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DWN	W.W.POWELL	4-17-75	CALCULATION OF FLOW LOSSES IN INLET AND DISCHARGE HEADERS ASSOCIATED WITH SAFETY RELIEF VALVES		
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1.0 ABSTRACT

During the study of resonant phenomena in pilot operated safety relief valves described in AGCO Report 2-0175-51, pressure differentials between the reservoir and valve inlet for different riser lengths were measured. Attempts to correlate these measured numbers with those predicted by equations intended for use in low velocity flow (such as Darcy and Weymouth) were not particularly successful.

Additionally, we at Anderson, Greenwood have for years been interested in obtaining a method of calculating the pressure existent at any point in discharge headers (tailpipes) or if the header has zero length, safety valve outlets. The problem has not been in obtaining a theoretical method but rather a theoretical method which gives good correlation with test results and is easily used.

We contacted Dr. Allan J. Chapman of the Rice University for assistance. The method recommended by Dr. Chapman for pressures which produces nozzle choking involves the use of Fanno Lines and is presented here for the use of the Engineer who must install Relief Valves, who desires to obtain satisfactory performance from these valves, and who, like the writer, has been away from school long enough to realize that he requires assistance. The approach is pratical and those interested in theory are referred to Dr. Chapmans excellent book "Introductory Gas Dynamics" or any of the many other available texts.

For subsonic Flow, a method is presented in Paragraph 5.0 which, while extremely simple, gives excellent results.

2.0 NOMENCLATURE

Le = Total Equivalent Length of Pipe (ft)

Ma = Local Mach No. (dimensionless)

W = Weight Flow Rate (lb/hr)

t = Temperature (°F).

T = Temperature (° Rankine)

k = Ratio of Specific Heats $\frac{C_p}{C_v}$ (dimensionless)

M = Molecular Weight of Lading Fluid = 28.964 x G

P = Pressure (psia)

L = Equivalent Length of Pipe (inches)

G = Specific Gravity of Lading Fluid (dimensionless)

D = Pipe Diameter (ft)

d = Pipe Diameter (in)

A_n = Area Valve Nozzle (in.sq.)

K_d = Actual Coefficient of Dishcharge of Valve (NOTE: Does not include 90% derating factor required by ASME Section VIII code).

C = Constant for Gas or Vapor (ASME VIII Figure UA230)

K = Resistance coefficient ($\frac{fL}{d}$)

f = Moody Friction Factor for Turbulent Flow in a Pipe (dimensionless)

V = Volumetric flow rate (SCFM)

SUBSCRIPTS

0 Stagnation State (where velocity is zero)

1 Entrance of Inlet Pipe

2 Exit of Inlet Pipe (or inlet of safety valve)

3 Nozzle Exit

4 Valve Exit

5 Discharge Header Exit

SUPERSCRIPTS

* Point of Choking in Adiabatic Flow (Ma = 1.0)

DEFINITIONS

Adiabatic - Without Loss or Gain of Heat.

Isentropic - Constant Entropy.

Entropy - A Quantity That is the Measure of the Amount of Energy in a System Not Available for Doing Work; Numerical Changes in the Quantity Being Determinable From the Ratio dQ/T Where dQ is a Small Increment of Heat Added or Removed and T is Absolute Temperature.

Fanno Lines - The locus of points describing the irreversible adiabatic flow of a gas at constant mass flow per unit area.

3.0 INLET HEADERS

3.1 Limitations

The method as presented in Paragraphs 3.0 and 4.0 is limited to those valves having set pressures that produce critical or choked (Ma = 1.0) flow at the valve nozzle during the relief cycles. Subsonic valves and headers are covered in Paragraph 5.0. The examples will evaluate only static and stagnation pressure. Temperature may also be determined by Fanno Lines but will not be included in this report.

3.2 Header Evaluation

Evaluation of either inlet or discharge headers is accomplished by the well accepted method of Equivalent Lengths as outlined in Crane Technical Paper 410 and others. Factors which will influence flow such as entrances, elbows, tees, valves, etc. are expressed as Equivalent Lengths of Pipe. The Equivalent Length of all these pipe components is added to the actual length of straight pipe to obtain the Total Equivalent Length (Le) of the pipe. Some typical Equivalent Lengths are listed in Figures I and II. The resistance coefficient $K = \frac{f\,(Le)}{d}$ is then determined using Figure III to obtain f.

3.3 Condition One - Nozzle Area Same As Inlet Pipe Area

(A length of pipe only with no valve)

The pressure drop due to adiabatic frictional flow in the inlet pipe is calculated by assuming choked flow (Ma = 1.0) at the pipe (or nozzle) exit and by assuming that the inlet of the valve to the nozzle face (if a valve is used) is simply an extension of the pipe. This length is normally very short and practically can be neglected. The Mach No. at the exit may be checked from the equation:

$$Ma = \frac{.00245\,W}{P_2 d^2}\sqrt{\frac{To}{kM}} \qquad \text{(Note 1)}$$

Where: P_2 = Static pressure at exit, psia

T_0 = Stagnation temperature at inlet

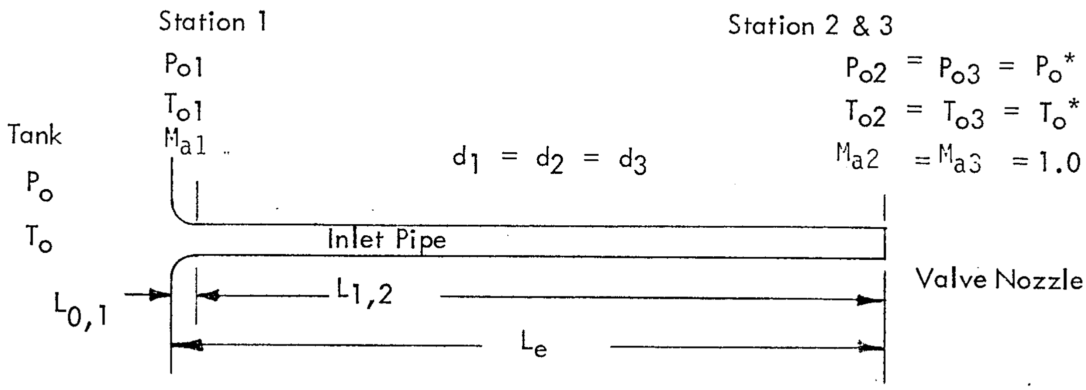
If the calculated Mach No. is greater than 1.0 (which cannot happen physically due to choking) the exit pressure will readjust itself so that M_{a2} will equal 1.0 Where the calculated Mach No. is equal to or greater than 1.0 this method is applicable. By knowing the Mach No. at the exit, stagnation pressure (Po) in the tank and properly evaluating the inlet pipe, fluid properties can be evaluated at any point in the pipe.

3.3.1 Example

Anderson, Greenwood Type 273, 3 x 44, Set Pressure 490 psig (505 psia). Header is 15ft (180 in.) of 3 inch schedule 80 pipe (2.9 inch I.D.). Header to tank connection is a 12 inch to 3 inch concentric reducer (L entry = 0 from Figure II) Lading Fluid is natural gas with k = 1.3, G = .60, t_o= 60°F (520°R).

This condition is a comparatively rare instance and will exist only with full bore pipeline valves, such as the AGCO Series 70, USI Type D and some Farriservo Valves. On these valves, the inlet piping should either be zero length (valve mounted directly on tank) or one pipe size larger and swaged down at the valve to obtain satisfactory performance. Note that the pressure drop between the tank (1) and valve (2) is the change in stagnation pressures.

3.3.2 Solution - Nozzle Same Area As Inlet Pipe Area



1) Known Parameters

Set Pressure,	P_{o1}	= 505 psia
Ratio Specific Heat,	k	= 1.3
Temperature,	t_o	= 60°F
Specific Gravity,	G	= 0.60
Moleclular Weight,	$M = 28.964(G)$	= 17.38
Inlet Pipe Diameter,	$d_1 = d_2 = d_3$	= 2.9 in.
Inlet Pipe Length,	$L_{1,2}$	= 180 in.

2) Header Evaluation

Header Section	L/d	L	K = fL/d
Entrance	0	0	0
Pipe	62.07	180	1.67
Fittings	0	0	0
Total	62.07	180	1.67

Where: $f = .027$ (Figure III)

$$L/d = \frac{180}{2.9} = 62.07$$

$$K = fL/d = .027\,(62.07) = 1.67$$

3) From Figure IV, The Mach No. At The Inlet For $k = 1.3$ and $K = 1.67$, is: $Ma_1 = .455$

4) From Figure V, The Stagnation Pressure Ratio For $k = 1.3$ and $Ma_1 = .455$ is: $\frac{P_{o1}}{P_o*} = 1.42$

Therefore, the stagnation pressure at the exit (Station 2)

is: $$P_o* = \frac{P_{o1}}{1.42} = \frac{505}{1.42} = 355.63$$

5) The Inlet Pressure Loss is: $P_{o1} - P_o* = 505 - 355.63 = 149.37$ psi

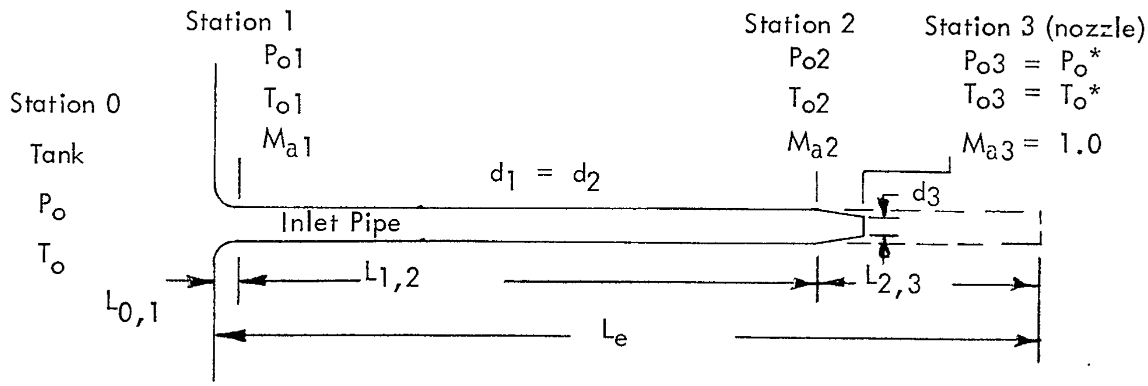
3.4 Condition Two - Nozzle Area Smaller Than Inlet Pipe Area

This condition is far more common than Condition One and is found in all "Nozzle" Valves (D thru T Orifice - API 526) or, as in this instance, where the inlet header is one pipe size larger than the valve. The pressure drop due to adiabatic frictional flow in the pipe is calculated by assuming choked flow (sonic) at the nozzle exit as in Condition One and isentropic flow in the nozzle. Knowing the Mach No. ($M_{a3}^* = 1.0$) at the nozzle exit and the effective area ratio A2/A3, the Mach No. (M_{a2}) at the valve entrance is found from Figure VII. Note that the actual nozzle area of the valve (A_n) is multiplied by the Discharge Coefficient K_d to obtain A3. Utilizing the Mach No. (M_{a2}) just found, the Friction Length Parameter $\frac{fL}{d}$ for a theoretical pipe length which would be required to accelerate the flow from M_{a2} to $M_{a3} = 1.0$ is found from Figure IV. This theoretical length replaces the nozzle. The Friction Length Parameter of the inlet pipe $\frac{fL_{1,2}}{d_1}$ is added to the theoretical expression $\frac{fL_{2,3}}{d_1}$ to obtain $\frac{fL_e}{d_1}$. Conditions at any point in the pipe (Such as at Station 2) are then readily determined.

3.4.1 Example

Same as 3.3.1 except that inlet header size has been changed from 3 inch to 4 inch (3.9 inch Inside Diameter) The actual coefficient of discharge K_d = .809/.9 = .90. If desired, Temperatures and/or Static Pressures may be simularly found in the same manner from the appropriate curves which are included. Additionally, now that the actual conditions at the valve inlet during flow are known, the capacity which the valve will deliver can be readily determined from standard sizing formulae.

3.4.2 Solution – Nozzle Area Smaller Than Area of Inlet Pipe



1) Known Parameters

Set Pressure,	P_{o1}	= 505 psia
Ratio Specific Heat,	k	= 1.3
Temperature,	t_o	= 60°F
Specific Cravity,	G	= 0.60
Molecular Weight,	M = 28.964(G)	= 17.38
Inlet Pipe Diameter,	$d_1 = d_2$	= 3.9 in.
Valve Nozzle Diameter,	d_3	= 2.9 in.
Valve Coefficient of Discharge	K_d	= 0.90
Inlet Pipe Length,	$L_{1,2}$	= 180 in.

2) Header Evaluation

Header Section	L/d	L	K = fL/d
Entrance	0	0	0
Pipe	46.15	180	1.15
Fittings	0	0	0
Total	46.15	180	1.15

Where: f = .025 (Figure III)

$$L/d = \frac{180}{3.9} = 46.15$$

$$K = fL/d = .025\,(46.15) = 1.15$$

3) The Isentropic Area Ratio of the Valve Inlet to the Valve Nozzle is: $\frac{A2}{A3} = \frac{(d_2)^2}{(d_3)^2 K_d} = \frac{(3.9)^2}{(2.9)^2(.90)} = \underline{2.00}$

From Figure VII, the Mach No. at Station 2 (M_{a2}) = $\underline{0.31}$ for k = 1.3 and Isentropic Area Ratio of 2.00.

4) From Figure IV, the Friction Length Parameter (fL/d) of a theoretical pipe length required to accelerate the flow from $M_{a2} = 0.31$ to $M_{a3} = 1.00$ is $\underline{5.25}$ for k = 1.3.

5) From Figure V, the Stagnation Pressure Ratio $\frac{P_{o2}}{P_{o*}}$ is $\underline{2.00}$ for k = 1.3 and a Mach No. (M_{a2}) of 0.31 as determined in Step 3.

6) The Total Friction Length Parameter (fL/d) of the actual plus theoretical pipe is $\frac{fL_{1,2}}{d_1} + \frac{fL_{2,3}}{d_2} = 1.15 + 5.25 = \underline{6.40}$

$\frac{fL_{1,2}}{d_1}$ was determined in Step 2, $\frac{fL_{2,3}}{d_2}$ was determined in Step 4.

7) From Figure IV, the Mach No. at Station 1 (M_{a1}) is $\underline{0.29}$ for a Friction Length Parameter of 6.40.

8) From Figure V, the Stagnation Pressure Ratio $\frac{P_{o1}}{P_o*}$ is 2.13 for k = 1.3 and a Mach No. (M_{a1}) of 0.29 as determined in Step 7.

9) The Stagnation Pressure at Station 3 (the exit) is:

$$P_o* = \frac{P_{o1}}{2.13} = \frac{505}{2.13} = \underline{237} \text{ psia}$$

10) The Stagnation Pressure at Station 2 (the valve inlet) is $P_{o2} = 2.00\ P_o^*$ from Step 5. Using $P_o^* = 23$ as determined in Step 9, $P_{o2} = 237\,(2.00) = \underline{474}$ psia

11) The Inlet Pressure Loss is: $P_{o1} - P_{o2} = 505 - 474 = \underline{31}$ psi

4.0 DISCHARGE HEADERS (TAILPIPES)

4.1 Theory

The approach used in Discharge Headers is different from that used on Inlet Headers. The Inlet Header calculation described in Paragraph 3.0 uses two known conditions. These are 1) Tank Stagnation Pressure and 2) Mach No. at the valve nozzle (Ma = 1.0). Neither of these conditions is known to the engineer when designing discharge headers. As presented here, one must first calculate the weight flow rate of the safety valve from the normal valve sizing formula using the actual inlet condition to the valve as determined in Section 3.0. Note that again the actual coefficient of the valve K_d is used.

Knowing the Weight Flow Rate, W and applying the equation from Paragraph 3.3 the Exit Mach No. (Ma_5) can be determined by assuming P = absolute atmospheric pressure. T_o is the inlet stagnation temperature.

If the expression yields a Mach No. equal to one or less, the assumed exit pressure was correct. If the Mach No. calculates to be greater than one, the assumed exit pressure was incorrect. The correct Static exit pressure may be determined by letting Ma_5 = 1.0. The exit pressure may also be calculated by multiplying the assumed pressure used in the first trial by the calculated Mach No. We therefore know the Exit Mach No. (Ma_5) and the Exit Static Pressure (P_5). See Paragraph 4.3 for Ma_5 less than 1.0.

Knowing these conditions, Fanno Lines can be used to determine the static pressures anywhere in the discharge header. If Stagnation pressures are desired, they may be readily obtained using the Mach No. at the Station in question and calculating the Stagnation Pressure using Figure VIII, which is a plot of

$$P_o = P\left[1+\left(\frac{k-1}{2}\right)Ma^2\right]^{k/(k-1)} \qquad \text{(Note 2)}$$

Considering the effect of back pressure on a theoretically perfect nozzle in a duct, critical (choked) flow will exist until the ratio of static pressure/stagnation pressure is $\cong$ 0.5, depending on the gas. Chapman states that for subsonic flow, the exit plane static pressure must equal the imposed static back pressure, thus the static back pressure has an effect on the flow rate through the passage. In the classic equations for sonic or subsonic flow, the test for sonic flow is the ratio of static downstream pressure/stagnation pressure in the valve nozzle and the ΔP used in calculating flow rate in subsonic flow is the stagnation pressure in the nozzle throat minus the static downstream pressure.

4.2 Condition One - Theoretical Mach No. at Discharge Header Outlet Equal to or Greater Than One

4.2.1 Example

Anderson, Greenwood Type 2J3, Series 100, Set Pressure is 175 psia. Valve nozzle diameter is 1.347 inches and nozzle coefficient is .975. Discharge header is 29.5 inches of 3 inch Schedule 40 pipe (3.06 inch I.D.). Lading fluid is natural gas with $k = 1.3$, $G = 0.60$, $C = 345$ and $t_0 = 45°F$.

4.2.2 Solution – Discharge Header Exit Mach No., M_{c5} = 1.0

Station 4

M_{a4}

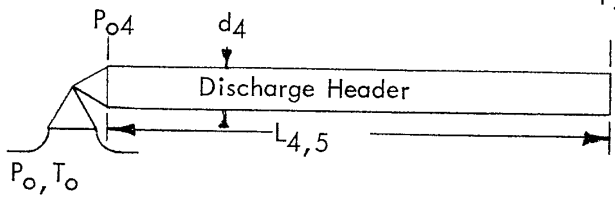
P_4

P_{o4}

Station 5

$M_{a5} \geqq 1.0$

$P_5 = P^*$ 14.7 psia (assumed)



NOTE: If valve discharges into a constant pressure header, use known pressure.

1) Known Parameters

Set Pressure,	P_o		= 175 psig
Ratio Specific Heat,	k		= 1.3
Temperature,	t_o	= 45°F	= 505°R
Specific Gravity,	G		= 0.60
Molecular Weight,	M	= 28.964(G)	= 17.38
Gas Constant,	C		= 345
Valve Nozzle Diameter,	d_3		= 1.347 in.
Valve Coefficient of Discharge,		K_d	= 0.975
Discharge Header Diameter,		d_4	= 3.06 in.
Discharge Header Length,		$L_{4,5}$	= 29.5 in.

2) Header Evaluation

Header Section	L/d	L	K = fL/d
Pipe	9.6	29	.24
Fittings	0	0	0
Total	9.6	29	.24

Where: f = .025 (Figure III)

$$L/d = \frac{29.5}{3.06} = 9.6$$

$$K = fL/d = .025\,(9.6) = .24$$

3) Capacity,

$$W = \frac{CK_d(d_3)^2\pi \quad [P_0\ (1.1) + 14.7]}{4} \frac{\sqrt{M}}{\sqrt{T_0}} \quad (\text{lb/hr})$$

$$W = \frac{345(.975)(1.34)^2\pi[175(1.1) + 14.7]}{4} \frac{\sqrt{17.38}}{\sqrt{505}}$$

$$W = 18425 \text{ lb/hr}$$

4) Mach No. $Ma_5 = \dfrac{.00245\ W}{P_5 d_4^{\,2}} \sqrt{\dfrac{T_0}{kM}}$

Assuming P_5 = 14.7 psia,

$$Ma_5 = \frac{.00245(18425)}{(14.7)(3.06)^2} \sqrt{\frac{505}{(1.3)(17.38)}}$$

$$Ma_5 = 1.550$$

Since the Mach No., Ma_5 is greater than one, the actual Static exit pressure,

$$P_5 = Ma_5\ (14.7) = 1.550\ (14.7)$$

$$= 22.79 \text{ psia}$$

5) From Figure IV, the Mach No. at the inlet to the discharge header (valve outlet) fork = 1.3 and K .24 is:

$$Ma_4 = .695$$

6) From Figure VI, the Static Pressure Ratio at the inlet to the outlet of the header for k = 1.3 and Ma_4 = .695 is:

$$\frac{P_4}{P*} = 1.46$$

Therefore, Static Pressure at valve outlet (discharge header inlet),

$P_4 = P^* (1.46) = P_5 (1.46) = 22.79 (1.46)$

P_4 = 33.24 psia = 18.54 psig†

7) From Figure VIII, the ratio of Stagnation Pressure/Static Pressure at the valve outlet (discharge header inlet) for k = 1.3 and Ma_4 = .695 Is:

$$\frac{P_{o4}}{P_4} = 1.35$$

Therefore, the Stagnation Pressure at the valve outlet

$P_{o4} = 1.35\ P_4 = 1.35\ (33.15)$ = 44.75 psia = 30 psig

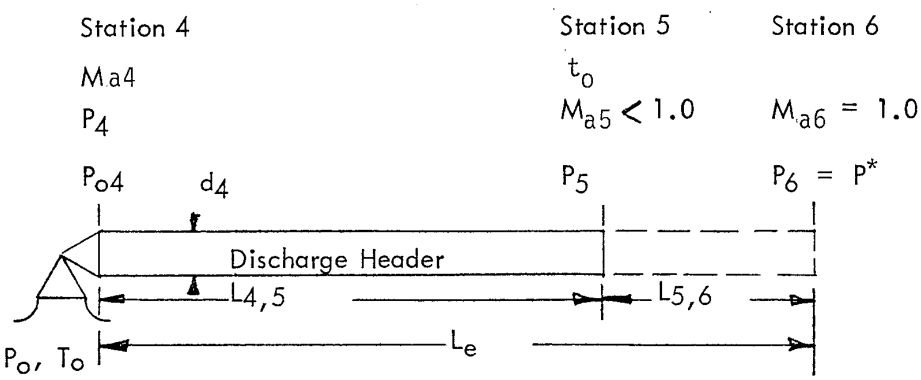
4.3 Condition Two - Theoretical Mach No. at Discharge Header Outlet Less Than 1.0.

4.3.1 Example

Anderson, Greenwood Type 2J3, Series 100 Set Pressure is 15 psig, valve nozzle diameter is 1.347 in., nozzle coefficient is .975. Discharge header is 29.5 inches of natural gas with k = 1.3, G = 0.60, C = 345 and t_o 45°F.

† The gauge value of the discharge header inlet static pressure (P4) is used in establishing if sufficient tailpipe induced built-up back pressure could occur such that the pressure rating of the outlet connection on the pressure relief valve would be exceeded.

4.3.2 Solution - Discharge Header Exit Mach No. $M_{c5} < 1.0$



1) Known Parameters

Set Pressure,	P_o	= 15 psig
Ratio Specific Heat,	k	= 1.3
Temperature,	t_o	= 45°F
Specific Gravity,	G	= 0.60
Molecular Weight,	M = 28.964(G)	= 17.38
Gas Constant,	C	= 345
Valve Nozzle Diameter,	d_3	= 1.347 in.
Valve Coefficient of Discharge,	K_d	= .975
Discharge Header Diameter,	d_4	= 3.06 in.
Discharge Header Length,	$L_{4,5}$	= 29.5 in.

2) Header Evaluation

Header Section	L/d	L	K = fL/d
Pipe	9.6	29.5	.24
Fittings	0	0	0
Total	0	0	.24

Where: f = .025 (Figure III)

$$L/d = \frac{29.5}{3.06} = 9.6$$

$$K = fL/d = .025\,(9.6) = .24$$

3) Capacity, $W = \dfrac{CK_d(d_3)^2\pi \quad [P_0\ (1.1) + 14.7]}{4} \sqrt{\dfrac{M}{T_0}}$ (lb/hr)

$$W = \frac{345(.975)(1.34)^2\pi[15(1.1) + 14.7]}{4} \sqrt{\frac{17.38}{505}}$$

$$W = 2774.5 \text{ lb/hr}$$

4) Mach No. $Ma_5 = \dfrac{.00245\ W}{P_5 d_4^{\,2}} \sqrt{\dfrac{T_0}{kM}}$

Assuming P_5 = 14.7 psia,

$$Ma_5 = \frac{.00245(2774.5)}{(14.7)(3.06)^2} \sqrt{\frac{505}{(1.3)(17.38)}}$$

$$Ma_5 = .233$$

Since the Mach No., Ma_5 is less than one the assumption that P_5, the Static exit pressure was 14.7 psia is correct.

5) From Figure IV, the Friction Length Parameter of a theoretical pipe attached to the header exit in which the flow is accelerated to Mach one at its exit for k = 1.3 and Ma_5 = .233

$$K = 11.0$$

6) From Figure VI, the Static Pressure Ratio from the inlet to the outlet of the theoretical pipe exit for k = 1.3 and Ma_5 = .233

$$\frac{P_5}{P^*} = 4.5$$

7) The Friction Length Parameter of Actual Header plus theoretical extension is $K = \dfrac{fL_{4\ 5}}{d_4} + \dfrac{fL_{5\ 6}}{d_4} = .24 + 11.0 = 11.24$

8) From Figure IV, the Mach No., Ma_4, at the inlet of the discharge header (valve outlet) for k= 1.3 and K = 11.24 is:

$$Ma_4 = .230$$

9) From Figure VI, the Static Pressure Ratio from the inlet of the actual header to the outlet of the theoretical extension for K = 1.3 and M_a 4 = .230 is

$$\frac{P_4}{P^*} = 4.55$$

10) The Static Pressure at the exit of the theoretical pipe extension is:

$$p^* = \frac{P_5}{4.5} = \frac{14.7}{4.5} = 3.266 \text{ psia}$$

11) The Static Pressure at the actual header inlet (valve outlet) is:

P4 = 4.55 P* = 4.55 (3.266) = 14.86 psia

= .16 psig

12) From Figure VIII, the ratio of Stagnation pressure to Static pressure at Station 4 for k = 1.3 and M_a = .230 is:

$$\frac{Po^4}{P_4} = 1.15$$

Po4 = 1.15 (14.86)

= 17.09 psia

2.39 psig

5.0 <u>SUBSONIC FLOW</u>

5.1 Theory

In subsonic flow all valves (including the safety valve), pipe, and fittings in both the inlet and discharge headers affect the flow rate through the system (since choked flow does not exist). The approach presented here is somewhat unconventional in that after properly evaluating the equivalent lengths of all involved plumbing, the pressure loss is assumed to be linear between the static tank pressure and the static pressure into which the system is discharging (normally atmosphere). The pressure at any desired point may now be found by pressure-length ratio. Knowing the pressures existant at the valve inlet and outlet, the capacity of the valve is readily calculated by appropriate formula and the pressure drop between the valve and tank is determined by simple subtraction.

This method is applicable only to installations where the tank pressure does not exceed 15 psig and where the Mach No. (Ma_5) at Station 5, discharge header exit, does not exceed 1.0 The Mach No., Ma_5, may be determined using the equation from paragraph 3.3, where d is the diameter of the <u>inlet pipe</u> (d_1, 2) and P is the absolute pressure at the discharge header exit. The inlet pipe ID must be used since the actual discharge header, being normally larger than the inlet pipe is replaced by its equivalent length of inlet pipe in the calculations. The Mach No. (Ma_5) in this equivalent pipe must be lower than 1.0 for the method to be applicable.

This approach to subsonic headers has been verified by extensive tests at the Anderson, Greenwood ASME accepted laboratory in El Campo and agreement between measured and calculated results are within +/-2%.

Another approach for calculating frictional piping losses is to calculate the resistance coefficient K for all pipe and fittings, adding them and calculating the flow using Darcy's equation.

$$V = \frac{40700}{60} \gamma \, d1^2 \frac{\sqrt{\Delta P \, P_1}}{\sqrt{K T_1 G}} \qquad \text{(Note 5)}$$

Where V = Volumetric flow, SCFM

γ = Expansion factor for the calculated K, ΔP and P_1

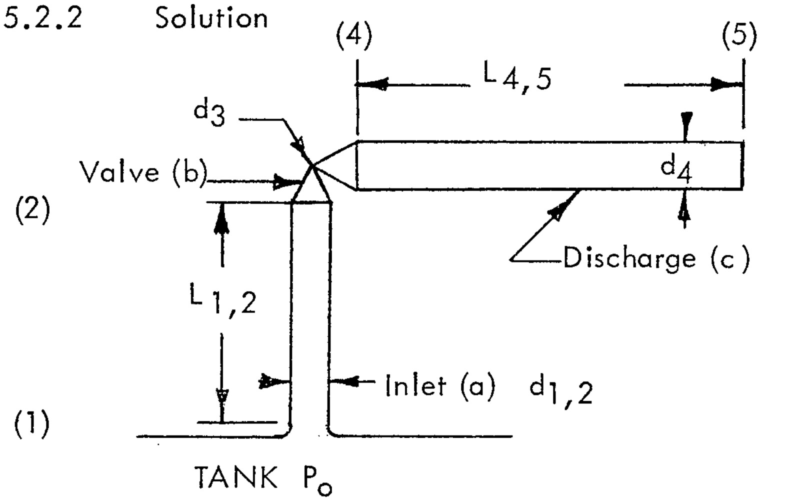
Figure X contains the resistance coefficient for AGCO Series 90 POSRV's. The expansion factor for air for various K's. ΔP's and P_1's is contained in Figure XI.

5.2 Method One - Equivalent L/D of Pipe, Fittings And Valve

5.2.1 Example

Anderson, Greenwood 2 x 3 Type 93, set pressure 5.0 psig. L/D from Figure X is 79, Inlet Header is 45.0 inches, 2 inch schedule 40 steel pipe (2.067 in ID), Discharge Header 91 inches, 3 inch schedule 40 pipe (3.068 ID). Lading Fluid is natural gas with k = 1.3, G = .60, T_o = 60°F. Valve is assumed to have remote pickup so that it is fully open at set and stays fully open irrespective of inlet pressure loss.

5.2.2 Solution



1) <u>Known Parameters</u>

Set Pressure,	P_o	= 5.0 psig
Ratio Specific Heat,	k	= 1.3
Temperature,	t_0 = 60°F	= 520°R
Specific Gravity,	G	= .60
Molecular Weight, M	= 28.964 (G)	= 17.38
Equivalent Length,	L/D	= 79
Inlet Header Length,	$L_{1,2}$	= 45.0 inches
Inlet Header Diameter,	$d_{1,2}$	= 2.067 inches
Discharge Header Length,	$L_{4,5}$	= 91.0 inches
Discharge Header Diameter,	$d_{4,5}$	= 3.068 inches
Coefficient of Discharge	$K_d = \frac{K}{.9} \quad \frac{.845}{.9}$	= .939
Header Exit Pressure	P_5	= 0 psig
Valve Orifice Area	A	= 2.29 in^2

2) System Evaluation

Section	L/d	L
Inlet Header	21.8	45.0
Valve	129 (a)	267
Discharge Header	6.1	12.63 (b)
Inlet	0	0
Total		324.63

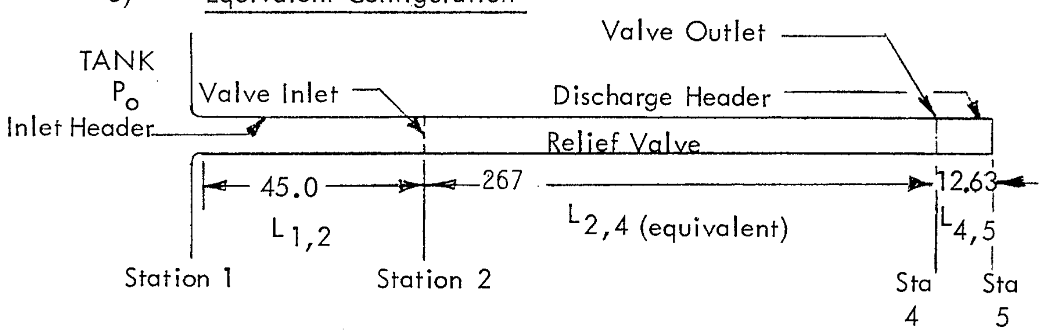
(a) From Figure X

(b) Discharge header length = 91.0 inches of 3.068 I D pipe. Since the equivalent length given in terms of pipe diameter, is a function of the fourth power of the pipe diameter,

$$\frac{L}{D} = \left(\frac{91.0}{3.068}\right)\left(\frac{2.067}{3.068}\right)^4 = 6.111$$

$$L = 2.067\,(6.111) = 12.63 \text{ in.}$$

3) Equivalent Configuration



4) Pressure Calculation Stations 1, 2, 4, & 5

$$P_1 = P_o = 5.0 \text{ psig}$$

$$P_5 = P_{atmos.} = 0 \text{ psig}$$

$$P_2 = 5.0 - 5.0\,\frac{(45.0)}{(324.63)} = 4.307 \text{ psig}$$

$$P_4 = 5.0 - 5.0\,\frac{(267 + 45.0)}{(324.63)} = .195 \text{ psig}$$

Note that the pressure differential across the valve, $P_2 - P_4$, is 4.112 psi, not 5.0 psi as would have been assumed without piping calculations.

5) The Mach no. of the theoretical 2 inch discharge pipe is:

$$Ma_5 = \frac{.00245W}{P_5 d^2}\sqrt{\frac{T_o}{kM}}$$

Where W = 2112 lbs./hr.

P_5= 14.7 psia, pressure at discharge header exit.

d = 2.068 in., dia. of inlet pipe.

T_o = 520°R, Temperature of Lading Fluid

k = 1.3, Ratio of Specific Heats.

M = 17.38, Molecular Wt. of Lading Fluid

Q_d = 807 SCFM

$$Ma_5 = \frac{.00245\,(2112)}{(14.7)\,(2.068)^2}\sqrt{\frac{520}{(1.3)\,(17.38)}}$$

Ma_5 = .395

Therefore this approach is applicable since Mach No. is less than one 1.0 (choked flow does not exist).

NOTE (1)

The temperature used to calculate the static discharge header exit pressure using the exit Mach No. should be the static temperature of the gas in the header. Normally this temperature is not known. Using the temperature at the valve inlet will yield reasonably accurate results.

The exit static temperature could be expressed in terms of stagnation temperature and the inlet stagnation temperature used since the flow is assumed to be adiabatic. Doing this however, introduces an appreciable error when compared with actual test results because the static temperature does not decrease as much as theory would predict.

To confirm the validity of using static inlet temperature, a 2J3 POSRV was flow tested at inlet pressures of 50 to 500 psig using 58°F natural gas with a specific gravity of 0.58. Table I lists the results.

TABLE I

INLET PRESSURE (PSIG)	**P_2 CALCULATED (PSIG)**	**P_2 MEASURED (PSIG)**
50	0	0
100	0	0
200	6.75	7.25
300	16.5	17.25
400	26.0	27.5
500	36.5	38.5

NOTE (1)

MACH NUMBER DERIVATION

$$Ma = \left(\frac{W}{A}\right)\left(\frac{1}{P}\right)\sqrt{\frac{\overline{R}T}{kg}}$$

[NASA COMPRESSED GAS HANDBOOK, 1969, p.85]

UNITS: W - lb_m/s
A - ft^2
P - lb_f/ft^2
T - °R, STATIC

CONSTANTS: $\overline{R}$, INDIVIDUAL GAS CONST.
R, UNIVERSAL GAS CONST.
$\overline{R} = R/M$
$R = 1544$ ft lb_f/mole$_{lb}$ °R
$g = 32.174$ ft $lb_m/s^2\, lb_f$

UNIT CONVERSIONS:

$$Ma = \left(\frac{\frac{W}{3600}}{\frac{\pi}{4}d^2\frac{1}{144}}\right)\left(\frac{1}{144P}\right)\sqrt{\frac{1544\,T}{Mk(32.174)}}$$

$$= 0.00245\,\frac{W}{d^2 P}\sqrt{\frac{T}{Mk}}$$

(2) Equation 5.31 Compressed Gas Handbook SP 3045 NASA

(3) Figure 6.10 Chapman

(4) Pages 37 and 38 used with permission of Crane Co.

(5) Crane Technical Papaer 410, p.

NOTE (6)

DERIVATION OF L/D AND K EQUATIONS FOR SAFETY RELIEF VALVES

(1) $q_h = 40{,}700\, Y d^2 \sqrt{\dfrac{\Delta P\, P_1}{K T_1 G}}$ [CRANE TECHNICAL PAPER #410 p. 3-4]

WHERE: q_h = VOLUMETRIC FLOW RATE, SCFH

Y = EXPANSION FACTOR

d = PIPE INTERNAL DIAMETER, in

ΔP = PRESSURE DROP ACROSS PIPE (VALVE), PSI.

P_1 = INLET PRESSURE, PSIA

K = RESISTANCE COEFFICIENT

T_1 = INLET TEMPERATURE, °R

G = SPECIFIC GRAVITY OF FLOWING MEDIA

(2) $V = 863\, K_d A F \sqrt{\dfrac{\Delta P' P_2'}{T_1 G}}$

WHERE: V = VOLUMETRIC FLOW RATE THROUGH VALVE, SCFM

K_d = VALVE NOZZLE COEFFICIENT

A = VALVE NOZZLE AREA, in^2

$\Delta P'$ = PRESSURE DROP ACROSS VALVE NOZZLE, PSI

P_2' = PRESSURE AT NOZZLE EXIT, PSIA

$= P_1 - 0.62(P_1 - P_2)^{1.04}$ FOR TYPE 91/94/95

$= P_1 - 0.55(P_1 - P_2)^{0.98}$ FOR TYPE 93/93T

P_2 = PRESSURE AT VALVE OUTLET, PSIA

$$F = \sqrt{\frac{\left(\frac{P_1}{P_2'}\right)^{(k-1)/k}\left[\left(\frac{P_1}{P_2'}\right)^{(k-1)/k} - 1\right]}{\frac{k-1}{k}\left(\frac{P_1}{P_2'} - 1\right)}}$$

SOLVING FOR K IN EQUATION (1):

$$K = \frac{(40{,}700)^2 Y^2 d^4 \Delta P \, P_1}{q_h^2 \, T_1 \, G}$$

$$q_h = 60V$$

SUBSTITUTING V, USING EQUATION (2), FOR q_h:

$$K = \frac{(40{,}700)^2 Y^2 d^2 \Delta P \, P_1 \, T_1 \, G}{(60)^2 (863)^2 K_d^2 A^2 F^2 \Delta P' P_2' \, T_1 \, G}$$

COMBINING NUMERICAL COEFFICIENTS AND SIMPLIFYING TERMS:

$$K = \frac{0.61782 \, Y^2 d^4 \Delta P \, P_1}{K_d^2 A^2 F^2 \Delta P' P_2'}$$

FOR A GIVEN VALVE OR PIPE GEOMETRY K IS CONSTANT. Y, THE EXPANSION FACTOR, IS 1.0 FOR INCOMPRESSIBLE FLOW AT VERY LOW PRESSURES. THEREFORE, K CAN BE DETERMINED BASED ON THE LIMITING VALUES OF F^2, ΔP, $\Delta P'$ AND P_2' AS P_1 APPROACHES P_2. THESE LIMITING VALUES ARE 1.0 FOR F^2 AND $\Delta P P_1 / \Delta P' P_2'$. THEREFORE, AT THE LIMIT K IS:

$$K = \frac{0.61782 \, d^4}{K_d^2 A^2}$$

WHERE: d = PIPE INTERNAL DIAMETER, in.

K_d = VALVE NOZZLE COEFFICIENT

A = VALVE NOZZLE AREA, in^2

$$\frac{L}{D} = \frac{K}{f}$$

WHERE: f = FRICTION FACTOR

Resistance of Valves and Fittings to Flow of Fluids

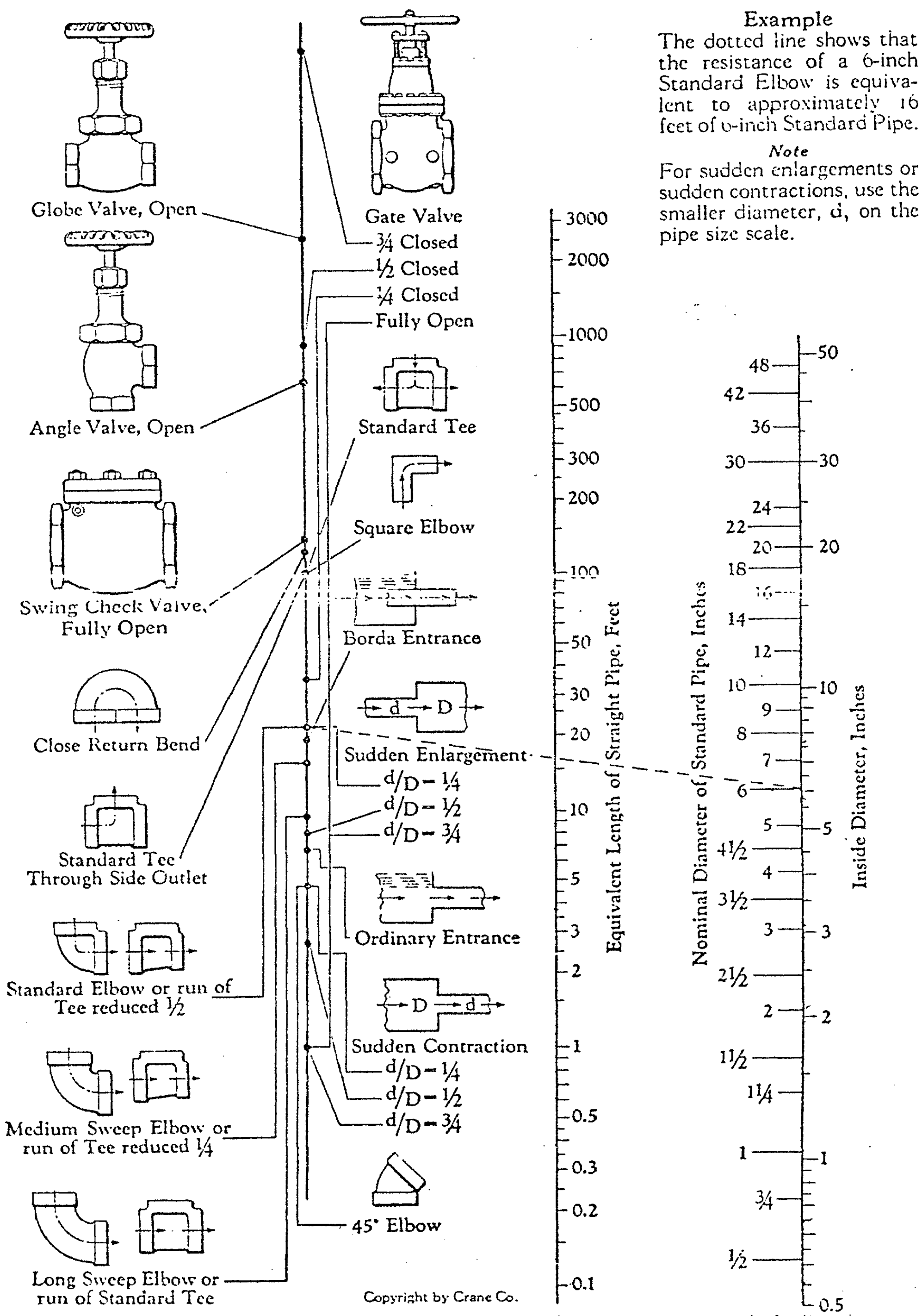
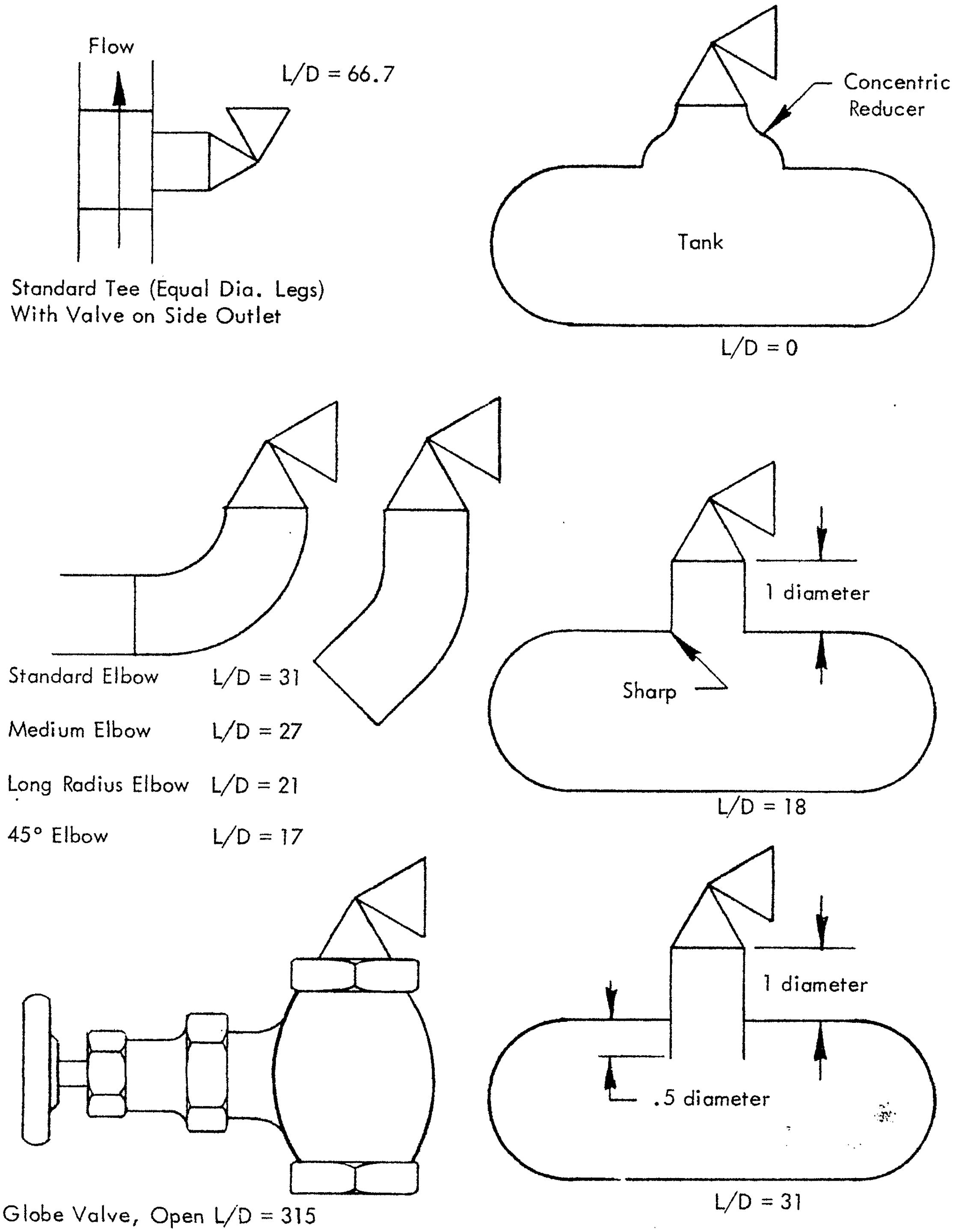


FIGURE 1

EQUIVALENT LENGTHS OF VARIOUS FITTINGS (1) (4)



(1) Crane "Resistance of Valves and Fittings to the Flow of Fluids"

FIGURE II

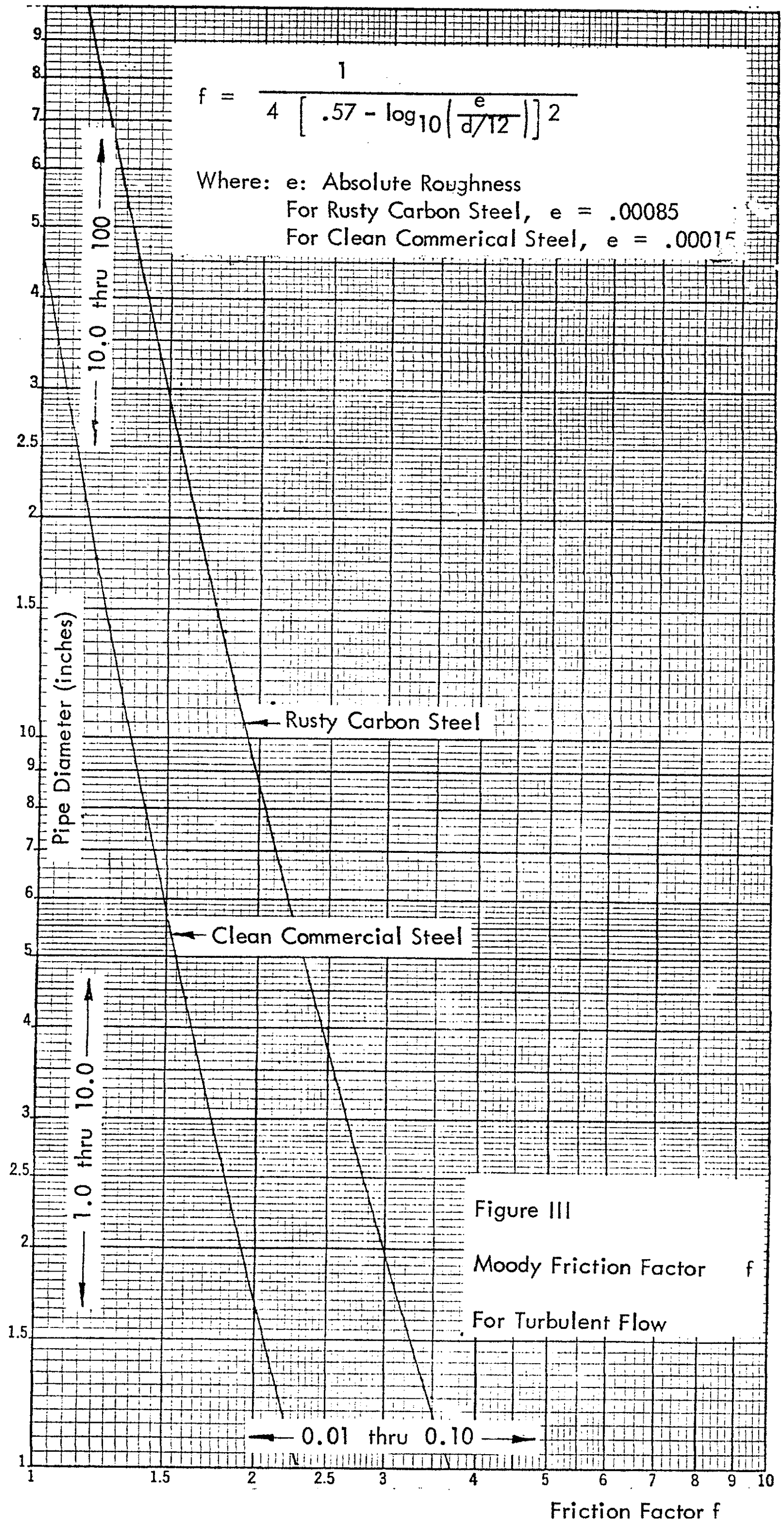


Figure III
Moody Friction Factor f
For Turbulent Flow

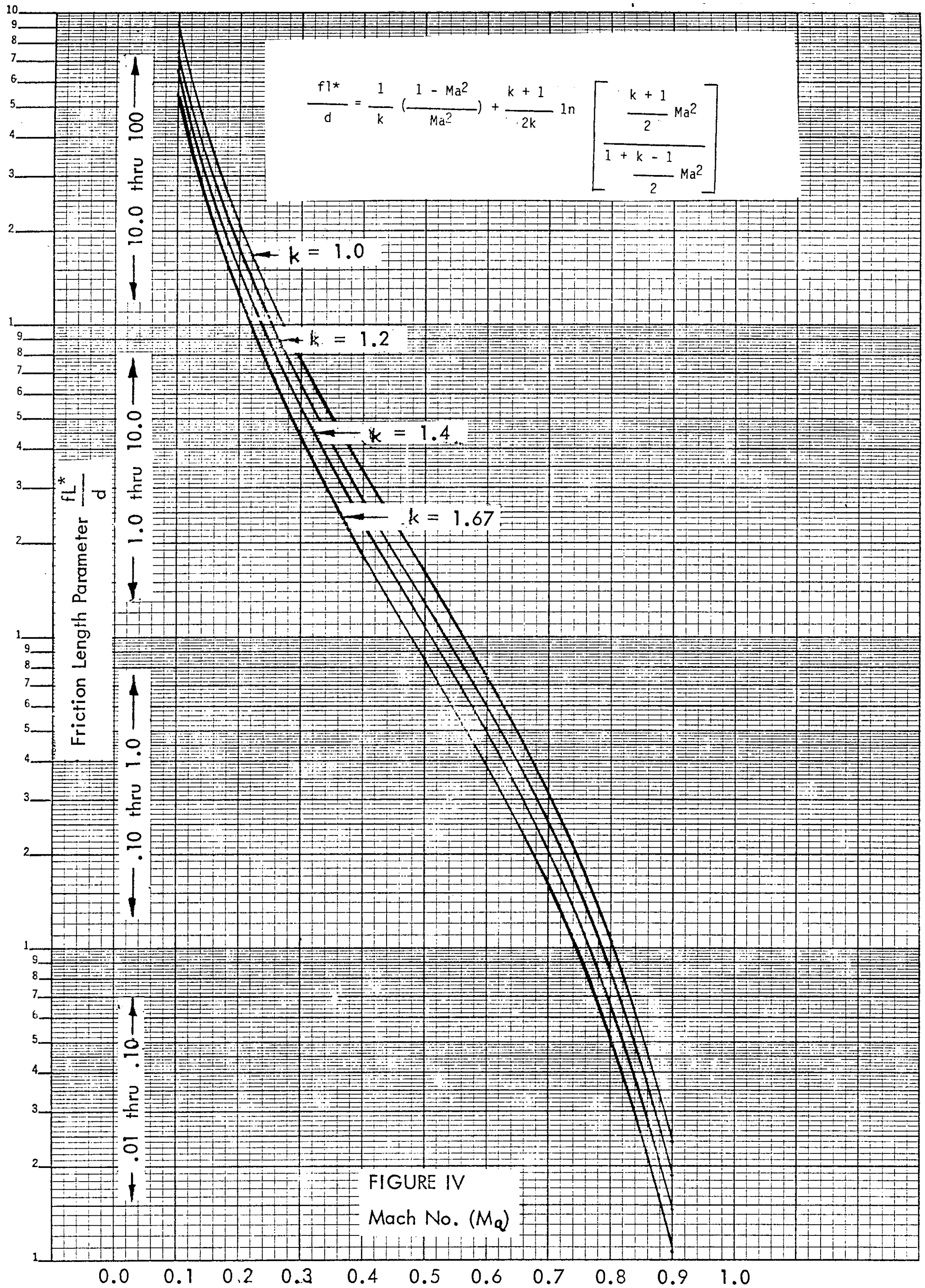


FIGURE IV
Mach No. (M_a)

Stagnation Pressure Ratio

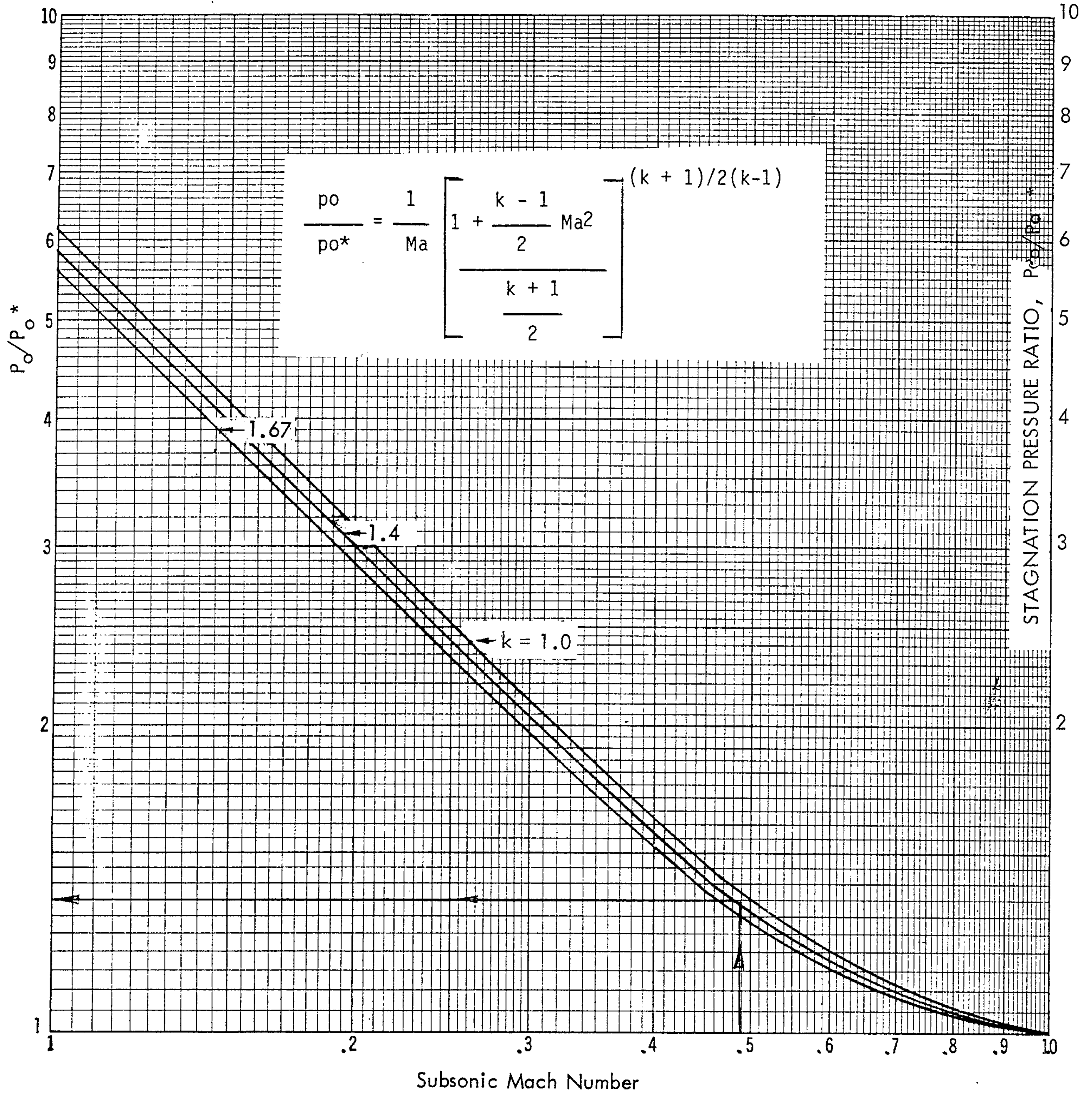
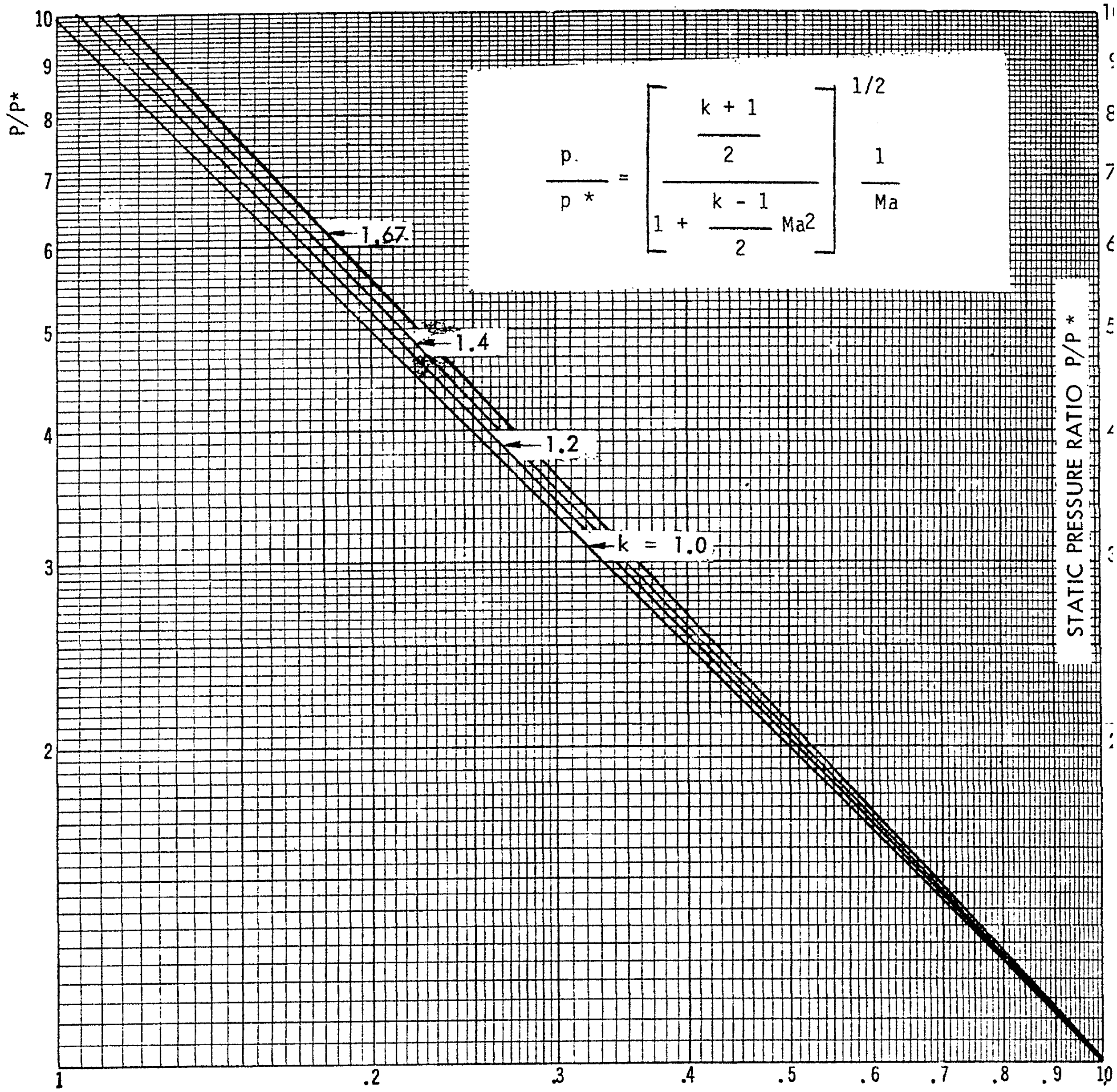


FIGURE V-...

Static Pressure Ratio



Subsonic Mach Number

FIGURE VI

Isentropic Area Ratio

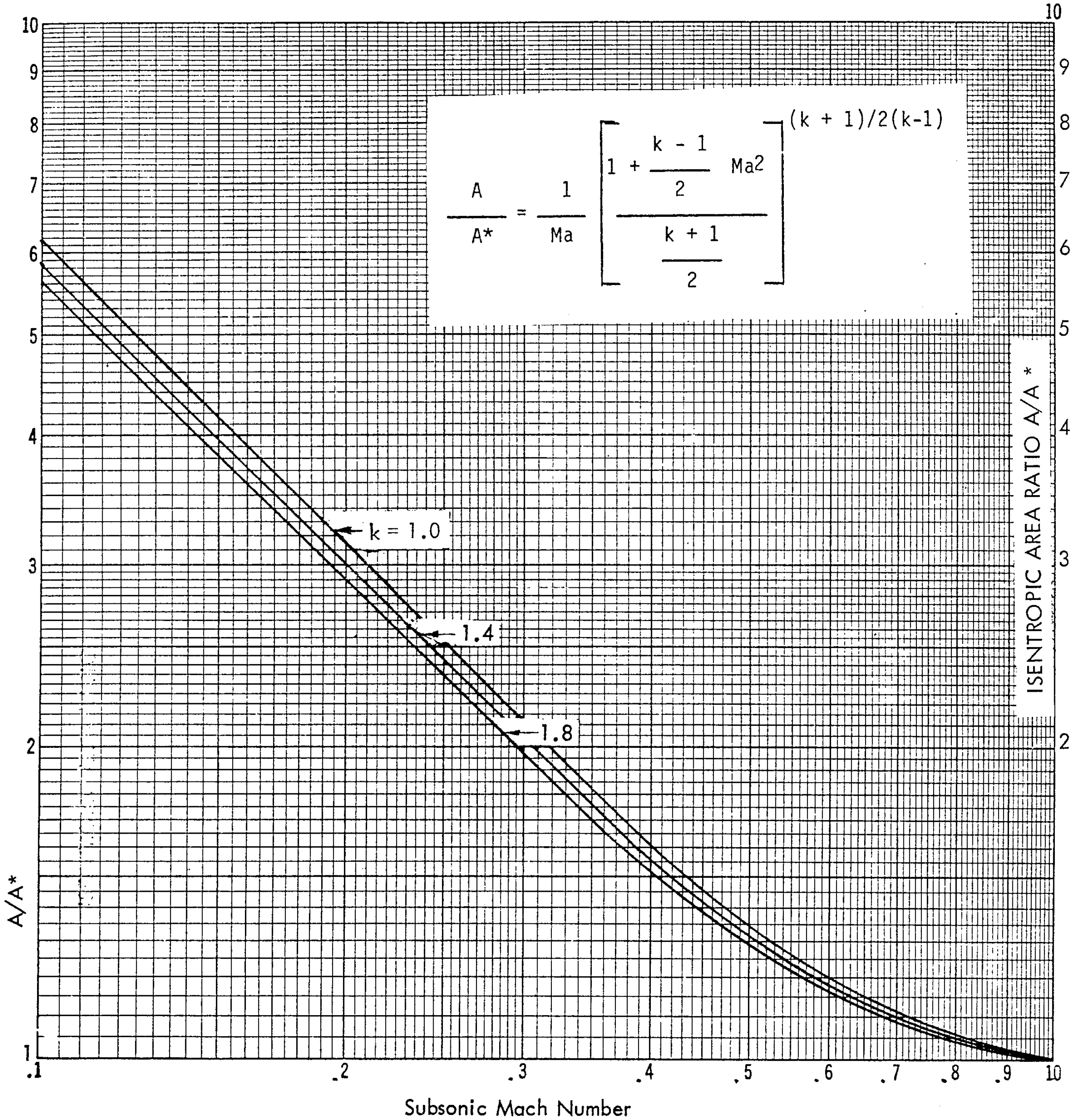


FIGURE V.11

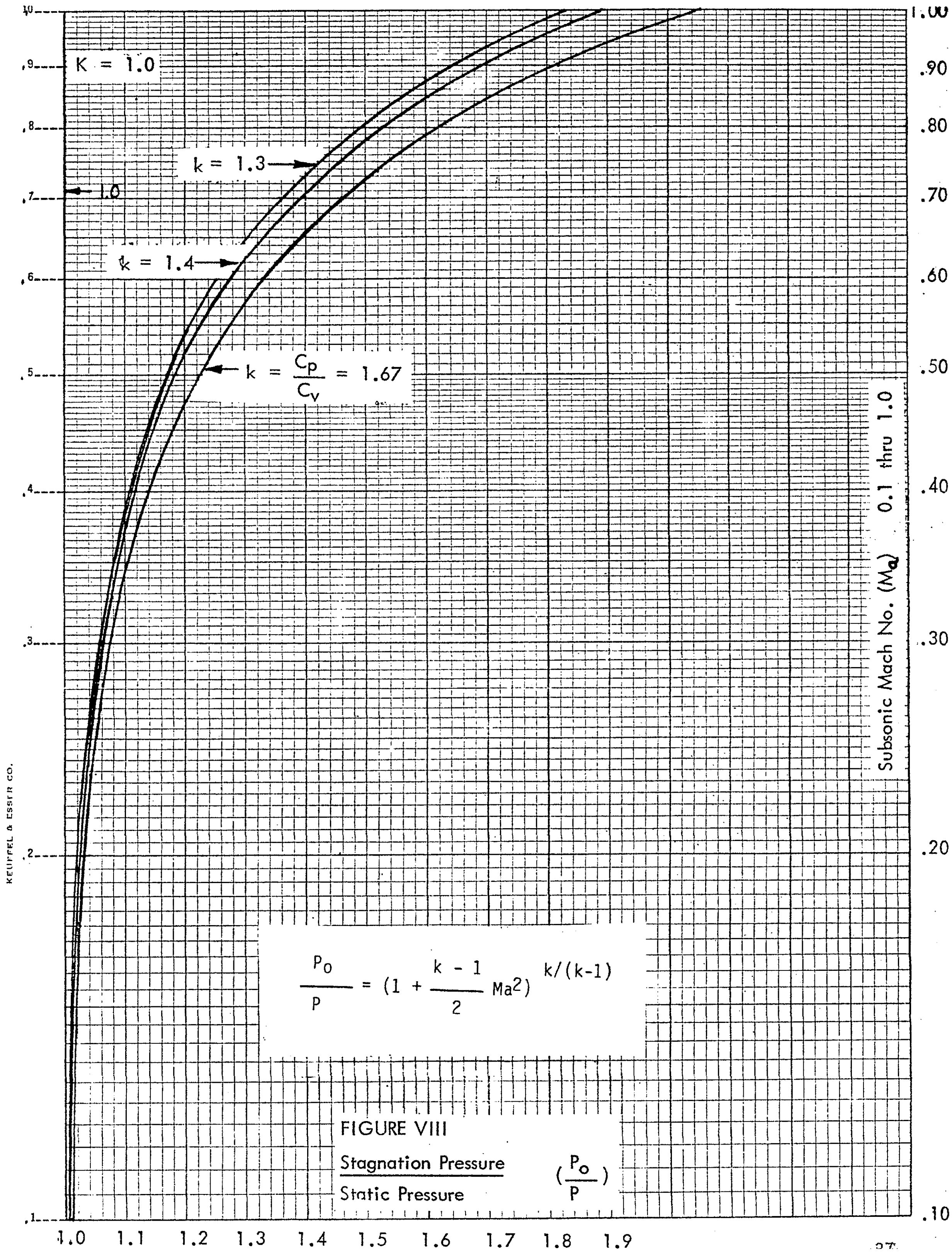


FIGURE VIII

Stagnation Pressure / Static Pressure $\left(\frac{P_o}{P}\right)$

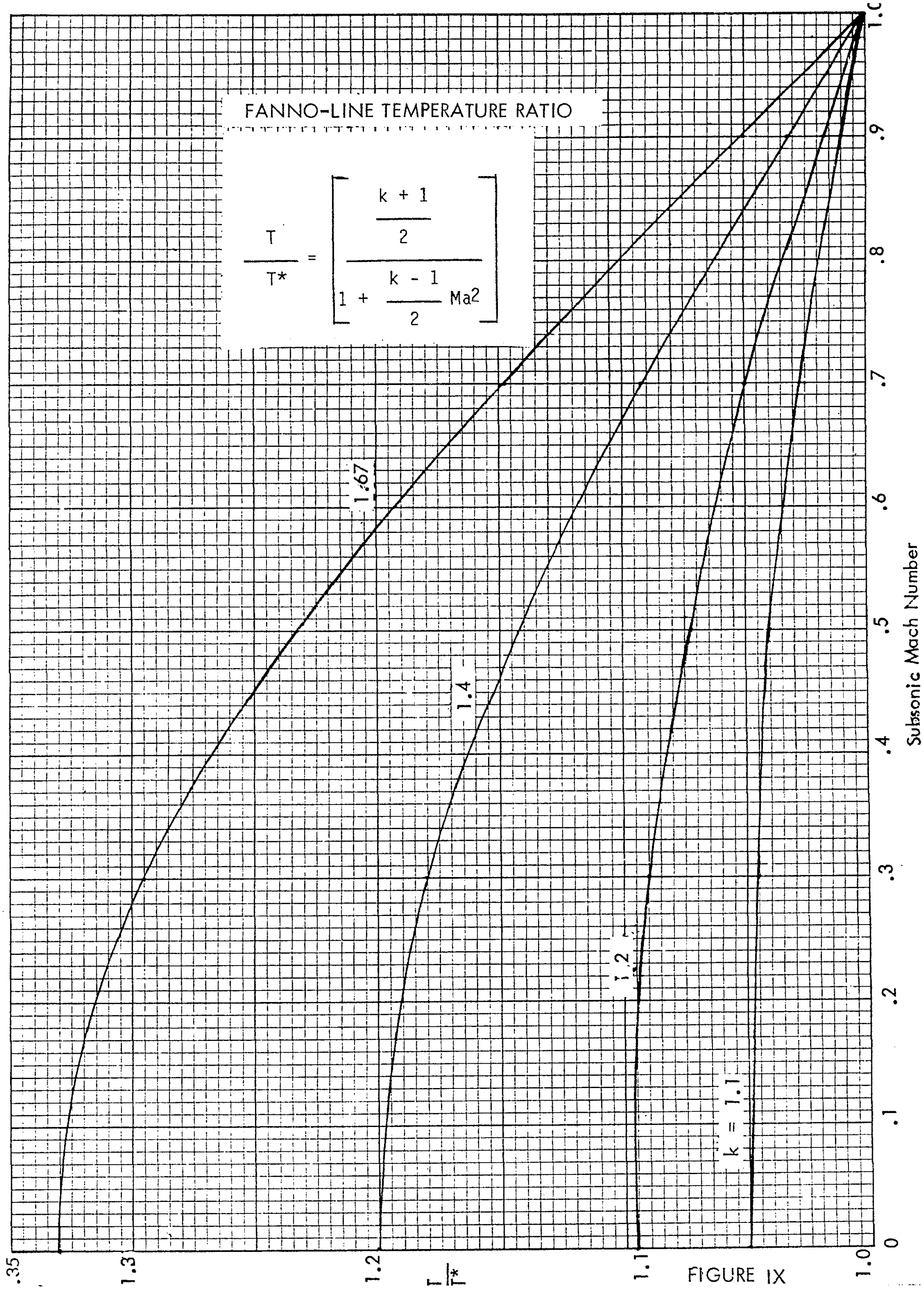

FANNO-LINE TEMPERATURE RATIO
$$\frac{T}{T^*} = \left[\frac{\frac{k+1}{2}}{1 + \frac{k-1}{2} Ma^2}\right]$$
1.67
1.4
1.2
k = 1.1
Subsonic Mach Number
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0
$\frac{T}{T^*}$
1.35
1.3
1.2
1.1
1.0

FIGURE IX

FIGURE X

TYPE 91/94/95
$K_d = .855$

Valve Size	L/D	L (In.)	K
2 x 3	95	197	1.81
3 x 4	111	341	1.92
4 x 6	128	516	2.08
6 x 8	156	947	2.32
8 x 10	156	1245	2.19
10 x 12	197	1974	2.65
12 x 16	168	2016	2.17

TYPE 93
$K_d = .939$

Valve Size	L/D	L (In.)	K
2 x 3	129	267	2.44
3 x 4	135	414	2.33
4 x 6	148	596	2.41
6 x 8	166	1008	2.48
8 x 10	153	1221	2.15
10 x 12	202	2024	2.72
12 x 16	159	1908	2.06

Equivalent Lengths (L/D) determined using:

$$\frac{L}{D} = .61782 \frac{d^4}{K_d^{\,2} A^2 f} \qquad \text{(Note 6)}$$

Where:
$$f = \frac{1}{4\left[.57 - \log 10 \left(\frac{e}{d/12}\right)\right]^2}$$

$e = .00015$

d = inlet pipe diameter, in (Schedule 40 size used)

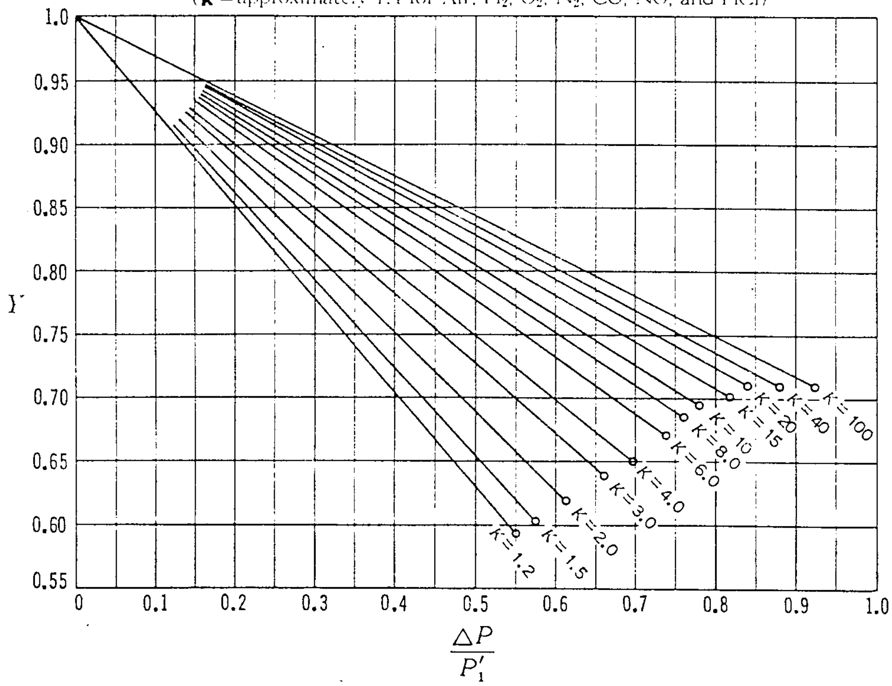
A = orifice area, in^2

$K = f\ L/D$

Net Expansion Factor Y for Compressible Flow Through Pipe to a Larger Flow Area

$k = 1.4$

(k = approximately 1.4 for Air, H_2, O_2, N_2, CO, NO, and HCl)



Limiting Factors For Sonic Velocity
$k = 1.4$

K	$\frac{\Delta P}{P'_1}$	Y
1.2	.552	.588
1.5	.576	.606
2.0	.612	.622
3	.662	.639
4	.697	.649
6	.737	.671
8	.762	.685
10	.784	.695
15	.818	.702
20	.839	.710
40	.883	.710
100	.926	.710

FIGURE XI