FLUID FLOW BASICS OF THROTTLING VALVES

"We simply make it right."

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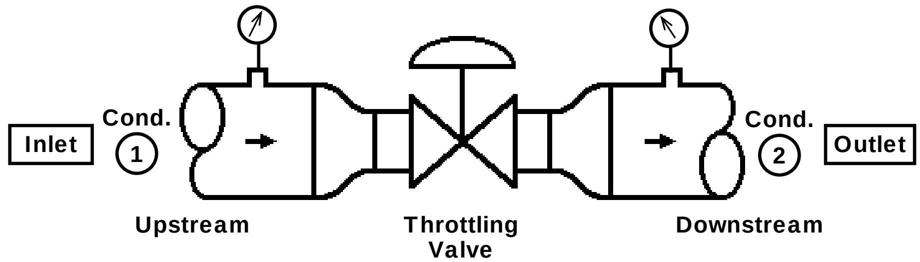
FLUID PARAMETERS –

The following fluid parameters are frequently associated with throttling valves –



$\Delta P_{Size} = P_1 - P_2$

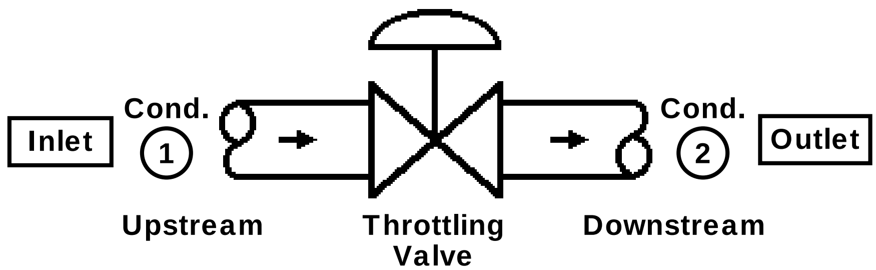
$\Delta T = T_1 - T_2$

	CONDITION 1		CONDITION 2
Flow Quantity	$\dot{m}_1$	Mass Flow Rate	$\dot{m}_2$
	$\dot{Q}_1, \dot{V}_1$	Volumetric Flow Rate	$\dot{Q}_2, \dot{V}_2$
Pressures	$P_1, P_{S1}, P_{1(Abs)}$	Static Pressure	$P_2, P_{S2}, P_{2\,(Abs)}$
	P_{V1}	Velocity Pressure	P_{V2}
		P_{VC} – Pressure @ Vena Contracta	
		ΔP_{size} – Sizing Pressure Drop	
	P_{VP1}, P_{SAT1}	Vapor Pressure / Saturation Pressure	P_{VP2}, P_{SAT2}
Thermo-dynamic	$T_1, T_{1(Abs)}, T_{1@xx^\circ SH}$	Temperature	$T_2, T_{2(Abs)}, T_{2@xx^\circ SH}$
	T_{SAT1}	Saturation Temperature	T_{SAT2}
	H_1, h_1	Enthalpy	H_2, h_2
	$\mathcal{V}_1$	Specific Volume	$\mathcal{V}_2$
Relative Weight/Mass	ρ_1	Liquid Density	ρ_2
	γ_1	Actual Gas Density	γ_2
	SG_1	Specific Gravity	SG_2
Geometric	$Ø_{1P}$	Pipe Diameter	$Ø_{2P}$
	$Ø_{1V}$	Valve Body Size	$Ø_{2V}$
	Z_1	Elevation Head	Z_2
	A_1	Cross-Section Area	A_2
	$\bar{v}_{1V}$	Avg. Valve Velocity	$\bar{v}_{2V}$
	$\bar{v}_{1P}$	Avg. Pipe Velocity	$\bar{v}_{2P}$
Fluid "Resistance to Flow"	μ_1	Absolute Viscosity	μ_2
	ν_1	Kinematic Viscosity	ν_2

THE BASICS OF THROTTLING VALVES

THROTTLING VALVES

Valves that are utilized as fluid control devices are typically “throttling valves”.



$\Delta P = P_1 - P_2$

Such valves experience <u>internal velocity and internal pressure gradients</u> (both positive and negative) that conclude with a <u>permanent pressure loss</u> (ΔP) from the inlet pipe-to-outlet pipe connections. Throttling valve trim (plug-seat) experiences relatively <u>high internal velocities</u> nearly 100% of operating time. In comparison, ON-OFF automated or manual valves experience velocity changes <u>ONLY</u> when being actuated from “open-to-closed”, or vice versa; i.e. a few seconds or minutes.

Bernoulli’s Theorem is the most useful tool in analyzing what is going on physically within the walls of a throttling valve, which includes —

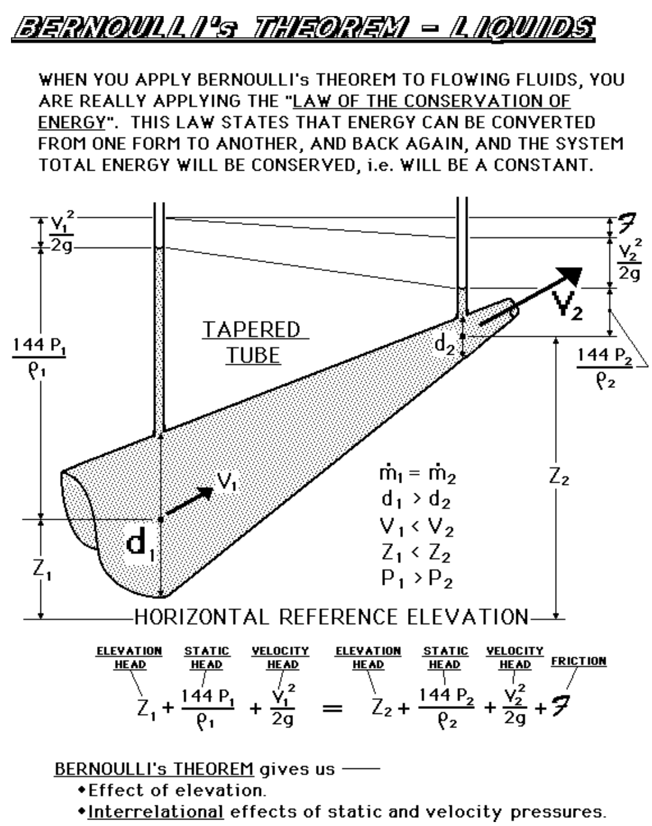
- velocity gradients
- pressure gradients

The other important tool is the 1st Law of Thermodynamics which allows analyzing —

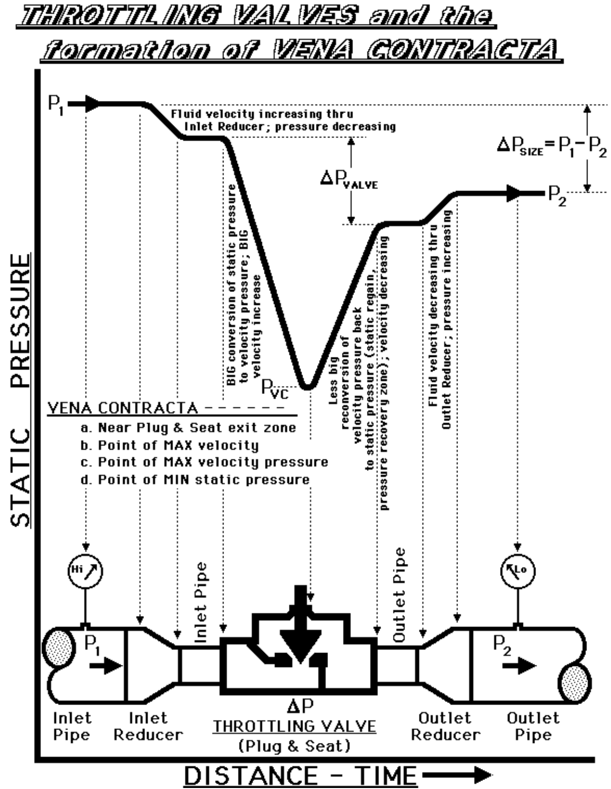
- fluid state
- thermal effects

Bernoulli’s principles apply to the following for throttling valves —

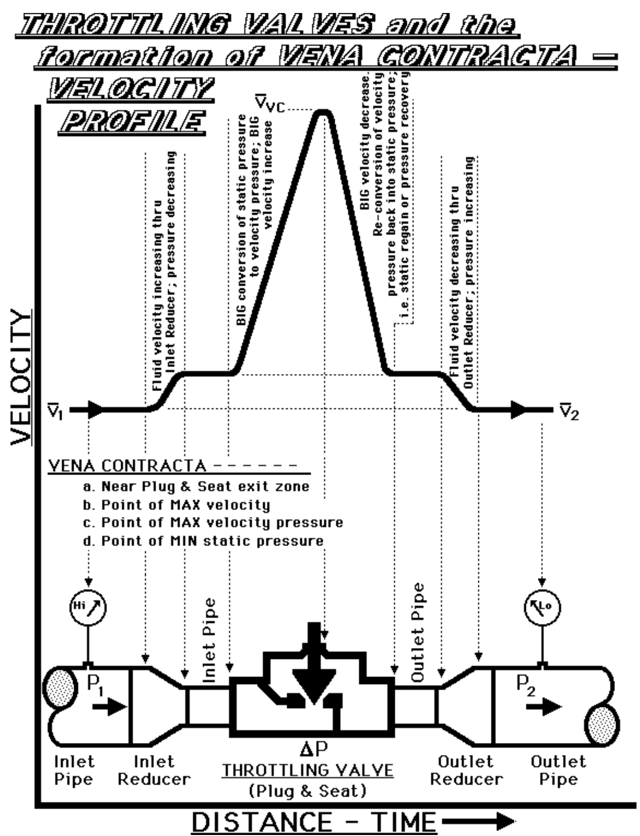
- inlet pipe reducer
- pressure drop to main orifice
- pressure recovery to outlet
- outlet pipe reducer



When the pressure gradients are graphically shown, one ends up with the rather typical "vena contracta" curve —

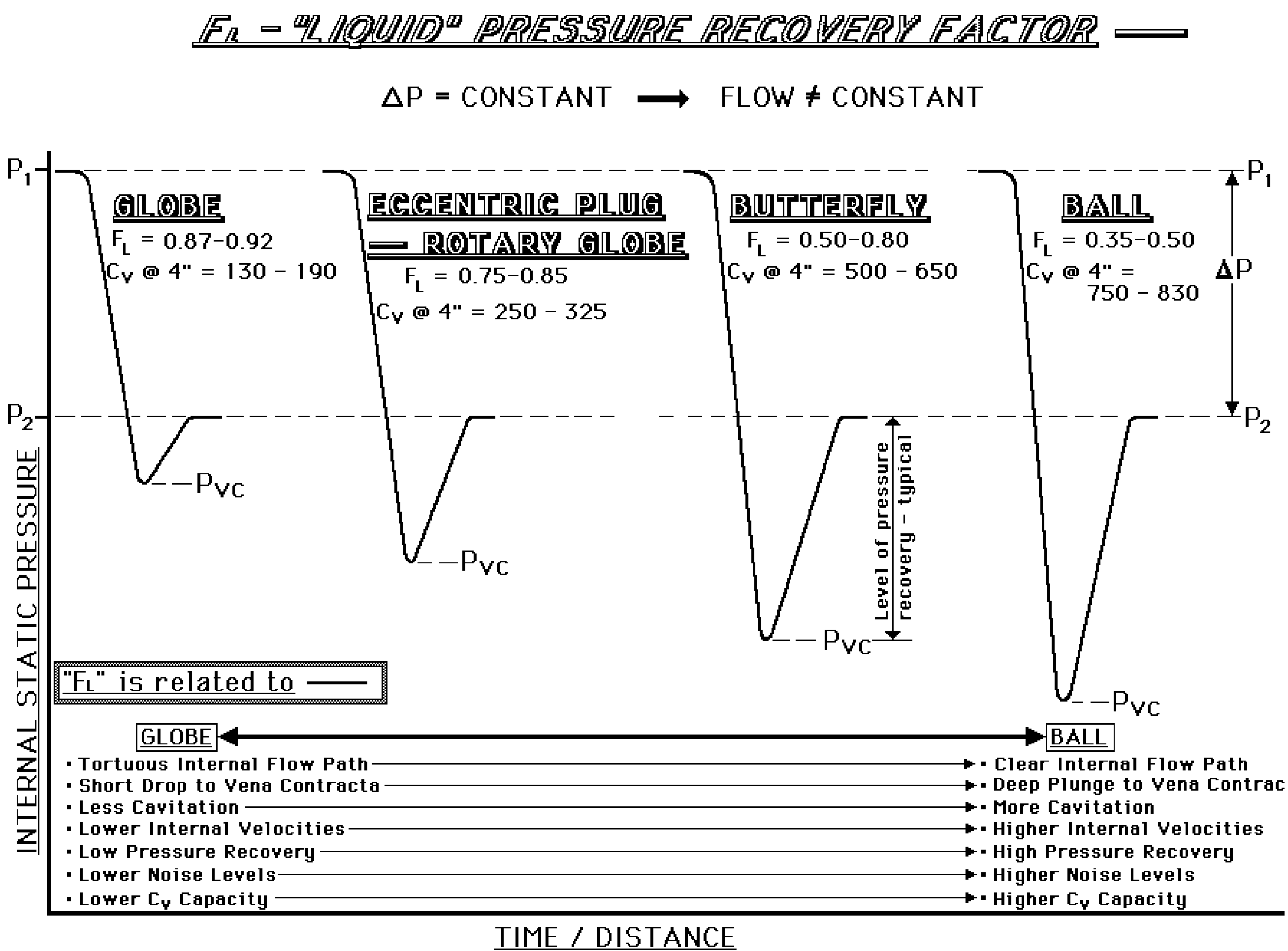


The velocity gradients form a sort of "inverse" of the vena contracta curve —



The depth of the vena contracta "dip" is primarily a function of a throttling valve's geometry; globe vs. butterfly, etc. The important parameter in determining the P_{VC} is – "F_L – Liquid Pressure Recovery Factor". As the name implies, the F_L factor is a measure of the effectiveness of the reconversion of velocity pressure into static pressure from the main orifice of the throttling valve (@ vena contracta) to the valve's outlet.

The following graphic attempts to give relative representation of the four major valve styles used for throttling service.



Both butterfly and ball valves are sub-classified as "high recovery valves". As a general rule, globe and eccentric plug (rotary globe) styles tend to make "better" throttling control valves.

FLUID STATES

Fluid flow is classified into two basic fluid states at the inlet.

LIQUID (Non-Compressible)	GAS–VAPOR (Compressible)

As pressure changes occur within a throttling valve, it is possible to produce 2-phase flow at the valve's outlet for either a liquid or gas-vapor at the inlet.

A "vapor" is a "gas" that is at, or relatively near, its "saturation" (boiling) conditions of pressure and temperature; i.e. saturated vapor or slightly superheated vapor. A "gas" is a fluid that does not liquify at reduced temperatures, or is a highly superheated vapor.

Throttling valves operate as "steady state, steady flow" devices. The entering and exiting mass flow rates are the same; i.e. flow is "continuous", and the Continuity Equation is applicable —

$$\dot{m}_1 = \dot{m}_2$$

(EQ. #1) $$\boxed{\rho_1 A_1 \bar{v}_1 = \rho_2 A_2 \bar{v}_2} \quad \text{(no phase change)}$$

$$\rho_1 A_1 \bar{v}_1 = \underbrace{\rho_{2V} A_{2V} \bar{v}_{2V}}_{\text{Vapor}} + \underbrace{\rho_{2L} A_{2L} \bar{v}_{2L}}_{\text{Liquid}} \quad \text{(2-phase outlet)}$$

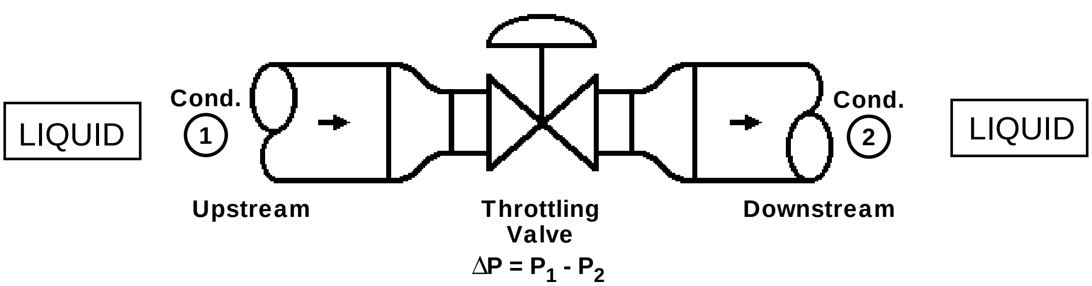
It is a thermodynamic principle that whenever there is a phase change between a throttling valve's entering and exiting fluid state, there is also a temperature change (i.e. decrease or cooling) in all such applications —

$$T_1 > T_2$$

LIQUIDS. For simple "liquid-in and liquid-out" flow there is no density change of the liquid —

$$\rho_1 = \rho_2$$

This constant density results in other parameters being typically affected —



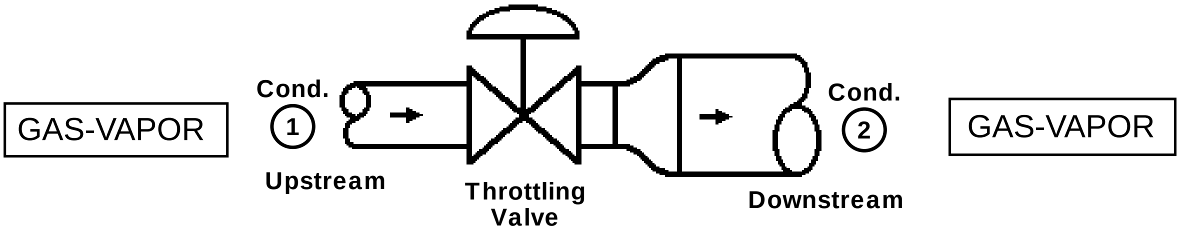
$$\dot{m}_1 = \dot{m}_2 \qquad A_1 = A_2 \qquad T_1 = T_2 \text{ — no phase change}$$

$$\bar{v}_1 = \bar{v}_2 \qquad \varnothing_1 = \varnothing_2 \qquad \rho_1 = \rho_2$$

GAS-VAPORS. For simple “gas-vapor-in and gas-vapor-out” flow, there is a density change (i.e. decrease) of the gas-vapor as the fluid decompresses (i.e. expands) —

$$\rho_1 > \rho_2$$

This changing density results in other parameters being typically affected —

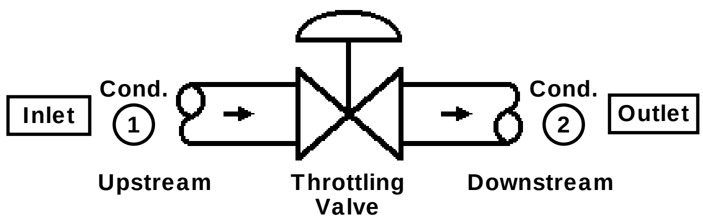


$\Delta P = P_1 - P_2$

$\dot{m}_1 = \dot{m}_2$	$A_1 < A_2$	$T_1 \approx T_2 \longrightarrow$	GAS or HIGHLY SUPERHEATED VAPOR
$\bar{v}_1 > \bar{v}_2$	$\varnothing_1 < \varnothing_2$ $\rho_1 > \rho_2$	$T_1 > T_2 \longrightarrow$	VAPOR

THERMODYNAMIC PRINCIPLES

THROTTLING PROCESS. In looking into the thermodynamic principles of a “throttling process”, we know —



$\Delta h = h_1 - h_2$
$\Delta h = 0$

THE **CHANGE IN ENTHALPY** ACROSS A RESTRICTION IN A PIPE — ORIFICE, REGULATOR, CONTROL VALVE — IS **“ZERO”** FOR A **THROTTLING PROCESS**.

By the continuity equation —

$$\dot{m}_1 = \dot{m}_2$$
$$\dot{m}_1 h_1 - \dot{m}_2 h_2 = 0 \quad \textbf{OR}$$

(EQ. #2) $$\boxed{\dot{m}(h_1 - h_2) = 0}$$

Parameter	English Units	Metric Units
h_1 = Valve Inlet Enthalpy	Btu/#	kJ/kg
h_2 = Valve Outlet Enthalpy	Btu/#	kJ/kg
$\dot{m}_1$ = Inlet Mass Flow	#/Hr	kg/Hr
$\dot{m}_2$ = Outlet Mass Flow	#/Hr	Kg/Hr

It is the use of fluid thermodynamic data and the thermodynamic principles of the "constant enthalpy throttling process" that throttling valves experience which allows an accurate determination of a fluid's state while internal to the valve as well as at the valve's outlet. In particular, we want to know what the fluid is physically doing at the throttling valve's main orifice (plug-seat); i.e. what is occurring at the vena contracta and elsewhere within the valve?

SATURATION STATE. A fluid is said to be "saturated" when —

- Liquid - when at the boiling temperature – T_{sat} – for a given pressure – P_{sat}

  Examples: Water @ P_{sat} = 14.7 psia $\longrightarrow$ T_{sat} = 212°F
  P_{sat} = 1.0135 BarA $\longrightarrow$ T_{sat} = 100°C

  Water @ P_{sat} = 145 psig $\longrightarrow$ T_{sat} = 355.8°F
  P_{sat} = 10 BarA $\longrightarrow$ T_{sat} = 179.9°C

- Vapor - when at the boiling temperature – T_{sat} – for a given pressure – P_{sat}

  Examples: Steam @ P_{sat} = 14.7 psia $\longrightarrow$ T_{sat} = 212°F
  P_{sat} = 1.0135 BarA $\longrightarrow$ T_{sat} = 100°C

  Steam @ P_{sat} = 145 psig $\longrightarrow$ T_{sat} = 355.8°F
  P_{sat} = 10 BarA $\longrightarrow$ T_{sat} = 179.9°C

Restating the above examples, we have both saturated liquid water (condensate) and saturated steam at the same P_{sat} and T_{sat}. Further, for any given fluid in its "saturation" state, when we have its pressure (P_{sat}), we KNOW its temperature (T_{sat}). To say a fluid is "saturated" is to give a property of the fluid. Only two extensive properties of a fluid will locate the fluid in the physical universe. We know exactly where a fluid is when we say the fluid is —

- saturated water at P_{sat} = 29 psia = 2.0 BarA, we know that T_{sat} = 248.4°F = 120.1°C.
- saturated steam at T_{sat} = 212°F = 100°C, we know that P_{sat} = 14.7 psig = 1.013 BarA.

SUPERHEATED VAPOR. A fluid is a superheated vapor when its temperature is greater than T_{sat} corresponding to the flowing pressure.

Examples: Steam @ P_1 = 145 psia & T_1 = 425°F
(T_{sat} = 355.8°F)
69.2°F SH

P_1 = 10 BarA & T_1 = 219°C
(T_{sat} = 179.9°C)
39.1°C SH

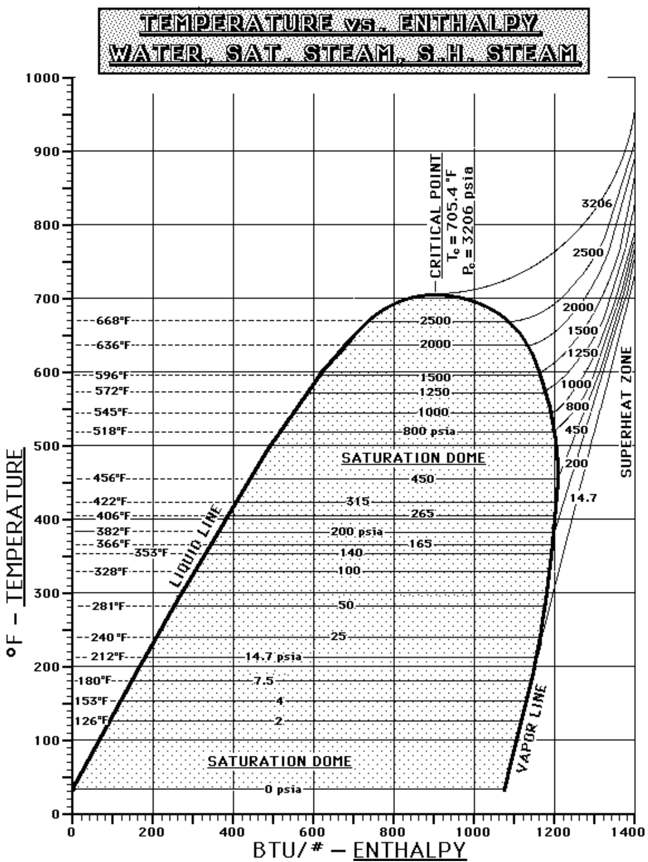
To say a vapor is “superheated” does NOT give an extensive property of the fluid; so, a second property must also be known to physically locate a superheated vapor in the universe.

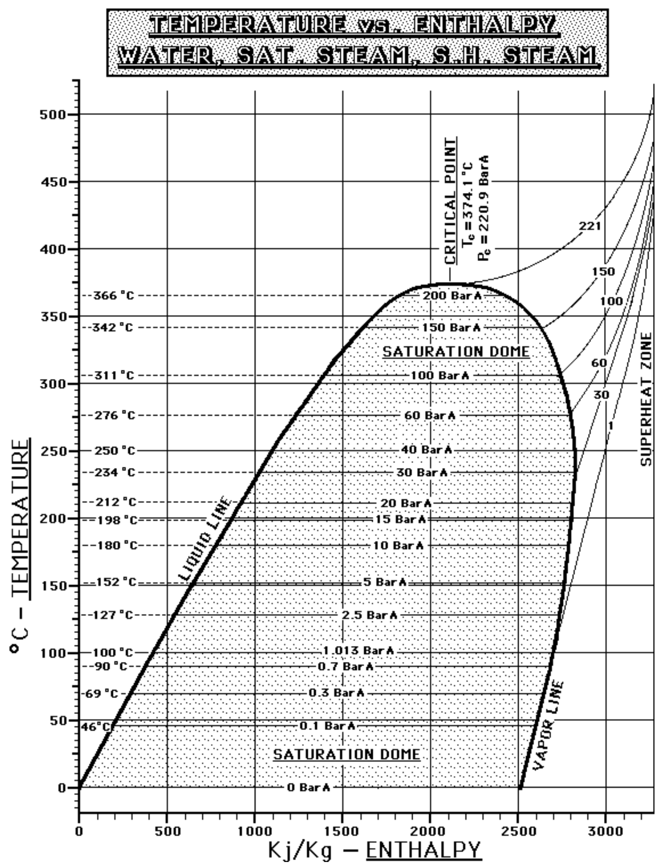
SUB-COOLED LIQUID. A liquid is sub-cooled when its temperature is less than T_{sat} corresponding to the flowing pressure.

Example: Water @ P = 145 psia & T = 60°F (T_{sat} = 355.8°F)
P = 10 BarA & T = 15.5°C (T_{sat} = 179.9°C)

To say a liquid is “sub-cooled” is NOT to give an extensive property of the fluid; so, a second property must also be known to physically locate a sub-cooled liquid in the universe.

THUMB CURVE – T vs. H GRAPH. The following graph is plotted using thermodynamic data for steam condensate; i.e. straight out of the “Steam Tables”.





These graphs are NOT useful for sub-cooled liquids; they are ONLY useful for analyzing boiling (vaporizing) liquid/vapors and superheated vapors.

The “critical” properties of “Critical Pressure - P_c” and “Critical Temperature - T_c” are located at the “peak” of the T vs. H Curve.

The important consideration for throttling processes is —

$$\Delta H = 0$$

This means that from Condition 1-to-Condition 2 for throttling valves, you move <u>downwards along a vertical line, as the enthalpy does not vary</u>.

<u>Examples</u>:

	Condition	
<u>Hot Condensate</u> P_1 = 150 psig ≈ 165 psia T_1 = saturated P_2 = 35 psig ≈ 50 psia	1	<u>Hot Condensate</u> P_1 = 9 Barg ≈ 10 BarA T_1 = saturated P_2 = 1.5 Barg ≈ 2.5 BarA
$\left.\begin{matrix} T_1 \approx 366°F \\ T_2 \approx 290°F \end{matrix}\right\}$ ΔT = -76°F $Desc_2$ = "Flashing"–2-phase	2	$\left.\begin{matrix} T_1 \approx 180°C \\ T_2 \approx 127°C \end{matrix}\right\}$ ΔT = -53°C $Desc_2$ = "Flashing"–2-phase
<u>Steam</u> P_1 = 250 psig ≈ 265 psia T_1 = Saturated P_2 = Atm = 14.7 psia	1	<u>Steam</u> P_1 = 14 Barg ≈ 15 BarA T_1 = Saturated P_2 = Atm = 1.013 BarA
$\left.\begin{matrix} T_1 \approx 406°F \\ T_2 \approx 310°F \end{matrix}\right\}$ ΔT = -96°F $Desc_2$ = Superheated Steam	2	$\left.\begin{matrix} T_1 \approx 198°C \\ T_2 \approx 150°C \end{matrix}\right\}$ ΔT = -48°C $Desc_2$ = Superheated Steam
<u>Steam</u> P_1 = 435 psig ≈ 450 psia T_1 = 770°F P_2 = 185 psig ≈ 200 psia	1	<u>Steam</u> P_1 = 29 Barg ≈ 30 BarA T_1 = 410°C P_2 = 14 Barg ≈ 15 BarA
T_{sat} @ P_1 = 456°F $Desc_1$ = Superheated Steam T_2 ≈ 755°F ⟶ ΔT ≈ -15°F $Desc_2$ = Superheated Steam	2	T_{sat} @ P_1 = 234°C $Desc_1$ = Superheated Steam T_2 ≈ 400°C ⟶ ΔT ≈ -10°C $Desc_2$ = Superheated Steam

TEMPERATURE. By reviewing the T vs. H Graphs on pg. 9, we can make a few temperature generalizations for saturated steam —

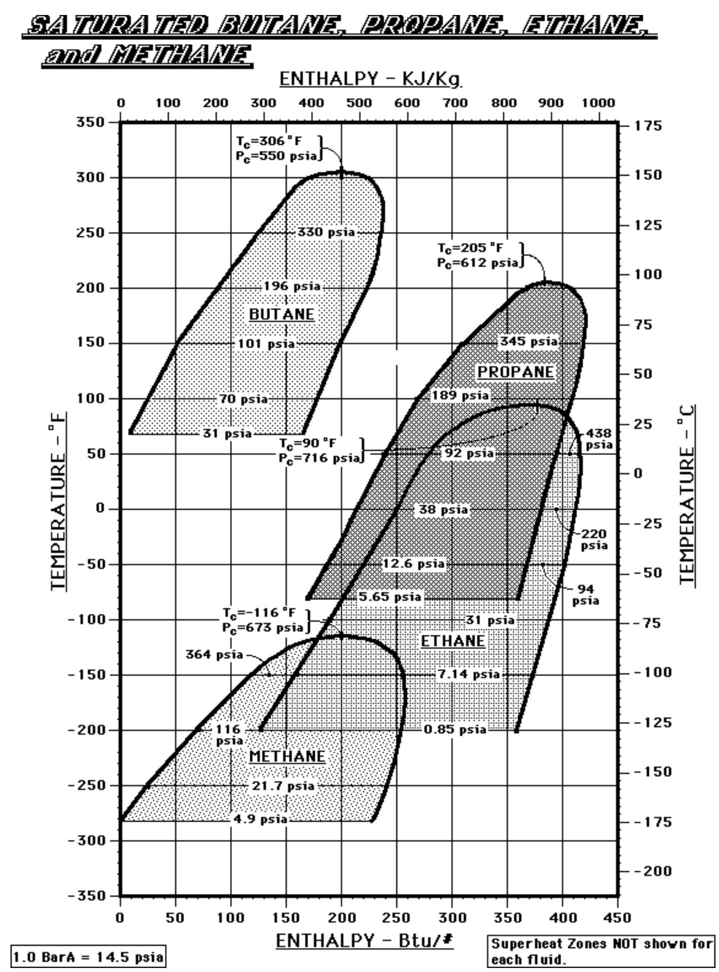
- A throttling ΔT (i.e. $T_1 - T_2$) is always present with a throttling ΔP (i.e. $P_1 - P_2$).
- Greater ΔT's occur for throttling ΔP's within the saturation dome.
- A ΔT is always associated with a throttling ΔP that causes phase change.
- Highly superheated vapor has a relatively small ΔT for a throttling ΔP.
- Slightly superheated vapor has a higher ΔT for a throttling ΔP than a highly superheated vapor.
- Vapors carry higher "heat contents".

This cooling effect due to throttling is frequently referred to as "Joule-Thompson cooling".

When a liquid is sub-cooled (below T_{sat}) both in and out of a throttling valve —

- There is no ΔT (i.e. $T_1 = T_2$) for a throttling ΔP.

BASIC PRINCIPLES. If one learns the thermodynamic principles of water-steam, then the same principles can be applied to many other fluids, even those that typically exist as gas-vapor at ambient pressure and temperature conditions. A basic understanding of these principles will help in understanding the process industry, because fluid separation by differing boiling points is a common occurrence in the Chemical Process Industry. Notice the similarity of the saturation domes of butane, propane, ethane, and methane plotted together at right; there is a striking resemblance to the earlier water-steam H vs. T plot. As very few processes operate at ambient pressures, one must be aware of the "pressure" conditions as well as the "temperature" conditions TOGETHER; i.e. T_{sat} & P_{sat}. Air separation plants and crude oil distillation processes are examples of application of these principles.

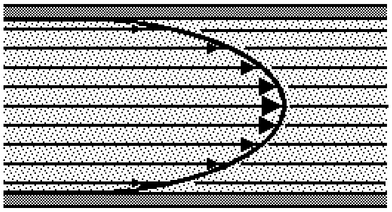


For throttling valves, an understanding of these principles will help in understanding both "Cavitation" and "Flashing" in the liquid flow realm, and will also help with understanding refrigeration and cryogenic applications.

LIQUID FLOW REALM

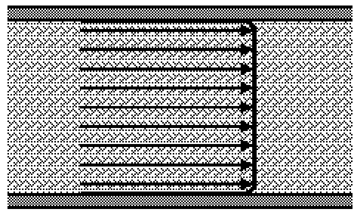
FLOW DESCRIPTION. Liquid flow is non-compressible flow. The first consideration in liquid throttling service is whether the fluid is acting in the "turbulent" or "non-turbulent" realm. "Non-turbulent" can be sub-categorized as "laminar" or "transitional".

The fluid parameter that is most predictive of non-turbulent flow is the fluid's viscosity; a second predictive parameter is very low velocity, which can occur even for relatively non-viscous fluids.



LAMINAR FLOW

1. $\overline{V}_{L-AVG}$ < 1 ft/sec (0.3 M/sec)
2. V_L is NOT constant across the pipe's cross-section; a "velocity profile" is formed, where at the pipe wall, $V_L \approx 0$ ft/sec.
3. Flow "streamlines" are formed.



TURBULENT FLOW

1. $\overline{V}_T$ > 2 ft/sec (0.6 M/sec)
2. V_T is nearly constant; no velocity profile is formed. Pipe wall velocity is only slightly less than V_T.
3. No flow streamlines are formed.

REYNOLDS NUMBER. A calculated dimensionless number value — the Reynold's No. – N_{Re} — is used in both pipes and throttling valves to categorize the differing realms —

Pipe Reynold's No. – N_{Rep}

(EQ #3)

$$\boxed{N_{Rep} = \frac{D \bullet \overline{V} \bullet \rho}{N_o \bullet \mu}}$$

Parameter	English Units	Metric Units
D = Pipe Internal Diameter	ft	M
$\overline{V}$ = Average Velocity	ft/sec	M/sec
ρ = Density	$\#/ft^3$	kg/M^3
μ = Absolute Viscosity	—	cP (gm/cm•sec)
μ_e = Absolute Viscosity	cP (#/ft•sec)	—
N_o = Units Correlation Constant	0.0672 dimensionless	1.00 dimensionless

Determination of Pipe Turbulence Realm

$N_{Rep} < 2100 \longrightarrow$ "Laminar Flow"

$2100 < N_{Rep} < 10{,}000 \longrightarrow$ "Transitional Flow"

$N_{Rep} > 10{,}000 \longrightarrow$ "Turbulent Flow"

Valve Reynold's No. - N_{Rev}

$$N_{Rev} = \frac{N_4 \bullet F_d \bullet \dot{Q}}{\nu \sqrt{F_L \bullet C_v}} \left[\frac{F_L{}^2 \bullet C_v{}^2}{N_2 \bullet d^4} + 1 \right]^{0.25}$$

(EQ #4)

$$\boxed{N_{Rev} \approx \frac{N_4 \bullet F_d \bullet \dot{Q} \bullet SG}{N_o \bullet \mu \sqrt{F_L \bullet C_v}}}$$

where: $\mu = \nu \bullet SG$

$$\left[\frac{F_L{}^2 \bullet C_v{}^2}{N_2 \bullet d^4} + 1 \right]^{0.25} \approx 1$$

Parameter			English Units	Metric Units
F_d	=	Valve Style Modifier	dimensionless	dimensionless
$\dot{Q}$	=	Volumetric Flow Rate	US GPM	M^3/Hr
μ	=	Absolute Viscosity	—	cP (gm/cm•sec)
μ_e	=	Absolute Viscosity	cP (#/ft•sec)	—
F_L	=	Liquid Pressure Recovery Factor	dimensionless	dimensionless
C_v	=	Valve Sizing Coefficient	dimensionless	dimensionless
ν	=	Kinematic Viscosity	#/ft•sec	Cst (gm/cm•sec)
N_4	=	Units Correlation Constant	17,300 dimensionless	76,000 dimensionless
N_o	=	Units Correlation Constant	0.0672 dimensionless	1.00 dimensionless

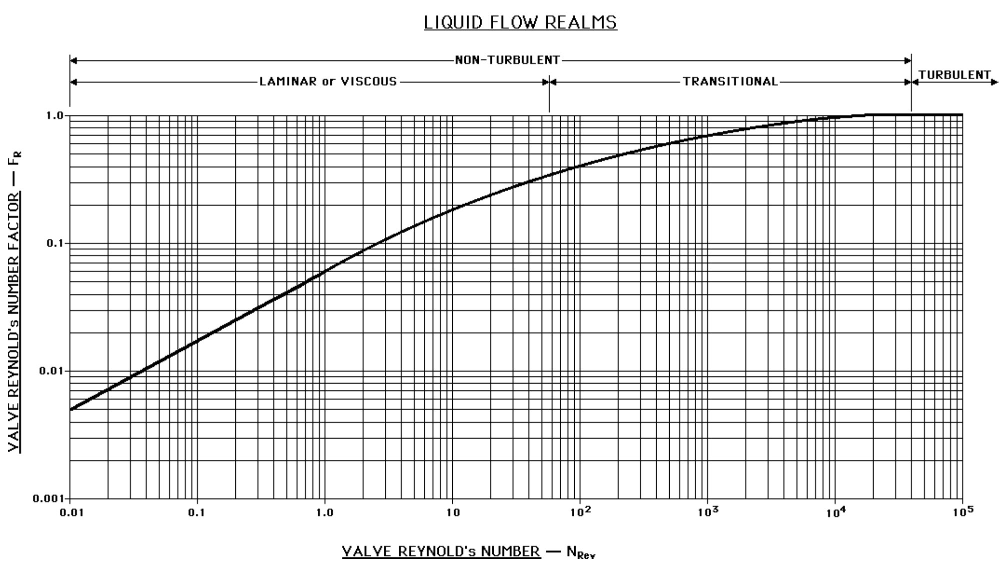
Determination of Valve Turbulence Realm

$N_{Rev} < 56$ ⟶ "Laminar Flow"

$56 < N_{Rev} < 40,000$ ⟶ "Transitional Flow"

$N_{Rev} > 40,000$ ⟶ "Turbulent Flow"

The vast number — greater than 98% — of applications of throttling valves are in the "turbulent" realm.



TURBULENT LIQUID FLOW – Non Vaporizing

“Turbulent” liquid flow is the most common type of liquid flow. It is in this realm’s context that the English “Valve Sizing Coefficient – C_V” originated. By definition —

(EQ #5)

$$\boxed{C_V = \dot{Q}\sqrt{\frac{SG}{\Delta P}}}$$

Where:

$$\left.\begin{array}{l} \text{fluid = water} \\ \Delta P = P_1 - P_2 = 1.00 \text{ psid} \\ \dot{Q} = 1.00 \text{ US GPM} \\ T = 60°F \\ SG = 1.00 \end{array}\right\} \quad C_V = 1.00$$

There is a “Metric Valve Sizing Coefficient - k_V” defined as —

(EQ #6)

$$\boxed{k_V = \dot{Q}\sqrt{\frac{SG}{\Delta P}}}$$

Where:

$$\left.\begin{array}{l} \text{fluid = water} \\ \Delta P = P_1 - P_2 = 1.00 \text{ BarD} \\ \dot{Q} = 1.00 \text{ M}^3\text{/Hr} \\ T = 4°C \\ SG_N = 1.000 \end{array}\right\} \quad k_V = 1.00$$

By correlating the units with conversion factors —

$$k_v = 1.00 \quad \left\{ \begin{array}{lcl} \dot{Q} = 1\ NM^3/Hr & = & 4.414\ \text{US GPM} \\ \Delta P = 1\ \text{Barg} & = & 14.5\ \text{psid} \\ T = 4°C & = & 38.6°F \\ SG_N = 1.00 & = & 1.0013 \end{array} \right.$$

Substituting into EQ. #5 —

$$C_v = 4.414 \sqrt{\frac{1.0013}{14.5}}$$

$$C_v = 1.16$$

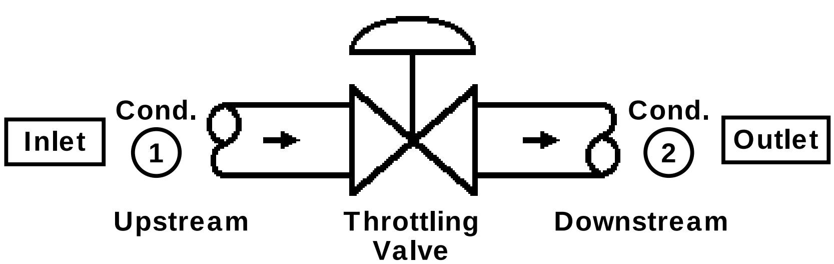
(EQ #7) $\therefore$ $\boxed{\mathbf{k_v = 1.16\ C_v}}$

or

$\boxed{\mathbf{C_v = 0.865\ k_v}}$

As the "C_v" coefficient is more universally utilized, this book will remain with "C_V" hereafter.

By using the correlation factor of EQ #7 (1/1.16 = 0.865), a single equation can be used for both English and Metric units —



$\Delta P = P_1 - P_2$

(EQ #8) $$\dot{Q} = N_1 \bullet C_v \sqrt{\frac{(P_1 - P_2)}{SG}}$$

Parameter	English Units	Metric Units
C_v = English Valve Sizing Coefficient	dimensionless	dimensionless
$\dot{Q}$ = Volumetric Flow Rate	US GPM	M^3/Hr
SG = Specific Gravity	dimensionless	dimensionless
P_1 = Valve Upstream Pressure	psig, psia	Barg, BarA
P_2 = Valve Downstream Pressure	psig, psia	Barg, BarA
N_1 = Units Correlation Constant	1.00 dimensionless	0.865 dimensionless

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As N_1, C_v, and SG are each a "constant", the "liquid sizing equation" (EQ. #7) can be represented as —

$$\dot{Q} = K\sqrt{\Delta P}$$ Where: K = combined constant

$$\dot{Q} \propto \sqrt{\Delta P}$$

Thus, for non-vaporizing liquid flow, <u>FLOW IS PROPORTIONAL TO THE SQUARE ROOT OF PRESSURE DROP</u>.

(EQ #8)

$$\dot{Q} = N_1 \bullet C_v \sqrt{\frac{\Delta P}{SG}}$$

$$\dot{Q} = \frac{N_1 \bullet C_v}{\sqrt{SG}}\sqrt{\Delta P}$$

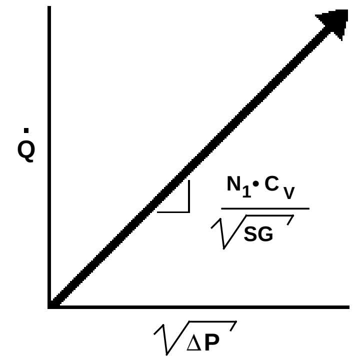
The above equation can be represented by a straight line as follows —

x-axis = $\sqrt{\Delta P}$

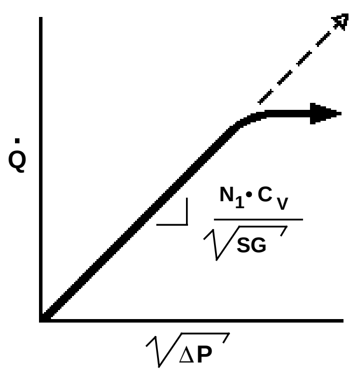
y-axis = $\dot{Q}$

slope = $N_1 \bullet C_V / \sqrt{SG}$

y-intercept = 0

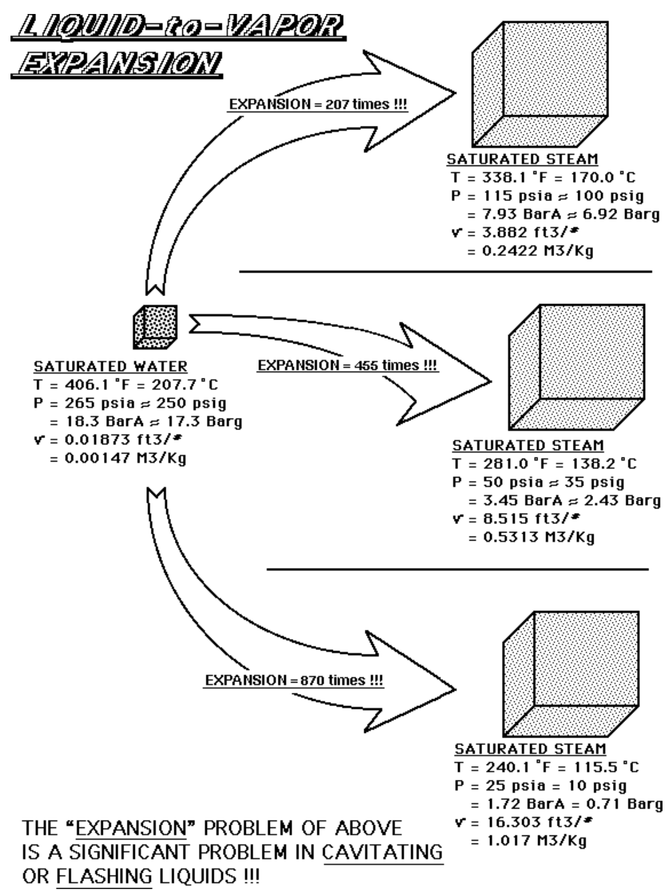


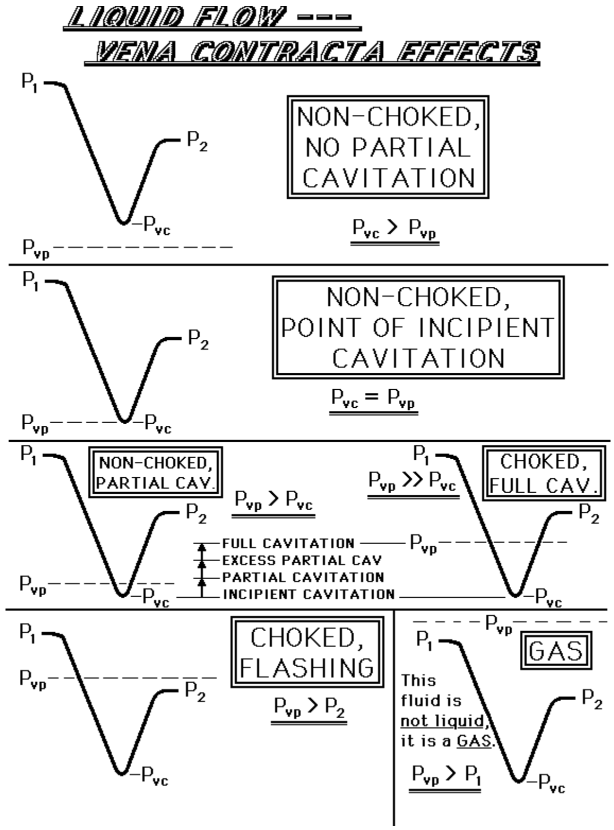
CAVITATION. If a throttling valve is put into a test apparatus using water with adequate flow and pressure capability, it can be observed that in reality the straight-line result does not occur, but begins to show a "deviation" at the higher flow rates (also higher ΔP's). If the ΔP is increased further, a point will be reached where added ΔP will no longer increase flow significantly. In fact, if the ΔP is increased by lowering the P_2-outlet pressure, the flow will remain constant. Conditions have arisen that has the flow described as "CHOKED FLOW". Here, flow is no longer proportional to the square root of pressure drop. The pressure drop corresponding to Choked Flow is "ΔP_{Allow}" (Also known as $\Delta P_{Critical}$ or ΔP_{Choked}).

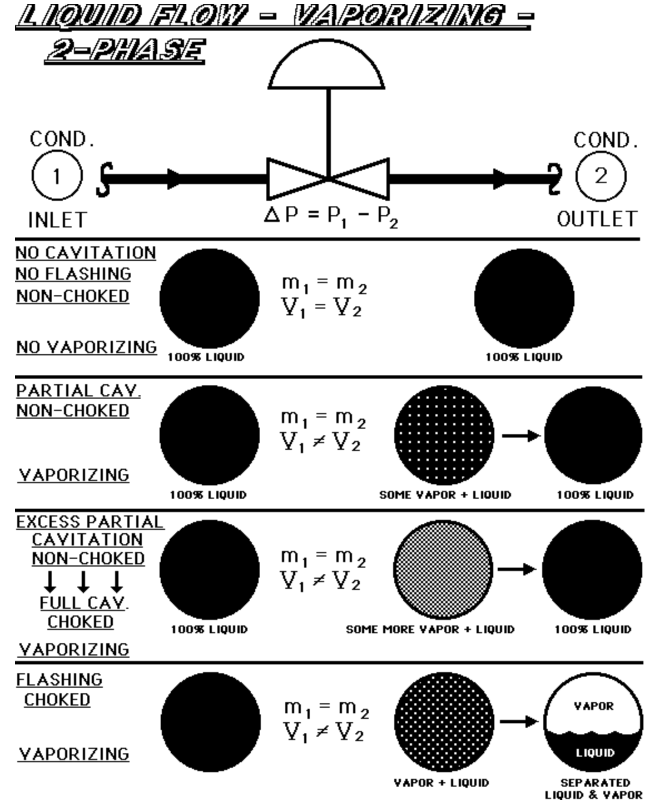


The deviation results because vapor bubbles are forming within the flow stream. A bubble requires 200-800 times as much volume as a comparable liquid mass. The end result is that the average liquid + vapor stream velocity increases dramatically even though only a small percentage of liquid vaporizes. These bubbles are forming because the throttling valve's internal static pressure decreases to a level below the fluid's vapor (saturation) pressure. When this occurs, there are several 2-phase zones that are used to sub-categorize the flow.

"Choked flow" is a condition normally associated with gas-vapor service. "Choked Flow - Liquid" means from a practical viewpoint that the fluid is acting more like a gas vapor than a liquid.







There are three different zones describing “cavitation” that are used by Cashco —

- Partial Cavitation
- Excess Partial Cavitation
- Full Cavitation

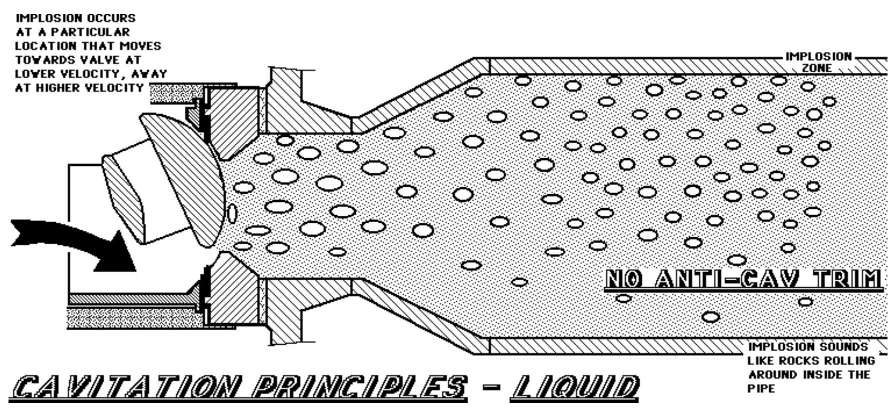
and one category described as “Flashing”.

Cavitating liquids can be broken down into two steps —

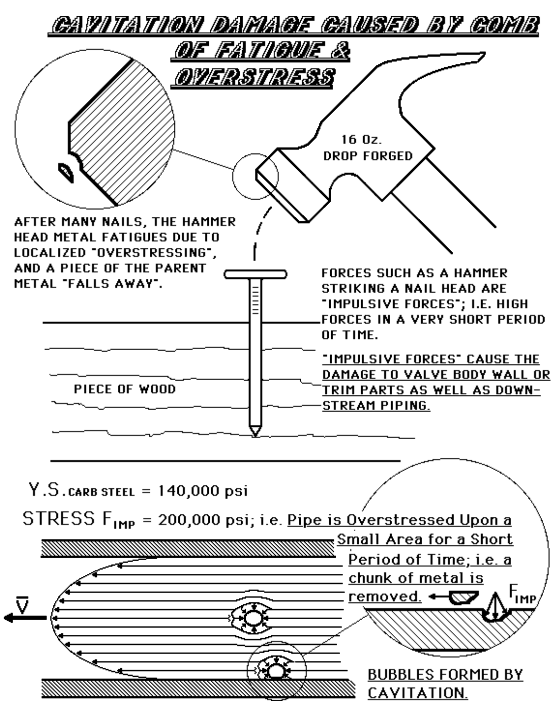
- 1st Step — Formation of vapor bubbles.
- 2nd Step — Collapse or implosion of vapor bubbles.

Cavitation 1st Step. As previously described, the space required for vapor bubbles greatly increases average fluid velocity. The liquid velocity is raised sufficiently to cause increased trim erosion. Fluids that reach descriptions “Excess Partial Cavitation” or “Full Cavitation” should be supplied only with metallic parts; i.e. no soft seats. It would be best for these parts to be hardened against the erosive effects of liquid at high velocity. This is recommended because the vapor bubbles form prior to the flow passing through the main orifice. Fluids that are described in “Partial Cavitation” do not need special trim material considerations. It must only be recognized that trim life will be reduced in comparison to a non-cavitating liquid application. (NOTE: Unfortunately, there exists some nomenclature problems that “confuse” the descriptions given to the vaporizing liquid flow zones. When the vapor bubbles begin to form, this is at times expressed as “flashing”. Thus, all cavitation would begin with flashing, and “Flashing” would also begin and end with “flashing”. This book will ONLY use “Flashing” to describe the liquid zone where $P_2 < P_{VP}$, and the outlet of valve and pipe will contain permanent 2-phase flow.)

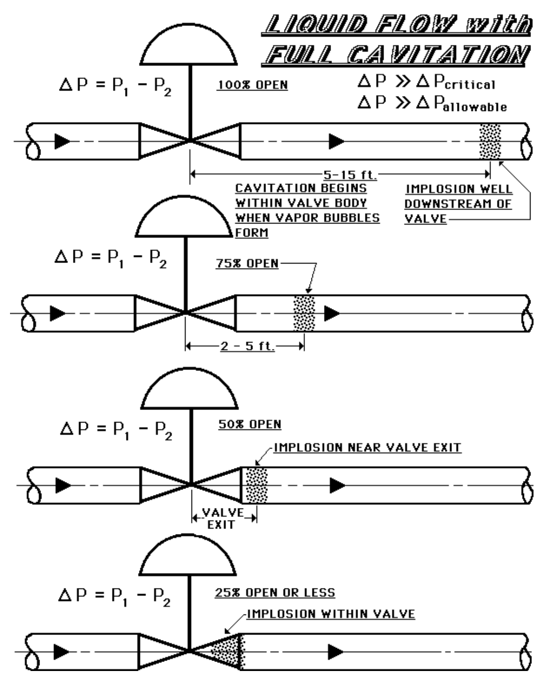
Cavitation 2nd Step. An “implosion” is the opposite of an “explosion”. As the name implies, it is an intense, sudden vapor bubble collapse. Whereas the vapor bubbles smoothly form according to the thermodynamic physical principles, the collapse does not. When static pressure is recovering, it once again crosses the vapor pressure level where it would be expected for the bubbles to disappear; however, unexplainably the bubbles remain beyond the valve internals zone and in many cases will be outside the valve body and in the downstream pipe before implosion occurs. No one has come up with a plausible explanation of this phenomenon; it can only be observed, predicted and accounted for.



Physical Mechanism. The “implosion” of vapor bubbles causes no harm (except vibration) when in the middle of the flow stream. However, when the collapse occurs at the pipe wall, body wall, or trim parts surfaces, mechanical material damage can be observed. The mechanism for failure is fatigue, and is very similar to the impulsive forces imparted by a hammer blow depicted to the left.



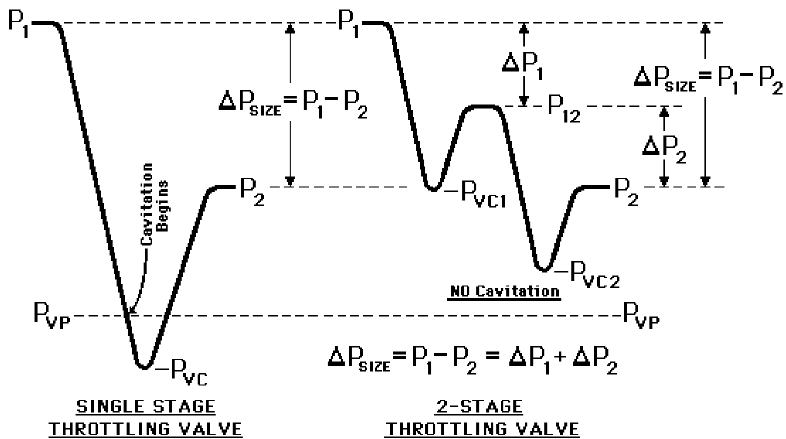
Cavitation is caused by internal fluid pressure conditions, and not by flow rate. As depicted at right, cavitation “moves” around within a valve or its downstream piping. In reality, it creates more “expensive” problems when a fluid is in the high end of the



"Excess Partial Cavitation" zone or barely into the "Full Cavitation" zone, for the implosion can easily occur within the valve body, leading to premature valve "problems" —

- body wall is penetrated
- trim is eroded
- guides and bearings become worn
- packing leaks
- diaphragms fail
- soft seats fail.

Preventing Cavitation. It is the internal static pressure "dip" depth to the vena contracta that causes the initial cavitation. By "staging" the overall ΔP into multiple steps, the reduced depth of the multiple "dips" can result in not crossing below the fluid's vapor pressure.

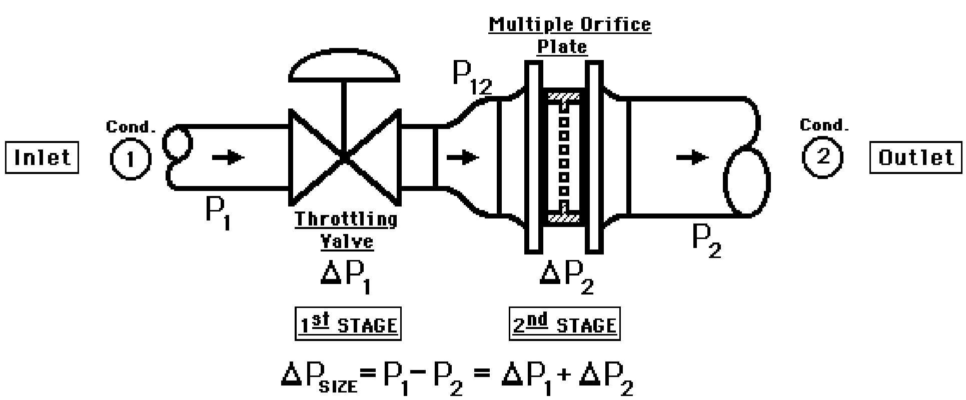


The depth of the vena contracta dip is related to the throttling valve's "F_L factor". Thus, simply by applying a globe or eccentric plug (rotary globe) throttling valve rather than a ball or butterfly throttling valve, you can eliminate cavitation. (See page 5.)

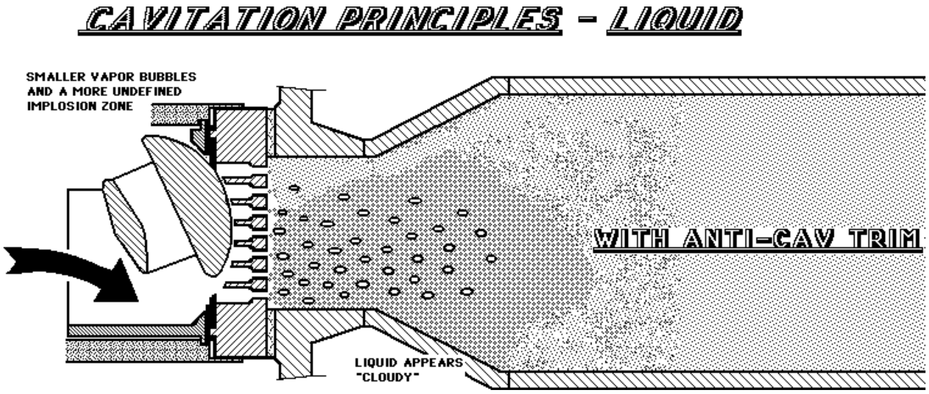
It is possible in liquid systems with a narrow turndown ratio (4:1) to use "multiple orifice plates (MOP)" located either upstream and/or downstream of the throttling valve to eliminate cavitation. Each MOP acts as a stage of ΔP and acts as "fixed orifices"; i.e. as flow rate varies, ΔP_{stage} also varies in accordance with —

$$\dot{Q} \propto \sqrt{\Delta P}$$

Such a system should NOT be allowed to develop any level of cavitation at any stage throughout its turndown range.



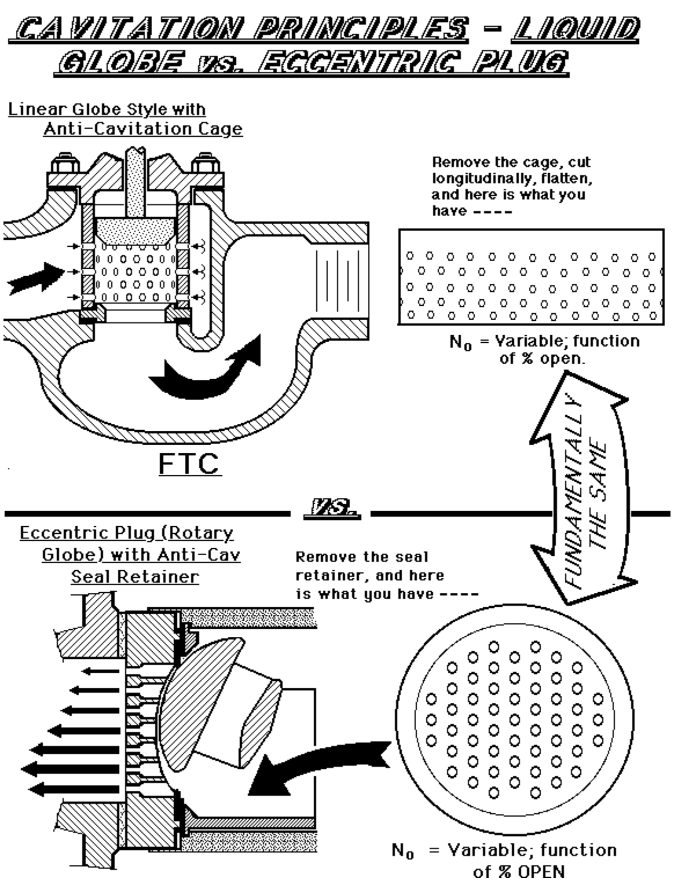
Reducing Cavitation. Most "anti-cavitation" throttling valve trim is somewhat of a misnomer. Such trim provides "multiple orifices" that are blocked-off or uncovered as the valve plug travels. This multiple orifice method still produces cavitation but it develops smaller vapor bubbles within the fluid stream. Thus, when the smaller bubbles implode —



- implosion normally occurs "sooner" rather than "later".
- implosion impulsive forces are smaller.
- better flow stream cross-sectional distribution of the bubbles is developed.

- Higher percentage of vapor bubbles imploding in the midst of the flow stream and away from metallic or seat surfaces; i.e. "collapsing upon itself".
- reduces noise level.
- reduces vibration.

As globe valves and eccentric plug (rotary globe) throttling valves represent the style of throttling valves most frequently applied, the graphic below shows the method used to control cavitation —



In general, the advantage of one style would be the disadvantage of the other style.

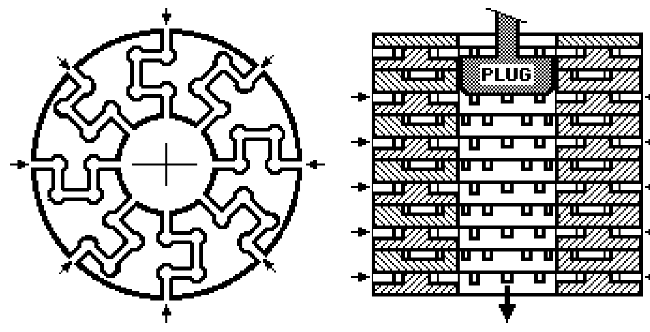
GLOBE Advantages

1. Smaller orifice diameters
2. More orifices
3. Higher rangeability
4. Better orifice "block-off"
5. Higher F_L
6. Lower noise level

ECCENTRIC PLUG Advantages

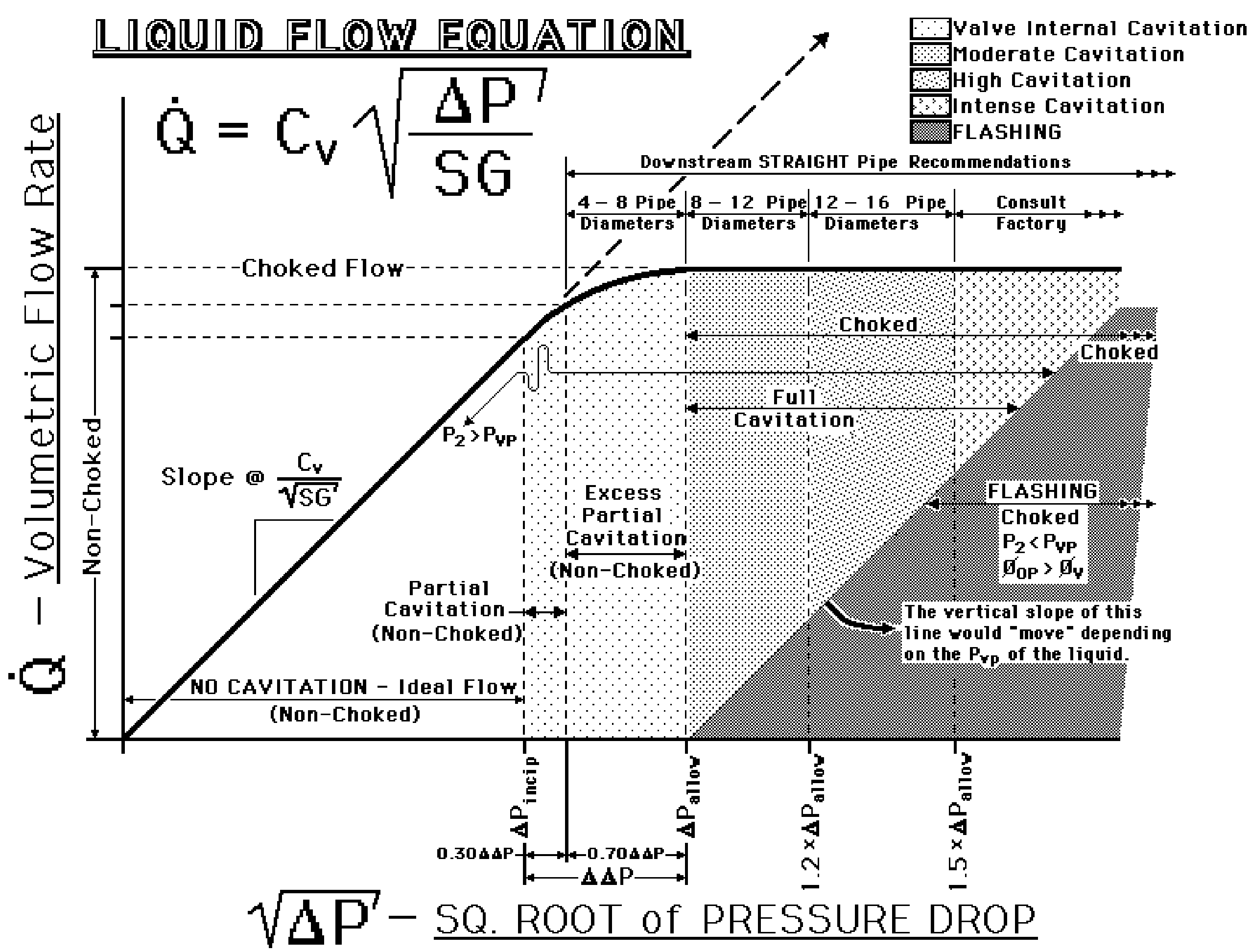
1. All vapor bubbles outside body; no body damage.
2. Seat ring and plug out of presence of vapor bubbles.
3. Higher C_V.
4. Straight-thru flow design.
5. Seal retainer serves as straightening vanes to direct flow to center of downstream pipe; locates implosion.
6. Less expensive parts (1/3 the cost of globe parts).

Combination. For high pressure drops it is possible to utilize in combination both "multi-stage and multi-path "throttling valves. Such valves have been identified as "drag valves", and usually consist of notched flat plates that have multiple channels milled within that take a direction changing circuitous path that can best be described as "tortuous". Each of the notched flat plates are then stacked in a staggered pattern into a "disc stack" to form the multi-stage, multi-path design.



MULTI-STAGE, MULTI-PATH, "DRAG VALVE"

Such drag valves can normally be designed to eliminate cavitation, or at least minimize it. This valve design is very, very expensive and is normally applied in only the most "severe service" instances.



PARTIAL CAVITATION. As shown in the graphics on pages 18 and 22, the "intensity" of the vapor bubble collapse is not so high as to create serious internal mechanical damage, nor is the percentage of fluid mass that converts to vapor high enough to significantly effect the flow rate passed through the throttling valve. A "partially cavitating liquid" is further described as "Non-Choked". There is no special sizing equation that is used for partially cavitating liquid service. The ΔP to cause the "beginning of cavitation (i.e. point of incipient cavitation)" is known as ΔP_{Incip}.

FULL CAVITATION. As shown in the graphics on pages 18 and 22, the "intensity" of the vapor bubble collapse is high enough to create very serious internal mechanical damage to the throttling valve and its downstream piping. The percentage of fluid mass that converts to vapor is high enough to dominate the flow stream, "choking" the flow at the throttling valve's main orifice. This level of cavitation is called "Full Cavitation, Choked Flow". As long as the throttling $\Delta P > 50$ psig (3.5 BarD), throttling valves experiencing such cavitation should be —

- Equipped with "anti-cavitation" trim to reduce cavitation intensity.
- Equipped with drag valve trim to eliminate cavitation.
- Change $\Delta P_{Throttle}$ conditions.
- Incorporate hardened trim.

The ΔP to cause full cavitation is identified as ΔP_{Allow}.

EXCESS PARTIAL CAVITATION. It is common sense to expect that as flow that is partially cavitating, but not yet choked, nears full cavitation and choked conditions, such flow could be nearly as damaging as full cavitation because —

- internal velocities are very high.
- implosion would be more likely to occur within the throttling valve.

Cashco has opted to introduce another level of cavitation intensity identified as "Excess Partial Cavitation". Both ΔP_{Incip} and ΔP_{Allow} can be mathematically calculated. Thus the two empirical values can be subtracted —

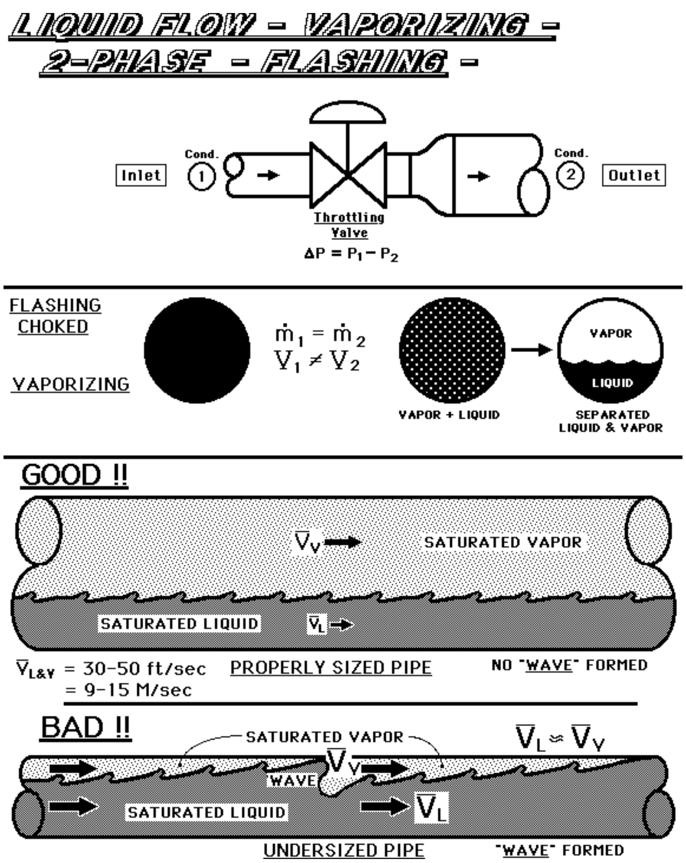
$$\Delta\Delta P = \Delta P_{Allow} - \Delta P_{Incip}$$

Cashco uses the number of 30% of the $\Delta\Delta P$ as a break point between "Partial Cavitation" and "Excess Partial Cavitation". Thus, the zones of cavitation separate as follows —

- Non-choked, Partial Cavitation (0.0–30% $\Delta\Delta P$).
- Non-choked, Excess Partial Cavitation (30.1%–99.9% $\Delta\Delta P$).
- Choked, Full Cavitation ($\Delta P_{Actual} \geq \Delta P_{Allow}$).

Valves experiencing "Excess Partial Cavitation" should be considered for applying anti-cavitational throttling valve trim. This is a "judgement call"; if ΔP_{Actual} is greater than 65% $\Delta\Delta P$ level, anti-cav trim is recommended.

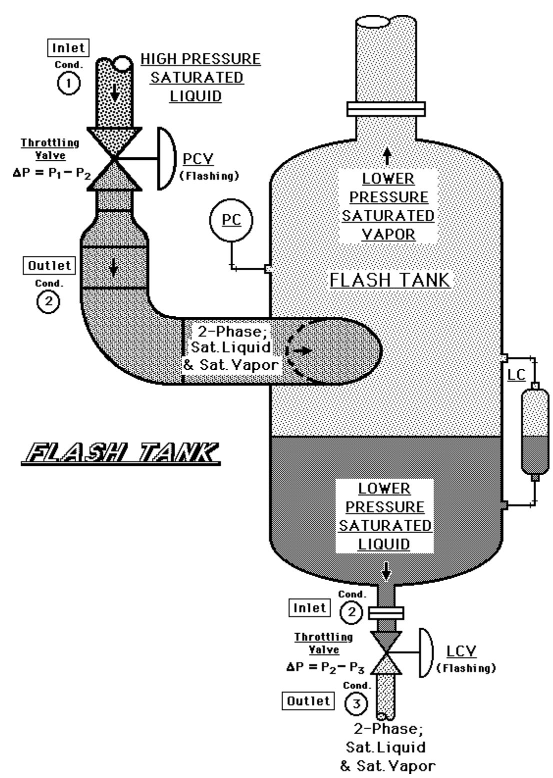
FLASHING. "Flashing" occurs as shown in the graphics on pages 18 and 22 when the $P_{VP} > P_2$, and the liquid portion and vapor portion of the flow stream remain separated as liquid and vapor in the downstream piping. Because vapors typically move at higher velocities than liquids, care has to be taken to keep average combined pipe velocities low enough to prevent formation of a "wave" inside the pipe, like that shown at left. If a wave would eventually cover across the pipe cross-section, it would accelerate the average liquid velocity upwards, approaching the average vapor velocity. Should this occur and the wave would hit an elbow, it could cause a catastrophic failure of the piping support system that could put the piping into a "heap" on the floor! Why? Because water is more dense than vapor, and the momentum of the high mass liquid generates very high impulsive forces, which the pipe elbow and pipe hangers must withstand.



The severity of flashing is increased when the downstream P_2 static pressure is low because –

- Greater expansion of vapor as pressure approaches ambient.
- Greater mass of vapor-to-liquid mass percentage.

Rather than install oversized pipes, typically the liquid is separated from the vapor at a "Flash Tank" as shown at right.



LIQUID SIZING EQUATIONS —

NON-TURBULENT FLOW. The following equations are used for "laminar" or "transitional" flow realms —

(EQ #9)

$$\dot{Q} = N_1 \bullet F_R \bullet C_V \sqrt{\frac{(P_1 - P_2)}{SG}}$$

Parameter			English Units	Metric Units
C_v	=	English Valve Sizing Coefficient	dimensionless	dimensionless
$\dot{Q}$	=	Volumetric Flow Rate	US GPM	M^3/Hr
SG	=	Specific Gravity	dimensionless	dimensionless
P_1	=	Valve Upstream Pressure	psig, psia	Barg, BarA
P_2	=	Valve Downstream Pressure	psig, psia	Barg, BarA
N_1	=	Units Correlation Constant	1.00 dimensionless	0.865 dimensionless
F_R	=	Reynolds No. Correction Factor	dimensionless	dimensionless

F_R can be determined from the graphic on page 13.

There is no known scientific method to evaluate the pipe reducer effect for non-turbulent flow. Thus, the pipe reducer effect is neglected.

TURBULENT FLOW, NON-CHOKED. This is the basic liquid flow equation —

(EQ #8- repeated)

$$\dot{Q} = N_1 \bullet C_v \sqrt{\frac{(P_1 - P_2)}{SG}}$$

(No pipe reducers)

(EQ #10)

$$\dot{Q} = N_1 \bullet F_P \bullet C_v \sqrt{\frac{(P_1 - P_2)}{SG}}$$

(With pipe reducers)

Parameter			English Units	Metric Units
C_v	=	English Valve Sizing Coefficient	dimensionless	dimensionless
$\dot{Q}$	=	Volumetric Flow Rate	US GPM	M^3/Hr
SG	=	Specific Gravity	dimensionless	dimensionless
P_1	=	Valve Upstream Pressure	psig, psia	Barg, BarA
P_2	=	Valve Downstream Pressure	psig, psia	Barg, BarA
N_1	=	Units Correlation Constant	1.00 dimensionless	0.865 dimensionless
F_P	=	Pipe Reducer Effect	dimensionless	dimensionless

F_P makes a correction for the changes in static and velocity pressure as a result of two pipe diameters associated with pipe reducers as well as resistance (i.e. friction) coefficients.

This equation is applied for **"Non-Choked"** turbulent flow zone with the following ΔP "tests" —

$\Delta P_{Actual} < \Delta P_{Incip}$ — **"No Partial Cavitation"**

$\Delta P_{Incip} < \Delta P_{Actual}$
$\Delta P_{Actual} < \left[\Delta P_{Incip} + 0.3\,(\Delta\Delta P)\right]$ — **"Partial Cavitation"**

$\Delta P_{Actual} < \Delta P_{Allow}$
$\Delta P_{Actual} > \left[\Delta P_{Incip} + 0.3\,(\Delta\Delta P)\right]$ — **"Excess Partial Cavitation"**

<u>TURBULENT FLOW, CHOKED</u>. There are several equations that pertain to this zone –

(EQ #11) $$\boxed{\Delta P_{Allow} = F_L^2\,(P_1 - P_{VC})}$$ **(No pipe reducers)**

(EQ #12) $$\boxed{\Delta P_{Allow} = \frac{F_{LP}^2}{F_p^2}\,(P_1 - P_{VC})}$$ **(With pipe reducers)**

Where: $P_{VC} = F_f \bullet P_{VP}$

Where: $F_f = 0.96 - 0.28\sqrt{\dfrac{P_{VP}}{P_C}}$

(**NOTE**: See page 43 for determining F_{LP}.)

ΔP_{Allow} is used as a "test" to determine the turbulent flow zone. If —

$\Delta P_{Actual} \geq \Delta P_{Allow}$ and $P_{VP} < P_2$ → **"Full Cavitation"**

$P_{VP} > P_2$ → **"Flashing"**

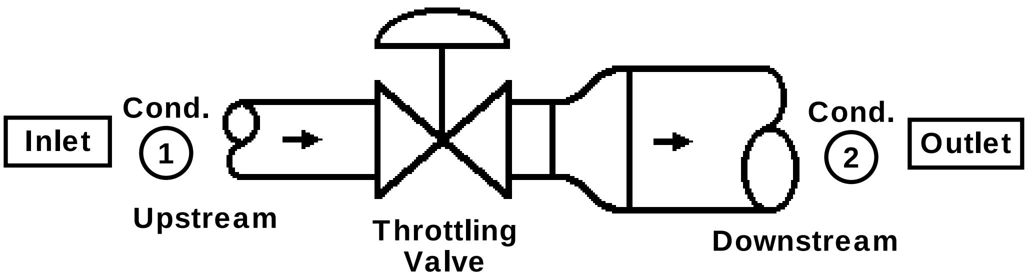
(EQ #13) $$\boxed{\dot{Q} = N_1 \bullet F_L \bullet C_V \sqrt{\frac{P_1 - (F_f \bullet P_{VP})}{SG}}}$$ **(No pipe reducers)**

(EQ #14) $$\boxed{\dot{Q} = N_1 \bullet F_{LP} \bullet C_V \sqrt{\frac{P_1 - (F_f \bullet P_{VP})}{SG}}}$$ **(With pipe reducers)**

Please note that for either "Full Cavitation" or "Flashing", the flow is "choked".

Parameter	**English Units**	**Metric Units**
C_V = English Valve Sizing Coefficient	dimensionless	dimensionless
$\dot{Q}$ = Volumetric Flow Rate	US GPM	M^3/Hr
SG = Specific Gravity	dimensionless	dimensionless
P_1 = Valve Upstream Pressure	psig, psia	Barg, BarA
P_2 = Valve Downstream Pressure	psig, psia	Barg, BarA
N_1 = Units Correlation Constant	1.00 dimensionless	0.865 dimensionless
F_L = Liquid Pressure Recovery Factor	dimensionless	dimensionless
P_{VC} = Estimated Pressure @ Vena Contracta	psia	BarA
F_f = Liquid Critical Pressure Ratio Factor	dimensionless	dimensionless
P_{VP} = Liquid Vapor Pressure	psia	BarA
P_C = Thermodynamic Critical Pressure	psia	BarA
F_{LP} = Combined Liquid Pressure Recovery Factor	dimensionless	dimensionless

GASEOUS FLOW REALM —



$\Delta P = P_1 - P_2$

FLOW DESCRIPTION. Gaseous flow is "compressible flow" as previously discussed in **FLUID STATES**. This means that as the throttling valve experiences its internal pressure drop, the gas is decompressing (expanding), followed in many cases by recompression in the pressure recovery zone. Gas expansion explains why outlet pipe sizes – $Ø_{2P}$ – are typically larger than inlet pipe sizes - $Ø_{1P}$; more "space" is needed.

Laminar or viscous gaseous flow is so very rare an occurrence that there are no "special" sizing routines used for throttling valves. All gaseous flow is regarded as "turbulent". (NOTE: High temperature/pressure polymers are an example of very viscous gaseous flow. Pilot plant testing is used to verify throttling valve suitability/sizing.)

Whereas "liquid choked flow" is problematic, "gaseous choked flow" service is a common occurring set of conditions that cause no particular "mechanical problems", unless accompanied by a high noise level. "Choked Flow" is a self-limiting flow rate for a throttling valve, and it occurs when the gas velocity reaches the speed of sound near the main orifice (i.e. vena contracta) of the valve, which acts to set up a "barrier" to additional flow. "Choked Flow" is sometimes also called "Critical Flow" or "Sonic Flow".

SONIC VELOCITY. The "speed of sound", also known as "sonic velocity" or "Mach 1.0", is a function of the particular type of gas and its temperature —

(EQ #15)
$$\boxed{C = \sqrt{\frac{k \cdot g_c \cdot \bar{R} \cdot T}{MW}}}$$

Parameter			English Units	Metric Units
C	=	Gas Sonic Velocity	ft/sec	M/sec
k	=	Gas Specific Heat Ratio	dimensionless	dimensionless
g_c	=	Gravitational Constant	32.2#m - ft/#f-Sec ^{2}	9.81M/sec^2
$\bar{R}$	=	Universal Ideal Gas Constant	1545#f - ft/# mole - °R	846.8 kg-M/kg mole - °K
T	=	Absolute Gas Temperature	°R	°K
MW	=	Gas Molecular Weight	#/# mole	kg/kg mole

In observing EQ #15 previous, one can see that the sonic velocity is proportional to the square root of the absolute temperature; note that pressure is not involved.

As an example, the sonic velocity of air at ambient conditions is —

$k = 1.4$ MW = 28.95 **T** = 60°F = 520°R = 289°K = 15°C

(English)

$$C = \sqrt{\frac{(1.4)\,(32.2)\,(1545)\,(520)}{28.95}}$$

C = 1118 ft/sec

(Metric)

$$C = \sqrt{\frac{(1.4)\,(9.81)\,(846.8)\,(289)}{(28.95)}}$$

C = 340.7 M/sec

1M = 3.281 ft
Checks Out.

This is a very high velocity, and with throttling valves being frequently in "choked flow" conditions, this is the magnitude of what must be physically considered.

There is a useful factor frequently used to evaluate gaseous service throttling valves —

(EQ #16)

$$\boxed{P_{Ratio} = \left.\frac{P1}{P2}\right)_{Abs}}$$

Parameter	**English Units**	**Metric Units**
P_1 = Valve Upstream Pressure	psia	BarA
P_2 = Valve Downstream Pressure	psia	BarA
P_{Ratio} = Valve Pressure Ratio	dimensionless	dimensionless

The first pass "rule of thumb" estimate gives —

$P_{Ratio} < 2.0$ ➡ "Non-choked Flow"

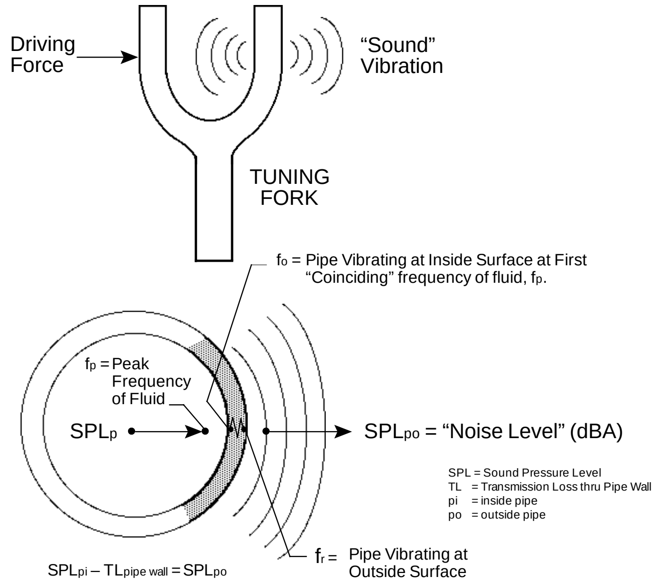
$P_{Ratio} > 2.0$ ➡ "Choked Flow"

This factor is useful because it guides one to know "when" to expect "Choked Flow", and thus when to expect throttling valve noise.

NOISE. Throttling valves are perceived as "noisy" when the measured "Sound Pressure Level – SPL" one meter downstream of the valve outlet and one meter perpendicular to the pipe wall exceeds 85 dBA. The perceived noise is actually the vibration of the pipe itself, much as a tuning fork vibrates. The pressure waves originate from the vena contracta zone of the throttling valve. Obviously, if the pipe is vibrating, then so are the throttling valve internals. There is more to a throttling valve manufacturer attempting to limit noise than just the human ear, the level of internal vibration is also being reduced in order to —

- reduce stem guide wear.
- reduce fatigue effects.
- increase soft seat life.
- reduce packing leaks.
- reduce diaphragm failures.

FUNDAMENTALS OF VALVE DOWNSTREAM PIPE NOISE PREDICTION

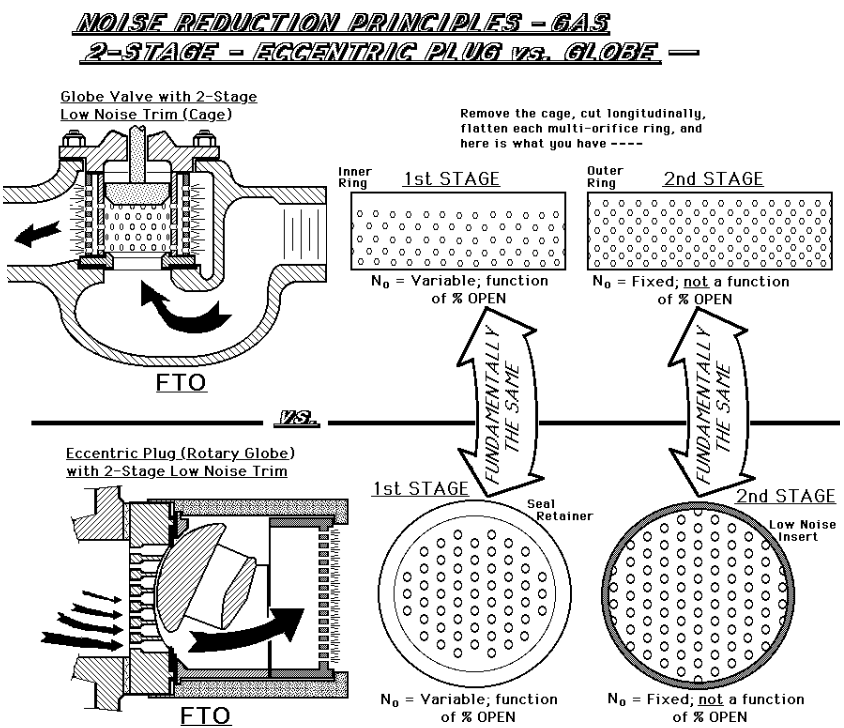


$TL_{pipe\ wall}$ is a function of –

a. pipe material (steel vs. fiberglass)
b. wall thickness
c. f_p – fluid generated peak frequency
d. f_r – external pipe ring frequency
e. f_o – first pipe coincidence frequency

Note: If $f_p \approx f_r \approx f_o$, "noise" level is at worst, as destructive wave interference is maximized!

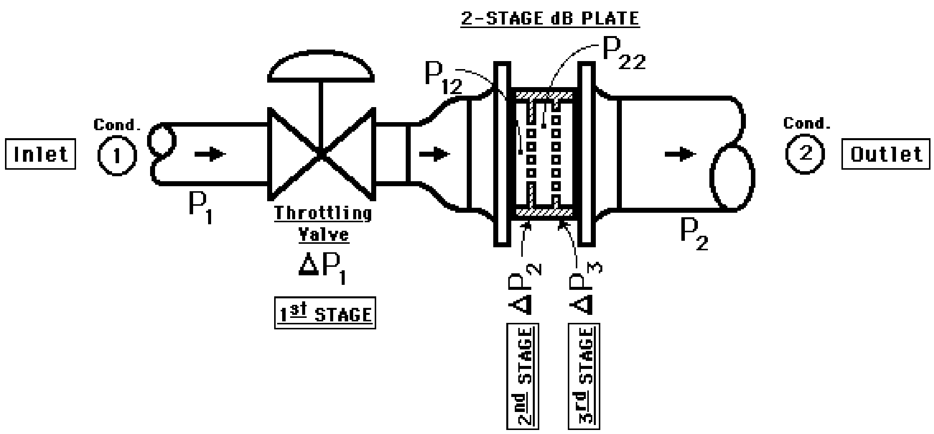
LOW NOISE TRIM. Calculation of throttling valve noise for gaseous service is a very complex issue. This book will only "touch the surface" of this topic. (Readers who are interested in a more detailed discussion of throttling valve noise control principles should request "TAT-001" directly from the Cashco factory.) Basically, throttling valve gaseous noise is controlled much as liquid cavitation is controlled by —



- staging pressure drop.
- controlling noise frequency bands by controlling velocity.
- using multiple orifices.
- using multiple paths.

Drag-type valves are also utilized for severe noise applications, and use passages that increase in cross-sectional area as flow "snakes" its way through the disc stack.

dB PLATES. A “dB Plate” is essentially a MOP as discussed on page 20, with the main difference being that multiple plates can be used to introduce additional ΔP stages. As flow decreases, the ΔP_{stg} decreases in a “square root” relationship, such that this type of device becomes less effective because the variable orifice(s) 1st stage throttling valve must increase its proportionate share of overall ΔP_{Total}, as the fixed orifices dB plates decrease in their proportionate share of overall ΔP_{total}. Such devices have a practical 4:1 turndown in flow rate to remain effective.



$$\Delta P_{SIZE} = P_1 - P_2 = \Delta P_1 + \Delta P_2 + \Delta P_3$$

Example:

ΔP_{Total}	Max Flow	ΔP_1	ΔP_2	ΔP_3
217.5 psid (15 BarD)	100% – High	72.5 psid (5 BarD)	87 psid (6 BarD)	58 psid (4 BarD)
217.5 psid (15 BarD)	50% – Medium	181.2 psid (12.5 BarD)	21.8 psid 1.5 BarD)	14.5 psid (1 BarD)
217.5 psid (15 BarD)	25% – Low	208.5 psid (14.4 BarD)	5.4 psid (0.38 BarD)	3.6 psid (0.25 BarD)

As one can observe, it is VERY IMPORTANT to know both maximum AND minimum conditions to properly size and select dB plates, because the ΔP's shift around as flow changes.

VENA CONTRACTA. Because gaseous flow is decompressing on ΔP through a throttling valve, the internally formed vena contracta takes on some significantly “different” shapes until the vena contracta completely loses it presence at high velocities. To control noise in a throttling valve, we want to keep the formed vena contracta present to the highest degree possible. The most recent ISA recommendations for controlling throttling valve noise use five different “regimes” to classify and describe each stage of ΔP through a throttling valve.

NOISE AND REGIMES

There are several contributors when a high noise level is generated in a throttling valve. The most obvious root causes are:

- High flow rate.
- High pressure drop.
- Low outlet pressure.
- Basic valve type.

Any one of the above can be sufficient to generate excessive noise alone. When two or more are together, one can expect up front that a noise level may be high (dBA > 85) or low (dBA < 85).

The following "rules-of-thumb" can be used as a tip-off to expect a noisy "throttling" application when the flow required is at Cv > 20:

- High Flow Rate. When the inlet pipe is 3" or 4" size, the flow carried is sufficiently high to probably cause high noise level with one other of the causes present also. A 6" inlet pipe alone can carry sufficient flow to cause high noise primarily due to the mass flow rate alone.

  High flow rates tend to generate broader frequency bands, including lower noise frequency levels which are difficult for the pipe wall to "absorb" (i.e. "alternate").

  If the C_v Required is greater than C_v = 50, start expecting high noise level. If the C_v Required is greater than C_v = 100, the noise level will more than likely be high.

- High Pressure Drop. If the ΔP_{Choked} is just reached and the flow is barely choked, noise level will likely be high. If the ΔP_{Actual} is greater than ΔP_{Choked} by 15% or greater, the noise level will likely be high.

  An approximation of ΔP_{Choked} can be as follows:

  $\Delta P_{Choked} \approx$ P1(absolute)/2.

  High pressure drops tend to generate higher noise frequency levels.

- Low Outlet Pressure. When the outlet pressure is P2 < 25 psig (1.7 Barg), the outlet density is relatively low, increasing the possibility of a high velocity, which in turn will generate high noise levels. This consideration alone will almost always cause a high noise level to be developed if the ΔP_{Choked} > 65 psid (4.5 Bard).

  Low outlet pressures tend to generate lower noise frequency levels.

- A better indicator of expected noise is a combination of pressure drop and outlet pressure as follows:

(EQ #16 Repeated)

$$\boxed{\left(\frac{P1}{P2}\right)_{Abs} = \text{Pressure Ratio}}$$

When the P_{Ratio} exceeds 2.3 – 2.7 and there is Cv > 20, noise levels will likely be high (dBA > 85) without the use of noise attenuating trim.

- <u>Basic Valve Type</u>. Different valve types with their different internal flow passage geometries will affect the noise levels generated. In general, the F_L factors below can be used as a guide.

VALVE TYPE	F_L FACTOR	RELATIVE NOISE LEVEL *
Globe	0.9	lowest
Eccentric plug - FTO	0.85	
Eccentric plug - FTC	0.68	
Butterfly	0.65	
Ball	0.45	highest

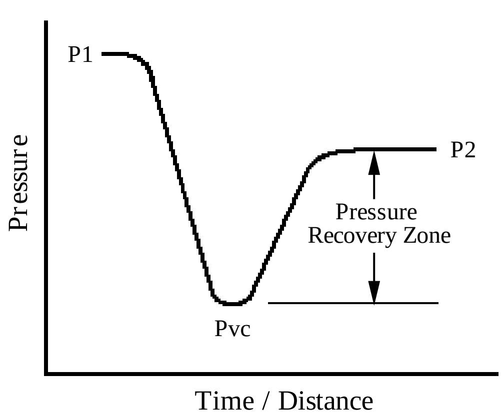
* For same flow conditions.

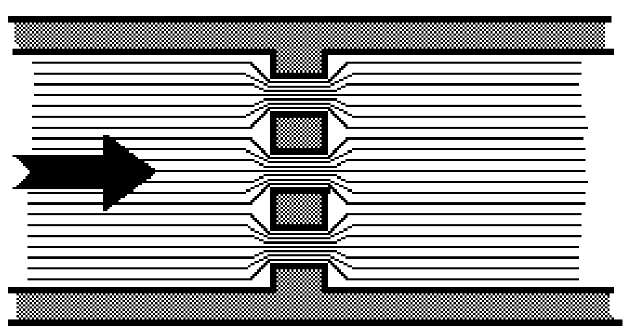
Again in general, globe or FTO eccentric plug (rotary globe) valves are less noisy than a butterfly or ball valve.

REGIMES.

The new terminology concerning "Noise Regimes" definitions are as follows:

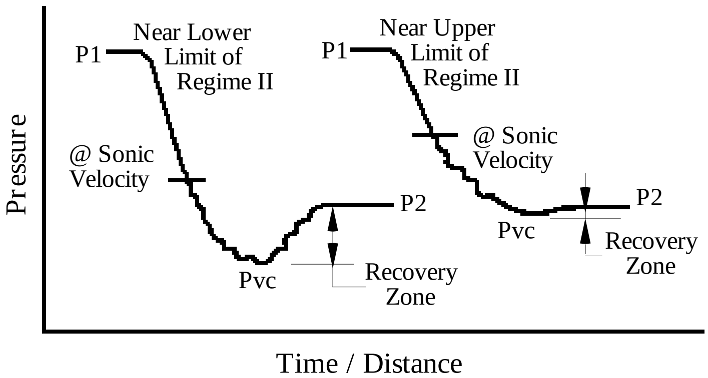
<u>Regime I</u> – Flow is <u>subsonic</u> and the P2 outlet pressure exhibits a high recovery (recompression) level; i.e. well formed, classical vena contracta. No "shock cells" formed.

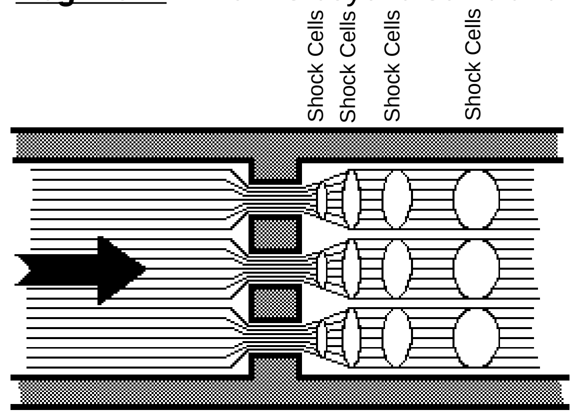




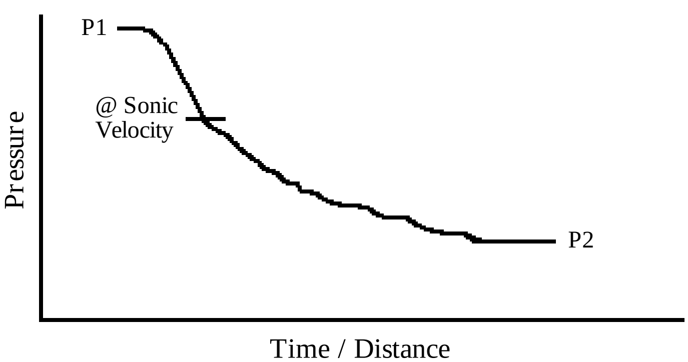
High noise levels would not be expected, except at higher flow rates.

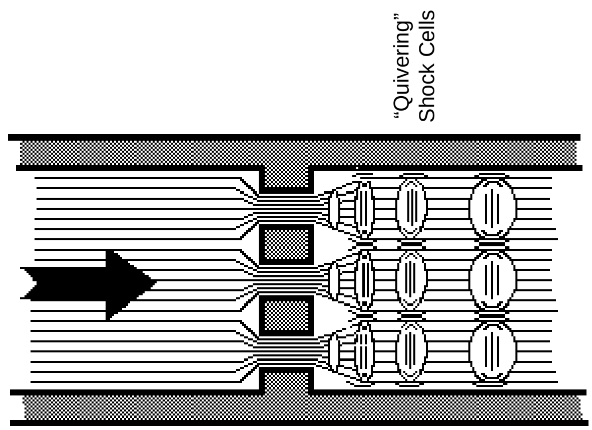
Regime II – Flow is sonic and slightly beyond. "Shock cells" (barriers) develop but do not interact. P2 outlet pressure exhibits some pressure recovery, but lower recovery as upper limit of regime is approached.



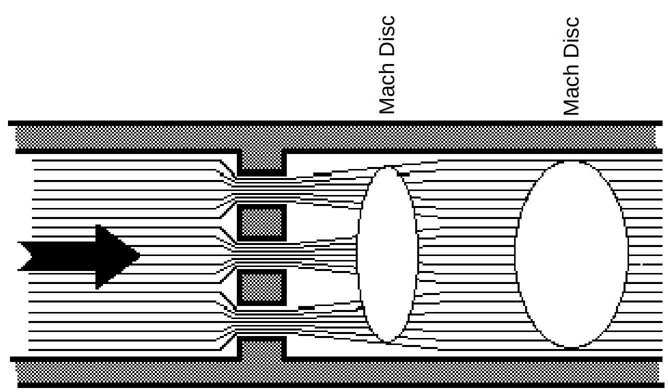


Regime III – Flow is beyond sonic and the inefficiency is such that no pressure recovery takes place. There is no clearly formed vena contracta point in the valve. There is a "strung-out", continuous pressure drop through the valve as flow traverses. Shock cells significantly interact.



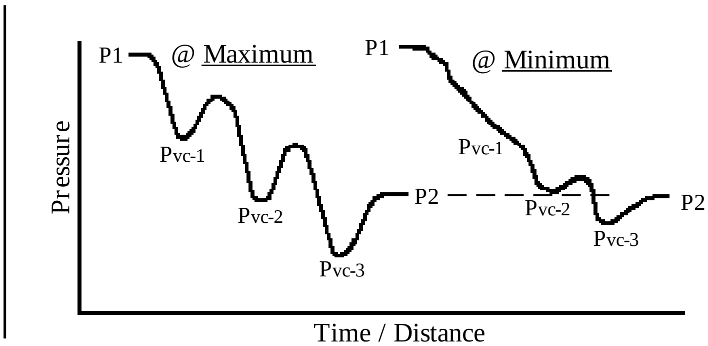


Regime IV – The individually formed shock cells merge together to form a single "Mach Disc". The pressure gradient curve is similar to Regime III above. A "jump" upwards in the noise level occurs after passing from Regime III to Regime IV.



Regime V – In this regime the flow reaches “constant acoustical efficiency”. When in Regime V, if the P2 outlet pressure is lowered, the noise level remains constant; this would not be true in any of Regimes I through IV.

REDUCING NOISE. To reduce noise generation in a throttling valve is best accomplished by controlling the internal velocities to be “sonic” but not “too much sonic”. This keeps the generated frequencies high enough for the downstream pipe wall to attenuate. This is essentially the methodology of most of the different types of noise attenuating trim designs.



The secret is to keep the throttling valve to 20-30% of the overall pressure drop at “maximum” flow condition. As the flow rate decreases and the throttling valve moves towards closing, the control valve's variable orifice(s) will take on a higher percentage of the overall pressure drop, and the fixed orifices a lower percentage.

For fixed downstream orifices within a valve body, noise can be brought down to an acceptable level at maximum conditions, but as flow decreases the valve would become noisier. A reasonable rule-of-thumb is that at about 1/3 maximum flow rate the noise level may “re-peak” (go from Regime III to Regime IV) for such a trim design.

Velocity. The velocity in the outlet pipe should always be kept to Mach No. —

$$M \le 0.30.$$

To exceed the above limit invalidates the calculated noise level prediction equation accuracy.

The velocity may exceed the $M \le 0.30$ limit at the valve outlet ONLY for valves without noise attenuating trim, but the noise prediction will be in error.

The velocity may NOT exceed the $M \le 0.30$ limit at the valve outlet when noise attenuating internal trim is applied. To exceed this limit invalidates the calculated noise level prediction equation accuracy.

VELOCITY CONSIDERATIONS ARE VERY IMPORTANT IN GASEOUS SERVICE!!

The following “limits” are recommended:

GAS VELOCITY LIMITS – Mach No.			
Location	**Recommended**	**Maximum**	**Limitations**
Inlet Pipe	<0.15	<0.225	No noise attenuation trim applied
Valve Body	<0.50	<0.75	
Outlet Pipe *	<0.30	<0.45	
Inlet Pipe	<0.15	<0.225	With noise attenuation trim applied
Valve Body	<0.225	<0.30	
Outlet Pipe *	<0.225	<0.30	

* After any pipe reducers.

OPTIONAL CONSTRUCTION. Generally speaking, gaseous service presents few problems other than noise in throttling service for either control valves or regulators. It is only when the fluid polymerizes, when solids (particulates) are in the flow stream or when liquids in a small percentage (2-phase flow) are present, that special considerations are necessary.

Stellited seating surfaces are seldom required for gaseous service.

If the fluid is “sour gas”, the presence of H_2S requires that the “NACE Construction be selected.

If temperatures are sustained at temperature levels T_1 > 450°F (230°C), live-loaded, high temperature packing or thermal radiation columns should be considered.

STEAM

FLOW DESCRIPTION. Steam is a common case of a gas. Thus, most of the principles of steam flow are the same as those for gaseous flow. The main differences are primarily due to the fact that the degree of expansion for steam is normally higher than other gases when going through the same level of pressure drop. This means that velocity limitations come up more frequently for steam than most gases.

There is a wealth of experience with steam, and this experience has come up with lower recommended pipe velocities for steam than other gases; this is due to the steam being “saturated” in many cases, and thus condensate (2-phase flow) will be present. When this is all put together, the end result is:

- more “noisy” throttling valves
- larger valve body sizes
- larger inlet and outlet pipe sizes.

NOISE & REGIMES. The principles for steam are the same as a gas as previously described.

VELOCITY. The velocity in the outlet pipe should always be kept to Mach Number —

$$M \leq 0.30$$

To exceed the above limit invalidates the calculated noise level prediction equation accuracy.

The velocity may exceed the $M \leq 0.30$ limit at the valve outlet ONLY for valves without noise attenuating trim.

The velocity may NOT exceed the $M \leq 0.30$ limit at the valve outlet when noise attenuating internal trim is applied. To exceed this limit invalidates the calculated noise level prediction equation accuracy.

VELOCITY CONSIDERATIONS ARE ***VERY*** IMPORTANT IN STEAM SERVICE!!

The following “limits” are recommended:

SATURATED @ INLET, SUPERHEATED @ OUTLET – VELOCITY LIMITS - Mach No.

Location	Recommended	Maximum	Limitations
Inlet Pipe	<0.125	<0.225	No noise attenuation trim applied
Valve Body	<0.225	<0.40	
Outlet Pipe *	<0.20	<0.30	
Inlet Pipe	<0.10	<0.20	With noise attenuation trim applied
Valve Body	<0.20	<0.30	
Outlet Pipe *	<0.20	<0.30	

* After any pipe reducers.

SUPER HEATED @ INLET, SUPERHEATED @ OUTLET – VELOCITY LIMITS - Mach No.

Location	Recommended	Maximum	Limitations
Inlet Pipe	0.175	<0.25	No noise attenuation trim applied
Valve Body	0.25	<0.40	
Outlet Pipe *	0.20	<0.30	
Inlet Pipe	0.15	<.225	With noise attenuation trim applied
Valve Body	0.225	<0.30	
Outlet Pipe *	0.20	<0.30	

* After any pipe reducers

SATURATED OR SUPERHEATED @ INLET, FLASHING @ OUTLET – VELOCITY LIMITS – Mach No.

Location	Recommended	Maximum	Limitations
Inlet Pipe	<0.05	<0.10	No noise attenuation trim applied
Valve Body	<0.075	<0.125	
Outlet Pipe *	<0.075	<0.125	
Inlet Pipe	<0.05	<0.10	With noise attenuation trim applied
Valve Body	<0.075	<0.125	
Outlet Pipe *	<0.075	<0.125	

* After any pipe reducers

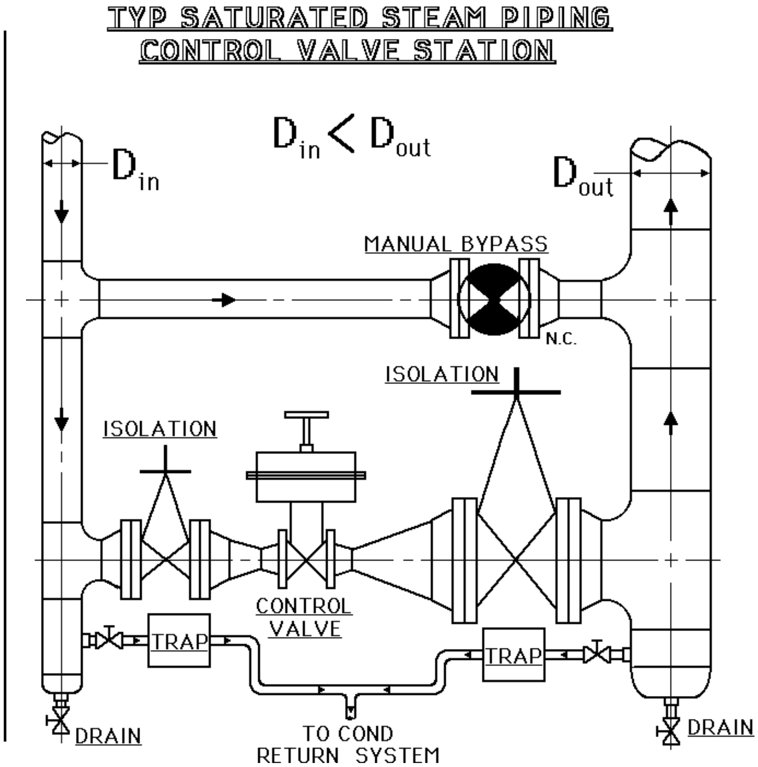
OPTIONAL CONSTRUCTION. Generally speaking, steam service presents few problems in throttling service for either control valves or regulators other than noise. Most problems for steam are caused by improper piping arrangements and inadequate/faulty steam trapping at inlet or outlet.

When pressure drops exceed 150 psid (10 BarD), consideration should be given to using hardened trim; i.e. models with stellite or hardened 416 SST.

If steam is flashing at the outlet, always use stellited or hardened trim. This is severe service.

TFE soft seats/seals are of marginal effectiveness in steam service. Results can be disappointing as to good shutoff. They should be limited to inlet pressures less than 150 psig (10.3 Barg). Metal seated designs for all steam services are recommended.

It is recommended to use stellited trim on applications for steam service where —



- ΔP > 200 psid (14 BarD) and saturated.
- ΔP > 150 psid (10 BarD) and superheated.
- Where valve is in “closed” position for extended time.

When steam pressure is greater than 500 psig (34.5 Barg) and “saturated”, trim should be stellited.

If temperatures at outlet are sustained at temperature levels T_2 > 450°F (230°C), live-loaded, high temperature packing or thermal radiation columns should be considered.

For steam systems it is mandatory that steam traps be located directly before the inlet to the control valve, particularly if the steam is supposed to be “saturated”. Any condensate (liquid) entrained in the flow stream at the inlet will cause accelerated trim wear due to erosion.

GASEOUS SIZING EQUATIONS

CHOKED VS. NON-CHOKED. The first step to gaseous throttling valve sizing is to determine whether the flow is "choked" (i.e. "critical" or "sonic") at the main orifice, or "non-choked" (i.e. "sub-critical" or "sub-sonic" at same. As pipe reducers are almost always included on both the upstream and downstream of most throttling valves, the gaseous equations include the "pipe reducer effects".

The following preliminary calculations and then tests are applied to determine the flow description at the throttling valve's main orifice and to determine the proper factors involved —

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

$$X_T \approx V_{GF} \bullet F_L^2$$

where: V_{GF} varies as to valve type.

Valve Type	$\mathbf{V_{GF}}$
Globe - Single Port	.85
Eccentric Plug	.84
Butterfly	.82
Ball	.81
Globe - Double Port	.83

NOTE: See page 44 for determining X_{TP}

$$X = \frac{\Delta P}{P_1} = \frac{(P_1 - P_2)}{P_1}$$

(EQ #17)

$$\boxed{Y = 1 - \left[\frac{X}{3 \bullet F_k \bullet X_{TP}}\right]}$$

With Limit: $2/3 \leq Y \leq 1.0$

and

Limit: $X_{Max} = F_k \bullet X_{TP}$

(Use "lesser value" of X or X_{max} in above EQ #17.)

When "Y" reaches its minimum value of 2/3 (0.67), the flow is described as "Choked Flow". If "Y" is greater than 2/3 (0.67), the flow is described as "Non-Choked Flow".

Parameter	English Units	Metric Units
k = Specific Heat Ratio	dimensionless	dimensionless
F_k = Specific Heat Ratio Factor	dimensionless	dimensionless
X = Pressure Drop Ratio	dimensionless	dimensionless
X_T = Pressure Drop Ratio Factor	dimensionless	dimensionless
X_{TP} = Combined Reducers/Pressure Drop Ratio Factor	dimensionless	dimensionless
F_L = Liquid Pressure Recovery Factor	dimensionless	dimensionless
V_{GF} = Valve Geometry Factor	dimensionless	dimensionless
P_1 = Upstream Pressure Absolute	psia	BarA
P_2 = Downstream Pressure Absolute	psia	BarA
ΔP = Sizing Pressure Drop	psid	BarD
Y = Expansion Factor	dimensionless	dimensionless
X_{max} = Choking Limit Factor	dimensionless	dimensionless

<u>STD. VOLUMETRIC FLOW BASIS</u>.

(EQ #18)

$$C_V = \frac{\dot{Q}}{N_7 \bullet F_P \bullet P_1 \bullet Y} \sqrt{\frac{SG \bullet T_1 \bullet Z_1}{X}}$$

(EQ #19)

$$C_V = \frac{\dot{Q}}{N_9 \bullet F_P \bullet P_1 \bullet Y} \sqrt{\frac{MW \bullet T_1 \bullet Z_1}{X}}$$

Parameter	English Units	Metric Units
C_v = English Valve Sizing Coefficient	dimensionless	dimensionless
$\dot{Q}$ = <u>MSC - Standard</u> Volumetric Flow Rate	SCFH	SM^3/Hr
* $\dot{Q}_N$ = <u>STP - Normal</u> Volumetric Flow Rate	—	NM^3/HR
Z_1 = Compressibility Factor	dimensionless	dimensionless
SG = Gas Specific Gravity	dimensionless	dimensionless
MW = Molecular Weight	#/# mole	kg/kg mole
T_1 = Absolute Temperature	°R	°K
P_1 = Upstream Pressure Absolute	psia	BarA
Y = Expansion Factor	dimensionless	dimensionless

Parameter (Cont.)			English Units	Metric Units
	F_P	= Piping Geometry Factor	dimensionless	dimensionless
	N_7	= Units Correlation Constant	1360 dimensionless	416 dimensionless
	N_9	= Units Correlation Constant	7320 dimensionless	2245 dimensionless
*	N_{7N}	= Units Correlation Constant	—	393 dimensionless
*	N_{9N}	= Units Correlation Constant	—	2121 dimensionless

* Use N_{7N} or N_{9N} in conjunction with $\dot{Q}_N$.
(NOTE: See page 45 for clarification of "STP–Normal" or "MSC–Standard" standard conditions.)

MASS FLOW BASIS.

(EQ #20)

$$C_V = \frac{\dot{m}}{N_6 \bullet F \bullet Y} \left[\frac{1}{\sqrt{X \bullet P_1 \bullet \gamma_1}} \right]$$

(EQ #21)

$$C_V = \frac{\dot{m}}{N_8 \bullet F_p \bullet P_1 \bullet Y} \sqrt{\frac{T_1 \bullet Z_1}{X \bullet MW}}$$

(EQ #22)

$$C_V = \frac{\dot{m}}{N_{10} \bullet F_p \bullet P_1 \bullet Y} \sqrt{\frac{T_1 \bullet Z_1}{X \bullet SG}}$$

Parameter		English Units	Metric Units
$\dot{m}$	= Mass Flow Rate	#/Hr	Kg/Hr
C_v	= English Valve Sizing Coefficient	dimensionless	dimensionless
MW	= Molecular Weight	#/# mole	kg/kg mole
SG	= Gas Specific Gravity	dimensionless	dimensionless
Z_1	= Compressibility Factor	dimensionless	dimensionless
X	= Pressure Drop Ratio	dimensionless	dimensionless
Y	= Expansion Factor	dimensionless	dimensionless
P_1	= Upstream Pressure Absolute	psia	BarA
T_1	= Absolute Temperature	°R	°K
F_p	= Piping Geometry Factor	dimensionless	dimensionless
γ	= Actual Density @ Upstream Cond.	$\#/ft^3$	kg/M^3
N_6	= Units Correlation Constant	63.3 dimensionless	27.3 dimensionless
N_8	= Units Correlation Constant	19.3 dimensionless	94.8 dimensionless
N_{10}	= Units Correlation Constant	104 dimensionless	510 dimensionless

STEAM. As steam is a vapor/gas that does not exist as a fluid within piping systems at standard conditions of pressure and temperature, Equation #20 is utilized using actual density — γ_{Actual} and mass flow basis. As this equation is based on "actual non-volumetric" conditions, there is no Compressibility Factor – Z_1 required to correct for deviation from the ideal gas laws. Should Equations #21 or #22 be utilized, these are based on the ideal gas laws, so Compressibility Factor – Z_1 is required as an input.

MISCELLANEOUS — PIPE REDUCERS CORRECTION FACTORS

TRIAL AND ERROR SOLUTION. A close look at EQ's #23 and #24 show that both "C_V" and "d_{1V}" are included in the formulas correcting for pipe reducer effects. Since determining "C_v and then "d_{1v}" is the ultimate purpose of the sizing equations, it should be obvious that the only possible method of a sizing calculation must be done iteratively — repeat trial and error calculations.

LIQUIDS AND GASES.

(EQ #23)

$$\boxed{F_p = \frac{\sum K \cdot C_v^2}{N_2 \cdot d_{1v}^4} + 1}$$

Where: $\sum K = K_1 + K_2 + K_{B1} - K_{B2}$

$$K_1 = 0.5\left[1 - \frac{d_{1v}^2}{D_{1P}^2}\right]^2$$

$$K_2 = 1.0\left[1 - \frac{d_{2v}^2}{D_{2P}^2}\right]^2$$

$$K_{B1} = 1 - \left[\frac{d_{1v}}{D_{1P}}\right]^4$$

$$K_{B2} = 1 - \left[\frac{d_{2v}}{D_{2P}}\right]^4$$

NOTE: Both the inlet and outlet pipe reducer effects are involved. If reducers are the same size for inlet and outlet, $K_{B1} = K_{B2}$ then the terms cancel out.

Parameter			English Units	Metric Units
F_P	=	Piping Geometry Factor	dimensionless	dimensionless
ΣK	=	Combined Head Loss Coefficient	dimensionless	dimensionless
C_v	=	English Valve Sizing Coefficient	dimensionless	dimensionless
N_2	=	Units Correlation Constant	890 dimensionless	0.00214 dimensionless
d_{1V}	=	Valve Inlet Body Size	in.	mm
d_{2V}	=	Valve Outlet Body Size	in.	mm
D_{1P}	=	Inlet Pipe Size (before reducer)	in.	mm
D_{2P}	=	Outlet Pipe Size (after reducer)	in.	mm
K_1	=	Inlet Resistance Coefficient	dimensionless	dimensionless
K_2	=	Outlet Resistance Coefficient	dimensionless	dimensionless
K_{1B}	=	Inlet Bernoulli Coefficient	dimensionless	dimensionless
K_{2B}	=	Outlet Bernoulli Coefficient	dimensionless	dimensionless

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LIQUIDS.

(EQ #24)

$$\boxed{F_{Lp} = F_L \left[\frac{\Sigma Ki \cdot F_L{}^2 \cdot C_v{}^2}{N_2 \cdot d_{1v}{}^4} + 1 \right]^{-1/2}}$$

Where: $\Sigma Ki = K_1 + K_{B1}$

$$K_1 = 0.5 \left[1 - \frac{d_{1v}{}^2}{D_{1P}{}^2} \right]^2$$

$$K_{B1} = 1 - \left[\frac{d_{1v}}{D_{1P}} \right]^4$$

NOTE: Only the <u>inlet</u> pipe reducer effect is involved.

Parameter			English Units	Metric Units
F_L	=	Liquid Pressure Recovery Factor	dimensionless	dimensionless
F_{LP}	=	Combined Reducers/Liquid Pressure Recovery Factor	dimensionless	dimensionless
ΣKi	=	Combined Inlet Head Loss Coefficient	dimensionless	dimensionless
C_v	=	English Valve Sizing Coefficient	dimensionless	dimensionless
N_2	=	Units Correlation Constant	890 dimensionless	0.00214 dimensionless

Parameter (Cont.)	English Units	Metric Units
d_{1V} = Valve Inlet Body Size	in.	mm
D_{1P} = Inlet Pipe Size (before reducer)	in.	mm
K_1 = Inlet Resistance Coefficient	dimensionless	dimensionless
K_{B1} = Inlet Bernoulli Coefficient	dimensionless	dimensionless

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GASES.

(EQ #25)

$$\boxed{X_{Tp} = \frac{X_T}{F_p^2}\left[\frac{X_T \bullet \Sigma Ki \bullet C_v^2}{N_5 \bullet d_{1v}} + 1\right]^{-1}}$$

Where: $\Sigma Ki = K_1 + K_{1B}$

$$K_1 = 0.5\left[1 - \frac{d_{1v}^2}{D_{1P}^2}\right]^2$$

$$K_{B1} = 1 + \left[\frac{d_{1v}}{D_{1P}}\right]^4$$

NOTE: Both the inlet and outlet pipe reducer effects are involved through F_p.

Parameter	English Units	Metric Units
X_{TP} = Combined Reducers / Pressure Drop Ratio Factor	dimensionless	dimensionless
X_T = Pressure Drop Ratio Factor	dimensionless	dimensionless
ΣKi = Combined Inlet Head Loss Coefficient	dimensionless	dimensionless
C_v = English Valve Sizing Coefficient	dimensionless	dimensionless
F_P = Piping Geometry Factor	dimensionless	dimensionless
N_5 = Units Correlation Constant	1000 dimensionless	0.00241 dimensionless
d_{1V} = Valve Inlet Body Size	in.	mm
D_{1P} = Inlet Pipe Size (before reducer)	in.	mm
K_1 = Inlet Resistance Coefficient	dimensionless	dimensionless
K_{B1} = Inlet Bernoulli Coefficient	dimensionless	dimensionless

STANDARD CONDITIONS FOR GASES

ENGLISH	METRIC
Std. Temp. = 60°F	"MSC" – Metric Std. Conditions
Std. Pressure = 14.7 psia	Std. Temp. = 15°C
	Std. Pressure = 1 Atm
	"STP" – Std. Temp. & Pressure
	or
	"N" = Normal
	Std. Temp = 0°C
	Std. Pressure = 1 Atm

"*Normal*" Metric Volume	"*Standard*" Metric Volume
NM^3/Hr	SM^3/Hr M^3/Hr
N lit/min	S lit/min lit/min

If the flow rate does not indicate as "N" or an "S",assume that it means "standard" metric volume.

IDEAL GAS vs. REALITY—

Use of the ideal gas law is extensive in the process industry with a correction factor to adjust for the "ideal vs. real" conditions. This correction factor is known as —

Z - the "compressibility factor"

Combined Ideal Gas Law	Combined Real Gas Law
$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$	$\boxed{Z_1 \times \frac{P_1V_1}{T_1} = Z_2 \times \frac{P_2V_2}{T_2}}$
$PV = n\bar{R}T$	$\boxed{PV = ZnRT}$
$PV = mRT$	$\boxed{PV = ZmRT}$

SPECIFIC GRAVITY – DENSITY – MOLECULAR WEIGHT

ρ OR γ (rho or gamma are used as density terms)

$\rho\ STD = \frac{MW}{V\ molar}$ SG gas = $\frac{\rho\ fluid}{\rho\ air}$

ENGLISH

ρ air = .07622 #/ft^3= ρ_{STD}

V molar = 379.8 ft^3/# mole

@ 60°F & 14.7 psia (1 ATM)

$\boxed{SG = 13.120 \times \rho_{Fluid}}$

$\boxed{MW = 379.8 \times \rho_{STD.}}$

$\boxed{MW = 28.95 \times SG_{STD.}}$

METRIC - STP

ρ air = 1.2924 kg/m^3 = ρ_{STP}

V molar = 22.40 m^3/kg - mole

@ 0°C & 1 ATM

$\boxed{SG = .7738 \times \rho_{Fluid}}$

$\boxed{MW = 22.40 \times \rho_{STP.}}$

$\boxed{MW = 28.95 \times SG_{STP.}}$

METRIC - MSC

ρ_{air} = 1.2225 kg /m^3= ρ_{MSC}

V $_{molar}$ = 23.68 m /kg ^{3}- mole

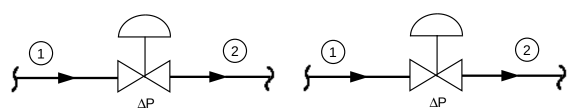
@ 15°C & 1 ATM

$\boxed{SG = .8180 \times \rho_{Fluid}}$

$\boxed{MW = 23.68 \times \rho_{MSC}}$

$\boxed{MW = 28.95 \times SG_{MSC}}$

VOLUMETRIC FLOW$_{STD\ CONDITIONS}$ vs. MASS FLOW—GAS



$\dot{Q}_{1E}$ = 15,000 SCFH	$\dot{Q}_{2E}$ = ??	$\dot{m}_{1E}$ = 5000 #/hr	$\dot{m}_{2E}$ = 5000 #/hr
$\dot{Q}_{1M}$ = 430 SM^3/hr	$\dot{Q}_{2M}$ = ??	$\dot{m}_{1M}$ = 2500 kg/hr	$\dot{m}_{2M}$ = 2500 kg/hr

By the "<u>law of conservation of mass"</u> we know that $m_{in} = m_{out}$, as a valve does not "store" any mass, and most processes experience "<u>steady-state, steady flow"</u> conditions. A gaseous <u>mass flow rate</u> represents an "<u>actual flow rate</u>".

When gaseous flow is expressed as a "<u>volumetric flow rate @ standard conditions"</u> — SCFH, SCFM, NM^3/Hr, SM^3/Hr, etc., this flow rate is a "<u>fictious</u>" flow rate that could not occur if a pressure drop takes place. Standard volumetric flow rates are used for ease of mathematical manipulation and application to the "<u>combined ideal gas law</u>" and the "<u>basic equation of state</u>".

$\dot{Q}_1$ above <u>must</u> be equal to $\dot{Q}_2$, for the above. It is a convenient way to calculate with a basis of $\dot{Q}_{in} = \dot{Q}_{out}$. Thus, both the $\dot{Q}_{in}$ and $\dot{Q}_{out}$ are "<u>fictious</u>". Any "<u>actual</u>" calculation — such as a velocity calculation — would have to be converted to $\dot{Q}_{in\ actual}$ or $\dot{Q}_{out\ actual}$ to do correctly. To calculate a velocity directly from SCFH or NM^3/Hr would be a waste of time as it would be meaningless.

IDEAL GAS LAWS —

Introducing <u>molar principles</u> allows for many <u>varying forms</u> to be made of the "<u>Combined Gas Law</u>" —

P	–	**absolute Pressure**	**T**	–	**absolute temperature**	$\acute{V}$	–	**specific volume**
V	–	**volume**	**MW**	–	**molecular weight**	$\mathbb{V}$	–	**molar volume**
n	–	**no. of moles**	**m**	–	**mass**	$\bar{\acute{V}}$	–	**molar specific volume**
$\bar{R}$	–	**molar (universal) gas constant**	ρ	–	**density**	**Z**	–	**compressibility**
T	–	**absolute temperature**	**SG**	–	**specific gravity**			

MOLAR PRINCIPLE 1 <u>1 mole of gas has a mass of 1 MW.</u>

Example – Nitogen gas; exists as a molecue in the form N_2.

MW = 2 x 14 = 28 # / #mole

1 # mole N_2 - 28 #

1 gm-mole N_2 = 28 gm

1 kg-mole N_2 = 28 kg

$$\boxed{n = \frac{m}{MW}}$$

MOLAR PRINCIPLE 2 <u>1 mole of gas occupies 1 molar volume at standard conditions of pressure and temperature</u>.

1#mole	occupies 379.8 ft^3	@	P=14.7 psia & T=60°F
1#mole	occupies 359.3 ft^3	@	P=14.7 psia & T=32°F
1#mole	occupies 393.4 ft^3	@	P=30" Hg & T = 80°F

<u>"N" – "NORMAL" CONDITIONS - METRIC STP</u>

1 gm-mole occupies 22.40 N•lit @ P = 1 atm & T = 0°C

1 kg-mole occupies 22.40 Nm^3 @ P = 1 atm & T = 0°C

<u>"S" – "METRIC "STD" CONDITIONS - METRIC MSC</u>

1 gm-mole occupies 23.68 S•lit @ P = 1 atm & T = 15°C

1 kg-mole occupies 23.68 SM^3 @ P = 1 atm & T = 15°C

MOLAR PRINCIPLE 3 <u>Gas density (SG, $\acute{V}_g$ also) is readily available if the MW of the gas is known.</u>

$$\boxed{\rho_{std} = \frac{MW}{\mathbb{V}_{molar}}} \qquad \boxed{\acute{V}_{std} = \frac{\mathbb{V}_{molar}}{MW}}$$

$$SG = \left.\frac{\rho_{gas}}{\rho_{air}}\right)_{std.} = \frac{MW_{gas} / \mathbb{V}_{molar}}{MW_{air} / \mathbb{V}_{molar}}$$

$$\boxed{SG = \frac{MW_{gas}}{28.95}}$$

where MW_{air} = 28.95# / #mole
= 28.95 kg / kg-mole
= 28.95 gm/gm-mole

MOLAR PRINCIPLE 4 <u>There is a "universal gas constant - $\bar{R}$" that is a function of the absolute pressure (P), molar volume ($\mathbb{V}$), and absolute temperature (T) that gives a "molar look" to the combined gas law.</u>

(Combined Gas Law)

$$\frac{P \cdot V}{T} = \left.\frac{P_{std} \cdot V_{std}}{T_{std}}\right)_{Std.\ Conditions}$$

$$Pv = \frac{P_{std}}{T_{std}} \times V_{std} \times T = \left(\frac{P}{T}\right)_{Std} \times [n \cdot \mathbb{V}_{molar}] \times T$$

where, $V = n \cdot \mathbb{V}_{molar}$

$$\frac{P_{std} \cdot \mathbb{V}_{molar}}{T_{std}} =$$ <u>Constant</u> with a "mass" component = $\bar{R}$

MOLAR PRINCIPLE 4 (Cont.)

$$R = \frac{(14.70\ \#f/in^2)\ (379.8\ ft^3/\#mole)\ (144\ in^2/ft^2)}{(60 + 460)°R}$$

$\bar{R}$ - 1545 $\dfrac{\#f \cdot ft}{\#mole \cdot °R}$

- 0.08206 $\dfrac{atm \cdot lit}{gm\text{-}mole \cdot °K}$
- 0.08206 $\dfrac{atm \cdot M^3}{kg\text{-}mole \cdot °K}$
- 0.08317 $\dfrac{BarA \cdot M^3}{kg\text{-}mole \cdot °K}$

(Basic Equation of State) $\boxed{Pv = n\bar{R}T}$

<u>Knowing these parameter relationships</u> —

$\rho = 1/\acute{V}$	$V = n\mathbb{V}_{molar}$	$V = m/\rho$
$R = \bar{R}/MW$	$\acute{V} = \bar{\acute{V}}/MW$	$\rho = m/V$
$V = m\acute{V}$	$n = m/MW$	

<u>rearranging and substituting</u>, the following equations can be developed —

$$\boxed{PV = mRT} \quad \boxed{P = \frac{RT}{\acute{V}}} \quad \boxed{P = \frac{\rho\bar{R}T}{P \cdot MW}}$$

$$\boxed{\rho = \frac{P}{RT}} \quad \boxed{P\acute{V} = RT} \quad \boxed{P = \frac{\bar{R}T}{\acute{V} \cdot MW}}$$

$$\boxed{\acute{V} = \frac{RT}{P}} \quad \boxed{\rho = \frac{P \cdot MW}{\bar{R}T}}$$

$$\boxed{P = \rho RT} \quad \boxed{\acute{V} = \frac{\bar{R}T}{P \cdot MW}}$$

MOLAR PRINCIPLE 5 <u>For a "balanced" chemical reaction formula, the expressed "coefficients" are in "moles".</u>

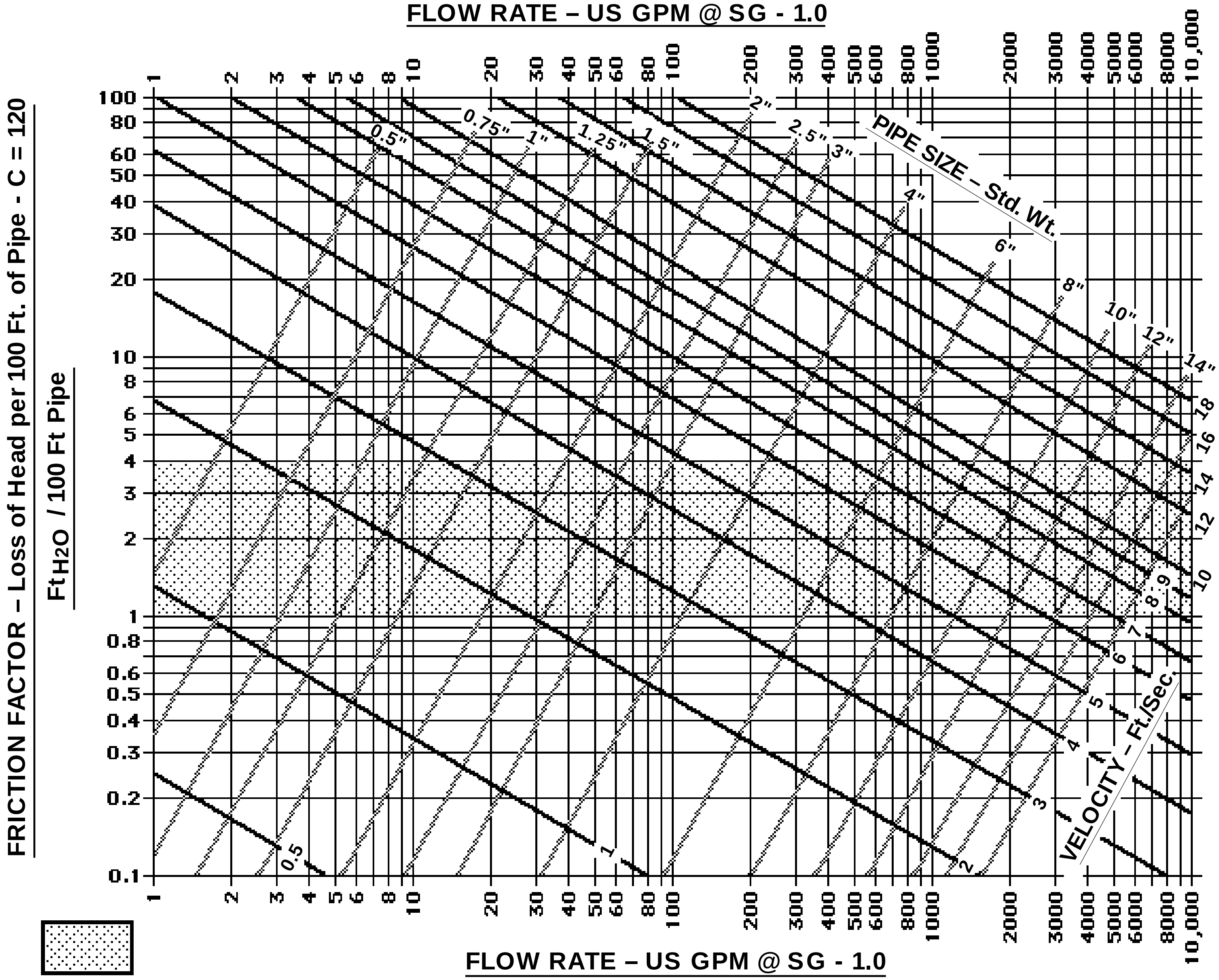
coefficient

<u>Example</u> –	2CO	+	O_2	$\longrightarrow$	$2Co_2$
	2 <u>moles</u> of carbon monoxide	+	1 <u>mole</u> of oxygen	yields	2 <u>moles</u> of carbon monoxide

MOLAR PRINCIPLE 6 <u>For a "balanced" chemical reaction formula, the expressed "coefficients" *can* be in "volumes" ONLY if the "after reaction" pressures and temperatures are brought to the same level</u>.

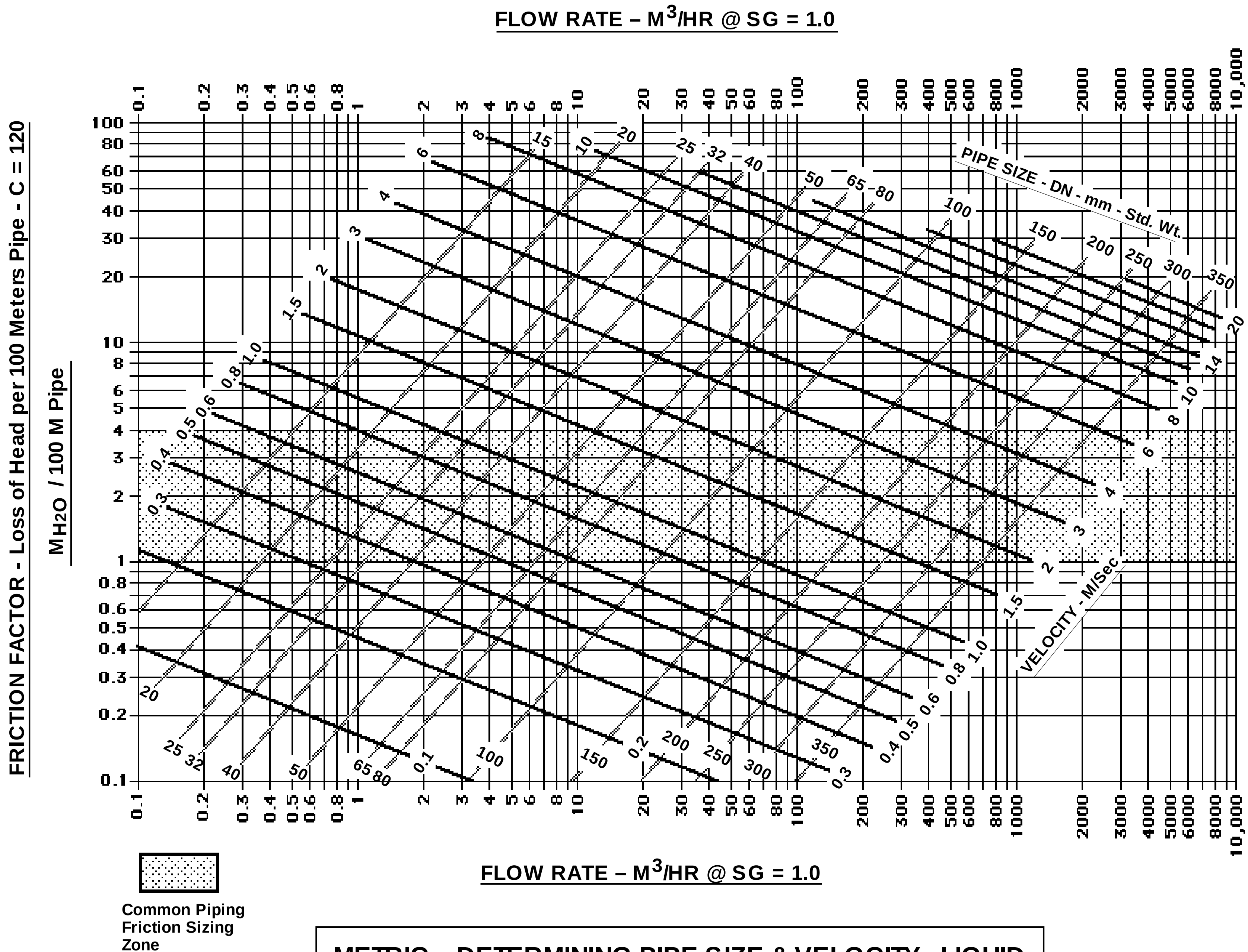
NOTES: As most chemical reactions are exothermic or endothermic, this approach is seldom used.

This principle is valid because 1 mole of gas does occupy 1 molar volume at standard conditions.

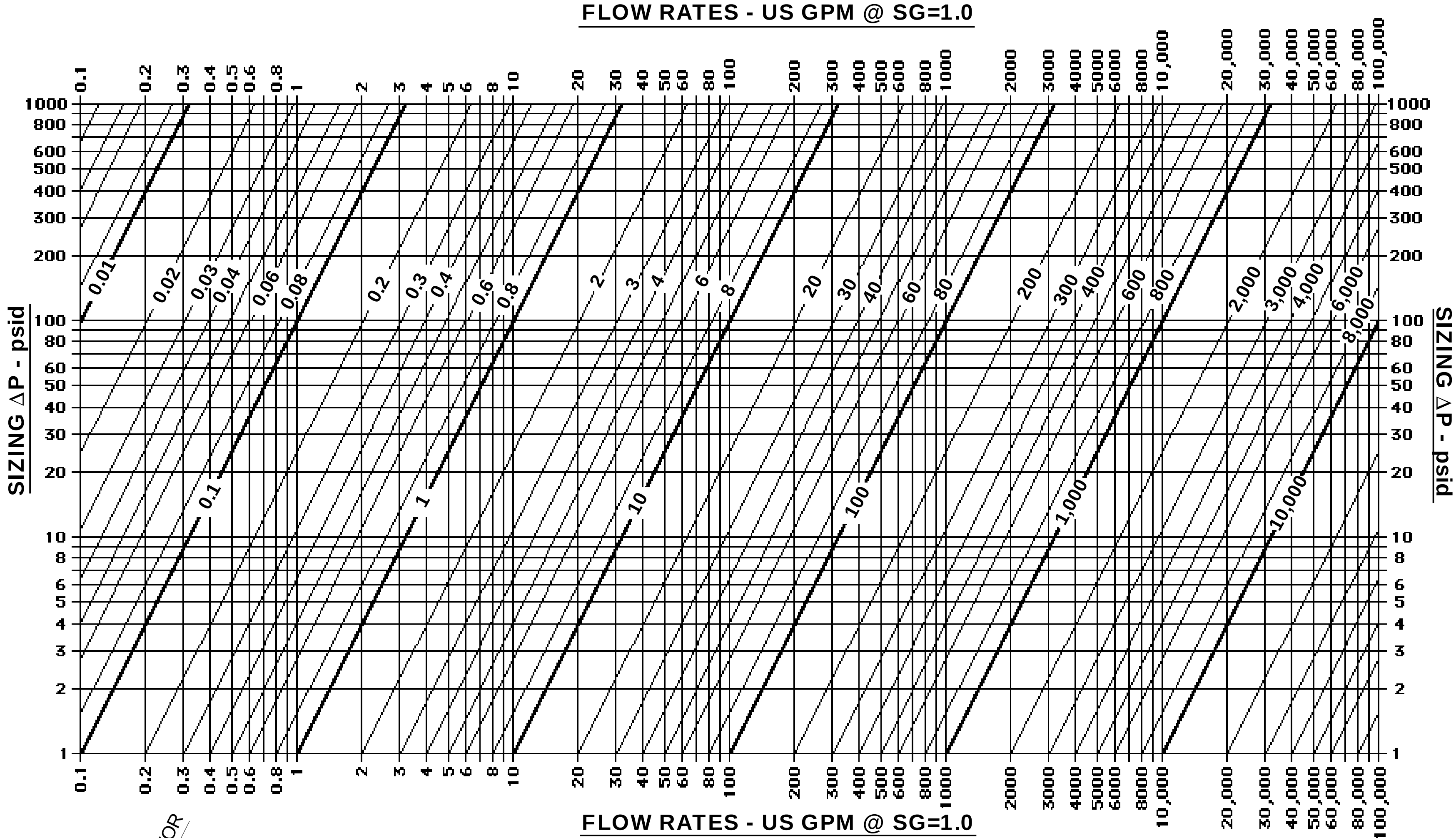


Common Piping Friction Sizing Zone

DETERMINING PIPE SIZE & VELOCITY – LIQUID



METRIC – DETERMINING PIPE SIZE & VELOCITY - LIQUID



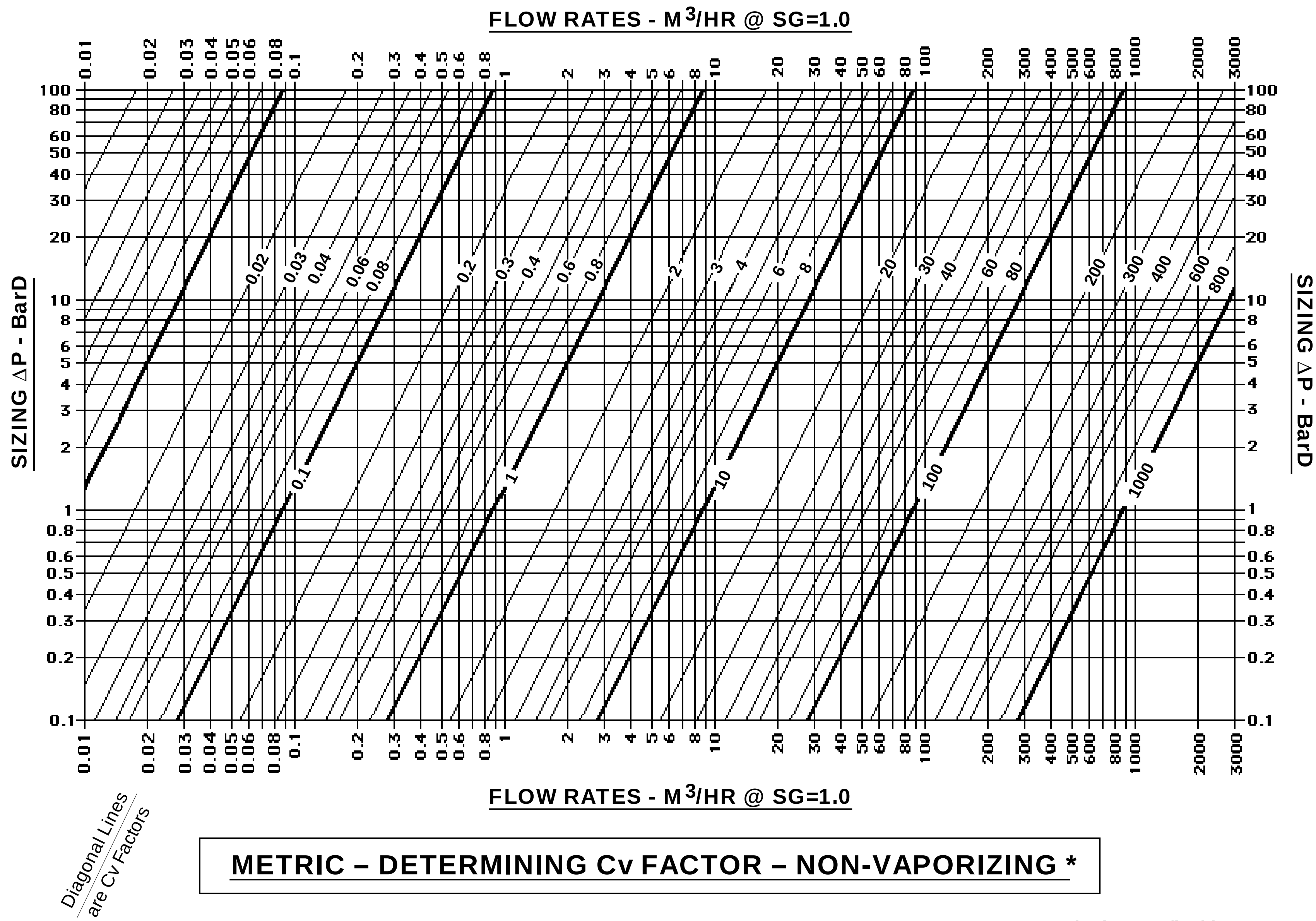
Diagonal Lines - Cv FACTOR

ENGLISH – DETERMINING Cv FACTOR – NON-VAPORIZING *

Example: $\dot{Q}$ = 200 GPM
P1 = 200 psig
P2 = 100 psig
ΔP - 100 psid

a. @ 200 GPM on x-axis, go vertically upward until crossing @ 100 psid on y-axis.
b. Determine by interpolating the diagonal exponential lines.
c. Cv = 20.

*** No cavitation; No flashing**

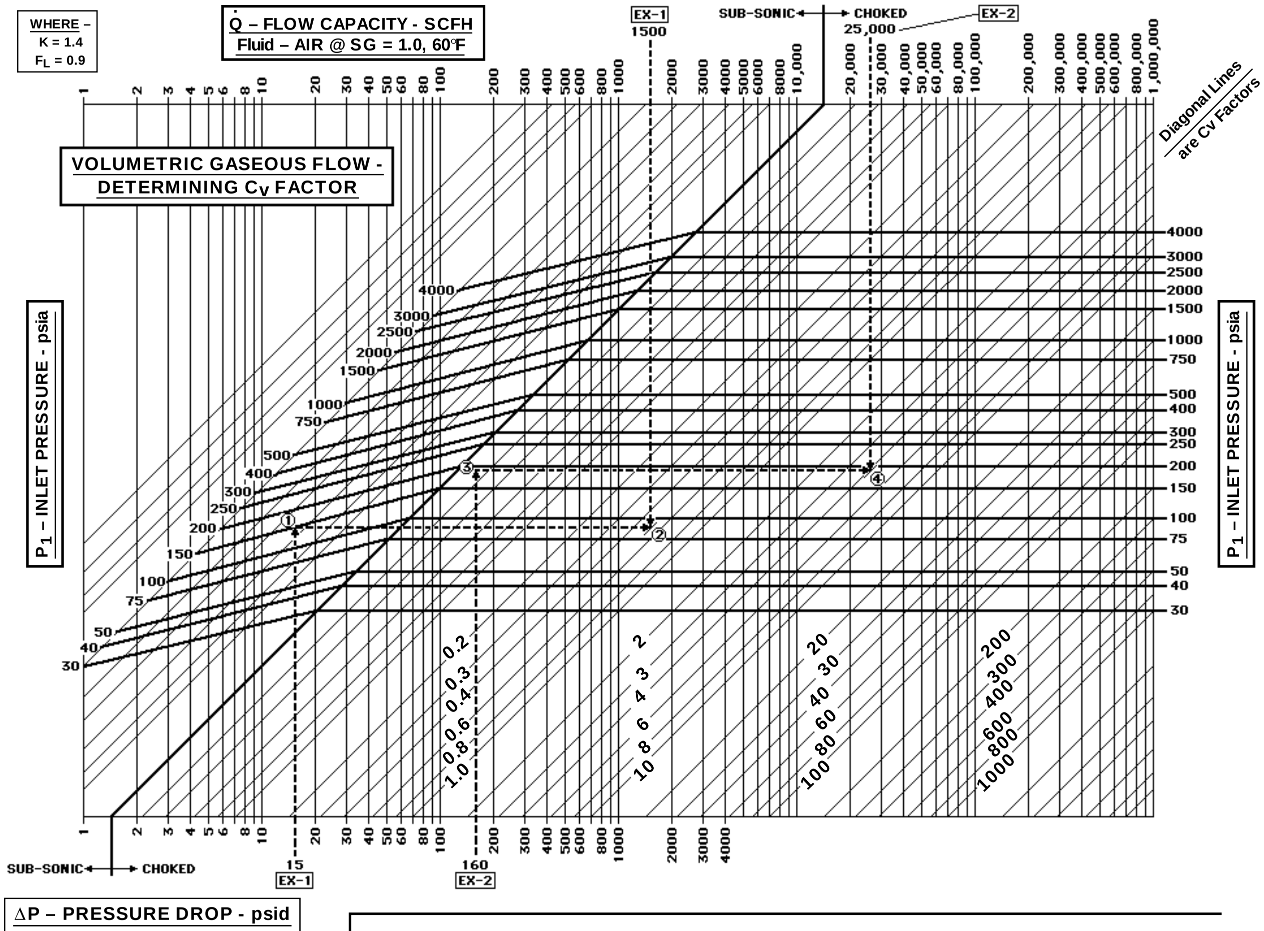


METRIC – DETERMINING Cv FACTOR – NON-VAPORIZING *

*** No cavitation; No flashing**

Example: $\dot{Q}$ = 10 M^3/Hr
P1 = 3.0 BarG
P2 = 1.0 BarG
ΔP - 2.0 BarD

a. @ 10 M^3/Hr on x-axis, go vertically upward until crossing @ 2 BarD on y-axis.
b. Determine by interpolating the diagonal exponential lines.
c. Cv ≈ 8.0.



EXAMPLE 1 -- EX-1

Fluid = AIR
P1 = 150 psia
P2 = 135 psia
ΔP = 15 psia
$\dot{Q}$ = 1500 SCFH

1. Locate 15 psid and draw a line vertically up until intersecting the 150 psia P1 line locating Point 1.
2. Draw a line horizontally from Point 1 to right.
3. Locate 1500 SCFH along top axis and draw a line vertically down until intersecting the horizontal line of Step 2 above at Point 2.
4. Read the inclined Cv Value. Cv ≈ 0.52.
5. Flow is sub-sonic; ie non-choked.
6. Computer gives 0.47 Cv.

EXAMPLE 2 -- EX-2

Fluid = AIR
P1 = 185 psia
P2 = 25 psia
ΔP = 160 psia
$\dot{Q}$ = 25,000 SCFH

1. Locate 160 psid and draw a line vertically up until intersecting the 185 psia P1 line locating Point 3.
2. Draw a line horizontally from Point 3 to right.
3. Locate 25,000 SCFH along top axis & draw a line vertically down until intersecting the horizontal line of Step 2 above at Point 4.
4. Read the inclined Cv value. Cv ≈ 4.2.
5. Flow is choked; i.e. sonic.
6. Computer gives 4.00 Cv.

DENSITY - SG - MW CORRECTION

$$\frac{\dot{Q}_{equiv}}{\dot{Q}_{gas}} = \sqrt{\frac{MW_{gas}}{28.95}} = \sqrt{\frac{\rho_{gas}}{.07622}} = \sqrt{\frac{SG_{gas}}{1.000}}$$

Where:
$\dot{Q}_{equiv}$ [=] SCFH
$\dot{Q}_{gas}$ [=] SCFH
ρ_{gas} [=] #/ft^3 @ Std. Cond.
MW_{gas} [=] #/#•mole

[Based on Std. Conditions of 60° F and 1 Atm.]

1. Choose the appropriate equation above that corresponds to the known relative mass units.
2. Solve the equation for $\dot{Q}_{equiv}$.
3. Use $\dot{Q}_{equiv}$ to enter the above graph to determine the Cv value.

EXAMPLE 3: $MW_{gas} = 40$, $\dot{Q}_{gas} = 25{,}000$ SCFH

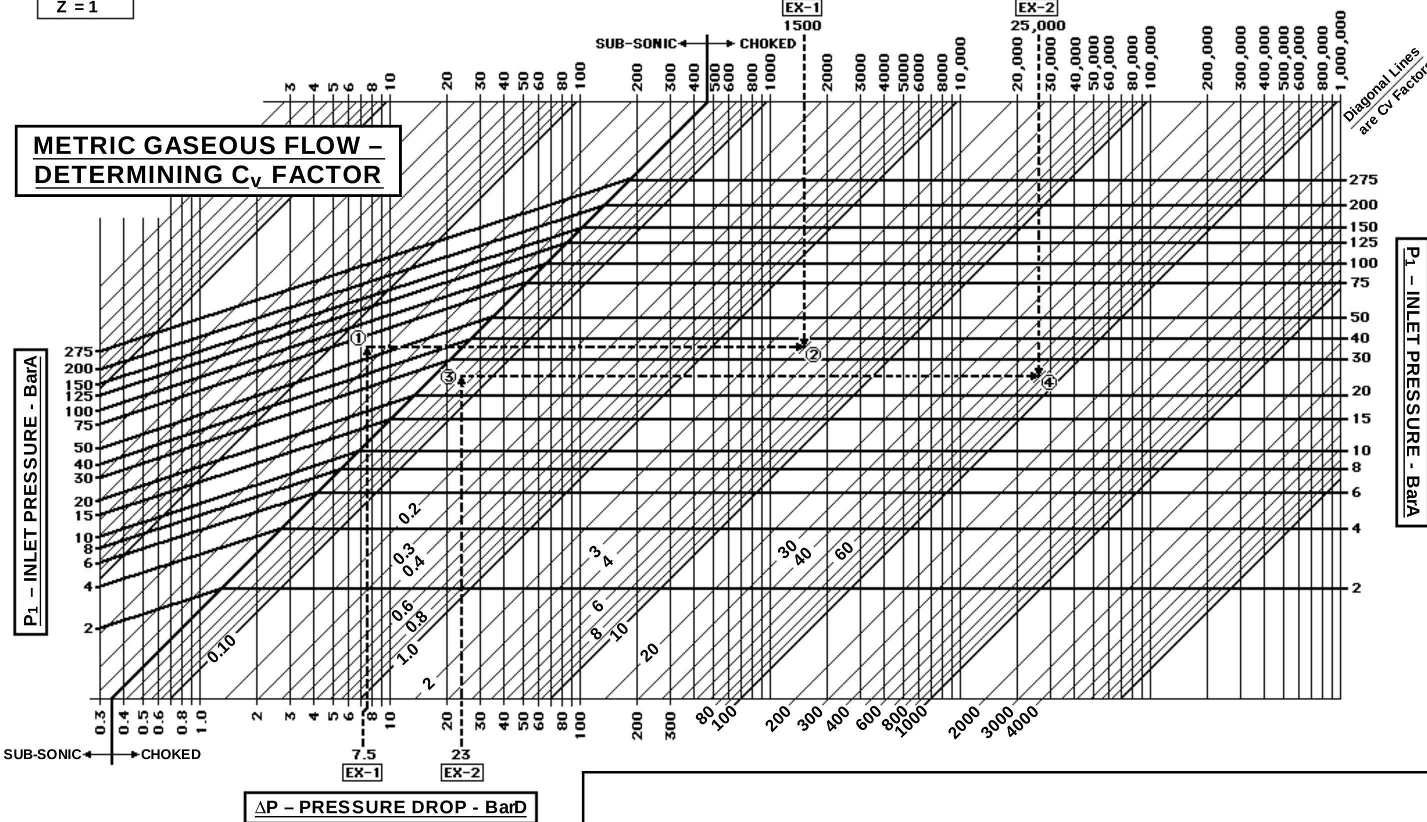
$$\frac{\dot{Q}_{equiv}}{\dot{Q}_{gas}} = \sqrt{\frac{MW_{gas}}{28.95}} \longrightarrow \dot{Q}_{equiv} = \dot{Q}_{gas}\sqrt{\frac{MW_{gas}}{28.95}}$$

$$\dot{Q}_{equiv} = 25{,}000 \times \sqrt{\frac{40}{28.95}} = 29{,}390 \text{ SCFH}$$

Use $\dot{Q}_{equiv}$ = 29,400 SCFH to enter graph above.

WHERE —
k = 1.4
F_L = 0.9
Z = 1

kv = 1.16 Cv



EXAMPLE 1 -- EX-1

Fluid = AIR
P1 = 60 BarA
P2 = 52.5 BarA
ΔP = 7.5 BarD
$\dot{Q}$ = 1500 SM3/Hr

1. Locate 7.5 BarD and draw a line vertically up until intersecting the 60 BarA P1 line locating Point (1).
2. Draw a line horizontally from Point (1) to right.
3. Locate 1500 SM3/Hr along top axis and draw a line vertically down until intersecting the horizontal line of Step 2 above at Point (2).
4. Read the inclined Cv Value. Cv ≈ 3.2
5. Flow is sub-sonic; ie non-choked.
6. Computer gives 3.1 Cv.

EXAMPLE 2 -- EX-2

Fluid = AIR
P1 = 25 BarA
P2 = 1 BarA
ΔP = 23 BarD
$\dot{Q}$ = 25,000 SM3/Hr

1. Locate 23 BarD and draw a line vertically up until intersecting the 25 BarA P1 line locating Point (3).
2. Draw a line horizontally from Point (3) to right.
3. Locate 25,000 SCFH along top axis & draw a line vertically down until intersecting the horizontal line of Step 2 above at Point (4).
4. Read the inclined Cv value. Cv ≈ 76.
5. Flow is choked; i.e. sonic.
6. Computer gives 74.2 Cv.

DENSITY - SG - MW CORRECTION

$$\frac{\dot{Q}_{equiv}}{\dot{Q}_{gas}} = \sqrt{\frac{MW_{gas}}{28.95}} = \sqrt{\frac{\rho_{gas}}{1.292}} = \sqrt{\frac{SG_{gas}}{1.000}}$$

Where: $\dot{Q}_{equiv}$ [=] SM^3/Hr
$\dot{Q}_{gas}$ [=] SM^3/Hr
ρ_{gas} [=] kg/M^3 @ MSC
MW_{gas} [=] kg/kg•mole

[Based on Metric Std. Conditions of 15° C and 1 Atm (MSC).]

1. Choose the appropriate equation above that corresponds to the known relative mass units.
2. Solve the equation for $\dot{Q}_{equiv}$.
3. Use $\dot{Q}_{equiv}$ to enter the above graph to determine the Cv value.

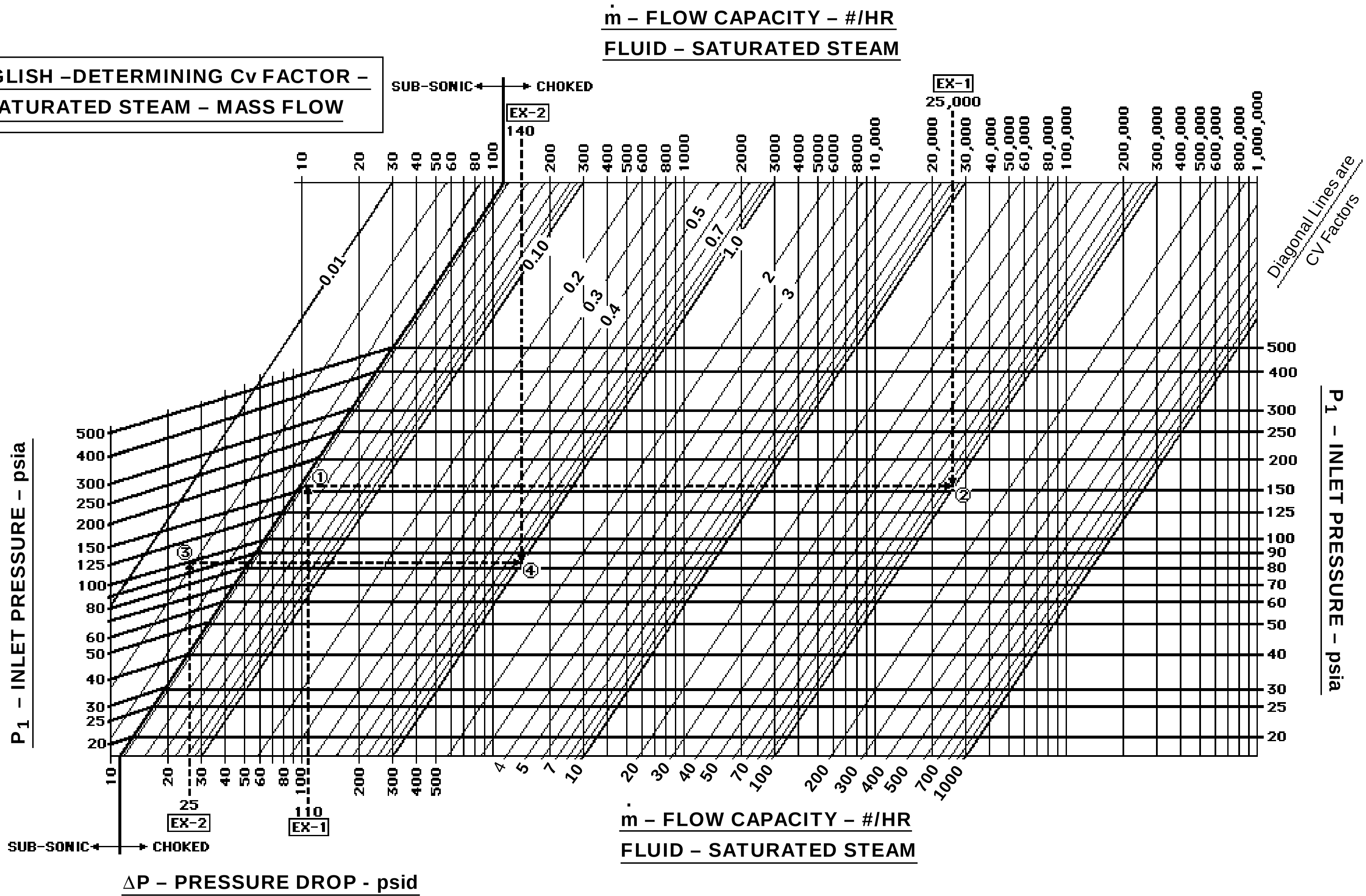
EXAMPLE 3: $MW_{gas} = 40$, $\dot{Q}_{gas} = 12{,}000\ M^3/Hr$

$$\frac{\dot{Q}_{equiv}}{\dot{Q}_{gas}} = \sqrt{\frac{MW_{gas}}{28.95}} \longrightarrow \dot{Q}_{equiv} = \dot{Q}_{gas}\sqrt{\frac{MW_{gas}}{28.95}}$$

$$\dot{Q}_{equiv} = 12{,}000 \times \sqrt{\frac{40}{28.95}} = 14{,}100\ M^3/Hr$$

Use $\dot{Q}_{equiv}$ = 14,100 to enter graph above.

ENGLISH –DETERMINING Cv FACTOR – SATURATED STEAM – MASS FLOW



EXAMPLE 1 -- EX-1

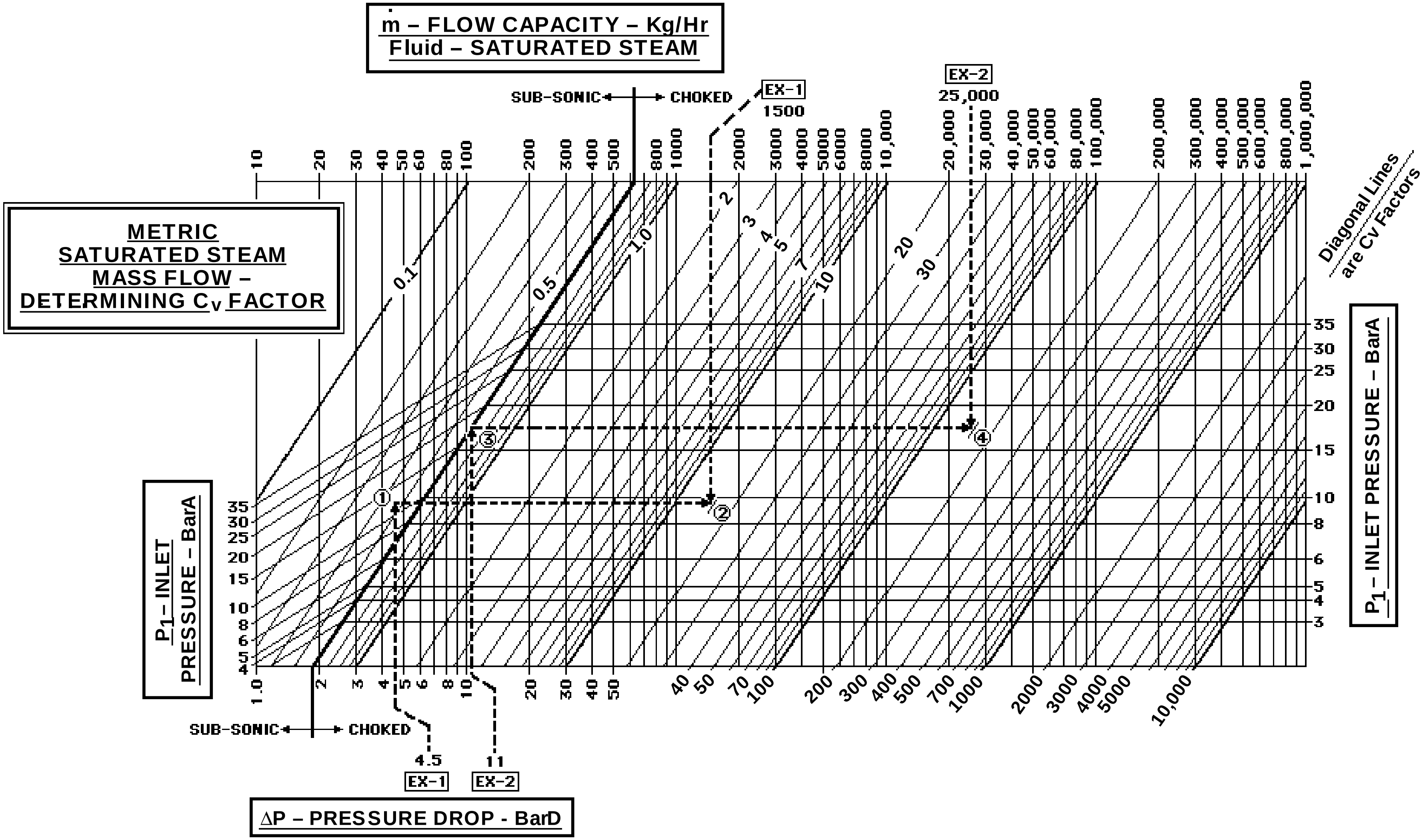
Fluid = SAT STM
P1 = 155 psia
P2 = 45 psia
ΔP = 110 psia
ṁ = 25,000 #/Hr
T1 = 361°F

1. Locate 110 psid and draw a line vertically up until intersecting the 155 psia P1 line locating Point ① .
2. Draw a line horizontally from Point ① to right.
3. Locate 25,000 #/Hr along top axis & draw a line vertically down until intersecting the horizontal line of Step 2 above at Point ② .
4. Read the inclined Cv value. Cv ≈ 100.
5. Flow is choked; i.e. sonic.
6. Computer gives 102.3 Cv.

EXAMPLE 2 -- EX-2

Fluid = SAT STM
P1 = 82 psia
P2 = 57 psia
ΔP = 25 psia
ṁ = 140 #/Hr
T1 = 314°F

1. Locate 25 psid and draw a line vertically up until intersecting the 82 psia P1 line locating Point ③ .
2. Draw a line horizontally from Point ③ to right.
3. Locate 140 #/Hr along top axis & draw a line vertically down until intersecting the horizontal line of Step 2 above at Point ④ .
4. Read the inclined Cv value. Cv ≈ 1.0.
5. Flow is choked; i.e. sonic.
6. Computer gives 1.22 Cv.



EXAMPLE 1 -- EX-1

Fluid = SAT STM
P1 = 9.5 BarA
P2 = 5 BarA
ΔP = 4.5 BarD
ṁ = 1500 Kg/Hr
T1 = 178°C

1. Locate 4.5 BarD and draw a line vertically up until intersecting the 9.5 BarA P1 line locating Point ①.
2. Draw a line horizontally from Point ① to right.
3. Locate 1500 Kg/Hr along top axis & draw a line vertically down until intersecting the horizontal line of Step 2 above at Point ②.
4. Read the inclined Cv value. Cv ≈ 16.
5. Flow is choked; i.e. sonic.
6. Computer gives 15.6 Cv.

EXAMPLE 2 -- EX-2

Fluid = SAT STM
P1 = 17 BarA
P2 = 6 BarA
ΔP = 11 BarD
ṁ = 25,000 Kg/Hr
T1 = 294°C

1. Locate 17 BarD and draw a line vertically up until intersecting the 17 BarA P1 line locating Point ③.
2. Draw a line horizontally from Point ③ to right.
3. Locate 25,000 Kg/Hr along top axis & draw a line vertically down until intersecting the horizontal line of Step 2 above at Point ④.
4. Read the inclined Cv value. Cv ≈ 150.
5. Flow is choked; i.e. sonic.
6. Computer gives 143.5 Cv.

VALVE SIZING COEFFICIENTS — C_v, C_g, and C_s.

A proprietary set of equations commonly found in throttling valve sizing uses a totally different empirical mathematical approach to calculate their C_g (gaseous service) and C_s (steam service) factors. The C_v factor used is identical to ISA formula's C_v factor.

Below are estimate "conversion factors" to relate the C_g and C_s factors to the more widely accepted C_v factor for ALL FLUIDS AND VALVES.

CONVERSION FACTORS :

C_v [=] Liquid	C_g [=] Gas	C_s [=] Steam
$C_v = 1.00$	$C_g = MF \times C_v$	$C_g = C_s \times 20$
	$MF \approx 28 - 38$	$C_s \approx 1.6 \times C_v$
	$C_g \approx 33 \times C_v$	

EXAMPLE : 2" (DN50) – Linear Characteristic, Globe-style Control Valve —

$$C_v = 72.9 \qquad C_g = 2330 \qquad C_s = 117$$

$$\frac{2330}{72.9} = 32.0 \approx 33 \qquad \frac{2330}{117} = 19.9 \approx 20$$

$$\frac{117}{72.9} = 1.61 \approx 1.6$$

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