III. FRICTION LOSS IN VALVES AND FITTINGS

Evaluation of the friction loss in valves and fittings involves the determination of the appropriate loss coefficient (K_f), which in turn defines the energy loss per unit mass of fluid:

$$e_f = \frac{K_f V^2}{2} \tag{7.33}$$

where V is (usually) the velocity in the pipe upstream of the fitting or valve. However, this is not always true and care must be taken to ensure that the value of V that is used is the one that is specified in the defining equation for K_f. The actual evaluation of K_f is done by determining the friction loss e_f from measurements of the pressure drop across the fitting (elbows, tees, valves, etc.). This is not straightforward, however, because the pressure in the pipe is influenced by the presence of the fitting for a considerable distance both upstream and downstream of the fitting. It is not possible, therefore, to obtain accurate values from measurements taken at pressure taps immediately adjacent to the fitting. The most reliable method is to measure the total pressure drop through a long run of pipe both with and without the fitting, at the same flow rate, and determine the fitting loss by difference.

There are several "correlation" expressions for K_f, which are described below (in Sections A through E) in the order of increasing accuracy. The "3-K" method (see Section E) is recommended because it accounts directly for the effect of both Reynolds number and fitting size on the loss coefficient and more accurately reflects the effect of fitting diameter than the 2-K method (Section D). For highly turbulent flow, the Crane method (Section C) agrees well with the 3-K method but is less accurate at low Reynolds numbers and is not recommended for laminar flow. The loss coefficient and $(L/D)_{eq}$ methods are more approximate but give acceptable results at high Reynolds (fully turbulent flow) numbers and when losses in valves and fittings are "minor losses" compared to the pipe friction. They are also appropriate for first estimates in problems that require iterative solutions.

A. Loss Coefficient

Values of K_f for various types of valves, fittings, etc., are found tabulated in various textbooks and handbooks. The assumption that these values are constant for a given type of valve or fitting is not accurate, however, because in reality the value of K_f varies with both the size (scale) of the fitting and the level of turbulence (Reynolds number). One reason that K_f is not the same for all fittings of the same type (e.g., all 90° elbows) is that all the dimensions of a fitting, such as the diameter and radius of curvature, do not scale by the same factor for large and small fittings. Most tabulated values for constant K_f values are close to the values of K_∞ from the 3-K method.

B. Equivalent L/D Method

The basis for the $(L/D)_{eq}$ method is the assumption that there is some length of pipe (L_{eq}) that has the same friction loss as that which occurs in the fitting, at a given (pipe) Reynolds number. Thus, the fittings are conceptually replaced by the equivalent additional length of pipe that has the same friction loss as the fitting:

$$e_f = \frac{4fV^2}{2} \sum \left(\frac{L}{D} \right)_{eq} \tag{7.34}$$

where f is the Fanning friction factor in the pipe at the given pipe Reynolds number and relative roughness. This is a convenient concept because it allows the solution of pipe flow problems with fittings to be carried out in a manner identical to that without fittings if L_{eq} is known. Values of $(L/D)_{eq}$ are tabulated in various textbooks and handbooks for a variety of fittings and valves (and are also listed in Table 7.3 here).

TABLE 7.3
3-*K* Constants[a] for Loss Coefficients for Valves and Fittings

Fitting		r/D	$(L/D)_{eq}$	K_1	K_i	K_d
Elbows						
90°	Threaded, standard	1	30	800	0.14	4.0
	Threaded, long radius	1.5	16	800	0.071	4.2
	Flanged, welded, bends	1	20	800	0.091	4.0
		2	12	800	0.056	3.9
		4	14	800	0.066	3.9
		6	17	800	0.075	4.2
	Mitered					
	1 weld (90°)		60	1000	0.27	4.0
	2 welds (45°)		30	800	0.136	4.1
	3 welds (30°)		24	800	0.105	4.2
45°	Threaded standard	1	16	500	0.071	4.2
	Long radius	1.5		500	0.052	4.0
	Mitered					
	1 weld (45°)		15	500	0.086	4.0
	2 welds (22.5°)		12	500	0.052	4.0
180°	Threaded, close return bend	1	50	1000	0.23	4.0
	Flanged	1		1000	0.12	4.0
	All	1.5		1000	0.10	4.0
Tees	Through branch (as elbow)					
	Threaded	1	60	500	0.274	4.0
		1.5		800	0.14	4.0
	Flanged	1	20	800	0.28	4.0
	Stub-in branch			1000	0.34	4.0
	Run-through threaded	1	20	200	0.091	4.0
	Flanged	1		150	0.05	4.0
	Stub-in branch			100	0	0
Valves						
Angle valve	Valve					
	45° full line size	$\beta = 1$	55	950	0.25	4.0
	90° full line size	$\beta = 1$	150	1000	0.69	4.0
Globe valve	Standard	$\beta = 1$	340	1500	1.70	3.6
Plug valve	Branch flow		90	500	0.41	4.0
	Straight through		18	300	0.084	3.9
	Three-way (flow through)		30	300	0.14	4.0
Gate valve	Standard	$\beta = 1$	8	300	0.037	3.9
Ball valve	Standard	$\beta = 1$	3	300	0.017	3.5
Diaphragm	Dam type			1000	0.69	4.9
Swing check	$V_{min} = 35[\rho(lb_m/ft^3)]^{-1/2}$ (ft/s)		100	1500	0.46	4.0
Lift check	$V_{min} = 40[\rho(lb_m/ft^3)]^{-1/2}$ (ft/s)		600	2000	2.85	3.8

Note: D_n is the nominal pipe size in inches.

[a] $K_f = \dfrac{K_1}{N_{Re}} + K_i\left(1 + \dfrac{K_d}{D_n^{0.3}}\right)$.

The method assumes that (1) sizes of all fittings of a given type can be scaled by the corresponding pipe diameter (D), and (2) the influence of turbulence level (i.e., Reynolds number) on the friction loss in the fitting is identical to that in the pipe (because the pipe f value is used to determine the fitting loss). Neither of these assumptions is accurate (as pointed out earlier), although the approximation provided by this method gives reasonable results at high turbulence levels (fully turbulent flow), especially if fitting losses are minor when compared to the total pipe friction loss.

C. Crane Method

The method given in the Crane Technical Paper 410 (1991) is a modification of the aforementioned methods. It is equivalent to the $(L/D)_{eq}$ method except that it recognizes that there is generally a higher degree of turbulence in the fitting than in the pipe at a given (pipe) Reynolds number. This is accounted for by always using the "fully turbulent" value for f (e.g., f_T) in the expression for the friction loss in the fitting, regardless of the actual Reynolds number in the pipe, that is,

$$e_f = \frac{K_f V^2}{2} \quad \text{where } K_f = 4 f_T \left(\frac{L}{D}\right)_{eq} \tag{7.35}$$

The value of f_T can be calculated from the Colebrook equation (Equation 6.40), for example,

$$f_T = \frac{0.0625}{[\log(3.7D/\varepsilon)]^2} \tag{7.36}$$

in which ε is the pipe roughness (0.0018 in. for new commercial steel). This is a two-constant model [f_T and $(L/D)_{eq}$], and values of these constants are tabulated in the Crane paper for a wide variety of fittings, valves, etc. This method gives satisfactory results for high turbulence levels (fully turbulent flow) but is less accurate at low Reynolds numbers and does not scale well with pipe size.

D. 2-*K* (Hooper) Method

The 2-K method by Hooper (1981, 1988) was based on experimental data from a variety of valves and fittings over a wide range of Reynolds numbers. The effect of both the Reynolds number and scale (fitting size) is reflected in the expression for the loss coefficient:

$$e_f = \frac{K_f V^2}{2}, \quad \text{where } K_f = \frac{K_1}{N_{Re}} + K_\infty\left(1 + \frac{1}{ID_{in.}}\right) \tag{7.37}$$

Here, $ID_{in.}$ is the internal diameter (in inches) of the pipe that contains the fitting. This method is valid over a much wider range of Reynolds numbers than the other methods. However, the effect of pipe size (e.g., $1/ID_{in.}$) in Equation 7.37 does not accurately reflect the scaling with pipe size, as discussed below in Section E.

E. 3-*K* (Darby) Method

Although the 2-K method applies over a wide range of Reynolds numbers, the scaling term (1/ID) does not accurately reflect data over a wide range of sizes for valves and fittings, as reported in a variety of sources (Crane, 1991; CCPS, 1998; Perry and Green, 2007; Darby, 2001; and references cited therein). Specifically, all the preceding methods tend to underpredict the friction loss for fittings of larger diameters. Darby (2001) has evaluated data from the literature for various valves and fittings and found that they can be represented more accurately by the following "3-K" equation:

$$K_f = \frac{K_1}{N_{Re}} + K_i\left(1 + \frac{K_d}{D_{n,in.}^{0.3}}\right) \tag{7.38}$$

Note that $D_{n,in.}$ is the nominal diameter, in inches. The values of the 3 K's (K_1, K_i, and K_d) are given in Table 7.3 (along with representative values of $(L/D)_{eq}$) for various valves and fittings. These values were determined from combinations of literature values from the references listed earlier and were all found to accurately follow the scaling law given in Equation 7.38. The values of K_1 are mostly those of the Hooper 2-K method, and the values of K_i were mostly determined from the Crane data. However, since there is no single comprehensive data set for many fittings over a wide range of sizes and Reynolds numbers, some estimation was necessary for some values.

Values of K_d are all very close to 4.0, and this value can be used to scale known values of K_f for a given pipe size to apply to other sizes. This method is the most accurate of the methods described for all Reynolds numbers and fitting sizes. Tables 7.4 and 7.5 list values for K_f for expansions and contractions and entrance and exit conditions, respectively (Hooper, 1981). The definition of K_f (i.e., $K_f = 2e_f/V^2$) involves the kinetic energy of the fluid, $V^2/2$. For sections that undergo area changes (e.g., pipe entrance, exit, expansions, or contractions), the entering and leaving velocities will be different. Because the value of the velocity used with the definition of K_f is arbitrary, it is very important to know which velocity is the reference value for a given loss coefficient. Values of K_f are usually based on the larger velocity entering or leaving the fitting (through the smaller cross section), but this should be verified if any doubt exists.

A note is in order regarding the exit loss coefficient, which is listed in Table 7.5 as equal to 1.0. Actually, if the fluid exits the pipe in a free jet into unconfined space, the loss coefficient is zero because the velocity of the fluid exiting the pipe is close to that of the fluid inside the pipe and thus the kinetic energy change is zero. However, when the fluid exits into a confined space so that the fluid leaving the pipe immediately mixes with the same fluid in the receiving vessel, the kinetic energy is dissipated as friction loss in the mixing process so the velocity goes to zero, and thus the loss coefficient is 1.0. In this case, the change in the kinetic energy and the friction loss at the exit cancel out.

IV. NON-NEWTONIAN FLUIDS

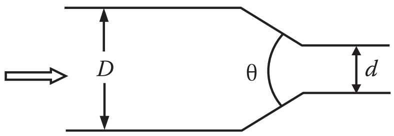
There are insufficient data in the literature to enable reliable correlation or prediction of friction loss in valves and fittings for non-Newtonian fluids. As a first approximation, however, it can be assumed that a correlation similar to the 3-K method should apply to non-Newtonian fluids if the (Newtonian) Reynolds number in Equation 7.38 could be replaced by a single corresponding dimensionless group that adequately incorporates the influence of the non-Newtonian properties. For the power law and Bingham plastic fluid models, two rheological parameters are required to describe the viscous properties, which generally results in two corresponding dimensionless groups ($N_{Re,pl}$ and n for the power law and N_{Re} and N_{He} for the Bingham plastic). However, it is possible to define an "effective viscosity" for a non-Newtonian fluid model that has the same significance in the Reynolds number as the viscosity has for a Newtonian fluid and incorporates all of the appropriate parameters for that model, which then can be used to define an equivalent non-Newtonian Reynolds number (see Darby and Forsyth, 1992). For a Newtonian fluid, the Reynolds number can be rearranged as follows:

$$N_{Re} = \frac{DV\rho}{\mu} = \frac{\rho V^2}{\mu V/D} = \frac{\rho V^2}{\tau_w/8} \tag{7.39}$$

TABLE 7.4
Loss Coefficients for Expansions and Contractions

K_f to be used with upstream velocity head, $V_1^2/2$. $\beta = d/D$

Contraction



$\theta < 45°$

$N_{Re,1} < 2500$:

$$K_f = 1.6\left[1.2 + \frac{160}{N_{Re,1}}\right]\left[\frac{1}{\beta^4} - 1\right]\sin\frac{\theta}{2}$$

$N_{Re,1} > 2500$:

$$K_f = 1.6\left[0.6 + 1.92 f_1\right]\left[\frac{1-\beta^2}{\beta^4}\right]\sin\frac{\theta}{2}$$

$\theta > 45°$

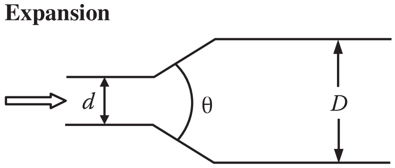
$N_{Re,1} < 2500$:

$$K_f = \left[1.2 + \frac{160}{N_{Re,1}}\right]\left[\frac{1}{\beta^4} - 1\right]\left[\sin\frac{\theta}{2}\right]^{1/2}$$

$N_{Re,1} > 2500$:

$$K_f = \left[0.6 + 1.92 f_1\right]\left[\frac{1-\beta^2}{\beta^4}\right]\left[\sin\frac{\theta}{2}\right]^{1/2}$$

Expansion



$\theta < 45°$

$N_{Re,1} < 4000$:

$$K_f = 5.2(1-\beta^4)\sin\frac{\theta}{2}$$

$N_{Re,1} > 4000$:

$$K_f = 2.6\left(1 + 3.2 f_1\right)\left(1-\beta^2\right)^2 \sin\frac{\theta}{2}$$

$\theta > 45°$

$N_{Re,1} < 4000$:

$$K_f = 2(1-\beta^4)$$

$N_{Re,1} > 4000$:

$$K_f = (1 + 3.2 f_1)(1-\beta^2)^2$$

Source: Hooper, W.B., "Calculate head Loss Caused by Change in Pipe Size", *Chem. Eng.*, 95, pp. 89–92, 1988.

Note: $N_{Re,1}$ is the upstream Reynolds number, and f_1 is the pipe friction factor at this Reynolds number.

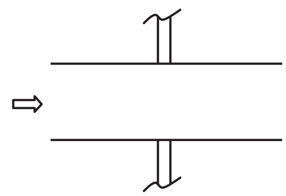
TABLE 7.5
Loss Coefficients for Pipe Entrance and Exit

$K_f = K_1/N_{Re} + K_\infty$

Entrance

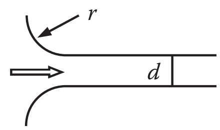
Inward projecting (Borda)

$K_1 = 160$, $K_\infty = 1.0$



Flush (rounded)

$K_1 = 160$



r/d	K_∞
0 (sharp)	*0.5*
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 and up	0.04

For pipe exit:

$K_\infty = 1.0$ for all geometries

$K_1 = 0$

Orifice: $K_\infty = \dfrac{2.91}{\beta^4}(1-\beta^2)(1-\beta^4) = \dfrac{(1-\beta^2)(1-\beta^4)}{C_o^2\beta^4}$

$\beta = D_o/D_p$

$K_1 = 0$

Source: Hooper, W.B., "The 2-K Method Predicts Head Loss in Pipe Fittings" *Chem. Eng.*, 88, pp. 96–100, 1981.

Introducing $\tau_w = m[(8V/D)(3n + 1)/4n]^n$ for the power law model results in

$$N_{Re,pl} = \frac{2^{(7-3n)}\rho Q^{(2-n)}}{m\pi^{(2-n)}D^{(4-3n)}}\left(\frac{n}{3n+1}\right)^n \tag{7.40}$$

which is identical to the expression derived in Chapter 6 (see Equation 6.71).

For the Bingham plastic, replacing τ_w for the Newtonian fluid in Equation 7.39 with $\tau_o + \mu_\infty\dot{\gamma}_w$ and using the approximation $\dot{\gamma}_w = 8V/D$, the corresponding expression for the Reynolds number is

$$N_{Re,BP} = \frac{4Q\rho}{\pi D\mu_\infty(1+\pi D^3\tau_o/32Q\mu_\infty)} = \frac{N_{Re}}{1+N_{He}/8N_{Re}} \tag{7.41}$$

The ratio $N_{He}/N_{Re} = D\tau_o/V\mu_\infty$ is also called the Bingham number (N_{Bi}). Darby and Forsyth (1992) showed experimentally that mass transfer in Newtonian and non-Newtonian fluids can be correlated by this method. That is, the same dimensionless correlation can be applied to both Newtonian and non-Newtonian fluids when the Newtonian Reynolds number is replaced by either Equation 7.40 for the power law fluid or Equation 7.41 for the Bingham plastic model. As a first approximation, therefore, we may assume that the same method would apply to friction loss in valves and fittings as described by the 3-*K* model, Equation 7.38. This approach is in agreement with the scant literature data on fitting losses with power law and Bingham plastic fluids (see, e.g., Chhabra and Richardson, 2008).