From: A. Amr
To: C. Jackson
Date: 3/28/96
Subject: Palo Verde USQ

The use of Fanno equations, which assumes adiabatic and constant area flow, to pipe components such as valves, orifices and cracks which are not of constant area has been the standard engineering practice as seen in the enclosed reference. This is equivalent to using Bernoulli equation for incompressible flow (assuming adiabatic constant area flow) and applying it to valves, orifices and cracks.

AAmr
3/28/96

Table 6-14. Trends in Compressible Subsonic Pipe Flow of Perfect Gases.[a]

	Isothermal[b]	Adiabatic[c]
Static pressure	decreases	decreases
Total pressure[d]	decreases	decreases
Velocity	increases	increases
Density	decreases	decreases
Temperature	constant	decreases
Mach number	increases[e]	increases[f]
Reynolds number	increases	increases[g]
Stagnation temperature[h]	increases	constant

[a] Uniform pipe or duct. Trends are with increasing distance as gas flows down the pipe. (Ref. 6-3, Vol. 1, pp. 166, 180.)
[b] The Mach number at the entrance is less than $1/\gamma^{1/2}$, where γ is the ratio of specific heat at constant pressure to that at constant volume, $\gamma = 1.4$ for air. Heat must be added to the flow to maintain constant temperature.
[c] The Mach number at the entrance is less than 1.0.
[d] Static pressure plus dynamic pressure, $p + \frac{1}{2}\rho U^2$.
[e] Tends toward the limit of $1/\gamma^{1/2}$ for long pipes.
[f] Tends toward the limit of 1 for long pipes.
[g] Kinematic viscosity decreases with temperature.
[h] Isentropic stagnation temperature.

The expressions of Table 6-13 for adiabatic incompressible flow are compared with experimental data for an air flow through a 0.375-in. (0.953-cm) pipe in Fig. 6-24. Note that the incompressible flow equation [Eq. (6-6)] significantly underestimates the pressure loss, the

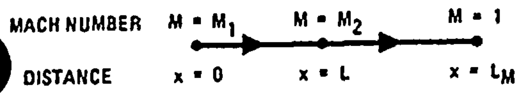

Fig. 6-23. Relationship between length L_M and Mach number M for adiabatic flow through a constant area duct.

simplified compressible flow expression (frame 3 of Table 6-13) is valid for small pressure drops, and frame 2 of Table 6-13, although computationally more difficult, provides an accurate prediction of the experimental data.

6.7.2. A Technique for Adiabatic Flow Through Pipeline Components

A relatively simple procedure can be employed to solve for subsonic adiabatic flow of a perfect gas through a series of pipeline components, such as shown in Fig. 6-25, based on the models of (1) an ideal adiabatic expansion or contraction at changes in pipe area (Chapter 7), and (2) adiabatic flow through constant-diameter pipes or ducts with friction (Tables 6-13 and 6-15). In essence, the procedure is as follows:

1. All frictional losses are smeared over the lengths of constant area piping.
2. The expansions or contractions between pipe sections are considered to be ideal, their frictional losses having been taken into account in step 1.

The procedure is most easily applied if the inlet flow rate is known (or postulated) and it is desired to compute the outlet conditions. For known pressure type boundary conditions with an unknown flow rate through a uniform pipe, it may be simpler to apply the simplified forms of Table 6-13.

In applying this procedure, it is useful to recall for steady flow of an ideal gas that (1) the mass rate of flow through each section is constant; (2) the isentropic

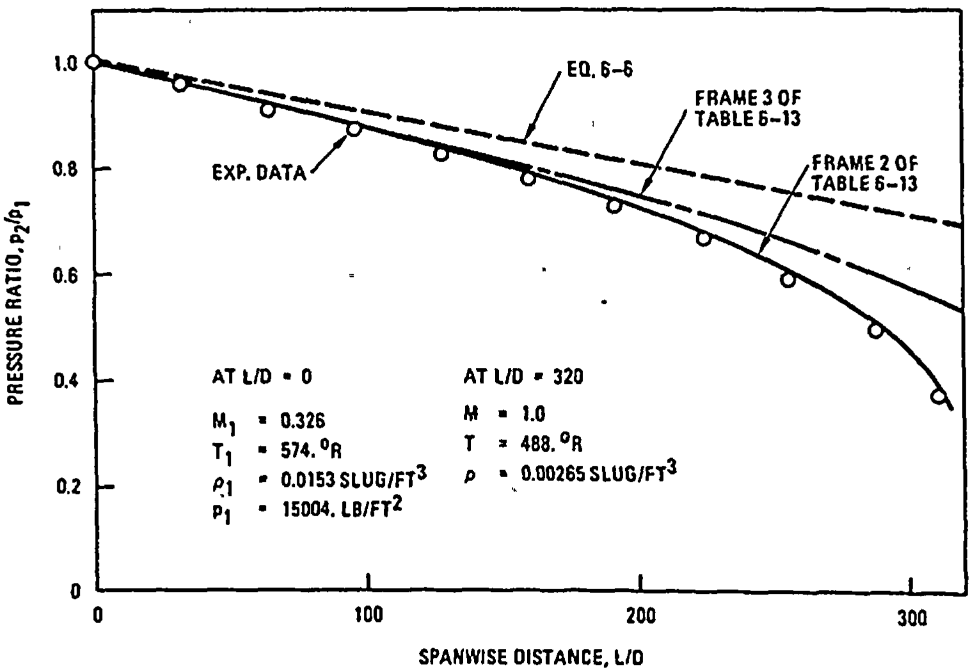

Fig. 6-24. Adiabatic compressible flow of air through a smooth pipe with $\gamma = 1.4$, $f = 0.0129$, $D = 0.375$ in., $Re = 4.63 \times 10^5$.

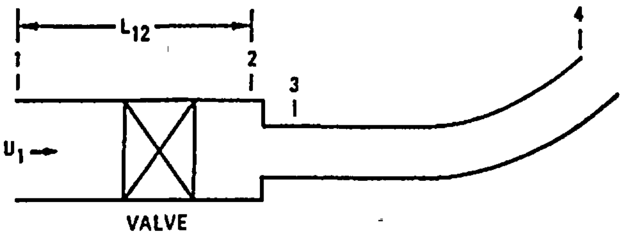

Fig. 6-25. A pipeline with adiabatic compressible flow.

stagnation temperature T_0 and isentropic stagnation speed of sound c_0 are constant for adiabatic flow regardless of the presence of absence of friction; (3) the isentropic stagnation temperature T_0, pressure p_0, density ρ_0, and speed of sound c_0 are constant across a frictionless adiabatic (i.e., isentropic) expansion or contraction and the ratio of these quantities to their local static values varies only with Mach number and γ (see Section 7.2); (4) using conservation of mass, $\dot{m} = \rho UA$ where $\dot{m}$ is the mass flow rate and A is the area of the pipe or duct, the perfect gas law $p = \rho RT$, and the speed of sound $c^2 = \gamma RT$, it can be shown that the quantity

$$\frac{\dot{m}T_0^{1/2}R^{1/2}}{p_0 A} = \gamma^{1/2}M\left(1 + \frac{\gamma - 1}{2}M^2\right)^{(\gamma+1)/2(1-\gamma)} \quad (6\text{-}68)$$

is only a function of Mach number M and the ratio of specific heats γ; and (5) the ratios of static pressure, temperature, and speed of sound to their isentropic stagnation values are functions only of Mach number M and ratio of specific heats γ,

$$\frac{T}{T_0} = \left(\frac{c}{c_0}\right)^2 = \left(1 + \frac{\gamma - 1}{2}M^2\right)^{-1}. \quad (6\text{-}69)$$

$$\frac{p}{p_0} = \left(1 + \frac{\gamma - 1}{2}M^2\right)^{\gamma/(1-\gamma)}. \quad (6\text{-}70)$$

Equations (6-68) through (6-70) are tabulated in Table 6-15. They are derived in Chapter 7, Section 7.2.

As an example of the procedure, consider the pipeline segment shown in Fig. 6-25. In this example, the inlet mass flow, inlet static pressure p_1, inlet absolute temperature T, and density ρ_1 are known, so the inlet velocity U_1 and Mach number M_1 can be easily computed. Given M_1, the value of fL_{M1}/D is read from Table 6-15. The ictional losses between 1 and 2 are the sum of the ight pipe loss and the valve loss. Smearing these losses r the pipe length between 1 and 2 yields

$$\frac{fL_{M2}}{D_1} = \frac{fL_{M1}}{D_1} - \left(\frac{fL_{12}}{D_1} + K_{valve}\right).$$

Table 6-15 then gives M_2 from fL_{M2}/D_1. (In applying this equation it has been implicitly assumed that the frictional losses are (1) independent of Mach number and (2) equivalent to pipe frictional loss. While the pipe friction effect is nearly independent of Mach number, the loss induced in bends, valves, and other fittings will have some, unknown, Mach number dependence. The approximation of smearing the frictional losses over the pipe is apt to be a valid approximation only if the frictional losses are not dominated by a single, isolated component.)

Knowing the Mach number at 2, M_2, the quantity $\dot{m}T_0^{1/2}R^{1/2}/(p_0A)$ at 2 can be read from column 5 of Table 6-15. Since T_0, p_0, $\dot{m}$, and R are all constant across the expansion, the Mach number at 3 is found from the line of Table 6-15 that contains

$$\left.\frac{\dot{m}T_0^{1/2}R^{1/2}}{p_0A}\right|_3 = \left.\frac{\dot{m}T_0^{1/2}R^{1/2}}{p_0A}\right|_2 \left(\frac{A_3}{A_2}\right).$$

A_3/A_2 is defined by the geometry of the contraction. Once M_3 is read from the table, the quantities T/T_0 and c/c_0 can be obtained from the same line, and $T_3 = T_0(T/T_0)$, $c_3 = c_0(c/c_0)$, and $U_3 = M_3c$ can be computed. ρ_3 can then be computed using conservation of mass, $\rho_3 = \dot{m}/U_3A_3$, and p_3 can be found from the perfect gas relationship.

The fluid properties between cross sections 3 and 4 are computed in the same manner as the properties between 1 and 2. A resultant friction loss is used which is the sum of the fL/D straight pipe loss, the contraction loss (Table 6-7), and the loss of the bend (Table 6-5). A numerical example is presented in Section 6.9.3.

6.8. TWO-PHASE FLOW

6.8.1. Fundamental Concepts

Many flows of practical importance are not homogeneous; they are two-phase mixtures of solids, liquids, and gases. Two-phase flows, which are the simplest of multiphase flows, generally cannot be accurately predicted by one-dimensional fluid dynamic analysis. While the integral equations for fluid dynamic analysis of two-phase flows are of the same form as for single-phase flow, their solution is much more formidable owing to (1) the multitude of parameters required to describe a two-phase flow, (2) the dynamic interaction of the two phases within a pipe or duct, and (3) the nonsteady dynamics that enter into most two-phase flows as water becomes steam or solid particles settle. Often the nature of the internal interaction between the two phases cannot be fully described and resort is made to descriptors such as "bubbly flow" or "slug flow."

As a result of the complexities of two-phase flows, empirical models based on experimental data must be

PALO VERDE STEAM GENERATOR ISSUE AND STEAM GENERATOR RULE

MEETING PURPOSE:

IDENTIFY TO ALL INDIVIDUALS WORKING ON THE SG RULE NEW POTENTIAL RADIOACTIVITY RELEASE PATHWAY DURING AN ACCIDENT. PATHWAY INVOLVES DEGRADED SG TUBES.

Meeting set for next Tuesday 3/5/96 @ 10:00 (tentative)

WHY IS THIS INFORMATION IMPORTANT:

POTENTIAL IMPACT UPON ACTIONS AND REQUIREMENTS INSTITUTED BY THE STAFF TO IMPLEMENT A SG RULE.

MEETING ACCOMPLISHMENTS:

1. PROVIDE BACKGROUND INFORMATION

2. IDENTIFY WHAT ASPECTS OF THE PROBLEM NEED TO BE REVIEWED AND BY WHOM

3. IDENTIFY WHAT QUESTIONS NEED TO BE ASKED AND WHO PROVIDES THE ANSWER.

4. ESTABLISH A DATE FOR MEETING TO DISCUSS THE IMPACT OF THE PALO VERDE ISSUE ON THE SG RULE.

DESCRIPTION OF THE PROBLEM

1. POST-LOCA COOLING - OPERATION OF ADVs AND TBVs

2. 1 HR AFTER LOCA, ADVs OR TBVs UTILIZED TO COOL SGs FOLLOWING A LOCA

3. SECONDARY PRESSURE CAN BE REDUCED BELOW PRIMARY COOLANT PRESSURE

4. BECAUSE ADVs OR TBVs ARE BEING UTILIZED, ISOLATION VALVES CONNECTED TO THE SECONDARY SIDE OF THE SG (GDC 57 VALVES) MAY NO LONGER BE WATER SEALED FOLLOWING A LOCA.

5. PALO VERDE HAS ASSUMED FOR THEIR LOCA ANALYSIS A SINGLE FAILURE OF A GDC 57 VALVE OR A STUCK OPEN ADV TO BOUND THE LOCA ANALYSIS.

6. SINCE THESE VALVES ARE NOT UTILIZED TO LIMIT DOSES TO MEET GDC 19 NOR PART 100, THESE VALVES ARE NOT APPENDIX J TESTED.

7. FOR THIS PATHWAY, CONTRIBUTION TO DOSE IS EQUIVALENT TO 50% OF THE CONTRIBUTION FROM CONTAINMENT LEAKAGE.

ARIZONA PUBLIC SERVICE COMPANY -
PALO VERDE NUCLEAR GENERATING STATION

5801 S. WINTERSBURG RD.,
MAIL STATION 7636,
TONOPAH, AZ. 85354

NUCLEAR REGULATORY AFFAIRS

FAX

DATE:	3/1/96
NUMBER OF PAGES, INCLUDING COVER SHEET:	5

To:	From:
Charles Thomas	Glenn Michael
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Fax phone:	Fax Phone: 602 393-5442
CC:	

REMARKS: ☐ Urgent ☐ For your Review ☐ Reply ASAP ☐ Please comment

Charles: Here are the responses to the questions about our LOCA dose calcs.

Glenn Michael

RESPONSE TO COMMENTS ON CALC. 13- NC- ZY-205

Per our telephone conversation on 2/28/1996, the following is provided to show that assumptions in calculation 13-NC-ZY-205 are conservative and to clarify how equations were derived:

1- CONSTANT IN EQUATION (1):

The constant in EQ(1) is derived from Crane page 3-4 and can be duplicated as follows.

The constant 1.11E-5 is equal to (144 X 0.002228 X0.002228) / (2 X 32.2) where g= 32.2 and CUBIC FOOT /SEC / GPM = 0.002228

2- EQUATION (2)

EQ(2) is showing a typo and the constant should read 3.32E-7 instead of 3.32E-6. Correct value of 3.32E-7 was used in solving the equations. This typo will be corrected in EQ(20) the calculation.

3- EQUATION (5)

There is a typo as discussed in EQ(5) where the plus is to be replaced by a minus in the coefficient of M^2.

Pressure ratio is defined as $P_e / P_0 = (P_e / P) (P/P_0)$ where P is the static pressure upstream.

From Streeter and Wylie "Fluid Mechanics" Equation 6.6.2

$$\frac{P_e}{P} = M\sqrt{\frac{2}{\gamma+1}}\sqrt{1+\frac{\gamma-1}{2}M^2} \qquad \text{EQ(A1)}$$

From Streeter and Wylie Equation 6.3.11

$$\frac{P}{P_o} = \left(1+\frac{\gamma-1}{2}M^2\right)^{-\frac{\gamma}{(\gamma-1)}} \qquad \text{EQ(A2)}$$

Multiplying EQ(A1) times EQ(A2) to obtain EQUATION (5) in the calculation.

$$Pr = \frac{P_e}{P_o} = M\sqrt{\frac{2}{\gamma+1}}\left(1+\frac{\gamma-1}{2}M^2\right)^{-\frac{-(\gamma+1)}{2(\gamma-1)}}$$

Typos in the calculation will be corrected.

4- CONSERVATISM IN METHODOLOGY.

Two sources of conservatism are assumed

A-Assuming air versus air / steam mixture properties provides conservative results. As seen in equation 4 the flow rate is directly proportional to γ and inversely proportional to R.

For a temperature of 300 F and a pressure of 50 PSIG the γ for air is 1.4 vs. 1.1 for steam. The R for air is 57.9 vs. 85 for steam. Therefore, assuming γ and R for air results in higher flow rates than would exist for any air / steam mixture. Hence using air is conservative.

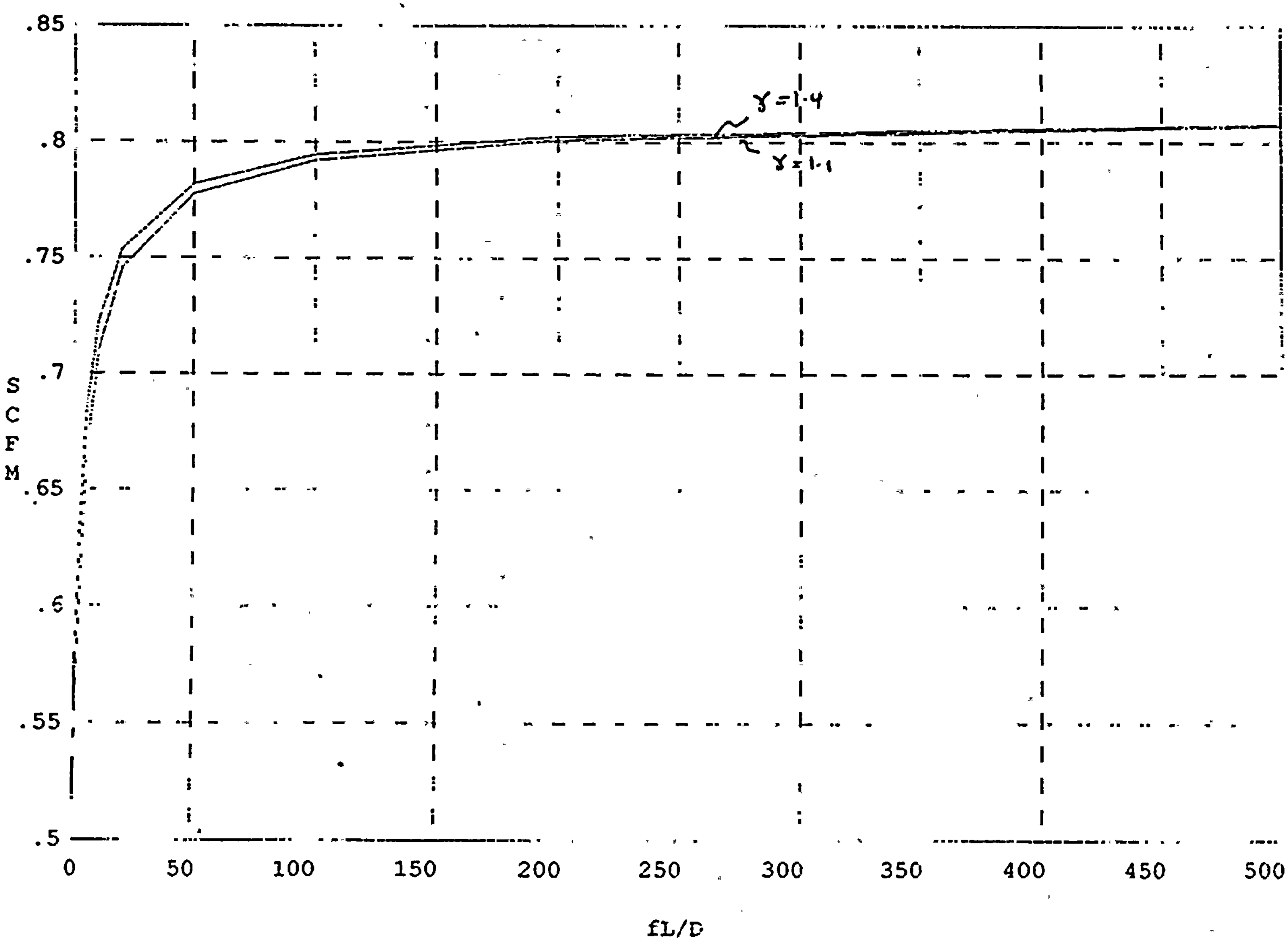
B- Assuming choked flow at exit implies that maximum flow is achieved for a given friction loss. Actual flow will not choke at higher friction losses and will be less (more conservative) than that assumed in the calculation.

5- EFFECTS OF MULTIPLE CRACKS

The area in TABLE 1 of Calculation 13-NC-ZY-205 (A bounding evaluation) is based on a leakage of 1 GPM per crack. Since the basis of the calculation is that this is the TECHNICAL SPECIFICATION limit, for multiple cracks the total flow will be also1 GPM. Hence the area in TABLE 1 is interpreted as the total area of all cracks for a given loss coefficient. The flow will also be the Air/steam flow from all cracks for a given loss coefficient. Since we take in the calculation the largest loss coefficient, the results of the calculation applies to the conservative case of multiple cracks each with the highest possible friction loss.

6- USING A LOSS COEFFICIENT OF 200

As discussed previously the increase of flow with loss coefficient is logarithmic as seen in attached Figure.Extrapolating the curve linearly to the SCFM value of 0.9 this corresponds to a loss coefficient of 1700. In essence the assumption in calculation is that the crack loss coefficient used is 1700. In the absence of an agreement on crack characteristic this assumption is acceptable based on engineering judgment.



for SCFM = 0.9
FL/D = 1700

Summary of Formulas — continued

● Head loss and pressure drop through valves and fittings

Head loss through valves and fittings is generally given in terms of resistance coefficient K which indicates static head loss through a valve in terms of "velocity head", or, equivalent length in pipe diameters L/D that will cause the same head loss as the valve.

From Darcy's formula, head loss through a pipe is:

$$h_L = f\,\frac{L}{D}\,\frac{v^2}{2g} \qquad \text{Equation 3-5}$$

and head loss through a valve is:

$$h_L = K\,\frac{v^2}{2g} \qquad \text{Equation 3-14}$$

therefore:

$$K = f\,\frac{L}{D} \qquad \text{Equation 3-15}$$

To eliminate needless duplication of formulas, the following are all given in terms of K. Whenever necessary, substitute $(f\,L/D)$ for (K).

$$h_L = \frac{522\,Kq^2}{d^4} = 0.002\,59\,\frac{KQ^2}{d^4} \qquad \text{Equation 3-14}$$

$$h_L = 0.001\,270\,\frac{KB^2}{d^4} = 0.000\,0403\,\frac{KW^2\bar{V}^2}{d^4}$$

$$\Delta P = 0.000\,1078\,K\rho v^2 = 0.000\,000\,0300\,K\rho V^2$$

$$\Delta P = 3.62\,\frac{K\rho q^2}{d^4} \overset{(\frac{\pi}{4})^2}{=} 0.000\,017\,99\,\frac{K\rho Q^2}{(\frac{\pi}{4}d^4)}\,A^2$$

$$\Delta P = 0.000\,008\,82\,\frac{K\rho B^2}{d^4}$$

$$\Delta P = 0.000\,000\,280\,\frac{KW^2\bar{V}}{d^4}$$

$$\Delta P = 0.000\,000\,000\,605\,\frac{K(q'_h)^2\,T\,S_g}{d^4\,P'}$$

$$\Delta P = 0.000\,000\,001\,633\,\frac{K\,(q'_h)^2\,S_g^2}{d^4\,\rho}$$

For compressible flow with h_L or ΔP greater than approximately 10% of inlet absolute pressure, the denominator should be multiplied by Y^2. For values of Y, see page A-22.

● Pressure drop and flow of liquids of low viscosity using flow coefficient

$$\Delta P = \left(\frac{Q}{C_v}\right)^2 \frac{\rho}{62.4} \qquad \text{Equation 3-16}$$

$$Q = C_v\sqrt{\Delta P\,\frac{62.4}{\rho}} = 7.90\,C_v\sqrt{\frac{\Delta P}{\rho}}$$

$$C_v = Q\sqrt{\frac{\rho}{\Delta P\,(62.4)}} = \frac{29.9\,d^2}{\sqrt{f\,L/D}} = \frac{29.9\,d^2}{\sqrt{K}}$$

$$K = \frac{891\,d^4}{(C_v)^2}$$

● Resistance coefficient, K, for sudden and gradual enlargements in pipes

If, $\theta \leq 45°$,

$$K_1 = 2.6\sin\frac{\theta}{2}\,(1-\beta^2)^2 \qquad \text{*Equation 3-17}$$

If, $45° < \theta \leq 180°$,

$$K_1 = (1-\beta^2)^2 \qquad \text{*Equation 3-17.1}$$

● Resistance coefficient, K, for sudden and gradual contractions in pipes

If, $\theta \leq 45°$,

$$K_1 = 0.8\sin\frac{\theta}{2}\,(1-\beta^2) \qquad \text{*Equation 3-18}$$

If, $45° < \theta \leq 180°$,

$$K_1 = 0.5\sqrt{\sin\frac{\theta}{2}}\;(1-\beta^2) \qquad \text{*Equation 3-18.1}$$

**Note:* The values of the resistance coefficients (K) in equations 3-17, 3-17.1, 3-18, and 3-18.1 are based on the velocity in the small pipe. To determine K values in terms of the greater diameter, divide the equations by β^4.

● Discharge of fluid through valves, fittings, and pipe; Darcy's formula

Liquid flow: Equation 3-19

$$q = 0.0438\,d^2\sqrt{\frac{h_L}{K}} = 0.525\,d^2\sqrt{\frac{\Delta P}{K\rho}}$$

$$Q = 19.65\,d^2\sqrt{\frac{h_L}{K}} = 236\,d^2\sqrt{\frac{\Delta P}{K\rho}}$$

$$w = 0.0438\,\rho d^2\sqrt{\frac{h_L}{K}} = 0.525\,d^2\sqrt{\frac{\Delta P\rho}{K}}$$

$$W = 157.6\,\rho d^2\sqrt{\frac{h_L}{K}} = 1891\,d^2\sqrt{\frac{\Delta P\rho}{K}}$$

Compressible flow:

$$q'_h = 40\,700\,Yd^2\sqrt{\frac{\Delta P\,P'_1}{KT_1\,S_g}} \qquad \text{Equation 3-20}$$

$$q'_h = 24\,700\,\frac{Yd^2}{S_g}\sqrt{\frac{\Delta P\,\rho_1}{K}}$$

$$q'_m = 678\,Yd^2\sqrt{\frac{\Delta P\,P'_1}{KT_1\,S_g}} = 412\,\frac{Yd^2}{S_g}\sqrt{\frac{\Delta P\,\rho_1}{K}}$$

$$q' = 11.30\,Yd^2\sqrt{\frac{\Delta P P'_1}{KT_1\,S_g}} = 6.87\,\frac{Yd^2}{S_g}\sqrt{\frac{\Delta P\rho_1}{K}}$$

$$w = 0.525\,Yd^2\sqrt{\frac{\Delta P}{K\bar{V}_1}} \qquad W = 1891\,Yd^2\sqrt{\frac{\Delta P}{K\bar{V}_1}}$$

Values of Y are shown on page A-22. For K, Y, and ΔP determination, see examples on pages 4-13 and 4-14.

[illegible]