	7.501	torr (mm of Hg)
kiloPascal	0.295	in of Hg
millibar	0.001	bar

Some units shown on this page are not recommended by SI, e.g., kilogram/sq. cm should be read as kilogram (force) /sq. cm

Table 8

Useful List of Equivalents (U. S. Customary Units)

1 U.S. gallon of water = 8.33 lbs @ std cond.
1 cubic foot of water = 62.34 lbs @ std cond. (= density)
1 cubic foot of water = 7.48 gallons
1 cubic foot of air = 0.076 lbs @ std cond. (= air density)
Air specific volume = 1/density = 13.1 cubic feet /lb
Air molecular weight M = 29
Specific gravity of air G = 1 (reference for gases)
Specific gravity of water = 1 (reference for liquids)
Standard conditions (US Customary) are at
14.69 psia & 60 DEG F*
G of any gas = density of gas/0.076
G of any gas = molecular wt of gas/29

G of gas at flowing temp $= \frac{G \times 520}{T + 460}$

Flow conversion of gas

$$\text{scfh} = \frac{\text{lbs/hr}}{\text{density}}$$

$$\text{scfh} = \frac{\text{lbs/hr} \times 379}{M}$$

$$\text{scfh} = \frac{\text{lbs/hr} \times 13.1}{G}$$

Flow conversion of liquid

$$\text{GPM} = \frac{\text{lbs/hr}}{500 \times G}$$

*Normal conditions (metric) are at 1.013 bar and 0 DEG. C & 4 DEG. C water

Note : Within this control valve handbook, the metric factors are at 1.013 bar and 15.6˚C.

Universal gas equation

		Metric
$P_v = mRTZ$ Where	P = press lbs/sq ft	P = Pascal
	v = volume in ft^3	v = m^3
	m = mass in lbs	m = kg
	R = gas constant	R = gas constant
	$= \frac{1545}{M}$	$= \frac{8314}{M}$
	T = temp Rankine	T = temp Kelvin
	Z = gas compressibility factor = Z	

Gas expansion $\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$

(perfect gas)

Velocity of sound C (ft/sec)

$$C = 223 \sqrt{\frac{k\ (T + 460)}{M}}$$

where T = temp DEG F
M = mol. wt
k = specific heat ratio C_p/C_v

Velocity of Sound C (m/sec)

$$C = 91.2 \sqrt{\frac{k\ (T + 273)}{M}}$$

where T = temp DEG C
M = mol. wt
k = specific heat ratio C_p/C_v

References

1. "The Introduction of a Critical Flow Factor for Valve Sizing," H. D. Baumann, ISA Transactions, Vol. 2, No. 2, April 1963
2. "Sizing Control Valves for Flashing Service," H. W. Boger, Instruments and Control Systems, January 1970
3. "Recent Trends in Sizing Control Valves," H. W. Boger, Proceedings Texas A&M 23rd Annual Symposium on Instrumentation for the Process Industries, 1968
4. "Effect of Pipe Reducers on Valve Capacity," H. D. Baumann, Instruments and Control Systems, December 1968
5. "Flow of a Flashing Mixture of Water and Steam through Pipes and Valves," W. F. Allen, Trans. ASME, Vol. 73, 1951
6. "Flow Characteristics for Control Valve Installations," H. W. Boger, ISA Journal, October 1966.
7. Flowmeter Computation Handbook, ASME, 1961
8. ANSI/ISA-75.01.01, Flow Equations for Sizing Control Valves
9. IEC 60534-2-1, Sizing Equations for Fluid Flow Under Installed Conditions
10. ISA-RP75.23-1995, Considerations for Evaluating Control Valve Cavitation
11. "Avoiding Control Valve Application Problems with Physics-Based Models," K. W. Roth and J. A. Stares
12. Masoneilan Noise Control Manual OZ3000

Notes

Notes

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